COTTER JOINT AND KNUCKLE JOINTS, KEYS AND COUPLINGS

Chapter 6

Cotter Joint

Cotter joints are used to connect two rods subjected to axial loads. The axial loads may be tensile or compressive in nature. The joint is made by using a wedge shaped steel plate of rectangular section called *Cotter.*

A cotter joint of *socket* and spigot type has three parts namely

i) Socket.
ii) Spigot and
iii) Cotter.

A *spigot* having forged circular ends on both sides with a *collar* provided to take up compressive loads goes inside the *socket,* another *tubular* forged component. The *socket* and *spigot* are connected by using a cotter tapered on one side in the slots provided in the *socket* and *spigot.* The taper in the cotter varies from 1 in 25 to 1 in 50 and a *clearance of* 1 to 3mm is provided between the *slot and the cotter.*

Note: ***Since the socket and spigot are produced by forging (a costly process), the material selected for cotter should be less stronger than that of socket and spigot so that in case of failure, cotter fails before the another components fail and can be replaced at a lesser cost.***

Applications:

i) Used for connecting piston rod with cross head of a double acting steam engine.
ii) For cotter foundation bolts.
iii) For connecting two halves of a fly wheel.
iv) For the rods of steel structure.
v) For jigs and fixtures.

Design of socket and spigot type cotter joint

The dimensions of various components of cotter joint shown in figure-6.1 are obtained as shown below.

Figure-6.1

a) ***Diameter of rod (d):*** The diameter of rod is determined by considering the failure of rod in tension or compression assuming that the strength of rod in tension or compression as same.

$$\text{Load,} \quad F = \frac{\pi d^2}{4}\sigma_t$$

$$\text{i.e.,} \quad \sigma_t = \frac{4F}{\pi d^2} \quad \text{... E(17-62)}$$

Where, $\sigma_t = Tensile\ stress\ in\ rod\ in\ N/mm^2$

$F = Tensile/compressive\ load, in\ Newton$

$d = Diameter\ of\ rod, in\ mm$

Hence knowing the *load F* and the *permissible tensile stress* σ_t, the *diameter of rod d* can be calculated.

b) Diameter of spigot and thickness of cotter (refer figure-6.2)

Figure-6.2

Where, $d_1 = Diameter\ of\ spigot$

$t = Thickness\ of\ cotter$

The weakest point in socket is the place at which the slot is provided to accommodate cotter of thickness *t*. Considering the failure of spigot in tension, the load *F* is given by

$$F = \left(\frac{\pi}{4}d_1^2 - d_1 t\right)\sigma_t$$

$$\text{i.e.,} \quad \sigma_t = \frac{4F}{\left(\pi d_1^2 - 4d_1 t\right)} \quad \text{... E(17-63)}$$

Note: By assuming $t = \frac{d_1}{4}$ ***or*** $d_1 = 4t$, ***and knowing*** σ_t ***and F, the diameter of socket*** d_1 ***and thickness of cotter may be calculated. Refer figure-6.3.***

Check for stress:

Based on d_1 and t, the *cotter* may be checked for *compressive stress* given by

$$F = d_1 t \sigma_c \quad \text{... E(17-69)}$$

Or, $\left.\sigma_c\right)_{induced} = \frac{F}{d_1 t}$

If $\left.\sigma_c\right)_{induced} > \left.\sigma_c\right)_{permissible}$, enhance the values of t (obviously d_1) such that $\left.\sigma_c\right)_{induced} \leq \left.\sigma_c\right)_{permissible}$

c) Outside diameter of socket

Figure-8.3

By considering the failure of *socket* in tension (or compression), the *tensile stress* σ_t in the socket is given by

$$\sigma_t = \frac{4F}{\pi\left(d_3^2 - d_1^2\right) - 4t\left(d_3 - d_1\right)} \qquad \text{... E(17-64)}$$

Hence knowing d_1, t *(previously calculated)*, load *F* and *tensile stress* σ_t, the outside diameter of *socket* d_3 can be determined.

d) Width of cotter (b) (Figure-8.4)

Figure-8.4

The *cotter* fails by *double shear* and hence the *shear stress* in the *cotter* is given by,

$$\tau = \frac{F}{2bt}$$

i.e., $F = 2bt\tau$ ... E(17-65)

Hence knowing *F*, τ and the thickness *t*, the width of cotter may be determined as

$$b = \frac{F}{2t\tau}$$

e) Length of spigot

The length of spigot from the slot is obtained by considering the *double shear* at the rod end is given by,

$$a = \frac{F}{2d_1\tau}$$

Obtained from $\tau = \dfrac{F}{2ad_1}$ ... E(17-66)

f) ***The diameter of socket collar*** (d_4)

The *diameter* of *socket collar* is obtained by considering the crushing failure of *socket.* Figure-8.5

$$F = (d_4 - d_1)t\sigma_c$$

i.e., $$\sigma_c = \frac{F}{(d_4 - d_1)t}$$ ... E(17.70)

Hence, knowing σ_c, F, d_1 and t, d_4 can be obtained.

g) ***Width of spigot collar (e)***

The width of spigot *collar 'e'* and is obtained by considering shear failure.
The *shear stress* τ in socket is given by,

$$\tau = \frac{F}{\pi d_1 e}$$ ... E(17-72)

i.e., $$e = \frac{F}{\pi d_1 \tau}$$

h) ***Diameter of spigot*** (d_2)

The *diameter of spigot* d_2 is obtained by considering the crushing of *collar* given by

$$F = \frac{\pi}{4}\left(d_2^2 - d_1^2\right)\sigma_c$$

i.e., $$\sigma_c = \frac{4F}{\pi\left(d_2^2 - d_1^2\right)}$$ ... E(17-68)

i) ***Width of collar at socket end*** (c)

The width of collar at socket end is given by

$$C = \frac{F}{2\tau\left(d_4 - d_1\right)}$$

From $$\tau = \frac{F}{2c\left(d_4 - d_1\right)}$$ ... E(17-67)

Example 6.1: Design a cotter joint for socket and spigot type to sustain an axial load of 100kN. The material selected for the joint has the following design stresses.

$\sigma_t = 120MPa,\quad \sigma_c = 160MPa,\quad \tau = 80MPa$ ***(VTU-Jan/Feb 2003)***

Solution:

Figure-6.6

a) The axial stress in the rods is given by

$$\sigma_t = \frac{4F}{\pi d^2}$$ ... E(17-62)

Therefore, Diameter of rod

$$d = \sqrt{\frac{4F}{\pi\sigma_t}}$$

$$= \sqrt{\frac{4\times100\times10^3}{\pi\times120}} = 32.573mm$$

Use, $d = 35mm$

b) Axial stress (tensile stress) across the slot of the rod is given by

$$\sigma = \sigma_t = \frac{4F}{\pi d_1^2 - 4d_1 t}$$ ... E(17-63)

Assuming thickness of cotter

$$t = \frac{d_1}{4}$$

$$\sigma_t = \frac{4F}{\pi d_1^2 - 4d_1\times\frac{d_1}{4}} = \frac{4F}{\pi d_1^2 - d_1^2}$$

$$120 = \frac{4 \times 100 \times 10^3}{\pi d_1^2 - d_1^2}$$

$$d_1 = 40mm$$

Therefore, $t = \frac{d_1}{4}$

$$= \frac{40}{4} = 10mm$$

Check for stress

Crushing strength of cotter or rod is

$$F = d_1 t \sigma_c$$

Using $\sigma_c)_{allowable} = 160MPa,\ calculate\ d_1 t$

Therefore, $d_1 t = \frac{F}{\sigma_c}$

$$= \frac{100 \times 10^3}{160} = 62.5mm^2$$

Assuming $t = \frac{d_1}{4}$

$$d_1 t = d_1 \times \frac{d_1}{4}$$

$$= \frac{d_1^2}{4} = 62.5$$

Therefore, $d_1 = 50mm$

And $t = \frac{d_1}{4}$

$$= \frac{50}{4} = 12.5mm$$

c) Tensile stress across the slot of socket is

$$\sigma = \frac{4F}{\pi\left(d_3^2 - d_1^2\right) - 4t\left(d_3 - d_1\right)} \quad \text{... E(17-64)}$$

Where, $\sigma = \sigma_t = 120 = \dfrac{4F}{\pi\left(d_3^2 - 50^2\right) - 4\times 12.5\left(d_3 - 50\right)}$

$$120 = \frac{4F}{\pi\left(d_3^2 - 2500\right) - 50\left(d_3 - 50\right)}$$

Therefore, $d_3^2 - 15.92d_3 + 2765.03 = 0$

Therefore, $d_3 = \dfrac{15.92 \pm \sqrt{(15.92)^2 + 4\times 1\times 2765}}{4}$

$$= 61.142mm$$

Use, $d_3 = 62mm$

d) The strength of cotter in double shear is

$$F = 2bt\tau$$

Therefore, $b = \dfrac{F}{2t\tau}$

$$= \frac{100\times 10^3}{2\times 12.5\times 80} = 50mm$$

$$b = 50mm$$

e) Shear stress due to double shear at rod end is

$$\tau = \frac{F}{2ad_1} \quad \text{... E(17-66)}$$

$$a = \frac{F}{2\tau d_1}$$

$$= \frac{100\times 1000}{2\times 80\times 50} = 12.5mm$$

$$a = 12.5mm$$

f) ***Crushing stress induced in the socket or cotter is***

$$\sigma_c = \frac{F}{(d_4 - d_1)t}$$

i.e.,
$$160 = \frac{100 \times 10^3}{(d_4 - 50)12.5}$$

$$d_4 = 100mm$$

g) ***Shear stress induced at the socket end is***

$$\tau = \frac{F}{2c(d_4 - d_1)} \quad \text{... E(17.67)}$$

$$80 = \frac{100 \times 10^3}{2c(100 - 50)}$$

Therefore, $c = 12.5mm$

h) ***Bending stress, σ_b in the cotter is***

$$\sigma_b = \frac{F(d_1 + 2d_4)}{4tb^2} \quad \text{... E(17.74)}$$

$$= \frac{100 \times 10^3 (50 + 2 \times 100)}{4 \times 12.5 \times 50^2}$$

$$\sigma_b = 200\, N/mm^2$$

i) ***Shear stress in collar is***

$$\tau = \frac{F}{\pi d_1 e} \quad \text{... E(17.72)}$$

$$e = \frac{F}{\pi d_1 \tau}$$

$$= \frac{100 \times 10^3}{\pi \times 50 \times 80} = 7.957mm$$

j) Shear stress in socket is

$$\tau = \frac{F}{\pi d_1 h} \qquad \text{... E(17.73)}$$

$$h = \frac{F}{\pi d_1 \tau}$$

$$= \frac{100 \times 10^3}{\pi \times 50 \times 80} = 7.957mm$$

$$h = 8mm$$

Knuckle Joint (Also known as Pin Joint) Refer figure-6.7

Knuckle joints are used to connect two rods subjected to tensile loads and also compressive loads when properly guided. The axes of the rods may be collinear or intersecting and lying in the same plane. This joint is also capable of transmitting rotary and transverse motion.

Knuckle joint has three parts namely

i) The Eye.

ii) The Fork and

iii) The Pin.

One end of the rod is forged to the shape of a Fork (or double eye) and the other rod is forged to the shape of an eye. The eye end of the rod is inserted between the jaws of fork. The hole in the *fork* and the *eye* are aligned and a *pin* having a *collar* is inserted in the holes. A *taper pin* or a *split pin* is inserted in the hole. Both the *fork eye ends* are octagonal in shape for some length and square for the remaining length.

Knuckle joint finds application in

i) Connecting tensions rods of structures.

ii) Rods of valve gears, valve rods and eccentric rods.

iii) For joining the links of suspension bridges.

iv) For diagonal strays in boilers.

v) Links of cycle chain, etc.

Figure-8.7: Dimensions of knuckle Joint

a) The diameter of rod (d) is obtained by considering the tensile stress in rod given by,

$$\sigma_t = \frac{4F}{\pi d^2} \qquad \text{... E(17.46)}$$

i.e., $$d = \sqrt{\frac{4F}{\pi\sigma_t}}$$

b) ***The diameter of Knuckle pin (d_2) is obtained by considering the double shear of Pin (figure-6.8)***

The load $$F = 2\frac{\pi d_2^2}{4}\tau$$

i.e., $$\tau = \frac{2F}{\pi d_2^2} \qquad \text{... E(17-53)}$$

$$d_2 = \sqrt{\frac{2F}{\pi\tau}}$$

c) ***Thickness of eye: The thickness of eye (b) is determined by considering the compressive stress in the eye due to bearing pressure of pin.***

i.e., $$F = \sigma_c d_2 b$$

$$b = \frac{F}{\sigma_c d_2} \qquad \text{... E(17-51)}$$

d) ***Thickness of Fork end: The thickness of Fork end (a) is determined by considering compressive stress in the fork due to bearing pressure on the pin.***

i.e., $$\sigma_c = \frac{F}{2d_2 a} \qquad \text{... E(17-52)}$$

or $$a = \frac{F}{2\sigma_c d_2}$$

e) ***The outer diameter of eye: The outer diameter of eye (d_4) is calculate by considering the tensile stress in the net area of eye given by***

$$\sigma_t = \frac{F}{(d_4 - d_2)b} \qquad \text{... E(17-47)}$$

For which, $\left(d_4 - d_2\right) = \dfrac{F}{\sigma_t b}$

Or $d_4 = \dfrac{F}{\sigma_t b} + d_2$

Check for shear stress (τ) in eye

The *shear stress in eye is checked using equation*

$$\tau = \frac{F}{b\left(d_4 - d_2\right)} \quad \text{... E(17-48)}$$

$\tau)_{induced}$ in eye from the above equation is to be $\leq \tau_{permissible}$. If this condition is not satisfied, *increase the net area* of eye. i.e., $b\left(d_4 - d_2\right)$

Check for shear stress in fork end

After knowing the dimensions of the fork ends namely, d_4, d_2 and a, the fork end is checked for shear stress which is given by

$$\tau = \frac{F}{2a\left(d_4 - d_2\right)} \quad \text{... E(17-50)}$$

τ obtained from equation (17-50) must be less than or equal to $\tau_{permissible}$ for safe design.

i.e., $\tau_{induced} \leq \tau_{permissible}$

Maximum bending moment $M_{b\max}$

Figure-6.9

The maximum bending moment in the Pin is given by

$$M_b)_{\max} = \frac{Fb}{8} \quad \text{... E(17-54)}$$

And maximum bending stress $\sigma_b)_{\max}$ in Pin is obtained by assuming that the Pin is supported and loaded as shown in figure-6.9 above and the maximum bending moment occurs at the centre of Pin given by

$$\tau_b)_{\max} = \frac{4F_b}{\pi d_3^2} \quad \text{... E(17-55)}$$

Example 6.2: Design and sketch the assemble of a knuckle joint to connect two mild steel rods subjected to an axial pull of 100kN. The allowable stress for rods and Pin are 100MPa, 130MPa and 60MPa in tension, crushing and shear respectively. The bending of Pin is prevented by selection of proper fit. (VUT-February 2002)

Solution: ***Refer figure-6.10***

a) ***The tensile stress in the rod is given by,***

$$\sigma_t = \frac{4F}{\pi d^2} \quad \text{... E(17-46)}$$

Where, $d = Diameter\ of\ rod$

i.e., $d = \sqrt{\frac{4F}{\pi \sigma_t}}$

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

$$d = 35.68mm$$

Adopt $d = 36mm$

b) ***Shear stress in the knuckle pin is given by***

$$\tau = \frac{2F}{\pi d_2^2} \quad \text{... E(17-53)}$$

Therefore, $d_2 = \sqrt{\frac{2F}{\pi \tau}}$

$$= \sqrt{\frac{2 \times 100 \times 10^3}{\pi \times 60}} = 32.57mm$$

Use, $d_2 = 33mm$

c) ***Compressive stress in the key due to bearing pressure of the Pin is***

$$\sigma_c = \frac{F}{d_2 b} \qquad \text{... E(17-51)}$$

Therefore, $b = \dfrac{F}{d_2 \sigma_c}$

$$= \frac{100 \times 10^3}{33 \times 130} = 23.31mm$$

Say $b = 24mm$

d) ***The tensile stress in the net area of eye is***

$$\sigma_t = \frac{F}{(d_4 - d_2)b} \qquad \text{... E(17-47)}$$

$$100 = \frac{100 \times 10^3}{(d_4 - 33)24}$$

$$d_4 = 74.66mm$$

Use, $d_4 = 75mm$

e) ***Tensile stress in the net area of fork ends is***

$$\sigma_t = \frac{F}{2a(d_4 - d_2)} \qquad \text{... E(17-49)}$$

i.e., $100 = \dfrac{100 \times 10^3}{2a(75 - 33)}$

Therefore, $a = 11.90mm$

Use $a = 12mm$

Check for bending stress in pin

The maximum bending stress in pin is given by

$$\sigma)_{b\ inclined} = \frac{4Fb}{\pi d_2^3} \qquad \text{... E(17-55)}$$

$$= \frac{4 \times 100 \times 10^3 \times 24}{\pi (33)^3}$$

$$= 85.03\, N/mm^2 < 100\, N/mm^2$$

Design of Turn Buckle

Example 6.3: Design a turn buckle/coupler to sustain an axial tensile load of 100kN by selecting materials for its components.

Solution: ***Refer figure-6.11***

Figure-6.11

Step I: Selection of Material

Select 45C8 as material for both screw and nut for which

$$\sigma_{ut} = 618 - 696\, N/mm^2 \quad \text{... T(1-25)}$$

Assume, $\sigma_{ut} = 650\, N/mm^2$

Taking $\sigma_{yt} = 0.6\sigma_{ut}$

$$\sigma_{yt} = 0.6 \times 650 = 390\, N/mm^2$$

$$\tau_y = \frac{\sigma_{yt}}{2}$$

$$= \frac{390}{2} = 195\, N/mm^2$$

Therefore, $\tau_{ed} = Design\ shear\ stress = \dfrac{\tau_y}{factor\ of\ safety}$

$$= \frac{195}{3} = 65\, N/mm^2$$

Step II: Calculation of dimensions of turn buckle

a) Strength of the rods based on a core diameter d_c ***is***

$$F = \frac{\pi}{4} d_c^2 \sigma_t \qquad \text{... E(17-75)}$$

$$F = A_c \sigma_t$$

Where, $A_c = Core\ area\ of\ screw = \frac{\pi}{4} d_c^2$

Assuming $\sigma_t = \frac{\sigma_{ut}}{factor\ of\ safety}$

$$= \frac{650}{3} = 216.67\ N/mm^2$$

$$A_c = \frac{F}{\sigma_t}$$

$$= \frac{100 \times 10^3}{216.67} = 461.53 mm^2$$

Referring Table (18-8), Using square threads of Normal series, for $A_c = 707 mm^2$, *adopt* $d = 36mm$, $d_c = 36mm$, *and pitch* $= 6mm$.

Note: One of the screwed rod has L.H. of threads and the other R.H. threads.

b) The resistance of screwed portion of the coupler at each end against shearing is

$$F = \pi a d \tau \qquad \text{... E(17.76)}$$

Therefore, $100 \times 10^3 = \pi \times a \times 36 \times 54.8$

$$a = 16.13 mm$$

Where, length $a = d\ to\ 12.5d\ \ for\ steel\ nuts$ ... E(17.77a)

$$= 1.25d = 1.25 \times 36 = 45mm$$

For practical purposes used $a = 45mm$

c) The length of outside diameter of the coupler at the nut portion is

$$F = \frac{\pi}{4}\left(d_1^2 - d^2\right)\sigma_t \qquad \text{... E(17.79)}$$

$$100 \times 10^3 = \frac{\pi}{4}\left(d_1^2 - 36^2\right) 216.67$$

$$d_1 = 43.4mm$$

Use $d_1 = 45mm$

d) The outside diameter of the turnbuckle at the middle is obtained from

$$F = \frac{\pi}{4}\left(d_3^2 - d_2^2\right)\sigma_t \qquad \text{... E(17.79)}$$

Note: Assuming $d_2 = d_1$ ***from figure-6.11***

$$100 \times 10^3 = \frac{\pi}{4}\left(d_3^2 - 45^2\right)216.67$$

$$d_3 = 51.11mm$$

Use $d_3 = 52mm$

e) Total length of coupler $= l = 6d$

$$= 6 \times 36 = 216mm$$

Press and shrink fits

When two cylindrical parts are assembled by *shrinking* or *press fitting* one part over the other, a *contact pressure* is developed between the two parts. Refer figure-6.12

Figure-6.12

Figure-6.7 shows a cylindrical members which have been assembled with a *shrink fit*. The contact pressure *'p'* between the members at radius *'b'* cause radial stress σ_r in each member at the contacting surfaces.

The *tangential stress* at the *outer surface* of *inner member* is

$$\left.\sigma_t\right)_i = -P\left(\frac{b^2 + a^2}{b^2 - a^2}\right) \qquad \text{... (i)}$$

Similarly the *tangential stress* at the *inner surface* of *outer member* is

$$\left.\sigma_t\right)_0 = P\left(\frac{c^2 + b^2}{c^2 - b^2}\right) \qquad \text{... (ii)}$$

The above equation (i) and (ii) can be solved only if the contact pressure is known.

In obtaining a *shrink*, the diameter of the *male member* is made larger than *diameter* of *female member*. The difference in diameters is called the *Interference* and is the *deformation* which the two members must experience. The diameter of cylinders are known and hence the deformation should be introduced to evaluate stresses.

Let, $\delta_0 = \textit{Increase in radius of hole.}$

$\delta_i = \textit{Decrease in radius of inner cylinder.}$

$$\delta_0 = \frac{bp}{E_0}\left(\frac{c^2+b^2}{c^2-b^2}+8\right)$$

$$\delta_i = \frac{-bp}{E_i}\left(\frac{b^2+a^2}{b^2-a^2}-8\right)$$

Then total deformation δ is

$$\delta = \delta_0 - \delta_i$$

$$= \frac{bp}{E_0}\left(\frac{c^2+b^2}{c^2-b^2}+8\right)+\frac{bp}{E_i}\left(\frac{b^2+a^2}{b^2-a^2}-8\right)$$

If both cylinders are made of same material,

$$E_0 = E_i = E$$

$$\delta = \frac{bp}{E}\left(\frac{c^2+b^2}{c^2-b^2}+8\right)+\frac{bp}{E}\left(\frac{b^2+a^2}{b^2-a^2}-8\right)$$

$$= \frac{bp}{E}\left[\frac{c^2+b^2}{c^2-b^2}+\frac{b^2+a^2}{b^2-a^2}\right]$$

$$= \frac{bp}{E}\left[\frac{\left(c^2+b^2\right)\left(b^2-a^2\right)+\left(c^2-b^2\right)\left(b^2+a^2\right)}{\left(c^2-b^2\right)\left(b^2-a^2\right)}\right]$$

$$= \frac{bp}{E}\left[\frac{c^2b^2-b^2a^2+c^2b^2-a^2b^2}{\left(c^2-b^2\right)\left(b^2-a^2\right)}\right]$$

$$= \frac{bp}{E}\left[\frac{2b^2c^2-2a^2b^2}{\left(c^2-b^2\right)\left(b^2-a^2\right)}\right]$$

$$= \frac{bp}{E}\left[\frac{2b^2\left(c^2-a^2\right)}{\left(c^2-b^2\right)\left(b^2-a^2\right)}\right]$$

$$p = \frac{E\delta}{b}\left[\frac{\left(c^2-b^2\right)\left(b^2-a^2\right)}{2b^2\left(c^2-b^2\right)}\right]$$

Substituting the value of P in $\left.\sigma_t\right)_i = P\left(\frac{b^2+a^2}{b^2-a^2}\right)$ we get

$$\left.\sigma_t\right)_i = \frac{-\delta E}{b}\left[\frac{\left(c^2-b^2\right)\left(b^2-a^2\right)}{2b^2\left(c^2-b^2\right)}\right]\left[\frac{b^2+a^2}{b^2-a^2}\right]$$

$$\left.\sigma_t\right)_i = \frac{-\delta E}{2b^3}\left[\frac{\left(c^2-b^2\right)\left(b^2-a^2\right)}{c^2-a^2}\right]$$

Similarly, $$\left.\sigma_t\right)_0 = \frac{E\delta}{b}\left[\frac{\left(c^2-b^2\right)\left(b^2-a^2\right)}{2b^2\left(c^2-b^2\right)}\right]\left[\frac{c^2+b^2}{c^2-b^2}\right]$$

$$\left.\sigma_t\right)_0 = \frac{E\delta}{2b^3}\left[\frac{\left(b^2+c^2\right)\left(b^2-a^2\right)}{c^2-a^2}\right]$$

Design of Keys

Keys are temporary fasteners used to hold Gears, Pulleys, Flanges, etc., on rotating shafts such that there is no relative rotational movement between them. These are generally made of same material (such as carbon steel) as that of shaft and help in transmission of power. The keys are generally of rectangular section with or without taper. Half the depth (or thickness) of key is accommodated in the key seat made in the shaft and remaining thickness in the key way (key slot) made in the hub of the member such as a Pulley to be keyed to the shaft. The main function of the keys is to transmit torque from a rotating shaft to the member mounted on the shaft such as Pulleys or Gears or flanges of a coupling.

The key fails by shear or crushing while transmitting torque from the shaft. Hence it acts as a safety device so that it fails earlier to the other members such as shaft or pulley. It is cheaper and can be replaced easily.

Classification of Keys

Based on the shape and purpose, for which keys are used, keys are broadly classified as

i) Sunk keys – used for heavy duty.

ii) Saddle keys, Woodruff keys, Pins – used for light duty.

i) Sunk Keys

It is a key which goes partly in the key seat made in the shaft and partly in the key way made in the hub. Most of the keys are included in this category. *Rectangular key* (Figure-6.13), and *square key* (figure-6.14), with or without taper are the commonly used sunk keys. A *rectangular key* without taper is called as a *rectangular parallel key* and *rectangular key* having taper on its top face along its length is called a *tapered key.* A *Gib head key* with taper provided along its length also falls under this category.

Rectangular Parallel Key (Figure-6.13)

Figure-6.13

Figure-6.14

Gib head Key (Figure-6.15 and Figure-6.16)

Gib head key is a sunk key tapered on the upper surface and provided with a Gib head on one end to facilitate the withdrawal of key from key way without disturbing the alignment of hub. Gib head keys are used in places where the thinner end of the key is inaccessible for removal. It can be removed by hammering a wedge shaped piece inserted between the hub and the Gib head in the space marked '*A*' in the figure-6.16.

Figure-6.15

Proportions of Gib head Key

If *'d' is the shaft diameter, then*

Key width $b = 0.25D + 2mm$

Thickness $h = Depth\ of\ key = 0.66b$

Total height of key $= h_1 = 1.75h$

Figure-6.16

Woodruff Key (Figure-6.17)

Woodruff key is a flat segmental metallic disc with ground or flat bottom. A key way of segmental type is cut in the shaft using a milling cutter such that the radius of the segment is equal to the diameter of the shaft. It is an adjustable sunk key and hence it can tilt in the key way so as to line up with the key way provided in the hub of the mating member such as Pulley or Gear. Woodruff keys are used for light loads. It's extra depth in the shaft prevents and tendency to turnover in its key way (which is generally found in rectangular keys). It is also to be noted that the increase in the depth of keyway in the shaft reduces the strength of the shaft.

Figure-6.17

Feather Keys (Figure-6.18 and Figure-6.19)

Feather keys are parallel keys and are used when it is required to have relative axial motion between the shaft and the mating member. The key is made a tight fit in the shaft and fastened in either the shaft or the hub and made a sliding fit in the mating member. Thus, a feather key enables turning moment to be transmitted from the shaft to the hub of mating member and vice-versa and also it enables the mating member such as Pulley or Gear to move axially along the shaft.

Figure-6.18

Figure-6.19

Designation of key

A key is designated by specifying the *width, thickness and length of key.*

A parallel key having width 20mm, thickness 12mm and length 120mm used with a shaft of diameter 70mm is designated as parallel key 20 (width) *(b)* $\times$ 12 (thickness) *(h)* $\times$ 120 (length) *(l)*.

Note: Of the three dimensions, the last dimension is the thickness, the maximum dimension is length of key and the remaining dimension is width.

Design of Splines

The torque transmitted by a straight sided spline is obtained from the equation,

$$M_t = \frac{1}{2} phli\left(D - h\right) \qquad \text{... E(17.20)}$$

The designations for splines is available in Table T(17-14)

The dimensions d, D and width of spline are obtained from T(17-14)

The bearing pressure required in equation (17-20) is obtained from T(17-19)

Example 6.4: Derive from fundamentals, the equations for torque transmitted by a key in shear and compression.

Figure-6.20

Solution:

Consider a shaft of radius r (Diameter, d) transmitting torque M_t. Let b, h and l be the width, thickness (depth) and length of the key respectively. Let F_t be the tangential force due to M_t.

Note: ***The key fails either by shear or compression. Considering the failure by shear (Figure-6.21)***

Figure-6.21

$$M_{t\,shear} = F_{shear} \times radius$$

$$= (shear\ \ stress \times Area\ \ under\ \ shear) \times radius$$

Area under shear $= bl$

Therefore, $M_{t\,shear} = (\tau bl) r$

Therefore, $M_{t\,shear} = bl\tau \dfrac{d}{2}$ '

Where, τ is the allowable shear stress.

Considering compression (or crushing) (figure-6.22)

$$M_{t\,compression} = F_{compression} \times radius$$

$$= (Compressive\ \ stress \times Area\ \ under\ \ crushing) \times radius$$

Area under compression (or crushing) [shown hatched]

$$= \frac{h}{2} l$$

$$= \left[\sigma_c \frac{h}{2} l\right] \frac{d}{2}$$

Where, $\sigma_c = Allowable\ compressive\ stress$

Therefore, $M_{t\,compression} = \dfrac{h}{2} l \sigma_c \dfrac{d}{2}$

Example 6.5: Prove that a square key is equally strong in shear and compression.

Solution:

Note: Refer figure-6.20

Consider a key of width *b*, thickness *h* and length *l* transmitting torque M_t.
The torque transmitted by the key in shear is

$$M_{t\,shear} = bl\tau \frac{d}{2} \qquad \text{... (1)}$$

Similarly, the torque transmitted by the key in compression is

$$M_{t\,compression} = \frac{h}{2} l\sigma_c \frac{d}{2} \qquad \text{... (2)}$$

Therefore,
$$\frac{M_{t\,shear}}{M_{t\,compression}} = \frac{bl\tau \frac{d}{2}}{\frac{h}{2} l\sigma_c \frac{d}{2}}$$

$$= \frac{2b\tau}{h\sigma_c}$$

$$= \frac{b(2\tau)}{h(\sigma_c)}$$

For a square key, $b = h$ and generally $\sigma_c = 2\tau$ for ductile materials. With this,

$$\frac{M_{t\,shear}}{M_{t\,compression}} = 1$$

i.e., $M_{t\,shear} = M_{t\,compression}$

Therefore, a square key is equally strong in shear and compression.

Example 6.6: Determine the length l of a square key required in terms of shaft diameter if the shaft and key are made of same material and of equal strength.

Solution:

Note: Refer figure-6.20

If *d* is the shaft diameter, then the torque transmitted by shaft is

$$M_{t\,shaft} = \frac{\pi d^3}{16} \eta \tau_s \qquad \text{... (1)}$$

Where, $\eta = Key\ way\ factor.$

$\tau_s = Shear\ stress\ for\ shaft\ material.$

Keyway factor

Considering two *shafts* of diameter *d* with one having a keyway cut along its length and the other without keyway. Obviously the shaft without keyway will be able to transmit more torque than the shaft with keyway due to stress concentration. This reduction in torque transmitting capacity is generally 25% and hence keyway factor, η is taken as 0.75.

Considering the key, from the stand point of shear

$$M_{t\,shear} = bl\tau_k \frac{d}{2} \quad \text{... (2)}$$

Where, $\tau_k = Shear\ stress\ in\ key.$

b *and* l *are width and length of key respectively.*

For equal strength,

$$Torque\ transmitted\ by\ shaft = Torque\ transmitted\ by\ key$$

i.e., $M_{t\,shaft} = M_{t\,key}$

i.e., $M_{ts} = M_{tk}$

$$\frac{\pi d^3}{16}\eta\tau_s = bl\tau_k \frac{d}{2}$$

Here, $\tau_s = \tau_k$, because shaft and key are made of same material.

Therefore, $\pi \frac{d^2}{8} \times \eta = bl$

Note: ***Generally, the width of key*** $b = \frac{1}{4}$ ***(shaft diameter) i.e.,*** $b = \frac{d}{4}$.

With this, $\frac{\pi d^2}{8} \times 0.75 = \frac{d}{4} l$

$$l = 1.178d.$$

This is the relation between the length of key and the shaft diameter for a square key.

Example 6.7: A rectangular key 12mm wide and 9mm thick is required to transmit a torque of 700N-m from a shaft of diameter 40mm. Taking allowable values of stress in

shear and compression as 58 and 110MPa, determine the length of the key required. If the key is 9mm wide and 12mm thick, what is the length of the key required?

Solution:

When the width of key is more than depth i.e., when $b > h$, the key will fail by compression before it fails by shear.

Therefore, $M_{t\ key\ in\ compression} = \frac{h}{2} l \sigma_c \frac{d}{2}$

$$700 \times 10^3 = \frac{9}{2} \times l \times 110 \times \frac{40}{2}$$

Therefore, *Length of key* $l = 70.707 mm$

When the depth of key is 12mm and width 9mm then $h > b,$ the key will fail by shear before it fails by crushing.

Therefore, $M_{t\ key\ in\ shear} = bl\tau_k \frac{d}{2}$

$$700 \times 10^3 = \frac{9}{2} \times l \times 58 \times \frac{40}{2}$$

Therefore, The length of key, $l = 67.05 mm$

Adopt $l = 70.707 mm$ (higher of the two values).

Example 6.8: Determine the dimensions of a steel tapered key to transmit 10kW at 1400rpm. The allowable shear and compressive stresses may be taken as 50 and 110MPa respectively. Also calculate the axial force required to drive the key home (keyway) or, the axial force required to assemble the key.

Solution:

i) To find the shaft diameter 'd'

We have torque

$$M_t = \frac{9550N}{1} \text{ N-m} \qquad \text{... E(19.3c)}$$

$$= \frac{9550 \times 10}{1400} = 68.21 \text{N-m}$$

$$= 68.21 \times 10^3 \text{ N-mm}$$

If *d* is the shaft diameter, then

$$M_t = \frac{\pi d^3}{16}\eta\tau_s$$

Where, $\eta = Keyway\ \ factor = 0.75(assumed)$

$\tau_s = 50MPa = 50\,N/mm^2$

Therefore, $68.21\times10^3 = \frac{\pi d^3}{16}\times0.75\times50$

$d = 21.002mm$

Adopt standard diameter $d = 25mm$

ii) To find the dimensions of key i.e., width b, thickness h and length l.

For shaft diameter $d = 25mm,$ referring to *Table of tapered key,* ... T(1-4)

We have, *Width* $b = 8mm$

Thickness $h = 7mm$

Note: Since the width of key is more than the depth i.e., $b > h,$ ***the key fails by compression before it fails by shear.***

$$M_{t\ key\ in\ compression} = \frac{h}{2}l\sigma_c\frac{d}{2}$$

$$68.21\times10^3 = \frac{7}{2}\times l\times110\times\frac{25}{2}$$

$l = 14.17mm$

Adopt length of key $l = 15mm$

Therefore, the key dimensions are,

Width, $b = 8mm$

Thickness $h = 7mm$

And *Length* $l = 15mm$

The axial force F_a necessary to drive the key home is given by

$$F_a = 2\mu_2 F + F\tan\beta \qquad \text{... E(17.11)}$$

Where, $\mu_2 = 0.1,\ \tan\beta = 0.0104$ if the taper is 1 in 100 (standard taper)

Therefore, $F_a = 2\times0.1\times F + F\times0.0104$

$$= 0.2104F$$

Note: ***For rectangular parallel key, the cross section of key is uniform throughout the length of key and no taper exists. i.e.,*** $\beta = 0$. ***For these keys,*** $\tan\beta = \tan 0 = 0$ ***and the equation for axial force becomes,***

$$F_a = 2\mu_2 F + 0 \text{ [from E(17.11)]}$$

$$= 2\times 0.1F = 0.2F$$

To find F, the pressure between the shaft and the hub

We have, $F_t = \mu_1 F$ ... E(17.7)

Therefore, $F = \dfrac{F_t}{\mu_1}$

Where, $F_t = \dfrac{M_t}{r}$

$$= \frac{68.21\times 10^3}{\frac{21.5}{2}} = 6345.12\,N$$

Assuming $\mu_1 = 0.3m$

$$F_t = \frac{6345.12}{0.3} = 21150.4N$$

$$F_a = 0.2104F \times 21150.4$$

$$= 4450N$$

Therefore, the axial force required to drive the key home $= 4450N$.

Example 6.9: A square key is required to transmit a torque of 400Nm from a shaft of 40mm diameter. If the length of the hub is 70mm, determine the dimensions of the side of the square key if key and shaft are made of same material having an allowable shear stress of $60\,N/mm^2$.

Solution:

$$M_{t\,key} = bl\tau_k \frac{d}{2} = M_{t\,shaft}$$

Here, $M_t = 400\text{N-m} = 400 \times 10^3 \text{ N-mm}$

$l = Length\ of\ key = Length\ of\ hub = 70mm$

$d = Shaft\ diameter = 40mm$

$b = Width\ of\ key$

$\tau_k = Shear\ stress\ in\ key = 60\ N/mm^2$

Substituting the values in the above equation we get,

$$400 \times 10^3 = b \times 70 \times 60 \times \frac{40}{2}$$

i.e., $b = 4.76mm$

Use $b = 5mm$

Therefore, $Side\ of\ square\ key = 5mm$

Example 6.10: If a shaft and key are made of same material, determine the length of key required in terms of shaft diameter taking width of key $b = \frac{d}{4}$ ***and thickness of key*** $h = \frac{3}{16}d$.

Solution:

If d is the shaft diameter, then the torque transmitted by the shaft is

$$M_{t\,shaft} = \frac{\pi d^3}{16}\eta\tau_s \qquad \text{... E(19.2)}$$

$$= \frac{\pi d^3}{16} \times 0.75 \times \tau_s \qquad \text{... (1)}$$

Note: Since the width of key is more than depth h, i.e., $b = \frac{d}{4} > h = \frac{3}{16}d$, ***the key fails by compression before if fails by shear***

Therefore, $M_{t\ key\ in\ compression} = \frac{h}{2} l \sigma_c \frac{d}{2}$

$$= \left(\frac{3d}{16} \times \frac{1}{2}\right) l \sigma_c \frac{d}{2} \qquad \text{... (2)}$$

Equating (1) and (2),

$$\frac{\pi d^3}{16} \times 0.75\tau_s = \left(\frac{3d}{16} \times \frac{1}{2}\right) l\sigma_c \frac{d}{2}$$

Using, $\sigma_c = 2\tau_s$,

$$\frac{\pi d^3}{16} \times 0.75\tau_s = \left(\frac{3d}{16} \times \frac{1}{2}\right) l\left(2\tau_s\right) \frac{d}{2}$$

$$\pi d \times 0.75 = 3 \times \frac{l}{2}$$

Therefore, $l = \frac{\pi d \times 0.75 \times 2}{3}$

Therefore, the length of the key $l = 1.57d$

Example 6.11: A 20kW, 1440rpm motor has a steel shaft. The diameter of the shaft is 45mm. The motor pull out torque is 3.5 times the average full load torque of motor. Assuming an yield shear stress of 55MPa and crushing stress of 108MPa for key material, design the key. Also determine the effect of keyways. Adopt a factor of safety of 2.5.

Solution:

Average torque of motor

$$M_t = \frac{9550 N \times 10^3}{n} \text{ N-mm}$$

$$= \frac{9550 \times 20 \times 10^3}{1440} \text{ N-mm}$$

$$= 132638.89 \text{ N-mm}$$

Pull out torque $= 3.5$ times average torque

$$= 3.5 \times 132638.89 \text{ N-mm}$$

$$= 464236.12 \text{ N-mm}$$

For shaft diameter $d = 45mm$

We have from table, T(17-4)

Width of key $b = 14mm$

Thickness of key $h = 9mm$

$$M_{t\ key\ in\ shear} = bl\tau_k \frac{d}{2}$$

Where, $\tau_y = Yield\ shear\ stress = 55MPa$

Design of Shear stress

$$\tau_{yd} = \frac{\tau_y}{factor\ of\ safety} = \tau_k$$

$$= \frac{55}{2.5} = 22MPa$$

Substituting in the above equation

$$464236.12 = 14 \times l \times 22 \times \frac{45}{2}$$

$$l = 66.99 \approx 67mm$$

$$M_{t\ key\ in\ compression} = \frac{h}{2} l \sigma_c \frac{d}{2}$$

$$464236.12 = \frac{9}{2} \times l \times 108 \times \frac{45}{2}$$

$$l = 42.45mm$$

Therefore the length of key $= 67mm$ (Higher of the two values)

Effect of Keyway

Torque transmitted by a solid shaft without keyway is given by

$$= \frac{\pi d^3}{16} \tau_s$$

Assuming the same material for both shaft and key

$$\tau_s = \tau_k$$

Therefore, Torque transmitted

$$= \frac{\pi (45)^3}{16} \times 55$$

$$= 393631.74 \text{ N-mm}$$

Torque transmitted by shaft with keyway

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

Taking keyway factor $\eta = 0.75$,

Torque $$= \frac{\pi (45)^3}{16} \times 0.75 \times 22$$

$$= 295223.81 \text{ N-mm}$$

Therefore, there is a 25% reduction in torque transmitted by the shaft with keyway.

Example 6.12: A 100mm diameter shaft running at 100rpm transmits 225kW. Power is taken off through a Gear whose hub is 200mm long. The key is made of steel having an ultimate shear stress of $350\ N/mm^2$ ***. Using a factor of safety of 5, determine the width of key required.***

Solution:

The torque transmitted by shaft is

$$M_{ts} = \frac{9550 N \times 10^3}{n} \qquad \text{... E(19.2)}$$

$$= \frac{9550 \times 225 \times 10^3}{100} = 21487500 \text{ N-mm}$$

$$M_{t\ key\ in\ shear} = bl\tau_k \frac{d}{2}$$

Where, length of key

$$= l = Length\ of\ \ hub = 200mm$$

$$\tau_k = \frac{\tau_{ult}}{factor\ \ of\ \ safety}$$

$$= \frac{350}{5} = 70MPa$$

Therefore, $$21487500 = b \times 200 \times 70 \times \frac{100}{2}$$

$$b = 30.7mm$$

Therefore width of key required $= 30.7mm$

Example 6.13: A splined connection in an automobile transmission consists of 10 splines cut on a 60mm shaft. The height of each spline is 6mm and the keyways in the hub are 50mm long. Determine the power transmitted at 3000rpm if the allowable normal pressure on the splines is limited to 5MPa.

Solution: (Refer figure)

The torque transmitted by a multi splined shaft is given by

$$M_t = \frac{1}{2} phli(D-h) \quad \text{... E(17.20)}$$

Where, $p = Normal\ pressure\ on\ splines = 5MPa$

$h = Thickness\ of\ splines = 6mm$

$l = Length\ of\ splines = 50mm = Length\ of\ hub$

$i = Number\ of\ splines = 10$

$D = Shaft\ diameter = 60mm$

Therefore, $M_t = \frac{1}{2} \times 5 \times 6 \times 50 \times 10 \times (60-6)$

$M_t = 405000\text{ N-mm} = 405\text{ N-m}$

But, $M_t = \frac{9550N}{3000};\ M_t\ is\ in\ \text{N-m}$

$N = 127.255kW$

Therefore, Power transmitted $= 127.255kW$

Example 6.14: Determine the power transmitted by a $6 \times 46 \times 52$ ***splined shaft at 2500rpm if the permissible normal pressure on splines is*** $7\ N/mm^2$. ***The keyways in the hub are 50mm long.***

Figure-6.23

Solution:

Note: *A* $6 \times 46 \times 52$ ***splined shaft has***

$i = 6$ splines on cut on a shaft of diameter $D = 52mm.$

$d = 46mm$

With depth of spline

$$h = \frac{D-d}{2}$$

$$= \frac{52-46}{2} = 3mm$$

The torque transmitted by a splined shaft is given by

$$M_t = \frac{1}{2}phli(D-h) \qquad \text{... E(17.20)}$$

$$= \frac{1}{2} \times 7 \times 3 \times 50 \times 6(52-3)$$

$$= 154350 \text{ N-mm}$$

But, $M_t = \frac{9550N}{n} \times 10^3;\ M_t\ is\ in$ N-mm

$$154350 = \frac{9550N}{2500} \times 10^3$$

$$N = 40.41kW$$

Therefore, Power transmitted by the splined shaft $= 40.41kW$

Example 6.15: A splined joint in an automobile has a specification of $8 \times 42 \times 48$***. The allowable bearing pressure on the surface of contact is*** $10\ N/mm^2$ ***. Determine the torque that can be transmitted over a contact length of 60mm. Also determine the power that can be transmitted at 750rpm.***

Solution:

i) To determine the torque transmitted by the splined shaft

$$M_{t\ splined\ shaft} = \frac{1}{2}phli(D-h)$$

Where, $p = Bearing\ pressure = 10\ N/mm^2$

$h = Thickness\ of\ spline, in\ mm$

Note: A spline designated by $8 \times 42 \times 48$ ***has*** $i = 8,\ d = 42mm$ ***and*** $D = 48mm.$

$$h = \frac{D-d}{2}$$

$$= \frac{48-42}{2} = 3mm$$

$$l = Length\ of\ \ spline = 60mm$$

Therefore, $\left.M_t\right)_{splined\ shaft} = \frac{1}{2} \times 10 \times 3 \times 60 \times 8(48-3)$

$$= 324000\text{N-mm}$$

ii) To find power transmitted, (N)

$$M_t = 9500 \times \frac{N}{n} \times 10^3$$

Therefore, $N = \frac{M_t n}{9550} \times 10^3 kW$

$$N = \frac{324000 \times 750}{9550} \times 10^3$$

$$= 25.45kW$$

Example 6.16: A square key is to be used to key a gear to a 35mm diameter shaft. The hub length of Gear is 60mm. Both shaft and key are made of same material having an allowable shear stress of 55MPa. What are the minimum dimensions of the sides of square key if 395N-m torque is to be transmitted? (VTU-Jan/Feb 2004)

Solution:

Note: A square key is equally strong in shear and compression.

i.e., $M_{t\ key\ in\ shear} = M_{t\ key\ in\ compression}$

$$M_{t\ key\ in\ shear} = bl\tau_k \frac{d}{2}$$

i.e., $395 \times 10^3 = b \times 60 \times 55 \times \frac{35}{2}$

Therefore, $b = 6.85mm = h$

Therefore, width of square key $b = 6.85mm$ = Depth of square key h.

Example 6.17: A square key is used to couple two commercial steel shafts of diameter 50mm in a flange coupling. Determine,

i) ***The side of key.***
ii) ***The power that can be transmitted at 1200rpm.***

Solution:

i) To find the side of key (b)

Torque transmitted by shaft

$$M_t)_{shaft} = \frac{\pi d^3}{16}\eta\tau_s$$

Where, $d = 50mm$

$\eta = Keyway\ \ factor = 0.75$

$\tau_s = 40MPa\ \ for\ commercial\ steel\ shaft\ as\ per\ ASME\ code$

Therefore, $$M_t)_{shaft} = \frac{\pi(50)^3}{16}\times 0.75\times 40$$

$$= 736310.8 \text{ N-mm}$$

$$M_t)_{key\ in\ shear} = bl\tau_k\frac{d}{2}$$

Assuming same material for both shaft and key and equation $M_t)_{shaft} = M_t)_{key}$

$$7363108.8 = bl\times 40\times\frac{50}{2}$$

Note: For square key, $l = 1.178d$

Therefore, $$736310.8 = b\times(1.178\times 50)\times 40\times\frac{50}{2}$$

Therefore, $b = 12.5mm = h$

ii) To determine the power transmitted (N)

Torque $$M_t = 9550\frac{N}{n}\times 10^3 \text{ N-mm}$$

i.e., $$736310.8 = 9550\times\frac{N}{1200}\times 10^3$$

$$N = 92.52kW$$

Example 6.18: A square key of length 75mm is used in a flange coupling to transmit power at 1440rpm. Determine,

- ***i) The diameter of solid shaft required.***
- ***ii) The power transmitted and***
- ***iii) The dimensions of key.***

Solution:

i) To determine the diameter of solid shaft (d)

The torque transmitted by shaft is

$$M_t)_{shaft} = \frac{\pi d^3}{16} \eta \tau_s \qquad \text{... E(19.2)}$$

Note: A square key is equally strong in shear and compression and the length of key in terms of shaft diameter d is given by

$$l = 1.178d \qquad (Proved\ already)$$

Therefore,
$$d = \frac{l}{1.178}$$

$$= \frac{75}{1.178} = 63.67mm$$

To find τ_s, ***the shear stress***

Assuming 40C8(C40) as the material of shaft and key,

$$\sigma_{ut} = 600MPa \qquad \text{... T(1.25)}$$

Assuming
$$\sigma_y = 0.6\sigma_{ut}$$

$$\sigma_y = 0.6 \times 600 = 360MPa$$

$$\tau_y = \frac{\sigma_y}{2}$$

$$= \frac{360}{2} = 180\,N/mm^2$$

$$\tau_s = \frac{\tau_y}{factor\ of\ safety}$$

$$= \frac{180}{3} = 60\ N/mm^2$$

Therefore, $\left.M_t\right)_{shaft} = \frac{\pi d^3}{16} \times 65^3 \times 0.75 \times 60$

$= 2426512.17$ N-mm

ii) To calculate the power transmitted (N)

Torque $\quad M_t = 9550 \frac{N}{n} \times 10^3$

i.e., $\quad 2426512.17 = \frac{9550 \times N \times 10^3}{1440}$

Therefore, $\quad N = 365.88kW$

iii) To find the dimensions of key $(i.e., b,\ h,\ l)$

$$\left.M_t\right)_{key\ in\ shear} = bl\tau_k \frac{d}{2}$$

i.e., $\quad 2426512.17 = b \times 75 \times 60 \times \frac{65}{2}$

Therefore, $\quad b = 16.59mm = 16.6mm = h$

Therefore, the key dimensions are,

Width $\quad b = 16.6mm.$

Length of key $\quad l = 75mm(given)$

Example 6.19: A square key of side 22mm coupling two commercial shafts of diameter 70mm in a coupling is to be replaced by a tapered key

i) Determine the dimensions of tapered key.

ii) Compare the torque transmitting capacity of square key with tapered key.

Solution:

i) To determine the dimensions of tapered key $(b,\ h\ and\ \ l)$

Corresponding to shaft diameter $d = 70mm,$

Width of key $b = 22mm$

Depth of key $h = 12mm$ ... T(17.4)

a) To calculate the length of key (l)

Note: Width of key $b = 22mm > h = 12mm$

i.e., the key fails by crushing before it fails by shear.

Therefore, $M_t)_{key\ in\ crushing} = \frac{h}{2} l \sigma_c \frac{d}{2}$ ...(i)

$M_t)_{shaft} = \frac{\pi d^3}{16} \eta \tau_s$ ... E(19.2)

Where, $d = 70mm.$

$h = Keyway\ \ factor = 0.75$

$\tau_s = 40MPa,\ for\ commercial\ steel\ as\ per\ ASME\ code$

Therefore, $M_t)_{shaft} = \frac{\pi}{16} \times 70^3 \times 0.75 \times 40$

$= 2020436.8$ N-mm

Equating $M_t)_{key}\ and\ M_t)_{shaft}$ and assuming $\sigma_c = 2\tau,$

$$2020436.8 = \frac{12}{2} \times l \left(2 \times 40\right) \frac{70}{2}$$

Therefore, *Length of key* $l = 120.26mm$

Therefore, the dimensions of tapered key are

Width $b = 22mm$

Depth $h = 12mm$

and Length $l = 120.26mm$

ii) To compare the torque capacities of tapered key and square key

a) $M_t)_{square\ key} = bl\tau_k \frac{d}{2}$

Where, $l = 1.178d$

$= 1.178 \times 70 = 82.46mm$

Therefore, $M_t)_{square\ key} = 22\times 82.46\times 40\times \frac{70}{2}$

$= 2539768$ N-mm

b) $M_t)_{tapered\ key} = 2020436.8$ N-mm

Therefore, $\frac{M_t)_{square\ key}}{M_t)_{tapered\ key}} = \frac{2539768}{2020436.8} = 1.257$

i.e., $M_t)_{square\ key} = 1.257\, M_t)_{tapered\ key}$

i.e., A square key having same width as that of tapered key will be able to transmit 25.7% more torque compared to tapered keys.

Example 6.20: A square key is used in a flange coupling to transmit 20kW at 1200rpm. By selecting same material for shaft, key and bolts, determine

i) The diameter of shaft required.
ii) The side of square key.
iii) The number and size of bolts required.

Solution:

i) To determine the diameter of shaft (d)

The torque transmitted by the coupling is

$$M_t = 9550\frac{N}{n}\times 10^3 \text{ N-mm}$$

$$= 9550\times \frac{20}{1200}\times 10^3 = 159166.67 \text{ N-mm}$$

If *d* is the shaft diameter, then Torque transmitted by shaft is

$$M_t)_{shaft} = \frac{\pi d^3}{16}\eta\tau_s \quad \text{... (i)}$$

Assuming 45C8 as the material of shaft, key and bolts, the ultimate stress σ_{ut} for 45C8 material is

$$\sigma_{ut} = 650MPa$$

Assuming, $\sigma_y = 0.6\sigma_{ut}$

$= 0.6\times 650 = 390MPa$

Yield shear stress,

$$\tau_y = \frac{\sigma_y}{2}$$

$$= \frac{390}{2} = 195\, N/mm^2$$

Design of shear stress τ_s

$$\tau_s = \frac{\tau_y}{factor\ of\ \ safety}$$

$$= \frac{195}{3} = 65\, N/mm^2$$

From equation (i)

$$159166.67 = \frac{\pi d^3}{16} \times 0.75 \times 65$$

i.e., $d = 25.52mm$

Adopt $d = 26mm\left(standard\ size\right)$

ii) To find the side of square key $\left(i.e.,\ b\ or\ h\right)$

Assuming $M_t)_{shaft} = M_t)_{key} = M_t)_{bolts}$, and equating $M_t)_{shaft} = M_t)_{key}$

$$159166.6 = bl\tau_k \frac{d}{2} \qquad \text{... (ii)}$$

Note: For a square key, the torque transmitted in shear = The torque transmitted in compression.

i.e., $M_t)_{key\ in\ shear} = M_t)_{key\ in\ compression}$

Where, the length of square key $l = 1.178d\left(\ proved\ already\right)$

Therefore, $l = 1.178 \times 26 = 30.63mm$

From equation (ii),

$$159166.67 = b\left(30.63\right) \times 65 \times \frac{26}{2}$$

i.e., $b = 6.15mm$

i.e. Use $b = h = 6.5mm$

iii) ***To find the number and size of bolts required*** (i, d_1)

a) ***The number of bolts,*** $i = 20d + 3$

Where, *'d'* is in meter

Therefore, $$i = 20\left(\frac{26}{1000}\right) + 3 = 3.52$$

Use, $i = 4$ bolts

b) ***To find size of bolts***

The torque transmitted through bolts is

$$\left.M_t\right)_{bolts} = i\left(\frac{\pi d_1^2}{4}\right)\tau_b \frac{D_1}{2} \qquad \text{... E(19-4)}$$

$$= i\left(A_c\right)\tau_b \frac{D_1}{2}$$

Where, $A_c = Core\ area\ of\ bolt = \dfrac{\pi d_1^2}{4}$

Here, $i = 4,\ \tau_b =_s = \tau_k = 63.33\, N/mm^2$

$D_1 = Bolt\ circle\ diameter = 2d + 0.05$ ... E(19-12a)

d is in meter

Therefore, $$D_1 = 2\left(\frac{26}{1000}\right) + 0.05$$

$$= 0.102m = 102mm$$

From equation (19-4)

$$159166.67 = 4\left(A_c\right) \times 63.33 \times \frac{102}{2}$$

i.e., $A_c = 12.32mm^2$

Referring Table (18-7), corresponding to $A_c = 14.2mm^2$ *(equal to or higher than* A_c *calculated). Select* $M\,5 \times 0.8$ *bolts.*

Example 6.21: A pulley is fastened to a 100mm diameter shaft running at 200rpm by means of a key 25mm wide and 150mm long. Taking the permissible values of shear and compressive stresses as 60MPa and 120MPa, determine the power that can be transmitted and also the thickness of key required.

Solution:

i) To determine the width of key required

$$M_t\Big)_{key\ in\ shear} = bl\tau_k \frac{d}{2}$$

$$= 25\times150\times60\times\frac{100}{2}$$

$$= 11250000 = 11.25\times10^6 \text{ N-mm}$$

$$M_t\Big)_{key\ in\ compression} = \frac{h}{2} l\sigma_c \frac{d}{2}$$

Assuming $M_t\Big)_{key\ in\ shear} = M_t\Big)_{key\ in\ compression}$

$$11.25\times10^6 = \frac{h}{2}\times150\times120\times\frac{100}{2}$$

i.e., $h = 31.25mm$

Note: $h = 31.25mm < b = 25mm$

ii) To calculate the power transmitted (N)

Torque transmitted

$$M_t = 9550\frac{N}{n}\times10^3 \text{ N-mm}$$

i.e., $$11.25\times10^6 = \frac{9550N\times10^3}{200}$$

Therefore, Power transmitted, $N = 235.6kW$

Example 6.22: It is required to design a square key for fixing a gear on a shaft of diameter 25mm. 15kW of power at 720rpm is transmitted from the shaft to the gear. The key is made of steel 50C4 $\sigma_{yt} = 460\,N/mm^2$ ***and the factor of safety is 3. The yield***

strength in compression can be assumed to be equal to the yield strength in tension, determine the dimension of key. ***(VTU-Jan/Feb 2005)***

Solution:

Torque transmitted by shaft

$$M_t = 9550 \frac{N}{n} \times 10^3 \text{ N-mm} \qquad \text{... E(19-3C)}$$

$$= \frac{9550 \times 15 \times 10^3}{720}$$

$$= 198958.3 \text{ N-mm}$$

Note: ***The length of a square key in terms of shaft diameter when both key and shaft are made of same material and of equal strength (proved in the earlier problem) is***

$$l = 1.178d$$

$$= 1.178 \times 25mm$$

Therefore, $\left.M_t\right)_{key\ in\ shear} = bl\tau_k \frac{d}{2}$

Where, $\sigma_{yt} = 460\, N/mm^2$

$$\tau_y = \frac{460}{2} = 230MPa$$

$$\tau_k = \frac{\tau_y}{factor\ of\ safety}$$

$$= \frac{230}{3} = 76.67MPa$$

$$198958.3 = b(1.178 \times 25)76.67 \times \frac{25}{2}$$

Therefore, Width of key $b = 7.05mm$

Use $b = 7.1mm = h = Thickness$

Example 6.23: A splined connection designated by $8 \times 52 \times 60$ ***is used for the gear and shaft assembly of a gear box and is to transmit 25kW at 400rpm. Assuming the normal pressure on spline as*** $5\, N/mm^2$***, coefficient of friction*** $\mu = 0.05$***, determine***

a) The length of hub of Gear and
b) The force required to shift the Gear.

Solution:

Note: A splined shaft designated by $8 \times 52 \times 60$ ***has*** $D = 60mm,\ d = 52mm$ ***and number of splines*** $i = 8$

Thickness of spline

$$h = \frac{D-d}{2}$$

$$= \frac{60-52}{2} = 4mm$$

a) To find length of hub (l)

$$The\ length\ of\ hub = Length\ of\ spline = l$$

The torque transmitted by splined shaft is given by

$$M_t = \frac{1}{2} phli(D-h) \qquad \text{... E(17.20)}$$

Where, $M_t = 9550 \frac{N}{n} \times 10^3$ N-mm

$$= \frac{9550 \times 25 \times 10^3}{400} = 596875 \text{ N-mm}$$

$$p = Pressure = 5\ N/mm^2$$

Substituting in E(17-20)

$$596875 = \frac{1}{2} \times 5 \times 4 \times l \times 8(60-4)$$

Therefore, $l = 133.23mm$

Use, $l = 134mm$

b) To find the force required to shift gear

Assuming the resultant force acting at the mean radius,

Torque $M_t = pr_m$

Where $r_m = \frac{60+52}{4} = 28mm$

Therefore, $p = \frac{M_t}{r_m}$

$$= \frac{596875}{28} = 21316.96\,N$$

Frictional force $= \mu p = 0.05 \times 21316.96 = 1065.85N$

Note: The force required to shift gear is equal an opposite to the frictional force $= 1065.85N$

Design of Couplings

Couplings are used to connect and to transmit power from the *driving shaft* to the *driven shaft. Couplings* are broadly classified into two types namely

a) *Rigid couplings.*
b) *Flexible couplings.*

Rigid couplings are used to transmit power between two accurately aligned shafts running at low speeds. Examples of this type of coupling are *Protected type Flange coupling* and *Marine coupling.*

Flexible couplings are used to take care of small un-intentional misalignment of shafts and the provide axial movement of shaft. It is to be noted here that these coupling also absorb vibration in the coupling to some extent.

Examples of flexible couplings are *Bushed Pin type coupling, Oldham coupling, etc.*

Design procedure for Flange Couplings

The components to be designed in a flange coupling are

i) *Shafts.*
ii) *Keys.*
iii) *Bolts and*
iv) *Flanges.*

Note: Refer figure (19.1 MDH)

i) The torque transmitted by the coupling is obtained from the equation

$$M_t = \frac{9550N}{n} \text{ N-m} \qquad \text{... E(19.3c)}$$

And is equated to, the torque transmitted by the shaft, given by

$$M_t)_{shaft} = \frac{\pi d^3}{16}\eta\tau_s \quad \text{... E(19.2)}$$

To obtain the shaft diameter *'d'*, the value of *'d'* is rounded off to the nearest *standard size. Refer chapter on "Design of Shafts" for standards shaft sizes.*

ii) ***Based on the shaft diameter 'd', width 'b' and thickness 'h' of key are selected from table (17-4)***

The length of the key is found by checking the key in *shear* and *compression.*

iii) ***Design of bolts used in the coupling includes calculation of the number of bolts*** $'i'$ ***and the size of bolts*** d_1.

The number of bolts $'i'$ is obtained from equation,

$$i = 20d + 3 \quad \text{... E(19.1a)}$$

Where, $d = Diameter\ of\ shaft\ in\ meter.$

The size of bolts is determined by calculating A_c, *the core area of bolts from equation (19.4) given by*

$$M_t)_{bolts} = i\left(\frac{\pi d_1^2}{4}\right)\tau_b \frac{D_1}{2} \quad \text{... E(19.4)}$$

Where, $\frac{\pi d_1^2}{4} = A_c = Core\ area\ of\ bolts; mm^2$

$\tau_b = Permissible\ shear\ stress\ in\ bolts; N/mm^2$

$D_i = Bolt\ circle\ diameter; mm$

Standard size of bolts is obtained from table (18.7) depending on *core area* A_c

Note: Unless otherwise specified, the shafts, keys and bots are made of same material and hence have the same value of stresses.

i.e. $\tau_{shaft} = \tau_{bolts} = \tau_{key}\ and$

$M_{shaft} = M_t)_{key} = M_t)_{bolts}$

iv) ***The hub diameter*** D_2, ***outside diameter of flange*** D ***and thickness of flange*** t ***are obtained from equation given by,***

Hub diameter $D_2 = 1.5d + 0.025$ ... E(19.13a)

Outside diameter of flange $D = 2.5d + 0.075$ ... E(19.14a)

Thickness of flange t

$$M_t)_{flange} = t(\pi D_2)\tau_f \frac{D_2}{2}$$ ... E(19.6)

Where, $\tau_f = Shear\ stress\ in\ Flange; N/mm^2$

Example 6.24: Design a protected type Cast Iron flange coupling to transmit 50kW at 200rpm. The maximum torque transmitted is to be 30% more than the mean torque. The permissible values of shear stress and compressive stress for shafts, bolts and key may be taken as 40MPa and 80MPa respectively. The permissible stress for Cast Iron may be taken as 10MPa. (VTU-Dec06/Jan07)

Solution:

a) ***Design of shaft***

The torque transmitted by shaft is given by

$$M_t = \frac{\pi d^3}{16}\eta\tau_s$$ ... E(19.2)

Where, Mean torque

$$M_t = \frac{9550N}{n} \text{ N-m}$$ ... E(19.3c)

$$= \frac{9550 \times 50 \times 10^3}{200} \text{ N-mm}$$

Therefore, $M_t = 2387500$ N-mm

The maximum torque transmitted $= 1.3M_t$

$$= 1.3 \times 2387500 = 3103750 \text{ N-mm}$$

Taking keyway factor $\eta = 0.75$, equation (19.2) becomes

$$3103750 = \frac{\pi d^3}{16} \times 0.75 \times 40; \text{ from which}$$

Shaft diameter, $d = 80.77mm$

Adopt standard size $d = 85mm$

b) ***Design of key (To find the dimension of key b, h and l)***

Referring to table (17.4) *(Tapered keys)*, corresponding to $d = 85mm,$ we get

Width of key $b = 22mm$

Depth of key $h = 14mm$

To find length of key

Note: ***Since the width of key*** $b >$ ***the depth h, the key fails by compression before it fails by shear.***

Therefore $$M_t)_{key\ in\ compression} = \frac{h}{2} l \sigma_c \frac{d}{2}$$

$$3103750 = \frac{14}{2} \times l \times 80 \times \frac{85}{2}$$

$$l = 130.41mm$$

Say $l = 131mm$

Therefore, Key dimensions are $b = 22mm,$ $h = 14mm$ and $l = 131mm$

c) ***Design of bolts (To find the number of bolts i and bolt diameter*** d_1***)***

i) ***To find the number of bolts i***

The number of bolts

$$i = 20d + 3$$

Where, d is the shaft diameter in meter.

$$= 20 \times 0.085 + 3$$

$$= 4.7$$

Adopt $i = 6$ (Preferable an even number)

ii) ***To find the bolt size*** d_1

The torque transmitted by bolts is given by

$$M_t)_{bolts} = i\left(\frac{\pi d_1^2}{4}\right)\tau_b \frac{D_1}{2} \qquad \text{... E(19.4)}$$

Where, $i = 6$

$$A_c = \frac{\pi d_1^2}{4} = Core\ are\ of\ bolts; mm^2$$

$$\tau_b = Shear\ stress\ in\ bolts = \tau_{shaft} = 40MPa$$

$$D_1 = Bolt\ circle\ diameter\ of\ pitch\ circle\ diameter\ of\ bolts$$

$$= 2d + 0.05 \quad \text{... E(19.12a)}$$

$$= 2 \times 0.085 + 0.05$$

$$= 0.17 + 0.05 = 0.22m$$

Where, $d\ is\ in\ meter$

Substituting the values in E(19.4), we get

$$3103750 = 6(A_c) \times 40 \times \frac{220}{2}$$

Therefore, $A_c = 117.57 mm^2$

Referring to Table (18.7), corresponding to $A_c = 125mm^2$, *(next higher value of* A_c*) Use,* $M14 \times 1.5$ *bolts*

Note: ***M stands for metric screw threads, 14 represents the*** **basic diameter** ***or*** **nominal diameter of bolts** ***14*****mm*****, 1.5 represents pitch of threads 1.5*****mm.**

d) The design the Flange (i.e., to find the flange thickness and flange diameter D)

$$M_t)_{flange} = t(\pi d_2)\tau_f \left(\frac{D_2}{2}\right) \quad \text{... E(19.6)}$$

Where, $t = Thickness\ of\ flange; mm$

$$\tau_f = Shear\ stress\ in\ flange = 10MPa$$

$$D_2 = Hub\ diameter = 15d + 0.025 \quad \text{... E(19.13a)}$$

$$d = Shaft\ diameter\ in\ meter$$

$$D_2 = 1.5(0.085) + 0.025$$

$$= 0.1525m = 152.5mm$$

Therefore, $3103750 = t(\pi \times 152.5) \times 10 \times \frac{152.5}{2}$

$$t = 8.5mm$$

Note: ***The value of thickness t obtained from the above formula is usually very small. Since the flanges are cast (made by casting process), for practical purposes,*** $t = \frac{d}{2}$ ***is adopted.***

Therefore, use $t = \frac{85}{2} = 42.5mm$

The outside diameter of flange D is given by

$$D = 2.5d + 0.075 \qquad \text{... E(19.14a)}$$

$$= 2.5 \times 0.085 + 0.075$$

$$= 0.2875m = 287.5mm$$

Example 6.25: A flange of coupling is used to connect two commercial shafts of diameter 50mm. four bolts of same material as that of shafts are used in the coupling on a bolt circle of 240mm diameter. The web thickness if 22mm. Determine

a) The size of bolts required and

b) The power transmitted at 200rpm.

Solution:

The torque transmitted by the shaft is given by

$$\left.M_t\right)_{shaft} = \frac{\pi d^3}{16}\eta\tau_s \qquad \text{... E(19.2)}$$

Where, $d = Shaft\ diameter = 50mm$

$\eta = Keyway\ \ factor = 0.75$

$\tau_s = 40MPa$ for commercial shafts with keyway according to ASME code.

Therefore, $\left.M_t\right)_{shaft} = \frac{\pi(50)^3}{16} \times 0.75 \times 40$

$$= 736310.78 \text{ N-mm}$$

a) To determine the size of bolts

$$\left.M_t\right)_{bolts} = i\left(\frac{\pi d_1^2}{4}\right)\tau_b\frac{D_1}{2} \qquad \text{... E(19.4)}$$

Here, $M_t)_{bolts} = M_t)_{shaft} = 736310.78$ N-mm

Number of bolts, $i = 4$

$$\frac{\pi d_1^2}{4} = A_c = Core\ area$$

$$\tau_b = \tau_s = 40MPa$$

Bolt circle diameter, $D_1 = 240mm$

Therefore, $736310.78 = 4\left(A_c\right)\times 40\times\frac{240}{2}$

i.e., $A_c = 38.35mm^2$

Corresponding to $A_c = 39.2mm^2$, use $M8\times 1.0\ bolts$

b) To determine the power transmitted

$$M_t = \frac{9550N\times 10^3}{n} \text{ N-mm}$$

$$736310.78 = \frac{9550N\times 10^3}{200}$$

$$N = 15.42kW$$

Therefore, Power transmitted $= 15.42kW$

Example 6.26: In a flange coupling used to connect two co-axial shafts of diameter 80mm to transmit 60kW at 200rpm. $6, M14\times 1.5$ ***bolts are used on bolt circle diameter of 240mm. The hub diameter is 150mm and flange thickness is 20mm. Determine***

a) The shear stress induced in the shaft.
b) The shear stress induced in the bolts.
c) The shear stress induced in the key taking compressive stress = 80MPa
d) The shear stress induced in the flanges.

Solution:

a) To find the shear stress induced in the shaft $\left(\tau_s\right)$

$$M_t)_{shaft} = \frac{\pi d^3}{16}\eta\tau_s \qquad \text{... E(19.2)}$$

Therefore, $2865000 = \frac{\pi (80)^3}{196} \times 0.75 \times \tau_s$

τ_s = *Shear stress induced in the shaft*

Therefore, $\tau_s = 37.99\, N/mm^2$

b) ***To find the shear stress induced in bolts*** (τ_b)

$$M_t)_{bolts} = i\left(\frac{\pi d_1^2}{4}\right)\tau_b \frac{D_1}{2} \qquad \text{... E(19.4)}$$

Where, $i = 6$

$A_c = \frac{\pi d_1^2}{4}$, For $M14 \times 1.5$ *bolts*, $A_c = 125mm^2$ ... T(18.7)

$D_1 = 240mm$

Therefore, $286500 = 6 \times 125 \tau_b \times \frac{240}{2}$

i.e., $\tau_b = 31.83\, N/mm^2$

c) ***To find the shear stress induced in key*** (τ_k)

Note: ***For shaft diameter*** $d = 80mm.$

Width of key $b = 22mm$

Depth of key $h = 14mm$ ... T(17.4)

To find the length of key l

Note: ***Since the width of key*** $b > h$, ***the key fails by compression before it fails by shear***

Therefore, $M_t)_{key\ in\ compression} = \frac{h}{2} l \sigma_c \frac{d}{2}$

$2865000 = \frac{14}{2} \times l \times 80 \times \frac{80}{2}$

$l = 127.9mm$

Now, $M_t)_{key\ in\ shear} = bl\tau_k \frac{d}{2}$

$$2865000 = 22 \times 127.9 \times \tau_k \frac{80}{2}$$

Where, $\tau_k = Shear\ stress\ induced\ in\ key$

$$\tau_k = 25.45\, N/mm^2$$

d) To find the shear stress induced in flanges (τ_f)

$$M_t)_{flange} = t(\pi d_2)\tau_f \left(\frac{D_2}{2}\right)$$

$$2865000 = 20(\pi \times 150)\tau_f \times \frac{150}{2}$$

$$\tau_f = 4.053\, N/mm^2$$

Therefore, Shear stress induced in the flange is

$$\tau_f = 4.053\, N/mm^2$$

Example 6.27: In a rigid flange coupling designed to transmit 50kW at 200rpm, a tapered key designated by $18 \times 11 \times 100$ ***is used to key the shafts of 60mm diameter to flanges. Five bolts are used in the coupling on a bolt circle of 170mm diameter. Taking the material of bolts same as that of shaft, Determine***

a) The shear stress induced in the shafts and key.

b) The size of the bolts required.

Solution:

Torque transmitted

$$M_t = \frac{9550N}{n} \text{ N-m}$$

$$= \frac{9550 \times 50 \times 10^3}{200} = 2387500 \text{ N-mm}$$

a) To determine the shear stress induced in shafts and key

$$M_t)_{shaft} = \frac{\pi d^3}{16}\eta\tau_s \quad \text{... E(19.2)}$$

Where, $d = Shaft\ diameter = 60mm$

$\eta = Keyway\ \ factor = 0.75(assumed)$

$\tau_s = Shear\ stress\ in\ shafts$

Therefore, $2387500 = \frac{\pi(60)^3}{16} \times 0.75\tau_s$

$\tau_s = 75.05\, N/mm^2$

Also, Torque transmitted by key in shear

$$M_t)_{key\ in\ shear} = bl\tau_k \frac{d}{2}$$

Where, $M_t)_{key} = M_t)_{shaft} = 2384500$ N-mm

Note: ***A key designated by*** $18 \times 11 \times 100$ ***has dimensions***

Width $b = 18mm$

Thickness $h = 11mm$

And Length $l = 100mm$

(The maximum dimension is l, and the minimum dimension is thickhess h)

Therefore, $2387500 = 18 \times 100 \times \tau_k \frac{60}{2}$

i.e., $\tau_k = Shear\ stress\ induced\ in\ key = 44.21\, N/mm^2$

b) To find the size of the bolts required

$$M_t)_{bolts} = i\left(\frac{\pi d_1^2}{4}\right)\tau_b \frac{D_1}{2} \quad \text{... E(19.4)}$$

$$= i(A_c)\tau_b \frac{D_1}{2}$$

Where, $i = Number\ of\ bolts = 5$

$A_c = Core\ area\ of\ each\, bolt\ in\ mm^2$

$\tau_b = Shear\ stress\ in\, bolts = Shear\ stress\ in\ shaft$

$$= \tau_s = 75.05\, N/mm^2$$

$$D_1 = Bolt\ circle\ diameter = 170mm$$

Therefore, $$2387500 = 5 \times A_c \times 75.05 \times \frac{170}{2}$$

$$A_c = 74.85mm^2$$

Referring table (18.7) corresponding to $A_c = 84.3mm^2$, select $M12 \times 1.75\,bolts$

Example 6.28: In a flange coupling having shaft diameter of 40mm, it is desired to limit the safe shear stress to $40\, N/mm^2$ ***. The outside diameter of the coupling is 200mm. There are 3 bolts situated on a bolt circle of 75mm radius. The diameter of bolts is 15mm. The thickness of flange is 20mm. Determine the power that can be transmitted by this coupling at 500rpm. Also determine the shear stress induced in the bolts.***

Solution:

$$M_t)_{shaft} = \frac{\pi d^3}{16} \eta \tau_s \qquad \text{... E(19.2)}$$

$$= \frac{\pi}{16} \times (40)^3 \times 0.75 \times 40$$

$$= 376991.12 \text{ N-mm}$$

i) To determine the shear stress induced in bolts i.e., τ_b

$$M_t)_{bolts} = i\left(\frac{\pi d_1^2}{4}\right)\tau_b \frac{D_1}{2} \qquad \text{... E(19.4)}$$

Where, $$M_t)_{shaft} = M_t\Big)_{bolts} = 376991.12 \text{ N-mm}$$

$$i = Number\ of\ bolts = 3$$

$$d_1 = Bolt\ diamter = 15mm$$

$$\tau_b = Shear\ stress\ induced\ in\ bolts; mm^2$$

$$\frac{D_1}{2} = Bolt\ circle\ radius = 75mm$$

Therefore, $376991.12 = 3\left(\frac{\pi}{4} \times 15^2\right)\tau_b \times 75$

i.e., $\tau_b = \textit{Shear stress induced in bolts} = 9.48\, N/mm^2$

ii) To determine the power transmitted (N)

Torque transmitted

$$M_t = \frac{9550N \times 10^3}{n} \text{ N-mm}$$

$$376991.12 = \frac{9550N \times 10^3}{500}$$

Therefore, $N = 19.74kW$

Therefore power transmitted by the coupling $= 19.74kW$

Design of Marine Flange Coupling (Solid Rigid Type)

Figure-6.4: Rigid Marine Coupling

i) The torque transmitted by the coupling is found from equation (19.3c) given by

$$M_t = \frac{9550N}{n}$$

Where, $N = \textit{Power transmitted in kW}$

$n = \textit{Speed in rpm}$

And is equated to the torque transmitted by the shaft given by

$$M_t = \frac{\pi d^3}{16}\eta\tau_s \quad \text{... E(19.2)}$$

***To obtain the shaft diameter* 'd'.**

Note: Since the flanges of the couplings are integral with the shaft, no keys are used and hence keyway factor η is taken as one.

ii) In the design of bolts, the number of bolts 'i' is obtained from equation

Number of bolts, $i = 33d + 5$

Where, $d = \textit{Shaft diameter in Meter}$

The diameter d_1 bolts is obtained from

$$d_1 = \sqrt{\frac{\eta d^2 \tau_s}{2iD_i \tau_b}} \qquad \text{... E(19.6a)}$$

Based on torque capacity of shaft,

$$d_1 = \sqrt{\frac{tD_2^2 \tau_f}{4iD_i \tau_b}} \qquad \text{... E(19.16b)}$$

Base on torque capacity of flange.

Note: ***The higher of the two values of d_1 is taken as required bolt diameter.***

iii) The outside diameter of the flange, bolt circle diameter and thickness of flanges are obtained from equations

Outside diameter of flange $D =_1 +2d \ to \ 3d$ ... E(19.19)

Bolt circle diameter $D_1 = 1.4d \ to \ 1.6d$ ... E(19.18)

Thickness of flange $t = 0.25d \ to \ 02.8d$ ... E(19.17)

Example 6.29: In a flange coupling shafts of diameter D, a square key of length 1.25D is used. Five bolts are used on a bolt circle of 5D. Assuming the shaft, key and bolts made of same material. Determine

a) The diameter of bolts d_1 in terms of D so that the torque transmitted by shaft, bolts and coupling are the same.

b) The key dimensions in terms of D.

Solution:

i) The torque transmitted by shaft is

$$M_t)_{shaft} = \frac{\pi D^3}{16} \eta \tau_s \qquad \text{... E(19-2)}$$

Assuming keyway factor $\eta = 0.75$

$$M_t)_{shaft} = \frac{\pi D^3}{16} \times 0.75 \times \tau_s$$

$$= 0.1473 D^3 \tau_s \qquad \text{... (i)}$$

ii) The torque transmitted through bolts is

$$M_t)_{bolts} = i\left(\frac{\pi d_1^2}{4}\right)\tau_b \frac{D_1}{2} \quad \text{... E(19-4)}$$

Where, $i = Number\ of\ bolts\ used\ in\ coupling = 5$

$d_1 = bolt\ \ diameter, mm$

$D_1 = Bolt\ circle\ diameter = 5D(given)$

Therefore, $M_t)_{bolts} = 5\left(\frac{\pi d_1^2}{4}\right)\tau_b\left(\frac{5D}{2}\right)$

$$= 9.8175\tau_b D d_1^2 \quad \text{.. (ii)}$$

Equating (i) and (ii)

$$0.1473D^2\tau_s = 9.8175\tau D d_1^2$$

Since shafts and bolts are of same material

$$\tau_s = \tau_b$$

Therefore, $d_1 = 0.1225D$

ii) To determine the square key dimensions (b, h and l)

Note: For a square key

$$M_t)_{key\ in\ shear} = M_t)_{key\ in\ compression}$$

Now, $M_t)_{key\ in\ shear} = bl\tau_k \frac{d}{2}$

Where, $l = Length\ of\ \ key = 1.25D(given)$

$d = D = Shaft\ diameter$

$\tau_s = \tau_k$

Therefore, $0.1473D^3\tau_s = b(1.25D)\tau_k \times \frac{D}{2}$

i.e., $b = Width\ of\ \ key = 0.2357D = Thickness\ of\ \ key\ h$

Therefore the key dimensions are

$Width \quad b = 0.2357D$

$Depth \quad h = 0.2357D$

$$Length\ l = 1.25D$$

Example 6.30: Two halves of flange coupling using shafts of diameter 100mm are fastened using six bolts of 20mm diameter on a bolt circle 250mm diameter. Determine the safe power that can be transmitted by this coupling at 200rpm assuming permissible shear stress = 50MPa.

Solution:

a) The torque transmitted by shaft is

$$M_t)_{shaft} = \frac{\pi d^3}{16}\eta\tau_s \qquad \text{... E(19.2)}$$

$$= \frac{\pi}{16}\times 100^3 \times 0.75 \times 50$$

$$= 7363107.8 \text{ N-mm}$$

b) The torque transmitted through bolts is

$$M_t)_{bolts} = i\left(\frac{\pi d_1^2}{4}\right)\tau_b \frac{D_1}{2} \qquad \text{... E(19-4)}$$

$$= 6\left(\frac{\pi}{4}\times 20^2\right)50\times\frac{250}{2}$$

$$= 11780972.45 \text{ N-mm}$$

Note: The safe power transmitted by coupling is based on least torque.

Therefore, $M_t = M_t)_{shaft} = 7363107.8$ N-mm

From $M_t = 9550\frac{N}{n}\times 10^3$

$$7363107.8 = 9550\times\frac{N}{200}\times 10^3$$

Therefore, the sage power *N* that can be transmitted is $N = 154.2kW$

Example 6.31: A flange coupling is used to connect two shafts of 100mm diameter. Bolts of 20mm diameter are equally spaced on a bolt circle of 300mm diameter. If the maximum shear stress in shafts and bolts is not to exceed $75\,N/mm^2$ ***and*** $60\,N/mm^2$

respectively, determine the number of bolts required.

Solution:

a) The torque transmitted by shafts in coupling is

$$M_t)_{shaft} = \frac{\pi d^3}{16}\eta\tau_s \qquad \text{... E(19-2)}$$

$$= \frac{\pi}{16}\times(100)^3 \times 0.75\times 75$$

$$= 11044661.67 \text{ N-mm}$$

b) The torque transmitted by bolts is

$$M_t)_{bolts} = i\left(\frac{\pi d_1^2}{4}\right)\tau_b \frac{D_1}{2} \qquad \text{... E(19-4)}$$

$$= i\left(\frac{\pi\times 20^2}{4}\right)60\times\frac{300}{2}$$

Equating $M_t)_{shaft} = M_t)_{bolts}$,

$$11044661.67 == i\left(\frac{\pi\times 20^2}{4}\right)60\times\frac{300}{2}$$

i.e., $i = 14\, bolts\left[A\ preferred\ even\ number\right]$

Example 6.32: In a flange coupling, 2 commercial steel shafts of 50mm diameter are used. The flanges are bolted together using four bolts of same material as that of shaft. The diameter of bolt circle is 240mm and the web thickness is 22mm. Determine

i) The minimum diameter of bolts required to transmit the same torque as the shaft.

ii) Calculate the power transmitted at 200rpm.

Solution:

i) The torque transmitted by shafts is

$$M_t)_{shaft} = \frac{\pi d^3}{16}\eta\tau_s \qquad \text{... E(19-2)}$$

Where, $d = 50mm$

$\eta = Keyway\ factor = 0.75(assumed)$

$\tau_s = 40MPa(For\ commercial\ steel\ as\ per\ ASME\ code)$

Therefore, $M_t)_{shaft} = \frac{\pi(50)^3}{16} \times 0.75 \times 40$

$= 736310.78$ N-mm

The torque transmitted through bolts is

$$M_t)_{bolts} = i\left(\frac{\pi d_1^2}{4}\right)\tau_b \frac{D_1}{2} \qquad \text{... E(19-4)}$$

$$= i(A_c)\tau_b \frac{D_1}{2}$$

Here, $i = Number\ of\ bolts = 4$

$A_c = Core\ are\ of\ bolts\ in\ mm^2 = \frac{\pi d_1^2}{4}$

$\tau_b = Shear\ stress\ in\ bolts = 40MPa(\ from\ data)$

$D_1 = Bolt\ circle\ diameter = 240mm$

Therefore, $M_t)_{bolts} = M_t)_{shaft} = 736310.78 = 4(A_c)40 \times \frac{240}{2}$

i.e., $A_c = 39.2mm^2$

Corresponding to $A_c = 39.2mm^2$, select $M8 \times 1.0\ bolts$

ii) To calculate the power transmitted $(i.e.,\ N)$

Torque $M_t = 9550\frac{N}{n} \times 10^3$

i.e., $736310.78 = \frac{9550N \times 10^3}{200}$

Therefore, Power transmitted $N = 15.42kW$

Example 6.33: A rigid flange coupling is assembled with five bolts on a 150mm bolt circle diameter. The shafts used in the coupling are of commercial steel and are of 50mm diameter. Determine the standard size of bolts required assuming permissible shear stress in bolts equal to 50MPa.

Solution:

The torque transmitted by shaft is

$$M_t)_{shaft} = \frac{\pi d^3}{16} \eta \tau_s \qquad \text{... E(19.2)}$$

Where, $d = Shaft\ diameter = 50mm(given)$

$\eta = Keyway\ \ factor = 0.75(assumed)$

$\tau_s = Shear\ stress\ \ for\ shaft\ material$

$= 40\, N/mm^2\ \ for\ commercial\ shafts\ as\ perASME\ code$

Therefore, $$M_t)_{shaft} = \frac{\pi(50)^3}{16} \times 0.75 \times 40$$

$$= 736310.78\ \text{N-mm}$$

The torque transmitted through bolts is

$$M_t)_{bolts} = i\left(\frac{\pi d_1^2}{4}\right)\tau_b \frac{D_1}{2} \qquad \text{... E(19-4)}$$

$$= i(A_c)\tau_b \frac{D_1}{2}$$

Where, $i = Number\ of\ \ bolts\ used\ in\ coupling = 5$

$A_c = \frac{\pi d_1^2}{4} = Core\ are\ of\ \ bolts\ in\ mm^2$

$\tau_b = Shear\ stress\ in\ bolts = 50MPa$

$D_1 = Bolt\ circle\ diameter = 150mm$

Equating the torque transmitted by shafts to torque transmitted through bolts

i.e., $$M_t)_{bolts} = M_t)_{shaft}$$

$$736310.78 = 5(A_c)50 \times \frac{150}{2}$$

i.e., $A_c = 39.27mm$

Corresponding to $A_c = 58mm^2$ Referring T(18-7)

$$\left[i.e.,\ A_c \geq 39.27mm^2\right]\ select\ M10\times1.5\ bolts$$

Example 6.34: In a flange coupling made using hollow shafts of outside diameter 50mm and inside diameter 30mm, the shear stress is to be limited to 60MPa. 6 bolts of size $M10\times1.5$ ***are used on a bolt circle of 150mm diameter. The hub diameter is 100mm and the thickness of flanges is 25mm. Determine***

- ***i) The power transmitted by the coupling at 1400rpm.***
- ***ii) The power transmitted at the same speed when the hollow shaft is replaced by a solid shaft of same material and of diameter equal to 50mm.***
- ***iii) The dimension of key required by limiting the crushing stress to 120MPa.***
- ***iv) The shear stress induced in flange.***
- ***v) The shear stress induced in bolts.***

Solution:

i) To determine the power transmitted by coupling using hollow shafts

$$M_t\Big)_{hollow\ shaft} = \frac{\pi}{16}\left(\frac{d_0^4 - d_i^4}{d_0}\right)\eta\tau_s$$

$$= \frac{\pi}{16}\left(\frac{50^4 - 30^4}{50}\right)0.75\times60$$

$$= 961327.325 \text{ N-mm}$$

But, $M_t = 9550\frac{N}{n}\times10^3$ N-mm

i.e., $961327.352 = \frac{9550N\times10^3}{1400}$

Therefore, Power $N = 140.93kW$

ii) To determine the power transmitted by solid shaft

$$M_t)_{solid\ \ shaft} = \frac{\pi d^3}{16}\eta\tau_s \qquad \text{... E(19-2)}$$

$$= \frac{\pi}{16}\times 50^3\times 0.75\times 60$$

$$= 1104466.17 \text{ N-mm}$$

$$1104466.17 = \frac{9550N\times 10^3}{1400}$$

i.e., Power transmitted by solid shaft is $N = 161.91kW$

iii) To determine the dimensions of key

For shaft diameter $d = 50mm,$

Width of key $b = 14mm$

Depth of key $h = 9mm$

To determine length of key (l)

Note: Width of key $b = 14mm > h = 9mm$

i.e., the key fails by compression or crushing before it fails by shear.

$$M_t)_{key\ in\ crushing} = \frac{h}{2}l\sigma_c\frac{d}{2}$$

$$961327.352 = \frac{9}{2}\times l\times 120\times\frac{d}{2}$$

Therefore, length of key $l = 71.21\ say\ l = 72mm$

The dimensions of key are

$$b = 14mm,\ h = 9mm\ and\ l = 72mm$$

iv) To determine the shear stress induced in flanges $(i.e., \tau_f)$

The torque transmitted by flanges is,

$$M_t)_{flanges} = t(\pi D_2)\tau_f\frac{d}{2} \qquad \text{... E(19-6)}$$

Where, $t = Thickness\ of\ \ flanges = 25mm$

$D_2 = Hub\ diameter = 100mm$

Therefore, $961327.352 = 25(\pi \times 100)\tau_f \times \frac{100}{2}$

Therefore, $\tau_f = 2.448\, N/mm^2$

***v) To determine the shear stress induced in bolts** $(i.e.\ \tau_b)$*

The torque transmitted through bolts is

$$M_t)_{bolts} = i\left(\frac{\pi d_1^2}{4}\right)\tau_b \frac{D_1}{2}$$

$$= i(Alc)\tau_b \frac{D_1}{2}$$

Where, $A_c = Core\ area\ of\ bolts = \frac{\pi d_1^2}{4};\ D_1 = 150mm$

***Note: For** $M10 \times 1.5$ bolts, $A_c = 58mm^2$* ... T(18-7)

$$961327.352 = 6(58)\tau_b \times \frac{150}{2}$$

Therefore, $\tau_b = 36.83\, N/mm^2$

***Example 6.35: A flange coupling made of shafts of diameter 50mm, 6 bolts of size** $M10 \times 1.5$ **are used on a bolt circle of 150mm diameter. A tapered key designated by** $20 \times 12 \times 100$ **connects the shafts to the flanges having 25mm thickness. Assuming the shear stress in flanges as** $10\, N/mm^2$**, determine the safe power that can be transmitted at 1400rpm based on the individual torque capacities of shaft, key, bolts and flanges. Assume** $\tau_s = \tau_b = \tau_k = 50MPa$ **and** $\sigma_c = 100MPa$.*

Solution:

Note: The safe power that is transmitted is based on the least torque capacity of the components in the coupling.

i) Torque capacity of shaft,

$$M_t)_{shaft} = \frac{\pi d^3}{16}\eta\tau_s$$ ... E(19-2)

$$= \frac{\pi (50)^3}{16} \times 0.75 \times 50$$

$$= 920388.47 \text{ N-mm}$$

ii) ***The torque transmitted by key in shear is***

a) $$M_t)_{key\ in\ shear} = bl\tau_k \frac{d}{2} \qquad \text{... (i)}$$

Note: ***A key designated by*** $20 \times 12 \times 100$ ***has***

Width $b = 20mm$

Thickness $h = 12mm$

Length $l = 100mm$

Therefore, $$M_t)_{key\ in\ shear} = 20 \times 100 \times 50 \times \frac{50}{2}$$

$$= 2500000 \text{ N-mm}$$

b) $$M_t)_{key\ in\ crushing} = \frac{h}{2} l\sigma_c \frac{d}{2}$$

$$= \frac{12}{2} \times 100 \times 100 \times \frac{50}{2}$$

$$= 1500000 \text{ N-mm}$$

iii) ***The torque transmitted through bolts is***

$$M_t)_{bolts} = i\left(\frac{\pi d_1^2}{4}\right)\tau_b \frac{D_1}{2} \qquad \text{... E(19-4)}$$

$$= i(A_c)\tau_b \frac{D_1}{2}$$

Note: ***Core area*** (A_c) ***of*** $M10 \times 1.5$ ***bolts is*** $A_c = 58mm^2$ ***... T(18-7)***

Therefore, $$M_t)_{bolts} = 6(58)50 \times \frac{150}{2}$$

$$= 1305000 \text{ N-mm}$$

Therefore, the least torque transmitted by any member in coupling is

$$M_t)_{\min} = M_t)_{shaft} = 920388.47 \text{ N-mm}$$

Safe power (N) transmitted by coupling based on $M_t = 920388.47$ N-mm is obtained from equation

$$M_t = 9550 \frac{N}{n} \times 10^3$$

i.e., $$920388.47 = 9550 \frac{N}{1400} \times 10^3$$

i.e., $$N = 134.93 kW$$

Example 6.36: A rigid coupling is used to transmit 20kW power at 720rpm. There are four bolts on a pitch circle diameter of 125mm. The bolts are made of steel 45C8 $\left(\sigma_{yt} = 380\, N/mm^2\right)$ and the factor of safety is 3. Determine the diameter of the bolts.

(VTU-Jan/Feb 2005 and Dec08/Jan 2009)

Solution:

The torque transmitted by coupling is

$$M_t = 9550 \frac{N}{n} \times 10^3 \text{ N-mm}$$

$$= \frac{9550 \times 20 \times 10^3}{720} = 265277.8 \text{ N-mm}$$

Torque transmitted through bolts is

$$M_t)_{bolts} = i \left(\frac{\pi d_1^2}{4} \right) \tau_b \frac{D_1}{2} \qquad \text{... E(19-4)}$$

Where, $i = Number\ of\ bolts = 4$

$\tau_b = Shear\ stress\ in\ bolts$

Now, $\sigma_{yt} = 380\, N/mm^2$ (Obtained from T(1-25)

$$\tau_y = \frac{\tau_{yt}}{2} = 190\, N/mm^2$$

$$\tau_{bolts} = \frac{\tau_y}{factor\ of\ safety}$$

$$= \frac{190}{3} = 63.33\ N/mm^2$$

$$D_1 = Pitch\ circle\ diameter\ of\ bolts = 125mm$$

Substituting in equation (19-4) we get

$$265277.8 = 4\left(\frac{\pi d_1^2}{4}\right)63.33 \times \frac{125}{2}$$

i.e., $d_1 = Core\ diameter\ of\ bolts = 4.62mm$

Standard size of bolts:

Corresponding to $A_c = \frac{\pi d_1^2}{4}$

$$= \frac{\pi}{4} \times (5)^2 = 19.63mm^2$$

Select $M6 \times 1\ bolts\ \left(correspondig\ to\ A_c = 20.1mm^2\right)$

Example 6.37: In a flange coupling used to couple two shafts of diameter d, 6 number of $M14 \times 2$ ***bolts are used on a pitch circle diameter 3d, the hub diameter = 2d. The shafts, key and bolts are made of same material with an allowable shear stress*** $= 50\ N/mm^2$. ***Determine***

- ***i)*** ***The diameter 'd' assuming torque transmitted by shaft and bolts are equal.***
- ***ii)*** ***The power transmitted by the coupling at 1200rpm.***
- ***iii)*** ***The shear stress induced in flanges taking flange thickness*** $= \frac{d}{2}$.
- ***iv)*** ***Dimensions of key by limiting crushing stress to*** $100\ N/mm^2$.
- ***v)*** ***The dimensions of hollow shaft, transmitting the same torque as the solid shaft required with outside diameter equal to diameter solid shaft and of same material.***

Solution:

i) ***To find d, the diameter of solid shaft***

If *d* is the diameter of solid shaft then,

$$M_t)_s = \frac{\pi d^3}{16}\eta\tau_s \qquad \text{... E(19-2)}$$

The torque transmitted by bolts

$$M_t)_b = i\left(\frac{\pi d_1^2}{4}\right)\tau_b \frac{D_1}{2} \qquad \text{... E(19-4)}$$

$$\tau_s = \tau_b \ \left(\textit{since same material}\right)$$

Keyway factor $\eta = 0.75,\ D_1 = 3d,\ A_c = \frac{\pi d_1^2}{4}$

For $M14\times 2$ bolts, from table (18-7)

$$A_c = 115mm^2$$

Therefore, $\frac{\pi d^3}{16}\times 0.75 = 6\left(1150\right)\times\frac{3d}{2}$

Therefore, $d = 83.834$

Adopt $d = 85mm$

ii) To determine power transmitted by coupling $\left(N\right)$

Torque, $M_t = 9550\frac{N}{n}\times 10^3$ N-mm

Where, $M_t = \frac{\pi d^3}{16}\eta\tau_s$

$$= \frac{\pi}{16}\times\left(85\right)^3\times 0.75\times 50$$

$$= 6029158.1 \text{ N-mm}$$

i.e., $6029158.1 = \frac{9550N}{1200}\times 10^3$

Therefore, Power transmitted $N = 757.6kW$

iii) To determine the shear stress induced in flanges i.e., τ_f

$$M_t)_{flange} = t(\pi d_2)\tau_f\left(\frac{D_2}{2}\right) \quad \text{... E(19-6)}$$

$$6029158.1 = \frac{85}{2}(\pi \times 2 \times 85)\tau_f\left(\frac{2\times 85}{2}\right)$$

Therefore, $D_2 = 2d\,(given) = 2 \times 85$

Therefore, $\tau_f = 3.125\, N/mm^2$

iv) To determine the dimensions of key (b, h and l)

For $d = 85mm$ from, T(17-4)

$$b = Width\ of\ key = 22mm$$

$$h = Thickness\ of\ key = 14mm$$

To determine length of key l

Note: Since $b > h$, the key fails by crushing before it fails by shear.

Therefore, $M_t)_{key\ in\ crushing} = \frac{h}{2} l \sigma_c \frac{d}{2}$

$$6029158.1 = \frac{14}{2} \times 6 \times 100 \times \frac{85}{2}$$

Therefore, $l = 202.66mm$

Therefore the key dimension are

Width	$b = 22mm$
Thickness	$h = 14mm$
Length	$l = 202.66mm$

v) To determine dimensions of hollow shaft $(d_0\ and\ d_i)$

If d_0 and d_i are the outside and inside diameters of hollow shaft, then

$$M_t)_{hollow\ shaft} = \frac{\pi}{16}\left(\frac{d_0^4 - d_i^4}{d_0}\right)\eta\tau_s$$

Here, $d_0 = d = 85mm$

$$6029158.1 = \frac{\pi}{16}\left(\frac{85^4 - d_i^4}{85}\right)0.75 \times 50$$

$$d_i = 64.586mm$$

Therefore, $d_0 = 85mm \; and \; d_i = 64.59mm$

Design of Marine Flange Coupling

Example 6.38: Design a Marine flange coupling of solid type to transmit 200kW at 100rpm. Assume the shafts and bolts made of same material with an allowable shear stress of 50MPa.

Solution:

a) To find the shaft diameter, d

The torque transmitted by coupling is given by

$$M_t = 9550\frac{N}{n} \times 10^3$$

$$= \frac{9550 \times 200 \times 10^3}{100}$$

$$= 191 \times 10^5 \text{ N-mm}$$

If '*d*' is the shaft diameter then, from equation

$$M_t = \frac{\pi d^3}{16}\eta\tau_s$$

Note: ***Here no keys are used since flanges are integral with shafts and hence key way factor*** $\eta = 1$

Therefore, $191 \times 10^5 = \frac{\pi d^3}{16} \times 1 \times 50$

Therefore, shaft diameter $d = 125mm$

b) To design the bolts (To find the number of bolts i and the bolt diameter d_1***)***

i) Number of bolts $i = 33d + 5$

$$= 33 \times 0.125 + 5$$

Where, *d* is in metre.

$i = 9.125$

Use $i = 10\ bolts$

To find size of bolts (d_1)

i) Bolt diameter based on torque capacity of shaft is

$$d_1 = \sqrt{\frac{\eta d^3 \tau_s}{2iD_1\tau_b}} \qquad \text{... E(19.16)}$$

Where, $\eta = 1$, $d = 125mm$, $\tau_s = \tau_b$ $i = 10$

$D_1 = Bolt\ circle\ diameter = 1.5d$

$= 1.5 \times 125 = 187.5mm$

$$d_1 = \sqrt{\frac{1 \times 125^3}{2 \times 10 \times 187.5}} = 22.82mm$$

ii) Based on torque capacity of flange

$$d_1 = \sqrt{\frac{tD^2\tau_f}{3iD_1\tau_b}} \qquad \text{... E(19.16b)}$$

Where, $t = Thickness\ of\ \ flange = 0.26d$ ... E(19.17)

$= 0.26 \times 125 = 32.5mm$

$\tau_f = \tau_b$, $i = 10$, $D_1 = 187.5mm$

$D_2 = Outside\ diameter\ of\ \ flange = D_1 + 2.5d$ ... E(19.19)

$= 187.5 + 2.5 \times 125$

$= 500mm$

$$d_1 = \sqrt{\frac{32.5 \times (500)^2}{4 \times 10 \times 187.5}}$$

$= 32.91mm$

Adopt diameter of bolts $d_1 = 33mm$ *(Higher of the two values of* d_1)

Taper of bolts is 1 in 100.

Muff of Sleeve Coupling

A muff or sleeve coupling consists of a cast iron sleeve (hollow cylinder) fitted over the ends of the shafts to be connected. The shafts are keyed to the sleeve by a sunk key. In this coupling, the torque is transmitted from one shaft to the sleeve and from the sleeve to the other shaft. There are no projecting components and hence this coupling has a smooth exterior.

Figure-6.25: Muff of Sleeve Coupling

Example 6.39: Design a Muff coupling with key to transmit 50kW at 500rpm. The permissible values of shear stress and compressive stress for shaft and key material may be taken as 60 and 120MPa respectively and for the cast iron, shear stress may be taken as 15MPa.

Solution:

a) To design the shaft (i.e., to find shaft diameter d)

$$M_t = \frac{9550 \times 50}{500} \text{ N-m}$$

$$= \frac{9550 \times 50}{500} \times 10^3 \text{ N-mm}$$

$$= 955 \times 10^3 \text{ N-mm}$$

The shaft diameter

$$d = \sqrt[3]{\frac{16M_t}{\pi\tau_d}} \qquad \text{... E(19.49)}$$

Where, $\tau_d = 60MPa$

Therefore, $d = \sqrt[3]{\frac{16 \times 955 \times 10^3}{\pi \times 60}} = 43.28mm$

Use, $d = 45mm$

b) Design of Key

For $d = 45mm$, referring T(17.4),

Width of key $b = 12mm$

Depth of key $h = 8mm$

Length of key = *Length of hub* $= 3.5d$

$$= 3.5 \times 45 = 157.5mm \qquad \text{... E(19.48)}$$

Check for stresses in key

Shear stress induced in key

$$\tau_{key} = \frac{2M_t}{bld}$$

$$= \frac{2 \times 995 \times 10^3}{12 \times 157.5 \times 45}$$

$$= 22.46\, N/mm^2 < \tau_{allowable} = 60\, N/mm^2$$

Similarly, compressive stress induced in key

$$\sigma_c = \frac{4M_t}{hld}$$

$$= \frac{4 \times 955 \times 10^3}{8 \times 157.5 \times 45}$$

$$= 67.32\, N/mm^2 < \sigma_{allowable} = 120\, N/mm^2$$

Hence, we see that the values of shear and compressive stresses induced in the key are much lesser than the permissible values and hence the design is safe.

c) Design of sleeve

The outside diameter of sleeve

$$D = 2d + 0.013 \qquad \text{... E(19.45a)}$$

$$= 2 \times 0.045 + 0.013$$

$$= 0.103m = 103mm$$

The length of sleeve = length of key $= 3.5d$

$$= 3.5 \times 45mm = 157.5mm$$

Note: The outside diameter of the sleeve may also be obtained from equation

$$D = \sqrt[3]{\frac{16M_t}{\pi\tau_{sleeve}\left(1-k\right)^4}} \qquad \text{... E(19.46)}$$

Where, $k = \frac{d}{D}$

Treating it as hollow shaft with outside diameter D,

Inside diameter = Shaft diameter d

Example 6.40: Design a muff coupling to connect two steel shafts of diameter 50mm.

The permissible shear and compressive stresses for the shaft and key may be taken as 50 and $100\,N/mm^2$ ***respectively. The permissible shear stress for cast iron sleeve may be taken as*** $10\,N/mm^2$.

Solution:

a) To find the torque transmitted (M_t)

Torque transmitted by the shafts is given by

$$M_t = \frac{\pi d^3}{16}\eta\tau_s \qquad \text{... E(19.2)}$$

$$= \frac{\pi}{16}(50)^3 \times 0.75 \times 50$$

$$= 920388.47 \text{ N-mm}$$

b) To find the key dimensions

For $d = 50mm,$ referring table (17.4),

$$\text{Width of key} \quad b = 14mm$$

$$\text{Depth of key} \quad h = 9mm$$

$$\text{Length of key} = \text{Length of sleeve} = 3.5d \qquad \text{... E(19.48)}$$

$$= 3.5 \times 150 = 175mm$$

Check for stresses induced in key

Shear stress induced in key

$$\tau_{key} = \frac{2M_t}{bld} \text{ form } M_t = bl\tau\frac{d}{2}$$

$$= \frac{2 \times 920388.47}{14 \times 175 \times 50}$$

$$= 15.03\ N/mm^2$$

Compressive stress σ_c induced in key from equation $M_t = \frac{h}{2}l\sigma_c\frac{d}{2}$ is

$$\sigma_c = \frac{4M_t}{hld}$$

$$= \frac{4 \times 920388.47}{9 \times 175 \times 50}$$

$$= 46.75 \ N/mm^2$$

We see that both of these stresses are well within the permissible values and hence the design is safe.

c) To find the dimensions of sleeve (Diameter D and length l)

The outside diameter of sleeve

$$D = 2d + 0.013 \qquad \text{... E(19.45a)}$$

$$= 2 \times \frac{5}{100} + 0.013 = 0.113m$$

$$D = 113mm$$

$$Length\ of\ sleevekey = Length\ of\ key = 3.5d$$

$$= 3.5 \times 50 = 175mm$$

Muff Coupling with Tapered Pins

These couplings are exactly identical to sleeve coupling excepting that the key is replaced by taper pins.

Figure-6.26

Proportions of Sleeve and Pins

If *d* is shaft diameter, then

Outside diameter of sleeve	$D = 1.5d$
Length of sleeve	$L = 3.5d$
Mean diameter of Pin	$d_p = 0.25d$

Example 6.41: Design a muff coupling with taper pins to transmit 25kW at 1000rpm limiting the shear stress in shaft and pin to 40MPa. Take shear stresses in sleeve as 20MPa.

Solution:

a) To find the torque transmitted and shaft diameter d

Torque transmitted by the shaft is

$$M_t = \frac{9550N}{n} \text{ N-m}$$

$$= \frac{9550N}{n} \times 10^3 \text{ N-mm}$$

$$= \frac{9550 \times 25 \times 10^3}{1000} = 238750 \text{ N-mm}$$

But, Torque $M_t = \dfrac{\pi d^3}{16} \tau_s$ ... E(19.2)

$$238750 = \frac{\pi d^3}{16} \times 40$$

$$d = 31.21 mm$$

Use shaft diameter $d = 35 mm$

b) To find the dimensions of sleeve

The outside diameter of sleeve

$$D = 1.5d$$

$$= 1.5 \times 35 = 52.5 mm$$

The length of sleeve

$$L = 3.5d$$

$$= 3.5 \times 35 = 122.5 mm$$

To check for shear stress in sleeve

The sleeve may be designed as a hollow shaft with outside diameter *D*, inside diameter *d* transmitting torque M_t.

$$M_t)_{sleeve} = \frac{\pi\left(D^4 - d^4\right)}{16D} \tau_{sleeve}$$

Therefore, $\tau_{sleeve} = \dfrac{M_t \times 16D}{\pi\left(D^4 - d^4\right)}$

Here, $M_t = 238750 \text{ N-mm}$

$$D = 48 mm$$

$$d = 35 mm$$

Therefore, $\tau_{sleeve} = \dfrac{238750 \times 16 \times 48}{\pi\left(48^4 - 35^4\right)} = 15.33 \; N/mm^2$

Which is less than the allowable value $20\,N/mm^2$ and hence the design is safe.

c) ***To find Pin diameter*** d_p

Now, Shear load F on Pin $= \frac{M_t}{r}$

$$= \frac{238750}{\frac{35}{2}} = 13642.9 \quad \text{Since } r = \frac{d}{2}$$

$$\tau_{pin} = \frac{F}{2A} = \frac{F}{2\frac{\pi}{4}\left(d_p^2\right)}$$

$$40 = \frac{13642.9}{2\frac{\pi}{4}\left(d_p^2\right)}$$

Therefore, $d_p = 14.74mm$

Use diameter of pin $d_p = 15mm$

Design of Pin or Bush type Flexible Coupling

Figure-6.27 (Figure-19.4 MDH)

i) ***The torque transmitted by the coupling is calculated from equation***

$$M_t = \frac{9550N}{n} \text{ N-m} \qquad \text{... E(19.3c)}$$

Where, $N = Power\ transmitted\ in\ kW$

$n = Speed\ in\ rpm$

And is equated to the torque transmitted by the shafts given by

$$M_t = \frac{\pi d^3}{16}\eta\tau_s \qquad \text{... E(19.2)}$$

***To obtain the shaft diameter* 'd'.**

ii) ***Based on shaft diameter 'd', the width 'b' and thickness 'h' of the key are se-***

lected from table (17.4). The key is checked for shear and compression and the length of key 'l' is found.

iii) The number of bolts **'i'** ***is usually taken as 4 or 6 (an even number preferably).***

The length of the rubber bush is obtained from figure(8.5) $\left(Note : l = 0.625d\right)$

Knowing (or assuming) bearing pressure P_b, the overall diameter d' at the enlarged end of pin is obtained from equation

$$M_t = i\left(F\right)\frac{D_1}{2} \qquad \text{... E(19.33a)}$$

$$= i\left(P_b l d'\right)\frac{D_1}{2} \qquad \text{... E(19.33b)}$$

From E(19.33a) and E(19.33b),

$$F = Pbld'$$

Obtain d' the overall diameter of the Pin (bolt at the enlarged end).

The size d_2 of the enlarged end of bolt is obtained by subtracting $2\left(t_r + t_b\right)$ from d'

Where, $t_r = Thickness\ of\ rubber\ bush, mm$

$t_b = Thickness\ of\ brass\ sleeve, mm$

Finally the diameter d_1 of the smaller end is approximated as $d_1 = 0.5d_2$

iv) The pin designed is checked for maximum shear stress using equations

Shear stress in Pin

$$\tau_p = \frac{F}{0.785 d_p^2} \qquad \text{... E(19.34)}$$

Where, $d_p = d_1 = Diameter\ of\ Pin$

Bending stress in Pin

$$\sigma_b = \frac{F\left(\frac{l}{2} b\right)}{\frac{\pi}{32} d_p^3} \qquad \text{... E(19.35)}$$

The shear stress induced in Pin is given by

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

And compared with the allowable shear stress in Pin and the Principal normal stress is given by

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

Example 6.42: Design a bushed Pin type flexible coupling to transmit 25kW at 500rpm by selecting suitable material for shafts, key and bolts. (VTU-Dec08/Jan09)

Solution:

a) To find the shaft diameter 'd'

$$M_t = \frac{9550N}{n} \text{ N-m}$$

$$= \frac{9550N}{n} \times 10^3 \text{ N-mm}$$

$$= \frac{9550 \times 25 \times 10^3}{500}$$

$$= 477.5 \times 10^3 \text{ N-mm}$$

Torque transmitted by the shaft $= \frac{\pi d^3}{16}\eta\tau_s$ ... E(19.2)

Where 'd' is the shaft diameter

$\eta = Keyway\ \ factor = 0.75$

$\tau_s = Shear\ stress\ \ for\ the\ shaft\ material\ in\ N/mm^2$

Assuming 40C8(C40) as the material of the shaft, bolts and key

$\sigma_u = 569MPa$ ... T(1.25)

$$\sigma_y = 0.6\sigma_u$$

$$= 0.6 \times 569 = 341.4MPa$$

$$\tau_y = \frac{\sigma_y}{2}$$

$$= \frac{341.4}{2} = 170.7\, N/mm^2$$

$$\tau_{ed} = \frac{\tau_y}{factor\ of\ safety}$$

$$= \frac{170.7}{3} = 56.9\, N/mm^2$$

Therefore, $M_t)_{shaft} = \frac{\pi d^3}{16} \times 0.75 \times 56.9$

Equating $M_t)_{coupling}\ to\ M_t)_{shaft}\ ,\ we\ get$

$$477.5 \times 10^3 = \frac{\pi d^3}{16} \times 0.75 \times 56.9$$

i.e., $d = 38.48mm$

Adopt Diameter of shaft $d = 40mm$

b) Design of key

Corresponding to $d = 40mm,$ Referring table (17.4)

Width of key, $b = 12mm$

Depth of key, $h = 8mm$

To find length of key l

Note: Since $b > h,$ ***the key fails by compression (crushing) before it fails by shear.***

Therefore, $M_t)_{key\ in\ compression} = \frac{h}{2} l \sigma_c \frac{d}{2}$

i.e., $477.5 \times 10^3 = \frac{8}{2} l (2 \times 56.9) \times \frac{40}{2}\ (assuming\ \sigma_c = 2\tau)$

Therefore, $l = 53.79mm$

Adopt $l = 54mm$

Therefore, the key dimensions are,

Width, $b = 12mm$

Thickness, $h = 8mm$

Length $l = 54mm$

c) To design bolts or Pins

Assuming number of bolts $i = 6,$ (preferably a minimum number of 4 bolts)

Torque, $M_t = iF\dfrac{D_1}{2}$ ... E(19.33a)

Where, $M_t = 477.5\times10^6$ N-mm

$i = 6$

$D_1 = Bolt\ circle\ diameter = 3d$

$= 3\times40 = 120mm$

Therefore, from equation (19.33a),

$$477.5\times10^3 = 6F\times\frac{120}{2}$$

Where, $F = Shear\ force\ on\ each\ Pin\ at\ the\ pitch\ circle\ diameter$

$= 1326.4N$

But from E(19.33b) and (19.33a),

$F = p_b l d'$

Where, $l = Length\ of\ bush = 0.625d\ \left(from\ figure - 5.39 \right)$

$= 0.625\times40 = 25mm$ (Figure-19.4MDH)

Assuming $P_b = 1N/mm^2$

$d' = Overall\ diameter\ at\ the\ enlarged\ end\ of\ Pin = 53.1mm$

From figure (5.61), the overall diameter

$d' = d_2 + 2\left(t_b + t_r\right)$

i.e., $53.1 = d_2 + 2\left(2+5\right)$

$d_2 = Diameter\ at\ enlarged\ end = 39.1mm$

Taking smaller diameter of Pin

$d_1 = 0.5d_2$

$= 0.5\times39.1 = 19.55mm$

d) Check for stress in Pin

The shear stress in Pin

$$\tau_p = \frac{F}{0.785 d_p^2} \qquad \text{... E(19.34)}$$

Where $d_p = d_1 = Diameter\ of\ Pin = 19.55mm$

Therefore, $\tau_p = \dfrac{1326.4}{0.785 \times (19.55)^2} = 4.42\ N/mm^2$

Bending stress in Pin

$$\sigma_b = \frac{F\left(\frac{l}{2} + b\right)}{\frac{\pi}{32} d_p^3} \qquad \text{... E(19.35)}$$

Where $b = Clearance\ between\ flanges = 0.1d\ (19.4MDH)$

$$= 0.1 \times 40 = 4mm$$

Therefore, substituting in E(19.35),

$$\sigma_b = \frac{1326.4\left(\frac{25}{2} + 4\right)}{\frac{\pi}{32}(19.55)^3} = 29.83\ N/mm^2$$

Now, Maximum shear stress induced in Pin is given by

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

Where, $\sigma = \sigma_b = 29.83\ N/mm^2$

$$\tau = \tau_p = 4.42\ N/mm^2$$

Therefore, $\tau_{induced} = \dfrac{1}{2}\sqrt{(29.83)^2 + 4(4.42)^2}$

$$= 15.56\ N/mm^2$$

Principal stress $\sigma' = \dfrac{1}{2}\left[\sigma + \sqrt{\sigma^2 + 4\tau^2}\right]$

$$= \frac{1}{2}\left[29.83 + \sqrt{29.83^2 + 4\left(4.42\right)^2}\right]$$

$$= 30.47\ N/mm^2$$

From the above analysis we see that the induced stresses are less than the permissible value $56.9\ N/mm^2$ and hence the design is safe.

e) To find the dimensions of flange

From figure (19.4MDH), outside diameter of flange

$$D = 4d$$

$$= 4 \times 110 = 160mm$$

Hub diameter of flange

$$D_2 = 2d$$

$$= 2 \times 40 = 80mm$$

Thickness of flange

$$t = 0.375d$$

$$= 0.375 \times 40 = 15mm$$

Design of Oldhan Coupling

This coupling consists of two flanges keyed to two shafts. A slot is cut in each flange (of some size) such that when assembled these slots are notmal to each other. A disc having perpendicular projections (called tongues) is inserted between the flanges. The tounges fit lossely in the flanges. The material of this coupling is steel or cast iron.

Oldham couplings can be used for any velocity or load and it generally causes small addition loads on the bearings. The advantages of this couplings is that it can be used for shafts which are not coaxial but whose axes are parallel to the extent of $0.05d$ where *'d'* is the shaft diameter. This coupling is generally used upto a maximum speed of 1500rpm.

Figure-6.29: Oldham's Coupling (Figure-19.5MDH)

The design of this coupling is based on the allowable pressure between the faces of the slots and the tongue. This pressure is not to exceed $8.3\ N/mm^2$.

Example 6.43: Design an Oldham coupling to transmit 50kW at 1200rpm. The permissible shear and compressive stresses for the shaft material are 50 and 100MPa respectively. The permissible pressure between the slots and tongues is not to exceed $8\ N/mm^2$.

Solution:

a) To find the shaft diameter (d)

The torque transmitted by coupling is obtained from equation

$$M_t = \frac{9550N}{n} \text{ N-m} \qquad \text{... E(19.3c)}$$

$$= \frac{9550 \times 50 \times 10^3}{1200} \text{ N-mm}$$

But, $M_t = \frac{\pi d^3}{16} \eta \tau_s$

Where $d = Shaft\ diameter$

$\eta = Keyway\ \ factor = 0.75$

$$397916.7 = \frac{\pi d^3}{16} \times 0.75 \times 50$$

Therefore, $d = 37.81mm$

Use shaft diameter $d = 40mm$

b) To find key dimensions (b, h and l)

For $d = 40mm,$ referring table (17.4)

Width of key, $b = 10mm$

Depth of key, $h = 8mm$

Note: Since $b > h,$ ***the key fails by compression before it fails by shear***

$$\left.M_t\right)_{key\ in\ compression} = \frac{h}{2} l \sigma_c \frac{d}{2}$$

$$397916.7 = \frac{8}{2} l(100) \times \frac{40}{2}$$

Therefore, $l = 50mm$

c) Dimension of Flanges and Tongue

i) Diameter of the disc $D = 3d + a$

Where, $a = Distance\ between\ centre\ lines\ of\ \ shafts$

$= 0.05d$

$$= 0.05 \times 40 = 2mm$$

Therefore, $D = 3 \times 40 + 1.9$

$$= 121.9mm$$

ii) The diameter of the boss

$$D_2 = 2d \qquad \text{... E(19.40)}$$

$$= 2 \times 40 = 80mm$$

iii) Length of Boss

$$l = 1.75d \qquad \text{... E(19.41)}$$

$$= 1.75 \times 40 = 70mm$$

iv) Width of groove

$$w = \frac{D}{6} \qquad \text{... E(19.42)}$$

$$= \frac{121.9}{6} = 20.31mm$$

v) Thickness of groove

$$h_1 = \frac{w}{2} \qquad \text{... E(19.43a)}$$

$$= \frac{20.31}{2} = 10.15mm$$

vi) Thickness of central disc

$$h = \frac{w}{2} \qquad \text{... E(19.43b)}$$

$$= \frac{20.31}{2} = 10.15mm$$

vii) The thickness of flanges

$$t = \frac{3}{4}d \qquad \text{... E(19.44)}$$

$$= \frac{3}{4} \times 40 = 30mm$$

Exercises

1) *Design a protected type CI flange coupling to transmit 20kW at 1400rpm. The maximum torrque is to be 25% more than the mean torque. Select suitable materials for shaft, key and bolts. Base the design on factor of safety of 3.*

2) *It is required to transmit 25kW at 1200rpm using a protected type flange coupling. The shaft key and bolts are made of same material 40C8. using factor of safety of 3, design the coupling.*

3) *In a flange coupling used to connect two coaxial shafts of diameter 100mm to transmit 75kW at 500rpm, 6,* $M \times 20 \times 2.5$ *bolts are used on a p.c.d. of 300mm. The hub diameter is 200mm and flange thickeness is 25mm. Determine*
   - i) *The shear stress induced in shafts and bolts.*
   - ii) *The shear stress induce in bolts using* $\sigma_c = 100 MPa$
   - iii) *The shear stress induced in flanges.*

4) *Design a marine flange coupling solid to transmit 250kW at 150rpm. Assume same material for shaft, bolts with an allowable stress of* $60\, N/mm^2$

5) *Design a bushed pin type flexible coupling to transmit 20kW at 100rpm. Select materials for shafts, key and bolts. Base the design on a factor of safety of 3.*

6) *Design a rigid flange coupling to transmit 10kW at 100rpm. The permissible stress for the coupling key and shaft are 65MPa in tension, 35MPa in shear and 100MPa in compression.* *(VTU-March/April 2000)*

7) *Design a cast iron flange coupling (protected type) to connect two shafts and transmit torque of 500N-m. The following stresses may be used. Permissible shear stress for shaft, bolt and key material is 50MPa. Permissible crushing stress for bolt and key material is 150MPa. Permissible shear stress for C.I. is 8MPa.*
(VTU-Sept/Oct 2000)

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

9) *Design a rigid flange coupling to transmit 10kW at 100rpm. The permissible stresses for the coupling, key and shaft are 65MPa in tension, 35MPa in shear and 100MPa in compression.* *(VTU-March/April 2000)*

10) *Determine the dimension of a tapered steel key required to transmit 15kW at 1200rpm. The permissible shear and compressive stresses for the material of key and shar are 55 and 110MPa respectively. Also calculate the axial force required to drive key home.*

Ans: $d = 25mm$

11) *Determine the power transmitted at 1200rpm by a tapered key designated by* $40 \times 20 \times 250mm$ *limiting the shear and compressive stresses to 60 and 110MPa respectively, Shaft diameter is 100mm.*

Ans: $N = 1727.75kW$

12) *Determine the width, thickness and length of a Gib head key required to transmit 20kW at 1400rpm by limiting the shear stress to 50MPa. The compressive stress may be taken as 1.9 times shear stress.*

Ans: $b = 8mm,\ h = 7mm$ *and* $l = 28mm$

13) *Determine the dimensions of a rectangular parallel key to transmit 20kW at 1400rpm and also the axial force required to assemble the key. Use* $\tau_{allowable} = 60MPa$ *and* $\sigma_{c\,allowable} = 110MPa$

Ans: $b = 8mm,\ h = 7mm$ *and* $l = 28.34mm$

14) *A splined connection in an automobile transmission consists of 6 splines cut on a shaft 100mm diameter. The keyways in the hub are 60mm long. Taking the permissible peressure on the splines as* $10\,N/mm^2$*, determine the power transmitted at 100rpm.*

Ans: $N = 42.22kW$

15) *Determine the power transmitted at 1200rpm by a splined shaft designated by* $6 \times 130 \times 145$ *having length 80mm by limiting the permissible normal pressure on splines to* $8\,N/mm^2$.

Ans: $N = 248.8kW$