04. Design of Power Screws

Marks – 12

Hours - 10

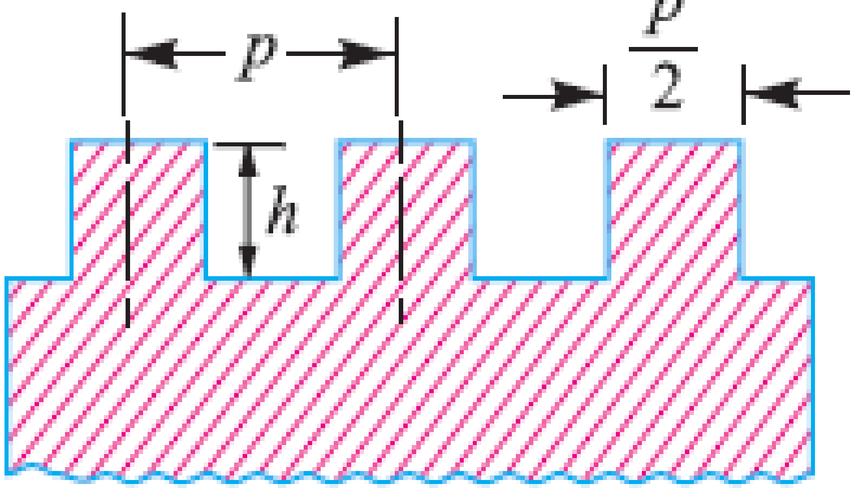
Introduction

- The power screws (also known as ***translation screws)*** are used to convert rotary motion into translatory motion.
- For example, in the case of the lead screw of lathe, the rotary motion is available but the tool has to be advanced in the direction of the cut against the cutting resistance of the material.
- In case of screw jack, a small force applied in the horizontal plane is used to raise or lower a large load.

- Power screws are also used in vices, testing machines, presses, etc.
- In most of the power screws, the nut has axial motion against the resisting axial force while the screw rotates in its bearings.
- In some screws, the screw rotates and moves axially against the resisting force while the nut is stationary and in others the nut rotates while the screw moves axially with no rotation.

Types of Thread Profile Used in Power Screws

1) Square Thread –



$h = 0.5\,p$

(*a*) Square thread.

Application – Screw jack, Mechanical Press, Clamping devices.

- This threads is adopted for the transmission of power in either direction.
- It is difficult to cut with taps and dies.
- It is generally cut on lathe machine with single point cutting tool.

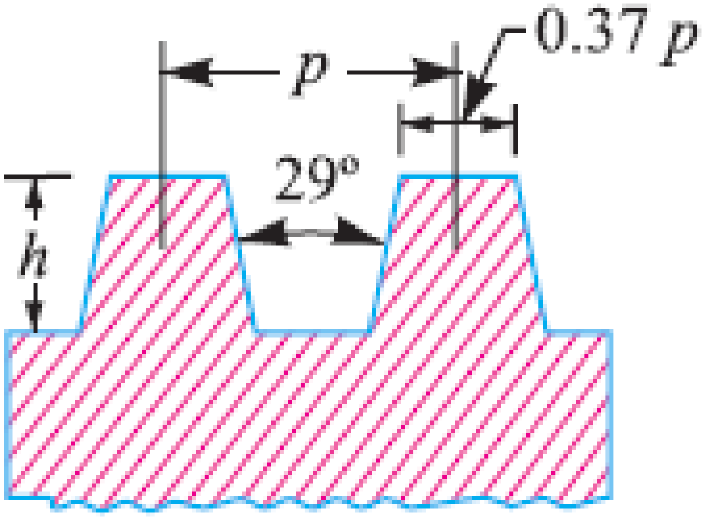
- **Advantages –**

1. It has maximum efficiency.
2. Minimum radial or bursting pressure on nut.
3. These are of self locking type.

- **Limitations –**

1. Strength of square thread is less as compared with other forms of threads.
2. These threads can not be used conveniently with split nut because of wear compensation is not possible and it is difficult in engagement & disengagement.

2) **Acme Threads** –



$h = 0.5\,p + 0.25$ mm

(b) Acme thread.

- The acme thread are not efficient as square thread.
- The acme threads are easier to cut and are stronger than square threads.
- **Application** – Lead screw of a lathe machine.

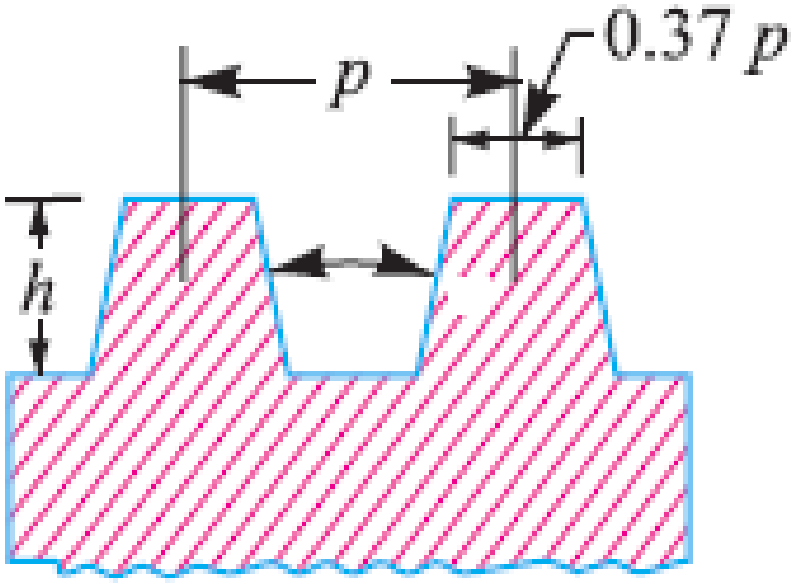
- **Advantages –**

1. These permits the use of split nut which can be easy for engagement & disengagement and compensate wear.
2. These are stronger than the square threads in shear because of the larger C/s at the root.

- **Limitations –**

1. The efficiency is lower than that of square thread due to slop given to the sides.
2. The slop on the sides introduces some brusting pressure on the nut.

3) **<u>Trapezoidal Threads</u>** –

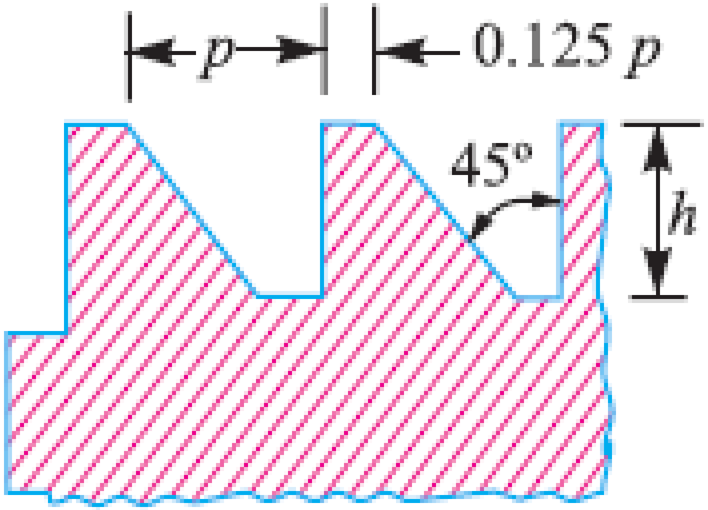


$h = 0.5\,p + 0.25$ mm

(*b*) Acme thread.

The trapezoidal threads are similar to Acme threads except the thread angle is 30^0 in trapezoidal threads.

4) **Buttress Threads** –

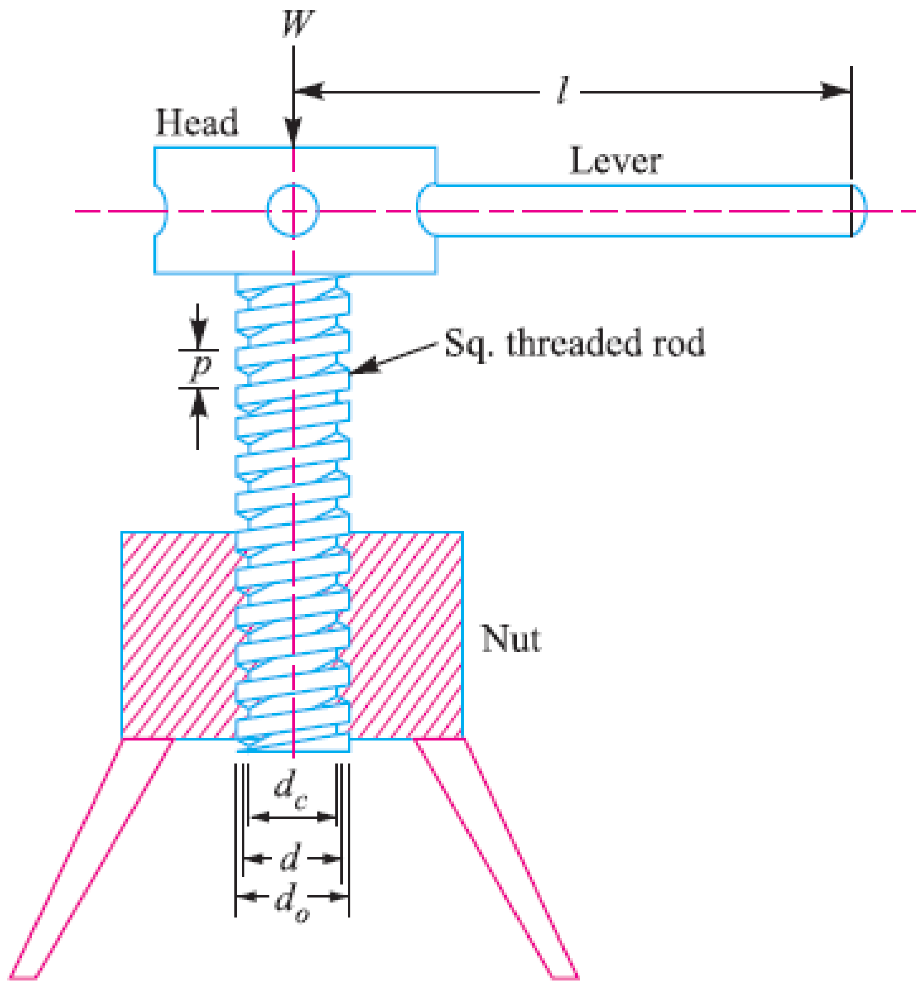


$h = 0.75\,p$

(*c*) Buttress thread.

- ➢A buttress thread *is used when large forces act along the screw axis in* one direction only.
- ➢This thread combines the higher efficiency of square thread and the ease of cutting and the adaptability to a split nut of acme thread.
- ➢ It is stronger than other threads because of greater thickness at the base of the thread.
- ➢The buttress thread has limited use for power transmission.
- ➢It is employed as the thread for light jack screws and vices.

Torque Required to Raise Load for Square Threaded Screws



$$W = Load\ to\ be\ raised\ in\ N.$$

$$P = Effort\ applied\ at\ the\ circumference\ of\ screw$$

$$p = Pitch\ of\ the\ screw\ in\ mm$$

$$d = mean\ diameter\ of\ the\ screw\ in\ mm$$

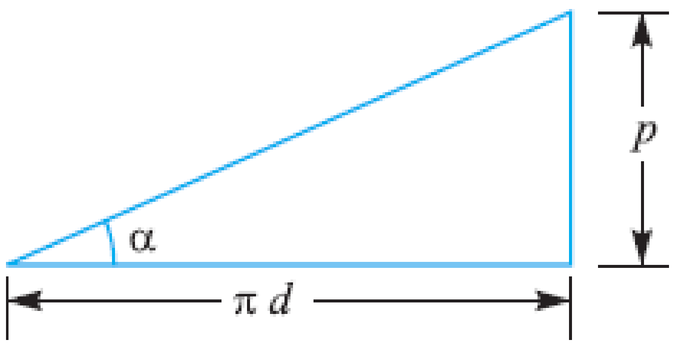
$$\alpha = Helix\ angle$$

$$\mu = coefficien\ t\ of\ friction\ between\ screw\ and\ nut$$

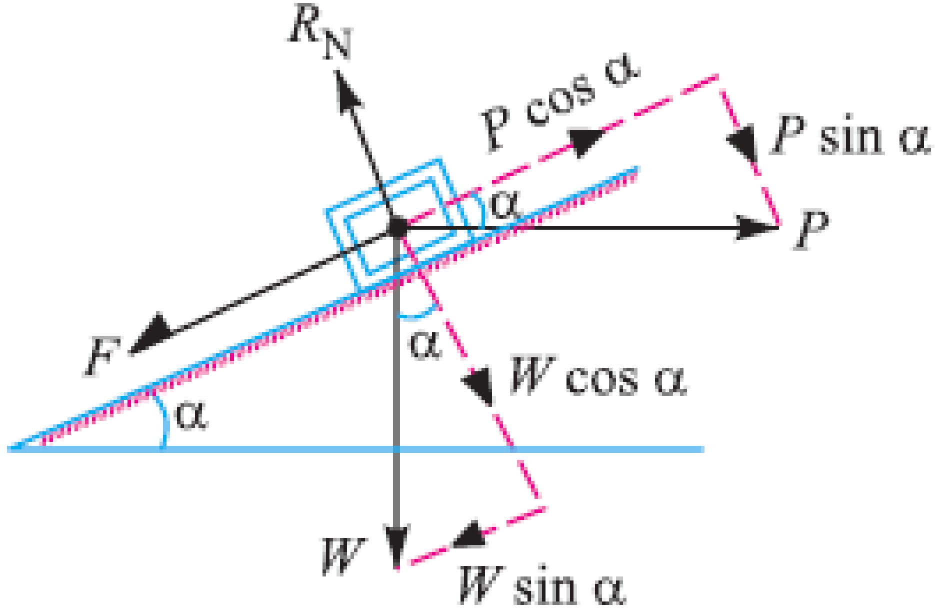
$$\mu = \tan \phi$$

$$\phi = friction\ angle$$

$$L = Lead\ of\ screw$$



(*a*) Development of a screw.



(*b*) Forces acting on the screw.

$$From\ the\ geometry\ of\ fig.$$

$$\tan\alpha = \frac{pitch}{\pi d} = \frac{p}{\pi\, d}$$

$$The\ force\ of\ friction\ F = \mu R_N\ will\ acts\ downwards$$

$$\text{Re}\,solving\ the\ forces\ along\ the\ path$$

$$\sum F_H = 0$$

$$P\cos\alpha - W\sin\alpha - F = 0$$

$$P\cos\alpha - W\sin\alpha - \mu\, R_N = 0$$

$$\therefore P\cos\alpha = W\sin\alpha + \mu\, R_N \;---(1)$$

$$\sum F_V = 0$$

$$R_N - P\sin\alpha - W\cos\alpha = 0$$

$$R_N = P\sin\alpha + W\cos\alpha \quad ---(2)$$

Put the value of R_N *in eq.1*

$$P\cos\alpha = W\sin\alpha + \mu(P\sin\alpha + W\cos\alpha)$$

$$P\cos\alpha = W\sin\alpha + \mu P\sin\alpha + \mu W\cos\alpha$$

$$P\cos\alpha - \mu P\sin\alpha = W\sin\alpha + \mu W\cos\alpha$$

$$P(\cos\alpha - \mu\sin\alpha) = W(\sin\alpha + \mu\cos\alpha)$$

$$P = W \times \frac{(\sin\alpha + \mu\cos\alpha)}{(\cos\alpha - \mu\sin\alpha)}$$

$$But\ \mu = \tan\phi$$

$$P = W \times \frac{(\sin\alpha + \tan\phi.\cos\alpha)}{(\cos\alpha - \tan\phi.\sin\alpha)}$$

$$But\ \tan\phi = \frac{\sin\phi}{\cos\phi}$$

$$P = W \times \frac{(\sin\alpha + \frac{\sin\phi}{\cos\phi} \times \cos\alpha)}{(\cos\alpha - \frac{\sin\phi}{\cos\phi} \times \sin\alpha)}$$

$$P = W \times \frac{(\sin\alpha.\cos\phi + \sin\phi.\cos\alpha)}{(\cos\alpha.\cos\phi - \sin\phi.\sin\alpha)}$$

$$But\ \sin(A+B) = \sin A.\cos B + \cos A.\sin B$$

$$\&\ \cos(A+B) = \cos A.\cos B - \sin A.\sin B$$

$$P = W \times \frac{\sin(\alpha+\phi)}{\cos(\alpha+\phi)}$$

$$P = W \times \tan(\phi+\alpha)$$

$$\therefore Torque\ required\ to\ overcome\ thread\ friction$$

$$T = P \times \frac{d}{2}$$

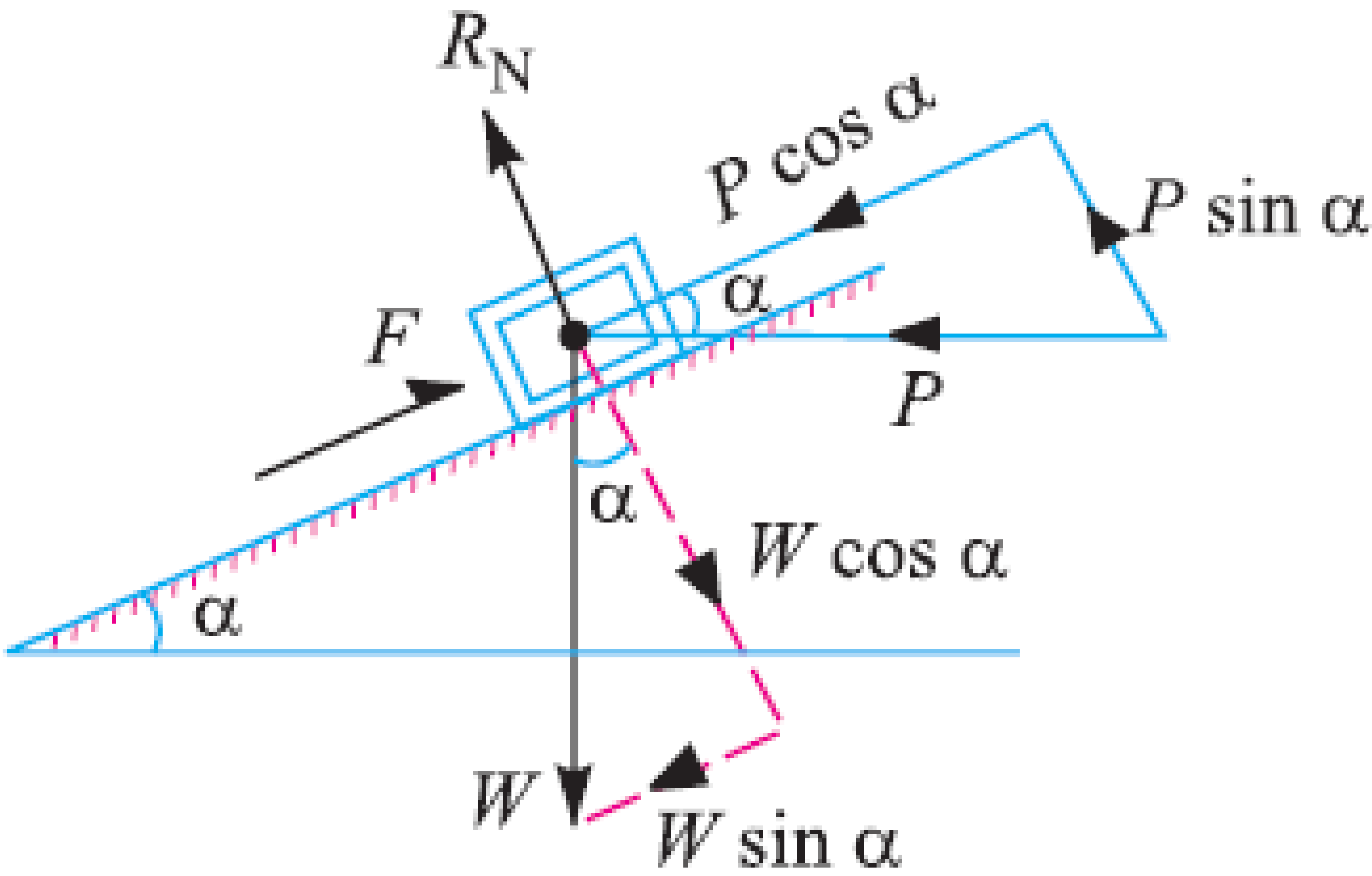
$$\therefore T = W \times \tan(\phi + \alpha) \times \frac{d}{2}$$

$$If\ effort\ is\ applied\ by\ lever\ of\ length\ 'l'$$

$$then, T = P \times l$$

$$\therefore T = W \times \tan(\phi + \alpha) \times l$$

Torque Required to Lower the Load



- Resolving the forces in horizontal & vertical direction.

$$\sum F_H = 0$$

$$P\cos\alpha = F - W\sin\alpha$$

$$P\cos\alpha = \mu R_N - W\sin\alpha ----(1)$$

$$\sum F_V = 0$$

$$R_N = W\cos\alpha - P\sin\alpha ----(2)$$

$$Put\, eq.2\, in\, eq.1$$

$$P\cos\alpha = \mu(W\cos\alpha - P\sin\alpha) - W\sin\alpha$$

$$P\cos\alpha = \mu W\cos\alpha - \mu P\sin\alpha - W\sin\alpha$$

$$P\cos\alpha + \mu P\sin\alpha = \mu W\cos\alpha - W\sin\alpha$$

$$P(\cos\alpha + \mu\sin\alpha) = W(\mu\cos\alpha - \sin\alpha)$$

$$P = W \times \frac{(\mu \cos\alpha - \sin\alpha)}{(\cos\alpha + \mu \sin\alpha)}$$

$$P = W \times \frac{(\tan\phi \cos\alpha - \sin\alpha)}{(\cos\alpha + \tan\phi \sin\alpha)} \ldots\ldots (as \mu = \tan\phi)$$

Multiply and divide by $\cos\phi$

$$P = W \times \frac{(\sin\phi \cos\alpha - \sin\alpha \cos\phi)}{(\cos\phi \cos\alpha + \sin\phi \sin\alpha)}$$

$$P = W \times \frac{\sin(\phi - \alpha)}{\cos(\phi - \alpha)}$$

$$P = W \times \tan(\phi - \alpha)$$

$\therefore$ *Torque required to lower the load in overcoming the friction* (T).

$$T = P \times \frac{d}{2}$$

$$T = W \tan(\phi - \alpha)\frac{d}{2}$$

Self Locking & Overhauling of Screw

- **Self Locking Screw –**

The torque required at the circumference of the screw to lower the load is given by

$$T = W\tan(\phi - \alpha)\frac{d}{2}$$

In the above equation, if friction angle is greater than helix angle (Ø > α) then the torque required to lower the screw is positive i.e. some effort is required to lower the load, such a screw is known as self **locking screw.**

Self Locking & Overhauling of Screw

- **Overhauling Screw –**

The torque required at the circumference of the screw to lower the load is given by

$$T = W\tan(\phi - \alpha)\frac{d}{2}$$

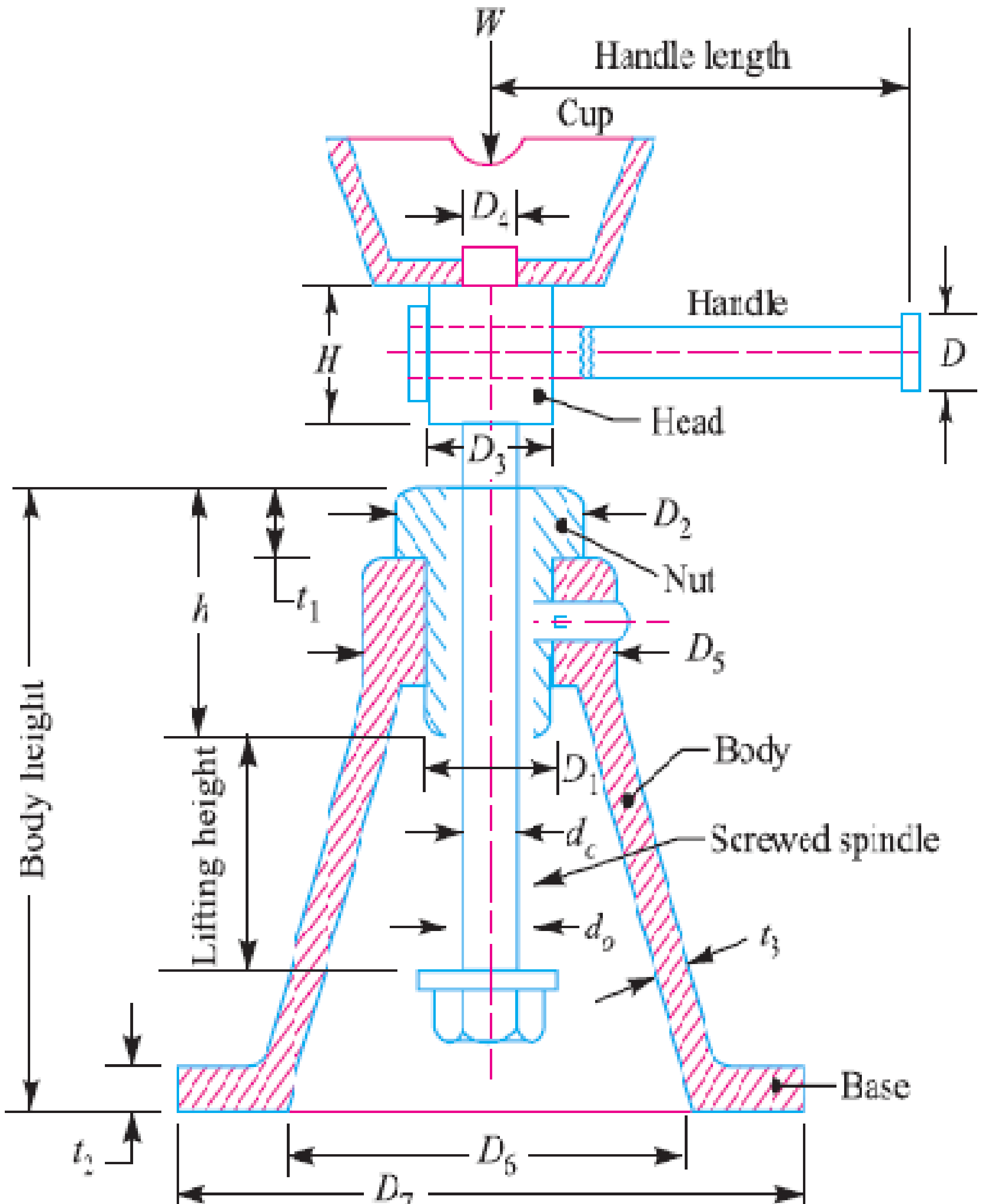
In the above equation, if friction angle is less than helix angle ($Ø < \alpha$) then the torque required to lower the screw is negative i.e. the load will start moving downward without application of any effort, such a screw is known as **overhauling screw.**

Design of Screw Jack

- The various parts of screw jack are

1. Screw spindle having square thread screw.
2. Nut and collar for nut.
3. Head at the top of the screw spindle for handle.
4. Cup at the top of head for the load.
5. Body of the screw jack.
6. A handle or Tommy bar.

Animation of Screw Jack


W
Handle length
Cup
D4
Handle
H
D
Head
D3
D2
Nut
t1
h
D5
Body height
Body
D1
Lifting height
dc
Screwed spindle
do
t3
Base
t2
D6
D7

1) Design of screw –

i) Find the core diameter of screw by considering the screw under pure compression.

$$\sigma_c = \frac{W}{A} = \frac{W}{\frac{\pi}{4} \times d_c^2}$$

$$From\, this\, equation\, d_c\, can\, be\, obtained$$

ii) Select the standard square threads for screw

$$Take (d_c + 6) = d_c \;\; for\, design\, in\, exam$$

$$Then\, d_o = \frac{d_c}{0.84} \;\&\; d_c = d_o - p$$

$$mean\, dia. = d = d_o - \frac{p}{2} = \frac{d_o + d_c}{2}$$

iii) In addition to direct compressive load, the screw is subjected to twisting moment (T_1), so for that the core diameter is increased and appropriately find the torque required to rotate the screw.

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

A) Checking of Screw –

i) Shear stress due to torque

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

$$\textit{from this eq. find shear stress}\ (\tau)$$

$ii)\ Direct\ compressive\ stress\ due\ to\ axial\ load -$

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

$Then, use$

$Maximum\ principle\ stress\ theory$

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

$Maximum\ shear\ stress\ theory$

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

$For\ safety$

$$\sigma_{c(\max)} < \sigma_{(Given)}\ \&$$

$$\tau_{\max} < \tau_{(Given)}$$

2) **Design of nut** –

i) The height of nut 'h' can be found by considering the bearing pressure on nut

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

where, n = no. of threads in contact with screw spindle

$$\therefore Height\ of\ nut = h = n \times p$$

where, p = pitch of threads

ii) Check for shear stresses in screw and nut threads

$$Shear\ stress\ in\ screw = \tau_s = \frac{W}{\pi d_c t \times n}$$

$$Shear\ stress\ in\ nut = \tau_n = \frac{W}{\pi d_o t \times n}$$

$$where, t = \frac{p}{2}$$

For safety of the screw & nut threads

$$\tau_s\ and\ \tau_n < \tau_{(Given)}$$

3) <u>Design of Nut Collar</u> –

Let, D_1 = Inner Diameter of nut collar or outer diameter of nut.

D_2 = Outer diameter of nut collar

t_1 = thickness of nut collar

a) Considering tearing of nut due to tensile strength

$$\sigma_{t(nut)} = \frac{W}{\frac{\pi}{4}(D_1^2 - d_o^2)}$$

Find out D_1

$b)$ *Considering the crushing of nut and collar*

$$\sigma_{cr(nut)} = \frac{W}{\frac{\pi}{4}(D_2^2 - D_1^2)}$$

Find D_2

$c)$ *Considering the shearing of nut collar*

$$\tau_{(nut)} = \frac{W}{\pi D_1 t_1}$$

Find t_1

4) <u>Design of (Spindle) Screw Head</u> –

$$To\ find\ the\ diameter\ of\ spindle\ head\ D_3$$

$$and\ diameter\ of\ cup$$

$$Use\ empirical\ relation\ as\ given\ below -$$

$$Diameter\ of\ head = D_3 = 1.75\, d_o$$

$$Diameter\ of\ cup = D_4 = \frac{D_3}{4}$$

5) **<u>To find the torque to overcome due to friction at the top of the screw (Collar friction)</u>** –

$$a)\ Assume\ uniform\ pressure\ condition$$

$$T_c = \frac{2}{3} \times \mu_c \times W \times [\frac{R_3^3 - R_4^3}{R_3^2 - R_4^2}]$$

$$b)\ Assume\ uniform\ wear\ condition$$

$$T_c = \mu_c \times W \times R$$

$$Where\ R = \frac{R_3 + R_4}{2}$$

6) **Design of Handle** –

a) To design the handle, consider the total torque to which the handle will be subjected is given by –

$T_{total} = T_1 + T_c$

Assume that one person can apply a force of 300 N to 400 N

$$\therefore Effective\ length\ of\ handle\ required$$

$$L_E = \frac{T_{total}}{300\ to\ 400\ N}$$

$$\therefore Length\ of\ handle = L = L_E + Gripping\ length$$

b) To find the diameter of handle (d_h) –

Consider the handle under bending

$$T_{total} = Total\ torque\ req.\ to\ raise\ the\ load$$

$$M = Maximum\ B.M.\ acting\ on\ handle$$

$$\sigma_b = Bending\ stress\ induced\ in\ handle$$

$$\therefore \sigma_b = \frac{M}{Z}$$

$$\sigma_b = \frac{M}{\frac{\pi}{32} \times d_h^3}$$

$$Find\ d_h$$

$$\therefore Height\ of\ nut = H = 2\, d_h$$

7) **<u>Buckling load of the screw</u>** –

Check the safety of the screw against buckling failure –

$$According\ to\ J.B.\ Johnson\ formula$$

$$Buckling\ load = W_{cr} = A_c \times \sigma_{yc}[1 - \frac{\sigma_{yc}}{4C\pi^2 E}(\frac{L}{k})^2]$$

$$Where, C = end\ fixity\ coefficient = 0.25$$

$$k = radius\ of\ gyaration = 0.25\, d_c$$

$$\sigma_{yc} = yield\ point\ stress$$

$$L = unsupported\ length\ of\ the\ screw$$

$$A_c = \frac{\pi}{4} d_c^2 = c/s\ area\ of\ screw$$

$$E = modulus\ of\ elasticity$$

Factor of safety against buckling failure is

$$N_{cr} = \frac{W_{cr}}{W}$$

For safe design of screw under buckling

$$N_{cr} \geq N_f$$

8) *Find the efficiency of the screw jack –*

$$\eta_0 = \frac{T_o}{T}$$

9) **Find dimensions of screw jack body by empirical relation** –

$$a)\ diameter\ of\ body\ at\ top = D_5 = 1.5\,D_2$$

$$b)\ Thickness\ of\ body = t_3 = 0.25\,d_o$$

$$c)\ Inside\ diameter\ at\ bottom = D_6 = 2.25\,D_2$$

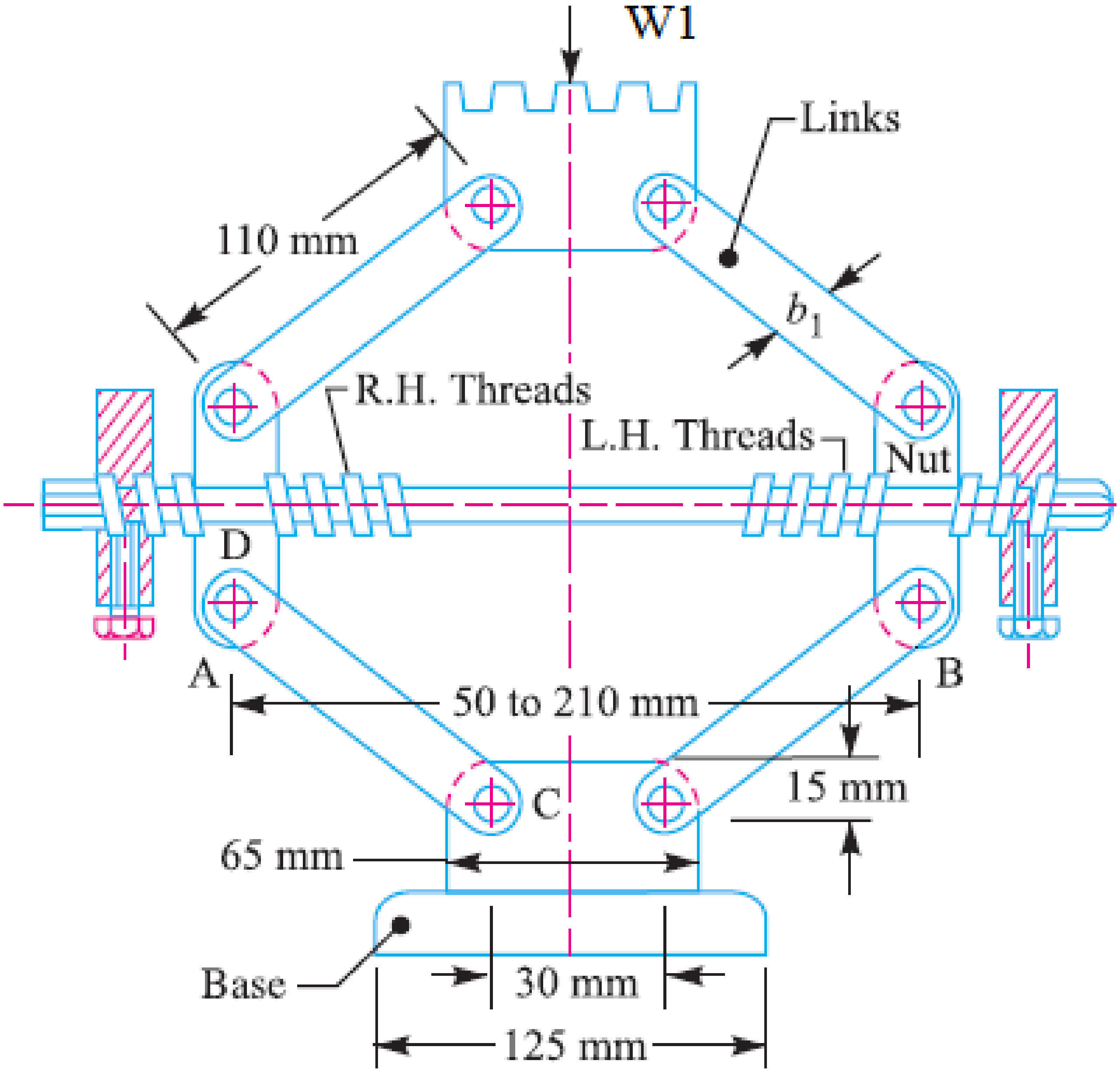
$$d)\ Outside\ diameter\ at\ bottom = D_7 = 1.75\,D_6$$

$$e)\ Thickness\ at\ the\ base = t_2 = 2\,t_1$$

$$f)\ Height\ of\ body = H_b = Max.lift + Ht.of\ nut + 100\,mm\ extra\ for\ clearance$$

$$H_b = (H_l + h + 100)$$

Design of Toggle Jack


W1
Links
110 mm
b_1
R.H. Threads
L.H. Threads
Nut
D
A
B
50 to 210 mm
C
15 mm
65 mm
Base
30 mm
125 mm

- The various parts of the toggle jack are

i) The compound screw

ii) Two nuts

iii) Eight links

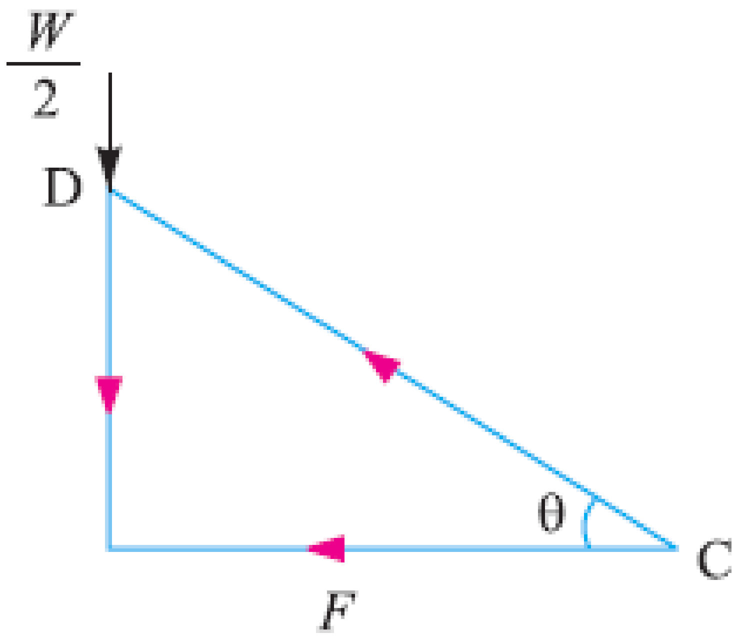
iv) The pins

v) The head

➢ Animation of Toggle Jack

1) <u>Find the maximum tensile force in screw</u> –

The maximum load on the screw thread occurs when the jack in bottom position.



From figure

$$\tan\theta = \frac{\frac{W}{2}}{F}$$

$$\therefore F = \frac{W}{2\tan\theta}$$

$Where\ \theta = angle\ of\ link\ with\ horizontal$

$\Rightarrow Maximum\ value\ of\ \theta\ is\ taken\ as\ 35^0$

$Due\ to\ force\ 'F'\ links\ are\ subjected\ to\ tension\ and$

$square\ thread\ screw\ under\ pull$

$\therefore Total\ tensile\ force\ on\ thread\ screw = 2F = W$

$Considering\ screw\ under\ tensile\ failure$

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

$d_c\ can\ be\ obtained.$

$Then\ d_0 = d_c + p$

$$\& \, d = \frac{d_o + d_c}{2}$$

2) Find the torque required to overcome the thread friction (T_1) –

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

$$\& T_1 = \frac{\pi}{16} \times \tau \times d_c^3$$

From this eq. find 'τ'

As the screw is subjected to both direct tensile stress and torsional shear stress

i) max*imum principle stress*

$$\sigma_{t(\max)} = \frac{1}{2}[\sigma_t + \sqrt{\sigma_t^2 + 4\tau^2}]$$

ii) max*imum shear stress*

$$\tau_{\max} = \frac{1}{2}[\sqrt{\sigma_t^2 + 4\tau^2}]$$

For safety

$$\sigma_{t(\max)} \leq \sigma_t \ \& \ \tau_{\max} \leq \tau$$

3) **Design of nut** -

a) Find the Height of nut by considering bearing pressure –

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

$$Find\ 'n'$$

$$For\ providing\ good\ stability\ \&\ avoid\ tearing$$

$$Taken' = n + 2$$

$$height\ of\ nut = n' \times p$$

$$Width\ of\ nut = b_1 = 1.5\, d_o$$

b) Check the stresses induced in nut and screw thread

$$Shear\ stress\ in\ screw = \tau_s = \frac{W}{\pi\, d_c\, t \times n}$$

$$Shear\ stress\ in\ nut = \tau_n = \frac{W}{\pi\, d_o\, t \times n}$$

4) <u>Calculate length of screw portion</u> –

Length of screw = dist. between the centered line of nut when toggle jack at the top position + thickness of nut + (2 x thickness of ring)

5) <u>Design of Pins</u> –

The pins are design under double shear.

$$\tau_p = \frac{F}{2A}$$

$$\tau_p = \frac{F}{2 \times \frac{\pi}{4} \times d_p^2}$$

$$Where, d_p = diameter\ of\ pin$$

$$\tau_p = Shear\ stress\ for\ pin$$

THE END