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MACHINE DESIGN

MECHANICAL ENGINEERING

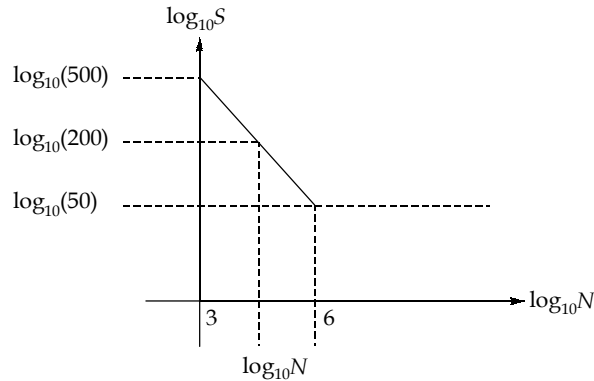
Date of Test : 11/10/2026

ANSWER KEY ➤

- | | | | | |
|--------|---------|---------|---------|---------|
| 1. (b) | 7. (d) | 13. (d) | 19. (b) | 25. (c) |
| 2. (b) | 8. (b) | 14. (b) | 20. (a) | 26. (c) |
| 3. (c) | 9. (d) | 15. (c) | 21. (c) | 27. (b) |
| 4. (c) | 10. (a) | 16. (c) | 22. (d) | 28. (b) |
| 5. (c) | 11. (d) | 17. (b) | 23. (a) | 29. (a) |
| 6. (c) | 12. (a) | 18. (d) | 24. (b) | 30. (c) |

DETAILED EXPLANATIONS

1. (b)

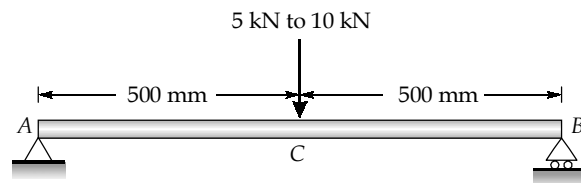


$$\frac{\log_{10}(500) - \log_{10}(50)}{3 - 6} = \frac{\log_{10}(200) - \log_{10}(50)}{\log_{10}(N) - 6}$$

$$\log_{10}(N) = 4.19385$$

$$N = 15626.07 \text{ cycles} \approx 15626 \text{ cycles}$$

2. (b)



$\therefore$ The bending moment will be maximum at mid span of shaft so point C will be critical section.

$\therefore$ At critical section, $M_{\max} = 5 \times 500 = 2500 \text{ Nm}$

$$\begin{aligned} (\sigma_b)_{\max} &= \frac{32M_{\max}}{\pi d^3} = \frac{32 \times 2500}{\pi d^3} \\ &= \frac{25464.79}{d^3} \text{ N/m}^2 \end{aligned}$$

$$M_{\min} = 2.5 \times 500 = 1250 \text{ N-m}$$

$$\begin{aligned} (\sigma_b)_{\min} &= \frac{32 \times M_{\min}}{\pi d^3} = \frac{32 \times 1250}{\pi d^3} \\ &= \frac{12732.39}{d^3} \text{ N/m}^2 \end{aligned}$$

$$(\sigma_b)_m = \frac{(\sigma_b)_{\max} + (\sigma_b)_{\min}}{2} = \frac{19098.59}{d^3} \text{ N/m}^2$$

$$(\sigma_b)_a = \frac{(\sigma_b)_{\max} - (\sigma_b)_{\min}}{2} = \frac{6366.2}{d^3} \text{ N/m}^2$$

$\therefore$ FOS for static loading and fluctuating loading are given

$$\therefore \frac{\frac{\sigma_m}{S_{yt}}}{\left(\frac{N_1}{N_2}\right)} + \frac{\frac{\sigma_a}{S_e}}{\left(\frac{N_2}{N_1}\right)} \leq 1$$

$$\frac{19098.59}{d^3 \left(\frac{550}{2} \right) \times 10^6} + \frac{6366.2}{d^3 \left(\frac{300}{4} \right) \times 10^6} \leq 1$$

$$d \geq 53.639 \text{ mm}$$

3. (c)

$$\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2} \right)^2 + (\tau_{xy})^2}$$

$$\sigma_{1,2} = \frac{200 + 100}{2} \pm \sqrt{(50)^2 + (50)^2}$$

$$\sigma_{1,2} = 150 \pm 70.7106$$

$$\sigma_1 = 220.71 \text{ MPa}$$

$$\sigma_2 = 79.289 \text{ MPa}$$

T.S.E.T for 2-D state of stress

$$\sigma_1^2 + \sigma_2^2 - 2\mu\sigma_1\sigma_2 \leq \left(\frac{S_{yt}}{N} \right)^2$$

$$\Rightarrow (220.71)^2 + (79.289)^2 - 2 \times 0.3 \times 220.71 \times 79.289 \leq \left(\frac{S_{yt}}{2} \right)^2$$

$$S_{yt} = 421.89 \text{ MPa} \approx 422 \text{ MPa}$$

4. (c)

Fatigue stress concentration factor,

$$k_f = \frac{1}{\text{Modified stress concentration factor}}$$

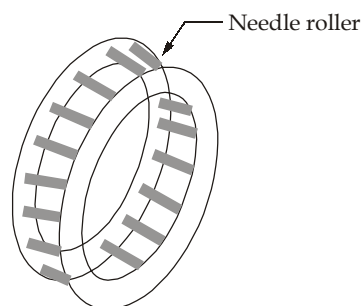
$$\Rightarrow k_f = \frac{1}{0.4} = 2.5$$

$$k_t = 3.5 \text{ (given)}$$

$$\text{Notch sensitivity factor, } q = \frac{k_f - 1}{k_t - 1} = \frac{2.5 - 1}{3.5 - 1}$$

$$q = 0.6$$

5. (c)



A Needle roller bearing is a special type of roller bearing which uses long and thin cylindrical rollers (like needles). Features of needle bearings are:

- $\frac{L}{D} \gg 1$
- Designed to bear heavy impact and fatigue load.
- Most preferred bearing where there is limitation in radial space.
- To minimize the radial space, both inner and outer race is absent.

Applications: Oscillating engines, impact loaded engines, piston pumps, power tool, etc.

6. (c)

7. (d)

$F_1 = 64 \text{ kN}$, $L_1 = 8 \times 10^6$ revolutions, $F_2 = ?$, $L_2 = 1 \times 10^6$ revolution

The expression for life of a bearing is given by,

$$L = \left(\frac{C}{F} \right)^K ; \text{ where } C \text{ and } K \text{ are constants.}$$

For ball bearing, $K = 3$ (Constant)

$$\begin{aligned} \Rightarrow F^3 \times L &= C \\ \Rightarrow F_1^3 \times L_1 &= C \text{ and } F_2^3 \times L_2 = C \\ \Rightarrow F_1^3 \times L_1 &= F_2^3 \times L_2 \\ \Rightarrow (64)^3 \times (8 \times 10^6) &= F_2^3 \times (1 \times 10^6) \\ \Rightarrow F_2 &= 128 \text{ kN} \end{aligned}$$

8. (b)

Load, $W = 2.5 \text{ kN}$

Bearing bore diameter, $D = 50 \text{ mm}$

Length of bearing, $L = 50 \text{ mm}$

$$\text{Pressure induced, } P = \frac{W}{LD} = \frac{2.5 \times 10^3}{(0.050 \times 0.050)} = 1 \times 10^6 \text{ Pa}$$

$$\text{Radial clearance, } C = 25 \mu\text{m} = 25 \times 10^{-3} \text{ mm}$$

$$\text{Bearing bore radius, } R = \frac{D}{2} = 25 \text{ mm}$$

Dynamic viscosity of lubricant, $\mu = 0.02 \text{ Pa.s}$

Rotational speed, $n = 2100 \text{ rpm} = 35 \text{ rps}$

$$\text{Sommerfeld number, } S = \frac{\mu n}{P} \times \left(\frac{R}{C} \right)^2 = \frac{0.02 \times 35}{1 \times 10^6} \times \left(\frac{25}{25 \times 10^{-3}} \right)^2 = 0.7$$

9. (d)

Explanation:

- To increase the length of a pressure vessel, circumferential lap joints are preferred because they are flexible.
- To increase the diameter of a pressure vessel, longitudinal butt joints are preferred because they are stronger.

10. (a)

Equivalent dynamic load $P = XVF_r + YF_a$

radial factor $X = 0.56$

thrust factor $Y = 1.79$

Since inner race rotates $V = 1$

$$P = 0.56 \times 1 \times 10 + 1.79 \times 4 \text{ kN} = 12.76 \text{ kN}$$

$$L_{10} = \frac{60 \times N \times L_{10h}}{10^6} = \frac{60 \times 1200 \times 20000}{10^6}$$

$$L_{10} = 1440 \text{ million revolution}$$

$$c = P(L_{10})^{1/3} = 12.76(1440)^{1/3}$$

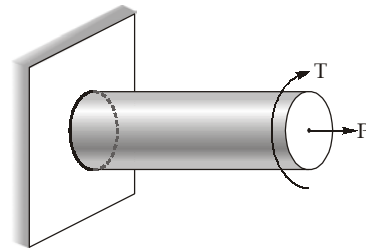
$$c = 144.09 \text{ kN} \simeq 144 \text{ kN}$$

11. (d)

$T = 800 \text{ N-m}$, $P = 120 \text{ kN}$, $d = 60 \text{ mm}$

$S_{ys} = 120 \text{ MPa}$, $\mu = 0.3$

$$\begin{aligned} \text{Axial stress due to } P; \sigma_x &= \frac{P}{A} = \frac{(120 \times 10^3)}{\left(\frac{\pi}{4} \times 60^2\right)} \\ &= 42.441 \text{ MPa} \end{aligned}$$



Torsional shear stress due to T ;

$$\tau_{xy} = \frac{16T}{\pi d^3} = \frac{16 \times 800 \times 10^3}{\pi \times (60)^3} = 18.863 \text{ MPa}$$

Using maximum distortion energy theory;

$$S_{yt} = \sqrt{3} S_{ys} = \sqrt{3} \times 120 = 207.85 \text{ MPa}$$

and

$$\frac{S_{yt}}{N} = \sqrt{\sigma_x^2 + 3\tau_{xy}^2}$$

$$\Rightarrow \frac{207.85}{N} = \sqrt{(42.441)^2 + 3 \times (18.863)^2}$$

$$\Rightarrow N = 3.88$$

12. (a)

$P_{\text{per}} = 0.4 \text{ MPa}$, $\mu = 0.15$, $D_0 = 140 \text{ mm}$, $R_0 = 70 \text{ mm}$

Since the clutch is old, so uniform wear theory will be applicable here.

Torque transmission using uniform wear theory is given as

$$T_f = \mu \cdot \pi \cdot P_{\text{per}} \cdot R_i (R_0^2 - R_i^2)$$

$$\text{Let } \frac{R_i}{R_0} = x \Rightarrow R_i = x R_0$$

$$\begin{aligned} \text{Now, } T_f &= \mu \cdot \pi \cdot P_{\text{per}} \cdot x R_0 \cdot (R_0^2 - x^2 R_0^2) \\ &= \mu \cdot \pi \cdot P_{\text{per}} \cdot R_0^3 \cdot (1 - x^2) \cdot x \end{aligned}$$

On differentiating the equation w.r.t. ' x '

$$\frac{dT_f}{dx} = \mu \cdot \pi \cdot P_{\text{per}} \cdot R_0^3 \cdot (1 - 3x^2)$$

$$\text{For maxima, } \frac{dT_f}{dx} = 0$$

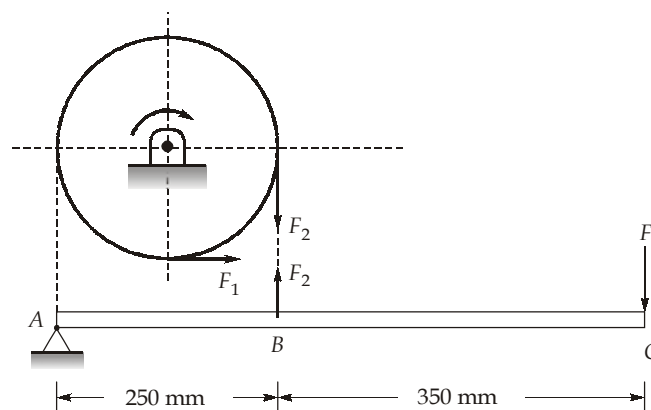
$$\Rightarrow 1 - 3x^2 = 0$$

$$\Rightarrow x = \frac{1}{\sqrt{3}}$$

Here, maximum torque is given as;

$$\begin{aligned}(T_f)_{\max} &= \mu \cdot \pi \cdot P_{\text{per}} \cdot R_0^3 \cdot \frac{1}{\sqrt{3}} \cdot \left[1 - \left(\frac{1}{\sqrt{3}} \right)^2 \right] \\ &= 0.15 \times \pi \times 0.4 \times (70)^3 \times \frac{1}{\sqrt{3}} \times \left[1 - \left(\frac{1}{\sqrt{3}} \right)^2 \right] \times 10^{-3} \\ &= 24.88 \text{ N-m} \simeq 24.9 \text{ N-m}\end{aligned}$$

13. (d)



$$\mu = 0.4, R = 250 \text{ mm}, \theta = 270^\circ$$

$$F_{\max} = 2 \text{ kN} = F_1$$

$$\frac{F_1}{F_2} = e^{\mu\theta}$$

$$\Rightarrow F_2 = \frac{F_1}{e^{\mu\theta}} = \frac{(2 \times 10^3)}{e^{\left(0.4 \times \frac{3\pi}{2}\right)}} = 303.67 \text{ N}$$

Moment about hinge point A;

$$\sum M_A = 0;$$

$$F_2 \times (250) - F \times (250 + 350) = 0$$

$$\Rightarrow (303.67) \times (250) - F \times (250 + 350) = 0$$

$$\Rightarrow F = 126.529 \text{ N} \simeq 127 \text{ N}$$

14. (b)

$$n_1 = 5, n_2 = 4, \text{ Power} = 12 \text{ kW}, N = 600 \text{ rpm}, \frac{D_0}{D_i} = \frac{R_0}{R_i} = 2.5, \mu = 0.18, P_{\text{per}} = 0.35 \text{ MPa}$$

$$\text{Power} = T \cdot \omega$$

$$(12 \times 10^3) = T_f \times \frac{2\pi(600)}{60}$$

$$\Rightarrow T_f = 190.986 \text{ N-m}$$

The number of pairs of contacting surfaces,

$$n = n_1 + n_2 - 1 = 5 + 4 - 1 = 8$$

Using uniform wear theory, friction torque is given as

$$\begin{aligned} T_f &= n \cdot \mu \cdot \pi \cdot P_{\text{per}} \cdot R_i (R_0^2 - R_i^2) \\ (190.986) &= 8 \times (0.18) \times \pi \times (0.35) \times R_i ((2.5 R_i)^2 - R_i^2) \times 10^{-3} \\ \Rightarrow R_i^3 &= 22975.3333 \\ \Rightarrow R_i &= 28.4285 \text{ mm} \\ \text{Outer radius, } R_0 &= (2.5)R_i = (2.5) \times (28.4285) = 71.07125 \text{ mm} \\ &\simeq 71.07 \text{ mm} \end{aligned}$$

15. (c)

Given : $P = 160 \text{ kN}$; $\tau_{\text{per}} = 70 \text{ MPa}$; $h = 10 \text{ mm}$

The total length (l) of the weld required to withstand the load of 160 kN is given by

$$\begin{aligned} P &= 0.707 h l \tau \\ \Rightarrow 160 \times 10^3 &= 0.707 (10) l (70) \\ \Rightarrow l &= 323.298 \text{ mm} \end{aligned}$$

$$\begin{aligned} \text{Weld length, } l_1 y_1 &= l_2 y_2 \\ \Rightarrow l_1 (300 - 71.8) &= l_2 (71.8) \\ \Rightarrow 228.2 l_1 &= 71.8 l_2 \quad \dots(i) \\ \text{Also, } l_1 + l_2 &= 323.298 \quad \dots(ii) \end{aligned}$$

On solving equation (i) and (ii);

$$\begin{aligned} l_1 &= 77.376 \text{ mm} \simeq 77.4 \text{ mm} \\ \text{and } l_2 &= 245.922 \text{ mm} \simeq 245.9 \text{ mm} \end{aligned}$$

16. (c)

Given : $P = 15 \text{ kN}$; $e = 600 + 200 = 800 \text{ mm}$; $\tau = 65 \text{ N/mm}^2$; $n = 5$

Primary shear force on rivets:

$$P'_1 = P'_2 = P'_3 = P'_4 = P'_5 = \frac{P}{n} = \frac{15 \times 10^3}{5} = 3000 \text{ N}$$

By Symmetry, the centre of gravity G is located at the centre of rivet-3. The radial distance of rivets centres from the centre of gravity G are as follow:

$$\begin{aligned} r_1 = r_2 = r_4 = r_5 &= \sqrt{150^2 + 200^2} = 250 \text{ mm} \\ r_3 &= 0 \end{aligned}$$

Secondary shear forces on rivets:

$$P'_3 = 0$$

$$\text{On rivet-1, P.e.} = \frac{P''_1}{r_1} [r_1^2 + r_2^2 + r_3^2 + r_4^2 + r_5^2]$$

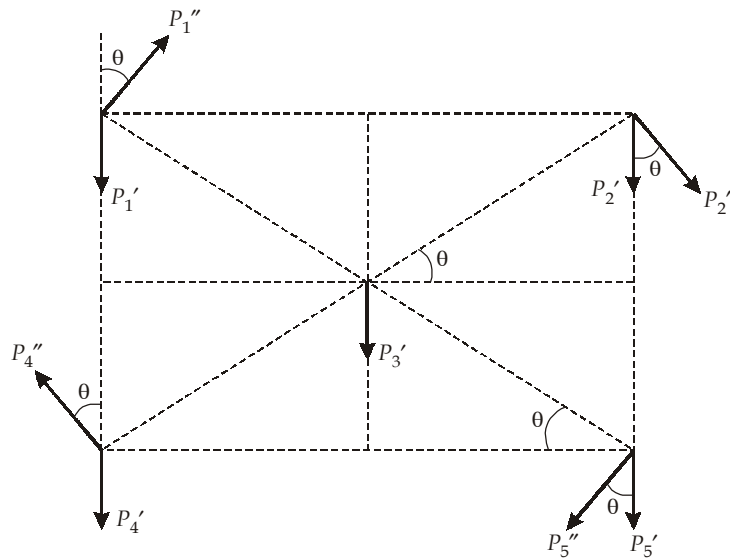
$$\Rightarrow (15 \times 10^3) \times 800 = \frac{P''_1}{250} [250^2 \times 4]$$

$$\Rightarrow P''_1 = 12000 \text{ N}$$

$$\text{Now, } P''_1 = P''_2 = P''_4 = P''_5 = 12000 \text{ N}$$

$$\tan \theta = \frac{150}{200}$$

$$\Rightarrow \theta = 36.87^\circ$$



Since, the angle between rivet-2 and rivet-5 is minimum. So they both are the worst loaded rivet. The resultant load on them is given as,

$$P_2 = P_5 = \sqrt{(3000)^2 + (12000)^2 + 2(3000)(12000)\cos(36.87^\circ)}$$

$$= 14512.06 \text{ N}$$

Now, equating the maximum shear force to the shear strength of the rivet

$$P_2 = A \cdot \tau$$

$$\Rightarrow 14512.06 = A \times 65$$

$$\Rightarrow A = 223.26 \text{ mm}^2 \approx 223.3 \text{ mm}^2$$

17. (b)

Given : $P = 4 \text{ kW}$; $n = 25 \text{ rps}$; $y = 0.3$; $b = 40 \text{ mm}$; $m = 4 \text{ mm}$; $z = 20$; $c_v = 1.5$; $\phi = 20^\circ$

$$\text{Power, } P = n \cdot 2\pi \cdot T$$

$$\Rightarrow T = \frac{P}{2\pi n} = \frac{4 \times 10^3}{2\pi(25)} = 25.465 \text{ Nm}$$

$$\text{Torque, } T = F_t \times \frac{D}{2} = F_t \frac{mz}{2}$$

$$\Rightarrow F_t = \frac{2T}{mz} = \frac{2 \times 25.465 \times 10^3}{4 \times 20} = 636.625 \text{ N}$$

For safe working of gear, $F_D \geq F_b$

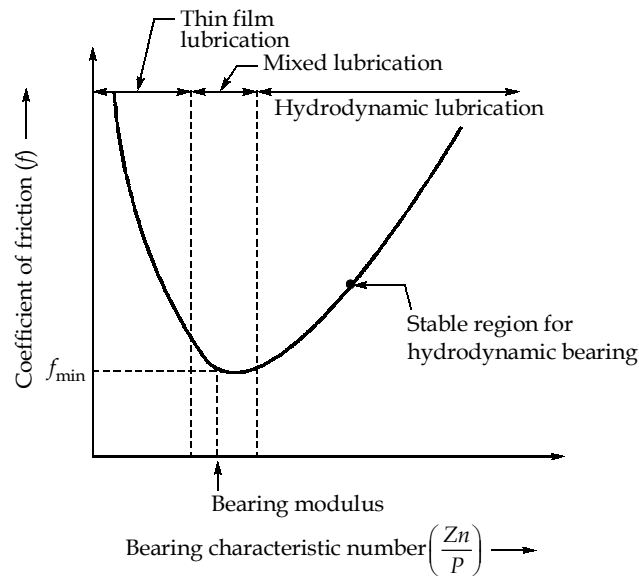
$$F_t \cdot c_v \leq \sigma_b \cdot m \cdot b \cdot y$$

$$\Rightarrow (636.625) \times 1.5 \leq \sigma_b \times (4) \times (40) \times (0.3)$$

$$\Rightarrow 19.89 \leq \sigma_b$$

$$\Rightarrow \sigma_b \geq 19.9 \text{ MPa} \approx 20 \text{ MPa}$$

18. (d)



- Bearing characteristic number $\left(\frac{Zn}{P}\right)$ is a dimensionless number. It defines the stability of hydrodynamic journal bearing and it helps to visualise the transition from boundary lubrication to thick film lubrication.
- In steady state operating condition, ϕ in order to avoid seizing the value of $\frac{Zn}{P}$ should be

atleast 5 to 6 times $\left(\frac{Zn}{P}\right)_{\text{minimum}}$ and for fluctuating condition it should be more than 15 times

$$\left(\frac{Zn}{P}\right)_{\text{minimum}}.$$

- Bearing characteristic number controls the design of journal bearing and it is dependent on the design parameters like operating condition (temperature, speed and load), geometrical parameters (length and diameter) and viscosity of the lubricant.

19. (b)

$$\begin{aligned}\text{Initial total KE} &= \frac{1}{2}MV^2 \times 1.2 \\ &= \frac{1}{2} \times 10^3 \times 35^2 \times 1.2 \text{ J} \\ &= 735000 \text{ J}\end{aligned}$$

Initial angular velocity of each wheel (ω_i) (pure rolling)

$$\omega_i = \frac{V}{R} = \frac{35}{0.25} = 140 \text{ rad/s}$$

$$\text{Angular deceleration } (\alpha) = \frac{a}{R} = \frac{7}{0.25} = \text{rad/s}^2$$

$$\omega_i^2 - \omega_f^2 = 2\theta\alpha$$

$$\theta = \frac{\omega_i^2 - \omega_f^2}{2\alpha} = \frac{140^2 - 0}{2 \times 28} = 350 \text{ rad}$$

From energy conservation, the friction between road and tyre is static, i.e. total kE of automobile will be goes in work done against breaking torque

$$kE = 4 \times T_f \theta$$

$$T_f = \frac{735000}{4 \times 350} = 525 \text{ N-m}$$

20. (a)

$P = 20 \text{ kN}$, $e = 150 \text{ mm}$, $\tau = 100 \text{ MPa}$, $l = 50 \text{ mm}$, $b = 100 \text{ mm}$.

s = size of weld (in mm)

t = Throat thickness (in mm)

$$\begin{aligned} A_{\text{throat}} &= A_t = 2hl \\ &= 2 \times 0.707 \times s \times 50 \\ &= 70.7s \text{ mm}^2 \end{aligned}$$

$$\text{Direct shear stress, } \tau_s = \frac{P}{A_t} = \frac{20 \times 10^3}{70.7s} = \frac{282.8854}{s} \text{ MPa}$$

$$\text{Polar MOI, } J = 2 \left[\frac{tl^3}{12} + tl \left(\frac{b}{2} \right)^2 \right]$$

$$= (tl) \left[\frac{l^2}{6} + \frac{b^2}{2} \right]$$

$$= (0.707s.l) \left(\frac{l^2 + 3b^2}{6} \right)$$

$$= (0.707s)(50) \left(\frac{50^2 + 3 \times 100^2}{6} \right) \text{ mm}^4 = 191479.1667s \text{ mm}^4$$

$$r_2 = \sqrt{\left(\frac{b}{2} \right)^2 + \left(\frac{l}{2} \right)^2} = \sqrt{\left(\frac{100}{2} \right)^2 + \left(\frac{50}{2} \right)^2} = 55.9017 \text{ mm}$$

$$\tau_{\text{sec}} = \tau_t = \frac{Pe.r_2}{J}$$

$$\sigma = \frac{20 \times 10^3 \times 150 \times 55.9017}{191479.1667s} = \frac{875.8399}{s} \text{ MPa}$$

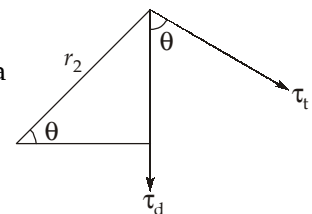
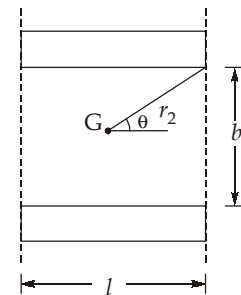
$$\cos \theta = \frac{(l/2)}{r_2} = \frac{(50/2)}{55.9017}$$

$$\tau_{\text{per}} = \sqrt{\tau_d^2 + \tau_t^2 + 2.\tau_d.\tau_t.\cos \theta}$$

$$= \sqrt{\left(\frac{282.8854}{s} \right)^2 + \left(\frac{875.8399}{s} \right)^2 + 2 \cdot \left(\frac{282.8854}{s} \right) \cdot \left(\frac{875.8399}{s} \right) \left(\frac{25}{55.9017} \right)}$$

$$100 = \frac{1033.7916}{s}$$

$$s = 10.34 \text{ mm}$$



21. (c)

$$T_{\max} = 6 \text{ kN}, T_{\min} = 2 \text{ kN}, R = 1 \text{ m}, L = 1.2 \text{ m}$$

$$\text{Maximum Torque} = (T_{\max} - T_{\min})R$$

$$= (6 - 2)1 = 4 \text{ kN-m}$$

$$\text{Maximum bending moment} = \frac{WL}{4} = \frac{8 \times 1.2}{4} = 2.4 \text{ kN-m}$$

$$\tau = \frac{16T}{\pi d^3} = \frac{16 \times 4 \times 10^6}{\pi d^3}$$

As per MDET,

$$\sqrt{\sigma_x^2 + 3\tau_{xy}^2} \leq \left(\frac{S_{yt}}{FOS} \right)$$

$$\frac{10^6}{\pi d^3} \left[\sqrt{(32 \times 2.4)^2 + 3(16 \times 4)^2} \right] \leq \frac{400}{2}$$

$$d \geq 59.8729$$

$$d_{\min} = 59.873 \text{ mm}$$

22. (d)

$$\tau_{\text{allow}} \geq \tau_{\text{ind}} = \frac{16T}{\pi d^3}$$

$$T \leq \frac{\tau_{\text{allow}} \pi d^3}{16} \leq \frac{60 \times \pi d^3}{16}$$

$$\theta_{\text{allow}} \geq \frac{TL}{GJ}$$

$$\frac{\theta_{\text{allow}}}{L} \geq \frac{T}{G \frac{\pi}{32} d^4}$$

$$\frac{0.02\pi}{180} \geq \frac{60 \cdot \frac{\pi d^3}{16}}{80 \times 10^3 \cdot \frac{\pi d^4}{32}} = \frac{60 \times 1}{80 \times 10^3 \times \frac{d}{2}}$$

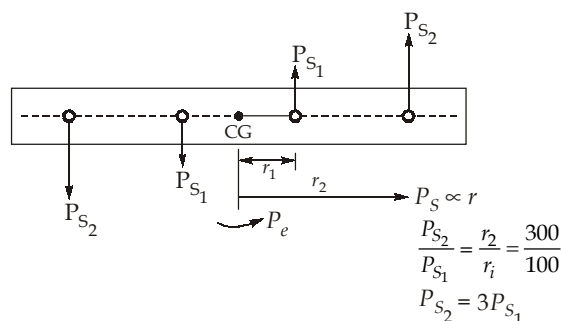
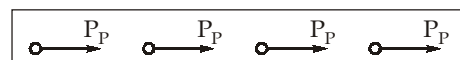
$$d \geq 4.29 \text{ mm}$$

$$d_{\min} = 4.3 \text{ mm}$$

23. (a)

Primary shear load (P_s), $P_p = \frac{P}{4}$

Secondary shear load



$$P_e = 2 \times 300 \times P_{S_2} + 200 \times P_{S_1} \times 2 \times 100$$

$$P \times 500 = 600 \times P_{S_2} + 200 \times \frac{P_{S_2}}{3}$$

$$5P = 6P_{S_2} + \frac{2}{3}P_{S_2} = \frac{20}{3}P_{S_2}$$

$$P_{S_2} = \frac{15P}{20}$$

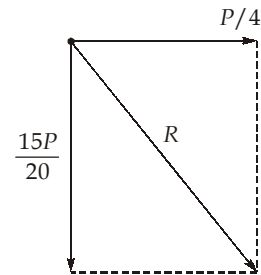
Rivet 1 and 4 are heavily loaded so,

$$\text{Resultant load, } R = \sqrt{\left(\frac{P}{4}\right)^2 + \left(\frac{15P}{20}\right)^2}$$

$$= P \sqrt{\frac{1}{16} + \frac{225}{400}} = \sqrt{\frac{5}{8}} \cdot P$$

$$\tau \cdot A = \sqrt{\frac{5}{8}} \cdot P$$

$$P = \sqrt{\frac{8}{5}} \times 80 \times \frac{\pi}{4} \times 20^2 = 31790.68 \text{ N} = 31.8 \text{ kN}$$



24. (b)

$$\omega_i = \frac{\pi \times 1000}{30} = 104.7198 \text{ rad/s}$$

$$\omega_f = \omega_i \times \alpha t$$

$$0 = 104.7198 + \alpha (10)$$

$$\alpha = -10.472 \text{ rad/s}^2$$

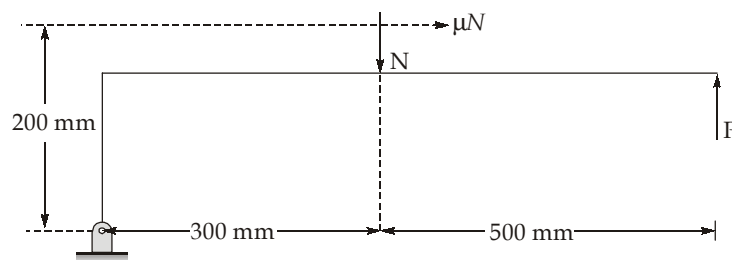
$$T_f = I\alpha$$

$$= 100 \times 0.15^2 \times 10.472 = 23.5619 \text{ N-m}$$

$$\mu NR = T_f$$

$$N = \frac{T_f}{\mu R} = \frac{23.5619}{0.3 \times 0.2} = 392.699 \text{ N}$$

FBD of lever



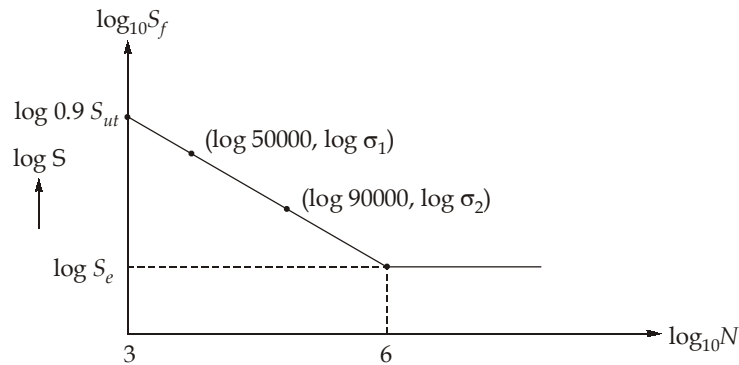
Taking moment about the hinge,

$$800P = 300 N + 200 \mu N = N(300 + 0.3 \times 200)$$

$$P = \frac{392.699 \times 360}{800}$$

$$\therefore P = 176.71 \text{ N} \approx 176.7 \text{ N}$$

25. (c)



Fatigue strength (σ_1) corresponding to 50,000 cycle (initial)

$$\frac{\log \frac{0.9 S_{ut}}{S_e}}{3-6} = \frac{\log \frac{\sigma_1}{300}}{\log 50000-6}$$

$$\Rightarrow \frac{\log \left(\frac{0.9 \times 600}{300} \right)}{-3} = \frac{\log \frac{\sigma_1}{300}}{\log 50000-6}$$

$$\Rightarrow \sigma_1 = 387.1034 \text{ MPa}$$

Fatigue strength (σ_2) corresponding to 90,000 cycle (final)

$$\frac{\log \frac{0.9 \times 600}{300}}{3-6} = \frac{\log \frac{\sigma_2}{300}}{\log 90000-6}$$

$$\sigma_2 = 368.2185 \text{ MPa}$$

$$\% \text{ change} = \frac{\sigma_2 - \sigma_1}{\sigma_1} \times 100$$

$$= \frac{368.2185 - 387.1034}{387.1034} \times 100$$

$$= -4.8785\% \approx -4.9\%$$

26. (c)

$W = 2000 \text{ N}$, $Z = 0.02 \text{ Pa.s}$, $D = L = 50 \text{ mm}$, $C = 2 \times 0.025 = 0.05 \text{ mm}$

$$P = \frac{W}{DL} = \frac{2000}{50 \times 50} = 0.8 \text{ MPa}$$

Power loss = μWV

$$= (2\pi^2) \cdot \left(\frac{Zn}{P} \right) \left(\frac{D}{C} \right) W \cdot \frac{D}{2} (2\pi n)$$

$$= (2\pi^3) \cdot \left(\frac{Zn^2}{P} \right) \left(\frac{D}{C} \right) WD$$

$$50 = (2\pi^3) \cdot \left(\frac{0.02n^2}{0.8 \times 10^6} \right) \left(\frac{50}{0.05} \right) (2000)(0.05)$$

$$n = 17.96 \text{ rev/s}$$

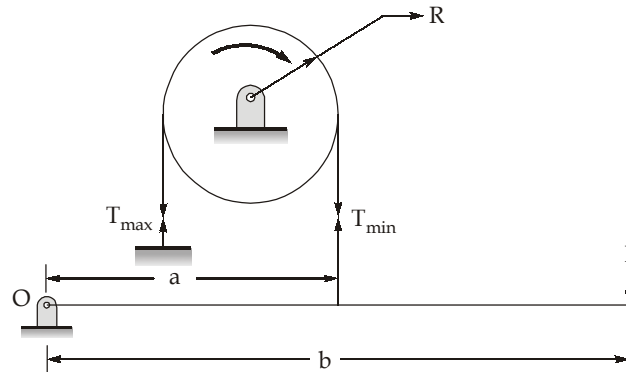
$$N = 60 n = 1077.5 \text{ rpm}$$

$$S = \left(\frac{Zn}{P} \right) \left(\frac{D}{C} \right)^2 = \left(\frac{0.02 \times 17.96}{0.8 \times 10^6} \right) \left(\frac{50}{0.05} \right)^2$$

$$= 0.4489 \simeq 0.45$$

27. (b)

Case 1 : CW rotation



$$\Sigma M_0 = 0$$

$$Fb = aT_{\min}$$

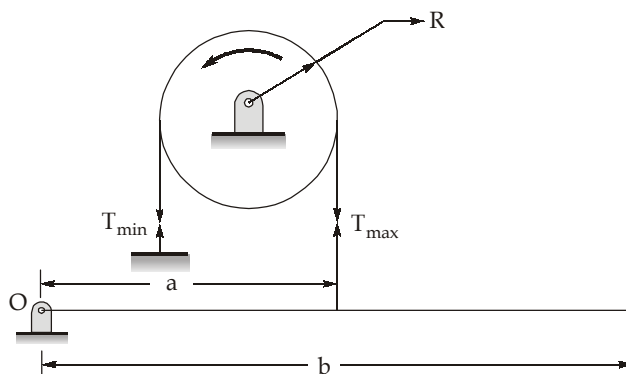
$$T_{\min} = \frac{Fb}{a}$$

$$T_{\max} = T_{\min} \cdot e^{\mu\theta}$$

$$\text{Brake torque} = (T_{\max} - T_{\min})R$$

$$= T_{\min} (e^{\mu\theta} - 1)R = \frac{Fb}{a} (e^{\mu\theta} - 1)R$$

Case 2 : ACW rotation



$$\Sigma M_0 = 0$$

$$Fb = aT_{\max}$$

$$T_{\max} = \frac{Fb}{a}$$

$$T_{\min} = \frac{T_{\max}}{e^{\mu\theta}}$$

$$\text{Brake torque} = (T_{\max} - T_{\min})R = T_{\max} \left(1 - \frac{1}{e^{\mu\theta}}\right)R$$

$$= \frac{Fb}{a} \left(\frac{e^{\mu\theta} - 1}{e^{\mu\theta}}\right)R$$

$$\frac{(\text{brake torque})_{CW}}{(\text{brake torque})_{ACW}} = \frac{1}{e^{\mu\theta}} < 1$$

28. (b)

Torque transmitted by each pair of contact surface

$$= \frac{30p}{2\pi N} = \frac{30 \times 50 \times 10^3}{2\pi \times 2000}$$

$$= 119.3662 \text{ N-m} = 119366.2 \text{ N-mm}$$

$$T = \pi \mu P_{\text{per}} r_i (r_0^2 - r_i^2)$$

$$P_{\text{per}} = \frac{T}{\pi \mu r_i (r_0^2 - r_i^2)} = \frac{119366.2}{\pi \times 0.3 \times 150 (300^2 - 150^2)}$$

$$= 0.0125 \text{ MPa}$$

29. (a)

Given : $\sigma_x = 90 \text{ MPa}$, $\sigma_y = 30 \text{ MPa}$, $\tau_{xy} = 20 \text{ MPa}$, $S_{yt} = 120 \text{ MPa}$, $\mu = 0.3$

$$\text{Principal stress, } \sigma_{1,2} = \left(\frac{\sigma_x + \sigma_y}{2}\right) \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$= \left(\frac{90 + 30}{2}\right) \pm \sqrt{\left(\frac{90 - 30}{2}\right)^2 + 20^2}$$

$$= 60 \pm 36.06$$

$$\Rightarrow \sigma_1 = 96.06 \text{ MPa and } \sigma_2 = 23.94 \text{ MPa}$$

Using Tresca's maximum shear stress theory;

$$\max \left\{ \left| \frac{\sigma_1 - \sigma_2}{2} \right|, \left| \frac{\sigma_1}{2} \right|, \left| \frac{\sigma_2}{2} \right| \right\} = \frac{S_{yt}}{2N}$$

$$\Rightarrow \max \left\{ \left| \frac{96.06 - 23.94}{2} \right|, \left| \frac{96.06}{2} \right|, \left| \frac{23.94}{2} \right| \right\} = \frac{120}{2N}$$

$$\Rightarrow \max\{36.6, 48.03, 11.97\} = \frac{120}{2N}$$

$$\Rightarrow 48.03 = \frac{120}{2N}$$

$$\Rightarrow N = 1.249$$

$$\Rightarrow \simeq 1.249$$

Using Von-Mises maximum distortion energy theory;

$$\sigma_1^2 + \sigma_2^2 - \sigma_1 \sigma_2 = \left(\frac{S_{yt}}{N}\right)^2$$

$$\Rightarrow 96.06^2 + 23.94^2 - 96.06 \times 23.94 = \left(\frac{120}{N} \right)^2$$

$$\Rightarrow N = 1.385 \simeq 1.386$$

30. (c)

$$W_{\max} = 500 \text{ kN}, W_{\min} = 300 \text{ kN}, S_{ut} = 850 \text{ MPa}, S_{yt} = 650 \text{ MPa}, \sigma_e = 450 \text{ MPa}, N = 2,$$

$$\text{Mean load, } W_m = \frac{W_{\max} + W_{\min}}{2} = \frac{500 + 300}{2} = 400 \text{ kN}$$

$$\text{Mean stress, } \sigma_m = \frac{W_m}{b^2} = \frac{400 \times 10^3}{b^2}$$

$$\text{load amplitude, } W_a = \left| \frac{W_{\max} - W_{\min}}{2} \right| = \left| \frac{500 - 300}{2} \right| = 100 \text{ kN}$$

$$\text{stress amplitude, } \sigma_a = \frac{W_a}{b^2} = \frac{100 \times 10^3}{b^2}$$

Using Soderberg relation;

$$\frac{\sigma_m}{S_{yt}} + \frac{\sigma_a}{\sigma_e} = \frac{1}{N}$$

$$\Rightarrow \frac{400 \times 10^3}{b^2 \times 650} + \frac{100 \times 10^3}{b^2 \times 450} = \frac{1}{2}$$

$$\Rightarrow b = 40.92937 \simeq 40.93 \text{ mm}$$

