SCHEME OF EVALUATION

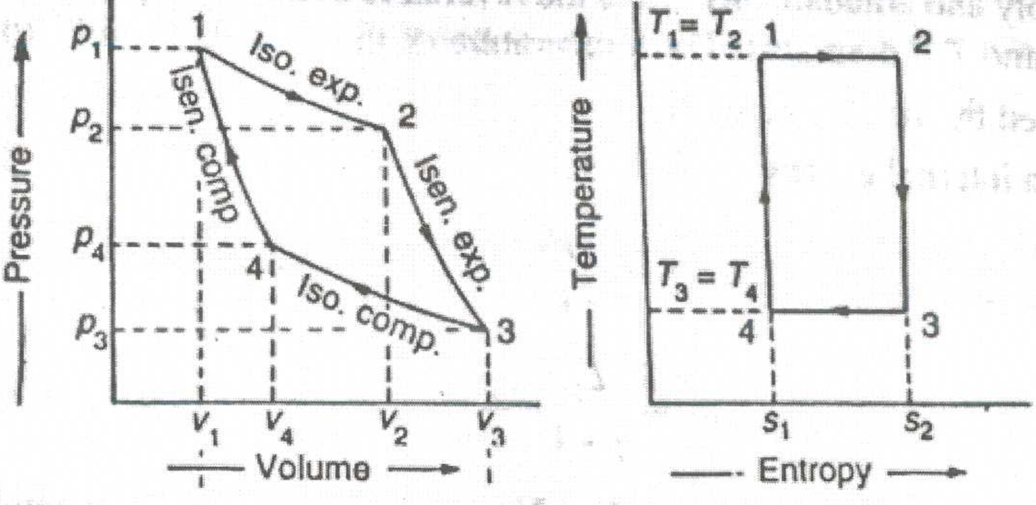
REVISION - 2015 COURSE CODE - 4024

COURSE TITLE :- THERMAL ENGINEERING

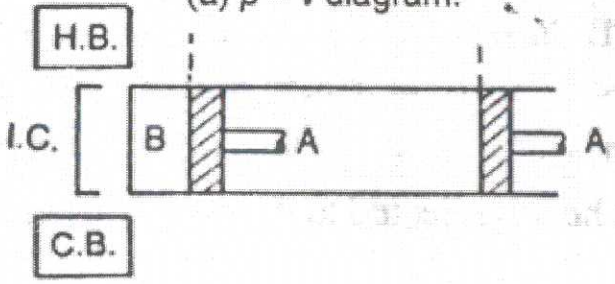
Q No.	Scoring Indicators	Split-up	Sub-
I.1)	Definite area or a space where some thermodynamic process is taking place 2		
I.2)	Ratio of WD to heat supplied or, $\eta = \frac{workdone}{heat\ supplied}$ 2		
I.3)	Steam at a temperature higher that of saturation temperature at that Pr. 2		
I.4)	Conduction , Convection and Radiation 2		
I.5)	Two stroke engine , Four stroke engine 2		
II. 1)	a). Charles law :- The volume of a given mass of perfect gas varies directly as its abs.temp when the abs.pr. remains constant mathematically, v α T or $\frac{v}{T} = constant$ or, $\frac{v1}{T1} = \frac{v2}{T2} = \frac{v3}{T3} = \cdots = constant$ 3 where , 1,2,3,…..refer to different set of conditions OR, All perfect gases change in volume by 1/273th of its original volume at $0^{o}C$ for every change of $1^{o}C$ in temp., when the pr. remains constant. b). Joules law :- The change of internal energy of a perfect gas is directly proportional to the change of temp. 3 $dE \propto dT$ or, $dE \propto mcdT = mc(T_2 - T_1)$ Where, m = mass of the gas & c = constant of proportionality known as specific heat	6	

2

II. 2)



(a) $p-v$ diagram. (b) $T-S$ diagram.



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HINTS :

1-process 1-2 : Isothermal expansion

Heat supplied = work done by the air

$$Q_{1\text{-}2} = p_1 v_1 \log_e\left(\frac{v2}{v1}\right) = mRT_1 \log_e\left(\frac{v2}{v1}\right) = 2.3\ mRT_1 \log\left(\frac{v2}{v1}\right) = 2.3\ mRT_1 \log \boldsymbol{r}$$

Where, $r = \frac{v2}{v1} =$ Expansion ratio

2- process 2-3 : Isentropic exp :

$Q_{1\text{-}2} = 0$

3. -process 3-4 : Isothermal compression :

$$\text{Heat rejected} = Q_{3\text{-}4} = p_3 v_3 \log_e\left(\frac{v3}{v4}\right) = mRT_3 \log_e\left(\frac{v3}{v4}\right) = 2.3\ mRT_3 \log\left(\frac{v3}{v4}\right)$$

$$= 2.3\ mRT_3 \log \boldsymbol{r}$$

4. Process 4-1 : Isentropic compression

$Q_{4\text{-}1} = 0$

Workdone = heat supplied - heat rejected

$= 2.3\ mRT_1 \log \boldsymbol{r} - 2.3\ mRT_3 \log \boldsymbol{r} = 2.3\ mR\ log\ r\ [\ T1 - T3\ ]$

Efficency $= \eta = \dfrac{Workdone}{Heat\ supplied}$ 4

$$= \frac{2.3\ mR\ log\ r\ [\ T1 - T3\ ]}{2.3\ mRT1\ \log r}$$ 6

$$= \frac{T1 - T3}{T1}$$

II. 3) IP – The power actually developed by the engine cylinder. It is based on the information obtained from the indicator diagram of the engine. It can also be found by conducting morse test. 2

SFC – is the ratio of TFC to BP

SFC = $\frac{TFC}{BP}$ 2 6

Indicated Thermal efficiency – is the ratio of IP to Heat supplied by fuel.

$\eta = \dfrac{Indicated\ power}{Heat\ supplied} = \dfrac{IP}{TFC\ x\ CV}$ 2

where CV = calorific value of fuel

4

II. 4) If Q_i is the incident radiant energy falling on a body as shown in fig, some part of it

it will be absorbed say Q_a, some part of it reflected from the surface say Q_r and some will be transmitted through the body say Q_t, then energy balance yields

$$Q_i = Q_a + Q_r + Q_t$$

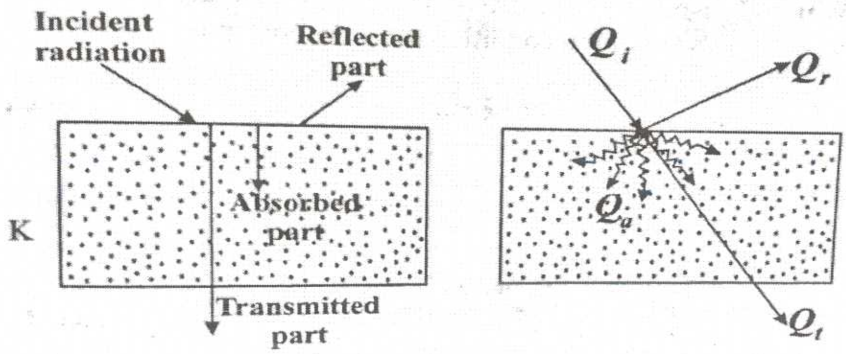
Dividing the above equation by Q_i, we have.

$$\frac{Q_a}{Q_i} + \frac{Q_r}{Q_i} + \frac{Q_t}{Q_i} = 1$$

If α is the fraction of total energy absorbed, ρ is the fraction of total energy reflected and τ is the fraction of total energy transmitted, then we have.

$$\alpha + \rho + \tau = 1$$

3



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Absorption, reflection and transmission of incident radiation

The fraction of incident radiation absorbed by a body is called *absorptivity* (α), the fraction reflected is called *reflectivity* (ρ) and the fraction transmitted through the body is the *transmissivity* (τ). These fractions should add up to unity.

The above definitions can be mathematically written as follows.

3

$$\textit{Absorptivity}, \alpha = \frac{\text{Amount of radiation absorbed}, Q_a}{\text{Total incident radiation } Q_i}$$

$$\textit{Reflectivity}, \rho = \frac{\text{Amount of radiation reflected}, Q_r}{\text{Total incident radiation } Q_i}$$

$$\textit{Transmissivity}, \tau = \frac{\text{Amount of radiation transmited}, Q_t}{\text{Total incident radiation } Q_i}$$

From the law of conservation of energy it follows that $\alpha + \rho + \tau = 1$

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II. 5)

Solution : Since temperature remains constant, Boyle's law applies and therefore,

$$p_1 v_1 = p_2 v_2 \, ; \quad v_2 = \frac{p_1 v_1}{p_2}$$

$$\therefore \quad v_2 = \frac{0.98 \times 0.45}{0.6} = 0.735 \text{ m}^3/\text{kg}$$

6

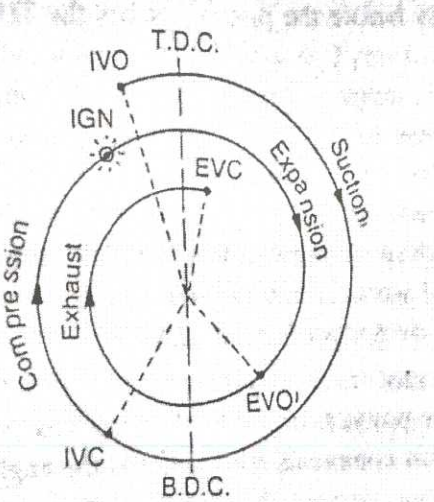
Density is the reciprocal of specific volume and therefore,

$$\rho_2 = \frac{1}{v_2} = \frac{1}{0.735} = 1.36 \text{ kg/m}^3$$

II. 6)

26.11. ... Stroke Cycle Petrol Engine

In the valve timing diagram, as shown in Fig. 26.6, we see that the inlet valve opens before the piston reaches *TDC* ; or in other words, while the piston is still moving up before the beginning of the suction stroke. Now the piston reaches the *TDC* and the suction stroke starts. The piston reaches the *BDC* and then starts moving up. The inlet valve closes, when the crank has moved a little beyond the *BDC*. This is done as the incoming charge continues to flow into the cylinder although the piston is moving upwards from *BDC*. Now the charge is compressed (with both valves closed) and then ignited with the help of a spark plug before the end of compression stroke. This is done as the charge



TDC : Top dead centre

BDC : Bottom dead centre

IVO : Inlet valve opens (10°-20° before *TDC*)

IVC : Inlet valve closes (30°- 40° after *BDC*)

IGN : Ignition (20°-30° before *TDC*) 2

EVO : Exit valve opens (30°-50° before *BDC*) 6

EVC : Exit valve closes (10°-15° after *TDC*)

Valve timing diagram for a four stroke cycle petrol engine.

Expln

requires some time to ignite. By the time, the piston reaches *TDC*, the burnt gases (under high pressure and temperature) push the piston downwards with full force and the expansion or working stroke takes place. Now the exhaust valve opens before the piston again reaches *BDC* and the burnt gases start leaving the engine cylinder. Now the piston reaches *BDC* and then starts moving up, thus performing the exhaust stroke. The inlet valve opens before the piston reaches *TDC* to start suction stroke. This is done as the fresh incoming charge helps in pushing out the burnt gases. Now the piston again reaches *TDC*, and the suction stroke starts. The exit valve closes after the crank has moved a little beyond the *TDC*. This is done as the burnt gases continue to leave the engine cylinder although the piston is moving downwards. It may be noted that for a small fraction of a crank revolution, both the inlet and outlet valves are open. This is known as valve overlap. 4

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II. 7)	Let us consider a four cylinder petrol engine coupled with hydraulic dynamomete measure brake power. Let IP_1, IP_2, IP_3, IP_4 are indicated power of cylinders 1,2,3, and 4 BP = Total brake power of all 4 cylinders BP_1 = Total brake power three cylinders with cylinder 1 cut off BP_2 = Total brake power three cylinders with cylinder 2 cut off BP_3 = Total brake power three cylinders with cylinder 3 cut off BP_4 = Total brake power three cylinders with cylinder 4 cut off FP_1, FP_2, FP_3, FP_4 frictional power each cylinders. If all cylinders are working $BP = (IP_1 - FP_1) + (IP_2 - FP_2) + (IP_3 - FP_3) + (IP_4 - FP_4)$ $= (IP_1 + IP_2 + IP_3 + IP_4) - (FP_1 + FP_2 + FP_3 + FP_4)$----(1) Now if one cylinder is cut out, the IP of that cylinder is cut out. But, FP of this cylinder sti exists. Therefore if the first cylinder is cutout, brake power of the other three cylinders BP_1 given by $BP_1 = (IP_2 + IP_3 + IP_4) - (FP_1 + FP_2 + FP_3 + FP_4)$ --(2) Substracting equation (2) from equation (1), we get $BP - BP_1 = IP_1$ (indicated power of first cylinder) Similarly by cutting cylinders 2,3 and 4 in turn, IP_2, IP_3 and IP_4 can be calculated as below. $BP - BP_2 = IP_2$ (indicated power of second cylinder) $BP - BP_3 = IP_3$ (indicated power of third cylinder) $BP - BP_4 = IP_4$ (indicated power of fourth cylinder) $\therefore$ Total indicated power developed, $IP = IP_1 + IP_2 + IP_3 + IP_4$	6	6

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III
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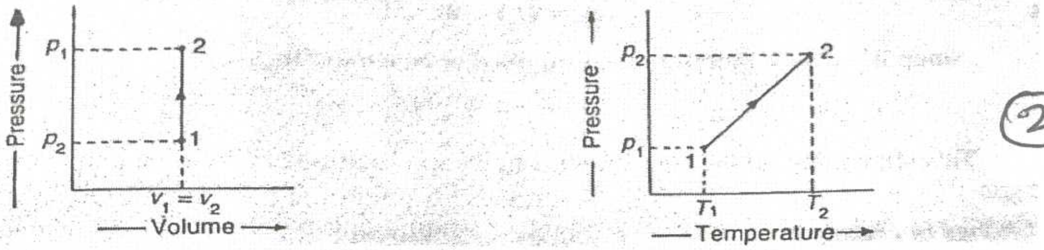
PART - C

The process is shown on the pressure-volume (p-v) and pressure-temperature (p-T) diagrams in Fig. 3.4 (a) and (b) respectively.



(a) p-v diagram. (b) p-T diagram.

Constant volume process.

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Now let us derive the following relations for the reversible constant volume process.

1. *Pressure-volume-temperature (p-v-T) relationship*

We know that the general gas equation is

$$\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2} \qquad \ldots$$

1

Since the gas is heated at constant volume, therefore $v_1 = v_2$.

$$\therefore \qquad \frac{p_1}{T_1} = \frac{p_2}{T_2} \quad \text{or} \quad \frac{p}{T} = \text{Constant} \qquad \ldots\text{[From equation}$$

Thus the constant volume process is governed by Gay-Lussac law.

2. *Workdone by the gas*

We know that $\delta W = p\, dv$

On integrating from state 1 to state 2,

1

$$\int_1^2 \delta W = \int_1^2 p\, dv = p \int_1^2 dv$$

or $$W_{1-2} = p\,(v_2 - v_1) = 0 \qquad \ldots(\because v_1 =$$

3. *Change in internal energy*

We know that change in internal energy,

$$dU = m\, c_v\, dT \qquad \ldots\text{(Joule's}$$

On integrating from state 1 to state 2,

$$\int_1^2 dU = m\, c_v \int_1^2 dT$$

1

or $$U_2 - U_1 = m\, c_v\, (T_2 - T_1)$$

4. *Heat supplied or heat transfer*

We know that $\delta Q = dU + \delta W$

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On integrating from state 1 to state 2,

$$\int_1^2 \delta Q = \int_1^2 dU + \int_1^2 \delta W$$

or $$Q_{1-2} = (U_2 - U_1) + W_{1-2}$$ (2)

Since $W_{1-2} = 0$, therefore heat supplied or heat transfer,

$$Q_{1-2} = U_2 - U_1 = m\,c_v\,(T_2 - T_1)$$

This shows that all the heat supplied to the gas is utilised in increasing the internal energy the gas.

(9)

5. *Change in enthalpy*

We know that the change in enthalpy,

$$dH = dU + d\,(pv)$$

On integrating from state 1 to state 2,

$$\int_1^2 dH = \int_1^2 dU + \int_1^2 d\,(p\,v)$$ (2)

or
$$H_2 - H_1 = (U_2 - U_1) + (p_2 v_2 - p_1 v_1)$$

$$= m\,c_v\,(T_2 - T_1) + m\,R\,(T_2 - T_1)$$

$$\ldots (\because p_1 v_1 = m\,R\,T_1\text{; and } p_2 v_2 = m\,R$$

$$= m\,(T_2 - T_1)\,(c_v + R) = m\,c_p\,(T_2 - T_1) \qquad \ldots (\because c_p - c_v =$$

III.

b) Given, $P_1 = 2\text{bar} = 2\times10^5\ N/m^2$, $T_1 = 77°C = 350K$

$P_2 = 7\text{ bar} = 7\times10^5\ N/m^2$, $V_1 = 0.3m^3$.

$C_p = 1.005\ kJ/kgk$., $C_v = 0.712\ kJ/kgk$.

$R = 0.287\ kJ/kgk. = 287\ J/kg\,k$

Final Temp $= T_2$.

We have $\frac{P_1}{T_1} = \frac{P_2}{T_2}$, $T_2 = \frac{P_2 T_1}{P_1} = \frac{7\times10^5\times350}{2\times10^5}$ (1)

$= 1225\,K = 952°C$

Change in IE $= du = U_2 - U_1 = mC_v(T_2 - T_1)$

mass of gas $= m$.

we have $P_1V_1 = mRT_1$ & $m = \frac{P_1V_1}{RT_1}$

$= \frac{2\times10^5\times0.3}{287\times350} = 0.597\ kg$ (2)

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IV.

a)

A process, in which the temperature of the working substance remains constant during its expansion or compression, is called *constant temperature process* or *isothermal process*. This will happen when the working substance remains in a perfect thermal contact with the surroundings, so that the heat 'sucked in' or 'squeezed out' is compensated exactly for the work done by the gas or on the gas respectively. It is thus obvious that in an isothermal process :

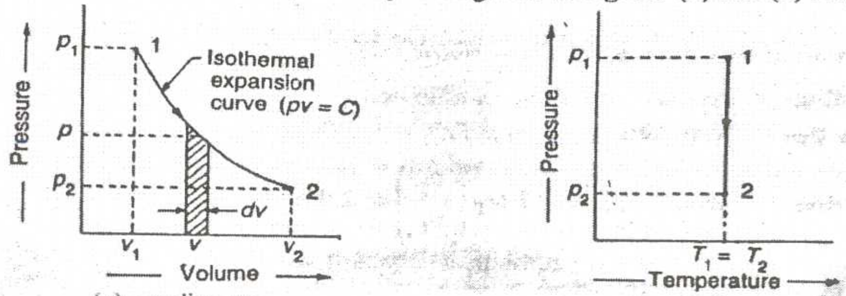
1. there is no change in temperature,
2. there is no change in internal energy, and
3. there is no change in enthalpy.

Now consider m kg of a certain gas being heated at constant temperature from an initial state 1 to final state 2.

Let p_1, v_1 and T_1 = Pressure, volume and temperature at the initial state 1, and

p_2, v_2 and T_2 = Pressure, volume and temperature at the final state 2.

The process is shown on the p-v and p-T diagrams in Fig. 3.9 (a) and (b) respectively. 2



(a) p-v diagram. (b) p-T diagram.

Constant temperature (Isothermal) process.

Now let us derive the following relations for the reversible constant temperature process or isothermal process.

1. *Pressure-volume-temperature (p-v-T) relationship*

We know that the general gas equation is

$$\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2} \quad \ldots (i)$$ 1

Since the gas is heated at constant temperature, therefore $T_1 = T_2$.

$$\therefore \quad p_1 v_1 = p_2 v_2 \quad \text{or} \quad p\,v = \text{Constant} \quad \ldots \text{[From equation } (i)]$$

Thus, the constant temperature process or isothermal process is governed by Boyle's law.

2. *Workdone by the gas*

We know that $\delta W = p\,dv$

On integrating from state 1 to state 2,

$$\int_1^2 \delta W = \int_1^2 p\,dv$$

or

$$W_{1-2} = \int_1^2 p\,dv \quad \ldots (ii)$$

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Since the expansion of the gas is isothermal, *i.e.* $p\,v = C$, therefore

$$p\,v = p_1\,v_1 \quad \text{or} \quad p = \frac{p_1\,v_1}{v}$$

(3)

Substituting this value of p in equation (*ii*), we have

$$W_{1-2} = \int_{v_1}^{v_2} \frac{p_1\,v_1}{v}\,dv = p_1\,v_1 \int_{v_1}^{v_2} \frac{dv}{v}$$

$$= p_1\,v_1\left[\log_e v\right]_{v_1}^{v_2} = p_1\,v_1\,\log_e\left(\frac{v_2}{v_1}\right)$$

The above equation may be expressed in terms of corresponding logarithm to the

$$W_{1-2} = 2.3\,p_1\,v_1\,\log\left(\frac{v_2}{v_1}\right) = 2.3\,p_1\,v_1\,\log r$$

where $r = \dfrac{v_2}{v_1}$, and is known as *expansion ratio.*

The equation (*iv*) may also be written as follows :

We know that $p_1\,v_1 = p_2\,v_2 = m\,R\,T$

$\therefore$ Work done, $W_{1-2} = 2.3\,m\,R\,T\log\left(\dfrac{v_2}{v_1}\right) = 2.3\,m\,R\,T\log r$

Since $p_1\,v_2 = p_2\,v_2$, therefore $\dfrac{v_2}{v_1} = \dfrac{p_1}{p_2}$

$\therefore$ Work done, $W_{1-2} = 2.3\,p_1\,v_1\,\log\left(\dfrac{p_1}{p_2}\right)$ (9)

Notes : (*a*) Expansion ratio, $r = \dfrac{\text{Volume at the end of expansion}}{\text{Volume at the beginning of expansion}}$

(*b*) Compression ratio, $r = \dfrac{\text{Volume at the beginning of compression}}{\text{Volume at the end of compression}}$

3. *Change in internal energy*

We know that change in internal energy,

$$dU = U_2 - U_1 = m\,c_v\,(T_2 - T_1)$$ (2)

Since it is a constant temperature process, *i.e.* $T_1 = T_2$, therefore

$$dU = U_2 - U_1 = 0 \quad \text{or} \quad U_1 = U_2$$

4. *Heat supplied or heat transferred*

We know that heat supplied or heat transferred from state 1 to state 2,

$$Q_{1-2} = dU + W_{1-2} = W_{1-2}$$ (2)

This shows that all the heat supplied to the gas is equal to the workdone by

By the gas.

5. *Change in enthalpy*

We know that change in enthalpy,

$$dH = H_2 - H_1 = m\,c_p\,(T_2 - T_1)$$ (1)

Since, it is constant temp. process, i.e, $T_1 = T_2$, Therefore, $dH = H_1 = H_2 = 0$, or $H_1 = H_2$

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IV. b)

Given, $\vartheta_1 = 0.1\,m^3$, $P_1 = 1.5\,bar = 1.5\times10^5\,\frac{N}{m^2}$

$\vartheta_2 = 0.5\,m^3$,

1. Final pr: of gas $= P_2$

We have $\frac{P_1V_1}{T_1} = \frac{P_2\vartheta_2}{T_2}$ (1)

$\because T = C$, $P_1\vartheta_1 = P_2\vartheta_2$ $\therefore P_2 = \frac{P_1V_1}{\vartheta_2}$

$= \frac{1.5\times10^5\times0.1}{0.5} = 30000\,\frac{N}{m^2}$

$= 0.3\,bar$

2. Heat supplied during the process (6)

$\because$ process is isothermal Heat supplied $=$ W.D.

i.e. $W_{1-2} = 2.3P_1\vartheta_1\log r$ (2)

and $r = \frac{\vartheta_2}{\vartheta_1} =$ Expn. ratio $= \frac{0.5}{0.1} = 5$

$Q_{1-2} = W_{1-2} = 2.3\times1.5\times10^5\times0.1\log5$

$= 24114.46 = 24115\,J$ (2)

$= 24.115\,kJ$.

3. W.D Since WD = Heat supplied

$W.D = 24.115\,kJ$ (1)

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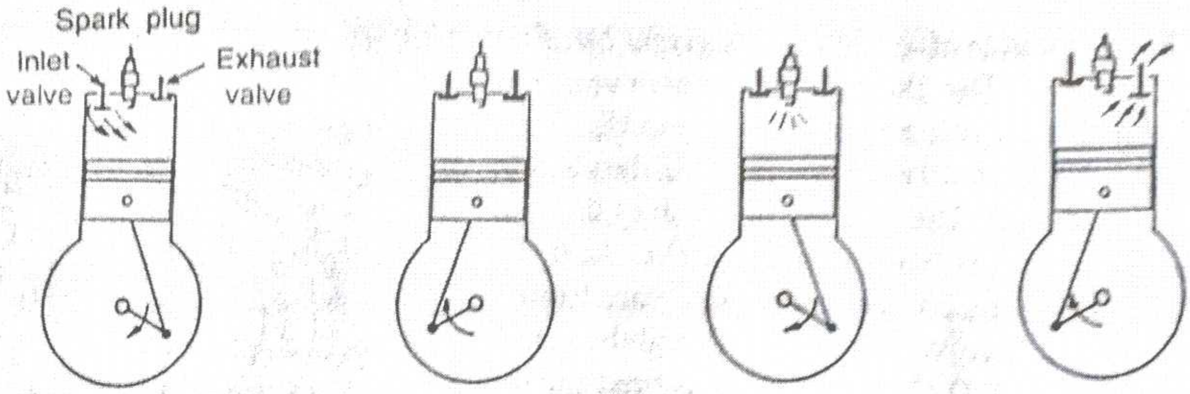
V.

a)



(*a*) Suction or charging stroke. (*b*) Compression stroke. (*c*) Expansion or working stroke. (*d*) Exhaust stroke.

OR, SIMILLAR Figures

HINTS - For EXPLANATION of working

Suction stroke

Compression stroke

Expansion stroke

Exhaust stroke

fig / 5

Expn / 4

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V. b)

Solution. Given : $W = 0.6 \times$ Heat rejected $= 0.6\, Q_{3-4}$; $T_1 - T_3 = 200°$ C

Thermal efficiency

We know that the thermal efficiency,

$$\eta = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{\text{Work done}}{\text{Work done + Heat rejected}}$$ 2

$$= \frac{0.6\, Q_{3-4}}{0.6\, Q_{3-4} + Q_{3-4}} = \frac{0.6}{1.6} = 0.375 \text{ or } 37.5\%$$

Source and sink temperatures

Let T_1 = Source temperature, and

T_3 = Sink temperature.

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We know that thermal efficiency (η),

$$0.375 = \frac{T_1 - T_3}{T_1} = \frac{200}{T_1}$$

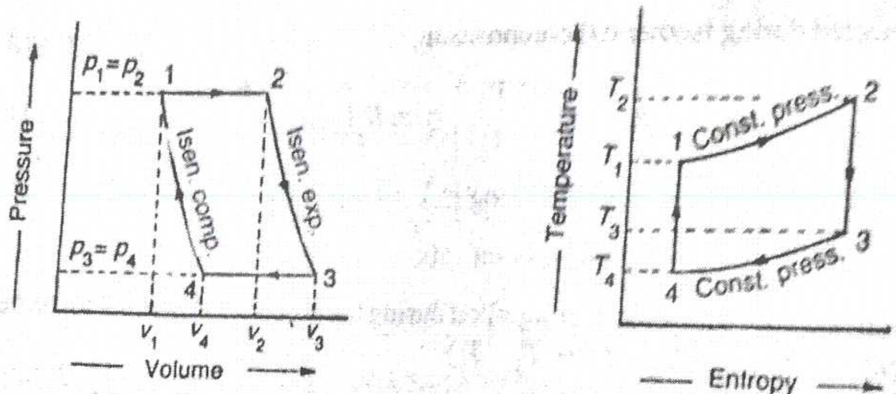
$\therefore$ $T_1 = 200/0.375 = 533.3 \text{ K} = 260.3° \text{ C}$ **Ans.** 2

and $T_3 = T_1 - 200 = 260.3 - 200 = 60.3° \text{ C}$ **Ans.** 2

VI. a)

Joule's Cycle

It consists of two constant pressure and two reversible adiabatic or isentropic processes shown on *p-v* and *T-S* diagrams in Fig. 6.8 (*a*) and (*b*).



(*a*) *p-v* diagram. (*b*) *T-S* diagram.

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Fig. 6.8. Joule's cycle.

Now, let us consider the four stages of the Joule's cycle. Let the engine cylinder contain *m* of air at its original condition represented by point 1 on *p-v* and *T-S* diagram. At this point, p_1, T_1 and v_1 be the pressure, temperature and volume of the air.

1. *First stage (Constant pressure heating).* The air is heated at a constant pressure from initial temperature T_1 to a temperature T_2 represented by the curve 1-2 in Fig. 6.8 (*a*) and (*b*).

$\therefore$ Heat supplied to the air,

$$Q_{1-2} = m\, c_p\, (T_2 - T_1)$$

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2. *Second stage (Reversible adiabatic or isentropic expansion)* The air is allowed to expand reversibly and adiabatically from v_2 to v_3. The reversible adiabatic expansion is represented by the curve 2-3 in Fig. 6.8 (*a*) and (*b*). The temperature of the air falls from T_2 to T_3. In this process, no heat is absorbed or rejected by the air.

3. *Third stage (Constant pressure cooling)* The air is now cooled at constant pressure from temperature T_3 to a temperature T_4 represented by 3-4 in Fig. 6.8 (*a*) and (*b*).

∴ Heat rejected by the air,

$$Q_{3-4} = m\,c_p\,(T_3 - T_4)$$

4. *Fourth stage (Reversible adiabatic or isentropic compression)*. The air is now compressed reversibly and adiabatically from v_4 to v_1. The reversible adiabatic compression is represented by the curve 4-1 in Fig. 6.8 (*a*) and (*b*). The temperature of the air increases from T_4 to T_1. Again, no heat is absorbed or rejected by the air.

We see, from above, that there is no interchange of heat during the two reversible adiabatic processes. The only interchange of heat takes place during constant pressure processes.

$$\therefore \text{Work done} = \text{Heat supplied} - \text{Heat rejected}$$

$$= m\,c_p\,(T_2 - T_1) - m\,c_p\,(T_3 - T_4)$$

Derivation

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and efficiency,

$$\eta = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{m\,c_p\,(T_2 - T_1) - m\,c_p\,(T_3 - T_4)}{m\,c_p\,(T_2 - T_1)}$$

$$= 1 - \frac{T_3 - T_4}{T_2 - T_1} = 1 - \frac{T_3\left(1 - \dfrac{T_4}{T_3}\right)}{T_2\left(1 - \dfrac{T_1}{T_2}\right)} \quad \ldots (i)$$

We know that for reversible adiabatic or isentropic expansion 2-3,

$$\frac{T_3}{T_2} = \left(\frac{v_2}{v_3}\right)^{\gamma - 1} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma - 1}{\gamma}} \quad \ldots (ii)$$

and for reversible adiabatic or isentropic compression 4-1,

$$\frac{T_4}{T_1} = \left(\frac{v_1}{v_4}\right)^{\gamma - 1} = \left(\frac{p_4}{p_1}\right)^{\frac{\gamma - 1}{\gamma}} \quad \ldots (iii)$$

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From equations (*ii*) and (*iii*), we find that

$$\frac{T_3}{T_2} = \frac{T_4}{T_1} \quad \text{or} \quad \frac{T_4}{T_3} = \frac{T_1}{T_2} \quad \ldots (\because p_1 = p_2 \text{ and } p_3 = p_4)$$

Substituting the value of T_4 / T_3 in equation (*i*),

$$\eta = 1 - \frac{T_3}{T_2} = 1 - \frac{T_4}{T_1} \quad \ldots \left(\because \frac{T_3}{T_2} = \frac{T_4}{T_1}\right)$$

From equations (*ii*) and (*iii*), we find that

$$\left(\frac{v_2}{v_3}\right)^{\gamma - 1} = \left(\frac{v_1}{v_4}\right)^{\gamma - 1} \quad \ldots \left(\because \frac{p_3}{p_2} = \frac{p_4}{p_1}\right)$$

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$$\frac{v_2}{v_3} = \frac{v_1}{v_4} = \frac{1}{r} \qquad \ldots \left(\because r = \frac{v}{v} \right.$$

or

$$\therefore \quad \frac{T_3}{T_2} = \frac{T_4}{T_1} = \left(\frac{1}{r}\right)^{\gamma-1} = \frac{1}{(r)^{\gamma-1}}$$

and

$$\eta = 1 - \frac{1}{(r)^{\gamma-1}}$$

VI.

b)

The mechanical cycle of petrol and diesel engines can be modelled into a thermodynamic cycle if the following assumptions are made.

(1) The engine operates in a closed cycle. The cylinder is filled with constant amount of working medium and the same fluid is used repeatedly.

(2) The working fluid is pure air; it behaves as a perfect gas and its properties at any state can be calculated by applying the characteristic gas equation.

(3) The working fluid is homogeneous throughout at all times and no chemical reaction takes place.

(4) The compression and expansion processes are truly adiabatic; no heat is gained or lost.

(5) All the processes are internally reversible; no mechanical or friction loss.

(6) The specific heats, c_p and c_v, and their ratio γ have constant values.

(7) The actual combustion of fuel is replaced by heat supply from a heat source. The source is brought into contact with the cylinder head at appropriate time. The cylinder head is perfect conductor during that operation. Heat is thus added from external heat source and not at the expense of combustion of fuel.

(8) The exhaust process is replaced by an equivalent amount of heat rejection to a heat sink by which the working fluid is restored to the initial state.

Both the source and sink are of infinite capacity so that when the heat is transferred from or to them, their temperatures remain unchanged.

6

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VII. a)

1. *Indicated thermal efficiency*

We know that indicated power,

$$I.P. = \frac{B.P.}{\eta_m} = \frac{22}{0.85} = 25.88 \text{ kW}$$

$\therefore$ Indicated thermal efficiency,

$$\eta_i = \frac{I.P. \times 3600}{m_f \times C} = \frac{25.88 \times 3600}{6.5 \times 30\,000} = 0.48 \text{ or } 48\%$$

2

2. *Brake thermal efficiency*

We know that brake thermal efficiency,

$$\eta_b = \frac{B.P. \times 3600}{m_f \times C} = \frac{22 \times 3600}{6.5 \times 30\,000} = 0.406 \text{ or } 40.6$$

2 6

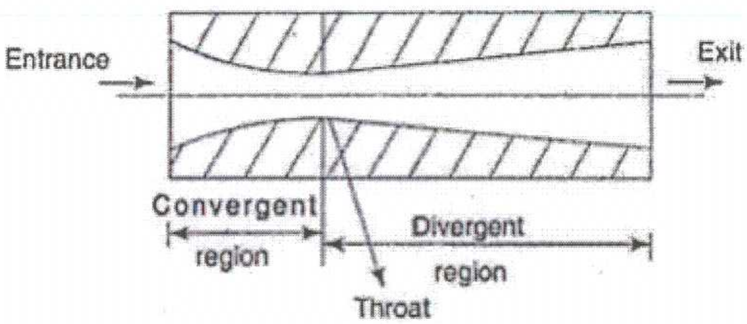
3. *Specific fuel consumption*

We know that specific fuel consumption

$$= \frac{m_f}{B.P.} = \frac{6.5}{22} = 0.295 \text{ kg / B.P. / h Ans.}$$

2

VII. b)



3

Convergent-Divergent Nozzle

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The steam enters the nozzle with a high pressure, but with a negligible velocity. In the converging portion (*i.e.* from the inlet to the throat), there is a drop in the steam pressure with a rise in its velocity. There is also a drop in the enthalpy or total heat of the steam. This drop of heat is not utilised in doing some external work, but is converted into kinetic energy. In the divergent portion (*i.e.* from the throat to outlet), there is further drop of steam pressure with a further rise in its velocity. Again, there is a drop in the enthalpy or total heat of steam, which is converted into kinetic energy. 3

It will be interesting to know that the steam enters the nozzle with a high pressure and negligible velocity. But leaves the nozzle with a high velocity and small pressure. The pressure, at which the steam leaves the nozzle, is known as back pressure. Moreover, no heat is supplied or rejected by the steam during flow through a nozzle. Therefore, it is considered as isentropic flow, and the corresponding expansion is considered as an isentropic expansion. 6

VIII Given : p= 6bar ; $t_w = 25°C$; $x = 0.9$; $t_{sup} = 250°C$; $Cp = 2.3$ kJ/kg K

a) From steam tables , corresponding to the pressure of 6 bar, we get ,

$h_f = 670.4$ kJ/kg $\quad h_{fg} = 2085$ kJ/kg $\quad$ and $\quad$ saturation temp. $T_{sat} = 158.8°C = 431.8K$

1). When the steam is wet :

Enthalpy per kg of wet steam $h = h_f + xh_{fg} = 670.4 + 0.9 \times 2085 = 2546.9$ kJ 3

We have the water at a temp. of 30°C and the enthalpy of water at that state is determined as

Enthalpy of water = 4.2 x 30 = 126 kJ

Actual amount of enthalpy required per kg of steam = 2546.9—126 = 2420.9 kJ

Total heat required to produce 2kg of steam = 2 x 2420.9 = 4841.8 kJ

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2). When the steam is dry saturated ie dryness fraction x = 1

Enthalpy required per kg of steam $hg = h_f + h_{fg} = 670.4 + 2085 = 2755.4$ kJ

Actual enthalpy required per kg of steam = 2755.4 – 126 = 2629.4 kJ 3

Total Enthalpy req'd to produce 2 kg of steam = 2 x 2629.4 = 5258.8 kJ

3). When the steam is super heated to 250°C

Enthalpy required per kg of steam $h_{sup} = hg + Cp\,(T_{sup} - T_{sat}) = 2755.4 + 2.3(523 - 431.8)$

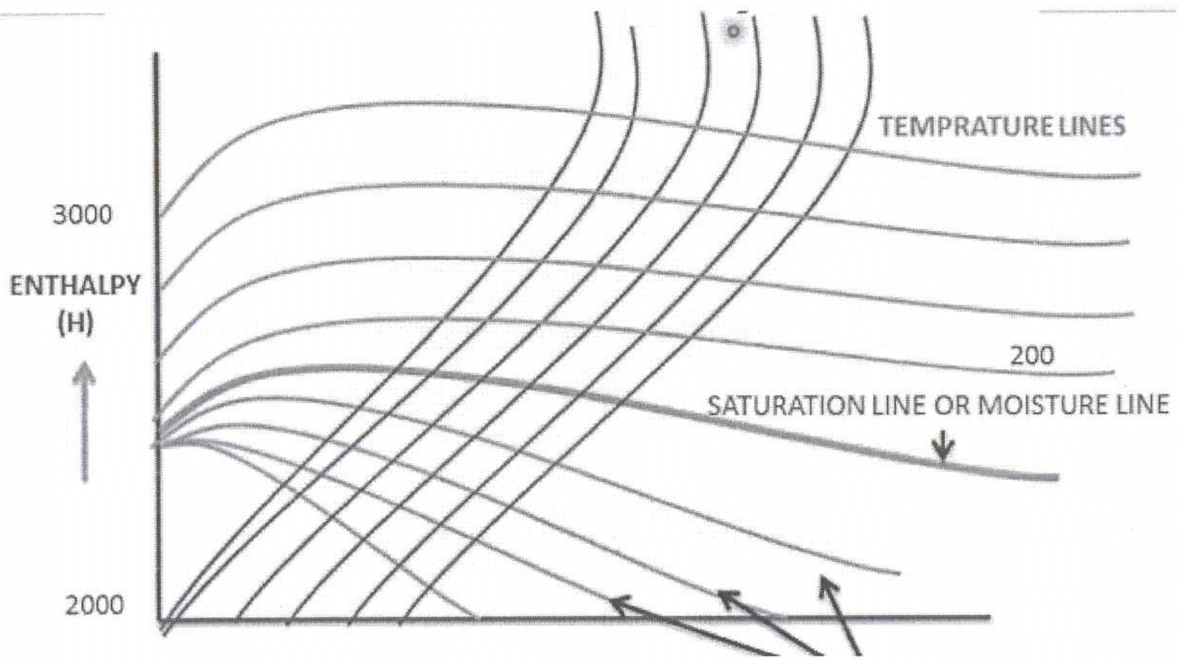
$= 2965.16\text{kJ}$

Actual enthalpy required per kg of steam $= 2965.16 - 126 = 2839.16$ kJ (3)

Total Enthalpy req'd to produce 2 kg of steam = 2 x 2839.16 = 5678.32 kJ

VIII b)

CONSTANT PRESSURE LINES



DRYNESS FRACTION LINES (6)

OR, SIMILLAR FIGURES…..

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IX.

a)

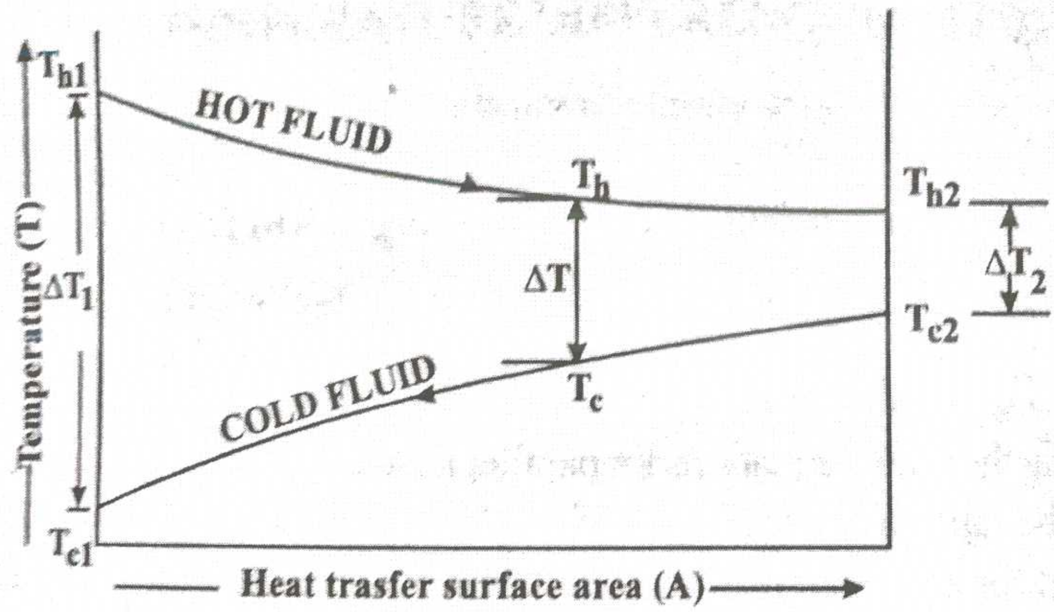

fig (3)

The hot fluid is transferring a part of its heat energy to the cold fluid, there will be an increase in enthalpy of the cold fluid and a corresponding decrease in enthalpy of the hot fluid.

The hot fluid transferring heat energy to cold fluid,

$$Q_h = m_h c_h (T_{h1} - T_{h2})$$

The cold fluid absorbs heat energy from hot fluid,

$$Q_c = m_c c_c (T_{c2} - T_{c1})$$

If we denote the temperature difference between the hot and cold fluids as ΔT

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$\therefore \Delta T = T_h - T_c$

Since ΔT is varying with position in the heat exchanger. Hence the heat exchanger transfers the heat from the hot fluid to the cold fluid is calculated by the equation.

$$Q = UA\,\Delta T_m$$

(9)

Where ΔT_m is a suitable mean temperature difference across the heat exchanger. It is obtained by analysis of heat exchanger and it will be in the form of logarithmic relation. This mean temperature difference is known as *logarithmic mean temperature difference* or *LMTD*. This LMTD value must be determined before using the actual heat transfer rate equation.

ΔT_m or logarithmic mean temperature difference (LMTD) for parallel flow or counter flow heat exchangers is given by.

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{l_n \dfrac{\Delta T_1}{\Delta T_2}}$$

Explu (6)

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IX. b)

Given , $x = 12\ mm = 0.012\ m$; $T_1 = 120°\ C = 273+120 = 393\ K$; $T_2 = 100°C = 373\ K$

Time $t = 1\ h$; area $A = 5\ m^2$; $k = 84 W/m\ K$

We have the quantity of heat transferred $= Q = \frac{kA\,(T1-T2)t}{x}$

$$= \frac{84 \times 5\ (393-373)3600}{0.012}$$ 3

$$= 2520 \times 10^6\ J/h$$

$$= 2520 \times 10^3\ kJ/h$$ 6

We have from the data that the heat required to evaporate 1kg of water at 100° C

Is equal to its latent heat ie 2260 kJ

$\therefore$ mass of water evaporated per hour $= \frac{total\ amount\ of\ heat\ transfered}{heat\ req'd\ to\ evaporate\ 1\ kg\ of\ water}$ 3

$$= (2520 \times 10^3)/2260 = 1115\ kg$$

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X.

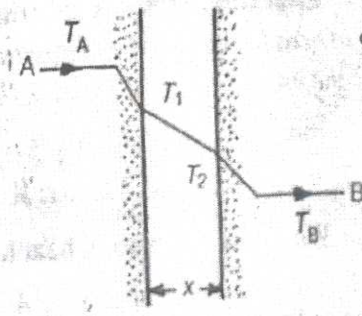
a) Consider a wall through which heat is transferred from a hot surface to a cold surface as shown in Fig. 34.6.

Let $(T_1 - T_2)$ = Difference of temperatures,

A = Surface area of the wall,

x = Thickness of the wall, and

k = Thermal conductivity of the wall material.



Overall coefficient of heat transfer.

fig (2)

As a matter of fact, there will be a thin film of air on both the hot as well as cold faces of the wall, which will act as transition layers adjacent to the wall surface, and through which the heat also has to flow in addition to the wall as shown in Fig. 34.6. Let A and B be the effective film of air for the heat flow.

Let T_A and T_B = Temperatures at ends of two thin films of air A and B respectively.

h_A and h_B = Coefficients of heat transfer for A and B respectively.

U = Overall coefficient of heat transfer.

(9)

We know that the rate of heat flow through air film A,

$$Q = h_A A (T_A - T_1) \quad \text{or} \quad (T_A - T_1) = \frac{Q}{h_A A}$$

derivation (7)

Similarly, rate of heat flow through the wall,

$$Q = \frac{kA(T_1 - T_2)}{x} \quad \text{or} \quad (T_1 - T_2) = \frac{Qx}{kA}$$

rate of heat flow through the film B,

$$Q = h_B A (T_2 - T_B) \quad \text{or} \quad (T_2 - T_B) = \frac{Q}{h_B A}$$

Adding equations (*i*), (*ii*) and (*iii*), we get

$$(T_A - T_B) = \frac{Q}{A}\left[\frac{1}{h_A} + \frac{x}{k} + \frac{1}{h_B}\right]$$

$$\therefore \quad Q = \frac{A(T_A - T_B)}{\left(\frac{1}{h_A} + \frac{x}{k} + \frac{1}{h_B}\right)}$$

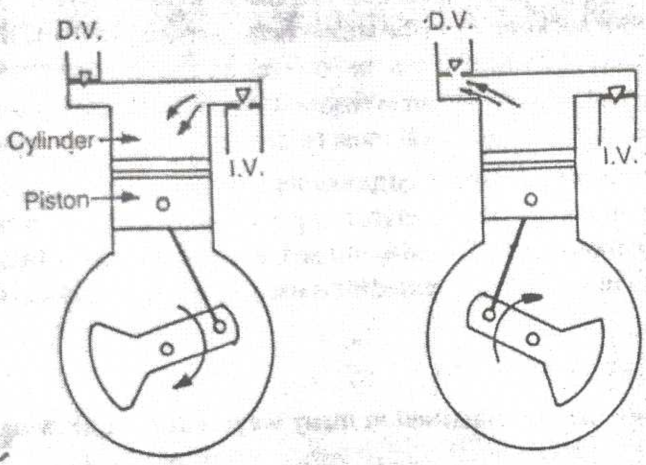
We know that the rate of heat flow,

$$Q = UA(T_A - T_B)$$

where $U = \dfrac{1}{\left[\dfrac{1}{h_A} + \dfrac{x}{k_1} + \dfrac{1}{h_B}\right]}$

X.
b)

A single stage reciprocating air compressor, in its simplest form, consists of a cylinder, piston, inlet and discharge valves, as shown in Fig. 28.1. From the geometry of the compressor, we find that when the piston moves downwards (or in other words, during outward or suction stroke), the pressure



(a) Suction stroke. (b) Delivery stroke.

Single stage reciprocating air compressor.

OR
Simillar figures

fig
3

6

inside the cylinder falls below the atmospheric pressure. Due to this pressure difference, the inlet valve (I.V.) gets opened and air is sucked into the cylinder, at inlet pressure until the piston completes the outward stroke. Now when the piston moves upwards (or in other words, during inward or delivery stroke), the pressure inside the cylinder goes on increasing till it reaches the discharge pressure. At this stage, the discharge valve (D.V.) gets opened and air is delivered to the container. At the end of delivery stroke, a small quantity of air, at high pressure, is left in the clearance space. As the piston starts its suction stroke, the air contained in the clearance space expands till its pressure falls below the atmospheric pressure. At this stage, the inlet valve gets opened as a result of which fresh air is sucked into the cylinder, and the cycle is repeated.

Explanation
3

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