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Detailed Solutions

ESE-2022 Mains Test Series

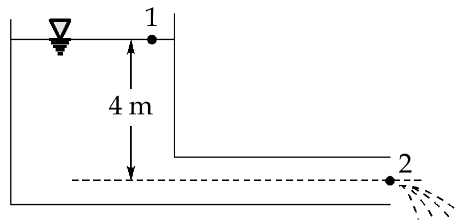
Mechanical Engineering Test No : 12

Full Syllabus Test (Paper-I)

Section : A

1. (a)

Given : d = 20 cm, L = 50 m, H = 4 m, f = 0.009



Applying Bernoulli's equation at the top of the water surface in tank and at the outlet of pipe.

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 + h_i + h_f$$

$$0 + 0 + 4 = 0 + \frac{V_2^2}{2g} + 0 + 0.5\frac{V_2^2}{2g} + \frac{4fLV_2^2}{2gd}$$

or
$$4 = \frac{V_2^2}{2g} + 0.5\frac{V_2^2}{2g} + \frac{4fLV_2^2}{2gd}$$

$$4 = \frac{V_2^2}{2g}\left[1 + 0.5 + \frac{4 \times 0.009 \times 50}{0.2}\right]$$

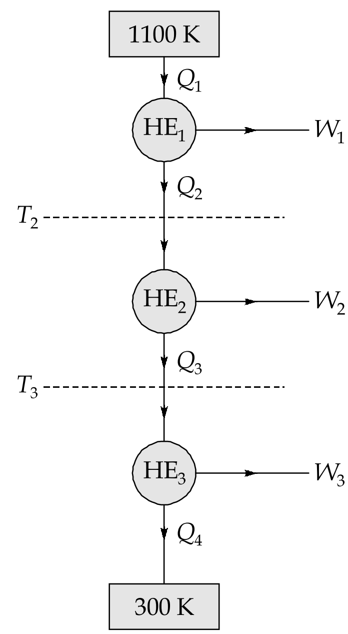
$$4 = \frac{V_2^2}{2g}[1.5+9]$$

$$V_2 = 2.734 \text{ m/s}$$

$\therefore$ Rate of flow, $Q = A \times V_2 = \frac{\pi}{4} \times 0.2^2 \times 2.734$

$$= 85.89\ l/\text{s}$$ Ans.

1. (b)



Here, $W_1 : W_2 : W_3 = 3 : 2 : 1$

Efficiency of $HE_1 = \frac{W_1}{Q_1} = \left(1 - \frac{T_2}{1100}\right)$

$\Rightarrow$ $$Q_1 = \frac{1100W_1}{1100 - T_2}$$ ...(i)

For HE_2, $$\eta_2 = \frac{W_2}{Q_2} = 1 - \frac{T_3}{T_2}$$ ...(ii)

For HE_3, $$\eta_3 = \frac{W_3}{Q_3} = 1 - \frac{300}{T_3}$$ ...(iii)

From energy balance of HE_1,

$$Q_1 = W_1 + Q_2$$

$\Rightarrow$ $$Q_2 = Q_1 - W_1$$

From equation (i), we get

$\Rightarrow$ $$Q_2 = \left\{\frac{1100W_1}{1100 - T_2} - W_1\right\}$$

$$Q_2 = W_1\left\{\frac{T_2}{1100 - T_2}\right\}$$ ...(iv)

Substituting Q_2 in equation (ii), we get

$$\frac{W_2}{W_1\left(\frac{T_2}{1100 - T_2}\right)} = 1 - \frac{T_3}{T_2}$$

$$\frac{W_2}{W_1} = \left(\frac{T_2}{1100 - T_2}\right)\left(\frac{T_2 - T_3}{T_2}\right)$$

$$\frac{W_2}{W_1} = \frac{2}{3} = \frac{T_2 - T_3}{1100 - T_2}$$

$\Rightarrow$

$$2200 - 2T_2 = 3T_2 - 3T_3$$

$$5T_2 - 3T_3 = 2200 \quad \text{...(v)}$$

Now, energy balance of HE_2, we get

$$Q_2 = W_2 + Q_3$$

and from equation (ii), we get

$$\frac{W_2}{W_2 + Q_3} = 1 - \frac{T_3}{T_2}$$

$$\frac{W_2}{W_2 + Q_3} = \frac{T_2 - T_3}{T_2}$$

$$W_2T_2 = (W_2 + Q_3)(T_2 - T_3)$$

$$W_2T_2 = W_2T_2 + Q_3T_2 - W_2T_3 - Q_3T_3$$

$$Q_3 = \frac{W_2T_3}{T_2 - T_3} \quad \text{...(vi)}$$

Substituting Q_3 in equation (iii), we get

$$\frac{W_3}{\left(\frac{W_2T_3}{T_2 - T_3}\right)} = \frac{T_3 - 300}{T_3}$$

$$\frac{W_3}{W_2} = \left(\frac{T_3}{T_2 - T_3}\right)\left(\frac{T_3 - 300}{T_3}\right)$$

$$\frac{1}{2} = \frac{T_3 - 300}{T_2 - T_3}$$

$$T_2 - T_3 = 2T_3 - 600$$

$$3T_3 - T_2 = 600 \quad \text{...(vii)}$$

From equation (v) and (vii), we have

$$5T_2 - 3T_3 = 2200$$

$$3T_3 - T_2 = 600$$

Solving these equations, we get

$$T_2 = 700 \text{ K}$$

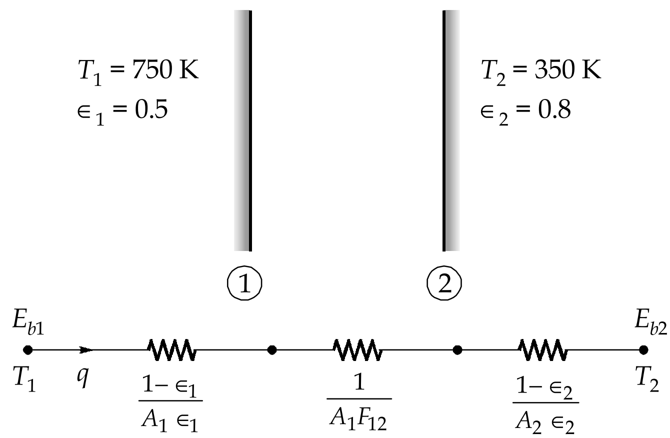
$$T_3 = 433.33 \text{ K}$$ **Ans.**

1. (c)

Assumptions:

1. Steady state condition exist.
2. All surfaces are diffuse, gray.
3. Convection effects are negligible.

We have,



Net rate of radiation heat transfer between two large parallel plates ($F_{12} = 1$) per unit area,

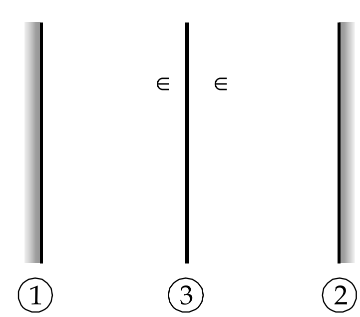
$$(q_{1-2})_{\text{without shield}} = \frac{\sigma\left(T_1^4 - T_2^4\right)}{\frac{1-\in_1}{A_1 \in_1} + \frac{1}{A_1 F_{12}} + \frac{1-\in_2}{A_2 \in_2}}$$

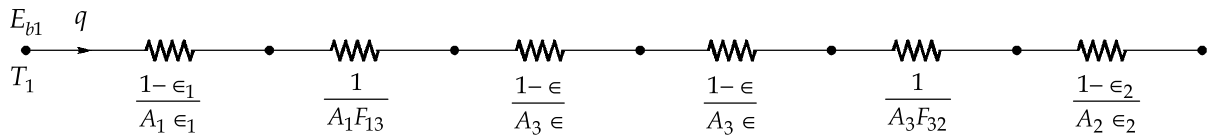
$$= \frac{\sigma\left(T_1^4 - T_2^4\right)}{\frac{1}{\in_1} + \frac{1}{\in_2} - 1} \qquad \left(\because A_1 = A_2 = 1 \text{ m}^2\right)$$

$$= \frac{5.67 \times 10^{-8}\left(750^4 - 350^4\right)}{\frac{1}{0.5} + \frac{1}{0.8} - 1}$$

$$= 7595.28 \text{ W/m}^2$$

Now, let the emissivity of radiation shield be '$\in$'





Here, $F_{13} = F_{32} = 1$ [Parallel plates]

$$A_1 = A_2 = A_3 = 1 \text{ m}^2$$

Now, net rate of radiation heat transfer between two large parallel plates per unit area,

$$(q_{1\text{-}2})_{\text{with shield}} = \frac{\sigma\left[T_1^4 - T_2^4\right]}{\left(\frac{1}{\epsilon_1}+\frac{1}{\epsilon_2}-1\right)+\left(\frac{2}{\epsilon}-1\right)}$$

$$= \frac{5.67\times 10^{-8}(750^4 - 350^4)}{\left(\frac{1}{0.5}+\frac{1}{0.8}-1\right)+\left(\frac{2}{\epsilon}-1\right)} = \frac{17089.38}{1.25+\frac{2}{\epsilon}}$$

As per the condition,

$$(q_{1\text{-}2})_{\text{with shield}} = 0.15\,(q_{1\text{-}2})_{\text{without shield}}$$

$$\frac{17089.38}{1.25+\frac{2}{\epsilon}} = 0.15(7595.28)$$

$$15 = 1.25+\frac{2}{\epsilon}$$

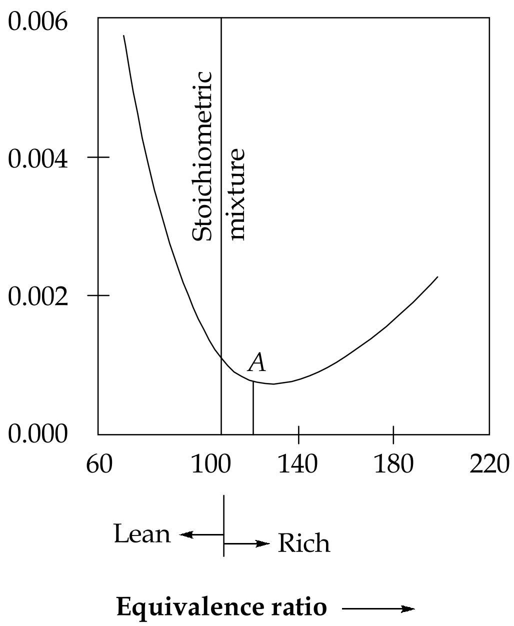
$$\epsilon = \frac{2}{13.75} = 0.1454$$

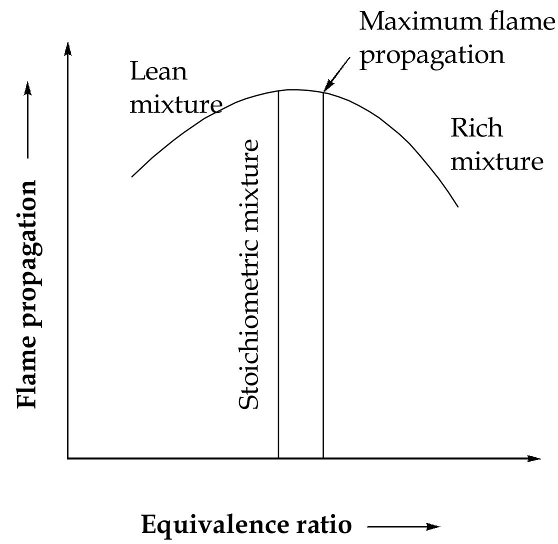
1. (d)

Effect of the following on the flame speed during combustion in an SI engine is:

1. **Compression ratio:** A high compression ratio increases the pressure and temperature of the working mixture which reduces the initial preparation phase of combustion and hence less ignition advance is needed. High pressure and temperature of the compressed mixture also speed up the second phase of combustion. Thus engine with higher compression ratio have higher flame speeds.
2. **Intake pressure:** Flame speed increases with the increase in the intake pressure and temperature. A higher initial pressure and temperature may help to form a better homogenous air-vapour mixture which helps in increasing the flame speed.
3. **Air-fuel ratio:** The air-fuel ratio has a very significant influence on the flame speed. The highest flame velocities are obtained with somewhat richer mixture. When the mixture is made leaner or richer the flame speed decreases. Less thermal energy is released in the case of lean mixtures resulting in lower flame temperature. Very

rich mixtures lead to incomplete combustion which results again in the release of less thermal energy.

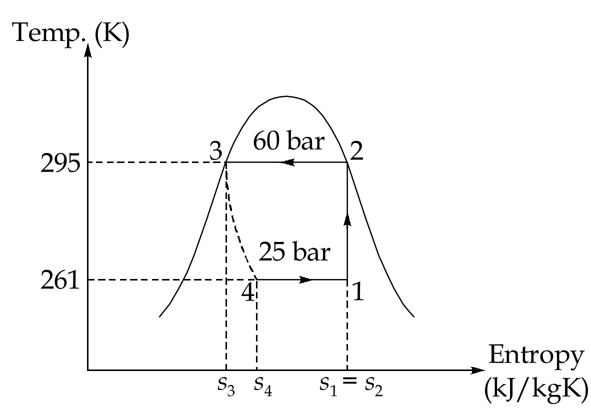




4. **Engine load:** The cycle pressure increases when the engine output is increased or the load on engine is increased. With the increased throttle opening the cylinder gets filled to a higher density. This results in increased flame speed. When the output is decreased by throttling, the initial and final compression pressures decrease and dilution of the working mixture increases. The smooth development of self-propagation nucleus of flame becomes unsteady and difficult.

1. (e)

Given: $P_2 = P_3 = 60$ bar, $P_1 = P_4 = 25$ bar, $T_2 = T_3 = 295$ K, $T_1 = T_4 = 261$ K



COP of cycle:

Let, x_1 = Dryness fraction of vapour entering compressor at point 1.

So, $s_1 = s_2$

$\Rightarrow$ $1.0332 = s_{f1} + x_1 s_{fg1}$ $[\because s_2 = s_{g@60\text{ bar}}]$

$$\Rightarrow \quad x_1 = \frac{1.0332 - 0.226}{1.2464 - 0.226} = 0.791$$

So, enthalpy at point 1

$$h_1 = h_{f1} + xh_{fg1}$$

$$= 56.32 + 0.791(322.58 - 56.32)$$

$$= 266.93 \text{ kJ/kg}$$

So, $$\text{COP of cycle} = \frac{h_1 - h_3}{h_2 - h_1} = \frac{h_1 - h_{f@60\text{ bar}}}{h_{g@60\text{ bar}} - h_1}$$

$$\text{COP} = \frac{266.93 - 151.96}{293.29 - 266.93} = 4.3612$$ **Answer**

Capacity of refrigerator:

We know that heat extracted or RE produced per kg of refrigerant,

$$\text{RE} = h_1 - h_4$$

As we know that during throttling enthalpy remains constant so,

$$h_3 = h_4 = h_{f@60\text{ bar}}$$

So, $\text{RE} = 266.93 - 151.96 = 114.97$ kJ/kg

Since the refrigerant flow is 5 kg/min,

So, Refrigerant capacity, RC = $\dot{m} \times RE$

$= 5 \times 114.97 = 574.85$ kJ/min

So, capacity of refrigerator (in TR) = $\frac{574.85}{211} = 2.724$ TR **Answer**

2. (a)

Given: d = 80 mm = 0.08 m, L = 800 m, Q = 0.48 m^3/min = 0.008 m^3/sec, ν = 0.015 stokes = 0.015×10^{-4} m^2/s, ρ = 1000 kg/m^3

$$\text{Mean velocity, } V = \frac{Q}{\text{Area}} = \frac{0.008}{\frac{\pi}{4}(0.08)^2} = 1.591 \text{ m/s}$$

$$\text{Reynolds number, Re} = \frac{V \times d}{\nu} = \frac{1.591 \times 0.08}{0.015 \times 10^{-4}} = 8.485 \times 10^4$$

As the Reynolds number is more than 4000, the flow is turbulent.

Now, $$f = \frac{0.0791}{(8.485 \times 10^4)^{1/4}} = 0.004636$$

$$\text{Head lost, } h_f = \frac{4fLV^2}{2gd} = \frac{4 \times 0.004636 \times 800 \times 1.591^2}{2 \times 9.81 \times 0.08}$$

$$h_f = 23.92 \text{ m}$$ Ans.

Wall shearing stress, $$\tau_0 = \frac{\rho f V^2}{2} = \frac{10^3 \times 0.004636 \times 1.591^2}{2}$$

$$\tau_0 = 5.866 \text{ N/m}^2$$ Ans.

Centre line velocity, at $y = 0.04$ m, $u = U_{max}$

$$\therefore \quad \frac{U_{max}}{U^*} = 5.75 \log_{10} \frac{U^* y}{\nu} + 5.55$$

$$U^* = \sqrt{\frac{\tau_0}{\rho}} = \sqrt{\frac{5.866}{1000}} = 0.0765 \text{ m/s}$$

$$\therefore \quad \frac{U_{max}}{0.0765} = 5.75 \log_{10} \frac{0.0765 \times 0.04}{0.015 \times 10^{-4}} + 5.55$$

$$= 5.75 \log_{10} 2040 + 5.55 = 24.58$$

$$\therefore \quad U_{max} = 0.0765 \times 24.58$$

$$= 1.88 \text{ m/s}$$ Ans.

Shear stress at 30 mm from pipe wall,

$$\tau = \tau_0 \left(1 - \frac{y}{r_0}\right), \text{ where } y = \text{distance from pipe wall}$$

$$\tau = 5.866 \left(1 - \frac{0.03}{0.04}\right)$$

$$\tau = 1.4665 \text{ N/m}^2$$ Ans.

Now, velocity at $y = 0.03$ m, is given by

$$\frac{u}{U^*} = 5.75 \log_{10} \frac{U^* y}{\nu} + 5.55$$

or $$\frac{u}{0.0765} = 5.75 \log_{10} \frac{0.0765 \times 0.03}{0.015 \times 10^{-4}} + 5.55$$

$$\frac{u}{0.0765} = 23.86$$

$\therefore \quad u = 1.825 \text{ m/s}$ Ans.

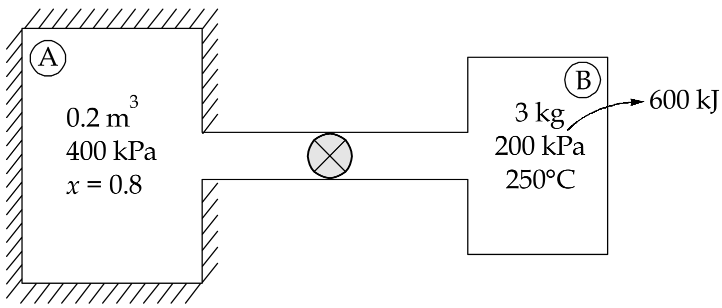
Thickness of laminar sublayer,

$$\delta' = \frac{11.6\nu}{U^*} = \frac{11.6 \times 0.015 \times 10^{-4}}{0.0765}$$

$$\delta' = 2.274 \times 10^{-2} \text{ cm}$$

or $\delta' = 0.02274 \text{ cm}$ Ans.

2. (b)



In tank (A),

Process is reversible adiabatic,

$\Rightarrow \quad \Delta s_A = 0$

$\Rightarrow$ Initial entropy of A = Final entropy of A

$$(s_i)_A = (s_f)_A$$

$$1.7765 + 0.8 \times 5.1191 = 1.6717 + x_2 \times 5.32$$

$$x_2 = 0.7895$$

Since, final condition is a mixture, so the temperature in tank A is T_{sat} corresponding to P = 300 kPa i.e.

$$T_{sat} = 133.52°\text{C}$$

Now, at initial state, $v_{1A} = v_f + xv_{fg}$

$$= 0.001084 + 0.8(0.46242)$$

$$= 0.37102 \text{ m}^3/\text{kg}$$

Initial mass of steam in tank A,

$$m_{1A} = \frac{V_{1A}}{v_{1A}} = \frac{0.2}{0.37102} = 0.539 \text{ kg}$$

At final state,

$$\nu_{2A} = v_f + xv_{fg}$$
$$= 0.001073 + 0.7895(0.60582)$$
$$= 0.4794 \text{ m}^3/\text{kg}$$

So, $$m_{2A} = \frac{V_{2A}}{v_{2A}} = \frac{0.2}{0.4794} = 0.4172 \text{ kg}$$

So, change in the mass of the mixture in tank A = 0.539 – 0.4172 = 0.1218 kg, this mass is transferred to tank B.

So, final mass in tank B = 3 + 0.1218 = 3.1218 kg

Now, consider tank A and tank B as a single system.

From first law of thermodynamics for closed system

$$\delta Q = \Delta U + \delta W$$

But, $\delta W = 0$ [$\because$ no work transfer]

$$-Q_{out} = \Delta U_A + \Delta U_B$$

$$-Q_{out} = (m_2u_2 - m_1u_1)_A + (m_2u_2 - m_1u_1)_B \quad \text{...(i)}$$

For tank A,

$$u_2 = 561.11 + 0.7895(1982.1)$$
$$= 2125.97 \text{ kJ/kg}$$
$$u_1 = 604.22 + 0.8(1948.9)$$
$$= 2163.34 \text{ kJ/kg}$$

For tank B,

$$u_1 = 2731.2 \text{ kJ/kg}$$

and $Q_{out} = 600$ kJ

Now, putting values in equation (i), we get

$$-600 = (0.4172 \times 2125.97 - 0.539 \times 2163.84) + (3.1218 \times u_2 - 3 \times 2731.2)$$
$$u_2 = 2521.93 \text{ kJ/kg}$$

2. (c)

In cylindrical co-ordinates, the radial variation of temperature at steady state, when there is heat generation is given by

$$\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) = \frac{-q_g}{k}$$

and $$q_g = q_0\left(1-\left(\frac{r}{r_0}\right)^2\right)$$

$$\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) = \frac{-q_0 r}{k}\left[1-\left(\frac{r}{2}\right)^2\right]$$

$\Rightarrow$ $$r\frac{\partial T}{\partial r} = \frac{-q_0}{k}\left[\frac{r^2}{2}-\frac{r^4}{4r_0^2}\right]+C_1$$

$$k\frac{\partial T}{\partial r} = -q_0\left(\frac{r}{2}-\frac{r^3}{4r_0^2}\right)+\frac{C_1 k}{r}$$

$$k\frac{\partial T}{\partial r} = \frac{-q_0 r}{2}\left(1-\frac{r^2}{2r_0^2}\right)+\frac{C_1 k}{r}$$

at $r = 0$, $$\frac{\partial T}{\partial r} = 0$$

$\Rightarrow$ $$C_1 = 0$$

$$k\frac{dT}{\partial r} = \frac{-q_0 r}{2}\left(1-\frac{r^2}{2r_0^2}\right)$$

Heat transferred from the rod surface per unit area,

$$q_F = -k\left(\frac{\partial T}{\partial r}\right)_{r=r_0}$$

$$q_F = -k\left(\frac{-q_0 r}{2k}\left(1-\frac{r^2}{2r_0^2}\right)\right)_{r=r_0}$$

$$q_F = \frac{q_0 r_0}{2}\left(1-\frac{1}{2}\right)$$

$$q_F = \frac{q_0 r_0}{4} \qquad \text{...(i)}$$

We have, heat removal from surface,

$$q_F = 1650 \text{ W/m}^2$$
$$k = 25 \text{ W/mK}$$
$$r_0 = 1.25 \text{ cm}$$

So, from equation (i), we get

$$1650 = \frac{q_0 \times 1.25\times 10^{-2}}{4} \quad = 528000 \text{ W/m}^3$$

Now, $$\frac{dT}{dr} = \frac{-q_0 r}{2k}\left(1-\frac{1}{2}\left(\frac{r}{r_0}\right)^2\right)$$

$$T = \frac{-q_0}{2k}\left(\frac{r^2}{2}-\frac{r^4}{8r_0^2}\right)+C_2$$

at $r = 0$, $T = T_c$, centreline temperature

$\Rightarrow$ $$C_2 = T_c$$

then, $$T = T_c - \frac{q_0}{2k}\left(\frac{r^2}{2}-\frac{r^4}{8r_0^2}\right)$$

At $r = r_0$

$$T_0 = T_c - \frac{q_0}{2k}\left(\frac{r_0^2}{2}-\frac{r_0^4}{8r_0^2}\right)$$

$$T_0 = T_c - \frac{q_0}{2k}\left(\frac{3}{8}r_0^2\right)$$

$$T_0 = T_c - \frac{3q_0 r_0^2}{16k}$$

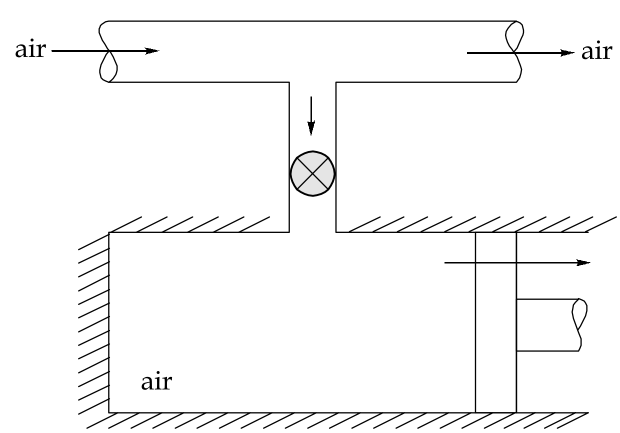
$$T_c - T_0 = \frac{3}{16}\left(\frac{q_0 r_0^2}{k}\right)$$

$$= \frac{3}{16}\left(\frac{528000\times(1.25\times10^{-2})^2\times10^3}{25}\right) = 618.75°C$$

So, temperature drop from the centreline to the surface is $T_c - T_0 = 618.75°C$

3. (a)

Given : Initial pressure = P_1; Initial volume = V_1; Final pressure = $P_2 = P_1$;
Final volume = $V_2 = 2V_1$; Pipeline pressure = P_P; Pipeline temperature = T_P



Now, amount of air entered into the cylinder = $m_2 - m_1$

From energy equation

$$h_p(m_2 - m_1) = m_2u_2 - m_1u_2 - W_{1\text{-}2} \quad \text{...(i)}$$

Assuming, air an ideal gas,

We get, $h = C_P T$

$u = C_V T$

and $m = \frac{PV}{RT}$

Putting values in equation (i), we get

$$C_P T_P\left(\frac{P_2V_2}{RT_2} - \frac{P_1V_1}{RT_1}\right) = \frac{P_2V_2}{RT_2}C_V T_2 - \frac{P_1V_1}{RT_1}C_V T_1 + P(V_2 - V_1) \qquad [\because P_2 = P_1]$$

$$\frac{C_P T_P}{R}\left(\frac{V_2}{T_2} - \frac{V_1}{T_1}\right) = \frac{C_V}{R}(V_2 - V_1) + (V_2 - V_1)$$

$$(V_2 - V_1) - \frac{C_P T_P}{R}\left(\frac{V_2}{T_2} - \frac{V_1}{T_1}\right) + \frac{C_V}{R}(V_2 - V_1) = 0$$

$$(V_2 - V_1)\left(\frac{C_V}{R} + 1\right) - \frac{C_P T_P}{R}\left(\frac{V_2}{T_2} - \frac{V_1}{T_1}\right) = 0$$

$$(V_2 - V_1)\left(\frac{C_P}{R}\right) - \left(\frac{C_P}{R}\right)T_P\left(\frac{V_2}{T_2} - \frac{V_1}{T_1}\right) = 0$$

$\because$ $$\frac{C_P}{R} \neq 0$$

$$(V_2 - V_1) - T_P\left(\frac{V_2}{T_2} - \frac{V_1}{T_1}\right) = 0$$

$$\frac{T_P V_2}{T_2} = (V_2 - V_1) + \frac{T_P V_1}{T_1}$$

$$T_2 = \frac{T_P V_2}{(V_2 - V_1) + \frac{T_P V_1}{T_1}}$$

$$T_2 = \frac{V_2}{\left(\frac{V_2 - V_1}{T_P}\right) + \frac{V_1}{T_1}}$$

and $V_2 = 2V_1$

$\therefore$ $$T_2 = \frac{2V_1}{\frac{2V_1 - V_1}{T_P} + \frac{V_1}{T_1}}$$

$$T_2 = \frac{2}{\frac{1}{T_1} + \frac{1}{T_P}}$$

3. (b)

Given : $A = 1.5 \times 3.5$ cm^2, $V = 1.2$ m/s, $T_s = 85°C$, $T_i = 40°C$, $T_0 = 70°C$, $\eta_p = 60\%$,
$\rho = 985.5$ kg/m^3, $k = 0.654$ W/mK, $Pr = 3.26$

Now, equivalent diameter of the duct,

$$D = \frac{4A}{P} = \frac{4ab}{2(a+b)}$$

$$= \frac{4 \times 1.5 \times 3.5}{2(1.5+3.5)} = 2.1 \text{ cm}$$

Reynolds number, $$Re = \frac{VD}{\nu} = \frac{1.2 \times 2.1 \times 10^{-2}}{0.517 \times 10^{-6}}$$

$$= 48742.747 > 4000$$

So, flow is turbulent

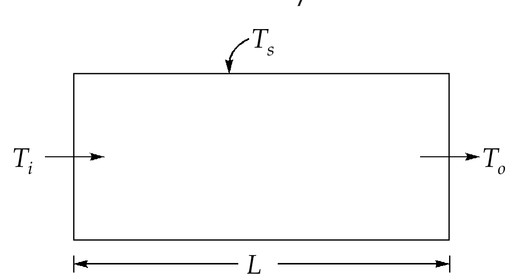
Using Dittus-Boilter equation for heating of fluid,

$$Nu = 0.023 Re^{0.8} Pr^{0.4}$$

$$\frac{hD}{k} = 0.023(48742.747)^{0.8}(3.26)^{0.4}$$

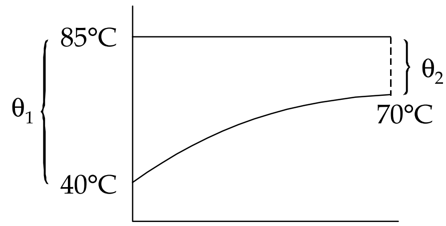
$$\frac{h \times 2.1 \times 10^{-2}}{0.654} = 207.655$$

$$h = 6466.97 \text{ W/m}^2\text{K}$$



$$\theta_1 = 85 - 40 = 45°C$$

$$\theta_2 = 85 - 70 = 15°C$$



$$\text{So, LMTD } (\theta_m) = \frac{\theta_1 - \theta_2}{\ln\left(\frac{\theta_1}{\theta_2}\right)} = \frac{45 - 15}{\ln\left(\frac{45}{15}\right)} = 27.307°\text{C}$$

and mass flow rate of water, $\dot{m} = \rho A V$

$$= 985.5 \times 1.5 \times 3.5 \times 10^{-4} \times 1.2$$

$$= 0.62086 \text{ kg/s}$$

We know, heat transfer rate, $q = \dot{m} C_P \Delta T$

$$= h(PL)\theta_m$$

$$0.62086 \times 4.18 \times 10^3 \times 30 = 6466.97 \times \frac{2(1.5 + 3.5)}{100} \times L \times 27.307$$

$$L = 4.409 \text{ m}$$

Using Blasius correlation for friction factor (f),

$$f = \frac{0.316}{\text{Re}^{1/4}} = \frac{0.316}{(48742.747)^{1/4}} = 0.02130$$

$\therefore$ Pressure drop in tube, $\Delta P = f \times \frac{L}{D} \times \frac{\rho V^2}{2}$

$$\Delta P = 0.02130 \times \frac{4.409}{0.021} \times \frac{985.5 \times 1.2^2}{2}$$

$$\Delta P = 3173.14 \text{ Pa}$$

and, pumping power $= \frac{\dot{m}\Delta P}{\rho \eta_P}$

$$P = \frac{0.62086 \times 3173.14}{985.5 \times 0.6}$$

$$= 3.3318 \text{ W}$$

3. (c)

Given: $m_1 = 60$ kg/min, $T_1 = 5°C = T_2'$, $\phi_1 = 100\% = 1$

From steam table given, at $T_1 = 5°C$

$$P_{vs1} = 0.00872 \text{ bar}$$

$$\phi_1 = \frac{P_{v1}}{P_{vs1}}$$

$\Rightarrow$ $$P_{v1} = 1 \times P_{vs1} = 0.00872 \text{ bar}$$

$$\omega_1 = \frac{0.622 P_{v1}}{P - P_{v1}} = \frac{0.622 \times 0.00872}{1 - 0.00872}$$

$$= 5.47 \times 10^{-3} \text{ kg/kg d.a}$$

$$h_1 = 1.005T_1 + \omega_1(2500 + 1.88T_1)$$

$$= 1.005 \times 5 + 5.47 \times 10^{-3}(2500 + 1.88 \times 5)$$

$$= 18.75 \text{ kJ/kg}$$

For sensible heating,

$\therefore$ $$\omega_1 = \omega_2 \Rightarrow P_{v1} = P_{v2}$$

So, $$P_{v2} = 0.00872 \text{ bar}$$

Now, $$P_{v2} = \phi_2 P_{vs2}$$

$\Rightarrow$ $$P_{vs2} = \frac{0.00872}{0.2} = 0.0436 \text{ bar}$$

Now, at $$P_{vs2} = 0.0436 \text{ bar}$$

By interpolation

$$\frac{T_2 - 30}{31 - 30} = \frac{0.0436 - 0.04246}{0.04496 - 0.04246}$$

$$T_2 = 30.456°C$$

$$h_2 = 1.005 \times 30.456 + 5.47 \times 10^{-3}(2500 + 1.88 \times 30.456)$$

$$= 44.60 \text{ kJ/kg}$$

$$T_3 = 24°C \text{ (given)}$$

(i) Let m_2 is mass flow rate through the heating coil and m_2' is mass flow rate by passed.

Now after mixing of heated air and bypassed air,

$$m_3T_3 = m_2T_2 + m_2'T_2'$$

where, $$m_3 = m_1 = 60 \text{ kg/min}$$

$$m_2 = m_1 - m_2' = 60 - m_2'$$

So, $$60 \times 24 = (60 - m_2') \times 30.456 + m_2' \times 5$$

$$1440 = 1827.36 - 30.456\, m_2' + 5\, m_2'$$

$\Rightarrow$ $$m_2' = \frac{1440 - 1827.36}{-25.456} = 15.22 \text{ kg/min}$$ **Answer**

(i) Heat added by coil, $\dot{Q} = m_2(h_2 - h_1) = (m_1 - m_2')(h_2 - h_1)$

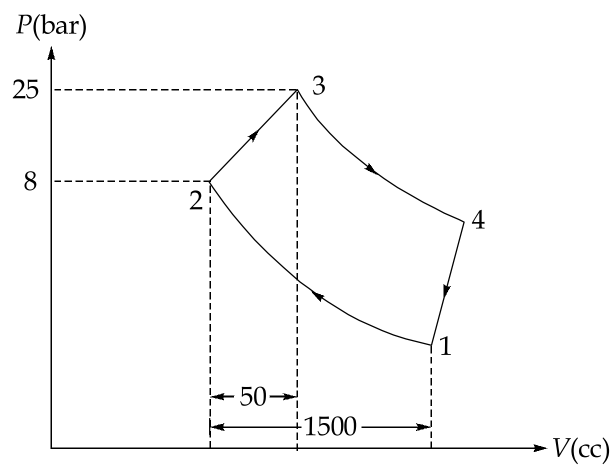
$= (60 - 15.22)\,(44.60 - 18.75)$

$= 1157.56$ kJ/min

$$\dot{Q} = 19.29 \text{ kW}$$ **Answer**

4. (a)

Given : $r = 6$, $V_s = 0.0015$ m^3, $P_2 = 8$ bar, $T_2 = 350°C$, $CV = 42$ MJ/kg, AFR = 16 : 1, $P_3 = 25$ bar



Now, $$V_1 - V_2 = 0.0015 \times 10^6 = 1500 \text{ cc}$$

$$V_1 = 6V_2$$

$$V_2 = \frac{1500}{6-1} = 300 \text{ cc}$$

$$V_3 = \frac{1}{30}(\text{Stroke volume}) + V_2$$

$$= \frac{1}{30}(1500) + 300 = 350 \text{ cc}$$

$$T_2 = 350 + 273 = 623 \text{ K}$$

and $$\frac{P_3V_3}{T_3} = \frac{P_2V_2}{T_2}$$

$$T_3 = T_2\left(\frac{P_3V_3}{P_2V_2}\right)$$

$$= 623\times\left(\frac{25\times 350}{8\times 300}\right)$$

$$= 2271.35 \text{ K}$$

We need to calculate work done and increase in internal energy between 2 and 3, so that we can find heat added by using first law of thermodynamics,

$$W_{23} = \text{Area under 2 - 3}$$

$$= \left(\frac{P_2+P_3}{2}\right)(V_3 - V_2)$$

$$= \left(\frac{8+25}{2}\right)\times 10^5 \times (350-300)\times 10^{-6}$$

$$= 82.5 \text{ J}$$

Now, mass of the mixture,

$$m = \left(\frac{PV}{RT}\right)_2 = \frac{8\times 10^5\times 300\times 10^{-6}}{287\times 623}$$

$$= 1.342\times 10^{-3} \text{ kg}$$

and $\Delta U = U_3 - U_2$

$$= mC_V(T_3 - T_2)$$

$$= m(C_P - R)(T_3 - T_2)$$

$$= 1.342\times 10^{-3}(1 - 0.287)(2271.35 - 623)$$

$$= 1.5575 \text{ kJ}$$

So, $Q_t = \Delta U + W$

$$= 1.5775 + \frac{82.5}{1000} = 1.66 \text{ kJ}$$

Actual heat liberated in one cycle,

$$Q_a = \left(\frac{m}{17}\right)\times CV \qquad \left(\because m_{gasoline} = \frac{m}{17}\right)$$

$$= \frac{1.342\times 10^{-3}}{17}\times 42000 = 3.3155 \text{ kJ}$$

So, heat lost during combustion = $Q_a - Q_t$

$$= 3.3155 - 1.66$$

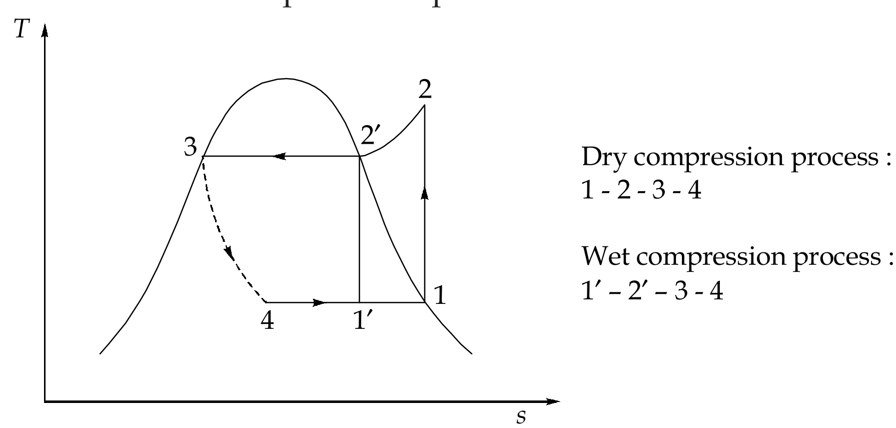
$$= 1.655 \text{ kJ}$$

$$\therefore \qquad \text{Heat lost in kJ/kg} = \frac{1.655}{1.342 \times 10^{-3}} = 1233.628 \text{ kJ/kg charge}$$

4. (b)

(i)

The liquid refrigerant absorbs heat from the cold space and changes to vapour state. The refrigerant has to dissipate this heat to the surroundings. To do so, its temperature has to be higher than the surrounding coolant. In order to increase the temperature of the refrigerant vapour greater than the surrounding, its pressure has to be increased. This is achieved with the help of a compressor.



Dry and wet compression of refrigerant on T-s diagram

In case the compression process involves the compression of dry saturated vapour or with slightly superheated vapour, it is called dry compression. In this case the complete compression process remains in superheated state. In case the compression process involves the compression of wet refrigerant, the compression is called wet compression.

The refrigerant vapour would be compressed efficiently if there is a perfect sealing between the piston and the cylinder bore. This would be satisfied to a certain extent with better surface finish obtained in the manufacturing process. In a reciprocating compressor, the lubricating oil serves two purposes: (i) It reduces the friction between the rubbing surfaces, and (ii) it acts as a sealing agent between the piston and the cylinder. In reciprocating compressors, wet compression is avoided due to the following reasons.

1. Liquid droplets present in the wet vapour wash away the lubricating oil from the cylinder walls of the compressor. Then there will not be any sealing agent between the piston and the cylinder bore. It results into a large friction between the piston

and the cylinder. Subsequently the driving motor will be overloaded and may lead to burning of the same. Since there is no proper sealing between the piston and the cylinder bore, the compression efficiency will be very poor. The blow-by loss will increase leading to poor volumetric efficiency.

2. Liquid droplets in the refrigerant would enter the compressor and damage the valves and other moving parts.

Dry compression is preferred over wet compression since it gives high volumetric efficiency and the mechanical efficiency of the compressor is increased with less chances of damage to it.

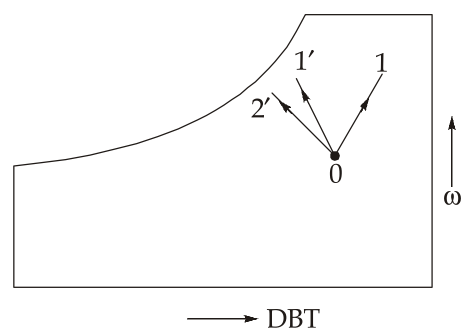
(ii)

Adiabatic saturation (0-2′): [T_w = WBT]

1. Temperature decreases.
2. Specific humidity increases.
3. Enthalpy and wet bulb temperature are constant.
4. Specific volume decreases.

Heating and Humidification: (0-1): [T > DBT]

1. Dry bulb temperature increases.
2. Specific humidity increases.
3. Enthalpy increases.
4. Dew point temperature increases.
5. Wet bulb temperature increases.
6. Specific volume increases.
7. It is used in winter air conditioning.



Cooling and Humidification (0-1′): [WBT < T < DBT]

1. Dry bulb temperature decreases.
2. Specific humidity increases.
3. Enthalpy increases.
4. Dew point temperature increases.
5. Wet bulb temperature increases.
6. Specific volume decreases.
7. It is used in desert cooler, cooling tower, etc.

Note: All the three processes are possible in air washer.

4. (c)

Engine 1 :

$$V_s = 3300 \times 10^{-6}\ \text{m}^3$$

$$\dot{V}_s = \frac{V_s \times N}{2 \times 60} = \frac{3300 \times 10^{-6} \times 4500}{2 \times 60}$$

$$= 0.12375\ \text{m}^3/\text{s}$$

$$\text{BMEP} = 9.3\ \text{bar}$$

$$\text{BP} = \text{BMEP} \times \dot{V}_s$$

$$= 9.3 \times 10^2 \times 0.12375$$

$$= 115.0875\ \text{kW}$$

and $$\eta_{\text{otto}} = 1 - \frac{1}{r^{\gamma-1}} = 1 - \frac{1}{(8.2)^{0.4}} = 0.569$$

Now, $$\text{efficiency ratio} = \frac{\eta_{ith}}{\eta_{otto}}$$

$$0.5 = \frac{\eta_{ith}}{0.569}$$

$$\eta_{ith} = 0.2845$$

also, $$\eta_{ith} = \frac{IP}{\dot{m}_f \times CV}$$

$$0.2845 = \frac{\dfrac{115.0875}{0.9}}{\dot{m}_f \times 44000}$$

$$\dot{m}_f = 0.010215\ \text{kg/s}$$

$$= 36.775\ \text{kg/hr}$$

If the test duration is t hours, then specific mass per kW of brake power is

$$\text{Specific mass} = t(\dot{m}_f) + m_{engine}$$

$$= (36.775t + 200)\ \text{kg}$$

and, $$\frac{\text{Specific mass}}{\text{kW of BP}} = \frac{36.775t + 200}{115.0875} \quad \text{...(i)}$$

Engine 2:

$$\text{Swept volume, } V_s = 3300 \text{ cc}$$
$$= 3300 \times 10^{-6} \text{ m}^3$$

$$\dot{V}_s = \frac{V_s \times N}{2 \times 60} = \frac{3300 \times 10^{-6} \times 4500}{2 \times 60}$$

$$= 0.12375 \text{ m}^3/\text{s}$$

$$\text{BMEP} = 12 \text{ bar}$$

$$\text{BP} = \text{BMEP} \times \dot{V}_s$$

$$= 12 \times 10^2 \times 0.12375 = 148.5 \text{ kW}$$

So,
$$\text{IP} = \frac{BP}{\eta_m} = \frac{148.5}{0.92} = 161.41 \text{ kW}$$

and
$$\eta_{\text{otto}} = 1 - \frac{1}{(r)^{\gamma - 1}} = 1 - \frac{1}{(5.5)^{0.4}} = 0.4943$$

Also,
$$\eta_{\text{ratio}} = \frac{\eta_{ith}}{\eta_{otto}}$$

$$\eta_{\text{ith}} = 0.5 \times 0.4943 = 0.2471$$

Now,
$$\eta_{\text{ith}} = \frac{IP}{\dot{m}_f \times CV}$$

$$0.2471 = \frac{161.41}{\dot{m}_f \times 44000}$$

$$\dot{m}_f = 0.01484 \text{ kg/s}$$

$$= 53.44 \text{ kg/hr}$$

and
$$\frac{\text{Specific mass}}{\text{kW of BP}} = \frac{53.44t + 220}{148.5} \quad \text{...(ii)}$$

From equation (i) and (ii) , we get

$$\frac{36.775t + 200}{115.0875} = \frac{53.44t + 220}{148.5}$$

$$47.4516t + 258.0645 = 53.44t + 220$$

$$t = 6.356 \text{ hours}$$

Now,

Engine I : Naturally aspirated engine with more η_{th} and less power should be used in passenger automobile for economical purposes, as less power is demanded.

Engine 2 : Supercharged engine with less η_{th} but with high power (BP) is recommended for high speed application automobile for higher speed and torque.

5. (a)

Given : $H = 200$ m, $A = 8300$ mm^2, $\eta_m = 0.93$, $Q = 0.5$ m^3/s, $P = 820$ kW

$$\text{Power at base of nozzle} = \rho g Q H = 10^3 \times 9.81 \times 0.5 \times 200 = 981 \text{ kW}$$

$$\text{Velocity of jet, } V_1 = \frac{Q}{A} = \frac{0.5}{8300 \times 10^{-6}}$$

$$\therefore \quad V_1 = 60.24 \text{ m/s}$$

$$\text{Kinetic energy of jet} = \rho g Q \times \frac{V_1^2}{2g}$$

$$= 10^3 \times 9.81 \times 0.5 \times \frac{60.24^2}{2 \times 9.81}$$

$$= 907.21 \text{ kW}$$

$$\therefore \quad \text{Power loss in nozzle} = 981 - 907.21$$

$$= 73.7856 \text{ kW} \qquad \text{Ans.}$$

$$\text{Nozzle efficiency, } \eta_n = \frac{907.21}{981} \times 100 = 92.48\% \qquad \text{Ans.}$$

Power input into the runner = Power of jet = 907.21 kW

Now, power developed by runner,

$$\text{R.P} = \frac{S.P}{\eta_{mech}} = \frac{820}{0.93} = 881.72 \text{ kW}$$

$$\therefore \quad \text{Power lost in runner} = 907.21 - 881.72$$

$$= 24.49 \text{ kW} \qquad \text{Ans.}$$

$$\therefore \quad \text{Wheel efficiency, } \eta_w = \frac{881.72}{907.21} \times 100$$

$$\eta_w = 97.19\% \qquad \text{Ans.}$$

$$\text{Power lost in friction} = \text{R.P} - \text{S.P}$$
$$= 881.72 - 820$$
$$= 61.72 \text{ kW} \qquad \text{Ans.}$$

$$\text{Overall efficiency, } \eta_0 = \eta_m \times \eta_w \times \eta_{mech}$$
$$\eta_0 = 0.9248 \times 0.9719 \times 0.93$$
$$= 0.8359 \qquad \text{Ans.}$$

5. (b)

Given : B.P. = 200 kW, $\eta_{Bth} = 0.35$, $\eta_{gasifier} = 0.6$, $(C.V.)_{producer\ gas} = 12500$ kJ/kg, $(C.V)_{diesel} = 45000$ kJ/kg

$$\text{Brake thermal efficiency, } \eta_{Bth} = \frac{B.P}{H.A/\text{sec}}$$

$$0.35 = \frac{200}{H.A/\text{sec}}$$

$$\therefore \quad \frac{H.A}{\text{sec}} = \frac{200}{0.35}$$
$$= 571.428 \text{ kW}$$

For 80% diesel replacement

$$0.8 \times \frac{H.A}{\text{sec}} = \dot{m}_p \times (C.V)_P$$

$$\therefore \quad 0.8 \times 571.428 = \dot{m}_p \times 12500$$

$$\therefore \quad \dot{m}_p = 131.657 \text{ kg/hr}$$

Now, gasifier efficiency,

$$\eta_{gasifier} = \frac{\dot{m}_p}{\dot{m}_s}$$

$$\therefore \quad 0.6 = \frac{131.657}{\dot{m}_s}$$

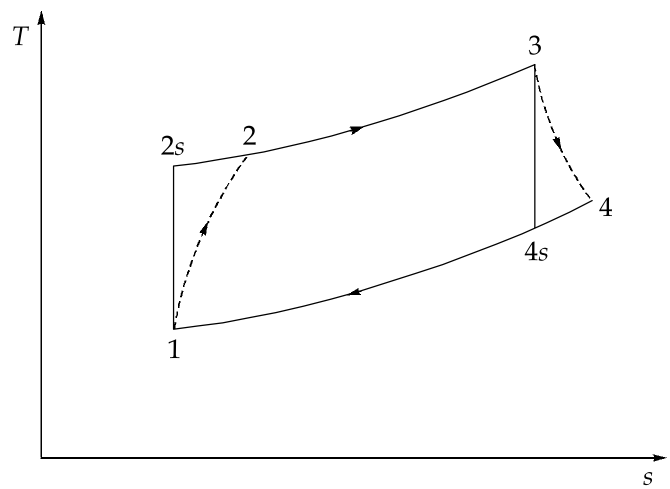
$$\therefore \quad \dot{m}_s = 219.42 \text{ kg/hr} \qquad \text{Ans.}$$

5. (c)

Given : $\eta_c = 70.42\% = 0.7042$; $\eta_T = 71\% = 0.71$; $Q_S = 476.354$ kJ/kg

$$\text{Work ratio } (r) = 0.0544$$
$$T_1 = 300 \text{ K}$$

Let pressure ratio, $r_p = \frac{P_2}{P_1} = \frac{P_3}{P_4}$



$$\text{Work ratio } (r) = \frac{W_T - W_C}{W_T} = 1 - \frac{W_C}{W_T}$$

and $$W_C = m_a C_P (T_2 - T_1)$$

$$W_T = m_a C_P (T_3 - T_4)$$

$\because$ Heat added in combustion chamber per kg of air

$$Q_S = C_P (T_3 - T_2)$$

$$476.354 = 1.005 (T_3 - T_2)$$

$$T_3 - T_2 = 474 \quad \text{...(A)}$$

and $$r = 0.0544$$

$$1 - \frac{m_a C_P (T_2 - T_1)}{m_a C_P (T_3 - T_4)} = 0.0544$$

$$\frac{T_2 - T_1}{T_3 - T_4} = 1 - 0.0544$$

$$\frac{T_2 - 300}{T_3 - T_4} = 0.9456 \quad \text{...(i)}$$

Now, compression process in compressor,

$$\frac{T_{2s}}{T_1} = (r_p)^{\frac{\gamma-1}{\gamma}}$$

$$T_{2s} = T_1 (r_p)^{\frac{\gamma-1}{\gamma}}$$

and $$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1}$$

$$T_2 = T_1 + \frac{T_{2s} - T_1}{\eta_c}$$

$$T_2 = 300 + \frac{300\left[\left(r_p\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{0.7042}$$

$$T_2 = 300 + 426\left(r_p^{2/7} - 1\right) \qquad [\because\ r = 1.4] \qquad \text{...(ii)}$$

Now, expansion process in turbine

$$\frac{T_3}{T_{4s}} = \left(r_p\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_{4s} = \frac{T_3}{\left(r_p\right)^{\frac{\gamma-1}{\gamma}}}$$

and $$\eta_T = \frac{T_3 - T_4}{T_3 - T_{4s}}$$

$$0.71 = \frac{T_3 - T_4}{T_3 - \frac{T_3}{\left(r_p\right)^{\frac{\gamma-1}{\gamma}}}}$$

$$T_3 - T_4 = 0.71(T_3)\left(1 - \frac{1}{\left(r_p\right)^{\frac{\gamma-1}{\gamma}}}\right) \qquad \text{...(iii)}$$

Now, from equation (i) and equation (iii), we get

$$T_2 - 300 = 0.9456(0.71)T_3\left(1 - \frac{1}{r_p^{2/7}}\right)$$

$$T_2 = 300 + 0.672796T_3\left(1 - \frac{1}{r_p^{2/7}}\right)$$

and, putting value of T_2 from equation (ii)

$$300 + 426\left(r_p^{2/7} - 1\right) = 300 + 0.672796T_3\left(1 - \frac{1}{r_p^{2/7}}\right)$$

$$T_3 = \frac{426}{0.672796}\left(r_p^{2/7}\right)$$

$$T_3 = 633.18\ r_p^{2/7} \quad \text{...(iv)}$$

Now, putting value of T_3 from equation (iv) and T_2 from equation (ii) in equation (A), we get

$$633.18\left(r_p^{2/7}\right) - \left[300 + 426\left(r_p^{2/7} - 1\right)\right] = 474$$

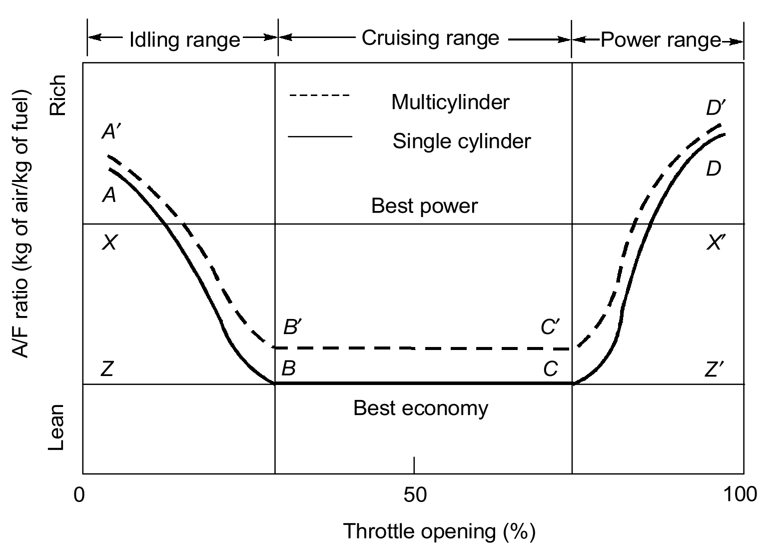
$$207.18 r_p^{2/7} = 348$$

$$r_p^{2/7} = 1.6797$$

$$r_p \simeq 6$$

5. (d)

(i) Rich mixture used during idling: During idling, engine runs without load and produces power only to overcome friction between parts. Rich mixture is required in this case to sustain combustion. During idling, pressure in the intake manifold is about 20 to 25 percent of atmospheric pressure. At suction stroke, inlet valve opens and the products of combustion get trapped in the clearance volume and expands in the inlet manifold. Later when piston moves downwards, the gases along with the fresh charges go into the cylinder. A very rich mixture must be supplied during idling to counteract the tendency of dilution and get combustible mixture.



(ii) Ignition timing should be advanced with increase in engine speed: Ignition timing needs to be increasingly advanced (relative to top dead centre, TDC) as the engine speed increases because of the following reasons:

1. To ensure that the air fuel mixture has the correct amount of time to burn completely. As the engine speed increases, the time available to burn the mixture decreases but the burning itself proceeds at the same speed, so it needs to be started increasingly earlier to complete in time.
2. The correct timing advance for a given engine speed will allow for maximum cylinder pressure to be achieved at correct crankshaft angular position.
3. Poor volumetric efficiency at high engine speed also requires increased advancement of ignition timing.

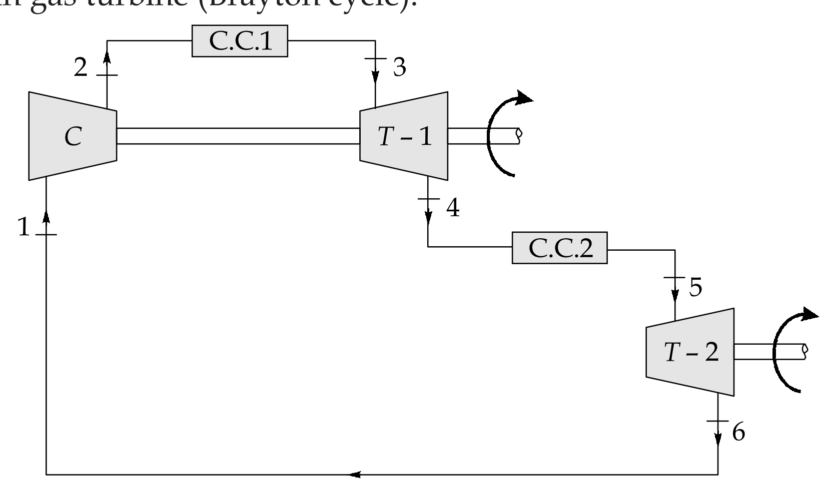
(iii) SI engines are generally not supercharged: Supercharging is forced induction in which the density of air charge is raised before it enters the cylinder, thus inducting the increased mass of air and then compressing it in each cylinder. The increased mass of the inducted charge into the cylinder raises both the temperature and density of the charge at the time of ignition. Moreover the supercharging increases the temperature and pressure of the inducted air. As a result, the temperature of the unburnt mixture in the cylinder might exceed the self-ignition temperature of the fuel and might remain at or above this temperature during the period of pre-flame reactions resulting in auto-ignition ,collision of flame fronts and knocking. Knocking may cause loss of engine power, damage to engine components and structure and might lead to complete engine failure.

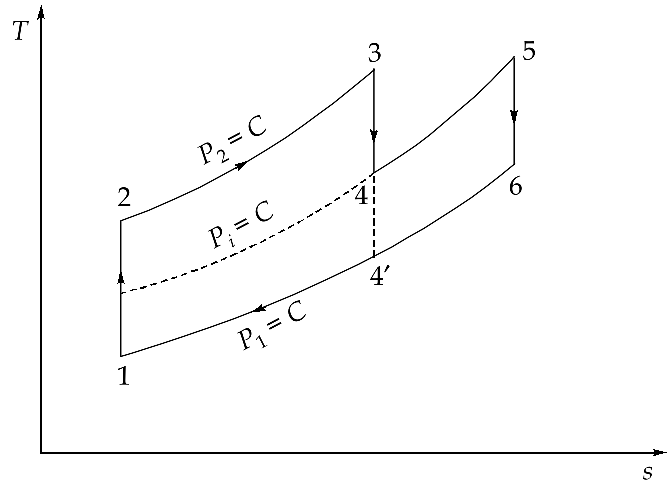
(iv) Two stroke engines find wide applications in marine propulsion because of the following reasons:

1. A two stroke engine provides a working stroke in every revolution. Hence a more uniform turning moment is obtained on the crankshaft. Therefore the mass of the flywheel used is lesser.
2. For the same power output, less space is occupied by a two stroke engine.

5. (e)

Reheating in gas turbine (Brayton cycle):



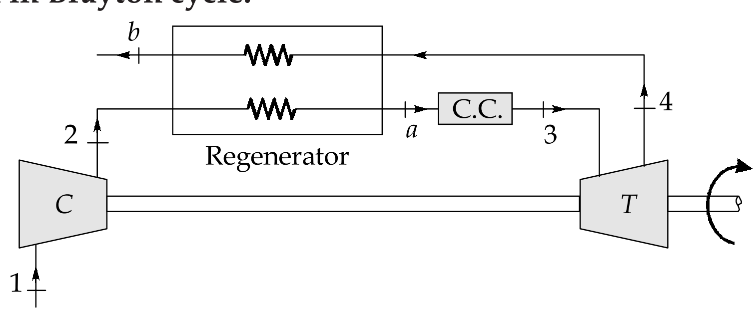


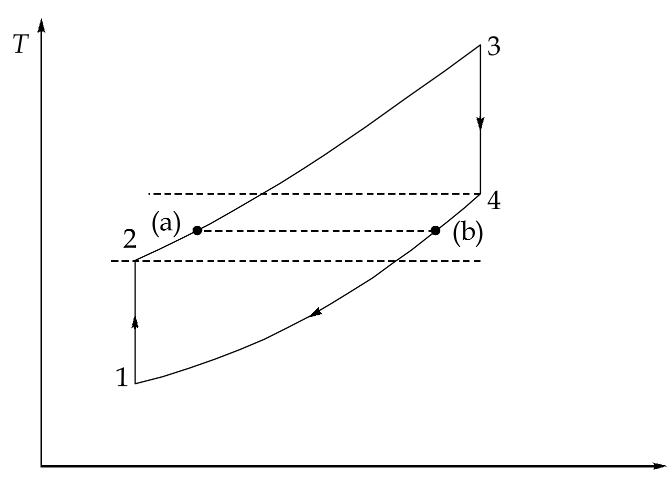
In reheat cycle the total expansion from higher pressure (P_2) to lower pressure (P_1) is done in more than one stages with reheating between the stages.

Effects of reheating:

1. Compressor work remains same while turbine work output increases, hence net work output increases.
2. Heat supplied and heat rejection both increases which causes increased mean temperature of heat addition and rejection hence, efficiency decreases.
3. Mean pressure ratio decreases.
4. Scope of regeneration increases because turbine exit temperature increases.

Regeneration in Brayton cycle:





It is the process in which the high temperature gas coming out from turbine is utilized to heat up the cold air coming out of compressor, before entering combustion chamber.

This preheating of cold air decreases the fuel required. Thus increases the efficiency.

Effects of regeneration:

1. Compressor work, turbine work output remain same, hence net work remain unchanged.
2. Heat supplied as well as heat rejection in the cycle decrease.
3. Mean temperature of the heat addition increases and mean temperature of heat rejection decreases, hence efficiency increases.
4. Specific fuel consumption decreases.

6. (a)

Given : P = 20 MW, H = 20 m, D_1 = 4.5 m, D_h = 2.0 m, N = 140 rpm, η_0 = 0.85, η_n = 0.94

(i)

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

$$0.85 = \frac{20 \times 10^6}{10^3 \times 9.81 \times Q \times 20}$$

$$\therefore \quad Q = 119.92 \text{ m}^3/\text{s} \qquad \text{Ans.}$$

(ii)

$$119.92 = \frac{\pi}{4}(4.5^2 - 2^2) \times V_{f_1}$$

$\therefore$ Velocity of flow,
$$V_{f_1} = \frac{119.92}{\frac{\pi}{4}(4.5^2 - 2^2)}$$

$$V_{f_1} = 9.4 \text{ m/s}$$

At the tip of the blade,

$$U_{1t} = \frac{\pi D_1 N}{60} = \frac{\pi \times 4.5 \times 140}{60}$$

$$U_{1t} = 32.99 \text{ m/s}$$

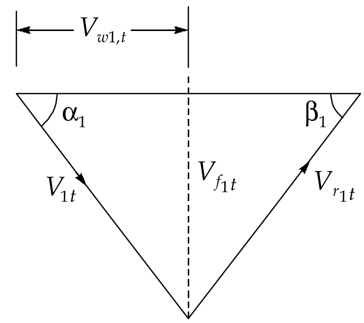
Hydraulic efficiency,
$$\eta_h = \frac{\left(V_{w_1} u_1\right)_{tip}}{gH}$$

$$\therefore \quad 0.94 = \frac{V_{w_{1t}} \times 32.99}{9.81 \times 20}$$

$$V_{w_{1t}} = 5.59 \text{ m/s}$$

Since, $V_{w_{1t}} < U_{1t}$, the inlet velocity triangle is an acute angled triangle, as shown in figure.

Inlet



From inlet velocity triangle,

$$\tan\alpha_{1t} = \left(\frac{V_{f_{1t}}}{V_{w_{1,t}}}\right) = \frac{9.4}{5.59}$$

$\therefore$ $\alpha_{1,t} = 59.26°$ **Ans.**

(iii) At mid radius,

$$D_m = \frac{D_1 + D_h}{2} = \frac{4.5+2}{2} = 3.25 \text{ m}$$

Tangential velocity, $U_{i,m} = \dfrac{\pi D_m N}{60} = \dfrac{\pi \times 3.25 \times 140}{60}$

$$U_{i,m} = 23.823 \text{ m/s}$$

Since the whirl velocity varies inversely with the radius,

$$\therefore \quad V_{w_{1,m}} = \frac{V_{w_{1,t}} \times U_{1,t}}{U_{1,m}} = \frac{5.59 \times 32.99}{23.823}$$

$$\therefore \quad V_{w_{1,m}} = 7.741 \text{ m/s}$$

From the velocity triangle at tip, considering appropriate suffix i.e. m = mid radius

$$\tan\alpha_{1,m} = \frac{V_{f_{1,m}}}{V_{w_{1,m}}} = \frac{9.4}{7.741} \qquad \left[\because (V_{f_1})_t = (V_{f_1})_m\right]$$

or $\tan\alpha_{1,m} = 1.214$

$\therefore$ $\alpha_{1,m} = 50.52°$ **Ans.**

Inlet blade angle at mid radius,

$$\tan\beta_{1,m} = \frac{(V_{f1})_m}{(U_{1m} - V_{w_1,m})} = \frac{9.4}{23.823 - 7.741}$$

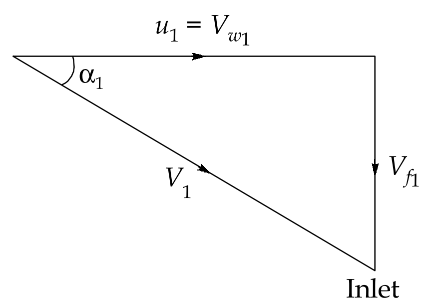
$$\tan\beta_{1,m} = 0.5845$$

$$\beta_{1,m} = 30.3°$$ Ans.

6. (b)

(i)

The velocity diagram is as shown below



$$\tan\alpha_1 = \frac{V_{f1}}{u_1} = \frac{V_{f1}}{V_{w1}}$$

$$\therefore \quad V_{w1} u_1 = u_1^2 = \frac{V_{f1}^2}{\tan^2\alpha_1}$$

Neglecting losses and assuming,

$$V_{f2} = V_{f1}$$

$$\text{Work input} = u_1 V_{w1} + \frac{V_{f1}^2}{2}$$

Thus, hydraulic efficiency is

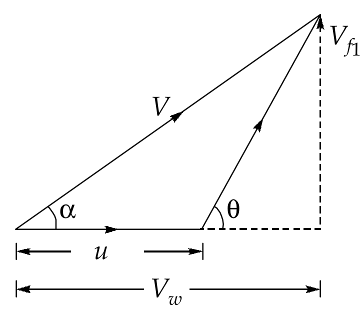
$$\eta_H = \frac{\dfrac{V_{f1}^2}{\tan^2\alpha_1}}{\dfrac{V_{f1}^2}{\tan^2\alpha_1} + \dfrac{V_{f1}^2}{2}}$$

Rearranging above equation,

$$\eta_H = \frac{2V_{f1}^2}{2V_{f1}^2 + V_{f1}^2\tan\alpha_1}$$

$$\eta_H = \frac{2}{2 + \tan^2\alpha_1}$$

(ii)



Inlet velocity triangle

We know, $$u = \frac{\pi DN}{60} = \frac{\pi \times 1.2 \times 450}{60} = 28.27 \text{ m/s}$$

From inlet velocity triangle, we have

$$V_f \cot\alpha = u + V_f \cot\theta$$

$$V_f \cot 20° = 28.27 + V_f \cot 60°$$

⇒ $$V_f = 13.03 \text{ m/s}$$

Hence, $$Q = 0.4 \times 13.03 = 5.212 \text{ m}^3\text{/s}$$

Work done by the runner per second is equal to the power developed which is given by,

$$P = \rho g Q \left(\frac{V_w u}{g}\right)$$

$$V_w = V_f \cot 20°$$

$$= 13.03 \times 2.7475$$

$$= 35.8 \text{ m/s}$$

∴ $$P = (9810 \times 5.212) \times \left(\frac{35.80 \times 28.27}{9.81}\right)$$

$$P = 5274.89 \times 10^3 \text{ W} \simeq 5275 \text{ kW}$$

Hydraulic efficiency, $$\eta = \frac{V_w u}{gH} = \frac{35.80 \times 28.27}{9.81 \times 120} = 0.86 (86\%)$$

6. (c)

Given : Hot water required = 6000 l/day, A = 120 m^2, I = 700 W/m^2, η_c = 0.6,

Cost = ₹8/kWh, Day length = 12 hour, η_{geyser} = 0.9

Total heat collected per second

$$= \eta_c \times A \times I = 0.6 \times 120 \times 700$$
$$= 50.4 \text{ kW}$$

Total heat collected in a day $= 50.4 \times 12 \times 3600$

$$= 2177.28 \times 10^3 \text{ kJ/day}$$
$$= \frac{2177.28 \times 10^3}{3600} \text{kWh/day} = 604.8 \text{ kWh/day}$$

Actual electricity consumption $= \frac{604.8}{0.9} = 672$ kWh/day

∴ Per day saving in electricity bill = 672 × 8 = ₹5376/-

Yearly saving in electricity = 5376 × 365

= ₹1962240/- per year **Ans.**

Now, heat received by solar collector $= mC_p\Delta T$

or $2177.28 \times 10^3 = 6000 \times 4.18 \times \Delta T$

∴ Temperature rise of water $= \frac{2177.28 \times 10^3}{6000 \times 4.18}$

$$\Delta T = 86.81°\text{C}$$ Ans.

7. (a)

Power produced without MPPT = 12 × 9 = 108 W

Maximum power production capability of the module

$$= V_{max} \times I_{max} = 6 \times 27$$
$$= 162 \text{ W}$$

∴ As the efficiency of the MPPT is 95%

Actual power produced with MPPT = 162 × 0.95 = 153.9 W

Surplus power produced by use of MPPT = 153.9 – 108 = 45.9 W

Surplus energy produce in 't' hours $= \frac{45.9 \times t}{1000}$ kWh

∴ Cost of surplus energy $= 3 \times 0.0459 \times t$

$$= 0.1377t$$

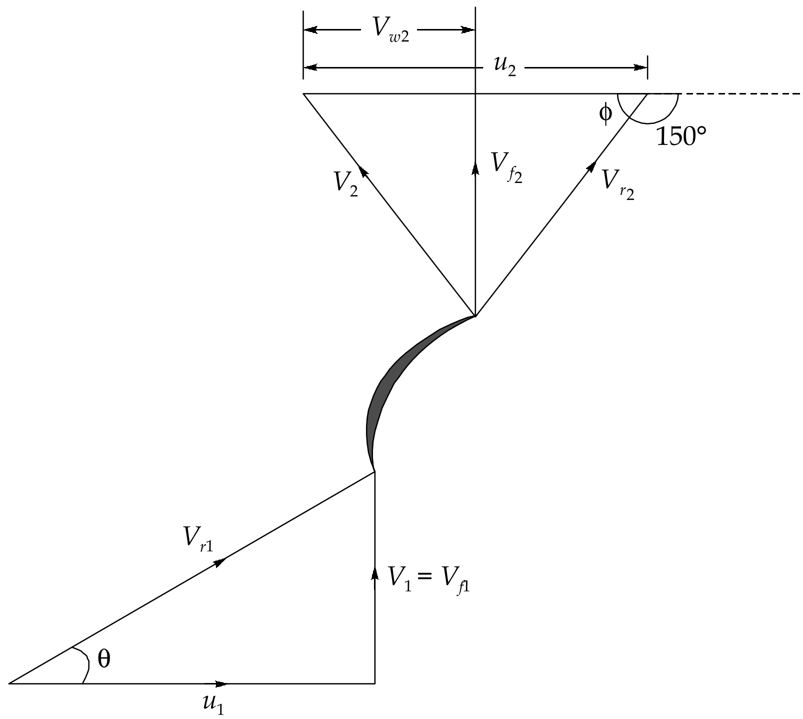
Cost of MPPT = ₹4200

Time (in hours) required to recover the cost of MPPT,

$$t = \frac{4200}{0.01377} = 30501.09 \text{ hours}$$

7. (b)

(i)



(ii)

Lift capacity, Q = 8000 litres/sec = 8 m^3/s

Manometric head, H_m = 6 m

N = 500 rpm

$$V_{f_1} = V_{f_2} = 2 \text{ m/s}$$

Manometric efficiency, $\eta_m = 0.6$

Angle of vane tip with the direction of motion = 150°

$$\phi = 180° - 150° = 30°$$

$$\eta_m = \frac{gH_m}{V_{w_2}u_2} \Rightarrow \frac{9.81 \times 6}{V_{w_2}u_2} = 0.6$$

$\therefore$ $$V_{w_2}u_2 = 98.1 \text{ m}^2/\text{s}^2 \quad \text{...(i)}$$

From outlet velocity triangle,

$$\tan\phi = \frac{V_{f_2}}{u_2 - V_{w_2}} = \frac{2}{u_2 - V_{w_2}}$$

$$u_2 - V_{w_2} = \frac{2}{\tan 30°} = 3.464 \text{ m/s} \quad \text{...(ii)}$$

On solving (i) and (ii), we get

$$V_{w_2} u_2 = u_2(u_2 - 3.464) = 98.10$$

$$u_2^2 - 3.464u_2 - 98.10 = 0$$

On solving, $u_2 = 11.7868$ m/s

Also, $u_2 = \frac{\pi D_2 N}{60}$

$$11.7868 = \frac{\pi D_2 \times 500}{60}$$

$$D_2 = 0.449 \text{ m} \simeq 0.45 \text{ m}$$

$$Q = \pi D_2 b_2 V_{f2}$$

$$8 = \pi \times 0.45 \times B_2 \times 2$$

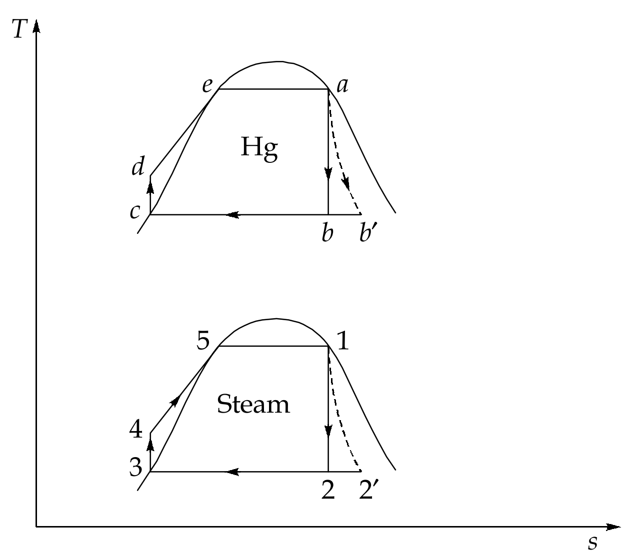
$$B_2 = 2.83 \text{ m}$$

∴ Width of impeller is 2.83 m and its diameter is 0.45 m.

7. (c)

$P_a = 10$ bar, $P_b = P_{b'} = P_c = 0.1$ bar, $P_1 = 40$ bar, $P_2 = P_{2'} = P_3 = 0.06$ bar

$\mu_m = 0.75$, $\eta_s = 0.80$



Cycle $a - b' - c - d - e$ = Mercury cycle

Cycle 1 – 2′ – 3 – 4 – 5 = Steam cycle

For mercury cycle,

$$s_a = s_b$$
$$0.5158 = 0.089 + x(0.6604 - 0.089)$$
$$x = 0.747$$
$$(h_b)_{0.1\text{ bar}} = h_f + x(h_g + h_f)$$
$$h_b = 34.485 + 0.747\,(332.975 - 34.485)$$
$$h_b = 257.457 \text{ kJ/kg}$$

Isentropic work by mercury turbine = $h_a - h_b$ = 362.406 – 257.457

$$= 104.95 \text{ kJ/kg}$$

Actual work done by mercury turbine = $h_a - h_{b'}$

$$W_T = 104.95 \times 0.75$$
$$= 78.7125 \text{ kJ/kg}$$
$$h_{b'} = h_a - 78.7125$$
$$= 362.406 - 78.7125$$
$$= 283.6935 \text{ kJ/kg}$$

Neglecting the pump work,

Heat supplied in the mercury power cycle,

$$Q_s = h_a - h_d$$
$$= h_a - h_d \qquad [\because \; h_d = h_c]$$
$$= 362.406 - 34.485$$
$$= 327.921 \text{ kJ/kg}$$

Heat rejected in the mercury power cycle,

$$Q_R = Q_s - W_T$$
$$= 327.921 - 78.7125$$
$$= 249.2085 \text{ kJ/kg}$$

For steam power cycle,

$$s_2 = s_1$$
$$0.521 + x(8.331 - 0.521) = 6.070$$
$$x = 0.71$$

$$(h_2)_{0.06\text{ bar}} = h_f + x(h_g - h_f)$$
$$= 151.5 + 0.71(2567 - 151.5)$$
$$= 1866.505 \text{ kJ/kg}$$

Isentropic turbine work $= h_1 - h_2$
$$= 2801 - 1866.505$$
$$= 934.495 \text{ kJ/kg}$$

Actual turbine work of steam power cycle,
$$W_T = 934.495 \times 0.80$$
$$= 747.596 \text{ kJ/kg}$$

Neglecting pump work,

Heat supplied in the steam power cycle,
$$Q_s = h_1 - h_4$$
$$= h_1 - h_3 \qquad [\because\ h_3 = h_4]$$
$$= 2801 - 151.5$$
$$= 2649.5 \text{ kJ/kg}$$

(1) By energy conservation,

Heat rejection by mercury power cycle in condenser = Heat received by feed water in steam power cycle

$$\therefore \qquad \dot{m}_m(h_{b'} - h_c) = 1 \times (h_1 - h_3)$$
$$\dot{m}_m = \frac{2801 - 151.5}{249.2085}$$
$$= 10.6332 \text{ kg of mercury per kg of steam}$$

(2) Thermal efficiency of cycle $= \dfrac{\text{Net work done}}{\text{Total heat supplied in mercury power cycle}}$

$$= \frac{78.7125 \times 10.6332 + 747.596}{10.6316 \times 327.921}$$
$$= 0.4544 = 45.44\%$$

(3) Work done by the mercury turbine
$$= 10.6332 \times 78.7125$$
$$= 836.97 \text{ kJ/kg of steam}$$

Work done by steam turbine = 747.596 kJ/kg of steam

8. (a)

(i)

Given : U_0 = 10 m/s, P = 1 atm = 1.01325 × 10^5 Pa, T = 15 + 273 = 288 K,

R = 60 m, N = 40 rpm

$$\omega = \frac{2\pi N}{60} = \frac{2\pi \times 40}{60} = 4.1887 \text{ rad/s}$$

$$\rho = \frac{P}{RT_1} = \frac{1.01325 \times 10^5}{287 \times 288} = 1.2258 \text{ kg/m}^3$$

$$\text{Area of rotor} = \pi R^2 = \pi \times (60)^2 = 11309.7335 \text{ m}^2$$

$$\text{Tip speed ratio, } \lambda = \frac{\omega R}{U_0} = \frac{4.1887 \times 60}{10} = 25.1322$$

$$\text{Total power available} = \frac{1}{2}\rho A U_0^3$$

$$= \frac{1}{2} \times 1.2258 \times 11309.7335 \times 10^3$$

$$P_{total} = 6.9317 \text{ MW}$$

$$T_{max} = P_{total} \times \frac{\lambda}{\omega} = 6.9317 \times \frac{25.1322}{4.1887}$$

$$T_{max} = 41.59 \text{ MNm}$$

Maximum torque coefficient,

$$C_{T,\,max} = \frac{C_{P_{max}}}{\lambda} = \frac{0.593}{25.1322}$$

$$C_{T,\,max} = 0.02359$$

Torque produced at the shaft,

$$T_{shaft} = C_{T_{max}} \times T_{max}$$

$$= 0.02359 \times 41.59$$

$$= 0.9811 \text{ MNm}$$ **Ans.**

Alternatively,

$$P_{max} = T_{shaft} \times \omega$$

or $$C_{P_{max}} \times P_{total} = T_{shaft} \times \omega$$

$$0.593 \times 6.9317 = T_{shaft} \times 4.1887$$

$$T_{shaft} = 0.9813 \text{ MNm}$$ **Ans.**

(ii)

The operation of a biogas plant or digestion process is affected by following factors :

1. **Temperature :** Methane forming bacteria work best in temperature range of 20 - 55°C. Digestion at higher temperature proceeds more rapidly than at lower temperatures with gas yield rates doubling at about every 5°C increase in temperature. Gas production decreases rapidly below 20°C and almost stops at 10°C.
2. **Pressure :** A minimum pressure of 6 - 10 cm of Hg column, i.e. 1.2 bar (absolute) is considered ideal for proper functioning of the plant. Excess pressure inhibits release of gas from slurry. It also leads to leakage in masonry through microspores.
3. **Solid to moisture ratio in the biomass :** Water is essential for survival and activity of microorganisms, hydrolysis process and activity of extracellular enzymes. It helps in better mixing of various constituents of the biomass, movement of bacteria and faster digestion rate. The optimum total solid concentration is 7 to 9%.
4. **pH value :** In the initial acid-forming stage of the digestion process, pH value may be around 6 or less. However, during the methane-formation stage, a pH value of 6.5 to 7.5 is maintained.
5. **Feeding rate :** A uniform feeding rate should be maintained because at higher feeding rate, the retention period will be less and undigested slurry may come out. Also if the digester is fed with too much raw material, at a time, acid will accumulate and the digestion process may stop.
6. **Carbon to nitrogen ratio (C/N) :** Carbon (in carbohydrate) and nitrogen (in proteins, nitrate etc.) are the main nutrients for anaerobic bacteria. Carbon supplies energy and nitrogen is needed for building up cell structure. The optimum C : N ratio for maximum microbiological activity is 30 : 1.
7. **Mixing or stirring :** Mixing has three important effects :
   (i) Maintains uniformity in substrate concentration temperature and other environmental factors.
   (ii) Minimizes the formation of scum at the surface.
   (iii) Prevents the deposition of solids at the bottom.

   Hence stirring brings the masses floating on top and depositing at bottom to the reach of bacteria.
8. **Seeding of biomass with bacteria :** To start and accelerate the fermentation process, a small amount of digested slurry, containing methane forming bacteria is added to the freshly charged plant. This is known as seeding. Seeding helps to accelerate the starting of the digestion process.

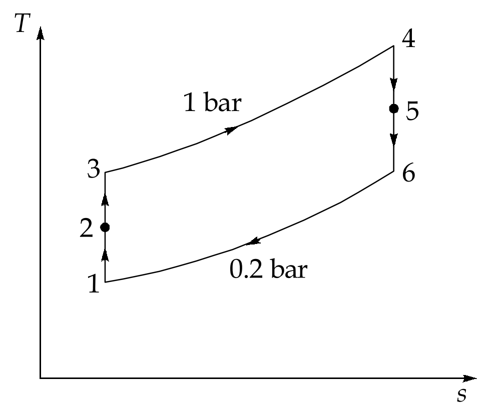
9. **Retention time :** Retention time is the time duration for which the slurry remains in the plant or the time that is available for biodigestion. It is determined by the volume of a digester divided by the volume of slurry added per day. Retention time is optimized to achieve a 70 - 80% complete digestion.

10. **Effect of Toxic substances :** High concentration of ammonia, antibiotics, pesticides, detergents, heavy metals like chromium, copper, nickel and zinc are toxic to bacteria responsible for biodigestion.

8. (b)

Given : V_1 = 268 m/s; $\eta_{comb} = \eta_c = \eta_T$ = 100%; T_1 = 220 K; C_{P_a} = 1.005 kJ/kgK; γ = 1.4;

T_4 = 1350 K; C_{P_g} = 1.102 kJ/kgK; n = 1.33



Process 1 - 2 (diffuser)

$$T_2 = T_1 + \frac{V_1^2}{2C_P} \qquad [h_{01} = h_{02} \text{ and } V_2 = 0]$$

$$T_2 = 220 + \frac{(268)^2}{2(1.005)\times 10^3}$$

$$T_2 = 255.733 \text{ K}$$

and

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

$$P_2 = 0.2\left(\frac{255.733}{220}\right)^{\frac{1.4}{0.4}} = 0.3387 \text{ bar}$$

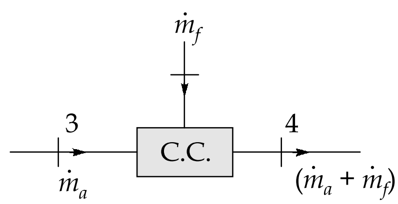
Process 2 - 3 (compression):

$$\frac{T_3}{T_2} = \left(\frac{P_3}{P_2}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_3 = 255.733\left(\frac{1}{0.3387}\right)^{\frac{0.4}{1.4}}$$

$$= 348.44 \text{ K}$$

Process 3 - 4 (combustion) :



$$\dot{m}_a C_P T_3 + \dot{m}_f CV \times \eta_{comb} = \left(\dot{m}_a + \dot{m}_f\right) C_{P_g} T_4$$

$$\dot{m}_f (43000 - 1.102 \times 1350) = \dot{m}_a (1.102 \times 1350 - 1.005 \times 348.44)$$

$$\dot{m}_f = \dot{m}_a \left(\frac{1137.518}{41512.3}\right)$$

$$\text{AFR} = \frac{\dot{m}_a}{\dot{m}_f} = \frac{1}{0.0274} = 36.4964$$

We know, $W_C = W_T$

$$\dot{m}_a C_P \left(T_3 - T_2\right) = \left(\dot{m}_a + \dot{m}_f\right) C_{P_g} \left(T_4 - T_5\right)$$

$$1.005(348.44 - 255.733) = (1 + 0.0274) \times 1.102 \times (1350 - T_5)$$

$$T_5 = 1267.708 \text{ K}$$

and

$$\frac{P_5}{P_4} = \left(\frac{T_5}{T_4}\right)^{\frac{n}{n-1}}$$

$$P_5 = 1 \times \left(\frac{1267.708}{1350}\right)^{\frac{1.33}{0.33}} = 0.7761 \text{ bar}$$

Now,

$$\frac{T_6}{T_5} = \left(\frac{P_6}{P_5}\right)^{\frac{n-1}{n}}$$

$$T_6 = 1267.708\left(\frac{0.2}{0.7761}\right)^{\frac{0.33}{1.33}}$$

$$= 905.533 \text{ K}$$

Exit velocity, $T_5 = T_6 + \frac{V_e^2}{2C_P}$

$$V_e = \sqrt{2 \times 1102(1267.708 - 905.533)}$$

$$V_e = 893.44 \text{ m/s}$$

and, thrust power $(TP) = \left\{\left(\dot{m}_a + \dot{m}_f\right) \times V_e - \dot{m}_a V_1\right\} V_1$

$$\frac{TP}{\dot{m}_a} = \{(1 + 0.0274) \times 893.44 - 268\} \times 268$$

$$\frac{TP}{\dot{m}_a} = 174178.63 \text{ J/kg}$$

$\therefore$ Specific thrust power = 174.178 kJ/kg

Now, Propulsive power $(PP) = \frac{1}{2}\dot{m}_a\left\{\left(1 + \frac{1}{AFR}\right)V_e^2 - V_1^2\right\}$

$$\frac{PP}{\dot{m}_a} = \frac{1}{2}\left\{(1 + 0.0274)(893.44)^2 - 268^2\right\}$$

$\therefore$ Specific propulsive power = 374.1413 kJ/kg

hence, $\eta_{\text{prop}} = \frac{TP/\dot{m}_a}{PP/\dot{m}_a}$

$$= \frac{174.178}{374.1413} = 0.46554$$

$$= 46.554\%$$

8. (c)

(i)

Specific speed for turbine is defined as the speed in rpm of a turbine geometrically similar to the actual turbine but of such a size that under corresponding conditions it will develop 1 kW power when working under unit head.

It is given as,

$$N_s = \frac{N\sqrt{P}}{H^{5/4}}$$

where, P = Power in kW, H = Head available in m, N = Speed of rotation in RPM.

(ii) Specific speed

It is given by,
$$N_s = \frac{N\sqrt{P}}{H^{5/4}} = \frac{120\sqrt{11000}}{(100)^{5/4}} = 39.8$$

(iii)

$$\eta_0 = \frac{P}{\rho_g QH}$$

$$0.88 = \frac{11\times10^6}{9810\times Q\times100}$$

$$Q = \frac{11\times10^6}{9810\times0.88\times100} = 12.742 \text{ m}^3/\text{s}$$

(iv)

Torque developed, $T = \rho Qr(V_{u_1} - V_{u_2})$

where, V_{u_1} = swirl velocity at inlet, r = radius of runner, $V_{u_2} = 0$ (for maximum efficiency)

$\therefore$
$$T = \frac{\rho QDV_{u_1}}{2}$$

Also,
$$P = \rho QDV_{u_1}$$

where,
$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi\times5\times120}{60} = 31.42 \text{ m/s}$$

$\Rightarrow$
$$11\times10^6 = 1000\times12.742\times31.42\, V_{u1}$$

$$V_{u1} = 27.48 \text{ m/s}$$

Thus,
$$T = 875.352 \text{ kN.m}$$

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