POWER SCREWS

31.1. INTRODUCTION :

Power screw is a machine member consisting of large sized screw and nut and is employed for converting rotary motion into translating motion, (hence called as ***translation screw***). Also since this type of screws exert some force and power, they are specified as power screws.

The most typical fields of application of power screw drives are for raising loads (i.e., ***screw jacks***), for loading the specimens in testing machines, for obtaining the required motions during machining as in the case of lathe and screw presses etc. Based on the places of applications, rotary motion is imposed upon the screw for obtaining axial movement of nut (e.g. ***Lead screw of*** a lathe) or rotary motion to the nut for receiving axial movement of screw (e.g. screw jack). Power screws can also be used in ***vices, robots*** and ***sluice gate of dam etc.***

31.2. MATERIALS :

In power-screw drives, the power is transmitted from screw to nut or from nut to screw by the sliding contact and hence the material, of which they can be made should have low coefficient of friction in order to reduce wear, heat and power loss due to friction. For fulfilling the above requirement, usually heterogeneous materials may be adopted for making power screw drive components similar to worm gear drive. In practice, screw is made of hardened steel and the nut is made of cast-iron or bronze.

Sometimes power is transmitted in power-screw by rolling friction. In such arrangement, the external load is transmitted from screw to nut or nut to screw through a number of steel balls interposed between the concave helical races made in screw and nut. This set up reduce the frictional losses by sliding contact. The ball bearing screw drives are mainly preferred for ***automatic door closers, power*** *actuators* etc. The schematic diagrams for sliding contact and rolling contact power screws are shown in figure 31.1.

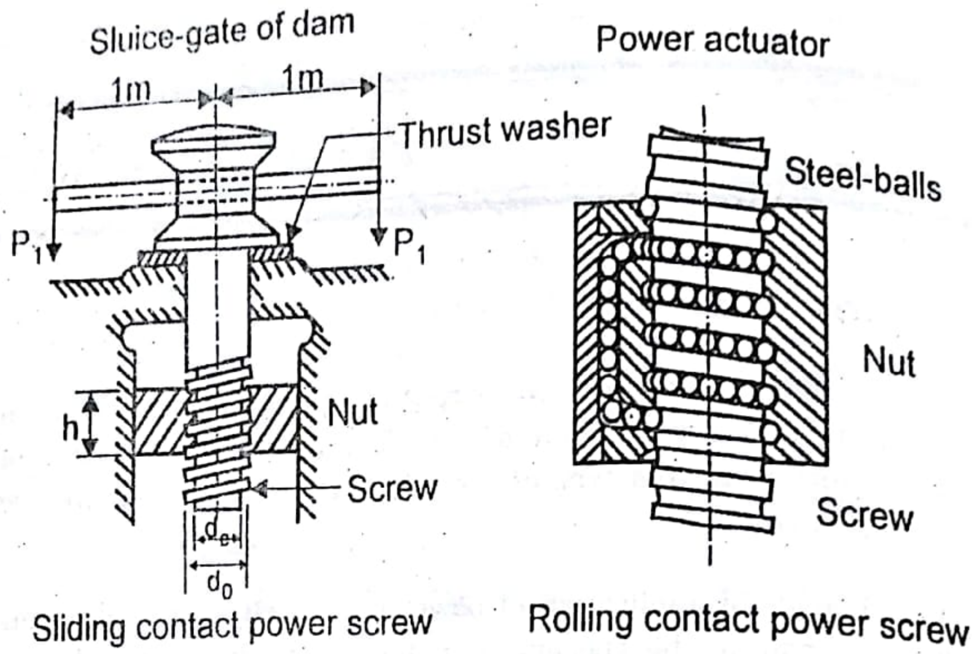

Fig. 31.1 Types of power screws

31.3. TYPES OF POWER-SCREW THREADS :

In contrast to fastening screws, requiring increased reliability against unintentional unscrewing, power screws require low friction. Consequently, screws with a smaller angle of thread may be preferred. For making power screws, three types of threads are commonly employed such as,

a) Square thread.

b) Trapezoidal thread

c) Buttress thread.

as shown in figure 31.2.

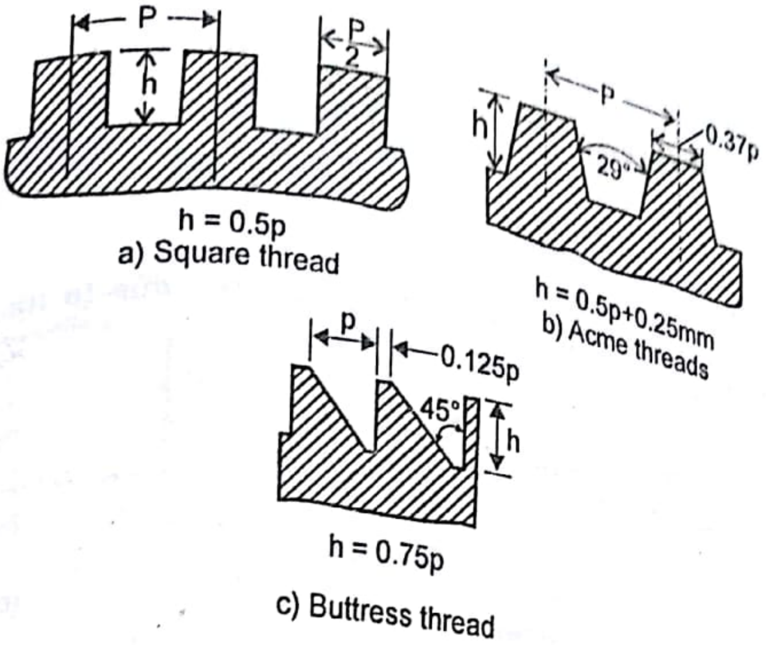

Fig. 31.2: Types of power-screw threads

Each thread is having its own merits. Among the above three types, square thread is considered for the transmission of power in either direction. It possess high efficiency and is employed for making lead screw of lathe, radial drilling machine column screw, screw jack etc.

Trapeazoidal or Acme thread can be produced easily than square thread but its efficiency is lower. They can be applied for making lead-screw of lathe and some vices etc.

Buttress threads are used for loading in one direction only. Due to their specific construction, if the screw is operated in the opposite direction, the applied load can easily be released. Hence this thread finds applications mainly in screw jack. The efficiency of buttress thread is more comparing to square thread and Acme thread.

The above threads may be formed into single start, double start and triple start threads in order to improve their efficiency and operating speed.

31.4. NOMENCLATURE OF SCREW THREAD :

1. Pitch (P) : It is the distance, measured parallel to the axis, between corresponding points of two adjacent threads.

2. Lead (L) : It is the axial distance between the corresponding points of two adjacent threads in the same helix. It is equal to the distance moved by a screw in the axial direction for one revolution of the screw. In the multi-start thread, lead is equal to the product of pitch by the number of starts.

i.e., Lead = 1 × Pitch (for single start thread)

= 2 × Pitch (for double start thread)

and so on.

3. Lead angle (α) :

It is the angle for the axial movement of the screw due to its one revolution. That is,

$$\alpha = \tan^{-1}\frac{\text{Lead}}{\text{Circumference}} = \tan^{-1}\frac{L}{\pi d}$$

where L = Lead

d = Pitch circle diameter of screw

4. Tip diameter (d_0) : It is the diameter of a co-axial cylinder that would just touch the crest of an external thread as in the case of bolt or root of an internal thread as in the case of nut. It is the maximum diameter of screw also called as ***nominal diameter*** or ***outside diameter*** and is used for designating the thread.

5. Root diameter (d_c) :

It is the diameter of a co-axial cylinder that would just touch the root of an external thread (i.e., bolt) and crest of an internal thread (nut). This is the minimum diameter of the screw and is also called as ***core diameter***.

6. Pitch diameter (d) :

It is the diameter of a co-axial cylinder the surface of which would pass through the threads at such points which make the width of threads equal to the width of space between threads. It is also called as ***effective diameter***.

31.5. WORKING PRINCIPLE OF POWER-SCREW DRIVE :

The axial movement of the nut (or load) for one revolution of screw is similar to raising the load through an inclined plane for a horizontal length equal to the circumference of the screw. Hence for the design analysis of power screws, the working principle of inclined plane may be adopted.

Let us consider an inclined plane at an angle (α) with the horizontal whose length is equal to the circumference of screw as shown in figure 31.3. Also assume a load (W) is raised along that inclined plane by a horizontal force (F). Let R be the normal reaction.

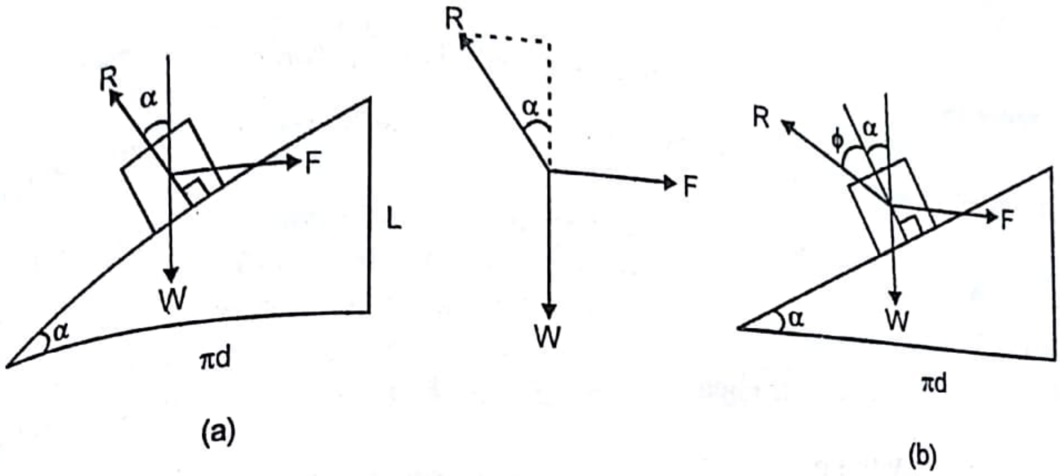

Fig. 31.3

If there is no friction, then, for equilibrium condition,

$F = R \sin \alpha$

and $W = R \cos \alpha$

Hence $F = \dfrac{W}{\cos \alpha} \sin \alpha = W \tan \alpha$ (Fig. 31.3(a))

In practice, there will always be some frictional force which acts in the opposite direction to applied force.

Hence due to friction, the reaction force will not be perpendicular to the inclined plane, but at an angle ϕ with the normal to the incline plane for this condition,

$F = R \sin (\alpha + \phi)$

and $W = R \cos (\alpha + \phi)$

Therefore $F = W \tan (\alpha + \phi)$ (Fig. 31.3(b))

In the above formula, $\alpha = \tan^{-1} \dfrac{L}{\pi d}$

and the angle of friction $\phi = \tan^{-1} \mu$ where μ is the coefficient of friction, L is lead and d is the pitch diameter of thread.

In power screw, the force on the inclined plane to raise the load is given in the form of torque and hence, the torque required to raise the load is given by,

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

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

If there is no friction, then, torque required, $T_r = \frac{Wd}{2} \tan \alpha$

Efficiency of power screw,

$$\eta = \frac{\text{Ideal effort}}{\text{Actual effort}}$$

$$= \frac{\text{Force for nil friction}}{\text{Force with friction}}$$

$$= \frac{W \tan \alpha}{W \tan(\alpha + \phi)}$$

$$= \frac{\tan \alpha}{\tan(\alpha + \phi)}$$

Similarly we can determine the force required to lower the load along the inclined plane.

Refer figure 31.4

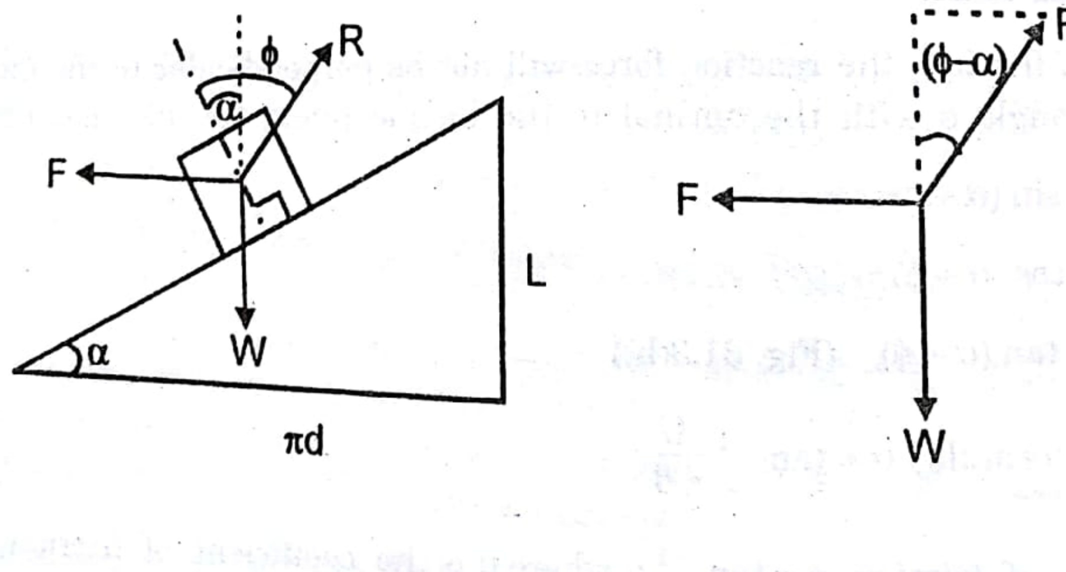

Fig 31.4

Now $F = R \sin(\phi - \alpha)$

and $W = R \cos(\phi - \alpha)$

Therefore $F = W \tan(\phi - \alpha)$

and the torque required to lower the load $T_l = \frac{Wd}{2} \tan(\phi - \alpha)$

From the above formula, we can understand that, if there is no friction, then $\phi = 0$ and thus the force F will become negative. That is, without any effort, the load will be lowered. Such a condition of lowering the load without any effect is known as ***overhauling***. *For self locking condition, (i.e., requiring some force for lowering the load), the friction angle ϕ should be always be greater than lead angle α.*

31.6. DESIGN CONSIDERATION FOR POWER SCREWS :

When designing the power-screw drive, the following factors should be considered.

1. Since the power-screw drive transmits power through sliding friction, its components such as screw and nut should be made of heterogenous materials (i.e., different materials). It is advisible that the *screw may be made of steel and the nut may be made of cast-iron or bronze.*

2. Even though the screw and nut are subjected to different types of stresses, attention must be given to nut design, because the nut material is softer than screw materials. The reason for selecting softer materials for nut is to allow the wear due to sliding contact on the nut only since the replacement of nut is easier and less costly than screw rod.

3. In order to reduce heat due to friction, sufficient lubrication must be provided.

4. In the case of screw drive for raising load such as screw jack the screw rod should have sufficient size in order to escape from buckling under heavy load.

5. If the screw rod is longer one, then proper supporting arrangements will be provided.

31.7. DESIGN OF POWER SCREWS :

When designing the power screws, we should take care of the various stresses induced in the screw rod and the nut. When the power is transmitted from screw to nut or nut to screw, the following stresses are induced in the screw and nut.

Stresses induced in the screw rod : (Refer figure 31.5)

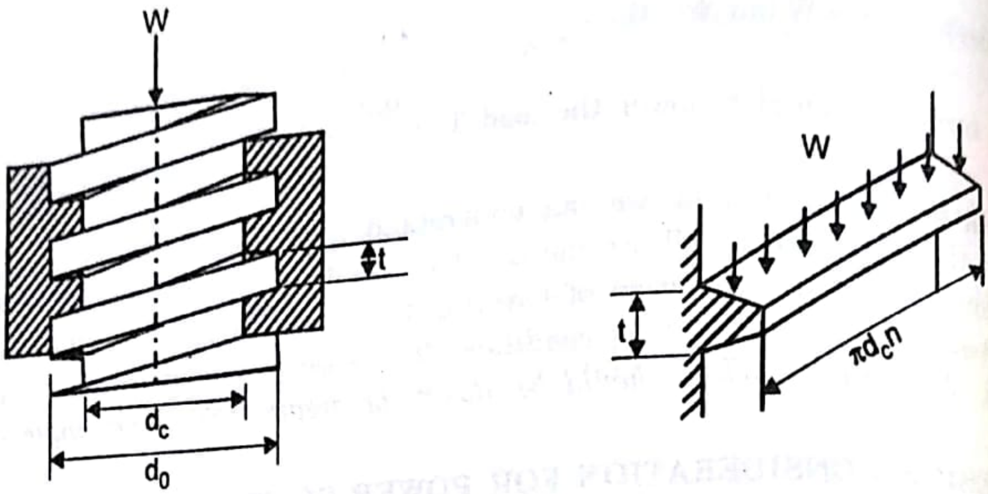

Fig. 31.5: Axial loaded power screw

Let d_0 = Outer (or tip) diameter of screw rod.

d_c = Inner (or core) diameter of screw rod.

d = Pitch diameter of screw rod.

t = Thickness of each thread.

n = Number of threads in contact with nut.

W = Axial load

There is no unique application of power screw. It can be operated to lift the load (e.g. screw jack) or to clamp the workpiece (e.g. vices) or to move the tool during machining (e.g. lead screw of lathe). Hence based on the type of operations, we should consider the concerned stresses. ***Some of such stresses induced in the screw rod are as follows.***

a) Tensile or compressive stress (i.e., normal stress) induced in the body of screw rod due to axial (i.e., direct) load,

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

If the length of rod is too great, then there is a chance for buckling under compressive load and in this situation, design should be based on column theory using Rankine's formula, according to which the induced stress is given by

$$S_n = \frac{W\left[1 + a\left(\frac{L}{K}\right)^2\right]}{\frac{\pi}{4} d_c^2} = \frac{4\,W\left[1 + a\left(\frac{L}{K}\right)^2\right]}{\pi\, d_c^2}$$

where 'a' is Rankine's constant, which is equal to 1/7500 for steel.

and $L =$ Equivalent length of screw rod.

$= 2\,l$ (for the column whose one end is fixed and other end is free)

$l =$ A-tual length of screw rod

$k =$ Lea t radius of gyration

Note : According to Rankine's formula, the buckling or crippling load is given by

$$F_{cr} = \frac{S_c \cdot A}{1 + a\left(\frac{L}{K}\right)^2}$$

In actual practices, the core diameter is first determined by considering the screw under simple compression and then it is checked for buckling for stability of the screw.

b) Torsional shear stress induced in the body of screw-rod

$$S_s = \frac{16\,T}{\pi\, d_c^3} \text{ where } T = \frac{W\,d}{2}\tan(\alpha + \phi)$$

Since the screw rod is subjected to both direct and shear stresses, there will be maximum (i.e., equivalent) axial stress and shear stress acting on the body which can be evaluated using maximum principal stress theory and maximum shear stress theory. According to them, the maximum principal stress.

$$S_{p1} = \frac{1}{2}\left[S_n + \sqrt{S_n^2 + 4\,S_s^2}\right]$$

and the maximum shear stress.

$$S_{sm} = \frac{1}{2}\sqrt{S_n^2 + 4\,S_s^2}$$

c) Direct shear stress induced in the threads of screw-rod,

$$\tau_s = \frac{W}{n \pi d_c t}$$

d) Bearing or crushing stress induced in the threads,

$$P_b = \frac{W}{n \frac{\pi}{4}(d_0^2 - d_i^2)} = \frac{4W}{n \pi (d_0^2 - d_i^2)}$$

Similarly, the stresses induced in the nut are as follows.

a) Direct shear stress,

$$\tau_n = \frac{W}{n \pi D_0 t}; \quad D_0 = \text{Major diameter of nut.}$$

b) Bearing stress,

$$P_b = \frac{W}{n \frac{\pi}{4}(d_0^2 - d_i^2)} = \frac{4W}{n \pi (d_0^2 - d_i^2)}$$

$$\text{(or)}\ P_b = \frac{W}{n \pi d t} \qquad \therefore \frac{(d_0^2 - d_i^2)}{4} = \left(\frac{d_0 + d_i}{2}\right)\left(\frac{d_0 - d_i}{2}\right)$$

$$= d \cdot \frac{p}{2} = d.t$$

where d = Pitch diameter.

p = Pitch

t = Thickness of square thread

For safe and optimum design, the above induced stresses should be less than their design (i.e. allowable) values.

For the design reference, the basic dimensions of square thread, allowable bearing pressure, coefficient of friction are given in tables 31.1, 31.2 and 31.3 respectively.

Table 31.2: Allowable Bearing pressure [P_b]

Sl.No.	Material combination	Allowable Bearing pressure (P_b) N/mm^2
1.	Steel screw-Bronze nut	12
2.	Steel screw-Cast iron nut	8
3.	Steel screw-Bronze nut for machine tool lead screws	3

Table 31.3 Coefficient of friction (μ)

Power screw Material combination	Sliding velocity (m/s)			
	0.1 m/s	0.5 m/s	1.0 m/s	2.0 m/s
Steel screw and Bronze nut	0.09	0.065	0.055	0.045
Steel screw and Cast-iron nut.	0.106	0.082	0.072	0.067

Note : The design yield stress for steel in tension or compression will be taken as 300 to 400 N/mm^2 and the design yield stress in shear will be approximately equal to one half of design yield stress in tension or compression.

31.8. CALCULATION OF THREAD WEAR :

As we know that, the thread wear is allowed to nut only in order for reducing the cost of replacement, the amount of bearing pressure induced in the nut, which causes for wear, have to be evaluated and is to be checked with its allowable values.

Consider a nut of power screw drive in which the bearing pressure is given by,

$$P_b = \frac{W}{n \pi d t} \text{ [Already derived]}$$

where W = Axial load,

n = Number of threads in contact

d = Pitch circle diameter of nut

t = Thickness of thread

From the above single formula, using W and P_b we can not determine three unknown parameters such as n, d, and t. Hence the following method can be adopted

Let H = Height of nut (i.e., Nut thickness)

S = Lead of nut

Z = number of starts

t_e = Depth of thread engagement

Now, $$n = \frac{H}{\left(\frac{S}{Z}\right)} = \frac{HZ}{S}$$

$$\therefore P_b = \frac{W}{n \pi d t_e} = \frac{WS}{\pi d t_e HZ}$$

For square thread, $t_e = \frac{1}{2} \times \text{pitch} = \frac{1}{2} \times \frac{S}{Z}$

Now $$P_b = \frac{2W}{\pi d H}$$

For safe design, this induced stress should be less than allowable value.

i.e., $$\frac{2W}{\pi d H} \leq [P_b]$$

Now take $\frac{H}{d} = \psi =$ Nut height factor

$= 1.2$ to 2.5 for solid nuts

$= 2.5$ to 3.5 for split nuts

Then $$\frac{2W}{\pi \psi d^2} \leq [P_b]$$

(or) $$d \geq \sqrt{\frac{2W}{\pi \psi [P_b]}}$$

After determining the pitch circle diameter d, the next standard dimension of d and other proportions of screw and nut may be selected from the table 31.1.

31.9. CALCULATION OF COLLAR FRICTION IN SCREW JACK :

When the axial load is taken up by a thrust collar as shown in figure 31.6 so that the load does not rotate with screw, then the torque required to overcome friction at the collar is given by

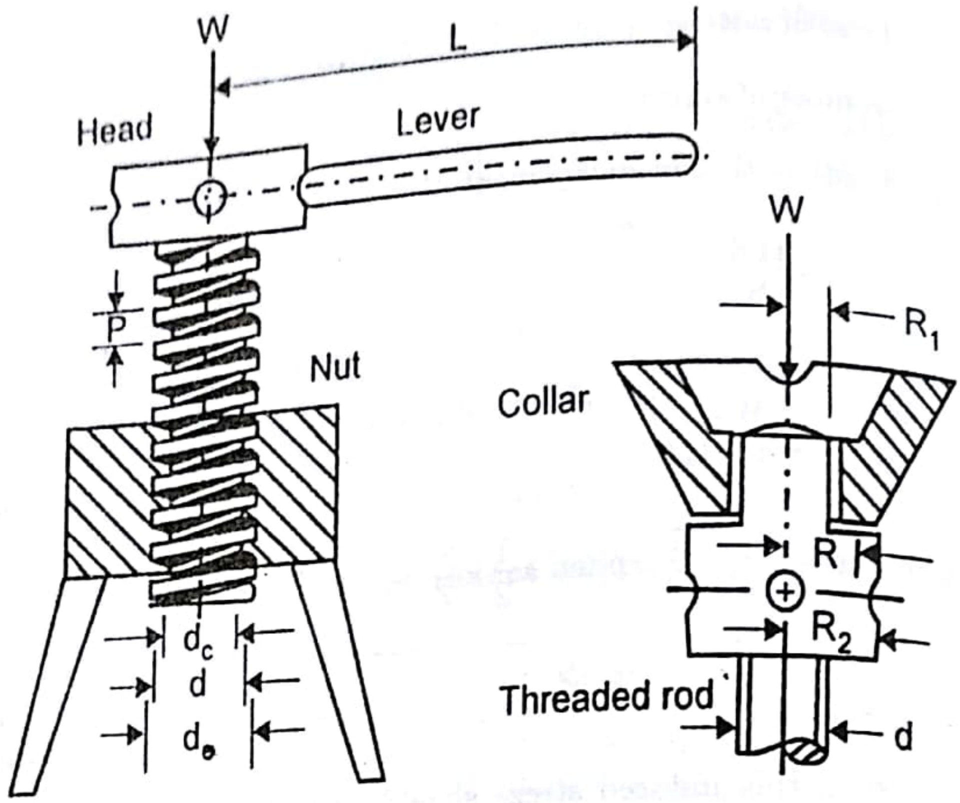

Fig. 31.6: Collar friction

$T = \mu_c W R$ (Assuming uniform wear condition)

(or) $T = \frac{2}{3} \mu_c W \left[\frac{R_1^3 - R_2^3}{R_1^2 - R_2^2} \right]$ (Assuming uniform pressure condition)

Where R_1 and R_2 = Outside and inside radii of collar.

R = Mean radius $= \frac{R_1 + R_2}{2}$

μ_c = Coefficient of friction for the collar.

31.10. DESIGN PROCEDURE :

The principal reason for power screw failure is due to thread wear which is occured by crushing (i.e., by bearing load). Hence the primary consideration should be given to design bearing pressure. For the design of power screw, the following procedure may be adopted.

1. From the given data, note down the axial load, operating speed, power etc.
2. Select suitable materials for screw and nut. Usually for normal duty applications, C45 steel may be used for screw rod and bronze for nut. For

heavy duty applications, steel is employed for screw and cast-iron is used for nut.

3. Determine the pitch circle diameter of nut by wear consideration with respect to allowable bearing stress given in table 31.2, using the formula,

$$d \geq \left[\frac{2\,W}{\pi\,\psi\,[P_b]} \right]^{1/2}$$

4. Select proper standard dimensions of screw and nut from the table 31.1.

5. Check the screw for axial stress, direct shear stress, and torsional shear stress.

6. Also check the screw for stability condition if it is very long, i.e., check for screw buckling.

7. For this, find the buckling load using Euler's formula or Rankine's formula and find out stability factor as

$$\text{Stability factor} = \frac{\text{Buckling load}}{\text{Applied load}}.$$

If the above factor is more than 2.5, buckling will not occur.

8. Check the nut for induced shear stress.

9. Calculate the efficiency of the drive.

10. Draw the arrangement of power screw drive.

Note : ***In screw jack design, the collar friction may also be taken into account.***

Hence total torque, $T = T_1 + T_2$

Where $T_1 =$ Torque between screw and nut

$T_2 =$ Torque between screw and collar

31.11. SOLVED PROBLEMS :

Problem 31.1 :

Design a lead-screw and split nut for a lathe for the following specifications.

Maximum axial load = 10 KN

Operating speed of nut movement = 0.3 m/min

Stroke required = 1250 mm

Also calculate the efficiency and the power required to drive the screw.

Solution :

Given :

Axial load, $W = 10\text{ KN} = 10000\text{ N}$

Nut speed, $N = 0.3\text{ m/min}$

Stroke $= 1250$ mm.

Let the material for screw is C45 steel, and for nut is bronze. In practice split nut will be considered for operation. Assume the design stresses for screw and nut are as,

a)	For screw in tension or compression	$[S_{n1}]$	=	150 N/mm^2
b)	For screw in shear	$[S_{s1}]$	=	75 N/mm^2
c)	For nut in tension	$[S_{n2}]$	=	50 N/mm^2
d)	For nut in shear	$[S_{s2}]$	=	25 N/mm^2
e)	Bearing pressure for steel and bronze combination	$[P_b]$	=	3 N/mm^2

Also assume square thread for our design.

Since the thread failure is mainly by thread wear, design should start from bearing pressure.

Design of nut :

Based on bearing pressure, the pitch circle diameter of nut for square thread is given by the formula,

$$d \geq \sqrt{\frac{2\,W}{\pi\,\psi\,[P_b]}} \qquad \text{(JDB 8.9)}$$

Where $\psi =$ Nut height factor $= \dfrac{\text{Nut thickness}}{\text{Pitch diameter}}$

$= \dfrac{H}{d} = 3$ (Assume) (for split nut) (PSG 7.87)

$[P_b] = 3\ N/mm^2$ [Table -31.2, (PSG 7.87)]

$$\therefore\ d \geq \sqrt{\frac{2 \times 10000}{\pi \times 3 \times 3}} \geq 26.6\text{ mm}$$

Take next standard pitch diameter $= 27$ mm. Corresponding to this diameter, the other proportions are, Major diameter for screw, (i.e., bolt), $d_0 = 30$ mm

Minor diameter for screw, $d_c = 24$ mm

Major diameter for nut, $D_0 = 30.5$ mm

Minor diameter for nut, $D_c = d_c = 24$ mm

Pitch, $p = 6$ mm

Thickness (or) depth of thread, $t = 3$ mm

The induced shear stress in the nut is given by,

$$\tau_n = \frac{W}{n\,\pi\,D_0\,t}$$

where $n =$ Number of threads in contact

$$= \frac{\text{Nut thickness}}{\text{pitch}} = \frac{H}{p}$$

Now, $H = \psi \cdot d = 3 \times 27 = 81$

$$\therefore n = \frac{81}{6} = 13.5 = 14 \text{ (say)}$$

Corrected nut thickness, $H = n.p = 14 \times 6 = 84$ mm.

$$\text{Hence } \tau_n = \frac{10000}{14 \times \pi \times 30.5 \times 3} = 2.5 \text{ N/mm}^2 < [S_{s2}] = 25 \text{ N/mm}^2$$

Since the induced shear stress is less than its design stress, our design is safe.

Outer diameter of nut (D_1) may be determined considering the tearing strength of nut. We know that

$$W = \frac{\pi}{4}(D_1^2 - D_0^2) \cdot S_{n2}$$

i.e., $10000 = \frac{\pi}{4}(D_1^2 - 30.5^2)\,50$

$$D_1 = \left[\left(\frac{4 \times 10000}{\pi \times 50}\right) + 30.5^2\right]^{1/2} = 34.4 \text{ mm}$$

Take $D_1 = 35$ mm

Design of screw :

Length of screw rod, l = Stroke + clearance on both sides of screw

$$= 1250 + (2 \times 15) = 1280 \text{ mm.}$$

The various induced stresses in the screw are checked as follows.

Direct stress induced in the body of screw,

$$S_n = \frac{W}{\frac{\pi}{4} d_c^2} = \frac{4\,W}{\pi\, d_c^2} = \frac{4 \times 10000}{\pi \times 24^2}$$

$$= 22.1 \text{ N/mm}^2 < [S_{n1}] = 150 \text{ N/mm}^2$$

Checking for buckling :

Now, the ratio of length of screw to pitch diameter

$$= \frac{l}{d} = \frac{1280}{27} = 47.4$$

Since it is very high value, buckling may occur. The buckling load according to Rankine's formula,

$$F_{cr} = \frac{S_c \cdot A}{1 + a\left(\frac{L}{K}\right)^2}$$

where S_c = Compressive strength of screw

$= 330 \text{ N/mm}^2$ (Assumed)

$A = \frac{\pi}{4} d_c^2 = \frac{\pi}{4} \times 24^2 = 452 \text{ mm}^2$

a = Rankine's constant $= \frac{1}{7500}$ for steel

L = Equivalent length $= \frac{l}{2}$

$= \frac{1280}{2} = 640 \text{ mm}$

K = Least radius of gyration $= \sqrt{\frac{I}{A}}$

$$= \sqrt{\frac{\frac{\pi}{64} d_c^4}{\frac{\pi}{4} d_c^2}} = \frac{d_c}{4} = \frac{24}{4} = 6 \text{ mm.}$$

Substituting all the values, we get

$$F_{cr} = \frac{330 \times 452}{1 + \left(\frac{1}{7500}\right)\left(\frac{640}{6}\right)^2} = 59260 \text{ N}$$

Stability factor of safety $= \dfrac{\text{Crippling load}}{\text{Applied load}}$

$$= \frac{59260}{10000} = 5.9$$

Since this is more than minimum requirement (i.e., 2.5), buckling will not occur and hence our design is safe.

Calculation of efficiency :

Efficiency of screw drive, $\eta = \dfrac{\tan \alpha}{\tan (\alpha + \phi)}$

Where $\alpha =$ Lead angle of screw

$$= \tan^{-1}\left(\frac{\text{Lead}}{\text{circumference}}\right) = \tan^{-1}\left(\frac{L}{\pi d}\right)$$

Assuming single start thread,

Lead $=$ Pitch $= 6$ mm

Hence, $\alpha = \tan^{-1}\left(\dfrac{6}{\pi \times 27}\right) = 4°$

$\phi =$ Friction angle $= \tan^{-1} \mu$

$= \tan^{-1} 0.1$ (Assuming coefficient of friction $\mu = 0.1$)

$= 5.7°$

$$\therefore \eta = \frac{\tan 4}{\tan (4 + 5.7)} = 0.41 = 41\%$$

Calculation of input power :

Power required, $P = \dfrac{2 \pi N T}{60}$ watts.

where $N = \text{rpm} = \dfrac{\text{Linear speed}}{\text{Lead}}$ [Linear speed $= 0.3$ m/min $= 300$ mm/min]

$$= \frac{300}{6} = 50 \text{ rpm}$$

T = Torque transmitted by the screw

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

$$= \frac{10000 \times 27 \times \tan(4 + 5.7)}{2} = 23076 \text{ N} - \text{mm}$$

$$= 23 \text{ N} - \text{m}$$

Now, power $P = \frac{2 \times \pi \times 50 \times 23}{60} = 120 \text{ W}$

Note : The input power can be calculated by another method.

Input power $= \frac{\text{Output power}}{\text{Efficiency}}$

Output power = Load × Linear speed

$$= 10000 \times \left(\frac{0.3}{60}\right) = 50 \text{ N} - \text{m/s}$$

$$= 50 \text{ W}$$

Input power $= \frac{50}{0.41} = 120 \text{ W}$

Specifications :

1.	Material for screw	=	C45 steel
2.	Material for nut	=	Bronze.
3.	Nominal (or major) diameter of screw	=	30 mm
4.	Major diameter of nut	=	30.5 mm
5.	Minor diameter of screw	=	24 mm
6.	Minor diameter of nut	=	24 mm
7.	Lead for screw and nut	=	6 mm
8.	Number of starts	=	1
9.	Type of thread	=	Square thread
10.	Efficiency of drive	=	41%
11.	Input power required	=	120 W
12.	Length of nut	=	84 mm
13.	Length of screw	=	1280 mm
14.	Lead screw speed	=	50 rpm

Problem 31.2 :

Design a screw jack to lift a load of 50 KN through 200 mm. The screw is made of C45 steel and the nut is made of cast-iron grade 20. Calculate the effort required to lift the load at a leverage of 500 mm.

Solution : (Refer the figure 31.7)

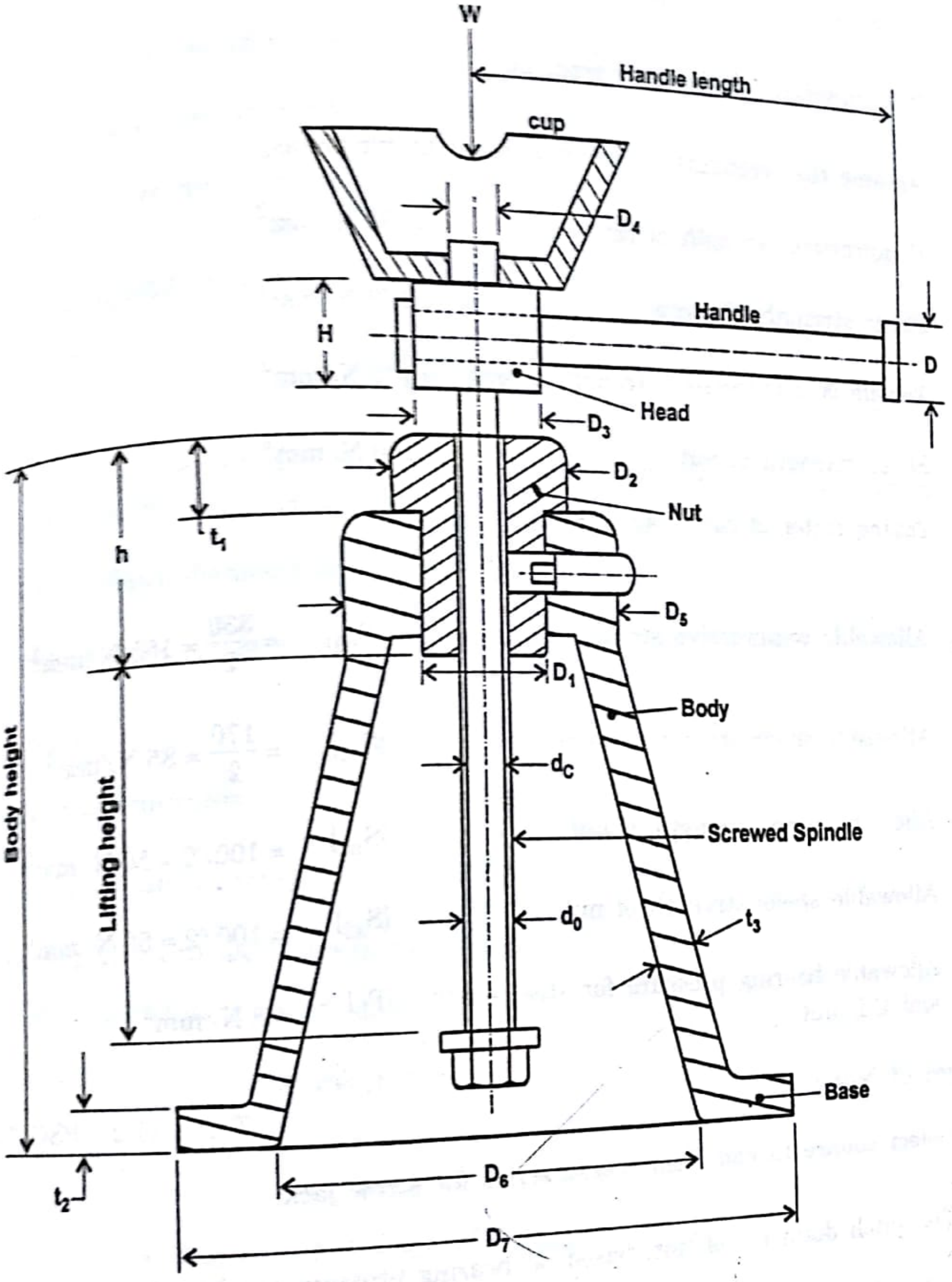

Fig. 31.7: Screw jack