- Two rods are connected by means of a cotter joint. The inside diameter of the socket and outside diameter of the socket collar are 50 and 100 mm respectively. The rods are subjected to a tensile force of 50 kN. The cotter is made of steel 30C8 (Syt = 400 N/mm2) and the factor of safety is 4. The width of the cotter is fi ve times of thickness. Calculate: (i) width and thickness of the cotter on the basis of shear failure; and (ii) width and thickness of the cotter on the basis of bending failure.
- Solution Given Syt = 400 N/mm2 (fs) = 4 P = (50 × 103) N d4 = 100 mm d2 = 50 mm

Step I Permissible stresses for cotter

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{400}{4} = 100 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5S_{yt}}{(fs)} = \frac{0.5(400)}{4} = 50 \text{ N/mm}^2$$

Step II Width and thickness on the basis of shear failure

$$b = 5t$$

From Eq. (4.25e),

$$P = 2bt\tau \quad \text{or} \quad 50 \times 10^3 = 2\,(5t)\,t\,(50)$$

$$\therefore \quad t = 10 \text{ mm} \quad \text{and} \quad b = 5t = 50 \text{ mm} \qquad \text{(i)}$$

Step III Width and thickness on the basis of bending failure

$$d_4 = 100 \text{ mm} \qquad d_2 = 50 \text{ mm}$$

From Eq. (4.25j),

$$\sigma_b = \frac{\dfrac{P}{2}\left[\dfrac{d_2}{4} + \dfrac{d_4 - d_2}{6}\right]\dfrac{b}{2}}{\left(\dfrac{tb^3}{12}\right)}$$

$$100 = \frac{\dfrac{50 \times 10^3}{2}\left[\dfrac{50}{4} + \dfrac{(100 - 50)}{6}\right]\dfrac{(5t)}{2}}{\left[\dfrac{t(5t)^3}{12}\right]}$$

$$\therefore \ t = 10.77 \text{ or } 12 \text{ mm} \quad \text{and} \quad b = 5t = 60 \text{ mm} \qquad \text{(ii)}$$

- Two rods, made of plain carbon steel 40C8 (Syt = 380 N/mm2), are to be connected by means of a cotter joint. The diameter of each rod is 50 mm and the cotter is made from a steel plate of 15 mm thickness. Calculate the dimensions of the socket end making the following assumptions: (i) the yield strength in compression is twice of the tensile yield strength; and (ii) the yield strength in shear is 50% of the tensile yield strength. The factor of safety is 6.
- Solution Given Syt = 380 N/mm2 (fs) = 6 t = 15 mm d = 50 mm

Step I Permissible stresses

$$\sigma_c = \frac{S_{yc}}{(fs)} = \frac{2S_{yt}}{(fs)} = \frac{2(380)}{6} = 126.67 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5S_{yt}}{(fs)} = \frac{0.5(380)}{6} = 31.67 \text{ N/mm}^2$$

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{380}{6} = 63.33 \text{ N/mm}^2$$

Step II Load acting on rods

$$P = \frac{\pi}{4} d^2 \sigma_t \quad \text{or} \quad P = \frac{\pi}{4}(50)^2 (63.33) = 124\,348.16 \text{ N}$$

Step III Inside diameter of socket (d_2)

From Eq. (4.25b),

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

$$124\,348.16 = \left[\frac{\pi}{4} d_2^2 - d_2(15)\right](63.33)$$

or $\quad d_2^2 - 19.1 d_2 - 2500 = 0$

Solving the above quadratic equation,

$$d_2 = \frac{19.1 \pm \sqrt{19.1^2 - 4(-2500)}}{2}$$

$\therefore \quad d_2 = 60.45$ or 65 mm (i)

Step IV Outside diameter of socket (d_1)

From Eq. (4.25d),

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

$$124\,348.16 = \left[\frac{\pi}{4}(d_1^2 - 65^2) - (d_1 - 65)\,(15)\right](63.33)$$

or $\quad d_1^2 - 19.1\,d_1 - 5483.59 = 0$

Solving the above quadratic equation,

$$d_1 = \frac{19.1 \pm \sqrt{19.1^2 - 4(-5483.59)}}{2}$$

$\therefore \quad d_1 = 84.21$ or 85 mm (ii)

Step V Diameter of socket collar (d_4)

From Eq. (4.25i),

$$\sigma_c = \frac{P}{(d_4 - d_2)\,t}$$

or $\quad 126.67 = \dfrac{124\,348.16}{(d_4 - 65)(15)}$

$\therefore \quad d_4 = 130.44$ or 135 mm (iii)

Step VI Dimensions a and c

From Eq. (4.25f),

$$a = \frac{P}{2\,d_2 \tau} = \frac{124\,348.16}{2\,(65)\,(31.67)} = 30.20 \text{ or } 35 \text{ mm} \quad \text{(iv)}$$

From Eq. (4.25g),

$$c = \frac{P}{2\,(d_4 - d_2)\tau} = \frac{124\,348.16}{2\,(135 - 65)\,(31.67)} = 28.04 \text{ or } 30 \text{ mm} \quad \text{(v)}$$

- KNUCKLE JOINT
- Knuckle joint is used to connect two rods whose axes either coincide or intersect and lie in one plane. The knuckle joint is used to transmit axial tensile force. The construction of this joint permits limited angular movement between rods, about the axis of the pin. This type of joint is popularmachines and structures. Typical applications of knuckle joints are as follows:
- (i) Joints between the tie bars in roof trusses.
- (ii) Joints between the links of a suspension bridge.
- (iii) Joints in valve mechanism of a reciprocating engine.
- (iv) Fulcrum for the levers.
- (v) Joints between the links of a bicycle chain. A knuckle joint is unsuitable to connect two rotating shafts, which transmit torque. The construction of a knuckle joint, used to connect two rods A and B subjected to tensile force P, is shown in Fig. 4.18. An eye is formed at the end of rod- B, while a fork is formed at the end of rod-A.
- The eye fi ts inside the fork and a pin passes through both the fork and the eye. This pin is secured in its place by means of a split-pin. Due to this type of construction, a knuckle joint is sometimes called a forked-pin joint. In rare applications, a knuckle joint is used to connect three rods—two with forks and a third with the eye. in The knuckle joint offers the following advantages:
- (i) The joint is simple to design and manufacture.
- (ii) There are a few parts in the knuckle joint, which reduces cost and improves reliability.
- (iii) The assembly or dismantling of the parts of a knuckle joint is quick and simple.
- The assembly consists of inserting the eye of one rod inside the fork of the other rod and putting the pin in their common hole and fi nally putting the split-pin to hold the pin. Dismantling consists of removing the split-pin and taking the pin out of the eye and the fork.

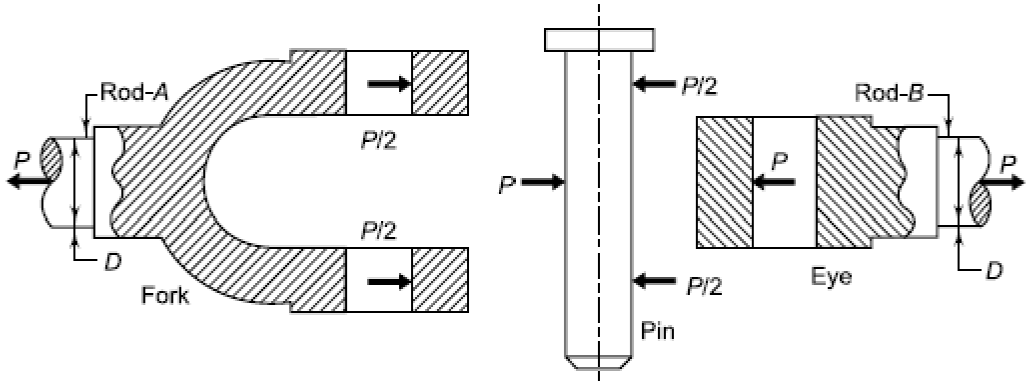

Fig. 4.19 *Free Body Diagram of Forces*

In Fig. 4.18, the following notations are used. D = diameter of each rod (mm) D1 = enlarged diameter of each rod (mm) d = diameter of knuckle pin (mm) d0 = outside diameter of eye or fork (mm) a = thickness of each eye of fork (mm) b = thickness of eye end of rod-B (mm) d1 = diameter of pin head (mm) x = distance of the centre of fork radius R from the eye (mm) For the purpose of stress analysis of a knuckle joint, the following assumptions are made:

(i) The rods are subjected to axial tensile force.

(ii) The effect of stress concentration due to holes is neglected.

(iii) The force is uniformly distributed in various parts. Free body diagram of forces acting on three components of the knuckle joint, viz., fork, pin and eye is shown in Fig. 4.19. This diagram is constructed by using the principle that actions and reactions are equal and opposite. The forces are determined in the following way,

(i) Consider rod-A with the fork end. The rod is subjected to a horizontal force P to the left. The sum of all horizontal forces acting on rod-A must be equal to zero. Therefore, there should be a force P to the right acting on the fork end. The force P is divided into two parts, each equal to (P/2) on the fork end.

(ii) (ii) Consider rod-B with the eye end. The rod is subjected to a horizontal force P to the right side. The sum of all horizontal forces acting on rod-B must be equal to zero. Therefore, there should be a force P to the left acting on the eye end. Fig. 4.19 Free Body Diagram of Forces

(iii) (iii) The forces shown on the pin are equal and opposite reactions of forces acting on the fork end of rod-A and the eye end of rod-B. In order to fi nd out various dimensions of the parts of a knuckle joint, failures in different parts and at different cross-sections are considered. For each type of failure, one strength equation is written. Finally, these strength equations are used to fi nd out various dimensions of the knuckle joint.

- (i) Tensile Failure of Rods Each rod is subjected
- to a tensile force P. The tensile stress in the rod is
- given by,

$$\sigma_t = \frac{P}{\left(\frac{\pi}{4} D^2\right)} \quad \text{or} \quad D = \sqrt{\frac{4P}{\pi \sigma_t}} \qquad (4.26a)$$

where σ_t is the permissible tensile stress for the rods. The enlarged diameter D_1 of the rod near the joint is determined by the following empirical relationship,

$$D_1 = 1.1\, D \qquad (4.26b)$$

(ii) Shear Failure of Pin The pin is subjected to double shear as shown in Fig. 4.20. The area of each of the two planes that resist shear failure is $\left(\frac{\pi}{4} d^2\right)$. Therefore, shear stress in the pin is given by,

$$\tau = \frac{P}{2\left(\frac{\pi}{4} d^2\right)} \quad \text{or} \quad d = \sqrt{\frac{2P}{\pi \tau}} \qquad (4.26c)$$

where τ is the permissible shear stress for the pin. The standard proportion for the diameter of the pin is as follows,

$$d = D \qquad (4.26d)$$

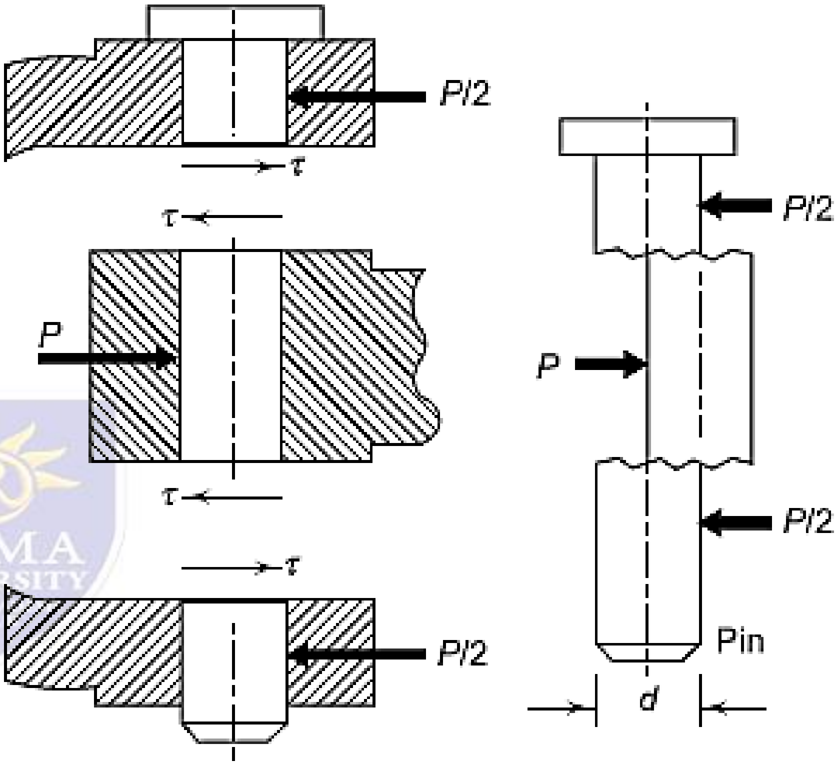

Fig. 4.20 *Shear Failure of Pin*