Mechanical Couplings

Couplings are used to connect and to transmit power from the driving shaft to the driven shaft.

A Coupling is used to connect two shafts permanently. It is disconnected only for repairs or to make a change in the instantiation. Permanent couplings are referred to simply as couplings, while those which may be rapidly engaged to transmit power, or disengaged when desire, are called clutches.

Too long shafts are difficult to produce and transport so long transmission shafts are made by joining one or more lengths with the help of couplings. Couplings are also used to connect driving unit (turbine, electric motor or an engine) and driven unit (pump, compressors etc.)- Couplings are available for joining shafts at angles or with misalignment, and have ability to provide damping.

SELECTION

The choice of a coupling is based on the following considerations:

(i) Loading. Torque to be transmitted and type of load, *i.e.*, static, variable or shock.
(ii) Misalignment. The maximum parallel and/or angular misalignment or relative position of the shafts to be joined. Requirement of compensation of axial displacement.
(iii) Length. To get the length of shaft.
(iv) Repair. To provide for disconnection for repairs or alterations.
(v) Requirement of damping ability.

CLASSIFICATION

A wide variety of couplings are commercially available, ranging from a simple rigid coupling to elaborate flexible couplings using gears, elastomer and fluids for transmission of torque from one shaft to another shaft or from a shaft to a device.

Couplings may be classified in two broad classes: rigid and flexible couplings.

Rigid Couplings: Rigid couplings are used to connect two shafts which are collinear. They do not permit any relative rotation and axial motion between them. The most widely used rigid couplings are:

(i) Sleeve *or* Box or Muff coupling.
(ii) Split Muff coupling.
(iii) Cast iron flange coupling.
(iv) Marine or Solid flange coupling.

Flexible coupling: Flexible couplings are used to connect two shafts which are non-collinear i.e., shafts having slight parallel or angular misalignment. A flexible coupling permits relative rotation and variation in the alignment of shafts within certain limits. The most widely used Flexible couplings are:

(i) *Pin or bush type flexible coupling*
(ii) Oldham's coupling
(iii) Universal coupling
(iv) Band type flexible coupling.
(v) Fabric flexible coupling.

A protective cover is provided on each flange, to ensure that the clothes of a worker do not get entangled with running bolts and nuts.

1. **1.** Design a flanged coupling, to transmit a power of 32 kW at 960 rpm. The overall torque is 20% greater than the mean torque. The allowable shear stress for the shaft, key and bolt is 40 MPa. The allowable shear stress in the CI flange is 15 MPa. Bearing pressure in the bush is 0.8 MPa. **(Dec. 2011) Marks 12**
2. Design a CI flange coupling for a steel shaft transmitting 15 kW at 200 rpm. The allowable sheer stresses in shaft, bolt and key materials is 40 MPa and the allowable sheer stress in flange is 20 MPa. The maximum torque is 25% greater than the mean torque. **(June/July 2011) Marks 12**
3. Design a protected type cast iron flange coupling for a steel shaft transmitting 30 kW at 200 rpm. The allowable shear stress in the shaft and key material is 40 MPa. The maximum torque transmitted to be 20% greater than full load torque. The allowable shear stress in the bolt is 60 MPa and allowable shear stress in the flange is 40 MPa. **(Dec. 08/Jan.09) Marks 12**
4. Design a flange coupling to transmit 14 kW at 600 rpm. Select C40 steel for the shaft and C35 steel for bolts, with a factor of safety = 2. Use allowable shear stress in cast iron flanges as 15 MPa. Also draw the sketch of the coupling. **(Dec. 2010) Marks 10**
5. A mild steel has to transmit 75 kW at 200 rpm. The allowable stress in the shaft material is limited to 40 MPa and the angle of twist is not to exceed 1^0 in a length of 20 diameters. Calculate the suitable diameter of the shaft and also design a cast iron flange coupling for this shaft. Assume the allowable stress in the material of the bolts to be 30 MPa and the bolts are fitted in reamed holes. Assume the allowable shear stress in the cast iron flange equal to 15 MPa. Assume $G = 80 \times 10^3\ {N}/{mm^2}$. **(Dec. 09/Jan.10) Marks 15**

Cast Iron Flange Coupling (Protected and Un – Protected CI flange coupling)

It consists of two similar cast iron flanges. Each flange is mounted on the shaft end and keyed to shaft. Sunk keys of rectangular or square cross – section are commonly used for the purpose. The flanges are connected by means of bolts and nuts as shown in Figs (13.2) and (13.3) P- 233. one of the flanges has a projected portion and the other flange has a corresponding recess. To ensure correct alignment, one of the shaft is extended so that its end partially enter the flange of the other shaft. this helps to bring the shafts to be in line and maintain alignment.

In un-protected CI flange coupling as shown in Figs (13.2). The bolt heads and nuts are open and hence liable to cause injury to the operator.

In protected CI flange coupling the bolt heads and nuts are covered by the projecting flanges as shown in Figs (13.3)

Design procedure for (protected / unprotected) cast iron rigid flange coupling

The component to be designed in this coupling are (i). Shafts (ii). Bolts (iii). Key (iv). Hub and (v) Flange. **Note :** Refer Fig (13.2)(P- 233)

i). Shaft design

Torque transmitted by the coupling

$$T = \frac{9.55 \times 10^6 (P)}{n} \quad N-mm \qquad E(3.3a)P\ 42$$

Power, P is in kW

And is equated to the equation for torque transmitted by the shaft diameter given by

Based on strength,

$$\tau = \frac{16\,T}{\pi\,\eta\,D^3} \qquad E(3.1)P\ 42$$

And *based on rigidity (stiffness)*

$$\theta = \frac{584\ T\ L}{G\ D^4} \qquad E(3.2)P\ 42$$

Where D = Shaft diameter.

The value of 'D' is round off to the next standard size Ref Table T(3.5a) P-48

ii). Key design

Based on the shaft diameter 'D' the value of width 'b' and thickness 'h' of taper key are selected from Table T(4.2) P- 61

Calculate *the effective length of key* = L = Hub length

$$\mathrm{L} = 1.2\mathrm{D} + 20\ \mathrm{mm} \qquad E(13.6)P\ 210$$

After selecting b, h and L for the key. Give a check for the induced shear stress and induced crushing stress in the key,

Check for key

For *Induced crushing stress in key*

$$T = \frac{1}{4}\sigma_{b1} hdL \qquad E(4.5\ b)P\ 53$$

$$\sigma_{k_{(ind)}} = \frac{4\ T}{h\ d\ L} \quad MPa$$

For *Induced shear stress in key*

$$T = \frac{1}{2}\tau_1 bdL \qquad E(4.6)P\ 54$$

$$\tau_{k_{(ind)}} = \frac{2\ T}{b\ d\ L} \quad MPa$$

$$if \qquad \tau_{k_{(ind)}} < \tau_{k_{(allow)}}, \qquad Design\ is\ Safe.$$

$$\sigma_{k_{(ind)}} < \sigma_{k_{(allow)}}, \qquad Design\ is\ Safe.$$

If the induced shear stress $\tau_{k_{(ind)}}$ and crushing stress $\sigma_{k_{(ind)}}$ in the key are less than this allowable (permissible) shear stress $\tau_{k_{(allow)}}$ and allowable (permissible) crushing stress $\sigma_{k_{(allow)}}$, the design of key is safe. Otherwise increase the key dimensions and check once again.

iii). Hub design

The hub diameter $D_1 = 1.8D + 20\ mm$ $\quad E(13.2)P\ 209$

The hub length $\mathrm{L} = 1.2\mathrm{D} + 20\ \mathrm{mm}$ $\quad$ E(13.6)P 210

iv). Bolt design

Number of bolts $i = \frac{1}{40}D + 2 \quad to \quad \frac{3}{80}D + 2$ $\quad E(13.1)P\ 209$

Even no of bolts as preferred where 'D' is diameter of shaft.

Determine *the minor diameter of the bolt* 'd' by the empirical formula

$$d = \frac{0.423D}{\sqrt{i}} + 7.5\ mm \qquad E(13.5)P\ 210$$

Select the size of bolt based on the core area of bolt (A_C)

Standardize the bolt size using table T(9.8) P- 113

Note: unless otherwise specified the shaft, key and bolts are made of same material and hence have same value of stress

$$T_{Shaft} = T_{Key} = T_{Bolt}$$

$$i.e., \ \tau_{Shaft} = \tau_{Key} = \tau_{Bolt}$$

Bolt circle diameter

$$D_2 = D_1 + 3.2d\ mm \qquad E(13.3)P\ 210$$

Check for bolt

Check for *induced shear stress in bolt*

$$T = \frac{\tau_{b(ind)}\ \pi\ i\ d^2 D_2}{8} \qquad E(13.9)P\ 210$$

$$\tau_{b(ind)} = \frac{T\ \times 8}{\pi\ i\ d^2 D_2}$$

$$if \qquad \tau_{b(ind)} < \tau_{b(allow)}, \qquad Design\ is\ Safe.$$

If the induced shear stress $\tau_{b(ind)}$ in the bolt are less than this allowable (permissible) shear stress $\tau_{b(allow)}$, the design of key is safe. Otherwise increase the bolt dimensions and check once again.

v). Flange design

Outer diameter of flange

$$D_3 = D_1 + 6d\ mm \qquad E(13.4)P\ 209$$

The flange thickness

$$t = 0.35D + 9\ mm \qquad E(13.7)P\ 210$$

Check for flange

$$T = \frac{\tau_{f(ind)}\ \pi\ {D_1}^2\ t}{2} \qquad E(13.11)P\ 210$$

$$\tau_{f(ind)} = \frac{2\ \times T}{\pi\ {D_1}^2\ t}$$

$$if \qquad \tau_{f(ind)} < \tau_{f(allow)}, \qquad Design\ is\ Safe.$$

If the induced shear stress $\tau_{f(ind)}$ in the flange are less than this allowable (permissible) shear stress $\tau_{f(allow)}$, the design of key is safe. Otherwise increase the flange dimensions and check once again.

	Symbols (Un Protected & protected)			
		Formulae	Equations	Table
Design of Shaft				
T	Torque transmitted in N-mm	$\frac{9.55 \times 10^6 (P)}{n}$ P in kW	(3.3a) P-42	
D	Shaft Diameter	$\tau = \frac{16\ T}{\pi\ \eta\ D^3}$	(3.1)P-48	(3.5a)P-48
		$\theta = \frac{584\ T\ L}{G\ D^4}$	(3.2)P-48	
Design of Key				
b	Width of taper key			(4.2) P-61
h	Height of taper key			
L	Length of taper key ≈ Length of Hub	L= 1.2D + 20 mm	(13.6) P-210	(4.2) P-61
Check for key				
$\sigma_{k(induced)}$	Induced crushing stress in key	$T = \frac{1}{4}\sigma_{k(ind)} hdL$	(4.5b) P-53	
$\tau_{k(induced)}$	Induced shear stress in key	$T = \frac{1}{2}\tau_{k(ind)} bDL$	(4.6) P-54	
Design of Hub				
D_1	hub diameter	$D_1 = 1.8D + 20\ mm$	(13.2) P-209	
L	hub length	L= 1.2D + 20 mm	(13.6) P-210	

Design of Bolt				
No of bolts / pins	I	$i = \frac{1}{40}D + 2$ to $\frac{3}{80}D + 2$	(13.1) P-209	
D	Diameter of bolt	$d = \frac{0.423D}{\sqrt{i}} + 7.5\ mm$	(13.5) P-210	(9.8) P-113
D_2	Bolt circle diameter	$D_2 = D_1 + 3.2d\ mm$	(13.3) P-209	
Check for bolt				
$\tau_{b(induced)}$	Induced shear stress in bolt	$T = \frac{\tau_{b(ind)}\ \pi\ i\ d^2 D_2}{8}$	(13.9) P-210	
Design of flange				
D_3	Outer diameter of flange	$D_3 = D_1 + 6d\ mm$	(13.4) P-209	
t	Flange thickness	$t = 0.35D + 9\ mm$	(13.7) P-210	
Check for flange				
$\tau_{f(induced)}$	Induced shear stress in Flange	$T = \frac{\tau_{f(ind)}\ \pi\ D_1{}^2\ t}{2}$	(13.11) P-210	
t_1	Thickness of protective circumferential flange	Assume t_1 = 0.25D mm		
η	Key way factor	Assume η = 0.75 mm		

1. Design a protected type cast iron flange coupling for a steel shaft transmitting 30 kW at 200 rpm. The allowable shear stress in the shaft and key material is 40 MPa. The maximum torque transmitted to be 20% greater than full load torque. The allowable shear stress in the bolt is 60 MPa and allowable shear stress in the flange is 40 MPa.

(Dec. 08/Jan.09) Marks 12

Data: P = 30 kW , n= 200 rpm , $\tau_s = \tau_k = 40\ MPa$, $\tau_f = 40\ MPa$,
$T = T_{Max} = 1.2\ T_{Mean}$

Normally for ductile material $\sigma_{yt} = \sigma_{yc}$ and ∴ $\sigma_s = 2 \times \tau_s$

Allowable crushing stress in the key $\sigma_s = 2 \times \tau_s = 2 \times 40 = 80\ MPa$

Assume keyway factor $\eta = 0.75$

i). Shaft design

a) *Max. torque transmitted* by the coupling

$$T_{Mean} = \frac{9.55 \times 10^6 (P)}{n} \quad N - mm \qquad E(3.3a)P\ 42$$

$$T_{Mean} = \frac{9.55 \times 10^6 (30 \times 10^3)}{200}$$

$$T_{Mean} = 1.4325 \times 10^6\ N - mm$$

$$T = T_{Max} = 1.2\ T_{Mean} = 1.2 \times 1.4325 \times 10^6$$

Torque $T = 1.719 \times 10^6 N - mm$

b) Shaft diameter ' D'

The equation for torque transmitted by the shaft diameter given by

Based on strength, $\tau = \frac{16\ T}{\pi\ \eta\ D^3}$ $E(3.1)P\ 42$

$$\tau_s = \frac{16\ T}{\pi\ \eta\ D^3}$$

$$40 = \frac{16 \times 1.719 \times 10^6}{\pi \times 0.75 \times D^3}$$

$$D = 66.33\ mm$$

Select standard diameter of shaft 'D' from T(3.5a) P- 48

$$D = 71\ mm$$

ii). Key design

a) Key dimensions (b , h and L)

Select taper key and taper 1:100 from Table T(4.2) P-61 for shaft diameter 'D' = 71mm

Width of key b = 20 mm *Thickness of key* h = 12 mm

Length of key = length of hub

$$\mathrm{L} = 1.2\mathrm{D} + 20\ \mathrm{mm} \qquad E(13.6)P\ 210$$

$$\mathrm{L} = 1.2 \times 71 + 20$$

$$\mathrm{L} = 105.2\ \mathrm{mm}$$

Preferred *length of key* L = 110 mm

(Length of key should not be less than the hub length)

b) Check for key

For *Induced crushing stress in key*

$T = \frac{1}{4}\sigma_{b1} hdL \qquad E(4.5\ b)P\ 53$

$$\sigma_{k_{(ind)}} = \frac{4\,T}{h\ d\ L} \quad MPa$$

$$\sigma_{k_{(ind)}} = \frac{4 \times 1.719 \times 10^6}{20 \times 12 \times 110}$$

$$\sigma_{k_{(ind)}} = 73.37\ MPa < 80\ MPa \left(\sigma_{k_{(allow)}}\right)$$

For *Induced shear stress in key*

$T = \frac{1}{2}\tau_1 bdL \qquad E(4.6)P\ 54$

$$\tau_{k_{(ind)}} = \frac{2\,T}{b\ d\ L} \quad MPa$$

$$\tau_{k_{(ind)}} = \frac{2 \times 1.719 \times 10^6}{12 \times 71 \times 110}$$

$$\tau_{k_{(ind)}} = 22.010 < 40\ MPa \left(\tau_{k_{(allow)}}\right)$$

$$22.101\ MPa \left(\tau_{k_{(ind)}}\right) < 40\ MPa \left(\tau_{k_{(allow)}}\right) \qquad Design\ is\ Safe.$$

$$73.37\ MPa \left(\sigma_{k_{(ind)}}\right) < 80\ MPa \left(\sigma_{k_{(allow)}}\right), \qquad Design\ is\ Safe.$$

as the induced shear stress $\tau_{k_{(ind)}}$ and crushing stress $\sigma_{k_{(ind)}}$ in the key are less than this allowable (permissible) shear stress $\tau_{k_{(allow)}}$ and allowable (permissible) crushing stress $\sigma_{k_{(allow)}}$, the design of key is safe.

iii). Hub design

a) *The hub diameter* $D_1 = 1.8D + 20\ mm \qquad E(13.2)P\ 209$

$$D_1 = 1.8 \times 71 + 20$$

$$D_1 = 147.8mm$$

b) *The hub length* $L = 1.2D + 20\ mm \qquad E(13.6)P\ 210$

$$L = 1.2 \times 71 + 20$$

$$L = 105.2\ mm$$

iv). Design of bolt

a) *Number of bolt*

$$i = \frac{1}{40}D + 2 \;\; to \;\; \frac{3}{80}D + 2 \qquad E(13.1)P\ 209$$

$$i = \frac{1}{40} \times 71 + 2 \quad to \quad \frac{3}{80} \times 71 + 2$$

$$i = 3.7752 \quad to \quad 4.6625$$

Take the number of bolts, i = 6 [even number of bolts are preferred]

b) Bolt diameter

$$d = \frac{0.423D}{\sqrt{i}} + 7.5\ mm \qquad E(13.5)P\ 210$$

$$d = \frac{0.423 \times 71}{\sqrt{6}} + 7.5$$

$$d = 19.76\ mm$$

From table T(9.8) P 114 for core diameter d = 20 mm, the select diameter of bolt is 20 mm.

Standard bolt size is M20 x 2.5 & its core area $A_c = 245\ mm^2$

c) Bolt circle diameter

$$D_2 = D_1 + 3.2d\ mm \qquad E(13.3)P\ 209$$

$$D_2 = 147.8 + 3.2 \times 20$$

$$D_2 = 211.8\ mm$$

d) Check for bolt

Shear stress induced in bolt

$$T = \frac{\tau\, \pi\, i\, d^2 D_2}{8} \qquad E(13.9)P\ 210$$

$$T = \frac{\tau_{b_{(ind)}}\, \pi\, i\, d^2 D_2}{8}$$

$$\tau_{b_{(ind)}} = \frac{8\,T}{\pi\, i\, d^2 D_2}$$

$$\tau_{b_{(ind)}} = \frac{8 \times 1.719 \times 10^6}{\pi \times 6 \times 20^2 \times 211.8}$$

$$\tau_{b_{(ind)}} = 8.61\ {}^{N}\!/_{mm^2}$$

$$\tau_{b_{(ind)}} = 8.61\ {}^{N}\!/_{mm^2} < 60\ {}^{N}\!/_{mm^2} \left(\tau_{b_{(allow)}}\right)$$

$$8.61\ {}^{N}\!/_{mm^2} \left(\tau_{b_{(ind)}}\right) < 60MPa \left(\tau_{b_{(allow)}}\right) \qquad Design\ is\ Safe.$$

as the induced shear stress $\tau_{b_{(ind)}}$ in the bolt are less than this allowable (permissible) shear stress $\tau_{k_{(allow)}}$, therefore the design of bolt is safe.

vi). Flange design

a. *Outer diameter of flange*

$$D_3 = D_1 + 6d\ mm \qquad E(13.4)P\ 209$$

$$D_3 = 147.8 + 6 \times 20$$

$$D_3 = 367.8\ mm$$

b. *Flange thickness*

$$t = 0.35D + 9\ mm \qquad E(13.7)P\ 210$$

$$t = 0.35 \times 71 + 9$$

$$t = 33.85\ mm$$

c. Check for flange

$$T = \frac{\tau\ \pi\ {D_1}^2\ t}{2} \qquad E(13.11)P\ 210$$

$$T = \frac{\tau_{f_{(ind)}}\ \pi\ {D_1}^2\ t}{2}$$

$$\tau_{f_{(ind)}} = \frac{2\,T}{\pi\ {D_1}^2\ t}$$

$$\tau_{f_{(ind)}} = \frac{2 \times 1.719\ \times 10^6}{\pi\ \times\ 147.8^2 \times 33.8}$$

$$\tau_{f_{(ind)}}\ = 8.61\ {}^{N}/_{mm^2}$$

$$\tau_{f_{(ind)}}\ = 1.48\ {}^{N}/_{mm^2}$$

$$\tau_{f_{(ind)}} = 1.48\,{}^{N}/_{mm^2}\ <\ 40\ {}^{N}/_{mm^2}\left(\tau_{f_{(allow)}}\right)$$

$$1.48\ {}^{N}/_{mm^2}\left(\ \tau_{f_{(ind)}}\right) <\ 40\ \ {}^{N}/_{mm^2}\left(\ \tau_{f_{(allow)}}\right) \qquad Design\ is\ Safe.$$

as the induced shear stress $\tau_{f_{(ind)}}$ in the flange are less than this allowable (permissible) shear stress $\tau_{f_{(allow)}}$, therefore the design of flange(thickness) is safe.

2. Design a flange coupling to transmit 14 kW at 600 rpm. Select C40 steel for the shaft and C35 steel for bolts, with a factor of safety = 2. Use allowable shear stress in cast iron flanges as 15 MPa. Also draw the sketch of the coupling. **(Dec. 2010) Marks 10**

Data: P = 14 kW , n= 600 rpm , $\tau_{f(allow)} = 15\ MPa$, refer Table T(1.8) P – 418 for material C40 & C35 steel , FoS = n = 2

Assume keyway factor $\eta = 0.75$

For material C40 Steel $\sigma_y = \ 324\ {}^{N}/_{mm^2}$ $\qquad T(1.8)P\ 419$

Normally for ductile material $\sigma_{yt} = \ \sigma_{yc}$ and ∴ $\sigma_s = 2\ \times\ \tau_k$

$$\sigma_{S_{(allow)}}\ =\ \ \sigma_{K_{(allow)}} = \frac{\sigma_y}{n} = \frac{324}{2} = 162\ {}^{N}/_{mm^2}$$

$$\tau_{S_{(allow)}}\ =\ \tau_{K(allow)} = \frac{\sigma_{S_{(allow)}}}{n} = \frac{162}{2} = 81\ {}^{N}/_{mm^2}$$

For material C35 Steel $\sigma_y = \ 304\ {}^{N}/_{mm^2}$ $\qquad T(1.8)P\ 419$

$$\sigma_{b_{(allow)}} = \frac{\sigma_y}{n} = \frac{324}{2} = 152\ {}^{N}/_{mm^2}$$

$$\tau_{b_{(allow)}} = \frac{\sigma_{b_{(allow)}}}{n} = \frac{152}{2} = 76\ {}^{N}/_{mm^2}$$

i). Shaft design

a) Max. torque transmitted by the coupling

$$T = \frac{9.55 \times 10^6 (P)}{n} \quad N - mm \qquad E(3.3a)P\ 42$$

$$T = \frac{9.55 \times 10^6 (14 \times 10^3)}{600}$$

Torque $T = 2.23 \times 10^5 N - mm$

b) Shaft diameter ‘ D’

The equation for torque transmitted by the shaft diameter given by

Based on strength,

$$\tau = \frac{16\,T}{\pi\,\eta\,D^3} \qquad E(3.1)P\ 42$$

$$\tau_s = \frac{16\,T}{\pi\,\eta\,D^3}$$

$$40 = \frac{16\ \times 2.23 \times 10^5}{\pi\ \times 0.75\ \times\ D^3}$$

$$D = 24.11\ \ mm$$

Select standard diameter of shaft ‘D’ from T(3.5a) P- 48

$$D = 25\,mm$$

iii). Key design

a) Key dimensions (b , h and L)

Select taper key and taper 1:100 from Table T(4.2) P-61 for shaft diameter ‘D’ = 25mm

Width of key b = 8 mm *Thickness of key* h = 7 mm

Length of key = length of hub

$$\mathrm{L} = 1.2\mathrm{D} + 20\ \mathrm{mm} \qquad E(13.6)P\ 210$$

$$\mathrm{L} = 1.2\ \times 25\ +\ 20$$

$$\mathrm{L} = 50\ \mathrm{mm}$$

Preferred *length of key* L = 56 mm

(Length of key should not be less than the hub length)

b) Check for key

For *Induced crushing stress in key*

$$T = \tfrac{1}{4}\sigma_{b1} hdL \qquad E(4.5\ b)P\ 53$$

$$\sigma_{k_{(ind)}} = \frac{4\,T}{h\ d\ L} \quad MPa$$

$$\sigma_{k_{(ind)}} = \frac{4\ \times 2.23 \times 10^5}{7\ \times 25\ \times 56}$$

$$\sigma_{k_{(ind)}} = 91.02\ MPa$$

$$\sigma_{k_{(ind)}} = 91.02\ MPa\ <\ 162\ MPa \left(\sigma_{k_{(allow)}}\right)$$

For *Induced shear stress in key*

$$T = \tfrac{1}{2}\tau_1 bdL \qquad E(4.6)P\ 54$$

$$\tau_{k_{(ind)}} = \frac{2\,T}{b\ d\ L} \quad MPa$$

$$\tau_{k_{(ind)}} = \frac{2\ \times 2.23 \times 10^5}{8\ \times 25\ \times 56}$$

$$\tau_{k_{(ind)}} = 39.82\ MPa$$

$$\tau_{k_{(ind)}} = 39.82\ MPa\ <\ 81\ MPa \left(\tau_{k_{(allow)}}\right)$$

$$39.82\ MPa\left(\ \tau_{k_{(ind)}}\right) <\ 81\ MPa\left(\ \tau_{k_{(allow)}}\right) \qquad Design\ is\ Safe.$$

$$91.02\ MPa\left(\ \sigma_{k_{(ind)}}\right) <\ 162\ MPa\left(\ \sigma_{k_{(allow)}}\right), \qquad Design\ is\ Safe.$$

as the induced shear stress $\tau_{k_{(ind)}}$ and crushing stress $\sigma_{k_{(ind)}}$ in the key are less than this allowable (permissible) shear stress $\tau_{k_{(allow)}}$ and allowable (permissible) crushing stress $\sigma_{k_{(allow)}}$, the design of key is safe.

iv). Hub design

a) *The hub diameter*

$$D_1 = 1.8D + 20\ mm \qquad E(13.2)P\ 209$$

$$D_1 = 1.8 \times 25 + 20$$

$$D_1 = 65\ mm$$

c) *The hub length* $L = 1.2D + 20\ mm \qquad E(13.6)P\ 210$

$$L = 1.2 \times 25 + 20$$

$$L = 50\ mm$$

v). Design of bolt

a) *Number of bolt*

$$i = \frac{1}{40}D + 2 \ \ to \ \ \frac{3}{80}D + 2 \qquad E(13.1)P\ 209$$

$$i = \frac{1}{40} \times 25 + 2 \quad to \quad \frac{3}{80} \times 25 + 2$$

$$i = 2.6 \quad to \quad 2.9375$$

Take the number of bolts, i = 4 [even number of bolts are preferred]

b) *Bolt diameter*

$$d = \frac{0.423D}{\sqrt{i}} + 7.5\ mm \qquad E(13.5)P\ 210$$

$$d = \frac{0.423 \times 25}{\sqrt{4}} + 7.5$$

$$d = 12.78\ mm$$

From table T(9.8) P 114 for core diameter d = 14 mm, the select diameter of bolt is 14 mm.

Standard bolt size is M14 x 2 & its core area $A_c = 115\ mm^2$

c) *Bolt circle diameter*

$$D_2 = D_1 + 3.2d\ mm \qquad E(13.3)P\ 209$$

$$D_2 = 65 + 3.2 \times 14$$

$$D_2 = 109.8\ mm$$

d) *Check for bolt*

Shear stress induced in bolt

$$T = \frac{\tau\ \pi\ i\ d^2 D_2}{8} \qquad E(13.9)P\ 210$$

$$T = \frac{\tau_{b_{(ind)}}\ \pi\ i\ d^2 D_2}{8}$$

$$\tau_{b_{(ind)}} = \frac{8\ T}{\pi\ i\ d^2 D_2}$$

$$\tau_{b_{(ind)}} = \frac{8 \times 2.23 \times 10^5}{\pi \times 4 \times 14^2 \times 109.8}$$

$$\tau_{b_{(ind)}} = 6.59\ {}^{N}/_{mm^2}$$

$$\tau_{b_{(ind)}} = 6.59\ {}^{N}/_{mm^2} < 76\ {}^{N}/_{mm^2} \left(\tau_{b_{(allow)}}\right)$$

$$6.59\ {}^{N}/_{mm^2} \left(\tau_{b_{(ind)}} \right) < 76\ MPa \left(\tau_{b_{(allow)}} \right) \qquad Design\ is\ Safe.$$

as the induced shear stress $\tau_{b_{(ind)}}$ in the bolt are less than this allowable (permissible) shear stress $\tau_{k_{(allow)}}$, therefore the design of bolt is safe.

vii). Flange design

a) Outer diameter of flange $D_3 = D_1 + 6d\ mm$ $E(13.4)P\ 209$

$$D_3 = 65 + 6 \times 14$$

$$D_3 = 149\ mm$$

b) Flange thickness $t = 0.35D + 9\ mm$ $E(13.7)P\ 210$

$$t = 0.35 \times 25 + 9$$

$$t = 17.75\ mm$$

c) Check for flange

$$T = \frac{\tau\ \pi\ {D_1}^2\ t}{2} \qquad E(13.11)P\ 210$$

$$T = \frac{\tau_{f_{(ind)}}\ \pi\ {D_1}^2\ t}{2}$$

$$\tau_{f_{(ind)}} = \frac{2\,T}{\pi\ {D_1}^2\ t}$$

$$\tau_{f_{(ind)}} = \frac{2 \times 2.23 \times 10^5}{\pi \times 658^2 \times 17.75}$$

$$\tau_{b_{(ind)}} = 8.61\ {}^{N}/_{mm^2}$$

$$\tau_{f_{(ind)}} = 1.893\ {}^{N}/_{mm^2}$$

$$\tau_{f_{(ind)}} = 1.893\,{}^{N}/_{mm^2} < 15\ {}^{N}/_{mm^2}\left(\tau_{f_{(allow)}}\right)$$

$$1.893\ {}^{N}/_{mm^2}\left(\tau_{f_{(ind)}}\right) < 15\ MPa\left(\tau_{f_{(allow)}}\right) \qquad Design\ is\ Safe.$$

as the induced shear stress $\tau_{f_{(ind)}}$ in the flange are less than this allowable (permissible) shear stress $\tau_{f_{(allow)}}$, therefore the design of flange(thickness) is safe.

3. A mild steel has to transmit 75 kW at 200 rpm. The allowable stress in the shaft material is limited to 40 MPa and the angle of twist is not to exceed 1^0 in a length of 20 diameters. Calculate the suitable diameter of the shaft and also design a cast iron flange coupling for this shaft. Assume the allowable stress in the material of the bolts to be 30 MPa and the bolts are fitted in reamed holes. Assume the allowable shear stress in the cast iron flange equal to 15 MPa. Assume $G = 80 \times 10^3\ {}^{N}/_{mm^2}$. **(Dec. 09/Jan.10) Marks 15**

Data: P = 75 kW , n= 200 rpm , $\tau_{s_{(allow)}} = 40\ MPa$ $\tau_{b_{(allow)}} = 30\ MPa$,

$\tau_{f_{(allow)}} = 15\ MPa$, $\theta = 1^0, G = 80 \times 10^3\ MPa, l = 20\ d$

Solution :

Assume keyway factor $\eta = 0.75$

Allowable crushing stress in the key and shaft

$$\sigma_{s_{(allow)}} = 2 \times \tau_{s_{(allow)}} = 2 \times 40 = 80\ MPa$$

Take $\tau_{s_{(allow)}} = \tau_{k_{(allow)}} = 40\ MPa$

Take $\sigma_{s_{(allow)}} = \sigma_{k_{(allow)}} = 80\ MPa$

i). Shaft design

a) Max. torque transmitted by the coupling

$$T = \frac{9.55 \times 10^6 (P)}{n}\ \ N - mm \qquad E(3.3a)P\ 42$$

$$T = \frac{9.55 \times 10^6 (75 \times 10^3)}{600}$$

Torque $T = 3.58125 \times 10^6 N - mm$

b) Shaft diameter ' D'

The equation for torque transmitted by the shaft diameter given by

Based on strength,

$$\tau = \frac{16\,T}{\pi\,\eta\,D^3} \qquad E(3.1)P\ 42$$

$$\tau_s = \frac{16\,T}{\pi\,\eta\,D^3}$$

$$40 = \frac{16\ \times 3.58125 \times 10^6}{\pi\ \times 0.75\ \times\ D^3}$$

$$D = 84.72\ \ mm$$

Based on rigidity,

$$\theta = \frac{584\ \ T\ L}{G\ \eta\ D^4} \qquad E(3.2)P\ 42$$

$$D^4 = \frac{584\ \ T\ L}{G\ \eta\ \theta}$$

$$D^4 = \frac{584\ \ \times 3.58125 \times 10^6\ \ \times 20\ D}{80 \times 10^3\ \ \times 0.75\ \ \times 1}$$

$$D = \sqrt[3]{6.9715 \times 10^5}$$

$$D = 88.67\ mm$$

Adopt (permissible) Diameter of the shaft is the large among the two values

∴ Select *standard diameter of shaft* 'D' from T(3.5a) P- 48

$$D = 90\ mm$$

iv). Key design

a) Key dimensions (b , h and L)

Select taper key and taper 1:100 from Table T(4.2) P-61 for shaft diameter 'D' = 90 mm

Width of key b = 25 mm *Thickness of key* h = 14 mm

Length of key = length of hub

$$\mathrm{L} = 1.2\mathrm{D} + 20\ \mathrm{mm} \qquad E(13.6)P\ 210$$

$$\mathrm{L} = 1.2\ \times 90\ +\ 20$$

$$\mathrm{L} = 128\ \mathrm{mm}$$

Preferred *length of key* L = 140 mm

(Length of key should not be less than the hub length)

c) Check for key

For *Induced crushing stress in key*

$$T = \tfrac{1}{4}\sigma_{b1} hdL \qquad E(4.5\ b)P\ 53$$

$$\sigma_{k_{(ind)}} = \frac{4\ T}{h\ d\ L} \quad MPa$$

$$\sigma_{k_{(ind)}} = \frac{4\ \times 3.58125 \times 10^6}{14\ \times 90\ \times 140}$$

$$\sigma_{k_{(ind)}} = 81.202\ MPa$$

$$\sigma_{k_{(ind)}} = 81.202\ MPa\ >\ 80\ MPa\left(\sigma_{k_{(allow)}}\right)$$

For *Induced shear stress in key*

$$T = \tfrac{1}{2}\tau_1 bdL \qquad E(4.6)P\ 54$$

$$\tau_{k_{(ind)}} = \frac{2\ T}{b\ d\ L} \quad MPa$$

$$\tau_{k_{(ind)}} = \frac{2 \times 3.58125 \times 10^6}{25 \times 90 \times 140}$$

$$\tau_{k_{(ind)}} = 22.738\ MPa$$

$$\tau_{k_{(ind)}} = 22.738\ MPa < 40\ MPa\left(\tau_{k_{(allow)}}\right) \qquad Design\ is\ fail.$$

$$key\ failure\ in\ crusing\ stress, so\ increase\ in\ dimension\ of\ key$$

Select next bigger value.

Refer Table T(4.2) P-61

Width of key b = 28 mm *Thickness of key* h = 16 mm

Preferred length of key L = 140 mm

For *Induced crushing stress in key*

$$T = \frac{1}{4}\sigma_{b1} hdL \qquad E(4.5\ b)P\ 53$$

$$\sigma_{k_{(ind)}} = \frac{4\,T}{h\,d\,L} \quad MPa$$

$$\sigma_{k_{(ind)}} = \frac{4 \times 3.58125 \times 10^6}{16 \times 90 \times 140}$$

$$\sigma_{k_{(ind)}} = 81.202\ MPa$$

$$\sigma_{k_{(ind)}} = 71.057\ MPa > 80\ MPa\left(\sigma_{k_{(ind)}}\right)$$

$$22.738\ MPa\left(\tau_{k_{(ind)}}\right) < 40\ MPa\left(\tau_{k_{(allow)}}\right) \qquad Design\ is\ Safe.$$

$$71.057\ MPa\left(\sigma_{k_{(ind)}}\right) < 80\ MPa\left(\sigma_{k_{(allow)}}\right), \qquad Design\ is\ Safe.$$

as the induced shear stress $\tau_{k_{(ind)}}$ and crushing stress $\sigma_{k_{(ind)}}$ in the key are less than this allowable (permissible) shear stress $\tau_{k_{(allow)}}$ and allowable (permissible) crushing stress $\sigma_{k_{(allow)}}$, the design of key is safe.

v). Hub design

a) *The hub diameter* $D_1 = 1.8D + 20\ mm \qquad E(13.2)P\ 209$

$$D_1 = 1.8 \times 90 + 20$$

$$D_1 = 182\ mm$$

b) *The hub length* $L = 1.2D + 20\ mm \qquad E(13.6)P\ 210$

$$L = 1.2 \times 90 + 20$$

$$L = 110\ mm$$

vi). Design of bolt

a) *Number of bolt*

$$i = \frac{1}{40}D + 2\ \ to\ \ \frac{3}{80}D + 2 \qquad E(13.1)P\ 209$$

$$i = \frac{1}{40} \times 90 + 2\ \ to\ \ \frac{3}{80} \times 90 + 2$$

$$i = 4.250\ \ to\ \ 5.375$$

Take the number of bolts, i = 6 [even number of bolts are preferred]

b) *Bolt diameter*

$$d = \frac{0.423D}{\sqrt{i}} + 7.5\ mm \qquad E(13.5)P\ 210$$

$$d = \frac{0.423 \times 90}{\sqrt{4}} + 7.5$$

$$d = 23.042\ mm$$

From table T(9.8) P 114 for core diameter d = 24 mm, the select diameter of bolt is 24 mm.

Standard bolt size is M24 x 3 & its core area $A_c = 385\ mm^2$

c) Bolt circle diameter

$$D_2 = D_1 + 3.2d\ mm \qquad E(13.3)P\ 209$$

$$D_2 = 182 + 3.2 \times 24$$

$$D_2 = 258.8\ mm$$

d) Check for bolt

Shear stress induced in bolt

$$T = \frac{\tau\ \pi\ i\ d^2 D_2}{8} \qquad E(13.9)P\ 210$$

$$T = \frac{\tau_{b_{(ind)}}\ \pi\ i\ d^2 D_2}{8}$$

$$\tau_{b_{(ind)}} = \frac{8\ T}{\pi\ i\ d^2 D_2}$$

$$\tau_{b_{(ind)}} = \frac{8 \times 3.58125 \times 10^6}{\pi \times 6 \times 24^2 \times 258.8}$$

$$\tau_{b_{(ind)}} = 10.196\ {}^{N}/_{mm^2}$$

$$\tau_{b_{(ind)}} = 10.196\ {}^{N}/_{mm^2} < 30\ {}^{N}/_{mm^2} \left(\tau_{b_{(allow)}}\right)$$

$$10.196\ {}^{N}/_{mm^2} \left(\ \tau_{b_{(ind)}}\right) < 30\ MPa \left(\ \tau_{b_{(allow)}}\right) \qquad Design\ is\ Safe.$$

as the induced shear stress $\tau_{b_{(ind)}}$ in the bolt are less than this allowable (permissible) shear stress $\tau_{k_{(allow)}}$, therefore the design of bolt is safe.

viii). Flange design

a. Outer diameter of flange

$$D_3 = D_1 + 6d\ mm \qquad E(13.4)P\ 209$$

$$D_3 = 182 + 6 \times 24$$

$$D_3 = 326\ mm$$

b. Flange thickness

$$t = 0.35D + 9\ mm \qquad E(13.7)P\ 210$$

$$t = 0.35 \times 90 + 9$$

$$t = 40.5\ mm$$

c. Check for flange

$$T = \frac{\tau\ \pi\ {D_1}^2\ t}{2} \qquad E(13.11)P\ 210$$

$$T = \frac{\tau_{f_{(ind)}}\ \pi\ {D_1}^2\ t}{2}$$

$$\tau_{f_{(ind)}} = \frac{2\ T}{\pi\ {D_1}^2\ t}$$

$$\tau_{f_{(ind)}} = \frac{2 \times 3.58125 \times 10^6}{\pi \times 182^2 \times 40.5}$$

$$\tau_{f_{(ind)}} = 1.699\ {}^{N}/_{mm^2}$$

$$\tau_{f_{(ind)}} = 1.699\ {}^{N}/_{mm^2} < 15\ {}^{N}/_{mm^2} \left(\tau_{f_{(allow)}}\right)$$

$$1.699\ {}^{N}/_{mm^2} \left(\ \tau_{f_{(ind)}}\right) < 15\ MPa \left(\ \tau_{f_{(allow)}}\right) \qquad Design\ is\ Safe.$$

as the induced shear stress $\tau_{f_{(ind)}}$ in the flange are less than this allowable (permissible) shear stress $\tau_{f_{(allow)}}$, therefore the design of flange(thickness) is safe.

FLEXIBLE COUPLING (BUSH-PIN TYPE)

The perfect alignment is rare to achieve in two shafts, The axial, angular and parallel are (he most common misalignments that exist either independently or in combination. The misalignments are shown in Figure 13. In such cases the flexible couplings are used and also absorb impact loads due to fluctuations in Torque and speed. The flexibly in a coupling can also be achieved1 due to the presence of some member itself and it named as incorporated flexibility. Pin or bush type coupling is the common example of flexible coupling.

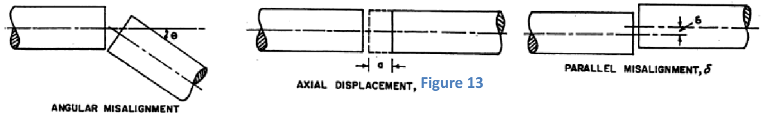
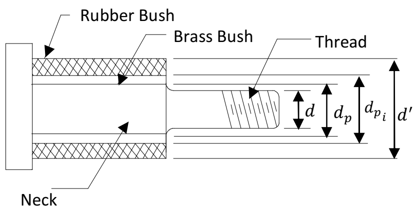


Figure 13

Bush type flexible coupling

This coupling is a modification of a rigid-type flange coupling in the sense that the design of the shaft and the hub of flanges are similar to that of rigid flange coupling (see Fig. 13.4c). In this, rubber bushes are provided to act as flexible elements, and to allow for a minor angular misalignment between the two shafts.



A brass sleeve of 2-3 mm thickness is used to reduce wear and tear of the rubber bush with pin. The rubber (6 to 10 mm thickness) with brass sleeve (3 to 5 mm thickness) is used over the pin and provides the desired flexibility in the Coupling

The flanges of the Coupling are not identical as shown in the figure. No socket and spigot are provided on the two flanges. Rather, there is a clearance of about 5 mm between the two faces of flanges. The designs of the two flanges are different because one flange contains holes for pins and bushes,

and the other flange contains holes for the threaded portion of the pin and the nut.

The diameter of the pin is also increased to increase the bearing area (bearing capacity) between bush and the flange hole. It should also be notice that the thickness of the flange containing bush is more to compensate the larger diameter of the hole made for bush. The pitch circle diameter of the pin is also to be on larger side to reduce the bearing pressure between bush and flange. the increased flexibility at the bush induce the possibility of bending of the pin; hence. the pin for bush type flexible coupling should also be designed for bending consideration. The driving flange carries the larger portion of the pin with rubber bush and brass sleeve. A protective cover is provided on this flange to ensure safety. The flanges are made of CI. The shaft, key, bolts/Pins are made of steel.

Design procedure for (bush pin) cast iron flexible flange coupling

The components to be designed in this coupling are (i). Shafts (ii). Bolts (iii). Key (iv). Hub (v) Flange and (vi) Bush.

Note : Refer Fig (13.4 c)P-233 A

i). Shaft design

Torque transmitted by the coupling

$$T = \frac{9.55 \times 10^6 (P)}{n} \quad N - mm \qquad E(3.3a)P\ 42$$

Power, P is in kW

and is equated to the equation for torque transmitted by the shaft diameter given by

Based on strength,

$$\tau = \frac{16\,T}{\pi\,\eta\,D^3} \qquad E(3.1)P\ 42$$

And *based on rigidity (stiffness)*

$$\theta = \frac{584\ T\ L}{G\ D^4} \qquad E(3.2)P\ 42$$

Where D = Shaft diameter.

The value of 'D' is round off to the next standard size Ref Table T(3.5a) P-48

ii). Key design

Based on the shaft diameter 'D' the value of width 'b' and thickness 'h' of taper key are selected from Table T(4.2) P- 61

Calculate *the effective length of key* = L = Hub length

$$\text{L} = 1.2\text{D} + 20\text{ mm} \qquad E(13.6)P\ 210$$

After selecting b, h and L for the key. Give a check for the induced shear stress and induced crushing stress in the key,

Check for key

For *Induced crushing stress in key*

$T = \frac{1}{4}\sigma_{b1} hdL \qquad E(4.5\ b)P\ 53$

$$\sigma_{k_{(ind)}} = \frac{4\ T}{h\ d\ L} \quad MPa$$

For *Induced shear stress in key*

$T = \frac{1}{2}\tau_1 bdL \qquad E(4.6)P\ 54$

$$\tau_{k_{(ind)}} = \frac{2\ T}{b\ d\ L} \quad MPa$$

$$if \qquad \tau_{k_{(ind)}} < \tau_{k_{(allow)}}, \qquad Design\ is\ Safe.$$

$$\sigma_{k_{(ind)}} < \sigma_{k_{(allow)}}, \qquad Design\ is\ Safe.$$

If the induced shear stress $\tau_{k_{(ind)}}$ and crushing stress $\sigma_{k_{(ind)}}$ in the key are less than this allowable (permissible) shear stress $\tau_{k_{(allow)}}$ and allowable (permissible) crushing stress $\sigma_{k_{(allow)}}$, the design of key is safe. Otherwise increase the key dimensions and check once again.

iii). Hub design

The hub diameter

$$D_1 = 1.8D + 20\ mm \qquad E(13.2)P\ 209$$

The hub length $\quad L = 1.2D + 20\ mm \qquad E(13.6)P\ 210$

iv). Pin / Bolt design

Number of bolts

$$i = \frac{1}{40}D + 2 \quad to \quad \frac{3}{80}D + 2 \qquad E(13.1)P\ 209$$

Even no of pins/ bolts as preferred where 'D' is diameter of shaft.

Determine *the minor diameter of the pins/ bolts 'd'* by the empirical formula

$$d = \frac{0.423D}{\sqrt{i}} + 7.5\ mm \qquad E(13.5)P\ 210$$

Select the size of bolt based on the core area of pins/ bolts (A_C)

Standardize the pins/ bolts size using table T(9.8) P- 113

Note: unless otherwise specified the shaft, key and pins/ bolts are made of same material and hence have same value of stress

$$T_{Shaft} = T_{Key} = T_{Bolt}$$

$$i.e.,\ \ \tau_{Shaft} = \tau_{Key} = \tau_{Bolt}$$

Calculate *the major diameter of the pin* by empirical relation

$$d_P = 2 \times Preliminary\ bolt\ diameter\ of\ the\ threade\ portion$$

$$d_P = 2 \times d$$

Assume the thickness of the brass bush t_b & t_r the rubber bush

Thickness of the brass bush $t_b = 1\ to\ 3\ mm$

Thickness of the rubber bush $t_r = 5\ to\ 10\ mm$

Inner diameter of the rubber bush $d_{Pi} = d_P + 2\ t_b$

Outer diameter of the rubber bush $d' = d_{Pi} + 2\ t_r$

Bolt circle diameter

$$D_2 = D_1 + 3.2d\ mm \qquad E(13.3)P\ 210$$

Check for bolt

Check for *induced shear stress in bolt*

$$T = \frac{\tau_{b_{(ind)}}\ \pi\ i\ d^2 D_2}{8} \qquad E(13.9)P\ 210$$

$$\tau_{b_{(ind)}} = \frac{T \times 8}{\pi\ i\ d^2 D_2}$$

$$if \qquad \tau_{b_{(ind)}} < \tau_{b_{(allow)}}, \qquad Design\ is\ Safe.$$

If the induced shear stress $\tau_{b_{(ind)}}$ in the bolt are less than this allowable (permissible) shear stress $\tau_{b_{(allow)}}$, the design of key is safe. Otherwise increase the bolt dimensions and check once again.

v). Flange design

Outer diameter of flange $D_3 = D_1 + 6d\ mm$ $E(13.4)P\ 209$

The flange thickness $t = 0.35D + 9\ mm$ $E(13.7)P\ 210$

Check for flange

$$T = \frac{\tau_{f_{(ind)}}\ \pi\ D_1^{\ 2}\ t}{2} \qquad E(13.11)P\ 210$$

$$\tau_{f_{(ind)}} = \frac{2 \times T}{\pi\ D_1^{\ 2}\ t}$$

$$if \qquad \tau_{f_{(ind)}} < \tau_{f_{(allow)}}, \qquad Design\ is\ Safe.$$

If the induced shear stress $\tau_{f_{(ind)}}$ in the flange are less than this allowable (permissible) shear stress $\tau_{f_{(allow)}}$, the design of key is safe. Otherwise increase the flange dimensions and check once again.

	Symbols (Flexible Protected)			
		Formulae	Equations	Table
(i). Design of Shaft				
T	Torque transmitted in N-mm	$\frac{9.55 \times 10^6 (P)}{n}$ P in kW	(3.3a) P-42	
η	Key way factor	Assume η = 0.75 mm		
D	Shaft Diameter	$\tau = \frac{16\ T}{\pi\ \eta\ D^3}$	(3.1)P-48	(3.5a)P-48
		$\theta = \frac{584\ T\ L}{G\ D^4}$	(3.2)P-48	

Design of Key				
b	Width of taper key			(4.2) P-61
h	Height of taper key			
L	Length of taper key ≈ Length of Hub	L= 1.2D + 20 mm	(13.6) P-210	(4.2) P-61
Check for key				
$\sigma_{k(induced)}$	Induced crushing stress in key	$T = \frac{1}{4}\sigma_{k(ind)}hdL$	(4.5b) P-53	
$\tau_{k(induced)}$	Induced shear stress in key	$T = \frac{1}{2}\tau_{k(ind)}bDL$	(4.6) P-54	
Design of Hub				
D_1	hub diameter	$D_1 = 1.8D + 20\ mm$	(13.2) P-209	
L	hub length	L= 1.2D + 20 mm	(13.6) P-210	
Design of Bolt				
No of bolts / pins	i	$i = \frac{1}{40}D + 2$ $to\ \frac{3}{80}D + 2$	(13.1) P-209	
d	Minor Diameter of bolt	$d = \frac{0.423D}{\sqrt{i}} + 7.5\ mm$	(13.5) P-210	(9.8) P-113
d_p	major Diameter of bolt	Assume $d_p = 2 \times d$		
t_b	Thickness of brass bush	Take $t_b = 3\ mm$		
t_r	Thickness of rubber bush	Take $t_r = 6\ mm$		
d_{pi}	Inside diameter of rubber bush	$d_{pi} = d_p + 2t_b$		

d'	Inside diameter of rubber bush	$d' = d_{pi} + t_r$		
D_2	Bolt circle diameter	$D_2 = D_1 + 3.2d\ mm$	(13.3) P-209	
F	Force on each pin	$T = i\ F\left(\frac{D_2}{2}\right)$		
F	Force on each pin	$F = P_p\ l\ d'$		
Check for bolt				
τ_p	shear stress in pin	$\tau_p = \frac{F}{\frac{\pi}{4}{d_p}^2}$		
C	Gap between flange	Take C = 5 mm		
σ_b	Bending stress in pin	$\sigma_b = \frac{F\left(\frac{l}{2} + C\right)}{\frac{\pi}{32}{d_p}^3}$		
σ_{Max}	Maximum normal stress in pin	$\sigma_{Max} = \sigma_1 = \frac{\sigma_b}{2} + \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + {\tau_p}^2}$	(1.11a)P-02	
τ_{Max}	Maximum shear stress in pin	$\tau_{Max} = \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + {\tau_p}^2}$	(1.12)P-02	
Design of flange				
D_3	Outer diameter of flange	$D_3 = D_1 + 6d\ mm$	(13.4) P-209	
t	Flange thickness	$t = 0.35D + 9\ mm$	(13.7) P-210	

Check for flange				
$\tau_{f(induced)}$	Induced shear stress in Flange	$T = \frac{\tau_{f(ind)}\ \pi\ D_1{}^2\ t}{2}$	(13.11) P-210	
t_1	Thickness of protective circumferential flange	Assume t_1 = 0.25D mm		

4. Design a flanged coupling, to transmit a power of 32 kW at 960 rpm. The overall torque is 20% greater than the mean torque. The allowable shear stress for the shaft, key and bolt is 40 MPa. The allowable shear stress in the CI flange is 15 MPa. Bearing pressure in the bush is 0.8 MPa.

(Dec. 2011) Marks 12

Data: P = 32 kW , n= 960 rpm , $\tau_s = \tau_k = \tau_b = 40\ MPa$, $\tau_f = 15\ MPa,\ P_b = 0.8\ MPa$, $T = T_{Max} = 1.2\,T_{Mean}$

Normally for ductile material $\sigma_{yt} = \sigma_{yc}$ and ∴ $\sigma_s = 2 \times \tau_s$

Allowable crushing stress in the key $\sigma_f = 2 \times \tau_f = 2 \times 40 = 80\ MPa$

Assume keyway factor $\eta = 0.75$

i). Shaft design

a) Max. torque transmitted by the coupling

$$T_{Mean} = \frac{9.55 \times 10^6 (P)}{n} \quad N-mm \qquad E(3.3a)P\ 42$$

$$T_{Mean} = \frac{9.55 \times 10^6 (32)}{960}$$

$$T_{Mean} = 3.1833 \times 10^5\ N-mm$$

$$T = T_{Max} = 1.2\,T_{Mean} = 1.2 \times 3.1833 \times 10^5$$

Torque $$T = 3.816 \times 10^5 N-mm$$

b) Shaft diameter ' D'

The equation for torque transmitted by the shaft diameter given by

Based on strength,

$$\tau = \frac{16\,T}{\pi\,\eta\,D^3} \qquad E(3.1)P\ 42$$

$$\tau_s = \frac{16\,T}{\pi\,\eta\,D^3}$$

$$40 = \frac{16 \times 3.816 \times 10^5}{\pi \times 0.75 \times D^3}$$

$$D = 36.5\ mm$$

Select standard diameter of shaft 'D' from T(3.5a) P- 48

$$D = 40\ mm$$

ii). Key design

a) Key dimensions (b , h and L)

Select taper key and taper 1:100 from Table T(4.2) P-61 for shaft diameter 'D' = 40 mm.

Width of key b = 12 mm *Thickness of key* h = 8 mm

Length of key = length of hub

$$\text{L} = 1.2\text{D} + 20\ \text{mm} \qquad E(13.6)P\ 210$$

$$\text{L} = 1.2 \times 40 + 20$$

$$\text{L} = 68\ \text{mm}$$

Preferred *length of key* L = 70 mm

(Length of key should not be less than the hub length) *b) Check for key*

For *Induced crushing stress in key*

$T = \frac{1}{4}\sigma_{b1} hdL$ $\quad E(4.5\, b)P\ 53$

$$\sigma_{k_{(ind)}} = \frac{4\,T}{h\,d\,L} \quad MPa$$

$$\sigma_{k_{(ind)}} = \frac{4\ \times 3.816 \times 10^5}{8\ \times 40\ \times 70}$$

$$\sigma_{k_{(ind)}} = 68.214\ MPa\ <\ 80\ MPa\left(\sigma_{k_{(allow)}}\right)$$

For *Induced shear stress in key*

$T = \frac{1}{2}\tau_1 bdL$ $\quad E(4.6)P\ 54$

$$\tau_{k_{(ind)}} = \frac{2\,T}{b\,d\,L} \quad MPa$$

$$\tau_{k_{(ind)}} = \frac{2\ \times 3.816 \times 10^5}{12\ \times 40\ \times 70}$$

$$\tau_{k_{(ind)}} = 22.738\ MPa\ <\ 40\ MPa\left(\tau_{k_{(allow)}}\right)$$

$$22.738\ MPa\left(\ \tau_{k_{(ind)}}\right) <\ 40\ MPa\left(\ \tau_{k_{(allow)}}\right) \qquad Design\ is\ Safe.$$

$$68.214\ MPa\left(\ \sigma_{k_{(ind)}}\right) <\ 80\ MPa\left(\ \sigma_{k_{(allow)}}\right), \qquad Design\ is\ Safe.$$

as the induced shear stress $\tau_{k_{(ind)}}$ and crushing stress $\sigma_{k_{(ind)}}$ in the key are less than this allowable (permissible) shear stress $\tau_{k_{(allow)}}$ and allowable (permissible) crushing stress $\sigma_{k_{(allow)}}$, the design of key is safe.

iii). Hub design

a) *The hub diameter*

$$D_1 = 1.8D + 20\ mm \qquad E(13.2)P\ 209$$

$$D_1 = 1.8 \times 40 + 20$$

$$D_1 = 92\ mm$$

b) *The hub length* $\quad L = 1.2D + 20\ mm \qquad E(13.6)P\ 210$

$$L =\ 1.2 \times 40 + 20$$

$$L = 68\ \ mm$$

iv). Design of bolt

a) *Number of bolt*

$$i = \frac{1}{40}D + 2\ \ to\ \ \frac{3}{80}D + 2 \qquad E(13.1)P\ 209$$

$$i = \frac{1}{40} \times 40\ + 2\quad to\quad \frac{3}{80} \times 40 + 2$$

$$i = 3\quad to\quad 3.5$$

It is preferable to provide minimum 6 pins for a flexible coupling to withstand the bending stress is 6.

Take *the number of bolts*, i = 6 [even number of bolts are preferred]

b) Bolt diameter

$$d = \frac{0.423D}{\sqrt{i}} + 7.5\ mm \qquad E(13.5)P\ 210$$

$$d = \frac{0.423 \times 40}{\sqrt{6}} + 7.5$$

$$d = 14.408\ mm$$

From table T(9.8) P 114 for core diameter d = 16 mm, the select diameter of bolt is 16 mm.

Standard bolt size is M16 x 2 & its core area $A_c\ = 157\ mm^2$

c) Major Diameter of bolt

Calculate the major diameter of the pin by empirical relation

$$d_P = 2 \times preliminary\ bolt\ diameter\ of\ the\ threade\ portion$$

Assume $d_p = 2 \times d$

$$d_p = 2 \times 16$$

$$d_p = 32\ mm$$

Assume the thickness of the brass bush t_b& $t_r$the rubber bush

Thickness of the brass bush $t_b = 3\ mm$

Thickness of the rubber bush $t_r = 6\ mm$

Inner diameter of the rubber bush $d_{Pi} = d_P + 2\ t_b$

$$d_{Pi} = 32 + 2 \times 3$$

$$d_{Pi} = 38\ mm$$

Outer diameter of the rubber bush $d' = d_{Pi} + 2\ t_r$

$$d' = 38 + 2 \times 5$$

$$d' = 50\ mm$$

d) Bolt circle diameter

$D_2 = D_1 + 3.2d\ mm$ $E(13.3)P\ 209$

$$D_2 = 92 + 3.2 \times 16$$

$$D_2 = 143.2\ mm$$

Force on each pin

$$T = i\ F\left(\frac{D_2}{2}\right)$$

$$F = \frac{2\ T}{i\ D_2}$$

$$F = \frac{2 \times 3.816 \times 10^5}{6 \times 143.2}$$

$$F = 889.2\ N$$

Also $F = P_b\ l\ d'$

$$889.2 = 0.8 \times l \times 48$$

$$l = 22.23\ mm$$

$$Say\ Length, \quad l = 24\ mm$$

Let the clearance between the faces of the flange as ‘C’ as 5 mm

Total length of rubber bush $l_1 = l + c$

$$l_1 = 24 + 5$$

$$l_1 = 29\ mm$$

e) Check for Pin

Shear stress induced in Pin

$$\tau_p = \frac{F}{\frac{\pi}{4} {d_p}^2}$$

$$\tau_p = \frac{889.2}{\frac{\pi}{4} \times 32^2}$$

$$\tau_p = 1.106\ MPa$$

Bending stress induced in Pin

$$\sigma_b = \frac{F\left(\frac{l}{2} + C\right)}{\frac{\pi}{32} {d_p}^3}$$

$$\sigma_b = \frac{889.2\left(\frac{24}{2}+5\right)}{\frac{\pi}{32}\times 32^2}$$

$$\sigma_b = 4.7\ MPa$$

∴ *Maximum normal stress induced in the pin*

$$\sigma_{Max} = \sigma_1 = \frac{\sigma_b}{2} + \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau_p{}^2}$$

$$\sigma_{Max} = \sigma_1 = \frac{4.7}{2} + \sqrt{\left(\frac{4.7}{2}\right)^2 + 1.106^2}$$

$$\sigma_{Max} = 4.947\ MPa < 80\ MPa$$

∴ *Maximum shear stress induced in the pin*

$$\tau_{Max} = \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau_p{}^2}$$

$$\tau_{Max} = \sqrt{\left(\frac{4.7}{2}\right)^2 + 1.106^2}$$

$$\tau_{Max} = 2.6\ MPa < 40\ MPa$$

$4.947\ MPa(\ \sigma_{Max}) < 80\ MPa(\ \sigma_{(allow)}),\qquad Design\ is\ Safe.$

$2.6\ {}^{N}/_{mm^2}(\ \tau_{Max}) < 60MPa(\ \tau_{(allow)})\qquad Design\ is\ Safe.$

as the maximum shear stress τ_{Max} and crushing stress σ_{Max} in the key are less than this allowable (permissible) shear stress $\tau_{(allow)}$ and allowable (permissible) crushing stress $\sigma_{(allow)}$, the design of key is safe.

v). Flange design

a. *Outer diameter of flange* $D_3 = D_1 + 6d\ mm \qquad E(13.4)P\ 209$

$$D_3 = 92 + 6\times 16$$

$$D_3 = 188\ mm$$

b. Flange thickness $\quad t = 0.35D + 9\ mm \qquad E(13.7)P\ 210$

$$t = 0.35\times 40 + 9$$

$$t = 23\ mm$$

c. Check for flange

$$T = \frac{\tau\ \pi\ D_1{}^2\ t}{2} \qquad E(13.11)P\ 210$$

$$T = \frac{\tau_{f(ind)}\ \pi\ D_1{}^2\ t}{2}$$

$$\tau_{f(ind)} = \frac{2\,T}{\pi\ D_1{}^2\ t}$$

$$\tau_{f(ind)} = \frac{2\times 3.816\times 10^5}{\pi\ \times\ 92^2\times 23}$$

$$\tau_{f(ind)} = 1.25\ {}^{N}/_{mm^2}$$

$$\tau_{f(ind)} = 1.25\ {}^{N}/_{mm^2} < 15\ {}^{N}/_{mm^2}\left(\tau_{f(allow)}\right)$$

$1.25\ {}^{N}/_{mm^2}\left(\ \tau_{f(ind)}\right) < 15\ {}^{N}/_{mm^2}\left(\ \tau_{f(allow)}\right)\qquad Design\ is\ Safe.$

as the induced shear stress $\tau_{f(ind)}$ in the flange are less than this allowable (permissible) shear stress $\tau_{f(allow)}$, therefore the design of flange(thickness) is safe.