DHANALAKSHMI COLLEGE OF ENGINEERING

(Dr.VPR Nagar, Manimangalam, Tambaram)

Chennai - 601 301

DEPARTMENT OF MECHANICAL ENGINEERING

III YEAR MECHANICAL - V SEMESTER

GE6503 – DESIGN OF MACHINE ELEMENTS

ACADEMIC YEAR (2017 - 2018) - ODD SEMESTER

UNIT – 3 (STUDY NOTES)

Prepared by:

R. SENDIL KUMAR, AP/MECH

P.V. ELUMALAI, AP/MECH

P. SIVA KUMAR, AP/MECH

Unit 3: DESIGN OF TEMPORARY AND PERMANENT JOINTS TORSION

PART – A

1. How is a bolt designated? **(A/M 2009)**

A bolt is designated by a letter M followed by nominal diameter and pitch in mm.

2. What factors influence the amount of initial tension?

1. External load 2. Material used 3. Bolt diameter

3. What is bolt of uniform strength?

A bolt of uniform strength has equal strength at the thread and shank portion.

4. What are the ways to produce bolts of uniform strength?

1. Reducing shank diameter equal to root diameter 2.Drilling axial hole

5. What stresses act on screw fastenings?

1.Initial stresses due to screwing up 2.Stresses due to external forces 3.Combined stresses

6. What are the different applications of screwed fasteners?

The different applications of screwed fasteners are

a. For readily connecting & disconnecting machine parts with out damage

b. The parts can be rigidly connected.

c. Used for transmitting power.

7. What are the advantages of screwed fasteners? **(A/M 2009)**

a. They are highly reliable in operation.

b. They are convenient to assemble & disassemble.

c. A wide range of screws can be used for various operating conditions.

d. They are relatively cheap to produce.

8. Define Pitch.

Pitch is defined as the distance from appoint on one thread to the corresponding on the adjacent thread in the same axis plane.

9. Define lead.

Lead is defined as the distance, which a screw thread advances axially in one rotation of the nut.

10. What are the different types of metric thread?

1. BSW (British standard whit worth)

2. BSE (British standard End)

11. Define term of self-locking of power screws. **(A/M 2004) (N/D 2012) (M/J 2013)**

If the friction angle (φ) is greater than helix angle (α) of the power screw, the torque applied to lower the load will be positive, indicating that an effort is applied to lower the load. This type of screw is known as self-locking screw.

12. How is bolt designated? Give example. **(A/M 2009, N/D 2007)**

M30 × 2.5

$M_d \times P$

M – Major diameter of bolt

d – Nonimal diameter in mm

P – Pitch in mm.

13. What is stud? **(N/D 2009)**

A stud is a bolt in which the head is replaced by a threaded end. It passes through one of the parts to be connected and is screwed into the other part.

14. What is meaning of bolt M24 X 2? **(A/M 2008)**

d = 24 – Nonimal diameter, P = 2 – Pitch of the bolt

15. State the advantages of threaded joints. **(N/D 2007)**

a. High clamping b. Small tightening force requirement.

16. Define welding.

Welding can be defined as a process of joining two similar or dissimilar metals with or without application of pressure along with or without addition of filler material.

17. What are the types of welded joints?

1. Butt joint 2. Lap joint 3. T joint 4. Corner joint 5. Edge joint

18. What are the two types of stresses are induced in eccentric loading of loaded jont?

1. Direct shear stress 2. Bending or torsional shear stress

19. Define Butt and Lap joint.

The joint is made by welding the ends or edges of two plates.

The two plates are overlapping each other for a certain distance. Then welded, such welding is called fillet weld.

20. When will the edge preparation need? **(A/M 2006)**

If the two plates to be welded have more than 6 mm thickness, the edge preparation should be carried out.

21. What are the two types of fillet weld? **(A/M 2007, 2006)**

1. Longitudinal or parallel fillet weld. 2. Transverse fillet weld.

22. State the two types of eccentric welded connections.

1. Welded connection subjected to moment in a plane of the weld.

2. Welded connection subjected to moment in a plane normal to the plane of the weld.

23. What are the practical applications of welded joints?

It has employed in manufacturing of machine frames, automobile bodies, aircraft, and structural works.

24. What is Tee-joint?

Two plates are arranged in T shape which means the plates are located at right angles to each other.

25. What is corner joint?

Two plates are arranged at right angles such that it forms an angle.

26. When will the weld deposit be weaker?

When the components are made of high carbon steel or alloy steel, the weld becomes weaker.

27. Define eccentrically loaded welded joints.

The external loaded where applied may not pass through the geometric centre in structural joints are called as eccentrically loaded joints.

28. Why are welded joints preferred over riveted joints? (A/M 2009, 08, 03)

Material is saved in welding and hence the machine element will be light if welded joints are used instead of riveted joints. Leak proof joints can be easily obtained by welded joints compared riveted joints.

29. What is the minimum size for fillet weld? If the required weld size from strength consideration is too small how will you fulfill the condition of minimum weld size?

(A/M 2008)

It is defined as the minimum size of the weld for a given thickness of the thinner part joined or plate to avoid cold cracking by escaping the rapid cooling.

Size of fillet weld h = $\sqrt{2}\ X\ Throat\ thickness$

= 1.4142t

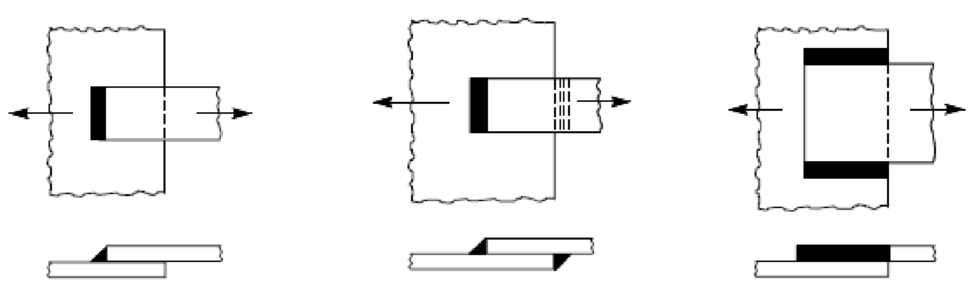
30. Name the possible modes of failure of riveted joint. (A/M 2008) (N/D 2012) (M/J 2012)

1. Crushing of rivets 2.Shearing of rivet 3.Tearing of the plate at the edge 4.Tearing of the plate between rivets.

31. How is welding classified? (A/M 2005)

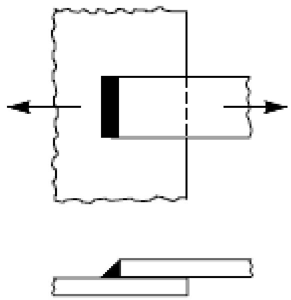
1. Forge welding 2. Electric resistance welding 3. Fusing welding.

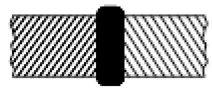
32. Differentiate with a neat sketch the fillet welds subjected to parallel loading and transverse loading. (A/M 2004)



(a) Single transverse. (b) Double transverse. (c) Parallel fillet.

33. Sketch at least two types of welded joint. **(A/M 2007)**





Lap joint **Butt joint**

34. What do you understand by the single start and double start threads? **(N/D 2011)**

A screw made by cutting a single helical groove on the cylinder is known as single threaded (or single-start) screw and if a second thread is cut in the space between the grooves of the first, a double threaded (or double-start) screw is formed.

35. Classify the rivet heads according to IS specifications. **(N/D 2011)**

1. Rivet heads for general purposes (below 12 mm diameter) according to IS : 2155 – 1982 (Reaffirmed 1996).
2. Rivet heads for general purposes (From 12 mm to 48 mm diameter) according to IS : 1929 – 1982 (Reaffirmed 1996).
3. Rivet heads for boiler work (from 12 mm to 48 mm diameter, according to IS : 1928 – 1961 (Reaffirmed 1996).

36. Determine the safe tensile load for a bolt of M20, assuming a safe tensile stress of 40 MPa? **(M/J 2012)**

For M 20 bolt the nominal diameter d_p = 18.37 6mm & the effective diameter is 16.933 mm.

Safe tensile load = Permissible stress × Cross-sectional area at bottom of the thread

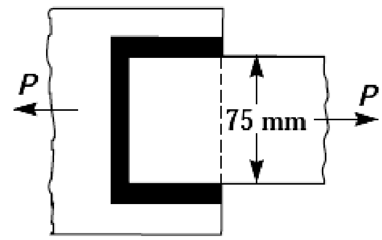
$$= 40 \text{ x } \frac{\pi}{4}\left(\frac{d_p+d_c}{2}\right)^2 = 40 \text{ x } \frac{\pi}{4}\left(\frac{18.376+16.933}{2}\right)^2 = 9786.759 \text{ N}$$

37. Write any two advantages and disadvantages of welded joints over riveted joints. **(M/J 2013)**

Material is saved in welded joints and hence the machine element will be light if welded joints are used instead of riveted joints. Leak proof joints can easily obtained by welded joints compared riveted joints

PART – B

1. A plate 75 mm wide and 12.5 mm thick is joined with another plate by a single transverse weld and a double parallel fillet weld as shown in Fig. the maximum tensile and shear stresses are 70 MPa and 56 MPa respectively. Find the length of each parallel fillet weld, if the joint is subjected to both static and fatigue loading. (APR/MAY 2008)



Solution. Given : Width = 75 mm ; Thickness = 12.5 mm ; σ_τ = 70 MPa = 70 N/mm^2 ; τ = 56 MPa = 56 N/mm^2.

The effective length of weld (l_1) for the transverse weld may be obtained by subtracting 12.5 mm from the width of the plate.

$$\therefore \quad l_1 = 75 - 12.5 = 62.5 \text{ mm}$$

Length of each parallel fillet for static loading

Let l_2 = Length of each parallel fillet.

We know that the maximum load which the plate can carry is

$$P = \text{Area} \times \text{Stress} = 75 \times 12.5 \times 70 = 65\,625 \text{ N}$$

Load carried by single transverse weld,

$$P_1 = 0.707\, s \times l_1 \times \sigma_t = 0.707 \times 12.5 \times 62.5 \times 70$$

and the load carried by double parallel fillet weld,

$$P_2 = 1.414\, s \times l_2 \times \tau = 1.414 \times 12.5 \times l_2 \times 56 = 9$$

$\therefore$ Load carried by the joint (P),

$$65\,625 = P_1 + P_2 = 38\,664 + 990\, l_2 \quad \text{or} \quad l_2 = 27.2 \text{ mm}$$

Adding 12.5 mm for starting and stopping of weld run, we have

$$l_2 = 27.2 + 12.5 = 39.7 \text{ say } 40 \text{ mm } \textbf{Ans.}$$

Length of each parallel fillet for fatigue loading

From Table 10.6, we find that the stress concentration factor for transverse welds is 1.5 and for parallel fillet welds is 2.7.

$\therefore$ Permissible tensile stress,

$$\sigma_t = 70 / 1.5 = 46.7 \text{ N/mm}^2$$

and permissible shear stress,

$$\tau = 56 / 2.7 = 20.74 \text{ N/mm}^2$$

Load carried by single transverse weld,

$$P_1 = 0.707\, s \times l_1 \times \sigma_t = 0.707 \times 12.5 \times 62.5 \times 46.7 = 25\,795 \text{ N}$$

and load carried by double parallel fillet weld,

$$P_2 = 1.414\, s \times l_2 \times \tau = 1.414 \times 12.5\, l_2 \times 20.74 = 366\, l_2 \text{ N}$$

$\therefore$ Load carried by the joint (P),

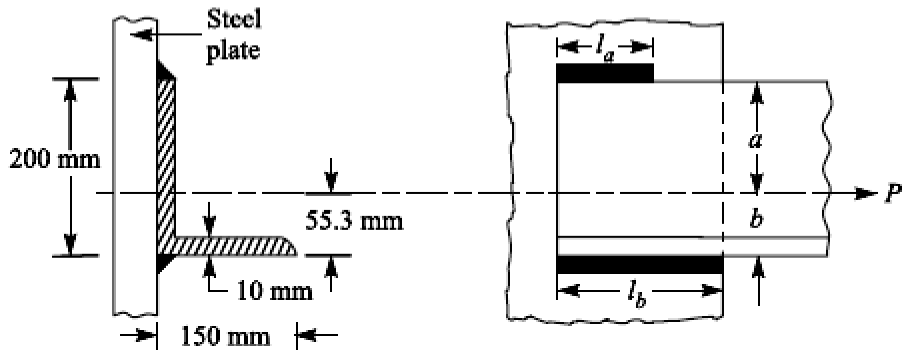
$$65\,625 = P_1 + P_2 = 25\,795 + 366\, l_2 \quad \text{or} \quad l_2 = 108.8 \text{ mm}$$

Adding 12.5 mm for starting and stopping of weld run, we have

$$l_2 = 108.8 + 12.5 = 121.3 \text{ mm} \quad \textbf{Ans.}$$

2. A 200 × 150 × 10 mm angle is to be welded to a steel plate by fillet welds as shown in Fig. If the angle is subjected to a static load of 200 kN, find the length of weld at the top and bottom. The allowable shear stress for static loading may be taken as 75 MPa.

(NOV/DEC 2004)



Solution. Given : $a + b = 200$ mm ; $P = 200$ kN $= 200 \times 10^3$ N ; $\tau = 75$ MPa $= 75$ N/mm^2

Let l_a = Length of weld at the top,

l_b = Length of weld at the bottom, and

l = Total length of the weld $= l_a + l_b$

Since the thickness of the angle is 10 mm, therefore size of weld,

$$s = 10 \text{ mm}$$

We know that for a single parallel fillet weld, the maximum load (P),

$$200 \times 10^3 = 0.707\, s \times l \times \tau = 0.707 \times 10 \times l \times 75 = 530.25\, l$$

$$\therefore \quad l = 200 \times 10^3 / 530.25 = 377 \text{ mm}$$

or $$l_a + l_b = 377 \text{ mm}$$

Now let us find out the position of the centroidal axis.

Let b = Distance of centroidal axis from the bottom of the angle.

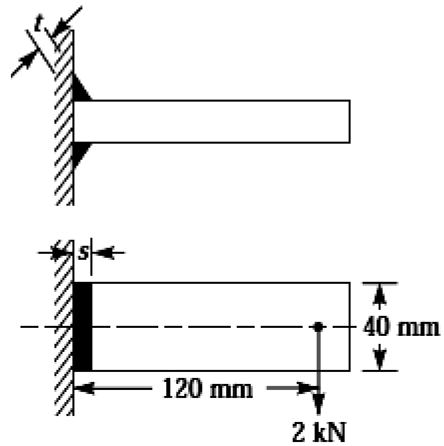
$$\therefore \quad b = \frac{(200 - 10)\, 10 \times 95 + 150 \times 10 \times 5}{190 \times 10 + 150 \times 10} = 55.3 \text{ mm}$$

and $$a = 200 - 55.3 = 144.7 \text{ mm}$$

We know that $$l_a = \frac{l \times b}{a + b} = \frac{377 \times 55.3}{200} = 104.2 \text{ mm} \quad \textbf{Ans.}$$

and $$l_b = l - l_a = 377 - 104.2 = 272.8 \text{ mm} \quad \textbf{Ans.}$$

3. A welded joint as shown in Fig. 10.24, is subjected to an eccentric load of 2 kN. Find the size of weld, if the maximum shear stress in the weld is 25 MPa. (NOV/DEC 2011)



Solution. Given: $P = 2\text{kN} = 2000$ N ; $e = 120$ mm ; $l = 40$ mm ; $\tau_{max} = 25$ MPa $= 25$ N/mm^2

Let s = Size of weld in mm, and

t = Throat thickness.

The joint, as shown in Fig. 10.24, will be subjected to direct shear stress due to the shear force, $P = 2000$ N and bending stress due to the bending moment of $P \times e$.

We know that area at the throat,

$$A = 2t \times l = 2 \times 0.707\, s \times l = 1.414\, s \times l = 1.414\, s \times 40 = 56.56 \times s \text{ mm}^2$$

$\therefore$ Shear stress,

$$\tau = \frac{P}{A} = \frac{2000}{56.56 \times s} = \frac{35.4}{s} \text{ N/mm}^2$$

Bending moment, $M = P \times e = 2000 \times 120 = 240 \times 10^3$ N-mm

Section modulus of the weld through the throat,

$$Z = \frac{s \times l^2}{4.242} = \frac{s\,(40)^2}{4.242} = 377 \times s \text{ mm}^3$$

$\therefore$ Bending stress,

$$\sigma_b = \frac{M}{Z} = \frac{240 \times 10^3}{377 \times s} = \frac{636.6}{s} \text{ N/mm}^2$$

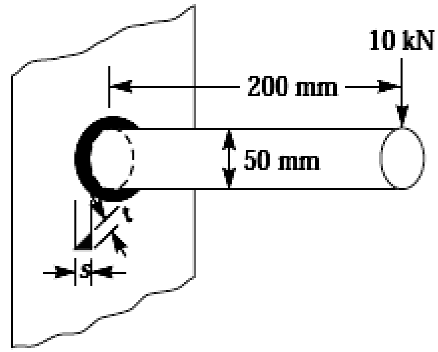
We know that maximum shear stress (τ_{max}),

$$25 = \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\,\tau^2} = \frac{1}{2}\sqrt{\left(\frac{636.6}{s}\right)^2 + 4\left(\frac{35.4}{s}\right)^2} = \frac{320.3}{s}$$

$\therefore$ $s = 320.3 / 25 = 12.8$ mm **Ans.**

4. A 50 mm diameter solid shaft is welded to a flat plate as shown in Fig. If the size of the weld is 15 mm, find the maximum normal and shear stress in the weld.

(NOV/DEC 2006/2007)



Solution. Given : $D = 50$ mm ; $s = 15$ mm ; $P = 10$ kN $= 10\,000$ N ; $e = 200$ mm

Let t = Throat thickness.

The joint, as shown in Fig. 10.25, is subjected to direct shear stress and the bending stress. We know that the throat area for a circular fillet weld,

$$A = t \times \pi D = 0.707\, s \times \pi D = 0.707 \times 15 \times \pi \times 50 = 1666 \text{ mm}^2$$

∴ Direct shear stress,

$$\tau = \frac{P}{A} = \frac{10\,000}{1666} = 6 \text{ N/mm}^2 = 6 \text{ MPa}$$

We know that bending moment,

$$M = P \times e = 10\,000 \times 200 = 2 \times 10^6 \text{ N-mm}$$

From Table 10.7, we find that for a circular section, section modulus,

$$Z = \frac{\pi\, t\, D^2}{4} = \frac{\pi \times 0.707\, s \times D^2}{4} = \frac{\pi \times 0.707 \times 15\,(50)^2}{4} = 20\,825 \text{ mm}^3$$

∴ Bending stress,

$$\sigma_b = \frac{M}{Z} = \frac{2 \times 10^6}{20\,825} = 96 \text{ N/mm}^2 = 96 \text{ MPa}$$

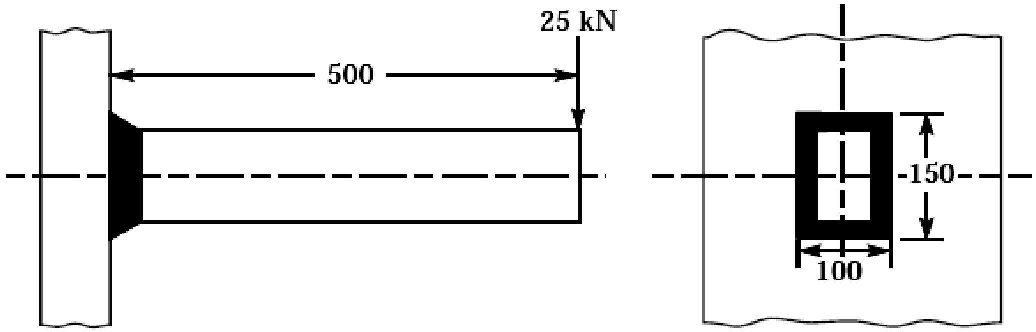
Maximum normal stress

We know that the maximum normal stress,

$$\sigma_{t(max)} = \frac{1}{2}\,\sigma_b + \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\,\tau^2} = \frac{1}{2} \times 96 + \frac{1}{2}\sqrt{(96)^2 + 4 \times 6^2}$$

$$= 48 + 48.4 = 96.4 \text{ MPa} \quad \textbf{Ans.}$$

5. A rectangular cross-section bar is welded to a support by means of fillet welds as shown in Fig. Determine the size of the welds, if the permissible shear stress in the weld is limited to 75 MPa. **(NOV/DEC 2011)**



All dimensions in mm

Solution. Given : P = 25 kN = 25 × 10^3N ; τ_{max} = 75 MPa = 75N/mm^2 ; l = 100 mm ; b = 150 mm ; e = 500 mm

Let s = Size of the weld, and

t = Throat thickness.

The joint, as shown in Fig. 10.26, is subjected to direct shear stress and the bending stress. We know that the throat area for a rectangular fillet weld,

$$A = t\,(2b + 2l) = 0.707\,s\,(2b + 2l)$$

$$= 0.707s\,(2 \times 150 + 2 \times 100) = 353.5\,s \text{ mm}^2 \qquad \ldots (\because\ t = 0.707s)$$

$\therefore$ Direct shear stress, $\tau = \dfrac{P}{A} = \dfrac{25 \times 10^3}{353.5\,s} = \dfrac{70.72}{s}$ N/mm^2

We know that bending moment,

$$M = P \times e = 25 \times 10^3 \times 500 = 12.5 \times 10^6 \text{ N-mm}$$

From Table 10.7, we find that for a rectangular section, section modulus,

$$Z = t\left(b.l + \frac{b^2}{3}\right) = 0.707\,s\left[150 \times 100 + \frac{(150)^2}{3}\right] = 15\,907.5\,s \text{ mm}^3$$

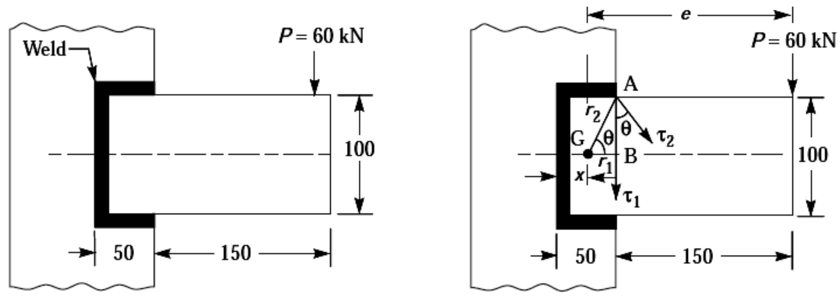
$\therefore$ Bending stress, $\sigma_b = \dfrac{M}{Z} = \dfrac{12.5 \times 10^6}{15\,907.5\,s} = \dfrac{785.8}{s}$ N/mm^2

We know that maximum shear stress (τ_{max}),

$$75 = \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\,\tau^2} = \frac{1}{2}\sqrt{\left(\frac{785.8}{s}\right)^2 + 4\left(\frac{70.72}{s}\right)^2} = \frac{399.2}{s}$$

$\therefore$ s = 399.2 / 75 = 5.32 mm **Ans.**

6. A rectangular steel plate is welded as a cantilever to a vertical column and supports a single concentrated load P, as shown in Fig. Determine the weld size if shear stress in the same is not to exceed 140 MPa. (NOV/DEC 2012, MAY/JUNE 2013)



All dimensions in mm.

Solution. Given : $P = 60$ kN $= 60 \times 10^3$ N ; $b = 100$ mm ; $l = 50$ mm ; $\tau = 140$ MPa $= 140$ N/mm^2

Let s = Weld size, and

t = Throat thickness.

First of all, let us find the centre of gravity (G) of the weld system, as shown in Fig. 10.31.

Let x be the distance of centre of gravity (G) from the left hand edge of the weld system. From Table 10.7, we find that for a section as shown in Fig. 10.31,

$$x = \frac{l^2}{2l + b} = \frac{(50)^2}{2 \times 50 + 100} = 12.5 \text{ mm}$$

and polar moment of inertia of the throat area of the weld system about G,

$$J = t\left[\frac{(b + 2l)^3}{12} - \frac{l^2\,(b + l)^2}{b + 2l}\right]$$

$$= 0.707\,s\left[\frac{(100 + 2 \times 50)^3}{12} - \frac{(50)^2\,(100 + 50)^2}{100 + 2 \times 50}\right] \quad \ldots (\because\ t = 0.707\,s)$$

$$= 0.707s\,[670 \times 10^3 - 281 \times 10^3] = 275 \times 10^3\,s \text{ mm}^4$$

Distance of load from the centre of gravity (G) *i.e.* eccentricity,

$$e = 150 + 50 - 12.5 = 187.5 \text{ mm}$$

$$r_1 = BG = 50 - x = 50 - 12.5 = 37.5 \text{ mm}$$

$$AB = 100 / 2 = 50 \text{ mm}$$

We know that maximum radius of the weld,

$$r_2 = \sqrt{(AB)^2 + (BG)^2} = \sqrt{(50)^2 + (37.5)^2} = 62.5 \text{ mm}$$

$$\therefore \quad \cos\theta = \frac{r_1}{r_2} = \frac{37.5}{62.5} = 0.6$$

We know that throat area of the weld system,

$$A = 2 \times 0.707s \times l + 0.707s \times b = 0.707\,s\,(2l + b)$$

$$= 0.707s\,(2 \times 50 + 100) = 141.4\,s \text{ mm}^2$$

$\therefore$ Direct or primary shear stress,

$$\tau_1 = \frac{P}{A} = \frac{60 \times 10^3}{141.4\,s} = \frac{424}{s} \text{ N/mm}^2$$

and shear stress due to the turning moment or secondary shear stress,

$$\tau_2 = \frac{P \times e \times r_2}{J} = \frac{60 \times 10^3 \times 187.5 \times 62.5}{275 \times 10^3\ s} = \frac{2557}{s} \text{ N/mm}^2$$

We know that the resultant shear stress,

$$\tau = \sqrt{(\tau_1)^2 + (\tau_2)^2 + 2\,\tau_1 \times \tau_2 \times \cos\theta}$$

$$140 = \sqrt{\left(\frac{424}{s}\right)^2 + \left(\frac{2557}{s}\right)^2 + 2 \times \frac{424}{s} \times \frac{2557}{s} \times 0.6} = \frac{2832}{s}$$

$\therefore$ $s = 2832 / 140 = 20.23$ mm **Ans.**

7. Design a knuckle joint to transmit 150 kN. The design stresses may be taken as 75 MPa in tension, 60 MPa in shear and 150 MPa in compression.

(NOV/DEC 2011, NOV/DEC 2012)

Solution. Given : $P = 150$ kN $= 150 \times 10^3$ N ; $\sigma_t = 75$ MPa $= 75$ N/mm^2 ; $\tau = 60$ MPa $= 60$ N/mm^2 ; $\sigma_c = 150$ MPa $= 150$ N/mm^2

The knuckle joint is shown in Fig. 12.16. The joint is designed by considering the various methods of failure as discussed below :

1. ***Failure of the solid rod in tension***

Let d = Diameter of the rod.

We know that the load transmitted (P),

$$150 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 75 = 59\,d^2$$

$\therefore$ $d^2 = 150 \times 10^3 / 59 = 2540$ or $d = 50.4$ say 52 mm **Ans.**

Now the various dimensions are fixed as follows :

Diameter of knuckle pin,

$$d_1 = d = 52 \text{ mm}$$

Outer diameter of eye, $d_2 = 2\,d = 2 \times 52 = 104$ mm

Diameter of knuckle pin head and collar,

$$d_3 = 1.5\,d = 1.5 \times 52 = 78 \text{ mm}$$

Thickness of single eye or rod end,

$$t = 1.25\,d = 1.25 \times 52 = 65 \text{ mm}$$

Thickness of fork, $t_1 = 0.75\,d = 0.75 \times 52 = 39$ say 40 mm

Thickness of pin head, $t_2 = 0.5\,d = 0.5 \times 52 = 26$ mm

2. ***Failure of the knuckle pin in shear***

Since the knuckle pin is in double shear, therefore load (P),

$$150 \times 10^3 = 2 \times \frac{\pi}{4} \times (d_1)^2\,\tau = 2 \times \frac{\pi}{4} \times (52)^2\,\tau = 4248\,\tau$$

$$\therefore \quad \tau = 150 \times 10^3 / 4248 = 35.3 \text{ N/mm}^2 = 35.3 \text{ MPa}$$

3. ***Failure of the single eye or rod end in tension***

The single eye or rod end may fail in tension due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\,t \times \sigma_t = (104 - 52)\,65 \times \sigma_t = 3380\,\sigma_t$$

$$\therefore \quad \sigma_t = 150 \times 10^3 / 3380 = 44.4 \text{ N / mm}^2 = 44.4 \text{ MPa}$$

4. ***Failure of the single eye or rod end in shearing***

The single eye or rod end may fail in shearing due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\,t \times \tau = (104 - 52)\,65 \times \tau = 3380\,\tau$$

$$\therefore \quad \tau = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

5. ***Failure of the single eye or rod end in crushing***

The single eye or rod end may fail in crushing due to the load. We know that load (P),

$$150 \times 10^3 = d_1 \times t \times \sigma_c = 52 \times 65 \times \sigma_c = 3380\,\sigma_c$$

$$\therefore \quad \sigma_c = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

6. ***Failure of the forked end in tension***

The forked end may fail in tension due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\,2\,t_1 \times \sigma_t = (104 - 52)\,2 \times 40 \times \sigma_t = 4160\,\sigma_t$$

$$\therefore \quad \sigma_t = 150 \times 10^3 / 4160 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

7. ***Failure of the forked end in shear***

The forked end may fail in shearing due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\,2\,t_1 \times \tau = (104 - 52)\,2 \times 40 \times \tau = 4160\,\tau$$

$$\therefore \quad \tau = 150 \times 10^3 / 4160 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

8. ***Failure of the forked end in crushing***

The forked end may fail in crushing due to the load. We know that load (P),

$$150 \times 10^3 = d_1 \times 2\,t_1 \times \sigma_c = 52 \times 2 \times 40 \times \sigma_c = 4160\,\sigma_c$$

$$\therefore \quad \sigma_c = 150 \times 10^3 / 4180 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

From above, we see that the induced stresses are less than the given design stresses, therefore the joint is safe.

8. Design and draw a cotter joint to support a load varying from 30 kN in compression to 30 kN in tension. The material used is carbon steel for which the following allowable stresses may be used. The load is applied statically. Tensile stress = compressive stress = 50 MPa; shear stress = 35 MPa and crushing stress = 90 MPa. (MAY/JUNE 2013)

Solution. Given : $P = 30$ kN $= 30 \times 10^3$ N ; $\sigma_t = 50$ MPa $= 50$ N / mm^2 ; $\tau = 35$ MPa $= 35$ N / mm^2 ; $\sigma_c = 90$ MPa $= 90$ N/mm^2

1. ***Diameter of the rods***

Let d = Diameter of the rods.

Considering the failure of the rod in tension. We know that load (P),

$$30 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 50 = 39.3\, d^2$$

$\therefore$ $d^2 = 30 \times 10^3 / 39.3 = 763$ or $d = 27.6$ say 28 mm **Ans.**

2. ***Diameter of spigot and thickness of cotter***

Let d_2 = Diameter of spigot or inside diameter of socket, and

t = Thickness of cotter. It may be taken as $d_2 / 4$.

Considering the failure of spigot in tension across the weakest section. We know that load (P),

$$30 \times 10^3 = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times \frac{d_2}{4}\right]50 = 26.8\,(d_2)^2$$

$\therefore$ $(d_2)^2 = 30 \times 10^3 / 26.8 = 1119.4$ or $d_2 = 33.4$ say 34 mm

and thickness of cotter, $t = \dfrac{d_2}{4} = \dfrac{34}{4} = 8.5$ mm

Let us now check the induced crushing stress. We know that load (P),

$$30 \times 10^3 = d_2 \times t \times \sigma_c = 34 \times 8.5 \times \sigma_c = 289\,\sigma_c$$

$\therefore$ $\sigma_c = 30 \times 10^3 / 289 = 103.8$ N/mm^2

Since this value of σ_c is more than the given value of $\sigma_c = 90$ N/mm^2, therefore the dimensions d_2 = 34 mm and $t = 8.5$ mm are not safe. Now let us find the values of d_2 and t by substituting the value of $\sigma_c = 90$ N/mm^2 in the above expression, *i.e.*

$$30 \times 10^3 = d_2 \times \frac{d_2}{4} \times 90 = 22.5\,(d_2)^2$$

$\therefore$ $(d_2)^2 = 30 \times 10^3 / 22.5 = 1333$ or $d_2 = 36.5$ say 40 mm **Ans.**

and $t = d_2 / 4 = 40 / 4 = 10$ mm **Ans.**

3. ***Outside diameter of socket***

Let d_1 = Outside diameter of socket.

Considering the failure of the socket in tension across the slot. We know that load (P),

$$30 \times 10^3 = \left[\frac{\pi}{4}\left\{(d_1)^2 - (d_2)^2\right\} - (d_1 - d_2)\,t\right]\sigma_t$$

$$= \left[\frac{\pi}{4}\left\{(d_1)^2 - (40)^2\right\} - (d_1 - 40)\,10\right]50$$

$$30 \times 10^3/50 = 0.7854\,(d_1)^2 - 1256.6 - 10\,d_1 + 400$$

or $$(d_1)^2 - 12.7\,d_1 - 1854.6 = 0$$

$$\therefore \quad d_1 = \frac{12.7 \pm \sqrt{(12.7)^2 + 4 \times 1854.6}}{2} = \frac{12.7 \pm 87.1}{2}$$

$$= 49.9 \text{ say } 50 \text{ mm } \textbf{Ans.} \qquad \text{...(Taking +ve sign)}$$

4. *Width of cotter*

Let b = Width of cotter.

Considering the failure of the cotter in shear. Since the cotter is in double shear, therefore load (P),

$$30 \times 10^3 = 2\,b \times t \times \tau = 2\,b \times 10 \times 35 = 700\,b$$

$$\therefore \quad b = 30 \times 10^3 / 700 = 43 \text{ mm } \textbf{Ans.}$$

5. *Diameter of socket collar*

Let d_4 = Diameter of socket collar.

Considering the failure of the socket collar and cotter in crushing. We know that load (P),

$$30 \times 10^3 = (d_4 - d_2)\,t \times \sigma_c = (d_4 - 40)10 \times 90 = (d_4 - 40)\,900$$

$$\therefore \quad d_4 - 40 = 30 \times 10^3 / 900 = 33.3 \text{ or } d_4 = 33.3 + 40 = 73.3 \text{ say } 75 \text{ mm } \textbf{Ans.}$$

6. *Thickness of socket collar*

Let c = Thickness of socket collar.

Considering the failure of the socket end in shearing. Since the socket end is in double shear, therefore load (P),

$$30 \times 10^3 = 2(d_4 - d_2)\,c \times \tau = 2\,(75 - 40)\,c \times 35 = 2450\,c$$

$$\therefore \quad c = 30 \times 10^3 / 2450 = 12 \text{ mm } \textbf{Ans.}$$

7. *Distance from the end of the slot to the end of the rod*

Let a = Distance from the end of slot to the end of the rod.

Considering the failure of the rod end in shear. Since the rod end is in double shear, therefore load (P),

$$30 \times 10^3 = 2\,a \times d_2 \times \tau = 2a \times 40 \times 35 = 2800\,a$$

$$\therefore \quad a = 30 \times 10^3 / 2800 = 10.7 \text{ say } 11 \text{ mm } \textbf{Ans.}$$

8. *Diameter of spigot collar*

Let d_3 = Diameter of spigot collar.

Considering the failure of spigot collar in crushing. We know that load (P),

$$30 \times 10^3 = \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]\sigma_c = \frac{\pi}{4}\left[(d_3)^2 - (40)^2\right]90$$

or $$(d_3)^2 - (40)^2 = \frac{30 \times 10^3 \times 4}{90 \times \pi} = 424$$

$$\therefore \quad (d_3)^2 = 424 + (40)^2 = 2024 \text{ or } d_3 = 45 \text{ mm } \textbf{Ans.}$$

9. *Thickness of spigot collar*

Let t_1 = Thickness of spigot collar.

Considering the failure of spigot collar in shearing. We know that load (P),

$$30 \times 10^3 = \pi d_2 \times t_1 \times \tau = \pi \times 40 \times t_1 \times 35 = 4400 t_1$$

$\therefore$ $t_1 = 30 \times 10^3 / 4400 = 6.8$ say 8 mm **Ans.**

10. The length of cotter (l) is taken as 4 d.

$\therefore$ $l = 4d = 4 \times 28 = 112$ mm **Ans.**

11. The dimension e is taken as 1.2 d.

$\therefore$ $e = 1.2 \times 28 = 33.6$ say 34 mm **Ans.**

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. (NOV/DEC 2007/2011)

Solution. Given : D = 300 mm ; p = 1.5 N/mm^2 ; n = 8 ; σ_y = 330 MPa = 330 N/mm^2; σ_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,

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

$\therefore$ 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.

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,

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

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

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

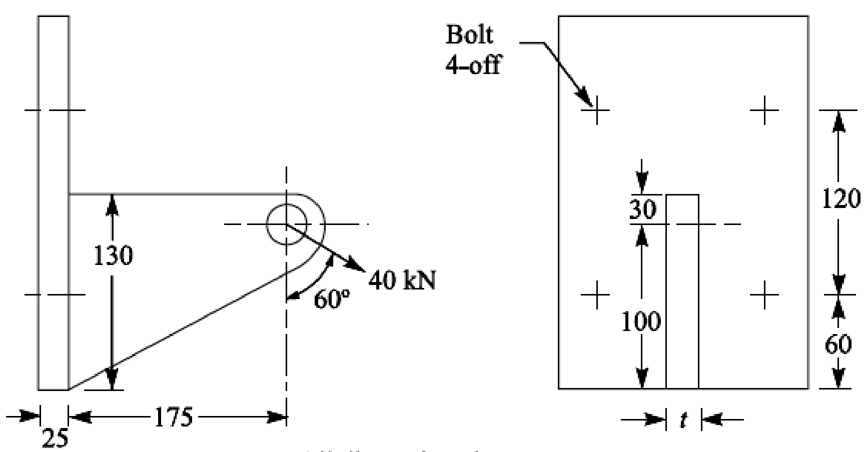
$$\frac{4220}{(d_c)^2} = 240\left(\frac{1}{2} - \frac{29\,534}{(d_c)^2\,330}\right) = 120 - \frac{21\,480}{(d_c)^2}$$

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.**

10. Determine the size of the bolts and the thickness of the arm for the bracket as shown in Fig. if it carries a load of 40 kN at an angle of 60° to the vertical. The material of the bracket and the bolts is same for which the safe stresses can be assumed as 70, 50 and 105 MPa in tension, shear and compression respectively. (NOV/DEC 2008)



All dimensions in mm.

Solution. Given : $W = 40$ kN $= 40 \times 10^3$ N ; $\sigma_t = 70$ MPa $= 70$N/mm^2 ; $\tau = 50$ MPa $= 50$ N/mm^2; $\sigma_c = 105$ MPa $= 105$ N/mm^2

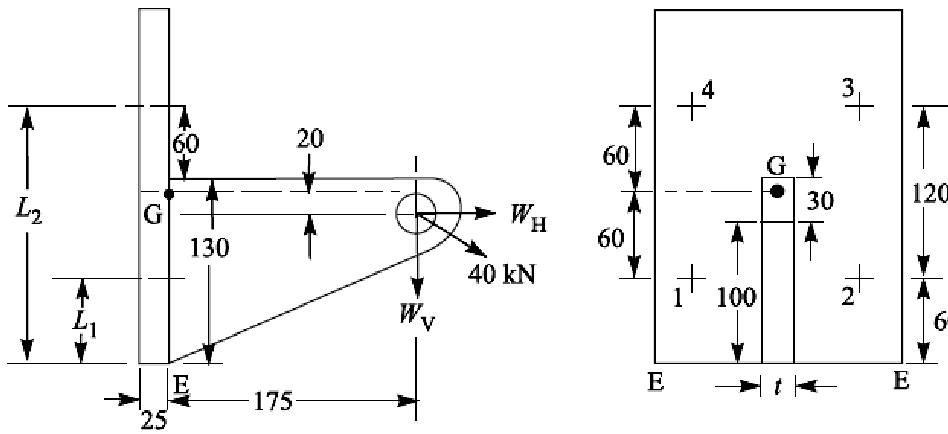
Since the load $W = 40$ kN is inclined at an angle of 60° to the vertical, therefore resolving it into horizontal and vertical components. We know that horizontal component of 40 kN,

$$W_H = 40 \times \sin 60° = 40 \times 0.866 = 34.64 \text{ kN} = 34\,640 \text{ N}$$

and vertical component of 40 kN,

$$W_V = 40 \times \cos 60° = 40 \times 0.5 = 20 \text{ kN} = 20\,000 \text{ N}$$

Due to the horizontal component (W_H), which acts parallel to the axis of the bolts as shown in Fig. 11.37, the following two effects are produced :



1. A direct tensile load equally shared by all the four bolts, and
2. A turning moment about the centre of gravity of the bolts, in the anticlockwise direction.

$\therefore$ Direct tensile load on each bolt,

$$W_{t1} = \frac{W_H}{4} = \frac{34\,640}{4} = 8660 \text{ N}$$

Since the centre of gravity of all the four bolts lies in the centre at G (because of symmetrical bolts), therefore the turning moment is in the anticlockwise direction. From the geometry of the Fig. 11.37, we find that the distance of horizontal component from the centre of gravity (G) of the bolts

$$= 60 + 60 - 100 = 20 \text{ mm}$$

$\therefore$ Turning moment due to W_H about G,

$$T_H = W_H \times 20 = 34\,640 \times 20 = 692.8 \times 10^3 \text{ N-mm} \qquad \text{...(Anticlockwise)}$$

Due to the vertical component W_V, which acts perpendicular to the axis of the bolts as shown in Fig. 11.37, the following two effects are produced:

1. A direct shear load equally shared by all the four bolts, and
2. A turning moment about the edge of the bracket in the clockwise direction.

$\therefore$ Direct shear load on each bolt,

$$W_s = \frac{W_V}{4} = \frac{20\,000}{4} = 5000 \text{ N}$$

Distance of vertical component from the edge E of the bracket,

$$= 175 \text{ mm}$$

$\therefore$ Turning moment due to W_V about the edge of the bracket,

$$T_V = W_V \times 175 = 20\,000 \times 175 = 3500 \times 10^3 \text{ N-mm} \quad \text{(Clockwise)}$$

From above, we see that the clockwise moment is greater than the anticlockwise moment, therefore,

Net turning moment $= 3500 \times 10^3 - 692.8 \times 10^3 = 2807.2 \times 10^3$ N-mm (Clockwise) ...*(i)*

Due to this clockwise moment, the bracket tends to tilt about the lower edge E.

Let w = Load on each bolt per mm distance from the edge E due to the turning effect of the bracket,

L_1 = Distance of bolts 1 and 2 from the tilting edge E = 60 mm, and

L_2 = Distance of bolts 3 and 4 from the tilting edge E

$= 60 + 120 = 180$ mm

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

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

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

$$= 2w\,(L_1)^2 + 2w(L_2)^2 = 2w\,(60)^2 + 2w(180)^2$$

$$= 72\,000\,w \text{ N-mm} \quad \text{...(ii)}$$

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

$$w = 2807.2 \times 10^3 / 72\,000 = 39 \text{ N/mm}$$

Since the heavily loaded bolts are those which lie at a greater distance from the tilting edge, therefore the upper bolts 3 and 4 will be heavily loaded. Thus the diameter of the bolt should be based on the load on the upper bolts. We know that the maximum tensile load on each upper bolt,

$$W_{t2} = w.L_2 = 39 \times 180 = 7020 \text{ N}$$

$\therefore$ Total tensile load on each of the upper bolt,

$$W_t = W_{t1} + W_{t2} = 8660 + 7020 = 15\,680 \text{ N}$$

Since each upper bolt is subjected to a tensile load (W_t = 15 680 N) and a shear load (W_s = 5000 N), therefore equivalent tensile load,

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

$$= \frac{1}{2}\left[15\,680 + \sqrt{(15\,680)^2 + 4(5000)^2}\right] \text{N}$$

$$= \frac{1}{2}\left[15\,680 + 18\,600\right] = 17\,140 \text{ N} \quad \text{...(iii)}$$

Size of the bolts

Let d_c = Core diameter of the bolts.

We know that tensile load on each bolt

$$= \frac{\pi}{2} (d_c)^2 \sigma_t = \frac{\pi}{4} (d_c)^2 70 = 55 (d_c)^2 \text{ N} \quad \text{...}(iv)$$

From equations ***(iii)*** and ***(iv)***, we get

$$(d_c)^2 = 17\,140 / 55 = 311.64 \quad \text{or} \quad d_c = 17.65 \text{ mm}$$

From Table 11.1 (coarse series), we find that the standard core diameter is 18.933 mm and corresponding size of the bolt is M 22. **Ans.**

Thickness of the arm of the bracket

Let t = Thickness of the arm of the bracket in mm, and

b = Depth of the arm of the bracket = 130 mm ...(Given)

We know that cross-sectional area of the arm,

$$A = b \times t = 130\, t \text{ mm}^2$$

and section modulus of the arm,

$$Z = \frac{1}{6} t\, (b)^2 = \frac{1}{6} \times t\, (130)^2 = 2817\, t \text{ mm}^3$$

Due to the horizontal component W_H, the following two stresses are induced in the arm :

1. Direct tensile stress,

$$\sigma_{t1} = \frac{W_H}{A} = \frac{34\,640}{130\, t} = \frac{266.5}{t} \text{ N/mm}^2$$

2. Bending stress causing tensile in the upper most fibres of the arm and compressive in the lower most fibres of the arm. We know that the bending moment of W_H about the centre of gravity of the arm,

$$M_H = W_H \left(100 - \frac{130}{2}\right) = 34\,640 \times 35 = 1212.4 \times 10^3 \text{ N-mm}$$

$\therefore$ Bending stress, $\sigma_{t2} = \dfrac{M_H}{Z} = \dfrac{1212.4 \times 10^3}{2817\, t} = \dfrac{430{\cdot}4}{t} \text{ N/mm}^2$

Due to the vertical component W_V, the following two stresses are induced in the arm :

1. Direct shear stress,

$$\tau = \frac{W_V}{A} = \frac{20\,000}{130\, t} = \frac{154}{t} \text{ N/mm}^2$$

2. Bending stress causing tensile stress in the upper most fibres of the arm and compressive in the lower most fibres of the arm.

Assuming that the arm extends upto the plate used for fixing the bracket to the structure. This assumption gives stronger section for the arm of the bracket.

$\therefore$ Bending moment due to W_V,

$$M_V = W_V (175 + 25) = 20\ 000 \times 200 = 4 \times 10^6 \text{ N-mm}$$

and bending stress,
$$\sigma_{t3} = \frac{M_V}{Z} = \frac{4 \times 10^6}{2817\ t} = \frac{1420}{t} \text{ N/mm}^2$$

Net tensile stress induced in the upper most fibres of the arm of the bracket,

$$\sigma_t = \sigma_{t1} + \sigma_{t2} + \sigma_{t3} = \frac{266.5}{t} + \frac{430.4}{t} + \frac{1420}{t} = \frac{2116.9}{t} \text{ N/mm}^2 \quad \text{...}(v)$$

We know that maximum tensile stress $[\sigma_{t(max)}]$,

$$70 = \frac{1}{2}\sigma_t + \frac{1}{2}\sqrt{(\sigma_t)^2 + 4\tau^2}$$

$$= \frac{1}{2} \times \frac{2116.9}{t} + \frac{1}{2}\sqrt{\left(\frac{2116.9}{t}\right)^2 + 4\left(\frac{154}{t}\right)^2}$$

$$= \frac{1058.45}{t} + \frac{1069.6}{t} = \frac{2128.05}{t}$$

$\therefore$ $t = 2128.05 / 70 = 30.4$ say 31 mm **Ans.**

Let us now check the shear stress induced in the arm. We know that maximum shear stress,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_t)^2 + 4\tau^2} = \frac{1}{2}\sqrt{\left(\frac{2116.9}{t}\right)^2 + 4\left(\frac{154}{t}\right)^2}$$

$$= \frac{1069.6}{t} = \frac{1069.6}{31} = 34.5 \text{ N/mm}^2 = 34.5 \text{ MPa}$$

Since the induced shear stress is less than the permissible stress (50 MPa), therefore the design is safe.

Notes : 1. The value of 't' may be obtained as discussed below :

Since the shear stress at the upper most fibres of the arm of the bracket is zero, therefore equating equation (v) to the given safe tensile stress (*i.e.* 70 MPa), we have

$$\frac{2116.9}{t} = 70 \quad \text{or} \quad t = 2116.9 / 70 = 30.2 \text{ say 31 mm } \textbf{Ans.}$$

2. If the compressive stress in the lower most fibres of the arm is taken into consideration, then the net compressive stress induced in the lower most fibres of the arm,

$$\sigma_c = \sigma_{c1} + \sigma_{c2} + \sigma_{c3}$$

$$= -\sigma_{t1} + \sigma_{t2} + \sigma_{t3}$$

... ($\because$ The magnitude of tensile and compressive stresses is same.)

$$= -\frac{266.5}{t} + \frac{430.4}{t} + \frac{1420}{t} = \frac{1583.9}{t} \text{ N/mm}^2$$

Since the safe compressive stress is 105 N/mm^2, therefore

$$105 = \frac{1583.9}{t} \quad \text{or} \quad t = 1583.9 / 105 = 15.1 \text{ mm}$$

This value of thickness is low as compared to 31 mm as calculated above. Since the higher value is taken, therefore

$$t = 31 \text{ mm } \textbf{Ans.}$$

11. 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. (NOV/DEC 2009)

Solution. Given : D = 300 mm ; p = 1.5 N/mm^2 ; n = 8 ; σ_y = 330 MPa = 330 N/mm^2; σ_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 $\quad \dfrac{25\,700}{(d_c)^2} = 120$

$\therefore \quad (d_c)^2 = 25\,700 / 120 = 214 \quad$ 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.**

<u>Review 16 marks question</u>

12. Two lengths of mild steel tie rod having width 200 mm are to be connected by means of Lozenge joint with two cover plates to withstand a tensile load of 180 kN. Completely design the joint, if the permissible stresses are 80 MPa in tension; 65 MPa in shear and 160 MPa in crushing. Draw a neat sketch of the joint. (MAY/JUNE 2012)

13. Design a knuckle joint to connect two mild steel bars under a tensile load of 25 kN. The allowable stresses are 65 MPa in tension, 50 MPa in shear and 83 MPa in crushing.

(MAY/JUNE 2012)

14. What is an eccentric loaded welded joint? Describe procedure for designing such a joint. (MAY/JUNE 2013)