Unit –III
Design of Joints

Introduction

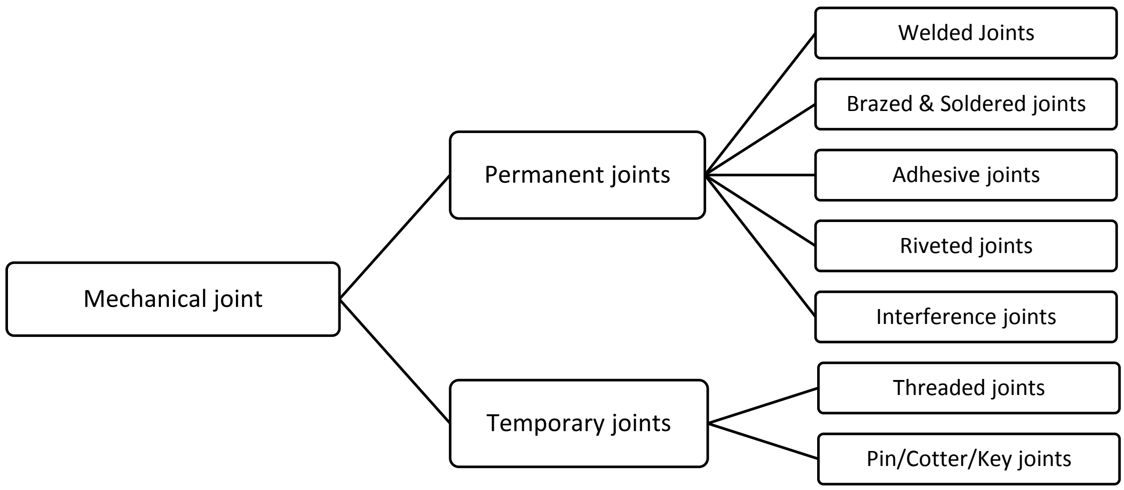
Mechanical joints or fasteners are used for making connections between different elements of machine or structure. A machine or a structure is made of a large number of parts and they need be joined suitably for the machine to operate satisfactorily. In manufacturing industries, joining of two or more components is necessary for assembly purposes. Joining makes the production system more reliable, efficient and profitable. In fact, joining can be defined as one of the manufacturing processes by which two or more solid components can be assembled together.

Types of joints

Mechanical joints are broadly classified into following two categories:

1. **Permanent joints**
   - Permanent joints can not be easily disassembled without damaging the connecting elements.
   - Different types of permanent joints are welded joints, brazed joints, soldered joints, adhesive joints, riveted joints and interference joints.

2. **Temporary or detachable joints**
   - Temporary joints can be easily disassembled without damaging the connecting elements.
   - Different types of detachable joints are threaded joints, pin joints, cotter joints and key joints.



Difference between permanent and temporary joints

Permanent Joint	Temporary Joint
Permanent joints don’t allow dismantling of assembled components without rupturing them.	Temporary joints allow easy dismantling of assembled components without breaking them.
Permanent joints are usually leak-proof.	Temporary joints are not necessarily leak-proof.
Strength of permanent joint is high. Usually joint strength is same with that of the components.	Strength of temporary joint is comparatively less.
As permanent joints cannot be disassembled easily, so inspection is difficult and costly. Often destructive testing is carried out, which damages the assembled structures.	It facilitates fast, easy and cost efficient inspection. No destructive testing is required for inspection of joints.
Repair and replacement are difficult and costly.	Repair and replacement are also easy.
Permanent joints are suitable for such applications where separation is usually not desired in the service life.	Temporary joints are suitable where frequent separation of assembled components is required.
Examples of various permanent joints: • Welding • Brazing and soldering • Riveting • Adhesive joining	Examples of various temporary joints: • Fasteners • Press fit • Cotter joints • Knuckle joints, etc.

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 force.It is usually used in connecting a piston rod to the cross head of a reciprocating steam engine.

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 Fig. 12.1 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. The compressive load is taken up by the collar on the spigot.

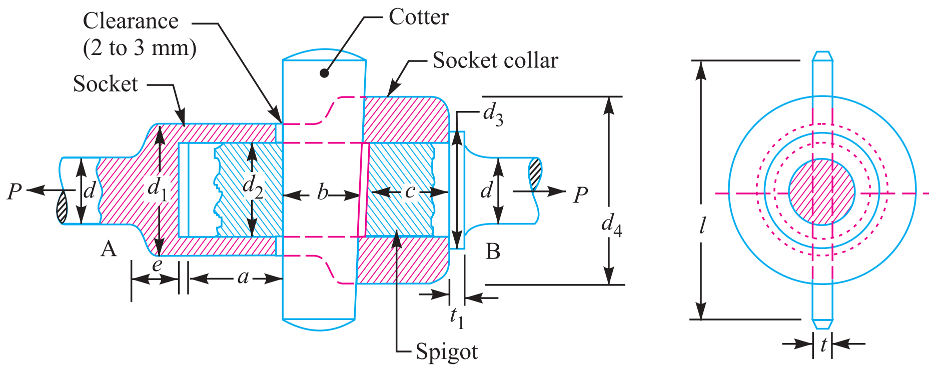


Fig. 12.1. Socket and spigot cotter joint.

12.4 Design of Socket and Spigot Cotter Joint

The socket and spigot cotter joint is shown in Fig. 12.1.

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 rod,

σ_t = Permissible tensile stress for the rods material,

τ = Permissible shear stress for the cotter material, and

σ_c = Permissible crushing stress for the cotter material.

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

1. *Failure of the rods in tension*

The rods may fail in tension due to the tensile load *P*. We know that

Area resisting tearing

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

∴ Tearing strength of the rods,

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

Equating this to load (P), we have

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

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

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

Since the weakest section of the spigot is that section which has a slot in it for the cotter, as shown in Fig. 12.2, therefore

Area resisting tearing of the spigot across the slot

$$= \frac{\pi}{4} (d_2)^2 - d_2 \times t$$

and tearing strength of the spigot across the slot

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

Equating this to load (P), we have

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

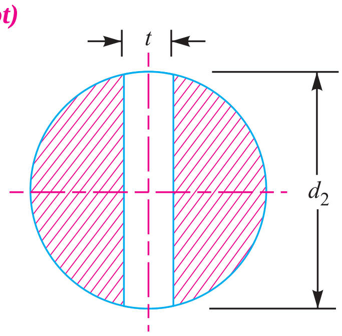


Fig. 12.2

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 $d_2 / 4$.

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

We know that the area that resists crushing of a rod or cotter

$$= d_2 \times t$$

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

Equating this to load (P), we have

$$P = d_2 \times t \times \sigma_c$$

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

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

We know that the resisting area of the socket across the slot, as shown in Fig. 12.3

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

$\therefore$ Tearing strength of the socket across the slot

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

Equating this to load (P), we have

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

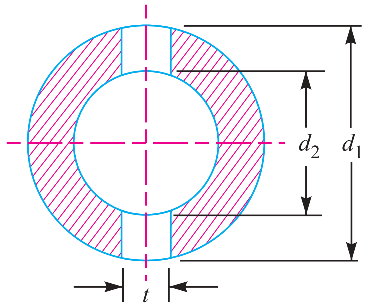


Fig. 12.3

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

5. *Failure of cotter in shear*

Considering the failure of cotter in shear as shown in Fig. 12.4. Since the cotter is in double shear, therefore shearing area of the cotter

$$= 2\ b \times t$$

and shearing strength of the cotter

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

Equating this to load (P), we have

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

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

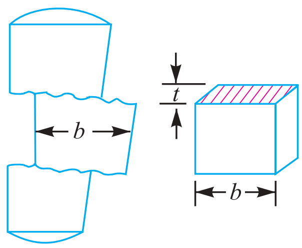


Fig. 12.4

6. *Failure of the socket collar in crushing*

Considering the failure of socket collar in crushing as shown in Fig. 12.5.

We know that area that resists crushing of socket collar

$$= (d_4 - d_2)\ t$$

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

Equating this to load (P), we have

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

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

7. *Failure of socket end in shearing*

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

$$= 2\ (d_4 - d_2)\ c$$

and shearing strength of socket collar

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

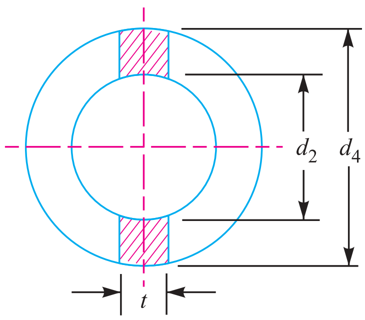


Fig. 12.5

Equating this to load (P), we have

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

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

8. *Failure of rod end in shear*

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

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

and shear strength of the rod end

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

Equating this to load (P), we have

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

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

9. *Failure of spigot collar in crushing*

Considering the failure of the spigot collar in crushing as shown in Fig. 12.6. We know that area that resists crushing of the collar

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

and crushing strength of the collar

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

Equating this to load (P), we have

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

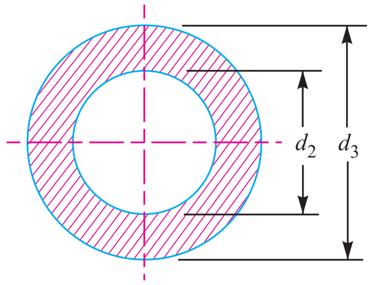


Fig. 12.8

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

10. *Failure of the spigot collar in shearing*

Considering the failure of the spigot collar in shearing as shown in Fig. 12.7. We know that area that resists shearing of the collar

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

and shearing strength of the collar,

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

Equating this to load (P) we have

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

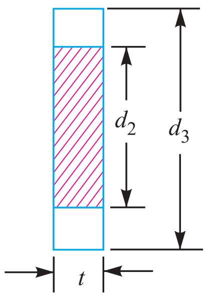


Fig. 12.7

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

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. 12.8.

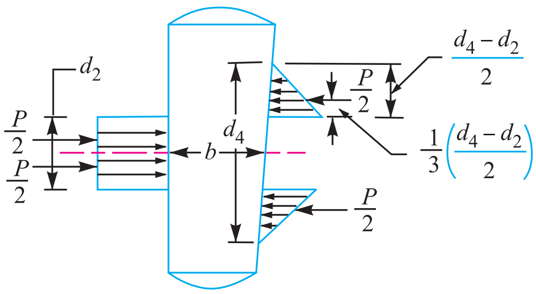


Fig. 12.6

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,

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

$\therefore$ Bending stress induced in the cotter,

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

This bending stress induced in the cotter should be less than the allowable bending stress of the cotter.

12. The length of cotter (l) in 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 : $P = 30\text{ kN} = 30 \times 10^3\text{ N}$; $\sigma_t = 50\text{ MPa} = 50\text{ N/mm}^2$; $\tau = 35\text{ MPa} = 35\text{ N/mm}^2$; $\sigma_c = 90\text{ MPa} = 90\text{ N/mm}^2$

The cotter joint is shown in Fig. 12.1. The joint is designed as discussed below :

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 $t = d_2 / 4 = 40 / 4 = 10\text{ mm}$ **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),

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

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

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

$$\text{or} \quad (d_1)^2 - 12.7\, d_1 - 1854.6 = 0$$

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

$= 49.9$ 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 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 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 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 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

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

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

9. Thickness of spigot collar

Let t_1 = Thickness of spigot collar.

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

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

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

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

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

11. The dimension e is taken as 1.2 d.

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

12.12 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. A knuckle joint may be readily disconnected for adjustments or repairs. Its use may be found 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 lever and rod connections of various types.

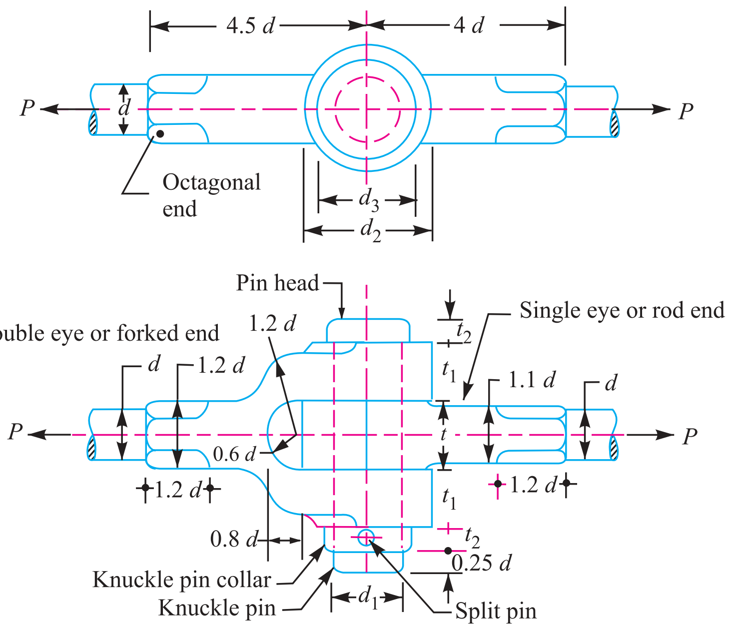
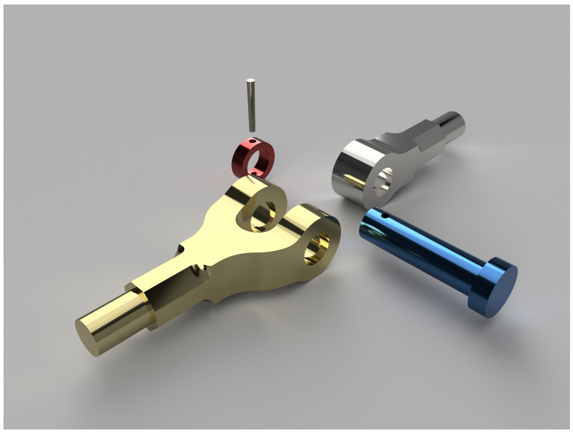


Fig. 12.16. Kunckle joint.



In knuckle joint (the two views of which are shown in Fig. 12.16), 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.

12.13 Dimensions of Various Parts of the Knuckle Joint

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,

$$d_2 = 2d$$

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, $t_2 = 0.5\,d$

Other dimensions of the joint are shown in Fig. 12.16.

12.14 Methods of Failure of Knuckle Joint

Consider a knuckle joint as shown in Fig. 12.16.

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.

In determining the strength of the joint for the various methods of failure, it is assumed that

1. There is no stress concentration, and
2. The load is uniformly distributed over each part of the joint.

Due to these assumptions, the strengths are approximate, however they serve to indicate a well proportioned joint. Following are the various methods of failure of the joint :

1. *Failure of the solid rod in tension*

Since the rods are subjected to direct tensile load, therefore tensile strength of the rod,

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

Equating this to the load (P) acting on the rod, we have

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

From this equation, diameter of the rod (d) is obtained.

2. *Failure of the knuckle pin in shear*

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

$$= 2 \times \frac{\pi}{4}\,(d_1)^2$$

and the shear strength of the pin

$$= 2 \times \frac{\pi}{4}\,(d_1)^2\ \tau$$

Equating this to the load (P) acting on the rod, we have

$$P = 2 \times \frac{\pi}{4}\,(d_1)^2\ \tau$$

From this equation, diameter of the knuckle pin (d_1) is obtained. This assumes that there is no slack and clearance between the pin and the fork and hence there is no bending of the pin. But, in actual practice, the knuckle pin is loose in forks in order to permit angular movement of one with respect to the other, therefore the pin is subjected to bending in addition to shearing. By making the diameter of knuckle pin equal to the diameter of the rod (*i.e.*, $d_1 = d$), a margin of strength is provided to allow for the bending of the pin.

In case, the stress due to bending is taken into account, it is assumed that the load on the pin is uniformly distributed along the middle portion (*i.e.* the eye end) and varies uniformly over the forks as shown in Fig. 12.17. Thus in the forks, a load $P/2$ acts through a distance of $t_1 / 3$ from the inner edge and the bending moment will be maximum at the centre of the pin. The value of maximum bending moment is given by

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

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

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

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

∴ Maximum bending (tensile) stress,

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

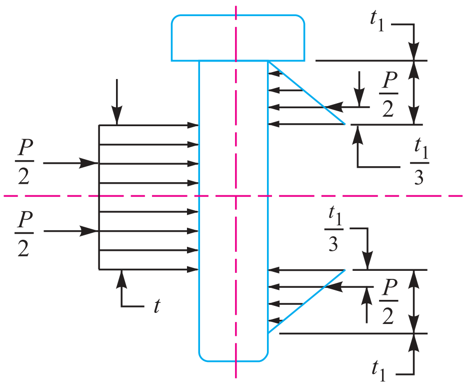


Fig. 12.17. Distribution of load on the pin.

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

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

The single eye or rod end may tear off due to the tensile load. We know that area resisting tearing $= (d_2 - d_1)\, t$

∴ Tearing strength of single eye or rod end

$$= (d_2 - d_1)\, t \times \sigma_t$$

Equating this to the load (P) we have

$$P = (d_2 - d_1)\, t \times \sigma_t$$

From this equation, the induced tensile stress (σ_t) 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).

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

The single eye or rod end may fail in shearing due to tensile load. We know that area resisting shearing $= (d_2 - d_1)\, t$

∴ Shearing strength of single eye or rod end

$$= (d_2 - d_1)\, t \times \tau$$

Equating this to the load (P), we have

$$P = (d_2 - d_1)\, t \times \tau$$

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

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

The single eye or pin may fail in crushing due to the tensile load. We know that area resisting crushing $= d_1 \times t$

$\therefore$ Crushing strength of single eye or rod end

$$= d_1 \times t \times \sigma_c$$

Equating this to the load (P), we have

$$\therefore \quad P = d_1 \times t \times \sigma_c$$

From this equation, the induced crushing stress (σ_c) for the single eye or pin may be checked. In case the induced crushing stress in more than the allowable working stress, then increase the thickness of the single eye (t).

6. *Failure of the forked end in tension*

The forked end or double eye may fail in tension due to the tensile load. We know that area resisting tearing

$$= (d_2 - d_1) \times 2\, t_1$$

$\therefore$ Tearing strength of the forked end

$$= (d_2 - d_1) \times 2\, t_1 \times \sigma_t$$

Equating this to the load (P), we have

$$P = (d_2 - d_1) \times 2t_1 \times \sigma_t$$

From this equation, the induced tensile stress for the forked end may be checked.

7. *Failure of the forked end in shear*

The forked end may fail in shearing due to the tensile load. We know that area resisting shearing

$$= (d_2 - d_1) \times 2t_1$$

$\therefore$ Shearing strength of the forked end

$$= (d_2 - d_1) \times 2t_1 \times \tau$$

Equating this to the load (P), we have

$$P = (d_2 - d_1) \times 2t_1 \times \tau$$

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

8. *Failure of the forked end in crushing*

The forked end or pin may fail in crushing due to the tensile load. We know that area resisting crushing $= d_1 \times 2\, t_1$

$\therefore$ Crushing strength of the forked end

$$= d_1 \times 2\, t_1 \times \sigma_c$$

Equating this to the load (P), we have

$$P = d_1 \times 2\, t_1 \times \sigma_c$$

From this equation, the induced crushing stress for the forked end may be checked.

Note: From the above failures of the joint, we see that the thickness of fork (t_1) should be equal to half the thickness of single eye ($t / 2$). But, in actual practice $t_1 > t / 2$ in order to prevent deflection or spreading of the forks which would introduce excessive bending of pin.

12.15 Design Procedure of Knuckle Joint

The empirical dimensions as discussed in Art. 12.13 have been formulated after wide experience on a particular service. These dimensions are of more practical value than the theoretical analysis. Thus, a designer should consider the empirical relations in designing a knuckle joint. The following procedure may be adopted :

1. First of all, find the diameter of the rod by considering the failure of the rod in tension. We know that tensile load acting on the rod,

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

where d = Diameter of the rod, and

σ_t = Permissible tensile stress for the material of the rod.

2. After determining the diameter of the rod, the diameter of pin (d_1) may be determined by considering the failure of the pin in shear. We know that load,

$$P = 2 \times \frac{\pi}{4} (d_1)^2 \ \tau$$

A little consideration will show that the value of d_1 as obtained by the above relation is less than the specified value (*i.e.* the diameter of rod). So fix the diameter of the pin equal to the diameter of the rod.

3. Other dimensions of the joint are fixed by empirical relations as discussed in Art. 12.13.

4. The induced stresses are obtained by substituting the empirical dimensions in the relations as discussed in Art. 12.14.

In case the induced stress is more than the allowable stress, then the corresponding dimension may be increased.

Example 12.7. *Design a knuckle joint to transmit 150 kN. 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 $= 150 \times 10^3$ N ; $\sigma_t = 75$ MPa $= 75$ N/mm^2 ; $\tau = 60$ MPa $= 60$ N/mm^2 ; $\sigma_c = 150$ MPa $= 150$ N/mm^2

The knuckle joint is shown in Fig. 12.16. The joint is designed by considering the various 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),

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

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

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*),

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

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

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*),

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

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

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*),

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

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

6. *Failure of the forked end in tension*

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

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

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

7. *Failure of the forked end in shear*

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

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

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

8. *Failure of the forked end in crushing*

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

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

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

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

Example 12.8. *Design a knuckle joint for a tie rod of a circular section to sustain a maximum pull of 70 kN. The ultimate strength of the material of the rod against tearing is 420 MPa. The ultimate tensile and shearing strength of the pin material are 510 MPa and 396 MPa respectively. Determine the tie rod section and pin section. Take factor of safety = 6.*

Solution. Given : $P = 70$ kN $= 70\,000$ N ; σ_{tu} for rod $= 420$ MPa ; *σ_{tu} for pin $= 510$ MPa ; $\tau_u = 396$ MPa ; $F.S. = 6$

We know that the permissible tensile stress for the rod material,

$$\sigma_t = \frac{\sigma_{tu} \text{ for rod}}{F.S.} = \frac{420}{6} = 70 \text{ MPa} = 70 \text{ N/mm}^2$$

and permissible shear stress for the pin material,

$$\tau = \frac{\tau_u}{F.S.} = \frac{396}{6} = 66 \text{ MPa} = 66 \text{ N/mm}^2$$

* Superfluous data.