**Problem 5**

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**

$D$ = 120 mm or $r$ = 60 mm ; $p$ = 6 N/mm2 ; $\sigma_t$ = 60 MPa = 60 N/mm2 ; $\sigma_{tb}$ = 40 MPa = 40 N/mm$^2$

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

***<u>Design of bolts</u>***

We know that the total upward force acting on the cover plate

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

Let the nominal diameter of the bolt is 24 mm. From Table 1 (coarse series), we find that the corresponding core diameter ($dc$) of the bolt is 20.32 mm.

∴ Resisting force offered by $n$ number of bolts,

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

$n$ = 67 860 / 12 973 = 5.23 say 6

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

$D_p = D + 2t + 3d_1 = 120 + 2 \times 10 + 3 \times 25 = 215$ 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 $\sqrt{d1}$ to 30$\sqrt{d1}$ , where $d1$ is the diameter of the bolt hole in mm.

∴ Minimum circumferential pitch of the bolts

$= 20\sqrt{d1} = 20\sqrt{25} = 100$ mm

and maximum circumferential pitch of the bolts

$= 30\sqrt{d1} = 30\sqrt{25} = 150$ 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

***<u>Design of cover plate</u>***

Let $t_1$ = Thickness of the cover plate. The semi-cover plate is shown in Figure.

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

$$M = 0.053\, P \times D_p$$

$$= 0.053 \times 67\,860 \times 215$$

$$= 773\,265 \text{ N-mm}$$

Outside diameter of the cover plate,

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

Width of the plate,

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

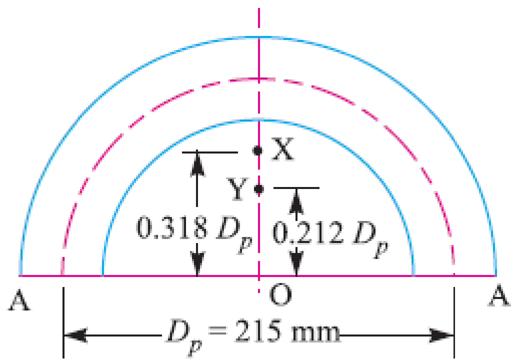
∴ Section modulus,

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

We know that bending (tensile) stress,

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

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




**Problem 6**

A steam engine of effective diameter 300 mm is subjected to a steam pressure of 1.5 N/mm$^2$. 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**

$D$ = 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 $d_c$ = Core diameter of the bolt in mm.

∴ 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,

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

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 $\quad \dfrac{25\,700}{(d_c)^2} = 120$

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

E-mail: LuayHashemAbbud@mustaqbal-college.edu.iq

From Table 1 (coarse series), the standard core diameter is $dc$ = 14.933 mm and the corresponding size of the bolt is M18