MADE EASY

Leading Institute for ESE, GATE & PSUs

Detailed Solutions

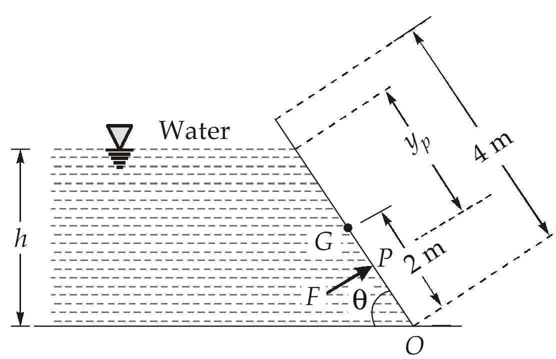
ESE-2024 Mains Test Series

Mechanical Engineering Test No : 10

Full Syllabus Test (Paper-1)

Section : A

1. (a)



Let F be the hydrostatic force acting on the gate at point P. Then,

F = (Pressure at the centroid of the submerged portion of gate) × (Submerged area of gate)

$$= \frac{h}{2} \times 9810 \times \frac{h}{\sin\theta} \times 1 = \frac{4905 \times h^2}{\sin\theta} \quad \text{...(i)}$$

The distance of the pressure centre P from the free surface along the gate can be found out as below:

$$y_p = \frac{h}{2\sin\theta} + \frac{\left(\frac{h}{\sin\theta}\right)^3}{12\times 1\times\left(\frac{h}{\sin\theta}\right)\times\left(\frac{h}{2\sin\theta}\right)}$$

$$y_p = \frac{h}{2\times\sin\theta} + \frac{h}{6\times\sin\theta} = \frac{2h}{3\sin\theta}$$

$$OP = \frac{h}{\sin\theta} - \frac{2h}{3\sin\theta} = \frac{h}{3\sin\theta}$$

Taking the moment of all forces about the hinge O,

$$F\times\left(\frac{h}{3\sin\theta}\right) - 2100\times(2\cos\theta) = 0$$

$$\Rightarrow \left(\frac{4905\times h^2}{\sin\theta}\right)\times\left(\frac{h}{3\sin\theta}\right) - 4200\times\cos\theta = 0$$

$$\Rightarrow \quad h^3 = \frac{4200\times 3}{4905}\times\cos\theta\sin^2\theta$$

$$\Rightarrow \quad h = 1.37\times(\cos\theta\cdot\sin^2\theta)^{1/3}$$ **Ans.**

1. (b)

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.

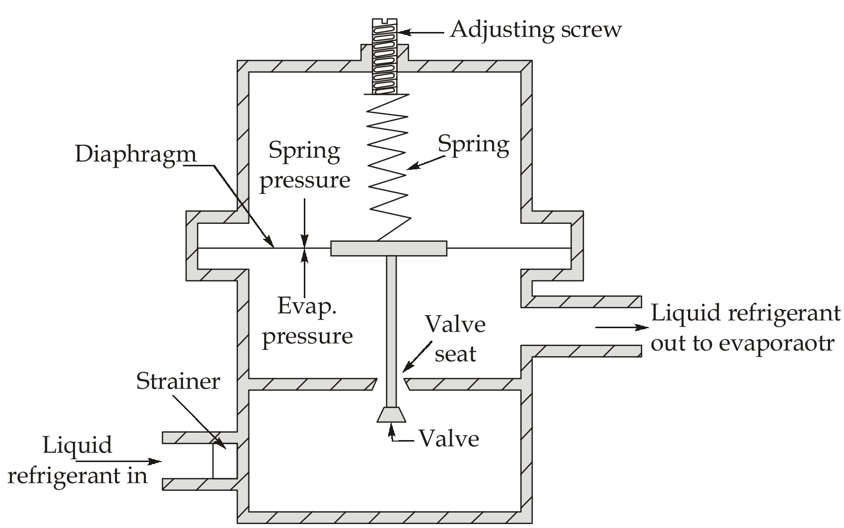


Figure: Automatic (or constant pressure) expansion valve

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.

1. (c)

Given : R_i = 6 cm = 0.06 m; R_o = 18 cm = 0.18 m; q_g = 1 kW/m^3; k = 0.5 W/m-K

At r = 0.18; T = 50°C

For uniform heat generation and steady state uni-directional heat flow in the radial direction, the differential equation is given by,

$$\frac{d}{dr}\left(r\frac{dt}{dr}\right) = \frac{-q_g \cdot r}{k}$$

Upon integration,

$$r\frac{dt}{dr} = \frac{-q_g \cdot r^2}{2k} + C_1$$

$$\Rightarrow \quad \frac{dt}{dr} = \frac{-q_g r}{2k} + \frac{C_1}{r}$$

Integrating the above equation once again,

$$t = \frac{-q_g r^2}{4k} + C_1 \ln(r) + C_2 \quad \text{...(ii)}$$

Boundary conditions:

(i) At $r = 0.12$ m, the temperature is to be maximum,

So, $$\frac{dt}{dr} = 0$$

$$\Rightarrow \quad 0 = \frac{-1000 \times 0.12}{2 \times 0.5} + \frac{C_1}{0.12}$$

$$\Rightarrow \quad C_1 = 14.4$$

(ii) At $r_2 = 0.18$ m; $T_2 = 50°C$

$$\therefore \quad 50 = \frac{-1000 \times 0.18^2}{4 \times 0.5} + 14.4 \ln(0.18) + C_2$$

$$\Rightarrow \quad C_2 = 90.89$$

The temperature distribution through the cylinder is given by,

$$t = \frac{-q_g r^2}{4k} + 14.4 \ln(r) + 90.89$$

At $r_1 = 0.06$ m

$$t_1 = \frac{-1000 \times 0.06^2}{4 \times 0.5} + 14.4 \times \ln(0.06) + 90.89$$

$$t_1 = 48.58°C$$ **Ans.**

At $r = 0.12$ m, $t = t_{max}$,

$$t_{max} = \frac{-1000 \times 0.12^2}{4 \times 0.5} + 14.4 \times \ln(0.12) + 90.89$$

$$t_{max} = 53.16°C$$ **Ans.**

1. (d)

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 these 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.

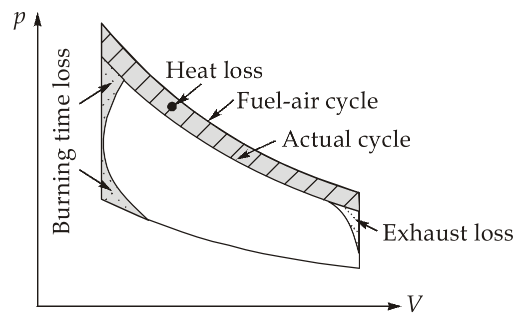


Figure: Comparison of constant volume fuel-air cycle with

Figure shows the comparison of the constant volume fuel-air cycle with the actual cycle, indicating the burning time loss, heat loss and exhaust loss. It is a power cycle and does not include gas exchanges.

Loss ratio : It is the ratio of the loss area to the fuel-air cycle area. It is used for the sake of comparison of different losses. The following loss ratios are defined:

(i) Time loss ratio: The ratio of the burning time loss area to the area of the fuel-air cycle diagram is called the time loss ratio.

(ii) Heat loss ratio: The ratio of the heat loss area to the area of fuel-air cycle diagram is called the heat loss ratio.

(iii) Blowdown loss ratio: The ratio of the exhaust blowdown area to the area of the fuel-air cycle diagram is called the blowdown loss ratio.

(iv) Lost work ratio: It can be expressed as the ratio of the difference of area of fuel-air cycle and area of actual cycle to the area of fuel-air cycle, i.e.

$$\text{Lost work ratio} = 1 - \frac{\text{Area of actual cycle}}{\text{Area of fuel-air cycle}}$$

1. (e)

$$\text{Bulk mean temperature, } T_m = \frac{82+18}{2} = 50°\text{C}$$

So, the properties of water at 50°C can be used for calculation.

$$\text{Reynold's number, Re} = \frac{4\dot{m}}{\pi\mu d} = \frac{4\times 0.01}{\pi\times\left(990\times 0.5675\times 10^{-6}\right)\times 0.06} = 377.71$$

Since, Re < 2000, so the flow is laminar flow.

Also, $\dot{q}\times\pi DL = \dot{m}\cdot c_p\cdot \Delta T_m$

$$\Rightarrow \quad L = \frac{0.01\times 4181\times(82-18)}{2600\times\pi\times 0.06} = 5.46\text{ m}$$ **Ans.**

For a fully developed laminar flow condition with constant heat flux,

$$\text{Nusselt number, Nu} = 4.36$$

$$\Rightarrow \quad \frac{h\cdot D}{k} = 4.36$$

$$\Rightarrow \quad h = \frac{4.36\times 0.64}{0.06} = 46.51\text{ W/m}^2\text{K}$$

$$\therefore \quad \text{Heat flux, } \dot{q} = h\times\left(T_{w,exit} - T_{b,exit}\right)$$

$$\Rightarrow \quad T_{w,exit} = \frac{\dot{q}}{h} + T_{b,exit} = \frac{2600}{46.51} + 82$$

$$= 137.90°\text{C}$$ **Ans.**

2. (a) (i)

The hemisphere in its floating condition is shown in figure. Let $\forall$ be the submerged volume. Then from equilibrium under floating condition.

$$\frac{2}{3}\pi r^3\times\rho = \forall\times\rho_l$$

or

$$\forall = \frac{2}{3}\pi r^3\times\frac{\rho}{\rho_l}$$

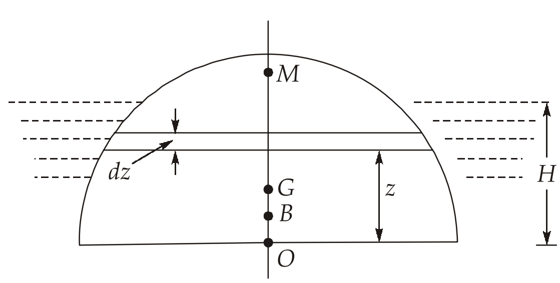


Figure: A solid hemisphere floating in a liquid

The centre of gravity G will lie on the axis of symmetry of the hemisphere. The distance of G along this line from the base of the hemisphere can be found by taking moments of elemental circular strips as shown in figure about the base as

$$OG = \frac{\int_0^r \pi\left(r^2 - z^2\right) z\,dz}{\frac{2}{3}\pi r^3} = \frac{\int_0^r \pi\left(r^2 z - z^3\right) dz}{\frac{2}{3}\pi r^3}$$

$$= \frac{3}{2} \times \frac{\left[\frac{r^2 z^2}{2} - \frac{z^4}{4}\right]_0^r}{r^3} = \frac{3}{2} \times \frac{\left(\frac{r^4}{4}\right)}{r^3}$$

$$= \frac{3}{8} r$$

In a similar way, the location of centre of buoyancy which is the centre of immersed volume $\forall$ is found as

$$OB = \frac{\int_0^H \pi\left(r^2 - z^2\right) z\,dz}{\frac{2}{3}\pi r^3 \frac{\rho}{\rho_l}} = \frac{3}{2} \frac{\rho_l}{r^3 \rho} \times \int_0^H \left(r^2 z - z^3\right) dz$$

$$= \frac{3}{2} \frac{\rho_l}{r^3 \rho} \left[\frac{r^2 z^2}{2} - \frac{z^4}{4}\right]_0^H$$

$$= \frac{3}{8} \frac{\rho_l}{\rho} r \frac{H^2}{r^2} \left(2 - \frac{H^2}{r^2}\right) \quad \text{...(i)}$$

where H is the depth of immersed volume as shown in figure.

If r_h is the radius of cross-section of the hemisphere at water line, then we can write

$$H^2 = r^2 - r_h^2$$

Substituting the value of H in equation (i), we have

$$OB = \frac{3}{8} \frac{\rho_l}{\rho} r \left(1 - \frac{r_h^4}{r^4}\right)$$

The height of the metacentre M above the centre of buoyancy B is given by

$$BM = \frac{I}{\forall} = \frac{\pi r_h^4}{4\left\{\left(\frac{2}{3}\pi r^3\right)\frac{\rho}{\rho_l}\right\}} = \frac{3}{8} \frac{\rho_l}{\rho} r \frac{r_h^4}{r^4}$$

Therefore, the metacentric height MG becomes

$$MG = MB - BG = MB - (OG - OB)$$

$$= \frac{3}{8}\frac{\rho_l}{\rho} r \frac{r_h^4}{r^4} - \frac{3}{8}r + \left[\frac{3}{8}\frac{\rho_l}{\rho} r\left(1 - \frac{r_h^4}{r^4}\right)\right]$$

$$= \frac{3}{8}r\left(\frac{\rho_l}{\rho} - 1\right)$$

Since, $\rho_l > \rho$, $MG > 0$, and hence, the equilibrium is stable.

2. (a) (ii)

From the given velocity components:

$$\frac{\partial u}{\partial x} = 0; \quad \frac{\partial v}{\partial y} = 0; \quad \frac{\partial w}{\partial z} = 0$$

$$\therefore \quad \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0$$

The velocity components satisfy the flow continuity equation, and, therefore, the given field is a possible case of fluid flow.

The curl of velocity vector is

$$\text{Curl } V = \nabla \times V$$

$$= \begin{bmatrix} \hat{i} & \hat{j} & \hat{k} \\ \frac{\partial}{\partial x} & \frac{\partial}{\partial y} & \frac{\partial}{\partial z} \\ a + by - cz & d - bx - ez & f + cx - ey \end{bmatrix}$$

$$= \hat{i}\left[\frac{\partial}{\partial y}(f + cx - ey) - \frac{\partial}{\partial z}(d - bx - ez)\right]$$

$$-\hat{j}\left[\frac{\partial}{\partial x}(f + cx - ey) - \frac{\partial}{\partial z}(a + by - cz)\right]$$

$$+\hat{k}\left[\frac{\partial}{\partial x}(d - bx - ey) - \frac{\partial}{\partial y}(a + by - cz)\right]$$

$$= \hat{i}(-e + e) - \hat{j}(c + c) + \hat{k}(-b - b)$$

$$= -2\left(c\hat{j} + b\hat{k}\right)$$

Thus the curl of velocity vector is not zero, and hence the flow is not irrotational.

$$\text{Vorticity, } \xi = \nabla \times V$$

$$= -2\left(c\hat{j} + b\hat{k}\right) = 2\sqrt{c^2 + b^2}$$

Vorticity equals twice the value of rotation

$$\therefore \quad \text{Rotation, } \omega = \sqrt{c^2 + b^2}$$

2. (b)

Given : $m = 0.6$ kgm^2; $N_1 = 3000$ rpm; $m = 2$ kg; $T_0 = 17°C = 290$ K

The initial angular velocity of the flywheel,

$$\omega_1 = \frac{2\pi N_1}{60} = \frac{2\pi \times 3000}{60} = 314.16 \text{ rad/s}$$

Initial available energy of the flywheel = Initial kinetic energy,

$$A_1 = \frac{1}{2}I\omega_1^2 = \frac{1}{2} \times 0.6 \times 314.16^2$$

$$A_1 = 29.61 \text{ kJ}$$

When this kinetic energy is dissipated as frictional heat then there will be rise in temperature of shaft and bearing.

$$A_1 = mc_{p,w}\Delta T$$

$$\Rightarrow \quad \Delta T = \frac{29.61 \times 10^3}{2 \times 4.187 \times 10^3} = 3.54°C$$ **Ans.**

Thus, the final temperature of bearings,

$$T_2 = T_0 + \Delta T = 17 + 3.54 = 20.54°C$$

$$= 293.54 \text{ K}$$

The maximum energy returned to flywheel as available energy,

$$A_2 = \Delta H - T_0\Delta S$$

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

$$= 2 \times 4.187 \times \left[3.54 - 290 \times \ln\left(\frac{293.54}{290}\right)\right] = 0.18 \text{ kJ}$$ **Ans.**

The amount of kinetic energy becoming unavailable,

$$UE = A_1 - A_2 = 29.61 - 0.18 = 29.43 \text{ kJ}$$ **Ans.**

For the final rpm of flywheel,

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

$$0.18 \times 10^3 = \frac{1}{2}\times 0.60\times \omega_2^2$$

$$\Rightarrow \quad \omega_2 = 24.49 \text{ rad/s}$$

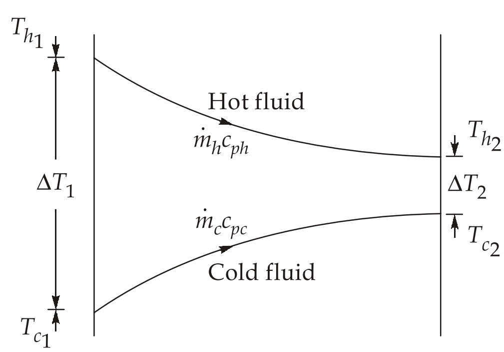
$$\Rightarrow \quad \frac{2\pi N_2}{60} = 24.49$$

$$N_2 = 233.86 \text{ rpm}$$

Ans.

2. (c)

For parallel flow, we have



$$\Delta T_1 = T_{h_1} - T_{c_1} = 160 - 15 = 145°\text{C}$$

$$\Delta T_2 = T_{h_2} - T_{c_2} = 30 - 25 = 5°\text{C}$$

$$\text{LMTD, } (\Delta T_m)_{\text{PF}} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{145-5}{\ln\left(\frac{145}{5}\right)} = 41.576°\text{C}$$

By neglecting the resistance offered by tube wall and scale formed, the overall heat transfer coefficient based on outer surface is given by,

$$U_0 = \frac{1}{\left(\frac{r_0}{r_i}\right)\times\frac{1}{h_i}+\frac{1}{h_o}} = \frac{1}{\left(\frac{9}{7.5}\right)\times\frac{1}{105}+\frac{1}{2095}} = 83.992 \text{ W/m}^2\text{K}$$

The mass flow rate of air, $\dot{m}_a = \frac{PV}{RT} = \frac{101.3 \times \left(\frac{8}{60}\right)}{0.287 \times 288} = 0.164$ kg/s

The heat exchange rate is given by,

$$Q = \dot{m}_a \cdot c_{p,air}\left(T_{c_1} - T_{c_2}\right) = 0.164 \times 1.005 \times (160 - 30)$$

$$= 21.427 \text{ kW}$$

$$Q = U_0 A (\Delta T_m)_{PF}$$

$$\Rightarrow \quad A_P = \frac{21.427 \times 10^3}{83.992 \times 41.576} = 6.136 \text{ m}^2$$

The density of air flowing through the intercooler tubes at 6.5 bar at an average temperature of $\left[\left(\frac{160+30}{2}\right) = 95°\text{C} = 368 \text{ K}\right]$ is

$$\rho_{air} = 6.154 \text{ kg/m}^3$$

Using continuity equation, $\dot{m} = \rho \cdot A_c \cdot V_\infty$

$$\Rightarrow \quad A_C = \frac{0.164}{6.154 \times 5.9} = 4.517 \times 10^{-3} \text{m}^2$$

The cross-section area per tube = $\pi r^2 = \pi \times 0.0075^2$

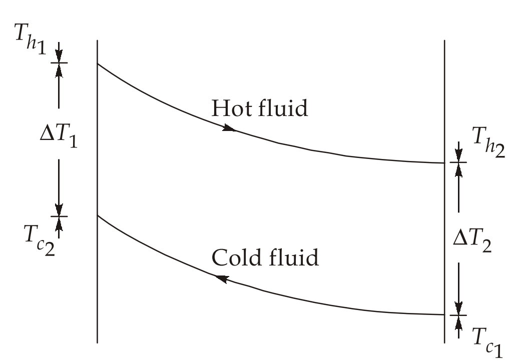
$$= 1.767 \times 10^{-4} \text{ m}^2$$

$\therefore$ The number of tubes, $n = \frac{4.517 \times 10^{-3}}{1.767 \times 10^{-4}} = 25.56 \simeq 26$ tubes **Ans.**

$$A_P = n \cdot \pi \cdot D_0 L$$

$$\Rightarrow \quad L = \frac{6.136}{26 \times \pi \times 0.018} = 4.173 \text{ m per tube}$$

For counter flow arrangement :



$$\Delta T_1 = T_{h_1} - T_{c_1} = 160 - 25 = 135°C$$

$$\Delta T_2 = T_{h_2} - T_{c_2} = 30 - 15 = 15°C$$

$$(\Delta T_m)_{CF} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{135 - 15}{\ln\left(\frac{135}{15}\right)} = 54.614°C$$

We know that for the same heat exchange rate,

$$A \propto \frac{1}{\Delta T_m}$$

$$\frac{A_{CF}}{A_{PF}} = \frac{(\Delta T_m)_{PF}}{(\Delta T_m)_{CF}}$$

$$\Rightarrow \quad \frac{L_{CF}}{L_{PF}} = \frac{(\Delta T_m)_{PF}}{(\Delta T_m)_{CF}}$$

$$\Rightarrow \quad L_{CF} = 4.173 \times \frac{41.576}{54.614} = 3.177 \text{ m/tube}$$

$$\text{Saving in length} = \frac{L_{PF} - L_{CF}}{L_{PF}} = \left(\frac{4.173}{3.177} - 1\right) \times 100 = 31.35\%$$ **Ans.**

3. (a)

Given : $P = 10000$ kW; $H = 350$ m; $N = 600$ rpm; $\phi = 0.48$; $\frac{D}{d} = 10$; $\eta_0 = 0.85$

$$\text{Power, } \eta_0 = \frac{P}{\rho g Q H}$$

$$\Rightarrow \quad 0.85 = \frac{10^7}{10^3 \times 9.81 \times Q \times 350}$$

$$\therefore \quad Q = 3.426 \text{ m}^3/\text{sec}$$ **Ans.**

Assuming coefficient of velocity,

$$C_V = 1$$

$$\text{Velocity of the jet, } V_1 = C_V\sqrt{2gH} = 1 \times \sqrt{2 \times 9.81 \times 350} = 82.867 \text{ m/s}$$

Also, $u = 0.48 \times 82.867 = 39.776$ m/s or $$\frac{\pi DN}{60} = 39.776$$

∴ $$D = \frac{39.776 \times 60}{\pi \times 600} = 1.266 \text{ m}$$ **Ans.**

Diameter of the jet, $$d = \frac{D}{10} = \frac{1.266}{10} = 0.1266 \text{ m}$$

Discharge through one jet is given by,

$$q = \frac{\pi}{4} d^2 \times V_1 = \frac{\pi}{4} \times (0.1266)^2 \times 82.867 = 1.043 \text{ m}^3/\text{s}$$

∴ The number of jets, $$n = \frac{Q}{q} = \frac{3.426}{1.043}$$

∴ n = 3.28, say 4 jets **Ans.**

The number of vanes is determined from an empirical relationship

$$N = \frac{1}{2}\left(\frac{D}{d}\right) + 15 = \frac{1}{2} \times 10 + 15 = 20$$ **Ans.**

Width of bucket, $b = 5d = 5 \times 0.1266 = 0.633$ m

and depth of bucket, $d = 1.2d = 1.2 \times 0.1266 = 0.152$ m **Ans.**

3. (b)

Given : $U_{max} = 2.25$ m/s; $R = \frac{D}{2} = \frac{0.25}{2} = 0.125$ m;

At $y = R - r = 12.5 - 8 = 4.5$ cm $= 0.045$ m; $u = 1.95$ m/s

$$\frac{U_{max} - U}{U^*} = 5.75 \log_{10}\left(\frac{R}{y}\right)$$

⇒ $$\frac{2.25 - 1.95}{U^*} = 5.75 \log_{10}\left(\frac{0.125}{0.045}\right)$$

⇒ $U^* = 0.1176$ m/s **Ans.**

Friction velocity, $$U^* = \sqrt{\frac{\tau_0}{\rho}}$$

⇒ $\tau_0 = (U^*)^2 \times \rho = (0.1176)^2 \times 1000$

⇒ $\tau_0 = 13.83$ N/m^2 **Ans.**

Also, $$\frac{u - \bar{U}}{U^*} = 3.75 + 5.75 \log_{10}\left(\frac{y}{R}\right)$$

When, $y = R$, $u = U_{max}$

$$\frac{U_{max} - \bar{U}}{U^*} = 3.75 + 5.75\log_{10}\left(\frac{R}{R}\right)$$

$$\Rightarrow \quad \frac{2.25 - \bar{U}}{0.1176} = 3.75$$

$$\Rightarrow \quad \bar{U} = 1.809 \text{ m/s}$$

$$\text{Discharge, } Q = A \times \bar{U}$$

$$= \frac{\pi}{4} \times (0.25)^2 \times 1.809 = 0.089 \text{ m}^3\text{/s}$$ **Ans.**

Also, friction velocity, $U^* = \bar{U}\sqrt{\frac{f}{8}}$

$$\Rightarrow \quad \text{Friction factor, } \rho = \left(\frac{u^*}{\bar{U}}\right)^2 \times 8 = \left(\frac{0.1176}{1.809}\right)^2 \times 8 = 0.0338$$ **Ans.**

For turbulent flow through rough pipes,

$$\frac{u}{u^*} = 8.5 + 5.75\log_{10}\left(\frac{y}{\in}\right)$$

At $y = R = 0.125$ m, $u = U_{max} = 2.25$ m/s

$$\frac{2.25}{0.1176} = 8.5 + 5.75\log_{10}\left(\frac{0.125}{\in}\right)$$

$$\in = 1.769 \times 10^{-3} \text{ m} = 1.769 \text{ mm}$$ **Ans.**

3. (c)

Given : $x_{O_2} = 0.06$; $x_{CO_2} = 0.63$; $x_{He} = 0.31$

$$c_p = x_{O_2} \cdot c_{pO_2} + x_{O_2} \cdot c_{pCO_2} + x_{O_2} \cdot c_{pHe}$$

$$= 0.06 \times 0.918 + 0.63 \times 0.846 + 0.31 \times 5.1926$$

$$= 2.1978 \text{ kJ/kgK}$$

$$c_v = x_{O_2} \times c_{vO_2} + x_{O_2} \cdot c_{vCO_2} + x_{He} \cdot c_{vHe}$$

$$= 0.06 \times 0.658 + 0.63 \times 0.657 + 0.31 \times 3.1156$$

$$= 1.4192 \text{ kJ/kgK}$$

The apparent gas constant of the mixture and the specific heat ratio are

$$R = c_p - c_v = 2.1978 - 1.4192 = 0.7786 \text{ kJ/kgK}$$

$$\gamma = \frac{c_p}{c_v} = \frac{2.1978}{1.4192} = 1.55$$

The temperature at the end of the expansion for the isentropic process is

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 600 \times \left(\frac{100}{1000}\right)^{\frac{0.55}{1.55}} = 265.04 \text{ K}$$

Actual outlet temperature, $T_2 = T_1 - \eta_T(T_1 - T_{2s})$

$$= 600 - 0.89 \times (600 - 265.04)$$

$$= 301.88 \text{ K}$$

As the heat transfer is adiabatic and thus there is no heat transfer, the actual work output is determined to be,

$$W_{out} = h_1 - h_2$$

$$= c_p(T_1 - T_2) = 2.1978 \times (600 - 301.88)$$

$$= 655.2 \text{ kJ/kg}$$ **Ans.**

The entropy change of the gas mixture and the exergy destruction in the turbine,

$$s_2 - s_1 = c_p \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right)$$

$$= 2.1978 \times \ln\left(\frac{301.88}{600}\right) - 0.7786\ln\left(\frac{100}{1000}\right)$$

$$= 0.2831 \text{ kJ/kgK}$$

$$X_{dest} = T_0 \cdot s_{gen} = 300 \times (0.2831) = 84.93 \text{ kJ/kg}$$ **Ans.**

The expended energy is the sum of the work output of turbine (exergy recovered) and the exergy destruction (exergy) wasted,

$$X_{expended} = X_{recovered} + X_{dest}$$

$$= 655.2 + 84.93$$

$$= 740.13 \text{ kJ/kg}$$

The second law of efficiency is the ratio of the recovered to expended exergy,

$$\eta_{II} = \frac{X_{recovered}}{X_{expended}} = \frac{X_{out}}{X_{expended}}$$

$$= \frac{655.2}{740.13} = 0.8853 \text{ or } 88.53\%$$ **Ans.**

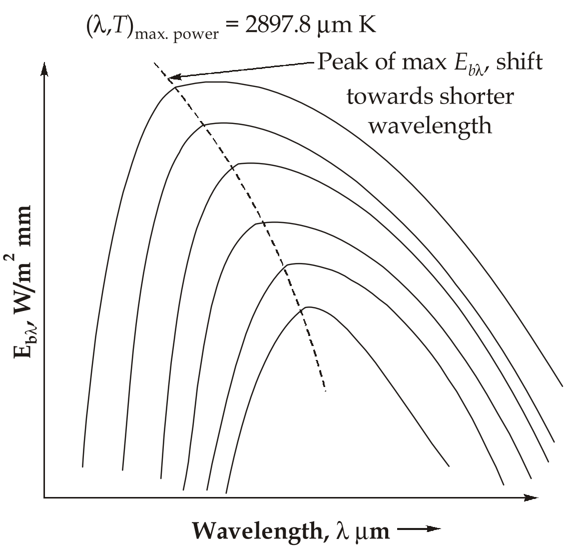
4. (a)

Wein's displacement law: As the temperature of a blackbody increases, peak of spectral emissive power shift towards shorter wavelengths. The wavelength at which the peak occurs for a specified temperature is given by Wein's displacement law as follows:

Assume wavelength at which sun has maximum spectral emissive power is λ_m.

From Wein's displacement law, we know that for blackbody,

$$\lambda_m T = 2897.8\ \mu\text{mK}$$



Refer to figure,

$$\text{The total emissivity, } \in = \frac{\int_0^{\infty} \in_{\lambda,b}\, d\lambda}{E_b}$$

$$= \in_{\lambda_1} \frac{\int_0^{\lambda_1} \in_{\lambda,b}\cdot d\lambda}{E_b} + \in_{\lambda_2} \frac{\int_{\lambda_1}^{\lambda_1} \in_{\lambda,b}\cdot d\lambda}{E_b} + \in_{\lambda_3} \frac{\int_{\lambda_2}^{\infty} \in_{\lambda,b}\cdot d\lambda}{E_b}$$

Using the black body function, we get

$$\in = \in_{\lambda_1} F_{0\to\lambda_1} + \in_{\lambda_2}\left[F_{(0\to\lambda_2)} - F_{(0\to\lambda_1)}\right] + \in_{\lambda_3}\left[1 - F_{(0\to\lambda_2)}\right]$$

Using table, we get

$$\text{For } \lambda_1\cdot T_s = 2 \times 800 = 1600\ \mu\text{m}\cdot\text{K};\quad F_{0-\lambda_1} = 0.019718$$

$$\text{For } \lambda_2\cdot T_s = 10 \times 800 = 8000\ \mu\text{m}\cdot\text{K};\quad F_{0-\lambda_2} = 0.856288$$

$\therefore \quad \epsilon = 0.1 \times 0.019718 + 0.5 \times (0.856288 - 0.019718) + 0.8 \times (1 - 0.856288)$

$= 0.535$ **Ans.**

The total emissive power of a brick wall is given by,

$$E = \epsilon\sigma T_s^4 = 0.535 \times 5.67 \times 10^{-8} \times 800^4$$

$$= 12.425 \text{ kW/m}^2$$ **Ans.**

Total absorptivity of the wall due to radiation from the coal bed is given by,

$$\alpha = \frac{\int_0^\infty \alpha_\lambda \cdot G_\lambda \cdot d\lambda}{\int_0^\infty G_\lambda \cdot d\lambda}$$

Since, the surface is diffuse, hence, $\alpha_\lambda = \epsilon_\lambda$. Further, we assume $G_\lambda \simeq E_{\lambda,b}$ as spectral distribution of the irradiation approximates that due to emission from a black body at 2400 K, hence,

$$\alpha = \frac{\int_0^\infty \alpha \cdot \epsilon_\lambda \cdot E_{\lambda,b}\, d\lambda}{\int_0^\infty E_{\lambda,b} \cdot d\lambda}$$

Breaking the integrals into parts, we get

$$\alpha = \epsilon_{\lambda_1} \cdot F_{0\to\lambda_1} + \epsilon_{\lambda_2} \cdot \left[F_{0\to\lambda_2} - F_{0\to\lambda_1}\right] + \epsilon_{\lambda_3}\left[1 - F_{0\to\lambda_2}\right]$$

From table;

For $\lambda_1 \cdot T_c = 2 \times 2000 = 4000\ \mu\text{mK} = 0.480877$

For $\lambda_2 \cdot T_c = 10 \times 2000 = 20000\ \mu\text{mK} = 0.985602$

Substituting the values of fraction, we get,

$\alpha = 0.1 \times 0.480877 + 0.5 \times (0.985602 - 0.480877) + 0.8 \times (1 - 0.985602) = 0.3119$ **Ans.**

4. (b)

From given data,

$$\omega = \frac{2\pi N}{60} = 0.1047\text{ N rad/s}, \quad r = \frac{50}{2} = 25\text{ cm} = 0.25\text{ m}$$

$$\frac{A}{A_s} = \left(\frac{30}{12}\right)^2 = 6.25$$

Case I : No Air vessel

At $\theta = 0°$, $h_{fs} = 0$

$$H_{as} = \frac{L_s}{g}\left(\frac{A}{A_s}\right)\omega^2 r$$

$$= \frac{5}{9.81} \times 6.25 \times (0.1047N)^2 \times 0.25$$

$$= 8.73 \times 10^{-3}\ N^2 m$$

At limiting condition for a suction pipe

$$H_{as} + H_r + H_s = H_{atm}$$

$$H_{as} + 2.5 + 3.5 = 10$$

or $$8.73 \times 10^{-3} N^2 = 4$$

$\therefore$ $$N = 21.405 \text{ rpm}$$ **Ans.**

Case II : With air vessel

When an air vessel is fitted in the suction pipe at 2.0 m from the cylinder. Acceleration pressure head is confined to a 2.0 m length next to the cylinder. Friction loss in remaining 3 m length of suction pipe is constant over time, as the flow is steady.

$\therefore$ $$H_{as}' = \frac{L_s}{g}\left(\frac{A}{A_s}\right)\omega^2 r$$

$$= \frac{2}{9.81} \times 6.25 \times (0.1047N)^2 \times 0.25$$

$$H_{as}' = 3.492 \times 10^{-3}\ N^2 m$$

$$V_s = \left(\frac{A}{As}\right)\frac{\omega r}{\pi} = \frac{6.25 \times 0.1047N \times 0.25}{\pi}$$

$$V_s = 0.052\ N\ m/s$$

$\therefore$ $$h_{fs} = \frac{fL_sV_s^2}{2gd_s} = \frac{0.02 \times 3 \times (0.052N)^2}{2 \times 9.81 \times 0.12}$$

$\therefore$ $$h_{fs} = 6.89 \times 10^{-5}\ N^2 m$$

At limiting condition for a suction pipe

$$H_{as}' + H_v + h_{fs} + H_s = H_{atm}$$

$$3.492 \times 10^{-3} N^2 + 2.5 + (6.89 \times 10^{-5} N^2) + 3.5 = 10$$

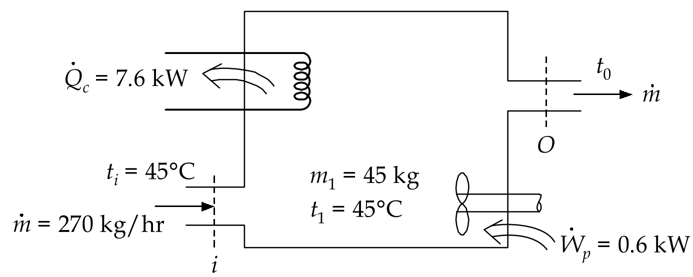
or $$N = 33.515 \text{ rpm}$$ **Ans.**

$$\text{Discharge, } Q = \frac{ALN}{60}$$

$$\text{Ratio of discharge } \frac{Q_2}{Q_1} = \frac{N_2}{N_1} = \frac{33.515}{21.405} = 1.5657$$

$\therefore$ Percentage change in discharge = 56.57 increase after fitting the air vessel.

4. (c)



Let suffix '*i*' and '*o*' represents the inlet and outlet conditions respectively and suffix '1' and '2' represent initial and final condition of water in the tank.

Let after '*t*' hours, the temperature inside tank is *T*.

According to mass balance,

$$\dot{m}_i - \dot{m}_0 = \left(\frac{dm}{dt}\right)_{C.V.}$$

$$\Rightarrow \quad \dot{m} - \dot{m} = \dot{m}_2 - \dot{m}_1$$

$$\Rightarrow \quad m_1 = m_2 = 45 \text{ kg} \quad \text{...(i)}$$

$$\Rightarrow \quad \dot{m}_i = \dot{m}_0 = 270 \text{ kg/hr} \quad \text{...(ii)}$$

According to energy balance,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \left(\frac{dU}{dt}\right)_{cv}$$

$$\Rightarrow \quad \frac{d}{dt}(mc_v \cdot dT) = \dot{E}_{\text{in}} - \dot{E}_{\text{out}}$$

$$\Rightarrow \quad m_1 c_v \frac{dT}{dt} = \dot{m}_i h_i - m_o h_o + (\dot{W}_p - \dot{Q}_c) \times 3600$$

$$\Rightarrow \quad 45 \times 4.18 \times \frac{dT}{dt} = 270 \times 4.18 \times (45 + 273) - 270 \times 4.18 \times T + (0.6 - 7.6) \times 3600$$

$$\Rightarrow \quad \frac{dT}{dt} = 1908 - 6T - 133.97$$

$$\Rightarrow \quad \frac{dT}{dt} = 1774.03 - 6T$$

$$\Rightarrow \quad \int_{T_1}^{T} \frac{dT}{1774.03 - 6T} = \int_0^t dt$$

$$\Rightarrow \quad t = \left.\frac{\ln(1774.03 - 6T)}{-6}\right]_{318}^{T}$$

$$\Rightarrow \quad -6t = \ln\left(\frac{1774.03 - 6T}{1774.03 - 6 \times 318}\right)$$

$$\Rightarrow \quad e^{-6t} = \frac{1774.03 - 6T}{-133.97}$$

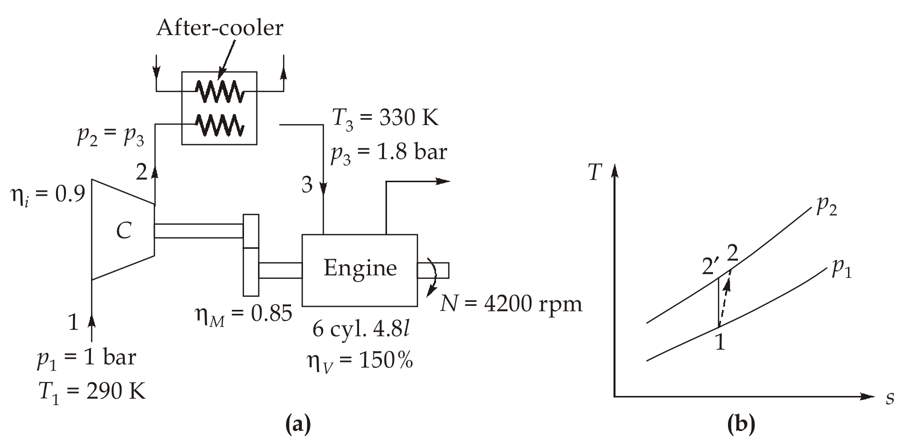
$$\Rightarrow \quad 6T - 1774.03 = 133.97e^{-6t}$$

$$\Rightarrow \quad T - 295.67 = 22.33e^{-6t}$$

$$\Rightarrow \quad T = 295.67 + 22.33e^{-6t}$$

Section : B

5. (a)



For isentropic compression process 1 – 2′.

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

$$\Rightarrow \quad T_{2'} = 290 \times \left(\frac{1.8}{1}\right)^{\frac{0.4}{1.4}} = 343.03 \text{ K}$$

$$\eta_C = \frac{T_{2'} - T_1}{T_2 - T_1}$$

$$\Rightarrow \quad 0.90 = \frac{343.03 - 290}{T_2 - 290}$$

$\Rightarrow \quad T_2 = 348.92$ K

For engine, swept volume, $V_s = 4.8 \times 10^{-3}$ m^3

$$\dot{V}_S = V_S \times \frac{N}{2 \times 60} = 4.8 \times 10^{-3} \times \frac{4200}{2 \times 60} = 0.168 \text{ m}^3/\text{s}$$

Volume of air induced, $\dot{V}_a = \eta_v \times \dot{V}_s = 1.5 \times 0.168 = 0.252 \text{ m}^3/\text{s}$

The overall volumetric efficiency is referred at ambient conditions.

Density of air at ambient conditions, $\rho = \frac{P}{RT} = \frac{100}{0.287 \times 290} = 1.2 \text{ kg/m}^3$

Mass of air inducted, $\dot{m}_a = \rho \times \dot{V}_a = 1.2 \times 0.252 = 0.3024 \text{ kg/s}$

Heat rejected from after-cooler $= \dot{m}_a c_p (T_2 - T_3)$

$= 0.3024 \times 1.005 \times (348.92 - 330)$

$= 5.75$ kW

Power needed to run the compressor $= \dot{m}_a c_p (T_2 - T_1)$

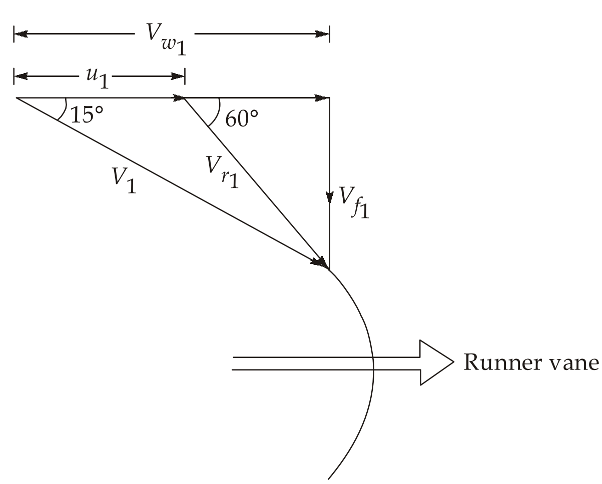
$= 0.3024 \times 1.005 \times (348.92 - 290)$

$= 17.9$ kW

Power absorbed from the engine $= \frac{17.9}{\eta_m} = \frac{17.9}{0.85} = 21.07$ kW

5. (b)

From the data,



$$u_1 = \frac{\pi DN}{60} = \frac{\pi \times 1 \times 450}{60} = 23.56 \text{ m/s}$$

From the velocity triangle at runner inlet,

$$\frac{V_1}{\sin 120°} = \frac{u_1}{\sin 45°}$$

$$\therefore \quad V_1 = \frac{23.56 \times \sin 120°}{\sin 45°} = 28.85 \text{ m/s}$$

$$\therefore \quad V_{w_1} = V_1 \cos 15° = 28.85 \cos 15°$$

$$= 27.87 \text{ m/s}$$

$$\text{Hydraulic efficiency, } \eta = \frac{V_{w_1} \cdot u_1}{gH}$$

$$= \frac{27.87 \times 23.56}{9.81 \times 70} = 0.9562 \text{ or } 95.62\%$$ **Ans.**

Now,
$$V_{f_1} = V_1 \sin 15° = 28.55 \sin 15°$$

$$V_{f_1} = 7.467 \text{ m/s}$$

$$\therefore \quad \text{Discharge, } Q = A\,V_{f_1} = 0.3 \times 7.467 = 2.24 \text{ m}^3\text{/sec}$$

$$\text{Draft tube inlet velocity, } V_2 = \frac{2.24 \times 4}{\pi \times 0.6^2} = 7.92 \text{ m/s}$$

and
$$V_3 = \frac{2.24 \times 4}{\pi \times 0.8^2} = 4.45 \text{ m/s}$$

Head saved by the draft tube through diffusion,

$$h = \frac{V_2^2 - V_3^2}{2g} = \frac{7.92^2 - 4.45^2}{2 \times 9.81} = 2.19 \text{ m}$$ **Ans.**

5. (c)

Figure below shows these losses as represented on the graph between stage efficiency and flow coefficient.

(i) Profile Losses : By profile losses, we mean the total pressure loss of two dimensional rectilinear cascade arising from the skin friction on the blade surface and due to the mixing of flow particles after the blades.

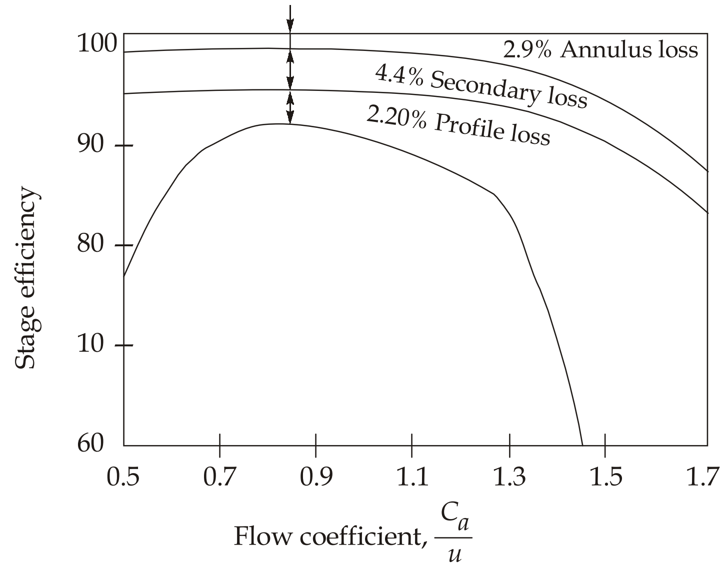


Figure : Losses in Axial Flow Compressor Stage

(ii) Skin Friction Losses on the Annulus Walls : This loss is from skin friction on the blade surface and on the annular walls. The wall friction total pressure losses from the skin friction on the annulus walls are difficult to analysis as the boundary layer growth on these walls is a complex three-dimensional phenomenon.

(iii) Secondary Flow Losses : Secondary flows in an axial flow compressor blade channels are produced by the combined effects of curvature and boundary layer. Secondary flow is produced when a streamwise component of velocity is developed from the deflection of an initially sheared flow.

As a result of curved flow path, the pressures on the concave side of passage will be higher than those on the convex side. Due to boundary layer, the flow velocities, at the inner and outer radii, i.e. at the upper and lower boundary walls of the passage are lower than those in the centre or mid-height. As result, the pressure gradients across the section are reduced and the pressure changes at the upper and lower ends of the passage, with the result, two secondary flows are set up in the blade channels.

5. (d)

(i) **Savonious type:**

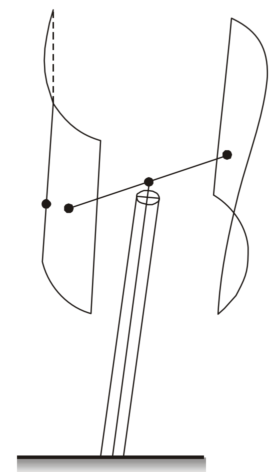


Figure: Savonious vertical-axis rotor

- The savonious rotor comprises two identical hollow semi-cylinders fixed to a vertical axis. The inner side of two half-cylinders face each other to have an 'S' shaped cross-section.
- Rotor rotates due to pressure difference between the two sides, irrespective of wind direction.
- It is self starting and the driving force is mainly of drag type.
- The rotor possesses high solidity so as to produce a high starting torque and hence this rotor is suitable for water pumping, using positive displacement pumps.
- It can extract power even from very slow winds, making it working most of the time.
- It has low speed and low efficiency.
- These are used for low power applications.

(ii) **Darreus Rotor :**

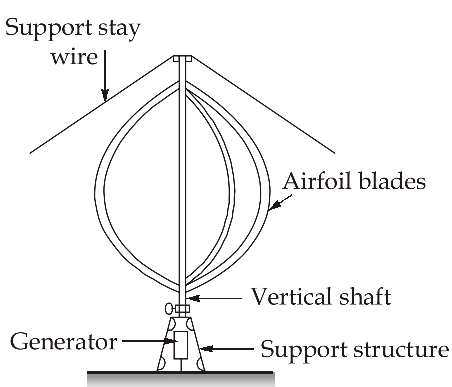


Figure: Darreus rotor

- This rotor has two or three thin curved blades of flexible metal strips, it looks like an egg beater and operates with the wind coming from any direction.
- It can be installed close to the ground eliminating the cost of the tower structure.
- Lift is the driving force, creating maximum torque when the blade moves across the wind.

- It is used for large-scale power generation.
- Power coefficient is considerably better than S-rotor.
- It runs at a large tip-speed ratio.
- As the pitch of the blade cannot change, rotor frequency increases with wind speed and power output keeps on increasing till the blades stall. Hence, at high wind speed it becomes difficult to control the output.
- One of the draw backs of this rotor is that it is usually not self-starting. Movement may be initiated by using electrical generator as motor.

(iv) Musgrove rotor:

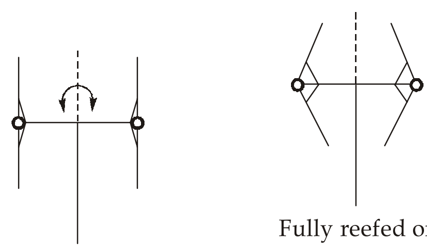


Figure: Musgrove rotor

- Musgrove suggested H shaped rotor where blades with fixed pitch are attached vertically to a horizontal cross arm.
- Inclining the blades to the vertical provides an effective means of altering the blades angle of attack and hence controlling the power output.

5. (e)

(i) Run-on Surface Ignition : When the ignition is switched off and the throttle is closed (fuel-air mixture is supplied through the idling jet), the condition in which the engine continues to fire is called run-on. It might be due to a hot surface in the cylinder, but the major cause is spontaneous ignition of the fuel-air mixture. The physical factors influencing spontaneous ignition are:

(a) an elevated temperature of the inlet mixture,

(b) poor cooling of the combustion chamber surface,

(c) duration of the valve overlap, and

(d) a high compression ratio.

The inlet temperature is elevated at the low speed condition by the low rate of air flow through the induction system, often in close proximity to the hot exhaust. At idling speed, the combustion chamber surface is not property cooled due to poor coolant circulation.

(ii) **Rumble :** Rumble is the name assigned to intermittent roughness caused by combustion chamber deposits which create secondary flame fronts. It is a low pitched noise distinctly different from spark knock. It follows that the rate of pressure rise and the maximum pressure become very high. Rumble develops early and at multiple points.

Rumble is avoided or minimized by eliminating deposits usually by fuel additives. The type of lubricating oil and gasoline without tetra ethyl lead can also reduce deposits and therefore rumble. Rumble causes vibrations of the crank shaft arising from a high rate of pressure rise with consequent deflection of mechanical parts.

Brake specific fuel consumption,

$$bsfc = \frac{\dot{m}_f}{BP} = \frac{5 \times 10^{-3}}{80} \times 3600 = 0.225 \text{ kg/kWh}$$

$$\eta_m = \frac{BP}{IP}$$

$$\Rightarrow \quad IP = \frac{80}{0.75} = 106.67 \text{ kW}$$

Indicated specific fuel consumption,

$$isfc = \frac{\dot{m}_f}{IP} = \frac{5 \times 10^{-3} \times 3600}{106.67} = 0.169 \text{ kg/kWh}$$ **Ans.**

Brake specific energy consumption,

$$bsec = \frac{kW \text{ heat input}}{kW \text{ work output}}$$

$$= \frac{\dot{m}_f \times CV}{BP} = bsfc \times CV$$

$$= \frac{0.225 \times 43000}{3600} = 2.69$$ **Ans.**

6. (a) (i)

The octane number requirement tends to go down when

(a) the ignition timing is retarded.

(b) the engine is operated at higher altitudes or smaller throttle openings or lower ambient pressures.

(c) the humidity of the air increases.

(d) the inlet air temperature is decreased.

(e) the fuel/air ratio is richer or leaner than that required for producing maximum knock.

(f) the exhaust gas recycle (EGR) system operates at part throttle.

(g) the engine load is reduced.

6. (a) (ii)

Given : Bore, D = 10 cm; Stroke, L = 10 cm; Connecting rod length, l = 17.5 cm; Crank radius, $r' = \frac{L}{2} = \frac{10}{2} = 5$ cm

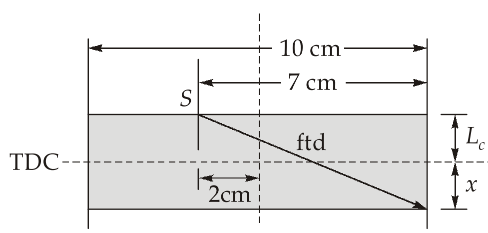
The piston displacement from TDC position,

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

$$x = 17.5 + 5 - 5\cos 13° - \sqrt{(17.5)^2 - 5^2\sin^2 13°}$$

$$= 0.164 \text{ cm}$$ **Ans.**

Referring to above figure,



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

or $$L_C = \frac{L}{r-1} = \frac{10}{8} = 1.25 \text{ cm}$$

$$L_C + x = 1.25 + 0.164 = 1.414 \text{ cm}$$

$$\text{Flame travel distance, } ftd = \sqrt{7^2 + (1.414)^2} = 7.141 \text{ cm}$$

As ignition lag is 8°, the combustion start at angle = 20° – 8° = 12° bTDC.

Rotation angle during flame propagation from 12° bTDC to 13° aTDC

$$= 12 + 13 = 25°$$

$$N = \frac{2160}{60} = 36 \text{ rps} = 36 \times 360° = 12960°/\text{sec}$$

Time of flame propagation, $t = \frac{25}{12960} = 1.93 \times 10^{-3}$ sec

$$\text{Effective flame speed, } v_f = \frac{ftd}{t} = \frac{7.141}{1.93 \times 10^{-3}} = 3700 \text{ cm/s}$$

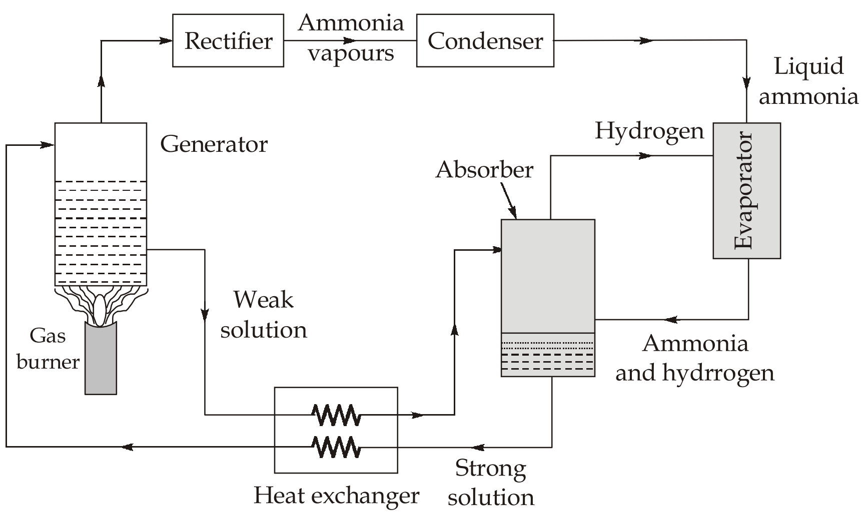
$$= 37 \text{ m/s}$$ **Ans.**

6. (b)

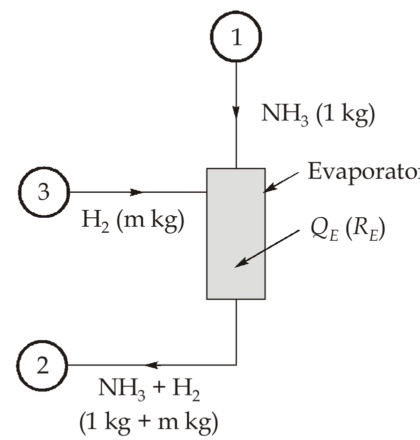
Given : $P_T = 14.71$ bar; $T_E = -15°C = 258$ K, $Q_{G_1} = 418.7$ kJ/kg; $Q_{G_2} = 1465.4$ kJ/kg

For NH_3, $h_1 = 335$ kJ/kg; $P_2 = 2.45$ bar; $h_2 = 1666$ kJ/kg, $v = v_s = 0.5$ m^3/kg

For H_3, $T_3 = 25°C = 298$ K; $R = 4.218$ kJ/kgK; $c_p = 12.77$ kJ/kgK



Domestic electrolux type refrigerator



Consider the flow of 1 kg of NH_3, heat given to the generator,

$$Q_G = Q_{G_1} + Q_{G_2} = 418.7 + 1465.4 = 1884.1 \text{ kJ/kg}$$

Since in the evaporator, $P_T = 14.71$ bar and $P_2 = P_{NH_3} = 2.45$ bar

Therefore, pressure of hydrogen (H_2),

$$P_{H_2} = P_T - P_2 = 14.71 - 2.45 = 12.26 \text{ bar}$$

Let m kg of H_2 flow through the evaporator per kg of NH_3

The ammonia (NH_3) occupies $v = 0.5$ m^3/kg. The same is the volume of H_2 at $T = 258$ K.

$$m = \frac{P_{H_2} \times v}{RT} = \frac{1226 \times 0.5}{4.218 \times 258} = 0.5633 \text{ kg/kg of } NH_3$$

Considering the energy balance, for the evaporator,

$$1 \times (h_2 - h_1) = c_p \times m(T_3 - T_2) + Q_E$$

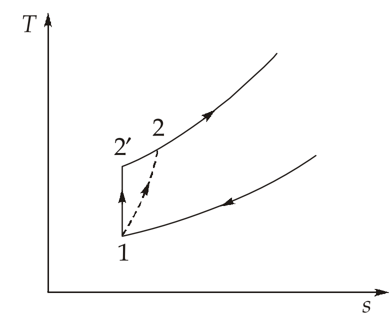
$$1 \times (1666 - 335) = 12.77 \times 0.5633 \times (298 - 258) + Q_E$$

$$\Rightarrow \quad Q_E = 1043.27 \text{ kJ/kg of } NH_3$$

$$\text{COP} = \frac{Q_E}{Q_G} = \frac{1043.27}{1884.1} = 0.55$$

6. (c)

For isentropic compression, we have



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

$$\therefore \quad T_2' = 320 \times (7.5)^{\frac{1.4-1}{1.4}}$$

$$= 569.072 \text{ K}$$

$$\text{Isentropic efficiency, } \eta_{isen,c} = \frac{T_2' - T_1}{T_2 - T_1}$$

$$\therefore \quad T_2 = T_1 + \frac{T_2' - T_1}{\eta_{isen,c}}$$

$$\therefore \quad T_2 = 320 + \frac{569.072 - 320}{0.9} = 596.746 \text{ K}$$

Therefore, power given to the air, $P = \dot{m}c_p\Delta T = 3.5 \times 1.005 \times (596.746 - 320)$

$$P = 973.454 \text{ kW}$$ **Ans.**

Temperature change per stage,

$$\Delta T_S = \frac{\Delta T}{10} = \frac{596.746 - 320}{10} = 27.67 \text{ K}$$

Work done/ kg of air second

$$\dot{P} = \dot{m} \cdot U \cdot \Delta V_w$$

or

$$c_p \Delta T_S = U \cdot \Delta V_w$$

$$\therefore \quad \Delta V_w = \frac{1.005 \times 10^3 \times 27.67}{180}$$

$$V_F(\tan\beta_1 - \tan\beta_2) = 154.49$$

$$\therefore \quad \tan\beta_1 - \tan\beta_2 = \frac{154.49}{110} = 1.4044 \quad \text{...(i)}$$

Also, Degree of reaction, $R = \frac{V_F}{2u}(\tan\beta_1 + \tan\beta_2)$

$\therefore \quad 0.5 = \frac{110}{2\times180}(\tan\beta_1 + \tan\beta_2)$

$\therefore \quad \tan\beta_1 + \tan\beta_2 = 1.636$ ...(ii)

From equation (i) and equation (ii),

$2\tan\beta_1 = 1.636 + 1.4044$

$\therefore \quad \beta_1 = 56.66°$

and from equation (ii), we get

$\beta_2 = 6.61°$

For symmetric blades,

$\alpha_1 = \beta_2 = 6.61°$

$\alpha_2 = \beta_1 = 56.66°$ **Ans.**

7. (a)

Given : mass flow rate of condensate, $\dot{m}_s$ = 18 kg/min; Mass flow rate of cooling work, $\dot{m}_w$ = 780 kg/min; Specific volume of steam at mean condensate temperature of 35°C = 25.245 m^3/kg

Partial pressure of steam at 35°C = 0.0056 MPa = 0.056 bar

Absolute condenser pressure = Barometric pressure – Vacuum reading

= 1.03 – 0.95 = 0.08 bar

Partial pressure of air = 0.08 – 0.056 = 0.024 bar

From steam table,

At 0.08 bar : h_f = 173.84 kJ/kg; h_{fg} = 2402.4 kJ/kg

Enthalpy of condensate corresponding to hot well temperature of 29°C = 121.55 kJ/kg

By energy balance, heat lost by steam = Heat gained by water

$$\dot{m}_s \times \left[h_f + xh_{fg} - h_{hot\,well}\right] = \dot{m}_w \times c_{pw} \times \left[t_{w_2} - t_{w_1}\right]$$

$$18\times[173.84 + x\times2402.4 - 121.55] = 780 \times 4.186 \times (30 - 20)$$

$$x = 0.7333$$

Hence, entering condition of steam to the condenser = 73.33% **Ans.**

Saturation pressure corresponding to 35°C = 0.056 bar

$$\text{Vacuum efficiency} = \frac{\text{Condenser vacuum}}{\text{Barometric reading} - \text{Reading pressure of steam}}$$

$$\eta_{vacuum} = \frac{0.95}{1.03 - 0.056} = 0.9753 \text{ or } 97.53\%$$ **Ans**

$$v_a = v_f + x(v_g - v_f)$$

$$= 0.001 + 0.7333 * (25.245 - 0.001) = 18.512 \text{ m}^3/\text{kg}$$

$$\text{Mass of air present, } M_a = \frac{P_a(v_a)}{RT} = \frac{0.024 \times 10^2 \times 18.512}{0.287 \times 308}$$

$$\therefore \quad M_a = 0.503 \text{ kg}$$ **Ans.**

7. (b) (i)

$$\text{Daily water requirement} = 25 \text{ m}^3/\text{day}$$

$$\text{Total dynamic head, } H_T = \text{Total vertical lift + Frictional losses}$$

$$= 12 + 12 \times 0.05 = 12.6 \text{ m}$$

Hydraulic energy required to raise water level

$$= \rho Q g H_T = 1000 \times 25 \times 9.81 \times 12.6 \times \frac{1}{3600}$$

$$= 858.375 \text{ Wh/day}$$

As solar radiation available at given location is 6 h/day, so the total wattage of PV panel = Total hydraulic energy/ Number of hours of peak sunshine/ day

$$= \frac{858.375}{6} = 143.06 \text{ W}$$

By considering system losses, the wattage of PV panel

$$= \frac{\text{Total PV panel wattage}}{\eta_{pump} \times \text{mismatch factor}} = \frac{143.06}{0.3 \times 0.85} = 561.02 \text{ W}$$

By considering operating factor, the actual value of total wattage of PV panel

$$= \frac{\text{Total PV wattage after losses}}{\text{Operating factor}}$$

$$= \frac{561.02}{0.75} = 748.03 \text{ W}$$

Number of solar PV panel required of 75 W = $\frac{748.03}{75} = 9.97 \simeq 10$ **Ans.**

So, 10 solar PV panels are required of 75 W.

$$\text{Power rating of motor} = \frac{748.03}{746} = 1.003 \text{ HP}$$ **Ans.**

(ii)

Stand alone solar PV system: The main components of a general stand-alone solar PV system are shown in figure. The MPPT senses the voltage and current outputs of the array and adjusts the operating point to extract maximum power under the given climatic conditions. The output of the array after converting to ac is fed to loads. The array output in excess of load requirement is used to charge the battery. If excess power is still available after fully charging the battery, it may be shunted to dump heaters. When the sun is not available, the battery supplies the load through an inverter. The battery discharge diode D_B prevents the battery from being overcharged after the charger is opened. The array diode D_A, is to isolate the array from the battery to prevent battery discharge through array during nights. A mode controller is a central controller for the entire system. It collects the system signals and keeps track of charge/discharge state of the battery, matches the generated power and load and commands the charger and dump heater on-off operation.

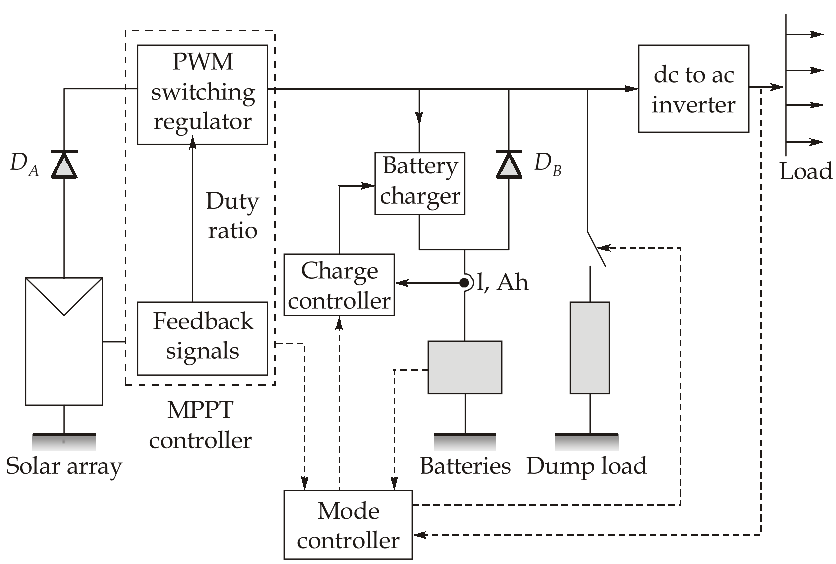
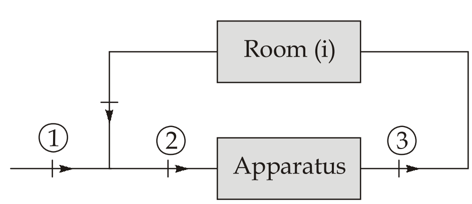


Figure: A general stand-alone solar PV system

7. (c)

$$\dot{m}_i = 45 \text{ kg/s}; t_i = 25°\text{C}; \phi_i = 50\%$$



$$\dot{m}_1 = 4 \text{ kg/s}; t_1 = 35°\text{C}; \phi_1 = 30\%$$

From mass conservation,

$$\dot{m}_i + \dot{m}_1 = m_2$$

$$\Rightarrow \quad \dot{m}_2 = 4 + 45 = 49 \text{ kg/s}$$

$$\dot{m}_i t_i + \dot{m}_1 t_1 = \dot{m}_2 t_2$$

$$\Rightarrow \quad t_2 = \frac{\dot{m}_i t_i + \dot{m}_1 t_1}{m_2} = \frac{4 \times 35 + 45 \times 25}{49} = 25.81°\text{C}$$

$$\phi_i = \frac{P_{v_i}}{P_{vs_i}}$$

$$\Rightarrow \quad P_{v_i} = 0.5 \times 0.03166 = 0.01583 \text{ bar}$$

$$\omega_i = \frac{0.622 \times P_{v_i}}{P_t - P_{v_i}}$$

$$= \frac{0.622 \times 0.01583}{1.01325 - 0.01583} = 0.0098717 \text{ kg/kg of d.a.}$$

$$\phi_1 = \frac{P_{v_1}}{P_{vs_1}}$$

$$\Rightarrow \quad P_{v_1} = 0.3 \times 0.0456291 = 0.0136887 \text{ bar}$$

$$\omega_1 = \frac{0.622 P_{v_1}}{P_t - P_{v_1}}$$

$$\omega_1 = \frac{0.622 \times 0.0136887}{1.01325 - 0.0136887} = 0.00852 \text{ kg/kg of d.a.}$$

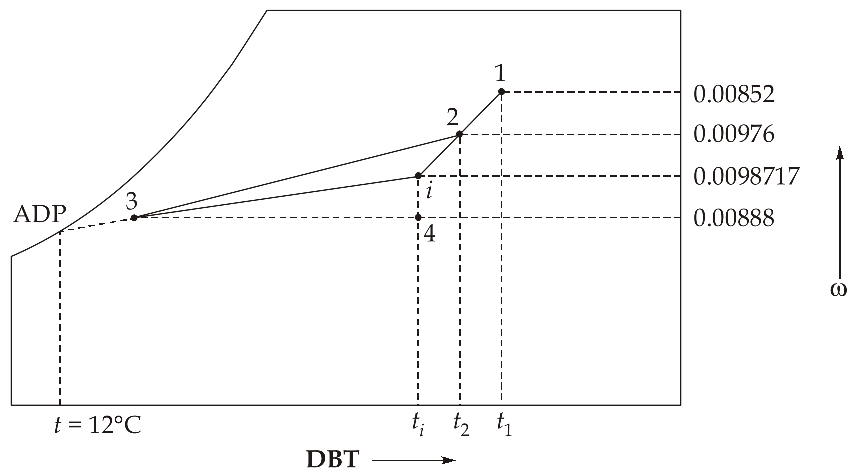
$$\omega_2 = \frac{\dot{m}_i \omega_i + \dot{m}_1 \omega_1}{\dot{m}_2} = \frac{45 \times 0.0098717 + 4 \times 0.00852}{49}$$

$$= 0.00976 \text{ kg/kg of d.a.}$$

$$\text{BPF} = \frac{t_3 - t_{ADP}}{t_2 - t_{ADP}}$$

$$\Rightarrow \quad 0.15 = \frac{t_3 - 12}{25.81 - 12}$$

$$\Rightarrow \quad t_3 = 14.0724°\text{C}$$



$$\omega_{ADP} = \frac{0.622 \times 0.014016}{1.01325 - 0.014016} \quad [\because \; \phi_{ADP} = 100\% \Rightarrow P_v = P_{vs}]$$

$$\omega_{ADP} = 0.008724 \text{ kg/kg of d.a.}$$

$$BPF = 0.15 = \frac{\omega_3 - 0.008724}{0.00976 - 0.008724}$$

$$\Rightarrow \quad \omega_3 = 0.00888 \text{ kg/kg of d.a.}$$

$$h = 1.005t + \omega \times (2500 + 1.88t)$$

$$\therefore \quad h_1 = 1.005 \times 35 + 0.00852 \times (2500 + 1.88 \times 35)$$

$$= 57.03 \text{ kJ/kg}$$

$$h_2 = 1.005 \times 25.81 + 0.00976 \times (2500 + 1.88 \times 25.81)$$

$$= 50.81 \text{ kJ/kg}$$

$$h_i = 1.005 \times 25 + 0.0098717 \times (2500 + 1.88 \times 25)$$

$$= 50.268 \text{ kJ/kg}$$

$$h_3 = 1.005 \times 14.0724 + 0.00888 \times (2500 + 1.88 \times 14.0724)$$

$$= 36.57 \text{ kJ/kg}$$

$$\omega_4 = \omega_3 = 0.00888 \text{ kg/kg of d.a.}; \; t_4 = t_i = 25°C$$

$$h_4 = 1.005 \times 25 + 0.00888 \times (2500 + 1.88 \times 25)$$

$$= 47.74 \text{ kJ/kg}$$

$$\text{Room sensible heat (RSH)} = \dot{m}_2 (h_4 - h_3) = 49(47.74 - 36.57)$$

$$= 547.33 \text{ kW}$$ **Ans.**

$$\text{Room sensible heat (RLH)} = \dot{m}_2 (h_i - h_4) = 49 \times (50.268 - 47.74)$$

$$= 123.87 \text{ kW}$$ **Ans.**

8. (a) (i)

A block diagram of MSW-to-energy incineration plant showing the sequence of various steps is shown in figure. The dry biomass is shredded to pieces of about 2.5 cm diameter.

An air stream segregates the refuse-derived fuel (RDF), which is lighter from heavier metal and glass pieces. These are reclaimed and recycled. The RDF thus obtained is burnt in a furnace at about 1000°C to produce steam in the boiler. Combustion process may be assisted by a required amount of auxiliary fuel when RDF does not burn properly by itself. The superheated steam obtained from boiler is used in a steam turbine coupled with an alternator to produce electrical output in the same way as in a conventional thermal plant. The flue gases are discharged to the atmosphere through a stack after removal of pollutants such as particulate matter, SO_x and NO_x. A heat-recovery steam generator extracts maximum possible heat from flue gases to form thermal output. The ash is removed and disposed of to landfills.

HRSG: (Heat recovery steam generator)

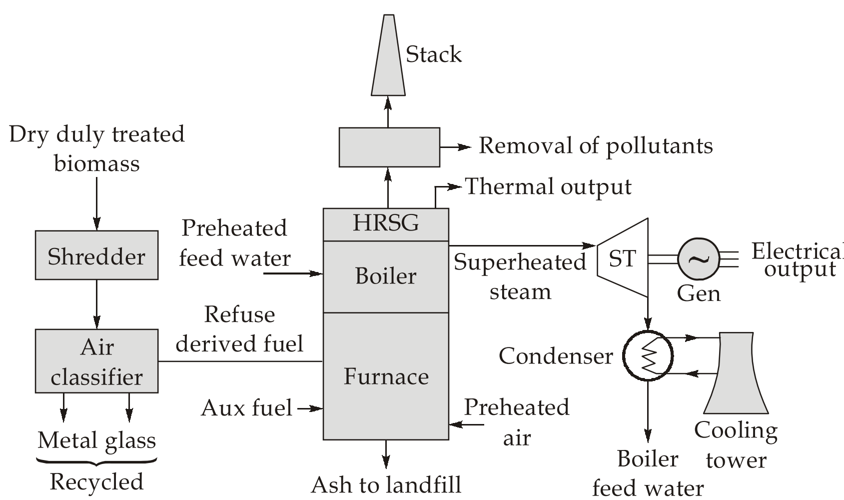


Figure: MSW to energy incineration plant

8. (a) (ii)

As the efficiency of engine is 32%, so the power to be produced = $\frac{240}{0.32} = 750$ kW

The engine is operating in dual-fuel mode, with 80% of diesel replacement, so power delivered by biomass = 0.8 × 750 = 600 kW

$$\text{Biomass feed rate, } \dot{m}_{feed} = \frac{\text{Power delivered by biomass}}{CV \times \eta_{gasifier}}$$

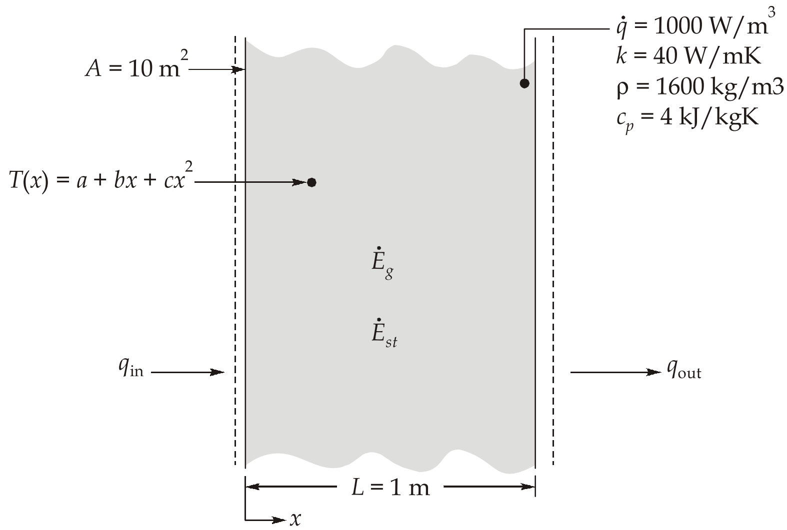
$$= \frac{600}{0.8 \times 16800} = 0.0446 \text{ kg/s} = 160.56 \text{ kg/h}$$ **Ans.**

8. (b)

Given : $T = a + bx + cx^2$

where $a = 920°C$, $b = -310°C/m$, $c = -50°C/m^2$; $x = L = 1$ m; $\dot{q} = 1000\ W/m^3$;

$\rho = 1600\ kg/m^3$; $k = 40$ W/mK; $c_p = 4$ kJ/kgK



Assumptions :

(i) One-directional conduction in the x-direction.

(ii) Isotropic medium with constant properties.

(iii) Uniform internal heat generation.

$$q_{in} = q\big|_{x=0} = -kA\cdot \frac{\partial T}{\partial x}\bigg|_{x=0}$$

$$= -kA\frac{\partial}{\partial x}\left(a + bx + cx^2\right)\bigg|_{x=0} = -kA(b + 2cx)_{x=0}$$

$$= -k\cdot A\cdot b = -40 \times 10 \times (-310) = 124000 \text{ W}$$

$$= 124 \text{ kW}$$ **Ans.**

$$q_{out} = q\big|_{x=1\text{m}} = -kA\frac{\partial T}{\partial x}\bigg|_{x=1\text{ m}}$$

$$= -kA\frac{\partial}{\partial x}\left(a + bx + cx^2\right)\bigg|_{x=1\text{ m}}$$

$$= -k\cdot A\cdot (b + 2cx)_{x = 1\text{ m}}$$

$$= -40 \times 10 \times [-310 + 2 \times (-50) \times 1]$$

$$= 164000 \text{ W} = 164 \text{ kW}$$ **Ans.**

The rate of change of energy storage in the wall $\dot{E}_{st}$ can be determined by applying an overall energy balance to the wall. For the control volume about the wall,

$$\dot{E}_{in} + \dot{E}_g - \dot{E}_{out} = \dot{E}_{st}$$

where, $$\dot{E}_g = \dot{q} \cdot A L$$

$$\Rightarrow \quad \dot{E}_{st} = \dot{E}_{in} + \dot{E}_g - \dot{E}_{out} = q_{in} + \dot{q} \cdot A L - q_{out}$$

$$= 124 + (1 \times 10) - 164$$

$$= -30 \text{ kW}$$ **Ans.**

The time rate of change of temperature at any point in the medium may be determined from the heat equation,

$$\frac{\partial T}{\partial t} = \frac{k}{\rho c_p}\frac{\partial^2 T}{\partial x^2} + \frac{\dot{q}}{\rho c_p}$$

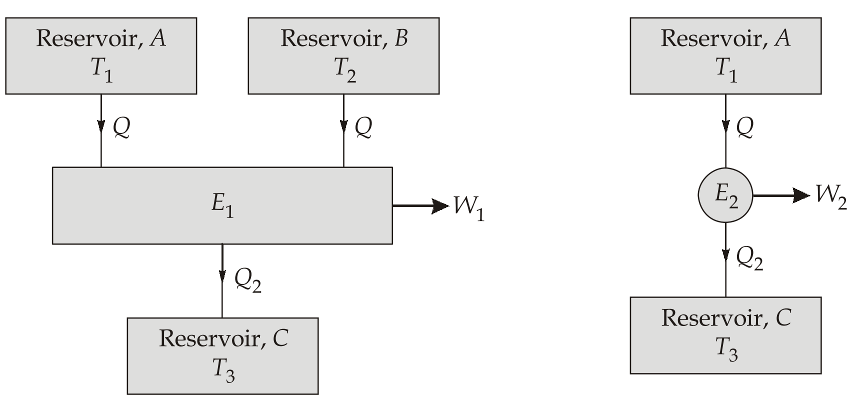
The derivative is independent of position in the medium. Hence, the time rate of temperature change is also independent of position and is given by,

$$\frac{\partial^2 T}{\partial x^2} = \frac{\partial^2}{\partial x^2}\left(a + bz + cx^2\right) = 2c = 2 \times (-50) = -100°\text{C/m}^2$$

$$\frac{\partial T}{\partial t} = \frac{40}{1600 \times 4000} \times (-100) + \frac{1000}{1600 \times 4000}$$

$$= -4.6875 \times 10^{-4}\,°\text{C/s}$$ **Ans.**

8. (c)



Now, $$\eta_I = k \cdot \eta_I$$ ...(i)

For case I, $$\eta_I = 1-\frac{Q_2}{Q+Q} = 1-\frac{Q_2}{2Q} \quad \text{...(ii)}$$

For reversible engine,

$$\frac{Q}{T_1}-\frac{Q_2}{T_3}+\frac{Q}{T_2} = 0$$

$$\Rightarrow \quad Q\left[\frac{1}{T_1}+\frac{1}{T_2}\right] = \frac{Q_2}{T_3}$$

$$\Rightarrow \quad \frac{Q_2}{Q} = T_3\left[\frac{1}{T_1}+\frac{1}{T_2}\right]$$

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

$$\eta_I = 1-\frac{1}{2}\left[T_3\left(\frac{1}{T_1}+\frac{1}{T_2}\right)\right] \quad \text{...(iii)}$$

For case II, $$\eta_{II} = 1-\frac{Q_2}{Q} = 1-\frac{T_3}{T_1} \quad \text{...(iv)}$$

Putting the values of equation (iii) and (iv) in equation (i), we get

$$\Rightarrow \quad 1-\frac{1}{2}\left[T_3\left(\frac{1}{T_1}+\frac{1}{T_2}\right)\right] = k\left[1-\frac{T_3}{T_1}\right]$$

$$\Rightarrow \quad 1-\frac{1}{2}\left[\frac{T_3\cdot T_1+T_2\cdot T_3}{T_1\cdot T_2}\right] = k\left[\frac{T_1-T_3}{T_1}\right]$$

$$\Rightarrow \quad k\left[\frac{T_1-T_3}{T_1}\right] = \frac{2T_1T_2-T_3T_1-T_2T_3}{2T_1T_2}$$

$$\Rightarrow \quad k = \frac{1}{2}\left(\frac{T_1}{T_1-T_3}\right)\times\left[\frac{T_1T_2-T_1T_3+T_1T_2-T_2T_3}{T_1T_2}\right]$$

$$\Rightarrow \quad k = \frac{1}{2}\left(\frac{T_1}{T_1-T_3}\right)\times\left[T_1(T_2-T_3)-T_2(T_3-T_1)\right]\times\frac{1}{T_2}\times\frac{1}{T_1}$$

$$\Rightarrow \quad k = \frac{1}{2}\left(\frac{T_1}{T_2}\right)\times\left[\frac{T_1(T_2-T_3)-T_2(T_3-T_1)}{T_1(T_1-T_3)}\right]$$

$$\Rightarrow \quad k = \frac{1}{2}\left(\frac{T_1}{T_2}\right)\times\left[\frac{T_2-T_3}{T_1-T_3}+\frac{T_2}{T_1}\right]$$

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