CHAPTER 4

TORQUE TRANSMITING JOINTS

1. Introduction

So that parts may rotate together with the shafts or axles on which they are mounted, use is made of keys, splines, pins, fits etc.; All these change the original geometry of the shafts and axles and give rise to stress concentration. Interference fits result in radial pressures on the mounting surfaces, which are non-uniformly distributed over the hub length. These joints will be discussed in detail as follows.

2. Keys

A key can be defined as a machine element that is used to connect the transmission shaft to rotating machine elements like pulley, gear; sprocket or flywheel. A keyed joint consisting of shaft, hub and key is illustrated in Fig. 1. There are two basic functions of the key. They are as follows:

(i) The primary function of the key is to transmit the torque from the shaft to the hub of mating element and vice versa.
(ii) The second function of the key is to prevent relative rotational motion between the shaft and the joined machine element like gear or pulley. In most of the cases, key also prevents axial motion between two elements, except in case of feather key or splined connection.

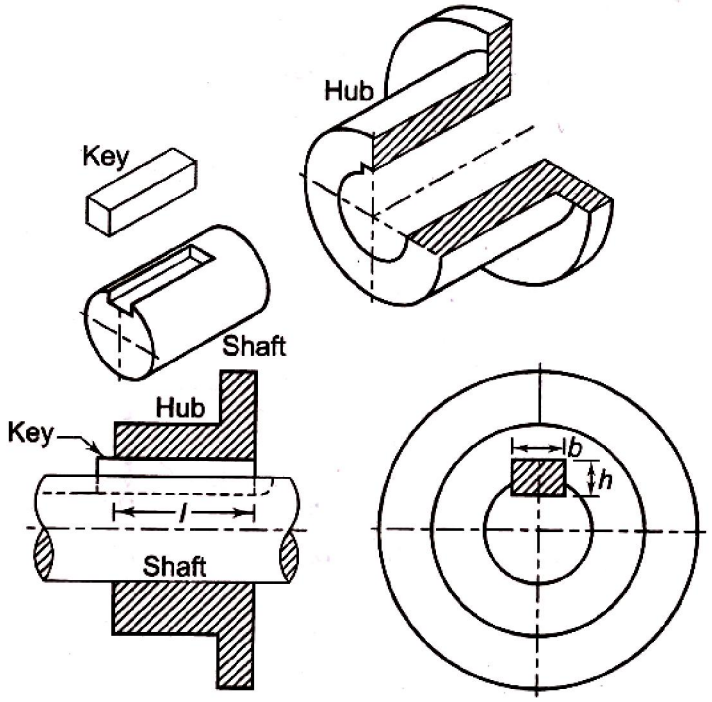


Fig.1. Key joint

A recess or slot machined either on the shaft or in the hub to accommodate the key is called keyway. The keyway is usually cut by vertical or horizontal milling cutter. The keyway results in stress concentration in the shaft and the part becomes weak. This is the main drawback of keyed joint. Keys are made of plain carbon steels like 45C8 or 50C8 in order to withstand shear and compressive stresses resulting from transmission of torque. According to IS, steel of tensile strength not less than 600 N/mm^2 shall be used as the material for the key.

Types of Keys

Many types of key are available and there are number of standards that specify the dimensions of the key. There are different ways to classify the keys. Some of them are as follows:

(i) Saddle key and sunk key;
(ii) Square key and flat key;
(iii) Taper key and parallel key; and
(iv) Key with and without Gib-head.

In addition, there are special types of keys such as Woodruff key, Kennedy key or feather key. The selection of the type of key for a given application depends upon following factors:

(i) power to be transmitted;
(ii) tightness of fit;
(iii) stability of connection; and
(iv) cost.

In this chapter, only popular types of key are discussed.

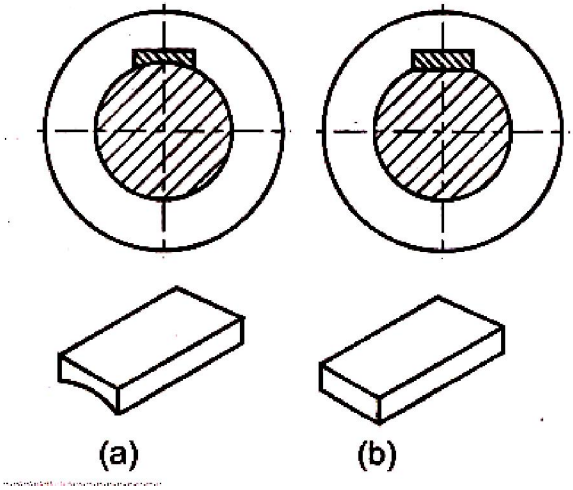


Fig.2. (c Hollow Saddle Key (b) Flat Saddle Key

Saddle keys

A saddle key is a key that fits in the keyway of the hub only. In this case there is no keyway on the shaft. There are two types of saddle key, namely, hollow and flat, as shown in Fig. 2.

A **hollow saddle** key has concave surface at the bottom to match the circular surface of the shaft.

A **flat saddle** key has flat surface at the bottom and it sits on the flat surface machined on the shaft.

In both types of saddle key, friction between shaft, key and hub prevents relative motion between the shaft and the hub. The power is transmitted by means of friction. Therefore, saddle keys are suitable for light duty or low power transmission as compared with sunk keys. The resistance to slip in case of flat key is slightly more than that of hollow key with concave surface. Therefore, flat saddle key is slightly superior to hollow saddle key as far as power transmitting capacity is concerned.

Saddle key requires keyway only on the hub. Therefore, cost of saddle key joint is less than that of sunk key joint. This is the main advantage of the saddle key. The disadvantage of saddle key is its low power transmitting capacity. Saddle key is liable to slip around the shaft when subjected to heavy torque. Therefore, it cannot be used in medium and heavy duty application.

Sunk keys

A sunk key is a key, in which half the thickness of the key fits into the keyway on the shaft and the remaining half in the keyway on the hub. Therefore, keyways are required both on the shaft as well as the hub of the mating element. This is a standard form of key and may be either of rectangular or square cross-section, as shown in Fig. 2. The standard dimensions of square and rectangular cross-section sunk keys are given in Table1. In sunk key, power is transmitted due to shear resistance of the key. The relative motion between the shaft and the hub is also prevented by the shear resistance of key. Therefore, sunk key is suitable for heavy duty application, since there is no possibility of the key to slip around the shaft. It is a positive drive. This is the main advantage of the sunk key over the saddle key. However, it is necessary to cut keyways both on the shaft and the hub. Therefore, the cost of sunk key joint is more than that of saddle key joint.

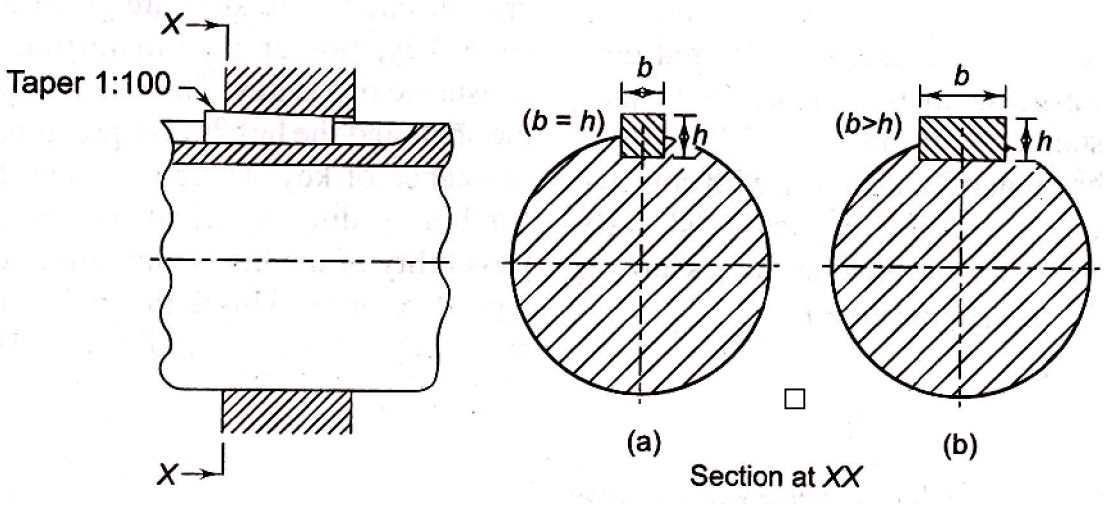


Fig.2. (a) Square Key (b) Flat Key

Table 1. Dimensions of square and rectangular sunk keys (in mm)

Shaft diameter		*Key size*	*Keyway depth*
Above	*Upto & including*	*b × h*	
6	8	2 × 2	1.2
8	10	3 × 3	1.8
10	12	4 × 4	2.5
12	17	5 × 5	3.0
17	22	6 × 6	3.5
22	30	8 × 7	4.0
30	38	10 × 8	5.0
38	44	12 × 8	5.0
44	50	14 × 9	5.5
50	58	16 × 10	6.0
58	65	18 × 11	7.0
65	75	20 × 12	7.5
75	85	22 × 14	9.0
85	95	25 × 14	9.0
95	110	28 × 16	10.0
110	130	32 × 18	11.0
130	150	36 × 20	12.0
150	170	40 × 22	13.0
170	200	45 × 25	15.0
200	230	50 × 28	17.0

Flat keys

Sunk keys with square or rectangular cross- section are widely used in practice. Sunk key with rectangular cross-section is called flat key. The flat key has more stability as compared with square key. Flat keys are more suitable for machine tool applications, where additional stability of the connection is desirable.

Square keys

Square keys are used in general industrial machinery. Selecting the square key without stress analysis, following rule of thumb may be used in industries. “The industrial practice is to use a square key with sides equal to one-quarter of the shaft diameter and length at least 1.5 times the shaft diameter.”

or, $b = h = d/4$ and $l = 1.5d$

where,

b = width of key (mm)

h = height or thickness of key (mm)

l = length of key (mm)

d = diameter of shaft (mm)

For flat key, the thumb-rule dimensions are as follows:

$b = d/4$, $h = 2b/3 = d/6$, $l = 1.5d$

Parallel key

A parallel key is a sunk key which is uniform in width as well as height throughout the length of the key. The parallel sunk key may be of rectangular or square section uniform in width and thickness throughout. It may be noted that a parallel key is a taper-less and is used where the pulley, gear or other mating piece is required to slide along the shaft.

Taper key

A taper key is uniform in width but tapered in height. The standard taper is1 in 100. The bottom surface of the key is straight and the top surface is given a taper. The taper is provided for following two reasons:

(i) When the key is inserted in the keyways of shaft and the hub and pressed by means of hammer, it becomes tight due to wedge action. This insures tightness of joint in operating conditions and prevents loosening of the parts.

(ii) Due to taper, it is easy to remove the key and dismantle the joint.

The taper of key is on one side. Machining taper on two sides of key is more difficult than making taper on one side. Also, there is no specific advantage of taper on two sides.

Gib-head keys

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. 3 (*a*) and its use is shown in Fig. 3 (*b*).

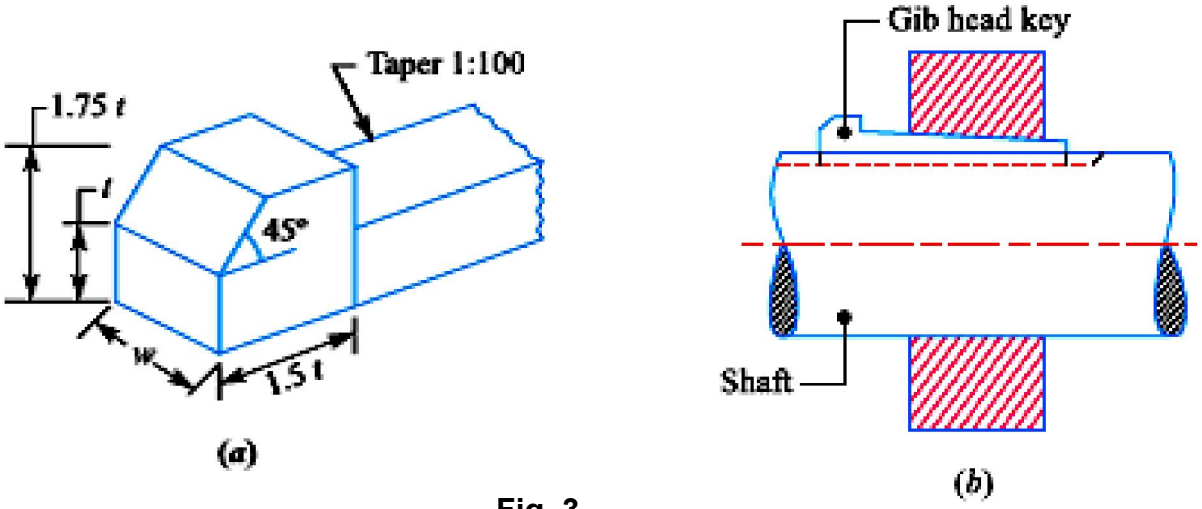


Fig. 3

Feather key

A feather key is a parallel key that is fixed either to the shaft or to the hub and that permits relative axial movement between them. . Figure 4 shows a feather key, which is fixed to the shaft by means of two cap screws, having countersunk-heads. The feather key transmits the torque and at the same time permits some axial movement of the hub. Feather keys are used where the parts mounted on the shaft are required to slide along the shaft such as clutches or gear shifting devices. It is an alternative to splined connection.

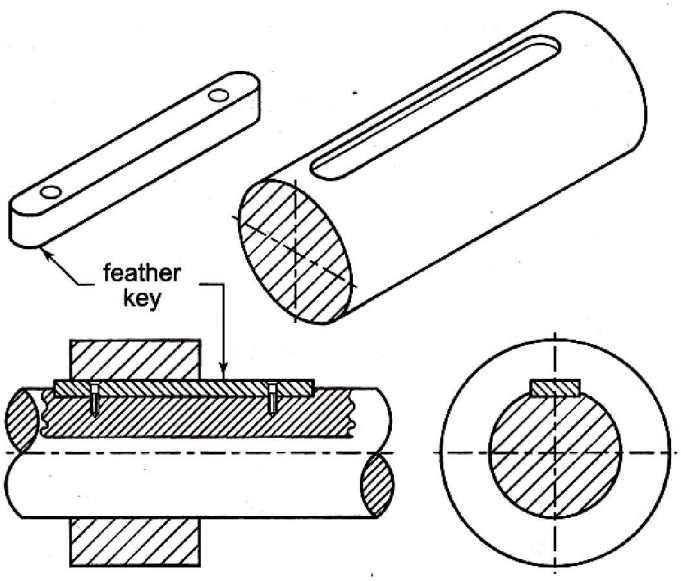


Fig. 4. Feather Key

Woodruff key

A Woodruff key is a sunk key, in the form of an almost semi-circular disk of uniform thickness as shown in Fig. 5. The keyway in the shaft is in the form of a semi-circular recess with the same curvature as that of the key. The bottom portion of the Woodruff key fits into circular keyway in the shaft. The keyway in the hub is made in usual manner.

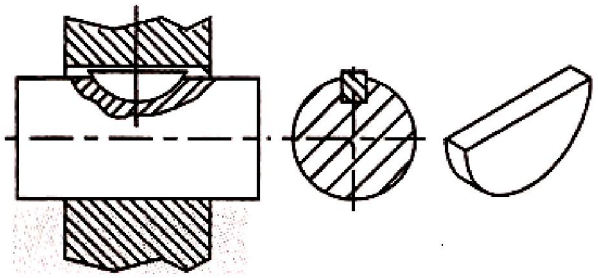


Fig. 5

The advantages of Woodruff key are as follows:
(i) The Woodruff key can be used on tapered shaft because it can be aligned by slight rotation in the seat.
(ii) The extra depth of key in the shaft prevents its tendency to slip over the shaft.

The disadvantages of Woodruff key are as follows:
(i)The extra depth of keyway in the shaft increase stress concentration and reduces its strength.
(ii) The key does not permit axial movement between the shaft and the hub.
Woodruff keys are used on tapered shafts in machine tools and automobiles.

3. Design of square and flat keys

Although there are many types of key, only square and flat keys are extensively used in practice. Therefore, the discussion in this chapter is restricted to square and flat keys. A square key is a particular type of flat key, in which the height is equal to the width of the cross-section. Therefore, for the purpose of analysis, a flat key is considered.
The forces acting on a flat key, with width as b and height as h, are shown in Fig. 6. The transmission of torque from the shaft to the hub results in two equal and opposite forces denoted by P. The torque M_t is transmitted by means of a force P acting on the left surface AC of the key. The equal and opposite force F, acting on the right surface DB of the key is reaction of the hub on the key. It is observed that force P on left surface AC and its equal and opposite reaction P on right surface DB are not in the same plane. Therefore, forces P'(P' = P) act as resisting couple preventing the key to roll in the keyway.

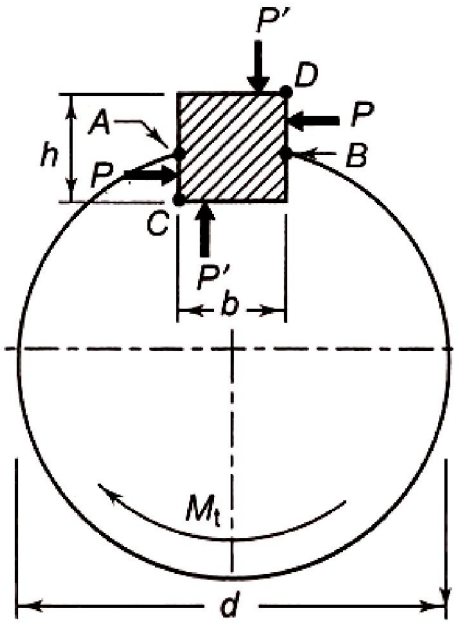


Fig. 6

The exact location of force P on surface AC is unknown. In order to simplify the analysis, it is assumed that the force P is tangential to the shaft diameter.
Therefore,

$$P = \frac{M_t}{(d/2)} = \frac{2M_t}{d} \qquad \text{...(a)}$$

where, M_t = transmitted torque (N-mm)
d = shaft diameter (mm)
P = force on key (N)

The design of square or flat key is based on two criteria, viz, failure due to shear stresses and failure due to compressive stresses. The shear failure will occur in plane AB. It is illustrated in Fig. 7 (a). The shear stress τ in the plane AB is given by,

$$\tau = \frac{P}{Area\ of\ plane\ AB} = \frac{P}{bl} \qquad \text{...(b)}$$

where, b = width of key (mm)
l = length of key (mm)
From (a) and (b),

$$\tau = \frac{2M_t}{dbl} \qquad \text{...(i)}$$

The failure due to compressive stresses will occur on surfaces AC or DB. The crushing area between shaft and key is shown in Fig. 7 (b).
It is assumed that,

$$\overline{AC} \cong \overline{BD} \cong h/2$$

where, h = height of key (mm)
The compressive stress σ_c in the key is given by,

$$\sigma_c = \frac{P}{Area\ of\ surface\ AC} = \frac{P}{(h/2)l} = \frac{2p}{hl} \qquad \text{...(c)}$$

From (a) and (c), $\sigma_c = \frac{2M_t}{dhl}$ ...(ii)

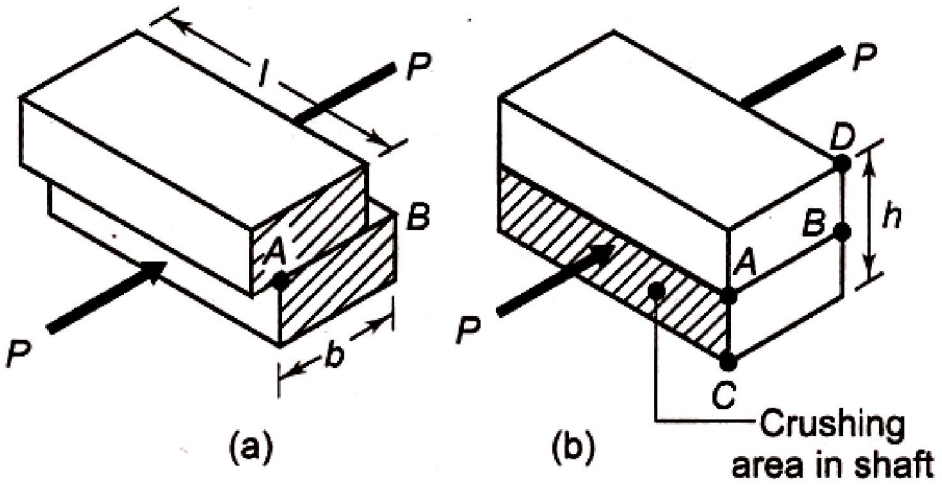


Fig. 7.Failure of Key (a) Shear Failure (b) Crushing Failure

Equations (i) and (ii) are stress equations of flat key.
For square key,

h = b

Substituting above relationship in Eqs (i) and (ii),

$$\tau = \frac{2M_t}{dbl} \quad \text{...(iii)}$$

$$\sigma_c = \frac{4M_t}{dbl} \quad \text{...(iv)}$$

From (iii) and (iv),

$$\sigma_c = 2\tau$$

Therefore, the compressive stress induced in a square key due to the transmitted torque is twice of shear stress.

Examples

1. It is required to design a square key for fixing a gear on a shaft of 25 mm diameter The shaft is transmitting 15 kW power at 720 rpm to the gear. The key is made of steel 50C4 (S_{yt} = 460 N/mm^2) and the factor of safety is 3. For key material the yield strength in compression can be assumed to be equal to the yield strength in tension. Determine the dimensions of the key

Solution: For the key material,

$$S_{yc} = S_{yt} = 460 \text{ N/mm}^2$$

$$\sigma_c = \frac{S_{yc}}{(fs)} = \frac{460}{3} = 153.33 \text{ N/mm}^2$$

According to maximum shear stress theory of failure,

$$S_{sy} = 0.5S_{yt} = 0.5(460) = 230 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{230}{3} = 76.67 \text{ N/mm}^2$$

The torque transmitted by the shaft is given by,

$$M_t = \frac{60 \times 10^6 (\text{kW})}{2\pi n} = \frac{60 \times 10^6 (15)}{2\pi (720)}$$

$$= 198\,943.68 \text{ N-mm}$$

The industrial practice is to use a square key with sides equal to one-quarter of the shaft diameter. Therefore,

$$b = h = \frac{d}{4} = \frac{25}{4} = 6.25 \quad \text{or} \quad 6 \text{ mm}$$

From equation (i)

$$l = \frac{2M_t}{\tau d b} = \frac{2(198\,943.68)}{(76.67)(25)(6)} = 34.60 \text{ mm} \qquad \text{(a)}$$

From equation (ii)

$$l = \frac{4M_t}{\sigma_c dh} = \frac{4(198\,943.68)}{(153.33)(25)(6)} = 34.60 \text{ mm} \qquad \text{(b)}$$

From (a) and (b), the length of the key should be 35 mm. The dimensions of the key are 6 x 6 x 35 mm.

2. The standard cross-section for a fiat key, which is fitted on a 50mm diameter shaft, is 16 x 10 mm. The key is transmitting 475 N-m torque from the shaft to the hub. The key is made of commercial steel ($S_{yt} = S_{yc} = 230$ N/mm^2), Determine the length of the key, if the factor of safety is 3.

Solution : For the key material,

$$\sigma_c = \frac{S_{yc}}{(fs)} = \frac{230}{3} = 76.67 \text{ N/mm}^2$$

According to maximum shear stress theory of failure,

$$S_{sy} = 0.5S_{yt} = 0.5(230) = 115 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{115}{3} = 38.33 \text{ N/mm}^2$$

From equation (i)

$$l = \frac{2M_t}{\tau d b} = \frac{2(475 \times 10^3)}{(38.33)(50)(16)} = 30.98 \text{ mm} \qquad \text{(a)}$$

From equation (ii)

$$l = \frac{4M_t}{\sigma_c dh} = \frac{2(475 \times 10^3)}{(76.67)(50)(10)} = 49.56 \text{ mm} \qquad \text{(b)}$$

From (a) and (b), the length of the key should be 5 mm.

4. Splines

Splines are keys that are made integral with the shaft. They are used when there is a relative axial motion between the shaft and the hub. The gear shifting mechanism in automobile gearboxes requires such type of construction. Splines are cut on the shaft by milling and on the hub by broaching. A splined connection, with straight splines, is shown in Fig. 8. Following notations are used:

D = major diameter of splines (mm)
d = minor diameter of splines (mm)
l = length of hub (mm)
n = number of splines

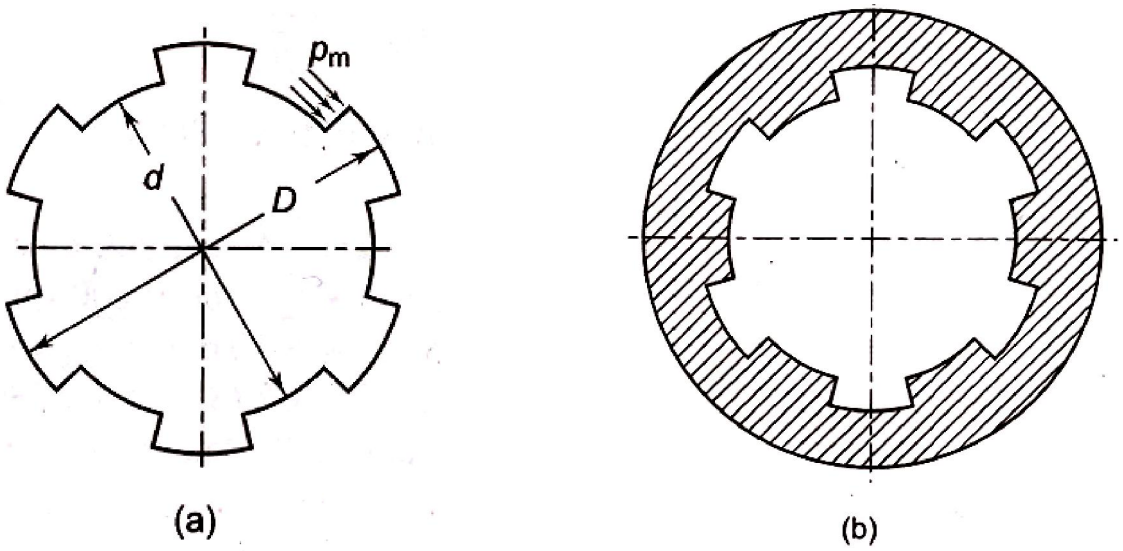


Fig. 8. Splines, (a) Shaft (b) Hub

The torque transmitting capacity of splines is given by,

$$M_t = p_m A R_m \qquad \text{(a)}$$

where,

M_t = transmitted torque (N-mm)

P_m = permissible pressure on spline (N/mm^2)

A = total area of splines (mm^2)

R_m = mean radius of splines (mm)

The area A is given by,

$$A = \frac{1}{2}(D - d)ln \qquad \text{(b)}$$

$$R_m = \frac{D + d}{4} \qquad \text{(c)}$$

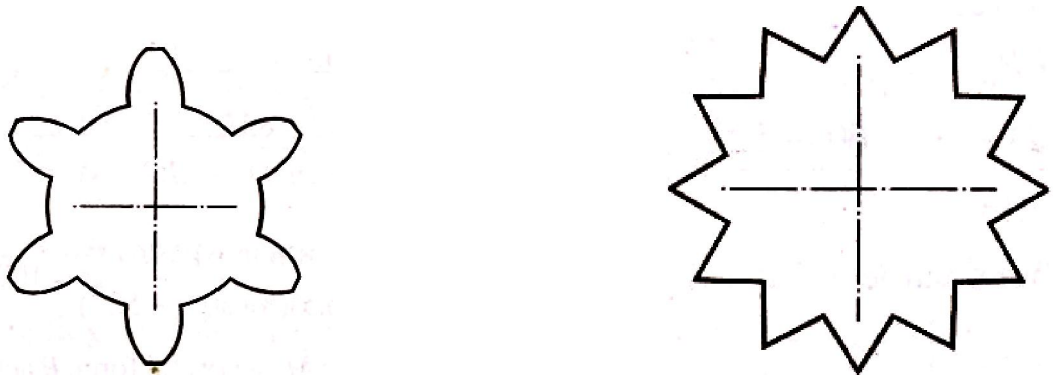
Substituting the above values in Eq. (a),

$$M_t = \frac{1}{8} p_m ln(D^2 - d^2) \qquad \text{(v)}$$

The permissible pressure on the splines is limited to 6.5 N/mm^2.
The above analysis and Fig. 8 refer to straight-sided splines. In addition, there are two other types of splines: involute splines and serrations, as shown in Fig. 9.

(i) Involute Splines: Involute splines are in the form of concentric external and internal gear teeth. They are stub teeth with pressure angle of 30^0. These splines are specified by module. Involute splines are more popular than straight splines due to greater strength relative to their size. Involute splines are self centering and tend to adjust to an even distribution of load. However, the cost of involute splines is more than straight-sided splines.

(ii) Serrations: Straight-sided serrations are used in applications where it is important to keep the overall size of the assembly as small as possible. They are used as interference joints. Serration joints are also used to obtain small angular relative adjustment between the joined members.



(a) (b)

Fig. 9. Types of Spline Profiles

Example

3. A standard splined connection 8 x 52 x 60 mm is used for the gear and the shaft assembly of a gearbox. The splines transmit 20 kW power at 300 rpm. The dimensions of the splines are as follows:

Major diameter = 60 mm
Minor diameter = 52 mm
Number of splines = 8

Permissible normal pressure on splines is 6,5 N/mm^2.
The coefficient of friction is 0.06. Calculate:

(i) The length of hub of the gear
(ii) The force required for shifting the gear.

Solution: The torque transmitted by the shaft is given by,

$$M_t = \frac{60 \times 10^6 (\text{kW})}{2\pi n}$$

$$= \frac{60 \times 10^6 (20)}{2\pi (300)} = 636\,619.76 \text{ N-mm}$$

From Eq. (v),

$$l = \frac{8M_t}{p_m n(D^2 - d^2)}$$

$$= \frac{8(636\,619.76)}{(6.5)(8)(60^2 - 52^2)} = 109.31 \text{ or } 110 \text{ mm}$$

Due to torque M_t, a normal force P acts on the splines. It is assumed that the force P acts at the mean radius of the splines. Therefore,

$$M_t = PR_m \qquad \text{(a)}$$

$$R_m = \frac{D + d}{4} = \frac{60 + 52}{4} = 28 \text{ mm}$$

Substituting above value in Eq. (a),

$$P = \frac{M_t}{R_m} = \frac{636\,619.76}{28} = 22736.42 \text{ N}$$

Friction force = μP = 0.06(22736.42) = 1364.19 N

The force required to shift the gear is equal and opposite of the friction force. Therefore, the force required to shift the gear is 1364.19 N.

5. Interference-Fit Joints

Introduction

Interference-fit joints depend for their operation on friction developed between the mating surfaces. The necessary normal pressure between the two surfaces is obtained due to elastic deformations undergone by the mating parts during assembly. To this end, an adequate amount of interference is created in the course of manufacture. By an interference is meant the negative difference in diameter between the hole in the hub (Fig. 1a) and the shaft (Fig. 1b). The frictional forces (F_{fr} in N) arising when the two parts are forced together (Fig. 1c) prevent the parts from relative displacement and enable the joint to resist both twisting and axial loads. In order for this to take place, the following condition must be satisfied

$$F_{fr} \geqslant K \sqrt{(10^3 \times 2T/d)^2 + F_a^2} \qquad \text{(i)}$$

where K = 1.5 to 2 is the grip safety factor; T is the transmitted torque (Nm); d is the mounted surface diameter (mm); and F_a is the axial force (N).

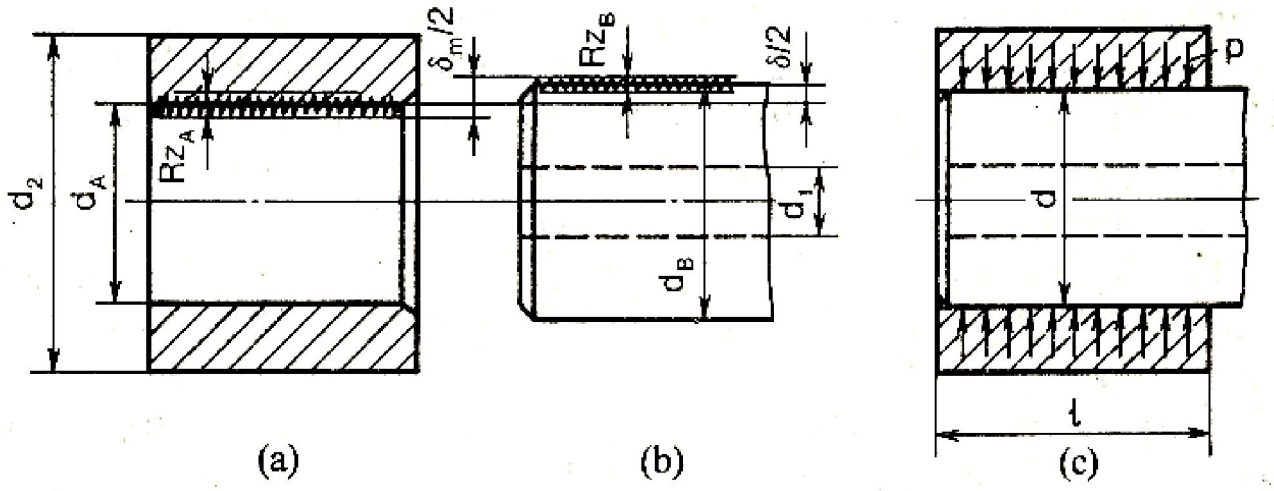


Fig. 1

Application of interference fit joints

Interference-fit joints are widely used to connect gear rings to hubs, hubs to shafts, carriers to planet axes and shafts, etc. They permit easy assembly, ensure good alignment, but give rise to appreciable stress concentration. Also, they are very sensitive to poor workmanship, the actual amount of interference being nearly always different from the theoretical value. This adversely affects the load-carrying capacity of the joint, so selective assembly is often required.

Types of interference fits

There are three types of interference fits in use by the industry today. They are:

(a) a shrink fit, where the outer member is heated, slipped onto the inner member, and the grip is secured by contraction of the outer member on cooling;

(b) an expansion fit, wherein the inner member is cooled, slipped into the outer member, and the grip is secured by expansion of the inner member when the assembly attains room temperature;

(c) a force or press fit, produced by pressing a shaft into a smaller hole. The necessary pressure is developed hydraulically.

Owing to their low cost and ease of manufacture, press fits are used most commonly. However, they suffer from grave shortcomings, in particular crushing and scouring of the contact surfaces and non-uniform deformation of the parts being joined. As compared with shrink or expansion fits, the latter factor may reduce the strength of a press fit assembly to two thirds of its normal value.

The above drawbacks are nonexistent in shrink fits (where the hub is heated to a temperature of 200 to 400°C) and expansion fits (where the shaft is cooled in dry ice (solid carbon dioxide at -80°C or in liquid air at about -200°C) which, however, have disadvantages of their own. In shrink fits, for example, the metal may undergo structural changes, scale may be raised, or the parts being fitted together may warp during heating. Being free from these shortcomings, expansion fits are finding ever increasing application with progress in cryogenic engineering.

To design an interference-fit joint means primarily to choose the type of it that would guarantee the transmission of a specified load.

When they are fitted together, the mating parts undergo diametral deformation equal to the amount of interference δ created during manufacture

$$\delta = d_B - d_A$$

where d_B and d_A are the diameters of the inner and outer members (shaft and hub hole, respectively) in mm.

The pressure p (MPa) exerted on the mating surfaces is given by the Lamé formula

$$p = \frac{\delta}{d\,(C_B/E_B + C_A/E_A)} \qquad \text{(ii)}$$

where d is the common diameter (mm); E_B and E_A are the elastic moduli of the inner and outer members (MPa); C_B and C_A are the Lamé constants for the inner and outer parts

$$C_B = (d^2 + d_1^2)/(d^2 - d_1^2) - \mu_B$$
$$C_A = (d_2^2 + d^2)/(d_2^2 - d^2) + \mu_A$$

μ_B and μ_A are the Poisson's ratios for the inner and outer members; d_1 is the diameter of the hole in the shaft; and d_2 is the outside diameter of the hub.
When the parts are fitted together, their surface irregularities are cut flush, so the effective interference δ becomes smaller than the measured value δ_m:

$$\delta = \delta_m - 1.2\,(Rz_B + Rz_A)\,10^3 \qquad \text{(iii)}$$

where Rz_B and Rz_A are the total heights of peak and valleys on the mating surfaces (μm).
In expansion and shrink fits, $\delta = \delta_m$
The pressure exerted on the mating surfaces should be such that the resulting frictional force exceeds the applied load

$$p = F_{fr}/(\pi dlf) \qquad \text{(iv)}$$

where l is the length of the mounting surface and f is the coefficient of friction (for steel members, it is taken as 0.06 to 0.22 with force fits and 0.08 to 0.19 with shrink and expansion fits).
In designing an interference-fit joint according to the condition in Eq. (i), we determine, as a point of departure, the required frictional force F_{fr}. Then, we find the necessary pressure p from Eq. (iv), the design interference δ from Eq. (ii), and the measured (actual) interference δ_m from Eq. (iii). Finally, we consult appropriate tables to choose a suitable standard fit for which

$$\delta_{min} \geq \delta_m$$

The shaft and hub making up an interference-fit joint should be checked for strength. Here, the hub is loaded to a greater extent than the shaft. The maximum safe amount of interference (mm) at which the outer part will not undergo plastic deformation is given by

$$\delta_{max} = P_{max}\delta/P$$

where $P_{max} = 0.5\sigma_{yA}\,[1 - (d/d_2)^2]$, being the yield strength of the hub material (MPa).
The force (N) necessary to make a press-fit joint is

$$F_P = \pi dl P_{max} f \qquad \text{(v)}$$

Example

Determine the parameters of a press-fit joint for a gear wheel and a shaft , using the following data: torque T = 5 x 10^2 N m; axial force F_a = 715 N; common diameter of the shaft and hub d = 50 mm; outside diameter of the hub d_2 = 80 mm; the shaft is solid, d_1 = 0; hub length l = 56 mm; yield strength of the shaft and hub material (steel 0.40C-Cr) σ_y = 500 MPa; maximum surface roughness $R_{a1} = R_{a2}$ = 1.25 μm; elastic modulus E = 2.2 x 10 MPa;
Poisson's ratio μ_B = 0.3; coefficient of friction f = 0.1.

Solution

(1) According to Eq. (i), the required frictional force is

$$F_{fr} = K\sqrt{(10^3 \times 2T/d)^2 + F_a^2} = 3 \times 10^4 \text{ N}$$

where K = 1.5 is the grip safety factor.

(2) Use Eq. (iv), to compute the required pressure

$$p = F_{fr}/(\pi dlf) = 34 \text{ MPa}$$

(3) Calculate the effective interference from Eq. (ii)

$$\delta = pd\,(C_B/E_B + C_A/E_A) = 0.025 \text{ mm}$$

Here, the Lamé constants are C_B = 1- μ_B= 0.7, and

$$C_A = \left(d_2^2 + d^2\right)/\left(d_2^2 - d^2\right) + \mu_A = 2.58$$

(4) Determine the measured interference, using Eq. (iii)

$$\delta_m = \delta + 1.2 \times 10^{-3}\,(Rz_B + Rz_A) = 0.040 \text{ mm}$$

Here, the height of the surface irregularities is $R_{z\,B} = R_{zA}$ = 5R_A = 6.25 μm.

(5) Enter standard tables with d = 50 mm to choose a standard fit (Ø50 H7/s6) having a minimum interference δ_{min} = 0.043 mm and a maximum interference δ_{max} = 0.084 mm.

(6) Check the mating parts for strength. The maximum pressure at which the hub will still be free from plastic deformation is

$$[p_{max}] = 0.5\sigma_{yA}\,[1 - (d/d_2)^2] = 152\ \text{MPa}$$

The respective maximum interference is

$$[\delta_{max}] = [p_{max}]\,\delta/p = 0.112\ \text{mm}$$

The hub strength is ensured because $\delta_{max} < |\delta_{max}|$

(7) The force necessary to make the press-fit joint is

$$F_p = \pi dlpf = 29\,900\ \text{N}$$