Example 11.8. *The cylinder head of a steam engine is subjected to a steam pressure of 0.7 N/mm². It is held in position by means of 12 bolts. A soft copper gasket is used to make the joint leak-proof. The effective diameter of cylinder is 300 mm. Find the size of the bolts so that the stress in the bolts is not to exceed 100 MPa.*

Solution. Given: $p = 0.7$ N/mm^2 ; $n = 12$; $D = 300$ mm ; $\sigma_t = 100$ MPa $= 100$ N/mm^2

We know that the total force (or the external load) acting on the cylinder head *i.e.* on 12 bolts,

$$= \frac{\pi}{4}(D)^2 \; p = \frac{\pi}{4}(300)^2 \; 0.7 = 49\,490 \text{ N}$$

∴ External load on the cylinder head per bolt,

$$P_2 = 49\,490 / 12 = 4124 \text{ N}$$

Let d = Nominal diameter of the bolt, and

d_c = Core diameter of the bolt.

We know that initial tension due to tightening of bolt,

$$P_1 = 2840 \, d \text{ N} \qquad \text{... (where } d \text{ is in mm)}$$

From Table 11.2, we find that for soft copper gasket with long through bolts, the minimum value of $K = 0.5$.

∴ Resultant axial load on the bolt,

$$P = P_1 + K \,.\, P_2 = 2840\, d + 0.5 \times 4124 = (2840\, d + 2062) \text{ N}$$

We know that load on the bolt (P),

$$2840\, d + 2062 = \frac{\pi}{4}(d_c)^2 \; \sigma_t = \frac{\pi}{4}(0.84d)^2 \; 100 = 55.4\, d^2 \qquad \text{...(Taking } d_c = 0.84\, d)$$

$$\therefore \quad 55.4\, d^2 - 2840d - 2062 = 0$$

or

$$d^2 - 51.3d - 37.2 = 0$$

$$\therefore \quad d = \frac{51.3 \pm \sqrt{(51.3)^2 + 4 \times 37.2}}{2} = \frac{51.3 \pm 52.7}{2} = 52 \text{ mm}$$

..(Taking + ve sign)

Thus, we shall use a bolt of size M 52. **Ans.**

Example 11.9. *A steam engine of effective diameter 300 mm is subjected to a steam pressure of 1.5 N/mm². The cylinder head is connected by 8 bolts having yield point 330 MPa and endurance limit at 240 MPa. The bolts are tightened with an initial preload of 1.5 times the steam load. A soft copper gasket is used to make the joint leak-proof. Assuming a factor of safety 2, find the size of bolt required. The stiffness factor for copper gasket may be taken as 0.5.*

Solution. Given : $D = 300$ mm ; $p = 1.5$ N/mm^2 ; $n = 8$; $\sigma_y = 330$ MPa $= 330$ N/mm^2; $\sigma_e = 240$ MPa $= 240$ N/mm^2 ; $P_1 = 1.5\, P_2$; $F.S. = 2$; $K = 0.5$

We know that steam load acting on the cylinder head,

$$P_2 = \frac{\pi}{4}(D)^2 \; p = \frac{\pi}{4}(300)^2 \; 1.5 = 106\,040 \text{ N}$$

∴ Initial pre-load,

$$P_1 = 1.5\, P_2 = 1.5 \times 106\,040 = 159\,060 \text{ N}$$

We know that the resultant load (or the maximum load) on the cylinder head,

$$P_{max} = P_1 + K.P_2 = 159\,060 + 0.5 \times 106\,040 = 212\,080 \text{ N}$$

This load is shared by 8 bolts, therefore maximum load on each bolt,

$$P_{max} = 212\ 080 / 8 = 26\ 510 \text{ N}$$

and minimum load on each bolt,

$$P_{min} = P_1 / n = 159\ 060/8 = 19\ 882 \text{ N}$$

We know that mean or average load on the bolt,

$$P_m = \frac{P_{max} + P_{min}}{2} = \frac{26\ 510 + 19\ 882}{2} = 23\ 196 \text{ N}$$

and the variable load on the bolt,

$$P_v = \frac{P_{max} - P_{min}}{2} = \frac{26\ 510 - 19\ 882}{2} = 3314 \text{ N}$$

Let $\quad d_c$ = Core diameter of the bolt in mm.

$\therefore$ Stress area of the bolt,

$$A_s = \frac{\pi}{4} (d_c)^2 = 0.7854\ (d_c)^2 \text{ mm}^2$$

We know that mean or average stress on the bolt,

$$\sigma_m = \frac{P_m}{A_s} = \frac{23\ 196}{0.7854\ (d_c)^2} = \frac{29\ 534}{(d_c)^2} \text{ N/mm}^2$$

and variable stress on the bolt,

and variable stress on the bolt,

$$\sigma_v = \frac{P_v}{A_s} = \frac{3314}{0.7854\ (d_c)^2} = \frac{4220}{(d_c)^2} \text{ N/mm}^2$$

According to *Soderberg's formula, the variable stress,

$$\sigma_v = \sigma_e \left(\frac{1}{F.S} - \frac{\sigma_m}{\sigma_y} \right)$$

$$\frac{4220}{(d_c)^2} = 240 \left(\frac{1}{2} - \frac{29\ 534}{(d_c)^2\ 330} \right) = 120 - \frac{21\ 480}{(d_c)^2}$$

or $\quad \dfrac{4220}{(d_c)^2} + \dfrac{21\ 480}{(d_c)^2} = 120 \quad$ or $\quad \dfrac{25\ 700}{(d_c)^2} = 120$

$\therefore \quad (d_c)^2 = 25\ 700 / 120 = 214 \quad$ or $\quad d_c = 14.6$ mm

From Table 11.1 (coarse series), the standard core diameter is d_c = 14.933 mm and the corresponding size of the bolt is M18. **Ans.**

11.17 Design of a Nut

When a bolt and nut is made of mild steel, then the effective height of nut is made equal to the nominal diameter of the bolt. If the nut is made of weaker material than the bolt, then the height of nut should be larger, such as 1.5 *d* for gun metal, 2 *d* for cast iron and 2.5 *d* for aluminium alloys (where *d* is the nominal diameter of the bolt). In case cast iron or aluminium nut is used, then *V*-threads are permissible only for permanent fastenings, because threads in these materials are damaged due to repeated screwing and unscrewing. When these materials are to be used for parts frequently removed and fastened, a screw in steel bushing for cast iron and cast-in-bronze or monel metal insert should be used for aluminium and should be drilled and tapped in place.

11.18 Bolted Joints under Eccentric Loading

There are many applications of the bolted joints which are subjected to eccentric loading such as a wall bracket, pillar crane, etc. The eccentric load may be

1. Parallel to the axis of the bolts,
2. Perpendicular to the axis of the bolts, and
3. In the plane containing the bolts.

We shall now discuss the above cases, in detail, in the following articles.

11.19 Eccentric Load Acting Parallel to the Axis of Bolts

Consider a bracket having a rectangular base bolted to a wall by means of four bolts as shown in Fig. 11.31. A little consideration will show that each bolt is subjected to a direct tensile load of

$W_{t1} = \frac{W}{n}$, where n is the number of bolts.

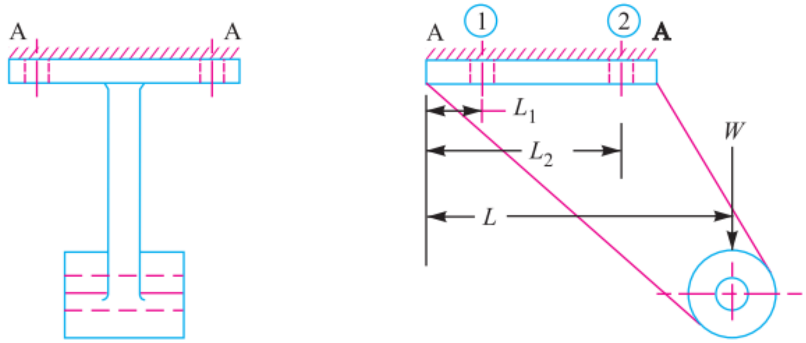

Fig. 11.31. Eccentric load acting parallel to the axis of bolts.

Further the load W tends to rotate the bracket about the edge A-A. Due to this, each bolt is stretched by an amount that depends upon its distance from the tilting edge. Since the stress is a function of *elongation, therefore each bolt will experience a different load which also depends upon the distance from the tilting edge. For convenience, all the bolts are made of same size. In case the flange is heavy, it may be considered as a rigid body.

Let w be the load in a bolt per unit distance due to the turning effect of the bracket and let W_1 and W_2 be the loads on each of the bolts at distances L_1 and L_2 from the tilting edge.

$\therefore$ Load on each bolt at distance L_1,

$$W_1 = w.L_1$$

and moment of this load about the tilting edge

$$= w_1.L_1 \times L_1 = w\,(L_1)^2$$

Similarly, load on each bolt at distance L_2,

$$W_2 = w.L_2$$

and moment of this load about the tilting edge

$$= w.L_2 \times L_2 = w\,(L_2)^2$$

$\therefore$ Total moment of the load on the bolts about the tilting edge

$$= 2w\,(L_1)^2 + 2w\,(L_2)^2 \qquad \text{...(i)}$$

... ($\because$ There are two bolts each at distance of L_1 and L_2)

Also the moment due to load W about the tilting edge

$$= W.L \qquad \text{...(ii)}$$

From equations **(*i*)** and **(*ii*)**, we have

$$W.L = 2w\,(L_1)^2 + 2w(L_2)^2 \quad \text{or} \quad w = \frac{W.L}{2\,[(L_1)^2 + (L_2)^2]} \qquad \text{...(iii)}$$

It may be noted that the most heavily loaded bolts are those which are situated at the greatest distance from the tilting edge. In the case discussed above, the bolts at distance L_2 are heavily loaded.

$\therefore$ Tensile load on each bolt at distance L_2,

$$W_{t2} = W_2 = w.L_2 = \frac{W.L.L_2}{2[(L_1)^2 + (L_2)^2]} \qquad \text{... [From equation (iii)]}$$

and the total tensile load on the most heavily loaded bolt,

$$W_t = W_{t1} + W_{t2} \qquad \text{...(iv)}$$

If d_c is the core diameter of the bolt and σ_t is the tensile stress for the bolt material, then total tensile load,

$$W_t = \frac{\pi}{4}\,(d_c)^2\,\sigma_t \qquad \text{...(v)}$$

From equations **(*iv*)** and **(*v*),** the value of d_c may be obtained.