

ESE

GATE

State Engg. Exams

MADE EASY
WORKBOOK 2026



**Detailed Explanations of
Try Yourself *Questions***

Mechanical Engineering
Design of Machine Elements



1

Static & Fluctuating Stresses



Detailed Explanation of Try Yourself Questions

T1 : Solution

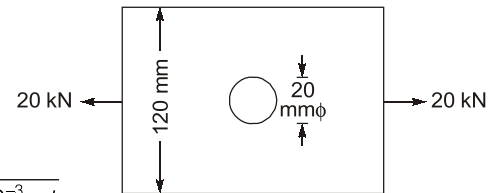
⇒

⇒

$$\sigma = \frac{P}{A}$$

$$\frac{600 \times 10^6}{3 \times 5} = \frac{20 \times 10^3}{(120 - 20) \times 10^{-3} \times t}$$

$$t = \frac{2 \times 10^5}{4 \times 10^7} = 5 \times 10^{-3} \text{ m} = 5 \text{ mm}$$



T2 : Solution

Mean stress,

$$\sigma_m = \frac{200 + (-100)}{2} = 50 \text{ MPa}$$

Variable stress,

$$\sigma_v = \frac{200 - (-100)}{2} = 150 \text{ MPa}$$

Soderberg's formula,

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

$$\frac{1}{2} = \frac{50}{0.55 \times \sigma_u} + \frac{150}{0.5 \times \sigma_u} = \frac{1}{\sigma_u} \left[\frac{50}{0.55} + \frac{150}{0.5} \right]$$

⇒

$$\sigma_u = 781.818 \text{ MPa} \approx 781.82 \text{ MPa}$$

T3 : Solution

$$\sigma_{\max} = \frac{32M_{\max}}{\pi d^3} = \frac{32 \times 500}{\pi d^3} = \frac{5092.96 \times 10^3}{d^3} \text{ MPa}$$

$$\sigma_{\min} = \frac{32M_{\min}}{\pi d^3} = \frac{32 \times -200}{\pi d^3} = \frac{-2037.18 \times 10^3}{d^3} \text{ MPa}$$

$$\sigma_{\text{mean}} = \frac{\sigma_{\max} + \sigma_{\min}}{2} = \frac{1527.89 \times 10^3}{d^3} \text{ MPa}$$

$$\sigma_v = \frac{\sigma_{\max} - \sigma_{\min}}{2} = \frac{3565.07 \times 10^3}{d^3} \text{ MPa}$$

For ductile material-using Soderberg equation

$$\frac{1}{N} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v}{\sigma_e} \quad [\sigma_e = 0.5 \sigma_u]$$

$$\frac{1}{2.5} = \frac{1527.89 \times 10^3}{400d^3} + \frac{3565.07 \times 10^3}{\frac{540}{2} \times d^3}$$

$$\Rightarrow d^3 = 2.5 \times 10^3 \times \left[\frac{1527.89}{400} + \frac{3565.07 \times 2}{540} \right] = 34.91 \text{ mm}$$

$$\Rightarrow d \simeq 35 \text{ mm}$$

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2

Welded, Riveted and Bolted Joint



Detailed Explanation of Try Yourself Questions

T1 : Solution

Pitch of rivets,

$$p = \frac{\pi(D+t)}{\text{No. of rivets/row}} = \frac{\pi(1600+30)}{45}$$

$$= 113.8 \text{ mm} \simeq 115 \text{ mm}$$

Efficiency,

$$\eta_c = 1 - \frac{d}{p} = 1 - \frac{35}{115} = 69.56 \% \simeq 70\%$$

T2 : Solution

Area of two welds,

$$A = 2(100 \times t) = (200 t) \text{ mm}^2$$

Primary shear stress

$$\tau_1 = \frac{P}{A} = \frac{20 \times 10^3}{200t} = \left(\frac{100}{t} \right) \text{ MPa}$$

Moment of inertia of two welds,

$$I = 2 I_{xx}$$

$$I_{xx} = \frac{100t^3}{12} + Ay_1^2 = \frac{100t^3}{12} + (100t)(100)^2 \text{ mm}^4$$

Since dimension t is very small compared with 100, so term t^3 can be neglected.

Therefore,

$$I_{xx} = (100^3 t) \text{ mm}^4$$

$$I = 2 I_{xx} = t \times \left(\frac{bd^2}{2} \right) = t \times \left(\frac{100 \times 200^2}{2} \right) = 2 \times 10^6 t \text{ mm}^4$$

$$= 2 \times (10^6 t) \text{ mm}^4$$

Bending stress in the top weld

$$\sigma_b = \frac{M_y}{I} = \frac{(20 \times 10^3 \times 200) \times 100}{2 \times 10^6 t} = \frac{200}{t} \text{ MPa}$$

Maximum shear stress

$$\tau = \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau_1^2} = \sqrt{\left(\frac{200}{2t}\right)^2 + \left(\frac{100}{t}\right)^2}$$

⇒

$$\tau = \frac{141.42}{t} \text{ MPa}$$

⇒

$$100 = \frac{141.42}{t}$$

⇒

$$t = 1.4142 \text{ mm}$$

⇒

$$h = \frac{t}{0.707} = 2 \text{ mm}$$

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Detailed Explanation of Try Yourself Questions

T1 : Solution

Life of bearing in millions of revolutions

$$= \left(\frac{C}{W} \right)^3 = \left(\frac{35}{45} \right)^3 = 0.4705$$

$$L = 60 NL_H$$

$$\Rightarrow \text{Life in hours, } L_H = \frac{L}{60N} = \frac{0.4705 \times 10^6}{60 \times 1800} = 4.356 \text{ hrs}$$

This the life expected for 90% of the bearings.

The average life expectancy is 5 times of the above life.

i.e. Average life = $5 \times 4.356 = 21.78 \text{ hrs}$

T2 : Solution

$$L = 60 NL_H = 60 \times 900 \times 2000 = 108 \times 10^6 \text{ revolutions}$$

$$L = \left(\frac{C}{W} \right)^3 \times 10^6$$

$$108 \times 10^6 = \left(\frac{C}{2.0} \right)^3 \times 10^6$$

$$C = 9.5 \text{ kN} = 9500 \text{ N}$$

$\therefore$ 6204 bearing is suitable

T3 : Solution

Given, $W = 6 \text{ kN} = 6000 \text{ N}$, $N = 1500 \text{ rpm}$, $D = 0.05 \text{ m}$, $L = 0.05 \text{ m}$, $S = 0.121$, $C = 50 \times 10^{-6} \text{ m}$

$$P = \frac{W}{LD} = \frac{6000}{50 \times 10^{-3} \times 50 \times 10^{-3}} = 2.4 \times 10^6 \text{ Pa}$$

Sommerfeld number,

$$S = \left(\frac{D}{C} \right)^2 \times \frac{ZN}{P}$$

$\Rightarrow$

$$0.121 = \left(\frac{50 \times 10^{-3}}{50 \times 10^{-6}} \right)^2 \times \frac{Z \times 1500}{60 \times 2.4 \times 10^6}$$

$\Rightarrow$

$$Z = 0.011616 \text{ Ns/m}^2 = 0.11616 \text{ Poise} = 11.62 \text{ cP}$$

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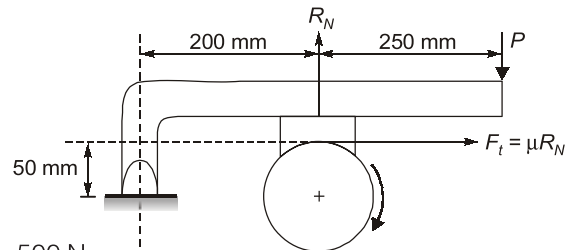
Brakes



Detailed Explanation of Try Yourself Questions

T1 : Solution

$$\begin{aligned}
 D &= 300 \text{ mm} = 0.3 \text{ m} \\
 T &= 75 \text{ N-m} \\
 T &= F_t \times r = \mu R_N \times r \\
 \Rightarrow 75 &= 0.35 \times R_N \times 0.15 \\
 \Rightarrow R_N &= 1428.57 \text{ N} \\
 F_T &= \mu R_N = 0.35 \times 1428.57 = 500 \text{ N} \\
 P \times (250 + 200) + F_T \times 50 &= R_N \times 200 \\
 \Rightarrow P \times 450 + 500 \times 50 &= 1428.57 \times 200 \\
 \Rightarrow P &= 579.365 \text{ N}
 \end{aligned}$$



T2 : Solution

Tension ratio,

$$\frac{T_1}{T_2} = e^{\mu\theta}$$

$$= e^{\frac{\pi \times 270}{180} \times 0.2} = 2.566 \quad \dots(i)$$

Torque equation

$$T = (T_1 - T_2)$$

$$\Rightarrow \frac{P \times 60}{2\pi N} = T_1 - T_2$$

$$\Rightarrow \frac{30 \times 10^3 \times 60}{2\pi \times 400} = (T_1 - T_2) \times r \quad \Rightarrow r = 1 \text{ m}$$

$$(T_1 - T_2) = 716.197 \text{ N} \quad \dots(ii)$$

From Eq (i) and (ii) we get

$$\begin{aligned}
 T_1 &= 1173.54 \text{ N} \\
 T_2 &= 457.34 \text{ N}
 \end{aligned}$$

So, maximum tension = $T_1 = 1173.54 \text{ N}$

T3 : Solution

Drum diameter = 850 mm

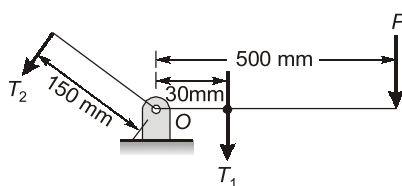
Thickness of block = 75 mm

$$\mu = 0.4$$

$$\frac{T_1}{T_2} = \left(\frac{1 + \mu \tan 7.5^\circ}{1 - \mu \tan 7.5^\circ} \right)^{12}$$

$$\frac{T_1}{T_2} = \left(\frac{1 + 0.4 \tan 7.5^\circ}{1 - 0.4 \tan 7.5^\circ} \right)^{12}$$

$$\frac{T_1}{T_2} = 3.5432$$



By taking moment about 'O'

$$P \times 500 + T_1 \times 30 = T_2 \times 150$$

$$P \times 500 + 3.5432 \times 30 T_2 = T_2 \times 150$$

$$P \times 500 = 43.704 T_2$$

Power absorbed by blocks

$$P = (T_1 - T_2) \omega \times \frac{d}{2}$$

$$d = 850 + 2 \times 75 = 1000$$

$$225 \times 10^3 \times 10^3 = (T_2 \times 3.5432 - T_2) \times \frac{2\pi \times 240}{60} \times \frac{1000}{2}$$

$$T_2 = 7040.27 \text{ N}$$

$$P \times 500 = 43.704 \times 7040.27$$

$$P = 615.35 \text{ N}$$

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5

Friction Clutches



Detailed Explanation of Try Yourself Questions

T1 : Solution

Pressure variation for uniform wear is given by

$$P = \frac{W}{2\pi(r_0 - r_i)r}$$

Maximum pressure occurs at smallest radius,

$$P_{\max} = \frac{W}{2\pi(r_0 - r_i)r_i} = \frac{8000}{2\pi(0.2 - 0.1)0.1} = 127323.9 \text{ N/m}^2$$

$$= 127.3 \text{ kN/m}^2$$

Minimum pressure occurs at largest radius,

$$P_{\min} = \frac{W}{2\pi(r_0 - r_i)r_0} = \frac{8000}{2\pi(0.2 - 0.1)0.2} = 63.66 \text{ kN/m}^2$$

T2 : Solution

Design torque,

$$T = 2 \times 50 = 100 \text{ N-m}$$

For uniform wear conditions, pressure variation is given by

$$P = \frac{C}{r} = \frac{W}{2\pi(r_0 - r_i)r}$$

Maximum pressure occurs at inner radius i.e. at $r = r_i$

$$\text{So, } P_{\max} = \frac{W}{2\pi(r_0 - r_i)r_i}$$

$\Rightarrow$

$$1 \times 10^6 = \frac{W}{2\pi\left(\frac{0.1}{2} - \frac{0.065}{2}\right) \times \frac{0.065}{2}}$$

$\Rightarrow$

$$W = 3573.56 \text{ N}$$

$$T = \mu WR_m \cdot n$$

$$100 = 0.08 \times 3573.56 \times \left(\frac{\frac{0.1}{2} + \frac{0.065}{2}}{2} \right) \times n$$

⇒

$$n = 8.479 \simeq 9 \text{ pair of contact surface}$$

Generally we take friction disk in such a way so that total number of contact pair are even, so answer should be 10.

T3 : Solution

$$P = T \cdot \omega$$

$$873.8 = T \times \frac{2\pi \times 900}{60}$$

$$T = 9.272$$

Also,

$$T = I \alpha$$

$$T = (mk^2) \alpha$$

$$9.272 = 14 \times (160)^2 \times \alpha$$

$$\alpha = 25.87 \text{ rad/se c}^2$$

Also,

$$\omega_F = \omega_i + \alpha t$$

$$t = \frac{2\pi \times 900}{60 \times 25.87} = 3.64 \text{ sec}$$

T4 : Solution

$$T = \frac{P \times 60}{2\pi N} = \frac{20 \times 10^3 \times 60}{2\pi \times 4000} = 47.746 = \mu W R_m$$

⇒

$$R_m = \frac{T}{\mu W}$$

⇒

$$\frac{2}{3} \left(\frac{r_0^3 - r_i^3}{r_0^2 - r_i^2} \right) = \frac{T}{\mu \times P \times \pi (r_0^2 - r_i^2)}$$

⇒

$$\begin{aligned} (r_0^3 - r_i^3) &= \frac{3T}{2\mu P\pi} \Rightarrow r_0^3 = r_i^3 + \frac{3T}{2\mu P\pi} \\ &= (50 \times 10^{-3})^3 - \frac{3 \times 47.746}{2 \times 0.3 \times 1.2 \times 10^6 \times \pi} \\ r_i &= 0.0395 \text{ m} = 39.5 \text{ mm} \end{aligned}$$

T5 : Solution

$$r_i = r_o - b \sin \alpha = 350 - 130 \sin 12.5 = 321.863 \text{ mm}$$

$$r_m = \frac{350 + 321.863}{2} = 335.93 \text{ mm}$$

Torque,

$$T = \frac{\mu W}{\sin \alpha} \times r_m$$

$$\begin{aligned} W &= \frac{T \sin \alpha}{\mu r_m} = \frac{400 \times 1000 \times 0.21644}{0.4 \times 335.93} \\ &= 644.298 \text{ N} \simeq 644.3 \text{ N} \end{aligned}$$





Detailed Explanation of Try Yourself Questions

T1 : Solution

Wear strength,

$$\frac{N_p}{N_g} = 2 = \frac{T_g}{T_p}$$

$$S_w(\text{or}) P_w = kQwd_p$$

$$k = 1.5 \text{ N/mm}^2$$

$$Q = \frac{2T_g}{T_p + T_g} = \frac{2}{1 + \frac{T_p}{T_g}} = \frac{2}{1 + \frac{1}{2}} = \frac{4}{3}$$

Now,

$$w = 100 \text{ mm}$$

$$d_p = 400 \text{ mm}$$

$$P_w = kQwd_p$$

$$= 1.5 \times \frac{4}{3} \times 100 \times 400 = 80000 \text{ N} = 80 \text{ kN}$$

T2 : Solution

$$\phi = 20^\circ$$

$$\text{Power} = 20 \text{ kW} = 20 \times 10^3 \text{ W}$$

$$N_p = 300 \text{ rpm}$$

$$\text{Velocity ratio} = \frac{T_g}{T_p} = 3$$

$$\sigma_{OG} = 100 \text{ MPa} = 100 \times 10^6 \text{ N/m}^2$$

$$= 100 \text{ N/mm}^2$$

$$\sigma_{OP} = 120 \text{ MPa} = 120 \times 10^6 \text{ N/m}^2$$

$$T_P = 15$$

$$T_G = 3 T_P = 3 \times 15 = 45$$

$$b = 14 \text{ m}$$

$$\sigma_{es} = 600 \text{ MPa} = 600 \text{ N/mm}^2$$

$$E_P = 200 \text{ GPa} = 200 \times 10^3 \text{ N/mm}^2$$

$$E_s = 100 \text{ GPa} = 100 \times 10^3 \text{ N/mm}^2$$

Module:

let

m = Module in mm and

D_P = Pitch circle diameter of the pinion in mm

Pitch line velocity,

$$V = \frac{\pi D_P N_P}{60} = \frac{\pi m T_P N_P}{60} = \frac{\pi \times m \times 15 \times 300}{60}$$

$$= 0.235 \text{ m/s}$$

Assuming steady load condition and 8-10 hours of service per day, the service factor C_s is taken as 1, i.e. $C_s = 1$.

We know that the design tangential tooth load,

$$W_T = \frac{P}{V} \times C_s = \frac{20 \times 10^3}{0.235m} \times 1 = \frac{85.10 \times 10^3}{m} \text{ N}$$

and Velocity factor,

$$C_v = \frac{3}{3 + v} = \frac{3}{3 + 0.235m}$$

Tooth form factor for pinion

$$y_p = 0.154 - \frac{0.912}{T_P} = 0.154 - \frac{0.912}{15} = 0.0932$$

And tooth form factor for gear

$$y_g = 0.154 - \frac{0.912}{T_g} = 0.154 - \frac{0.912}{45} = 0.133$$

$\therefore$

$$\sigma_{PO} \times y_p = 120 \times 0.0932 = 11.184$$

$$\sigma_{OG} \times y_g = 100 \times 0.133 = 13.3$$

Since $(\sigma_{OP} \times y_p)$ is less than $(\sigma_{OG} \times y_g)$ therefore the pinion is weaker. Now using the Lewis equation to the pinion, we have

$$W_T = \sigma_{wp} \times b \pi m y_p = \sigma_{OP} \times C_v b \pi m y_p \quad [\because \sigma_{wp} = \sigma_{OP} \times C_v]$$

$$\frac{85.1 \times 10^3}{m} = 120 \times \left(\frac{3}{3 + 0.235m} \right) \times (14m \times \pi m \times 0.0932)$$

$$\frac{85.1 \times 10^3}{m} = \frac{1475.69}{(3 + 0.235m)}$$

$$\Rightarrow \frac{57.66}{m} = \frac{m^2}{3 + 0.235m} = 172.98 + 13.55 m = m^3$$

$$\Rightarrow m^3 - 13.55 m - 172.98 = 0$$

$$m = 6.37 \text{ mm} \approx 7 \text{ mm}$$

Face width:

$$b = 14 m = 14 \times 7 = 98 \text{ mm}$$

Pitch diameter of gears,

$$D_P = m T_P = 7 \times 15 = 105 \text{ mm}$$

$$D_g = m T_g = 7 \times 45 = 315 \text{ mm}$$

Checking the gears for wear:-

Ratio factor,

$$Q = \frac{2 \times VR}{VR + 1} = \frac{2 \times 3}{3 + 1} = 1.5$$

$$\begin{aligned} \text{Load stress factor} &= \frac{(\sigma_{es})^2 \sin \phi}{1.4} \left[\frac{1}{E_p} + \frac{1}{E_g} \right] \\ &= \frac{600^2 \times \sin 20^\circ}{1.4} \left[\frac{1}{200 \times 10^3} + \frac{1}{100 \times 10^3} \right] \\ &= 1.3192 \text{ N/mm}^2 \end{aligned}$$

The maximum or limiting load for wear

$$\begin{aligned} W_w &= D_p \times b Q K \\ &= 105 \times 98 \times 1.5 \times 1.319 \\ &= 20358.765 \text{ N} = 20.358 \text{ kN} \end{aligned}$$

Tangential load on tooth,

$$\begin{aligned} W_T &= \frac{85.1 \times 10^3}{m} = \frac{85.1 \times 10^3}{7} \\ &= 12157 \text{ N} = 12.157 \text{ kN} \end{aligned}$$

Since maximum wear load (20.358 kN) is more than the tangential load (12.157 kN) on the tooth, the design is satisfactory from the wear point of view.

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