



MACHINE DESIGN

TOPIC:-

INTRODUCTION, ADVANTAGES AND DISADVANTAGES OF SCREW JOINTS

LOCATION OF SCREW JOINTS

IMPORTANT TERMS USED IN SCREW THREADS

11.1 Introduction

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 screwed joint is mainly composed of two elements *i.e.* a bolt and nut. The screwed joints are widely used where the machine parts are required to be readily connected or disconnected without damage to the machine or the fastening. This may be for the purpose of holding or adjustment in assembly or service inspection, repair, or replacement or it may be for the manufacturing or assembly reasons.

11.2 Advantages and Disadvantages of Screwed Joints

Following are the advantages and disadvantages of the 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

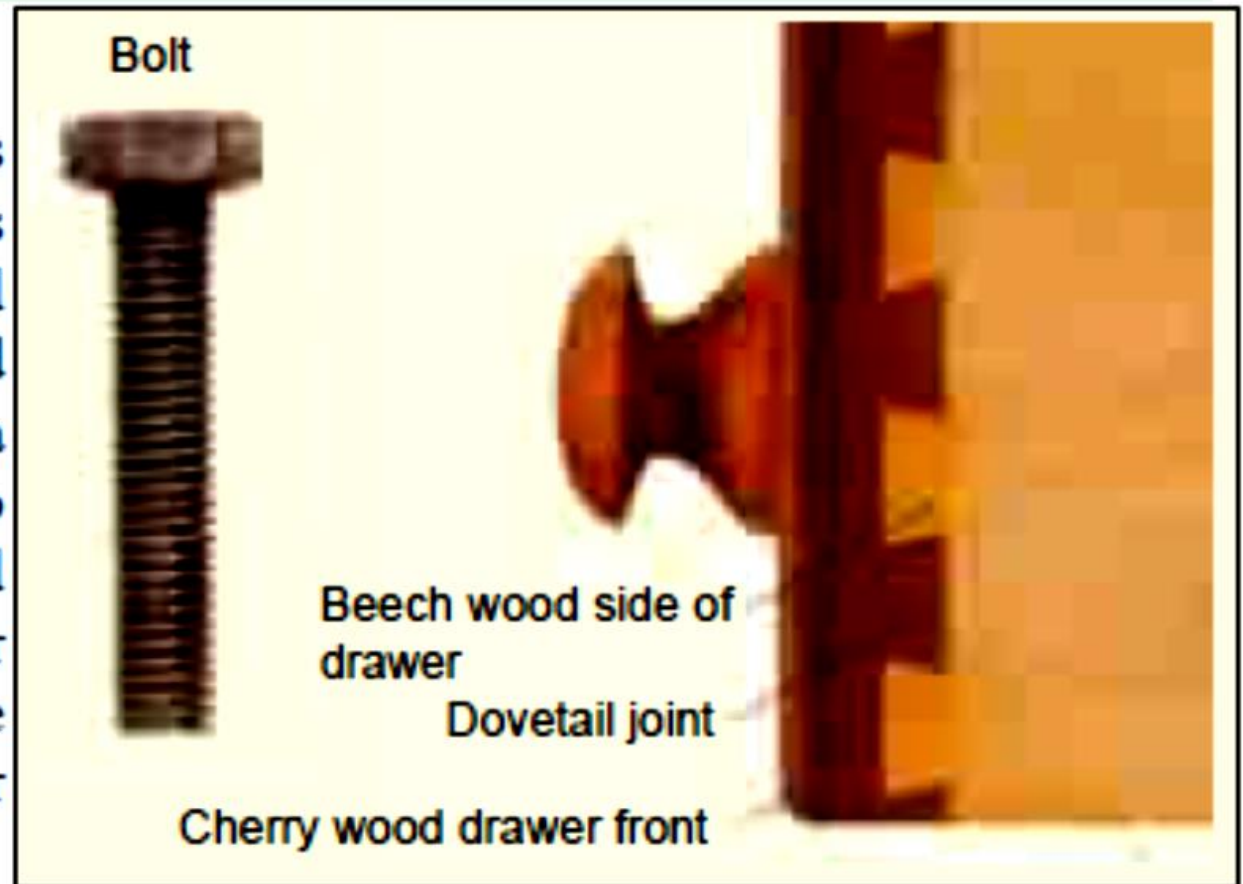
The main disadvantage of the screwed joints is the stress concentration in the threaded portions which are vulnerable points under variable load conditions.

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



11.5 Location of Screwed Joints

The choice of type of fastenings and its location are very important. The fastenings should be located in such a way so that they will be subjected to tensile and/or shear loads and bending of the fastening should be reduced to a minimum. The bending of the fastening due to misalignment, tightening up loads, or external loads are responsible for many failures. In order to relieve fastenings of bending stresses, the use of clearance spaces, spherical seat washers, or other devices may be used.



11.3 Important Terms Used in Screw Threads

The following terms used in screw threads, as shown in Fig. 11.1, are important from the subject point of view :

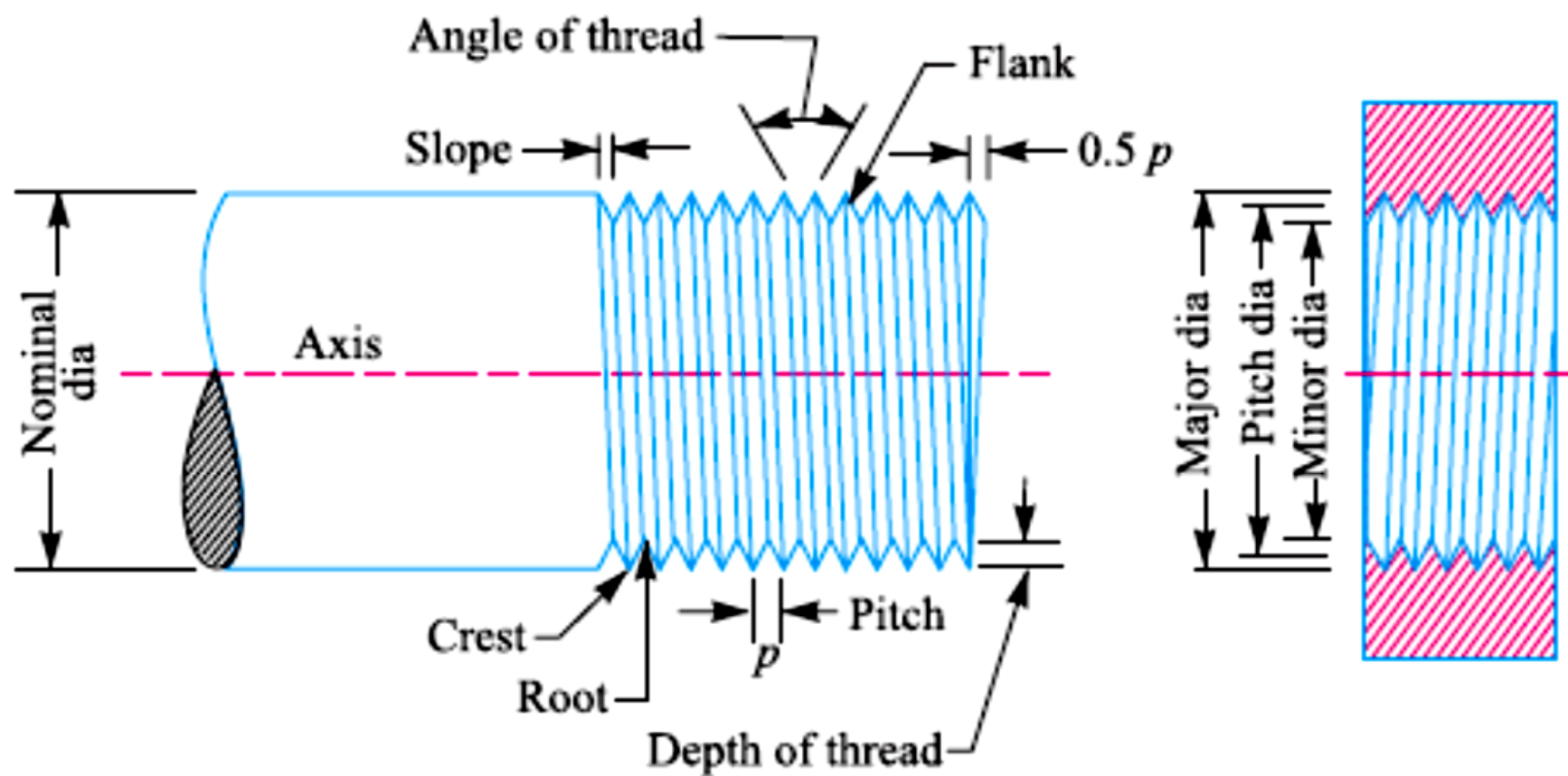


Fig. 11.1. Terms used in screw threads.

1. Major diameter. It is the largest diameter of an external or internal screw thread. The screw is specified by this diameter. It is also known as *outside* or *nominal diameter*.

2. Minor diameter. It is the smallest diameter of an external or internal screw thread. It is also known as *core* or *root diameter*.

3. Pitch diameter. It is the diameter of an imaginary cylinder, on a cylindrical screw thread, the surface of which would pass through the thread at such points as to make equal the width of the thread and the width of the spaces between the threads. It is also called an *effective diameter*. In a nut and bolt assembly, it is the diameter at which the ridges on the bolt are in complete touch with the ridges of the corresponding nut.

4. Pitch. It is the distance from a point on one thread to the corresponding point on the next. This is measured in an axial direction between corresponding points in the same axial plane. Mathematically,

$$\text{Pitch} = \frac{1}{\text{No. of threads per unit length of screw}}$$

5. Lead. It is the distance between two corresponding points on the same helix. It may also be defined as the distance which a screw thread advances axially in one rotation of the nut. Lead is equal to the pitch in case of single start threads, it is twice the pitch in double start, thrice the pitch in triple start and so on.

6. Crest. It is the top surface of the thread.

7. Root. It is the bottom surface created by the two adjacent flanks of the thread.

8. Depth of thread. It is the perpendicular distance between the crest and root.

9. Flank. It is the surface joining the crest and root.

10. Angle of thread. It is the angle included by the flanks of the thread.

11. Slope. It is half the pitch of the thread.



MACHINE DESIGN

TOPIC:-

1. DESIGNATION OF SCREW THREADS
2. INITIAL STRESSES DUE TO SCREW UP FORCES
3. NUMERICALS

11.8 Designation of Screw Threads

According to Indian standards, IS : 4218 (Part IV) 1976 (Reaffirmed 1996), the complete designation of the screw thread shall include

1. *Size designation.* The size of the screw thread is designated by the letter 'M' followed by the diameter and pitch, the two being separated by the sign \times . When there is no indication of the pitch, it shall mean that a coarse pitch is implied.

2. *Tolerance designation.* This shall include

(a) A figure designating tolerance grade as indicated below:

'7' for fine grade, '8' for normal (medium) grade, and '9' for coarse grade.

(b) A letter designating the tolerance position as indicated below :

'H' for unit thread, 'd' for bolt thread with allowance, and 'h' for bolt thread without allowance.

For example, A bolt thread of 6 mm size of coarse pitch and with allowance on the threads and normal (medium) tolerance grade is designated as *M6-8d*.

11.11 Initial Stresses due to Screwing up Forces

The following stresses are induced in a bolt, screw or stud when it is screwed up tightly.

1. **Tensile stress due to stretching of bolt.** Since none of the above mentioned stresses are accurately determined, therefore bolts are designed on the basis of direct tensile stress with a large factor of safety in order to account for the indeterminate stresses. The initial tension in a bolt, based on experiments, may be found by the relation

$$P_i = 2840 d \text{ N}$$

where

P_i = Initial tension in a bolt, and

d = Nominal diameter of bolt, in mm.

The above relation is used for making a joint fluid tight like steam engine cylinder cover joints etc. When the joint is not required as tight as fluid-tight joint, then the initial tension in a bolt may be reduced to half of the above value. In such cases

$$P_i = 1420 d \text{ N}$$

The small diameter bolts may fail during tightening, therefore bolts of smaller diameter (less than M 16 or M 18) are not permitted in making fluid tight joints.

If the bolt is not initially stressed, then the maximum safe axial load which may be applied to it, is given by

P = Permissible stress \times Cross-sectional area at bottom of the thread
(i.e. stress area)

The stress area may be obtained from Table 11.1 or it may be found by using the relation

$$\text{Stress area} = \frac{\pi}{4} \left(\frac{d_p + d_c}{2} \right)^2$$

where

d_p = Pitch diameter, and

d_c = Core or minor diameter.

2. Torsional shear stress caused by the frictional resistance of the threads during its tightening. The torsional shear stress caused by the frictional resistance of the threads during its tightening may be obtained by using the torsion equation. We know that

$$\frac{T}{J} = \frac{\tau}{r}$$

$$\therefore \tau = \frac{T}{J} \times r = \frac{T}{\frac{\pi}{32} (d_c)^4} \times \frac{d_c}{2} = \frac{16 T}{\pi (d_c)^3}$$

where

τ = Torsional shear stress,

T = Torque applied, and

d_c = Minor or core diameter of the thread.

It has been shown during experiments that due to repeated unscrewing and tightening of the nut, there is a gradual scoring of the threads, which increases the torsional twisting moment (T).

3. Shear stress across the threads. The average thread shearing stress for the screw (τ_s) is obtained by using the relation :

$$\tau_s = \frac{P}{\pi d_c \times b \times n}$$

where

b = Width of the thread section at the root.

The average thread shearing stress for the nut is

$$\tau_n = \frac{P}{\pi d \times b \times n}$$

where

d = Major diameter.

4. Compression or crushing stress on threads. The compression or crushing stress between the threads (σ_c) may be obtained by using the relation :

$$\sigma_c = \frac{P}{\pi [d^2 - (d_c)^2] n}$$

where

d = Major diameter,

d_c = Minor diameter, and

n = Number of threads in engagement.

5. Bending stress if the surfaces under the head or nut are not perfectly parallel to the bolt axis. When the outside surfaces of the parts to be connected are not parallel to each other, then the bolt will be subjected to bending action. The bending stress (σ_b) induced in the shank of the bolt is given by

$$\sigma_b = \frac{x \cdot E}{2l}$$

where

x = Difference in height between the extreme corners of the nut or head,

l = Length of the shank of the bolt, and

E = Young's modulus for the material of the bolt.

Example 11.2. Two machine parts are fastened together tightly by means of a 24 mm tap bolt. If the load tending to separate these parts is neglected, find the stress that is set up in the bolt by the initial tightening.

Solution. Given : $d = 24$ mm

From Table 11.1 (coarse series), we find that the core diameter of the thread corresponding to M 24 is $d_c = 20.32$ mm.

Let $\sigma_t =$ Stress set up in the bolt.

We know that initial tension in the bolt,

$$P = 2840 d = 2840 \times 24 = 68\ 160 \text{ N}$$

We also know that initial tension in the bolt (P),

$$68\ 160 = \frac{\pi}{4} (d_c)^2 \sigma_t = \frac{\pi}{4} (20.30)^2 \sigma_t = 324 \sigma_t$$

$$\therefore \sigma_t = 68\ 160 / 324 = 210 \text{ N/mm}^2 = 210 \text{ MPa} \text{ Ans.}$$



MACHINE DESIGN

TOPICS:-

DESIGN OF SCREW JACK

17.14 Design of Screw Jack

A bottle screw jack for lifting loads is shown in Fig. 17.11. The various parts of the screw jack are as follows:

1. Screwed spindle having square threaded screws,
2. Nut and collar for nut,
3. Head at the top of the screwed spindle for handle,
4. Cup at the top of head for the load, and
5. Body of the screw jack.

In order to design a screw jack for a load W , the following procedure may be adopted:

1. First of all, find the core diameter (d_c) by considering that the screw is under pure compression, *i.e.*

$$W = \sigma_c \times A_c = \sigma_c \times \frac{\pi}{4} (d_c)^2$$

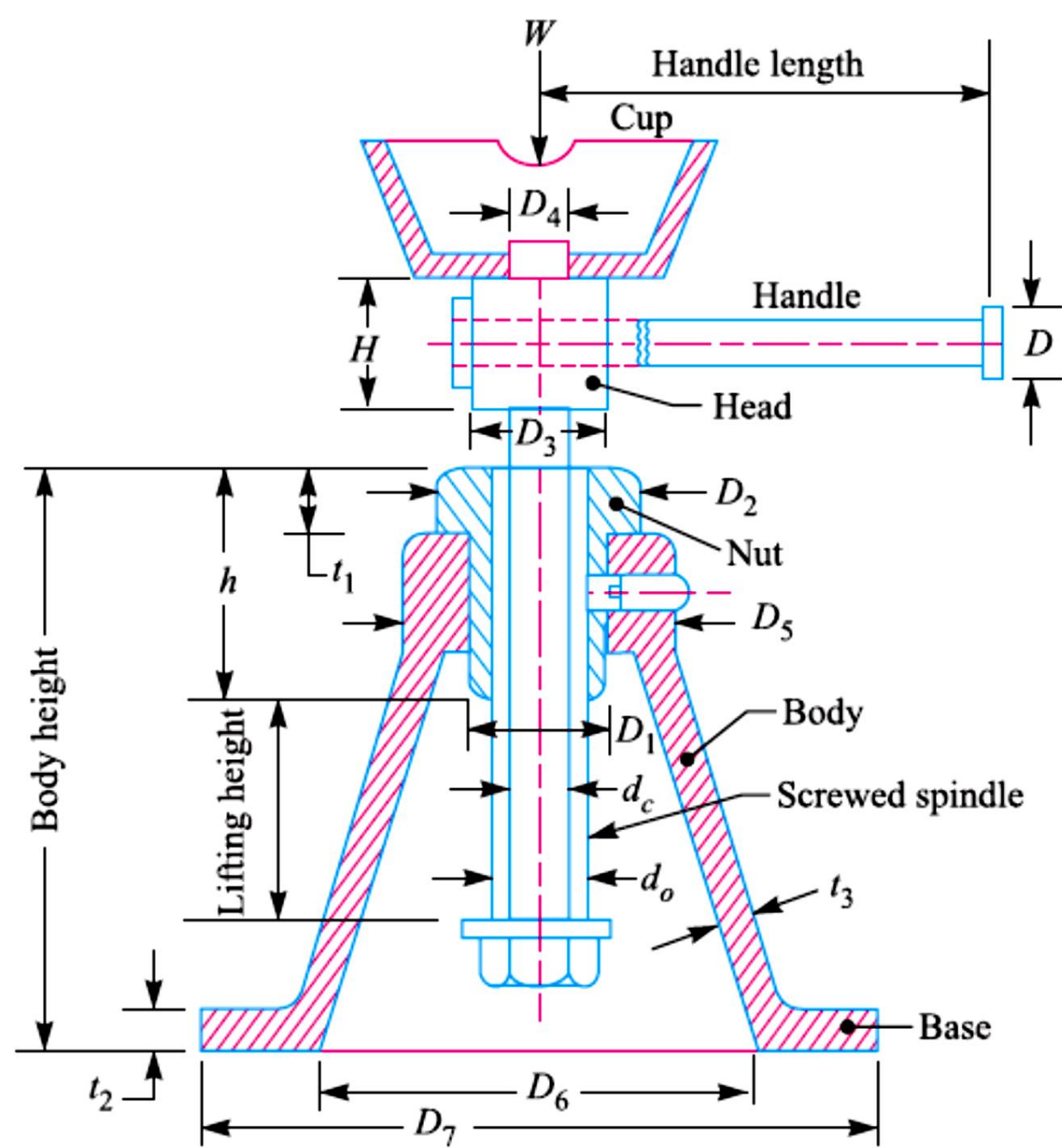


Fig. 17.11. Screw jack.

2. Find the torque (T_1) required to rotate the screw and find the shear stress (τ) due to this torque.

We know that the torque required to lift the load,

$$T_1 = P \times \frac{d}{2} = W \tan (\alpha + \phi) \frac{d}{2}$$

where

P = Effort required at the circumference of the screw, and

d = Mean diameter of the screw.

\therefore Shear stress due to torque T_1 ,

$$\tau = \frac{16 T_1}{\pi (d_c)^3}$$

Also find direct compressive stress (σ_c) due to axial load, *i.e.*

$$\sigma_c = \frac{W}{\frac{\pi}{4} (d_c)^2}$$

3. Find the principal stresses as follows:

Maximum principal stress (tensile or compressive),

$$\sigma_{c(max)} = \frac{1}{2} \left[\sigma_c + \sqrt{(\sigma_c)^2 + 4\tau^2} \right]$$

and maximum shear stress,

$$\tau_{max} = \frac{1}{2} \sqrt{(\sigma_c)^2 + 4\tau^2}$$

These stresses should be less than the permissible stresses.

4. Find the height of nut (h), considering the bearing pressure on the nut. We know that the bearing pressure on the nut,

$$P_b = \frac{W}{\frac{\pi}{4} [(d_o)^2 - (d_c)^2] n}$$

where

n = Number of threads in contact with screwed spindle.

\therefore Height of nut,

$h = n \times p$

where

p = Pitch of threads.

5. Check the stressess in the screw and nut as follows :

$$\tau_{(screw)} = \frac{W}{\pi n.d_c.t}$$

$$\tau_{(nut)} = \frac{W}{\pi n.d_o.t}$$

where

$$t = \text{Thickness of screw} = p / 2$$

6. Find inner diameter (D_1), outer diameter (D_2) and thickness (t_1) of the nut collar.

The inner diameter (D_1) is found by considering the tearing strength of the nut. We know that

$$W = \frac{\pi}{4} [(D_1)^2 - (d_o)^2] \sigma_t$$

The outer diameter (D_2) is found by considering the crushing strength of the nut collar. We know that

$$W = \frac{\pi}{4} [(D_2)^2 - (D_1)^2] \sigma_c$$

The thickness (t_1) of the nut collar is found by considering the shearing strength of the nut collar. We know that

$$W = \pi D_1.t_1.\tau$$

7. Fix the dimensions for the diameter of head (D_3) on the top of the screw and for the cup. Take $D_3 = 1.75 d_o$. The seat for the cup is made equal to the diameter of head and it is chamfered at the top. The cup is fitted with a pin of diameter $D_4 = D_3 / 4$ approximately. This pin remains a loose fit in the cup.

8. Find the torque required (T_2) to overcome friction at the top of screw. We know that

$$T_2 = \frac{2}{3} \times \mu_1 W \left[\frac{(R_3)^3 - (R_4)^3}{(R_3)^2 - (R_4)^2} \right] \quad \dots \text{(Assuming uniform pressure conditions)}$$

$$= \mu_1 W \left[\frac{R_3 + R_4}{2} \right] = \mu_1 W R \quad \dots \text{(Assuming uniform wear conditions)}$$

where $R_3 =$ Radius of head, and
 $R_4 =$ Radius of pin.

9. Now the total torque to which the handle will be subjected is given by

$$T = T_1 + T_2$$

Assuming that a person can apply a force of 300 – 400 N intermittently, the length of handle required

$$= T / 300$$

The length of handle may be fixed by giving some allowance for gripping.

10. The diameter of handle (D) may be obtained by considering bending effects. We know that bending moment,

$$M = \frac{\pi}{32} \times \sigma_b \times D^3 \quad \dots (\because \sigma_b = \sigma_t \text{ or } \sigma_c)$$

11. The height of head (H) is usually taken as twice the diameter of handle, *i.e.* $H = 2D$.
12. Now check the screw for buckling load.

Effective length or unsupported length of the screw,

$$L = \text{Lift of screw} + \frac{1}{2} \text{ Height of nut}$$

We know that buckling or critical load,

$$W_{cr} = A_c \cdot \sigma_y \left[1 - \frac{\sigma_y}{4 C \pi^2 E} \left(\frac{L}{k} \right)^2 \right]$$

where

σ_y = Yield stress,

C = End fixity coefficient. The screw is considered to be a strut with lower end fixed and load end free. For one end fixed and the other end free, $C = 0.25$

k = Radius of gyration = $0.25 d_c$

The buckling load as obtained by the above expression must be higher than the load at which the screw is designed.

13. Fix the dimensions for the body of the screw jack.
14. Find efficiency of the screw jack.

