Chapter 2

Cotter and Knuckle Joints

2.1 Introduction

- A cotter is a flat wedge shaped 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.
- The taper varies from I in 48 to 1 in 24.
- The cotter is usually made of mild steel or wrought iron.
- 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.
- It is usually used in connecting a piston rod to the cross-head of a reciprocating steam engine, a piston rod and its extension as a tail or pump rod, etc.

2.2 Types of Cotter Joints

- Following are the three commonly used cotter joints to connect two rods by a cotter:
  1. Socket and spigot cotter joint,
  2. Sleeve and cotter joint, and
  3. Gib and cotter joint.

2.2.1. Socket and Spigot Cotter Joint

- In a socket and spigot cotter joint, one end of the rods (say A) is provided with a socket type of end as shown in following Fig. and the other end of the other rod (say B) is inserted into a socket.
- The end of the rod which goes into a socket is also called spigot.
- A rectangular hole is made in the socket and spigot.
- A cotter is then driven tightly through a hole in order to make the temporary connection between the two rods.
- The load is usually acting axially, but it changes its direction and hence the cotter joint must be designed to carry both the tensile and compressive loads.

Design of Socket and Spigot Cotter Joint

- The socket and spigot cotter joint is shown in following Fig.

Let ,

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,
σt = Permissible tensile stress for the rods material,
τ = Permissible shear stress for the cotter material, and
σc = Permissible crushing stress for the cotter material.

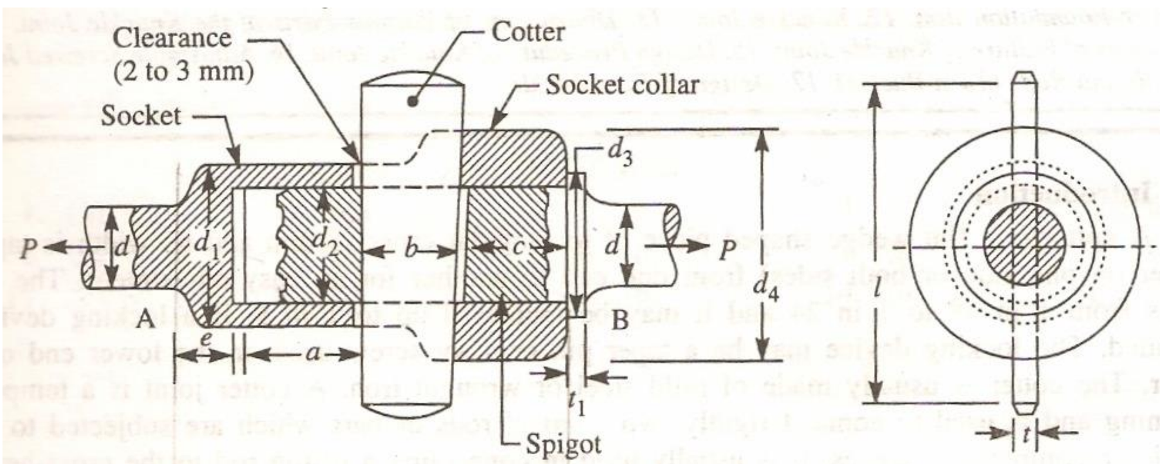
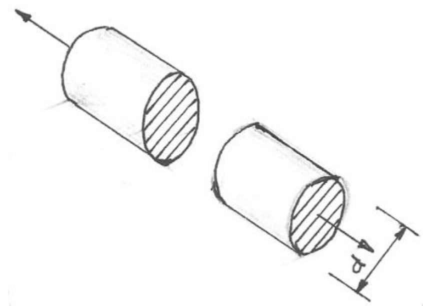


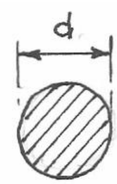
Fig. Socket and Spigot Cotter Joint

- ❖ The dimensions for a socket and spigot cotter joint may be obtained by considering the various modes of failure as discussed below.

<u>Step-1 Failure of the rods in tension</u>

- ❖ The rods may fail in tension due to the tensile load P. We know that Area resisting tension (tearing)





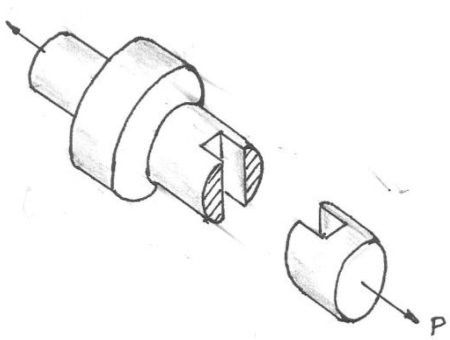
$$A = \frac{\pi}{4} d^2$$

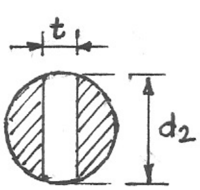
- ❖ Now tensile stress in the rods is given as

$$\boxed{\sigma t = \frac{P}{\frac{\pi d^2}{4}}}$$

- ❖ From this equation, diameter of the rods **(d)** may be determined.

<u>Step-2 Failure of spigot in tension across the weakest section (at slot)</u>





$$A = \left(\frac{\pi}{4} d_2^2 - d_2 \cdot t\right)$$

- ❖ Since the weakest section of the spigot is that section which has a slot in it for the cotter. The spigot will fail in tension across the wickest portion .

- ❖ Tensile stress in the spigot across the slot is given as

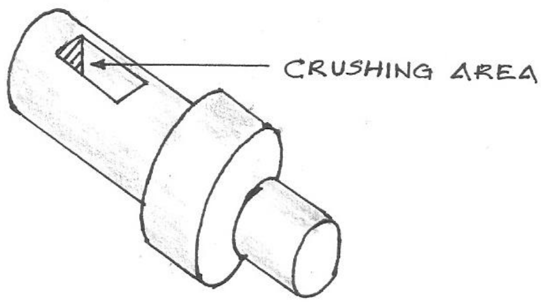
$$\sigma t = \frac{P}{\qquad}$$

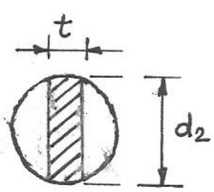
- ❖ From this equation, the diameter of spigot or inside diameter of socket (d_2) may be determined.

Note : In actual practice, the thickness of cotter is usually taken as $t = \frac{d2}{4}$

<u>Step-3 Failure of spigot end or cotter in crushing</u>

- ❖ We know that the area that resists crushing of spigot end or cotter is





A = d_2 . t

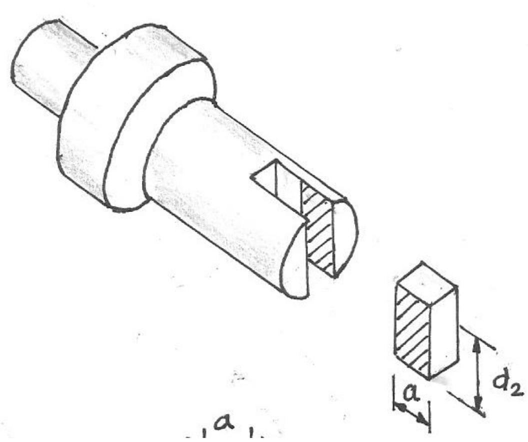
- ❖ Hence Crushing Stress is

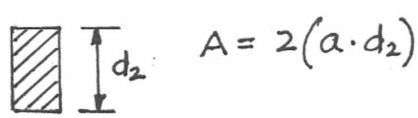
$$\sigma c = \frac{P}{d2\ .\ t}$$

- ❖ From this equation, the induced crushing stress σc may be checked.

<u>Step-4 Failure of spigot end in shearing</u>

- ❖ Since the rod end is in double shear, therefore the area resisting shear of the rod end





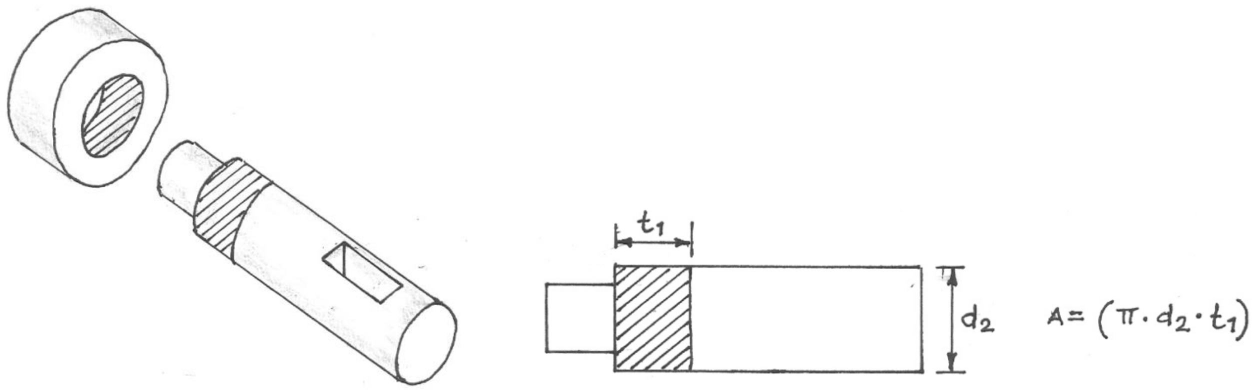
- ❖ Now shear stress in the rod end is

$$\tau = \frac{P}{2(a.d2)}$$

- ❖ From this equation, the distance from the end of the slot to the end of the spigot (a) may be obtained.

Step-5 Failure of spigot collar in shearing

- Considering the failure of the spigot collar in shearing as shown in following Fig. We know that area that resists shearing of the collar



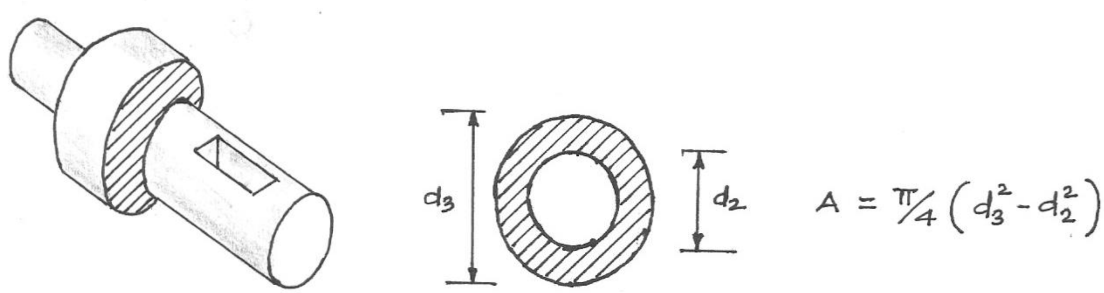
- shear stress in the collar is given as –

$$\tau = \frac{P}{\pi.\ d2.\ t1}$$

- From this equation, the thickness of spigot collar (t_1) may be obtained.

Step-6 Failure of spigot collar in crushing

- Considering the failure of the spigot collar in crushing as shown in following Fig. We know that area that resists crushing of the collar -



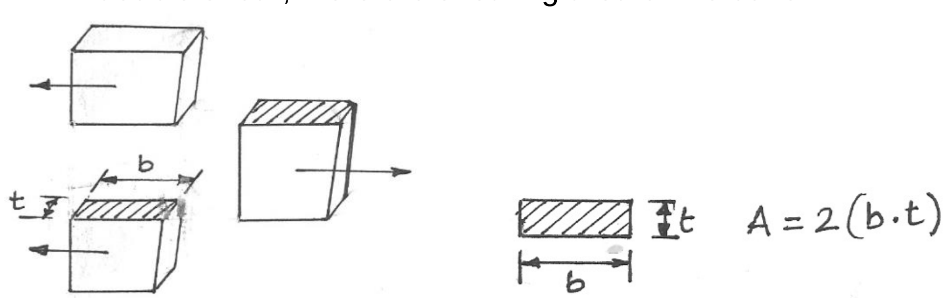
- crushing strength of the collar is –

$$\sigma c = \frac{P}{\qquad}$$

- From this equation, the diameter of the spigot collar (d_3) may be obtained.

Step-7 Failure of cotter in shear

- Considering the failure of cotter in shear as shown in following Fig. Since the cotter is in double shear, therefore shearing area of the cotter



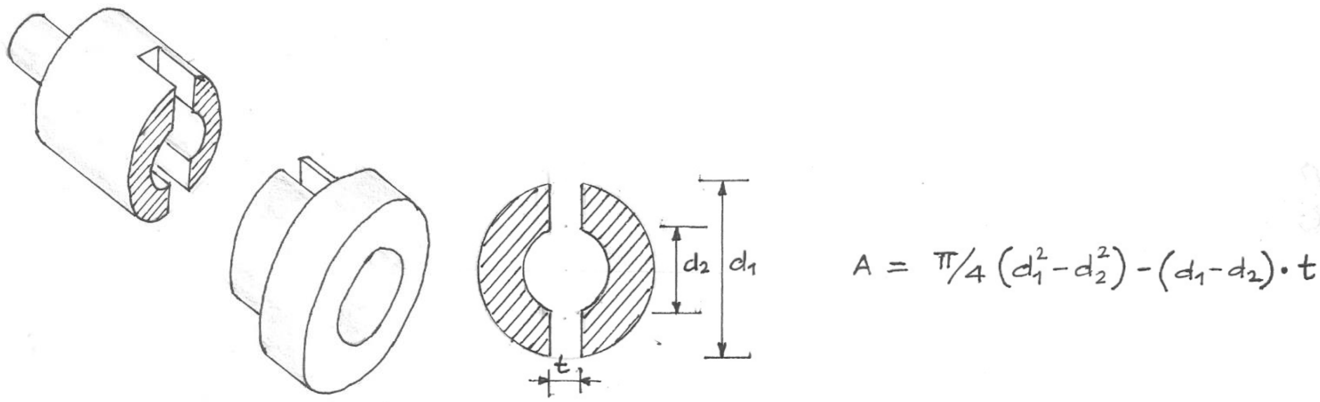
❖ shear stress of the cotter is given as –

$$= \frac{P}{2(b.t)}$$

❖ From this equation, width of cotter (b) is determined.

Step-8 Failure of the socket in tension across the slot

❖ We know that the resisting area of the socket across the slot, as shown in following Fig



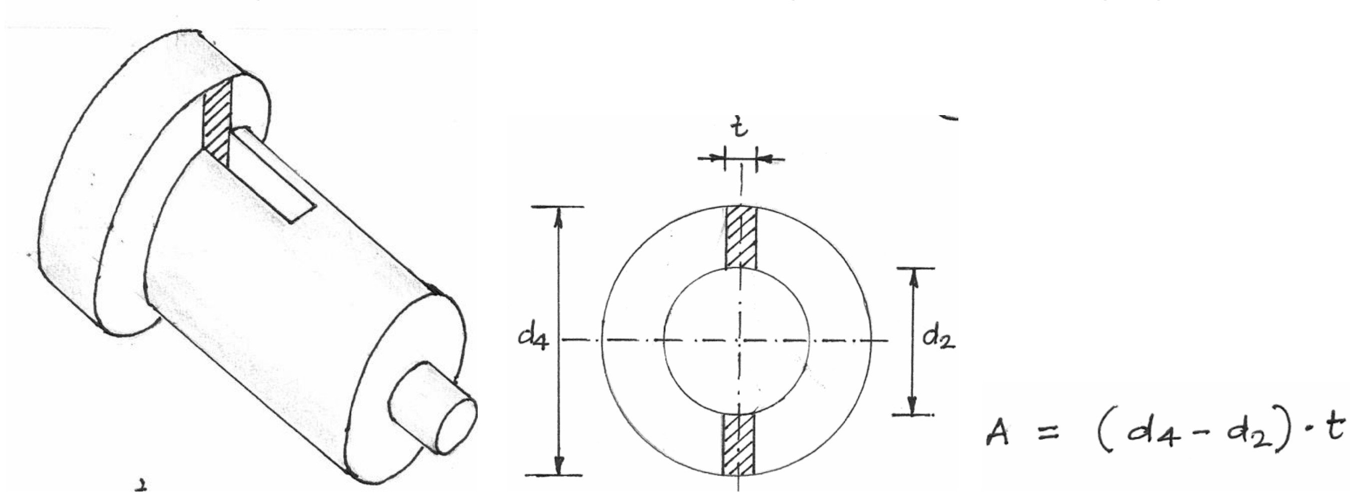
❖ Tearing stress of the socket across the slot is given as –

$$\sigma t = \frac{P}{\quad}$$

❖ From this equation, outside diameter of socket (d_1) may be determined.

Step-9 Failure of the socket in Crushing

❖ Considering the failure of socket collar in crushing as shown in following Fig.



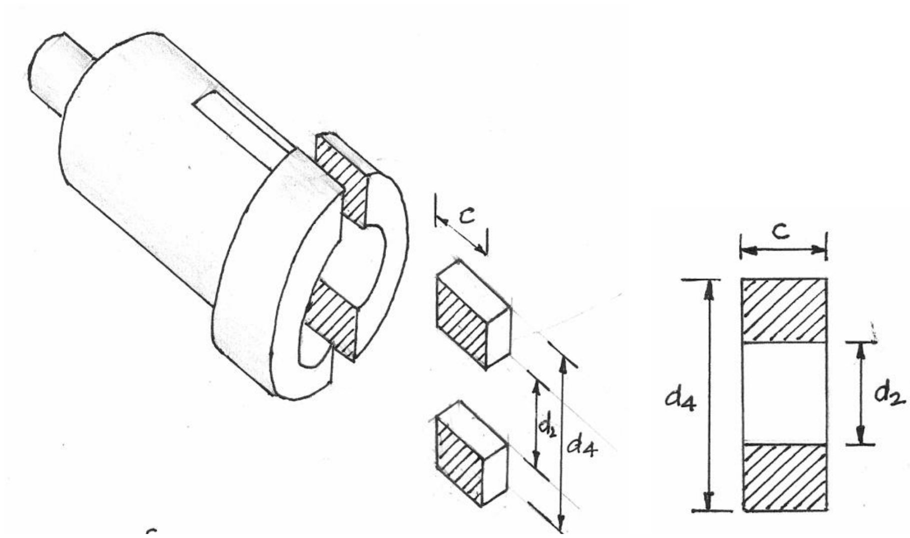
❖ crushing stress is given as –

$$\sigma c = \frac{P}{(d4 - d2).t}$$

❖ From this equation, the diameter of socket collar (d_4) may be obtained.

Step-10 Failure of the socket in Shearing

- Since the socket end is in double shear, therefore area that resists shearing of socket collar is



$$A = 2\,(d_4 - d_2)\cdot c$$

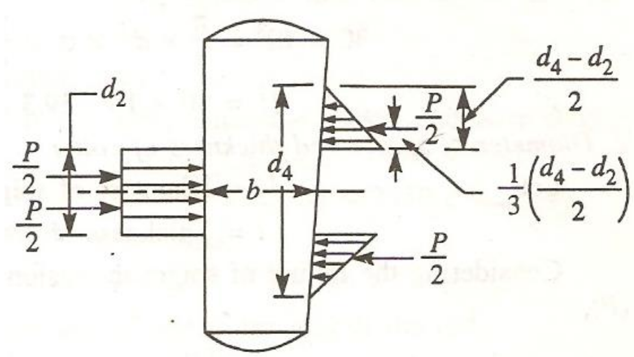
- shearing stress of socket collar is

$$\boxed{= \frac{P}{2(d4 - d2)c}}$$

- From this equation, the thickness of socket collar (c) may be obtained.

Step-11 Failure of cotter in bending

- In all the above relations, it is assumed that the load is uniformly distributed over the various cross-sections of the joint.
- But in actual practice, this does not happen and the cotter is subjected to bending.
- In order to find out the bending stress induced, it is assumed that the load on the cotter in the rod end is uniformly distributed while in the socket end it varies from zero at the outer diameter (d_4) and maximum at the inner diameter (d_2), as shown in Fig.



- Bending stress induced in the cotter is

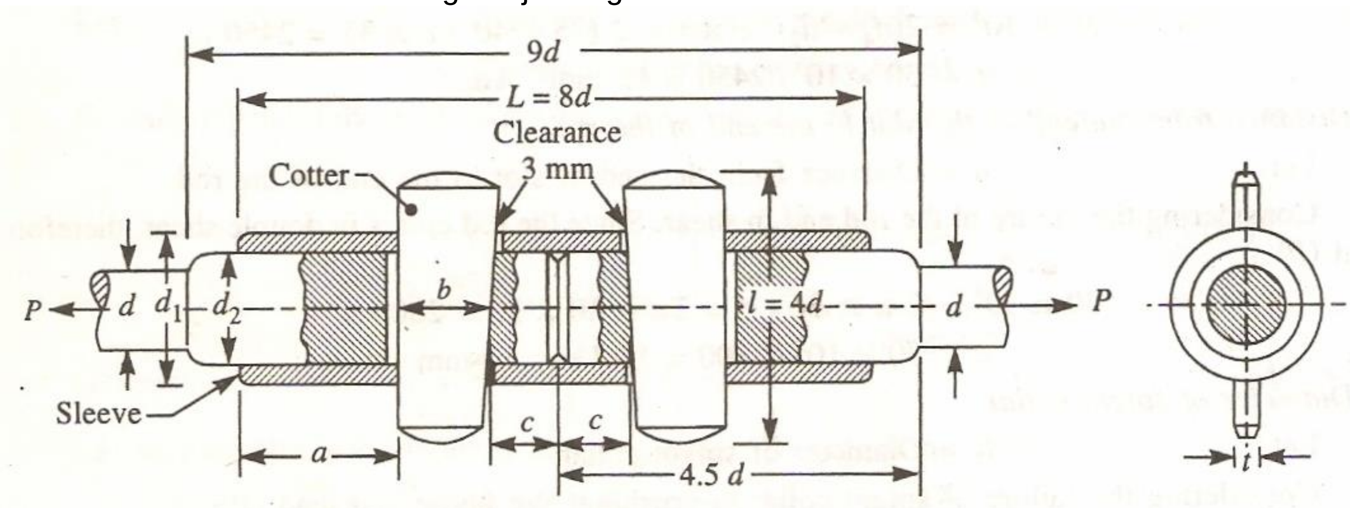
$$\sigma_b = \frac{M_{max}}{Z} = \frac{\frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{4}\right)}{t \times b^2/6} = \frac{P\,(d_4 + 0.5\,d_2)}{2\,t \times b^2}$$

Step-12 Other proportions

- The length of cotter (l) in taken as , l = 4 d.
- The taper in cotter should not exceed 1 in 24. In case the greater taper is required, then a locking device must be provided.

2.2.2 . Sleeve and Cotter Joint

- Sometimes, a sleeve and cotter joint as shown in following Fig. is used to connect two round rods or bars.
- In this type of joint, a sleeve or muff is used over the two rods and then two cotters (one on each rod end) are inserted in the holes provided for them in the sleeve and rods.
- The taper of cotter is usually 1 in 24. It may be noted that the taper sides of the two cotters should face each other as shown in Fig.
- The clearance is so adjusted that when the cotters are driven in, the two rods come closer to each other thus making the joint tight.



Let

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),

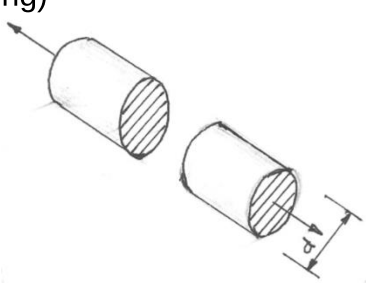
c = Distance of the rod end from its end to the cotter hole,

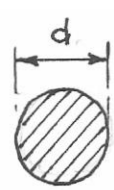
σ_t, τ and σ_c = Permissible tensile, shear and crushing stresses respectively for the material of the rods and cotter.

- The dimensions for a sleeve and cotter joint may be obtained by considering the various modes of failure as discussed below

Step-1 Failure of the rods in tension

- The rods may fail in tension due to the tensile load P. We know that Area resisting tension (tearing)





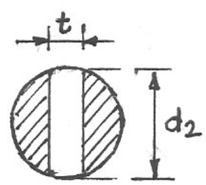
$$A = \frac{\pi}{4} d^2$$

- ❖ Now tensile stress in the rods is given as

$$\sigma t = \frac{P}{\frac{\pi d^2}{4}}$$

- ❖ From this equation, diameter of the rods **(d)** may be determined.

<u>Step-2 Failure of rod in tension across the weakest section (at slot)</u>



$$A = \left(\frac{\pi}{4} d_2^2 - d_2 \cdot t\right)$$

- ❖ Since the weakest section of the rod is that section which has a slot in it for the cotter. The rod will fail in tension across the wickest portion .
- ❖ Tensile stress in the rod across the slot is given as

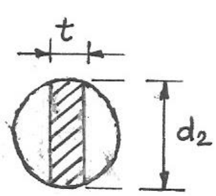
$$\sigma t = \frac{P}{\qquad}$$

- ❖ From this equation, the diameter of rod (d_2) may be determined.

Note : In actual practice, the thickness of cotter is usually taken as $t = \frac{d2}{4}$

<u>Step-3 Failure of rod end or cotter in crushing</u>

- ❖ We know that the area that resists crushing of rod end or cotter is



$A = d_2 \, . \, t$

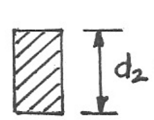
- ❖ Hence Crushing Stress is

$$\sigma c = \frac{P}{d2\,.\,t}$$

- ❖ From this equation, the induced crushing stress σc may be checked.

<u>Step-4 Failure of rod end in shearing</u>

- ❖ Since the rod end is in double shear, therefore the area resisting shear of the rod end



$$A = 2(a \cdot d_2)$$

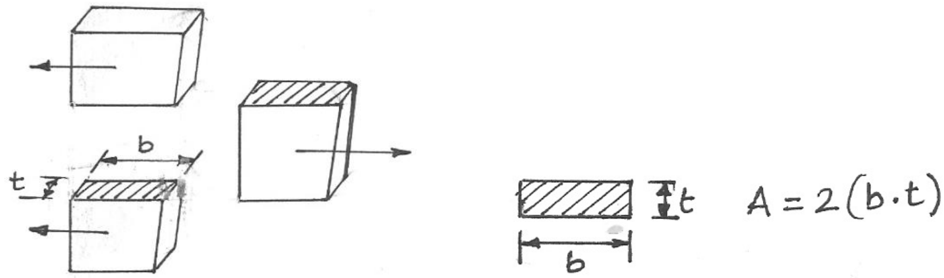
- ❖ Now shear stress of the rod end is

$$\tau = \frac{P}{2(a.d2)}$$

- From this equation, the distance from the end of the slot to the end of the spigot (a) may be obtained.

<u>Step-5 Failure of cotter in shear</u>

- Considering the failure of cotter in shear as shown in following Fig. Since the cotter is in double shear, therefore shearing area of the cotter



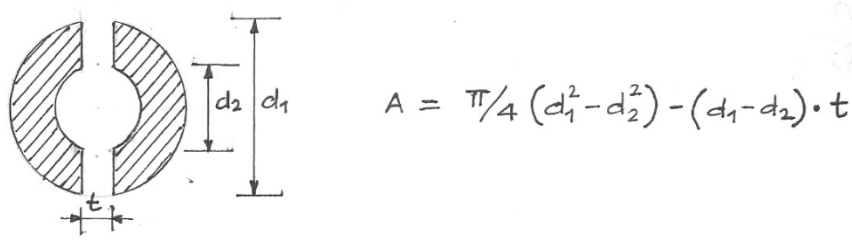
- Shear stress in the cotter is given as –

$$= \frac{P}{2(b.t)}$$

- From this equation, width of cotter (b) is determined.

<u>Step-6 Failure of the sleeve in tension across the slot</u>

- We know that the resisting area of the sleeve across the slot, as shown in following Fig



- Tearing stress of the sleeve across the slot is given as –

$$\sigma t = \frac{P}{\qquad\qquad}$$

- From this equation, outside diameter of sleeve (d_1) may be determined.

<u>Step-7 Failure of the sleeve end in shear</u>

- Since the socket end is in double shear, therefore area that resists shearing of socket collar is

$$= 2\ (d_1 - d_2)\ c$$

- shearing stress of socket collar is

$$= \frac{P}{2(d1 - d2)c}$$

- From this equation, the thickness of socket collar (c) may be obtained.

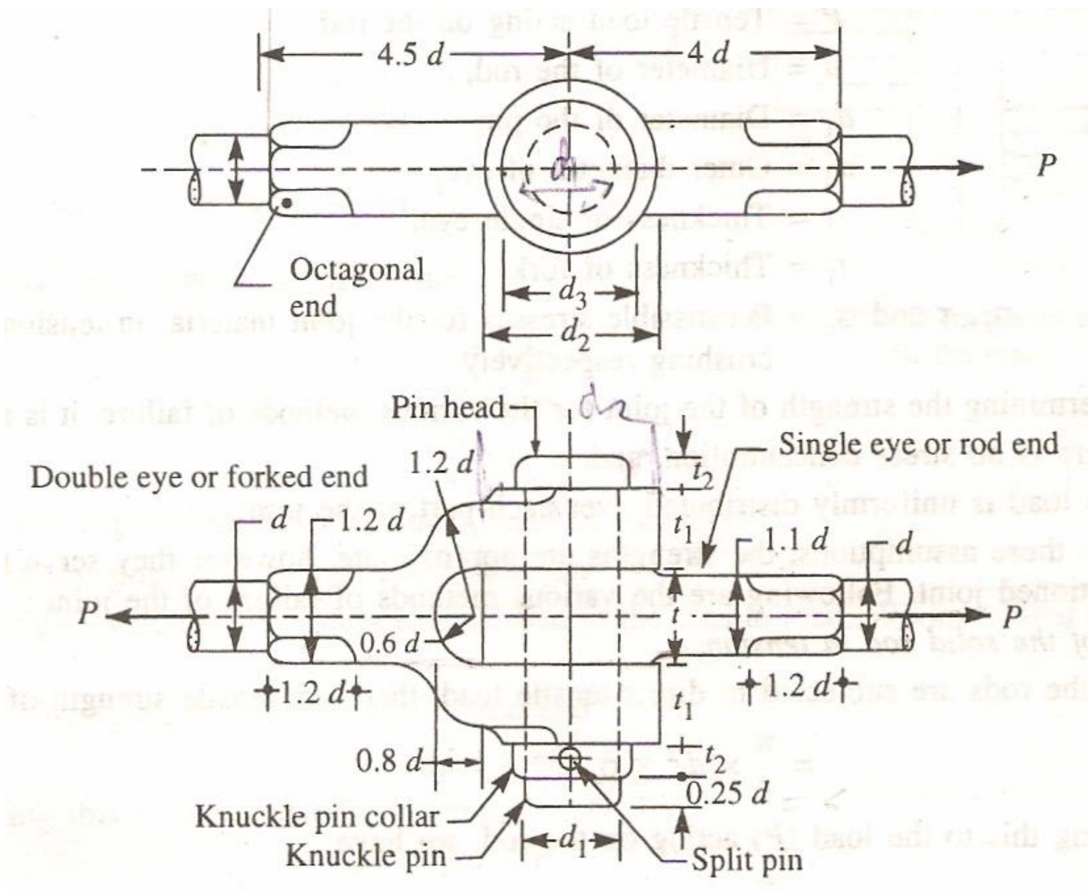
Step-8 The other proportions are

- Length of sleeve , L=8d
- Length of cotter , l=4d

2.3 Knuckle Joint

- A knuckle joint is used to connect two rods which are under the action of tensile loads. However, if the joint is guided, the rods may support a compressive load.
- Knuckle joint is also called as pin joint .
- Joint is not rigid and permits the angular motion of the rods
- A knuckle joint may he readily disconnected for adjustments or repairs.
- ***Uses***

  In the link of a cycle chain, tie rod joint for roof truss,
  valve rod joint with eccentric rod, pump rod joint,
  tension link in bridge structure and rod connections of various types.



- In knuckle joint (the two views of which are shown in above Fig.), one end of one of the rods is made into an eye and the end of the other rod is formed into a fork with an eye in each of the fork leg.
- The knuckle pin passes through both the eye hole and the fork holes and may be secured by means of a collar and taper pin or spilt pin.
- The knuckle pin may be prevented from rotating in the fork by means of a small stop, pin, peg or snug.
- In order to get a better quality of joint, the sides of the fork and eye are machined, the hole is accurately drilled and pin turned.
- The material used for the joint may be steel or wrought iron.

Design of knuckle joint

- Consider a knuckle joint as shown in above Fig.

Let

P = Tensile load acting on the rod,

d = Diameter of the rod,

d_1 = Diameter of the pin,

d_2 = Outer diameter of eye,

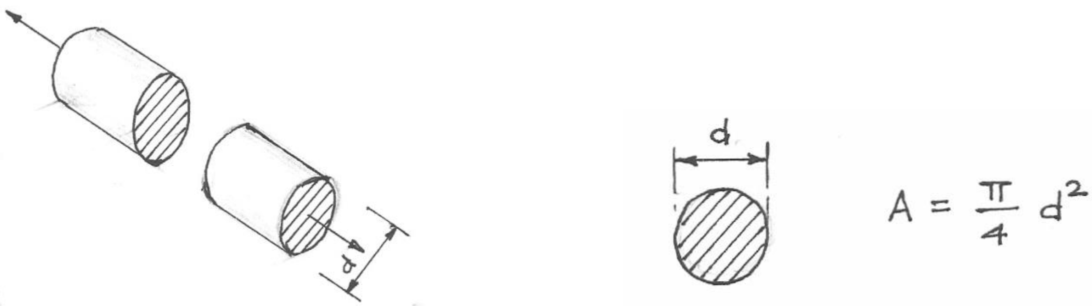
t = Thickness of single eye,

t_1 = Thickness of fork.

σ_t, τ and σ_c = Permissible stresses for the joint material in tension, shear and crushing respectively.

Step-1 Failure of the rods in tension

- The rods may fail in tension due to the tensile load P. We know that Area resisting tension (tearing)



- Now tensile stress in the rods is given as

$$\boxed{\sigma t = \frac{P}{\frac{\pi d^2}{4}}}$$

- From this equation, diameter of the rods **(d)** may be determined.

Step-2 Various proportions of knuckle joint are as below

If d is the diameter of rod, then diameter of pin,

$$d_1 = d$$

Outer diameter of eye,

$$d_2 = 2\ d$$

Diameter of knuckle pin head and collar,

$$d_3 = 1.5\ d$$

Thickness of single eye or rod end,

$$t = 1.25\ d$$

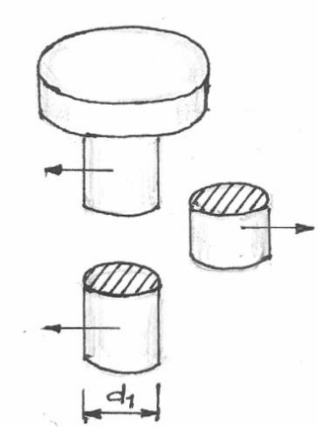
Thickness of fork, $t_1 = 0.75\ d$

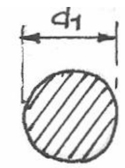
Thickness of pin head,

$$t_2 = 0.5\ d$$

Step-3 Failure of knuckle pin in shear

- Since the pin is in double shear, therefore cross-sectional area of the pin under shearing is





$$A = 2\left(\frac{\pi}{4} d_1^2\right)$$

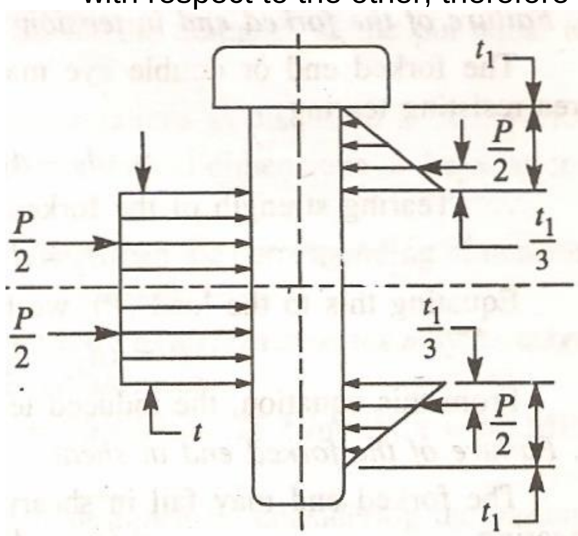
- The shear stress in the pin is given as –

$$= \frac{P}{}$$

- From this equation, diameter of the knuckle pin (d_1) is obtained.

Step-4 Failure of knuckle pin in bending

- In actual practice, the knuckle pin is loose in forks in order to permit angular movement of with respect to the other, therefore the pin is subjected to bending in addition to shearing.

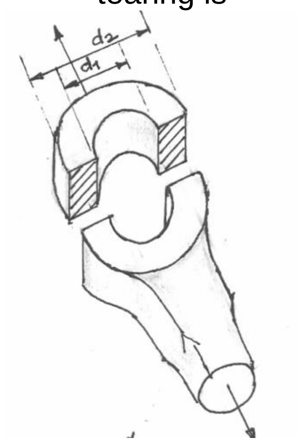


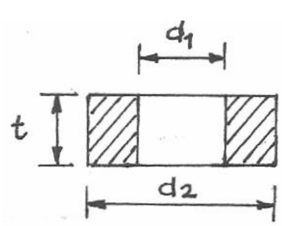
∴ Maximum bending (tensile) stress,

$$\sigma_t = \frac{M}{Z} = \frac{\frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{4}\right)}{\frac{\pi}{32}(d_1)^3}$$

Step-5 Failure of single eye end in Tension

- The single eye or rod end may tear off due to the tensile load. We know that area resisting tearing is





$$A = (d_2 - d_1)\,t$$

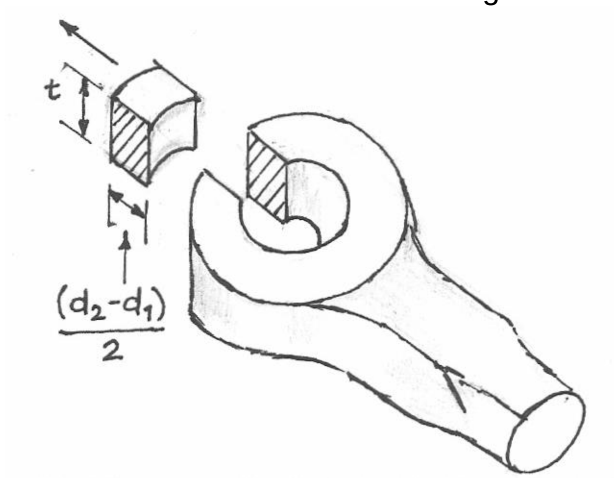
- Tensile stress of single eye end is

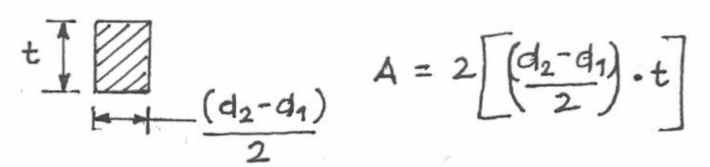
$$\boxed{\sigma t = \frac{P}{(d2 - d1)t}}$$

- From this equation, the induced tensile stress (σt) for the single eye or rod end may be checked.
- In case the induced tensile stress is more than the allowable working stress, then increase the outer diameter of the eye (d_2).

Step-6 Failure of single eye end in shear

- The single eye or rod end may fail in shearing due to load.
- We know that area resisting shearing





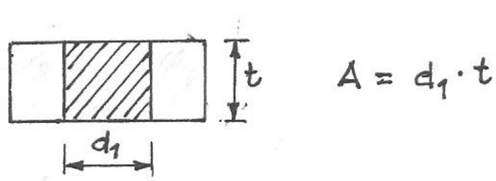
- Shear stress in single eye end is

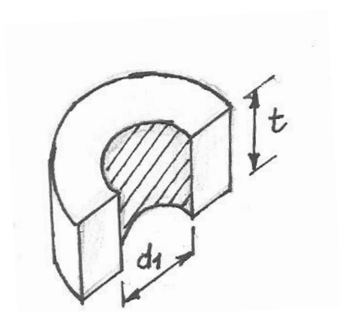
$$\boxed{= \frac{P}{(d2 - d1)\ t}}$$

- From this equation, the induced shear stress (τ) for the single eye or rod end may be checked.

Step-7 Failure of single eye end in crushing

- The single eye or pin may fail in crushing due to the tensile load.
- We know that area resisting crushing is





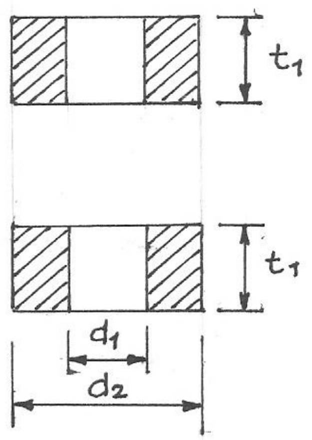
- ❖ Crushing stress of single eye end is

$$\boxed{\sigma c = \frac{P}{d1\,.\,t}}$$

- ❖ From this equation, the induced crushing stress (σc) for the single eye may be checked.

<u>Step-8</u> ***<u>Failure of double eye end in Tension</u>***

- ❖ The double eye may fail in tension due to the tensile load.
- ❖ We know that area resisting tearing



$$A = 2\left[(d_2 - d_1)\cdot t_1\right]$$

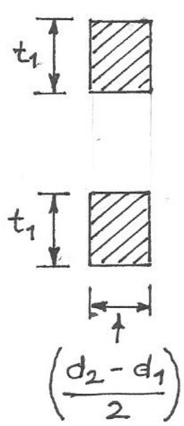
- ❖ Therefore Tensile stress in the double eye end is –

$$\boxed{\sigma t = \frac{P}{2(d2 - d1)\,t1}}$$

- ❖ From this equation, the induced tensile stress (σt) for the double eye (forked) end may be checked.

<u>Step-9</u> ***<u>Failure of double eye end in shear</u>***

- ❖ The double eye or rod end may fail in shearing due to load.
- ❖ We know that area resisting shearing



$$A = 2\left[2\left(\frac{d_2 - d_1}{2}\right)\cdot t_1\right]$$

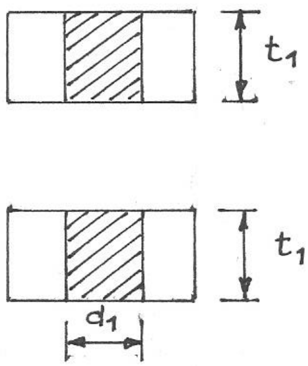
- Shear stress in double eye end is

$$= \frac{P}{2\,(d2 - d1)\; t1}$$

- From this equation, the induced shear stress (τ) for the double eye or rod end may be checked.
- In case induced shear stress is more than the allowable working stress, then thickness of the fork (t_1) is increased.

Step-10 Failure of double eye end in crushing

- The double eye may fail in crushing due to the tensile load.
- We know that area resisting crushing is



$$A = 2\,(d_1 \cdot t_1)$$

- Crushing stress of double eye end is

$$\sigma c = \frac{P}{2.\,d1\,.\;t1}$$

- From this equation, the induced crushing stress (σc) for the double eye may be checked.

Example ::: Design a knuckle joint to transmit 150 kW. The design stresses may be taken as 75 MPa in tension, 60 MPa in shear and 150 MPa in compression.

Solution. Given : P = 150 kN = 150 × 10^3 N ; σ_t = 75 MPa = 75 N/mm^2 ; τ = 60 MPa = 60 N/mm^2 ; σ_c = 150 MPa = 150 N/mm^2

1. *Failure of the solid rod in tension*

Let d = Diameter of the rod.

We know that the load transmitted (P),

$$150 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 75 = 59\, d^2$$

$\therefore$ $d^2 = 150 \times 10^3 / 59 = 2540$ or d = 50.4 say 52 mm **Ans.**

Now the various dimensions are fixed as follows :

Diameter of knuckle pin,

$$d_1 = d = 52 \text{ mm}$$

Outer diameter of eye, $d_2 = 2\,d = 2 \times 52 = 104$ mm

Diameter of knuckle pin head and collar,

$$d_3 = 1.5\,d = 1.5 \times 52 = 78 \text{ mm}$$

Thickness of single eye or rod end,

$$t = 1.25\,d = 1.25 \times 52 = 65 \text{ mm}$$

Thickness of fork, $t_1 = 0.75\,d = 0.75 \times 52 = 39$ say 40 mm

Thickness of pin head, $t_2 = 0.5\,d = 0.5 \times 52 = 26$ mm

2. *Failure of the knuckle pin in shear*

Since the knuckle pin is in double shear, therefore load (P),

$$150 \times 10^3 = 2 \times \frac{\pi}{4}(d_1)^2\,\tau = 2 \times \frac{\pi}{4}(52)^2\,\tau = 4248\,\tau$$

$$\therefore \quad \tau = 150 \times 10^3/4248 = 35.3 \text{ N/mm}^2 = 35.3 \text{ MPa}$$

3. *Failure of the single eye or rod end in tension*

The single eye or rod end may fail in tension due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\,t \times \sigma_t = (104 - 52)\,65 \times \sigma_t = 3380\,\sigma_t$$

$$\therefore \quad \sigma_t = 150 \times 10^3/3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

4. *Failure of the single eye or rod end in shearing*

The single eye or rod end may fail in shearing due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\,t \times \tau = (104 - 52)\,65 \times \tau = 3380\,\tau$$

$$\therefore \quad \tau = 150 \times 10^3/3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

5. *Failure of the single eye or rod end in crushing*

The single eye or rod end may fail in crushing due to the load. We know that load (P),

$$150 \times 10^3 = d_1 \times t \times \sigma_c = 52 \times 65 \times \sigma_c = 3380\,\sigma_c$$

$$\therefore \quad \sigma_c = 150 \times 10^3/3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

6. *Failure of the forked end in tension*

The forked end may fail in tension due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\,2\,t_1 \times \sigma_t = (104 - 52)\,2 \times 40 \times \sigma_t = 4160\,\sigma_t$$

$$\therefore \quad \sigma_t = 150 \times 10^3/4160 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

7. *Failure of the forked end in shear*

The forked end may fail in shearing due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\,2\,t_1 \times \tau = (104 - 52)\,2 \times 40 \times \tau = 4160\,\tau$$

$$\therefore \quad \tau = 150 \times 10^3/4160 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

8. *Failure of the forked end in crushing*

The forked end may fail in crushing due to the load. We know that load (P),

$$150 \times 10^3 = d_1 \times 2\ t_1 \times \sigma_c = 52 \times 2 \times 40 \times \sigma_c = 4160\ \sigma_c$$

$$\therefore \qquad \sigma_c = 150 \times 10^3/4180 = 36\ \text{N/mm}^2 = 36\ \text{MPa}$$

2.4 Adjustable Screwed Joint for Round Rods (Turnbuckle)

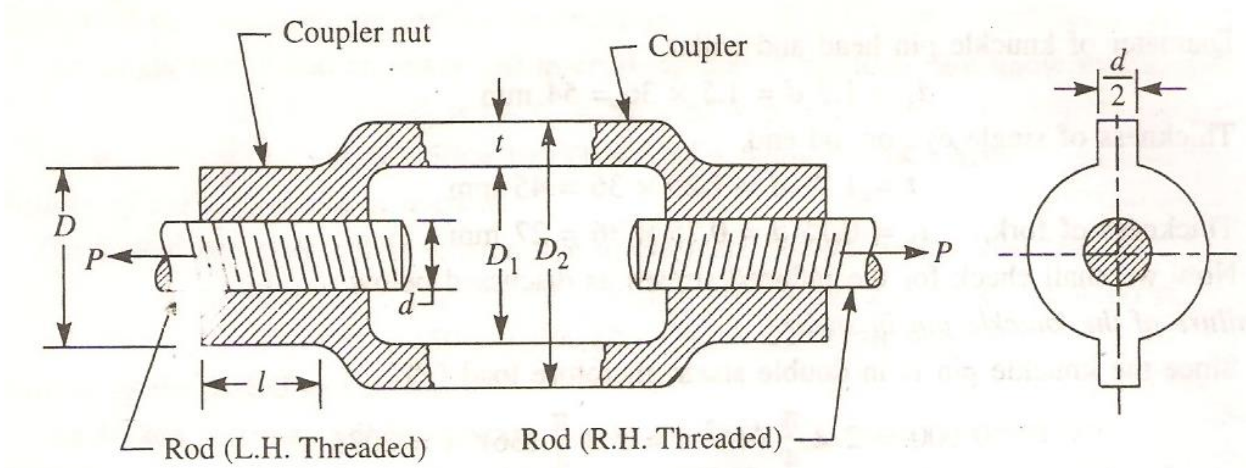
- Sometimes, two round tie rods, as shown in following Fig. are connected by means of a coupling Known as a turnbuckle.
- In this type of joint, one of the rods has right hand threads and the other rod has left hand threads.
- The rods are screwed to a coupler which has a threaded hole.
- The coupler is of hexagonal or rectangular shape in the centre and round at both the ends in order to facilitate the rods to tighten or loosen with the help of a spanner when required.
- Sometimes instead of a spanner, a round iron rod may be used

Applications:

1. It can be used as link between the railway wagon
2. Used in roof structure

Design of Turnbuckle

- Consider a turnbuckle, subjected to an axial load P, as shown in following Fig.



- Let ,

  dc = core diameter of tie rod

  d = nominal (maximum) diameter of rod

  D = outside diameter of coupler nut

  l = Length of coupler nut

  D1 = Inside diameter of coupler

  D2 = Outside diameter of coupler

  L = Total length of coupler

  σt = tensile stress

  σc = Crushing stress and

  τ = Shear stress , for coupler and tie rod material .

- Giving a margin for higher coefficient of friction, the maximum principal stress may be taken as 1.3 times the normal stress.

- Therefore for designing a threaded section, we shall take the design load as 1.3 times the normal load, i.e.,

$$\boxed{\text{Design load,} \quad P_d = 1.3\,P}$$

- ***The following procedure may be adopted in designing a turn-buckle:***

<u>Step-1 Diameter of the rods in tension</u>

- The diameter of the rods (d) may be obtained by considering the tearing of the threads of the rods at their roots.
- We know that area resisting Tearing

$$= \frac{\pi}{4}\,(d_c)^2$$

- Now the tensile stress in the threads is given as –

$$\boxed{\sigma t = \frac{1.3P}{\frac{\pi (dc)^2}{4}}}$$

- From the above expression, the core diameter of the threads (dc) may be obtained.
- The nominal diameter (d) may be obtained by using relation –

$$\boxed{\frac{dc}{d} = 0.84}$$

<u>Step-2 Length of the coupler nut</u>

- The length of the coupler nut (l) is obtained by considering the shearing of the threads at their roots in the coupler nut.
- We know that area resisting Shearing at the roots of the threads of the coupler nut is

$$= (\pi\, d_c \times l)$$

- Hence shear stress in nut is given as

$$\boxed{= \frac{1.3\ P}{\pi.d.l}}$$

- From this expression, the value of (l) may be calculated.
- In actual practice, the length of coupler nut (l) is taken ***<u>d to 1.25</u> d*** for steel nuts and ***<u>1.5 d to 2 d</u>*** for cast iron and softer material

<u>Step-3 Crushing of threads of coupler nut</u>

- We know that area resisting crushing is

$$= \frac{\pi}{4}\left[(d)^2 - (d_c)^2 \right] n \times l$$

- Now Crushing stress induced in the coupler nut is

$$\sigma c = \frac{1.3P}{\quad}$$

- From this expression, the induced crushing stress (σc) may be checked.

Step-4 Outside diameter of the coupler nut (D)

- The outside diameter of the coupler nut (D) may be obtained by considering the tearing at the coupler nut.
- We know that area resisting tearing at the coupler nut is –

$$= \frac{\pi}{4}(D^2 - d^2)$$

- Now tearing stress induced is given as –

$$\boxed{\sigma t = \frac{P}{\qquad\qquad}}$$

- From this expression, the value of D may be calculated.
- In actual practice, the diameter of the coupler nut (D) is taken from ***1.25 d to 1.5 d.***

Step-5 Outside diameter of the coupler

- Inside diameter of the coupler is generally taken as D1 = (d + 6 mm),
- The outside diameter of the coupler (D2) may be obtained by considering the tearing of the coupler.
- Now tearing stress induced is given as –

$$\boxed{\sigma t = \frac{P}{\qquad\qquad}}$$

- From this equation find diameter D2.
- In actual practice, the outside diameter of the coupler (D2) is taken as ***1.5 d to 1.7 d***.
- If the section of the coupler is to be made hexagonal or rectangular to fit the spanner, it may be circumscribed over the circle of outside diameter D2.

Step-6 Other proportions are

- The length of the coupler between the nuts (L) depends upon the amount of adjustment required. It is usually taken as

L = 6 d.

- The thickness of the coupler is usually taken as

t = 0.75 d

Example The pull in the tie rod of an iron roof truss is 50 kN. Design a suitable adjustable screwed joint. The permissible stresses are 75 MPa in tension, 3Z5 MPa in shear and 90 MPa in crushing.

Solution. Given : $P = 50$ kN $= 50 \times 10^3$ N ; $\sigma_t = 75$ MPa $= 75$ N/mm^2 ; $\tau = 37.5$ MPa $= 37.5$ N/mm^2

We know that the design load for the threaded section,

$$P_d = 1.3\ P = 1.3 \times 50 \times 10^3 = 65 \times 10^3 \text{ N}$$

An adjustable screwed joint, as shown in Fig. 12.19, is suitable for the given purpose. The various dimensions for the joint are determined as discussed below :

1. *Diameter of the tie rod*

Let d = Diameter of the tie rod, and

d_c = Core diameter of threads on the tie rod.

Considering tearing of the threads on the tie rod at their roots.

We know that design load (P_d),

$$65 \times 10^3 = \frac{\pi}{4}(d_c)^2\ \sigma_t = \frac{\pi}{4}(d_c)^2\ 75 = 59\ (d_c)^2$$

$$\therefore \qquad (d_c)^2 = 65 \times 10^3 / 59 = 1100 \text{ or } d_c = 33.2 \text{ mm}$$

From Table 11.1 for coarse series, we find that the standard core diameter is 34.093 mm and the corresponding nominal diameter of the threads or diameter of tie rod,

d = 39 mm **Ans.**

2. *Length of the coupler nut*

Let l = Length of the coupler nut.

Considering the shearing of threads at their roots in the coupler nut. We know that design load (P_d),

$$65 \times 10^3 = (\pi\ d_c.l)\ \tau = \pi \times 34.093 \times l \times 37.5 = 4107\ l$$

$$\therefore \qquad l = 65 \times 10^3 / 4017 = 16.2 \text{ mm}$$

Since the length of the coupler nut is taken from d to $1.25\ d$, therefore we shall take

$l = d$ = 39 mm **Ans.**

We shall now check the length of the coupler nut for crushing of threads.

From Table 11.1 for coarse series, we find that the pitch of the threads is 4 mm. Therefore the number of threads per mm length,

$$n = 1/4 = 0.25$$

We know that design load (P_d),

$$65 \times 10^3 = \frac{\pi}{4}\left[(d)^2 - (d_c)^2\right] n \times l \times \sigma_c$$

$$= \frac{\pi}{4}\left[(39)^2 - (34.093)^2\right] 0.25 \times 39 \times \sigma_c = 2750\ \sigma_c$$

$$\therefore \qquad \sigma_c = 65 \times 10^3 / 2750 = 23.6 \text{ N/mm}^2 = 23.6 \text{ MPa}$$

Since the induced crushing stress in the threads of the coupler nut is less than the permissible stress, therefore the design is satisfactory.

3. *Outside diameter of the coupler nut*

Let D = Outside diameter of the coupler nut.

Considering tearing of the coupler nut. We know that axial load (P),

$$50 \times 10^3 = \frac{\pi}{4}(D^2 - d^2)\,\sigma_t$$

$$= \frac{\pi}{4}\left[D^2 - (39)^2\right] 75 = 59\left[D^2 - (39)^2\right]$$

or $$D^2 - (39)^2 = 50 \times 10^3 / 59 = 848$$

$\therefore$ $$D^2 = 848 + (39)^2 = 2369 \text{ or } D = 48.7 \text{ say } 50 \text{ mm} \quad \textbf{Ans.}$$

Since the minimum outside diameter of coupler nut is taken as 1.25 d (*i.e.*, 1.25 × 39 = 48.75 mm), therefore the above value of D is satisfactory.

4. *Outside diameter of the coupler*

Let D_2 = Outside diameter of the coupler, and

D_1 = Inside diameter of the coupler = d + 6 mm = 39 + 6 = 45 mm

Considering tearing of the coupler. We know that axial load (P),

$$50 \times 10^3 = \frac{\pi}{4}\left[(D_2)^2 - (D_1)^2\right]\sigma_t = \frac{\pi}{4}\left[(D_2)^2 - (45)^2\right] 75 = 59\left[(D_2)^2 - (45)^2\right]$$

$\therefore$ $$(D_2)^2 = 50 \times 10^3 / 59 + (45)^2 = 2873 \quad \text{or} \quad D_2 = 53.6 \text{ mm}$$

Since the minimum outside diameter of the coupler is taken as 1.5 d (*i.e.*, 1.5 × 39 = 58.5 say 60 mm), therefore we shall take

$$D_2 = 60 \text{ mm} \quad \textbf{Ans.}$$

5. Length of the coupler between nuts,

$$L = 6\,d = 6 \times 39 = 234 \text{ mm} \quad \textbf{Ans.}$$

6. Thickness of the coupler,

$$t_1 = 0.75\,d = 0.75 \times 39 = 29.25 \text{ say } 30 \text{ mm} \quad \textbf{Ans.}$$

and thickness of the coupler nut,

$$t = 0.5\,d = 0.5 \times 39 = 19.5 \text{ say } 20 \text{ mm} \quad \textbf{Ans.}$$