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/mm² ; $n = 12$; $D = 300$ mm ; $\sigma_t = 100$ MPa = 100 N/mm²

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/mm² ; $n = 8$; $\sigma_y = 330$ MPa = 330 N/mm²; $\sigma_e = 240$ MPa = 240 N/mm² ; $P_1 = 1.5\ P_2$; $F.S. = 2$; $K = 0.5$

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

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

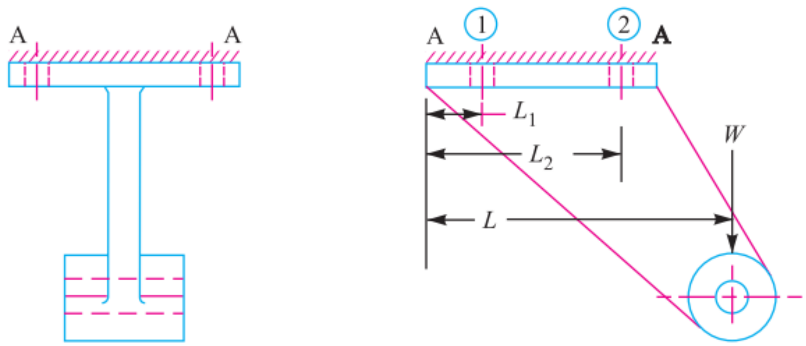

Fig. 11.31. Eccentric load acting parallel to the axis of bolts.

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

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

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

Also the moment due to load W about the tilting edge

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

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

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

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,

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

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

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

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,

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

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

Example 11.12. *A bracket, as shown in Fig. 11.31, supports a load of 30 kN. Determine the size of bolts, if the maximum allowable tensile stress in the bolt material is 60 MPa. The distances are :*

L_1 = 80 mm, L_2 = 250 mm, and L = 500 mm.

Solution. Given : W = 30 kN ; σ_t = 60 MPa = 60 N/mm^2 ; L_1 = 80 mm ; L_2 = 250 mm ; L = 500 mm

We know that the direct tensile load carried by each bolt,

$$W_{t1} = \frac{W}{n} = \frac{30}{4} = 7.5 \text{ kN}$$

and load in a bolt per unit distance,

$$w = \frac{W.L}{2[(L_1)^2 + (L_2)^2]} = \frac{30 \times 500}{2[(80)^2 + (250)^2]} = 0.109 \text{ kN/mm}$$

Since the heavily loaded bolt is at a distance of L_2 mm from the tilting edge, therefore load on the heavily loaded bolt,

$$W_{t2} = w.L_2 = 0.109 \times 250 = 27.25 \text{ kN}$$

∴ Maximum tensile load on the heavily loaded bolt,

$$W_t = W_{t1} + W_{t2} = 7.5 + 27.25 = 34.75 \text{ kN} = 34\ 750 \text{ N}$$

Let d_c = Core diameter of the bolts.

We know that the maximum tensile load on the bolt (W_t),

$$34\ 750 = \frac{\pi}{4}(d_c)^2\ \sigma_t = \frac{\pi}{4}(d_c)^2\ 60 = 47\ (d_c)^2$$

or d_c = 27.2 mm

From Table 11.1 (coarse series), we find that the standard core diameter of the bolt is 28.706 mm and the corresponding size of the bolt is M 33. **Ans.**

11.20 Eccentric Load Acting Perpendicular to the Axis of Bolts

A wall bracket carrying an eccentric load perpendicular to the axis of the bolts is shown in Fig. 11.34.

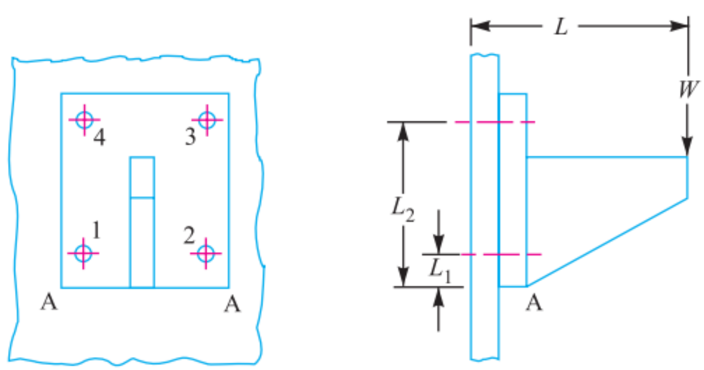

Fig. 11.34. Eccentric load perpendicular to the axis of bolts.

In this case, the bolts are subjected to direct shearing load which is equally shared by all the bolts. Therefore direct shear load on each bolts,

$$W_s = W/n, \text{ where } n \text{ is number of bolts.}$$

A little consideration will show that the eccentric load W will try to tilt the bracket in the clockwise direction about the edge A-A. As discussed earlier, the bolts will be subjected to tensile stress due to the turning moment. The maximum tensile load on a heavily loaded bolt (W_t) may be obtained in the similar manner as discussed in the previous article. In this case, bolts 3 and 4 are heavily loaded.

∴ Maximum tensile load on bolt 3 or 4,

$$W_{t2} = W_t = \frac{W.L.L_2}{2\,[(L_1)^2 + (L_2)^2]}$$

When the bolts are subjected to shear as well as tensile loads, then the equivalent loads may be determined by the following relations :

Equivalent tensile load,

$$W_{te} = \frac{1}{2}\left[W_t + \sqrt{(W_t)^2 + 4(W_s)^2}\right]$$

and equivalent shear load,

$$W_{se} = \frac{1}{2}\left[\sqrt{(W_t)^2 + 4(W_s)^2}\right]$$

Knowing the value of equivalent loads, the size of the bolt may be determined for the given allowable stresses.

Example 11.14. *For supporting the travelling crane in a workshop, the brackets are fixed on steel columns as shown in Fig. 11.35. The maximum load that comes on the bracket is 12 kN acting vertically at a distance of 400 mm from the face of the column. The vertical face of the bracket is secured to a column by four bolts, in two rows (two in each row) at a distance of 50 mm from the lower edge of the bracket. Determine the size of the bolts if the permissible value of the tensile stress for the bolt material is 84 MPa. Also find the cross-section of the arm of the bracket which is rectangular.*

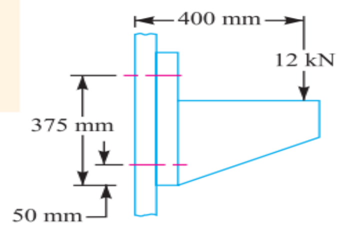

Fig. 11.35

Solution. Given : $W = 12$ kN $= 12 \times 10^3$ N ; $L = 400$ mm ; $L_1 = 50$ mm ; $L_2 = 375$ mm ; $\sigma_t = 84$ MPa $= 84$ N/mm^2 ; $n = 4$

We know that direct shear load on each bolt,

$$W_s = \frac{W}{n} = \frac{12}{4} = 3 \text{ kN}$$

Since the load W will try to tilt the bracket in the clockwise direction about the lower edge, therefore the bolts will be subjected to tensile load due to turning moment. The maximum loaded bolts are 3 and 4 (See Fig. 11.34), because they lie at the greatest distance from the tilting edge A–A (*i.e.* lower edge).

We know that maximum tensile load carried by bolts 3 and 4,

$$W_t = \frac{W.L.L_2}{2\,[(L_1)^2 + (L_2)^2]} = \frac{12 \times 400 \times 375}{2\,[(50)^2 + (375)^2]} = 6.29 \text{ kN}$$

Since the bolts are subjected to shear load as well as tensile load, therefore equivalent tensile load,

$$W_{te} = \frac{1}{2}\left[W_t + \sqrt{(W_t)^2 + 4(W_s)^2}\right] = \frac{1}{2}\left[6.29 + \sqrt{(6.29)^2 + 4 \times 3^2}\right] \text{kN}$$

$$= \frac{1}{2}(6.29 + 8.69) = 7.49 \text{ kN} = 7490 \text{ N}$$

Size of the bolt

Let d_c = Core diameter of the bolt.

We know that the equivalent tensile load (W_{te}),

$$7490 = \frac{\pi}{4}(d_c)^2 \sigma_t = \frac{\pi}{4}(d_c)^2\, 84 = 66\,(d_c)^2$$

$$\therefore \quad (d_c)^2 = 7490 / 66 = 113.5 \quad \text{or} \quad d_c = 10.65 \text{ mm}$$

From Table 11.1 (coarse series), the standard core diameter is 11.546 mm and the corresponding size of the bolt is M 14. **Ans.**

Cross-section of the arm of the bracket

Let t and b = Thickness and depth of arm of the bracket respectively.

∴ Section modulus,

$$Z = \frac{1}{6}\, t.b^2$$

Assume that the arm of the bracket extends upto the face of the steel column. This assumption gives stronger section for the arm of the bracket.

∴ Maximum bending moment on the bracket,

$$M = 12 \times 10^3 \times 400 = 4.8 \times 10^6 \text{ N-mm}$$

We know that the bending (tensile) stress (σ_t),

$$84 = \frac{M}{Z} = \frac{4.8 \times 10^6 \times 6}{t.b^2} = \frac{28.8 \times 10^6}{t.b^2}$$

$$\therefore \quad t.b^2 = 28.8 \times 10^6 / 84 = 343 \times 10^3 \quad \text{or} \quad t = 343 \times 10^3 / b^2$$

Assuming depth of arm of the bracket, b = 250 mm, we have

$$t = 343 \times 10^3 / (250)^2 = 5.5 \text{ mm}$$ **Ans.**

11.21 Eccentric Load on a Bracket with Circular Base

Sometimes the base of a bracket is made circular as in case of a flanged bearing of a heavy machine tool and pillar crane etc. Consider a round flange bearing of a machine tool having four bolts as shown in Fig. 11.40.

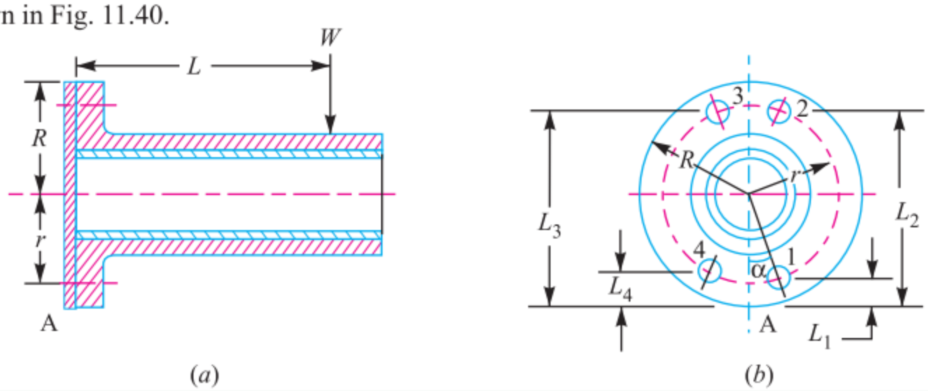

Fig. 11.40. Eccentric load on a bracket with circular base.

Let R = Radius of the column flange,

r = Radius of the bolt pitch circle,

w = Load per bolt per unit distance from the tilting edge,

L = Distance of the load from the tilting edge, and

L_1, L_2, L_3, and L_4 = Distance of bolt centres from the tilting edge A.

As discussed in the previous article, equating the external moment $W \times L$ to the sum of the resisting moments of all the bolts, we have,

$$W.L = w\,[(L_1)^2 + (L_2)^2 + (L_3)^2 + (L_4)^2]$$

$$\therefore \quad w = \frac{W.L}{(L_1)^2 + (L_2)^2 + (L_3)^2 + (L_4)^2} \quad \ldots(i)$$

Now from the geometry of the Fig. 11.40 (*b*), we find that

$$L_1 = R - r\cos\alpha \qquad L_2 = R + r\sin\alpha$$

$$L_3 = R + r\cos\alpha \quad \text{and} \quad L_4 = R - r\sin\alpha$$

Substituting these values in equation (*i*), we get

$$w = \frac{W.L}{4\,R^2 + 2\,r^2}$$

$$\therefore \text{ Load in the bolt situated at 1} = w.L_1 = \frac{W.L.L_1}{4\,R^2 + 2\,r^2} = \frac{W.L\,(R - r\cos\alpha)}{4\,R^2 + 2\,r^2}$$

This load will be maximum when $\cos\alpha$ is minimum *i.e.* when $\cos\alpha = -1$ or $\alpha = 180°$.

$\therefore$ Maximum load in a bolt

$$= \frac{W.L\,(R + r)}{4\,R^2 + 2\,r^2}$$

In general, if there are n number of bolts, then load in a bolt

$$= \frac{2W.L\,(R - r\cos\alpha)}{n\,(2R^2 + r^2)}$$

and maximum load in a bolt,

$$W_t = \frac{2\,W.L\,(R + r)}{n\,(2R^2 + r^2)}$$

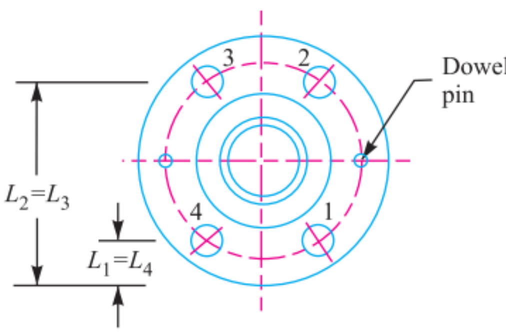

Fig. 11.41

The above relation is used when the direction of the load W changes with relation to the bolts as in the case of pillar crane. But if the direction of load is fixed, then the maximum load on the bolts may be reduced by locating the bolts in such a way that two of them are equally stressed as shown in Fig. 11.41. In such a case, maximum load is given by

$$W_t = \frac{2\,W.L}{n}\left[\frac{R + r\cos\left(\frac{180}{n}\right)}{2R^2 + r^2}\right]$$

Knowing the value of maximum load, we can determine the size of the bolt.

Note : Generally, two dowel pins as shown in Fig. 11.41, are used to take up the shear load. Thus the bolts are relieved of shear stress and the bolts are designed for tensile load only.

Example 11.17. *The base of a pillar crane is fastened to the foundation (a level plane) by eight bolts spaced equally on a bolt circle of diameter 1.6 m. The diameter of the pillar base is 2 m. Determine the size of bolts when the crane carries a load of 100 kN at a distance of 5 m from the centre of the base. The allowable stress for the bolt material is 100 MPa. The table for metric coarse threads is given below :*

Major diameter (mm)	*20*	*24*	*30*	*36*	*42*	*48*
Pitch (mm)	*2.5*	*3.0*	*3.5*	*4.0*	*4.5*	*5.0*
Stress area (mm^2)	*245*	*353*	*561*	*817*	*1120*	*1472*

Solution. Given : $n = 8$; $d = 1.6$ m or $r = 0.8$ m ; $D = 2$m or $R = 1$m ; $W = 100$ kN $= 100 \times 10^3$ N ; $e = 5$ m; $\sigma_t = 100$ MPa $= 100$ N/mm^2

The pillar crane is shown in Fig. 11.42.

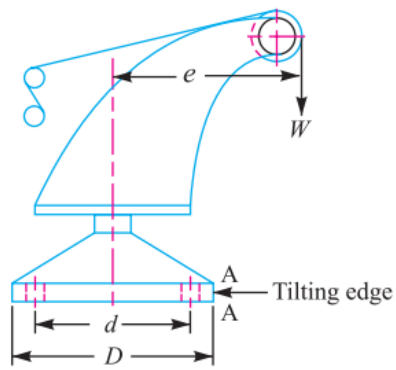

Fig. 11.42

We know that the distance of the load from the tilting edge *A-A*,

$$L = e - R = 5 - 1 = 4 \text{ m}$$

Let d_c = Core diameter of the bolts.

We know that maximum load on a bolt,

$$W_t = \frac{2\,W.L\,(R + r)}{n\,(2R^2 + r^2)} = \frac{2 \times 100 \times 10^3 \times 4\,(1 + 0.8)}{8\,[2 \times 1^2 + (0.8)^2]} = \frac{1440 \times 10^3}{21.12} = 68.18 \times 10^3 \text{ N}$$

We also know that maximum load on a bolt (W_t),

$$68.18 \times 10^3 = \frac{\pi}{4}\,(d_c)^2\,\sigma_t = \frac{\pi}{4}\,(d_c)^2\,100 = 78.54\,(d_c)^2$$

$$\therefore \quad (d_c)^2 = 68.18 \times 10^3 / 78.54 = 868 \quad \text{or} \quad d_c = 29.5 \text{ mm}$$

From Table 11.1 (coarse series), we find that the standard core diameter of the bolt is 31.093 mm and the corresponding size of the bolt is M 36. **Ans.**

11.22 Eccentric Load Acting in the Plane Containing the Bolts

When the eccentric load acts in the plane containing the bolts, as shown in Fig. 11.44, then the same procedure may be followed as discussed for eccentric loaded riveted joints.

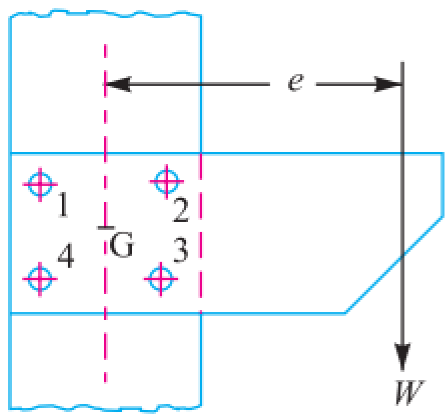

Fig. 11.44. Eccentric load in the plane containing the bolts.

Example 11.20. *Fig. 11.45 shows a solid forged bracket to carry a vertical load of 13.5 kN applied through the centre of hole. The square flange is secured to the flat side of a vertical stanchion through four bolts. Calculate suitable diameter D and d for the arms of the bracket, if the permissible stresses are 110 MPa in tension and 65 MPa in shear.*

Estimate also the tensile load on each top bolt and the maximum shearing force on each bolt.

Solution. Given : $W = 13.5$ kN = 13 500 N ; $\sigma_t = 110$ MPa = 110 N/mm^2 ; $\tau = 65$ MPa = 65 N/mm^2

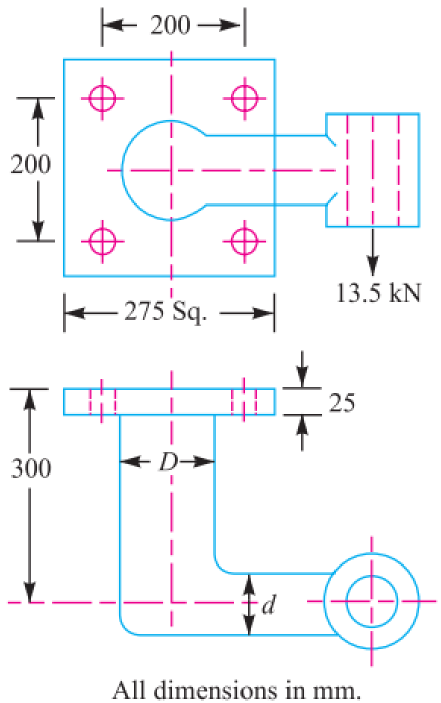

Fig. 11.45

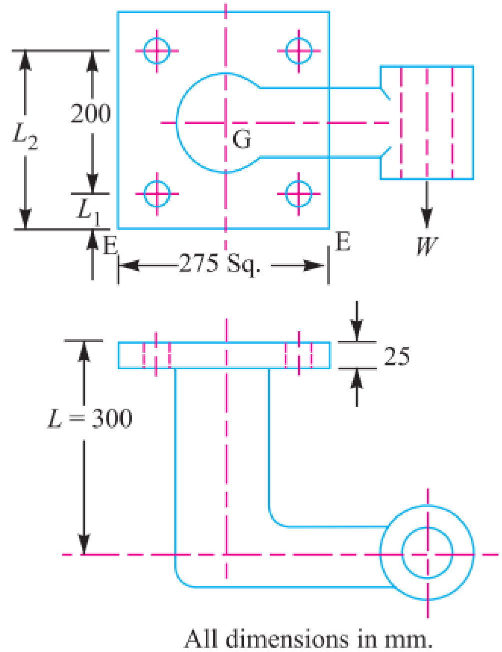

Fig. 11.46

Diameter D for the arm of the bracket

The section of the arm having D as the diameter is subjected to bending moment as well as twisting moment. We know that bending moment,

$$M = 13\,500 \times (300 - 25) = 3712.5 \times 10^3 \text{ N-mm}$$

and twisting moment, $T = 13\,500 \times 250 = 3375 \times 10^3$ N-mm

∴ Equivalent twisting moment,

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(3712.5 \times 10^3)^2 + (3375 \times 10^3)^2} \text{ N-mm}$$

$$= 5017 \times 10^3 \text{ N-mm}$$

We know that equivalent twisting moment (T_e),

$$5017 \times 10^3 = \frac{\pi}{16} \times \tau \times D^3 = \frac{\pi}{16} \times 65 \times D^3 = 12.76\, D^3$$

∴ $D^3 = 5017 \times 10^3 / 12.76 = 393 \times 10^3$

or $D = 73.24$ say 75 mm **Ans.**

Diameter (d) for the arm of the bracket

The section of the arm having d as the diameter is subjected to bending moment only. We know that bending moment,

$$M = 13\,500\left(250 - \frac{75}{2}\right) = 2868.8 \times 10^3 \text{ N-mm}$$

and section modulus, $Z = \frac{\pi}{32} \times d^3 = 0.0982\, d^3$

We know that bending (tensile) stress (σ_t),

$$110 = \frac{M}{Z} = \frac{2868.8 \times 10^3}{0.0982\, d^3} = \frac{29.2 \times 10^6}{d^3}$$

$\therefore$ $d^3 = 29.2 \times 10^6 / 110 = 265.5 \times 10^3$ or $d = 64.3$ say 65 mm **Ans.**

Tensile load on each top bolt

Due to the eccentric load W, the bracket has a tendency to tilt about the edge E–E, as shown in Fig. 11.46.

Let w = Load on each bolt per mm distance from the tilting edge due to the tilting effect of the bracket.

Since there are two bolts each at distance L_1 and L_2 as shown in Fig. 11.46, therefore total moment of the load on the bolts about the tilting edge E–E

$$= 2\,(w.L_1)\,L_1 + 2(w.L_2)\,L_2 = 2w\,[(L_1)^2 + (L_2)^2]$$

$$= 2w\,[(37.5)^2 + (237.5)^2] = 115\,625\, w \text{ N-mm} \quad \text{...}(i)$$

...($\because$ L_1 = 37.5 mm and L_2 = 237.5 mm)

and turning moment of the load about the tilting edge

$$= W.L = 13\,500 \times 300 = 4050 \times 10^3 \text{ N-mm} \quad \text{...}(ii)$$

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

$$w = 4050 \times 10^3 / 115\,625 = 35.03 \text{ N/mm}$$

$\therefore$ Tensile load on each top bolt

$$= w.L_2 = 35.03 \times 237.5 = 8320 \text{ N}$$ **Ans.**

Maximum shearing force on each bolt

We know that primary shear load on each bolt acting vertically downwards,

$$W_{s1} = \frac{W}{n} = \frac{13\,500}{4} = 3375 \text{ N}$$

...($\because$ No. of bolts, n = 4)

Since all the bolts are at equal distances from the centre of gravity of the four bolts (G), therefore the secondary shear load on each bolt is same.

Distance of each bolt from the centre of gravity (G) of the bolts,

$$l_1 = l_2 = l_3 = l_4 = \sqrt{(100)^2 + (100)^2} = 141.4 \text{ mm}$$

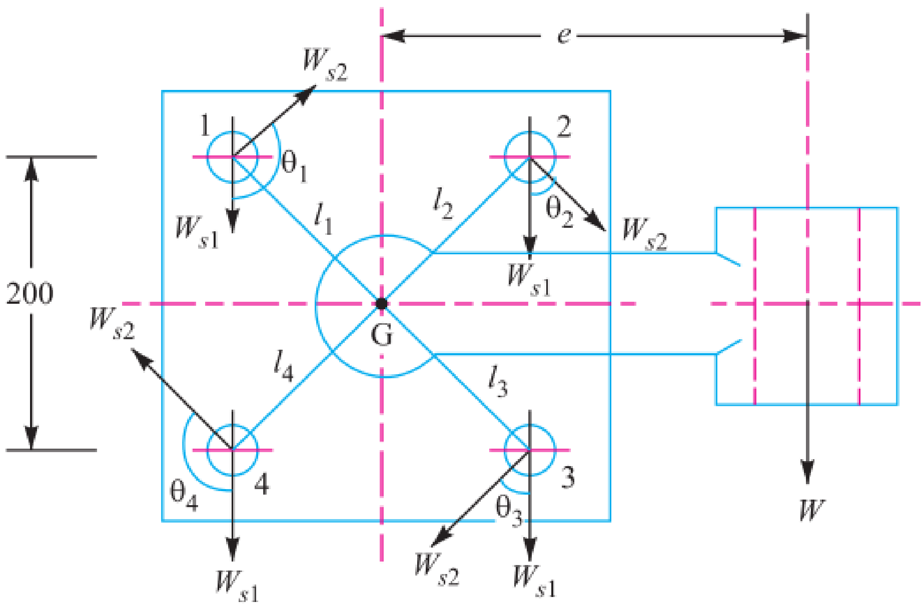

Fig. 11.47

∴ Secondary shear load on each bolt,

$$W_{s2} = \frac{W.e.l_1}{(l_1)^2 + (l_2)^2 + (l_3)^3 + (l_4)^2} = \frac{13\,500 \times 250 \times 141.4}{4\,(141.4)^2} = 5967 \text{ N}$$

Since the secondary shear load acts at right angles to the line joining the centre of gravity of the bolt group to the centre of the bolt as shown in Fig. 11.47, therefore the resultant of the primary and secondary shear load on each bolt gives the maximum shearing force on each bolt.

From the geometry of the Fig. 11.47, we find that

$$\theta_1 = \theta_4 = 135°, \text{ and } \theta_2 = \theta_3 = 45°$$

∴ Maximum shearing force on the bolts 1 and 4

$$= \sqrt{(W_{s1})^2 + (W_{s2})^2 + 2\, W_{s1} \times W_{s2} \times \cos 135°}$$

$$= \sqrt{(3375)^2 + (5967)^2 - 2 \times 3375 \times 5967 \times 0.7071} = 4303 \text{ N}$$ **Ans.**

and maximum shearing force on the bolts 2 and 3

$$= \sqrt{(W_{s1})^2 + (W_{s2})^2 + 2\, W_{s1} \times W_{s2} \times \cos 45°}$$

$$= \sqrt{(3375)^2 + (5967)^2 + 2 \times 3375 \times 5967 \times 0.7071} = 8687 \text{ N}$$ **Ans.**