Design of Machine Elements – 1

Module – III

Design of Joints, Couplings and Keys

Design Data Hand Book Referred – Dr. K. Lingaiah Volume1

Design of Keys

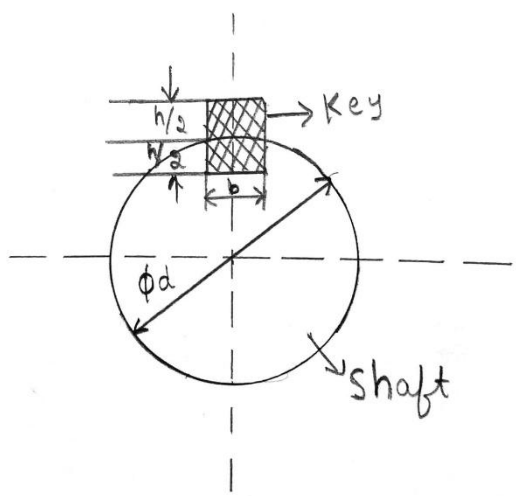
A key is a machine element which is used to prevent the relative rotational motion between the shaft and the connecting member such as flanges, discs, coupling, gears etc. through which the process is being transmitted.

The different types of keys are sunk key, saddle keys taper pin, Pink key etc.

The different types of Sunk keys are:

1. Rectangular or square sunk key
2. Wood ruff key
3. Gib head key
4. Feather key etc.

Derive torque transmitted by a key in shear and in compression



$Fig\ \frac{17.1}{17.2}$

Let d be the dia of the shaft which transmits torque M_t , Let 'l', 'b', 'h' be the length, width and depth of the key respectively. The key way fails either by compressive loads or shear loads.

WKT, $M_{t_{comp}} = F_{comp} \times Radius$ of Shaft

$$\sigma_c = \frac{F_c}{A}$$

$$F_c = \sigma_c \times A$$

$$= \sigma_{comp} \times \frac{h}{2} \times l \times \frac{d}{2}$$

$$M_{t_{comp}} = \sigma_c \times \frac{h}{2} \times l \times \frac{d}{2}$$

$$h = \frac{4M_{t_{comp}}}{\sigma_c \times l \times d} \qquad \Rightarrow 19.51$$

Similarly

$M_{t_{shear}} = F_s \times Radius$ of shaft

$$\tau_s = \frac{F_s}{A}$$

$$F_s = \tau_s \times A$$

$$= \tau_s \times b \times l$$

$$M_{t_{shear}} = \tau_s \times b \times l \times \frac{d}{2}$$

$$b = \frac{2M_{t_{shear}}}{\tau_s \times l \times d} \qquad \Rightarrow 19.50$$

Note:

1. Generally, the compressive stress is twice the shear stress. $\sigma_c = 2\tau_s$
2. If b>h, then the design is due to compression, Since the member is going to fail by compression.
3. If h>b, then the design is due to shear, Since the member is going to fail by shear.
4. Find dia of the shaft by considering keyway factor or shaft factor i.e.

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

where η = Keyway factor or shaft factor =0.75

η = 0.75 to 0.9

$\tau_s = allowable$ shear stress

Problems:

1. Prove that a square key is equally strong in both M_t compression and in shear.

   Sol:

   WKT, for a square key b=h and also

   WKT $\sigma_c = 2\tau_s$

   WKT strength of the key in compression is given by

$$M_{t_{comp}} = \sigma_c \times \frac{h}{2} \times l \times \frac{d}{2}$$

   Similarly WKT the strength of the key in shear is given by

$$M_{t_{shear}} = \tau_s \times b \times l \times \frac{d}{2}$$

$$\frac{M_{t_{comp}}}{M_{t_{shear}}} = \frac{\sigma_c \times \frac{h}{2} \times l \times \frac{d}{2}}{\tau_s \times b \times l \times \frac{d}{2}}$$

$$= \frac{2\tau_s}{\tau_s} \times \frac{h}{2b} \qquad (\because \sigma_c = 2\tau_s)$$

$$\frac{M_{t_{comp}}}{M_{t_{shear}}} = \frac{h}{b} = 1 \qquad (\because Square\ key\ h = b)$$

$$M_{t_{comp}} = M_{t_{shear}}$$

2. A rectangular key of 15mm width and 12mm thickness is required to transmit a torque of 800Nm from a shaft 40mm dia, taking allowable values of stresses in shear and compression as 58MPa and 110MPa respectively. Find the length of the key required, also find the length of the key, If 12mm width and 15mm thickness.

   Sol:

   Case 1. Given b=15mm, h=12mm, M_t =800×10^3 N-mm,

   d=40mm, $\tau_s = 58MPa$, $\sigma_c = 110MPa$, $l = ?$

   Since b is greater than h $(b > h)$

$$M_{t_{comp}} = \sigma_c \times \frac{h}{2} \times l \times \frac{d}{2}$$

$$800 \times 10^3 = 110 \times \frac{12}{2} \times l \times \frac{40}{2}$$

$$l = 60.6mm$$

Case 2. If b=12mm, h=15mm

Since h is greater than b $(h > b)$

$$M_{t_{shear}} = \tau_s \times b \times l \times \frac{d}{2}$$

$$800 \times 10^3 = 58 \times 12 \times l \times \frac{40}{2}$$

$$l = 57.47mm$$

3. Find the dimensions of the steel tapered key to transmit 20KW at 1800rpm, allowable shear and compressive stresses are 80MPa and 170MPa respectively. Also calculate the axial force required to drive the key way.

Sol:

N = 20kw, n=1800rpm, $\tau_s = 80Mpa$, $\sigma_c = 170Mpa$, $F_a = ?$

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

$$= \frac{9550 \times 20 \times 10^3}{1800} = 106.111 \times 10^3 \, Nmm$$

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

$$\eta = 0.75$$

$$106.111 \times 10^3 = \frac{\pi d^3}{16} \times 0.75 \times 80$$

$$d = 20.81mm$$

but, $d_{std} = 22mm$ $\Rightarrow T \rightarrow \frac{16.10}{16.15}$

$T \rightarrow \frac{17.4}{17.13}$ for $d_{std} = 22mm$

$h = 6mm$ b=6mm

Since b=h= 6mm,

The strength is going to be the same in compression and shear.
Hence design is either due to compression or due to shear.
Taking,

$$M_{t_{shear}} = \tau_s \times b \times l \times \frac{d}{2}$$

$$106.11 \times 10^3 = 80 \times 6 \quad \times l \times \frac{22}{2}$$

$$l = 20.09 mm$$

To find Axial Force:

$$F_a = 2\mu_2 F + F \ \tan\beta \qquad\qquad \Rightarrow \frac{17.11}{17.2}$$

$Take \ \ \mu_2 = 0.10$ and $\tan\beta = 0.0104$ if the taper is 1:100

Where F = normal frictional force

WKT,

$$F = \frac{F_t}{\mu}$$

F_t = tangential force

μ=Co-effcient of friction

Take μ=0.3 always

Wkt, $M_t = F_t \times \frac{d}{2}$

$$F_t = \frac{2M_t}{d}$$

$$= \frac{2 \times 106.11 \times 10^3}{22}$$

$$F_t = 9.64 \times 10^3 N$$

$$\therefore F = \frac{9.646 \times 10^3}{0.3}$$

$$F = 32.153 \times 10^3 N$$

$$F_a = 2 \times 0.10 \times 32.153 \times 10^3 + 32.153 \times 10^3 \times 0.0104$$

$$F_a = 6.764 \times 10^3 N$$

Design of Couplings

A coupling is a machine element which is used to connect 2 shaft and to increase the length of the shafts for transmitting more power.

The different types of shaft couplings are rigid coupling and flexible coupling.

RIGID COUPLING:

These are used to connect 2 shafts which are perfectly aligned to each other. It will be further classified into

1. Sleeve coupling or Muff coupling

2. Flange coupling {Protected or Unprotected}

FLEXIBLE COUPLING:

These are used to connect 2 shafts which are not perfectly aligned either by angular miss alignment or lateral miss alignment. It can be further classified into

1. Bush pin flexible coupling

2. Universal flexible coupling

3. Old ham’s coupling

PROCEDURE FOR FLANGE COUPLING

$$\text{Fig}\,\frac{19.1}{19.15\text{ pg.no }271}$$

Step 1 : Find the torque

$$M_t = \frac{9550N}{\eta} \qquad \Rightarrow \frac{19.3c}{19.3\text{ pg.no }259}$$

Step 2 : Find the dia for the shaft

$$M_t = \frac{\pi d^3}{16} \times \eta \times \tau_s \quad \Rightarrow \frac{19.2}{19.3\text{ pg.no }259}$$

$$\eta = 0.75$$

$$\text{d}_{\text{std}} \rightarrow \text{T} \Rightarrow \frac{16.10}{16.15}$$

Step 3 : Find the dimensions of the key based on

$$d_{std} \rightarrow T \Rightarrow \frac{17.4}{17.13}$$

Step 4 : Find the length of key based on compression or shear

Step 5 : Find the no of bolts (i)

$$i = 0.02d + 3mm \qquad \Rightarrow \frac{19.1b}{19.3 \text{ pg.no } 259}$$

Step 6 : Find the bolt circle dia or pitch circle dia

$$D_1 = 2d + 50mm \qquad \Rightarrow \frac{19.12b}{19.4 \text{ pg.no } 260}$$

Step 7 : Find the bolt dia

$$d_1 = \frac{0.5d}{\sqrt{i}} \qquad \Rightarrow \frac{19.8}{19.3 \text{ pg.no } 259}$$

$$d_1 = T \rightarrow \frac{18.7}{18.17}$$

Step 8 : Check the stress in the bolt

$$M_{t_{bolts}} = i \times \frac{\pi d_1^2}{4} \times \tau_b \times \frac{D_1}{2} \qquad \Rightarrow \frac{19.4}{19.3 \text{ pg.no } 259}$$

if $\tau_b \langle \tau_{by}$ The design is safe

where:

τ_b = developed shearin the bolts

τ_{by} = allowable shearin the bolts

Step 9 : Hub dimensions

a) hub dia (D_2) = 1.5d + 25mm $\Rightarrow \frac{19.13b}{19.4 \text{ pg.no } 260}$

b) hub length (l) = 1.25d + 18.75mm $\Rightarrow \frac{19.14d}{19.4 \text{ pg.no } 260}$

c) Outside dia (D) =2.5d + 75mm $\Rightarrow \frac{19.14b}{19.4 \text{ pg.no } 260}$

d) thickness of the flange [t] = 0.25d $\Rightarrow \frac{19.17}{19.4 \text{ pg.no } 260}$

Step 10 : check the stresses in the flange based on thickness

$$M_t = t \times \pi D_2 \times \tau_b \times \frac{D_2}{2} \quad \Rightarrow \frac{19.6}{19.3 \text{ pg.no } 259}$$

if $\tau_b \langle \tau_{by}$ The design is safe

Problems:

1. Design a rigid flange coupling to transmit 50 kw at 500 rpm, assume C-40 material for shaft and bolts and assume cast iron for flange. Take C-40 steel for keys as that of shaft since not given.

Sol.

N=50kw, n=500rmp

Shaft, bolts and keys = C-40

Flange = Cast iron

Step 1 Find the torque

$$M_t = \frac{9550N}{\eta} \qquad \Rightarrow \frac{19.3c}{19.3 \text{ pg.no } 259}$$

$$M_t = \frac{9550 \times 50 \times 10^3}{500} = 9.55 \times 10^5\, Nmm$$

Step 2 Find the dia for the shaft

$$M_t = \frac{\pi d^3}{16} \times \eta \times \tau_d \qquad \Rightarrow \frac{19.2}{19.3 \text{ pg.no } 259}$$

$$9.55 \times 10^5 = \frac{\pi \times d^3}{16} \times 0.75 \times \tau_c$$

Given C-40 material for shaft

$\therefore$ Fos=2.5

$$\sigma_y = 328 Mpa \qquad \Rightarrow \text{T} \rightarrow \frac{1.5}{6}$$

$$\sigma_{yd} = \frac{\sigma_y}{Fos} = \frac{328.6}{2.5} = 131.44 Mpa$$

$$\therefore \tau_s = \frac{\sigma_y d}{2} = \frac{131.44}{2} = 65.72 Mpa$$

$$\therefore 9.55 \times 10^5 = \frac{\pi \times d^3}{16} \times 0.75 \times 65.72$$

$$d = 46.210 mm$$

$$d_{std} = 48 \qquad \Rightarrow \text{T} \rightarrow \frac{16.10}{16.15}$$

Step 3 Find the dimensions of the key based on

$$\text{d}_{\text{std}} \rightarrow \text{T} \Rightarrow \frac{17.4}{17.13}$$

Take h = 9mm, b=14mm

Step 4 Since b > h

$$M_{t_{comp}} = \sigma_c \times \frac{h}{2} \times l \times \frac{d}{2}$$

Wkt σ_c=131.44 Mpa Q C-40 Material calculated in step 2

$$\therefore 9.55\times10^5 = 131.44\times\frac{9}{2}\times l\times\frac{48}{2}$$

$$l = 67.27mm$$

Step 5 Find the no of bolts (i)

$$i = 0.02d + 3mm \qquad \Rightarrow \frac{19.1b}{19.3 \text{ pg.no } 259}$$

$$i = 0.02(48) + 3mm$$

$$i = 3.96\ ;\ \ 4bolts$$

Step 6 Find the bolt circle dia or pitch circle dia

$$D_1 = 2d + 50mm \qquad \Rightarrow \frac{19.12b}{19.4 \text{ pg.no } 260}$$

$$D_1 = 2(48) + 50mm$$

$$D_1 = 146mm$$

Step 7 Find the bolt dia

$$d_1 = \frac{0.5d}{\sqrt{i}} \qquad \Rightarrow \frac{19.8}{19.3 \text{ pg.no } 259}$$

$$d_1 = \frac{0.5\times48}{\sqrt{i}}$$

$$d_1 = \frac{0.5\times48}{\sqrt{4}} = 12mm$$

$$d_{std} = 12mm \qquad \text{T} \rightarrow \frac{18.7}{18.17}$$

Step 8 Check the stress in the bolt

$$M_{t_{bolts}} = i\times\frac{\pi d_1^2}{4}\times\tau_b\times\frac{D_1}{2} \qquad \Rightarrow \frac{19.4}{19.3 \text{ pg.no } 259}$$

$$9.55\times10^5 = i\times\frac{\pi 12^2}{4}\times\tau_b\times\frac{146}{2}$$

$$\tau_b = 28.917Mpa$$

Wkt τ_{yd} =65.72 Mpa Q C-40 Material calculated in step 2

$\therefore \; \tau_b \langle \tau_{yd} \, (\tau_{by})$

28.917⟨65.72 Mpa

Hence design is safe

Step 9 Hub dimensions

a) hub dia (D_2) = 1.5d + 25mm $\Rightarrow \dfrac{19.13b}{19.4 \text{ pg.no } 260}$

= 1.5(48)+25

=97mm

b) hub length (l) = 1.25d + 18.75mm $\Rightarrow \dfrac{19.14d}{19.4 \text{ pg.no } 260}$

= 1.25(48) + 18.75mm

= 78.75mm

c) Outside dia (D) =2.5d + 75mm $\Rightarrow \dfrac{19.14b}{19.4 \text{ pg.no } 260}$

=2.5(48) + 75mm

=195mm

d) thickness of the flange [t] = 0.25d $\Rightarrow \dfrac{19.17}{19.4 \text{ pg.no } 260}$

=0.25(48)

=12mm

Step 10 Check the stresses in the flange based on thickness

$$M_t = \text{t} \times \pi D_2 \times \tau_b \times \frac{D_2}{2} \quad \Rightarrow \frac{19.6}{19.3 \text{ pg.no } 259}$$

$$9.55 \times 10^5 = 12 \times \pi (97) \times \tau_b \times \frac{97}{2}$$

$$\tau_f = 5.384 Mpa$$

Given Cast iron for flange

Wkt $\sigma_{ut} = 124.5 Mpa \qquad \Rightarrow \text{T} \rightarrow \dfrac{1.3}{Pg.no \ 4}$

$$\sigma_y = \frac{\sigma_{ut}}{2} = \frac{124.5}{2} = 62.25 Mpa$$

$$\sigma_{yd} = \frac{\sigma_y}{Fos} = \frac{62.25}{2.5} = 24.9 Mpa$$

$$\tau_{yd} = \frac{\sigma_{yd}}{2} = \frac{24.9}{2} = 12.45 Mpa$$

$\therefore \; \tau_f \langle \tau_{yd}$

i.e 5.384 Mpa$\langle$12.45Mpa

$\therefore$ Design is safe

2. Design a Cast iron flange coupling for transmitting a torque of 5000Nm by assuming suitable materials for shafts, bolts and keys.

Sol.

Given $M_t = 5000 Nm$

$= 5000 \times 10^3 Nmm$

Flange $\rightarrow$ CastIron

Assume the material as C-40 for shaft, Keys and bolts with FOS=2.5

Step 1 Torque

Given $M_t = 5000 Nm$

Step 2 Find the dia for the shaft

$$M_t = \frac{\pi d^3}{10} \times \eta \times \tau_d$$

Given C-40 material for shaft

$\therefore$ Fos = 2.5

$\sigma_y = 328.6 Mpa$ $\qquad \Rightarrow \text{T} \rightarrow \frac{1.5}{6}$

$$\sigma_{yd} = \frac{\sigma_{yd}}{2} = \frac{131.44}{2} = 65.72 Mpa$$

$$5000\times10^{3}=\frac{\pi d^{3}}{16}\times0.75\times65.72$$
$$d=80.24mm$$

$d_{std}=85mm$ $\quad$ $T\rightarrow\frac{16.10}{16.16}$

Step 3 Find the dimensions of the key based on

$$d_{std}\rightarrow T\Rightarrow\frac{17.4}{17.13}$$

Take h = 14mm, b=22mm

Step 4 Since b > h

$$M_{t_{comp}}=\sigma_c\times\frac{h}{2}\times l\times\frac{d}{2}$$

$\sigma_c=131.44Mpa$ $\quad$ Q C-40 Material already calculated in step 2

$$5000\times10^{3}=131.44\times\frac{14}{2}\times l\times\frac{85}{2}$$
$$l=127.86mm$$

Step 5 Find the no of bolts (i)

$i=0.02d+3mm$ $\quad$ $\Rightarrow\frac{19.1b}{19.3\text{ pg.no }259}$

$$i=0.02(85)+3mm$$
$$i=4.7\;;\;5bolts$$

Step 6 Find the bolt circle dia or pitch circle dia

$D_1=2d+50mm$ $\quad$ $\Rightarrow\frac{19.12b}{19.4\text{ pg.no }260}$

$$D_1=2(85)+50mm$$
$$D_1=220mm$$

Step 7 Find the bolt dia

$d_1=\frac{0.5d}{\sqrt{i}}$ $\quad$ $\Rightarrow\frac{19.8}{19.3\text{ pg.no }259}$

$$d_1 = \frac{0.5 \times 85}{\sqrt{5}} = 19mm$$

$$d_{std} = 20mm \qquad \text{T} \rightarrow \frac{18.7}{18.17}$$

Step 8 Check the stress in the bolt

$$M_{t_{bolts}} = i \times \frac{\pi d_1^2}{4} \times \tau_b \times \frac{D_1}{2} \qquad \Rightarrow \frac{19.4}{19.3 \text{ pg.no } 259}$$

$$5000 \times 10^3 = 5 \times \frac{\pi 20^2}{4} \times \tau_b \times \frac{220}{2}$$

$$\tau_b = 28.93 Mpa$$

Wkt τ_{yd} =65.72 Mpa ∵ C-40 Material calculated in step 2

$\therefore \ \tau_b \langle \tau_{yd} \ (\tau_{by})$

28.93⟨65.72 Mpa

Hence design is safe

Step 9 Hub dimensions

a) hub dia (D_2) = 1.5d + 25mm $\Rightarrow \frac{19.13b}{19.4 \text{ pg.no } 260}$

= 1.5(85)+25

=152.5mm

b) hub length (l) = 1.25d + 18.75mm $\Rightarrow \frac{19.14d}{19.4 \text{ pg.no } 260}$

= 1.25(85) + 18.75mm

= 125mm

c) Outside dia (D) =2.5d + 75mm $\Rightarrow \frac{19.14b}{19.4 \text{ pg.no } 260}$

=2.5(85) + 75mm

=287.5mm

d) thickness of the flange [t] = 0.25d $\Rightarrow \frac{19.17}{19.4 \text{ pg.no } 260}$

=0.25(85)

=21.25mm

Step 10 Check the stresses in the flange based on thickness

$$M_t = t \times \pi D_2 \times \tau_b \times \frac{D_2}{2} \quad \Rightarrow \frac{19.6}{19.3 \text{ pg.no } 259}$$

$$5000 \times 10^5 = 21.25 \times \pi (152.5) \times \tau_b \times \frac{152.5}{2}$$

$$\tau_f = 6.4409 Mpa$$

Given Cast iron for flange

$$\sigma_y = \frac{\sigma_{ut}}{2} = \frac{124.5}{2} = 62.25 Mpa$$

$$\sigma_{yd} = \frac{\sigma_y}{Fos} = \frac{62.25}{2.5} = 24.9 Mpa$$

$$\tau_{yd} = \frac{\sigma_{yd}}{2} = \frac{24.9}{2} = 12.45 Mpa$$

$\therefore \ \tau_f \langle \tau_{yd}$

Wkt $\sigma_{ut} = 124.5 Mpa$ for grey cast iron $\Rightarrow T \rightarrow \frac{1.3}{Pg.no\ 6}$

i.e 6.4409 Mpa$\langle$12.45Mpa

$\therefore$ Design is safe

3. Design a flange coupling to connect a shaft to a motor with following specifications. Take pump output 3000 liters/min, Total head 20 mm, Pump speed 600 rpm. Take efficiency 70%. Select C-40 steel for shaft, C-35 for key with Fos = 2. Assume shear stress in cast iron flange is 15Mpa.

Sol.

Given Q = 3000litres/min

$$= \frac{3000}{60} = 50 \text{ litres/sec}$$

$$= 0.05\ m^3/\sec$$

h = 20mm

$\eta = 0.7$

Wkt $P = \frac{WQH}{1000} \ \frac{Nm}{Sec}$

Where w = mg (weigth density)

m = 1000Kg/m^3 (mass density)

$w = 1000 \times 9.81$

$= 9810\ N/m^3$

$$Q = \text{Discharge}$$

$$\therefore\ P = \frac{9810 \times 0.05 \times 20}{1000}$$

$$P = 9.81\ \text{Kw}$$

$$\therefore\ P_{total} = \frac{p}{\eta} = \frac{9.81}{0.7} = 14\text{Kw}$$

Given

$$n = 600\ \text{rpm},\ \text{Fos} = 2$$

Shaft and Bolts $\rightarrow$ C-40

Keys $\rightarrow$ C-35

Flange $\rightarrow$ Cast iron $\Rightarrow \tau_{yd} = 15Mpa$

Step 1 Find the torque

$$M_t = \frac{9550N}{\eta} \qquad \Rightarrow \frac{19.3c}{19.3\ \text{pg.no}\ 259}$$

$$M_t = \frac{9550 \times 14 \times 10^3}{600} = 222.83 \times 10^3\ Nmm$$

Step 2 Find the dia for the shaft

$$M_t = \frac{\pi d^3}{16} \times \eta \times \tau_d \qquad \Rightarrow \frac{19.2}{19.3\ \text{pg.no}\ 259}$$

$$9.55 \times 10^5 = \frac{\pi d^3}{16} 0.75 \times \tau_d$$

Given C-40 material for shaft

$\therefore$ Fos=2.5

$$\tau_d = ?$$

$$\sigma_y = 328Mpa \qquad \Rightarrow \text{T} \rightarrow \frac{1.5}{6}$$

$$\sigma_{yd} = \frac{\sigma_y}{Fos} = \frac{328.6}{2} = 164.3Mpa$$

$$\therefore\ \tau_{yd} = \frac{\sigma_y d}{2} = \frac{164.3}{2} = 82.15 Mpa$$

$$\therefore\ 222.83\times10^3 = \frac{\pi\times d^3}{16}\times0.75\times82.15$$

$$d = 26.40mm$$

$$d_{std} = 28 \qquad \Rightarrow \text{T}\rightarrow\frac{16.10}{16.15}$$

Step 3 Find the dimensions of the key based on

$$\text{d}_{\text{std}} \rightarrow \text{T} \Rightarrow \frac{17.4}{17.13}$$

Take h = 7mm, b=8mm

Step 4 Since b > h

$$M_{t_{comp}} = \sigma_c \times \frac{h}{2}\times l\times\frac{d}{2}$$

Given C-35 for keys Fos=2

$$\sigma_c = \frac{\sigma_{yd}}{2} = \frac{152}{2} = 76Mpa$$

$$\therefore 222.83\times10^3 = 152\times\frac{7}{2}\times l\times\frac{28}{2}$$

$$l = 29.91mm$$

Step 5 Find the no of bolts (i)

$$i = 0.02d + 3mm \qquad \Rightarrow \frac{19.1b}{19.3 \text{ pg.no } 259}$$

$$i = 0.02(28) + 3mm$$

$$i = 3.56\ ;\ \ 4bolts$$

Step 6 Find the bolt circle dia or pitch circle dia

$$D_1 = 2d + 50mm \qquad \Rightarrow \frac{19.12b}{19.4 \text{ pg.no } 260}$$

$$D_1 = 2(22) + 50mm$$

$$D_1 = 106mm$$

Step 7 Find the bolt dia

$$d_1 = \frac{0.5d}{\sqrt{i}} \qquad \Rightarrow \frac{19.8}{19.3 \text{ pg.no } 259}$$

$$d_1 = \frac{0.5 \times 25}{\sqrt{4}} = 7mm$$

$$d_{std} = 7mm \qquad \text{T} \rightarrow \frac{18.7}{18.17}$$

Step 8 Check the stress in the bolt

$$M_{t_{bolts}} = i \times \frac{\pi d_1^2}{4} \times \tau_b \times \frac{D_1}{2} \qquad \Rightarrow \frac{19.4}{19.3 \text{ pg.no } 259}$$

$$222.83 \times 10^3 = 4 \times \frac{\pi 7^2}{4} \times \tau_b \times \frac{106}{2}$$

$\tau_b = 27.31 Mpa$

but Wkt for C-40 steel

Take $\tau_{by} = \tau_s = 82.15 Mpa$ (calculated in step 2)

$\therefore \ \tau_b \langle \ \tau_{by}$

i.e 27.31Mpa < 82.15Mpa

$\therefore$ Design is safe

Step 9 Hub dimensions

a) hub dia (D_2) = 1.5d + 25mm $\Rightarrow \frac{19.13b}{19.4 \text{ pg.no } 260}$

= 1.5(28)+25

=67mm

b) hub length (l) = 1.25d + 18.75mm $\Rightarrow \frac{19.14d}{19.4 \text{ pg.no } 260}$

= 1.25(28) + 18.75mm

= 53.75mm

c) Outside dia (D) =2.5d + 75mm $\Rightarrow \frac{19.14b}{19.4 \text{ pg.no } 260}$

=2.5(28) + 75mm

=145mm

d) thickness of the flange [t] = 0.25d $\Rightarrow \frac{19.17}{19.4 \text{ pg.no } 260}$

=0.25(28)

=7mm

Step 10 check the stresses in the flange based on thickness

$$M_{t_{flange}} = \text{t} \times \pi D_2 \times \tau_b \times \frac{D_2}{2} \quad \Rightarrow \frac{19.6}{19.3 \text{ pg.no } 259}$$

$$222.83 \times 10^3 = 7 \times \pi (67) \times \tau_b \times \frac{67}{2}$$

$$\tau_f = 4.514 Mpa$$

Given $\tau_{by} = 15 Mpa$

$\therefore \tau_f \langle \tau_{yd}$

i.e 4.514 Mpa⟨15Mpa

$\therefore$ Design is safe

BUSH PIN TYPE FLEXIBLE COUPLING

From Fig $\frac{19.4}{19.15}$

From fig [b] or [c] = clearance

d = dia of the shaft

D_1 = pitch circle dia

D_2 = hub dia

D = Outside dia of the flange

L = length of the hub

Problems:

1. A pin and bush type flexible coupling is to be designed to transmit 30KW at 300rpm, allowable shear stress for shaft (τ_s) made of carbon steel is 55MPa and cast-iron flange of 15MPa. Pin is made of same material as that of shaft and bearing pressure in bush is 20MPa. Design a coupling.

Sol:

Given: N = 30KW, n = 300rpm, Shaft and pin = τ_{yd} = 55MPa,

Flange = τ_{yd} = 15MPa, P = 20 MPa

Considering for keys same material as that of shaft

$$M_t = \frac{9550N}{n} = \frac{9550 \times 30 \times 10^3}{300} = 9.55 \times 10^5\, Nmm$$

$$M_t = \frac{\pi d^3}{16} \times \eta \times \tau_{yd}$$ (Consider η = 0.75)

$$9.55 \times 10^5 = \frac{\pi \times d^3}{16} \times 0.5 \times 55$$

d = 49.03mm

$$d_{std} = 50mm \quad \Rightarrow \quad T - \frac{16.10}{16.15}$$

From $T-\frac{17.4}{17.13}$ for $d_{std} = 50mm$, b = 14mm, h = 9mm

Since b $>$ h

The torque is due to compression

$$M_{tcomp} = \sigma_c \times \frac{h}{2} \times l \times \frac{d}{2}$$

Given $\tau_{yd} = 55MPa$ (Q key and shaft are same material)

WKT $\sigma_c = \sigma_{yd} = 2\tau_{yd}$

$= 2 \times 55MPa$

$\sigma_c = 110MPa$

$$\therefore \quad 9.55 \times 10^5 = 110 \times \frac{9}{2} \times l \times \frac{50}{2}$$

l = 77.17mm

Dimensions Of Coupling

From table $\frac{19.6}{19.24}$ for $d_{std} = 50$mm

Take Coupling number = B_5

Outside dia of flange (D) = 170mm

Hub dia (D_2) = 80mm

Hub length (l) = 45mm

Flange width (l_1) = 35mm

Dia of the bolt (d_b) or Dia of the pin (d_p) = 12mm = d_1

No of bolts (i) = 4

Pitch circle dia of the bolts (D_1) = 120mm

Nominal gap between coupling holes = c = b = 4mm

Check the pin for both shear and bending

WKT $M_t = i \times F \times \frac{D_1}{2}$ $\Rightarrow$ $\frac{19.33a}{19.5}$

$$9.55 \times 10^5 = 4 \times F \times \frac{120}{2}$$

$$F = 3.979 \times 10^3 N/mm^2$$

WKT Shear in the pin

$$\tau_p = \frac{F}{0.785(d_p)^2}$$

$$= \frac{3.979 \times 10^3}{0.785(12)^2}$$

$$\tau_p = 35.19 N/mm^2$$

Given for pin $\tau_{yd} = 55MPa$ (Shaft and pin are same material)

$$\therefore \tau_p < \tau_{yd}$$

35.19MPa $<$ 55MPa

Hence design is safe

WKT The bending stress in the pin

$$\sigma_b = \frac{F \times [\frac{l}{2} + b]}{\frac{\pi}{32} \times d_p^3}$$ $\Rightarrow$ $\frac{19.35}{19.5}$

$$= \frac{3.979 \times 10^3 \times [\frac{45}{2} + 4]}{\frac{\pi}{32} \times (12)^3}$$

$$\sigma_b = 621.55MPa$$

Where σ_b = developed bending stress

But WKT For the pin , Given

$\tau_{yd} = 55MPa$

$\therefore \sigma_{yd} = 2(\sigma_{yd})$

$\sigma_{yd} = 2 \times 55 = 110MPa$

$\therefore \sigma_b > \sigma_{by} \left(\sigma_{yd}\right)$

$621.55\text{MPa} > 110\text{MPa}$

Hence design is not safe

Taking dia of the pin

$d_p = d_1 = 22mm$

(By solving trial and error method considering $d_1 = 22mm$, $\sigma_b =$ or $< \sigma_{by}$)

$\therefore d_p = d_1 = 22mm$ is taken

$$\sigma_b = \frac{F \times [\frac{l}{2} + b]}{\frac{\pi}{32} \times d_p^3}$$

$$= \frac{3.979 \times 10^3 \times [\frac{45}{2} + 4]}{\frac{\pi}{32} \times (22)^3}$$

$\sigma_b = 99.37MPa$

$\sigma_b < \sigma_{by}$

Hence design is safe.

OLD HAM'S COUPLING

Refer fig $\frac{19.5}{19.15}$

Refer from equation 19.36 to equation 19.44

It is based on the principle of double slider crank mechanism which is used to connect 2 shafts which are parallel but not collinear. It consists of 2 flanges with a diametric rectangle groove.

PROBLEM:

1] Design an Old ham's coupling from the following data

N = 5KW, n = 200rpm, distance between axis of shafts a = 5mm, allowable shear stress in the shaft = 40MPa, allowable bearing pressure between the slot and the tongue of the central disk =8MPa.

Sol:

Given N=5KW

n=200rpm

a=5mm

shaft τ_{yd} =40MPa

P=8MPa

$M_t = \frac{9550N}{n} = \frac{9550 \times 5 \times 10^3}{200} = 2.387 \times 10^5 N-m$

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

[take n=0.75]

$2.387 \times 10^5 = \frac{\pi d^3}{16} \times 0.75 \times 40$

d=34.34mm

d_{std}=35mm $\Rightarrow$ T- $\frac{16.10}{16.15}$

from T- $\frac{17.4}{17.13}$ for d_{std}=35mm

b=10mm   h=8mm

WKT

Length of boss l =1.75d $\Rightarrow \frac{19.41}{19.6}$

= 1.75 × 35

l = 61.25mm

WKT

Diameter of the disc D =3d+a $\Rightarrow \frac{19.39}{19.6}$

= 3×35+5

D = 110mm

WKT

Torque transmitted by the coupling

$M_t = 2FL = \frac{PD^2h}{6}$ $\Rightarrow \frac{19.37}{19.6}$

$2.387 \times 10^5 = \frac{8 \times 110^2 \times h}{6}$

Breadth of the groove h = 14.79mm

WKT

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

$15 = \frac{w}{2}$

w = 30mm [w=width]

WKT

Diameter of the boss $D_2 = 2d$ $\Rightarrow$ $\frac{19.40}{19.6}$

$= 2 \times 35$

$= 70mm$

WKT

Thickness of the flange $t = \frac{3}{4}d$ $\Rightarrow$ $\frac{19.44}{19.6}$

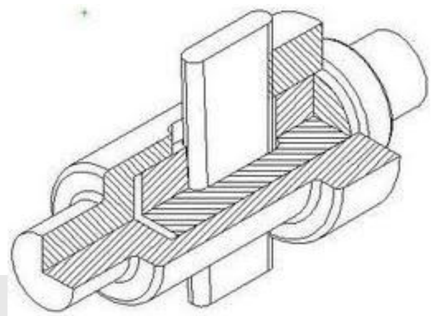
$= \frac{3}{4} \times 35$

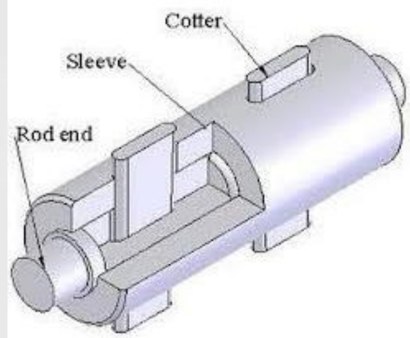
$= 26.25mm$

$T \approx 27mm$

Design of Joints

Design of Socket and Spigot Cotter joint





Refer fig. $\frac{17.10}{17.9}$

Equation to be used

From $\frac{17.62}{17.6}$ to $\frac{17.74}{17.6}$

From fig. F=Load

d=dia of the rod

d_{std} – T- $\frac{16.10}{16.15}$

d_1 = dia of spigot end or inside dia of the socket

d_2= dia of spigot collar

d_3= outside dia of socket

d_4= dia of socket collar

e=thickness of spigot collar

a=distance from the end of the slot to the rod end

b=mean width of the cotter

t=thickness of the cotter

c=thickness of the socket collar

h=thickness of the socket at the end of the rod

σ =permissible tensile stress

τ =permissible shear stress

σ_c =permissible crushing stress

Procedure to solve :

Step-1

$$\sigma = \frac{4F}{\pi d^2} \qquad \Rightarrow \frac{17.62}{17.6}$$

$$d_{std} - T. \frac{16.10}{16.15}$$

d=

Step-2

$$F = d_1 t \sigma_c \qquad \Rightarrow \frac{17.69}{17.6}$$

d_1 t=

Step-3

-

$$\sigma = \frac{4F}{(\pi d_1^2 - 4 d_1 t)} \qquad \Rightarrow \frac{17.63}{17.6}$$

$d_1 =$

Step-4

$$F = 2bt\,\tau \qquad \Rightarrow \frac{17.65}{17.6}$$

b=

Step-5

$$\sigma = \frac{4F}{\pi(d_2^2 - d_1^2)} \qquad \Rightarrow \frac{17.68}{17.6}$$

$d_2 =$

Step-6

$$\tau = \frac{F}{2ad_1} \qquad \Rightarrow \frac{17.66}{17.6}$$

a=

Step-7

$$\tau = \frac{F}{\pi d_1 e} \qquad \Rightarrow \frac{17.72}{17.6}$$

e=

Step-8

$$\sigma = \frac{4F}{\pi\left(d_3{}^2 - d_1^2\right) - 4t(d_3 - d_1)} \qquad \Rightarrow \frac{17.64}{17.6}$$

$d_3 =$

Step-9

$$\sigma_c = \frac{F}{(d_4 - d_1)t} \qquad \Rightarrow \frac{17.70}{17.6}$$

$$d_4 =$$

Step-10

$$\tau = \frac{F}{2c(d_4 - d_1)} \qquad \Rightarrow \frac{17.67}{17.6}$$

$$c =$$

Step11

$$\tau = \frac{F}{\pi d_1 h} \qquad \Rightarrow \frac{17.73}{17.6}$$

$$h =$$

Step12

$$\sigma_b = \frac{F(d_1 + 2d_4)}{4tb^2} \qquad \Rightarrow \frac{17.74}{17.6}$$

Problems:

1. Design a socket and spigot cotter joint to connect two rods subjected to a tensile load of 120 KN. The permissible stresses for joint may be taken as 100MPa in tension 60MPa in shear and 120MPa in crushing.

Sol :

Given : F=120 $\times 10^3$ N

σ = 100MPa

τ = 60 MPa

σ_c = 120Mpa

Step-1

$$\sigma = \frac{4F}{\pi d^2} \qquad \Rightarrow \frac{17.62}{17.6}$$

$$100 = \frac{4 \times 120 \times 10^3}{\pi \times d^2}$$
$$d = 39.08mm$$
$$d_{std} = 40mm$$

Step-2

$$F = d_1 t \sigma_c \qquad \Rightarrow \frac{17.69}{17.6}$$

$$120 \times 10^3 = d_1 \times t \times 120$$
$$d_1 \times t = 1000mm^2$$

Step-3

-

$$\sigma = \frac{4F}{(\pi d_1^2 - 4 d_1 t)} \qquad \Rightarrow \frac{17.63}{17.6}$$

$$100 = \frac{4 \times 120 \times 10^3}{\pi \times d_1^2 - 4(1000)}$$
$$d_1 = 52.92mm \approx 53mm$$
$$d_1 \times t = 1000mm^2$$
$$53mm \times t = 1000mm^2$$
$$t = 18.86mm \approx 19mm$$

Step-4

$$F = 2bt\tau \qquad \Rightarrow \frac{17.65}{17.6}$$

$$120\times10^3 = 2\times b\times19\times60$$
$$b = 52.63mm \approx 53mm$$

Step-5

$$\sigma = \frac{4F}{\pi(d_2^2 - d_1^2)} \qquad \Rightarrow \frac{17.68}{17.6}$$

$$120 = \frac{4\times120\times10^3}{\pi(d_2{}^2 - 53^2)}$$
$$d_2 = 63.89mm \approx 64mm$$

Step-6

$$\tau = \frac{F}{2ad_1} \qquad \Rightarrow \frac{17.66}{17.6}$$

$$60 = \frac{120\times10^3}{2\times a\times53}$$
$$a = 18.86mm \approx 19mm$$

Step-7

$$\tau = \frac{F}{\pi d_1 e} \qquad \Rightarrow \frac{17.72}{17.6}$$

$$60 = \frac{120\times10^3}{\pi\times53\times e}$$
$$e = 12.01mm \approx 12mm$$

Step-8

$$\sigma = \frac{4F}{\pi\left(d_3{}^2 - d_1^2\right) - 4t(d_3 - d_1)} \qquad \Rightarrow \frac{17.64}{17.6}$$

$$100 = \frac{4 \times 120 \times 10^3}{\pi\left(d_3{}^2 - 53^2\right) - 4 \times 19 \times (d_3 - 53)}$$

$$d_3 = 68.67mm \approx 69mm$$

Step-9

$$\sigma_c = \frac{F}{(d_4 - d_1)t} \qquad \Rightarrow \frac{17.70}{17.6}$$

$$120 = \frac{120 \times 10^3}{(d_4 - 53)19}$$

$$d_4 = 105.63mm \approx 106mm$$

Step-10

$$\tau = \frac{F}{2c(d_4 - d_1)} \qquad \Rightarrow \frac{17.67}{17.6}$$

$$60 = \frac{120 \times 10^3}{2c(106 - 53)}$$

$$c = 18.86mm \approx 19mm$$

Step11

$$\tau = \frac{F}{\pi d_1 h} \qquad \Rightarrow \frac{17.73}{17.6}$$

$$60 = \frac{120 \times 10^3}{\pi \times 53 \times h}$$

$$h = 12mm$$

Step12

$$\sigma_b = \frac{F(d_1 + 2d_4)}{4tb^2} \qquad \Rightarrow \frac{17.74}{17.6}$$

$$\sigma_b = \frac{120 \times 10^3 (53 + 2(106))}{4 \times 19 \times 53^2}$$
$$\sigma_b = 148.95 MPa$$

2. Design a socket and a spigot cotter joint to connect 2 rods subjected to steady axial pull of 100Kn.the material used for spigot end, socket end and the cotter is C-40 steel having tensile yield strength of 328.6MPa ,take FOS as 4 for tension, 6 for shear and 3 for crushing based on tensile yield strength.

Sol:

Given F=100kN

$$\sigma_y = 328.6 MPa$$

FOS=4 in tension

FOS=6 in shear

FOS=3 in crushing

$$\sigma = \frac{\sigma_y}{(FOS)_{tension}} = \frac{328.6}{4} = 82.15 MPa$$

$$\tau = \frac{\sigma_y}{(FOS)_{shear}} = \frac{328.6}{6} = 54.76 MPa$$

$$\sigma_c = \frac{\sigma_y}{(FOS)_{crushing}} = \frac{328.6}{3} = 109.533 MPa$$

Step-1

$$\sigma = \frac{4F}{\pi d^2} \qquad\qquad \Rightarrow \frac{17.62}{17.6}$$

$$82.15 = \frac{4 \times 120 \times 10^3}{\pi \times d^2}$$

$$d = 39.36mm$$

Step-2

$$F = d_1 t \sigma_c \qquad\qquad \Rightarrow \frac{17.69}{17.6}$$

$$100 \times 10^3 = d_1 \times t \times 109.533$$

$$d_1 \times t = 912.966mm^2$$

Step-3

-

$$\sigma = \frac{4F}{(\pi d_1^2 - 4d_1 t)} \qquad\qquad \Rightarrow \frac{17.63}{17.6}$$

$$82.15 = \frac{4 \times 100 \times 10^3}{\pi \times d_1^2 - 4(912.966)}$$

$$d_1 = 52.07mm \approx 52mm$$

$$d_1 \times t = 912.966mm^2$$

$$52mm \times t = 912.966mm^2$$

$$t = 17.55mm \approx 18mm$$

Step-4

$$F = 2bt\tau \qquad\qquad \Rightarrow \frac{17.65}{17.6}$$

$$100 \times 10^3 = 2 \times b \times 18 \times 54.76$$

$$b = 50.72mm \approx 51mm$$

Step-5

$$\sigma = \frac{4F}{\pi(d_2^2 - d_1^2)} \qquad\qquad \Rightarrow \frac{17.68}{17.6}$$

$$82.15 = \frac{4 \times 100 \times 10^3}{\pi(d_2{}^2 - 51^2)}$$

$$d_2 = 64.42mm \approx 65mm$$

Step-6

$$\tau = \frac{F}{2ad_1} \qquad \Rightarrow \frac{17.66}{17.6}$$

$$54.76 = \frac{100 \times 10^3}{2 \times a \times 52}$$

$$a = 17.55mm \approx 18mm$$

Step-7

$$\tau = \frac{F}{\pi d_1 e} \qquad \Rightarrow \frac{17.72}{17.6}$$

$$54.76 = \frac{100 \times 10^3}{\pi \times 52 \times e}$$

$$e = 11.17mm$$

Step-8

$$\sigma = \frac{4F}{\pi\left(d_3{}^2 - d_1^2\right) - 4t(d_3 - d_1)} \qquad \Rightarrow \frac{17.64}{17.6}$$

$$82.15 = \frac{4 \times 100 \times 10^3}{\pi(d_3^2 - 52^2) - 4(18)(d_3 - 52)}$$

$$d_3 = 67.96mm \approx 68mm$$

Step-9

$$\sigma_c = \frac{F}{(d_4 - d_1)t} \qquad \Rightarrow \frac{17.70}{17.6}$$

$$82.15 = \frac{100 \times 10^3}{(d_4 - 52)18}$$

$$d_4 = 67.96mm \approx 68mm$$

Step-10

$$\tau = \frac{F}{2c(d_4 - d_1)} \qquad \Rightarrow \frac{17.67}{17.6}$$

$$54.76 = \frac{100 \times 10^3}{2c(120 - 51)}$$

$$c = 13.23mm$$

Step11

$$\tau = \frac{F}{\pi d_1 h} \qquad \Rightarrow \frac{17.73}{17.6}$$

$$54.76 = \frac{100 \times 10^3}{\pi \times 52 \times h}$$

$$h = 11.178mm$$

Step12

$$\sigma_b = \frac{F(d_1 + 2d_4)}{4tb^2} \qquad \Rightarrow \frac{17.74}{17.6}$$

$$\sigma_b = \frac{100 \times 10^3 (52 + 2(120))}{4 \times 18 \times 50^2}$$

$$\sigma_b = 162.22MPa$$

3. Design a cotter joint to resist a load of 50kN which acts along the axis of the rods connected by a cotter. The material of the cotter and rod is same and has ultimate tensile strength of 400MPa, the shear strength may be taken as half the tensile strength, the compressive strength maybe assumed as 1.5 times the tensile strength. Take FOS=4.

Given

F=50kN

$\sigma_{ut} = 400MPa$

FOS=4

$\tau = \frac{1}{2}\sigma$ $\qquad$ $\sigma_c = 1.5\sigma$

$$\sigma = \frac{\sigma_{ut}}{FOS} = \frac{400}{4} = 100MPa$$

$\tau = 0.5(100)$ σ_c =1.5(100)

=50MPa =150MPa

SAME AS ABOVE PROCEDURE

DESIGN OF KNUCKLE JOINTS

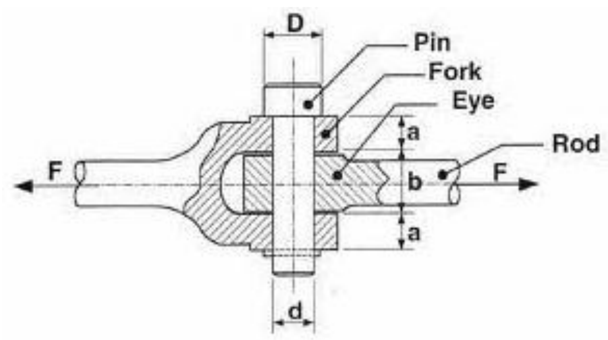


Fig $\frac{17.9}{17.9}$

$$\frac{17.46}{17.5} to \frac{17.57}{17.5}$$

From Figure:

F = Load

d = Dia of rod

$d_{std} = T_{6.10/16.15}$

d_2 = Dia of the pin

d_3 = Dia of the pin head

d_4 = Outside dia of eye

a = thickness of fork

b = thickness of single eye

h = thickness of the pin head

$\sigma = \sigma_t$ = permissible tensile stress

σ_c = permissible crushing stress

τ = permissible shear stress

PROCEDURE :

1. $\sigma = \sigma_t = \dfrac{4F}{\pi d^2}$ $\Rightarrow$ $\dfrac{17.46}{17.5}$

   $d_{std} \rightarrow T_{16.10}$

2. $\tau = \dfrac{2F}{\pi d_2^2}$ $\Rightarrow$ $\dfrac{17.53}{17.5}$

   d_2 we get

3. $\sigma_c = \dfrac{F}{d_2 b}$ $\Rightarrow$ $\dfrac{17.51}{17.5}$

   b we get

4. $\tau = \dfrac{F}{b(d_4 - d_2)}$ $\Rightarrow$ $\dfrac{17.48}{17.5}$

   d_4 we get

5. $\tau = \dfrac{F}{2a(d_4 - d_2)}$ $\Rightarrow$ $\dfrac{17.50}{17.5}$

6. d_3 = 1.5d (It is not available in data handbook)

7. h = 0.5d

8. Check the stress

   a. To check tensile stress

$$\sigma_t = \frac{F}{(d_4 - d_2)b} \qquad \Rightarrow \qquad \frac{17.47}{17.5}$$

If $\sigma_t < \sigma_{ty}$

$\therefore$ If σ_t is less than σ_{ty} design is safe

Where σ_t = developed tensile stress

σ_{ty} = allowable tensile stress (given)

   b. To check crushing stress

$$\sigma_c = \frac{F}{2d_2 a} \qquad \Rightarrow \qquad \frac{17.52}{17.5}$$

If $\sigma_c < \sigma_{cy}$ then design is safe

Where σ_c = developed crushing stress

σ_{cy} = allowable rushing stress (given)

   c. To check for shear stress

$$\tau = \frac{F}{2a(d_4 - d_2)} \qquad \Rightarrow \qquad \frac{17.50}{17.5}$$

If $\tau < \tau_{fy}$ then design is safe

Where τ = developed shear

τ_{fy} = allowable shear (given)

Problems:

1. Design a knuckle joint for a tie rod of circular cross section to sustain a maximum pull of 70KN, the ultimate tensile strength of a rod is 450MPa, the ultimate crushing and shear strength of the pin material is 510MPa and 396MPa respectively, take FOS=6.

Sol.

F = 70KN, σ_{ut} = 450MPa, σ_{uc} = 510MPa, τ_{us} = 396MPa, FOS=6

$$\therefore \ \sigma = \sigma_t = \frac{\sigma_{ut}}{FOS} = \frac{450}{6} = 75MPa$$

$$\sigma_c = \frac{\sigma_{uc}}{FOS} = \frac{510}{6} = 85MPa$$

$$\tau_u = \frac{\tau_{us}}{FOS} = \frac{396}{6} = 66MPa$$

Step 1

$$\sigma = \sigma_t = \frac{4F}{\pi d^2} \qquad \Rightarrow \qquad \frac{17.46}{17.5}$$

$$75 = \frac{4 \times 70 \times 10^3}{\pi \times d^2}$$

d = 34.47mm

d_{std} = 35mm $\Rightarrow$ $\frac{16.10}{16.15}$

Step 2

$$\tau = \frac{2F}{\pi d_2^2} \qquad \Rightarrow \qquad \frac{17.53}{17.5}$$

$$66 = \frac{2 \times 70 \times 10^3}{\pi \times d_2^2}$$

d_2 = 25.98 $\approx$ 26mm

Step 3

$$\sigma_c = \frac{F}{d_2 b} \qquad \Rightarrow \qquad \frac{17.51}{17.5}$$

$$85 = \frac{70 \times 10^3}{26 \times b}$$

$$b = 31.67 \approx 32mm$$

Step 4

$$\tau = \frac{F}{b(d_4 - d_2)} \qquad \Rightarrow \qquad \frac{17.48}{17.5}$$

$$66 = \frac{70\times10^3}{32(d_4 - 26)}$$

$$d_4 = 59.14 \approx 59mm$$

Step 5

$$\tau = \frac{F}{2a(d_4 - d_2)} \qquad \Rightarrow \qquad \frac{17.50}{17.5}$$

$$66 = \frac{70\times10^3}{2\times a(59-26)}$$

$$a = 16mm$$

Step 6

d_3 = 1.5d
d_3 = 52.5mm

Step 7

h = 0.5d
h = 17.5mm

Step 8 To check for stresses

To check tensile stress

a.

$$\sigma_t = \frac{F}{(d_4 - d_2)b}$$

$$\sigma_t = \frac{70\times10^3}{(59-26)32}$$

$$\sigma_t = 66.28MPa$$

But given $\sigma_{ty} = 75MPa$

$\therefore \sigma_t < \sigma_{ty}$

The design is safe 66.28MPa $<$ 75MPa

Since the developed tensile stress is less than allowable tensile stress the design is safe

b. To check for crushing stress

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

$$= \frac{70 \times 10^3}{2 \times 26 \times 16}$$

$$\sigma_c = 84.13MPa$$

But given σ_{cy} = 85MPa

$\therefore \sigma_c < \sigma_{cy}$

i.e. 84.13MPa $<$ 85MPa

Since the developed crushing stress is less than the allowable crushing stress the design is safe

c. To check for shear stress

$$\tau = \frac{F}{2a(d_4 - d_2)}$$

$$= \frac{70 \times 10^3}{2 \times 16(59 - 26)}$$

$$\tau = 66.28MPa$$

But given $\sigma_{fy} = 66.28MPa$

$\sigma < \sigma_{fy}$

i.e. $66.28MPa \, ; \, 66MPa$

Since the developed shear stress is approximately equal to allowable shear the design is considered to be safe

Step 9 Copy fig 17.9

2. Design a knuckle joint to sustain an axial load of 100KN, the allowable stresses of the material of the joint area as follows, take σ = 100MPa, τ = 60MPa, σ_c = 120MPa.

Sol:

Same as the previous problem