THE GATE COACH

Revised Study Material

For

GATE – PSU

Mechanical Engineering

Machine Design

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CONTENTS

CHAPTER 8 THREADED JOINTS

INTRODUCTION

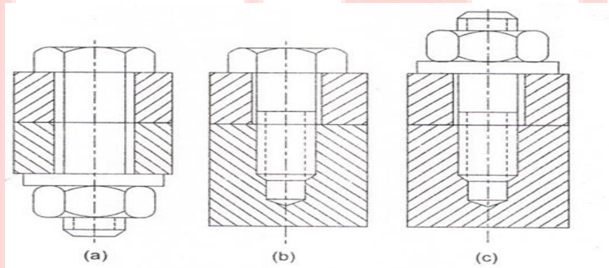
Threaded joint is defined as a separable joint of two or more machine parts that are held together by means of a threaded fastening such as a bolt and a nut. The salient features of this definition are as follows:

Threaded joints are used to hold two or more machine parts together. These parts can be dismantled, if required, without any damage to machine parts or fastening. Therefore, threaded joints are detachable joints, unlike welded joints.

BASIC TYPES OF SCREW FASTENING

There are three parts of a threaded fastening, viz., a bolt or screw, a nut and washer. There is a basic difference between the bolt and the screw.

- A bolt is a fastener with a head and straight threaded shank and intended to be used with a nut to clamp two or more parts.
- The same bolt can be called screw when it is threaded into a tapped hole in one of the parts and not into the nut.



Types of Screw Fastening: (a) Through Bolt (b) Tap Bolt (c) Stud

Simple washers are thin annular shaped metallic disks. The functions of a washer are as follows: -

1. It distributes the load over a large area on the surface of clamped parts.
2. It prevents marring of clamped parts during assembly.
3. It prevents marring of the bolt head and nut surface during assembly.
4. It provides bearing surface over large clearance holes.

Through Bolts: A through bolt is simply called a 'bolt' or a bolt and nut'. The bolt consists of a cylindrical rod with head at one end and threads at the other. The cylindrical portion between the head and the threads is called shank. Hexagonal head bolt and nut are popular in the machine building industry. Square head and nut are used mostly with rough type of bolts in construction work.

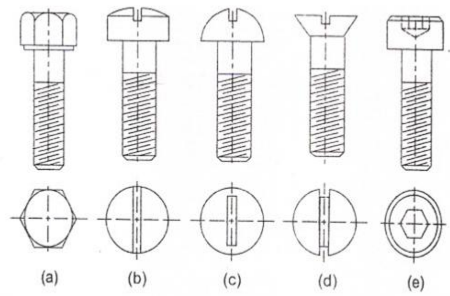
Tap Bolts and Cap Screws: There is basic difference between through bolt and tap bolt. The tap bolt is turned into a threaded (tapped) hole in one of the parts being connected and not into a nut. On the other hand, the through bolt is turned into the nut.

Studs: A stud is a cylindrical rod threaded at both end. One end of the stud is screwed into the tapped hole in one of the connecting parts. The other end of the stud receives a nut. Stud joints are used under the following conditions:

CAP SCREWS

Cap screws belong to the category of tap bolts. A wide variety of shapes are available for the head of cap screw. On the other hand, tap bolt has hexagonal or square head.Depending upon the shape of the head, cap screws are divided into the following two groups:

1. Cap screws in which the head is engaged externally by a spanner; and
2. Cap screws in which the head is engaged internally and from the end face

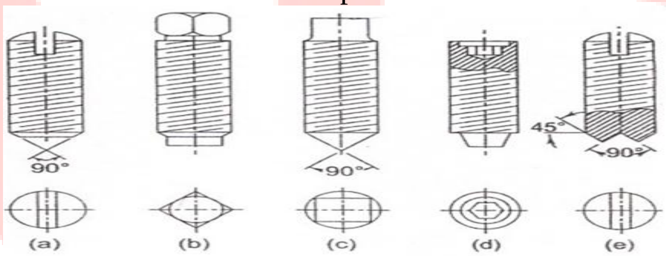


Cap Screws with Different Heads: (a) Hexagonal (b) Filister (c) Button (d) Flat (e) Fexagonal S

(a) (b) (c)

SETSCREWS

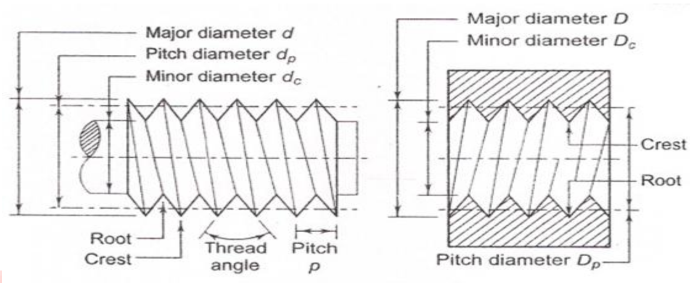
Setscrew is used to prevent relative motion between two parts. The threaded portion of the setscrew passes through a tapped hole in one of the parts and the end of the screw presses against the other part. The end of the screw is called the point of the screw.



Types of Set Screws

- *Flat Point*: Flat point is used when lateral force, which tens to displace one part with respect to another, is randomly applied.
- *Dog Point*: Dog point is used when the lateral force, which tends to displace one part with respect to other, is large.
- *Cone Point*: Cone point is used when the lateral force is small.
- *Hanger Point*: Hanger point has a smaller taper. It is used when the lateral force is large.
- *Cut Point*: Cut point is used when the part being held cannot be drilled or hardened.

TERMINOLOGY OF SCREW THREADS



Terminology of Threads

The right handed threads are always used unless there is special reason for requiring left hand thread.

Various dimensions of external and internal threads. They are as follows:

Major Diameter

The major diameter is the diameter of an imaginary cylinder that bounds the crest of an external thread (d) or the root of an internal thread (D). The major diameter is the largest diameter of the screw thread. It is also called the nominal diameter of the thread.

Minor Diameter

The minor diameter is the diameter of an imaginary cylinder that bounds the roots of an external thread (D_c) or the crest of an internal thread (D_c). The minor diameter is the smallest diameter of the screws thread. It is also called core or root diameter of the thread.

Pitch Diameter

The pitch diameter is the diameter of an imaginary cylinder, the surface of which would pass through the threads at such points as to make the width of the threads equal to the width of spaces cut by the surface of the cylinder. It is also called the effective diameter of the thread.

Pitch

Pitch is the distance between two similar points on threads measured parallel to the axis of the threads measured parallel to the axis of the thread. It is denoted by the letter p.

Lead

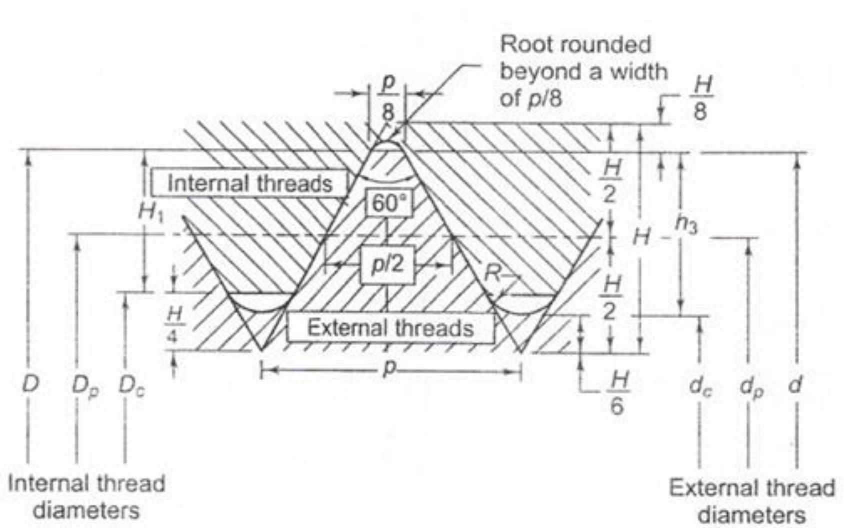
Lead is the distance that the nut moves parallel to the axis of the screw, when the nut is given one turn.

Thread Angle

Thread angle is the angle included between the sides of the thread measured in an axial plane. Thread angle is 60^0 for ISO metric threads.

Tensile Stress Area

It has been observed during testing of the threaded rods that an unthreaded rod, having a diameter equal to the mean of the pitch diameter and the minor diameter [i.e. $(d_p+d_c)/2$] has the same tensile strength as the threaded rod.



Profile of Internal and External Threads

BOLT JOINT-SIMPLE ANALYSIS

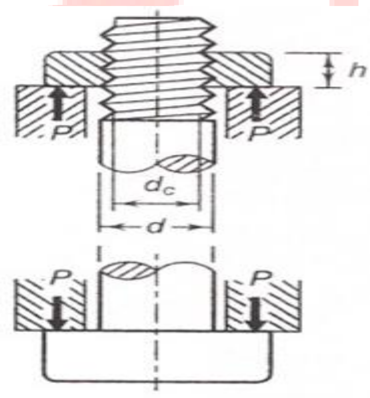
A bolted joint subjected to tensile force p. The cross section at the core diameter d_c is the weakest section. The maximum tensile stress in the bolt at this cross section is give by,

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

The height of the nut h can be determined by equating the strength of the bolt in tension with the strength in shear.

Assumptions:

1. Each turn of the thread in contact with the nut supports an equal amount of load.
2. There is no stress concentration in the threads.
3. The yield strength in shear is equal to half of the yield strength in tension ($S_{sy} = 0.5S_{yt}$).
4. Failure occurs in the threads of the bolt and not in the threads of the nut.



Bolt in Tension

Since

$$\sigma_t = \frac{S_{yt}}{(fos)}$$

The strength of the bolt in tension is given by:

$$P = \frac{\pi}{4} d_c^2 \frac{S_{yt}}{(fs)}$$

The threads of the bolt in contact with the nut are sheared at the core diameter d_c. The shear area is equal to ($\pi d_c h$), where h is the height of the nut. The strength of the bolt in shear is given by,

$$P = (\pi d_c h) \frac{S_{yt}}{(fs)}$$

$$= (\pi d_c h) \frac{S_{yt}}{(2fs)}$$

Equating

$$h = 0.5\ d_c$$

Assuming (d_c=0.8d),

$$h = 0.4d$$

Therefore, for standard coarse threads, the threads are equally strong in failure by shear and failure by tension, if the height of the nut is approximately 0.4 times of the nominal diameter of the bolt. The height of the standard hexagonal nut is (0.8d). Hence, the threads of the bolt in the standard nut will not fail by shear. Rewriting the height of the standard nut,

$$h = 0.8d$$

ECCENRTICALLY LOADED RIVETED JOINT

The primary shear forces are given by

$$P_1' = P_2' = P_3' = P_4' = \frac{P}{No.of\ bolts}$$

The secondary shear forces are given by

$$P_1'' = Cr_1$$
$$P_2'' = Cr_2$$
$$P_3'' = Cr_3$$
$$P_4'' = Cr_4$$

Where C = constant of proportionality.

$$C = \left(\frac{Pe}{{r_1}^2 + {r_2}^2 + {r_3}^2 + {r_4}^2}\right)$$

Hence

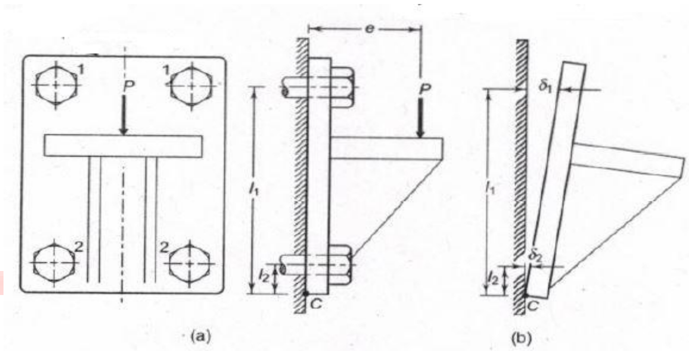
$$P_1'' = \frac{Per_1}{{r_1}^2 + {r_2}^2 + {r_3}^2 + {r_4}^2}$$
$$P_2'' = \frac{Per_2}{{r_1}^2 + {r_2}^2 + {r_3}^2 + {r_4}^2}$$
$$P_3'' = \frac{Per_3}{{r_1}^2 + {r_2}^2 + {r_3}^2 + {r_4}^2}$$
$$P_4'' = \frac{Per_4}{{r_1}^2 + {r_2}^2 + {r_3}^2 + {r_4}^2}$$

The primary and secondary shear forces are added vectorially added to get the resultant shear stress.

The maximum resultant shear force is equated to shear strength of bolt to find out diameter of rivet.

$$P_{max} = \left[\frac{\pi}{4} d^2\right] \tau$$

ECCENTRICLOAD PERPENDICULAR TO AXIS OF BOLT



Assumption:

1. The bracket and the steel structure are rigid.
2. The bolts are fitted in reamed and ground holes.
3. The bolts are not pre-loaded and there is no tensile stress due to initial tightening.
4. The stress concentration in the threads is neglected.
5. All bolts are identical.

Direct shear stress in each bolt due to force P,

$$P_1' = P_2' = \frac{P}{No.of\ bolts}$$

The moment (P×e) tend to tilt the bracket about C. Each bolt is stretched by an amount (δ) which is proportional to its vertical distance from the point C.

$$\delta_1 \propto l_1$$
$$\delta_2 \propto l_2$$

From stress-strain relationships, it can be concluded that

$$P_1'' \propto l_1$$
$$P_2'' \propto l_2$$

The secondary shear forces are given by

$$P_1'' = Cl_1$$
$$P_2'' = Cl_2$$

Where C = constant of proportionality.

$$C = \left(\frac{Pe}{{l_1}^2 + {l_2}^2}\right)$$

Hence

$$P_1'' = \frac{Pel_1}{{l_1}^2 + {l_2}^2}$$
$$P_2'' = \frac{Pel_2}{{l_1}^2 + {l_2}^2}$$

The bolt which is located at the farthest distance from the tilting edge C, is subjected to maximum force.

Shear stress is given by

$$\tau = \frac{P'_{max}}{A}$$

The tensile stress is given by

$$\sigma_t = \frac{P''_{max}}{A}$$

The principal stresses are given by

$$\sigma_1 = \frac{\sigma_t}{2} + \sqrt{\left(\frac{\sigma_t}{2}\right)^2 + \tau^2}$$

The principal shear stress is given by

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

ECCENTRIC LOAD ON CIRCULAR BASE

Assumption:

1. All bolts are identical.
2. The bearing and structure are rigid.
3. The bolts are not pre-loaded and there is no tensile stress due to initial tightening.
4. The stress concentration in the threads is neglected.
5. The bolts are relieved of shear stresses by dowel pins.

The resisting force acting on any due to tendency of the bearing to tilt, is proportional to its distance from tilting edge.

$$P_1 \propto l_1$$
$$P_1 = Cl_1$$

Similarly

$$P_2 = Cl_2$$
$$P_3 = Cl_3$$
$$P_4 = Cl_4$$

C = constant of proportionality.

$$Pl = P_1 l_1 + P_2 l_2 + P_3 l_3 + P_4 l_4$$

From both equations

$$Pl = C{l_1}^2 + C{l_2}^2 + C{l_3}^2 + C{l_4}^2$$
$$C = \frac{Pl}{{l_1}^2 + {l_2}^2 + {l_3}^2 + {l_4}^2}$$

Forces acting on bolt 1

$$P_1 = \frac{Pl l_1}{{l_1}^2 + {l_2}^2 + {l_3}^2 + {l_4}^2}$$

If a = radius of flange, b = radius of pitch circle of bolts

$$l_1 = a - b\cos\alpha$$
$$l_2 = a + b\sin\alpha$$
$$l_3 = a + b\cos\alpha$$
$$l_4 = a - b\sin\alpha$$

$$l_1{}^2 + l_2{}^2 + l_3{}^2 + l_4{}^2 = 4a^2 + 2b^2$$

Hence

$$P_1 = \frac{Pl(a - b\cos\alpha)}{2(2a^2 + b^2)} = \frac{2Pl(a - b\cos\alpha)}{4(2a^2 + b^2)}$$

Hence general expression for 'n' bolts becomes

$$P_1 = \frac{2Pl(a - b\cos\alpha)}{n(2a^2 + b^2)}$$

The maximum value will be when $\cos\alpha = -1$

$$P_{max} = \frac{2Pl(a + b)}{n(2a^2 + b^2)}$$

EXAMPLE

A crane-runway bracket is fastened to the roof truss by means of two identical bolts as shown in figure. Determine the size of the bolts, if permissible tensile stress in the bolts is limited to 75 N/mm^2.

SOLUTION

Given P = 20 kN, e = 550 mm, $(\sigma_t)_{max} = 75\ N/mm^2$

STEP I: Direct tensile stress

$$P_1 = P_2 = \frac{P}{No.of\ bolts} = \frac{20 \times 1000}{2} = 10000\ N$$

STEP II: Tensile force due to the tendency of bracket to tilt.

$$P_1" \propto l_1 \text{and}$$
$$P_2" \propto l_2$$

Then

$$P_1" = Cl_1 \text{and}$$
$$P_2" Cl_2$$

Moment of resisting forces

$$P \times 550 = P_1" l_1 + P_2" l_2$$

From two eq

$$P \times 550 = C\left(l_1{}^2 + l_2{}^2\right)$$

$$C = \frac{P \times 550}{l_1{}^2 + l_2{}^2}$$

Hence

$$P_1" = \frac{P \times 550}{l_1{}^2 + l_2{}^2} l_1$$

Substituting values

$$P_1" = 24146.24\ N$$

STEP III: Resultant tensile force

Bolt 1 is at farthest point, hence it is subjected to maximum tensile force.

Hence total force on bolt 1 is

= (10000+24146.34) = 34146.34 N

STEP IV: Size of bolts

$$A(\sigma_t)_{max} = 34146.34$$
$$A(75) = 34146.34$$
$$A = 455.28\,mm^2$$

Hence standard size of bolt is M30.