

# LINEAR MOTION

## Leadscrew Assemblies

### TFE Coating

The general purpose in TFE coating is to supply a more even distribution of lubricant than is normally found when using standard self-lubricating plastics on steel. The wear life, coefficient of friction and resulting torque to drive a lead screw assembly are highly governed by the ability of the engineered plastic to supply sufficient lubrication to the nut/screw interface. The inability of the internal lubricating agents in some plastics to consistently migrate to the surface may result in erratic drag torques and unpredictable wear. TFE coating the entire screw surface thus results in an extremely even lubrication distribution. Test results indicate system torque requirements to be consistent with little or no change in required frictional driving torque, even with changes in motor R.P.M.

### Handling

Lubrication to the interface occurs by the nut picking up the TFE particles from the coating as well as from the migration of the internal lubricant within the plastic nut.

Although care should be taken to ensure that chips and voids do not occur in the coating, small voids have been shown to have little effect on system performance. The lubricant, although solid, has some "spreading" ability as in fluid lubricants. When machining for bearing ends, soft fixturing is recommended.

The manufacture has developed a custom composition TFE specifically for our lead screws and nut materials. It is applied at high temperatures in our facilities and provides extended nut life and smooth operation at very low additional cost.

TFE coated screws provide the maximum level of self-lubrication and should not be additionally lubricated or used in environments where oils or other lubricant contamination is possible. Screws up to 8ft (2.4m) can be supplied coated.

### Design and Engineering Data

#### Screw Accuracy

A unique precision rolling process for screw manufacturing is used.

Positional repeatability is within 0.0013mm.

Lead accuracy for standard screws is .0006 mm/mm. Lead accuracies are available up to .0001 mm/mm.

We can provide a laser-interferometer generated mapping of high lead screw accuracy screws. Assemblies have an extremely high repeatability of .0013mm.

#### Lead

Advancement per revolution

Lead = Pitch x Number of Starts

#### End Machining

We can custom machine screws to your requirements (quote to your drawings) or cut to length screws for your own machining.

Lengths can be specified up to 4m, or up to 2.4m with TFE coating.

### Pitch

Crest to crest distance or one divided by threads per inch. (on a multiple start thread, the pitch equals the lead divided by the number of starts.)

### Critical Speed

This is the rotational speed at which a screw will experience vibration or other dynamic problems. See Critical Speed Chart to determine if application parameters result in speed approaching critical.

To minimize critical speed problems: use a longer lead, choose a larger diameter or increase bearing mount support.

Prices for specials to drawings available.

Please fax for a quote.

### Traverse Speed

The polyacetal nut material used provides long wear life over a wide variety of conditions. However, very high loads and/or speeds will accelerate nut wear. We recommend the following linear traversing speeds for optimum life:

### Maximum Load

Although these Anti-Backlash assemblies are capable of withstanding relatively high loads without catastrophic failure, they have been designed to operate under the loading shown in the size charts on the product pages (Design load kg).

Lead mm	Traverse Speed mm/sec
2.5 - 12	100
12 - 25	250
25 - 60	760

### Efficiency

Efficiency of a lead screw varies highly with lead angle (see size listings). For example the efficiency is low when the lead screw is either very small or very large. For example high helix screws approach maximum efficiency because of their optimum lead angle.

### Torque

The required motor torque to drive a lead screw assembly is the sum of three components: The inertia torque, static frictional torque, and torque to move load. It must be noted that this is the torque necessary to drive the lead screw assembly alone. Additional torque associated with driving frictional bearings and motor shafts, moving components, and drag due to general assembly misalignment must also be considered.

### Inertia Torque

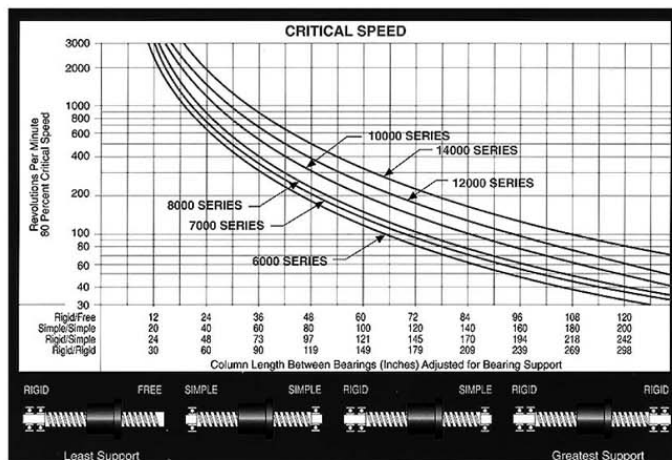
$T = I \alpha$  Where  $I$  = screw inertia

$\alpha$  = angular acceleration

# LINEAR MOTION

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TFE Coating



### Static Frictional Torque

Our Anti-Backlash Assemblies are typically supplied with static frictional torque of 0.007Nm to 0.05Nm. The magnitude of the frictional torque is dependant on the standard factory settings or settings specified by the customer. Generally the higher the preset force, the better the Anti-Backlash characteristics.

### Torque To Move Load

The torque to move a certain load is a function of the lead and the efficiency of the lead screw assembly.

$$\text{TORQUE} = \frac{\text{Load} \times \text{Lead}}{2\pi \times \text{Efficiency}}$$

### Back Driving

Sometimes referred to as reversibility, this is the ability of a screw to be turned by a thrust load applied to the nut. Generally, when the screw lead is less than  $\frac{1}{8}$  the diameter, back driving will not occur. For higher leads, where back driving is likely, the torque required for holding a load is:

$$\text{Tb} = \frac{\text{Load} \times \text{Lead} \times \text{Efficiency}}{2\pi}$$

### PV Calculation (Material Limits)

O.D. (Inches)  
Root Dia. R.D. (Inches)  
Lead (Inches)  
Load - Axial (Lbs.)  
Linear Speed (Inches/sec.)

Use the Critical Speed chart for limits of Rpm and then the PV Calculation for Material Limits. The lowest value must not be exceeded when converted into Rpm or linear speed.

### Helix Travel

H.T. = (Surface Travel per Rev.)

$$= \frac{\% (\text{Lead})^2 + (P_o)^2}{(\text{Thread Length})}$$

$$\text{Where } P_o = \frac{O.D. + R.D.}{2}$$

$$\begin{aligned} \text{AT} &= \text{Area per Thread} \\ &= \text{H.T.} \times \text{Thread Thickness} \\ &= \frac{\text{H.T.} \times O.D. - R.D.}{2} \end{aligned}$$

$$\begin{aligned} \text{ATA} &= \text{Total Thread Area} \\ &= \text{AT} \times 3 \quad (\text{Assuming 3 Thread Contact}) \end{aligned}$$

$$\text{RS} = \frac{\text{Linear Speed}}{\text{Lead}} = \text{Rev./sec.}$$

$$\text{RPM} = \text{RS} \times 60 = \text{Rev./min.}$$

$$\text{Travel Speed} = \text{Ft/min} = \text{Rpm} \times \frac{\text{H.T.}}{12} = \text{Ts}$$

$$\text{PV} = \text{TS} \times \frac{\text{Load}}{\text{ATA}} \quad \text{lb./in}^2 \text{ ft/min}$$

# LINEAR MOTION

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### Screw Straightness

Screw straightness is indicated by Total Indicator Runout. (TIR).  
The standard straightness for lead screws is 0.075mm/300mm.  
We can provide tighter specifications on customer request.

SCREW / NUT SERIES	STATIC FRICTIONAL DRAG TORQUE (Nm)	SCREW INERTIA $\text{kgm}^2/\text{inch}$	ANTI-BACKLASH LIFE*	ANTI BACKLASH LIFE W/TFE COATING
B4000	FREE WHEELING	$2.11 \times 10^{-8}$	N/A	N/A
B6000		$1.06 \times 10^{-9}$	Typical Backlash	Typical Backlash
B7000		$2.47 \times 10^{-9}$		
B8000		$3.67 \times 10^{-9}$		
B10000		$1.00 \times 10^{-10}$		
B12000		$2.15 \times 10^{-10}$		
B14000		$4.09 \times 10^{-10}$	0.076 - 0.25mm	0.076 - 0.25mm
ZBX4000	0.007 - 0.03	$2.11 \times 10^{-8}$	100-150 million cm	380 - 500 million cm
ZBX6000	0.010 - 0.03	$1.06 \times 10^{-9}$		
ZBX7000	0.020 - 0.04	$2.47 \times 10^{-9}$		
ZBX8000	0.020 - 0.04	$3.67 \times 10^{-9}$		
ZBX10000	0.030 - 0.05	$1.00 \times 10^{-10}$	200-250 million cm	450 - 580 million cm
KHD 600	0.007 - 0.02	$1.06 \times 10^{-9}$		
VHD 8000	0.010 - 0.04	$3.67 \times 10^{-9}$	500-570 million cm	760 - 880 million cm
VHD 10000	0.010 - 0.04	$1.00 \times 10^{-9}$		
VHD 12000	0.020 - 0.05	$2.15 \times 10^{-10}$		

### PHYSICAL PROPERTIES

LEAD SCREWS		NUTS			ASSEMBLY	
MATERIAL	SURFACE FINISH	MATERIAL	TENSILE STRENGTH	COEFFICIENT OF EXPANSION	STANDARD OPERATING TEMPERATURE RANGE	COEFFICIENT OF FRICTION NUT TO SCREW
303 Stainless Steel	Better than 400 microns (0.4 $\mu\text{m}$ Ra)	Polyacetal with lubricating additive	1.4N/m <sup>2</sup> (9,700 psi)	$6.0 \times 10^{-5}$ in./in./°F (3.3 $\times 10^{-5}$ mm/mm/°C)	32 - 200 °F (0 - 93°C)	Static = 0.08 Dynamic = 0.09 (un-coated Static 0.08 Dynamic 0.15)

\*Life will vary with loading, operating environment and duty cycle. The longer screw leads generally provide longer life.

Degassing in a Vacuum at  $10^{-4}$  TORR.

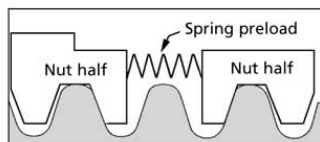
Time	Weight loss %
24 hrs	0.20%
48 hrs	0.26%
96 hrs	0.35%
330 hrs	0.37%

# LINEAR MOTION

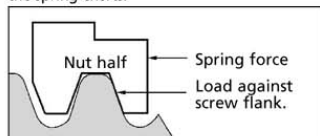
## Leadscrew Assemblies

### Axial Take-Up Mechanism

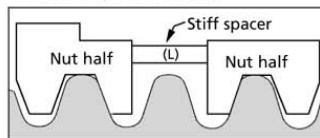
**A patented - backlash compensation without high drag torque.**



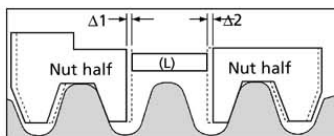
The standard method for taking up backlash is to bias two nut halves axially using some type of compliant spring (wavy washer, compression spring, rubber washer, etc.) The unit is very stiff in the direction in which the nut half is loaded against the flank of the screw thread. However, in the direction away from the screw thread, the nut is only as axially stiff as the amount of preload which the spring exerts.



For example, if the maximum axial load which the system is subjected to is 50lbs, the amount of spring preload must be equal to or greater than 50lbs. In order to maintain intimate screw/nut contact. The problems arising from preloading in this manner are increased torque and nut wear.

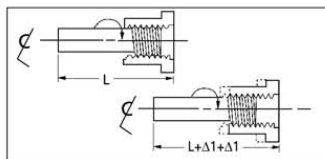


Obviously, the higher the load at the screw/nut interface, the higher the required torque to drive the nut on the screw and the more susceptible the unit is to nut wear. An alternate method would thus be to replace the spring with a stiff spacer sized to fit exactly between the two nut halves. There would then be no excessive preload force at the interface and the unit would be capable of carrying high axial loads in either direction with no backlash. This is fine initially, however, as use time increases, wear begins on the nut threads causing a gap to develop between the spacer (L) and the nut halves.



This gap ( $\Delta 1 + \Delta 2$ ) is now the amount of backlash which has developed in the unit. This backlash can be removed by replacing the stiff spacer with a new spacer equal to  $(L + \Delta 1 + \Delta 2)$ . This process, although effective, would be extremely costly and difficult to implement on a continuous basis.

### THE MANUFACTURER'S SOLUTION



What is needed, then, is a stiff spacer which will continually expand to accommodate the wear which occurs during use. This is done by creating a spacer threaded at one end with a complimentary nut torsionally biased to advance when a gap develops. The thread at the end of the spacer is a fine helix such that an axial load will not backdrive the nut once space growth has occurred. The amount of preload on the unit is only that necessary to turn the spacer nut on the spacer rod and is independent of the external system loadings.

**THEREFORE, WE HAVE A SELF-WEAR COMPENSATING UNIT WHICH HAS EXTREMELY LOW FRICTIONAL DRAG TORQUE YET HIGH AXIAL STIFFNESS.**