- DESIGN OF SIMPLE MACHINE PARTS

- design of simple machine parts is illustrated with the help of examples. The following points should be remembered in solving such problems:
- (i) The dimensions of simple machine parts are determined on the basis of pure tensile stress, pure compressive stress, direct shear stress, bending stress or torsional shear stress. The analysis is simple but approximate, because a number of factors such as principal stresses, stress concentration, and reversal of stresses is neglected. Therefore, a higher factor of safety of up to 5 is taken to account for these factors.
- (ii) It is incorrect to assume allowable stress as data for design. The allowable stress is to be obtained from published values of ultimate tensile strength and yield strength for a given material by Eqs (4.1) and (4.2) respectively.
- (iii) It will be proved at a later stage that according to maximum shear stress theory, the yield strength in shear is 50% of the yield strength in tension. Therefore, Ssy = 0.5 Syt and the permissible shear stress (t) is given by

$$\tau = \frac{S_{sy}}{(fs)}$$

- The above value of allowable shear stress is used in determination of dimensions of the component. The design analysis in this chapter is approximate and incomplete. The purpose of such problems is to illustrate the application of design equations to fi nd out the geometric dimensions of the component. In practice, much more stress analysis is involved in the design of machine parts.

- Two plates, subjected to a tensile force of 50 kN, are fi xed together by means of three rivets as shown in Fig. 4.10 (a). The plates and rivets are made of plain carbon steel 10C4 with a tensile yield strength of 250 N/mm2. The yield strength in shear is 50% of the tensile yield strength, and the factor of safety is 2.5. Neglecting stress concentration, determine (i) the diameter of the rivets; and (ii) the thickness of the plates.

Step I Permissible shear stress for rivets

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5\,S_{yt}}{(fs)} = \frac{0.5\,(250)}{(2.5)} = 50\ \text{N/mm}^2$$

Step II Diameter of rivets

Since there are three rivets,

$$3\left[\frac{\pi}{4}d^2\right]\tau = P \quad \text{or} \quad 3\left[\frac{\pi}{4}d^2\right]50 = 50\times10^3$$

$$\therefore\ d = 20.60 \text{ or } 22\text{ mm} \qquad \text{(i)}$$

Step III Permissible tensile stress for plates

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{250}{2.5} = 100\ \text{N/mm}^2$$

Step IV Thickness of plates

As shown in Fig. 4.10(b),

$$\sigma_t\,(200 - 3d)\,t = P$$

$$\text{or} \quad 100(200 - 3\times22)\,t = 50\times10^3$$

$$\therefore\ t = 3.73 \text{ or } 4\text{ mm} \qquad \text{(ii)}$$

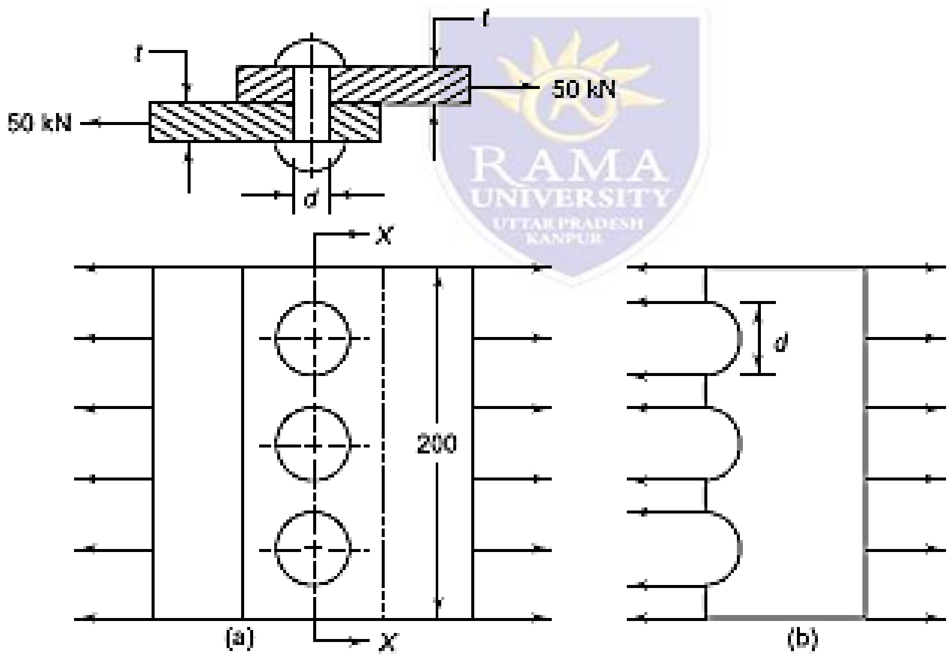

Fig. 4.10 *(a) Riveted Joint (b) Tensile Stress in Plate*

Lecture Machine Design

- COTTER JOINT
- A cotter joint is used to connect two co-axial rods, which are subjected to either axial tensile force or axial compressive force. It is also used to connect a rod on one side with some machine part like a crosshead or base plate on the other side. It is not used for connecting shafts that rotate and transmit torque. Typical applications of cotter joint are as follows:
- (i) Joint between the piston rod and the crosshead of a steam engine
- (ii) Joint between the slide spindle and the fork of the valve mechanism
- (iii) Joint between the piston rod and the tail or pump rod Step III Permissible tensile stress for plates st = S fs yt = 250 = 2 5 100 2 N/mm Step IV Thickness of plates As shown in Fig. 4.10(b), st (200 – 3d) t = P or 100(200 – 3 × 22) t = 50 × 103 \ t = 3.73 or 4 mm (ii) Fig. 4.10 (a) Riveted Joint (b) Tensile Stress in Plate
- (iv) Foundation bolt The principle of wedge action is used in a cotter joint. A cotter is a wedge-shaped piece made of a steel plate. The joint is tightened and adjusted by means of a wedge action of the cotter. The construction of a cotter joint, used to connect two rods A and B is shown in Fig. 4.11. Rod-A is provided with a socket end, while rod-B is provided with a spigot end. The socket end of rod-A fi ts over the spigot end of rod-B. The socket as well as the spigot is provided with a narrow rectangular slot. A cotter is tightly fi tted in this slot passing through the socket and the spigot. The cotter has uniform thickness and the width dimension b is given a slight taper. The taper is usually 1 in 24. The taper is provided for the following two reasonst.
- (i) When the cotter is inserted in the slot through the socket and the spigot and pressed by means of hammer, it becomes tight due to wedge action. This ensures tightness of the joint in operating condition and prevents loosening of the parts
- . (ii) Due to its taper shape, it is easy to remove the cotter and dismantle the joint.
- The amount by which the two rods are drawn together is called the draw of the cotter. The cotter joint offers the following advantages:
- (i) The assembly and dismantling of parts of the cotter joint is quick and simple. The assembly consists of inserting the spigot end into the socket end and putting the cotter into their common slot. When the cotter is hammered, the rods are drawn together and tightened. Dismantling consists of removing the cotter from the slot by means of a hammer.
- (ii) The wedge action develops a very high tightening force, which prevents loosening of parts in service.
- (iii) The joint is simple to design and manufacture.

- Free body diagram of forces acting on three components of cotter joint, viz., socket, cotter and spigot is shown in Fig. 4.12. 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 a socket end. The rod is subjected to a horizontal force P to the left. The sum of all horizontal forces acting on the rod A with socket must be equal to zero. Therefore, there should be a force P to the right acting on the socket. This force is shown by two parts, each equal to (P/2) on the socket end.
- (ii) Consider rod-B with the spigot end. The rod is subjected to a force P to the right. 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 on the spigot end.
- (iii) The forces shown on the cotter are equal and opposite reactions of forces acting on the spigot end of rod-B and the socket end of rod-A.

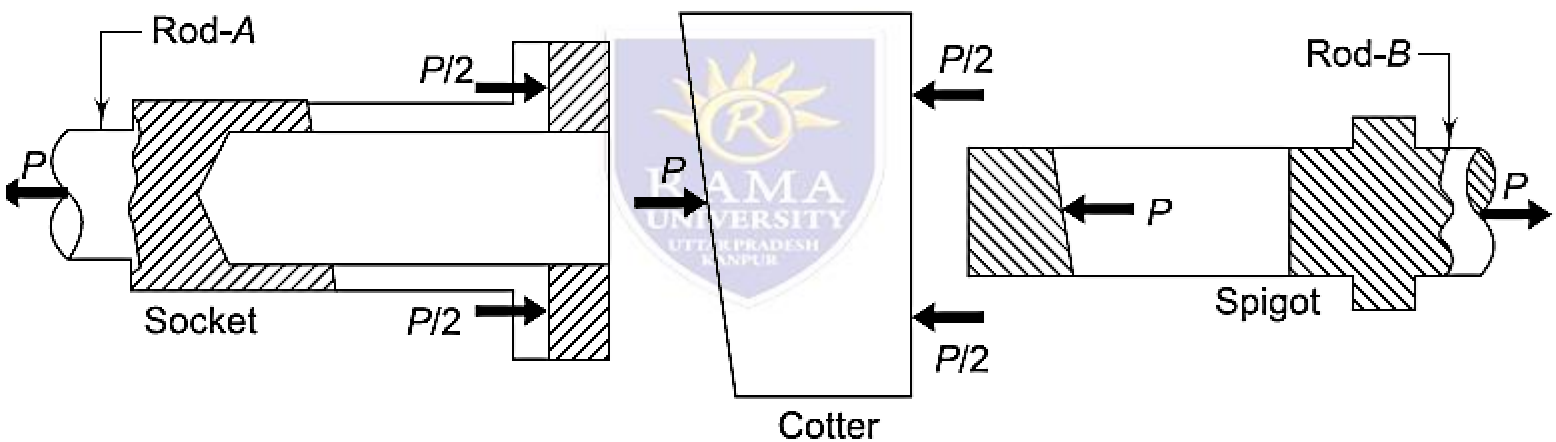

Fig. 4.12 *Free Body Diagram of Forces*

- For the purpose of stress analysis, the following assumptions are made:
- (i) The rods are subjected to axial tensile force.
- (ii) The effect of stress concentration due to the slot is neglected.
- (iii) The stresses due to initial tightening of the cotter are neglected.
- In Fig. 4.11, the following notations are used
- P = tensile force acting on rods (N)
- d = diameter of each rod (mm)
- d1 = outside diameter of socket (mm)
- d2 = diameter of spigot or inside diameter of socket (mm)
- d3 = diameter of spigot-collar (mm)
- d4 = diameter of socket-collar (mm)
- a = distance from end of slot to the end of spigot on rod- (mm)
- b = mean width of cotter (mm)
- c = axial distance from slot to end of socket collar (mm)
- t = thickness of cotter (mm)
- t1 = thickness of spigot-collar (mm)
- l = length of cotter (mm)
- In order to design the cotter joint and fi nd out
- the above dimensions, failures in different parts and
- at different cross-sections are considered. Based
- on each type of failure, one strength equation iswritten.
- Finally, these strength equations are used
- to determine various dimensions of the cotter joint.

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

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

or $$d = \sqrt{\frac{4P}{\pi \sigma_t}} \qquad (4.25a)$$

where σ_t is the permissible tensile stress for the rods.

(ii) Tensile Failure of Spigot Figure 4.13(a) shows the weakest cross-section at XX of the spigot end, which is subjected to tensile stress.

$$\text{Area of section at } XX = \left[\frac{\pi}{4} d_2^2 - d_2 t\right]$$

Therefore, tensile stress in the spigot is given by,

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

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

From the above equation, the diameter of spigot or inner diameter of socket (*d*2) can be determined by assuming a suitable value of *t*. The thickness of the cotter is usually determined by the following empirical relationship,

t = 0.31*d*