THERMODYNAMICS

MAY-18

Q.1 Attempt any FOUR out of the following: [20]

Q.1(a) State kelvin-plank statement and clausius statement of the second law of thermodynamics. [5]

Ans: Efficiency of the engine is given by,

$$n = \frac{W_{net}}{Q_1} = 1 - \frac{Q_2}{Q_1}$$

We know that for any engine, $w_{net} < Q_1$ sine heat Q_1 is transferred to a system cannot be completely converted to work in a cycle. Therefore, the efficiency is always less then unity. Hence, $Q_2 > 0$i.e there has always to be a heat rejection. To produce network in a thermodynamic cycle, a heat engine has thus to exchange heat with two reservoirs, the source and the sink.

Kelvin-plank statement of second law of thermodynamics:

It is impossible to heat engine to produce Net Work in complete cycle if it exchanges heat only wit bodies at a single fixed temperature. Heat flows from a high temperature body to ow temperature body. The reverse process can never occur spontaneously.

Clausius statement of second law of thermodynamics:

It is impossible to construct a device which, operating in a cycle, will produce no effect other than the transfer of heat from a low temperature body to high temperature body.

Q.1(b) Draw a neat diagram of roots blower and explain its working. [5]

Ans**:** a) **Roots Blower:** Is an extension of the idea of a gear pump, popular in engines for pumping oil. There are two lobes on each rotor, and their shape is of cycloidal or involute form (refer fig (a) one of the lobes is connected to the drive and the second is gear driven from the first, the two rotating in opposites directions. very small clearances between the lobes and between the casing and the lobes are provided to prevent leakages and to reduce wear.

b) Four times the volume between the casing and one side of the rotor will be displaced in each revolution of the driving shaft.as each side of each lobe faces its side of the casing volume of gas V. at pressure $p_{1,}$ is displaced towards the delivery side at constant pressure.

c) A further rotation of the rotor opens this volume to the receiver and the gas flows back from the receiver, since the gas is at a higher pressure.

d) The gas induced is compressed irreversibly by that from the receiver to the pressure p_2, and then delivery begins. this process is carried out four time per revolution the driving shaft. The p-

V diagram for this machine is shown in fig (c). In which the pressure risers irreversibly from p_1 and p_2 at constant volume.

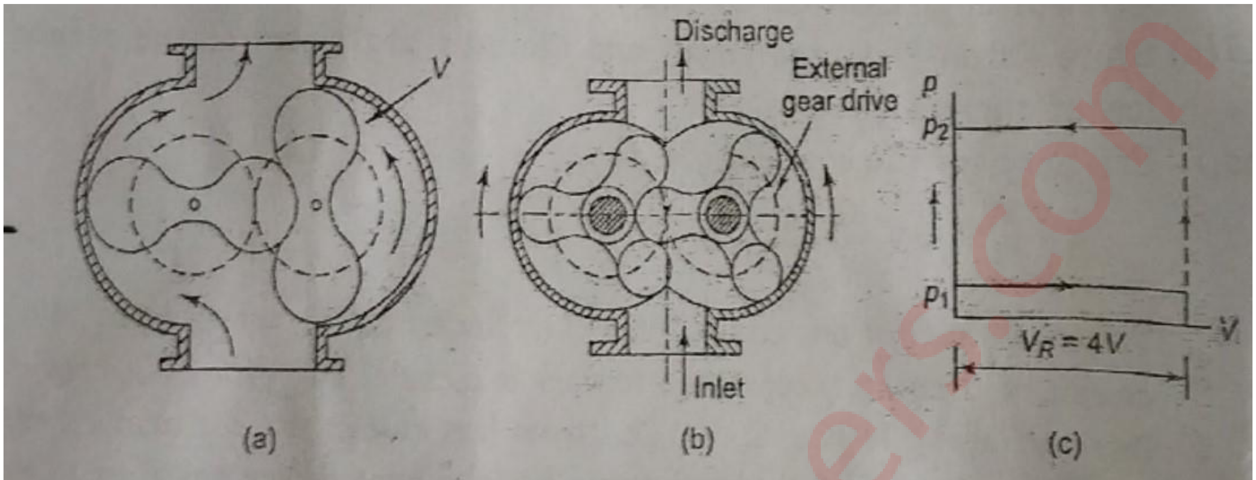


Fig.: Roots blower with (a) Two lobe rotor, (b) Thee lobe rotor, and (c) P-C diagram

Fig (a)

For pressure ratio of 1.2,1.6 and 2.0, the roots blower efficiency becomes 0.945,0.84 and 0.765 respectively, which show that the efficiency decreases as the pressure ratio increases.

Q.1(c) show that energy is a property of a system. [5]

Ans: Energy - A property of a system

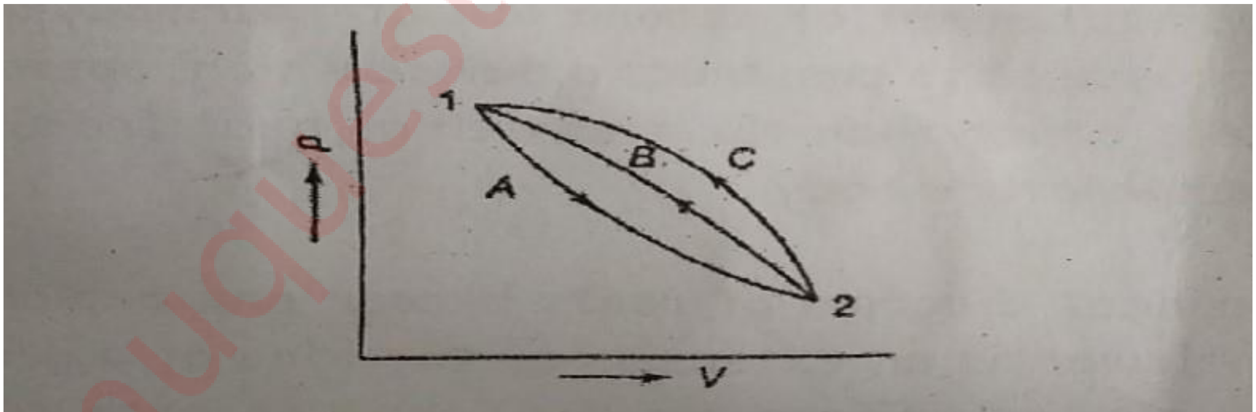


Fig.: Energy – A property of a system

Fig(a)

Consider a system which changes its state from state 1 to state 2 by following the path A, and the returns from state 2 to state 1 by following the path B (refer fig (a)). so the system undergoes a cycle. Writing the first or path A

$$Q_A = \Delta E_\Delta + W_A \qquad (1)$$

For path

$$Q_B = \Delta E_B + W_B \qquad (2)$$

The process A and B together constitute a cycle, for which

$$(\sum W)_{CYCLE} = (\sum Q)_{CYCLE}$$

OR $W_A + W_B = Q_A + Q_B$

OR $Q_A - W_A = W_B - Q_B \qquad (3)$

From eq (1), (2) and (3) its yields

$$\boldsymbol{\Delta E_A = -\Delta E_B} \qquad (4)$$

Similarly, had the system returned from the state 2 to state 1 but following the path C instead of path B

$$\boldsymbol{\Delta E_A = -\Delta E_C} \qquad (5)$$

From eq (4) and (5)

$$\boldsymbol{\Delta E_B = \Delta E_C} \qquad (6)$$

Therefore, it is seen that the change in energy between two states of a system is the same, whatever the path the system may follow in undergoing that change of state. If some arbitrary value of energy is assigned to state 2 the value of energy at state 1 is fixed independent of the path the system follows. Therefore, energy has a definite value of energy state of the system. Hence it is a point function and a property of the system

The energy E is an extensive property. The specific energy, e = E/m (j/kg). is an extensive property.

The cyclic integral of any property is zero, because the final state is identical with the initial state.

Q1 (d) Draw a simple schematic diagram of a thermal power plant with one reheated. Also represent this on T-S diagram. [5]

Ans: Reheat cycle: the reheat cycle is designed to take advantage of high boiler pressure by eliminating the problem of excessive moisture content in the exhaust system.

In the reheat cycle, the steam is expanded in number of stages. After each stages of expansion, the system is reheated on the boiler, then it expands in the next stage of turbine and is finally exhausted to condenser. In the reheat cycle the expansion of steam from the initial state 1 to the condenser pressure is carried out in two stages, depending upon the number of reheats used. In the first step, steam expands in high pressure turbine from initial state to approximately the saturated vapors line (process 1-2S). the steam is then reheated at a constant pressure in boiler (process 2S-3) and the remaining expansion process (process 3-4s) is carried out in the low pressure turbine. in the case of 2 re-heaters, steam is reheated twice at two different constant pressure.

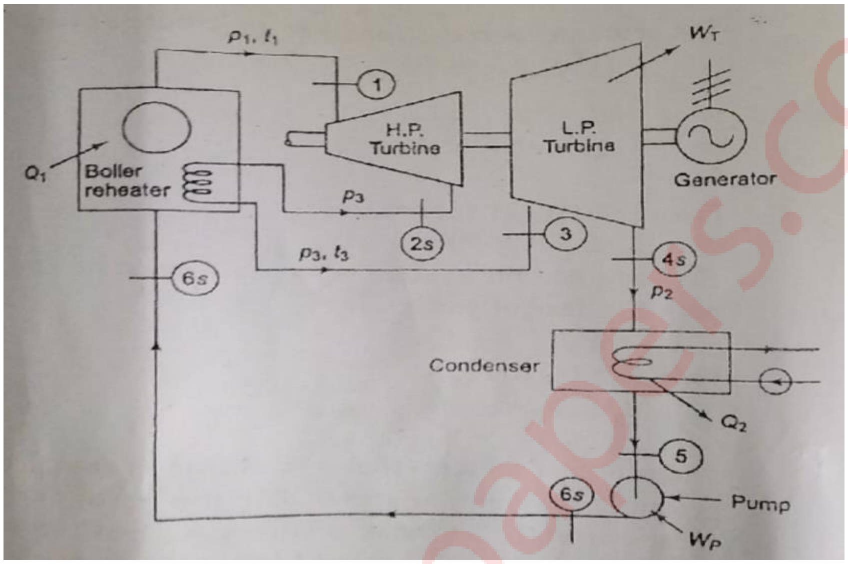


Fig (a)

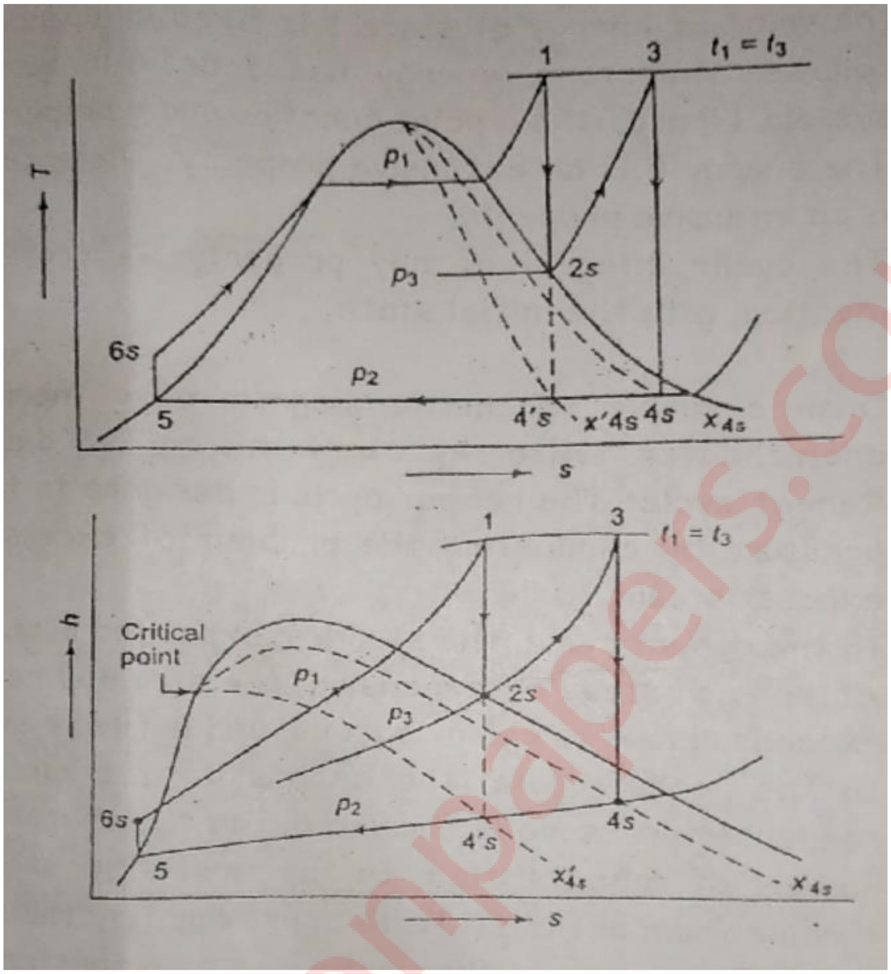


Fig (b)

Effects of reheat cycle:

1. Turbine work (W) increases.
2. Pump work (W) is constant.
3. Net work done (W) increases.
4. Steam rate decreases.
5. Heat supplied increases.
6. Dryness fraction increases.
7. Condenser load increases.
8. Thermal efficiency may increase or decrease or remains constant.

Q1(e) define dryness faction saturation temperature work ratio and the specific steam consumption.
[5]

Ans: Dryness fraction : the quality of steam as regards its dryness is termed as dryness fraction.

Dryness fraction is usually expressed by the symbol ‘X’.
Dryness fraction often spoken as the quality of wet steam. If M_S
= mass of dry steam contained in the steam considered, and M
= masses of water in suspension in the steam considered.

$$x = \frac{m_s}{m_s + m}$$

Thus, if dryness faction of wet steam, X = 0.8 then one kg of wet steam contains 0-2 kg moisture (water) in suspension and 0-8 kg of dry steam.

Saturation temperature: it is the maximum temperature corresponding to a given pressure at which a substance can exist in liquid form.
When a liquid and its vapor are in equilibrium at a certain pressure and temperature, only the pressure or the temperature is sufficient to identify the saturation state. If the pressure is given, the temperature of the fixture gets fixed, which is known as the ‘saturation temperature’.

Work ratio: the work ratio for a power plant is defined as the ratio of the network output of the cycle to the work developed by the turbine.

$$work\ ratio = \frac{network\ output}{turbine\ work} = \frac{Wnet}{W_T}$$

$$= 1 - Back\ work\ ratio$$

Specific steam consumption
It is also called as steam rate. It relates the power output to amount of steam necessary to produce it. It is amount of steam required to produce 1 kwh (3600 KJ) of power. It is denoted by SSC (specific steam consumption) is expressed as

$$SSC = \frac{Mass\ pf\ steam\ in\ (\frac{kg}{h})}{power\ output\ in\ kw^h} = \frac{[m_s(\frac{kg}{h})]}{[m_s(\frac{kg}{s})w_{net}\frac{kg}{kj}]}$$

$$SSC = \frac{[3600\ (\frac{kj}{kwh})]}{\frac{kj}{}}$$

$[W_{net}\ (kg)]$

Q.2 (a) 'steam enters a nozzle at a pressure of 7 bar and 200° C with an initial enthalpy of 2850 kj/kg and leaves at a pressure of 1.5 bar. The initial velocity of steam at the entrance is 40 m/s and the exit velocity from the nozzle is 700 m/s. the mass flow rate through the nozzle is 14000 kg/hr. the heat loss from the nozzle is 11705 kJ/hr. find the final enthalpy of steam and the nozzle area at exit if the specific volume at exit is 1.24 m^3/kg. [10]

Ans: P_1= 7 Bar

T_1= 200° C

H_2= 2850 KJ/kg

V_1= 40 m/s

P_2 = 1.5 bar

V_2 = 700 m/s

h_2 =?

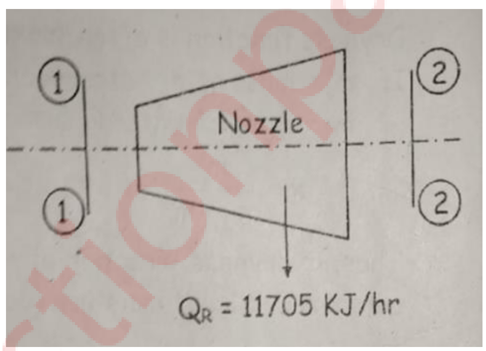


Fig (a)

Heat rejected from nozzle

$(Q_R) = 11705 \frac{kj}{hr}$ $\frac{11705}{3600}$ =

$= \mathbf{3.25\ kw}$ Mass

flow rate.

$M = 14000 \frac{kg}{hr}$

$\frac{14000}{3600}$ $= =$ $\mathbf{3.889 \frac{kg}{s}}$

Specific volume at exit (V_2) = **1.24 m^3/kg**

Applying the steady flow energy equation at nozzle inlet and outlet.

$$Q + m\,[h_1 + \frac{v12}{2000} + gz_1] = W + m\,[h_2 + \frac{v22}{2000} + gz_2]$$

Here, W = PE = 0

$$Q + m\,[h_1 + \frac{v12}{2000}] = m\,[h_2 + \frac{v22}{2000}]$$

$$3.25 + 3.889\,[2850 + \frac{40^2}{2000}] = 3.889\,[h_2 + \frac{700^2}{2000}]$$

$$\boldsymbol{h_2 = 2606.64\,\frac{kj}{kg}}$$

Exit nozzle area (A_2)

We have,

$$A_2 = \frac{mv^2}{V_2}$$

$$A_2 = 3.\frac{889 \times 1.24}{700}$$

$$A_2 = 6.88 \times 10^{-3}\,m^2$$

Exit area

$$\boldsymbol{A_2 = 68.8\,cm^2}$$

Q.2 (b) state and prove the clausius inequality. **[5]**

Ans: let us consider a cycle ABCD Let A be a general process, either reversible or irreversible, while the other process in the cycle are reversible. Let the cycle be divided inti number of elementary $\square = 10 - \frac{dQ_2}{dQ}$.

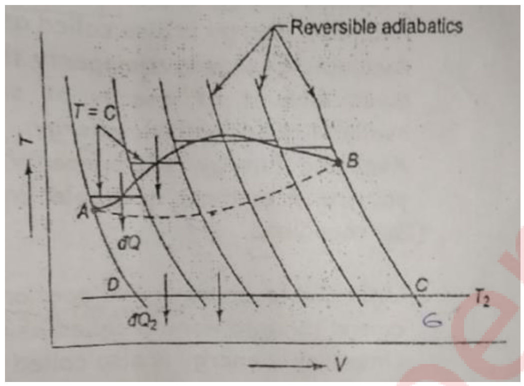


Fig (a)

Where dQ is the heat supplied at t and dQ_2 is the heat rejected at T_2 now the efficiency of the general cycle will be less than or equal to the efficiency of the reversible cycle.

$$n_{Irr} \geq n_{Rev}$$

$$1 - \frac{dQ_2}{dQ} \leq \left(1 - \frac{dQ_2}{dQ}\right)_{Rev}$$

Or, $$\frac{dQ_2}{dQ} \geq \left(\frac{dQ_2}{dQ}\right)_{Rev}$$

Or, $$\frac{dQ}{dQ_2} \leq \left(\frac{dQ}{dQ_2}\right)_{Rev}$$

But, $$\left(\frac{dQ}{dQ_2}\right)_{Rev} = \frac{T}{T_2}$$

Hence, $$\frac{dQ}{dQ_2} \leq \frac{T}{T_2}$$

For reversible process,

$$ds = \frac{dQ_{rev}}{\quad} \quad \frac{dQ_2}{\quad} =$$

$$T \qquad T_2$$

Hence for any process AB,

$$\frac{dQ}{T} \leq ds$$

Then for cycle,

$$\oint \left(\frac{dQ}{T}\right) \leq \oint ds$$

Since entropy is a property and cyclic integral of any property is zero.

$$\oint \left(\frac{\boldsymbol{dQ}}{\boldsymbol{T}}\right) \leq \mathbf{0}$$

This equation is known as clausius inequality.

It provides the criterion for the reversibility of the cycle. If,

$\oint\left(\frac{\boldsymbol{dQ}}{\boldsymbol{T}}\right) = \mathbf{0}$, the cycle is reversible.

$\oint\left(\frac{\boldsymbol{dQ}}{\boldsymbol{T}}\right) < \mathbf{0}$, the cycle is possible and reversible.

$\oint\left(\frac{\boldsymbol{dQ}}{\boldsymbol{T}}\right) > \mathbf{0}$, the cycle is possible.

Q.2(c) What is available energy and unavailable energy? [5]

Ans: **Available energy:** It is the portion of energy supplied as heat energy which can be converted into the useful work by ideal process, which reduces the system to a dead state.

Available energy is a property of system.

Available energy is also called as energy.

Available energy is a property that determined the useful work potential at given amount of energy at some specified state. It is also known as availability or available energy.

Available energy is a property which is used to identify the potential pf particular system. Available energy is a property which links system and surroundings.

Unavailable energy: the portion of energy supplied as heat which cannot be converted into work is called as unavailable energy.

Unavailable energy is also called as energy.

Work output is maximized when the process between two specified state is executed in reversible manner.

The system must be dead state at the end of the process to maximize the work output.

A system that is in equilibrium with its environmental is said to be at the dead state.

At dead state, the useful work potential of a system is zero.

The system must be dead state at th end of the process to maximize the work output.

Q3(a) A heat engine is used to drive a heat pump. The heat transfers from the heat engine and the heat pump are used to heat the water circulating through the radiators of a building. The efficiency of the heat engine is 27% and COP of the heat pump is 4. Evaluate the ratio of the heat transfers to the circulating water to the heat transfer to the heat engine. [10]

Ans: Efficiency of the heat engine.

η_{HE} = 27 %

C.O.P = 4

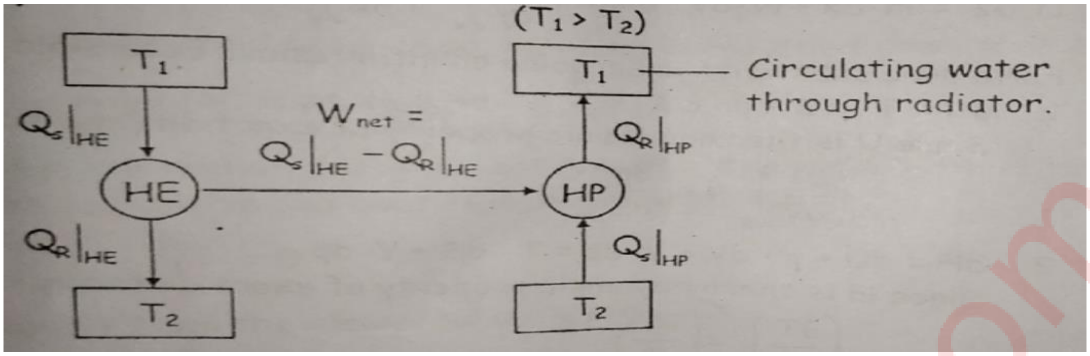


Fig (a)

We know that,

Efficiency of heat engine

$\eta_{HE} = \frac{W_{net}}{\text{heat supplied}} = [\,(Q_S)_{HE} - (Q_R)_{HE}\,] \,/\, [\,(Q_S)_{HE}\,] = 0.27$

$0.27\,(Q_S)_{HE} = (Q_S)_{HE} - (Q_R)_{HE}$ (1) Now,

$(C.O.P)_{HP} = \frac{(\text{heat supplied to circulating water})}{W_{net}}$

$4 = \frac{(Q_R)_{HP}}{(Q_S)_{HE} - (Q_R)_{HE}}$

$4 = \frac{(Q_R)_{HP}}{0.27\,(Q_S)_{HE}}$ from (1)

$\therefore \frac{(Q_R)_{HP}}{(Q_S)_{HE}} = 0.27 \times 4 = 1.08$

The ratio of heat transfer to circulating water $(Q_R)_{HP}$ to the heat transfer to the heat engine $(Q_S)_{HE}$

$$\boldsymbol{\frac{(Q_R)_{HP}}{(Q_S)_{HE}} = 1.08}$$

Q.3(b) Write Maxwell's equations. **[5]**

Ans: A pure substance existing in single phase has only two independent variables of the 8 quantities P, V, T, S, U, H, F (Helmholtz function|) and G (Gibbs function) anyone may be expressed as a function of any two others.

If dz = m. dx + N. dy then, $\left(\frac{\partial M}{\partial Y}\right)_X = \left(\frac{\partial N}{\partial X}\right)_Y$

For a pure substance undergoing an infinitesimal reversible process,

1. dU = T. dS – p. dV since U is thermodynamic property of exact differentials.

$$\left(\frac{\partial T}{\partial V}\right)_s = \left(\frac{\partial p}{\partial S}\right)_V$$

2. dH = dU + p. dv + V. dp = T. ds + V. dp since H Is thermodynamics property of exact differentials,

$$\left(\frac{\partial T}{\partial p}\right)_S = \left(\frac{\partial v}{\partial S}\right)_P$$

3. dF = dU-T. dS – S. dT = -P. dV – S. dT since F is the thermodynamics property of exact differentials,

$$\left(\frac{\partial p}{T}\right)_V = \left(\frac{\partial S}{\partial V}\right)_T$$

4. dG = dH – T. dS – S. dT = V. dp – S. dt
   Since G is the thermodynamics property of exact differentials,

$$\left(\frac{\partial V}{\partial T}\right)_P = \left(\frac{\partial S}{\partial p}\right)_T$$

Q.3(c) State and prove clausius theorem. **[5]**

Ans: Let a smooth close curve representing a reversible cycle be considered. Let the closed cycle be divided into large number of strips by means of reversible adiabatic. Each strip may be closes at top and bottom by reversible isotherms.

The original closed cycle is thus replaced by zig zag closed path consisting of alternate adiabatic and isotherm process, such that heat transfer during all the isothermal process is equal to the heat transferred in the original cycle.

Thus an original cycle is replaced by a larger number of Carnot cycles, if the adiabatic are closed to one another and the number of Carnot cycle is large, the zig zag line will coincide the original cycle.

For the elemental cycle abcd, dQ_1 heat is absorbed reversibly at T_1 and Dq_2 heat is rejected reversibly at T_2, $\frac{dQ_1}{T_1} = \frac{dQ_2}{T_2}$

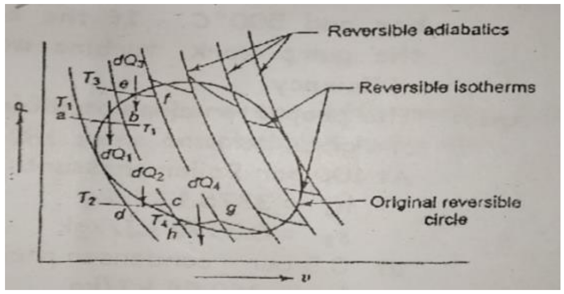


Fig (a)

If the sign convention for the heat is applied then, (heat supplied is taken as positive and heat is rejected negative.

$$\frac{dQ_1}{T_1} + \frac{dQ_2}{T_2} = 0$$

Similarly, from elemental cycle efgh,

$$\frac{dQ_3}{T_3} + \frac{dQ_4}{T_4} = 0$$

If similar equations are written for all elements,

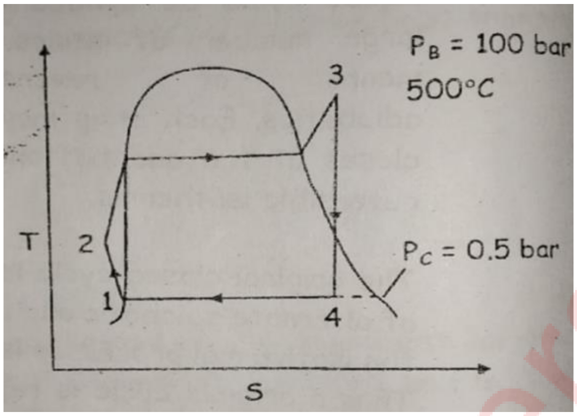
$$\frac{dQ_1}{T_1} + \frac{dQ_2}{T_2} + \frac{dQ_3}{T_3} + \frac{dQ_4}{T_4} + .. = 0$$

Or $$\oint_R \left(\frac{dQ}{T}\right) = 0$$

The cyclic integral of $\left(\frac{dQ}{T}\right)$ or a reversible cyclic is equal to zero. This is known as clausius theorem.

Q.4(a) In a Rankine cycle the stream at the inlet to the turbine is at 100 bar and 500° C. If the exhaust pressure is 0.5 bar, determine the pump work, turbine work, condenser heat flow and Rankine efficiency. [10]

Ans: The property values at state points found from the steam tables are given below.



At 100 bar- Boiler pressure:

h_3 = 3375.1 kJ/kg

S_3 = 6.5995 kJ/kg

At 0.5 bar- condenser pressure.

h_F =340.54 kJ/kg

S_f = 1.0912 kJ/kg

h_g= 2645.2 kJ/kg

S_g= 7.5931 kJ/kg

V_f = 0.001030 m^3/kg

Now, $S_3 = S_4$ =6.5995 kJ/kg k

$S_4 = S_f + X (S_g - S_F)$

6.5995 = 1.0912 + X (7.5931 – 1.0912)

X = 0.847 (dryness fraction)

Now $h_4 = h_f + X (h_g - h_f)$ h_4 =

340.54+0.847 (2645.2 -340.54) **h_4**

= 2292.587 kJ/kg

At 0.5 bar $h_1 = h_f$ = 340.54

Pump work is given by,

$(h_2 - h_1) = -\int vdp$

$= 0.001030\ (10000 - 50)$

$h_2 - 340.54 = 10.249$ **h_2 =**

3507896 kJ/kg pump work

$(W_P) = 10.249$ kJ/kg

- turbine work $(W_T) = h_3 - h_4$

$= 3375.1 - 2292.587$

W_T = 1082.513 kJ/kg

- Condenser heat flow (Q) = $(h_4 - h_1)$

$= 2292.587340.54$

Q = 1952.047 kJ/kg

Rankine efficiency $(\eta) = \frac{W_T - W_P}{Q_S}$

Here, Q_S = heat supplied by boiler.

$Q_S = (h_3 - h_2) = 3375.1 - 350.789$

$=$ **3024.311 kJ/kg**

$$\eta = \frac{1082.513 - 10.249}{3024.311} = 0.3545$$

η =35.45 %

Q.4(b) With the help of P-V and T-S diagram, compare the efficiencies of Otto, diesel and dual cycle for the same compression ratio and the same heat rejection. [5]

Ans: Otto: 1234

Diesel: 123'4

Dual: 122"3"4

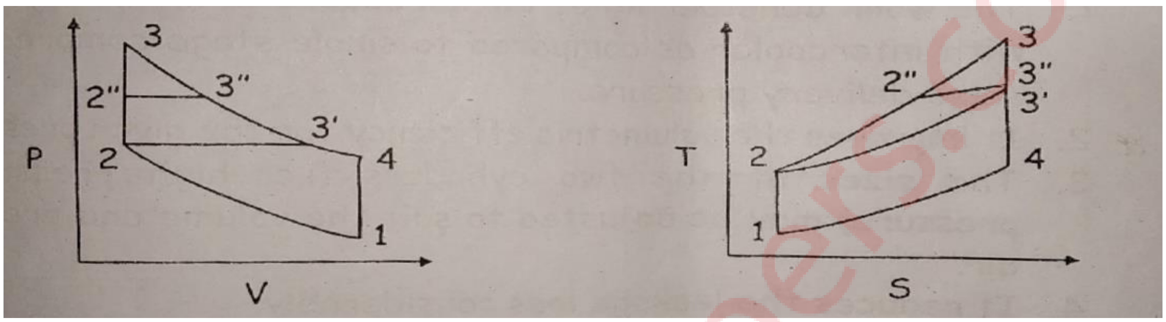


Fig (a)

From above figures

It is seen that for the same heat rejection the heat input in Otto cycle is maximum and heat in diesel cycle is minimum hence, Otto cycle has the highest efficiency and the diesel has least. Dual cycle having the efficiency between two.

Q.4(C) What do you understand by multistage compression? What are its merits over single stage compression? [5]

Ans: In single stage compressor, air is sucked, compressed in the cylinder and then delivered at a high pressure. But sometimes, the air is required at a high pressure.

In such cases, either we employ a large pressure ratio (in single cylinder) all compress the air in two or more cylinders in series. It has been experience that if we employ single stage compression for or producing high pressure air (say 8 to 10 bar) **It suffers the following drawbacks:**

1. The size of the cylinder will be too large
2. Due to compression there is a rise in temperature of air. It is difficult to reject heat from the air in the small time available during compression.
3. Sometimes the temperature of air at end of compression is too high. it may heat of the cylinder head or burn the lubricating oil.

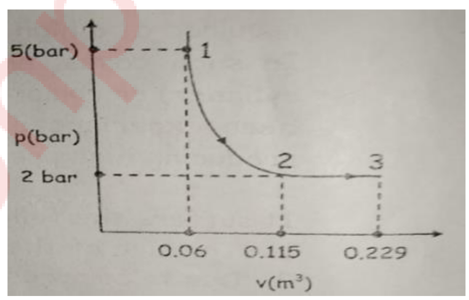
In order to overcome the above mentioned difficulties two or more cylinders are provided in series which intercooling arrangement between them. Such an arrangement is known as multistage compassion.

Advantages of multistage compression:

1. the work done per kg of air is reduced in multistage compression with intercooler as compared to single stage compression for the same delivery pressure.
2. It improves the volumetric efficiency for the given pressure ratio.
3. The sizes of two cylinders (i.e. high pressure and low pressure) may be adjusted to suit the volume and pressure of the air.
4. It reduces the leakage loss considerably.
5. It gives more uniform torque and hence smaller size flywheel is required.
6. it provides effective lubrication because of lower temperature range.
7. It reduces the cost of compressor.

Q.5(a) 0.06 m^3 of air at 5 bar and 200° C expands isentropic until the pressure becomes 2 bar. It is then heated at constant pressure until the enthalpy increase during this process is 80 KJ. Calculate the work done in each process and the total work done. [10]

Ans:



T_1 = 200° C = 200+273 **= 473 K** mass flow rate of air (m) using ideal gas equation.

$P_1v_1 = mRT_1$

$500 \times 0.06 = m \times 0.287 \times 473$

m = 0.221 kg

Process (1 – 2), isentropic expansion :

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_2 = T_1 \times \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$= 473 \times \left(\frac{2}{5}\right)^{\frac{1.4-1}{1.4}}$$

$T_2 = 364$ K

Also $\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{v_1}{v_2}\right)^{\gamma-1}$

$$\left(\frac{2}{5}\right)^{\frac{1.4-1}{1.4}} = \left(\frac{0.06}{v_2}\right)^{1.4-1}$$

$V_2 = 0.115$ m^3

Process 2 – 3: constant pressure (p = c)

During constant pressure, enthalpy (h) of air is increased by 80 kj

$\Delta H = 80 = m.\ C_p\ (T_3 - T_2)$

$80 = 0.221 \times 1.005\ (T_3 - 364)$

$T_3 = 724.19$ K

$$\frac{V_2}{T_2} = \frac{V_3}{T_3}$$

$$V_3 = \frac{T_3}{T_2} \times V_2 = \frac{724.19}{364} \times 0.115$$

$V_3 = 0.229$ m^3

Work done process 1-2 isentropic expansion

$$(W_{1\text{-}2}) = \frac{P_1V_1 - P_2V_2}{\gamma - 1} = \left[\frac{(500\times0.06)-(200\times0.115)}{1.4-1}\right]$$

$W_{1\text{-}2} = 17.5$ KJ

Work done in process 2-3: constant pressure (P =C)

$W_{2\text{-}3} = P.\ dv = P\ (V_3 - V_2)$

$= 200(0.229 - 0.115)$

$\mathbf{W_{2.3} = 22.8\ kJ}$

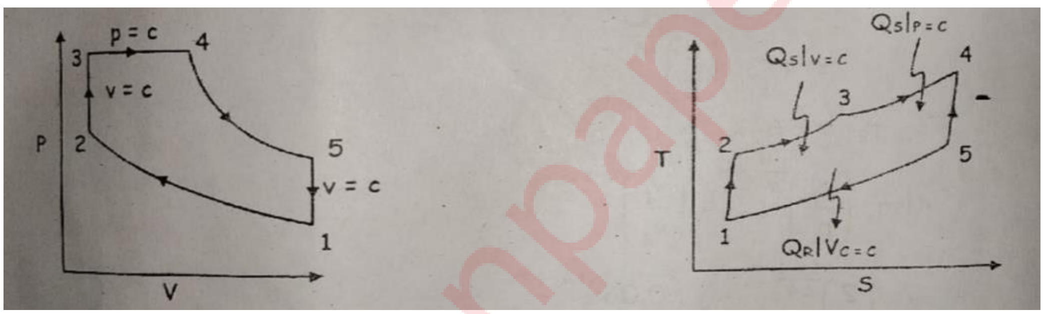
Net work done (W) = $W_{1\text{-}2} + W_{2\text{-}3}$

$= 1705 + 22.8$

(W) = 40.3 KJ

Q.5(B)In an I.C. engine operating on the dual cycle, the temperature of working fluid (air) at the beginning of the compression is 27° C. The ratio of the maximum and minimum pressure of the cycle is 70 and compression ratio 15. The amount of heat added at constant volume and constant pressure are equal, compute the air standard thermal efficiency of the cycle. [10]

Ans:



Air standard dual cycle:

Compression ratio (r_c) (r_c)

$= \frac{V_1}{V_2} = \mathbf{15}$

T_1 = 27° C = 27+273 **= 300 K**

γ **= 1.4**

$Q_{S(V=C)} = Q_{S(P=C)}$

Ratio of the maximum and minimum pressure of the cycle.

$\frac{P_3}{p_1}\ OR\ \frac{P_4}{P_1} = \mathbf{70\ Bar}$

Process 1 – 2: isentropic compression :

$\left(\frac{V_1}{V_2}\right)^{\gamma-1} = \left(\frac{T_2}{T_1}\right)$

$(15)^{1.4-1} = \frac{T_2}{300}$

T_2 = 886.25 K

Similarly,

$\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{v_1}{v_2}\right)^{\gamma-1}$

$\frac{P_2}{P_1} = (15)^{1.4} = \mathbf{44.31}$ ∴ $\boldsymbol{P_2 = 44.31\, P_1}$

Process 2 -3 : $\frac{P_2}{T_2} = \frac{P_3}{T_3}$ constant volume process (v=c)

∴ $\frac{p_3}{p_2} = \frac{T_3}{T_2}$ ∴ $\frac{P_3}{44.31 P_1} = \frac{T_3}{T_2}$ From (1)

But, $\frac{p_3}{p_1} = 70$

∴ $\frac{T_3}{T_2} = \frac{70}{44.31} = 1.58$

$T_3 = 1.58 \times T_2 = 1.58 \times 886.25 = 1400.275$ K

T_3 = 1400.275 K

Now heat addition at constant volume is equal to heat addition at constant pressure,

∴ $Q_{S(v=c)} = Q_{S(p=c)}$

$C_v (T_3 - T_2) = C_P = (T_4 - T_3)$

$0.718(1400.275 - 886.25) = 1.005 (T_4 - 1400.275)$

T_4 = 1767.51 K

Process 3 – 4 : $\frac{V_3}{V_4} = \frac{T_3}{T_4} = \frac{1400.275}{1767.51} = 0.79$

Cut off ratio (ρ) = $\frac{V_4}{V_3} = 1.27$

Process 5 – 1 : r_e – expansion ratio = $\frac{V_5}{V_4}$

$$r_e = \frac{V_5}{V_4} = \frac{R_C}{\rho} = \frac{15}{1.27} = \mathbf{11.81}$$

Now $\left(\frac{V_4}{V_5}\right)^{\gamma-1} = \left(\frac{T_5}{T_4}\right)$

$T_5 = \frac{1767.51}{(11.81)^{0.4}} = 658.36$

T_5 = 658.36 K

Efficiency of the cycle (η dual)

$$= \frac{W_{net}}{heat\ supplied\ (Q_S)}$$

$$= \frac{Q_S - Q_R}{Q_S}$$

$$= \frac{[C_V(T_3-T_2)+C_P(T_4-T_3)-C_V(T_5-T_1)]}{[C_V(T_3-T_2)+C_P(T_4-T_3)]}$$

$$\eta_{Dual} = 1 - \frac{T_5 - T_1}{(T_3-T_2)+\gamma(T_4-T_3)}$$

$$= 1 - \frac{658.36-300}{(1400.275-886.25)+1.4(1767.51-1400275)}$$

η_{Dual} = 0.6514 = 65.14 %

Q.6(a)Derive an expression of air standard efficiency for diesel cycle. (10)

Ans: The limitation of compression ratio in The SI engine can be overcome by compressing air alone, instead of air fuel mixture, and then injecting the fuel into the cylinder in the spray form when combustion is desired. The CI engine, first proposed by Rudolph Diesel (1890) very similar to SI engine differing mainly in the method of initiating combustion. Diesel cycle operate at much higher compression ratio, typically between 12 and 24.

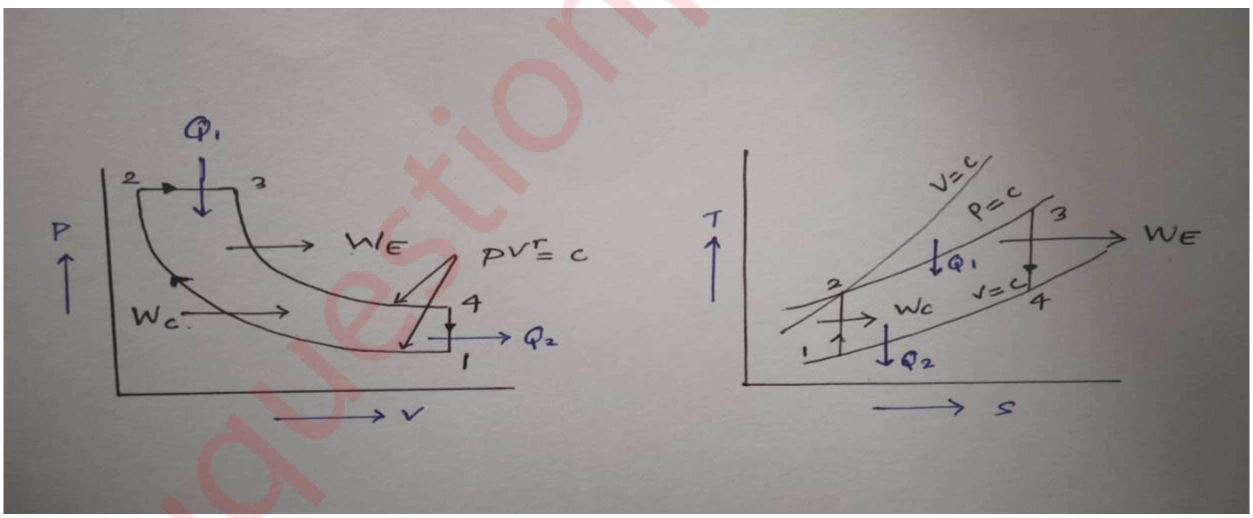
The air standard diesel cycle Consist of following processes.

Process 1-2: Reversible adiabatic compression of air when piston moves upward.

Process 2-3: Reversible constant pressure het addition.

Process 3-4: Reversible adiabatic expansion.

Process 4-1: Reversible constant volume heat rejection.



Air standard efficiency for diesel cycle:

1. **Compression ratio:** It is defined as the ratio of volume at the beginning of the compression to the volume at the end of compression.

$r_k = \text{volume at the begining of the compression} = v_1$

$$\text{volume at the end of compression} \qquad v_2$$

2. **Expansion Ratio:** It is defined as the ratio of volume after expansion to the volume before expansion.

$$r_e = \frac{\text{Volume after expannsion}}{\text{volume before expnsion}} = \frac{v_4}{v_3}$$

3. **Cut of ratio:** It is defined as the ratio between end and start volume for the combustion phase.

$$r_c = \frac{v_3}{v_2}$$

$$r_k = r_c . r_e$$

The thermal efficiency of diesel cycle can be written as.

$$n_{Diesel} = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1}$$

Heat supplied, $Q_{(2-3)} = Q_1 = m .C_P (T_3 - T_2)$

Heat rejected, $Q_{(4-1)} = Q_2 = m .C_P (T_4 - T_1)$

Hence efficiency can be given as,

$$n_{Diesel} = 1 - \frac{Q_2}{Q_1} = 1 - \frac{m.c_v(T_4 - T_1)}{m.c_p(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{\gamma(T_3 - T_2)} \quad \text{……... (1)}$$

Process 2-3: $\frac{T_2}{T_3} = \frac{p_2 . v_2}{p_3 . v_3} = \frac{v_2}{v_3} = \frac{1}{r_c} \; (p_2 = p_3)$ **OR** $T_2 = T_3 . (\frac{1}{r_c})$

Process 1-2: $\frac{T_1}{T_2} = (\frac{v_2}{v_1})^{\gamma - 1}$

$$T_1 = T_2 . \frac{1}{r_k^{\gamma-1}} = \frac{T_3}{r_c} . \frac{1}{r_k^{\gamma-1}}$$

Process 3-4: $\frac{T_4}{T_3} = (\frac{v_3}{v_4})^{\gamma - 1} = (\frac{1}{r_e})^{\gamma-1}$ **OR** $T_4 = T_3 (\frac{r_c}{r_k})^{\gamma-1}$

Substituting the value of T_1, T_2 and T_4 in the equation (1)

$$n_{Diesel} = 1 - \{T_3 (\frac{r_c^{\gamma-1}}{r_k^{\gamma-1}}) - \frac{T_3}{r_c} (\frac{1}{r_k^{\gamma-1}})\}$$

1

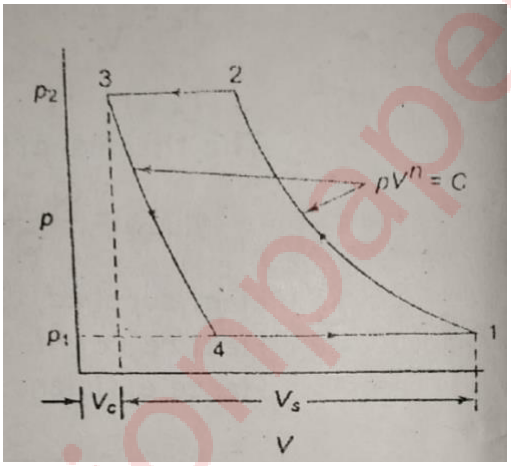
$\{□(T_3 - T_3\,(r_c))\}$

$$n_{Diesel} = 1 - \frac{1}{□}.\left(\frac{1}{r_k^{□-1}}\right).\frac{r_c^{□}-1}{r_c-1}$$

Efficiency for the **diesel cycle** is **less** than that of **Otto cycle** for same compression ratio.

Q.6(b) A single stage reciprocating air compressor has a swept volume of 2000cm° and runs at 800 rpm. It operates on a pressure ratio of 8 with the clearance of 5% of the swept volume. Assume NTP room conditions and at inlet (p =101.3 kPa, T = 15° C) and polytrophic compression and expansion with n= 1.25. calculate i.) Indicated power ii.) Volumetric efficiency iii.) Mass and flow rate iv.) Free air delivery v.) Isothermal efficiency and vi.) The actual power needed to drive the air compressor if the mechanical efficiency is 85%. [10]

Ans:



p_1 = 101.3 KPa, $p_2 = 81p_1$ = 810.4 kpa

T_1 = 288 K, V_S = 2000 cm^3

$V_3 = V_C = 0.05\ V_S$ = 100 cm^3

$V_1 = V_C + V_S$ = 2100 cm^3

$p_3 v_3^n = p_4 v_4^n$

$\therefore V_4 = \left(\frac{p_3}{p_4}\right)^{\frac{1}{n}}, V_3 = (8)^{\frac{1}{1.25}} \times 100$

= 528 cm^3

$V_1 - V_4$ = 2100- 528 = 1572 cm^3

$W = \frac{n}{n-1} p_1(v_1 - v_4)\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$

$= \frac{1.25}{0.25} \times 101.3 \times 10^3 \times 1572 \times 10^{-6}\left[(8)^{\frac{0.25}{1.25}} - 1\right] = 411\ J$

(a) Indicated power = $\frac{411 \times 800 \times 10^{-3}}{60} = \mathbf{5.47\ kw}$

(b) Volumetric efficiency = $\frac{1572}{2000} \times 100 = \mathbf{78.6}\ \%$

(c) Mass of air compressed per cycle

$M = \frac{pV}{RT} = \frac{101.3 \times 10^3 \times 1572 \times 10^{-6}}{287 \times 288} = \mathbf{1.93 \times 10^{-3} kg}$

(d) FAD = free air delivery = $1572 \times 10^{-6} \times 800 = \mathbf{1.26\ m^3/min}$

(e) $W_r = p_1 (V_1 - V_4) \ln \frac{P_2}{P_1} = 101.3 \times 10^3 \times 1572 \times 10^{-6} \ln 8$

= 331 J

$\eta_{ISOTHERMAL} = \frac{0.331 \times 800}{5.47 \times 60} = \mathbf{80.7}\%$

(f) Input power = $\frac{5.47}{0.85} = \mathbf{6.44\ kw}$