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DEPARTMENT OF AGRICULTURE ENGINEERING

R2019-MACHINE DESIGN

III YEAR V SEM

UNIT 2 –DESIGN OF FASTENERS

TOPIC –Stresses due to Combined forces & Problems

11.13 Stress due to Combined Forces

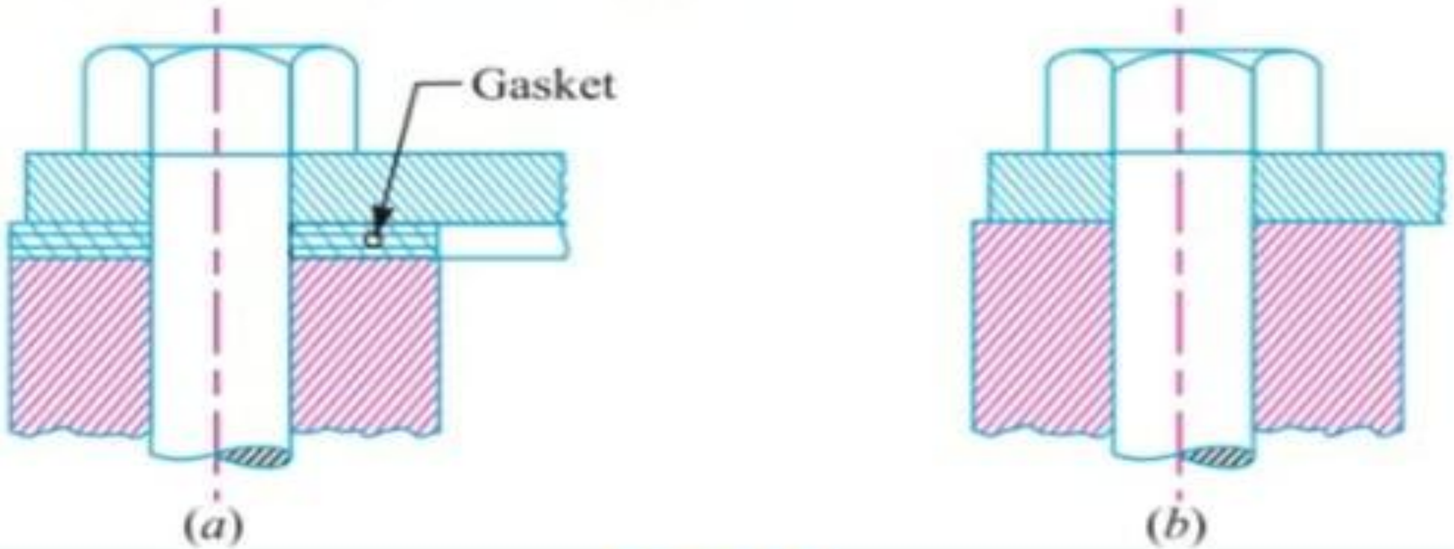

Fig. 11.23

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

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

When the connected members are very yielding as compared with the bolt, which is a soft gasket, as shown in Fig. 11.23 (*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. 11.23 (*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 :

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

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 11.2. 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.

Table 11.2. Values of *K* for various types of joints.

Type of joint	$K = \frac{a}{1+a}$
Metal to metal joint with through bolts	0.00 to 0.10
Hard copper gasket with long through bolts	0.25 to 0.50
Soft copper gasket with long through bolts	0.50 to 0.75
Soft packing with through bolts	0.75 to 1.00
Soft packing with studs	1.00

11.14 Design of Cylinder Covers

The cylinder covers may be secured by means of bolts or studs, but studs are preferred. The possible arrangement of securing the cover with bolts and studs is shown in Fig. 11.24 (*a*) and (*b*) respectively. The bolts or studs, cylinder cover plate and cylinder flange may be designed as discussed below:

1. *Design of bolts or studs*

In order to find the size and number of bolts or studs, the following procedure may be adopted.

Let D = Diameter of the cylinder,

p = Pressure in the cylinder,

d_c = Core diameter of the bolts or studs,

n = Number of bolts or studs, and

σ_{tb} = Permissible tensile stress for the bolt or stud material.

We know that upward force acting on the cylinder cover,

$$P = \frac{\pi}{4}(D^2)\,p \quad \text{...}(i)$$

This force is resisted by n number of bolts or studs provided on the cover.

∴ Resisting force offered by n number of bolts or studs,

$$P = \frac{\pi}{4}(d_c)^2\,\sigma_{tb} \times n \quad \text{...}(ii)$$

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

$$\frac{\pi}{4}(D^2)\,p = \frac{\pi}{4}(d_c)^2\,\sigma_{tb} \times n \quad \text{...}(ii)$$

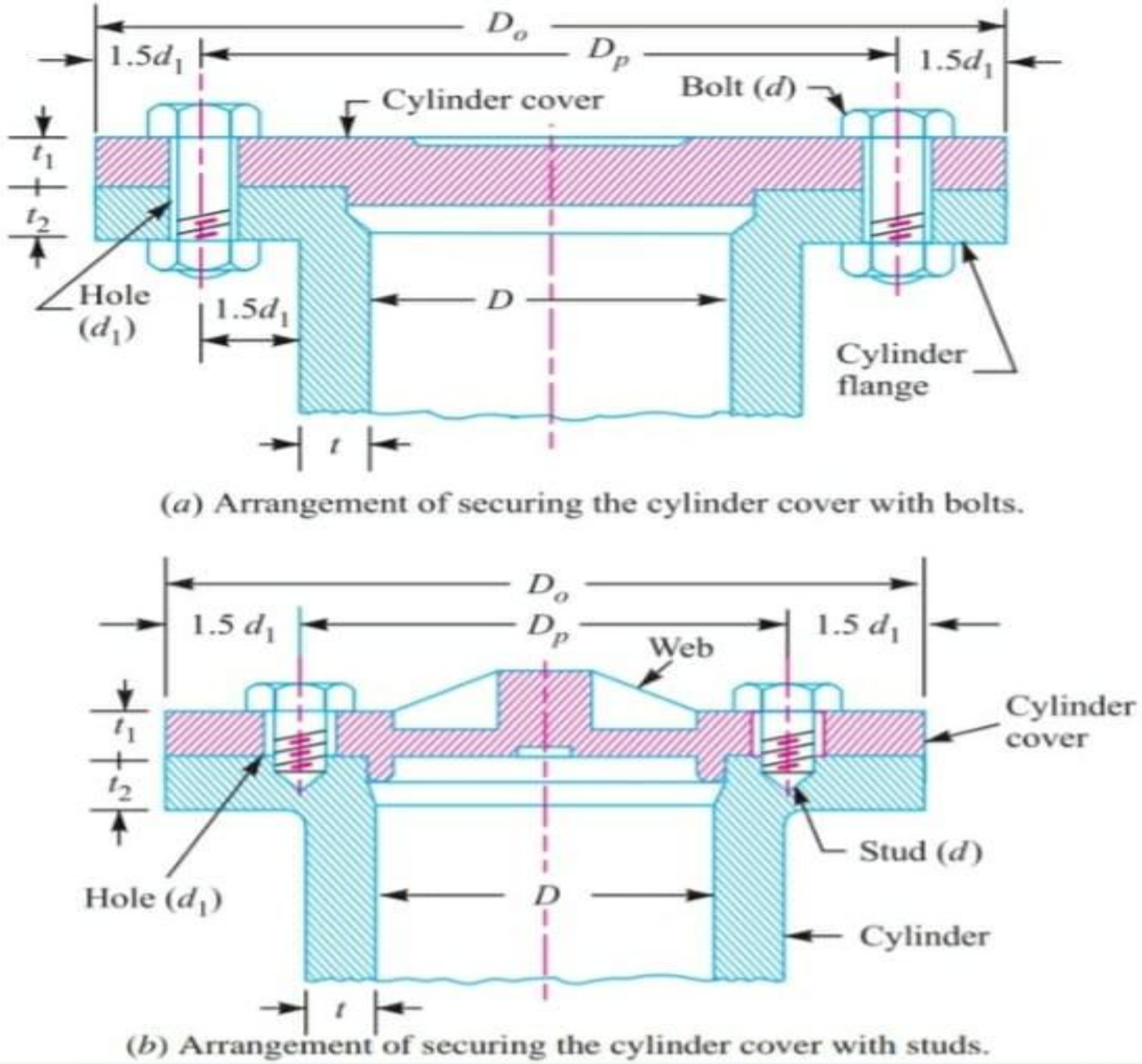

Fig. 11.24

From this equation, the number of bolts or studs may be obtained, if the size of the bolt or stud is known and *vice-versa*. Usually the size of the bolt is assumed. If the value of n as obtained from the above relation is odd or a fraction, then next higher even number is adopted.

The bolts or studs are screwed up tightly, along with metal gasket or asbestos packing, in order to provide a leak proof joint. We have already discussed that due to the tightening of bolts, sufficient tensile stress is produced in the bolts or studs. This may break the bolts or studs, even before any load due to internal pressure acts upon them. Therefore a bolt or a stud less than 16 mm diameter should never be used.

The tightness of the joint also depends upon the circumferential pitch of the bolts or studs. The circumferential pitch should be between $20\sqrt{d_1}$ and $30\sqrt{d_1}$, where d_1 is the diameter of the hole in mm for bolt or stud. The pitch circle diameter (D_p) is usually taken as $D + 2t + 3d_1$ and outside diameter of the cover is kept as

$$D_o = D_p + 3d_1 = D + 2t + 6d_1$$

where t = Thickness of the cylinder wall.

2. *Design of cylinder cover plate*

The thickness of the cylinder cover plate (t_1) and the thickness of the cylinder flange (t_2) may be determined as discussed below:

Let us consider the semi-cover plate as shown in Fig. 11.25. The internal pressure in the cylinder tries to lift the cylinder cover while the bolts or studs try to retain it in its position. But the centres of pressure of these two loads do not coincide. Hence, the cover plate is subjected to bending stress. The point X is the centre of pressure for bolt load and the point Y is the centre of internal pressure.

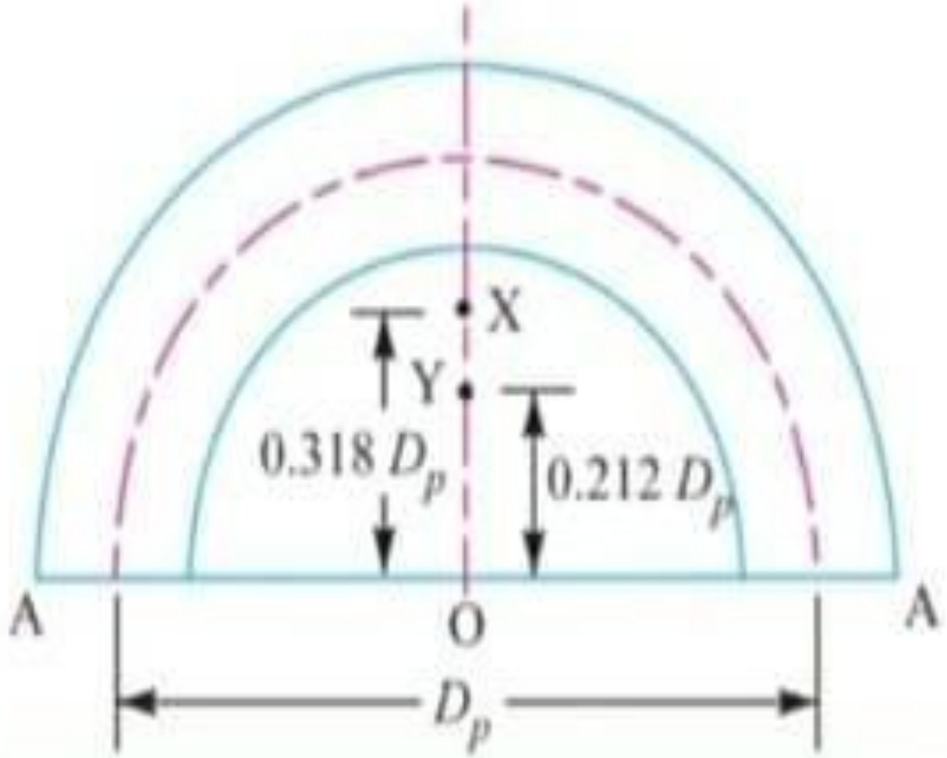

Fig. 11.25. Semi-cover plate of a cylinder.

We know that the bending moment at A-A,

$$M = \frac{\text{Total bolt load}}{2}(OX - OY) = \frac{P}{2}(0.318\,D_p - 0.212\,D_p)$$

$$= \frac{P}{2} \times 0.106\,D_p = 0.053\,P \times D_p$$

Section modulus, $Z = \frac{1}{6} w\,(t_1)^2$

where w = Width of plate

= Outside dia. of cover plate – 2 × dia. of bolt hole

$= D_o - 2d_1$

Knowing the tensile stress for the cover plate material, the value of t_1 may be determined by using the bending equation, *i.e.*, $\sigma_t = M / Z$.

3. *Design of cylinder flange*

The thickness of the cylinder flange (t_2) may be determined from bending consideration. A portion of the cylinder flange under the influence of one bolt is shown in Fig. 11.26.

The load in the bolt produces bending stress in the section X-X. From the geometry of the figure, we find that eccentricity of the load from section X-X is

e = Pitch circle radius – (Radius of bolt hole + Thickness of cylinder wall)

$$= \frac{D_p}{2} - \left(\frac{d_1}{2} + t\right)$$

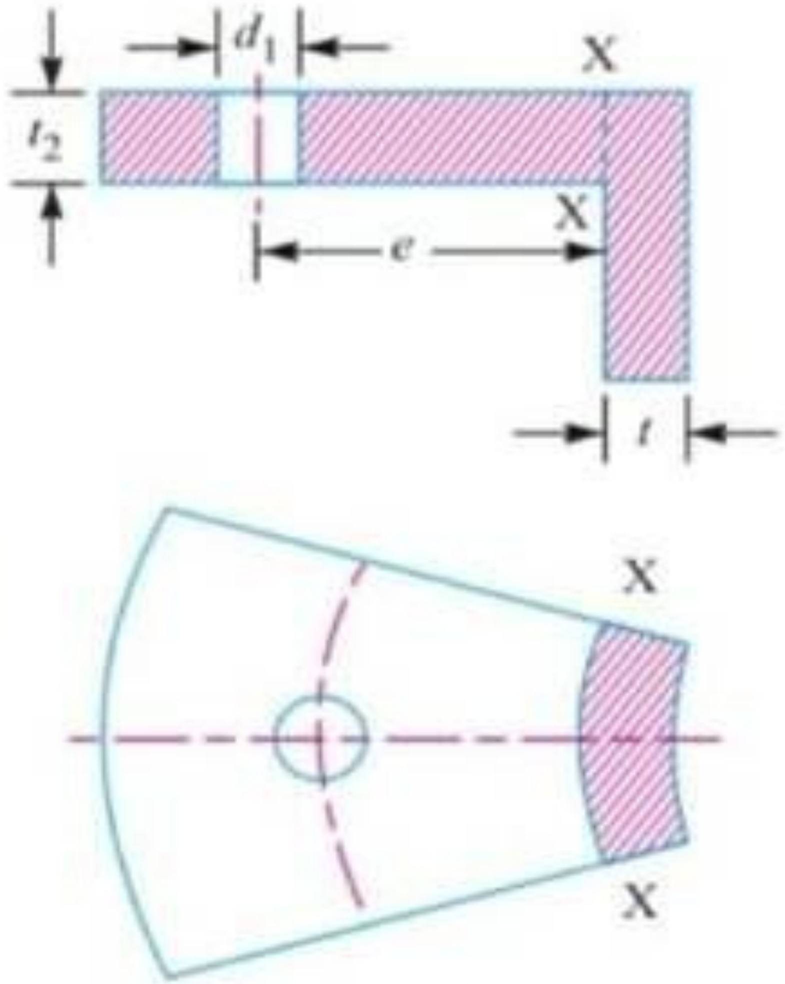

Fig. 11.26. A portion of the cylinder flange.

$\therefore$ Bending moment, $M = \text{Load on each bolt} \times e = \frac{P}{n} \times e$

Radius of the section X-X,

$$R = \text{Cylinder radius} + \text{Thickness of cylinder wall} = \frac{D}{2} + t$$

Width of the section X-X,

$$w = \frac{2\pi R}{n}$$, where n is the number of bolts.

Section modulus, $Z = \frac{1}{6} w (t_2)^2$

Knowing the tensile stress for the cylinder flange material, the value of t_2 may be obtained by using the bending equation *i.e.* $\sigma_t = M / Z$.

Example 11.6. *A steam engine cylinder has an effective diameter of 350 mm and the maximum steam pressure acting on the cylinder cover is 1.25 N/mm². Calculate the number and size of studs required to fix the cylinder cover, assuming the permissible stress in the studs as 33 MPa.*

Solution. Given: $D = 350$ mm ; $p = 1.25$ N/mm² ; $\sigma_t = 33$ MPa $= 33$ N/mm²

Let d = Nominal diameter of studs,

d_c = Core diameter of studs, and

n = Number of studs.

We know that the upward force acting on the cylinder cover,

$$P = \frac{\pi}{4} \times D^2 \times p = \frac{\pi}{4} (350)^2 \, 1.25 = 120\,265 \text{ N} \quad ...(i)$$

Assume that the studs of nominal diameter 24 mm are used. From Table 11.1 (coarse series), we find that the corresponding core diameter (d_c) of the stud is 20.32 mm.

$\therefore$ Resisting force offered by n number of studs,

$$P = \frac{\pi}{4} \times (d_c)^2 \sigma_t \times n = \frac{\pi}{4} (20.32)^2 \, 33 \times n = 10\,700\, n \text{ N} \quad ...(ii)$$

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

$n = 120\,265 / 10\,700 = 11.24$ say 12 **Ans.**

Taking the diameter of the stud hole (d_1) as 25 mm, we have pitch circle diameter of the studs,

$$D_p = D + 2t + 3d_1 = 350 + 2 \times 10 + 3 \times 25 = 445 \text{ mm}$$

...(Assuming $t = 10$ mm)

∴ *Circumferential pitch of the studs

$$= \frac{\pi \times D_p}{n} = \frac{\pi \times 445}{12} = 116.5 \text{ mm}$$

We know that for a leak-proof joint, the circumferential pitch of the studs should be between $20\sqrt{d_1}$ to $30\sqrt{d_1}$, where d_1 is the diameter of stud hole in mm.

∴ Minimum circumferential pitch of the studs

$$= 20\sqrt{d_1} = 20\sqrt{25} = 100 \text{ mm}$$

and maximum circumferential pitch of the studs

$$= 30\sqrt{d_1} = 30\sqrt{25} = 150 \text{ mm}$$

Since the circumferential pitch of the studs obtained above lies within 100 mm to 150 mm, therefore the size of the stud chosen is satisfactory.

∴ Size of the stud = M 24 **Ans.**

Example 11.7. *A mild steel cover plate is to be designed for an inspection hole in the shell of a pressure vessel. The hole is 120 mm in diameter and the pressure inside the vessel is 6 N/mm². Design the cover plate along with the bolts. Assume allowable tensile stress for mild steel as 60 MPa and for bolt material as 40 MPa.*

Solution. Given : D = 120 mm or r = 60 mm ; p = 6 N/mm² ; σ_t = 60 MPa = 60 N/mm² ; σ_{tb} = 40 MPa = 40 N/mm²

First for all, let us find the thickness of the pressure vessel. According to Lame's equation, thickness of the pressure vessel,

$$t = r\left[\sqrt{\frac{\sigma_t + p}{\sigma_t - p}} - 1\right] = 60\left[\sqrt{\frac{60+6}{60-6}} - 1\right] = 6 \text{ mm}$$

Let us adopt t = 10 mm

Design of bolts

Let d = Nominal diameter of the bolts,

d_c = Core diameter of the bolts, and

n = Number of bolts.

We know that the total upward force acting on the cover plate (or on the bolts),

$$P = \frac{\pi}{4}(D)^2\ p = \frac{\pi}{4}(120)^2 6 = 67\ 860 \text{ N} \qquad \text{...(i)}$$

Let the nominal diameter of the bolt is 24 mm. From Table 11.1 (coarse series), we find that the corresponding core diameter (d_c) of the bolt is 20.32 mm.

∴ Resisting force offered by n number of bolts,

$$P = \frac{\pi}{4}(d_c)^2\ \sigma_{tb} \times n = \frac{\pi}{4}(20.32)^2\ 40 \times n = 12\ 973\ n \text{ N} \qquad \text{...(ii)}$$

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

$$n = 67\,860 / 12\,973 = 5.23 \text{ say } 6$$

Taking the diameter of the bolt hole (d_1) as 25 mm, we have pitch circle diameter of bolts,

$$D_p = D + 2t + 3d_1 = 120 + 2 \times 10 + 3 \times 25 = 215 \text{ mm}$$

∴ Circumferential pitch of the bolts

$$= \frac{\pi \times D_p}{n} = \frac{\pi \times 215}{6} = 112.6 \text{ mm}$$

We know that for a leak proof joint, the circumferential pitch of the bolts should lie between $20\sqrt{d_1}$ to $30\sqrt{d_1}$, where d_1 is the diameter of the bolt hole in mm.

∴ Minimum circumferential pitch of the bolts

$$= 20\sqrt{d_1} = 20\sqrt{25} = 100 \text{ mm}$$

and maximum circumferential pitch of the bolts

$$= 30\sqrt{d_1} = 30\sqrt{25} = 150 \text{ mm}$$

Since the circumferential pitch of the bolts obtained above is within 100 mm and 150 mm, therefore size of the bolt chosen is satisfactory.

∴ Size of the bolt = M 24 **Ans.**

Design of cover plate

Let t_1 = Thickness of the cover plate.

The semi-cover plate is shown in Fig. 11.27.

We know that the bending moment at *A-A*,

$$M = 0.053\, P \times D_p$$
$$= 0.053 \times 67\,860 \times 215$$
$$= 773\,265 \text{ N-mm}$$

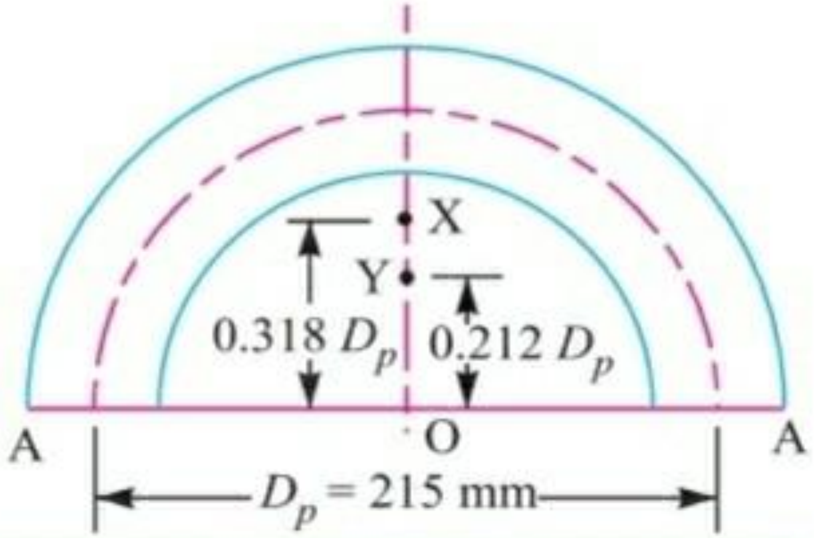

Fig. 11.27

Outside diameter of the cover plate,

$$D_o = D_p + 3d_1 = 215 + 3 \times 25 = 290 \text{ mm}$$

Width of the plate,

$$w = D_o - 2d_1 = 290 - 2 \times 25 = 240 \text{ mm}$$

∴ Section modulus,

$$Z = \frac{1}{6} w (t_1)^2 = \frac{1}{6} \times 240 (t_1)^2 = 40 (t_1)^2 \text{ mm}^3$$

We know that bending (tensile) stress,

$$\sigma_t = M/Z \quad \text{or} \quad 60 = 773\,265 / 40 (t_1)^2$$

$$\therefore \quad (t_1)^2 = 773\,265 / 40 \times 60 = 322 \quad \text{or} \quad t_1 = 18 \text{ mm}$$ **Ans.**

Thank You