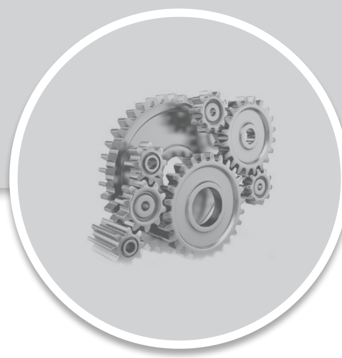
MECHANICAL ENGINEERING

Machine Design



Comprehensive Theory
with Solved Examples *and* Practice Questions

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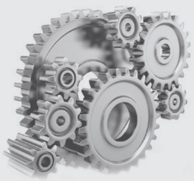

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CHAPTER

1

Design Against Fluctuating Load

1.1 INTRODUCTION

The mechanism by which a requirement is converted into meaningful and functional plan is called a design. The design is an innovative, iterative and decision making process. This book deals with analysis and design of machine elements.

Purpose and scope of design

- It presents a body of knowledge that will be useful is components design for performance, strength and durability.
- It provides treatment of design to meet strength requirements of members and other aspects of design involving prediction of displacement, buckling of a component.

1.1.1 Design against fluctuating load

In many engineering application a machine component is subjected to a fluctuating load due to which it fails at stress levels subsequently lower than the yield strength of material.

The phenomenon of progressive fracture due to repeated loading is called as fatigue and it is observed that fatigue failure begins with a crack at some point inside the component. These cracks are not visible till they reach the surface of component which results in sudden and total failure of component.

1.2 TYPES OF STRESSES

1.2.1 Fluctuating Stress

The stresses which vary from a minimum value to a maximum value of the same nature (i.e. tensile or compressive) are called fluctuating stresses.

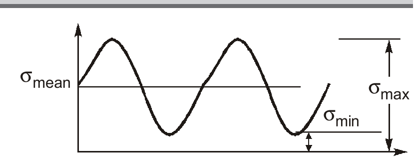


Figure: Fluctuating stress cycle

1.2.2 Repeated Stress

Stress variation is such that the minimum stress is zero, mean and amplitude stress have the same value for repeated loading.

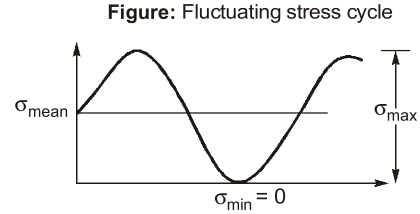


Figure: Repeated stress cycle

1.2.3 Cyclic Stress

The stresses which vary from one value of compressive to the same value of tensile or vice versa, are known as completely reversed or cyclic stresses.

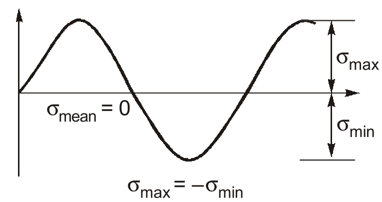


Figure: Completely reversed or cyclic stress

1.2.4 Alternating Stress

The stresses which vary from a minimum value to a maximum value of the opposite nature (i.e. from a certain minimum compressive to a certain maximum tensile or from a minimum tensile to a maximum compressive) are called alternating stresses.

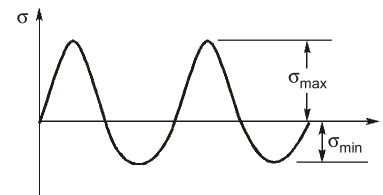


Figure: Alternating Stress

1.2.5 Fluctuating Stress Cycle

$$\text{Mean stress,} \quad \sigma_{mean} \text{ or } \sigma_{avg} = \frac{\sigma_{max} + \sigma_{min}}{2}$$

$$\text{Stress amplitude, } \sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

$$\text{Range of stress, } \sigma_R = \sigma_{max} - \sigma_{min}$$

$$\text{Stress ratio} = \frac{\sigma_{min}}{\sigma_{max}}$$

$$\text{Amplitude ratio} = \frac{\sigma_a}{\sigma_{mean}}$$

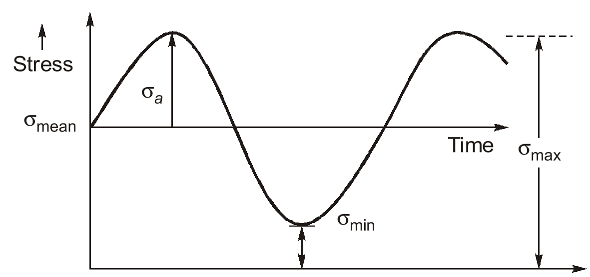


Figure: Fluctuating Stress Cycle

1.3 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. In order to define stress concentration let as consider an example of rectangular cross section plate of thickness *t* with a circular hole of diameter *d* at centre subjected to tensile stress'

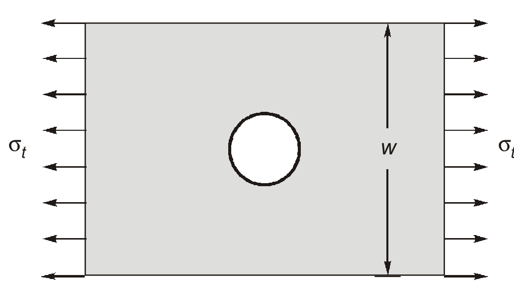


Figure: Plate with circular hole

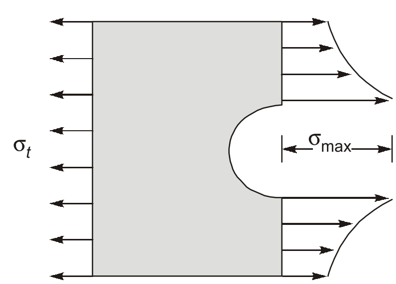


Figure: Stress distribution around circular hole

By using photoelastic technique it is observed that the distribution of stresses near the hole are greater than the nominal stress (σ_o), where nominal stress,

$$\sigma_o = \frac{\sigma_t wt}{(w-d)t} = \frac{\sigma_t w}{(w-d)}$$

The main causes of stress concentration are:

(a) Variation in properties of material due to impurities, cavities etc.

(b) Abrupt changes in section in order to mount gears, sprokets etc.

(c) Discontinuities in component due to certain machine features such as oil holes, keyways etc.

So in order to consider the effect of stress concentration, a factor called stress concentration factor is used.

1.3.1 Stress Concentration Factor

Geometric stress concentration factors can be used to estimate the stress amplification in the vicinity of a geometric discontinuity.

- k_t (theoretical stress concentration factor) is used to relate the maximum stress at the discontinuity to the nominal stress.
- k_{ts} is used for shear stresses
- k_t is based on the geometry of the discontinuity

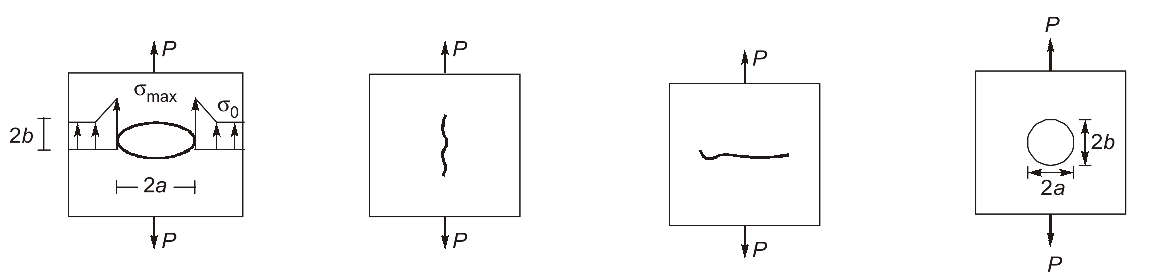
$$k_t = \frac{\text{Highest value of actual stress near discontinuity}}{\text{Nominal stress obtained by elementary equations for minimum cross-section}}$$

or

$$k_t = \frac{\sigma_{max.}}{\sigma_0} = \frac{\tau_{max.}}{\tau_0}$$

NOTE : Stress concentration in a machine component of ductile materials not so harmful as it is in brittle material because in ductile material local yielding may distribute stress concentration.

1.3.2 Stress concentration factor for different geometry



$$k_t = \frac{\sigma_{max}}{\sigma_0} = \left(1 + \frac{2a}{b}\right)$$

(a) Elliptical Hole

$a = 0$

$$k_t = \frac{\sigma_{max}}{\sigma_0} = 1$$

$$\sigma_{max} = \sigma_0$$

(b) Crack parallel to load

$b = 0$

$$k_t = \frac{\sigma_{max}}{\sigma_0} = \infty$$

$$\sigma_{max} = \infty$$

(c) Crack perpendicular to load

$a = b$

$$k_t = \frac{\sigma_{max}}{\sigma_0} = 1 + \frac{2a}{b} = 3$$

(d) Circular Hole

Figure: Stress concentration factor for different geometry

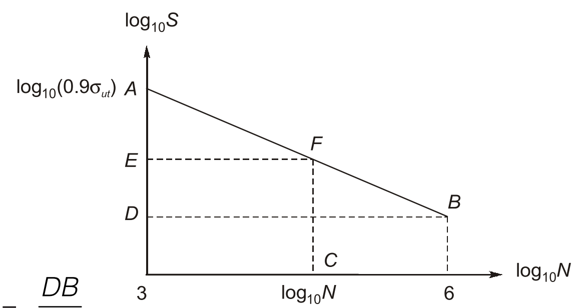
1.3.3 Methods to find theoretical stress concentration factor (k_t)

Experimental Method	Mathematical Method
• Photo-elasticity method • Brittle-coating method • Strain gauge	• Finite element method

EXAMPLE : 1.3 A cylindrical shaft is subjected to an alternating stress of 100 MPa. Fatigue strength to sustain 100 cycles is 490 MPa. If the corrected endurance strength is 70 MPa, then the estimated shaft life will be _____ cycles.

Solution:

It is a finite life problem. The line AB is the failure line, where A [3, $\log_{10}(0.9\sigma_{ut})$] But here it will be A[3, $\log_{10}(490)$] and B {6, $\log_{10}(70)$} Therefore, F {$\log_{10}$ N, $\log_{10}(100)$} we have to find N



$$\frac{EF}{AE} = \frac{DB}{AD}$$

or
$$\frac{\log_{10} N - 3}{\log_{10} 490 - \log_{10} 100} = \frac{6-3}{\log_{10} 490 - \log_{10} 70}$$

or
$$N = 281914 \text{ cycles.}$$

OBJECTIVE BRAIN TEASERS

Q.1 Equation of Goodman line is given by

(a) $\frac{\sigma_m}{s_{yt}} + \frac{\sigma_a}{s_e} = 1$

(b) $\frac{s_{yt}}{\sigma_m} + \frac{\sigma_a}{s_e} = 1$

(c) $\frac{\sigma_m}{s_{ut}} + \frac{\sigma_a}{s_e} = 1$

(d) $\frac{\sigma_m}{s_{ut}} + \frac{s_e}{\sigma_a} = 1$

Q.2 Ratio of increase of actual stress over nominal stress to increase of theoretical stress over nominal stress is called

(a) Endurance limit
(b) Fatigue strength
(c) Mean fluctuating stress
(d) Notch sensitivity

Q.3 Stress concentration factors are used for components made of brittle material subjected to

(a) Static load (b) Fluctuating load
(c) Both (a) & (b) (d) None of these

Q.4 Theoretical stress concentration factor at the edge of hole is given by

(a) $1+\frac{a}{b}$

(b) $1+\frac{b}{a}$

(c) $1+2\left(\frac{b}{a}\right)$

(d) $1+2\left(\frac{a}{b}\right)$

where, a = Semi-axis of ellipse perpendicular to direction of load,
b = Semi-axis of ellipse indirection of load.

Q.5 Stress concentration is due to

(a) Irregularities present in the component
(b) Abrupt change of concentration
(c) Both (a) and (b)
(d) None of these

Q.6 Reduction of stress concentration can be achieved by

(a) Additional notches in member under tension
(b) Addition holes in member under tension
(c) Both (a) and (b)
(d) None of these

Q.7 A link is under a pull which lies on one of the faces as shown in the figure below. The magnitude of maximum compressive stress in the link would be

$$\frac{1}{FOS} = \frac{\sigma_a \times k_f}{\sigma_e} + \frac{\sigma_m}{\sigma_{yt}}$$

(k_t is not required for ductile material)

$$k_f = 1 + q(k_t - 1)$$
$$= 1 + 0.4(2.5 - 1) = 1.6$$
$$\frac{1}{FOS} = \frac{1.6 \times 50}{150} + \frac{150}{250}$$
$$\Rightarrow \quad FOS = 0.8823$$

(Hence material will fail after certain no. of cycles)

12. (a)

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

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

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

Principal stresses, σ_1, σ_2

$$\sigma_{1,2} = \frac{1}{2}\left[(\sigma_x + \sigma_y) \pm \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}\right]$$
$$\sigma_{1,2} = \frac{1}{2}\left[150 \pm \sqrt{50^2 + 4 \times 20^2}\right] = \frac{1}{2}[150 \pm 64.03]$$
$$\sigma_{1,2} = 107.015 \text{ MPa}, 42.984 \text{ MPa}$$

From maximum shear stress theory,

$$\text{Max}\,[(\sigma_1 - \sigma_2), (\sigma_1), (\sigma_2)] \le \frac{S_{yt}}{N_1}$$
$$\Rightarrow \quad 107.015 \le \frac{400}{N_1}$$
$$N_1 \le 3.737$$

From maximum distortion energy theory,

$$\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2 \le \left(\frac{S_{yt}}{N_2}\right)^2$$
$$\Rightarrow \quad 8699.9 \le \left(\frac{400}{N_2}\right)^2$$
$$\Rightarrow \quad N_2 = 4.288$$

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

13. (c)

When a material is fully sensitive to notches

$$\Rightarrow \quad q = 1$$

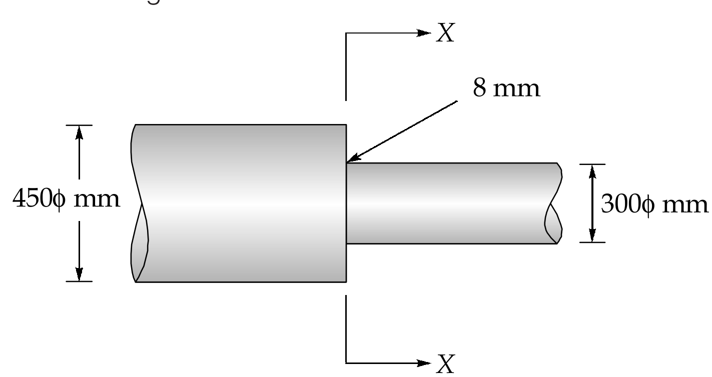
So, $\quad k_f = 1 + q\,(k_t - 1) = 1 + (k_t - 1)$

$$k_f = k_t$$

CONVENTIONAL BRAIN TEASERS

Q.1 The section of a steel shaft is shown in figure. The shaft is machined by a turning process. The shaft is machined by a turning process. The section at *XX* is subjected to a constant bending moment of 500 kN-m. The shaft material has ultimate tensile strength of 500 MN/m^2, yield point 350 MPa and endurance limit in bending is 210 MN/m^2. The notch sensitivity factor can be taken as 0.8. The values of surface finish factor, size factor and reliability factor are 0.79, 0.75 and 0.897 respectively. The theoretical stress concentration factor may be interpolated from following tabulated values:

$\frac{r_f}{d}$	0.025	0.05	0.1
k_t	2.6	2.05	1.66



where r_f is the fillet radius and 'd' is the shaft diameter. Determine the life of the shaft.

CHAPTER

2

Bolted, Welded and Riveted Joints

2.1 INTRODUCTION

Fastener is an important member since machine is a huge structure, so for joining two plates together joining elements are needed. Because machine can't be made as a single component. It is made up by different components joined together. If joining element is weak it will give some effect on machine assembly which reduces the performance of machine.

There are generally two types of joints:

- **Non permanent joint :** If we disassemble that there is no effect on mating component.
- **Permanent joint :** If we disassemble that there is some negative effect on the mating component (strength may be reduced).

Permanent Joint	Non-permanent Joint
• Welded joint • Riveted joint • Brazing, soldering • Adhesive bolding	• Bolted joint • Screwed joint • Cotter joint • Keys, coupling

2.2 BOLTED JOINT

- Threaded fastener designed to pass through holes in mating members and to be secured by tightening a nut from the end opposite to the head of the bolt.
- ANSI standard bolts and nuts of equal grades are designed to have the bolt fail before the threads in the nut are stripped.

2.2.1 Eccentric loading

There are many applications in which bolted joint is subjected to eccentric loading. For example, wall bracket, pillar crane, etc.

Following are the different cases of eccentric loading:

- Eccentric load in the plane containing the bolts.
- Eccentric load perpendicular to the axis of bolts.
- Eccentric load parallel to the axis of bolts.

2.2.1.1 Eccentric Load in Plane Containing the Bolts

- When an eccentric force is acting in the plane of bolts, it will produce two effects.
  - Direct shear stress
  - Shear due to moment set up at C.G.
- In this case, the line of action of the force does not pass through the C.G. of the bolt system.
- For the design of bolt subjected to eccentric load in the plane of bolt, following procedure is used:

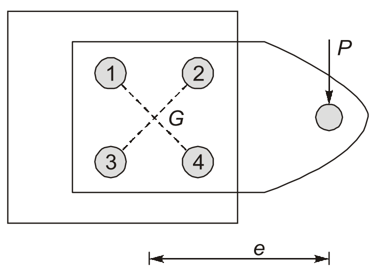


Figure: Eccentric load on a bolted joint

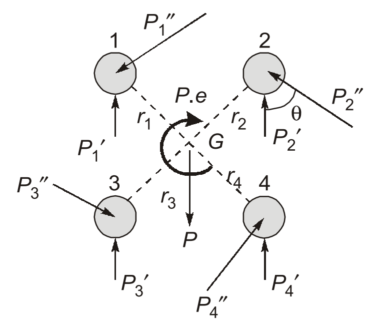


Figure: Direct and torsional shear load on bolts

Direct shear load:

$$P_1' + P_2' + P_3' + P_4' = P$$

and

$$P_1' = P_2' = P_3' = P_4' = P/4$$

also

$$\frac{T}{J} = \frac{\tau}{R} = \frac{G\theta}{L}$$

$$\therefore \quad \tau \propto \frac{TR}{J}$$

$$\tau \propto R$$

$$P \propto R$$

$$\therefore \quad P_1'' \times r_1 + P_2'' \times r_2 + P_3'' \times r_3 + P_4'' \times r_4 = P.e$$

Torsional shear load:

$P_1'' = kr_1$, $P_2'' = kr_2$, $P_3'' = kr_3$ and $P_4'' = kr_4$

$$\therefore \quad kr_1^2 + kr_2^2 + kr_3^2 + kr_4^2 = P.e$$

$$\therefore \quad k = \frac{Pe}{r_1^2 + r_2^2 + r_3^2 + r_4^2}$$

From resultant force / geometry 2nd and 4th bolts are more loaded.

$$\therefore \quad P_{2\,\text{res}} = \sqrt{P_2'^2 + P_2''^2 + 2P_2' P_2'' \cos\theta} = \tau_{max} \times \frac{\pi}{4} d_c^2$$

By maximum shear stress theory, $\tau_{max} = \dfrac{\tau_{yt}}{\text{FOS}} = \dfrac{\sigma_{yt}}{2\text{FOS}}$

$$P_1' = P_2' = P_3' = P_4' = \frac{P_V}{4} = \frac{10.392}{4} = 2.598\,\text{kN}$$

Bending stress load

$$= P_V \times 120 = \frac{P_1''(2\times 250^2 + 2\times 50^2)}{250}$$

$$P_1'' = P_2'' = 2.398\text{ kN}$$

Bending stress will be maximum at 1 & 2 bolt

So,

Resultant stress on bolt 1 →

$$P_b = P_1 + P_1'' = 1.5 + 2.398$$

$$P_b = 3.898\text{ kN}$$

So, maximum shear stress,

$$\tau_{max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + (\tau)^2}$$

$$\sigma = \frac{P_b}{A} \quad \text{and } \tau = \frac{P_1'}{A}$$

$$\tau_{max} = \frac{S_{yt}}{2} = \frac{400}{2} = 200\,\text{MPa}$$

$$\tau_{permissible} = \frac{\tau_{max}}{FOS} = \frac{200}{5} = 40\text{ MPa}$$

$$40 = \sqrt{\left(\frac{3.898\times 10^3}{2\times A}\right)^2 + \left(\frac{2.598\times 10^3}{A}\right)^2}$$

$$40 = \frac{3.2478\times 10^3}{A}$$

$$A = 81.195\text{mm}^2$$

$$A = \frac{\pi}{4}\times d^2 = 81.195$$

$$d = 10.168\text{ mm}$$

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CONVENTIONAL BRAIN TEASERS

Q.1 Two lengths of mild steel tie rod having width 200 mm and thickness 12.5 mm are to be connected by means of a butt joint with double cover plates. Find out the diameter of rivet and number of rivets for economical design. Use the following data:

Permissible stress in tension = 80 MPa

Permissible stress in shear = 65 MPa

Permissible stress in crushing = 160 MPa

Resistance of rivet in double shear = 1.75 times resistance of rivet in single shear.

Diameter of rivet hole (mm)	19.5	21.5	23.5	25.5	29	32	35	38
Diameter of rivet (mm)	18	20	22	24	27	30	33	36

Solution:

Diameter of rivet hole, $d = 6\sqrt{t} = 6\sqrt{12.5} = 21.2$ mm

As per table, for the rivet hole of 21.5 mm size, diameter of rivet is 20 mm.

∴ Diameter of rivet, $d = 20$ mm

Let, n = number of rivets

P_t = Maximum pull on the joint i.e. tearing resistance of plate at the outer row which has only one rivet.

$P_t = (b - d)\,t \times \sigma_t = (200 - 21.5) \times 12.5 \times 80$

$= 178{,}500$ N

CHAPTER

3

Shaft and Key

3.1 SHAFT

It is rotating machine element generally circular in cross-section, which supports transmission element (Gears, pulleys, sprockets) and transmits power.

3.1.1 Specific Categories of Shaft

1. **Axle**
   - Axle is a kind of shaft that supports rotating elements (wheels, hoisting drum, rope sheaves) and this is fitted to the housing by means of bearing.
   - In general, axle is subjected to only bending moment due to transverse loads (like bearing reaction) and does not transmit any useful torque. Example: Rear axle of railway wagon; sometimes axle also transmits torque (automobile rear axle).
   - An axle may rotate with the wheel or simply support the wheel.
2. **Spindle:** A spindle is a short rotating shaft. It is used in all machine tools such as small drive shaft of a lathe or the spindle of a drilling machine.
3. **Counter Shaft:** It is a secondary shaft, which is driven by the main shaft and from which the power is supplied to the machine component.
4. **Jack-Shaft:** It is an auxiliary or intermediate shaft between two shafts that are used in transmission of power.
5. **Line-Shaft:** It consist of number of shaft, which are connected in axial direction by means of coupling. A number of pulleys is mounted on line shaft and power is transmitted to individual machine by different belt.

3.1.2 Shaft Material and Method of Production

Ordinary transmission shafts are made of medium carbon steels with a carbon content from 0.15 to 0.40 percent [30C8, 40C8].

- Commercial shafts are made of low carbon steels. They are produced by hot-rolling and finished to size either by cold-drawing or by turning and grinding.
- Cold drawing produces a stronger shaft than hot-rolling but tolerances on their diameters and straightness are not very close compared with shaft finished by turning and grinding process cold drawing also result in residual stress at the surface of the shaft.

Assumption

Force '*F*' act tangential to shaft diameters

For each key

$$\therefore \quad \frac{T}{2} = F \times \frac{d}{2} \Rightarrow F = \frac{T}{d}$$

Design Based on Failure Due to Shear Stress

$$\tau = \frac{F}{\text{Area(AC)}} = \frac{F}{\sqrt{2}\, b\, l} = \frac{T}{\sqrt{2}\, d\, b\, l}$$

Design Based on Failure Due to Crushing Stress

$$\sigma_c = \frac{F}{\text{Project of area (OB)}} = \frac{F}{\frac{b}{\sqrt{2}} \times L} = \frac{\sqrt{2}\, F}{b.l} = \frac{\sqrt{2}\, T}{d.b.l}$$

Here, l = Length of key

■■■■

OBJECTIVE BRAIN TEASERS

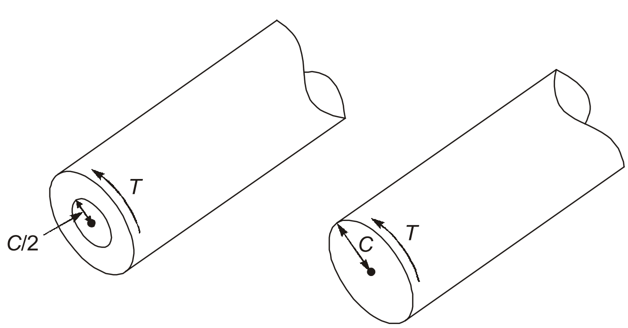
Q.1 Consider the following statements with respect to flexible shafts:

1. They have high rigidity in bending and low rigidity in tension.
2. The applications of flexible shafts are restricted to low power drives.

Which of the above statement(s) is/are correct?

(a) 1 only (b) 2 only
(c) Both 1 and 2 (d) Neither 1 nor 2

Q.2 A shaft is subjected to a torque *T*. Compairing the effectiveness of using the tube shown in the figure below with that of solid section of radius C, compute the percent increase in maximum allowable shear stress for the tube versus the solid section.



Q.3 A cold rolled steel shaft is designed on the basis of maximum shear stress theory. The principal stresses induced at its critical section are 60 MPa and –60 MPa respectively. If the yield stress for the shaft material is 360 MPa, the factor of safety of the design is______.

Q.4 A shaft is subjected to maximum bending moment of 80 N/mm^2 and a maximum shear stress equal to 30 N/mm^2 at a particular section. If the yield point in tension of the material is 200 N/mm^2 and maximum shear stress theory is used, the factor of safety will be

(a) 1.5 (b) 2.0
(c) 3.5 (d) 4.0

Q.5 A static load is mounted at the center of a shaft rotating at uniform angular velocity. This shaft will be designed for

(a) the maximum compressive stress (static)
(b) the maximum tensile stress (static)
(c) the maximum bending moment (static)
(d) fatigue loading

Q.6 In the assembly design of shaft, pulley and key the weakest member is

(a) Pulley (b) Key
(c) Shaft (d) None

Q.20 A hollow transmission shaft having inner diameter to outer diameter ratio 0.6 transmits 40 kW power at 720 rpm. For this shaft, the permissible angle of twist is 3° per meter length. The modulus of rigidity for the shaft material is 79300 MPa. What is the diameter of the shaft?

(a) 43.97 mm (b) 24.97 mm
(c) 54.97 mm (d) 34.98 mm

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ANSWER KEY

1. (b)	2. (6.67) (6.60 to 6.70)			3. (3)
4. (b)	5. (d)	6. (b)	7. (d)	8. (c)
9. (a)	10. (c)	11. (a)	12. (c)	13. (d)
14. (c)	15. (d)	16. (c)	17. (b)	
18. (80)(79 to 81)	19. (d)	20. (d)		

HINTS & EXPLANATIONS

1. (b)

Flexible shafts have low rigidity in bending, making them flexible and highly rigid in torsion making them capable of transmitting torque.

2. (6.67) (6.60 to 6.70)

Maximum allowable shear stress for tube (t) and solid shaft (s) is

$$(\tau_t)_{max} = \frac{Tc}{J_t}\ ;\ (\tau_s)_{max} = \frac{Tc}{J_s}$$

Percentage increase in shear stress,

$$\frac{(\tau_s)_{max} - (\tau_t)_{max}}{(\tau_t)_{max}} \times 100 = \frac{J_s - J_t}{J_t} \times 100$$

$$= \frac{\frac{\pi}{2}c^4 - \left[\frac{\pi}{2}\left\{c^4 - \left(\frac{c}{2}\right)^4\right\}\right]}{\left[\frac{\pi}{2}\left\{c^4 - \left(\frac{c}{2}\right)^4\right\}\right]} \times 100 = \frac{1 \times 16}{16 \times 15} \times 100$$

$= 6.67\%$

3. (3)

$$\tau_{max} = \frac{\sigma_1 - \sigma_2}{2} = \frac{60 - (-60)}{2} - 60 \text{ MPa}$$

According to maximum shear stress theory yield stress in shear $= \sigma_{se} = 0.5 \times \sigma_{yt}$

$= 0.5 \times 360 = 180$

$$\therefore \quad \tau_{all} = \frac{\sigma_{se}}{Fos}$$

$$\therefore \quad FOS = \frac{\sigma_{se}}{\tau_{allowable}} = \frac{180}{60} = 3$$

4. (b)

$$\tau_{max} = \sqrt{\left(\frac{80-0}{2}\right)^2 + 30^2} = 50 \text{ N/mm}^2$$

$$\therefore \quad t = \frac{\sigma_y}{2} = \frac{200}{2} = 100 \text{ N/mm}^2$$

$$FOS = \frac{10}{50} = 2$$

5. (d)

Fatigue loading is primarily the type of loading which causes cyclic variations in the applied stress or strain on a component. Thus, any variable loading is basically a fatigue loading Example: Rotating shaft with a static point load at the centre.

6. (b)

Key is the weakest member in the assembly of shaft, pulley and key. Key acts as a safety device. Whenever excess load appears on the pulley, key fails first and it keeps shaft and pulley safer.

7. (d)

Saddle key is a key that fits in the keyway of the hub only. In this case there is no keyway provided on the shaft and friction between shaft, key and hub prevents relative motion between the shaft and the hub and power is transmitted by means of friction only.

10. (c)

$$\text{Maximum shear stress} = \frac{16T}{\pi d^3} = 240 \text{ MPa} = \tau$$

$$\frac{T}{J} = \frac{\tau}{R}$$

Let D is the diameter for both the shaft,

For thin circular shaft,

$$J = \frac{\pi D^3 t}{4}$$

$$T_c = \frac{\pi D^3 t}{4 \times D} \times 2 \times \tau$$

$$T_c = \frac{\pi D^2 t}{2}\tau$$

For solid shaft, $T_s = \frac{\pi}{16}\tau D^3$

Given $T_c = T_s$ and τ is same for both shaft as material is same.

$$\frac{\pi D^2 t}{2}\tau = \frac{\pi}{16}\tau \times D^3$$

$$t = \frac{D}{8}$$

20. (d)

We know that, $\frac{T}{J} = \frac{G\theta}{L}$ ... (i)

Torque, $T = \frac{P \times 60}{2\pi N} = \frac{40 \times 10^3 \times 60}{2\pi \times 720}$

$= 530.516$ N-m

$$J = \frac{\pi D^4}{32}\left(1 - k^4\right)$$

$$\frac{\theta}{L} = \frac{3 \times \pi}{180} = 0.0523 \text{ radian/m}$$

Equation (i),

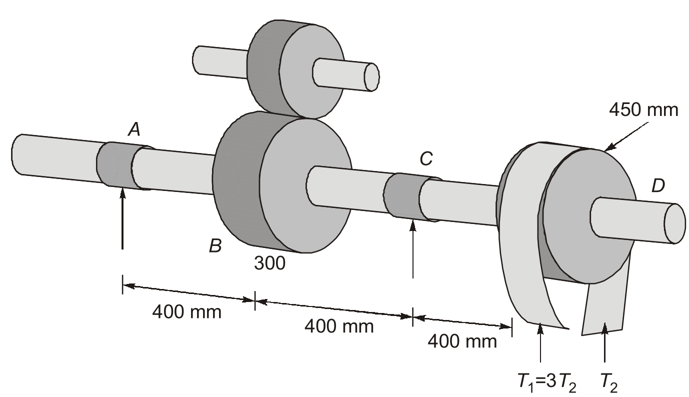
$$\frac{530.516 \times 32}{\pi \times D^4\left(1 - k^4\right)} = 79300 \times 10^6 \times 0.0523$$

$$D = 0.034978 \text{ m} = 34.978 \text{ mm}$$

CONVENTIONAL BRAIN TEASERS

Q.1 A shaft *ABCD* is supported at end *A* and at point *C* and other end *B* consists of pulley which has belt tension in vertical direction, $AD = 1200$ mm, $AC = 800$ mm. A spur gear having pressure angle 20° is mounted at point *B* which is at a distance 400 mm from *A*. D_{pulley} = 450 mm and that of gear is 300 mm. The gear supplied power = 20 kW at 500 rpm by another gear at the top of it. The tension on tight side is 3 times, the tension on slack side, combined shock and fatigue factor for bending and torsion is 1.5, $(\sigma_{yt})_{shaft}$ = 700 MPa design the shaft dia using FOS = 2.

Solution:



$$\tau_{max} = \frac{16}{\pi d^3}\left[\sqrt{(k_b M)^2 + (k_T T)^2}\right]$$