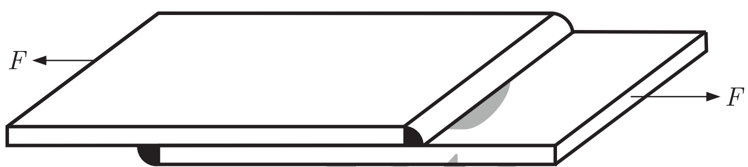
CHAPTER 5

MACHINE DESIGN

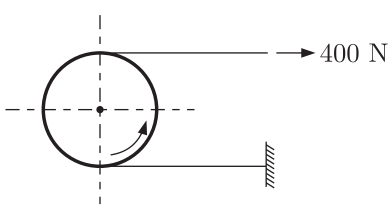
YEAR 2012 **TWO MARKS**

MCQ 5.1 A fillet welded joint is subjected to transverse loading F as shown in the figure. Both legs of the fillets are of 10 mm size and the weld length is 30 mm. If the allowable shear stress of the weld is 94 MPa, considering the minimum throat area of the weld, the maximum allowable transverse load in kN is



(A) 14.44 (B) 17.92
(C) 19.93 (D) 22.16

MCQ 5.2 A force of 400 N is applied to the brake drum of 0.5 m diameter in a band-brake system as shown in the figure, where the wrapping angle is 180°. If the coefficient of friction between the drum and the band is 0.25, the braking torque applied, in Nm is



(A) 100.6 (B) 54.4
(C) 22.1 (D) 15.7

MCQ 5.3 A solid circular shaft needs to be designed to transmit a torque of 50 Nm. If the allowable shear stress of the material is 140 MPa, assuming a factor of safety of 2, the minimum allowable design diameter is mm is
(A) 8 (B) 16

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(C) 24 (D) 32

YEAR 2011 **TWO MARKS**

MCQ 5.4 Two identical ball bearings P and Q are operating at loads 30 kN and 45 kN respectively. The ratio of the life of bearing P to the life of bearing Q is

(A) $\frac{81}{16}$ (B) $\frac{27}{8}$

(C) $\frac{9}{4}$ (D) $\frac{3}{2}$

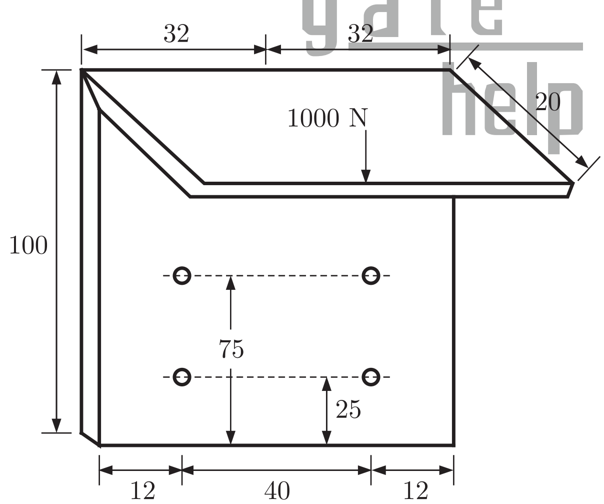
YEAR 2010 **TWO MARKS**

MCQ 5.5 A band brake having band-width of 80 mm, drum diameter of 250 mm, coefficient of friction of 0.25 and angle of wrap of 270 degrees is required to exert a friction torque of 1000 Nm. The maximum tension (in kN) developed in the band is

(A) 1.88 (B) 3.56

(C) 6.12 (D) 11.56

MCQ 5.6 A bracket (shown in figure) is rigidly mounted on wall using four rivets. Each rivet is 6 mm in diameter and has an effective length of 12 mm.



Direct shear stress (in MPa) in the most heavily loaded rivet is

(A) 4.4 (B) 8.8

(C) 17.6 (D) 35.2

MCQ 5.7 A lightly loaded full journal bearing has journal diameter of 50 mm, bush bore of 50.05 mm and bush length of 20 mm. If rotational speed of journal is 1200 rpm and average viscosity of liquid lubricant is 0.03 Pa s, the power loss (in W) will be

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(A) 37 (B) 74
(C) 118 (D) 237

YEAR 2009 **TWO MARKS**

MCQ 5.8 A forged steel link with uniform diameter of 30 mm at the centre is subjected to an axial force that varies from 40 kN in compression to 160 kN in tension. The tensile (S_u), yield (S_y) and corrected endurance (S_e) strengths of the steel material are 600 MPa, 420 MPa and 240 MPa respectively. The factor of safety against fatigue endurance as per Soderberg's criterion is
(A) 1.26 (B) 1.37
(C) 1.45 (D) 2.00

• Common Data For Q.9 and Q.10

A 20° full depth involute spur pinion of 4 mm module and 21 teeth is to transmit 15 kW at 960 rpm. Its face width is 25 mm.

MCQ 5.9 The tangential force transmitted (in N) is
(A) 3552 (B) 2611
(C) 1776 (D) 1305

MCQ 5.10 Given that the tooth geometry factor is 0.32 and the combined effect of dynamic load and allied factors intensifying the stress is 1.5; the minimum allowable stress (in MPa) for the gear material is
(A) 242.0 (B) 166.5
(C) 121.0 (D) 74.0

YEAR 2008 **TWO MARKS**

MCQ 5.11 A journal bearing has a shaft diameter of 40 mm and a length of 40 mm. The shaft is rotating at 20 rad/s and the viscosity of the lubricant is 20 mPa-s. The clearance is 0.020 mm. The loss of torque due to the viscosity of the lubricant is approximately.
(A) 0.040 N-m (B) 0.252 N-m
(C) 0.400 N-m (D) 0.652 N-m

MCQ 5.12 A clutch has outer and inner diameters 100 mm and 40 mm respectively. Assuming a uniform pressure of 2 MPa and coefficient of friction of liner material is 0.4, the torque carrying capacity of the clutch is
(A) 148 N-m (B) 196 N-m

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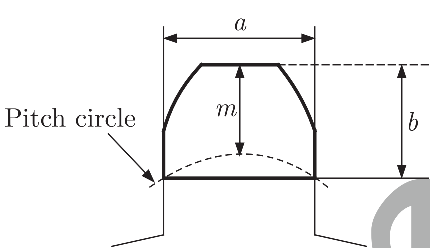
(C) 372 N-m (D) 490 N-m

MCQ 5.13 A spur gear has a module of 3 mm, number of teeth 16, a face width of 36 mm and a pressure angle of $20°$. It is transmitting a power of 3 kW at 20 rev/s. Taking a velocity factor of 1.5 and a form factor of 0.3, the stress in the gear tooth is about.

(A) 32 MPa (B) 46 MPa

(C) 58 MPa (D) 70 MPa

MCQ 5.14 One tooth of a gear having 4 module and 32 teeth is shown in the figure. Assume that the gear tooth and the corresponding tooth space make equal intercepts on the pitch circumference. The dimensions 'a' and 'b', respectively, are closest to



(A) 6.08 mm, 4 mm (B) 6.48 mm, 4.2 mm

(C) 6.28 mm, 4.3 mm (D) 6.28 mm, 4.1 mm

MCQ 5.15 Match the type of gears with their most appropriate description.

Type of gear

P. Helical

Q. Spiral Bevel

C. Hypoid

S. Rack and pinion

Description

1. Axes non parallel and non intersecting
2. Axes parallel and teeth are inclined to the axis
3. Axes parallel and teeth are parallel to the axis
4. Axes are perpendicular and intersecting, and teeth are inclined to the axis.
5. Axes are perpendicular and used for large speed reduction
6. Axes parallel and one of the gears has infinite radius

(A) P-2, Q-4, R-1, S-6 (B) P-1, Q-4, R-5, S-6

(C) P-2, Q-6, R-4, S-2 (D) P-6, Q-3, R-1, S-5

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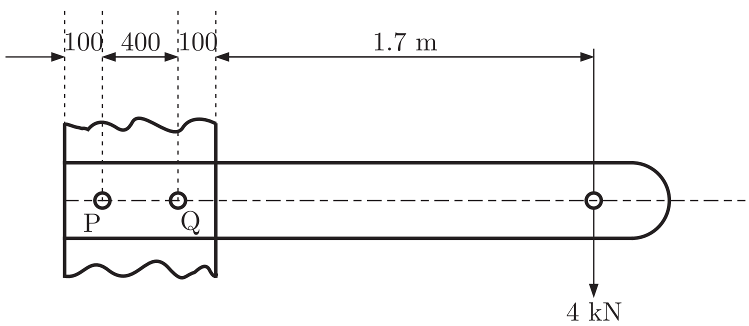
• Common Data For Q.16 and Q.17

A steel bar of 10 × 50 mm is cantilevered with two M 12 bolts (P and Q) to support a static load of 4 kN as shown in the figure.



MCQ 5.16 The primary and secondary shear loads on bolt P, respectively, are

(A) 2 kN, 20 kN (B) 20 kN, 2 kN

(C) 20 kN, 0 kN (D) 0 kN, 20 kN

MCQ 5.17 The resultant shear stress on bolt P is closest to

(A) 132 MPa (B) 159 MPa

(C) 178 MPa (D) 195 MPa

YEAR 2007 ONE MARK

MCQ 5.18 A ball bearing operating at a load F has 8000 hours of life. The life of the bearing, in hours, when the load is doubled to $2F$ is

(A) 8000 (B) 6000

(C) 4000 (D) 1000

YEAR 2007 TWO MARKS

MCQ 5.19 A thin spherical pressure vessel of 200 mm diameter and 1 mm thickness is subjected to an internal pressure varying form 4 to 8 MPa. Assume that the yield, ultimate and endurance strength of material are 600, 800 and 400 MPa respectively. The factor of safety as per Goodman's relation is

(A) 2.0 (B) 1.6

(C) 1.4 (D) 1.2

MCQ 5.20 A natural feed journal bearing of diameter 50 mm and length 50 mm operating at 20 revolution/ second carries a load of 2 kN. The lubricant used has a viscosity of 20 mPas. The radial clearance is 50 μm. The Sommerfeld number for the bearing is

(A) 0.062 (B) 0.125

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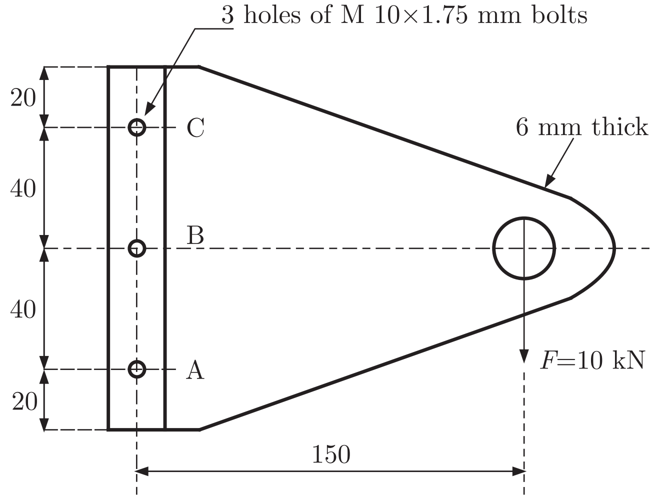
(C) 0.250 (D) 0.785

MCQ 5.21 A bolted joint is shown below. The maximum shear stress, in MPa in the bolts at A and B , respectively are

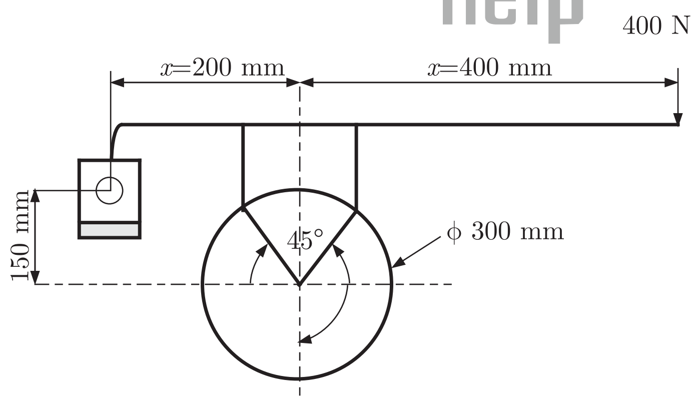


(all dimensions in the figure are in mm)

(A) 242.6, 42.5 (B) 42.5, 242.6

(C) 42.5, 42.5 (D) 18.75, 343.64

MCQ 5.22 A block-brake shown below has a face width of 300 mm and a mean co-efficient of friction of 0.25. For an activating force of 400 N, the braking torque in Nm is



(A) 30 (B) 40

(C) 45 (D) 60

MCQ 5.23 The piston rod of diameter 20 mm and length 700 mm in a hydraulic cylinder is subjected to a compressive force of 10 kN due to the internal pressure. The end conditions for the rod may be assumed as guided at the piston end and hinged at the other end. The Young's modulus is 200 GPa. The factor

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of safety for the piston rod is

(A) 0.68 (B) 2.75

(C) 5.62 (D) 11.0

• Common Data For Q.24 and Q.26

A gear set has a pinion with 20 teeth and a gear with 40 teeth. The pinion runs at 30 rev/s and transmits a power of 20 kW. The teeth are on the 20° full-depth system and have a module of 5 mm. The length of the line of action is 19 mm.

MCQ 5.24 The center distance for the above gear set in mm is

(A) 140 (B) 150

(C) 160 (D) 170

MCQ 5.25 The contact ratio of the contacting tooth is

(A) 1.21 (B) 1.25

(C) 1.29 (D) 1.33

MCQ 5.26 The resultant force on the contacting gear tooth in N is

(A) 77.23 (B) 212.20

(C) 2258.1 (D) 289.43

YEAR 2006 TWO MARKS

MCQ 5.27 A disc clutch is required to transmit 5 kW at 2000 rpm. The disk has a friction lining with coefficient of friction equal to 0.25. Bore radius of friction lining is equal to 25 mm. Assume uniform contact pressure of 1 MPa. The value of outside radius of the friction lining is

(A) 39.4 mm (B) 49.5 mm

(C) 97.9 mm (D) 142.9 mm

MCQ 5.28 Twenty degree full depth involute profiled 19 tooth pinion and 37 tooth gear are in mesh. If the module is 5 mm, the centre distance between the gear pair will be

(A) 140 mm (B) 150 mm

(C) 280 mm (D) 300 mm

MCQ 5.29 A cylindrical shaft is subjected to an alternating stress of 100 MPa. Fatigue strength to sustain 1000 cycles is 490 MPa. If the corrected endurance strength is 70 MPa, estimated shaft life will be

(A) 1071 cycles (B) 15000 cycles

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(C) 281914 cycles (D) 928643 cycles

MCQ 5.30 A 60 mm long and 6 mm thick fillet weld carries a steady load of 15 kN along the weld. The shear strength of the weld material is equal to 200 MPa . The factor of safety is

(A) 2.4 (B) 3.4

(C) 4.8 (D) 6.8

YEAR 2005 ONE MARK

MCQ 5.31 Which one of the following is criterion in the design of hydrodynamic journal bearings ?

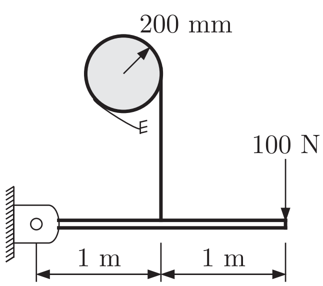
(A) Sommerfeld number (B) Rating life

(C) Specific dynamic capacity (D) Rotation factor

YEAR 2005 TWO MARKS

• Common Data For Q.32 and Q.33

A band brake consists of a lever attached to one end of the band. The other end of the band is fixed to the ground. The wheel has a radius of 200 mm and the wrap angle of the band is 270°. The braking force applied to the lever is limited to 100 N and the coefficient of friction between the band and the wheel is 0.5. No other information is given.



MCQ 5.32 The maximum tension that can be generated in the band during braking is

(A) 1200 N (B) 2110 N

(C) 3224 N (D) 4420 N

MCQ 5.33 The maximum wheel torque that can be completely braked is

(A) 200 Nm (B) 382 Nm

(C) 604 Nm (D) 844 Nm

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YEAR 2004 ONE MARK

MCQ 5.34 Two mating spur gears have 40 and 120 teeth respectively. The pinion rotates at 1200 rpm and transmits a torque of 20 Nm. The torque transmitted by the gear is

(A) 6.6 Nm (B) 20 Nm

(C) 40 Nm (D) 60 Nm

MCQ 5.35 In terms of theoretical stress concentration factor (K_t) and fatigue stress concentration factor (K_f), the notch sensitivity 'q' is expressed as

(A) $\frac{(K_f - 1)}{(K_t - 1)}$ (B) $\frac{(K_f - 1)}{(K_t + 1)}$

(C) $\frac{(K_t - 1)}{(K_f - 1)}$ (D) $\frac{(K_f + 1)}{(K_t + 1)}$

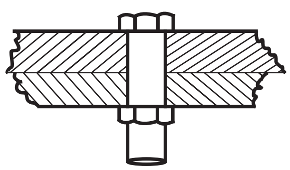
MCQ 5.36 The S-N curve for steel becomes asymptotic nearly at

(A) 10^3 cycles (B) 10^4 cycles

(C) 10^6 cycles (D) 10^9 cycles

YEAR 2004 TWO MARKS

MCQ 5.37 In a bolted joint two members are connected with an axial tightening force of 2200 N. If the bolt used has metric threads of 4 mm pitch, the torque required for achieving the tightening force is



(A) 0.7 Nm (B) 1.0 Nm

(C) 1.4 Nm (D) 2.8 Nm

MCQ 5.38 Match the following

	Type of gears		Arrangement of shafts
P.	Bevel gears	**1.**	Non-parallel off-set shafts
Q.	Worm gears	**2.**	Non-parallel intersecting shafts
R.	Herringbone gears	**3.**	Non-parallel, non-intersecting shafts
S.	Hypoid gears	**4.**	Parallel shafts

(A) P-4 Q-2 R-1 S-3

(B) P-2 Q-3 R-4 S-1

(C) P-3 Q-2 R-1 S-4

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(D) P-1 Q-3 R-4 S-2

YEAR 2003 ONE MARK

MCQ 5.39 A wire rope is designated as 6×19 standard hoisting. The numbers 6×19 represent

(A) diameter in millimeter $\times$ length in meter

(B) diameter in centimeter $\times$ length in meter

(C) number of strands $\times$ numbers of wires in each strand

(D) number of wires in each strand $\times$ number of strands

YEAR 2003 TWO MARKS

MCQ 5.40 Square key of side "$d/4$" each and length 'l' is used to transmit torque "T" from the shaft of diameter "d" to the hub of a pulley. Assuming the length of the key to be equal to the thickness of pulley, the average shear stress developed in the key is given by

(A) $\frac{4T}{ld}$ (B) $\frac{16T}{ld^2}$

(C) $\frac{8T}{ld^2}$ (D) $\frac{16T}{\pi d^3}$

MCQ 5.41 In a band brake the ratio of tight side band tension to the tension on the slack side is 3. If the angle of overlap of band on the drum is $180°$, the coefficient of friction required between drum and the band is

(A) 0.20 (B) 0.25

(C) 0.30 (D) 0.35

• Common Data For Q.42 and Q.43

The overall gear ratio in a 2 stage speed reduction gear box (with all spur gears) is 12. The input and output shafts of the gear box are collinear. The counter shaft which is parallel to the input and output shafts has a gear (Z_2 teeth) and pinion ($Z_3 = 15$ teeth) to mesh with pinion ($Z_1 = 16$ teeth) on the input shaft and gear (Z_4 teeth) on the output shaft respectively. It was decided to use a gear ratio of 4 with 3 module in the first stage and 4 module in the second stage.

MCQ 5.42 Z_2 and Z_4 are

(A) 64 and 45 (B) 45 and 64

(C) 48 and 60 (D) 60 and 48

MCQ 5.43 The centre distance in the second stage is

(A) 90 mm (B) 120 mm

(C) 160 mm (D) 240 mm

YEAR 2002 ONE MARK

MCQ 5.44 The minimum number of teeth on the pinion to operate without interference in standard full height involute teeth gear mechanism with 20° pressure angle is

(A) 14 (B) 12

(C) 18 (D) 32

YEAR 2002 TWO MARKS

MCQ 5.45 The coupling used to connect two shafts with large angular misalignment is

(A) a flange coupling

(B) an Oldham's coupling

(C) a flexible bush coupling

(D) a Hooke's joint

MCQ 5.46 A static load is mounted at the centre of a shaft rotating at uniform angular velocity. This shaft will be designed for

(A) the maximum compressive stress (static)

(B) the maximum tensile (static)

(C) the maximum bending moment (static)

(D) fatigue loading

MCQ 5.47 Large speed reductions (greater than 20) in one stage of a gear train are possible through

(A) spur gearing

(B) worm gearing

(C) bevel gearing

(D) helical gearing

MCQ 5.48 If the wire diameter of a closed coil helical spring subjected to compressive load is increased from $1\,\text{cm}$ to $2\,\text{cm}$, other parameters remaining same, the deflection will decrease by a factor of

(A) 16 (B) 8

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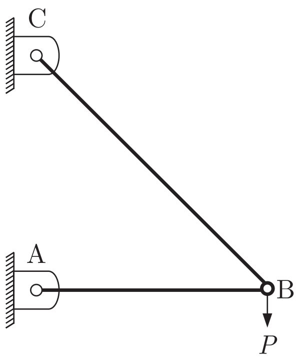
(C) 4 (D) 2

YEAR 2001 ONE MARK

MCQ 5.49 Bars AB and BC, each of negligible mass, support load P as shown in the figure. In this arrangement,



(A) bar AB is subjected to bending but bar BC is not subjected to bending.

(B) bar AB is not subjected to bending but bar BC is subjected to bending.

(C) neither bar AB nor bar BC is subjected to bending.

(D) both bars AB and BC are subjected to bending.

YEAR 2001 TWO MARKS

MCQ 5.50 Two helical tensile springs of the same material and also having identical mean coil diameter and weight, have wire diameters d and $d/2$. The ratio of their stiffness is

(A) 1 (B) 4

(C) 64 (D) 128

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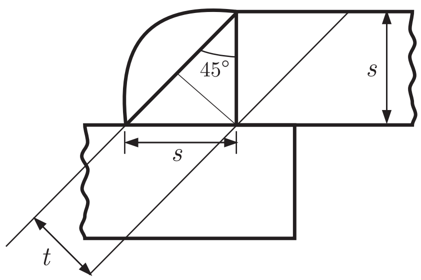
SOLUTION

SOL 5.1 Option (C) is correct.

Given : Width of fillets $s = 10\,\text{mm}$, $l = 30\,\text{mm}$, $\tau = 94\,\text{MPa}$



The shear strength of the joint for single parallel fillet weld is,

$$P = \text{Throat Area} \times \text{Allowable stress}$$

$$= t \times l \times \tau$$

From figure $$t = s\sin 45^\circ = 0.707\,s$$

$$P = 0.707 \times s \times l \times \tau$$

$$= 0.707 \times (0.01) \times (0.03) \times (94 \times 10^6)$$

$$= 19937\,\text{N or } 19.93\,\text{kN}$$

SOL 5.2 Option (B) is correct.

Given : $$T_1 = 400\,\text{N},\ \mu = 0.25,\ \theta = 180^\circ = 180^\circ \times \frac{\pi}{180^\circ} = \pi\,\text{rad}.$$

$$D = 0.5\,\text{m},\ r = \frac{D}{2} = 0.25\,\text{m}$$

For the band brake, the limiting ratio of the tension is given by the relation,

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

$$\frac{400}{T_2} = e^{0.25 \times \pi} = 2.19$$

$$T_2 = \frac{400}{2.19} = 182.68\,\text{N}$$

For Band-drum brake, Braking Torque is

$$T_B = (T_1 - T_2) \times r$$

$$= (400 - 182.68) \times 0.25 = 54.33\,\text{Nm} \cong 54.4\,\text{Nm}$$

SOL 5.3 Option (B) is correct.

$$F.O.S = \frac{\text{Allowable shear stress}}{\text{Design shear stress}}$$

Design shear stress for solid circular shaft

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$$\tau = \frac{16T}{\pi d^3} = \frac{16 \times 50 \times 10^3}{\pi d^3} \qquad \text{From } \frac{T}{J} = \frac{\tau}{r}$$

Therefore
$$F.O.S = \frac{140 \times \pi d^3}{16 \times 50 \times 10^3}$$

or,
$$2 = \frac{140 \times \pi d^3}{16 \times 50 \times 10^3}$$

$$d^3 = \frac{2 \times 16 \times 50 \times 10^3}{140 \times \pi}$$

$$d = 15.38 \text{ mm} \cong 16 \text{ mm}$$

SOL 5.4 Option (B) is correct.

Given : $W_P = 30$ kN, $W_Q = 45$ kN

Life of bearing,
$$L = \left(\frac{C}{W}\right)^k \times 10^6 \text{ revolutions}$$
$$C = \text{Basic dynamic load rating} = \text{Constant}$$

For ball bearing, $k = 3$

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

These are the identical bearings. So for the Life of P and Q.
$$\left(\frac{L_P}{L_Q}\right) = \left(\frac{W_Q}{W_P}\right)^3 = \left(\frac{45}{30}\right)^3 = \left(\frac{3}{2}\right)^3 = \frac{27}{8}$$

SOL 5.5 Option (D) is correct.

Given : $b = 80$ mm, $d = 250$ mm, $\mu = 0.25$, $\theta = 270°$, $T_B = 1000$ N-m

Let, $T_1 \rightarrow$ Tension in the tight side of the band (Maximum Tension)

$T_2 \rightarrow$ Tension in the slack side of the band (Minimum Tension)

Braking torque on the drum,
$$T_B = (T_1 - T_2)\, r$$
$$T_1 - T_2 = \frac{T_B}{r} = \frac{1000}{0.125} = 8000 \text{ N} \qquad \text{...(i)}$$

We know that limiting ratio of the tension is given by,
$$\frac{T_1}{T_2} = e^{\mu\theta} = e^{\left(0.25 \times \frac{\pi}{180} \times 270\right)} = 3.246$$
$$T_2 = \frac{T_1}{3.246}$$

Substitute T_2 in equation (i), we get
$$T_1 - \frac{T_1}{3.246} = 8000 \quad \Rightarrow \quad 3.246\,T_1 - T_1 = 25968$$
$$2.246\,T_1 = 25968 \quad \Rightarrow \quad T_1 = \frac{25968}{2.246} = 11.56 \text{ kN}$$

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SOL 5.6 Option (B) is correct.

Given : $d = 6$ mm, $l = 12$ mm, $P = 1000$ N

Each rivets have same diameter, So equal Load is carried by each rivet.

Primary or direct force on each rivet,

$$F = \frac{P}{4} = \frac{1000}{4} = 250 \text{ N}$$

Shear area of each rivet is,

$$A = \frac{\pi}{4}(6 \times 10^{-3})^2 = 28.26 \times 10^{-6} \text{ mm}^2$$

Direct shear stress on each rivet,

$$\tau = \frac{F}{A} = \frac{250}{28.26 \times 10^{-6}} = 8.84 \times 10^6 \simeq 8.8 \text{ MPa}$$

SOL 5.7 Option (A) is correct.

Given : $d = 50$ mm, $D = 50.05$ mm, $l = 20$ mm, $N = 1200$ rpm, $\mu = 0.03$ Pa s

Tangential velocity of shaft,

$$u = \frac{\pi dN}{60} = \frac{3.14 \times 50 \times 10^{-3} \times 1200}{60} = 3.14 \text{ m/sec}$$

And Radial clearance, $$y = \frac{D-d}{2} = \frac{50.05 - 50}{2} = 0.025 \text{ mm}$$

Shear stress from the Newton's law of viscosity,

$$\tau = \mu \times \frac{u}{y} = 0.03 \times \frac{3.14}{0.025 \times 10^{-3}} = 3768 \text{ N/m}^2$$

Shear force on the shaft,

$$F = \tau \times A = 3768 \times (\pi \times d \times l)$$
$$= 3768 \times 3.14 \times 50 \times 10^{-3} \times 20 \times 10^{-3} = 11.83 \text{ N}$$

Torque, $$T = F \times \frac{d}{2} = 11.83 \times \frac{50}{2} \times 10^{-3} = 0.2957 \text{ N-m}$$

We know that power loss,

$$P = \frac{2\pi NT}{60} = \frac{2 \times 3.14 \times 1200 \times 0.2957}{60}$$
$$= 37.13 \text{ W} \simeq 37 \text{ W}$$

SOL 5.8 Option (A) is correct.

Given : S_u or $\sigma_u = 600$ MPa, S_y or $\sigma_y = 420$ MPa, S_e or $\sigma_e = 240$ MPa,
$d = 30$ mm $F_{max} = 160$ kN (Tension), $F_{min} = -40$ kN (Compression)

Maximum stress, $$\sigma_{max} = \frac{F_{max}}{A} = \frac{160 \times 10^3}{\frac{\pi}{4}(30)^2} = 226.47 \text{ MPa}$$

Minimum stress, $$\sigma_{min} = \frac{F_{min}}{A} = -\frac{40 \times 10^3}{\frac{\pi}{4} \times (30)^2} = -56.62 \text{ MPa}$$

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Mean stress, $\sigma_m = \frac{\sigma_{\max} + \sigma_{\min}}{2} = \frac{226.47 - 56.62}{2} = 84.925\text{ MPa}$

Variable stress, $\sigma_v = \frac{\sigma_{\max} - \sigma_{\min}}{2} = \frac{226.47 - (-56.62)}{2}$

$= 141.545\text{ MPa}$

From the Soderberg's criterion,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v}{\sigma_e}$$

$$\frac{1}{F.S.} = \frac{84.925}{420} + \frac{141.545}{240} = 0.202 + 0.589 = 0.791$$

So, $F.S. = \frac{1}{0.791} = 1.26$

SOL 5.9 Option (A) is correct.

Given : $m = 4\text{ mm}$, $Z = 21$, $P = 15\text{ kW} = 15 \times 10^3\text{ W}$ $N = 960\text{ rpm}$
$b = 25\text{ mm}$, $\phi = 20^\circ$

Pitch circle diameter, $D = mZ = 4 \times 21 = 84\text{ mm}$

Tangential Force is given by,

$$F_T = \frac{T}{r} \qquad \text{...(i)}$$

Power transmitted, $P = \frac{2\pi NT}{60} \Rightarrow T = \frac{60P}{2\pi N}$

Then $F_T = \frac{60P}{2\pi N} \times \frac{1}{r}$ $\qquad r =$ Pitch circle radius

$$= \frac{60 \times 15 \times 10^3}{2 \times 3.14 \times 960} \times \frac{1}{42 \times 10^{-3}}$$

$$= 3554.36\text{ N} \simeq 3552\text{ N}$$

SOL 5.10 Option (B) is correct.

From Lewis equation

$$\sigma_b = \frac{F_T p_d}{by} = \frac{F_T}{b \times y \times m} \qquad p_d = \frac{\pi}{p_c} = \frac{\pi}{\pi m} = \frac{1}{m}$$

$$= \frac{3552}{25 \times 10^{-3} \times 0.32 \times 4 \times 10^{-3}}$$

$$\sigma_b = 111\text{ MPa}$$

Minimum allowable (working stress)

$$\sigma_W = \sigma_b \times C_v = 111 \times 1.5 = 166.5\text{ MPa}$$

SOL 5.11 Option (A) is correct.

Given : $d = 40\text{ mm}$, $l = 40\text{ mm}$, $\omega = 20\text{ rad/sec}$
$Z(\mu) = 20\text{ mPa-s} = 20 \times 10^{-3}\text{ Pa-s}$, $c(y) = 0.020\text{ mm}$

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Shear stress, $\tau = \mu \frac{u}{y}$ From the Newton's law of viscosity...(i)

$$u = r\omega = 0.020 \times 20 = 0.4 \text{ m/sec}$$

$$\tau = \frac{20 \times 10^{-3} \times 0.4}{0.020 \times 10^{-3}} = 400 \text{ N/m}^2$$

Shear force is generated due to this shear stress,

$$F = \tau A = \tau \times \pi dl \qquad A = \pi dl = \text{Area of shaft}$$

$$= 400 \times 3.14 \times 0.040 \times 0.040 = 2.0096 \text{ N}$$

Loss of torque, $T = F \times r = 2.0096 \times 0.020$

$$= 0.040192 \text{ N-m} \simeq 0.040 \text{ N-m}$$

SOL 5.12 Option (B) is correct.

Given : $d_1 = 100$ mm $\Rightarrow r_1 = 50$ mm, $d_2 = 40$ mm $\Rightarrow r_1 = 20$ mm

$p = 2$ MPa $= 2 \times 10^6$ Pa, $\mu = 0.4$

When the pressure is uniformly distributed over the entire area of the friction faces, then total frictional torque acting on the friction surface or on the clutch is given by,

$$T = 2\pi\mu p\left[\frac{(r_1)^3 - (r_2)^3}{3}\right]$$

$$= \frac{2}{3} \times 3.14 \times 0.4 \times 2 \times 10^6[(50)^3 - (20)^3] \times 10^{-9}$$

$$= 195.39 \text{ N-m} \simeq 196 \text{ N-m}$$

SOL 5.13 Option (B) is correct.

Given : $m = 3$ mm, $Z = 16$, $b = 36$ mm, $\phi = 20°$, $P = 3$ kW

$N = 20$ rev/sec $= 20 \times 60$ rpm $= 1200$ rpm, $C_v = 1.5$, $y = 0.3$

Module, $m = \frac{D}{Z}$

$$D = m \times Z = 3 \times 16 = 48 \text{ mm}$$

Power, $P = \frac{2\pi NT}{60}$

$$T = \frac{60P}{2\pi N} = \frac{60 \times 3 \times 10^3}{2 \times 3.14 \times 1200}$$

$$= 23.88 \text{ N-m} = 23.88 \times 10^3 \text{ N-mm}$$

Tangential load, $W_T = \frac{T}{R} = \frac{2T}{D} = \frac{2 \times 23.88 \times 10^3}{48} = 995 \text{ N}$

From the lewis equation Bending stress (Beam strength of Gear teeth)

$$\sigma_b = \frac{W_T P_d}{by} = \frac{W_T}{bym} \qquad \left[P_d = \frac{\pi}{P_C} = \frac{\pi}{\pi m} = \frac{1}{m}\right]$$

$$= \frac{995}{36 \times 10^{-3} \times 0.3 \times 3 \times 10^{-3}}$$

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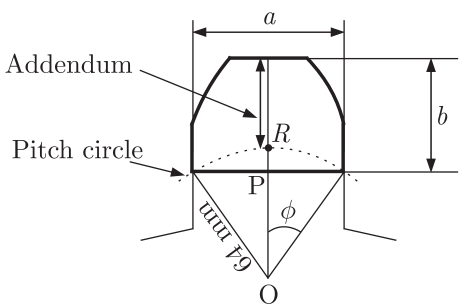
$$\sigma_b = \frac{995}{3.24 \times 10^{-5}} = 30.70 \times 10^6 \text{ Pa} = 30.70 \text{ MPa}$$

Permissible working stress

$$\sigma_W = \sigma_b \times C_v = 30.70 \times 1.5 = 46.06 \text{ MPa} \cong 46 \text{ MPa}$$

SOL 5.14 Option (D) is correct.



Given : $m = 4$, $Z = 32$, Tooth space = Tooth thickness = a

We know that, $m = \frac{D}{Z}$

Pitch circle diameter, $D = mZ = 4 \times 32 = 128$ mm

And for circular pitch, $P_c = \pi m = 3.14 \times 4 = 12.56$ mm

We also know that circular pitch,

$$P_c = \text{Tooth space} + \text{Tooth thickness}$$
$$= a + a = 2a$$
$$a = \frac{P_c}{2} = \frac{12.56}{2} = 6.28 \text{ mm}$$

From the figure, $b = \text{addendum} + PR$

or $\sin\phi = \frac{PQ}{OQ} = \frac{a/2}{64} = \frac{3.14}{64}$

$$\phi = \sin^{-1}(0.049) = 2.81^\circ$$
$$OP = 64\cos 2.81^\circ = 63.9 \text{ mm}$$
$$PR = OR - OP = 64 - 63.9 = 0.1 \text{ mm}$$
$$OR = \text{Pitch circle radius}$$

And $b = m + PR = 4 + 0.1 = 4.1$ mm

Therefore, $a = 6.28$ mm and $b = 4.1$ mm

SOL 5.15 Option (A) is correct.

	Types of Gear		Description
P.	Helical	**2.**	Axes parallel and teeth are inclined to the axis
Q.	Spiral Bevel	**4.**	Axes are perpendicular and intersecting, and teeth are inclined to the axis

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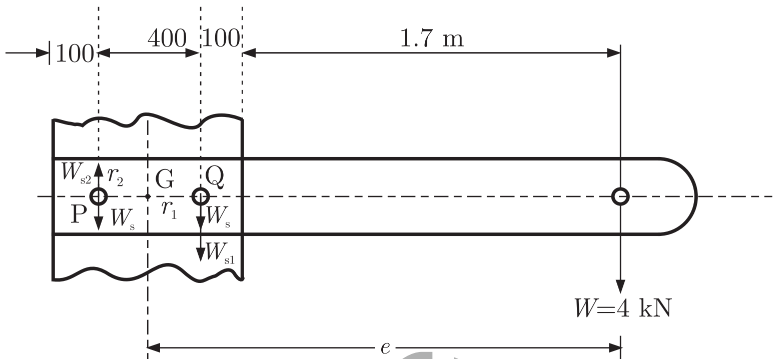
R.	Hypoid	**1.**	Axes non parallel and non-intersecting
S.	Rack and pinion	**6.**	Axes are parallel and one of the gear has infinite radius

So, correct pairs are P-2, Q-4, R-1, S-6

SOL 5.16 Option (A) is correct.



In this figure W_S represent the primary shear load whereas W_{S1} and W_{S2} represent the secondary shear loads.

Given : $A = 10 \times 50\text{ mm}^2$, $n = 2$, $W = 4\text{ kN} = 4 \times 10^3\text{ N}$

We know that primary shear load on each bolt acting vertically downwards,

$$W_s = \frac{W}{n} = \frac{4\text{ kN}}{2} = 2\text{ kN}$$

Since both the bolts are at equal distances from the centre of gravity G of the two bolts, therefore the secondary shear load on each bolt is same.

For secondary shear load, taking the moment about point G,

$$W_{s1} \times r_1 + W_{s2} \times r_2 = W \times e$$

$$r_1 = r_2 \text{ and } W_{s1} = W_{s2}$$

So,

$$2r_1 W_{s1} = 4 \times 10^3 \times (1.7 + 0.2 + 0.1)$$

$$2 \times 0.2 \times W_{s1} = 4 \times 10^3 \times 2$$

$$W_{s1} = \frac{8 \times 10^3}{2 \times 0.2} = 20 \times 10^3 = 20\text{ kN}$$

SOL 5.17 Option (B) is correct.

From the figure, resultant Force on bolt P is

$$F = W_{s2} - W_s = 20 - 2 = 18\text{ kN}$$

Shear stress on bolt P is,

$$\tau = \frac{F}{\text{Area}} = \frac{18 \times 10^3}{\frac{\pi}{4} \times (12 \times 10^{-3})^2} = 159.23\text{ MPa} \simeq 159\text{ MPa}$$

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SOL 5.18 Option (D) is correct.

Given : $W_1 = F$, $W_2 = 2F$, $L_1 = 8000$ hr

We know that, life of bearing is given by

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

For ball bearing, $k = 3$, $L = \left(\frac{C}{W}\right)^3 \times 10^6$ revolution

For initial condition life is,

$$L_1 = \left(\frac{C}{F}\right)^3 \times 10^6$$

$$8000 \text{ hr} = \left(\frac{C}{F}\right)^3 \times 10^6 \qquad \text{...(i)}$$

For final load, $L_2 = \left(\frac{C}{2F}\right)^3 \times 10^6 = \frac{1}{8} \times \left(\frac{C}{F}\right)^3 \times 10^6$

$$= \frac{1}{8}(8000 \text{ hr}) = 1000 \text{ hr} \qquad \text{From equation (i)}$$

SOL 5.19 Option (B) is correct.

Given : $d = 200$ mm, $t = 1$ mm, $\sigma_u = 800$ MPa, $\sigma_e = 400$ MPa

Circumferential stress induced in spherical pressure vessel is,

$$\sigma = \frac{p \times r}{2t} = \frac{p \times 100}{2 \times 1} = 50p \text{ MPa}$$

Given that, pressure vessel is subject to an internal pressure varying from 4 to 8 MPa.

So, $\sigma_{\min} = 50 \times 4 = 200$ MPa

$$\sigma_{\max} = 50 \times 8 = 400 \text{ MPa}$$

Mean stress, $\sigma_m = \frac{\sigma_{\min} + \sigma_{\max}}{2} = \frac{200 + 400}{2} = 300$ MPa

Variable stress, $\sigma_v = \frac{\sigma_{\max} - \sigma_{\min}}{2} = \frac{400 - 200}{2} = 100$ MPa

From the Goodman method,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_v}{\sigma_e} = \frac{300}{800} + \frac{100}{400} = \frac{3}{8} + \frac{1}{4} = \frac{5}{8} \quad \Rightarrow \quad \text{F.S.} = \frac{8}{5} = 1.6$$

SOL 5.20 Option (B) is correct.

Given : $d = 50$ mm, $l = 50$ mm, $N = 20$ rps,

$Z = 20$ mPa - sec $= 20 \times 10^{-3}$ Pa - sec

Radial clearance $= 50\,\mu\text{m} = 50 \times 10^{-3}$ mm, Load $= 2$ kN

We know that,

p = Bearing Pressure on the projected bearing area

$$= \frac{\text{Load on the journal}}{l \times d}$$

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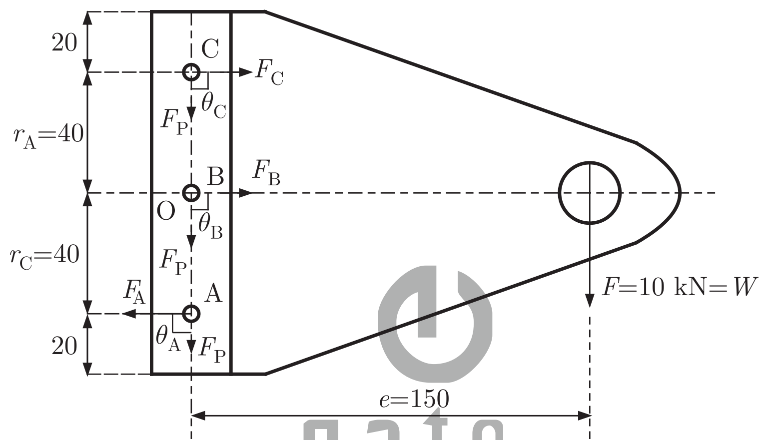
$$= \frac{2 \times 10^3}{50 \times 50} = 0.8\ \text{N/mm}^2 = 0.8 \times 10^6\ \text{N/m}^2$$

$$\text{Sommerfeld Number} = \frac{ZN}{p}\left(\frac{d}{c}\right)^2 \qquad c = \text{diameteral clearance} = 2 \times \text{radial clearance}$$

$$S.N. = \frac{20 \times 10^{-3} \times 20}{0.8 \times 10^6} \times \left(\frac{50}{100 \times 10^{-3}}\right)^2$$

$$= \frac{20 \times 10^{-3} \times 20}{0.8 \times 10^6} \times \left(\frac{1}{2}\right)^2 \times 10^6 = 0.125$$

SOL 5.21 Option (A) is correct.



Given : Diameter of bolt $d = 10$ mm, $F = 10$ kN, No. of bolts $n = 3$

Direct or Primary shear load of each rivet

$$F_P = \frac{F}{n} = \frac{10 \times 10^3}{3}\ \text{N}$$

$$F_P = 3333.33\ \text{N}$$

The centre of gravity of the bolt group lies at O (due to symmetry of figure).

$$e = 150\ \text{mm} \qquad \text{(eccentricity given)}$$

Turning moment produced by the load F due to eccentricity

$$= F \times e = 10 \times 10^3 \times 150$$

$$= 1500 \times 10^3\ \text{N-mm}$$

Secondary shear load on bolts from fig. $r_A = r_C = 40$ mm and $r_B = 0$

$$\text{We know that } F \times e = \frac{F_A}{r_A}[(r_A)^2 + (r_B)^2 + (r_C)^2]$$

$$= \frac{F_A}{r_A} \times [2(r_A)^2] \qquad (r_A = r_C \text{ and } r_B = 0)$$

$$1500 \times 10^3 = \frac{F_A}{40} \times [2(40)^2] = 80F_A$$

$$F_A = \frac{1500 \times 10^3}{80} = 18750\ \text{N}$$

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$$F_B = 0 \qquad (r_B = 0)$$

$$F_C = F_A \times \frac{r_C}{r_A} = 18750 \times \frac{40}{40}$$

$$= 18750\,\text{N}$$

From fig we find that angle between

$$F_A \text{ and } F_P = \theta_A = 90^\circ$$

$$F_B \text{ and } F_P = \theta_B = 90^\circ$$

$$F_C \text{ and } F_P = \theta_C = 90^\circ$$

Resultant load on bolt A,

$$R_A = \sqrt{(F_P)^2 + (F_A)^2 + 2F_P \times F_A \cos\theta_A}$$

$$= \sqrt{(3333.33)^2 + (18750)^2 + 2 \times 3333.33 \times 18750 \times \cos 90^\circ}$$

$$R_A = 19044\,\text{N}$$

Maximum shear stress at A

$$\tau_A = \frac{R_A}{\frac{\pi}{4}(d)^2} = \frac{19044}{\frac{\pi}{4}(10)^2} = 242.6\,\text{MPa}$$

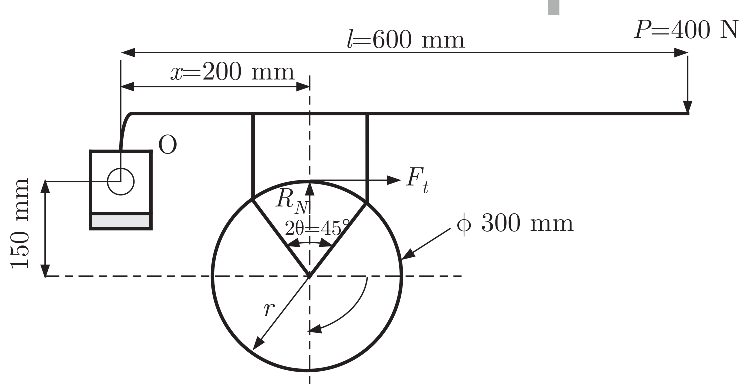
Resultant load on Bolt B,

$$R_B = F_P = 3333.33\,\text{N} \qquad (F_B = 0)$$

Maximum shear stress at B,

$$\tau_B = \frac{R_B}{\frac{\pi}{4}(d)^2} = \frac{3333.33}{\frac{\pi}{4} \times (10)^2} = 42.5\,\text{MPa}$$

SOL 5.22 Option (C) is correct.



Given : $P = 400\,\text{N},\ r = \frac{300}{2}\,\text{mm} = 150\,\text{mm},\ l = 600\,\text{mm}$

$x = 200\,\text{mm},\ \mu = 0.25$ and $2\theta = 45^\circ$

Let, $R_N \rightarrow$ Normal force pressing the brake block on the wheel

$F_t \rightarrow$ Tangential braking force or the frictional force acting at the contact surface of the block & the wheel.

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Here the line of action of tangential braking force F_t passes through the fulcrum O of the lever and brake wheel rotates clockwise. Then for equilibrium, Taking the moment about the fulcrum O,

$$R_N \times x = P \times l$$

$$R_N = \frac{P \times l}{x} = \frac{400 \times 0.6}{0.2} = 1200 \text{ N}$$

Tangential braking force on the wheel,

$$F_t = \mu R_N$$

Braking Torque, $$T_B = F_t \times r = \mu R_N \times r$$

$$= 0.25 \times 1200 \times 0.15 = 45 \text{ N-m}$$

SOL 5.23 Option (C) is correct.

Given : $d = 20$ mm, $l = 700$ mm,

$E = 200 \text{ GPa} = 200 \times 10^9 \text{ N/m}^2 = 200 \times 10^3 \text{ N/mm}^2$

Compressive or working Load $= 10$ kN

According to Euler's theory, the crippling or buckling load (W_{cr}) under various end conditions is given by the general equation,

$$W_{cr} = \frac{c\pi^2 EI}{l^2} \qquad ...(i)$$

Given that one end is guided at the piston end and hinged at the other end.

So, $c = 2$

From equation (i),

$$W_{cr} = \frac{2\pi^2 EI}{l^2} = \frac{2\pi^2 E}{l^2} \times \frac{\pi}{64} d^4 \qquad I = \frac{\pi}{64} d^4$$

$$= \frac{2 \times 9.81 \times 200 \times 10^3}{(700)^2} \times \frac{3.14}{64} \times (20)^4$$

$$= 62864.08 \text{ N} = 62.864 \text{ kN}$$

We know that, factor of safety (FOS)

$$\text{FOS} = \frac{\text{Crippling Load}}{\text{Working Load}} = \frac{62.864}{10} = 6.28$$

The most appropriate option is (C).

SOL 5.24 Option (B) is correct.

Given : $Z_P = 20$, $Z_G = 40$, $N_P = 30$ rev/sec, $P = 20 \text{ kW} = 20 \times 10^3$ W, $m = 5$ mm

Module, $$m = \frac{D}{Z} = \frac{D_P}{Z_P} = \frac{D_G}{Z_G}$$

$$D_P = m \times Z_P = 5 \times 20 = 100 \text{ mm}$$

or, $$D_G = m \times Z_G = 5 \times 40 = 200 \text{ mm}$$

Centre distance for the gear set,

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$$L = \frac{D_P + D_G}{2} = \frac{100 + 200}{2} = 150 \text{ mm}$$

SOL 5.25 Option (C) is correct.

Given :

Length of line of action, $L = 19$ mm

Pressure angle, $\phi = 20°$

$$\text{Length of arc of contact} = \frac{\text{Length of path of contact}\,(L)}{\cos\phi}$$

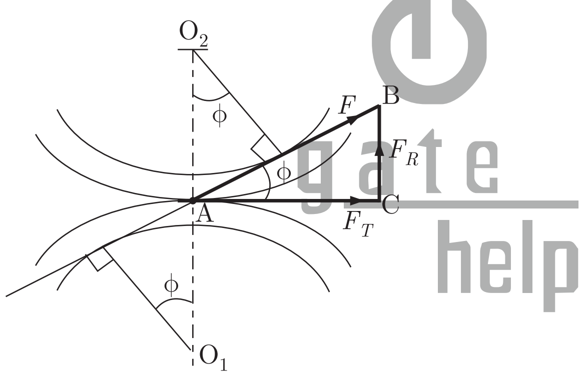
$$= \frac{19}{\cos 20°} = 20.21 \text{ mm}$$

Contact ratio or number of pairs of teeth in contact,

$$= \frac{\text{Length of arc of contact}}{\text{circular pitch}}$$

$$= \frac{20.21}{\pi m} = \frac{20.21}{3.14 \times 5} = 1.29$$

SOL 5.26 Option (C) is correct.



Let, $T \rightarrow$ Torque transmitted in N - m

We know that power transmitted is,

$$P = T\omega = T \times \frac{2\pi N}{60}$$

$$T = \frac{60P}{2\pi N} = \frac{60 \times 20 \times 10^3}{2 \times 3.14 \times 1800} = 106.157 \text{ N-m}$$

$$F_T = \frac{T}{R_P} \qquad \text{Tangential load on the pinion}$$

$$= \frac{106.157}{0.05} = 2123.14 \text{ N}$$

From the geometry, total load due to power transmitted,

$$F = \frac{F_T}{\cos\phi} = \frac{2123.14}{\cos 20°} \simeq 2258.1 \text{ N}$$

SOL 5.27 Option (A) is correct.

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Given : $P = 5\,\text{kW},\ N = 2000\,\text{rpm},\ \mu = 0.25,\ r_2 = 25\,\text{mm} = 0.025\,\text{m},$
$p = 1\,\text{MPa}$

Power transmitted, $$P = \frac{2\pi NT}{60}$$

Torque, $$T = \frac{60P}{2\pi N} = \frac{60 \times 5 \times 10^3}{2 \times 3.14 \times 2000} = 23.885\,\text{N-m}$$

When pressure is uniformly distributed over the entire area of the friction faces, then total frictional torque acting on the friction surface or on the clutch,

$$T = 2\pi\mu p\left[\frac{(r_1)^3 - (r_2)^3}{3}\right]$$

$$23.885 \times 3 = 2 \times 3.14 \times 0.25 \times 1 \times 10^6 \times [r_1^3 - (0.025)^3]$$

$$r_1^3 - (0.025)^3 = \frac{23.885 \times 3}{2 \times 3.14 \times 0.25 \times 10^6}$$

$$r_1^3 - 1.56 \times 10^{-5} = 45.64 \times 10^{-6} = 4.564 \times 10^{-5}$$

$$r_1^3 = (4.564 + 1.56) \times 10^{-5} = 6.124 \times 10^{-5}$$

$$r_1 = (6.124 \times 10^{-5})^{1/3} = 3.94 \times 10^{-2}\,\text{m} = 39.4\,\text{mm}$$

SOL 5.28 Option (A) is correct.

Given : $Z_P = 19,\ Z_G = 37,\ m = 5\,\text{mm}$

Also, $$m = \frac{D}{Z}$$

For pinion, pitch circle diameter is,

$$D_P = m \times Z_P = 5 \times 19 = 95\,\text{mm}$$

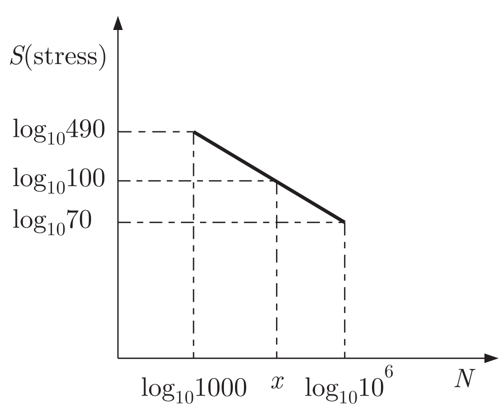
And pitch circle diameter of the gear,

$$D_G = m \times Z_G = 5 \times 37 = 185\,\text{mm}$$

Now, centre distance between the gear pair (shafts),

$$L = \frac{D_P}{2} + \frac{D_G}{2} = \frac{95 + 185}{2} = 140\,\text{mm}$$

SOL 5.29 Option (C) is correct.



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We know that in S-N curve the failure occurs at 10^6 cycles (at endurance strength)

We have to make the S-N curve from the given data, on the scale of $\log_{10}$.

Now equation of line whose end point co-ordinates are

$$(\log_{10}1000, \log_{10}490) \text{ and } (\log_{10}10^6, \log_{10}70)$$

or $$(3, \log_{10}490) \text{ and } (6, \log_{10}70),$$

$$\frac{y-\log_{10}490}{x-3} = \frac{\log_{10}70-\log_{10}490}{6-3} \qquad \left(\frac{y-y_1}{x-x_1} = \frac{y_2-y_1}{x_2-x_1}\right)$$

$$\frac{y-2.69}{x-3} = \frac{1.845-2.69}{3}$$

$$y-2.69 = -0.281(x-3) \qquad \text{...(i)}$$

Given, the shaft is subject to an alternating stress of 100 MPa

So, $$y = \log_{10}100 = 2$$

Substitute this value in equation (i), we get

$$2-2.69 = -0.281(x-3)$$

$$-0.69 = -0.281x+0.843$$

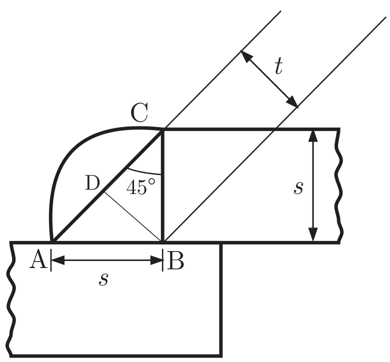
$$x = \frac{-0.843-0.69}{-0.281} = 5.455$$

And $$\log_{10}N = 5.455$$

$$N = 10^{5.455} = 285101$$

The nearest shaft life is 281914 cycles.

SOL 5.30 Option (B) is correct.



Given : $l = 60\text{ mm} = 0.06\text{ m}$, $s = 6\text{ mm} = 0.006\text{ m}$, $P = 15\text{ kN} = 15\times10^3\text{ N}$

$$\text{Shear strength} = 200\text{ MPa}$$

We know that, if τ is the allowable shear stress for the weld metal, then the shear strength of the joint for single parallel fillet weld,

$$P = \text{Throat Area} \times \text{Allowable shear stress}$$

$$= t\times l\times\tau$$

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$$P = 0.707s \times l \times \tau \qquad t = s\sin 45^\circ = 0.707s$$

$$\tau = \frac{P}{0.707 \times s \times l} = \frac{15 \times 10^3}{0.707 \times 0.006 \times 0.06} = 58.93 \text{ MPa}$$

Factor of Safety,

$$FOS = \frac{\text{Shear strength}}{\text{Allowable shear stress}} = \frac{200 \text{ MPa}}{58.93 \text{ MPa}} = 3.39 \simeq 3.4$$

SOL 5.31 Option (A) is correct.

The coefficient of friction for a full lubricated journal bearing is a function of three variables, i.e.

$$\mu = \phi\left(\frac{ZN}{p}, \frac{d}{c}, \frac{l}{d}\right)$$

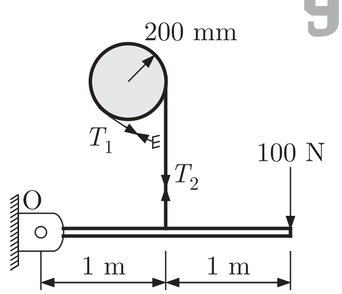
Here, $\frac{ZN}{p}$=Bearing characteristic Number, d=Diameter of the bearing

l=Length of the bearing, c=Diameteral clearance

$$\text{Sommerfeld Number} = \frac{ZN}{p}\left(\frac{d}{c}\right)^2$$

It is a dimensionless parameter used extensively in the design of journal bearing. i.e. sommerfeld number is also function of $\left(\frac{ZN}{p}, \frac{d}{c}\right)$. Therefore option (A) is correct.

SOL 5.32 Option (B) is correct.



Given : $r = 200 \text{ mm} = 0.2 \text{ m}$, $\theta = 270^\circ = 270 \times \frac{\pi}{180} = \frac{3\pi}{2}$ radian, $\mu = 0.5$

At the time of braking, maximum tension is generated at the fixed end of band near the wheel.

Let, $T_2 \rightarrow$ Tension in the slack side of band

$T_1 \rightarrow$ Tension in the tight side of band at the fixed end

Taking the moment about the point O,

$$T_2 \times 1 = 100 \times 2 \qquad \Rightarrow \quad T_2 = 200 \text{ N}$$

For the band brake, the limiting ratio of the tension is given by the relation

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

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$$T_1 = 200 \times e^{0.5 \times \frac{3\pi}{2}} = 200 \times 10.54 = 2108 \text{ N}$$
$$\simeq 2110 \text{ N}$$

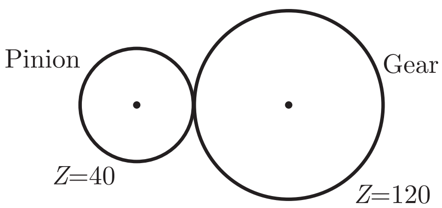
So, maximum tension that can be generated in the band during braking is equal to 2110 N

SOL 5.33 Option (B) is correct.

Maximum wheel torque or braking torque is given by,

$$T_W = (T_1 - T_2)\,r = (2110 - 200) \times 0.2 = 382 \text{ N-m}$$

SOL 5.34 Option (D) is correct.



Given : $Z_P = 40$ teeth, $Z_G = 120$ teeth, $N_P = 1200$ rpm, $T_P = 20$ N-m

Velocity Ratio, $\dfrac{Z_P}{Z_G} = \dfrac{N_G}{N_P}$

$$N_G = \frac{Z_P}{Z_G} \times N_P = \frac{40}{120} \times 1200 = 400 \text{ rpm}$$

Power transmitted is same for both pinion & Gear.

$$P = \frac{2\pi N_P T_P}{60} = \frac{2\pi N_G T_G}{60}$$
$$N_P T_P = N_G T_G$$
$$T_G = \frac{N_P T_P}{N_G} = \frac{1200}{400} \times 20 = 60 \text{ N-m}$$

So, the torque transmitted by the Gear is 60 N-m

SOL 5.35 Option (A) is correct.

When the notch sensitivity factor q is used in cyclic loading, then fatigue stress concentration factor may be obtained from the following relation.

$$K_f = 1 + q(K_t - 1)$$
$$K_f - 1 = q(K_t - 1)$$
$$q = \frac{K_f - 1}{K_t - 1}$$

SOL 5.36 Option (C) is correct.

The S-N curve for the steel is shown below :

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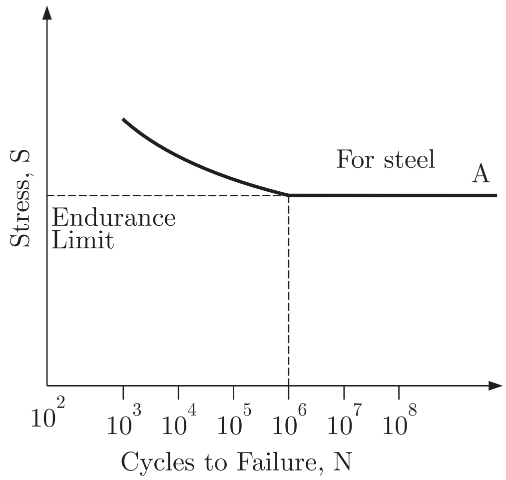

We can easily see from the S-N curve that, steel becomes asymptotic nearly at 10^6 cycles.

SOL 5.37 Option (C) is correct.

Given : $F_t = 2200\text{ N},\ p = 4\text{ mm} = 0.004\text{ m}$

Torque required for achieving the tightening force is,

$$T = F_t \times r = F_t \times \frac{\text{Pitch}}{2\pi} = 2200 \times \frac{0.004}{2 \times 3.14} = 1.4\text{ N-m}$$

SOL 5.38 Option (B) is correct.

	Type of Gears		Arrangement of shafts
P.	Bevel gears	**2.**	Non-parallel intersecting shafts
Q.	Worm gears	**3.**	Non-parallel, non-intersecting shafts
R.	Herringbone gears	**4.**	Parallel shafts
S.	Hypoid gears	**1.**	Non-parallel off-set shafts

So, correct pairs are P-2, Q-3, R-4, S-1.

SOL 5.39 Option (C) is correct.

The wire ropes are designated by the number of strands multiplied by the number of wires in each strand. Therefore,

$$6 \times 19 = \text{Number of strands} \times \text{Number of wires in each strand.}$$

SOL 5.40 Option (C) is correct.

Given : Diameter of shaft $= d$

Torque transmitted $= T$

Length of the key $= l$

We know that, width and thickness of a square key are equal.

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i.e. $$w = t = \frac{d}{4}$$

Force acting on circumference of shaft

$$F = \frac{T}{r} = \frac{2T}{d} \qquad (r = d/2)$$

Shearing Area, $$A = \text{width} \times \text{length} = \frac{d}{4} \times l = \frac{dl}{4}$$

Average shear stress, $$\tau = \frac{\text{Force}}{\text{shearing Area}} = \frac{2T/d}{dl/4} = \frac{8T}{ld^2}$$

SOL 5.41 Option (D) is correct.

Let, $T_1 \rightarrow$ Tension in the tight side of the band,

$T_2 \rightarrow$ Tension in the slack side of the band

$\theta \rightarrow$ Angle of lap of the band on the drum

Given : $\frac{T_1}{T_2} = 3$, $\theta = 180° = \frac{\pi}{180} \times 180 = \pi$ radian

For band brake, the limiting ratio of the tension is given by the relation,

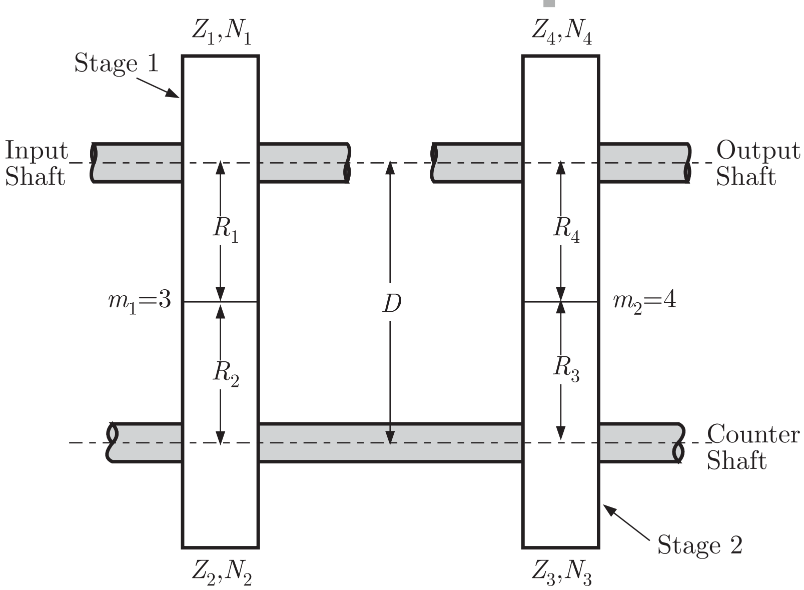
$$\frac{T_1}{T_2} = e^{\mu\theta} \text{ or } 2.3\log\left(\frac{T_1}{T_2}\right) = \mu\theta$$

$$2.3 \times \log(3) = \mu \times \pi$$

$$2.3 \times 0.4771 = \mu \times 3.14$$

$$\mu = \frac{1.09733}{3.14} = 0.349 \simeq 0.35$$

SOL 5.42 Option (A) is correct.



Let N_1, N_2, N_3 and N_4 are the speeds of pinion 1, gear 2, pinion 3 and gear 4 respectively.

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Given : $Z_1 = 16$ teeth , $Z_3 = 15$ teeth and $Z_4 = ?$, $Z_2 = ?$

Velocity ratio $$\frac{N_1}{N_4} = \frac{Z_2/Z_1}{Z_3/Z_4} \qquad N \propto 1/Z$$

$$= \frac{Z_2}{Z_1} \times \frac{Z_4}{Z_3} = 12 \qquad ...(i)$$

But for stage 1, $$\frac{N_1}{N_2} = \frac{Z_2}{Z_1} = 4 \qquad ...(ii)$$

So, $$4 \times \frac{Z_4}{Z_3} = 12 \qquad \text{from eq. (i)}$$

$$\frac{Z_4}{Z_3} = 3, \qquad \Rightarrow \quad Z_4 = 3 \times 15 = 45 \text{ teeth}$$

From equation (ii), $$Z_2 = 4 \times Z_1 = 4 \times 16 = 64 \text{ teeth}$$

SOL 5.43 Option (B) is correct.

Let centre distance in the second stage is D.

$$D = R_4 + R_3 = \frac{D_4 + D_3}{2}$$

But, $$\frac{D_4}{Z_4} = \frac{D_3}{Z_3} = 4 \qquad m = D/Z \text{ module}$$

$$D_4 = 4 \times Z_4 = 4 \times 45 = 180$$

Or, $$D_3 = 4 \times Z_3 = 4 \times 15 = 60$$

So, $$D = \frac{180 + 60}{2} = 120 \text{ mm}$$

SOL 5.44 Option (C) is correct.

In standard full height involute teeth gear mechanism the arc of approach is not be less than the circular pitch, therefore

Maximum length of arc of approach = Circular pitch

...(i)

where Maximum length of the arc of approach

$$= \frac{\text{Max. length of the path of approach}}{\cos\phi}$$

$$= \frac{r\sin\phi}{\cos\phi} = r\tan\phi$$

Circular pitch, $$P_C = \pi m = \frac{2\pi r}{Z} \qquad m = \frac{2r}{Z}$$

Hence, from equation (i), we get

$$r\tan\phi = \frac{2\pi r}{Z}$$

$$Z = \frac{2\pi}{\tan\phi} = \frac{2\pi}{\tan 20^\circ} = 17.25 \simeq 18 \text{ teeth}$$

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SOL 5.45 Option (D) is correct.

(A) **Flange coupling :-** It is used to connect two shaft having perfect coaxial alignment and no misalignment is allowed between them.

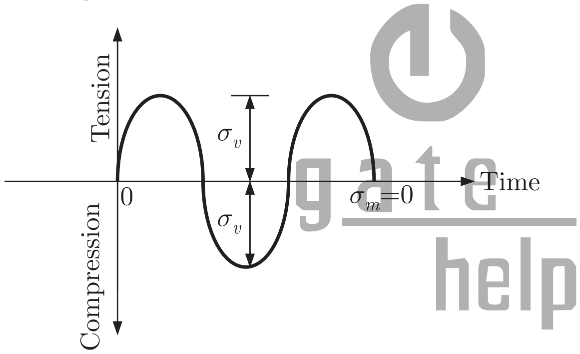
(B) **Oldham's coupling :-** It is used to join two shafts which have lateral misalignment.

(C) **Flexible bush coupling :-** It is used to join the abutting ends of shafts when they are not in exact alignment.

(D) **Hook's joint :-** It is used to connect two shafts with large angular misalignment.

SOL 5.46 Option (D) is correct.

When the shaft rotates, the bending stress at the upper fibre varies from maximum compressive to maximum tensile while the bending stress at the lower fibres varies from maximum tensile to maximum compressive. The specimen subjected to a completely reversed stress cycle. This is shown in the figure.



When shaft is subjected to repeated stress, then it will be designed for fatigue loading.

SOL 5.47 Option (B) is correct.

For a worm gear the velocity ratio ranges between 10 : 1 to 100 : 1. So, Large speed reductions (greater than 20) in one stage of a gear train are possible through worm gearing.

SOL 5.48 Option (A) is correct.

For Helical spring, deflection is given by,

$$\delta = \frac{64PR^3n}{Gd^4} = \frac{8PD^3n}{Gd^4}$$

where, P = Compressive load

d = Wire diameter

R = Coil diameter

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$$G = \text{Modulus of rigidity}$$

From the given conditions

$$\delta \propto \frac{1}{d^4}$$

Given $d_1 = 1\,\text{cm}$ and $d_2 = 2\,\text{cm}$

$$\frac{\delta_2}{\delta_1} = \left(\frac{d_1}{d_2}\right)^4$$

$$\frac{\delta_2}{\delta_1} = \left(\frac{1}{2}\right)^4 = \frac{1}{16}$$

$$\delta_2 = \frac{\delta_1}{16}$$

So, deflection will decrease by a factor of 16.

SOL 5.49 Option (C) is correct.

Bars AB and BC have negligible mass. The support load P acting at the free end of bars AB and BC. Due to this load P, In bar AB compressive stress and in bar BC tensile stress are induced.

However, none of these bars will be subjected to bending because there is no couple acting on the bars.

SOL 5.50 Option (C) is correct.

Let L_1 & L_2 are lengths of the springs and n_1 & n_2 are the number of coils in both the springs.

Given : $$W_1 = W_2$$

$$m_1 g = m_2 g$$

$$\rho\nu_1 g = \rho\nu_2 g \qquad m = \rho\nu$$

$$A_1 \times L_1 \times \rho g = A_2 \times L_2 \times \rho g$$

$$\frac{\pi}{4} d_1^2 \times \pi D_1 n_1 = \frac{\pi}{4} d_2^2 \times \pi D_2 n_2 \qquad L = \pi D n$$

$$d_1^2 \times n_1 = d_2^2 \times n_2 \qquad D_1 = D_2$$

Given : $$d_1 = d \;\&\; d_2 = \frac{d}{2}$$

$$d^2 \times n_1 = \frac{d^2}{4} \times n_2$$

or, $$n_1 = \frac{n_2}{4}$$

The deflection of helical spring is given by,

$$\delta = \frac{8PD^3 n}{Gd^4}$$

Spring stiffness, $$k = \frac{P}{\delta} = \frac{Gd^4}{8D^3 n}$$

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From the given conditions, we get

$$k \propto \frac{d^4}{n}$$

So,

$$\frac{k_1}{k_2} = \left(\frac{d_1}{d_2}\right)^4 \times \left(\frac{n_2}{n_1}\right)$$

$$\frac{k_1}{k_2} = \left(\frac{d}{d/2}\right)^4 \times \frac{n_2}{n_2/4}$$

$$\frac{k_1}{k_2} = 16 \times 4 = 64$$

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