VTU e-Shikshana Programme 2

V Semester: B.E. (Mechanical Engineering)
Design of Machine Elements:1 (17ME54)

Module – 5
Design of Threaded Fasteners
and Power Screws

By

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Topics to Learn

- Design of Threaded Fasteners
- Terminology/Type/Manufacture
- Strength of bolts/Materials used
- Design of Bolts for static loads
- Eccentrically loaded bolted joints
- Design of Power Screws/Applications
- Design of Screw Jack

Organisation of Lecture Notes

Section 1: Introduction, Relevance of Threaded Fasteners in Design and Development of Machines, Reasons for extensive use of Threaded fasteners, commonly used threaded fasteners. Terminology, Forms of screw threads, Specifications of ISO Metric Threads, Materials, Design of threaded fasteners under static loads, Preload due to initial tightening, Stresses in bolts, Stresses due to initial tightening, Numerical Examples.

Section 2: Threaded joints under external loads, Joints for clamping and joints for pressure vessels/ cylinder heads, relative stiffness, resultant load, design procedure for cylinder head bolts/studs, Numerical Examples.

Section 3: Eccentrically loaded bolted joints: Introduction, Assumptions, General theory, Types of Eccentrically loaded bolted joints, Load / stress analysis, Design equations, Numerical examples.

Section 4: Power Screws: Introduction, Advantages and disadvantages, Form of threads: Square, Acme, trapezoidal and buttress threads, Single start and Multi start threads, Designation of Power screws, Analysis of square threads, Torque and efficiency of Power Screws: Torque required to raise the load, Torque required to lower the load, Collar friction, Efficiency, Over-hauling and self-locking, Number of threads in the nut, Height of nut, Stresses induced in power screws. Numerical examples.

Section 1

Introduction

Threaded Elements: Machine elements on which threads are cut are called threaded elements. They are broadly classified as 1) Threaded Fasteners / Threaded Joints, 2) Power screws,

Threaded Fasteners are threaded elements used in threaded joints for fastening or clamping parts. Usually parts are connected or clamped by using multiple number of threaded fasteners depending upon the load to be sustained by the joint. Examples: Screws, Bolt and Nut, studs etc,.

Power screws are threaded elements which are not used for fastening purposes. They are used to convert rotary motion in to linear motion. They are also called as transmission screws. Examples: Screws of a screw jack, C clamp, Machine vice, Lead screw of a lathe, etc,.

Threaded joints are of two types:

Screwed joints: In these joints screw is threaded in to a tapped hole in the part itself. Torque is applied on the screw.

Bolted joints: Nut and bolt are used to secure the joint: Used where the thickness of the part to be connected is not sufficient to accommodate the screw. Torque is applied on the nut.

In the above joints washers may also be used to ensure distribution of load, to prevent damage of clamped parts and to provide bearing surface over large clearance holes.

Relevance of Threaded Fasteners in Design and Development of Machines

Design and development of machines is a vast field which involves several inter related activities. Major activities are: Need analysis for defining design requirements, to set up specifications, concept generation, concept screening, concept evaluation and concept selection, Detail design, prototype building and testing and evaluation of prototype and finally production/manufacture. Once the systems/sub systems/components are manufactured they are assembled to perform the intended task. Assembly of the systems/sub systems/components hold them together to transmit or transform motion and forces. Different fastening methods are employed in assembly process. Fasteners used may be permanent or removable. Rivets and welds are permanent fasteners. Screws, blots and nuts, keys etc, are removable or separable fasteners. Among all these fasteners threaded fasteners are more extensively used. It is reported that 60% of machine elements are threaded elements.

Reasons for extensive use of Threaded fasteners

- ✓ Joints are removable / separable
- ✓ Easy to install and use
- ✓ Available in variety of standard sizes as off - the - shelf items
- ✓ Standardization reduces cost
- ✓ Parts are held together by large clamping force due to wedge action at the threads thereby loosening of parts avoided
- ✓ Reliable, self-locking : can be used in any orientation

Disadvantages

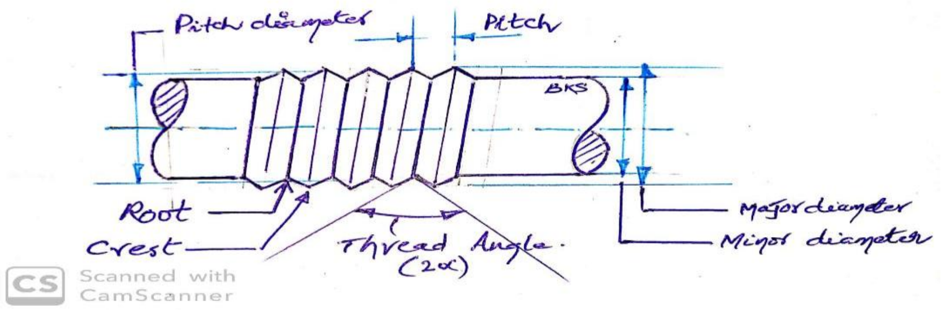
Require holes in parts to be clamped. Hence stress concentration weakens the parts.

Commonly used threaded fasteners:

- Machine screws, Wood screws
- Tapping screws
- Nuts and bolts: U bolts , Eye bolts, Carriage bolts, Through bolts, Tap bolts, Studs etc,

Terminolgy

Figure shows a treaded element. Following terms are used to specify a threaded element.



Major diameter: Nominal diameter: Diameter of an imaginary cylinder that bounds the crests of an external thread or roots of an internal thread.

Minor diameter: Core or Root diameter: Diameter of an imaginary cylinder that envelopes the roots of an external thread or crests of an internal thread.

Pitch diameter: Effective diameter: Diameter of an imaginary cylinder the surface of which pass through such points where the width of the threads is equal to the space between the threads.

Pitch: Axial distance between two similar points on adjacent threads.

Lead: Distance moved by a nut along the bolt axis in one revolution.

Thread angle: Angle included between the sides of the thread.

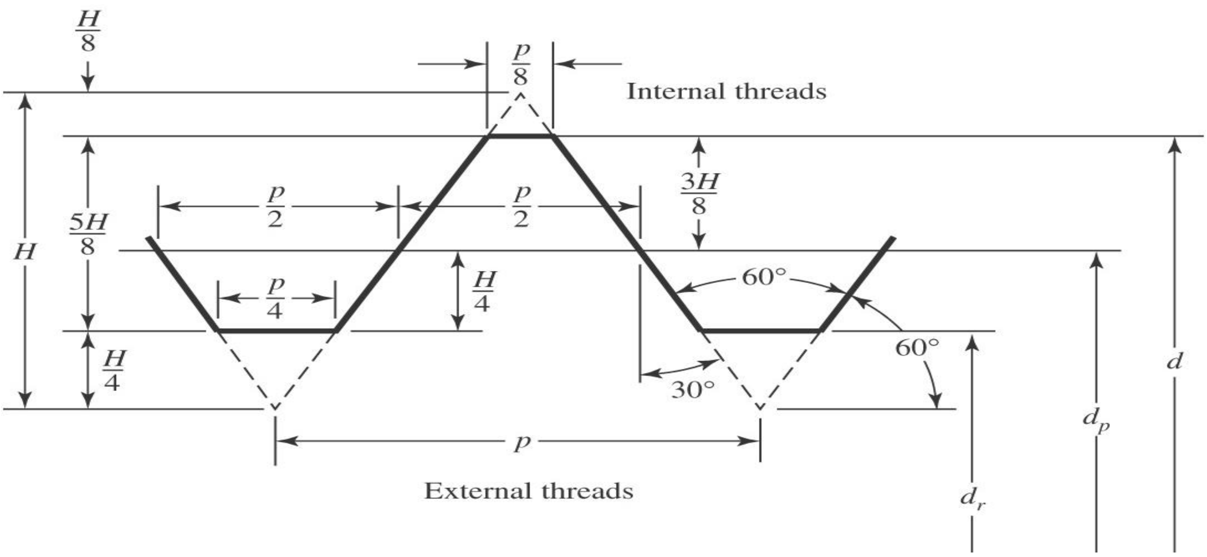
Tensile stress area: It is observed that an unthreaded rod having a diameter equal to the mean of the pitch diameter and minor diameter of a threaded rod has the same tensile strength as that of the threaded rod. The cross section of this unthreaded rod is called tensile stress area. This is the area which is used for calculating the tensile strength of the bolt.

Main forms of screw threads: Triangular or V threads and Square threads are widely used. Others are modification of the above.

V threads: Widely used in threaded fasteners due to following reasons.

- Offer higher friction and lesser the possibility of loosening
- Offer higher strength
- Easy to manufacture

Figure below shows the profile of an ISO Metric thread. Profile is an equilateral triangle with thread angle of 60 degrees. Base of this triangle is equal to the pitch. Crests and roots are flattened or rounded to reduce stress concentration.



Coarse and fine threads: Coarse threads are used in applications free from vibrations: Fine threads are used under dynamic loads.

Specifications of ISO Metric Threads: They are specified as follows.

By the letter M followed by the value of nominal diameter and pitch in mm and separated by the symbol X, for fine series. Example: M12 X 1.25 represents a thread (Fine series) with a nominal diameter of 12mm and pitch of 1.25 mm. Coarse series threads are simply specified as M12.

Basic dimensions for design profiles of ISO Metric Threads are available in Design data hand books in which dimensions like nominal diameter, root diameter, pitch, root area etc., are given.

If standards are not available, following relations can be used.

- For coarse thread : Root diameter = 0.84 x Nominal diameter
- For fine thread : Root diameter = 0.88 x Nominal diameter

Right hand and left hand threads: Right hand threads are always used unless there is a requirement for left hand thread

Methods of manufacture: Threads can be cut by Automatic thread cutting machines. They can also be produced by thread rolling. Thread rolling is superior to thread cutting because of the following reasons.

- Residual compressive stresses increase fatigue and impact resistance
- Lesser waste of material
- Improved grain structure

Materials: Several materials are used for threaded fasteners. Most fasteners are made from free cutting steels and plain carbon steels like 40C8 or 45C8 or alloy steels like 40 Ni 14, 35Mn5Mo5, 40Cr4Mo2 etc, Selected based on Strength, weight, corrosion resistance and cost. Stain less steel for corrosion resistance. Aluminium, brass, copper, titanium are also used. Coatings are used for corrosion protection and to reduce thread friction and wear.

Design of threaded fasteners under static loads

The usual design criterion given below, employed for designing machine elements under static loads, is used here also.

It states that:

- At Failure: Induced Stresses > Ultimate Strength (US) or Yield Strength (YS).
- For Safe Design: Induced Stresses < Ultimate Strength or Yield Strength.
- When Factor of safety (FoS) is introduced: Induced Stresses < Allowable strength.

- Allowable strength or Design strength or Limiting strength = $\frac{US}{FoS}$ or $\frac{YS}{FoS}$
- **Induced Stresses or working stresses < Allowable strength or Design strength or Limiting strength**

Stresses in bolts

Nomenclature: Following nomenclature is used in the following sections.

σ_{all}: Allowable Tensile strength

τ_{all}: Allowable shear strength

d: Nominal diameter

d_1: Core or root diameter

F_i = Pre-load due to initial tightening

F_{safe} = Safe load or bolt load

A_C = Cross sectional area at root

A_t= Tensile stress area

σ_i = Tensile stress due to initial tightening

In threaded joints following stresses are induced.

- Stresses due to initial tightening (Pre load)
- Stresses due to externally applied loads
- Stresses due to combination of the above.

Necessity for initial tightening:

In any threaded assembly the threaded elements are initially tightened before they take the service loads. Initial tightening is necessary for the following reasons.

- To ensure that parts never separate and held together by sufficient clamping force
- To maintain friction (No sliding to shear forces)
- to ensure even distribution of loads
- To prevent warpage of parts
- To prevent bolts from loosening

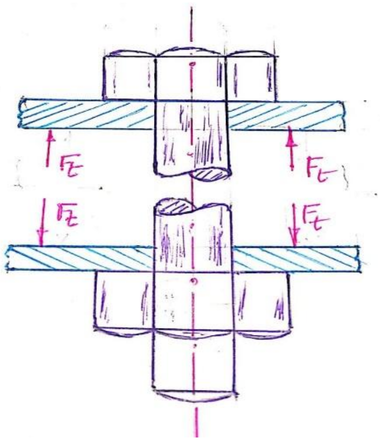
Stresses due to initial tightening:

Due to initial tightening following stresses are induced. The net stress due to preload is the resultant of all. Following gives the details of the stresses.

- Tensile stresses due to stretching of the bolt
- Torsional stresses due to friction between threads
- Shear stresses across threads
- Crushing stresses on threads
- Bending stresses if the head or nut are not perfectly normal to the bolt axis

The above stresses cannot be accurately determined.

Tensile stresses due to stretching of the bolt: As seen in the figure given below when the nut is screwed on to the bolt it tends to bring the plates closer inducing compressive loads on the plates. But the reactive forces tend to separate the plates as shown which in turn induces tensile loads resulting stretching of the bolts. Ths tensile stresses are induced.



Torsional stress due to friction between threads: When torque is applied on the screw rod either to tighten the nut or to drive in a screw torsional shear stresses are induced in the body of the screw. It is given by:

- $\tau_{\text{ind}} = \dfrac{16\,T}{\pi d_1^3}$

Shear stress across threads: When torque is applied on the screw rod during initial tightening, it results in axial load acting on the screw threads. Consequently shear stresses are induced in the threads which is given by:

- $\boldsymbol{\tau_{\text{ind}} = \dfrac{Shear\ Load}{Shear\ Area} = \dfrac{F}{b.i.\pi d_1}}$

F= axial load, b = width of the thread at the root, i – Number of threads d_1: Core or root diameter

Crushing stress on threads: When torque is applied on the screw rod during initial tightening, it results in axial load acting on the screw threads which induce crushing stresses.

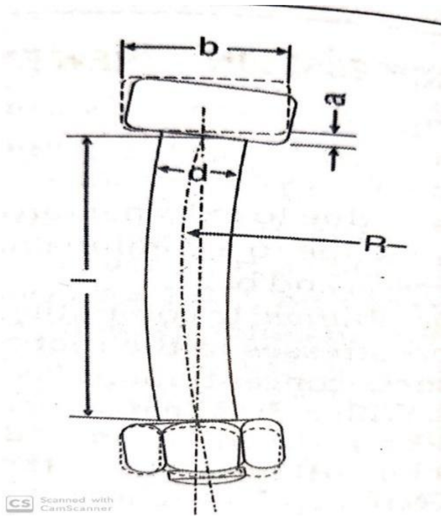
It is computed by; $\sigma_c = \frac{Crushing\ Load}{Crushing\ Area} = \frac{4F}{\pi\, i\,(d^2 - d_1^2)}$

F= axial load, d = nominal diameter, d_1: Core or root diameter i – number of threads

Bending stresses

During fastening if the head or nut are not perfectly normal to the bolt axis bending stresses are induced.

- $\sigma_b = \frac{aE}{2l}$



E = Modulus of elasticity

l = Length of shank of bolt

a = difference in between extreme corners of nut or head

Empirical relation for preload stresses[AS1]

Since the initial stresses cannot be accurately determined bolts are designed based on a direct tensile load with high factor of safety, based on experimental results at Cornell University, are as follows.

a. When the parts have metal to metal contact initial preload is taken as a tensile load given by:

F_i = 2804.69d -----------(5.1)

b. When flexible seal or gasket is used

F_i = 1402.34d -----------(5.2)

Where, d is the nominal diameter of the bolt in mm and F_i is the initial load in Newtons

If the bolts are not initially stressed

Safe axial load:

$F_{safe} = \sigma_{all} \times A_t$ ---- (5.3)

Stress due to initial tightening

$\sigma_i = \frac{F_i}{A_C}$ ---- (5.4)

Where, A_C = Cross sectional area at root, A_t= Tensile stress area

Maximum safe holding force of a set screw

$F = 6.344\,(d)^{2.31}$ ---- (5.5)

Relation between torque applied to the nut and axial tensile load in a bolt

$T = 0.2\,F_i d$ ---- (5.6)

Numerical Examples

1. M 30 x 2 bolts are manufactured using steel with yield strength of 464 MPa. Assuming that the bolts are not initially stressed, compare the load capacity (safe load) when they are designed on the basis of a) factor of safety of 4 and b) factor of safety of 5.

Solution

a) Factor of safety of 4

Given: M 30 x 2 bolts:
Nominal diameter d = 30 mm; pitch p = 2mm

Safe load or load capacity:

$F_{safe} = \sigma_{all} \times A_t$

A_t = Tensile area = 621 mm^2 (From standard Design data handbooks)

$$\sigma_{all} = \frac{464}{4} = 116MPa$$

F = 621 x 116 = 72036 N = 7.2 kN

b) Factor of safety of 5

$$\sigma_{all} = \frac{464}{5} = 92.8MPa$$

F = 621 x 92.8 = 57628 N = 5.7 kN

2. Determine stress induced in a M24 x 3 bolt due to pre load.

Given: M 24 x 3 bolt:

Nominal diameter = 24 mm, pitch = 3 mm

Pre – load stresses

$$\sigma_i = \frac{pre - load}{core\ area} = \frac{4F_i}{\pi d_1^2}$$

F_i = 2804.69 d = 2804.69 (24) = 67312.56 N

d_1: Core or root diameter = 23.319392 mm (From standard Design Data Handbooks: DDHB)

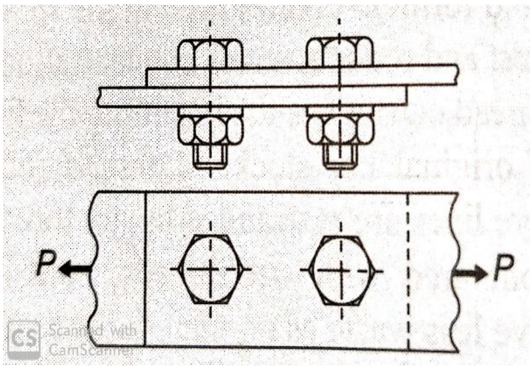
σ_i = 158.7 MPa

3. Two plates are connected by means of two bolts as shown in figure. Determine the size of the bolts on the basis of a factor of safety of 4, if the bolts are made of heat treated Nickel steel with tensile yield strength of 714 MPa. Given P = 8 kN

Solution: Bolts fail by shear of shank

For safe design $\tau_{ind} \leq \tau_{all}$

τ_{all} = Allowable shear strength



$$\tau_{all} = \frac{\sigma_{yp} tension}{2(FoS)} = \frac{714}{2(4)} = 89.2\ MPa$$

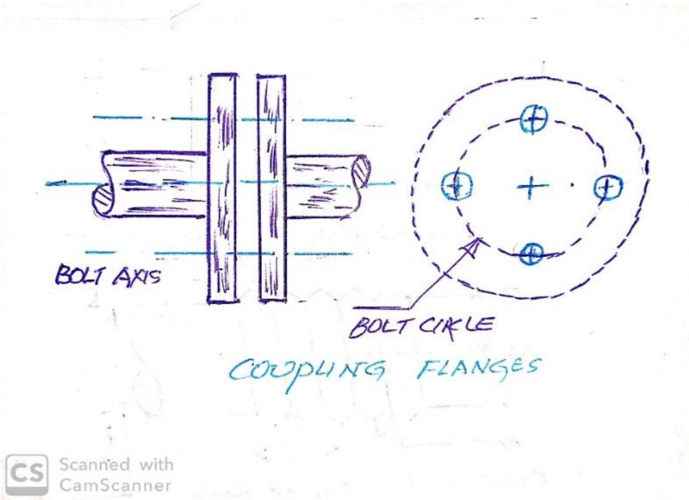
$$\tau_{ind} = \frac{P}{\frac{2(\pi d^2)}{4}} = \frac{2P}{\pi d^2} = 89.2 MPa$$ d: nominal / shank diameter

$$d^2 = \frac{2 \times 8 \times 10^3}{\pi(89.2)} = 57.1$$ d = 7.55 mm

Standard size: M8 bolts can be employed

4. In a flanged coupling the flanges are to be connected by 4 bolts, on a pitch circle of diameter of 100mm. The coupling transmits 3000Nm of torque. Determine the size of the bolts if the allowable shear strength is 40 MPa.

Solution: Assumption: Pre-load due to initial tightening is ignored.



Bolts are subjected to shear load. Shank of the bolts are sheared off.

Shank diameter

$\tau_{ind} \leq \tau_{all}$

$\tau_{all} = 40\ MPa\ (given)$

$\tau_{ind} = \frac{F_S}{\left(\frac{\pi d^2}{4}\right)}$ F_S = Shear load or each bolt

$$F_S = \frac{Torque}{i(Bolt\ circle\ radius)} = \frac{3000 \times 10^3}{4(50)} = 15 \times 10^3 N$$

$$\therefore \tau_{ind} = \frac{15 \times 10^3 \times 4}{\pi . d^2} = \frac{19108.2}{d^2} = 40$$

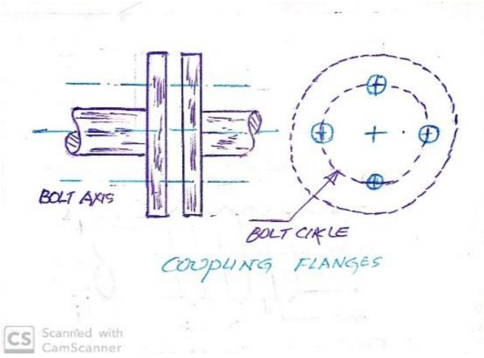
$\therefore d^2 = 477.55$ d = 21.5 mm

Four M 22 bolts are recommended.

5. Determine the power that can be transmitted by a CI flange coupling having four M4 x 0.7 bolts positioned on a pitch circle of diameter 150 mm.

The shaft runs at 1000 rpm. Design strength for the bolt material is 50 MPa.

Solution:



Shear force that can be transmitted by each bolt

$F_S = A \times \tau_{all}$ (Based on Nominal diameter)

$$= \left(\frac{\pi d^2}{4}\right) \times \tau_{all}$$

$$= \frac{\pi (4)^2 \times \tau_{all}}{4}$$

$= 4\,\pi \times 50$

$F_S = 200\,\pi N$

Torque = $F_S * R * i$

M4 bolts

d = 4 mm

P = 0.7 mm

τ_{all} = 50 MPA

N = 1000 rpm

R = 75 mm

T = 200 π x 75 π 4 = 10048 Nmm

$$Power = P = \frac{TN}{9.55 \times 10^6} = \frac{10048 \times 1000}{9.55 \times 10^6}$$

P = 10 kW

6. A machine weighing 10 kN is provided with an eye bolt for lifting and transporting purposes. The bolt is made of 30C8 steel with yield strength of 400 MPa. Determine the size of the bolt based on a factor of safety of 5. Neglect pre load.

Solution:
F = 10 kN, Steel C38

σ_{yp} = 400 MPa, FoS = 5

$$\sigma_{all} = \frac{400}{5} = 80\ MPa$$

Core diameter of the bolt:

$$\sigma_{ind} \leq \sigma_{all}$$

$$\sigma_{ind} = \frac{4F}{\pi d_1^2} = \frac{4 \times 10^3 \times 10}{\pi d_1^2} = 80$$

$$d_1^2 = \frac{4 \times 10 \times 10^3}{3.14 \times 80} = 159.2$$

d_1 = 12.6 mm, Corresponding to this standard bolt M16 x 2 is sufficient, for which: d = 16 mm, d_1 = 13.54 mm

Section 2

Threaded Joints under External loads:

Threaded joints are often required to take external loads. Entire load is not taken up by the screwed fastener (Say, Bolt and nut) alone. A part of the load is taken up by the connected parts. To understand how the load is shared between them it is required to know their elastic behaviour. When the nut is tightened initially, the bolt is subjected to tension and thus elongates whereas the parts connected are subjected to compression. When the parts are clamped together and put in to service and subjected to external load the bolt is further elongated whereas the clamped parts are relieved of compressed load. The amount by which the load is reduced depends on the stiffness of the bolt and connected parts. Two types of joints can be considered based on functional requirements.

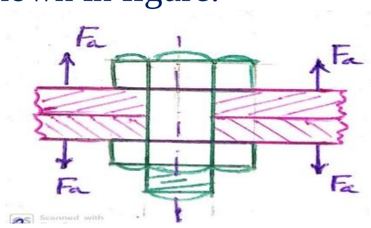
Joints for clamping: Functional requirement is **only to hold the parts together** against forces which tend to pull or slide them apart

Joints for cylinder heads: Functional requirement is to hold the parts together and also making the **joint leak proof**. Gaskets are used.

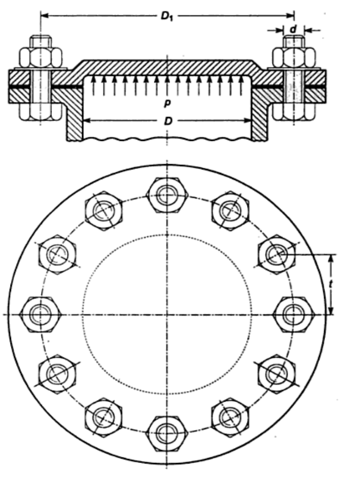
Resultant load depends on

- Pre-load
- External load
- Relative elastic yielding of the bolt and connected member

a) **Joints for Clamping:** External load acts in addition to Pre-load. No gaskets are used as shown in figure.



b) **Joints for pressure vessels and cylinder heads:** Functional requirement is to hold the parts together and also making the joint leak proof. Gaskets are used as shown Eg; Cylinder Head. Bolts are subjected to direct tensile stresses due to pressure of the fluid inside the cylinder. The gas forces tend to push the cylinder head outwards which is resisted by the bolts.



Resultant load: The resultant load is given by the following expression.

$F_R = F_i + K\,F_a$ ---- (5.7)

F_R = Resultant load
F_a = externally applied load
K = Relative stiffness of the bolts and connected parts
F_i = Pre-load

Values of K are available in Standard DDHB and also they can be computed Sample values are given below.

Relative stiffness of the bolts and connected parts, K

Type of Joint	**K**
Soft packing with studs	1.00
Soft packing with through bolts	0.75
Asbestos	0.60
Soft Copper gasket with through bolts	0.50
Hard Copper gasket with through bolts	0.25
Metal to metal Joints with through bolts	0.00

Two specific terms are used in the above table viz..soft gasket (Packing) and hard gasket (Packing), When the stiffness of the parts held together by the bolts is too small compared to stiffness of the bolt it is termed as soft gasket. In this case, the resultant load is sum of the initial tension and external load. If it is too large, then it is termed as hard gasket. In this case, the resultant load is equal to the initial tension.

When the stiffness values are known K can be computed as follows.

<u>When there is no gasket:</u>

$$K = \frac{K_b}{K_b + K_m}$$

K_b = stiffness of bolt
K_m = stiffness of the part

<u>When a gasket is used:</u>

K = Relative stiffness of the bolts and connected parts (Gasket)

$$K = \left[\frac{\frac{E_b A_b}{L}}{\frac{E_b A_b}{L} + \frac{E_g A_g}{l_g}} \right] \qquad ----(5.8)$$

Where,
E_b = Modulus elasticity of bolt
E_g = Modulus elasticity of gasket
A_b = Cross-sectional area of bolt
A_g = Cross-sectional area of gasket
L = Length of the bolt
L_g = Length of the gasket (thickness)

For safe Design, in both the above cases:

$$\sigma_{ind} \le \sigma_{all}$$

$$\sigma_{ind} = \frac{4F_R}{\pi d_1^2}$$

$$F_R = F_i + K\,F_a \qquad \text{--- (5.7)}$$

Design of cylinder head bolts

For safe design:

$$\sigma_{ind} \le \sigma_{all}$$

Resultant load on the cylinder head due to gas load and initial tension = Tensile resistance of bolts

$$F_R = F_i + k\,F_{gas}$$

F_{gas} = Cross sectional area of the cylinder x maximum pressure = external load

$$= \frac{\pi}{4}(D^2) \times P_{max} \qquad D = effective\ diameter\ of\ the\ cylinder$$

Tensile resistance of bolts = Root area x No. of bolts x Allowable tensile strength

$$= \frac{\pi d_1^2}{4} \times i \times \sigma_{all}$$

Equating the above two equations the core diameter of the bolt can be determined if the number of bolts are known and vice versa.

Numerical Examples

1. **Recommend suitable ISO metric threaded bolts for a joint in which initial pre-load on the bolt is 4kN and the load on the joint is 10kN. The ratio of member stiffness to the bolt stiffness is 3. The bolt with coarse thread is made of plain carbon steel whose yield strength is 324 MPa. Use a factor of safety of 3.**

$$\sigma_{ind} \leq \sigma_{all}$$

$$\sigma_{all} = \frac{\sigma_{yp}}{PoS} = \frac{324}{3} = 108\ MPa$$

$$\sigma_{ind} = \frac{4F_R}{\pi d_1^2}$$

$F_R = F_i + K\ F_a$

$F_i = 4 \text{ x } 10^3$ N

$F_a = 10 \text{ x } 10^3$ N

$$K = \frac{K_b}{K_b + K_m}$$

Given: $K_m = 3K_b$

$\therefore K = 0.25$

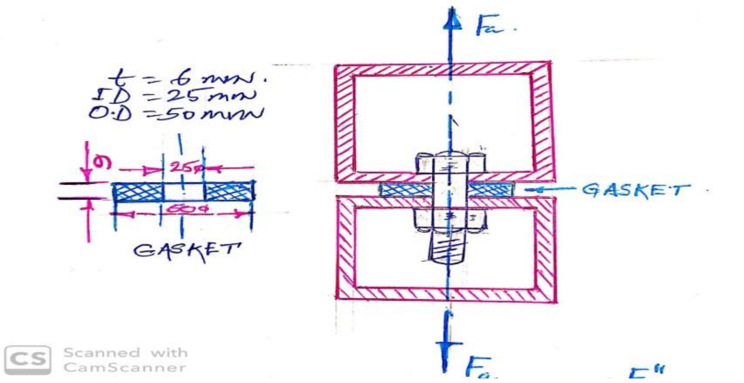
F_R $(4 \text{ x } 10^3) + (0.25 \text{ x } 10 \text{ x } 10^3) = 6.5 \text{ x } 10^3$ N

$$\sigma_{ind} = \frac{4 \times 6.5 \times 10^3}{\pi d_1^2} = 108\text{MPa}$$

$d_1 = 8.75$ mm

Standard sized coarse thread:
M12 x 1.75 for which d_1 = 9.852 mm is recommended.

2. **In a bolted assembly as shown M20 x 2.5 annealed medium carbon steel bolts are used. Is it safe to use bolts if the nut is tightened initially by a spanner? Modulus elasticity of the bolt is 206 x 10^3 MPa and that of the gasket is 118 x 10^3 MPa. Load on the assembly is 10 kN. Yield strength of bolt material is 316 MPa.**



Solution

To check safe or not

$$\sigma_{ind} = \left(\frac{4F_R}{\pi d_1^2}\right)$$

For M 20 x 2.5 bolts: d = 20 mm, d_1 = 16.932827 mm

$F_R = F_i + KF_a$ $\quad\quad$ F_a = 10 kN

F_i = 2804.69d = 2804.69 x 20

F_i = 56093.8 N

$$K = \left[\frac{\frac{E_b A_b}{L}}{\frac{E_b A_b}{L} + \frac{E_g A_g}{l_g}}\right]$$

E_b = 206 x 10^3 MPa
E_g = 118 x 10^3 MPa
l_g = 6 mm

L = 30 mm

$A_b = \frac{\pi}{4} \times (16.932887)^2$

= 225.11 mm^2

$A_g = \frac{\pi}{4} \times (50^2 - 25^2)$ = 1471.87 mm^2

K = 0.055

F_R = 56093.8 + 0.055 (10 x 10^3)

F_R = 56643.8 N

$$\sigma_{ind} = \frac{56643.8}{225.11} = 251.6\ MPa$$

$$(FoS)_{realised} = \frac{\sigma_{yp}}{\sigma_{ind}} = \frac{316}{251.6} = 1.25$$

FoS = 1.25 > 1 ∴ Marginally safe.

3. A pressure vessel of inner diameter 200 mm is subjected to an internal pressure of 10 MPa. A cover plate is fastened to the flanged end by means of 6 bolts. Selecting carbon steel with 320 MPa as yield strength determine the size of the bolts required considering initial tension in the following cases.

i. **Metal to metal joint**
ii. **Joint with a copper gasket**

Solution:

D_i = 200 mm: P_i = 10 MPa: σ_{yp} = 320 MPa:

FoS (not given) = FoS = 2.5 (assumed)

$$\sigma_{all} = \frac{320}{2.5} = 128\ MPa$$

Case (1): Metal to Metal Joint

$$\sigma_{ind} \le \sigma_{all}$$

$$\frac{4F_r}{\pi d_1^2} \le \sigma_{all}$$

$F_r = F_i + K\,F_a$ K = 0 (for metal to metal joint)

$$F_i = 2804.69\left(\frac{d_1}{0.84}\right);\quad Let\ d = \left(\frac{d_1}{0.84}\right)$$

$F_r = 3338.9\ d_1$

$$\frac{4\ X\ 3338.9\ \mathrm{d1}}{\pi d_1^2} \leq 108$$

$d_1 = 33.2$ mm,   Bolts M36 x 2, with $d_1 = 33.54$ mm are recommended.

Case (2): When copper gasket is used

$$\frac{4F_r}{\pi d_1^2} \leq \sigma_{all}$$

$F_R = F_i + K\,F_a$ K = 0.5 (for copper gasket)

$$F_i = 1402.34\left(\frac{d_1}{0.84}\right);\quad Let\ d = \left(\frac{d_1}{0.84}\right)$$

$F_R = 1669.4\ d_1$

$$Steam\ load, F_a = \frac{\pi D_1^2}{4} * P_i = \frac{\pi * 200^2}{4} * 10 = 313.8kN$$

Steam load per bolt = $\frac{F_a}{6}$ = 52.3kN

$F_R = F_i + K\,F_a$

$F_R = 1669.4\ d_1 + 0.5 * 52.3 \times 10^3$

$\frac{4F_R}{\pi d_1^2} = 128$ Results in a Quadratic equation,   Solve for d_1,

$d_1 = 26.5$ mm

Bolts M30 x 3.5, with $d_1 = 27.7$ mm are recommended.

4. **An engine cylinder has an effective diameter of 350mm and the maximum gas pressure acting on the cylinder cover is 1.25 MPa. Determine the number and size of the studs required to fix the cylinder cover. The material of the stud has an allowable tensile strength of 70MPa. Assume that the preload due to initial tightening is 25% larger than the gas load.**

Solution

D_i = 350 mm: P_i = 1.25 MPa: σ_{tall} = 70 MPa

$$Gas\ load, F_a = \frac{\pi D_1^2}{4} * P_i = \frac{\pi * 350^2}{4} * 1.25 = 120.26kN$$

$F_r = F_i + K\ F_a$

$F_i = 1.25\ F_a$ (Given)
$= 1.25 * 120.26 = 150.3$ kN

Let a copper gasket be used

So, K = 0.5

$F_r = 150.3 + 0.5 * 120.26 = 210.43$ kN

Size and number of bolts

a. Assuming size, number can be determined
b. Assuming number, size can be determined

Given: σ_{tall} = 70 MPa

Total resisting area to be provided by the bolts

$$\frac{F_r}{\sigma_{tall}} = \frac{210.43 \times 10^3}{70} = 3006mm^2$$

This should be provided by “i” number of bolts’ say,

Let us try M24 x 2 bolts, with d_1 = 21.546 mm

$$A_c = \frac{\pi * 21.546^2}{4} = 325.9 \approx 326mm^2$$

$$i = \frac{3006}{326} = 9.2 \approx 10\ bolts$$

Therefore: 10, M 24 x 2 bolts are recommended.

Also try by assuming number, to determine size.

5. (Exercise) The piston rod of a hydraulic cylinder exerts pressure of 10 MPa inside the cylinder. The diameter of the cylinder is 40 mm. The flange thickness is 10mm and a cast iron cover plate of thickness 10 mm is fixed to the cylinder by means of 4, M12 bolts having a minor diameter of 9.853mm and a zinc gasket of 3 mm thickness. Bolts are made of steel for which the yield strength is 400 MPa. Determine the factor of safety if E for steel is 207 GPa, E for CI is 100 GPa, E for Zinc is 90 GPa.

Section 3

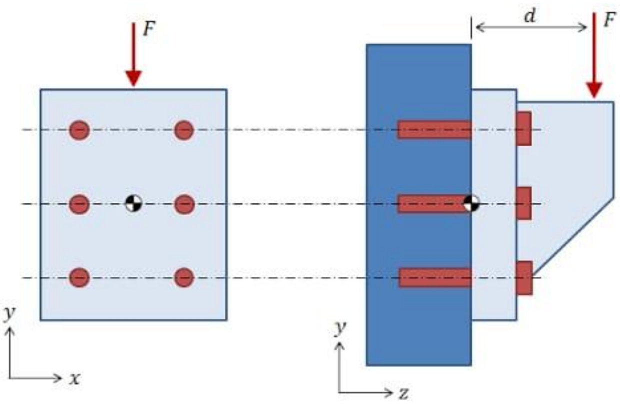
Eccentrically loaded bolted joints

Introduction: In many engineering applications bolted joints are subjected to eccentric loads. Example: wall bracket, wall hangers. Pillar cranes, J Hooks etc,. A few examples are as shown in figure given below. Examples: As seen in these figures there are three different ways in which eccentrically loaded bolted joints can be loaded.

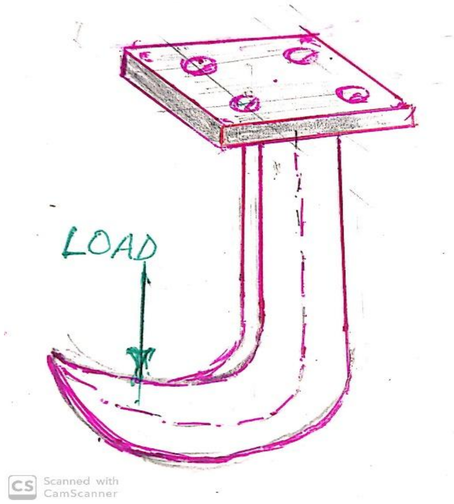
Three ways of eccentric loading:

- Load axis parallel to the bolt axis
- Load axis perpendicular to the bolt axis
- Loads acting in the plane containing the bolts

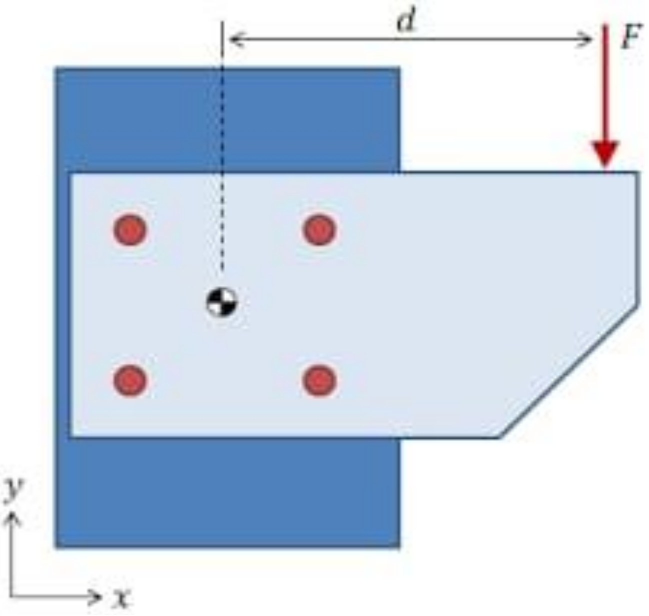
Load axis perpendicular to the bolt axis



Load axis parallel to the bolt axis



Loads acting in the plane containing the bolts



Assumptions:

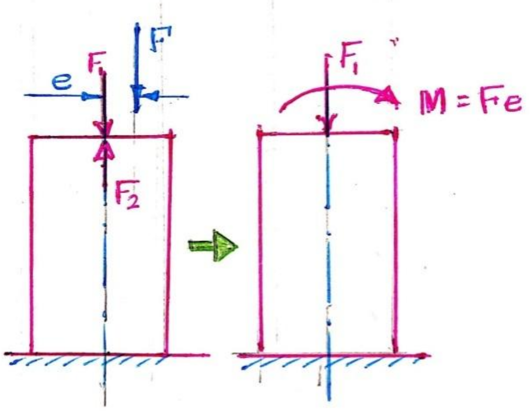
In the analysis of eccentrically loaded bolted joints following assumptions are made.

- ➢ All bolts are identical and design is based on heavily loaded bolt
- ➢ Structure to which the bracket is fixed and the bracket are rigid
- ➢ Bolts are fitted in reamed and ground holes so as to avoid bending
- ➢ Bolts are not preloaded.
- ➢ Stress concentration effect ignored

Eccentrically loaded Members (General case)

Figure shows a member subjected to an eccentric load. As seen an eccentric load can be thought of as a direct load and moment.

An Eccentric load = Direct load + Moment

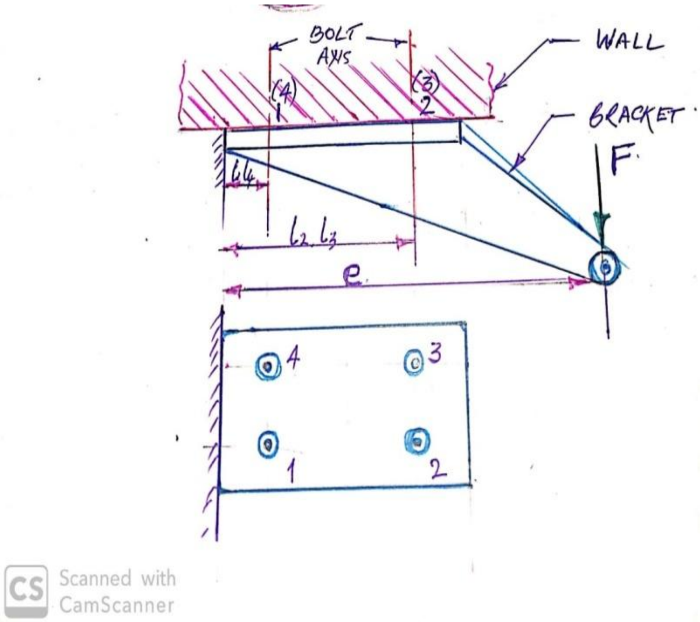


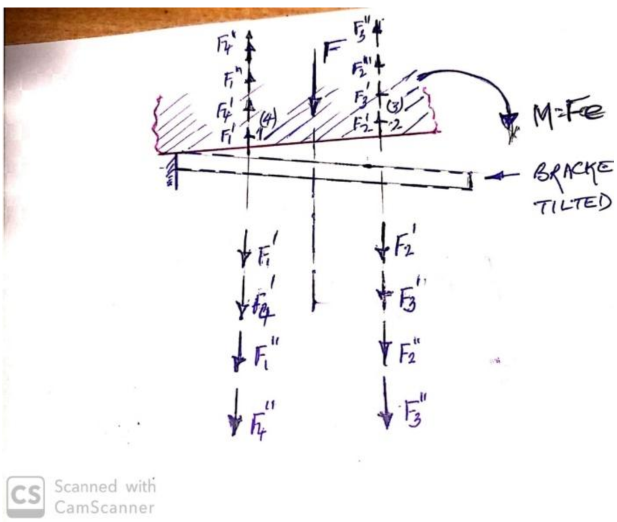
Effect of Direct load and Moment need to be considered in designing eccentrically loaded bolted joints

Load analysis when Load axis is parallel to the bolt axis

Figure shows a bracket fastened to the roof. Applied load tends to turn the base of the bracket about the tilting edge. Each bolt is stretched. Bolts are stressed to different degree which depends on their distance from the tilting edge which is shown by hatched line.

Let F is the load acting on the bracket at a distance **e** from the tilting edge.
e is eccentricity.





Primary load or direct load (F'): It is due to the direct load. is equal at all bolts. It is tensile.

$$F' = \frac{F}{i}\ (equal\ at\ all\ bolts)$$

$$F_1' = F_2' = F_3' = F_4' = \cdots = \frac{F}{4}$$

Secondary load (F”): It is also a tensile load and is due to the moment. It is given by:

$$F" = \frac{F\ e\ l}{\sum l^2}\ ;$$

$l = l_1, l_2, l_3,.$ = distance of individual bolts from XX ie., tilting edge.

From the above expression it can be shown that the secondary load is proportional to the distance l. Therefore, secondary load is maximum at those bolts which are farthest from the tilting edge.

Since all the bolts are identical and design is based on heavily loaded bolt, there is no need to determine the load at each bolt. Primary load being same at all bolts heavily loaded bolt is that at which secondary load is maximum. Thus the heavily loaded bolt can be identified. Both primary and secondary loads are as shown in the figure.

Design is based on resultant load (F_R).

$F_R = F' + F"$, at the heavily loaded bolt.

For safe design

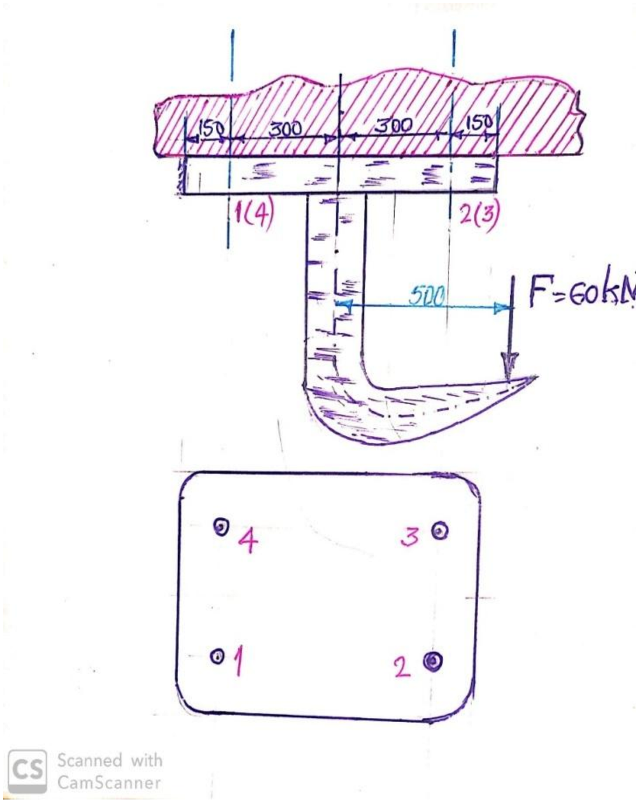
$$\sigma_{ind} \leq \sigma_{all}$$

$$\frac{4F_R}{\pi d_1^2} \leq \sigma_{all}$$

Size of the heavily loaded bolt is determined from the above relation.
Other bolts (loaded to a lesser extent) are also of the same size.

Numerical Example

1. Determine the size of the bolts required to fasten a wall hanger as shown in figure. The bolts are made of C-30 steel use a factor of safety of 4.



Bolt axis and load axis are parallel. Primary load (F') is tensile, secondary load (F'') is also tensile. Therefore the resultant load is:

$F_R = F' + F''$ at the heavily loaded bolt.

$$F' = \frac{F}{4} = \frac{60 * 10^3}{4} = 15\ kN$$

$F_1' = F_2' = F_2' = F_4' = 15$ kN

$F'' = \frac{F\,e\,L}{\sum L^2}$: will be maximum at bolt 2 and 3

$$F_2'' = F_3'' = \frac{60 * 10^3 * 500 * 750}{2(150)^2 + 2(750)^2} = 19.23\ kN$$

Resultant load at bolt 2 and 3

$F_{R2} = F_{R3} = F_2' + F_2'' = F_3' + F_3'' = 15 + 19.23 = 34.23$ kN;

$F_{R2} = F_{R3} = 34.23$ kN;
For safe design

$$\sigma_{ind} \leq \sigma_{all} \qquad\qquad \frac{4F_R}{\pi d_1^2} \leq 79$$

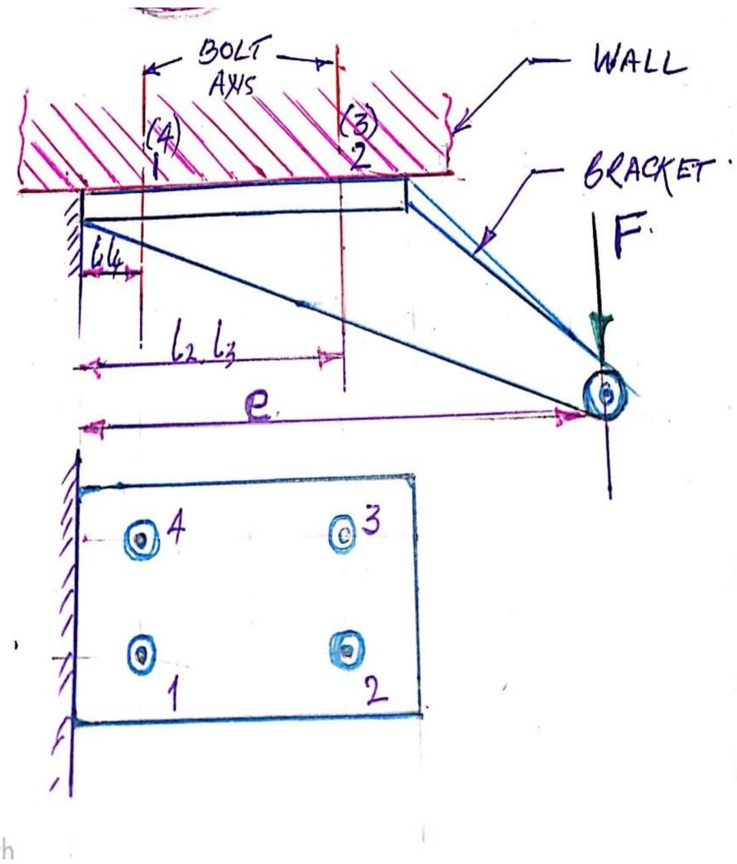
$$\frac{4 * 34.23 * 10^3}{\pi * 79} \leq d_1^2$$

Therefore $d_1^2 = 551.96$ and $d_1 = 23.5$ mm

4, M30 x 3.5 bolts are recommended for which d = 30 mm, d_1 = 27.62 mm

2. A bracket as shown is used to support a load P. Given l_1 = 30mm, l_2 = 300mm, e= 500mm, Maximum allowable stress in the bolt is 75MPa. Size of the bolts is M 27X3. Determine the load P.

Solution:



To determine the load P

$$\sigma_{ind} \le \sigma_{all}$$

$$\frac{4F_R}{\pi d_1^2} \le 75 - (A)$$

For M 27 x 3 bolts,

d = 27 mm, d_1 = 23.31 mm

$F_R = F' + F''$ at the heavily loaded bolt.

Bolt 2 and 3 are critically loaded bolts.

Primary load,

$$F' = \frac{P}{4}$$

$$F" = \frac{P\ e\ l_2}{\sum L^2} = \frac{P * 500 * 300}{2(30)^2 + 2(300)^2} = 0.825\ P$$

$$F_R = \frac{P}{4} + 0.825\ P = 1.89\ P$$

$$\frac{4 * 0.825\ P}{\pi * 23.31^2} = 75$$

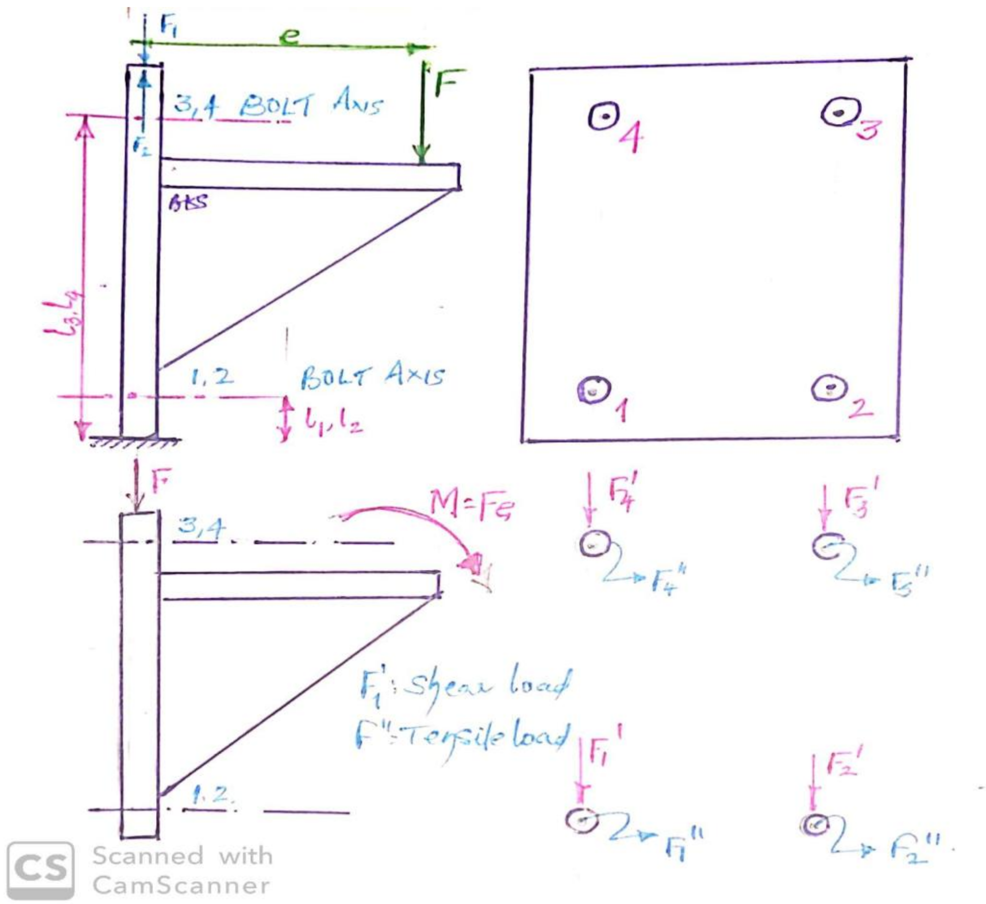
$$P = \frac{75 * \pi 23.31^2}{4 * 0.825} = 38.775\ kN$$

P = 38.775 kN

Load analysis when Load axis is perpendicular to the bolt axis

Figure shows a bracket that can be fastened to a structure by means of bolts. Applied load tends to turn the bracket about the tilting edge. Bolts are stressed to different degree which depends on their distance from the tilting edge.

Let F is the load acting on the bracket at a distance **e** , e is the eccentricity.



Load axis perpendicular to the bolt axis. Applied load tends to turn the bracket about the titling edge which is shown by hatched line.

- Primary load is shear load $(F') = \frac{Eccentric\ load}{No.of\ bolts}$
- $F' = \frac{F}{i}\ (Equal\ at\ all\ bolts)$
- $F_1' = F_2' = F_3' = F_4' = \cdots \ldots \ldots . = \frac{F}{i}$

Secondary load due to the moment is tensile load (F")

$$F" = \frac{F\ e\ l}{\sum l^2}$$

l = l_1, l_2, l_3, …. Distance of individual bolts from the tilting edge XX. Secondary load is maximum at those bolts which are farthest from the tilting edge. Both primary and secondary loads are as shown in the figure.

Design is based on resultant load. It is a case of combined loads. Hence the resultant load is expressed in terms of:

- Equivalent or Maximum normal load
- Equivalent or Maximum shear load

$$Maximum\ normal\ load, F_{Nmax} = \frac{F''}{2} + \sqrt{\left(\frac{F''}{2}\right)^2 + (F')^2}$$

$$Maximum\ shear\ load, = \sqrt{\left(\frac{F''^2}{2}\right) + (F')^2}$$

For safe design:

$$\sigma_{max} \leq \sigma_{all}$$

$$\frac{4F_{Nmax}}{\pi d_1^2} \leq \sigma_{all}$$

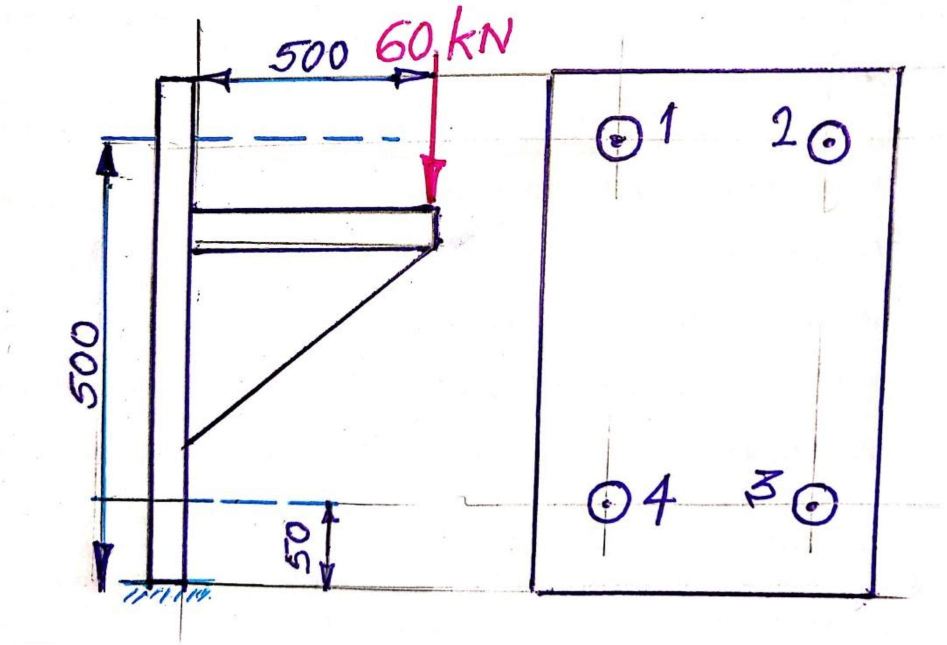
$$\tau_{max} \leq \tau_{all}$$

$$\frac{4F_{Smax}}{\pi d_1^2} \leq \tau_{all}$$

Higher of the two values to be adopted as the required diameter of the bolt.

Numerical Example

1. **Determine the size of the bolts required to fasten a bracket as shown in figure subjected to an eccentric load 60kN. Bolts have an allowable tensile strength of 80MPa.**



Primary load is shear load

$$(F'), F_1' = F_2' = F_3' = F_4' = \frac{60 * 10^3}{4} = 15 * 10^3 N$$

Secondary tensile load (F") will be maximum at 1 and 2

$$F_4" = \mathrm{F}_3" = \frac{F\ e\ l}{\sum L^2} = \frac{60 \times 10^3 \times 500 \times 550}{50^2 + 50^2 + 550^2 + 550^2} = 27.049\ kN$$

Since it is a case of combined load, maximum stresses are computed based on Equivalent or Maximum normal load (F_{Nmax}) and Equivalent or maximum shear load (F_{Smax})

For safe design:

$$\sigma_{max} \leq \sigma_{all}$$

$$\frac{4F_{Nmax}}{\pi d_1^2} \leq \sigma_{all}$$

$$\tau_{max} \leq \tau_{all}$$

$$\frac{4F_{Smax}}{\pi d_1^2} \leq \tau_{all}$$

$$Maximum\ normal\ load,\quad F_{Nmax} = \frac{F_3''}{2} + \sqrt{\left(\frac{F_3''}{2}\right)^2 + (F_3')^2}$$

$$= \frac{27}{2} + \sqrt{\left(\frac{27}{2}\right)^2 + (15)^2} = 34.86\text{kN}$$

$$Maximum\ shear\ load, F_{Smax} = \sqrt{\left(\frac{F_3''}{2}\right)^2 + (F_3')^2}$$

$$= \sqrt{\left(\frac{27}{2}\right)^2 + (15)^2} = 20.86\text{ kN}$$

Core diameter based on F_{Nmax}

$$\frac{4F_{Nmax}}{\pi d_1^2} \leq \sigma_{all}$$

$$\frac{4 * 34.86 * 10^3}{\pi d_1^2} = 80 \qquad and \quad d_1 = 23.56\ mm$$

Core diameter based on F_{Smax}

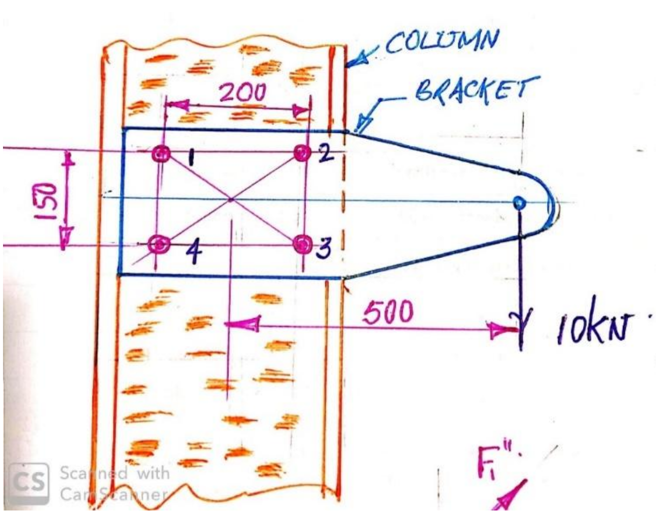
$$\frac{4F_{Smax}}{\pi d_1^2} \leq \sigma_{all}$$

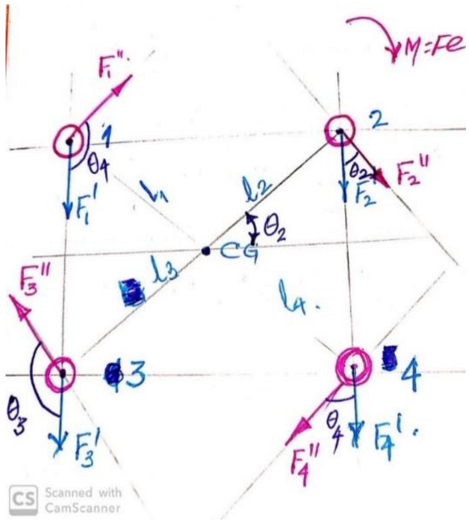
$$\frac{4 * 20.86 * 10^3}{\pi d_1^2} = 40 \qquad and \quad d_1 = 25.7\ mm$$

Adopt higher of the two values, Therefore, d_1 = 25.7mm
4 bolts M 27 X 2 with **d = 27mm and d_1 = 25.7 mm recommended**

Load analysis when the load axis is in the same plane containing the bolts

Figure shows a bracket that is fastened to a structure by means of bolts. Here, the applied load tends to turn the bracket about the centre of gravity of the bolt group. Bolts are stressed to different degree which depends on their distance from the centre of gravity. Let F is the load acting on the bracket at a distance **e** , e is the eccentricity. Procedure is illustrated as follows.





- Primary load is shear load $(F') = \frac{Eccentric\ load}{No.of\ bolts}$
- $F' = \frac{F}{i}\ (Equal\ at\ all\ bolts)$
- $F_1' = F_2' = F_3' = F_4' = \cdots \ldots \ldots . = \frac{F}{i}$

- Secondary loads $(F")$ are also shear loads acting at right angle to the line joining the center of gravity of the bolt group and individual bolt axis (bolt centre), so as to induce a couple of the same sense as that M = F e

$$F" = \frac{F\ e\ l}{\sum l^2}$$

- l is the distance of the individual bolt centre from the centre of gravity of the bolt group.

- Both primary and secondary shear load act in the same plane (coplanar force)

- θ: Angle between the primary and secondary shear load

Since they are coplanar force the resultant load of the each bolt will be

$$F_R = \sqrt{F'^2 + F"^2 + 2.F'.F".cos\theta}$$

- Design would be based on the most heavily load bolt.

- Heavily loaded bolt can be identified by using the following conditions:

  ✓ **F" α l**
  ✓ **F" is maximum at that bolt which is farthest from the centre of gravity of the bolt group**
  ✓ **F_R will be maximum at that bolt where θ is acute**.

For safe design

$$\tau_{max} \leq \tau_{all}$$

$$\frac{4F_{Smax}}{\pi d_1^2} \leq \tau_{all}$$

Numerical Examples

1. **In a steel bridge a bracket is connected to a column as shown in figure. The bracket carries a hoisting drum which lifts a load of 10 kN. Eccentricity is 500 mm. Suggest suitable bolts of ISO Metric threads if they are made of plain carbon steel based on a factor of 2.5. Yield strength in tension for bolts is 400MPa.**

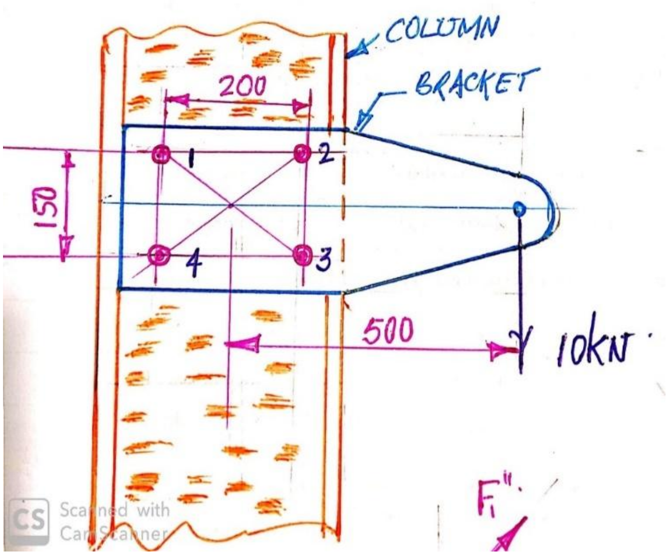
Solution:

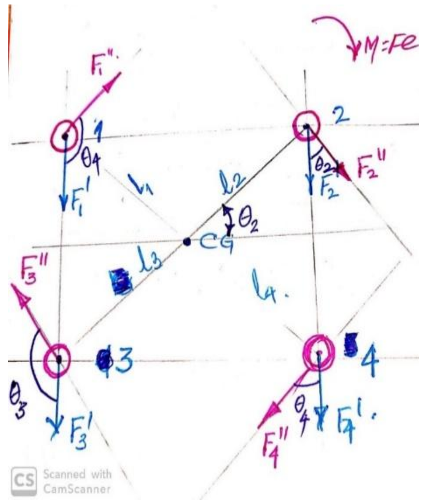
F = 50 kN, e = 500 mm

1. *Determine CG of the bolt group*

$$X = \frac{x1 + x2 + x3 + x4}{4} = \frac{0 + 200 + 0 + 200}{4} = 100\,mm$$

$$Y = \frac{y1 + y2 + y3 + y4}{4} = \frac{150 + 150 + 0 + 0}{4} = 75\,mm$$





2. Draw the loading diagram showing the primary and secondary forces as shown above

3. Determine l and θ

   Since bolts are placed symmetrically

$$l_1 = l_2 = l_3 = l_4 = \sqrt{75^2 + 100^2} = 125\,mm$$

   From the loading diagram

$$cos\theta 2 = \frac{100}{125} = 0.8$$

$$cos\theta 4 = \frac{100}{125} = 0.8$$

Hence the resultant load will be maximum at 2 and 4

4. F' = Primary load (direct load)

$$F_1' = F_2' = F_3' = F_4' = \frac{F}{i} = \frac{10000}{4} = 2500N$$

5. Secondary load F'' = shear load

$$F" = \frac{f\ e\ l_2}{\sum L^2} = \frac{1000 * 500 * 125}{4\ (125)^2} = 10000\ N$$

6. Secondary shear load being equal at all the bolts the most heavily loaded bolts are 2 and 4 where θ is acute

Maximum resultant load

$$F_{R2} = \sqrt{F2'^2 + F2"^2 + 2.F2'.F2".cos\theta 2} \quad = 12093N$$

7. Bolt size

   For safe design

$$\tau_{max} \leq \tau_{all}$$

$$\frac{4F_{R2}}{\pi d_1^2} \leq \tau_{all}$$

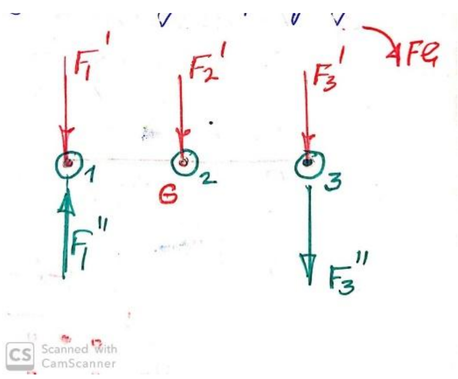
$$\frac{4 * 12093 * 10^3}{\pi d_1^2} = 80$$

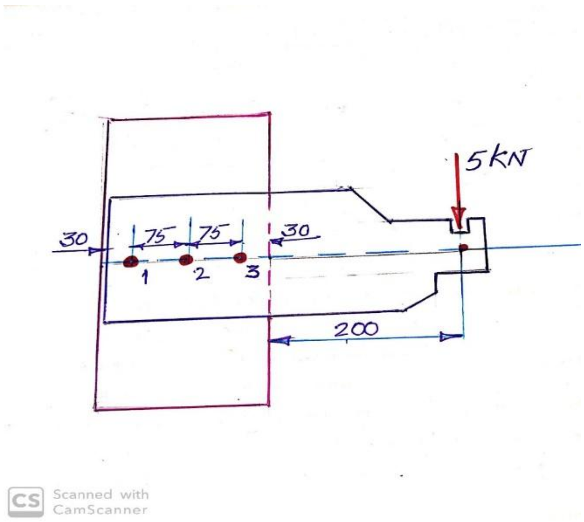
$$d_1^2 = 192.56\ mm \quad and \quad d_1 = 13.8\ mm$$

<u>Four bolts M 18 x 2.5 with d = 18 mm and d_1 = 14.93 mm are recommended.</u>

2. A steel plate is connected to a channel by means of three bolts in a food storage cabinet as shown. Bolts are made of plain carbon steel with tensile yield strength of 380 MPa. Determine the size of the bolts based on a factor of safety of 3.

Design data: Load= 5kN, e = 305 mm





Primary load is shear load

$$F' = \frac{F}{3} = \frac{5 * 10^3}{3} = 1.667\ kN$$

Secondary loads are also shear loads acting in the same plane as that of F'

$$F'' = \frac{F\ e\ l}{\sum L^2}$$

Bolts 1 and 3 are at the same distance from cg.

$$F_1 = F_3 = \frac{5 * 10^3 * 305 * 75}{(75)^2 + (75)^2} = 10.166\ kN$$

θ is zero at bolt 3

F_R is maximum at bolt 3

$F_R = F_3' + F_3''$, at the heavily loaded bolt

$F_R = F_3' + F_3'' = 1.667 + 10.166 = 11.83$ kN

For safe design

$$\tau_{max} \leq \tau_{all}$$

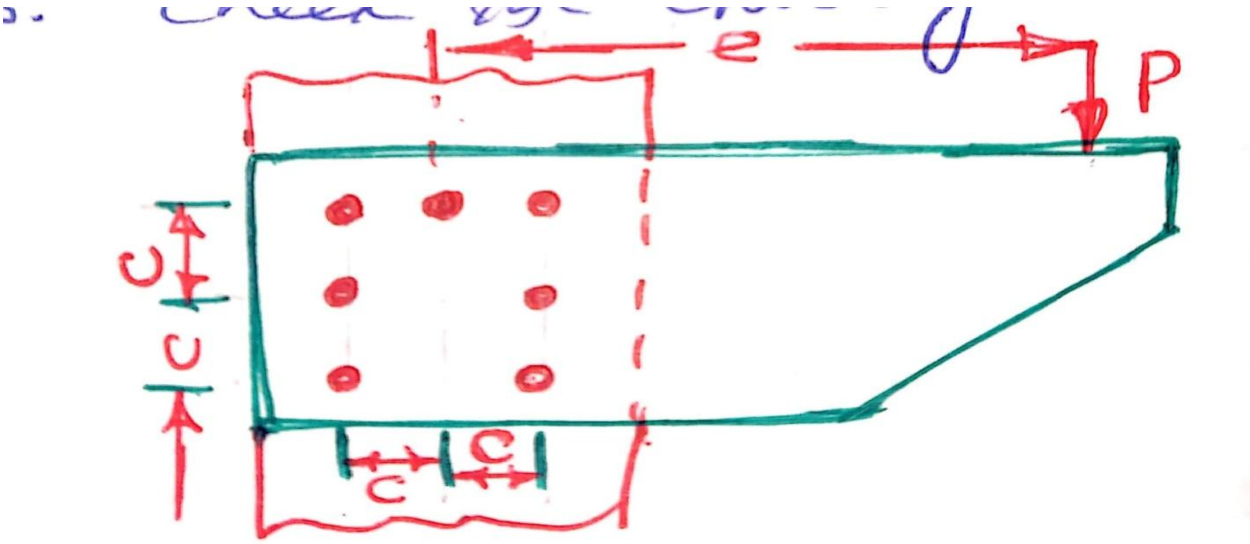
$$\frac{4F_R}{\pi d_1^2} \leq 63.3$$

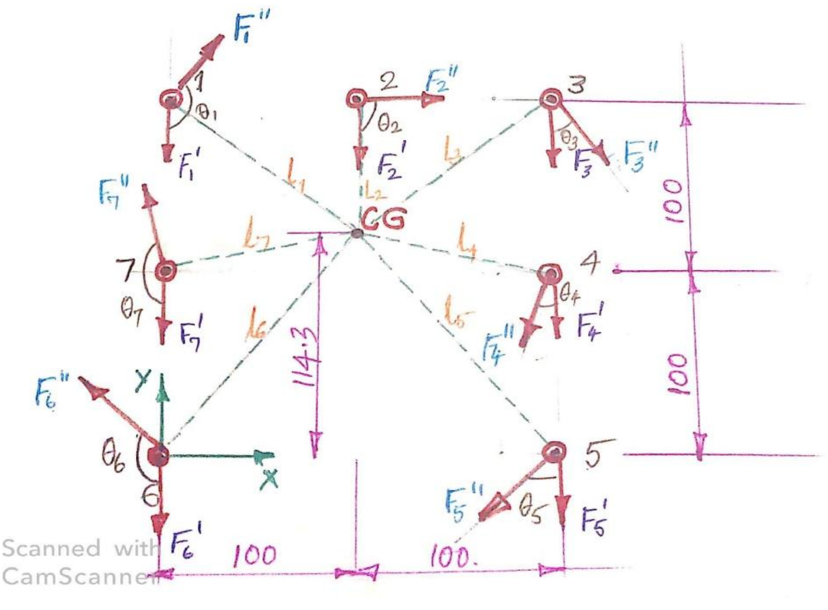
Therefore d_1^2 = 238 sq. mm   and d_1 = 15.42 mm

<u>Three M18 x 2 bolts are recommended for which d = 18 mm, d_1 = 15.4 mm</u>

3. A bracket of thickness 30mm is fastened to a vertical wall as shown. All the bolts are to be of same diameter. Load on the bracket is 50kN. If the shear stress is limited to 65MPa determine the size of the bolts. Check your design against crushing failure if allowable crushing strength is 120 MPa.

P = 50kN,   C = 100 mm, e = 400 mm





Solution:

C.G. position: X = 100 mm: Y = 114.3 mm

Primary load is shear load

$$F' = \frac{F}{7} = \frac{50 * 10^3}{7} = 7.143\ kN$$

$l_1 = l_3 = 131.7$ mm: $l_2 = 85.7$ mm:

$l_4 = l_7 = 101$ mm: $l_5 = l_6 = 152$ mm

Secondary load is maximum at 5 and 6

$$F_5' = F_6'' = \frac{F\ e\ l}{\sum l^2} = 27981\ N$$

θ is small at 5: at which F_R will be maximum

Cos $\theta 5$ = 0.658
F_{R5} = 33121 N

$$\frac{4F_R}{\pi d_1^2} \leq 65$$

Therefore d_1^2 = 649.1 sq.mm and d_1 = 25.48 mm

Seven M30 x 2 bolts are recommended for which d = 30mm, d_1 = 25.7 mm

$$Crushing\ stress = \frac{F_R}{dt} = \frac{33121}{30 * 30} = 36.8\ MPa$$

Less than 120 MPa, design is safe

Section 4

Power Screws

Introduction

- Power screws are threaded elements which are not used for fastening purposes.
- Used to transmit and transform motion and power
- Used to convert rotary motion in to linear motion
- Torque is converted in to force
- Also called as translation screws

Examples:
Automotive vehicle Jacks, C clamp, Fly press, Machine vice, Lead screw of a lathe, linear actuators, Nuclear reactor control devices, etc

There are two main two main applications: in which we require the use of power screws.

- High efficiency requirements such as lead screws and screws in presses
- Low efficiency requirements to achieve self-locking conditions such as in screw jack etc.

Following are the essential parts of a power screw system.

- Screw
- Nut
- A part (Frame) to hold either the screw or nut in place.

Power screws can be operated in two different ways. Possible combination of power and motion transmission is given.

- ➢ Screw rotating in a bearing and nut to have axial motion: Lead screw
- ➢ Nut fixed to the frame and screw to have axial motion: Vehicle jacks and machine vices

Following are the advantages and disadvantages of power screws.

Advantages

- ➢ Large load carrying capacity
- ➢ Compact in construction; lesser number of parts

- Provides large mechanical advantage: for example, 15 kN can be raised by an effort of 400N
- Precise and accurate linear motion
- Noiseless operation
- Can be designed with self-locking feature

Disadvantages

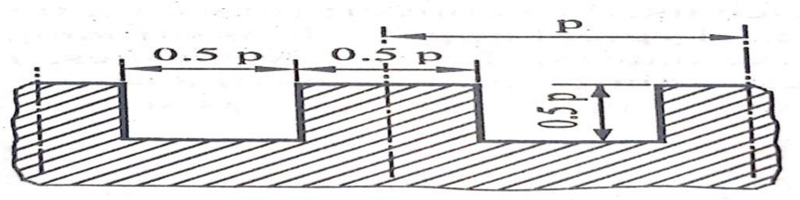
- Poor efficiency (40-70%)
- High friction at threads and rapid wear

Form of threads:

The most popular threads which are used in power screw applications are:

- Square
- Trapezoidal
- Acme threads and
- Buttress threads

Square threads

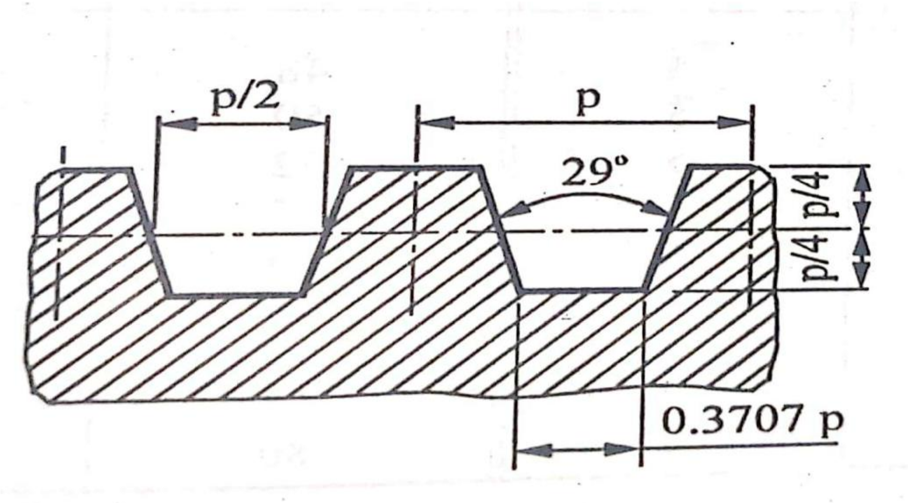


Square Thread

Square threads: Features

- Flanks are normal to the axis
- Difficult to manufacture: Difficult to cut with taps and dies
- Used for power transmission in Machine tools
- Can transmit power in any direction
- Maximum efficiency

Acme threads

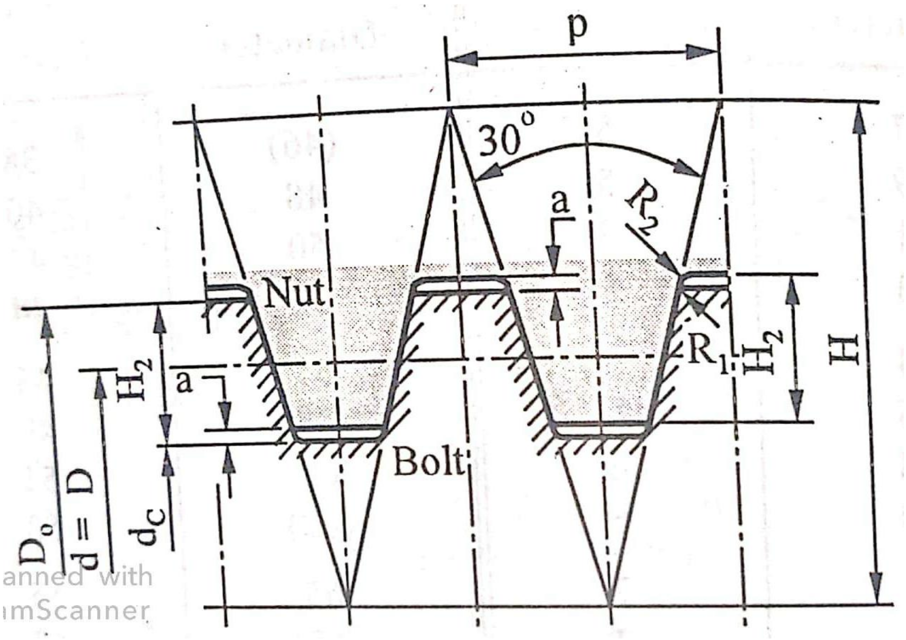


Acme Threads

Acme threads: Features

- ✓ Modified form of square thread
- ✓ Much stronger than square threads
- ✓ Can be cut with taps and dies and easy to manufacture
- ✓ Offer increased area in shear at the root
- ✓ Used in lead screws and bench vices

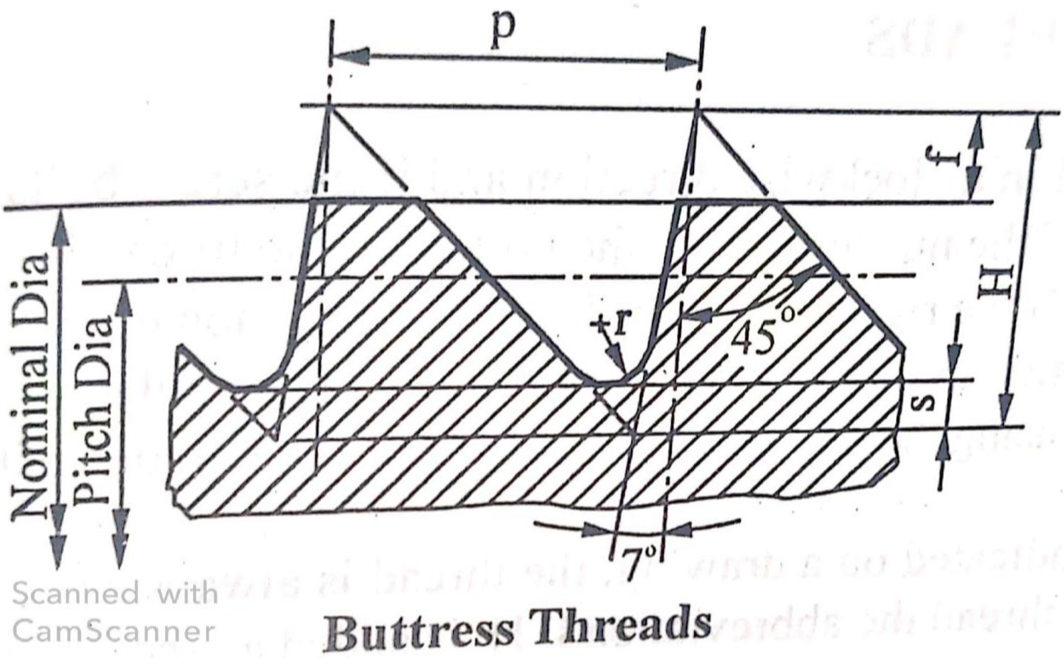
Trapezoidal threads



➢ Similar to ACME threads but thread angle is 30 degrees

➢ Standardised by ISO

Buttress threads



Buttress threads : Features

- ❑ Can transmit large forces only in one direction
- ❑ Stronger compared to other threads
- ❑ Combine high efficiency of Square threads and ease of cutting of Acme or Trapezoidal threads

Single start threads

- ✓ Pitch = lead
- ✓ Depth of thread depends on pitch
- ✓ Large lead requires larger pitch
- ✓ Large pitch results in small core diameter: weakens the screw
- ✓ Multi start threads are used

Multi–start threads

- Two or more parallel threads are employed to increase the travel of the nut per revolution: multi –start threads
- Two or three helix with same pitch are offset parallel to each other
- Lead = pitch x Number of starts

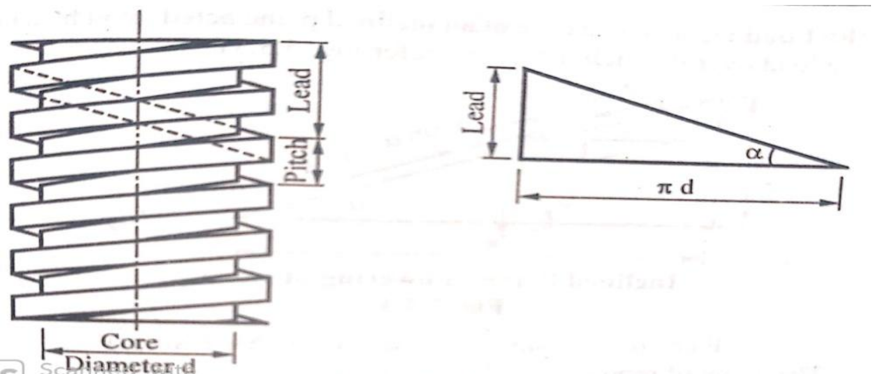
Designation of Power screws

- <u>Single start</u>
- Square thread: Sq 30 x 6
- ISO trapezoidal: Tr 30 x 6
- <u>Multi start</u>
- Square thread: Sq 30 x 14 (P7)
- ISO trapezoidal: Tr 30 x 21 (P7)
- Standard square screw threads dimensions are available in any Design Data Handbooks.

Analysis of square threads

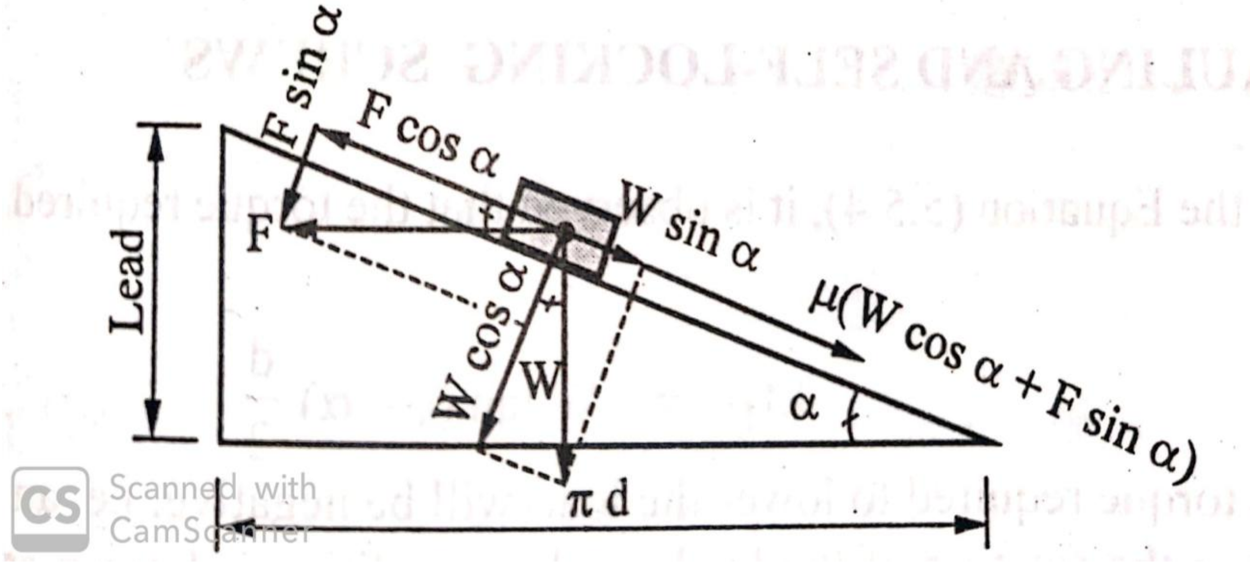
Torque and efficiency of Power Screws

- Analysis is done by considering it as equivalent to raising or lowering of a load/weight on an inclined plane.
- An unwound thread is imagined to represent an inclined plane
- Movement of nut and screw against applied axial load is analogous to movement of a load/ weight on an inclined plane
- This is the basis for determining the torque required to raise or lower the load



Force Analysis

Inclined plane: Lifting of load



- Lifting of load does not necessarily mean that power screw is moving upwards
- When the direction of motion and direction of load are opposite, it can be considered as lifting the load.
- Work is done against the load

Nomenclature

- d: Nominal Diameter
- d_1: Core diameter
- d_2: Mean diameter

$$d2 = \left(\frac{d + d_1}{2}\right)$$

- l = lead
- α = lead angle or helix angle
- f = coefficient of friction at screw threads

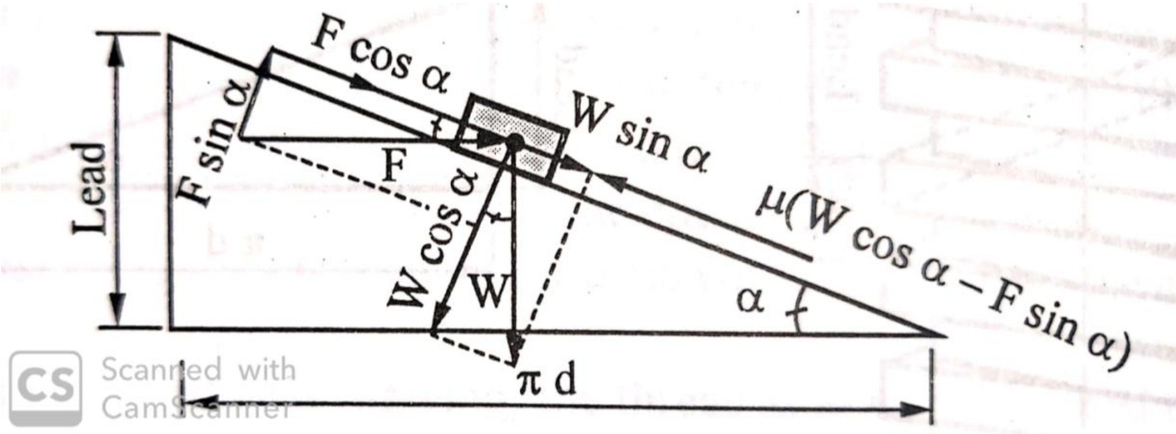
Torque required to raise the load

$$T = \frac{W d_2}{2} . tan(\alpha + \emptyset)$$

$$= \frac{Wd_2}{2} \cdot \left[\frac{tan\emptyset + tan\alpha}{1 - tan\emptyset\ tan\alpha}\right]$$

$$= \frac{Wd_2}{2}\left[\frac{f + tan\alpha}{1 - f\ tan\alpha}\right]$$

Inclined plane: Lowering of load



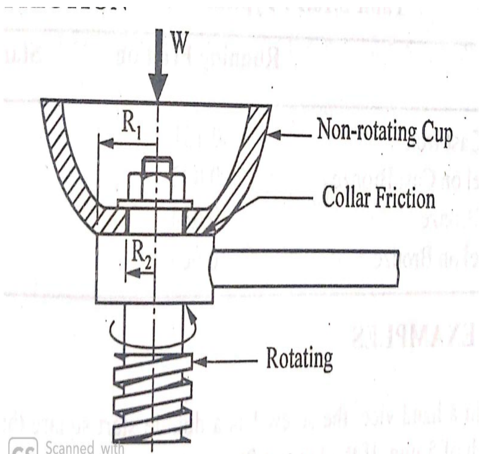
Torque required to lower the load

$$T_L = \frac{Wd_2}{2}\tan(\emptyset - \alpha)$$

$$= \frac{Wd_2}{2}\left[\frac{f - tan\alpha}{1 + f\ tan\alpha}\right]$$

Collar friction

- ✓ Power screws always operate with a collar or a bearing surfaces
- ✓ Continuous rubbing: frictional losses
- ✓ Frictional Power loss



Collar Friction

Collar friction: Torque required to overcome collar friction

$$T_c = \frac{W f_c d_c}{2}$$

d_c = Mean collar diameter which can be computed based on two theories:

Based on uniform pressure theory

$$d_c = \frac{2}{3}\left[\frac{d'^3 - d''^3}{d'^2 - d''^2}\right]$$

Based on Uniform wear theory

$$d_c = \left(\frac{d' - d''}{2}\right)$$

d′ = larger diameter

d″ = smaller diameter

f_c = coefficient of friction at the collar

- Uniform pressure theory predicts larger torque and relatively safer results
- Uniform wear theory is less complicated to use

Efficiency: There are frictional power losses at the screw threads and collar. Hence the torque required to operate the screw with friction is higher than that without friction. Therefore efficiency is given by:

$$\eta = \frac{d_2 . tan\alpha}{d_2 \left(\frac{f + tan\alpha}{1 - f\ tan\alpha}\right) + f_c d_c}$$

Over–hauling and self–locking: These are the two features which can be built in to the power screw while designing.

In applications like load lifting devices such as screw jack, load should not descends on its own. Some additional torque is required to descend the load. This condition is called **Self-locking** which prevents the load descending on its own.

In applications like fly press, load should descends on its own. Some additional torque is required to stop the descent of the load. This condition is called **Over–hauling** which makes the mass and the screw to come down on their own and strike the component to produce the require impact.

Thus,

Torque required to lower the load

$$T_L = \frac{\boldsymbol{W d_2}}{\mathbf{2}} \mathbf{tan}(\emptyset - \boldsymbol{\alpha})$$

If α is greater than ϕ

T_L becomes negative: Additional torque is required to stop the descent of the load: This condition is called Over - hauling: load descends on its own.

For Over hauling

$$tan\alpha \geq \frac{f d_2 + f_c d_c}{(d_2 - f . f_c . d_c)}$$

<u>Example: FLY PRESS</u>

Screw with the fly wheel masses are raised and released: If the screw is designed to overhaul, the mass and the screw come down on their own and strike the component.

If α is less than ϕ

T_L becomes positive: Additional torque is required to descend the load: This condition is called Self-locking load does not descends on its own.

Example: Screw Jack

- Load is raised and released: If the screw is designed to self-lock the load does not comes down on its own. Extra effort is required to descend the load. Further, the efficiency of a self-locked screw is less than 50%.

Number of threads: The number of threads in the nut is determined based on bearing pressure. It is given by:

$$\boldsymbol{n = \frac{4W}{P_b.\pi\left(d^2 - D_1^2\right)}}$$

Height of nut

$$l_h = \frac{\boldsymbol{4W}}{P_b.\pi(d^2 - D_1^2)}$$

Stresses induced in power screws: While designing a power screw following stresses need to be considered

- Due to axial load
  - ✓ Compression
  - ✓ Buckling: Column action
- Torsion
- Shearing across threads
- Crushing or bearing across threads

Direct Tensile or Compressive stress due to axial load

$$\sigma_t \ or \ \sigma_c = \frac{4W}{\pi d_1^2}$$

Stresses due to buckling: When the screw rod is slender it is likely to buckle. It act as a column. Then the stresses due to buckling has to be computed using any of the column equation. Rankine's formula is used here, as given below.

$$\sigma_c = \frac{W}{A}\left[1 + a\left(\frac{1}{k}\right)^2\right]$$

$$\frac{l}{k} = Slemderness\ ratio: l = length$$

$$k = \sqrt{\frac{I}{A}} \qquad I = \frac{\pi d^4}{64}$$

α = Rankine's constant
k = radius of gyration
$I =$ moment of inertia of cross sectional area

Torsional shear stresses

$$\tau = \frac{16T}{\pi d_1^3}$$

Maximum Principal stress and Maximum shear stress: Power screws usually operate under combined loading conditions. Hence the maximum stresses due to these combined loads is given in terms of Maximum Principal Stress and Maximum shear stress

$$\text{Maximum Principal stress} = \sigma_1 = \frac{\sigma}{2} + \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

$$\text{Maximum shear stress} = \tau_{max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

Shear stresses in threads: Threads are subjected to direct shear at their roots. The direct shear stress at the root of a thread is given by:

$$\tau_{th} = \frac{W}{\pi d_1 n t}$$

t = thickness of the thread at the root = $\frac{p}{2}$, p = pitch, n= number of threads

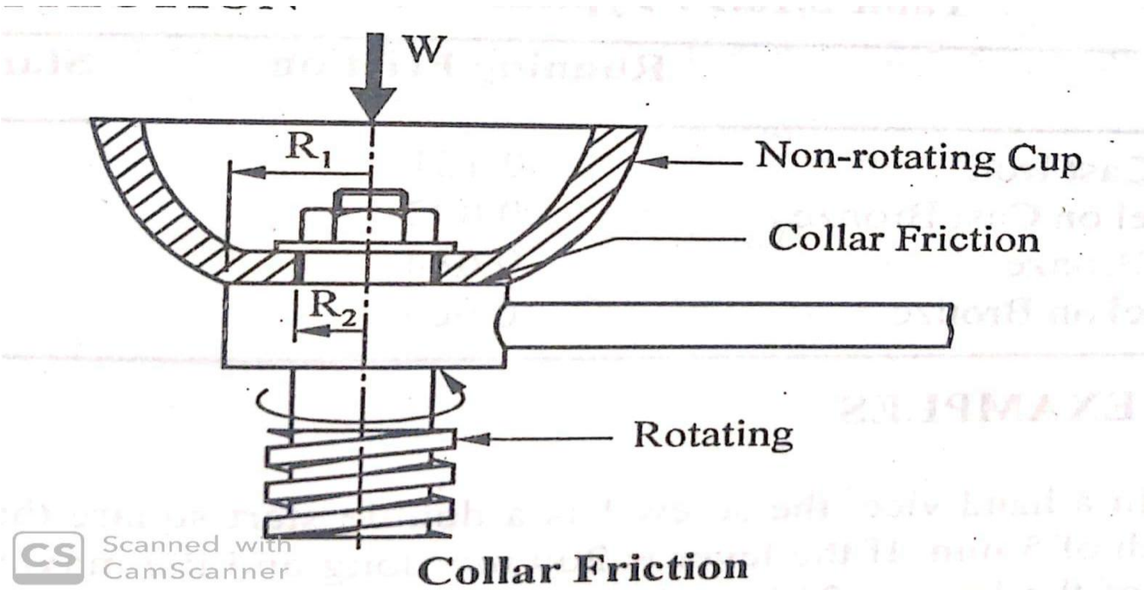
Bearing stresses in threads: Threads are subjected to bearing loads due relative motion. Bearing stresses (P_b) induced is given by:

$$P_b = \frac{4W}{n.\pi(d^2 - D_1^2)}$$

D_1= Root diameter of internal threads

Numerical examples

1. **A lifting device has a double start square threaded screw of 24 mm outer diameter and pitch 5mm. It is operated by a lever of 200 mm length. Maximum effort that can be applied at the end of the lever is 250N. Co efficient of friction at the screw threads is 0.12. Determine the load that can be lifted using this device, neglecting collar friction. Is the device self-locked?**



Solution:

Given

d = 24 mm, p = 5 mm

d_1 = d – p = 19 mm

d_2 = mean diameter = $\left(\frac{d+d_1}{2}\right) = 21.5mm$

tanϕ = f = 0.12, ϕ = 6.84

Lead = Pitch x No. of starts = 5 x 2 = 10mm

$\tan\alpha$ = Lead / πd_2 = 0.1481 α = 8.42

Torque to be applied at the handle:

T = Effort x Handle length = 250 x 200 = 50000 Nmm

Torque required to raise the load

$$= \frac{Wd_2}{2}\left[\frac{f + tan\alpha}{1 - f\ tan\alpha}\right]$$

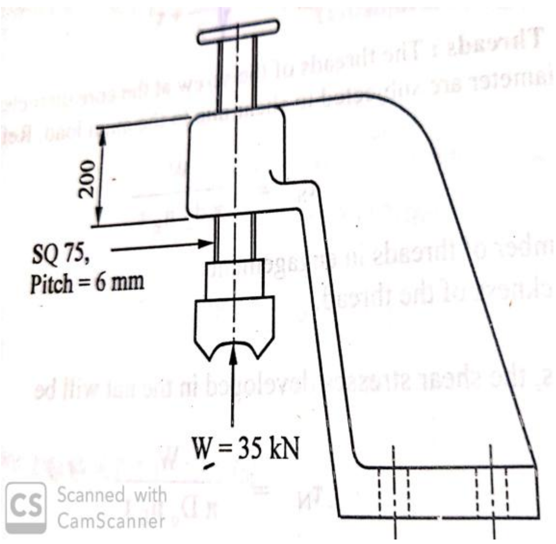
$= \frac{W.21.5}{2}\left[\frac{0.12+0.1481}{1-0.12*0.1481}\right] = 2.932W$ Nm

Therefore,
50000 = 2.932 W
Load that can be lifted W = 17053 N

Further, α is greater than ϕ: screw overhauls allowing the load to decend on its own : Design is not satisfactory

2. **A shaft straightener is as shown in figure. It is required to apply a load of 35kN to straighten the shaft. Determine the force required at the rim of the hand wheel of 300 mm diameter. The coefficient of friction at the threads is 0.12. Neglecting buckling of the screw rod determine:**
1) **Max. compressive and shear stress**
2) **Shear and bearing stress in the threads**
3) **Efficiency of the device neglecting collar friction**

Shaft Straightener



Data given:

- W = 35 kN
- d= 75mm
- Diameter of the hand wheel = D= 300mm
- P=6 mm
- f= 0.12

Solution:

d_1 = d-p = (75-6) = 69mm

$$d_2 = \left(\frac{d + d_1}{2}\right) = 72\ mm$$

$$\tan\alpha = \frac{lead}{\pi d_2} = \frac{6}{\pi * 72} = 0.0266$$

α = 1.519 degree

tan ϕ = 0.12

ϕ = 6.843 degree

1. Effort at the hand wheel

Torque at the hand wheel = Effort x Radius of the hand wheel

Torque required to produce an axial load of 35 kN

$T_H = \frac{Wd_2}{2}\left[\frac{f+tan\alpha}{1-f\ tan\alpha}\right] =$

$$= \frac{35000 * 72}{2}\ \left[\frac{0.12 + 0.0266}{1 - 0.12 * 0.0266}\right]$$

= 185206.81 Nmm

185206.81 = P* 150

P = 1234.72 N

2, Maximum stresses in the screw rod

Compressive stress: $\sigma_c = \frac{4W}{\pi d_1^2}$

$$= \frac{4 * 35000}{\pi * 69^2} = 9.36\ MPa$$

Torsional shear stress

$$= \tau = \frac{16T}{\pi d_1^3} = 2.87\ MPa$$

Maximum compressive stress =

$$\sigma_1 = \frac{\sigma}{2} + \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2} = 10.16\ MPa$$

Maximum shear stress

$$\tau_{max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2} = 5.48\ MPa$$

3.For a triple start screw nominal diameter is 50 mm and pitch is 8 mm. The load on screw is 7.5 kN and coefficient of friction is 0.12, Neglecting collar friction and assuming length of nut as 48 mm. Determine:

i. **Maximum stress developed in the screw body. Bearing pressure,**
ii. **State whether the screw is self locking or overhauling.**

Data given:

- W = 7.5 kN
- d= 50 mm: P=8 mm , Triple start
- f= 0.12
- H= 48 mm

Solution:

d_1 = d-p = (50-8) = 42mm

$$d_2 = \left(\frac{d + d_1}{2}\right) = 46\ mm$$

$$tan\alpha = \frac{lead}{\pi d_2} = \frac{3 * 8}{\pi * 46} = 0.1661$$

α = 9.4 degree

f = tanϕ = 0.12

ϕ = 6.843 degree

Maximum stresses in the screw rod

Compressive stress:

$$\sigma_c = \frac{4W}{\pi d_1^2}$$

$$= \frac{4 * 7500}{\pi * 42^2} = 5.46\ MPa$$

Torsional shear stress =

$$\tau = \frac{16T}{\pi d_1^3}$$

Torque required to produce an axial load of 7.5 kN

$$= \frac{W d_2}{2}\left[\frac{f + tan\alpha}{1 - f\ tan\alpha}\right] = 50246\ Nmm$$

tanα = 0.1661

α = 9.4 degree

f = tanϕ = 0.12

ϕ = 6.843 degree

$$\tau = \frac{16T}{\pi d_1^3} = 3.45\ MPa$$

T = 50246 Nmm
d_1 = 42 mm

Maximum compressive stress =

$$\sigma_1 = \frac{\sigma}{2} + \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2} = 7.06\ MPa$$

Maximum shear stress

$$\tau_{max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2} = 4.39\ MPa$$

Bearing pressure

H = p X n:

48 = 8 X n: n = 6

d_1 = 42 mm : d = 50 mm : W = 7500 N

$$P_b = \frac{4W}{n.\pi\ (d^2 - D_1^2)} = 2.16\ MPa$$

α is greater than ϕ : Overhauls

4. A machine vice has single start, square threads with 22 mm nominal diameter and 5mm pitch. The outer and inner diameters of the friction collar are 55 and 45 mm respectively. The coefficient of friction for thread and collar are 0.15 and 0.17 respectively. The machinist can comfortably exert a force of 125 N on the handle at a mean radius of 150 mm. Assuming uniform wear for the collar, calculate the the clamping force developed between the jaws and the Overall efficiency of the vice

5. The lead screw of a lathe has single-start ISO metric trapezoidal threads of 52 mm nominal diameter and 8 mm pitch. The screw is required to exert an axial force of 2 kN in order to drive the tool carriage during turning operation. The thrust collar of 100 mm outer diameter and 60 mm inner diameter is provided to take the thrust load. The coefficient of friction at the screw threads and the

collar are 0.15 and 0.12 respectively. The carriage moves at a speed of 4mm/sec.

Calculate :

- The power required to drive the lead screw and
- The efficiency of the screw

Evaluate the results using uniform wear theory and uniform pressure theory

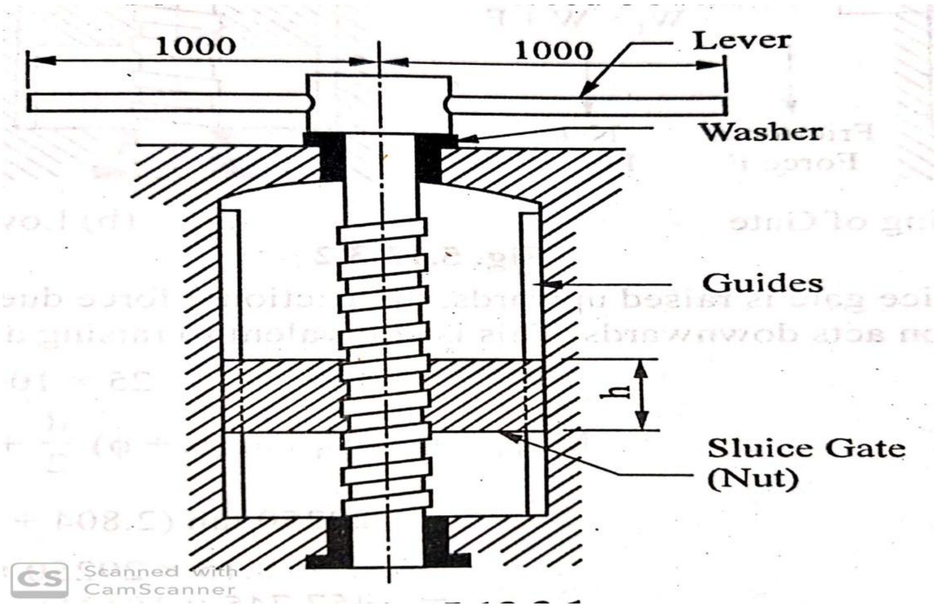
6.A power transmission screw of a screw press is required to transmit maximum load of 10 tonnes and rotates at 60 rpm. Trapezoidal threads employed The screw thread friction coefficient is 012. Torque required for collar friction and journal bearing is about 10% of the torque to drive the load considering screw friction. Determine screw dimensions and its efficiency. Also determine power required to drive the screw. Maximum permissible compressive stress in screw is 100MPa.

8. Show that the efficiency of a self-locking square threaded screw is less than 50%.

9. The cutter of a broaching machine is pulled by a square threaded screw of 50mm external diameter and 8mm pitch. The operating nut takes a load of 42 kN on a flat surface of 84mm external diameter and 56mm internal diameter. Coefficient of friction is 0.15 for all contact surfaces. Determine the power required to rotate the nut when the cutting speed is 15m/min. What is the efficiency of the screw?

10. A sluice gate weighing 25kN as shown in figure is to be raised or lowered by a square threaded screw of outside diameter 70mm and pitch of 10mm.The frictional resistance induced by water pressure against the gate when it is in its lowest position is 4250N. A washer having outer diameter of 200mm and an inner diameter of 70mm is used as a collar. The coefficient of friction at the threads is 0.12 and at the collar is 0.15. Determine the maximum force to be exerted at the end of the lever for raising and lowering the gate. What is the efficiency of the system? Also determine the required height of the nut if the permissible bearing pressure is 6MPa.

Sluice Gate



Solution: Steps

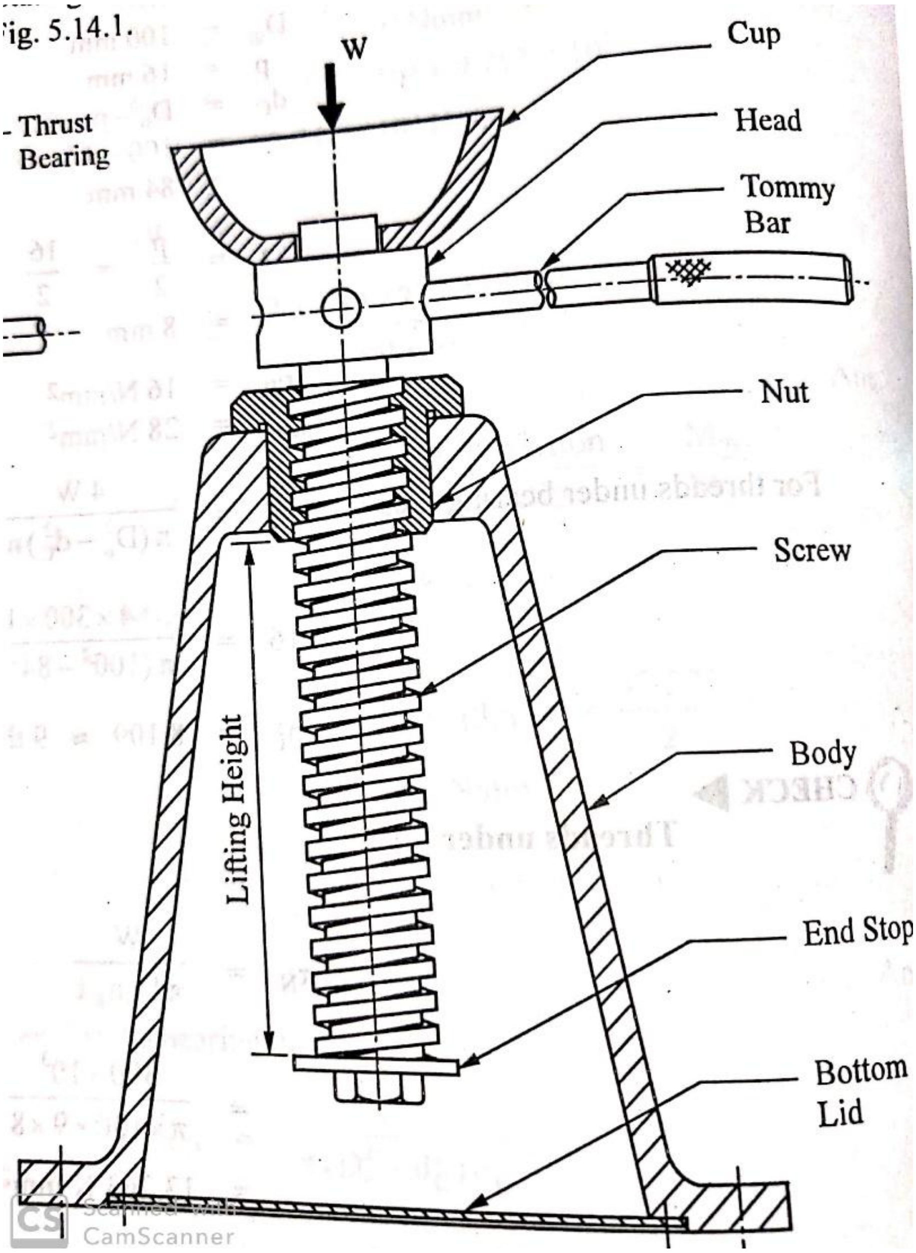
- Parameters of Screw:
- Torque and effort required to raise the gate and efficiency calculation
- Torque and effort required to lower the gate and efficiency calculation
- Height of the nut: No. of threads

Design of a Screw Jack

Design of screw jack to lift a load of 42kN with a lift of 150mm. Allowable tensile strength and shear strength of the screw rod are 82 MPa and 50 MPa respectively. Make suitable assumptions wherever necessary

Data:

W = 42 kN: Lift = 150mm
σ_{all} = 82 MPa τ_{all} = 50 MPa



Solution: Design of Screw Rod

The screw rod is subjected to both compressive and torsional load simultaneously and also acts as a column. Let the preliminary diameter be determined based on direct compressive stress. Increase this diameter by 20-25% to resist the combined compressive and shear stresses. Carryout the design check in terms of induced stresses

Preliminary diameter (Exploratory)

Let a square threaded screw is used.

For safe design

$\sigma_{ind} \leq \sigma_{all}$ Given: σ_{all} = 82 MPa: W = 42 kN

$\sigma_{ind} = \frac{W}{A_c}$ = σ_{all} = 82 MPa
A_c = 512.19mm^2

Nearest standard size: Sq 32 x 6: with d = 32 mm from DDHB
Increase it to 1.25 x 32 = 40 mm

Let Standard sized square threaded screw be used: Sq 40 x 7: with d_1 = 33 mm: p = 7 mm,
A_c = 855 mm^2. However, it should be checked for induced stresses.

Design check: When this screw is used under the given load following stresses are induced:

Direct Compressive stress:

$$\sigma_{C,ind} = \frac{W}{A_C} = \frac{42\ x\ 10^3}{855} = 49.12\ MPa$$

Torsional shear stress: Induced due to the torque required (applied) to raise the load by overcoming the friction at the screw threads.

$$T = \frac{Wd_2}{2}.tan\ (\alpha + \emptyset)$$

d = 40 mm, d_1: 33 mm,

Mean diameter:

$$d_2 = \left(\frac{d + d_1}{2}\right) = 36.5\ mm$$

Let the co efficient of friction at the screw thread = f = 0.14 from DDHB

tan (ϕ) = 0.14, ϕ = 7.96 degrees

$$tan\alpha = \frac{lead}{\pi d_2} = \frac{7}{\pi * 36.5} = 0.061$$

α = 3.49 degrees

α is less than ϕ, therefore SELF – LOCKED. This is the necessary condition for a screw jack. If this condition is not satisfied, it should redesigned.

Torque: torque required to raise the load by overcoming the friction at the screw threads:

$$T_S = \frac{Wd_2}{2} tan\,(\emptyset + \alpha)$$

$$= \frac{42 * 1000 * 36.5}{2} tan\,(3.49 + 7.96)$$

T_S = 155249 Nmm

Torsional shear stress:

$$\tau = \frac{16T}{\pi d_1^3} = \frac{16 * 155249}{\pi * 33^3} = 22\ MPa$$

Stresses due to buckling: The screw rod acts as a column and hence likely to buckle. The induced stresses are computed using:

Rankine – Gordon formula:

$$\sigma_C = \frac{W}{A}\left[1 + a\left(\frac{l}{k}\right)^2\right]$$

$$\frac{l}{k} = Slenderness\ ratio\ .k = radius\ of\ gyration = \frac{d_1}{4} = \frac{33}{4} = 8.25\ mm$$

Length of the screw rod =l = 2* lift = 2* 150 = 300 mm

a = end fixity constant $= \frac{1}{6250}$ for a column free to bend in any plane

$$\sigma_{cr} = \frac{42 * 1000}{855}\left[1 + \frac{1}{6250}\left(\frac{300}{8.25}\right)^2\right]$$

σ_{cr} = 59.48 MPa

As evident from the above a state of combined stress occurs in the screw rod. Hence Maximum Principal Stress and Maximum shear stress should be computed and checked against allowable stresses.

Maximum Principal Stress:

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

$$= \frac{59.48}{2} + \sqrt{\left(\frac{59.48}{2}\right)^2 + 22^2}$$

σ_1 = 66.5 MPa

Maximum shear stress:

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

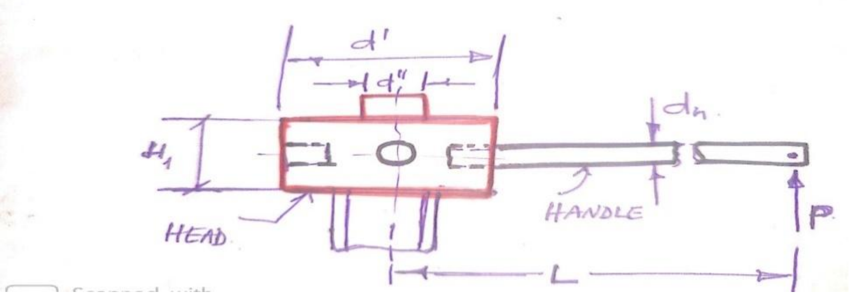
$$= \sqrt{\left(\frac{59/48}{2}\right)^2 + 22^2}$$

τ_{max} = 36.5 MPa

σ_1 = 66.5 MPa τ_{max} = 36.5 MPa Less than σ_{all} = 82 MPa τ_{all} = 50 MPa. Therefore, the proposed screw rod with specifications Sq 40 x 7 is safe.

Design of Head (Thrust Collar) and Handle

Thrust collar is extension of screw rod.



Dimensions of the collar are fixed as follows:

- d′ = larger diameter = 1.5 d = 1.5 x 40 = 60 mm
- d″ = smaller diameter = 0.5 d = 0.5 x 40 = 20 mm

Mean collar diameter: Uniform wear theory

- $d_C = \left(\frac{d' + d''}{2}\right) = 40\ mm$
- f_c = coefficient of friction at the collar = 0.12

Torque required to overcome the friction at the collar:

$$T_C = \frac{42 * 1000 * 0.12 * 40}{2} = 100.8 * 10^3\ Nmm$$

Total torque T = (155.249 + 100.8) x 10^3
= 256 x 10^3 Nmm

Length of the Handle

Effort applied on the handle produces the torque required.

If P = effort = 400 – 500N (Effort that can be applied by an average person)

L = length

P x L = T
400 x L = 256 x 10^3
L = 640 mm

Diameter of the Handle

From bending consideration
M = (640 – 30) x 400 = 256 x 10^3 Nmm

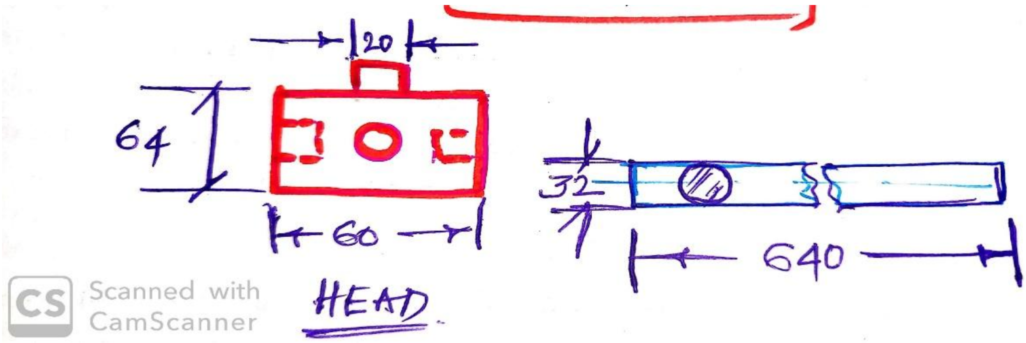
Material same as screw rod

$$\sigma_b = \frac{32M}{\pi d_h^3} = \frac{32 \times 256 \times 10^3}{\pi d_h^3} = 82\ MPa$$

d_h = 31.17 mm = 32 mm

Height of Thrust Collar

$h_t = 2 \times d_h = 2 \times 32 = 64$ mm

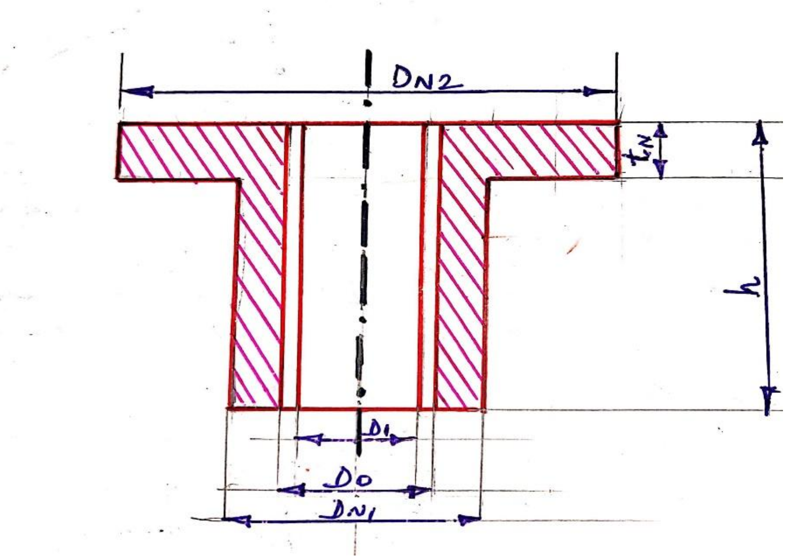


Design of Nut

Material:
CI / Phosphor Bronze
Phosphor Bronze

- $\sigma_{t,all} = 50$ MPa
- $\tau_{all} = 25$ MPa
- $P_b = 15$ MPa

- ➢ D_o = Nominal diameter
- ➢ D_1 = core diameter
- ➢ D_{N1} =O.D of nut body
- ➢ D_{N2} = O.D of nut collar
- ➢ t_n = thickness of nut collar
- ➢ h = height of nut



Number of threads and Height of Nut

From bearing consideration

$$n = \frac{4W}{P_b\pi\,(D^2 - D_1^2)}$$

$$= \frac{4 \times 42 \times 1000}{15 \times \pi\,(40^2 - 33^2)} = 6.98 = 7\ threads$$

h = 7 x 7 = 49 mm

Design check

Shear stress in the threads of the screw

t = root thickness of the thread = p/2

$$\tau_{th} = \frac{W}{\pi d_1 nt} = \frac{42 \times 1000}{\pi \times 33 \times 7 \times 3.5} = 16.536\ MPa$$

Shear stress in the threads of the nut

$$\tau_{th} = \frac{W}{\pi D_o nt} = \frac{42 \times 1000}{\pi \times 40 \times 7 \times 3.5} = 13.62\ MPa$$

τ_{ind} less than τ_{all}

D_{N1} = O.D of nut body D_o = 40 mm
= 2 x 40 = 80 mm

Thickness and diameter of the nut collar

1. **Thickness**

By shear considerations

$$t_n = \frac{W}{\pi D_{N1\pi N}} = \frac{42 \times 1000}{\pi \times 80 \times\times 25} = 6.685 = 7\ mm$$

2. **Diameter D_{N2}**

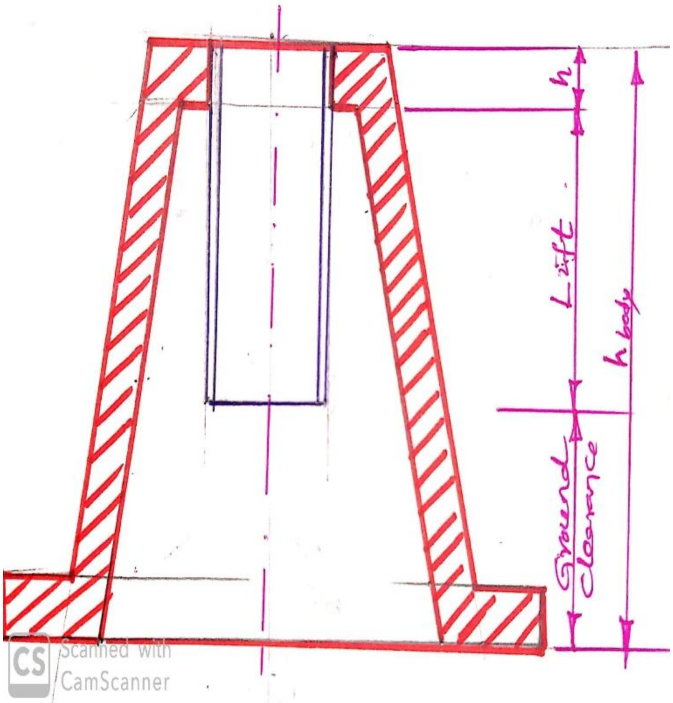
By compression considerations

$$\sigma_{c,all} = \frac{4W}{\pi(D_{N2}^2 - D_{N1}^2)}$$

$$50 = \frac{4 \times 42 \times 1000}{\pi(D_{N2}^2 - 80^2)}$$

D_{N2} = 86.426 mm = 87 mm

Body: Body will generally made of CI . It is hollow as shown and tapered to provide stability.



Height = Lift + h + ground clearance
= 150 + 49 + 20 = 219 mm = 220 mm. Other dimensions proportionately fixed
Efficiency

$$\eta = \frac{d_2.tan\alpha}{d_2\left(\frac{f + tan\alpha}{1 - f\ tan\alpha}\right) + f_c d_c} = 16\%$$

Since efficiency is less than than 50% the screw is SLF – LOCKED

CUP: Non rotating element made of Cast Iron: Dimensions can be fixed proportionately.

Bibliography

1. K.Mahadevan and K. Balaveera Reddy, Design Data Handbook, Fourth Edition, CBS Publishers and Distributors, 2016.
2. Dr.K.Lingaiah and Prof. B.R.Narayana Iyengar, Machine Design Data Handbook, Suma publishers.Volume 1, Fourth Edition, 2006.
3. Maleev and Hartman's Machine Design, Revised and Edited by O.P. Grover, CBS Publishers and Distributors, 2015.
4. V.B.Bhandari, Design of Machine Elements, Fourth Edition, Mc Graw Hill Education (India) Private ltd. 2017.
5. Farazdak Haideri, Machine Design, second Edition, Nirali Prakashan, Pune, 2002.

Wish you all the best