UTS Faculty of Engineering and I. T.

49316 Materials Handling Assignment 4 – 2009

Investigate the claims of the attached article about the “Olds Elevator”

Read the article carefully noting how the Olds elevator may have advantages over its competitors.

In addition go to the website: http://www.oldselevator.com/
You may look at the machine in operation and obtain further helpful information from the downloads.

Consider the elevation of foundry sand to a height of 8 m at a rate of 10 tonnes per hour.

After reading the papers and viewing the website decide on a nominal diameter and a rotational speed for a suitable Olds elevator.

Assume all steel construction.

Using fundamental reasoning and calculations estimate the power requirement to drive the machine. Take into account digging power, friction power to overcome friction at all surfaces, elevation power and exit losses. Also make an allowance for power to operate the machine with no load.

You may find the attached notes on power screws helpful in devising your fundamental theory.

Comment on the proposed test rig at Wollongong.
A common requirement in mechanical design is to move components in a straight line. Elevators move vertically up or down. Machine tools move cutting tools or parts to be machined in straight lines, either horizontally or vertically, to shape metal into desired forms. A precision measuring device moves a probe in straight lines to determine electronically the dimensions of a part. Assembly machines require many straight-line motions to insert components and to fasten them together. A packaging machine moves products into cartons, closes flaps, and seals the cartons.

Some examples of components and systems that facilitate linear motion are:

<table>
<thead>
<tr>
<th>Power screws</th>
<th>Ball screws</th>
<th>Jacks</th>
<th>Fluid power cylinders</th>
</tr>
</thead>
<tbody>
<tr>
<td>Linear actuators</td>
<td>Linear slides</td>
<td>Ball bushings</td>
<td>Rack and pinion sets</td>
</tr>
<tr>
<td>Linear solenoids</td>
<td>Positioning stages</td>
<td>X-Y-Z tables</td>
<td>Gantry tables</td>
</tr>
</tbody>
</table>

Figure 17–1(a) shows a cutaway view of a jack that employs a power screw to produce linear motion. Power is delivered to the input shaft by an electric motor. The worm, machined integral with the input shaft, drives the wormgear, resulting in a reduction in the speed of rotation. The inside of the wormgear has machined threads that engage the external threads of the power screw, driving it vertically. Figure 17–1(b) shows the screw jack mated with an external gear reducer and motor to provide a complete linear motion system. Limit switches, position sensors, and programmable logic controllers can be used to control the motion cycle. These and other types of linear actuators can be seen on Internet sites 1–10.

Rack and pinion sets are discussed in Chapter 8. Fluid power actuators employ oil hydraulic or pneumatic fluid pressure to extend or retract a piston rod within a cylinder as discussed in textbooks on fluid power. Positioning stages, X-Y-Z tables, and gantry tables typically are driven by precision stepper motors or servomechanisms that allow precision location of components anywhere within their control volume. Linear solenoids are devices that cause a rodlike core to be extended or retracted as power is supplied to an electrical coil, producing rapid movement over small distances. Applications are seen in office equipment, automation devices, and packaging systems. See Internet site 11.
FIGURE 17–1 Examples of linear motion machine elements (Joyce/Dayton Corporation, Dayton, OH)

Linear slides and ball bushings are designed to guide mechanical components along a precise linear track. Low friction materials or rolling contact elements are used to produce smooth motion with low power required. See Internet sites 1, 3, 5, and 7–9.

Power screws and ball screws are designed to convert rotary motion to linear motion and to exert the necessary force to move a machine element along a desired path. Power screws operate on the classic principle of the screw thread and its mating nut. If the screw is supported in bearings and rotated while the nut is restrained from rotating, the nut will translate along the screw. If the nut is made an integral part of a machine, for example, the tool holder for a lathe, the screw will drive the tool holder along the bed of the machine to take a cut. Conversely, if the nut is supported while it is rotating, the screw can be made to translate. The screw jack uses this approach.

A ball screw is similar in function to a power screw, but the configuration is different. The nut contains many small, spherical balls that make rolling contact with the threads of the screw, giving low friction and high efficiencies when compared with power screws. Modern machine tools, automation equipment, vehicle steering systems, and actuators on aircraft use ball screws for high precision, fast response, and smooth operation.

Visit a machine shop where there are metal-cutting machine tools. Look for examples of power screws that convert rotary motion to linear motion. They are likely to be on manual lathes moving the tool holder. Or look at the table drive for a milling machine. Inspect the form of the threads of the power screw. Are they of a form similar to that of a screw thread with sloped sides? Or are the sides of the threads straight? Compare the screw threads with those shown in Figure 17–2 for square, Acme, and buttress forms.

While in the shop, do you see any type of material-testing equipment or a device called an arbor press that exerts large axial forces? Such machines often employ square-thread power screws to produce the axial force and motion from rotational input, through
either a hand crank or an electric motor drive. If they are not in the machine shop, look for them in the metallurgy lab or another room where materials testing is done.

Now look further in the machine shop. Are there machines that use digital readouts to indicate position of the table or the tool? Are there computer numerical control machine tools? Any of these types of machines should have ball screws rather than the traditional power screws because ball screws require significantly less power and torque to drive them against a given load. They can also be moved faster and positioned more accurately than power screws. You may or may not be able to see the recirculating balls in the nut of the power screw, as illustrated in Figure 17–3. But you should be able to see the different-shaped threads looking like grooves with circular bottoms in which the spherical balls roll.

Have you seen such power screws or ball screws anywhere outside a machine shop? Some garage door openers employ a screw drive, but others use chain drives. Perhaps your home has a screw jack or a scissors jack for raising the car to change a tire. Both use power screws. Have you ever sat in a seat on an airplane where you can see the mechanisms that actuate the flaps on the rear edge of the wings? Try it sometime, and observe the actuators during takeoff or landing. It is likely that you will see a ball screw in action.

This chapter will help you learn the methods of analyzing the performance of power screws and ball screws and to specify the proper size for a given application.
You are the Designer

You are a member of a plant engineering team for a large steel processing plant. One of the furnaces in the plant in which steel is heated prior to final heat treatment is installed beneath the floor, and the large ingots are lowered vertically into it. While the ingots are soaking in the furnace, a large, heavy hatch is placed over the opening to minimize the escape of heat and provide more uniform temperature. The hatch weighs 111.2 kN.

You are asked to design a system that will permit the hatch to be raised at least 381 mm above the floor within 12.0 s and to lower it again within 12.0 s.

What design concept would you propose? Of course, there are many feasible concepts, but suppose that you proposed a system like that sketched in Figure 17–4. An overhead support structure is suggested on which a worm/wormgear drive system would be mounted. One shaft would be driven directly by the gear drive while a second shaft would be driven simultaneously by a chain drive. The shafts are power screws, supported in bearings at the top and bottom. A yoke is connected to the hatch and is mounted on the screws with the nuts that mate with the screw integral with the yoke. Therefore, as the screw rotates, the nuts carry the yoke and the hatch vertically upward or downward.

As the designer of the hatch lift system, you must make several decisions. What size screw is required to ensure that it can safely raise the 111.2 kN hatch? Notice that the screws are placed in tension as they are supported on the collars on the upper support system. What diameter, thread type, and thread size should be used? The sketch suggests the Acme thread form. What other styles are available? At what speed must the screws be rotated to raise the hatch in 12.0 s or less? How much power is required to drive the screws? What safety concerns exist while the system is handling this heavy load? What advantage would there be to using a ball screw rather than a power screw?

The material in this chapter will help you make these decisions, along with providing methods of computing stresses, torques, and efficiencies.

**Figure 17–4** An Acme screw-driven system for raising a hatch

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**17–1 Objectives of This Chapter**

After completing this chapter, you will be able to:

1. Describe the operation of a power screw and the general form of square threads, Acme threads, and buttress threads as they are applied to power screws.
2. Compute the torque that must be applied to a power screw to raise or lower a load.
3. Compute the efficiency of power screws.
4. Compute the power required to drive a power screw.
5. Describe the design of a ball screw and its mating nut.
6. Specify suitable ball screws for a given set of requirements of load, speed, and life.
7. Compute the torque required to drive a ball screw, and compute its efficiency.

17–2

POWER SCREWS

Figure 17–2 shows three types of power screw threads: the square thread, the Acme thread, and the buttress thread. Of these, the square and buttress threads are the most efficient. That is, they require the least torque to move a given load along the screw. However, the Acme thread is not greatly less efficient, and it is easier to machine. The buttress thread is desirable when force is to be transmitted in only one direction.

Table 17–1 gives the preferred combinations of basic major diameter, \( D \), and number of threads per inch, \( n \), for Acme screw threads. The pitch, \( p \), is the distance from a point on one thread to the corresponding point on the adjacent thread, and \( p = 1/n \).

Other pertinent dimensions listed in Table 17–1 include the minimum minor diameter and the minimum pitch diameter of a screw with an external thread. When you are performing stress analyses on the screw, the safest approach is to compute the area corresponding to the minor diameter for tensile or compressive stresses. However, a more accurate stress computation results from using the tensile stress area (listed in Table 17–1), found from

\[
A_t = \frac{\pi}{4} \left[ \frac{D_r + D_p}{2} \right]^2
\]  
(17–1)

**TABLE 17–1** Preferred Acme screw threads

<table>
<thead>
<tr>
<th>Nominal major diameter, ( D ) (in)</th>
<th>Threads per in, ( n )</th>
<th>Pitch, ( p = 1/n ) (in)</th>
<th>Minimum minor diameter, ( D_n ) (in)</th>
<th>Minimum pitch diameter, ( D_p ) (in)</th>
<th>Tensile stress area, ( A_t ) (in²)</th>
<th>Shear stress area, ( A_s ) (in²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/4</td>
<td>16</td>
<td>0.0625</td>
<td>0.1618</td>
<td>0.2043</td>
<td>0.0263 32</td>
<td>0.3355 32</td>
</tr>
<tr>
<td>5/16</td>
<td>14</td>
<td>0.0714</td>
<td>0.2140</td>
<td>0.2614</td>
<td>0.0443 38</td>
<td>0.4344 38</td>
</tr>
<tr>
<td>3/8</td>
<td>12</td>
<td>0.0833</td>
<td>0.2632</td>
<td>0.3161</td>
<td>0.0659 89</td>
<td>0.5276 89</td>
</tr>
<tr>
<td>7/16</td>
<td>12</td>
<td>0.0833</td>
<td>0.3253</td>
<td>0.3783</td>
<td>0.0972 90</td>
<td>0.6396 90</td>
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<tr>
<td>1/2</td>
<td>10</td>
<td>0.1000</td>
<td>0.3594</td>
<td>0.4306</td>
<td>0.1225 10</td>
<td>0.7278 10</td>
</tr>
<tr>
<td>5/8</td>
<td>8</td>
<td>0.1250</td>
<td>0.4570</td>
<td>0.5408</td>
<td>0.1955 5</td>
<td>0.9180 5</td>
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<tr>
<td>3/4</td>
<td>6</td>
<td>0.1667</td>
<td>0.5371</td>
<td>0.6424</td>
<td>0.2732 8</td>
<td>1.084 8</td>
</tr>
<tr>
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<td>0.7663</td>
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<td>1.313 12</td>
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<tr>
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<td>5</td>
<td>0.2000</td>
<td>0.7509</td>
<td>0.8726</td>
<td>0.5175 15</td>
<td>1.493 15</td>
</tr>
<tr>
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<td>0.2000</td>
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<td>0.6881 18</td>
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<td>0.9998</td>
<td>1.1210</td>
<td>0.8831 21</td>
<td>1.952 21</td>
</tr>
<tr>
<td>1 1/2</td>
<td>4</td>
<td>0.2500</td>
<td>1.0719</td>
<td>1.2188</td>
<td>1.030 24</td>
<td>2.110 24</td>
</tr>
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<td>1 1/4</td>
<td>4</td>
<td>0.2500</td>
<td>1.1965</td>
<td>1.3429</td>
<td>1.266 27</td>
<td>2.341 27</td>
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<td>2</td>
<td>4</td>
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<td>2.454 33</td>
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<td>3</td>
<td>0.3333</td>
<td>1.8572</td>
<td>2.0450</td>
<td>2.982 36</td>
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<td>2.1065</td>
<td>2.2939</td>
<td>3.802 39</td>
<td>4.075 39</td>
</tr>
<tr>
<td>2 1/2</td>
<td>3</td>
<td>0.3333</td>
<td>2.3558</td>
<td>2.5427</td>
<td>4.711 42</td>
<td>4.538 42</td>
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<tr>
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<td>2</td>
<td>0.5000</td>
<td>2.4326</td>
<td>2.7044</td>
<td>5.181 45</td>
<td>4.757 45</td>
</tr>
<tr>
<td>3 1/8</td>
<td>2</td>
<td>0.5000</td>
<td>2.9314</td>
<td>3.2026</td>
<td>7.388 50</td>
<td>5.700 50</td>
</tr>
<tr>
<td>4</td>
<td>2</td>
<td>0.5000</td>
<td>3.4302</td>
<td>3.7008</td>
<td>9.985 55</td>
<td>6.640 55</td>
</tr>
<tr>
<td>4 1/2</td>
<td>2</td>
<td>0.5000</td>
<td>3.9291</td>
<td>4.1991</td>
<td>12.972 60</td>
<td>7.577 60</td>
</tr>
<tr>
<td>5</td>
<td>2</td>
<td>0.5000</td>
<td>4.4281</td>
<td>4.6973</td>
<td>16.351 65</td>
<td>8.511 65</td>
</tr>
</tbody>
</table>

*Per (inch) of length of engagement.*
This is the area corresponding to the average of the minor (or root) diameter, \( D_r \), and the pitch diameter, \( D_p \). The data reflect the minimums for commercially available screws according to recommended tolerances.

Another failure mode for a power screw is the shearing of the threads in the axial direction, cutting them away from the main shaft near the pitch diameter. The shear stress is computed from the direct stress formula,

\[
\tau = F/A_s
\]

The shear stress area, \( A_s \), listed in Table 17–1 is also found in published data and represents the area in shear approximately at the pitch line of the threads for a 25.4 mm length of engagement. Other lengths would require that the area be modified by the ratio of the actual length to 25.4 mm.

**Torque Required to Move a Load**

When using a power screw to exert a force, as with a jack raising a load, you need to know how much torque must be applied to the nut of the screw to move the load. The parameters involved are the force to be moved, \( F \); the size of the screw, as indicated by its pitch diameter, \( D_p \); the lead of the screw, \( L \); and the coefficient of friction, \( f \). Note that the lead is defined as the axial distance that the screw would move in one complete revolution. For the usual case of a single-threaded screw, the lead is equal to the pitch and can be read from Table 17–1 or computed from \( L = p = 1/n \).

In the development of Equation (17–2) for the torque required to turn the screw, Figure 17–5(a), which depicts a load being pushed up an inclined plane against a friction force, is used. This is a reasonable representation for a square thread if you think of the thread as being unwrapped from the screw and laid flat. The torque for an Acme thread is slightly different from this due to the thread angle. The revised equation for the Acme thread will be shown later.

The torque computed from Equation (17–2) is called \( T_u \), implying that the force is applied to move a load up the plane, that is, to raise the load. This observation is completely appropriate if the load is raised vertically, as with a jack. If, however, the load is horizontal or at some angle, Equation (17–2) is still valid if the load is to be advanced along the screw "up the thread." Equation (17–4) shows the required torque, \( T_d \), to lower a load or move a load "down the thread."

The torque to move a load up the thread is

\[
T_u = \frac{FD_p}{2} \left[ \frac{L + \pi fD_p}{\pi D_p - fL} \right]
\]

(17–2)
This equation accounts for the force required to overcome friction between the screw and the nut in addition to the force required just to move the load. If the screw or the nut bears against a stationary surface while rotating, there will be an additional friction torque developed at that surface. For this reason, many jacks and similar devices incorporate antifriction bearings at such points.

The coefficient of friction for use in Equation (17–2) depends on the materials used and the manner of lubricating the screw. For well-lubricated steel screws acting in steel nuts, \( f = 0.15 \) should be conservative.

An important factor in the analysis for torque is the angle of inclination of the plane. In a screw thread, the angle of inclination is referred to as the lead angle, \( \lambda \). It is the angle between the tangent to the helix of the thread and the plane transverse to the axis of the screw. It can be seen from Figure 17–5 that

\[
\tan \lambda = \frac{L}{\pi D_p} \tag{17–3}
\]

where \( \pi D_p \) = circumference of the pitch line of the screw

Then if the rotation of the screw tends to raise the load (move it up the incline), the friction force opposes the motion and acts down the plane.

Conversely, if the rotation of the screw tends to lower the load, the friction force will act up the plane, as shown in Figure 17–5(b). The torque analysis changes, producing Equation (17–4):

\[
T_d = \frac{FD_p}{2} \left[ \frac{\pi f D_p - L}{\pi D_p + f L} \right] \tag{17–4}
\]

If the screw thread is steep (that is, if it has a high lead angle), the friction force may not be able to overcome the tendency for the load to "slide" down the plane, and the load will fall due to gravity. In most cases for power screws with single threads, however, the lead angle is rather small, and the friction force is large enough to oppose the load and keep it from sliding down the plane. Such a screw is called self-locking, a desirable characteristic for jacks and similar devices. Quantitatively, the condition that must be met for self-locking is

\[
f > \tan \lambda \tag{17–5}
\]

The coefficient of friction must be greater than the tangent of the lead angle. For \( f = 0.15 \), the corresponding value of the lead angle is 8.5°. For \( f = 0.1 \), for very smooth, well-lubricated surfaces, the lead angle for self-locking is 5.7°. The lead angles for the screw designs listed in Table 17–1 range from 1.94° to 5.57°. Thus, it is expected that all would be self-locking. However, operation under vibration should be avoided, as this may still cause movement of the screw.

**Efficiency of a Power Screw**

*Efficiency* for the transmission of a force by a power screw can be expressed as the ratio of the torque required to move the load without friction to that with friction. Equation (17–2) gives the torque required with friction, \( T_w \). Letting \( f = 0 \), the torque required without friction, \( T' \), is

\[
T' = \frac{FD_p}{2} \left( \frac{L}{\pi D_p} \right) = \frac{FL}{2\pi} \tag{17–6}
\]

Then the efficiency, \( e \), is

\[
e = \frac{T'}{T_w} = \frac{FL}{2\pi T_w} \tag{17–7}
\]
Alternate Forms of the Torque Equations

Equations (17–2) and (17–4) can be expressed in terms of the lead angle, rather than the lead and the pitch diameter, by noting the relationship in Equation (17–3). With this substitution, the torque required to move the load would be

\[ T_u = \frac{FD_p}{2} \left[ \frac{(\tan \lambda + f)}{(1 - f \tan \lambda)} \right] \]  

(17–8)

and the torque required to lower the load is

\[ T_u = \frac{FD_p}{2} \left[ \frac{(f - \tan \lambda)}{(1 + f \tan \lambda)} \right] \]  

(17–9)

Adjustment for Acme Threads

The difference between Acme threads and square threads is the presence of the thread angle, \( \phi \). Note from Figure 17–1 that \( 2\phi = 29^\circ \), and therefore \( \phi = 14.5^\circ \). This changes the direction of action of the forces on the thread from that depicted in Figure 17–5. Figure 17–6 shows that \( F \) would have to be replaced by \( F \cos \phi \). Carrying this through, the analysis for torque would give the following modified forms of Equations (17–8) and (17–9). The torque to move the load up the thread is

\[ T_u = \frac{FD_p}{2} \left[ \frac{(\cos \phi \tan \lambda + f)}{(\cos \phi - f \tan \lambda)} \right] \]  

(17–10)

and the torque to move the load down the thread is

\[ T_u = \frac{FD_p}{2} \left[ \frac{(f - \cos \phi \tan \lambda)}{(\cos \phi + f \tan \lambda)} \right] \]  

(17–11)

Power Required to Drive a Power Screw

If the torque required to rotate the screw is applied at a constant rotational speed, \( \omega \), then the power in horsepower to drive the screw is

\[ P = T(n) \]  

(U.S.)

\[ = P_w \]  

(S.I.)

References 1 and 2 include more detail about the formulas that characterize the performance of power screws.
Example Problem 17–1

Two Acme-threaded power screws are to be used to raise a heavy access hatch, as sketched in Figure 17–4. The total weight of the hatch is 111.2 kN, divided equally between the two screws. Select a satisfactory screw from Table 17–1 on the basis of tensile strength, limiting the tensile stress to 68.95 MPa. Then determine the required thickness of the yoke that acts as the nut on the screw to limit the shear stress in the threads to 34.475 MPa. For the screw thus designed, compute the lead angle, the torque required to raise the load, the efficiency of the screw, and the torque required to lower the load. Use a coefficient of friction of 0.15.

Solution

The load to be lifted places each screw in direct tension. Therefore, the required tensile stress area is

\[ A_t = \frac{F}{\sigma_d} = \frac{55.6 \text{ kN}}{68.95 \times 10^6 \text{ N/m}^2} = 806.5 \text{ mm}^2 \]

From Table 17–1, a 38.1 mm diameter Acme thread screw with four threads per 25.4 mm would provide a tensile stress area of 816.8 mm².

For this screw, each 25.4 mm of length of a nut would provide 1510.4 mm² of shear stress area in the threads. The required shear area is then

\[ A_s = \frac{F}{\tau_d} = \frac{55.6 \text{ kN}}{34.475 \times 10^6 \text{ N/m}^2} = 1613 \text{ mm}^2 \]

Then the required length of the yoke would be

\[ h = 1613 \text{ mm}^2 \left( \frac{25.4 \text{ mm}}{1510.4 \text{ mm}^2} \right) = 27.18 \text{ mm} \]

For convenience, let’s specify \( h = 31.75 \text{ mm} \).

The lead angle is (remember that \( L = p = 1/n = 1/4 = 6.35 \text{ mm} \))

\[ \lambda = \tan^{-1} \left( \frac{L}{\pi D_p} \right) = \tan^{-1} \left( \frac{6.35}{\pi(34.11)} \right) = 3.39^\circ \]

The torque required to raise the load can be computed from Equation (17–10):

\[ T_u = \frac{FD_p}{2} \left[ \frac{(\cos \phi \tan \lambda + f)}{\cos \phi - f \tan \lambda} \right] \quad (17–10) \]

Using \( \cos \phi = \cos(14.5^\circ) = 0.968 \), and \( \tan \lambda = \tan(3.39^\circ) = 0.0592 \), we have

\[ T_u = \frac{(55.6 \text{ kN})(34.11 \text{ mm})}{2} \left[ \frac{(0.968)(0.0592) + 0.15}{0.968 - (0.15)(0.0592)} \right] = 204.4 \text{ N} \cdot \text{m} \]

The efficiency can be computed from Equation (17–7):

\[ e = \frac{FL}{2\pi T_u} = \frac{(55.6 \text{ kN})(6.35 \text{ mm})}{2(\pi)(204.4 \text{ N} \cdot \text{m})} = 0.275 \text{ or } 27.5\% \]

The torque required to lower the load can be computed from Equation (17–11):

\[ T_d = \frac{FD_p}{2} \left[ \frac{(f - \cos \phi \tan \lambda)}{\cos \phi + f \tan \lambda} \right] \quad (17–11) \]

\[ T_d = \frac{(55.6 \text{ kN})(34.11 \text{ mm})}{2} \left[ \frac{0.15 - (0.968)(0.0592)}{0.968 + (0.15)(0.0592)} \right] = 89.95 \text{ N} \cdot \text{m} \]
Example Problem 17–2

It is desired to raise the hatch in Figure 17–4 a total of 381 mm in no more than 12.0 s. Compute the required rotational speed for the screws and the power required.

Solution

The screw selected in the solution for Example Problem 17–1 was a 38.1 mm Acme-threaded screw with four threads per 25.4 mm. Thus, the load would be moved 6.35 mm with each revolution. The linear speed required is

\[ V = \frac{381 \text{ mm}}{12.0 \text{ s}} = 31.75 \text{ mm/s} \]

The required rotational speed would be

\[ \omega = \frac{31.75 \text{ mm/s}}{6.35 \text{ mm}} \cdot \frac{1 \text{ rev}}{60 \text{ s/min}} = 31.41 \text{ rad/s} \]

Then the power required to drive each screw would be

\[ P = T \omega = (204.4 \text{ N\cdotm}) (31.41 \text{ rad/s}) = 6420.5 \text{ W} \]

Alternative Thread Forms for Power Screws

While the standard Acme thread is probably the most widely used, others are available. The stub Acme thread has a similar form with a 29° angle between the sides, the depth of the thread is shorter, providing a stronger, more rigid thread. Metric power screws are typically made according to the ISO trapezoidal form that has a 30° included angle.

The relatively low efficiency of standard single-thread Acme screws (approximately 30% or less) can be a strong disadvantage. Higher efficiencies can be achieved using high lead, multiple thread designs. The higher lead angle produces efficiencies in the 30% to 70% range. It should be understood that some mechanical advantage is lost so that higher torques are required to move a particular load as compared with single thread screws. See Internet site 10.

17–3 BALL SCREWS

The basic action of using screws to produce linear motion from rotation was described in Section 17–2 on power screws. A special adaptation of this action that minimizes the friction between the screw threads and the mating nut is the ball screw.

Figure 17–3 shows a cutaway view of a commercially available ball screw. It replaces the sliding friction of the conventional power screw with the rolling friction of bearing balls. The bearing balls circulate in hardened steel races formed by concave helical grooves in the screw and the nut. All reactive loads between the screw and the nut are carried by the bearing balls which provide the only physical contact between these members. As the screw and the nut rotate relative to each other, the bearing balls are diverted from one end and are carried by the ball-guide return tubes to the opposite end of the ball nut. This recirculation permits unrestricted travel of the nut in relation to the screw. (See Internet site 3.)

Applications of ball screws occur in automotive steering systems, machine tool tables, linear actuators, jacking and positioning mechanisms, aircraft controls such as flap actuating devices, packaging equipment, and instruments. Figure 17–7 shows a machine with a ball screw installed to move a component along the ways of the bed.
The application parameters to be considered in selecting a ball screw include the following:

- The axial load to be exerted by the screw during rotation
- The rotational speed of the screw
- The maximum static load on the screw
- The direction of the load
- The manner of supporting the ends of the screw
- The length of the screw
- The expected life
- The environmental conditions

**Load-Life Relationship**

When transmitting a load, a ball screw experiences stresses similar to those on a ball bearing, as discussed in Chapter 14. The load is transferred from the screw to the balls, from the balls to the nut, and from the nut to the driven device. The contact stress between the balls and the races in which they roll eventually causes fatigue failure, indicated by pitting of the balls or the races.

Thus, the rating of ball screws gives the load capacity of the screw for a given life which 90% of the screws of a given design will survive. This is similar to the $L_{10}$ life of ball
bearings. Because ball screws are typically used as linear actuators, the most pertinent life parameter is the distance traveled by the nut relative to the screw.

Manufacturers usually report the rated load that a given screw can exert for 1 million inches (25.4 km) of cumulative travel. The relationship between load, \( P \), and life, \( L \), is also similar to that for a ball bearing:

\[
\frac{L_2}{L_1} = \left( \frac{P_1}{P_2} \right)^3
\]  

(17–12)

Thus, if the load on a ball screw is doubled, the life is reduced to one-eighth of the original life. If the load is cut in half, the life is increased by eight times. Figure 17–8 shows the nominal performance of ball screws of a small variety of sizes. Many more sizes and styles are available.

**Torque and Efficiency**

The efficiency of a ball bearing screw is typically taken to be 90\%. This far exceeds the efficiency for power screws without rolling contact that are typically in the range of 20\% to 30\%. Thus, far less torque is required to exert a given load with a given size of screw. Power is correspondingly reduced. The computation of the torque to turn is adapted from Equation (17–7), relating efficiency to torque:

\[
e = \frac{FL}{2\pi T_u}
\]

(17–7)

Then, using \( e = 0.90 \),

\[
T_u = \frac{FL}{2\pi e} = 0.177FL
\]

(17–13)

**FIGURE 17–8** Ball screw performance
Because of the low friction, ball screws are virtually never self-locking. In fact, designers also use this property to advantage by purposely using the applied load on the nut to rotate the screw. This is called backdriving; backdriving torque can be computed from

\[
T_b = \frac{FLe}{2\pi} = 0.143FL
\]  
\hspace{1cm} (17-14)

**Example Problem 17-3**

Select suitable ball screws for the application described in Example Problem 17-1 and illustrated in Figure 17-4. The hatch must be lifted 381 mm to open it eight times per day, and then it must be closed. The design life is 10 years. The lifting or lowering is to be completed in no more than 12.0 s.

For the screw selected, compute the torque to turn the screw, the power required, and the actual expected life.

**Solution**

The data required to select a screw from Figure 17-8 are the load and the travel of the nut on the screw over the desired life. The load is 55.6 kN on each screw:

\[
\text{Travel} = \frac{381 \text{ mm}}{\text{stroke}} \times \frac{2 \text{ strokes}}{\text{cycle}} \times \frac{8 \text{ cycles}}{\text{day}} \times \frac{365 \text{ days}}{\text{year}} \times \frac{10 \text{ years}}{} = 222.5 \times 10^5 \text{mm}
\]

From Figure 17-8, the 50.8 mm screw with two threads per inch and a lead of 12.7 mm is satisfactory.

The torque required to turn the screw is

\[
T_u = 0.177FL = 0.177(55.6 \text{ kN})(12.7 \text{ mm}) = 125 \text{ N.m}
\]

The rotational speed required is

\[
\omega = \frac{1 \text{ rev}}{12.7 \text{ mm}} \times \frac{381 \text{ mm}}{12.0 \text{ s}} \times \frac{60 \text{ s}}{\text{min}} = 15.7 \text{ rad/s}
\]

The power required for each screw is

\[
P = T_u = (125 \text{ N.m})(15.7 \text{ rad/s}) = 1.961 \text{ kW}
\]

Compare this with the 6.42 kW required for the Acme screw in Example Problem 17-1.

The actual travel life expected for this screw at a 55.6 kN load would be approximately \(81.28 \times 10^6\) mm, using Figure 17-8. This is 3.65 times longer than required.

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**17-4 APPLICATION CONSIDERATIONS FOR POWER SCREWS AND BALL SCREWS**

This section discusses additional application considerations that apply to both power screws and ball screws. Details may change according to the specific geometry and manufacturing processes. Supplier data should be consulted.

**Critical Speed**

The proper application of ball screws must take into account their vibration tendencies, particularly when operating at relatively high speeds. Long slender screws may exhibit the
phenomenon of critical speed at which the screw would tend to vibrate or whirl about its axis, possibly reaching dangerous amplitudes. Therefore, it is recommended that the operating speed of the screw be below 0.80 times the critical speed. An estimate for the critical speed, offered by Roton Products, Inc. (Internet site 10), is:

$$n_c = \frac{4.76 \times 10^6 d K_s}{(SF)L^2} \quad \text{(U.S.)}$$

$$= 187.6 \times 10^3 \frac{d k s}{(SF)L^2} \quad \text{(S.I.)}$$

where,

- $d$ = Minor diameter of the screw (mm)
- $K_s$ = End fixity factor
- $L$ = Length between supports (mm)
- $SF$ = Safety factor

The end fixity factor, $K_s$, depends on the manner of supporting the ends of the screw with the possibilities being:

1. Simply supported at each end by one bearing: $K_s = 1.00$
2. Fixed at each end by two bearings that prevent rotation at the support: $K_s = 2.24$
3. Fixed at one end and simply supported at the other: $K_s = 1.55$
4. Fixed at one end and free at the other: $K_s = 0.32$

The value of the safety factor is a design decision, often taken to be in the range from 1.25 to 3.0. Note that the screw length is squared in the denominator, indicating that a relatively long screw would have a low critical speed. The best designs would employ a short length, rigid fixed supports, and large diameter.

**Column Buckling**

Ball screws that carry axial compressive loads must be checked for column buckling. The parameters, similar to those discussed in Chapter 6, are the material from which the screw is made, the end fixity, the diameter, and the length. Long screws should be analyzed using the Euler formula, Equation (6–5) or (6–6), while the J. B. Johnson formula, Equation (6–7), is used for shorter screws. End fixity depends on the rigidity of supports similar to that described above for critical speed. However, the factors are different for column loading.

1. Simply supported at each end by one bearing: $K_s = 1.00$
2. Fixed at each end by two bearings that prevent rotation at the support: $K_s = 4.00$
3. Fixed at one end and simply supported at the other: $K_s = 2.00$
4. Fixed at one end and free at the other: $K_s = 0.25$

Suppliers of commercially available ball screws include data for allowable compressive load in their catalogs. See Internet sites 3, 5, and 10.

**Material for Screws**

Ball screws are typically made from carbon or alloy steels using thread-rolling technology. After the threads are formed, induction heating improves the hardness and strength of the surfaces on which the circulating balls roll for wear resistance and long life. Ball screw nuts are made from alloy steel that is case hardened by carburizing.
Power screws are typically produced from carbon or alloy steels such as AISI 1018, 1045, 1060, 4130, 4140, 4340, 4620, 6150, 8620, and others. For corrosive environments or where high temperatures are experienced, stainless steels are used, such as AISI 304, 305, 316, 384, 430, 431, or 440. Some are made from aluminum alloys 1100, 2014, or 3003.

Power screw nuts are made from steels for moderate loads and when operating at relatively low speeds. Grease lubrication is recommended. Higher speeds and loads call for lubricated bronze nuts that have superior wear performance. Applications requiring lighter loads can use plastic nuts that have inherently good lubricity without external lubrication. Examples of such applications are food processing equipment, medical devices, and clean manufacturing operations.

REFERENCES


INTERNET SITES FOR LINEAR MOTION ELEMENTS

2. Ball-screws.net. www.ball-screws.net Site lists numerous manufacturers of ball screws, some with online catalogs.
3. Thomson Industries, Inc. www.thomsonindustries.com Manufacturer of ball screws, ball bushings, linear motion guides, and a variety of other linear motion elements. Site includes catalog information and design data for load, life, and travel rate. Thomson is a part of the Linear Motion Systems Division of Danaher Motion Group.
4. Danaher Motion Group. www.danahermcg.com
Part of Danaher Corporation. Manufactures and markets motion control components under the Thomson, BS&A (Ball Screws & Actuators), Deltran, Warner linear, and other brands.
5. THK Linear Motion Systems. www.thk.com
Manufacturer of ball screws, ball splines, linear motion guides, linear bushings, linear actuators, and other motion control products.
6. Joyce/Dayton Company. www.joycejacks.com Manufacturer of a wide variety of jacks for commercial and industrial applications. Included are power screw and ball screw types with integral gear drives and complete motorized actuator systems.
7. SKF Linear Motion. www.linearmotion.skf.com Manufacturer of high efficiency screws, linear guiding systems, and actuators.

PROBLEMS

1. Name three types of threads used for power screws.
2. Make a scale drawing of an Acme thread having a major diameter of 38.1 mm and four threads per 25.4 mm. Draw a section 50.8 mm long.
3. Repeat Problem 2 for a buttress thread.
4. Repeat Problem 2 for a square thread.
5. If an Acme-thread power screw is loaded in tension with a force of 133.44 kN, what size screw from Table 17-1 should be used to maintain a tensile stress below 68.95 MPa?