MADE EASY
Leading Institute for ESE, GATE & PSUs

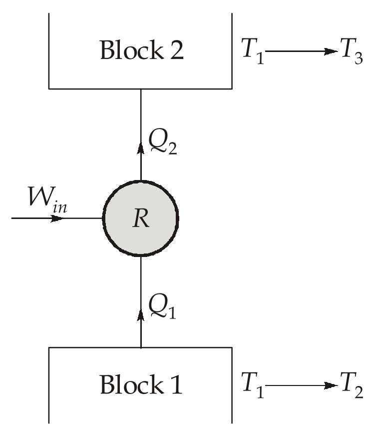
Detailed Solutions

ESE-2026 Mains Test Series

Mechanical Engineering Test No : 10

Section : A

1. (a) Solution:



Given : Mass of each block, m = 20 kg; Specific heat of block, C = 0.5 kJ/kgK

The initial temperature of both blocks is T_1 = 50°C = 323 K

Assumption : Specific heat of the metal remains constant throughout the process.

Let the final temperature of first block be T_2 and that of second block be T_3 and $T_3 > T_2$

i.e. $$T_3 - T_2 = 150°C$$

Heat absorbed from block 1;

$$Q_1 = mc(T_1 - T_2)$$

Heat rejected to block 2;

$$Q_2 = mc(T_3 - T_1)$$

From the energy balance equation;

$$Q_1 + W_{in} = Q_2$$

$$\therefore \quad W_{in} = Q_2 - Q_1$$

For minimum work input the process must be reversible, so for reversible process:

$$(\Delta s)_{universe} = 0$$

$$\Rightarrow (\Delta s)_1 + (\Delta s)_2 + (\Delta s)_R + (\Delta s)_{surr} = 0$$

$$\Rightarrow \quad mc\ln\frac{T_2}{T_1} + mc\ln\frac{T_3}{T_1} + 0 + 0 = 0$$

$$\Rightarrow \quad mc\ln\frac{T_2T_3}{T_1^2} = 0$$

$$\Rightarrow \quad T_2T_3 = T_1^2$$

$$\Rightarrow \quad T_2T_3 = (323)^2 = 104329 \quad \text{...(i)}$$

So, work required $= W_{in} = Q_2 - Q_1$

$$\Rightarrow \quad (W_{in})_{min} = mc\,(T_3 - T_1 - T_1 + T_2)$$

$$= mc\,(T_3 + T_2 - 2T_1) \quad \text{...(ii)}$$

Also, we have

$$T_3 - T_2 = 150°\text{C} = 150 \text{ K} \quad \text{...(iii)}$$

From equation (i) and (iii)

$$T_3 = 406.593 \text{ K and } T_2 = 256.593 \text{ K}$$

Hence, $(W_{in})_{min} = 20 \times 0.5(406.593 + 256.593 - 2 \times 323)$

$= 171.86$ kJ **Ans.**

1. (b) Solution:

Given : Number of cylinders = 4; Orifice diameter = 6 cm; Coefficient of discharge (C_d) = 0.65; Bore (D) = 12 cm, Stroke (L) = 14 cm; N = 1500 rpm; Brake torque = 157 N-m; Fuel consumption = 6.5 kg/hr; Head across orifice (h_w) = 6.7 cm of water

(i)

$$\text{B.P.} = \frac{2\pi NT}{60} = \frac{2\pi \times 1500 \times 157}{60} = 24.66 \text{ kW}$$

$$\text{Thermal efficiency} = \frac{B.P. \times 3600}{\dot{m}_f \times C.V}$$

$$\text{Efficiency, } \eta_{th} = \frac{24.66 \times 3600}{6.5 \times 43100} = 0.31688 = 31.688\% \qquad \text{Ans.}$$

(ii)

$$\text{bmep} = \frac{B.P.}{LAN \times \text{Number of cylinders}}$$

$$\text{bmep} = \frac{24.66 \times 10^3}{0.14 \times \frac{\pi}{4} \times (0.12)^2 \times \frac{1500}{2 \times 60} \times 4}$$

$$\text{bmep} = 3.1145 \text{ bar} \qquad \text{Ans.}$$

(iii)

$$\text{Volume flow rate of air} = C_d A \sqrt{2 g h_w \times \frac{\rho_w}{\rho_a}}$$

$$\rho_a = \frac{P}{RT} = \frac{1 \times 10^5}{0.287 \times (22 + 273) \times 10^3} = 1.181 \text{ kg/m}^3$$

$$\therefore \quad Q_{air} = 0.65 \times \frac{\pi}{4} \times (0.06)^2 \sqrt{2 \times 9.81 \times 0.067 \times \frac{1000}{1.18}}$$

$$Q_{air} = 0.0613 \text{ m}^3/\text{s}$$

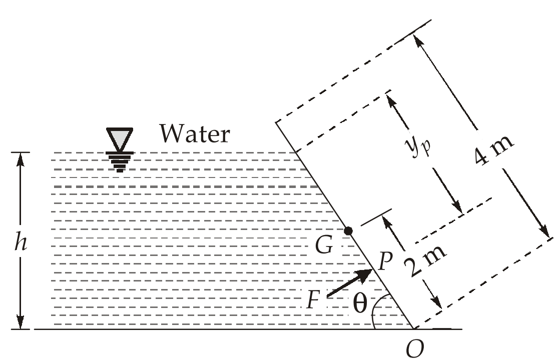
Swept volume = Number of cylinders × AL × Number of working cycles

$$= 4 \times \frac{\pi}{4} \times (0.12)^2 \times 0.14 \times \frac{1500}{2 \times 60} = 0.079168 \text{ m}^3/\text{s}$$

$$\text{Volumetric efficiency} = \frac{\text{Volume inhaled}}{\text{Swept volume}}$$

$$= \frac{0.0613}{0.079168} = 0.7743 = 77.43\% \qquad \text{Ans.}$$

1. (c) Solution:



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)

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

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. (d) Solution:

Given : Vertical Kiln diameter (d) = 6.5 m; Thickness of chrome brick = 40 cm; Thermal conductivity (k) = 1.18 W/mK; Kiln inside temperature (T_1) = 1000°C; Surrounding temperature (T_∞) = 30°C; Convective heat transfer coefficient (h) = 15 W/m^{2}K; Inside radius (r_1) = $\frac{6.5}{2} = 3.25$ m ; Outside radius (r_2) = 3.25 + 0.4 = 3.65 m

(i) The heat loss from hemispherical dome (q) = $\frac{1}{2} \times \frac{(T_1 - T_2)}{\frac{1}{4\pi k}\left(\frac{1}{r_1} - \frac{1}{r_2}\right)}$

or
$$q = \frac{1}{2} \times \frac{4\pi k (T_1 - T_2) r_1 r_2}{(r_2 - r_1)}$$

$$= \frac{0.5 \times 4\pi \times 1.18(1000 - T_2) \times 3.25 \times 3.65}{3.65 - 3.25}$$

or, $$q = 219.8761\ (1000 - T_2)\text{W}$$

The convective heat transfer from the dome to outside air

$$q = hA(T_2 - T_0) = 15 \times 2 \times \pi \times (3.65)^2 \times (T_2 - 30)$$

$$q = 1255.6159\,(T_2 - 30)\ \text{W}$$

Since same heat loss occurs from each section

$$219.8761(1000 - T_2) = 1255.6159\,(T_2 - 30)$$

or, $$1000 - T_2 = 5.7105T_2 - 171.3168$$

or, $$T_2 = \frac{1171.3168}{6.7105} = 174.549°\text{C}$$ Ans.

So, $$q = 1255.6159\ (174.544 - 30) = 181.5\ \text{kW}$$ Ans.

(ii)

For a dome with flat top

$$\frac{kA(T_1 - T_2)}{L} = hA(T_2 - T_0)$$

or, $$\frac{1.18 \times (1000 - T_2)}{0.4} = 15(T_2 - 30)$$

or, $$1000 - T_2 = 5.0847T_2 - 152.5423$$

$$T_2 = \frac{1152.5423}{6.0847} = 189.416°\text{C}$$

Hence, $$q' = hA(T_2 - 30)$$

$$= 15 \times \pi \times 3.25^2 \times (189.416 - 30)$$

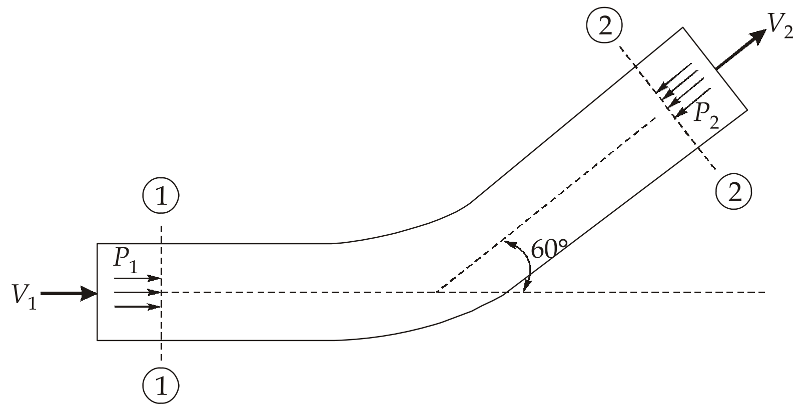
$$= 79.35\ \text{kW}$$

$$\text{Reduction in heat loss} = \frac{q - q'}{q} \times 100 = \frac{181.5 - 79.35}{181.5} \times 100$$

$$= 56.28\%$$ **Ans.**

1. (e) Solution:

Given : Pipe diameter (D) = 350 mm; Head (H) = 25 m; Velocity (V) = 4.5 m/s



The pipe is of uniform cross-sectional area. Therefore, the flow velocities at sections 1-1 and 2-2 are same.

$$V_1 = V_2 = 4.5 \text{ m/s}$$

$$\text{Flow rate, } Q = A_1V_1 = \frac{\pi}{4}\times(0.35)^2\times 4.5$$

$$Q = 0.4329 \text{ m}^3/\text{s}$$

The pressure intensity is same at the two sections.

$$P_1 = P_2 = \rho g H = 9810 \times 25 = 245250 \text{ N/m}^2$$

Force exterted by the bend along the *x*-axis

$$F_x = \text{Dynamic force} + \text{Static force}$$
$$= \rho Q(V_1 - V_2\cos 60°) + P_1A_1 - P_2A_2\cos 60°$$

$$F_x = 1000 \times 0.4329(4.5 - 4.5\times 0.5) + 245250\times\frac{\pi}{4}(0.35)^2 - 245250\times\frac{\pi}{4}\times(0.35)^2\times 0.5$$

or $$F_x = 974.025 + 23595.8152 - 11797.9076$$

$$F_x = 12771.9326 \text{ N}$$

Force exterted by the bend along the *y*-axis

$$F_y = \text{Dynamic force} + \text{Static force}$$
$$= \rho Q(0 + V_2\sin 60°) + P_2A_2\sin 60°$$

$$= 1000\times 0.4329\times\left(4.5\times\frac{\sqrt{3}}{2}\right) + 245250\times\frac{\pi}{4}\times(0.35)^2\times\frac{\sqrt{3}}{2}$$

$$= 1687.0607 + 20434.57538$$

$\therefore$ $$F_y = 22121.63608 \text{ N}$$

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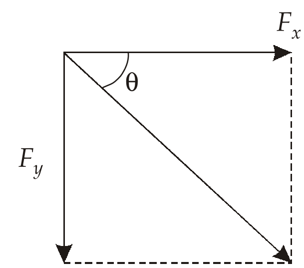
The magnitude of resultant force acting on the bend is



$$F = \sqrt{F_x^2 + F_y^2}$$

$$F = \sqrt{(12771.9326)^2 + (22121.63608)^2}$$

$$F = 25543.865 \text{ N}$$ **Ans.**

and the direction of resultant force with positive x-axis is

$$\tan\theta = \frac{F_y}{F_x} = \frac{22121.63608}{12771.9326} = 1.732$$

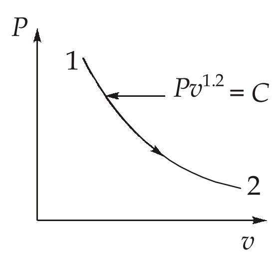
or, $\theta = 60°$ **Ans.**

An equal and opposite force will be required to hold the duct in position.

2. (a)

Given: $P_1 = 15$ bar; $T_1 = 300°\text{C} = 573$ K; $n = 1.2$; $P_2 = 1.3$ bar

(i) If the fluid is air:



The process is reversible polytropic

$$\therefore \quad P_1 v_1^n = P_2 v_2^n$$

Final volume:

$$\Rightarrow \quad v_2 = v_1 \times \left(\frac{P_1}{P_2}\right)^{\frac{1}{n}} = \frac{RT_1}{P_1} \times \left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$$

$$= \frac{0.287 \times 573}{1500} \times \left(\frac{15}{1.3}\right)^{\frac{1}{1.2}} = 0.8415 \text{ m}^3/\text{kg}$$

Final temperature:

$$\frac{P_1 v_1}{T_1} = \frac{P_2 v_2}{T_2}$$

$\Rightarrow$

$$T_2 = T_1 \times \frac{P_2 v_2}{P_1 v_1}$$

$$= 573 \times \frac{1.3 \times 0.8415}{15 \times 0.1096} = 381.298 \text{ K}$$

$$\left(v_1 = \frac{RT_1}{P_1} = 0.1096 \text{ m}^3/\text{kg}\right)$$

Heat transfer:

$$\delta Q = dU + \delta W$$

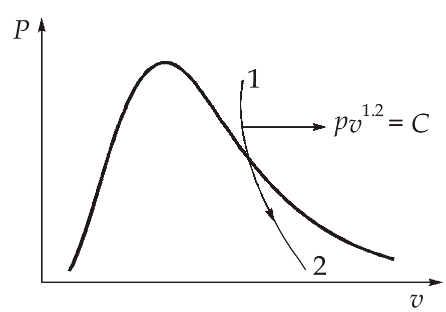
$$= c_v (T_2 - T_1) + \frac{P_2 v_2 - P_1 v_1}{1 - n}$$

$$= 0.718 \times (381.298 - 573) + \frac{1.3 \times 100 \times 0.8415 - 15 \times 100 \times 0.1096}{1 - 1.2}$$

$$= -137.642 + 275.025$$

$$= 137.383 \text{ kJ/kg}$$

(ii) If the fluid is steam:



From steam table:

At 15 bar and 300°C

$$v_1 = 0.16971 \text{ m}^3/\text{kg}; \ u_1 = 2783.6 \text{ kJ/kg}$$

$\therefore$ **Final volume:**

$$v_2 = v_1 \left(\frac{P_1}{P_2}\right)^{\frac{1}{n}} = 0.16971 \times \left(\frac{15}{1.3}\right)^{\frac{1}{1.2}} = 1.30266 \text{ m}^3/\text{kg}$$

At 1.3 bar; $v_f = 0.00104917$ m^3/kg

$v_g = 1.3253$ m^3/kg; $u_f = 449.05$ kJ/kg; $u_g = 2514.3$ kJ/kg

Since, $v_2 < v_g$ so the state 2 is in the wet region.

$$\therefore \quad v_2 = v_f + x_2(v_g - v_f)$$

$$\Rightarrow \quad 1.30266 = 0.00104917 + x_2 \times (1.3253 - 0.00104917)$$

$$\Rightarrow \quad x_2 = 0.983$$

$$\therefore \quad u_2 = u_f + x_2(u_g - u_f)$$

$$= 449.05 + 0.983 \times (2514.3 - 449.05)$$

$$= 2478.99 \text{ kJ/kg}$$

Final temperature:

$$T_2 = (T_{sat})_{@1.3 \text{ bar}} = 107.109°\text{C} = 380.109 \text{ K}$$

Heat transfer:

$$\delta q = du + \delta w$$

$$= (u_2 - u_1) + \frac{P_1 v_1 - P_2 v_2}{n-1}$$

$$= (2478.99 - 2783.6) + \frac{15 \times 100 \times 0.16971 - 1.3 \times 100 \times 1.30266}{1.2 - 1}$$

$$= -304.61 + (426.096) = 121.486 \text{ kJ/kg}$$

2. (b) Solution:

Given: $p = 2$, $\dot{Q} = 20$ MW, SSC = 5 kg/kWh, $x = 0.8$, $p_{sat} = \frac{76-66}{76} \times 1.0133 = 0.133$ bar,

$t_{sat} = 51°$C, $h_{fg} = 2590$ kJ/kg, $D_i = 30$ mm, $D_o = 40$ mm, $V = 10$ m/s, $t_{c1} = 25°$C,

$t_{c2} = 51 - 7 = 44°$C, $U = 5000$ W/m^{2}K

Steam condensed, $\dot{m}_h = \frac{SSC \times \dot{Q}}{3600} = \frac{5 \times 20000}{3600} = 27.777$ kg/sec

From energy balance,

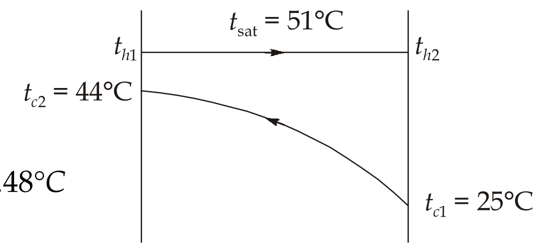
$$\dot{m}_h \times (x h_{fg}) = \dot{m}_c c_c \times (t_{c2} - t_{c1})$$

where, suffix 'h' and suffix 'c' are for hot fluid and cold fluid respectively.

$$\therefore \quad 27.777 \times 0.8 \times 2590 = \dot{m}_c \times 4.18 \times (44 - 25)$$

$$\therefore \quad \dot{m}_c = 724.7 \text{ kg/sec}$$ Ans. (i)

Now, assuming counter flow arrangement,



$$\theta_1 = 51 - 44 = 7°C$$

$$\theta_2 = 51 - 25 = 26°C$$

∴ LMTD, $$\theta_m = \frac{\theta_1 - \theta_2}{\ln\frac{\theta_1}{\theta_2}} = \frac{7-26}{\ln\frac{7}{26}} = 14.48°C$$

∴ Heat exchanger, $Q = \dot{m}_h \cdot x \cdot h_{fg} = UA\theta_m$

∴ $27.777 \times 0.8 \times 2590 = 5 \times A \times 14.48$

∴ $A = 794.94 \text{ m}^2$ Ans. (ii)

Mass flow rate of water through each tube,

$$\dot{m} = \rho A_i V = 10^3 \times \frac{\pi}{4} \times 0.03^2 \times 10 = 7.068 \text{ kg/s}$$

∴ Number of tubes per pass,

$$n = \frac{\dot{m}_c}{\dot{m}} = \frac{724.7}{7.068}$$

$$n = 102.5 \text{ or } 103 \text{ tubes per pass}$$

Now, Area of heat transfer, $A = 794.94 \text{ m}^2$

or, $n\, p\, \pi\, D_o\, l = 794.94$

$103 \times 2 \times \pi \times 0.04 \times l = 794.94$

∴ $$l = \frac{794.94}{103 \times 2 \times \pi \times 0.04} = 30.7 \text{ m}$$ Ans. (iii)

Thus, 103 tubes of length 30.7 m are required in each pass.

2. (c) Solution:

$$\text{Indicated power (IP)} = \frac{P_{im} \times L \times A \times n}{60 \times 1000 \times 30}$$

$$= \frac{4.5 \times 10^5 \times 0.3 \times \frac{\pi}{4} \times (0.2)^2 \times 3950}{60 \times 1000 \times 30} = 9.307 \text{ kW}$$

$$\text{Brake power (BP)} = T \times \omega$$

$$= W \times r \times \frac{2\pi N}{60}$$

$$= 50 \times 9.81 \times 0.5 \times \frac{2\pi \times 8000}{60 \times 30 \times 1000} = 6.848 \text{ kW}$$

$$\text{Heat supplied at NTP} = \frac{2.5}{30}\times 20000 = 1666.67 \text{ kJ/min}$$

Heat equivalent of brake power (BP) = 6.848 × 60 = 410.88 kJ/min

$$\text{Heat lost to cooling medium} = (mc\Delta T)_W$$

$$= \frac{85}{30}\times 4.18\times 32 = 378.986 \text{ kJ/min}$$

Total air used = 35 m^3 at 710 mm of Hg

$$\text{Volume of air used at NTP} = 35\times\frac{273}{291}\times\frac{710}{760} \qquad \left(\because \frac{PV}{T} = C\right)$$

$$= 30.675 \text{ m}^3$$

$$\text{Mass of air used} = \frac{30.675\times 1.25}{30} = 1.278 \text{ kg/min}$$

$$\text{Mass of gas at NTP, } m_g = \frac{PV}{RT} = \frac{1.0132\times 10^5\times 2.5}{287\times 273} = 3.233 \text{ kg}$$

$$\text{Mass of gas/min} = \frac{3.233}{30} = 0.1077 \text{ kg/min}$$

Total mass of exhaust gas = 1.278 + 0.1077 = 1.385 kg/min

Heat lost to exhaust gas = 1.385 × 1.1 × (360 – 18) = 521.037 kJ/min

Unaccounted heat lost = 1666.67 – (410.88 + 378.986 + 521.037)

= 355.767 kJ/min

$$\text{Mechanical efficiency, } \eta_m = \frac{BP}{IP}\times 100$$

$$= \frac{6.848}{9.307}\times 100 = 73.58\%$$

Indicated thermal efficiency,

$$\eta_{ith} = \frac{IP}{\text{Heat supplied}}\times 100$$

$$= \frac{9.307\times 60}{1666.67}\times 100 = 33.505\%$$

Heat Balance Sheet :

Heat input (per minute)	kJ	Heat expenditure (per minute)	kJ
Heat supplied by fuel	1666.67	1. Heat equivalent to Brake power (BP)	410.88
		2. Heat lost to cooling medium	378.986
		3. Heat lost in exhaust	521.037
		4. Unaccounted loss	355.767
		Total	1666.67

3. **(a) (i) Solution:**

Assumptions:

1. Steady incompressible flow
2. Neglect frictional losses other than the loss due to sudden enlargement.

For a sudden enlargement, the loss of head is given by

$$h_1 = \left(1 - \frac{A_1}{A_2}\right)^2 \frac{u_1^2}{2g} \qquad \text{[As by continuity equation } u_2 = u_1 \frac{A_1}{A_2}\text{]}$$

where, 1 represents smaller pipe and 2 represents larger diameter pipe.

Applying the energy equation between a point just before enlargement and a point just after enlargement.

$$\frac{u_1^2}{2g} + \frac{P_1}{\rho g} = \frac{u_2^2}{2g} + \frac{P_2}{\rho g} + h_L$$

$$\frac{u_1^2}{2g} + \frac{P_1}{\rho g} = \frac{u_2^2}{2g} + \frac{P_2}{\rho g} + \left(1 - \frac{A_1}{A_2}\right)^2 \frac{u_1^2}{2g}$$

By continuity equation,

$$u_1 A_1 = u_2 A_2 = Q$$

$$u_2 = u_1 \frac{A_1}{A_2}$$

then,

$$\frac{P_2 - P_1}{\rho g} = \frac{u_1^2}{2g} - \frac{u_1^2}{2g}\left(\frac{A_1}{A_2}\right)^2 - \left(1 - \frac{A_1}{A_2}\right)^2 \frac{u_1^2}{2g}$$

$$\frac{\Delta P}{\rho g} = \left[1 - \left(\frac{A_1}{A_2}\right)^2 - \left(1 - \frac{A_1}{A_2}\right)^2\right] \frac{u_1^2}{2g}$$

Assuming $\frac{A_1}{A_2} = r$

$$\frac{\Delta P}{\rho g} = \left[1 - r^2 - (1-r)^2\right]\frac{u_1^2}{2g}$$

For a given pipe and a given discharge, u_1 is constant. The pressure rise would be maximum when

$$\frac{d}{dr}\left(\frac{\Delta P}{\rho g}\right) = 0$$

$$0 - 2r + 2(1-r) = 0$$

$$2 - 4r = 0$$

$$r = \frac{1}{2} = \frac{A_1}{A_2}$$

$\because$
$$\frac{A_1}{A_2} = \left(\frac{D_1}{D_2}\right)^2 = \frac{1}{2}$$

$\Rightarrow$
$$\frac{D_1}{D_2} = \frac{1}{\sqrt{2}}$$

The maximum pressure rise is given by

$$\frac{\Delta P}{\rho} = \left[1 - \frac{1}{4} - \left(1 - \frac{1}{2}\right)^2\right]\frac{u_1^2}{2}$$

$$\frac{\Delta P}{\rho} = \frac{1}{4}u_1^2$$

The maximum pressure head may be expressed as

$$h = \frac{\Delta P}{\rho g} = \frac{1}{2}\frac{u_1^2}{2g} \qquad \left[\text{As by continuity equation } u_2 = u_1\frac{A_1}{A_2}\right]$$

Which is half of velocity head in smaller pipe

3. (a) (ii) Solution:

Applying Euler's equation along r, we obtain

$$\frac{1}{\rho}\frac{\partial P}{\partial r} = \frac{V^2}{r}$$

$$\frac{\partial P}{\partial r} = \rho\frac{V^2}{r} = \rho\frac{V_\theta^2}{r} \qquad \text{where, } V_\theta = \frac{c}{r}$$

$$\frac{\partial P}{\partial r} = \frac{\rho c^2}{r^3}$$

On integrating,

$$P_2 - P_1 = \rho c^2 \int_{R_1}^{R_2} r^{-3} dr$$

$$P_2 - P_1 = \rho c^2 \left[\frac{r^{-2}}{-2}\right]_{R_1}^{R_2}$$

$$= \frac{\rho c^2}{2}\left[\frac{R_2^2 - R_1^2}{R_1^2 R_2^2}\right] \quad \text{...(i)}$$

The constant c can be written in terms of the mass flow rate m is given by

$$\dot{m} = \int_{R_1}^{R_2} \rho V_\theta drb = \rho b \int_{R_1}^{R_2} \frac{c}{r} dr$$

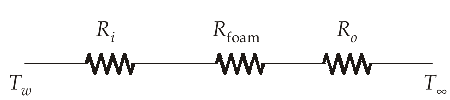
$$\dot{m} = \rho bc \ln\frac{R_2}{R_1}$$

$$c = \frac{\dot{m}}{\rho b \ln\frac{R_2}{R_1}} \quad \text{...(ii)}$$

Substituting c in equation (i) we can write

$$P_2 - P_1 = \frac{\dot{m}^2}{2\rho b^2 \left(\ln\frac{R_2}{R_1}\right)^2}\left[\frac{R_2^2 - R_1^2}{R_1^2 R_2^2}\right]$$

3. (b) Solution:



$T_w = 55°C, \quad r_1 = 20 \text{ cm} = r_i$

$T_\infty = 27°C, \quad r_2 = 23 \text{ cm} = r_o$

$h_i = 50 \text{ W/m}^2 °C$

$h_o = 12 \text{ W/m}^2 °C$

$K_{insulation} = 0.03 \text{ W/m°C}$

Price of electricity = ₹ 0.08/kWh

$D_o = 6 \text{ cm} + 40 \text{ cm} = 46 \text{ cm}$

$$D_i = 40 \text{ cm}$$

$$A_i = \pi D_i L = \pi \times 0.4 \times 2 = 2.5132 \text{ m}^2$$

$$R_i = \frac{1}{h_i A_i} = \frac{1}{50 \times 2.5132} = 0.00795 \text{ °C/W}$$

$$A_o = \pi D_o L = \pi \times 0.46 \times 2 = 2.89 \text{ m}^2$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{12 \times 2.89} = 0.0288 \text{ °C/W}$$

$$R_{\text{foam}} = \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi KL} = \frac{\ln\left(\frac{23}{20}\right)}{2\pi \times 0.03 \times 2} = 0.37073 \text{ °C/W}$$

$$R_{\text{total}} = R_i + R_{\text{foam}} + R_o$$

$$= 0.00795 + 0.0288 + 0.37073$$

$$= 0.40748 \text{ °C/W}$$

Rate of heat loss from the hot water tank is

$$\dot{Q} = \frac{T_w - T_\infty}{R_{\text{total}}} = \frac{55 - 27}{0.40748} = 68.7150 \text{ W}$$

The amount and cost of heat loss per year

$$Q = \dot{Q}.\Delta t$$

$$= 0.068715 \times 365 \times 8 \text{ h/yr.kW}$$

$$= 200.6478 \text{ kWh/yr}$$

Cost of energy = amount of energy × unit cost

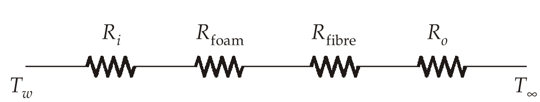
= 200.6478 × ₹0.08 kWh

Cost of energy = ₹16.05 per year

Fraction of the annual heating cost

$$= \frac{16.05}{280} = 5.73\%$$

If 3 cm thick fibre glass insulation is used to wrap the entire tank, the individual resistances becomes.



$$r_3 = 46 + 6 = 52 \text{ cm,}$$

$$A_o = \pi D_o L = \pi \times 0.52 \times 2 = 3.267 \text{ m}^2$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{12 \times 3.267} = 0.0255 \text{°C/W}$$

$$R_{\text{foam}} = \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi k_1 L} = \frac{\ln\left(\frac{23}{20}\right)}{2\pi \times 0.03 \times 2} = 0.3707°\text{C/W}$$

$$R_{\text{fibre glass}} = \frac{\ln\left(\frac{r_3}{r_2}\right)}{2\pi k_2 L} = \frac{\ln\left(\frac{26}{23}\right)}{2\pi \times 0.035 \times 2} = 0.2787°\text{C/W}$$

$$R_{\text{total}} = R_o + R_{\text{foam}} + R_{\text{fibre}} + R_o$$
$$= 0.00795 + 0.3707 + 0.2787 + 0.0255$$
$$= 0.68285°\text{C/W}$$

Rate of heat loss from the hot water heater in this case

$$\dot{Q} = \frac{T_w - T_\infty}{R_{\text{total}}} = \frac{55 - 27}{0.68285} = 41.0046 \text{ W}$$

The energy saving is = 68.7150 – 41.0046 = 27.7104 W

The time necessary for this additional insulation to pay for its cost of \$30 is then

$$\text{Cost} = \frac{27.7104 \times 8}{1000} \times t \times 0.08$$

$$30 = 0.2216832 \times t \times 0.08$$

$$t = 1691.6 \text{ days or } 4.63 \text{ years}$$

3. (c) Solution:

Given data : Bore, d = 60 mm; 4 stroke engine, $N_{cycle} = N/2$, $k = 4$

Stroke, L = 90 mm; Speed, N = 3000 rpm; torque arm, r = 0.35 m; W = 150 N

$\dot{V}_f$ = 5.5 litre/hr; ρ_f = 800 kg/m^3; CV = 45 MJ/kg fuel

Morse-Test results:

Engine cut out	1	2	3	4
Corresponding brake-load	104 N	100 N	102 N	108 N

$$\dot{m}_f = \dot{V}_f \cdot \rho = \left(\frac{5.5}{1000}\right)\frac{1}{3600} \times 800$$

$$= \frac{11}{9000} \text{ kg/s} \simeq 1.2222 \times 10^{-3} \text{kg/s}$$

Torque : $T = Wr = 150 \times 0.35 \times 52.5$ N-m

(i) Brake power : $(BP) = \frac{2\pi NT}{60} = \frac{P_{bmep}LAN_{cycle}k}{60}$

$$BP = \frac{2\pi NT}{60} = \frac{2\pi \times 3000 \times 52.5}{60}$$

$$= 5250\,\pi \simeq 16.4934 \text{ kW}$$ **Ans.**

(ii) $$P_{b\,mep} = \frac{(BP)60}{LAN_{cycle}\,k} = \frac{(16.4934)\times 60}{0.09 \times \frac{\pi}{4} \times 0.06^2 \times \frac{3000}{2} \times 4}$$

$$P_{b\,mep} = 6.48 \text{ bar}$$ **Ans.**

(iii) Brake thermal efficiency : $\eta_b = \frac{BP}{\dot{m}_f \times CV} = \frac{16.4934}{1.2222 \times 10^{-3} \times 45 \times 10^3}$

$$\eta_b \simeq 29.988\%$$ **Ans.**

(v) Brake specific fuel consumption :

$$\text{bsfc} = \frac{\dot{m}_f}{BP} = \frac{1.2222 \times 10^{-3}(kg/s)}{\left(\frac{16.4934\,kWh}{3600}\right)}$$

$$\text{bsfc} = 0.26677 \text{ kg/kWH}$$ **Ans.**

Let, BP_i denotes BP when engine i is cutoff

IP_i denotes IP when engine i is cutoff

IP(x) denotes IP of x^{th} cylinder

From Morse Test :

Engine Cutout	Corresponding Brake load	Corresponding $BP_i = \frac{2\pi N\,(W_i r)}{60}$	Corresponding IP [IP(I)= IP – IP_i = BP – BP_i]
1	104 N	11.4354 kW	IP(1) = 5.00580 kW
2	100 N	10.9956 kW	IP(2) = 5.4978 kW
3	102 N	11.2155 kW	IP(3) = 5.2779 kW
4	108 N	11.8752 kW	IP(4) = 4.6182 kW
			IP = 20.4519 kW

(vi) Mechanical efficiency, $\eta_m = \frac{BP}{IP} = 0.80644$

$$\eta_m = 80.6448\%$$ **Ans.**

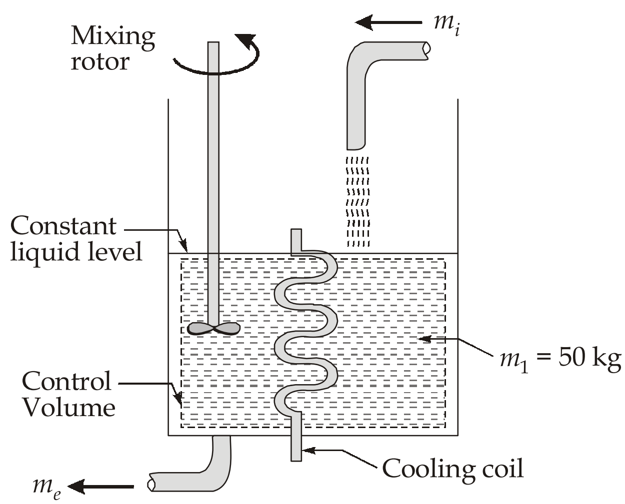
(vii)

$$P_{imep} = \frac{P_{bmep}}{\eta_m} \text{ or } \frac{\text{IP} \times 60}{\text{LAN}_{\text{cycle}}\, k}$$

$$P_{imep} = \frac{6.48}{0.806448} = 8.035 \text{ bar}$$ **Ans.**

4. (a) Solution:

Given : m_1 = 50 kg; T_1 = 50°C; $\dot{m}$ = 300 kg/h; $\dot{Q}$ = –7.5 kW; $\dot{W}$ = –0.6 kW



Assumptions:

1. Control Volume is defined by dashed line.
2. Water temperature is uniform with position throughout. $T = T(t)$
3. c_p value will remain constant,

m_i = mass entering control volume

m_e = mass exit control volume

For this control volume process, mass balance

$$\left(\frac{d_m}{dt}\right) = m_i - m_e$$

$$m_i = m_e = m$$

Energy balance, $$\frac{dU_{cv}}{dt} = \dot{m}h_i + \dot{Q}_{cv} - \dot{m}h_e - W_{cv} \quad ...(i)$$

Since, mass contained within control volume remains constant with time,

$$\frac{dU_{cv}}{dt} = \frac{d(m_{cv}u)}{dt} = m_{cv}\frac{du}{dt} = m_{cv}c_w\frac{dT}{dt}$$

For water, $h_i - h_e = c_w (T_i - T_e)$

Since water is well mixed, the temperature at the exit equals the temperature of overall quantity of liquid influx, so

$h_i - h_e = c_w \times (T_i - T)$ [‘T’ is general temperature at any time ‘t’]

From equation (i)

$$m_{cv} c_w \frac{dT}{dt} = \dot{m} c_w (T_i - T) + \dot{Q}_{cv} - \dot{W}_{cv}$$

$$\frac{dT}{dt} + \frac{\dot{m}}{m_{cv}} T = \frac{m}{m_{cv}} T_i + \frac{\dot{Q}_{cv} - \dot{W}_{cv}}{m_{cv} c_w}$$

$$\frac{dT}{dt} + \frac{300}{3600 \times 50} T = \frac{300}{3600 \times 50} \times 323 + \frac{-7.5 + 0.6}{50 \times 4.18}$$

$$\frac{dT}{dt} + \frac{T}{600} = 0.5053$$ **Ans.**

$$\text{I.F.} = e^{\int \frac{1}{600} dt} = e^{\frac{t}{600}}$$

$$T \times e^{\frac{t}{600}} = \int 0.5053 \times e^{\frac{t}{600}} dt + c$$

$$T \times e^{\frac{t}{600}} = \frac{0.5053 \times e^{\frac{t}{600}}}{\frac{1}{600}} + c$$

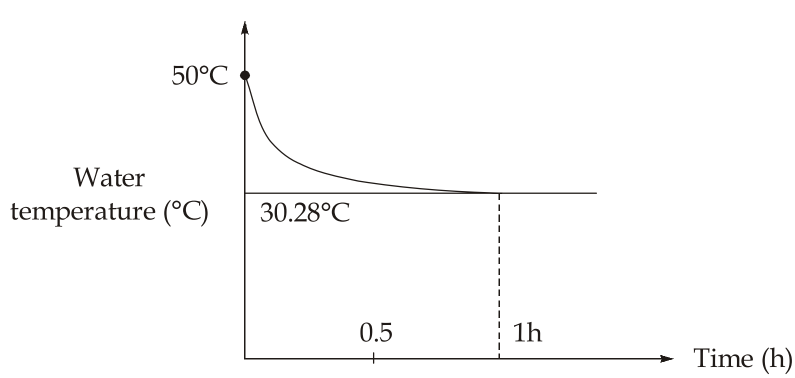
$$T = 303.19 + c e^{\frac{t}{600}}$$

At $t = 0$, $T = 323$ K

$$323 - 303.19 = c$$

$$c = 19.809$$

$\therefore$ $$T = 303.19 + 19.809 e^{\frac{t}{600}}$$ **Ans.**



at $t = 0$, $T = 50°C$

at $t = \infty$, $T = 30.286°C$

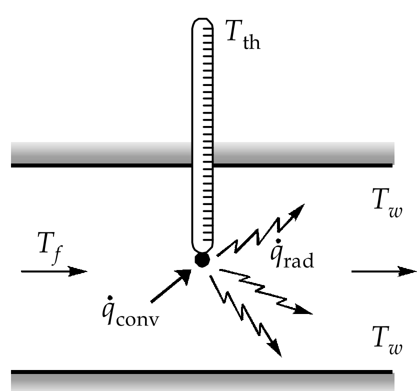
4. (b) (i) Solution:

A temperature measuring device indicates the temperature of its sensor, which is supposed to be, but is not necessarily, the temperature of the medium that the sensor is in. When a thermometer (or any other temperature measuring device such as a thermocouple) is placed in a medium, heat transfer takes place between the sensor of the thermometer and the medium by convection until the sensor reaches the temperature of the medium.

But when the sensor is surrounded by surfaces that are at a different temperature than the fluid, radiation exchange will take place between the sensor and the surrounding surfaces.

When the heat transfers by convection and radiation balance each other, the sensor will indicate a temperature that falls between the fluid and surface temperatures. Below we develop a procedure to account for the radiation effect and to determine the actual fluid temperature.

Consider a thermometer that is used to measure the temperature of a fluid flowing through a large channel whose walls are at a lower temperature than the fluid. Equilibrium will be established and the reading of the thermometer will stabilize when heat gain by convection, as measured by the sensor, equals heat loss by radiation (or vice versa). That is, on a unit area basis,



$$\dot{q}_{conv,\ to\ sensor} = \dot{q}_{rad,\ from\ sensor}$$

$$h(T_f - T_{th}) = \varepsilon_{th}\,\sigma\,(T_{th}^4 - T_w^4)$$

$$T_f = T_{th} + \frac{\varepsilon_{th}\sigma\left(T_{th}^4 - T_w^4\right)}{h} \quad \text{(K)}$$

where, T_f = actual temperature of the fluid, K

T_a = temperature value measured by the thermometer, K

T_w = temperature of the surrounding surfaces, K

h = convection heat transfer coefficient, W/m^2.K

ε = emissivity of the sensor of the thermometer

The last term in equation is due to the radiation effect and represents the radiation correction. Note that the radiation correction term is most significant when the convection heat transfer coefficient is small and the emissivity of the surface of the sensor is large. Therefore, the sensor should be coated with a material of high reflectivity (low emissivity) to reduce the radiation effect.

Placing the sensor in a radiation shield without interfering with the fluid flow also reduces the radiation effect. The sensors of temperature measurement devices used outdoors must be protected from direct sunlight since the radiation effect in that case is sure to reach unacceptable level.

4. (b) (ii) Solution:

Assumptions:

1. Steady-state conditions
2. Neglecting conduction effects through the thermocouple wires

As per given information

$$T_w = 500 \text{ K}$$
$$T_{th} = 850 \text{ K}$$
$$\varepsilon = 0.6$$
$$h = 60 \text{ W/m}^2 \text{ °C}$$
$$\sigma = 5.67 \times 10^{-8} \text{ W/m}^2 \text{ K}^4$$

The actual air temperature :

$$T_f = T_{th} + \frac{\varepsilon_{th}\sigma\left(T_{th}^4 - T_w^4\right)}{h}$$

$$T_f = 850 + \frac{0.6 \times 5.67 \times 10^{-8}\left(850^4 - 500^4\right)}{60}$$

$$T_f = 1110.54 \text{ K}$$

4. (c) Solution:

$$Q = 0.15 \text{ m}^3/\text{s}; \text{ Windage losses} = (0.5\, u^2/2g) \text{ m}$$
$$V = 70 \text{ m/s}; \theta = 180° - 150° = 30°$$

Relative velocity with which, the jet strikes the vane = $(V - u)$

Relative velocity at exit = $(V - u)$ (1 – friction losses)

$$= (V-u)\left(1-\frac{8}{100}\right) = 0.92\,(V-u)$$

Force of water per units mass in X-direction

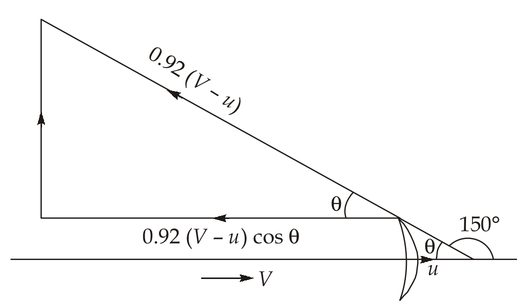
$$F_X = (V-u) - [-0.92(V-u)\cos\theta]$$
$$= (V-u)[1 + 0.92 \times \cos\theta)$$

(Work done/N) of water, $W = F_X.u$ – windage losses

$$= \frac{(V-u)}{g}\times(1+0.92\cos\theta)\times u - \frac{0.5}{2g}u^2$$

$$\eta = \frac{W}{\left[\frac{V^2}{2g}\right]} = \frac{1}{V^2}[2(V-u)(1+0.92\cos\theta)u - 0.5u^2]$$

$$= \frac{1}{V^2}[2(V.u-u^2)(1+0.92\cos\theta) - 0.5u^2] \quad \text{...(i)}$$



(i) Velocity of vanes corresponding to maximum efficiency.

Condition for maximum efficiency is that $\frac{d\eta}{du} = 0$,

Therefore from equation (i),

$$0 = [2(V - 2u)(1 + 0.92 \cos\theta) - 0.5 \times 2u]$$

On substituting the values,

$$0 = 2(70 - 2u)(1 + 0.92 \cos 30) - u$$

$\Rightarrow \quad 251.544 - 7.187u - u = 0$

$$\Rightarrow \qquad u = \frac{251.544}{8.187} = 30.725 \text{ m/s}$$

(ii) Value of maximum efficiency, η_{max} :

From equation (i) at u = 30.725 m/s, we get :

$$\eta_{max} = \frac{1}{V^2}[2(V-u)(1+0.92\cos\theta)u - 0.5u^2]$$

$$= \frac{1}{(70)^2}[2(70-30.725)(1+0.92\cos 30)\times 30.725 - 0.5\times 30.725^2]$$

$$= 0.7886 \text{ or } 78.86\%$$

(iii)Force on vanes at right angles to direction of motion.

$$\dot{m} = \rho.Q = 1000 \times 0.15 = 150 \text{ kg/s}$$

$$F_X = \dot{m}\,[(V-u)(1+0.92\cos\theta)]$$

$$= 150[(70 - 30.725)(1 + 0.92 \times \cos 30)] = 10585.06 \text{ N}$$

$$F_Y = m[(V-u)(0+0.92\times\sin\theta)]$$

$$= 150(70 - 30.725)(0.92 \times \sin 30°)] = 2709.98 \text{ m}$$

(iv) Power developed, P :

$$P = F_X.u \times \eta_{max}$$

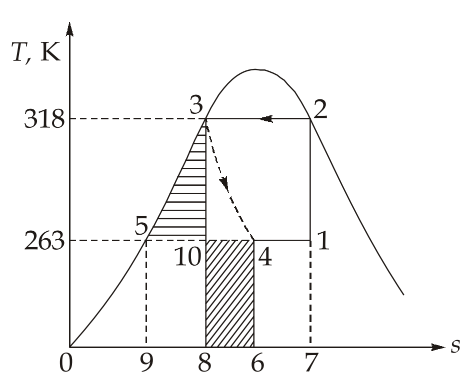
$$= 10585.06 \times 30.725 \times 0.7886 \text{ Nm/s or W}$$

$$= 256473 \text{ W} = 256.473 \text{ kW}$$

Section : B

5. (a) Solution:

Given : Evaporator coil temperature (T_E) = –10°C; Condenser coil temperature (T_C) = 45°C; Mean specific heat (C_m) = 0.983 kJ/kgK



From R-12 refrigeration table:

$$h_2 = 204.9 \text{ kJ/kg}$$

$$h_3 = h_4 = 79.7 \text{ kJ/kg}$$
$$s_2 = s_1 = 0.6812 \text{ kJ/kg}$$
$$s_3 = s_{10} = 0.2877 \text{ kJ/kgK}$$

From figure for T-s representation

$$s_3 - s_5 = s_{10} - s_5 = c_m \ln \frac{T_C}{T_E}$$
$$= 0.983 \ln \frac{318}{263}$$

or, $$s_3 - s_5 = s_{10} - s_5 = 0.1867 \text{ kJ/kgK}$$

$$\text{Area } 3 - 5 - 10 = c_m(T_C - T_E) - (s_{10} - s_5)T_E$$
$$= 0.983(45 - (-10)) - 0.1867 \times 263$$

or, $$\text{Area } 3 - 5 - 10 = 54.065 - 49.1021 = 4.9629 \text{ kJ/kg}$$

Also, $$s_4 - s_{10} = \frac{\text{Area } 4 - 6 - 8 - 10}{T_E} = \frac{\text{Area } 3 - 5 - 10}{T_E}$$

$$s_4 - s_{10} = \frac{4.9629}{263} = 0.01887 \text{ kJ/kgK}$$

and $$s_2 - s_3 = \frac{h_2 - h_3}{T_C} = \frac{204.9 - 79.7}{318} = 0.3937 \text{ kJ/kgK}$$

$$\text{Work done} = \text{Area } 1 - 2 - 3 - 5 - 1$$
$$= \text{Area } 3 - 5 - 10 + \text{Area } 1 - 2 - 3 - 10$$
$$= 4.9629 + (s_2 - s_3)(T_C - T_E)$$
$$\text{Work done} = 4.9629 + 0.3937\,(55)$$
$$= 26.6164 \text{ kJ/kg}$$
$$\text{Refrigeration effect} = \text{Area } 4 - 1 - 7 - 6$$
$$= \left[(s_2 - s_3) - (s_4 - s_{10})\right] \times T_E$$
$$= [0.3937 - 0.01887] \times 263$$
$$\text{R.E.} = 98.580 \text{ kJ/kg}$$

Hence, $$\text{COP} = \frac{R.E}{\text{Work input}} = \frac{98.580}{26.6164} = 3.7036$$ **Ans.**

Alternatively,

$$h_3 = h_4 = (h_f)_{\text{at } 45^\circ\text{C}} = 79.7 \text{ kJ/kg}$$

$$h_2 = (h_g)_{\text{at } 45^\circ\text{C}} = 204.9 \text{ kJ/kg}$$

$$s_2 = (s_g)_{at\ 45^\circ C} = 0.6812 \text{ kJ/kg.K}$$

Also, $s_2 = s_1 \Rightarrow 0.6812 = 0.1080 + x_1[0.7020 - 0.1080]$

$$x_1 = 0.9649$$

$$h_1 = 26.9 + 0.9649[183.2 - 26.9]$$

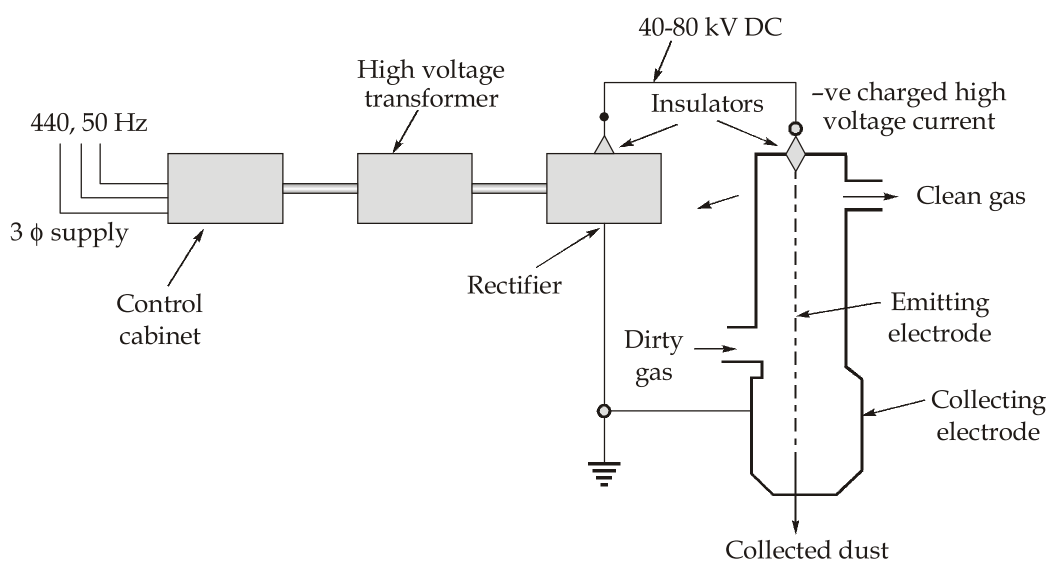
$$= 177.714 \text{ kJ/kg}$$

$$\text{C.O.P of plant} = \frac{\text{Desired effect}}{\text{Work input}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{177.714 - 79.7}{204.9 - 177.714}$$

$$= 3.605$$ **Ans.**

5. (b) Solution:

The principal components of an electrostatic precipitator (ESP) are two sets of electrodes insulated from each other. The first set is composed of rows of electrically grounded vertical parallel plates, called the collection electrodes, between which the dust-laden gas flows. The second set of electrodes consists of wires, called the discharge or emitting electrodes that are centrally located between each pair of parallel plates. The wires carry a unidirectional negatively charged high-voltage current from an external DC source. The applied high voltage generates a unidirectional, non-uniform electrical field. When that voltage is high enough, a blue luminous glow called a corona, is produced around them. Electrical forces in the corona accelerate the free electrons present in the gas so that they ionize the gas molecules, thus forming more electrons and positive gas ions.



The positive ions travel to the negatively charged wire electrodes. The electrons follows the electrical field toward the grounded electrodes but their velocity decreases toward the plates. Gas molecules capture the low velocity electrons and become negative ions.

As these ions move to the collecting electrode, they collide with the fly ash particles in the gas stream and give them negative charge. The negatively charged fly ash particles are driven to the collecting plate.

Collected particulate matter must be removed from the collecting plates on a regular schedule to ensure efficient collector operation. Removal is usually accomplished by a mechanical hammer scrapping system.

An electrostatic precipitator like a cyclone separator, has an overall collection efficiency, η_o is defined by

$$\eta_o = \frac{\text{mass of all particles retained by collector}}{\text{mass of all particles entering collector}}$$

$$= 1 - \exp\left(-\frac{AV_{mo}}{Q}\right)$$

where, A = Area of collector plate (m^2);

V_{mo} = effective migration velocity of particles (m/sec)

Q = Flue gas volume flow rate for each plate (m^3/sec)

Hence,

With increase in collector area, collection efficiency increases.

With increase in migration velocity, collection efficiency increases.

With decrease in mass flow rate, collection efficiency increases.

5. (c) Solution:

The specific speed of centrifugal pump may be defined as the speed in revolutions per minute of a geometrically similar pump of such a size that under corresponding conditions it would deliver 1 litre of liquid per second against a head of 1 metre. It is represented by N_S.

$$\text{Discharge, } Q = (k\pi B_1 D_1)V_{f_1}$$

$$Q \propto B_1 D_1 V_{f_1}$$

$$Q \propto D_1^2 V_{f_1}\text{; Since } B_1 \propto D_1$$

Also $$V_{f_1} = \psi\left(\sqrt{2gH_m}\right)\text{; or } V_{f_1} \propto \left(\sqrt{H_m}\right)$$

Thus, $$Q \propto D_1^2\left(\sqrt{H_m}\right)$$

$$\frac{Q}{D_1^2\left(\sqrt{H_m}\right)} = \text{Constant} \quad \text{...(i)}$$

Further, $$U_1 = \frac{\pi D_1 N}{60}$$

$$D_1 \propto \left(\frac{U_1}{N}\right)$$

$$U_1 = K_u\left(\sqrt{2gH_m}\right),\ U_1 \propto \left(\sqrt{H_m}\right)$$

Thus, $$D_1 \propto \left(\frac{\sqrt{H_m}}{N}\right)$$

$$\frac{\sqrt{H_m}}{D_1 N} = \text{Constant} \quad \text{...(ii)}$$

Substituting the value of D_1 from equation (ii) in equation (i),

$$Q \propto \left(\frac{H_m^{3/2}}{N^2}\right)$$

or $$N^2 \propto \left(\frac{H_m^{3/2}}{Q}\right)$$

or $$N \propto \left(\frac{H_m^{3/4}}{\sqrt{Q}}\right)$$

or $$N = \left(\frac{H_m^{3/4}}{\sqrt{Q}}\right);\ \text{where } C \text{ is a constant}$$

or $$\frac{N\sqrt{Q}}{H_m^{3/4}} = C$$

Now, according to the definition of the specific speed put Q = 1 litre/second and H_m = 1 m, then $C = N = N_S$.

$$N_S = \frac{N\sqrt{Q}}{H_m^{3/4}}$$

5. **(d) Solution:**

The photovoltaic effect is the creation of voltage and electric current in a material upon exposure to light and is a physical and chemical phenomenon

When a solar cell is under solar radiation, then electron hole pairs are generated and electric current I is obtained in the circuit.
Electric current I is the difference between the solar light generated current (I_L) and diode junction current (I_J)

$$I = I_L - I_J$$

$$I_J = I_0\left[\exp\left(\frac{eV}{kT}\right) - 1\right]$$

Where I_0 = Dark or saturation current.

$$I = I_{sc} - I_J$$

I_{sc} is current at short circuit and at open circuit $I = 0$

$$\therefore \quad 0 = I_{sc} - I_J$$

$$0 = I_{sc} - I_0\left[\exp\left(\frac{eV}{kT}\right) - 1\right]$$

V_{oc} is open circuit voltage

$$V_{oc} = \frac{kT}{e}\ln\left(\frac{I_{sc} + I_0}{I_0}\right)$$

Power can be defined as,

$$P = VI = V \times \left\{I_{sc} - I_0 \exp\left(\frac{eV}{KT}\right) + I_0\right\}$$

Now, in above expression power is function of V.

$$P = f(V)$$

For maximization of power

$$\frac{dP}{dV} = 0 \text{ and } \frac{d^2P}{dV^2} < 0$$

$$= I_{sc} - I_0 \exp\left(\frac{eV}{KT}\right) + I_0 - VI_0 \exp\left(\frac{eV}{kT}\right).\frac{e}{kT} = 0$$

$$\frac{I_{sc} + I_0}{I_0} = \exp\left(\frac{eV_m}{kT}\right)\left\{1 + \frac{eV_m}{kT}\right\}$$

In above expression V_m can be calculated and hence

$$I_m = I_{sc} - I_0\left\{\exp\left(\frac{eV_m}{kT} - 1\right)\right\}$$

$$P_{max} = V_m I_m$$

5. (e) Solution:

Given : V_1 = 900 m/s; u = 410 m/s; α_1 = 18°; $\dot{m}$ = 0.75 kg/s; k_b = 0.8

$$\beta_1 = \tan^{-1}\left(\frac{V_f}{V_{w_1} - u}\right)$$

$$\beta_1 = \beta_2 = 31.95°$$

$$V_{f_1} = V_1 \sin\alpha = 278.115 \text{ m/s}$$

$$V_{w_2} = V_1 \cos\alpha = 855.95 \text{ m/s}$$

$$V_{r_1} = \sqrt{V_{f_1} + \left(V_{w_1} - u\right)^2} = 525.56 \text{ m/s}$$

$$V_{r_2} = 0.8V_{r_1} = 0.8 \times 525.56 = 420.448 \text{ m/s}$$

$$V_{w_2} = V_{r_2} \cos\beta_2 - u$$

$$= 420.448 \cos 31.95 - 410 = 53.24 \text{ m/s}$$

$$\Delta V_w = V_{w_1} + V_{w_2}$$

$$= 855.95 + (-53.24) = 802.71 \text{ m/s}$$

Tangential, $F_t = \dot{m}\Delta V_w$

$= 0.75 \times 802.71 = 602.03$ N **Ans.**

Diagram power, $\dot{W}_D = 602.03 \times 410 \times 10^{-3}$ kW = 246.833 kW

Diagram efficiency, $\eta_D = \dfrac{W_D}{\frac{1}{2}\dot{m}V_1^2 \times 10^{-3}} = \dfrac{246.833 \times 2}{0.75 \times 900^2 \times 10^{-3}}$

$= 0.8126$ or 81.26% **Ans.**

Axial thrust, $F_a = \dot{m}\Delta V_a$

$$F_a = 0.75\left[V_{r_1} \sin\beta_1 - V_{r_2} \sin\beta_2\right]$$

$$F_a = 0.75[525.56 \times \sin 31.95 - 420.448 \times \sin 31.95]$$

$F_a = 41.72$ N **Ans.**

6. (a) Solution:

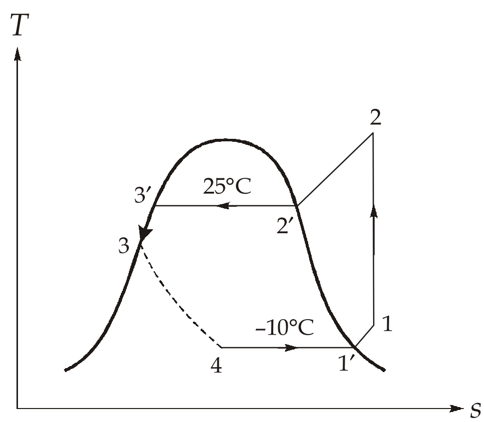
Given data : Refrigerating capacity, RC = 15TR

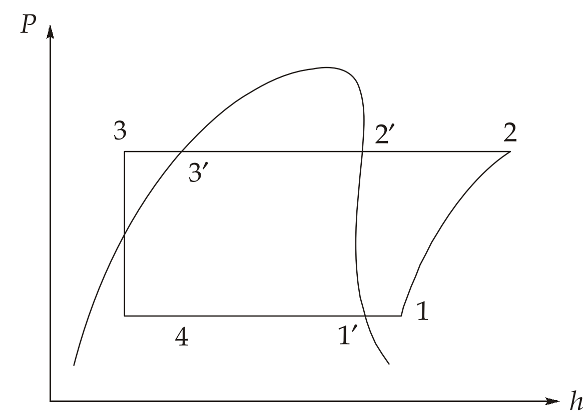
$$T_{1'} = -10°C = -10 + 273 = 263 \text{ K}$$

$$T_{2'} = 25°C = 25 + 273 = 298 \text{ K}$$

$$T_1 = -3°C = -3 + 273 = 270 \text{ K}$$

$$T_{3'} - T_3 = 6°C$$





From R-12 table,

$$h_{1'} = 183.2 \text{ kJ/kg}; \quad h_{3'} = 59.7 \text{ kJ/kg}; \quad h_{2'} = 197.7 \text{ kJ/kg};$$

$$s_{1'} = 0.7020 \text{ kJ/kgK}; \quad s_{2'} = 0.6869 \text{ kJ/kgK}$$

$$v_{1'} = 0.0767 \text{ m}^3\text{/kg}; \quad v_{2'} = 0.0269 \text{ m}^3\text{/kg}$$

Entropy at point 1, $s_1 = s_{1'} + c_{p_v} \ln\left(\frac{T_1}{T_{1'}}\right)$

$$= 0.7020 + 0.733 \ln\left(\frac{270}{263}\right) = 0.7212 \text{ kJ/kgK}$$

Entropy at point 2, $s_2 = s_{2'} + c_{p_v} \ln\left[\frac{T_2}{T_{2'}}\right]$

$$s_2 = 0.6869 + 0.733 \ln\left[\frac{T_2}{298}\right]$$

Compression 1 → 2 is isentropic

So, $s_1 = s_2$

$$0.7212 = 0.6869 + 0.733 \ln\left[\frac{T_2}{298}\right]$$

$$T_2 = 312.27 \text{ K}$$

Enthalpy at point 1,

$$h_1 = h_{1'} + c_{p_v}(T_1 - T_{1'})$$

$$h_1 = 183.2 + 0.733 \times (270 - 263)$$

$$h_1 = 188.33 \text{ kJ/kg}$$

Enthalpy at point 2,

$$h_2 = h_{2'} + c_{p_v}(T_2 - T_{2'})$$

$$h_2 = 197.7 + 0.733 \times (312.27 - 298)$$

$$h_2 = 208.16 \text{ kJ/kg}$$

Enthalpy at point 3,

$$h_3 = h_{3'} - c_{p_l}(T_{3'} - T_3)$$

$$h_3 = 59.7 - 1.235 \times 6$$

$$h_3 = 52.29 \text{ kJ/kg}$$

(i) $$\text{COP} = \frac{h_1 - h_3}{h_2 - h_1} = \frac{188.33 - 52.29}{208.16 - 188.33} = 6.86$$ **Ans.**

(ii) Theoretical power per tonne of refrigeration

$$\text{RE} = h_1 - h_3$$

$$= 188.33 - 52.29 = 136.04 \text{ kJ/kg}$$

∵ Mass flow rate of refrigerant,

$$\dot{m}_r = \frac{RC}{RE} = \frac{15 \times 3.5}{136.04} = 0.3859 \text{ kg/sec}$$

Power during compression, $P = \dot{m}_r(h_2 - h_1)$

$$= 0.3859 \times [208.16 - 188.33]$$

$$= 7.652 \text{ kW}$$

Theoretical power per tonne of refrigeration = $\dfrac{Power}{RC}$

$$= \frac{7.652}{15} = 0.5101 \text{ kW/tonne}$$ **Ans.**

(iii)

Let, D = Bore of compressor, L = Stroke of compressor = 1.5D and N = Speed of compressor = 900 rpm

between 1 and 1′

$$\frac{v_1}{T_1} = \frac{v_{1'}}{T_{1'}}$$

$$v_1 = 0.0767 \times \left[\frac{270}{263}\right] = 0.07874 \text{ m}^3/\text{kg}$$

1. When there is no clearance

$$\frac{\pi}{4} \times D^2 \times L \times \frac{N}{60} \times K = \dot{m}_r \times v_1$$

$$\frac{\pi}{4} \times D^2 \times 1.5D \times \frac{900}{60} \times 2 = 0.3859 \times 0.07874$$

$$D = 0.09508 \text{ m}$$

$$D = 95.08 \text{ mm}$$

$$L = 1.5D = 1.5 \times 95.08 = 142.63 \text{ mm}$$ **Ans.**

2. When there is a clearance of 2% between 2 and 2′

$$\frac{v_2}{T_2} = \frac{v_{2'}}{T_{2'}}$$

$$\Rightarrow \quad v_2 = v_{2'} \times \frac{T_2}{T_{2'}}$$

$$v_2 = 0.0269 \times \frac{312.27}{298} = 0.02818 \text{ m}^3/\text{kg}$$

$$\eta_v = 1 + c - c\left[\frac{v_1}{v_2}\right]$$

$$= 1 + 0.02 - 0.02\left[\frac{0.07874}{0.02818}\right] = 0.9641$$

Also,

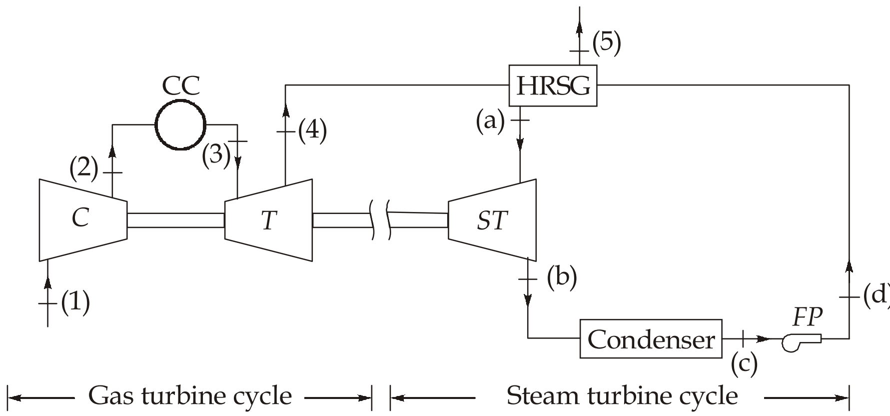
$$\eta_v = \frac{\dot{m}v_1}{\frac{\pi}{4}D^2 L \frac{N}{60} \times k}$$

$$0.9641 = \frac{0.3859 \times 0.7874}{\frac{\pi}{4}D^3 \times 1.5 \times \frac{900}{60} \times 2}$$

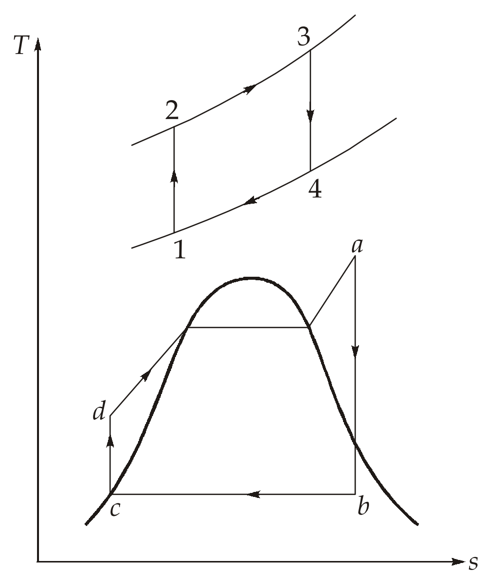
$$D = 0.09625 \text{ m}$$

$$D = 96.25 \text{ mm}$$
$$L = 1.5D = 1.5 \times 96.25$$
$$= 144.37 \text{ mm}$$ **Ans.**

6. (b) Solution:



Given: $T_1 = 17°C = 290$ K, $P_1 = 1$ bar, $T_3 = 1400$ K, $T_a = 400°C = 673$ K, $P_a = 6$ MPa, $r_p = 10$, $P_b = 15$ kPa, $P_{output} = 37.3$ MW, $C_{P\ air} = 1.0032$ kJ/kgK, $T_5 = 420$ K



In gas turbine cycle,
$$\frac{T_2}{T_1} = (r_p)^{\frac{\gamma-1}{\gamma}} = (10)^{\frac{1.4-1}{1.4}}$$
$$T_2 = 290 \times 1.93 = 559.9 \text{ K}$$
$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\Rightarrow \qquad T_4 = 1400\left(\frac{1}{10}\right)^{\frac{1.4-1}{1.4}} = 725.13 \text{ K}$$

$$\text{Compressor work, } W_C = C_P(T_2 - T_1) = 1.0032(559.9 - 290)$$
$$= 270.76 \text{ kJ/kg}$$

$$\text{Turbine work, } W_T = C_P(T_3 - T_4) = 1.0032(1400 - 725.13)$$
$$= 677.03 \text{ kJ/kg}$$

and heat added in combustion chamber,

$$q_s = C_P(T_3 - T_2) = 1.0032(1400 - 559.9)$$
$$= 842.79 \text{ kJ/kg}$$

$$\text{Net work per kg, } (W_{net})_{GT} = W_T - W_C = 677.03 - 270.76$$
$$= 406.27 \text{ kJ/kg}$$

Heat recovered in HRSG,

$$q_{HRSG} = C_P(T_4 - T_5) = 1.0032(725.13 - 420)$$
$$= 306.11 \text{ kJ/kg}$$

In steam turbine cycle, $h_a = 3177.2$ kJ/kg

$$s_a = 6.5408 \text{ kJ/kg}$$

For expansion in steam turbine,

$$s_a = s_b$$
$$6.5408 = s_{f\text{ at15 kPa}} + xs_{fg\text{ at 15 kPa}}$$
$$6.5408 = 0.73664 + x(8.0311 - 0.73664)$$

$$\Rightarrow \qquad x = 0.7957$$

and $h_b = h_{f\text{ at15 kPa}} + xh_{fg\text{ at 15 kPa}}$

$$= 219.98 + 0.7957(2598.8 - 219.98) = 2110.40 \text{ kJ/kg}$$

$$\text{Feed pump work, } W_P = v_c\,(P_d - P_c)$$
$$W_P = v_{f\text{ at 15 kPa}}(6000 - 15)$$
$$= 0.001013\,(6000 - 15) = 6.062 \text{ kJ/kg}$$

and $h_c = h_{f\text{ at 15 kPa}} = 219.98$ kJ/kg

and $h_d = h_c + W_P = 219.98 + 6.062 = 226.04$ kJ/kg

So, $\text{Mass of steam generated} = \dfrac{q_{HRSG}}{h_a - h_d} = \dfrac{306.11}{3177.2 - 226.04}$

$$= 0.1037 \text{ kg steam per kg of air}$$

Now, Net steam turbine output,

$$(W_{net})_{ST} = (h_a - h_b) - (h_d - h_c)$$
$$= (3177.2 - 2110.40) - (226.54 - 219.98)$$
$$= 1060.738 \text{ kJ/kg}$$

So, Steam cycle output per kg of air = 1060.738 × 0.1037

$$= 110 \text{ kJ/kg of air}$$

Now, Total combined cycle output = $(W_{net})_{GT} + (W_{net})_{ST}$

$$W_{CC} = (406.27) + (110)$$
$$= 516.27 \text{ kJ/kg of air}$$

Combined cycle efficiency, $\eta_{CC} = \dfrac{W_{CC}}{q_s} = \dfrac{516.27}{842.79} = 61.25\%$

6. (c) Solution:

Rate of air consumption, $\dot{m}_a = 50$ kg/s

$\Delta h = 300$ kJ/kg

Velocity coefficient, $z = 0.96$

Air fuel rate = 70 : 1

$\eta_{combustion} = 95\%$

Calorific value = 42000 kJ/kg

Aircraft velocity, $C_a = \dfrac{1800 \times 1000}{3600} = 500$ m/s

(i) Exit velocity of jet $C_j = z\sqrt{2 \times \Delta h \times 1000}$

$$= 0.96\sqrt{2 \times 300 \times 1000} = 743.61 \text{ m/s}$$

(ii) Fuel flow rate:

Rate of fuel consumption, $\dot{m}_f = \dfrac{\text{Rate of air consumption}}{\text{Air-fuel ratio}} = \dfrac{50}{70}$

$$\dot{m}_f = 0.71428 \text{ kg/s}$$

(iii) Thrust is force produced due to change of momentum.

$$\text{Thrust produced} = \dot{m}_a\left(C_j - C_a\right)$$
$$= 50(743.61 - 500) = 12180.5 \text{ N}$$

Thrust specific fuel consumption

$$= \frac{\text{Fuel consumption}}{\text{Thrust}}$$

$$= \frac{0.71428}{12180.5} = 5.864 \times 10^{-5} \text{ kg/N of thrust/s}$$

(iv) Thermal efficiency, η_{thermal}

$$\eta_{th} = \frac{\text{Work output}}{\text{Heat supplied}}$$

$$= \frac{\text{Gain in kinetic energy per kg of air}}{\text{Heat supplied by fuel per kg of air}}$$

$$= \frac{\left(C_j^2 - C_a^2\right)}{2\left(\frac{m_f}{m_a}\right) \times C.V \times \eta_{\text{combustion}} \times 1000}$$

$$= \frac{\left(743.61^2 - 500^2\right)}{2\left(\frac{1}{70}\right) \times 42000 \times 1000 \times 0.95} = 26.57\%$$

(v) Propulsive power

$$\text{Propulsive power} = \dot{m}_a \times \frac{\left(C_j^2 - C_a^2\right)}{2}$$

$$= \frac{50}{1000} \times \frac{\left(743.61^2 - 500^2\right)}{2} \text{kW}$$

$$= 7573.89 \text{ kW}$$

(vi) Propulsive efficiency pump,

$$\eta_{\text{prop}} = \frac{\text{Thrust power}}{\text{Propulsive power}} = \frac{2C_a}{C_j + C_a}$$

$$= \frac{2 \times 500}{743.61 + 500} = 0.8041 = 80.411\%$$

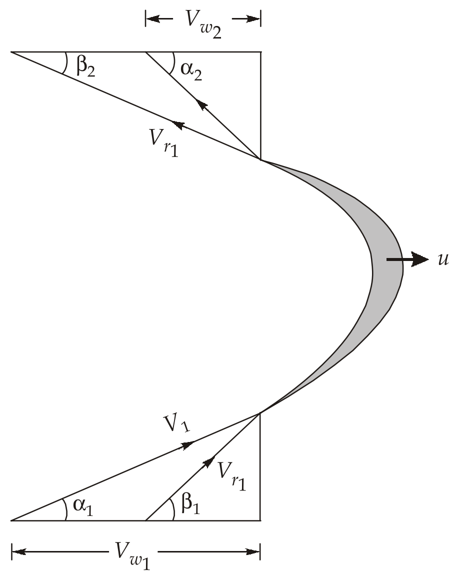
(vii) Overall efficiency,

$$\eta_0 = \frac{\left(C_j - C_a\right)C_a}{\left(\frac{m_f}{m_a}\right) \times CV \times \eta_{combustion}}$$

$$= \frac{(743.61 - 500) \times 500}{\left(\frac{1}{70}\right) \times 42000 \times 0.95 \times 1000} = 0.2137 = 21.37\%$$

7. **(a) Solution:**

Given : $\alpha_1 = 20°$; $\eta_n = 0.88$; $\beta_2 = 29°$; $u = 140$ m/s; $v_1 = 350$ m/s; RHF = 1.05



(i)

$$\tan\beta_1 = \frac{V_1 \sin\alpha_1}{V_1 \cos\alpha_1 - u}$$

$$= \frac{350 \times \sin 20}{350\cos 20 - 140} = 0.63373$$

$$\beta_1 = 32.363°$$

$$V_{r_1} \sin\beta_1 = V_1 \sin\alpha_1$$

$$V_{r_1} = \frac{350 \times \sin 20}{\sin 32.363} = 223.629 \text{ m/s}$$

$$V_{r_2} = 0.8 \times V_{r_1}$$

$$= 0.8 \times 223.629 = 178.903 \text{ m/s}$$

$$V_{w_1} = V_1 \cos\alpha_1 = 350 \times \cos 20 = 328.89 \text{ m/s}$$

$$V_{w_2} = V_{r_2} \cos\beta_2 - u$$

$$= 178.903 \times \cos 29 - 140 = 16.472 \text{ m/s}$$

Work done per kg, $W_D = \Delta V_w u = \left(V_{w_1} + V_{w_2}\right)u$

$$= (328.89 + 16.472) \times 140 \times 10^{-3} \text{ kJ/kg} = 48.35 \text{ kJ/kg}$$

$$\text{Blade efficiency, } \eta_b = \frac{\text{Work done per kg}}{\frac{1}{2}V_1^2} = \frac{2\Delta V_w u}{V_1^2}$$

$$\eta_b = \frac{48.35 \times 2 \times 10^3}{350^2} = 0.7893$$

$$\eta_b = 78.93\%$$

$$\text{Stage efficiency, } \eta_s = \eta_n \times \eta_b$$

$$= 0.88 \times 0.7893$$

$$\eta_s = 0.6946$$

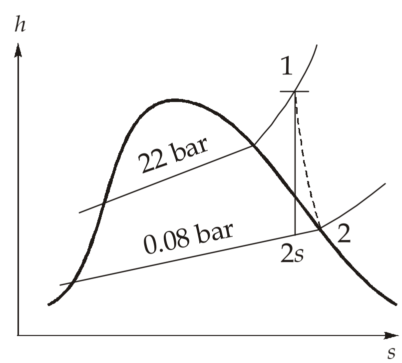
$$= 69.465\%$$

(ii)

$$\eta_{internal} = RHF \times \eta_{stage}$$

$$= 1.05 \times 0.6946$$

$$= 0.7294 \text{ or } 72.94\%$$



From steam table,

$$h_1 = 2894.5 \text{ kJ/kg}$$

$$s_1 = 6.4903 \text{ kJ/kgK}$$

At 0.08 bar,

$$h_f = 173.84 \text{ kJ/kg}, \; s_f = 0.59249 \text{ kJ/kgK}$$

$$h_{fg} = 2402.4 \text{ kJ/kg}, \; s_{fg} = 7.6348 \text{ kJ/kgK}$$

$$s_1 = s_{2s} = s_f + x_{2s} s_{fg}$$

$$6.4903 = 0.59249 + x_{2s}\, 7.6348$$

$$x_{2s} = 0.7724$$

$$h_{2s} = 173.84 + 0.7724 \times 2402.4$$

$$= 2029.67 \text{ kJ/kg}$$

$$h_1 - h_2 = \eta_{internal} \times (h_1 - h_{2s})$$
$$= 0.7294 \times (2894.5 - 2029.67)$$
$$= 630.806 \text{ kJ/kg}$$

$$\text{Number of stages} = \frac{(\Delta h)_{total}}{(\Delta h)_{stages}} = \frac{630.806}{48.35}$$

$$n = 13.046$$
$$= 14 \text{ stages}$$ **Ans.**

7. (b) Solution:

Given : $\beta = 40°$, $\phi = 30.73°$, $n = 142$ (22nd may), $\rho = 0.4$, $\overline{H}_g = 15275.8 \text{ kJ/m}^2$,

$\overline{H}_d = 3256.7 \text{ kJ/m}^2$

For 22nd may, $n = 142$

$$\delta = 23.45 \times \sin\left(\frac{360}{365}(142 + 284)\right) \text{ degrees}$$

$$\delta = 20.34° \text{ or } 0.355 \text{ rad}$$

Hour angle at sunrises (or sunset) for horizontal surface

$$\omega_{sh} = \pm\cos^{-1}(-\tan\phi\tan\delta)$$
$$\omega_{sh} = 102.73° \text{ or } 1.792 \text{ radians}$$

Hour angle at sunrise (or sunset) at tilted surface

$$\omega_{st} = \pm\min\left[\left|\cos^{-1}(-\tan\phi\tan\delta)\right|, \left|\cos^{-1}(-\tan(\phi-\beta)\tan\delta)\right|\right]$$

$$\omega_{st} = \pm\min\left[\left|\cos^{-1}(-\tan(30.73)\tan(20.34)\right|, \left|\cos^{-1}(-\tan(30.73-40)\tan(20.34)\right|\right]$$

$$\omega_{st} = \pm\min[102.73°, 86.53°]$$

$\therefore$ $$\omega_{st} = 86.53° \text{ or } 1.509 \text{ radians}$$

Now, tilt factor

$$\overline{R}_b = \frac{\omega_{st}\sin\delta\sin(\phi-\beta) + \cos\delta\cos(\phi-\beta)\sin\omega_{st}}{\omega_{sh}\sin\delta\sin\phi + \cos\delta\cos\phi\sin\omega_{sh}}$$

Putting the valves, we get

$$\overline{R}_b = \frac{1.509\sin(20.34°)\sin(30.73° - 40°) + \cos(20.34°)\cos(30.73° - 40°) \times \sin(86.53°)}{1.792\sin(20.34°)\sin(30.73°) + \cos(20.34°)\cos(30.73°) \times \sin(102.73°)}$$

$\therefore \quad \bar{R}_b = 0.759$

For diffuse radiation,

$$\bar{R}_d = \frac{1+\cos 40°}{2} = 0.883$$

For reflected radiation,

$$\bar{R}_r = \frac{\rho(1-\cos\beta)}{2} = \frac{0.4(1-\cos 40)}{2} = 0.0467$$

Now, monthly average total daily radiation on tilted surface,

$$\frac{\bar{H}_T}{\bar{H}_g} = \left(1-\frac{\bar{H}_d}{\bar{H}_g}\right)\times\bar{R}_b + \left(\frac{\bar{H}_d}{\bar{H}_g}\right)\times\bar{R}_d + \bar{R}_r$$

$$\bar{H}_T = 15275.8\left[\left(1-\frac{3256.7}{15275.8}\right)\times 0.759 + \frac{3256.7\times 0.883}{15275.8} + 0.0467\right]$$

$$\bar{H}_T = 12711.54 \text{ kJ/m}^2\text{-day}$$ **Ans.**

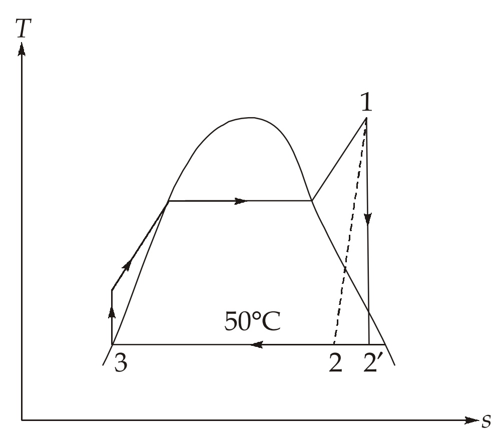
7. (c) Solution:

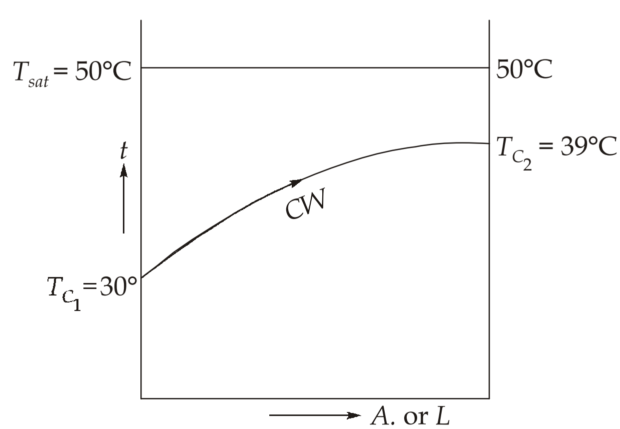
Given : Steam flow rate from turbine exit = 40 t/h

$$\text{Air leakage, } \dot{m}_a = \frac{0.4 \text{ kg}}{1000 \text{ kg of steam}}$$

Water inlet temperature $(T_{c_1}) = 30°C$

Water exit temperature $(T_{c_2}) = 39°C$





(i) From steam table,

At 50°C, $P_{sat} = 0.12352$ bar, $v_g = 12.027 \text{ m}^3/\text{kg}$

Using perfect gas law,

$$P_a V_a = \dot{m}_a R_a T_a$$

Since air will occupy the condensor volume thus,

$$\dot{V}_a = \dot{V}_s = \dot{m}_s v_s = \dot{m}_s v_g$$

$\therefore$ $$P_a \dot{m}_s v_g = \dot{m}_a R_a T_a$$

or, $$P_a = \frac{\dot{m}_a R_a T_a}{\dot{m}_s v_g} = \frac{0.4 \times 0.287 \times (50+273) \times 10^3}{1000 \times 12.027}$$

$$= 3.0383 \times 10^{-5} \text{ bar (negligible)}$$

This shows that at entry, P_a is negligible and total pressure in the condensor is 0.12352 bar.

The condensate leaves at 39°C

So, from steam table

At 39°C, P_{sat} = 0.07 bar, v_g = 20.524 m^3/kg

The total pressure in the condensor

$$P = P_{sat} + P_{air}$$

$$P_{air} = P - P_{sat} = 0.12352 - 0.07 = 0.05352 \text{ bar}$$

The mass flow rate of air removed at 39°C,

$$\dot{m}_a = \dot{m}_s \times 0.4 \times 10^{-3} = 40 \times 10^3 \times 0.4 \times 10^{-3}$$

$$= 16 \text{ kg/h}$$

Volume of air removed per hour at 39°C

$$V_a = \frac{\dot{m}_a R_a T_a}{P_a} = \frac{16 \times 287 \times 312}{0.05352 \times 10^5} = 267.7 \text{ m}^3/\text{h}$$

The mass flow rate of steam accompanying this air at 39°C is

$$\dot{m}_s = \frac{\dot{V}}{v_g} = \frac{267.7}{20.524} = 13.0432 \text{ kg/h}$$

Due to installation of separate air extraction in cooler side, air along with some steam leaves the air cooler at 26°C.

At 26°C from steam table

$$P_{sat} = 0.033639 \text{ bar}$$

$$v_g = 40.973 \text{ m}^3/\text{kg}$$

$\therefore$ $$P_{air} = 0.12352 - 0.033639 = 0.089881 \text{ bar}$$

The volume of air removed at 26°C from condensor air cooler section

$$\dot{V}_a = \frac{\dot{m}_a R_a T}{P_{air}} = \frac{16 \times 287 \times 299}{0.089881 \times 10^5}$$

$$\dot{V}_a = 152.758 \text{ m}^3/\text{h} = \dot{V}$$

Mass of steam accompanying this air

$$\dot{V}_a = \dot{V}_s$$

$$\dot{m}_s = \frac{\dot{V}}{v_g} = \frac{152.758}{40.973} = 3.7282 \text{ kg/h}$$

Hence, The saving in condensate by using separate extraction pump in cooler steam of the condensor.

$$(\dot{m}_c)_{saved} = 13.0432 - 3.7282 = 9.3149 \text{ kg/h}$$ Ans.

Saving in heat supply in the boiler = $(\dot{m}_c)_{saved} \times c_{p_w}\left(t_{c_2} - t_{makeup}\right)$

$$= 9.3149 \times 4.18 \times (39 - 28)$$

$$= 428.3 \text{ kJ/h} = 0.1189 \text{ kW}$$ Ans.

(ii) Air ejector pump capacity without air cooler section = 267.7 m^3/h

Air ejector pump capacity with air cooler section = 152.758 m^3/h

Thus, percentage reduction in air ejector load = $\frac{267.7 - 152.758}{267.7} \times 100 = 42.93\%$

(iii) Now, Taking the energy balance around the control volume of condenser and using subscripts *s*, *a* and *c* representing steam, air and condensate respectively we have

$$Q_{rej} + \dot{m}_c h_c + \dot{m}_{a_2} h_{a_2} + \dot{m}_{s_2} h_{s_2} = \dot{m}_{a_1} h_{a_1} + \dot{m}_{s_1} h_{s_1}$$

or

$$Q_{rej} = \dot{m}_{a_1} h_{a_1} + \dot{m}_{s_1} h_{s_1} - \dot{m}_c h_c - \dot{m}_{a_2} h_{a_2} - \dot{m}_{s_2} h_{s_2}$$

Here,

$$\dot{m}_{a_1} = \dot{m}_{a_2} = 16 \text{ kg/h}$$

$$\dot{m}_c = (40 \times 10^3 - 3.7282) \text{ kg/h}$$

$$\dot{m}_{s_2} = 3.7282 \text{ kg/h}$$

From steam table

$$h_{s_1} = h_g \text{ at } 50°\text{C} = 2591.3 \text{ kJ/kg}$$

$$h_{s_2} = h_g \text{ at } 26°\text{C} = 2548.3 \text{ kJ/kg}$$

$$h_c = h_f \text{ at } 39°\text{C} = 163.35 \text{ kJ/kg}$$

$\therefore$ $$Q_{rej} = 16 \times 1.005(50 - 26) + 40 \times 10^3 \times 2591.3 - (40 \times 10^3 - 3.7282) \times 163.35 - 3.7282 \times 2548.3$$

$$= 385.92 + 103652000 - 6533390.999 - 9500.572$$

$$Q_{rej} = 97109494.35 \text{ kJ/h} = 26.974 \text{ MW}$$

But $$Q_{rej} = 26.974 \times 10^3 = m_w c_{p_w} \left(t_{c_2} - t_{c_1}\right)$$

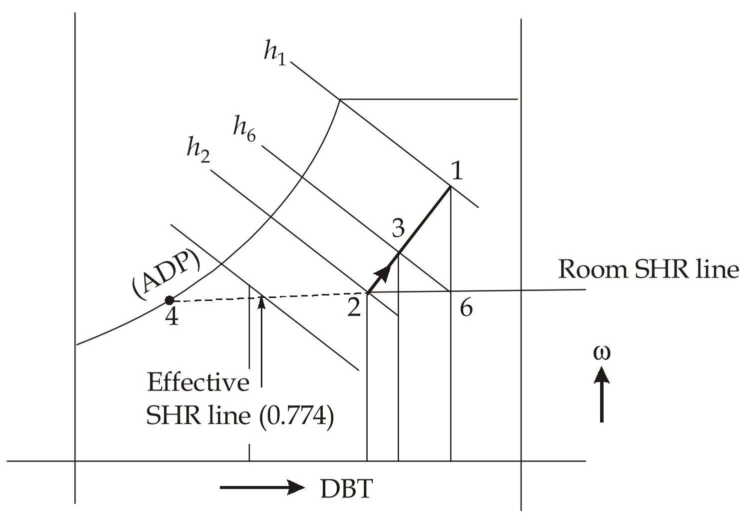
or $$m_w = \frac{26.974 \times 10^3}{4.18 \times (39 - 30)} = 717.012 \text{ kg/s}$$ **Ans.**

8. (a) Solution:

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,

$$m_{a1} = \frac{75}{60} \times 0.9 = 1.125 \text{ kg/s}$$

(i) Sensible heat load of outside air

$= 1.125(h_6 - h_2)$

$= 1.125(63 - 50) = 14.625$ kW

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

$= 1.125(80 - 63) = 19.125$ kW

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

(ii) Total sensible load taken by the plant

$= 14.625 + 58 = 72.625$ kW

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

∴ Total latent heat load taken by the plant

$= 72.625 + 33.125 = 105.75$ kW

(iii) Effective room sensible heat load

$= 58 + 0.2(14.625)$

$= 60.925$ kW

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

$= 17.825$ kW

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

(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

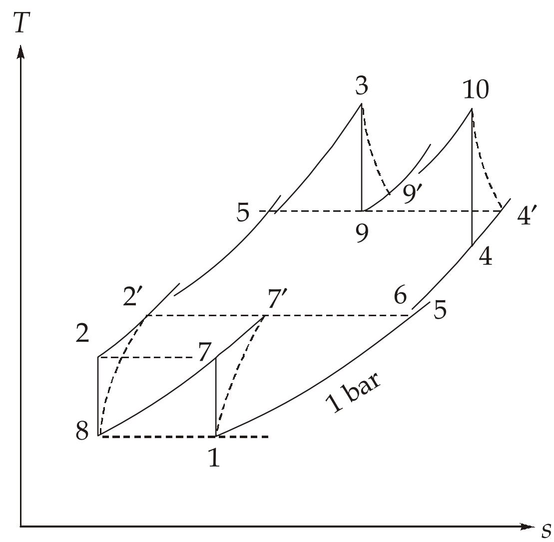
$$\therefore \text{ Percentage recirculated air} = \frac{193.60}{268.6} \times 100 = 72.07\%$$

The air temperature entering the coil,

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

$$= 28.21°\text{C}$$

8. (b) Solution:



Given: $P = 2000$ kW; $T_1 = 25 + 273 = 298$ K; $r_{p'} = 2.6$

$T_3 = 730 + 273 = 1003$ K

$$T_7 = T_1\left[r_{p'}\right]^{\frac{\gamma-1}{\gamma}} = 298 \times [2.6]^{\frac{1.4-1}{1.4}}$$

$$= 391.54 \text{ K}$$

$$\eta_c = 0.82 = \frac{T_7 - T_1}{T_{7'} - T_1}$$

$$0.82 = \frac{391.54 - 298}{T_{7'} - 298}$$

$$T_{7'} = 412.07 \text{ K} = T_{2'}$$

Work of low pressure compressor,

W_{LPC} = Work output of high pressure turbine (W_{HPT})

$$= c_{p_a} \times (T_{7'} - T_1)$$

$$W_{LPC} = 1.005 \times (412.07 - 298)$$

$$W_{LPC} = 114.64 \text{ kJ/kg}$$

$$W_{HPT} = 114.64 = c_{p_g} \times (T_3 - T_{9'})$$

$$114.64 = 1.147 \times (1003 - T_{9'})$$

$$T_{9'} = 903.05 \text{ K}$$

$$\eta_T = \frac{T_3 - T_{9'}}{T_3 - T_9}$$

$$0.82 \times (1003 - T_9) = 1003 - 903.05$$

$$T_9 = 881.109 \text{ K}$$

$$P_2 = P_{2'} = 1 \times 2.6 \times 2.6 = 6.76 \text{ bar}$$

Pressure after regeneration and combustion = $0.98 \times 0.98 \times P_2$

$$P_3 = 0.98 \times 0.98 \times 6.76$$

$$= 6.492 \text{ bar}$$

$$\frac{T_3}{T_9} = \left(\frac{P_3}{P_9}\right)^{\frac{\gamma_g - 1}{\gamma_g}}$$

$$\frac{1003}{881.109} = \left(\frac{6.492}{P_9}\right)^{\frac{1.33-1}{1.33}}$$

$$P_9 = 3.85 \text{ bar}$$

Pressure after reheating, $P_{10} = 0.98 \times P_9$

$$= 0.98 \times 385 = 3.773 \text{ bar}$$

$$P_4 = 1 \times 1.02 = 1.02 \text{ bar}$$

$$\frac{T_{10}}{T_4} = \left(\frac{P_{10}}{P_4}\right)^{\frac{\gamma_g - 1}{\gamma_g}}$$

$$\frac{1003}{T_4} = \left(\frac{3.773}{1.02}\right)^{\frac{1.33-1}{1.33}}$$

$$T_4 = 725.02 \text{ K}$$

$$\eta_T = 0.82 = \frac{1003 - T_{4'}}{1003 - 725.02}$$

$$T_{4'} = 775.05 \text{ K}$$

Work done by low pressure turbine,

$$W_{LPT} = c_{pg}(T_{10} - T_{4'})$$

$$= 1.147 \times (1003 - 775.05)$$

$$= 261.46 \text{ kJ/kg}$$

Work output low pressure turbine,

$$W_{net} = W_{LPT} - W_{HPT}$$

$$= 261.46 - 114.64$$

$$= 146.82 \text{ kJ/kg}$$

Mass flow rate of air, $\dot{m}_a = \dfrac{\text{Power}}{W_{net}} = \dfrac{2000}{146.82} = 13.62 \text{ kg/s}$

Now, Effectiveness, $\in = \dfrac{T_{5'} - T_{2'}}{T_{4'} - T_{2'}}$

$$0.78 = \frac{T_5 - 412.07}{775.05 - 412.07}$$

$$T_5 = 695.19 \text{ K}$$

Total heat added, $Q_{in} = c_{p_g}(T_3 - T_5) + c_{p_g} \times (T_{10} - T_{9'})$

$$Q_{in} = 1.147 \times [1003 - 695.19 + 1003 - 881.109]$$

$$= 492.88 \text{ kJ/kg}$$

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = \frac{146.82}{492.88} = 0.298$$ **Ans.**

$$W_{LPC} = W_{HPC} = W_{HPT} = 13.62 \times 114.64 = 1561.39 \text{ kW}$$

$$W_{LPT} = 13.62 \times 261.46 = 3561.085 \text{ kW}$$ **Ans.**

$$\dot{m}_f = \frac{\dot{m}_a \times \text{Heat input}}{\eta_{comb} \times CV} = \frac{13.62 \times 492.88}{0.98 \times 44 \times 10^3} = 0.1557 \text{ kg/s}$$

$$\dot{m}_f = 0.1557 \times 3600 = 560.46 \text{ kg/h}$$

$$\text{sfc} = \frac{\dot{m}_f}{\text{Power}} = \frac{560.46}{2000} \text{ kg/kWh}$$

$$= 0.2802 \text{ kg/kWh}$$ **Ans.**

8. (c) Solution:

Heat energy required for cooking,

$$E_1 = 50 \times 1750 = 87500 \text{ kJ/day}$$

Heat energy required for breakfast snacks, etc.

$$E_2 = 0.5 \times 87500$$
$$= 43750 \text{ kJ/day}$$

Heat energy required for bathing purpose,

$$E_3 = 50 \times 20 \times 4.2 \times (55 - 25)$$
$$= 126000 \text{ kJ/day}$$

Energy required for 3 lamps, for 12 families

$$E_4 = \frac{3 \times 40 \times 12 \times 5 \times 3600}{1000 \times 0.4}$$
$$= 64800 \text{ kJ/day}$$

∴ Total thermal energy to be supplied,

$$E = \frac{E_1 + E_2 + E_3}{0.6} + E_4$$

$$E = \frac{87500 + 43750 + 126000}{0.6} + 64800$$

$$E = 493550 \text{ kJ/kg}$$

∴ $$\text{Biogas required} = \frac{493550}{17500} = 28.2 \text{ m}^3/\text{day}$$

Required dry matter to produce 28.2 m^3 of biogas = $\frac{28.2}{0.34}$ = 82.95 kg of dry matter

Considering 18% of solid matter in cow dung, the required wet dung

$$= \frac{82.95}{0.18} = 460.83 \text{ kg/day}$$

∴ Number of cattles required to produce 460.83 kg of dung per day,

$$n = \frac{460.83}{7} = 65.83 \simeq 66$$ Ans.

Required volume of gas holder = 0.6 × 28.2 = 16.92 m^3 Ans.

Daily feed of slurry = dung + water

$= 460.83 + 460.83$

$= 921.66$ kg/day

$\therefore$ Volume of slurry $= \dfrac{921.66}{1090} = 0.845\ \text{m}^3/\text{day}$

As retention period is 50 days,

$\therefore$ Volume of digester $= \dfrac{50 \times 0.845}{0.9} = 46.94 \simeq 47\ \text{m}^3$ Ans.