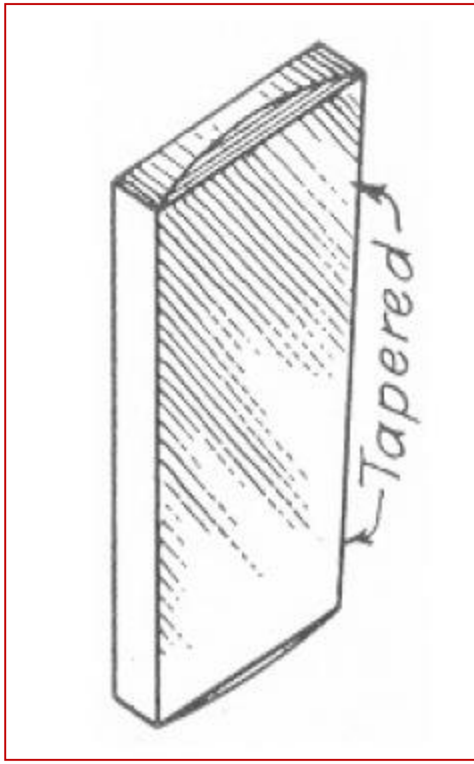
Cotter joints

- A cotter is a flat wedge-shaped piece of steel
- This is used to connect rigidly two rods which transmit motion in the axial direction, without rotation.
- These joints may be subjected to tensile or compressive forces along the axes of the rods
- It is uniform in thickness but tapering in width , generally on one side only. Usually the taper is 1 in 30.

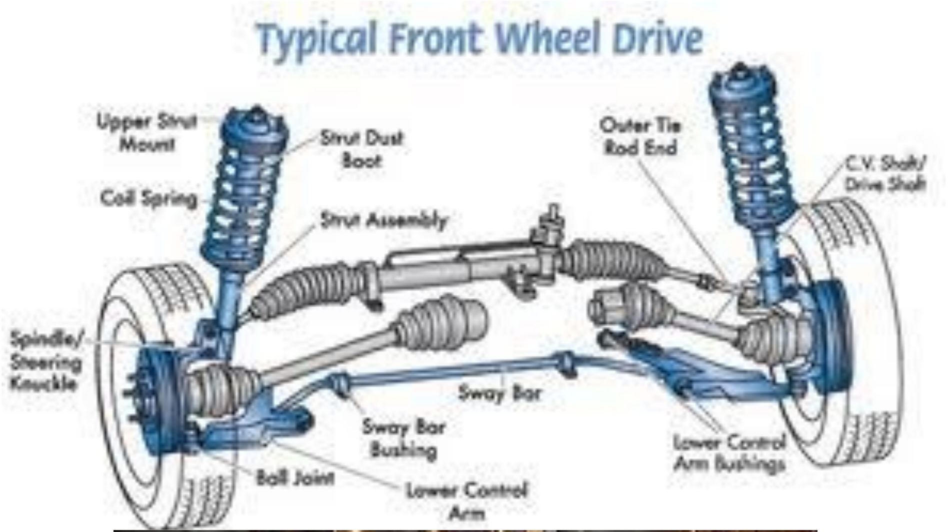


Cotter joints

Applications:

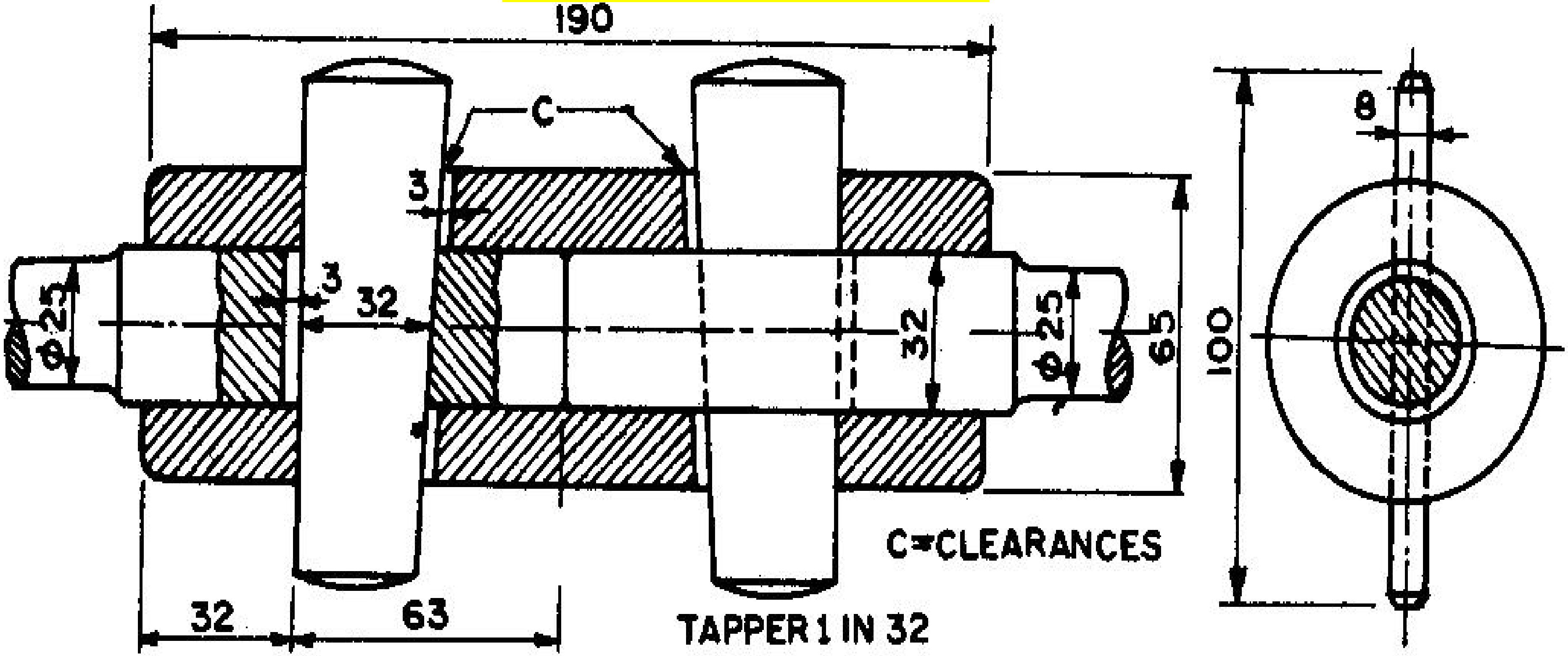
- Piston rod to the crosshead of a steam engine,
- Piston rod and its extension as a tail or pump rod
- Valve rod and its stem
- Strap end of connecting rod
- Foundation bolts to fasten heavy machinery to foundations …..etc.,

Cotter joints



Sleeve and cotter joint

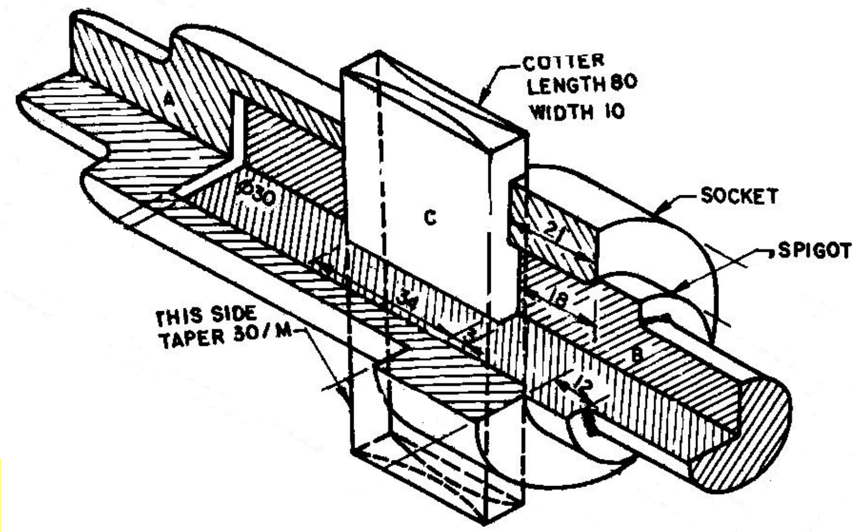
For circular rods



The enlarged ends of the rods butt against each other with a common sleeve over them

- The rod ends are enlarged to take care of the weakening effect caused by slots

Socket and Spigot Cotter Joint



Slots are wider than the cotter.

Cotter pulls the rod and socket tightly together

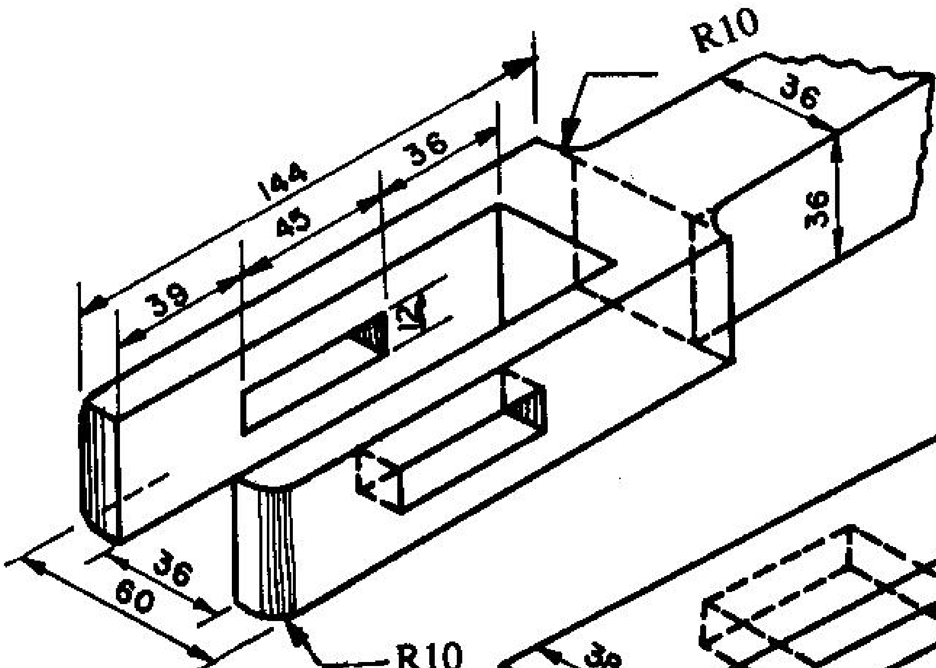
Clearance: must be provided for adjustment.(2 to 3 mm)

Proportions

cotter thickness = (1/3)diameter of rod

cotter width = rod diameter

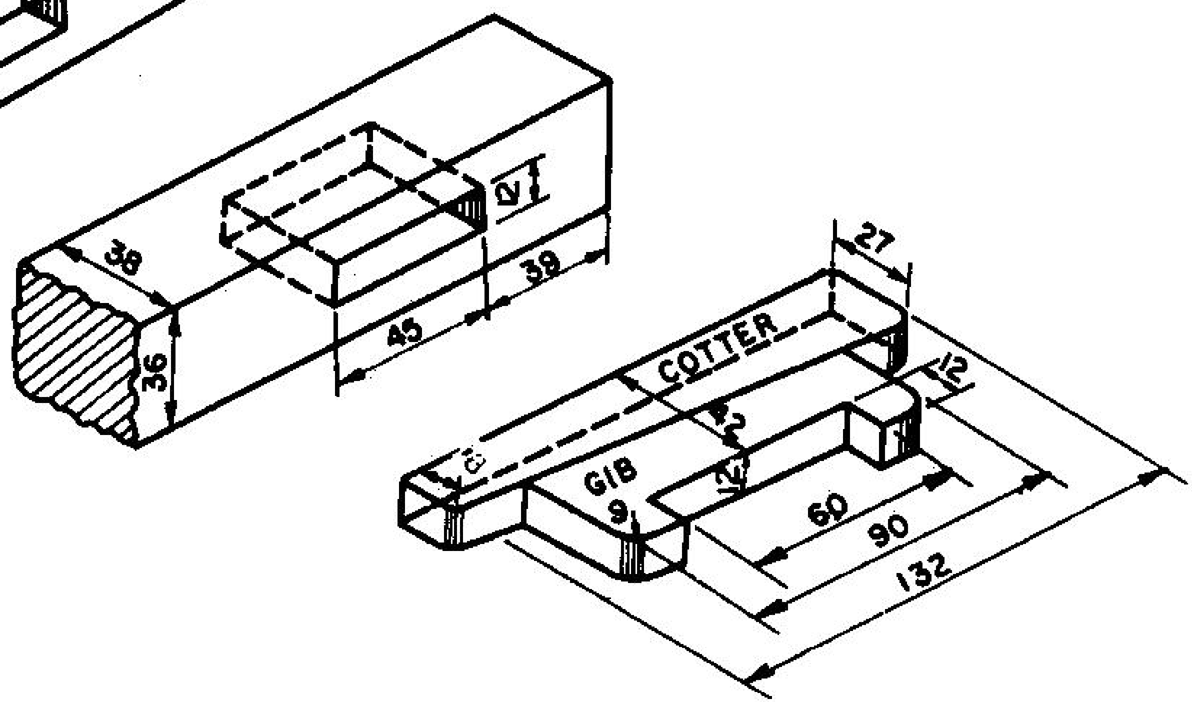
Gib and cotter joint for rectangular rods



One bar end is made in the form of a strap

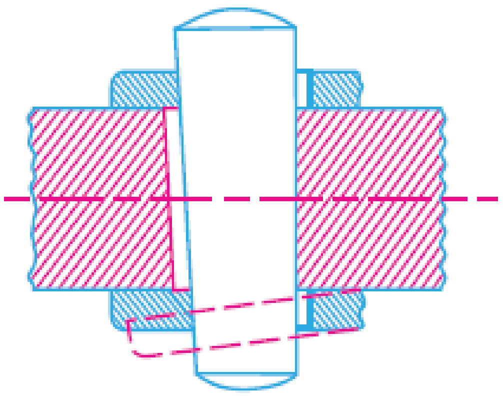
A Gib is used along with the cotter.

The thickness of the gib and cotter are same

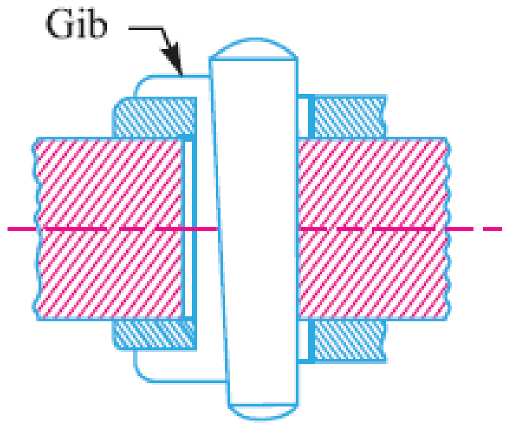


Gib and cotter joint for rectangular rods

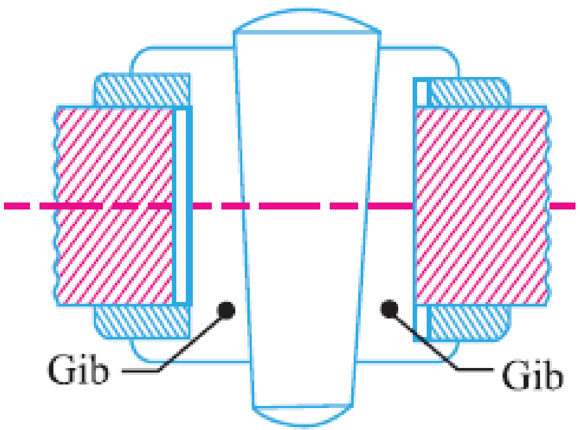
When the cotter alone (*i.e. without gib) is driven, the friction between its* ends and the inside of the slots in the strap tends to cause the sides of the strap to spring open (or spread) outwards as shown dotted in Fig.



(a) Cotter without gib.



(b) Cotter with one gib.



(c) Cotter with double gib.

Design of Socket and Spigot Cotter Joint

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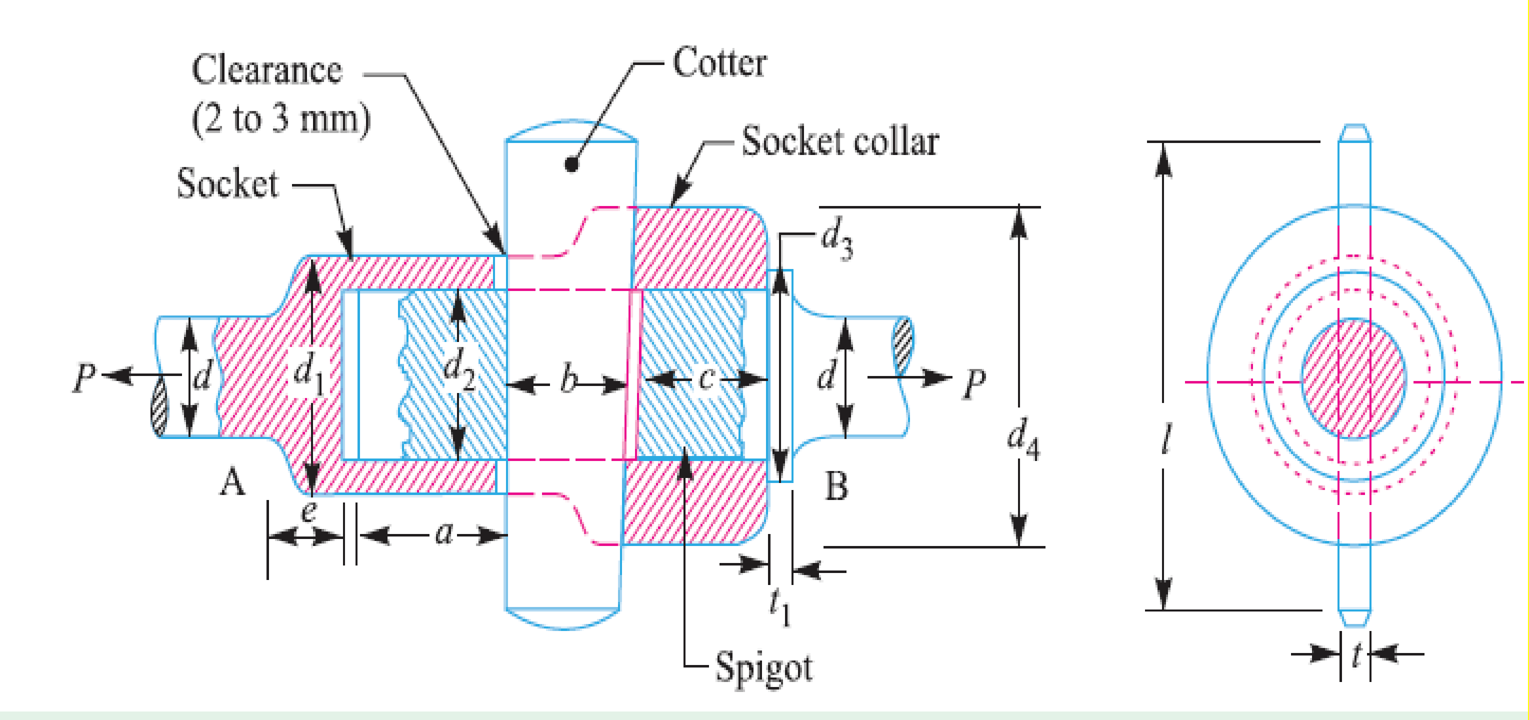
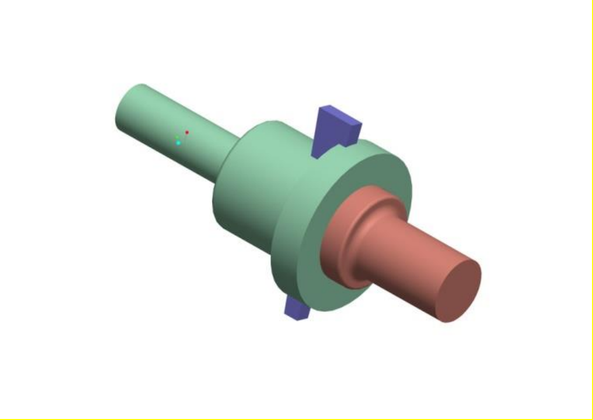


Design of Socket and Spigot joint

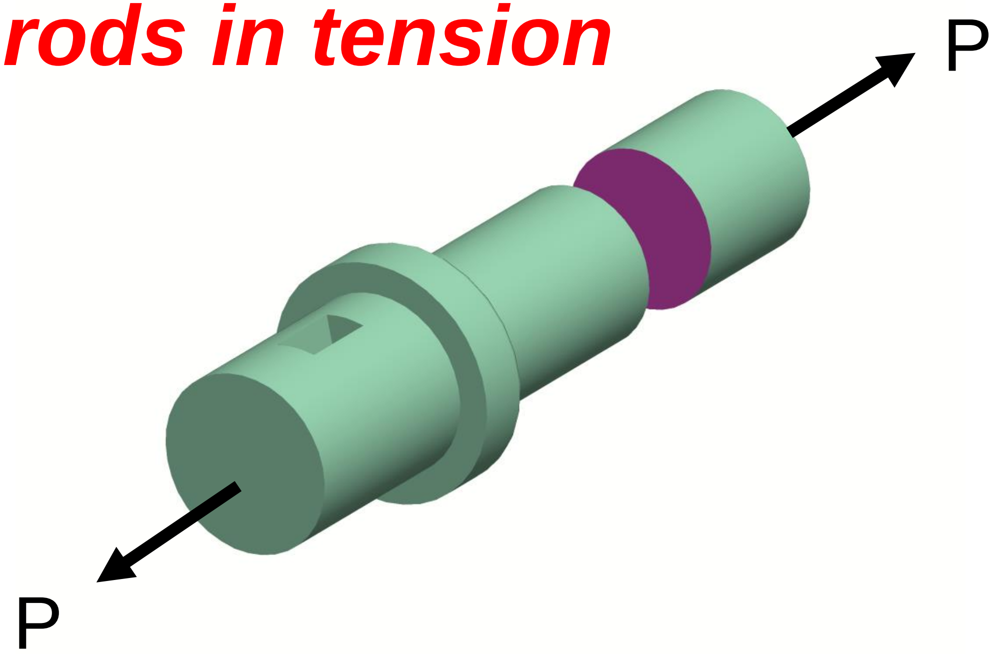
Let P = *Load carried by the rods,*

- d = *Diameter of the rods,*
- $d1$ = *Outside diameter of socket,*
- $d2$ = *Diameter of spigot or inside diameter of socket,*
- $d3$ = *Outside diameter of spigot collar,*
- $t1$ = *Thickness of spigot collar,*
- $d4$ = *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 rod,*
- σ_t = *Permissible tensile stress for the rods material*
- τ = Permissible shear stress for the cotter material
- σ_c = *Permissible crushing stress for the cotter material*



Design of Socket and Spigot Cotter Joint

1.Failure of the rods in tension



Area resisting tearing

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

$\therefore$ Tearing strength of the rod

$$= \frac{\pi}{4} \times d^2 \times \sigma_t$$

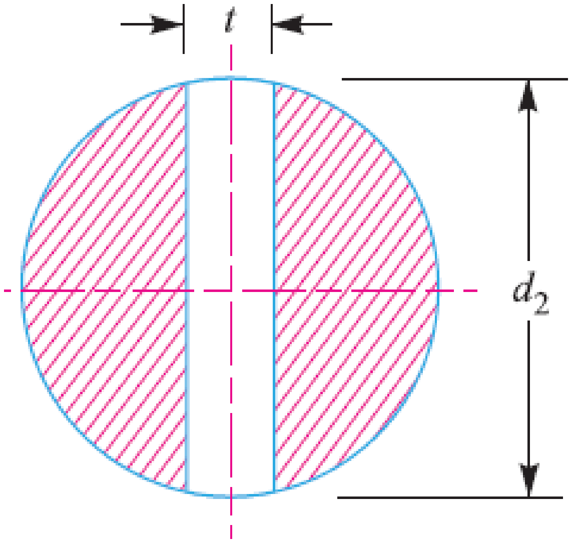
Equating this to load (P), we have

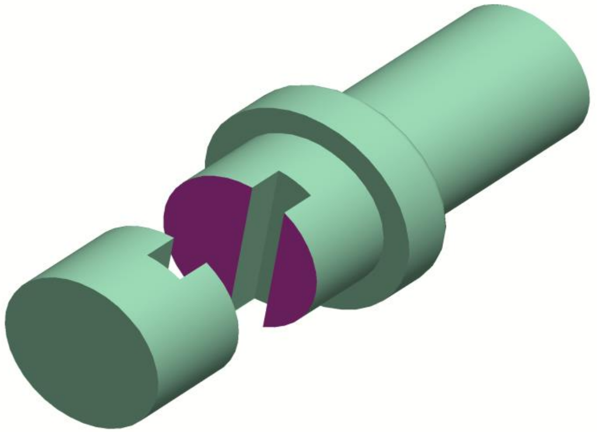
$$P = \frac{\pi}{4} \times d^2 \times \sigma_t$$

Design of Socket and Spigot Cotter Joint

2. *Failure of spigot in tension across the weakest section (or slot)*

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

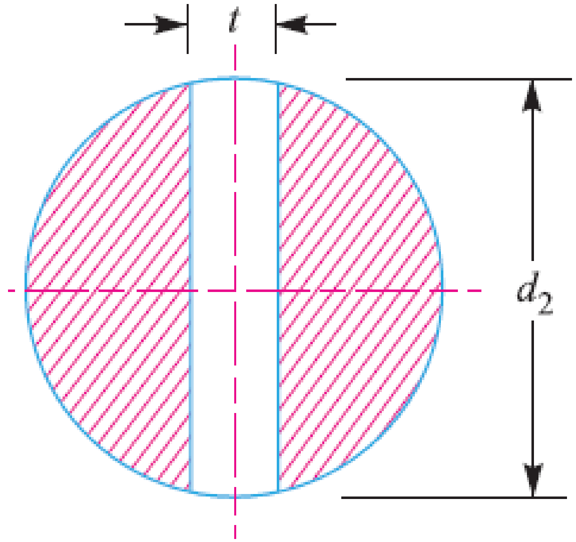


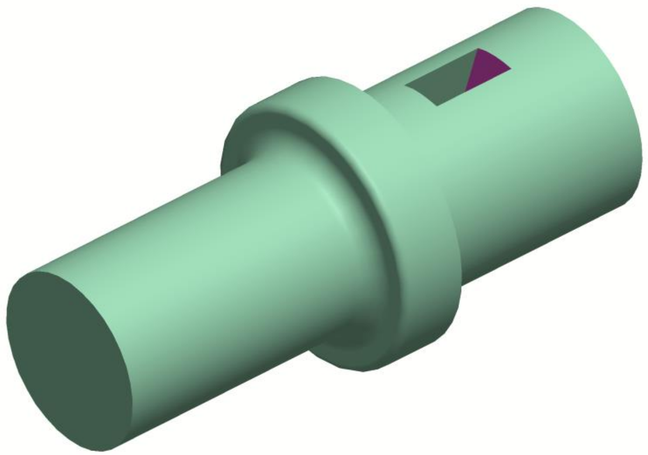


Design of Socket and Spigot Cotter Joint

3. *Failure of the rod or cotter in crushing*

$$P = d_2 \times t$$

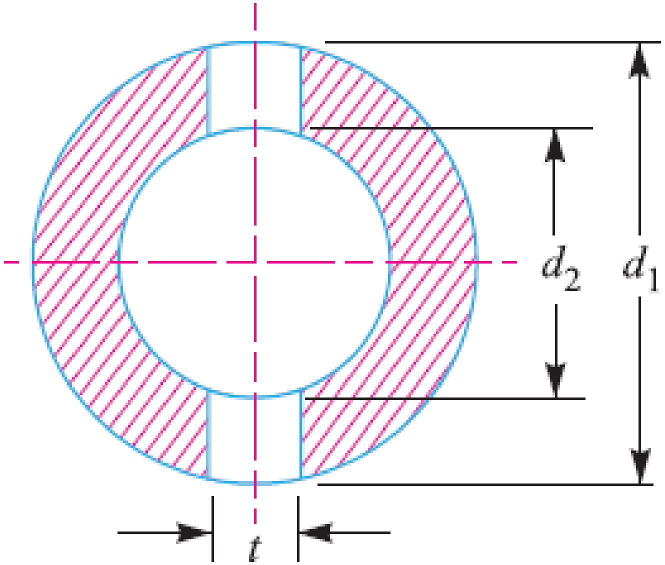


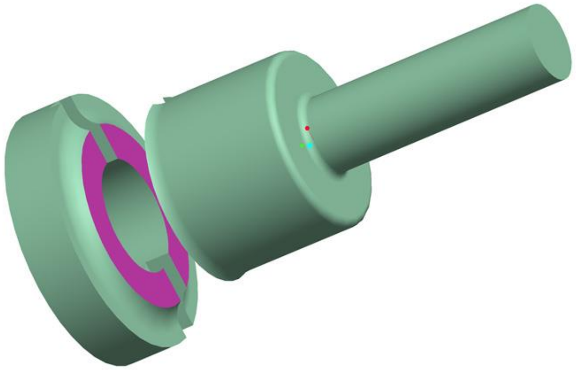


Design of Socket and Spigot Cotter Joint

4. *Failure of the socket in tension across the slot*

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

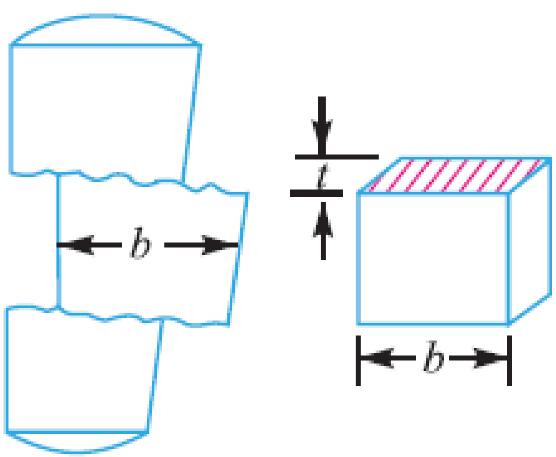


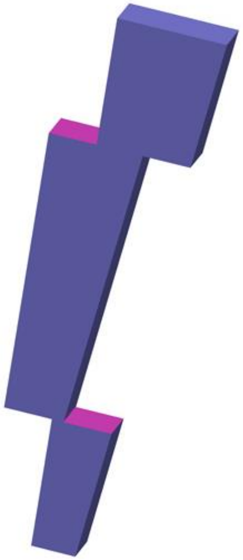


Design of Socket and Spigot Cotter Joint

5. *Failure of cotter in shear*

$$P = 2b \times t \times \tau$$

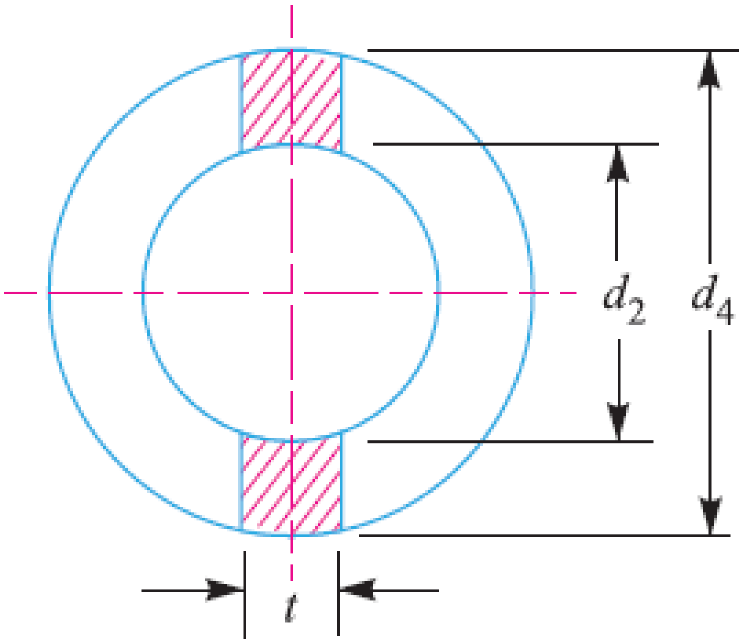


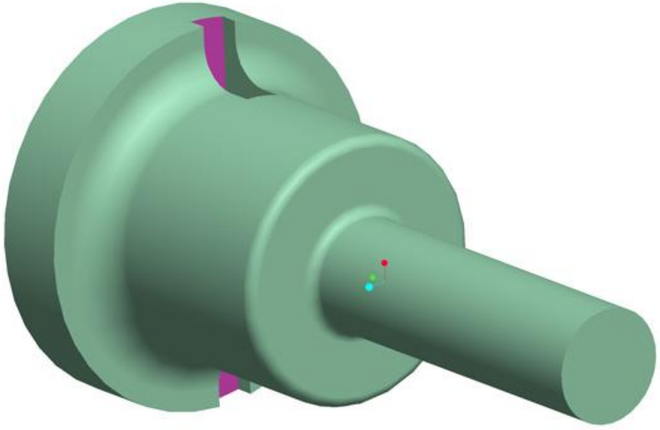


Design of Socket and Spigot Cotter Joint

6. *Failure of the socket collar in crushing*

$$P = (d_4 - d_2)\, t \times \sigma_c$$



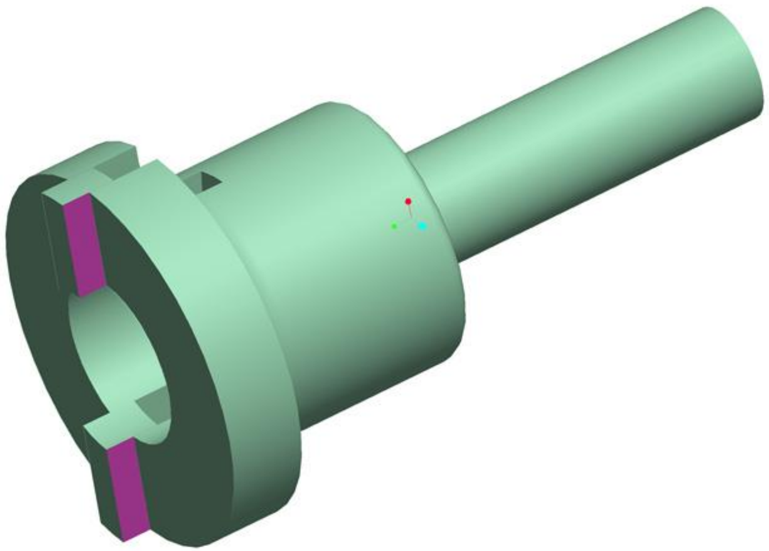


Design of Socket and Spigot Cotter Joint

7. *Failure of socket end in shearing*

$$P = 2\,(d_4 - d_2)\,c \times \tau$$

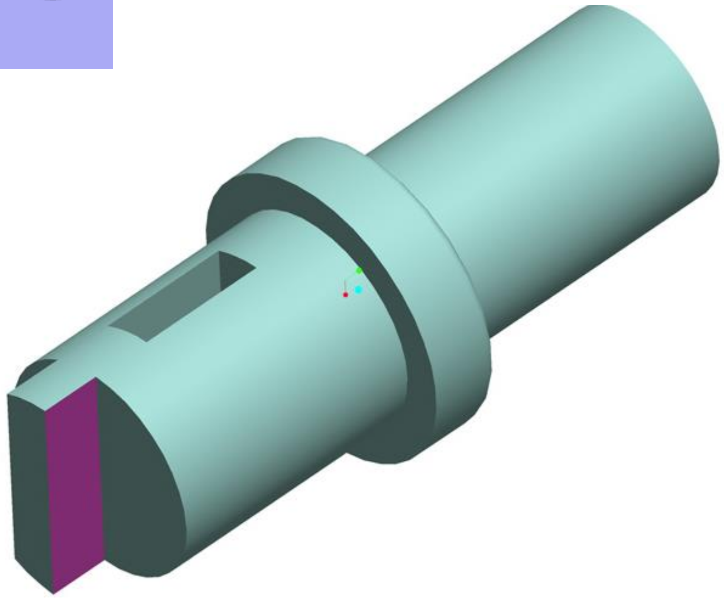
From this equation, the thickness of socket collar (c) may be obtained.



Design of Socket and Spigot Cotter Joint

8. *Failure of rod end in shear*

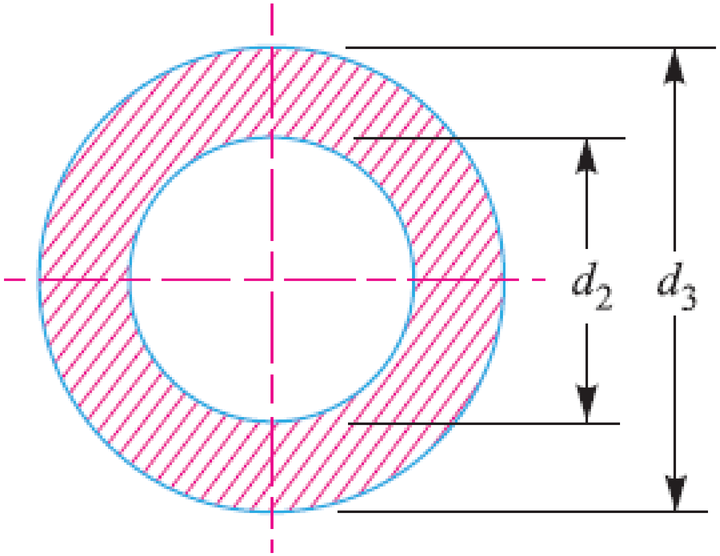
$$P = 2\,a \times d_2 \times \tau$$



Design of Socket and Spigot Cotter Joint

9.Failure of spigot collar in crushing

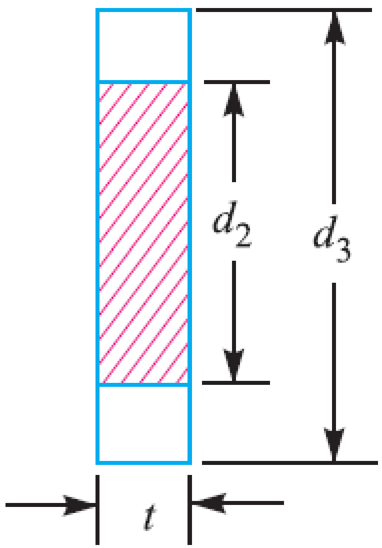
$$P = \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]\sigma_c$$

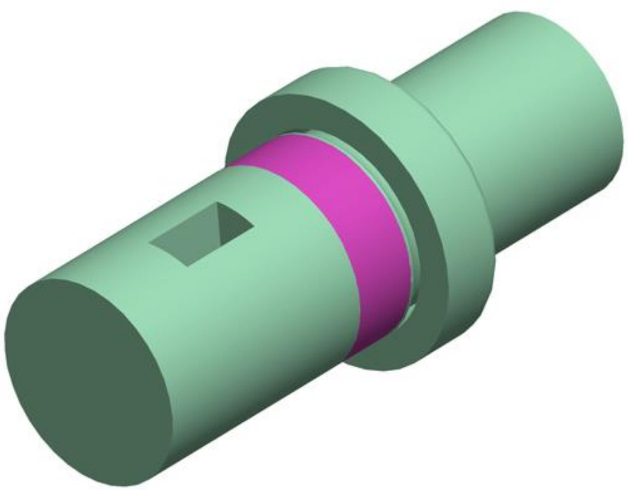


Design of Socket and Spigot Cotter Joint

10. *Failure of the spigot collar in shearing*

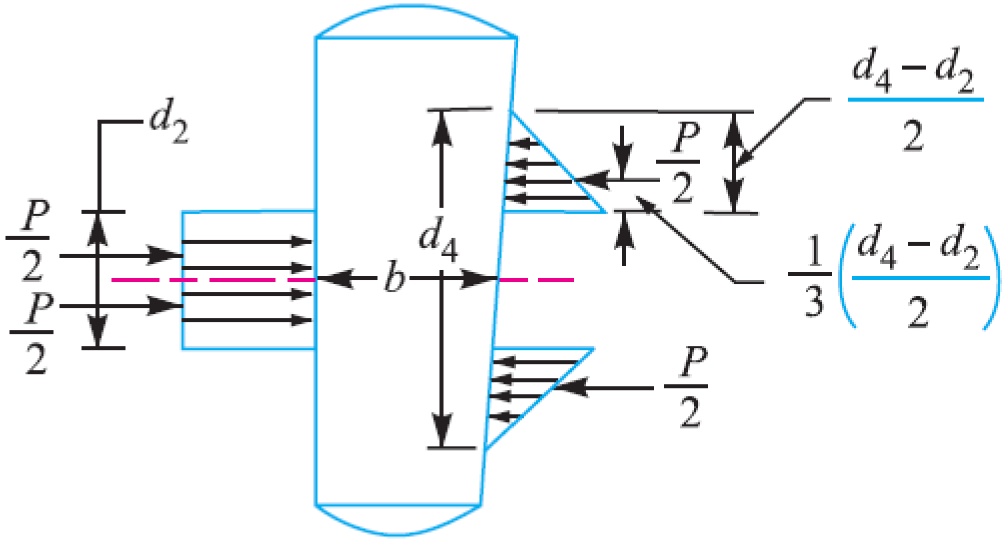
$$P = \pi d_2 \times t_1 \times \tau$$





Design of Socket and Spigot Cotter Joint

11. Failure of cotter in bending



The maximum bending moment occurs at the centre of the cotter and is given by

$$M_{max} = \frac{P}{2}\left(\frac{1}{3}\times\frac{d_4 - d_2}{2} + \frac{d_2}{2}\right) - \frac{P}{2}\times\frac{d_2}{4}$$

$$= \frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{2} - \frac{d_2}{4}\right) = \frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{4}\right)$$

We know that section modulus of the cotter,

$$Z = t \times b^2 / 6$$

∴ Bending stress induced in the cotter,

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

12. The length of cotter (l) is taken as 4 d.

13. The taper in cotter should not exceed 1 in 24. In case the greater taper is required, then a locking device must be provided.

14. The draw of cotter is generally taken as 2 to 3 mm.

Notes: 1. When all the parts of the joint are made of steel, the following proportions in terms of diameter of the rod (d) are generally adopted :

$d_1 = 1.75\,d$, $d_2 = 1.21\,d$, $d_3 = 1.5\,d$, $d_4 = 2.4\,d$, $a = c = 0.75\,d$, $b = 1.3\,d$, $l = 4\,d$, $t = 0.31\,d$, $t_1 = 0.45\,d$, $e = 1.2\,d$.

Taper of cotter = 1 in 25, and draw of cotter = 2 to 3 mm.

2. If the rod and cotter are made of steel or wrought iron, then $\tau = 0.8\,\sigma_t$ and $\sigma_c = 2\,\sigma_t$ may be taken.

Example 12.1. *Design and draw a cotter joint to support a load varying from 30 kN in compression to 30 kN in tension. The material used is carbon steel for which the following allowable stresses may be used. The load is applied statically.*

Tensile stress = compressive stress = 50 MPa ; shear stress = 35 MPa and crushing stress = 90 MPa.

Solution. Given : $P = 30 \text{ kN} = 30 \times 10^3 \text{ N}$; $\sigma_t = 50 \text{ MPa} = 50 \text{ N/mm}^2$; $\tau = 35 \text{ MPa} = 35 \text{ N/mm}^2$; $\sigma_c = 90 \text{ MPa} = 90 \text{ N/mm}^2$

1. *Diameter of the rods*

Let d = Diameter of the rods.

Considering the failure of the rod in tension. We know that load (P),

$$30 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 50 = 39.3\, d^2$$

$$\therefore \quad d^2 = 30 \times 10^3 / 39.3 = 763 \quad \text{or} \quad d = 27.6 \text{ say } 28 \text{ mm} \quad \textbf{Ans.}$$

2. *Diameter of spigot and thickness of cotter*

Let d_2 = Diameter of spigot or inside diameter of socket, and

t = Thickness of cotter. It may be taken as $d_2 / 4$.

Considering the failure of spigot in tension across the weakest section. We know that load (P),

$$30 \times 10^3 = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times \frac{d_2}{4}\right] 50 = 26.8\,(d_2)^2$$

$$\therefore \quad (d_2)^2 = 30 \times 10^3 / 26.8 = 1119.4 \quad \text{or} \quad d_2 = 33.4 \text{ say } 34 \text{ mm}$$

and thickness of cotter, $t = \dfrac{d_2}{4} = \dfrac{34}{4} = 8.5 \text{ mm}$

Let us now check the induced crushing stress. We know that load (P),

$$30 \times 10^3 = d_2 \times t \times \sigma_c = 34 \times 8.5 \times \sigma_c = 289\, \sigma_c$$

$$\therefore \quad \sigma_c = 30 \times 10^3 / 289 = 103.8 \text{ N/mm}^2$$

Since this value of σ_c is more than the given value of $\sigma_c = 90 \text{ N/mm}^2$, therefore the dimensions $d_2 = 34$ mm and $t = 8.5$ mm are not safe. Now let us find the values of d_2 and t by substituting the value of $\sigma_c = 90 \text{ N/mm}^2$ in the above expression, *i.e.*

$$30 \times 10^3 = d_2 \times \frac{d_2}{4} \times 90 = 22.5\,(d_2)^2$$

$$\therefore \quad (d_2)^2 = 30 \times 10^3 / 22.5 = 1333 \quad \text{or} \quad d_2 = 36.5 \text{ say } 40 \text{ mm} \quad \textbf{Ans.}$$

and $\quad t = d_2 / 4 = 40 / 4 = 10 \text{ mm}$ **Ans.**

3. *Outside diameter of socket*

Let d_1 = Outside diameter of socket.

Considering the failure of the socket in tension across the slot. We know that load (P),

$$30 \times 10^3 = \left[\frac{\pi}{4}\left\{(d_1)^2 - (d_2)^2\right\} - (d_1 - d_2)\, t\right]\sigma_t$$

$$= \left[\frac{\pi}{4}\left\{(d_1)^2 - (40)^2\right\} - (d_1 - 40)\, 10\right] 50$$

$$30 \times 10^3/50 = 0.7854\,(d_1)^2 - 1256.6 - 10\, d_1 + 400$$

or $(d_1)^2 - 12.7\, d_1 - 1854.6 = 0$

$$\therefore \quad d_1 = \frac{12.7 \pm \sqrt{(12.7)^2 + 4 \times 1854.6}}{2} = \frac{12.7 \pm 87.1}{2}$$

$= 49.9$ say 50 mm **Ans.** ...(Taking +ve sign)

4. *Width of cotter*

Let b = Width of cotter.

Considering the failure of the cotter in shear. Since the cotter is in double shear, therefore load (P),

$$30 \times 10^3 = 2\,b \times t \times \tau = 2\,b \times 10 \times 35 = 700\,b$$

$$\therefore \quad b = 30 \times 10^3 / 700 = 43 \text{ mm } \textbf{Ans.}$$

5. Diameter of socket collar

Let $\quad d_4$ = Diameter of socket collar.

Considering the failure of the socket collar and cotter in crushing. We know that load (P),

$$30 \times 10^3 = (d_4 - d_2)\,t \times \sigma_c = (d_4 - 40)10 \times 90 = (d_4 - 40)\,900$$

$$\therefore \quad d_4 - 40 = 30 \times 10^3 / 900 = 33.3 \text{ or } d_4 = 33.3 + 40 = 73.3 \text{ say } 75 \text{ mm } \textbf{Ans.}$$

6. Thickness of socket collar

Let $\quad c$ = Thickness of socket collar.

Considering the failure of the socket end in shearing. Since the socket end is in double shear, therefore load (P),

$$30 \times 10^3 = 2(d_4 - d_2)\,c \times \tau = 2\,(75 - 40)\,c \times 35 = 2450\,c$$

$$\therefore \quad c = 30 \times 10^3 / 2450 = 12 \text{ mm } \textbf{Ans.}$$

7. Distance from the end of the slot to the end of the rod

Let $\quad a$ = Distance from the end of slot to the end of the rod.

Considering the failure of the rod end in shear. Since the rod end is in double shear, therefore load (P),

$$30 \times 10^3 = 2\,a \times d_2 \times \tau = 2a \times 40 \times 35 = 2800\,a$$

$$\therefore \quad a = 30 \times 10^3 / 2800 = 10.7 \text{ say } 11 \text{ mm } \textbf{Ans.}$$

8. *Diameter of spigot collar*

Let $\quad d_3$ = Diameter of spigot collar.

Considering the failure of spigot collar in crushing. We know that load (P),

$$30 \times 10^3 = \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]\sigma_c = \frac{\pi}{4}\left[(d_3)^2 - (40)^2\right]90$$

or
$$(d_3)^2 - (40)^2 = \frac{30 \times 10^3 \times 4}{90 \times \pi} = 424$$

$$\therefore \quad (d_3)^2 = 424 + (40)^2 = 2024 \quad \text{or} \quad d_3 = 45 \text{ mm}$$ **Ans.**

9. *Thickness of spigot collar*

Let $\quad t_1$ = Thickness of spigot collar.

Considering the failure of spigot collar in shearing. We know that load (P),

$$30 \times 10^3 = \pi\, d_2 \times t_1 \times \tau = \pi \times 40 \times t_1 \times 35 = 4400\, t_1$$

$$\therefore \quad t_1 = 30 \times 10^3 / 4400 = 6.8 \text{ say } 8 \text{ mm}$$ **Ans.**

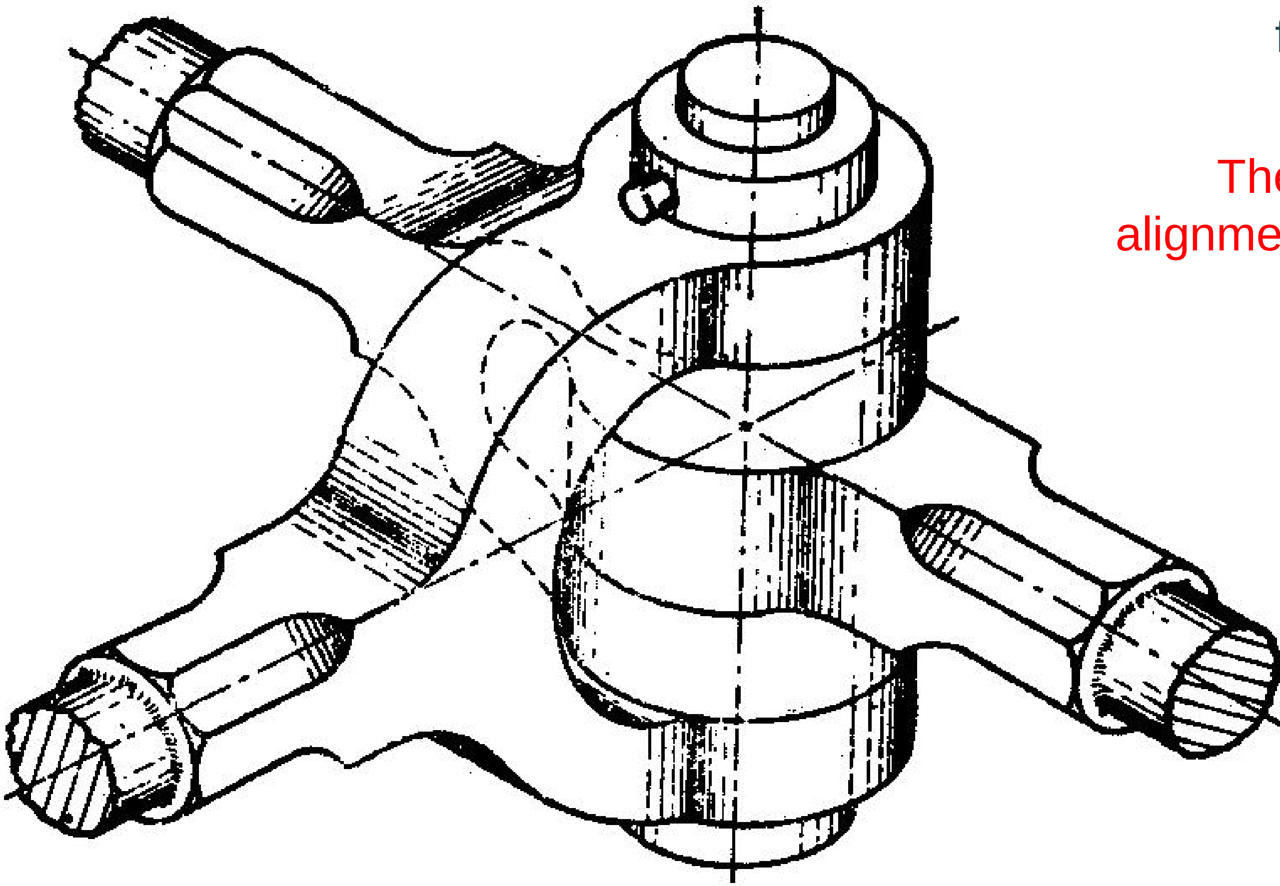
10. The length of cotter (l) is taken as 4 d.

$$\therefore \quad l = 4\,d = 4 \times 28 = 112 \text{ mm}$$ **Ans.**

11. The dimension e is taken as 1.2 d.

$$\therefore \quad e = 1.2 \times 28 = 33.6 \text{ say } 34 \text{ mm}$$ **Ans.**

Knuckle joint



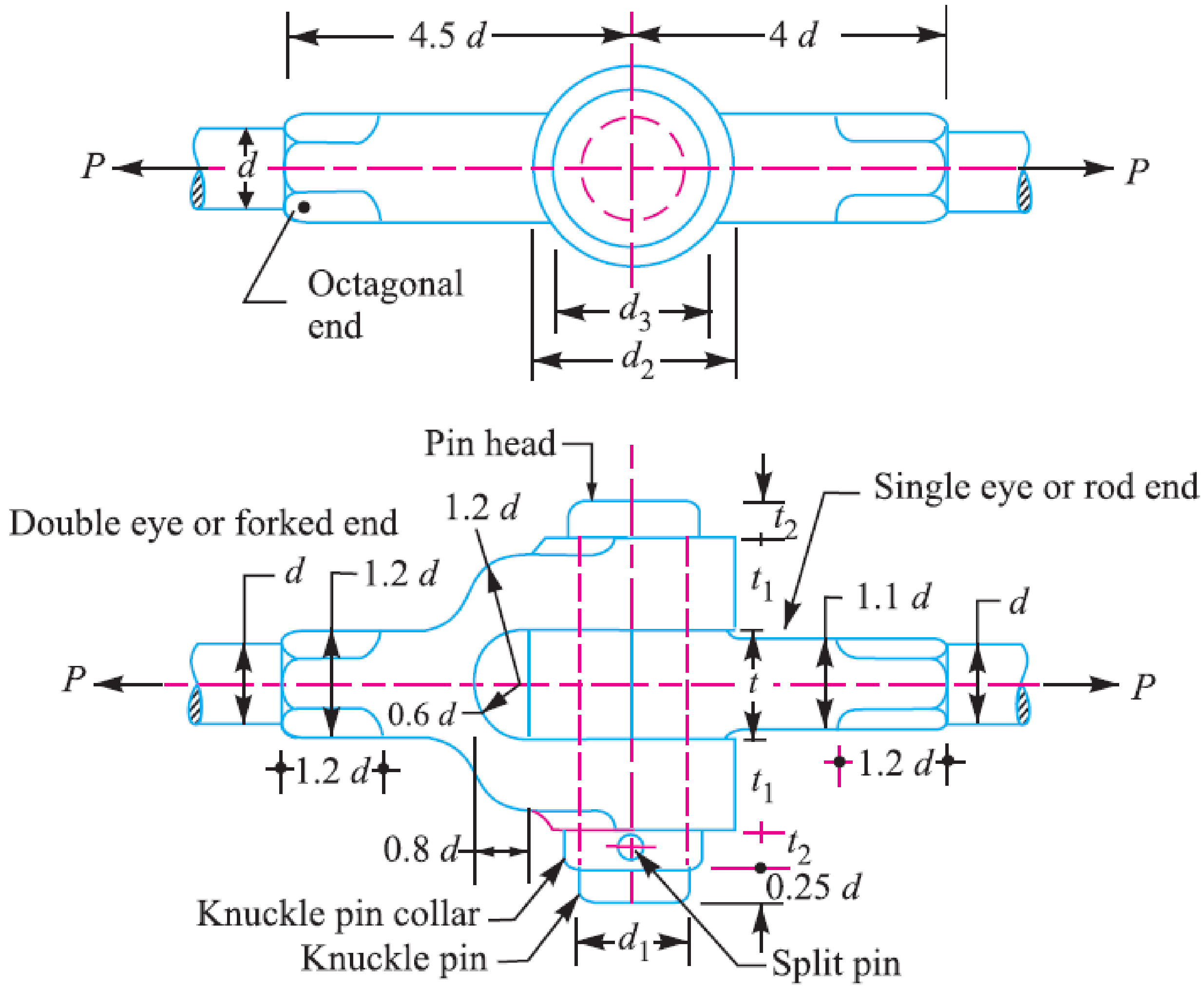
Two or more rods subjected to tensile and compressive forces are fastened together

Their axes are not in alignments but meet in a point

The joint allows a small angular moment of one rod relative to another

It can be easily connected and disconnected

Applications: Elevator chains, valve rods, etc


4.5 d
4 d
P
d
P
Octagonal end
d3
d2
Pin head
Single eye or rod end
1.2 d
t2
Double eye or forked end
d
1.2 d
t1
1.1 d
d
P
t
P
0.6 d
1.2 d
1.2 d
t1
t2
0.8 d
0.25 d
Knuckle pin collar
Knuckle pin
d1
Split pin

The dimensions of various parts of the knuckle joint are fixed by empirical relations as given below. It may be noted that all the parts should be made of the same material *i.e.* mild steel or wrought iron.

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,

$$t = 1.25\,d$$

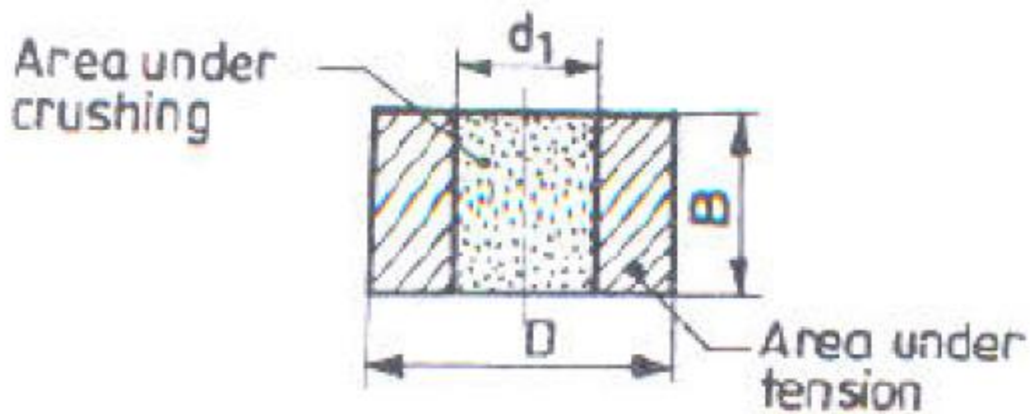
Thickness of fork, $t_1 = 0.75\,d$

Thickness of pin head, $t_2 = 0.5\,d$

1. Tension failure of the rod, across the section of diameter, d

$$F = \frac{\pi d^2}{4} \tau$$

2. Tension failure of the eye (fig.1)



$$F = (D-d_1)\, B\, \sigma_t$$

3. Tension failure of the fork (fig.2)

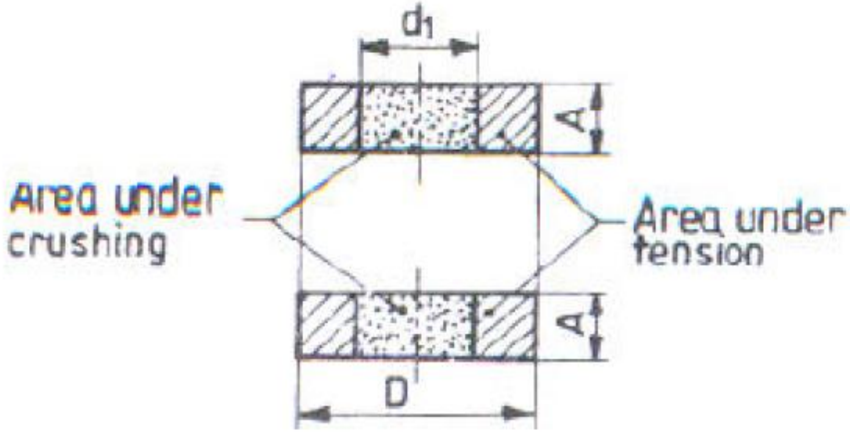


Fig.2

$$F = 2\,(D - d_1)\,A\,\sigma_t$$

4. Shear failure of the eye (Fig.3)

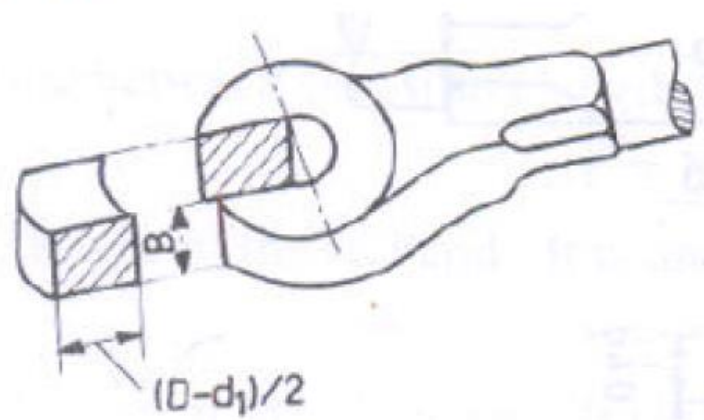


Fig.3

$$F = (D - d_1)\,B\,\tau$$

5. Shear failure of the fork (Fig.4)

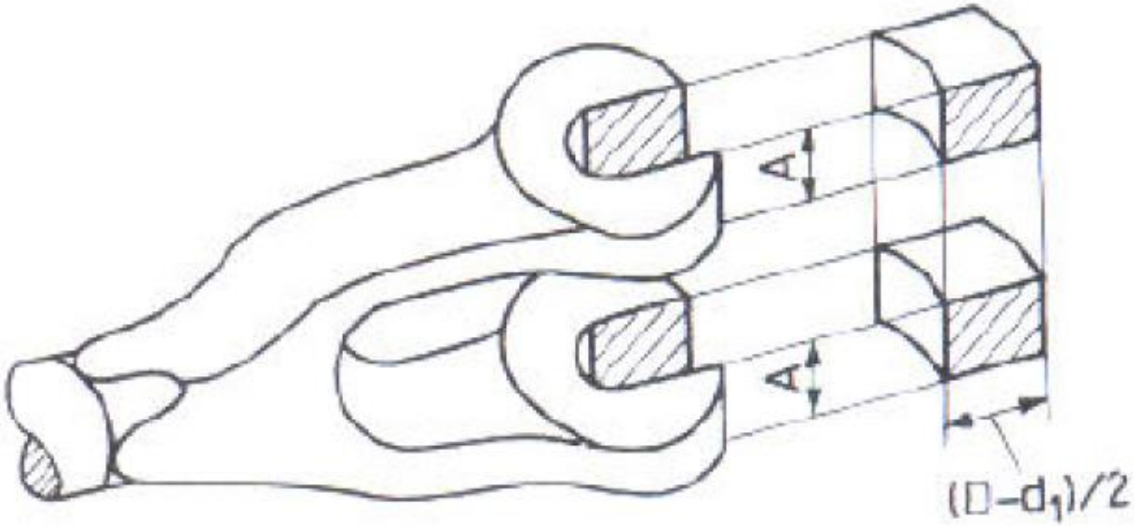


Fig.4

$$F = 2\,(D\text{-}d_1)\,A\,\tau$$

6. Shear failure of the pin. It is under double shear.

$$F = 2x\frac{\pi}{4}d^2 x\tau$$

7. Crushing between the pin and eye (fig.1)

$$F = d_1 B \sigma_c$$

8. Crushing between the pin and fork (fig.2)

$$F = 2 d_1 A \sigma_c$$

$$M = \frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{2}\right) - \frac{P}{2} \times \frac{t}{4}$$

$$= \frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{2} - \frac{t}{4}\right)$$

$$= \frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{4}\right)$$

and section modulus, $Z = \frac{\pi}{32}(d_1)^3$

$\therefore$ Maximum bending (tensile) stress,

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

From this expression, the value of d_1 may be obtained.

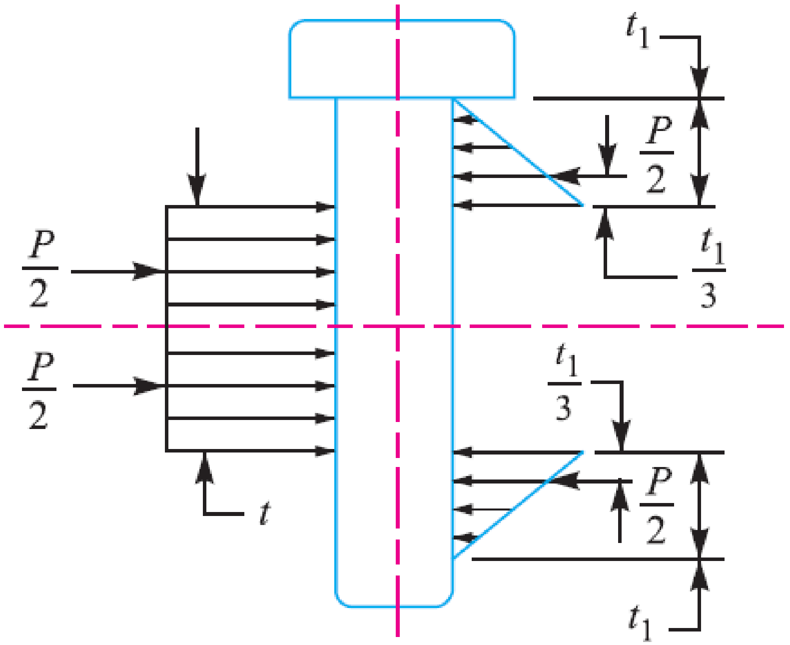


Fig. 12.17. Distribution of load on the pin.

Example 12.7. *Design a knuckle joint to transmit 150 kN. The design stresses may be taken 75 MPa in tension, 60 MPa in shear and 150 MPa in compression.*

Solution. Given : $P = 150$ kN $= 150 \times 10^3$ N ; $\sigma_t = 75$ MPa $= 75$ N/mm^2 ; $\tau = 60$ MPa $= 60$ N/mn
$\sigma_c = 150$ MPa $= 150$ N/mm^2

The knuckle joint is shown in Fig. 12.16. The joint is designed by considering the varic methods of failure as discussed below :

1. *Failure of the solid rod in tension*

Let d = Diameter of the rod.

We know that the load transmitted (P),

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

$$\therefore \quad d^2 = 150 \times 10^3 / 59 = 2540 \quad \text{or} \quad d = 50.4 \text{ say } 52 \text{ mm} \quad \textbf{Ans.}$$

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),

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

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

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),

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

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

From above, we see that the induced stresses are less than the given design stresses, therefore the joint is safe.


Thank
you!