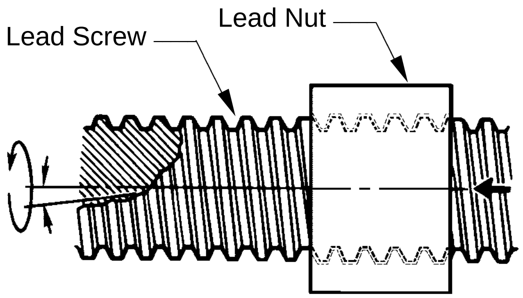
4) Drive Mechanisms

This section will introduce most of the more common types of drive mechanisms found in linear motion machinery. Ideally, a drive system should not support any loads, with all the loads being handled by a bearing system. Topics discussed will include, but not be limited to, the mechanism of actuation, efficiency, accuracy, load transfer, speed, pitch, life cycle, application and maintenance. Each type of drive system will be accompanied by a diagram and useful equations when applicable. Some of the terms used with screws, the most common drive component, are as follows:

lead — advance of the nut along the length of the screw per revolution
pitch — distance between corresponding points on adjacent thread forms (pitch = lead / # of starts)
# of threads — number of teeth found along a unit length of the screw (1 / pitch)
# of starts — number of helical grooves cut into the length of the shaft
outer diameter — largest diameter over the threaded section (at top of threads)
root diameter — smallest diameter over the threaded section (at base of threads)
stub — specific type of ACME thread where the root diameter is larger to provide for a more heavy duty screw (the threads look “stubby”)
critical shaft speed — operating speed of spinning shaft that produces severe vibrations during operation. This is a function of length, diameter, and end supports.
maximum compressive load — maximum load that can be axially applied to the screw before buckling or permanent deformation is experienced. Also referred to as column strength.
end bearing supports — the screw must be supported at one or both ends with thrust type bearings. Depending upon the application, it may also be desirable to provide for a stiffer system by incorporating angular contact bearings (fixed support).

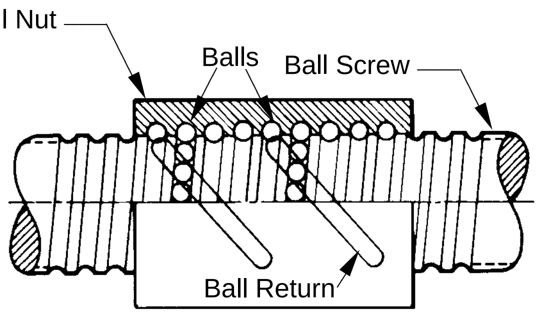
Although shafts, gear trains, belt and pulley, rack and pinion, and chain and sprocket drives are practical in other applications, they require special consideration when used in CNC machinery. This is because there is typically backlash associated with these types of drives, which increases the system error. Thorough technical descriptions of these types of drives can be found in the Stock Drive Products Components Library.



Lead Screw System

Lead screws are threaded rods that are fitted with a nut. There are many types of threads used, but the most prevalent in industry is the ACME lead screw. Because the ACME thread is an industry standardized thread style, it is easily interchanged with parts from various manufacturers. The basic function of a screw is to convert rotary input motion to linear output motion. The nut is constrained from rotating with the screw, so as the screw is rotated the nut travels back and forth along the length of the shaft. The friction on the nut is a function of environment, lubrication, load, and duty cycle; therefore, practical life cycle is difficult to quantify. Lead screw/nut drive systems are available in a variety of sizes and tolerances. Contact is primarily sliding, resulting in relatively low efficiency and a wear rate proportional to usage. **Advantages** include the self-locking capability in back drive mode which is good for vertical applications, low initial costs, near silent operation, manufacturing ease, and a wide choice of a materials. **Disadvantages** of ACME screws include lower efficiencies (typically 30-50%, depending on nut preload) which require larger motor drives, and unpredictable service life.

Ball Screws are very similar to lead screws with the exception of a ball bearing train riding between the screw and nut in a recirculating raceway. This raceway is generally lubricated, which allows for predictable service life. Due to the increased number of mating and moving parts, matching tolerances becomes more critical. The screw threads have rounded shapes to conform to the shape of the balls. The function, terminology, and formulas are the same as found with lead screws, however the performance of ball screws is far superior. The rolling action of the balls versus the sliding action of the ACME nut provides significant advantages. **Advantages** of ball screw drives are increased efficiency (typically up to 90 – 95%) which allows required motor torque to be lower, predictable service life, low wear rate and maintenance costs. **Disadvantages** include limited material choice, higher initial cost, and an auxiliary brake is required to prevent back driving with vertical applications.



Ball Screw System

Helpful Formulas: When determining the amount of input torque required to produce an amount of output linear force, there are many factors to consider. The following equations provide a practical approach in making force and torque calculations.

Force Calculations:

$$\boldsymbol{F_T} = \boldsymbol{F_A} + \boldsymbol{F_E} + \boldsymbol{F_F} \qquad (1)$$

where: F_T = Total Force
F_A = Acceleration Force
F_E = External Force
F_F = Friction Force

$$F_A = \frac{W}{g} \times \frac{a}{12} \text{ lb} \qquad (2)$$

where: **W** = total weight to accelerate (lb)
a = linear acceleration (in/sec^2)
g = acceleration from gravity (ft/sec^2)

External Force (F_E) may be due to gravity in vertical applications, or may be from external work requirements (feeding material, stretching material, etc.)

Friction Force (F_F) required to overcome all of the friction in the load bearing system (with a low friction bearing system, this can be negligible)

NOTE: *The Total force must be below the compressive (thrust) rating of the screw chosen. A modest factor of safety should be added to the total force so that unexpected dynamic loads are safely handled by the screw system.*

Torque Calculations:

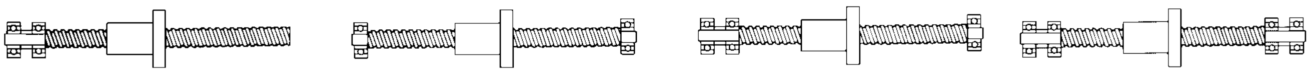
For most typical applications, rotary inertia, motor rotor inertia, and screw inertia are negligible, therefore left out of the torque calculations.

$$T = F_T \times \frac{1}{2\pi pe} \qquad (3)$$

where: F_T = Total Force
p = screw pitch (revolutions per inch)
e = efficiency (no units)

NOTE: *The Torque required should be well below the torque rating of the motor chosen. A modest factor of safety should be added to the torque required so that unexpected dynamic loads are safely handled by the driving system.*

Selecting and Sizing Screw Drive Systems: When choosing a particular screw for a given application, there are several factors to be considered. Required rpm, critical speed and maximum compressive strength are the most important design features that determine screw design parameters, and can be calculated according to the following equations. Since thread style design is irrelevant in these calculations, the same equations and charts can be used for both lead screws and ball screws. Bearing configuration must be considered when using these equations. The following diagrams represent the typical bearing end support arrangements.



A. Fixed-Free   **B.** Simple-Simple   **C.** Fixed-Simple   **D.** Fixed-Fixed

$$\text{rpm} = \frac{\text{linear velocity (in/min)}}{\text{lead (in/rev)}} \tag{4}$$

Maximum Speed:

$$C_s = F(4.76 \times 10^6)\frac{d}{L^2} \tag{5}$$

where:

- C_s = critical speed (rpm)
- d = root diameter of screw (inches)
- L = length between supports (inches)
- F = end support factor (see diagram)
  - case **A.**: 0.36
  - case **B.**: 1.00
  - case **C.**: 1.47
  - case **D.**: 2.23

Maximum Load

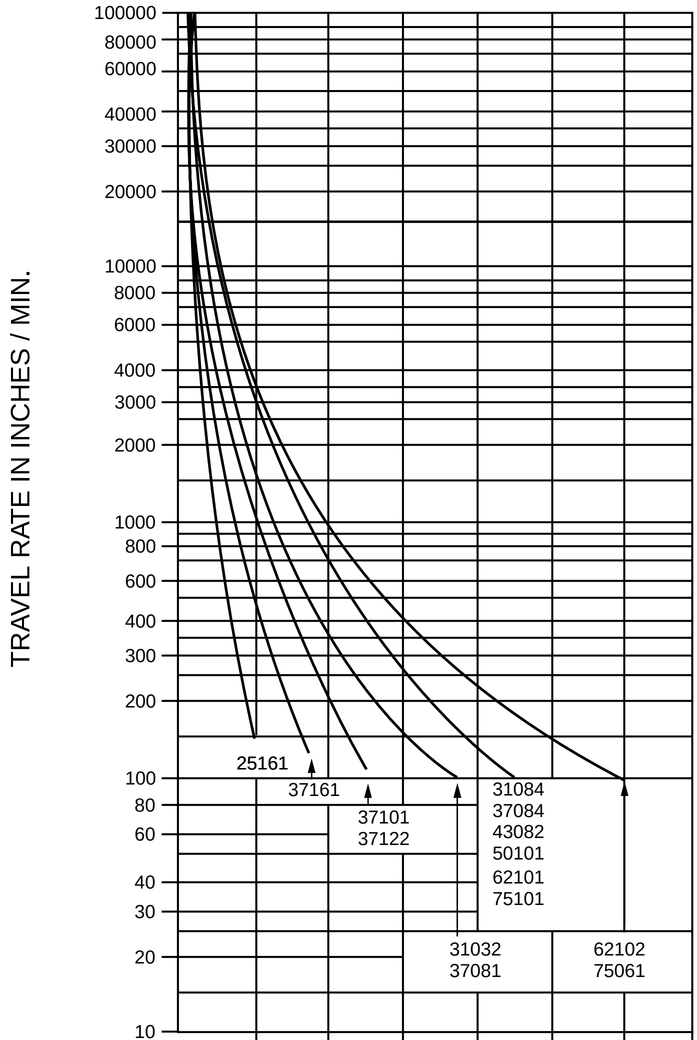
$$P = F(14.03 \times 10^6)\frac{d^4}{L^2} \tag{6}$$

where:

- P = maximum load (lbs) (critical load)
- d = root diameter of screw (inches)
- L = maximum distance between nut and load carrying bearing
- F = end support factor (see diagram)
  - case **A.**: 0.25
  - case **B.**: 1.00
  - case **C.**: 2.00
  - case **D.**: 4.00

The formulas above can be represented graphically by the charts on following pages. These charts have been compiled for screws made of stainless steel. Speeds, loads, diameters, bearing arrangements and products are referenced. It must be realized that a screw may be able to rotate at very high rpm's, but the nut may have more strict limitations. For this reason, we have truncated the ball screw rpm diagrams to a top end of 4000 rpm, and provided each type screw with their own charts. Please note that the ball screw charts are only represented for screws of 16 mm and 25 mm diameter.

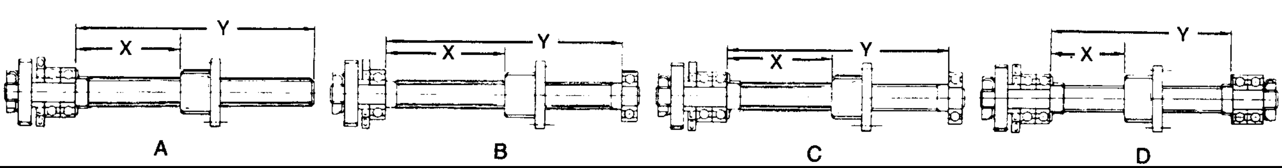
Travel Rate vs. Length
For Standard ACME Screws



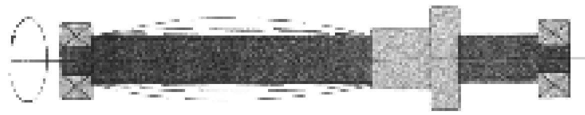
	REF								
ONE END FIXED OTHER END FREE	A	6	12	18	24	30	36	42	INCHES
BOTH ENDS SUPPORTED	B	10	20	30	40	50	60	70	INCHES
ONE END FIXED OTHER END SUPPORTED	C	12	24	36	48	61	73	85	INCHES
BOTH ENDS FIXED	D	15	30	45	60	75	90	105	INCHES

LENGTH

MAXIMUM LENGTH (IN.) ADJUSTED FOR BEARING SUPPORT
"Y" DIMENSION



CRITICAL SPEED

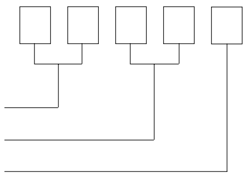


PURPOSE

This graph was designed to simplify the selection of the proper lead screw so as to avoid lengths and speeds which will result in vibration of the assembly (critical speed). The factors which can be controlled after a particular maximum length is determined are: method of bearing support and choice of lead screw diameter.

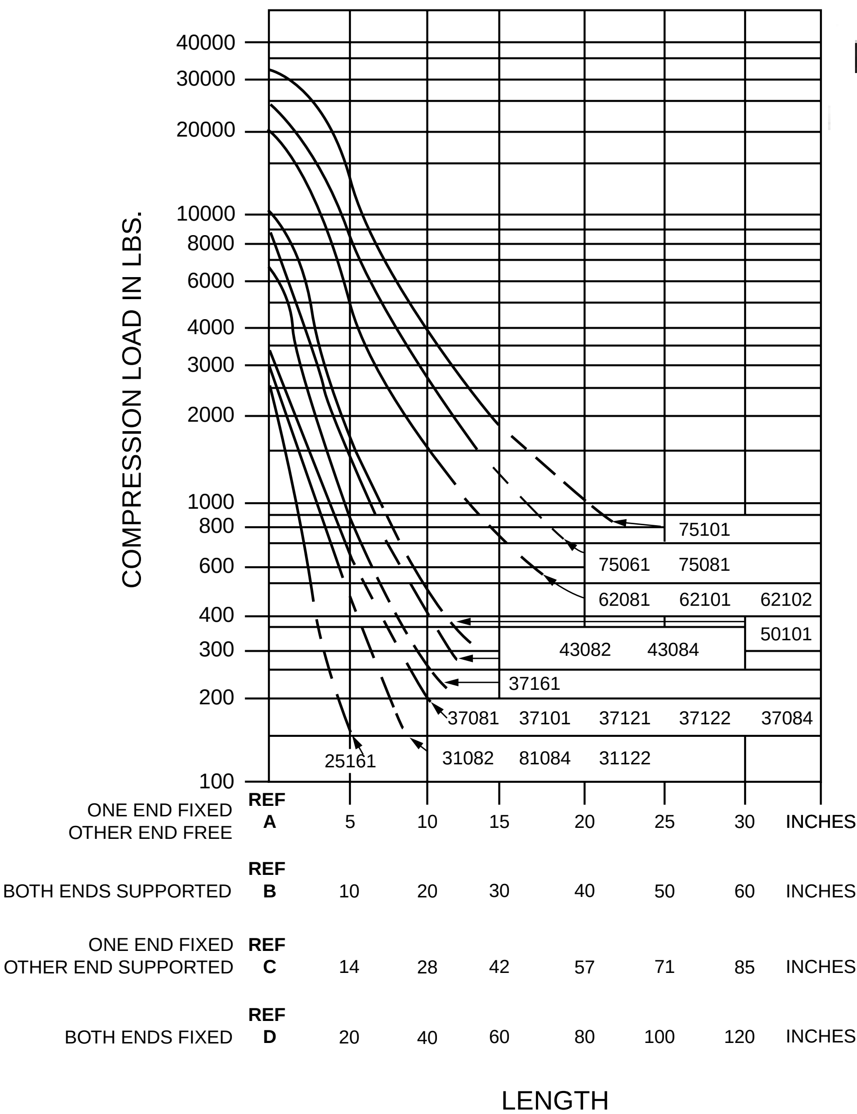
USE OF THE GRAPH

1. Choose preferred bearing support means, based on design considerations.
2. On the proper bearing support horizontal line (A, B, C or D) choose length of lead screw.
3. Draw vertical line at the lead screw length, determined at (2.), and draw a horizontal line at the travel rate.
4. All screw diameters to the right and above the intersection point in (3.) are suitable for this application.
5. Screw sizes are coded as follows:

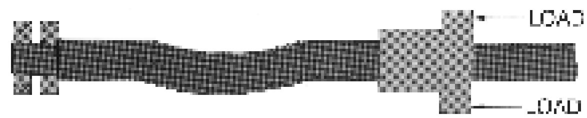


Diameter (in)
Threads / in
Starts

Compression Load vs. Length
For Standard Ball Screws and ACME Screws



COLUMN LOADS

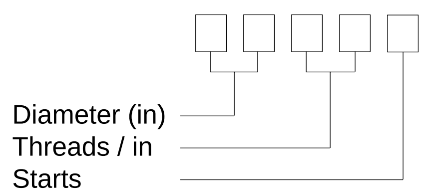


PURPOSE

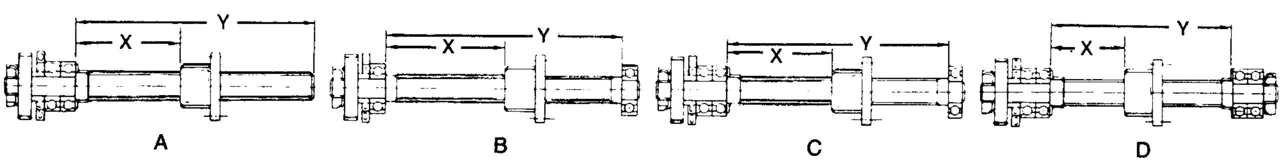
This graph was designed to simplify the selection of the proper lead screw so as to avoid buckling when subjected to the axial loading by means of the nut. The factors which can be controlled after a particular maximum length is determined are: method of bearing support and choice of lead screw diameter.

USE OF THE GRAPH

1. Choose preferred bearing support means, based on design considerations.
2. On the proper bearing support horizontal line (A, B, C or D) choose length of lead screw.
3. Draw vertical line at the lead screw length, determined at (2.), and draw a horizontal line at the compression load the unit is exerting on the screw.
4. All screw diameters to the right and above the intersection point in (3.) are suitable for this application.
5. Screw sizes are coded as follows:



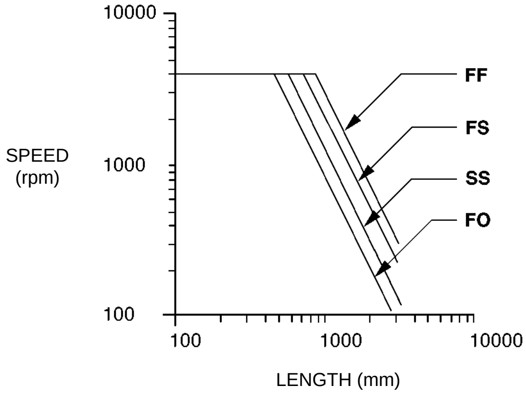
MAXIMUM LENGTH (IN.) ADJUSTED FOR BEARING SUPPORT
"X" DIMENSION



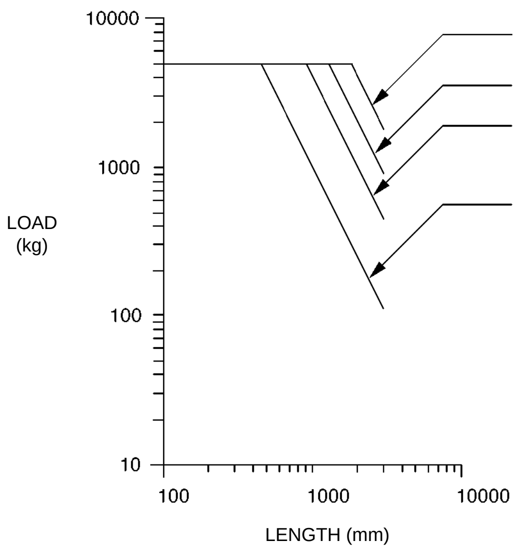
CRITICAL SPEED & LOAD

LOAD AND SPEED LIMITS ON 16 mm BALL SCREWS

CRITICAL SPEED

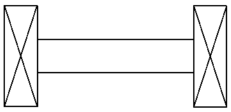


CRITICAL LOAD

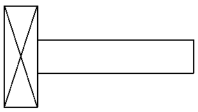


BEARING SUPPORT TYPES

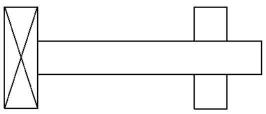
FF – Fixed, Fixed



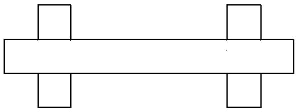
FO – Fixed, Open



FS – Fixed, Simple



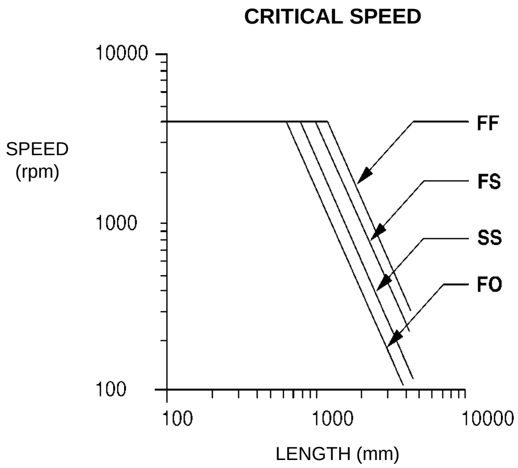
SS – Simple, Simple

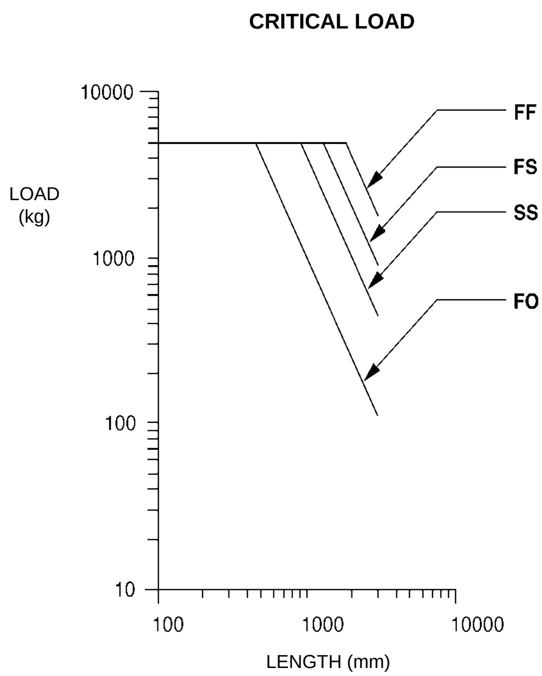


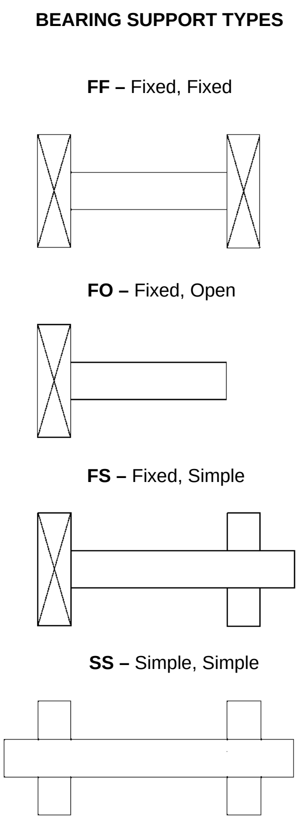
TECHNICAL INFORMATION

CRITICAL SPEED & LOAD

LOAD AND SPEED LIMITS ON 25 mm BALL SCREWS

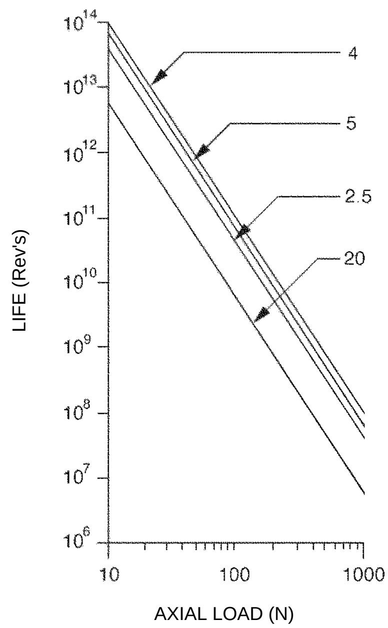




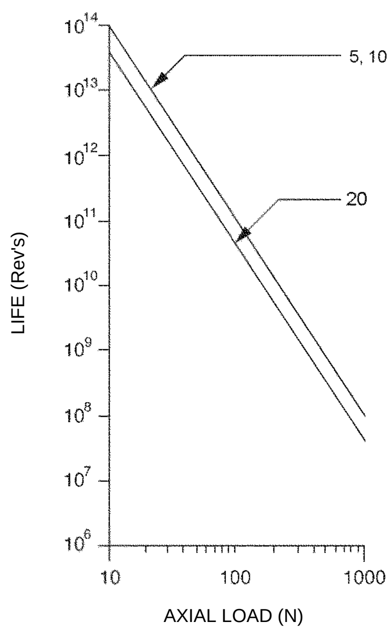


BALL & ACME SCREW ASSEMBLY LIFE EXPECTANCY

16 mm LIFE EXPECTANCY



25 mm LIFE EXPECTANCY



SPECIFICATIONS			
Pitch	Screw Dia.	Axial Load (N)	
		Dynamic (C_a)	Static
2.5	16	3500	5500
4	16	2600	4200
5	16	4600	7200
5	25	5100	12600
10	16	4200	6500
10	25	5100	12600
20	16	1900	2500
20	25	3570	8800

$$L = \left[\frac{C_a}{F_m}\right]^3 \times 10^6$$

L = life expectancy expressed in number of revolutions

C_a = dynamic load rating (N) [for acme nuts, see design load column on catalog pages].

F_m = average axial load (N).

Example: For 10 mm pitch screw,16 mm dia., C_a = 4200 N carrying an average axial load, F_m = 200 N (45 lbs.) the expected life is:

$$L = \left[\frac{4200}{200}\right]^3 \times 10^6 = 9.261 \times 10^9 \text{ revolutions.}$$

At an average of 1000 rpm this will result in:

$$\frac{9.261 \times 10^9 \text{ revolutions}}{1000 \text{ rpm}} \times \frac{1 \text{ hour}}{60 \text{ minutes}} = 154{,}000 \text{ hours}$$

of expected operational life. Note that the nature of the motion (jerky, smooth, etc.) will affect the life expectancy.

TECHNICAL INFORMATION