USN 1GR16ME071

Fourth Semester B.E. Degree Examination, June/July 2018
Applied Thermodynamics

Time: 3 hrs. Max. Marks: 80

Note: 1. Answer any FIVE full questions, choosing one full question from each module.
2. Use of thermodynamics data handbook is permitted.

Module-1

1 a. Derive an expression for mean effective pressure in an air standard Otto cycle. **(08 Marks)**

b. Compression ratio of an air standard dual cycle is 8. Air is at 100 kPa, 300 K at the beginning of the compression process. The temperature of air at the end of constant pressure heat addition process is 1300 K. The net heat transfer to the cycle is 480 kJ/kg. Determine:
i) Heat added during constant volume per kg of air
ii) Air standard cycle efficiency and
iii) m.e.p. **(08 Marks)**

OR

2 a. For a simple gas turbine cycle, the optimum pressure ratio for maximum work output of cycle is given by

$$r_p = \left\{\eta_C \eta_T \frac{T_3}{T_1}\right\}^{\frac{\gamma}{2(\gamma-1)}}$$

where η_C and η_T are the isentropic efficiency of compressor and turbine respectively. T_3 and T_1 = maximum and minimum temperature of the cycle respectively. $\gamma = C_p/C_v$ **(08 Marks)**

b. Determine the network output and thermal efficiency of an ideal gas turbine cycle having two stages of compression with perfect intercooling, two stages of expansion with perfect reheating between the stages and an ideal regenerator. The overall pressure ratio of the cycle is 4 and the maximum temperature of the cycle is 900°C. Assume that the atmospheric temperature is 15°C and the cycle is designed for maximum work output. Draw the schematic and T-S diagrams for the cycle. **(08 Marks)**

Module-2

3 a. Why is Carnot cycle not practicable for steam power plant? Explain briefly with the help of T-S diagram. **(06 Marks)**

b. Discuss the effect of (i) Boiler pressure and (ii) Superheat on the performance of a Rankine cycle. **(06 Marks)**

c. A steam power plant operates on a theoretical reheat cycle. Steam at boiler with 150 bar, 550°C expands through the high pressure turbine. It is reheated at a constant pressure of 40 bar to 550°C and expands through the low pressure turbine to a condenser at 0.1 bar. Draw h-s diagram and find:
i) Quality of steam at turbine exit
ii) Cycle efficiency
iii) Steam rate in kg/KW.h **(04 Marks)**

OR

4 a. With the help of flow and h-s diagram, derive an expression for cycle efficiency and also for mass of steam bled in a practical regenerative steam cycle with one open feed water heater. (08 Marks)

b. Steam at 30 bar, 350°C is supplied to a steam turbine in a practical regenerative cycle and the steam is bled at 4 bar. The bled steam comes out as dry saturated steam and heats the feed water in an direct contact type feed water heater to its saturated liquid state. The rest of the steam in the turbine expands to condenser pressure of 0.1 bar. Assuming the turbine efficiency to be same before and after bleeding, determine:
 i) The turbine efficiency
 ii) Steam quality at the condenser inlet
 iii) Mass of steam bled per kg of boiler steam
 iv) Cycle efficiency. (08 Marks)

Module-3

5 a. With neat sketch, explain the Orsat's apparatus used for exhaust gas analysis. (06 Marks)

b. The products of combustion of an unknown hydrocarbon C_xH_y have the following composition as measured by an Orsat apparatus: CO_2 = 8.0%, CO = 0.9%, O_2 = 8.8% and rest is N_2. Determine:
 i) Composition of the fuel
 ii) The air-fuel ratio
 iii) Percentage of excess air
 iv) Dew point temperature of the products if the total pressure is 1.0 bar. (10 Marks)

OR

6 a. Explain the principle of conducting Morse test on IC engines for determining frictional power. (04 Marks)

b. List the factors affecting the detonation. (02 Marks)

c. A 4-cylinder 2-stroke petrol engine has a bore of 57 mm and stroke of 90 mm. Its rated speed is 2800 rpm and is tested at this speed against a brake, which has a torque arm of 0.356 m. The net brake load is 155 N and the fuel consumption is 6.74 lit/h. The specific gravity of the petrol is 0.735 and it has a calorific value of 44200 kJ/kg. A Morse test is carried out and the cylinders are cut-out in order 1, 2, 3, 4 with corresponding brake loads 111, 106.5, 104.2 and 111.3 N respectively. Calculate for this speed
 i) The engine torque
 ii) Brake mean effective pressure
 iii) Brake thermal efficiency
 iv) BSFC
 v) Mechanical efficiency
 vi) Indicated thermal efficiency. (10 Marks)

Module-4

7 a. A vapour compression plant uses R-12 and is to develop 5 tonnes of refrigeration. The condenser and evaporator temperatures are to be 40°C and −10°C respectively. Determine:
 i) The refrigerant flow rate in kg/s
 ii) Heat rejected in the condenser in KW
 iii) COP
 iv) Power required to drive the compressor (06 Marks)

b. An air refrigeration system working on Reversed Brayton Cycle with 15 tonnes capacity has its pressure range 1 bar to 10 bar. Air enters the compressor at −5°C and enters the expander at 25°C. Assuming the isentropic efficiency of expander and compressor each has 85%, find: i) COP ii) Air flow rate and iii) Power required. (06 Marks)

c. What are the desirable properties of good refrigerant? (04 Marks)

OR

8 a. With a neat sketch explain the working of air conditioning system for hot and dry summer condition. Show the processes on psychrometric chart (08 Marks)

b. It is required to design an air conditioning plant for a office room with the following conditions:

Outdoor conditions: 14°C DBT and 10°C WBT

Required conditions: 20°C DBT and 60% RH

Amount of air circulation = 0.3 m^3/min/person

Seating capacity of office = 60

The required condition is achieved first by heating and then by adiabatic humidifying. Determine:

i) Heating capacity of the coil in KW and surface temperature required if the by-pass factor of the coil is 0.4.

ii) The capacity of the humidifier. (08 Marks)

Module-5

9 a. Derive the condition for minimum work required by a two stage air compressor with perfect intercooling between stages. Assume the compression follows the law $PV^n = C$ for stage-1 and for the stage-2 follows $PV^m = C$. Reduce this equation when n = m. (08 Marks)

b. A single stage, double acting air compressor, required to deliver 14 m^3 of air per minute measured at 1.013 bar and 15°C. The delivery pressure is 7 bar and speed is 300 rpm. Take the clearance volume as 5% of swept volume with the compression and expansion index, n = 1.3. Calculate:

i) the bore and stroke of the cylinder assuming L = 1.2 D

ii) Delivery temperature

iii) Indicated power required. (08 Marks)

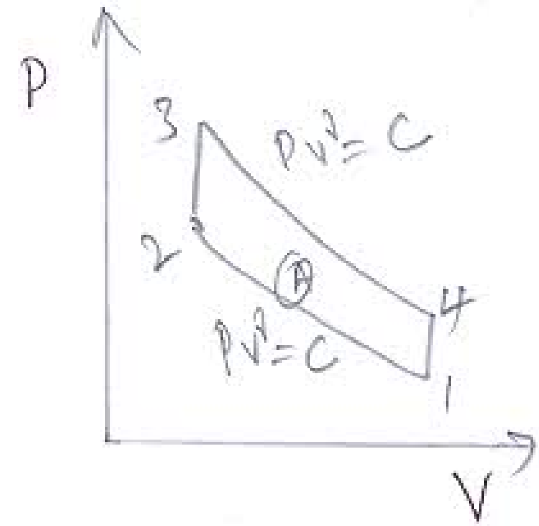
OR

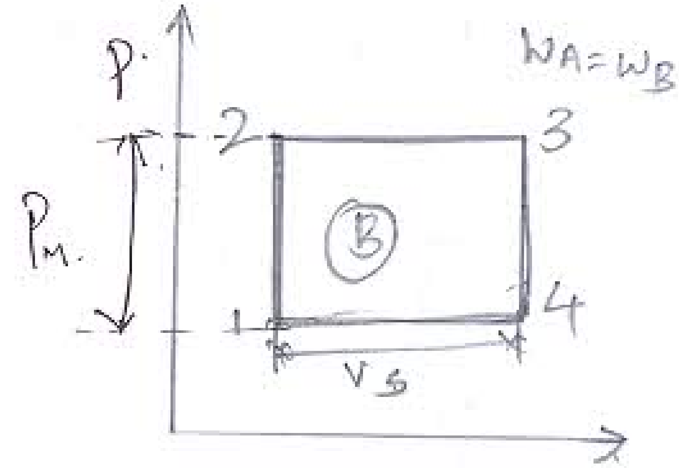
10 a. Prove that maximum flow rate of steam per unit area through a nozzle occurs when the ratio of pressure at throat to the inlet pressure is equal to $\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$ where n = isentropic index of expansion. (08 Marks)

b. An adiabatic steam nozzle is to be designed for a discharge rate of 10 kg/s of steam from 10 bar and 400°C to a back pressure of 1 bar. The nozzle efficiency is 0.92 and the frictional loss is assumed to take place in the diverging portion of the nozzle only. Calculate i) Velocity of steam at throat and exit of the nozzle. ii) Throat and exit area. Assume index of expansion = 1.3. (08 Marks)

* * * * *

1 a





Mean effective Pressure is calculated by

$$P_M = \frac{\text{work done per cycle}}{V_s}$$

$$= \frac{(P_3V_3 - P_4V_4)}{\gamma - 1} - \frac{(P_2V_2 - P_1V_1)}{\gamma - 1}$$

$$= \frac{1}{\gamma-1}\left[P_4V_4\left[\frac{P_3V_3}{P_4V_4} - 1\right] - P_1V_1\left[\frac{P_2V_2}{P_1V_1} - 1\right]\right]$$

$$\therefore \frac{P_3}{P_2} = \frac{P_4}{P_1} = \alpha.$$

$$= \frac{1}{\gamma-1}\left[P_4V_4\left[\frac{P_3}{P_4}\cdot\frac{1}{r_c} - 1\right] - P_1V_1\left[\frac{P_2}{P_1}\cdot\frac{1}{r_c} - 1\right]\right]$$

$$= \frac{VV}{\gamma-1}\left[P_4 (r_c^{\gamma-1} - 1) - P_1 (r_c^{\gamma-1} - 1)\right]$$

$$= \frac{P_1 V_1}{\gamma - 1}\left[\left(r_c^{\gamma-1} - 1\right)\left(P_4 - P_1\right)\right]$$

$$= \frac{P_1 V_1}{\gamma - 1}\left[\left(r_c^{\gamma-1} - 1\right)\left(\alpha - 1\right)\right]$$

$$P_M = \frac{\frac{P_1 V_1}{\gamma - 1}\left(r_c^{\gamma-1} - 1\right)\left(\alpha - 1\right)}{V_s}$$

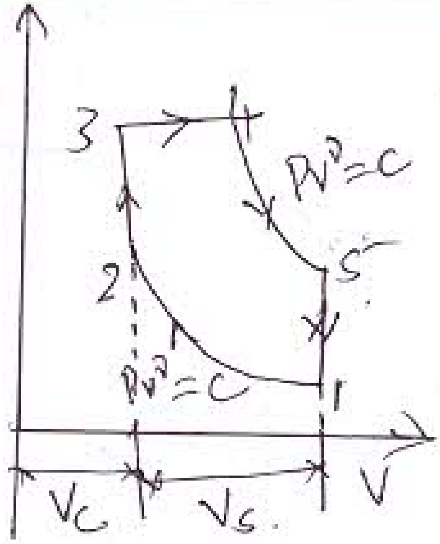
$$= \frac{\frac{P_1 V_1}{\gamma - 1}\left(r_c^{\gamma-1} - 1\right)\left(\alpha - 1\right)}{V_1 - V_2}$$

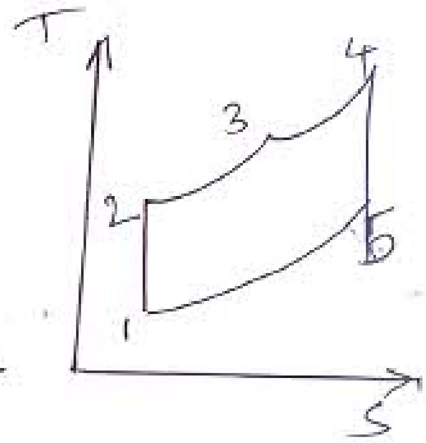
$$= \frac{\frac{P_1 V_1}{\gamma - 1}\left(r_c^{\gamma-1} - 1\right)\left(\alpha - 1\right)}{V_1 \cdot \left(1 - \frac{V_2}{V_1}\right)} = \frac{P_1 \left(r_c^{\gamma-1} - 1\right)\left(\alpha - 1\right)}{\left(\gamma - 1\right)\left(1 - \frac{1}{r_c}\right)}$$

$$P_M = \frac{P_1 r_c \left(r_c^{\gamma-1} - 1\right)\left(\alpha - 1\right)}{\left(\gamma - 1\right) \cdot \left(r_c - 1\right)}$$

1 Given data.

b) $r_c = 8$





$P_1 = 100\ kPa$

$T_1 = 300K$

$T_4 = 1300K$

$Q_{net} = 480\ kJ/kg$

$Q_{net} = Q_s - Q_R$

Sol:

$$\frac{T_2}{T_1} = r_c^{\gamma-1} = \frac{T_2}{300} = 8^{1.4-1}$$

$$= T_2 = 689.2K.$$

$$Q_{net} = Mc_v(T_3 - T_2) + Mc_p(T_4 - T_3)$$

$$= 1 \times 0.71(T_3 - 689.2) + 1 \times 1.005(1300 - T_3)$$

$$0.71T_3 - 489.33 + 1306.5 - 1.005T_3$$

$$480 = +817.17 - 0.295T_3$$

$$T_3 = 1142.949.K$$

$$\beta = \frac{T_3}{T_2} = \frac{1142.94}{689.2} = \underline{\underline{1.658}}$$

$\therefore T_4 = \rho T_3$

$1300 = \rho \times 1142.94$

$$\underline{\underline{\rho = 1.1374}}$$

* Heat added during constant volume per kg of air:

$$Q_{s\,cons\,volu} = mc_v(T_3 - T_2)$$

$$= 1 \times 0.71(1142.94 - 689.2)$$

$$= \underline{\underline{322.15 \frac{kJ}{kg}}}$$

$$\eta = 1 - \left[\frac{\rho^{\gamma}(\beta - 1)}{r^{\gamma-1}(\beta - 1) + \beta\gamma(\rho - 1)}\right]$$

$$\eta = 1 - \left[\frac{(1.1374)^{1.4}(1.658 - 1)}{8^{1.4-1}\left((1.658-1) + 1.658 \times 1.4 (1.137-1)\right)} \right]$$

$$= 1 - \left[\frac{0.787}{1.511 + 0.318} \right]$$

$$\eta = 56.97\ \%$$

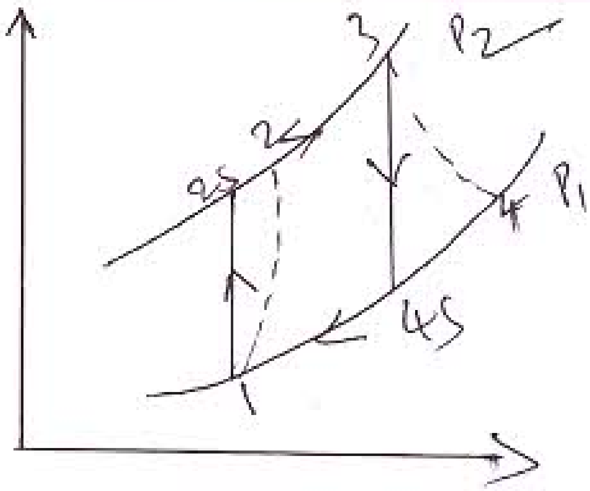
$$P_M = \frac{P_1 r^{\gamma}}{(\gamma-1)(r-1)} \left(\beta \gamma (\rho-1) + (\beta-1) - r^{1-\gamma} (\beta \rho^{\gamma} - 1) \right)$$

$$= \frac{100 \times 8^{1.4}}{(1.4-1)(8-1)} \left[1.658 \times 1.4 \,(1.137-1) + (1.658-1) - 8^{1-1.4} (1.65 \cdot 1.137^{1.4} - 1) \right]$$

$$656.39 \left[0.3180 + 0.658 - 0.4243 \right]$$

$$= 362.13 \text{ KPA}$$

Q
a.



$T_1 = T_{MIN}$. $T_3 = T_{Max}$.

we know that

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

let $r_p^{\frac{\gamma-1}{\gamma}} = x$.

if isentropic efficiency of compressor is η_c

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

$$= T_1 \left[1 + \frac{x-1}{\eta_c}\right]$$

$$T_{4s} = T_3 \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} = T_3 \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_3}{x}$$

if isentropic efficency of turbine is η_T

Then $\eta_T = \frac{T_3 - T_4}{T_3 - T_{4s}} = -T_3 + T_4 = \eta_T (T_{4s} - T_3)$

$T_4 = T_3 + \eta_T \left(\frac{T_3}{x} - T_3 \right)$

$= T_3 \left[1 + \eta_T \left(\frac{1}{x} - 1 \right) \right]$

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

$= c_p \left[\eta_T \left(T_3 - \frac{T_3}{x} \right) - \frac{x T_1 - T_2}{\eta_c} \right]$ kJ/kg

$= c_p \left[\eta_T T_{max} \left(1 - \frac{1}{r_p^{\frac{\gamma-1}{\gamma}}} \right) - \frac{T_{min}}{\eta_c} \left(r_p^{\frac{\gamma-1}{\gamma}} \right) \right]$

For maximum sp. work $\frac{\partial w}{\partial x} = 0$

$\frac{\partial w}{\partial x} = c_p \left(\frac{\eta_T T_3}{x^2} - \frac{T_1}{\eta_c} \right) = 0$

$x^2 = \eta_T \eta_c \cdot \frac{T_3}{T_1} = \sqrt{\eta_T \eta_c \, T_3 / T_1}$

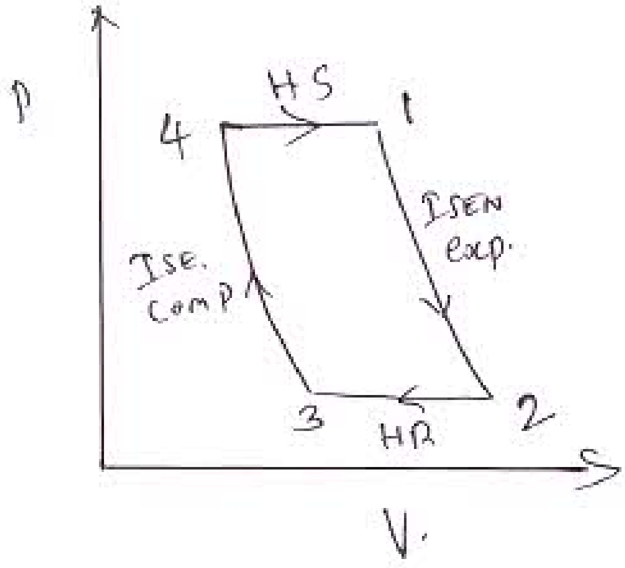
2a.

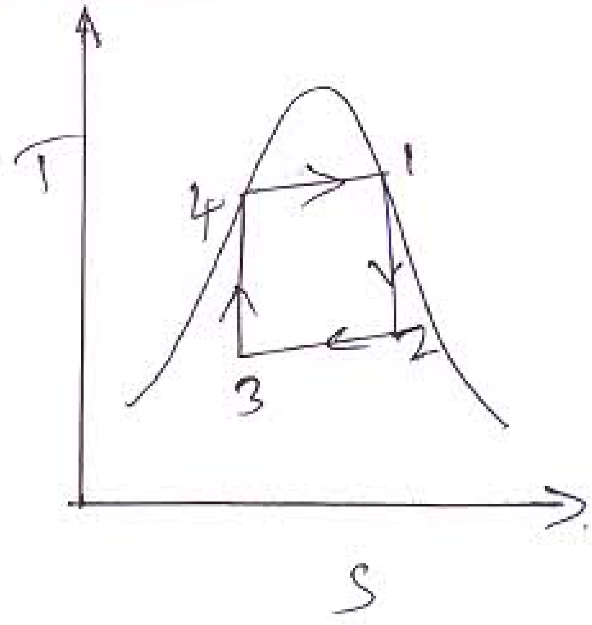
$$r_p = \left[\eta_T \eta_C \frac{T_3}{T_1} \right]^{\frac{\gamma}{2(\gamma - 1)}}$$

Proved

2.
b.

3 a.





Carnot cycle is ~~the~~ not practically possible because

1> Iso thermal heat addition in boiler takes place only in wet region.

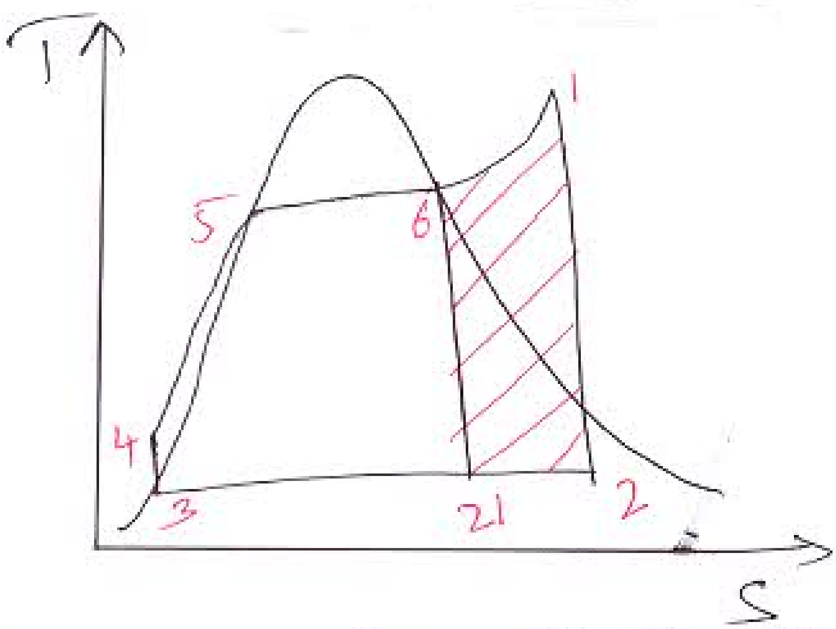
2> ~~it is diff~~ during isothermal condensation process the condensation is stopped at 3 which should be vertically below state 4. due to this, ~~so~~ the steam will not go complete condensation.

3) The compression has to deal with a non-homogenous mixture.

4. The higher Power requirement for compression reduce the plant efficiency.

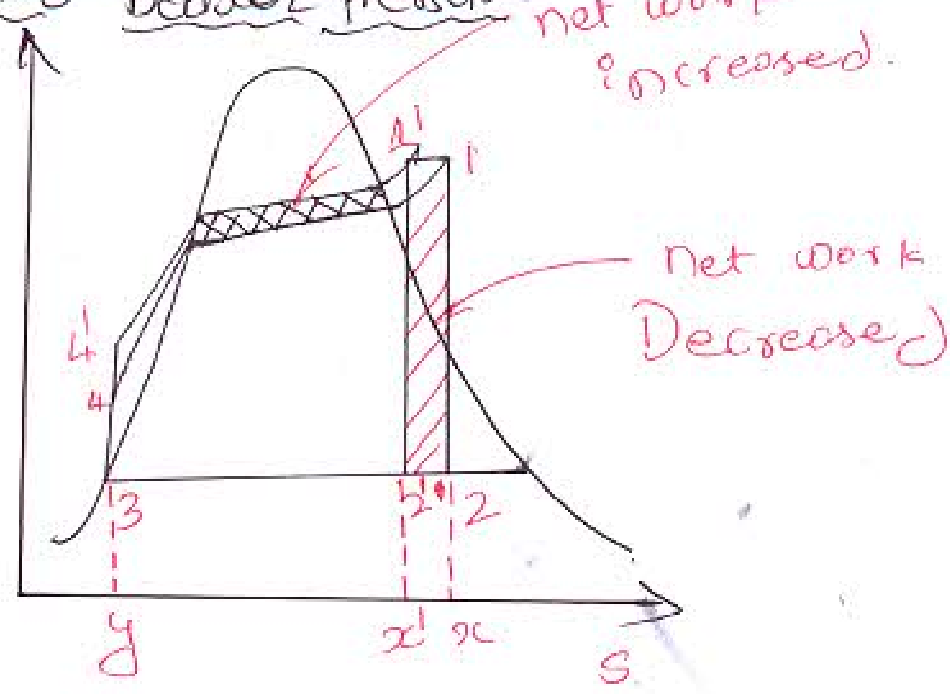
much Change in the net work, but since heat rejected decreases. This results in an increase in Rankine efficiency with an increase in maximum pressure.

iii) **effect of super heat:-**



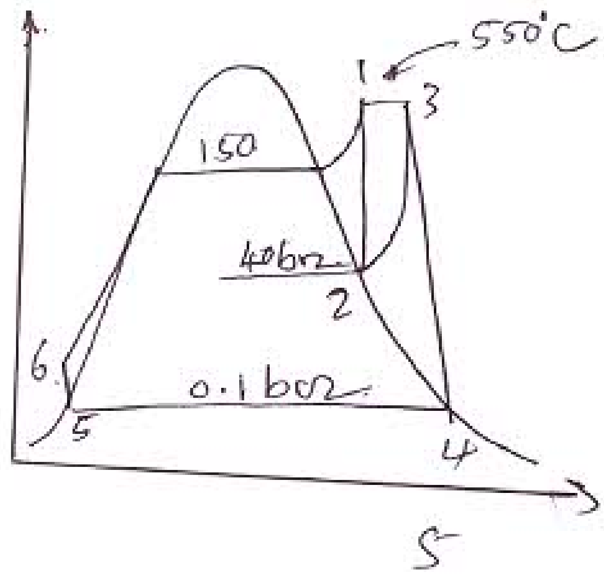
The above fig shows the effect of super heat on the Rankine cycle. it is evident that. The work is increased by area 6-1-2-2'-6 The heat transfered in the boiler is increased by area 6-1-b-b'-6. the ratio of these two areas is greater then the ratio of net work to heat supplied for the rest of the cycle.

3 b) i) effect of boiler pressure



The effect of increasing the boiler Pressure on the Rankine cycle efficiency is illustrated on a T-S diagram. in this case the maximum temperature of the steam as well as exhaust Pressure are kept constant. increase in boiler Pressure results in decrease in heat rejected by given area 2-x-x'-2'-2. Further the net work increases by the amount shown by area of double cross hatching, however it decreases by the amount shown by area of single cross hatching. Thus there will hence for a given pressure, superheating the steam increases the Rankine cycle efficien, further when the steam is superheated, the quality of the steam leaving turbine.

3.
c.



$h_1 = 346.5$ kJ/kg $\quad h_3 = 3565$ kJ/kg

$h_2 = 3065$ kJ/kg $\quad h_4 = 2300$ kJ/kg

$x_{4s} = 0.88$. $\quad h_5$ (steam tabel) $= 191.83$ kJ/kg.

$W_p = V\Delta P = 10^{-3} \times 150 \times 10^2 = 15$ kJ/kg.

$h_{6s} = 206.83$ kJ/kg.

$Q_1 = (h_1 - h_6) + (h_3 - h_{2s})$

$= (3465 - 206.83) + (3565 - 3065)$

$= 3758.17$ kJ/kg.

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

$= (3465 - 3065) + (3505 - 2300).$

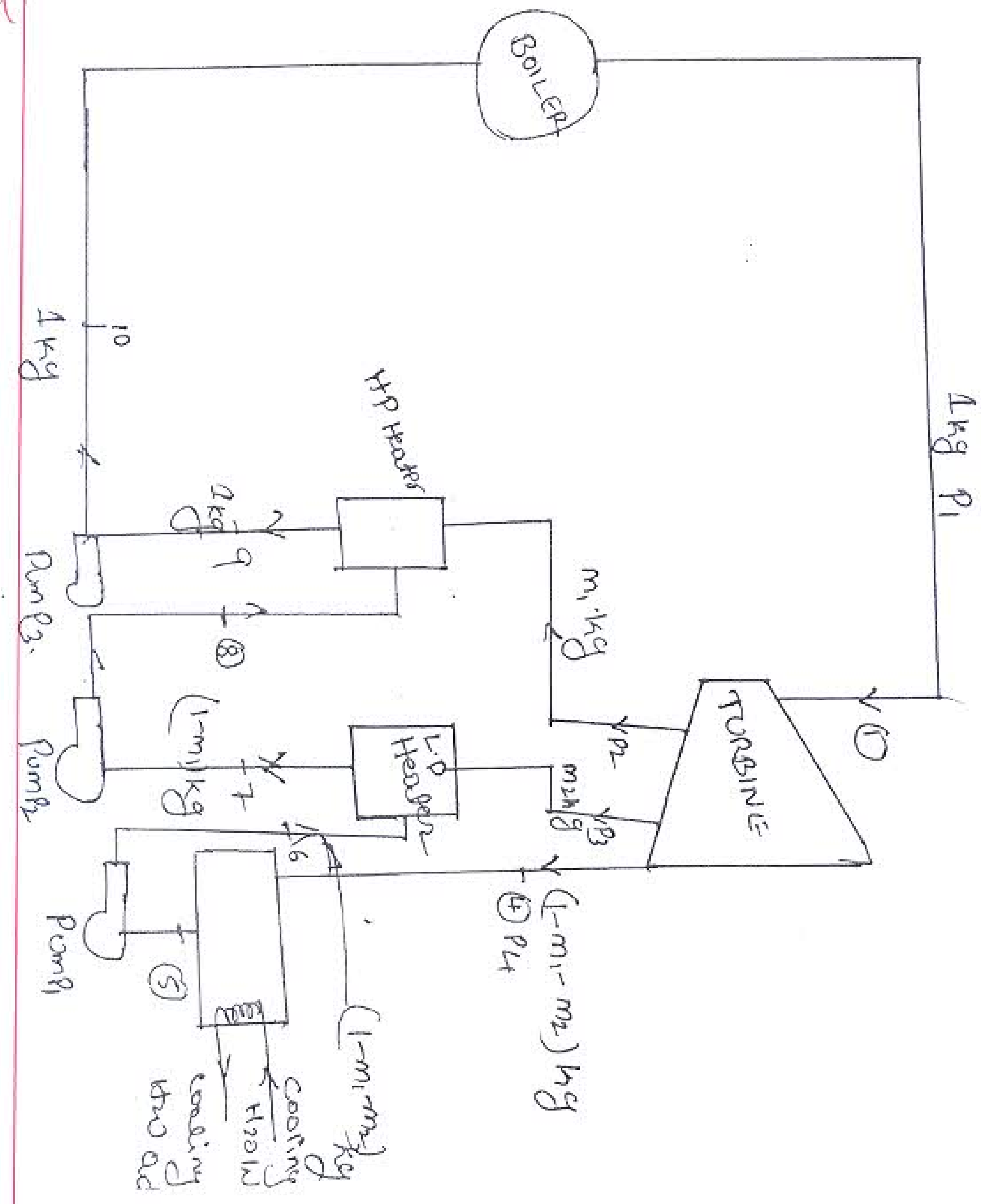
$= 1665 \frac{kJ}{kg}.$

$W_{net} = W_T - W_P = 1665 - 15 = 1650 \frac{kJ}{kg}.$

$\eta = \frac{W_{net}}{Q_1} = \frac{1650}{3758.1}$

$= 43.93\%$

steam rate $= \frac{3600}{1560} = 2.18 kg/kWh$


BOILER
TURBINE
1kg P1
1
m1 kg
P2
m2 kg
P3
(1-m1-m2) kg
4 P4
HP Heater
LP Heater
1kg
9
8
(1-m1) kg
7
6
(1-m1-m2) kg
5
cooling H2O in
cooling H2O out
Pump3
Pump2
Pump1
1 kg
10

4. a

$$m_2 = (1-m_1)\,\frac{H_7 - H_5}{H_3 - H_5}$$

%

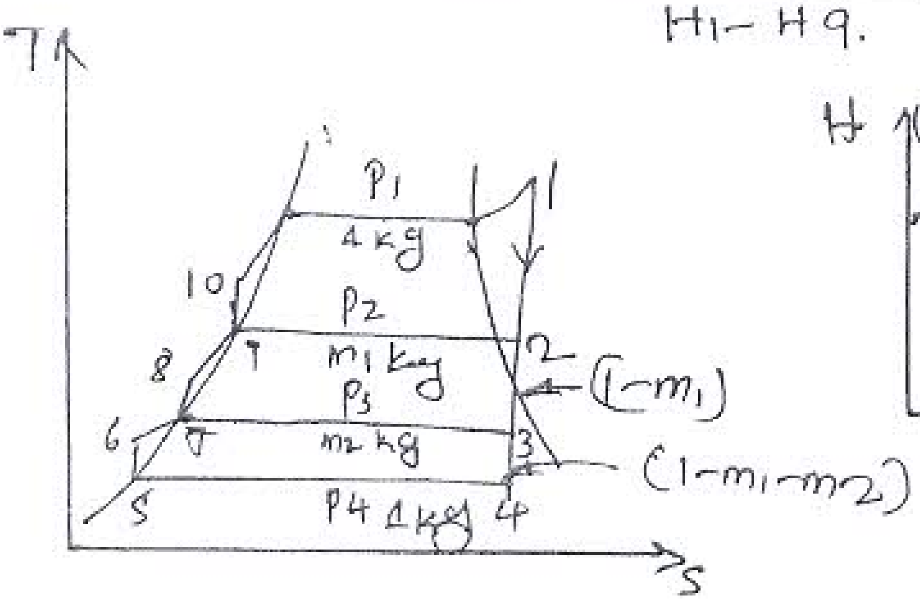
Isentropic turbine work

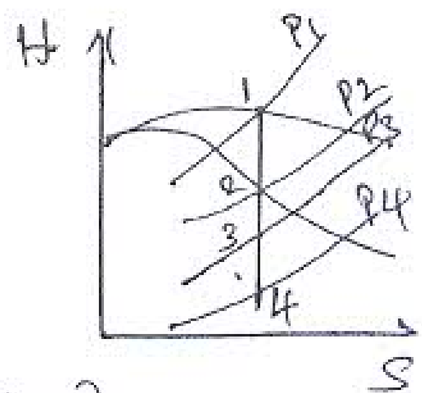
$$\omega_T = m(H_1 - H_2) + m_2(H_1 - H_3) + (1 - m_1 - m_2)(H_1 - H_4)$$

(or) $$\omega_T = m(H_1 - H_2) + (1 - m_1)(H_2 - H_3) + (1 - m_1 - m_2)(H_3 - H_4)$$

$$Q_s = H_1 - H_9$$

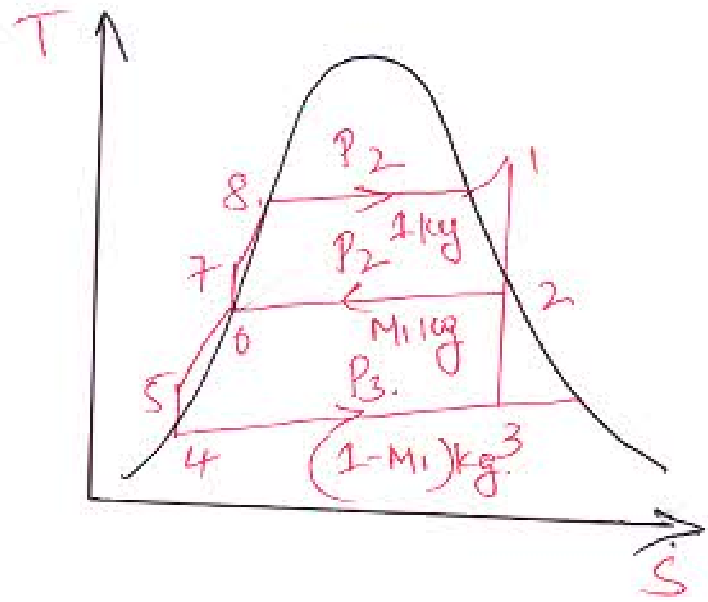
$$\eta_{reg} = \frac{\omega_T}{Q_s} = \frac{m_1(H_1 - H_2) + m_2(H_1 - H_3) + (1 - m_1 - m_2)(H_1 - H_6)}{H_1 - H_9}$$





4 b.

From mollier Chart.



$P_1 = 30$bar $t_{sup} = 350°C$ we get

$H_1 = 3115$ kJ/kg.

$P_2 = 4$bar $x_2^1 = 1$ we get. $H_2 = 2670$ kJ/kg

$H_2^1 = 2738.5$ kJ/kg

$P_3 = 0.1$bar $H_3 = 2190 \frac{kJ}{kg}$

$$\eta_{HPr} = \frac{H_1 - H_2^1}{H_1 - H_2} = \frac{3115 - 2738.5}{3115 - 2670} = 84.6\%.$$

~~$\eta_{LPT} = 0.824$~~ $\eta_{HPT} = \eta_{LPT}$

$$\eta_{HPt} = \eta_{LPT} = \frac{H_2' - H_3'}{H_2' - H_3} = \frac{2738.5 - H_3'}{2738.5 - 2190}$$

$H_3' = 2274.5 \, kJ/kg$

$x_3' = 0.87$

$M_1 H_2' + (1 - m_1) H_4 = H_6$

$H_4 = h_{f3}$ at 0.1 bar $= 191.83 \, kJ/kg$

$H_6 = h_{f2}$ at 4 bar $= 604.7 \, kJ/kg$

$M_1 (2738.5) + (1 - m_1) 191.83 = 604.7$

$M_1 = 0.162 \, kg/kg$ of steam supplied

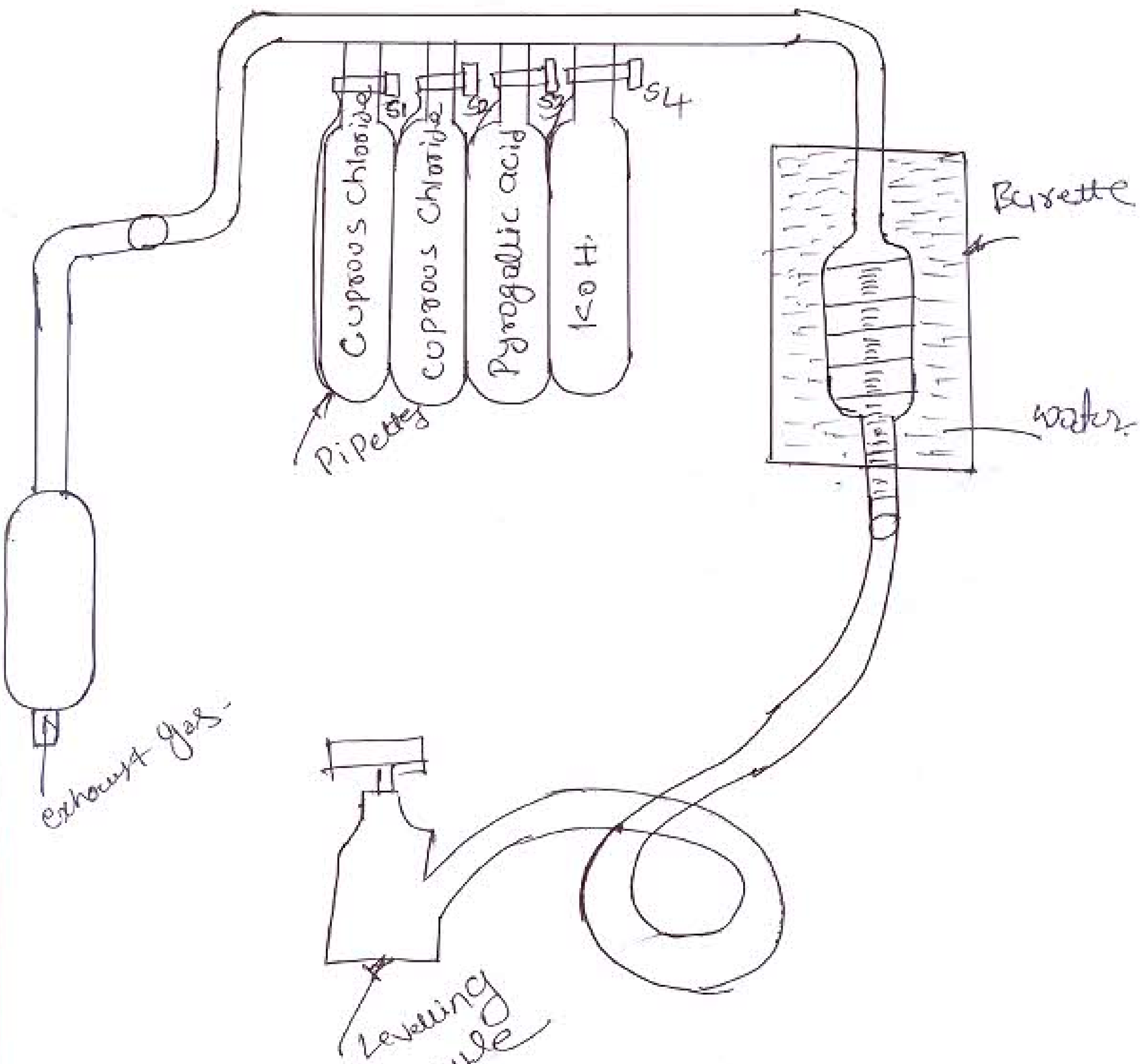
$W_T = (H_1 - H_2') + (1 - m_1)(H_2' - H_3')$

$= 765.33 \, kJ/kg$

$Q_s = (H_1 - H_6) = 2510$

$$\eta_{cy} = \frac{W_T}{Q_s} = \frac{765.33}{2510} = 30.49\%$$

5
A



it is a practical method of analysing exhaust gases.

The main parts of the apparatus are show in fig.

100 cm3 of exhaust gas is drawn into the bottle by lowering levelling bottle.

The stop cock S_4 is then opened and the whole exhaust gas is forced to pipette1. The gas remain in this pipette for some time and most of the CO_2 is absorbed by the KOH present in the pipette. The leveling bottle is then lowered to allow the chemical to come to its original level. The volume of gas thus absorbed is read on the scale of measuring bottle. The flue gas is then forced through the pipette 1 for a number of times to ensure that the whole of the CO_2 is absorbed. Further the remaining flue gas is then forced to the pipette 2 which contains pyrogallic acid to absorb whole O_2, the reading on the measuring burette will be the sum of volume of CO_2 & O_2. The oxygen content can be found out by subtraction. Finally gas is forced through the pipette3 & 4 containing cuprous chloride which absorbs CO completely. The amount of N_2 in the sample can be determined by subtracting from total volume of the gas the sum of CO_2, CO_2 & O_2 contents.

5.

6. Sol

let α moles of O_2 be supplied per mole of fuel. Then the chemical reaction can be written as follows.

$$C_xH_y + \alpha O_2 + 3.76\alpha N_2 \rightarrow 8CO_2 + 0.9CO + 8.8O_2 + \frac{y}{2}H_2O$$

Oxygen balance gives.

$$2\alpha = 16 + 0.9 + 17.6 + \frac{y}{2} \rightarrow (1)$$

Nitrogen balance gives

$$3.76\alpha = 82.3$$

$$\alpha = 21.89 \rightarrow (2)$$

$$\therefore 2 \times 21.89 = 34.5 + \frac{y}{2}$$

$$y = 18.56.$$

Carbon balance gives

$$x = 8 + 0.9$$

$$= 8.9.$$

∴ Chemical formula for the fuel is

$C_{8.9}H_{18.56}$

$$\% \text{ Carbon} = \frac{8.9 \times 12}{(8.9 \times 12) + (18.56 + 1)} \times 100$$

$$= 0.8519.$$

$$\% \text{ Hydrogen} = \frac{18.56 \times 1}{(18.56 \times 1) + (8.9 \times 12)} \times 100$$

$$= \underline{\underline{0.148}} = 14.8\%$$

ii) Air fuel ratio.

$$A:F = \frac{32d + 3.76d \times 28}{12x + y}$$

$$= \frac{(32 + 3.76 \times 28) \times 21.89}{(12 \times 8.9) + 18.56}$$

$$A:F = 23.97.$$

% Excess air = ~~67.22~~%

$$\frac{8.8 \times 32}{(21.89 \times 32) - (8.8 \times 32)} \times 100$$

% Excess air = 67.22%

iv) Dew Point temperature of the Product.

$$C_{8.9}H_{18.50} + 21.89\,O_2 + 82.3\,N_2 \rightarrow 8CO_2 + 82.3N_2 + 9.28H_2O + 0.9CO + 8.8O_2$$

Total moles of the Product.

$= 8 + 0.9 + 8.8 + 82.3 + 9.28 = 109.28$

Mole fraction $X_{H_2O} = \frac{9.28}{109.28} = 0.0849$

$P_{H_2O} = 0.849 \times 1.01325$

$P_{H_2O} = 0.086\ bar$

$T_{DP} = 42.89^\circ C$

6 a)

Morse test:

This test is only applicable to multi-cylinder engines.

The engine is run at the required speed. The torque is measured. One cylinder is cut out by shorting the plug if an S.I. engine is under test. The speed falls [illegible] with one cylinder cut out, and is restored by reducing the load. The torque is measured again when the speed has reached its original value. If the values of I.P. of the cylinders are denoted by I_1, I_2, I_3 & I_4 (considering 4-cylinder engine), and power losses (friction power) in each cylinder are denoted by F_1, F_2, F_3 & F_4, then the value of BP, B at the test speed with all cylinders firing is given by

$$B = (I_1 - F_1) + (I_2 - F_2) + (I_3 - F_3) + (I_4 - F_4) \rightarrow (1)$$

If number 1 cylinder is cut out, then the contribution I_1 is lost; and if the losses due to that cylinder remain the same as when it is firing, then the BP, B_1 now obtained at the same speed is

Put $I_1 = 0$ in the above equation we get

This method is also known as fuel rate extrapolation Method. A graph conecting fuel consumption (y-axis) and brake power (x-axis) at constant speed is drawn & it is extrapolated on the negative axis of the Brake power. The intercept of the -ve axis is taken as the friction power of the engine at that speed. The method of extrapolation is shown in the above fig. In the above fig the fuel consumption & brake power is linear which permits extrapolation. Further, when the engine dose not develop any power i.e BP = 0, it consumes a certain amount of fuel, the energy would have been spent in over coming the friction. hence, the extrapolated -ve intercept of the x-axis will be the work representing the combined losses due to mechanical friction, pumping work & blowby and as a whole it is termed the frictional loss of the engine. it should be noted that the measured frictional power by this method will hold good only for a practicular speed & is appricable mainly to C.I engines

$B_1 =$ [illegible]

Subtracting eqn (2) from eqn (1), we get

$B =$ [illegible]

[illegible]

$B - B_1 = I_1$ ⟶ (2) [illegible]

Similarly

$B - B_2 = I_2$ when cylinder number 2 is cut out

$B - B_3 = I_3$ ⟶ 〃 〃 3

$B - B_4 = I_4$ ⟶ 〃 〃 4

Then for the engine,

$I = I_1 + I_2 + I_3 + I_4$ ⟶ (4)

$FP = (IP)_n - (BP)_n$

where n is the number of cylinders

6.7441 Liter per hr = 0.00131 kg/sec

$M_f = V_f \times \rho_f =$ ~~6.74×10^{-6}~~ $\times 1000 \times 0.735$

0.00131

$M_f = 4.9539 \times 10^{-3}$ kg/sec $= 9.55 \times 10^{-4}$

iii) $\eta_{bth} = \frac{BP}{m_f \times CV} = \frac{16.187}{\text{~~}4.953 \times 10^{-3}\text{~~ } 9.55 \times 10^{-4} \times 44200}$

~~7.39~~ 6 38.34 %

(iv) BSFC $\frac{m_f}{BP} \times 3600 = \frac{\text{~~}4.9539 \times 10^{-3}\text{~~ } 9.55 \times 10^{-4}}{16.187 \times \text{~~}44200\text{~~}} \times 3600$

0.212 ~~0.089~~ kg / kw.h

$BP_1 = \frac{2 \pi N T_1}{60} = \frac{2 \times \pi \times 2800 \times 39.516}{60 \times 1000}$

$= 11.588$ kw.

$BP_2 = \frac{2 \pi N T_2}{60} = \frac{2 \times \pi \times 2800 \times 37.914}{60 \times 1000}$

$= 11.11$ kw.

$T_1 = W_1 \times R$

$= 39.516$ N-M

$T_2 = 37.914$

$T_3 = 37.0952$ Nm

$T_4 = 39.6228$ Nm

6.
b) Sol

i Engine torque.

T = Net brake load × length of torque arm.

= ~~150×0.~~ 155 × 0.356

= 55.18 NM.

$BP = \frac{2\pi NT}{60} = \frac{2\pi \times 2800 \times 55.18}{60 \times 1000}$

BP = 16.187 kW

0.047

$BP = \frac{P_{Mb} LAK \times N}{60}$

$A = \frac{\pi d^2}{4}$

$= \frac{\pi \times (60 \times 10^{-3})^2}{4}$

$= 0.002827\ m^2$

2.822

$16.187 = \frac{P_{Mb} \times 90 \times 10^{-3} \times 0.0028 \times 1 \times 2800 \times 4}{60}$

~~13.76 bar~~ 344.16 kPa

= 3.44 bar

$$BP_3 = \frac{2\pi N T_3}{60\times1000} = \frac{2\times\pi\times2800\times37.095}{60\times1000}$$

$$\underline{\underline{10.878\,kW}}$$

$$BP_4 = \frac{2\pi N T_4}{60\times1000} = \frac{2\times\pi\times2800\times39.622}{60\times1000}$$

$9.55\times T_4$

$$BP_4 = \underline{\underline{11.619}}\ kW.$$

$IP_1 = BP - BP_1$

$= 16.187 - 11.888 = 4.599$

$IP_2 = 16.187 - 11.11 = 5.077$

$IP_3 = 16.187 - 10.878 = 5.309$

$IP_4 = 16.187 - 11.619 = 4.568$

$$\eta_{mech} = \frac{BP}{IP} = \frac{16.187}{19.553} = 82.78\%$$

$IP = 19.553$

$$\eta_{th} = \frac{IP}{m_f \times C_V} = \frac{19.553}{9.55\times0.4\times44200}$$

$\underline{\underline{46.32\%}}$

i)

ii)

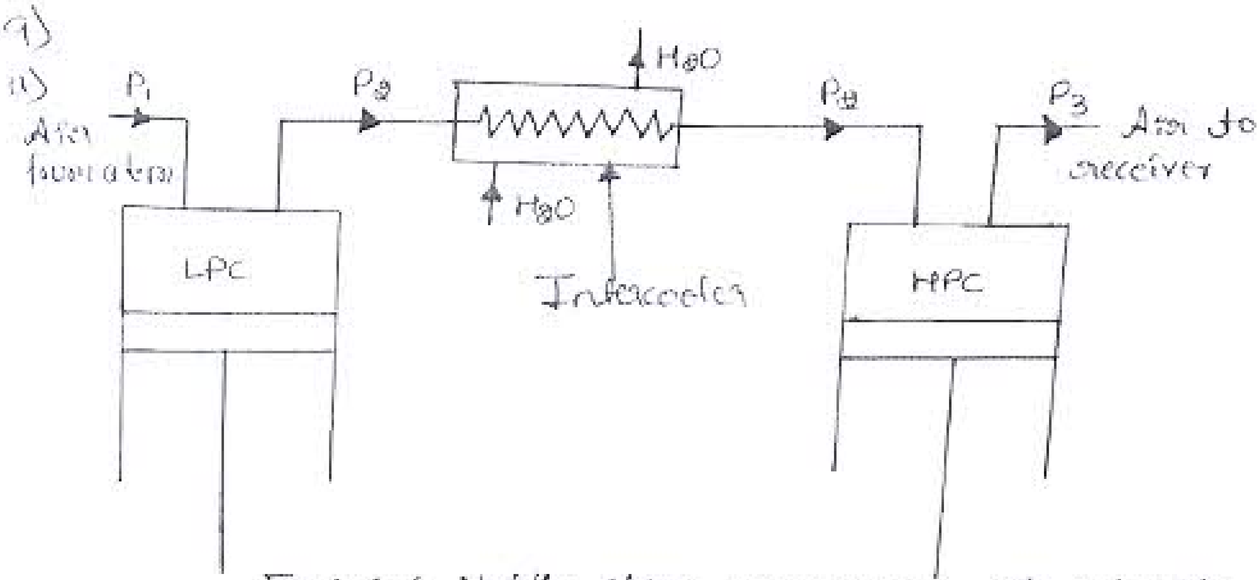


Fig :- 8.6 Multi-stage compression with intercooling

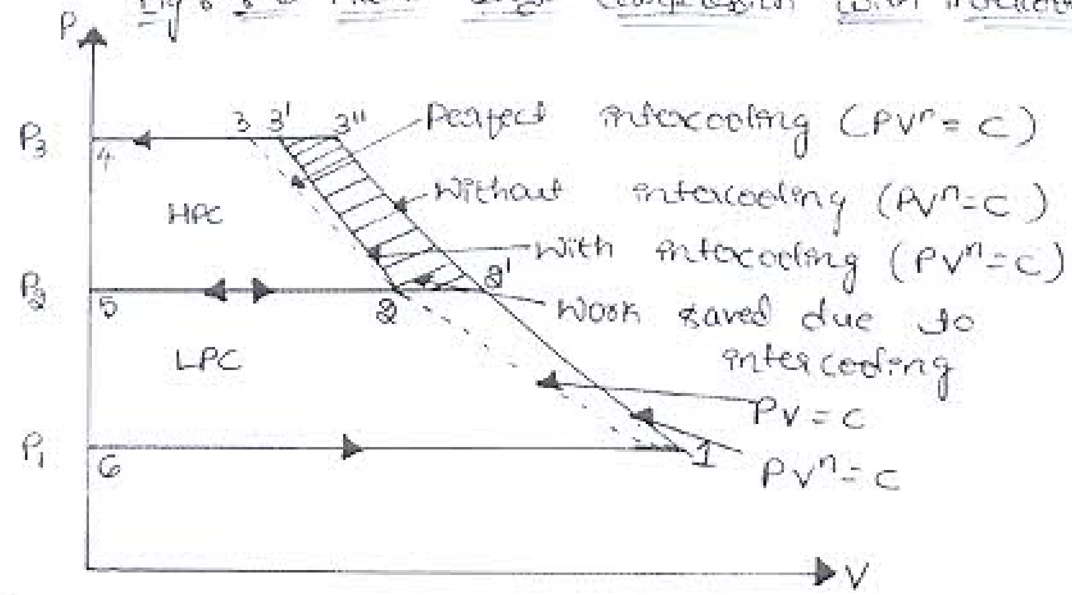


Fig : 8.6 P-V diagram of two stage Air compressor

Consider a two stage air compressor as shown in fig.

Case (i) : Without intercooling :-

Work done /cycle = $W_{LPC} + W_{HPC}$

$$= \left(\frac{n}{n-1}\right) P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] + \left(\frac{n}{n-1}\right) P_2 V_2' \left[\left(\frac{P_3}{P_2}\right)^{\frac{n-1}{n}} - 1\right]$$

Now work done / cycle for two stage compression:

$$W=\left(\frac{n}{n-1}\right)P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}+\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}-2\right]$$

$$=\left(\frac{n}{n-1}\right)P_1V_1\left[2\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}-2\right]$$

$$=\left(\frac{2n}{n-1}\right)P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}-1\right]$$

But $P_2=\sqrt{P_1P_3}=(P_1P_3)^{1/2}$

$$\therefore W=\left(\frac{2n}{n-1}\right)P_1V_1\left[\left(\left(\frac{P_1P_3}{P_1}\right)^{1/2}\right)^{\frac{n-1}{n}}-1\right]$$

$$\boxed{W=\left(\frac{2n}{n-1}\right)P_1V_1\left[\left(\frac{P_3}{P_1}\right)^{\frac{n-1}{2n}}-1\right]}$$

Similarly for 3 stages

$$W=\left(\frac{3n}{n-1}\right)P_1V_1\left[\left(\frac{P_4}{P_1}\right)^{\frac{n-1}{3n}}-1\right]$$

For x - stages

$$\boxed{W=\left(\frac{xn}{n-1}\right)P_1V_1\left[\left(\frac{P_{x+1}}{P_1}\right)^{\frac{n-1}{xn}}-1\right]}$$ //

Note:-

$$\boxed{\text{To show } z=\left(\frac{P_{x+1}}{P_1}\right)^{1/x}}$$

where, z = pressure ratio
x = no of stages.
P = initial pressure of the air.

7>

a) Ans. The absorption refrigeration system is a heat operated unit which uses a refrigerant that is alternately absorbed and liberated from the absorbent. In the basic absorption system, the compressor in the vapour compression cycle is replaced by an absorber - generator assembly involving less mechanical work.

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