LINEAR MOTION DEVICES

Trends in the last few decades have led designers to rethink concepts of industrial motion. Companion gains in control theory and hardware allow schemes for complete motion systems not previously possible. And motion technologies from fields as diverse as national defense and medical diagnostics are available for exploitation in other fields. The primary effect is that today's designers must consider not just industrial "power transmission" along shafts and through reductions, but also the broader concept of *industrial motion and control*. And linear motion is an essential part of it.

Many modern processes call for unattended operation, exceptional precision, high throughput, flexibility for short runs, and total manufacturing integration. Often in such cases, humans can't perform well enough. Modern sensors and controls, coupled with diverse and precise linear and rotary motion devices, combine to fill the needs. Thus, designers must consider linear motion as an integral part of industrial motion and control.

The major componentry of linear motion systems can be categorized as:

- Actuators.
- Support systems (bearings).
- Control systems and components.

Many equipment manufacturers supply complete systems that include all major types of components.

ACTUATORS

Common linear actuation devices for single-axis motion include, but are not limited to:

- Various complex linkages such as a walking beam or slider-crank mechanism.
- Gear rack and pinion set.
- Plate or disc-cam drive with fixed-axis follower.
- Cylindrical-cam drive with fixed-axis follower.
- Chain, belt, or cable drive in the linear part of its path, with or without special attachments.
- Plastic drive tape.
- Sliding-action leadscrew (usually acme screw), with nut.
- Ball-bearing leadscrew (ballscrew) with nut. The balls recirculate into and out of the load zone.
- Planetary roller screw, in which a nut engaging several planetary threaded rollers mounts on a threaded shaft. The threaded rollers also engage the shaft. Axes of shaft, nut, and rollers are parallel. Threads on the nut and planetary rollers are of the same helix angle.
- Recirculating roller screw, in which a nut engaging several grooved rollers mounts on a threaded shaft. The rollers also engage the shaft. Differs from the planetary roller screw in that rollers recirculate into and out of the load zone.
- Skewed rollers on a rotating cylindrical rod (traction motion on a threadless rod).
- Fluid-power cylinder with direct-driving rod, rodless cylinder, or cable cylinder.
- Electric solenoid.
- Electric linear motor, such as linear induction motor or linear step motor.
- Adapted electric rotary motor. A common type has no shaft. The rotor also serves as the mating nut on a leadscrew with axis coincident with rotor axis.

Preceding listing order does not imply relative importance.

Many linear actuation devices come as complete packages. For example, you can get a motorized ball screw powered through spur or worm gearing, and fully self-contained so that you need only mount and wire the assembly to your machine. Similarly, you can get an actuator that is synchronous-belt-driven for the linear actuation, motorized, and with integral geared speed reducer. You can also get electrohydraulically actuated devices that use the precision of digital control with the high force of a hydraulic cylinder. You can get any such packages with integral control hardware and software.

Screw jacks

One of the older conventional actuation packages, the screw jack, comes in a single housing containing the input rotating shaft, the output linear motion shaft (screw), all bearings, and the lubricant. The user connects a motor and the load attachment. There are three major types.

Machine-screw jack. In the common machine-screw jack, Figure 1, rotation of the input (worm) shaft turns the worm gear and drive nut, which connects rigidly to the worm gear. The leadscrew (also called the lifting screw or stem) is of acme or modified square-thread form. It is threaded through the drive nut and converts rotary motion of the nut to linear motion, if the screw is kept from turning with the drive nut.

Rolling-element bearings support the input shaft and worm gear to minimize frictional loss. Thrust bearings support the load. The stem cover stores lubricant and helps protect the stem from damage and contamination.

Machine-screw jacks come in many stock sizes with load ratings from less than 1 to more than 250 tons. A jack can mount stationary so the stem reciprocates, or an external nut can be used so the stem rotates and the nut (attached to the load) reciprocates. Most machine-screw jacks are self-locking. Thus, the load remains stationary in the event of a power failure.

The major limitation of machine-screw jacks is low efficiency, typically

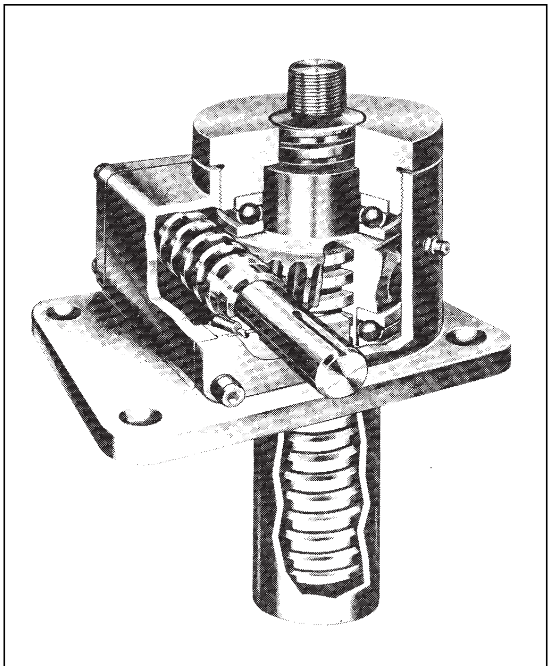


Figure 1—Typical machine screw jack.

25% or less. Sliding between drive nut and screw generates heat. This heating restricts duty cycles of machine screws to 5 to 7 ½ min/hr at full load.

Ball screw jack. If an application calls for the advantages of a machine-screw jack, but needs a longer duty cycle or higher efficiency, a better choice may be a ball screw jack, Figure 2, or a roller-screw jack. See also discussions "Ball screws" and "Roller screws" which follow. The heart of a ball screw jack is an assembly composed of a screw and nut, separated by a recirculating series of balls. The ball screw jack works like a ball bearing and has similar life predictability.

Rolling friction of the ball screw, compared with sliding friction of the machine screw, generates little heat. This allows higher lifting speeds in the ball screw jack and much better efficiency, typically 92 to 95%. The higher efficiency reduces input power requirement to about two thirds that of a machine-screw jack for a given load. Besides higher efficiency than machine-screw jacks, ball screw jacks have a lower ratio of starting to running torque. Because of low rolling friction, many ball screw jacks can be back-driven. See the discussion on back driving under "Ball screws."

Roller-screw jack. You can also improve on a machine-screw jack's advantages by using a roller-screw jack. See the discussion under "Roller screws" which follows.

Screw-jack application. Figure 3 depicts a typical lifting arrangement, sometimes called a T-system. It shows how one motor can power several synchronized lifting points by using couplings and right-angle gearboxes. Selection of the best arrangement for an application is usually based on space availability and motor accessibility.

Here are some guidelines:

- Keep screw load direction parallel to the screw axis as much as possible.
- Keep the span between drive components as short as practical. That keeps interconnecting shafting short and limits the chance of a critical-speed problem.
- When necessary, use pillow-block supports, dynamically balanced shafting, or both, to help avoid critical-speed vibrations.
- Select shaft couplings with high strength-to-bore ratios (such as gear couplings) to minimize system inertia.
- Use three and four-way miter boxes whenever possible, to shorten interconnecting shafts.
- Use limit switches to restrict extremes of stem travel, or connect rotary switches directly to one unused side of the double-extending input shaft of a jack.
- Use a slip coupling between motor and jack input shaft to provide overload protection.

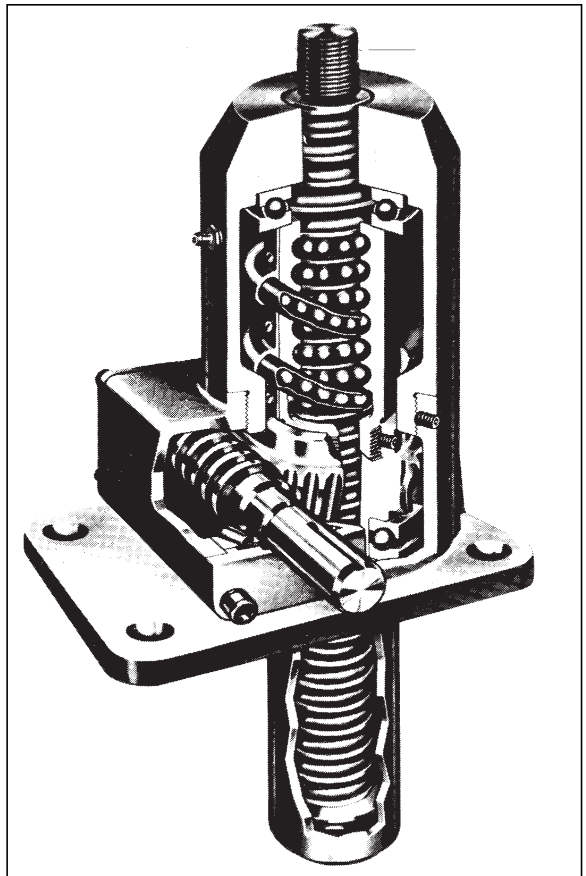


Figure 2 — Typical ball screw jack.

- You can include other torque-sensing devices into a jack or a motor to protect the whole system from overload.

To select a screw jack, use a system design manual having illustrations, column strength charts, power and torque data, and sample calculations.

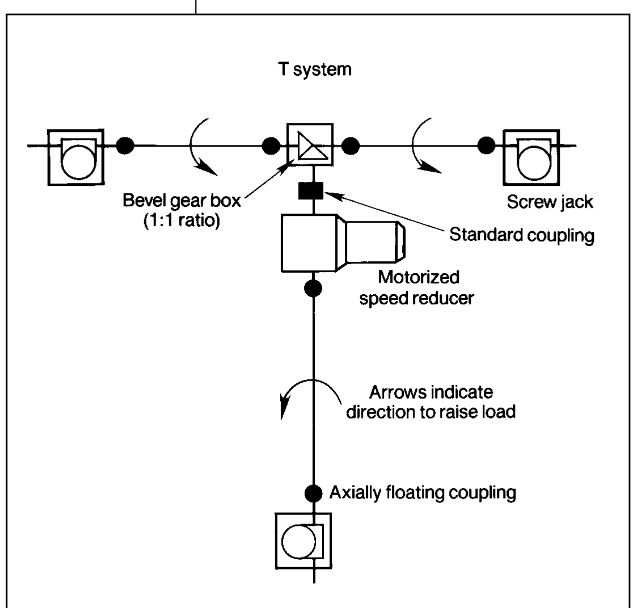


Figure 3 — Typical screw jack system arrangement.

Before selecting any jack equipment, determine:

- Number of lifting points.
- Total load per jack.
- Load direction (tension or compression).
- Speed at which load must move.
- Total distance load must move.

Also, consider duty cycle and environmental conditions such as temperature and contaminants. Next, using a system design manual, make calculations to determine input torque, speed, and power requirements. Then check column strength and select the jack.

Ball screws

The ball screw jack is a specific use of a more general linear actuation device called a ball screw or ball-bearing screw. The first ball screws appeared a half-century ago, but their popularity surged in the last few decades as needs for efficient mechanical linear actuation grew rapidly. Early models were seen as low-friction actuators in automotive steering gear and similar uses. Sliding friction of threads on the nut and screw is replaced by rolling friction of balls in approximately circular-form threads on the nut and screw, instead of Acme or modified square threads of the machine-screw actuator. A deflector forces the balls out of the screw's threads and into the ball guide on the nut. The guide directs them diagonally back to the opposite end of the nut and rechannels them to the screw's thread, so the balls circulate continuously. This self-containment allows use of fewer balls. The ball guide can be an external tube that mounts on the nut or, for more compactness, an integral interior part of the nut.

Formed vs. machined threads. Most ball screws are of steel or stainless steel. Threads are machined and ground for high precision, or roll-formed for lesser precision and lower cost. In recent years, thread-forming techniques have improved to where better grade rolled-thread ball screws approach the precision of many machined ball screws of several years ago.

Backlash. The precision of a ball screw assembly depends strongly on the amount of backlash in it. Backlash is the measured play between nut and screw. Ball size and ball-and-ball-groove conformity dictate the amount of backlash. For ordinary non-preloaded ball screws, a common backlash range is 0.002 to 0.013 in., depending on screw size and ball diameter. To avoid backlash, one nut can be preloaded against another on the screw so there is no play. Similarly, one ball circuit can be preloaded against another within the one nut. In some assemblies, slots cut in the nut make it act as a preloaded spring when mounted, or a clamping device can draw it up against the screw thread.

Mountings. Critical speed of any long shaft, including a ball screw, is a speed at which it vibrates violently in a transverse direction. It results from rotating-system unbalance. The member can have several critical speeds, all multiples or submultiples of one predominating "first-order" speed. A machine cannot operate long at critical speed. Bidirectional thrust bearings at each end of a ball screw provide greatest support for critical-speed and stiffness considerations; bidirectional bearings at one end with the other end totally free, the least support.

Stiffness. A ball screw's diameter and the number and size of its load-carrying balls determine assembly stiffness. Preload and end configuration can also affect stiffness.

When a screw mounts with single-direction thrust bearings at each end or thrust bearings at only one end and freely at the other, then its spring rate varies directly with its cross-sectional area at the root diameter; directly with operating load; and inversely with length. The spring rate of the ball screw system varies inversely with the sum of the reciprocals of the spring rates of the screw, the nut, and the bearings. Total system spring rate is always less than that of the most compliant member.

Drive torque. For rotary-to-linear motion, ball screw assembly drive torque is:

$$T = \frac{PL}{2\pi E}$$

where:
T = Torque input, lb-in.
P = Operating load, lb
L = Lead, in./rev
E = Efficiency (about 90%)

Torque values from the equation do not account for drag or inefficiencies due to mounting or to drive components, to wiper drag, or to torque due to preload.

Preload torque. Preloading a ball screw does add resisting torque. In some applications such as numerically controlled machines, in order to specify the screw's motor drive completely, you must know preload torque in addition to load torque. Consult the ball screw manufacturer.

Life expectancy. A ball screw is like a ball bearing in many ways, and life expectancy is based on similar principles. For a ball screw, life expectancy depends on:

- Applied force, including acceleration and friction loads.
- Number and length of load strokes.

For best life, the load should be applied along the same axis as that of the ball screw. Side loads and loads that tend to overturn the nut cause uneven load distribution on the balls and reduce life expectancy.

Life expectancy is based on a standard rating of 1 million in. of travel. Most ball screw manufacturers offer charts to determine life in inches of travel for various loads. For example, if the load is halved from rated load, expected life increases by about nine times.

Other factors affecting life rating are screw and nut materials and hardnesses. If hardness of a stainless steel ball screw is reduced from, say, 56 to 45 Rockwell C, then load capacity is reduced to about 30% of what it had been.

Column load. Excessive compression load on a ball screw may cause it to buckle. Column-load capacity of a ball screw varies with length, type of end mounting (fixity), and screw root diameter. In general, these parameter changes increase the tendency for the screw to buckle as a column:

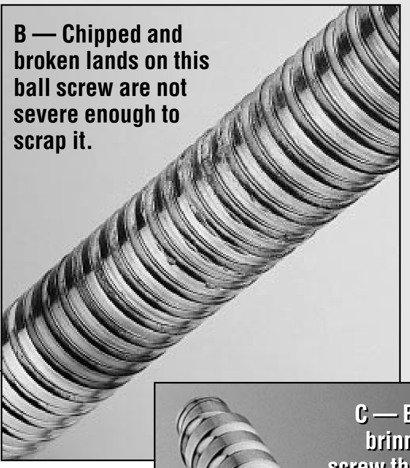
- Increasing axial load.
- Increasing length-to-diameter (slenderness) ratio.
- Reducing end-support fixity.

Another problem that axial load can cause, unrelated to column buckling, is that load makes the ball screw lead change slightly. In most cases it is insignificant. However, for heavy load in tension or compression in a high-precision application, you may need to account for lead variation.

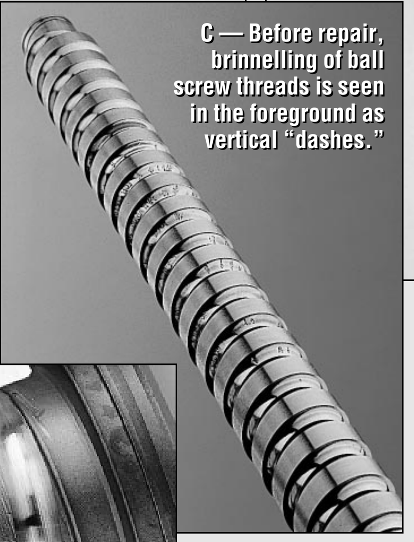
Installation. Misalignment can shorten life of a ball screw assembly to a small fraction of the expected life. Avoid misalignment by making mounting surfaces of the nut parallel or perpendicular to the ball screw centerline, or provide a gimbal housing if appropriate.

Get more life out of ball screws

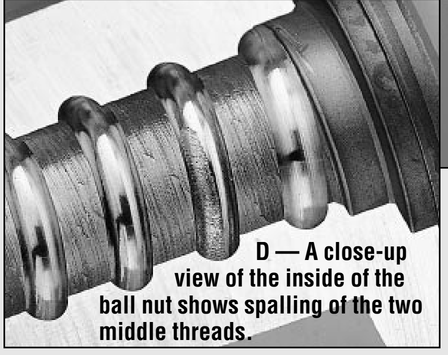
The ball screw on the right has seen many hours of service, but it is more economical to repair it than to discard it. After repair, it will look like the assembly at left.



B — Chipped and broken lands on this ball screw are not severe enough to scrap it.



C — Before repair, brinnelling of ball screw threads is seen in the foreground as vertical "dashes."



D — A close-up view of the inside of the ball nut shows spalling of the two middle threads.

Figure A — Common forms of wear or damage on the threads of ball screws.

Many ball screws are repairable. Common problems, such as loss of repeatability due to wear can be fixed by regrinding the ball thread grooves and then using larger balls in the ball-screw assembly. If a minor crash bends the screw a bit, it can often be straightened to its original accuracy and returned to service. Surface problems such as spalling, brinelling, chipping, or feathering, Figure A, can be reground and replated. All of these repairs may prove more economical than replacement with a whole new ball-screw assembly.

To determine when a ball screw needs repair, and approximate how much repair, measure its diametral backlash, or lash. Diametral lash (not axial backlash) is a measurement that can be taken in the plant. The ball-screw assembly is placed in V blocks. An engineer lifts the ball nut vertically and with a gage, measures the play between the ball nut and screw. For a preloaded assembly, a diametral lash of 0.0005 in. indicates a wear factor of 50%. Minimal, or Level 1, repair is needed to bring the screw back into use. A lash of 0.0035 in. represents 80% wear and indicates either a Level IV repair or a dead ball screw. Similarly for nonpreloaded assemblies, a diametral lash of 0.009 in. is 50% wear and needs Level 1 and 0.015 in. is 80% wear and needs Level IV or replacement.

For ball-circle diameters ranging from 0.03934 to 0.04875 in., the acceptable lash is 0.002 to 0.004 in. For diameters ranging from 0.5625 to 0.6250 in., the acceptable lash is 0.004 to 0.008 in.

As noted, a ball screw with 80% or more wear is likely irreparable. Four repairs is about the maximum for any ball screw. After that, bury the ball screw with honors.

Levels of repair. When a ball screw arrives at a repair facility, it is inspected and evaluated for the type of needed repair. This process can take up to three or four days.

In general, there are four levels of cost-effective repair. While each successive level adds cost, this cost is still less than a new ball-screw assembly.

All repair levels involve the same four basic steps: inspect, clean, reball, and straighten.

- Level 1 (three days) repairs loss of repeatability due to wear. Balls are the first to wear. Thus, new, larger balls are used at every level to restore preload and repeatability. The secret to proper ball replacement: for every 0.003 in. of wear, use a 0.001 in. larger ball. The screw is also straightened because a bow as little as a thousandth of an inch can put excess moment on the ball nut, which can later result in failure.
- Level 2 (seven days) adds regrinding of the ball nut to the steps taken in Level 1. Ball-nut thread grooves wear faster than the screw threads because they are subject to more ball travel.
- Level 3 (seven days) adds regrinding of the ball screw threads, and as required, rebuilding of the journal diameters with eutectic spraying and grinding them back to size.
- Level 4 (14 days) adds regrinding of the ball nut and ball screw. This level may cost 55% of a new assembly, but the repaired ball screw will have a normal new-screw life. When the repair cost goes over 65%, buy a new assembly.

These four levels of repair are classified by the most common repairs and do not cover all contingencies.

Excerpted from an article by Thomson Saginaw Ball Screw Co., in the June 1996 issue of PTD.

A common installation procedure: Mount the ball screw in its bearing at either end; loosely connect the nut to its mounting; traverse the nut from one end to the other so it can seek its own center; then tighten the nut mounting. This tends to alleviate induced side loads on the ball screw assembly. If torque from one end to the other is constant, there is no significant binding in the mounting system.

Several manufacturers now supply complete ball screw assemblies, including nut and screw properly preloaded, bearings, slides or rails, and frame. The user aligns and bolts down the entire frame. You need not align or preload any component. You can get motorized versions, too.

Back driving. The ball screw assembly has the capability of either the

Selecting lubricants for ball screws

Lubricants maintain the low-friction advantage of ball-screw assemblies by minimizing rolling resistance between balls and tracks and sliding friction between adjacent balls. Proper lubrication helps keep most contaminants out, which reduces the damage foreign matter can cause.

Oil and grease provide corrosion protection, but lubricant choice depends on the advantages and disadvantages of each. Oil can be applied at a controlled flow rate directly to the point of need. It will clean out moisture and other contaminants as it runs through the ball nut and provides cooling. Oil disadvantages include:

- Possibility of excess oil contaminating the process.
- Cost of a pump and metering system to apply oil properly.

Grease is less expensive than oil to apply and requires less frequent application, and it does not contaminate process fluids. Grease disadvantages:

- It is hard to keep inside the ball nut and has a tendency to build up at the ends of ball nut travel, where it accumulates chips and abrasive particles.
- Incompatibility of old grease with relubrication grease can create a problem.

Oil lubrication. Operating temperature, load, and speed determine the oil viscosity and application rate for an installation. If the oil is too viscous or if you use too much, heat may be generated. If the oil is too thin or you use too little, parts may not be coated adequately; friction and wear may result.

The following guidelines are appropriate for most applications, but if extremes of temperature, load, or speed are involved, consult a lubrication specialist.

The recommended nominal viscosity of the oil at 40 C is based on the mean speed of the ball screw, its diameter, and the temperature at which the ball nut is likely to stabilize. Viscosity is expressed in centistrokes (1 cSt = 1 mm^2/sec.) Various grades have been selected for standardization (DIN 51512).

To determine the nominal viscosity of the oil for an application, establish the mean speed of rotation of the ball screw and, from it, the limiting speed, dn_m, factor. You also need the temperature at which the ball nut is likely to stabilize.

Mean speed of rotation accounts for the ball screw's duty cycle:

$$n_m = n_1(q_1 \div 100) + n_2(q_2 \div 100) + n_3(q_3 \div 100) + \ldots$$

where:

n_m = mean speed, rpm

$n_{1,2,3\ldots}$ = speed for time $q_{1,2,3\ldots}$, rpm

$q_{1,2,3}$ = time at speed $n_{1,2,3\ldots}$, % of total

For typical applications, n_m ranges from 200 to 500 rpm.

The dn_m factor is given by:

$$dn_m = (d)(n_m)$$

where:

d = ball screw nominal diameter, mm

Typical values of dn_m range from 15,000 to 25,000 mm/min. Values of dn to 100,000, where n is the maximum speed of rotation, are becoming more common, and in such cases the lower viscosity should be used if the oil selection guide indicates a grade midway between two adjacent viscosity curves.

Ball nut operating temperature should be about 20 C, however, it usually stabilizes a few degrees above screw-shaft operating temperature. If you can't measure nut temperature, assume it to be 30 C for your initial selection of oil viscosity.

Required oil flow rate is a function of:

- Number of ball circuits.
- Ball-screw orientation.
- Operating environment.
- Load.
- Speed.
- Judgments based on knowledge of the application.

In addition:

- If the ball screw is horizontal, add nothing to the flow rate, Q_{min}, to account for orientation; if vertical, add 25%.
- If the application is clean and dry, add nothing to Q_{min} to account for environment; if not, add 25%.
- If the screw is not subject to high loads or speeds, add nothing to Q_{min} to account for severe running conditions; if it is, add 50%.

Grease lubrication. Grease is not so widely used as oil for ball-nut lubrication, though it lubricates acceptably. Speeds that are high for ball screws are no problem for grease, so speed is no criterion for selection.

One problem with grease: It tends to be fed out of the nut and onto the ball screws, accumulating at the extremities of travel where it collects contaminants. It must be replenished regularly.

Grease is a complex subject. Greases consist of a mineral or synthetic oil, additives, and a thickening agent such as lithium, bentonite, aluminum, and barium complexes. For most applications, use a grease with a drop point above 220 C, a service temperature range of -30 to 130 C, and a limiting speed factor (dn) above 1,000,000. Such a grease is classified as HL91 Grade 2 (DIN 51818) and is based on Mil-9-7711A.

As a rule of thumb, replenish grease at least every 800 hr. However, because conditions vary so widely, you should confirm this interval by inspection, and readjust if needed. For extreme conditions, such as dn values above 50,000, consult a lubrication expert.

Excerpted from an article by Thomson Industries Inc., in the February 1995 issue of PTD.

nut or screw rotating when a thrust load is applied to the other member of the assembly. However, not all ball screws can back drive. The thread's helix angle determines if the assembly can back drive. Generally, a ball screw with a helix angle of 6 deg or more will back drive; those of 4 to 6 deg are marginal; those under 4 deg probably will not do so. Be aware that, in some situations in which you would not expect back driving, continuous machine vibration with the ball screw unpowered and unrestrained might cause slow back driving.

In many situations, you would not want the ball screw to back drive. For example, should power fail on a lift, it could be disastrous for the load to run back. You must assure that either the ball screw cannot back drive or, if it can, that you provide a holding means such as a spring-applied, electrically released brake to prevent screw rotation on power loss.

Variations. Many variations in ball screws and optional equipment let you adapt them to special requirements. For example, hollow screws are available for situations where low system weight is important, such as in actuators on aircraft.

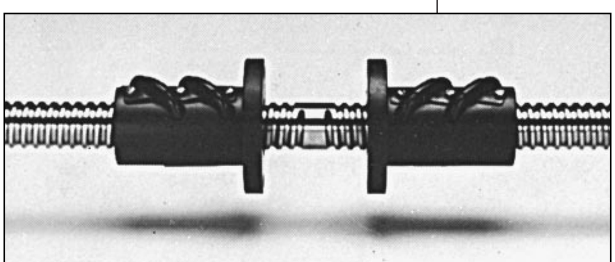


Figure 4—Bidirectional 1-piece ball screw needs no joint to connect and synchronize left and right-hand screws.

Options in seals, wipers, mounting arrangements, housings, preload devices, and similar characteristics help also. And you can get specialty systems such as the bidirectional 1-piece ball screw of Figure 4. It lets two nuts move in opposite directions simultaneously, an advantage in applications such as clamping devices and robotics. Also, a trunnion-mounted nut can be helpful in some applications; a telescoping screw in others.

Roller screws

The first roller screws appeared in the early 1950s. However, only in the last decade or so has their popularity increased significantly. Roller screws are more costly to produce than ball screws and they are applied mostly where application requirements of load-carrying capacity, axial stiffness, linear speed, or acceleration and deceleration rates are especially stringent.

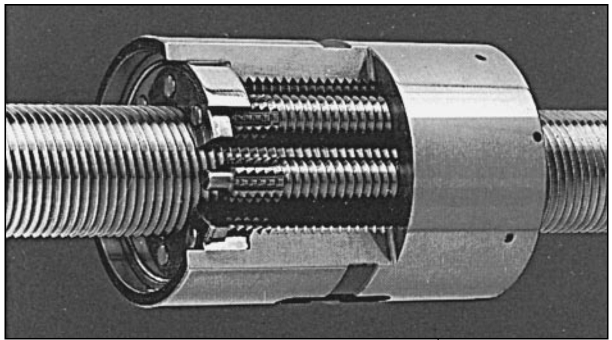


Figure 5—Typical planetary roller screw. In this style, roller screws remain in constant contact with the threaded haft. In another style, roller screw recirculate into and out of the load zone.

Overall, roller screws are similar to ball screws in preload configuration, backlash, lost motion, left-hand and right-hand thread, back driving, efficiency, torque, and power requirements.

Figure 5 shows a planetary roller screw.

Linear slides and races

Not all linear motion applications consist of straight lines. Some applications require an occasional curve or the circular motion of pure radial movement, such as that found in tool changing mechanisms, measurement of turbine blades, rotating manipulators, and movement of prisms in laser measurement machines.

These curved linear systems consist of slides or races, and rings (360 degrees of rotation) or segments of rings (90 or 180 degrees of rotation). One type of slide uses opposing female bearings with V-shaped outer rollers in a two-and-two arrangement. The bearings ride on a track with matching V-shaped rails. A carriage plate on top of the two-and-two bearings is the mounting platform, Figure 6. Thus, the carriage assembly effectively runs on eight line-contact points on a track with varying circumferential diameters.

For a fixed segment of a ring, fixed center carriage plates are the most popular. A bogie carriage, Figure 7, is used around S-bends, slideways with differing bend radii, and curves where looseness in the movement between straight and curved sections is not desirable. The bogie carriage runs on swivel bearings, which operate on a principal similar to that used in train and tram bogies to negotiate bends in the track.

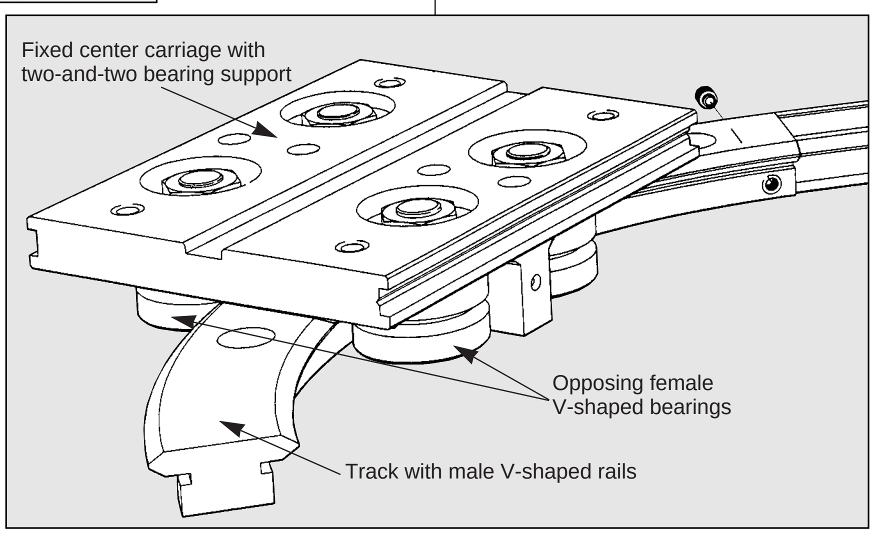


Figure 6 — V-ball bearing systems use opposing female bearings with V shaped outer rollers in a two-and-two bearing support arrangement. A fixed center carriage uses the two-and-two arrangement to support the mounting plate. Two of the V bearings have eccentric studs to facilitate adjustment.

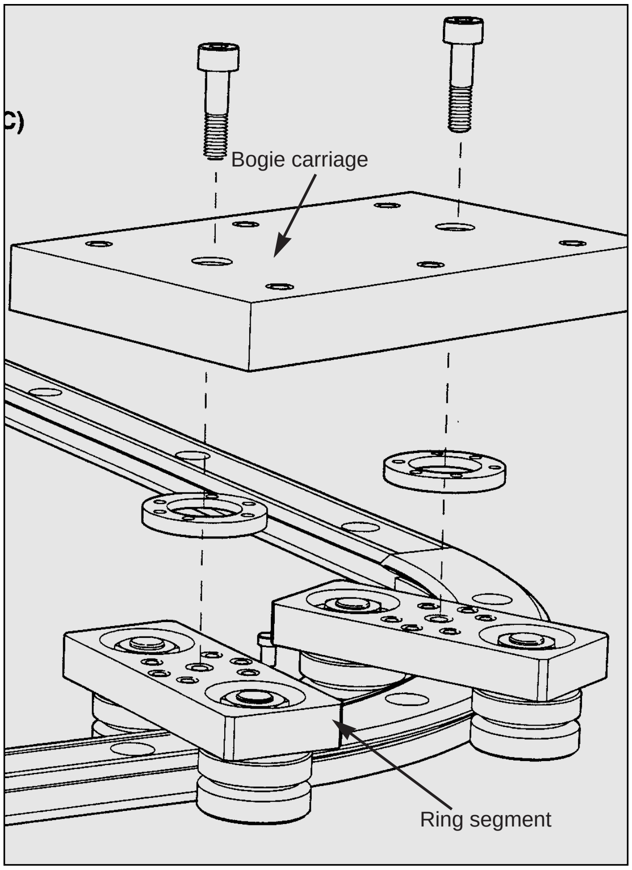


Figure 7 — A bogie carriage carries loads around S bends or slideways with differing bend radii. The bogie carriage runs on swivel bearings, which operate on a principal similar to that used in train and tram bogies to negotiate bends in the track.

V-ball bearing track systems are best suited to light loads — direct loads from 120 to 3,800 N (26.98 to 854.38 lb) and moment loads from 0.6 to 220 Nm (0.53 to 1,946.90 lb-in.) in a lubricated system. Refer to manufacturers' tables for precise load handling capabilities.

Rings offer stability with support as near to the load as possible. A gearcut rack on the outside or inside diameter of the register face of the ring serves as the drive mechanism.

Linear slideways are available in lengths to 4 m; for longer lengths, slide segments are matched and butted together.

V-ball bearing systems can run dry or lubricated. Lubrication, through lubricator blocks, can prolong system life by as much as 150%. Every time a ring slide rotates it is wiped with oil, which is also imparted to the female V of the bearing surfaces.

For rings or segments of rings, it is possible to achieve circular motion with radial run-out no greater than 0.05 mm per 360 deg (pro rata over angle of segment). Axial run-out is 5 microns.

For multiple carriage track systems, the greatest repeatability error is in the direction of travel and is dependent on the play in the drive mechanism. However, it is possible to achieve repeatability with 0.2 mm with a belt-driven system.

With all radial motion, engineers must consider centrifugal force. The force is proportional to the square of the tangential velocity. Doubling the carriage speed quadruples the force. It is also inversely proportional to the radius. Doubling the radius halves the force. Often, this force will also cause a moment load about the carriage plate.

Excerpted from an article by Bishop-Wisecarver Corp., in the August 1996 issue of PTD.

Other mechanical actuators

As listed earlier, there are many linear actuators besides screws. Among the oldest—still much used in specific machines—is the slider-crank mechanism and its many variations. Perhaps in its eldest form it converted rotary to linear motion to drive pump pistons. Later versions became popular converting linear motion to rotary motion on steam-engine-driven railroad locomotives and paddlewheel-propelled ships. The simplest—the classical Scotch yoke—converts rotary to linear motion in a characteristic linear-motion profile called simple harmonic motion. Imagine looking down at right angles to the crank's axis and following the motion of a point on its circumference. As the point travels across the axis its entire velocity vector is perpendicular to your line of sight; the point seems to be moving fastest. When the point nears an extreme of horizontal crank throw, its velocity vector is nearly parallel to your line of sight. The point seems nearly motionless, then motionless, then it reverses direction. If the crank turns at constant angular velocity, that projected action of the point is simple harmonic motion. Though simple harmonic motion is fairly easy to produce, it has high linear acceleration at the extremes of travel. High acceleration means high force—which generally means increased tendency for machine-part wear or breakage, and for workpiece or product damage. Cams, bar-type linkages, and similar devices can modify crank-generated motions into profiles of lesser peak acceleration. "Cycloidal motion" is such a profile.

Moreover, modern electric motors and their controls modify displacement, velocity, and acceleration profiles of mechanisms so that you can readily get the best profiles for a process without danger to components.

For example, a cam-and-screw mechanism, Figure 8, adapts a constant-lead cam and a ball spline to provide linear motion according to predetermined programs that the motor and its control execute. The cam mounts rigidly on two ball-spline bushings that can traverse the length of the spline shaft. The bushings are preloaded against each other to prevent backlash.

The spline bushing can move axially on ball tracks on the spline shaft, but it cannot rotate relative to the shaft. Thus if torque turns the spline shaft, the bushing turns with the shaft, causing the cam to rotate. A motor-and-reducer package turns the spline shaft, which therefore turns the bushing and cam. The cam then acts much like it would in a typical rotary index drive except that the traditional roles of cam and cam follower are reversed: An axially mobile cam engages a straight-line array of stationary cam followers. The cam rotates through the row of followers to achieve linear actuation. The cam seemingly pulls itself along the single-file row of standing followers, contacting at least two at any instant to prevent backlash.

A carriage housing mounts on and travels with the cam, but does not rotate. Tapered roller bearings let the cam turn inside the housing but support thrust load only; two linear bearing systems over the actuator's length carry

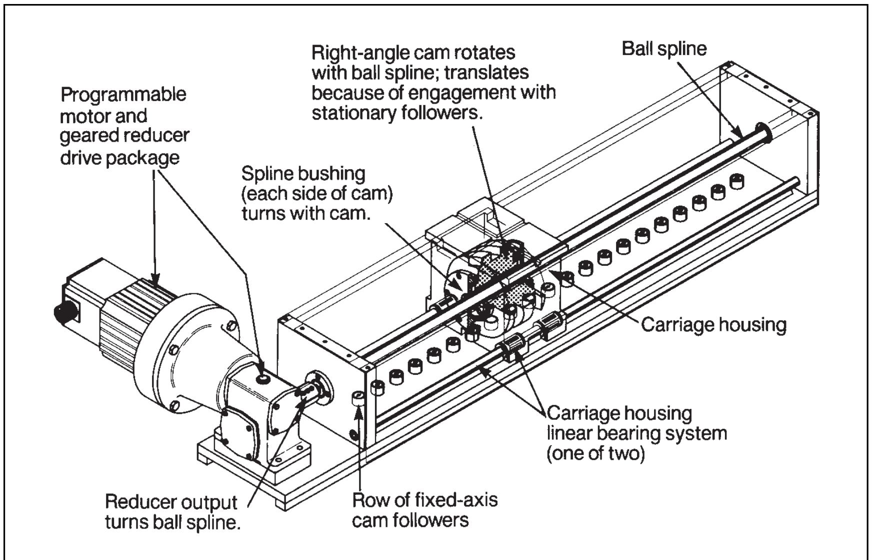


Figure 8—Cam-and-screw mechanism reverses usual roles of cam and followers. As reducer output turns ball spline, constant-lead rotary cam engages cam followers standing in a row. Positive cam pitch makes cam move axially, engaging succeeding followers and disengaging receding followers.

the weight of the cam, carriage, and payload, and resist tipping moments.

Standard leads for such systems can be up to five times more than those of ball screws.

Other types of mechanical linear actuation include belt, chain, and cable drives in their linear travel paths. For example, Figure 9 shows a linear actuator specifically for conveying (linearly pushing) products. The driving elements are wire-reinforced polyurethane synchronous belts with pusher-cleat attachments. Many cable drives and power-transmission roller chain drives work similarly with attachments. Many positioning tables, single stages, and self-contained linear actuation systems use the reinforced synchronous belt as the linear-motion element.

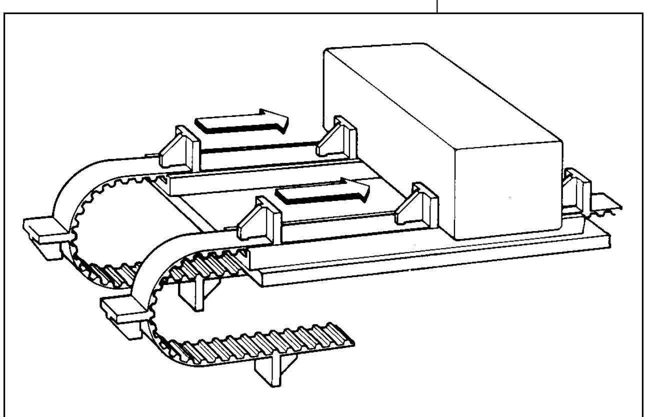


Figure 9—Wire-reinforced polyurethane synchronous belts with attachments provide linear motion in straight-line portion of belt travel. Various cleats, pushers, or lugs provide contact. Many positioning tables or stages use such a device for linear motion. With attachments, roller chain drives and cable drives can work similarly. (Note: Not shown is the belt-driving mechanism, which can be various devices, such as sprockets.)

The popularity of rodless cylinders, Figure 10, for linear motion applications is increasing, especially in applications with space restrictions. It can challenge mechanical and electrical actuators in many uses, and product variety has grown over the last few years.

The band cylinder is a popular version of the rodless cylinder. The cylinder wall has a slot running its full length. The slot lets the piston connect mechanically to an external carriage. A dynamic strip-type seal over the slot's outer surface and another over its inner surface seal-in cylinder pressure and seal-out contaminants. Carriage travel opens the seal section beneath the carriage, but maintains the seal. As the carriage passes, the seals reseat on the slot and continue to seal.

Magnetically coupled rodless cylinders make slots and dynamic seals unnecessary; the piston couples magnetically to the external carriage.

Recent versions of these rodless cylinders can now handle tipping or transverse loads. Most vendors offer fully pre-engineered, out-of-the-box, bolt-it-down, hook-it-up systems. Options include position sensors, position and velocity controls, end-of-stroke bumpers, shock absorbers, and other snubbing devices, brakes, external and guides.

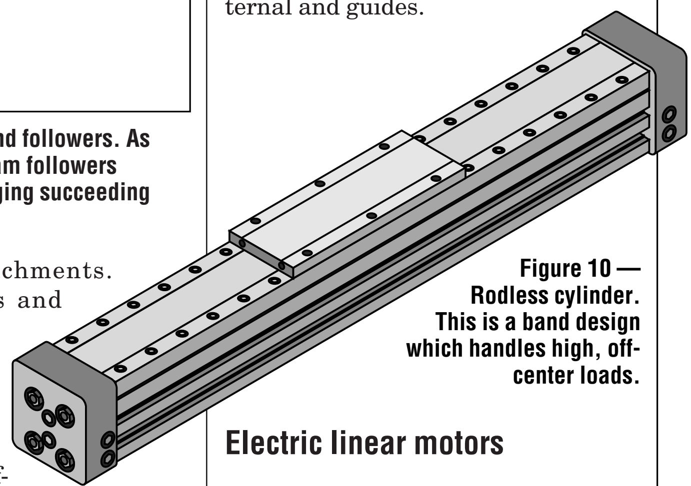


Figure 10 — Rodless cylinder. This is a band design which handles high, off-center loads.

Electric linear motors

An electric linear motor makes conversion of rotary to linear motion unnecessary. Thus, linear motors can eliminate shafts, belts, and gears to minimize space, energy, and costs. For information on rotary motors, see the Motors Products Department in this handbook.

Linear motors generate force rather than torque. Force to inertia ratio and stiffness — depending on the type of linear motor — can be as high as 30:1, and to 0.9 million N/mm or 5 million lb/in., respectively. Some versions offer smooth motion to within a fraction of a percent of their nominal velocity, which can range between 1 micron/sec to over 5 m/sec. Others can operate continuously, provide 5 g (or higher) acceleration, and offer a low settling time, in some cases, less than 50 msec.

Linear motors all have the same basic structure. Imagine a rotary ac or dc motor that has been sliced along its axis and opened up flat, resulting in two sections. One section, the primary, is a set of electrical coils embedded in a core (typically of steel, epoxy, or aluminum). The structure of the second section (the secondary) depends on the type of linear motor. A typical air gap of 0.024 in. (0.6 mm) separates the two sections for noncontact force transmission.

While there are several types of linear motors, many rely on the interaction of magnetic flux that produces forces on the moving and stationary members. For the voice-coil, dc force, and step-motor types, part of this flux comes from coil current. For the induction motor, ac excites a coil that produces flux. In turn, this flux interacts with flux produced by induction (like a transformer) and generates a force proportional to the relative strengths and distribution of the interacting fluxes.

Voice-coil motor. Constructed like solenoids, these motors come in moving-coil and moving-magnet configurations and produce more precise motion than solenoids. Voice-coil motors maintain high linearity between applied current and developed force; operate efficiently; and, when built with high-energy magnet materials, develop high forces and acceleration rates, to 100 g in some versions, and can oscillate at high frequencies. However, as with a solenoid, the simple mechanical structure precludes long strokes.

Single-axis actuators. Another short-stroke linear device is the single-axis actuator. Recent developments have focused on the actuation driver that provides the linear motion. One driver is a polymer that expands under applied current. The other driver is an alloy material that expands when subjected to a magnetic field.

The polymer-based actuator combines the small size of solenoids, the forces of hydraulic cylinders (to 500 lb), and the proportional control of electric motors. One of its benefits is that it lets engineers stay under the 20-lb limit of traditional short-stroke actuators without buying custom solenoids. These actuators can operate hydraulic valves and brakes, replace a ball screw system, actuate controls in car engines, and control robotic gripper manipulators.

Rather than magnetics, it uses a thermally reactive polymer. Heat is supplied by an electric heating element that consists of an electrically and thermally conductive carbon-fiber or silicon-carbide grid. The heat expands the polymer that pushes against a piston, Figure 11. As the polymer cools, the piston retracts.

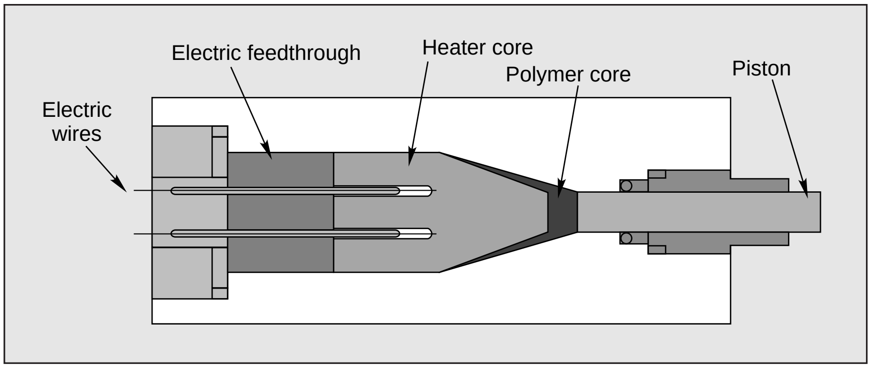


Figure 11 — Cross-section of a solid-state actuator. Heat expands a polymer, which pushes against a piston.

Actuation speed depends on the heat sink temperature, applied power, and applied load. Polymers are available to react at temperatures from -50 to 625 F. In some cases, the heat generated in the application can be used to activate the actuator.

Polymer expansion is directly proportional to the received electric power. It can be precisely positioned anywhere in its range of travel by varying the electric power from 60 to 200 W. Depending on heat sink temperature and the weight of the load (minimum of 25 lb), continuous power input is needed to maintain a specific temperature for the piston to hold position.

Cooling occurs when electrical power is reduced or removed from the device. The polymer is stiff, with a bulk modulus (hydraulic stiffness) of 1 million psi.

The other new linear actuator has a drive rod made of Terfenol-D, a magnetostrictive metal alloy of terbium, dysprosium, and iron. It uses magnetostriction, where an applied magnetic field causes the material to change its geometric dimensions.

The actuator has copper wire coil and permanent magnets housed in aluminum or stainless steel, Figure 12. The copper wire is wound around the drive rod. When current of 1.4 to 3.4 A from an external power supply is applied to the coil, it creates a magnetic north-south orientation at the molecular level. This new orientation causes the drive rod to lengthen as the diameter shrinks. When current is removed, the rod returns to its original shape.

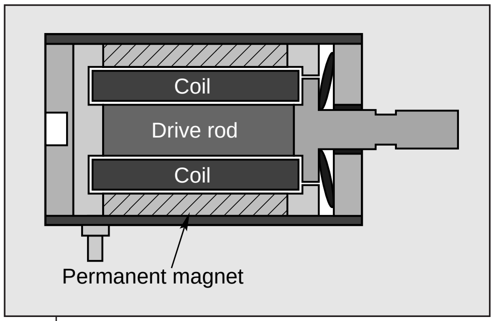


Figure 12 — The actuator consists of a drive rod made of the magnetostrictive alloy Terfenol-D, copper wire coil, Alnico permanent magnets, and a magnetic return circuit housed in aluminum or stainless steel.

The rod responds to the application or withdrawal of current almost instantaneously. The expansion is proportional and repeatable. The amount of force that the actuator supplies to linearly move an object depends on the size of the rod. A 12-mm diam rod can exert at least 200 lb of force. A 75-mm diam rod can exert at least 9,000 lb. Commercial versions of this actuator have available displacements in the thousandths of an inch to over 2 in.

Thus, these actuators are for applications that need high speed and high force, such as machining.

Linear induction motor. A linear induction motor resembles a common rotary induction motor that has been split axially and rolled out flat. Its speed-thrust curve, Figure 13, resembles the speed-torque curve of a standard ac NEMA Design B rotary motor. In operation, the moving part of the motor lags its synchronous speed to develop thrust. Also, speed depends on power-supply frequency. Efficiency is low except when the motor operates near design rating.

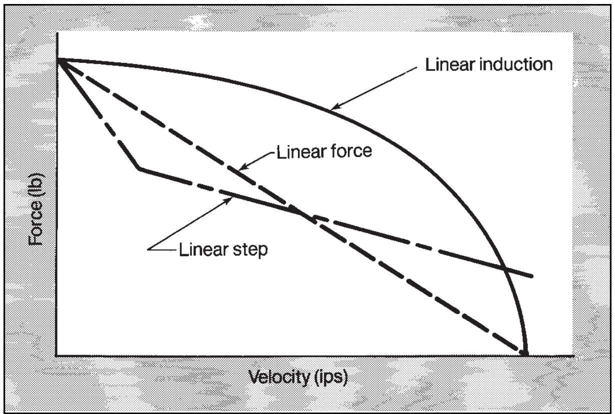


Figure 13—Typical speed-thrust curves of linear induction, force, and step motors.

The primary is a wound structure, much like a conventional motor stator. The secondary is a metallic structure, much like a rotor. Either structure can be the moving part of a linear induction motor.

The motor can operate directly from line current with a fixed speed thrust characteristic. It can also operate in an open loop from an adjustable-speed source for adjustable-speed applications. And it can operate as a servo system with a closed position loop. Figure 14 shows such a servomotor system in which the linear measuring system could be an encoder feeding position data back to an ac vector drive. (For more on vector drives, see the "AC vector" section in the Adjustable Speed Drives Product Department in this handbook).

The linear induction motor in Figure 14, sometimes called an asynchronous linear motor, is a single-coil motor — the primary part (similar to a stator in a rotary motor) holds the windings. The secondary part (like a rotor in a rotary ac motor) consists of iron and short-circuit rods of copper or aluminum. The magnetic field in the secondary is generated by the current induced by the moving and alternating magnetic field of the primary. In most cases, the secondary coil is stationary; the primary, with the moving part of the system.

The thrust produced in an ac linear motor is approximately proportional to the face area between the primary and secondary parts. A modification of the single-coil primary system is a dual-coil system, in which a primary part mounts at either side of a flat secondary. In effect, that doubles the working face area and thus the thrust of a similarly sized single primary system. In most applications, the dual primaries are fixed and the secondary coil is with the moving part of the system. Thus, you would consider first a single-coil primary system for long-stroke applications; a dual-coil primary for shorter-stroke, higher-thrust applications.

Linear force motor. Like the linear induction motor, you can think of a linear force motor as a conventional dc motor that has been slit axially and rolled out flat, Figure 15. It comes in moving-magnet, moving-winding, brush, and brushless types, and with various core materials.

Brush-type motor is the least expensive. Operating from direct current, each motor has a stationary coil assembly and moving magnets. The motor cable does not move in this configuration. Velocity goes to 1 m/sec; and acceleration to 0.5 g. Above these values there is excessive arcing and rapid deterioration of the brushes. These motors are excluded from clean room and vacuum applications.

Brushless, aluminum-core linear motor operates from three phase power, with a moving coil and cable. In applications requiring short travel lengths, the coil can be stationary and the magnets moving. The core of the primary encloses the windings in aluminum. There is no magnetic attraction between the two motor parts. Therefore, it may require a double-sided magnet assembly to close the magnetic circuit effectively.

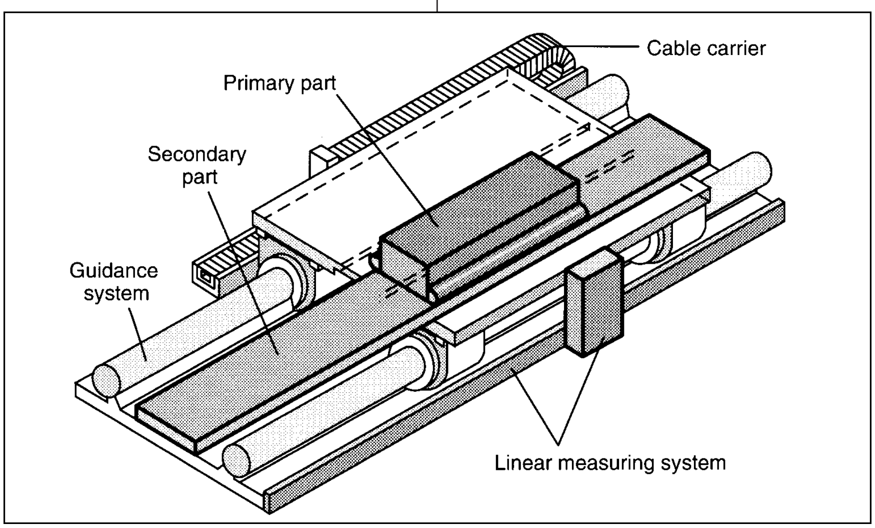


Figure 14—Typical ac linear servomotor. It could be powered by a vector drive that receives position and speed information from the linear measuring system.

Table 1–Typical linear motor performance specifications

Type	Max stroke	Max force (lb)	HP	Acceleration (g)	Resolution (in.)
Voice coil	0.5 in.	100	Less than 1	Several g	Less than 0.001
AC linear induction	Several ft	1-3000	1-10	Less than 1	Less than 0.1
AC linear ind.servo	100 ft	25-1800	1-10	10	Less than 0.0002
DC linear force	0.5-2 ft	500-1000	1-5	1	Less than 0.001
Step	0.5-2 ft	10-25	Less than 1	1	0.025
Microstep	Several ft	5	Less than 1	1	0.00004

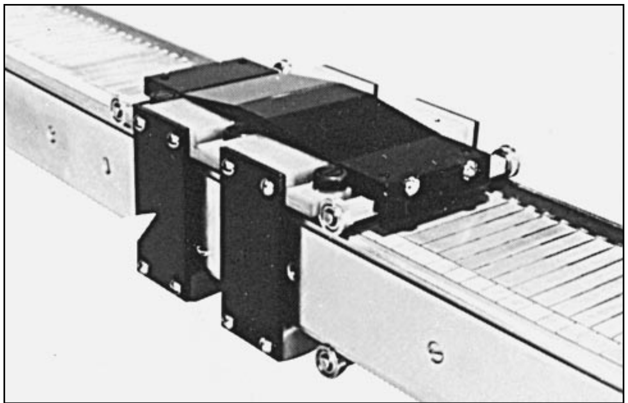


Figure 15—Typical linear force motor.

This type of linear motor generates smooth motion.

Depending on the moving load, it can accelerate to 4 g's, but can also develop eddy currents at speeds over 1 m/sec. It is ideal for vacuum applications.

Brushless, epoxy-core linear motor is also non-ferrous with its epoxy-based core. It too, provides smooth motion. It has an advantage over aluminum in that it does not experience eddy currents at high speed. It has low stiffness (typically 35,000 N/mm) at high coil temperatures (to 125 C) and it tends to give off gases in high vacuum environments.

Brushless, steel-core linear motor uses steel lamination in the main body of the primary to enclose the copper windings. This motor uses a single-sided magnet assembly and is air or water cooled for high duty cycle applications. It functions well in applications that require such duty cycles and velocities up to 200 ips (5 m/sec). But this motor is subject to cogging. It can generate strong magnetic attraction between the two motor parts, which must be accounted for in the load carrying capacity of the bearing system. Typical applications are machine tools where high force, 5,000 lb (23,000 N), may be required, or in general automation where speed of several m/sec is needed, 120 to 200 ips (3 to 5 m/sec).

Moving-magnet linear motors. This motor is a special case of a linear force motor. It uses an unconventional magnetic circuit in a cylindrical armature, similar to the shape of hydraulic and pneumatic actuators, and a three-phase wound stator assembly, Figure 16. The design helps its speed approach that of voice coil actuators and is spatially efficient to accommodate an increase in the amount of magnet material and coil windings in the actuator. Minimum magnetic circuit lengths and short air gaps aid the efficiency and force capability.

The cylindrical design aids acceleration, which ranges from 1,000 to 1,500 ft/sec^2. The working gap length, or circumference of the armature, is long versus the amount of material in the armature. A 600 lb force output version demonstrated speeds to 90 in./sec. The force/mass ratio was 1,360 ft/sec^2. Maximum velocity is limited by the drive voltage available and the travel distance, Figure 17.

These motors can be synchronized with the electrical sequencing of the field coils. This ability provides controlled velocity, acceleration, and positioning without additional sensors.

They achieve high detent forces through the permanent magnet design and very small air gaps. Detent forces are the result of changes in the reluctance of the magnetic circuit in the linear motor. Permanent magnets, even with the coils off, still produce flux. The armature seeks a position of least reluctance and attempts to remain in that position. The force needed to move the armature, with the coils off, is the detent force, which cannot be completely eliminated in these motors. It can be varied over a range of about 5 to 15% of the maximum force capability.

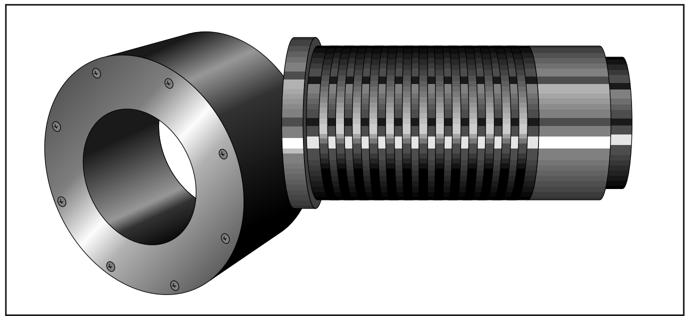


Figure 16 — The stator assembly (left) shows the windings that are sequentially activated to move the armature (right). The actuator, including its own weight, can accelerate at a rate of 8,600 in./sec^2 when driving a load of 23 lb. The stator is 9.0 in. OD $\times$ 4.9 in. ID $\times$5 in. long. The armature is just under 4.9 in. in diameter $\times$ 8 in. in length. Armatures can be made to almost any length to accommodate the needed travel distance.

Low detent motors are available. Reducing the detent involves the same techniques used in dc motors to reduce cogging, i.e. use of smaller pitch for the coils and magnets, a larger pole width relative to pitch, a larger air gap, or skewing the stator poles relative to the armature. Generally, however, lowering the detent force increases motor size and cost. These techniques also tend to reduce the force capability of the motor, re-

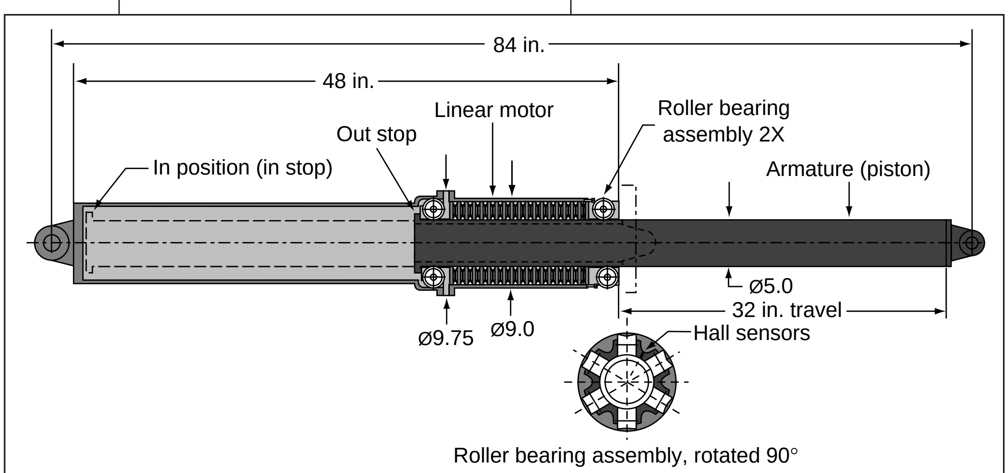


Figure 17 — This cylinder shaped linear actuator can develop 1,800 lb of force with a travel of 32 in.

quiring the use of a larger motor for a given force.

These motors are suited to applications requiring high acceleration, synchronized speed or acceleration, and high detent forces and reliability.

Linear step motor. This motor, Figure 18, provides the same incremental point-to-point precision as its rotating counterpart.

A linear step motor has a toothed, magnetic pole structure on the stator (platen) and on the slider (forcer). Platen and forcer tooth structure almost match. For example, the platen may have 11 teeth in the same length in which the forcer has 10. By sequentially energizing two coils that operate in conjunction with a permanent magnet (oriented parallel to the axis of motion), the step motor can be made to move in one-quarter-tooth-pitch increments. You can get extremely fine resolution (to 25,000 steps/in.) with microstep controls.

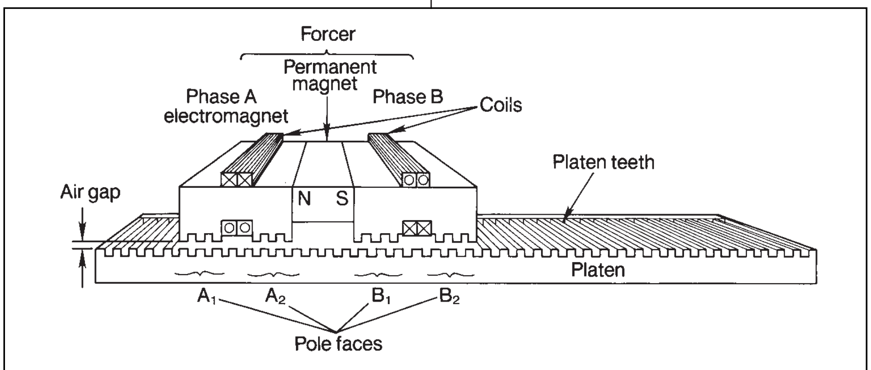


Figure 18—Linear step motor works on same principle as rotary step motor. Here, four sets of teeth on forcer are spaced in quadrature so only one set at a time lines up with any set of platen teeth.

Linear step motors are well-suited for positioning applications requiring high acceleration and high-speed, low-mass moves. Motor systems offer:

- High throughput. Linear step motors are capable of speeds to 100 ips, and low forcer mass allows fast acceleration.
- Mechanical simplicity.
- Reliability. Few moving parts and in some models, air bearings, make for long life and low maintenance compared with rotary systems.
- Precise open-loop operation. Linear systems allow open-loop unidirectional repeatability to 1 micron (0.00004 in.).
- Small work envelope. Most linear step motors need less space than equivalent rotary conversion systems.
- Long stroke. Travel length is limited only by platen length — and increasing length does not lessen performance.
- Multiple motion. By overlapping trajectories, more than one forcer can operate on one platen at one time.

BEARINGS

As with any other power transmission system, a linear motion system must be supported and guided. Generally, moving parts exert some force, and the force must be resisted for the system to remain stable. That is the chief reason for any bearing: It must bear a load. For information on bearings for rotary systems, see the Bearings Product Department in this handbook.

Linear-motion bearings are of many types, some much like rotary bearings. One of the most common ways to classify bearings is by type of bearing-to-load contact:

- Plain bearings. Surfaces slide on each other or on a lubricant film between them.
- Rolling-element bearings. Elements such as balls or rollers between moving surfaces provide the lower resistance of rolling friction instead of sliding friction.

Plain bearings. The simplest linear-motion plain bearing to visualize is the flat way, perhaps the oldest device that lets one machine element, such as the bed of a planer, move easily on another. However, it is also difficult to make well, because nearly perfect flatness is hard to maintain over a long distance. Early machine ways were hand-scraped by craftsmen to remove high spots. The flat way must also have a means to keep the payload from running off due to any transverse load. A common method is to provide not one flat surface, but two, butted at an angle to each other to form one V-shaped way. A companion V-shaped way mates with it. A variation of this technique uses a continuous V-shaped way with companion wheels or sheaves with mating circumferential cross sections. They roll on and are guided by the way. The wheels, however, make such a system a rolling-element system. Dovetail ways are another variation of the V-shaped way.

By replacing the flat way with a cylinder with axis parallel to the direction of motion and making the companion moving piece a cylindrical rod, you create a sleeve bearing. Now, however, you can use short, well-aligned sleeve bearings in series to support the linear-motion device.

Such a bearing looks like the sleeve bearing used to support and guide rotary motion. However, it supports axial motion, and the mechanics of motion may differ. A major difference: the hydrodynamic oil-film "wedge" that develops between a rotating shaft and a sleeve bearing above a certain minimum rotary speed isn't there when the shaft moves axially. Well-lubricated linear-motion sleeve bearings can serve well at low speed.

Chief among the differences in linear sleeve bearings are type of bearing material. Common bearing bronzes are often used, and so are graphites. So, too, are metal sleeve bearings with solid-lubricant inserts. Another type is the ceramic linear bearing—a metal sleeve coated with a hard ceramic. Solid plastic or metal-backed plastic sleeve bearings are also in common linear-bearing use.

Rolling-element bearings. You can gain the advantage of lower rolling friction and, thus, higher speed capacity by substituting rolling-element linear bearings for plain linear bearings, much like you can with rotary bearings. For example, you can put rolling elements between the simple flat-way plain mating surfaces to reduce friction. With proper restraining systems, the elements could be balls or rollers. Likewise for more complex systems such as V and dovetail ways. Ball, roller, and crossed-roller systems are in service there. Also, the flat way bearing can become more like a box beam with cam-follower bearings or similar bearings on two or more sides to guide a carriage along the length of the beam. All these devices are used successfully in various applications, depending on precision, complexity, and cost constraints.

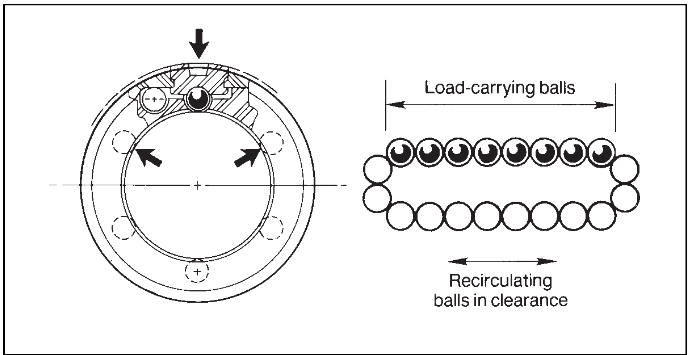


Figure 19—Principle of recirculating linear ball bearing.

Perhaps the simplest rolling-element linear bearing is the linear ball bearing. Figures 19 and 20 show a basic arrangement, with recirculating balls that travel an oblong path. A linear ball bearing has three or more oblong circuits of balls. Each circuit holds balls in one of it straight sides in rolling contact between the shaft and the bearing race. The load rolls along the balls in this part of the circuit. Balls in the rest of the circuit recirculate freely in the clearance in the sleeve. A retainer guides the balls and keeps them from falling out if the bearing is removed from the shaft. The retainer makes the linear ball bearing as easy to handle and install as a plain bearing.

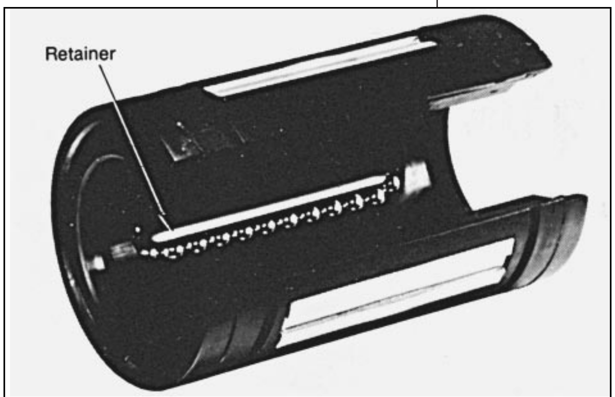


Figure 20—Typical linear ball bearing. Retainer guides balls and keeps them from dropping out should bearing be removed from shaft.

An example of a more complex rolling element linear bearing is a “smart” bearing, Figure 21. The smart designation comes from this bearing’s ability to recognize types of misalignment and to self-align around any or all of three axes independently. It can handle load capacities without giving up the ability to absorb torsional misalignment with no increase in stress levels.

Each double-track bearing plate in-

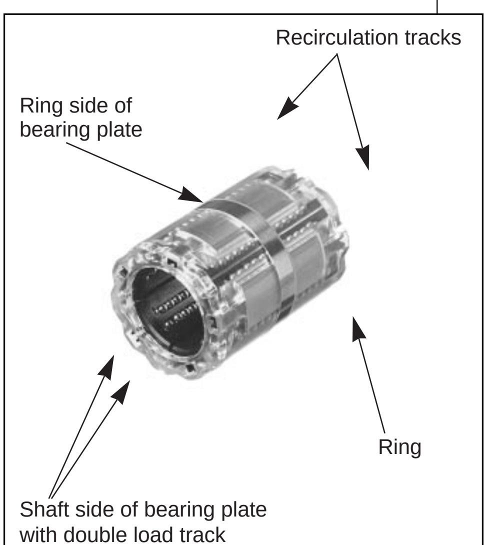


Figure 21— A transparent version of the Super Smart Ball Bushing bearing shows provisions for high capacity and for pitch, roll, and yaw. Steady-state speed can be up to 3 m/sec; coefficient of friction, down to 0.001.

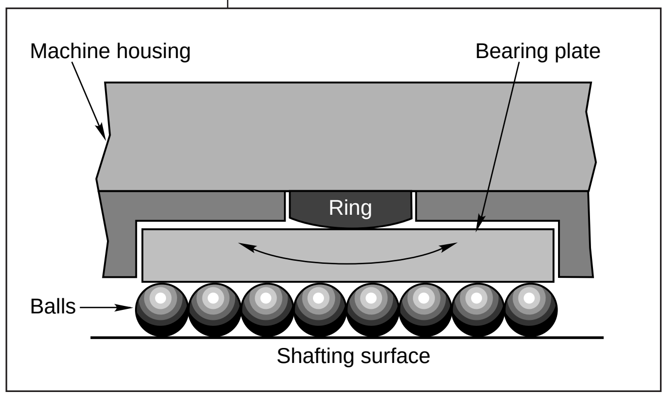


Figure 22 — Schematic axial cross section shows how bearing plate can self-align (“pitch”) about curved inner surface of hardened ring.

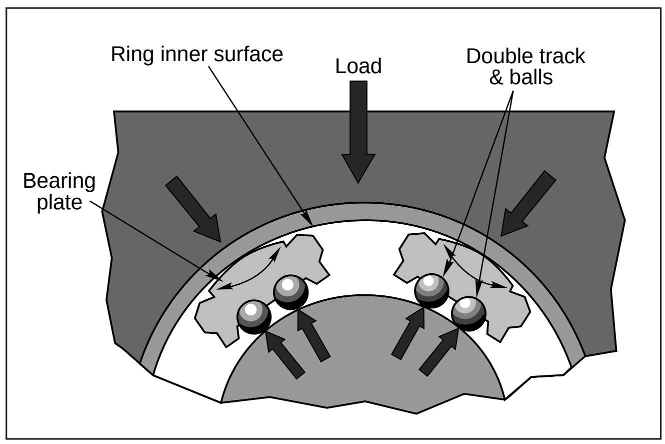


Figure 23— Double-track bearing plates have ring-side curvature that lets them self-align (“roll”) to distribute load evenly on their ball tracks.

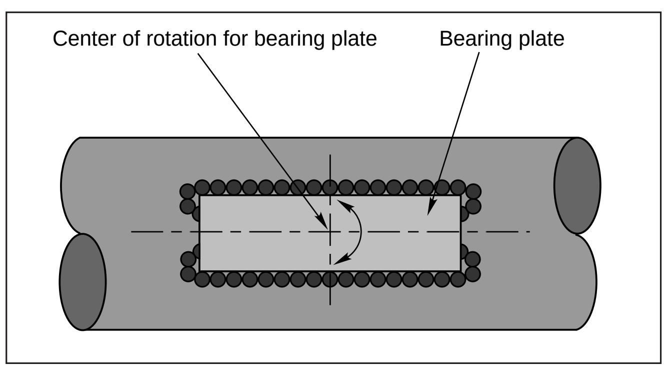


Figure 24 — Each bearing plate can turn about its geometric plan center to prevent skewing relative to shaft surface (“yaw”).

Table 2 — Linear ball bearing design variables

If you wish to determine:	And you know:	Then solve for:
Allowable load capacity	Bearing size Travel life required Shaft hardness	Allowable load capacity = rolling load rating $\times K_L \times K_H$
Travel life expectancy	Bearing size Load capacity required Shaft hardness	$K_L = \frac{\text{load capacity required}}{\text{rolling load rating} \times K_H}$ and read travel life from Figure 25
Minimum allowable bearing size	Load capacity required Travel life required Shaft hardness	Rolling load capacity = $\frac{\text{load capacity required}}{K_L \times K_H}$ and choose bearings with the next highest rating from catalog tables
Minimum allowable shaft hardness	Bearing size Load capacity required Travel life required	$K_H = \frac{\text{load capacity required}}{\text{rolling load rating} \times K_L}$ and read shaft hardness from Figure 26

dividually compensates for three types of misalignment:

- It compensates for shaft angular deflection or misaligned housing bore, or pitch, Figure 22.
- It evenly distributes load on its two ball tracks, compensating for roll, Figure 23.
- It rotates on a radial axis to eliminate skewing between ball tracks and shaft, or yaw, Figure 24.

Self-alignment minimizes friction, which holds performance and bearing life and simplifies installation.

Life expectancy and shaft hardness influence load capacity of a linear bearing-and-shaft combination. Circumferential positioning of load-carrying working tracks relative to applied-load direction is important. Life expectancy (travel life) is expressed as total inches of linear movement between bearing and shaft. Manufacturers' catalogs give normal and maximum rolling load ratings. Generally, they are based on travel life of 2 million in. Table 2 and Figures 25 and 26 show a way to compute expected life.

A linear ball bearing may form a continuous cylinder around a shaft, or it may have an axial split that lets it be preloaded on the shaft when the bearing is forced into an interference-fit housing.

Some linear ball bearings have "rolling keyways." Balls roll in a semicircular-cross-section axial groove to prevent rotation of the entire bearing unit on the shaft. This is much like a ball spline, but with only one ball channel.

A variation of the linear ball bearing is the linear-and-rotary unit, made up of a sleeve that holds a cylindrical array of balls in contact between the shaft and housing. It permits individual or simultaneous linear and rotary movement—a combination needed in many applications.

Precise, high-load, linear travel can be gained in many applications by linear roller bearings instead of ball bearings. Rollers pass over a flat channel, rolling in line contact between channel and load. At the end of the channel, rollers recirculate, running unloaded behind the channel to return between channel and load.

Roller skewing could cause instability and high friction. To prevent it, precise roller end guides, central sta-

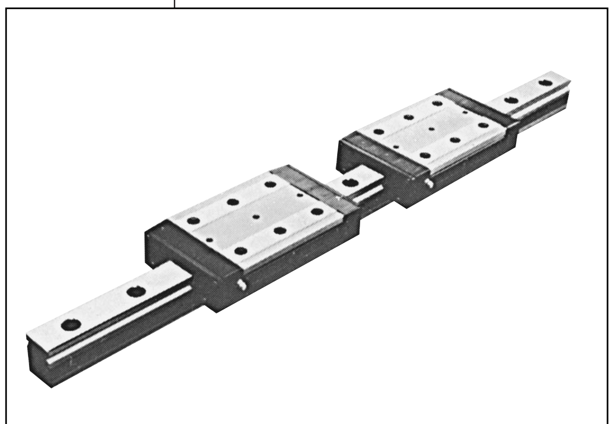


Figure 27 — Cylindrical rollers provide higher load capacity than balls, because they are in theoretical line contact instead of point contact. Linear roller way here is a rolling guide unit. Cylindrical rollers run on a track rail to get endless linear motion while circulating in a slide unit.

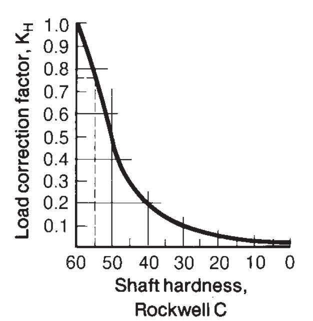


Figure 26 — Load correction factor for linear ball bearings based on shaft hardness.

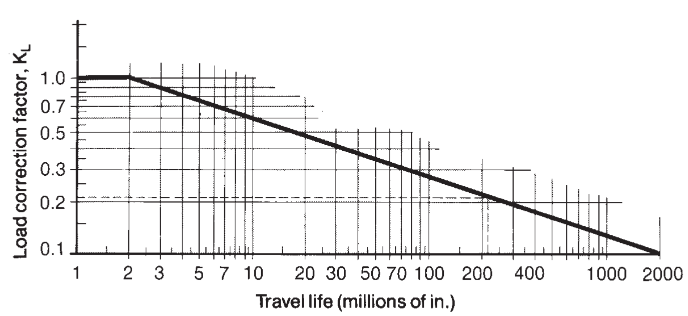


Figure 25 — Load correction factor for linear ball bearings based on required travel life.

bilizer bands, or similar means are provided.

Dynamic load capacity of linear roller bearings is generally quoted in terms of L_{10} life for a given travel life, such as 10 million in.

CONTROLS

Some sensing devices, such as proximity switches, work equally well for rotary or linear systems. Others, such as linear variable differential transformers, optical linear encoders, and force transducers, are especially for linear application.

Much control technology applicable to rotary motion systems applies as well to linear systems. Digital microprocessing allows it. For example, programmable controls and computer software simplify motion control and permit simultaneously controlled motion in many degrees of freedom. A case in point is a multiaxis positioning system that includes rotary as well as linear movement.

In many complex linear motion systems today, quality of movement depends essentially on system stiffness and resolution of the position-measuring equipment. For example, with the ac linear servomotor shown in Figure 14, it is possible to position within ± 1 increment of the encoder system.

The linear measuring system shown in Figure 14 is generic. It could serve many kinds of linear motion systems, and it could be:

- An optical linear encoder. Here a scanning unit consists of a light source, photovoltaic cells, condensing lens, and grating reticule. On a linear motor, it usually mounts on the primary unit. It moves relative to a linear scale with line grating, producing sinusoidal signals. The controller counts the resulting signals to establish position and speed.
- An encoder system using cable and pulley. With an industrial rotary encoder having a pulley on its shaft and a tension-controlled cable, you can sense position and speed.
- Rack-and-pinion encoder system. Here, a gear rack transmits linear movement. A pinion rotates as its meshing rack moves. A rotary encoder couples to the pinion shaft and provides position and speed information. Criteria for high accuracy are straightness and evenness of rack and pinion.

Interferential optical encoder systems are for applications demanding very high accuracy and resolution. Laser interferometer measuring systems can offer resolution of 0.01 μm. Common applications are inspection machines, extreme-precision machine tools, and wafer-slicing machines for semiconductor manufacturing. These devices require special consideration of the lasers' environment, such as air temperature, pressure, and cleanliness.

A growing tendency among linear-motion-device suppliers is to offer complete systems as well as components for linear motion. You can get some systems that include the actuator, its bearings, framework, drive and control, and sensors for control and safety limits. You need only bolt the system down and wire it up. Little or no adjustment or tuning may be needed.

For more about control components, see the Controls and Sensors Department in this handbook. ■