Motion Control Handbook

The following section covers a variety of technical topics related to precision positioning and motion control. It is our goal here to provide material that will help prospective users reach an informed decision concerning the suitability of specific positioning components for their application. We have noted the general dearth of information of this type, and are familiar with the tendency to hype "specs" at the expense of substance. Glib claims of "sub-micron accuracies" in particular, only have meaning when a comprehensive error budget is prepared, taking an integrated approach to both the positioning components and the specifics of the application.

We take our responsibility seriously as vendors of secondary reference standards for dimension. We also feel that our customers are better served by having more information at their disposal, not less. Feel free to contact us should you wish to discuss any of the material presented here, or examine the specifics of your application. We look forward to serving you.

In addition to the material presented in this section, more extensive application notes are available, which provide greater depth on a variety of positioning system topics. The application notes listed below are available free of charge through our Sales Department and on our website–www.NEAT.com.

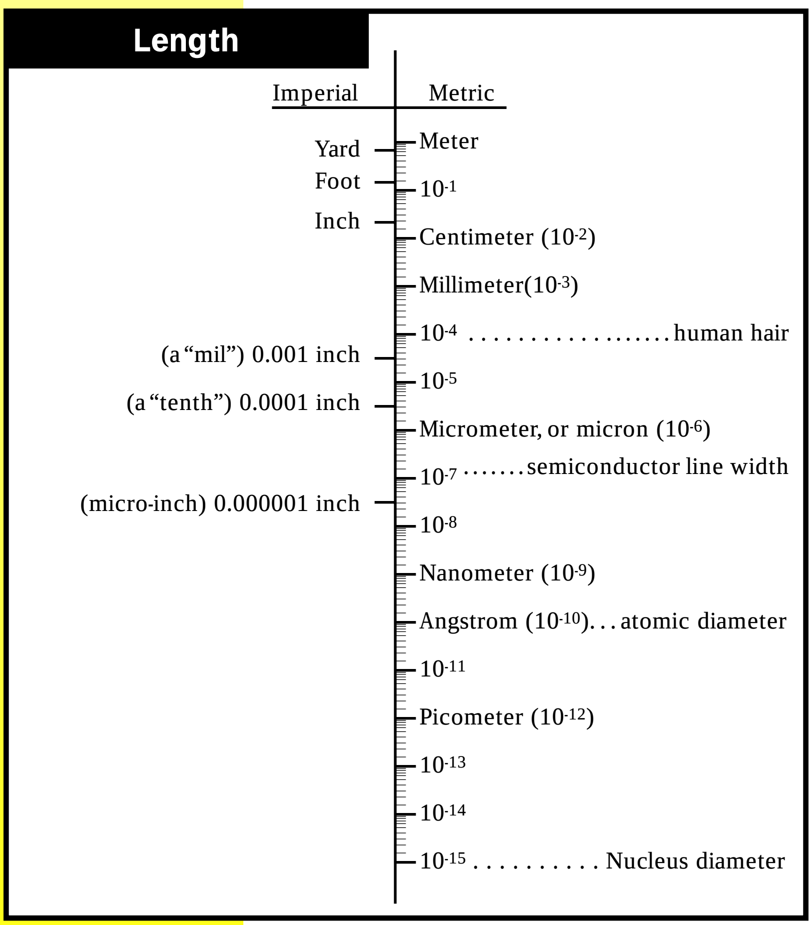
1. **Slow Down to Speed Up!**
   A discussion on how to optimize linear motor performance through careful consideration of move profiles.
2. **Accuracy in Positioning Systems**
   A look at the factors limiting accuracy, with emphasis on interferometer-based systems.
3. **Positioning Systems Overview**
   A broad overview of positioning technologies, examined from the standpoint of the individual components with which positioning systems are designed.
4. **Linear Motor Applications Note**
   A detailed look at linear motor systems, with several accompanying MathCAD spreadsheets that analyze system performance.

Contents

Units of Measure

While Imperial dimensions and thread standards remain popular in America, engineering calculations benefit from the metric system, and in particular the SI, or MKS (meter-kilogram-second) system. A number of dimensional units are employed when discussing positioning systems, and some may not be familiar to all users. The fundamental units are those of distance, mass, time, and temperature; all other units can be derived from these (we neglect here the equally fundamental Ampere, mole and Candela).

Time is employed uniformly in both the Imperial and SI systems; we have the second, millisecond (10^{-3}), microsecond (10^{-6}), and nanosecond (10^{-9}). Confusion creeps in where mass and force are concerned: in the SI system, the unit of mass is the kilogram, and the unit of force is the Newton. Weight is the gravitational force on a body and is proportional to its mass, W=mg. A kilogram force is the weight of 1-kg mass, and is equal to 9.81 Newtons or 2.2 pounds. The Imperial pound, ounce, etc. are actually units of force despite the fact that you can "convert" kilograms to pounds by multiplying by ~2.2; the units have taken a beating. The Imperial unit of mass is, of course, the slug; in general, it's better to forget the Imperial units, and stick to SI.



The SI unit of angle is the dimensionless radian, which is the plane angle whose arc length is equal to its radius. A full circle has 2π radians, and common subdivisions include the milliradian (10^{-3}), microradian (10^{-6}), and nanoradian (10^{-9}). The Imperial system, which is probably more familiar, divides a circle into 360 degrees; each degree into 60 minutes of arc; and each minute of arc into 60 seconds of arc (or more simply, arc-seconds). For comparison purposes, a radian is ~57.3 degrees, one arc-second is very nearly 5 microradians, and there are 1,296,000 arc-seconds in a full circle. The positioning community is fairly fond of degrees and arc-seconds, and as there is less ground for confusion here than was the case for mass and force, we employ both systems of angular units, as we wish.

The most common positioning units are those of length, for which the SI unit is the meter. Common subdivisions include the millimeter (10^{-3} m), micrometer (also called the micron, at 10^{-6} m), and nanometer (10^{-9} m). Since we are now working with applications whose resolutions are sub-nanometer, we should probably include the picometer, at 10^{-12} m. The chart below places these units in relation to both their Imperial counterpart (the inch), and recognizable objects of matching dimensions.

Torque is expressed in SI units by the Newton-meter; the corresponding Imperial unit is either the ounce-inch or the foot-pound. The SI units for linear and torsional stiffness are Newtons/meter and Newton-meters/radian, respectively; the Imperial equivalents are pounds/inch and ounce-inches/degree.

Accuracy

Positioning system accuracy can be conveniently divided into two categories: the accuracy of the way itself, and the linear positioning accuracy along the way. The former describes the degree to which the ways (ball and rod, crossed roller, air bearing, etc.) provide an ideal single-axis translation, while the latter is concerned with the precision of incremental motion along the axis (typically related to the leadscrew, linear encoder, or other feedback device).

WAY ACCURACY

Any moving object has six available degrees of freedom (Figure 1). These consist of translation, or linear movement, along any of three perpendicular axes (X, Y, and Z), as well as rotation around any of those axes (θx, θy, and θz). The function of a linear positioning way is to precisely constrain the movement of an object to a single translational axis only (typically described as the X axis). Any deviations from ideal straight line motion along the X axis are the result of inaccuracy in the way assembly.

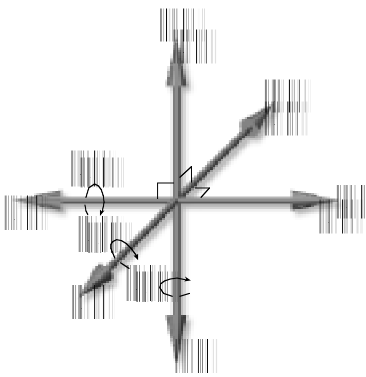


Figure 1 – Six Degrees of Freedom

There are five possible types of way inaccuracy, corresponding to the five remaining degrees of freedom (Figure 2): translation in the Y axis; translation in the Z axis; rotation around the X axis (roll); rotation around the Y axis (pitch); and rotation around the Z axis (yaw). Since there are interrelations between these errors (angular rotation, for example, produces a translational error at any point other than the center of rotation), it is worthwhile to carefully examine the effects of each type of error and its method of measurement.

Way Translation Errors

Since all useful methods of producing linear motion average over a number of points (due to multiple balls or rollers, or the area of an air bearing), "pure" translational errors from straight line motion (that is, without any underlying angular error) are usually minor. An exaggerated sine wave error in rolling element ways could achieve a pure translational error without rotation, as would the case of each roller in a way running over a contaminant particle at the same time; both of these cases are never encountered in practice. If a rolling element stage has been subjected to a large impact, the ways may be brinelled (dented) at each ball or roller location; this can result in a pure translational error that occurs periodically along the travel.

Positioning tables do nonetheless, exhibit some vertical and horizontal runout (typically referred to as errors of flatness and straightness, respectively), as can be measured by placing a sufficiently sensitive indicator on a table and measuring the vertical or horizontal displacement along its travel. A typical high-resolution measurement technique would mount a conductively coated optical flat on the stage under test, and monitor the runout with a capacitance gauge. This will reveal errors which can be divided into three categories:

1. **A potentially large component, which is roughly linear with distance.**
   This is due to a lack of parallelism between the optical flat and the ways. This can be eliminated by adjusting the flat so as to be parallel to the ways of the stage. Note, however, that to minimize vertical runout (flatness errors), the customer part must be similarly aligned parallel to the ways, which is not necessarily exactly parallel to either the base of the stage or its top.

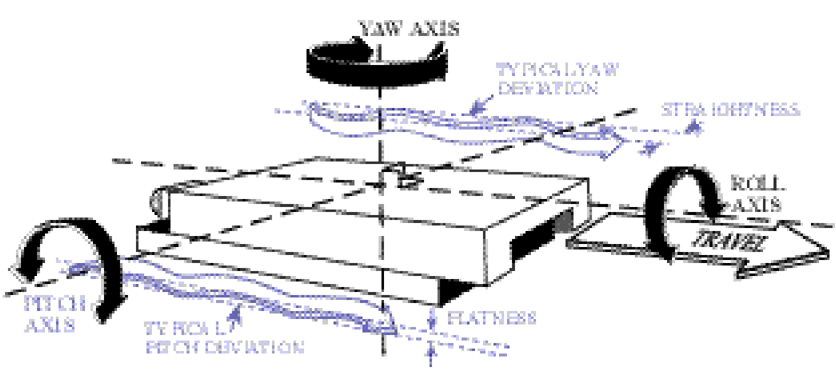


Figure 2 – Possible Way Inaccuracies

Accuracy (Cont.d)

2. **A low frequency component, which cannot be eliminated by adjustment of the optical flat.**
   This is rarely a "pure" translational error, but is rather a consequence of the underlying angular errors (pitch, roll, and yaw) in the ways. Since the moving portion of the stage follows (at some level) a curved trajectory, there is a corresponding linear deviation from a straight line. The angular and linear errors correlate quite well, and one can be obtained from the other by the process of integration or differentiation.

3. **Higher frequency components, which can arise from a variety of sources, not necessarily errors of the ways.**
   If a ballscrew is used, a once-per-revolution rise and fall of the table top can occur, especially near each end of travel. The use of flexurally coupled nuts and/or friction nuts can reduce this effect. Additional sources of higher frequency flatness errors can include microstructure in the ways or rolling elements, drive and/or motor induced vibration, and structural resonances in the stage top.

Since a number of optical positioning applications have limited depths of field, it is important to understand the magnitude of each of the above effects, and to modify the stage design so as to minimize the effects. The use of air bearings and linear motors can reduce total errors of flatness and straightness to less than 0.5 micron over 250 mm, and to less than 20 nanometers over 10 mm.

Way Angular Errors

The angular errors of roll, pitch, and yaw (θx, θy, and θz, respectively) are always present at some level in positioning tables, and degrade performance in several ways. Their direct effect is to vary the angular orientation of a user payload; due to the relative ease with which these errors can be maintained at low levels (1 - 50 arc-seconds, depending on stage technology), the effects of changing payload angle are of little consequence in many applications. Certain optical positioning tasks, however, may be directly impacted by angular errors.

Of somewhat greater concern are the translational errors resulting from underlying angular errors. The simple pitch error of ±16.5 arc-seconds shown in Figure 3, corresponding to a radius of curvature of 1 kilometer, will produce a Z axis translation of 20 microns in a half meter travel stage at either end of travel, relative to its centered position. Such simple pitch errors are typically found in non-recirculating table designs, due to the overhanging nature of the load at both extremes of travel. More complex curvatures, involving roll, pitch, and yaw, as well as multiple centers of curvature can also be encountered.

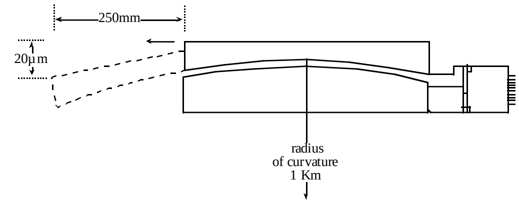


Figure 3 – Pitch Error

The worst impact of angular errors is the resulting Abbé (offset) error, which affects linear positioning accuracy. Unlike the simple translational error described in the above example, Abbé error increases as the distance between the precision determining element and the measurement point increases. This effect is described in detail on page 174.

Way angular errors are easily affected by the method of mounting the positioning stage (see page 179). In general, air bearings provide the ultimate in angular accuracy, as they have an inherently averaging effect, and their reference surfaces can be made very flat. The best stages can hold angular errors to as low as 1 arc-second per 250 mm.

Angular errors of a way assembly can best be measured using a laser interferometer. We employ a dual path optical assembly to eliminate sensitivity to linear translation, while providing 6.5 milli-arc-second (32 nano-radian) resolution for either pitch or yaw. The measurement of roll requires the use of a rectangular optical flat and either an autocollimator or a pair of capacitance gauges operated differentially.

LINEAR POSITIONING ACCURACY

A variety of techniques are available to incrementally position a user payload along a linear axis. Leadscrews and ballscrews are by far the most common, although linear motors, piezoelectric mechanisms, and belt drives are also used. Linear positioning accuracy is simply the degree to which commanded moves match internationally defined units of length. Ultimately, all length measurements are tied to the meter, as defined by the Comitee Consultif pour Definition du Metre. Its current value is the distance which light in a vacuum travels in 1/299,792,458 second.

Leadscrew-Based Systems

Low to moderate accuracy systems typically depend on a leadscrew or ballscrew to provide accurate incremental motion. Such systems are often operated open loop via stepping motors; if closed loop operation is employed, it is frequently with a rotary encoder. In either case, the leadscrew is a principal accuracy determining element. Leadscrews exhibit a cumulative lead error, which is usually monotonic in nature, together with a periodic

Accuracy (Cont.d)

component, which is cyclic and varies over each revolution of the screw. In addition, there can be backlash in the nut, which will reveal itself upon direction reversal. Precision positioning stages generally employ either a preloaded ballscrew, or a leadscrew with an anti-backlash friction nut. Ballscrews are preferred for high speed applications, and offer a high natural frequency due to their inherent stiffness. Leadscrews with anti-backlash nuts provide very high repeatability at modest cost, and are appropriate for most applications. Our leadscrews are available in both commercial and precision grades, with cumulative lead errors of 0.0001"/inch (1 micrometer/cm) for the precision grade, and 0.0004"/inch (4 micrometers/cm) for the commercial grade. Periodic error values are 0.0004" (10 micrometers) and 0.001" (25 micrometers) respectively. The above cumulative lead errors correspond to 100 and 400 ppm for precision and commercial grades, respectively.

It is important to realize that use of a leadscrew with a specified cumulative lead error, periodic error, and repeatability does not ensure that the positioning table will provide that level of accuracy. Among the factors which conjoin to degrade overall performance are thermal expansion, due both to ambient temperature changes and nut-friction induced heating, and Abbé error. Both of the latter effects produce different error values, depending on the location on the user payload. In the case of leadscrew thermal expansion, the position of the nut relative to the stage duplex bearing is important, while for Abbé error, it is the distance from the leadscrew centerline to the customer payload.

Geometry and Multi-Axis Errors

As mentioned above, angular errors in the stage ways degrade linear positioning accuracy through Abbé error. X-Y Tables have an additional parameter that impacts accuracy to a substantial degree: orthogonality, or the degree of squareness between the two axes. This parameter is held to less than 50 arc-seconds on our commercial grade tables, and less than 20 arc-seconds for precision models. For the latter case, a 300 mm travel corresponds to 30 microns of error due to orthogonality alone. We can, upon request, prepare tables which are square to within 10 arc-seconds; note, however, that trying to get the level of orthogonality lower than the value for yaw has limited meaning. Custom systems (typically air bearing designs) can hold orthogonality errors to below 2 arc-seconds. Another error source in systems with two or more axes is opposite axis error, which results when one axis has a straightness error. It is the job of the leadscrew or encoder on the other axis to provide accuracy in this direction, but since they are on two separate axes, this error is not corrected. Cosine error, or inclination of the leadscrew or encoder to the ways, is usually slight, but grows in importance with short travel, interferometer based stages. All of the above geometry errors are amenable to cancellation through mapping.

Linear Encoder-Based Systems

Use of a linear encoder eliminates concern over the leadscrew cumulative and periodic error, as well as friction induced thermal expansion. In many systems, the leadscrew can be dispensed with altogether and replaced with a non-contacting linear motor. With intrinsic accuracies on the order of 5 microns per meter, linearly encoded stages offer a significant increase in accuracy over leadscrew based systems, as well as much higher resolution (typically 0.1 to 1 micron). A number of error sources remain, however, and are often overlooked when specifying an encoder. The single largest error is often Abbé error, which can easily degrade accuracy by tens of microns. With a thermal expansion coefficient of ~10 ppm/degree C, linear encoders must be carefully controlled thermally to utilize their potential accuracy. An ambient temperature change of 1 degree C produces a 10 micron per meter error, double the encoders intrinsic 5 micron per meter accuracy. Contacting encoders are convenient, but read-head wind-up can be about half a micron, and higher if rubber sealing wipers are left in place. Non-contact encoders eliminate read-head wind-up, but can have tighter alignment requirements during installation. The encoder resolution itself defines an error source; a 1 micron resolution encoder moving from zero to +5 microns may display +2 microns when the read-head is actually at +2.7 microns, resulting in a 0.7 micron worst case error. Increasing the resolution below 2-5 microns generally requires electronic interpolation, which can also contribute low-level errors. In X-Y tables, each encoder fails to detect horizontal run-out in the other axis (opposite axis error), thereby ignoring translation along its measurement axis of potentially large magnitude (1 to 10 microns, depending on stage design, precision, and travel). Linear encoders are also incapable of correcting for orthogonality errors, which can range from 1 to 20 microns, again dependent on stage design, precision, and travel. Properly specified, linear encoders can significantly improve system accuracy, particularly if mapping is employed, but their limitations are frequently understated. In recent years, a variety of encoder designs have emerged which employ scattering or diffraction to determine position. The former employ a steel tape as the reference surface, resulting in a very convenient non-contact encoder system with resolutions to 0.1 micron. Linearity (slope) errors are present, on the order of 20 microns/meter, but these can be compensated for with a two point slope error correction. Diffraction based encoders permit the use of very fine grating pitches, and allow resolutions of as little as 10 nanometers. Despite these features, they remain subject to the error sources described above.

Accuracy (Cont.d)

Laser Interferometer-Based Systems

Laser interferometers are the ultimate position feedback device. They offer very high resolution, typically 10 nanometers in single pass and 5 nanometers in double pass. Intrinsic accuracy is better than 1 ppm for unstabilized sources, and as high as 0.01 ppm for stabilized designs. Abbé error can be virtually eliminated by appropriate location of the retroreflector or plane mirrors. Opposite axis error and table orthogonality error, intrinsic to encoders, can be eliminated in X-Y tables by the use of two plane mirrors, as shown on page 200. Among the barriers to achieving the very high intrinsic accuracy possible with laser interferometers is the variability of the speed of light in air. This value, constant only in a vacuum, is a function of atmospheric pressure, temperature, and humidity, as well as the concentration of other trace gases. The impact amounts to about 1 ppm per degree Centigrade, 0.4 ppm per mm-Hg pressure, and 0.1 ppm per 10% change in relative humidity. In actuality, the relationship (the Edelin equation) is non-linear, but the above linear approximations are valid for small changes near S.T.P. (760 mm-Hg, and 20 degrees C). Compensation for varying atmospheric conditions can be performed by manual entry, or by automatic sensing and correction term calculation, using precision environmental sensors and the system computer. Since atmospheric effects influence the entire air path between the polarizing beamsplitter and retroreflector (or plane mirror), it is important to minimize the "dead path" between the positioning table and the stationary beamsplitter.

Assuming that the beam path has been chosen so as to eliminate Abbé error, the remaining error sources (other than atmospheric effects) are thermal expansion of the user's part, the positioning table parts, and the base which mounts the table relative to the optics; differential flexing of the table top as it travels; cosine error; and imperfect squareness and flatness of the plane mirrors in X-Y assemblies. The use of "L" mirrors can replace two adjustable plane mirrors with a single glass L mirror; while this avoids concern about misadjustment, neither case can readily assure squareness below the ±1 arc-second level. This limits X-Y systems to a minimum of 5 ppm inaccuracy due to this effect alone; over a 300 mm travel, this accumulates to 1.5 microns. Single-axis systems, which do not have squareness to contend with, can achieve overall accuracies approaching several ppm (1-3 microns/meter), assuming exacting thermal management and atmospheric compensation, as well as beam angle trimming to minimize cosine error. At this level, positioning system design becomes a fairly elaborate exercise in HVAC (heating, ventilation, and air conditioning).

To illustrate the degree to which thermal issues complicate system design, consider a 300 mm travel single-axis table which seeks to achieve "tenth micron accuracy". One tenth of a micron over 300 mm is equal to 0.3 ppm. Recall that atmospheric compensation for laser interferometers is 1 ppm per degree C and 0.4 ppm per mm-Hg pressure. There will also be ~350 mm of base material (we will presume granite) between the table center and the stationary beamsplitter. The thermal expansion coefficient of granite is 6.3 ppm per degree C. If we choose to allocate our "error budget" of 0.3 ppm, assigning 0.1 ppm to atmospheric temperature, 0.1 ppm to atmospheric pressure, and 0.1 ppm to granite thermal expansion, then we have the following result: air temperature must be measured with 0.1 degree C absolute accuracy; pressure must be measured to within 0.25 mm-Hg accuracy; and the granite must be maintained at a constant temperature within 0.02 degrees C.

This analysis neglects thermal expansion of the user's part or the positioning table top, as well as cosine error, humidity changes, table top differential flexure, etc. Temperature changes in the interferometer optics alter the path length of the reference beam, introducing another error source, although specialized optics are available which reduce this effect. If the user's part is not maintained at exactly 20 degrees C, back correcting to that temperature requires precise knowledge of its thermal expansion coefficient, which is rarely available. Proper estimation and inclusion of all these error sources further exacerbates the thermal control requirements, often raising them to largely unachievable levels. Given that a fairly expensive laser interferometer fails to approach the needed accuracy levels in this application, the application of appropriate skepticism to advertising claims for stage accuracy is warranted.

Free copies of our Applications Note "Accuracy in Positioning Systems" are available upon request; contact our Sales Department to obtain a copy or visit our website — www.NEAT.com.

Repeatability

The repeatability of a positioning system is the extent to which successive attempts to move to a specific location vary in position. A highly repeatable system (which may or may not also be accurate) exhibits very low scatter in repeated moves to a given position, regardless of the direction from which the point was approached. Figures 4a, b, and c illustrate the difference between repeatability and accuracy.

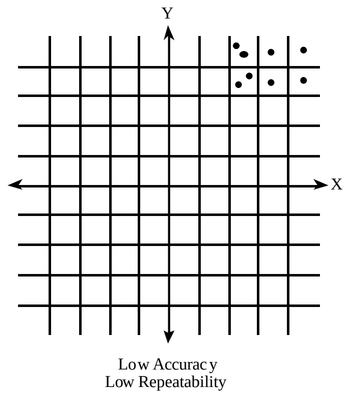


Figure 4a

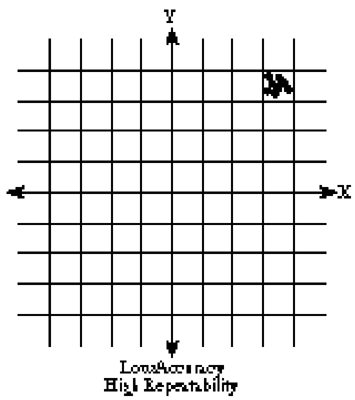


Figure 4b

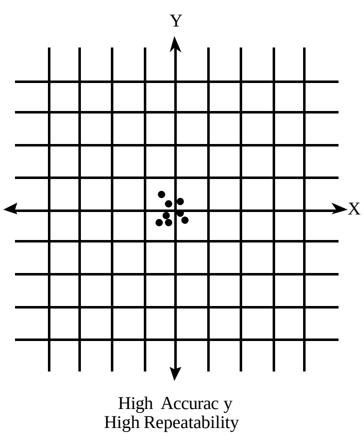


Figure 4c

A distinction can be drawn between the variance in moves to a point made from the same direction (uni-directional repeatability) and moves to a point from opposing directions (bi-directional repeatability). In general, the positional variance for bi-directional moves is higher than that for uni-directional moves. Quoting uni-directional repeatability figures alone can mask dramatic amounts of backlash.

Our repeatability testing is performed in the following sequence: the table is indexed to a point from one direction (say, from –10 mm to 0.000 mm). The measuring instrument (typically a laser interferometer) is then “zeroed”. The table then continues in the same direction to +10 mm, returns to 0.000 mm, and continues on to –10 mm. The move sequence is then repeated for 5 cycles, with positional data acquired at each approach to “zero”. Approaches to zero alternately display the uni-directional and bi-directional values, and the worst case deviations are recorded as the respective repeatabilities. There is a natural tendency to want to collect data from a large number of cycles, and statistically process these to prepare a 3 sigma value of repeatability. While this can be done to characterize closed loop positioning systems using a linear feedback sensor, repetitive move sequences with open loop or rotary encoded stages tend to generate some frictionally induced leadscrew heating, with consequent thermal expansion and positional drift. Accordingly, any of this catalog's repeatability figures for standard positioning tables (as opposed to complete servo systems) reflect the specific properties of the leadscrew and nut. The short-term nature of the repeatability test also eliminates any influence due to ambient temperature changes.

The degree of concern displayed above to eliminate thermal effects from the measurement of repeatability may seem overly exacting; it is driven, however, by the desire to properly showcase the very high intrinsic repeatability of our anti-backlash nut design (used with leadscrew-driven stages). Extensive testing with a laser interferometer reveals typical uni-directional values of below 0.5 micron (with many in the 0.1 to 0.3 micron range), and bi-directional values below one micron (with many in the 0.2 to 0.5 micron range). In addition, the self-compensating nature of the anti-backlash nut design results in little degradation of these values over service lifetimes in excess of 5 million meters.

When a very high level of repeatability is required, it is better to dispense with the use of leadscrews altogether, and substitute a linear motor as the actuating element. While this requires the addition of a linear encoder, and operation of the stage in closed loop mode with a servo controller, the resulting performance is greatly enhanced, and is limited only by the resolution of the linear encoder and the inevitable presence of thermal effects.

Resolution

Resolution is defined as the smallest positional increment which can be commanded of a motion control system. The mechanical positioning components, motor, feedback device, and electronic controller each play a role in determining overall system resolution.

In stepping motor systems, the resolution is set by the leadscrew pitch, motor step angle, and drive electronics. For any given pitch, two full step resolutions can be achieved through the use of either 1.8 degree or 0.9 degree stepping motors (which provide 200 and 400 full steps/revolution, respectively). This full step resolution can be further increased by microstepping (see page 210). Our microsteppers electronically subdivide each full step into 10 or 50 microsteps, producing 2,000 or 10,000 microsteps per revolution with 1.8 degree steppers, and 4,000 or 20,000 microsteps per revolution with 0.9 degree motors, respectively. While microstepping can be implemented with higher division ratios than 50, the increased resolution is often of limited use (see below). This chart provides resolutions for our full line of leadscrews and ballscrews, together with motor options and microstepping drives.

Popular resolutions for step motor stages include 0.0001 inch (achieved with a 0.20" leadscrew and a 200 step/revolution motor operated in ÷10 microstep mode), and 1 micron (by substituting a 2mm leadscrew). The key question to ask in determining the required system resolution is: "What are the minimum incremental moves which must be performed in a given application?" Resolution is easily over-specified, or confused with accuracy and/or repeatability. In general, it is appropriate to specify a resolution that is about five times smaller than the position error that is required by the application.

The resolution of servo systems which utilize rotary encoders is a function of the leadscrew pitch and the encoder resolution. Rotary encoders are characterized by the number of lines per revolution; our control electronics, however, can perform a 4x multiplication of the line count. For example, our RE-2000 rotary encoder has 500 lines, which is translated to 2000 counts per revolution in the counting electronics. The resulting linear resolution is shown in the accompanying chart. Use of the RE-2000 rotary encoder provides the same resolution for a given leadscrew as that of our divide by 10 microstepper.

The resolution of servo systems incorporating linear encoders or laser interferometers is independent of the screw pitch, and is strictly a function of the positional feedback device. In some cases, the leadscrew is replaced with a linear motor, which requires the use of a linear encoder. Standard DPS linear encoders provide resolutions of 5, 2, 1, 0.5, 0.25, or 0.1 micron, with interpolation electronics built into the encoder read head. Diffraction based linear encoders are optionally available, providing resolutions as low as 20 nanometers. Finally, laser interferometers can be supplied as feedback devices, providing a single pass resolution of 10 nanometers, and a double pass resolution of 5 nanometers.

Resolutions of Rotary Motor Systems

	Drive Screw	Stepper Motor												Servo Motor			
		1.8 Degree (200 Full Steps/Rev.)						0.9 Degree (400 Full Steps/Rev.)						With RE-2000[1]		With RE-4000	
	Lead	Full Step		÷10 Microstep		÷50 Microstep		Full Step		÷10 Microstep		÷50 Microstep					
ENGLISH	(inch)	(inch)	(um)	(inch)	(um)	(inch)	(um)	(inch)	(um)	(inch)	(um)	(inch)	(um)	(inch)	(um)	(inch)	(um)
	0.5 (2TPI)	0.002500	63.5	0.000250	6.35	0.000050	1.27	0.001250	31.75	0.000125	3.175	0.000025	0.635	0.000250	6.35	0.000125	3.175
	0.4 (2.5TPI)	0.002000	50.8	0.000200	5.08	0.000040	1.016	0.001000	25.4	0.000100	2.54	0.000020	0.508	0.000200	5.08	0.000100	2.54
	0.2 (5TPI)	0.001000	25.4	0.000100	2.54	0.000020	0.508	0.000500	12.7	0.000050	1.27	0.000010	0.254	0.000100	2.54	0.000050	1.27
	0.1 (10TPI)	0.000500	12.7	0.000050	1.27	0.000010	0.254	0.000250	6.35	0.000025	0.635	0.000005	0.127	0.000050	1.27	0.000025	0.635
	0.05 (20TPI)	0.000250	6.35	0.000025	0.635	0.000005	0.127	0.000125	3.175	0.000013	0.3175	0.000003	0.0635	0.000025	0.635	0.000013	0.3175
	0.025 (40TPI)	0.000125	3.175	0.000013	0.3175	0.000003	0.0635	0.000063	1.5875	0.000006	0.15875	0.000001	0.03175	0.000013	0.3175	0.000006	0.15875
	0.02 (50TPI)	0.000100	2.54	0.000010	0.254	0.000002	0.0508	0.000050	1.27	0.000005	0.127	0.000001	0.0254	0.000010	0.254	0.000005	0.127
METRIC	(mm)	(um)	(inch)	(um)	(inch)	(um)	(inch)	(um)	(inch)	(um)	(inch)	(um)	(inch)	(um)	(inch)	(µm)	(inch)
	10	50	0.001969	5	0.000197	1	0.000039	25	0.000984	2.5	0.000098	0.5	0.000020	5	0.000197	2.5	0.000098
	5	25	0.000984	2.5	0.000098	0.5	0.000020	12.5	0.000492	1.25	0.000049	0.25	0.000010	2.5	0.000098	1.25	0.000049
	3	15	0.000591	1.5	0.000059	0.3	0.000012	7.5	0.000295	0.75	0.000030	0.15	0.000006	1.5	0.000059	0.75	0.000030
	2	10	0.000394	1	0.000039	0.2	0.000008	5	0.000197	0.5	0.000020	0.1	0.000004	1	0.000039	0.5	0.000020
	1.4	7	0.000276	0.7	0.000028	0.14	0.000006	3.5	0.000138	0.35	0.000014	0.07	0.000003	0.7	0.000028	0.35	0.000014

[1]The RE-2000 rotary encoder (with standard 4x interpolation) has the same resolution as a 1.8 degree stepper with a ÷10 microstep drive. Using this combination can simplify quasi closed-loop systems (using the special 310M command).

Resolution (Cont.d)

As finer and finer resolutions are sought, an important distinction arises between the smallest increment which can be commanded, and the smallest increment which can be achieved. Unachievable, or "empty", resolution allows dramatic product claims to be made, but provides no useful benefit. For example, driving a 0.5 mm leadscrew with a 50,000 step per revolution microstepper, or a 50,000 count/revolution rotary encoded servo motor, produces a nominal 25 nanometer resolution. However, the friction present in the motor, nut, and ways (especially those of recirculating type), renders bi-directional moves of this magnitude impossible. Meaningful resolutions below the 100 nanometer level require a minimization of mechanical friction, typically through the use of air bearing ways and non-contact linear servo motor drives. An integrator is also required in the servo loop, to avoid the excessive gains that would otherwise be necessary to achieve such small incremental moves. Another difficulty with very high resolutions is the resulting limit on top speed, as the electronic counting circuitry sets a cap on the number of counts per second that can be processed.

Perhaps the ultimate level of positioning resolution has been achieved in the Scanning Tunneling Microscope, for which a Nobel Prize in Physics was awarded in 1986. In this device, and the related Atomic Force Microscope, piezoelectric technology and elaborate vibration isolation measures are used to achieve better than 0.1 Angstrom resolution (<0.00001 micron, or 0.0000000004 inch!), allowing detailed pictures of surface atomic structures to be viewed.

Note the missing atom in the following picture (Figure 5). This STM image shows a single-atom defect in iodine adsorbate lattice on platinum.

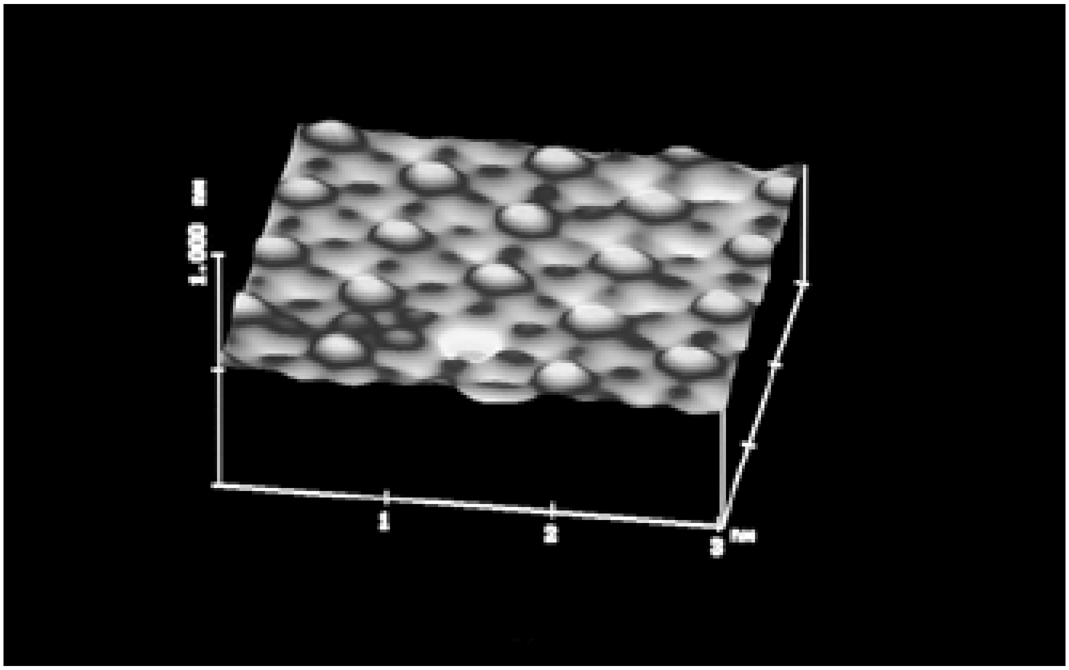


Figure 5 – Iodine Atoms on a Platinum Substrate

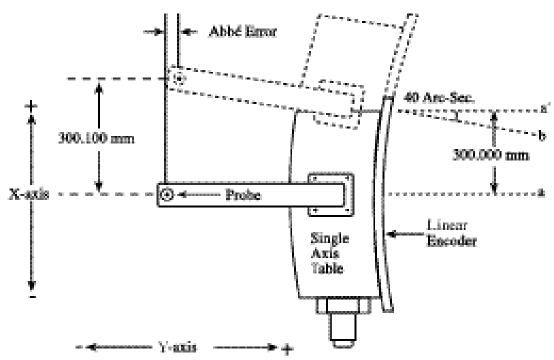
Image taken with NanoScope® SPM (Digital Instruments, Veeco Metrology Group, Santa Barbara, CA).
2.5 nanometer Scan Courtesy of Purdue University.

Abbé Error

Abbé error (pronounced ab-ā) can be a significant source of error in positioning applications. Named after Ernst Abbé, a noted optical designer, it refers to a linear error caused by the combination of an underlying angular error (typically in the ways which define the motion) and a dimensional offset between the object being measured and the accuracy determining element (typically a leadscrew or encoder). In open loop systems (or closed loop systems employing rotary feedback), the accuracy is nominally determined by the precision of the leadscrew. Similarly, in systems with linear encoders or interferometers, it is that device which determines the accuracy. It is important, however, to recall exactly what information these devices provide: Leadscrews really tell us nothing but the relative position of the nut and screw, and encoders tell us only the position of the read head relative to the glass scale. Extrapolating this to include the position of an item of interest, despite its firm mechanical connection to the nut or encoder read-head, is ill founded.

To illustrate this, consider Figure 6, which shows a single-axis stage with a linear encoder. The stage carries an offset arm which positions a probe over a sample. The apparent distortion in the stage is intentional; it is intended to illustrate, in exaggerated fashion, a stage whose ways have a curvature (in this case, yaw). Someone using this stage, and in possession of appropriate test instruments, would measure an error between the stage position as determined by the encoder read-head, and the actual linear position of the probe.

Top View



End View

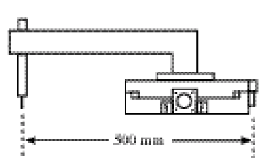


Figure 6 – Abbé Error Example

Suppose the curvature is sufficient to produce an angle a'b in Figure 6 of 40 arc-seconds (a' is drawn parallel to a). If the stage moves forward 300 mm, the probe at the end of the arm will be found to have moved +300.100 mm, resulting in an X-axis error of +100 microns. If the ways were, in fact, curved in a circular arc as shown, there would also be a Y-axis shift of +25 microns. This Y-axis error would be eliminated (while the X-axis error would remain) if the angular error were a purely local property of the ways at the +300 mm location. 100 microns is quite a large error, and Abbé error is accordingly important among the error sources to be considered.

Abbé error is insidious, and can best be countered by assuming the presence of angular error in a system and then working to minimize both the underlying error and its effect, through design optimization and appropriate placement of leadscrews, encoders, etc. The best tool to analyze angular error is the laser interferometer, which, when used with special dual path optics, measures pitch or yaw with 0.025 arc-second (0.125 micro-radian) resolution. We measure roll using a rectangular optical flat, in conjunction with an autocollimator or two capacitance gauges in differential mode.

Sources of angular error include the following:

1. Curvature of ways
2. Entry and exit of balls or rollers in recirculating ways
3. Variation in preload along a way
4. Insufficient preload or backlash in a way
5. Contaminants between rollers and the way surface
6. Finite torsional stiffness in a way, leading to angular deflection driven by:
   a. external forces acting on the load
   b. overhang torques due to the load's travel
   c. overhang torques due to stage components
   d. an offset leadscrew mounting position
   e. friction due to wipers in a linear encoder
7. Mounting the stage to an imperfectly flat surface

Abbé Error (Cont.d)

In the example shown in Figure 6, Abbé error could be lessened by moving the encoder to the left side of the stage. Reducing the arm's length, or mounting the encoder at the edge of the sample (with the read head connected to the arm) would be more effective. Virtual elimination of Abbé error could be achieved by using a laser interferometer and mounting the moving retroreflector on the probe assembly. Note that the component positions shown in Figure 6 effectively control Abbé error due to pitch error of the stage, since the height of the probe and encoder are roughly equal. While the stage might exhibit a pitch error (rotation around the Y-axis), there is no corresponding vertical (Z-axis) offset needed to produce Abbé error. The third degree of rotational freedom, roll, corresponds in the illustration to the rotation around the axis of motion (X-axis). This would result in the gap between the probe and the sample varying as the stage moved.

In general, try to estimate or measure the magnitude of all three possible angular errors (roll, pitch, and yaw) in any given system under actual load bearing conditions. Then, look for any offsets between driving or measuring devices and the point of interest on the load. Calculate the Abbé error, and if it proves unacceptable, optimize the design to reduce either the offset or the underlying angular error. In general, systems built using precision lapped granite and air bearings, which do not extend the load beyond the table base at any point in the travel, are best at minimizing angular errors.

To determine the magnitude of Abbé error, simply multiply the offset by the tangent of the angle. In the example, this was: 500 mm x tan (40 arc-seconds) = 500 x tan (0.011 degrees) = 500 x 0.000194 = 0.100 mm. If the angle is known in radians instead of degrees, the problem is that much easier:

Abbé error = angle x offset.

For example, an angular error of 194 micro-radians, or 0.000194 radians, in conjunction with an offset of 500 mm will result in an Abbé error of 0.000194 x 500 mm = 0.100 mm (not coincidentally, the angle of 194 micro-radians was chosen to match the 40 arc-seconds of the previous example). Finally, a helpful rule of thumb is that the Abbé error will equal about 5 microns per meter of offset and arc-second of angular error; in Imperial units, this amounts to about 5 micro-inches per inch of offset and arc-second of angular error. Once again, 40 arc-seconds x 20 inches x 5 = 4000 micro-inches, or 0.004". The accompanying chart and figure may prove helpful in determining which offsets produce Abbé error for a given angular error.

Angular Error	Offset Axis	Error Axis
U x (roll)	X	none
U x	Y	Z
U x	Z	Y
U y (pitch)	X	Z
U y	Y	none
U y	Z	X
U z (yaw)	X	Y
U z	Y	X
U z	Z	none

Thermal Expansion

Thermal expansion poses significant constraints on the accuracy achievable in positioning applications. It becomes increasingly important as the table travel is increased. Ambient temperature changes are one obvious source of thermal expansion. As the positioning table warms, its constituent parts undergo expansion, at a specific rate for any material. Accordingly, to permit any two users to agree on what constitutes an "accurate" positioning system, all critical dimensional measurements world-wide are understood to take place at 20 degrees C (68 degrees F). Measurements performed at other than 20 degrees C must be corrected to their 20 degree C value, using the thermal expansion coefficient for the material being tested. Leadscrews, fabricated from stainless or low carbon steel, expand at a rate of 12 parts per million per degree C, while the aluminum stage body expands at 23 ppm/degree C. Mechanical stress due to the differential between these two expansion rates is eliminated by the design of our positioning tables, which captivates one end of the leadscrew using a duplex, angular contact bearing set, while allowing the other end to slide freely through a bearing providing only radial support.

The leadcrew is the principal accuracy determining element of many positioning systems. At 12 ppm/degree C, a 3 degree change in ambient temperature will result in an expansion of 36 ppm, and amounts to about 7 microns for a 200 mm travel stage. A one meter long leadscrew would experience a total expansion five times as great, or 36 microns. In some cases, ambient temperature can be easily regulated, while in others, it may be possible to simply measure the temperature and calculate a compensation value. An additional source of thermal expansion, however, is less easily avoided. As the leadscrew spins within the nut, it experiences a friction induced heating which can easily exceed ambient temperature changes. For example, to achieve 250 mm per second translational velocities with a 5 mm leadscrew requires that the screw rotate at 3000 rpm. Any screw/nut combination (including ball screws) which is designed to provide very high repeatability will have some level of frictional torque. At the torque of our standard anti-backlash nut, the leadscrew will experience an 8 degree C temperature rise at this speed, resulting in 10 microns of thermal expansion for every 100 mm of screw length. This amounts to 20 and 100 microns for the preceding 200 mm and one meter examples, respectively. This expansion is also hard to correct; it builds over a 5-10 minute period as the leadscrew gradually warms, and may grow and shrink as the duty cycle of the application varies. Motor heating may also superimpose an additional thermal profile across a positioning table, and can elevate leadscrew temperature. Shutting motor current down to an idle level while not moving can reduce this effect.

Note that the expansion is measured from the stationary (duplex bearing) end of the leadscrew; in our positioning tables, this has been located at the table end opposite the motor, to facilitate leadscrew assembly and removal. Thermal expansion often sets a reasonable "upper limit" on the accuracy with which leadscrews should be specified. If the system is to be operated open loop, there is no point in paying for leadscrew accuracy in excess of the expected thermal expansion. If, on the other hand, linear encoders are used (as described below), then leadscrew accuracy becomes even less relevant.

Use of a linear encoder to determine position, using either a servo control or "quasi-closed loop control" with a stepping motor, will obviate any leadscrew induced thermal (and other) errors. Since encoders generate negligible heat as the read head moves over the glass scale, friction induced heating can be ignored. Our linear encoders have a thermal expansion coefficient of ~10 ppm/degree C. To avoid the introduction of errors due to thermal expansion, ambient temperature must be carefully controlled.

Alloys are available (eg., Invar) with much lower thermal expansion coefficients at room temperature. The difficulty and cost of fabricating stage components from these materials, however, generally favors the use of other techniques to minimize error (typically, the use of linear encoders and/or tight temperature regulation, and laser interferometers). Thermal expansion can become a significant error source in very high accuracy systems, which will typically employ laser interferometers. In this case, the user's part, the positioning table components between the user's part and the retroreflector or plane mirror, the base plate between the table and the polarizing beamsplitter, and the interferometer optics themselves are all subject to thermal expansion and its resulting inaccuracy.

Thermal Expansion Coefficients for Various Materials		
6061 aluminum	23.4 ppm/°C	13.0 ppm/°F
low carbon steel	11.7 ppm/°C	6.5 ppm/°F
304 stainless steel	17.3 ppm/°C	9.6 ppm/°F
Invar	0.6 ppm/°C	0.3 ppm/°F
granite	6.3 ppm/°C	3.5 ppm/°F
NEAT linear encoders	9.4 ppm/°C	5.2 ppm/°F
fused quartz	0.6 ppm/°C	0.3 ppm/°F
Zerodur	0.1 ppm/°C	0.05 ppm/°F

Mapping

Mapping can be an effective tool to reduce errors in positioning systems. Sources of error amenable to correction via mapping include those due to leadscrew cumulative error, leadscrew periodic error, Abbé error, nut backlash, cosine error, and deviations from orthogonality in multiple axis systems. Essentially, mapping consists of measuring and recording the actual position of a stage, for later use in returning to that point. In most cases, the measuring instrument is used only to acquire data on the stage, and is not present during actual operation. Common calibration sources include laser interferometers and precision glass grid plates. The positioning system must have sufficient resolution to implement a corrective move to the desired degree of accuracy. As an example, consider a single-axis positioning table with 1 micron resolution. Nominally, a 100 mm move would require 100,000 steps. In this case, due to a cumulative leadscrew error, 100,000 steps actually results in a 100.013 mm move. Programming a move of 100 / 100.013 x 100,000 = 99,987 steps, will produce the desired 100 mm move.

Mapping is especially effective when a relatively small number of positions are required; in this case, a unique measured value can be used for each location. In other cases, one or more points can be recorded, and subsequent points inferred, or "interpolated" from the nearest measured values. In the above example, a 50 mm move would require 49,994 steps, under the assumption that the screw error is linear. Compensation for leadscrew periodic error requires several points for each revolution, substantially increasing the storage requirements. In leadscrew based systems, thermal expansion often sets a limit on the level of accuracy worth reducing by mapping techniques. Systems with linear encoders and/or linear motors are better suited to high accuracy mapping techniques.

Single-axis stages are mapped with the use of a laser interferometer and automated data acquisition software. X-Y tables require the use of a two axis laser with an L-mirror assembly, or a precision grid plate, with the latter technique being easier to implement on production stages. We have devoted significant resources to the acquisition of very high precision grid plates. We currently have plates of dimensions 190 x 190 mm, 300 x 300 mm, and 600 x 600 mm. These plates were exposed on one of the worlds' most accurate photoplotters, and the two smaller plates are fabricated on Zerodur, a glass-ceramic with nearly zero thermal expansion. The gridplates consist of 12 micron wide chrome lines on a uniform grid every 10 mm, and are mounted in a fine adjustment tip-tilt-yaw stage, which is in turn placed on the X-Y stage under test. The stage is mounted on a large granite surface plate, under a bridge with a centrally mounted, precision Z-axis focusing stage. A long working distance 100X microscope objective, infinity corrected optical system, and machine vision camera and software complete the system. This mapping station allows us to provide a file of stage X-Y errors, and permits numerous error sources to be compensated. The net result is an improvement in X-Y accuracy of from ten to twenty times that of an unmapped stage.

Cosine Error

Cosine error results from an angular misalignment between the motion of a positioning table, and the accuracy determining element (leadscrew, encoder, or laser interferometer beam path). Under most circumstances, it has a negligible effect on overall accuracy, owing to the significant degree of misalignment needed to influence accuracy. Consider, for example, the case of a 300 mm travel positioning table with a linear encoder. The encoder is pitched so as to be inclined to the direction of motion, and the encoder will accordingly measure a larger move than has actually occurred. Pythagoras's theorem ($a^2 + b^2 = c^2$) yields the magnitude of the error. At a 100 micron misalignment, the encoder path equals $\sqrt{300^2 + 0.1^2} = \sqrt{90{,}000.01}$, or 300.0000167; the error is only 17 nanometers. If the misalignment is specified in terms of angle, then the error will equal: travel x (1 – cosine of theta), hence the name, cosine error. In the above example, the angle was 68.75 arc-seconds, and the cosine of theta equals 0.999999944.

If the encoder resolution is one micron, then a misalignment of 800µm would be necessary to generate a cosine error equivalent to a single count. Our stage design, fixturing, and inspection procedures hold leadscrew and encoder alignment to levels far below this value, rendering cosine error of negligible consequence in most NEAT positioning stages. In systems using laser interferometers for positional feedback however, simple visual alignment with a reduced aperture can introduce cosine error on the order of 5 ppm. This is significant when compared with the intrinsic interferometer accuracy of <1 ppm, and may necessitate careful adjustment of the beam angle in pitch and yaw to maximize the measured distance. Note that with laser interferometers, cosine error results in a distance measurement smaller than the actual move; this is opposite to the effect of cosine error for a linear encoder.

Mounting Issues

The care which must be taken when mounting a precision positioning stage is often underestimated, especially if the full accuracy intrinsic to the stage must be realized. When investigating claims of stage inaccuracy, we regularly determine that inadequate mounting provisions are the root cause of the errors. To better appreciate the sensitivity of mounting errors, consider the idealized example illustrated below (figure 7). A linear stage is bolted to a perfectly flat surface, using fasteners separated by 200 mm. A human hair, which is 75 microns in diameter (and presumed, for discussion purposes, to be incompressible) happens to lie on the mounting surface. The precision ways will now follow an arc, and the resulting Abbé error (see page 174) will produce an X axis positioning error of 75 microns at a position 100 mm above the leadscrew (or linear encoder), or 38 microns if the customer payload was mounted 50 mm above the leadscrew. When you may have spent some money to obtain a positioning accuracy well below this level, the need to carefully consider the mounting surface takes on new meaning.

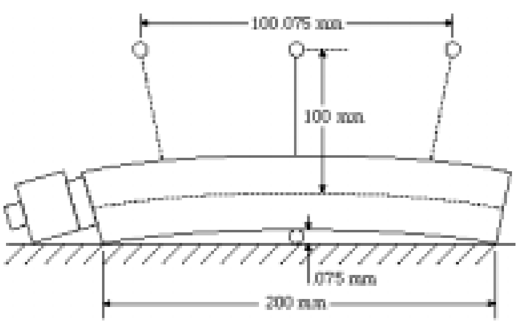


Figure 7 – Mounting Error

In general, positioning stages are best mounted to precisely flat surfaces of reasonable cross-section. A useful guide is to make sure that the mounting surface flatness exceeds the desired stage flatness. If the stage can hold pitch to below 10 arc-seconds (50 microradians), then the surface should be flat to better than 5 microns per 100 mm. As previously illustrated, care should be taken to ensure that the surface is clean, and that no foreign particles lie under the stage. Tapped holes should be carefully de-burred, and fasteners should be set to designated (and uniform) torques. Lapped and properly plated aluminum and/or steel surface plates of adequate thickness make ideal mounting surfaces. Granite or ceramic surfaces can be made extremely flat, but it is imperative that the threaded inserts for fastening to the surface be installed prior to final lapping. If these inserts are installed afterwards, they may lie slightly above or below the mounting surface, with the potential to seriously degrade accuracy.

Some customers prefer to avoid issues of surface flatness through the use of spherical washers. While these can, in fact, eliminate concerns over surface flatness, this is only the case if three (not four) spherical washers are employed, and most stock stages do not offer a triangular hole pattern. Stage flexure when supported on only three of the four mounting points will degrade accuracy, and even if three equally spaced holes could be provided, flexure will occur in conventional ball or crossed roller stages, as the center of gravity shifts during motion. When mounted to a properly flat and thick base plate, the stiffness of the base of the positioning stage is increased, an advantage which is lost when spherical washers are employed.

In some cases, it is necessary to mount two single-axis stages together. Our product line has been designed with this in mind; existing holes permit any linear single-axis stage to be easily mounted to any other such stage, in either an X-Y or coaxial orientation. The only rule is that the underlying stage must be of equal or larger cross section; for example, an RM stage will mount to another RM™, RMS™, TM™, TMS™, LM™, FM™, XM™, HM™, HMS™ stages, etc, but an HM stage cannot be directly mounted to an RM stage.

Move and Settle Time

Many applications emphasize throughput, and accordingly seek to minimize the move and settle time required for point-to-point moves. One of our differentiating characteristics is our ability to provide in-depth mathematical analysis of the dynamic behavior of our stages. We have developed extensive Mathcad and MathLab® spreadsheets which model and predict the performance of our positioning stages. To measure the dynamic response of our tables, and provide real-world feedback to our models, we use a tool with both high spatial and temporal resolution: the laser interferometer. Its combination of 1.25 nanometer resolution, 100 kHz position update rate, and non-contact optical position sensing are ideal for examining stage dynamics. We have written software which samples the laser interferometer at variable rates to 100 kHz, and the resulting data file is analyzed using Mathcad and MathLab software to reveal specific details of the system dynamics.

It is important to note that it is meaningless to specify either a move and settle time, or simply a settling time, without also providing a position window within which to settle. The first entry into this target window does not necessarily define the completion of the move and settle, as subsequent ringing may move the payload outside the window. Move completion is defined when the position is within the target window, and it remains there until the next move. In general, the smaller the target window is defined, the longer the move and settle time will be.

Move Time

The move and settle time for a given application can be conveniently separated into two halves: the move time, and the settling time. The time required to make a given move is a function of the commanded move trajectory, the available torque or force, and the inertia and friction of the load. The move trajectory includes the acceleration and deceleration profiles, the percent of the move spent at constant velocity, and the top speed which can be achieved. Short moves may never reach top speed, and will simply consist of an acceleration and deceleration phase. When stepping motors are used to drive a stage, the fact that their available torque falls off with speed requires either that a lower acceleration be commanded, or that the acceleration be decreased as the shaft speed increases. Stepping motors also have a very noticeable way of letting you know that they have inadequate torque for a given application – they stall. Once a specific acceleration profile and top speed have been selected, either by consulting a model or by empirical testing, the move time for any given move distance is easily calculated. Note that since stepping motor driven stages are open loop devices, the position to which they settle may not be the precise one that was desired (due, for example, to leadscrew error). A linear encoder can be added, and it can be interrogated at the end of the move to see if a small corrective move is in order; the next obvious step is to a servo system.

For rotary or linear servo motors, linear acceleration trajectories are typical, although an "S curve" profile may be used to minimize jerk. At low duty cycles, servo move times are limited by the available electrical resources (amplifier current and voltage), or by the risk of de-polling the permanent magnets of the motor. At high duty cycles, thermal limits on the servo motor coils set the minimum move times. In general, the calculations for move time are reasonably simple and deterministic, although a summary of the system inertias is required.

Settling Time, Stepper-Based Systems

The transient performance of a stepping motor driven positioning table is governed by two or more spring-mass systems, the first of which is composed of the moving mass (stage and load), together with the compliance of the nut, leadscrew, and duplex bearings. The other consists of the stepping motor rotor, knob, coupling, leadscrew, and reflected payload inertia, together with the compliance of the stepping motor's holding torque curve. Since the permissible following error of a stepping motor is limited to at most two full steps, in most cases the axial resonance will dominate, especially as the payload mass is increased. If the flexible coupling which connects the motor shaft to the leadscrew has inadequate torsional stiffness, or if long or large diameter leadscrews are chosen, then this constitutes a third possible resonance.

As in any spring-mass system, two key parameters define the behavior: the natural frequency f_0, and the damping Q. Typical natural frequencies for positioning tables lie in the 50 to 250 Hz range, with "Q"'s ranging from 10 to 40. The axial natural frequency can be observed by simply tapping a positioning table along its axis; such an "impulse" test is shown in Figure 8 (with an f_0 of 200 Hz). The Q is equal to 2π times the number of cycles required for the oscillation to decay to 1/e (37%) of its initial amplitude; in the case of Figure 8, the Q is ~25.

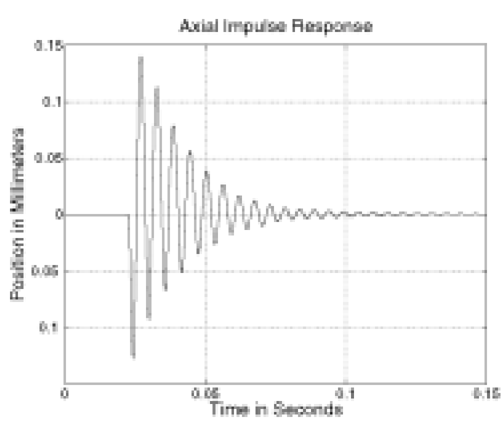


Figure 8

Move and Settle Time (Cont.d)

Since f_0 is proportional to the square root of the stiffness/mass ratio, significant reductions in moving mass, or increases in screw-nut and duplex bearing stiffness, are required to appreciably change the natural frequency. In some cases, such efforts can be counterproductive; switching from our anti-backlash nut to a stiffer ball nut can increase the natural frequency, but also increase the Q, resulting in a longer overall settling time. The inherent damping of our spring loaded, anti-backlash nut is effective in minimizing settling times. Figure 9a shows the settling performance of a single-axis stage executing a 1 millimeter move. As the graph indicates, moves may require several tens of milliseconds in which to settle to a rest; depending on the size of the target position window, this can take even longer. In the case of small moves (for example, disk drive track-to-track testing), the step excitation is smaller, and the settling time to within a given target window improves commensurately. We have developed extensive models of stepping motor driven stages, and can use these to predict stage performance.

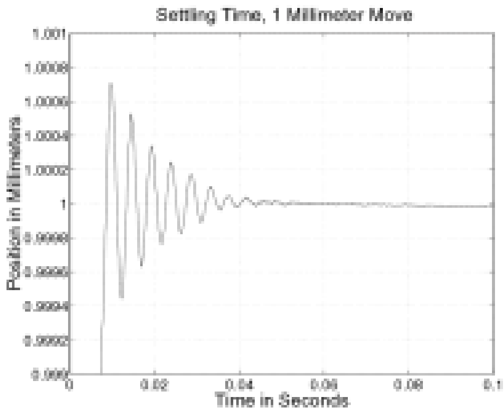


Figure 9a

Dynamic performance of a positioning table becomes key when short, high speed moves are performed repetitively, with a brief user function performed at each location. The overall throughput is then closely tied to how much of the cycle must be spent waiting for the stage to settle. Note that in such cases (Figure 9b), a high degree of repeatability can be obtained without waiting for the stage to settle; while not yet in its final location, the stage may be in a highly predictable position at a fixed time interval after the move termination. While useful for brief tasks such as firing a laser, longer tasks (such as a video frame grab, which requires 30 milliseconds) will see a "smeared" stage position. A useful rule of thumb in high repetition rate systems is to compare the natural frequency with the desired repetition rate: expecting a table with a 100 Hz natural frequency to move and "stop" at 10 Hz is achievable (assuming a well damped system); similar operation at 50 Hz is unrealistic. At or above 100 Hz, essentially sinusoidal motion with no discernible pauses will result.

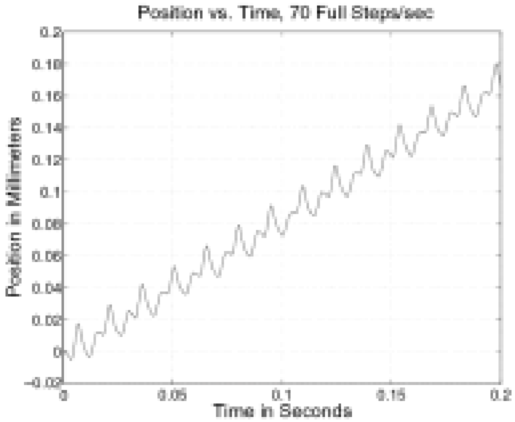


Figure 9b

Settling Time, Servo-Based Systems

Servo systems exhibit a number of dynamic behaviors that differ from those of stepper driven positioners. In most cases, servo controllers execute a PID loop, in which the commanded output is proportional to the error (the P term), the rate of change of the error (the D term), and the errors at previous samples of the system (the I term). A fundamental parameter of any servo loop is the servo bandwidth f_0, which is that frequency at which the servo loop's ability to counteract disturbances begins to roll off. In general, the response curve of a properly tuned (well-damped) servo loop is flat from D.C. out to the servo bandwidth, at which point the system response rolls off as $1/f_0^2$. The goal in selecting servo tuning parameters is usually to maximize the servo bandwidth, with the upper bound on the bandwidth usually set by phase lag from the lowest frequency mechanical resonance. Two other fundamental properties, related to the servo bandwidth, are the system natural frequency, ω_0, which is $2 \times \pi \times f_0$, and the time constant, τ_0, which is simply the reciprocal of ω_0.

The effect of the proportional term can be thought of as a torsional or axial spring (Figure 10). In this graph, the error, in microns, is plotted against the absolute value of torque or force. In all but air bearing systems, there will be a frictional component present. In systems without an integrator, the proportional term can only correct error down to the level at which friction is present; in the graph of Figure 10, the system would be left with a following error of 5 counts. The function of an integrator is to use the memory of the error in previous samples to push the command output above that produced by the proportional term alone. This will result in zero steady state error, but the effective time constant, τ_{int}, can be five to ten times the system time constant τ.

Since torque (or, in a linear motor system, force) is only produced in response to an error, a servo system will always exhibit a lag during acceleration and a lead during deceleration (Figure 11). The servo lead, or inertial following error, which is present at the end of the deceleration phase will equal:

$$\mathbf{4 \times Acc./\ \omega_0^2\ (in\ meters)}$$

Move and Settle Time (Cont.d)

where ω_0 is the natural frequency described above. For a given moving mass and servo bandwidth, we can also calculate the servo stiffness, (in Newtons per meter), which is the slope of the V shaped lines in Figure 10; this turns out to be equal to mass x $\omega_0^2/4$. The limit (in meters) to the following error that can be corrected by the proportional term can now be calculated — it is simply the friction (in Newtons), divided by the stiffness, (in Newtons per meter). We will refer to this as frictional error.

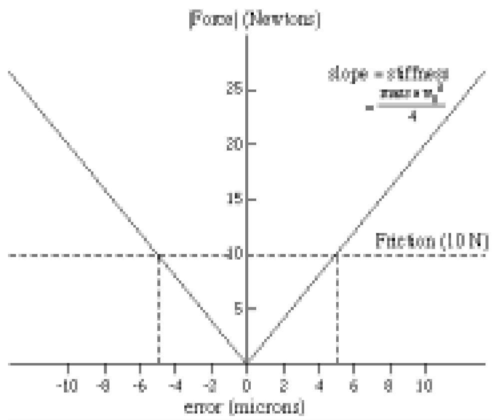


Figure 10 – Force vs. Error

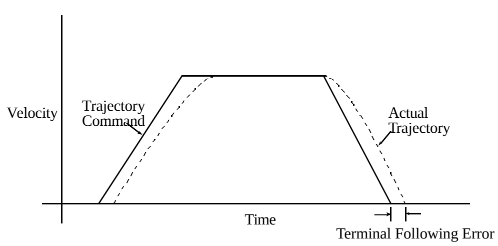


Figure 11 – Servo Lag

We can now examine the settling behavior of the system over time, with the key inputs being the desired settling window, the inertial following error, the frictional error, and the two relevant time constants, τ_0 and τ_{int}. In the absence of friction (the easiest case), the settling time is simply τ_0 times the natural log of the ratio (inertial following error / desired settling window). Put more simply, the inertial following error will drop by the factor of l/e (0.36) for every time constant, τ_0. If the servo bandwidth is 50 Hz, then τ_0 is 3.2 milliseconds. In a system with an acceleration of 2G (~20 meters/sec^2), the inertial following error will equal 811 microns. If our goal is to settle to within 2 microns, the time required will be 3.2 x ln(811/2), or 19.2 milliseconds. The price paid for friction will now become apparent. The servo stiffness of this system, if the mass is 10 kg, will be 246,500 N/m. With a friction of 10 Newtons, the frictional following error will be 41 microns. The settling behavior now consists of two phases; in the proportional phase, which settles to 41 microns at τ_0, the time required is 3.2 x ln(811/41), or 9.6 milliseconds. In the integrator phase, however, the time constant is 5-10x longer, and the system settles from 41 microns to 2 microns at the considerably slower τ_{int}, for a settling time in this last phase of ~20 x ln(41/2), or 60 milliseconds. The overall settling time is then ~ 70 milliseconds, compared to ~10 in the absence of friction (Figure 12). The above examples assume that no acceleration feed-forward is employed; this has the benefit of reducing (in some cases significantly) the inertial following error, and hence reducing settling times. Figures 13a and 13b show the effect of acceleration feed forward in reducing the overshoot (and hence improving the settling time) in a high acceleration (2G) move. Also assumed in the example above were a sufficiently fine encoder resolution, and the absence of external perturbing forces or, vibration.

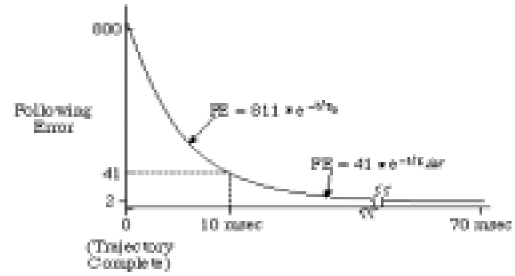


Figure 12 – Settling Time

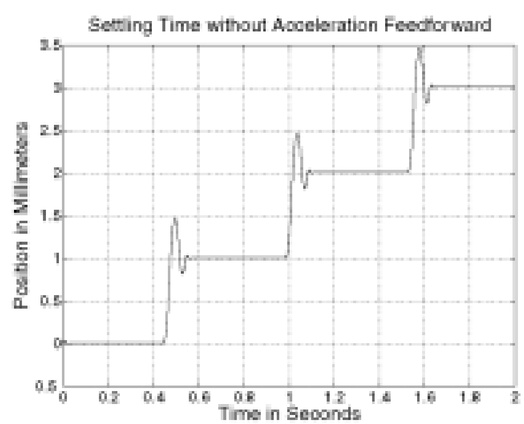


Figure 13a

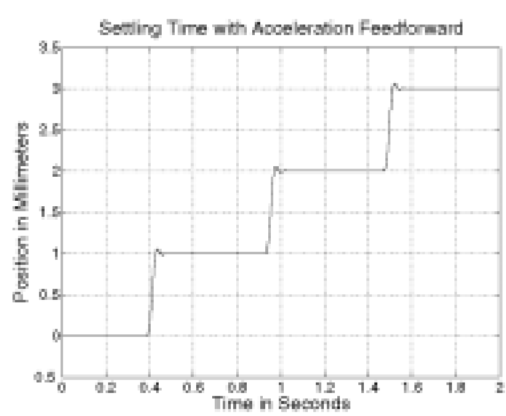


Figure 13b

Move and Settle Time (Cont.d)

While all servo positioning systems share the basic properties previously described, specific variants require additional scrutiny. For the sake of simplicity, the previous section was tailored to linear motor servo systems. In leadscrew driven systems, the payload mass is seen as a reflected inertia, scaled down by the square of the lead. The leadscrew itself often dominates the system inertia, and the motor inertia and coupling stiffness must also be considered. Rotary encoded servo stages share some attributes with the stepping motor driven stages discussed previously. There is still a strong component of the settling behavior which is simply the axial spring-mass resonance, and the ringing due to this is largely ignored by the servo loop. It consequently damps out by purely mechanical means.

A linearly encoded, leadscrew based servo stage shares some attributes with both its rotary counterpart, and linear motor systems. Leadscrew inaccuracy is no longer an issue, but the axial resonance is now directly part of the position servo loop. While it is typically higher in frequency than the servo bandwidth, it contributes phase shift at the lower frequencies that can affect loop stability. This has the effect of reducing the achievable servo bandwidth, so it is valuable (as always) to explore design changes that will raise the frequency of the axial resonance. In some cases, a “dual loop” architecture may prove useful; in this case, the derivative term is provided by a tightly coupled, motor mounted rotary encoder, with the linear encoder used only for position feedback. Another useful stratagem is to employ notch filters, implemented in either the analog signal path or in digital domain, to lower the gain of the servo loop at the axial resonance and hence allow the servo bandwidth to be increased.

Servo systems based on linear motors have some similarities with linearly encoded leadscrew-based systems, with the distinct advantage that there is no leadscrew. The elimination of this high-Q mechanical resonance, with its unwelcome phase shift, allows the servo bandwidth to be increased significantly. The limit to the achievable servo bandwidth in linear motor systems arises from phase shift due to the inevitable presence of structural resonances, as well as the zero order hold resulting from the discretely sampled digital loop filter. When analyzing position servo systems, it is customary to close a weak position loop, and inject a swept sine wave to detect the various system resonances. The design can then be re-evaluated, with an eye towards identifying and improving any existing resonances, either by increasing their frequency or lowering their “Q”. Multiple notch filters, of varying attenuation and frequency response, can also be added to allow the bandwidth to be pushed as far as is practical. The simple expedient of acceleration feed-forward can be added to reduce the inertial following error, with a commensurate reduction in the settling time. This term does not affect servo stability, as it is outside of the position loop. Additional feed-forward methods based on convolutional algorithms can further reduce the time needed to settle. High resolution systems often require vibration isolation to minimize the effect of external vibration. Conventional isolation systems employ sluggish pneumatic actuators; when the center of gravity of the stage moves, they introduce a tilt into the stage mounting surface. While leadscrew based systems largely ignore this effect, linear motor stages see a force equal to the moving mass times the sine of the angle. This force acts to induce a following error, and degrade the settling time. Active isolation systems can be effective in addressing this issue.

While some of the graphs in this section can serve as rough guidelines, the factors that contribute to move and settle times vary widely with a large number of factors. These factors are driven by both the application requirements, as well as the specifics of the design of the positioning stage, servo loop filter, feed-forward strategy, and external equipment. We have developed relatively sophisticated mathematical models, which allow us to optimize a given system for a specific application and budget.

Constant Velocity Systems

While the term "velocity ripple" is commonly used, and often appears in application requirements, it is somewhat misleading, and we prefer other measures to characterize stage performance. In general, the need to move at constant velocity arises because a customer action occurs at a fixed frequency, and their goal is to have the result occur at uniform position intervals on some moving product. Clearly, if we can move at a perfectly constant velocity, this goal will be achieved. It should be noted that modern motion control electronics generate velocity command profiles using digital circuitry locked to precise crystal oscillators. As a result, the commanded motion profiles generated by these controllers are essentially perfect; virtually all of the velocity ripple measured in moving stages is due to the stage failing (for a wide variety of reasons) to follow the controller's commanded motion profile accurately. Over long periods of time, the average velocity error will be that of the crystal oscillator, which is typically accurate to better than 0.01%. More typically, the challenge is to provide velocity uniformity on time scales ranging from a few milliseconds to a minute, and in this case the crystal accuracy is only one of a number of sources of velocity error.

In a fair fraction of applications, the "fixed frequency" customer process is fixed simply because it is triggered by a digital strobe signal. This is typically provided by a crystal oscillator, which is an obvious and simple choice, provided that a positioning system vendor can be found who can provide a "perfect" constant velocity stage. Since perfection is potentially costly to attain, we frequently recommend a fairly low cost alternative: add a linear encoder to the positioning stage, and derive a digital strobe from the actual position of the product, as opposed to a crystal-based clock. In this case, variations in the speed of the positioning stage are irrelevant, and the spatial position of the customer action on the product will be extremely uniform. With the use of digital PLDs and/or PLLs, fairly arbitrary ratios between encoder resolution and event spacing can be accommodated. A typical application for which this works very well would be a laser marking system, for which laser pulses must occur at specific intervals on a part. In certain optical scanning applications, this approach is considered unacceptable, since features will be overexposed if the stage were to slow down at any point. Some of these applications can also use this technique, however, if their light source is capable of high-speed modulation. In this case, the illumination can be turned on and off so as to allow a fixed integration period for the detector, irrespective of changes in stage velocity. Since the strobes to the detector A:D system originate from the linear encoder, position uniformity of the samples is also assured. All of this is to say that in many cases, applications do not need constant velocity, and a better system architecture may allow the use of a lower cost stage with easily measurable "velocity ripple". We're not trying to duck the technical challenge here, but merely to save the customer money.

There is a subset of the complete range of constant velocity applications for which the above techniques will not suffice. These typically involve a scanning measurement system in which a continuous illumination source cannot be modulated, or a component with inertia is in the customer process, examples of which are a high speed rotating monogon and polygon in optical scanning applications. In this case, there is no ability to slow down or speed up the polygon to match variations in the speed of the positioning stage. In these applications, it is in fact important that our stage move at a constant velocity. The concept of "velocity ripple", however, is not the best way to characterize stage performance. To better visualize this, consider a typical application, in which a customer process is writing optical data onto (or reading optical data from) a moving medium. In this case, the requirement is that the stage meet a "20 mm/sec velocity, ±1%" specification.

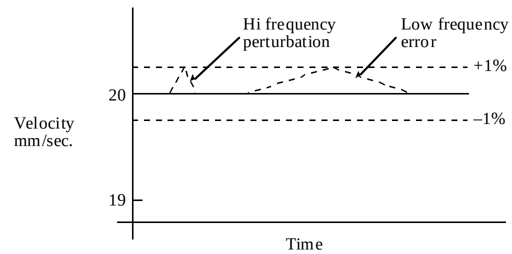


Figure 14a – Velocity vs. Time

If we were to plot velocity vs. time, we would obtain a straight line (Figure 14a), with the dotted lines representing the ±1% tolerance. The resulting plot of position vs. time is a simple straight line whose slope is the velocity, but the effect of two "legal" 1% variations are quite different (Figure 14b). The high frequency perturbation produces a small change in the intended position trajectory, while the position error of the low frequency perturbance is quite large. Since most applications of this sort cannot directly sense any of the derivatives of position, **it is the position error that matters.**

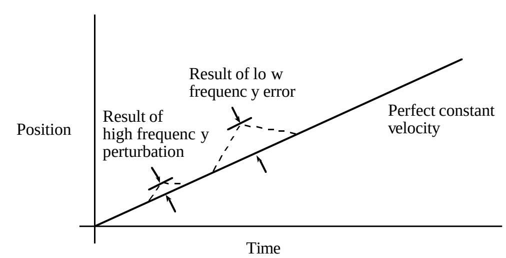


Figure 14b – Position vs. Time

Accordingly, we prefer to measure position directly, using laser interferometers with resolutions of up to 1.25 nanometers and sampling rates of up to 100 kHz. When plotted, the result is a nearly perfect straight line whose slope equals the

Constant Velocity Systems (Cont.d)

velocity. To better see the deviations from the intended position trajectory, we then subtract a best-fit straight line to the data and greatly increase the position sensitivity, providing a graph of Position Error versus Time. (Figure 14c).

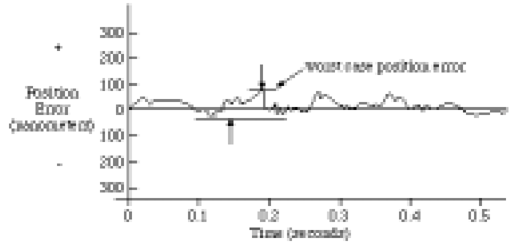


Figure 14c – Position Error vs. Time

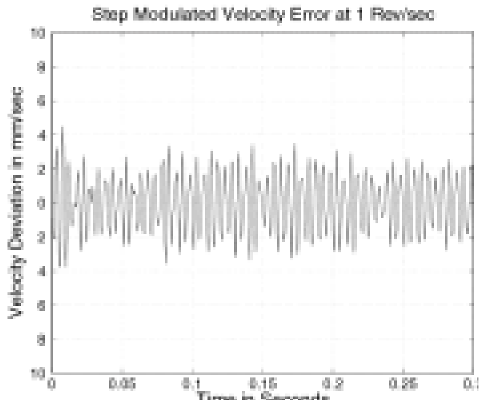


Figure 15a

This is the most physically significant means of representing the data, and it reveals the deviation from the intended position trajectory at any given time. For further analysis, we perform FFTs on both the position and velocity vs. time data, and analyze the spectral content of the data. This can identify structural resonances to modify via design changes, as well as suggest the appropriate frequencies at which to implement notch filters, to further suppress positional error. In some cases, we convolve the resulting spectral content against a customer-provided weighting function, and then transform back into position domain to see the position error through the filter of the customers' sensitivities. For those applications where stage velocity is actually significant, our Zygo laser interferometer performs a "time-stamping" of position data, such that the time of each position measurement is known to an accuracy of 16 nanoseconds. Together with its resolution of 1.25 nanometers and sample rate of up to 100 kHz, this is a powerful tool for analyzing stage dynamics.

Stage design for constant velocity systems (again, we prefer to think in terms of "deviation from intended position") must be based upon an understanding of root causes. These causes are different depending upon the basic conformation of the stage; whether the stage is leadscrew or linear motor driven, uses stepping or servo motors, has recirculating, non-recirculating, or air bearing ways, etc. The paragraphs below describe some of the root causes for various stage conformations.

Stepping motors' inherently discrete mode of operation causes velocity ripple at the motor step rate, especially at low step rates, although microstepping can greatly reduce low speed velocity ripple. In Figure 15a, the primary source of velocity error is caused by individual full steps.

Above the motor's primary resonance (~1 revolution/second) the ripple amplitude driven by step rate falls off rapidly, until, at intermediate speeds (~5 revolutions/second) the effects of individual steps disappear, to be replaced with a component dominated by the manufacturing tolerances of the stepping motor's 50 magnetic poles (Figure 15b). Microstepping cannot reduce this effect. At still higher speeds (~25 revolutions/second), as pole modulation exceeds 1 KHz, it is swamped by the system's mechanical inertia, and no longer produces a signature. This leaves a residuum of velocity ripple due to leadscrew periodic error and torque variation, as well as the fine structure of the bearings and ways.

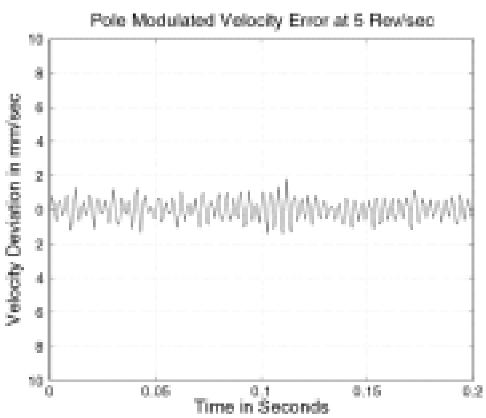


Figure 15b

Servo systems using leadscrews with rotary encoders for position feedback exhibit measurable levels of velocity ripple, synchronous with leadscrew rotation. Some of this error may be attributed to leadscrew pitch variation. In addition, since servo systems typically have lower torsional stiffnesses than steppers do, they exhibit variable following error (and, hence, velocity ripple) due to leadscrew torque variation. The use of linear encoders avoids errors due to leadscrew pitch variation, but the decoupling of the motor and encoder due to compliance in the leadscrew, nut, stage, and encoder read head introduces

Constant Velocity Systems

phase shift into the servo loop, necessitating a lower servo bandwidth. Lower servo bandwidth decreases the torsional stiffness of the servo system, and may increase errors due to leadscrew torque variations. In both Figures 16a and 16b, the larger spikes are due to the use of Hall Sensors for digital commutation. Sinusoidal commutation (described below) can eliminate this effect.

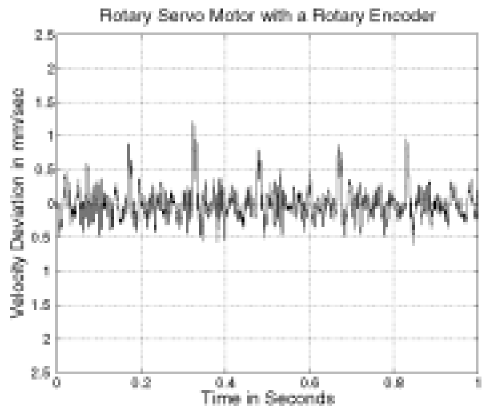


Figure 16a

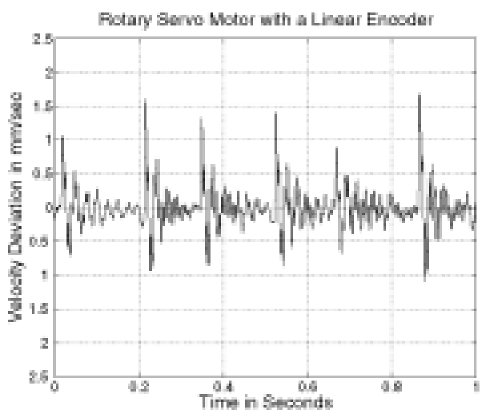


Figure 16b

Stages using recirculating way bearings exhibit noticeable force variation due to ball exit/entrance, which will cause velocity "spikes" at random intervals as the stage moves. This effect is particularly noticeable in linear motor driven stages, which typically have much lower moving inertia than leadscrew driven stages (the moving inertia of most leadscrew driven stages is dominated by the rotational inertia of the screw, not by moving mass of the stage). For this reason, non-recirculating bearings, or, better still, air bearing ways are preferred in staging designed for extremely low velocity ripple.

Extremely low levels of velocity ripple can be achieved using sinusoidally commutated linear motors, driving low friction rolling element or air bearing ways. Here, the residual error sources are linear motor magnetic field variations relative to an ideal sinusoidal field, variations in way friction, encoder interpolation errors, forces due to moving cables, amplifier current loop zero crossing errors, digital quantization effects in the control electronics, etc. A graph of velocity versus time for a sinusoidally commutated, linear motor/air bearing system moving at 4 millimeters/second is shown in Figure 17a. As previously mentioned, however, it is not velocity error per se that is important. Of greater interest is the resulting positional tracking error, shown in Figure 17b, with an enlarged view of a section in Figure 17c. In this case, the system is following its intended trajectory with an error of only ±20 nanometers (~50 atoms)! After the stage design is optimized, custom modifications to the servo loop filter permit errors to be further reduced. We have developed extensive experience in the design of constant velocity systems, and can configure a system to specifically address the technical and budgetary needs of your application.

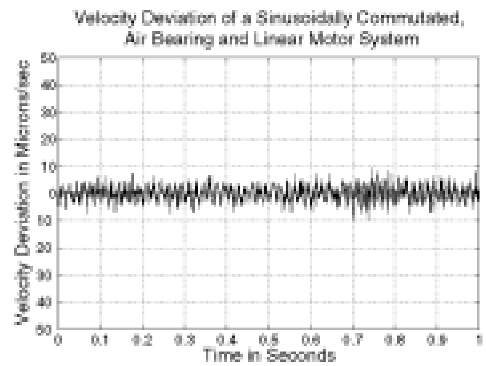


Figure 17a

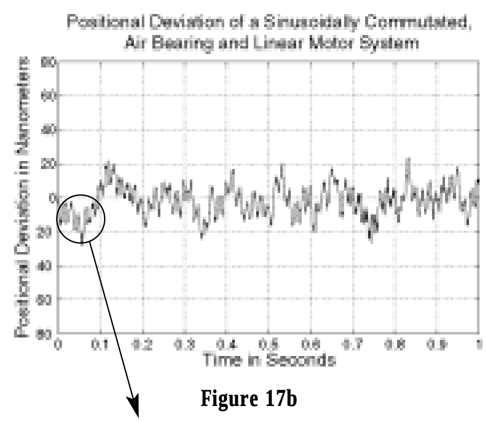


Figure 17b

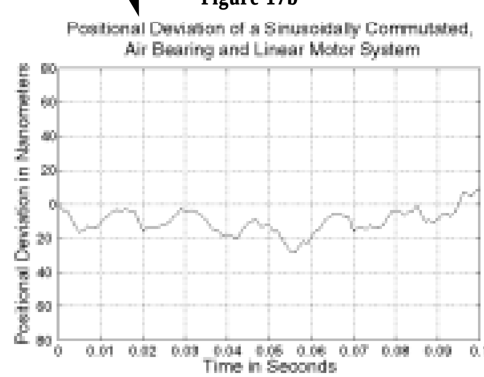


Figure 17c

Interpolated Motion

For many positioning system applications, simple point-to-point moves or constant velocity motion are sufficient. In other applications, more sophisticated movement is required. This is especially the case when the path an object takes through space is important. For example, when a simple point-to-point system moves two axes at any angle other than 45°, the path is as shown (Figure 18a), with one axis finishing before the other. More sophisticated motion controllers, which offer linear interpolation, can move at an arbitrary angle, and yet have the point-to-point path be a straight line (Figure 18b). More general path requirements can include circular interpolation, continuous path motion along arbitrary combinations of linear and arc moves (Figure 18c), and smooth, cubic spline motion through a series of points (Figure 18d). One common application requires that a third theta axis be maintained tangent to the path, as the X and Y axes follow an arbitrary curved path in the X-Y plane. Another computational problem we encounter consists of three linear axes with bearings at one end, pushing a planar payload in X, Y, and Theta axes. In this case, retaining a fixed (but virtual) "axis of rotation" as the system moves in the X and Y axes requires extensive trigonometric transforms. A similar case involves three linear Z axis stages with gimbals supporting a single payload, with the need to perform arbitrary Z axis motion, together with tip and tilt motions. Another common problem is to take a Linear - Theta stage and use it to perform X-Y motion. Higher order interpolations, of more limited use, include helical, spherical, and elliptical interpolation.

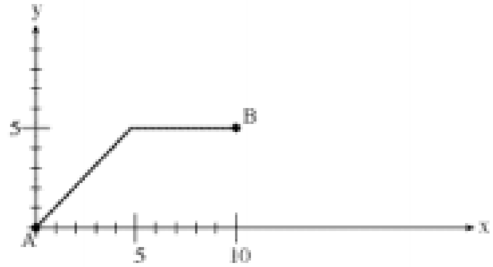


Figure 18a – Point-to-Point

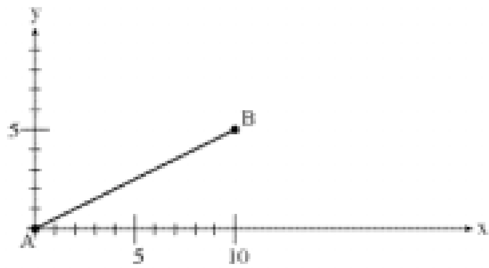


Figure 18b – Linear Interpolation

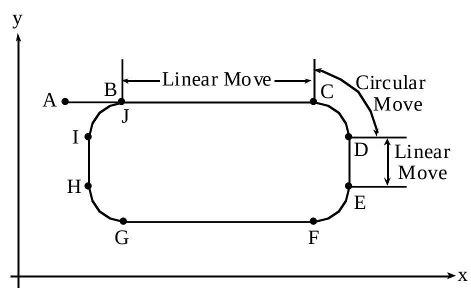


Figure 18c – Circular Interpolation

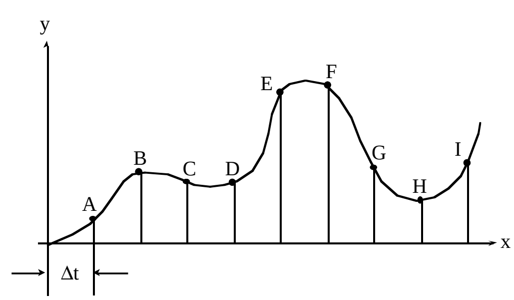


Figure 18d – Cubic Spline Interpolation

Stacking Order and Eucentric Motion

The term "stacking order" refers to the sequence in which stages are mounted to each other, starting at mechanical "ground", and continuing on up to the user payload. When linear axes are mounted to each other, the stacking order is largely immaterial, but this situation changes markedly when linear and rotary tables are intermixed. As a simple example, consider a combination of an X-Y stage and a rotary table. If an optical system is to view a part mounted on the positioner, the simplest choice would be to place the rotary table below the X-Y stage, with its rotary axis coincident with the optical axis of the viewing optics. Any point on the X-Y stage, when brought into the field of view, will now rotate around the center of that field. The drawbacks to this choice are that the X-Y stage cables will wind up, and the stiffness of the rotary table bearing may be inadequate as the X-Y stage extends to full travel. Inverting the stacking order, by placing the rotary table on top of the X-Y stage, solves the previous two problems, but the image whirls out of the field as soon as the rotary table is activated. Appropriate "eucentric" software, however, can create a virtual center of rotation at the optical axis. If point-to-point eucentric control is used, the image will leave the field, but return to the field center at the end of the move to the desired angle. If interpolated eucentric control is used, the image rotation will be indistinguishable from the case in which the rotary table was mounted below the X-Y stage. In the wafer stage of Figure 19, the tilt axis is designed such that its rotary axis is coincident with the wafer plane, and it is mounted below the X-Y stage. In an alternate design, the X-Y stage could be mounted below a much smaller tilt axis, although more complex software and additional Z axis travel would also be required. Our Design Engineers have extensive experience with positioning systems of the above types, and can design a solution for most application requirements.

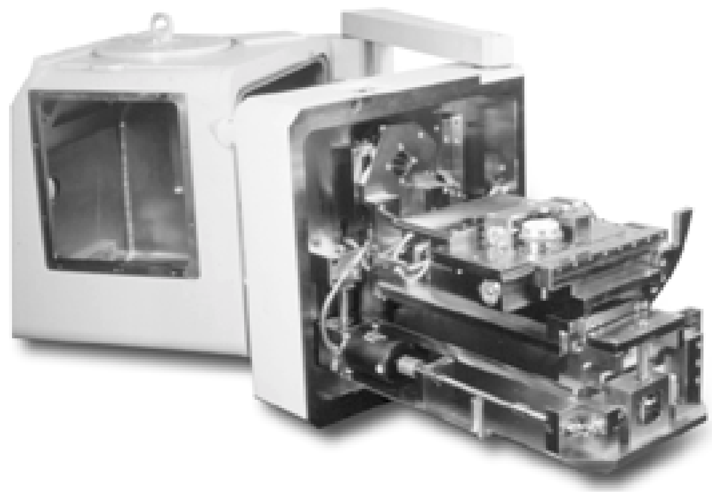


Figure 19

Torque and Force Requirements

Torque is required to both overcome friction in the nut, and accelerate the motor and load to the required top speed. Similarly, in linear motor applications, force is required to overcome friction in the ways, cable bending forces, and to accelerate the moving mass of the stage and the user payload. In general, calculations of this sort make the most sense (and are a lot easier) when the MKS system of units is employed. The MKS unit of torque is the Newton-meter (N-m), and the corresponding unit of rotational inertia is kilogram-meters squared (kg-m^2).

Rotary Motor Example

The load which must be accelerated consists of the manual positioning knob, the motor rotor, the flexible shaft coupling, the leadscrew, the moving portion of the positioning table, and the user's payload. The rotational inertia (J) of the knob used on our positioning tables is 6.3×10^{-6} kg-m^2 while the helical shaft coupling has an inertia of 2.2×10^{-6} kg-m^2. The rotor inertia varies with motor frame size and length; values for all of our standard stepping motors are included on page 195. Specific values of the rotor inertia for six standard motors are as follows:

Rotor Inertias

Stepping Motors:	
17 frame, 0.16 N-m (23 oz-in) holding torque:	3×10^{-6} kg-m^2
23 frame, 0.38 N-m (50 oz-in) holding torque:	11×10^{-6} kg-m^2
23 frame, 0.71 N-m (100 oz-in) holding torque:	23×10^{-6} kg-m^2
Servo Motors:	
40 mm square brushless:	5.65×10^{-6} kg-m^2
50 mm square brushless:	38.1×10^{-6} kg-m^2
Brush servomotor:	26.1×10^{-6} kg-m^2

Depending on size, our tables can be provided with either ~12 mm or ~18 mm outer diameter (OD) leadscrews, whose rotational inertias are 2.7×10^{-6} and 1.2×10^{-5} kg-m^2 per 100 mm of travel, respectively. The process of determining the required torque for a given application begins by adding the rotary inertia of the knob, motor rotor, coupling, and leadscrew. The user payload mass and the moving mass of the positioning table must then be summed, and converted into an equivalent rotary inertia, via the following formula:

Rotational Inertia of Table & Load

$$J = \frac{m \times \left(\frac{lead}{2\pi}\right)^2}{e}$$

J: rotational inertia, in kg-m^2
m: total moving mass, in kg
lead: screw lead, in m
e: screw efficiency

The efficiency of the leadscrew is typically 0.6 for our leadscrews with anti-backlash nuts, and 0.9 for ballscrews. The moving masses of single-axis stages, as well as the upper and lower axis moving masses for X-Y tables, are listed with the specifications for each table. Finally, the total rotational inertia is converted to a torque which, when summed with the friction torque, equals the total required torque. The frictional torque of our positioning tables is held between 0.03 and 0.06 N-m for 12 mm OD leadscrews, and between 0.06 and 0.09 N-m for 18 mm OD screws. While the nut can be adjusted to lower torque values, this can reduce its otherwise excellent (1-2 micron) repeatability, and decrease the axial stiffness. Due to the presence of a lubricant film, the friction between leadscrew and nut increases with rpm, as well as at lower temperatures.

As an example, consider our XYL-1515-SM, a 300 mm x 300 mm (12" x 12") travel X-Y table. It is supplied with a standard motor, and has an upper axis moving mass of 8.6 kg. We will assume a screw lead of 5 mm (0.005 m), a leadscrew diameter of 18 mm, and a user payload of 23 kg. This load must be accelerated at 2 meters per second squared. To begin, we sum the rotational inertias of the respective components:

Sum of Rotational Inertias

J Knob = 6.3×10^{-6} kg-m^2
J Coupling = 2.2×10^{-6} kg-m^2
J Motor = 23×10^{-6} kg-m^2
J Leadscrew = 12×10^{-6} kg-$m^2 \times$ (300 mm/100mm) = 36×10^{-6} kg-m^2

$$J\ Table + Load = \frac{(8.6\ kg + 23\ kg) \times (0.005/2\pi)^2}{0.6} = 33 \times 10^{-6}\ kg\text{-}m^2$$

Total Rotational Inertia = 1.0×10^{-4} kg-m^2

Note that the rotational inertia of the leadscrew is greater than that of the payload.

The formula to convert rotational inertia to torque is as follows:

Torque

$$T = J \times A / Lead$$

T: torque, in N-m
J: rotational inertia, in kg-m^2
A: acceleration, in m/s^2
Lead: screw lead, in m/rad

In this case,

$$T = \frac{1.0 \times 10^{-4}\ kg\text{-}m^2 \times 2\ m/s^2}{0.005m/2\pi} = 0.25\ N\text{-}m$$

(Remember, a Newton is a kg-m/s^2). Summed with a frictional torque of 0.09 N-m, this results in a total torque requirement of 0.34 N-m. This is less than the motor's holding torque of 0.70 N-m. However, motor torque falls off with speed; the intersection of the torque requirement with the motor's speed-torque curve determines the maximum speed to which this

Torque and Force Requirements (Cont.d)

load could be accelerated without stalling (about 22 revolutions/second in the above case, using a 310M). The increase in frictional torque at 22 revs/second due to lubricant viscous drag will reduce this achievable top speed, to perhaps 12 revs/second. Adopting a safety margin of 20%, operation at speeds of up to 10 revolutions per second would be acceptable. Lowering the acceleration will reduce the torque requirement, allowing higher speeds to be obtained, but at an increase in overall move duration. The speed-torque curve for a given motor is a function of the drive design and operating voltage.

The foregoing rotary motor example provides specific numerical relationships between the inertias of various table components, and the formulas relating inertia to required torque. The degree of exactness may prove misleading; a variety of small, hard to quantify effects, including lubricant viscosity, nut pitch vs. efficiency, etc., conspire to defeat a purely quantitative approach to load and motor sizing. In particular, opinions vary as to what constitutes an acceptable safety margin for step motor systems, which may range from 10 to 30 percent. Whenever possible, actual simulation of the application is recommended before committing to a set of performance criteria. Vertically oriented applications, for example, alter nut efficiency in a manner difficult to predict in advance.

We regularly “get the lead out”, setting up dummy loads and duplicating a proposed configuration as closely as possible. Consultation with our Applications Engineers is recommended when sizing a motor to achieve specific results.

Linear Motor Example

These systems are considerably more straightforward than rotary motor based stages. Newton’s second law is just about all you need: **F = m x a**. The moving mass of the stage and the mass of the customer payload are added to obtain the total moving mass (in kg), which is multiplied by the desired acceleration, in meters per second squared. One “G” is 9.8 m/s squared. A moving mass of 10 kilograms and an acceleration of 5 m/s^2 will require a force of 50 Newtons. To this must be added the frictional force of the ways, and any other forces, such as cable loop bending forces. If these total 10 Newtons (for a total of 60 Newtons), and the force constant of the linear motor is 15 Newtons per Amp, then the peak coil current would be 4.0 Amperes. Since the force constant and back-emf constant are the same in MKS units, the stage back-emf will be 15 volts per meter/sec. At a top speed of 0.5 meter per second, the back-emf will therefore total 7.5 volts. If our coil resistance is 4.0 ohms, we will require 16 volts to drive 4 amps through this coil (Ohms law, V = I x R), so the total supply voltage required will be 16 + 7.5 + 3 volts (for FET, cable, and connector losses), for a total of 26.5 volts.

Motion Calculations

Most numeric questions about positioning applications boil down to one of two types: "How long does it take to get there?" and "How fast am I going (at top speed; at time t; point x; etc.)?" The related question of whether the motor has sufficient torque or force to perform the required profile is covered on page 189 (Torque and Force Requirements).

There are four physical quantities associated with motion calculations: position (x); its first derivative, velocity (v); its second derivative, acceleration (a), and of course, time (t). These quantities are related by several familiar equations, which can in turn be expanded by substitution to cover any combination. The full set of possible combinations follows.

Uniform Motion

No.:	To Find:	As a Function of:	Use:
1	x	v and t	$x = vt$
2	v	x and t	$v = x/t$
3	t	x and v	$t = x/v$

Accelerated Motion

No.:	To Find:	As a Function of:	Use:
4	x	a and t	$x = at^2/2$
5	x	v and a	$x = v^2/2a$
6	v	a and t	$v = at$
7	v	x and a	$v = \sqrt{2ax}$
8	a	v and t	$a = v/t$
9	a	x and t	$a = 2x/t^2$
10	a	v and x	$a = v^2/2x$
11	t	v and a	$t = v/a$
12	t	x and a	$t = \sqrt{2x/a}$

As an example, consider the simple move profile of Figure 20. A positioning table makes a move of 100 mm (0.1 meter), accelerating at 0.2 "G" to a top speed of 400 mm/sec (0.4 m/sec), and then decelerating at 0.5 G. Since one "G" is 9.8 m/s^2, the acceleration occurs at 1.96 m/s^2, and the deceleration occurs at 4.9 m/s^2. Using formula #11 ($t=v/a$), the accel time is found to be 0.204 second; using formula #5, the accel distance is found to be 40.8 mm. Similarly, the decel time and distance are found to be 0.082 second and 16.3 mm, respectively. The constant velocity distance is found by subtracting the sum of the accel and decel distances from the overall move size of 100 mm, leading to a 42.9 mm length of the move at constant velocity. Using formula #3, this phase has a duration of 0.107 second, for an overall move time of 0.393 second. If the move distance was less than 57.1 mm, and the acceleration and deceleration values were unchanged, then the stage would not be capable of reaching the 400 mm/sec top speed, and the velocity profile would be triangular, rather than the trapezoidal shape shown below.

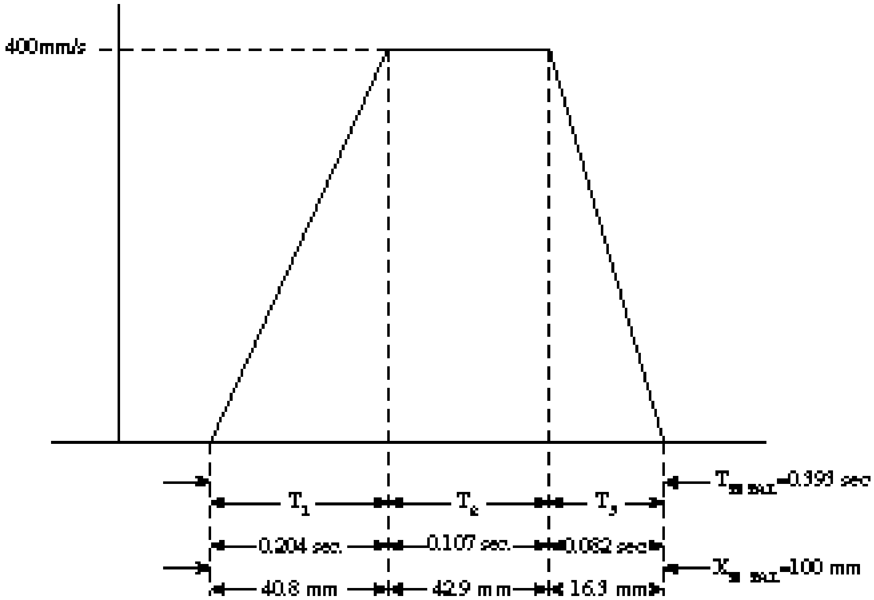


Figure 20 – Speed Profile

Retainer Creep

Linear positioning systems which utilize rolling elements can be divided into two classes: recirculating and non-recirculating. To illustrate the distinction, our TM Series is a non-recirculating design (see Figure 21) while our TMS Series is a recirculating design (see Figure 22). The potential exists in any non-recirculating design, whether ball or crossed roller, for the retainer and its ball/roller complement to undergo a slow, long term creep from its original centered position. A variety of causes exist, including operation in a vertical orientation, minor variations in preload along the way, high acceleration moves, etc.

The effect manifests itself when, after a series of short moves, a long move or end of travel registration is performed. Partway along the move, the retainer encounters the end stop, and the rolling friction must now be replaced with sliding friction to reset the retainer to its center position. If the preload is too high, the motor may be incapable of developing the required torque, and the move will not be completed.

Retainer creep is rarely addressed and often mistaken for insufficient motor torque, misalignment, etc. While it can be prevented by introducing a variable preload along the way, this technique adversely impacts flatness and straightness. Other schemes based on pulleys and wires have been proposed, but may impact long term reliability. We have developed sensitive techniques to measure and set the preload forces at a specific design level, which allow the motor and leadscrew to overcome any retainer creep should it arise. In final testing, we intentionally offset all retainers and verify that motor low speed torque is sufficient to reset the retainer(s). Under these circumstances, the system is self-compensating, and retainer creep will be of no consequence. In rare cases with low torque motors and high lead screws, we will advise customers of the effect, and work to modify the design to eliminate its adverse impact. In linear motor systems, where the motors' continuous force rating may be insufficient to reset a retainer, we can provide crossed roller ways with geared retainers. This eliminates the retainer creep issue for this class of systems.

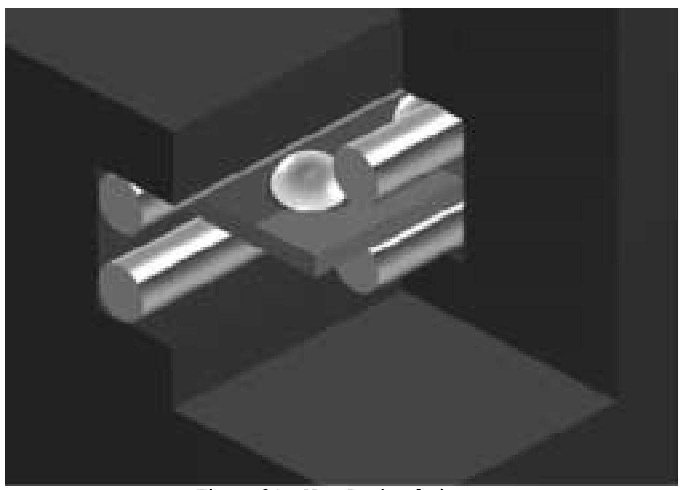


Figure 21 – Non-Recirculating

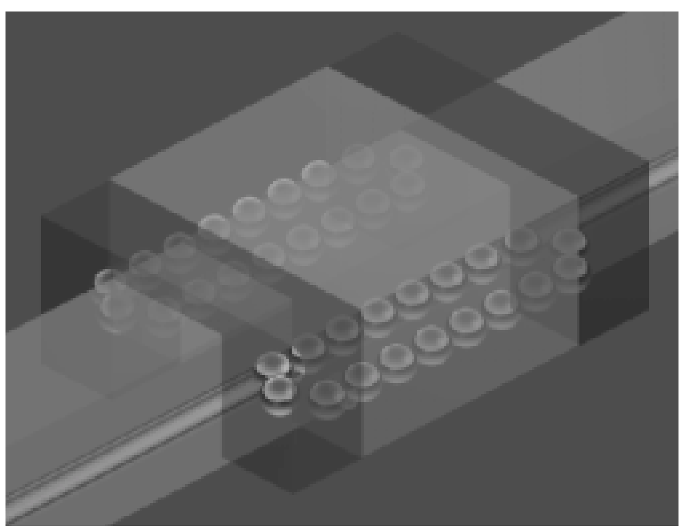


Figure 22 – Recirculating

Leadscrews and Ballscrews

Despite the recent trend toward the use of linear motors, leadscrews remain the most popular and cost-effective choice in many applications. A careful consideration of the various types of leadscrews, together with their strengths and weaknesses, can lead to optimal choices when configuring a system. Our stages are offered with three basic types of leadscrews; in each case, axial movement is constrained by a suitably chosen duplex angular contact bearing at one end of the leadscrew.

Our standard leadscrew provides very high repeatability, with a useful balance between compactness, stiffness, and speed. It is offered in two diameters (12 and 18 mm), two thread accuracy classes, and a variety of Imperial and metric leads. It achieves its high repeatability via an anti-backlash mechanism (Figure 23a), in which the nut body is cut into four flexural segments, which are radially spring loaded via a circumferential spring. This design achieves a unidirectional repeatability of less than one micron; a uniform, well damped torque of 30 to 120 milli-Newton-meters (depending on lead and diameter); and an axial stiffness of ~$2x10^6$ Newtons per meter. The nut is quite compact, with an overall length of less than 25 mm, but its lower stiffness may make it less appropriate for closed loop control with linear encoders. Our standard leadscrew and nut are compact and cost-effective, provide very high repeatability, and are best used with light to moderate loads and duty cycles, at speeds of up to 900 rpm (15 rps).

Figure 23a – The NEAT Friction Nut

For applications that require higher accuracy, we offer a range of leadscrews manufactured by Universal Threadgrinding. These also use an anti-backlash nut, which is longer and stiffer than our standard nut. Due to the metal-to-metal contact inherent in this design, periodic lubrication is required, and the speed range is limited to 1200 rpm and below. The principal attribute of this leadscrew and nut is very high accuracy, both over full travel and over each revolution (periodic error). The torque variations are also quite low, permitting uniform scanning speeds to be achieved. Stiffness is ~$5x10^6$ Newtons per meter, and torque is ~60 milli-Newton-meters.

In applications that require either high speeds, a high duty cycle, or both, a ballscrew is the best choice. Ballscrews use recirculating balls to couple the nut to the screw, and eliminate backlash through the elastic preloading of oversize balls (Figure 23b). In addition to their suitability for high speeds and duty cycles, ballscrews are much stiffer than anti-backlash leadscrews and nuts, and can achieve stiffnesses of $5x10^7$ Newtons per meter. To estimate the natural frequency for a given axis, the stiffnesses of the nut, duplex bearing, nut mount, bearing mounts, and the column stiffness of the leadscrew itself must be taken into account. Simply maximizing the stiffness may not be optimal, as there may be a trade-off between stiffness and thermal expansion due to frictional heating. Due to the entry and exit of balls into the preloaded region, torque uniformity of ballscrews is not as smooth as with a conventional anti-backlash design. Most ballscrew torques are in the 60 to 150 milli-Newton-meter range. We offer ballscrews in three leads: 2mm, for high resolution; 5mm, for most applications; and 10 mm, for high-speed applications.

Figure 23b – Ball Nut

Rotary Stepping Motors

Every one of our positioning tables is supplied complete with a motor, together with a rear shaft mounted knob for fine manual positioning. Unless otherwise specified, this will be a stepping motor, and the motor provided with any particular stage model is shown in the chart that follows. Certain applications may require that tables be supplied less motors, with brush or brushless DC servomotors, or with an optional stepping motor. Any exceptions to our standard motor should be clearly specified when an order is placed. The following describes our standard and optional stepping motors.

The primary characteristics of a stepping motor which determine its suitability for any given application are its frame size, the number of steps per revolution, and its holding torque. The motor size is normally described by its corresponding NEMA (National Electrical Manufacturers Association) frame size, together with the motor length. Our stepping motors are available in two frame sizes: 17 and 23. To a reasonable approximation, the cross section of the motor, in inches, is equal to the frame size divided by 10 (for example, 23 frame motors are about 2.3 inches square; the actual dimension is 2.24").

The number of steps per revolution for a motor determines the resolution of the positioning table which it drives. Our standard stepping motors provide 200 full steps per revolution, while optional motors provide 400 full steps per revolution. All of our stepping motor drives implement microstepping, with a standard division ratio of 10, which results in 2000 or 4000 microsteps per revolution from 200 and 400 step per revolution motors, respectively. High resolution divide-by-50 drive modules are optionally available, providing 10,000 or 20,000 microsteps per revolution.

The holding torque is the maximum torque the motor can develop before swinging to a new pole position. Note that an energized motor has no torque when it is in position; it develops increasing torque as the motor shaft is displaced from its nominal position. A stepping motor's capability of developing full torque over a small angular displacement (1.8 degrees for a 200 step motor) would require high gain (and currents) in a servo motor of equal overall size. This advantage of high stiffness per unit volume, achievable at low motor currents, is a key stepping motor feature. Unlike servo motors, steppers are inherently stable, although they can stall if their torque capability is exceeded.

The holding torque of a stepping motor increases with increasing motor volume. For a given length, increasing the cross section (frame size) results in more holding torque, as does increasing the length within a given frame size. Motor costs also scale as the motor volume is increased. In many applications, the requirement is not holding torque, but available torque at a specific rotation rate or step frequency. In general, a higher holding torque translates to greater available torque at any specific rotation rate, but this is complicated by additional factors such as the motor's inductance and the drive voltage.

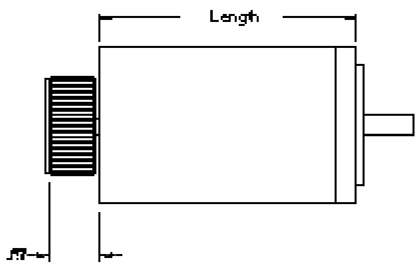
High speed performance is enhanced by the combination of a low motor inductance and a high drive voltage. Low inductance results in higher motor currents, however, which may tax the power supply or driver output capability. Too high a drive voltage, in turn, may raise safety considerations, can result in excessive drive and motor heating, and may increase radiated electrical noise.

All of our standard stepping motors are supplied in a six lead configuration (two center tapped coils, bifiliar wound). The manner in which leads are wired to a drive can be critical in determining overall system performance. The chart on the next page indicates which motors have their electrical leads brought out the face of the motor, parallel to the motor shaft, where a DE-9 connector is attached 4 inches from the motor face. This arrangement is specifically designed for use with our rotary motor mount; it will often prove useful in other applications by providing a locking connector at the motor mounting plate. All other motors listed have leads exiting the motor barrel on the outer diameter and may be wired to a connector if specified.

Rotary Stepping Motors (Cont.d)

The rotor inertias of our stepping motors are provided to assist in calculating achievable motor (and load) acceleration. As mentioned, all of our standard stepping motors are double shafted, and include knobs for manual positioning. This knob is 1.2” in diameter x 0.62” long, and has a rotary inertia of 6.3 x 10^{-6} kg-m^2. In applications where maximum acceleration is required the knob can be removed, providing a modest increment in achievable acceleration.

The accompanying table lists our most popular stepping motors, together with their key parameters. Footnotes indicate the standard motors shipped with our positioning tables; the remaining motors are optionally available when higher torque or resolution is required. Motors are normally available from stock. In addition to incorporation into our positioning tables, these stepping motors are available for uncommitted use in positioning or motion control applications; use the provided part numbers when ordering motors separately. Speed-torque curves for our three standard motors also follow.

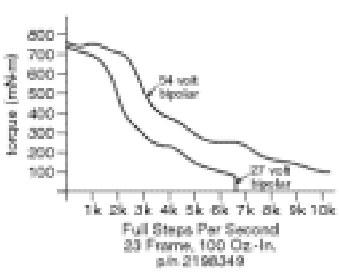


Rotary Stepping Motors

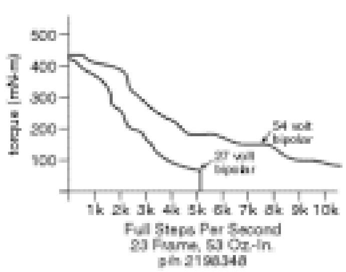
Part Number	Frame Size	Steps Per Rev.	Torque oz-in (N-m)	Voltage (volts)	Current Per Phase (amps)	Inductance (millihenries)	Rotor Inertia oz-in^2 (kg-m^2)	Length (inches)	Face-Wired Connector
2198366	17	400	23 (0.16)	4.0	1.2	3.1	0.190 (3x10^{-6})	1.85	NO
2198377[1]	17	200	36 (0.25)	6.0	0.8	6.5	0.300 (5x10^{-6})	1.54	NO
2198376[2]	17	200	44 (0.31)	4.0	1.2	2.0	0.370 (7x10^{-6})	1.85	NO
2198352	23	400	80 (0.57)	6.0	1.2	8.8	0.740 (14x10^{-6})	2.13	NO
2198364	23	400	118 (0.84)	5.4	1.5	6.5	1.090 (20x10^{-6})	2.99	NO
2198348[3]	23	200	53 (0.38)	5.0	1.0	10.0	0.620 (11x10^{-6})	2.00	YES
2198349[4]	23	200	100 (0.71)	4.7	1.6	5.7	1.280 (23x10^{-6})	3.25	YES
2198350	23	200	150 (1.06)	3.4	2.9	2.9	1.740 (32x10^{-6})	4.00	YES

[1]Standard on X-Theta
[2]Standard on Mini XYR, XYMR, OFS, RM and RMS
[3]Standard on XY, XYR, TM, TMS, LM, FM, XM, ZE, Z-Theta, RT and RTR
[4]Standard on XYL, Ballscrew XYR, OFL, HM and HMS

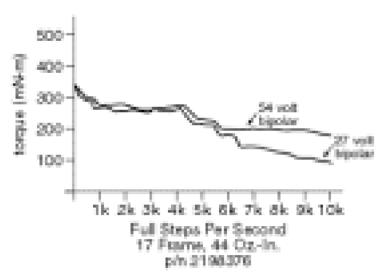
23 Frame, 100 oz.-in.



23 Frame, 53 oz.-in.



17 Frame, 44 oz.-in.

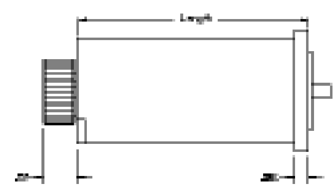


Rotary Servo Motors

Our Positioning Tables can be optionally equipped with either brush or brushless rotary DC servo motors. While brush type rotary servo motors have some of the same drawbacks as their linear counterparts (as described in the Linear Servo Motors section on the next page), their enclosed design reduces concerns over brush particulates and radiated electrical noise; moreover, they offer reasonable cost savings. Our brushless motors provide higher performance; however, in addition to using rare earth magnets, their construction places the heat producing coils in the stator, where waste heat can much more readily be dissipated. Both our brush and brushless motors include rear shaft mounted knobs, and they are wired directly to our DE-9™ motor connector. There are two brushless motors, for different size stages: the 40mm motor is best suited to RM™, RMS™, Mini-XYR™, XYMR™, X-Theta™, and OFS™ stages; while the 50mm model mates to the motor mount of the majority of our stages.

As with linear motors the basic quality factor of the motor is K_{mw}, expressed in Newton-meters per $\sqrt{\text{watt}}$. This relates the continuous torque that the motor can produce, with the waste heat which must be dissipated, in watts. Since thermal characteristics set the ultimate performance limit of the motor, it is this parameter which designers seek to maximize. Other relevant parameters are the Torque Constant, in units of Newton-meters per Amp, which relates the output torque to the motor current, and the Back-emf Constant, in volts per radian per second, which relates the number of volts that the motor produces as a generator, with its rotational speed. In SI units, these two constants are identical. In Imperial units, the equivalent units are ounce-inches per Amp, and volts per thousand RPM. To convert between these units, multiply the ounce-inches per amp by 0.74 to obtain volts per thousand RPM.

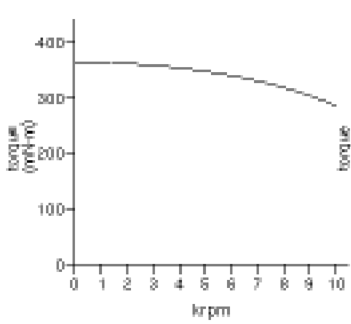
Specifications and speed-torque curves for our three motors follow; while the specified windings are carried in stock, alternate windings are also available, providing varied torque and back-emf constants. Note that it is permissible (and expected) that the motor briefly exceed its “safe” operating area during acceleration, as long as the duty cycle is such that the average current or torque is within safe limits.



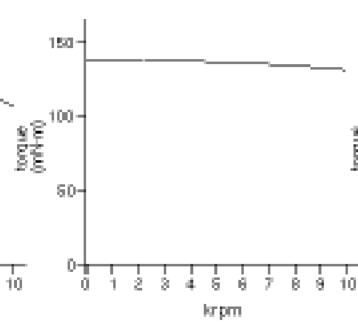
Rotary Servo Motors

Part Number	Type	Torque-Constant N-m/A	Back-EMF Constant V/(rad/s)	Resistance ohms	Inductance mH	Length inches	Cross-Section mm
2198369	50mm brushless	86.5×10^{-3}	86.5×10^{-3}	1.21	0.413	4.30	50mm square
3930144	40mm brushless	31.4×10^{-3}	31.4×10^{-3}	1.21	0.198	2.92	40mm square
2198368	brush	61.2×10^{-3}	61.2×10^{-3}	1.01	1.6	4.33	54mm diameter

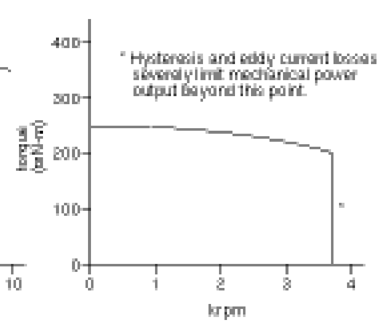
50mm brushless



40mm brushless



Brush



Linear Servo Motors

Linear servo motors have become very important components of precision positioning systems, with numerous advantages over traditional mechanical actuators (such as ballscrews). Linear servo motors (synchronous designs aside) consist of a permanent magnet assembly which establishes a magnetic flux, and a coil assembly which generates a force proportional to coil current. While linear servo motors can be implemented in both brush and brushless configurations, the multiple drawbacks of the brush type (limited brush life, contaminants, and electrical noise) strongly favor the brushless design. Another design choice relates to the presence or absence of iron in the coil assembly. The use of iron in the coil assembly produces strong attractive forces between the coil and magnet assembly, on the order of 600 to 20,000 Newtons (135 to 4,500 lbs.), as well as a periodic "cogging" force. While the focusing of the magnetic flux can increase the continuous force per unit volume by 10-30%, of far greater consequence is the increased inductance. The electrical time constant of brushless motors with iron-based coil assemblies is 5 to 20 times that of ironless designs, the effect of which is to significantly lower the achievable servo bandwidth. This parameter is the key to achieving excellent dynamic performance, and directly determines system settling time (and hence throughput). Servo bandwidth also determines the achievable level of servo stiffness; if an application does not have sufficient time to allow a sluggish integrator term to sum, the stiffness will directly affect the position error. Accordingly, we only manufacture linear brushless servo motors, with no iron in the coil assembly. The distinction as to which component moves and which is fixed is application dependent; a moving magnet track imposes a penalty in moving mass and dimensional envelope, but allows a stationary coil cable. Moving the coil, on the other hand, lowers the moving mass and is more compact, but requires that an appropriate service loop be provided for the coil's electrical cable.

A single-phase coil can only function for certain limited stroke applications. Traditionally, linear motor designs capable of arbitrary travels have adopted a three-phase coil, with an amplifier that excites the windings in a six-step sequence, based on three digital magnetic field sensors mounted in the coil assembly. If pure, constant velocity motion is required, sinusoidal commutation can be substituted for the simpler six-step sequencing. This can be accomplished either by using the linear encoder to determine coil current relationships, or by utilizing two analog magnetic field (Hall) sensors mounted in the coil assembly to proportion coil currents. The encoder-based method requires that the servo controller provide two analog (DAC) outputs per axis, which may reduce the total number of axes which can be controlled, and a "phase-finding" initialization routine must be performed upon power-up. Both the encoder-based and the analog sensor-based methods of sinusoidal commutation require specialized servo amplifiers to accommodate the new inputs.

Our motor design, among the various models presented above, is accordingly the brushless, ironless, three phase linear servo motor. This is a product which can be fully defined with a relatively small set of specifications. The simplest of these are dimensional (i.e., magnet track/coil cross-section, coil length, mounting patterns, etc.). The required length of magnet track is essentially the coil length plus the travel, with some allowance for over-travel. The weight of the magnet track and coil assembly are also of interest, with the latter needed for moving mass calculations if the coil is the moving component. The key characteristics, however, are electromechanical, and make far more sense if expressed in MKS units. These key attributes are:

Linear Servo Motors

Attribute	Symbol	Units	LM1-3-1D	LM1-3-2D	LM2-3-1D	LM2-3-2D
Fundamental Motor Constant	Kmw	Newtons/$\sqrt{\text{Watt}}$	4.32	6	5.9	8.5
Force Constant	Kma	Newtons/Amp	8.9	13	13	18.5
Back-emf Constant	Kme	Volts/meter/second	8.9	13	13	18.5
Resistance (@ 20 C)	R	Ohms	4.25	4.7	5.2	4.5
Inductance	L	Milli-Henries	1.65	2.3	2	2.8
Continuous Current	Ic	Amps	2.5	3.7	3	4.5
Peak Current	Ip	Amps	7.5	11	9	13.5
Continuous Force	Fc	Newtons	20	40	40	80
Peak Force	Fp	Newtons	60	120	120	160

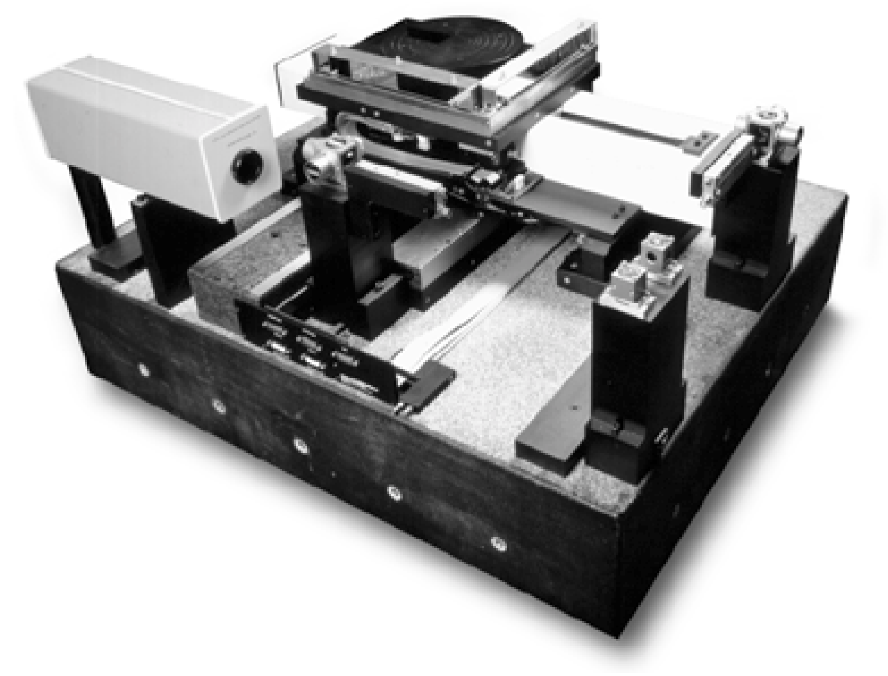
Linear Servo Motors (Cont.d)

The fundamental motor constant, K_{mw}, is normally omitted by linear motor vendors, in favor of the more obvious force constant, K_{ma}. The latter constant, however, can be arbitrarily varied by an appropriate choice of wire gauge. Since most users would like “a lot of force per amp”, manufacturers can comply by simply winding coils for a large force constant. Some manufacturers claim to have the “highest force per motor area”. This is quite meaningless, as one could easily wind a coil that would have two, five, or a hundred times as much force per amp. No one would want such a coil, however, because the effect of this choice is to exact a large penalty on the user in terms of amplifier voltage requirements. When expressed in MKS units, the force constant and back-emf constant are identical: for every Newton/Amp of force constant, the coil will generate 1 Volt per meter per second. This back-emf acts to oppose the supply voltage, and requires proportionally higher supply voltages (and amplifier voltage ratings) to achieve a given velocity. In addition, a high force constant results in a proportionally higher coil resistance, which also requires higher voltages to drive a given current into the winding.

The fundamental motor constant, $K_{mw,}$ expressed in units of Newtons per square-root watt, is more meaningful. It relates the force the motor can produce to the waste heat which must be dissipated. Since thermal characteristics set the ultimate performance limit to the motor, it is this parameter which designers seek to maximize. It is to first order independent of the wire gauge, and is simply a function of the magnetic flux level in the gap, the coil volume, and the efficiency of copper packing. It accordingly represents the “quality” of a linear motor. It can also be used to predict the optimum wire gauge and coil resistance for any specific application. In the preceding table the continuous and peak force are also largely independent of the linear motor coil wire gauge. The continuous and peak currents are widely variable through suitable choices of wire gauge; the continuous current (or force) rating is determined by the maximum allowable coil temperature rise, while the peak current (or force) rating is selected to avoid any possible de-polling of the permanent magnets.

It is important to note that the operation of a linear motor at its continuous force or current rating is equivalent to operation at its maximum temperature rating, which is materials limited to 120°C for our linear motors. At this temperature, the resistance of the coil windings is increased by 39% over its nominal value (specified for 20°C), and the fundamental motor constant Kmw is similarly reduced by 18%. This change must be taken into consideration when designing systems that will operate near the coil's maximum temperature limit. The specified continuous current and continuous force rating already take this into account, since by definition operating at these levels is synonymous with operating at maximum allowed temperature.

Interferometer Feedback Systems



Laser interferometers provide the ultimate in position feedback, combining very high resolution, non-contact sensing, high update rates, and intrinsic accuracies of 0.1 ppm. They can be used in positioning systems as either passive position readouts, or as active feedback components in a position servo loop. Unlike linear encoders, the interferometer beam path can usually be arranged to coincide with the item or point being measured, eliminating, or greatly reducing errors due to Abbé offset.

Laser interferometers can be divided into two categories: fringe counting and two-frequency systems. The former is similar in operation to a Michaelson interferometer while the latter uses two closely spaced frequencies, one of which experiences a Doppler shift from the moving reflector. Upon recombination, the two frequencies are heterodyned to generate a beat frequency within the range of counting electronics. The two frequency design, while more costly to implement, is considered the higher performance system, especially for velocity feedback. In both cases polarization selective optics are used to route one beam to and from the moving workpiece while retaining a fixed path for the reference beam.

Single-axis systems utilize a beam path as shown in Figure 24a and consist of the laser head, polarizing beamsplitter with retroreflector, the moving retroreflector, and a photodiode receiver. XY systems (Figure 24b) replace the moving retroreflector with a plane mirror and add a quarter-wave plate and an additional retroreflector to the separation optics. The quarter-wave plate circularly polarizes the workpiece beam, causing it to perform two passes, with a corresponding doubling of resolution and halving of achievable top speed. This configuration eliminates errors due to Abbé offset, yaw, and opposite axis horizontal runout, and ignores orthogonality errors in the X-Y tables (the plane mirrors, however, must be precisely square to each other). The reflectors can consist of two “stick mirrors” in adjustable mounts, or a single “L mirror” (as shown in the photo). The latter eliminates concerns over stick mirror misadjustments but carries cost penalties which grow rapidly with increasing travel.

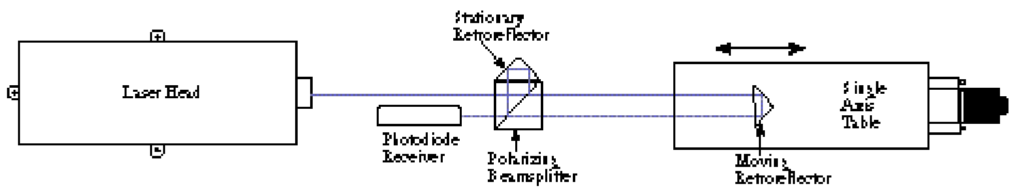


Figure 24a – Single-Axis Interferometer Beam Path

Interferometer Feedback Systems (Cont.d)

As the N.B.S. (now N.I.S.T.) pointed out in the mid-seventies, any He-Ne laser provides frequency stability equal to or better than 1 part in 10^6, 1 ppm (any greater error would inhibit the lasing process due to the narrow neon line width). Frequency stabilization systems can improve this, achieving accuracies of better than 1 part in 10^7 (0.1 ppm). The following error sources, however, conjoin to degrade this very high intrinsic accuracy:

1. Speed of light variations due to temperature, pressure, etc.
2. Pressure, temperature and humidity sensor accuracy
3. Plane mirror squareness and flatness
4. Thermal expansion of workpiece, positioning table, base plate and interferometer optics
5. Cosine error
6. Accuracy of workpiece thermal expansion coefficient
7. Differential flexure of positioning table top through its travel
8. Edelin and Jones equation accuracy
9. Accuracy of deadpath value

While the individual contributions of these effects may be small, their aggregate effect on error budgets can be significant. Attempts to wring increasing accuracy from a positioning system rapidly degrade into elaborate thermal management exercises. Our accuracy claims, while modest in light of industry practice (ads claiming “tenth micron accuracy” from open loop or encoder-based products are particularly amusing), are based on reasonable estimation of the consequences of a number of error sources. In some applications, however, a number of the above error sources can be side-stepped. One such application is a wafer positioning stage, where fiducials on each wafer can be located to high precision, and used to align the lasers X-Y coordinate frame to the wafer frame for the duration of a single set of measurements (typically several minutes or less). Our Applications Engineers can be relied upon to produce realistic estimates of the achievable accuracy in your specific application. For a more detailed discussion of interferometer based positioning, including four-pass and differential interferometers, wavelength compensators, etc., call for our free Application Note, “Accuracy in Positioning Systems”.

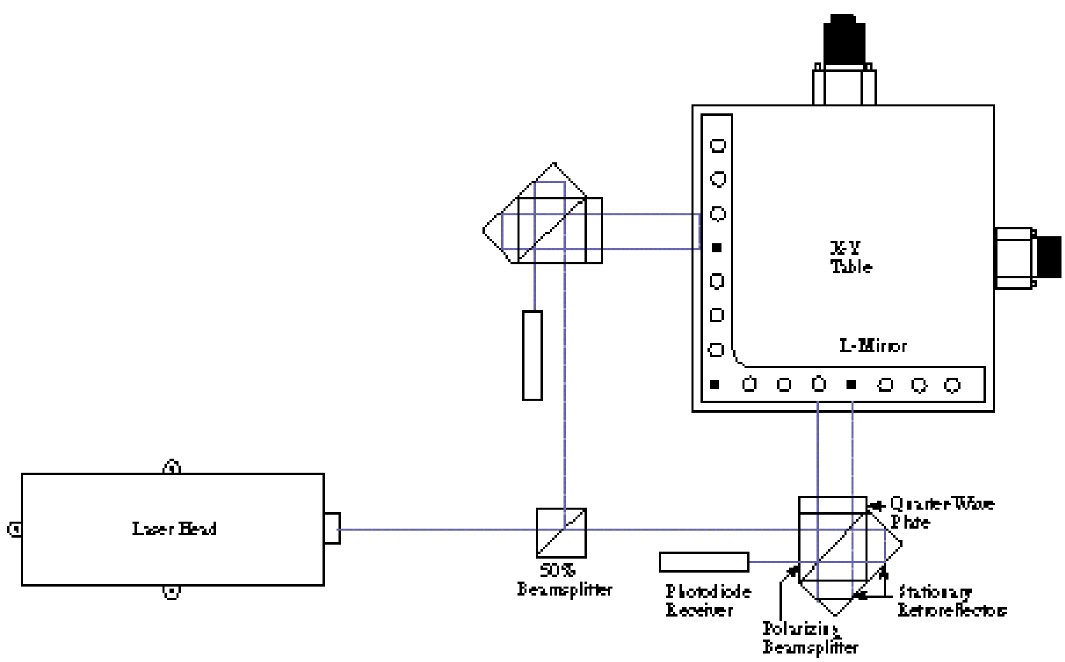


Figure 24b – Two-Axis Interferometer Beam Path

Limit Sensors

Our limit sensors are typically used for two purposes: as a highly repeatable position reference, and to help prevent overtravel. A limit sensor trip point is often referred to as a "home" position, and can be referenced at the beginning of a program or a move sequence.

Our motorized rotary tables incorporate a single sensor to provide a "home" position for registration (without limiting travel). Note that since a magnet in the rotary table is used to activate the limit switch, this home position will be different, depending on whether the limit is approached from a CW or CCW direction. We can optionally provide two limit sensors in a rotary table to restrict motion to a particular angular range. Our motorized single-axis and X-Y stages feature a limit sensor at each end of travel. In addition to providing two position references per axis, these limits can help prevent overtravel. They change state when encountered, providing a signal to the motion controller to stop motion. The limit trip point is typically set to activate 0.5 to 1.0 mm (0.020 to 0.040 inch) beyond nominal travel. All move profiles should include an appropriate distance to decelerate to stop within the nominal travel. Additional travel beyond the limit trip point is provided (before the rubber stop at the mechanical end of travel), but this may not be enough to allow the use of a limit sensor as a means of stopping a high speed move (since our stages emphasize compactness). In some applications, users may wish to delimit the travel of one of our stages. In this case, we install two limit magnets during the assembly of the stage; please notify the Sales Department at the time of order if this option is desired.

Our standard limit sensors operate via the Hall effect; they detect the position of a magnet that moves relative to the sensor. With the exception of those used in our RM and RMS Series positioners, our limits include the Hall sensor chip, a pull-up resistor, and integral leadwires. Due to space constraints, our RM and RMS stage models have no integral pull-up resistor. They are the open collector type, which means they switch low (sink current) upon activation. See the chart and figure 25 for details. The pull-up resistor provides a fail-safe function; if a limit cable is missing or broken, the controller will see a logic low, preventing motion until the cable is installed or repaired. If our RM and RMS Series positioners are used with controllers that are not our design, then external pull-up resistors in the motion control electronics may be required.

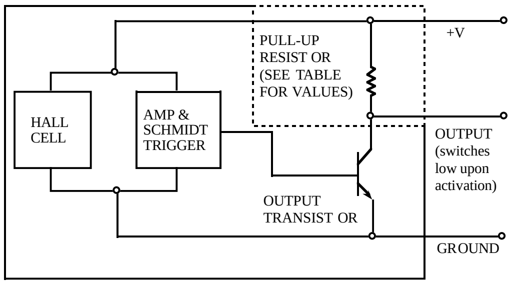


Figure 25

The limit switch signals are brought out on the Limit/Encoder connector, which is a sub-mini DE-9-S (socket) connector in most of our stage models (see the "Rotary Motor Mount" section). While our limit switches can operate at voltage levels of up to 24 volts, their nominal operating voltage is +5V.

PLEASE NOTE THAT THE LIMIT SWITCHES AND ANY ENCODERS SHARE THEIR SUPPLY VOLTAGE! IF YOUR STAGE IS EQUIPPED WITH ENCODERS, USE OF ANY SUPPLY VOLTAGE OTHER THAN +5 VOLTS WILL DAMAGE THE ENCODER. DO NOT USE A LIMIT SWITCH VOLTAGE OTHER THAN +5 VOLTS UNLESS YOUR SYSTEM HAS NO ENCODERS!

Positional repeatability of these sensors is ±1-2 microns (<0.0001 inch) at constant temperature. Homing algorithms

NEAT Limit Sensors

Positioner Model #	Limit P/N	Operating Voltage	Max. Sink Current	Pull Up Resistor	Supply Current (Max)
XY, XYR, XYL, MAS, OFL, XYMR, X-Theta, TM, TMS, LM, FM, XM, HM, HMS, SAS, Impulse, AirBeam, AirGlide, ZE, Z-Theta, RTR-4, RTR-6, & RTR-8	2087405	4.5-24 VDC	15 ma @ 24 VDC	6KΩ, 1/8 watt (internal)	8 milliamps
OFS, RT, RTR-10, & RTR-12	2087354	4.5-5.5 VDC	3 ma @ 5.5 VDC	4.7 KΩ, 1/4 watt (external)	4 milliamps
RM & RMS	1144066	4.5-24 VDC	20 ma @ 45 VDC	none	8 milliamps

Limit Sensors (Cont.d)

should engage the limit sensor briefly and then pull out, stopping at the limit release point, to avoid potential thermal drift incurred by continuous activation. Approaching the sensors at slow, consistent speeds will allow optimal repeatability. Despite the high intrinsic accuracy of our limit sensor, if its trip point is very close to a step or count boundary (especially in low resolution systems), small oscillations in the load position may result in a ±1 step ambiguity in the home position. The use of a once per revolution signal from a rotary encoder in conjunction with the limit switch, or the index mark on a linear encoder, can provide yet more accurate home position repeatability.

We have designed a PC board that allows compatibility with controllers that look for alternate limit sensors (not of our design). This surface mount board fits within the motor mount of most of our stage models, and is directly connected to the DE-9 Limit/Encoder connector. This board has been designed so as to allow our standard limit switches to achieve compatibility with virtually all motion controllers available in the marketplace. Since this board is implemented in surface mount technology, it is not configurable by our customers. If you plan on operating a stage with a motion controller from another vendor, it is important that you inform the salesperson of the intended controller, or its electrical input requirements, at the time of order. We can also reconfigure stages should you change your motion controller, but this will require that the stage be returned to our Customer Service Department.

Rotary Motor Mount

Our 23 frame motor mount is made of aluminum, and (among other things) mounts a motor to a positioning table. We feel that the careful attention to detail and continuous design improvement in our motor mount exemplifies the quality and consideration that go into our entire product line. Key features of the motor mount include integral motor and limit/encoder connectors, built-in encoder and index signal capabilities, and positive motor alignment.

Our standard 23 frame stepping motors are custom manufactured with the motor leads exiting the motor face, and terminated in a locking, strain-relieved DE-9 connector. When bolted to the motor mount, the leads perform a 180 degree bend within the mount, and the connector is secured with two #4-40 jackscrews. This is superior to conventional motor wiring, which requires offset lead cutting, soldering the main cable, heat shrinking the solder joints, sliding back the insulation jacket, etc. Such cable assemblies, common among competitive designs, are permanently wired to the motor, have no strain-relief, and are prone to wire chaffing and shorts. With our motor mount connector, the cable is easily detachable, with an electrical shield terminated at the motor case, a secure strain-relief, and a metal connector hood with locking jackscrews for a positive interconnection. The motor connector is a male (pin type); this choice of connector polarities assures that the cable coming from the motor drive electronics has socket contacts, and hence cannot short out on exposed metal surfaces.

On the other side of the motor mount, an internal surface-mount PC board combines the limit switch and (if optionally selected) the encoder signals and brings these out on the limit/encoder connector. As with the motor connector, a secure interconnection is provided via this DE-9 connector with its locking jackscrews. The limit/encoder connector is of opposite polarity (socket contacts) to the motor connector, preventing inadvertent misconnection.

The motor mount also has an internal chamber for a flexible shaft coupling, which connects the motor drive shaft to the leadscrew shaft. Access slots at the top and bottom of the motor mount allow two-sided access to the clamp screws, which secure the helical coupling. Since the coupling diameter (1.00") is less than the clearance hole which accepts the motor mounting boss, the coupling can be removed along with the motor, should service be necessary. The torsional stiffness of our most widely used coupling is 100 N-m/rad (0.07 oz-in/arc-sec); higher stiffness couplings are optionally available.

We have developed 2000 or 4000 count per revolution optical encoders, which can be mounted within the motor mount. This option (see Rotary Encoders) consists of a modular read head, which mounts to the stage body, and a 500 or 1000 line/revolution code disk, which mounts to the leadscrew shaft via an aluminum hub. The encoder is powered through the limit/encoder connector, resides entirely within the motor mount, and outputs A and B channel position information on pins 6 and 7 of this connector. Mounting the encoder on the table side of the flexible coupling provides more accurate positional tracking, and its location within the motor mount keeps the encoder safely out of harm's way. An index on the 2000 count per revolution encoder provides a convenient signal once for each motor revolution. This can improve the accuracy of an end-of-travel limit sensor used as a "home" reference position (among other things). Our standard rotary encoder and internal PC board provided differential outputs. When a linear encoder is specified, the output signals are wired to our standard limit/encoder PC board, and are present on the same pins of the DE-9 connector as with our rotary encoder.

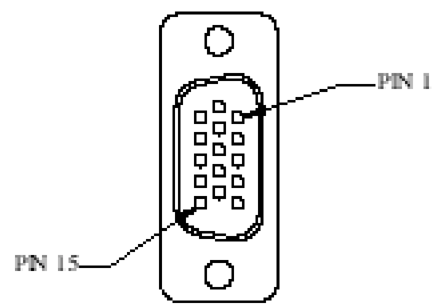
In applications where positioning stages are shipped without motors, a slot is machined in the motor mount to allow the user to dress the leads forward to a connector. This technique is recommended over direct cabling to the motor, since it allows the use of separate cables, and provides a locking, strain-relieved connection.

Standard Product Pinouts

The following pinouts detail the assignment of motor and limit/encoder connector pins. Note that where applicable, two connectors are used; they are opposite polarity, so that their pin numberings are mirror images of each other. Refer to the tables and figures (26a-e) below for details.

Our linear motor driven units utilize a DA-15P connector for the motor signals and a DE-9S connector for the limit and encoder signals of each axis. This includes our SAS™, Impulse™, and AirBeam™ products.

Male Connector



Linear Motor Units

Pin	Motor Connector (DA-15P)	Limit/Encoder Connector (DE-9S)	Limit/Encoder Connector (HD-15P)
1	Phase 1	+5 Volts	Limit +5V
2	Phase 1	+ Limit Output [1]	Limit Out – Plus
3	Phase 3	– Limit Output	Limit Out – Minus
4	Phase 2	Index Output [2]	Shield (Spare)
5	Phase 2	Ground	Limit Ground
6	Ground	Encoder Channel A	Encoder +5V
7	Hall 1	Encoder Channel B	Encoder A
8	Hall 2	Encoder Channel $\bar{A}$	Encoder A
9	Phase 1	Encoder Channel $\bar{B}$	Encoder B
10	Phase 3		Encoder B
11	Phase 3		Encoder Z
12	Phase 2		Encoder Z
13	+5 Volts		Home / Reference
14`	Fault		Encoder Ground
15	Hall 3		Shield

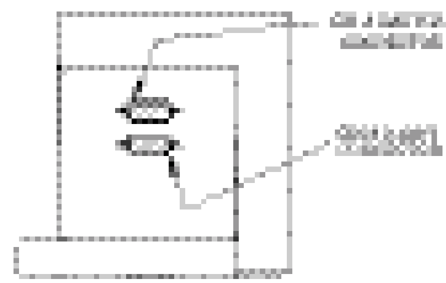


Figure 26a

Products utilizing a 23 frame rotary motor include a DE-9P connector for each motor and a DE-9S for the limits/encoder for each axis. Note that the motor mount is inverted on the lower axis of monolithic X-Y tables (2" to 10" travel). The following pinouts apply to our XY™, XYR™, XYL™, OFL™, TM™, TMS™, LM™, FM™, XM™, HM™, HMS™, Z-Elevator, Z-Theta™, RT™, and RTR™ Series units.

23 Frame Rotary Motor Units

Pin	Motor Connector (DE-9P)			Limit/Encoder Connector (DE-9S)
	Stepper	Servo		
		Brushless	Brush	
1	Coil A	Motor Phase 1	Motor +V	+5 Volts
2	Coil $\bar{A}$	Motor phase 2	Not connected	+ Limit Output [1]
3	Not connected	Ground	Not connected	– Limit Output
4	Coil B	Hall input 1	Not connected	Index Output [2]
5	Coil $\bar{B}$	Hall input 2	Not connected	Ground
6	Coil A, center tap	Motor phase 3	Motor -V	Encoder Channel A
7	Not connected	+5 volts	Not connected	Encoder Channel B
8	Not connected	Motor Fault Input	Not connected	Encoder Channel $\bar{A}$
9	Coil B, center tap	Hall input 3	Not connected	Encoder Channel $\bar{B}$

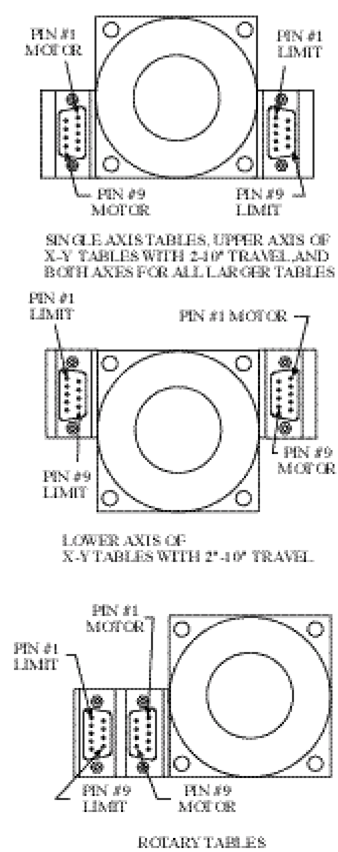


Figure 26b

[1] The + Limit Output is activated by moves which result from counterclockwise rotation as viewed facing the motor knob drive shaft. Movement away from the motor trips the "+" limit.

[2] The index output is a once per revolution signal for 2,000 count/revolution rotary encoders, and a once per travel signal for linear encoders.

Standard Product Pinouts (Cont.d)

Our miniature XYR-3030™ unit uses the same DE-9 connectors and pinouts as our products that use 23 frame rotary motors.

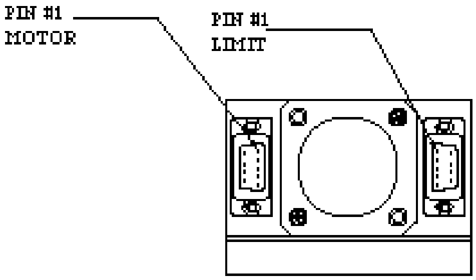
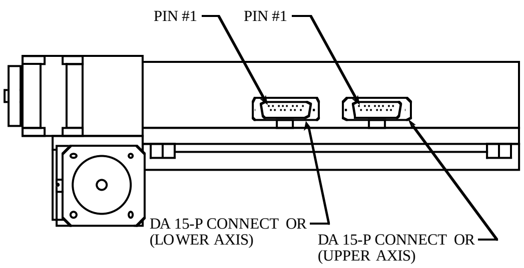


Figure 26c

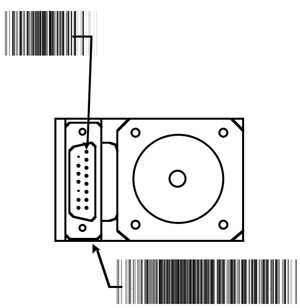
XYR-3030 Stage

Pin	Motor Connector (DE-9P) Upper Axis	Motor Connector (DE-9P) Lower Axis	Limit/Encoder Connector (DE-9S)
1	Coil A	Coil $\bar{A}$	+5 Volts
2	Coil $\bar{A}$	Coil A	+ Limit Output[1]
3	Not connected	Not connected	– Limit Output
4	Coil B	Coil B	Index Output[2]
5	Coil $\bar{B}$	Coil $\bar{B}$	Ground
6	Coil A, center tap	Coil A, center tap	
7	Not connected	Not connected	
8	Not connected	Not connected	
9	Coil B, center tap	Coil B, center tap	

Our XYMR™, X-Theta™, and OFS™ Series units use a single DA-15P connector for each axis.



XYMR & X-Theta Stages



OFS Series

Figure 26d

XYMR, X-Theta, & OFS Series

Pin	Motor/Limit/Encoder Connector (DA-15P) Stepper	Servo Brushless	Servo Brush
1	Coil A	Motor Phase	Motor +V
2	Coil $\bar{A}$	Motor Phase	
3	Coil B, center tap	Hall +5 Volts	
4	+5 Volts	Hall Input 3	
5	+ Limit Output	Logic +5 Volts	+5 Volts
6	Encoder Channel $\bar{A}$	+ Limit Output[1]	+ Limit Output[1]
7	Encoder Channel A	– Limit Output	– Limit Output
8	Index Output	Index Output[2]	Index Output[2]
9	Coil A, center tap	Motor Phase 3	Motor -V
10	Coil B	Hall Ground	Hall Ground
11	Coil $\bar{B}$	Hall Input 1	Hall input 1
12	Ground	Hall Input 2	Hall input 2
13	– Limit Output	Logic Ground	Logic Ground
14	Encoder Channel $\bar{B}$	Encoder Channel A	Encoder Channel A
15	Encoder Channel B	Encoder Channel B	Encoder Channel B

Our RM™ and RMS™ Series units utilize latching in-line connectors.

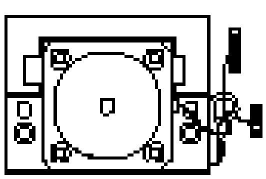


Figure 26e

RM & RMS Series

Pin	Motor Connector	Limit Connector
1	Coil A	+5 Volts
2	Coil A, center tap	+ Limit Output[1]
3	Coil $\bar{A}$	– Limit Output
4	Coil B	Ground
5	Coil B, center tap	
6	Coil $\bar{B}$	

[1] The + Limit Output is activated by moves which result from counterclockwise rotation as viewed facing the motor knob drive shaft. Movement away from the motor trips the "+" limit.

[2] The index output is a once per revolution signal for 2,000 count/revolution rotary encoders, and a once per travel signal for linear encoders.

Stepping Motor Drives

Standard 1.8 degree stepping motors (Figure 27) consist of a laminated, toothed stator wound with two center tapped coils, surrounding a 50 pole hybrid rotor. The rotor consists of an axially magnetized permanent magnet, with two laminated iron cups. Unlike DC motors, applying current to the motor windings generates a torque which resists rotation (the holding torque).

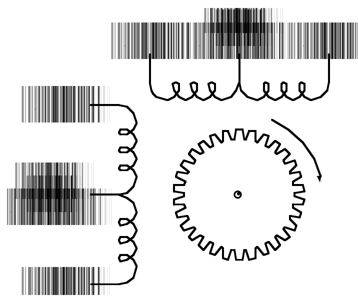


Figure 27 – 1.8° Stepping Motor

However, by switching coils on and off in a specific four step sequence, (Figure 28a), the rotor will “step” 1.8 degrees per current change. An optional eight step sequence, (Figure 28b), doubles the resolution to 0.9 degrees (400 steps per revolution). Rotation is therefore achieved by simply applying an appropriate sequence of winding currents.

Full Step Current Switching Sequence

	Phase 1	Phase 2	Phase 3	Phase 4
step 1	on	on	off	off
step 2	on	off	off	on
step 3	off	off	on	on
step 4	off	on	on	off

Figure 28a

Half Step Current Switching Sequence

	Phase 1	Phase 2	Phase 3	Phase 4
step 1	on	on	off	off
step 2	on	off	off	off
step 3	on	off	off	on
step 4	off	off	off	on
step 5	off	off	on	on
step 6	off	off	on	off
step 7	off	on	on	off
step 8	off	on	off	off

Figure 28b

To reverse direction, simply read from the bottom of each figure.

Several factors complicate this otherwise simple scheme. An energized stepping motor exhibits a rotary stiffness which resists deflection from its current position. Coupled with the rotary inertia of the rotor, this spring-mass system produces a fundamental resonance in the 50 to 150 Hz range. Operation at step rates near this natural frequency increases noise and vibration, and may cause the motor to drop out of synchronization (lose position). The use of microstepping, which is implemented on all of our stepping motor drives, dramatically reduces or eliminates this effect.

Due to the motor rotor inertia, there is a limit to the step rate that can be applied to a stationary motor without it stalling, or failing to follow the step train. This rate, called the STOP-START RATE, is a function of the motor's holding torque, rotor inertia, and load inertia. It ranges from 400 to 1000 full steps per second (2 to 5 revolutions per second); a typical value for a lightly loaded 23 frame motor is 700 full steps per second. To operate at step rates above this value, the step frequency must be accelerated, or “ramped” from a rate below the start-stop rate to the desired top speed. The starting frequency is usually chosen to be above the fundamental resonance and safely below the stop-start rate; a value of 400 full steps per second (2 revolutions per second) is typically employed with 23 frame motors.

In addition to resisting instantaneous starts at high step rates, the rotor (and load) inertia can produce overshoot if the pulse train is abruptly terminated. Accordingly, the stopping point must be anticipated and the motor ramped down to an appropriate frequency (again, 400 full steps/second is typical) before stopping. The overall velocity profile for a move is shown in Figure 29a; short moves, which may not reach the programmed top speed, result in triangular moves, as shown in Figure 29b. The allowable acceleration and deceleration values are determined by the motor's torque, drive type, and the total inertia.

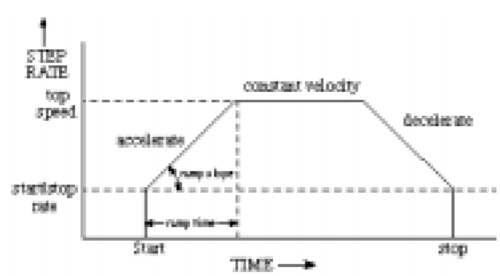


Figure 29a – Trapezoidal Velocity Profile

Stepping Motor Drives (Cont.d)

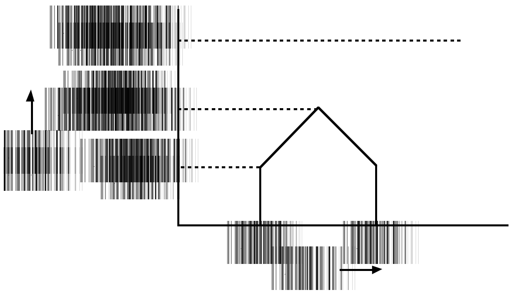


Figure 29b – Triangular Velocity Profile

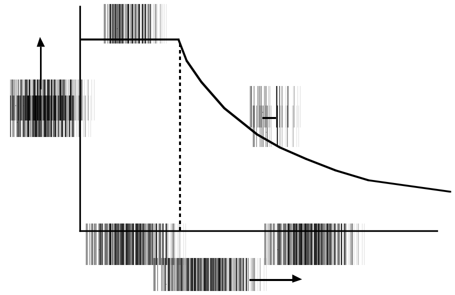


Figure 31 – Speed-Torque Curve

Having implemented appropriate start-stop rates and acceleration/deceleration ramps, the next concern is usually the maximum achievable slew rate. This is dominated by the inductance of the stepping motor's windings. As a result of this inductance, the current in the motor windings does not instantly rise to its expected value of V/R. Instead, it follows the formula:

$$I=\frac{V}{R}\left(1-e^{-R/(Lt)}\right)$$

which starts out linearly and then asymptotically approaches the level expected from the winding resistance and the applied voltage. At low step rates, (Figure 30a) the winding current has sufficient time to reach its full value, providing rated torque. As the step rate is increased, however, the winding current can only build to a fraction of its full torque value before it is switched off (Figure 30b). As a result, motor torque falls with increasing step frequency; eventually, there is insufficient torque to drive the load and the motor stalls. Speed-torque curves (Figure 31) can be broken into two regions; a low speed region within which torque is constant, and a high speed region, within which torque is inversely proportional to frequency.

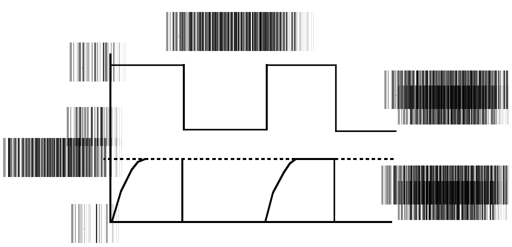


Figure 30a – Low Step Rates

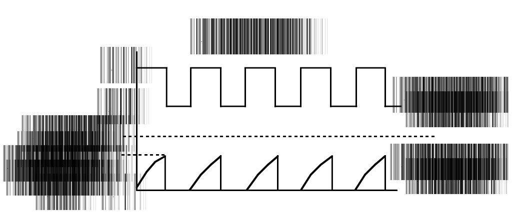


Figure 30b – High Step Rates

The challenge in driving stepping motors is to get the current to rise as quickly as possible, thereby providing more high speed torque. One technique, called unipolar L/R, sends current into the center tap of each coil, and alternately switches one end of each winding to ground (Figure 32a). The unipolar L/R drive gets around the slow current built-up by operating the motor from a voltage many times higher than its rating. Large dropping resistors are used to limit the motor current to its rated value. The effect of the added resistors is to make the load more resistive and less inductive in nature (changing the L/R term in the equation), which causes the current to build up quickly. This technique is limited in practice by the resulting large power dissipation in the dropping resistors, and has been superceded by chopping drives (see below).

A more advanced technique, referred to as a bipolar chopper, uses twice as many transistors in an H bridge configuration (Figure 32b). By turning on diagonally opposed transistors, current can be made to flow in both directions through the coil. Generally, only half of each coil (center tap to one end) is driven. To produce the fastest possible rise of current in the winding, the drive voltage is set at 10-20 times the rated coil voltage. A sensing resistor is then used to shut off current to the motor when the rate current is reached. Current then recirculates through the winding until it decays below the sensing threshold, when the transistors are again turned on. The motor current is therefore “chopped”, typically at an inaudible 20 KHz, providing optimum high speed performance without exceeding rated motor current. All of our stepping motor drives employ bipolar choppers, as well as microstepping; we also offer the drive modules separately (MDM7 and HRDM20), for OEM users interested in incorporating them into their own systems. All of our chopper drives incorporate full protection from midrange instability, an otherwise potentially serious stepping motor resonance.

When operated for optimum performance, stepping motors should be expected to run hot. While it may seem alarming to first-time users, case temperatures of 100 - 150 degrees F are of no concern, given the 125 degree C (257 degree F) rating of the motor coils. A cool or luke-warm motor is, in fact, oper-

Stepping Motor Drives (Cont.d)

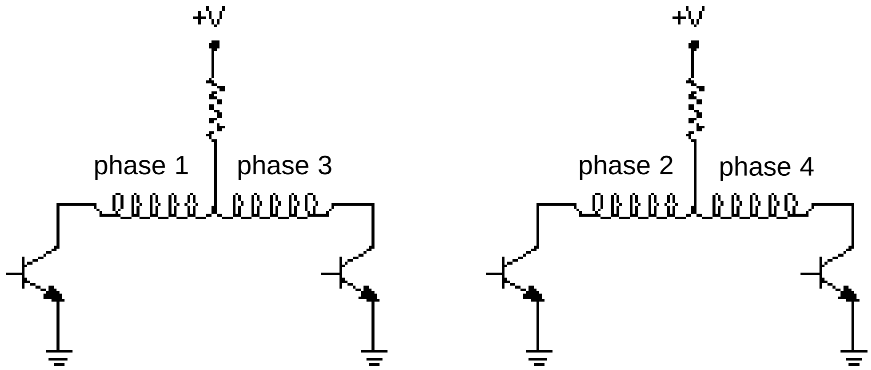


Figure 32a – L/R Drive

ating at below peak performance. Some controls (the 100 and 300 Series, for example) incorporate automatic logic to reduce current and hence heating, when not moving. Another common concern regards the 'stalling' of stepping motors. While highly undesirable in any given application, stalling will not, even if prolonged, result in damage to the motor or drive.

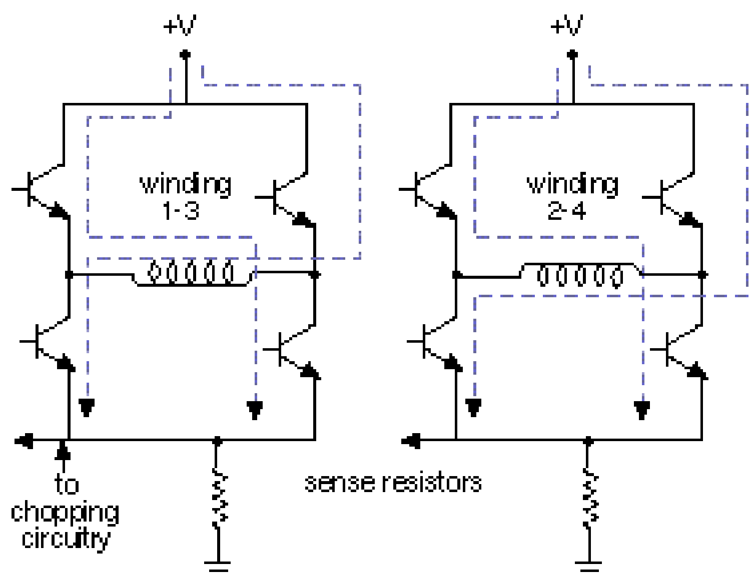


Figure 32b – H-Bridge Chopper Drive

Full Coil vs. Half Coil

Stepping motor drivers of bipolar chopper design have four output terminations. The vast majority of stepping motors, on the other hand, are wired in a six lead configuration (two coils, each with a center-tap). Users are therefore faced with a dilemma: which two wires should be left unterminated? The choice has a number of effects on system performance, and we have found significant confusion among users on the correct course of action. Hence, this section.

Frequently, users wire the drive across the full coil, with the center tap unconnected. The motivation is usually to "use all the copper", or to "get more torque". To see the actual effect more clearly, we need to consider how stepping motors are wound and rated. A very simple drive method, called unipolar, operates by sending current into the center tap and alternately switching either end of the coil to ground. Current thereby flows in one half of the winding at a time. Since unipolar drives are simpler and use half as many transistors as bipolar types (see Stepping Motor Drives), they have historically been more popular. Stepping motor current ratings have therefore adopted a unipolar convention: the rated motor torque will be generated if the rated current is applied through half the winding. Half coil operation is occasionally referred to as a "parallel" connection, while full coil operation is referred to as a "series" connection.

Since torque is proportional to magnetic field strength, and the magnetic field is proportional to the coil current times the number of turns, operation across the full coil should be performed at half the rated motor current. Driving the full coil at rated current saturates the iron, with negligible increase over rated torque but four times the ohmic heating. While we see some cases of users attempting to drive the full coil at rated current, most use the correct half current value.

One motivation in driving the full coil is to reduce ohmic heating: the IR losses of driving half the current into the full coil (twice the resistance) are half that of operating the motor in half coil mode. While this is true, the ohmic losses are a small fraction of the potential power dissipation of a stepping motor, and driving the full coil has a serious drawback - it presents four times the inductance of a half coil. This decreases high speed torque, and hence performance, to a substantial degree. "Wait…", I hear you say, "putting two equal inductors in series doubles, not quadruples, the inductance". While this holds with separated inductors, the inductance of a single coil increases as the square of the number of turns (doubling the number of turns also doubles the flux through the original turns).

As discussed in the Stepping Motor Drives section, the speed-torque curve can be broken into two regions: the low speed region, within which torque remains constant, and the high speed region, where torque is inversely proportional to step frequency. Half coil operation doubles the frequency to which torque remains constant. In the high speed region, half coil drive will produce twice the torque (and twice the power) of full coil drive. Accordingly, any application requiring high speed operation should employ half coil drive. The only downside from this "free lunch" is a concomitant doubling of current required from the power supply, as well as increased drive and motor heating.

Driving motors across the full coil has one specific application: it allows low speed operation of high current motors whose current ratings exceed the capacity of the drive and/or power supply. Typically, this is associated with a high torque requirement, which can only be met with a high current motor. For example, the 310M can supply up to 3.5 amperes at 42 volts. An application requiring 250 oz-in of low speed torque could be addressed with our motor P/N 2198365, which requires 4.6 amps. If operated in half coil mode, the 310M could only be set to 3.5 amps, and the low speed torque would be significantly reduced. By setting the 310M to 2.3 amps and running the motor across the full coil, full torque would be generated for low speed moves. Note, however, that the size of the "constant torque" region would be half of that obtained with half coil drive. In addition, if the application required high speed operation, half coil drive would still be superior, since that region is dominated by motor inductance and drive voltage.

While most commercially available stepping motors do have 6 leads, some 8 lead motors are available. With these motors, the "half coils" of coil A and B are wired independently of each other (no "center tap"), allowing one additional option when connected to bipolar chopper stepping motor drives - a full coil connection with the half coils in parallel with each other. In this mode of operation, the proper drive current is the same as for half coil drive, twice that of series full coil drive, and the coil static power dissipation is 1/2 that of half coil and equal to that of series full coil drive. The coil load inductance is equal to that for half coil and 1/4 that of series full coil drive, and the coil load resistance is 1/2 that of the half coil and 1/4 that of the series full coil connection. This mode of operation offers the speed vs. torque performance of half coil drive (due to its low coil inductance) with the lower static power dissipation of full coil drive (due to its low coil resistance), and is often the preferred mode of operation for 8 lead motors. The only significant drawback to this mode of operation is that it does require the full half-coil static current, and there are more wires (and hence more ways to mis-wire the motor).

Microstepping

An important variation on conventional stepping motor drives is that of microstepping. Conventional bipolar drives alternate the current direction in one coil at every step, resulting in a rotor displacement of 1.8 degrees. In microstepping, the coil current is changed in much smaller increments, increasing in one coil as it decreases in the other. The rotor responds by swinging to its new magnetic equilibrium, which can be a small fraction of a full step.

Microstepping has two principal benefits: it provides increased resolution without a sacrifice in top speed, and it provides smoother low speed motion. For example, to achieve a resolution of 5 microns with a full step system requires the use of a screw with a 1.0 mm lead. This places substantial constraints on top speed. A shaft speed of 40 revolutions per second results in a linear velocity of only 40 mm per second. Use of a divide-by-10 microstepper provides the same 5 micron resolution with a 10 mm leadscrew, but the linear velocity in this case is now 400 mm per second. Alternately, the resolution can be increased, to 0.5 micron with a 1 mm lead, or 1.0 micron with a 2 mm lead.

Since stepping motors, by definition, move in discrete angular increments, operation at low step rates (especially near the fundamental resonance) generates noise and vibration. Microstepping decreases the size of these increments, and increases their frequency for a given rotation rate. This results in significantly smoother low speed operation. A laser interferometer was used to produce the graphs in Figures 33a-d, which show the reduction in positional oscillations and velocity ripple for a positioner at a low (0.05 revolution/second) step rate.

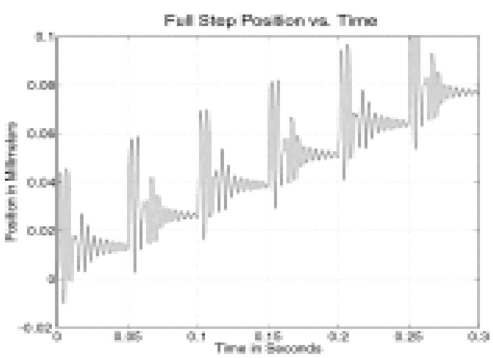


Figure 33a

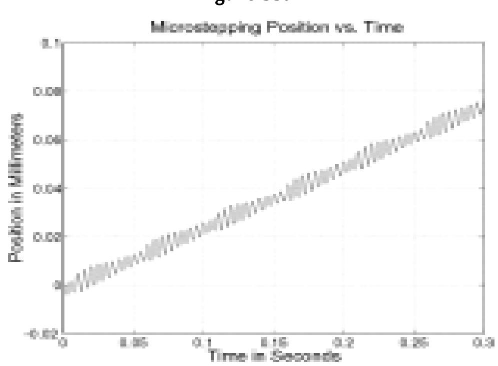


Figure 33b

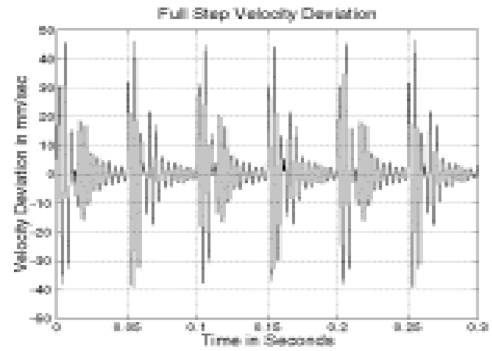


Figure 33c

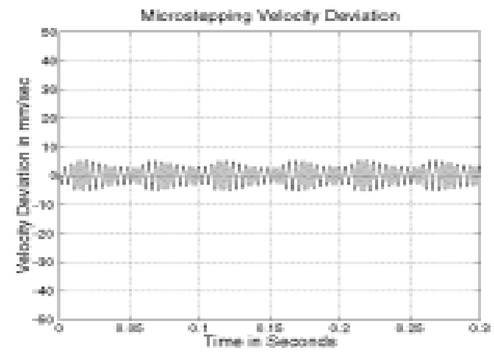


Figure 33d

Despite the apparent benefits of microstepping, it is frequently implemented with excessive degrees of subdivision. A key attribute of many commercial systems is “empty resolution”, where the apparent microstepping resolution can not be achieved. The torque which any microstep generates is found as follows: torque per microstep = motor holding torque x sine (90 degrees/S.D.R.), where S.D.R. is the step division ratio.

In the case of 50,000 step/revolution systems, the S.D.R. is 250, and each microstep produces a torque change of 0.3 oz-in (with a standard 53 oz-in motor). Most high repeatability, preloaded leadscrew systems have torques in the 3-6 oz-in range; accordingly, 10-20 microsteps must be taken before the torque builds to a level which results in leadscrew motion. Direction reversals behave similarly; what appears to be backlash in the positioning table is actually excessive microstepping resolution. High division ratios have the additional effect of limiting the achievable top speed, by the inability to produce very high speed pulse trains (100 revolutions per second at 50,000 steps per revolution requires 5 MHz ramped pulse trains).

We have chosen a standard microstepping division level of 10; this provides 2000 microsteps per revolution, matching the resolution of our built-in rotary encoders. The resulting torque per microstep of 3 oz-inches is also a close match to existing leadscrews, providing a tight coupling of command and response. For systems which require higher resolution, we offer an optional division level of 50. This is as high a value as is practical, and requires pulse rates of up to 1 MHz, but provides exceptional speed and resolution: 0.25 microns with a 400 step/revolution motor and a 5 mm leadscrew.

Midrange Resonance

Stepping motors exhibit an additional idiosyncrasy, which complicates drive design: midrange instability. This phenomena, which varies in severity with the nature of the load being driven, appears as an oscillation of the motor rotor from its intended position. It generally sets in at step rates of 1200 to 3000 full steps/second (6 to 15 revolutions per second). The oscillation itself is in the 50-150 Hz range, and often builds in amplitude over a number of cycles, causing a stall condition within 0.1 to 1 second. Unlike the fundamental motor resonance, half stepping and microstepping do not alleviate the problem (in fact, some popular microstepping systems are prone to this condition). Our bipolar chopper, microstepping drives completely suppress midrange resonance, by sensing the deviation from intended position and electronically introducing viscous damping to counteract the effect. In so doing, all the motor torque is made available to accelerate the load, instead of being wasted on spurious oscillations. Bizarre mechanical fixes to the problem, such as Lancester dampers, drill chucks on the motor rear shaft, etc., are accordingly unnecessary.

Servo Motor Drives

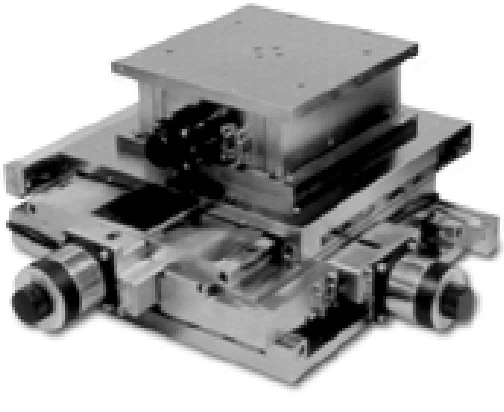
Our servo motors are available in rotary brushed, rotary brushless, and linear brushless models. The servo amplifiers to drive these motors can be categorized by their output stage type (Linear or Pulse Width Modulated), and the method of commutation (brushes, digital Halls, analog Halls, or encoder-based). The standard input to a servo amplifier is a single analog voltage, which typically can vary from +10 volts to –10 volts. While amplifiers can be configured as either voltage or current output, we only employ current mode amplifiers; in this case, an input voltage commands an output current. Operation in this mode provides less undesirable phase shift in the servo loop, and makes the amplifier less sensitive to changes in motor inductance.

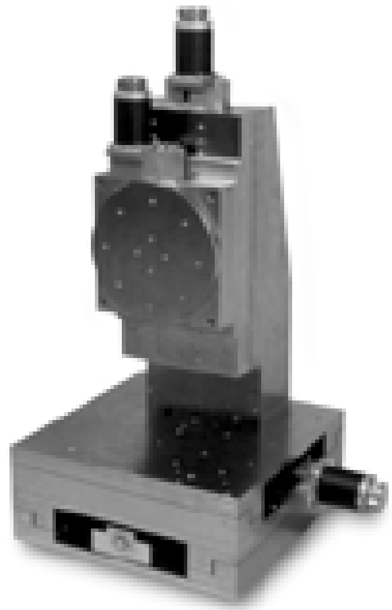
Amplifiers with a linear output stage have certain advantages, but must be carefully matched to the application. They have none of the electrical noise associated with PWM amplifiers, making them ideal candidates for noise sensitive applications. They can also provide very low errors as they cross through zero current. Their primary disadvantage is that they are quite limited in power, since the linear output stage is inefficient. In addition, care must be taken to avoid secondary breakdown, by remaining within the SAFE OPERATING AREA at all times. Accordingly, linear amplifiers are best suited to low power applications which can benefit from low zero-crossing errors and noise.

With PWM output stages, the output switches between full positive and full negative at ~20 kHz, and the duty cycle is varied to produce the desired average current. This technique is very efficient, and hence much higher electrical power is available in a compact, easy to cool amplifier. Since any polyphase motor must be commutated (coils energized alternately to maintain uniform torque or force) the method of commutation further differentiates servo amplifiers. Brush motors are self-commutating, and from the perspective of the amplifier, only a single two-wire coil is being driven. Our brushless motors are three phase motors, with three coils wound in a delta (triangular) fashion. Our BDM6 servo amplifier accepts three digital Hall-effect sensors, which are mounted on the motor and sense coil position. It includes three half bridges, which drive the three phases. By the simple expedient of jumpering the Hall input lines to particular states, this brushless amplifier can now drive a brush type motor.

In linear motor systems, the small force irregularities that result from the discrete switching of the motor coils can be undesirable. Sinusoidal commutation, which is analogous to the microstepping of step motors, replaces discrete coil switching with smooth, sinusoidally varying coil currents. There are two techniques to achieve this. In one, the three digital Hall sensors are replaced with two analog Hall devices, which sense both the magnetic field polarity and strength. The single servo amplifier input signal is multiplied by these two signals, and the resulting two signals command a specialized amplifier. This technique conserves DAC outputs, and requires no phase finding routine upon power-up. The other technique eliminates the analog multiplication, and relies on the controller to calculate the commutation signal values, based on the position of the linear encoder. While potentially more accurate, this technique requires more computational resources in the controller, uses two DAC outputs per axis, and requires a phase finding routine upon power-up.

High Vacuum Positioning Tables





Most of our standard positioning tables are available with optional preparation for use in high vacuum systems operating down to 10^{-7} torr. These tables follow specific, high vacuum compatible design rules resulting in outgassing rates below 1×10^{-5} torr-liter/second per axis.

In conventional positioning tables, a number of outgassing sources are present which corrupt high vacuum systems. Among these are the motors, lubricants, metal finishes, table materials and design. Our vacuum prepared stepping motors are extensively modified and incorporate 220 degree C magnet wire, teflon leadwire insulation and lacing, and a specialized bearing lubricant. All remaining stepping motor materials are selected for high vacuum compatibility and the finished motor undergoes a bake-out at 85 degrees C for 8 hours at 10^{-7} torr.

Lubricants are especially prone to outgassing and, if not properly selected, may condense on critical optical or semiconductor surfaces within the vacuum chamber. We use specialized perflorinated polyether lubricants that achieve vapor pressures of 9×10^{-9} torr at 100 degrees C.

Conventional anodization, the standard surface treatment for our products, vastly increases the tables' surface area and its tendency to absorb water vapor. This entrapped water vapor then constitutes a virtual leak which may require days to pump away. Vacuum prepared tables are finished with an electroless nickel plating that minimizes water vapor retention.

Vented, stainless steel fasteners are used on all blind tapped holes, eliminating trapped pockets of air. Any remaining cavities are similarly relieved to allow a rapid pump-down. Teflon insulated wires with fluxless solders are used on all connectors. A final three-stage cleaning process, followed by nylon-gloved assembly, further ensures a low overall outgassing rate. Upon request we can supply complete materials and procedures lists detailing design features of vacuum prepared tables.

In addition to the basic task of ensuring that the positioning stage does not corrupt the vacuum, there is also a need to avoid damage to the stage from the limited heat dissipation presented by the vacuum environment. All of our stepping motor controllers have an "idle" capability, which allows the motor phase currents to be brought to a small fraction of their nominal value when not moving. Both stepper and servo drives can be filtered so as to remove heating due to hysteretic iron losses, and we have additional means of ensuring that thermal issues are properly managed, to avoid any component failures due to overheating.

High Vacuum Positioning Tables (Cont.d)

Whenever positioning stages are mounted within a high vacuum enclosure, there is a need to route the motor, limit, and encoder lines through the wall of the vacuum chamber. While some users are proficient at this, for others the sourcing and installation of appropriate connectors can be a real problem. We have designed dedicated feedthrough plates, which can route up to three axes of positioning stages through the chamber wall, and which are suitable for vacuum levels above 1 x 10^{-7} Torr. See Figure 34. We offer both rectangular and circular versions. Unlike conventional circular connectors, these feedthrough plates separate the motor lines from limit and encoder signals for maximum noise immunity, and provide locking, strain relieved interconnects on both sides of the vacuum chamber. They also interface directly to our standard cable sets and motion controllers on the atmospheric pressure side, and accept specialized, vacuum compatible cables for the run from the chamber wall to the high vacuum stage.

Figure 34 – Vacuum Feedthru Plate

Vibration Isolation Systems

For many positioning systems of low to moderate resolution, no particular effort must be expended to isolate the positioning stage from environmental vibration. As the system resolution increases, however, the need to provide isolation from external vibration increases. It is not a simple matter to determine if a system needs isolation, or the degree of sophistication of that isolation, since the problem has a number of contributing factors. Primary among these is the amplitude and spectral content of the background vibration itself. While in some cases, identifying and removing or abating the vibration source(s) can be accomplished, in other cases the sources are an unavoidable component of the immediate environment. The next issue is the set of natural frequencies (resonances) of both the positioning stage and the other structural members of the overall system. In general, we strive to make these as high and well damped as is practical. We are then left with the convolution of the external excitation, whose amplitude and spectral content vary with the various resonances of the stage and structural members. The resulting unwanted relative motion between the customer payload on the stage, and the customer process (optical head, SEM column, etc.) is then evaluated relative to the application requirements for stability. If system performance is degraded in an unsatisfactory manner, then some means of attenuating the external vibration sources is indicated.

Isolation techniques vary widely. In less demanding applications, simple rubber mounting devices provide adequate relief. More commonly, active pneumatic isolation modules are used as three or four supports for the system. Each of these includes a low frequency (1-3 Hz) horizontal and vertical isolation mechanism, together with a mechanical pneumatic servo valve to maintain a level condition. These systems provide substantially lower natural frequencies, and therefore greater attenuation at the frequencies of interest, than simple rubber isolators. The recent trend towards the use of linear motors and higher speeds raises conflicts with traditional pneumatic isolators; unlike leadscrew driven stages, which are relatively insensitive to platform tilts, linear motor systems see a tilt as a direct force on the servo loop, of magnitude: Mass x sine (θ). More advanced isolation systems augment conventional pneumatic isolators with multiple axes of linear motors and velocity sensors. These can maintain a “rigid” yet isolated platform in response to background vibrations and stage movements. Sophisticated DSP based controls can also communicate with the stage motion controller, anticipating accelerations, forces, and center of gravity shifts, and compensating accordingly. Our Design Engineers can help advise you on the appropriate isolation solution for your application, as well as provide complete systems consisting of an integrated and tested stage and isolation system.

Low Magnetic Field Tables

Certain applications require that our positioning tables have minimal external magnetic fields. Some of these, such as mapping the field strengths of "wigglers" for synchrotron studies, take place at atmospheric pressure, while others (E-beam and focused ion beam systems, for example) require high vacuum preparation as well. We have employed a sensitive flux-gate gaussmeter to map the field strength around our positioning tables. Since the primary construction material is non-magnetic aluminum, the overall field strength is low. The worst component proved to be the small Nd-Fe-B magnet used to activate our Hall-effect limit sensors. The stepping or servo motors proved to be the only other significant field source, although their strength is quite low (in retrospect, effective motor design requires that the field be kept internal to the motor). Both the leadscrew and the ways, which are fabricated from tool steel (leadscrews are optionally available in 304 stainless) showed no clearly discernible magnetic field. The gaussmeter resolution is <0.5 mG, but the earth's field, together with fields resulting from building materials requires differential measurements as stage components are moved relative to the flux-gate detector.

For moderate sensitivity applications, we have prepared a 1 gauss level map of the fields surrounding our tables. The motor's field strength fell to this level at a cylinder 1.5" from the motor surface, and 3.5" from the motor ends. On the stage top surface, the limit magnet fell to the one gauss level on a hemisphere of 1½" radius, centered above the magnet location. Our limit sensors can easily be replaced with opto-interrupters, eliminating this field source.

In many cases, the motor's external field strength will have fallen to acceptable levels at the point of interest on the user's payload. Alternate procedures would include extending the motor shaft(s) and thereby moving the motor(s) away from the stage, and enclosing the motor in a can constructed from mu-metal (this can would have to be somewhat larger than the motor to avoid saturating the mu-metal). In high vacuum X-Y systems, custom spline-drive tables can be supplied which allow both motors to be located outside the vacuum chamber. While the steel components (leadscrew and ways) are prime candidates for increased field levels, the hardening process takes these materials above their Curie point. As mentioned, leadscrews can be provided in 304 stainless, and the rod and ball ways can be made from the mildly magnetic 440C stainless alloy. We regularly design custom stages with piezoelectric actuators, which have no external magnetic fields. Our air bearing stages are another means of minimizing ferrous components, although they are not vacuum compatable.

In some applications, typically involving electron beams, sensitivity to magnetic fields can be severe. While the testing methodology is rigorous, we can provide stage measurements below the 0.005 Gauss level. As mentioned above, mu-metal shielding and other design techniques may be required. Given the fact that the external field drops off as $1/r^3$, appropriate positioning of stage components can also be quite useful.

High Rad Tables

Our positioning tables can be configured to allow operation in a variety of radioactive environments. Such environments may contain alpha particles (doubly ionized helium), beta particles (electrons), X rays and gamma rays (photons), and/or neutrons (uncharged nucleons). Radiation can be measured in terms of dose, using roentgens (a unit of gamma ray exposure), or rads (a generalized radiation unit, depositing 100 ergs per gram of absorbing material), as well as in terms of source intensity, in curies (a source strength which produces 3.7×10^{10} disintegration per second).

In general, the limited penetration capabilities of alpha and beta particles require little or no modification to positioning tables to allow operation in such environments. X and gamma rays, which possess considerable penetrating power, present more of a problem. In particular, they can degrade most hydrocarbons and man-made polymers, and produce spurious noise in digital circuitry. Typical preventative measures include the use of specialized lubricants, motor coil varnishes, and wire insulations. Solid state limit sensors are replaced with mechanical or magnetic reed switches, and positional feedback, when required, uses multi-turn, high rad-rated resolvers in place of optical encoders. While gamma rays cannot activate or induce radioactivity in exposed materials, positioning table components can become "hot" if contaminated with radioactive particulates. In this case, special design rules may be required to allow the simple removal of replaceable components (motors, leadscrews, etc.) by means of remote manipulators. For example, we have provided stages with both end of travel limit sensors (reed switches) mounted in a small removable module, activated by a push rod when the table reached either end of travel. Such design techniques become increasingly important for operation in neutron rich environments. Neutron absorption can induce significant radioactivity in positioning table components, by converting the elements comprising the table into radioactive isotopes.

In all cases, we assume that the control electronics are not exposed to radiation, as we do not produce rad-hardened positioning controllers. Since our motor controls are completely functional with extended cable runs (up to 200 feet), they can usually be located away from the area housing the mechanical components.

We can supply positioning tables, complete with motors, limit switches, and feedback components, which are capable of operation at cumulative dose levels up to 1×10^7 rads. Since we do not stock rad-hardened motors, delivery times will be longer than for standard tables; in some cases, customers prefer the regular replacement of stock motors to minimize cost and delivery impacts. Careful consultation with our Applications Engineers is recommended to determine the best match of positioning table to your specific application requirements. Needless to say, our warranty procedure is modified for high-rad positioning systems; return authorizations will not be issued!