2019

RANK IMPROVEMENT WORKBOOK

Mechanical Engineering

Machine Design

Answer Key of Objective & Conventional Questions

MADE EASY Publications

1 Design for Static Loading

LEVEL 1 Objective Questions

1.	(a)
2.	(a)
3.	(a)
4.	(d)
5.	(b)
6.	(b)
7.	(b)
8.	(d)
9.	(b)
10.	(c)
11.	(a)

LEVEL 2 Objective Questions

12.	(a)
13.	(a)

■■■■

LEVEL 3 Conventional Questions

Solution : 14

$$\sigma_t = 33.87 \text{ N/mm}^2 \text{ (compressive)}$$
$$\sigma_b = 52.26 \text{ N/mm}^2 \text{ (tensile)}$$

Solution : 15

$$d = 41.55 \text{ mm}$$

Solution : 16

$$d = 17.315 \text{ mm}$$
$$\text{Diameter of shaft} = 20 \text{ mm}$$

Solution : 17

$$t = 11.458 \text{ mm}$$
$$\text{thickness, } t = 11.5 \text{ mm}$$

■■■■

2 Design Against Fluctuating Loading

LEVEL 1 Objective Questions

1.	(a)
2.	(a)
3.	(c)
4.	(d)
5.	(b)
6.	(d)
7.	(b)
8.	(a)
9.	(b)
10.	(c)

LEVEL 2 Objective Questions

11	(a)
12	(b)
13.	(b)
14.	(a)
15.	(b)
16.	(c)
17.	(d)
18.	(a)
19.	(2.7)
20.	(b)
21.	(5)
22.	(c)

■■■■

LEVEL 3 Conventional Questions

Solution : 23

$$d = 30.83 \text{ mm}$$

Solution : 24

from equation (i):

$$\sigma_i = 193.68 \text{ N/mm}^2$$
$$\sigma_i < 200 \text{ MPa}$$

Hence, design is safe.

Solution : 25

$$D_i = \frac{D_0}{2} = 11 \text{ mm}$$

Solution : 26

Stress concentration is the localized stress considerably higher than average, even in uniformly loaded cross-section of uniform thickness due to abrupt change in the geometry or localised loading. Stress concentration factor is not considered harmful for ductile materials in static loading because of the phenomenon of local yielding in ductile materials when relieves the stress concentration. When the stress in the vicinity of the discontinuity reaches the yield point there is a plastic deformation, resulting in re-distribution of stresses. This plastic deformation prevents the harmful effects of stress concentration in ductile materials.

While in Brittle materials, stress concentration factor is important in both static and dynamic loading. Brittle materials fail due to fracture. So there is little deformation to relax the concentrated stresses and thus has damaging effects.

■■■■

3 Theory of Failure & Spring

LEVEL 1 Objective Questions

1.	(d)
2.	(c)
3.	(d)
4.	(b)
5.	(d)
6.	(b)
7.	(c)
8.	(b)
9.	(c)
10.	(b)

LEVEL 2 Objective Questions

11.	(c)
12.	(a)
13.	(b)
14.	(d)
15.	(b)
16.	(c)
17.	2118 (2115 to 2121)
18.	(c)
19.	(c)
20.	(a)
21.	(a)

■■■■

LEVEL 3 Conventional Questions

Solution : 22

$$\tau_A = 20 \text{ N/mm}^2$$
$$\tau_B = 26.3 \text{ N/mm}^2.$$

Solution : 23

(i) If axial force P is applied on the spring work done by axial force P is converted into strain energy and stored in the spring.

strain energy U = work done by P

$$= \text{(Average torque)} \times \text{(angular displacement)} = \frac{M_t}{2} \times \theta$$

But $\theta = \dfrac{M_t \times l}{JG}$ (from torsion equation)

Where M_t = torque acting on spring wire = $\left(\dfrac{PD}{2}\right)$

l = length of the spring wire = $(\pi D n)$

J = polar moment of inertia of the wire = $\left(\dfrac{\pi d^4}{32}\right)$

G = Shearing modulus

Substituting these values in equation we get

Strain energy $U = \dfrac{4P^2D^3N}{Gd^4}$

Now according to Castigliano's theorem, the displacement corresponding to force P is obtained by partially differentiating strain energy w.r.t. that force.

$$\delta = \frac{\partial U}{\partial P} = \frac{\partial}{\partial P}\left(\frac{4P^2D^3N}{Gd^4}\right) = \frac{8PD^3N}{Gd^4}$$

Where δ = axial deflection of the spring (mm)

The stiffness of the spring (k) is defined as the force required to produce unit deflection. Therefore,

$$K = \frac{P}{\delta} = \frac{Gd^4}{8D^3 n} = \frac{Gd^4}{8D^3 n}$$

$$n = 14$$

Length of spring wire = $\pi \, n \, D = \pi \times 14 \times 30.3 = 1.33$ m

Solution : 24

$$D_B = 60 \text{ mm}$$

Solution : 25

$$N = 3.25 \text{ or } 4 \text{ coils}$$
$$k = 162.74$$

Solution : 26

$$d = 17.5863 \text{ mm} \simeq 18 \text{ mm}$$
$$D = Cd = 6 \times 18 = 108 \text{ mm}$$
$$n = 10 \text{ turns}$$
$$\delta_{max} = 85.8032 \text{ mm}$$
$$L_F = 315 \text{ mm}$$
$$p = 28.6067 \text{ mm}$$

Solution: 27

$$T = 2.872 \text{ kNm}$$

Solution : 28

$$T = 2.0955 \text{ kN.m}$$

Solution : 29

(i) $\text{fos} = 1.67$

(ii) $\text{fos} = 1.04$

(iii) $\text{fos} = \dfrac{100}{79.69} = 1.27$

(iv) $\text{fos} = \dfrac{100}{84} = 1.19$

■■■■

4 Welded joint, Riveted joint and Bolted Joints

LEVEL 1 Objective Questions

1.	(c)
2.	(b)
3.	(c)
4.	(d)
5.	(a)
6.	(d)
7.	(b)
8.	(b)
9.	(a)
10.	(a)
11.	(a)
12.	(c)
13.	(d)
14.	(c)
15.	(b)
16.	(d)

LEVEL 2 Objective Questions

17.	(d)
18.	(b)
19.	(b)
20.	(d)
21.	(d)
22.	(a)
23.	(a)
24.	(b)
25.	(42.43)
26.	(a)
27	(c)
28.	(c)
29.	3.22(3.15 to 3.30)
30.	0.75 (0.743 to 0.758)

■■■■

LEVEL 3 Conventional Questions

Solution : 31

$$n = 5$$

Solution : 32

But maximum shear stress occurs at the throat of the weld which is inclined at 45° to the horizontal plane

$$\therefore \quad \tau_{max} = \frac{2.83T}{\pi s d^2}$$

Solution : 33

$$d = 21.140 \text{ mm}$$

$$d = \text{diameter of rivet} = 22 \text{ mm}$$

Solution : 34

Efficiency of the joint = 0.625 or 62.5 %

Solution : 35

(i)

$$s = 7.21 \text{ mm or } 8 \text{ mm}$$

(ii)

$$s = 9.14 \text{ mm or } 10 \text{ mm}$$

Solution : 36

$$\sigma_{ta} = 100 \text{ MPa}$$

$$\therefore \quad \tau_a = 76.4 \text{ MPa}$$

$$\therefore \quad \sigma_{ca} = \frac{75000}{750} = 100 \text{ N/mm}^2 = 100 \text{ MPa}$$

Solution : 37

$$l = 163.675 \text{ mm}$$

Design length of weld = 163.675 + 12.5 = 176.175 mm

Solution : 38

$$\therefore \quad d = 52 \text{ mm}$$

Size of bolt = M52

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5 Clutches and Brakes

LEVEL 1 Objective Questions

1.	(b)
2	(a)
3..	(c)
4.	(b)
5.	(d)
6.	(a)
7.	(a)
8.	(c)
9.	(d)
10.	(a)
11.	0.127(0.122 to 0.130)
12.	(c)

LEVEL 2 Objective Questions

13.	(d)
14.	(d)
15.	(1738.13)
16.	(c)
17.	(a)
18.	(39.5)
19.	(1173.54)
20.	(a)
21.	(d)
22.	(d)
23.	(b)

■■■■

LEVEL 3 Conventional Questions

Solution : 24

(i) $E_v = 15.2783 \times 10^3$ J

(ii) $E_w = 498.8861$ J

Initial angular velocity of engine is,

$$\omega_{E1} = 3 \times \omega_1 = 3 \times 47.62 = 142.86 \text{ rad/sec}$$

(iii) $E_E = 7.1431 \times 10^3$ J

(iv) $E_T = 22.9202 \times 10^3$ J

$E = 11.4601 \times 10^3$ J

Solution : 25

$$a_t = 0.1971 \text{ m/s}^2$$

Final angular acceleration, $\alpha_f = 2.288$ rad/s^2

$$\Delta T = 0.2321 \text{ sec.}$$

Solution : 26

$$n = 3.616 \simeq 4 \text{ (take)}$$

Number of active turns of the coil, $n = 4$.

Solution : 27

$$\theta = 0.667 \text{ rad}$$

$$M_t = 22686 \text{ N-m}$$

Solution : 28

$$\theta = 27.49 \text{ rad}$$

$$M_t = 3453.86 \text{ N-m}$$

Solution : 29

$$\theta = 1.3533 \text{ radians or } \theta = 1.3533\left(\frac{180}{\pi}\right) = 77.54°$$

The angular dimension of of pad can be taken as 80°.

Solution : 30

(i) Mass of each shoe $m = 6.234$ kg

(ii) Size of the shoe

$$l = 170.16 \text{ mm}$$

$$b = 60 \text{ mm}$$

Solution : 31

(i) Face width

$$b = 54.7 \text{ say } 55 \text{ mm}$$

(ii) $W_n = 8640$ N

$W_e = 2292$ N

Solution : 32

$$H_g = 106.59 \text{ kJ}$$

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6 Gears

LEVEL 1 Objective Questions

1. (b)
2. (c)
3. (c)
4. (c)
5. (c)
6. (a)
7. (a)
8. (b)
9. (a)

LEVEL 2 Objective Questions

10. (c)
11. (b)
12. (4)
13. (a)
14. (b)
15. (144.22)
16. (c)

■■■■

LEVEL 3 Conventional Questions

Solution : 17

module = 8 mm
dia. of pinion = 200 mm
dia. of gear = 480 mm
Face width, b = 10 m = 80 mm

Solution : 18

Given,

$$\phi = 20^\circ$$
$$\text{Power} = 20 \text{ kW} = 20 \times 10^3 \text{ W}$$
$$N_P = 300 \text{ rpm}$$
$$\text{Velocity ratio} = \frac{T_G}{T_P} = 3,$$
$$\sigma_{OG} = 100 \text{ MPa} = 100 \times 10^6 \text{ N/m}^2 = 100 \text{ N/mm}^2$$
$$\sigma_{OP} = 120 \text{ MPa} = 120 \times 10^6 \text{ N/m}^2 = 120 \text{ N/mm}^2$$
$$T_P = 15,$$

and
$$T_G = 3T_P = 3 \times 15 = 45$$
$$b = 14 \text{ m}$$
$$\sigma_{es} = 600 \text{ MPa} = 600 \text{ N/mm}^2$$
$$E_P = 200 \text{ GPa} = 200 \times 10^9 \text{ N/m}^2 = 200 \times 10^3 \text{ N/mm}^2$$
$$E_G = 100 \text{ GPa} = 100 \times 10^9 \text{ N/m}^2 = 100 \times 10^3 \text{ N/mm}^2$$

Module:

Let, m = module in mm and

D_P = pitch circle diameter of the pinion in mm

Pitch line velocity,
$$V = \frac{\pi D_P N_P}{60} = \frac{\pi m T_P N_P}{60} \qquad \left(\because m = \frac{D}{T}\right)$$
$$= \frac{\pi \times m \times 15 \times 300}{60} = (235.61\ m) \text{ mm/s} = (0.235\ m) \text{ m/s}$$

Assuming steady load conditions 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,

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

and velocity factor,
$$C_V = \frac{3}{3+V} = \frac{3}{3+0.235\,m}$$

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,
$$y_G = 0.154 - \frac{0.912}{T_G} = 0.154 - \frac{0.912}{45} = 0.133$$

$\therefore$
$$\sigma_{OP} \times y_P = 120 \times 0.0932 = 11.184$$
$$\sigma_{OG} \times y_G = 100 \times 0.133 = 13.3$$

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,

$$W_T = (\sigma_{WP}\, b\pi m\, y_P) = (\sigma_{OP} \times C_V)\, b\pi\, m y_P \qquad (\because\ \sigma_{WP} = \sigma_{OP} \times C_V)$$

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

$$\frac{85.1\times 10^3}{m} = \frac{1475.69m^2}{(3+0.235m)}$$

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

$$\Rightarrow \qquad m^3 - 13.55\text{ m} - 172.98 = 0$$

By trial and error method, we get $m = 6.37$ mm

We can consider the standard module, (of second choice), $m = 7$

Face width:

Given that face width $b = 14$ m $= 14 \times 7 = 98$ mm

Pitch diameters of gears:

We know that pitch diameter of the pinion,

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

Pitch diameter of the gear,

$$D_G = mT_G = 7 \times 45 = 315 \text{ mm}$$

Checking the gears for wear:

We know that the ratio factor,
$$Q = \frac{2\times VR}{VR+1} = \frac{2\times 3}{3+1} = 1.5$$

Load stress factor,
$$K = \frac{(\sigma_{es})^2 \sin\phi}{1.4}\left(\frac{1}{E_P}+\frac{1}{E_G}\right)$$
$$= \frac{600^2 \sin 20^\circ}{1.4}\left(\frac{1}{200\times 10^3}+\frac{1}{100\times 10^3}\right)$$
$$= 1.3192 \text{ N/mm}^2$$

We know that the maximum or limiting load for wear,

$$W_W = D_P\, b\, QK$$
$$= 105 \times 98 \times 1.5 \times 1.319 = 20358.765 \text{ N}$$
$$= 20.358 \text{ kN}$$

Tangential load on tooth (or beam strength of the tooth)

$$W_T = \frac{85.1\times 10^3}{m} = \frac{85.1\times 10^3}{7}$$
$$= 12157.14 \text{ N} = 12.157 \text{ kN}$$

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

Solution : 19

Rated Power = 5.29 kW

Solution : 20

New pitch circle diameter, $d_P' = 184.5$ mm

and $d_G' = 461.42$ mm

$\phi' = 23.6°$

$F_t = 2180.4$ N

$F_r = 952.6$ N

$F_N = 2379.4$ N

Solution : 21

Beam strength, $P_b = 6604.88$ N

Wear strength, $P_w = 64589$

Rated power = 18.92 kW

Solution : 22

$m = 8$ mm

$T_P = 15$

$T_G = 150$ mm

■■■■

7 Bearings

LEVEL 1 Objective Questions

1.	(d)
2.	(c)
3.	(b)
4.	(d)
5.	(a)
6.	(b)
7.	(d)
8.	(c)
9.	(c)
10.	(a)
11.	(a)
12.	(d)
13.	(c)
14.	(d)
15.	(b)

LEVEL 2 Objective Questions

16.	(a)
17.	(*)
18.	(10.22)
19.	(a)
20.	(157.46)
21.	(3.55)
22.	(d)
23.	(4.4)
24.	(b)
25.	(21.78)
26.	(64.45)
27.	(c)
28.	(c)
29.	(b)

■■■■

LEVEL 3 Conventional Questions

Solution : 30

$$S = 0.627 \simeq 0.630$$

Power loss in friction, (kW) = 0.1911 kW

$$\Delta T = 23.082°C$$

$$\frac{h_0}{c} = 0.8$$

Minimum film thickness

$$h_0 = 40 \text{ microns}$$

Solution : 31

$$S = 0.2488$$

$$h_o = 0.0110 \text{ mm}$$

Frictional force = 6.088 N

Frictional torque = 83.6 N-mm

Solution : 32

$$C = 36.8 \text{ kN}$$

Solution : 33

$$F_r = P = 5854.16 \text{ N}$$

Solution : 34

$$S = 0.122$$

$$P_{loss} = 40.46 \text{ watt}$$

$$h_0 = 0.02 \text{ mm}$$

Bearing modulus = 0.4883×10^{-6}

Solution : 35

Standard tolerance for hole = 0.046 mm

Standard tolerance for shaft = 0.03 mm

Safe bearing load = 12393 N

Solution : 36

Amount of artificial cooling required

$$= Q_g - Q_d = 71.5 - 28 = 43.5 \text{ W}$$

Let, m = mass of the lubricating oil required in kg/s

$$\therefore \quad Q_g = Q_t$$

$$m = 0.00386 \text{ kg/s} = 0.23 \text{ kg/min}$$

■■■■

8 Shafts

LEVEL 1 Objective Questions

1.	(c)
2.	(d)
3.	(b)
4.	(d)
5.	(c)
6.	(a)
7.	(b)
8.	(c)
9.	(d)
10.	26.89(26 to 27)

LEVEL 2 Objective Questions

11.	(d)
12.	(d)
13.	(a)
14.	(a)
15.	(d)
16.	(1280) (No range)
17.	(d)
18.	(30) (29 to 31)
19.	(d)

■■■■

LEVEL 3 Conventional Questions

Solution : 20

Given

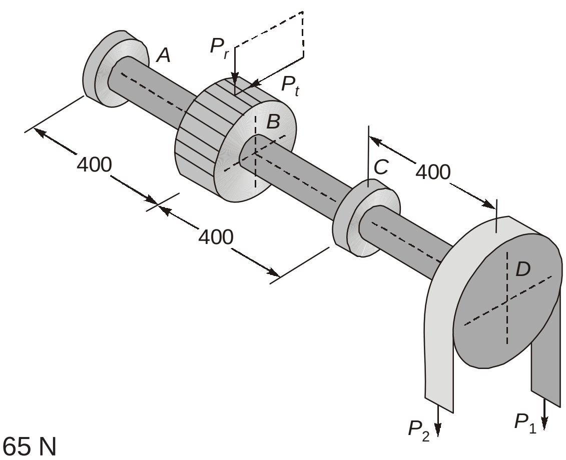
For pulley, $T_1 = 3T_2$

For gear, pressure angle , $\phi = 20°$

$D_2 = 300$ mm

power, $P = 20$ kW

$N = 500$ rpm



Torque transmitted, $T = \dfrac{P}{\omega} = \dfrac{20\times10^3}{\left(\dfrac{2\pi\times500}{60}\right)}$

$= \dfrac{20\times10^3}{52.36} = 381.97$ N-m

$= 381971$ N-mm

The same torque applies for pulley

i.e. $(T_1 - T_2)225 = 381971 = T_1 - T_2 = 1697.65$ N

$\Rightarrow \quad 3T_2 - T_2 = 1697.65 \qquad (\because\ T_1 = 3T_2)$

$\Rightarrow \quad T_2 = 848.83$N

$T_1 = 2546.48$ N

$= T_1 + T_2 = 3395.31$ N

For the gear, torque $T = F_t \times 150$, $\quad F_t \rightarrow$ Tangential force (horizontal)

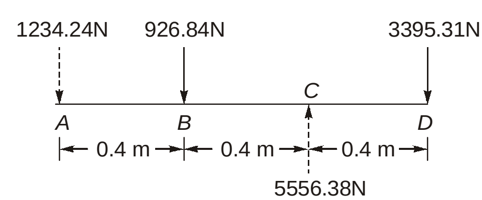
$381971 = F_t \times 150$

$\Rightarrow \quad F_t = 2546.48$ N

$F_r = F_t \times \tan 20°$, $\quad F_t \rightarrow$ Radial force (vertical)

$= 2546.48 \times \tan 20° = 926.84$ N

Invertical plane:



$R_{AV} + R_{CV} = 926.84 + 3395.31$

$= 4322.15$ N

$\Sigma M_A = 0$

$R_{CV} \times 0.8 = 926.84 \times 0.4 + 3395.31 \times 1.2$

$R_{CV} = 5556.38$ N (upwards)

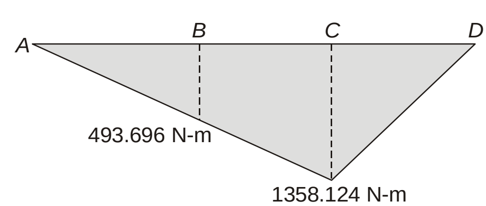
$\Rightarrow \quad R_{AV} = 4322.15 - 5556.38$

$= 1234.24$ N (downwards) i.e. -1234.24 N

$BM_B = -1234.24 \times 0.4 = -493.696$ N-m

$BM_C = 3395.31 \times 0.4$

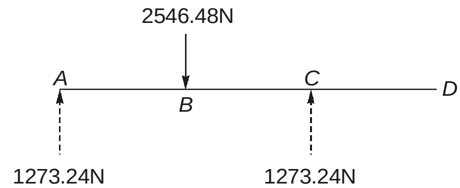
$= 1358.124$ N-m



In horizontal plane:

$R_{AH} + R_{CH} = 2546.48$, $\qquad R_{DH} = 0$

$$R_{AH} = R_{CH} = \frac{2546.48}{2} = 1273.24 \text{ N}$$

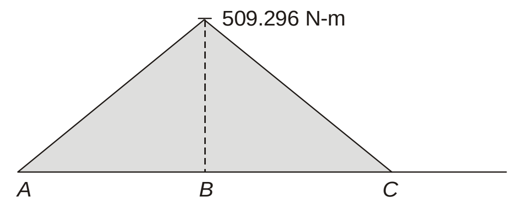


$$BM_B = 1273.24 \times 0.4$$
$$= 509.296 \text{ N-m}$$

The maximum *B.M* is at point *C*,

i.e. $M_C = 1358.124$ N-m

and maximum torque, $T = 381.971$ N-m



Equivalent torque, $T_e = \sqrt{(k_b M)^2 + (k_c T)^2}$

$$= \sqrt{(1.5 \times 1358.124)^2 + (1.5 \times 381.971)^2}$$

$T_e = 2116.224$ N-m [$\because$ $k_b = k_t = 1.5$ (given)]

mostly shafts are designed on the basis of maximum shear stress theory, given tensile strength of the shaft and key is 700 MPa.

$$= 700 \times 10^6 \text{ N/m}^2$$

Allowance for the key way for stress as 0.75 and factor of safety = 5

$$\therefore \quad \tau = \frac{0.75 \times 700 \times 10^6}{5} = 52.5 \times 10^6$$

$$\therefore \quad 52.5 \times 10^6 = \frac{16 T_e}{\pi d^3} = \frac{16 \times 2116.224}{\pi \times d^3} \qquad (\because \tau = 16\, T/\pi d^3)$$

$$d = 0.05899 \text{ m}$$
$$= 58.99 \text{ mm} \qquad \text{(diameter of the shaft)}$$
$$\simeq 60 \text{ mm}$$

maximum shear stress for the key

$$\tau = \frac{\sigma}{2 \times F.S} \times 0.75 = \frac{700}{2 \times 5} \times 0.75 = 52.5 \text{ N/mm}^2$$

For the key, take width w = thickness, $t = \dfrac{d}{4}$

$$w = t = \frac{60}{4} = 15 \text{ mm}$$

$\therefore$ Torque $T = \tau \times \dfrac{d}{2} lw$ (for crushing)

$$= 381971 \text{ N-m} = 52.5 \times \frac{60}{2} \times l \times 15$$

$\Rightarrow$ $l = 16.16$ mm

Solution : 21

Section B-B is the most critical section among B, E and C.

$$\text{FOS} = 2.25$$

Solution : 22

According to maximum strain energy theory,

$$d = 90.8 \text{ mm}$$

Solution : 23

$$d = 6.45 \times 10^{-2} \text{ m or } 64.5 \text{ mm}$$

The next preferred size is d = 80 mm

Outer diameter = 80 mm

Inner diameter = 60 mm

shear stress, τ = 58.9 MPa

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