University of Anbar

College of Engineering

Mechanical Eng. Department

Power Plants

Chapter one

Steam Power Plant

by

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Power Plants

A Power Plant may be defined as a machine or assembly of equipment that generates and delivers a flow of mechanical or electrical energy.

Classification of Power Plants

1. Conventional Power Plants

- Steam Turbine Power Plant
- Gas Turbine Power Plant
- Diesel Power Plant
- Nuclear Power Plant

2. Non-conventional Power Plants

- Wind Energy Power System
- Solar Thermal Power Plant
- Ocean Thermal Energy Conversion
- Biomass Energy Power System
- Geothermal Energy

Energy sources

1. Renewable energy sources

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- Salar energy
- Wind energy

- Tidal energy
- Flowing of stream of water

2. Non-renewable energy Sources

- Coal
- Petroleum
- Natural gas
- Nuclear power

Fuels

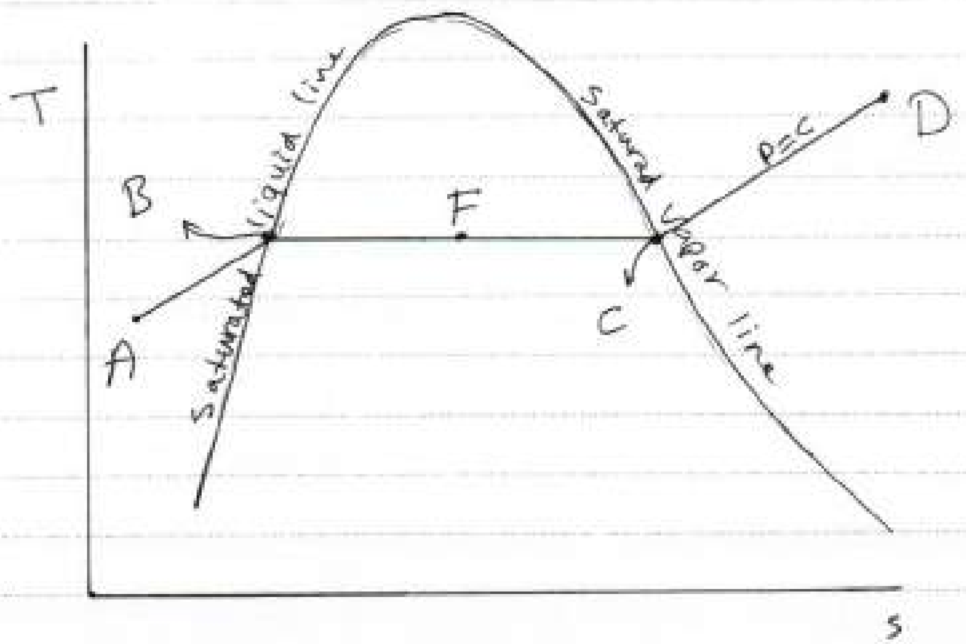
It is defined as any material which when burn will produce heat. Various fuels commonly used are as follows :-

- Solid fuels (Wood , lignite , Coal)
- Liquid fuels (Petroleum)
- Gasous fuels (Natural gas)

Definitions :~

- subcooled water (A) :~ It refers to water existing below its normal saturation temp.

- Saturated liquid (B) :~ a liquid that exists at the saturation temperature or boiling Point that corrosponding to the existing pressure. If any energy is added to the liquid and the pressure remain constant, some of the liquid would boil. During boiling, temp. will remain constant.



- Saturated Vapor (C) :~

A vapor that exists at the saturation temprature that corrosponding to the existing pressure. If any energy was removed while the pressure is constant, some the vapour would condense and the temp. reamins constant.

- Superheated Steam (D) :~

Steam existing at a temp. above the saturation temprature that corrosponding to the existing pressure. The removal of a small amount of energy will not cause the vapor to dendence, its temprature will just decrease.

- Wet Steam (F) :~

It is a mixture of ~~water~~ liquid and vapor. If additional heat is added to the wet steam at constant pressure, the temprature remains constant untill all liquid is evaporated (saturated steam)

- Dryness Fraction:-

It is a ratio of vapor mass to the total mass.

$$X = \frac{m_v}{m_v + m_L}$$

Where m_v: vapor mass
m_L: liquid mass

- Moisture Content:-

It is the ratio of liquid mass to the total mass

$$y = \frac{m_L}{m_L + m_v}$$

$$\therefore \quad X + y = 1$$

h_f : Enthalpy of water (KJ/Kg.K)

h_g : Enthalpy of dry steam (KJ/Kg.K)

h_x : Enthalpy of wet steam (KJ/Kg.K)

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

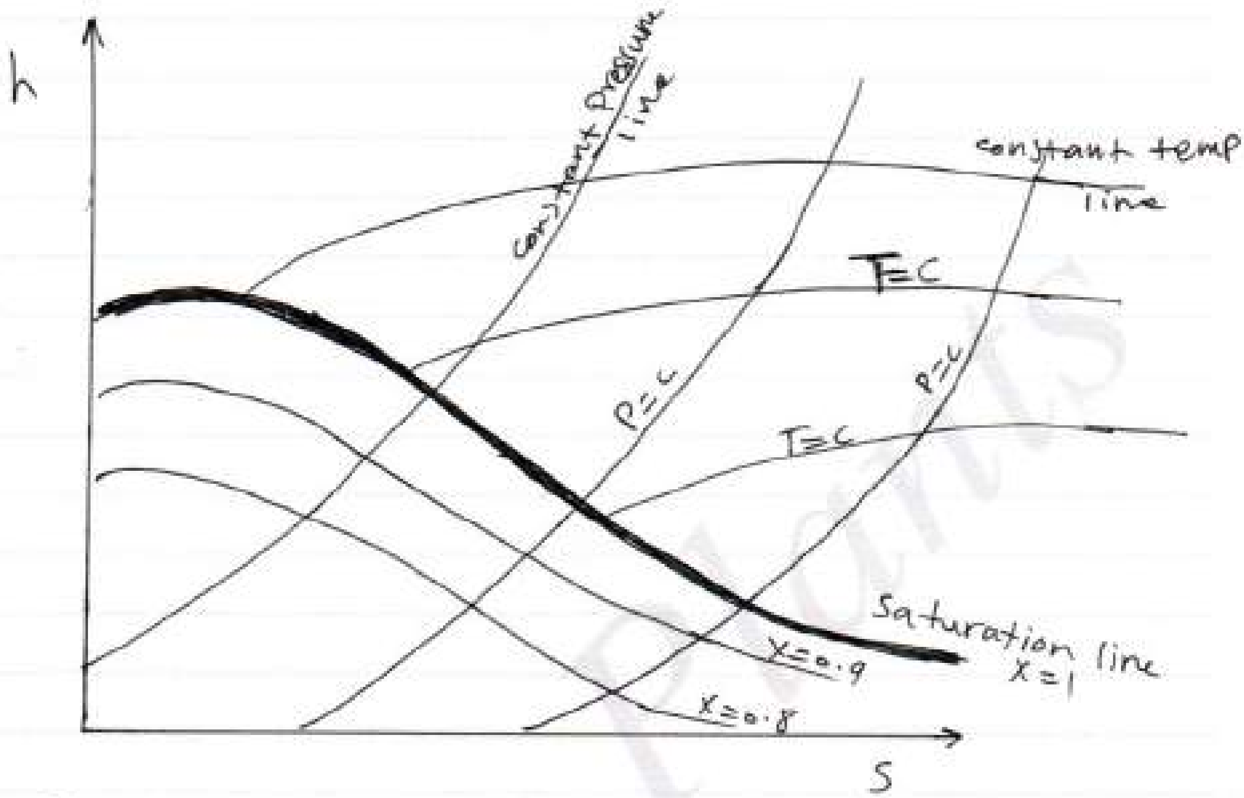
where $h_{fg} = h_g - h_f$

So, $$x = \frac{h_x - h_f}{h_g}$$

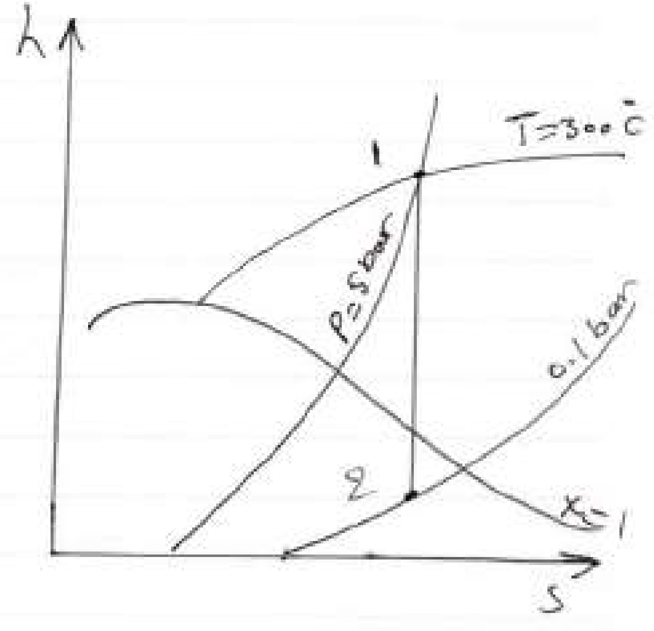
* h-s daigram (Mollier Chart)

The Mollier Chart is a chart on which enthalpy (h) versus entropy (s) is plotted. The chart contains a series of constant tempreture lines, a series of constant pressure, a series of constant quality lines.

The Mollier chart is used when quality (x) is greater than 50% and for ~~exp~~ superheated steam



Ex/ Find the enthalpy drop when steam expand from 5 bar, 300 °C to 0.1 bar Using Molier chart



$h_1 = 3070$ KJ/Kg

$h_2 = 2370$ KJ/Kg

$\Delta h = h_1 - h_2$

$= 3070 - 2370$

$= 700$ KJ/Kg

Steam Power Plant Cycles

Steam is most common working fluid used in vapor power plant cycles because of its many desirable characteristics such as :-

- low cost
- availability
- high enthalpy of vaporization

Steam power plant are commonly referred to as coal plant, nuclear plant, or natural gas plant depending on type of fuel used to apply heat to the steam.

- Vapor Power Cycles classifications :-

  - Carnot vapour Power Plant
  - Rankine cycle
  - Reheat cycle
  - Regenerative cycle

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- Performance Parameters:

  - Thermall efficiency:

    Thermall efficiency is the parameter which gauges the extent to which the energy input to the device is converted to net work output from it.

$$\text{Thermal efficiency } \eta = \frac{\text{Net Work in cycle}}{\text{Heat added in cycle}}$$

  - Work ratio

    It refer to the ratio of net work to the Positive work.

$$\text{Work ratio} = \frac{W_{net}}{W_{turbine}}$$

  - Specific Steam Consumption (SSC):

    It indicates the steam requirment per unit Power output, and it is given in Kg/Kw.h

$$SSC = \frac{3600}{W_{net}} \text{, Kg/KWh}$$

* Carnot vapour Power cycle

Carnot cycle can be defined as an ideal cycle having highest thermodynamic efficieney.

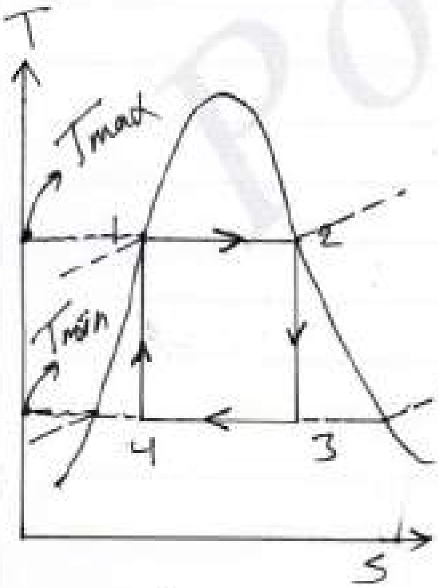
Arrangment proposed for using carnot vapour Power cycle is as follows :-

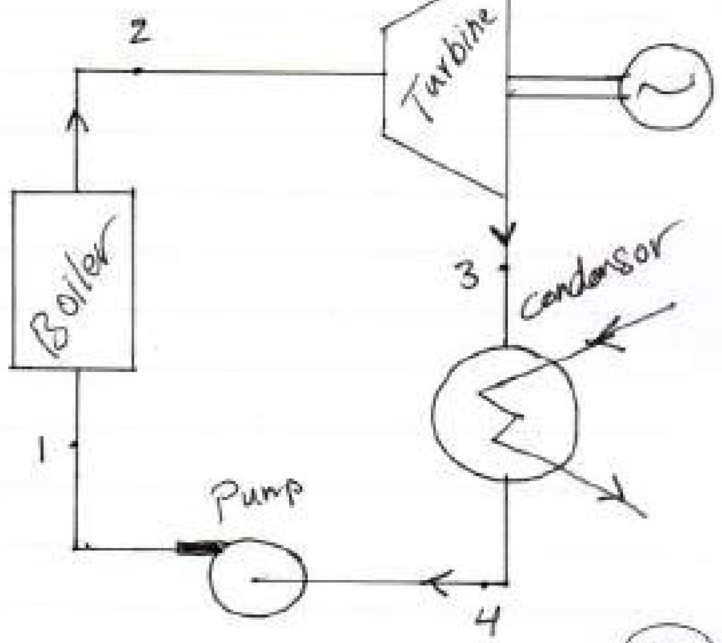
1-2 = Reversible isothermal heat addition in the boiler

2-3 = Reversible adiabatic expansion in steam turbine

3-4 = Reversible isothermal heat rjection in the condensor.

4-1 = Reversible adiabatic compression or Pumping in feed water pump.





Assuming steady-state flow processes in the cycle and neglecting changes in kinetic and potential energy:

$$\text{Thermal efficiency} = \frac{\text{Net work}}{\text{Head added}}$$

Net work = Turbine work − Pumping work

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

Head added $= h_2 - h_1$

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

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

$Q_{rejected} = h_3 - h_4$

$$\eta_{carnot} = 1 - \frac{Q_{rej}}{Q_{add}}$$

Also, heat added and rejected may be given as function of temp. and entropy as follows :-

$$Q_{add} = T_1 * (S_2 - S_1)$$

$$Q_{rej} = T_3 * (S_3 - S_4)$$

$$S_1 = S_4 \quad , \quad S_2 = S_3$$

Therefore, Substituting values,

$$\boxed{\eta_{Carnot} = 1 - \frac{T_3}{T_1}}$$

$$T_1 = T_2 \quad , \quad T_3 = T_4$$

* Limitations of Carnot cycle

Although Carnot cycle is simple thermodynamically and has the highest thermal efficiency, it is difficult to operate in practice because of the following reasons :-

1. It is diffucult to compress a wet vapour isentropically to the saturated state as required by the process 4 - 1

2. It is difficult to control the quality of the condensate coming out of the condensor, so that the state (4) is hard to obtain.

3. It is difficult for a pump to handle wet mixture ~~steam~~ which undergoes simultanous change in ~~its~~ phase as its pressure increases

Ex/ In a steam power plant, the steam supply is at 15 bar and dry saturated. The condenser pressure is 0.4 bar. Calculate the carnot efficiency

Solutions:~

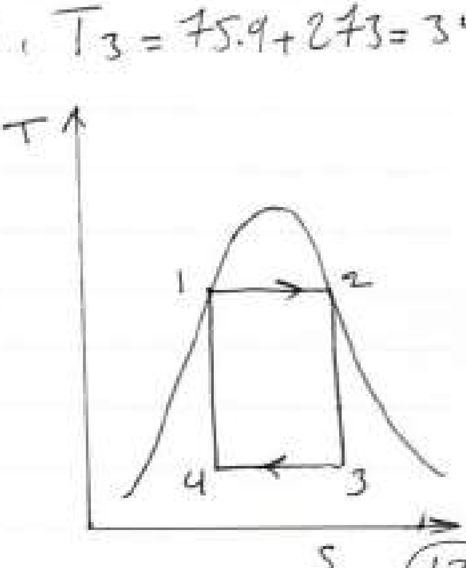
From steam table:

At 15 bar: $t_s = 198.3^\circ C = T_1 = T_2$, $T_1 = 198.3 + 273 = 471.3\ K$

At 0.4 bar: $t_s = 75.9^\circ C = T_3 = T_4$, $T_3 = 75.9 + 273 = 348.9\ K$

$$\eta_{carnot} = 1 - \frac{T_{min}}{T_{max}}$$

$$= 1 - \frac{348.9}{471.3} = 25.9\ \%$$



* Rankine Cycle

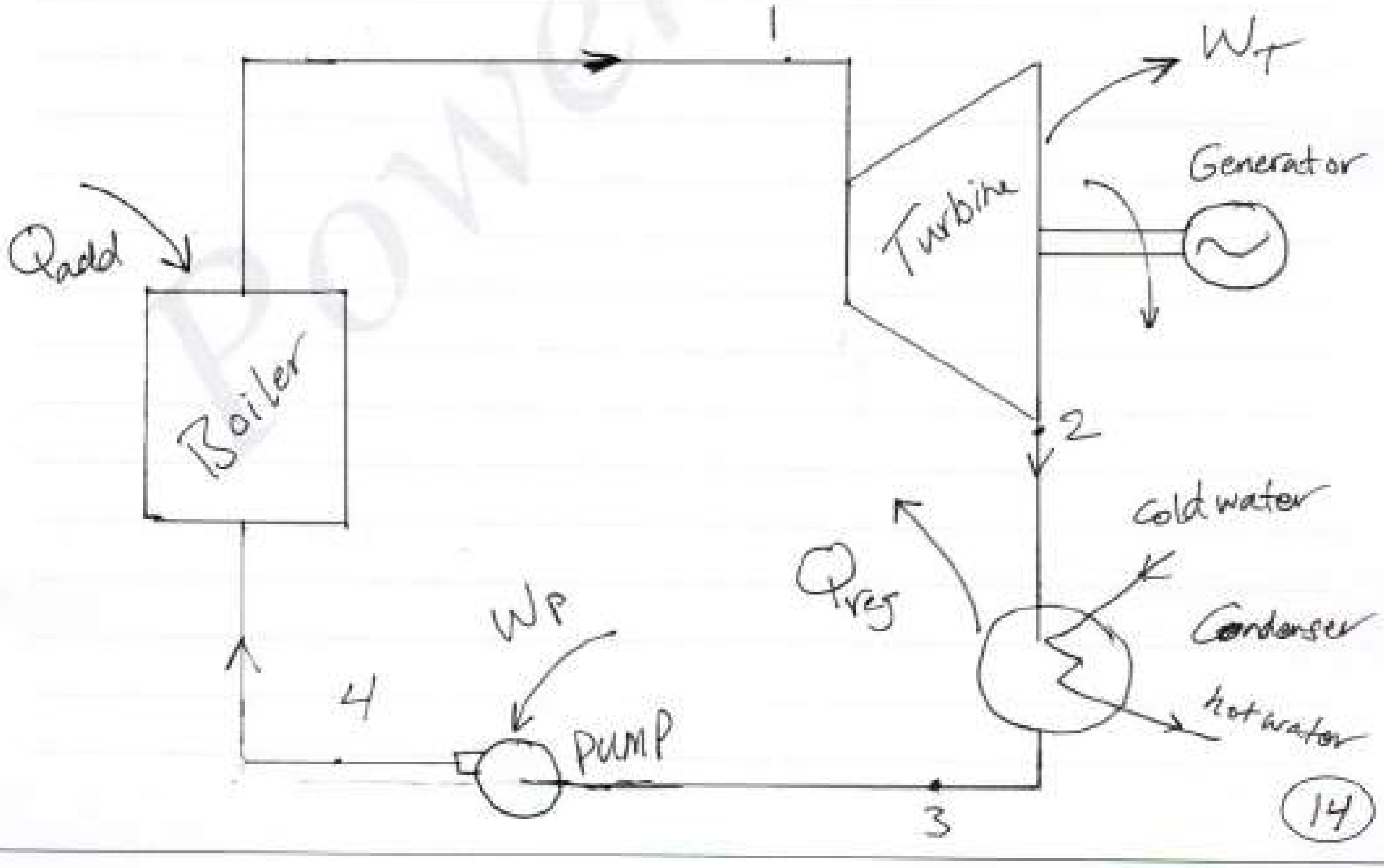
Rankine cycle is a thermodynamic cycle derived from Carnot vapour power cycle for overcoming its limitations. Rankine cycle has the following thermodynamic processes:

1-2 = Reversible adiabatic expansion in the turbine

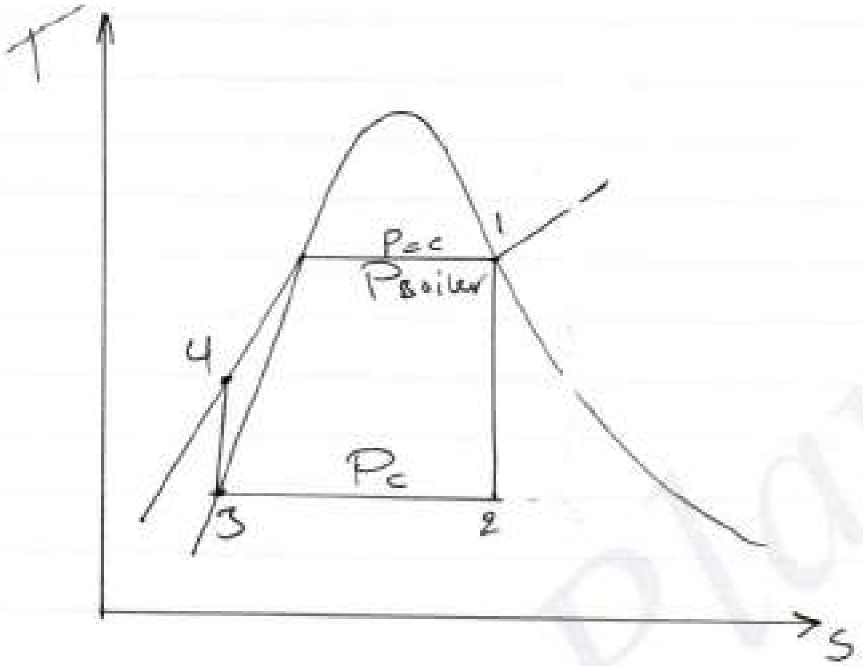
2-3 = Constant-pressure transfer of heat in the Condenser.

3-4 = Reversible adiabatic pumping process in the feed pump

4-1 = Constant-pressure transfer of heat in the boiler.



Power Plants



By applying steady flow energy equation to boiler, turbine, condenser and pump, we get:-

(i) For boiler

$$\frac{dE_{cv}}{dt} = \dot{Q} - \dot{W} + \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gz_i\right) - \dot{m}_o \left(h_o + \frac{V_o^2}{2} + gz_o\right)$$

Assumptions:-

* Steady state $\left(\frac{dE_{cv}}{dt} = 0\right)$
* No change in velocity $\left(\frac{V_i^2}{2} - \frac{V_o^2}{2} = 0\right)$
* No change in elevation $(gz_i - gz_o = 0)$

$\therefore$

$$0 = \dot{Q} - \dot{W} + \dot{m}_i h_i - \dot{m}_o h_o \quad , \dot{m}_i = \dot{m}_o$$

$$\boxed{0 = q - w + h_i - h_o} \text{ ———— equ (1)}$$

In boiler, there is no work added or rejected
$W = 0$

$$q_{add} = h_o - h_i$$

or $\boxed{q = h_1 - h_4}$ (for boiler Q_{add})

(ii) For turbine

Same steps are applied as we did above.

for turbine, there is heat added or rejected ($Q = 0$)

from equ (1)

$$0 = -W + h_i - h_o$$

or

$$\boxed{W_{turbine} = h_1 - h_2} \text{ (for turbine } W_t\text{)}$$

(iii) For Condenser

In condenser, there is no work added or rejected ($W=0$)

So, from equation (1)

$0 = -q + h_i - h_o$ (the signal(-) is accounted fro heat rejection)

Or $\boxed{Q_{cond.} = h_2 - h_3}$ (for Condenser Q_{rej})

(iv) For feed pump

In pump, there is no heat added or rejected ($Q=0$)

So, from equ. (1)

$0 = +W + h_i - h_o$

the signal (-) was changed to (+) beacuse the pump work is input

So

$W_{pump} = h_o - h_i$

$\boxed{W_{pump} = h_4 - h_3}$ (for pump W_p)

Now, efficiency of Rankine cycle is given by :-

$$\eta_{Rankine} = \frac{W_{net}}{Q_{add}}$$

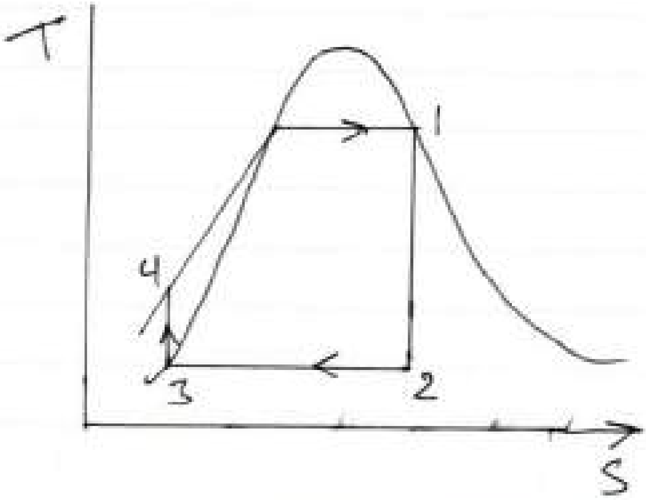
$$= \frac{W_T - W_P}{Q_{add}}$$

$$\boxed{\eta_{Rankine} = \frac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_4)}}$$

$$W_{pump} = v\,(P_{Boiler} - P_{Cond.}) = h_4 - h_3$$

Ex/ The same example above, Calculate Rankine efficiency cycle ?

Solution:-



From steam table

at 15 bar $\quad h_1 = h_g = 2789.9$ KJ/Kg
$\quad S_1 = S_g = 6.4406$ KJ/Kg K

at 0.4 bar $\quad h_3 = h_f = 317.7$ KJ/Kg
$\quad h_{fg} = 2319.2$ KJ/Kg
$\quad S_f = 1.0261$ KJ/Kg K
$\quad S_{fg} = 6.6448$ KJ/Kg K
$\quad v = 0.001$ m^3/Kg

$S_1 = S_2$ (steam expands isentropically)

$$S_2 = S_{f_2} + X_2 S_{fg}$$
$$6.6406 = 1.0261 + X_2 * 6.6448$$

$$X_2 = 0.815$$

$$h_2 = h_f + X h_{fg}$$
$$= 317.7 + 0.815 * 2319.2$$
$$= 2207.8 \text{ KJ/Kg}$$

$$v(P_2 - P_1) = h_4 - h_3$$

1 bar = 100 KPa

$$0.001(1500 - 40) = h_4 - 317.7$$
$$h_4 = 319.16 \text{ KJ/Kg}$$

$$W_{done} = W_T - W_C = (h_1 - h_2) - (h_4 - h_3)$$

$$= (2789.9 - 2207.8) - (319.16 - 317.7)$$

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$W_{done} = 580.64$ KJ/Kg

$Q_{add} = h_1 - h_4$

$= (2789.9 - 319.16)$

$= 2470.74$ KJ/Kg

$$\eta_{Rankine} = \frac{W_{done}}{Q_{add}}$$

$$= \frac{580.64}{2470.74} = 23.5\ \%$$

Irreversibilities and losses in Rankine Cycle

In Rankine cycle, the major irreversibility is encountered during the expansion through turbine. Irreversibities in turbine significantly reduce the expansion work.

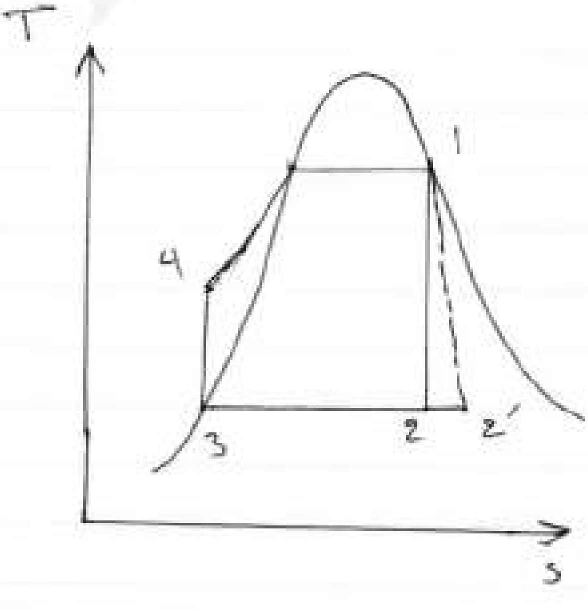
This deviation of expansion from ideal to actual process can be accounted for by isentropic turbine efficiency.

Ideal expansion in steam turbine is shown by 1-2 on T-s representation. Actual expansion process is shown by 1-2′

$$\eta_{isen,t} = \frac{W_{t,actual}}{W_{t,ideal}}$$

$$= \frac{W_{1-2'}}{W_{1-2}}$$

$$W_{1-2'} < W_{1-2}$$



$$\boxed{\eta_{isen.,t} = \frac{h_1 - h_{2'}}{h_1 - h_2}}$$

Ex/ A steam power plant operates between a boiler pressure of 42 bar and a condenser pressure of 0.035 bar. Calculate for these limites the cycle efficiency

(i) for a Rankine cycle with dry saturated steam at entry to the turbine

(ii) for Rankine cycle of (i) when the expansion process has an isentropic efficiency of 80%

Solution:

(i) From steam table

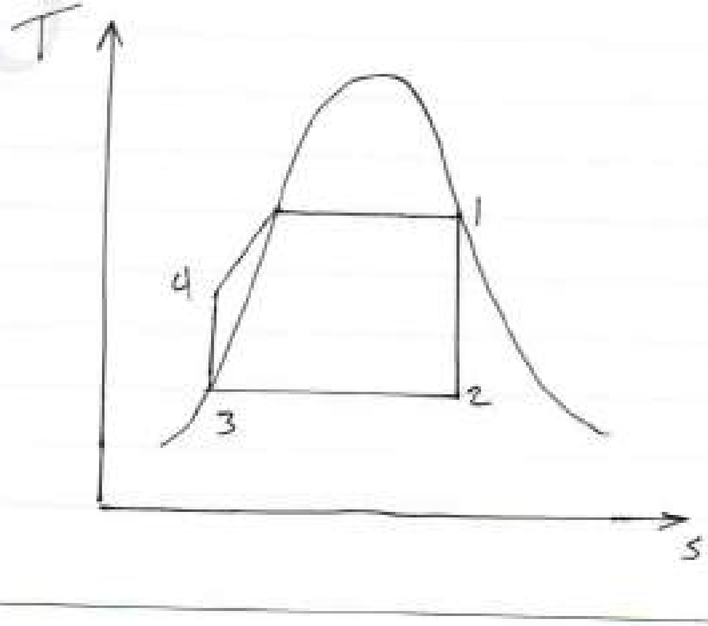
At 42 bar $T_{saturation} = 253.2\,^{\circ}C$

$h_{fg} = 1698\ KJ/Kg$

$h_1 = h_g = 2800\ KJ/Kg$

$S_1 = S_g = 6.049\ KJ/Kg\,K$

$S_1 = S_2$



At 0.035 bar $\quad S_f = S_3 = 0.391$ KJ/kgK
$S_{fg} = 8.13$ KJ/kgK

$S_2 = 6.049 = S_f + X_2 S_{fg}$
$6.049 = 0.391 + X_2 * 8.13$

$X_2 = 0.696$

$h_2 = h_f + X_2 h_{fg}$

At 0.035 bar $\quad h_f = 112$ KJ/kg $= h_3$
$h_{fg} = 2438$ KJ/kg

$h_2 = 112 + 0.696 * 2438$
$= 1808$ KJ/kg

$v(P_2 - P_1) = h_4 - h_3$
$0.001(4200 - 3.5) = h_4 - 112$

$h_4 = 116.2$ KJ/kg

$W_{net} = W_t - W_c$

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

$= (2800 - 1808) - (116.2 - 112)$

$= 987.8$ KJ/kg

$Q_{add} = h_1 - h_4$

$= 2800 - 116.2$

$= 2683.8$ kJ/kg

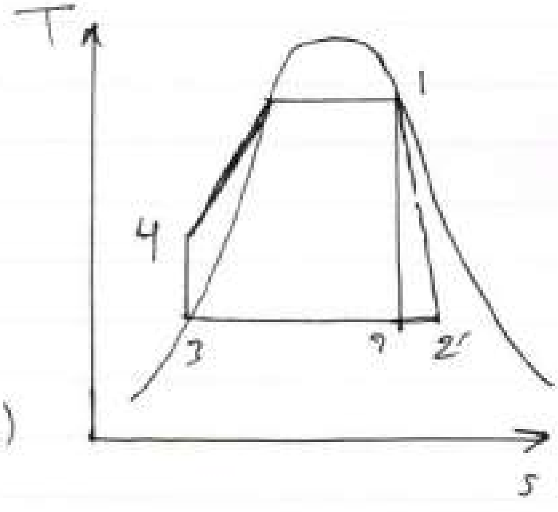
$$\eta_{Rankine, ideal} = \frac{W_{net}}{Q_{add}} = \frac{987.8}{2683.8} = 36.8\%$$

(ii) When turbine isen. efficiency is 80%

$$\eta_{isen, t} = \frac{h_1 - h_2'}{h_1 - h_2}$$

$$0.8 = \frac{2800 - h_2'}{2800 - 1808}$$

$h_2' = 2006.4$ kJ/kg



$W_{net} = (h_1 - h_2') - (h_4 - h_3)$

$= (2800 - 2006.4) - (116.2 - 112)$

$= 789.4$ kJ/kg

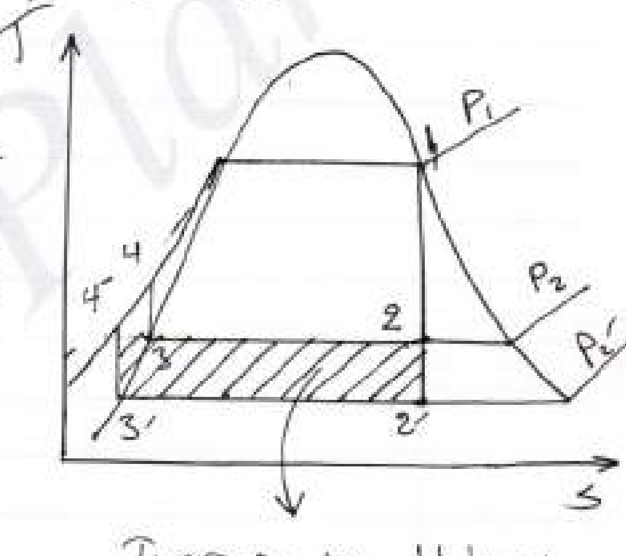
$$\eta_{Rankine, actual} = \frac{789.4}{2683.8} = 29.4\%$$

Increasing Rankine Cycle efficiency

Steam power plants are used for the Production of most of the electric power in the word, therefore, small increase in thermal efficiency can mean large saving of fuel requirment

1. Lowering the Condenser pressures

The dashed area in this daigram represents the increase in W_{net}.
The heat input also increases (4-4') but this increase is very small



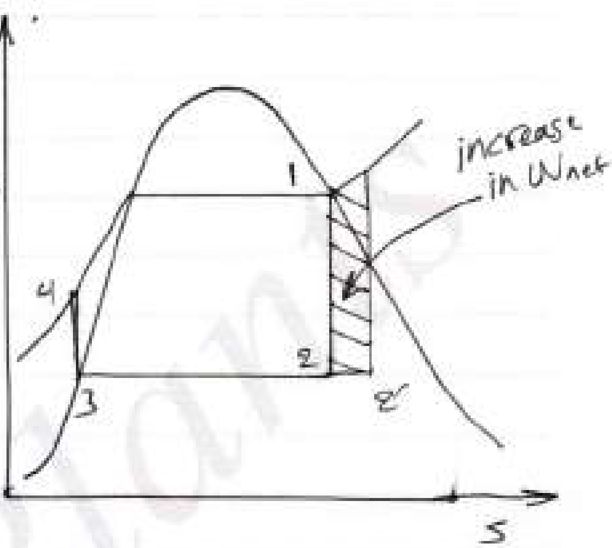
Dis advantage

- It increases the moisture content of the steam at the final stages of the turbine.

The large quantities of moisture are highly undesirable because it erodes the turbine blades

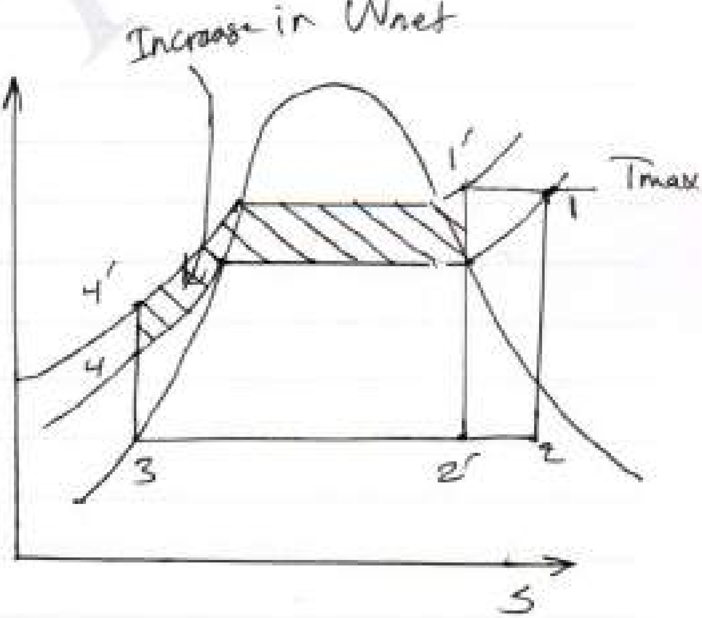
2. Superheating the steam to high temperature :

Both the W_{net} and heat input increase as a result of superheating. The overall effect is an increas in η



3. Increasing the boiler pressure

It raises the average temp at which heat is added to the steam and thus raises the η.



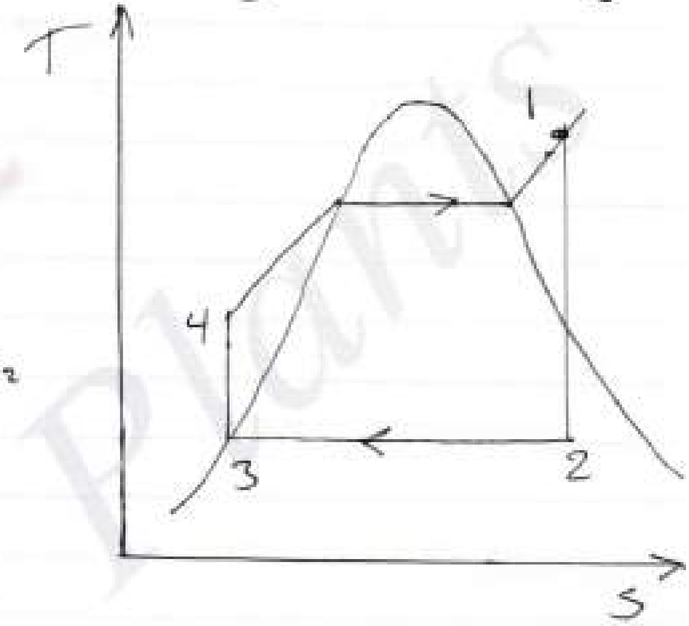
For a fixed turbine inlet temp., the cycle shifts to the left and moisture content of the steam at the turbine exit increases.

This side effect can be corrected by reheating the system

Power Plants

EX / In steam power plant, steam enters turbine at conditions of 42 bar, 500°C and exits at 0.035 bar. Calculate efficiency of the cycle?

Solution:~



From steam table
At 42 bar, 500°C
$h_1 = 3442.6$ KJ/Kg
$S_1 = 7.066$ Kg/Kg $= S_2$

At 0.03 bar
$h_f = 112$ KJ/Kg
$h_{fg} = 2438$ KJ/Kg
$S_f = 0.391$ KJ/KgK
$S_{fg} = 8.13$ KJ/KgK

$S_2 = S_f + X_2 S_{fg}$
$7.066 = 0.391 + X_2 * 8.13$

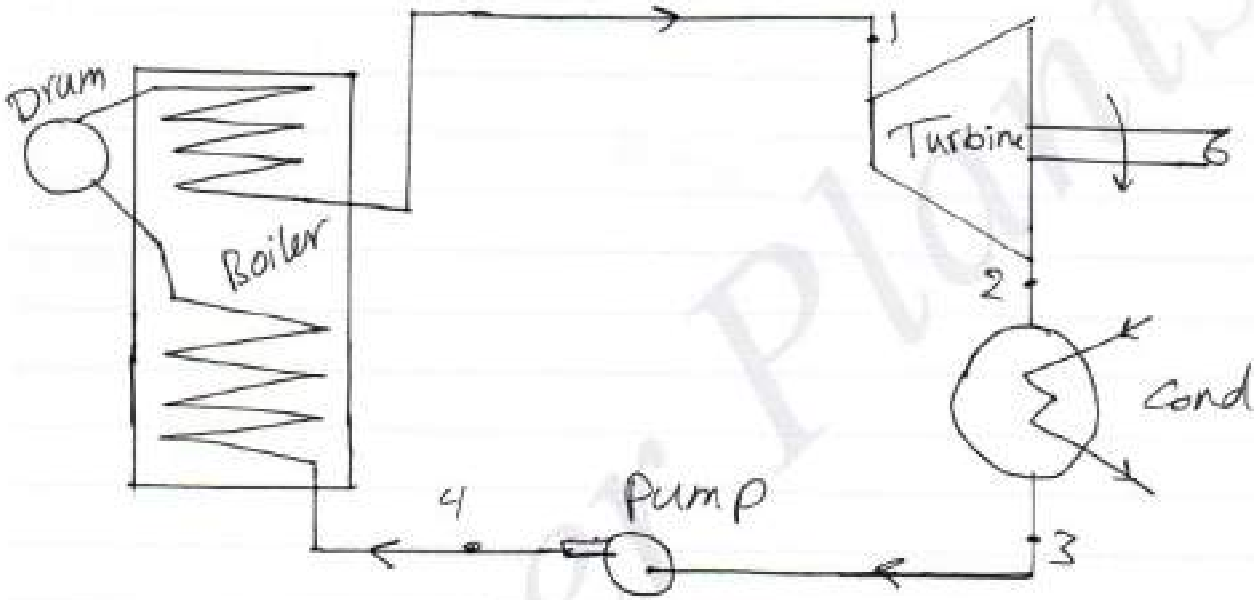
$X_2 = 0.821$

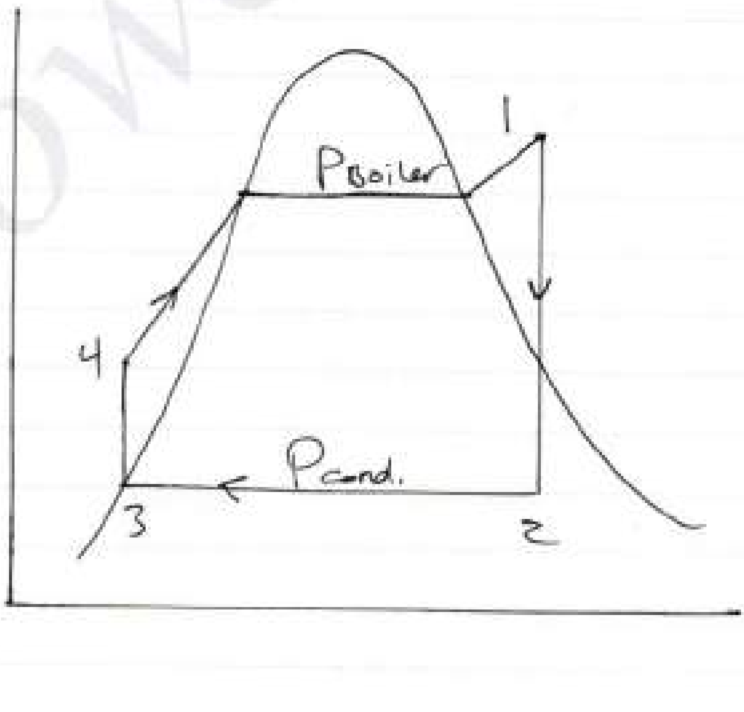
$h_2 = h_f + X_2 h_{fg}$
$= 112 + 0.821 * 2438$
$= 2113$ KJ/Kg

$h_3 = h_f = 112$ KJ/Kg

* Rankine Cycle with superheating

The dry saturated steam from the boiler drum is passed through a second bank of smaller bore tubes within the boiler.





$v(P_2 - P_1) = h_4 - h_3$

$0.001\,(4200 - 3.5) = h_4 - 112$

$h_4 = 116.2$

$$W_{net} = W_t - W_c$$
$$= (h_1 - h_2) - (h_4 - h_3)$$
$$= (3442.6 - 2113) - (116.2 - 112)$$
$$= 1325.4 \text{ KJ/Kg}$$

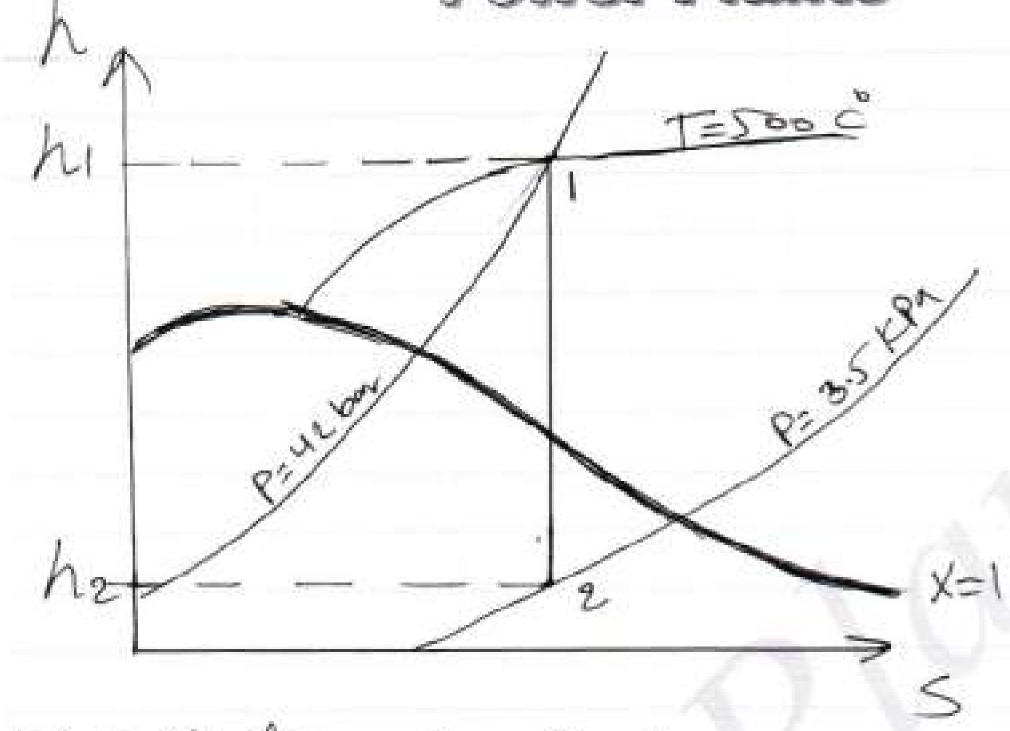
$$Q_{add} = h_1 - h_4$$
$$= 3442.6 - 116.2$$
$$= 3326.4 \text{ KJ/Kg}$$

$$\eta_{Rankine} = \frac{W_{net}}{Q_{add}}$$

$$= \frac{1325.4}{3326.4} = 39.8\,\%$$

Or,

If we use Mollier Chart to estimate the values of enthalpy instead of steam stable, the results will be as follows:-



Directly from the chart

$h_1 = 3450$ KJ/Kg
$h_2 = 2120$ KJ/Kg

$h_3 = 112$ from steam-table
$h_4 = 116.2$

$W_{net} = (3450 - 2120) - (116.2 - 112)$
$= 1325.8$ KJ/Kg

$Q_{add} = 3450 - 116.2$
$= 3333.8$

$\eta = \dfrac{1325.8}{3333.8} = 39.7\%$

you can notice that the diffrence between the two η are very small = 0.1 % which is acceptable

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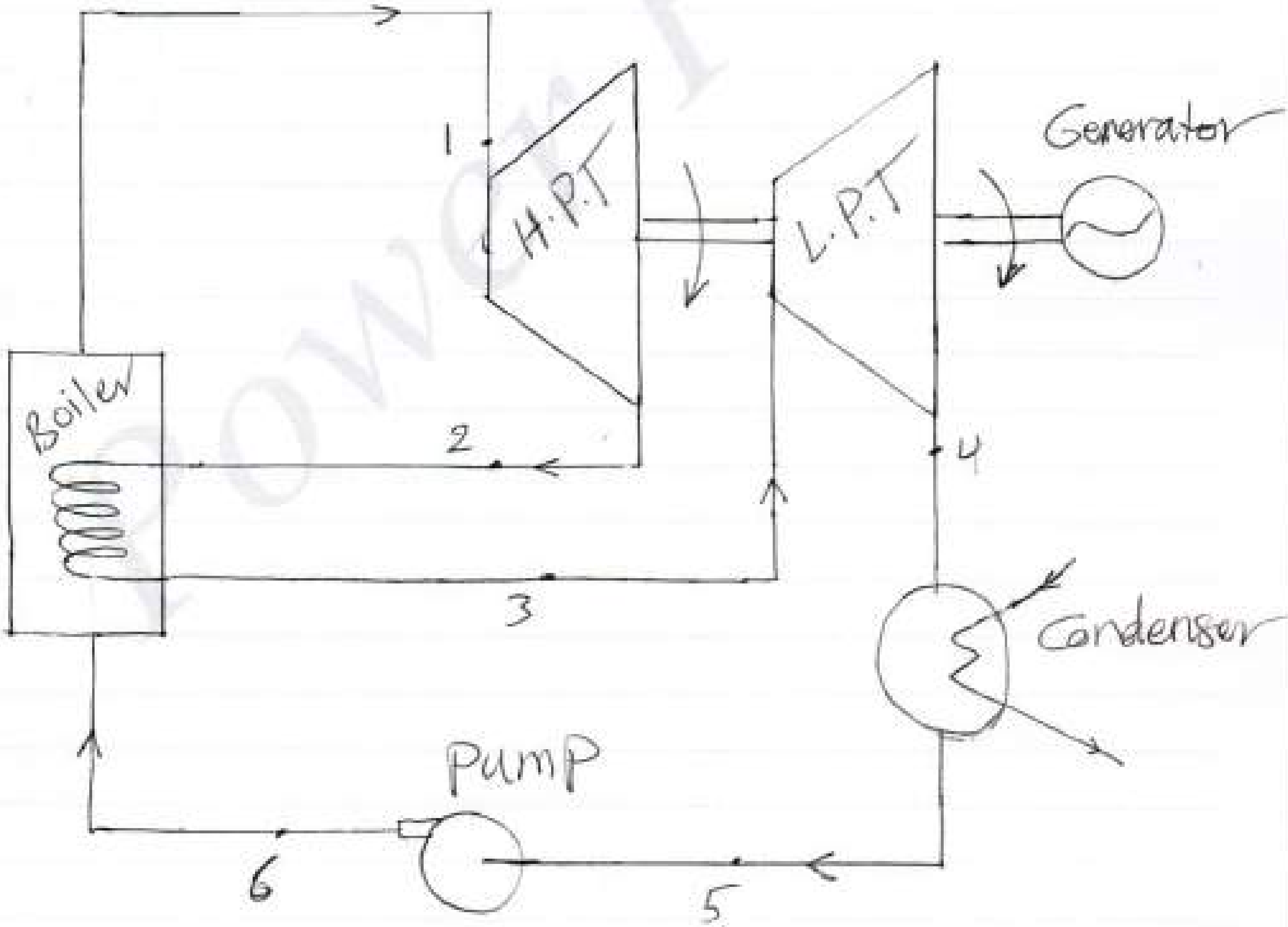
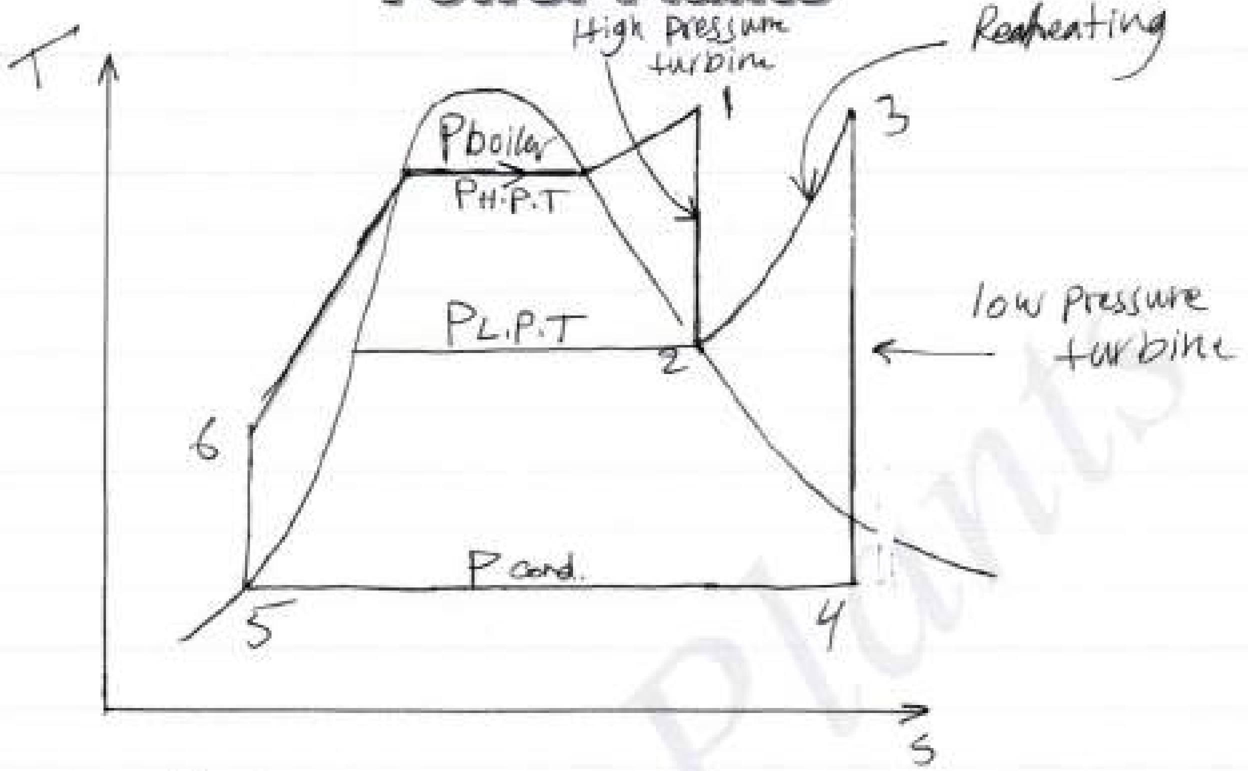
* Reheat Rankine cycle

It is desirable to increase the temprature at which heat is supplied to the steam, and also to keep the steam as dry as possible in the lower stages of the turbine.

The ideal Rankine cycle differs from the simple ideal Rankine cycle in that the expansion process takes place in two stages.





(1-2): high-pressure turbine stage, Steam is expanded isentropically to an intermediate pressure

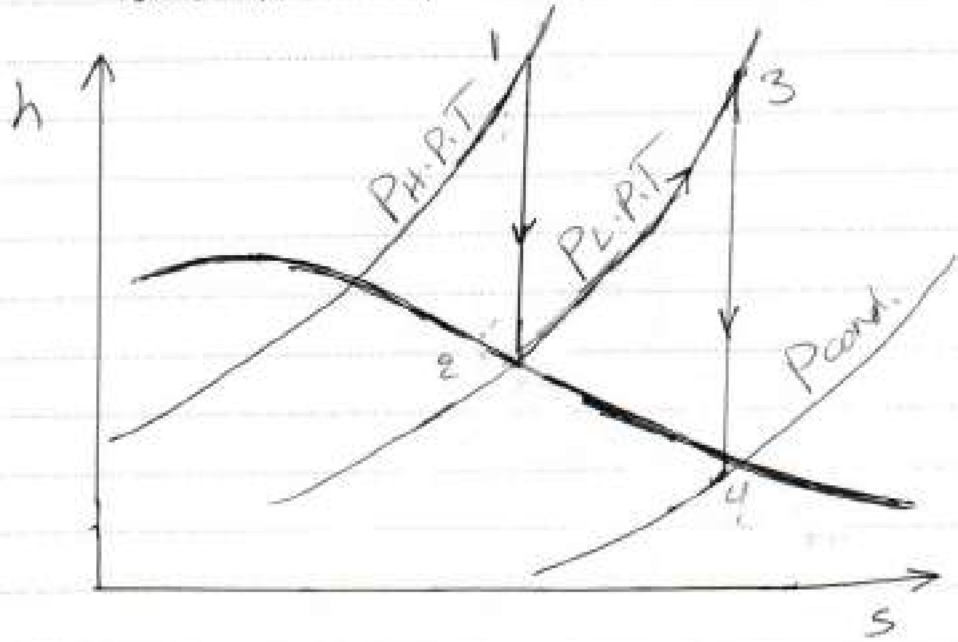
(2-3): Intermediate pressure, Steam is sent back to the boiler where it is reheated at Constant pressure

(3-4): Low-pressure turbine stage, steam is expanded isentropically to the condenser pressure.

(4-5): Condenser pressure, Steam is cooled down to saturated liquid

(5-6): Pump, water is pumped from condenser pressure to boiler Pressure.

(6-1): Boiler pressure, heat is added to water and turn it to superheated steam.



Thermodynamic analysis of reheat cycle as shown on T-S and h-S representation are carried out as follows:-

$$W_{H.P.T} = h_1 - h_2$$

$$W_{L.P.T} = h_3 - h_4$$

$$W_{pump} = h_6 - h_5$$

$$W_{net} = (W_{H.P.T} + W_{L.P.T}) - W_{pump}$$

$$\boxed{W_{net} = (h_1 - h_2) + (h_3 - h_4) - (h_6 - h_5)}$$

$$Q_{add} = Q_{boiler} + Q_{reheating}$$

$$\boxed{Q_{add} = (h_1 - h_6) + (h_3 - h_2)}$$

$$\eta = \frac{W_{net}}{Q_{add}}$$

$$\boxed{\eta_{reheating} = \frac{(h_1 - h_2) + (h_3 - h_4) - (h_6 - h_5)}{(h_1 - h_6) + (h_3 - h_2)}}$$

output power of the plant $= \dot{m}_s \cdot W_{net}$

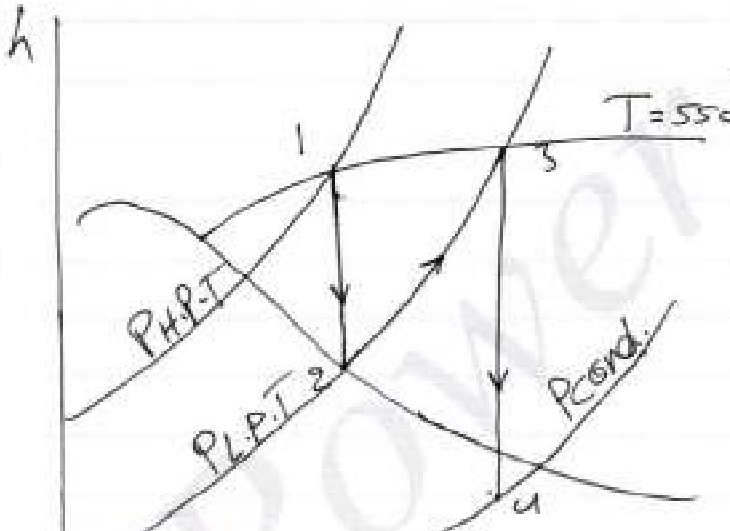
$\dot{m}_s$ = mass flow rate of the steam

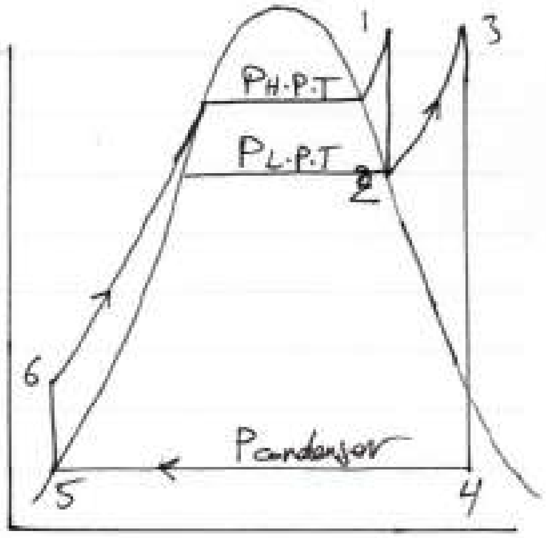
$$\boxed{P_{out} = \dot{m}_s \cdot W_{net}}$$

EX/ In a steam power plant, steam enters HPT at 150 bar and 550°C and leaves as saturated vapor. the Condenser Pressure is 0.1 bar. Assume that the reheating occures up to the original tempressure Determine

a) work done
b) heat added
c) out put Power if the mass flow rate of steam is 50 kg/s
d) Thermal efficiency of the cycle

Solution:-





From Mollier chart

At 150 bar and 550 °C

$h_1 = 3455$ KJ/Kg
$h_2 = 2785$ KJ/Kg

reheating pressure is 1.25 Mpa

$h_3 = 3585$ KJ/Kg

$h_4 = 2465$ KJ/Kg

From Steam table

At 0.1 bar $h_5 = 191.83$ KJ/Kg

$$v(P_2 - P_1) = h_6 - h_5$$
$$0.001(15000 - 10) = h_6 - 191.83$$

$$h_6 = 206.82 \text{ KJ/Kg}$$

$$W_T = W_{H.P.T} + W_{L.P.T}$$

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

$= (3455 - 2785) + (3585 - 2465)$

$= 1790 \text{ KJ/Kg}$

$W_P = h_6 - h_5$

$= 206.82 - 191.83$

$= 15 \text{ KJ/Kg}$

$W_{done} = W_T - W_P$

$= 1790 - 15$

$= 1775 \text{ KJ/Kg}$

$Q_{add} = (h_1 - h_6) + (h_3 - h_2)$

$= (3455 - 206.82) + (3585 - 2785)$

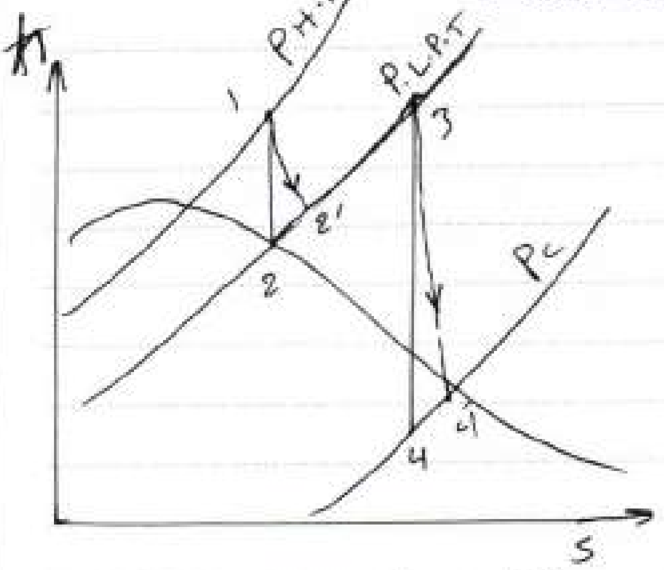
$= 4048.18 \text{ KJ/Kg}$

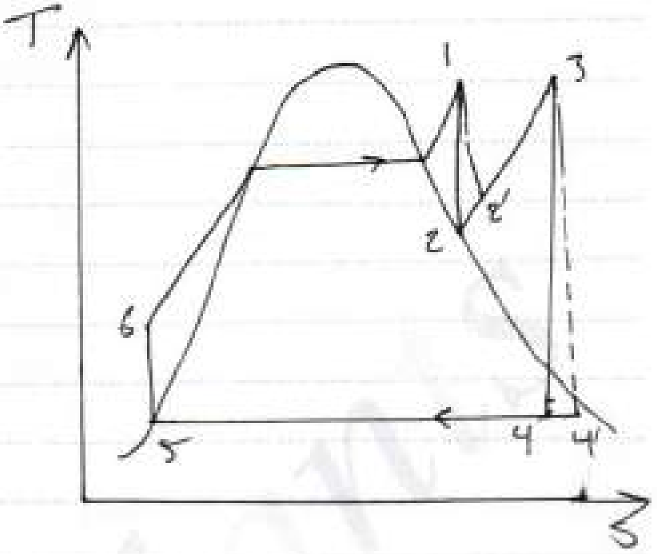
$$\eta_{ideal} = \frac{W_{done}}{Q_{add}} = \frac{1775}{4048.18} = 43.84\ \%$$

If the isentropic efficiency of the two turbines are 85 %, Find the new efficiency?

$P_{output} = \dot{m}_s . WD$

$= 50 * 1775 = 88.75 \text{ MW}$





$$\eta_{ise.\,H.P.T} = \frac{h_1 - h_2'}{h_1 - h_2}$$

$$0.85 = \frac{3455 - h_2'}{3455 - 2785} \Rightarrow h_2' = 2885.5 \text{ KJ/Kg}$$

$$\eta_{ise,\,L.P.T} = \frac{h_3 - h_4'}{h_3 - h_4}$$

$$0.85 = \frac{3585 - h_4'}{3585 - 2465} \Rightarrow h_4' = 2633 \text{ KJ/Kg}$$

$$W_T = (h_1 - h_2') + (h_3 - h_4') - (h_6 - h_5)$$

$$= (3455 - 2885.5) + (3585 - 2633) - 15$$

$$= 1506.5 \text{ KJ/Kg}$$

$$Q_{add} = (h_1 - h_6) + (h_3 - h_2')$$

$$= (3455 - 266.82) + (3585 - 2885.5)$$

$$= 3947.7 \text{ KJ/kg}$$

$$\eta_{actual} = \frac{W_{done}}{Q_{add}} = \frac{1506.5}{3947.7} = 38.16\%$$

$$\Delta\eta = \eta_{ideal} - \eta_{actual}$$

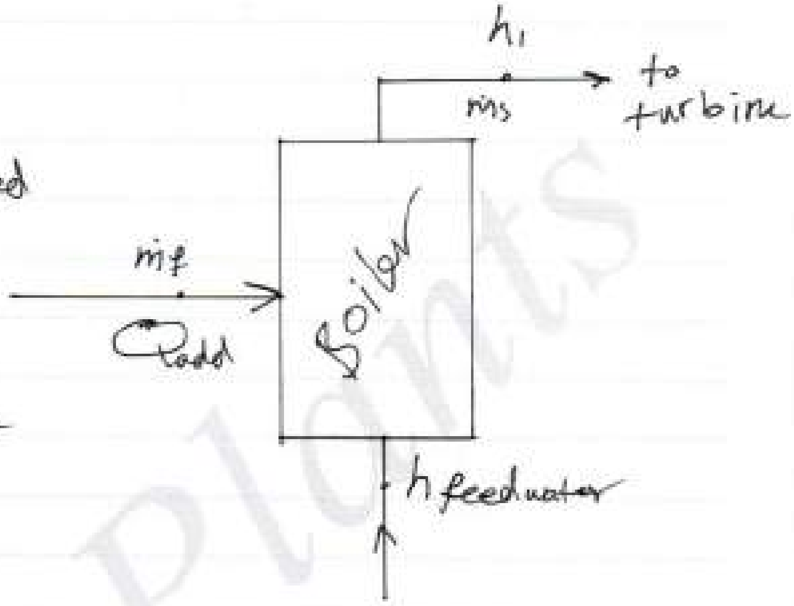
$$= 43.84\% - 38.16\%$$

$$= 5.68\%$$

You can notice that the cycle efficiency has decreased 5.68% when taking into account isentropic efficiency of the H.PT and L.P.T.

* Efficiencies in Steam Power plant

- Boiler efficiency



It is the heat supplied to the steam in the boiler expressed as a percentage of the chemical energy of the fuel which is available in the combustion.

ie,

$$\boxed{\eta_b = \frac{(h_1 - h_{F.W})\,\dot{m}_s}{\dot{m}_f \times C.V}}$$

where, h_1 = Enthalpy of steam entering the turbine (KJ/Kg)

$h_{F.W}$ = Enthalpy of feed water (KJ/Kg)

$\dot{m}_s$ = steam mass flow rate (Kg/s)

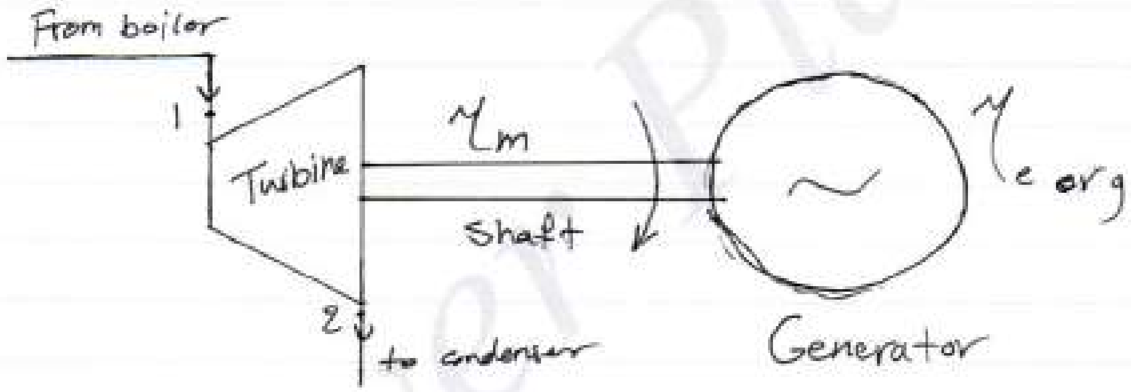
$\dot{m}_f$ = fuel mass flow rate (Kg/s)

$C.V$ = Calorific value of the fuel (KJ/Kg)

- Mechanical and electrical (generator) efficiency

In steam plants, the work done by turbine is transferred to the generator to generate electricity by a shaft.

Due to friction losses in the shaft and iron losses in the generator, there are going to be two efficiencies mechanical efficiency and electrical efficiency



So, the output Power of Plant in this case

called "terminal Power"

$$\text{Terminal Power} = \dot{m}_s * W.D * \eta_g * \eta_m$$

- Cycle efficiency

Cycle efficiency which is explanied well previously takes into account equipment of the clycle: turbine, condenser, pump and boiler

$$\eta_{cycle} = \frac{W.D}{Q_{add}}$$

- Specific steam consumption

When taking into account mechanical and electrical efficiency of the shaft and the generator, S.S.C will be as follows,

$$S.S.C = \frac{3600}{W.D * \eta_g * \eta_m} \quad kg_{fuel}/kw.hr$$

- Plant efficiency

Plant efficiency includes cycle equipment and both mechanical and electrical efficiencies along with boiler efficiency :-

$$\eta_{plant} = \eta_{cycle} * \eta_{boiler} * \eta_m * \eta_g$$

Power Plants

EX/ Steam power plant has the following data:-

Steam condition at turbine 6 MPa, 450°C

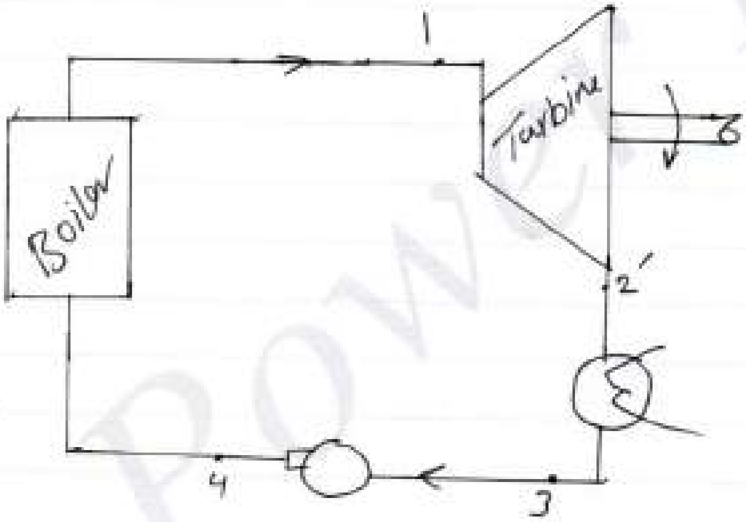
Condensor pressure 0.008 MPa

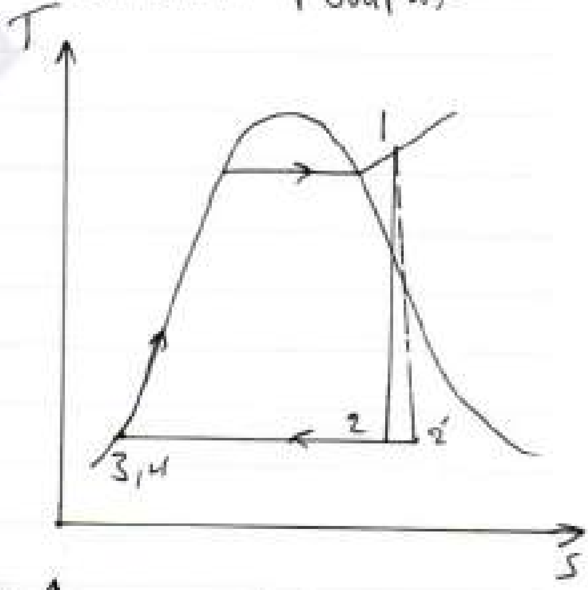
$\eta_{turbine} = 0.83$, $\eta_m = 0.92$, $\eta_g = 0.96$, $\dot{m}_s = 40$ kg/s

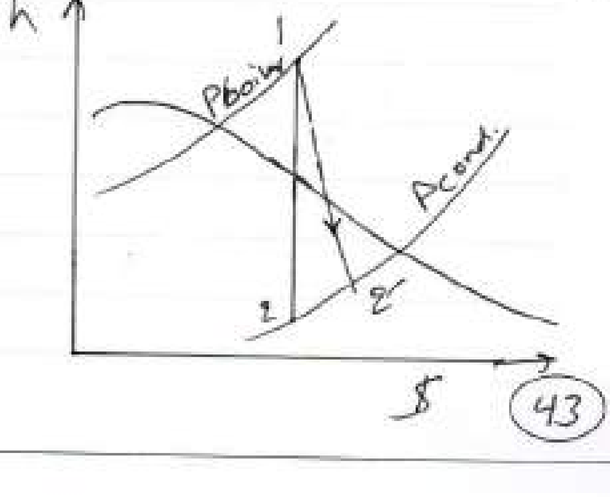
CV of fuel = 37000 KJ/kg , $\dot{m}_f = 4$ kg/s

Neglect pump work. Find

a) Work done
b) heat added
c) boiler efficiency
d) cycle efficiency
e) Plant efficiency
c) terminal Poutput







Solution :-

From Mollier charts

$h_1 = 3305$ KJ/Kg
$h_2 = 2105$ KJ/Kg

$$\eta_t = \frac{h_1 - h_2'}{h_1 - h_2} \Longrightarrow 0.83 = \frac{3305 - h_2'}{3305 - 2105}$$

$h_2' = 2881.7$ KJ/Kg

From Steam table

at 0.008 MPa , $h_3 = h_4 = h_f = 173.7$ KJ/Kg

$$W.D = h_1 - h_2'$$
$$= 3305 - 2105$$
$$= 1200 \text{ KJ/Kg}$$

$$Q_{add} = h_1 - h_4$$
$$= 3305 - 173.7$$
$$= 3131.3 \text{ KJ/Kg}$$

$$\eta_b = \frac{(h_1 - h_4)\, m_s}{m_f \; CV}$$

$$= \frac{(3305 - 173.7) * 40}{4 * 37000} = 84.6\ \%$$

$$\text{Cycle efficiency} = \frac{W.D}{Q_{add}}$$

$$= \frac{1200}{3131.3} = 38.3\%$$

$$\eta_{Plant} = \eta_{cycle} \times \eta_b \times \eta_m \times \eta_e$$

$$= 0.383 \times 0.846 \times 0.92 \times 0.96$$

$$= 28.6\ \%$$

You can notice that $\eta_{Plant} < \eta_{cycle}$

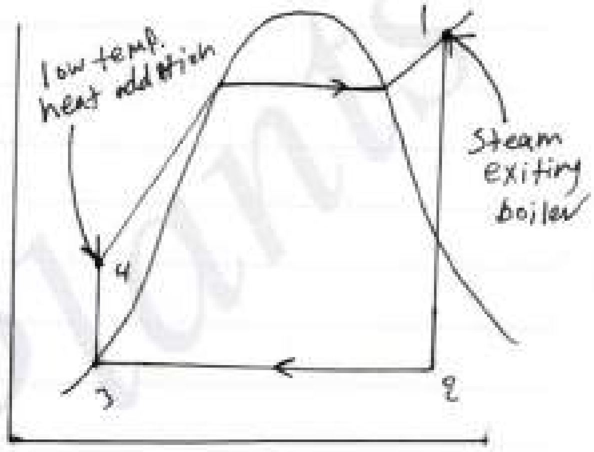
$$\text{Terminal } P_{output} = \dot{m}_s \times W.D \times \eta_g \times \eta_m$$

$$= 40 \times 1200 \times 0.92 \times 0.96$$

$$= 42.4\ MW$$

* Regenerative Cycle

In simple Rankine cycle, the heat is added to the working fluid at relatively low temprature. at which heat is added and thus lowers the cycle efficiency as seen in the figure.



The Rankine efficiency can be improved by bleeding off some of the steam at an intermediat pressure during the expansion, and mixing this steam with feed water which has been pumped to the same pressure.

The purpose of this is to increase the mean temprature at which heat is added, so heat added in the boiler is reduced thus efficiency iss increased.

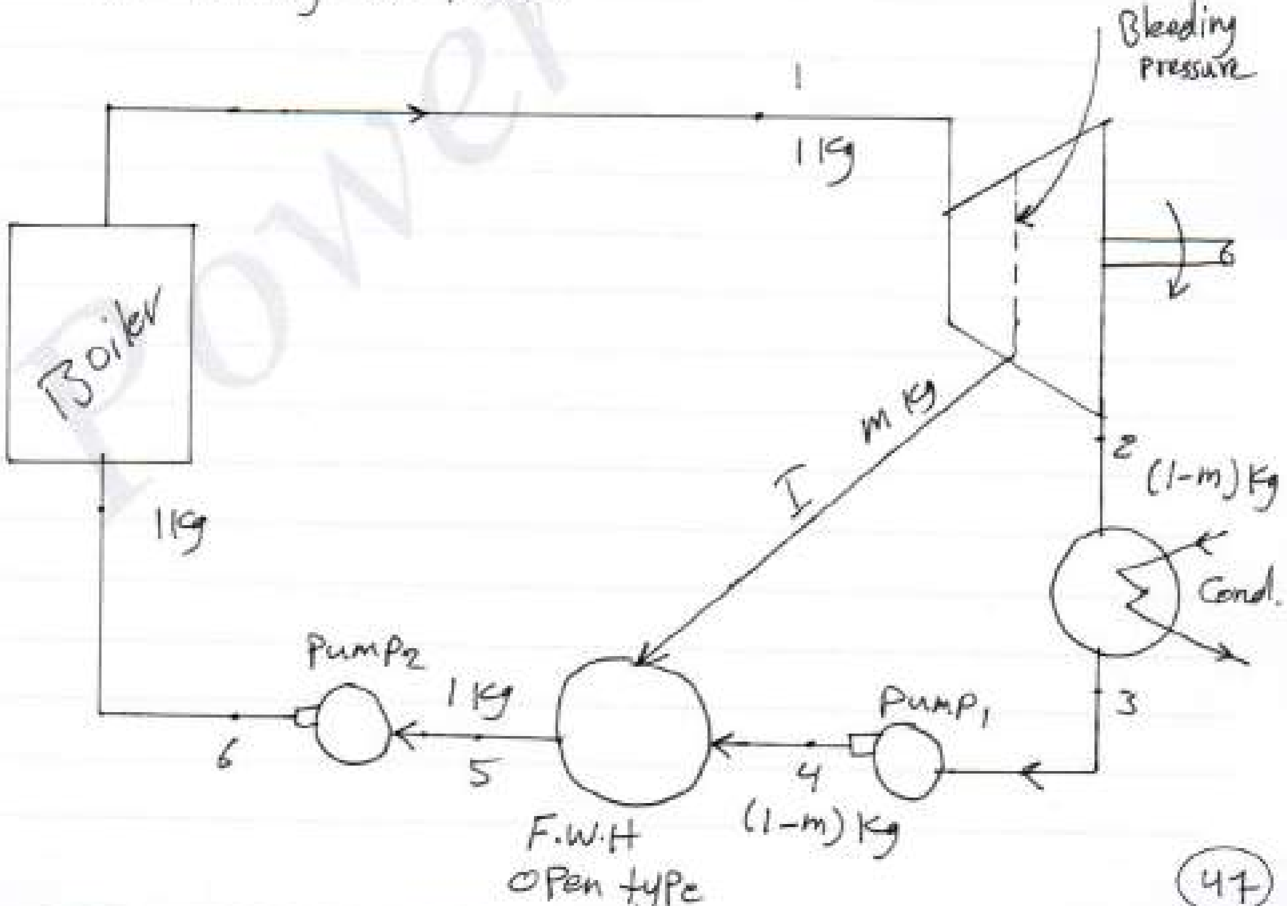
Mohanad Alheety

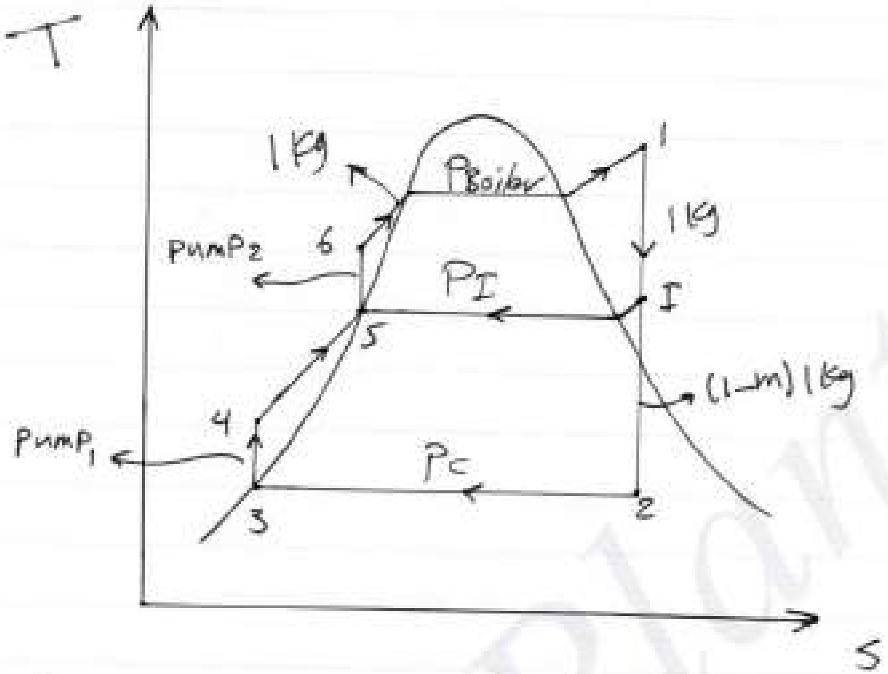
* Feed water heaters

A feed water heater is basically a heat exchanger where heat is transfferd from the steam to the feed water either by the two fluid stream (open feed water heater) or without mixing (closed feed water heater).

- Open feed-water heater

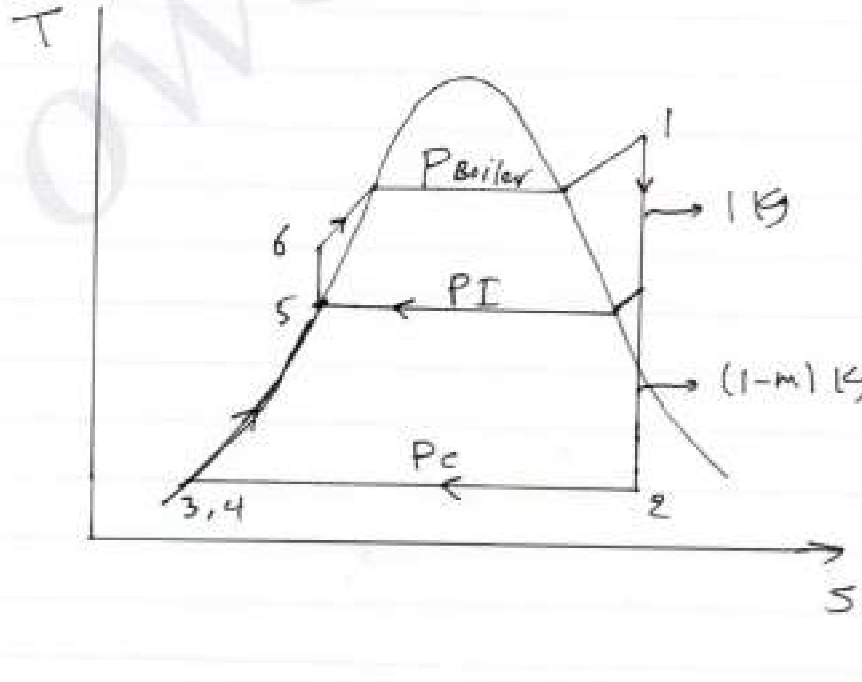
An open feed water heater is basically a mixing chamber where steam extracted from the turbine and mixed with the feed water exiting the pump.

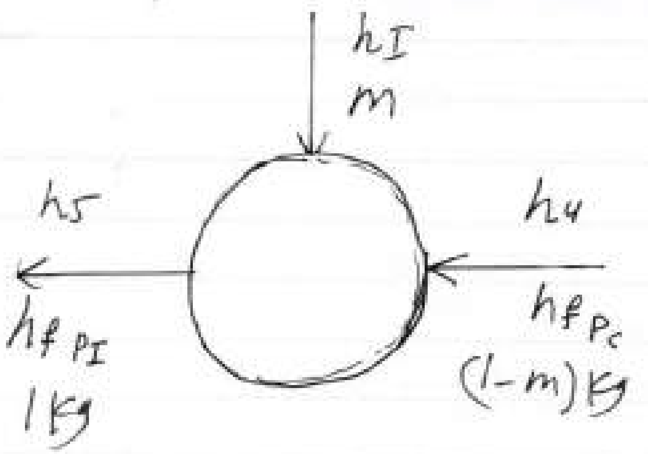


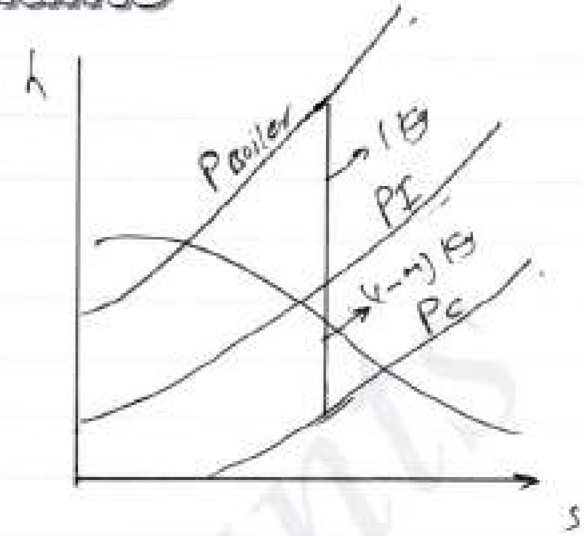


In order to make the calculations eaiser, we neglect work of pump1, so that:

$h_4 = h_3$







Heat balance of the heater

Energy input = Energy output

$$m\,h_I + (1-m)\,h_4 = 1kg \times h_5$$
$$m\,h_I + (1-m)\,h_{f_{P_c}} = 1kg \times h_{f_{P_I}}$$

$$\therefore\; m = \frac{h_{f_{P_I}} - (1-m)\,h_{f_{P_c}}}{h_I}$$

$$W.D = W_{1-I} + W_{I-2} - W_{6-5}$$

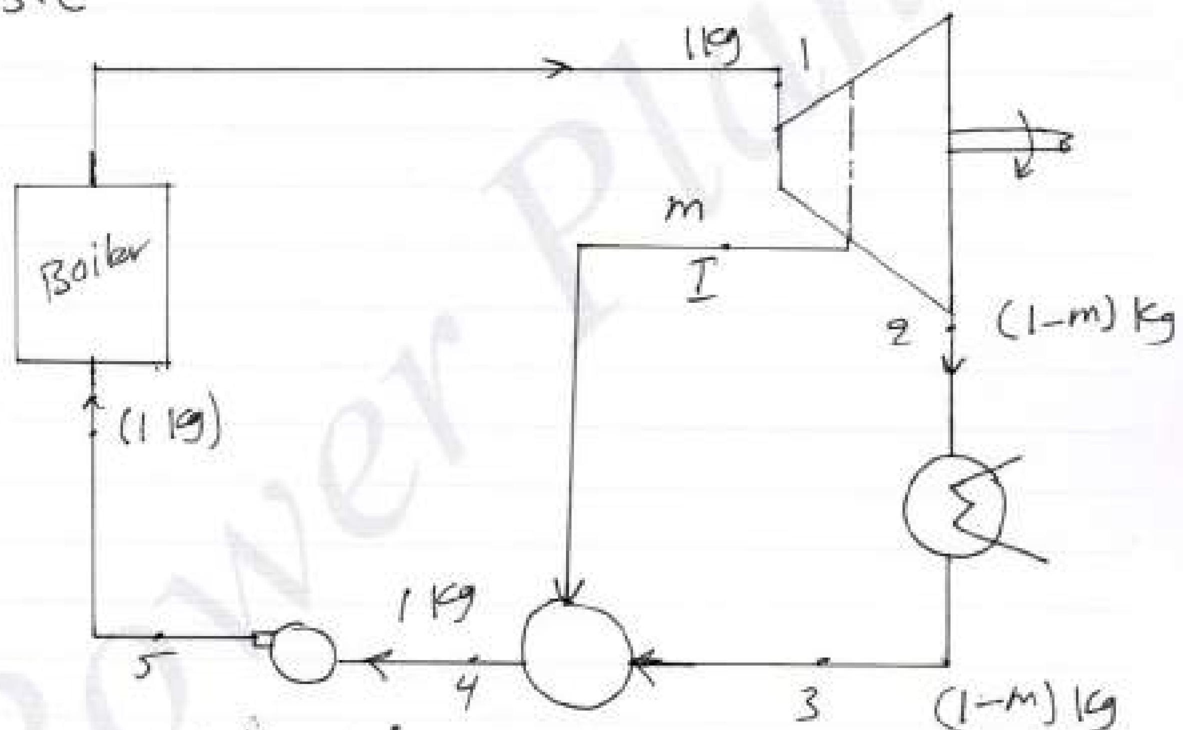
$$= 1kg\,(h_1 - h_I) + (1-m)(h_I - h_2) - (h_6 - h_5)$$

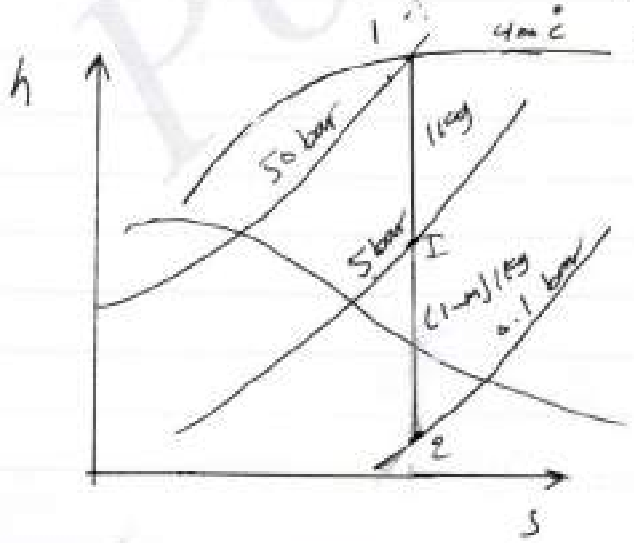
$$Q_{add} = (h_1 - h_6) \times 1\,kg$$

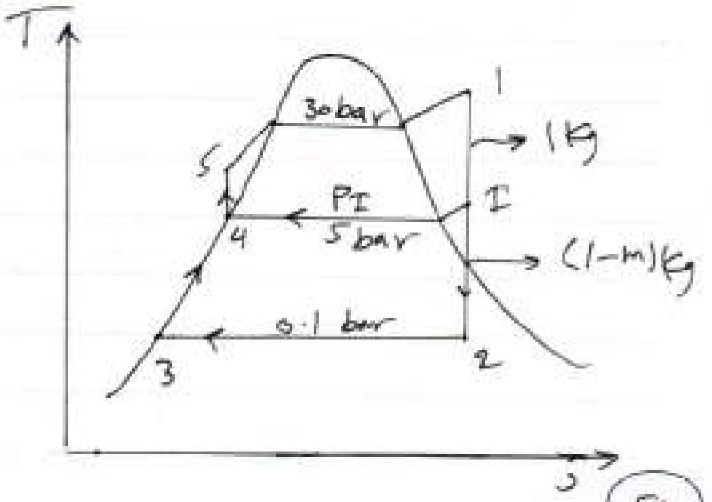
$$\eta_{cycle} = \frac{W.D}{Q_{add}}$$

EX/ In a single-heater regenerative cycle, the steam enters the turbine at 30 bar, 400°C and the exhust pressure is 0.1 bar. The feed water heater is a direct contact type which operate at 5 bar. Find

a) Amount of mass used for bleeding
b) Efficiency of the cycle
c) S.S.C







Solutions:-

From Mollier chart

At 30 bar, 400°C $h_1 = 3230$ KJ/Kg

$h_I = 2790$ KJ/Kg

$h_2 = 2190$ KJ/Kg

From steam table

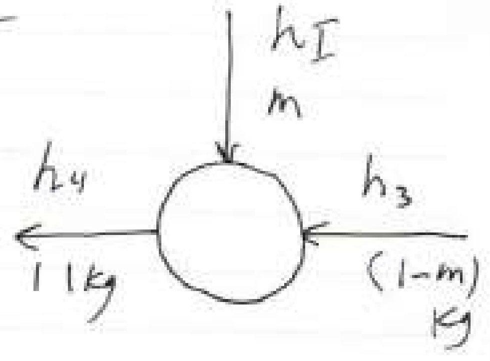
At 0.1 bar $h_3 = h_{f_{P_c}} = 191.8$ KJ/Kg
At 5 bar $h_4 = h_{f_{P_I}} = 640.1$ KJ/Kg

$v(P_{Boiler} - P_I) = h_5 - h_4$
$0.001(3000 - 500) = h_5 - 640.1$
$h_5 = 642.6$ KJ/Kg

Heat balance for the heater

$m h_I + (1-m) h_3 = h_4 * 1$
$m * 2790 + (1-m) * 191.8 = 640.1$
$2598.2\, m = 448.3$

$m = 0.1725$ Kg



$$W.D = 1kg(h_1 - h_I) + (1-m)(h_I - h_2) - (h_5 - h_4)$$

$$= (3230 - 2790) + (1 - 0.1725)(2790 - 2190) - (640.6 - 640.1)$$

$$= 936 \ KJ/kg$$

$$Q_{add} = (h_1 - h_5) * 1kg$$

$$= (3230 - 642.6) * 1$$

$$= 2587.4 \ KJ/kg$$

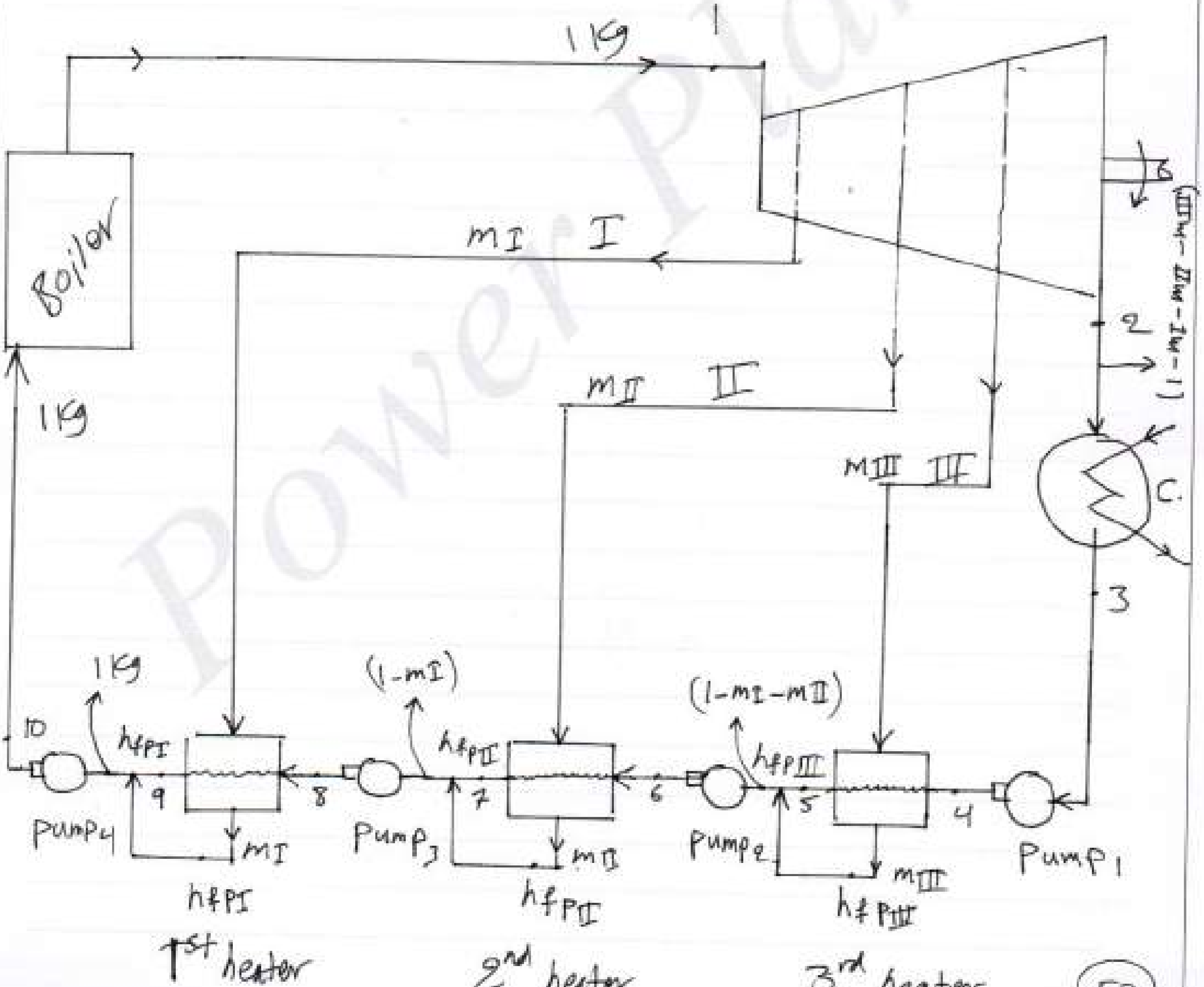
$$\eta_{cycle} = \frac{W.D}{Q_{add}} = \frac{936}{2587.4} = 36.18 \%$$

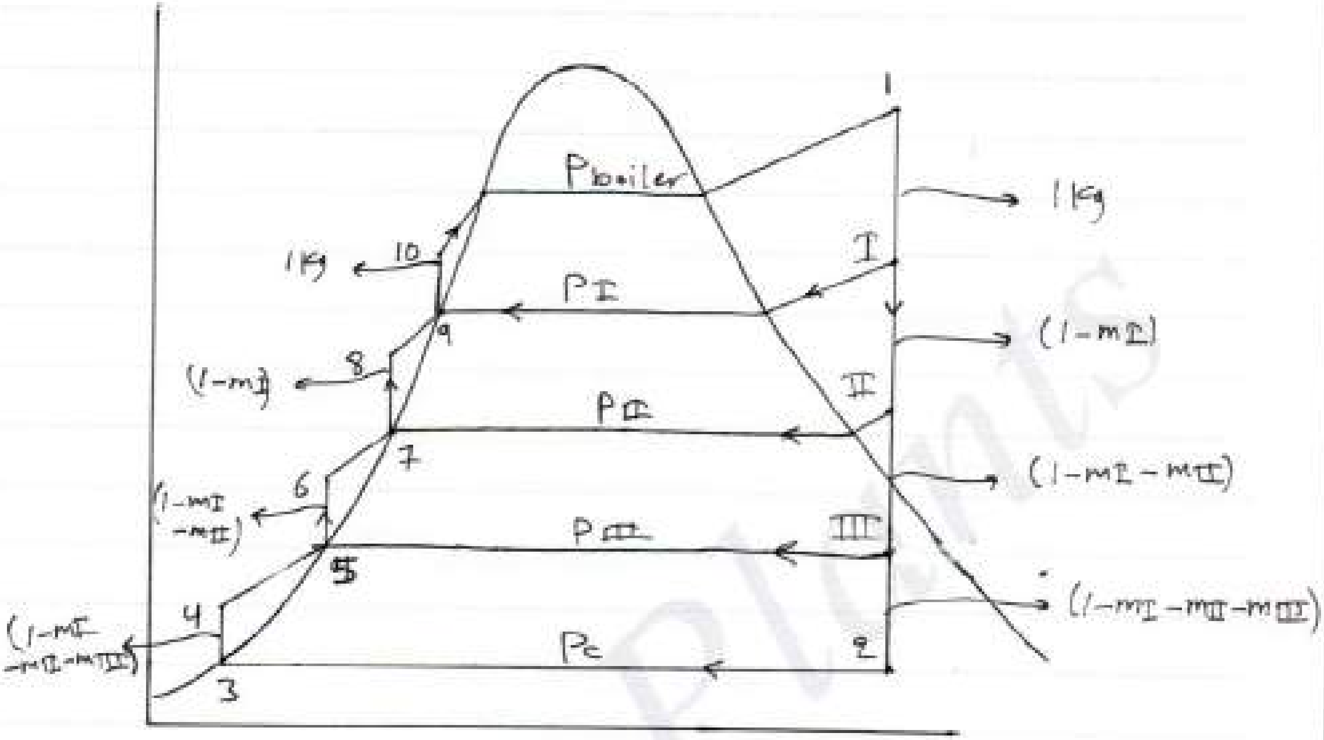
$$S.S.C = \frac{3600}{W.D} = \frac{3600}{936} = 3.846 \ kg_{fuel}/kW.h$$

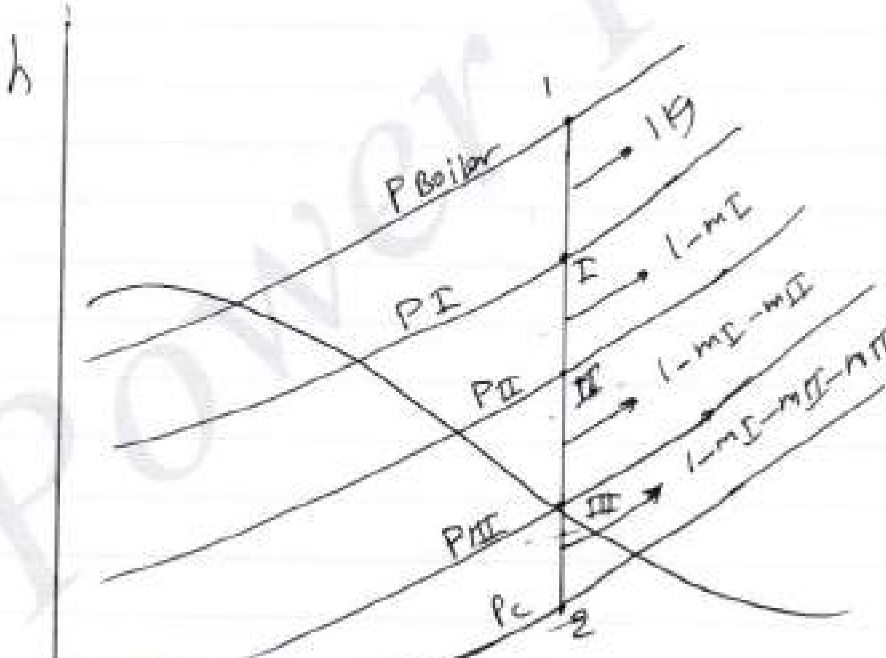
- Closed feed water heaters

In this type, the heat is transferred from the extracted (bleeded) steam to the feed water without mixing taking place. The two streams can be at different pressure since they donot mix

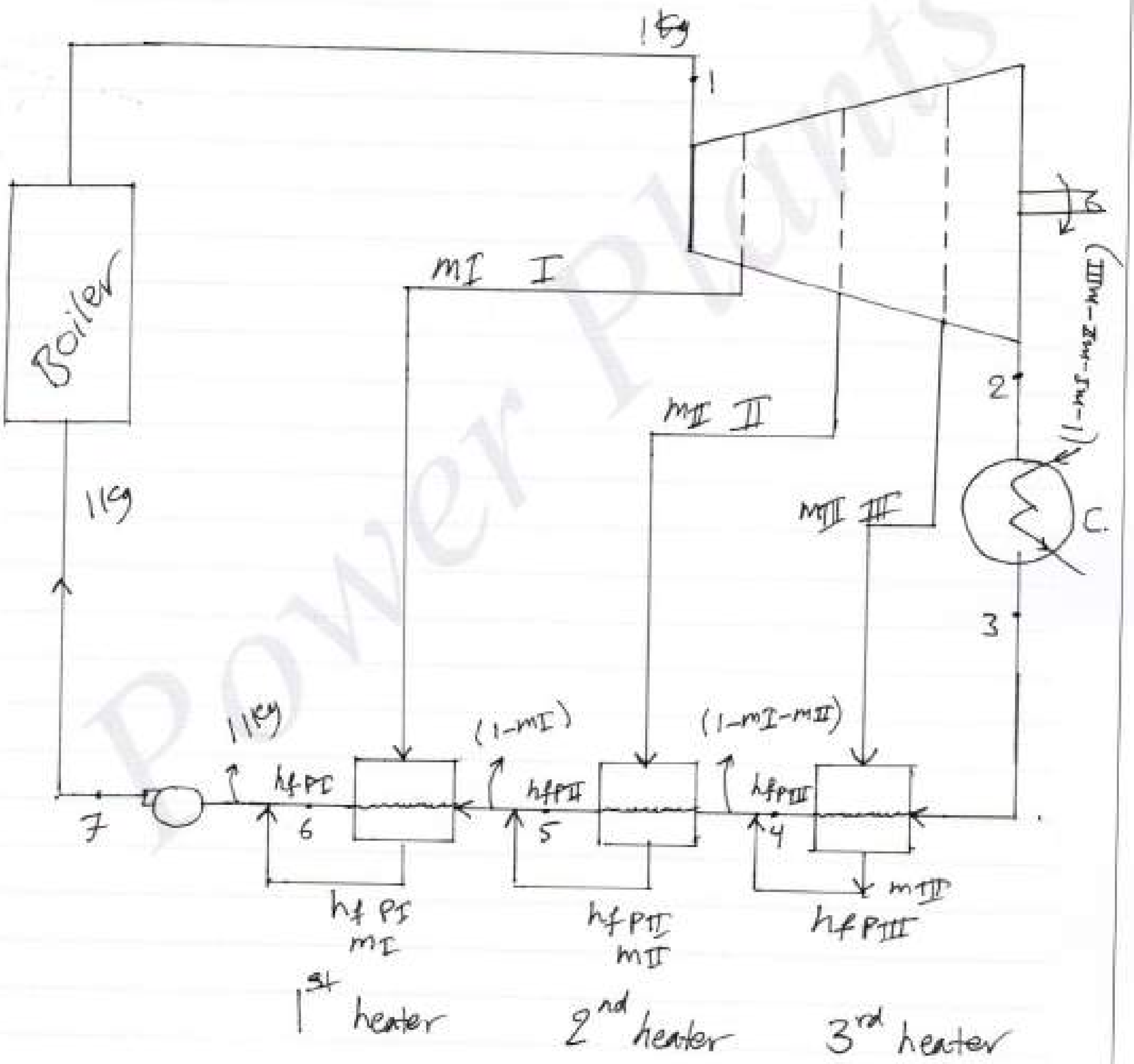
- Forward flow heaters :-

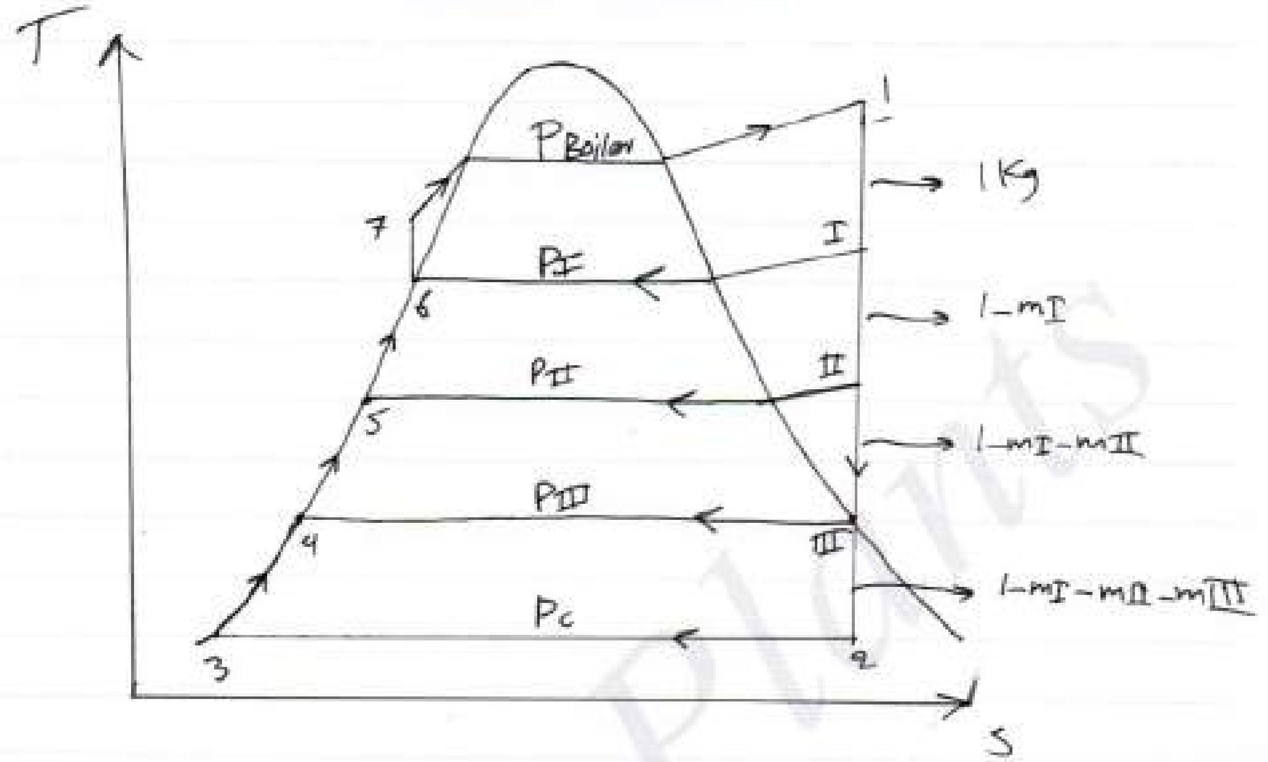



1
Pboiler
1kg
10
1kg
I
PI
(1-mI)
9
8
(1-mI)
II
PII
7
(1-mI-mII)
6
(1-mI -mII)
PIII
III
5
(1-mI-mII-mIII)
4
(1-mI -mII-mIII)
Pc
2
3


h
1
PBoiler
1kg
I
1-mI
PI
PII
II
1-mI-mII
1-mI-mII-mIII
PIII
III
Pc
2
s

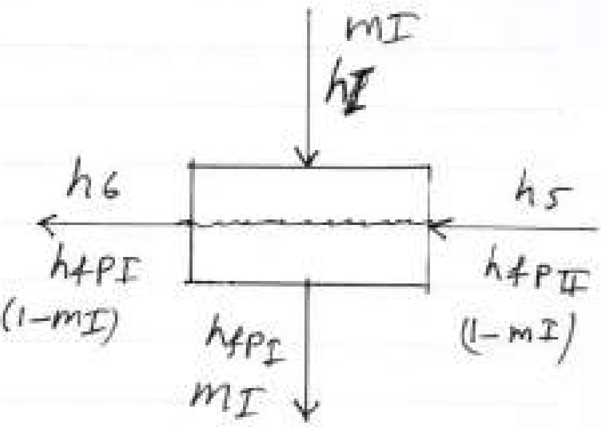
In order to make the calculations eaiser, we can neglect work of $pump_1$, $pump_2$, $pump_3$, so the new cycle is as follows:-





* Calculations :-

- 1st heater



Heat rejected = heat gained

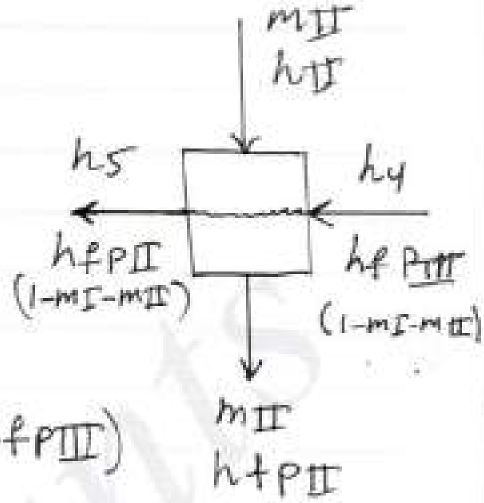
$$m_I (h_I - h_{fP_I}) = (1 - m_I) * (h_6 - h_5)$$

$$m_I (h_I - h_{fP_I}) = (1 - m_I)(h_{fP_I} - h_{fP_{II}})$$

Then, m_I is calculated from the equation above

- 2nd heater

Heat rejected = Heat gained



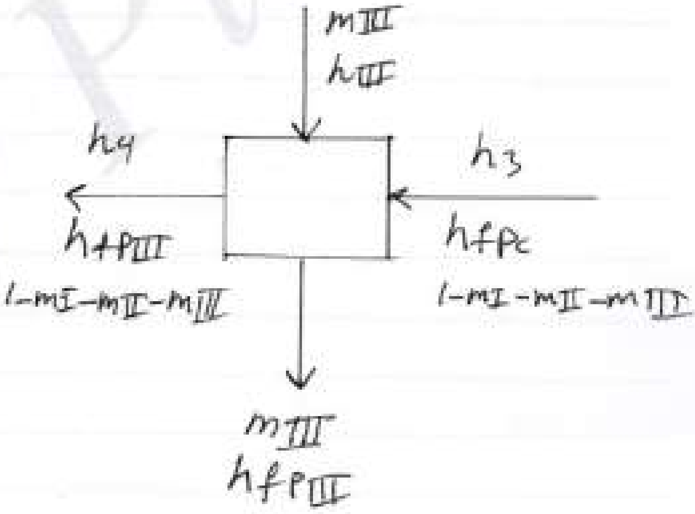
$$m_{II}(h_{II} - h_{fPII}) = (1 - m_I - m_{II})(h_5 - h_4)$$

$$m_{II}(h_{II} - h_{fPII}) = (1 - m_I - m_{II})(h_{fPII} - h_{fPIII})$$

Then, m_{II} is calculated from equation above.

- 3rd heater

Heat rejected = heat gained



$$m_{III}(h_{III} - h_{fPIII}) = (1 - m_I - m_{II} - m_{III})(h_4 - h_3)$$

$$m_{III}(h_{III} - h_{fPIII}) = (1 - m_I - m_{II} - m_{III})(h_{fPIII} - h_{fPc})$$

m_{III} is also calculated from equation above.

Now, after all extracted masses are evaluated, we can calculate the cycle efficiency :-

$$W.D = W_{1-I} + W_{I-II} + W_{II-III} + W_{III-2} - W_{7-6}$$

$$= (h_1 - h_I) * 1\,kg + (h_I - h_{II}) * (1 - m_I) + (h_{II} - h_{III}) * (1 - m_I - m_{II}) + (h_{III} - h_2) * (1 - m_I - m_{II} - m_{III})$$

$$Q_{add} = (h_1 - h_7) * 1\,kg$$

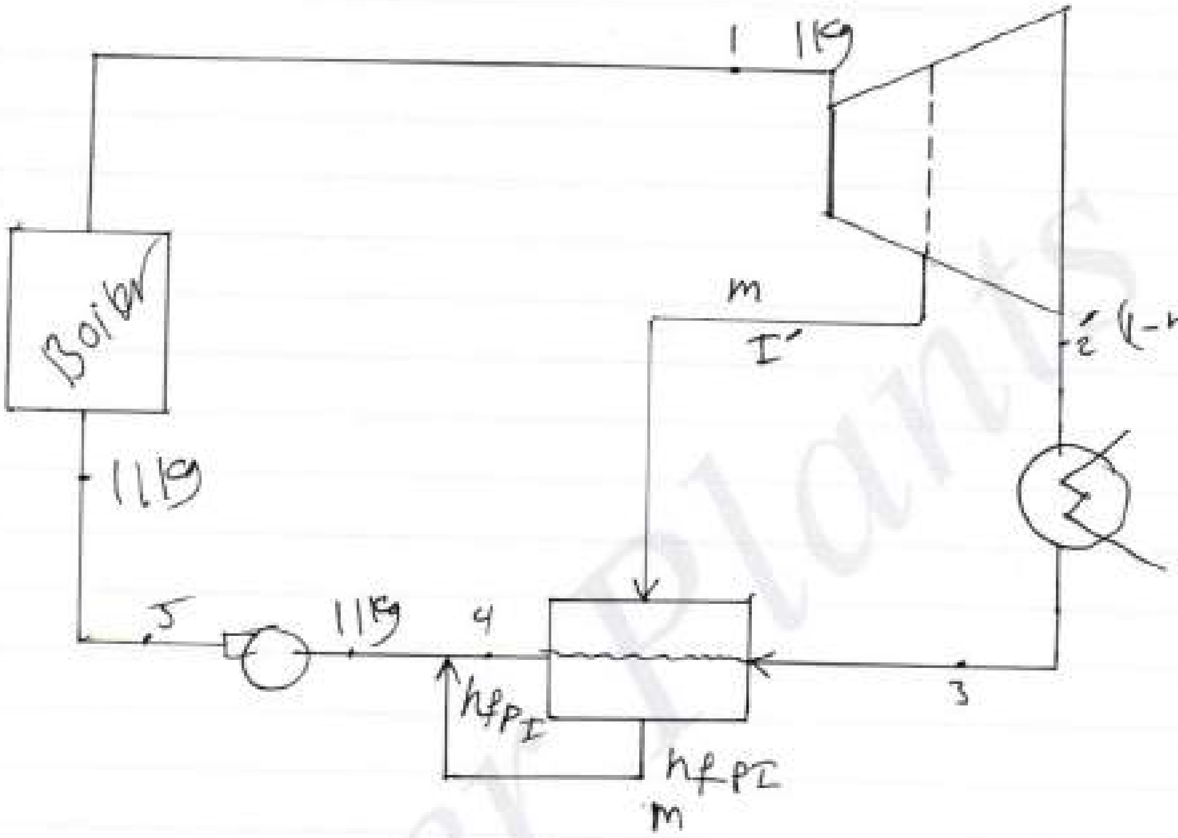
Now, $\eta_{cycle} = \dfrac{W.D}{Q_{add}}$

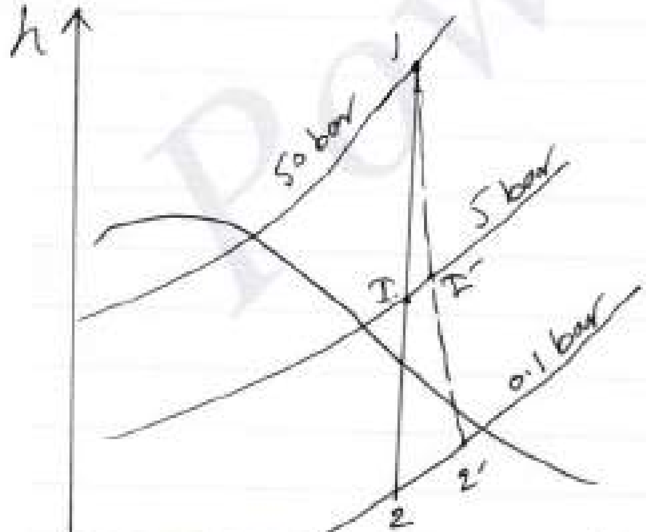
Ex/ For example on page 50, use the same conditions. The feed water heater is closed type (forward). Calculate

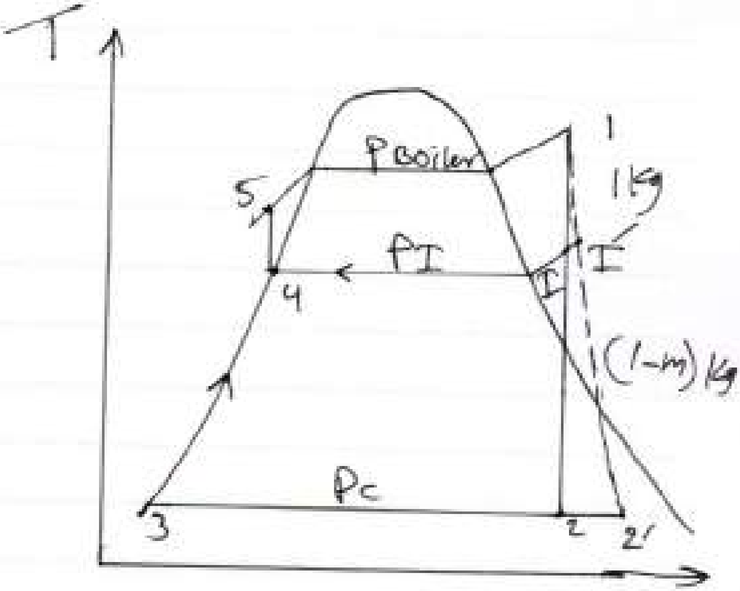
a) amount of mass used for bleeding
b) η_c

turbine efficiency is 85 %

Solutions_


1
1 kg
Boiler
m
I'
2' (1-m)
1 kg
5
1 kg
4
3
h_{fPI}
h_{fPI}
m


h
1
50 bar
5 bar
I
I'
0.1 bar
2
2'
s


T
1
P Boiler
1 kg
5
PI
I
I
4
(1-m) kg
Pc
3
2
2'
s

From chart: $h_1 = 3230$ KJ/Kg

$h_I = 2790$ KJ/Kg

$h_2 = 2190$ KJ/Kg

From steam table: $h_3 = h_f = 191.8$ KJ/Kg

$h_4 = h_{fP_I} = 640.1$ KJ/Kg

$$\eta_t = \frac{h_1 - h_2'}{h_1 - h_2} \Rightarrow 0.85 = \frac{3230 - h_2'}{3230 - 2190}$$

$h_2' = 2346$ KJ/Kg

$$0.85 = \frac{h_1 - h_I'}{h_1 - h_I} \Rightarrow 0.85 = \frac{3230 - h_I'}{3230 - 2790}$$

$h_I' = 2856$ KJ/Kg

$h_5 = 642.6$ KJ/Kg , $v(P_b - P_I) = h_5 - h_4$

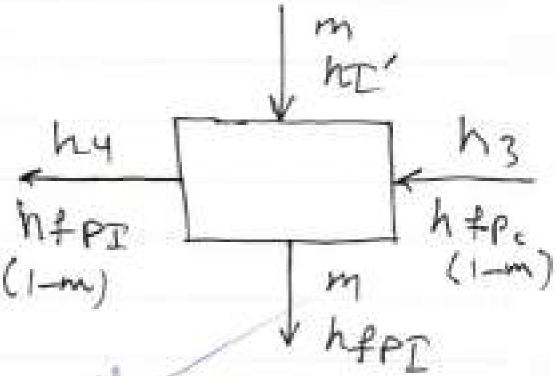
Heat balance for heater

$m(h_I' - h_{fP_I}) = (1-m)(h_4 - h_3)$

$m(2856 - 640.1) = (1-m)(640.1 - 191.8)$

$\therefore 2664.2\, m = 448.3$

$m = 0.1682$ Kg



Mohaned Alheety

$W.D = 1Kg\,(h_1 - h_I') + (h_I' - h_e')(1-m) - (h_5 - h_4)$

$= 1*(3230 - 2856) + (2856 - 2346)(1 - 0.1682) - (642.6 - 640.1)$

$= 795.7 \text{ KJ/Kg}$

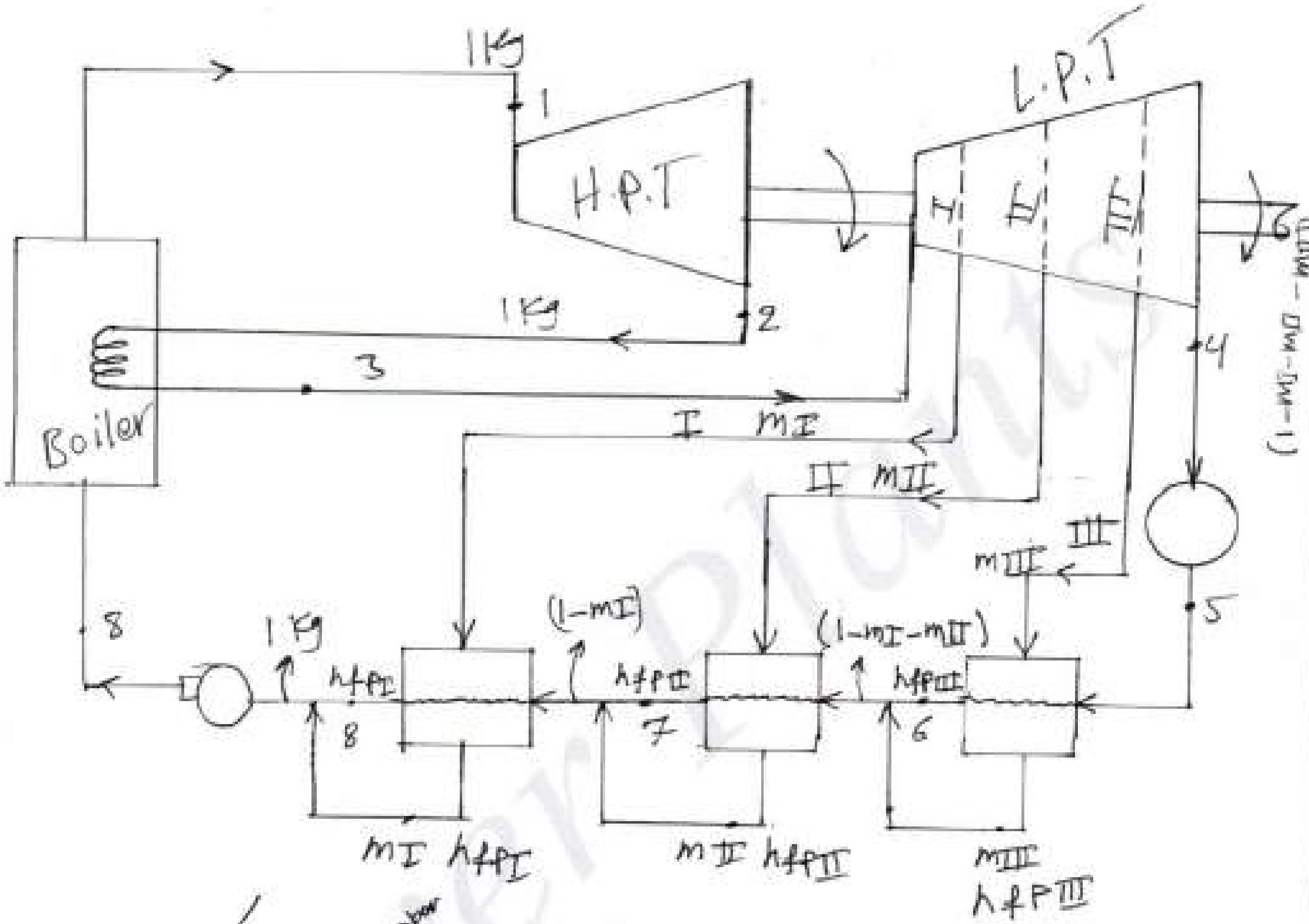
$Q_{add} = (h_1 - h_5) 1Kg$
$= (3230 - 642.6) * 1$
$= 2587.4 \text{ KJ/Kg}$

$\eta_{cycle} = \frac{795.7}{2587.4} = 30.75\%$

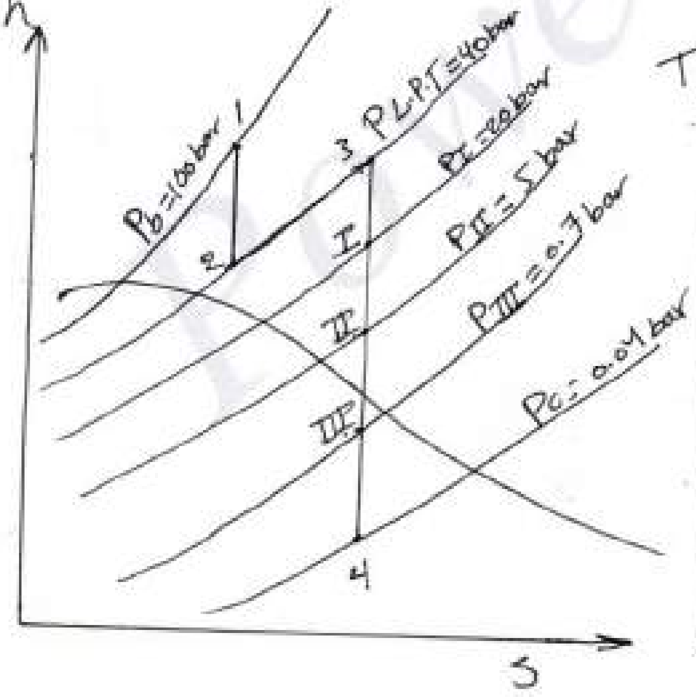
EX/ The output power of a steam plant is 100 MW. The turbine consists of high and low pressure cylinders with a reheating between them using boiler fuel gases. There are three forward flow feed water heaters (closed type) located in the cycle and use bleed steam from the low pressure turbine.
Bleedings occure at 20 bar, 5 bar, 0.7 bar
Boiler steam pressure and tempreture are 100 bar, 550°C
high pressure turbine exhausts at 40 bar.
steam admitted to low pressure turbine at 40 bar, 510 C
Steam exhausts from low pressure turbine at 0.04 bar
Find: a) amounts of steam used for bleeding
b) mass flow rate of steam
c) cycle efficiency

Neglect all pumps work.

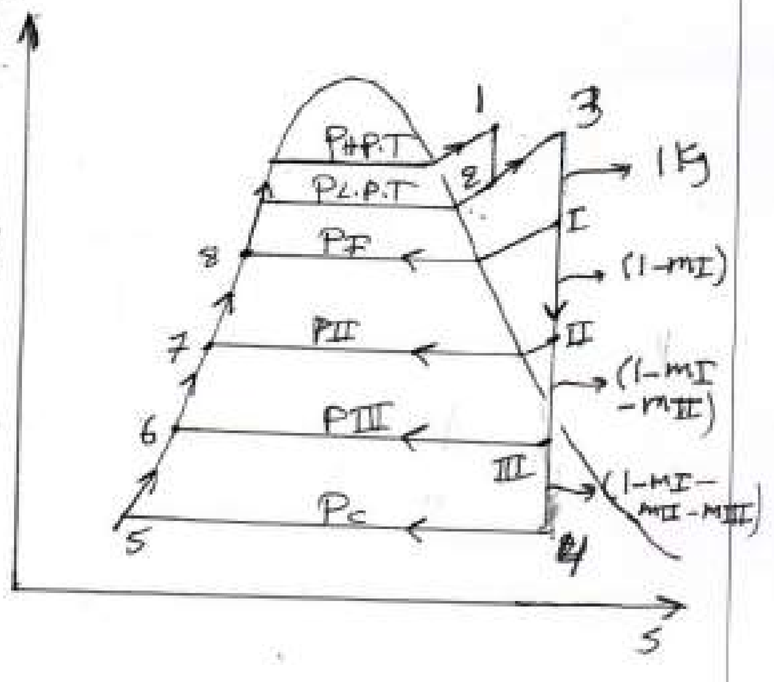
solution:-



1 kg
1
H.P.T
L.P.T
I
II
III
2
1 kg
3
4
Boiler
I mI
II mII
III
mIII
5
8
1 kg
(1-mI)
(1-mI-mII)
(1-mI-mII-mIII)
hfPI
hfPII
hfPIII
8
7
6
mI hfPI
mII hfPII
mIII hfPIII



h
Pb=100 bar
1
3
PL.P.T=40 bar
PI=20 bar
PII=5 bar
PIII=0.7 bar
Pc=0.04 bar
2
I
II
III
4
s



T
1
3
1 kg
PH.P.T
PL.P.T
2
PI
I
(1-mI)
8
PII
II
7
(1-mI-mII)
6
PIII
III
(1-mI-mII-mIII)
Pc
5
4
s

From h-s Chart: $h_1 = 3500$ KJ/Kg

$h_2 = 3200$ KJ/Kg

$h_3 = 3470$ KJ/Kg

$h_I = 3250$ KJ/Kg

$h_{II} = 2880$ KJ/Kg

$h_{III} = 2527$ KJ/Kg

$h_4 = 2135$ KJ/Kg

From steam table: $h_5 = h_{f P_c} = 121.4$ KJ/Kg

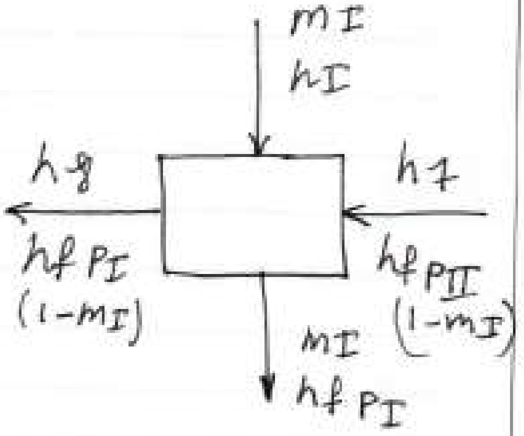
$h_6 = h_{f P_{III}} = 376.8$ KJ/Kg

$h_7 = h_{f P_{II}} = 640.1$ KJ/Kg

$h_8 = h_{f P_I} = 908.6$ KJ/Kg

Heat balance

- 1st heater



$$m_I(h_I - h_{f P_I}) = (1 - m_I)(h_8 - h_7)$$

$$m_I(3250 - 908.6) = (1 - m_I)(908.6 - 640.1)$$

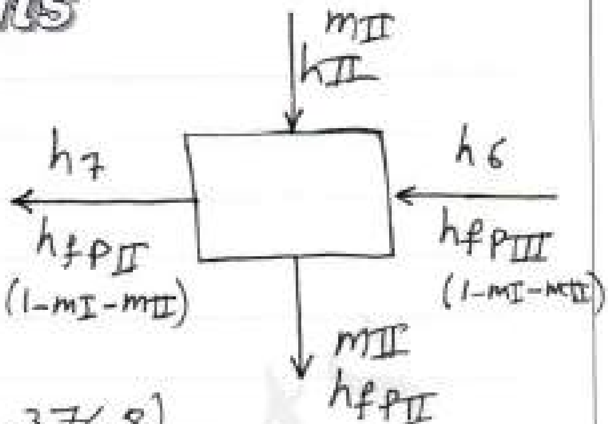
$$2341.4\, m_I = (1 - m_I) * 268.5$$

$$2609.9\, m_I = 268.5$$

$$m_I = 0.1028 \text{ Kg}$$

- 2^{nd} heater

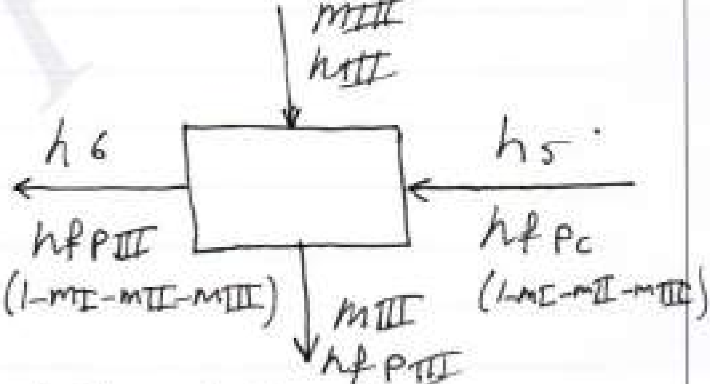
$$m_{II}(h_{II} - h_{fP_{II}}) = (1 - m_I - m_{II}) \times (h_7 - h_6)$$



$$m_{II}(2880 - 640.1) = (1 - 0.1028 - m_{II}) \times (640.1 - 376.8)$$
$$2239.9\, m_{II} = (0.8972 - m_{II}) \times 263.3$$
$$2503.2\, m_{II} = 236.237$$

$$m_{II} = 0.0943 \text{ Kg}$$

- 3^{rd} heater



$$m_{III}(h_{III} - h_{fP_{III}}) = (1 - m_I - m_{II} - m_{III})(h_6 - h_5)$$

$$m_{III}(2527 - 376.8) = (1 - 0.1028 - 0.0943 - m_{III})(376.8 - 121.4)$$
$$2150.2\, m_{III} = (0.8029 - m_{III}) \times 255.4$$
$$m_{III} \times 2405.6 = 205.06$$

$$m_{III} = 0.0852 \text{ Kg}$$

$$W.D = W_{H.P.T} + W_{L.P.T}$$
$$= W_{H.P.T} + W_{3-I} + W_{I-II} + W_{II-III} + W_{III-4}$$

$$= 1kg(h_1 - h_2) + 1kg(h_3 - h_I) + (1 - m_I)(h_I - h_{II}) + (1 - m_I - m_{II})(h_{II} - h_{III}) + (1 - m_I - m_{II} - m_{III})(h_{III} - h_4)$$

$$= (3500 - 3200) + (3470 - 3250) + (1 - 0.1028)(3250 - 2880)$$

$$+ (1 - 0.1028 - 0.0943)(2880 - 2527) + (1 - 0.1028 - 0.0943 - 0.0852) * (2527 - 2135)$$

$$= 1416.7 \ kJ/kg$$

$$Q_{add} = 1kg(h_1 - h_8) + 1kg(h_3 - h_2)$$

$$= (3500 - 908.6) + (3470 - 3200)$$

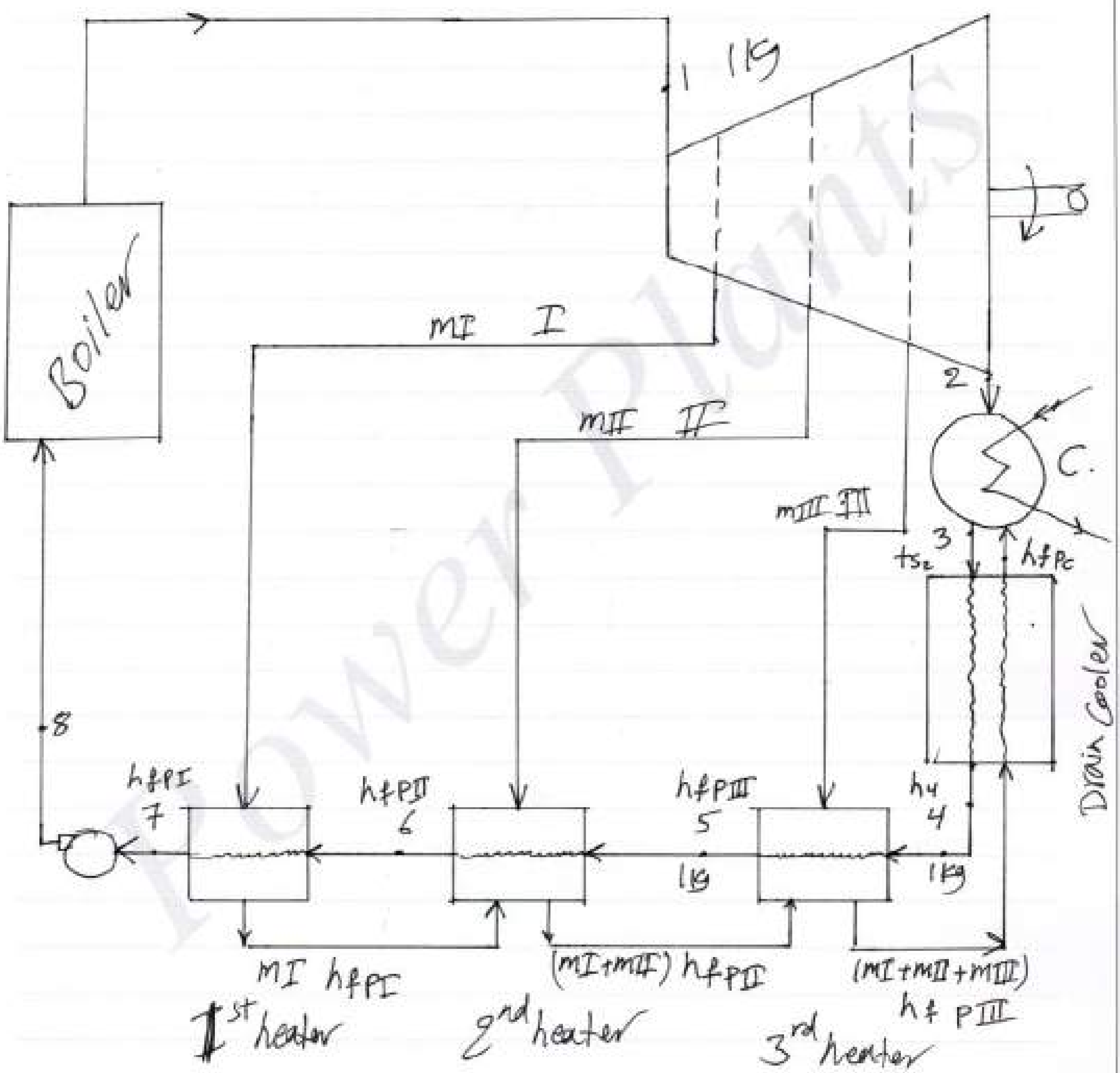
$$= 2861.4 \ kJ/kg$$

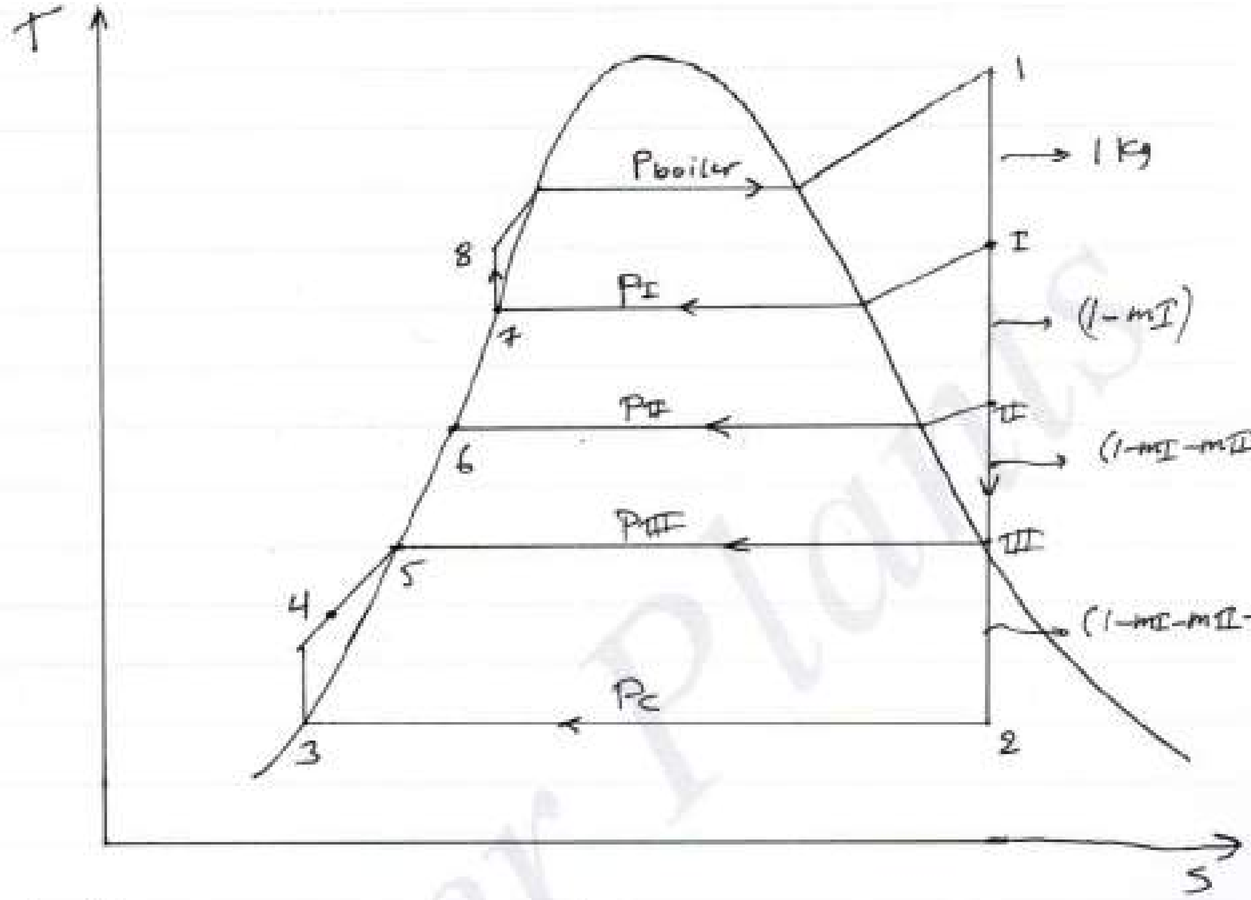
$$P_{output} = \dot{m}_s * W.D$$
$$100\,000 = \dot{m}_s * 1416.7$$

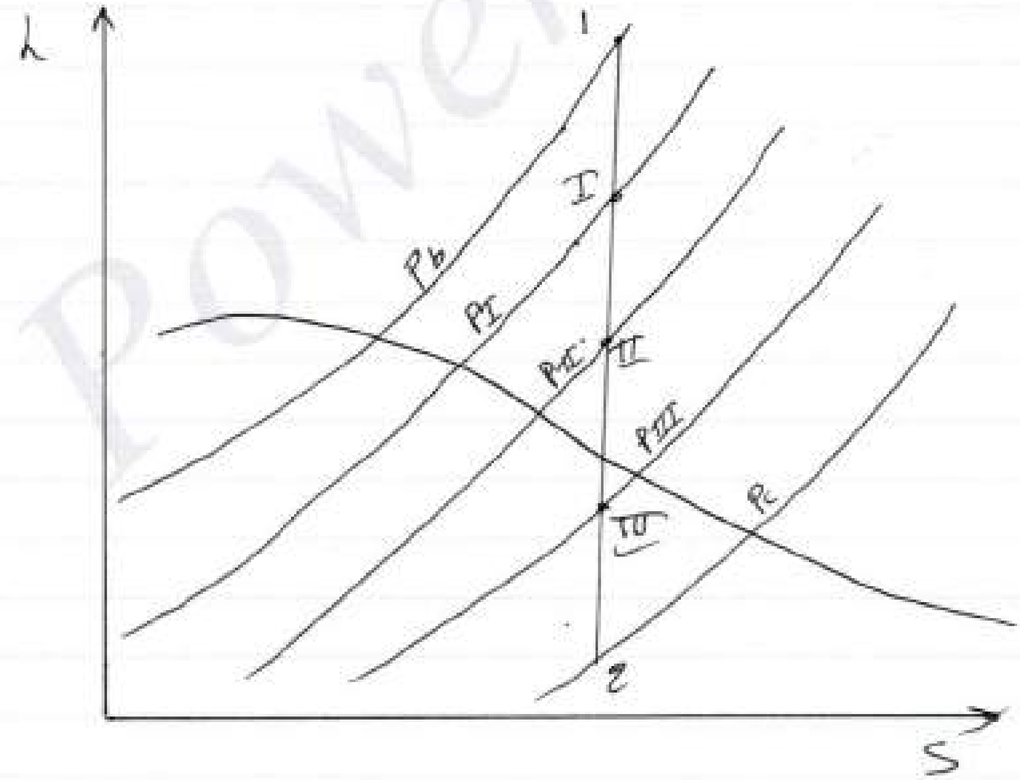
$$\dot{m}_s = 70.58 \ kg/s$$

$$\eta_{cycle} = \frac{1416.7}{2861.4} = 49.5\ \%$$

- Backward flow heaters (Cascaded)

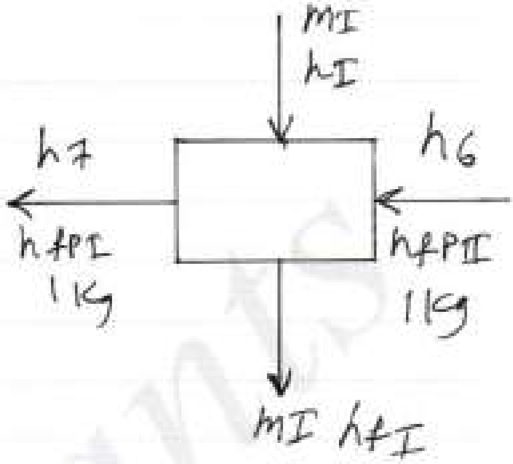



T
1
Pboiler
1 Kg
I
8
PI
(1-mI)
7
PII
II
6
(1-mI-mII)
PIII
III
5
4
(1-mI-mII-mIII)
Pc
3
2
S


h
1
I
Pb
PI
PII
II
PIII
III
Pc
2
S

Heat balance for heaters

- 1^{st} heater

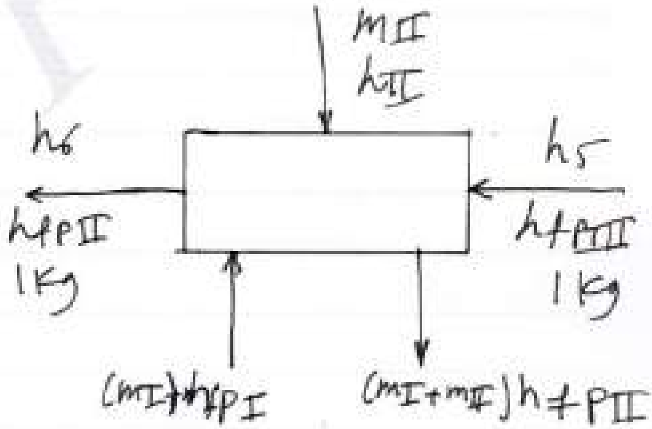


$$1kg(h_7 - h_6) = m_I(h_I - h_{fI})$$

$$1kg(h_{fpI} - h_{fpII}) = m_I(h_I - h_{fI})$$

$$m_I = \frac{h_{fpI} - h_{fpII}}{h_I - h_{fpI}}$$

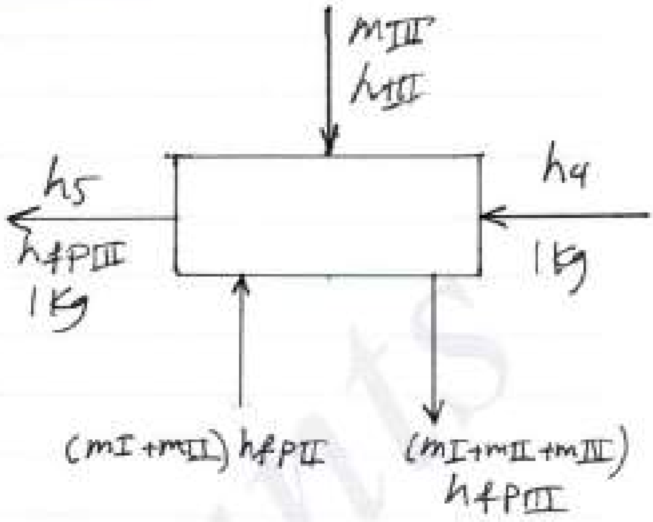
- 2^{nd} heater



$$h_5 * 1kg + m_{II} h_{II} + m_I h_{fpI} = h_6 * 1kg + (m_I + m_{II}) h_{fpII}$$

We already evaluated m_I

So, m_{II} can be calculated.



$$h_4 \times 1kg + m_{III} h_{III} + (m_I + m_{II}) h_{fpII} = h_5 \times 1kg + (m_I + m_{II} + m_{III}) h_{fpIII} \quad — ①$$

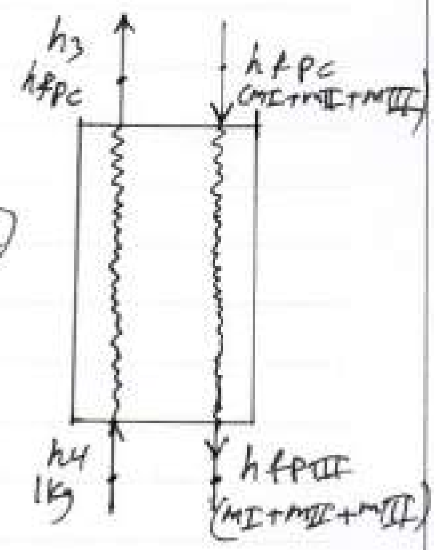
m_I, m_{II} are evaluated previously
In order to calculate m_{III}, we need to estimat (h_4), so we can make heat balance on the drain cooler

- Drain Cooler

$$1kg\,(h_4 - h_3) = (m_I + m_{II} + m_{III})(h_{fpIII} - h_{fpc})$$

$$1kg\,(h_4 - h_{fpc}) = (m_I + m_{II} + m_{III})(h_{fpIII} - h_{fpc})$$

$$h_4 = (m_I + m_{II} + m_{III})(h_{fpIII} - h_{fpc}) + h_{fpc} \quad — ②$$



by subistituting equ ② in equ ①

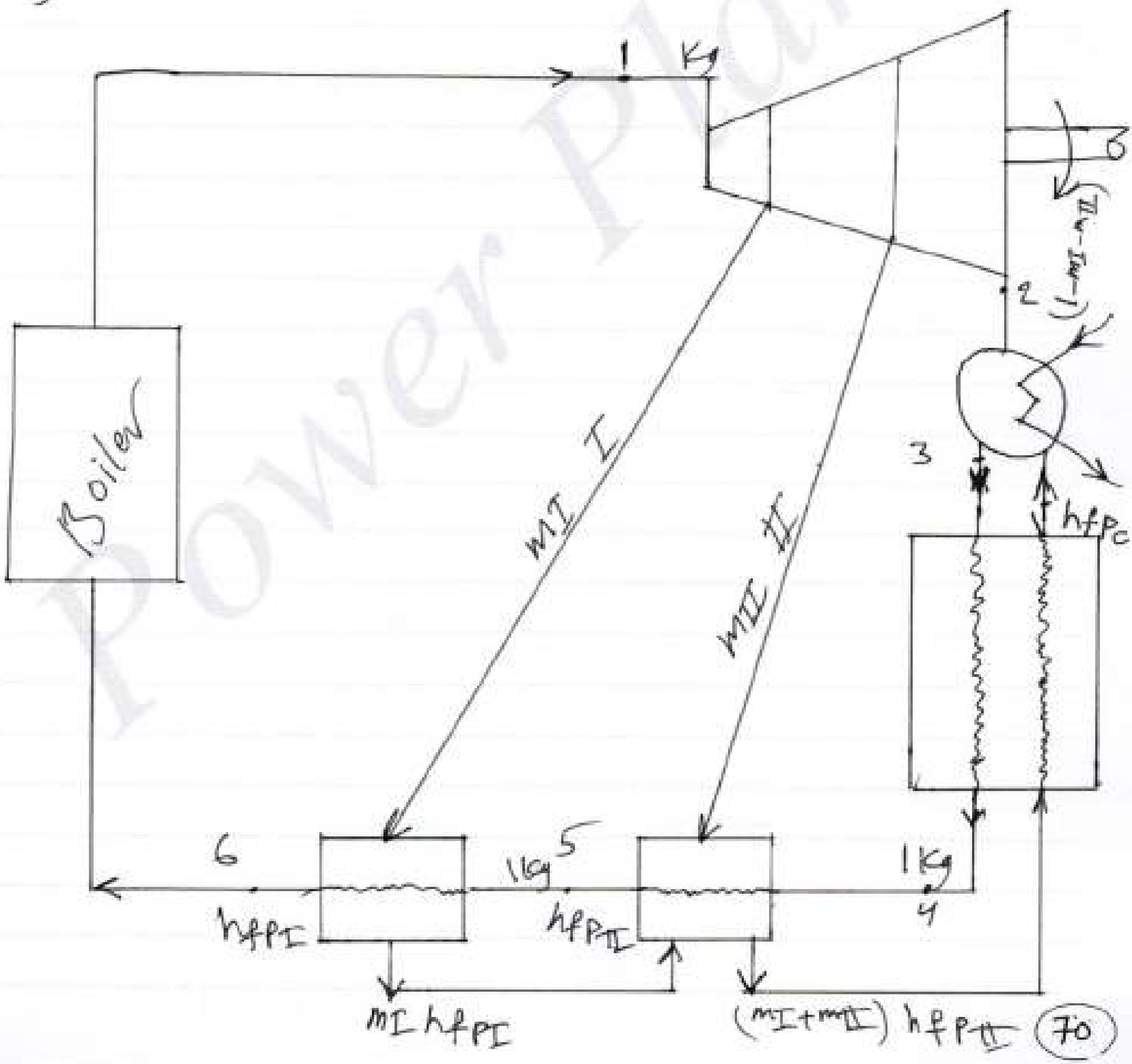
we can find m_{III}.

EX/ Steam is supplied to a turbine at 30 bar and 350°C. The turbine exhaust pressure is 0.08 bar. The main condensate is heated in two stages of backward closed feed water heaters. The steam bled from the turbine at 5 bar and 1 bar. Calculate:

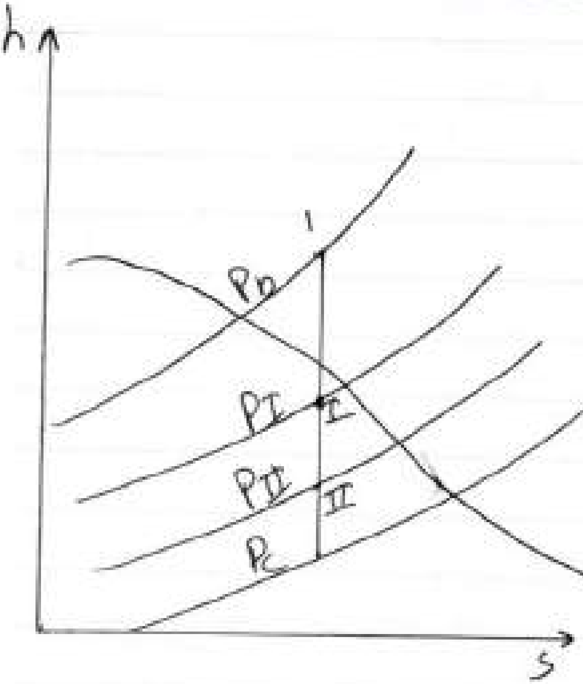
a) masses of bled steam of each heater.

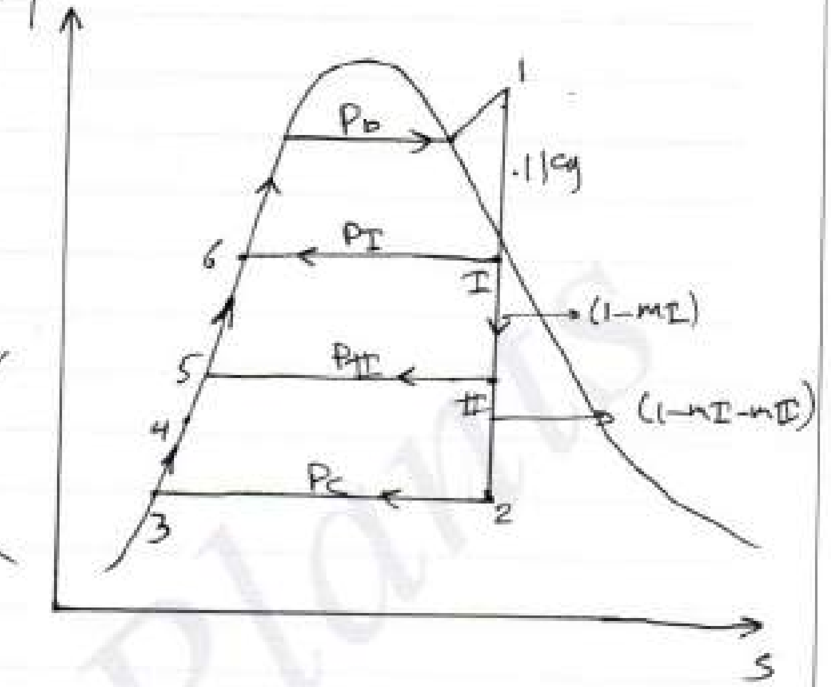
b) Thermal efficiency of the cycle.

Neglect all pumps work.



Power Plants





From Mollier chart

At 30 bar, 350 C

$h_1 = 3115$ KJ/Kg

$h_I = 2720$ KJ/Kg

$h_{II} = 2450$ KJ/Kg

$h_2 = 2120$ KJ/Kg

From steam table

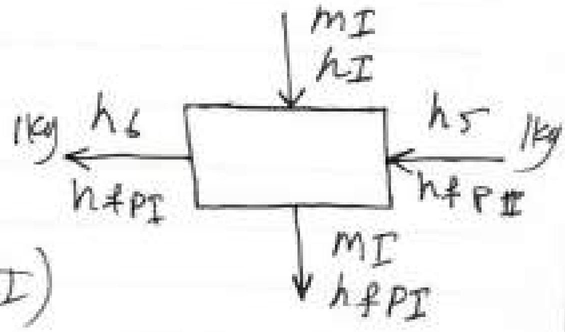
$h_3 = h_{fP_c} = 173.9$ KJ/Kg

$h_5 = h_{fP_{II}} = 417.5$ KJ/Kg

$h_6 = h_{fP_I} = 640.1$ KJ/Kg

Heat balance

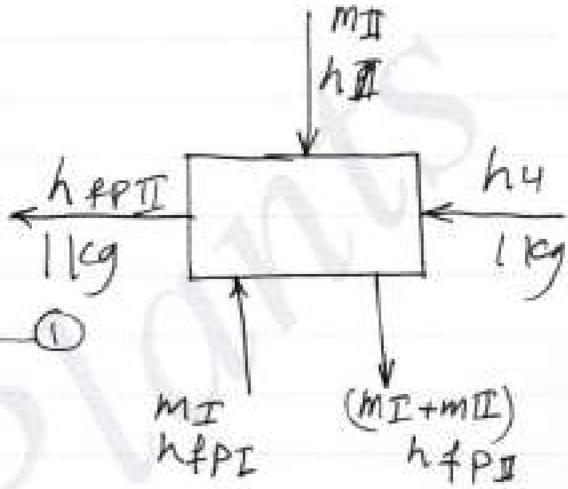
1st heater



$$m_I(h_I - h_{fP_I}) = 1\,Kg\,(h_{fP_I} - h_{fP_{II}})$$

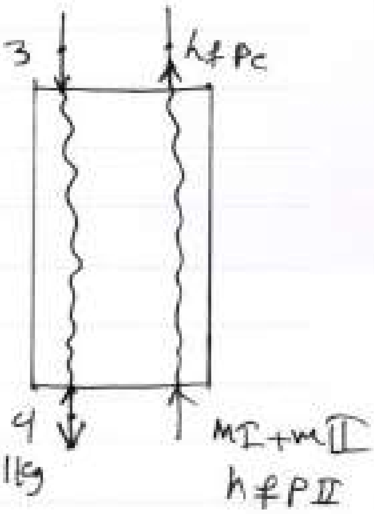
$$m_I = \frac{640.1 - 417.5}{2720 - 640.1} = 0.107 \text{ kg}$$

- 2nd heater



$$h_{II} m_{II} + m_I h_{fPI} + h_4 * 1\text{kg}$$
$$= h_{fPII} * 1\text{kg} + (m_I + m_{II})\, h_{fPII} \quad \text{——} ①$$

- Drain Cooler



$$1\text{kg}\,(h_{\cancel{4}} - h_3) = (m_I + m_{II})(h_{fPII} - h_{fPc})$$

$$h_4 = (m_I + m_{II})(h_{fPII} - h_{fPc}) + h_3 \quad \text{——} ②$$

Inserting the value of h_4 in equ ① :-

$$m_{II} h_{II} + m_I h_{fPI} + (m_I + m_{II})(h_{fPII} - h_{fPc}) + h_3$$
$$= (m_I + m_{II})\, h_{fPII} + h_{fPII}$$

$$2456\, m_{II} + 0.107 * 640.1 + (0.107 + m_{II})(417.5 - 173.9) + 173.9$$
$$= (0.107 + m_{II}) * 417.5 + 417.5$$

$$2450 m_{II} + 68.4967 + 243.6 m_{II} + 26.06 + 173.9 = 417.5 m_{II} + 44.673 + 417.5$$

$$2276.1\, m_{II} = 193.72$$

$$m_{II} = 0.085 \text{ Kg}$$

$$W.D = 1\text{Kg}(h_1 - h_I) + (1 - m_I)(h_I - h_{II}) + (1 - m_I - m_{II})(h_{II} - h_2)$$

$$= (3115 - 2720) + (1 - 0.107)(2720 - 2450) + (1 - 0.107 - 0.085)(2450 - 2120)$$

$$= 903 \text{ KJ/Kg}$$

$$Q_{add} = (h_1 - h_6) \times 1\text{Kg}$$

$$= (3115 - 640.1) = 2475.2 \text{ KJ/Kg}$$

$$\eta_{cycle} = \frac{W.D}{Q_{add}}$$

$$= \frac{903}{2475.2} = 36.48\ \%$$

Mohanad Alheety

Power Plants

Ex/ In a steam power plant, steam enters a H.P.T at 100 bar and 500°C and exits at 25 bar. Steam is re-heated to 490°C. Then, it enters intermediate pressure turbine and expands to 3 bar. Then, it is reheated again to 311°C, and it enters the L.P.T and expand to the condenser pressure of 0.05 bar. A closed feed water heater is located in the feed cycle and uses bleed steam from the H.P.T at 40 bar. The net output power is 100 MW.

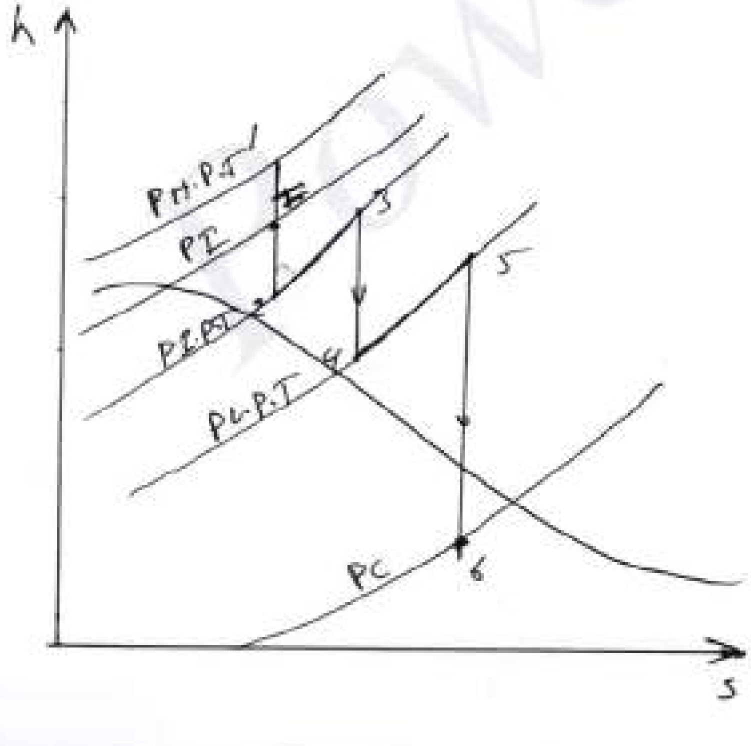
Determine :-

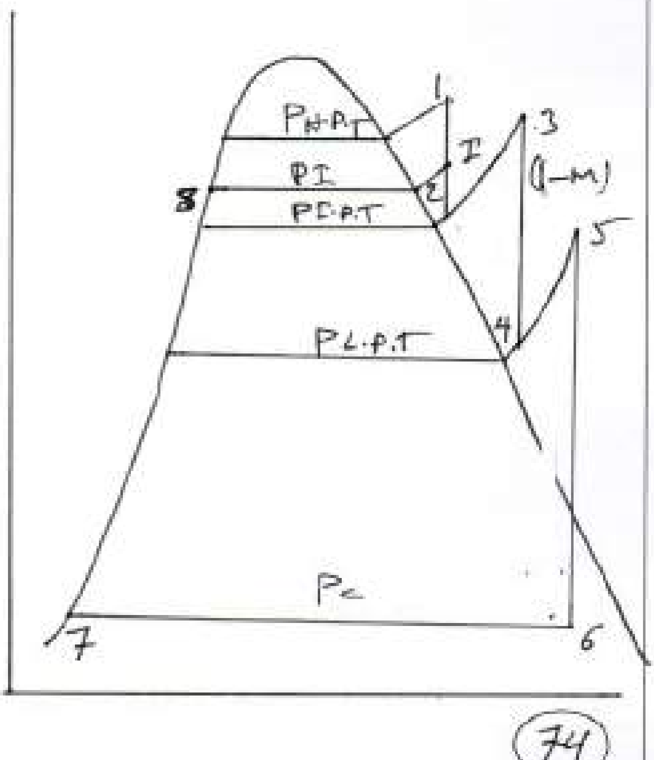
a) amount of steam used for bleeding.

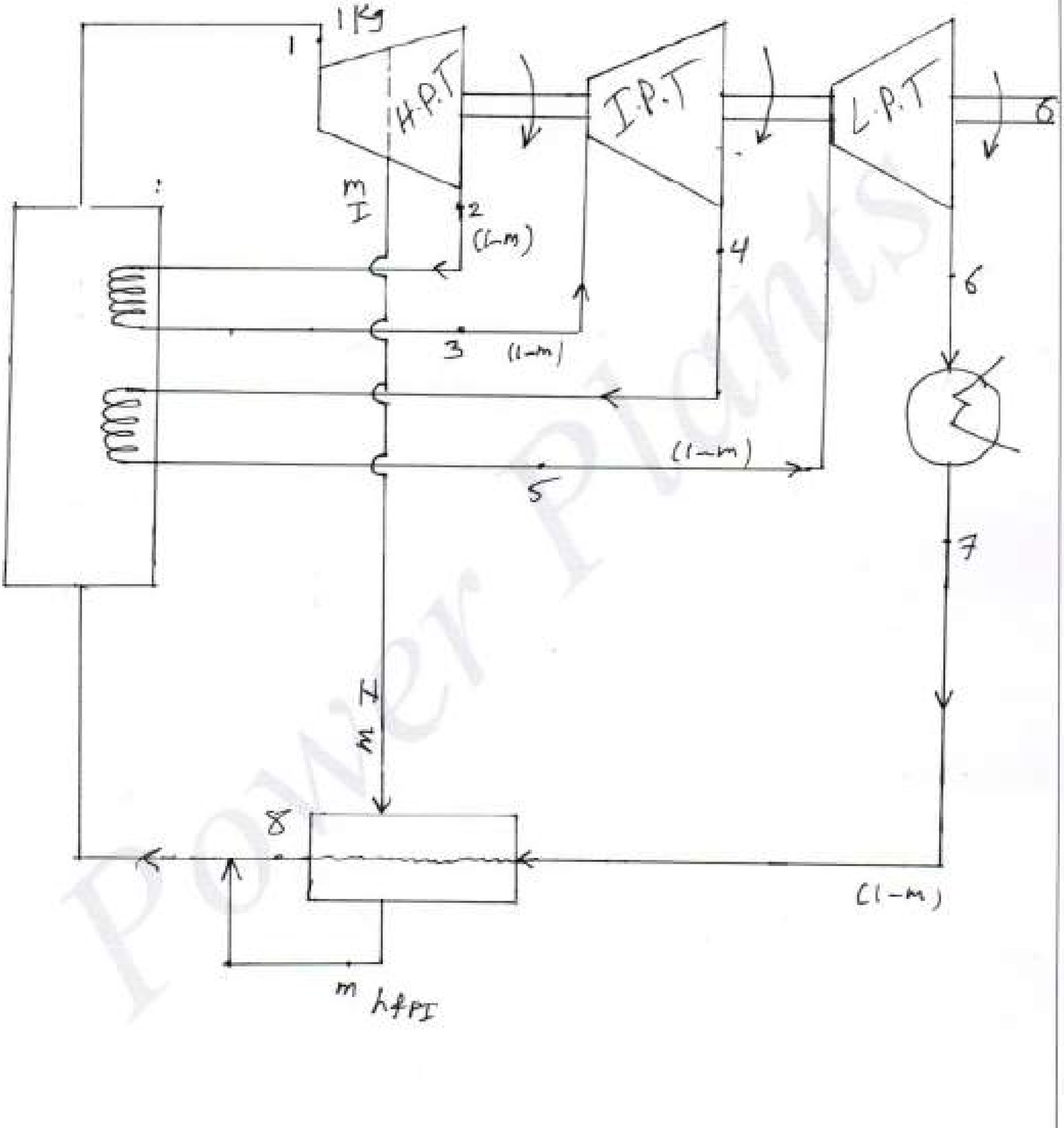
b) mass flow rate of the steam

c) Work done

d) Cycle efficiency.






1 Kg
1
H.P.T
I.P.T
L.P.T
G
m I
2
(1-m)
4
3
(1-m)
6
(1-m)
5
7
m I
8
(1-m)
m h4PI

From Mollier Chart

$h_1 = 3375$ KJ/Kg

$h_2 = 2985$ KJ/Kg

$h_I = 3095$ KJ/Kg

$h_3 = 3440$ KJ/Kg

$h_4 = 2860$ KJ/Kg

$h_5 = 3090$ KJ/Kg

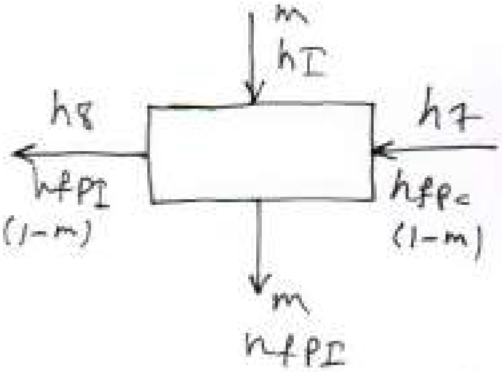
$h_6 = 2360$ KJ/Kg

From Steam table

$h_7 = h_{fPc} = 137.82$ KJ/Kg

$h_8 = h_{fPI} = 1087.3$ KJ/Kg

Heat balance for heater



$$m(h_I - h_{fPI}) = (1-m)(h_{fPI} - h_{fPc})$$

$$m(3095 - 1087.3) = (1-m)(1087.3 - 137.82)$$

$$2007.7\,m = 949.48 - 949.48\,m$$

$$m = 0.321 \text{ Kg}$$

$$W.D = (h_1 - h_I)\,1\text{kg} + (1-m)\left[(h_I - h_2) + (h_3 - h_4) + (h_5 - h_6)\right]$$

$$= (3375 - 3095) + (1 - 0.321)\left[(3375 - 2985) + (3440 - 2860) + (3090 - 2360)\right]$$

$$= 1434.3 \text{ KJ/Kg}$$

$$Q_{added} = (h_1 - h_8)1kg + (1-m)[(h_3-h_2) + (h_5-h_4)]$$

$$= (3375 - 1087.3) + (1-0.321)[(3440-2985) + (3090-2860)]$$

$$= 2752.8 \ KJ/kg$$

$$P_{output} = \dot{m}_s * W.D$$

$$\dot{m}_s = \frac{100\,000}{1434.3} = 69.7 \ Kg/s$$

$$\eta_{cycle} = \frac{1434.3}{2752.8} = 52.1\%$$

Power Plants

Sheet 1

Q1/ Steam is supplied to a two-stage at 40 bar and 350°C. It expands in the first turbine until it is just dry saturated, then it is reheated to 350°C and expanded through the second turbine. The condenser pressure is 0.035 bar. Calculate S.S.C and η_{cycle}

[2.79 Kg/kw hr, 38.4%]

Q2/ A steam power plant operates on ideal regenerative cycle with one open feed water heater. Steam enters the turbine at 15 MPa and 600°C. Some steam leaves the turbine at a pressure of 1.2 MPa and enters the open heater. Determine the fraction of steam extraction from the turbine and thermal efficiency of the cycle

[0.227 Kg, 46.3%]

Q3/ In a regenerative steam cycle emplying three closed feed water heaters, the steam is supplied to the turbine at 42 bar and 500°C and is exhusted to the condenser at 0.035 bar. The bleed steam for feed heating is taken at pressures of 15, 4 and 0.5 bar. Neglect work of the pumps, calculate:

a) Masses of bled steam
b) Work done
c) Cycle efficiency

[0.0952, 0.0969, 0.0902, 1183.6 KJ/Kg, 43.6%]

Q4/ A steam power plant operates on an ideal reheat-regenerative cycle and has a net ~~work~~ power output of 120 MW. Steam enters the high-pressure turbine at 10 MPa and 550°C and leaves at 0.8 MPa. Some steam is extracted at this pressure to heat the feed water in an open feedwater heater. The rest is reheated to 500°C and expanded in the low pressure turbine to the condenser pressure of 10 KPa. Determine

a) masses of bleed steam

b) cycle efficiency

[81.9 kg/s, 44.4%]

Q5/ In a steam power plant, the steam mass flow rate is 80 ton/hr, the steam enter the turbine at 6 MPa and 480°C. Condenser pressure is 0.06 MPa. Mechanical and electrical efficiencies are 0.96 and 0.95. Find

a) S.S.C

b) Output power

c) Cycle efficiency

Steam Generator

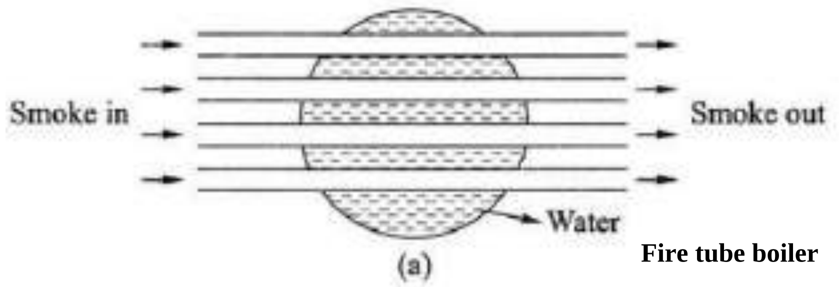
Steam generator is a closed vessel that is used to generate steam at constant pressure as per the process requirement. The steam generated may be wet, dry saturated or superheated in state. In modern power plants, it is very common to use one boiler (single unit) per turbine, which leads to simpler piping systems and relatively easier boiler and turbine control. These boilers are usually designed to operate either at critical pressure (221.2 bar) or above or below the critical pressure.

In a steam generator or boiler, constant pressure is maintained by balancing the rate of steam generated with the rate of steam consumed. In a thermal power station, coal is the main source of combustion. The heat generated by burning coal is utilized to generate steam, which in turn runs the turbo-generator.

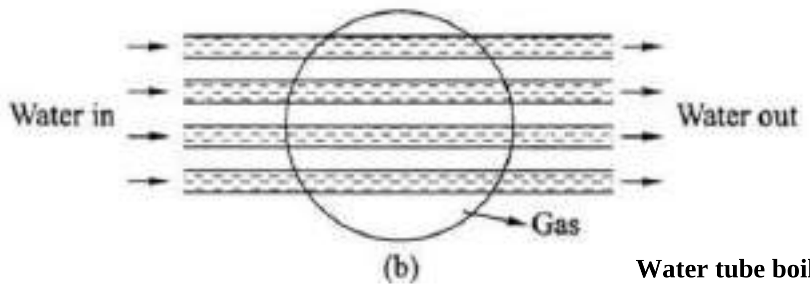
Classification of Boilers

- **Fire tube and water tube boilers**

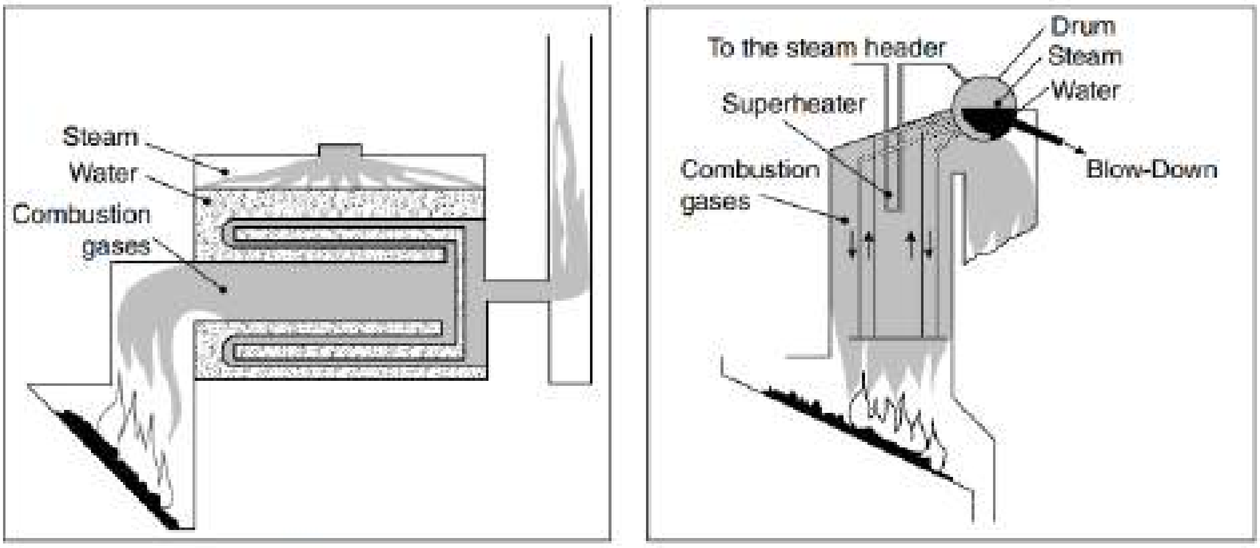
In the fire tube boilers, the hot gases produced after the combustion of fuel pass through the tubes and the water surrounds these tubes. In the water tube boilers, water flows inside tubes and gas remains outside these tubes.



Fire tube boiler



Water tube boiler



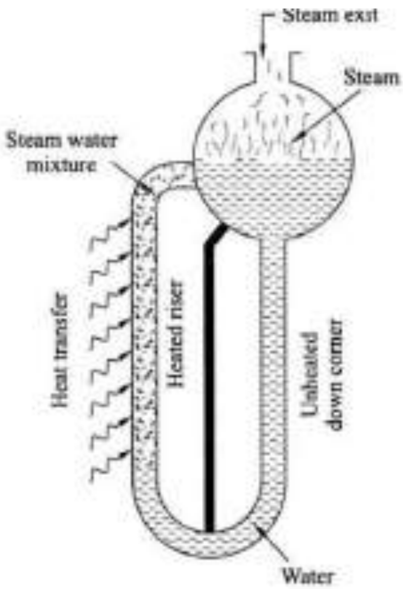
(a) **Fire tube boiler**

(b) **Water tube boiler**

➢ Natural and Forced Circulation Boiler:

All conventional boilers in which the circulation of water is done by thermo-symphonic method, are known as natural circulation boilers. In natural circulation boiler, the circulation of water takes place due to natural convection current and density variation of fluid by the application of heat, i.e., thermo-siphon.

This is based on the principle that the density of water is more than steam and as such requires a boiler drum. The method is effective only up to a pressure less than 20 bar. After this pressure, the difference in density of water and steam becomes less and so this system fails to work.



In forced circulation type boilers, the circulation of water inside the tube is done by a forced circulating pump. Water is forced inside the tube by mechanical means. This method is applicable in high-capacity water tube boilers.

- **Based on Heat Sources**

Boilers may be classified based on the fuel used for combustion or heat generation source. Various heat sources used may be the following:

(i) Heat generated by the combustion of fuel in solid, liquid or gaseous forms

(ii) Heat generated by hot waste gases as byproducts of other chemical processes

(iii) Heat generated by nuclear energy

- **Stationary and Mobile Boilers**

If the boilers are used at one place only they are termed stationary boilers. These boilers are used either for process heating in industries or for power generation in steam power plants.

Mobile boilers are portable and are used in locomotives and ships. Examples: Locomotive boiler and marine boiler

Essentials of a Good Steam Boiler

A good boiler must possess the following features:

i. The boiler should be capable of producing large quantity of steam of the required quality.
ii. The steam must be generated at minimum cost.
iii. The rate of steam generation must match with the requirement.
iv. It must occupy less floor area.
v. It must be started quickly.
vi. Construction should be so simple and well-designed that the repair and inspection can be done easily.

Accessories of Steam Boiler:

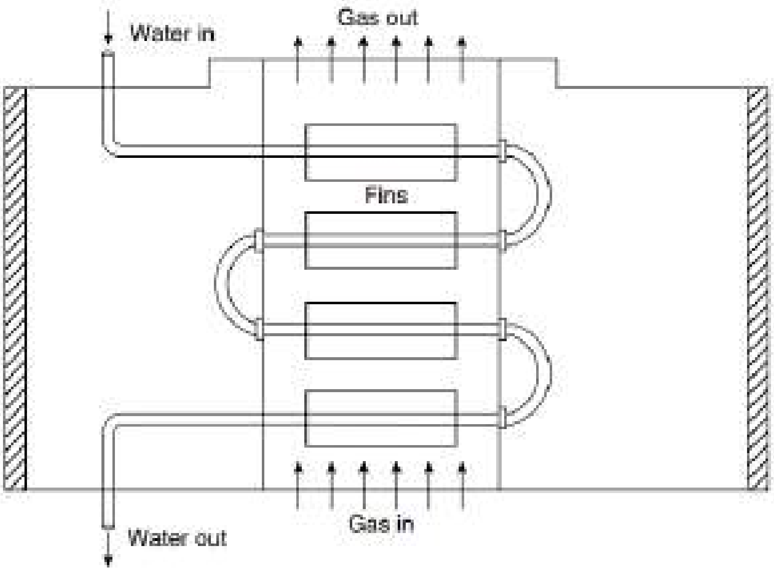
Boiler accessories are auxiliary systems in boiler installation for the proper functioning and increase of the thermal efficiency of the boiler.

The essential boiler accessories are as follows:

- **Economizer**

The purpose of economizer is to heat the feed water by the direct use of the heat of flue gas discharged to the atmosphere through chimney. The economizer reduces the temperature of flue gas. The feed water temperature is increased substantially.

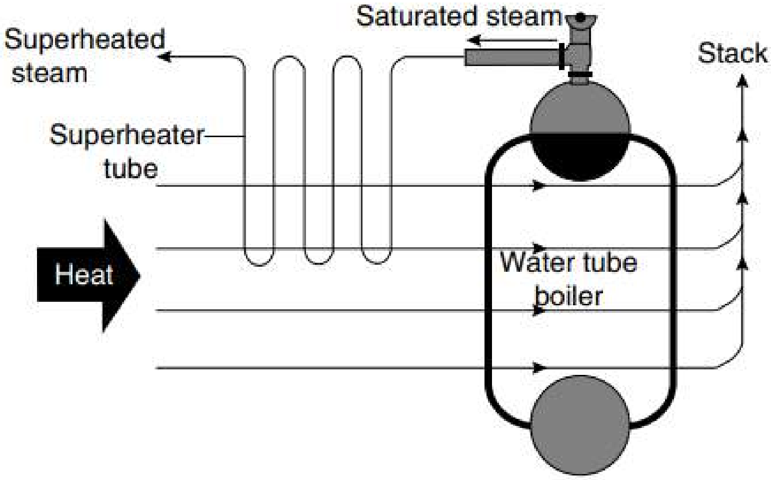
Thus, there is a saving in heat. The boiler efficiency is increased substantially. The economizer is placed in the path of the exit gas nearer to the boiler.



- **Superheaters**

The functions of the superheater are to remove the last traces of moisture from the saturated steam coming out of boiler and to increase its temperature above saturation temperature. Superheating increases overall cycle efficiency as well as avoids too much condensation in the last stages of the turbine, which avoids the blade corrosion.

Superheater is made of coils of tubes forming parallel tube circuits connected between heaters. The superheater tubes are made of high-temperature strength special alloy steels. The coils are heated by the heat of combustion gas during their passage from the furnaces to the chimney.



- **Steam separator**

The function of the steam separator is to separate the water particles in suspension that are carried by the steam coming from the boiler. If suspended water particles enter the turbine or engine, they cause erosion and corrosion of blades and other parts. It is always installed as close to the engine or turbine as possible on the main pipeline.

Steam Turbines

A Steam Turbine is an engine that converts heat energy from pressurized steam into mechanical energy where the steam is expanded in the turbine in multiple stages to generate the required work. Steam turbine engines are used to produce electricity.

Types of Steam Turbines

- **Impulse Steam Turbine**

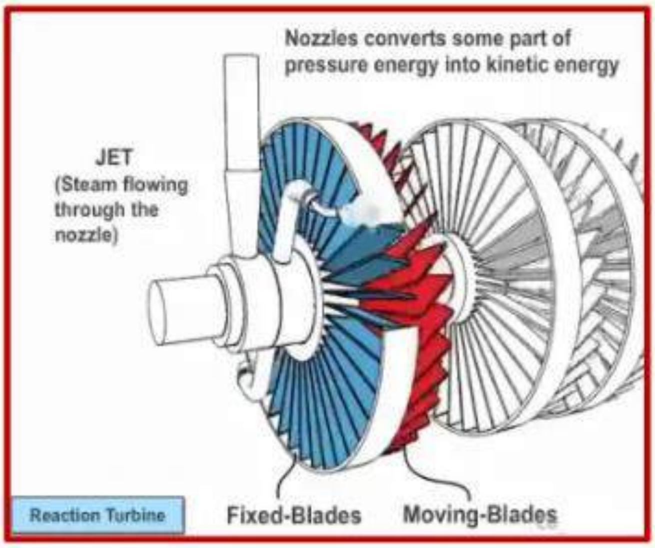
The basic idea of an impulse steam turbine is that a jet of steam from a fixed nozzle pushes against the rotor blades and impels them forward. So the impulse force of high-velocity steam exerts a force on the blade to turn the rotor. The kinetic energy of the steam is transferred to the rotating wheel by momentum transfer within the blades. Pelton Wheel, Banki Turbine, etc are typical examples of Impulse turbine.

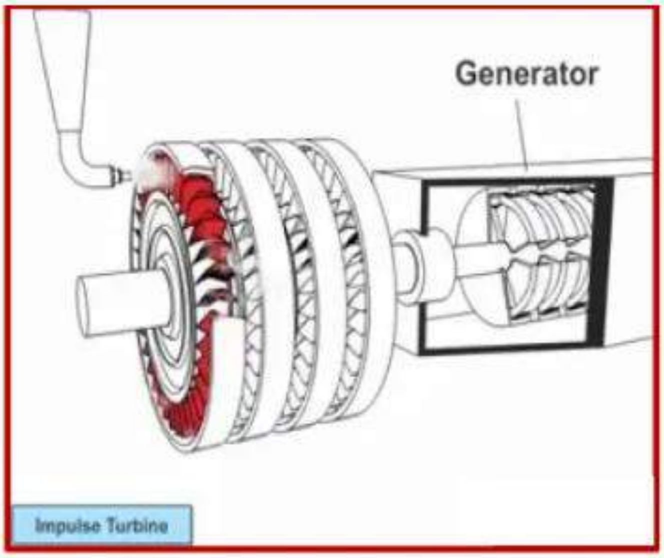
- **Reaction Steam Turbine**

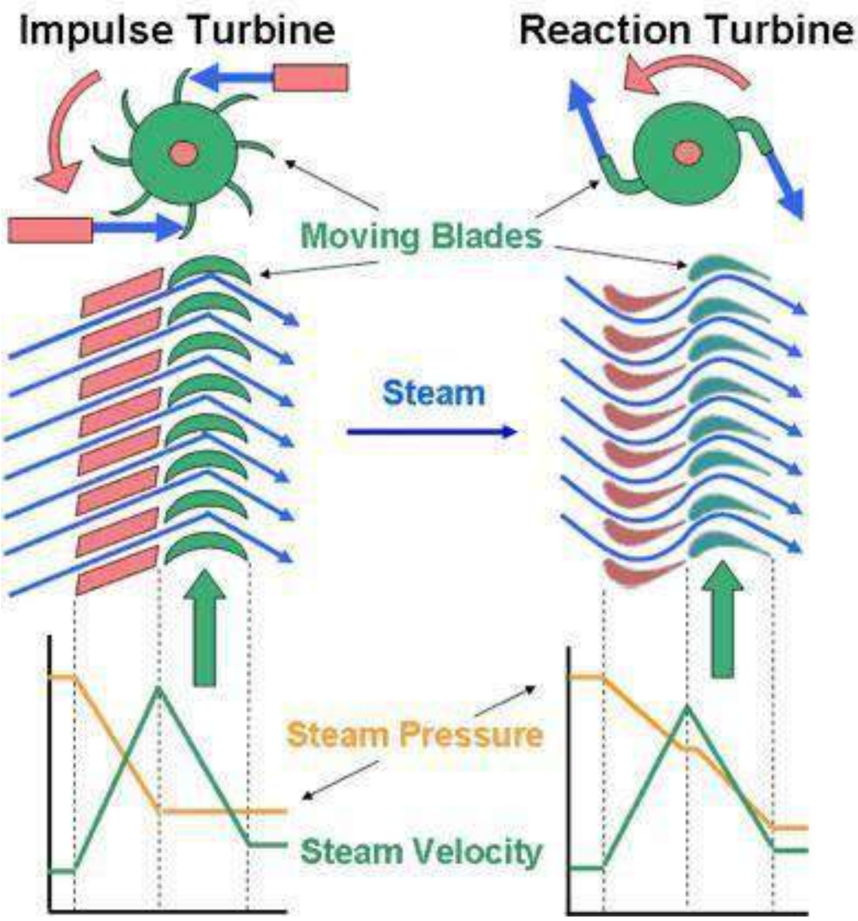
In the reaction steam turbine, a jet of steam flows from a nozzle on the rotor (the moving blades) by fixed blades designed to expand the steam. The rotor gets its rotational force from the steam as it leaves the blades. Roughly 50% of the output power is generated by the impact force and the other 50% from the reaction force by the steam expansion. Francis Turbine, Kaplan Propeller turbine, Deriaz turbine, etc are examples of reaction turbine.

The main difference between impulse and reaction turbine lies in the way in which steam is expanded while it moves through them such that:

- In the impulse type steam turbine, the steam expands in the nozzle and its pressure doesn’t change as it moves over the blades.
- In the reaction type, the steam expands continuously as it passes over the blades and thus there is a gradual fall in pressure during expansion.


Nozzles converts some part of pressure energy into kinetic energy
JET
(Steam flowing through the nozzle)
Reaction Turbine
Fixed-Blades
Moving-Blades


Generator
Impulse Turbine


Impulse Turbine
Reaction Turbine
Moving Blades
Steam
Steam Pressure
Steam Velocity

LOSSES IN STEAM TURBINE

Losses are all-time very important for manufacturing any machine. That's why Manufacturer takes special attention for manufacturing any machine. We know an ideal machine which has 100% gross efficiency will do the equivalent work to the isentropic enthalpy. It means turbine uses every single bit of heat drop produced by steam.

But practically turbine's work done is much less than isentropic heat drop of the steam used. Because some internal losses occurred at the time of its operation. These losses are directly affected by the turbines output as well as its efficiency

Though there are several losses in a turbine, but here we will discuss some important internal losses in the turbine.

- **Nozzle Friction Loss :-**

It is a very important loss for Impulse Turbine. When steam passes through the nozzles, friction loss occurs and the formation of eddies. Friction occurs in the nozzle due to the factor of nozzle efficiency and it is the ratio of actual enthalpy drop to isentropic enthalpy drop.

- **Blade Friction Loss :-**

This loss is important for both Impulse and Reaction turbine . Blade friction loss is due to the steam's gliding over the blades and friction of the surface of the blades

- **Wheel Friction Loss :-**

When steam passes through the rotating turbine wheel, it produces some resistance on the turbine wheel. As a result, it rotates in lower speed from its original speed. It is the loss in both Impulse and Reaction turbine. The total frictional loss is about 10% of total turbine loss.

- **Losses due to mechanical friction :-**

This loss is for turbine's bearing. Mechanical friction loss is due to the friction between the shaft and wheel bearing and also the regulating valve of the turbine. This loss may be reduced by proper lubrication of the moving parts of the turbine. This loss occurs both Impulse and Reaction turbine.

Steam Condensers

Steam condensers are devices in which the exhaust steam from the steam turbine is condensed by means of cooling water. Condensation can be done by removing heat from exhaust steam using circulating cooling water. During condensation, the working substance (steam) changes its phase from vapour to liquid and rejects latent heat.

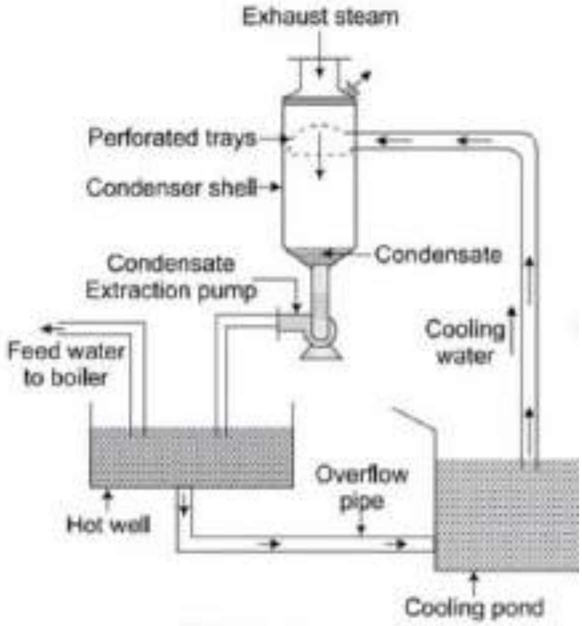
The main advantages of a steam condenser in a steam power plant are as follows:

- It increases the efficiency of the power plant due to increased enthalpy drop.
- It reduces temperature of the exhaust steam which also results in more work output.
- The condensed steam can be reused as feed water for boiler which reduces the cost of power generation.

Types of condensers

Mixing type condensers (Parallel flow jet condenser)

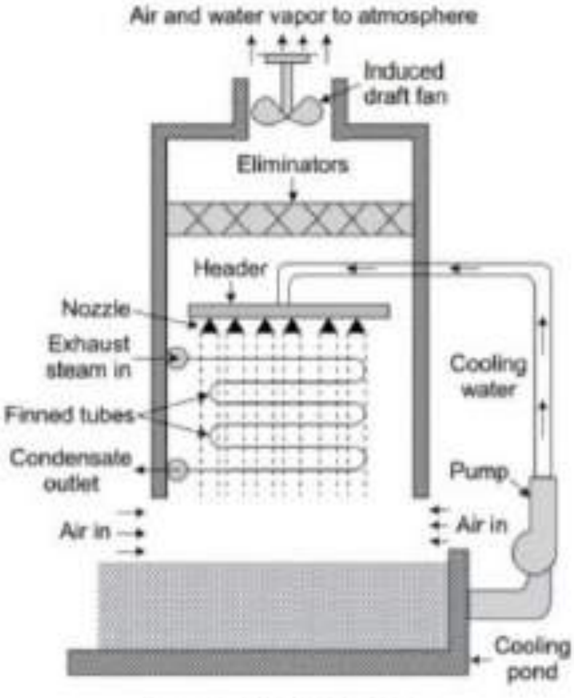
In parallel flow jet condenser both the steam and the water enters from the top and flows in the same direction. The exhaust steam is condensed when it mixes up with water. The condensate and the cooling water are delivered to the hot well from where surplus water flows to the cooling pond through an overflow pipe. Sometimes a single pump know as wet air pump is used to remove both air and the condensate but generally separate air pump is used to remove air as it gives a great vacuum.



Parallel Flow Jet Condenser

Non-mixing type condensers (Evaporative condenser)

In evaporative condenser the steam flows enters the gilled pipes and flows backwards and forwards in a vertical plane. The water pump sprays water on the pipes which condenses the steam. The quantity of cooling water needed to condense the steam can be reduced by causing the circulating water to evaporate which decrease the temperature. The remaining water is collected in the cooling pond.



Evaporative Condenser

University of Anbar

College of Engineering

Mechanical Eng. Department

Power Plants

Chapter two

Gas Turbine

Tutor

Mohanad A. Alheety

Gas Turbine

A gas turbine, also called a combustion turbine, is a type of internal combustion engine. It has an upstream rotating compressor coupled to an downstream turbine, and a combustion chamber in between.

The basic operation of the gas turbine is similer to that of the steam power plant expcept that ~~the~~ air is used instead of water.

Applications:-

- Aircraft field
- Electrical Power generation
- Marine propulsion

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Advantages :-

- Very high power-to-weight ratio
- Smaller than most reciprocating engines of the same power rating.
- Fewer moving parts than reciporocating engines.
- Very low toxic emissions

Disadvantages :-

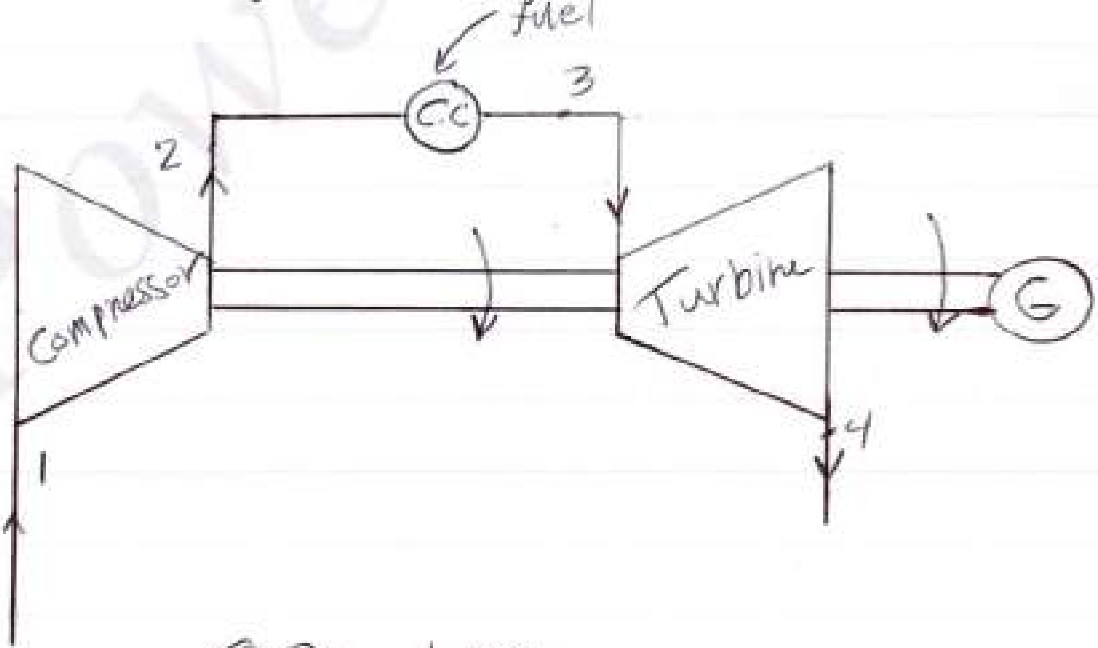
- Cost is very high
- Less efficient than reciprocating engines at idle speed.
- Very sensitive to changes of the components.

Bryton Cycle

Brayton Cycle popularly used for gas turbine plants. It comprises of adiabatic comp-ression process, constant pressure heat addition adiabatic expansion process and constant pressure heat release process.

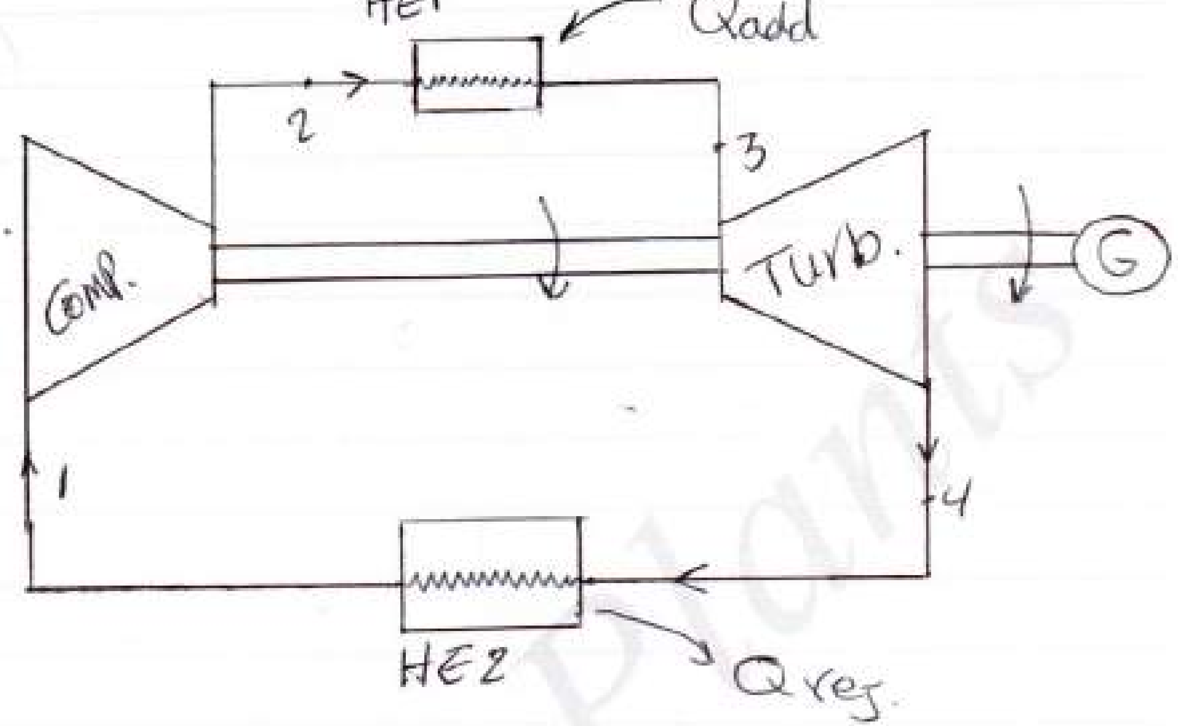
There are two types of gas turbine power plants:-

- Open type.
- Closed type.

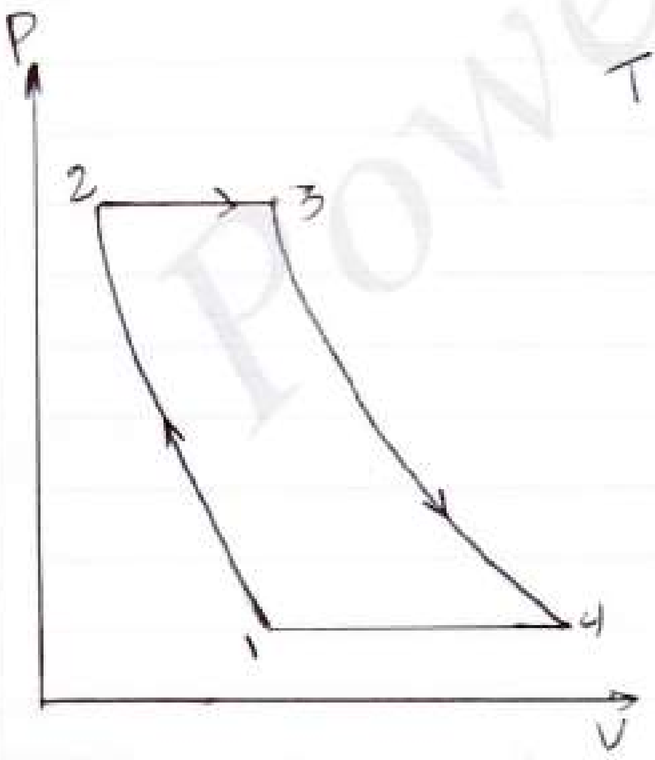


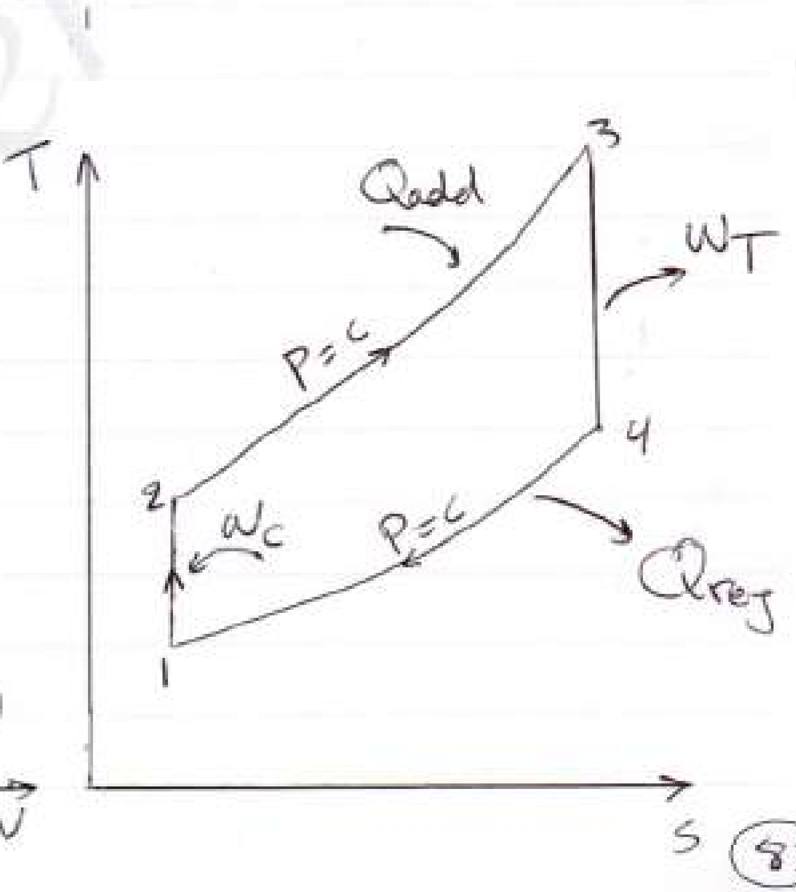
Open type

Power Plants



Closed type





Thermodynamic cycle shows following processes:

1-2 : Adiabatic compression, involving (-ve) work, W_C in compressor.

2-3 : Constant pressure heat addition, involving heat addition Q_{add} in combustion chamber of heat exchanger.

3-4 : Adiabatic expansion, involving (+ve) work, W_T in turbine

4-1 : Constant pressure heat rejection, involving heat Q_{rej} in atmosphere or heat exchanger.

It will be assumed that kinetic energy is zero. After applying the flow equation to each part of the plant, we get :-

For compressor :

$$W_c = Cp_a (T_2 - T_1) \quad \frac{kJ}{kg}$$

For the combustion chamber :

$$Q_{add} = Cp_g (T_3 - T_2)$$

For the turbine :

$$W_T = Cp_g (T_3 - T_4)$$

Then,

$$W_{net} = W_T - W_c$$
$$= Cp_g (T_3 - T_4) - Cp_a (T_2 - T_1)$$

Thermal eff. $\eta = \dfrac{\text{net work output}}{\text{heat supplied}}$

$$= \frac{Cp_g (T_3 - T_4) - Cp_a (T_2 - T_1)}{Cp_g (T_3 - T_2)}$$

For a perfect gas law, in terms of pressure ratio across the turbine (r_{PT}) given by:

$$r_{PT} = \frac{P_3}{P_4}$$

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma_g - 1}{\gamma_g}} \Rightarrow \frac{T_3}{T_4} = \left(r_{PT}\right)^{\frac{\gamma_g - 1}{\gamma_g}}$$

Where γ is the ratio of specific heat at constant pressure and constant volume

$$\gamma = \frac{CP}{CV}$$

The same thing is applied across the compressor ,

$$r_{pc} = \frac{P_2}{P_1}$$

$$\frac{T_2}{T_1} = (r_{pc})^{\frac{\gamma_a - 1}{\gamma_a}}$$

$$P_{output} = \dot{m} * W_{net}$$

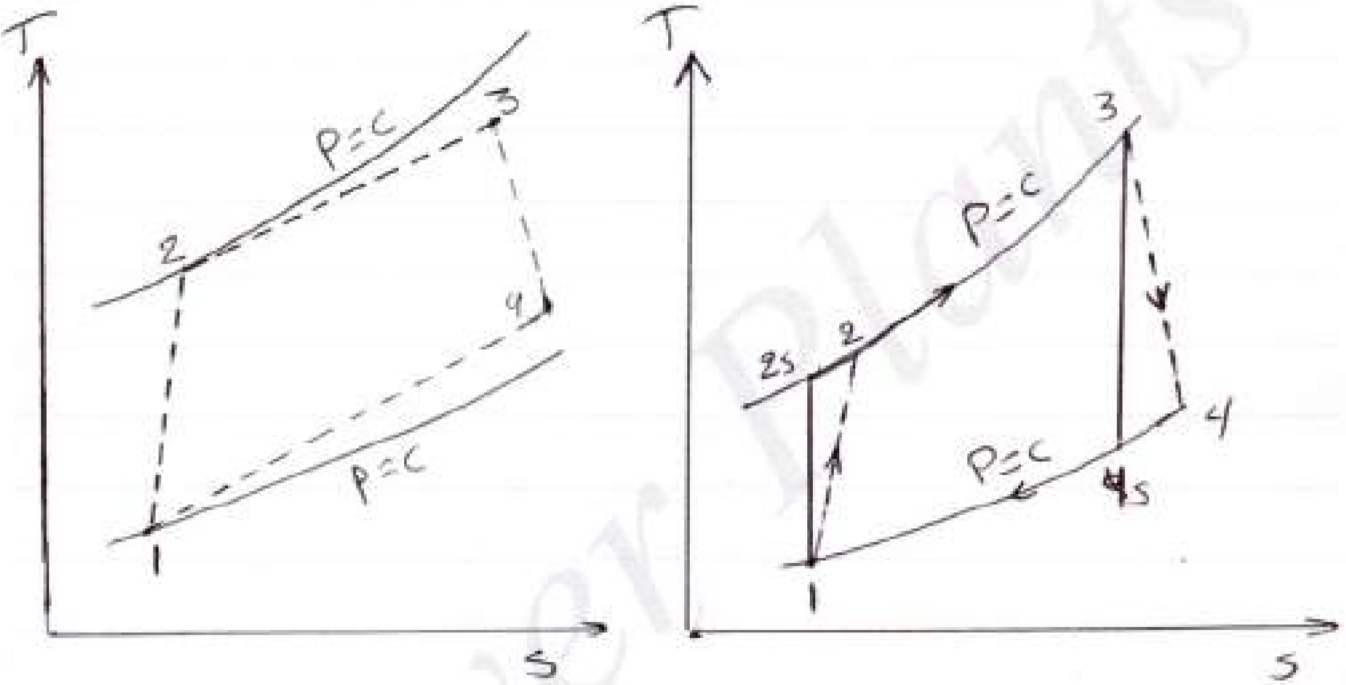
Where, $\dot{m}$ = mass flow rate of air (kg/s)

Gas turbine irreversibilites and losses :-

Because of frictional effects within the compressor and turbine, the working fluid would experience increases in specific entropy across these components.

Owing to friction, there also would be pressure drops as the working fluid passes through the heat exchangers.

However, because frictional pressure drops are less significant sources of irreversibility, We ignore them and for simplicity show the flow through the heat exchangers as occuring at constant pressure.

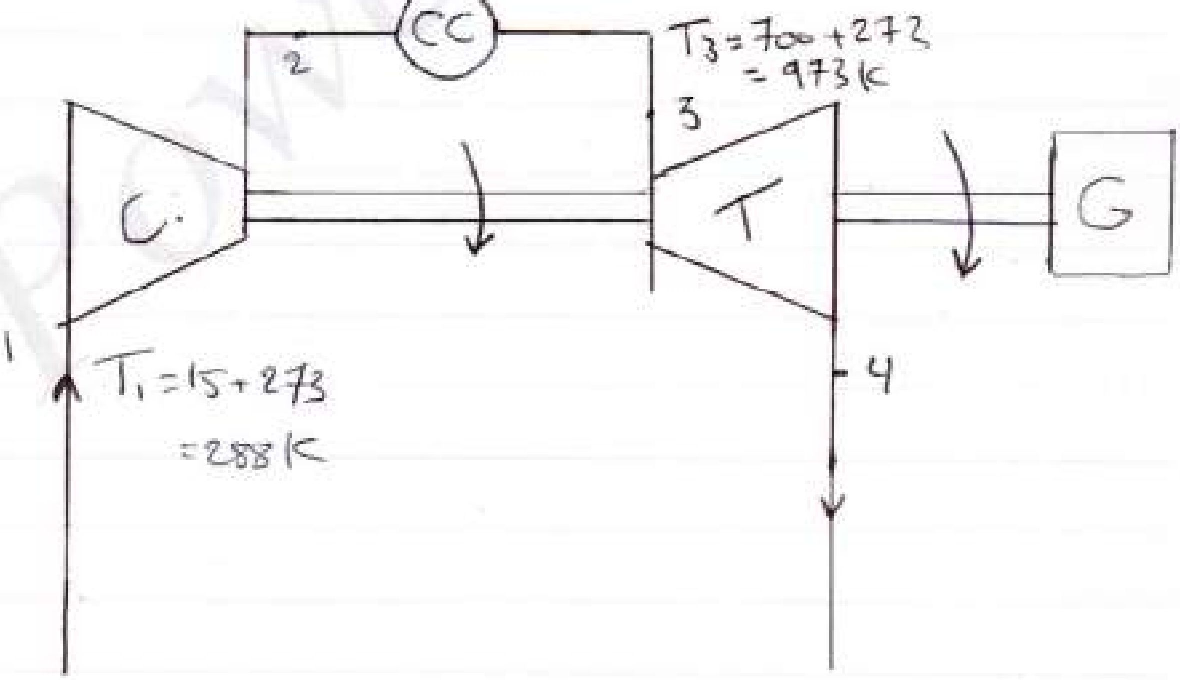


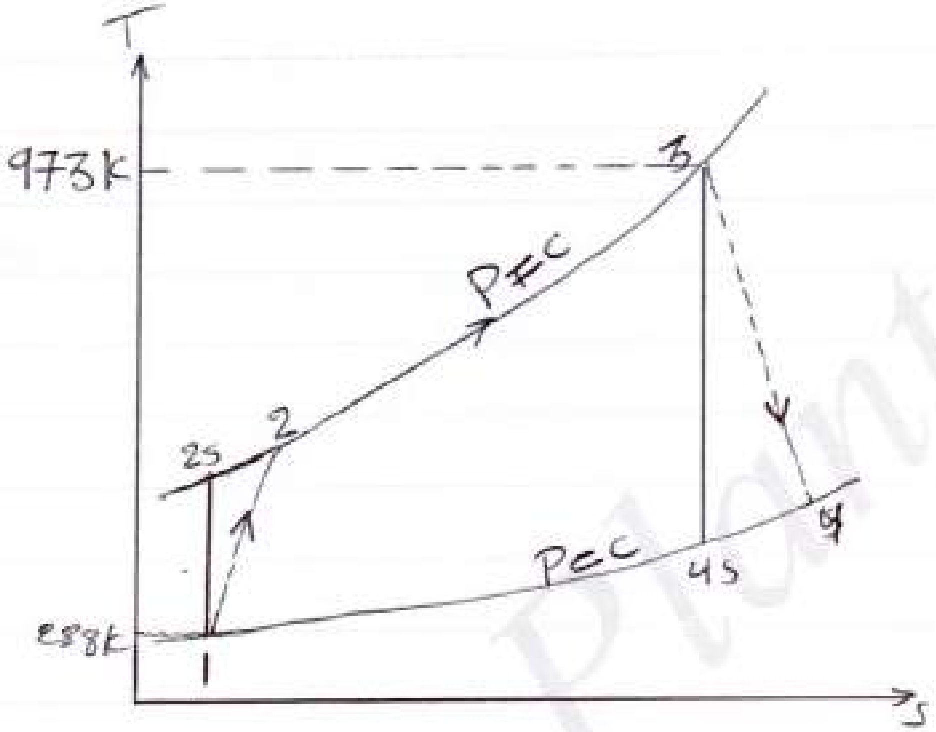
$$\eta_t = \frac{Cp_g(T_3 - T_4)}{Cp_g(T_3 - T_{4s})} = \frac{T_3 - T_4}{T_3 - T_{4s}}$$

$$\eta_c = \frac{Cp_a(T_{2s} - T_1)}{Cp_a(T_2 - T_1)} = \frac{T_{2s} - T_1}{T_2 - T_1}$$

Ex/ A gas turbine unit has a pressure ratio of 10/1 and a maximum cycle tempressure of 700 °C. The isentropic efficiencies of the compressor and turbine are 0.82 and 0.85 respectively. Calculate the power output of the turbine when the air enters the compressor at 15 °C at the rate of 15 Kg/s. Take $C_P = 1.005$ Kj/Kg K and $\gamma = 1.4$ for the compression process, and take $C_P = 1.11$ Kj/Kg K and $\gamma = 1.33$ for the expansion process. Also calculate η_{th}?

Solution:-





$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\gamma-1/\gamma}$$

$$T_{2s} = 288 \times (10)^{\frac{0.4}{1.4}} = 556\ K$$

$$\eta_c = \frac{T_{2s}-T_1}{T_2-T_1}$$

$$0.82 = \frac{556-288}{T_2-288} \longrightarrow T_2 = 614.8\ K$$

For turbine

$$\eta_t = \frac{T_3-T_4}{T_3-T_{4s}}$$

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$$\frac{T_3}{T_{4s}} = \left(\frac{P_2}{P_1}\right)^{\gamma-1/\gamma}$$

$$T_{4s} = \frac{973}{10^{\frac{0.333}{1.333}}} = 547.4\ K$$

$$0.85 = \frac{973 - T_4}{973 - 547.4} \Rightarrow T_4 = 611.2\ K$$

$$W_c = CP\,(T_2 - T_1)$$
$$= 1.005\,(614.8 - 288)$$
$$= 328.4\ KJ/kg$$

$$W_t = CP\,(T_3 - T_4)$$
$$= 1.11\,(973 - 611.2)$$
$$= 401.6\ KJ/kg$$

Thus, $W_{net} = W_t - W_c$
$$= 401.6 - 328.4$$
$$= 73.2\ KJ/kg$$

$$P_{output} = \dot{m}_a * W_{net}$$
$$= 15 * 73.2 = 1098\ kW$$

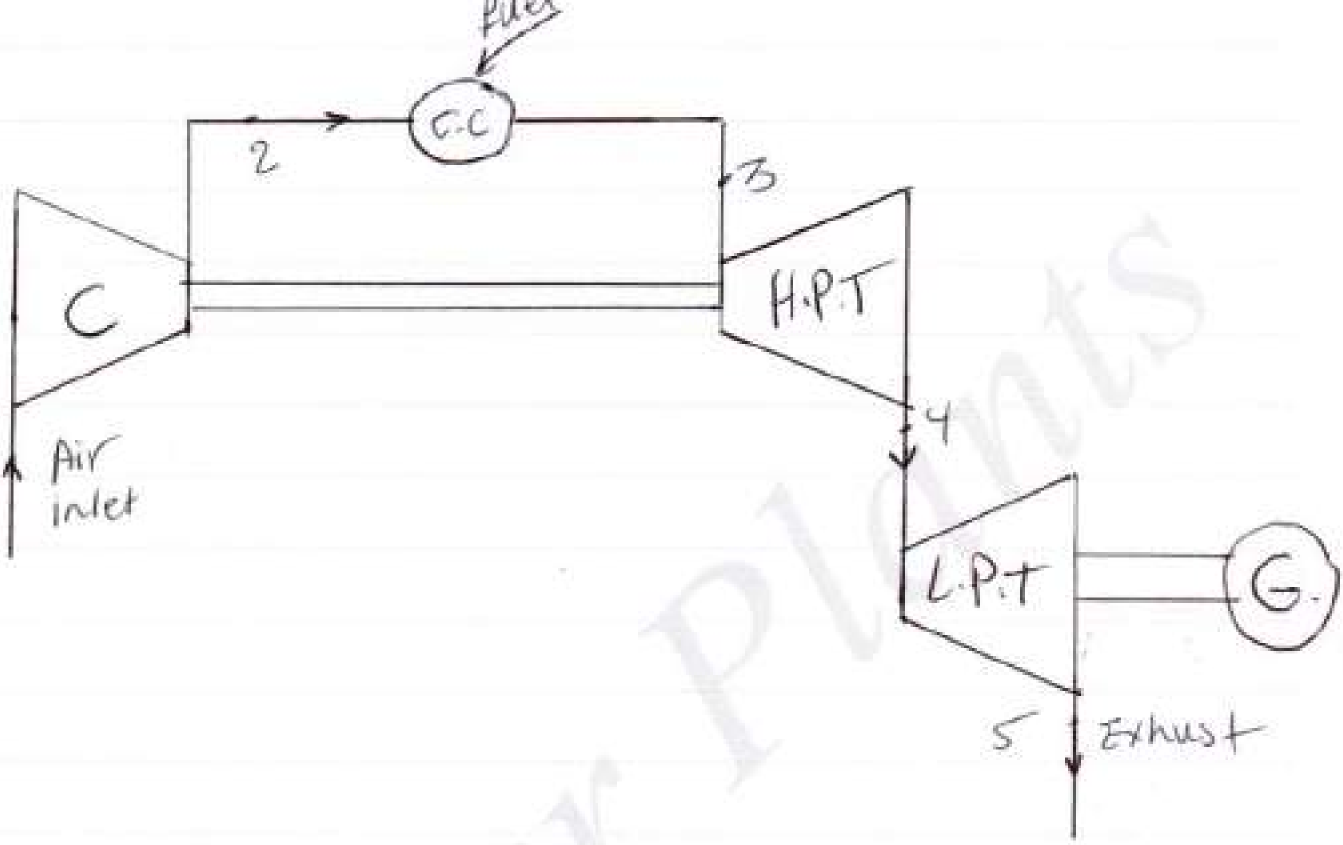
$$Q_{add} = CP_g(T_3 - T_2)$$
$$= 1.11(973 - 614.8)$$
$$= 397.6 \ KJ/kg$$

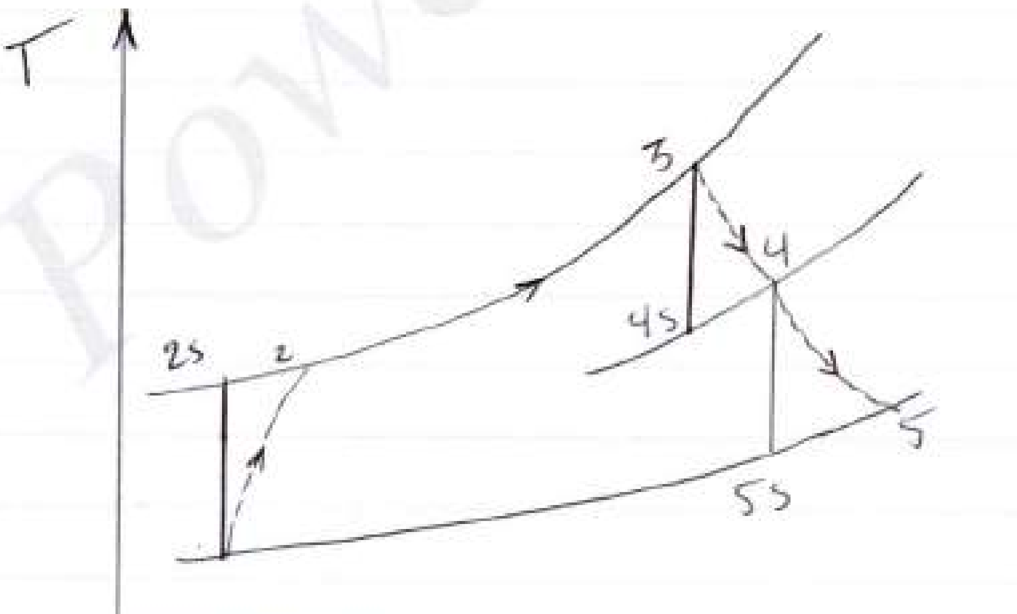
$$\eta_{th} = \frac{W_{net}}{Q_{add}} = \frac{73.2}{397.6} = 18.4\%$$

Use of a power turbine

It is sometimes more convenient to have two seperate turbines, one of which drives the compressor while the other provides the power output.

The first, or high-pressure (HP) turbine, is called as the compressor turbine while the other, or low-pressure (LP) turbine is called as the power turbine.


Fuel
C.C
2
3
C
H.P.T
4
Air
inlet
L.P.T
G.
5
Exhust


T
3
4
4s
2s
2
5
5s
s

as stated before, work of HP turbine is equal to the work of the compressor.

Thus, $W_{H.P.T} = W_c$

$$C_{Pg}(T_3 - T_4) = C_{Pa}(T_2 - T_1)$$

And, $W_{net} = W_{L.P.T}$

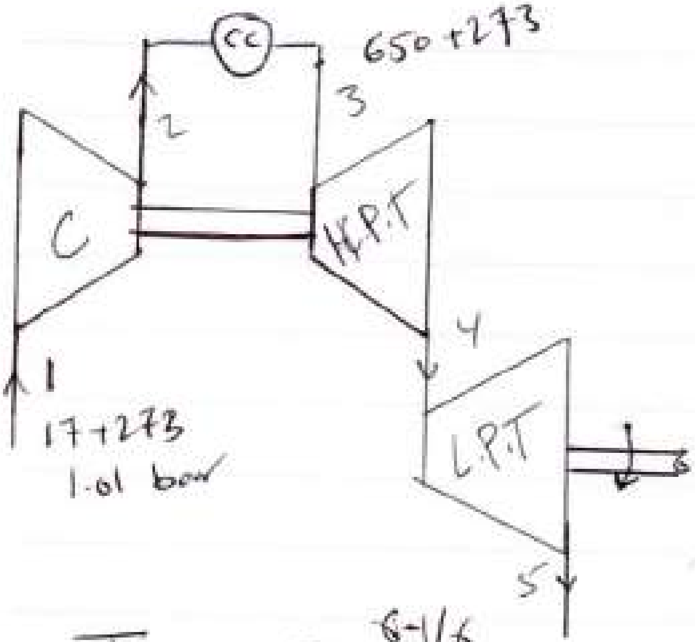
$$W_{net} = C_{Pg}(T_4 - T_5)$$

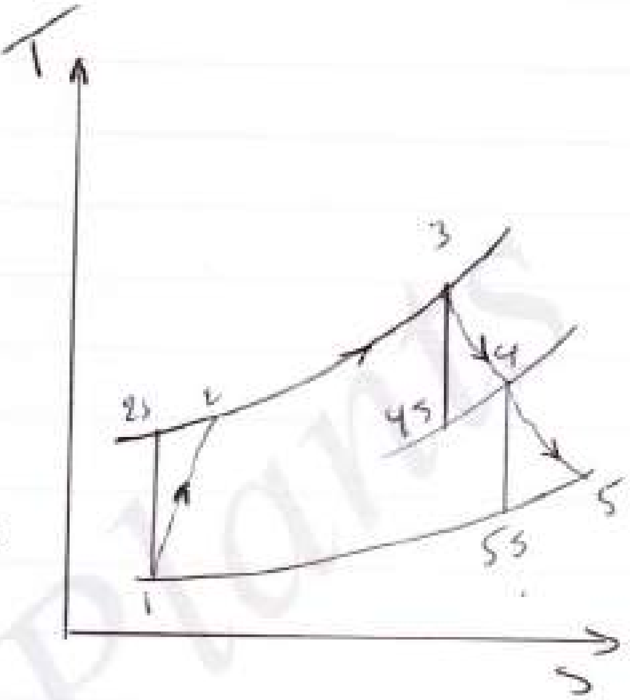
Ex/ A gas turbine unit takes in air at 17°C and 1.01 bar and the pressure ratio is 8/1. the compressor is driven by the H.P.T and the L.P.T drives a sperate shaft. The isentropic efficiencies of the compressor, and the HP and LP turbine are 0.8, 0.85, and 0.83 respectively Calculate the pressure and temprature of the gases entering the power turbine, the net power developed by the unit per kg/s mass flow rate, and cycle efficiency

$C_{Pg} = 1.15$, $\gamma_g = 1.333$, Maximum Cycle temp. = 650°C

$C_{Pa} = 1.005$, $\gamma_a = 1.4$

Solution :-





$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\gamma-1/\gamma}$$

$$T_{2s} = 290 \times 8^{\frac{0.4}{1.4}} = 525\ K$$

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1}$$

$$0.8 = \frac{525 - 290}{T_2 - 290} \Longrightarrow T_2 = 584\ K$$

Then, $W_c = C_{pa}(T_2 - T_1)$

$$= 1.005 \times 294 = 295.5\ KJ/Kg$$

Now, the work output from H.P.T must be sufficient to drive the compressor,

$$W_{H.P.T} = Cp_g (T_3 - T_4) = 295.5$$

$$1.11 (923 - T_4) = 295.5$$

$$T_4 = 666 \text{ K}$$

$$\eta_{t\,H.P.T} = \frac{T_3 - T_4}{T_3 - T_{4s}} \Rightarrow 0.85 = \frac{923 - 666}{923 - T_{4s}}$$

$$T_{4s} = 620.5 \text{ K}$$

$$\frac{T_3}{T_{4s}} = \left(\frac{P_3}{P_4}\right)^{\gamma - 1/\gamma}$$

or, $$\frac{P_3}{P_4} = \left(\frac{T_3}{T_{4s}}\right)^{\frac{\gamma}{\gamma - 1}}$$

$$\frac{8 + 1.01}{P_4} = \left(\frac{923}{620.5}\right)^{\frac{1.333}{0.333}} \Longrightarrow P_4 = 1.65 \text{ bar}$$

To find the power output, it is necessary to evaluate T_5.

$$\frac{P_4}{P_5} = \frac{P_4}{P_3} \times \frac{P_2}{P_1} \quad (\text{since } P_2 = P_3 \text{ and } P_5 = P_1)$$

Therefore, $$\frac{P_4}{P_5} = \frac{8}{4.9} = 1.63$$

Then, $\frac{T_4}{T_{5s}} = \left(\frac{P_4}{P_5}\right)^{\frac{\gamma-1}{\gamma}}$

$T_{5s} = \frac{666}{1.131} = 588\ K$

$\eta_{L.P.T} = \frac{T_4 - T_5}{T_4 - T_{5s}}$

$0.83 = \frac{666 - T_5}{666 - 588} \Longrightarrow T_5 = 601.2\ K$

Then, $W_{L.P.T} = Cp_g\,(T_4 - T_5)$

$= 1.15\,(666 - 601.2)$

$= 74.5\ KJ/Kg$

$P_{output} = m_a * W_{L.P.T}$

$= 1 * 74.5 = 74.5\ KW$

$Q_{add} = Cp_g\,(T_3 - T_2)$

$= 1.15 * (923 - 584)$

$= 390\ KJ/Kg$

$\eta_{cycle} = \frac{W_{net}}{Q_{add}} = \frac{74.5}{390} = 19.1\ \%$

Modifications of Brayton Cycle

Generally, the thermal efficiency of brayton cycle is low. Thermal efficienay can be increased by incorporating thermal refinements in the simple cycle suchas :-

- Intercooling between compressor stages
- Reheating between turbine stages
- Regeneration

The thermal refinements can raise the plant efficiency to over 30% and thereby obliterate the advantage of fuel possessed by diesel or steam power plants.

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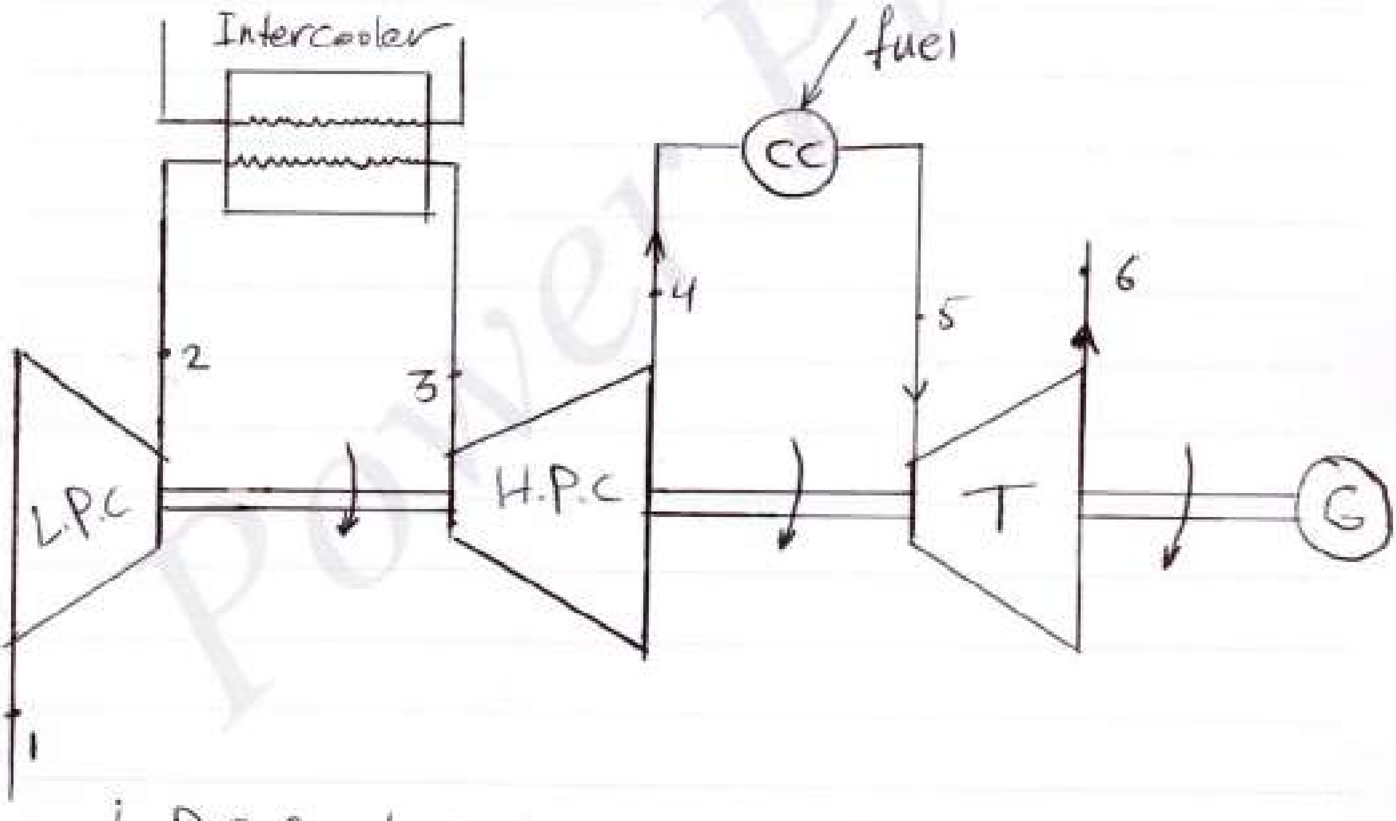
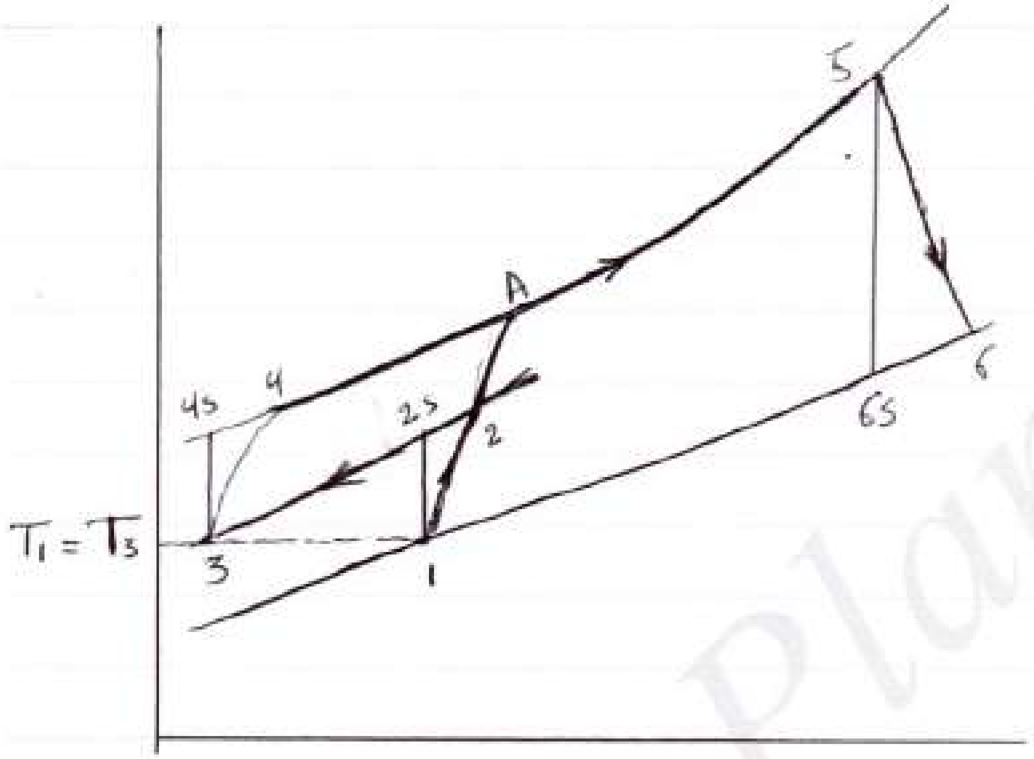
Intercooling

Net work of gas turbine can be increased by reducing negative work i.e. compressor work.

Multistaging of compression process with intercooling in between is one of the approaches for reducing compression work.



L.P.C : low-pressure compressor
H.P.C : High-pressure compressor



1-2 : Air is compressed ~~the~~ across the L.P.C

2-3 : Air is sent to intercooler where tempreture is reduced from T_2 to T_3

3-4 : Air is compressed across the H.P.C

4-5 : Heat is added to the compressed air

5-6 : Air is expanded across the turbine to produce work.

In case of perfect intercooling,

$$T_1 = T_3 \quad \text{and} \quad \frac{P_2}{P_1} = \frac{P_4}{P_3}$$

T-S daigram shows that with absence of intercooling, the exiting temprature would be T_A while with intercooling this state ia at 4

$$T_A > T_4$$

The reduced temprature at compressor exit leads to lowering work of the compressor and thus increasing net work.

$$W_c = W_{L.P.C} + W_{H.P.C}$$
$$= Cp_a (T_2 - T_1) + Cp_a (T_4 - T_3)$$

$$W_t = Cp_g (T_5 - T_6)$$

$$W_{net} = W_t - W_c$$
$$= W_t - (W_{L.P.C} + W_{H.P.C})$$

$$Q_{add} = Cp_g (T_5 - T_4)$$

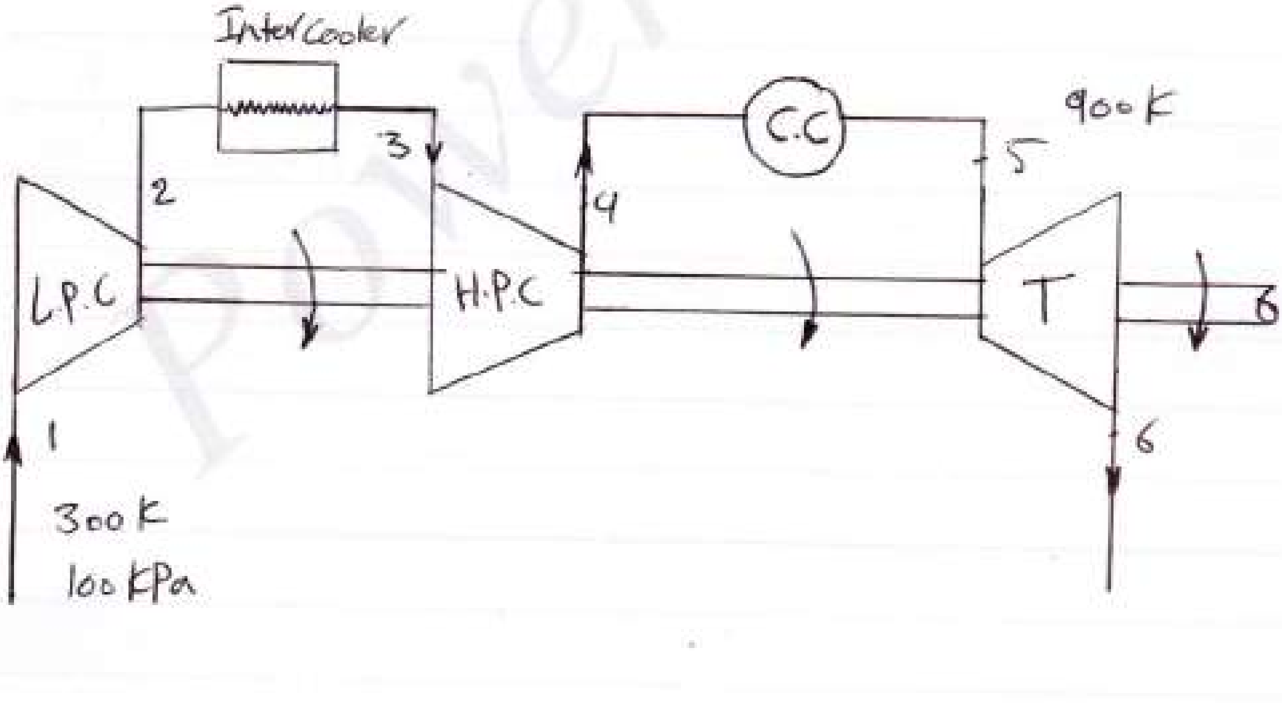
$$\eta = \frac{Cp_g (T_5 - T_6) - Cp_a [(T_2 - T_1) + (T_4 - T_3)]}{Cp_g (T_5 - T_4)}$$

Ex/ In a brayton cycle gas turbine power plant, the minimum and maximum tempretures are 300 K and 900 K respectively. Air is compressed from 100 KPa to 1000 KPa in a two-stage compressor with intercooling between. Consider that the inter-cooling tempreture is equal to the inlet temp. to the first compressor. Also, compression ratio between the two stages are equal.

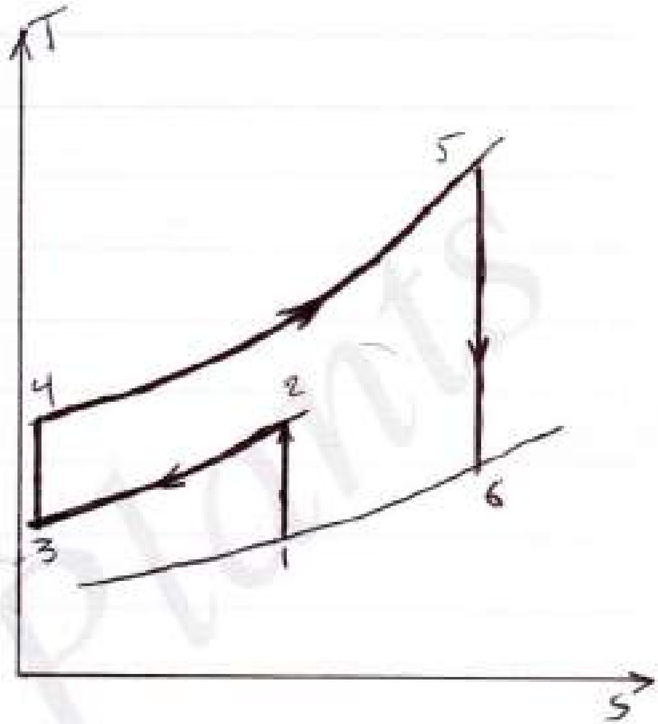
Take $Cp_g = 1.15$, $\gamma_g = 1.333$

$Cp_a = 1.005$, $\gamma_a = 1.4$

Calculate thermal efficiency of the plant?



Solution:—



It is given that

$$T_1 = T_3$$

$$\frac{P_2}{P_1} = \frac{P_4}{P_3}$$

$$\frac{P_4}{P_1} = \frac{1000}{100} = 10$$

$$\frac{P_4}{P_1} = \frac{P_4}{P_3} \cdot \frac{P_2}{P_1}$$

$$\therefore \frac{P_2}{P_1} = \sqrt{\frac{P_4}{P_1}} = \sqrt{10} = \frac{P_4}{P_3}$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\gamma-1/\gamma} \Rightarrow T_2 = 300 \times \left(\sqrt{10}\right)^{\frac{0.4}{1.4}} = 416.8\,K$$

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} = 300\left(\sqrt{10}\right)^{\frac{0.4}{1.4}} = 416.8\,K$$

$$W_C = W_{L.P.C} + W_{H.P.C}$$

$W_c = Cp_a(T_2 - T_1) + Cp_a(T_4 - T_3)$

$= 1.005(416.8 - 300) * 2$

$= 234.7$ KJ/Kg

$T_5 = 900$ (Max temp.)

$$\frac{T_5}{T_6} = \left(\frac{P_5}{P_6}\right)^{\frac{\gamma - 1}{\gamma}}$$

$$T_6 = \frac{900}{\left(\frac{1000}{100}\right)^{\frac{0.333}{1.333}}} = 506.3 \text{ K}$$

$W_t = Cp_g(T_5 - T_6)$

$= 1.15(900 - 506.3)$

$= 452.75$ KJ/Kg

$W_{net} = W_t - W_c$

$= 452.75 - 234.7$

$= 218$ KJ/Kg

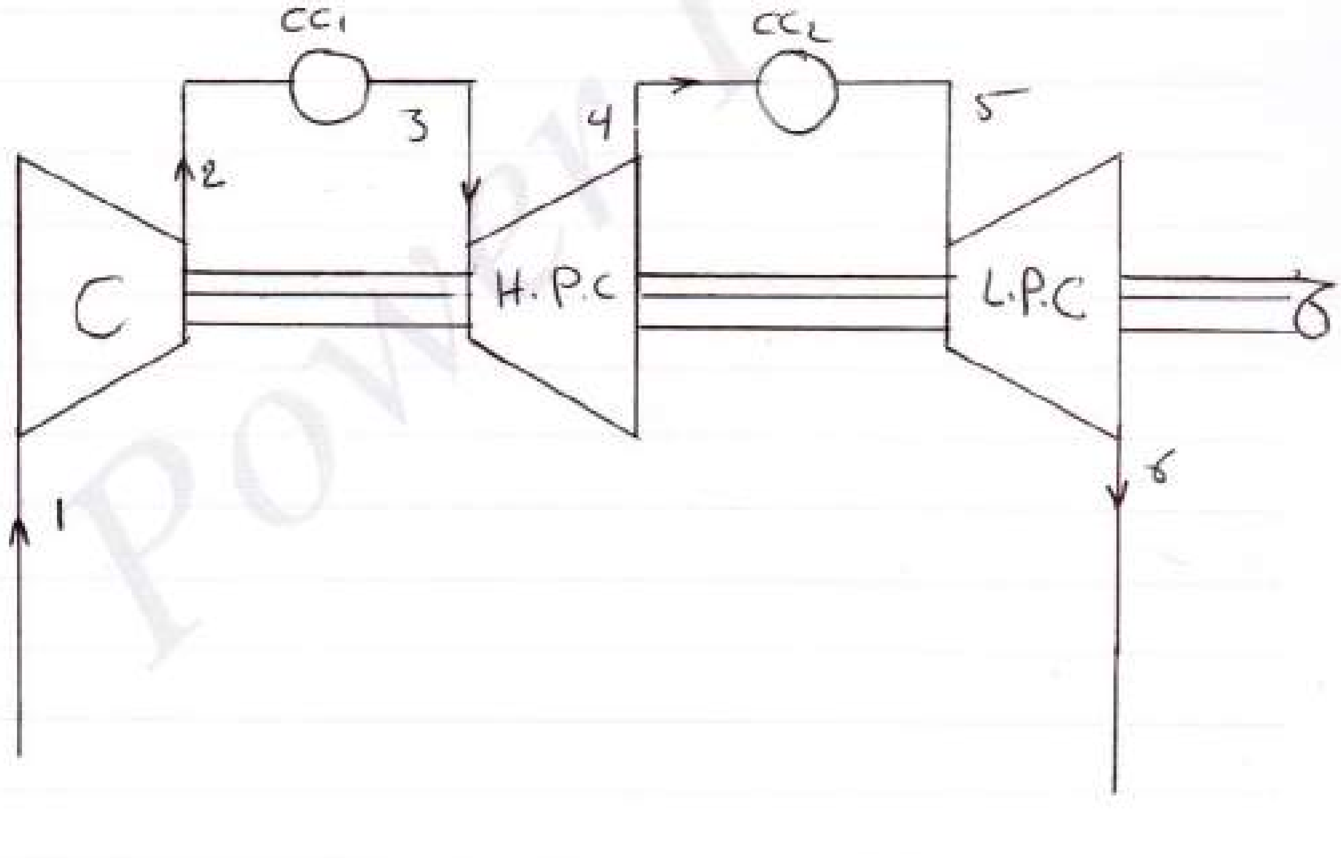
$Q_{add} = Cp_g(T_5 - T_4)$

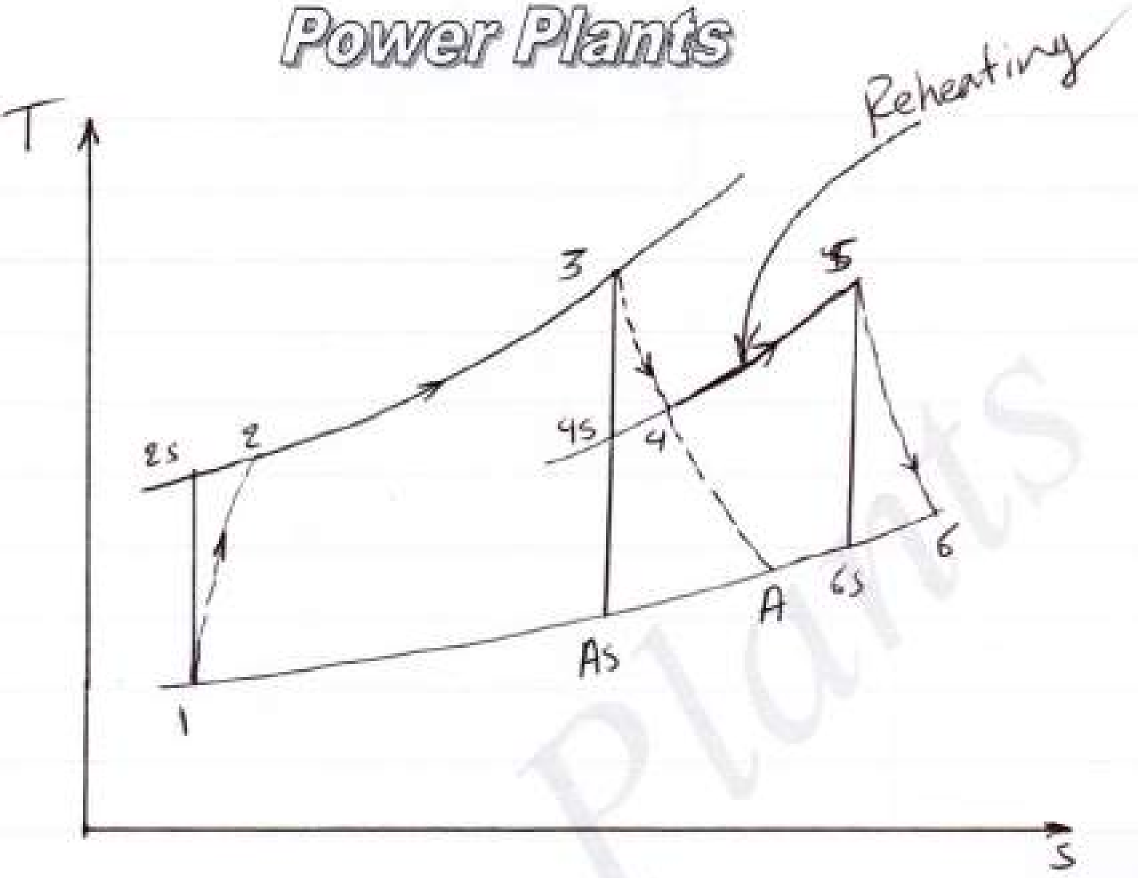
$= 1.15(900 - 416.8) = 555.7$ KJ/Kg

$$\eta_{cycle} = \frac{W_{net}}{Q_{add}} = \frac{218}{555.7} = 39.22\%$$

Reheating

In order to maximize the work available from the simple gas turbine cycle, enthalpy of fluid entering gas turbine is increased. Also, expand its expansion upto the lowest possible enthalpy value.





1-2 : Air is compressed across the compressor
2-3 : Heat is added through first Combustion chamber
3-4 : Air is expanded across H.P. turbine
4-5 : Air is reheated by adding heat through the second Comb. chamber.
5-6 : Air is expanded across L.P. turbine

The work of the two stages turbine is greater that of single expansion.

$$\therefore (T_3 - T_4) + (T_5 - T_6) > (T_3 - T_A)$$

$$W_c = Cp_a (T_2 - T_1)$$

$$W_{net} = W_{H.P.T} + W_{L.P.T} - W_c$$

$$W_{H.P.T} = Cp_g (T_3 - T_4)$$
$$W_{L.P.T} = Cp_g (T_5 - T_4)$$

For perfect reheating, $T_3 = T_5$

$$Q_{add} = Q_{add\,cc_1} + Q_{add\,cc_2}$$
$$= Cp_g (T_3 - T_2) + Cp_g (T_5 - T_4)$$

So, Efficiency for reheat cycle

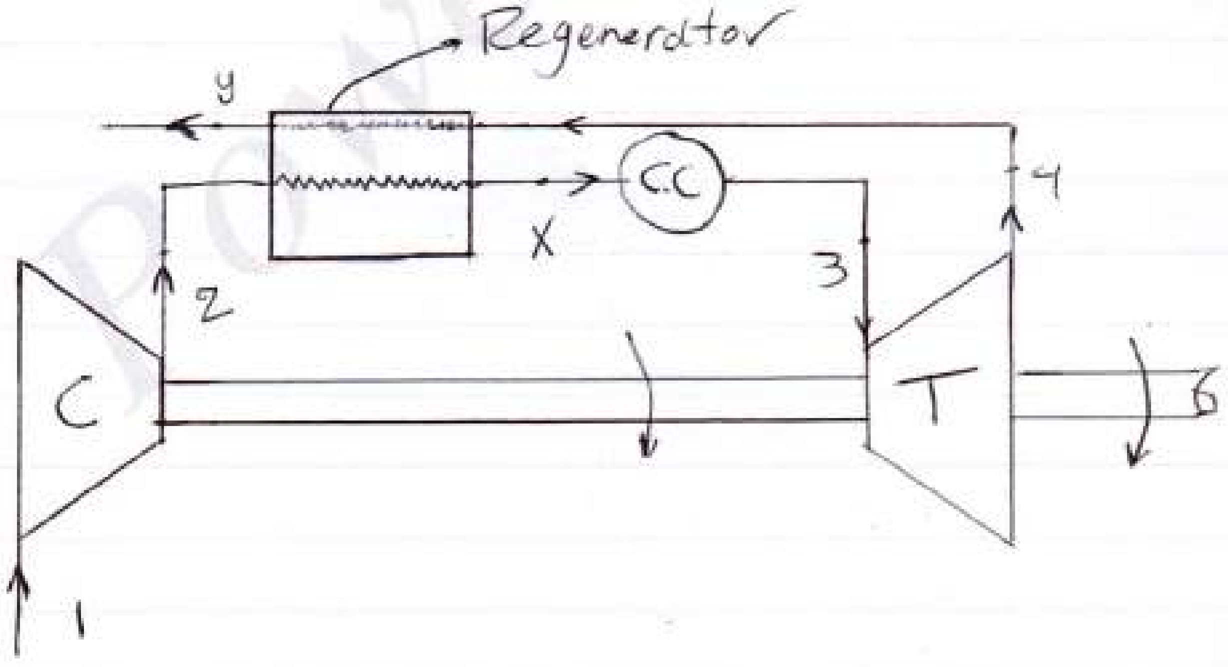
$$\eta_{reheat} = \frac{Cp_g \left[(T_3 - T_4) + (T_5 - T_6) \right] - Cp_a (T_2 - T_1)}{Cp_g \left[(T_3 - T_2) + (T_5 - T_4) \right]}$$

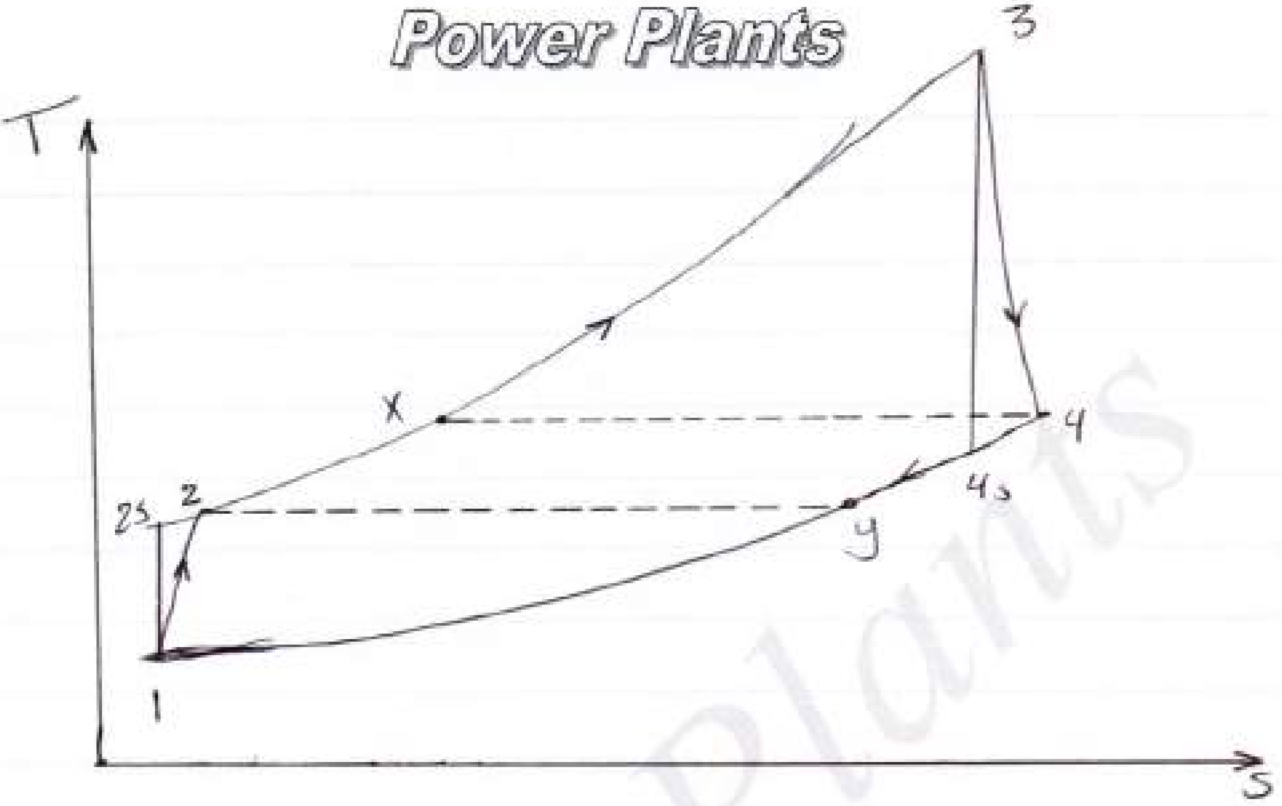
Mohanad Alheety

Regeneration

The turbine exhaust temprature of a gas turbine is normally above ambient temprature. Accordingly, the hot turbine exhaust has a potential for using. One way of utilizing this potential is by means of a heat exchanger called a regenerator

A regenerator allows the air exiting the compressor to be preheated before entering the combustor, thereby reducing the amount of fuel that must be burned in the combustor.





The regenerator shown is a counterflow heat exchanger. Ideally, no frictional pressure drop occures in either stream.

The turbine exhaust gas is cooled from state 4 to state y while the air exiting the compressor is heated from state 2 to state x.

If there is no losses in the heat exchanger, Then,

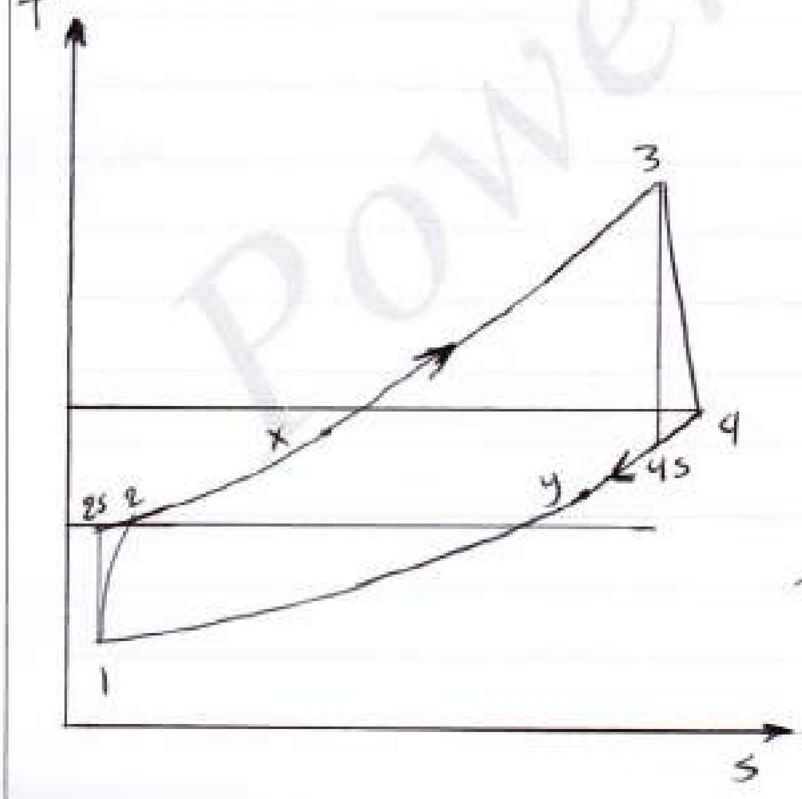
$$\dot{m}_a \, Cp_a \, (T_x - T_2) = \dot{m}_a \, Cp_g \, (T_4 - T_y)$$

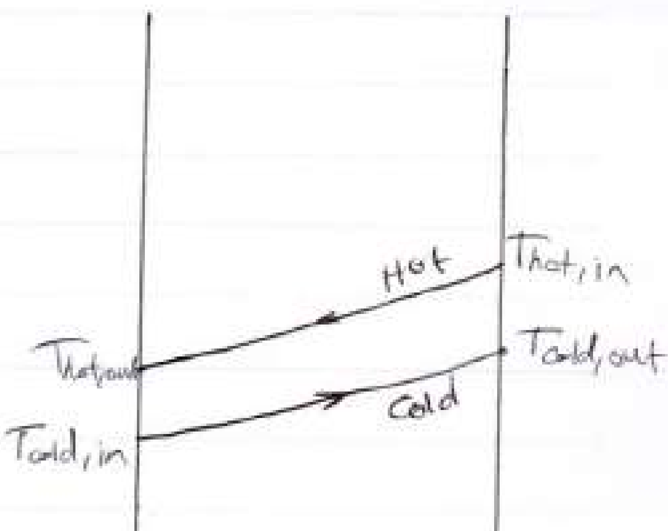
which means the heat given up by the gases must be equal to the heat recieved by the air.

$$\left.\begin{aligned} T_4 &= T_x \\ T_y &= T_2 \end{aligned}\right\} \text{ For ideal heat exchanger}$$

In practice, this is impossible

$T_4 > T_x$, $T_y > T_2$





Heat exchanger effectivness

It is defined as heat recieved by the air to the maximum possible heat which could be transferred from the gases in the heat exchanger

There fore,

$$\text{Effectivness } \epsilon = \frac{\dot{m}_a C_{Pa}(T_x - T_2)}{m_a C_{Pg}(T_4 - T_2)}$$

Thermal ratio

$$\text{Thermal ratio} = \frac{\text{temperature rise of the air}}{\text{Max. temperature difference available}}$$

$$= \frac{T_x - T_2}{T_4 - T_2}$$

$$Q_{add} \text{ (without HX)} = C_{Pg}(T_3 - T_2)$$

$$Q_{add} \text{ (with HX)} = C_{Pg}(T_3 - T_x)$$

It is obvious that

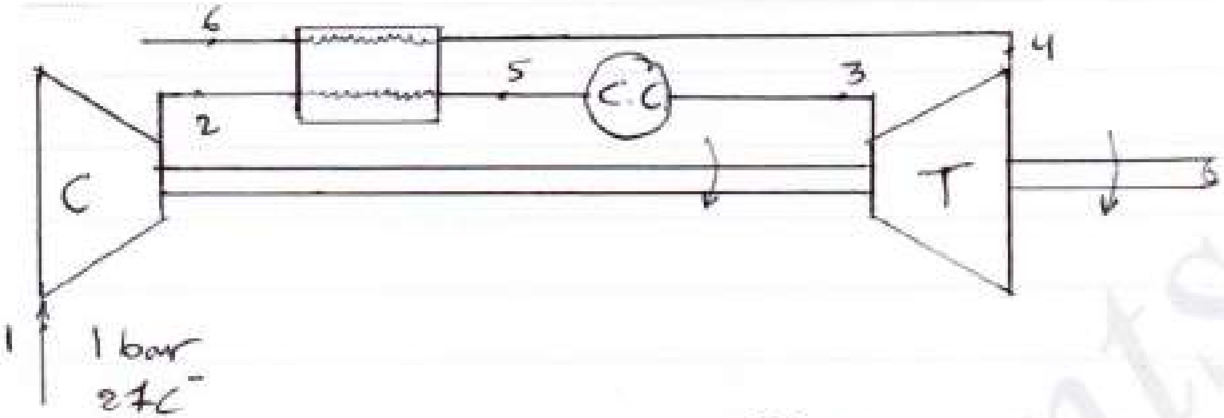
$$Q_{add\,(with\,HX)} < Q_{add\,(without\,HX)}$$

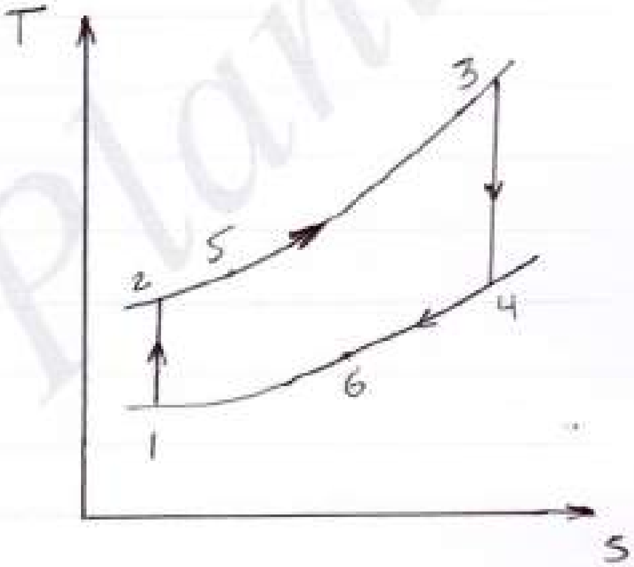
$$W_{net} = Cp_g(T_3 - T_4) - Cp_a(T_2 - T_1)$$

$$\eta_{regen.} = \frac{Cp_g(T_3 - T_4) - Cp_a(T_2 - T_1)}{Cp_g(T_3 - T_x)}$$

Ex/ In a gas turbine, air is supplied at 1 bar, 27°C into compressor having compression ratio of 8. The air leaving combustion chamber is heated upto 1100 k and expanded upto 1 bar. A heat exchanger having effectivness of 0.8 is fitted at exit of turbine for heating the air before its inlet into combustion chamber. Assume that,
$CP = 1.0032$ kJ/kg.k (for all processes)
$\gamma_a = 1.506$, $\gamma_g = 1.396$

Determine: specific work output of the plant
thermal efficiency of the plant.
work ratio





$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\gamma-1/\gamma}$$

$$T_2 = 300\,K\,(8)^{\frac{0.506}{1.506}}$$

$$T_2 = \cancel{1100\,K}\ 603.32\,K$$

$$T_3 = 1100\,K$$

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\gamma-1/\gamma} \Rightarrow T_4 = \frac{1100}{(8)^{\frac{0.346}{1.346}}}$$

$$T_4 = 644.53\,K$$

$$\epsilon = \frac{Cp_a(T_5 - T_2)}{Cp_g(T_4 - T_2)}$$

$$0.8 = \frac{T_5 - 603.32}{644.53 - 603.32}$$

$$\Rightarrow T_5 = 636.28\,K$$

$$W_c = C_{pa}(T_2 - T_1)$$
$$= 1.0032\,(603.32 - 300) = 304.3 \text{ KJ/Kg}$$

$$W_t = C_{pg}\,(T_3 - T_4)$$
$$= 1.0032\,(1100 - 644.53) = 456.93 \text{ KJ/Kg}$$

$W_{net} = W_t - W_c = 152.6$ KJ/Kg (specific work output)

$$\eta_{cycle} = \frac{W_{net}}{Q_{add}}$$

$$Q_{add} = C_{pg}(T_3 - T_5)$$
$$= 1.0032\,(1100 - 636.28) = 465.2 \text{ KJ/Kg}$$

$$\eta = \frac{152.6}{465.2} = 32.81\,\%$$

$$\text{Work ratio} = \frac{W_t - W_c}{W_t} = \frac{W_{net}}{W_t}$$

$$= \frac{152.6}{456.93} = 0.334$$

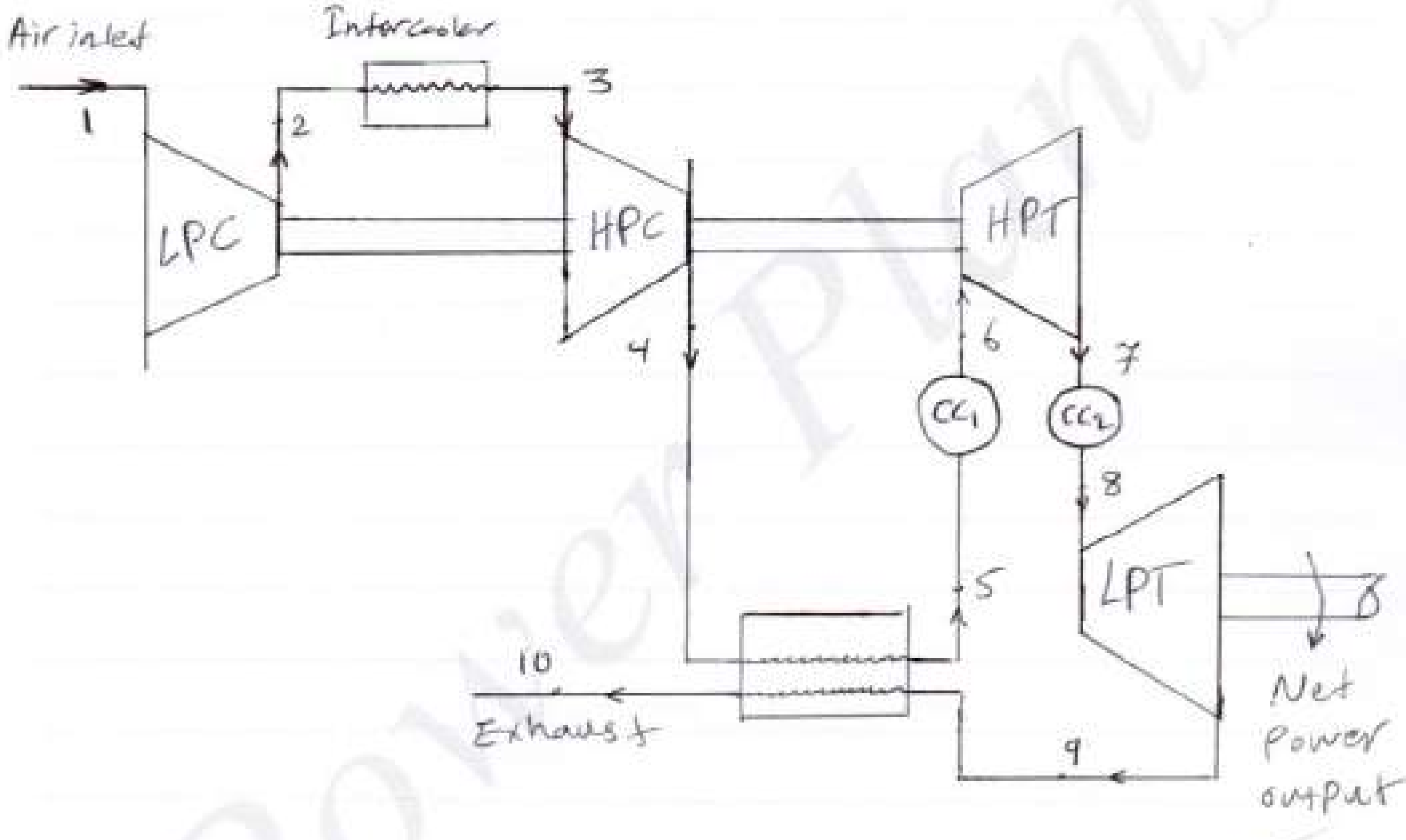
Ex/ A 5000 KW gas turbine generating set operates with two compressor stages with intercooling between stages; the overall pressure ratio is 9/1. A HP turbine is used to drive the compressors. and a LP turbine drives the generator. The temperature of gases at entry to the HP turbine is 650°C and the gases leaving the LP turbine are passed through a heat exchanger to heat the air leaving the HP stage compressor. Gases are reheated to 650°C after expansion in H.P.T The compressors have equal pressure ratios and intercooling is complete between stages. The air inlet temperature to the unit is 15°C.

Assume that: η_c for both compressors is 0.8
η_t = = turbines is 0.85
The heat exchanger thermal ratio is 0.75.

A mechanical efficiency of 98% can be assumed for both the power shaft and the compressor turbine shaft. Determine :-
(i) the cycle efficiency (ii) the work ratio
(iii) the mass flow rate.

For air take $CP = 1.005$ KJ/Kg , $\gamma = 1.4$
For gas take $CP = 1.15$ KJ/Kg , $\gamma = 1.333$

Solution :-



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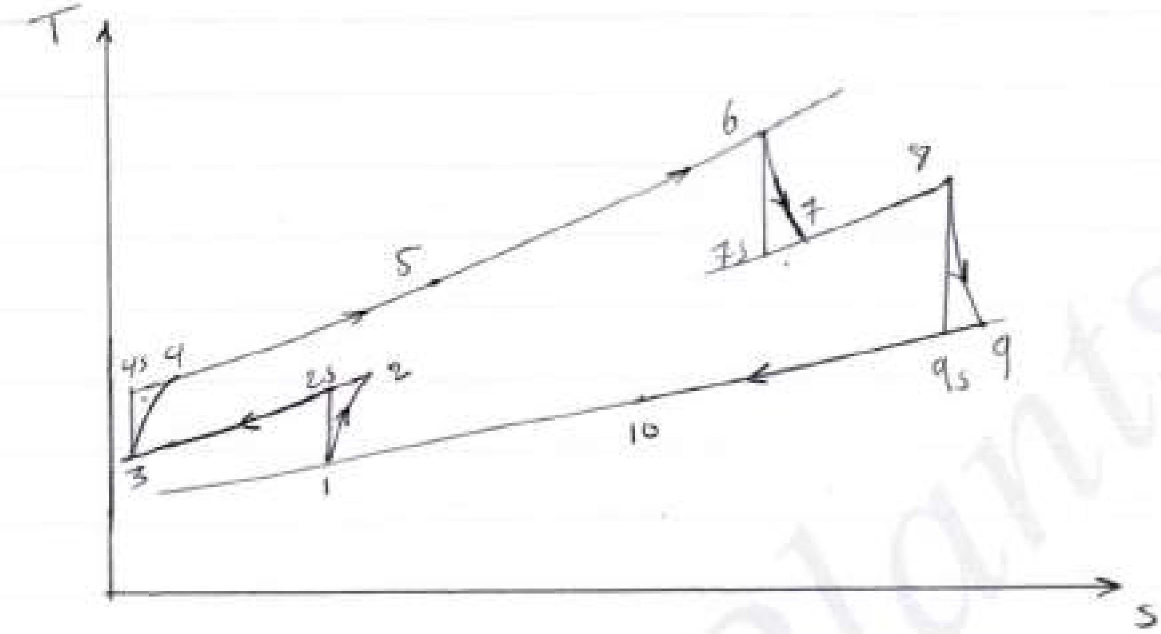

Since the pressure ratio and isentropic efficiency of each compressor is the same, then the work input required for each compressor is the same.

$$T_1 = T_3 \quad , \quad T_2 = T_4$$

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\gamma-1/\gamma}$$

$$\frac{P_2}{P_1} = \sqrt{9} = 3$$

$$T_{2s} = 288 + (3)^{\frac{0.4}{1.4}} = 394\,K$$

$$\eta_{c_{LPC}} = \frac{T_{2s} - T_1}{T_2 - T_1}$$

$$0.8 = \frac{T_{2s} - 288}{T_2 - 288} \longrightarrow T_2 = 420.5 \text{ K}$$

$$W_{LPC} = C_{Pa} (T_2 - T_1)$$
$$= 1.005 (420.5 - 288)$$
$$= 133.1 \text{ KJ/Kg}$$ [For each compressor]

$$T_{4s} = T_{2s} = 394$$
$$T_4 = T_2 = 420.5$$

The HP turbine is required to drive both compressors and to overcome mechanical friction

$$W_c = W_{c_1} + W_{c_2}$$
$$= 133.1 + 133.1$$
$$= 266.2 \text{ KJ/Kg}$$

$$W_{HPT} = \frac{W_c}{\eta_m} = \frac{266.2}{0.98} = 272 \text{ KJ/Kg}$$

Therefore,

$$272 = Cp_g\,(T6 - T7)$$
$$272 = 1.15\,(923 - T7)$$
$$T7 = 686.5\,K$$

$$\eta_{H.P.T} = \frac{T6 - T7}{T6 - T7s}$$

$$0.85 = \frac{923 - 686.5}{923 - T7s} \Rightarrow T7s = 645\,K$$

$$\frac{T6}{T7s} = \left(\frac{P6}{P7}\right)^{\gamma-1/\gamma} \quad \text{or} \quad \frac{P6}{P7} = \left(\frac{T6}{T7s}\right)^{\frac{\gamma}{\gamma-1}}$$

$$\frac{P6}{P7} = \left(\frac{923}{645}\right)^{\frac{1.333}{0.333}} = 4.19$$

$$\frac{P6}{P9} = \frac{P6}{P7} \cdot \frac{P8}{P9} \quad , \quad P7 = P8$$

$$\frac{P8}{P9} = \frac{\cancel{4.19}\ 9}{4.19} = 2.147$$

$$\frac{T_8}{T_{9s}} = \left(\frac{P_8}{P_9}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_{9s} = \frac{923}{(2.147)^{\frac{0.333}{1.333}}} = 762.6\ K$$

$$\eta_{LPT} = \frac{T_8 - T_9}{T_8 - T_{9s}}$$

$$0.85 = \frac{923 - T_9}{923 - 762.9} \Rightarrow T_9 = 786.7\ K$$

$$\text{Net work output} = Cp_g (T_8 - T_9) * 0.98$$
$$= 1.15 (923 - 786.7) * 0.98$$
$$= 153.7\ KJ/Kg$$

$$\text{Thermal ratio} = \frac{T_5 - T_4}{T_9 - T_4}$$

$$0.75 = \frac{T_5 - 420.5}{786.7 - 420.5}$$

$$T_5 = 695.2\ K$$

Heat supplied $= Cp_g (T_6 - T_5) + Cp_g (T_8 - T_7)$

$= 1.15 [(923 - 695.2) + (923 - 686.5)]$

$= 534 \ kJ/kg$

$$\eta_{cycle} = \frac{W_{net}}{Q_{add}} = \frac{153.7}{534} = 28.8\%$$

$$\text{Work ratio} = \frac{\text{Work net}}{\text{gross work}}$$

gross work $= W_{H.P.T} + W_{L.P.T}$

$$= 272 + \frac{153.7}{0.98} = 429 \ kJ/kg$$

$$\text{Work ratio} = \frac{153.7}{429} = 0.358$$

$P_{output} = \dot{m} * W_{net}$

$$\dot{m} = \frac{5000}{153.7} = 32.6 \ kg/s$$

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Power Plants

Sheet 2

Q1/ A gas turbine has an overall pressure ratio of 5 and maximum cycle temperature of 550 C. The turbine drives the compressor and an electric generator, the mechanical efficiency of the drive being 97%. The ambient temperature is 20°C and air enters the compressor at rate of 15 Kg/s : $\eta_c = 80\%$

$\eta_t = 83\%$

$Cp_a = 1.005$, $\gamma = 1.4$
$Cp_g = 1.15$, $\gamma = 1.333$

Determine : Power output, Cycle efficiency work ratio

(660.3W, 12.1%, 0.169)

Q2/ In a marine gas turbine unit, a HP stage turbine drives the compressor, and a LP stage turbine drives the propeller through suitable gearing. The overall pressure ratio is 4/1, the mass flow rate is 60 Kg/s

Maximum temperature is 650C, and the air intake conditions are 1.01 bar and 25°C.
$\eta_C = 0.8$, $\eta_{H.P.T} = 0.83$, $\eta_{LP.T} = 0.85$.
The Mechanical ~~losses~~ efficiency of both shafts is 98%. Calculate:
(i) Pressure between turbine stages
(ii) Cycle efficiency
(iii) Shaft power

(1.57 bar, 14.9%, 4560W)

Q3/ For the same unit in Q2, Calculate cycle efficiency when a heat exchanger is fitted. Assume a thermal ratio of 0.75

(23.4%)

Q4/ In a gas turbine generating set, two stages of compression are used with an intercooling between stages. The HP turbine drives the HP compressor, and LP turbine drive the LP compressor and the generator. The exhaust from the LP turbine passes through a heat exchanger which transfers heat to the air leaving the HP compressor. There is a reheat combustion chamber between turbine stages which rises the gas temperature to 600C, which is also the gas temperature at entry to the HP turbine. the overall pressure ratio is 10/1, each compressor having the same pressure ratio, and the air temperature at entry to the unit is 20C. The heat exchanger thermal ratio may be taken as 0.7 and intercooling is complete between compressor stages.

Assume isentropic efficiencies of 0.8 for both compressor stages, and 0.85 for turbine stages and 2% of the work of each turbine is used in overcoming friction. Calculate:

(i) power output for a mass flow rate of 115 kg/s

(ii) overall efficiency of the plant.

(14460 kW, 25.7%)