RPSC AEn-2024 Main Test Series

MECHANICAL ENGINEERING | **Test 3**

Test Mode : • Offline • Online

Subjects : Machine Design and Strength of Materials

DETAILED EXPLANATIONS

1. Solution:

S.No.	Thick cylinder	Thin cylinder
1	The thickness of cylindrical vessel is greater than (1/20) of its internal diameter.	The thickness of cylindrical vessel is less than (1/15) to (1/20) of its internal diameter.
2	The stresses are not uniform rather it varies along the thickness.	Stresses are assumed to be uniform throughout the wall thickness.

2. Solution:

If the angle of the helix of the coil is so small that the bending effects can be neglected, then the spring is called a closed coiled spring. If the angle of the helix of the coil is quite appreciable that the both bending as well as torsional shear stresses are introduced in the spring, then the spring is called as open coiled spring.

3. Solution:

For maximum deflection, $y_{max} = \frac{5wl^4}{384EI}$

For maximum slope, $\theta_{max} = \frac{wl^3}{24EI}$

4. Solution:

Slenderness ratio: The slenderness ratio of a column is the ratio of its effective length to its radius of gyration. It is dimensionless and denoted by 'k'.

5. Solution:

The work done by the load in straining the material of a body is stored within it in the form of energy known as strain energy.

6. Solution:

The relationship between modulus of elasticity, modulus of rigidity and Poisson's ratio is given by

$$E = 2G(1 + \mu)$$
$$E = 3K(1 - 2\mu)$$

Where, E = Modulus of elasticity, G = Modulus of rigidity, K = Bulk modulus, μ = Poisson's ratio

7. Solution:

The point of contraflexure is the location on a beam where the bending moment is zero and the curvature changes sign from positive to negative or vice-versa.

8. Solution:

1. **Journal bearing:** The sliding contact bearings in which the sliding action is along the circumference of a circle or an arc of a circle and carying radial loads are known as journal or sleeve bearings.
2. **Classification of Journal Bearing:** The types of journal bearing are as follows:
   (i) Full Journal Bearing
   (ii) Partial Journal Bearing
   (iii) Fitted Journal Bearing

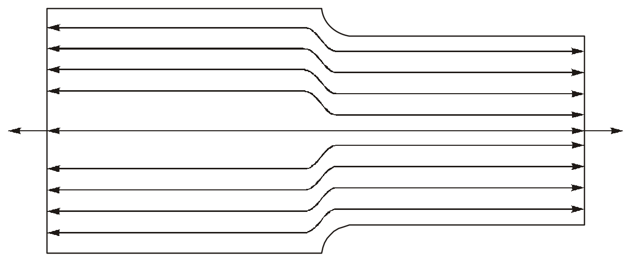
9. Solution:

Factors that reduce the fatigue strength of materials:

1. Size of the component
2. Shape of the component
3. Surface finish
4. Temperature of the material
5. Notch sensitivity of the material

10. Solution:

Stress concentration: Stress concentration is defined as the localization of high stresses due to the irregularities present in the component and abrupt changes of the cross-section.



Stress concentration

11. Solution:

1. The material of the shaft is uniform throughout.
2. The twist along the shaft is uniform.
3. Normal cross sections of the shaft, which were plane and circular before twist; remain plane and circular after twist.
4. All diameters of the normal cross section which were straight before twist, remain straight with their magnitude unchanged, after twist.

12. Solution:

(i) **Life of bearing:** The life of a bearing is the total number of revolutions (or total hours of operation at a given constant speed) that a bearing completes before the first sign of fatigue failure occurs on any of its rolling contact surfaces. It represents the actual service life of an individual bearing under given operating conditions.

(ii) **Rating life of bearing:** The rated life, also called L10 life, is the number of revolutions (or operating hours at a constant speed) that 90% of a group of identical bearings can complete without fatigue failure under specified load and speed conditions.

(iii) **Median life of bearing:** The median life, also called L50 life, is the number of revolutions (or operating hours) that 50% of a group of identical bearings are expected to complete without fatigue failure.

13. Solution:

Given: W_{max} = 160 kN, W_{min} = –160 kN, FOS = 2, σ_{ut} = 1100 MPa, σ_{yt} = 930 MPa.

Mean or average load, $W_m = \dfrac{W_{max}+W_{min}}{2} = \dfrac{160+(-160)}{2} = 0$

$\therefore$ Mean stress, $\sigma_m = \dfrac{W_m}{A} = 0$

$$\text{Variable load, } W_v = \frac{W_{max} - W_{min}}{2} = \frac{160-(-160)}{2} = 160 \text{ kN}$$

$$\therefore \quad \text{Variable stress, } \sigma_v = \frac{W_v}{A} = \frac{160 \times 10^3}{\frac{\pi}{4}d^2} = \frac{160 \times 10^3}{0.7854 d^2} \text{ MPa}$$

We know that according to Soderberg's formula for reversed axial loading,

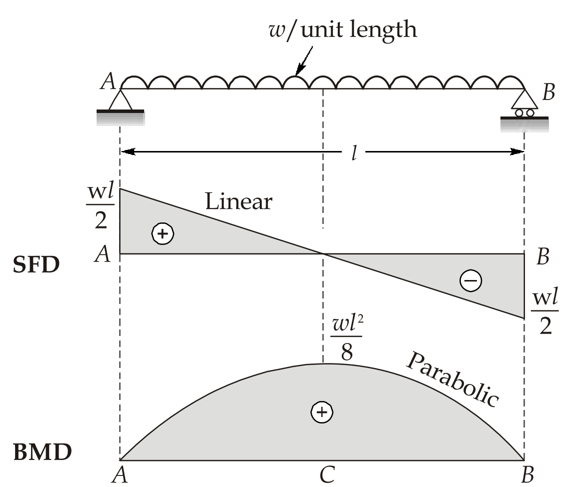
$$\frac{1}{FOS} = \frac{\sigma_m}{\sigma_{yt}} + \frac{\sigma_v}{\sigma_e}$$

$$\frac{1}{2} = \frac{0}{930} + \frac{160 \times 10^3}{0.7854 d^2 \times 0.5 \times 1100} \quad [\because \sigma_e = 0.5\,\sigma_{ut}]$$

$$d^2 = 740.79$$

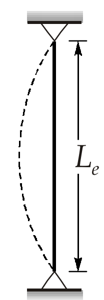
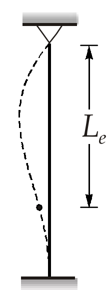
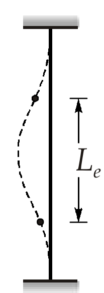
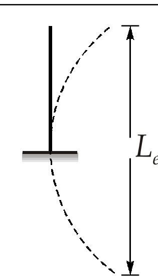
$$d = 27.22 \text{ mm}$$

14. Solution:



15. Solution:

The end conditions of a column determine its effective length and hence its buckling load. The common types are:

End Condition	Diagram	Effective Length
Both ends hinged (both ends are held in position but not restrained against rotation)		$L_e = L$
One end fixed and other hinged: (One end is held in position and restrain against rotation but other end is held in position but not restrained against rotation.)		$L_e = \frac{L}{\sqrt{2}}$
Both ends fixed: (Both ends held in position and restrained against rotation)		$L_e = \frac{L}{2}$
One end fixed and other end free: (One end held in position and restrained against rotation but other end neither held in position nor restrained against rotation)		$L_e = 2L$

16. Solution:

Given: Width, $b = 5$ cm, Thickness, $t = 8$ mm, Length, $l = 700$ mm, Central load, $W = 45$ kN, Maximum allowable stress, σ = 200 MPa.

Bending stress (σ) is given as

$$\sigma = \frac{3Wl}{2nbt^2}$$

$$200 \times 10^6 = \frac{3 \times 45 \times 10^3 \times 700 \times 10^{-3}}{2 \times n \times 50 \times 10^{-2} \times \left(8 \times 10^{-3}\right)^2}$$

$$n = 7.38 = 8$$

17. Solution:

Diameter, d = 15 cm or 150 mm

$$\text{Power transmitted, } P = 150 \text{ kW}$$

$$\text{Speed, } N = 180 \text{ rpm}$$

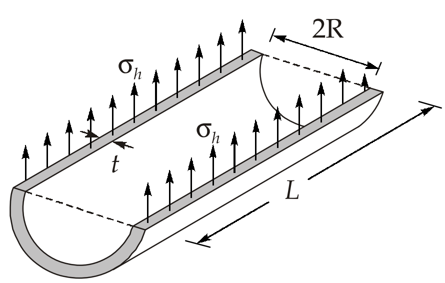
$$\text{Angular velocity, } \omega = \frac{2\pi N}{60} = 6\pi = 18.85 \text{ rad/s}$$

$$\text{Torque, } T = \frac{P}{\omega} = \frac{150 \times 10^3}{18.85} = 7957.75 \text{ Nm}$$

$$\tau_{max} = \frac{16T}{\pi d^3} = \frac{16 \times 7957.75 \times 10^3}{\pi \times (150)^3} = 12 \text{ N/mm}^2$$

18. Solution:

Consider a cylinder with insides pressure equal to P closed at both end. If we consider lower half then if σ_h is circumferential stress, then

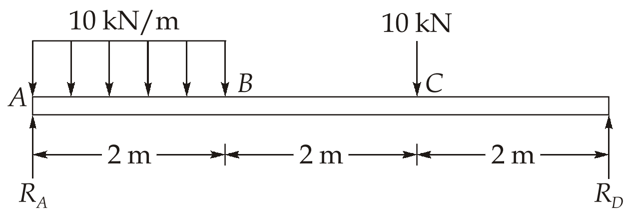


σ_h × Area on which circumferential stress is acting = P × Projected area

$$(\sigma_h \times L \times t) \times 2 = P \times (2R \times L)$$

$$\Rightarrow \quad \sigma_h = \frac{PR}{t} \text{ or } \frac{PD}{2t}$$

19. Solution:



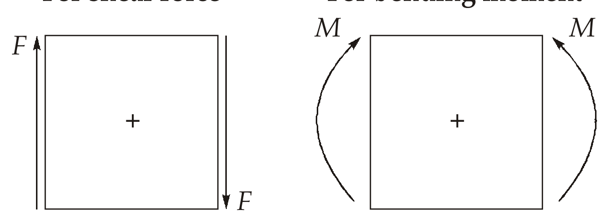
(i) $\Sigma M_A = 0 \Rightarrow R_A(0) + (10 \times 2) \times 1 + 10(4) - R_D(6) = 0$

$$\Rightarrow \quad R_D = \frac{60}{6} \Rightarrow R_D = 10 \text{ kN}$$

$$\Sigma F_y = 0 \Rightarrow \quad R_A + R_D = 10 \times 2 + 10 = 30 \text{ kN}$$

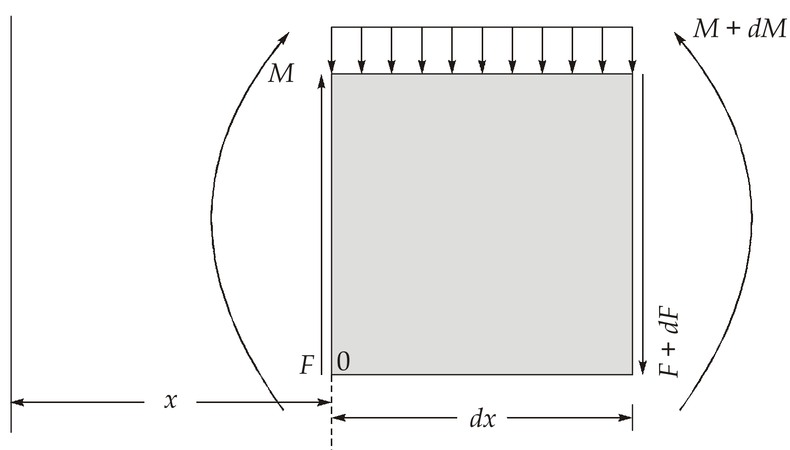
$$\Rightarrow \quad R_A + 10 = 30 \text{ kN} \Rightarrow R_A = 20 \text{ kN}$$

Sign convection to be used: (Meriam sign convention)



General relationship between shear force and bending moment

For rotational equilibrium $\Sigma M_D = 0$



$$\Rightarrow \quad +M + F(0) - (F + dF)dx - (M + dM) = 0$$

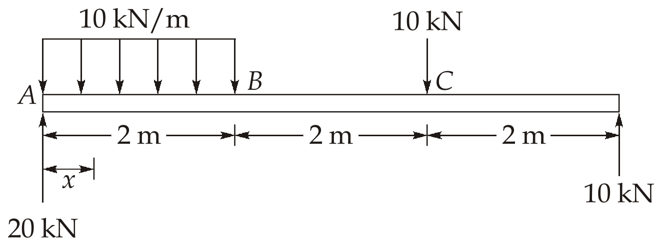
$$\Rightarrow \quad (F + dF)dx = dM$$

$$\Rightarrow \quad Fdx + dF.dx = dM$$

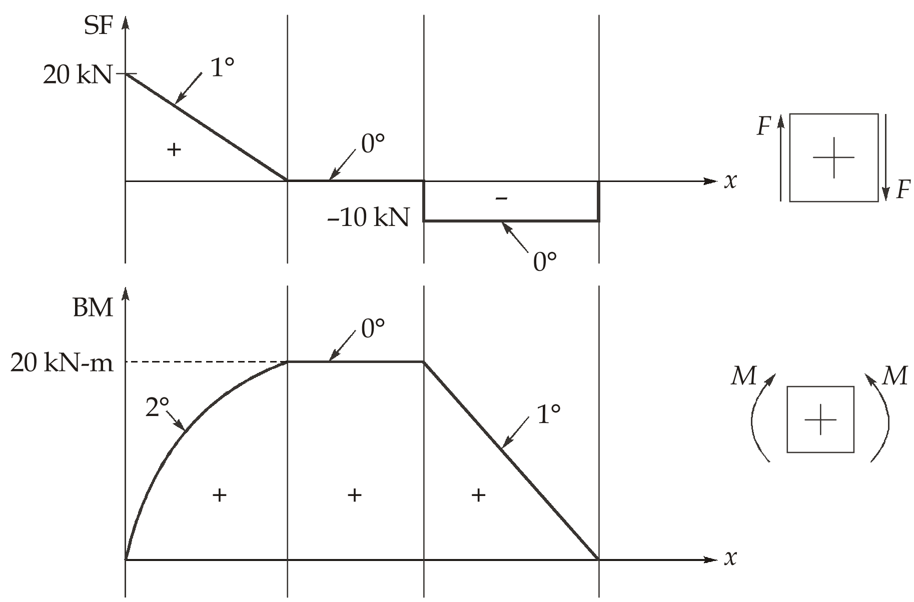
$$\Rightarrow \quad F = \frac{dM}{dx}$$

General relationship between SF and BM when Meriam sign convention is followed.

(ii)



Section $AB : x \in (0\text{ m}, 2\text{ m})$	Section $BC : x \in (2\text{ m}, 4\text{ m})$	Section $CD : x \in (4\text{ m}, 6\text{ m})$
$F_x = 20 - 10x : (\text{linear})$ $(F)_{x=0^+} = 20\text{ kN}$ $(F)_{x=2^-} = 0\text{ kN}$	$F_x = 0\text{ kN}$ $(F)_{x=2^+} = 0\text{ kN}$ $(F)_{x=4^-} = 0\text{ kN}$	$F_x = -10\text{ kN}$ $(F)_{x=4^+} = -10\text{ kN}$ $(F)_{x=6^-} = -10\text{ kN}$
$M_x = 20x - 10x\left(\frac{x}{2}\right) : (\text{Parabolic})$ $(M)_{x=0} = 0$ $(M)_{x=2} = 20\text{ kN}-\text{m}$	$M_x = 20\text{ kN}-\text{m}$ $(M)_{x=2^+} = 20\text{ kN}-\text{m}$ $(M)_{x=4^-} = 20\text{ kN}-\text{m}$	$M_x = 20 - 10x : (\text{linear})$ $(M)_{x=4^+} = 20\text{ kN}-\text{m}$ $(M)_{x=6^-} = 0\text{ kN}-\text{m}$



20. Solution:

Input data: ϕ = 20(FD), Number of teeth on pinion (z_1) = 20.

Number of teeth on gear (z_2) = 50

Power, P = 5 kW at 1200 rpm

Speed of pinion, N_1 = 1200 rpm

S_{ut} = Ultimate strength for pinion and gear = 500 MPa

Face width (b) = 44 mm

Module = 4 mm

Service factor, C_s = 2

$$\text{Velocity factor, } C_V = \frac{3}{3+V}$$

Lewis form factor for pinion, $Y_1 = 0.32$

Lewis form factor for gear, $Y_2 = 0.408$

To determine FOS (N) = ?

T_1 = Torque to be transmitted by pinion (or) Rated torque for pinion

$$= \frac{P \times 60}{2\pi N_1} \times 10^6 = 39788.736 \text{ N-mm}$$

Design torque for pinion (T_{max}) = $T_1 \times C_s$

$$(T_{max})_1 = 79577.472 \text{ N-mm}$$

$$\text{Velocity of pinion, } V_1 = \frac{\pi(D_1 = mz_1)N_1}{60} = 5.027 \text{ m/s}$$

$$F_t = \text{Tangential force} = \frac{2(T_{max})_1}{D_1} = 1989.4368 \text{ N}$$

$$C_V = \frac{3}{3+V} = 0.374$$

$$F_d = \text{Dynamic load} = \frac{F_t}{C_V} = 5319.35 \text{ N}$$

F_s = Beam strength for weaker gear, pinion is the weaker gear because pinion and gear are made of same material.

Hence,

$$F_s = \left[\sigma_b = \frac{S_{ut}}{N}\right][Y_1]bm = \left(\frac{500}{N}\right) \times 0.32 \times 44 \times 4 = \frac{28160}{N} \text{ N}$$

Safe condition with respect to bending failure, $F_d \le F_s$

$$5319.35 \le \frac{28160}{N} \Rightarrow N \le 5.29$$

Hence, FOS (N) = 5.29

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