UNIT 7 SHAFTS

Structure

7.1 INTRODUCTION

Shafts form the important elements of machines. They are the elements that support rotating parts like gears and pulleys and in turn are themselves supported by bearings resting in the rigid machine housings. The shafts perform the function of transmitting power from one rotating member to another supported by it or connected to it. Thus, they are subjected to torque due to power transmission and bending moment due to reactions on the members that are supported by them. Shafts are to be distinguished from axles which also support rotating members but do not transmit power. Axles are thus subjected to only bending loads and not to the torque.

Shafts are always made to have circular cross-section and could be either solid or hollow. The shafts are classified as straight, cranked, flexible or articulated. Straight shafts are commonest to be used for power transmission. Such shafts are commonly designed as stepped cylindrical bars, that is, they have various diameters along their length, although constant diameter shafts would be easy to produce. The stepped shafts correspond to the magnitude of stress which varies along the length. Moreover, the uniform diameter shafts are not compatible with assembly, disassembly and maintenance. Such shafts would complicate the fastening of the parts fitted to them, particularly the bearings, which have to be restricted against sliding in axial direction. While determining the form of a stepped shaft it is borne in mind that the diameter of each cross-section should be such that each part fitted on to the shaft has convenient access to its seat.

The parts carried by axle or shaft are fastened to them by means of keys or splines and for this purpose the shaft and axle are provided with key ways or splines. The bearings that support the shafts or axle may be of sliding contact or rolling contact type. In the former case the journal of the shaft rotates freely on thin lubricant layer between itself and bearing, while in the latter case the inner race of the bearing is force fitted on the journal of the shaft and rotates with the shaft while outer race is supported in the housing and remains stationary.

A shaft is joined with another in different ways and configurations. The coaxial shafts are connected through couplings which may be rigid or flexible.

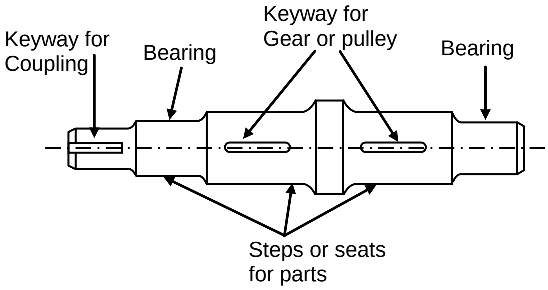
Objectives

After studying this unit, you should be able to

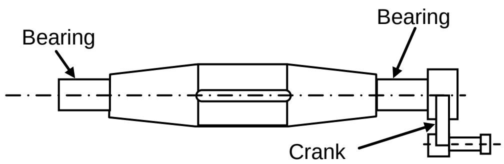
- describe types of shafts,
- take decisions to select the materials for shaft,
- estimate shaft diameters in different segments along length, and
- design couplings for shafts.

7.2 TYPES OF SHAFT

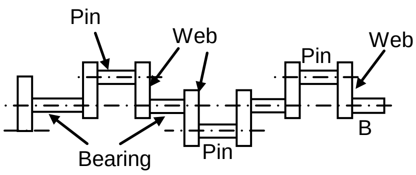
The types of shaft are mentioned in introduction. Figure 7.1(a) shows a stepped shaft with three seats for supported parts which can be pulleys, gears or coupling. Two seats for bearings are also indicated. These bearings will be rolling contact type. Figure 7.1(b) shows a single crank shaft. The crank may be connected to another element like connecting rod which may have a combined rotary and reciprocating motion. The connection is through a bearing often called crank pin. The straight part of the shaft may support a pulley or a gear. The connection will be through a key. Multiple crank shaft is shown in Figure 7.1(c). Each crank pin would carry a connecting rod and each crank pin will be between the supporting bearings. The other shaft types are explanatory.



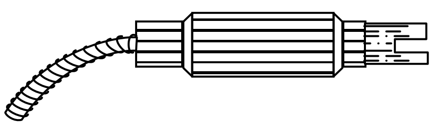
(a) A Stepped Shaft



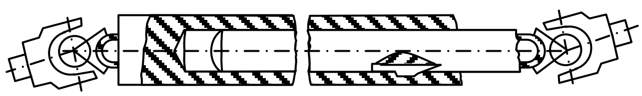
(b) A Single Crank Shaft



(c) Multiple Crank Shaft



(d) Flexible Shaft



(e) Articulated Shaft

Figure 7.1 : Different Types of Shafts

The adjacent sections of shafts with different diameters are joined by smooth transition fillet with as large radius as permitted by supported part or bearing that supports the shaft. The larger radius of fillets will reduce stress concentration factor.

7.3 MATERIALS FOR SHAFTS

From the above discussion the materials for the shaft would be required to possess

(a) high strength,

(b) low notch sensitivity,

(c) ability to be heat treated and case hardened to increase wear resistance of journals, and

(d) good machinability.

Shafts could be made in mild steel, carbon steels or alloy steels such as nickel, nickel-chromium or chrome-vanadium steels.

Commercial shaftings (available in stock) are generally made in low carbon steel by hot rolling. Such shaftings could be finished to size by cold drawing are machining (turning and grinding). Cold drawing produces stronger shaft but generally introduces residual stresses which may result in distortion of the shaft when subjected to unsymmetrical machining like cutting a keyway. Table 7.1 describes shafting available commercially.

Table 7.1 : Standard Sizes of Commercial Shafting (Diameter)

Upto 25 mm in increment of 0.5 mm
25 to 50 mm in increment of 1.0 mm
50 to 100 mm in increment of 2.0 mm
100 to 200 mm in increment of 5.0 mm

Carbon steel is frequently used as a shafting material and this material can be subjected to heat treatment which can result into ultimate strength of about 800 MPa with yield strength exceeding 550 MPa. Such steels can be tempered and hardened to a hardness of 40 to 50 RC to get a good wear resistance in the journal. Most common material is medium carbon steel with C between 0.27% and 0.57%.

Heavily loaded shafts are often made in alloy steels which because of their high strength would result in smaller diameters. These steels are amenable to heat treatment and especially high wear resistance in journal is obtainable by case hardening treatment. However, designer has to be careful while choosing such steels because they will be highly notch sensitive. These steels also are costly. Further, the smaller diameters of shafts may not always be advantageous, because a strong enough shaft may not have sufficient rigidity. Due to these reasons, the high strength alloy steels have limited scope of utility. Carbon steels are largely replacing alloy steels because of development of methods of heat treatment and case hardening. Special purpose and large diameter shafts are often forged. Diameters larger than 125 mm are regarded as large in this respect. The forged shafts are subsequently machined to size.

Ductile cast iron is also finding use as a shaft material because of its low notch sensitivity and damping capacity.

Steel castings are also used as shaft material and their strength is comparable to mild steel.

7.4 SHAFT STRENGTH UNDER TORSIONAL LOAD

The shafts are always subjected to fatigue load hence they must be calculated for fatigue strength under combined bending and torsion loading. However, the initial estimate of diameter is obtained from the torque that is transmitted by the shaft. The bending moment variation along the length of the shaft is established after fixing some structural features like distance between supporting bearings and distance between points of application of forces and bearings.

Following notations will be used for shaft.

d = diameter of shaft,

M_t = torque transmitted by the shaft,

H = power transmitted by the shaft (W),

N = rpm of the shaft,

τ_s = permissible shearing stress,

σ_b = permissible bending stress, and

M_b = bending moment.

Considering only transmission of torque by a solid shaft.

The power transmitted by shaft and the torque in the shaft are related as

$$H = M_t \, \omega$$

If H is in Watts and M_t in Nm.ω is angular velocity in rad/s and equals $\frac{2\pi N}{60}$

$$H = \frac{M_t \, 2\pi N}{60}$$

$$\therefore \qquad M_t = \frac{30 H}{\pi N} \text{ Nm} \qquad \ldots (7.1)$$

The shearing stress and the torque are related as

$$\tau = \frac{16 M_t \times 10^3}{\pi d^3} \text{ N/mm}^2$$

If M_t is in Nm and d in mm.

$$\therefore \qquad M_t = \frac{\pi}{16} \times 10^{-3} \, \tau \, d^3 \qquad \ldots (7.2)$$

From Eqs. (7.1) and (7.2)

$$\frac{30 H}{\pi N} = \frac{\pi}{16} 10^{-3} \, \tau \, d^3$$

or

$$d^3 = \frac{16 \times 30}{\pi^2 N} \frac{H}{\tau} \times 10^3$$

$$\therefore \qquad d = 36.5 \left(\frac{H}{\tau N} \right)^{0.33} \text{ mm} \qquad \ldots (7.3)$$

In Eq. (7.3) H is in W, τ in N/mm^2, N in rpm and d in mm.

For calculating shaft diameter, d, we substitute the permissible value of shearing stress in place of τ. Table 7.2 describes permissible values for steel shaft under various service conditions, when the bending are much smaller then torsional loads.

Table 7.2 : Allowable Shear Stress for Shafts

Service Condition	τ_s (MPa)
Heavily loaded short shafts carrying no axial load	48-106
Multiple bearing long shafts carrying no axial load	13-22
Axially loaded shafts (bevel gear drive or helical gear drive)	8-10
Shafts working under heavy overloads (stone crushers, etc.)	4.5-5.3

Manufacturers, sometimes making shaft routinely, like to use Eq. (7.3) with value for τ substituted. For example for a heavily loaded short shaft, the first in Table 7.2, Eq. (7.3) will yield

$$d = 36.5 \times \frac{1}{(78)^{\frac{1}{3}}} \left(\frac{H}{N}\right)^{0.33}$$

or $$d = 8.543 \left(\frac{H}{N}\right)^{0.33}$$

For shaft working under heavy overload, the last of Table 7.2, using τ = 4.9

$$d = 36.5 \times \frac{1}{(4.9)^{\frac{1}{3}}} \left(\frac{H}{N}\right)^{0.33}$$

or $$d = 21.5 \times \left(\frac{H}{N}\right)^{0.33}$$

Suppose the manufacturer wants to find diameter of the shaft of machine which is short and likely to be overloaded with nominal power of 15 kW at 300 rpm.

$$d = 21.5 \times \left(\frac{15 \times 10^3}{300}\right)^{0.33} = 79.2 \text{ mm}$$

Thus, with handy formula the engineer can calculate the diameter in no time. Yet the detailed calculations may have to be done for carefully designed shafts. We will consider such design procedure now.

7.5 STRESSES IN BENDING AND TORSION

The shafts are circular section cylindrical parts that are rotating and supported in bearings. Most shafts are subjected to bending moment and torque simultaneously. The bending moment at different sections has to be calculated and a bending moment diagram is drawn to locate section where bending moment is highest. The torque at this section is also calculated. We will see in solved example how the bending moment diagram is plotted along with torque. In this section let us assume that at any section the BM is M and torque is M_t. Then stresses due to M and M_t can be calculated by normal bending and torsion theories, at any point on the surface of the shaft. Figure 7.2 shows the stress distribution over the cross-section and state of stress at a point on the surface at a radius $\frac{d}{2}$. Apparently both bending stress σ and shearing stress τ (respectively due to M

and M_t) have highest magnitudes σ_1 and τ_1 at surface or point A.

$$\sigma_1 = \frac{32M}{\pi d^3}$$

$$\tau_1 = \frac{16M_t}{\pi d^3}$$

The maximum principal stress for state of stress at point A shown in Figure 7.2 is written as

$$\sigma_{p1} = \frac{\sigma_1}{2} + \sqrt{\left(\frac{\sigma_1}{2}\right)^2 + \tau_1^2}$$

$$= \frac{16M}{\pi d^3} + \sqrt{\left(\frac{16M}{\pi d^3}\right)^2 + \left(\frac{16M_t}{\pi d^3}\right)^2}$$

$$= \frac{16}{\pi d^3} + [M + \sqrt{M^2 + M_t^2}] \qquad \text{. . . (i)}$$

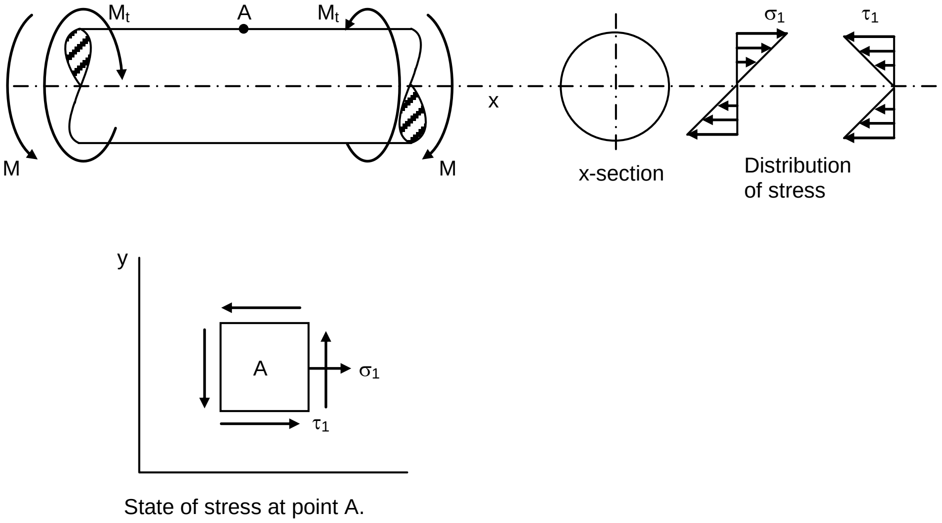


Figure 7.2 : A Shaft under Bending and Torsion

If we assume that a bending moment M_e, acting alone would induce a bending stress σ_{p1} at point A, then

$$\sigma_{p1} = \frac{32 M_e \sigma_1}{\pi d^2} \qquad \text{. . . (ii)}$$

Then the right hand sides of (i) and (ii) being equal we obtain

$$M_e = \frac{1}{2} [M + \sqrt{M^2 + M_t^2}] \qquad \text{. . . (7.4)}$$

Using equivalent bending moment for designing the shaft is same as using *maximum normal stress theory of failure*. If σ_b denote the permissible bending stress for the steel shaft, then

$$\sigma_b = \frac{32 M_e}{\pi d^3}$$

$$\therefore \qquad d = 2.17 \left(\frac{M_e}{\sigma_b} \right)^{0.33} \qquad \text{. . . (7.5)}$$

For example, we can find diameter of a shaft at a section where M = 40 kNm and M_t = 20 kNm. Take σ_b = 50 N/mm^2.

You should be careful about units. M_e is in Nmm and σ_b in N/mm^2 in Eq. (7.5).

$$M_e = \frac{1}{2} [40 \times 16^6 + 10^6 \sqrt{40^2 + 20^2}] = \frac{10^6 \times 20}{2} [2 + \sqrt{4 + 1}]$$

$$= 10^7 \times 4.236 \text{ Nmm}$$

Using calculated value of M_e and given value of σ_b in Eq. (7.5)

$$d = 2.17 \left(4.236 \times \frac{10^6}{50} \right)^{0.33}$$

or $\qquad d = 94.6$ mm

If we use maximum shearing stress theory then failure will occur when

$$\tau_{max} = \frac{16}{\pi d^3} \sqrt{M^2 + M_t^2}$$

If we assume that a torque M_{te} acting along will cause same shearing stress as τ_{max}, then

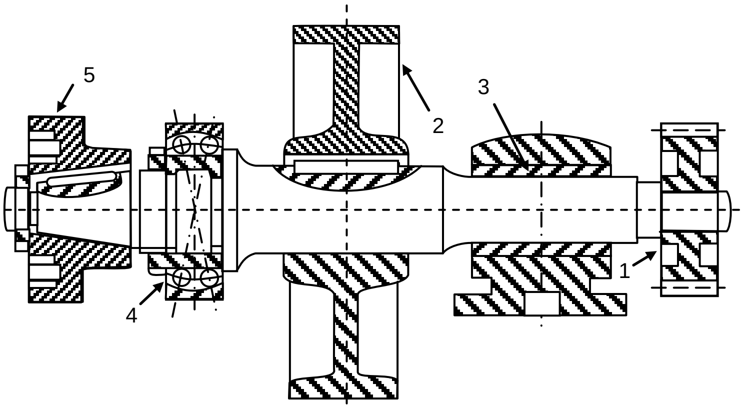
$$M_{te} = \sqrt{M^2 + M_t^2} \qquad \ldots (7.6)$$

M_{te} is called equivalent torque. Both M_e and M_{te} can be used to calculate shaft diameter.

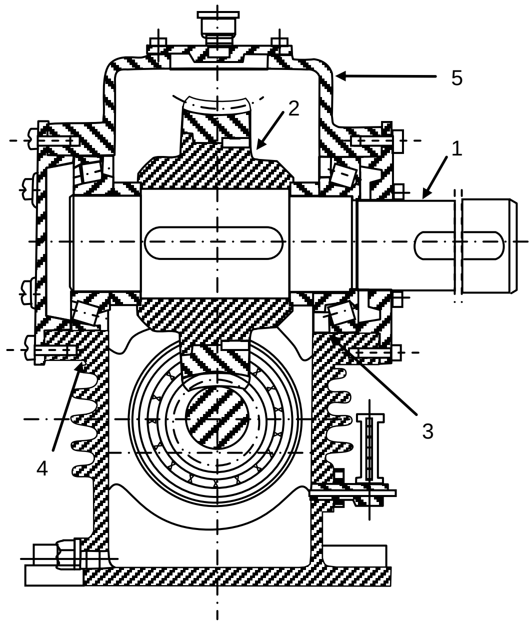
While solving an actual problem the designer will have to find bending moment and torque at various sections of shaft. It may require complete understanding of how the forces are transmitted to shafts from attached parts like gears, pulleys and chain sprockets or coupling. It also needs understanding as to how bearings provide support to the shaft. The shaft can be regarded as simply supported or fixed beam for determining the bending moments. We will consider the shaft loading now.

7.6 SHAFT LOADING

The parts that are supported by shaft have already been mentioned as gear, pulley and coupling. Figure 7.19(a) showed the seats for these parts on shaft. Figure 7.3(a) shows the shaft on which gear, pulley and coupling are supported. It is also shown that the pulley and coupling are connected to shaft through key which sits in the keyway. Figure 7.3(b) shows yet another shaft that supports a worm wheel. Two roller bearings support the shaft and themselves are supported in the casing.



(a) Shaft Carrying 1 Gear, 2 Pulley, 5 Coupling and Supported in 3 Sliding Contact Bearing and in 4 Rolling Contact Bearing



(b) Shaft 1 Supports Worm Wheel, 2 and is Supported in Bearings, 3 and 4. Bearings are Supported in the Housing 5

Figure 7.3

Calculations of forces coming upon the shaft from gear may not be explained in detail. The simplest gear is straight tooth spur gear, i.e. the pair of gears mounted on parallel shaft with teeth parallel to the axes of the shafts. In helical spur gear the teeth are inclined to the parallel axes of the shafts. In case of straight tooth spur gears the contacting teeth are subjected to tangential and radial force components (tangent and radial in respect of pitch circle of gear) denoted by P_t and P_r. P_t will cause the torque and a transverse force on the shaft. P_r will act as a transverse force on the shaft. These forces are shown in line diagram in Figure 7.4. The forces P_r and P_t will act in two planes which are mutually perpendicular. Thus, they will cause bending moment in mutually perpendicular plane. The resultant bending moment can be found by combining bending moments by usual method of finding resultant of two vectors along two mutually perpendicular directions. Alternatively the resultant of P_t and P_r can be found as P_n such that

$$P_t = P_n \cos \alpha \text{ and } P_r = P_n \sin \alpha \text{ so that } P_r = P_t \tan \alpha \qquad \ldots (7.6)$$

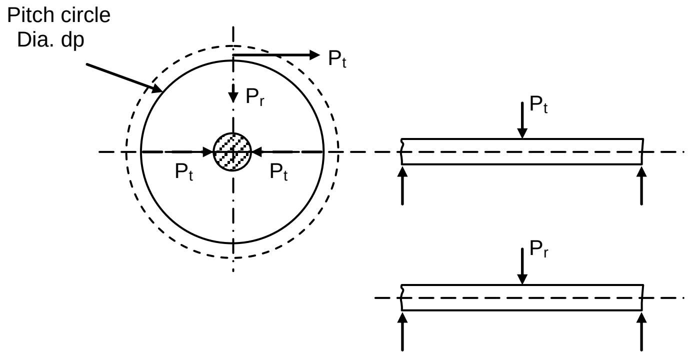


Figure 7.4 : Loading of Shaft by Gear

Here α is the pressure angle of gear teeth (You may like to revise your understanding of pressure angle). For understanding occurrence of shaft torque two fictitious forces each equal to P_t and opposite to each, may be assumed to act on the shaft parallel to P_t at gear pitch circle as shown in Figure 7.4. Then the couple P_t with arm equal to $\frac{1}{2}$ of pitch circle diameter will cause a torque equal to $P_t \frac{d}{2}$. This is the torque transmitted by the shaft. Another P_t at centre of the gear will act transversely to cause bending moment. Perpendicular to the plane of P_t, P_r will act on the shaft in similar manner. The bending moments M_b and M_r will be calculated depending upon the distance between supports. If the supporting bearings are narrow like ball or roller bearing, the supports may be regarded as simple. If bearings are long like sliding contact then supports are regarded as fixed.

As against gear a pulley is pulled by belt tension on two sides. The tensions on tight and slack sides of belt, called T_1 and T_2 are related as $\frac{T_1}{T_2} = e^{\mu\theta}$ where μ is coefficient of friction between pulley surface and belt and θ is angle of contact between belt and pulley (Figure 7.5). This figure is representing the simplest case in which two belt sides are vertical, θ being π. The shaft will be subjected to load $T_1 + T_2$ acting transversely. Additionally the weight of pulley will also act at the same section. If the centre line of T_1 and T_2 is inclined then components in vertical and horizontal planes can be found and bending moment in vertical plane is calculated by combining weight of the pulley with vertical component of the tension ($T_1 + T_2$). BM in horizontal plane is separately calculated and BM in horizontal and vertical planes are combined, thereafter. The torque on shaft is calculated as $(T_1 + T_2)\frac{D}{2}$ where D is the diameter of pulley.

In case of chain only force tangent to sprocket will act. If the force is P, then it will result in torque of $\frac{P\,d\,p}{2}$ and transverse force P on shaft.

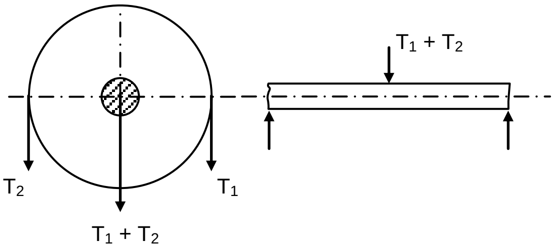


Figure 7.5 : Loading of Shaft by Pulley

The coupling is the connection between coaxial shafts through discs (called flanges as they are integral with hub). These discs are connected to shafts through keys, the bolts connect the discs with each other. The shear force develops in each bolt and of such shear forces exert torque on the shaft. If there are n bolts at the pitch circle of diameter d_p then the torque is $\frac{n\,F\,d\,p}{2}$ where F is shear force in each bolt. Figure 7.5 shows schematic of coupling. No BM is caused by coupling.

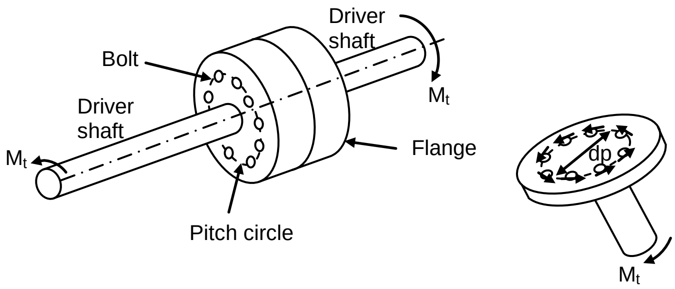


Figure 6.6 : Coupling

With calculation of load understood and supports of bearings known we can proceed to make some calculations through solved examples.

Example 7.1

A shaft is required to transmit a power of 25 kW at 360 rpm. The force analysis due to attached parts results in BM of 830 Nm at a section between bearings. If permissible stresses in the shaft are : 60 N/mm^2 in bending and 40 N/mm^2 in shear calculate the diameter of the shaft.

Solution

M = 830 Nm = 0.83 × 10^6 Nmm

$$\text{Power, } H = 25 \times 10^3 = M_t\,\omega = \frac{2\pi N}{60}\,M_t = \frac{2\pi\;360}{60}\,M_t = 12\pi\,M_t$$

$$\therefore \quad M_t = \frac{25 \times 10^3}{12\pi} = 663.15 \text{ Nm} = 0.663 \times 10^6 \text{ Nmm}$$

$\therefore$ Equivalent BM from Eq. (7.4)

$$M_e = \frac{1}{2}\,[M + \sqrt{M^2 + M_t^2}\,]$$

$$= \frac{1}{2}\,[830 + \sqrt{(830)^2 + (663)^2}\,]$$

$$= \frac{1}{2}\,[830 + 1062.3]$$

$$\text{or} \qquad M_e = 946.15 \text{ Nm} = 0.946 \times 10^6 \text{ Nmm} \qquad \ldots \text{(i)}$$

Equivalent torque from Eq. (7.6)

$$M_{te} = \sqrt{M^2 + M_t^2}$$

$$= 1062.3 \text{ Nm} = 1.062 \times 10^6 \text{ Nmm} \quad \text{. . . (ii)}$$

The equivalent BM will cause bending stress which is not allowed to exceed permissible value.

$$\therefore \quad 60 = \frac{32 M_e}{\pi d^3} = \frac{32 \times 0.946 \times 10^6}{\pi d^3}$$

$$\text{or} \quad d^3 = \frac{9.6 \times 10^6}{60} = 0.161 \times 10^6$$

$$\therefore \quad d = 54.4 \text{ mm} \quad \text{. . . (a)}$$

The equivalent torque will cause shearing stress which is not allowed to exceed permissible value.

$$\therefore \quad 40 = \frac{16 M_e}{\pi d^3} = \frac{16 \times 1.062 \times 10^6}{\pi d^3}$$

$$\text{or} \quad d^3 = \frac{5.41 \times 10^6}{40} = 0.1352 \times 10^6$$

$$\therefore \quad d = 51.33 \text{ mm} \quad \text{. . . (b)}$$

Out of (a) and (b) the larger diameter will be selected.

$\therefore$ Shaft diameter d = 54.4 mm.

Example 7.2

A shaft carries a 1000 N pulley in the centre of two ball bearings which are 2000 mm apart. The pulley is keyed to the shaft and receives 30 kW of power at 150 rpm. The power is transmitted from the shaft through a flexible coupling just outside the right bearing. The belt derive is horizontal and the sum of the belt tension is 8000 N. Calculate the diameter of the shaft if permissible stress in bending is 80 N/mm^2 and in shear it is 45 N/mm^2.

Solution

The belt tensions T_1 and T_2 cause horizontal transverse force while weight of the pulley causes vertical transverse force in the middle of the span as shown in Figure 7.7. The BM diagrams in vertical and horizontal planes and torque diagrams are also shown.

Force in vertical plane = F_V = 1000 N

Force in horizontal plane = F_H = 8000 N

Both F_V and F_H act at the mid span. Maximum BM occurs at mid span, assuming that the bearings behave as simple support

$$M_V = \frac{1000 \times 2000}{4} = 0.5 \times 10^6 \text{ Nmm}$$

$$M_H = \frac{8000 \times 2000}{4} = 4 \times 10^6 \text{ Nmm}$$

$\therefore$ Resultant BM, $M = \sqrt{M_V^2 + M_H^2} = 10^6 \sqrt{0.5^2 + 4^2} = 4.031 \times 10^6 \text{ Nmm}$

The shaft torque $\quad M_t = \dfrac{H}{\omega} = \dfrac{30 \times 10^3}{2\pi \times \dfrac{150}{60}} = 1.91 \times 10^3 = 1.9 \times 10^6 \text{ Nmm}$

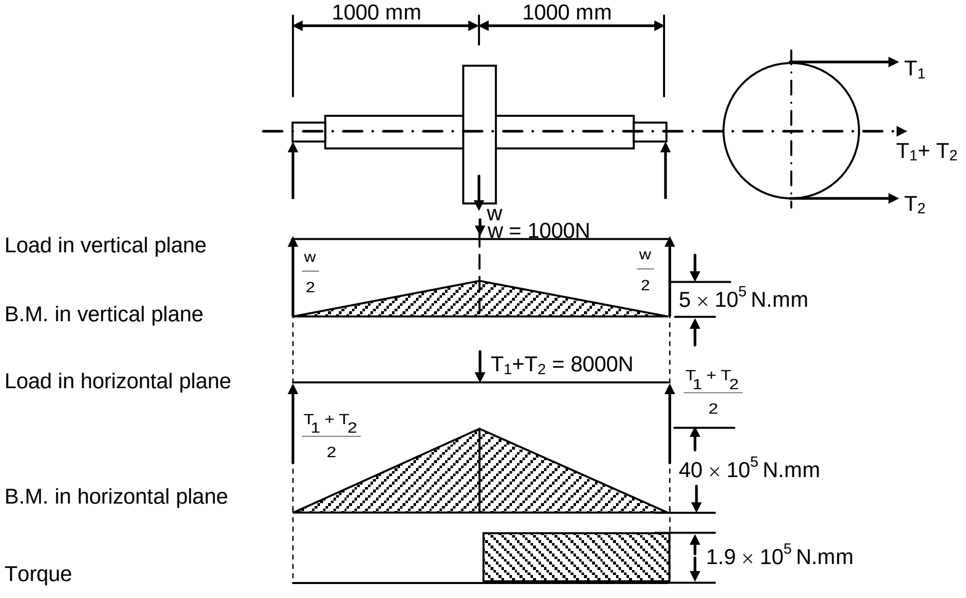


Figure 7.7

$\therefore$ Eq. BM $\quad M_e = \frac{1}{2}[M + \sqrt{M^2 + M_t^2}] = \frac{1}{2}[4.031 + \sqrt{4.031^2 + 1.91^2}] \times 10^6$

$= 4.25 \times 10^6$ Nmm

and Eq. torque $\quad M_{te} = \sqrt{M^2 + M_t^2} = \sqrt{4.031^2 + 1.91^2} \times 10^6$ Nmm

$= 4.46 \times 10^6$ Nmm

$\therefore$ Permissible bending stress $= 80$ N/mm$^2 = \frac{32 M_e}{\pi d^3}$

$$\therefore \qquad d^3 = \frac{32 \times 4.25 \times 10^6}{\pi \times 80} = 0.54 \times 10^6$$

$$\therefore \qquad d = 81.5 \text{ mm} \qquad \text{. . . (a)}$$

Permissible shearing stress $= 45 = \frac{16 M_{te}}{\pi d^3}$

$$\therefore \qquad d^3 = \frac{16 \times 4.46 \times 10^6}{\pi \times 45} = 0.5048 \times 10^6$$

$$d = 79.6 \text{ mm} \qquad \text{. . . (b)}$$

(a) being larger diameter is acceptable, d = 81.5 mm

SAQ 1

(a) Describe different types of shafts. Sketch a stepped shaft to support a gear, a pulley and coupling at one end. The shaft will be supported in ball bearings.

(b) Describe materials for shaft.

(c) What are the loads that come upon shaft?

(d) How will you calculate load upon a shaft if it supports a pulley or when it supports a gear?

(e) Define equivalent bending moment and equivalent torque and state upon which theories of failure they depend.

(f) A machine shaft is supported in ball bearings placed at a distance of 750 mm. The shaft carries a 450 mm diameter pulley at a distance of 200 mm on the right of right hand bearing and a straight tooth spur gear 200 mm pitch circle diameter on the right of left hand bearing at a distance of 250 mm from it. 15 kW of power is supplied at spur gear at 600 rpm and taken off at the pulley on which belt is mounted making an angle of 60^o with the horizontal. The ratio of the belt tensions is 3 : 1 and pulley weighs 800 N. The gear meshes with another gear directly located above the shaft. The permissible bending stress is 100 MPa and permissible shearing stress is 55 MPa. The pressure angle of gear is 20^o. Determine the diameter of the shaft.

7.7 SHAFTS UNDER TORSION AND BENDING

Most frequently shafts are loaded under torque and bending moment simultaneously. In addition, the shafts may be loaded by axial load of tensile or compressive nature. The bending moments on shaft may act in different planes and they have to be solved in two mutually perpendicular planes and their resultant could be obtained by usual method.

If at any section of the shaft a bending moment M_b and a torque M_t are acting, then by maximum principal stress theory, the equivalent bending moment can be expressed as

$$M_{eq} = \frac{1}{2}\left[M_b + \sqrt{M_b^2 + M_t^2}\right] \qquad \ldots (7.7)$$

The diameter of the shaft then can be calculated from well known equation of bending of beam, that is,

$$\sigma_b = \frac{32\, M_{eq}}{\pi\, d^3}$$

$$d = 2.15 \left(\frac{M_{eq}}{\sigma_b}\right)^{0.33} \qquad \ldots (7.8)$$

σ_b here is permissible bending stress which could be taken as fatigue strength divided by factor of safety.

Table 7.3 : Allowable Bending Stress for Steels

Material	**σ_u (MPa)**	**σ_b (MPa)**		
		Constant Load σ_{b1}	**Pulsating Load σ_{b2}**	**Reversible Load σ_{b3}**
Carbon Steel	400	130	70	40
	500	170	75	45
	600	200	95	55
	700	230	110	65
Alloy Steel	800	270	130	75
	1000	330	150	90
Steel Casting	400	100	50	30
	500	120	70	40

The torque transmitted by the shaft remains constant over a long period of time. It varies only when power changes and power changes occurs only occasionally. Thus, the shearing stress on the shaft cross-section changes much less frequently. On the other hand the bending stress on the shaft cross-section changes in each cycle, which means bending stress changes with frequency, which is equal to rpm of the shaft. Eq. (7.4), though expresses bending stress, is function of both bending moment and torque and hence is equally dependent upon both. It will be more logical to make it more dependent upon bending moment and less upon the torque. This is done by multiplying torque by a factor α where $\alpha < 1$. Thus, the equivalent BM will be

$$M_{eq} = \sqrt{M_b^2 + (\alpha M_t)^2} \qquad \text{. . . (7.9)}$$

where $\alpha = \dfrac{\sigma_{b3}}{\sigma_{b2}}$ for pulsating torque and $\alpha = 1$ for reversible torque.

σ_{b1}, σ_{b2} and σ_{b3} are described as permissible stresses under three respective conditions of (i) constant load, (ii) pulsating load, and (iii) reversible load. These permissible stresses for some steels are described in Table 7.3.

Thus, Eq. (7.5) is rewritten as

$$d = 2.15 \left(\frac{M_{eq}}{\sigma_{b3}} \right)^{0.33} \qquad \text{. . . (7.10)}$$

Eq. (7.8) is used to determine shaft diameters at various sections along its length. The method consists in plotting the bending moment diagrams in the plane of the forces, resolving these bending moment diagrams into two mutually perpendicular planes and combining them to calculate resultant bending moment. The torque diagram is similarly plotted and equivalent bending moment is then calculated, which when substituted in Eq. (7.10) would give the shaft diameter, *d*.

The axial thrust, that acts as a compressive force on the shaft cross-section, usually causes stress that is in insignificat in comparison with bending stresses. However, if it is considered, it reduces the tensile stress. If the axial load is tensile in nature, the resultant stress may be taken as

$$\sigma = \sigma_b + \alpha\, \sigma_t \qquad \text{. . . (7.11)}$$

where σ_t is the tensile stress and α, the same factor as defined earlier. Such designs of shafts in which compressive axial force is produced are often preferred.

The diameters of such sections which carry keyed parts are often increased by 8 to 10% in excess of those calculated by Eq. (7.10) to take care of the stress concentration produced and cross-section reduced by keyway. The shaft journal diameters are also calculated in the same way but their lengths are determined in conjunction with the bearing. For rolling contact bearing, the length is selected, depending upon proper bearing width. For sliding contact bearing, generally, the length varies from 0.4 to 1.5 times diameter. In revised calculation factor of safety at each critical cross-section is checked.

7.7.1 ASME Formula

Yet another approach for shaft calculation is based upon maximum shearing stress theory whereby equivalent torque is given by

$$M_{teq} = \sqrt{M_b^2 + M_t^2} \qquad \text{. . . (7.12)}$$

when M_{teq} is used in torsion equation, shaft diameter

$$d = \left(\frac{16 M_{teq}}{\pi \tau_s} \right)^{\frac{1}{3}} \qquad \text{. . . (7.13)}$$

τ_s is permissible shearing stress.

The equivalent torque method is recommended by American Society of Mechanical Engineering for calculation of shaft diameters. They suggest the modification of equivalent torque as

$$M_{teq} = \sqrt{(K_m\ M_b)^2 + (K_t\ M_t)^2} \qquad \ldots (7.13)$$

Recommended values of K_m and K_t are described in Table 7.4 while permissible shearing stress τ_s to be used in Eq. (7.12) is chosen smaller of the following :

$$\tau_s = 0.3\ \sigma_Y \qquad \text{or} \qquad \tau_s = 0.18\ \sigma_u \qquad \ldots (7.14)$$

where σ_Y is the yield strength and σ_u is the ultimate tensile strength.

These values may be further reduced by 25% if keyway is present.

Table 7.4 : Values of K_m and K_t in ASME Formula

Type of Loading	K_m	K_t
Stationary Shaft :		
Load applied gradually	1.0	1.0
Load applied suddenly	1.5-2.0	1.5-2.0
Rotating Shaft :		
Load applied gradually	1.5	1.0
Steady load	1.5	1.0
Load applied suddenly		
Minor shock	1.5-2.0	1.0-1.5
Heavy shock	2.0-3.0	1.5-3.0

The design stress in Eq. (7.12) can be further reduced by 25% if shaft failure would cause serious consequences.

Some times it is preferred to increase the torque by using a dividing factor, K to account for presence of a keyway. If width of the key is w and h is its depth, then

$$K = 1 - 0.2\,\frac{w}{d} - 1.1\,\frac{h}{d} \qquad \ldots (7.15)$$

Example 7.3

A shaft is supported in ball bearings which are placed 200 mm apart. The shaft carries a straight tooth spur gear of 20° pressure angle at a distance of 50 mm from right hand bearing between the supports. 3.9 kW of power is transmitted by the shaft at 90 rpm. The pitch circle diameter of the gear is 125 mm which receives power from a pinion placed in the same vertical plane above the gear and power is taken off from right hand through a coupling. The shaft is to be made in steel (carbon) for which ultimate tensile strength is 700 MPa and permissible bending stresses in pulsating and reversible bending loading respectively are 110 and 65 MPa. These permissible values take care of stress concentration, size and surface finish. Find diameter of the section where gear is fitted on shaft through a key, using both bending and torsional equivalence.

Solution

Before proceeding to calculate diameter shaft loading has to be calculated.

$$\omega = \frac{2\pi N}{60} = \frac{2\pi\ 90}{60} = 9.425 \text{ rad/s}$$

$$H = 3.9 \times 10^3\ W = M_t\,\omega \quad M_t \text{ in Nm}$$

$$\therefore \qquad M_t = \frac{3.9 \times 10^3}{9.425} = 413.8 \text{ Nm} = 0.414 \times 10^6 \text{ Nmm} \qquad \ldots \text{(i)}$$

The torque M_t acts upon the gear at a radius of $\frac{125}{2}$ mm .

If a tangential force P_t acts upon the gear at this radius

$$P_t = \frac{M_t}{\frac{d_p}{2}} = \frac{0.414 \times 10^6}{\frac{125}{2}} = 6.621 \times 10^6 \text{ N}$$

This force will act on shaft transversely in horizontal plane (tangential force on gear) at a distance of 50 mm from right hand bearing, which is regarded as simple support along with left hand bearing. The schematic of the shaft is shown in Figure 7.8. The bending moment due to P_t is calculated below.

Reaction at RH bearing $R_1 = \frac{P_t \times 150}{200} = \frac{6.621 \times 10^3 \times 150}{200} = 4.97 \times 10^3$ N

BM at section where gear sits, $M_1 = 4.97 \times 10^3 \times 50 = 248.3 \times 10^3$ Nmm . . . (ii)

The gear will be subjected to a radial force component which will be transmitted to the shaft as transverse load in vertical plane.

The radial force, $P_r = P_t \tan \alpha = 6.621 \times 10^3 \times \tan 20$

$$= 6.621 \times 10^3 \times 0.364$$

or $P_r = 2.41 \times 10^3$ N . . . (iii)

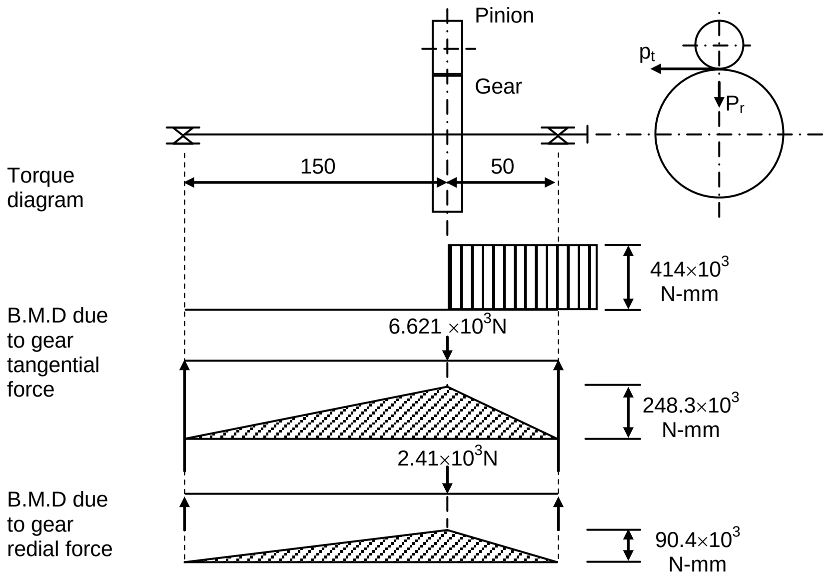


Figure 7.8

BM due to P_r in vertical plane in gear section

$$M_2 = \frac{P_r \times 150 \times 50}{200} = \frac{2.41 \times 10^3 \times 150 \times 50}{200} = 90.4 \times 10^3 \text{ Nmm}$$

The torque BM in horizontal plane and BM in vertical plane are drawn in Figure 7.8.

Hence, resultant BM in shaft at section where gear is mounted

$$M = \sqrt{M_1^2 + M_2^2} = 10^3 \sqrt{249.3^2 + 90.4^2} = 264.24 \times 10^3 \text{ Nmm}$$

$\therefore$ Eq. BM $M_{eq} = \frac{1}{2} [M + \sqrt{M^2 + (\alpha\, M_t)^2}\,]$

$$\alpha = \frac{\text{reversible fat strength}}{\text{pulsating fat strength}} = \frac{65}{110} = 0.59$$

$$M_{eq} = \frac{1}{2}[264.24 + \sqrt{264.24^2 + (0.59 \times 414)^2}] \times 10^3 = 254.7 \times 10^3 \quad \text{. . . (iv)}$$

Using M_{teq} as defined in ASME formula. From Table 7.4 read for rotating shaft and higher value for minor shock

$$K_m = 2.0, \quad K_t = 1.5$$

$$\therefore \qquad M_{teq} = \sqrt{(K_m\ M)^2 + (K_t\ M_t)^2}$$

$$= \sqrt{(2 \times 264.24)^2 + (1.5 \times 414)^2} \times 10^3 = \sqrt{0.28 + 0.386} \times 10^6$$

$$= 816 \times 10^3 \text{ Nmm} \quad \text{. . . (v)}$$

The values of M_{eq} and M_{teq} will be used for calculating diameter. With M_{eq} the permissible stress will be fatigue strength in reversible stress cycle, i.e. 65 N/mm^2 (given)

$$\therefore \qquad 65 = \frac{32\,M_{eq}}{\pi\,d^3}$$

$$\text{or} \qquad d = \left(\frac{32 \times 254.7 \times 10^3 M_{eq}}{\pi\ 65}\right)^{\frac{1}{3}} = 34.17 \text{ mm} \quad \text{. . . (a)}$$

With M_{teq} the permissible stress will be fatigue strength in shear, i.e. 0.18 σ_u

$$\tau_s = 0.19 \times 700 = 126 \text{ N/mm}^2$$

To take care of keyway stress concentration this stress is reduced by 25%.

Hence, $\qquad \tau_s = 0.75 \times 126 = 94.5 \text{ N/mm}^2$

$$\therefore \qquad 94.5 = \frac{16\,M_{eq}}{\pi\,d^3}$$

$$\text{or} \qquad d = \left(\frac{16 \times 0.816 \times 10^6}{\pi \times 94.5}\right)^{\frac{1}{3}} = 35.3 \text{ mm} \quad \text{. . . (b)}$$

Out of two diameters (a) and (b) the higher value will be chosen.

$\therefore \qquad d$ = 35.3 mm $\qquad$ say 35.5 mm

The designed shaft will look like one shown in Figure 7.9.

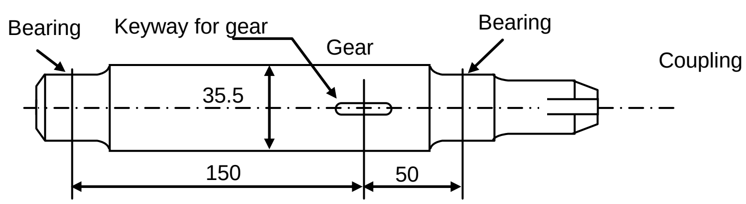


Figure 7.9

7.8 STIFFNESS OF SHAFT

Shafts are often designed for strength as illustrated in theory and solved examples so far. But all shafts have to be stiff and rigid so that their deflection and twist are within permissible limits. If the shaft exceeds in deflection and twist limits the diameter has to be increased. We must remember that the deflection and twists are inversely proportional to cube of the diameter hence, lesser diameter will result in greater deflection and twist. The problem becomes important when high strength steel is used for shaft. Such shaft will result in smaller diameter and hence, larger deflection. Moreover, using high strength steel requires greater care for its greater notch sensitivity.

The permissible values of displacement (in bending and torsion) are decided with respect to the requirements of machine in which shaft is placed, hence, such values vary from machine to machine. For example, permissible deflection of shaft in machine tool may depend upon module of the gear fitted on the shaft while the limit in shaft of the rotor of an electric motor will be in function of air gap. In general, however, the maximum deflection in shaft must not exceed 0.2% of the span between the bearings in case of machines with gears mounted on shafts. The slope due to bending at the bearings must also be limited. Following are the limits for precision machines :

Slope $\leq$ 0.001 rad if bearing sliding contact type.

Slope $\leq$ 0.008 rad if bearing rolling contact type.

Slope $\leq$ 0.050 rad if bearing self aligning type.

The angular twist may become basic design consideration for shaft such as in drilling machine where the twist should not be greater than 0.035 radius over a length of 25 $\times$ diameter. The transmission shaft in a gantry crane is not allowed to twist more than 0.012 rad per meter length.

In general, the deflection of shaft is reduced by

(a) making mounted parts lighter,

(b) keeping mounted parts balanced, and

(c) mounting parts close to bearing.

The angular displacement or twist θ in radians is given by

$$\theta = \frac{32 M_t}{\pi G d^4} l \qquad \text{. . . (7.16)}$$

where M_t is torque acting over length l.

The deflection can be calculated by such simple formula as

$$\delta = \frac{W l^3}{48 EI} = \frac{4 W l^3}{3E \pi d^4} \qquad \text{. . . (7.17)}$$

where W is the central load on shaft of span l and δ is under the load.

G and E in above equations are modulus of rigidity and modulus of elasticity respectively. The support slope of beam is calculated as

$$i = \frac{W l^2}{16 EI} = \frac{4 W l^2}{E \pi d^4} \qquad \text{. . . (7.18)}$$

It is not surprising to note that most shafts in practice may not coincide with conditions of simple supported beams for which Eqs. (7.14) and (7.15) have been written. For one thing their diameter is not uniform along length hence you may have to resort to method of integration or area moment method for calculation of slope and deflection. The equations for area moment method are :

$$i_2 - i_1 = \Sigma \frac{A}{EI} \qquad \text{. . . (7.19)}$$

$$(x_2 i_2 - y_2)(x_1 i_1 - y_1) = \Sigma \frac{A\bar{x}}{EI} \qquad \text{. . . (7.20)}$$

where 1 and 2 refer to two sections along the shaft, *i* and *y* denote the slope and deflection and *x* is the distance of the section from the origin. *A* is the area of bending moment diagram between sections 1 and 2 and $\bar{x}$ is the distance of centre of gravity of *A* from the origin. The calculation of *i* and *y* will depend upon judicious choice of the origin.

Example 7.4

For the shaft of Example 7.2 of Section 7.7 calculate the maximum values of angular twist, deflection and slope. Assume E = 200 GPa and G = 80 GPa.

Solution

This is a simple case in which shaft is loaded like a simply supported beam with l = 200 mm while d was calculated as 81.5 mm. The central load is 8000 N in horizontal and 1000 N in vertical plane.

Use Eq. (7.14) for δ

$$\delta_H = \frac{W l^3}{48 EI}$$

with l = 2000 mm, W = 8000 N, $I = \frac{\pi}{64} d^4$, d = 81.5 mm

$$\therefore \qquad \delta_H = \frac{8000 \times (2)^3 \times 10^9 \times 64}{48 \times 2 \times 10^5 \times \pi\,(81.5)^4} = 3.1 \text{ mm}$$

$$\text{and} \qquad \delta_V = \frac{1000 \times (2)^3 \times 10^9 \times 64}{48 \times 2 \times 10^5 \times \pi\,(81.5)^4} = 0.3875 \text{ mm}$$

Hence, resultant deflection

$$\delta = \sqrt{\delta_H^2 + \delta_V^2} = \sqrt{3.1^2 + 0.3875^2} = 3.124 \text{ mm} \qquad \text{. . . (i)}$$

This deflection is $\left(\frac{\delta}{l} \times 100\right)$ % of span $= \left(\frac{3.124}{2000} \times 100\right) = 0.156\%$ span . . . (ii)

The slope at the bearing, from Eq. (7.15)

$$i_H = \frac{4W l^2}{EI\,\pi d^4} = \frac{4 \times 8000 \times (2)^2 \times 10^6}{2 \times 10^5 \times \pi \times (81.5)^4} = 0.046 \text{ rad}$$

$$i_V = \frac{4 \times 1000 \times (2)^2 \times 10^6}{2 \times 10^5 \times \pi \times (81.5)^4} = 0.0058 \text{ rad}$$

$$\therefore \qquad i = \sqrt{i_H^2 + i_V^2} = \sqrt{0.046^2 + 0.0058^2} = 0.04636 \text{ rad} \qquad \text{. . . (iii)}$$

The torque is constant from pulley to the coupling which is assumed at 10% l from RH bearing. So that the length of shaft to be twisted is

$$1000 + 0.1 \times 1000 = 1100 \text{ mm.}$$

Use l = 1100 mm, M_t = 19.1 × 10^5 Nmm, G = 80 × 103 N/mm^2 in Eq. (7.13)

$$\therefore \qquad \theta = \frac{32 \times 19.1 \times 10^5 \times 1100}{80 \times 10^3 \times (81.5)^4\ \pi} = 0.00605 \text{ rad} \qquad \text{. . . (iv)}$$

Example 7.5

A hollow shaft of diameter ratio $\frac{3}{8}$ is required to transmit 600 kW at 110 rpm, the maximum torque being 20% greater than mean. The shearing stress is not to exceed 62 MN/m^2 and twist in length of three metres is not to exceed 1.4 degrees. Determine the diameter of the shaft. Assume modulus of rigidity for shaft material as 84 GN/m^2.

Solution

Note that this problem requires consideration of stress and angle of twist. We have to keep the angle of twist, θ within limit. We may also understand at this point that Eq. (7.13) is applicable to solid shaft only. If we want to find θ for hollow shaft we have to recall the basic torsion formulae, i.e.

$$\frac{M_t}{J} = \frac{G\theta}{l}$$

in which J is the moment of inertia of the shaft section. And if we denote outside and inside diameters of hollow shaft with suffixes o and i on d,

$$J = \frac{\pi}{32}(d_o^4 - d_i^4) = \frac{\pi}{32} d_o^4 \left[1 - \left(\frac{d_i}{d_o}\right)^4\right]$$

The given value of $\frac{d_i}{d_o} = \frac{3}{8} = 0.375$

$$\therefore \qquad J = \frac{\pi}{32} d_o^4 \, [1 - (0.375)^4] = 0.0982\,[1 - 0.02]\, d_o^4 = 0.0963\, d_o^4$$

$$\therefore \qquad \theta = 0.0244 \text{ rad} = \frac{M_t\, l}{G\, J}$$

where l = 3000 mm

$$M_{t1} = \frac{H}{\omega} = \text{mean torque} = \frac{600 \times 10^3}{2\pi \frac{110}{60}} = 52.1 \times 10^3 \text{ Nm} = 52.1 \times 10^6 \text{ Nmm}$$

The starting torque $= 1.2\, M_{t1} = 62.52 \times 10^6$ Nmm

$$\therefore \qquad 0.0244 = \frac{62.52 \times 10^6 \times 3000}{84 \times 10^3 \times 0.0963\, d_o^4} = \frac{23.3 \times 10^6}{d_o^4}$$

$$\therefore \qquad d_o = \left(\frac{23.2 \times 10^6}{0.0244}\right)^{\frac{1}{4}} = 175.6 \text{ mm} \qquad \ldots \text{(a)}$$

The second consideration is based on stress,

$$\tau_s = 62 \text{ MN/m}^2 \text{ or N/mm}^2$$

$$\therefore \quad \text{Using} \qquad \tau = \frac{16 M_t}{\pi d^3}$$

$$d_o = \left(\frac{16 \times 62.52 \times 10^6}{\pi \times 62}\right)^{\frac{1}{3}} = 172.5 \text{ mm} \qquad \ldots \text{(b)}$$

From (a) and (b) we see that diameter from consideration of angle of twist is larger hence, this should be accepted. And then the stress will be lower than permissible at

$$\tau = \frac{16 \times 62.52 \times 10^6}{\pi \times (175.6)^3} = 58.8 \text{ N/mm}^2$$

$$\therefore \qquad d_o = 175.6 \text{ mm}, \;\; d_i = 65.85 \text{ mm}$$

SAQ 2

(a) How do you modify equivalent bending moment to take into consideration that the bending moment varies at much higher frequency than the torque on the shaft?

(b) How do you account for the manner in which load is applied upon a shaft-static, sudden or shock?

(c) What is the stiffness of shaft in bending and torsion? How do you consider the deflection and twist in design of shaft?

(d) Under what condition the deflection and twist of shaft become important?

(e) A hollow shaft with diameter ratio 0.7 is required to transmit 500 kW at 300 rpm with a uniform twisting moment. Allowable shearing stress is 60 N/mm^2 and twist in 2.0 m length is not to exceed 1 degree. Calculate the minimum external diameter and internal diameter of the shaft satisfying these conditions and find actual value. G = 8.2 × 10^4 N/mm^2.

7.9 SUMMARY

Shaft is an important machine element and transmits power. Shafts are many types and are made cylindrical. They are subjected to torque and bending moment, hence, at any point in the section of shaft there exists direct bending stress due to bending moment and shearing stress due to torque. They are designed against maximum principal stress or maximum shearing stress. The load (comprising bending moment and torque) is converted into equivalent bending moment or equivalent torque. The diameters are calculated by modifying the expressions for equivalent bending moment and equivalent torque by considering condition and manner of loading. The keyways become essential feature of shafts because some part like gear or pulley has to be attached on it to transmit power. The keys are standardised and can be selected from relevant table. There is yet simpler method to use a square key of depth of $\frac{1}{4}$ diameter of shaft. The shafts are often made in medium carbon steel which can be heat treated. Alloy steel shafts are not uncommon if corrosive atmosphere exists. Cast iron shaft, though used rarely, will tend to become heavier.

Couplings connect coaxial shafts. They are formed by two discs attached to shafts through key and jointed by bolts, parallel to shaft axis. The discs are made as flanges integral with the hub. The flanges are often made in cast iron. Muff couplings are thick cylinders which could be used as sleeves or split to be bolted around the shaft. The driving force in muff coupling is friction between the inner surface of muff and outer surface of shaft. The muff can be a single piece sleeve keyed to shafts or split in halves which are tightened by the bolts. The muff is made in cast iron.

7.10 ANSWERS TO SAQs

SAQ 2

(e) Follow Example 7.2 of Section 7.12.

$$D_o = D, \quad D_i = 0.7\,D$$

$$J = \frac{\pi}{32}\,(D_o^4 - D_i^4) = \frac{\pi}{32}\,D^4\ (1 - 0.24) = 74.6 \times 10^{-3}\ D^4$$

$$M_t = \frac{H}{\omega} = \frac{500 \times 10^3}{2\pi\,\frac{300}{60}} = \frac{106.5 \times 10^3}{\pi} = 15.9 \times 10^3 \text{ Nm} = 15.9 \times 10^6 \text{ Nmm}$$

$$\tau = \frac{M_t}{J} \cdot \frac{D}{2} = \frac{15.9 \times 10^6 \times 0.5}{74.6 \times 10^{-3}\ D^3} = \frac{106.5 \times 10^6}{D^3} = 61 \text{ N/mm}^2$$

$$\therefore \qquad D = \left(\frac{106.5 \times 10^6}{60}\right)^{\frac{1}{3}} = 12.12 \text{ mm} \qquad \text{. . . (a)}$$

Angle of twist, $\theta = \dfrac{M_t\, l}{G\, J} = \dfrac{15.9 \times 10^6 \times 2.0 \times 10^3}{8.2 \times 10^4 \times 74.6 \times 10^{-3}\ D^4} = \dfrac{\pi}{180}$

$$D = \left(\frac{180 \times 5.20 \times 10^6}{\pi}\right)^{\frac{1}{4}} = 131.4 \text{ mm} \qquad \text{. . . (b)}$$

The diameter at (b) will be selected.

$$D_o = 131.5 \text{ mm}, \quad d_i = 92 \text{ mm}$$

FURTHER READING

Joseph Edward Shigley (1986), *Mechanical Engineering Design*, First Metric Edition, McGraw-Hill Book Company, New York.

V. M. Faires (1965), *Design of Machine Elements*, 4th Edition, The Macmillan Company, New York.

W. Cawthorne and A. L. Mellanby, *The Elements of Machine Design*, Longmans Green and Company Limited, London.

Rajendra Karwa (2006), *Machine Design*, 2nd Edition, Laxmi Publication (P) Ltd., New Delhi.

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P. C. Sharma and D. K. Aggarwal (1987), *Machine Design*, 4th Edition, Katson Publishing House, Ludhiana.

MACHINE DESIGN

Design is a process that ends in creation of something which will satisfy some need of a person, group of persons or society. The homes and buildings in which we reside, the dams which store water for irrigation or generation of electricity, an engine which is used for pumping water or a hoist for lifting loads are the things that are designed before they are made. The Course on Machine Design consists of seven units. First unit will give to you an Introduction to Machine Design. In this unit you will study about the properties of engineering materials and procedure of designing machine elements. Second unit is on

Design of Temporary Connections, this unit will provide you detail about the calculation of Diameter of a Bar, Knuckle Joint, and Cotter Joint.

In third unit, you will study about Rivet Joint. In engineering practice it is often required that two sheets or plates are joined together and carry the load in such ways that the joint is loaded. Many times such joints are required to be leak proof so that gas contained inside is not allowed to escape. A riveted joint is easily conceived between two plates overlapping at edges, making holes through thickness of both, passing the stem of rivet through holes and creating the head at the end of the stem on the other side.

Fourth unit is devoted to Welded Joint. The unit consists of Welded Connections, Types of Welding Joints, Strength, T-Joint, Unsymmetrical Section Loaded Axially, and Eccentrically Loaded Welded Joint.

In Fifth unit you will study about Design of Screws, Fasteners and, Power Screws. Screws are used for power transmission or transmission of force. A screw is a cylinder on whose surface helical projection is created in form of thread. In this unit, you will study about Geometry of Thread, Mechanics of Screw and Nut Pair, Power Screw Mechanics, Application of Power Screw, Standard Threads, Design of Screw and Nut, Threaded Fastener, Failure of Bolts and Screws, and Permissible Stresses in Bolts.

Sixth unit is on Keys and Couplings in which you will study about types of keys, Forces Acting on a Sunk Key, Strength of a Sunk Key, Effect of Keyways, and Couplings.

A key is a piece of steel inserted between the shaft and hub or boss of the pulley to connect these together in order to prevent relative motion between them. It is always inserted parallel to the axis of the shaft.

Last unit is on Shafts. Shafts form the important elements of machines. They are the elements that support rotating parts like gears and pulleys and in turn are themselves supported by bearings resting in the rigid machine housings. In this unit you will study about types of Shaft, Materials for Shafts, Shaft Strength under Torsional Load, Stresses in Bending and Torsion, Shaft Loading, Shafts under Torsion and Bending, and Stiffness of Shaft