UNIT III

TEMPORARY AND PERMANENT JOINTS

CHAPTER 3

Knuckle Joint

A knuckle joint is used to connect two rods which are under the action of tensile loads. However, if the joint is guided, the rods may support a compressive load. 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.

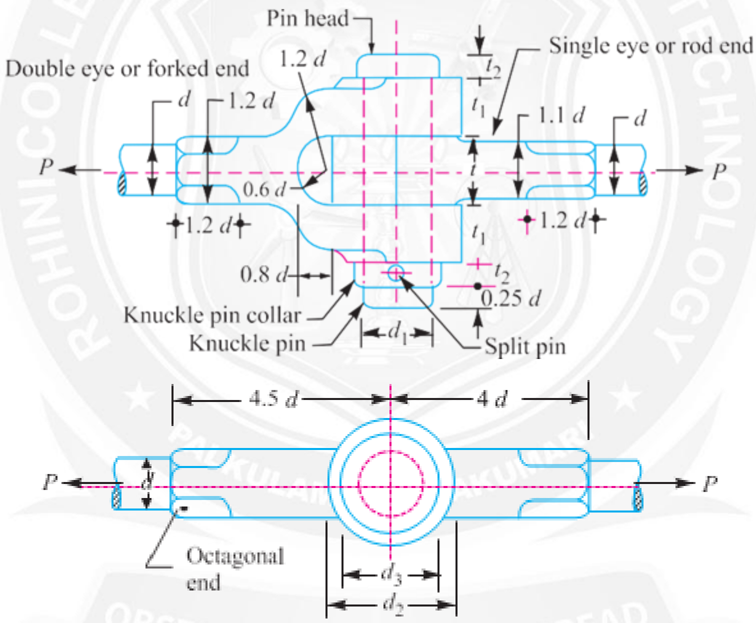

Fig 3.1Kunckle joint.

[Source: "A Textbook of Machine Design by R.S. Khurmi J.K. Gupta, Page: 456]

In knuckle joint (the two views of which are shown in Fig. 3.1), 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.

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, $d_3 = 1.5d$

Thickness of single eye or rod end, $t = 1.25d$

Thickness of fork, $t1 = 0.75d$

Thickness of pin head, $t2 = 0.5d$

Other dimensions of the joint are shown in Fig. 3.1

Methods of Failure of Knuckle Joint

Consider a knuckle joint as shown in Fig. 3.1.

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

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

and the shear strength of the pin

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

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

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

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. 3.1. 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

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

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

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

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

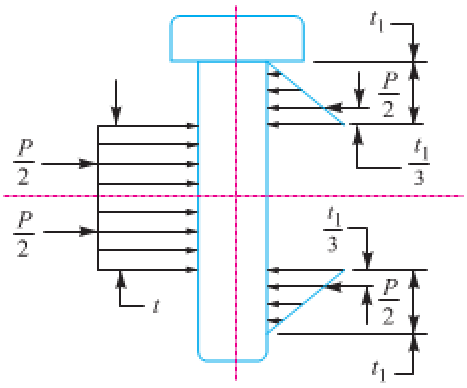

Fig 2.2 Distribution of load on the pin.

[Source: "A Textbook of Machine Design by R.S. Khurmi J.K. Gupta, Page: 458]

∴ Maximum bending (tensile) stress,

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

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

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

$$= (d_2 – d1)\ t \times \sigma_t$$

Equating this to the load (P) we have

$$P = (d_2 – d1)\ t \times \sigma_t$$

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

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

From this equation, the induced crushing stress (σc) 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

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

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.

Design Procedure of Knuckle Joint

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,

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

A little consideration will show that the value of d1 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.

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

Problem 3.1

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.

Given Data:

$P = 150 \text{ kN} = 150 \times 10^3 \text{ N}$

$\sigma_t = 75 \text{ MPa} = 75 \text{ N/mm}^2$

$\tau = 60 \text{ MPa} = 60 \text{ N/mm}^2$

$\sigma_c = 150 \text{ MPa} = 150 \text{ N/mm}^2$

The knuckle joint is shown in Fig. 2.1 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),

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

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

$$150 \times 10^3 = 59\, d^2$$

$$d^2 = 150 \times 10^3 / 59 = 2540 \text{ or}$$

$$d = 50.4 \text{ say } 52 \text{ mm.}$$

Now the various dimensions are fixed as follows:

` Diameter of knuckle pin, $d_1 = d$

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

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

$$d_2 = 104 \text{ mm}$$

Diameter of knuckle pin head and collar, $d_3 = 1.5\, d = 1.5 \times 52$

$$d_3 = 78 \text{ mm}$$

Thickness of single eye or rod end, $t = 1.25\, d = 1.25 \times 52$

$$t = 65 \text{ mm}$$

Thickness of fork, $t_1 = 0.75\, d = 0.75 \times 52$

$$t_1 = 39 \text{ say } 40 \text{ mm}$$

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

$t_2 = 26$ mm

2. Failure of the knuckle pin in shear

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

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

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

$$150 \times 10^3 = 4248\ \tau$$

$$\tau = 150 \times 10^3 / 4248$$

$$\therefore \tau = 35.3\ \text{N/mm}^2 = 35.3\ \text{MPa}$$

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),

$$150 \times 10^3 = (d_2 - d_1)\ t \times \sigma_t$$

$$150 \times 10^3 = (104 - 52)\ 65 \times \sigma_t$$

$$150 \times 10^3 = 3380\ \sigma_t$$

$$\sigma_t = 150 \times 10^3 / 3380$$

$$\therefore \sigma_t = 44.4\ \text{N / mm}^2 = 44.4\ \text{MPa}$$

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),

$$150 \times 10^3 = (d_2 - d_1)\ t \times \tau$$

$$150 \times 10^3 = (104 - 52)\ 65 \times \tau$$

$$150 \times 10^3 = 3380\ \tau$$

$$\therefore \tau = 150 \times 10^3 / 3380$$

$$\tau = 44.4\ \text{N/mm}^2 = 44.4\ \text{MPa}$$

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),

$$150 \times 10^3 = d_1 \times t \times \sigma_c$$

$$150 \times 10^3 = 52 \times 65 \times \sigma_c$$

$$150 \times 10^3 = 3380\ \sigma_c$$

$\sigma_c = 150 \times 10^3 / 3380$

$\therefore \sigma_c = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$

6. Failure of the forked end in tension

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

$150 \times 10^3 = (d_2 - d_1)\ 2\ t_1 \times \sigma_t$

$150 \times 10^3 = (104 - 52)\ 2 \times 40 \times \sigma_t$

$150 \times 10^3 = 4160\ \sigma_t$

$\sigma_t = 150 \times 10^3 / 4160$

$\therefore \sigma_t = 36 \text{ N/mm}^2 = 36 \text{ MPa}$

7. Failure of the forked end in shear

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

$150 \times 10^3 = (d_2 - d_1)\ 2\ t_1 \times \tau$

$150 \times 10^3 = (104 - 52)\ 2 \times 40 \times \tau$

$150 \times 10^3 = 4160\ \tau$

$\tau = 150 \times 10^3 / 4160$

$\therefore \tau = 36 \text{ N/mm}^2 = 36 \text{ MPa}$

8. Failure of the forked end in crushing

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

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

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

$150 \times 10^3 = 4160\ \sigma_c$

$\sigma_c = 150 \times 10^3 / 4180$

$\therefore \sigma_c = 36 \text{ N/mm2} = 36 \text{ MPa}$

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