CHAPTER – 4 - TORQUE TRANSMITTING JOINTS

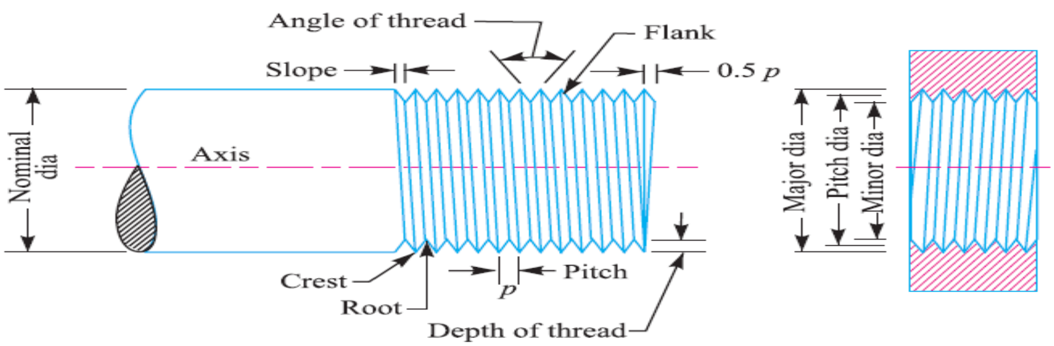
SCREW THREADS

4.1 Introduction

A screw thread is formed by cutting a continuous helical groove on a cylindrical surface. A screw made by cutting a single helical groove on the cylinder is known as ***single threaded*** (or single-start) screw and if a second thread is cut in the space between the grooves of the first, a ***double threaded*** (or double-start) screw is formed. Similarly, triple and quadruple (*i.e.* multiple-start) threads may be formed.The helical grooves may be cut either ***right hand*** or ***lefthand*.**

4.2 Important Terms Used in Screw Threads

The following terms used in screw threads, as shown in Fig. 11.1, are important from the subject point of view:



1. *Major diameter*. It is the largest diameter of an external or internal screw thread. The screw is specified by this diameter. It is also known as ***outside*** or ***nominal diameter*.**

2. *Minor diameter*. It is the smallest diameter of an external or internal screw thread. It is also known as ***core*** or ***root diameter*.**

3. *Pitch diameter*. It is the diameter of major diameter and minor diameter.

4. *Pitch*. It is the distance from a point on one thread to the corresponding point on the next. Mathematically, Pitch =1/No. of threads per unit length of screw

5. *Crest*. It is the top surface of the thread.

6. *Root*. It is the bottom surface created by the two adjacent flanks of the thread.

7. *Depth of thread*. It is the perpendicular distance between the crest and root.

8. *Flank*. It is the surface joining the crest and root.

9. *Angle of thread*. It is the angle included by the flanks of the thread.

10. *Slope*. It is half the pitch of the thread.

4.3 Forms of Screw Threads

The following are the various forms of screw threads.

1. *British standard whit worth (B.S.W.) thread*. This is a British standard thread profile and has coarse pitches. It is a symmetrical *V*-thread in which the angle between the flanks, measured in an axial plane, is 55°. These threads are found on bolts and screwed fastenings for special purposes. The various proportions of B.S.W. threads are shown in Fig. 11.2.

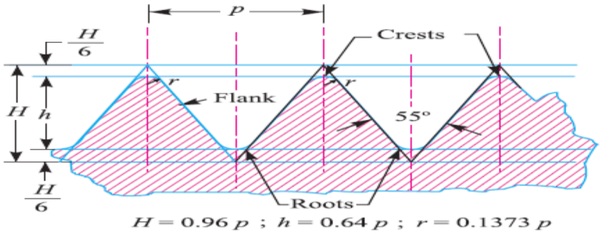


Fig. 11.2. British standard whitworth (B.S.W) thread.

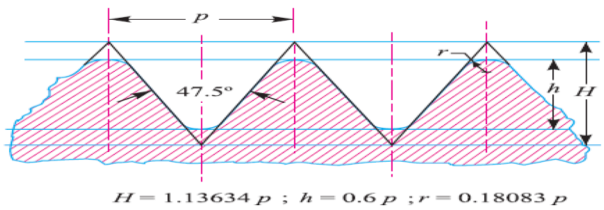


Fig. 11.3. British association (B.A.) thread.

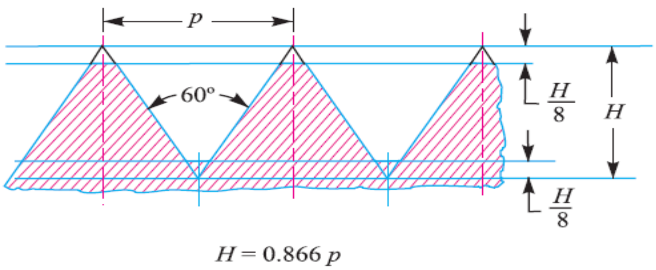


Fig. 11.4. American national standard thread.

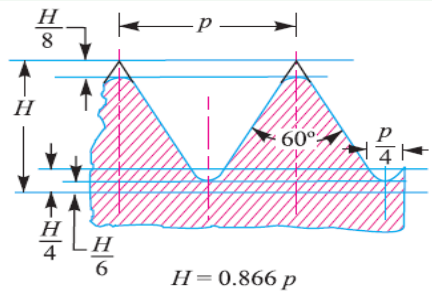


Fig. 11.5. Unified standard thread.

2. *British association (B.A.) thread*. This is a B.S.W. thread with fine pitches. The proportions of the B.A. thread are shown in Fig. 11.3. These threads are used for instruments and other precision works.

3. *American national standard thread*. The American national standard or U.S. or Seller's thread has flat crests and roots. The flat crest can withstand more rough usage than sharp *V*-threads. These threads are used for general purposes *e.g.* on bolts, nuts, screws and tapped holes. The various proportions are shown in Fig. 11.4.

4. *Unified standard thread*. The three countries *i.e.*, Great Britain, Canada and United States came to an agreement for a common screw thread system with the included angle of 60°, in order to facilitate the exchange of machinery. The thread has rounded crests and roots, as shown in Fig. 11.5.

5. *Square thread*. The square threads, because of their high efficiency, are widely used for transmission of power in either direction. Such types of threads are usually found on the feed mechanisms of machine tools, valves, spindles, screw jacks etc. The square threads are not so strongas V-threads but they offer less frictional resistance to motion than Whitworth threads. The pitch of the square thread is often taken twice that of a B.S.W. thread of the same diameter. The proportions of the thread are shown in Fig. 11.6.

6. *Acme thread*. It is a modification of square thread. It is much stronger than square thread and can be easily produced. These threads are frequently used on screw cutting lathes, brass valves, cocks and bench vices. When used in conjunction with a split nut, as on the lead screw of a lathe, the tapered sides of the thread facilitate ready engagement and disengagement of the halves of the nut when required. The various proportions are shown in Fig. 11.7.

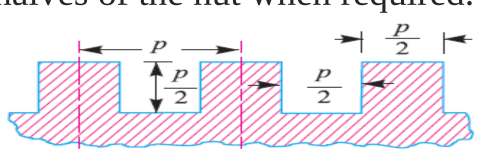


Fig. 11.6. Square thread.

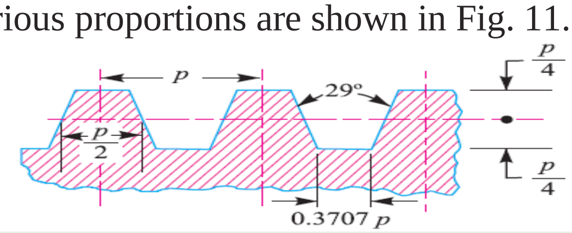


Fig. 11.7. Acme thread.

7. *Knuckle thread.* It is also a modification of square thread. It has rounded top and bottom. It can be cast or rolled easily and cannot economically be made on a machine. These threads are used for rough and ready work. They are usually found on railway carriage couplings, hydrants, necks of glass bottles and large molded insulators used in electrical trade.

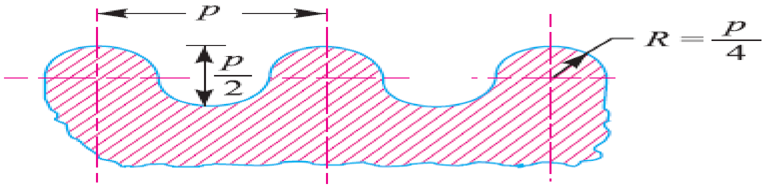


Fig. 11.8. Knuckle thread.

4.4 Designation of Screw Threads

According to Indian standards, IS : 4218 (Part IV) 1976 (Reaffirmed 1996), the complete designation of the screw thread shall include

1. *Size designation.* The size of the screw thread is designated by the letter `*M*' followed by the diameter and pitch, the two being separated by the sign ×. When there is no indication of the pitch, it shall mean that a coarse pitch is implied.

2. *Tolerance designation.* This shall include

(*a*) A figure designating tolerance grade as indicated below:

'7' for fine grade, '8' for normal (medium) grade, and '9' for coarse grade.

(*b*) A letter designating the tolerance position as indicated below:

'*H*' for unit thread, '*d*' for bolt thread with allowance, and '*h*' for bolt thread without allowance.

For example, A bolt thread of 6 mm size of coarse pitch and with allowance on the threads and normal (medium) tolerance grade is designated as *M*6-8*d*.

4.5 Stresses in Screwed Fastening due to Static Loading

The following stresses in screwed fastening due to static loading are important from the subject point of view:

1. Internal stresses due to screwing up forces,

2. Stresses due to external forces

1. *Tensile stress due to stretching of bolt.* Since none of the above mentioned stresses are accurately determined, therefore bolts are designed on the basis of direct tensile stress with a large factor of safety in order to account for the indeterminate stresses. The initial tension in a bolt, based on experiments, may be found by the relation

$$Pi = 2840\, d \text{ N}$$

Where Pi = Initial tension in a bolt, and d = Nominal diameter of bolt, in mm.

The above relation is used for making a joint fluid tight like steam engine cylinder cover joints etc. When the joint is not required as tight as fluid-tight joint, then the initial tension in a bolt may be reduced to half of the above value. In such cases

$$Pi = 1420\, d \text{ N}$$

The small diameter bolts may fail during tightening, therefore bolts of smaller diameter (less than M 16 or M 18) are not permitted in making fluid tight joints. If the bolt is not initially stressed, then the maximum safe axial load which may be applied to it,
is given by

P = Permissible stress × Cross-sectional area at bottom of the thread

$$\text{Stress area} = \frac{\pi}{4}\left(\frac{d_p + d_c}{2}\right)^2$$

Where d_p = Pitch diameter, and d_c = Core or minor diameter

2. *Torsional shear stress caused by the frictional resistance of the threads during its tightening*.

The torsional shear stress caused by the frictional resistance of the threads during its tightening may be obtained by using the torsion equation. We know that

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

$$\tau = \frac{T}{J} \times r = \frac{T}{\frac{\pi}{32}(d_c)^4} \times \frac{d_c}{2} = \frac{16\,T}{\pi\,(d_c)^3}$$

Where τ = Torsional shear stress, *T* = Torque applied, and d_c = Minor or core diameter of the thread.

It has been shown during experiments that due to repeated unscrewing and tightening of the nut, there is a gradual scoring of the threads, which increases the torsional twisting moment (*T*).

3. *Shear stress across the threads*. The average thread shearing stress for the screw (τ_s) is obtained by using the relation:

$$\tau_s = \frac{P}{\pi d_c \times b \times n}$$

where b = Width of the thread section at the root.

The average thread shearing stress for the nut is

$$\tau_n = \frac{P}{\pi d \times b \times n}$$

where d = Major diameter.

4. *Compression or crushing stress on threads*. The compression or crushing stress between the threads (σ_c) may be obtained by using the relation:

$$\sigma_c = \frac{P}{\pi\,[d^2 - (d_c)^2]\,n}$$

Where *d* = Major diameter, d_c = Minor diameter, and *n* = Number of threads in engagement.

5. *Bending stress if the surfaces under the head or nut are not perfectly parallel to the bolt axis*. When the outside surfaces of the parts to be connected are not parallel to each other, then the bolt will be subjected to bending action. The bending stress (σ_b) induced in the shank of the bolt is given by

$$\sigma_b = \frac{x\,.\,E}{2l}$$

Where *x* = Difference in height between the extreme corners of the nut or head,
l = Length of the shank of the bolt, and *E* = Young's modulus for the material of the bolt.

4.6 Stresses due to External Forces

The following stresses are induced in a bolt when it is subjected to an external load.

1. *Tensile stress*. The bolts, studs and screws usually carry a load in the direction of the bolt axis which induces a tensile stress in the bolt.

Let *dc* = Root or core diameter of the thread, and ʃ*t* = Permissible tensile stress for the bolt material.

We know that external load applied,

$$P = \frac{\pi}{4}(d_c)^2 \ \sigma_t \qquad \text{or} \qquad d_c = \sqrt{\frac{4\ P}{\pi\ \sigma_t}}$$

2. *Shear stress*. Sometimes, the bolts are used to prevent the relative movement of two or more parts, as in case of flange coupling, then the shear stress is induced in the bolts. The shear stresses should be avoided as far as possible. It should be noted that when the bolts are subjected to direct shearing loads, they should be located in such a way that the shearing load comes upon the body (*i.e.*

shank) of the bolt and not upon the threaded portion. In some cases, the bolts may be relieved of shear load by using shear pins. When a number of bolts are used to share the shearing load, the finished bolts should be fitted to the reamed holes.

Let *d* = Major diameter of the bolt, and *n* = Number of bolts.

Shearing load carried by the bolts,

$$P_s = \frac{\pi}{4} \times d^2 \times \tau \times n \qquad \text{or} \qquad d = \sqrt{\frac{4\ P_s}{\pi\,\tau\,n}}$$

3. *Combined tension and shear stress*. When the bolt is subjected to both tension and shear loads, as in case of coupling bolts or bearing, then the diameter of the shank of the bolt is obtained from the shear load and that of threaded part from the tensile load. A diameter slightly larger than that required for either shear or tension may be assumed and stresses due to combined load should be

checked for the following principal stresses.

Maximum principal shear stress,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_t)^2 + 4\tau^2}$$

and maximum principal tensile stress,

$$\sigma_{t(max)} = \frac{\sigma_t}{2} + \frac{1}{2}\sqrt{(\sigma_2)^2 + 4\tau^2}$$

4.7 Introduction

A key is a piece of mild 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. Keys are used as temporary fastenings and are subjected to considerable crushing and shearing stresses. A keyway is a slot or recess in a shaft and hub of the pulley to accommodate a key.

KEYS

KEYS

- ✓ A key is piece of mild steel inserted between transmission shafts and the hub of the rotating element like gear, pulley or sprocket to connect them together.
- ✓ It is used to transmit power for long distance
- ✓ Keys are temporary fasteners.
- ✓ The keys are subjected to crushing and shearing stresses.

Types of Keys

The following types of keys are important from the subject point of view:

1. Sunk keys, **2.**Saddle keys, **3.**Tangent keys, **4.**Round keys, and **5.**Splines.

Sunk Keys

1. *Rectangular sunk key*. A rectangular sunk key is shown in Fig. 13.1. The usual proportions of this key are:

Width of key, $w = d / 4$; and thickness of key, $t = 2w / 3 = d / 6$

where$d$ = Diameter of the shaft or diameter of the hole in the hub. The key has taper 1 in 100 on the top side only.

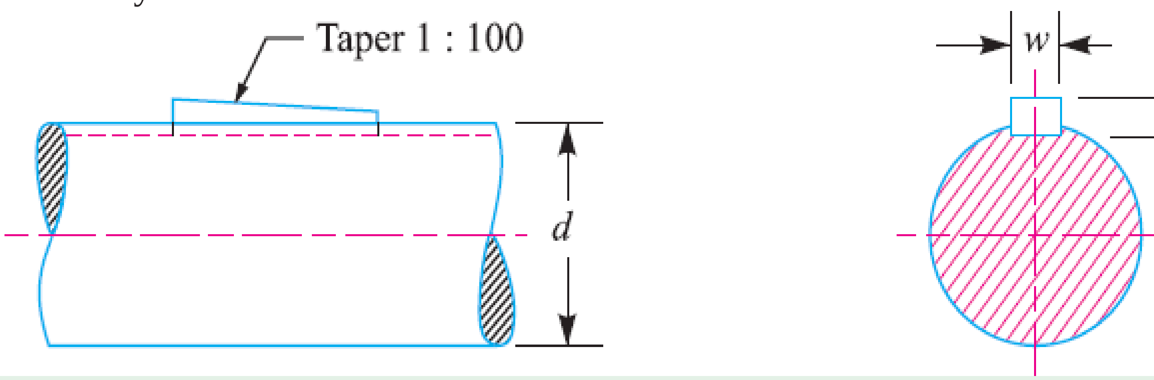


Fig. 13.1. Rectangular sunk key.

2. *Square sunk key*. The only difference between a rectangular sunk key and a square sunk key is that its width and thickness are

equal, *i.e.* $w = t = d / 4$

3. *Parallel sunk key*. The parallel sunk keys may be of rectangular or square section uniform in width and thickness throughout. It

may be noted that a parallel key is a taperless and is used where the pulley, gear or other mating piece is required to slide along the shaft.

4. *Gib-head key*. It is a rectangular sunk key with a head at one end known as ***gib head***. It is usually provided to facilitate the removal

of key. A gib head key is shown in Fig. 13.2 and its use in shown in Fig. 13.2 (*b*).

The usual proportions of the gib head key are :

Width, $w = d / 4$; and thickness at large end, $t = 2w / 3 = d / 6$

5. *Feather key*. A key attached to one member of a pair and which permits relative axial movement is known as ***feather key***. It is a special type of parallel key which transmits a turning moment and also permits axial movement. It is fastened either to the shaft or hub, the key being a sliding fit in the key way of the moving piece.

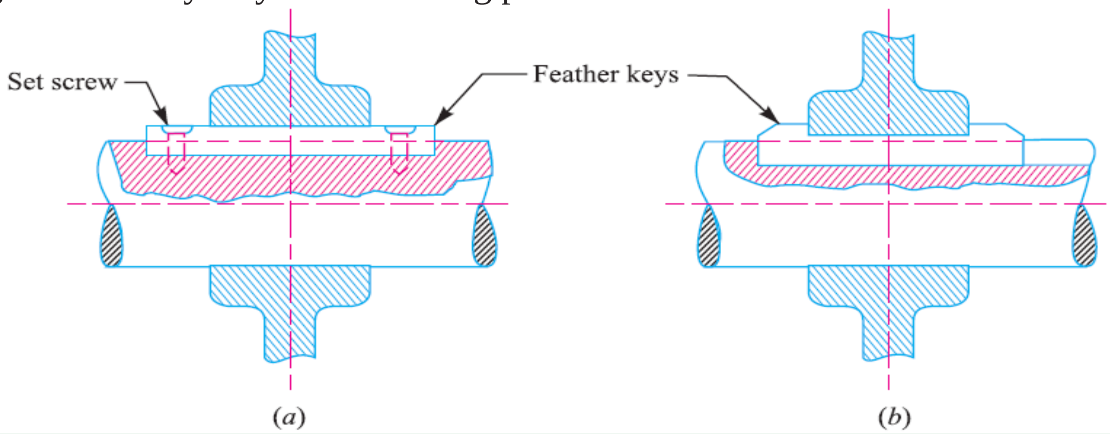


Fig. 13.3. Feather key.

6. *Woodruff key*. The woodruff key is an easily adjustable key. It is a piece from a cylindrical disc having segmental cross-section in front view as shown in Fig. 13.4. A woodruff key is capable of tilting in a recess milled out in the shaft by a cutter having the same curvature as the disc from which the key is made. This key is largely used in machine tool and automobile construction.

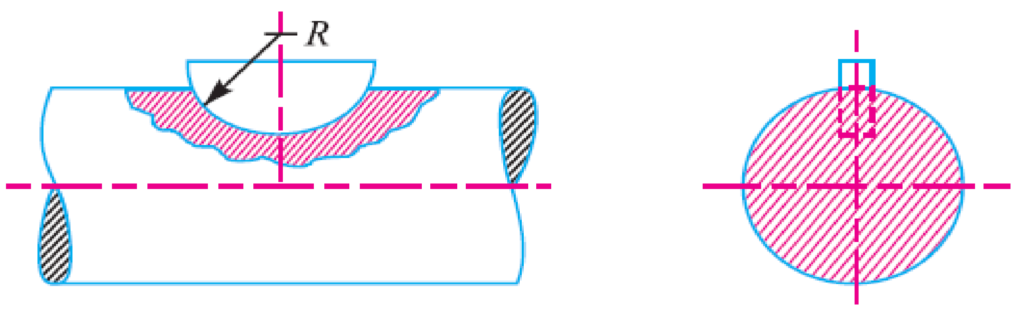


Fig. 13.4. Woodruff key.

Saddle keys

The saddle keys are of the following two types :

1. Flat saddle key, and **2.** Hollow saddle key.

A ***flat saddle key*** is a taper key which fits in a keyway in the hub and is flat on the shaft as shown in Fig. 13.5. It is likely to slip round the shaft under load. Therefore it is used for comparatively light loads.

A ***hollow saddle key*** is a taper key which fits in a keyway in the hub and the bottom of the key is shaped to fit the curved surface of the shaft. Since hollow saddle keys hold on by friction, therefore these are suitable for light loads. It is usually used as a temporary fastening in fixing and setting eccentrics, cams etc.

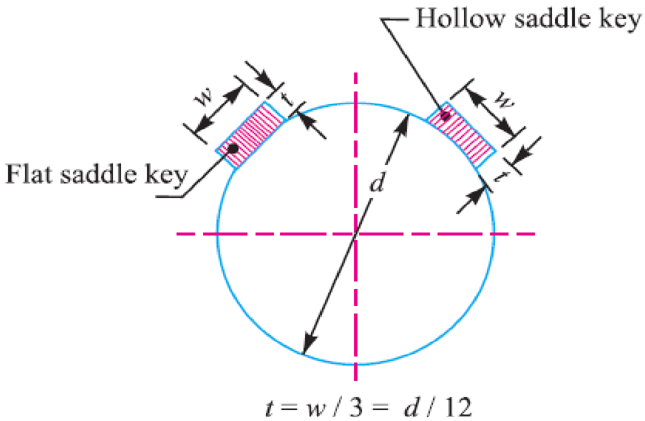


Fig. 13.5. Saddle key.

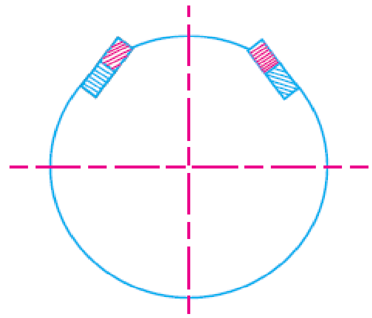


Fig. 13.6. Tangent key.

Tangent Keys

The tangent keys are fitted in pair at right angles as shown in Fig. 13.6. Each key is to withstand torsion in one direction only. These are used in large heavy duty shafts.

Round Keys

The round keys, as shown in Fig. 13.7(*a*), are circular in section and fit into holes drilled partly in the shaft and partly in the hub. They have the advantage that their keyways may be drilled and reamed after the mating parts have been assembled. Round keys are usually considered to be most appropriate for low power drives.

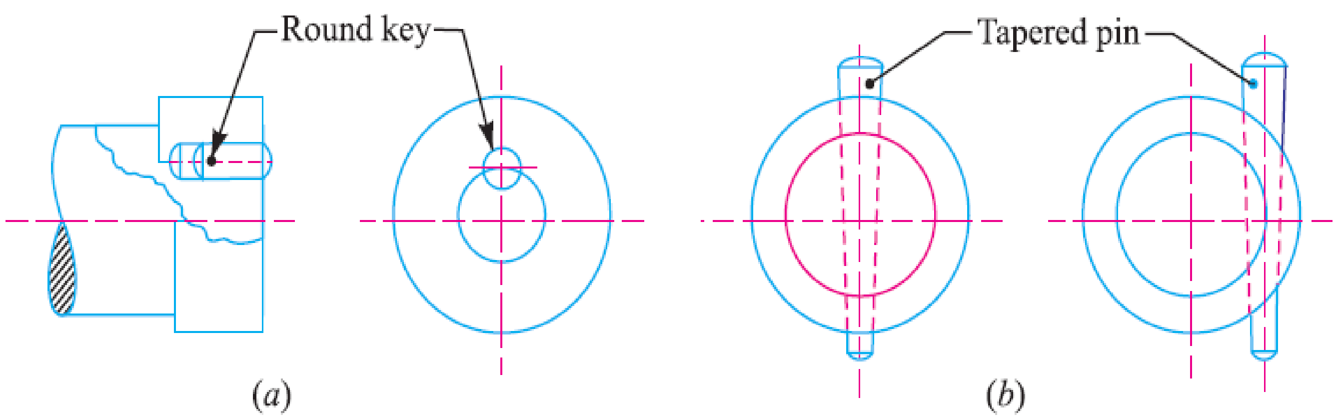


Fig. 13.7. Round keys.

Splines

Sometimes, keys are made integral with the shaft which fits in the keyways broached in the hub. Such shafts are known as ***splined shafts*** as shown in Fig. 13.8. These shafts usually have four, six, ten or sixteen splines. The splined shafts are relatively stronger than shafts having a single keyway. The splined shafts are used when the force to be transmitted is large in proportion to the size of the shaft as in automobile transmission and sliding gear transmissions. By using splined shafts, we obtain axial movement as well as positive drive is obtained.

DESIGN OF SUNK KEYS

DESIGN OF SUNK KEYS.

The Sunk Keys are provided half in the keyway of the shaft and half in the keyway of hub of pulley (or) gear.

Usual Proportion formulas:-

1) Rectangular Sunk Key:

Width of the Key $= w = \frac{d}{4}$ in mm.

Thickness of Key $= t = \frac{d}{6}$ in mm.

(or)

$t = \frac{2}{3} w$ mm

Length of the Key $= l = 4w$ mm.

(or)

$l = 1.57d$ mm.

2) Square Key:

Width of the Key = Thickness of the Key.

where,

d - Diameter of shaft [mm].

3) Shear stress induced in the key.

Tangential force $= F = \sigma_{sk} \times l \times w$.

Torque: $T = F \times R$.

Torque = Tangential force × Radius of shaft.

$T = \sigma_{sk} \times l \times w \times \frac{d}{2}$.

4) Crushing stress induced in key.

Tangential force = [illegible].

$F = \sigma_{ck} \times l \times \frac{t}{2}$

Torque = Tangential force × Radius of shaft

$T = F \times R$.

$T = \sigma_{ck} \times l \times \frac{t}{2} \times \frac{d}{2}$.

5) Considering shearing failure of key.

Length of the key $= l = \frac{2T}{d \times w \times \sigma_{sk}}$

width of the key $= w = \frac{2T}{d \times l \times \sigma_{sk}}$

6) Considering Crushing failure of key.

Length of the key $= l = \dfrac{4T}{d \times t \times \sigma_{ck}}$

Thickness of the key $= t = \dfrac{4T}{d \times l \times \sigma_{ck}}$.

7) Effect of Key ways.

Effect of Keyways.

$$e = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)$$

$$k_\theta = 1 + 0.4\left(\frac{w}{d}\right) + 0.7\left(\frac{h}{d}\right)$$

where,

$$h = \frac{t}{2}.$$

Problem of Keys.

ril- 2001.

(*) 1) A shaft 30mm diameter is transmitting power at a maximum shear stress of 80 Mpa. If a pulley is connected to the shaft by means of key. Find the dimensions of a key. So that the [illegible] stress in the key is not to exceed 50 Mpa and the length of key is 4 times the width of key.

Given data:-

Diameter of shaft $= d = 30mm$.

Shear stress of shaft $= \sigma_{ss} = 80$ Mpa.

$\sigma_{ss} = 80\ N/mm^2$

Shear stress in key $= \sigma_{sK} = 50$ Mpa.

$\sigma_{sK} = 50\ N/mm^2$.

Length of the key $= l = 4w$.

To Find:

Dimensions of the key. [l, t, w]

Solu.

Torque on Shaft:

Torque = $T = \frac{\pi}{16} \times d^3 \times \sigma_{ss}$.

$= \frac{\pi}{16} \times (30)^3 \times 80$

$T = 424.115 \times 10^3$ N.mm.

Torque in the shaft = Torque in the key.

Tangential force:

$T = F \times R$.

$F = \frac{T}{R} = \frac{424.115 \times 10^3}{15}$

$F = 28.274 \times 10^3$ N.

width of key:

Tangential force = $F = \sigma_{sk} \times l \times w$.

$28.274 \times 10^3 = 50 \times 4w \times w$.

$4w^2 = \frac{28.274 \times 10^3}{50}$

$4w^2 = 565.48$

$$w^2 = \frac{565.48}{4}$$

$$w^2 = 141.37$$

$$w = 11.889 \text{ mm.}$$

7.D.B. N.LP

Pg.no 21 → width of the Key = w = 12 mm.

$$\text{width} = w = \frac{d}{4}$$

$$= \frac{30}{4}$$

$$w = 7.5 \text{ mm.}$$

width = w = 8 mm.

Hence we take the highest value of width = w = 12mm.

Length of the Key $= l = 4w.$

$$= 4 \times 12$$

$$l = 48 \text{ mm.}$$

Length of Key $= l = 1.57\, d.$

$$= 1.57 \times 30$$

$$l = 47.1 \text{mm.}$$

Hence we take the highest value of length = l = 48 mm.

Thickness of the Key = $t = \frac{d}{6}$

$= \frac{30}{6}$

$t = 5\ mm.$

Thickness of Key = $t = \frac{2}{3}\ w.$

$= \frac{2}{3} \times 12.$

$t = 8\ mm.$

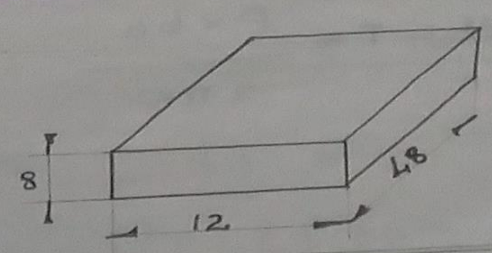
Hence we take the highest value of Thickness = t = 8 mm.

Result :

Length of the Key = L = 48 mm.

width of the Key = w = 12 mm.

Thickness of the Key = t = 8 mm.



(32) April - [1994-95]

Design a sunk key suitable for 60mm diameter of shaft to transmit 18 KW at 500 rpm. The permissible stresses in the key materials are 60 N/mm² in shearing and 150 N/mm² in crushing.

Given data:-

Diameter of shaft = d = 60 mm

Power = $P = 18 \times 10^3$ W.

speed = N = 500 rpm.

shearing stress in key = σ_{sk} = 60 N/mm²

Crushing stress in key = σ_{ck} = 150 N/mm²

To find:

Design of sunk key.

soln.

Torque on shaft:

$$\text{Torque: } T = \frac{P \times 60}{2 \pi N}$$

$$= \frac{18 \times 10^3 \times 60}{2 \times \pi \times 500}$$

$T = 343.775$ N-m.

$T = 343.775 \times 10^3$ N-mm.

From N.L.P. D.B. Pg.No - 21.

$w = 18$ mm

$t = 11$ mm.

$l = 4w$

$= 4 \times 18$

$l = 72$ mm.

From usual Proportions.

Width $= w = \frac{d}{4} = \frac{60}{4}$

$w = 15$ mm.

Thickness $= t = \frac{d}{6} = \frac{60}{6}$

$t = 10$ mm.

Thickness $= t = \frac{2}{3} w = \frac{2}{3} \times 15$

$t = 10$ mm.

Length $= l = 4w = 4 \times w = 4 \times 15$

$l = 60$ mm.

Length = $l = 1.57 d$ = 1.57×60

$l = 94.2$ mm.

using shearing stress :

$$T = \sigma_{sk} \times l \times w \times \frac{d}{2}$$

$$343.775 \times 10^3 = 60 \times l \times 18 \times 30.$$

$l = 10.6$ mm.

using Crushing Stress :

$$T = \sigma_{ck} \times l \times \frac{t}{2} \times \frac{d}{2}$$

$$343.775 \times 10^3 = 150 \times l \times 5.5 \times 30$$

$l = 13.889$ mm.

using shearing stress :

$$T = \sigma_{sk} \times l \times w \times \frac{d}{2}$$

$$343.775 \times 10^3 = 60 \times 94.2 \times w \times 30$$

$w = 2.027$ mm.

Using Crushing stress:

$$T = \sigma_{ck} \times l \times \frac{t}{2} \times \frac{d}{2}$$

$$343.775 \times 10^3 = 150 \times 94.2 \times \frac{t}{2} \times 30.$$

$$\frac{t}{2} = 0.8109$$

$$t = 1.62 \text{ mm}.$$

Result:-

Taking higher value dimensions:

Length of the key = l = 94.2mm.

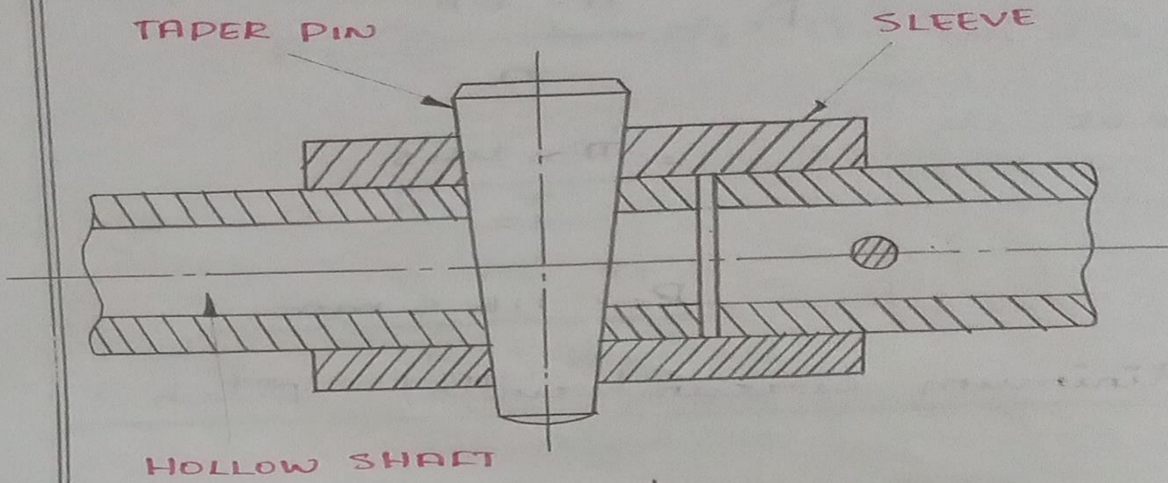
width of the key = w = 18mm.

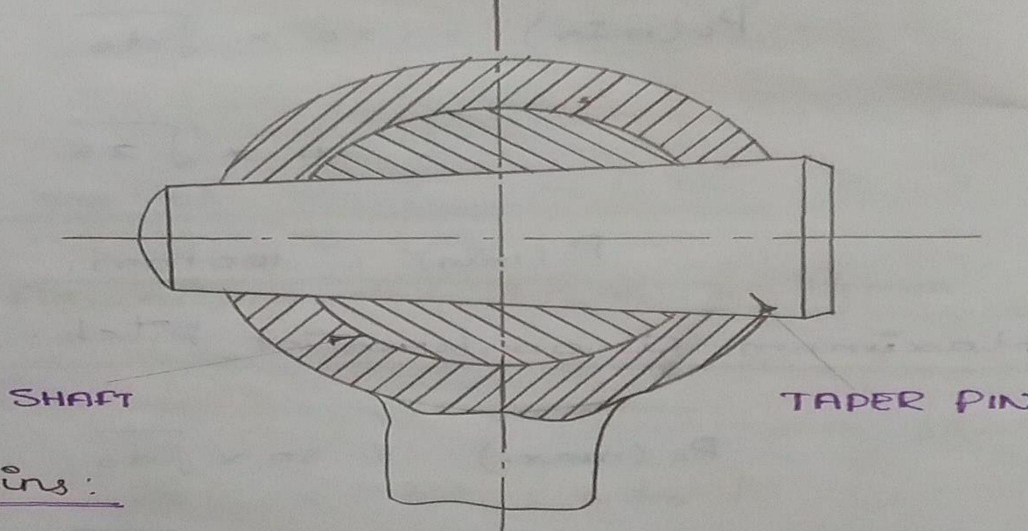
Thickness of the key = t = 11mm.

PIN JOINTS

DESIGN OF PINS

DESIGN OF PINS





Pins:

- Pins are generally used in levers, wheels and hollow shafts.
- They are usually tapered [1 in 50] for easy assembly.
- They are made up of steel.

The diameter of the taper pin [d_p]

is measured along the centre line of the shaft.

Where,

T - Torque transmitted on shaft [N-mm]

d - diameter of the shaft [mm]

d_p - Diameter of pin

τ_p - Shear stress in pin material [N/mm²]

$[\tau]_p$ - Allowable or permissible shear stress in pin material [N/mm²]

F - Tangential force by the pin.

Torque transmitted:

$$T = F \times \frac{d}{2}$$

Diameter of pin:

$$d_p = \sqrt{\frac{4T}{\pi \times [\tau]_p \times d}}$$

Tangential force on pin :-

F = Shearing area × Shearing stress.

$$F = 2 \times \frac{\pi}{4} d_p^2 \times \tau_p \quad [\text{For Pin}]$$

$$T = F \times l. \quad [\text{For lever}].$$

1) A tangential force of 5000 N is applied to the taper pin which fits on 40mm dia. of shaft. Determine the diameter of taper pin. Assume allowable shear stress of pin material as 275 N/mm². (280)

Given data:

Tangential force = F = 5000 N.

diameter of shaft = d = 40mm.

Permissible shear stress = $[\tau]_p$ = 275 N/mm².

To Find:

Diameter of taper pin.

solu.

$$\text{Torque} = T = F \times \frac{d}{2}$$

$$= 5000 \times \frac{40}{2}$$

$$T = 100 \times 10^3 \text{ N-mm.}$$

$$d_p = \sqrt{\frac{4T}{\pi \times [\tau]_p \times d}}$$

$$= \sqrt{\frac{4 \times 100 \times 10^3}{\pi \times 275 \times 40}}$$

$$d_p = 3.4$$

$$d_p = 4 \text{ mm.}$$

Result :

Diameter of pin = $d_p = 4$ mm.

2) A tangential force of 4800N is applied to the taper pin which fits on 40 mm dia. of shaft. Determine the diameter of taper pin. Assume allowable shear stress of pin material as 280 N/mm².

Given data :

Tangential force : $F = 4800 N$.

Diameter of shaft = $d = 40 mm$.

Permissible shear stress = $[\tau]_p = 280 N/mm^2$.

To find :

Diameter of taper pin.

Soln.

Torque = $T = F \times \frac{d}{2}$

$= 4800 \times \frac{40}{2}$

$T = 96 \times 10^3$ N-mm.

$$d_p = \sqrt{\frac{4T}{\pi \times [\tau]_p \times d}}$$

$$= \sqrt{\frac{4 \times 96 \times 10^3}{\pi \times 280 \times 40}}$$

$d_p =$ 3.30 mm.

$d_p =$ 4 mm.

Result :

Diameter of taper pin = d_p = 4 mm.

3) A propeller of an out board motor has 45 mm outside hub which fits on 35 mm diameter shaft. The hub and shaft are fastened by a brass shear pin. If the over load occurs at the propeller, the pin will shear thus avoiding damage to the mechanism. Calculate the dia. of shear pin which will fail at a torque of 90000 N-mm. with ultimate shear strength of 280 N/mm².

Given data:

outer hub = 45 mm.

Diameter of shaft = $d = 35$ mm.

Torque = $T = 90 \times 10^3$ N-mm.

ultimate shear strength = $[\tau]_p = 280$ N/mm².

To Find:

Diameter of shear pin.

Soln:

$$d_p = \sqrt{\frac{4T}{\pi \cdot [\tau]_p \cdot d}}$$

$$= \sqrt{\frac{4 \times 90 \times 10^3}{\pi \times 280 \times 35}}$$

$$= \sqrt{11.6930}$$

$$d_p = 3.41 \text{ mm.}$$

$$d_p = 4 \text{mm.}$$

Result:

Diameter of shear pin $= d_p = 4$ mm.

4) (*) A 300mm lever is fixed to 37.5mm shaft by means of a taper pin through its hub perpendicular to the axis. The mean diameter of pin is 10mm. What pull on the end of this lever cause a shearing stress on the pin of 65 N/mm² and what torsion stress will this produce on the shaft.

Given data:-

Length of lever = l = 300mm.

Diameter of shaft = d = 37.5mm.

Diameter of pin = d_p = 10mm.

Shear stress = τ_p = 65 N/mm².

To Find:

Torsion stress $[\sigma_T]$.

Tangential force [F].

Soln.

(*) NOTE:-

Torque = $T = F \times l$.

Tangential force:-

Torque:-

$$d_p = \sqrt{\frac{4T}{\pi \cdot \tau_p \cdot d}}$$

$$10 = \sqrt{\frac{4 \times T}{\pi \times 65 \times 37.5}}$$

$$100 = \frac{4 \times T}{\pi \times 65 \times 37.5}$$

$$100 = \frac{4T}{7657.632}$$

$$4T = 100 \times 7657.632$$

$$4T = 765.763 \times 10^3$$

$$T = \frac{765.763 \times 10^3}{4}$$

$$T = 191.441 \times 10^3 \text{ N-mm.}$$

$T = F \times l$.

$$F = \frac{T}{l} = \frac{191.441 \times 10^3}{300}$$

$F = 638.136$ N.

Torsion stress :

$$T_T = \frac{\pi}{16} \times d^3 \times \sigma_T.$$

$$\sigma_T = \frac{T \times 16}{\pi \times d^3}$$

$$= \frac{191.441 \times 10^3 \times 16}{\pi \times (37.5)^3}$$

$\sigma_T = 18.48$ N/mm².

Result :

Tangential force = $F = 638.136$ N.

Torsion stress = $\sigma_T = 18.48$ N/mm².