9.8 KEYS

A key can be defined as a machine element which is used to connect the transmission shaft to rotating machine elements like pulleys, gears, sprockets or flywheels. A keyed joint consisting of shaft, hub and key is illustrated in Fig. 9.16. 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 the 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, the key also prevents axial motion between two elements, except in case of feather key or splined connection.

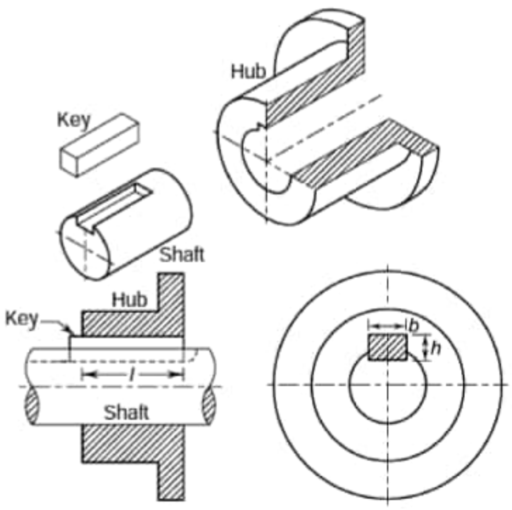

Fig. 9.16 *Key-joint*

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 a 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 a 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 Indian standards, steel of tensile strength not less than 600 N/mm^2 shall be used as the material for the key.

Many types of keys are available and there are a number of standards, which specify the dimensions of the key[3-6]. 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
(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 the 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.

9.9 SADDLE KEYS

A saddle key is a key which fits in the keyway of the hub only. In this case, there is no keyway on the shaft. There are two types of saddle keys, namely, *hollow* and *flat*, as shown in Fig. 9.17. A hollow saddle key has a concave surface at the bottom to match the circular surface of the shaft. A flat saddle

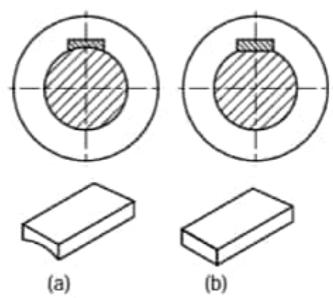

Fig. 9.17 *(a) Hollow Saddle Key (b) Flat Saddle Key*

key has a flat surface at the bottom and it sits on the flat surface machined on the shaft. In both types of saddle keys, friction between the shaft, key and

[3] IS 2048–1983: Specification for parallel keys and keyways.
[4] IS 2292–1974: Specification for taper keys and keyways.
[5] IS 2293–1974: Specification for Gib-head key and keyways.
[6] IS 2294–1980: Specification for Woodruff keys and keyways.

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 the saddle key joint is less than that of sunk key joint. This is the main advantage of the saddle key. The disadvantage of the 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 applications.

9.10 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. 9.18. The standard dimensions of square and rectangular cross-section sunk keys are given in Table 9.3 given on page 349. 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.

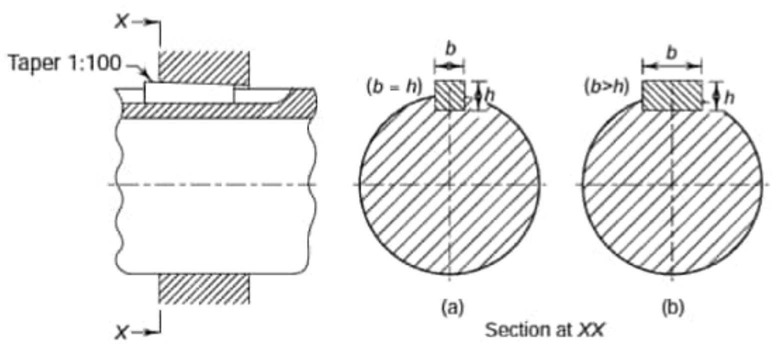

Fig. 9.18 *(a) Square Key (b) Flat Key*

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 the sunk key joint is more than that of the saddle key joint.

Sunk keys with square or rectangular cross-sections are widely used in practice. A sunk key with rectangular cross-section is called a *flat key*. The flat key has more stability as compared with square key. Square keys are used in general industrial machinery. Flat keys are more suitable for machine tool applications, where additional stability of the connection is desirable. While selecting the square key without stress analysis, the following rule of thumb may be used. *"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, $$b = h = \frac{d}{4}$$

and $$l = 1.5\, d$$

where,

b = width of key (mm)

h = height or thickness of key (mm)
l = length of key (mm)
d = diameter of shaft (mm)

Table 9.3 *Dimensions of square and rectangular sunk keys (in mm)*

Shaft diameter		*Key size*	*Keyway depth*
Above	*Up to and including*	$b \times h$	
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

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

$$b = \frac{d}{4}$$

$$h = \frac{2}{3}\,b = \frac{d}{6}$$

$$l = 1.5\,d$$

Sunk keys with square or rectangular cross-sections are classified into two groups, namely, parallel and taper keys. A *parallel key* is a sunk key which is uniform in width as well as height throughout the length of the key. A *taper key* is uniform in width but tapered in height. The standard taper is 1 in 100. The bottom surface of the key is straight and the top surface is given a taper. The taper is provided for the 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 the 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. Tapered keys are often provided with Gib-head to facilitate removal. The Gib-head taper key is shown in Fig. 9.19. The projection of Gib-head is hazardous in rotating parts. As compared with parallel key, taper key has the following advantages:

(i) The taper surface results in wedge action and increases frictional force and the tightness of the joint.

(ii) The taper surface facilitates easy removal of the key, particularly with Gib-head.

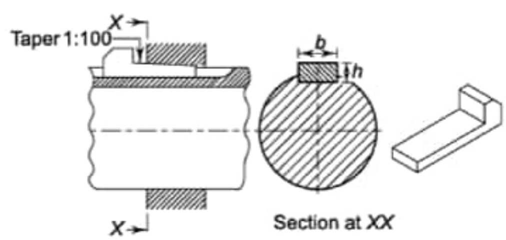

Fig. 9.19 *Gib-head Taper Key*

However, machining taper on the surface increases the cost.

9.11 FEATHER KEY

A feather key is a parallel key which is fixed either to the shaft or to the hub and which permits *relative axial movement between them*. The feather key is a particular type of sunk key with uniform width and height. There are number of methods to fix the key to the shaft or hub. Figure 9.20 shows a feather key, which is fixed to the shaft by means of two cap screws, having countersunk-heads. There is a clearance fit between the key and the keyway in the hub. Therefore, the hub is free to slide over the key. At the same time, there is no relative rotational movement between the shaft and the hub. Therefore, 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.

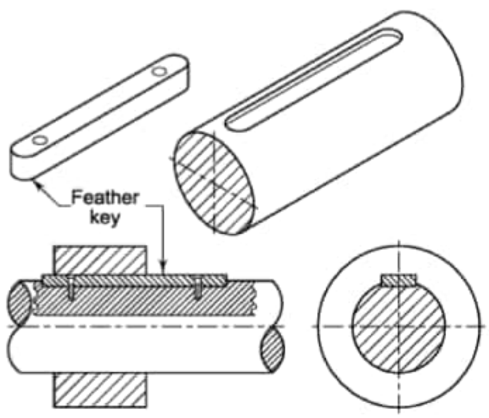

Fig. 9.20 *Feather Key*

9.12 WOODRUFF KEY

A Woodruff key is a sunk key in the form of an almost semicircular disk of uniform thickness as shown in Fig. 9.21. The keyway in the shaft is in the form of a semicircular recess with the same curvature as that of the key. The bottom portion of the Woodruff key fits into the circular keyway in the shaft. The keyway in the hub is made in the usual manner. The projecting part of Woodruff key fits in the keyway in the hub. Once placed in position, the Woodruff key tilts and aligns itself on the shaft. The advantages of Woodruff key are as follows:

(i) The Woodruff key can be used on tapered shaft because it can align 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.

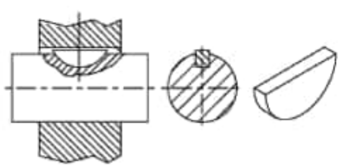

Fig. 9.21 *Woodruff Key*

9.13 DESIGN OF SQUARE AND FLAT KEYS

Although there are many types of keys, 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. 9.22. The

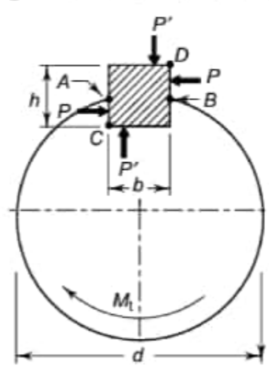

Fig. 9.22 *Forces Acting on Key*

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 P, acting on the right surface DB of the key is the reaction of the hub on the key. It is observed that the force P on left surface AC and its equal and opposite reaction P on the right surface DB are not in the same plane. Therefore, forces P^1 ($P^1 = P$) act as resisting couple preventing the key to roll in the keyway.

The exact location of the force P on the 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 stress and failure due to compressive stress. The shear failure will occur in the plane AB. It is illustrated in Fig. 9.23(a). The shear stress τ in the plane AB is given by,

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

where,

b = width of key (mm)
l = length of key (mm)

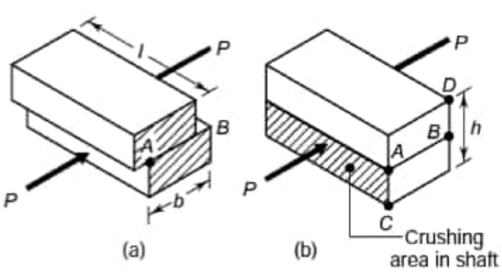

Fig. 9.23 *Failure of Key: (a) Shear Failure (b) Crushing Failure*

From (a) and (b),

$$\tau = \frac{2M_t}{dbl} \qquad (9.27)$$

The failure due to compressive stress will occur on surfaces AC or DB. The crushing area between shaft and key is shown in Fig. 9.23(b).

It is assumed that,

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

where,

h = height of key (mm)

The compressive stress σ_c in the key is given by,

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

From (a) and (c),

$$\sigma_c = \frac{4M_t}{dhl} \qquad (9.28)$$

Equations (9.27) and (9.28) are stress equations of flat key.

For square key,

$$h = b$$

Substituting the above relationship in Eqs (9.27) and (9.28),

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

$$\sigma_c = \frac{4M_t}{dbl} \qquad \text{(b)}$$

From (a) and (b),

$$\sigma_c = 2\tau \qquad (9.29)$$

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

Example 9.13 *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

Given kW = 15 $n = 720$ rpm $S_{yt} = 460$ N/mm^2
$(fs) = 3$ $d = 25$ mm

Step I *Permissible compressive and shear stresses*

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

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

According to maximum shear stress theory of failure,

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

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

Step II *Torque transmitted by the shaft*

$$M_t = \frac{60 \times 10^6 (\text{kW})}{2\pi n} = \frac{60 \times 10^6 (15)}{2\pi (720)}$$
$$= 198\,943.68 \text{ N-mm}$$

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

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

From Eq. (9.27),

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

From Eq. (9.28),

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

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

Example 9.14 *The standard cross-section for a flat key, which is fitted on a 50 mm diameter shaft, is 16 × 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

Given $M_t = 475$ N-m $S_{yt} = S_{yc} = 230$ N/mm^2
$(fs) = 3$ $d = 50$ mm $b = 16$ mm $h = 10$ mm

Step I *Permissible compressive and shear stresses*

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

According to maximum shear stress theory of failure,

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

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

Step II *Key length*
From Eq. (9.27),

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

From Eq. (9.28),

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

From (a) and (b), the length of the key should be 50 mm.

9.14 DESIGN OF KENNEDY KEY

The Kennedy key consists of two square keys as shown in Fig. 9.24. In this case, the hub is bored off the centre and the two keys force the hub and the shaft to a concentric position. Kennedy key is used for heavy duty applications. The analysis of the Kennedy key is similar to that of the flat key.

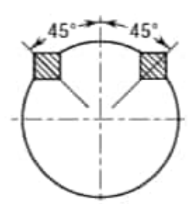

Fig. 9.24 *Kennedy Key*

It is based on two criteria, viz., failure due to shear stress and failure due to compressive stress. The forces acting on one of the two Kennedy keys are shown in Fig. 9.25. Since there are two keys, the torque transmitted by each key is one half of the total torque. The two equal and opposite forces P are due to the transmitted torque. The exact location of the force P is unknown. It is assumed to act tangential to the shaft diameter. Therefore,

$$\frac{M_t}{2} = P\left(\frac{d}{2}\right)$$

or

$$P = \frac{M_t}{d} \quad \text{(a)}$$

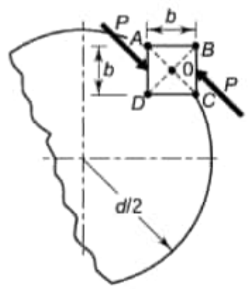

Fig. 9.25 *Forces acting on Kennedy Key*

The failure due to shear stress will occur in the plane AC. The area of the plane AC is $[\overline{AC} \times l]$ or $[\sqrt{2}bl]$. The shear stress is given by,

$$\tau = \frac{P}{\sqrt{2}bl} \quad \text{(b)}$$

From (a) and (b),

$$\tau = \frac{M_t}{\sqrt{2}dbl} \quad (9.30)$$

The compressive stress is given by,

$$\sigma_c = \frac{P}{(\text{Projected area})} = \frac{P}{[\overline{OB} \times l]} = \frac{P}{\left(\frac{b}{\sqrt{2}}\right) \times l} \quad \text{(c)}$$

From (a) and (c),

$$\sigma_c = \frac{\sqrt{2}M_t}{dbl} \quad (9.31)$$

where l is the length of the key.

Example 9.15 *A shaft, 40 mm in diameter, is transmitting 35 kW power at 300 rpm by means of Kennedy keys of 10 × 10 mm cross-section. The keys are made of steel 45C8 ($S_{yt} = S_{yc} = 380$ N/mm²) and the factor of safety is 3. Determine the required length of the keys.*

Solution

Given kW = 35 $n = 300$ rpm
$S_{yt} = S_{yc} = 380$ N/mm² $(fs) = 3$
$d = 40$ mm $b = h = 10$ mm

Step I *Permissible compressive and shear stresses*

$$\sigma_c = \frac{S_{yc}}{(fs)} = \frac{380}{3} = 126.67 \text{ N/mm}^2$$

According to distortion energy theory of failure,

$$S_{sy} = 0.577\, S_{yt} = 0.577\,(380) = 219.26 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{219.26}{3} = 73.09 \text{ N/mm}^2$$

Step II *Torque transmitted by shaft*

The torque transmitted by the shaft is given by,

$$M_t = \frac{60 \times 10^6 (\text{kW})}{2\pi n} = \frac{60 \times 10^6 (35)}{2\pi\,(300)}$$
$$= 1\ 114\ 084.6 \text{ N-mm}$$

Step III *Key length*

From Eq. (9.30),

$$l = \frac{M_t}{\sqrt{2}db\tau} = \frac{(1\ 114\ 084.6)}{\sqrt{2}(40)(10)(73.09)} = 26.95 \text{ mm}$$

From Eq. (9.31),

$$l = \frac{\sqrt{2}M_t}{db\sigma_c} = \frac{\sqrt{2}(1\ 114\ 084.6)}{(40)(10)(126.67)} = 31.10 \text{ mm}$$

Example 9.16 *The dimensions of a Woodruff key for a 30 mm diameter shaft are shown in Fig. 9.26(a). The shaft is transmitting 5 kW power*

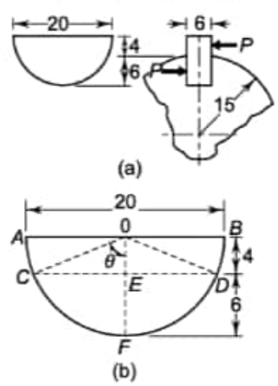

Fig. 9.26 *Woodruff Key*

at 300 rpm The key is made of steel 50C4 ($S_{yt} = S_{yc}$ = 460 N/mm²). Calculate the factor of safety used in design.

Solution

Given kW = 5 n = 300 rpm
$S_{yt} = S_{yc}$ = 460 N/mm² d = 30 mm

Step I *Torque transmitted by the shaft*
The torque transmitted by the shaft is given by,

$$M_t = \frac{60 \times 10^6 (\text{kW})}{2\pi n} = \frac{60 \times 10^6 (5)}{2\pi (300)}$$
$$= 159\ 154.94 \text{ N-mm}$$

Step II *Force induced by torque*
The torque is transmitted by two, equal and opposite, forces P. It is assumed that the force P acts at the shaft surface. Therefore,

$$P \times 15 = M_t$$
$$P = \frac{M_t}{15} = \frac{159\ 154.94}{15} = 10\ 610.33 \text{ N}$$

Step III *Key area*
Referring to Fig. 9.26(b),

$$\cos\theta = \frac{\overline{OE}}{\overline{OC}} = \frac{4}{10} \quad \text{or} \quad \theta = 66.42°$$
$$\overline{CD} = 2\,\overline{CE} = 2\,\overline{OC} \sin\theta$$
$$= 2(10) \sin (66.42°) = 18.33 \text{ mm}$$

Total area of the key

$$= \frac{1}{2}\left[\frac{\pi}{4}(20)^2\right] = 157.08 \text{ mm}^2$$

Area of sector $OCFD$

$$= \left(\frac{2\theta}{180}\right)(157.08) = \left(\frac{2(66.42)}{180}\right)(157.08)$$
$$= 115.93 \text{ mm}^2$$

$$\text{Area of } \Delta OCD = \frac{1}{2}\overline{CD} \times \overline{OE}$$
$$= \frac{1}{2}(18.33)(4) = 36.66 \text{ mm}^2$$

Area of key in contact with the shaft = 115.93 – 36.66 = 79.27 mm²

Area of key in contact with the hub = 157.08 – 79.27 = 77.81 mm²

Step IV *Factor of safety against compression failure*
The area of the key in contact with the hub is less and the compressive stress in the area is given by,

$$\sigma_c = \frac{P}{A} = \frac{10\ 610.33}{77.81} = 136.36 \text{ N/mm}^2$$
$$(fs) = \frac{S_{yc}}{\sigma_c} = \frac{460}{136.36} = 3.37 \qquad \text{(i)}$$

Step V *Factor of safety against shear failure*
The shear failure will occur in the plane CD, and its area is ($\overline{CD} \times 6$ or (18.33×6) mm² Therefore,

$$\tau = \frac{P}{A} = \frac{10\ 610.33}{(18.33 \times 6)} = 96.48 \text{ N/mm}^2$$
$$(fs) = \frac{S_{sy}}{\tau} = \frac{0.577\,S_{yt}}{\tau}$$
$$= \frac{0.577(460)}{96.48} = 2.75 \qquad \text{(ii)}$$

9.15 SPLINES

Splines are keys which 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. 9.27. The following notations are used:

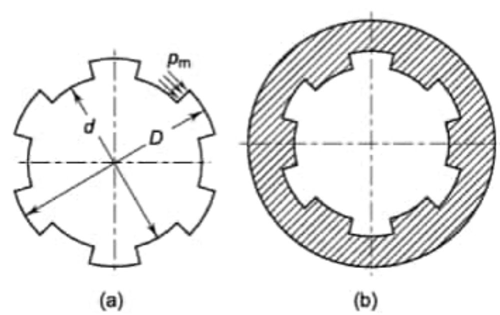

Fig. 9.27 *Splines: (a) Shaft (b) Hub*

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

The torque transmitting capacity of splines is given by,

$$M_t = p_m A R_m \tag{a}$$

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,

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

$$R_m = \frac{D+d}{4} \tag{c}$$

Substituting the above values in Eq. (a),

$$M_t = \frac{1}{8} p_m \, ln(D^2 - d^2) \tag{9.32}$$

The permissible pressure on the splines is limited to 6.5 N/mm^2.

The above analysis and Fig. 9.27 refer to straight-sided splines. In addition, there are two other types of splines, namely, involute splines and serrations as shown in Fig. 9.28.

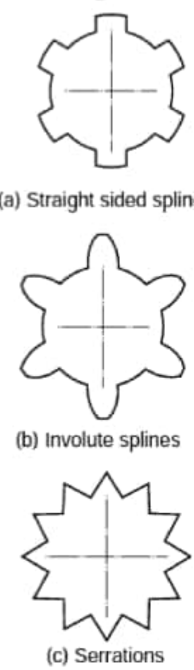

Fig. 9.28 *Types of Spline Profiles*

(i) Involute Splines Involute splines are in the form of concentric external and internal gear teeth. They are stub teeth with a pressure angle of 30°. 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.

Example 9.17 *A standard splined connection 8 × 52 × 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/mm². 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

Given kW = 20 n = 300 rpm p_m = 6.5 N/mm^2
For splines, D = 60 mm d = 52 mm n = 8
μ = 0.06

Step I *Torque transmitted by the shaft*

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

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

Step II *Length of hub*

From Eq. (9.32),

$$l = \frac{8M_t}{p_m n(D^2 - d^2)} = \frac{8(636\ 619.76)}{(6.5)(8)(60^2 - 52^2)}$$

$$= 109.31 \text{ or } 110 \text{ mm} \tag{i}$$

Step III *Force required to shift gear*

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,

$$M_t = P\,R_m \tag{a}$$

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

Substituting the above value in Eq. (a),

$$P = \frac{M_t}{R_m} = \frac{636\,619.76}{28} = 22\,736.42 \text{ N}$$

Friction force $= \mu P = 0.06(22\,736.42) = 1364.19$ 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.