CLASS TEST

S.No.: 01DLSK_ME_ABCDEFGH_20726

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MACHINE DESIGN

MECHANICAL ENGINEERING

Date of Test : 20/07/2026

ANSWER KEY

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

DETAILED EXPLANATIONS

1. (b)

$$T = \frac{P \times 60}{2\pi N} = \frac{12 \times 10^3 \times 60}{2\pi \times 1440} = 79.577 \text{ N} - \text{m}$$

$$T = P \times \frac{d}{2}$$

$$P = 1768.377 \text{ N}$$

$$P_{eff} = P \times FOS = P \times 5 = 8841.885 \text{ N} \quad \text{(where, } FOS \text{ is factor of safety = 5)}$$

So,

$$P_{eff} = m \times b \times \sigma_b \times Y$$

$$= m \times 10 \times m \times 200 \times 0.308$$

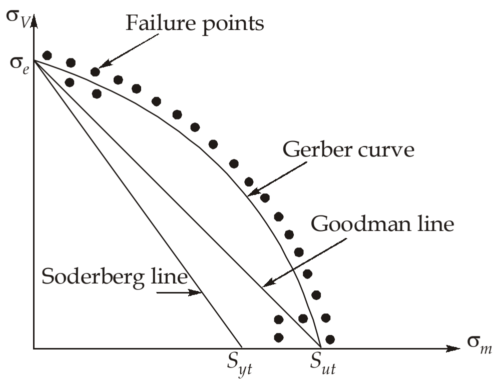
$$8841.885 = 10 \text{ m}^2 \times 200 \times 0.308$$

$$m = 3.788 \text{ mm}$$

But, module should be an integer i.e., m = 4 mm

2. (a)

Theories based on the Soderberg line or the Goodman line, as failure criteria are conservative theories. This results in increased dimensions of the components, thus they become uneconomical. The Gerber curve takes the mean path through failure points. It is, therefore, more accurate and economical in predicting fatigue failure.



3. (c)

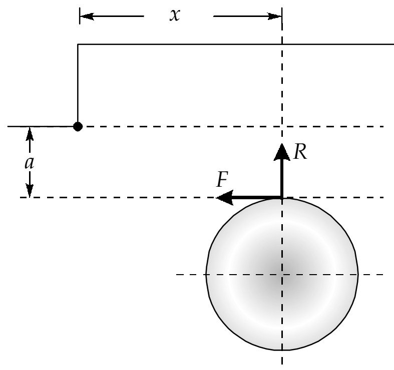
Principal stresses are given as

$$\sigma_1 = \frac{16}{\pi d^3}\left[M + \sqrt{M^2 + T^2}\right]$$

$$\sigma_2 = \frac{16}{\pi d^3}\left[M - \sqrt{M^2 + T^2}\right]$$

In σ_2, $M < \sqrt{M^2 + T^2}$ so σ_2 will be negative. σ_1 is always positive. So, both the principal stresses are of opposite nature.

4. (b)



$$F = \mu \cdot R$$
$$R \cdot x > F \cdot a$$
$$R \cdot x > \mu R \cdot a$$
$$x > \mu a$$

5. (d)

Intensity of pressure, p = 0.2 MPa
Outer diameter = 400 mm
Inner diameter = 200 mm
Axial force = p × Area

$$= 0.2 \times 10^6 \times \frac{\pi}{4}(0.4^2 - 0.2^2)$$
$$= 6000\,\pi$$

6. (c)

$$\text{Axial pitch} = \frac{\text{Circular pitch}}{\tan\alpha} = \frac{p_c}{\tan\alpha}$$

7. (b)

$$k_{eq}\ \text{(parallel)} = k + k = 2k$$

8. (c)

$$k_t = 1 + 2\left(\frac{a}{b}\right) = 1 + 2(1) = 3$$

So,
$$\sigma_{max} = 3 \times \left(\frac{50 \times 10^3}{(40-10)\times(10)}\right) = 500\,\text{MPa}$$

10. (d)

11. (b)

Standard endurance limit,

$$S_e = 0.5\ S_{ut}$$
$$(S_e)_{axial} = (S_e)_{bending} \times 0.8$$
$$= 0.5 \times 0.8 \times S_{ut}$$
$$= 0.5 \times 0.8 \times 440 = 176\ \text{MPa}$$

Corrected endurance limit,

$$(S'_e)_{\text{axial}} = k_a\, k_b\, k_c\, k_d\, (S_e)_{\text{axial}}$$

$$= \frac{k_a k_b k_c}{k_f} \times (S_e)_{axial}$$

$$q = \frac{k_f - 1}{k_t - 1}$$

$$\Rightarrow \quad 0.8 = \frac{k_f - 1}{2.5 - 1}, \; k_f = 2.2$$

$$(S'_e)_{\text{axial}} = \frac{0.67 \times 0.85 \times 0.897}{2.2} \times 176$$

$$= 40.86 \text{ MPa}$$

Permissible stress amplitude,

$$\sigma_a = \frac{(S'_e)_{axial}}{N} = \frac{40.86}{2} = 20.43 \text{ MPa}$$

$$\sigma_a = \frac{P}{(w-d)t}; \; t = \frac{30000}{40 \times 20.43} = 36.71 \text{ mm}$$

12. (b)

$$\sigma_{\text{min}} = 100 \text{ MPa}, \; \sigma_{\text{max}} = 150 \text{ MPa}$$

$$\text{Stress ratio } (R) = \frac{\sigma_{\text{min}}}{\sigma_{\text{max}}} = \frac{100}{150} = 0.67$$

$$\text{Amplitude ratio } (A) = \frac{\sigma_a}{\sigma_m} = \frac{\sigma_{\text{max}} - \sigma_{\text{min}}}{\sigma_{\text{min}} + \sigma_{\text{min}}}$$

$$= \frac{150 - 100}{150 + 100} = \frac{50}{250} = 0.2$$

13. (b)

Given : $(\sigma_x)_{\text{min}} = 40$, $(\sigma_x)_{\text{max}} = 100$

$$(\sigma_x)_m = \frac{40 + 100}{2} = 70 \text{ MPa}$$

$$(\sigma_x)_a = \frac{100 - 40}{2} = 30 \text{ MPa}$$

$(\sigma_y)_{\text{min}} = 10$ MPa, $(\sigma_y)_{\text{max}} = 80$ MPa

$$(\sigma_y)_m = \frac{(\sigma_y)_{\text{max}} + (\sigma_y)_{\text{min}}}{2} = 45 \text{ MPa}$$

$$(\sigma_y)_a = \frac{(\sigma_y)_{\text{max}} - (\sigma_y)_{\text{min}}}{2} = 35 \text{ MPa}$$

From distortion energy theory,

$$\sigma_m = \sqrt{(\sigma_x)_m^2 + (\sigma_y)_m^2 - (\sigma_x)_m \times (\sigma_y)_m}$$

$$= \sqrt{(70)^2 + (45)^2 - 70 \times 45} = 61.44 \text{ MPa}$$

$$\sigma_a = \sqrt{(\sigma_x)_a^2 + (\sigma_y)_a^2 - (\sigma_x)_a \times (\sigma_y)_a}$$

$$= \sqrt{(30)^2 + (35)^2 - 30 \times 35} = 32.79 \text{ MPa}$$

From Goodman diagram,

$$\frac{\sigma_m}{S_{ut}} + \frac{\sigma_a}{S_e} = \frac{1}{N}$$

$$\Rightarrow \quad \frac{61.44}{660} + \frac{32.79}{270} = \frac{1}{N}$$

$$N = 4.66$$

14. (b)

$$D_p = mT_p = 8 \times 16 = 128 \text{ mm}$$
$$T_G = 4 \times T_p = 4 \times 16 = 64$$
$$D_G = mT_G = 8 \times 64 = 512 \text{ mm}$$

We know that tooth form factor is:

$$y = 0.154 - \frac{0.912}{T}$$

For pinion

$$y_p = \left(0.154 - \frac{0.912}{16}\right) = 0.097$$

$$y_p \times \sigma_p = 0.097 \times 85 = 8.245$$

For gear,

$$y_G = 0.154 - \frac{0.912}{64} = 0.13975$$

$$y_G \times \sigma_G = 0.13975 \times 103 = 14.394$$

As $y_p \times \sigma_p$ is less than $y_G \times \sigma_G$ then design is to be done with respect to pinion,

$$P_{\text{eff}} = \pi \times y_p \times \sigma_p \times m \times b = \pi \times 8.245 \times 8 \times 90$$
$$= 18649.75 \text{ N}$$

$$P_{\text{eff}} = \frac{P_t}{C_V}$$

$$P_t = P_{\text{eff}} \times C_V$$

$$\text{Power} = P_t \times V$$

where,

$$V = \frac{\pi \times D_P \times N}{60} = \frac{\pi \times 0.128 \times 500}{60} = 3.351 \text{ m/s}$$

So,

$$\text{Power} = 18649.75 \times \frac{3}{3 + 3.351} \times 3.351$$

$$= 29520.69 \text{ W} = 29.521 \text{ kW}$$

15. (b)

As per given information, $P = 2400$ N

Reliability of the system, $R = 0.82$

Reliability of each bearing $= [0.82]^{1/4} = 0.95159$

$$\left[\frac{L}{L_{10}}\right] = \left[\frac{\ln\left(\frac{1}{R}\right)}{\ln\left(\frac{1}{R_{90}}\right)}\right]^{\frac{1}{1.17}}$$

$$\frac{5}{L_{10}} = \left[\frac{\ln\left(\frac{1}{0.95159}\right)}{\ln\left(\frac{1}{0.9}\right)}\right]^{\frac{1}{1.17}}$$

$$\frac{5}{L_{10}} = 0.5254$$

$$L_{10} = 9.5163 \text{ million revolutions}$$

$$\left(\frac{C}{P}\right)^3 = 9.5163$$

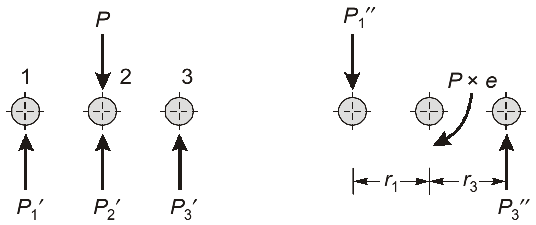
$$C = (9.5163)^{1/3} \times 2400$$

$$C = 5085.89 \text{ N}$$

16. (d)

Permissible shear stress, $\tau = \dfrac{\tau_y}{N} = \dfrac{0.5\,\sigma_y}{N} = \dfrac{0.5 \times 380}{3} = 63.33\,\text{MPa}$

The primary and secondary shear forces are shown in figure (a) and (b). The centre of gravity of three bolts will be at the centre of the bolt (2).



(a) (b)

$$P_1' = P_2' = P_3' = \frac{P}{3} = \frac{10000}{3} = 3333.33\,\text{N}$$

$$P_1'' = P_3'' = \frac{Per_1}{r_1^2 + r_3^2} = \frac{10000 \times (200 + 30 + 75) \times 75}{75^2 + 75^2} = 20333.33\,\text{N}$$

Resultant shear force on bolt (3) is maximum.

$$P_3 = P_3' + P_3''$$
$$= 3333.33 + 20333.33$$
$$= 23666.67 \text{ N}$$

Size of bolt, $\tau = \dfrac{P_3}{A} = \dfrac{23666.67}{\frac{\pi}{4} d_c^2}$ [Given $\tau_{yt} = 0.5\sigma_{yt}$]

$\Rightarrow$ $d_c = 21.8$ mm

17. (a)

Shear diagonal or line of pure shear is the locus of all points, corresponding to pure shear stress.

$$\sigma_1 = -\sigma_2 = \tau_{12}$$

$$\frac{\sigma_1}{\sigma_2} = -1 = -\tan 45°$$

In first and third quadrant, both stresses are of same nature.
In second and fourth quadrant, both stresses are of opposite nature.

18. (a)

$$K_2 = \frac{G\times(2d)^4}{8\times(2D)^3\times 2N} = \frac{Gd^4}{8D^3N} = K$$

$\therefore$ $$K_2 = K$$

19. (a)

$$\sigma_e = \frac{1}{2}\times 1.07 = 0.535\ \text{GN/m}^2 = 535\ \text{MPa}$$

for reversed axial loading, $\sigma_m = 0$
and variable stress

$$\sigma_v = \frac{F}{A} = \frac{4F}{\pi d^2} = \frac{4\times 180\times 10^3}{\pi d^2}$$

$$= \frac{229183.12}{d^2}$$

Substituting in design equation of Soderberg (for steel)

$$\frac{1}{N} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v}{\sigma_e}$$

$$\frac{1}{2} = 0 + \frac{229183.12}{d^2\times 535\times 10^6}$$

$\Rightarrow$ $$d = 0.02927\ \text{m}$$
$$= 29.27\ \text{mm}$$
$$\approx 30\ \text{mm}$$

20. (c)

Given, $(L_{80})_1 = 380$ MR

$(P_e)_1 = 12$ kN

We know that,

$$\because \frac{L_{80}}{L_{90}} = \left[\frac{\ln\left(1/0.8\right)}{\ln\left(1/0.9\right)}\right]^{1/1.17}$$

$$L_{80} = 1.90\ L_{90}$$

$$(L_{90})_1 = \frac{(L_{80})_1}{1.90} = \frac{380}{1.90} = 200\ \text{MR}$$

Rated life of bearing is bearing life with 90% reliability

$$L_{90} \;\alpha\; \left(\frac{1}{P_e}\right)^3$$

$$\frac{(L_{90})_1}{(L_{90})_2} = \left[\frac{(P_e)_2}{(P_e)_1}\right]^3 \qquad \because (P_e)_2 = 2(P_e)_1$$

$$\frac{(L_{90})_1}{(L_{90})_2} = (2)^3$$

$$(L_{90})_2 = \frac{200}{8} = 25 \text{ MR}$$

21. (c)

$$\sigma_{max} = \frac{32M_{max}}{\pi d^3} = \frac{32\times 500}{\pi d^3} = \frac{5092.96}{d^3}\text{MPa}$$

$$\sigma_{min} = \frac{32M_{min}}{\pi d^3} = \frac{32\times(-200)}{\pi d^3} = \frac{-2037.18}{d^3}\text{MPa}$$

$$\sigma_m = \frac{\sigma_{max}+\sigma_{min}}{2} = 1527.89\text{MPa}$$

$$\sigma_v = \frac{\sigma_{max}-\sigma_{min}}{2} = \frac{3565.07}{d^3}\text{MPa}$$

For ductile material-using Soderberg equation

$$\frac{1}{N} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v}{\sigma_e}$$

$$\frac{1}{2.5} = \frac{1527.89}{400d^3} + \frac{3565.07}{\frac{540}{2}\times d^3}$$

$$\Rightarrow \quad d^3 = 2.5\left[\frac{1527.89}{400} + \frac{3565.07\times 2}{540}\right]$$

$$\Rightarrow \quad d \simeq 3.5 \text{ mm}$$

22. (a)

Given: $S_{yt} = S_{yc} = 400$ MPa

$$\sigma = \begin{bmatrix} 100 & 20 \\ 20 & 50 \end{bmatrix}$$

$\sigma_x = 100$, $\sigma_y = 50$, $\tau_{xy} = 20$

Principal stresses, σ_1, σ_2

$$\sigma_{1,2} = \frac{1}{2}\left[(\sigma_x+\sigma_y) \pm \sqrt{(\sigma_x-\sigma_y)^2 + 4\tau_{xy}^2}\right]$$

$$\sigma_{1,2} = \frac{1}{2}\left[150 \pm \sqrt{50^2 + 4\times 20^2}\right] = \frac{1}{2}[150 \pm 64.03]$$

$$\sigma_{1,2} = 107.015 \text{ MPa}, 42.984 \text{ MPa}$$

From maximum shear stress theory,

$$\text{Max }[(\sigma_1 - \sigma_2), (\sigma_1), (\sigma_2)] \le \frac{S_{yt}}{N_1}$$

$$\Rightarrow \quad 107.015 \le \frac{400}{N_1}$$

$$N_1 \le 3.737$$

From maximum distortion energy theory,

$$\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2 \le \left(\frac{S_{yt}}{N_2}\right)^2$$

$$\Rightarrow \quad 8699.9 \le \left(\frac{400}{N_2}\right)^2$$

$$\Rightarrow \quad N_2 = 4.288$$

The ratio of factor of safety = $\dfrac{N_1}{N_2} = 0.87$

23. (c)

Load sharing in upper bolt will be W_1 each and in lower bolt is W_2.

If moment is taken about XX, then

$$W \times 400 = 2W_1 \times 400 + W_2 \times 50$$

Also, proportionate load is taken by bolts ...(i)

$$\frac{2W_1}{400} = \frac{W_2}{50}$$

$$\Rightarrow \quad W_1 = 4W_2 \quad \text{...(ii)}$$

Load capacity of each upper bolt

$$W_1 = \sigma \times A$$

$$= 60 \times 10^6 \times 353 \times 10^{-6} = 21180 \text{ N}$$

$$W_2 = \frac{21180}{4} = 5295 \text{ N}$$

From equation (i)

$$W = \frac{2 \times 21180 \times 400 + 5295 \times 50}{400} = 43.021 \text{ kN}$$

24. (c)

$$K_w = \frac{4C-1}{4C-4} + \frac{0.615}{C}; \quad C - \text{spring index} = \frac{D}{d}$$

K_w = Wahl's stress factor

D = Mean diameter of the spring coil

d = Diameter of spring wire

$$D = D_0 - d = 75 - 6 = 69 \text{ mm}$$

$$C = \frac{69}{6} = 11.5$$

$$K_w = \frac{(4\times11.5)-1}{(4\times11.5)-4}+\frac{0.615}{11.5}=1.124$$

$$\tau_{max} \text{ (max. shear stress)} = \frac{8K_wFC}{\pi d^2}$$

$$F = \frac{350\times\pi\times6^2}{8\times11.5\times1.124} = 382.7950 \text{ N}$$

$$\delta = \frac{8FC^3N}{Gd}$$

δ – Deflection

F – Applied load

N – Number of turn

G – Modulus of rigidity

$$\frac{\delta}{N} = \frac{8FC^3}{Gd}$$

$$\Rightarrow \quad \frac{8\times382.7950\times11.5^3}{84\times10^3\times6} = 9.24 \text{ mm}$$

25. (a)

$$F_m = \sqrt[3]{\frac{F_1^3t_1+F_2^3t_2+F_3^3t_3}{T}}$$

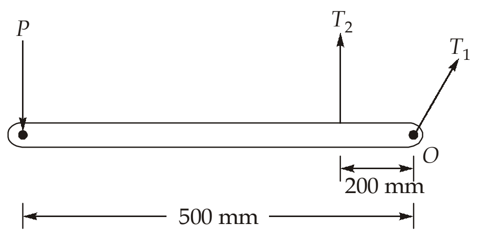
$$= \sqrt[3]{\frac{3^3(1)+(1.5)^3(2)+(0.5)^3(3)}{1+2+3}}$$

$$= 1.785 \text{ kN}$$

26. (c)

Maximum force exerted by the cylinder,

$$P = p\times A = 0.7\times\frac{\pi}{4}\times50^2 = 1374.45 \text{ N}$$



$\Sigma M_0 = 0$

$$P\times500 = T_2\times200$$

$$T_2 = 1374.45\times\frac{500}{200} = 3436.12 \text{ N}$$

$$\frac{T_1}{T_2} = e^{\mu\theta} = e^{0.25\times\frac{200\pi}{180}} = 2.393$$

$$T_1 = 8223.59 \text{ N}$$

$$\text{Brake moment} = (T_1 - T_2) \times r$$
$$= (8223.59 - 3436.12) \times 0.2$$
$$= 957.49 \text{ N-m}$$
$$= 957.5 \text{ N-m}$$

27. (a)

Given: M_t = 200 Nm, FOS = 2, μ = 0.3, P_a = 350 kPa

$$M_t \times \text{FOS} = \frac{\pi}{8} \times \mu \times P_a \times d\left(D^2 - d^2\right)$$

$$\Rightarrow \quad 200 \times 2 = \frac{\pi}{8} \times 0.3 \times 350 \times 1000\, d\left(4d^2 - d^2\right)$$

$$\Rightarrow \quad 400 = \frac{\pi}{8} \times 0.3 \times 350000 \times 3d^3$$

$$\Rightarrow \quad d = 0.148 \text{ m} = 148 \text{ mm}$$

28. (b)

Given: M_t = 2500 N-m, τ = 140 MPa

$$\text{Torsional shear stress,} \quad \tau = \frac{M_t}{2\pi t r^2}$$

$$140 = \frac{2500 \times 10^3}{2 \times \pi \times t \times 25^2}$$

$$t = 4.55 \text{ mm}$$

$$\text{Size of weld, } h = \frac{t}{0.707} = \frac{4.55}{0.707} = 6.43 \text{ mm}$$

29. (d)

$$\text{Principal stresses } \sigma_1, \sigma_2 = \frac{16}{\pi d^3}\left[M \pm \sqrt{M^2 + T^2}\right] = \frac{16}{\pi d^3}\left[300 \pm \sqrt{300^2 + 225^2}\right] \times 10^3 \text{MPa}$$

$$\sigma_1 = \frac{10.8 \times 10^6}{\pi d^3} \text{MPa}$$

$$\sigma_2 = \frac{-1.2 \times 10^6}{\pi d^3} \text{MPa}$$

According to maximum principal stress theory,

$$\sigma_1 \le \frac{\sigma_y}{F.O.S}$$

$$\frac{10.8 \times 10^6}{\pi d^3} \le \frac{210}{2}$$

$$d \ge 31.99$$

Minimum diameter required,

$$d = 31.99 \text{ mm} \approx 32 \text{ mm}$$

30. (b)

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

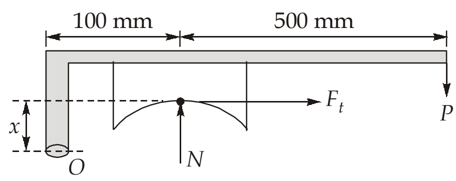
$$300 = F_t \times \frac{0.3}{2}$$

$$F_t = 2000 \text{ N}$$

$$F_t = \mu N$$

$$2000 = \mu N$$

$$N = \frac{2000}{0.2} = 10000 \text{ N} = 10 \text{ kN}$$



For brake to be self lock, P should be zero. Take moment about O,

$$F_t \times x = N \times 100$$

$$x = \frac{10000 \times 100}{\mu \times 10000} = \frac{100}{0.2} = 500 \text{ mm}$$

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