Module
6
Power Screws

Lesson
1
Power Screw drives and their efficiency

Instructional Objectives

At the end of this lesson, the students should be able to understand

- Power screw mechanism.
- The thread forms used in power screws.
- Torque required to raise and lower a load in a power screw
- Efficiency of a power screw and condition for self locking.

6.1.1 Introduction

A power screw is a drive used in machinery to convert a rotary motion into a linear motion for power transmission. It produces uniform motion and the design of the power screw may be such that

(a) Either the screw or the nut is held at rest and the other member rotates as it moves axially. A typical example of this is a screw clamp.

(b) Either the screw or the nut rotates but does not move axially. A typical example for this is a press.

Other applications of power screws are jack screws, lead screws of a lathe, screws for vices, presses etc.

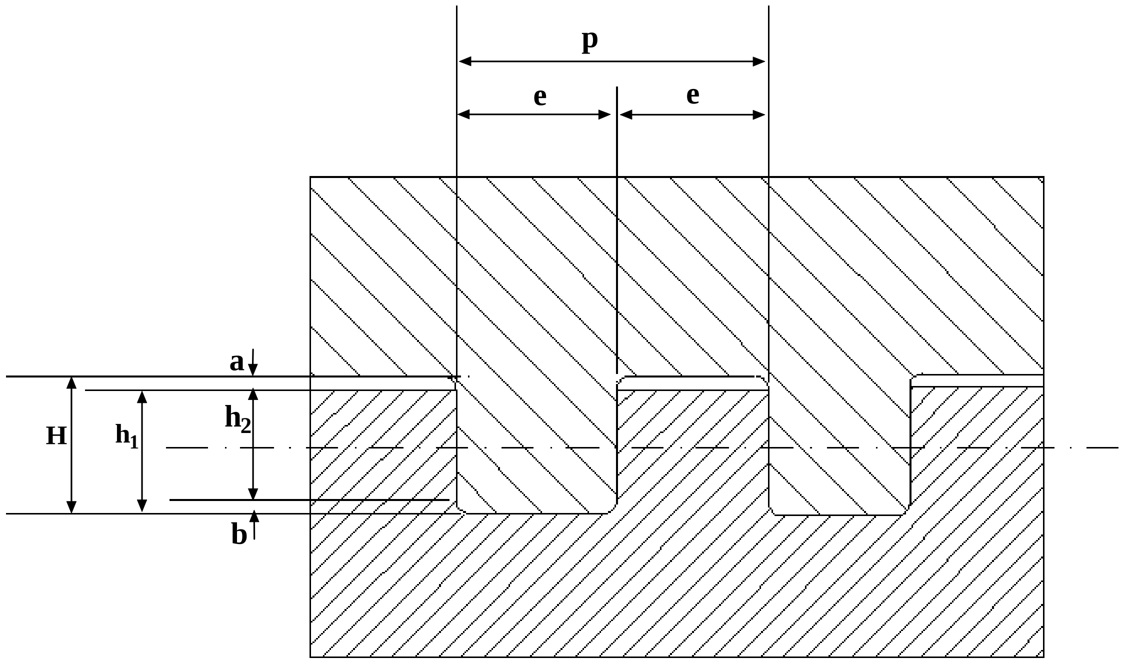
Power screw normally uses square threads but ACME or Buttress threads may also be used. Power screws should be designed for smooth and noiseless transmission of power with an ability to carry heavy loads with high efficiency. We first consider the different thread forms and their proportions:

Square threads-

The thread form is shown in **figure-6.1.1.1**. These threads have high efficiency but they are difficult to manufacture and are expensive. The proportions in terms of pitch are:

$h_1 = 0.5\ p$; $h_2 = 0.5\ p - b$; $H = 0.5\ p + a$; $e = 0.5\ p$

a and b are different for different series of threads.



***6.1.1.1F** – Some details of square thread form*

There are different series of this thread form and some nominal diameters, corresponding pitch and dimensions a and b are shown in **table-6.1.1.1** as per I.S. 4694-1968.

6.1.1.1T – Dimensions of three different series of square thread form.

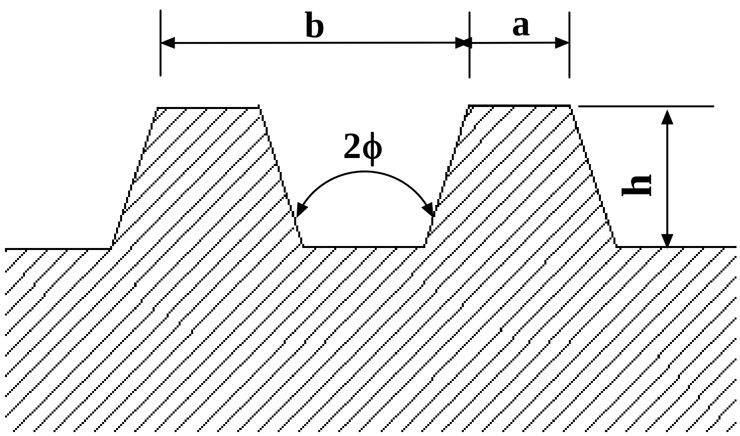
Fine Series					Normal Series					Coarse Series				
Nominal Dia (mm)	Steps (mm)	Pitch (mm)	a (mm)	b (mm)	Nominal Dia (mm)	Steps (mm)	Pitch (mm)	a (mm)	b (mm)	Nominal Dia (mm)	Steps (mm)	Pitch (mm)	a (mm)	b (mm)
10-22	2	2	0.25	0.25	22-28	2	5	0.25	0.5	22-28	2	8	0.25	0.5
22-62	2	3	0.25	0.25	30-36	2	6	0.25	0.5	30-38	2	10	0.25	0.5
115-175	5	6	0.25	0.5	115-145	5	14	0.5	1	115-130	5	22	0.5	1
250-300	10	12	0.25	0.5	240-260	10	22	0.5	1	250-280	10	40	0.5	1
420-500	20	18	0.5	1	270-290	10	24	0.5	1	290-300	10	44	0.5	1

According to IS-4694-1968, a square thread is designated by its nominal diameter and pitch, as for example, SQ 10 x 2 designates a thread form of nominal diameter 10 mm and pitch 2 mm.

Acme or Trapezoidal threads

The Acme thread form is shown in **figure- 6.1.1.2**. These threads may be used in applications such as lead screw of a lathe where loss of motion cannot be tolerated. The included angle $2\phi = 29^o$ and other proportions are

$a = \frac{p}{2.7}$ and h = 0.25 p + 0.25 mm



***6.1.1.2F** – Some details of Acme or Trapezoidal thread forms.*

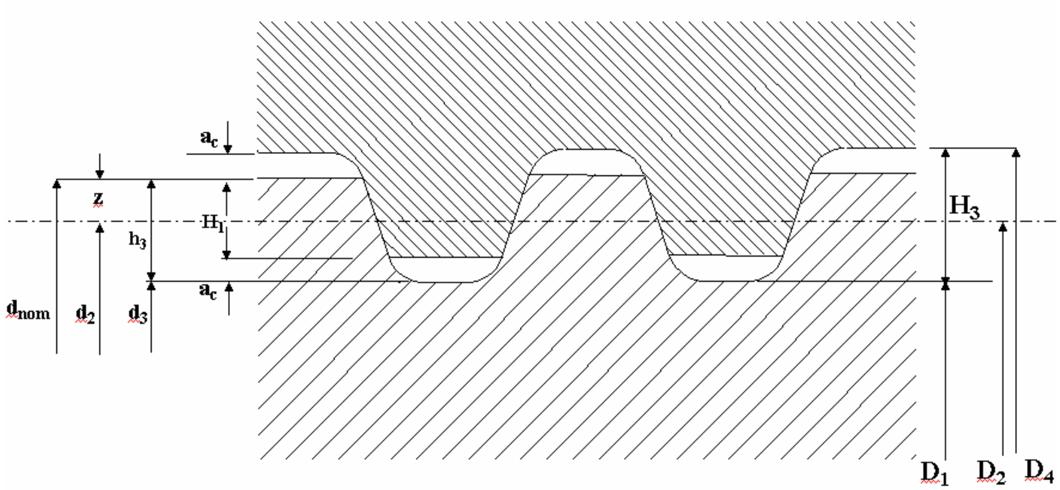
A metric trapezoidal thread form is shown in **figure- 6.1.1.3** and different proportions of the thread form in terms of the pitch are as follows:

Included angle = 30^o ; H_1= 0.5 p ; z = 0.25 p + $H_1/2$; $H_3 = h_3 = H_1 + a_c$ = 0.5 p + a_c

a_c is different for different pitch, for example

a_c = 0.15 mm for p = 1.5 mm ; a_c = 0.25 mm for p = 2 to 5 mm;

a_c = 0.5 mm for p = 6 to 12 mm ; a_c = 1 mm for p = 14 to 44 mm.



***6.1.1.3F**- Some details of a metric Trapezoidal thread form.*

Some standard dimensions for a trapezoidal thread form are given in **table- 6.1.1.2** as per IS 7008 (Part II and III) - 1973:

6.1.1.2T- Dimensions of a trapezoidal thread form.

Nominal Diameter (mm)	8	10	5	25	50	75	100	150	200	250	300
pitch (mm)	1.5	2	4	5	8	10	12	16	18	22	24

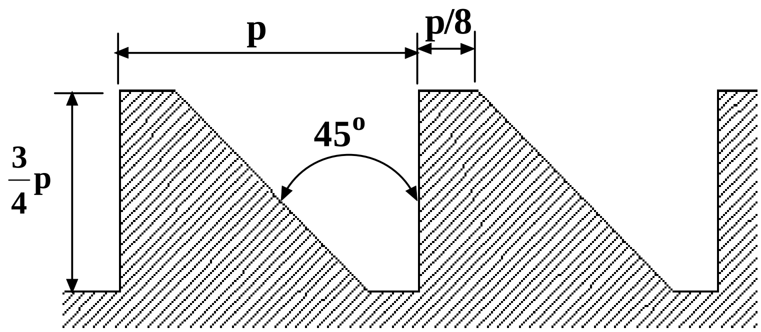
According to IS7008-1973 trapezoidal threads may be designated as, for example, Tr 50 x 8 which indicates a nominal diameter of 50 mm and a pitch of 8 mm.

Buttress thread

This thread form can also be used for power screws but they can transmit power only in one direction. Typical applications are screw jack, vices etc. A Buttress thread form is shown in **figure- 6.1.1.4**. and the proportions are shown in the figure in terms of the pitch.

On the whole the square threads have the highest efficiency as compared to other thread forms but they are less sturdy than the trapezoidal thread forms and the adjustment for wear is difficult for square threads.

When a large linear motion of a power screw is required two or more parallel threads are used. These are called multiple start power drives.



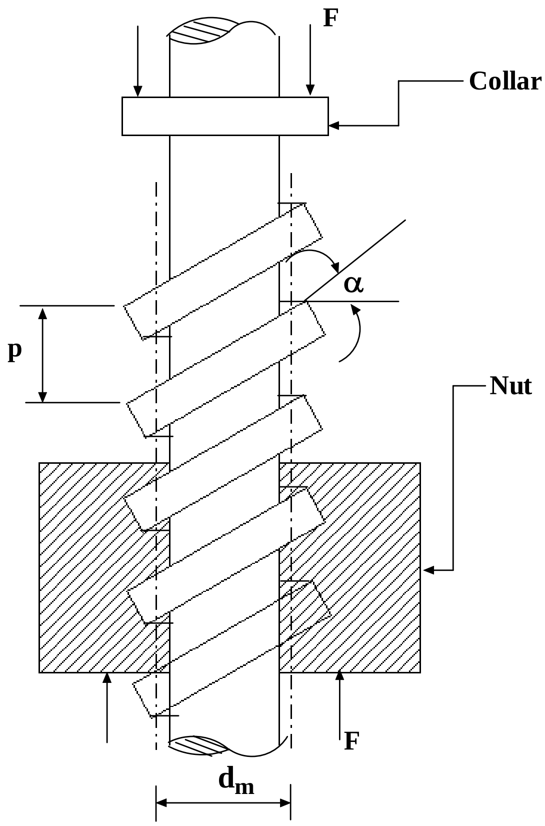
6.1.1.4F *– Some details of a Buttress thread form*

6.1.2 Efficiency of a power screw

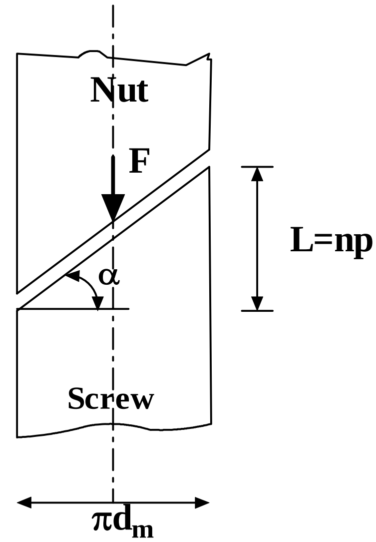
A square thread power screw with a single start is shown in **figure-6.1.2.1**. Here p is the pitch, α the helix angle, d_m the mean diameter of thread and F is the axial load. A developed single thread is shown in **figure- 6.1.2.2** where L = n p for a multi-start drive, n being the number of starts. In order to analyze the mechanics of the power screw we need to consider two cases:

(a) Raising the load

(b) Lowering the load.



6.1.2.1F *– A square thread power screw*



6.1.2.2F*- Development of a single thread*

Raising the load

This requires an axial force P as shown in **figure- 6.1.2.3**. Here N is the normal reaction and μN is the frictional force.

For equilibrium

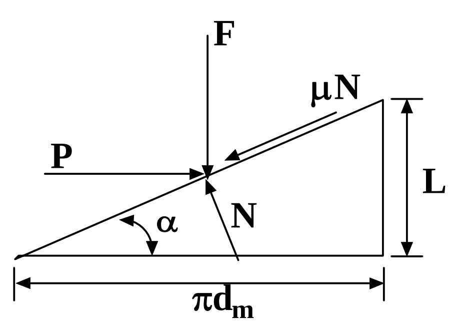
$P - \mu N \cos \alpha - N \sin \alpha = 0$

$F + \mu N \sin \alpha - N \cos \alpha = 0$

This gives

$$N = F/(\cos\alpha - \mu\sin\alpha)$$

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



6.1.2.3 F*- Forces at the contact surface for raising the load.*

Torque transmitted during raising the load is then given by

$$T_R = P\frac{d_m}{2} = F\frac{d_m}{2}\frac{(\mu\cos\alpha + \sin\alpha)}{(\cos\alpha - \mu\sin\alpha)}$$

Since $\tan\alpha = \frac{L}{\pi d_m}$ we have

$$T_R = F\frac{d_m}{2}\frac{(\mu\pi d_m + L)}{(\pi d_m - \mu L)}$$

The force system at the thread during lowering the load is shown in **figure- 6.1.2.4**. For equilibrium

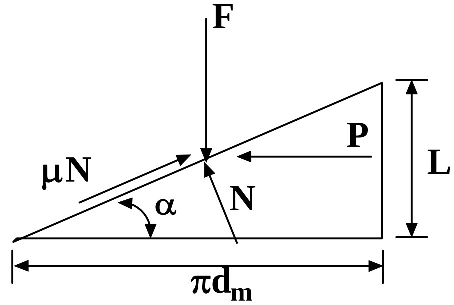
P - μ N cos α + N sin α = 0

F - N cos α -μ N sin α = 0

This gives

$$N = F/(\cos\alpha + \mu\sin\alpha)$$

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



6.1.2.4F*- Forces at the contact surface for lowering the load.*

Torque required to lower the load is given by

$$T_L = P\frac{d_m}{2} = F\frac{d_m}{2}\frac{(\mu\cos\alpha - \sin\alpha)}{(\cos\alpha + \mu\sin\alpha)}$$

And again taking $\tan\alpha = \frac{L}{\pi d_m}$ we have

$$T_L = F\frac{d_m}{2}\frac{(\mu\pi d_m - L)}{(\pi d_m + \mu L)}$$

Condition for self locking

The load would lower itself without any external force if

$\mu\pi d_m < L$

and some external force is required to lower the load if

$\mu\pi d_m \geq L$

This is therefore the **condition for self locking**.

Efficiency of the power screw is given by

$$\eta = \frac{\text{Work output}}{\text{Work input}}$$

Here work output = F. L

Work input = p. πd_m

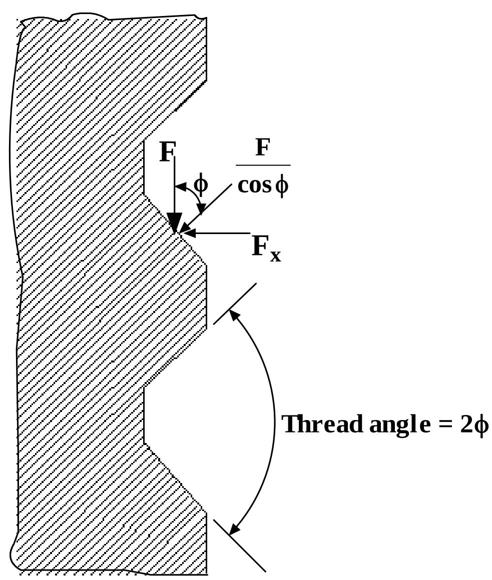
This gives

$$\eta = \frac{F}{P}\tan\alpha$$

The above analysis is for square thread and for trapezoidal thread some modification is required. Because of the thread angle the force normal to the thread surface is increased as shown in **figure- 6.1.2.5.** The torque is therefore given by

$$T = F\frac{d_m}{2}\frac{\left(\mu\pi d_m \sec\phi + L\right)}{\left(\pi d_m - \mu L \sec\phi\right)}$$

This considers the increased friction due to the wedging action. The trapezoidal threads are not preferred because of high friction but often used due to their ease of machining.



*6.1.2.5 **F** – Normal force on a trapezoidal thread surface*

Bursting effect on the nut

Bursting effect on the nut is caused by the horizontal component of the axial load F on the screw and this is given by (**figure- 6.1.2.5**)

$F_x = F \tan \phi$

For an ISO metric nut $2\phi = 60^o$ and $F_x = 0.5777$ F.

Collar friction

If collar friction μ_c is considered then another term $\mu F d_c/2$ must be added to torque expression. Here d_c is the effective friction diameter of the collar. Therefore we may write the torque required to raise the load as

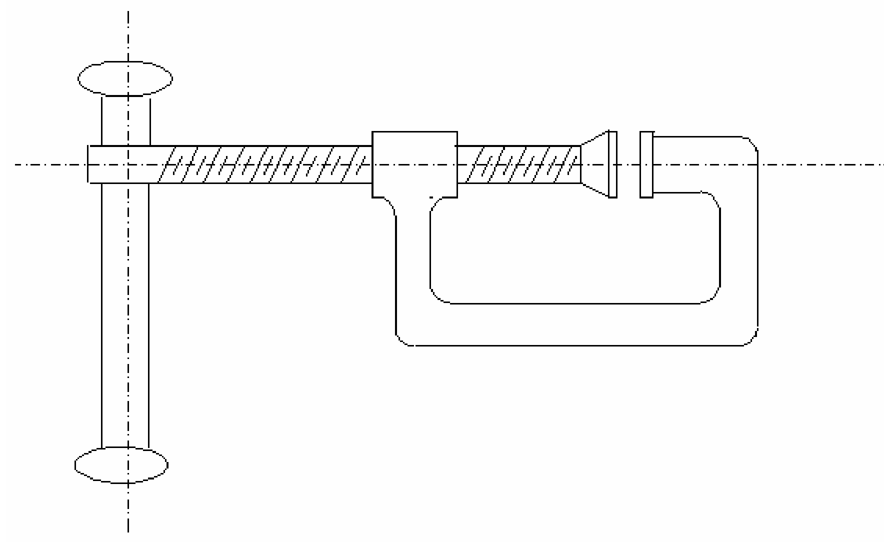
$$T = F\frac{d_m}{2}\frac{(\mu\pi d_m + L)}{(\pi d_m - \mu L)} + \mu_c F\frac{d_c}{2}$$

6.1.3 Problems with answers

Q.1: The C-clamp shown in **figure-6.1.3.1** uses a 10 mm screw with a pitch of 2 mm. The frictional coefficient is 0.15 for both the threads and the collar. The collar has a frictional diameter of 16 mm. The handle is made of steel with allowable bending stress of 165 MPa. The capacity of the clamp is 700 N.

(a) Find the torque required to tighten the clamp to full capacity.

(b) Specify the length and diameter of the handle such that it will not bend unless the rated capacity of the clamp is exceeded. Use 15 N as the handle force.



6.1.3.1 F*- A typical C- clamp.*

A.1.

(a) Nominal diameter of the screw, d = 10 mm.

Pitch of the screw, p = 2 mm.

Choosing a square screw thread we have the following dimensions:

Root diameter, $d_3 = d_{nominal} - 2h_3$ = 7.5 mm (since a_c = 0.25 mm and $h_3 = 0.5p + a_c$)

Pitch diameter, $d_2 = d_{nominal} - 2z$ = 8 mm. (since z = 0.5 p)

Mean diameter, d_m = (7.5+8)/2 = 7.75 mm.

Torque, $T = F\frac{d_m}{2}\frac{(\mu\pi d_m + L)}{(\pi d_m - \mu L)} + \mu_c F\frac{d_c}{2}$

Here F = 700 N, $\mu = \mu_c$ = 0.15, L = p = 2 mm (assuming a single start screw thread) and d_c = 16 mm. This gives T = 1.48 Nm.

Equating the torque required and the torque applied by the handle of length L we have 1.48 = 15 L since the assumed handle force is 15 N. This gives L= 0.0986 m. Let the handle length be 100 mm.

The maximum bending stress that may be developed in the handle is

$\sigma = \frac{My}{I} = \frac{32M}{\pi d^3}$ where d is the diameter of the handle.

Taking the allowable bending stress as 165 MPa we have

$$d = \left(\frac{32M}{\pi\sigma_y}\right)^{1/3} = \left(\frac{32x1.48}{\pi x165x10^6}\right)^{1/3} = 4.5x10^{-3}m = 4.5\ mm$$

With a higher factor of safety let d = 10 mm.

Q.2. A single square thread power screw is to raise a load of 50 KN. A screw thread of major diameter of 34 mm and a pitch of 6 mm is used. The coefficient of friction at the thread and collar are 0.15 and 0.1 respectively. If the collar frictional diameter is 100 mm and the screw turns at a speed of 1 rev s^{-1} find

(a) the power input to the screw.

(b) the combined efficiency of the screw and collar.

A.2.

(a) Mean diameter, $d_m = d_{major} - p/2 = 34\text{-}3 = 31$ mm.

Torque $T = F\frac{d_m}{2}\frac{(\mu\pi d_m + L)}{(\pi d_m - \mu L)} + \mu_c F\frac{d_c}{2}$

Here F = 5 x 10^3 N, d_m = 31 mm, μ = 0.15, μ_c = 0.1, L = p = 6 mm and d_c = 100 mm

Therefore $T = 50x10^3 x \frac{0.031}{2}\left(\frac{0.15\pi x 0.031 + 0.006}{\pi x 0.031 - 0.15x0.006}\right) + 0.1x50x10^3 x \frac{0.1}{2}$

= 416 Nm

Power input = T ω = 416 x 2π x 1 = 2613.8 Watts.

(b) The torque to raise the load only (T_0) may be obtained by substituting $\mu = \mu_c = 0$ in the torque equation. This gives

$$T_0 = F\frac{d_m}{2}\left(\frac{L}{\pi d_m}\right) = \frac{FL}{2\pi} = \frac{50x10^3 x0.006}{2\pi} = 47.75$$

Therefore $\eta = \frac{FL/2\pi}{T} = \frac{47.75}{416} = 0.1147 \ \text{ i.e. } 11.47\%$

6.1.4 Summary

Power screw drive in machinery is firstly discussed and some details of the thread forms used in such drives are given. The force system at the contact surface between the screw and the nut is analyzed and the torque required to raise and lower a load, condition for self locking and the efficiency of a power screw are derived. Typical problems on power screw drives are taken up and discussed.