CHAPTER

12

Cotter and Knuckle Joints

12.1 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 1 in 48 to 1 in 24 and it may be increased up to 1 in 8, if a locking device is provided. The locking device may be a taper pin or a set screw used on the lower end of the cotter. 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, strap end of connecting rod etc.

12.2 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.

The design of these types of joints are discussed, in detail, in the following pages.

12.3 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 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.

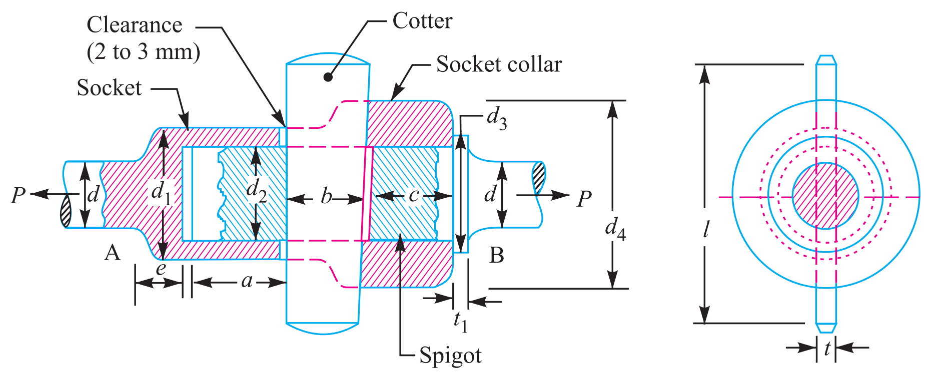

Fig. 12.1. Socket and spigot cotter joint.

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.

Contents

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

Fork lift is used to move goods from one place to the other within the factory.

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

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

$\therefore$ 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

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

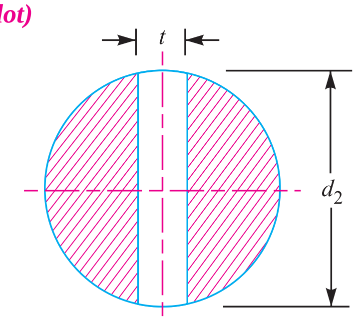

Fig. 12.2

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

$$= d_2 \times t$$

$\therefore$ 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.

Top

Contents

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

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

∴ Tearing strength of the socket across the slot

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

Equating this to load (P), we have

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

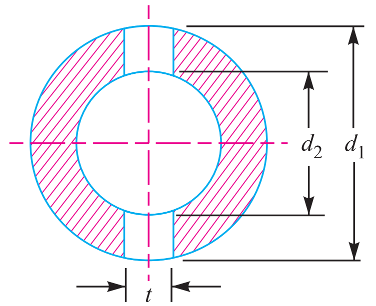

Fig. 12.3

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.

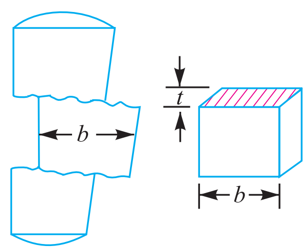

Fig. 12.4

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.

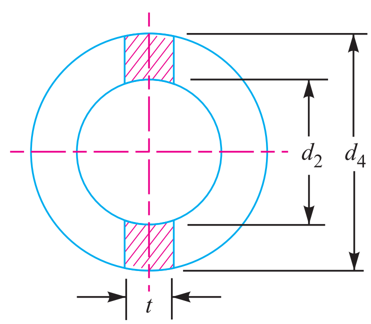

Fig. 12.5

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

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

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.

Top

Contents

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

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

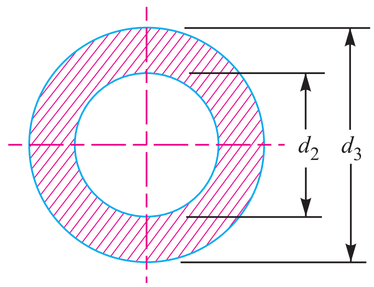

Fig. 12.6

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

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

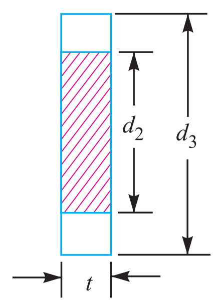

Fig. 12.7

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.

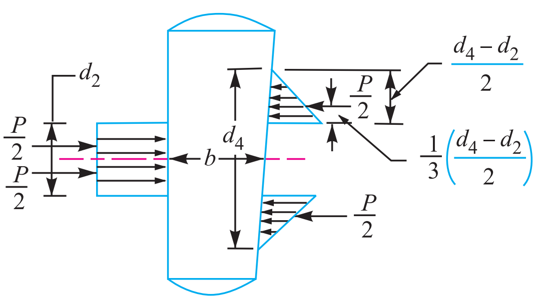

Fig. 12.8

Top

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,

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

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

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 : $P = 30$ kN $= 30 \times 10^3$ N ; $\sigma_t = 50$ MPa $= 50$ N / mm^2 ; $\tau = 35$ MPa $= 35$ N / mm^2 ; $\sigma_c = 90$ MPa $= 90$ N/mm^2

Accessories for hand operated sockets.

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),

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

$\therefore$ $d^2 = 30 \times 10^3 / 39.3 = 763$ or $d = 27.6$ say 28 mm **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),

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

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

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

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

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

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

Since this value of σ_c is more than the given value of $\sigma_c = 90$ 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 $\sigma_c = 90$ N/mm^2 in the above expression, *i.e.*

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

$\therefore$ $(d_2)^2 = 30 \times 10^3 / 22.5 = 1333$ or $d_2 = 36.5$ say 40 mm **Ans.**

and $t = d_2 / 4 = 40 / 4 = 10$ 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 d_4 = Diameter of socket collar.

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

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

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

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

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

A. T. Handle, B. Universal Joint

Contents

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),

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

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

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

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

11. The dimension e is taken as 1.2 d.

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

12.5 Sleeve and Cotter Joint

Sometimes, a sleeve and cotter joint as shown in Fig. 12.9, 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. 12.9. 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.

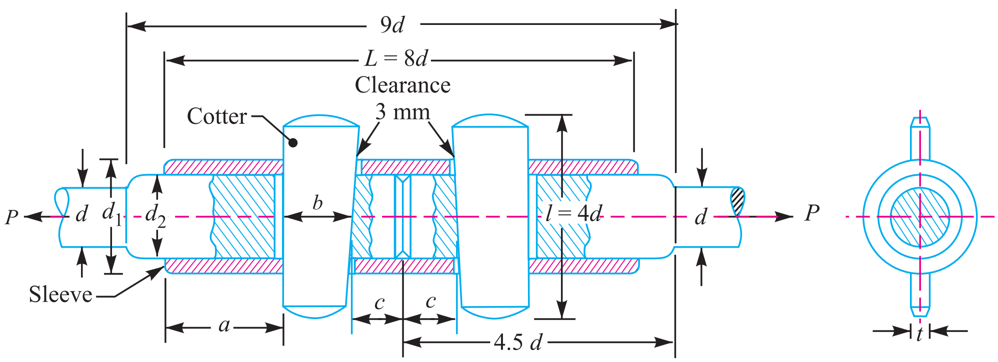

Fig. 12.9. Sleeve and cotter joint.

The various proportions for the sleeve and cotter joint in terms of the diameter of rod (d) are as follows :

Outside diameter of sleeve,

$$d_1 = 2.5\,d$$

Diameter of enlarged end of rod,

$$d_2 = \text{Inside diameter of sleeve} = 1.25\,d$$

Length of sleeve, $L = 8\,d$

Thickness of cotter, $t = d_2/4$ or $0.31\,d$

Width of cotter, $b = 1.25\,d$

Length of cotter, $l = 4\,d$

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

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

= $1.25\,d$

Top

Contents

12.6 Design of Sleeve and Cotter Joint

The sleeve and cotter joint is shown in Fig. 12.9.

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 :

1. *Failure of the rods in tension*

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

$$\text{Area resisting tearing} = \frac{\pi}{4} \times d^2$$

∴ 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 obtained.

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

Since the weakest section is that section of the rod which has a slot in it for the cotter, therefore area resisting tearing of the rod across the slot

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

and tearing strength of the rod across the slot

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

Equating this to load (P), we have

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

From this equation, the diameter of enlarged end of the rod (d_2) may be obtained.

Note: The thickness of cotter is usually taken as $d_2 / 4$.

Top

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

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

$$= d_2 \times t$$

$\therefore$ 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 sleeve in tension across the slot*

We know that the resisting area of sleeve across the slot

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

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

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

Equating this to load (P), we have

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

From this equation, the outside diameter of sleeve (d_1) may be obtained.

5. *Failure of cotter in shear*

Since the cotter is in double shear, therefore shearing area of the cotter

$$= 2b \times t$$

and shear strength of the cotter

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

Equating this to load (P), we have

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

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

6. *Failure of rod end in shear*

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

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

Offset handles.

Contents

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, distance (a) may be determined.

7. Failure of sleeve end in shear

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

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

and shear strength of the sleeve end

$$= 2\,(d_1 - d_2)\, c \times \tau$$

Equating this to load (P), we have

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

From this equation, distance (c) may be determined.

Example 12.2. *Design a sleeve and cotter joint to resist a tensile load of 60 kN. All parts of the joint are made of the same material with the following allowable stresses :*

σ_t = 60 MPa ; τ = 70 MPa ; and σ_c = 125 MPa.

Solution. Given : P = 60 kN = 60 × 10^3 N ; σ_t = 60 MPa = 60 N/mm^2 ; τ = 70 MPa = 70 N/mm^2 ; σ_c = 125 MPa = 125 N/mm^2

1. Diameter of the rods

Let d = Diameter of the rods.

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

$$60 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 60 = 47.13\, d^2$$

$$\therefore \quad d^2 = 60 \times 10^3 / 47.13 = 1273 \quad \text{or} \quad d = 35.7 \text{ say } 36 \text{ mm} \quad \textbf{Ans.}$$

2. Diameter of enlarged end of rod and thickness of cotter

Let d_2 = Diameter of enlarged end of rod, and

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

Considering the failure of the rod in tension across the weakest section (*i.e.* slot). We know that load (P),

$$60 \times 10^3 = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times \frac{d_2}{4}\right]60 = 32.13\,(d_2)^2$$

$$\therefore \quad (d_2)^2 = 60 \times 10^3 / 32.13 = 1867 \quad \text{or} \quad d_2 = 43.2 \text{ say } 44 \text{ mm} \quad \textbf{Ans.}$$

and thickness of cotter,

$$t = \frac{d_2}{4} = \frac{44}{4} = 11 \text{ mm} \quad \textbf{Ans.}$$

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

$$60 \times 10^3 = d_2 \times t \times \sigma_c = 44 \times 11 \times \sigma_c = 484\, \sigma_c$$

$$\therefore \quad \sigma_c = 60 \times 10^3 / 484 = 124 \text{ N/mm}^2$$

Since the induced crushing stress is less than the given value of 125 N/mm^2, therefore the dimensions d_2 and t are within safe limits.

3. Outside diameter of sleeve

Let d_1 = Outside diameter of sleeve.

Top

Contents

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

$$60 \times 10^3 = \left[\frac{\pi}{4}[(d_1)^2 - (d_2)^2] - (d_1 - d_2)\, t\right] \sigma_t$$

$$= \left[\frac{\pi}{4}[(d_1)^2 - (44)^2] - (d_1 - 44)\, 11\right] 60$$

$$\therefore \quad 60 \times 10^3 / 60 = 0.7854\,(d_1)^2 - 1520.7 - 11\, d_1 + 484$$

or $\quad (d_1)^2 - 14\, d_1 - 2593 = 0$

$$\therefore \quad d_1 = \frac{14 \pm \sqrt{(14)^2 + 4 \times 2593}}{2} = \frac{14 \pm 102.8}{2}$$

$= 58.4$ say 60 mm **Ans.** ...(Taking +ve sign)

4. *Width of cotter*

Let b = Width of cotter.

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

$$60 \times 10^3 = 2\, b \times t \times \tau = 2 \times b \times 11 \times 70 = 1540\, b$$

$$\therefore \quad b = 60 \times 10^3 / 1540 = 38.96 \text{ say } 40 \text{ mm} \textbf{ Ans.}$$

5. *Distance of the rod from the beginning to the cotter hole (inside the sleeve end)*

Let a = Required distance.

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

$$60 \times 10^3 = 2\, a \times d_2 \times \tau = 2\, a \times 44 \times 70 = 6160\, a$$

$$\therefore \quad a = 60 \times 10^3 / 6160 = 9.74 \text{ say } 10 \text{ mm} \textbf{ Ans.}$$

6. *Distance of the rod end from its end to the cotter hole*

Let c = Required distance.

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

$$60 \times 10^3 = 2\,(d_1 - d_2)\, c \times \tau = 2\,(60 - 44)\, c \times 70 = 2240\, c$$

$$\therefore \quad c = 60 \times 10^3 / 2240 = 26.78 \text{ say } 28 \text{ mm} \textbf{ Ans.}$$

12.7 Gib and Cotter Joint

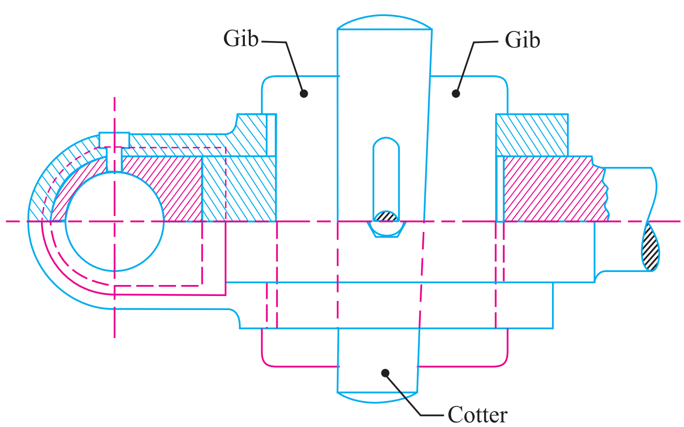

Fig. 12.10. Gib and cotter joint for strap end of a connecting rod.

Top

A *gib and cotter joint is usually used in strap end (or big end) of a connecting rod as shown in Fig. 12.10. In such cases, 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. 12.11 (*a*). In order to prevent this, gibs as shown in Fig. 12.11 (*b*) and (*c*), are used which hold together the ends of the strap. Moreover, gibs provide a larger bearing surface for the cotter to slide on, due to the increased holding power. Thus, the tendency of cotter to slacken back owing to friction is considerably decreased. The jib, also, enables parallel holes to be used.

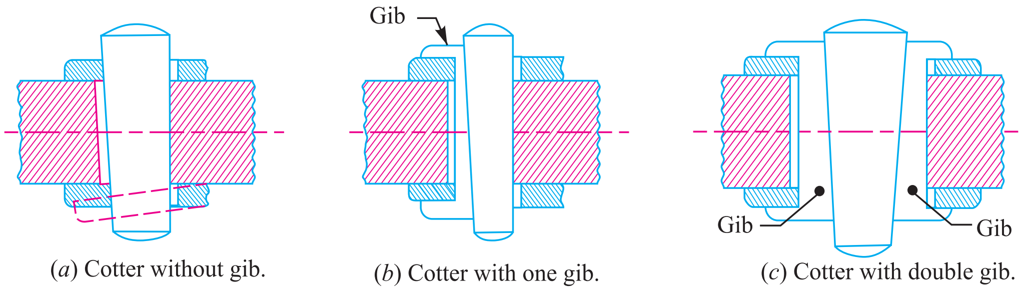

Fig. 12.11. Gib and cotter Joints.

Notes : 1. When one gib is used, the cotter with one side tapered is provided and the gib is always on the outside as shown in Fig. 12.11 (*b*).

2. When two jibs are used, the cotter with both sides tapered is provided.

3. Sometimes to prevent loosening of cotter, a small set screw is used through the rod jamming against the cotter.

12.8 Design of a Gib and Cotter Joint for Strap End of a Connecting Rod

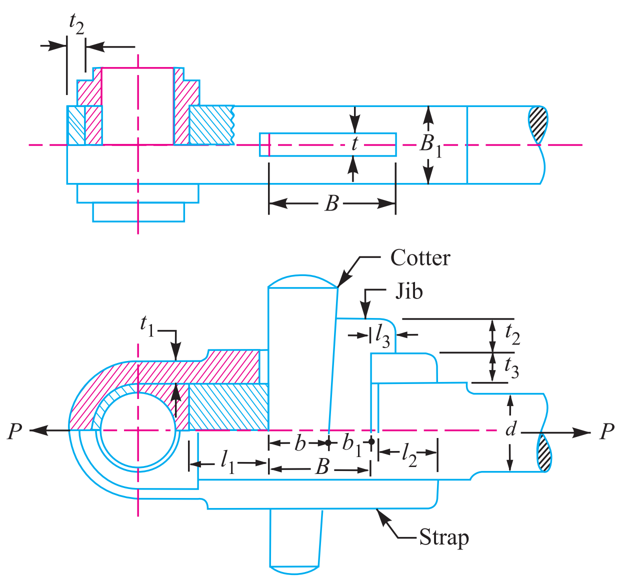

Fig. 12.12. Gib and cotter joint for strap end of a connecting rod.

Consider a gib and cotter joint for strap end (or big end) of a connecting rod as shown in Fig. 12.12. The connecting rod is subjected to tensile and compressive loads.

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* A gib is a piece of mild steel having the same thickness and taper as the cotter.

Let P = Maximum thrust or pull in the connecting rod,

d = Diameter of the adjacent end of the round part of the rod,

B_1 = Width of the strap,

B = Total width of gib and cotter,

t = Thickness of cotter,

t_1 = Thickness of the strap at the thinnest part,

σ_t = Permissible tensile stress for the material of the strap, and

τ = Permissible shear stress for the material of the cotter and gib.

The width of strap (B_1) is generally taken equal to the diameter of the adjacent end of the round part of the rod (d). The other dimensions may be fixed as follows :

Thickness of cotter,

$$t = \frac{\text{Width of strap}}{4} = \frac{B_1}{4}$$

Thickness of gib = Thickness of cotter (t)

Height (t_2) and length of gib head (l_3)

= Thickness of cotter (t)

In designing the gib and cotter joint for strap end of a connecting rod, the following modes of failure are considered.

1. *Failure of the strap in tension*

Assuming that no hole is provided for lubrication, the area that resists the failure of the strap due to tearing $= 2\, B_1 \times t_1$

$\therefore$ Tearing strength of the strap

$= 2\, B_1 \times t_1 \times \sigma_t$

Equating this to the load (P), we get

$$P = 2\, B_1 \times t_1 \times \sigma_t$$

From this equation, the thickness of the strap at the thinnest part (t_1) may be obtained. When an oil hole is provided in the strap, then its weakening effect should be considered.

The thickness of the strap at the cotter (t_3) is increased such that the area of cross-section of the strap at the cotter hole is not less than the area of the strap at the thinnest part. In other words

$$2\, t_3\, (B_1 - t) = 2\, t_1 \times B_1$$

From this expression, the value of t_3 may be obtained.

(a) Hand operated sqaure drive sockets (b) Machine operated sockets.

Note : This picture is given as additional information and is not a direct example of the current chapter.

2. *Failure of the gib and cotter in shearing*

Since the gib and cotter are in double shear, therefore area resisting failure

$$= 2\,B \times t$$

and resisting strength $= 2\,B \times t \times \tau$

Equating this to the load (P), we get

$$P = 2\,B \times t \times \tau$$

From this equation, the total width of gib and cotter (B) may be obtained. In the joint, as shown in Fig. 12.12, one gib is used, the proportions of which are

Width of gib, $b_1 = 0.55\,B$; and width of cotter, $b = 0.45\,B$

The other dimensions may be fixed as follows :

Thickness of the strap at the crown,

$$t_4 = 1.15\,t_1 \text{ to } 1.5\,t_1$$

$$l_1 = 2\,t_1;\text{ and } l_2 = 2.5\,t_1$$

Example 12.3. *The big end of a connecting rod, as shown in Fig. 12.12, is subjected to a maximum load of 50 kN. The diameter of the circular part of the rod adjacent to the strap end is 75 mm. Design the joint, assuming permissible tensile stress for the material of the strap as 25 MPa and permissible shear stress for the material of cotter and gib as 20 MPa.*

Solution. Given : $P = 50$ kN $= 50 \times 10^3$ N ; $d = 75$ mm ; $\sigma_t = 25$ MPa $= 25$ N/mm^2 ; $\tau = 20$ MPa $= 20$ N/mm^2

1. *Width of the strap*

Let B_1 = Width of the strap.

The width of the strap is generally made equal to the diameter of the adjacent end of the round part of the rod (d).

$\therefore$ $B_1 = d = 75$ mm **Ans.**

Other dimensions are fixed as follows :

Thickness of the cotter

$$t = \frac{B_1}{4} = \frac{75}{4} = 18.75 \text{ say } 20 \text{ mm} \quad \textbf{Ans.}$$

Thickness of gib = Thickness of cotter = 20 mm **Ans.**

Height (t_2) and length of gib head (l_3)

= Thickness of cotter = 20 mm **Ans.**

2. *Thickness of the strap at the thinnest part*

Let t_1 = Thickness of the strap at the thinnest part.

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

$$50 \times 10^3 = 2\,B_1 \times t_1 \times \sigma_t = 2 \times 75 \times t_1 \times 25 = 3750\,t_1$$

$$\therefore \quad t_1 = 50 \times 10^3 / 3750 = 13.3 \text{ say } 15 \text{ mm} \quad \textbf{Ans.}$$

3. *Thickness of the strap at the cotter*

Let t_3 = Thickness of the strap at the cotter.

The thickness of the strap at the cotter is increased such that the area of the cross-section of the strap at the cotter hole is not less than the area of the strap at the thinnest part. In other words,

$$2\,t_3\,(B_1 - t) = 2\,t_1 \times B_1$$

$$2\,t_3\,(75 - 20) = 2 \times 15 \times 75 \quad \text{or} \quad 110\,t_3 = 2250$$

$$\therefore \quad t_3 = 2250 / 110 = 20.45 \text{ say } 21 \text{ mm} \quad \textbf{Ans.}$$

4. Total width of gib and cotter

Let B = Total width of gib and cotter.

Considering the failure of gib and cotter in double shear. We know that load (P),

$$50 \times 10^3 = 2\,B \times t \times \tau = 2\,B \times 20 \times 20 = 800\,B$$

$$\therefore \quad B = 50 \times 10^3 / 800 = 62.5 \text{ say } 65 \text{ mm } \textbf{Ans.}$$

Since one gib is used, therefore width of gib,

$$b_1 = 0.55\,B = 0.55 \times 65 = 35.75 \text{ say } 36 \text{ mm } \textbf{Ans.}$$

and width of cotter, $b = 0.45\,B = 0.45 \times 65 = 29.25$ say 30 mm **Ans.**

The other dimensions are fixed as follows :

$$t_4 = 1.25\,t_1 = 1.25 \times 15 = 18.75 \text{ say } 20 \text{ mm } \textbf{Ans.}$$

$$l_1 = 2\,t_1 = 2 \times 15 = 30 \text{ mm } \textbf{Ans.}$$

and $l_2 = 2.5\,t_1 = 2.5 \times 15 = 37.5$ say 40 mm **Ans.**

12.9 Design of Gib and Cotter Joint for Square Rods

Consider a gib and cotter joint for square rods as shown in Fig. 12.13. The rods may be subjected to a tensile or compressive load. All components of the joint are assumed to be of the same material.

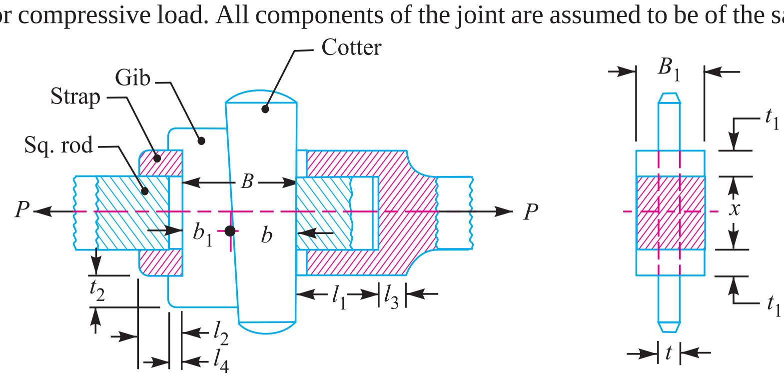


Fig. 12.13. Gib and cotter joint for square rods.

Let P = Load carried by the rods,

x = Each side of the rod,

B = Total width of gib and cotter,

B_1 = Width of the strap,

t = Thickness of cotter,

t_1 = Thickness of the strap, and

σ_t, τ and σ_c = Permissible tensile, shear and crushing stresses.

In designing a gib and cotter joint, the following modes of failure are considered.

1. Failure of the rod in tension

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

Area resisting tearing $= x \times x = x^2$

$\therefore$ Tearing strength of the rod

$$= x^2 \times \sigma_t$$

Equating this to the load (P), we have

$$P = x^2 \times \sigma_t$$

From this equation, the side of the square rod (x) may be determined. The other dimensions are fixed as under :

Width of strap, B_1 = Side of the square rod = x

Thickness of cotter, $t = \frac{1}{4}$ width of strap $= \frac{B_1}{4}$

Thickness of gib = Thickness of cotter (t)

Height (t_2) and length of gib head (l_4)

= Thickness of cotter (t)

2. Failure of the gib and cotter in shearing

Since the gib and cotter are in double shear, therefore,

Area resisting failure $= 2\,B \times t$

and resisting strength $= 2\,B \times t \times \tau$

Equating this to the load (P), we have

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

From this equation, the width of gib and cotter (B) may be obtained. In the joint, as shown in Fig. 12.13, one gib is used, the proportions of which are

Width of gib, $b_1 = 0.55\,B$; and width of cotter, $b = 0.45\,B$

In case two gibs are used, then

Width of each gib $= 0.3\,B$; and width of cotter $= 0.4\,B$

3. Failure of the strap end in tension at the location of gib and cotter

Area resisting failure $= 2\,[B_1 \times t_1 - t_1 \times t] = 2\,[x \times t_1 - t_1 \times t]$ ... ($\because B_1 = x$)

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

Equating this to the load (P), we have

$$P = 2\,[x \times t_1 - t_1 \times t]\,\sigma_t$$

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

4. Failure of the strap or gib in crushing

The strap or gib (at the strap hole) may fail due to crushing.

Area resisting failure $= 2\,t_1 \times t$

$\therefore$ Resisting strength $= 2\,t_1 \times t \times \sigma_c$

Equating this to the load (P), we have

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

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

5. Failure of the rod end in shearing

Since the rod is in double shear, therefore

Area resisting failure $= 2\,l_1 \times x$

$\therefore$ Resisting strength $= 2\,l_1 \times x \times \tau$

Equating this to the load (P), we have

$$P = 2\,l_1 \times x \times \tau$$

From this equation, the dimension l_1 may be determined.

6. Failure of the strap end in shearing

Since the length of rod (l_2) is in double shearing, therefore

Area resisting failure $= 2 \times 2\,l_2 \times t_1$

$\therefore$ Resisting strength $= 2 \times 2\, l_2 \times t_1 \times \tau$

Equating this to the load (P), we have

$$P = 2 \times 2\, l_2 \times t_1 \times \tau$$

From this equation, the length of rod (l_2) may be determined. The length l_3 of the strap end is proportioned as $\frac{2}{3}$ rd of side of the rod. The clearance is usually kept 3 mm. The length of cotter is generally taken as 4 times the side of the rod.

Example 12.4. *Design a gib and cottor joint as shown in Fig. 12.13, to carry a maximum load of 35 kN. Assuming that the gib, cotter and rod are of same material and have the following allowable stresses :*

$$\sigma_t = 20 \text{ MPa} ; \tau = 15 \text{ MPa} ; \text{and } \sigma_c = 50 \text{ MPa}$$

Solution. Given : $P = 35$ kN $= 35\,000$ N ; $\sigma_t = 20$ MPa $= 20$ N/mm^2 ; $\tau = 15$ MPa $= 15$ N/mm^2 ; $\sigma_c = 50$ MPa $= 50$ N/mm^2

1. Side of the square rod

Let x = Each side of the square rod.

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

$$35\,000 = x^2 \times \sigma_t = x^2 \times 20 = 20\, x^2$$

$$\therefore \quad x^2 = 35\,000 / 20 = 1750 \text{ or } x = 41.8 \text{ say } 42 \text{ mm } \textbf{Ans.}$$

Other dimensions are fixed as follows :

Width of strap, $B_1 = x = 42$ mm **Ans.**

Thickness of cotter, $t = \dfrac{B_1}{4} = \dfrac{42}{4} = 10.5$ say 12 mm **Ans.**

Thickness of gib = Thickness of cotter = 12 mm **Ans.**

Height (t_2) and length of gib head (l_4)

= Thickness of cotter = 12 mm **Ans.**

2. Width of gib and cotter

Let B = Width of gib and cotter.

Considering the failure of the gib and cotter in double shear. We know that load (P),

$$35\,000 = 2\, B \times t \times \tau = 2\, B \times 12 \times 15 = 360\, B$$

$$\therefore \quad B = 35\,000 / 360 = 97.2 \text{ say } 100 \text{ mm } \textbf{Ans.}$$

Since one gib is used, therefore

Width of gib, $b_1 = 0.55\, B = 0.55 \times 100 = 55$ mm **Ans.**

and width of cotter, $b = 0.45\, B = 0.45 \times 100 = 45$ mm **Ans.**

3. Thickness of strap

Let t_1 = Thickness of strap.

Considering the failure of the strap end in tension at the location of the gib and cotter. We know that load (P),

$$35\,000 = 2\,(x \times t_1 - t_1 \times t)\, \sigma_t = 2\,(42 \times t_1 - t_1 \times 12)\, 20 = 1200\, t_1$$

$$\therefore \quad t_1 = 35\,000 / 1200 = 29.1 \text{ say } 30 \text{ mm } \textbf{Ans.}$$

Now the induced crushing stress may be checked by considering the failure of the strap or gib in crushing. We know that load (P),

$$35\,000 = 2\, t_1 \times t \times \sigma_c = 2 \times 30 \times 12 \times \sigma_c = 720\, \sigma_c$$

$$\therefore \quad \sigma_c = 35\,000 / 720 = 48.6 \text{ N/mm}^2$$

Since the induced crushing stress is less than the given crushing stress, therefore the joint is safe.

4. Length (l_1) of the rod

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

$$35\,000 = 2\,l_1 \times x \times \tau = 2\,l_1 \times 42 \times 15 = 1260\,l_1$$

$$\therefore \quad l_1 = 35\,000 / 1260 = 27.7 \text{ say } 28 \text{ mm } \textbf{Ans.}$$

5. Length (l_2) of the rod

Considering the failure of the strap end in shearing. Since the length of the rod (l_2) is in double shear, therefore load (P),

$$35\,000 = 2 \times 2\,l_2 \times t_1 \times \tau = 2 \times 2\,l_2 \times 30 \times 15 = 1800\,l_2$$

$$\therefore \quad l_2 = 35\,000 / 1800 = 19.4 \text{ say } 20 \text{ mm } \textbf{Ans.}$$

Length (l_3) of the strap end

$$= \frac{2}{3} \times x = \frac{2}{3} \times 42 = 28 \text{ mm } \textbf{Ans.}$$

and length of cotter $= 4\,x = 4 \times 42 = 168$ mm **Ans.**

12.10 Design of Cotter Joint to Connect Piston Rod and Crosshead

The cotter joint to connect piston rod and crosshead is shown in Fig. 12.14. In such a type of joint, the piston rod is tapered in order to resist the thrust instead of being provided with a collar for the purpose. The taper may be from 1 in 24 to 1 in 12.

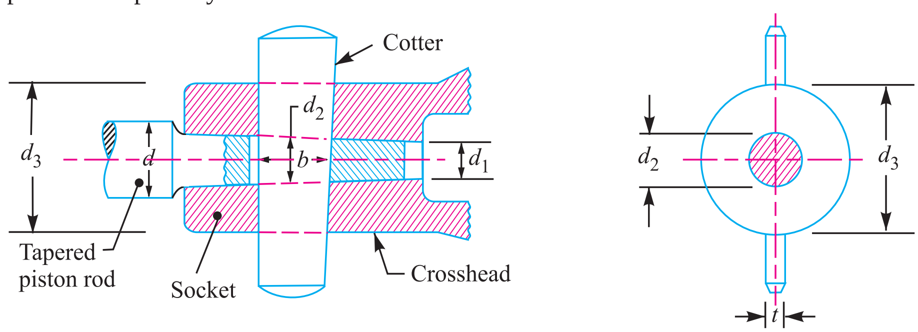


Fig. 12.14. Cotter joint to connect piston rod and crosshead.

Let d = Diameter of parallel part of the piston rod,

d_1 = Diameter at tapered end of the piston,

d_2 = Diameter of piston rod at the cotter,

d_3 = Diameter of socket through the cotter hole,

b = Width of cotter at the centre,

t = Thickness of cotter,

σ_t, τ and σ_c = Permissible stresses in tension, shear and crushing respectively.

We know that maximum load on the piston,

$$P = \frac{\pi}{4} \times D^2 \times p$$

where D = Diameter of the piston, and

p = Effective steam pressure on the piston.

Let us now consider the various failures of the joint as discussed below :

Contents

1. *Failure of piston rod in tension at cotter*

The piston rod may fail in tension at cotter due to the maximum load on the piston. We know that area resisting tearing at the cotter

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

∴ Tearing strength of the piston rod at the cotter

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

Equating this to maximum load (P), we have

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

From this equation, the diameter of piston rod at the cotter (d_2) may be determined.

Note: The thickness of cotter (t) is taken as 0.3 d_2.

2. *Failure of cotter in shear*

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 maximum load (P), we have

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

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

3. *Failure of the socket in tension at cotter*

We know that area that resists tearing of socket at cotter

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

and tearing strength of socket at cotter

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

Equating this to maximum load (P), we have

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

From this equation, diameter of socket (d_3) is obtained.

4. *Failure of socket in crushing*

We know that area that resists crushing of socket

$$= (d_3 - d_2)\,t$$

and crushing strength of socket

$$= (d_3 - d_2)\,t \times \sigma_c$$

Equating this to maximum load (P), we have

$$P = (d_3 - d_2)\,t \times \sigma_c$$

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

The length of the tapered portion of the piston rod (L) is taken as 2.2 d_2. The diameter of the parallel part of the piston rod (d) and diameter of the piston rod at the tapered end (d_1) may be obtained as follows :

Top

$$d = d_2 + \frac{L}{2} \times \text{taper} \; ; \text{ and } d_1 = d_2 - \frac{L}{2} \times \text{taper}$$

Note: The taper on the piston rod is usually taken as 1 in 20.

Example 12.5. *Design a cotter joint to connect piston rod to the crosshead of a double acting steam engine. The diameter of the cylinder is 300 mm and the steam pressure is 1 N/mm². The allowable stresses for the material of cotter and piston rod are as follows :*

$$\sigma_t = 50 \text{ MPa} \; ; \tau = 40 \text{ MPa} \; ; \text{and } \sigma_c = 84 \text{ MPa}$$

Solution. Given : $D = 300$ mm ; $p = 1$ N/mm² ; $\sigma_t = 50$ MPa $= 50$ N/mm² ; $\tau = 40$ MPa $= 40$ N/mm² ; $\sigma_c = 84$ MPa $= 84$ N/mm²

We know that maximum load on the piston rod,

$$P = \frac{\pi}{4} \times D^2 \times p = \frac{\pi}{4} (300)^2 \; 1 = 70\;695 \text{ N}$$

The various dimensions for the cotter joint are obtained by considering the different modes of failure as discussed below :

1. Diameter of piston rod at cotter

Let d_2 = Diameter of piston rod at cotter, and

t = Thickness of cotter. It may be taken as $0.3\, d_2$.

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

$$70\;695 = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t = \left[\frac{\pi}{4}(d_2)^2 - 0.3\,(d_2)^2\right] 50 = 24.27\,(d_2)^2$$

$\therefore \quad (d_2)^2 = 70\,695 / 24.27 = 2913 \quad$ or $\quad d_2 = 53.97$ say 55 mm **Ans.**

and $\quad t = 0.3\, d_2 = 0.3 \times 55 = 16.5$ mm **Ans.**

2. Width of cotter

Let b = Width of cotter.

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

$$70\;695 = 2\, b \times t \times \tau = 2\, b \times 16.5 \times 40 = 1320\, b$$

$\therefore \quad b = 70\,695 / 1320 = 53.5$ say 54 mm **Ans.**

3. Diameter of socket

Let d_3 = Diameter of socket.

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

$$70\;695 = \left\{\frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right] - (d_3 - d_2)\, t\right\}\sigma_t$$

$$= \left\{\frac{\pi}{4}\left[(d_3)^2 - (55)^2\right] - (d_3 - 55)\, 16.5\right\} 50$$

$$= 39.27\,(d_3)^2 - 118\,792 - 825\, d_3 + 45\,375$$

or $\quad (d_3)^2 - 21\, d_3 - 3670 = 0$

$$\therefore \quad d_3 = \frac{21 \pm \sqrt{(21)^2 + 4 \times 3670}}{2} = \frac{21 \pm 123}{2} = 72 \text{ mm} \quad \text{...(Taking + ve sign)}$$

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

$$70\;695 = (d_3 - d_2)\, t \times \sigma_c = (72 - 55)\, 16.5 \times \sigma_c = 280.5\, \sigma_c$$

$\therefore \quad \sigma_c = 70\,695 / 280.5 = 252$ N/mm²

Since the induced crushing is greater than the permissible value of 84 N/mm², therefore let us find the value of d_3 by substituting σ_c = 84 N/mm² in the above expression, *i.e.*

$$70\ 695 = (d_3 - 55)\ 16.5 \times 84 = (d_3 - 55)\ 1386$$

$$\therefore \quad d_3 - 55 = 70\ 695 / 1386 = 51$$

or $$d_3 = 55 + 51 = 106 \text{ mm} \textbf{ Ans.}$$

We know the tapered length of the piston rod,

$$L = 2.2\ d_2 = 2.2 \times 55 = 121 \text{ mm} \textbf{ Ans.}$$

Assuming the taper of the piston rod as 1 in 20, therefore the diameter of the parallel part of the piston rod,

$$d = d_2 + \frac{L}{2} \times \frac{1}{20} = 55 + \frac{121}{2} \times \frac{1}{20} = 58 \text{ mm} \textbf{ Ans.}$$

and diameter of the piston rod at the tapered end,

$$d_1 = d_2 - \frac{L}{2} \times \frac{1}{20} = 55 - \frac{121}{2} \times \frac{1}{20} = 52 \text{ mm} \textbf{ Ans.}$$

12.11 Design of Cotter Foundation Bolt

The cotter foundation bolt is mostly used in conjunction with foundation and holding down bolts to fasten heavy machinery to foundations. It is generally used where an ordinary bolt or stud cannot be conveniently used. Fig. 12.15 shows the two views of the application of such a cotter foundation bolt. In this case, the bolt is dropped down from above and the cotter is driven in from the side. Now this assembly is tightened by screwing down the nut. It may be noted that two base plates (one under the nut and the other under the cotter) are used to provide more bearing area in order to take up the tightening load on the bolt as well as to distribute the same uniformly over the large surface.

Variable speed Knee-type milling machine.

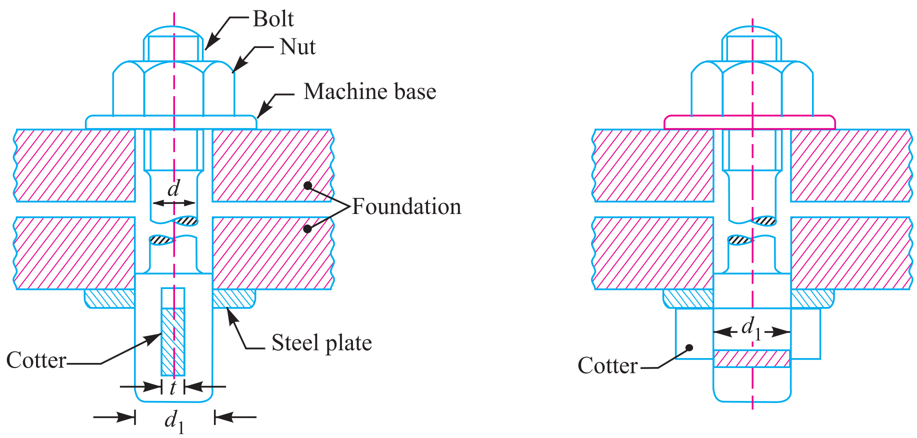


Fig. 12.15. Cotter foundation bolt.

Let d = Diameter of bolt,

d_1 = Diameter of the enlarged end of bolt,

t = Thickness of cotter, and

b = Width of cotter.

The various modes of failure of the cotter foundation bolt are discussed as below:

1. *Failure of bolt in tension*

The bolt may fail in tension due to the load (P). We know that area resisting tearing

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

∴ Tearing strength of the bolt

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

Equating this to the load (P), we have

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

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

2. *Failure of the enlarged end of the bolt in tension at the cotter*

We know that area resisting tearing

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

∴ Tearing strength of the enlarged end of the bolt

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

Equating this to the load (P), we have

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

From this equation, the diameter of the enlarged end of the bolt (d_1) may be determined.

Note: The thickness of cotter is usually taken as $d_1 / 4$.

3. *Failure of cotter in shear*

Since the cotter is in double shear, therefore area resisting shearing

$$= 2\, b \times t$$

∴ Shearing strength of cotter

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

Equating this to the load (P), we have

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

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

4. *Failure of cotter in crushing*

We know that area resisting crushing

$$= b \times t$$

∴ Crushing strength of cotter

$$= b \times t \times \sigma_c$$

Equating this to the load (P), we have

$$P = b \times t \times \sigma_c$$

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

Example 12.6. *Design and draw a cottered foundation bolt which is subjected to a maximum pull of 50 kN. The allowable stresses are :*

$\sigma_t = 80$ MPa ; $\tau = 50$ MPa ; and $\sigma_c = 100$ MPa

Solution. Given: $P = 50$ kN $= 50 \times 10^3$ N ; $\sigma_t = 80$ MPa $= 80$ N/mm^2 ; $\tau = 50$ MPa $= 50$ N/mm^2 ; $\sigma_c = 100$ MPa $= 100$ N/mm^2

1. *Diameter of bolt*

Let d = Diameter of bolt.

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

$$50 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 80 = 62.84\, d^2$$

$\therefore$ $d^2 = 50 \times 10^3 / 62.84 = 795.7$ or $d = 28.2$ say 30 mm **Ans.**

2. *Diameter of enlarged end of the bolt and thickness of cotter*

Let d_1 = Diameter of enlarged end of the bolt, and

t = Thickness of cotter. It may be taken as $d_1 / 4$.

Considering the failure of the enlarged end of the bolt in tension at the cotter. We know that load (P),

$$50 \times 10^3 = \left[\frac{\pi}{4}(d_1)^2 - d_1 \times t\right]\sigma_t = \left[\frac{\pi}{4}(d_1)^2 - d_1 \times \frac{d_1}{4}\right] 80 = 42.84\,(d_1)^2$$

$\therefore$ $(d_1)^2 = 50 \times 10^3 / 42.84 = 1167$ or $d_1 = 34$ say 36 mm **Ans.**

and $t = \dfrac{d_1}{4} = \dfrac{36}{4} = 9$ mm **Ans.**

3. *Width of cotter*

Let b = Width of cotter.

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

$$50 \times 10^3 = 2\, b \times t \times \tau = 2\, b \times 9 \times 50 = 900\, b$$

$\therefore$ $b = 50 \times 10^3 / 900 = 55.5$ mm say 60 mm **Ans.**

Let us now check the crushing stress induced in the cotter. Considering the failure of cotter in crushing. We know that load (P),

$$50 \times 10^3 = b \times t \times \sigma_c = 60 \times 9 \times \sigma_c = 540\, \sigma_c$$

$\therefore$ $\sigma_c = 50 \times 10^3 / 540 = 92.5$ N/mm^2

Since the induced crushing stress is less than the permissible value of 100 N/mm^2, therefore the design is safe.

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.

Contents

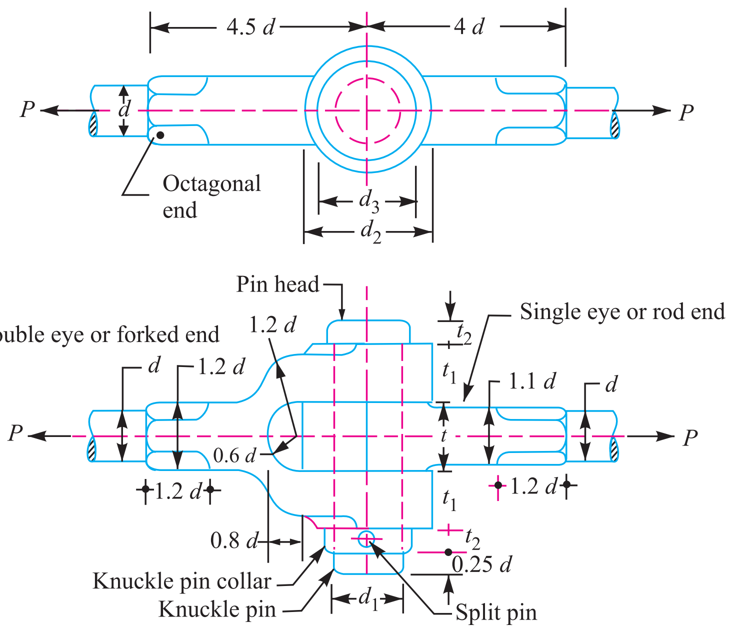
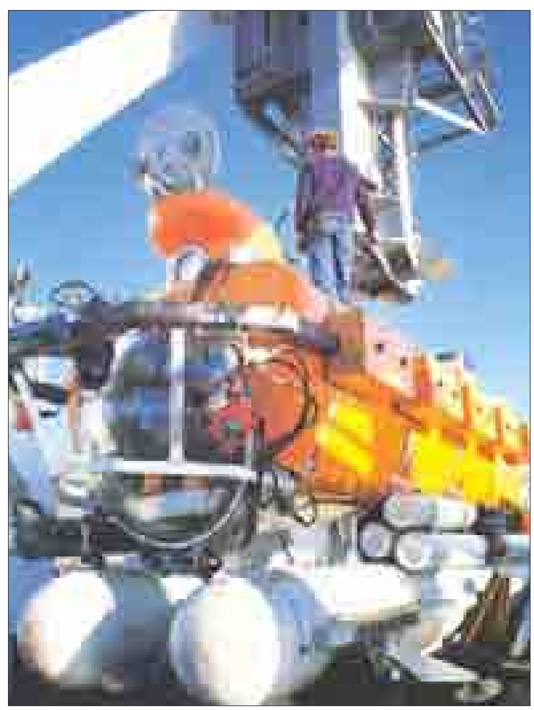


Fig. 12.16. Kunckle joint.

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.



Submersibles like this can work at much greater ocean depths and high pressures where divers cannot reach.

Note : This picture is given as additional information and is not a direct example of the current chapter.

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,

$$d_2 = 2\,d$$

Top

Contents

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

Top

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,

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

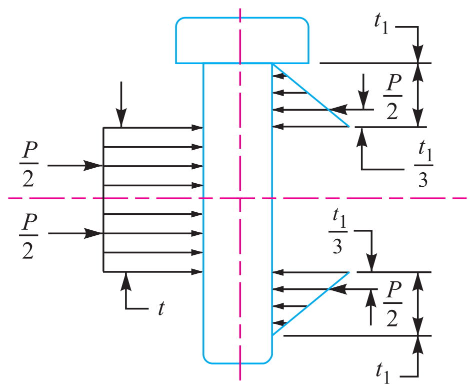


Fig. 12.17. Distribution of load on the pin.

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 $= d_1 \times t$

∴ 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

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

∴ 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

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

∴ 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 $= d_1 \times 2\, t_1$

∴ 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

Contents

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),

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

$\therefore$ $d^2 = 150 \times 10^3 / 59 = 2540$ 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),

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

Top

Contents

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

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.

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.

Top

We shall now consider the various methods of failure of the joint as discussed below:

1. *Failure of the rod in tension*

Let d = Diameter of the rod.

We know that the load (P),

$$70\,000 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 70 = 55\, d^2$$

$\therefore$ $d^2 = 70\,000 / 55 = 1273$ or $d = 35.7$ say 36 mm **Ans.**

The other dimensions of the joint are fixed as given below :

Diameter of the knuckle pin,

$$d_1 = d = 36 \text{ mm}$$

Outer diameter of the eye,

$$d_2 = 2\,d = 2 \times 36 = 72 \text{ mm}$$

Diameter of knuckle pin head and collar,

$$d_3 = 1.5\,d = 1.5 \times 36 = 54 \text{ mm}$$

Thickness of single eye or rod end,

$$t = 1.25\,d = 1.25 \times 36 = 45 \text{ mm}$$

Thickness of fork, $t_1 = 0.75\,d = 0.75 \times 36 = 27$ mm

Now we shall check for the induced streses as discussed below :

2. *Failure of the knuckle pin in shear*

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

$$70\,000 = 2 \times \frac{\pi}{4} (d_1)^2\, \tau = 2 \times \frac{\pi}{4} (36)^2\, \tau = 2036\, \tau$$

$\therefore$ $\tau = 70\,000 / 2036 = 34.4 \text{ N/mm}^2$

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),

$$70\,000 = (d_2 - d_1)\, t \times \sigma_t = (72 - 36)\, 45\, \sigma_t = 1620\, \sigma_t$$

$\therefore$ $\sigma_t = 70\,000 / 1620 = 43.2 \text{ N/mm}^2$

4. *Failure of the forked end in tension*

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

$$70\,000 = (d_2 - d_1)\, 2\, t_1 \times \sigma_t = (72 - 36) \times 2 \times 27 \times \sigma_t = 1944\, \sigma_t$$

$\therefore$ $\sigma_t = 70\,000 / 1944 = 36 \text{ N/mm}^2$

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

12.16 Adjustable Screwed Joint for Round Rods (Turnbuckle)

Sometimes, two round tie rods, as shown in Fig. 12.18, 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

Turnbuckle.

Contents

instead of a spanner, a round iron rod may be used. The iron rod is inserted in a hole in the coupler as shown dotted in Fig. 12.18.

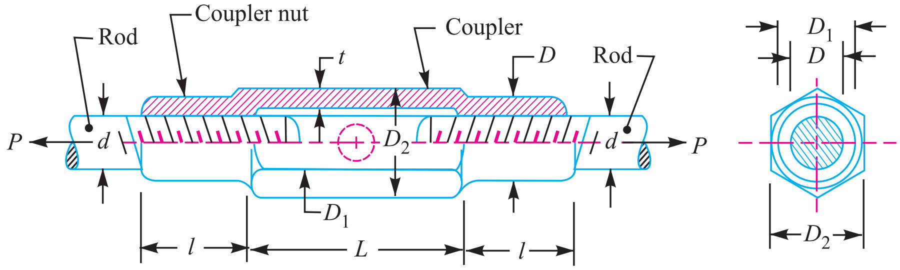


Fig. 12.18. Turnbuckle.

A turnbuckle commonly used in engineering practice (mostly in aeroplanes) is shown in Fig. 12.19. This type of turnbuckle is made hollow in the middle to reduce its weight. In this case, the two ends of the rods may also be seen. It is not necessary that the material of the rods and the turnbuckle may be same or different. It depends upon the pull acting on the joint.

12.17 Design of Turnbuckle

Consider a turnbuckle, subjected to an axial load P, as shown in Fig. 12.19. Due to this load, the threaded rod will be subjected to tensile stress whose magnitude is given by

$$\sigma_t = \frac{P}{A} = \frac{P}{\frac{\pi}{4}(d_c)^2}$$

where d_c = Core diameter of the threaded rod.

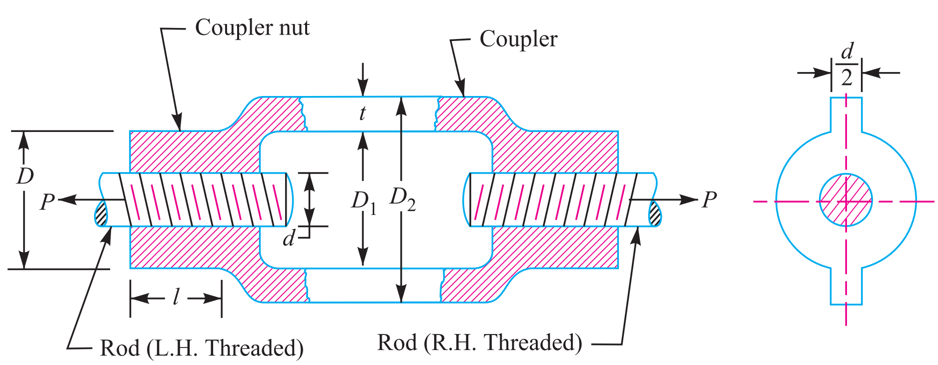


Fig. 12.19. Turnbuckle.

In order to drive the rods, the torque required is given by

$$T = P \tan(\alpha + \phi)\frac{d_p}{2}$$

where α = Helix angle,

$\tan\phi$ = Coefficient of friction between the threaded rod and the coupler nut, and

d_p = Pitch diameter or mean diameter of the threaded rod.

Top

∴ Shear stress produced by the torque,

$$\tau = \frac{T}{J} \times \frac{d_p}{2} = \frac{P \tan(\alpha + \phi)\dfrac{d_p}{2}}{\dfrac{\pi}{32}(d_p)^4} \times \frac{d_p}{2} = P \tan(\alpha + \phi) \times \frac{8}{\pi (d_p)^2}$$

$$= \frac{8P}{\pi (d_p)^2} \left(\frac{\tan\alpha + \tan\phi}{1 - \tan\alpha \times \tan\phi} \right)$$

The usual values of $\tan\alpha$, $\tan\phi$ and d_p are as follows :

$$\tan\alpha = 0.03, \tan\phi = 0.2, \text{ and } d_p = 1.08\, d_c$$

Substituting these values in the above expression, we get

$$\tau = \frac{8P}{\pi (1.08\, d_c)^2} \left[\frac{0.03 + 0.2}{1 - 0.03 \times 0.2} \right] = \frac{8P}{4\pi (d_c)^2} = \frac{P}{2A} = \frac{\sigma_t}{2}$$

$$\quad ...\left[\because A = \frac{\pi}{4}(d_c)^2 \right]$$

Since the threaded rod is subjected to tensile stress as well as shear stress, therefore maximum principal stress,

$$\sigma_{t\,(max)} = \frac{\sigma_t}{2} + \frac{1}{2}\sqrt{(\sigma_t)^2 + 4\tau^2} = \frac{\sigma_t}{2} + \frac{1}{2}\sqrt{(\sigma_t)^2 + (\sigma_t)^2} \qquad ...\left(\because \tau = \frac{\sigma_t}{2} \right)$$

$$= 0.5\,\sigma_t + 0.707\,\sigma_t = 1.207\,\sigma_t = 1.207\, P/A$$

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.*

Design load, $\quad P_d = 1.3\,P$

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

1. *Diameter of the rods*

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

Tearing resistance of the threads of the rod

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

Equating the design load (P_d) to the tearing resistance of the threads, we have

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

where $\quad d_c$ = Core diameter of the threads of the rod, and

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

From the above expression, the core diameter of the threads may be obtained. The nominal diameter of the threads (or diameter of the rod) may be found from Table 11.1, corresponding to the core diameter, assuming coarse threads.

2. *Length of the coupler nut*

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

Shearing resistance of the threads of the coupler nut

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

where τ = Shear stress for the material of the coupler nut.

Equating the design load to the shearing resistance of the threads in the coupler nut, we have

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

From this expression, the value of l may be calculated. In actual practice, the length of coupler nut (l) is taken d to 1.25 d for steel nuts and 1.5 d to 2 d for cast iron and softer material nut. The length of the coupler nut may also be checked for crushing of threads. We know that

Crushing resistance of the threads in the coupler nut

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

where σ_c = Crushing stress induced in the coupler nut, and

n = Number of threads per mm length.

Equating the design load to the crushing resistance of the threads, we have

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

From this expression, the induced σ_c may be checked.

3. *Outside diameter of the coupler nut*

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

Tearing resistance at the coupler nut

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

where σ_t = Permissible tensile stress for the material of the coupler nut.

Equating the axial load to the tearing resistance at the coupler nut, we have

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

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.

4. *Outside diameter of the coupler*

The outside diameter of the coupler (D_2) may be obtained by considering the tearing of the coupler. We know that

Tearing resistance of the coupler

$$= \frac{\pi}{4}\left[(D_2)^2 - (D_1)^2\right] \sigma_t$$

where D_1 = Inside diameter of the coupler. It is generally taken as (d + 6 mm), and

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

Equating the axial load to the tearing resistance of the coupler, we have

$$P = \frac{\pi}{4}\left[(D_2)^2 - (D_1)^2\right] \sigma_t$$

From this expression, the value of D_2 may be calculated. In actual practice, the outside diameter of the coupler (D_2) 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 D_2.

5. The length of the coupler between the nuts (L) depends upon the amount of adjustment required. It is usually taken as 6 d.

6. The thickness of the coupler is usually taken as t = 0.75 d, and thickness of the coupler nut, t_1 = 0.5 d.

Contents

Example 12.9. *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, 37.5 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),

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

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

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,

$$d = 39 \text{ mm}$$ **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),

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

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

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

$$l = d = 39 \text{ mm}$$ **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),

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

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

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

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),

Top

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

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

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

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

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

4. ***Outside diameter of the coupler***

Let $\quad D_2$ = 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),

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

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

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

$D_2 = 60$ mm **Ans.**

5. Length of the coupler between nuts,

$L = 6\,d = 6 \times 39 = 234$ mm **Ans.**

6. Thickness of the coupler,

$t_1 = 0.75\,d = 0.75 \times 39 = 29.25$ say 30 mm **Ans.**

and thickness of the coupler nut,

$t = 0.5\ d = 0.5 \times 39 = 19.5$ say 20 mm **Ans.**

EXERCISES

1. Design a cotter joint to connect two mild steel rods for a pull of 30 kN. The maximum permissible stresses are 55 MPa in tension ; 40 MPa in shear and 70 MPa in crushing. Draw a neat sketch of the joint designed.

   [Ans. d = 22 mm; d_2 = 32 mm ; t = 14 mm ; d_1 = 44 mm ; b = 30 mm ; a = 12 mm ; d_4 = 65 mm ; c = 12 mm ; d_3 = 40 mm ; t_1 = 8 mm]

2. Two rod ends of a pump are joined by means of a cotter and spigot and socket at the ends. Design the joint for an axial load of 100 kN which alternately changes from tensile to compressive. The allowable stresses for the material used are 50 MPa in tension, 40 MPa in shear and 100 MPa in crushing.

   [Ans. d = 51 mm ; d_2 = 62 mm ; t = 16 mm ; d_1 = 72 mm ; b = 78 mm ; a = 20 mm ; d_3 = 83 mm ; d_4 = 125 mm ; c = 16 mm ; t_1 = 13 mm]

3. Two mild steel rods 40 mm diameter are to be connected by a cotter joint. The thickness of the cotter is 12 mm. Calculate the dimensions of the joint, if the maximum permissible stresses are: 46 MPa in tension ; 35 MPa in shear and 70 MPa in crushing.

   [Ans. d_2 = 30 mm ; d_1 = 48 mm ; b = 70 mm ; a = 27.5 mm ; d_4 = 100 mm ; c = 12 mm ; d_3 = 44.2 mm ; t = 35 mm ; t_1 = 13.5 mm]

4. The big end of a connecting rod is subjected to a load of 40 kN. The diameter of the circular part adjacent to the strap is 50 mm.

Design the joint assuming the permissible tensile stress in the strap as 30 MPa and permissible shear stress in the cotter and gib as 20 MPa.

[Ans. B_1 = 50 mm ; t = 15 mm ; t_1 = 15 mm ; t_3 = 22 mm ; B = 70 mm]

5. Design a cotter joint to connect a piston rod to the crosshead. The maximum steam pressure on the piston rod is 35 kN. Assuming that all the parts are made of the same material having the following permissible stresses :

σ_1 = 50 MPa ; τ = 60 MPa and σ_c = 90 MPa.

[Ans. d_2 = 40 mm ; t = 12 mm ; d_3 = 75 mm ; L = 88 mm ; d = 44 mm ; d_1 = 38 mm]

6. Design and draw a cotter foundation bolt to take a load of 90 kN. Assume the permissible stresses as follows :

σ_t = 50 MPa, τ = 60 MPa and σ_c = 100 MPa.

[Ans. d = 50 mm ; d_1 = 60 mm ; t = 15 mm ; b = 60 mm]

7. Design a knuckle joint to connect two mild steel bars under a tensile load of 25 kN. The allowable stresses are 65 MPa in tension, 50 MPa in shear and 83 MPa in crushing.

[Ans. $d = d_1$ = 23 mm ; d_2 = 46 mm ; d_3 = 35 mm ; t = 29 mm ; t_1 = 18 mm]

8. A knuckle joint is required to withstand a tensile load of 25 kN. Design the joint if the permissible stresses are :

σ_t = 56 MPa ; τ = 40 MPa and σ_c = 70 MPa.

[Ans. $d = d_1$ = 28 mm ; d_2 = 56 mm ; d_3 = 42 mm ; t_1 = 21 mm]

9. The pull in the tie rod of a roof truss is 44 kN. Design a suitable adjustable screw joint. The permissible tensile and shear stresses are 75 MPa and 37.5 MPa respectively. Draw full size two suitable views of the joint. **[Ans. d = 36 mm ; l = 11 mm ; D = 45 mm ; D_2 = 58 mm]**

QUESTIONS

1. What is a cotter joint? Explain with the help of a neat sketch, how a cotter joint is made ?
2. What are the applications of a cottered joint ?
3. Discuss the design procedure of spigot and socket cotter joint.
4. Why gibs are used in a cotter joint? Explain with the help of a neat sketch the use of single and double gib.
5. Describe the design procedure of a gib and cotter joint.
6. Distinguish between cotter joint and knuckle joint.
7. Sketch two views of a knuckle joint and write the equations showing the strength of joint for the most probable modes of failure.
8. Explain the purpose of a turn buckle. Describe its design procedure.

OBJECTIVE TYPE QUESTIONS

1. A cotter joint is used to transmit

   (*a*) axial tensile load only
   (*b*) axial compressive load only
   (*c*) combined axial and twisting loads
   (*d*) axial tensile or compressive loads

2. The taper on cotter varies from

   (*a*) 1 in 15 to 1 in 10
   (*b*) 1 in 24 to 1 in 20
   (*c*) 1 in 32 to 1 in 24
   (*d*) 1 in 48 to 1 in 24

3. Which of the following cotter joint is used to connect strap end of a connecting rod ?
   (*a*) Socket and spigot cotter joint (*b*) Sleeve and cotter joint
   (*c*) Gib and cotter joint (*d*) none of these
4. In designing a sleeve and cotter joint, the outside diameter of the sleeve is taken as
   (*a*) 1.5 *d* (*b*) 2.5 *d*
   (*c*) 3 *d* (*d*) 4 *d*

   where *d* = Diameter of the rod.
5. The length of cotter, in a sleeve and cotter joint, is taken as
   (*a*) 1.5 *d* (*b*) 2.5 *d*
   (*c*) 3 *d* (*d*) 4 *d*
6. In a gib and cotter joint, the thickness of gib is ....thickness of cotter.
   (*a*) more than (*b*) less than (*c*) equal to
7. When one gib is used in a gib and cotter joint, then the width of gib should be taken as
   (*a*) 0.45 *B* (*b*) 0.55 *B*
   (*c*) 0.65 *B* (*d*) 0.75 *B*

   where *B* = Total width of gib and cotter.
8. In a steam engine, the piston rod is usually connected to the crosshead by means of a
   (*a*) knuckle joint (*b*) universal joint
   (*c*) flange coupling (*d*) cotter joint
9. In a steam engine, the valve rod is connected to an eccentric by means of a
   (*a*) knuckle joint (*b*) universal joint
   (*c*) flange coupling (*d*) cotter joint
10. In a turn buckle, if one of the rods has left hand threads, then the other rod will have
    (*a*) right hand threads (*b*) left hand threads
    (*c*) pointed threads (*d*) multiple threads

ANSWERS

1. (*d*)	**2.** (*d*)	**3.** (*c*)	**4.** (*b*)	**5.** (*d*)
6. (*c*)	**7.** (*b*)	**8.** (*d*)	**9.** (*a*)	**10.** (*a*)

Page 44