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Detailed Solutions

ESE-2023 Mains Test Series

Mechanical Engineering Test No : 8

Section A : Machine Design + Mechatronics & Robotics **[All Topics]**

Section B : IC Engine **[All Topics]**

Renewable Sources of Energy-2 + Industrial and Maintenance Engg.-2 **[Part syllabus]**

Section : A

1. (a)

Given : m = 1800 kg; V = 2 m/s; δ = 170 mm; c = 8, G = 82375 MPa; τ = 200 MPa

Wire diameter

The kinetic energy of the moving wagon is absorbed by the springs.

$$\text{K.E. of wagon} = \frac{1}{2}mv^2 = \frac{1}{2}\times 1800\times 2^2$$

$$= 3600 \text{ N-m} = 3600 \times 10^3 \text{ N-mm}$$

Suppose P is the maximum force acting on each spring and causing it to compress by 170 mm.

The strain energy absorbed by two springs.

$$E = \frac{1}{2}\times 2\times P\times \delta = 170P \text{ N-mm}$$

The strain energy absorbed by two springs is equal to the kinetic energy of the wagon.

$$170P = 3600 \times 10^3$$

or, $$P = 21176.4705 \text{ N}$$

$$K = \frac{4c-1}{4c-4}+\frac{0.615}{c} = \frac{4\times 8-1}{4\times 8-4}+\frac{0.615}{8} = 1.184$$

We know, $$\tau = k\left(\frac{8Pc}{\pi d^2}\right)$$

$$200 = 1.184\left(\frac{8 \times 21176.4705 \times 8}{\pi \times d^2}\right)$$

or, $$d = 50.5362 \text{ or } 51 \text{ mm}$$ Ans.

Mean coil diameter, $D = cd = 8 \times 51 = 408$ mm Ans.

Number of active coils,

We know, $$\delta = \frac{8PD^3N}{Gd^4}$$

$$170 = \frac{8 \times 21176.4705 \times (408)^3 N}{(82375) \times (51)^4}$$

or $$N = 8.2338 \text{ or } 9 \text{ coils}$$ Ans.

Actual spring rate, $$k = \frac{Gd^4}{8D^3N} = \frac{82375 \times (51)^4}{8 \times (408)^3 \times 9}$$

$$k = 113.9628 \text{ N/mm}$$ Ans.

1. (b)

In practice, reduction of stress concentration is achieved by following methods:

(i) Additional notches and holes in tension member : A flat plate with a V-notch subjected to tensile force shown in figure (a). It is observed that a single notch results in a high degree of stress concentration. The severity of stress concentration is reduced by three methods:

(a) use of multiple notches;

(b) drilling additional holes; and

(c) removal of undesired material.

These methods are illustrated in figure (b), (c) and (d) respectively. The method of removing undesired material is called the principle of minimization of the material. In these three methods, the sharp bending of force flow line is reduced and it follows a smooth curve.

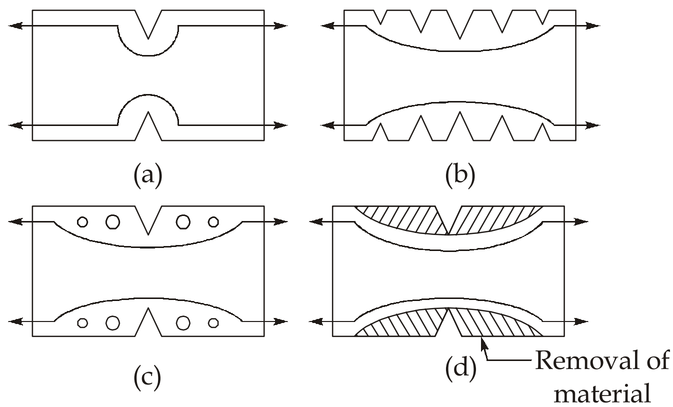


Figure: Reduction of stress concentration due to V-notch
(a) original (b) multiple notches (c) drilled holes
(d) removal of undesirable material

(ii) Fillet radius, undercutting and notch for member in bending : A bar of circular cross-section with a shoulder and subjected to bending moment is shown in figure (a). Ball bearings, gears or pulleys are seated against this shoulder. The shoulder creates a change in cross-section of the shaft, which results in stress concentration. There are three methods to reduce stress concentration at the base of this shoulder. Figure (b) shows the shoulder with a fillet radius *r*. This results in gradual transition from small diameter to larger diameter. The fillet radius should be as large as possible in order to reduce stress concentration. In practice, fillet radius is limited by the design of mating components. The fillet radius can be increased by undercutting the shoulder as illustrated in figure (c). A notch results in stress concentration. Surprisingly, cutting an additional notch is an effective way to reduce stress concentration. This is illustrated in figure (d).

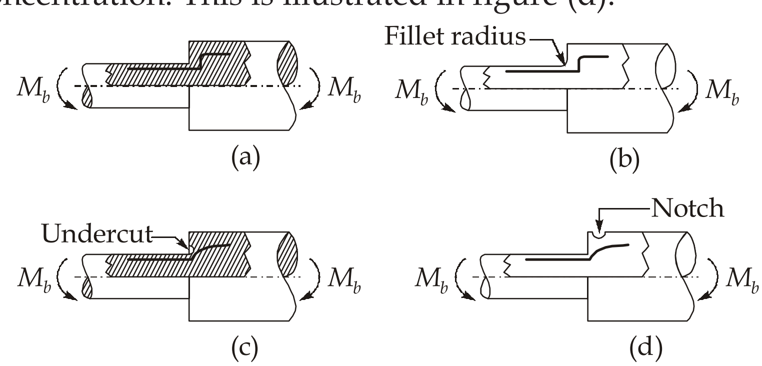


Figure: Reduction of stress concentration due to abrupt change in cross-section
(a) original component (b) fillet radius (c) undercutting (d) addtion of notch

(iii) Drilling additional holes for shaft : A transmission shaft with a keyway is shown in figure (a). The keyway is a discontinuity and results in stress concentration at the corners of the keyway and reduces torsional shear strength. An empirical relationship developed by H. F. Moore for the ratio 'C' of torsional strength of shaft having a keyway to torsional strength of same sized shaft without keyway is given by

$$C = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)$$

where w and h are width and height dimensions of the keyway respectively and d is the shaft diameter. The four corners of the keyway viz. m_1, m_2, n_1 and n_2 are shown in figure (c). It has been observed that torsional shear stresses at two points viz. m_1 and m_2 are negligibly small in practice and theoretically equal to zero. On the other hand, the torsional shear stresses at two points viz. n_1 and n_2 are excessive and theoretically infinite which means even a small torque will produce permanent set at these points. Rounding corners at two points viz. n_1 and n_2 by means of fillet radius can reduce the stress concentration. A stress concentration factor $K_t = 3$ should be used when a shaft with keyway is subjected to combined bending and torsional moments.

In addition to giving fillet radius at the inner corners of keyway, there is another method of drilling two symmetrically holes on the sides of keyway. These holes press the force flow lines and minimize their bending in the vicinity of the keyway. This method is illustrated in figure (b).

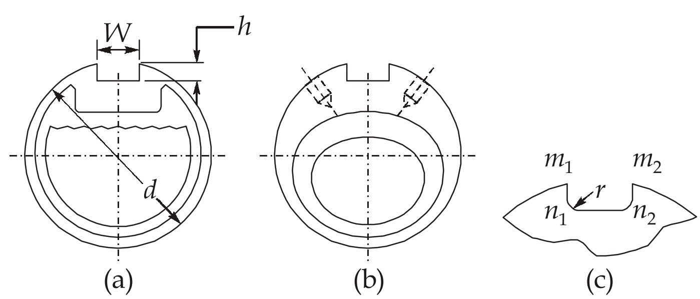


Figure: Reduction of stress concentration in shaft with keyway
(a) original shaft (b) drilled holes (c) fillet radius

1. (c)

Given: $P = 12$ kW, $N = 1000$ rpm, $D = 2.5$d, $\mu = 0.15$, $p = 0.4$ MPa.

Number of steel plates, $(N_1) = 6$, Number of bronze plate $(N_2) = 5$

Torque transmitting capacity, $M_t = \dfrac{60 \times 10^6 \times P}{2\pi N} = \dfrac{60 \times 10^6 \times 12}{2\pi \times 1000} = 114591.559$ N – mm

Force required to engage clutch,

$$F = \frac{\pi p d}{2}(D - d) = \frac{\pi p d}{2}(2.5d - d)$$

$$= \frac{1.5 \times \pi \times p \times d^2}{2} = 0.75\pi p d^2$$

Number of effective surfaces,

$$N = N_1 + N_2 - 1 = 6 + 5 - 1 = 10$$

So, from uniform wear theory, $M_t = \frac{\mu F N}{4}(D + d)$

Putting the value of F.

Or,
$$M_t = \frac{\mu \times 0.75\pi p d^2 \times 10}{4}(2.5d + d)$$

$$114591.559 = \frac{0.15 \times 0.75 \times \pi \times 3.5 \times 0.4 \times 10 \times d^3}{4}$$

$$d = 45.2474 \text{ mm}$$ **Ans.**

$$D = 2.5 \times 45.2474 = 113.1186 \text{ mm}$$ **Ans.**

1. (d)

Given: $B = \begin{bmatrix} 0.707 & ? & 0 & 2 \\ ? & 0 & 1 & 4 \\ ? & -0.707 & 0 & 5 \\ 0 & 0 & 0 & 1 \end{bmatrix}$

From
$$\vec{n} \times \vec{o} = \vec{a}$$

$$\begin{bmatrix} i & j & k \\ 0.707 & n_y & n_z \\ o_x & 0 & -0.707 \end{bmatrix} = J$$

And $i\,(-0.707\,n_y) - j\,(-0.5 - n_z o_x) + k\,(-n_y o_x) = J$

$\therefore$
$$n_y = 0$$

From length equations: $|n| = 1$

or,
$$0.707^2 + n_y^2 + n_z^2 = 1 \Rightarrow n_z = \pm 0.707$$
$$o_x^2 + 0.707^2 = 1 \Rightarrow o_x = \pm 0.707$$

Therefore, there are two possible acceptable solutions.

$$B = \begin{bmatrix} 0.707 & 0.707 & 0 & 2 \\ 0 & 0 & 1 & 4 \\ 0.707 & -0.707 & 0 & 5 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

and
$$B = \begin{bmatrix} 0.707 & -0.707 & 0 & 2 \\ 0 & 0 & 1 & 4 \\ -0.707 & -0.707 & 0 & 5 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

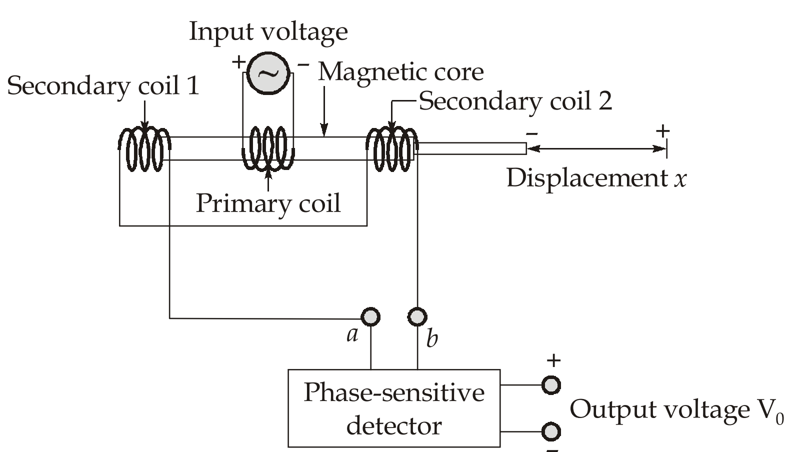
1. (e)

Linear variable differential transformer (LVDT)

The LVDT is a rugged electromagnetic transducer used to measure linear displacement. LVDT consists of an iron core which can move freely within a primary or power coil and two secondary coils as shown in figure. A movable magnetic core provides a variable coupling between windings. The secondary coils are connected in series configuration and are equally positioned with respect to the primary coil.

When the core is centrally located, the emfs generated in the secondary coils are equal and opposite and the net output voltage is zero. When the core is moved to one side, the voltage in the primary coil becomes larger and that in the other secondary coil becomes smaller. The magnitude of output voltage is proportional to the displacement of the core from the null position. The phase-sensitive detector converts the AC secondary voltage into a DC voltage, V_0. The magnitude of the DC voltage is proportional to the amplitude of the AC voltage.

At mid-position of the core, the induced voltage in each coil is of the same amplitude and 180° out of phase, producing a zero or null output. As the core moves from the null position, the output amplitude increases a proportional amount over a linear range around the null as shown in figure.



Linear variable differential transformer

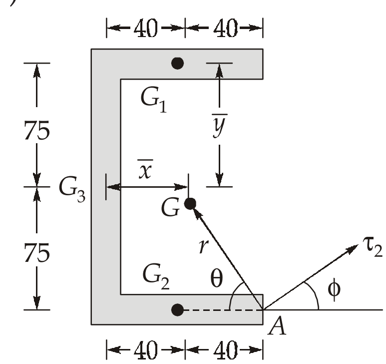
Advantages of LVDT

1. It has a high range, upto 1 m.
2. Friction is less and hence less is the wear problem.
3. It has low hysteresis.
4. The power consumption is less.
5. It has infinite resolution.
6. The output is highly linear and accurate.

2. (a)

Given: Eccentric load (P) = 80 kN

Permissible shear stress (τ) = 150 MPa



[All dimensions are in mm]

Primary shear stress:

By symmetry, the centre of gravity G of three welds is midway between two horizontal welds. Therefore,

$$\overline{y} = 75 \text{ mm}$$

Taking moment of three welds about the vertical line passing through G_3,

$$(80+150+80)\overline{x} = 80 \times 40 + 80 \times 40 + 150 \times 0$$

or, $$\overline{x} = 20.6451 \text{ mm}$$

The area of three welds are as follows:

$$A_1 = 80 \times t \text{ mm}^2$$
$$A_2 = 80 \times t \text{ mm}^2$$
$$A_3 = 150 \times t \text{ mm}^2$$
$$A = A_1 + A_2 + A_3 = 310t \text{ mm}^2$$

The primary shear stress in the weld is given by,

$$\tau_1 = \frac{P}{A} = \frac{80000}{310t} = \left(\frac{258.0645}{t}\right) \text{MPa}$$

Secondary shear stress

From figure, A is the farthest point from the centre of gravity G and its distance r is given by,

$$r = \sqrt{(80-20.6451)^2 + (75)^2} = 95.6451 \text{ mm}$$

Also,
$$\tan\theta = \frac{75}{(80-20.6451)} \text{ or } \theta = 51.6419°$$

$$\phi = 90 - \theta = 38.358°$$

Therefore, the secondary shear stress is inclined at 38.358° with horizontal.

$$e = (80 - 20.6451) + 200 = 259.3549 \text{ mm}$$

$$M = P \times e = 80 \times 10^3 \times 259.3549$$

or,
$$M = 20748.392 \times 10^3 \text{ N-mm}$$

G_1, G_2 and G_3 are the centre of gravity of three welds and their distance from common centre of gravity G are as follows

$$\overline{G_1G} = \overline{G_2G} = \sqrt{(40-20.6451)^2 + 75^2} = 77.4571 \text{ mm}$$

or,
$$r_1 = r_2 = 77.4571 \text{ mm}$$

$$r_3 = \overline{G_3G} = \bar{x} = 20.6451 \text{ mm}$$

Moment of inertia of the welds about G.

$$J_1 = J_2 = A_1\left[\frac{(80)^2}{12} + (77.4571)^2\right]$$

$$= (80t)\left[\frac{(80)^2}{12} + (77.4571)^2\right]$$

$$J_1 = J_2 = 522634.8539t \text{ mm}^4$$

$$J_3 = A_3\left[\frac{(150)^2}{12} + (20.6451)^2\right]$$

$$= (150t)\left[\frac{(150)^2}{12} + (20.6451)^2\right] = 345183.0231t \text{ mm}^4$$

$$J = 2J_1 + J_3 = 1390452.731 \text{ mm}^4$$

The secondary shear stress at point A is given by,

$$\tau_2 = \frac{Mr}{J}$$

$$\tau_2 = \frac{20748.392 \times 10^3 \times 95.6451}{1390452.731}$$

or $$\tau_2 = \left(\frac{1427.22}{t}\right)\text{MPa}$$

Resultant shear stress

The secondary shear stress is inclined at 38.358° with horizontal, it is resolved into vertical and horizontal components.

$$\text{Vertical component} = \tau_2 \sin\phi$$

$$= \left(\frac{1427.22}{t}\right)\sin 38.358° = \left(\frac{885.6944}{t}\right)\text{MPa}$$

$$\text{Horizontal component} = \tau_2 \cos\phi$$

$$= \left(\frac{1427.22}{t}\right)\cos 38.358° = \left(\frac{1119.1525}{t}\right)\text{MPa}$$

The primary shear stress is vertically upwards. Therefore, the total vertical component is given by,

$$\frac{885.6944}{t} + \frac{258.0645}{t} = \frac{1143.7589}{t}\text{MPa}$$

The resultant shear stress is given by

$$\tau = \sqrt{\left(\frac{1143.7589}{t}\right)^2 + \left(\frac{1119.1525}{t}\right)^2}$$

$$\tau = \frac{1600.2145}{t}\text{MPa}$$

Size of weld

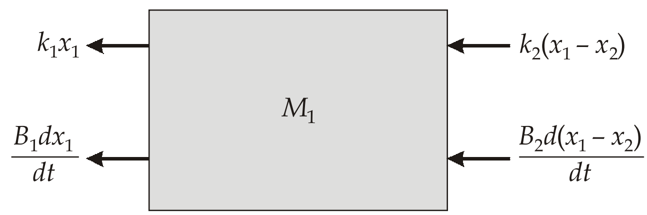
$$150 = \frac{1600.2145}{t}$$

or $$t = 10.668 \text{ mm}$$

and $$h = \frac{t}{0.707} = 15.0892 \text{ mm}$$

2. (b)

From a free body diagram, spring forces and damping force can be determined. Analyzing the free body diagram for mass M_1,



From Newton's second law of motion,

$$M_1\frac{d^2x_1}{dt^2} = -k_1x_1 - B_1\frac{dx_1}{dt} - k_2(x_1-x_2) - B_2\frac{d(x_1-x_2)}{dt}$$

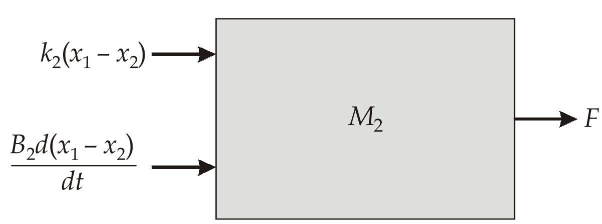
or, $M_1\frac{d^2x_1}{dt} + (B_1+B_2)\frac{dx_1}{dt} + (k_1+k_2)x_1 = B_2\frac{dx_2}{dt} + k_2x_2$

Taking Laplace transform, we get

$[M_1s^2 + (B_1 + B_2)s + (k_1 + k_2)]\, X_1(s) = [B_2s + k_2]X_2(s)$ ... (i)

Free body diagram for mass M_2, is shown below:

From Newtonian second law, we get



$$M_2\frac{d^2x_2}{dt^2} = k_2(x_1-x_2) + B_2\frac{d(x_1-x_2)}{dt} + F$$

Taking Laplace transform, we get

$$[M_2s^2 + B_2s + k_2]\, X_2(s) = [k_2 + B_2s]X_1(s) + F(s) \quad \text{... (ii)}$$

Substituting the value of $X_2(s)$ from equation (i) in equation (ii), we can relate $X_1(s)$ and $F(s)$ as

$$\left[\frac{\{M_1s^2 + (B_1+B_2)s + k_1 + k_2\}(M_2s^2 + B_2s + k_2) - (k_2 + B_2s)^2}{k_2 + B_2s}\right]X_1(s) = F(s)$$ **Ans.**

2. (c)

The three link transformation matrices ${}^0T_1$, ${}^1T_2$, ${}^2T_3$ and the overall arm transformation matrix ${}^0T_3$ are obtained as

$$ {}^0T_1(\theta_1) = \begin{bmatrix} C_1 & 0 & S_1 & 0 \\ S_1 & 0 & -C_1 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} $$

$$ {}^1T_2(\theta_2) = \begin{bmatrix} C_2 & 0 & -S_2 & 0 \\ S_2 & 0 & C_2 & 0 \\ 0 & -1 & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} $$

$$ {}^2T_3(\theta_3) = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & d_3 \\ 0 & 0 & 0 & 1 \end{bmatrix} $$

The kinematic model of the manipulator will be

$$ {}^0T_1 \times {}^1T_2 \times {}^2T_3 = {}^0T_3 $$

Substituting, the respective matrix values, we get

$$ {}^0T_3 = \begin{bmatrix} C_1 & 0 & S_1 & 0 \\ S_1 & 0 & -C_1 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} C_2 & 0 & -S_2 & 0 \\ S_2 & 0 & C_2 & 0 \\ 0 & -1 & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & d_3 \\ 0 & 0 & 0 & 1 \end{bmatrix} $$

On solving and simplifying, we get,

$$ {}^0T_3 = \begin{bmatrix} C_1C_2 & -S_1 & -C_1S_2 & -d_3C_1S_2 \\ S_1C_2 & C_1 & -S_1S_2 & -d_3S_1S_2 \\ S_2 & 0 & C_2 & d_3C_2 \\ 0 & 0 & 0 & 1 \end{bmatrix} $$

or,
$$ \begin{bmatrix} C_1C_2 & -S_1 & -C_1S_2 & -d_3C_1S_2 \\ S_1C_2 & C_1 & -S_1S_2 & -d_3S_1S_2 \\ S_2 & 0 & C_2 & d_3C_2 \\ 0 & 0 & 0 & 1 \end{bmatrix} = \begin{bmatrix} 0.354 & 0.866 & 0.354 & 0.106 \\ -0.612 & 0.500 & -0.612 & -0.184 \\ 0.707 & 0 & 0.707 & 0.212 \\ 0 & 0 & 0 & 1 \end{bmatrix} $$

The preferred solutions for joint displacement are obtained by comparing element (1, 4), (2, 4) and (3, 4). The resulting equations are:

$$-d_3C_1S_2 = 0.106 \quad \text{...(i)}$$
$$-d_3S_1S_2 = -0.184 \quad \text{...(ii)}$$
$$d_3C_2 = 0.212 \quad \text{...(iii)}$$

Dividing equation (ii) by equation (i), the solution for θ_1 is

$$\theta_1 = \text{A}\tan 2(0.184, \; -0.106) = -60°$$

Squaring and adding equation (i) and (ii)

$$d_3^2\, S_2^2\, (S_1^2 + C_1^2) = (0.106)^2 + (-0.184)^2$$

or,
$$d_3S_2 = \pm\sqrt{(0.106)^2 + (-0.184)^2} \quad \text{...(iv)}$$

Dividing equation (iv) by equation (iii), we get,

$$\theta_2 = A\tan 2\left[\pm\sqrt{(0.106)^2 + (-0.184)^2}, 0.212\right]$$
$$= \pm 45°$$

The joint displacement d_3 for joint 3 is obtained by squaring and adding equation (i), equation (ii) and equation (iii). Since the displacement d_3 cannot be negative, only positive sign is used. Thus,

$$d_3 = +\sqrt{(0.106)^2 + (-0.184)^2 + (0.212)^2} = +0.3$$

The two possible solutions are tabulated in table below:

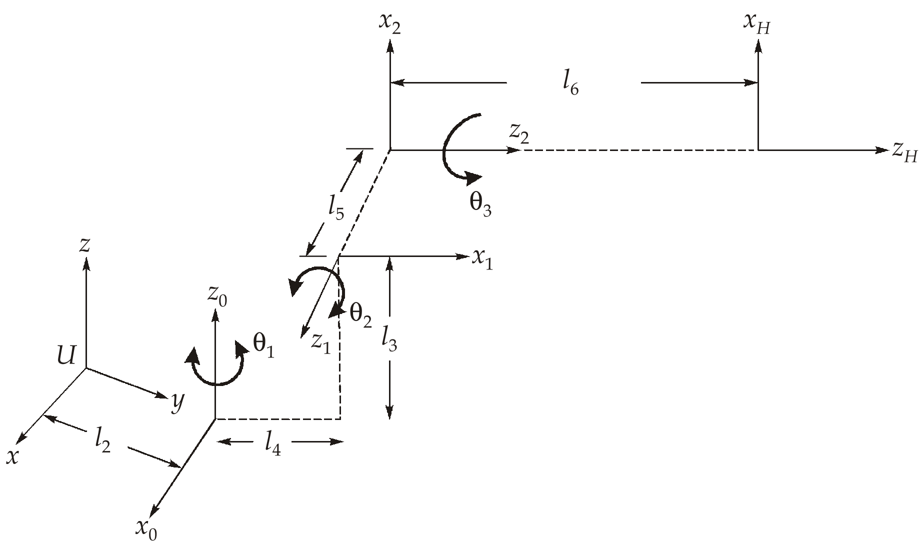
Solution No.	θ_1	θ_2	d_3
1	–60°	–45°	0.3
2	–60°	45°	0.3

The joint range specified for joint 2 is:

$-30° < \theta_2 < 70°$. The solution 1 specifies angle θ_2 as $\theta_2 = -45°$. This violates the joint range constraint and hence, solution 1 is not feasible.

3. (a)

In this solution, the locations of the origins of some frames are arbitrary. Therefore, intermediate matrices might be different for each case. However, the final answer should be the same.



Prepare table:

Link	θ_i	d_i	a_i	α_i	$c\theta_i$	$s\theta_i$	$c\alpha_i$	$s\alpha_i$
0 - 1	θ_i	l_3	l_4	90	c_1	s_1	0	1
1 - 2	$90+\theta_2$	$-l_5$	0	90	$-s_2$	c_2	0	1
2 - H	θ_3	l_6	0	0	c_3	s_3	1	0

Note that the rotation about z_1 is shown to be 90 + θ_2 and not θ_2. This is because even when θ_2 is zero, there is a 90° angle between x_1 and x_2.

The total transformation between the base of the robot and the hand will be

$$^{U}T_H = {^{U}T_O} \times {^{O}T_H} = {^{U}T_O} \times A_1A_2A_3$$

We have,

$$A_i = \begin{bmatrix} C\theta_i & -S\theta_i C\alpha_i & S\theta_i S\alpha_i & a_i C\theta_i \\ S\theta_i & C\theta_i C\alpha_i & -C\theta_i S\alpha_i & a_i S\theta_i \\ 0 & S\alpha_i & C\alpha_i & d_i \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Substituting, the values from D-H parameter table in above transformation matrix, we have,

$$A_1 = \begin{bmatrix} C_1 & 0 & S_1 & L_4C_1 \\ S_1 & 0 & -C_1 & L_4C_1 \\ 0 & 1 & 0 & l_3 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

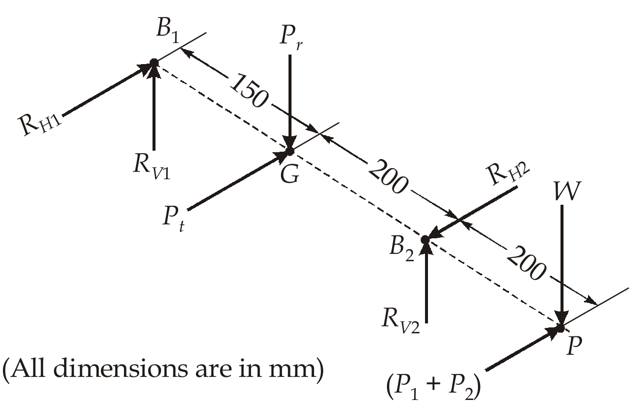
$$A_2 = \begin{bmatrix} -S_2 & 0 & C_2 & 0 \\ C_2 & 0 & S_2 & 0 \\ 0 & 1 & 0 & -l_5 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

$$A_3 = \begin{bmatrix} C_3 & -S_3 & 0 & 0 \\ S_3 & C_3 & 0 & 0 \\ 0 & 0 & 1 & l_6 \\ 0 & 0 & 1 & 1 \end{bmatrix}$$

$$^{U}T_O = \begin{bmatrix} 1 & 0 & 0 & l_1 \\ 0 & 1 & 0 & l_2 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

3. (b)

Given: N = 1000 rpm, d_1 = 15 mm, d_2 = 300 mm, L_{10h} = 10000 hr, load factor = 3.5



(All dimensions are in mm)

Radial and axial forces

The forces acting on the shaft are as shown in above figure.

Considering forces in the vertical plane and taking moments of forces about bearing B_1, we have

$$P_r(150) + W(550) - R_{V2}(350) = 0$$

or $205 \times 150 + 150 \times 550 - R_{V2} \times 350 = 0$

$\therefore$ $R_{V2} = 323.5714$ N

Also, $R_{V1} + R_{V2} = P_r + W$

$R_{V1} + 323.5714 = 205 + 150$

or $R_{V1} = 31.4286$ N

Considering forces in the horizontal plane and taking moments of forces about bearing B_1.

$P_t \times 150 + (P_1 + P_2)550 - R_{H2} \times 350 = 0$

$518 \times 150 + (520 + 200) \times 550 - R_{H2} \times 350 = 0$

or $$R_{H2} = 1353.4285 \text{ N}$$

Also, $$R_{H2} = R_{H1} + P_t + (P_1 + P_2)$$

$$1353.4285 = R_{H1} + 518 + (520 + 200)$$

$\therefore$ $$R_{H1} = 115.4285 \text{ N}$$

$\therefore$ The reaction on bearing B_1,

$$R_1 = \sqrt{\left(R_{V_1}\right)^2 + \left(R_{H_1}\right)^2} = \sqrt{(31.4286)^2 + (115.4285)^2}$$

$$= 119.6306 \text{ N}$$

and, the reaction on bearing B_2,

$$R_2 = \sqrt{\left(R_{V_2}\right)^2 + \left(R_{H_2}\right)^2} = \sqrt{(323.5714)^2 + (1353.4285)^2}$$

$$= 1391.57 \text{ N}$$

The bearing reactions are in the radial direction therefore,

$$F_{r_1} = R_1 = 119.6306 \text{ N}$$
$$F_{r_2} = R_2 = 1391.57 \text{ N}$$

There is no axial thrust on these bearings; hence,

$$F_{a_1} = F_{a_2} = 0$$

Dynamic load capacities

$$P_1 = F_{r_1} = 119.6306 \text{ N}$$
$$P_2 = F_{r_2} = 1391.57 \text{ N}$$

As we know, $$L_{10} = \frac{60 \times N \times L_{10h}}{10^6} = \frac{60 \times 1000 \times 10000}{10^6}$$

$$L_{10} = 600 \text{ million rev.}$$

Considering load factor,

$$C_1 = P_1(L_{10})^{1/3} \times \text{Load factor}$$

$$= 119.6306(600)^{1/3} \times 3.5 = 3531.5124 \text{ N}$$ Ans.

$$C_2 = P_2(L_{10})^{1/3} \times \text{Load factor}$$

$$C_2 = 1391.57(600)^{1/3} \times 3.5$$

$$C_2 = 41079.345 \text{ N} \qquad \text{Ans.}$$

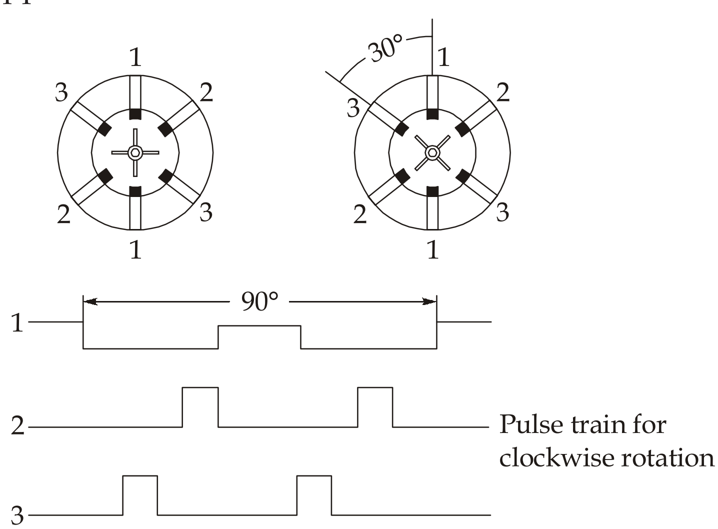
3. (c)

Stepper motor : A stepper motor is a rotating machine which converts a DC voltage pulse into a series of discrete rotational steps. Each step position is an equilibrium position without further excitation, this makes it ideally suitable for use with a digital control. The torque produced by the stepper motor is very small and it can be used to give an accurate positioning as in the case of computer printers, robots, machine tools, etc. The relationship between steps per revolution and step angle is given by the following formula:

$$\text{Step angle} = \frac{360°}{\text{Number of steps per revolution}}$$

There are a number of forms of stepper motors, but they are mainly of three types:

1. variable reluctance types of stepper motor
2. permanent magnetic stepper motor
3. hybrid stepper motor.



Stepper motor

Variable reluctance type

The operation of a simple variable reluctance type model shown in figure which has 30° per step.

The rotor has a number of soft iron teeth which is less than the number of poles on the stator. The stator has pairs of poles being activated by a series of coils. When one pair of poles (pair 1 - 1) is activated, a magnetic field is produced and the rotor will align itself with the energized coil to give the least magnetic reluctance. To move the rotor, the new pair is switched (2 - 2) and the rotor lines up with those poles. Thus, by sequentially switching the current from one pair of poles to the next, the rotor can be made to rotate in steps.

Application and Advantages of Stepper Motor

It is one of the motors that is essentially digital in nature and compactable for designing computers and computer peripheral equipments. Stepper motor is used where precise positioning is required, in combination with a microprocessor controller. Stepper motors have the following advantages:

1. Feedback is not essential for stepper motor and it is perfectly compactable with either analog or digital feedback.
2. Error is minimum and non-cumulative.
3. The stepper motor can accelerate its load easily as maximum torque occurs at low pulse rate.
4. It eliminates the use of gear reduction as low velocities can be obtained with a stepper motor.

However they have disadvantages of low efficiency and resilience problem.

4. (a)

Given : $W = 300$ mm, $t = 25$ mm, $\sigma_t = 90$ MPa, $\tau = 80$ MPa, $\sigma_c = 150$ MPa

(i) Diameter of rivets

$$d = 6\sqrt{t} = 6\sqrt{25} = 30 \text{ mm}$$

(ii) Number of rivets

The shear resistance of one rivet in double shear is

$$P_s = 1.875\left[\frac{\pi}{4}d^2\tau\right]$$

$$= 1.875\left[\frac{\pi}{4}\times(30)^2\times 80\right]$$

or, $$P_s = 106028.7521 \text{ N}$$

Crushing resistance of one rivet is

$$P_c = \sigma_c dt = 150 \times 30 \times 25 = 112500 \text{ N}$$

As $$P_s < P_c$$

Therefore, shear strength of the rivet is the criterion of design.

So, tensile strength of a plate in outer row is

$$P_t = (W - d)\,t\sigma_t = (300 - 30) \times 25 \times 90$$

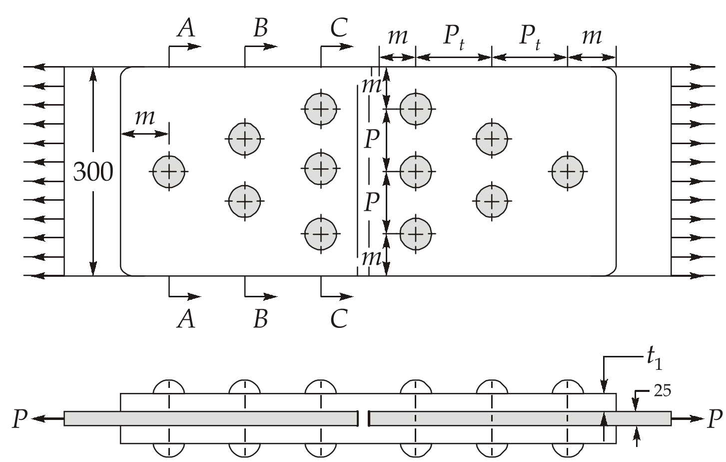
or, $$P_t = 607500 \text{ N}$$

Equating the strength of the plate with the shear strength of n rivets

$$607500 = n(106028.7521)$$

or, $$n = 5.7295 \text{ or } 6 \text{ rivets}$$

(iii) Arrangement of rivets



The dimensions given in figure has following numerical values.

$$m = 1.5d = 1.5 \times 30 = 45 \text{ mm}$$

$$P_t = 2d = 2 \times 30 = 60 \text{ mm}$$

$$t_1 = 0.625t = 0.625 \times 25 = 15.625 \text{ mm}$$

From figure, $$m + P + P + m = 300$$

$$45 + 2P + 45 = 300$$

$$P = 105 \text{ mm}$$

(iv) Efficiency of joint

The strength of the joint along three section is

Along section A-A,

$$\text{Strength} = (W - d)t\sigma_t = 607500 \text{ N}$$

Along section B-B,

$$\text{Strength} = (W - 2d)t\sigma_t + 1 \times P_s$$

$= (300 - 2 \times 30)25 \times 90 + 106028.7521$

$= 646028.7521 \text{ N}$

Along section *C-C*,

$$\text{Strength} = (W - 3d)t\sigma_t + 3 \times P_s$$

$$= (300 - 3 \times 30) \times 25 \times 90 + 3 \times 106028.7521$$

$$= 790586.2563 \text{ N}$$

$$\text{Shear resistance of all rivets} = 6 \times P_s$$

$$= 6 \times 106028.7521$$

$$= 636172.5126 \text{ N}$$

The strength of the joint is least along section *A-A*.

$$\text{Strength of solid plate} = W \times t \times \sigma_t$$

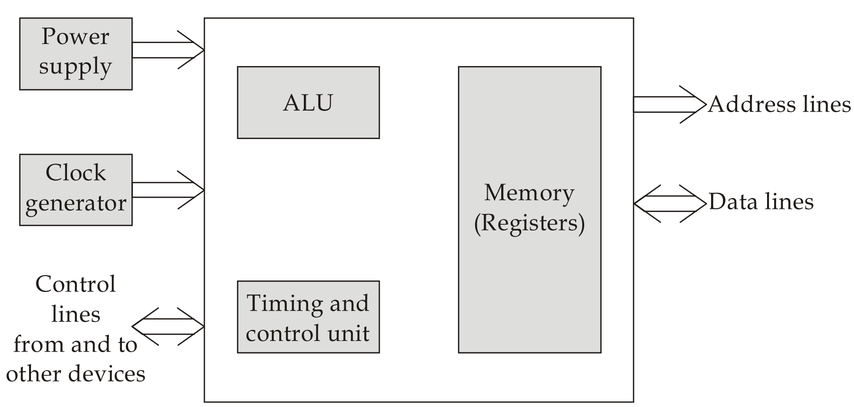
$$= 300 \times 25 \times 90$$

$$= 675000 \text{ N}$$

Therefore, $$\eta = \frac{607500}{675000} = 0.9 \text{ or } 90\%$$

4. (b)

(i)



General block diagram of a microprocessor in a system

Interface is the mid-system between I/O and a microprocessor and takes care of code conversion, synchronization and other associated problems. In other words, interface is the device which resolves the functional and constructional differences between I/O device and microprocessor. The differences are enumerated below,

1. Peripherals are electromechanical devices while microprocessors are electronic devices. So, their ways of operation are different.

2. Data transfer rates between I/O devices and microprocessor vary vastly.
3. Operation of I/O devices has to be synchronized with that of the microprocessors. Basically we mean here the protocols and mechanisms which should see that the information is transmitted between the I/O device and microprocessor without any problems such as loss of data or data corruption.
4. Data format between microprocessor and I/O is different.
5. Operation of each I/O has to be individually controlled so as not to disturb the operation of the microprocessor.

For these reasons an I/O interface is used. The I/O interface performs the following tasks:

1. It provides ways through which data from the each external device will be transferred properly with the microprocessor without causing interference to other devices connected to the system buses.
2. It resolves any difference that may exist regarding the timing (clock speeds) between the microprocessor and the peripheral device. The microprocessor CPU runs on its own internal clock while the peripherals may or may not have their own internal clocks.
3. It converts the format of the data of the peripherals to the format that is acceptable by the microprocessor and vice versa.
4. Also produce interrupt signals to force the microprocessor or react immediately in case some peripherals demand immediate action.

(ii) Capacitance, $C = \epsilon_o \epsilon_r \dfrac{A}{d}$

$$A = 2 \times 30 \times 30 = 1800 \text{ cm}^2 = 0.18 \text{ m}^2$$

Displacement sensitivity $= \dfrac{dC}{dd}$

$$\therefore \quad \frac{dC}{dd} = \epsilon_o \epsilon_r A\left(-\frac{1}{d^2}\right) = -\frac{\epsilon_o \epsilon_r A}{d^2}$$

$$= \frac{-8.854 \times 10^{-12} \times 1.005 \times 0.18}{\left(1.2 \times 10^{-3}\right)^2} = -1.112 \times 10^{-6} \text{ F/m}$$

Minus sign indicates that the capacitance will increase for decreasing value of d.

4. (c)

Given : $W = 12$ kN, $N = 1500$ rpm, $P = 1.2$ MPa, $\dfrac{l}{d} = 1$, $\dfrac{r}{c} = 900$, $\mu = 35$ m Pa-s

(i) Dimension of bearing

$$P = \frac{W}{ld} = \frac{W}{d^2}$$

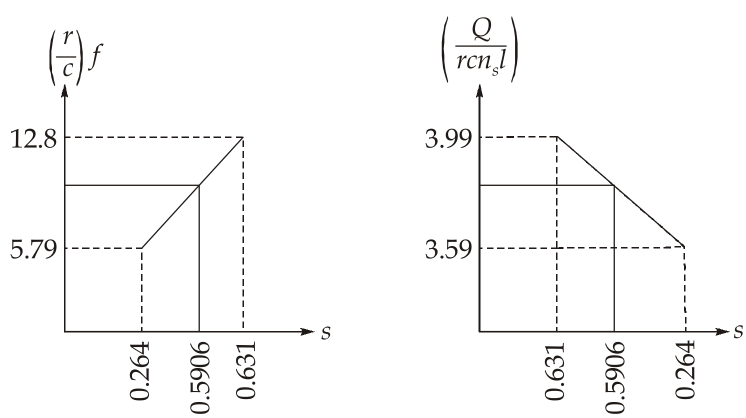
or, $$1.2 = \frac{12 \times 10^3}{d^2}, \; d = 100 \text{ mm} = l$$

(ii) Sommerfeld number $(s) = \left(\frac{r}{c}\right)^2 \frac{\mu n_s}{P}$

$$= \frac{(900)^2 (35 \times 10^{-9}) \left(\frac{1500}{60}\right)}{1.2}$$

$$s = 0.590625$$

The values of dimensionless performance parameters are obtained by linear interpolation from table given



$$\left(\frac{r}{c}\right)f = 5.79 + (12.8 - 5.79)\frac{(0.5906 - 0.264)}{(0.631 - 0.264)} = 12.0288$$

$$\left(\frac{Q}{rcn_s l}\right) = 3.99 - (3.99 - 3.59)\frac{(0.5906 - 0.264)}{(0.631 - 0.264)} = 3.634$$

$$\frac{Q_s}{Q} = 0.497 - (0.497 - 0.28)\frac{(0.5906 - 0.264)}{(0.631 - 0.264)} = 0.30388$$

Coefficient of friction, $$f = 12.0288\left(\frac{c}{r}\right) = 12.0288 \times \left(\frac{1}{900}\right)$$

or, $$f = 0.013365$$ Ans.

(iii) Power lost in friction

$$\text{Power} = \frac{2\pi n_s \times f \times W \times r}{10^6}$$

$$= \frac{2\pi \times \left(\frac{1500}{60}\right) \times 0.013365 \times 12 \times 10^3 \times 50}{10^6}$$

Power = 1.26 kW Ans.

(iv) Total flow of oil

$$Q = 3.634 r \times c \times n_s \times l$$

$$= 3.634 \times 50 \times \left(\frac{50}{900}\right)\left(\frac{1500}{60}\right) \times 100$$

$$= 25236.111 \text{ mm}^3/\text{s}$$

(v) Side leakage $Q_s = 0.30388 \times Q = 7668.749 \text{ mm}^3/\text{s}$

(vi) Temperature rise

$$\Delta t = \frac{8.3 \times P \times (CFV)}{(FV)}$$

$$\Delta t = \frac{8.3 \times 1.2 \times 12.0288}{3.634}$$

$$\Delta t = 32.9683°\text{C}$$ Ans.

Section : B

5. (a)

Given: I.P. = 60 kW, *bsfc* = 275 gm/kWh = 0.275 kg/kWh, $\eta_{b\,th} = 35\%$, $\eta_m = 65\%$ (at 75% load)

Let, the brake power at full load be x kW; the brake power at 75% of load be $0.75x$ kW; the indicated power at 75% of load = ($0.75x$ + F.P.) kW [F.P. = Friction power]

At 75% load, $\eta_m = 0.65$

$$\left[\frac{BP}{IP}\right]_{75\%} = 0.65$$

$$\Rightarrow \quad \frac{0.75x}{0.75x + F.P.} = 0.65$$

$$\Rightarrow \quad 0.75x = 0.4875x + 0.65 \times \text{F.P.}$$

$$\Rightarrow \quad \text{F.P.} = 0.4038x$$

Friction power (F.P.) remains constant at all loads.

At full load, I.P. = B.P. + F.P. = 60 kW

or, $x + 0.4038x = 60$

$\Rightarrow$ $x = 42.74$ kW

$\therefore$ Brake power, B.P. = 42.74 kW **Ans.**

Friction power, F.P. = 0.4038 × 42.74

= 17.259 kW **Ans.**

Mechanical efficiency at full load,

$$\eta_m = \frac{\text{B.P.}}{\text{I.P.}} = \frac{42.74}{60} = 0.7123 = 71.23\%$$ **Ans.**

Indicated thermal efficiency,

$$\eta_{ith} = \frac{\eta_{bth}}{(\eta_m)_{\text{at full load}}} = \frac{0.35}{0.7123} = 0.4913 = 49.13\%$$ **Ans.**

Indicated specific fuel consumption,

$$isfc = bsfc \times \eta_m = 0.275 \times 0.7123 = 0.1958 \text{ kg/kWh}$$ **Ans.**

At half load,

$$\text{B.P.} = \frac{42.74}{2} = 21.37 \text{ kW}$$

$$\text{F.P.} = 17.259 \text{ kW}$$

$$\eta_m = \frac{\text{B.P.}}{\text{B.P.+F.P.}} = \frac{21.37}{21.37 + 17.259}$$

$$= 0.5532 = 55.32\%$$ **Ans.**

5. (b)

General maintenance problems of Biogas plants and their remedies

(i) Handling of digested slurry : It may be a major problem if sufficient open space or compost pits are not available to get the slurry dry. For domestic plant, a 200-litre capacity oil drum can be used to carry this affluent to the fields.

(ii) Low gas production rate : The methane forming bacteria are very sensitive to temperature. During winter, as the temperature falls, the gas production rate also reduces. Following methods may be used to enhance rate of gas production.

- Using solar thermal heater to add hot water in the inlet slurry, but the water temperature should not exceed 60°C as higher temperature will kill the methane forming bacteria.
- Circulation of solar-heated hot water through pipes in the digester.
- Greenhouse effect may be used to trap solar radiation for heating.
- Covering the biogas plant by straw bags during night hours.
- Manual or auto stirring of digester slurry.
- Addition of some nutrients for bacteria.

(iii) Some problems arise due to the following :

- Increased loading rate.
- Not mixing sufficient water in the cattle dung.

Due to these reasons, the flow of slurry from inlet towards outlet is very slow or may even stop. This may cause accumulation of volatile fatty acids leading to decrease of pH value and ultimately failure of the plant. Also, due to lack of water in the slurry, it is not possible to stir the digester content of high solid concentration.

(iv) Some persons add urea fertilizer in large quantities, leading to reduction of bacterial activity due to toxicity.

(v) Leakage of gas from gasholder, especially in case of Janata plant is a major and common problem. Leakage should be checked and leakage points are marked and repaired. During repairing operation, there should be no gas inside the gasholder and the stopcock should remain open till the repaired points get dry.

5. (c)

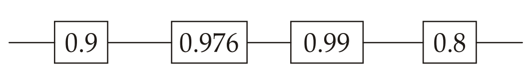
The reliability of the first parallel system,

$$R_{P1} = 1 - (1 - 0.8)(1 - 0.6)(1 - 0.7)$$
$$= 0.976$$

The reliability of the second parallel system,

$$R_{P2} = 1 - (1 - 0.9)(1 - 0.5)(1 - 0.8)$$
$$= 0.99$$

The above parallel series system can be reduced to equivalent single series system as given below:



The reliability of the system is given by

$$R_{eq} = 0.9 \times 0.976 \times 0.99 \times 0.8 = 0.6957$$ **Ans.**

If the reliability of the second component in first parallel system is changed to 0.7, the reliability of first parallel system would be

$$R'_{P1} = 1 - (1 - 0.8)(1 - 0.7)(1 - 0.7) = 0.982$$

The reliability of the combined system would be

$$R'_{eq} = 0.9 \times 0.982 \times 0.99 \times 0.8 = 0.69996$$

$\therefore$ Change in reliability = 0.69996 – 0.6957 = 4.269×10^{-3} = 0.004269

5. (d)

Maximum cell efficiency, $\eta = \dfrac{-\Delta G}{-\Delta H} = \dfrac{-237191 \times 100}{-285838} = 82.98\%$ Ans. (i)

Theoretical EMF, $|\Delta G| = nFE$

$\therefore$ $E = \dfrac{237191}{2 \times 96500} = 1.229 V$ Ans. (ii)

Voltage efficiency, $\eta_v = \dfrac{V_{act}}{V_{th}} = \dfrac{1.15 \times 100}{1.229} = 93.57\%$ Ans. (iii)

Losses $= V_{\text{th}} - V_{\text{act}}$

$= 1.229 - 1.15 = 0.079\text{V}$ Ans. (iv)

Maximum power output, $P_{\max} = \dfrac{m}{M} \times \Delta G$

$= \dfrac{10^{-3}}{2.02} \times 237.191 \times 10^3$

$P_{\max} = 117.42 \text{ kW}$ Ans. (v)

5. (e)

Given: Number of failures = 2, Number of units tested = 25

(i) Percentage of failures $= \dfrac{\text{Number of failures}}{\text{Number of units tested}}$

$= \dfrac{2}{25} \times 100 = 8\%$ **Ans.**

(ii) Total time = 1200 × 25 = 30000 unit-hour

Non-operating time = 800 hours for 1$^{\text{st}}$ failure + 600 hours for 2$^{\text{nd}}$ failure

= 1400 unit-hour

$\therefore$ Operating time = Total time – Non-operating time

= 30000 – 1400 = 28600 hours.

$\therefore$ Number of failures per operating hour

$= \dfrac{\text{Number of failures}}{\text{Operating time}} = \dfrac{2}{28600}$

$= 6.993 \times 10^{-5}$ failure/unit-hour Ans.

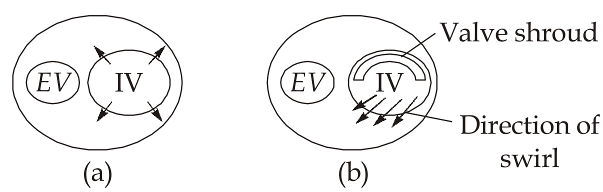
(iii) Mean time between failure,

$\text{MTBF} = \dfrac{1}{6.993 \times 10^{-5}} = 14300 \text{ hours}$ Ans.

6. (a)

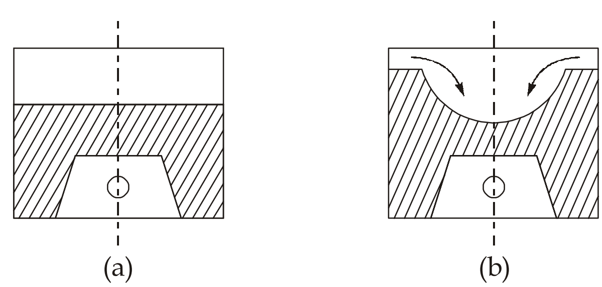
(i) The knocking tendency of an engine is affected by the following design consideration as described below:

(a) Effect of shrouded inlet valve: Plain valves and shrouded inlet valves are shown in figure (a) and figure (b) respectively. The use of a shrouded inlet valve provides the flow of charge in a definite direction, so that the combustion time is reduced. This will reduce the tendency to knock. The shrouded valve also tends to reduce the cycle-to-cycle variation, especially when oriented so as to give tangential flow into the cylinder.



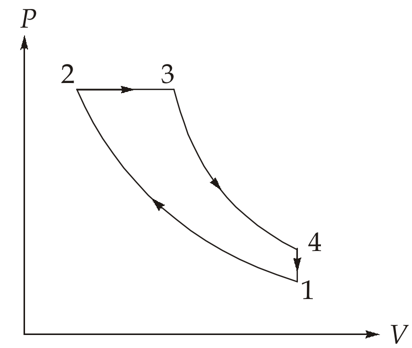
(a) Plain valves and (b) shrouded inlet valves
(IV–Inlet valve, EV–Exhaust valve)

(b) Effect of piston shape: Plain piston with flat top and squish piston are shown in figure (a) and figure (b) respectively. In squish piston, the charge is squeezed radially inwards, near the top dead centre and tendency to knock is less. The thin space above the piston in the combustion chamber is called the quench space. Near the end of the flame travel the end-gas is located in a thin space where it makes good contact with the cylinder walls, which are at a lower temperature than the end-gas. Thus the quench space is cooled which reduces the possibility of auto-ignition and hence the knocking. Because of the reduced space above the squish piston, the combustion chamber becomes effectively more compact and the possibility of turbulence increases. Both of these factors tend to decrease the knocking tendency.



(a) Flat Piston and (b) squish piston.

(ii)



Given: $r = 16$

Since cut-off takes place at 8% of the stroke,

$$V_3 - V_2 = 0.08\,(V_1 - V_2)$$

$$\frac{V_3}{V_2} - 1 = 0.03\left(\frac{V_1}{V_2} - 1\right)$$

$$\Rightarrow \quad \rho = 1 + 0.08\,(r - 1)$$

$$\Rightarrow \quad \rho = 2.2$$

For initial conditions,

$$C_V = 0.718 \text{ kJ/kg-K}$$

$$R = 0.287 \text{ kJ/kg-K}$$

$$\eta_{\text{diesel}} = 1 - \frac{1}{r^{\gamma-1}} \times \frac{\rho^{\gamma} - 1}{\gamma(\rho - 1)} = 1 - \frac{1}{(16)^{1.4-1}} \times \frac{(2.2)^{1.4} - 1}{1.4(2.2 - 1)}$$

$$= 0.6042$$

If specific heat at constant volume increases by 2.5%.

$$C_V' = 1.025 \times 0.718$$

$$= 0.73595 \text{ kJ/kg-K}$$

$$\therefore \quad C_P' = C_V' + R = 1.02295 \text{ kJ/kg-K}$$

$$\therefore \quad \gamma' = \frac{C_P'}{C_V'} = \frac{1.02295}{0.73595} = 1.38997$$

$$\eta_{\text{diesel}}' = 1 - \frac{1}{(16)^{1.38997-1}} \times \frac{(2.2)^{1.38997} - 1}{1.38997(2.2 - 1)}$$

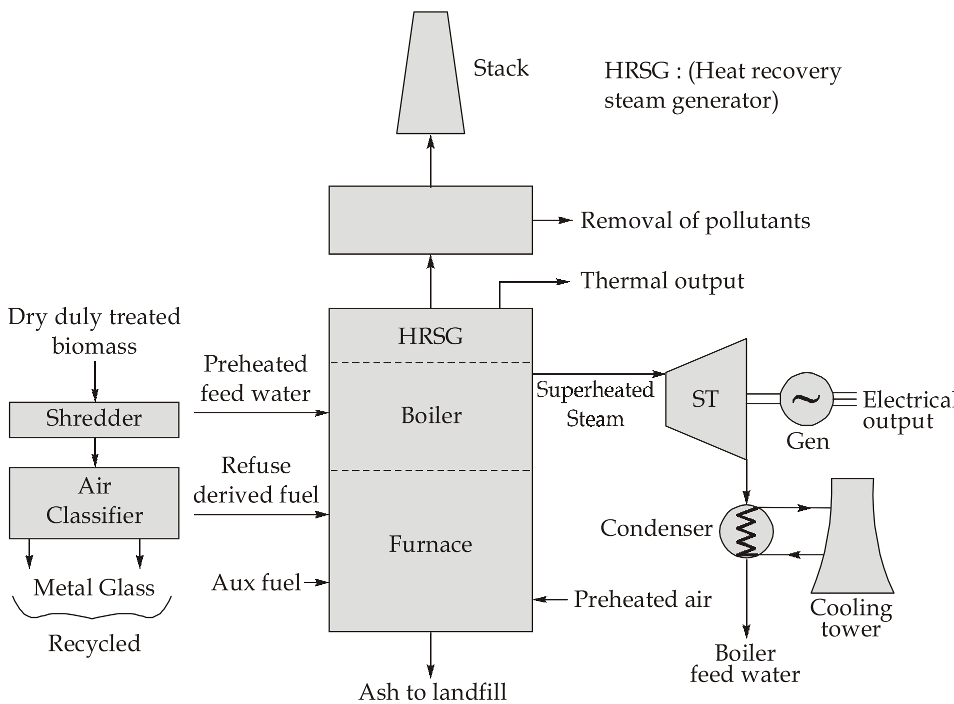
$$= 0.594934$$

Percentage change in efficiency $= \dfrac{0.6042 - 0.594934}{0.6042} \times 100 = 1.5335\%$ **Ans.**

6. (b)

(i)

A block diagram of MSW-to-energy incineration plant showing the sequence of various steps is shown in figure. The dry biomass is shredded to pieces of about 2.5 cm diameter. An air stream segregates the refuse-derived fuel (RDF), which is lighter from heavier metal and glass pieces. These are reclaimed and recycled. About 30% of the US waste stream is recycled. The RDF thus obtained is burnt in a furnace at about 1000°C to produce steam in the boiler. Combustion process may be assisted by a required amount of auxiliary fuel when RDF does not burn properly by itself. The superheated steam obtained from boiler is used in a steam turbine coupled with an alternator to produce electrical output in the same way as in a conventional thermal plant. The flue gases are discharged to the atmosphere through a stack removal of pollutants such as particulate matter, SO_x and NO_x. A heat-recovery steam generator extracts maximum possible heat from flue gases to form thermal output. The ash is removed and disposed off to landfills.



MSW to energy incineration plant

(ii)

Characteristics of various fuel cells

S.No.	Fuel cell	Op. Temp.	Fuel	Efficiency
1.	PEMFC	40 - 60°C	H_2	48 - 58%
2.	AFC	90°C	H_2	64%
3.	PAFC	150 - 200°C	H_2	42%
4.	MCFC	600 - 700°C	H_2 and CO	50%
5.	SOFC	600 - 1000°C	H_2 and CO	60 - 65%

Hydrogen is a primary fuel and the main source of energy for all fuel cells. Thus pure hydrogen or hydrogen-rich gases are generally used in fuel cells. Some fuel cells (MCFC and SOFC) can also CO along with H_2. All types of fuels can be classified into two categories: (i) direct type, and (ii) indirect type.

Direct type fuels are directly introduced in the cell as such, without any transformation or reforming, to serve as active material. Examples are pure hydrogen, mixture of hydrogen with other gases, hydrazine (N_2H_4) and methanol.

Indirect fuels are hydrogen-rich fuels, which are first converted (reformed) to a mixture of H_2 and some other products, e.g., CO, CO_2 and N_2. Hydrocarbon fuels are decomposed by reaction with steam at high temperature in the presence of a catalyst. The process is known as steam reforming of fuels. The products containing mainly H_2, CO and CO_2 are then supplied to the fuel cell.

6. (c)

(i) Carbon monoxide is generated in an engine when it is operated with a fuel-rich equivalence ratio as there is not enough oxygen to convert all carbon to carbon dioxide. For fuel-rich mixtures, CO concentrations in the exhaust increase steadily with the increasing equivalence ratio. The engine runs rich when it is started or when it is accelerated under load. For fuel-lean mixtures, CO concentrations in the exhaust are very low and are of the order 10^{-3} mole fraction. Poor mixing, local rich regions, and incomplete combustion create some CO. SI engines often operate close to stoichiometric at part load and operate fuel rich at full load. Under these conditions, the CO emissions are significant. However, CI engines operate well on the lean side of stoichiometric and therefore produce very little CO emissions.

(ii)

- The most widely accepted causes for hydrocarbon emissions in exhaust gases of spark ignition engines are:
  1. Flame quenching at the combustion chamber walls, leaving a layer of unburned fuel-air mixture adjacent to the walls.
  2. Crevices in the combustion chamber, small volumes with narrow entrances, which are filled with the unburned mixture during compression and remains unburned after flame passages, since the flame cannot propagate into the crevices. The main crevice regions are the spaces between the piston, the piston rings and the cylinder walls.
  3. The oil film and deposits on the cylinder walls absorb fuel during intake and compression and the fuel vapour is desorbed into the cylinder during expansion and exhaust.

4. Incomplete combustion, either partial burning or complete misfire, occurring when the combustion quality is poor, e.g. during engine transients when air-fuel, exhaust gas recirculation and spark timing may not be adequately controlled.

- The following are the major causes for hydrocarbon emissions in the exhaust of CI engines.

1. Diesel fuel contains components of higher molecular weights on average than those in a gasoline fuel, resulting in higher boiling and condensing temperatures. This causes some hydrocarbon particles to condense on the surface of the solid carbon soot generated during combustion. Most of this is burned as mixing continues and the combustion process proceeds but a small amount is exhausted out of the cylinder.
2. The air-fuel mixture in a CI engine is heterogeneous with fuel still being added during combustion. It causes local spots to range from very rich to very lean and many flame fronts exist at the same time unlike the homogeneous air-fuel mixture of an SI engine that essentially has one flame front. Incomplete combustion may be caused by undermixing or overmixing. With undermixing, in fuel-rich zone some fuel particles do not find enough oxygen to react with, and in the fuel-lean zones some local spots will be too lean for combustion to take place properly. With overmixing, some fuel particles may be mixed with burned gases and it will therefore lead to incomplete combustion.
3. A small amount of liquid fuel is often trapped on the tip of the injector nozzle even when injection stops. This small volume of fuel is called sac column. This sac volume of liquid fuel is surrounded by a fuel-rich environment and therefore it evaporates very slowly causing hydrocarbon emissions in the exhaust.
4. CI engines also have hydrocarbon emissions for some of the same reasons as SI engines do, e.g. flame quenching, crevice volume, oil-film and deposits on the cylinder wall, misfiring, etc.

- Hydrocarbon concentration from the exhaust of an SI engine can be decreased by the following methods:

1. **Increasing the exhaust gas temperature:** By increasing the exhaust gas temperature the oxidation reaction of HC increases, if sufficient oxygen is present in the exhaust, and this lowers the HC emission. The exhaust gas temperature can be increased by changing the variables like decreasing the compression ratio, retarding the spark, increasing the temperature of the coolant, increasing the speed, increasing the charge pressure and insulating the exhaust manifold.

2. **More oxygen in the exhaust:** Enough oxygen in the exhaust is necessary in order to carry out oxidation reaction of hydrocarbons. It can be achieved by using lean fuel-air mixture. However, the mixture should not be too lean to cause misfire.
3. **Lesser mass in the quench region:** The mass of the hydrocarbons in quench regions can be reduced by decreasing the surface to volume ratio, increasing the turbulence during combustion.
4. **More time for reaction:** The reaction time can be increased by decreasing the speed, using a more homogeneous mixture, increasing the exhaust pressure.

7. (a)

(i) For the unsupercharged engine:

Brake specific fuel consumption,

$$bsfc = \frac{\dot{m}_f}{\text{B.P.}} = 0.25 \text{ kg/kWh}$$

or, Mass of fuel consumed,

$$\dot{m}_f = bsfc \times \text{B.P.}$$

$$\Rightarrow \quad \dot{m}_f = 0.25 \times 280 = 70 \text{ kg/h} = \frac{70}{60} \text{ kg/min}$$

Mass of air used, $\dot{m}_a = \dot{m}_f \times \frac{A}{F} = \frac{70}{60} \times 20 = \frac{70}{3}$ kg/min

Volumetric efficiency,

$$\eta_v = \frac{\text{Actual mass of air taken in per cycle}}{\text{Mass of air corresponding to swept volume}}$$

$$0.82 = \frac{\dot{m}_a}{\rho_a \times V_d \times \frac{N}{2}} = \frac{70/3}{\frac{P}{RT} \times V_d \times \frac{2000}{2}}$$

$$0.82 = \frac{70/3}{\frac{1.013 \times 100}{0.287 \times 285} \times V_d \times 1000}$$

$$V_d = 0.022976 \text{ m}^3$$

$\therefore$ Engine capacity = 0.022976 m^3 **Ans.**

Brake power = $P_{\text{bmep}} \times V_s$

$$\Rightarrow \quad 280 \times 10^3 = P_{bmep} \times 0.022976 \times \frac{N}{2 \times 60}$$

$$\Rightarrow \qquad 280 \times 10^3 = P_{bmep} \times 0.022976 \times \frac{2000}{120}$$

$$P_{\text{bmep}} = 7.31 \text{ bar}$$ **Ans.**

(ii) For the supercharged engine:

Total power produced by engine = 280 + (0.08 × Total power)

$\Rightarrow$ (1 – 0.08) × Total power = 280

$$\Rightarrow \text{Total power produced} = \frac{280}{1-0.08} = 304.347 \text{ kW}$$

$\because$ For power output of 280 kW, the mass of air required

$$= \frac{70}{3} \text{ kg/min}$$

$\therefore$ For power output of 304.347 kW, the mass of air required is

$$= \frac{70 \times 304.347}{3 \times 280}$$

$$\dot{m}_a' = 25.362 \text{ kg/min.}$$

$$\text{Volumetric efficiency, } \eta_V = \frac{\dot{m}_a'}{\rho_a' \times V_d \times \frac{N}{2}}$$

Here, ρ_a' is the density of air at inlet to cylinder i.e.; outlet of supercharger.

$$0.82 = \frac{25.362}{\frac{P_2}{RT_2} \times 0.022976 \times \frac{2000}{2}} = \frac{25.362}{\frac{P_2}{0.287 \times (35+273)} \times 0.022976 \times 1000}$$

$$\Rightarrow \qquad P_2 = 118.995 \text{ kPa} = 1.18995 \text{ bar}$$

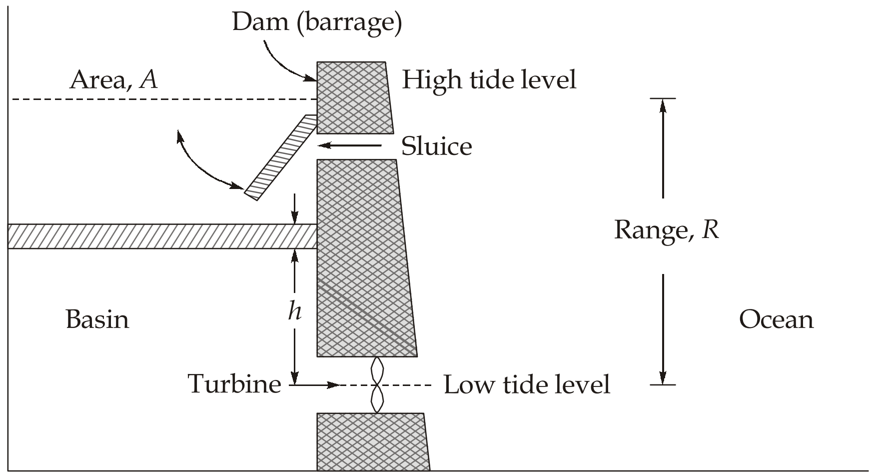
$\therefore$ Increase in air pressure required in the supercharger is given by,

$$(1.18995 - 0.7) = 0.48995$$

$$\approx 0.49 \text{ bar}$$ (Ans.)

7. (b)

Main components of a tidal plant, as shown in figure :



Power generation from tides

Main components of a tidal plant are:

(i) dam, barrage or dyke: a barrier constructed to hold water

(ii) sluice ways: rapid controlled gates, used to fill basin during high tides or emptying it during low tides and

(iii) a special, bulb type power turbine-generator set, Steel shell containing an alternator and special Kaplan turbine with variable pitch blades.

Tidal power associated with single filling or emptying of a basin may be estimated as follows:

Consider water trapped at high tide in a basin of area A, and allowed to run out through a turbine at low tide as shown in figure. Potential energy in the mass of water stored in incremental head dh above head h:

$$dW = dm.gh$$

But $$dm = \rho A dh$$

Thus, $$dW = \rho Aghdh$$

Total potential energy of water stored in the basin

$$W = \int_r^R \rho \cdot A \cdot g \cdot h \cdot dh \text{ Joules}$$

or $$W = \frac{1}{2}\rho Ag\left(R^2 - r^2\right)$$

where, ρ = Density of water; g = Gravitational constant

As the time between consecutive high and low tide is 6 hours 12.5 minutes (=22,350 seconds), this power is to be utilized within this period. Assuming an average sea water density of 1025 kg/m^3, the average theoretical power generated in one filling or emptying of the basin is given by:

$$P_{avg} = \frac{1025 \times 9.81 \times A \times \left(R^2 - r^2\right)}{2 \times 22350}$$

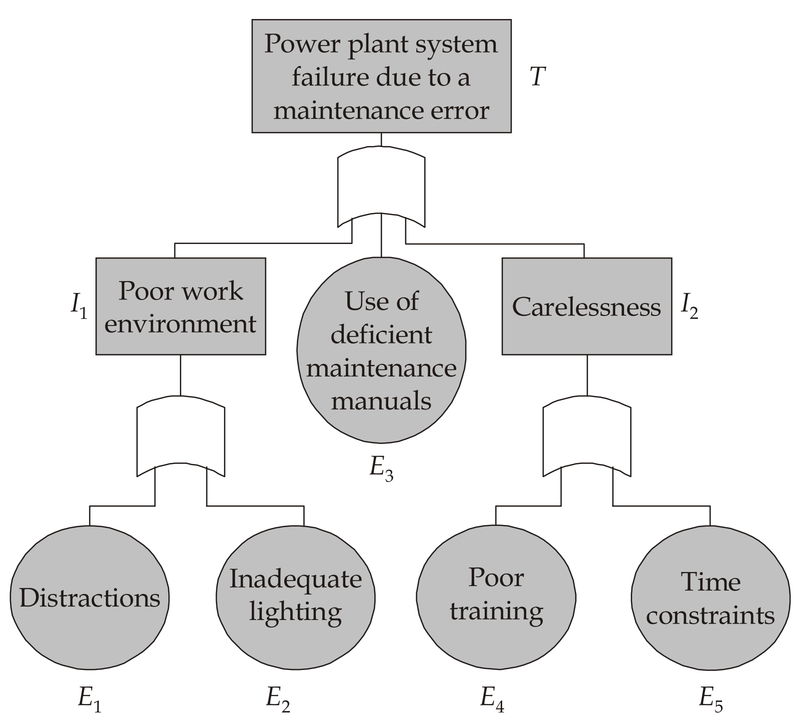
$$P_{avg} = 0.225 \times A \times (R^2 - r^2) \text{ Watt}$$ Proved.

7. (c)

(i) The main objectives of performing fault tree analysis (FTA) are:

- To understand the functional relationships of system failures.
- To verify the ability of the system to meet its imposed safety associated requirements.
- To highlight cost-effective improvements and critical areas.
- To comprehend the degree of protection that the design concept provides against the occurrence of failures.
- To meet jurisdictional requirements.

(ii)



8. (a)

(i) Since, a petrol engine is throttled to maintain a high fuel/air ratio with load, combustion is not complete within the cylinder and a plot of brake power versus the rate of fuel consumption does not yield a straight line. Hence, extrapolation is virtually impossible. So, the Willan's line method is not applicable to SI engine.

(ii) Given: d = 140 mm = 0.14 m, L = 150 mm = 0.15 m, N = 600 rpm,

$\dot{m}_f = 0.0425 \text{ kg/min} = \dfrac{0.0425}{60} \text{ kg/s}$, CV = 42 MJ/kg = 42000 kJ/kg

Difference in tension on either side of brake pulley, W = 300 N

Brake circumference = 2.4 m

Length of the indicator diagram = 55 mm

Area of positive loop = 490 mm^2

Area of negative loop = 30 mm^2

Spring constant = 0.9 bar/mm

(a) $$\text{Arm length} = \frac{\text{Brake circumference}}{2\pi}$$

$$\Rightarrow \quad r = \frac{2.4}{2\pi} = 0.382 \text{ m}$$

$$\text{Torque, } T = Wr = 300 \times 0.382 = 114.6 \text{ N-m}$$

$$\therefore \quad \text{Brake power, BP} = T \times \omega = 114.6 \times \frac{2\pi N}{60} = 114.6 \times 2\pi \times \frac{600}{60} = 7.2 \text{ kW}$$ **Ans.**

(b) Mean height of indicator diagram

$$= \frac{\text{Net area of indicator diagram}}{\text{Length of indicator diagram}}$$

$$= \frac{490 - 30}{55} = 8.363 \text{ mm}$$

Indicated mean effective pressure = Mean height of indicator diagram × Spring constant

$$imep = 8.363 \times 0.9 = 7.527 \text{ bar}$$

$$\therefore \quad \text{I.P.} = \frac{imep \times L \times A \times N}{2 \times 60}$$

$$= 7.527 \times 10^5 \times 0.15 \times \frac{\pi}{4} \times (0.14)^2 \times \frac{600}{2 \times 60}$$

$$= 8.69 \text{ kW}$$ **Ans.**

(c) Mechanical efficiency, $\eta_m = \dfrac{B.P.}{I.P.} = \dfrac{7.2}{8.69} = 0.8286 \text{ or } 82.86\%$ **Ans.**

(d) Brake thermal efficiency,

$$\eta_{bth} = \frac{B.P.}{\dot{m}_f \times CV} = \frac{7.2 \times 60}{0.0425 \times 42000} = 0.2420 \text{ or } 24.20\% \quad \textbf{Ans.}$$

(e) Brake specific fuel consumption,

$$\text{bsfc} = \frac{\dot{m}_f}{B.P.} = \frac{0.0425 \times 60}{7.2} = 0.354 \text{ kg/kWh} \quad \textbf{Ans.}$$

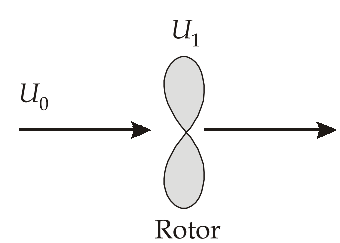
8. (b)

$U_H = 20$ m/s, $H = 50$ m, $z = 120$ m, $\rho = 1.23$ kg/m^3, $\alpha = 0.15$, $D = 100$ m,

$A = \frac{\pi D^2}{4} = 7853.98$ m^2, $U_1 = 0.7U_0$, $\eta_{gen} = 0.8$

Now, $$U_z = U_H\left(\frac{z}{H}\right)^\alpha = 20\left(\frac{120}{50}\right)^{0.15} = 22.8 \text{ m/s}$$

and $$U_1 = 0.7 \times 22.8 = 15.96 \text{ m/s}$$



Now, $$P_0 = \frac{1}{2}\rho A U_0^3$$

$$= \frac{1}{2} \times 1.23 \times 7853.98 \times 22.8^3$$

$$= 57.25 \text{ MW} \quad \text{Ans. (i)}$$

Interference factor, $$a = \frac{U_0 - U_1}{U_0} = \frac{22.8 - 15.96}{22.8} = 0.3$$

$\therefore$ Power coefficient, $c_p = 4a(1-a)^2 = 4 \times 0.3(1-0.3)^2$

$$c_p = 0.588$$

$\therefore$ Power extracted by the turbine,

$$P_T = c_p \cdot P_0 = 0.588 \times 57.25$$

$$P_T = 33.66 \text{ MW} \quad \text{Ans. (ii)}$$

Electrical power generated, $P_E = \eta_{gen} \times P_T = 0.8 \times 33.66$

$$P_E = 26.93 \text{ MW} \quad \text{Ans. (iii)}$$

Axial thrust on turbine, $$F_A = 4a(1-a)\rho A\frac{U_0^2}{2}$$

$$= 4 \times 0.3 \times 0.7 \times 1.23 \times 7853.98 \times \frac{22.8^2}{2}$$

$$F_A = 21 \times 10^5 \text{ N} \quad \text{Ans. (iv)}$$

Maximum axial thrust occurs when $a = 0.5$

$\therefore$ $$F_A = 4 \times 0.5 \times 0.5 \times 1.23 \times 7853.98 \times \frac{22.8^2}{2}$$

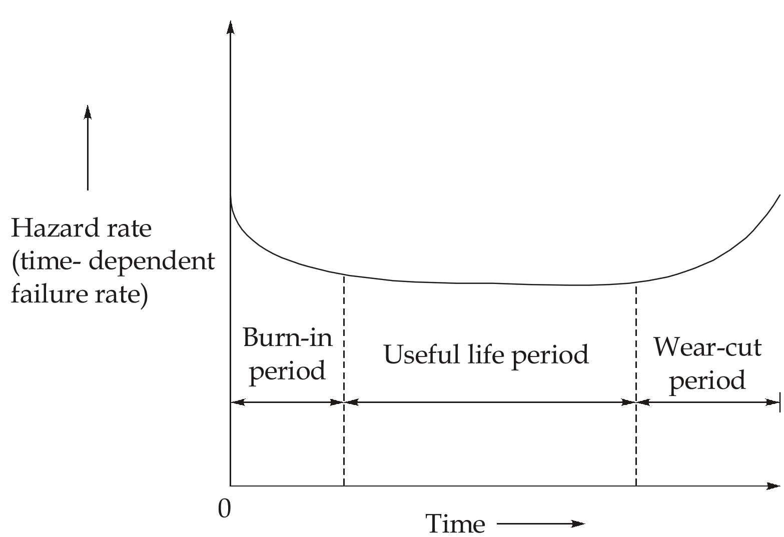
$$F_A = 25 \times 10^5 \text{ N}$$ Ans. (v)

8. (c)

(i) The commonly adopted objectives of maintenance and plant engineering are:

- To maximize the uninterrupted available time of the assets or machinery or equipment or services so that they can be used for the intended purpose they were procured.
- To preserve the value of assets by reducing the rate of deterioration by maintaining them to work in good condition. For example, lubricating, removing rust, dust, and dirt periodically and applying grease, etc.
- To facilitate maximisation of output of production by maintaining utilisation of machinery without affecting its deterioration.
- To perform the maintenance activities in the most economical way, such as restoring the condition of machine at the earliest possible time when it fails or is about to fail, planning and scheduling preventive maintenance without interrupting the production schedules.
- To update the machine and/or reconditioning with the latest technological or engineering features and/or management philosophies/styles.

(ii) As shown in figure, the curve is divided into three sections: burn-in period, useful life period, and wear-out period.



Bathtub hazard rate curve

(a) During the burn-in period, the system/item hazard rate decreases with time, and some of the reasons for the occurrence of failures during this time period are inadequate debugging, poor manufacturing methods and processes, poor quality control, human error, and substandard materials and workmanship. Three other terms used for this decreasing hazard rate region are debugging region, infant mortality region, and break-in-region.

(b) During the useful life period, the hazard rate remains constant. Some of the reasons for the occurrence of failures in this region are as follows:
- Higher random stress than expected
- Low safety factors
- Undetectable defects
- Abuse
- Natural failures
- Human errors

(c) Finally, during the wear-out period, the hazard rate increases with time t. Some of the reasons for the occurrence of failures, in this region are wear from aging, wrong overhaul practices, wear due to friction, corrosion, and creep; short designed-in life of the item/system under consideration; and poor maintenance practices.

Mathematically, the following equation can be used for representing above figure of bathtub hazard rate curve:

$$\lambda(t) = \alpha\beta(\alpha t)^{\beta-1} e^{(\alpha t)^{\beta}}$$

where, β is the shape parameter

α is the scale parameter

t is time.

$\lambda(t)$ is the hazard rate (time-dependent failure rate).

At $\beta = 0.5$, the above equation gives the shape of the bathtub hazard rate curve shown in figure above.

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