UNIT-2

- Keys
- Splines
- Socket and spigot cotter joint
- Sleeve and cotter joint
- Gib and cotter joint
- Knuckle joint

Keys

- A key is a machine element that is used to connect the transmission shaft to rotating machine element like pulley, gear, sprocket, flywheel
- ➢ To prevent relative motion between them and
- ➢ To transmit the torque from the shaft to the hub.
- According to Indian standards, steel of tensile strength not less than 600 N/mm^2 shall be used as material for the key.

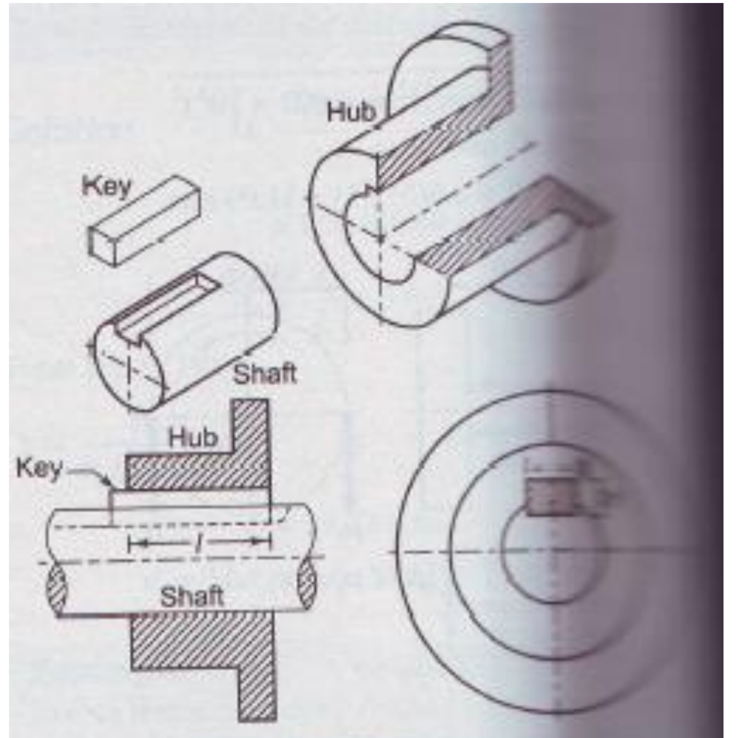


Fig. 9.16 *Key-joint*

Type of keys

- Saddle key and sunk key
- Square key and flat key
- Taper key and parallel key
- Key with and without Gib-head

Some special keys are:-

- ➢ Woodruff key
- ➢ Kennedy key
- ➢ Feather key

- Selection of key based on following factors:-

i. Power to be transmitted
ii. Tightness of fit
iii. Stability of connection
iv. cost

Saddle keys

- A Saddle key is a key that fits in the keyway of the hub only. (no keyway on the shaft)
- Cost is less.
- Low power transmission.
- Suitable for light duty works.
- Types:- (1) Hollow saddle key, (2) Flat saddle key.

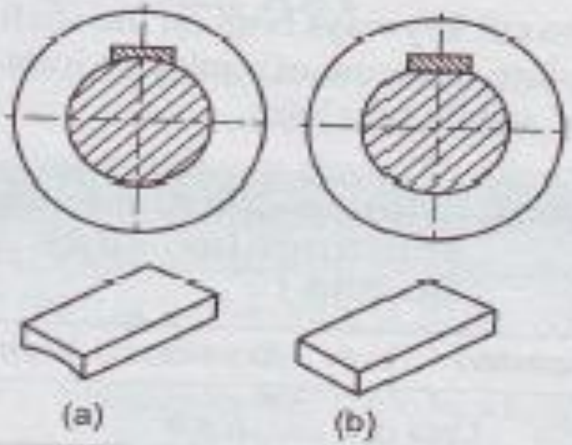


Fig. 9.17 *(a) Hollow Saddle Key*
(b) Flat Saddle Key

Sunk Keys

- A sunk key is a key, in which half the thickness of the key fits into the keyway of hub and the remaining half in the keyway on the hub.
- There is no possibility of the key to slip around shaft.
- Suitable for heavy duty works.

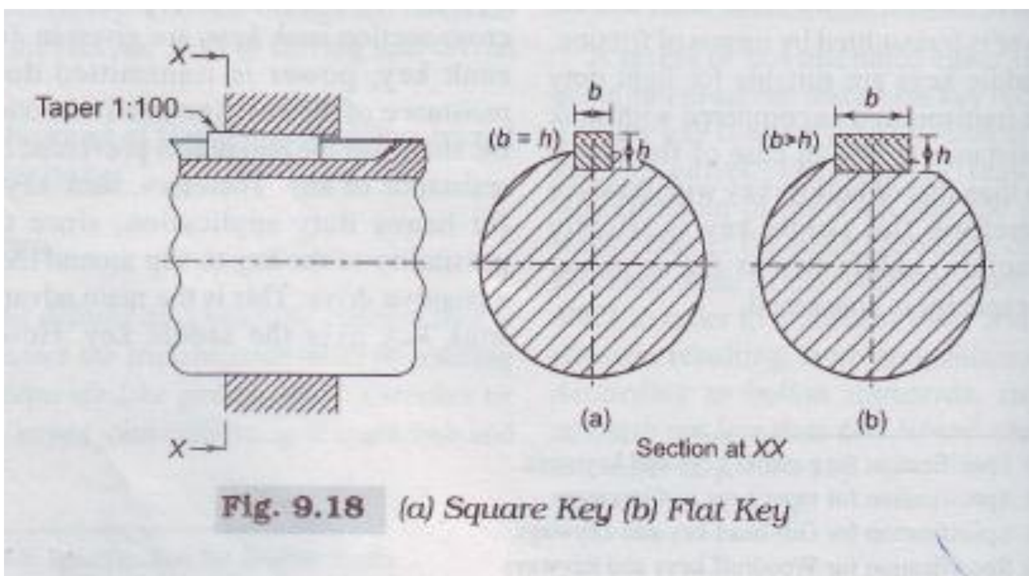
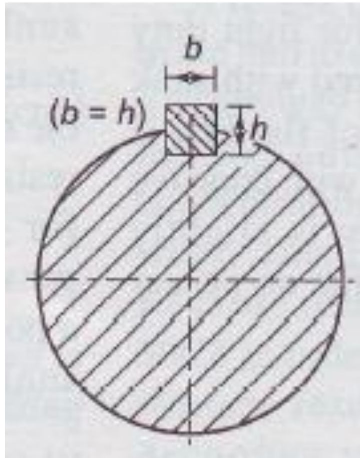


Fig. 9.18 *(a) Square Key (b) Flat Key*

- Square sunk key: sunk key having square cross section.



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

$$l = 1.5d$$

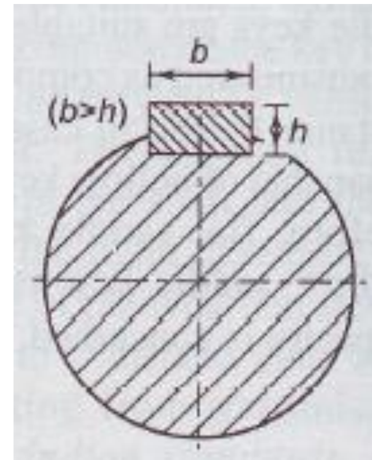
b = width of key (mm)

h = height or thickness of key

l = length of key (mm)

d = diameter of shaft (mm)

- Flat sunk key:- Sunk key with rectangular cross section is called flat key.



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

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

$$l = 1.5d$$

Parallel and Taper key

- Parallel key is a sunk key which is uniform in width as well as height throughout the length of the key.
- A taper key is uniform in width but tapered in height. The standard taper is 1 in 100.
- Tapered keys are often provided with Gib-head to facilitate removal.

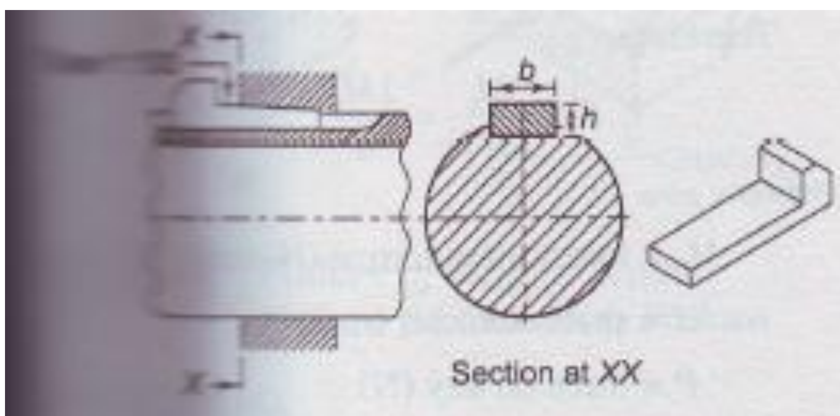


Fig. 9.19 *Gib-head Taper Key*

Feather key

- A feather key is a parallel that is fixed either to the shaft or to the hub and that permits relative axial movement between them.

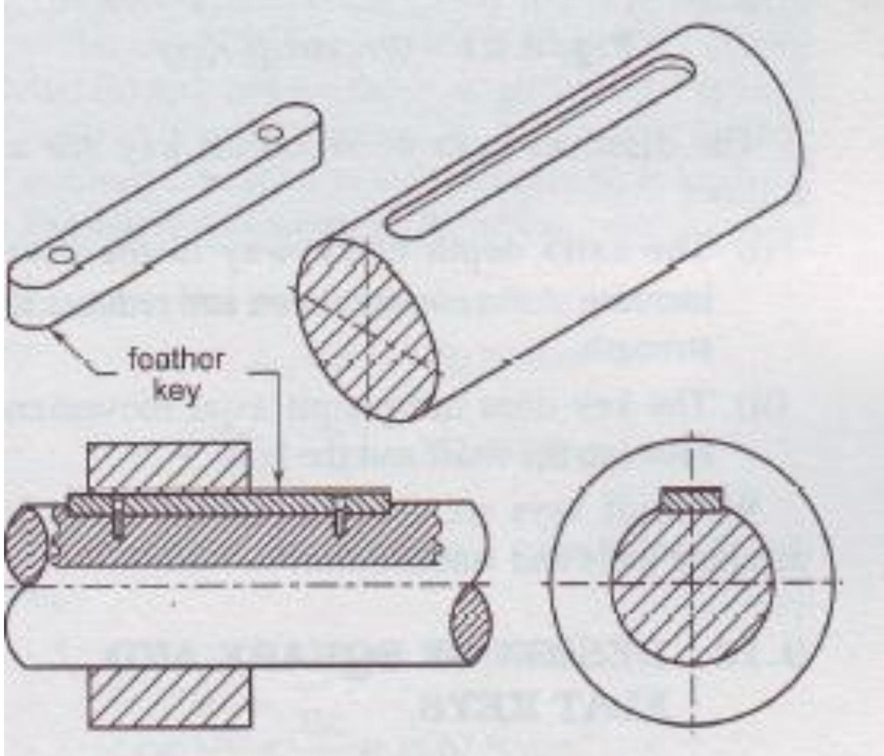


Fig. 9.20 *Feather Key*

Design of square and flat keys

- The transmission of torque from the shaft to hub results in two equal and opposite forces denoted by P-

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

Where M_t = transmitted torque (N-mm)

d = shaft diameter (mm)

P = force on key (N)

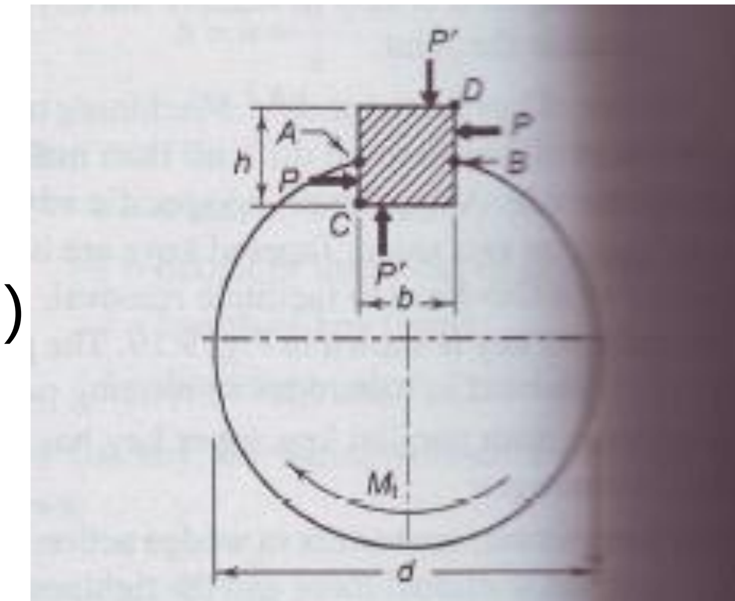


Fig. 9.22 *Forces Acting on Key*

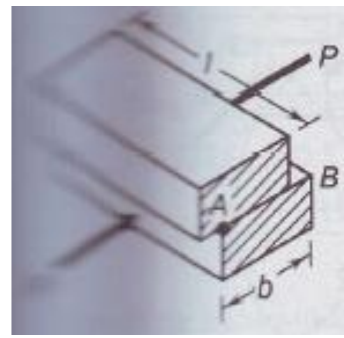
Design of keys based on two criteria-

1. Failure due to shear stress
2. Failure due to compressive stress

1. The shear stress will occur in plane AB

$$\tau = \frac{P}{\text{area of plane } AB} = \frac{P}{bl}$$

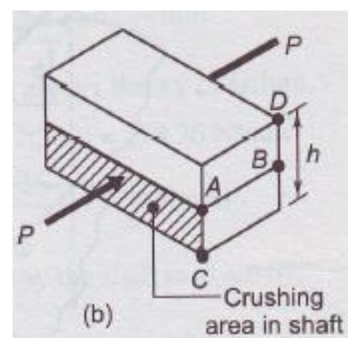
$$\tau = \frac{2M_t}{dbl}$$



2. The compressive stress will occur on surface AC or DB

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

$$\sigma_c = \frac{4M_t}{dhl}$$



Problem 1. It is required to design a square key for fixing a gear on a shaft of 25 mm diameter. The shaft is transmitting 15 kW power at 720 rpm to the gear. The key is made of steel 50C4 (S_{yt} = 460 N/mm^2) and the factor of safety is 3. For key material, the yield strength in compression can be equal to the yield strength in tension. Determine the dimensions of the key.

Problem 2. The cross section of a flat key for a 40 mm diameter shaft is 22 X 14 mm. The power transmitted by the shaft to the hub is 25 kW at 300 rpm. The key is made of steel ($S_{yc}=S_{yt}=300$ N/mm^2) and the factor of safety is 2.8. Determine the length of the key. Assume ($S_{Sy}=0.577\ S_{yt}$)

Design of Kennedy key

- The kennedy key consists of two square keys.

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

$$P = \frac{M_t}{d}$$

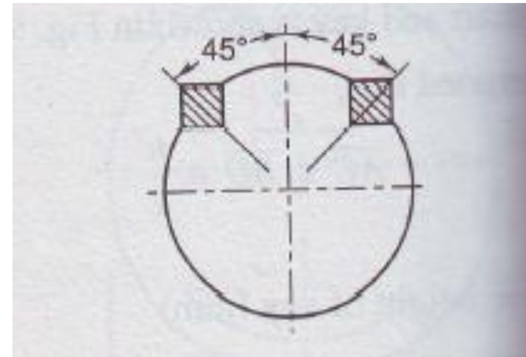


Fig. 9.24 *Kennedy Key*

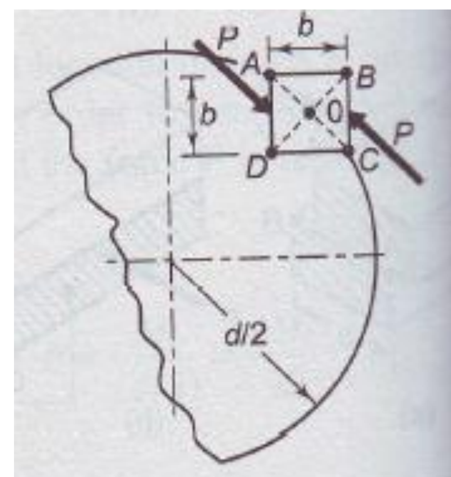
- Design is based on two criteria:-

1. Failure due to shear stress

$$\tau = \frac{M_t}{\sqrt{2}dbl}$$

2. Failure due to compressive stress

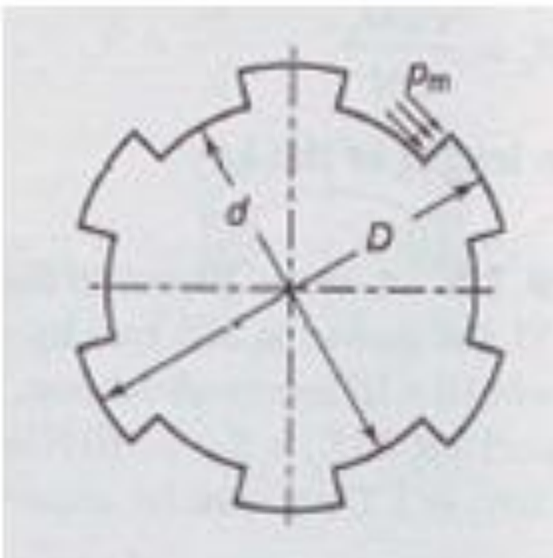
$$\sigma_c = \frac{\sqrt{2}M_t}{dbl}$$



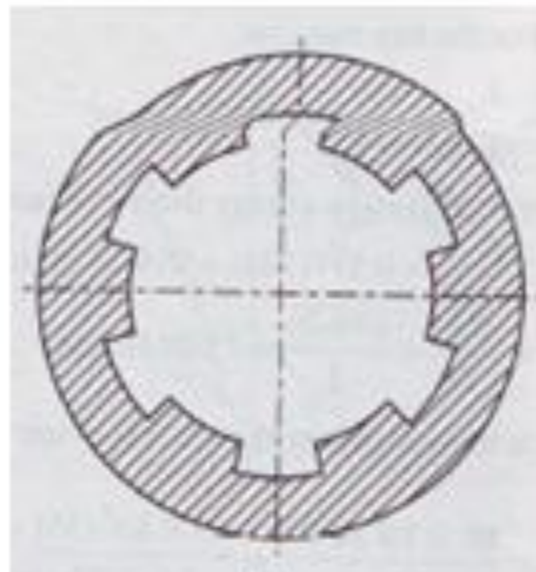
Problem 3. A shaft, 40 mm in diameter, is transmitting 35 kW power at 300 rpm by means of Kennedy keys of 10X10 mm cross section. The keys are made of steel 45C8 ($S_{yt}=S_{yc}=380$ N/mm^2) and the factor of safety is 3. Determine the required length of the keys.

Splines

- Sometimes, keys are made integral with the shaft which fits in the keyways broached in the hub. Such shafts are known as ***spline shaft*** as shown in Fig. These shafts usually have four, six, ten or sixteen splines.



(a)



(b)

- The torque transmitting capacity of splines is given by

$$M_t = P_m \times A \times R_m$$

Where M_t = torque transmitted (N-mm)

P_m = Permissible pressure ion spines (N/mm^2)

A = total area of splines (mm^2)

R_m = mean radius of splines (mm)

The area $A = \frac{1}{2}(D-d)\, \ell \times n$

Where ℓ = length of the hub

n = No. of splines

$$R_m = (D+d)/4$$

$$\boxed{M_t = 1/8\, P_m\, \ell\, n\, (D^2 - d^2)}$$

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

Problem 4. A standard splined connection 8 x 52 x 60 mm is used for the gear and the shaft assembly of a gear box. A 20 kW power at 300 rpm is transmitted by the splines.

The dimensions of the splines are

Major diameter = 60 mm

Minor diameter = 52 mm

No. of splines = 8

The normal pressure on the splines is limited to 6.5 N/mm^2. The coefficient of friction is 0.06.

Calculate (i) length of the hub of the gear &

(ii) The force required to shift the gear.

(2009, 2010)

Problem 5 A standard splined connection 8 x 36 x 40 mm is used for the gear and the shaft assembly rotating at 700 rpm. The dimensions of the splines are

Major diameter = 40 mm

Minor diameter = 36 mm

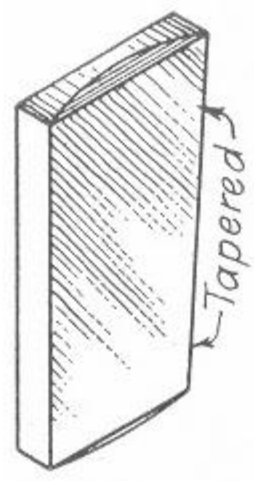
No. of splines = 8

The length of the gear hub is 50 mm and the normal pressure on the splines is limited to 6.5 N/mm^2 .

Calculate the power that can be transmitted from the gear to the shaft.

Cotter joint

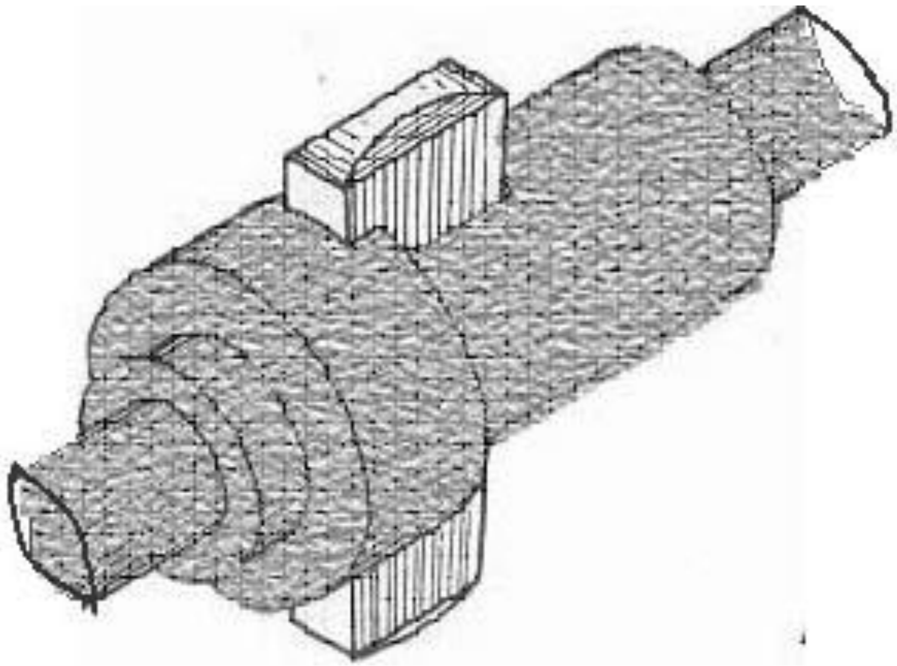
- A cotter joint is a temporary fastening and is used to connect rigidly two co-axial rods or bars which are subjected to axial tensile or compressive forces and not in rotation.
- A cotter is a flat wedge shaped steel piece of rectangular cross-section and its width is tapered (either on one side or both sides) from one end to another for an easy adjustment.



- It is usually used in connecting a piston rod to the crosshead of a reciprocating steam engine, piston rod to the tail or pump rod etc.

Types of cotter joints

1. Socket and spigot joint
2. Sleeve and cotter joint
3. Gib and cotter joint



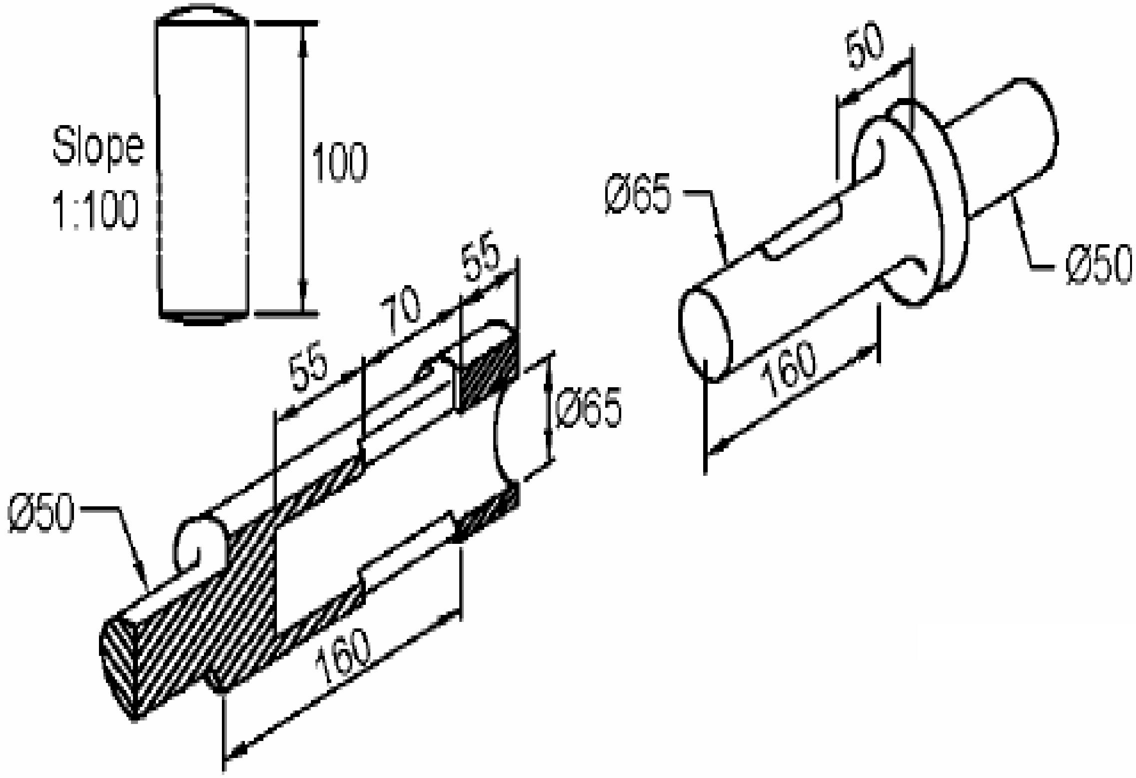


Figure 5.8 : Cotter, Socket and Spigot

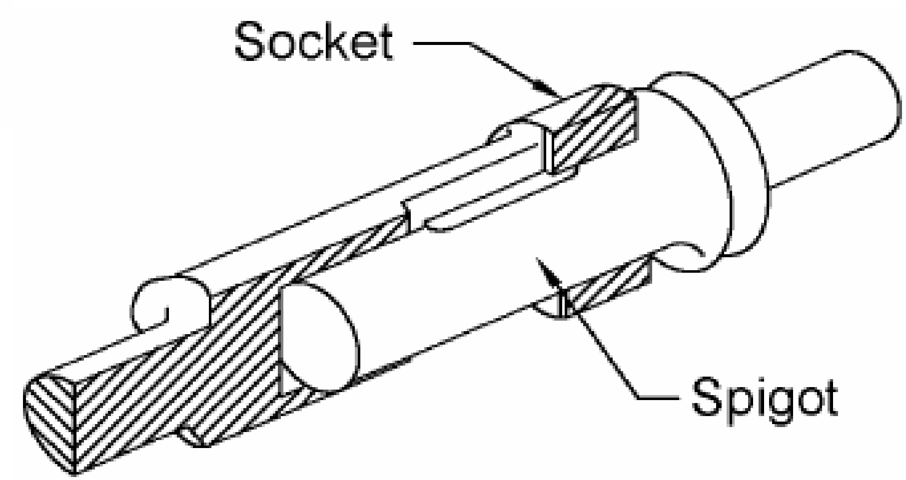
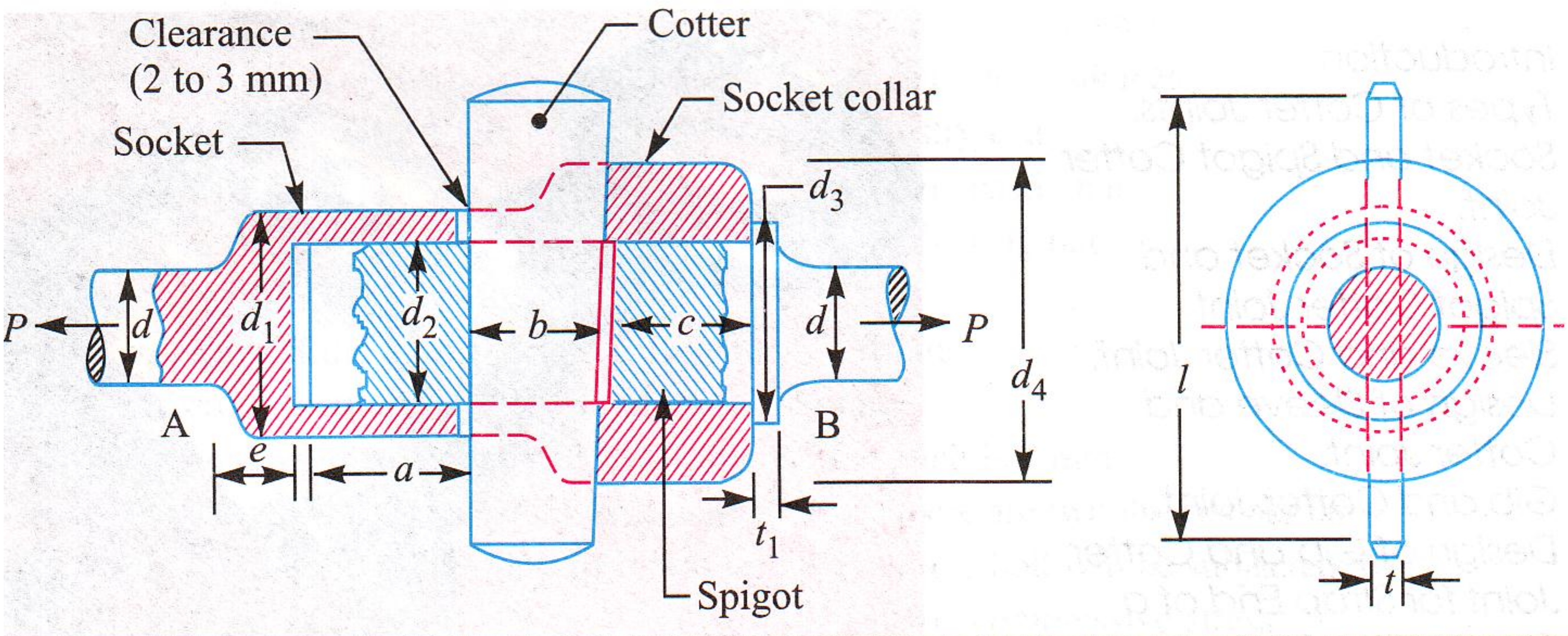


Figure 5.9 : Socket and Spigot Assembled

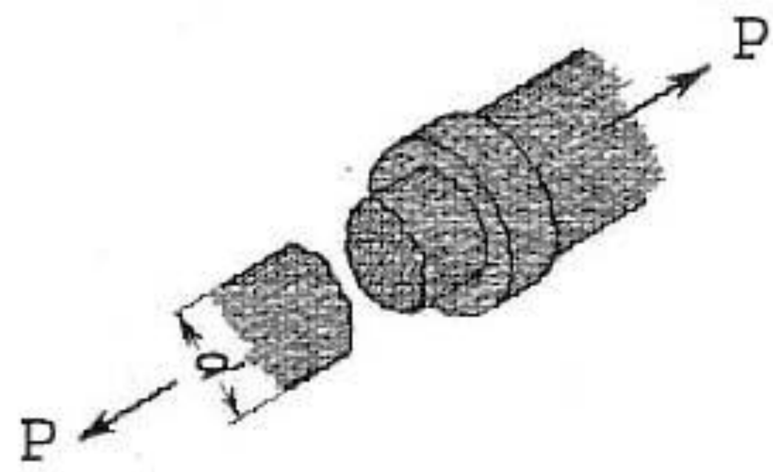
1.Socket and spigot cotter joint

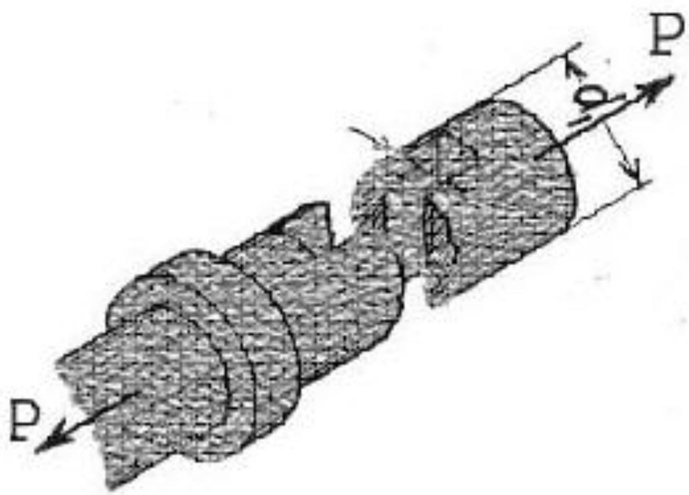


Design of socket and spigot joint

P= load carried by the rods
d= diameter of the rods
d_1= outside diameter of socket
d_2= diameter of spigot or inside diameter of socket
d_3= outside diameter of spigot collar
t_1= thickness of spigot collar
d_4= diameter of socket collar
c= thickness of socket collar
b= mean width of cotter
t= thickness of cotter
l= length of cotter
a= distance from the end of the slot to the end of spigot on rod B
σ_t= permissible tensile stress for the rods material
ζ= permissible shear stress for cotter material and
σ_c= permissible crushing stress for the cotter material

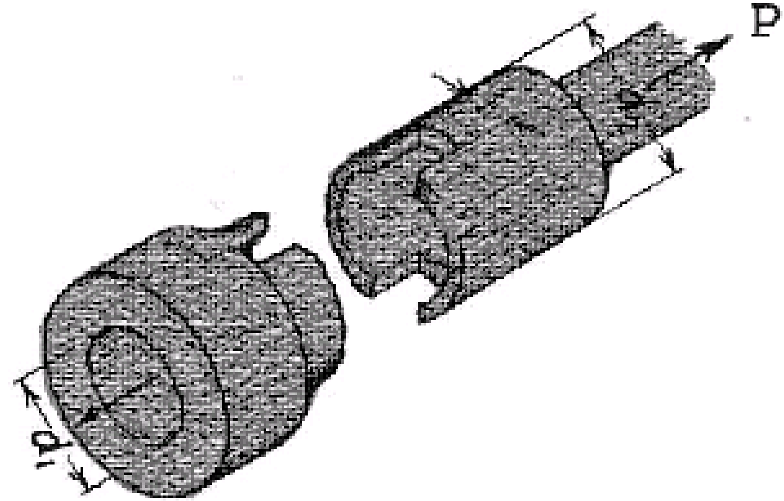
Tension failure of the rod

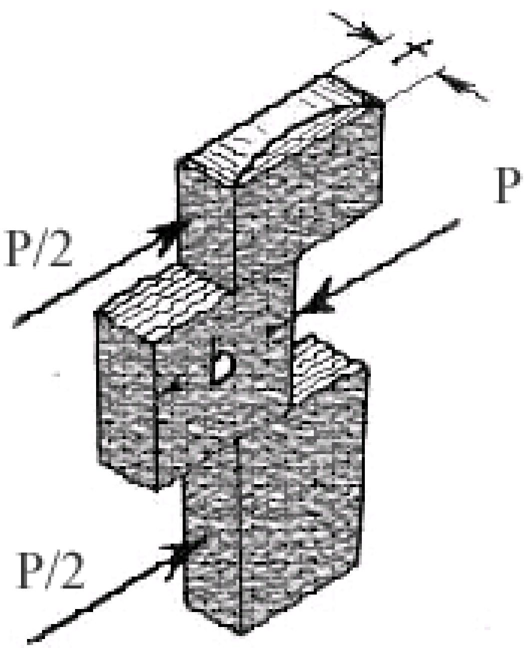




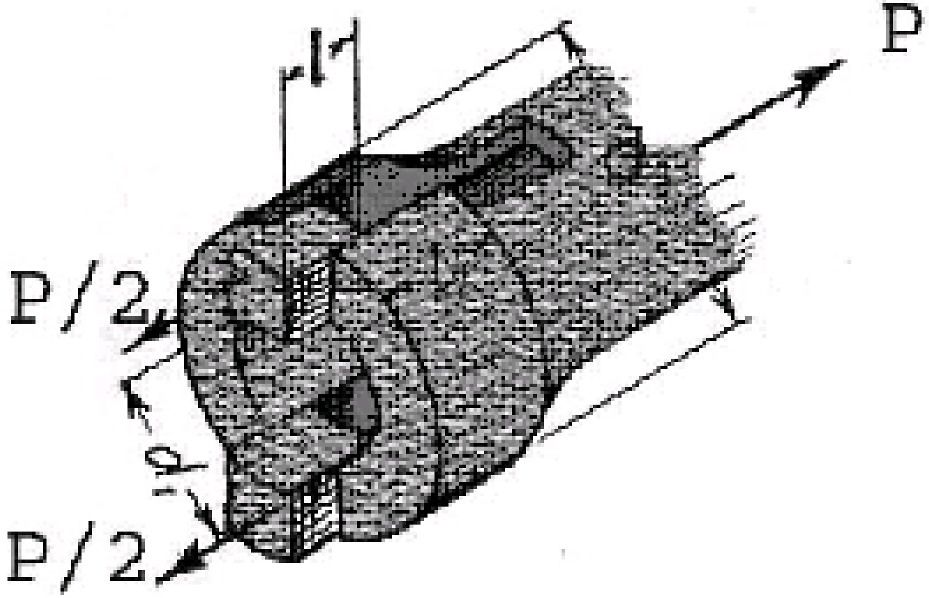
Tension failure of rod across slot

Tensile failure of socket across slot

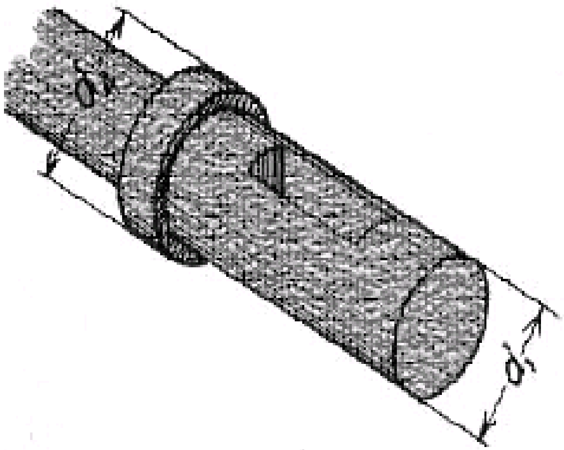




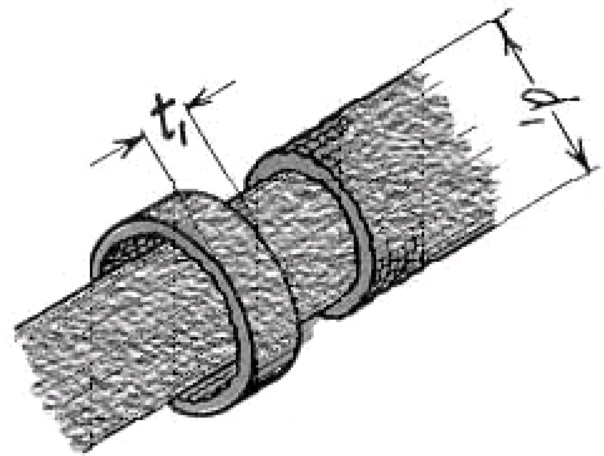
Shear failure of cotter



Shear failure of socket end



Crushing failure of collar



Shear failure of collar

1. Failure of rods in tension

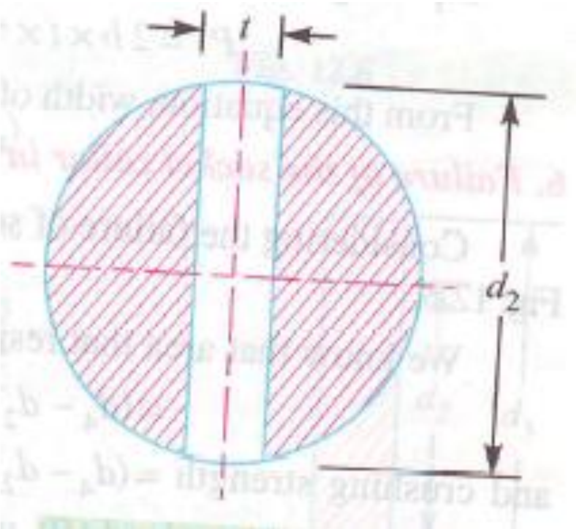
Resisting area = $\frac{\pi}{4} d^2$

Tearing strength of the rod = $\frac{\pi}{4} d^2 \times \sigma_t$

Equating to load (P) = $\frac{\pi}{4} d^2 \times \sigma_t$

From above equation diameter of rods (d) may be determined.

2. Failure of spigot in tension



Resisting area = $\frac{\pi}{4} d_2^{\,2} - d_2 \times t$

Tearing strength of the rod = $\left[\frac{\pi}{4} d_2^{\,2} - d_2 \times t\right] \times \sigma_t$

Equating to load (P) = $\left[\frac{\pi}{4} d_2^{\,2} - d_2 \times t\right] \times \sigma_t$

t may be taken as $d_2/4$.

From above equation diameter of spigot or inside diameter of socket (d_2) may be determined.

3. Failure of rod or cotter in crushing

Resisting area = $d_2 \times t$

Crushing strength = $d_2 \times t \times \sigma_c$

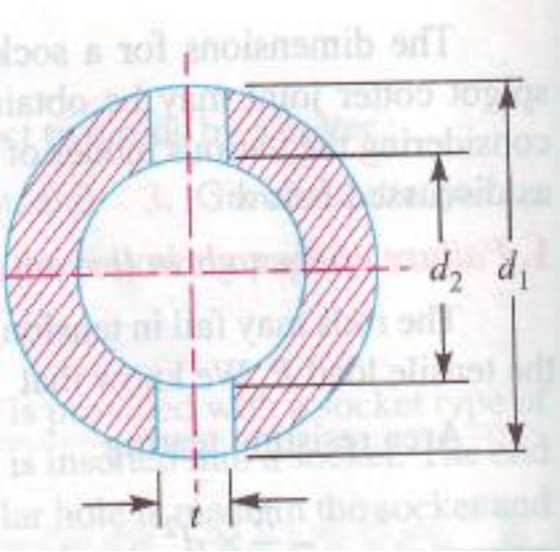
Equating this to load (P)

$$\boxed{P = d_2 \times t \times \sigma_c}$$

from this equation crushing strength may be checked.

4. Failure of socket in tension

Resisting area = $\frac{\pi}{4}\left(d_1^2 - d_2^2\right) - \left(d_1 - d_2\right)t$



Tearing strength of the rod = $\left[\frac{\pi}{4}\left(d_1^2 - d_2^2\right) - \left(d_1 - d_2\right)t\right] \times \sigma_t$

$$\text{Equating to load (P)} = \left[\frac{\pi}{4}\left(d_1^2 - d_2^2\right) - \left(d_1 - d_2\right)t\right] \times \sigma_t$$

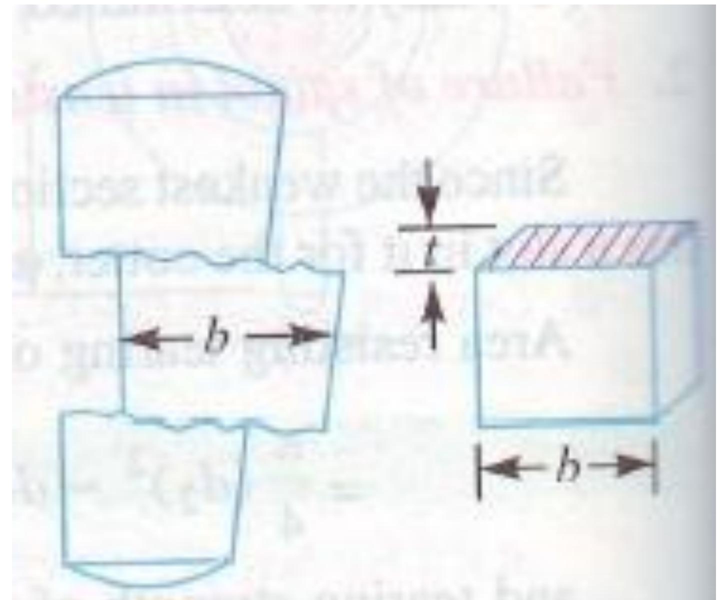
From this equation diameter of socket d_1 may be determined.

5. Failure of cotter in shear

Resisting area = 2 b x t

Shearing strength = 2 b x t x ζ

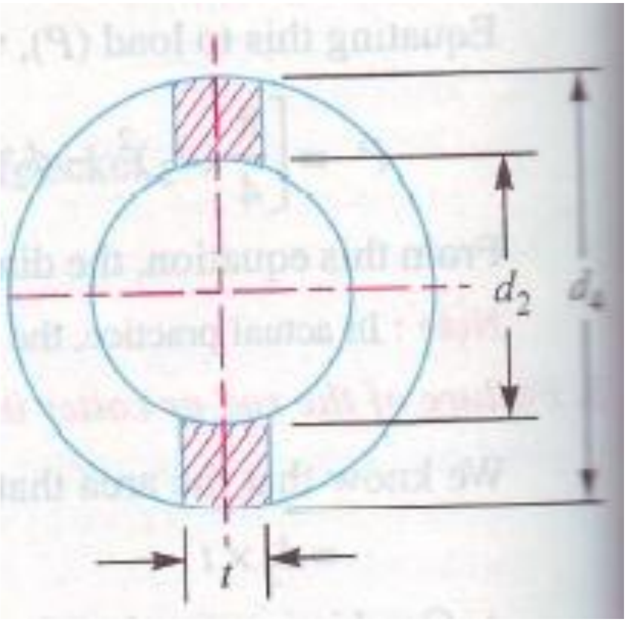
Equating to load P = 2 b x t x ζ



From this equation width of cotter (b) may be determined.

6. Failure of socket collar in crushing

Resisting area = $(d_4 - d_2) \times t$



Crushing strength = $(d_4 - d_2) \times t \times \sigma_c$

Equating to load $P = (d_4 - d_2) \times t \times \sigma_c$

From this equation diameter of socket Collar (d_4) may be determined.

7. Failure of socket end in shearing

Since the socket end is in double shear

Resisting area= $2(d_4 - d_2) \times c$

Shearing strength of the socket collar= $2(d_4 - d_2) \times c \times \zeta$

Load $P = 2(d_4 - d_2) \times c \times \zeta$

From this equation thickness of the socket collar (c) may be determined.

8. Failure of rod end in shear

Resisting area= 2 a x d_2

Shear strength of the rod end= 2 a x d_2 x ζ

Load P= 2 a x d_2 x ζ

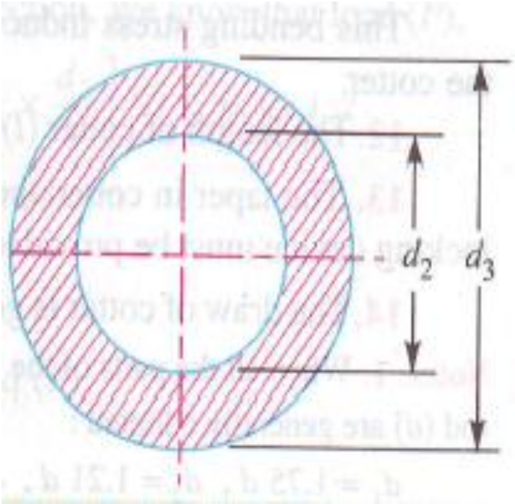
From this equation the distance from the end of the slot to the end of the rod (a) may be determined.

9. Failure of spigot collar in crushing

Resisting area= $\left[\frac{\pi}{4}\left(d_3{}^2 - d_2{}^2\right)\right]$

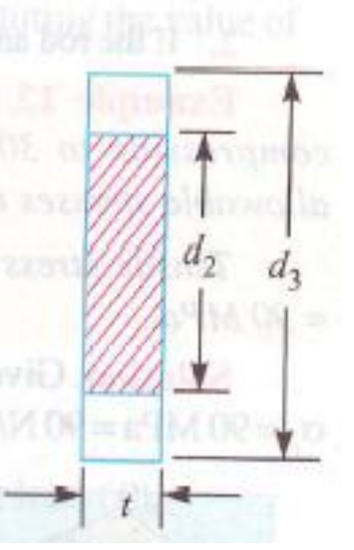
Crushing strength of the collar= $\left[\frac{\pi}{4}\left(d_3{}^2 - d_2{}^2\right)\right] \times \sigma_c$

Load P= $\left[\frac{\pi}{4}\left(d_3{}^2 - d_2{}^2\right)\right] \times \sigma_c$



From this equation diameter of the spigot collar (d_3) may be determined.

10. Failure of spigot collar in shearing



Resisting area= $\pi\, d_2 \times t_1$

Shearing strength of the collar= $\pi\, d_2 \times t_1 \times \zeta$

Load P= $\pi\, d_2 \times t_1 \times \zeta$

From this equation thickness of the spigot collar (t_1) may be determined.

11. Length of the cotter

$$\boxed{l = 4\,d}$$

12. The taper in cotter should not exceed 1 in 24.

13. Clearance of cotter is generally taken as 2 to 3 mm.

Problem 1. Design and draw a cotter joint to support a load from 30 kN in compression to 30 kN in tension. The material used is carbon steel for which the following allowable stress may be used. The load is applied statically.

Tensile stress= compressive stress= 50 MPa

shear stress= 35 MPa and

crushing stress= 90 MPa. **(2008)**

Problem 2. Design a socket and spigot cotter joint to connect two steel rods of equal diameter. Each rod is subjected to an axial tensile force of 50 kN. Material of two rods and cotter is steel 30 C8 (S_{yt} = 400N/mm^2 , S_{yc} = 2 S_{yt} and S_{sy} =0.5 S_{yt}) . The factor of safety for rods, spigot end and socket end is assumed as 6 while for cotter it is taken as 4.

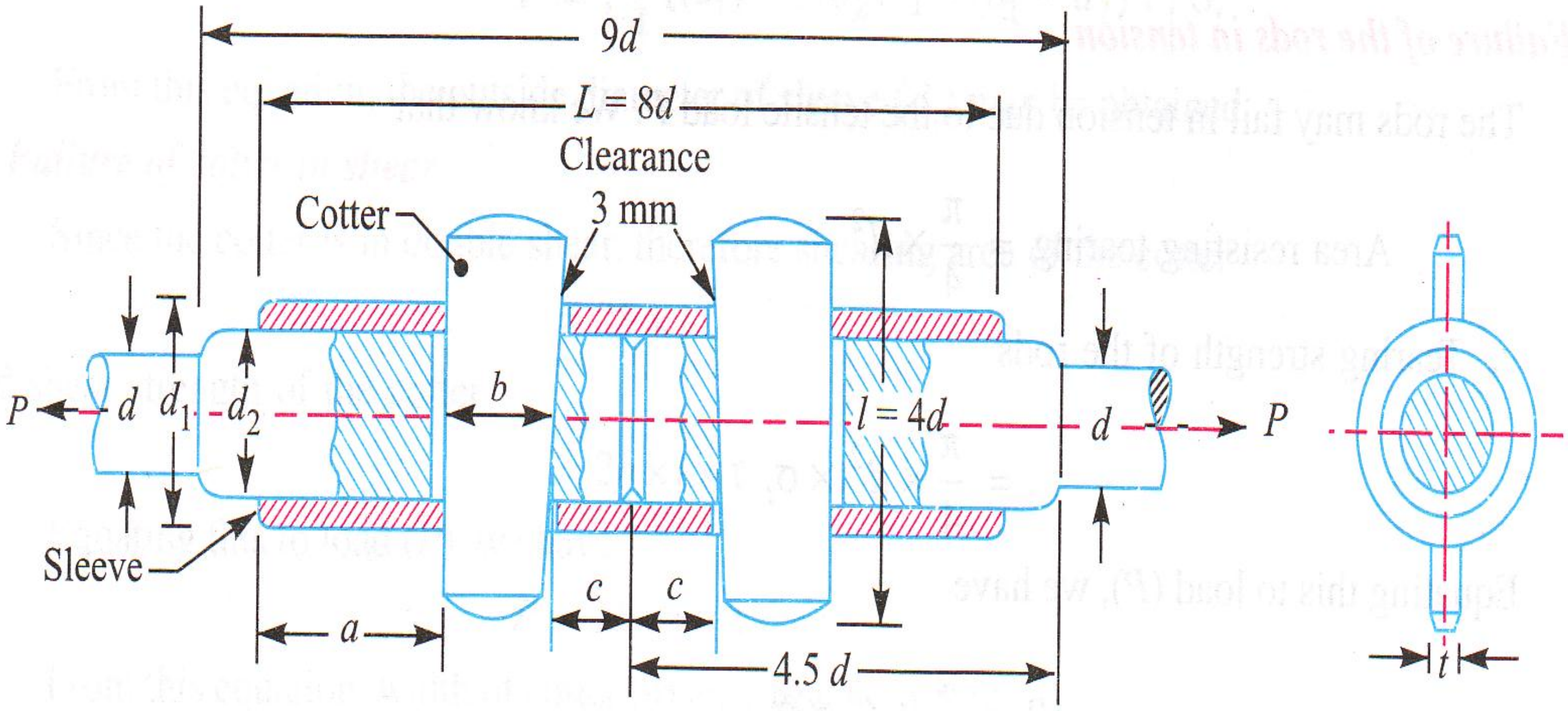
Draw sketch of joint showing designed dimensions. **(2011)**

Problem 3. Design and draw a cotter joint to support a load from 80 kN in compression to 80 kN in tension. The material used is carbon steel for which the following allowable stresses may be taken.

σ_t = 60 Mpa, $\sigma_{crushing}$ = 100 Mpa, ζ = 40 MPa

Go for socket and spigot cotter joint. **(2009)**

2. Sleeve and cotter joint



Design of sleeve and cotter joint

P = Load carried by the rods,

d = Diameter of the rods,

d_1 = Outside diameter of sleeve,

d_2 = Diameter of the enlarged end of rod,

t = Thickness of cotter,

l = Length of cotter,

b = Width of cotter,

a = Distance of the rod end from the beginning to the cotter hole (inside the sleeve end),

c = Distance of the rod end from its end to the cotter hole.

σ_t, τ and σ_c = Permissible tensile, shear and crushing stresses respectively for the material of the rods and cotter.

1. Failure of rods in tension

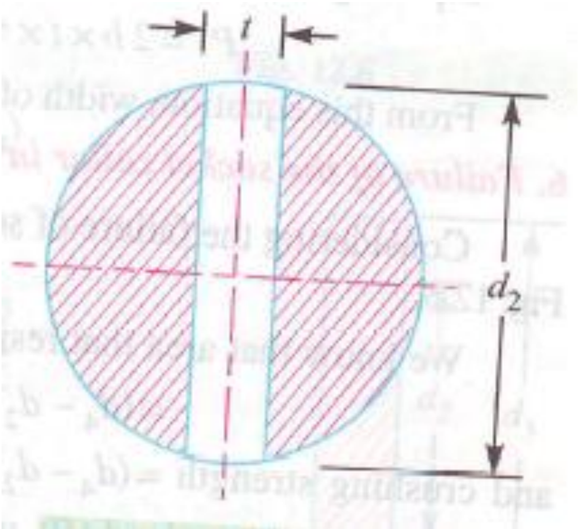
Resisting area = $\frac{\pi}{4} d^2$

Tearing strength of the rod = $\frac{\pi}{4} d^2 \times \sigma_t$

$$\boxed{\text{Equating to load (P)} = \frac{\pi}{4} d^2 \times \sigma_t}$$

From above equation diameter of rods (d) may be determined.

2. Failure of rod in tension across the weakest section (i.e. slot)



Resisting area = $\frac{\pi}{4}d_2^{\ 2} - d_2 \times t$

Tearing strength of the rod = $\left[\frac{\pi}{4}d_2^{\ 2} - d_2 \times t\right] \times \sigma_t$

Equating to load (P) = $\left[\frac{\pi}{4}d_2^{\ 2} - d_2 \times t\right] \times \sigma_t$

t may be taken as $d_2/4$.

From above equation diameter of the enlarged end of the rod (d_2) may be determined.

3. Failure of cotter in crushing

Resisting area = $d_2 \times t$

Crushing strength = $d_2 \times t \times \sigma_c$

Equating this to load (P)

$$\boxed{P = d_2 \times t \times \sigma_c}$$

from this equation crushing strength may be checked.

4. Failure of sleeve in tension across the slot

Resisting area = $\frac{\pi}{4}\left(d_1^{\,2}-d_2^{\,2}\right)-\left(d_1-d_2\right)t$

Tearing strength of the rod = $\left[\frac{\pi}{4}\left(d_1^{\,2}-d_2^{\,2}\right)-\left(d_1-d_2\right)t\right]\times\sigma_t$

Equating to load (P) = $\boxed{\left[\frac{\pi}{4}\left(d_1^{\,2}-d_2^{\,2}\right)-\left(d_1-d_2\right)t\right]\times\sigma_t}$

From this equation outside diameter of sleeve d_1 may be determined.

5. Failure of cotter in shear

Resisting area = 2 b x t

Shearing strength = 2 b x t x ζ

$$\boxed{\text{Equating to load } P = 2\, b \times t \times \zeta}$$

From this equation width of cotter (b) may be determined.

6. Failure of rod end in shear

Resisting area= 2 a x d_2

Shear strength of the rod end= 2 a x d_2 x ζ

$$\boxed{\text{Load } P = 2\, a \times d_2 \times \zeta}$$

From this equation the distance (a) may be determined.

7. Failure of sleeve end in shear

Since the sleeve end is in double shear

Resisting area= $2(d_1-d_2) \times c$

Shearing strength of the socket collar= $2(d_1-d_2) \times c \times \zeta$

$$\boxed{\text{Equating to Load } P= 2(d_1-d_2) \times c \times \zeta}$$

From this equation distance (c) may be determined.

Problem 4. Design a sleeve and cotter joint to resist a tensile load of 60 kN. All parts of the joint are made of the same material with the following allowable stresses:

Tensile stress = 60 MPa

Shear stress = 70 MPa

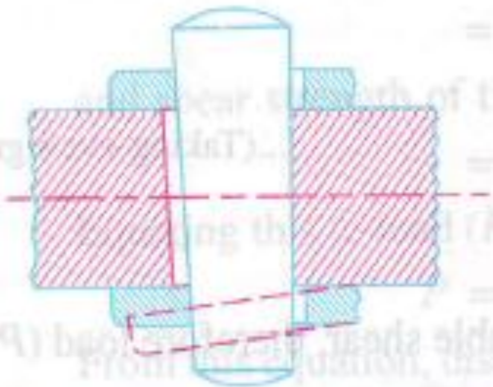
Crushing stress = 125 MPa

Problem 5. Two steel rods are to be connected by means of a steel sleeve, and two steel cotters. The rods are subjected to a tensile load of 40 kN. Design the joint by drawing a free hand sketch of the joint.

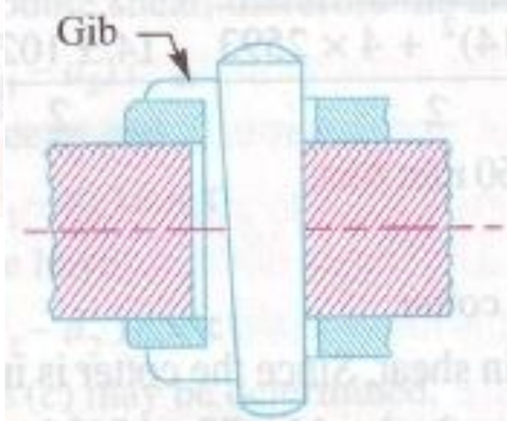
The permissible stress in tension σ_t = 60 MPa, in shear ζ = 50 MPa and in crushing σ_c = 90 Mpa may be assumed.

(2010)

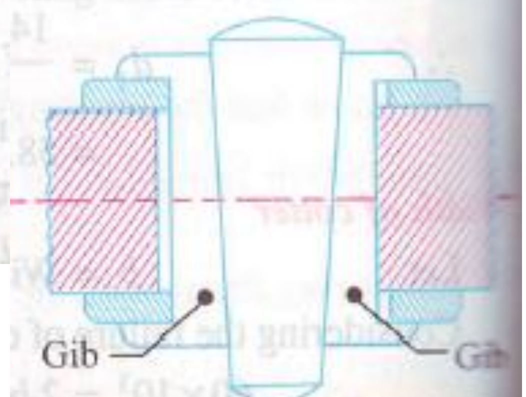
3. Gib and cotter joint



(a) Cotter without gib.

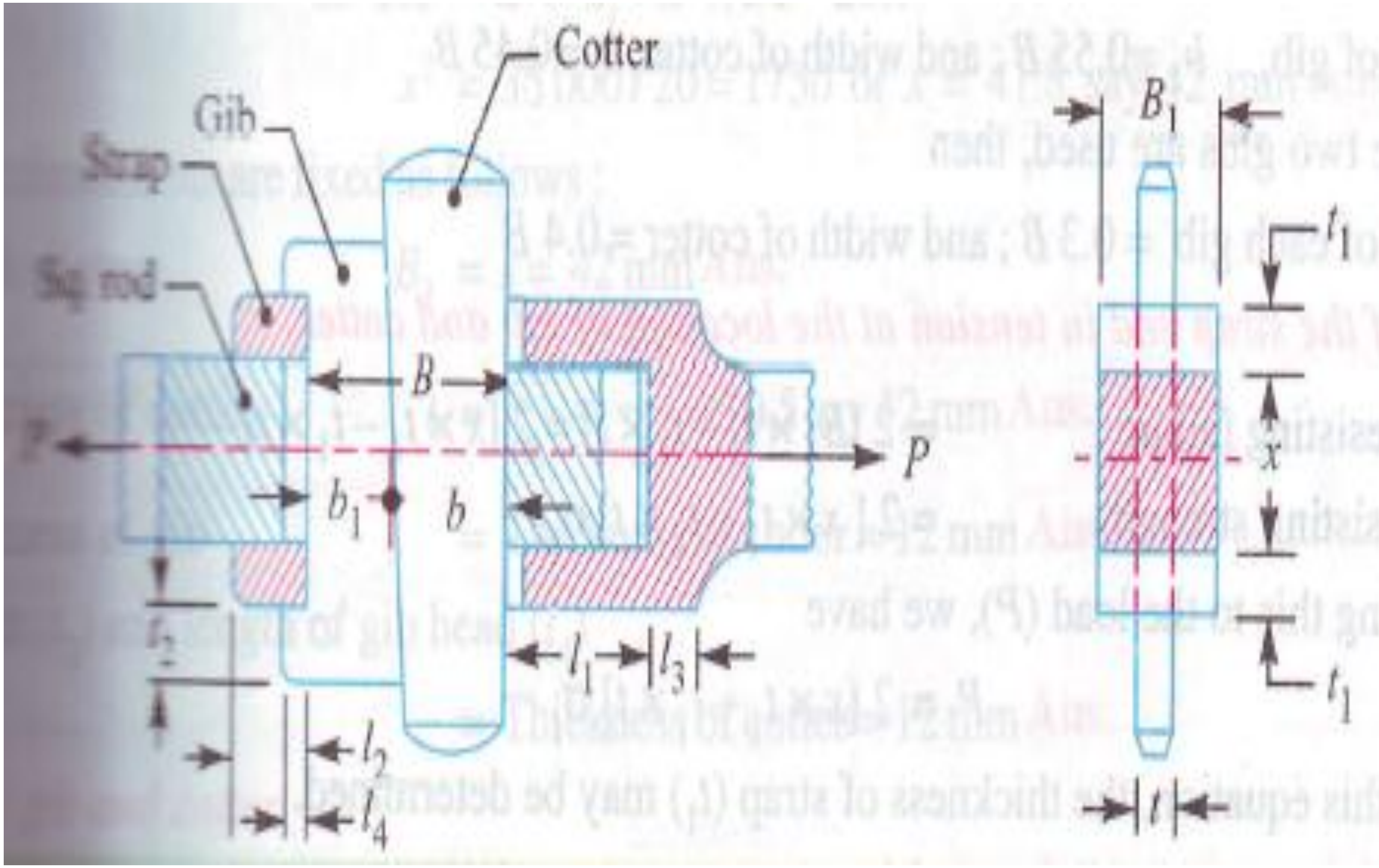


(b) Cotter with one gib.



(c) Cotter with double gib

Design of Gib and cotter joint for square rods



Cont…

P= Load carried by the rods,

x= each side of the rod,

B= total width of gib and cotter,

B_1= width of the strap,

t= thickness of the cotter,

t_1= thickness of the strap,

σ_t= permissible tensile stress for the rods material

ζ= permissible shear stress for cotter material and

σ_c= permissible crushing stress for the cotter material

1. Failure of rod in tension

Resisting area = x. x = x^2

Tearing strength of the rod = $x^2 . \sigma_t$

Equating to load P = $x^2 . \sigma_t$

From this equation side of the square rod may be determined.

Other dimensions are:-

Width of the strap, B_1= x

Thickness of the cotter, t = $B_1/4$

Thickness of the gib = thickness of cotter (t)

Height (t_2) and length of gib head (l_4) = thickness of cotter (t)

2. Failure of the gib and cotter in shearing

Resisting area = 2 B. t

Shearing strength of the rod = 2 B. t. ζ

Equating to load P = 2 B. t. ζ

From this equation width of the gib and cotter (B) may be obtained.

In case of one gib:-

Width of gib b_1= 0.55 B and width of cotter b= 0.45 B

In case of two gibs:-

Width of each gib b_1= 0.3 B and width of cotter b= 0.4 B

3. Failure of the strap end in tension:-

Resisting area = 2 $[x\ t_1 - t_1\ t]$

Resisting strength = 2 $[x\ t_1 - t_1 t]\ \sigma_t$

Equating to load $P = 2\ [x\ t_1 - t_1\ t] \times \sigma_t$

From this equation thickness of the strap (t_1) may be obtained.

4. Failure of the strap or gib in crushing:-

Resisting area = $2\ t_1\ t$

Resisting strength = $2\ t_1\ t\ \sigma_c$

Equating to load $P = 2\ t_1\ t\ \sigma_c$

From this equation crushing strength may be checked.

5. Failure of the rod end in shearing:-

Resisting area = 2 l_1.x

Resisting strength = 2 l_1.x.ζ

Equating to load P = 2 l_1.x.ζ

From this equation length of the rod (l_1) may be obtained.

6. Failure of the strap end in shearing:-

Resisting area = $2 .2\, l_2 .\, t_1$

Resisting strength = $2 .2\, l_2 .\, t_1 .\, \zeta$

Equating to load $P = 2 .2\, l_2 .\, t_1 .\, \zeta$

From this equation length of the rod (l_2) may be determined.

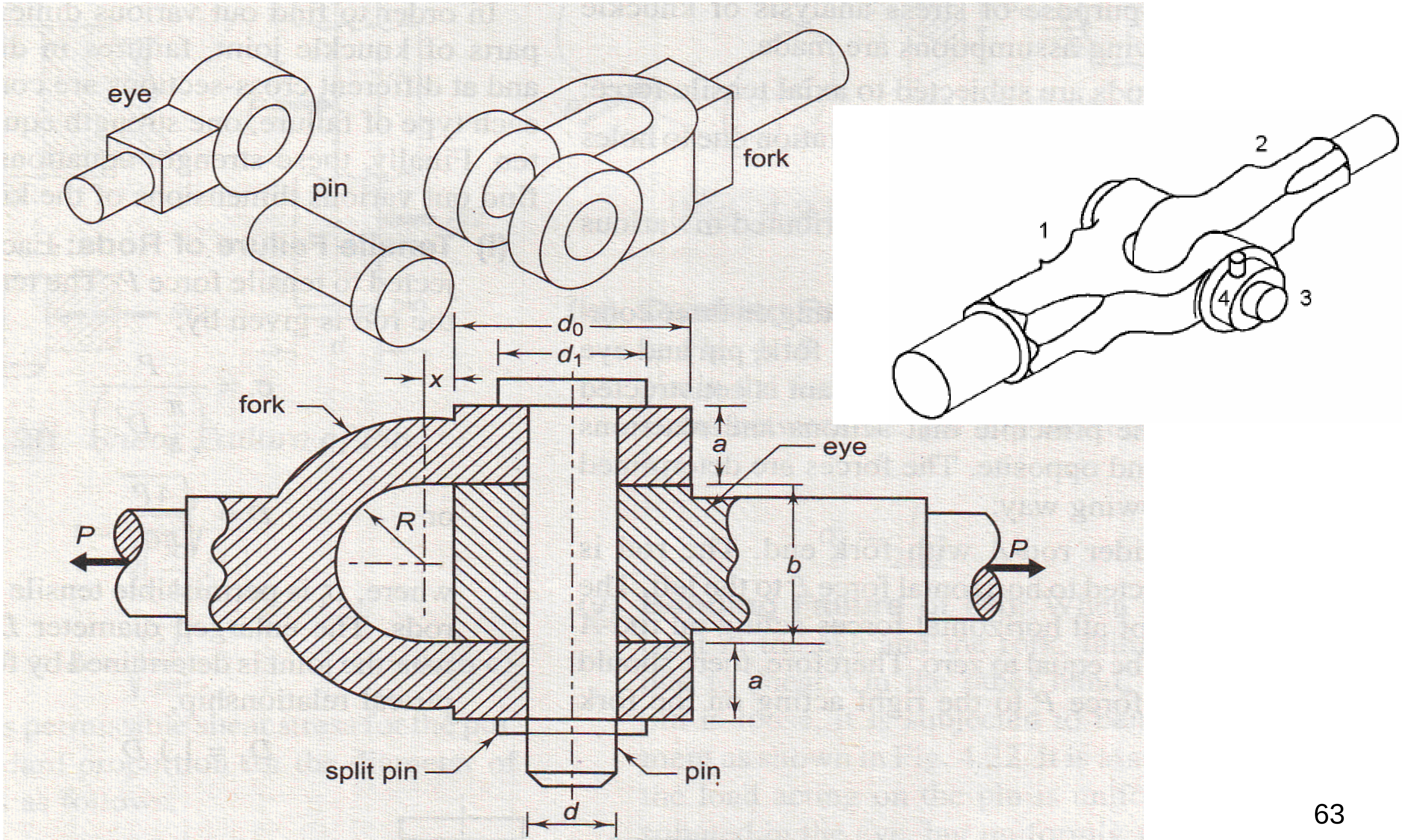
Problem 6. Design a gib and cotter joint to carry maximum load of 35 kN. Assuming that the gib, cotter and rod are made of the same material and have the following allowable stress.

Tensile stress = 20 MPa

Shear stress = 15 MPa

Crushing stress = 50 MPa

Knuckle joint



Cont…

P= tensile load acting on the rod,

D= diameter of each rod,

D_1= enlarged diameter of each rod,

d= diameter of the knuckle pin,

d_0= outside diameter of eye or fork,

a = thickness of each eye of fork,

b = thickness of eye end of rod-B

d_1= diameter of the pin head

x = distance of the centre of fork radius R from the eye

1. Tensile Failure of rods :

Load $$\boxed{P = \frac{\pi}{4} D^2 \sigma_t}$$

- The enlarged diameter D_1 of the rod is determined by the following empirical relationship

$$\boxed{D_1 = 1.1\ D}$$

2. Shear Failure of pin:

$$\boxed{P = 2 \times \frac{\pi}{4} d^2 \tau}$$

3. Crushing Failure of pin in Eye :

$$\boxed{\sigma_c = \frac{P}{bd}}$$

4. Crushing Failure of pin in fork :

$$\boxed{\sigma_c = \frac{P}{2ad}}$$

5. Bending Failure of pin :

$$\boxed{\sigma_b = \frac{32}{\pi d^3} \times \frac{P}{2}\left[\frac{b}{4} + \frac{a}{3}\right]}$$

6. Shear Failure of eye :

$$\boxed{\tau = \frac{P}{b(d_o - d)}}$$

7. Tensile Failure of fork :

$$\sigma_t = \frac{P}{2a(d_o - d)}$$

8. Shear Failure of fork :

$$\tau = \frac{P}{2a(d_o - d)}$$

9. Tensile Failure of eye :

$$\sigma_t = \frac{P}{b(d_o - d)}$$

- Standard proportions of the dimensions a and b are as follows,

$$a = 0.75\ D$$

$$b = 1.25\ D$$

- The diameter of the pin head is taken as,

$$d_1 = 1.5\ d$$

- The gap x shown in fig is usually taken as

$$x = 10\ mm$$

Design steps

1. Diameter of each rod (D) - by using equation

$$\boxed{D = \sqrt{\frac{4\,P}{\pi \sigma_t}}}$$

2. Enlarged diameter of each rod (D_1):

$$\boxed{D_1 = 1.1\ D}$$

3. Dimensions (a) and (b) :

$$\boxed{\begin{aligned} a &= 0.75\ D \\ b &= 1.25\ D \end{aligned}}$$

4. Diameter of pin (d) by using equations

$$\boxed{P = 2 \times \frac{\pi}{4} d^2 \tau}$$

&

$$\boxed{\sigma_b = \frac{32}{\pi d^3} \times \frac{P}{2}\left[\frac{b}{4} + \frac{a}{3}\right]}$$

Select the diameter whichever is maximum.

5. Outside diameter of eye (d_o) and diameter of pin head (d_1)

$$d_o = 2d$$

$$d_1 = 1.5\,d$$

6. **Check the tensile, crushing and shear stresses in the eye:** by using equations

$$\sigma_t = \frac{P}{b(d_o - d)}$$

$$\boldsymbol{\sigma_c = \frac{P}{bd}}$$

$$\tau = \frac{P}{b(d_o - d)}$$

7. **Check the tensile, crushing and shear stresses in the fork:** by using equations

$$\sigma_t = \frac{P}{2a(d_o - d)}$$

$$\sigma_c = \frac{P}{2ad}$$

$$\tau = \frac{P}{2a(d_o - d)}$$

Problem 7. Design a knuckle joint to transmit 150 kN. The design stress may be taken as 75 MPa in tension, 60 MPa in shear and 150 MPa in crushing. (2011)

Problem 8. A knuckle joint is to transmit a force of 140,000 N. Allowable stresses in tension, shear and compression are 75 N/mm^2, 65 N/mm^2 and 140 N/mm^2 respectively. Design the joint. (2010)

Assignment-2

Que.1 Design a Knuckle joint (with fork and eye) to with stand a load of 10,000 N. The materials used for all components have the following properties ultimate tensile strength = ultimate compressive strength = 480 N/mm2. Shear strength = 360 N/mm2. Factor of safety =6. After design, draw a neat proportioned sketch of joint giving all dimensions.

Que.2 Design a cotter joint to carry a maximum load of 50,000 N. All components are made of the same material having the following allowable stresses.

Tensile stress = 20 MN/m2

Compressive stress = 50 MN/m2

Shear stress = 15 MN/m2

Draw a proportioned neat sketch of the joint you have designed.

Que.3 It is required to design a square key for fixing a pulley on the shaft which is 50 mm in diameter. A 10 kW power at 200 r.p.m. is transmitted by the pulley to the shaft. The key is made of steel 45C8 (Syt= Syc= 380 N/mm2) and the factor of safety is 3. Determine the dimensions of the key.