16ME218
DESIGN OF MACHINE ELEMENTS

Semester - 6
B.E Mechanical Engineering

Academic Year : 2019-20

16ME218 DESIGN OF MACHINE ELEMENTS
Course Outcomes

On successful completion of the course, the student will be able to

CO1	Describe design process, perform design calculations and analyse the theories of failure.
CO2	Design hollow and solid shafts under variable loading used in automobiles industrial applications.
CO3	**Design permanent joints, screws and fasteners used in industrial applications.**
CO4	Design helical coil, leaf, disc and torsion springs under various loads used in automobiles and industrial applications.
CO5	Design and select suitable sliding and rolling contact bearings for static and dynamic load capacities used in automobiles and agricultural equipment
CO6	Design connecting rods and flywheels under various loads used in automobiles and industrial applications.

SYLLABUS

STEADY STRESSES AND VARIABLE STRESSES IN MACHINE MEMBERS 24

Introduction to the design process, Factors influencing machine design, Selection of materials based on mechanical properties, Impact and shock loading and calculation of Principal stresses for various load combinations, Eccentric loading, Factor of safety, Fatigue Strength, S-N diagram, Theories of failure, Stress concentration. Design of solid and hollow shafts based on strength and rigidity. Design for static, dynamic and variable loading, Soderberg, Goodman and Gerber relations

SYLLABUS

DESIGN OF WELDED JOINTS AND SCREWED FASTENERS 12
Types of welded joints, strength of a welded joint, Design of parallel and transverse fillet Welded Joints. Eccentric loading- Form of screw threads, Nomenclature, Bolts of uniform strength, Design of screwed fasteners

DESIGN OF SPRINGS 12
Design of Helical, Leaf, Disc and Torsional springs subject to constant loads and varying loads, Design of Concentric torsion springs

DESIGN OF BEARINGS, CONNECTING ROD AND FLYWHEEL 12
Design of bearings: Mckee's equation, Sommerfield Number, Rolling and Sliding contact bearings, Design of Connecting rod. Design of Flywheel

WELDED JOINTS

- A welded joint is a permanent joint which is obtained by the fusion of the edges of the two parts to be joined together, with or without the application of pressure and a filler material.
- The heat required for the fusion of the material may be obtained by burning of gas (in case of gas welding) or by an electric arc (in case of electric arc welding).
- Welding is extensively used in fabrication as an alternative method for casting or forging and as a replacement for bolted and riveted joints.
- It is also used as a repair medium e.g. to reunite metal at a crack, to build up a small part that has broken off such as gear tooth or to repair a worn surface such as a bearing surface.

WELDED JOINTS AND THEIR ADVANTAGES

Welding is a very commonly used permanent joining process.

A welded joint has following advantages:

(i) Compared to other type of joints, the welded joint has higher efficiency. An efficiency > 95 % is easily possible.

(ii) Since the added material is minimum, the joint has lighter weight.

(iii) Welded joints have smooth appearances.

(iv) Due to flexibility in the welding procedure, alteration and addition are possible.

(v) It is less expensive.

(vi) Forming a joint in difficult locations is possible through welding.

WELDED JOINTS AND THEIR DISADVANTAGES

1. Since there is an uneven heating and cooling during fabrication, therefore the members may get distorted or additional stresses may develop.

2. It requires a highly skilled labour and supervision.

3. Since no provision is kept for expansion and contraction in the frame, therefore there is a possibility of cracks developing in it.

4. The inspection of welding work is more difficult than riveting work.

WELDED JOINTS AND THEIR APPLICATIONS

- Pressure vessels, steel structures.
- Flanges welded to shafts and axles.
- Crank shafts
- Heavy hydraulic turbine shafts
- Large gears, pulleys, flywheels
- Gear housing
- Machine frames and bases
- Housing and mill-stands.

BASIC TYPES OF WELDED PROCESSES

Welding Process

Gas Welding

1. Oxy-acetylene
2. Air,-Acetylene
3. Oxy-Hydrogen

Resistance Welding

1. Butt
2. Spot
3. Seam
4. Projection

Arc Welding

1. Carbon arc
2. Metal arc
3. Plasma arc
4. Gas Metal Arc (MIG)
5. Gas Tungsten Arc (TIG)

Newer Welding

1. Electron beam
2. Laser

Solid State Welding

1. Friction
2. Ultrasonic
3. Diffusion
4. Explosive

STRENGTH OF WELDED JOINTS

Adequate care must be taken to enhance strength of the welded joint. It is seen that strength of a welded joint gets affected mainly by the following factors.

(i) ***Crack initiation***: it is possible that cracks form while cooling a melted metal.

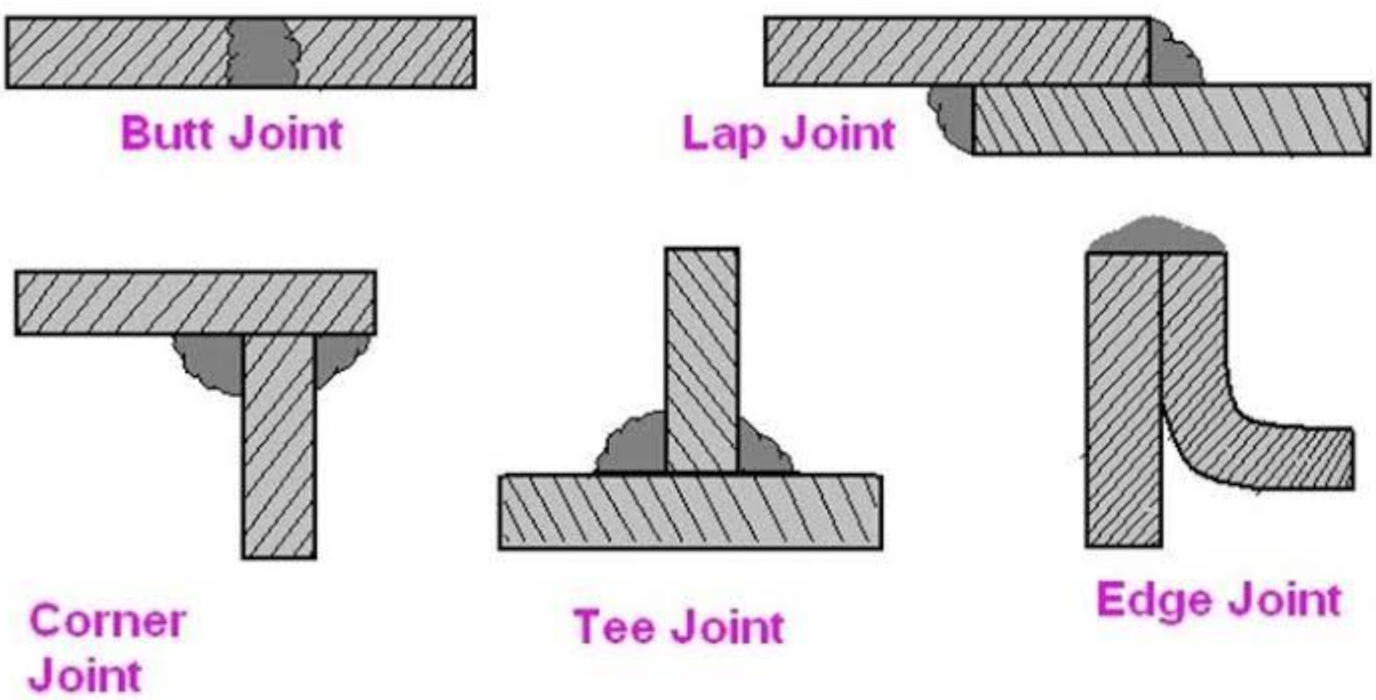
(ii) ***Residual stresses***: due to inhomogeneous heating of the base metals, residual stresses may exist upon cooling.

(iii) ***Metallurgical transformation***: in heat affected zone (HAZ) metallurgical properties may change leading to weakening of the joint.

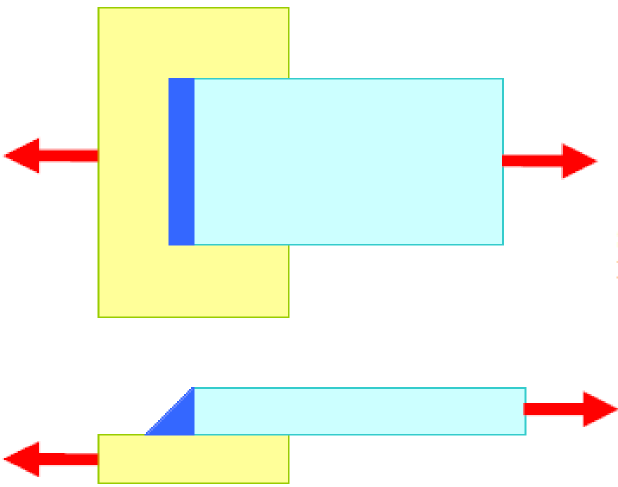
(iv) ***Defects***: of various kinds like incomplete penetration, porosity, slag inclusion which affect the strength of a welded joint.

(v) ***Stress concentration***: abrupt change in the geometry after welding may introduce stress concentration in the structure.

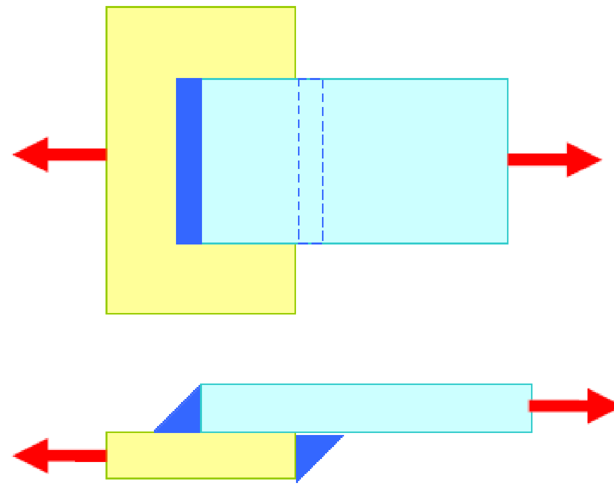
BASIC TYPES OF WELDED JOINTS



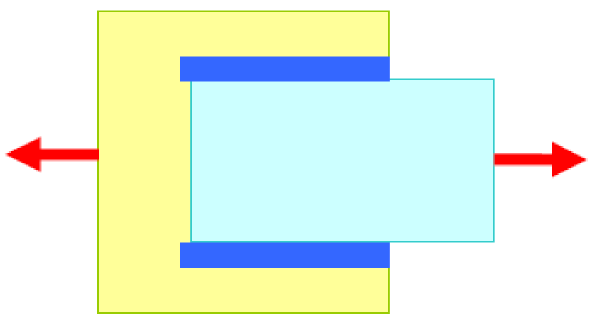
TYPES OF LAP JOINTS



Single transverse
lap joint

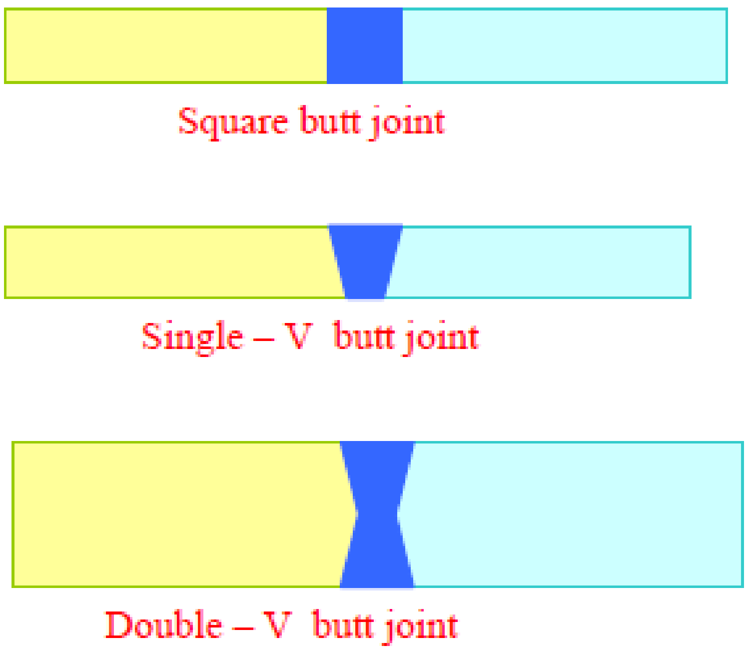


Double transverse
lap joint



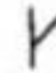
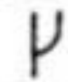
Parallel lap joint

TYPES OF BUTT JOINTS

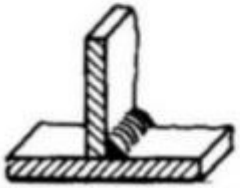
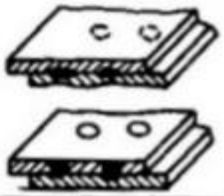
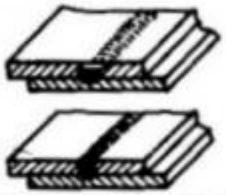


- Square butt joint
- Single V-butt joint, double V-butt joint
- Single U-butt joint, double U-butt joint
- Single J-butt joint, double J-butt joint
- Single bevel-butt joint, double bevel butt joint

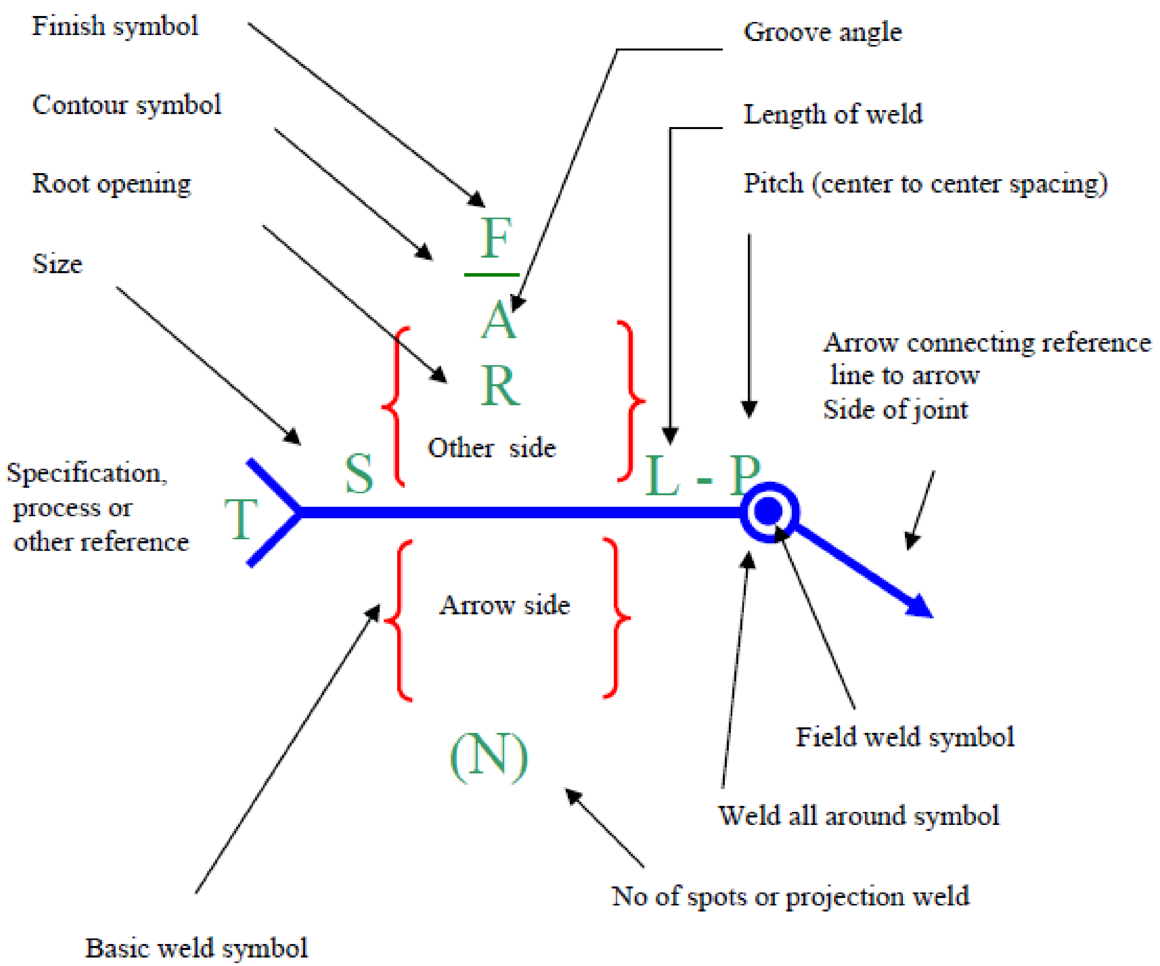
BASIC WELD SYMBOLS

No.	*Designation*	*Illustration*	*Symbol*
1.	Butt weld between plates with raised edges (the raised edges being melted down completely)		
2.	Square butt weld		
3.	Single-V butt weld		
4.	Single-bevel butt weld		
5.	Single-V butt weld with broad root face		
6.	Single-bevel butt weld with broad root face		
7.	Single-U butt weld (parallel or sloping sides)		
8.	Single-U butt weld		

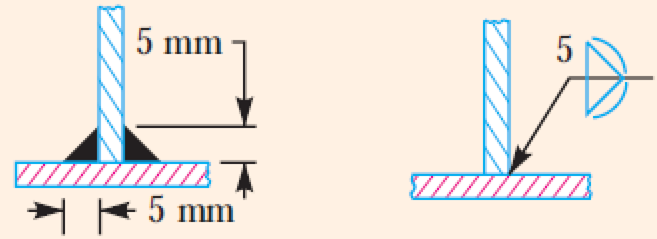
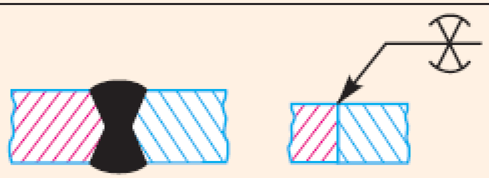
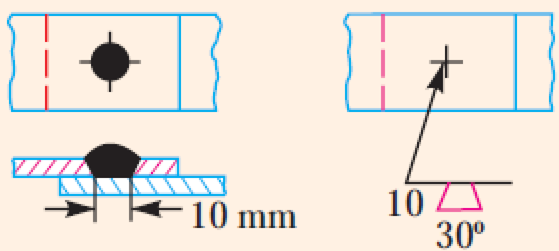
BASIC WELD SYMBOLS

9.	Backing run; back or backing weld		
10.	Fillet weld		
11.	Plug weld; plug or slot weld		
12.	Spot weld		
13.	Seam weld		

WELDING SYMBOL

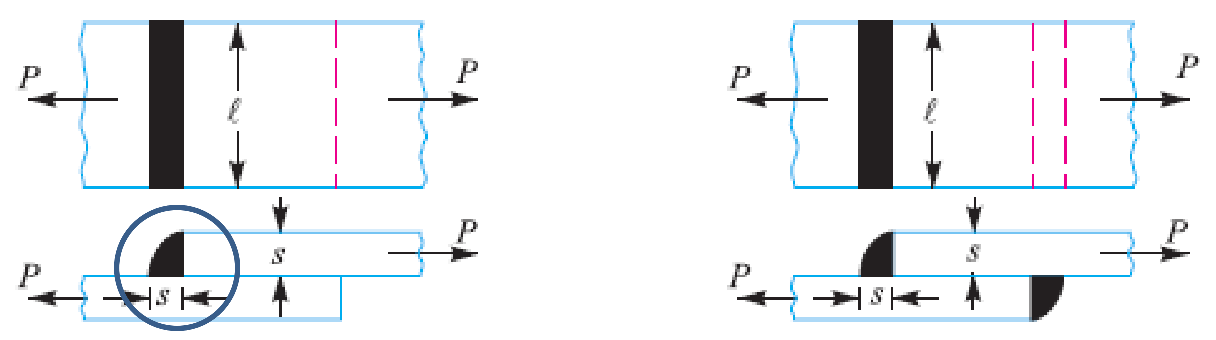


WELDING SYMBOL

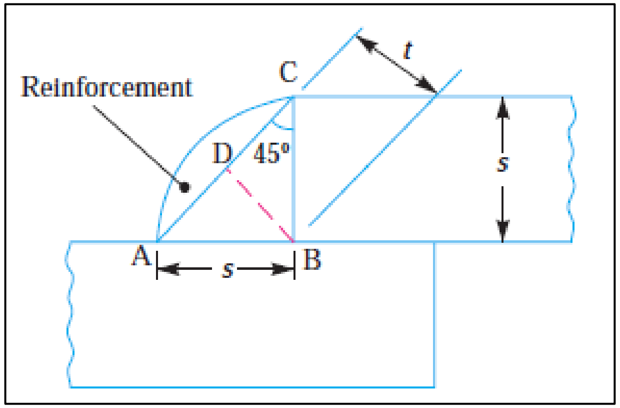
S. No.	*Desired weld*	*Representation on drawing*
1.	Fillet-weld each side of Tee- convex contour	
2.	Single V-butt weld -machining finish	
3.	Double V- butt weld	
4.	Plug weld - 30° Groove-angle-flush contour	

Strength of Transverse Fillet Welded Joints

- Main Failure Mechanism is Tensile Failure



(a) Single transverse fillet weld. (b) Double transverse fillet weld.



Tensile Strength of the joint for the single fillet weld is

P = Throat area × Allowable tensile stress

$= t \times l \times \sigma_t$

$= 0.707\ s \times l \times \sigma_t$

For double fillet weld

$P = 2 \times 0.707\ s \times l \times \sigma_t = 1.414\ s \times l \times \sigma_t$

Where,

t = Throat thickness (*BD*),

= s x sin 45 = 0.707 s

s = Leg or size of weld,

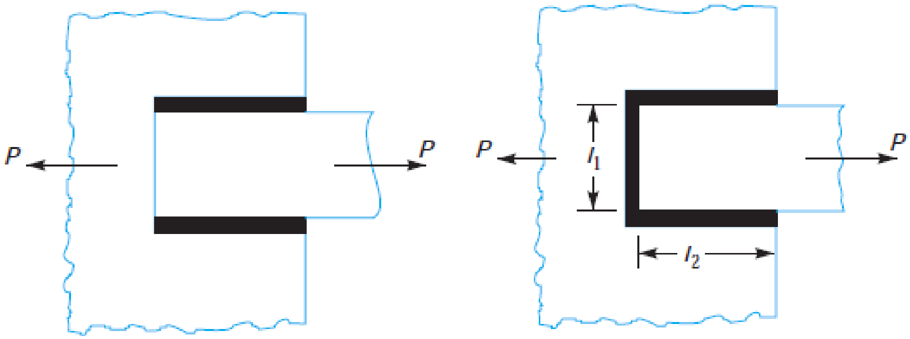
= Thickness of plate, and

l = Length of weld,

σ_t = Allowable Tensile stress

Strength of Parallel Fillet Welded Joints

- The parallel fillet welded joints are designed for shear strength



(*a*) Double parallel fillet weld.

(*b*) Combination of transverse and parallel fillet weld.

Combination of transverse and parallel fillet weld

$$P = 0.707s \times l_1 \times \sigma_t + 1.414\, s \times l_2 \times \tau$$

Shear Strength of the joint for the single parallel fillet weld is

P = Throat area × Allowable tensile stress

$$= t \times l \times \tau$$

$$= 0.707\, s \times l \times \tau$$

For double parallel fillet weld

$$P = 2 \times 0.707\, s \times l \times \tau = 1.414\, s \times l \times \tau$$

Where,

t = Throat thickness,

= s x sin 45 = 0.707 s

s = Leg or size of weld,

= Thickness of plate, and

l = Length of weld,

τ = Allowable Shear stress

NUMERICAL PROBLEMS

1. A plate 100 mm wide and 10 mm thick is to be welded to another plate by means of double parallel fillets. The plates are subjected to a static load of 80 kN. Find the length of weld if the permissible shear stress in the weld does not exceed 55 MPa.

Given:

Width = 100 mm ; Thickness = 10 mm ; P = 80 kN = 80 × 103 N; τ = 55 MPa = 55 N/mm^2

To Find:

Length of the weld, l

Solution:

For double parallel fillet weld

$$P = 2 \times 0.707\, s \times l \times \tau = 1.414\, s \times l \times \tau$$

Size of the weld = plate thickness = 10 mm

$$80 \times 10^3 = 1.414 \times 10 \times l \times 55$$

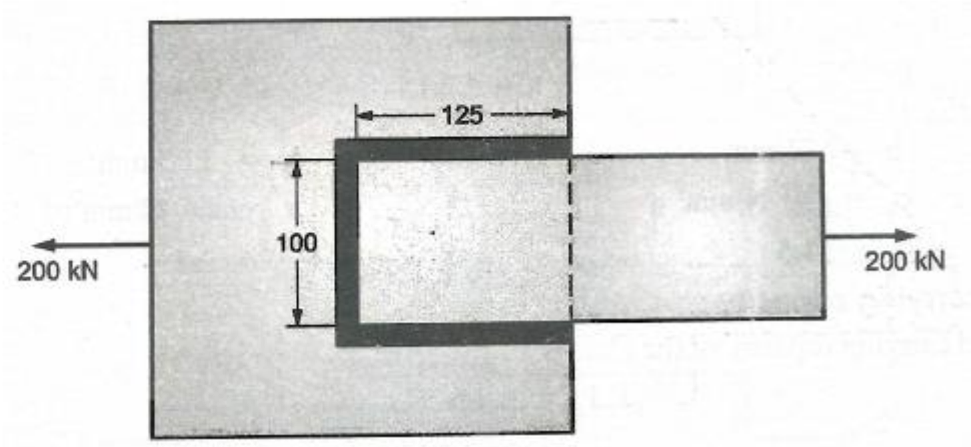
$$l = 103 \text{ mm}$$

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

l = 103 + 12.5 = 115.5 mm Ans.

NUMERICAL PROBLEMS

2. Two plates joined by fillet welds, as shown, are subjected to a tensile load of 200 kN. If the allowable shear stress for the weld material is 85 MPa, calculate the size of the weld.



Solution :

Given : $l_1 = 125$ mm ; $l_2 = 100$ mm;

$l_3 = 125$ mm ; $P = 200 \times 10^3$ N;

$\tau = 85$ N/mm^2.

Irrespective of type and direction of load, the stress induced in the fillet weld is considered to be shear stress.

Hence, shear stress induced in the fillet weld is,

$$\tau = \frac{P}{(l_1 + l_2 + l_3)\,t}$$

$$85 = \frac{200 \times 10^3}{(125 + 100 + 125)\,t}$$

$$\therefore \; t = 6.722 \text{ mm}$$

The weld size is,

$$h = \sqrt{2}\,t = \sqrt{2} \times 6.722$$

$$= 9.50 \text{ mm or } 10 \text{ mm}$$

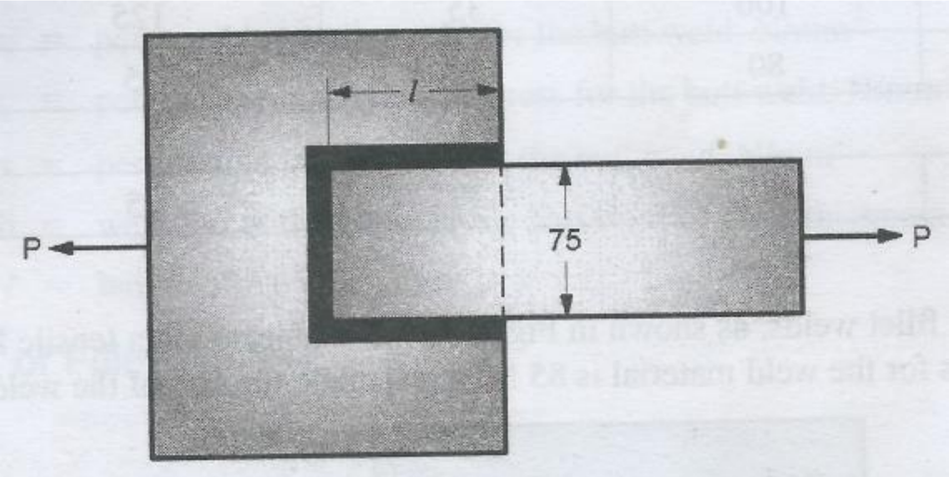
$$\mathbf{h = 10 \text{ mm}}$$ ...**Ans.**

NUMERICAL PROBLEMS

3. A plate, 75 mm wide and 12.5 mm thick, is joined to another plate by a transverse and parallel fillet welds, as shown. The permissible tensile stress for the plate is 70 MPa, while the permissible shear stress for the weld is 56 MPa. The stress concentration factor for the transverse weld is 1.5 and for parallel weld is 2.7, find the length of each parallel fillet weld if

(i) The joint is subjected to static loading and

(ii) The joint is subjected to fatigue loading.



NUMERICAL PROBLEMS

Given : $b = 75$ mm ; $h = 12.5$ mm;

$\sigma_t = 70\ N/mm^2$; $\tau = 56\ N/mm^2$;

$K_{ft} = 1.5$; $K_{fp} = 2.7$

- **Load carrying capacity :**

The load carrying capacity of the plate is,

$$P = \sigma_t\, bh$$

$$= 70 \times 75 \times 12.5$$

$$\text{or} \quad P = 65625\ N$$

- **Joint subjected to static loading :**

The load carried by single transverse weld is,

$$P_1 = \tau\, bt = \frac{\tau\, bh}{\sqrt{2}} = \frac{56 \times 75 \times 12.5}{\sqrt{2}} = 37123.1\ N$$

The load carried by double parallel welds is,

$$P_2 = \tau\,(l + l)\,t = \frac{2\,\tau\, l\, h}{\sqrt{2}} = \frac{2 \times 56 \times l \times 12.5}{\sqrt{2}} = 989.95\, l,\ N$$

The total load carried by welded joint is,

$$P = P_1 + P_2$$

$$65625 = 37123.1 + 989.95\, l$$

$$989.95\, l = 28501.9$$

$$\mathbf{l = 28.8\ mm} \qquad \text{...Ans.}$$

NUMERICAL PROBLEMS

- **Joint subjected to fatigue loading :**

The load carried by single transverse weld is,

$$P_1 = \frac{\tau}{K_{ft}}\, bt = \frac{\tau\, bh}{K_{ft}\sqrt{2}} = \frac{56 \times 75 \times 12.5}{1.5 \times \sqrt{2}} = 24748.74 \text{ N}$$

The load carried by double parallel welds is,

$$P_2 = \frac{\tau}{K_{fp}}\, (l + l)\, t = \frac{2\tau\, l\, h}{K_{fp}\sqrt{2}} = \frac{2 \times 56 \times l \times 12.5}{2.7 \times \sqrt{2}} = 366.65\, l \text{ N}$$

The total load carried by welded joint is,

$$P = P_1 + P_2$$

$$65625 = 24748.74 + 366.65\, l$$

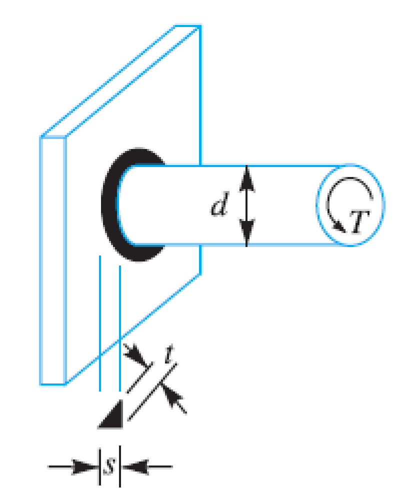
$$366.65\, l = 40876.26$$

$$l = \mathbf{111.5 \text{ mm}}$$

DESIGN OF WELDED JOINTS

Special Cases of Fillet Welded Joints

1. *Circular fillet weld subjected to torsion.*



Let d = Diameter of rod,

r = Radius of rod,

T = Torque acting on the rod,

s = Size (or leg) of weld,

t = Throat thickness,

*J = Polar moment of inertia of the weld section $= \dfrac{\pi t d^3}{4}$

We know that shear stress for the material,

$$\tau = \frac{T.r}{J} = \frac{T \times d/2}{J}$$

$$= \frac{T \times d/2}{\pi t d^3/4} = \frac{2T}{\pi t d^2} \qquad \ldots\left(\because \frac{T}{J} = \frac{\tau}{r}\right)$$

This shear stress occurs in a horizontal plane along a leg of the fillet weld. The maximum shear occurs on the throat of weld which is inclined at 45° to the horizontal plane.

$\therefore$ Length of throat, $t = s \sin 45° = 0.707\, s$

and maximum shear stress,

$$\tau_{max} = \frac{2T}{\pi \times 0.707\, s \times d^2} = \frac{2.83\, T}{\pi\, s\, d^2}$$

2. *Circular fillet weld subjected to bending moment.*

Let d = Diameter of rod,

M = Bending moment acting on the rod,

s = Size (or leg) of weld,

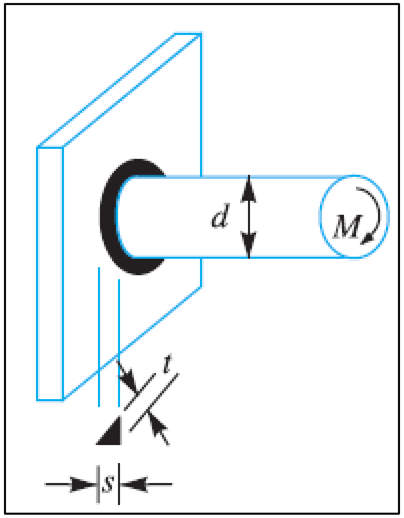
t = Throat thickness,

**Z = Section modulus of the weld section

$$= \frac{\pi t d^2}{4}$$

We know that the bending stress,

$$\sigma_b = \frac{M}{Z} = \frac{M}{\pi t d^2 / 4} = \frac{4M}{\pi t d^2}$$



This bending stress occurs in a horizontal plane along a leg of the fillet weld. The maximum bending stress occurs on the throat of the weld which is inclined at 45° to the horizontal plane.

$\therefore$ Length of throat, $t = s \sin 45° = 0.707\, s$

and maximum bending stress,

$$\sigma_{b(max)} = \frac{4\,M}{\pi \times 0.707\, s \times d^2} = \frac{5.66\,M}{\pi\, s\, d^2}$$

3. *Long fillet weld subjected to torsion.*

Let T = Torque acting on the vertical plate,

l = Length of weld,

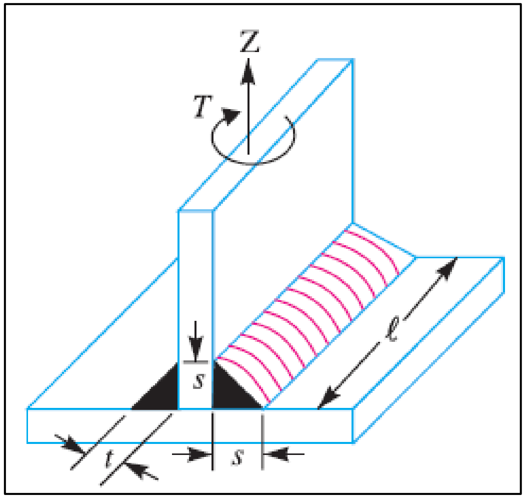
s = Size (or leg) of weld,

t = Throat thickness, and

J = Polar moment of inertia of the weld section

$$= 2 \times \frac{t \times l^3}{12} = \frac{t \times l^3}{6} \quad \ldots$$

($\because$ of both sides weld)



It may be noted that the effect of the applied torque is to rotate the vertical plate about the Z-axis through its mid point. This rotation is resisted by shearing stresses developed between two fillet welds and the horizontal plate. It is assumed that these horizontal shearing stresses vary from zero at the Z-axis and maximum at the ends of the plate. This variation of shearing stress is analogous to the variation of normal stress over the depth (l) of a beam subjected to pure bending.

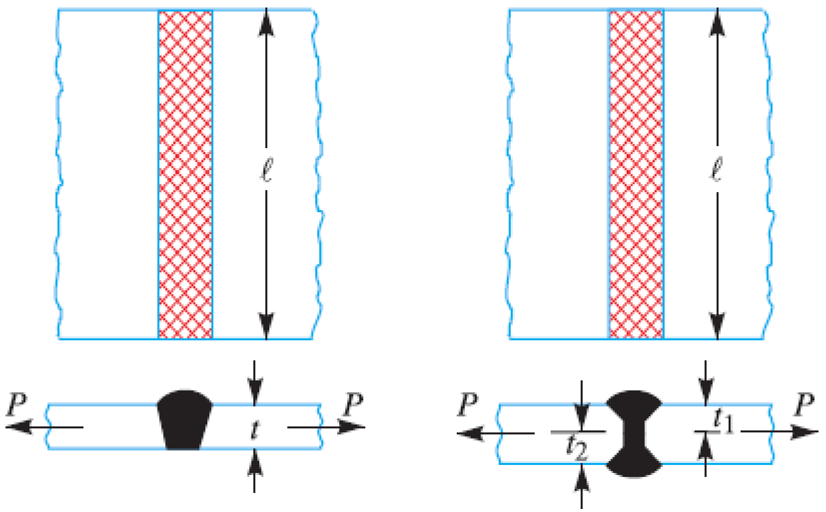
$\therefore$ Shear stress, $$\tau = \frac{T \times l/2}{t \times l^3/6} = \frac{3\,T}{t \times l^2}$$

The maximum shear stress occurs at the throat and is given by

$$\tau_{max} = \frac{3T}{0.707\,s \times l^2} = \frac{4.242\,T}{s \times l^2}$$

Strength of Butt Joints

-The butt joints are designed for tension or compression



(a) Single V-butt joint. (b) Double V-butt joint.

In case of butt joint, the length of leg or size of weld is equal to the throat thickness which is equal to thickness of plates.

Tensile strength of the butt joint (single-*V* or square butt joint),

$$P = t \times l \times \sigma_t$$

Tensile strength for double-V butt joint

$$P = (t_1 + t_2)\, l \times \sigma_t$$

Stresses for Welded Joints

The stresses in welded joints are difficult to determine because of the variable and unpredictable parameters like homogenuity of the weld metal, thermal stresses in the welds, changes of physical properties due to high rate of cooling etc. The stresses are obtained, on the following assumptions:

1. The load is distributed uniformly along the entire length of the weld, and
2. The stress is spread uniformly over its effective section.

Stresses for Welded Joints

The following table shows the stresses for welded joints for joining ferrous metals with mild steel electrode under steady and fatigue or reversed load.

Type of weld	*Bare electrode*		*Coated electrode*	
	Steady load (MPa)	*Fatigue load (MPa)*	*Steady load (MPa)*	*Fatigue load (MPa)*
1. Fillet welds (All types)	80	21	98	35
2. Butt welds				
Tension	90	35	110	55
Compression	100	35	125	55
Shear	55	21	70	35

Stress Concentration Factor for Welded Joints

The reinforcement provided to the weld produces stress concentration at the junction of the weld and the parent metal. When the parts are subjected to fatigue loading, the stress concentration factor as given in the following table should be taken into account.

Type of joint	*Stress concentration factor*
1. Reinforced butt welds	1.2
2. Toe of transverse fillet welds	1.5
3. End of parallel fillet weld	2.7
4. T-butt joint with sharp corner	2.0

SCREWED FASTENERS

A screw thread is formed by cutting a continuous helical groove on a cylindrical surface.

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. Similarly, triple and quadruple (*i.e.* multiple-start) threads may be formed.

The helical grooves may be cut either ***right hand*** or ***left hand.***

A threaded component consists of two components, a **bolt and nut**

Purpose

To connect the components together, which should be readily disassembled

To transmit the power or energy

As a means of adjustments for obtaining accurate movements in measuring instruments

Advantages and Disadvantages of Screwed Joints

Advantages

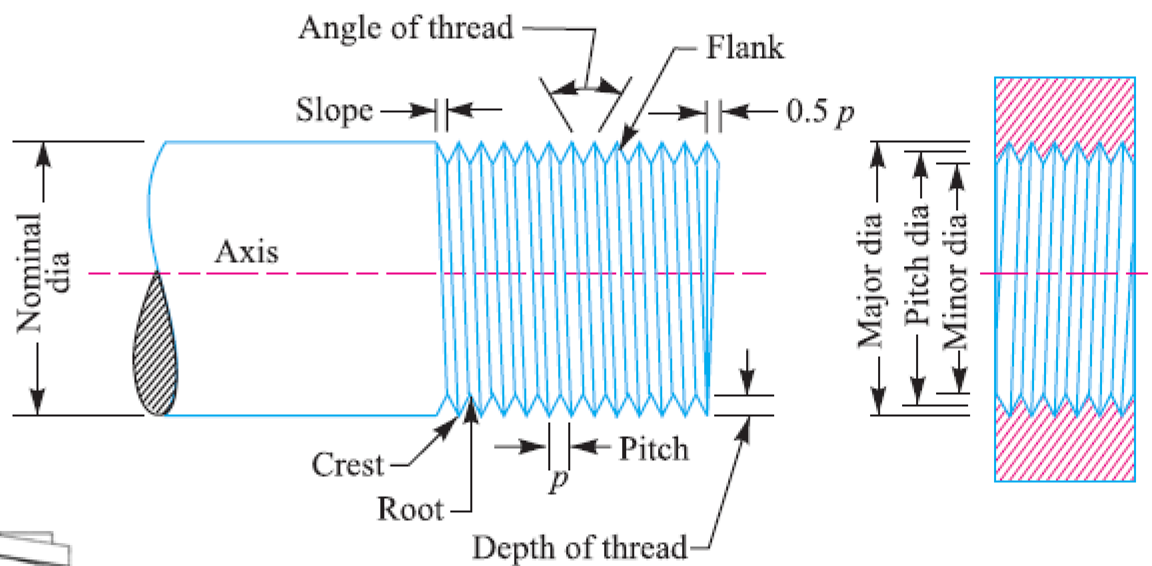
1. Screwed joints are highly reliable in operation.
2. Screwed joints are convenient to assemble and disassemble.
3. A wide range of screwed joints may be adopted to various operating conditions.
4. Screws are relatively cheap to produce due to standardisation and highly efficient manufacturing processes.

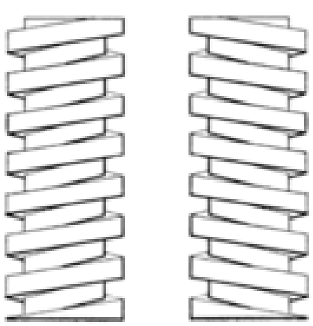
Disadvantages

1. The main disadvantage of the screwed joints is the stress concentration in the threaded portions which are vulnerable points under variable load conditions.
2. The components become weak because of the holes
3. There is possibility of loosening of joint due to excessive vibrations

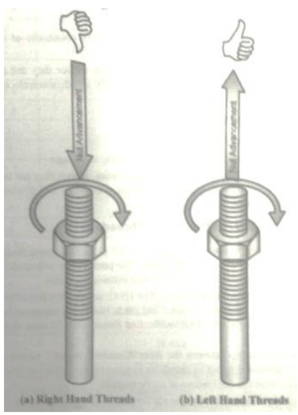
Note : The strength of the screwed joints is not comparable with that of riveted or welded joints.

Important Terms Used in Screw Threads





Right-hand Left-hand



Forms of Screw Threads

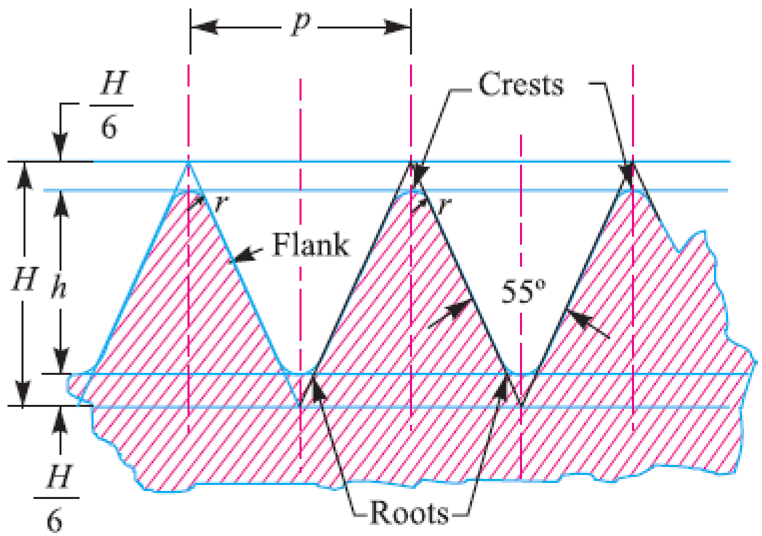
Various forms of threads

i. British Standard Whitworth (B.S.W) threads
ii. British Standard Fine (B.S.F) threads
iii. British Standard Pipe (B.S.P) threads
iv. British Association (B.A) threads
v. American National Standard (A.N.S) threads
vi. Unified National Threads
vii. I.S.O Metric threads

Forms of Threads used in power screws:

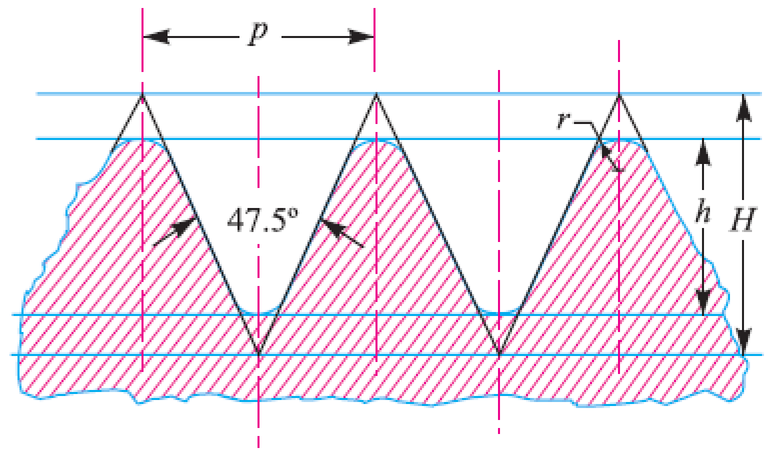
i. Square threads
ii. ACME threads
iii. Trapezoidal Threads
iv. Buttress Threads

Forms of Screw Threads



$H = 0.96\,p$; $h = 0.64\,p$; $r = 0.1373\,p$

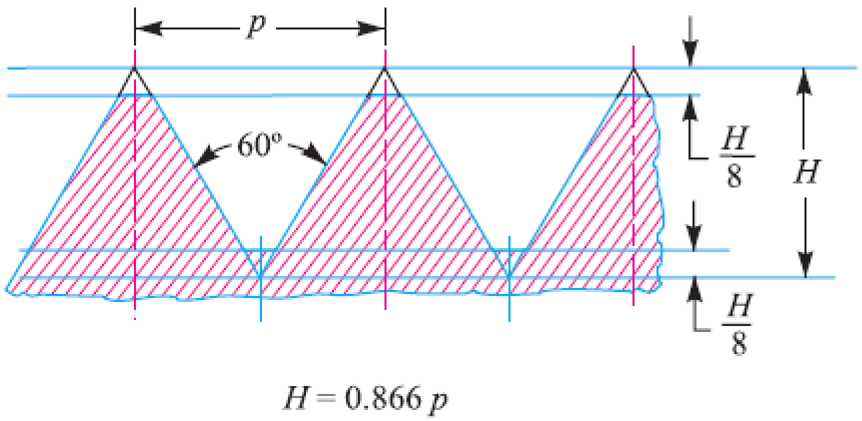
British standard whitworth (B.S.W) thread.



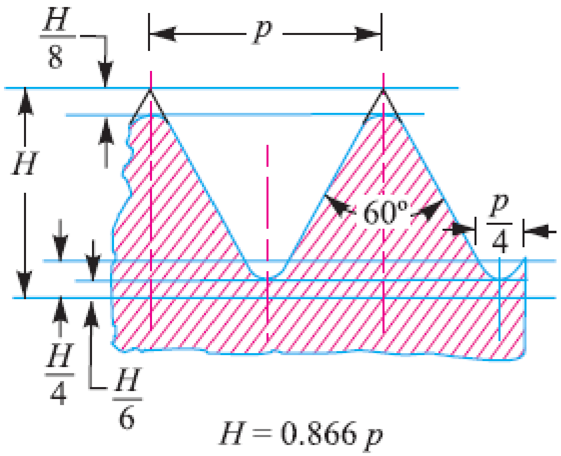
$H = 1.13634\,p$; $h = 0.6\,p$; $r = 0.18083\,p$

British association (B.A.) thread.

Forms of Screw Threads

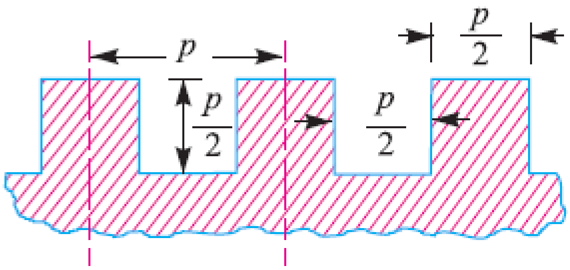


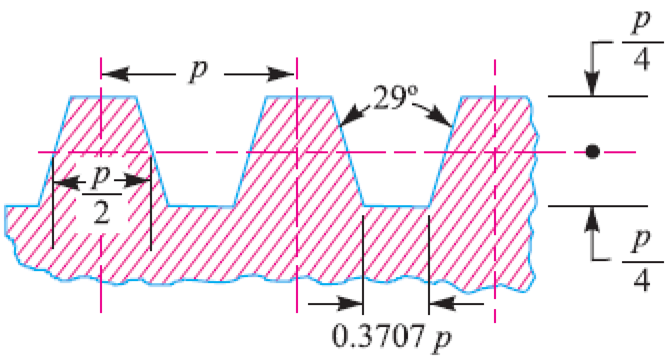
American National Standard Thread



Unified Standard Thread

Forms of Screw Threads

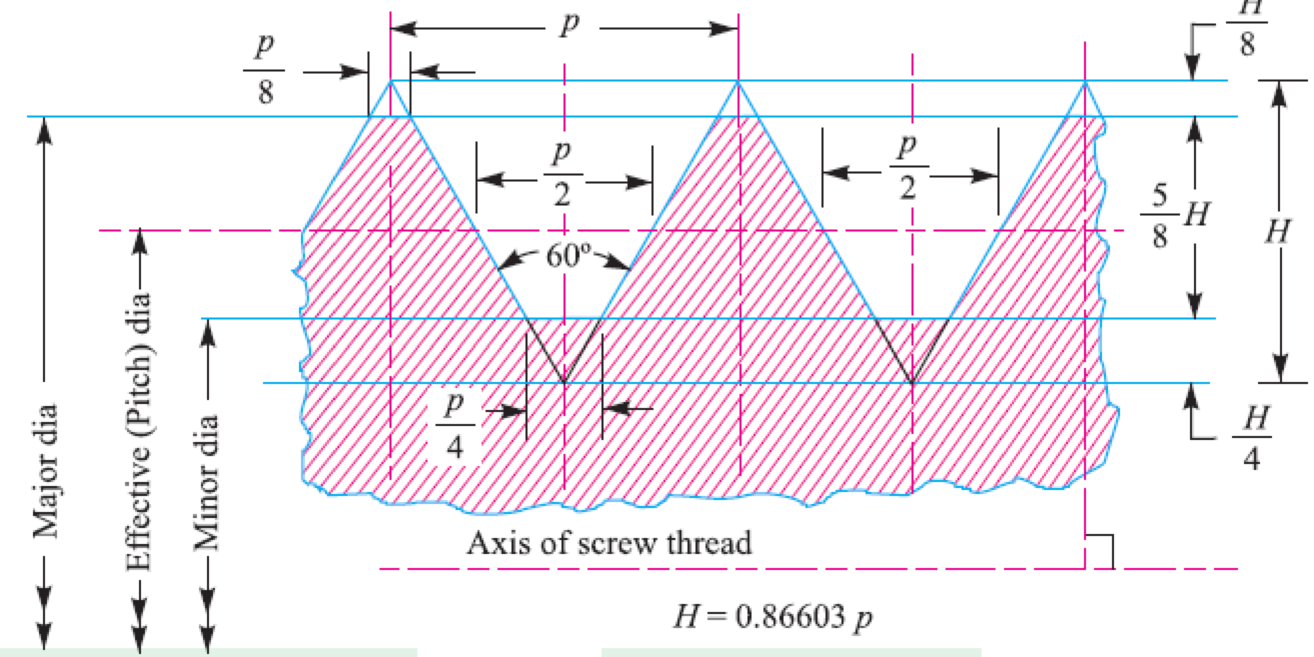




Square Thread

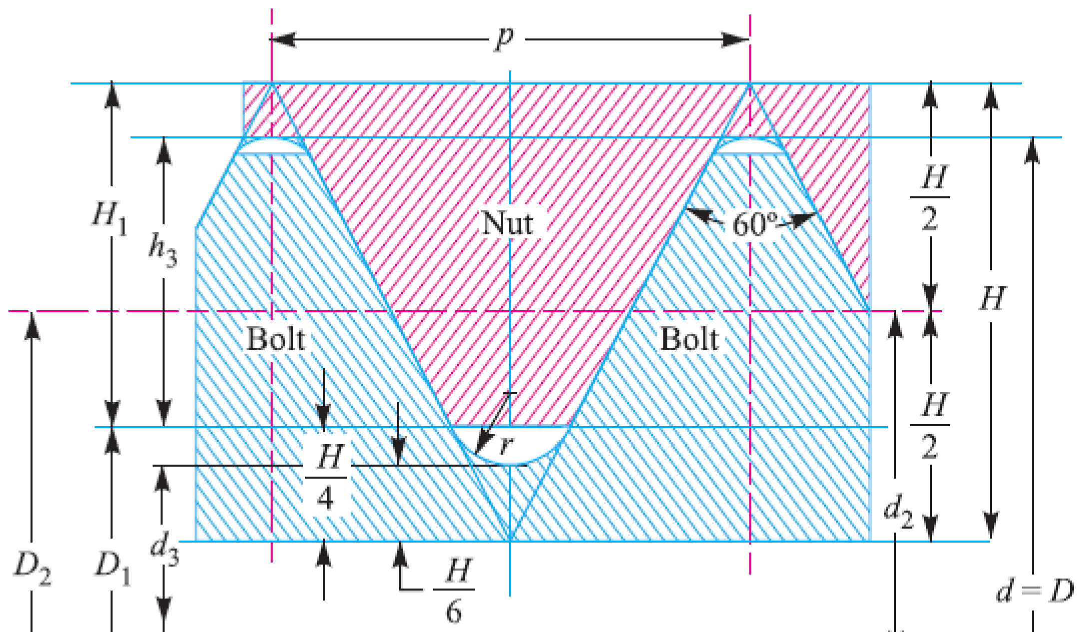
ACME Thread

Basic Profile of the Thread



Basic profile of the thread.

Design Profile of the Nut and Bolt



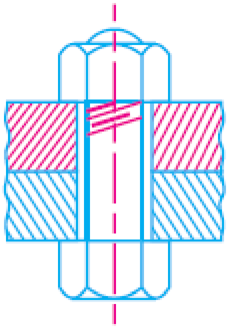
$H = 0.86603\,p;\qquad D_2 = d_2 = d - 0.6495\,p;$

$D_1 = d - 1.0825\,p;\qquad d_3 = d - 1.2268\,p;$

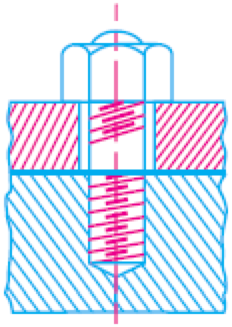
$H_1 = \frac{5}{8}H;\qquad h_3 = \frac{17}{24}H;\qquad r = \frac{H}{6}$

d = Diameter of nut; D = Diameter of bolt.

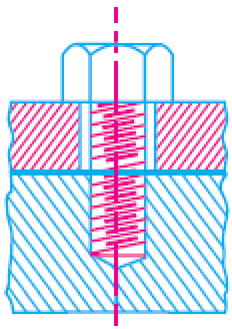
SCREWED FASTENERS



(*a*) Through bolt.

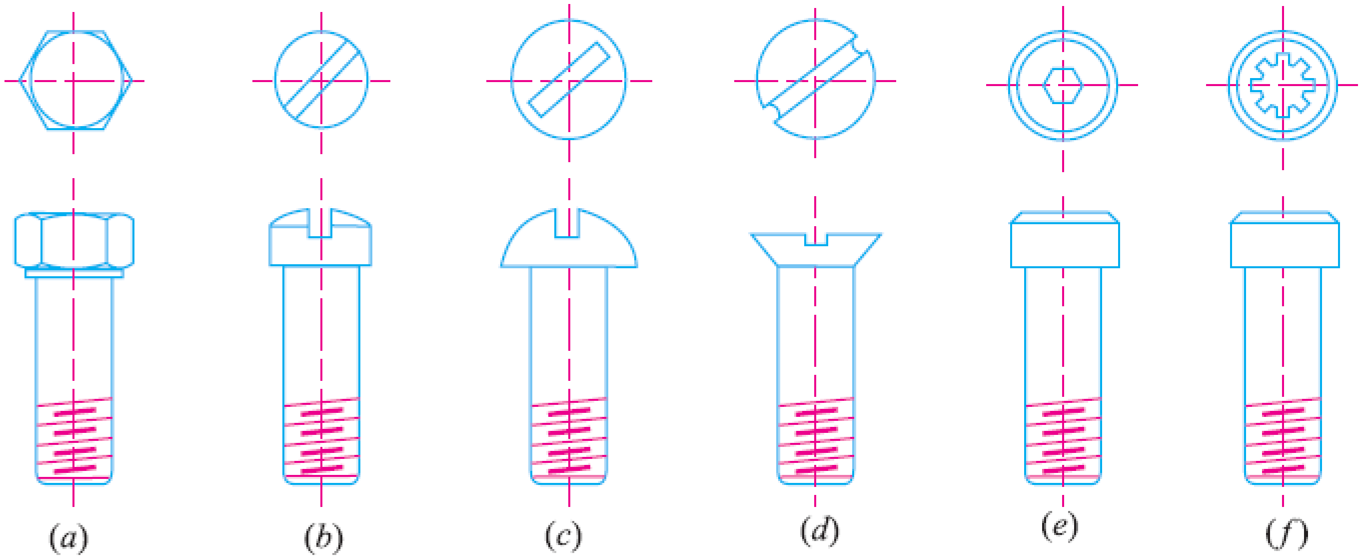


(*b*) Tap bolt.



(*c*) Stud.

SCREWED FASTENERS



(*a*) Hexagonal head; (*b*) Fillister head; (*c*) Round head; (*d*) Flat head;
(*e*) Hexagonal socket; (*f*) Fluted socket.

Stresses in Screwed Fastening due to Static Loading

The following stresses in screwed fastening due to static loading are important from the subject point of view :

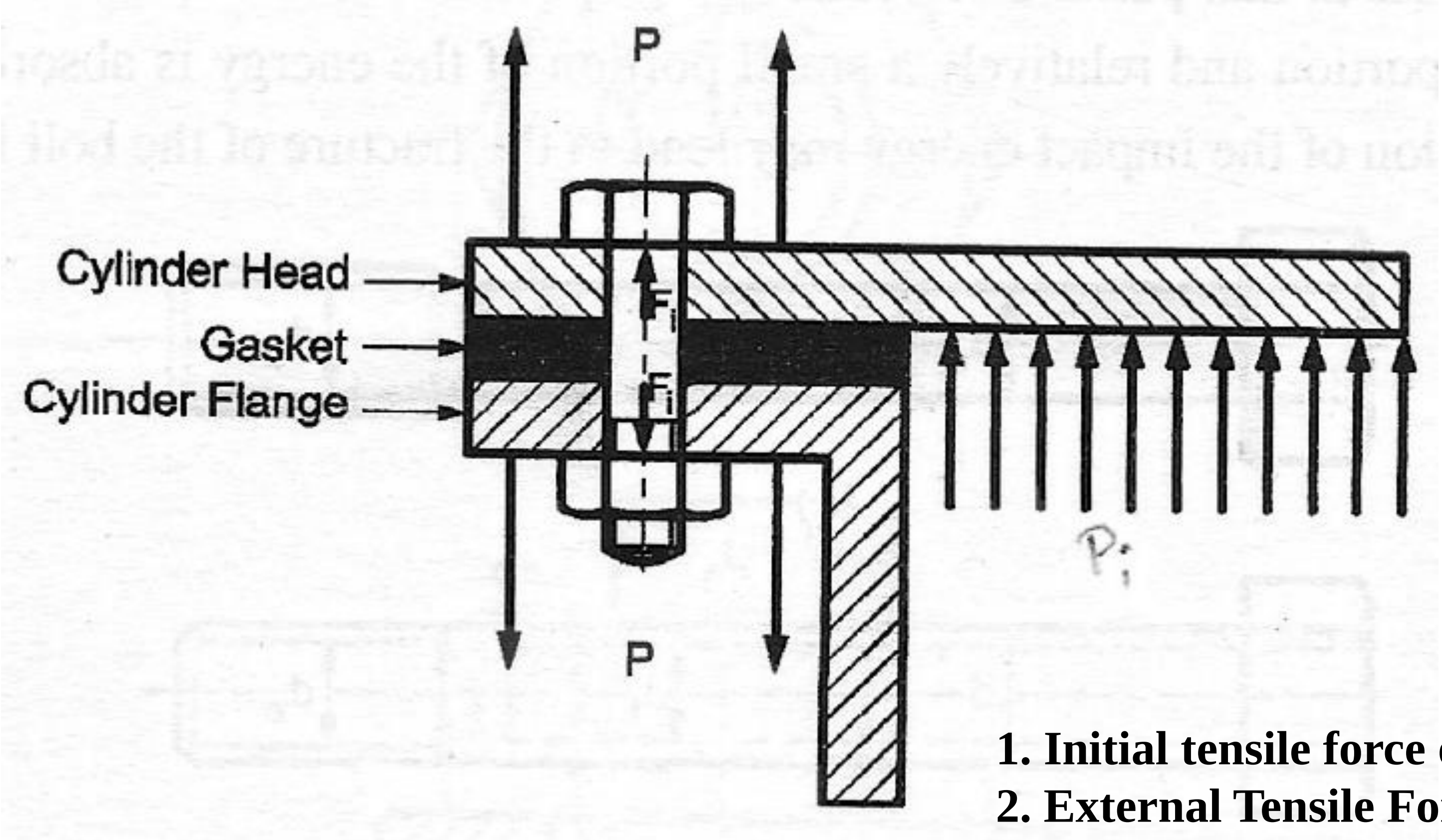
1. Internal stresses due to screwing up forces,

2. Stresses due to external forces, and

3. Stress due to combination of stresses at (1) and (2).

SCREWED FASTENERS

BOLTED JOINT UNDER TENSION



1. **Initial tensile force or Preload, F_i**
2. **External Tensile Force, P_b**

SCREWED FASTENERS

Initial Stresses due to Screwing up Forces

1. Tensile stress due to stretching of bolt.
2. Torsional shear stress caused by the frictional resistance of the threads during its tightening.
3. Shear stress across the threads
4. Compression or crushing stress on threads
5. Bending stress if the surfaces under the head or nut are not perfectly parallel to the bolt axis

Stresses due to External Forces

1. Tensile stress
2. Shear stress
3. Combined tension and shear stress

Resultant Tensile Force on Bolt:

The resultant tensile force on the bolt due to the combined effect of the initial tensile force (preload) and external tensile force, can be determined as follows :

Let, F_i = initial tensile force or preload on each bolt due to tightening, N

P = total external force on each bolted assembly, N

P_b = portion of P taken by each bolt, N

P_m = portion of P taken by connected members, under each bolt N

F_b = resultant force on each bolt, N

F_m = resultant force on connected members under each bolt, N

K_b = stiffness of the bolt under each bolt, N/mm

K_m = stiffness of the connected members, N/mm

- Due to the preload, the bolt is under tension whereas the connected members are under compression. Hence, the bolt is elongated and the connected members are compressed.
- When the external tensile force is applied to the preloaded assembly, there is a charge in the deformation of the bolt as well as the connected members.
- The bolt, which is initially in tension, gets elongated further whereas the connected members, which are in compression initially, are relieved of the compression partially.

SCREWED FASTENERS

Hence, the increase in the deformation of the bolt is,

$$\Delta \delta_b = \frac{P_b}{K_b} \quad \text{...(a)}$$

The decrease in the deformation of the connected member is,

$$\Delta \delta_m = \frac{P_m}{K_m} \quad \text{...(b)}$$

Assuming that, under the action of an external force, connected members are not separated.

Increase in deformation of bolt = Decrease in deformation of the connected members

i.e. $$\Delta \delta_b = \Delta \delta_m \quad \text{...(c)}$$

Substituting the values of Equations (a) and (b) in Equation (c),

$$\frac{P_b}{K_b} = \frac{P_m}{K_m} \quad \text{...(7.4)}$$

$$P_m = \left(\frac{K_m}{K_b}\right) P_b \quad \text{...(d)}$$

Now, the total external force on each bolted assembly is,

$$P = P_b + P_m \quad \text{...(e)}$$

Substituting Equation (d) in Equation (e),

$$P = P_b + \left(\frac{K_m}{K_b}\right) P_b$$

$$P = \left(\frac{K_b + K_m}{K_b}\right) P_b$$

$$\therefore \quad P_b = \left(\frac{K_b}{K_b + K_m}\right) P \quad \text{...(7.5)}$$

Substituting Equation (7.5) in Equation (d),

$$P_m = \frac{K_m}{K_b}\left(\frac{K_b}{K_b + K_m}\right) P$$

$$\therefore \quad P_m = \left(\frac{K_m}{K_b + K_m}\right) P \quad \text{...(7.6)}$$

Therefore the resultant force on each bolt is,

$$\mathbf{F_b = P_b + F_i} \quad \text{...(f)}$$

$$\mathbf{F_b = \left(\frac{K_b}{K_b + K_m}\right) P + F_i} \quad \text{...(7.7)}$$

and the resultant force on the connected members under each bolt is,

$$\mathbf{F_m = P_m - F_i} \quad \text{...(g)}$$

$$\mathbf{F_m = \left(\frac{K_m}{K_b + K_m}\right) P - F_i} \quad \text{...(7.8)}$$

Condition for avoiding Joint Separation:

In order to avoid the separation of the connected members, the resultant force on the connected members, 'F_m' must be compressive or negative. Hence, preload 'Fi' must be large enough to maintain the resultant force in connected members negative.

If the resultant force on connected members become zero or positive, then the two members will separate and the entire force will be carried by the bolt. In such case, as the basic assumption made in the analysis is violated, the preceding analysis becomes irrelevant.

Hence in order to avoid the joint separation,

SCREWED FASTENERS

$$F_m < 0$$

$$\text{or} \left(\frac{K_m}{K_b + K_m}\right) P - F_i < 0$$

$$\text{or} \left(\frac{K_m}{K_b + K_m}\right) P < F_i$$

This is the condition to avoid the joint separation,

By designating, $$C = \frac{K_b}{K_b + K_m}$$

$$F_b = CP + F_i$$

$$\text{and} \quad F_m = (1 - C) P - F_i$$

Where, C = gasket factor which depends upon the gasket and bolt materials

The approximate values of gasket factor 'C' used in preliminary calculations are given

Type of Joint	Gasket Factor 'C'
• Metal to metal joint	0.00 to 0.10
• Hard copper gasket	0.25 to 0.50
• Soft copper gasket	0.50 to 0.75
• Soft thick gasket	0.75 to 1.00

NUMERICAL PROBLEMS

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/mm^2. Calculate the number and size of studs required to fix the cylinder cover, assuming the permissible stress in the studs as 33 MPa.

NUMERICAL PROBLEMS

Solution. Given: $D = 350$ mm ; $p = 1.25$ N/mm^2 ; $\sigma_t = 33$ MPa $= 33$ N/mm^2

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,

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

Assume that the studs of nominal diameter 24 mm are used. From PSGDB 5.42(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,

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

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

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

NUMERICAL PROBLEMS

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)

$\therefore$ *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.

$\therefore$ 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.

$\therefore$ Size of the stud = M 24 **Ans.**

* The circumferential pitch of the studs can not be measured and marked on the cylinder cover. The centres of the holes are usually marked by angular distribution of the pitch circle into n number of equal parts. In the present case, the angular displacement of the stud hole centre will be 360°/12 = 30°.

NUMERICAL PROBLEMS

The cylinder head of a steam engine is subjected to a steam pressure of 5 bar and 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 250 mm. If the allowable tensile stress for the bolts is limited to 90 N/mm^2, determine the size of the bolts.

Solution :

Given : $P_i = 5\text{ bar} = 5 \times 10^5\text{ N/m}^2 = 0.5\text{ N/mm}^2$; $n = 12$;

$D = 250\text{ mm}$; $\sigma_t = 90\text{ N/mm}^2$.

1. External force on each bolt :

The total external force on bolted joints due to internal pressure is,

$$P_T = \frac{\pi D^2 P_1}{4} = \frac{\pi \times (250)^2 \times 0.5}{4}$$

or $$P_T = 24543.7\text{ N}$$

NUMERICAL PROBLEMS

The external force on each bolted joint is,

$$P = \frac{P_T}{n} = \frac{24543.7}{12} = 2045.3 \text{ N} \qquad \ldots(a)$$

2. Initial tension in each bolt :

The initial tension, due to tightening, in each bolt is,

$$F_i = 2840d \text{ N} \qquad \ldots(b)$$

3. Resultant force on each bolt :

For joint with soft copper gasket, the gasket factor is taken as,

$$C = 0.5$$

The resultant force on each bolt is,

$$F_b = CP + F_i = 0.5 \times 2045.3 + 2840d$$

$$\text{or} \quad F_b = 1022.65 + 2840d \qquad \ldots(c)$$

4. Bolt size :

$$\sigma_t = \frac{F_b}{A_c}$$

$$F_b = \sigma_t \cdot A_c$$

$$1022.65 + 2840d = 90 \times \frac{\pi d_c^2}{4}$$

$$1022.65 + \frac{2840\, d_c}{0.84} = \frac{90\, \pi\, d_c^2}{4} \qquad \left\{ \text{For course series, } d \approx \frac{d_c}{0.84} \right\}$$

$$1022.65 + 3380.95\, d_c = 70.68\, d_c^2$$

$$14.46 + 47.83\, d_c = d_c^2$$

$$\therefore \quad d_c^2 - 47.83\, d_c - 14.46 = 0$$

$$d_c = \frac{+47.83 \pm \sqrt{(47.83)^2 + 4 \times 1 \times 14.46}}{2}$$

$$= \frac{47.83 \pm 48.43}{2}$$

$$d_c = 48.13 \text{ or } -0.3$$

$$= 48.13 \text{ (taking positive value)}$$

$$d = \frac{d_c}{0.84} = \frac{48.13}{0.84} = 57.29 \text{ or } 60 \text{ mm}$$

$$\therefore \quad \mathbf{d = 60\ mm\ \ or\ M60}$$ **...Ans.**

NUMERICAL PROBLEMS

The cylinder head of a steam engine is subjected to a steam pressure of 0.7 N/mm^2. 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.

NUMERICAL PROBLEMS

Solution. Given: $p = 0.7$ N/mm^2 ; $n = 12$; $D = 300$ mm ; $\sigma_t = 100$ MPa $= 100$ N/mm^2

We know that the total force (or the external load) acting on the cylinder head *i.e.* on 12 bolts,

$$= \frac{\pi}{4}(D)^2\ p = \frac{\pi}{4}(300)^2\ 0.7 = 49\ 490 \text{ N}$$

$\therefore$ 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,

$$P_1 = 2840\ d \text{ N} \qquad \text{... (where } d \text{ is in mm)}$$

we find that for soft copper gasket with long through bolts, the minimum value of $K = 0.5$.

$\therefore$ Resultant axial load on the bolt,

$$P = P_1 + K \cdot P_2 = 2840\ d + 0.5 \times 4124 = (2840\ d + 2062) \text{ N}$$

We know that load on the bolt (P),

$$2840\ d + 2062 = \frac{\pi}{4}(d_c)^2\ \sigma_t = \frac{\pi}{4}(0.84d)^2\ 100 = 55.4\ d^2 \qquad \text{...(Taking } d_c = 0.84\ d\text{)}$$

$$\therefore \quad 55.4\ d^2 - 2840d - 2062 = 0$$

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.**

NUMERICAL PROBLEMS

STRESSES IN SCREW FASTENERS

The stresses induced in the screw fasteners due to external forces or due to resultant force (i.e. combination of preload and external force) are discussed below :

Stresses in Screw Body :

Following types of stresses are induced in a screw body :

1. Direct tensile stress (σ_t)
2. Direct shear stress (τ)
3. Maximum shear stress (τ_{max})

1. Direct tensile stress (σ_t) :

Direct tensile stress induced in a screw body due to tensile force 'F' is given by,

$$\sigma_t = \frac{F}{A_c} = \frac{F}{\left(\frac{\pi}{4} d_c^2\right)}$$

Where, σ_t = direct tensile stress in a screw body, N/mm^2

F = tensile force on screw fastener, N

A_c = core cross-sectional area of bolt, mm^2

d_c = core diameter of bolt, mm

2. Direct shear stress (τ) :

Occasionally the screw fasteners are also subjected to shear force in addition to direct tensile force (e.g. brackets). In such cases, the direct shear stress induced in a screw body is given by,

$$\tau = \frac{F_s}{A_c} = \frac{F_s}{\left(\frac{\pi}{4} d_c^2\right)}$$

Where, τ = direct shear stress in a screw body, N/mm^2

F_s = shear force on screw fastener, N

3. Maximum shear stress (τ_{max}) :

If a screw body is subjected to direct tensile stress 'σ_t' and direct shear stress 'τ', then the maximum shear stress induced in the screw body is calculated using maximum shear stress theory.

$\therefore$ The maximum shear stress in the screw body is given by,

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

NUMERICAL PROBLEMS

Stresses in Screw Threads :

The following types of stresses are induced in a screw threads :

1. Direct shear stress
   (a) direct shear stress in screw threads (τ_s)
   (b) direct shear stress in nut threads (τ_n)
2. Crushing stress.

1. Direct shear stress :

(a) Direct shear stress in screw threads (τ_s) :

The direct shear stress induced in a screw threads due to tensile force **'F'** is given by,

$$\tau_s = \frac{F}{\pi d_c t Z} = \frac{F}{\pi d_c \cdot h}$$

Where,
τ_s = direct shear stress induced in a screw threads, N/mm^2
t = thickness of the thread at the root, mm = p
p = pitch, mm
Z = number of threads in engagement
h = height of the nut, mm = $t \cdot Z$

(b) Direct shear stress in nut threads (τ_n) :

Similarly, the direct shear stress induced in a nut threads due to tensile force **'F'** is given by,

$$\tau_n = \frac{F}{\pi d t Z} = \frac{F}{\pi d h}$$

Where,
τ_n = direct shear stress induced in nut threads, N/mm^2
d = nominal diameter, mm

NUMERICAL PROBLEMS

2. Crushing stress :

The crushing stress induced in a threads is given by,

$$\sigma_c = \frac{F}{\pi/4\left(d^2 - d_c^2\right) Z}$$

Where, σ_c = Crushing stress induced in threads, N/mm^2

NUMERICAL PROBLEMS

The cylinder head of steam engine with 250 mm bore is fastened by eight stud bolts made of steel 30V8 (S_{yt}= 300 MPa). The maximum pressure inside the cylinder is 1 MPa. If the overload is 20 % , determine :

1. The bolt size
2. The Tightening torque

NUMERICAL PROBLEMS

Given :

$D_i = 250$ mm ; $n = 8$,

$S_{yt} = 300$ N/mm^2 ; $p_i = 1$ N/mm^2

$K_a = 1.2.$

- **Total force on cylinder head :**

The total force on cylinder head due to steam pressure is,

$$P_T = K_a \frac{\pi D_i^2}{4} \cdot p_i$$

$$= \frac{1.2 \times \pi \times (250)^2 \times 1}{4}$$

$$\text{or } P_T = 58.905 \times 10^3 \text{ N}$$

- **Total force on each stud bolt :**

It is assumed that soft thick gasket is used, for which the gasket factor is,

$$C = 0.75 \text{ to } 1.0$$

Let us take,

$$C = 1.0$$

The external force on each stud bolt due to steam pressure is,

$$P = \frac{P_T}{n} = \frac{58.905 \times 10^3}{8} = 7363.1 \text{ N}$$

The initial tension required in the stud bolt for a fluid tight joint is,

$$F_i = 2840 \text{ d}$$

$\therefore$ The total force on each stud bolt is,

$$F = CP + F_i = 1 \times 7363.1 + 2840 \text{ d}$$

$$\text{or } F = 7363.1 + 2840 \text{ d}$$

NUMERICAL PROBLEMS

Size of stud bolt :

$$\sigma_t = \frac{F}{A_c}$$

$$\frac{S_{yt}}{N_f} = \frac{F}{A_c}$$

Assuming $N_f = 1$ and substituting Equation (a) in Equation (b), we get,

$$\frac{300}{1} = \frac{7363.1 + 2840\,d}{A_c}$$

$$300\,A_c = 7363.1 + 2840\,d$$

$$300 \times \frac{\pi d_c^2}{4} = 7363.1 + 2840 \times 1.19\,d_c \quad \{\text{For coarse series, } d \approx 1.19\,d_c\}$$

$$235.62\,d_c^2 = 7363.1 + 3379.6\,d_c$$

$$\therefore\ d_c^2 = 14.34\,d_c + 31.25$$

$$d_c^2 - 14.34\,d_c - 31.25 = 0$$

$$d_c = \frac{14.34 \pm \sqrt{(-14.34)^2 + 4 \times 1 \times 31.25}}{2 \times 1}$$

$$= \frac{14.34 \pm 18.18}{2} = 16.26 \text{ or } -1.92$$

$$\text{or} \quad d_c = 16.26 \text{ mm} \qquad \text{(taking positive value)}$$

$$d = 1.19\,d_c = 1.19 \times 16.26$$

$$d = 19.35 \text{ mm}$$

Form PSGDD Page No. 5.42, **M20 bolt (coarse series) is selected.** **...Ans.**

NUMERICAL PROBLEMS

Tightening torque :

$$
\begin{aligned}
T &= K F_i d \\
&= 0.2 F_i d \qquad \text{(Take } K \approx 0.2\text{)} \\
&= 0.2 \times 2840\, d \times d \\
&= 568\, d^2 \\
&= 568 \times (20)^2 \\
&= 22720 \text{ N-mm}
\end{aligned}
$$

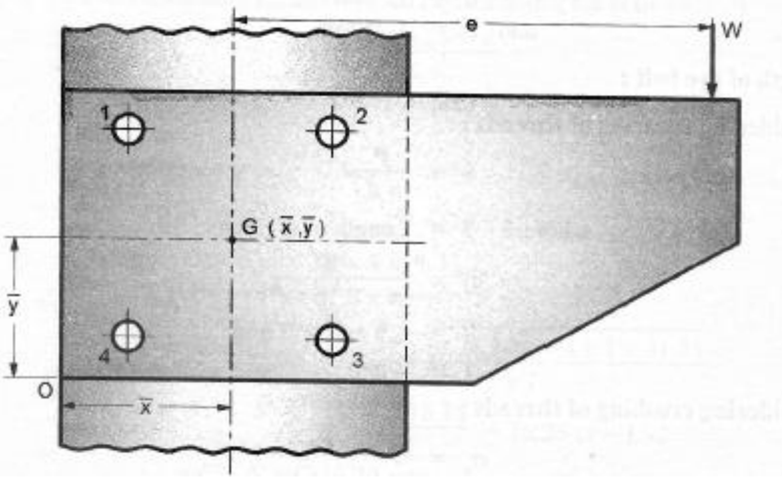
or $\mathbf{T = 227.2 \text{ N-m}}$

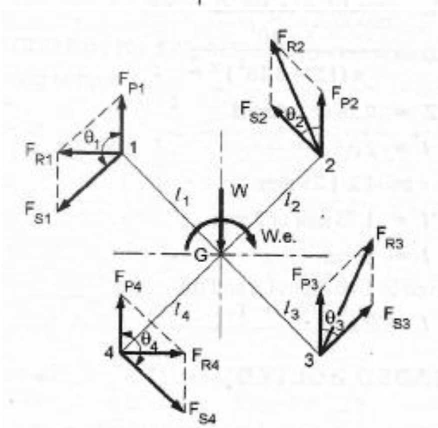
ECCENTIRCALLY LOADED BOLTED JOINTS

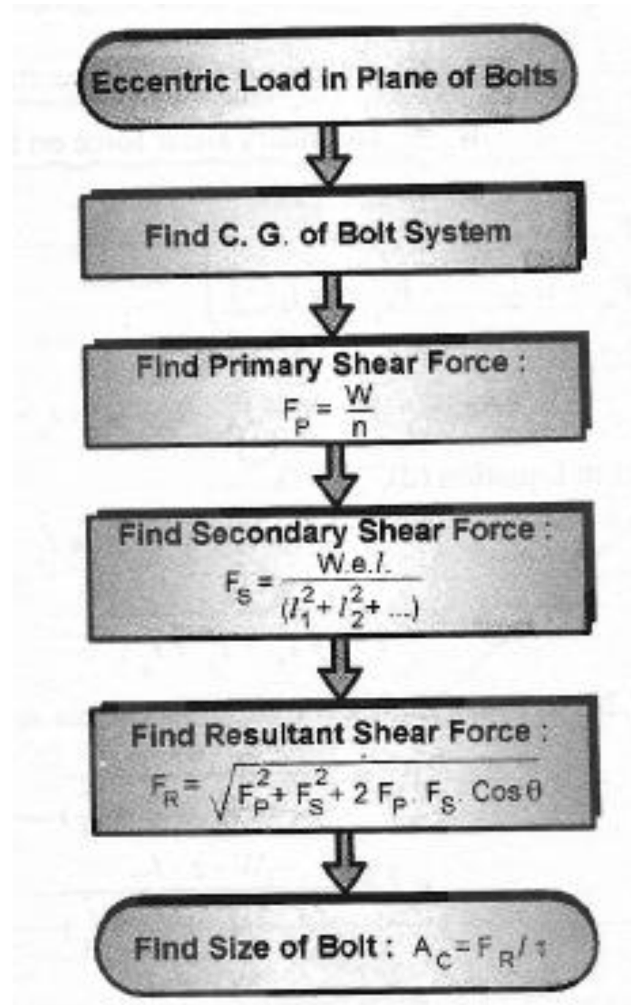
In many applications, the bolted joints are subjected to eccentric loading such as wall bracket, pillar crane etc., There are three possible cases.

1. **Eccentric load in the plane of bolts**
2. **Eccentric load perpendicular to the axis of bolt**
3. **Eccentric load parallel to the axis of bolt**

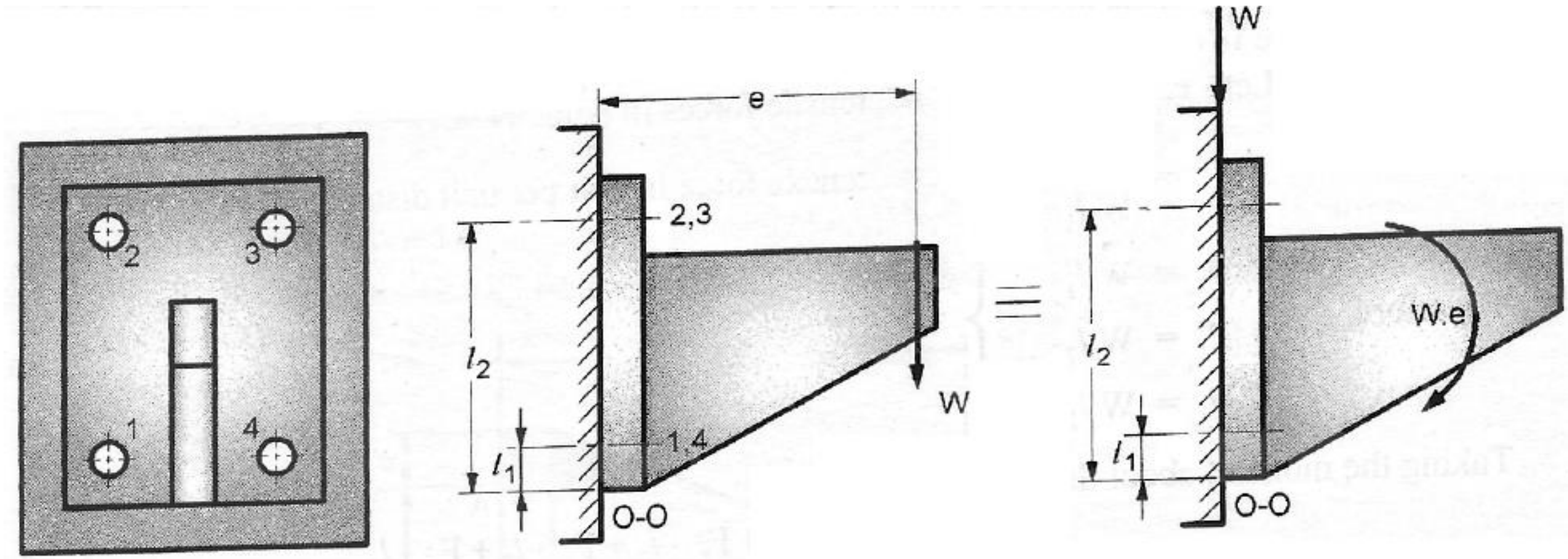
1. Eccentric load in the plane of bolts



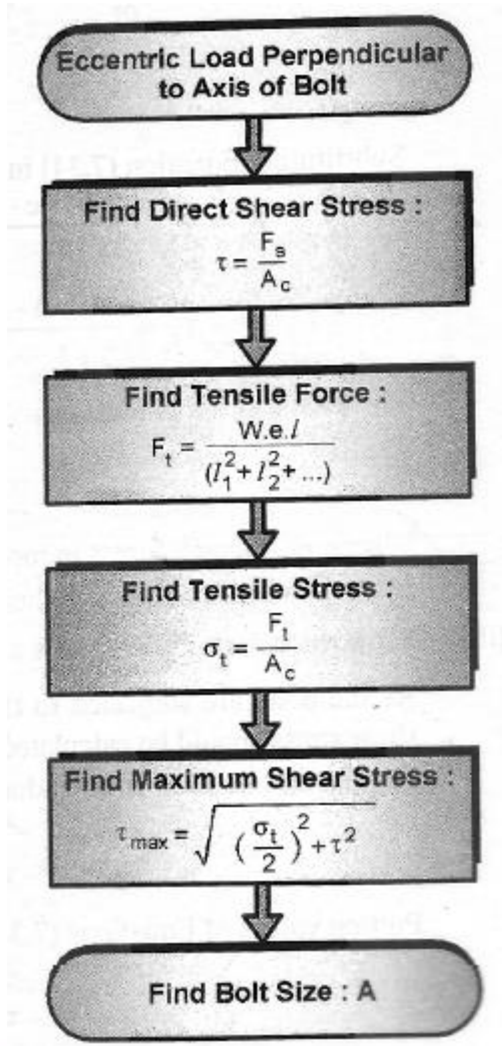




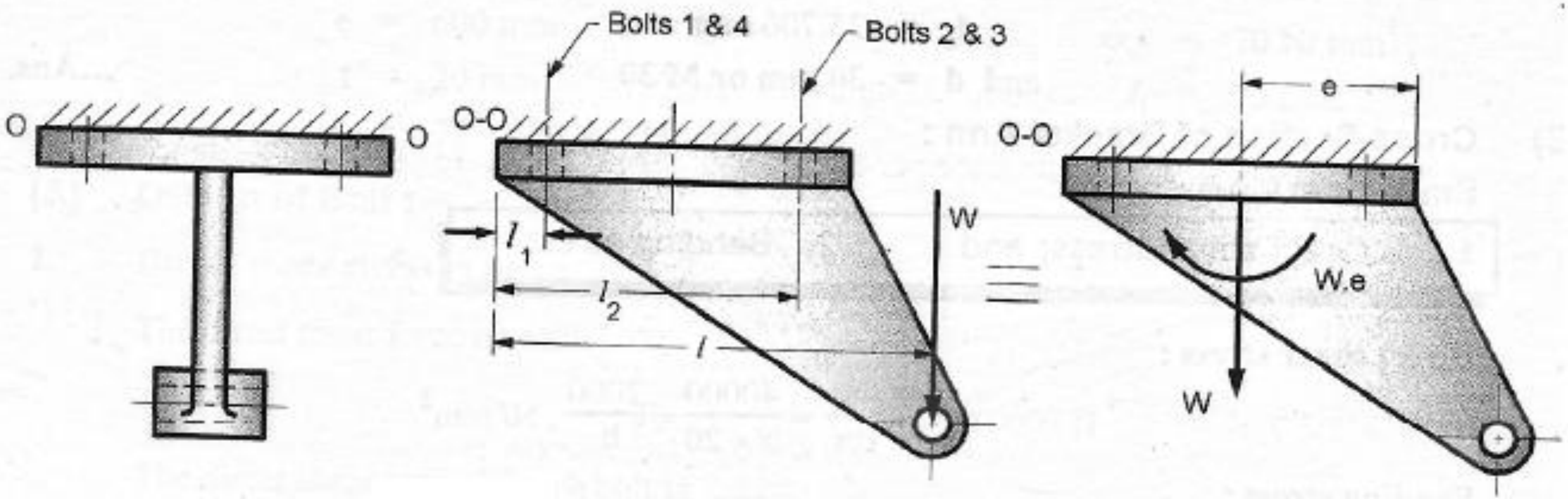
2. Eccentric load perpendicular to the axis of bolt



SCREWED FASTENERS

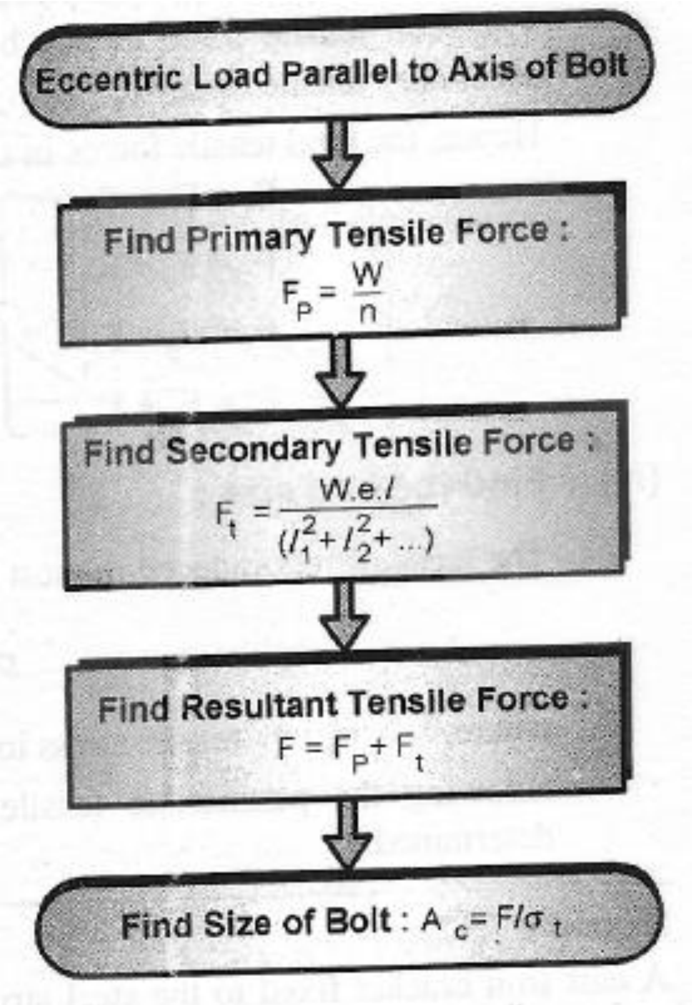


3. Eccentric load parallel to the axis of bolt.



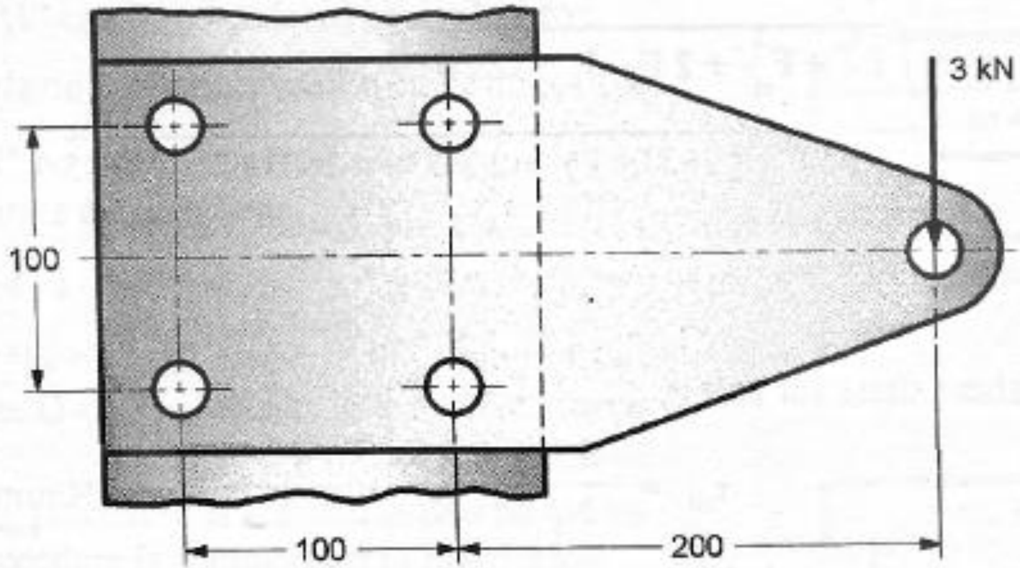
Eccentric Load Parallel to Axis of Bolt

SCREWED FASTENERS



NUMERICAL PROBLEMS

A steel plate subjected to a force of 3 kN and fixed to a vertical channel by means of four identical bolts is shown in Fig. The bolts are made of plain carbon steel 45C8 with yield strength of 380 N/mm^2. If the required factor of safety is 2, determine the diameter of bolts.

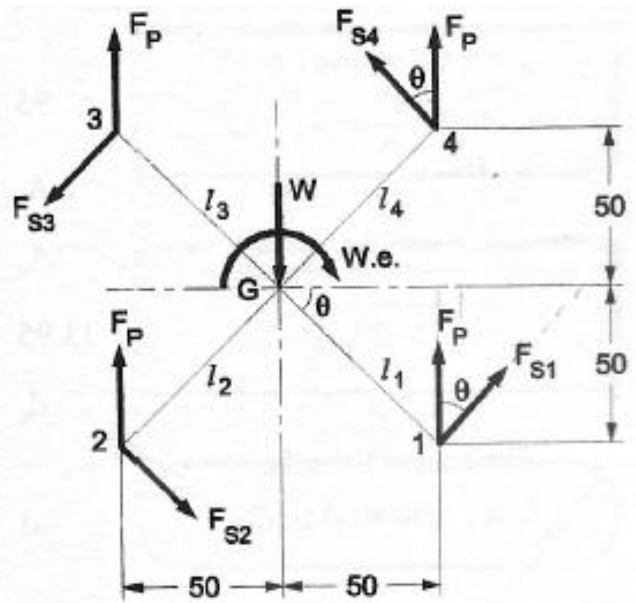


NUMERICAL PROBLEMS

Solution :

Given : $W = 3000$ N ; $S_{yt} = 380$ N/mm^2 ;

$n = 4$; $N_f = 2$.



1. Primary shear force :

Primary shear force on each bolt

$$F_p = \frac{W}{n} = \frac{3000}{4} = 750 \text{ N}$$

2. Secondary shear force :

From Fig.

$$e = 200 + 100/2 = 250 \text{ mm}$$

$$l_1 = l_2 = l_3 = l_4$$

$$= \sqrt{50^2 + 50^2} = 70.71 \text{ mm}$$

NUMERICAL PROBLEMS

$$W \cdot e = F_{s_1} \cdot l_1 + F_{s_2} \cdot l_2 + F_{s_3} \cdot l_3 + F_{s_4} \cdot l_4$$

$$W \cdot e = w\, l_1^2 + w\, l_2^2 + w\, l_3^2 + w\, l_4^2$$

$$\therefore \quad w = \frac{W \cdot e}{l_1^2 + l_2^2 + l_3^2 + l_4^2} = \frac{3000 \times 250}{4 \times (70.71)^2}$$

or $\quad w = 37.5$ N/mm

From Fig. 7.11.2, it is seen that the total load is maximum on bolt 1and bolt 4.

$$\theta = \tan^{-1}\left(\frac{50}{50}\right) = \tan^{-1}(1) = 45°$$

The secondary shear force on bolt 1 and 4 is,

$$F_{s_1} = F_{s_4} = w\, l_1 = 37.5 \times 70.71 = 2651.63 \text{ N}$$

3. Resultant shear force :

The resultant shear force on bolt 1 is,

$$F_{R_1} = \sqrt{F_p^2 + F_{s1}^2 + 2\, F_p \cdot F_{s_1} \cdot \cos\theta}$$

$$= \sqrt{(750)^2 + (2651.63)^2 + 2 \times 750 \times 2651.63 \times \cos 45}$$

or $\quad F_{R_1} = 3225.85$ N

NUMERICAL PROBLEMS

4. Bolt size :

The allowable shear stress for bolt is,

$$\tau_{all} = \frac{S_{sy}}{N_f} = \frac{0.5\, S_{yt}}{N_f} = \frac{0.5 \times 380}{2.6} = 95 \text{ N/mm}^2$$

The shear stress induced in bolt 1 is,

$$\tau = \frac{F_{R_1}}{A_c}$$

$$95 = \frac{3225.85}{A_c}$$

$$\therefore \quad A_c = 33.95 \text{ mm}^2$$

$$\text{Now,} \quad A_c = \pi d_c^2/4$$

$$33.95 = \pi d_c^2/4$$

$$\therefore \quad d_c = 6.575 \text{ mm}$$

For coarse threads,

$$d \approx 1.19\, d_c$$

$$\approx 1.19 \times 6.575$$

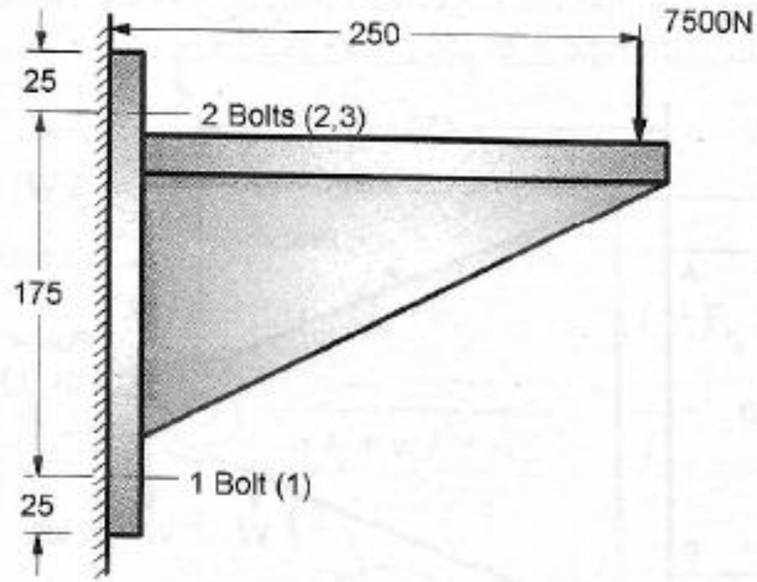
$$\therefore \quad d = 7.82 \text{ mm or } 8 \text{ mm}$$

$$\mathbf{d = 8 \text{ mm or M8}}$$

NUMERICAL PROBLEMS

A bracket, shown in Fig. is fixed to the support by means of three bolts. The dimensions given in figures are in mm. The bolts are made of plain carbon steel 45C8 (S_{yt} = 380 N/mm^2) and the factor of safety is 2.5. Specify the size of bolts assuming following relationship.

$$d = \frac{d_c}{0.84}$$



Solution :

Given : $W = 7500$ N ; $l_1 = 25$ mm;

$l_2 = l_3 = 200$ mm ; $S_{yt} = 380$ N/mm^2 ;

$N_f = 2.5$; $n = 3$.

1. Direct shear stress in bolt :

The direct shear force on each bolt is,

$$F_s = \frac{W}{n} = \frac{750}{3} = 2500 \text{ N}$$

The direct shear stress in each bolt is,

$$\tau = \frac{F_s}{A_c} = \frac{2500}{A_c}, \text{ N/mm}^2$$

2. Tensile stress in bolt :

Taking moment about tilting edge,

$$W \cdot e = w\,l_1^2 + w\,l_2^2 + w\,l_3^2$$

$$W \cdot e = w\,l_1^2 + 2w\,l_2^2$$

$$w = \frac{W \cdot e}{l_1^2 + 2\,l_2^2} = \frac{7500 \times 250}{25^2 + 2\,(200)^2}$$

$$\text{or} \quad w = 23.2558 \text{ N/mm}$$

The tensile force in most heavily loaded bolt is,

$$F_{t_2} = w\,l_2 = 23.2558 \times 200 = 4651.16 \text{ N}$$

NUMERICAL PROBLEMS

The tensile stress in most heavily loaded bolt is,

$$\sigma_t = \frac{F_{t_2}}{A_c} = \frac{4651.16}{A_c}, \text{ N/mm}^2$$

3. Maximum shear stress in bolt :

The maximum shear stress induced in most heavily loaded bolt is,

$$\tau_{max} = \sqrt{\left(\frac{\sigma_t}{2}\right)^2 + \tau^2} = \sqrt{\left(\frac{4651.16}{2\ A_c}\right)^2 + \left(\frac{2500}{A_c}\right)^2}$$

$$\text{or} \quad \tau_{max} = \frac{3414.43}{A_c}, \text{ N/mm}^2$$

4. Bolt size :

Now permissible shear stress for a bolt material is,

$$\tau_{all} = \frac{S_{sy}}{N_f} = \frac{0.5\ S_{yt}}{N_f} = \frac{0.5 \times 380}{2.5} = 76 \text{ N/mm}^2$$

$$\text{Now,} \quad 76 = \frac{3414.43}{A_c}$$

$$\therefore \quad A_c = 44.9266 \text{ mm}^2$$

$$\frac{\pi}{4} d_c^2 = 44.9266$$

$$\therefore \quad d_c = 7.563 \text{ mm}$$

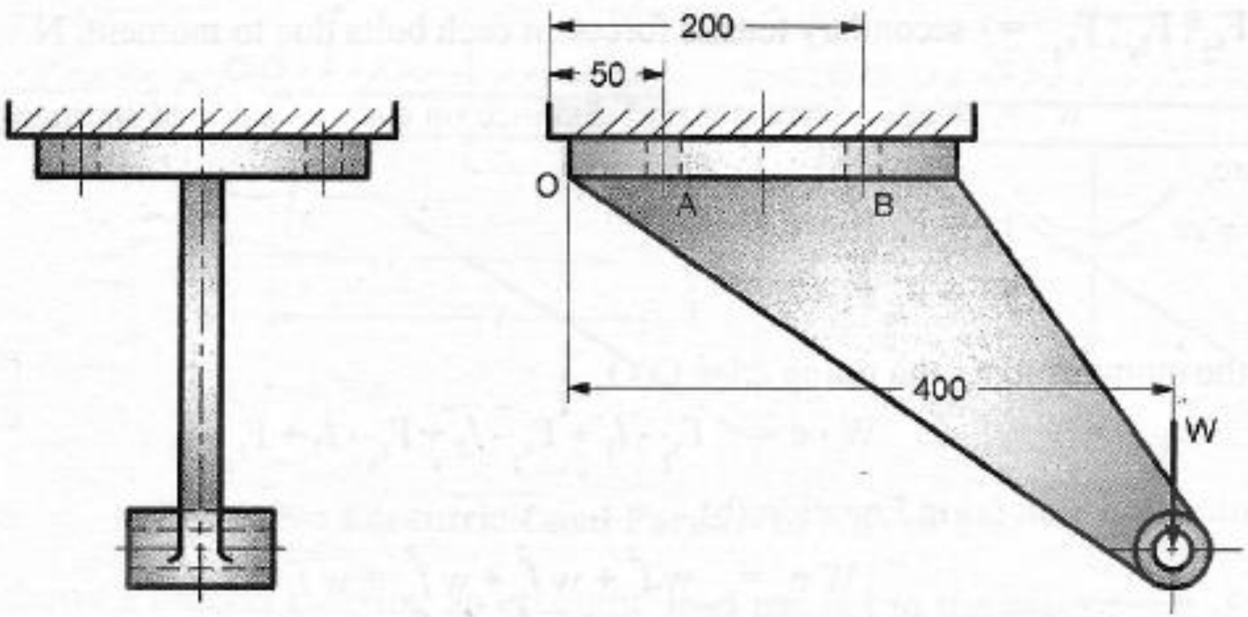
$$d = \frac{d_c}{0.84} = \frac{7.563}{0.84}$$

$$\therefore \quad d = 9.003 \text{ mm or } 10 \text{ mm}$$

$$\mathbf{d = 10 \text{ mm or } M10}$$

NUMERICAL PROBLEMS

A cast iron bracket fixed to the steel structure, as shown in Fig. supports a load **'W'** of 25 kN. There are two bolts each at A and B. If the permissible tensile stress for bolts is 50 N/mm^2, determine the size of the bolts.



NUMERICAL PROBLEMS

Solution :

Given : $W = 25 \times 10^3$ N ; $\sigma_t = 50$ N/mm^2

$l_1 = 50$ mm ; $l_2 = 200$ mm

$l = 400$ mm ; n = 4.

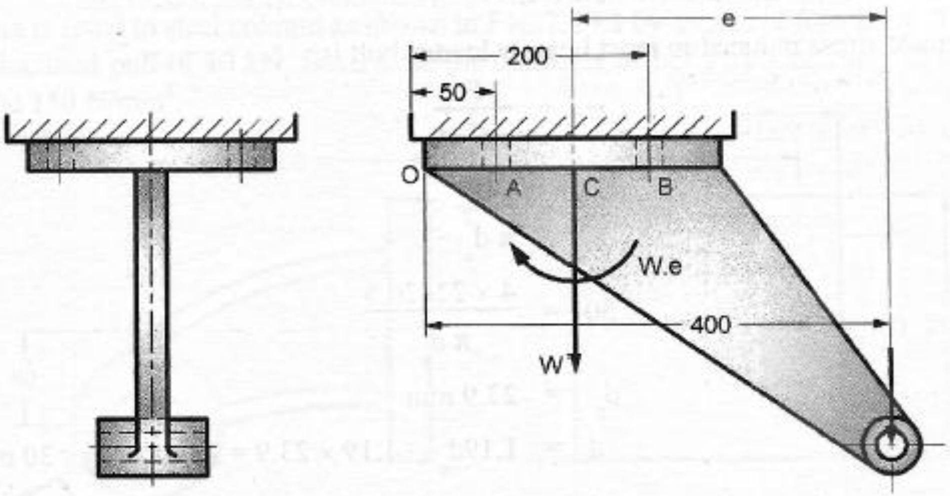


Fig. 7.17.2

1. **Primary tensile force :**

The primary tensile force in each bolt is,

$$F_p = \frac{W}{n} = \frac{25 \times 10^3}{4} = 6250 \text{ N}$$

NUMERICAL PROBLEMS

2. Secondary tensile force :

$$e = 400 - 200 + CB$$
$$= 200 + AB/2$$
$$= 200 + (200 - 50)/2$$

or

$$e = 275 \text{ mm}$$

Taking moment about O,

$$We = 2wl_1^2 + 2wl_2^2$$
$$25000 \times 10^3 \times 275 = 2w \times 50^2 + 2w \times 200^2$$
$$\therefore w = \frac{25 \times 10^3 \times 275}{2(50^2 + 200^2)}$$

or

$$w = 80.88 \text{ N/mm}$$

The second tensile force in most heavily loaded bolt is,

$$F_{t2} = wl_2 = 80.882 \times 200 = 16176.5 \text{ N}$$

3. Resultant tensile force :

$$F = F_p + F_{t2}$$
$$= 6250 + 16176.5$$

or

$$F = 22426.5 \text{ N}$$

4. Bolt size :

The tensile stress induced in most heavily loaded bolt is,

$$\sigma_t = \frac{F}{\pi d_c^2/4}$$

or

$$\sigma_t = \frac{4F}{\pi d_c^2}$$

$$\therefore \quad 50 = \frac{4 \times 22426.5}{\pi d_c^2}$$

$$\therefore \quad d_c = 23.9 \text{ mm}$$

$$\therefore \quad d = 1.19 d_c = 1.19 \times 23.9 = 28.44 \text{ mm or } 30 \text{ mm}$$

$$\mathbf{d = 30 \text{ mm or M30}}$$