UNIT III

TEMPORARY AND PERMANENT JOINTS

CHAPTER 2

Stresses due to External Forces

The following stresses are induced in a bolt when it is subjected to an external load.

1. Tensile stress.

The bolts, studs and screws usually carry a load in the direction of the bolt axis which induces a tensile stress in the bolt.

Let d_c = Root or core diameter of the thread, and

σ_t = Permissible tensile stress for the bolt material.

We know that external load applied,

$$P = \frac{\pi}{4}(d_c)^2\sigma_t$$

$$d_c = \sqrt{\frac{4P}{\pi\sigma_t}}$$

Now from Table, the value of the nominal diameter of bolt corresponding to the value of d_c may be obtained or stress area $[\frac{\pi}{4}(d_c)^2]2$ may be fixed.

2. Shear stress.

Sometimes, the bolts are used to prevent the relative movement of two or more parts, as in case of flange coupling, then the shear stress is induced in the bolts. The shear stresses should be avoided as far as possible. It should be noted that when the bolts are subjected to direct shearing loads, they should be located in such a way that the shearing load comes upon the body (i.e. shank) of the bolt and not upon the threaded portion. In some cases, the bolts may be relieved of shear load by using shear pins. When a number of bolts are used to share the shearing load, the finished bolts should be fitted to the reamed holes.

Let d = Major diameter of the bolt, and

n = Number of bolts.

∴ Shearing load carried by the bolts,

$$P_s = \frac{\pi}{4} \times d2 \times \tau \times n$$

$$d = \sqrt{\frac{4P_s}{\pi \tau n}}$$

3. Combined tension and shear stress.

When the bolt is subjected to both tension and shear loads, as in case of coupling bolts or bearing, then the diameter of the shank of the bolt is obtained from the shear load and that of threaded part from the tensile load. A diameter slightly larger than that required for either shear or tension may be assumed and stresses due to combined load should be checked for the following principal stresses.

Maximum principal shear stress,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_t)^2 + 4\tau^2}$$

and maximum principal tensile stress,

$$(\sigma_t)_{max} = \frac{\sigma_t}{2} + \frac{1}{2}\sqrt{(\sigma_t)^2 + 4\tau^2}$$

These stresses should not exceed the safe permissible values of stresses.

Problem 2.1

An eye bolt is to be used for lifting a load of 60 kN. Find the nominal diameter of the bolt, if the tensile stress is not to exceed 100 MPa. Assume coarse threads.

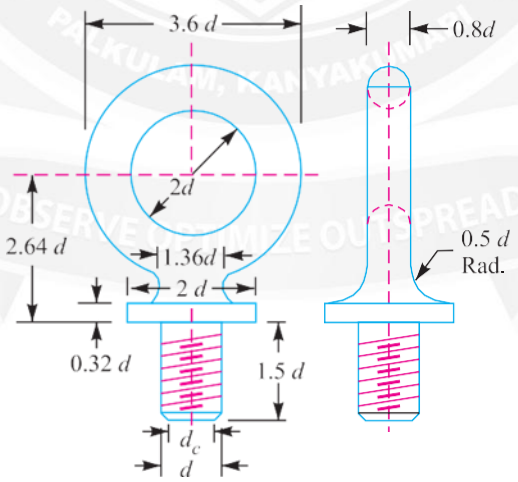

Fig 2.1

[Source: "A Textbook of Machine Design by R.S. Khurmi J.K. Gupta, Page: 392]

Given Data:

$P = 60 \text{ kN} = 60 \times 10^3 \text{ N}$

$\sigma_t = 100 \text{ MPa} = 100 \text{ N/mm}^2$

An eye bolt for lifting a load is shown in Fig. 2.1.

Let d = Nominal diameter of the bolt, and

d_c = Core diameter of the bolt.

We know that load on the bolt (P),

$$60 \times 10^3 = \frac{\pi}{4} (d_c)^2 \sigma_t$$

$$60 \times 10^3 = \frac{\pi}{4} (d_c)^2 100$$

$$(d_c)^2 = 60 \times 10^3 / 78.55$$

$$(d_c)^2 = 764 \text{ or}$$

$$\therefore d_c = 27.6 \text{ mm}$$

From Table (coarse series), we find that the standard core diameter (d_c) is 28.706 mm and the corresponding nominal diameter (d) is 33 mm

Stress due to Combined Forces

The resultant axial load on a bolt depends upon the following factors:

1. The initial tension due to tightening of the bolt,
2. The external load, and
3. The relative elastic yielding (springiness) of the bolt and the connected members.

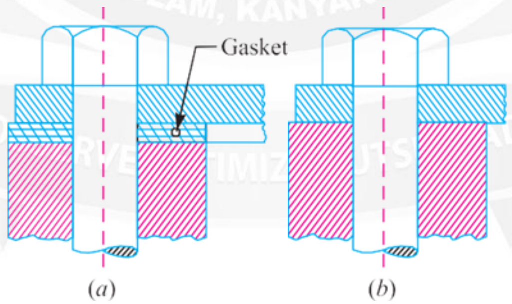

Fig 2.2

[Source: "A Textbook of Machine Design by R.S. Khurmi J.K. Gupta, Page: 394]

When the connected members are very yielding as compared with the bolt, which is a soft gasket, as shown in Fig. 2.2 (a), then the resultant load on the bolt is approximately equal to the sum of the initial tension and the external load. On the other hand, if the bolt is very yielding as compared with the connected members, as shown in

Fig. 2.2 (b), then the resultant load will be either the initial tension or the external load, whichever is greater. The actual conditions usually lie between the two extremes. In order to determine the resultant axial load (P) on the bolt, the following equation may be used:

$$P = P_1 + \frac{a}{1+a} \times P_2$$

$$P = P_1 + K.P_2 \quad \text{.......Substituting } \left(K = \frac{a}{1+a}\right)$$

where P_1 = Initial tension due to tightening of the bolt,

P_2 = External load on the bolt, and

a = Ratio of elasticity of connected parts to the elasticity of bolt.

For soft gaskets and large bolts, the value of a is high and the value of $\frac{a}{1+a}$ is approximately equal to unity, so that the resultant load is equal to the sum of the initial tension and the external load.

For hard gaskets or metal to metal contact surfaces and with small bolts, the value of a is small and the resultant load is mainly due to the initial tension (or external load, in rare case it is greater than initial tension).

The value of 'a' may be estimated by the designer to obtain an approximate value for the resultant load. The values of $\frac{a}{1+a}$ (i.e. K) for various type of joints are shown in Table 2.2. The designer thus has control over the influence on the resultant load on a bolt by proportioning the sizes of the connected parts and bolts and by specifying initial tension in the bolt.

Preloading of Bolt and Combined Stresses

In the applications such as pressure vessels and cylinder covers, it is an essential to apply an initial tightening torque to make a joint leak proof. It is called "Preloading". The preloading are as follows.

i) It stops leakage in pressure vessels.

ii) It improves the fatigue strength.

The amount of preloading tension is dependent on many factors such as

i) External Load

ii) Materials used

iii) Diameter of Bolts.

An approximate formula for preload (or) initial tension is given by

$$P_i = 2860d \text{ N}$$

Bolts of Uniform Strength

When a bolt is subjected to shock loading, as in case of a cylinder head bolt of an internal combustion engine, the resilience of the bolt should be considered in order to prevent breakage at the thread. In an ordinary bolt shown in Fig. 2.3 (a), the effect of the impulsive loads applied axially is concentrated on the weakest part of the bolt i.e. the cross-sectional area at the root of the threads. In other words, the stress in the threaded part of the bolt will be higher than that in the shank. Hence a great portion of the energy will be absorbed at the region of the threaded part which may fracture the threaded portion because of its small length.

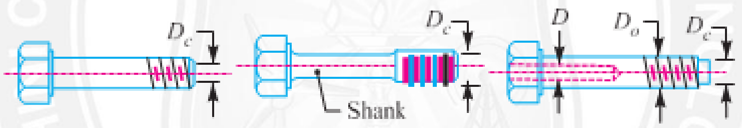

Fig 2.3 Bolts of uniform strength.

[Source: "A Textbook of Machine Design by R.S. Khurmi J.K. Gupta, Page: 404]

If the shank of the bolt is turned down to a diameter equal or even slightly less than the core diameter of the thread (D_c) as shown in Fig. 2.3 (b), then shank of the bolt will undergo a higher stress. This means that a shank will absorb a large portion of the energy, thus relieving the material at the sections near the thread. The bolt, in this way, becomes stronger and lighter and it increases the shock absorbing capacity of the bolt because of an increased modulus of resilience. This gives us bolts of uniform strength. The resilience of a bolt may also be increased by increasing its length. A second alternative method of obtaining the bolts of uniform strength is shown in Fig. 2.3 (c). In this method, an axial hole is drilled through the head as far as the thread portion such that the area of the shank becomes equal to the root area of the thread.

Let D = Diameter of the hole.

D_o = Outer diameter of the thread, and

D_c = Root or core diameter of the thread.

$$\frac{\pi}{4} D^2 = \frac{\pi}{4} [(D_o)^2 - (D_i)^2]$$

$$D^2 = [(D_o)^2 - (D_i)^2]$$

$$D = \sqrt{[(D_o{}^2) - (D_i)^2]}\backslash$$

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.

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.

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. 2.4. 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.

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.

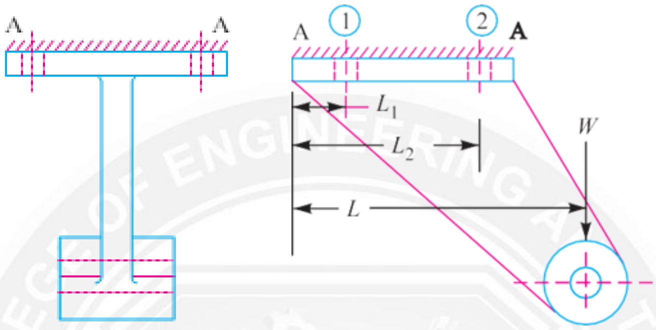

Fig 2.4 Eccentric load acting parallel to the axis of bolts.

[Source: "A Textbook of Machine Design by R.S. Khurmi J.K. Gupta, Page: 405]

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.

∴ 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

$$= w.\ L_2 \times L_2$$

$$= w\ (L_2)^2$$

∴ Total moment of the load on the bolts about the tilting edge

$$= 2w\ (L_1)^2 + 2w\ (L_2)^2 \qquad ...(i)$$

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

Also the moment due to load W about the tilting edge

$$= W.L \qquad ...(ii)$$

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

$$W.L = 2w\,(L_1)^2 + 2w(L_2)^2 \text{ or}$$

$$w = \frac{W.L}{2[(L_1)^2 + (L_2)^2]}$$

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.

∴ Tensile load on each bolt at distance L_2,

$$W_{t2} = W_2 = w\,L_2$$

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

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

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

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,

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

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

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. 2.5.

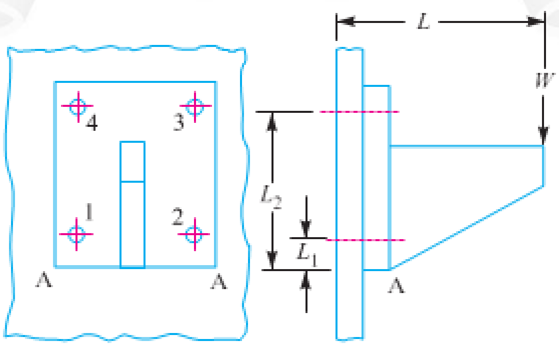

Fig 2.5 Eccentric load perpendicular to the axis of bolts.

[Source: "A Textbook of Machine Design by R.S. Khurmi J.K. Gupta, Page: 409]

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,

$$W_s = W/n,$$

where n 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,

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

and equivalent shear load,

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

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

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. 2.6

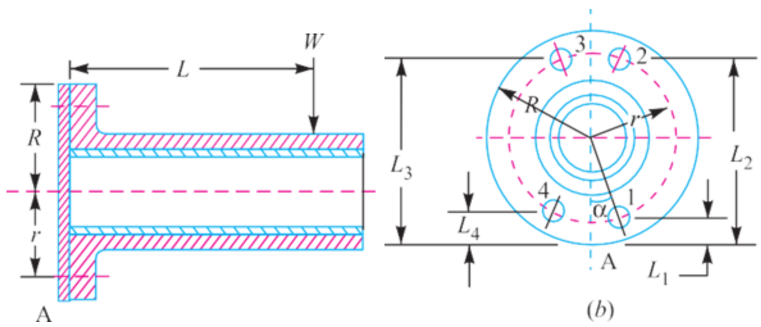

Fig 2.6 Eccentric load on a bracket with circular base.

[Source: "A Textbook of Machine Design by R.S. Khurmi J.K. Gupta, Page: 419]

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.

Equating the external moment W × L to the sum of the resisting moments of all the bolts, we have,

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

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

Now from the geometry of the Fig. 2.6 (b), we find that

$$L_1 = R - r\cos\alpha$$

$$L_2 = R + r\sin\alpha$$

$$L_3 = R + r\cos\alpha \text{ and}$$

$$L_4 = R - r\sin\alpha$$

Substituting these values in equation (i), we get

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

∴ Load in the bolt situated at 1 = w. L_1

$$= \frac{W.L.L_1}{[4R^2 + 2r^2]}$$

$$= \frac{W.L.(R - r\cos\alpha)}{[4R^2 + 2r^2]}$$

This load will be maximum when cos α is minimum i.e. when cos α = – 1 or α = 180°.

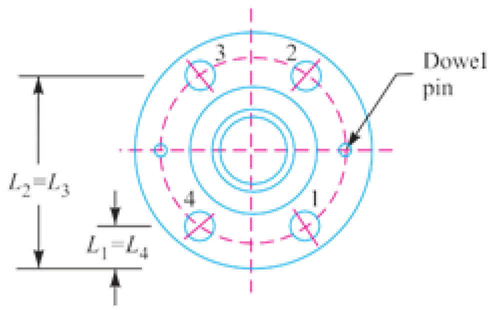

Fig 2.7

[Source: "A Textbook of Machine Design by R.S. Khurmi J.K. Gupta, Page: 420]

∴ Maximum load in a bolt

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

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

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

and maximum load in a bolt,

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

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. 2.7. In such a case, maximum load is given by

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

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

Problem 2.2

A pillar crane having a circular base of 600 mm diameter is fixed to the foundation of concrete base by means of four bolts. The bolts are of size 30 mm and are equally spaced on a bolt circle diameter of 500 mm. Determine: 1. The distance of the load from the centre of the pillar along a line X-X as shown in Fig. 2.8 (a). The load lifted by the pillar crane is 60 kN and the allowable tensile stress for the bolt material is 60 MPa.

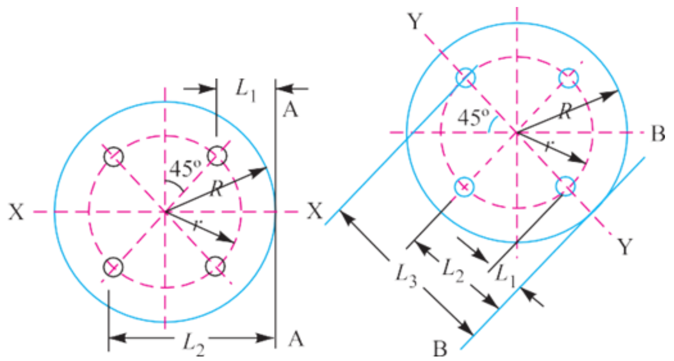

Fig 2.8

[Source: "A Textbook of Machine Design by R.S. Khurmi J.K. Gupta, Page: 422]

2. The maximum stress induced in the bolts if the load is applied along a line Y-Y of the foundation as shown in Fig. 2.8 (b) at the same distance as in part (1).

Given Data:

$D = 600$ mm or $R = 300$ mm

$n = 4$

$d_b = 30$ mm

$d = 500$ mm or $r = 250$ mm

$W = 60$ kN

$\sigma_t = 60$ MPa $= 60$ N/mm^2

Since the size of bolt (i.e. $d_b = 30$ mm), is given therefore from Table, we find that the stress area corresponding to M 30 is 561 mm^2.

We know that the maximum load carried by each bolt

$$= \text{Stress area} \times \sigma_t = 561 \times 60$$

$$= 33660 \text{ N}$$

$$= 33.66 \text{ kN}$$

and direct tensile load carried by each bolt

$$= \frac{W}{n} = \frac{60}{4} = 15 \text{ kN}$$

∴ Total load carried by each bolt at distance L_2 from the tilting edge A-A

$$= 33.66 + 15 = 48.66 \text{ kN} \quad \text{...(i)}$$

From Fig. 2.8 (a), we find that

$$L_1 = R - r \cos 45°$$

$$L_1 = 300 - 250 \times 0.707$$

$$L_1 = 123 \text{ mm} = 0.123 \text{ m}$$

and

$$L_2 = R + r \cos 45°$$

$$L_2 = 300 + 250 \times 0.707$$

$$L_2 = 477 \text{ mm} = 0.477 \text{ m}$$

Let w = Load (in kN) per bolt per unit distance.

∴ Total load carried by each bolt at distance L_2 from the tilting edge A-A

$= w. L_2 = w \times 0.477$ kN ...(ii)

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

$w = 48.66 / 0.477 = 102$ kN/m

∴ Resisting moment of all the bolts about the outer (i.e. tilting) edge of the flange along the tangent A-A

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

$= 2 \times 102 [(0.123)^2 + (0.477)^2]$

$= 49.4$ kN-m

1. Distance of the load from the centre of the pillar

Let e = Distance of the load from the centre of the pillar or eccentricity of the load, and

L = Distance of the load from the tilting edge A-A = e – R = e – 0.3

We know that turning moment due to load W, about the tilting edge A-A of the flange

$= W.L = 60 (e - 0.3)$ kN-m

Now equating the turning moment to the resisting moment of all the bolts, we have

$60 (e - 0.3) = 49.4$

$\therefore e - 0.3 = 49.4 / 60$

$e = 0.823$ or $e = 0.823 + 0.3$

$e = 1.123$ m.

2. Maximum stress induced in the bolt

Since the load is applied along a line Y-Y as shown in Fig. 11.43 (b), and at the same distance as in part (1) i.e. at $L = e - 0.3 = 1.123 - 0.3 = 0.823$ m from the tilting edge B-B, therefore

Turning moment due to load W about the tilting edge B–B

$= W.L = 60 \times 0.823 = 49.4$ kN-m

From Fig. 2.8 (b), we find that

$L_1 = R - r$

$L_1 = 300 - 250$

$L_1 = 50$ mm $= 0.05$ m

$L_2 = R = 300\text{ mm} = 0.3\text{ m}$

and $L_3 = R + r = 300 + 250$

$L_3 = 550\text{ mm} = 0.55\text{ m}$

∴ Resisting moment of all the bolts about B–B

$= w\,[(L_1)^2 + 2(L_2)^2 + (L_3)^2]$

$= w\,[(0.05)^2 + 2(0.3)^2 + (0.55)^2]$ kN-m

$= 0.485\,w$ kN-m

Equating resisting moment of all the bolts to the turning moment, we have

$0.485\,w = 49.4$

or $w = 49.4 / 0.485 = 102$ kN/m

Since the bolt at a distance of L3 is heavily loaded, therefore load carried by this bolt

$= w.\,L3$

$= 102 \times 0.55$

$= 56.1$ kN

and net force taken by the bolt

$= w.\,L_3 - \frac{W}{n}$

$= 56.1 - \frac{60}{4}$

$= 41.1\text{ kN} = 41100\text{ N}$

∴ Maximum stress induced in the bolt

$= \frac{\text{Force}}{\text{Stress Area}}$

$= \frac{41100}{516}$

$= 79.65\text{ N/mm}^2$

$= 79.65$ MPa

<u>Eccentric Load Acting in the Plane Containing the Bolts</u>

When the eccentric load acts in the plane containing the bolts, as shown in Fig. 2.9, then the same procedure may be followed as discussed for eccentric loaded riveted joints.

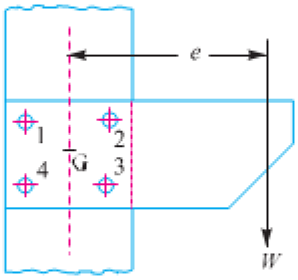

Fig 2.9 Eccentric load in the plane containing the bolts.

[Source: "A Textbook of Machine Design by R.S. Khurmi J.K. Gupta, Page: 424]

Problem 2.3

Fig. 2.10 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.

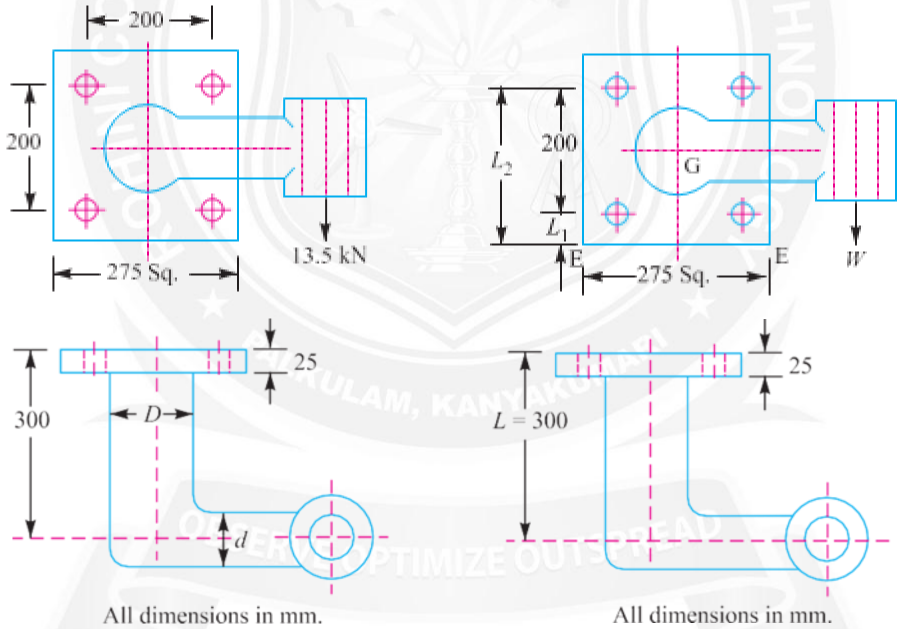

Fig 2.10 **Fig 2.11**

[Source: "A Textbook of Machine Design by R.S. Khurmi J.K. Gupta, Page: 424]

Given Data:

$W = 13.5 \text{ kN} = 13500 \text{ N}$

$\sigma_t = 110 \text{ MPa} = 110 \text{ N/mm}^2$

$\tau = 65 \text{ MPa} = 65 \text{ N/mm}^2$

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,

$$M = 13500 \times (300 - 25)$$

$$M = 3712.5 \times 10^3 \text{ N-mm}$$

and twisting moment,

$$T = 13\,500 \times 250$$

$$T = 3375 \times 10^3 \text{ N-mm}$$

∴ Equivalent twisting moment,

$$T_e = \sqrt{M^2 + T^2}$$

$$T_e = \sqrt{(3712.5 \times 10^3)^2 + (3375 \times 10^3)^2}$$

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

We know that equivalent twisting moment (T_e),

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

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

$$5017 \times 10^3 = 12.76\, D^3$$

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

or D = 73.24 say

D = 75 mm

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,

$$M = 13500\left[250 - \frac{75}{2}\right]$$

$$M = 2868.8 \times 10^3 \text{ N-mm}$$

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

$$= 0.0982\, d^3$$

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

$$110 = \frac{29.2 \times 10^3}{d^3}$$

$d^3 = 29.2 \times 106 / 110 = 265.5 \times 10^3$ or

$d = 64.3$ say 65 mm.

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 2.11.

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. 2.11, therefore total moment of the load on the bolts about the tilting edge E–E

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

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

$$= 2w\,[(37.5)2 + (237.5)2]$$

$$= 115\,625\ w\ \text{N-mm} \qquad \text{...(i)}$$

... (L_1 = 37.5 mm and L_2 = 237.5 mm)

and turning moment of the load about the tilting edge

$$= W.L = 13\,500 \times 300$$

$$= 4050 \times 10^3\ \text{N-mm} \qquad \text{...(ii)}$$

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

$$w = 4050 \times 10^3 / 115625 = 35.03\ \text{N/mm}$$

∴ Tensile load on each top bolt

$$= w.\,L_2 = 35.03 \times 237.5$$

$$= 8320\ \text{N.}$$

Maximum shearing force on each bolt

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

$$W_{s1} = \frac{W}{n} = \frac{13500}{4} = 3375\ \text{N.}$$

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,

$$l_1 = l_2 = l_3 = l_4 = \sqrt{100^2 + 100^2}$$

$$= 141.4\ \text{mm}$$

∴Secondary shear load on each bolt,

$$W_{s2} = \frac{W.el_1}{(l_1)^2+(l_2)^2+(l_3)^2+(l_4)^2}$$

$$W_{s2} = \frac{13500 \times 250 \times 141.4}{4(141.4)^2}$$

$$W_{s2} = 5967 \text{ N.}$$

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. 2.12, 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. 2.12, we find that

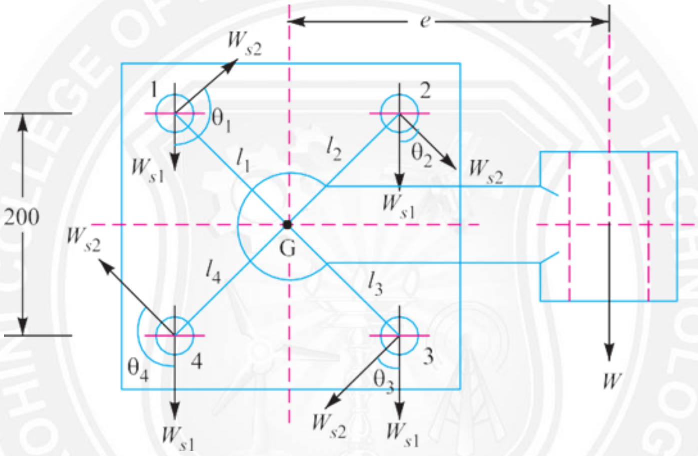

Fig 2.12

[Source: "A Textbook of Machine Design by R.S. Khurmi J.K. Gupta, Page: 424]

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

∴Maximum shearing force on the bolts 1 and 4

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

$$= \sqrt{3375)^2 + (5967)^2 - 2 \times 3375 \times 5967 \times 0.7071}$$

$$= 4303 \text{ N}$$

and maximum shearing force on the bolts 2 and 3

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

$$= \sqrt{3375)^2 + (5967)^2 + 2 \times 3375 \times 5967 \times 0.7071}$$

$$= 8687 \text{ N}$$