2.3 SHAFT COUPLING

2.3.1 Introduction

Shafts are usually available up to 7 metres length due to inconvenience in transport. In order to have a greater length, it becomes necessary to join two or more pieces of the shaft by means of a coupling.

Shaft couplings are used in machinery for several purposes, the most common of which are the following :

1. To provide for the connection of shafts of units that are manufactured separately such as a motor and generator and to provide for disconnection for repairs or alternations.
2. To provide for misalignment of the shafts or to introduce mechanical flexibility.
3. To reduce the transmission of shock loads from one shaft to another.
4. To introduce protection against overloads.
5. It should have no projecting parts.

Requirements of a Good Shaft Coupling

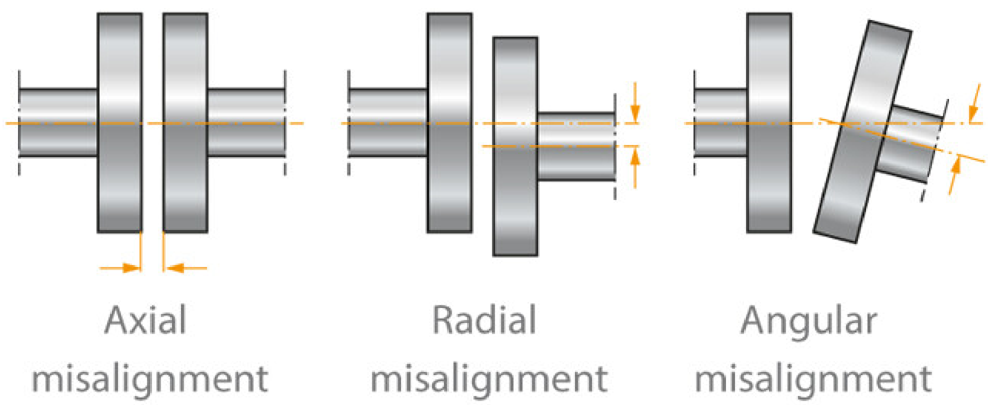
A good shaft coupling should have the following requirements :

1. It should be easy to connect or disconnect.
2. It should transmit the full power from one shaft to the other shaft without losses.
3. It should hold the shafts in perfect alignment.
4. It should reduce the transmission of shock loads from one shaft to another shaft.
5. It should have no projecting parts.

2.3.2 Types of Shaft Couplings

Shaft couplings are divided into two main groups as follows

1. ***Rigid coupling*.** It is used to connect two shafts which are perfectly aligned. Following types of rigid coupling are important from the subject point of view :
(*a*) Sleeve or muff coupling.



(*b*) Clamp or split-muff or compression coupling, and
(*c*) Flange coupling.
2. *Flexible coupling.* It is used to connect two shafts having both lateral and angular misalignment. Following types of flexible coupling are important from the subject point of view :

(*a*) Bushed pin type coupling,
(*b*) Universal coupling, and
(*c*) Oldham coupling.

We shall now discuss the above types of couplings, in detail,

2.3.4 Sleeve or Muff-coupling

It is the simplest type of rigid coupling, made of cast iron. It consists of a hollow cylinder whose inner diameter is the same as that of the shaft. It is fitted over the ends of the two shafts by means of a gib head key, as shown in Fig. 2.19. The power is transmitted from one shaft to the other shaft by means of a key and a sleeve. The usual proportions of a cast iron sleeve coupling are as follows :

Outer diameter of the sleeve, $D = 2d + 13$ mm
and length of the sleeve, $L = 3.5\ d$

where d is the diameter of the shaft.
In designing a sleeve or muff-coupling, the following procedure may be adopted.

1. *Design for sleeve*
The sleeve is designed by considering it as a hollow shaft

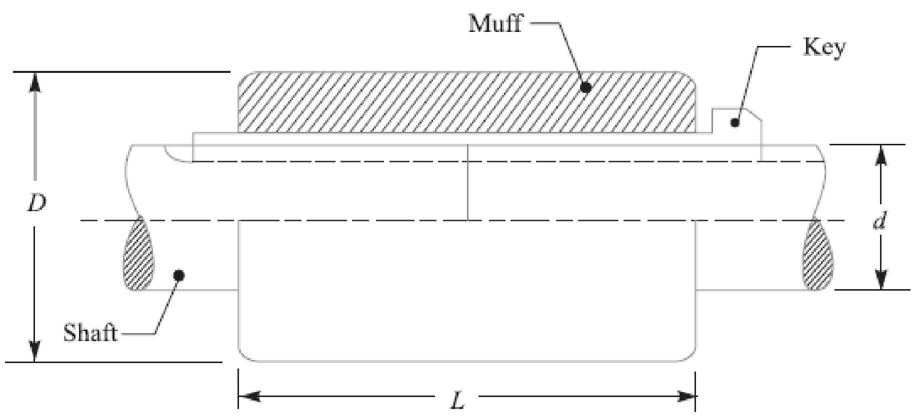


Fig. 2.19. Sleeve or muff coupling.

Let T = Torque to be transmitted by the coupling, and
τ_c = Permissible shear stress for the material of the sleeve which is cast rion.
The safe value of shear stress for cast iron may be taken as 14 MPa.
We know that torque transmitted by a hollow section,

$$T = \frac{\pi}{16} \times \tau_c \left(\frac{D^4 - d^4}{D}\right) = \frac{\pi}{16} \times \tau_c \times D^3\ (1 - k^4) \qquad \ldots (\because\ k = d/D)$$

From this expression, the induced shear stress in the sleeve may be checked.

2. *Design for key*
The key for the coupling may be designed in the similar way as discussed in Art. 2.2.9. The width and thickness of the coupling key is obtained from the proportions. The length of the coupling key is atleast equal to the length of the sleeve (*i.e.* 3.5 d). The coupling key is usually made into two parts so that the length of the key in each shaft,

$$l = \frac{L}{2} = \frac{3.5\ d}{2}$$

After fixing the length of key in each shaft, the induced shearing and crushing stresses may be checked. We know that torque transmitted,

$$T = l \times w \times \tau \times \frac{d}{2} \quad \text{... (Considering shearing of the key)}$$

$$= l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2} \quad \text{... (Considering crushing of the key)}$$

Example 2.13. *Design and make a neat dimensioned sketch of a muff coupling which is used to connect two steel shafts transmitting 40 kW at 350 r.p.m. The material for the shafts and key is plain carbon steel for which allowable shear and crushing stresses may be taken as 40 MPa and 80 MPa respectively. The material for the muff is cast iron for which the allowable shear stress may be assumed as 15 MPa.*

Given Data:

P = 40 kW = 40 × 10^3 W
N = 350 r.p.m.
τ_s = 40 MPa = 40 N/mm^2
σ_{cs} = 80 MPa = 80 N/mm^2
τ_c = 15 MPa = 15 N/mm^2

To Find:

The muff coupling is shown in Fig. 2.19. It is designed as discussed below :

1.*Design for shaft*

Let d = Diameter of the shaft.

We know that the torque transmitted by the shaft, key and muff,

$$T = \frac{P \times 60}{2\ \pi N} = \frac{40 \times 10^3 \times 60}{2\ \pi \times 350} = 1100 \text{ N-m}$$

$$= 1100 \times 10^3 \text{ N-mm}$$

We also know that the torque transmitted (T),

$$1100 \times 10^3 = \frac{\pi}{16} \times \tau_s \times d^3 = \frac{\pi}{16} \times 40 \times d^3 = 7.86\ d^3$$

$$\therefore \quad d^3 = 1100 \times 10^3 / 7.86 = 140 \times 10^3 \text{ or } d = 52 \text{ say } 55 \text{ mm } \textbf{Ans.}$$

2. *Design for sleeve*

We know that outer diameter of the muff,

D= 2d + 13 mm = 2 × 55 + 13 = 123 say 125 mm **Ans.**

and length of the muff,
L = 3.5 d = 3.5 × 55 = 192.5 say 195 mm **Ans.**

Let us now check the induced shear stress in the muff. Let τ_c be the induced shear stress in the muff which is made of cast iron. Since the muff is considered to be a hollow shaft, therefore the torque transmitted (T),

$$1100 \times 10^3 = \frac{\pi}{16} \times \tau_c \left(\frac{D^4 - d^4}{D}\right) = \frac{\pi}{16} \times \tau_c \left[\frac{(125)^4 - (55)^4}{125}\right]$$

$$= 370 \times 10^3 \tau_c$$

$$\therefore \quad \tau_c = 1100 \times 10^3 / 370 \times 10^3 = 2.97 \text{ N/mm}^2$$

Since the induced shear stress in the muff (cast iron) is less than the permissible shear stress of 15 N/mm2, therefore the design of muff is safe.

3. *Design for key*
From Table 2.2, we find that for a shaft of 55 mm diameter,
Width of key, w = 18 mm **Ans.**
Since the crushing stress for the key material is twice the shearing stress, therefore a square key may be used.

∴ Thickness of key, $t = w$ = 18 mm **Ans.**
We know that length of key in each shaft,
$l = L / 2 = 195 / 2 = 97.5$ mm **Ans.**
Let us now check the induced shear and crushing stresses in the key. First of all, let us consider shearing of the key. We know that torque transmitted (T),

$$1100 \times 10^3 = l \times w \times \tau_s \times \frac{d}{2} = 97.5 \times 18 \times \tau_s \times \frac{55}{2} = 48.2 \times 10^3 \tau_s$$

$$\therefore \quad \tau_s = 1100 \times 10^3 / 48.2 \times 10^3 = 22.8 \text{ N/mm}^2$$

Now considering crushing of the key. We know that torque transmitted (T),

$$1100 \times 10^3 = l \times \frac{t}{2} \times \sigma_{cs} \times \frac{d}{2} = 97.5 \times \frac{18}{2} \times \sigma_{cs} \times \frac{55}{2} = 24.1 \times 10^3 \sigma_{cs}$$

$$\therefore \quad \sigma_{cs} = 1100 \times 10^3 / 24.1 \times 10^3 = 45.6 \text{ N/mm}^2$$

Since the induced shear and crushing stresses are less than the permissible stresses, therefore the design of key is safe.

2.3.5 Clamp or Compression Coupling

It is also known as ***split muff coupling***. In this case, the muff or sleeve is made into two halves and are bolted together as shown in Fig. 2.20. The halves of the muff are made of cast iron. The shaft ends are made to abutt each other and a single key is fitted directly in the keyways of both the shafts. One-half of the muff is fixed from below and the other half is placed from above. Both the halves are held together by means of mild steel studs or bolts and nuts. The number of bolts may be two, four or six. The nuts are recessed into the bodies of the muff castings. This coupling may be used for heavy duty and moderate speeds. The advantage of this coupling is that the position of the shafts need not be changed for assembling or disassembling of the coupling. The usual proportions of the muff for the clamp or compression coupling are :
Diameter of the muff or sleeve, $D = 2d + 13$ mm
Length of the muff or sleeve, $L = 3.5\, d$
where d = Diameter of the shaft

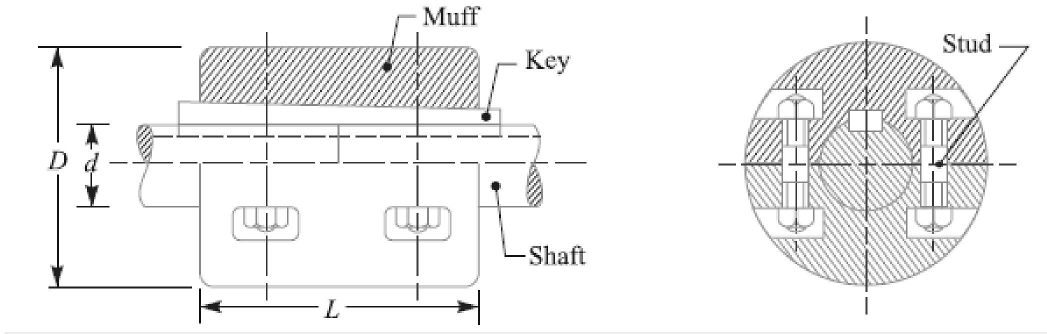


Fig. 2.20. Clamp or compression coupling.

In the clamp or compression coupling, the power is transmitted from one shaft to the other by means of key and the friction between the muff and shaft. In designing this type of coupling, the following procedure may be adopted.

1. *Design of muff and key*

The muff and key are designed in the similar way as discussed in muff coupling (Art. 13.14).

2. *Design of clamping bolts*

Let T = Torque transmited by the shaft,

d = Diameter of shaft,

d_b = Root or effective diameter of bolt,

n = Number of bolts,

σ_t = Permissible tensile stress for bolt material,

μ= Coefficient of friction between the muff and shaft, and

L = Length of muff.

We know that the force exerted by each bolt

$$= \mu \times \frac{\pi}{4}(d_b)^2 \sigma_t \times \frac{n}{2} \times \pi = \mu \times \frac{\pi^2}{8}(d_b)^2 \sigma_t \times n$$

and the torque that can be transmitted by the coupling,

$$T = F \times \frac{d}{2} = \mu \times \frac{\pi^2}{8}(d_b)^2 \sigma_t \times n \times \frac{d}{2} = \frac{\pi^2}{16} \times \mu\,(d_b)^2 \sigma_t \times n \times d$$

$$= \frac{\pi}{4}(d_b)^2 \sigma_t$$

∴ Force exerted by the bolts on each side of the shaft

$$= \frac{\pi}{4}(d_b)^2 \sigma_t \times \frac{n}{2}$$

Let p be the pressure on the shaft and the muff surface due to the force, then for uniform pressure distribution over the surface,

$$p = \frac{\text{Force}}{\text{Projected area}} = \frac{\frac{\pi}{4}(d_b)^2 \sigma_t \times \frac{n}{2}}{\frac{1}{2} L \times d}$$

∴ Frictional force between each shaft and muff,

$$F = \mu \times \text{pressure} \times \text{area} = \mu \times p \times \frac{1}{2} \times \pi d \times L$$

$$= \mu \times \frac{\frac{\pi}{4}(d_b)^2 \sigma_t \times \frac{n}{2}}{\frac{1}{2} L \times d} \times \frac{1}{2}\pi d \times L$$

From this relation, the root diameter of the bolt (*db*) may be evaluated.

Example 2.14. *Design a clamp coupling to transmit 30 kW at 100 r.p.m. The allowable shear stress for the shaft and key is 40 MPa and the number of bolts connecting the two halves are six. The permissible tensile stress for the bolts is 70 MPa. The coefficient of friction between the muff and the shaft surface may be taken as 0.3.*
Given :

$P = 30 \text{ kW} = 30 \times 10^3 \text{ W}$
$N = 100$ r.p.m.
$\tau = 40 \text{ MPa} = 40 \text{ N/mm}^2$
$n = 6$
$\sigma_t = 70 \text{ MPa} = 70 \text{ N/mm}^2$
$\mu = 0.3$

1. *Design for shaft*
Let d = Diameter of shaft.
We know that the torque transmitted by the shaft,

$$T = \frac{P \times 60}{2\pi N} = \frac{30 \times 10^3 \times 60}{2\pi \times 100} = 2865 \text{ N-m} = 2865 \times 10^3 \text{ N-mm}$$

We also know that the torque transmitted by the shaft (T),

$$2865 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 40 \times d^3 = 7.86\, d^3$$

$$\therefore \quad d^3 = 2865 \times 10^3 / 7.86 = 365 \times 10^3 \text{ or } d = 71.4 \text{ say } 75 \text{ mm } \textbf{Ans.}$$

2. *Design for muff*
We know that diameter of muff,
$D = 2d + 13 \text{ mm} = 2 \times 75 + 13 = 163$ say 165 mm **Ans.**
and total length of the muff,
$L = 3.5\, d = 3.5 \times 75 = 262.5$ mm **Ans.**

3. *Design for key*
The width and thickness of the key for a shaft diameter of 75 mm (from Table 13.1) are as follows :
Width of key, w = 22 mm **Ans.**
Thickness of key, t = 14 mm **Ans.**
and length of key = Total length of muff = 262.5 mm **Ans.**

4. *Design for bolts*
Let db = Root or core diameter of bolt.
We know that the torque transmitted (T),

$$2865 \times 10^3 = \frac{\pi^2}{16} \times \mu (d_b)^2 \sigma_t \times n \times d = \frac{\pi^2}{16} \times 0.3\, (d_b)^2 \; 70 \times 6 \times 75 = 5830 (d_b)^2$$

$$\therefore \quad (d_b)^2 = 2865 \times 10^3 / 5830 = 492 \quad \text{or} \quad d_b = 22.2 \text{ mm}$$

From Table **11.1,** we find that the standard core diameter of the bolt for coarse series is 23.32 mm and the nominal diameter of the bolt is 27 mm (M 27). **Ans.**

2.3.6 Flange Coupling

A flange coupling usually applies to a coupling having two separate cast iron flanges. Each flange is mounted on the shaft end and keyed to it. The faces are turned up at right angle to the axis of the shaft. One of the flange has a projected portion and the other flange has a corresponding recess.

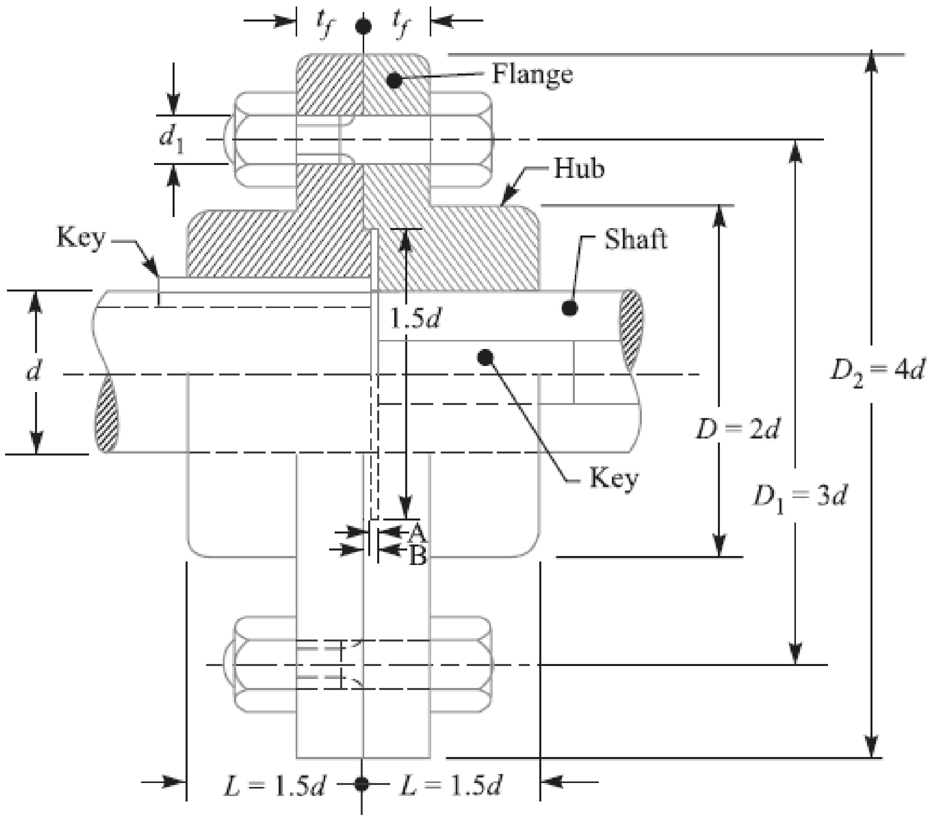


Fig. 2.21. Unprotected type flange coupling.

This helps to bring the shafts into line and to maintain alignment. The two flanges are coupled together by means of bolts and nuts. The flange coupling is adopted to heavy loads and hence it is used on large shafting. The flange couplings are of the following three types :

1. *Unprotected type flange coupling.* In an unprotected type flange coupling, as shown in Fig. 2.21, each shaft is keyed to the boss of a flange with a counter sunk key and the flanges are coupled together by means of bolts. Generally, three, four or six bolts are used. The keys are staggered at right angle along the circumference of the shafts in order to divide the weakening effect caused by keyways. The usual proportions for an unprotected type cast iron flange couplings, as shown in
Fig. 2.21, are as follows :

If *d* is the diameter of the shaft or inner diameter of the hub, then Outside diameter of hub,

$D = 2\,d$

Length of hub, $L = 1.5\,d$

Pitch circle diameter of bolts,

$D1 = 3d$

Outside diameter of flange,

$D2 = D1 + (D1 – D) = 2\,D1 – D = 4\,d$

Thickness of flange, $tf = 0.5\,d$

Number of bolts = 3, for *d* upto 40 mm

= 4, for *d* upto 100 mm

= 6, for *d* upto 180 mm

2. *Protected type flange coupling.* In a protected type flange coupling, as shown in Fig. 2.22, the protruding bolts and nuts are protected by flanges on the two halves of the coupling, in order to avoid danger to the workman.

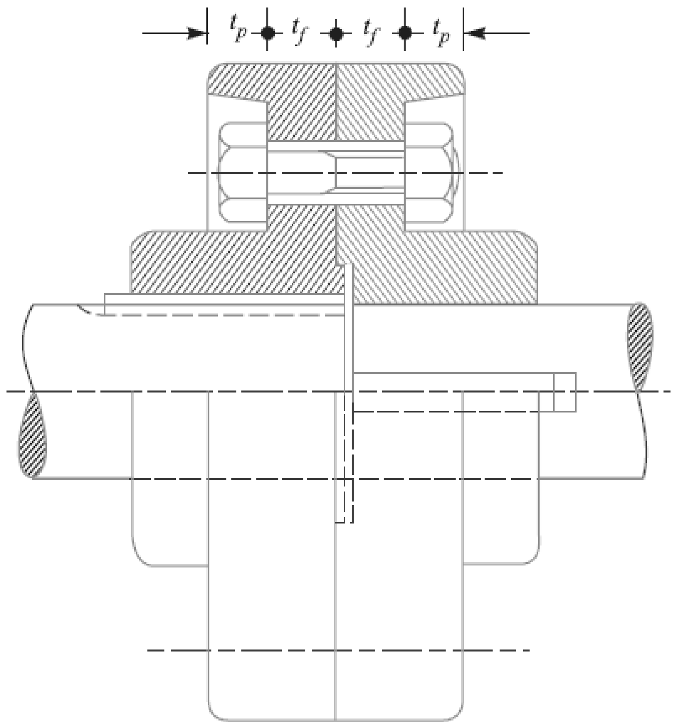


Fig. 2.22. Protective type flange coupling.

3. *Marine type flange coupling*. In a marine type flange coupling, the flanges are forged integral with the shafts as shown in Fig. 2.23. The flanges are held together by means of tapered headless bolts, numbering from four to twelve depending upon the diameter of shaft.
The number of bolts may be choosen from the following table.

Table 2.3. Number of bolts for marine type flange coupling.
[According to IS : 3653 – 1966 (Reaffirmed 1990)]

Shaft diameter (mm)	35 to 55	56 to 150	151 to 230	231 to 390	Above 390
No. of bolts	4	6	8	10	12

The other proportions for the marine type flange coupling are taken as follows :
Thickness of flange = d / 3
Taper of bolt = 1 in 20 to 1 in 40
Pitch circle diameter of bolts, $D1 = 1.6\ d$
Outside diameter of flange, $D2 = 2.2\ d$

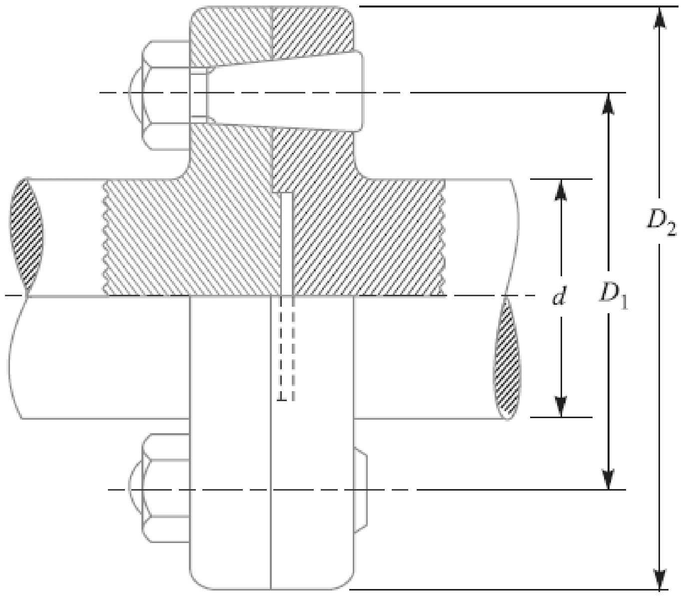


Fig. 2.23. Marine type flange coupling.

2.3.7 Design of Flange Coupling

Consider a flange coupling as shown in Fig. 2.22 and Fig. 2.23.

Let d = Diameter of shaft or inner diameter of hub,
D = Outer diameter of hub,
$d1$ = Nominal or outside diameter of bolt,
$D1$ = Diameter of bolt circle,
n = Number of bolts,
t_f = Thickness of flange,
τ_s, τ_b and τ_k = Allowable shear stress for shaft, bolt and key material respectively
τ_c = Allowable shear stress for the flange material *i.e.* cast iron,
σ_{cb}, and σ_{ck} = Allowable crushing stress for bolt and key material respectively.
The flange coupling is designed as discussed below :

1. *Design for hub*

The hub is designed by considering it as a hollow shaft, transmitting the same torque (T) as that of a solid shaft.

$$T = \frac{\pi}{16} \times \tau_c \left(\frac{D^4 - d^4}{D} \right)$$

The outer diameter of hub is usually taken as twice the diameter of shaft. Therefore from the above relation, the induced shearing stress in the hub may be checked. The length of hub (L) is taken as 1.5 d.

2. *Design for key*

The key is designed with usual proportions and then checked for shearing and crushing stresses.The material of key is usually the same as that of shaft. The length of key is taken equal to the length of hub.**3.**

3. *Design for flange*

The flange at the junction of the hub is under shear while transmitting the torque. Therefore, the troque transmitted,

T = Circumference of hub × Thickness of flange × Shear stress of flange × Radius of hub

$$= \pi D \times t_f \times \tau_c \times \frac{D}{2} = \frac{\pi D^2}{2} \times \tau_c \times t_f$$

The thickness of flange is usually taken as half the diameter of shaft. Therefore from the above relation, the induced shearing stress in the flange may be checked.

$$\text{Load on each bolt} = \frac{\pi}{4}(d_1)^2 \tau_b$$

$\therefore$ Total load on all the bolts

$$= \frac{\pi}{4}(d_1)^2 \tau_b \times n$$

and torque transmitted,

$$T = \frac{\pi}{4}(d_1)^2 \tau_b \times n \times \frac{D_1}{2}$$

From this equation, the diameter of bolt (d_1) may be obtained. Now the diameter of bolt may be checked in crushing.

We know that area resisting crushing of all the bolts

$$= n \times d_1 \times t_f$$

and crushing strength of all the bolts

$$= (n \times d_1 \times t_f)\,\sigma_{cb}$$

$\therefore$ Torque,

$$T = (n \times d_1 \times t_f \times \sigma_{cb})\,\frac{D_1}{2}$$

From this equation, the induced crushing stress in the bolts may be checked.

Example 2.15. *Design a cast iron protective type flange coupling to transmit 15 kW at 900 r.p.m. from an electric motor to a compressor. The service factor may be assumed as 1.35. The following permissible stresses may be used :*
Shear stress for shaft, bolt and key material = 40 MPa
Crushing stress for bolt and key = 80 MPa
Shear stress for cast iron = 8 MPa
Draw a neat sketch of the coupling.

Given : $P = 15$ kW $= 15 \times 10^3$ W
$N = 900$ r.p.m.
Service factor = 1.35
$\tau_s = \tau_b = \tau_k = 40$ MPa = 40 N/mm^2
$\sigma_{cb} = \sigma_{ck} = 80$ MPa = 80 N/mm^2
$\tau_c = 8$ MPa = 8 N/mm^2
The protective type flange coupling is designed as discussed below :

1. *Design for hub*

First of all, let us find the diameter of the shaft (d). We know that the torque transmitted by the shaft,

$$T = \frac{P \times 60}{2\pi N} = \frac{15 \times 10^3 \times 60}{2\pi \times 900} = 159.13 \text{ N-m}$$

Since the service factor is 1.35, therefore the maximum torque transmitted by the shaft,

$$T_{max} = 1.35 \times 159.13 = 215 \text{ N-m} = 215 \times 10^3 \text{ N-mm}$$

We know that the torque transmitted by the shaft (T),

$$215 \times 10^3 = \frac{\pi}{16} \times \tau_s \times d^3 = \frac{\pi}{16} \times 40 \times d^3 = 7.86\, d^3$$

$\therefore$ $d^3 = 215 \times 10^3 / 7.86 = 27.4 \times 10^3$ or $d = 30.1$ say 35 mm **Ans.**

We know that outer diameter of the hub,

$$D = 2d = 2 \times 35 = 70 \text{ mm } \textbf{Ans.}$$

and length of hub, $L = 1.5\, d = 1.5 \times 35 = 52.5$ mm **Ans.**

Let us now check the induced shear stress for the hub material which is cast iron. Considering the hub as a hollow shaft. We know that the maximum torque transmitted (T_{max}).

$$215 \times 10^3 = \frac{\pi}{16} \times \tau_c \left[\frac{D^4 - d^4}{D}\right] = \frac{\pi}{16} \times \tau_c \left[\frac{(70)^4 - (35)^4}{70}\right] = 63\,147\, \tau_c$$

$\therefore$ $\tau_c = 215 \times 10^3 / 63\,147 = 3.4 \text{ N/mm}^2 = 3.4$ MPa

Since the induced shear stress for the hub material (*i.e.* cast iron) is less than the permissible value of 8 MPa, therefore the design of hub is safe.

2. *Design for key*

Since the crushing stress for the key material is twice its shear stress (*i.e.* $\sigma_{ck} = 2\tau_k$), therefore a square key may be used. From Table 2.2, we find that for a shaft of 35 mm diameter,

Width of key, w = 12 mm **Ans.**

and thickness of key, $t = w$ = 12 mm **Ans.**

The length of key (l) is taken equal to the length of hub.

$\therefore$ $l = L$ = 52.5 mm **Ans.**

Let us now check the induced stresses in the key by considering it in shearing and crushing. Considering the key in shearing. We know that the maximum torque transmitted (*Tmax*),

$$215 \times 10^3 = l \times w \times \tau_k \times \frac{d}{2} = 52.5 \times 12 \times \tau_k \times \frac{35}{2} = 11\,025\, \tau_k$$

$\therefore$ $\tau_k = 215 \times 10^3 / 11\,025 = 19.5 \text{ N/mm}^2 = 19.5$ MPa

Considering the key in crushing. We know that the maximum torque transmitted (T_{max}),

$$215 \times 10^3 = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2} = 52.5 \times \frac{12}{2} \times \sigma_{ck} \times \frac{35}{2} = 5512.5\, \sigma_{ck}$$

$\therefore$ $\sigma_{ck} = 215 \times 10^3 / 5512.5 = 39 \text{ N/mm}^2 = 39$ MPa

Since the induced shear and crushing stresses in the key are less than the permissible stresses, therefore the design for key is safe.

3. *Design for flange*

The thickness of flange (*tf*) is taken as 0.5 *d*.

$\therefore$ $t_f = 0.5\, d = 0.5 \times 35 = 17.5$ mm **Ans.**

Let us now check the induced shearing stress in the flange by considering the flange at the junction of the hub in shear.

We know that the maximum torque transmitted (*Tmax*),

$$215 \times 10^3 = \frac{\pi D^2}{2} \times \tau_c \times t_f = \frac{\pi (70)^2}{2} \times \tau_c \times 17.5 = 134\,713\, \tau_c$$

$\therefore$ $\tau_c = 215 \times 10^3 / 134\,713 = 1.6 \text{ N/mm}^2 = 1.6$ MPa

Since the induced shear stress in the flange is less than 8 MPa, therefore the design of flange is safe.

4. *Design for bolts*

Let $d1$ = Nominal diameter of bolts.

Since the diameter of the shaft is 35 mm, therefore let us take the number of bolts,

$n = 3$

and pitch circle diameter of bolts,

$D1 = 3d = 3 \times 35 = 105$ mm

The bolts are subjected to shear stress due to the torque transmitted. We know that the maximum torque transmitted ($Tmax$),

$$215 \times 10^3 = \frac{\pi}{4}(d_1)^2 \ \tau_b \times n \times \frac{D_1}{2} = \frac{\pi}{4}(d_1)^2 \ 40 \times 3 \times \frac{105}{2} = 4950\ (d_1)^2$$

$$(d_1)^2 = 215 \times 10^3/4950 = 43.43 \text{ or } d_1 = 6.6 \text{ mm}$$

Assuming coarse threads, the nearest standard size of bolt is M 8. **Ans.**

Other proportions of the flange are taken as follows :

Outer diameter of the flange,

$D2 = 4\ d = 4 \times 35 = 140$ mm **Ans.**

Thickness of the protective circumferential flange,

$tp = 0.25\ d = 0.25 \times 35 = 8.75$ say 10 mm **Ans.**

Example 2.16. *Design a rigid flange coupling to transmit a torque of 250 N-m between two coaxial shafts. The shaft is made of alloy steel, flanges out of cast iron and bolts out of steel. Four bolts are used to couple the flanges. The shafts are keyed to the flange hub. The permissible stresses are given below:*

Shear stress on shaft =100 MPa

Bearing or crushing stress on shaft =250 MPa

Shear stress on keys =100 MPa

Bearing stress on keys =250 MPa

Shearing stress on cast iron =200 MPa

Shear stress on bolts =100 MPa

After designing the various elements, make a neat sketch of the assembly indicating the important dimensions. The stresses developed in the various members may be checked if thumb rules are used for fixing the dimensions.

Given :

$T = 250$ N-m $= 250 \times 10^3$ N-mm

$n = 4$; $\tau_s = 100$ MPa $= 100$ N/mm^2

$\sigma_{cs} = 250$ MPa $= 250$ N/mm^2

$\tau_k = 100$ MPa $= 100$ N/mm^2

$\sigma_{ck} = 250$ MPa $= 250$ N/mm^2

$\tau_c = 200$ MPa $= 200$ N/mm^2

$\tau_b = 100$ MPa $= 100$ N/mm^2

To Find : *Design a rigid flange coupling*

Design for hub

First of all, let us find the diameter of the shaft (d). We know that the torque transmitted by the shaft (T),

$$250 \times 10^3 = \frac{\pi}{16} \times \tau_s \times d^3 = \frac{\pi}{16} \times 100 \times d^3 = 19.64\ d^3$$

∴ d_3 = 250 × 103 / 19.64 = 12 729 or d = 23.35 say 25 mm **Ans.**

We know that the outer diameter of the hub,

$D = 2d = 2 \times 25 = 50$ mm

and length of hub, $L = 1.5 d = 1.5 \times 25 = 37.5$ mm

Let us now check the induced shear stress in the hub by considering it as a hollow shaft. We know that the torque transmitted (T),

$$250 \times 10^3 = \frac{\pi}{16} \times \tau_c \left(\frac{D^4 - d^4}{D} \right) = \frac{\pi}{16} \times \tau_c \left[\frac{(50)^4 - (25)^4}{50} \right] = 23\,013\ \tau_c$$

$$\therefore \quad \tau_c = 250 \times 10^3 / 23\,013 = 10.86 \text{ N/mm}^2 = 10.86 \text{ MPa}$$

Since the induced shear stress for the hub material (*i.e.* cast iron) is less than 200 MPa, therefore the design for hub is safe.

2. *Design for key*

From Table 2.2, we find that the proportions of key for a 25 mm diameter shaft are :

Width of key, w = 10 mm **Ans.**

and thickness of key, t = 8 mm **Ans.**

The length of key (l) is taken equal to the length of hub,

∴ $l = L = 37.5$ mm **Ans.**

Let us now check the induced shear and crushing stresses in the key. Considering the key in shearing. We know that the torque transmitted (T),

$$250 \times 10^3 = l \times w \times \tau_k \times \frac{d}{2} = 37.5 \times 10 \times \tau_k \times \frac{25}{2} = 4688\ \tau_k$$

$$\therefore \quad \tau_k = 250 \times 10^3 / 4688 = 53.3 \text{ N/mm}^2 = 53.3 \text{ MPa}$$

Considering the key in crushing. We know that the torque transmitted (T),

$$250 \times 10^3 = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2} = 37.5 \times \frac{8}{2} \times \sigma_{ck} \times \frac{25}{2} = 1875\ \sigma_{ck}$$

$$\therefore \quad \sigma_{ck} = 250 \times 10^3 / 1875 = 133.3 \text{ N/mm}^2 = 133.3 \text{ MPa}$$

Since the induced shear and crushing stresses in the key are less than the given stresses, therefore the design of key is safe.

3. *Design for flange*

The thickness of the flange (tf) is taken as 0.5 d.

∴ $t_f = 0.5 d = 0.5 \times 25 = 12.5$ mm **Ans.**

Let us now check the induced shear stress in the flange by considering the flange at the junction of the hub in shear. We know that the torque transmitted (T),

$$250 \times 10^3 = \frac{\pi D^2}{2} \times \tau_c \times t_f = \frac{\pi (50)^2}{2} \times \tau_c \times 12.5 = 49\,094\ \tau_c$$

$$\tau_c = 250 \times 10^3 / 49\,094 = 5.1 \text{ N/mm}^2 = 5.1 \text{ MPa}$$

Since the induced shear stress in the flange of cast iron is less than 200 MPa, therefore design of flange is safe.

4. *Design for bolts*

Let $d1$ = Nominal diameter of bolts.

We know that the pitch circle diameter of bolts,

∴ $D1 = 3d = 3 \times 25 = 75$ mm **Ans.**

The bolts are subjected to shear stress due to the torque transmitted. We know that torque transmitted (T),

$$250 \times 10^3 = \frac{\pi}{4}(d_1)^2 \tau_b \times n \times \frac{D_1}{2} = \frac{\pi}{4}(d_1)^2 100 \times 4 \times \frac{75}{2} = 11\ 780\ (d_1)^2$$

∴ $(d1)^2 = 250 \times 10^3 / 11\ 780 = 21.22$ or $d1 = 4.6$ mm

Assuming coarse threads, the nearest standard size of the bolt is M 6. **Ans.**

Other proportions of the flange are taken as follows :

Outer diameter of the flange,

$D2 = 4\ d = 4 \times 25 = 100$ mm **Ans.**

Thickness of the protective circumferential flange,

$t_p = 0.25\ d = 0.25 \times 25 = 6.25$ mm **Ans.**

2.3.8 Flexible Coupling

flexible coupling is used to join the abutting ends of shaftswhen they are not in exact alignment. In the case of a direct coupled drive from a prime mover to an electric generator, we should have four bearings at a comparatively close distance. In such a case and in many others, as in a direct electric drive from an electric motor to a machine tool, a flexible coupling is used so as to permit an axial misalignemnt of the shaft without undue absorption of the power which the shaft are transmitting. Following are the different types of flexible couplings :

1. Bushed pin flexible coupling, **2.** Oldham's coupling, and **3.** Universal coupling.

2.3.9 Bushed-pin Flexible Coupling

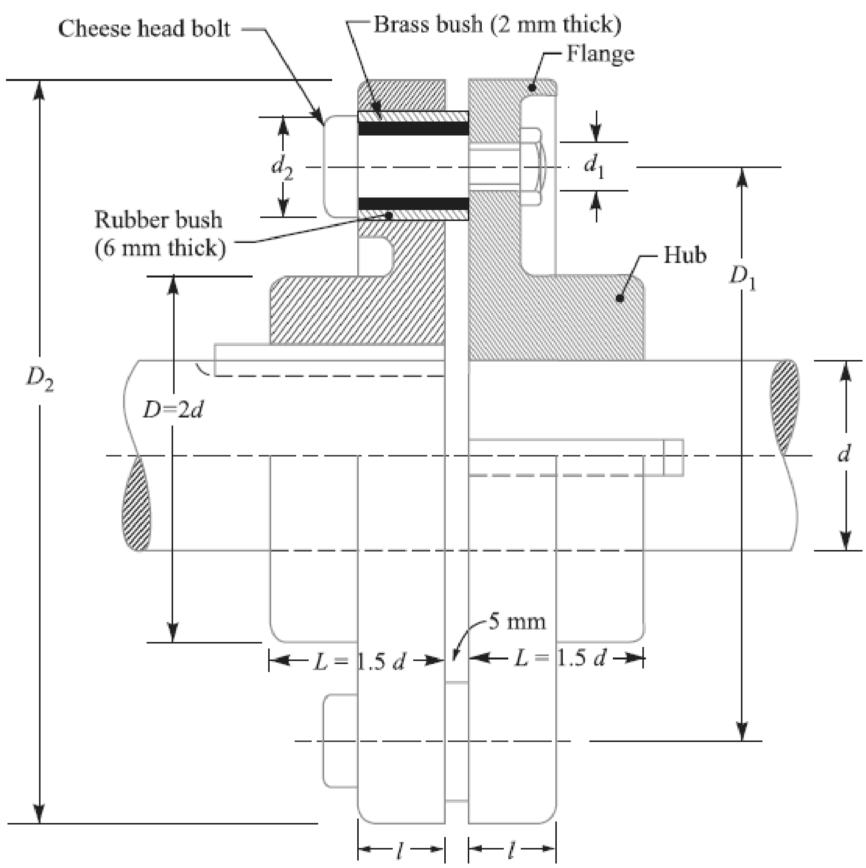


Fig. 2.24. Bushed-pin flexible coupling.

A bushed-pin flexible coupling, as shown in Fig. 2.24, is a modification of the rigid type of flange coupling. The coupling bolts are known as pins. The rubber or leather bushes are used over the pins. The two halves of the coupling are dissimilar in construction. A clearance of 5 mm is left between the face of the two halves of the coupling. There is no rigid connection between them and the drive takes place through the medium of the compressible rubber or leather bushes.

In designing the bushed-pin flexible coupling, the proportions of the rigid type flange coupling are modified. The main modification is to reduce the bearing pressure on the rubber or leather bushes and it should not exceed 0.5 N/mm2. In order to keep the low bearing pressure, the pitch circle diameter and the pin size is increased.

Let l = Length of bush in the flange,

$d2$ = Diameter of bush,

p_b = Bearing pressure on the bush or pin,

n = Number of pins, and

$D1$ = Diameter of pitch circle of the pins.

We know that bearing load acting on each pin,

$W = p_b \times d_2 \times l$

∴ Total bearing load on the bush or pins

$= W \times n = p_b \times d_2 \times l \times n$

and the torque transmitted by the coupling,

$$T = W \times n \left(\frac{D_1}{2}\right) = p_b \times d_2 \times l \times n \left(\frac{D_1}{2}\right)$$

The threaded portion of the pin in the right hand flange should be a tapping fit in the coupling hole to avoid bending stresses.

The threaded length of the pin should be as small as possible so that the direct shear stress can be taken by the unthreaded neck.

Direct shear stress due to pure torsion in the coupling halves,

$$\tau = \frac{W}{\frac{\pi}{4}(d_1)^2}$$

Since the pin and the rubber or leather bush is not rigidly held in the left hand flange, therefore the tangential load (W) at the enlarged portion will exert a bending action on the pin as shown in Fig. 2.25. The bush portion of the pin acts as a cantilever beam of length l. Assuming a uniform distribution of the load W along the bush, the maximum bending moment on the pin,

$$M = W\left(\frac{l}{2} + 5 \text{ mm}\right)$$

We know that bending stress,

$$\sigma = \frac{M}{Z} = \frac{W\left(\frac{l}{2} + 5 \text{ mm}\right)}{\frac{\pi}{32}(d_1)^3}$$

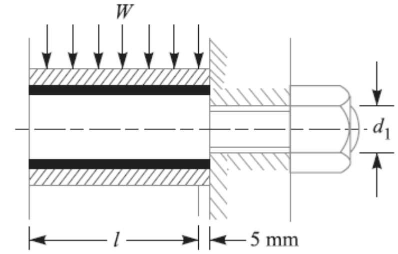


Fig. 2.25.

Since the pin is subjected to bending and shear stresses, therefore the design must be checked either for the maximum principal stress or maximum shear stress by the following relations :

Maximum principal stress

$$= \frac{1}{2}\left[\sigma + \sqrt{\sigma^2 + 4\tau^2}\right]$$

and the maximum shear stress on the pin

$$= \frac{1}{2}\sqrt{\sigma^2 + 4\tau^2}$$

The value of maximum principal stress varies from 28 to 42 MPa.

Example 2.17. *Design a bushed-pin type of flexible coupling to connect a pump shaft to a motor shaft transmitting 32 kW at 960 r.p.m. The overall torque is 20 percent more than mean torque.*
The material properties are as follows :
(*a*) *The allowable shear and crushing stress for shaft and key material is 40 MPa and 80 MPa respectively.*
(*b*) *The allowable shear stress for cast iron is 15 MPa.*
(*c*) *The allowable bearing pressure for rubber bush is 0.8 N/mm*^{2}.
(*d*) *The material of the pin is same as that of shaft and key.*
Draw neat sketch of the coupling.

Given : P = 32 kW = 32 × 10^3 W
N = 960 r.p.m.
$T_{max} = 1.2\ T_{mean}$
$\tau_s = \tau_k$ = 40 MPa= 40 N/mm2
$\sigma_{cs} = \sigma_{ck}$ = 80 MPa = 80 N/mm2
τ_c = 15 MPa = 15 N/mm2
P_b = 0.8 N/mm2

The bushed-pin flexible coupling is designed as discussed below :
1. *Design for pins and rubber bush*
First of all, let us find the diameter of the shaft (d). We know that the mean torque transmitted by the shaft,

$$T_{mean} = \frac{P \times 60}{2\pi N} = \frac{32 \times 10^3 \times 60}{2\pi \times 960} = 318.3 \text{ N-m}$$

and the maximum or overall torque transmitted,

$$T_{max} = 1.2\ T_{mean} = 1.2 \times 318.3 = 382 \text{ N-m} = 382 \times 10^3 \text{ N-mm}$$

We also know that the maximum torque transmitted by the shaft (T_{max}),

$$382 \times 10^3 = \frac{\pi}{16} \times \tau_s \times d^3 = \frac{\pi}{16} \times 40 \times d^3 = 7.86\ d^3$$

$$\therefore \quad d^3 = 382 \times 10^3 / 7.86 = 48.6 \times 10^3 \text{ or } d = 36.5 \text{ say } 40 \text{ mm}$$

We have discussed in rigid type of flange coupling that the number of bolts for 40 mm diameter shaft are 3. In the flexible coupling, we shall use the number of pins (n) as 6.

$$\therefore \quad \text{Diameter of pins, } d_1 = \frac{0.5\,d}{\sqrt{n}} = \frac{0.5 \times 40}{\sqrt{6}} = 8.2 \text{ mm}$$

In order to allow for the bending stress induced due to the compressibility of the rubber bush, the diameter of the pin ($d1$) may be taken as 20 mm. **Ans.**

The length of the pin of least diameter *i.e.* $d1$ = 20 mm is threaded and secured in the right hand coupling half by a standard nut and washer. The enlarged portion of the pin which is in the left hand coupling half is made of 24 mm diameter. On the enlarged portion, a brass bush of thickness 2 mm is pressed. A brass bush carries a rubber bush. Assume the thickness of rubber bush as 6 mm.

∴ Overall diameter of rubber bush,

$d2 = 24 + 2 \times 2 + 2 \times 6 = 40$ mm **Ans.**

and diameter of the pitch circle of the pins,

$D1 = 2\,d + d2 + 2 \times 6 = 2 \times 40 + 40 + 12 = 132$ mm **Ans.**

Let l = Length of the bush in the flange.

We know that the bearing load acting on each pin,

$W = p_b \times d2 \times l = 0.8 \times 40 \times l = 32\,l$ N

and the maximum torque transmitted by the coupling (T_{max}),

$$382 \times 10^3 = W \times n \times \frac{D_1}{2} = 32\,l \times 6 \times \frac{132}{2} = 12\,672\,l$$

$$\therefore \quad l = 382 \times 10^3 / 12\,672 = 30.1 \text{ say } 32 \text{ mm}$$

$$\text{and} \quad W = 32\,l = 32 \times 32 = 1024 \text{ N}$$

∴ Direct stress due to pure torsion in the coupling halves,

$$\tau = \frac{W}{\frac{\pi}{4}(d_1)^2} = \frac{1024}{\frac{\pi}{4}(20)^2} = 3.26 \text{ N/mm}^2$$

Since the pin and the rubber bush are not rigidly held in the left hand flange, therefore the tangential load (W) at the enlarged portion will exert a bending action on the pin. Assuming a uniform distribution of load (W) along the bush, the maximum bending moment on the pin,

$$M = W\left(\frac{l}{2} + 5\right) = 1024\left(\frac{32}{2} + 5\right) = 21\,504 \text{ N-mm}$$

and section modulus, $Z = \frac{\pi}{32}(d_1)^3 = \frac{\pi}{32}(20)^3 = 785.5 \text{ mm}^3$

We know that bending stress,

$$\sigma = \frac{M}{Z} = \frac{21\,504}{785.5} = 27.4 \text{ N/mm}^2$$

∴ Maximum principal stress

$$= \frac{1}{2}\left[\sigma + \sqrt{\sigma^2 + 4\tau^2}\right] = \frac{1}{2}\left[27.4 + \sqrt{(27.4)^2 + 4\,(3.26)^2}\right]$$

$$= 13.7 + 14.1 = 27.8 \text{ N/mm}^2$$

and maximum shear stress

$$= \frac{1}{2}\left[\sqrt{\sigma^2 + 4\tau^2}\right] = \frac{1}{2}\left[\sqrt{(27.4)^2 + 4\,(3.26)^2}\right] = 14.1 \text{ N/mm}^2$$

Since the maximum principal stress and maximum shear stress are within limits, therefore the design is safe.

2. *Design for hub*

We know that the outer diameter of the hub,

$D = 2\,d = 2 \times 40 = 80$ mm

and length of hub, $L = 1.5\,d = 1.5 \times 40 = 60$ mm

Let us now check the induced shear stress for the hub material which is cast iron. Considering the hub as a hollow shaft. We know that the maximum torque transmitted (T_{max}),

$$382 \times 10^3 = \frac{\pi}{16} \times \tau_c \left[\frac{D^4 - d^4}{D}\right] = \frac{\pi}{16} \times \tau_c \left[\frac{(80)^4 - (40)^4}{80}\right] = 94.26 \times 10^3\,\tau_c$$

$\tau_c = 382 \times 10^3 / 94.26 \times 10^3 = 4.05$ N/mm^2 = 4.05 MPa

Since the induced shear stress for the hub material (*i.e.* cast iron) is less than the permissible value of 15 MPa, therefore the design of hub is safe.

3. *Design for key*

Since the crushing stress for the key material is twice its shear stress (*i.e.* $\sigma_{ck} = 2\,\tau_k$), therefore a square key may be used. From Table 2.2, we find that for a shaft of 40 mm diameter,

Width of key, $w = 14$ mm **Ans.**

and thickness of key, $t = w = 14$ mm **Ans.**

The length of key (L) is taken equal to the length of hub, *i.e.*

$L = 1.5\,d = 1.5 \times 40 = 60$ mm

Let us now check the induced stresses in the key by considering it in shearing and crushing.

Considering the key in shearing. We know that the maximum torque transmitted (T_{max}),

$$382 \times 10^3 = L \times w \times \tau_k \times \frac{d}{2} = 60 \times 14 \times \tau_k \times \frac{40}{2} = 16\,800\,\tau_k$$

$$\therefore \quad \tau_k = 382 \times 10^3/16\,800 = 22.74 \text{ N/mm}^2 = 22.74 \text{ MPa}$$

Considering the key in crushing. We know that the maximum torque transmitted (T_{max}),

$$382 \times 10^3 = L \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2} = 60 \times \frac{14}{2} \times \sigma_{ck} \times \frac{40}{2} = 8400\,\sigma_{ck}$$

$$\therefore \quad \sigma_{ck} = 382 \times 10^3/8400 = 45.48 \text{ N/mm}^2 = 45.48 \text{ MPa}$$

Since the induced shear and crushing stress in the key are less than the permissible stresses of 40 MPa and 80 MPa respectively, therefore the design for key is safe.

4. *Design for flange*

The thickness of flange (t_f) is taken as 0.5 d.

$\therefore \quad t_f = 0.5\,d = 0.5 \times 40 = 20$ mm

Let us now check the induced shear stress in the flange by considering the flange at the junction of the hub in shear.

We know that the maximum torque transmitted (T_{max}),

$$382 \times 10^3 = \frac{\pi D^2}{2} \times \tau_c \times t_f = \frac{\pi (80)^2}{2} \times \tau_c \times 20 = 201 \times 10^3\,\tau_c$$

$$\therefore \quad \tau_c = 382 \times 10^3 / 201 \times 10^3 = 1.9 \text{ N/mm}^2 = 1.9 \text{ MPa}$$

Since the induced shear stress in the flange of cast iron is less than 15 MPa, therefore the design of flange is safe.

2.3.10 Oldham Coupling

It is used to join two shafts which have lateral mis-alignment. It consists of two flanges *A* and *B* with slots and a central floating part *E* with two tongues *T*1 and *T*2 at right angles as shown in Fig. 2.26. The central floating part is held by means of a pin passing through the flanges and the floating part. The tongue *T*1 fits into the slot of flange *A* and allows for 'to and fro' relative motion of the shafts, while the tongue *T*2 fits into the slot of the flange *B* and allows for vertical relative motion of the parts. The resultant of these two components of motion will accommodate lateral misalignment of the shaft as they rotate.

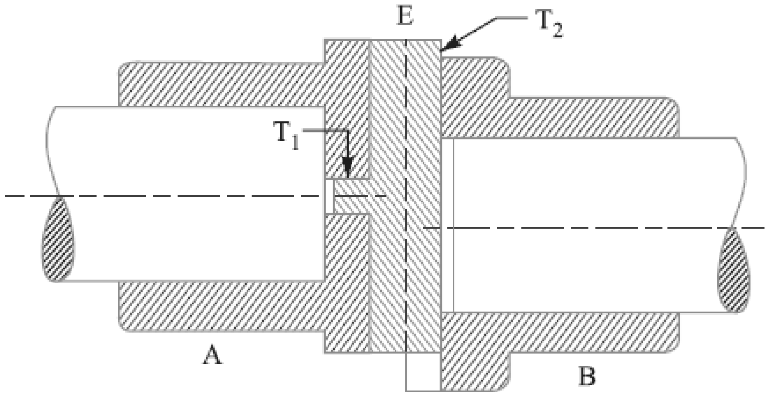


Fig. 2.26. Oldham coupling.

2.3.11 Universal (or Hooke's) Coupling

A universal or Hooke's coupling is used to connect two shafts whose axes intersect at a small angle. The inclination of the two shafts may be constant, but in actual practice, it varies when the motion is transmitted from one shaft to another. The main application of the universal or Hooke's coupling is found in the transmission from the gear box to the differential or back axle of the automobiles. In such a case, we use two Hooke's coupling, one at each end of the propeller shaft, connecting the gear box at one end and the differential on the other end. A Hooke's coupling is also used for transmission of power to different spindles of multiple drilling machine. It is used as a knee joint in milling machines. In designing a universal coupling, the shaft diameter and the pin diameter is obtained as discussed below. The other dimensions of the coupling are fixed by proportions as shown in Fig.

Let d = Diameter of shaft,

dp = Diameter of pin, and

$\tau$and τ_1 = Allowable shear stress for the material of the shaft and pin respectively.

We know that torque transmitted by the shafts,

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

From this relation, the diameter of shafts may be determined.

Since the pin is in double shear, therefore the torque transmitted.

$$T = 2 \times \frac{\pi}{4} (d_p)^2 \tau_1 \times d$$

From this relation, the diameter of pin may be determined.

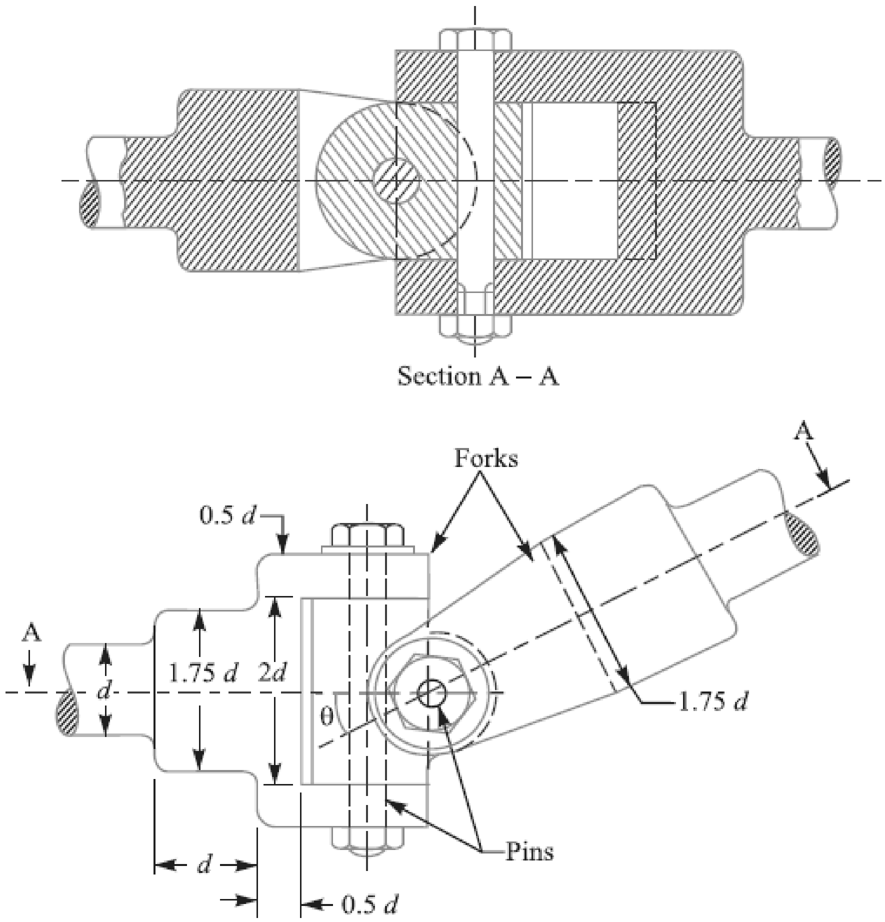


Fig. 2.27. Universal (or Hooke's) coupling.

EXERCISES

1. A shaft running at 400 r.p.m. transmits 10 kW. Assuming allowable shear stress in shaft as 40 MPa, find the diameter of the shaft.**[Ans. 35 mm]**

2. A hollow steel shaft transmits 600 kW at 500 r.p.m. The maximum shear stress is 62.4 MPa. Find the outside and inside diameter of the shaft, if the outer diameter is twice of inside diameter, assuming that the maximum torque is 20% greater than the mean torque.**[Ans. 100 mm ; 50 mm]**

3. A hollow shaft for a rotary compressor is to be designed to transmit a maximum torque of 4750 N-m. The shear stress in the shaft is limited to 50 MPa. Determine the inside and outside diameters of the shaft, if the ratio of the inside to the outside diameter is 0.4. **[Ans. 35 mm ; 90 mm]**

4. A motor car shaft consists of a steel tube 30 mm internal diameter and 4 mm thick. The engine develops 10 kW at 2000 r.p.m. Find the maximum shear stress in the tube when the power is transmitted through a 4 : 1 gearing. **[Ans. 30 MPa]**

5. A cylindrical shaft made of steel of yield strength 700 MPa is subjected to static loads consisting of a bending moment of 10 kN-m and a torsional moment of 30 kN-m. Determine the diameter of the shaft using two different theories of failure and assuming a factor of safety of 2.**[Ans. 100 mm]**

6. A line shaft rotating at 200 r.p.m. is to transmit 20 kW. The allowable shear stress for the material of the shaft is 42 MPa. If the shaft carries a central load of 900 N and is simply supported between bearing 3 metre apart, determine the diameter of the shaft. The maximum tensile or compressive stress is not to exceed 56 MPa. **[Ans. 50 mm]**

7. Two 400 mm diameter pulleys are keyed to a simply supported shaft 500 mm apart. Each pulley is 100 mm from its support and has horizontal belts, tension ratio being 2.5. If the shear stress is to be limited to 80 MPa while transmitting 45 kW at 900 r.p.m., find the shaft diameter if it is to be used for the input-output belts being on the same or opposite sides. **[Ans. 40 mm]**

8. A cast gear wheel is driven by a pinion and transmits 100 kW at 375 r.p.m. The gear has 200 machine cut teeth having 20° pressure angle and is mounted at the centre of a 0.4 m long shaft. The gear weighs 2000 N and its pitch circle diameter is 1.2 m. Design the gear shaft. Assume that the axes of the gear and pinion lie in the same horizontal plane. **[Ans. 80 mm]**

9. Fig. 2.28 shows a shaft from a hand-operated machine. The frictional torque in the journal bearings at *A* and *B* is 15 N-m each. Find the diameter (*d*) of the shaft (on which the pulley is mounted) using maximum distortion energy criterion. The shaft material is 40 C 8 steel for which the yield stress in tension is 380 MPa and the factor of safety is 1.5. **[Ans. 20 mm]**

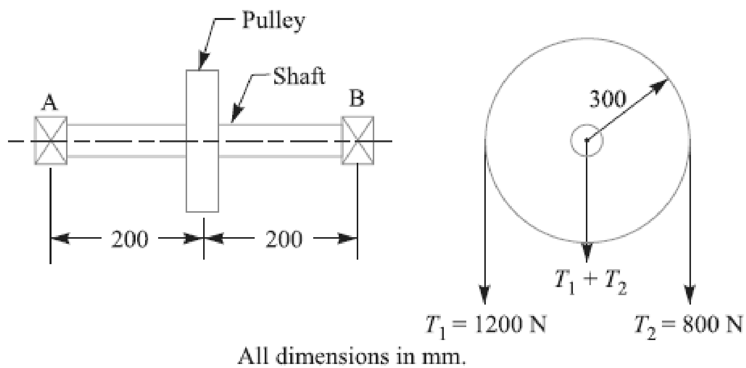


Fig. 2.28

10. A horizontal shaft *AD* supported in bearings at *A* and *B* and carrying pulleys at *C* and *D* is to transmit 75 kW at 500 r.p.m. from drive pulley *D* to off-take pulley *C*, as shown in Fig. 2.29. Calculate the diameter of shaft. The data given is : *P*1 = 2 *P*2 (both horizontal), *Q*1 = 2 *Q*2 (both vertical), radius of pulley *C* = 220 mm, radius of pulley *D* = 160 mm, allowable shear stress = 45 MPa.

[Ans. 100 mm]

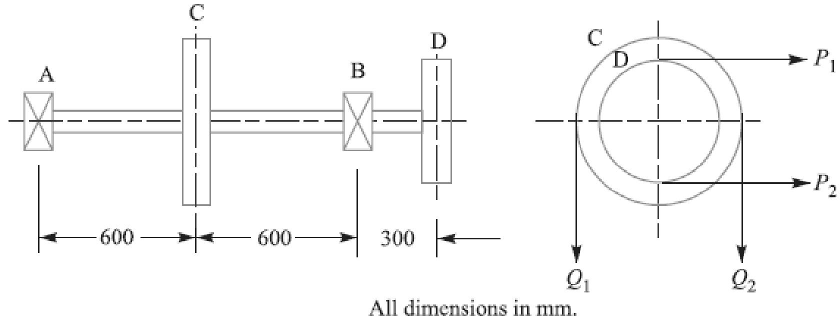


Fig. 2.29

11. A shaft made of steel receives 7.5 kW power at 1500 r.p.m. A pulley mounted on the shaft as shown in Fig. 2.30 has ratio of belt tensions 4. The gear forces are as follows :

Ft = 1590 N; *Fr* = 580 N
Design the shaft diameter by maximum shear stress theory. The shaft material has the following properties :
Ultimate tensile strength = 720 MPa; Yield strength = 380 MPa; Factor of safety = 1.5.

[Ans. 20 mm]

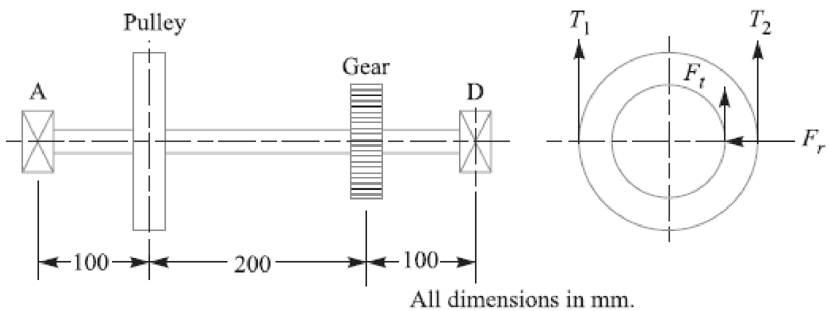


Fig. 2.30

12. A shaft 80 mm diameter transmits power at maximum shear stress of 63 MPa. Find the length of a 20 mm wide key required to mount a pulley on the shaft so that the stress in the key does not exceed 42 MPa.**[Ans. 152 mm]**

13. A shaft 30 mm diameter is transmitting power at a maximum shear stress of 80 MPa. If a pulley is connected to the shaft by means of a key, find the dimensions of the key so that the stress in the key is not to exceed 50 MPa and length of the key is 4 times the width. **[Ans. *l* = 126 mm]**

14. A steel shaft has a diameter of 25 mm. The shaft rotates at a speed of 600 r.p.m. and transmits 30 kW through a gear. The tensile and yield strength of the material of shaft are 650 MPa and 353 MPa respectively. Taking a factor of safety 3, select a suitable key for the gear. Assume that the key and shaft are made of the same material. **[Ans. *l* = 102 mm]**

15. Design a muff coupling to connect two shafts transmitting 40 kW at 120 r.p.m. The permissible shear and crushing stress for the shaft and key material (mild steel) are 30 MPa and 80 MPa respectively. The material of muff is cast iron with permissible shear stress of 15 MPa. Assume that the maximum torque transmitted is 25 per cent greater than the mean torque.
[Ans. *d* = 90 mm ; *w* = 28 mm, *t* = 16 mm, *l* = 157.5 mm ; *D* = 195 mm, *L* = 315 mm]

16. Design a compression coupling for a shaft to transmit 1300 N-m. The allowable shear stress for the shaft and key is 40 MPa and the number of bolts connecting the two halves are 4. The permissible tensile stress for the bolts material is 70 MPa. The coefficient of friction between the muff and the shaft surface may be taken as 0.3. **[Ans. *d* = 55 mm ; *D* = 125 mm ; *L* = 192.5 mm ; *db* = 24 mm]**

17. Design a cast iron protective flange coupling to connect two shafts in order to transmit 7.5 kW at 720 r.p.m. The following permissible stresses may be used :
Permissible shear stress for shaft, bolt and key material = 33 MPa
Permissible crushing stress for bolt and key material = 60 MPa
Permissible shear stress for the cast iron = 15 MPa **[Ans. *d* = 25 mm; *D* = 50 mm]**

18. A flanged protective type coupling is required to transmit 50 kW at 2000 r.p.m.. Find :
(*a*) Shaft diameters if the driving shaft is hollow with *di* / *d*0 = 0.6 and driven shaft is a solid shaft.
Take τ= 100 MPa.
(*b*) Diameter of bolts, if the coupling uses four bolts. Take $\sigma_c = \sigma_t$ = 70 MPa and τ= 25 MPa.

Assume pitch circle diameter as about 3 times the outside diameter of the hollow shaft.
(*c*) Thickness of the flange and diameter of the hub. Assume σ_c = 100 MPa and τ = 125 MPa.
(*d*) Make a neat free hand sketch of the assembled coupling showing a longitudinal sectional elevation with the main dimensions. The other dimensions may be assumed suitably.

19. A marine type flange coupling is used to transmit 3.75 MW at 150 r.p.m. The allowable shear stress in the shaft and bolts may be taken as 50 MPa. Determine the shaft diameter and the diameter of the bolts. **[Ans. 300 mm ; 56 mm]**

20. Design a bushed-pin type flexible coupling for connecting a motor shaft to a pump shaft for the following service conditions :
Power to be transmitted = 40 kW ; speed of the motor shaft = 1000 r.p.m. ; diameter of the motor shaft = 50 mm ; diameter of the pump shaft = 45 mm.
The bearing pressure in the rubber bush and allowable stress in the pins are to be limited to 0.45 N/mm^2 and 25 MPa respectively. **[Ans. d_1 = 20 mm; n = 6; d_2 = 40 mm ; l = 152 mm]**