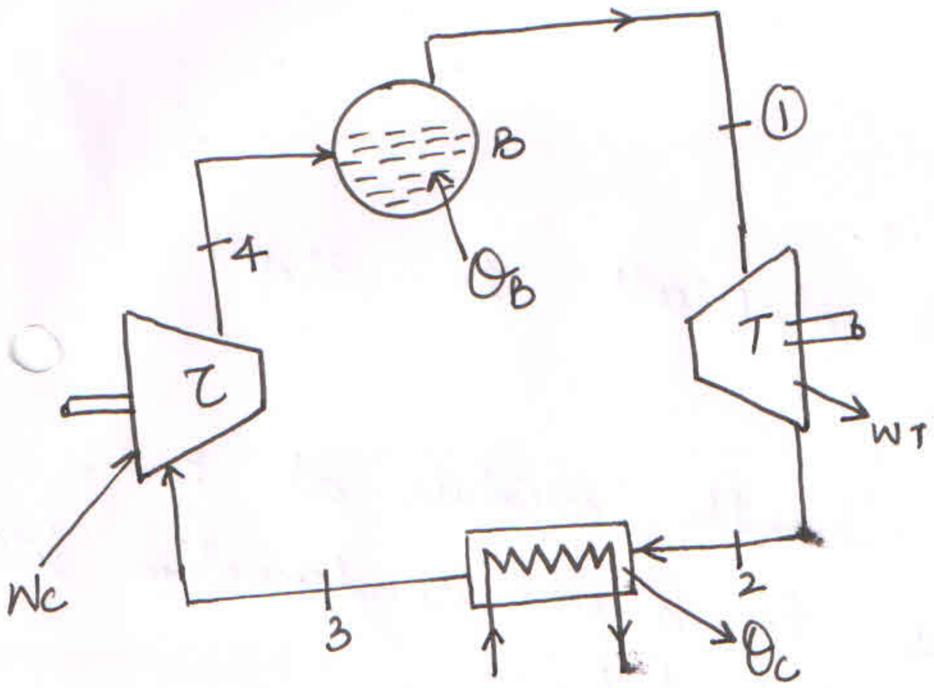
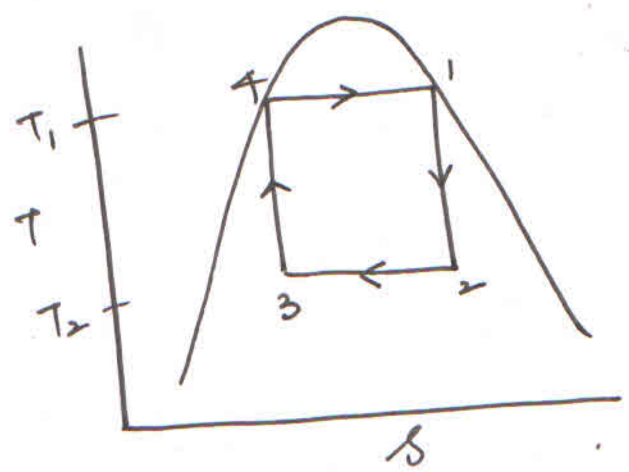
VAPOUR POWER CYCLE

Carnot Cycle:





$$\eta = \frac{W_T - W_C}{Q_B}$$

$\sum W = \sum Q$ or $W_T - W_C = Q_B - Q_C$

$$\eta = \frac{Q_B - Q_C}{Q_B} = 1 - \frac{Q_C}{Q_B}$$

Highest eff. But low work ratio

$$R = \frac{W_T - W_C}{W_T}$$

limitation:

1. low work ratio.
2. At T_3 it is diffi~~cult~~ to stop
   Condensation.
3. Difficult to handle mixture at 3.
   It is better to allow Condensation till
   it becomes Saturated at 3

Def: ~~Steam~~ Heat Rate is defined as amount of heat supplied to generate 1KWh of electricity or power.

Def: Steam Rate: It is defined as the rate of steam flow (kg/h) required to produce unit shaft power (1Kw)

$$SR = \frac{1}{W_T - W_P} = \frac{3600}{W_T - W_P} \text{ kJ/kWh}$$

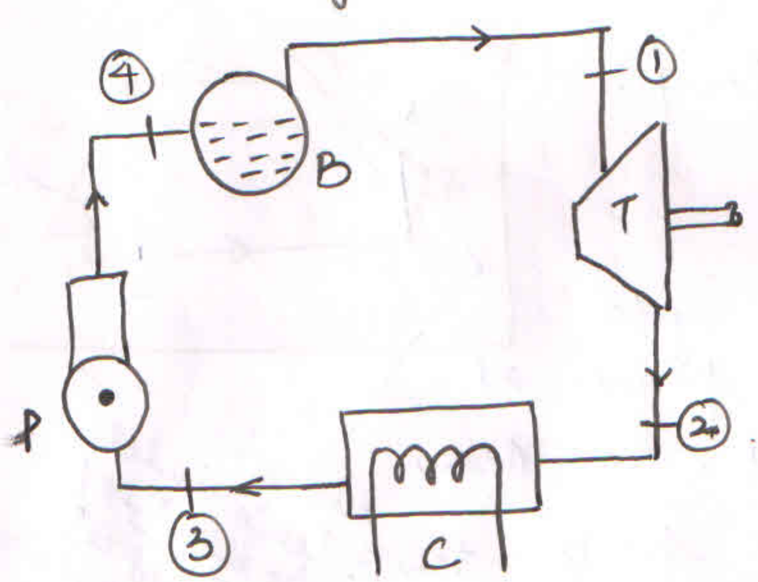
Refer to

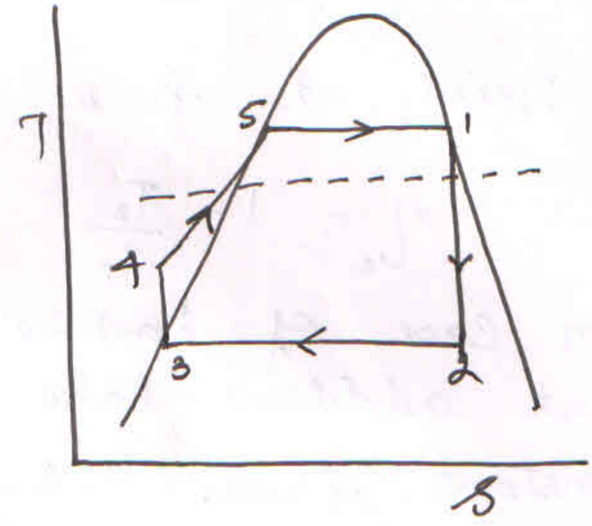
1. Applied thermodynamics for Engineering Technologists by Eastop & McConkey
2. Engg. Thermodynamics by PK Nag

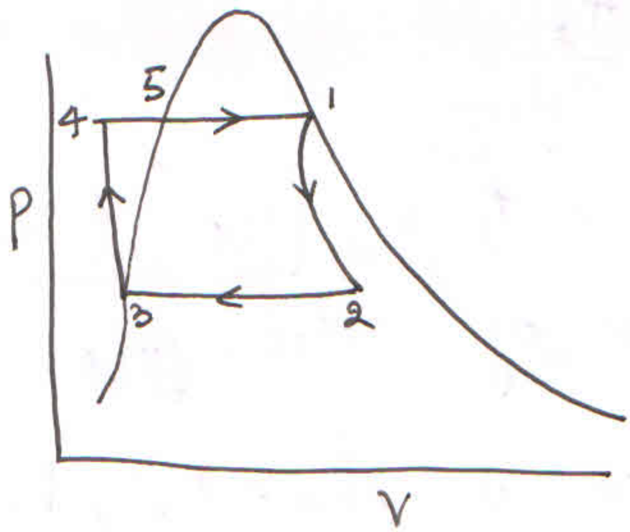
for

a) Isentropic Efficiency of Turbines
b) Definition of various Terms.

Rankine Cycle:-





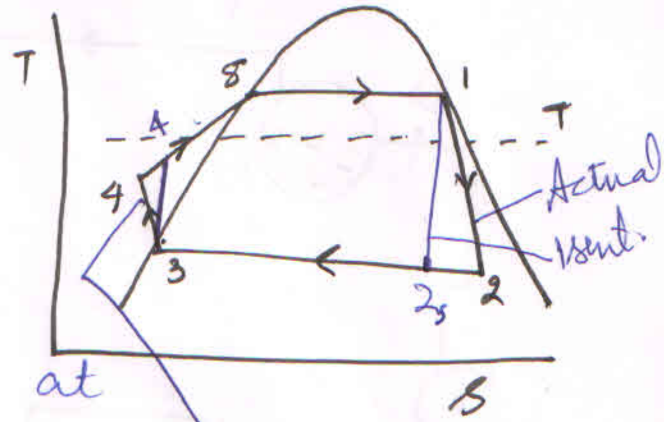


At (3) pump is Introduced - a Small volume of liquid.

Heating Starts from 4 to 1

Efficiency of Rankine Cycle is less than that of Carnot cycle operating between Same temperatures.

In case of Carnot cycle entire heat is supplied at peak temp.



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

In case of Rankine cycle heat addition take place at constant press. 4-5-1, i.e Mean temp is less than peak temp.

$$\eta_R = 1 - \frac{T_2}{T_{mean}}$$

Since $T_{mean} < T_2$ = $\boxed{\eta_R < \eta_c}$

For Unit mass of water SFEE

1. Boiler : SFEE is $h_4 + Q_1 = h_1$

$$Q_1 = h_1 - h_4$$

2. Turbine : $h_1 = W_T + h_2$ $W_T = h_1 - h_2$

3. Condenser : $Q_2 + h_3 = h_2$

$$Q_2 = h_2 - h_3$$

4. Pump : $W_p + h_4 = h_3$

$$W_p = h_3 - h_4$$

$$\eta = \frac{W_{net}}{Q_1} = \frac{W_T - W_C}{Q_1} = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4}$$

pump work W_c can be neglected because

$$T\,ds = dh - v\,dp \quad ds = 0$$

$$dh = v\,dp$$

$$\Delta h = \int v\,dp$$ since v is very very small for liquids.

$\int v\,dp$ can be neglected.

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

<u>Steam</u> <u>Rate</u>: It is defined as the rate of steam flow (kg/h) required to produce Unit shaft power (1 kW)

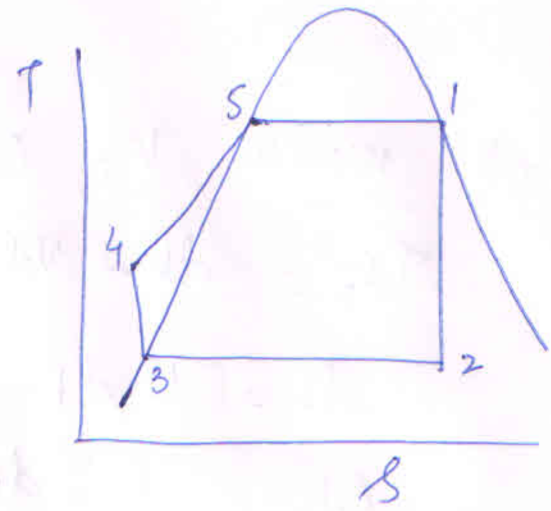
$$S.R = \frac{1}{W_T - W_P} \frac{kg}{kW.s} = \frac{3600}{W_T - W_P} \frac{kJ}{kWh}$$

<u>Heat</u> <u>Rate</u> :- Defined as the rate of heat I/p (Q) to produce Unit power.

$$1\,kW : HR = \frac{3600 Q_1}{W_T - W_P} = \frac{3600}{\eta_{cyc}} \frac{kg}{kWh}.$$

Mean Temp of heat addition

$Tds = dh + Vdp$



from 4-1 $dP = 0$

T is varying from 4-1

$Tds = dh$

Int. $\int T ds = \int dh$

Let there be some mean Temp T_m such that $\int dh = T_m \int ds$

$$T_m = \frac{h_1 - h_4}{S_1 - S_4}$$

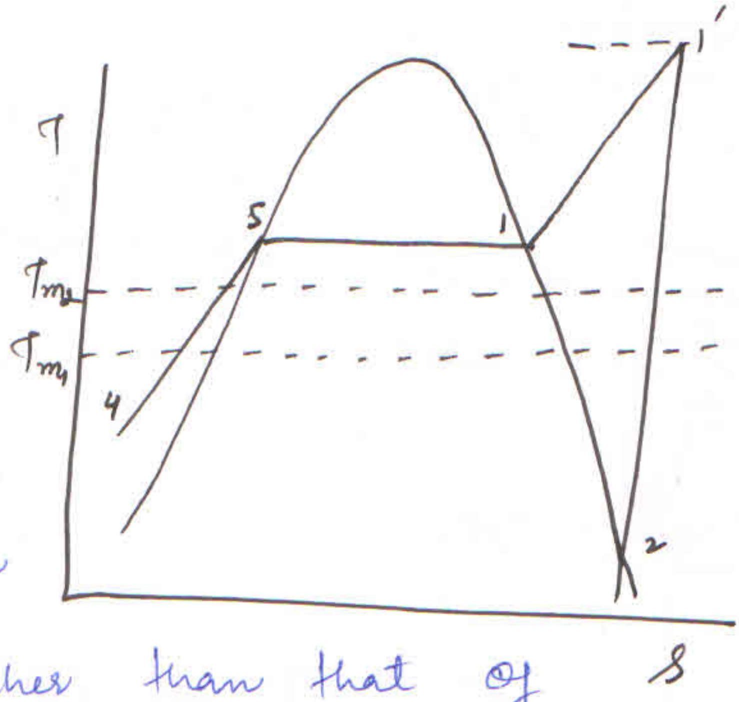
$$\eta_R = 1 - \frac{Q_2}{Q_1} = 1 - \frac{T_2 (s_1 - s_{4s})}{T_m (s_1 - s_{4s})}$$

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

keeping T_2 same

$$\eta_R = f(T_m)$$

Efficiecy of Rankine cycle can be improved by Increasing T_m. It is possible by super-heating

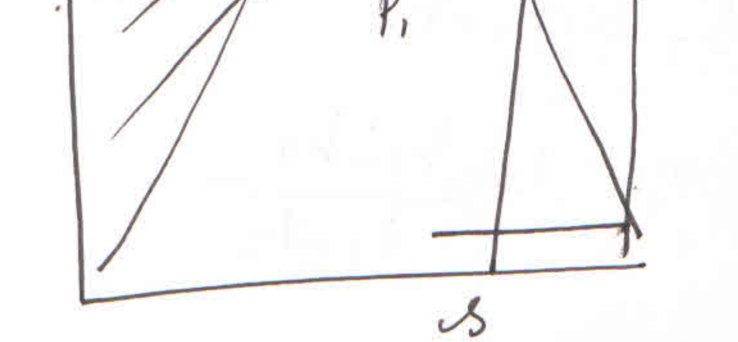


$T_{m_2} > T_{m_1}$

hence efficiency of Rankine cycle with superheat is higher than that of saturated steam.

But T_{max} (T_1') is restricted by metallurgical consideration.

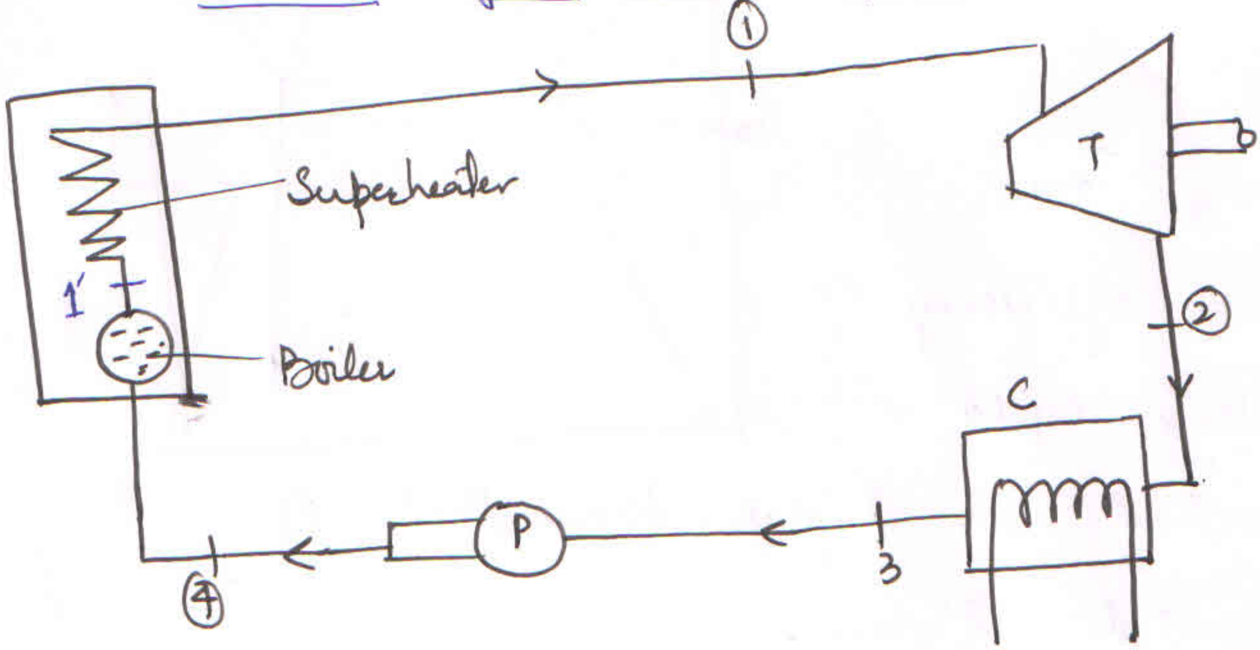
Tmean can be measured by Increasing pressure also. That leads to wet steam at the turbine exit. The droplets of water may erode the the turbine blades.

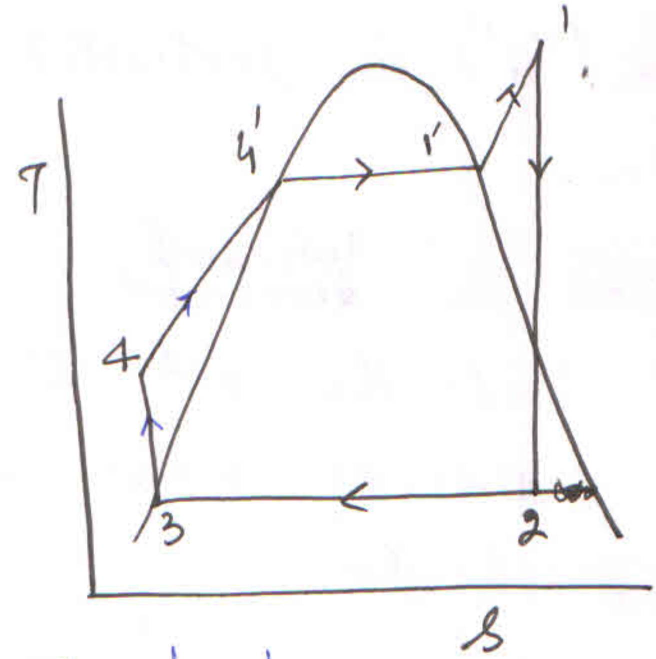


The dryness fraction at the turbine exits should be greater than 85% to prevent the damage to turbine blades.

SUPERHEATED STEAM

Rankine cycle with Superheat



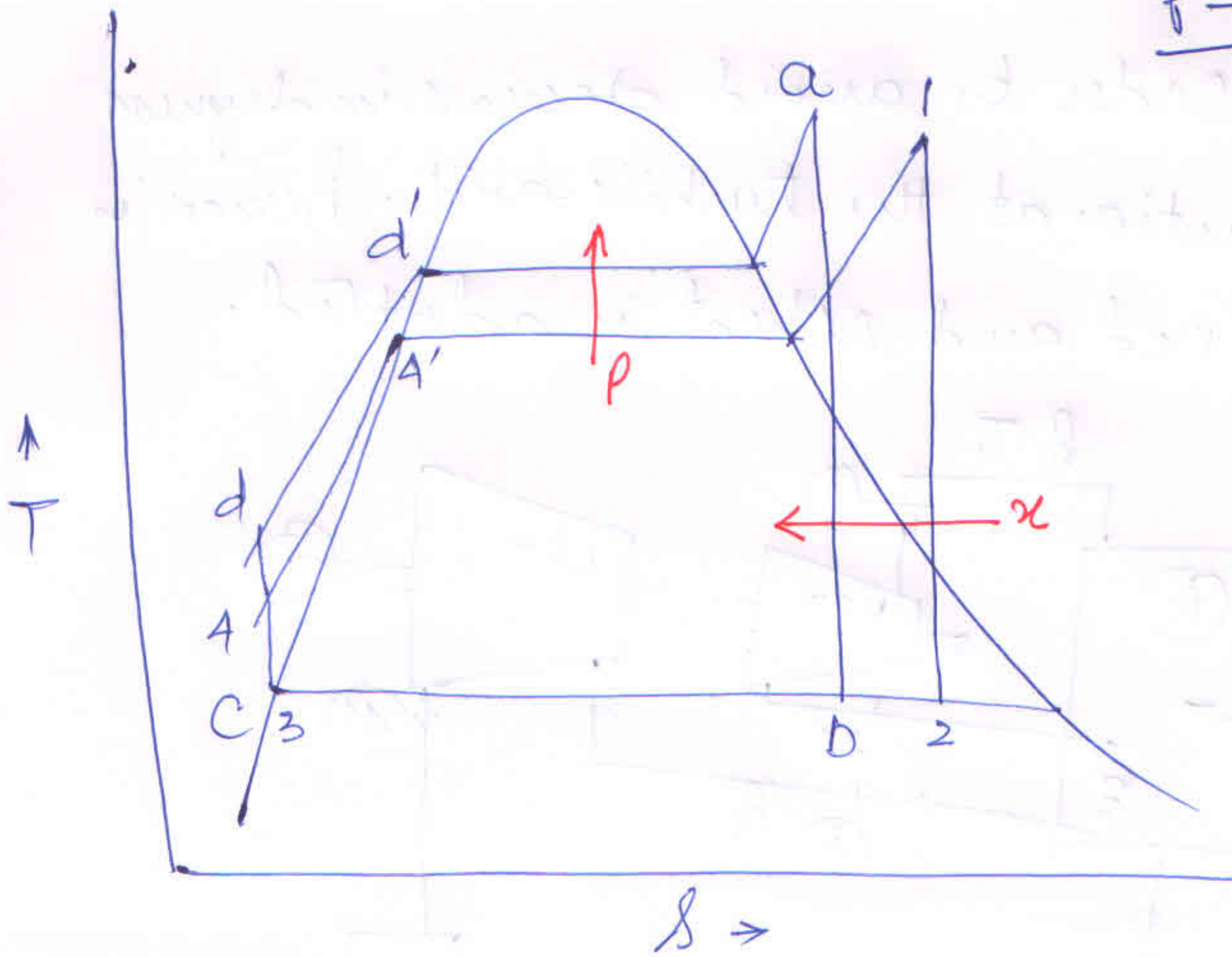


$Q_1 = h_1 - h_4$ $\qquad W_T = h_1 - h_2$

$Q_2 = h_2 - h_3$ $\qquad W_P = h_4 - h_3$

$$\eta = 1 - \frac{h_1 - h_4}{h_2 - h_3} \simeq \frac{h_1 - h_2}{h_1 - h_4}$$

W_P - very Small



Effect of INCREASING PRESSURE. on Rankine cycle

1. Average temp increases by increasing pressure as shown above. ↑P

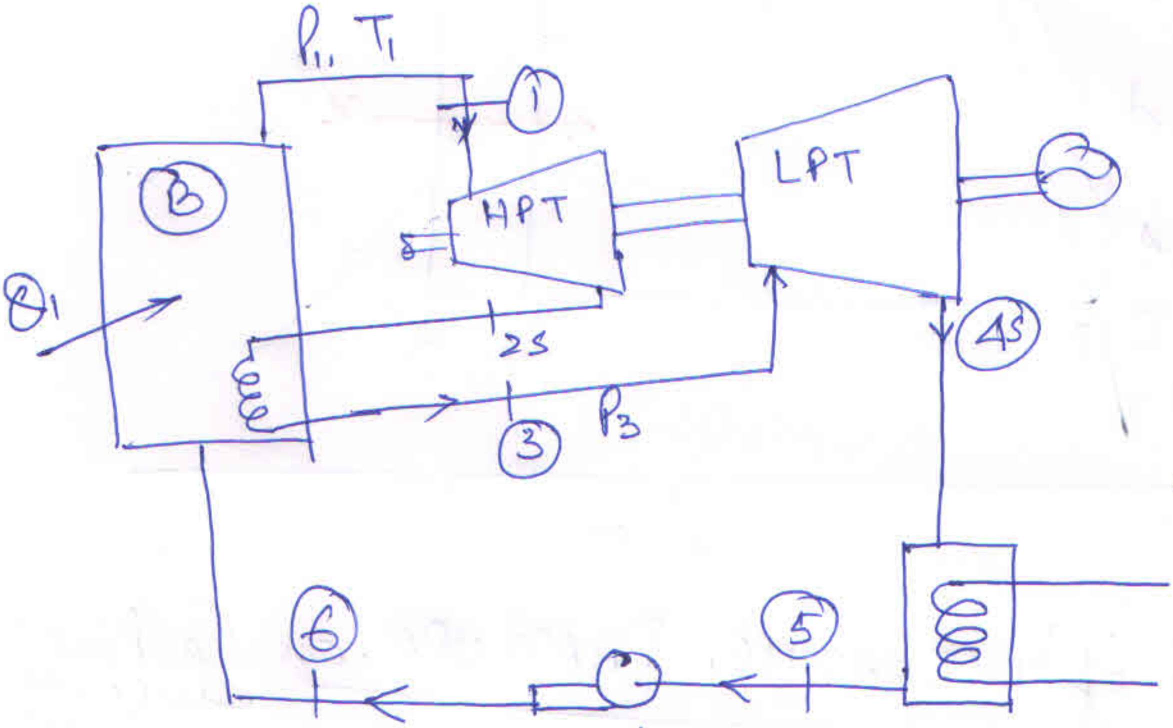
2. but dryness fraction at the turbine exit decreases from 2 to b.

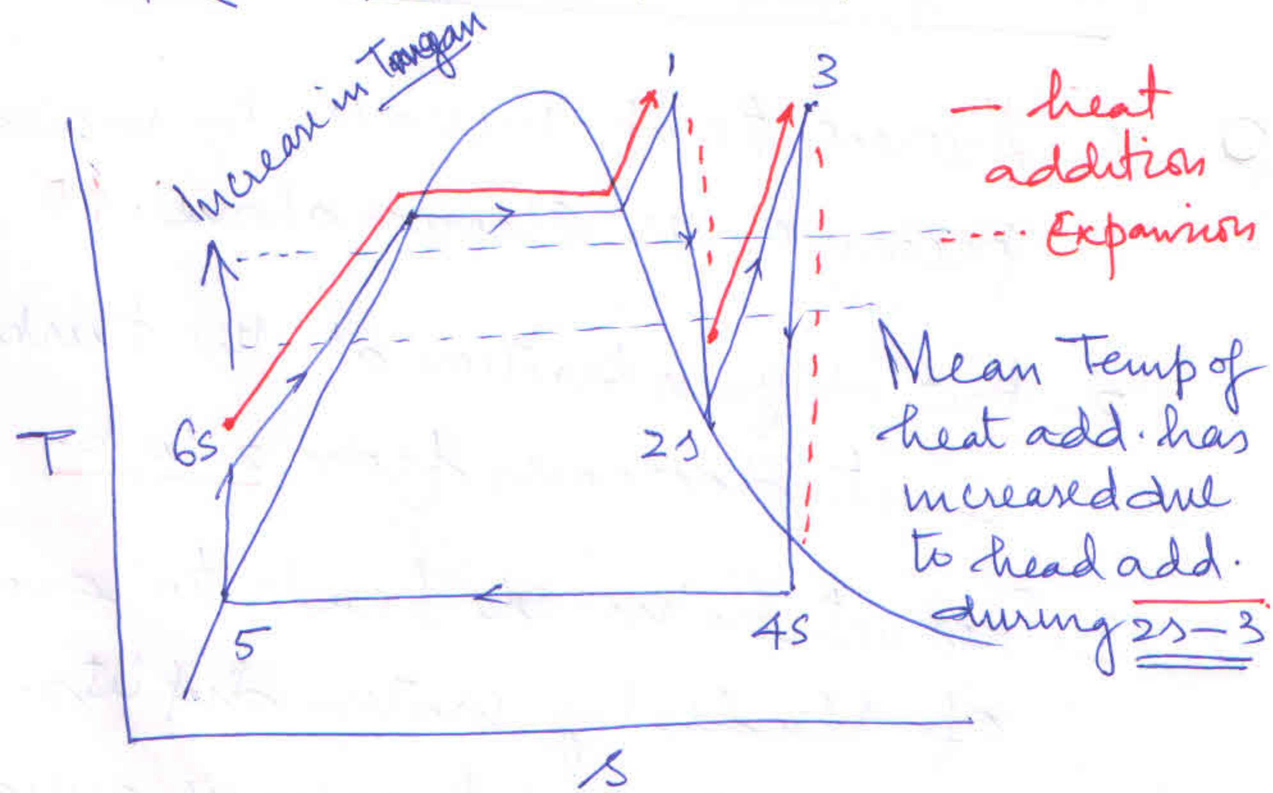
3. Reduction in x leads to erosion of blades by water droplets.

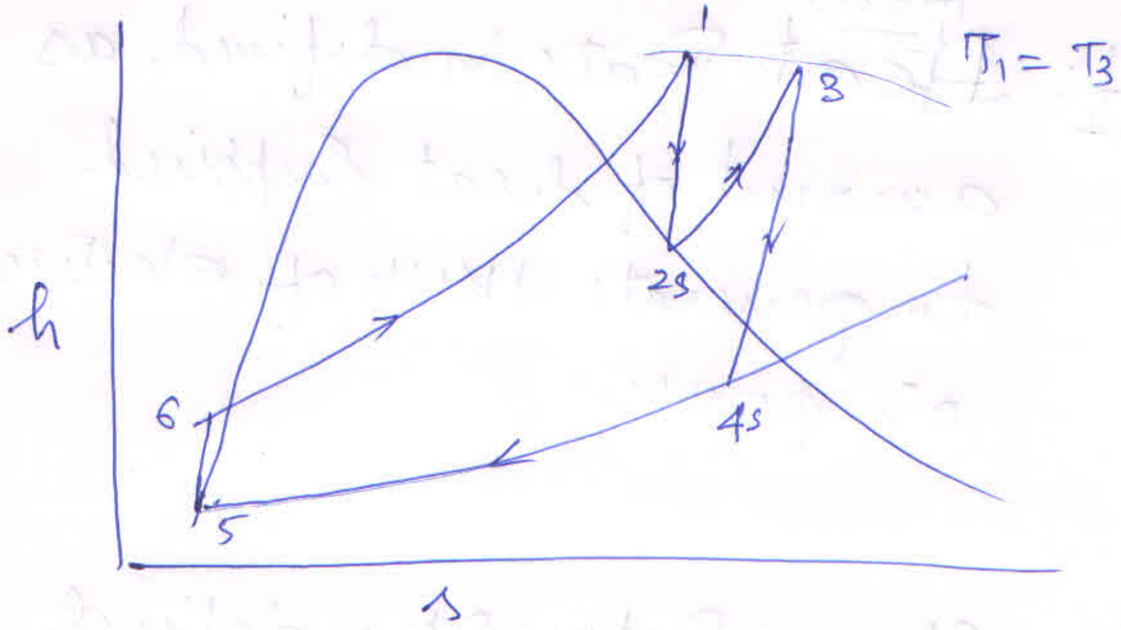
Increasing pressure to increase average temp is not admissable

Reheat Cycle

In order to avoid decrease in dryness fraction at the turbine exit, $P_{1(max)}$ is fixed and reheat is adopted.







$$Q_1 = h_1 - h_{6s} + h_3 - h_{2s}$$

$$Q_2 = h_{4s} - h_5$$

$$W_T = h_1 - h_{2s} + h_3 - h_{4s}$$

$$W_P = h_{6s} - h_5$$

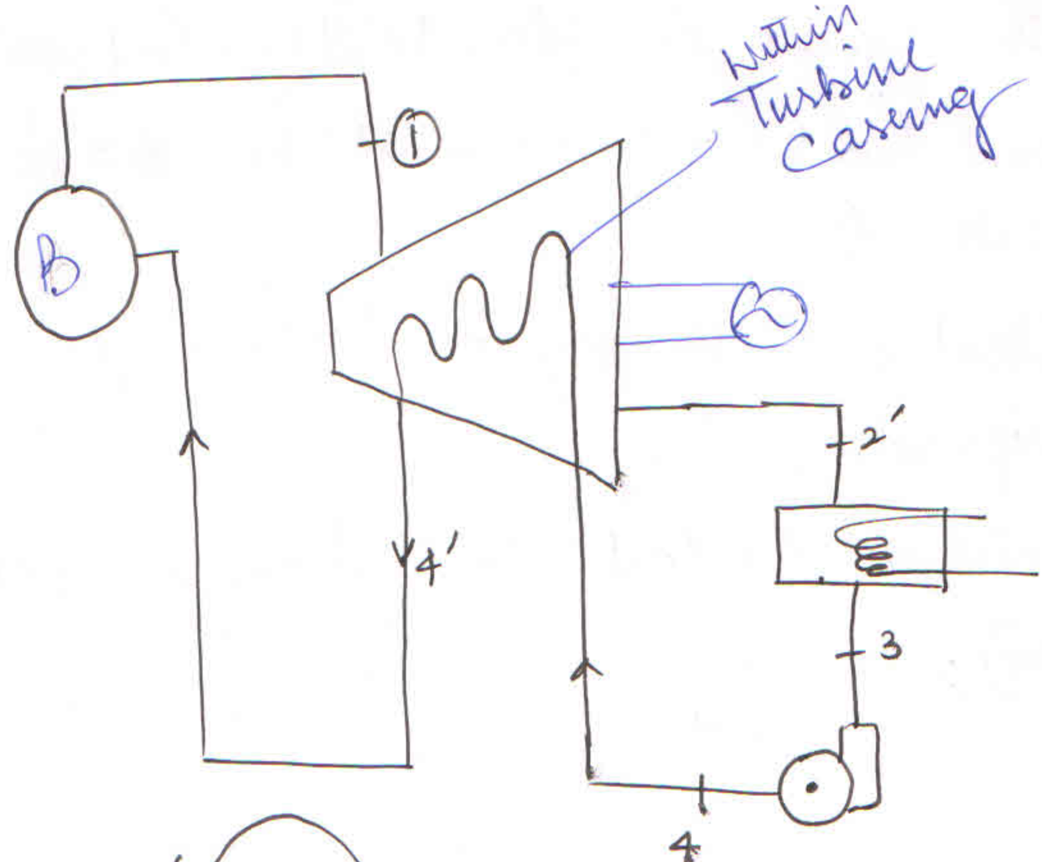
$$\eta = \frac{W_T - W_P}{Q_1} = \frac{(h_1 - h_{2s} + h_3 - h_{4s}) - (h_{6s} - h_5)}{h_1 - h_{6s} + h_3 - h_2}$$

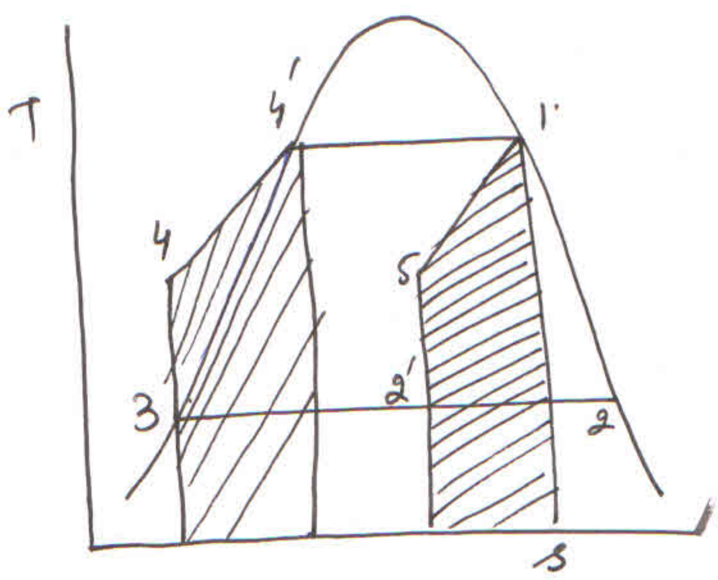
$$\text{Steam rate} = \frac{3600}{(h_1 - h_{2s} + h_3 - h_{4s}) - (h_{6s} - h_5)} \text{ Kg/Kwh}$$

* Since higher press. is used, pump work may be appreciable.

REGENERATIVE CYCLE

The mean Temp of heat addition can be Increased by decreasing the amount of heat added at low temp. In ideal Reg. cycle feed water is circulated in the turbine casing after the pump. The heat transfer is assumed to be reversible having infinitesimal temp grad.





11

$$Q_1 = h_1 - h_4 = T_1 (S_1 - S_{4'})$$

$$Q_2 = h_2' - h_3 = T_2 (S_2' - S_3)$$

Since $S_4' - S_3 = S_1 - S_2'$ & $S_1 - S_4' = S_2' - S_3$

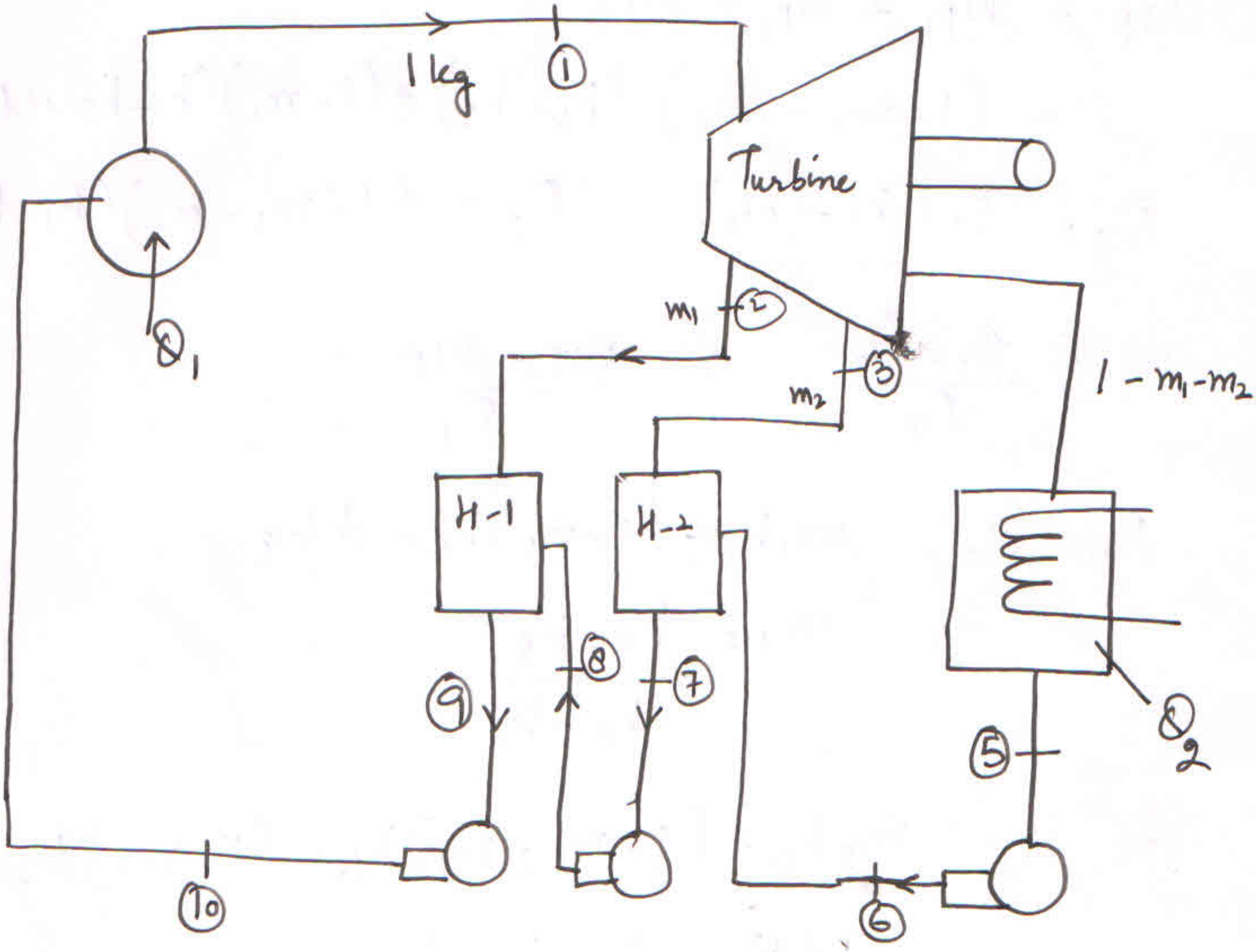
$$\eta = 1 - \frac{Q_2}{Q_1} = 1 - \frac{T_2}{T_1} \text{ (Carnot Eff).}$$

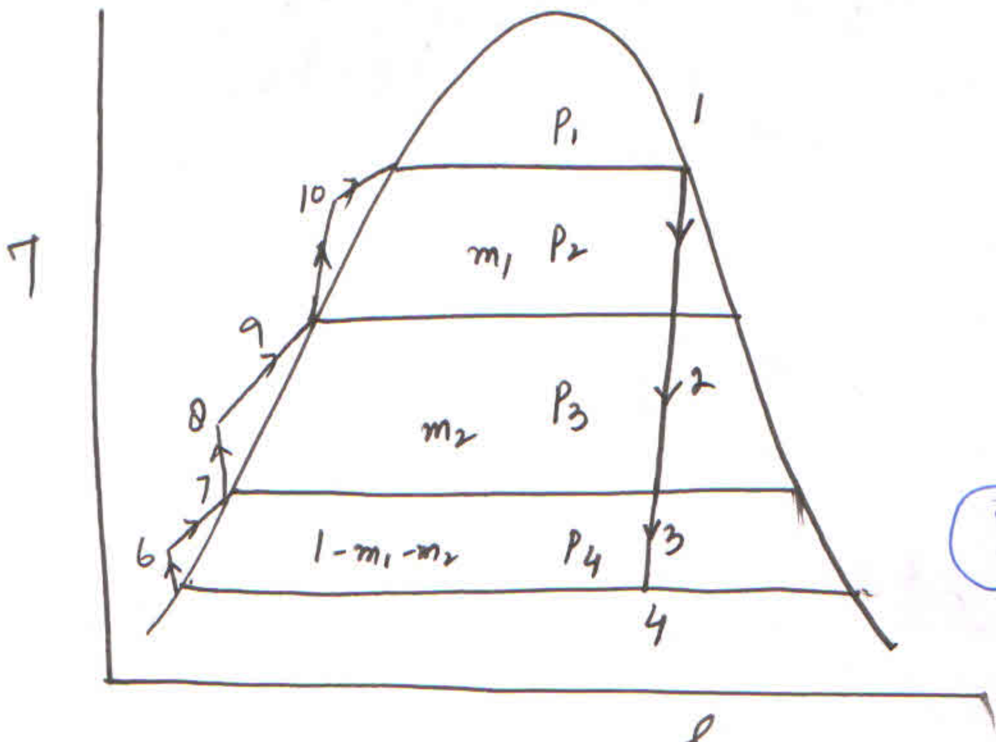
$$W_T = (h_1 - h_2') - (h_4' - h_4)$$

This cycle is practically impossible

1. Rev. heat x-fer cannot be obtained in finite time.
2. Heat x-changer in turbine is mechanically impractical.
3. Moisture content in turbine will be high.

Practical Regenerative Cycle :→





$$W_T = (h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4)$$

$$W_p = W_{p_1} + W_{p_2} + W_{p_3}$$

$$= (1 - m_1 - m_2)(h_6 - h_5) + (1 - m_1)(h_6 - h_7) + 1.(h_{10} - h_9)$$

$$Q_1 = 1.(h_1 - h_{10}) \qquad Q_2 = (1 - m_1 - m_2)(h_4 - h_5)$$

$$\eta = \frac{Q_1 - Q_2}{Q_1} = \frac{W_T - W_p}{Q_1}$$

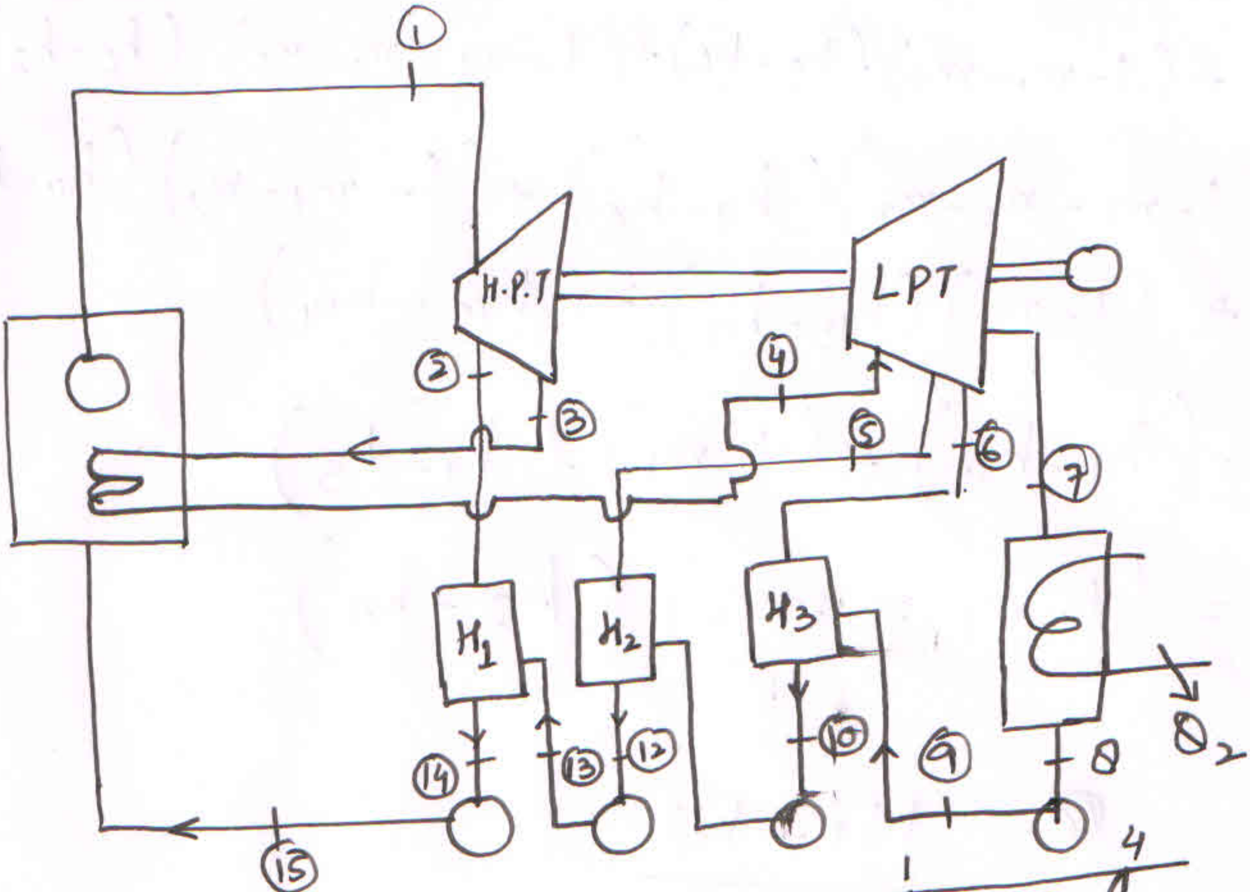
for H_2 $\quad m_1 h_2 + (1 - m_1) h_8 = 1 h_9$

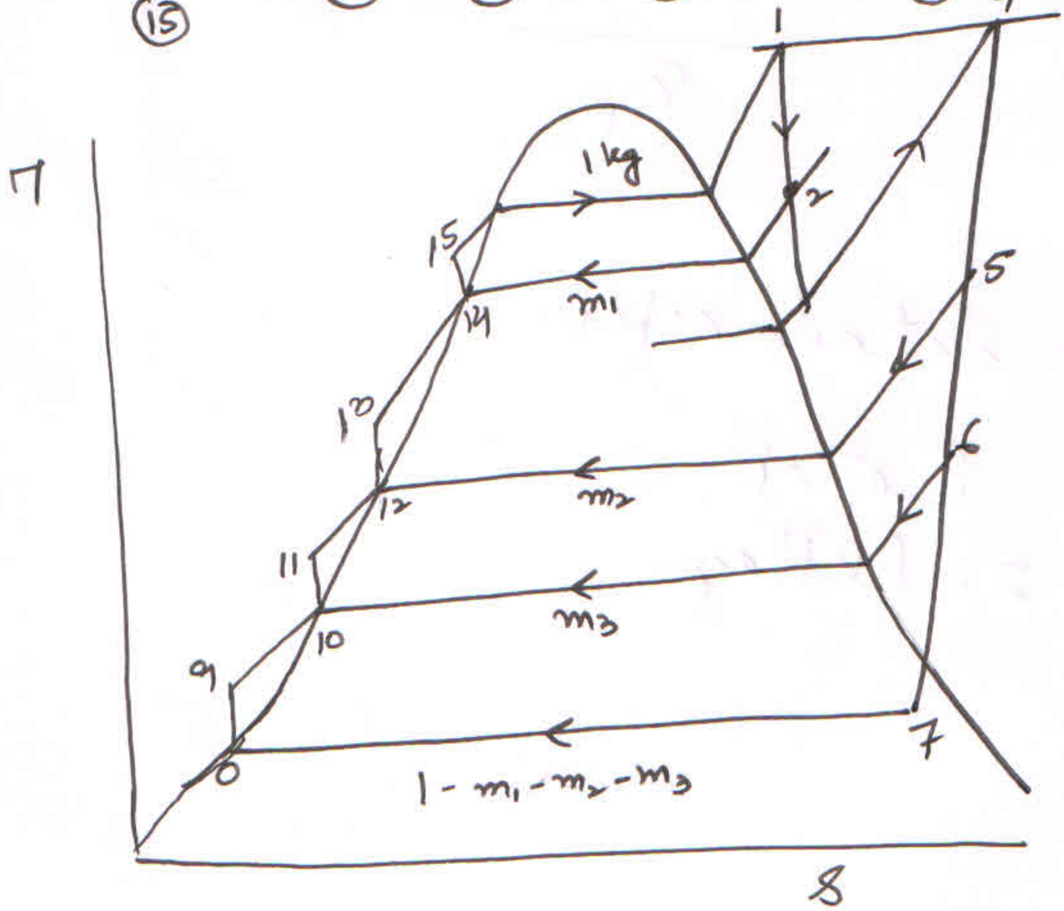
$$m_1 = \frac{h_9 - h_8}{h_2 - h_8}$$

for H_1 $\quad m_2 h_3 + (1 - m_1 - m_2) h_6 = (1 - m_1) h_7$

$$m_2 = (1 - m_1) \frac{h_7 - h_6}{h_3 - h_6}.$$

REHEAT REGENERATIVE CYCLE:





$$-W_T = (h_1-h_2)+(1-m_1)(h_2-h_3)+(1-m_1)(h_4-h_5)$$
$$+(1-m_1-m_2)(h_5-h_6)+(1-m_1-m_2-m_3)(h_6-h_7)$$

$$W_P = (1-m_1-m_2-m_3)(h_9-h_8)+(1-m_1-m_2)(h_{11}-h_{10})$$
$$+(1-m_1)(h_{13}-h_{12})+(h_{15}-h_{14})$$

$$Q_1 = (h_1-h_{15})+(1-m_1)\cdot(h_4-h_3)$$

$$Q_2 = (1-m_1-m_2-m_3)(h_7-h_8)$$

$$\eta = \frac{W_T - W_P}{Q_1}$$

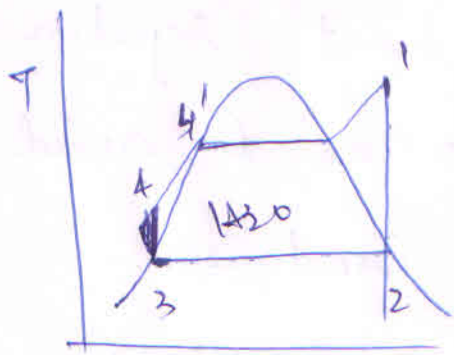
for details refer.

1. Eastop
2. PK Nag.

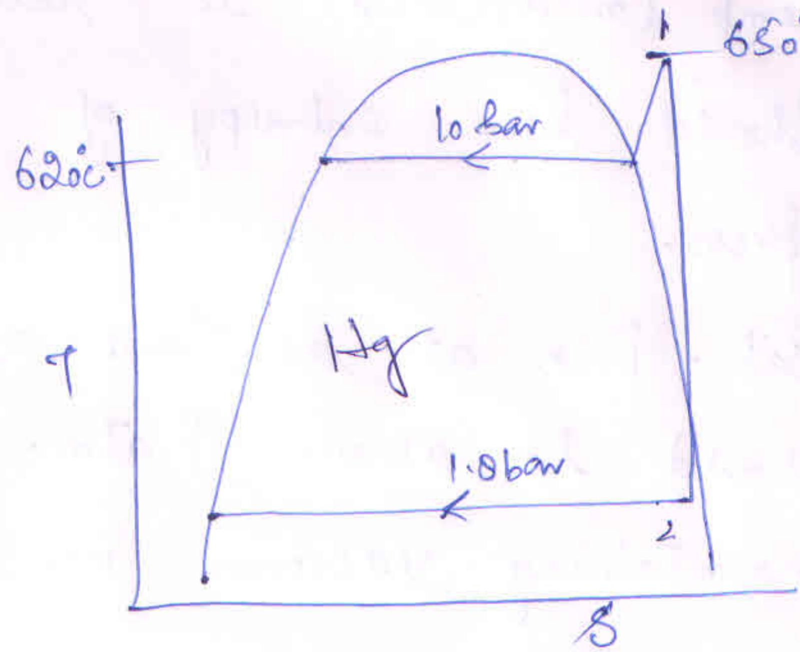
Desirable characteristics of working fluid.

1. Fluid should have high critical temp so that the sat. press at max permissible temp (mett. consid) is relatively low. It should have enthalpy of evaporation at that press.
2. Sat. press at the temp of heat rejection should be above P_{atm}. So as to avoid maintaining vacuum in the condenser.
3. Specific heat should be small so that little heat transfer is required to raise the liquid to boiling point.
4. Sat vapour line should be steep, very close to turbine expansion so that excessive moisture doesnot appear during expansion.
5. Freezing point of fluid should be below room temp.

6. Chemical Stability, Should not Contaminate with materials of Construction, non-toxic non Corrosive, not excessively viscous, low Cost.



Not ideal fluid



ideal working fluid.

Crit Temp = 374.15°C	Crit Temp = 1460°C
Crit Press = 221.2 bar	Crit Press = 1080 bar

Line 1-2 is away from Sat. line. Thus moisture appears at the end of expansion (at 2)	Line 1-2 is Closer to Sat. line. very little moisture appears at the end of expansion (at 2)

P-18

BINARY VAPOUR CYCLES:

No single fluid can meet all the requirements as mentioned. In overall evaluation water is a better choice. In higher temp range diphenyl ether $(C_6H_5)_2O$, Al. bromide, Al_2Br_6 mercury, Sodium, potassium are few fluids used.

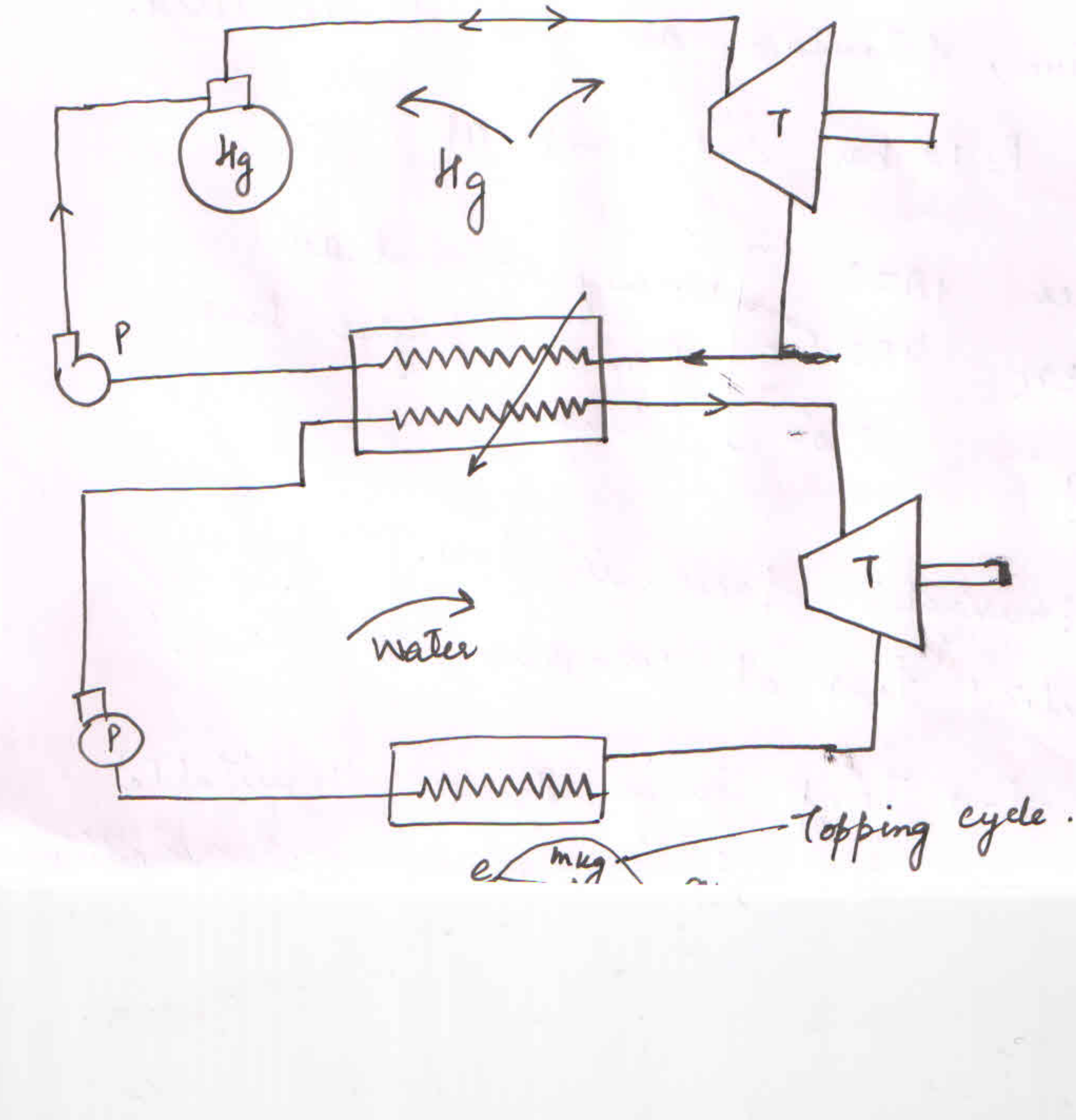
At P = 12 bar Sat. temp of

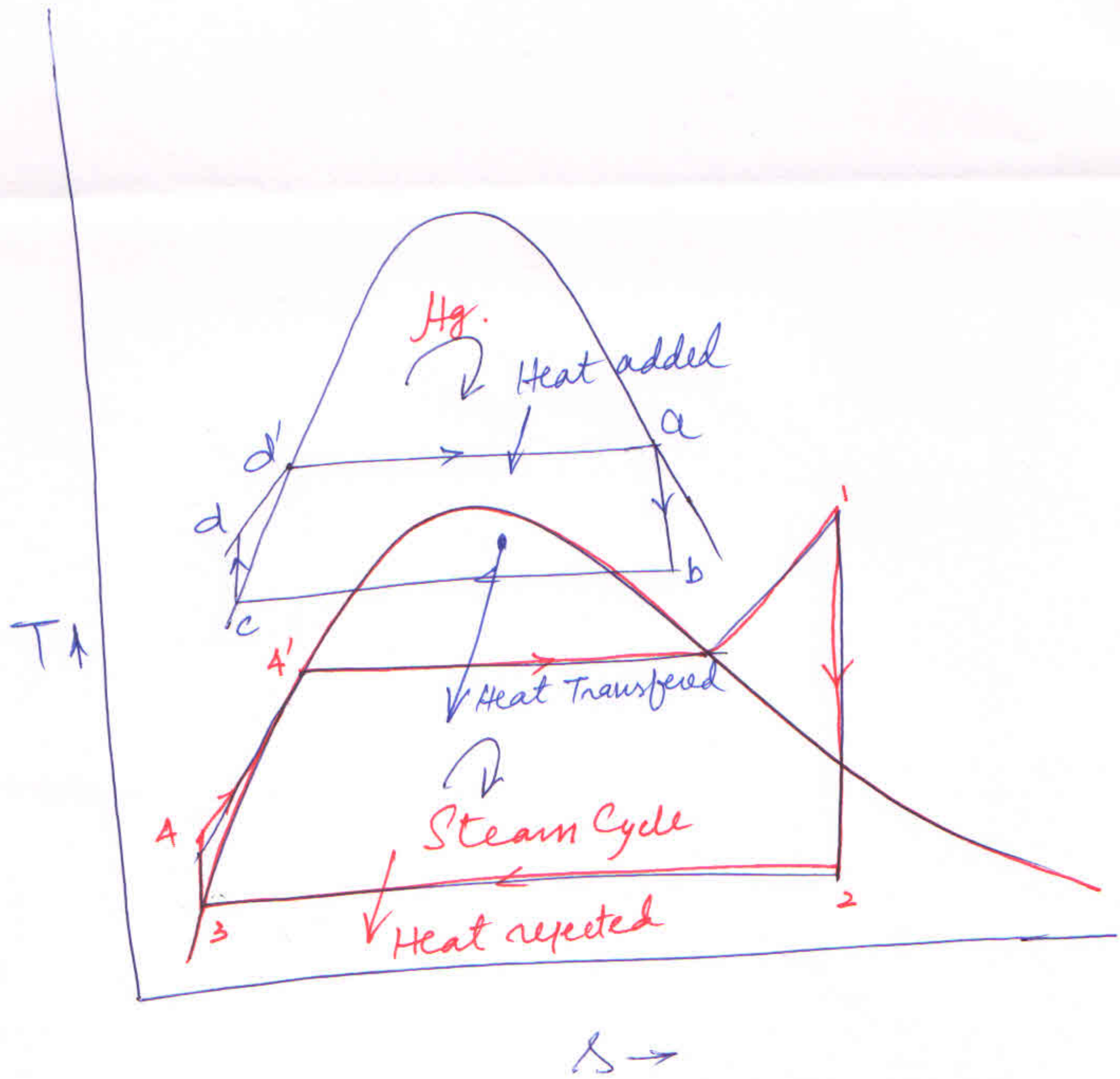
water	187°c	Mercury is a better fluid in higher temp range.
Al_2Br_6	482.5°c	
Hg	560°c	

Vapourization press is low.
Critical temp at 1080 bar is 1460°c

In low Temp range Hg is Unsuitable because its sat. press. becomes exceedingly low and it is very difficult to maintain high Vacuum. At 30°c Sat. press is 2.7×10^{-4} cm Hg. Its Sp. vol at such a low press is very large and it is difficult to accomodate it.

For this reason Hg vapours leaving the turbine is condensed at a higher temp and heat released during condensation is used for heating water to form steam.





Binary Vapour Power Cycle

Steam and Hg

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

$$\eta = \frac{Q_1 - Q_2}{Q_1} = \frac{W_T - W_P}{Q_1}$$

$$m(h_b - h_c) = h_6 - h_5$$

$$m = \frac{h_6 - h_5}{h_b - h_c}$$

To vapourise 1 kg of steam 7 to 8 kg Hg must Condense.

Schiller Station USA.

Efficiency of Coupled cycles.

$$\eta_1 = 1 - \frac{Q_2}{Q_1}, \quad \eta_2 = 1 - \frac{Q_3}{Q_2}$$

$$Q_2 = Q_1(1-\eta_1) \quad \text{—} (1)$$

$$Q_3 = Q_2(1-\eta_2) \quad \text{—} (2)$$

$$\eta = 1 - Q_3/Q_1$$

$$Q_2 Q_3 = Q_1 Q_2 (1-\eta_1)(1-\eta_2)$$

$$\frac{Q_3}{Q_1} = (1-\eta_1)(1-\eta_2)$$

$$\eta = 1 - (1-\eta_1)(1-\eta_2)$$

for n coupled cycles.

~~$\eta = 1 - \prod_{i=1}^{n}$~~

$$\eta = 1 - \prod_{i=1}^{n} (1 - \eta_i)$$

for $n = 2$

$$\eta = \eta_1 + \eta_2 - \eta_1 \eta_2$$

$\eta > \eta_1$, $\eta > \eta_2$ Always.

STEAM CYCLES.

Q.1 (a) Steam is supplied, dry saturated at 40 bar to a turbine and the condenser pressure is 0.035 bar. If the plant operates on the Rankine cycle, calculate, per kilogram of steam:

i) the work output neglecting the feed-pump work;
ii) the work required for the feed pump;
iii) the heat transferred to the condenser cooling water, and the amount of cooling water required through the condenser if the temperature rise of the water is assumed to be 5.5 K;
iv) the heat supplied;
v) the Rankine efficiency;

b) For the same steam conditions calculate the efficiency and the specific steam consumption for a Carnot cycle operating with wet steam.

SOL

Given,

$p_1 = 40$ bar.

$s_1 = 6.069$ kJ/kg-K $= s_2$.

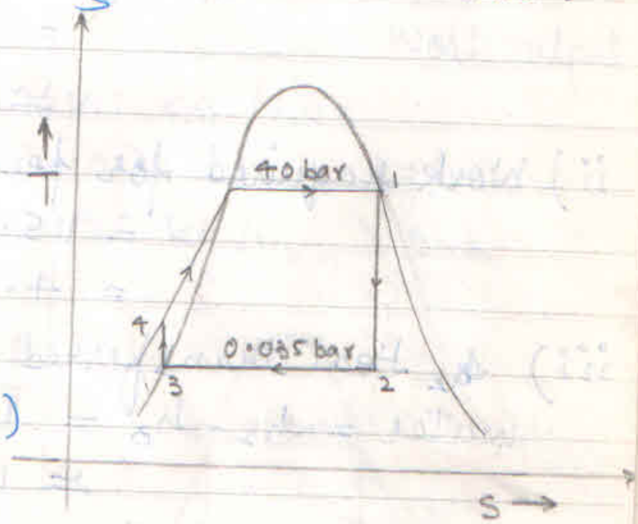
$s_2 = s_f + x_2 s_{fg}$.

$p_2 = 0.035$ bar

$s_2 = 0.391 + x_2(8.132)$

$6.069 - 0.391 = x_2(8.132)$

$\Rightarrow x_2 = 0.698$



$\therefore$, $h_2 = h_f + x_2 h_{fg}$

$= 111.8 + (0.698)(2438.6)$

$\Rightarrow h_2 = 1814.5017$ kJ/kg & $h_3 = 111.8$ kJ/kg

Also, $h_1 = 2800.3$ kJ/kg.

$\Rightarrow h_2' = 2026.888$ kJ/kg

Also, $\eta_{isen} = \frac{h_3 - h_4'}{h_3 - h_4} = 0.78$ (given)

$\Rightarrow h_3 - h_4' = (0.78)(h_3 - h_4)$

$h_4' = h_3 - (0.78)(h_3 - h_4)$

$= 3158.7 - (0.78)[3158.7 - 2185.57]$

$\Rightarrow h_4' = 2399.6586$ kJ/kg

Now, work output, $W = (h_1 - h_2') + (h_3 - h_4')$

$= (3095.1 - 2026.888)$

$+ (3158.7 - 2399.6586)$

$= 268.212 + 759.0414$

$\therefore W = 1027.2534$ kJ Ans

Heat supplied, $Q = (h_1 - h_6) + (h_3 - h_2')$

$= (3095.1 - 115.8085) + (3158.7 - 2026.888)$

$= 2979.2915 + 331.812$

$\therefore Q = 3311.1$ kJ Ans

Now, Efficiency, $\eta = \frac{W}{Q} = \frac{1027.2534}{3311.1} = 0.3102$

$\Rightarrow \eta = 31.02\%$ Ans

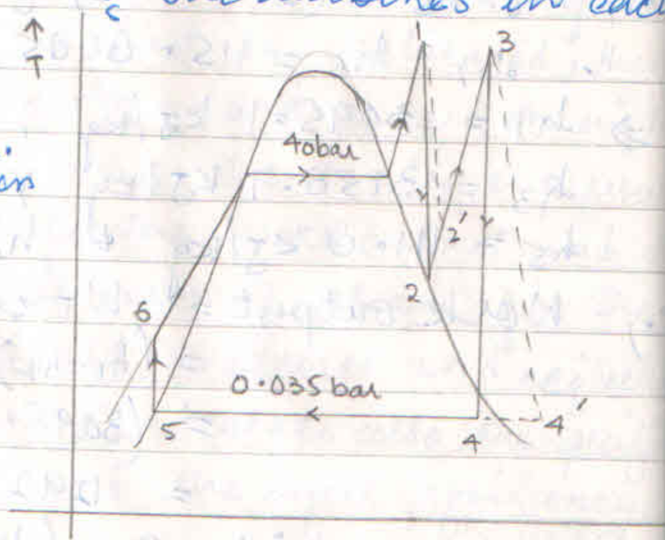
And, specific steam consumption (SSC) $= \frac{1}{W} \times 3600$

$= \frac{1}{1027.2534} \times 3600 = 3.5045$

$\Rightarrow SSC = 3.5$ kg/kWh Ans

Q8.4→ If the expansion in the Turbines of Problem 8.3 have isentropic efficiencies of 84% and 78% respectively, in the first and second stages. Calculate the work output and the heat supplied per kilogram of steam, the cycle efficiency, and the specific steam consumption.

Compare the efficiencies and specific steam consumptions obtained from problem 8.1, 8.2, 8.3, 8.4. Compare also the wetness of the steam leaving the turbines in each case.



SOL.

From previous question

$h_1 = 3095.1$ kJ/kg

$h_2 = 2775.8$ kJ/kg

$h_3 = 3158.7$ kJ/kg

$h_4 = 2185.57$ kJ/kg

$h_5 = 111.8$ kJ/kg

$h_6 = 115.8085$ kJ/kg.

We know, isentropic efficiency, $\eta_{isen} = \frac{\text{Actual enthalpy drop}}{\text{Isentropic enthalpy drop}}$

$$\eta_{isen} = \frac{h_1 - h_2'}{h_1 - h_2} = 0.84 \text{ (given)}$$

$$\Rightarrow h_1 - h_2' = 0.84(h_1 - h_2)$$

$$h_2' = h_1 - 0.84(h_1 - h_2)$$

$$= 3095.1 - 0.84[3095.1 - 2775.8]$$

We know, $Tds = dh - v dp$

During pump work $ds = 0$

$\therefore \; 0 = dh - v dp$

$\Rightarrow dh = v dp$.

$h_4 - h_3 = v_3 (p_4 - p_3)$

$\because v_3 = 0.001003 \, m^3/kg$.

$\therefore \; h_4 - h_3 = 0.001003 (40 - 0.035) \times 10^5 \times 10^{-3}$

$= 4.00848 \, kJ/kg$.

$h_4 = h_3 + 4.00848$

$= 111.8 + 4.00848$

$\Rightarrow h_4 = 115.808 \, kJ/kg$.

Hence, $h_1 = 2800.3 \, kJ/kg$.

$h_2 = 1814.5017 \, kJ/kg$.

$h_3 = 111.8 \, kJ/kg$

$h_4 = 115.808 \, kJ/kg$.

i) Work output neglecting the feed pump work

$= h_1 - h_2 = 2800.3 - 1814.5017$

$= 985.7983 \, kJ$

$\simeq 986 \, kJ$ Ans.

ii) Work required for feed pump $= h_4 - h_3$

$= 115.808 - 111.8$

$= 4.00848 \, kJ$ Ans

iii) Heat transferred to the condenser cooling water $= h_2 - h_3 = 1814.5017 - 111.8$

$\simeq 1703 \, kJ$ Ans

We know, $Q = m C_p \Delta T$

$C_p = 4.187 \, kJ/kg$

$\Delta T = 5.5 \, k$

$\therefore$, $1703 = m(4.187)(5.5)$

$\Rightarrow m = \frac{1703}{(4.187)(5.5)} = 73.952$ kg.

$\therefore$, $m = 73.952$ kg Ans

iv) heat supplied $= h_1 - h_4 = 2800.3 - 115.808$

$= 2684.492$ kJ Ans

v) Rankine efficiency, $\eta = \frac{\text{Work output}}{\text{heat supplied}}$

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

$= \frac{(2800.3 - 1814.5017) + (115.808 - 111.8)}{2800.3 - 115.808}$

$= \frac{986 + 4.00848}{2684.492}$

$= 0.3687$

$\Rightarrow \eta = 36.87\%$ Ans

vi) Specific steam consumption (ssc) $= \frac{1}{\text{Work output}}$

$= 0.00101$ kg/kWs

$= (0.00101)(3600)$ kg/kWh

$\therefore$, SSC $= 3.636$ kg/kWh Ans

(b) Now, $S_4 = 2.797$ kJ/kg-K

$= S_3$.

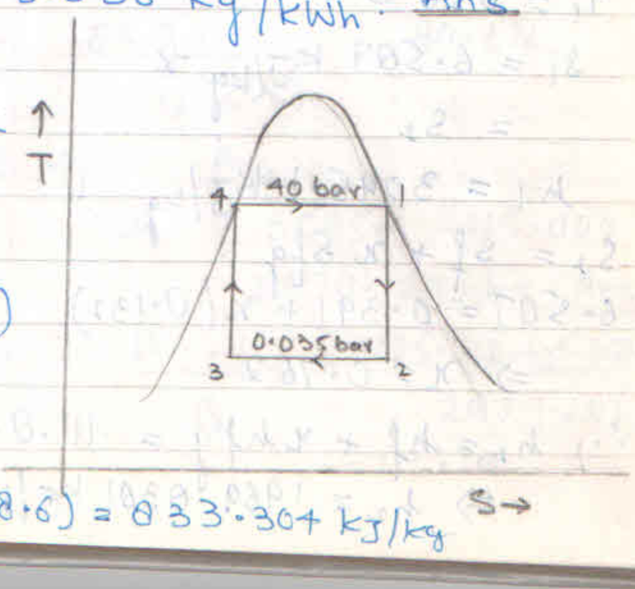
$S_3 = S_f + x_3 S_{fg}$

$2.797 = 0.391 + x_3(8.132)$

$\Rightarrow x_3 = 0.2959$.

$\therefore$ $h_3 = h_f + x_3 h_{fg}$

$= 111.8 + (0.2959)(2438.6) = 833.304$ kJ/kg



$\Rightarrow h_3 = 833.304$ kJ/kg

& $h_4 = 1087.4$ kJ/kg

Also, $h_1 = 2800.3$ kJ/kg

$h_2 = 1014.5017$ kJ/kg

Since $p_1 = 40$ bar $\Rightarrow T_1 = 250.3°C = 523.3K$

& $p_2 = 0.035$ bar $\Rightarrow T_2 = 26.69°C = 302.69K$

$\therefore \eta = \frac{T_1 - T_2}{T_1} = \frac{523.3 - 302.69}{523.3} = \frac{722.1151}{523.3}$

$= 0.4215746$

$\Rightarrow \eta = 42.15746\%$ Ans

$SSC = \frac{1}{722.11513} \times 3600 = 4.98$ kgs/kWh.

Ans

Students are advised to go through the lecture notes along with the following books

1. Engineering Thermodynamics by P K NAG

2. Applied Thermodynamics for Engineering Technologists by EASTOP & McCONKEY

In case of any typographic mistake, error or any difficulty, students are advised to call me on 9906763424,7006161837, hanief@nitsri.net .

students can call me for arranging video lectures

Students must complete this module within 5 days i.e before wednesday (6th May).

Two unsolved numerical have been solved from EASTOP (Prob. 1 &4) students are advised to attempt other problems from EASTOP and PK Nag.