MADE EASY
Leading Institute for ESE, GATE & PSUs

Detailed Solutions

ESE-2025
Mains Test Series

Mechanical Engineering
Test No : 1

Thermodynamics + Refrigeration and Air-conditioning + IC Engine

Section A : Thermodynamics + Refrigeration and Air-conditioning + IC Engine

1. (a)

Given :Heat transfer to gas, $Q_r = 1$ kW

Heat transfer from gas to surrounding, $Q_s = 50t$ W

Consider helium in the closed rigid tank as the system

$\therefore$ For closed rigid tank $\dot{W} = 0$

From energy rate balance, $\dfrac{dE}{dt} = \dot{Q} - \dot{W}$

$$\frac{dE}{dt} = \dot{Q} \qquad [\text{As } \dot{W} = 0]$$

As system receives by heat transfer from resistor at a rate of 1 kW and loses energy to surrounding at a rate of 50tW.

$$\therefore \qquad \dot{Q} = [1000 - 50t]\text{W}$$

So, $$\frac{dE}{dt} = 1000 - 50t \qquad (\text{where } t \text{ is in seconds})$$

Integrating the above equation, we have

$$\Delta E = \int_0^t \frac{dE}{dt} \cdot dt = \int_0^t (1000 - 50t)\, dt$$

$$= \left[1000t - \frac{50t^2}{2}\right]_0^t = 1000t - \frac{50t^2}{2} \text{Joule}$$

$$= \frac{\left[1000t - \frac{50t^2}{2}\right]}{1000} \text{kJ}$$

$$\Delta E = t - 0.025\, t^2 \text{ kJ} \qquad \text{...(i)}$$

Since the above equation is parabolic so the plot between change in energy and time would also be parabolic. We need to find where the ΔE value is maximum and zero.

$$\Delta E = t - 0.025\, t^2$$

for $\Delta E = 0$, we have

$$t - 0.025\, t^2 = 0$$

$$t(1 - 0.025t) = 0$$

$$t = 0 \text{ sec}; \; t = \frac{1}{0.025} = 40 \text{ sec}$$

At $t = 0$ sec, $\Delta E = 0$ and $t = 40$ sec, $\Delta E = 0$

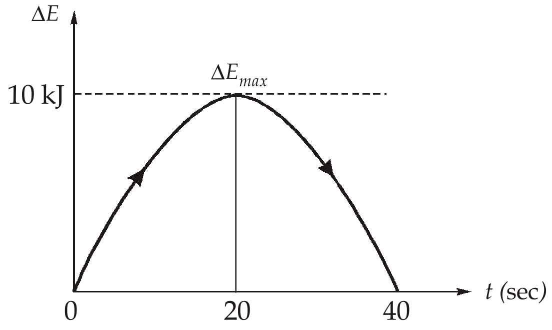
For maximum ΔE, differentiate equation (i) and equate it to zero

$$\frac{d(\Delta E)}{dt} = 1 - 0.05t = 0$$

$$t = \frac{1}{0.05} = 20 \text{ sec}$$

∴ ΔE will be maximum at $t = 20$ sec

$$(\Delta E)_{\max} = 20 - 0.025 \times (20)^2 = 10 \text{ kJ}$$



Following results can be concluded from the above change in energy CΔE Vs time plot

1. From 0 to 20 sec

   Energy increases from the initial value at $t = 0$ to maximum value at $t = 20$ sec.

2. From 20 sec to 40 sec

   Energy decreased from its maximum value at $t = 20$ sec to zero at $t = 40$ sec.

- Note that since any arbitrary value of E_0 can be assigned to the energy of the system at $t = 0$, no particular significance can be attached to the value of energy at the initial state or any other state. Only changes in the energy of the system has significance.

1. (b)

Given : $\dot{m}_1 = 4$ kg/sec; $\dot{m}_2 = 0.5$ kg/sec; $T_2 = 250°C$; $T_1 = 45°C$

From conservation of mass, $\dot{m}_3 = \dot{m}_2 + \dot{m}_1 = 4 + 0.5 = 4.5$ kg/sec

From first law, $\dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_1 h_1$

($\because$ $Q = 0$, because insulated, $\Delta KE = 0$, $\Delta PE = 0$)

From steam table,

$$h_1 = 188.95 \text{ kJ/kg and } h_2 = 2957.6 \text{ kJ/kg}$$

$$4.5\, h_3 = 0.5 \times 2957.6 + 4(188.95)$$

$$h_3 = \frac{0.5 \times 2957.6 + 4(188.95)}{4.5} = 496.58 \text{ kJ/kg}$$

The enthalpy value is less than that of saturated liquid at 600 kPa. Therefore, the existing water is also subcooled. Its temperature is interpolated from steam tables.

$$T_3 = \left(\frac{496.58 - 482.9}{504.09 - 482.9}\right) \times 5 + 115 = 118.23°C$$

The entropy at this temperature is interpolated from the steam tables.

$$s_3 = \frac{(15.275 - 1.4733)}{(120 - 115)} \times (118.23 - 115) + 1.4733$$

$$= 1.5083 \text{ kJ/kgK}$$

and again from steam tables, $s_1 = 0.63836$ kJ/kgK and $s_2 = 7.1832$ kJ/kgK

$$\text{Entropy generation, } s_{gen} = \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1$$

$$= 4.5(1.5083) - 0.5(7.1832) - 4(0.63836)$$

$$= 0.64231 \text{ kJ/kgK}$$

Entropy production or generation is positive, indicating that entropy is produced, a consequence of the 2nd law. Hence, the mixing process between the superheated steam and subcooled water is irreversible process.

For a process to be reversible process the entropy generation must be zero.

1. (c)

Given : $P_H = 10$ bar, $P_L = 2$ bar; $x_w = 0.35$; $Q_s = 84$ kJ/kg of NH_3 generated

(i)

The amount of weak solution leaving the generator per kg of anhydrous NH_3 generated

$$= 10 - 1 = 9 \text{ kg}$$

$$\text{Heat absorbed, } Q_a = 802.5(1 - x_w) - 928x_w^2$$

$$= 802.5(1 - 0.35) - 928(0.35)^2$$

$$= 407.95 \text{ kJ/kg of } NH_3 \text{ generated}$$

From enthalpy balance of generator,

Heat coming into generator = Heat going out of generator

∴ Heat carried by strong ammonia + Heat given by source = Heat carried by ammonia vapour + Heat carried by weak aqua + Heat absorption + Heat lost to the surrounding

$$10[418 + 4.78 \times 82] + Q_s = 1860 + 9[418 + 4.78 \times 100] + 407.95 + 79$$

$$Q_s = 2311.35 \text{ kJ/kg of } NH_3 \text{ generated}$$

(ii)

The processes are represented on P-h diagram,

From P-h chart (enthalpies in kJ/kg)

$$h_1 = 554 \text{ kJ/kg}; h_2 = 1647 \text{ kJ/kg and } h_3 = 1899 \text{ kJ/kg}$$

Refrigerating effect per kg of $NH_3 = h_2 - h_1$

$$= 1647 - 554 = 1093 \text{ kJ/kg}$$

∴ Ratio of refrigeration effect to heat supplied $= \dfrac{\text{Refrigeration effect}}{\text{Heat supplied}}$

$$= \frac{1093}{2311.35} = 0.473$$

Refrigerating capacity of system $= \dfrac{1093 \times 8}{60 \times 3.5} = 41.64$ tons

(iii)

The net work available out of 2311.35 kJ of heat for the compression of refrigerant in the compressor is

$$= 2311.35 \times 0.1$$
$$= 231.135 \text{ kJ/kg}$$

Work required in the compressor per kg of NH_3 compressed isentropically

$$= h_3 - h_2$$
$$= 1899 - 1647 = 252 \text{ kJ/kg}$$

The refrigeration effect possible with 231.135 kJ of thermal energy when used to drive the prime mover coupled to the compression refrigeration machine is

$$= \frac{231.135}{252} \times 1093 = 1002.5 \text{ kJ}$$

For same given input of thermal energy,

$$\frac{\text{Refrigeration effect in compression machine}}{\text{Refrigeration effect in absorption machine}} = \frac{1002.5}{1093} = 0.917$$

1. (d)

A complete heating, ventilating and air conditioning (HVAC) system has the facility to heat, cool, humidify, dehumidify, clean and distribute the conditioned air into the room so as to meet the indoor year-round human comfort or industrial applications. In a large building or in a large installation such as shopping complex, academic complex, research laboratories and office complex, etc. Different areas may require different levels of temperature, humidity and cleanliness. The regions with different requirements are divided into zones. Similarly, in a hotel too, each occupant in each room may desire a different level of temperature and humidity. In some cases, the temperature can be controlled by varying the air-volume flow rate or by supplying a little colder air and providing heaters for fine control of temperature. But in general, such vast systems are divided into zones of similar requirements and the system is integrated for energy optimization.

Sometimes the complex may be so far spread out that conditioned air from a single unit cannot be economically transported to various locations. The fan power requirement will be large and also the size of the duct to carry conditioned air and return air will occupy enormous space. In such cases, chilled water and hot water are generated at a central place and transported in underground- insulated pipes to individual buildings for efficient heating/cooling of the coils with air blown and recirculated in separate air handling units. Similarly, in high-rise buildings too, chilled water and hot water are pumped to all the floors where the individual cooling coils are used to heat/cool the air. These systems involve an heat exchanger in a central place where the water is heated/cooled.

A pump is used to transport water in a long pipe line involving heat loss/gain through insulation, followed by cooling/heating of air in another heat exchanger. It saves fan power, but involves more pump power, more heat loss/gain in pipes and an extra heat exchanger. On the other hand, if small individual units are used, the equipment will directly heat or cool the air, thus, eliminating the temperature drop in the second heat exchanger, saving pump power and also heat loss through insulation. Such systems will, however, be of maximum load that each zone needs. It may so happen that all the individual units may not utilize the maximum capacity simultaneously, but such maximum capacity has to be anyway provided for each unit. A central system of smaller total cooling/heating capacity can take care of this by meeting the requirement of maximum load in some zones while the loads in some other zones will be at their minimum. This concept is called diversity factor, which is the ratio of actual load of the whole system to the sum of the individual maximum loads in all the zones.

1. **(e)**

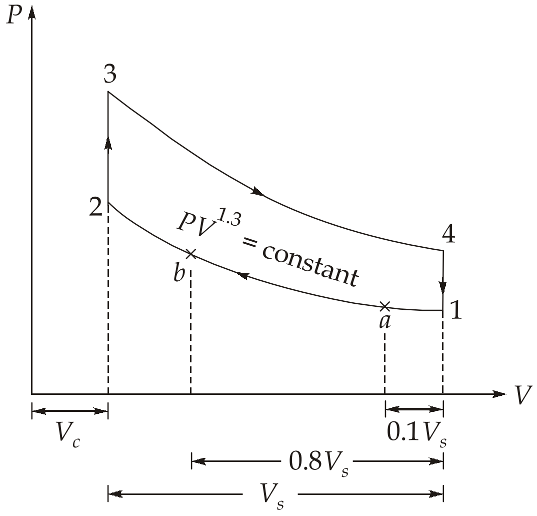
Given : CV = 42000 kJ/kg; $\eta_{\text{relative}} = 0.75$

The process 1 - 2 represents compression, following the law $PV^{1.3}$ = constant.

Point *a* represent 10% of compression stroke.

Point *b* represent 80% of the compression stroke.

$$\begin{aligned} V_a &= V_1 - 0.1V_s \\ &= V_s + V_c - 0.1V_s \\ V_a &= V_c + 0.9V_s \end{aligned}$$



and
$$V_b = V_1 - 0.8V_s$$
$$= V_s + V_c - 0.8V_s$$
$$V_b = V_c + 0.2V_s$$

a - b follows the law $PV^{1.3}$ = constant

So,
$$P_aV_a^{1.3} = P_bV_b^{1.3}$$
$$\frac{V_a}{V_b} = \left(\frac{P_b}{P_a}\right)^{1/1.3} = \left(\frac{4}{1}\right)^{1/1.3} = 2.905$$
$$\frac{V_c + 0.9V_s}{V_c + 0.2V_s} = 2.905$$

$$\Rightarrow \quad \frac{1 + 0.9\dfrac{V_s}{V_c}}{1 + 0.2\dfrac{V_s}{V_c}} = 2.905$$

$$\Rightarrow \quad 1 + 0.9\frac{V_s}{V_c} = 2.905 + 0.581\frac{V_s}{V_c}$$
$$\frac{V_s}{V_c} = 5.972$$

$$\text{Compression ratio, } r = \frac{V_1}{V_2} = \frac{V_c + V_s}{V_c} = 1 + \frac{V_s}{V_c}$$
$$r = 1 + 5.972 = 6.972$$

$\therefore$ Air standard efficiency,

$$\eta_0 = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(6.972)^{0.4}}$$
$$\eta_0 = 0.54 = 54\%$$

$$\text{Relative efficiency} = \frac{\text{Indicated thermal efficiency}}{\text{Air-standard efficiency}} = \frac{\eta_{th_i}}{\eta_o}$$
$$0.75 = \frac{\eta_{th_i}}{0.54}$$
$$\eta_{th_i} = 0.405 = \frac{\text{Indicated power}}{C.V \times \dot{m}} = \frac{I.P}{C.V \times \dot{m}}$$

Specific fuel consumption (indicated) is $\dfrac{\dot{m}}{I.P}$

$$\frac{\dot{m}}{I.P} = \frac{1}{C.V \times \eta_{th_i}} = \frac{1}{42000 \times 0.405} = 5.88 \times 10^{-5} \text{ kg/kWs}$$

$$\frac{\dot{m}}{I.P} = 5.88 \times 10^{-5} \times 3600 \text{ kg/kWh}$$

$$isfc = 0.212 \text{ kg/kWh}$$

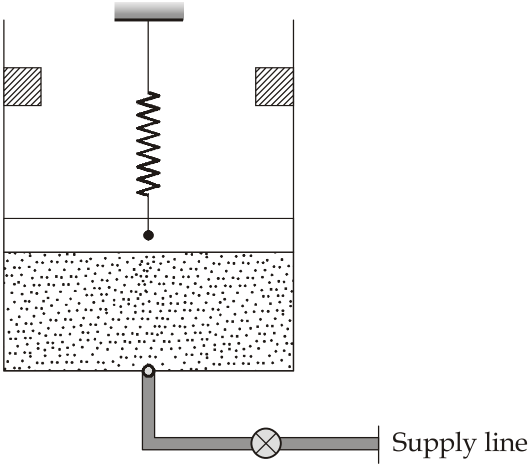
2. **(a)**

Given : $A_p = 0.1 \text{ m}^2$; $V = 20\text{L}$

Volume when piston hits the stops,

$$V_S = 50\,\text{L}$$

$$T_1 = 10°\text{C};\ P_1 = 200 \text{ kPa};\ P_3 = 800 \text{ kPa};\ T_3 = 80°\text{C}$$



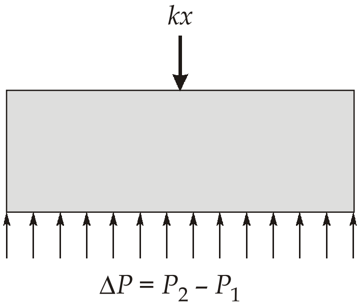
$$\text{Initial mass in the cylinder, } m_1 = \frac{P_1V_1}{RT_1} = \frac{200 \times \left(20 \times 10^{-3}\right)}{0.287 \times (10 + 273)}$$

$$m_1 = 0.0492 \text{ kg}$$

(i)

Let P_2 be the pressure when piston hits the stops.

FBD of piston,



$$A_P(P_2 - P_1) = kx$$

$$A_P(P_2 - P_1) = k\frac{(V_2 - V_1)}{A_P}$$

$$P_2 = P_1 + k\frac{(V_2 - V_1)}{A_P^2}$$

$$= 200 + 100 \times \frac{(50 - 20) \times 10^{-3}}{(0.1)^2} = 200 + 300$$

$$P_2 = 500 \text{ kPa}$$

Pressure required inside cylinder for the piston to hit the stops is 500 kPa or more.

$\because$ The piston will be at the stops when final pressure inside cylinder will be 800 kPa.

(ii)

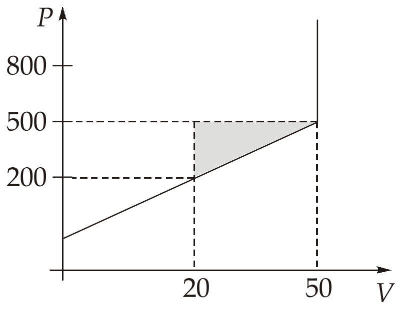
From mass balance, $m_2 - m_1 = m_i - m_e$

Since there is no air outlet so $m_e = 0$

$$\therefore \quad m_2 - m_1 = m_i \qquad \text{...(i)}$$

From 1st law,

$$m_2u_2 - m_1u_1 = m_ih_i + Q - m_eh_e - W_{cv} \qquad \text{...(ii)}$$



$$W_{cv} = \int PdV = \frac{1}{2}(P_1 + P_2)(V_2 - V_1)$$

$$= \frac{1}{2}(200 + 500)\left(50 \times 10^{-3} - 20 \times 10^{-3}\right)$$

$$= \frac{1}{2}\left(700 \times 30 \times 10^{-3}\right) = 10.5 \text{ kJ}$$

From equation (ii),

$$Q = m_2u_2 - m_1u_1 - m_ih_i + W_{cv}$$

$$= m_2c_vT_2 - m_1c_vT_1 - (m_2 - m_1)c_pT_i + W_{cv}$$

Here, $$m_2 = \frac{P_2V_2}{RT_2} = \frac{800 \times \left(50 \times 10^{-3}\right)}{0.287 \times (80 + 273)}$$

$$m_2 = 0.395 \text{ kg}$$

$\therefore$ $$Q = 0.395 \times 0.718\,(80 + 273) - 0.0492(0.718)(10 + 273)$$
$$- (0.395 - 0.0492)1.005(50 + 273) + 10.5$$

$$Q = -11.635 \text{ kJ}$$

(iii)

Net change in entropy,

$$\Delta s_{\text{net}} = m_2(s_2 - s_i) - m_1(s_1 - s_i) - \frac{Q}{T_o}$$

$$s_2 - s_i = c_p \ln\left(\frac{T_2}{T_i}\right) - R\ln\left(\frac{P_2}{P_i}\right)$$

$$= 1.005 \ln\left(\frac{353}{323}\right) - 0.287 \ln\left(\frac{800}{800}\right)$$

$$= 0.08926 \text{ kJ/kgK}$$

and $$s_1 - s_i = c_p \ln\left(\frac{T_1}{T_i}\right) - R\ln\left(\frac{P_1}{P_i}\right)$$

$$= 1.005 \ln\left(\frac{283}{323}\right) - 0.287 \ln\left(\frac{200}{800}\right)$$

$$= 0.265 \text{ kJ/kgK}$$

$$\Delta s_{\text{net}} = 0.395 \times 0.08926 - 0.0492 \times 0.265 - \left(\frac{-11.6}{283}\right)$$

$$\Delta s_{\text{net}} = 0.0632 \text{ kJ/K}$$

2. (b)

Given : $RC = 80$ kW; $T_E = -8°C$; $T_c = 42°C$, $\frac{d}{L} = \frac{1}{1.4}$, $N = 1250$ rpm

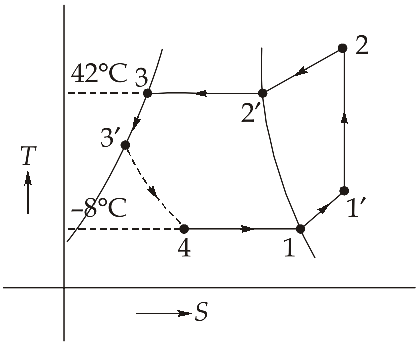
The process of the given system as represented on T-s diagram as shown.

Process, 4 - 1′ represent evaporator

1′ - 2 represent isentropic compression

2 - 3′ represent condensor

3′ - 4 represent throttling process



$$h_3' = h_3 - C_{pl}(T_3 - T_3')$$

$$= 252.4 - 1.24(5)$$

$$h_3' = 246.2 \text{ kJ/kg}$$

Now,

$$h_1' = h_1 + C_{pg}(T_1' - T_1)$$

$$= 402.3 + 0.74(10)$$

$$h_1' = 409.7 \text{ kJ/kg}$$

$$\text{Refrigeration effect (RE)} = h_1' - h_4 = 409.7 - 246.2 \quad [\text{As } h_4 = h_3', \text{ isenthelpic process}]$$

$$= 163.5 \text{ kJ/kg}$$

Now, considering isentropic compression 1′ – 2.

$$S_1' = S_2$$

$$S_{g1} + C_{pg}\ln\left(\frac{T_1'}{T_1}\right) = S_{g2} + C_{pg}\ln\left(\frac{T_2}{T_2'}\right)$$

$$1.764 + 0.74\ln\left(\frac{-8+10+273}{-8+273}\right) = 1.697 + 0.74\ln\left(\frac{T_2}{273+42}\right)$$

$$T_2 = 357.86 \text{ K}$$

$$T_2 = 84.86°\text{C}$$

$$h_2 = h_2' + C_{pg}(T_2 - T_2')$$

$$= 416.8 + 0.74(84.86 - 42)$$

$$h_2 = 448.52 \text{ kJ/kg}$$

(i)

$$\text{COP} = \frac{\text{Desired Effect or refrigeration effect}}{\text{Work done}}$$

$$\text{COP} = \frac{163.5}{h_2 - h_1'} = \frac{163.5}{448.52 - 409.7} = 4.21$$

(ii)

$$\text{Refrigeration capacity} = 80 \text{ kW}$$

$$80 = \dot{m}(h_1' - h_4) = \dot{m}(409.7 - 246.2)$$

$$\dot{m} = 0.489 \text{ kg/sec}$$

(iii) The condition of the vapour refrigerant at the inlet of compressor is represented by the condition 1′.

Volume handled by compressor, $V = \dot{m}v_1'$

where v_1' is the specific volume of the gas.

$$\frac{v_1'}{v_1} = \frac{T_1'}{T_1} = \frac{-8+10+273}{-8+273} = 1.038$$

$$v_1' = 1.038 \times 0.064$$

$$v_1' = 0.0664 \text{ m}^3/\text{kg}$$

$$V = 0.0664 \times 0.489 \times 60 \text{ m}^3/\text{min}$$

$$V = 1.948 \text{ m}^3/\text{min}$$

This is the required theoretical piston displacement.

(iv) As the compressor is having two cylinder, then the volume handled by each cylinder is

$$V_1 = \frac{1.948}{2} = 0.974 \text{ m}^3/\text{min}$$

Also, volume handled by each cylinder per minute is given by

$$V_1 = \left(\frac{\pi}{4}d^2 L \times \eta_V\right) \times N$$

$$0.974 = \frac{\pi}{4}\left(d^2\right)(1.4d) \times 0.85 \times 1250$$

$$d = 0.0941 \text{ m}$$

$$d = 9.41 \text{ cm}$$

and $L = 13.1$ cm

2. **(c)**

Given : $d = 0.05$ m; Stroke, $L = 0.07$ m; $N = 3200$ rpm; $n = 4$ (four cylinder)

Torque arm, $r = 0.35$, $W = 150$ N

Fuel consumption, $\dot{V}_f = 6\ l/\text{h} = \dfrac{6 \times 10^{-3}}{3600}$

$$\dot{V}_f = 1.667 \times 10^{-6} \text{ m}^3/\text{s}$$

$$\rho_f = 780 \text{ kg/m}^3$$

$$C.V = 44000 \text{ kJ/kg}$$

$$\dot{m}_f = \rho_f \times \dot{V}_f = 780 \times 1.667 \times 10^{-6}$$

$$= 1.3 \times 10^{-3} \text{ kg/s}$$

$$\text{Torque, } T = W \times r = 150 \times 0.35 = 52.5 \text{ Nm}$$

(i) $$\text{Brake power} = \frac{2\pi NT}{60} = \frac{2 \times \pi \times 3200 \times 52.5}{60} = 17.59 \text{ kW}$$ **Ans.**

(ii) $$\text{Brake power, BP} = \frac{(bmep)LANn}{2 \times 60} \qquad \left[\text{Four stroke} = \frac{\text{PLANn}}{120}\right]$$

$$17.59 \times 1000 = \frac{(bmep) \times 0.07 \times \frac{\pi}{4}(0.05)^2 \times 3200 \times 4}{2 \times 60}$$

$$bmep = 11.998 \text{ bar} \simeq 12 \text{ bar}$$ **Ans.**

(iii) Brake thermal efficiency,

$$\eta_b = \frac{B.P}{\dot{m}_f \times C.V} = \frac{17.59}{1.3 \times 10^{-3} \times 44000} = 30.75\%$$ **Ans.**

$$bsfc = \frac{\dot{m}_f}{B.P} = \frac{1.3 \times 10^{-3} \times 3600}{17.59}$$

$$bsfc = 0.266 \text{ kg/kWh}$$ **Ans.**

(iv) From Morse test,

$$\text{Indicated power, } IP_1 = 17.59 - \frac{2 \times \pi \times 3200 \times (100 \times 0.35)}{60 \times 1000} = 5.86 \text{ kW}$$

Similarly,
$$IP_2 = 17.59 - \frac{2 \times \pi \times 3200 \times (95 \times 0.35)}{60 \times 1000} = 6.45 \text{ kW}$$

$$IP_3 = 17.59 - \frac{2 \times \pi \times 3200 \times (98 \times 0.35)}{60 \times 1000} = 6.096 \text{ kW}$$

$$IP_4 = 17.59 - \frac{2 \times \pi \times 3200 \times (104 \times 0.35)}{60 \times 1000} = 5.39 \text{ kW}$$

$$IP = IP_1 + IP_2 + IP_3 + IP_4$$

$$= 5.86 + 6.45 + 6.096 + 5.39$$

$$IP = 23.79 \text{ kN}$$ **Ans.**

$$\text{Mechanical efficiency, } \eta_m = \frac{BP}{IP} = \frac{17.59}{23.79} = 0.7391$$

$$\eta_m = 73.91\%$$ **Ans.**

Also, $$imep = \frac{bmep}{0.7391} = \frac{12}{0.7391} = 16.23 \text{ bar}$$ **Ans.**

3. (a)

Given : $P_a = 7$ bar, $P_c = 0.06$ bar, $\eta_n = 0.9$, $\eta_e = 0.65$, $\eta_c = 0.82$, $x_e = 0.92$

From steam tables

P_e (Saturation pressure at 5°C) = 0.008726 bar

At $P_a = 7$ bar, $h_a = 2762.8$ kJ/kg

when $x_a = 1$ and $T_a = 164.946$°C

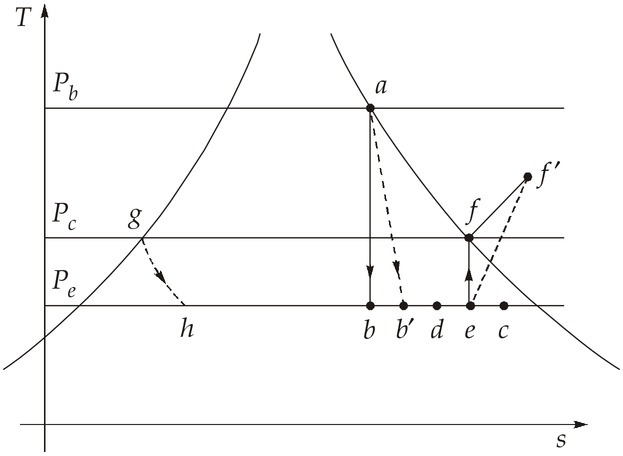
Entropy at 5°C for liquid = 0.07625 kJ/kgK

Entropy at 5°C for saturated vapour = 9.0248 kJ/kgK

Entropy at 7 bar pressure for saturated steam = 6.7071 kJ/kgK

As *ab* is isentropic expansion

$$\therefore \quad s_a = s_b$$



Dryness fraction of *b* in given by

$$x_b = \frac{6.7071 - 0.07625}{9.0248 - 0.07625} = 0.7409 \simeq 0.741$$

$$h_b = (h_f)_b + x_b(h_{fg})_b$$

$$h_b = 21.02 + 0.741[2489]$$

[Using steam tables find h_f and h_{fg} at 5°C]

$$h_b = 1865.369 \text{ kJ/kg}$$

Nozzle efficiency is given by

$$\eta_n = \frac{h_a - h_b'}{h_a - h_b}$$

$$h_b' = h_a - \eta_n(h_a - h_b)$$

$$= 2762.8 - 0.9(2762.8 - 1865.369)$$

$$h_b' = 1955.112 \text{ kJ/kg}$$

The points b and b' are on the same pressure line

$$\therefore \quad (h_{fg})_b = \left(h_{fg}\right)'_b \text{ and } \left(h_f\right)_b = \left(h_f\right)'_b$$

$$h_b' = \left(h_f\right)'_b + x_b\left(h_{fg}\right)'_b$$

$$x_b' = \frac{h_b' - \left(h_f\right)'_b}{\left(h_{fg}\right)'_b} = \frac{1955.112 - 21.02}{2489} = 0.777$$

The compression efficiency is given by

$$\eta_c = \frac{h_f - h_e}{h_f' - h_e}$$

$$h_f' = h_e + \frac{h_f - h_e}{\eta_c} = 2310.9 + \frac{2559.43 - 2310.9}{0.82}$$

$$h_f' = 2613.985 \text{ kJ/kg}$$

(i) The mass of motive steam required per kg of flashed vapour is given by,

$$m = \frac{h_f - h_e}{\eta_c\eta_e\eta_n(h_a - h_b) - \left(h_f - h_e\right)}$$

$$m = \frac{2559.43 - 2310.9}{0.82 \times 0.65 \times 0.9(2762.8 - 1865.369) - (2559.43 - 2310.9)}$$

$$m = 1.366 \text{ kg of motive steam per kg of flashed vapour}$$

(ii) Quality of flashed vapour from flash chamber:

As we know, $\quad h_c + m.h_d = (m+1)h_e$

$$h_c = (1.366 + 1)2310.9 - 1.366 \times 2237.80$$

$$= 2410.75 \text{ kJ/kg}$$

Also, $$h_c = (h_f)_c + x_c(h_{fg})_c$$

∴ $$x_c = \frac{2410.75 - 21.02}{2489} = 0.960$$

The entrainment efficiency is given by

$$\eta_e = \frac{h_a - h_d}{h_a - h_b'}$$

∴ $$h_d = h_a - \eta_e(h_a - h_b')$$

$$= 2762.8 - 0.65(2762.8 - 1955.112)$$

$$h_d = 2237.80 \text{ kJ/kg}$$

Also, $$h_d = \left(h_f\right)_d + x_d\left(h_{fg}\right)_d$$

$$x_d = \frac{h_d - \left(h_f\right)_d}{\left(h_{fg}\right)_d} = \frac{2237.80 - 21.02}{2489} = 0.891$$

At point *e*, $$x_e = 0.92$$

∴ $$h_e = 21.02 + 0.92(2489) = 2310.9 \text{ kJ/kg}$$

The compression *ef* is isentropic,

∴ $$s_e = s_f = \left(s_f\right)_e + \frac{x_f\left(h_{fg}\right)_f}{T_f} = \left(s_f\right) + x_f\left(s_{fg}\right)$$

Also, $$s_e = 0.07625 + 0.92(9.0248 - 0.07625) = 8.308 \text{ kJ/kgK}$$

$$x_f = \frac{s_e - \left(s_f\right)_{\text{at 0.06 bar}}}{\left(s_{fg}\right)_{\text{at 0.06 bar}}} = \frac{8.308 - 0.52082}{7.8082}$$

$$x_f = 0.997$$

$$h_f = \left(h_f\right)_f + x_f\left(h_{fg}\right)_f = 151.48 + 0.997(2415.2)$$

$$h_f = 2559.43 \text{ kJ/kg}$$

Refrigerating effect per kg of flashed vapour

$$= h_c - \text{enthalpy of make up water at } 20°\text{C}$$

$$= 2410.75 - 4.18 \times 20 = 2327.15 \text{ kJ/kg}$$

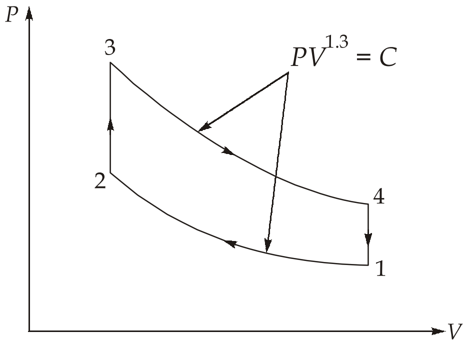
(iv) Refrigerating effect per ton refrigeration per hour

$$= 3.5 \times 3600 = 12600 \text{ kJ}$$

Mass of motive steam required per ton of refrigeration per hour

$$= \frac{12600}{2327.15} \times 1.366 = 7.396 \text{ kJ/ton-hr}$$

3. (b)



Given : Compression ratio, $r = 9$; Air fuel ratio = 14 : 1

The Stoichiometric equation can be written as

$$C_8H_{18} + \frac{25}{2}\left[O_2 + \frac{79}{21}N_2\right] \rightarrow 8CO_2 + 9H_2O + \frac{25}{2} \times \frac{79}{21}N_2$$

$$\text{Stoichiometric air/fuel ratio} = \frac{12.5 \times \left(32 + \frac{79}{21} \times 28\right)}{(12 \times 8) + (1 \times 18)} = 15.06$$

With the given air/fuel ratio as 14, the mixture is rich in fuel. Therefore the combustion will be incomplete.

The chemical reaction becomes:

$$C_8H_{18} + y\left[O_2 + \frac{79}{21}N_2\right] \rightarrow xCO + (8-x)CO_2 + 9H_2O + \frac{79}{21}yN_2$$

$$\frac{\text{Air}}{\text{Fuel}} = 14 = \frac{y\left(32 + \frac{79}{21} \times 28\right)}{(12 \times 8) + (1 \times 18)}$$

$$y = 11.62$$

By oxygen balance:

$$y = \frac{x}{2} + (8 - x) + 4.5$$

$$11.62 = 0.5x + 8 - x + 4.5$$

$$0.5x = 8 + 4.5 - 11.62$$

$$x = 1.76$$

The chemical equation now becomes:

$$C_8H_{18} + 11.62\left(O_2 + \frac{79}{21}N_2\right) \rightarrow 1.76CO + 6.24CO_2 + 9H_2O + \frac{79}{21} \times 11.62N_2$$

Number of moles before combustion $= 1 + 11.62 + \left(\frac{79}{21} \times 11.62\right) = 56.33$

Number of moles after combustion = 1.76 + 6.24 + 9 + 43.71 = 60.71

$$\text{Molecular expansion} = \frac{60.71 - 56.33}{56.33} \times 100 = 7.78\%$$ **Ans.**

(i) $P_1 = 1.2$ bar; $T_1 = 65 + 273 = 338$ K; $n = 1.3$; $r = 9$

$$\frac{T_2}{T_1} = (r)^{n-1}$$

$$T_2 = 338 \times (9)^{0.3} = 653.42 \text{ K}$$

$$q_{2\text{-}3} = c_v(T_3 - T_2)$$

$$\frac{44000}{15} = 0.71(T_3 - 653.42)$$

$$T_3 = 4784.87 \text{ K}$$ **Ans.**

For pressure,
$$\frac{P_1V_1}{T_1} = \frac{P_3V_3}{T_3}$$

$$P_3 = \frac{V_1}{V_3} \times \frac{T_3}{T_1} \times P_1$$

$$= 9 \times \frac{4784.87}{338} \times (1.2) = 152.89 \text{ bar}$$

(ii)

Since the mass of the reactants and products is the same and specific heats are assumed same, the temperature of the products with molecular expansion will remain the same as without molecular expansion, only pressure will change.

$$\therefore \qquad T_3 = 4784.87 \text{ K} \qquad \textbf{Ans.}$$

$$PV = n\bar{R}T$$

$$P \propto n$$

$$\frac{P_3'}{P_3} = \frac{n'}{n}$$

where n' is the number of moles after expansion

$$P_3' = 152.89 \times \frac{60.71}{56.33} = 164.78 \text{ bar} \qquad \textbf{Ans.}$$

3. (c)

Given : $P_0 = 100$ kPa; $D_0 = 1$ m; $T_i = 250$ K; $V_1 = 0.4$ m^3; $P_1 = 100$ kPa; $P_{max} = 200$ kPa; $V_2 = 2$ m^3

Volume of balloon when it becomes spherical,

$$V_0 = \frac{\pi}{6} \times (1)^3 = 0.5236 \text{ m}^3$$

$$\text{Initial mass of helium gas, } m = \frac{P_1V_1}{RT_1} = \frac{100 \times 0.4}{2.077 \times 250} = 0.077 \text{ kg}$$

(i)

As,
$$P = P_0 + C\left[1 - \left(\frac{D_0}{D}\right)^6\right]\left(\frac{D_0}{D}\right) = P_0 + C\left[\left(\frac{D_0}{D}\right) - \left(\frac{D_0}{D}\right)^7\right]$$

$$\frac{dP}{dD} = 0 + C\left[-\frac{D_0}{D^2} + 7\frac{D_0^7}{D^8}\right]$$

$$\frac{dP}{dD} = C\left[\frac{7D_0^7}{D^8} - \frac{D_0}{D^2}\right]$$

For pressure to be maximum $\frac{dP}{dD} = 0$ and $D = D_{max}$

$$\therefore \qquad C\left[\frac{7D_0^7}{D_{max}^8} - \frac{D_0}{D_{max}^2}\right] = 0$$

$$\frac{7}{D_{max}^8} - \frac{1}{D_{max}^2} = 0 \qquad [\text{As } D_0 = 1 \text{ m}]$$

$$7D_{max}^2 = D_{max}^8$$

$$7 = D_{max}^6$$

$$D_{max} = 7^{1/6} = 1.3831 \text{ m}$$

∴ $$V_{max} = \frac{\pi}{6}\times(D_{max})^3 = \frac{\pi}{6}\times(1.3831)^3 = 1.3853 \text{ m}^3$$

Also, $$\frac{P_1V_1}{T_1} = \frac{P_{max}\times V_{max}}{T_{max}}$$

$$T_{max} = \frac{P_{max}\times V_{max}\times T_1}{P_1\times V_1}$$

$$T_{max} = \frac{200\times 1.3853\times 250}{100\times 0.4} = 1731.625 \text{ K}$$

(ii)

On putting, D_{max} = 1.3831 m; P_{max} = 200 kPa and P_0 = 100 kPa in

$$P = P_0 + C\left[1-\left(\frac{D_0}{D}\right)^6\right]\left(\frac{D_0}{D}\right) \quad \text{...(i)}$$

We get, $$200 = 100 + C\left[1-\left(\frac{1}{1.3831}\right)^6\right]\left(\frac{1}{1.3831}\right)$$

$$C = 161.36$$

As $$V_2 = 2 \text{ m}^3$$

∴ $$\frac{\pi}{6}\times(D_2)^3 = 2$$

$$D_2 = 1.5632 \text{ m}$$

Again putting, $P = P_2$, C = 161.36 and D = 1.5632 in equation (i)

We get

$$P_2 = 100 + 161.36\left[1-\left(\frac{1}{1.5632}\right)^6\right]\left(\frac{1}{1.5632}\right) = 196.15 \text{ kPa}$$

Also, $$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

$$T_2 = \frac{P_2\times V_2\times T_1}{P_1\times V_1} = \frac{196.15\times 2\times 250}{100\times 0.4} = 2451.875 \text{ K}$$

Hence, the final pressure and temperature inside the balloon are P_2 = 196.15 kPa and T_2 = 2451.875 K

(iii) Work transfer during the process,

$$W = \int_1^2 PdV$$

$$W = P_0(V_0 - V_1) + \int_{V_0}^{V_2} PdV$$

$$W = P_0(V_0 - V_1) + P_0(V_2 - V_0) + C\int_{V_0}^{V_2}\left[\frac{D_0}{D} - \left(\frac{D_0}{D}\right)^7\right]dV$$

Also,

$$V = \frac{\pi}{6}D^3$$

$$dV = \frac{3\pi}{6}D^2 \cdot dD$$

∴

$$W = 100(0.5236 - 0.4) + 100(2 - 0.5236) + 161.6\int_{D=1}^{D=1.5632}\left[\frac{D_0}{D} - \left(\frac{D_0}{D}\right)^7\right] \times \frac{\pi}{2}D^2 \cdot dD$$

$$W = 100(2 - 0.4) + \frac{161.36 \times \pi}{2}\int_1^{1.5632}\left[D_0 D - \frac{D_0^7}{D^5}\right]dD$$

$$W = 160 + 253.462\left[\frac{D_0 D^2}{2} + \frac{D_0^7}{4D^4}\right]_1^{1.5632}$$

$$W = 160 + 253.462\left[\frac{(1.5632)^2}{2} + \frac{1}{4(1.5632)^4} - \frac{1}{2} - \frac{1}{4}\right] \qquad [\text{As } D_0 = 1]$$

$$W = 160 + 253.462 \times 0.5136$$

$$W = 290.19 \text{ kJ}$$

From first law,

$$Q = \Delta U + W$$

$$Q = m(u_2 - u_1) + W = m(C_v)_{He}[T_2 - T_1] + W$$

$$Q = 0.077 \times 3.1156 \times [2451.875 - 250] + 290.19$$

$$Q = 818.42 \text{ kJ}$$

Hence, the work transfer and heat transfer for the overall process are

$$W = 290.19 \text{ kJ and } Q = 818.42 \text{ kJ}$$

4. (a)(i)

For air at 35°C DBT and 60% RH

$$\phi = \frac{p_v}{p_{vs}}$$

from steam table at 35°C, $p_{vs} = 0.05629$ bar

$$p_v = \phi p_{vs}$$

$$= 0.6 \times 0.05629 = 0.03377 \text{ bar}$$

So,
$$\omega = \frac{0.622 \times p_v}{p_t - p_v} = \frac{0.622 \times 0.03377}{1.033 - 0.03377}$$

$$\omega = 0.02102 \text{ kg/kg of dry air}$$

and
$$\text{enthalpy, } h = c_p t + \omega(2500 + 1.88t) \text{ kJ/kg dry air}$$

$$= 1.005 \times 35 + 0.02102(2500 + 1.88 \times 35)$$

$$h = 89.11 \text{ kJ/kg of dry air}$$

For the air at 22°C DBT and 55% RH

$$p_v = p_{vs} \times \phi$$

From steam tables at 22°C,

$$p_{vs} = 0.026453 \text{ bar}$$

$$p_v = 0.55 \times 0.026453 = 0.01455$$

$$\omega = \frac{0.622 \times p_v}{p_t - p_v} = \frac{0.622 \times 0.01455}{1.033 - 0.01455}$$

$$\omega = 0.00889 \text{ kg/kg of dry air}$$

∴
$$h = 1.005 \times 22 + 0.00889(2500 + 1.88 \times 22)$$

$$h = 44.7 \text{ kJ/kg of dry air}$$

The due point temperature of air at 22°C DBT and 55% RH is the saturation temperature corresponding to the pressure $p_v = 0.01455$

From steam table, using interpolation

$$\text{DPT} = 12.03°\text{C}$$

The atmospheric air is cooled until the temperature of the air becomes 12.03°C DBT and saturated and then it is heated until it comes to the required condition.

For air at 12.03° and 100% RH

$$p_v = p_{vs} = 0.01455$$

and also
$$\omega = 0.00889 \text{ kg/kg of dry air}$$

$$\therefore \quad h = 1.005 \times 12.03 + 0.00889[2500 + 1.88 \times 12.03]$$

$$h = 34.52 \text{ kJ/kg of dry air}$$

Mass of air in 50 m^3/min of air,

$$\dot{m}_a = \frac{(p_t - p_v)v}{RT} = \frac{(1.033 - 0.03377) \times 10^5 \times 60}{287 \times (35 + 273)}$$

$$\dot{m}_a = 56.52 \text{ kg/min}$$

1. Capacity of cooling coil $= \dfrac{56.52(89.11 - 34.52)}{60 \times 3.5} = 14.692$ kW

2. Capacity of heating coil $= \dfrac{56.52(44.7 - 34.52)}{60} = 9.59$ kW

3. Amount of water vapour added to air per minute

$$= m_a \Delta w$$

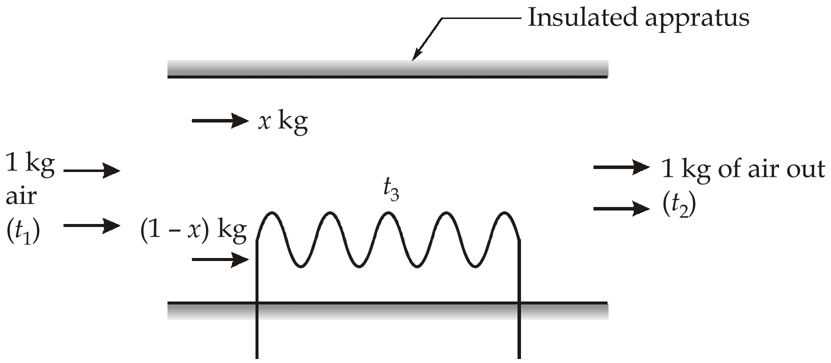
$$= 56.52(0.02102 - 0.00889) = 0.685 \text{ kg/min}$$

4. (a)(ii)

The bypass factor (BPF) of a coil in a cooling or heating system is a measure of how effectively that air is cooled or heated by the coil. It represents the fraction of air that bypasses the coil without being affected by the cooling or heating process. A higher BPF indicates that more air is bypassing the coil, resulting in less efficient cooling or heating. The amount of air that bypasses or the bypass factor depends upon the following factors.

1. The number of rows in a coil in the direction of flow.
2. The number fins provided in a unit lenth.
3. The velocity of air.

Let 1 kg of air at a temperature t_1 is passed over the coil having its temperature (i.e. coil surface temperature) t_3 as shown in the figure below.



Considering that when air passes over a coil, some of it (say x kg) just bypasses unaffected while the remaining $(1 - x)$ kg comes in direct contact with the coil.

Balancing enthalpies, we get

$$xc_pt_1 + (1-x)c_pt_3 = 1 \times 1 \times c_pt_2 \qquad \text{(where } c_p = \text{specific humid heat)}$$

$$xt_1 + t_3 - xt_3 = t_2$$

$$x(t_1 - t_3) = t_2 - t_3$$

$$x = \frac{t_2 - t_3}{t_1 - t_3}$$

where x is the bypass factor (BPF)

For heating $$\text{BPF} = \frac{t_3 - t_2}{t_3 - t_1}$$

and for cooling $$\text{BPF} = \frac{t_2 - t_3}{t_1 - t_3}$$

4. (b) (i)

Let suffix 1 denotes values for CO_2 and suffix 2 denote values for N_2.

Given $n_1 = 3$; $n_2 = 5$; $P_1 = 110$ kPa; $P_2 = 2.2$ MPa; $T_1 = 20°C = 293$ K

1. Mass of the mixture,

$$m = n_1M_1 + n_2M_2$$

$$m = 3(44) + 5(28) = 272 \text{ kg}$$

Specific heat of mixture at constant volume,

$$c_v = x_1c_{v_1} + x_2c_{v_2} \qquad \text{(where } x \text{ is the mole fraction)}$$

$$c_v = \left(\frac{132}{272}\right) \times (0.653) + \left(\frac{140}{272}\right) \times (0.745)$$

$$c_v = 0.7 \text{ kJ/kgK}$$

and specific heat of mixture at constant pressure

$$c_p = \left(\frac{132}{272}\right) \times (0.842) + \left(\frac{140}{272}\right) \times (1.042)$$

$$c_p = 0.945 \text{ kJ/kgK}$$

The ratio of specific heats, $\gamma = \dfrac{c_p}{c_v} = \dfrac{0.945}{0.7}$

$$\gamma = 1.35$$

As the mixture is compressed adiabatically, so using relation

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

We get, Final temperature, $T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$

$$T_2 = 293 \times \left(\frac{2.2 \times 1000}{110}\right)^{\frac{0.35}{1.35}} = 637.05 \text{ K}$$

or $T_2 = 364.05°\text{C}$

2. For work

Applying first law of thermodynamics,

$$\Delta Q = dU + \Delta W$$

$$\Delta Q = 0, \text{ for adiabatic process}$$

$$\Delta W = -dU$$

$$= -mc_v dT$$

$$= -272 \times (0.7)(364.05 - 20)$$

$$= -65.507 \text{ MJ}$$

Ans.

3. The entropy change

$$\Delta s = c_p \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right)$$

$$= 0.945 \ln\left(\frac{637.05}{293}\right) - (0.945 - 0.7)\ln\left(\frac{2.2 \times 1000}{110}\right)$$

$$= 3.974 \times 10^{-6} \text{ kJ/kgK}$$

The entropy change should be zero for this isentropic process. The above small value of entropy change is a measure of the error in the calculations.

4. (b) (ii)

Given : $I = 0.6$ kg.m^2; $N_1 = 2100$ rpm; $m = 0.8$ kg; $T_0 = 18°\text{C} = 291$ K

Initial angular velocity of the flywheel,

$$\omega_1 = \frac{2 \times \pi \times N_1}{60} = \frac{2 \times \pi \times 2100}{60} = 219.91 \simeq 220 \text{ rad/s}$$

$$\omega_1 = 220 \text{ rad/s}$$

Initial available energy of the flywheel is equal to the initial kinetic energy of the flywheel

$$AE_1 = \frac{1}{2} \times I\omega_1^2 = \frac{1}{2} \times 0.6 \times (220)^2$$

$$AE_1 = 14.52 \text{ kJ}$$

The kinetic energy is dissipated as frictional heat that rises the temperature of shaft and bearings.

$$KE = mc_{pw}(\Delta T)$$

$$14.52 = 0.8 \times 4.187 \times \Delta T$$

$$\Delta T = 4.335°\text{C}$$

Thus, the final temperature of the bearings

$$T_2 = T_0 + \Delta T$$

$$= 18 + 4.335 = 22.335 \text{ C}$$

- The maximum energy returned to flywheel as available energy.

$$AE_2 = \Delta h - T_0\Delta s$$

$$= mc_{pw}\left[(T_2 - T_0) - T_0 \ln\left(\frac{T_2}{T_0}\right)\right]$$

$$= 0.8 \times 4.187\left[(22.335 - 18) - 291 \ln\left(\frac{273 + 22.335}{291}\right)\right]$$

$$= 0.107 \text{ kJ}$$ **Ans.**

- The amount of kinetic energy becoming unavailable.

$$UE = AE_1 - AE_2$$

$$= 14.52 - 0.107 = 14.413 \text{ kJ}$$

- The final rpm of flywheel

Available energy retained by flywheel = $\frac{1}{2}I\omega_2^2$

$$0.107 \times 10^3 = \frac{1}{2} \times 0.6 \times \omega_2^2$$

$$\omega_2 = 18.89 \text{ rad/s}$$

or $$N_2 = \frac{\omega_2 \times 60}{2\pi}$$

$$N_2 = 180.38 \text{ rpm}$$

4. (c) (i)

The variables that are affecting the performance characteristics are as follows:

1. **Combustion Rate and Spark Timing** : The spark should be timed and the combustion rate controlled such that the maximum pressure occurs as close to the beginning of the power stroke as possible, consistent with a smooth running engine. As a general rule, the spark timing and combustion rate are regulated in such a way that approximately one half of the total pressure rise due to combustion has occurred as the piston reaches TDC on the compression stroke.
2. **Air-Fuel Ratio :** This ratio must be set to fulfill engine requirements. Consistent with these requirements, however, it is usually set as close as possible to the best economy proportions during normal cruising speeds, and as close as possible to the best power proportions when maximum performance is required.
3. **Compression Ratio :** An increase in compression ratio increases the thermal efficiency, and is, therefore, generally advantageous. The compression ratio in most SI engines is limited by knock, and the use of economically feasible antiknock quality fuels. Increasing compression ratio also increases the friction of the engine, particularly between piston rings and the cylinder walls, and there is a point at which further increase in compression ratio would not be profitable, though this point appears to be rather high.
4. **Engine Speed :** At low speeds, a greater length of time is available for heat transfer to the cylinder walls and therefore a greater proportion of heat loss occurs. Up to a certain point, higher speeds produce greater air consumption and therefore greater ip. Higher speeds, however, are accompanied by rapidly increasing fp and by greater inertia in the moving parts. Consequently, the engine speed range must be a compromise, although most present day designs appear to favour the higher speeds.
5. **Mass of Inducted Charge :** The greater the mass of the charge inducted, the higher the power produced. For a given engine, the geometry is fixed, and it is desirable to induct a charge to a maximum possible density giving the highest volumetric efficiency.
6. **Heat Losses :** It should be noted that the large proportion of the available energy is lost in a non-usable form, i.e., heat losses. Any method which can be employed to prevent the excessive heat loss and cause this energy to leave the engine is a usable form will tend to increase engine performance. Higher coolant temperatures, for instance, provide a smaller temperature gradient around combustion chamber walls and a reduction in heat loss, but are limited by the possibility of damage to engine parts.

7. **Scavenging Efficiency :** In case of two-stroke engines scavenging efficiency is defined as the ratio of the amount of air or gas-air mixture, which remains in the cylinder, at the actual beginning of the compression to the product of the total volume and air density of the inlet. Scavenging efficiency for most of the two-stroke engines varies from 40 to 95 per cent depending upon the type of scavenging provided.

8. **Charge Efficiency :** The charge efficiency shows how well the piston displacement of a four-stroke engine is utilized. Various factors affecting charge efficiency are:
   (i) the compression ratio.
   (ii) the amount of heat picked up during passage of the charge through intake manifold.
   (iii) the valve timing of the engine.
   (iv) the resistance offered to air-fuel charge during its passage through induction manifold.

9. **Combustion Efficiency :** Combustion efficiency is the ratio of heat liberated to the theoretical heat in the fuel. The amount of heat liberated is less than the theoretical value because of incomplete combustion either due to dissociation or due to lack of available oxygen. Combustion efficiency in a well adjusted engine varies from 92% to 97%.

4. (c) (ii)

Given : Fuel consumption = 2.3 kg/hr; N = 1400 rpm;

$$\rho_f = s_f \times 1000 = 0.88 \times 1000 = 880 \text{ kg/m}^3$$

The volume of the fuel injected per cycle $= \dfrac{2.3}{60 \times 880 \times \dfrac{1400}{2}} = 0.622 \times 10^{-7} \text{ m}^3/\text{cycle}$

Since fuel is an incompressible liquid, the velocity of injection can be obtained by using Bernoulli equation,

So,
$$V = c_d\sqrt{2 \times \left(\frac{\Delta P}{\rho_f}\right)} = 0.88 \times \sqrt{\frac{2 \times (135-25) \times 10^5}{880}}$$
$$= 139.14 \text{ m/s}$$

$$\text{Duration of injection} = \frac{25}{360 \times \dfrac{1400}{60}} = 0.298 \times 10^{-2} \text{ sec}$$

$$\therefore \quad \text{The nozzle orifice area} = \frac{\text{Volume of fuel injected per cycle}}{\text{Injection velocity} \times \text{Injection time}}$$

$$= \frac{0.622 \times 10^{-7}}{139.14 \times 0.298 \times 10^{-2}}$$

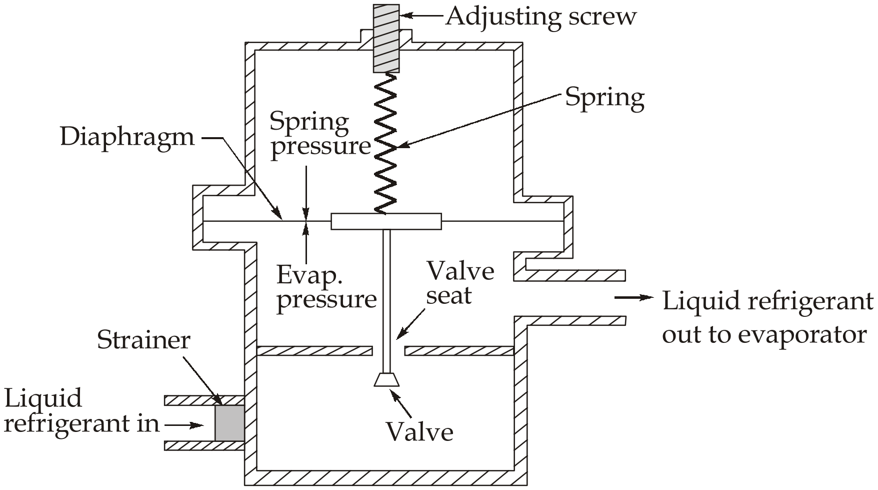
$$\frac{\pi}{4}d^2 = 1.5 \times 10^{-7}$$

$$d = \sqrt{\frac{1.5\times10^{-7}\times4}{\pi}}\times1000 \text{ mm}$$

$$d = 0.437 \text{ mm}$$

Section B : Thermodynamics + Refrigeration and Air-conditioning + IC Engine

5. (a)



Automatic (or constant pressure) expansion valve

The automatic expansion valve is also known as constant pressure expansion valve, because it maintains constant evaporator pressure regardless of the load on the evaporator. Its main moving force is the evaporator pressure. It is used with dry expansion evaporators where the load is relatively constant.

The automatic expansion valve, as shown in figure, consists of a needle valve and a seat (which forms an orifice), a metallic diaphragm or bellows, spring and an adjusting screw. The opening and closing of the valve with respect to the seat depends upon the following two opposing forces acting on the diaphragm :

1. The spring pressure and atmospheric pressure acting on the top of the diaphragm, and
2. The evaporator pressure acting below the diaphragm.

When the compressor is running, the valve maintains an evaporator pressure in equilibrium with the spring pressure and the atmospheric pressure. The spring pressure can be varied by adjusting the tension of the spring with the help of spring adjusting screw. Once the spring is adjusted for a desired evaporator pressure, then the valve operates automatically to maintain constant evaporator pressure by controlling the flow of refrigerant to the evaporator.

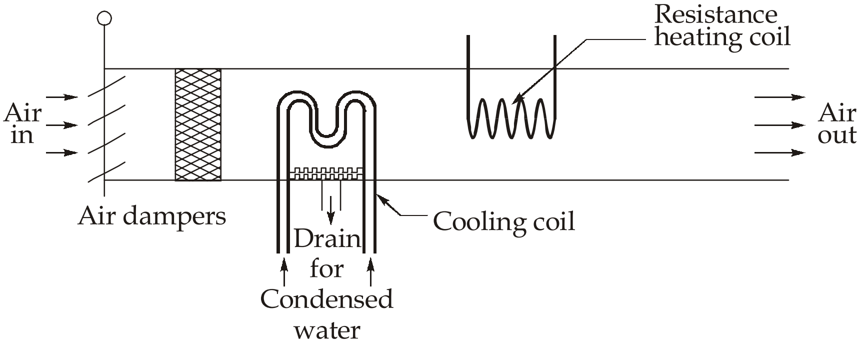
When the evaporator pressure falls down, the diaphragm moves downwards to open the valve. This allows more liquid refrigerant to enter into the evaporator and thus increasing the evaporator pressure till the desired evaporator pressure is reached. On the other hand, when the evaporator pressure rises, the diaphragm moves upwards to reduce the opening of the valve) This decreases the flow of liquid refrigerant to the evaporator which, in turn, lowers the evaporator pressure till the desired evaporator pressure is reached.

When the compressor stops, the liquid refrigerant continues to flow into the evaporator and increases the pressure in the evaporator. This increase in evaporator pressure causes the diaphragm to move upwards and the valve is closed. It remains closed until the compressor starts again and reduces the pressure in the evaporator.

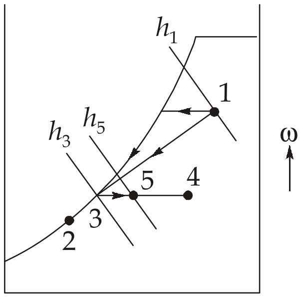
5. (b)

Summer Air-conditioning System for Hot and Humid Outdoor Conditions for eg. In summer conditions with hot and humid outdoors, air conditioning must be designed to remove both sensible and latent heat effectively, ensuring indoor thermal comfort, humidity control, and energy efficiency. Proper system design, component selection, and control strategies are essential for optimal performance.

The average summer conditions at Bombay are 32°C DBT and 75% Relative humidity. The required comfort conditions are same as 24°C DBT and 60% R.H. The arrangements of the equipments required and presentation of psychrometric processes are shown in figure.



Summer air-conditioning system for hot and wet weather



Representation of Psychrometric process

1 = Atmospheric condition

5 = Required comfort condition.

Such systems are used for hot and humid outdoor conditions like Bombay, Madras, Calcutta and identical places.

First air is passed through the cooling coil for dehumidification and it comes out at condition 3 leaving the cooling coil as the cooling coil temperature is below the dew point temperature of incoming air. The bypass factor of the cooling coil is given by $B_c = \frac{\text{Distance (2 - 3)}}{\text{Distance (2 - 1)}}$ and the capacity of the cooling coil is given by

$$\text{Capacity of cooling coil} = \frac{V(h_1 - h_3)}{v_s(3.5)}\text{tons}$$

where V is the volume of atmospheric air circulated in the air-conditioned system per second and v_s is the specific volume of atmospheric air.

The air enters the heating coil at condition 3 and comes out at condition 5 then the bypass factor of the heating coil and its capacity are given by

$$B_h = \frac{\text{Distance (5 - 4)}}{\text{Distance (3 - 4)}}$$

$$\text{and capacity of heating coil} = \frac{V}{v_s}(h_5 - h_3)\ \text{kJ/sec}$$

5. (c)

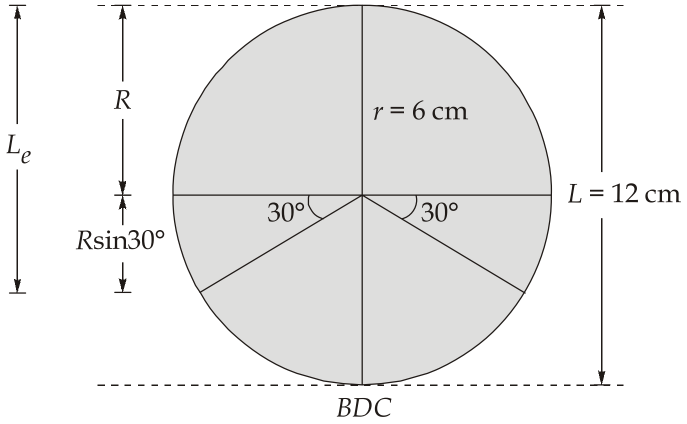
Given : $d = 9$ cm; $L = 12$ cm; $r = 9$; $\frac{\dot{m}_a}{\dot{m}_f} = 16$; $T_1 = 300$ K; $P = 1$ bar; $\dot{m}_a = 180$ kg/hr;

$R = 287$ J/kgK; N = 3800 rpm

$$\text{Mass of fuel, } \dot{m}_f = \frac{\dot{m}_a}{16} = \frac{180}{16} = 11.25$$

$$\text{Supply fuel, } \dot{m}_{\text{supp}} = \dot{m}_a + \dot{m}_f$$

$$= 180 + 11.25 = 191.25\ \text{kg/h}$$



As we have to make all the calculations according to the effective stroke.

$$R = \frac{L}{2} = \frac{12}{2} = 6 \text{ cm}$$

$$\text{Effective stroke, } L_e = R + R\sin\theta$$

$$= 6 + 6\sin 30° = 9 \text{ cm}$$

Swept volume corresponding to L_e,

$$V_{se} = \frac{\pi}{4} \times (0.09)^2 \times 0.09 = 5.73 \times 10^{-4} \text{ m}^3$$

Total cylinder volume corresponding to L_e,

$$V = \left(\frac{r}{r-1}\right) V_{se} = \frac{9}{8} \times \left(5.73 \times 10^{-4}\right) = 6.44 \times 10^{-4} \text{ m}^3$$

$$\text{Density, } \rho = \frac{P}{RT} = \frac{1 \times 10^5}{287 \times 300} = 1.16 \text{ kg/m}^3$$

$$\text{Mass of the mixture per cycle} = 6.44 \times 10^{-4} \times 1.16$$

$$m = 7.47 \times 10^{-4}$$

$$\text{Ideal rate of mass flow, } \dot{m}_{ideal} = m \times N \times 60$$

$$= 7.47 \times 10^{-4} \times 3800 \times 60 = 170.32 \text{ kg/hr}$$

$$\therefore \quad \text{Scavenging ratio, } R_{sc} = \frac{\dot{m}_{\text{supp}}}{\dot{m}_{\text{ideal}}} = \frac{191.25}{170.32} = 1.12$$ **Ans.**

$$\text{Scavenging efficiency, } \eta_{sc} = 1 - e^{-R_{sc}} = 1 - e^{-1.12} = 0.6737$$

$$\eta_{sc} = 67.37\%$$ **Ans.**

$$\text{Trapping efficiency, } \eta_{tr} = \frac{\eta_{sc}}{R_{sc}} = \frac{67.37}{1.12} = 60.15\%$$ **Ans.**

5. (d)

Given : P_1 = 20.98 MPa; T_c = 647.3 K; T_1 = 633.22°C = 906.22 K; P_c = 220.9 bar;
T_c = 439.03°C = 712.03; m = 18.02 kg/Kmol

Water vapour is a closed system because it is contained in a rigid closed container.

(i)

$$\text{Reduced temperature, } T_{R_1} = \frac{906.22}{647.3} = 1.4$$

and $$\text{Reduced pressure, } P_{R_1} = \frac{20.98}{22.09} = 0.949 \simeq 0.95$$

For these values of reduced temperature and reduced pressure, the value of z obtained from the compressibility chart is approximately 0.9.

We know that $$z = \frac{Pv}{RT}$$

So, the specific volume at state 1 can be determined as follows:

$$v_1 = z_1 \frac{RT_1}{P_1}$$

$$= 0.9\left[\frac{8314}{18.02}\right] \times \left[\frac{906.22}{20.98 \times 10^6}\right] = 0.01794 \text{ m}^3/\text{kg}$$

(ii)

Since both mass and volume remain constant, the water vapour cools at constant specific volume and thus, at constant V_R', using the value for specific volume determine in part (i), the constant V_R' value is

$$V_R' = \frac{vP_c}{RT_c}$$

$$= \frac{0.01794 \times \left(22.09 \times 10^6\right)}{\left(\frac{8314}{18.02}\right) \times 647.3} = 1.33$$

and $$T_{R_2} = \frac{712.03}{647.3} = 1.10$$

Locating the point on the compressibility chart where $V_R' = 1.33$ and $T_R = 1.10$, the corresponding value for P_R is approximately 0.68.

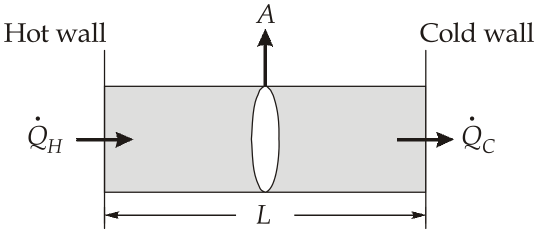
So,
$$P_2 = P_c(P_{R_2})$$
$$= 22.09 \times 0.68 = 15.02 \text{ MPa}$$ **Ans.**

5. **(e)**

(i)

Given : Energy conducted into the rod,

$$\dot{Q}_H = \frac{kA(T_H - T_C)}{L}$$



At steady state, entropy rate balance gives

$$\frac{ds}{dt} = s_i - s_e + s_{gen}$$

$$s_{gen} = s_e - s_i \qquad \left(\frac{ds}{dt} = 0, \text{ because steady state}\right)$$

$$= \frac{\dot{Q}_C}{T_C} - \frac{\dot{Q}_H}{T_H}$$

The energy transfer are positive in the directions of the arrows, energy rate balance gives

$$\dot{Q}_C = \dot{Q}_H$$

So,
$$s_{gen} = Q_H\left(\frac{1}{T_C} - \frac{1}{T_H}\right) = \frac{kA(T_H - T_C)}{L}\left[\frac{T_H - T_C}{T_H T_C}\right]$$

$$s_{gen} = \frac{kA(T_H - T_C)^2}{L T_H T_C}$$ **Ans.**

(ii)

We need to plot the heat transfer rate $\dot{Q}_H$ (in kW) and time rate of entropy s_{gen} in (kW/K) versus L.

$$\dot{Q}_H = \frac{kA(T_H - T_C)}{L}$$

Putting values of k, A, T_H and T_C in above equation we have

$$\dot{Q}_H = \frac{0.4 \times 0.1 \times (427 - 127)}{L} = \frac{12}{L}$$

The above equation is the equation of rectangular hyperbola (xy = constant) and is given by

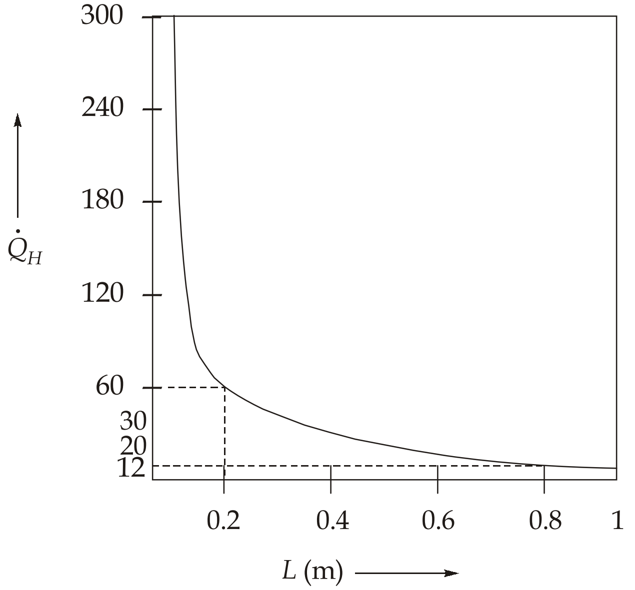
$$\dot{Q}_H \times L = 12$$

Hence, at $L = 0.2,\ \dot{Q}_H = 60$

$L = 0.4,\ \dot{Q}_H = 30$

$L = 0.6,\ \dot{Q}_H = 20$

and $L = 1,\ \dot{Q}_H = 12$



For time rate of entropy generation,

$$\dot{s}_{gen} = \frac{kA(T_H - T_C)^2}{LT_HT_C}$$

$$\dot{s}_{gen} = \frac{0.4 \times 0.1(300)^2}{700 \times 400 \times L} = \frac{0.01286}{L}$$

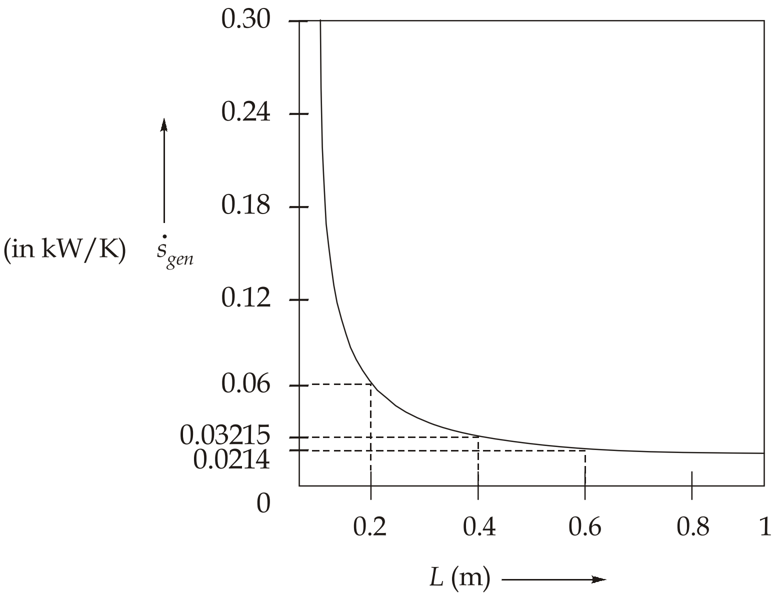
The above equation is also for rectangular hyperbola.

At $L = 0.2;\ \dot{s}_{gen} = 0.0643$

$L = 0.4;\ \dot{s}_{gen} = 0.03215$

$L = 0.6;\ \dot{s}_{gen} = 0.0214$

$L = 1;\ \dot{s}_{gen} = 0.257$



6. **(a)**

Given : For CO : $m = 4$ kg; $P = 2$ bar; $T = 450$ K

For Air : $m = 2$ kg; $P = 5$ bar; $T = 350$ K

Consider the container contents to be the system.

For the system, $W = Q = 0$

From first law, $\Delta Q = dU + \Delta W$

$$dU = 0$$

$$dU = (dU)_{\text{air}} + (dU)_{\text{CO}}$$

$$(dU)_{\text{air}} + (dU)_{\text{CO}} = 0$$

$(mc_v)_{\text{air}}\,(T - 350) + (mc_v)_{\text{CO}}(T - 450) = 0$ ...(i)

For specific heats at constant volume for both the gases,

$$(c_v)_{CO} = \frac{R}{\gamma - 1} = \frac{8.314/28}{1.395 - 1} = 0.752 \text{ kJ/kgK}$$

and
$$(c_v)_{air} = \frac{R}{\gamma - 1} = \frac{8.314/29}{1.395 - 1} = 0.726 \text{ kJ/kgK}$$

Putting value of $(c_v)_{air}$ and $(c_v)_{CO}$ in equation (i), we get

$$(2 \times 0.726)(T - 350) = (4 \times 0.752)(450 - T)$$

$$T = \frac{(1.452 \times 350) + (3.008 \times 450)}{3.008 + 1.452}$$

$$T = 417.44 \text{ K}$$ **Ans.**

(ii)

The partition can freely move, but the total volume remains constant,

where
$$V = V_{CO} + V_{Air}$$

and
$$V_{air} = \left(\frac{mRT}{P}\right)_{air} = \frac{2 \times \left(\frac{8.314}{29}\right) \times 350}{5 \times 100} = 0.401 \text{ m}^3$$

and
$$V_{CO} = \left(\frac{mRT}{P}\right)_{CO} = \frac{4 \times \left(\frac{8.314}{28}\right) \times 450}{2 \times 100} = 2.67 \text{ m}^3$$

$$\text{Total volume, } V = V_{CO} + V_{air} = 2.67 + 0.401 = 3.071 \text{ m}^3$$

At equilibrium, each gas is at $T = 417.44$ K and pressure P

Also,
$$V = \frac{(mR)_{air} T}{P} + \frac{(mR)_{CO} T}{P}$$

$$3.07 = \frac{T}{P}\left[(mR)_{air} + (mR)_{CO}\right]$$

$$P = \frac{417.44}{3.07}\left[2 \times \left(\frac{8.314}{29}\right) + 4 \times \left(\frac{8.314}{28}\right)\right]$$

$$= 239.463 \text{ kPa} = 2.394 \text{ bar}$$ **Ans.**

(iii)

The final volume occupied by the air and CO are respectively.

$$V_{air} = \frac{m_{air} \times RT}{P} = \frac{2 \times \left(\frac{8.314}{29}\right) \times (417.44)}{239.463} = 0.999$$

$$V_{air} \simeq 1 \text{ m}^3$$

$$V_{CO} = \frac{m_{CO} \times RT}{P} = \frac{4 \times \left(\frac{8.314}{28}\right) \times (417.44)}{239.463}$$

$$V_{CO} = 2.07 \text{ m}^3$$

6. (b) (i)

The possible causes of the observed differences between the actual cycle and the fuel air cycle include the following losses that are taken into account for the analysis of the actual cycle, and which are not considered in the analysis of the fuel-air cycle.

- **Leakage** : At higher piston speeds, leakage is usually insignificant in a well adjusted engine. However, at low piston speeds and high gas pressure, the gas flows into the regions between the piston, piston rings and cylinder walls and gets cooled by heat transfer through cylinder walls. These regions are called crevice regions. The gases flowing into there regions usually remain unburned and some of the gases return to the cylinder during the later part of the expansion stroke and the remaining gas leaks past the piston rings to the crank case. Leakage can be estimated by measuring blow by, that is, the mass of the gases flowing out from the crank case breather. This leakage loss reduces the cylinder pressure during combustion and during the early part of the expansion stroke, thus reducing the net power output of the engine.
- **Imperfect Mixing of Fuel and Air** : In practice, it is not possible to obtain a perfect homogeneous mixture of fuel, air and residual gases in the cylinder before the ignition takes place, because of insufficient turbulence. In one part of the cylinder, there may be excess oxygen and in another part excess fuel may be present. The excess fuel may not find enough oxygen for complete combustion, which may result in the appearance of CO, H_2 and unburned fuel in the exhaust. The efficiency of the engine will decrease because of the wastage of fuel. The wastage of fuel can be reduced by using a lean mixture. This will ensure the complete utilization of fuel, thus providing maximum economy.
- **Progressive Burning** : In the analysis of constant volume fuel-air cycle it is assumed that the ignition takes place at top dead centre and combustion is instantaneous. In an actual SI engine, combustion starts at a certain point and continues by moving

the flame front. Combustion is complete when the flame front has passed through the entire charge. It takes some time in doing so. The time required for this varies with the fuel composition, the combustion chamber shape and size including the number and position of the ignition point and engine operating conditions. Different amounts of charge burn at different times even if the piston is nearly stationery during combustion. The spark ignites a very small portion of the charge immediately adjacent to it. The flame then spreads progressively throughout the mixture. This phenomenon of burning is called progressive burning.

- **Burning Time Losses** : The crankshaft normally rotates through 40° or more between the time the spark is produced and the time the charge is completely burned. The time in degree crank angle (°CA) depends upon the flame speed and the distance between the position of spark plug and the farthest side of combustion chamber. The flame travel distance can be reduced by locating the spark plug at the centre of the cylinder head or by using more than one spark plug. A hemispherical combustion chamber often uses two spark plugs mounted on the cylinder head on opposite sides to reduce the flame travel distance. The motion of the flame front depends upon how fast the heat is transferred from the flame front to the unburned mixture just ahead of the flame front. Heat is generated by the chemical reaction at the flame front.
- **Heat Losses to the Cylinder Walls :** During combustion and early part of the expansion stroke, the temperature of the cylinder gases is high and a considerable amount of heat flows from the hot gases through the cylinder walls and cylinder head into the water jacket in water-cooled engines or cooling fins in air-cooled engines. Some heat enters into the piston head, and from there it flows to the piston rings. As the piston rings are in contact with the cylinder walls, so the heat flows to the cylinder walls. Out of this heat, some heat is taken away by the engine oil present between the piston rings and the cylinder walls for lubrication. The rest of the heat goes to the cooling media.

  As a result of loss of heat to the cylinder walls the work and efficiency of the cycle are reduced because some of the heat energy liberated by combustion is not utilized for producing work during expansion. However, the loss of heat during combustion reduces the maximum temperature and the dissociation of CO_2 and H_2O is also reduced, which results in complete combustion, but the improvement in work and efficiency is only marginal.
- **Exhaust Blowdown Loss** : The pressure at the end of the expansion stroke is much higher than the atmospheric pressure. If the exhaust valve is opened at bottom dead centre, the piston would have to do a large amount of work in order to expel

the high pressure exhaust gases during the exhaust stroke. If the exhaust valve is opened too early, part of the expansion work is lost. The best position is to open the exhaust valve 40° to 70° before BDC. In this case too, some expansion work is lost but the work spent by the piston during the exhaust stroke is reduced. The net result will be the gain in some work. The early opening of the exhaust valve releases the pressure of the gas before the piston reaches BDC. This process is called exhaust blowdown.

- **Fluid Friction** : At high engine speeds, turbulence inside the engine cylinder causes friction between fluid particles. During suction and exhaust processes, fluid friction is appreciable. Near the end of the compression process and during the initial part of the expansion process, fluid friction is high because of higher pressure between the fluid particles. During combustion, flame speed is turbulent, which causes more fluid friction. The overall effect of fluid friction on the actual cycle is very little.
- **Gas Exchange or Pumping Loss :** The purpose of the gas exchange process is to admit the fresh charge during the suction stroke and remove the burned gases at the end of the expansion stroke. During the induction process, pressure losses occur as the charge passes through the air filter, the carburettor and the intake manifold. There is additional pressure drop across the inlet valve. The drop in pressure along the intake system also depends on the engine speed. The pressure during the suction stroke is below atmospheric. The exhaust system consists of an exhaust manifold, an exhaust pipe, and often a catalytic converter for emission control, and a muffler or silencer. The burned cylinder gases get expelled because of the pressure difference between the cylinder and the exhaust system. The pressure during the exhaust stroke is much higher than the atmospheric.

6. (b) (ii)

Given : Bore, d = 10 cm; Stroke, $L = d$ = 10 cm (square engine)

$$\text{Crank radius, } a = \frac{L}{2} = 5 \text{ cm}$$

$$\text{Connecting rod length, } l = 18 \text{ cm}$$

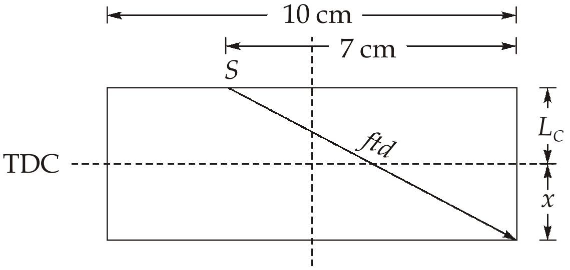
(1) The piston displacement from TDC position,

$$x = l + a - a\cos\theta - \sqrt{l^2 - a^2\sin^2\theta}$$

$$= 18 + 5 - 5\cos 14° - \sqrt{18^2 - 5^2\sin^2 14°}$$

$$x = 0.189 \text{ cm}$$ **Ans.**

(2)



As clearance volume,
$$V_C = \frac{V_S}{r-1}$$

$$L_C = \frac{L}{r-1} = \frac{10}{7} = 1.43 \text{ cm}$$

$$L_C + x = 1.43 + 0.189$$
$$= 1.62 \text{ cm}$$

Flame travel distance, $ftd = \sqrt{7^2 + (1.62)^2} = 7.185$ cm **Ans.**

(3)

$$\text{Ignition lag} = 7°$$

$$\text{Combustion starts at an angle} = 20 - 7 = 13° \text{ bTDC}$$

Rotational angle during flame propagation from

$$13° \text{ bTDC to } 14° \text{ aTDC} = 13 + 14 = 27°$$

$$N = \frac{2200}{60} = 36.67 \text{ rps}$$

$$= 36.67 \times 360° = 13200 \text{ degree/s}$$

Time of flame propagation, $t = \dfrac{27}{13200} = 0.002045$ sec

Effective flame speed, $V_f = \dfrac{ftd}{t} = \dfrac{7.185}{0.002045} = 3512.67$ cm/s

$V_f = 35.13$ m/s **Ans.**

6. (c)

For (U × R) to be minimum and the figure of merit to be maximum

we have,
$$x = \frac{A_p L_n}{A_n L_p} = \sqrt{\frac{k_n \rho_p}{k_p \rho_m}}$$

We have, $L_p = L_n, k_p = k_n \text{ and } \sigma = \dfrac{1}{\rho}$

Therefore, $$\frac{A_p}{A_n} = \sqrt{\frac{\rho_n k_n}{\rho_p k_p}} = \sqrt{\frac{1500}{1000}} = 1.225$$

$$A_p = \frac{\pi(0.01)^2}{4} = 7.85 \times 10^{-5} \text{m}^2$$

$$\therefore \quad A_n = \frac{A_p}{1.225} = 6.41 \times 10^{-5} \text{m}^2$$

$$d_n = \sqrt{\frac{4A_n}{\pi}} = \sqrt{\frac{4 \times 6.41 \times 10^{-5}}{\pi}}$$

$$d_n = 9.034 \times 10^{-3} \text{ m}$$

with $L_p = L_n, k_p = k_n$

We get, $$U = \frac{(A_p + A_n)k}{L} = \frac{(7.85 \times 10^{-5} + 6.41 \times 10^{-5}) \times 1.2}{0.01}$$

$$= 0.0171 \text{ W/K}$$

$$R = \frac{L\left(\dfrac{1}{\sigma_p A_p} + \dfrac{1}{\sigma_n A_n}\right)}{100}$$

$$= \frac{0.01\left(\dfrac{1}{1000 \times 7.85 \times 10^5} + \dfrac{1}{1500 \times 6.41 \times 10^{-5}}\right)}{100}$$

$$R = 2.314 \times 10^{-3} \text{ Ohm}$$

$$U \times R = 0.0171 \times 2.314 \times 10^{-3}$$

$$= 3.96 \times 10^{-5} \text{ W-Ohm/K}$$

$$\alpha_{pn} = \alpha_p - \alpha_n = 0.00035 \text{ V/K}$$

$$z = \frac{\alpha_{pn}^2}{UR} = \frac{0.00035^2}{3.96 \times 10^{-5}} = 3.09 \times 10^{-3} \text{k}^{-1}$$

$$T_m = \frac{T_h + T_c}{2} = \frac{258 + 313}{2} = 285.5 \text{ K}$$

For the case of maximum COP (ξ_{max})

$$\xi_{\max} = \frac{T_c\left[\sqrt{1+zT_m}-\frac{T_h}{T_c}\right]}{(T_h-T_c)\left[\sqrt{1+zT_m}+1\right]}$$

$$= \frac{258\left[\sqrt{1+3.09\times10^{-3}\times285.5}-\left(\frac{313}{258}\right)\right]}{(313-258)\left[\sqrt{1+3.09\times10^{-3}\times285.5}+1\right]}$$

$$= \frac{258\times(1.372-1.2132)}{55\times(1.372+1)} = 0.3141$$

Also,

$$I_{\max} = \frac{\alpha_{pn}\Delta T}{R\left[\sqrt{1+zT_m}-1\right]} = \frac{0.00035\times55}{2.314\times10^{-3}(1.372-1)}$$

$$= 22.36 \text{ amps}$$

$$(Q_c)_{\text{per pair}} = \alpha_{pn}I_{\max}T_c-\left(I_{\max}^2\frac{R}{2}\right)-U(T_h-T_c)$$

$$= 0.00035\times22.36\times258-\left(\frac{22.36^2\times2.314\times10^{-3}}{2}\right)-0.0171\times55$$

$$(Q_c)_{\text{per pair}} = 0.5001 \simeq 0.5 \text{ W per pair}$$

For 20 W cooling, the number of thermocouple pairs required $= \frac{20}{0.5} = 40$ pairs

$$\therefore \quad \text{Power} = \frac{Q_c}{COP} = \frac{20}{0.3141} = 63.67 \text{ W}$$

For the case of maximum cooling,

We have,

$$\xi_{\max} = \frac{\left(\frac{zT_c^2}{2}\right)-(T_h-T_c)}{zT_hT_c}$$

$$= \frac{\left(\frac{3.09\times10^{-3}\times(258)^2}{2}\right)-55}{3.09\times10^{-3}\times313\times258} = 0.192$$

Also,

$$I_{\max} = \frac{\alpha_{pn}T_c}{R} = \frac{0.00035\times258}{2.314\times10^{-3}} = 39.023 \text{ amps}$$

$$(Q_{c,\,\max})_{\text{per pair}} = U\left[\frac{zT_c^2}{2} - (T_h - T_c)\right]$$

$$(Q_{c,\,\max})_{\text{per pair}} = 0.0171\left[\frac{3.09\times10^{-3}\times(258)^2}{2} - (55)\right]$$

$$(Q_{c,\,\max})_{\text{per pair}} = 0.82 \text{ W per pair}$$

For 20 W cooling, the number of thermocouple pairs required $= \dfrac{20}{0.82} = 24.39 \simeq 25$ pairs

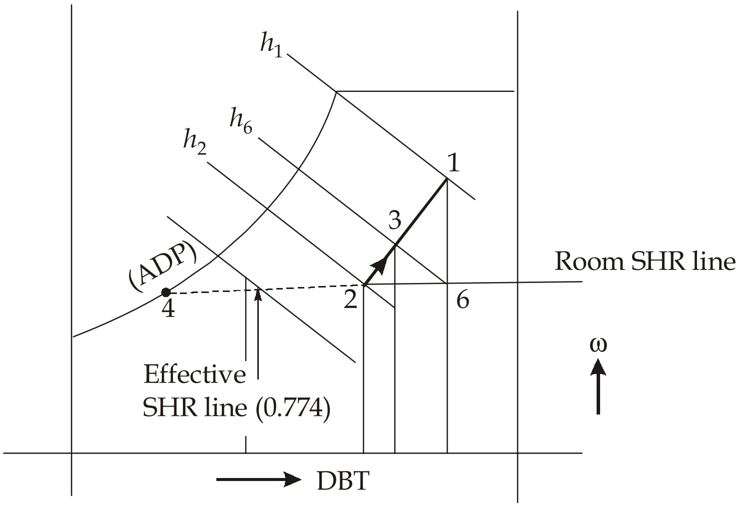
$$\text{Power} = \frac{Q_c}{COP} = \frac{20}{0.192} = 104.17 \text{ W}$$

7. **(a)**

Given : Inside conditions, 25°C DBT, 50% RH

Outside conditions, 36.5°C DBT, 26°C WBT, BPF = 0.2

Let us mark point (1) and (2) as outside and inside conditions on Psychrometric chart and obtain the following values.



$$h_1 = 80 \text{ kJ/kg}$$
$$h_2 = 50 \text{ kJ/kg}$$
$$h_6 = 63 \text{ kJ/kg}$$

and $v_1 = 0.9 \text{ m}^3\text{/kg}$

The mass of the fresh air circulated,

$$m_{a_1} = \frac{75}{60} \times 0.9 = 1.125 \text{ kg/s}$$

(i) Sensible heat load of outside air

$$= 1.125(h_6 - h_2)$$

$$= 1.125(63 - 50) = 14.625 \text{ kW}$$

Latent heat load of outside air $= 1.125(h_1 - h_6)$

$$= 1.125(80 - 63) = 19.125 \text{ kW}$$

Total heat load of outside air $= 14.625 + 19.125 = 33.75$ kW

(ii) Total sensible load taken by the plant

$$= 14.625 + 58 = 72.625 \text{ kW}$$

Total load to be taken by coil $= 19.125 + 14 = 33.125$ kW

∴ Total latent heat load taken by the plant

$$= 72.625 + 33.125 = 105.75 \text{ kW}$$

(iii) Effective room sensible heat load

$$= 58 + 0.2(14.625)$$

$$= 60.925 \text{ kW}$$

Effective room latent heat load $= 14 + 0.2(19.125)$

$$= 17.825 \text{ kW}$$

Effective sensible heat factor $= \dfrac{60.925}{60.925 + 17.825} = 0.774$

(iv) Now draw a SHR = 0.774 line through the point 2 as shown in figure which cuts the saturation line at point 4. The DBT at point 4 is apparatus dew point temperature (ADP).

$$(\text{ADP})_4 = 11.1°\text{C}$$

(v) Dehumidified air quality $= \dfrac{\text{Effective room sensible heat load}}{0.0204(T_s - T_{ADP})(1 - BPF)}$

$$= \frac{60.925}{0.0204(25 - 11.1)(1 - 0.2)} = 268.6 \text{ m}^3/\text{min}$$

(vi) Recirculated air $= 268.6 - 75 = 193.60$ m^3/min

∴ Percentage recirculated air $= \dfrac{193.60}{268.6} \times 100 = 72.07\%$

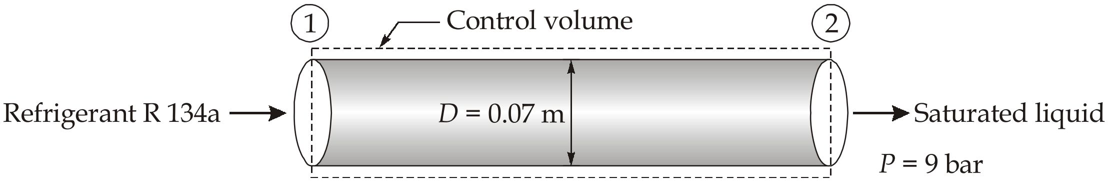
The air temperature entering the coil,

$$T = \frac{(193.60 \times 25) + (75 \times 36.5)}{268.6}$$

$$= 28.21°\text{C}$$

7. **(b)**

Given : $x_1 = 0.1$; $T_1 = 36°\text{C} = 309$ K; $C_1 = 12$ m/s



For the control volume

$$W_{cv} = 0$$

and potential energy change, $\Delta PE = 0$

(i)

For mass flow rate at inlet

$$\dot{m}_1 = \frac{A_1 C_1}{v_1} = \frac{\left(\frac{\pi}{4}D^2\right)C_1}{v_1} \quad \text{...(i)}$$

the quality of steam at inlet is, $x = 0.1$

$$v_1 = v_{f_1} + x_1(v_{g_1} - v_{f_1})$$

From refrigerant table, $v_{g_1} = 0.02238 \text{ m}^3/\text{kg}$

and $v_{f_1} = \frac{1}{1163.4} = 0.000859 \text{ m}^3/\text{kg}$

$$v_1 = 0.000859 + 0.1(0.02238 - 0.000859)$$

$$v_1 = 0.00301 \text{ m}^3/\text{kg}$$

Substituting the above value in equation (i), we have

$$\dot{m}_1 = \frac{\frac{\pi}{4}(0.07)^2 \times 12}{0.00301} = 15.343 \text{ kg/sec}$$

(ii)

From mass balance, $\dot{m}_1 = \dot{m}_2$

$$\frac{A_1 C_1}{v_1} = \frac{A_2 C_2}{v_2}$$

$$\frac{C_1}{v_1} = \frac{C_2}{v_2} \qquad \text{...(ii)}$$

As the exit of refrigerant is saturated liquid at 9 bar,

We need to find v_2

$$v_2 = v_{f2}$$

From refrigerant 134a table, using interpolation

$$\rho_{f_2} = \frac{1171.6 - 1163.4}{(0.86263 - 0.91185)} \times (0.9 - 0.91185) + 1163.4$$

$$\rho_{f_2} = 1165.37 \text{ kg/m}^3$$

So,

$$v_{f_2} = \frac{1}{\rho_{f_2}} = \frac{1}{1165.37} = 0.000858 \text{ m}^3\text{/kg} = v_2$$

Putting the above value in equation (ii), we have

$$\frac{12}{0.00301} = \frac{C_2}{0.000858}$$

$$C_2 = 3.42 \text{ m/s}$$ **Ans.**

(iii)

Rate of heat transfer,

From energy rate balance,

$$h_1 + \frac{C_1^2}{2} + gz_1 + Q_{cv} = h_2 + \frac{C_2^2}{2} + gz_2 + W_{cv}$$

Since $W_{cv} = 0$ and potential energy changes are negligible.

$$\dot{Q}_{cv} = \dot{m}\left[(h_2 - h_1) + \frac{C_2^2 - C_1^2}{2}\right]$$

Again from refrigerant 134a table at $T_1 = 36°\text{C}$, $h_{f_1} = 250.48$ kJ/kg and $h_{g_1} = 417.65$ kJ/kg

$$h_1 = h_{f_1} + x\left(h_{g_1} - h_{f_1}\right)$$

$$= 250.48 + 0.1(417.65 - 250.48)$$

$$= 267.197 \text{ kJ/kg}$$

Specific enthalpy for the saturated liquid refrigerant at the exit is obtained from the table

Using interpolation,

$$\frac{250.48-247.54}{0.91185-0.86263}\times(0.9-0.86263)+247.54 = h_{f_2} = h_2$$

$$h_2 = 249.77 \text{ kJ/kg}$$

$$\dot{Q}_{cv} = 15.343\times\left[(249.77-267.197)+\frac{(3.42)^2-(12)^2}{2000}\right]$$

$$\dot{Q}_{cv} = -268.397 \text{ kW}$$

Since the heat transfer is negative, therefore heat transfer is from the refrigerant.

7. **(c)**

Given : 4 stroke engine,

Bore, D = 16 cm; Stroke, L = 32 cm; Speed, N = 280 rpm

P_{im} = 8 bar; $\dot{m}$ = 2.5 kg/hr; C.V = 42 MJ/kg

$$\text{IP} = \frac{P_{im}\,LA\left(\frac{N}{2}\right)}{60\times1000} = \frac{8\times10^5\times0.32\times\frac{\pi}{4}(0.16^2)\times\frac{280}{2}}{60\times1000}$$

$$\text{IP} = 12 \text{ kW}$$

(i) Indicated thermal efficiency,

$$\eta_{\text{ith}} = \frac{IP}{\dot{m}_f\times C.V.}$$

$$\eta_{\text{ith}} = \frac{12\times3600}{2.5\times42\times10^3}\times100 = 41.14\%$$

For air, $$\dot{m}_a = \frac{2.5\times20}{60} = 0.833 \text{ kg/min}$$

$$\rho_a = \frac{P}{RT} = \frac{1\times10^5}{287\times293} = 1.19 \text{ kg/m}^3$$

$$\text{Volume of air consumed} = \frac{0.833}{1.19} = 0.7 \text{ m}^3\text{/min}$$

Swept volume per minute,

$$V_s \times \frac{N}{2} = \frac{\pi}{4}(0.16)^2 \times 0.32 \times \frac{280}{2} = 0.9 \text{ m}^3/\text{min}$$

$$\text{Volumetric efficiency, } \eta_v = \frac{0.7}{0.9} \times 100 = 77.78\%$$ **Ans.**

$$\text{Brake power, BP} = \frac{2\times\pi\times NT}{60\times1000} = \frac{2\times\pi\times280\times200}{60\times1000}$$

$$\text{BP} = 5.86 \text{ kW}$$

$$\text{Heat input} = \frac{2.5\times42000}{60} = 1750 \text{ kJ/min}$$

Heat equivalent of BP = 5.86 × 60 = 351.6 kJ/min

Heat in cooling water = 6 × 4.18 × 35 = 877.8 kJ/min

One kg of H_2 in the fuel will be converted to 9 kg of H_2O during combustion.

$$\text{Enthalpy of steam} = c_{pw}(100 - T_a) + h_{fg} + c_{pa}(t_{\text{sup}} - 100)$$

$$= 4.18(100 - 20) + 2250 + 2.1(400 - 100)$$

$$= 3214.4 \text{ kJ/kg}$$

$$\text{Heat carried by away by steam} = \dot{m}_s(h - c_{pw}t)$$

$$= 9\times0.16\times\frac{2.5}{60}[3214.4 - 4.18\times20]$$

$$= 187.85 \text{ kJ/min}$$

$$\text{Heat carried away by exhaust gases} = \left[0.833 + \frac{2.5}{60} - 9\times0.16\times\frac{2.5}{60}\right]\times1\times(400-20)$$

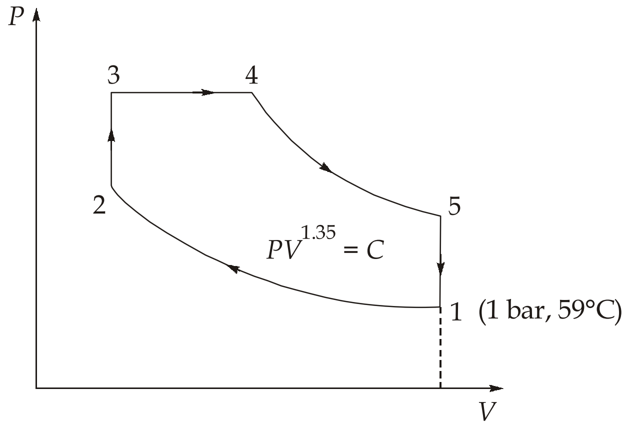
$$= 309.6 \text{ kJ/min}$$

Unaccounted loss = 1750 – (351.6 + 877.8 + 187.85 + 309.6)

= 23.15 kJ/min

Heat input (per minute)	kJ	Heat expenditure (per minute)	kJ
Heat Supplied by Fuel	1750	1. Heat equivalent to BP	351.6
		2. Heat loss to cooling water	877.8
		3. Heat carried away by steam	187.85
		4. Heat lost to exhaust	309.6
		5. Unaccounted loss	23.15
		Total	1750

8. (a)



Given : $r = 13$; cut off ratio, $\rho = 1.65$

$$P_{max} = P_3 = P_4 = 50 \text{ bar}$$
$$P_1 = 1 \text{ bar}$$
$$T_1 = 59°\text{C} = 332 \text{ K}$$
$$n = 1.35$$

(i)

Considering process 1 - 2

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{n-1}$$
$$T_2 = 332(13)^{1.35 - 1}$$
$$T_2 = 814.74 \text{ K}$$

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^{n}$$

$$P_2 = 1 \times (13)^{1.35} = 31.9 \text{ bar}$$

Considering process 2 - 3,

$$T_3 = T_2 \times \frac{P_3}{P_2} = 814.74 \times \frac{50}{31.9}$$
$$T_3 = 1277.022 \text{ K}$$

Considering process 3 - 4,

$$T_4 = T_3 \times \frac{V_4}{V_3} = T_3 \times \rho$$
$$= 1277.022 \times 1.65 = 2107.09 \text{ K}$$

Considering process 4 - 5,

$$\frac{T_5}{T_4} = \left(\frac{V_4}{V_5}\right)^{n-1} = \left(\frac{V_4}{V_3}\times\frac{V_3}{V_5}\right)^{n-1}$$

$$= \left(\frac{V_4}{V_3}\times\frac{V_3}{V_5}\right)^{n-1} = \left(\frac{\rho}{r}\right)^{n-1}$$

$$T_5 = 2107.09\times\left(\frac{1.65}{13}\right)^{0.35} = 1023.11 \text{ K}$$

(ii)

The compression and expansion process are polytropic, there will be heat transfer during the process as well.

Heat transfer during 1 - 2,

$$Q_{1-2} = \left(\frac{\gamma-n}{\gamma-1}\right)\left(\frac{P_1V_1-P_2V_2}{n-1}\right)$$

$$= \left(\frac{\gamma-n}{\gamma-1}\right)mR\left(\frac{T_1-T_2}{n-1}\right) \qquad [\gamma = 1.4 \text{ for air}]$$

For unit mass, $$Q_{1-2} = \left(\frac{1.4-1.35}{1.4-1}\right)\times 0.287\left(\frac{332-814.74}{1.35-1}\right)$$

$$Q_{1-2} = -49.48 \text{ kJ/kg}$$

Heat transfer during the process 2 - 3,

$$Q_{2-3} = c_v\,(T_3 - T_2)$$

$$= 1.005(1277.022 - 814.74)$$

$$Q_{2-3} = 464.593 \text{ kJ/kg}$$

For process 3 - 4,

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

$$= 1.005(2107.09 - 1277.022)$$

$$Q_{3-4} = 834.22 \text{ kJ/kg}$$

For process 4 - 5,

$$Q_{4-5} = \left(\frac{\gamma-n}{\gamma-1}\right)R\left(\frac{T_4-T_5}{n-1}\right)$$

$$= \left(\frac{\gamma - 1.35}{\gamma - 1}\right)(0.287)\left(\frac{2107.09 - 1023.11}{1.35 - 1}\right)$$

$$Q_{4-5} = 111.11 \text{ kJ/kg}$$

For process 5 - 1,

$$Q_{5-1} = c_v (T_1 - T_5)$$

$$= 0.718(332 - 1023.11)$$

$$Q_{5-1} = -496.22 \text{ kJ/kg}$$

$$\text{Work done, } W = \text{Heat supplied } (Q_S) - \text{Heat rejected } (Q_R)$$

$$= (464.593 + 834.22 + 111.11) - (496.22 + 49.48)$$

$$= 864.223 \text{ kJ/kg}$$

$$\text{Efficiency, } \eta = \frac{W}{Q_S} = \frac{864.223}{1409.923}$$

$$\eta = 0.6129 \text{ or } 61.29\%$$

(ii)

$$\text{Swept volume, } V_S = V_1 - V_2 = V_1\left(1 - \frac{1}{r}\right)$$

$$V_S = \frac{mRT_1}{P_1}\left(\frac{r-1}{r}\right) = \frac{287 \times 332}{1 \times 10^5}\left[\frac{12}{13}\right]$$

$$= 0.8795 \text{ m}^3/\text{kg}$$

$$\text{Mean effective pressure, } p_{mef} = \frac{W}{V_S} = \frac{864.223 \times 10^3}{0.8795}$$

$$= 9.83 \text{ bar}$$ **Ans.**

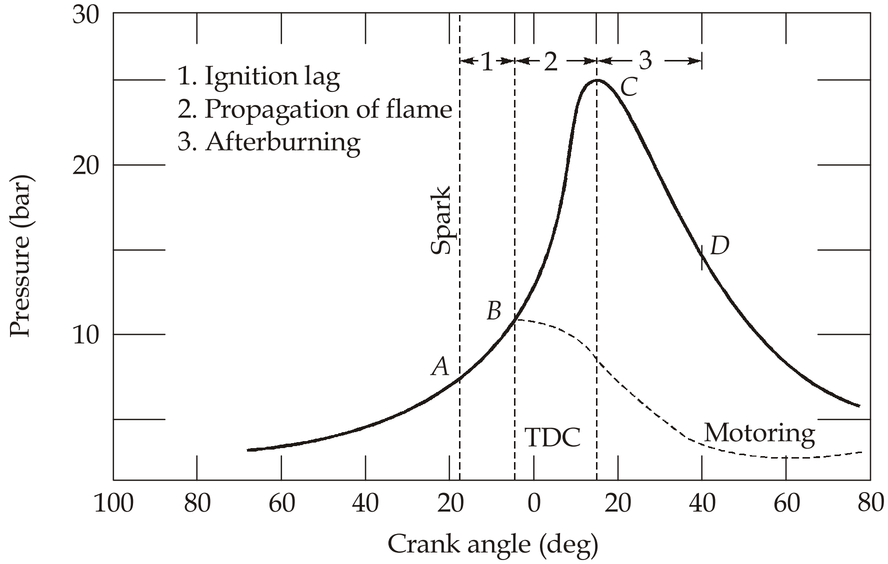
8. **(b)**

The combustion process in SI engine consisting of three stages.

(1) (A → B) Ignition lag

(2) (B → C) Main stage

(3) (C → D) After burning



Stages of combustion in an SI engine

1. The first stage (A → B) is referred to as the ignition lag or preparation phase in which growth and development of a self propagating nucleus of flame takes place. This is a chemical process depending upon both temperature and pressure, the nature of the fuel and the proportion of the exhaust residual gas. Further, it also depends upon the relationship between the temperature and the rate of reaction.
2. The second stage (B → C) is a physical one and it is concerned with the spread of the flame throughout the combustion chamber. The starting point of the second stage is where the first measurable rise of pressure is seen on the indicator diagram i.e., the point where the line of combustion departs from the compression line (point B). This can be seen from the deviation from the motoring curve.

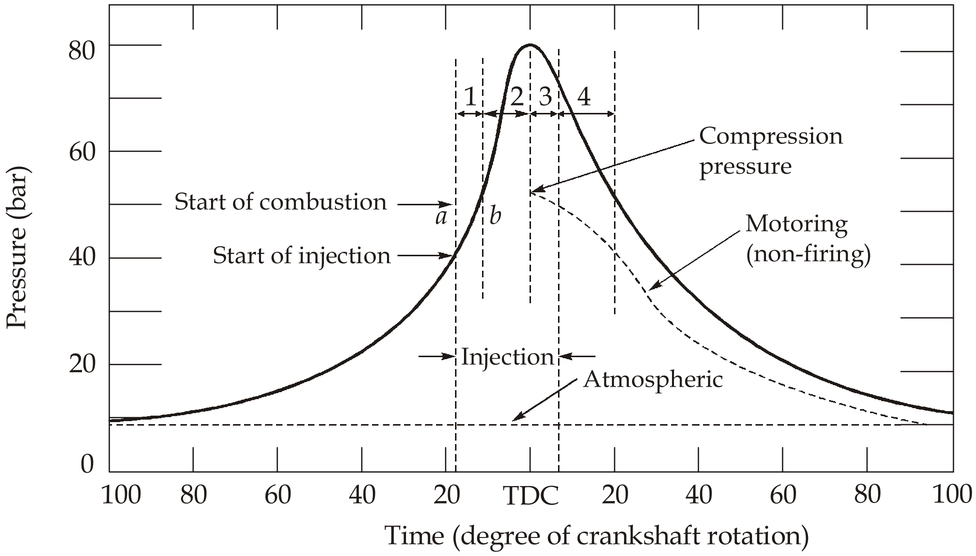
   During the second stage the flame propagates practically at a constant velocity. Heat transfer to the cylinder wall is low, because only a small part of the burning mixture comes in contact with the cylinder wall during this period. The rate of heat-release depends largely on the turbulence intensity and also on the reaction rate which is dependent on the mixture composition. The rate of pressure rise is proportional to the rate of heat-release because during this stage, the combustion chamber volume remains practically constant (since piston is near the top dead centre).
3. The starting point of the third stage is usually taken as the instant at which the maximum pressure is reached on the indicator diagram (point C). The flame velocity decreases during this stage. The rate of combustion be-comes low due to lower flame velocity and reduced flame front surface. Since the expansion stroke starts

before this stage of combustion, with the piston moving away from the top dead centre, there can be no pressure rise during this stage.

The combustion in a CI engine is considered to be taking place in four stages as shown in figure below. It is divided into the ignition delay period, the period of rapid combustion, the period of controlled combustion and the period of after-burning.

1. **Ignition Delay Period**

The ignition delay period is also called the preparatory phase during which some fuel has already been admitted but has not yet ignited. This period is counted from the start of injection to the point where the pressure-time curve separates from the motoring curve indicated as start of combustion in the figure.



Stages of combustion in an CI engine

The delay period in the CI engine exerts a very great influence on both engine design and performance. It is of extreme importance because of its effect on both the combustion rate and knocking and also its influence on engine starting ability and the presence of smoke in the exhaust. The fuel does not ignite immediately upon injection into the combustion chamber. There is a definite period of inactivity between the time when the first droplet of fuel hits the hot air in the combustion chamber and the time it starts through the actual burning phase. This period is known as the ignition delay period. In Figure the delay period is shown on pressure crank angle (or time) diagram between points a and b. Point a represents the time of injection and point b represents the time at which the pressure curve (caused by combustion) first separates from the motoring curve. The ignition delay period can be divided into two parts, the physical delay and the chemical delay.

Physical Delay: The physical delay is the time between the beginning of injection and the attainment of chemical reaction conditions.

Chemical Delay: During the chemical delay, reactions start slowly and then accelerate until inflammation or ignition takes place. Generally, the chemical delay is larger than the physical delay.

2. **Period of Rapid Combustion**

The period of rapid combustion also called the uncontrolled combustion, is that phase in which the pressure rise is rapid. During the delay period, the droplets have had time to spread over a wide area and fresh air is always available around the droplets. Most of the fuel admitted would have evapo-rated and formed a combustible mixture with air. By this time, the prefilame reactions would have also been completed. The period of rapid combustion is counted from end of delay period or the beginning of the combustion to the point of maximum pressure on the indicator diagram. The rate of heat-release is maximum during this period. It may be noted that the pressure reached during the period of rapid combustion will depend on the duration of the delay period (the longer the delay the more rapid and higher is the pressure rise since more fuel would have accumulated in the cylinder during the delay period).

3. **Period of Controlled Combustion**

The rapid combustion period is followed by the third stage, the controlled combustion. The temperature and pressure in the second stage is already quite high. Hence the fuel droplets injected during the second stage burn faster with reduced ignition delay as soon as they find the necessary oxygen and any further pressure rise is controlled by the injection rate. The period of controlled combustion is assumed to end at maximum cycle temperature.

4. **Period of After-Burning**

Combustion does not cease with the completion of the injection process. The unburnt and partially burnt fuel particles left in the combustion chamber start burning as soon as they come into contact with the oxygen. This process continues for a certain duration called the after-burning period. Usually this period starts from the point of maximum cycle temperature and continues over a part of the expansion stroke. Rate of after-burning depends on the velocity of diffusion and turbulent mixing of unburnt and partially burnt fuel with the air. The duration of the after-burning phase may correspond to 70-80 degrees of crank travel from TDC.

8. (c) (i)

Design and Operating variables that reduce NO_x concentration from the exhaust of SI Engine

The concentration of NO_x from the exhaust of an SI engine can be reduced by the following methods:

Decreasing the combustion temperature

The temperature of the combustion products can be decreased by:

(a) Decreasing the compression ratio
(b) Retarding the spark
(c) Avoiding knock
(d) Decreasing the inlet temperature of the charge
(e) Decreasing the speed (more heat loss, less turbulence, lower inlet temperature)
(f) Decreasing the inlet pressure of the charge (more heat loss, greater residual gas retention)
(g) Exhaust gas recirculation (EGR)
(h) Increasing the humidity of air, or water injection
(i) Using a very rich or very lean air/fuel ratio.

Decreasing the availability of oxygen in the flame front

The oxygen availability in the flame front can be reduced by:

(i) Using rich mixtures
(ii) Decreasing the homogeneity of the mixture
(iii) Using a stratified charge
(iv) Using the divided combustion chambers.

Effect of operating variables on CI engine NO_x exhaust emission

The CI engines operate with lean mixtures and therefore produce very little CO and HC concentrations.

The important operating variables that affect the NO_x concentration from direct injection CI engine are given below:

Injection timing

Injection advance results in a longer delay period because the fuel is injected in air at lower pressures and temperatures. A longer delay period causes more fuel evaporation and mixing in the lean flame region (LFR), resulting in a high NO_x concentration. The NO_x formation in the other regions of the spray increases with injection due to the high temperatures reached. Late injection is one of the effective ways of reducing NO_x emissions. However, this results in a loss of brake mean effective pressure (bmep) and in increase of brake specific fuel consumption (bsfc).

Fuel/air ratio

An increase in the fuel/air ratio (on leaner side of the stoichiometric) results in an increase in the maximum average gas temperature, and in an increase in NO_x concentration in the exhaust.

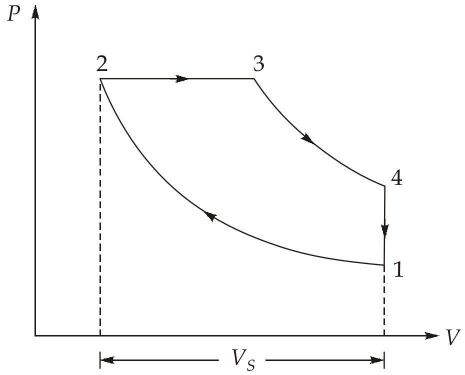
Type of fuel

The delay period is longer for the fuel with low cetane number and therefore more fuel is present in the lean flame region (LFR) when combustion starts. This larger quantity of fuel produces a higher gas temperature. Due to combustion early in the cycle, more NO_x is formed in the lean flame region.

Intake air charge dilution

The dilution of intake air charge with exhaust gases or water-injection reduces the combustion temperature and hence the NO_x emission.

8. (c) (ii)



Given : $r = 22$; C.V = 42000 kJ/kgK; $R = 0.287$ kJ/kgK

The air/fuel ratio is 28 : 1, therefore 1 kg mixture has $\frac{1}{29}$ kg of fuel.

$$\begin{aligned} c_p &= c_v + R \\ &= 0.74 + 25 \times 10^{-6}T + 0.287 \\ &= 1.027 + 25 \times 10^{-6}T \end{aligned}$$

Heat transfer during process 2 - 3, per kg of mixture

$$q_{2-3} = \frac{42000}{29} = 1448.3 \text{ kJ}$$

Also,
$$q_{2-3} = \int_{T_2}^{T_3} c_p dT$$

$$\int_{970}^{T_3} \left(1.027 + 25 \times 10^{-6} T\right) dT = 1448.3$$

$$1.027(T_3 - 970) + \frac{25 \times 10^{-6}}{2}\left[T_3^2 - 970^2\right] = 1448.3$$

$$1.027T_3 - 996.19 + 12.5 \times 10^{-6} T_3^2 - 11.76 = 1448.3$$

$$12.5 \times 10^{-6} T_3^2 + 1.027T_3 - 2456.25 = 0$$

$$T_3 = \frac{-1.027 + \sqrt{(1.027)^2 - 4\left(12.5 \times 10^{-6}\right)(-2456.25)}}{2 \times 12.5 \times 10^{-6}}$$

$$T_3 = 2325.83 \text{ K}$$

For constant pressure process 2 - 3,

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

$$V_3 = \frac{T_3}{T_2} \times V_2 = \frac{2325.83}{970} = 2.397V_2$$

Swept volume, $V_s = V_1 - V_2$

$$= V_2\left(\frac{V_1}{V_2} - 1\right) = V_2(r - 1) = V_2(22 - 1)$$

$$= 21\, V_2$$

Percentage of stroke during which the combustion is completed

$$= \frac{V_3 - V_2}{V_s} \times 100$$

$$= \frac{2.397V_2 - V_2}{21V_s} \times 100$$

$$= \frac{1.397V_2 - V_2}{21V_s} \times 100 = 6.65\%$$

❍❍❍❍