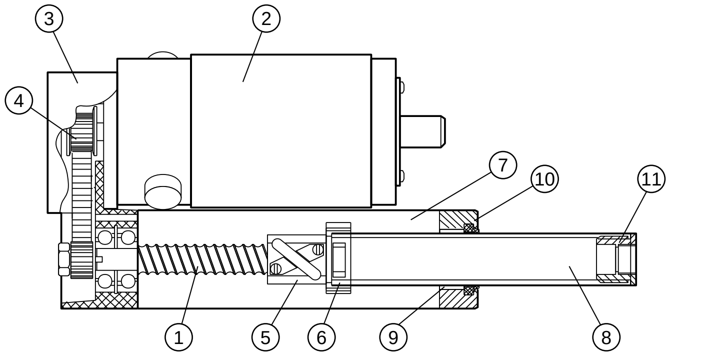
Rod Type or Rodless?

Rod Type actuators are similar in configuration to a hydraulic or pneumatic actuator and are preferred when you need to position an externally supported load, move a load that pivots, retrofit a hydraulic or pneumatic actuator, or have "reach in requirements".

EC and NV electric actuators (see the NV series figure, right) use *leadscrews* (1) to convert rotary motion into linear motion. The *motor* (2) is mounted to the *bearing housing* (3), and the motor's power is transmitted to the screw through a *gear*, or *timing-belt reduction* (4). The screw turns and moves the *drive nut* (5), which is connected to a *guide flange* (6). The guide flange keeps the nut from rotating, by sliding through the *guide cylinder* (7). The *thrust tube* (8) is threaded on to the nut, and is supported by the *sleeve bearing* (9) in the *rod-end housing* (10). The load is attached to the *rod end* (11).

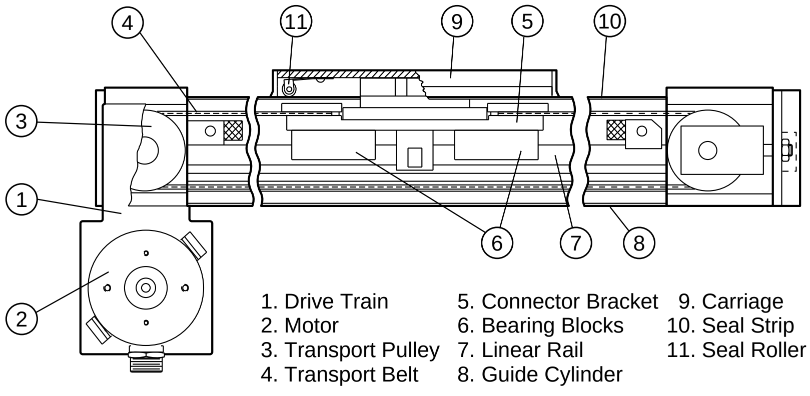
NV Series



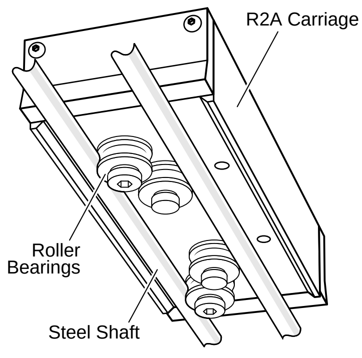
1. Leadscrew
2. Motor
3. Bearing Housing
4. Drive Train
5. Drive Nut
6. Guide Flange
7. Guide Cylinder
8. Thrust Tube
9. Sleeve Bearing
10. Rod End Housing
11. Rod End

Rodless actuators have a bearing support system and a carriage that runs the length of the actuator. This type of actuator is preferred when you need to save space by eliminating external guides and ways, when high speed and long stroke lengths are needed, when the shortest overall work envelope is needed, or when a multi axis Cartesian System is required.

R2, R3 and R4 rodless actuators use a leadscrew or a transport belt to convert the motor's power to linear thrust. Pictured below is a belt-drive actuator. As in the EC and NV actuators, there is a *timing belt*, or *gear reduction* (1) between the *motor* (2) and the *driven pulley* (3). The *transport belt* (4) runs over two pulleys and each end is connected to the *connector bracket* (5). The connector bracket is connected to two *bearing blocks* (6) that ride on the recirculating ball-bearing *rail* (7) that is mounted in *the guide cylinder* (8). The *carriage* (9) is mounted to the connector bracket and the *seal strip* (10) runs between them. The connector bracket lifts the seal as the carriage moves, while *roller wheels* (11) in the carriage push the seal back in place. **R2 actuators** have no bearing blocks, but instead have *roller wheels* for bearing support (as seen in the figure below). Four track-roller bearings run on two hardened and ground steel shafts, pressed into the extrusion.

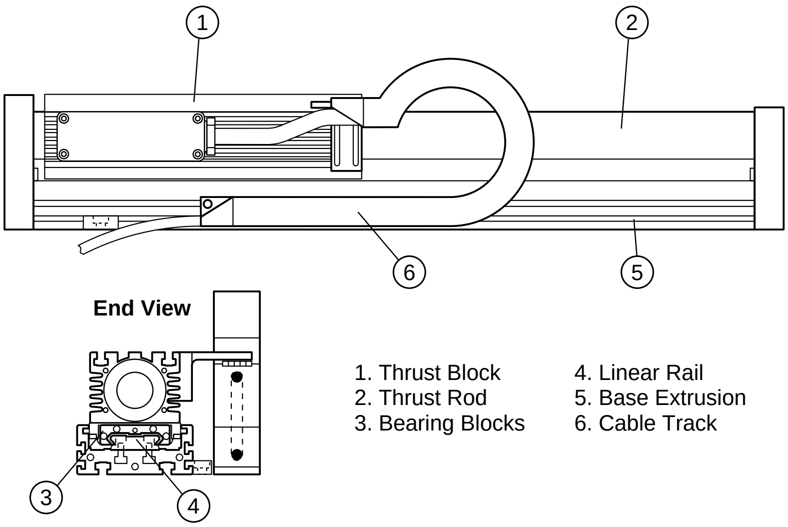


1. Drive Train
2. Motor
3. Transport Pulley
4. Transport Belt
5. Connector Bracket
6. Bearing Blocks
7. Linear Rail
8. Guide Cylinder
9. Carriage
10. Seal Strip
11. Seal Roller



Rod Type or Rodless?

Linear Modules are very efficient, high speed systems, that have repeatabilities in microns. They are similar in configuration to the R-series Rodless actuator family, but operate on a different principle. A linear motor converts electric current directly into linear force. There is no rotary motion to be converted. Motor coils, hall effect sensors, and the linear-encoder read head are mounted in the *Thrust Block* (1), while permanent magnets are in the *Thrust Rod* (2). The Thrust Block is connected to one or two *bearing blocks* (3) that ride on a wide linear *rail* (4), which is mounted to the *base extrusion* (5). The Thrust Rod runs through the Thrust Block and is mounted to brackets on either end of the base extrusion. The motor wiring is housed in a *cable track* (6) that flexes with the movement of the Thrust block.



1. Thrust Block
2. Thrust Rod
3. Bearing Blocks
4. Linear Rail
5. Base Extrusion
6. Cable Track

Rotary to Linear Conversion

Linear motion systems driven by rotating electric motors commonly employ one of three rotary-to-linear conversion systems: **ballscrew, belt drive, or Acme screw.**

Leadscrews

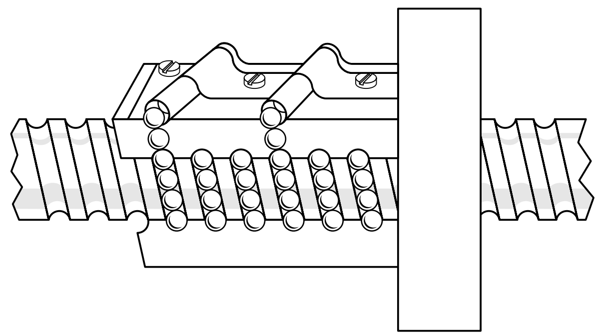
Screw-drive mechanisms, whether Acme screw or ballscrew provide high thrust (to thousands of pounds), but are often limited by critical speed, maximum recirculation speed of ball nut circuits, or sliding friction of Acme nut systems.

Ballscrew

The majority of linear motion applications convert motor torque to linear thrust using ballscrews due to their ability to convert more than 90% of the motor's torque to thrust. As seen below the ballnut uses one or more circuits of recirculating steel balls which roll between the nut and ballscrew threads. Ballscrews provide an effective solution when the application requires:

- High efficiency - low friction
- High duty cycle (> 50%)
- Long life - low wear

Ballscrew

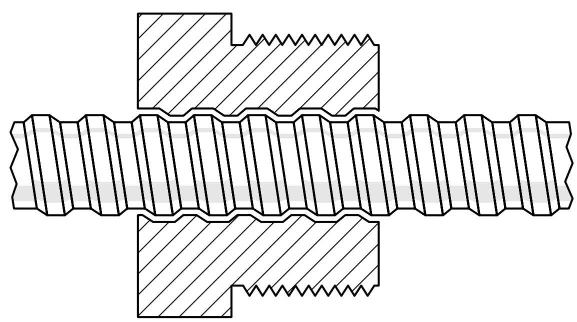


Acme Screw

The Acme Screw uses a plastic or bronze solid nut that slides along the threads of the screw, much like an ordinary nut and bolt. Since there are no rolling elements between the nut and the leadscrew, Acme screws yield only 30-50% of the motor's energy to driving the load. The remaining energy is lost to friction and dissipated as heat. This heat generation limits the duty cycle to less than 50%. A great benefit of the Acme screw is its ability to hold a vertical load in a power-off situation (refer to the Backdrive specifications for acme screw actuators). The Acme screw is a good choice for applications requiring:

- Low speeds
- Low duty cycles (50%)
- The ability to hold position while motor power is off

Acme Screw



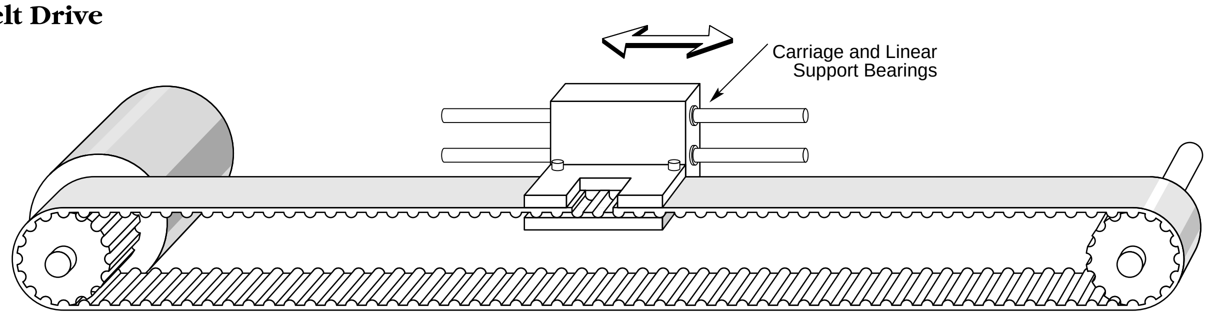
Timing Belt

Belt Drive systems offer many of the benefits of ballscrews, yet have fewer moving parts, and do not have the critical speed limits of leadscrew-driven systems. They generally provide more linear motion from the same motor movement, resulting in higher travel speeds with minimal component wear. In contrast, this design results in lower repeatability and accuracy. Thrust capability is also smaller compared to screw-drive systems due to the tensile strength limitation of the transport belt.

The general configuration can be seen in the figure. A toothed belt passes around a pulley in each end of the actuator and is attached to the carriage to pull it back and forth along the length of travel. The carriage is supported by a linear bearing system to provide load carrying capacity. The neoprene belt is reinforced with steel tensile elements to provide strength and minimize belt stretch. Timing belt systems are a good solution for applications requiring:

- High speeds
- Low thrusts
- High efficiency
- High duty cycle

Belt Drive

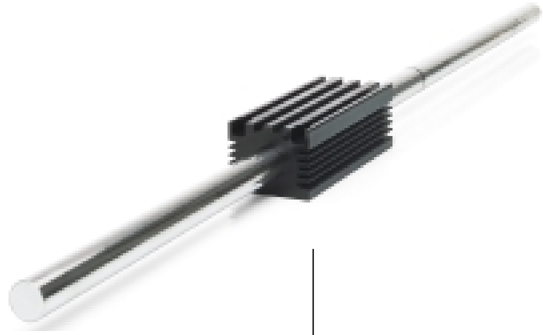


Mechanical Drive Comparison

The following chart will help pinpoint which linear drive mechanism is right for your application. IDC offers many actuator options, such as brakes, encoders, lubrication ports, preloaded nuts, and precision ground screws, that may help you meet your specification. If these standard options do not meet your requirements, please contact the factory for information regarding custom solutions.

Considerations	Acme Screw	Ballscrew	Belt Drive
Noise	Quiet	Noisy	Quiet
Back Driving	Self locking	Easily backdrives	Easily backdrives
Backlash	Increases with wear	Constant throughout screw life	Can increase with wear or stretching of belt
Repeatability	+/-0.005" to 0.0005"	+/-0.005" to 0.0005"	+/-0.004"
Duty Cycle	Moderate max. 60%	High max. 100%	High max. 100%
Mechanical Efficiency	Low Plastic Nut - 50% Bronze Nut - 40%	High - 90%	High - 90%
Life and Mechanical Wear	Shorter life due to high friction	Longer	Longer
Shock loads	Higher	Lower	Low
Smoothness	Smooth operation at lower speeds	Smooth operation at all speeds	Smooth operation at all speeds
Speeds	Low	High	Higher
Cost	$$ Lowest	$$$ Moderate	$$$ Moderate

Mechanical Drive Comparison



Linear Motor	Comments
Moderate	**Acme:** Sliding nut design provides quiet operation. **Ball:** Transmits audible noise as balls recirculate through nut during motion. **Belt:** the neoprene cover of the belt provides damping of noise. The support bearings will generate some noise.
Easily backdrives	**Acme:** Good for vertical applications; vibration may cause position loss. **Ball:** May require brake or holding device when no holding torque is applied to the screw. **Belt:** May require brake or holding device when no holding torque is applied to the drive pulley. **Linear motor:** Low friction system–backdrives easily.
Negligible	**Acme:** Considered worn-out when backlash exceeds 0.020". Typically 0.006" when shipped from the factory. **Ball:** Typically constant at 0.006" (lead screw/nut only). **Belt:** Typically at 0.010" when shipped. Can be adjusted to compensate for wear or stretching. **Linear motor:** with a preloaded rail there is zero backlash.
Best (microns)	**Linear motor:** repeatability depends upon the encoder used. Sub micron possible.
High max. 100%	**Acme:** Low duty cycle due to high friction from sliding surface design. **Ball:** High screw efficiency and low friction allow high duty cycle. **Belt:** High efficiency provides low heating and high duty cycle.
Highest - 90-95%	**Acme:** Low efficiency sliding friction surfaces. **Ball**: High efficiency smooth rolling contact. **Linear motor:** Efficient - no conversion needed between rotary and linear motion.
Longest life, least mechanical wear	**Acme:** Mechanical wear is a function of duty cycle, load and speed. **Ball:** Virtually no mechanical wear when operated within rated load specifications. **Belt:** High efficiency contributes to long life. Drive belts can be easily replaced to extend system life. **Linear motor:** Limited by cable flex, and bearing life.
Highest	**Acme:** Better suited because of larger surface area. **Ball:** Brunelling of steel balls limits shock load capability. **Belt:** Shock loads can cause fatigue and stretching of drive belts.
Smoothest	**Acme:** At extreme low speeds, units have a tendency to stop/start stutter (due to friction). **Ball:** Generally smoother than acme through the entire speed range. **Belt:** 180° engagement of belt provides continuous smooth contact throughout the speed range.
Highest	**Acme:** Extreme speeds and accelerations can generate excessive heat and deform the screw. **Belt:** Each revolution of the drive pulley provides several inches of travel. Speeds up to 120 in/sec can be achieved. **Linear motor:** LM to 196 in/sec. LD to 394 in/sec.
$$$$ Highest	

Electric Cylinders vs. Hydraulics & Pneumatics

For many applications, hydraulic or pneumatic linear cylinders are a better choice than their electromechanical alternatives. For example, when extremely heavy loads (>25,000 N [5,620 lb]) must be moved, hydraulic cylinders are usually the best solution.

Or, when very light loads must be moved rapidly and repeatedly from one fixed location to another fixed location, pneumatic cylinders may be the most economical solution.

	Industrial Devices Electric Cylinders	Hydraulic Cylinders	Pneumatic Cylinders
Installation	All electric operation requires simple wiring; directly compatible with other electronic controls.	Requires expensive plumbing, filtering, pumps, etc. Must pay close attention to compatibility of components.	Requires expensive plumbing, filtering, pumps, etc.
Precise Positioning	Cost-effective, repeatable (to ±0.013mm [±0.0005in]), rigid multi-stop capabilities.	Requires expensive position sensing and precise electro-hydraulic valving to implement; has tendency to creep.	Most difficult to achieve. Requires expensive position sensing and precise valving to implement; has tendency to creep.
Control	Solid-state microprocessor-based controls allow automatic operation of complex motion sequences.	Requires electronic/fluid interfaces and sometimes exotic valve designs. Hysteresis, dead zone, supply pressure and temperature changes complicate control.	Inherently non-linear, compressible power source severely complicates servo control. Compressibility can be an advantage in open loop operation.
Speed	Smooth, variable speed capabilities from 0.5 to 1330 mm/sec [0.02 to 52.5 in/sec].	Difficult to control accurately. Varies with temperature and wear. Stick slip can be a problem.	More susceptible to stick slip and varying load. Well-suited for high speed applications to 5 m/sec [200 in/sec].
Reliability	Repeatable, reproducible performance throughout useful life of product; little maintenance required.	Very contamination sensitive. Fluid sources require maintenance. Seals are prone to leak. Good reliability with diligent maintenance.	Very contamination sensitive. Air sources require proper filtration. Good reliability, but usually many system components are involved.
Power	Up to 25,000 N [5620 lb], 3kW [4 HP].	Virtually unlimited force. Most powerful.	Up to 5,000 lbs. Typically used below 1 HP.
Cycle Life	Up to millions of cycles at rated load. Easy to predict.	Dependent on design and seal wear; usually good.	Dependent on seal wear, usually good.
Environment	Standard models rated for -20° to 160° F. Inherently clean and energy efficient.	Temperature extremes can be a major problem. Seals are prone to leak. Waste disposal is increasingly problematic.	Temperature extremes can be a major problem. Seals prone to leak. Air-borne oil can be a problem.
Safe Load Holding	Acme screw units are self-locking if power fails. Fail-safe brakes available for ball screw models.	Complex back-up safety devices must be used.	Complex back-up safety devices must be used.
Cost	Moderate initial cost; very low operating cost.	Components often cost less, but installation and maintenance are increased. Hydraulic power unit cost is high if not pre-existing. Most economical above 10 HP.	Components often cost less, but installation and maintenance are increased. Most cost-effective for low power, simple point-to-point applications.

Electric Cylinders vs. Hydraulics & Pneumatics

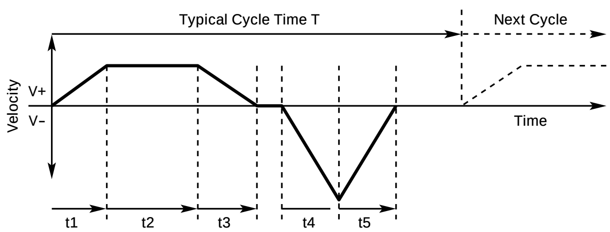
But when simplicity, flexibility, programmability, accuracy and reliability are important and loads are within the capacity of the technology, electromechanical solutions often are the most desirable.

Further, electromechanical systems are inherently more compatible with today's automation controls.

Move Profile

Rotary and linear actuator selection begins with the calculation of speed, thrust and torque requirements. In order to determine the torque required, the acceleration of the mass being moved must be calculated. A **"move profile"**, or a **plot of load velocity vs. time,** is sketched in order to simplify the **peak acceleration** and **peak velocity** calculations.

Typical Machine Cycle



(1) Total distance, $$d_{tot} = v_{MAX}\left[\frac{t_1}{2} + t_2 + \frac{t_3}{2}\right]$$

(2) Max velocity, $$v_{MAX} = \frac{d_{tot}}{\left(\frac{t_1 + t_3}{2}\right) + t_2}$$

(3) Acceleration, $$a = \frac{v_{MAX}}{t_{ACCEL}}$$

The figure above is an example of a typical machine cycle, and is made up of two Move Profiles; the first is an example of **a trapezoidal profile,** while the second is a **triangular profile.** The horizontal axis represents time and the vertical axis represents velocity (linear or rotary). The load accelerates for a time (t_1), has a constant velocity or slew section (t_2), and decelerates to a stop (t_3). There it dwells for a time, accelerates in the negative direction (t_4), and decelerates back to a stop (t_5) without a slew region. The equations needed to calculate Peak Velocity and Acceleration for a general trapezoidal profile are shown in the figure. A triangular profile can be thought of as a trapezoidal profile where $t_2 = 0$.

The Move Profile sketch contains some important information:

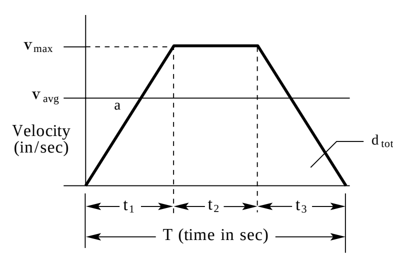
- **Peak acceleration** is the steepest slope on the curve, in this case during t_4 or t_5.
- **Maximum velocity** is at the highest or lowest point over the entire curve, here at the peak between t_4 and t_5.
- **Distance** is equal to the area under the curve. Area above the time axis represents distance covered in the positive direction, while negative distance falls below this axis. The distance equation (1) is just a sum of the areas of two triangles and a rectangle.

Trapezoidal and Triangular Profiles

A couple of assumptions can greatly simplify the general equations. For the Trapezoidal profile we assume $t_1=t_2=t_3$, and for the Triangular we assume $t_3=t_4$. Substituting these assumptions into equations (2) and (3) yields the equations shown in the figure below.

For a given distance (or area), a triangular profile requires lower acceleration than the trapezoidal profile. This results in a lower thrust requirement, and in turn, a smaller motor. On the other hand, the triangular profile's peak speed is greater than the trapezoidal, so for applications where the motor speed is a limiting factor, a trapezoidal profile is usually a better choice.

Trapezoidal Move Profile

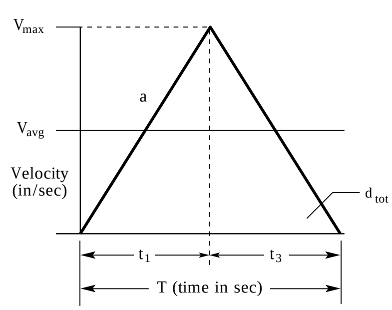


$$v_{AVE} = \frac{d_{tot}}{t_{tot}}$$

$$t_1 = t_2 = t_3 = \frac{t_{tot}}{3}$$

$$\boxed{v_{MAX} = 1.5\frac{d_{tot}}{t_{tot}} = 1.5\,v_{AVE} \qquad a = 4.5\frac{d_{tot}}{(t_{tot})^2}}$$

Triangular Move Profile



$$v_{AVE} = \frac{d_{tot}}{t_{tot}}$$

$$t_1 = t_3 = \frac{t_{tot}}{2} \qquad t_2 = 0$$

$$\boxed{v_{MAX} = \frac{2d_{tot}}{t_{tot}} = 2\,v_{AVE} \qquad a = \frac{4d_{tot}}{(t_{tot})^2} = \frac{2v_{MAX}}{t_{tot}}}$$

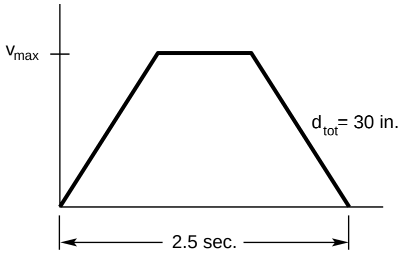
Engineering

Move Profile

Example 1

Calculate the peak acceleration and velocity for an object that needs to move 30 inches in 2.5 seconds. Assume a Trapezoidal Profile.

Solution



$$v_{AVE} = \frac{30\text{ in}}{2.5\text{ sec}} = 12\text{ in/sec}$$

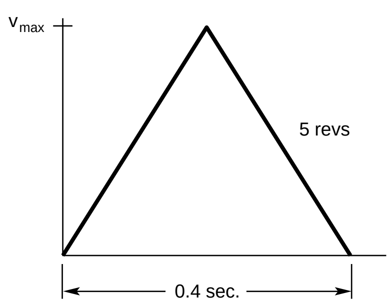
$$v_{MAX} = 1.5\,\frac{d_{tot}}{t_{tot}} = 18\text{ in/sec}$$

$$a = 4.5\,\frac{d_{tot}}{(t_{tot})^2} = 21.6\text{ in/sec}^2$$

Example 2

Calculate, in radians/sec, the peak acceleration and velocity for an cylinder that needs to move 5 revolutions in 0.4 seconds. Assume a Triangular Profile.

Solution



$$d_{tot} = 5\text{ revs} \times \frac{2\pi\text{ rad}}{\text{rev}} = 31.42\text{ rad}$$

$$v_{AVE} = \frac{31.42\text{ rad in}}{0.4\text{ sec}} = 78.55\,\frac{\text{rad}}{\text{sec}}$$

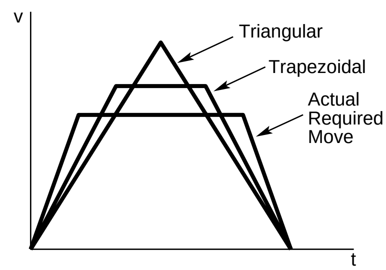
$$v_{MAX} = 2\,v_{AVE} = 157.1\,\frac{\text{rad}}{\text{sec}}$$

$$a = 4\,\frac{d_{tot}}{T^2} = 785.5\,\frac{\text{rad}}{\text{sec}^2}$$

Example 3

This is an example of a case when triangular and trapezoidal move profiles are not adequate approximations. Assume a maximum actuator speed is 6 inches/sec. Sketch a move profile that will complete a 10 inch move in 2 seconds. What is the minimum allowable acceleration rate in inches/sec^2?

Solution



Triangular

$$v_{AVE} = \frac{10\text{ in}}{2\text{ sec}} = 5\text{ in/sec}$$

$v_{MAX} = 2 \times v_{AVE} = 10$ in/sec ($v_{MAX} > 6$ in/sec – too fast)

Trapezoidal

$v_{MAX} = 1.5 \times v_{AVE} = 7.5$ in/sec ($v_{MAX} > 6$ in/sec – too fast)

These are too fast, so we need to find t_1 as follows:

Required Profile

$$d_{tot} = v_{MAX}\left(\frac{(t_1 + t_3)}{2} + t_2\right)$$

substitute $(t_1 + t_3) = t_{tot} - t_2$

$$\frac{d}{v_{MAX}} = \left(\frac{(t_{tot} - t_2)}{2}\right) + t_2 = \frac{t_{tot}}{2} + \frac{t_2}{2}$$

solving for t_2,

$$t_2 = \left(\frac{d_{tot}}{v_{MAX}} - \frac{t_{tot}}{2}\right) \times 2 = \left(\frac{10\text{ in}}{6\text{ in/sec}} - \frac{2\text{ sec}}{2}\right) \times 2$$

$$t_2 = 1.33\text{ sec}$$

Now assume $t_1 = t_3$, so

$t_1 = (t_{tot} - t_2)/2 = 0.33$ sec.

Finally, calculate acceleration

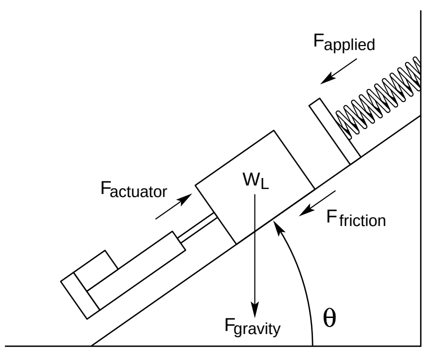
$$a = \frac{v_{MAX}}{t_1} = \frac{6\text{ in/sec}}{0.33\text{ sec}} = 18\,\frac{\text{in}}{\text{sec}^2}$$

Thrust Calculation

Thrust Calculation

The thrust required to move a mass a given distance within a given time may be calculated by summing all of the forces that act on the mass. These forces generally fall within the following four categories:

- **Gravity** is important when something is being raised or lowered in a system. Lifting a mass vertically is one example, as is sliding something on an incline.
- **Friction forces** exist in almost all systems and must be considered.
- **Applied forces** come from springs, other actuators, magnets, etc., and are the forces that act on the mass other than friction, gravity, and the actuator's thrust. The spring shown in the figure below is an example of an Applied force.
- **Actuator thrust** is the required actuator force, and is what we need to determine.



The figure above shows a general case where the force required by an actuator must be determined. All of the above forces are included, and it is important to note that all of these forces can change over time, so the thrust must be calculated for each section of the move profile. The worst case thrust and speed required should be used to pick the appropriate actuator. All of these forces added up (Σ) must be equal to mass × acceleration, or:

$$\Sigma F = m \times a \quad \text{, or,} \qquad (1)$$

$$F_{actuator} - F_{applied} - F_{friction} - F_{gravity} = ma = \left(\frac{W_t}{g}\right) a \qquad (2)$$

$$\boxed{F_{actuator} = \left(\frac{W_t}{g}\right) a + F_{applied} + F_{friction} + F_{gravity}} \qquad (3)$$

where $W_t = W_{load} + W_{actuator}$ (4)

$F_{friction} = \mu W_L \cos\theta$, and

$F_{gravity} = W_L \sin\theta$

$W_{actuator}$ becomes important when the acceleration force, $(W_t/g)a$, is a significant part of the thrust calculation. For simplicity, start by neglecting this weight, and calculate the required thrust without it. After selecting an actuator, add its mass to the mass of the load and recalculate. To make these equations clear, lets begin with an example.

Example 1

We would like to move a 200 lb weight a distance of 10 inches in 2 seconds. The mass slides up and incline with a friction coefficient of 0.1 at an angle of 45°. There is a spring that will be in contact with the mass during the last 0.5 inch of travel and has a spring rate of 100 lb/in. What is the maximum thrust and velocity?

Solution

We need to look at the thrust requirement during each part of the move, and find the points of maximum thrust and maximum speed. Choosing a trapezoidal profile we calculate that v_{max} is 7.5 in/sec and the peak acceleration is 11.25 in/sec² (see Move Profile Section).

Acceleration Section:

Ma = 200 lb/386 in/sec² × 11.25 in/sec²
= 5.83 lb

$F_{applied}$ = 0 lb
$F_{friction}$ = [200 lb × cos (45)] × 0.1 = 14.14 lb
$F_{gravity}$ = 200 lb × sin (45) = 141.4 lb
F_{total} = 161 lb

Slew Section:

Ma = 0 lb (since a=0)
$F_{applied}$ = 0 lb
$F_{friction}$ = [200 lb × cos (45)] × 0.1 = 14.14 lb
$F_{gravity}$ = 200 lb × sin (45) = 141.4 lb
F_{total} = 156 lb

Deceleration Section:

Ma = 200 lb/386 in/sec² × -11.25 in/sec²
= -5.83 lb
$F_{applied}$ = K × x = 0.5 in × 100 lb/in =50 lb (worst case)
$F_{friction}$ = [200 lb × cos (45)] × 0.1 = 14.14 lb
$F_{gravity}$ = 200 lb × sin (45) = 141.4 lb
F_{total} = 200 lb

So the worst case required thrust is <u>200 lb.</u> And the worst case velocity is <u>7.5 in/sec.</u>

Thrust Calculation

Actuator Mass

In applications where the **acceleration force**, $(W_t/g)a$, is a significant part of the required thrust, the actuator mass must be considered in the thrust calculation. After an actuator is chosen, the actuator weight (linear inertia), $W_{actuator}$, is added to the weight of the load. $W_{actuator}$ can be determined using the tables and equation in the actuator data section. To illustrate, we will use the previous example.

1. The first step is to pick an actuator with the above thrust and speed capability. One such actuator is an EC3-B32-20-16B-12. This is an EC3 actuator with a B32 motor, a 2:1 gear reduction, a 16mm lead ballscrew, and a 12 inch stroke.
2. The next step is to look up the effective **Actuator Linear Inertia** in the tables located in the particular actuator section (do not include the "load" term in the equation). An entry from this table can be seen in the table below. The B32 motor inertia is 1.00 x 10^{-3} in-lb-sec^2. The effective actuator weight, calculated from the table is 247 lb.
3. The final step is to add this weight to the weight of the load, W_L, and recalculate the peak thrust required for each section of the move profile **(do not add this weight to the gravity or friction terms)**:

Acceleration Section:

Ma = 447 lb/386 in/sec^2 × 11.25 in/sec^2

= 13.03 lb

F_{total} **= 169 lb**

Slew Section:

Ma = **0 lb** (since a=0)

Deceleration Section:

Ma = 447 lb/386 in/sec^2 × -11.25 in/sec^2)

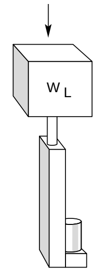
= -13.03 lb

F_{total} **= 193 lb**

We can see from this calculation that the addition of this extra "acceleration weight" increases the thrust required during acceleration, but reduces the peak thrust required during deceleration. The EC3-B32-20-16B-12 will work in the application.

Vertical and Horizontal Cases

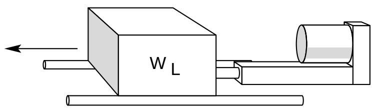
In a vertical system, θ is 90°, sin90 = 1, and $F_{gravity}$ is equal to W_L. Since cos90 = 0, $F_{friction}$ = 0.



$$F_{actuator} = (W_t/g)a + F_{applied} + F_{gravity}$$

$$\boxed{F_{actuator} = (W_t/g)a + F_{applied} + W_L}$$

In a horizontal system, sinθ = 0, so gravity would play no part ($F_{gravity}$ = 0), and cosθ =1, so $F_{friction}$ would be equal to μW_L, or 50 lb.

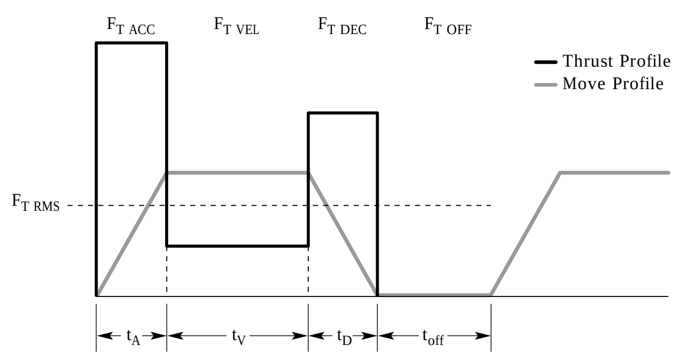


$$F_{actuator} = (W_t/g)a + F_{applied} + F_{friction}$$

$$\boxed{F_{actuator} = (W_t/g)a + F_{applied} + \mu W_L}$$

RMS Thrust

For all Servo Motor applications, the RMS Thrust needs to be calculated. This thrust must fall within the continuous duty region of the actuator. Use the following equation when calculating RMS Thrust:



$$F_{T\,RMS} = \sqrt{\frac{(F_{T\,ACC})^2 t_a + (F_{T\,VEL})^2 t_v + (F_{T\,DEC})^2 t_d + (F_{T\,OFF})^2 t_{off}}{t_a + t_v + t_d + t_{off}}}$$

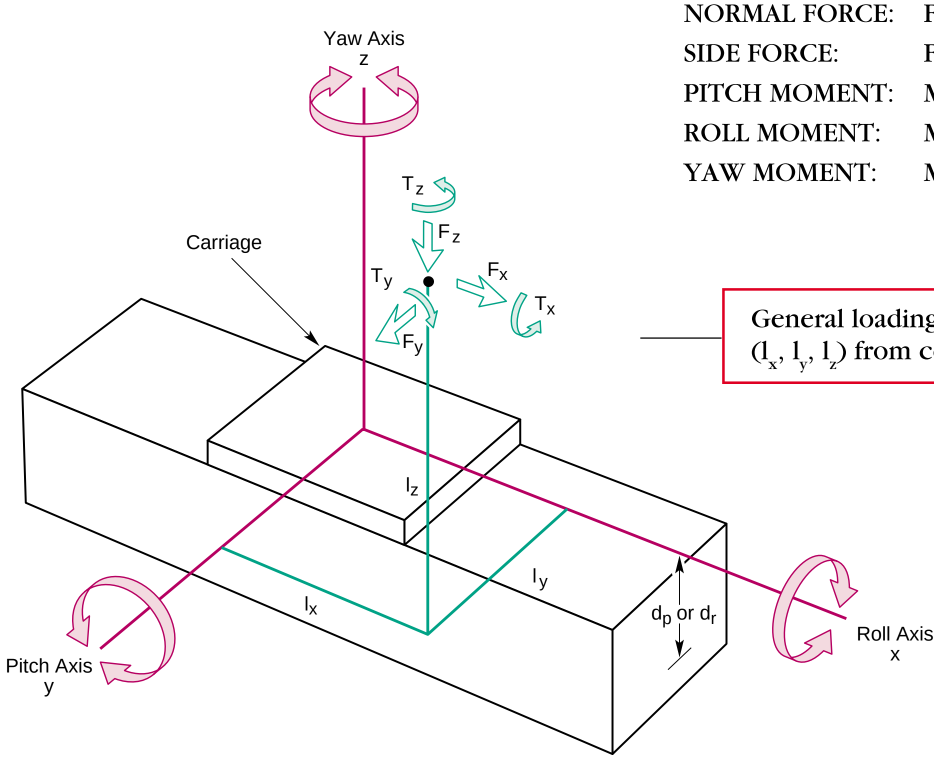
EC3 Cylinder — Actuator Linear Inertia = [A + B*(stroke, in) + D]/C

Leadscrew	Ratio	A (lb-in-s²)	B (lb-in-s²/in)	C (lb-in-s²/lb)
16x16	1:1	1.1909E-03	1.1760E-05	2.6039E-05
	1.5:1	7.4495E-04	5.2280E-06	1.1573E-05
	2:1	4.7866E-04	2.7650E-06	6.1212E-06

Motor Inertia (in-lb-sec²)

	D (lb-in-s²)
B32	1.00 E-03

Loading Limitations



NORMAL FORCE: $F_n = F_z$

SIDE FORCE: $F_s = F_y$

PITCH MOMENT: $M_p = F_z (l_x) + F_x (l_z + d_p) + T_y$

ROLL MOMENT: $M_r = F_z (l_y) + F_y (l_z + d_r) + T_x$

YAW MOMENT: $M_y = F_x (l_y) - F_y (l_x) + T_z$

General loading condition applied at point (l_x, l_y, l_z) from center of the carriage.

Rodless actuators have bearings designed to support significant loads, however actuator loading is limited, and **moment, normal, and side loads** on the rodless actuator and linear motor carriages need to be evaluated.

Moment loads are rotational forces (or torques) which are applied to the carriage assembly. They are defined as yaw, pitch, and roll and are measured in in-lb.

To evaluate the actuator you have chosen, use the diagram above. The three forces (F) and the three torques (T) are a general loading condition applied to the load a distance l_x, l_y, and l_z from the centerpoint of the carriage. These forces may be due to gravity, friction, applied loads, and actuator thrust. Each of these forces may act at different points of application. For example, gravity will act at the center of gravity of the load, while friction and applied loads will act at the edge of the load.

The equations for pitch, roll, and yaw assume that the directions of the arrows in the figure are POSITIVE. If the forces or torques in your application work in the opposite direction, they are NEGATIVE and they should be entered as negative numbers in the formulas. The same is true for the distances l_x, l_y, and l_z.

It is important to add the carriage to bearing distances, d_p (pitch) and d_r (roll), in the roll and pitch moment calculations above. These distances vary for each actuator and are listed in the table below.

Examples

To illustrate these calculations, refer to the facing page which shows six typical rodless actuator orientations and the equations for the calculation of moment, normal and side loads in each case.

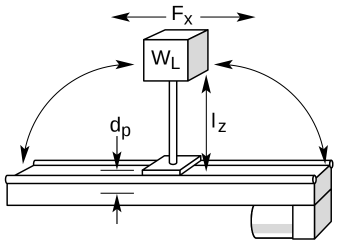
Table of Added Moment Distances (inches)

	R2A Series		R3 Series		R4 Series		Linear Motors	
	Screw	Belt	Screw	Belt	Screw	Belt	LM25	LM38
d_p - pitch	1.822	1.575	1.68	1.10	2.42	1.90	2.60	3.23
d_r - roll	1.11	1.11	2.76	1.97	3.94	3.06	2.60	3.23

Moment Loading

Load and Cylinder Orientation

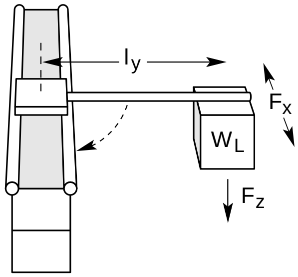
Note: THRUST = $F_{actuator}$ as explained on page K-12, equation (3)



$F_n = F_z = W_L$

$M_p = F_x\,(l_z + d_p)$

F_x = THRUST



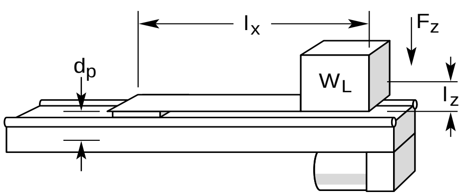
$F_n = F_z = W_L$

$M_p = F_x\,(l_z + d_p)$

$M_r = F_z\,(l_y)$

$M_y = F_x\,(l_y)$

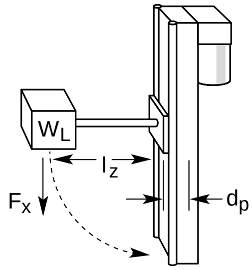
F_x = THRUST



$F_n = F_z = W_L$

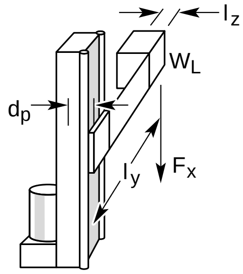
$M_p = F_z\,(l_x) + F_x\,(l_z + d_p)$

F_x = THRUST



$M_p = F_x\,(l_z + d_p)$

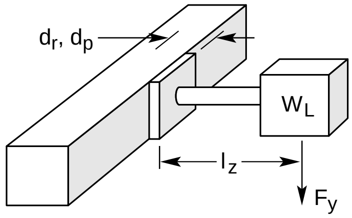
F_x = THRUST



$M_p = F_x\,(l_z + d_p)$

$M_y = F_x\,(l_y)$

F_x = THRUST



$F_y = W_L$

$M_p = F_x\,(l_z + d_p)$

$M_r = F_y\,(l_z + d_r)$

F_x = THRUST

Loading Limitations

Dual Carriage Option

The a dual carriage option is available to increase pitch, roll and yaw loading capability. This is generally needed in vertical applications with heavy loads. To get the most benefit from the extra carriage, the load should be distributed so that each carriage sees the same force. The Dual Carriage option is available for both belt and screw driven actuators. Follow the example in the figure to calculate the required distance between the two carriages with a given loading configuration. This distance must be added to the stroke length of the actuator. The option is specified as in the example below:

R4B32-104B-96-PL-ADE

Dual Carriage Option

Note: The second carriage does not increase the thrust load capability.

Idler

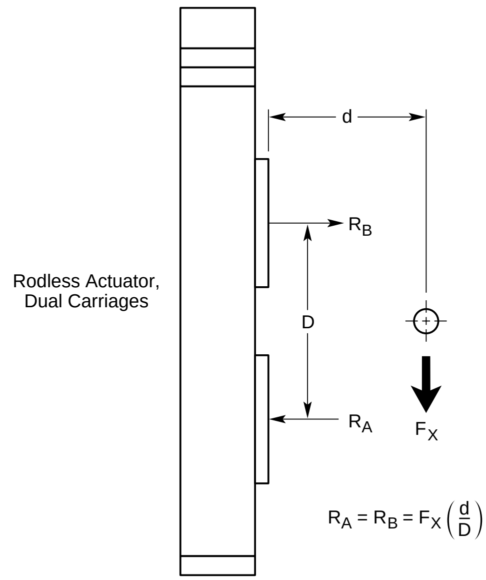
An idler is a rodless actuator with its power transmission parts removed and can be used in conjunction with a rodless actuator when roll, pitch or yaw moments are exceeded. It is dimensionally similar in an R2, R3, or R4 actuator except it has no motor, belt or, screw. An idler is specified:

Rx-IDLER-Stoke Length (in)-ASE

For example, an R3-IDLER-48-ASE is a 48-inch stroke R3 idler with angle brackets, a single carrige and English mounting dimensions. All idlers have the same load limitations as their actuator counterparts.

Customer Options

By changing the system design, and taking the loading off of the actuator, sometimes a smaller actuator than once thought can be used. **Counterbalancing** removes the need for the actuator to do work against gravity. **External rail support** removes moment loads that are too great for the actuator. This can be done with an IDC Idler, or any number of off-the-shelf linear rails and bearings.



$$\boxed{R_A = -R_B = F_x\left(\frac{d}{D}\right)}$$

Example:

To support a 300 lb load located 10 inches away from the carriage of an R4 actuator, how far apart should the centers of the carriages be placed if we know the maximum allowable normal load is 300 lb on each carriage?

It is assumed there is a rigid bracket between the load and the carriages (weight of the bracket is included in the 300 lb) and that there are no moment loads transmitted to the carriage.

Solution:

It is a good idea to leave some safety margin so we will use 150 lb as the maximum Normal Load on the R4 carriage (actual normal load is 300 lb).

$R_A = 150$ lb

$D = ?$

$d = 10$ in

$F_x = 300$ lb

$$R_A = R_B + F_x\left(\frac{d}{D}\right)$$

$$D = \frac{F_x\,(d)}{R_A} = \frac{300\text{ lb }(10\text{ in})}{150\text{ lb}}$$

$$\boxed{D = 20\text{ inches}}$$

Engineering

Duty Cycle

Duty Cycle is the ratio of motor-on time to total cycle time and is used to determine the acceptable level of running time so that the thermal limits of the motor or actuator components are not exceeded. Inefficiencies cause a temperature rise in a system, and when the temperature reaches a critical point, components fail. Letting the system to rest idle during the cycle allows these system components to cool. Duty cycle is limited by Acme screw and motor thermal limits. Use the following equation and example to determine Duty cycle:

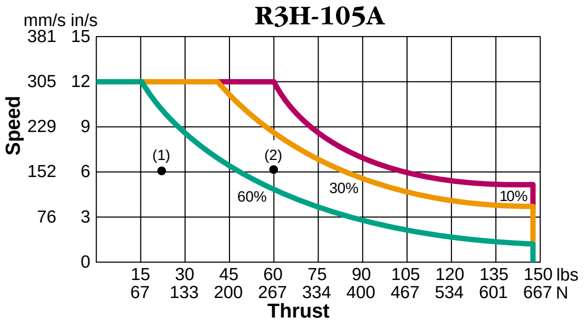
$$\text{Duty Cycle} = \frac{\text{ON TIME}}{\text{ON TIME} + \text{OFF TIME}} \times 100$$

Leadscrew Limitations

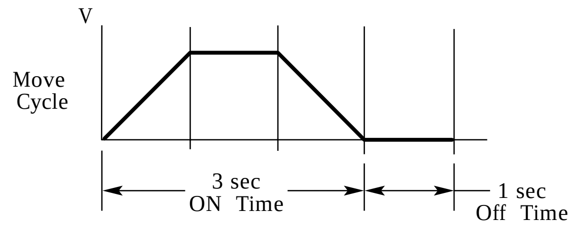
Cylinders with **Acme screws** have sliding friction surfaces and are limited to a maximum 60% duty cycle regardless of motor capability. The friction in the Acme screw causes rapid heating, and continuous operation is likely to end in a ruined nut or screw. For actuators with **ballscrews** the motor is the only duty-cycle limitation when used within the listed speed vs. thrust curves in the catalog.

Acme screw limitations are shown in the catalog speed thrust curves as a set of percentages. An example of an R3H-105A curve is below. If we wanted to run this actuator at 6 in/sec and 25 lb thrust, it would be limited to 60% Duty Cycle over any 10 minute period (1). If we wanted to increase the thrust to 60 lb, this would drop the duty cycle rating to 30% (2).

Acme screw duty cycle is a function of Pressure and Velocity, so operation at very low speeds or thrusts may allow duty cycles above 60%. If an Acme screw is required and Duty Cycle requirements exceed 60%, please call the factory for a custom evaluation. If power-off position holding is required, consider using a brake on a ballscrew or on a motor.



Example:



$$\text{Duty Cycle} = \frac{3 \text{ sec}}{3 + 1 \text{ sec}} \times 100$$

$$\boxed{= 75\%}$$

Motor Type

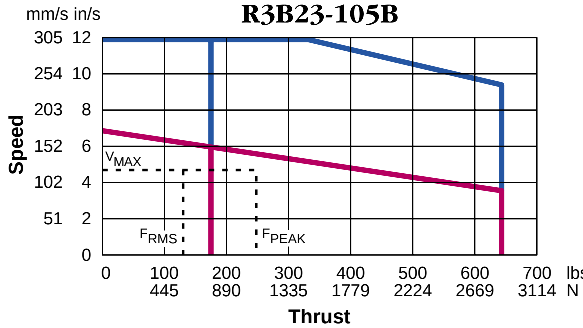
Electric motors incur heat losses via a number of paths, namely, friction, ohmic (I^2R) losses in copper windings, hysteresis and eddy current induction in magnetic core materials, and proximity and/or skin effect in windings. As a result duty cycle can be limited by the motor winding temperature limitations.

Step Motors

Series wound motors (S23T, S32T, S42T, etc.) are rated for 100% duty cycle, while the parallel wound motors (S23V, S32V, S42V, etc.) are limited to 50% above 5 RPS. Duty Cycle percentages for Stepper motor actuators are listed as percentages on the speed thrust curve.

Servo Motors

Actuators using D, H and B Series motors must have their peak (F_{peak}) and continuous (F_{RMS}) thrust requirements determined to establish their safe operation within an application. F_{RMS} can be determined using the RMS Thrust equation in the Thrust Calculation section. Plotting F_{RMS} on the actuator Speed vs. Thrust curve indicates the allowable Duty Cycle. For ball screw actuators, F_{RMS} must fall within the continuous duty region, while for Acme screws it must fall in the 60% duty region. F_{peak} must fall within remaining operating envelope. The speed vs. thrust curve below is an example of proper servo actuator sizing.



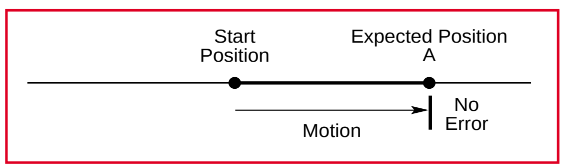
Actuator Precision

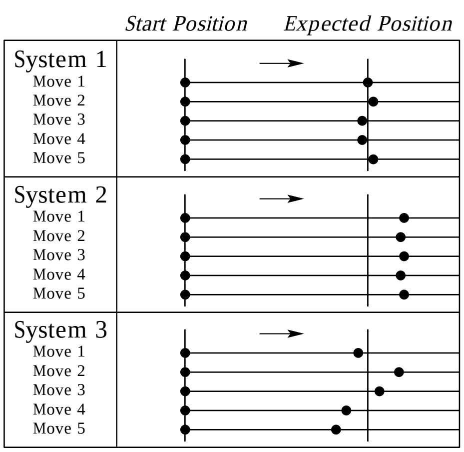
Parameter	Definition	Dominating Factors
Absolute Accuracy	The maximum error between expected and actual position.	• Accuracy of the motor/drive system • Leadscrew pitch error (lead accuracy) • System backlash (drive train, leadscrew and nut assembly)
Repeatability	The ability of a positioning system to return to a location during operation when approaching from the same direction, *at the same speed and deceleration rate.*	• Angular repeatability of the motor/ drive system • System friction • Changes in load, speed, and deceleration
Resolution	The smallest positioning increment achievable. *In digital control systems, resolution is the smallest specifiable position increment.*	• Angular resolution of the motor/drive system • Drive Train Reduction • Leadscrew Pitch
Backlash	The amount of play (lost motion) between a set of moveable parts.	• Leadscrew wear (acme) • Drive train wear • Spaces between moving parts

Accuracy and Repeatability

Assume three linear positioning systems each attempt five moves from an absolute zero position to absolute position "A". The individual end positions of each move are charted on a linear scale below to demonstrate their accuracy and repeatability by displaying their proximities to the expected position.

Ideal System





Degree of Accuracy	Degree of Repeatability	Comment
High	High	System 1 is both accurate and repeatable, the end positions are tightly grouped together and are close to the expected position
Low	High	System 2 is inaccurate but repeatable, the end positions are tightly grouped around a point but are not close to the expected position.
Low	Low	System 3 is neither accurate nor repeatable, the end positions are not tightly grouped and are not close to the expected position.

Actuator Precision

Backlash

The clearance between elements in a drive train or leadscrew assembly which produces a mechanical "dead band" or "dead space" when changing directions, is known as the **backlash** in a system.

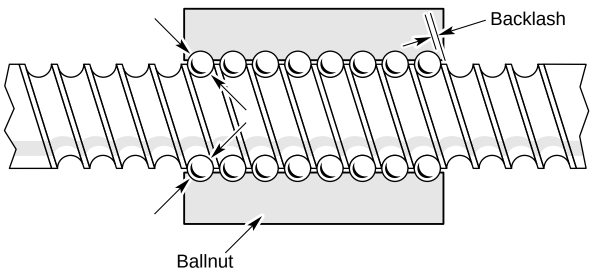
In most mechanical systems, some degree of backlash is necessary to reduce friction and wear. In an IDC Actuator System, system backlash will typically be 0.010 - 0.015 inches. Usually 0.006 - 0.008" is attributed to the leadscrew/nut assembly. For ballscrews this will remain constant throughout the life of a cylinder, while for acme screws it will increase with wear.

Reducing the Effects of Backlash

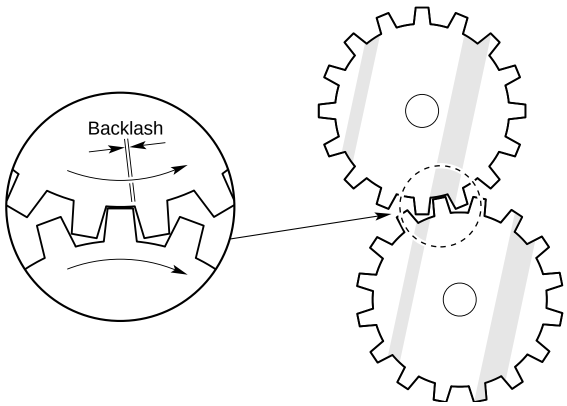
1. Approach a stop position from the same direction.
2. Apply a constant linear force on the cylinder thrust tube or carriage. This is done automatically for cylinders used in vertical orientations with a backdriving load.
3. For programmable positioning devices it is possible to program out backlash by specifying a small incremental move (enough to take out the backlash) prior to making your normal moves in a particular direction.
4. Use a preloaded nut on a leadscrew to counteract the backlash. *Contact IDC about the precision ground screw option which reduces backlash in the drive nut.*
5. An inline actuator with the motor directly coupled to the leadscrew has less backlash than parallel or reverse parallel units which utilize a gear train or drive belt/pulley.
6. Use an IDC LM or LD Linear Motor Module.

Primary Sources of Backlash

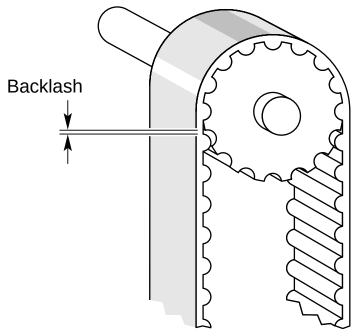
1. Drive Nut/Leadscrew Assembly



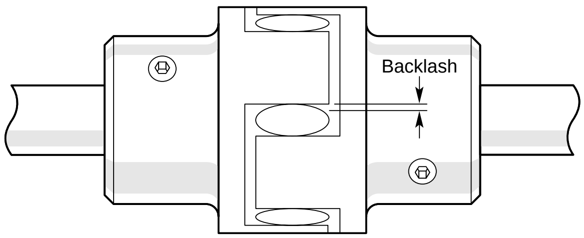
2. Drive Train (Gears, Timing Belt/Pulley)



3. Timing Belt/Pulley



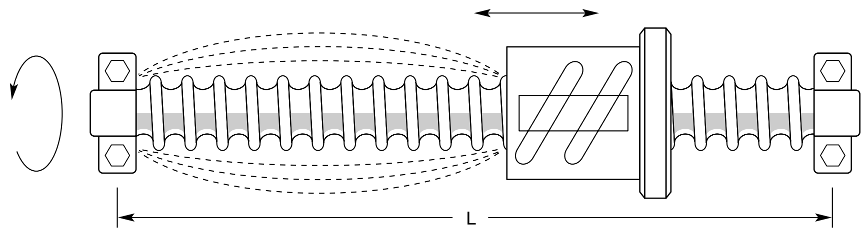
4. Coupling

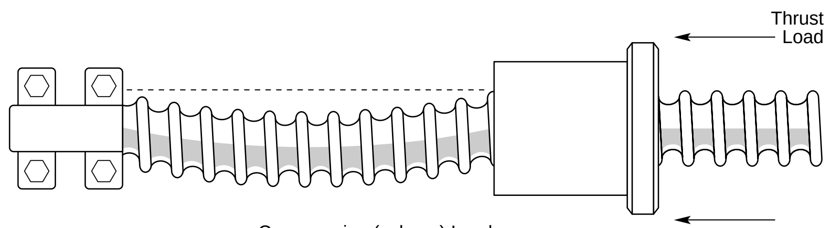


Critical Speed and Column Loading

Critical Speed

All leadscrew systems have a rotational speed limit where harmonic vibrations occur. In Industrial Devices cylinders, this limit is a function of unsupported leadscrew length. Operation beyond this critical speed will cause the leadscrew to vibrate (whip violently) eventually bending or warping the screw.

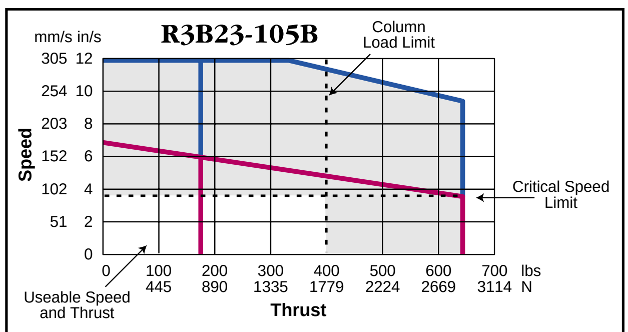




Compression (column) Load

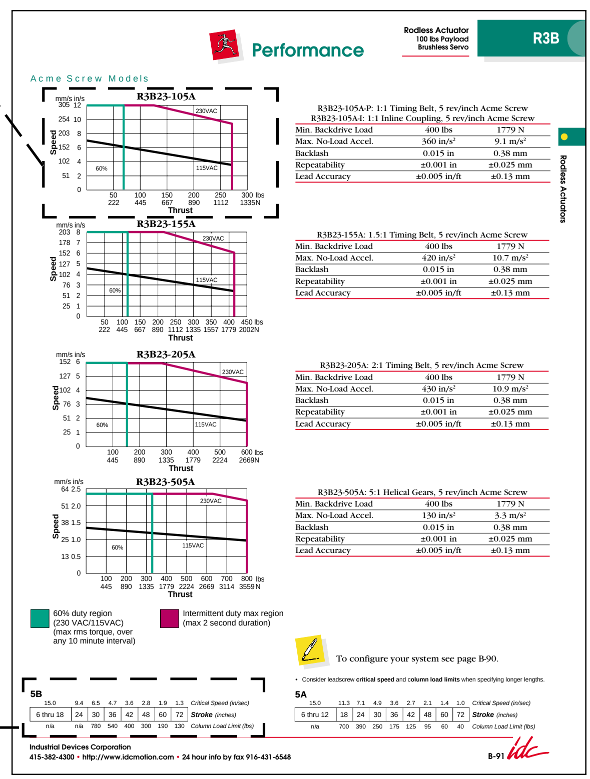
Column Strength

All leadscrews have a maximum column loading limit which causes the screw to compress as load increases. In IDC cylinders this limit is a function of unsupported leadscrew length. Exceeding this limit will cause the leadscrew to buckle and become permanently damaged.



Determining the Limits

Critical Speed and Column Loading information for each screw type (i.e. 2B, 5A, 8A, 5B …) can be found at the bottom of each "Performance" page in the particular actuator's section.



Performance

Rodless Actuator
100 lbs Payload
Brushless Servo

R3B

Acme Screw Models

R3B23-105A-P: 1:1 Timing Belt, 5 rev/inch Acme Screw
R3B23-105A-I: 1:1 Inline Coupling, 5 rev/inch Acme Screw

Min. Backdrive Load	400 lbs	1779 N
Max. No-Load Accel.	360 in/s²	9.1 m/s²
Backlash	0.015 in	0.38 mm
Repeatability	±0.001 in	±0.025 mm
Lead Accuracy	±0.005 in/ft	±0.13 mm

R3B23-155A: 1.5:1 Timing Belt, 5 rev/inch Acme Screw

Min. Backdrive Load	400 lbs	1779 N
Max. No-Load Accel.	420 in/s²	10.7 m/s²
Backlash	0.015 in	0.38 mm
Repeatability	±0.001 in	±0.025 mm
Lead Accuracy	±0.005 in/ft	±0.13 mm

R3B23-205A: 2:1 Timing Belt, 5 rev/inch Acme Screw

Min. Backdrive Load	400 lbs	1779 N
Max. No-Load Accel.	430 in/s²	10.9 m/s²
Backlash	0.015 in	0.38 mm
Repeatability	±0.001 in	±0.025 mm
Lead Accuracy	±0.005 in/ft	±0.13 mm

R3B23-505A: 5:1 Helical Gears, 5 rev/inch Acme Screw

Min. Backdrive Load	400 lbs	1779 N
Max. No-Load Accel.	130 in/s²	3.3 m/s²
Backlash	0.015 in	0.38 mm
Repeatability	±0.001 in	±0.025 mm
Lead Accuracy	±0.005 in/ft	±0.13 mm

60% duty region (230 VAC/115VAC) (max rms torque, over any 10 minute interval)

Intermittent duty max region (max 2 second duration)

To configure your system see page B-90.

• Consider leadscrew **critical speed** and **column load limits** when specifying longer lengths.

5A

15.0	11.3	7.1	4.9	3.6	2.7	2.1	1.4	1.0	*Critical Speed (in/sec)*
6 thru 12	18	24	30	36	42	48	60	72	***Stroke*** *(inches)*
n/a	700	390	250	175	125	95	60	40	*Column Load Limit (lbs)*

Industrial Devices Corporation
415-382-4300 • http://www.idcmotion.com • 24 hour info by fax 916-431-6548

B-91

Example

Find the Column Load and Critical Speed limits for an R3B23-105B, 42 inch stroke actuator. The actuator data can be found in the R3B Section. Reading off the chart at the bottom of page, the limits are 400 lb and 3.6 in/sec. The usable speed/ thrust is restricted to less than these values as seen in the modified speed vs. thrust curve.

5B

15.0	9.4	6.5	4.7	3.6	2.8	1.9	1.3	*Critical Speed (in/sec)*
6 thru 18	24	30	36	42	48	60	72	***Stroke*** *(inches)*
n/a	n/a	780	540	400	300	190	130	*Column Load Limit (lbs)*

Environment

Environmental conditions are an important design consideration when selecting an IDC actuator. Industrial Devices' units are self-contained systems which are protected from "direct contact" with harsh environments by an aluminum housing with a durable anodized and epoxy coated surface finish. However, extreme conditions can have an adverse effect on cylinder operation and life. Factors such as extreme temperature, liquids or abrasive contaminants (gaining internal access) can impede performance and cause premature wear of mechanical parts. Review the information below when sizing your application to choose appropriate options or protective measures.

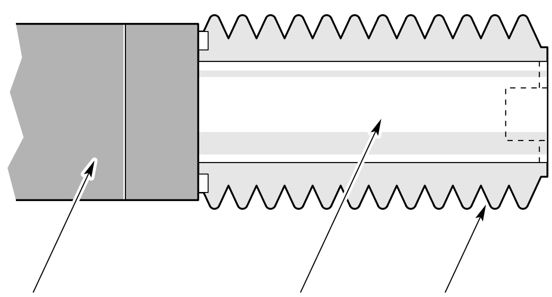
Primary Environmental Factors

- Temperature
- Liquid contaminants
- Particle contaminants

Rod Type	Rodless
Temperature	
• NV actuators are rated for use between 0 and 60°C (32 to 140°F). • EC actuators are rated for use between -30 and 70°C (-22 to 158°F)	• R-series rodless actuators are rated for use between -28°C and 60°C (-20°F to 140°F).
Particle and Liquid Contaminants	
• NV series actuators are protected against dust but are not protected against direct water (or any liquid) contact. Liquid or moisture can gain access into the housing, eventually corroding internal components. • EC actuators are sealed and gasketed and are rated to IP 54. They are protected against dust and light water sprays and splashing.	• Rodless actuators are protected by a flexible stainless-steel-backed flouro-silicon seal. They are IP44 rated, which protects the actuator where minimal liquid splash may occur. Units require a customer supplied bellows when placed in an environment containing liquid spray or washdown.

Protective Boot Option

The -PB option is available for EC actuators and increases the actuator's resistance to liquids. The diagram to the right shows a typical installation of an EC with the -PB option. This option protects the actuator to IP65. Note that some IDC motors are not protected to this level. The -PB option is not available with NV or R-series Rodless actuators.

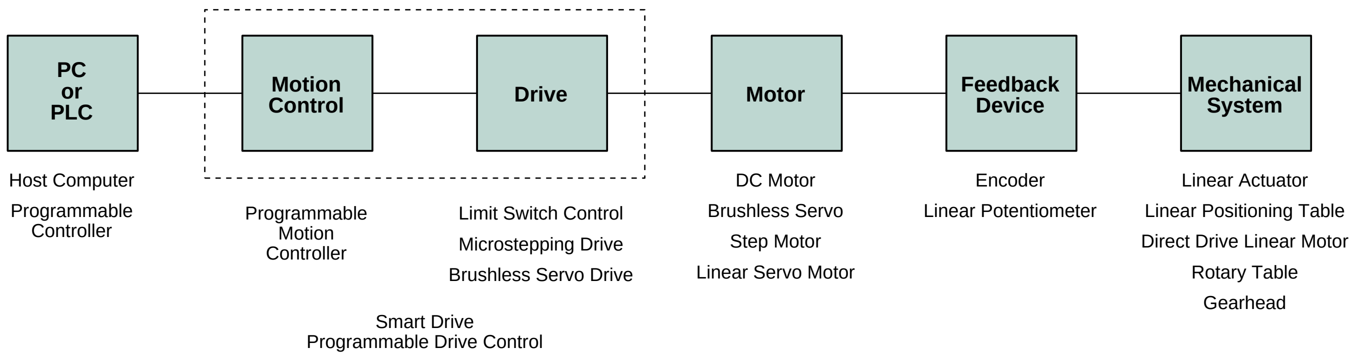


Custom Environmental Options

IDC has over 20 years experience designing custom actuators. We have designed fully encapsulated actuators for Corrosive, Food Processing, and Washdown environments, and have experience designing Cleanroom compatible actuators. Call IDC for more information regarding Custom environmental options.

Introduction to Motion Control Technology

Many different components are used in a variety of combinations to create a complete motion control or positioning system. IDC offers the broadest range of products spanning the complete spectrum from mechanical actuators to microstepping and brushless servo drives to programmable motion controllers. A successful application depends on choosing the right combination of actuator, motor, drive, and control technology. More than one technology may meet the requirements of your application. In this case, factors such as performance, cost, flexibility, and simplicity may determine your selection.

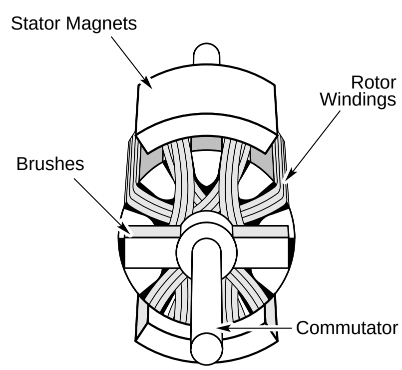


Introduction to Motor Technologies

DC Motor Systems

DC motors are used with IDC's DC motor controls to control velocity and position. With these simple controls, limit switches attached to the actuator or the customer's load provide position feedback. DC motors are also used with analog position controls and a linear potentiometer for position feedback. The result is an absolute positioning system. They can also be used in closed loop servo systems with an incremental encoder for position feedback.

DC motors have windings on the rotor of the motor. To supply current to the rotor, a set of brushes ride on a segmented commutator, which supplies current to the appropriate internal windings depending on the position of the rotor. Brushes may need to be replaced periodically, depending on the load, speed and duty cycle. The commutation method limits the top speed that can be reached before arcing over the commutator segments occurs. The heat generated in the windings must travel across the air gap and the through stator to be dissipated. This long thermal path results in a motor that is less thermally efficient and is therefore larger than a brushless motor with an equivalent power rating. The windings also add inertia to the rotor, which results in lower peak acceleration rates than a similar brushless motor.



Common Applications

- Air cylinder replacement
- Clamping, pressing
- Transfer mechanisms

Advantages

- Lower cost than brushless
- Simple controls

Disadvantages

- High maintenance (brushes).
- Larger motor
- Less responsive

Step Motor Systems

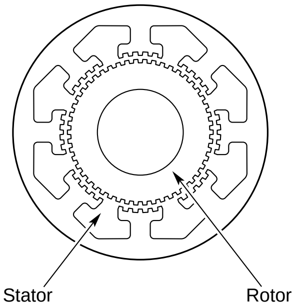
Step motors have the inherent ability to divide a revolution into discrete steps and to hold position at rest without the need for a feedback device. Electromagnets in the stator are energized and cause teeth in the rotor to line up with teeth in the stator. Sequencing the current in the windings causes motion. The rotor will "follow" the stator as if attached by a spring, as long as the available torque from the motor is not exceeded.

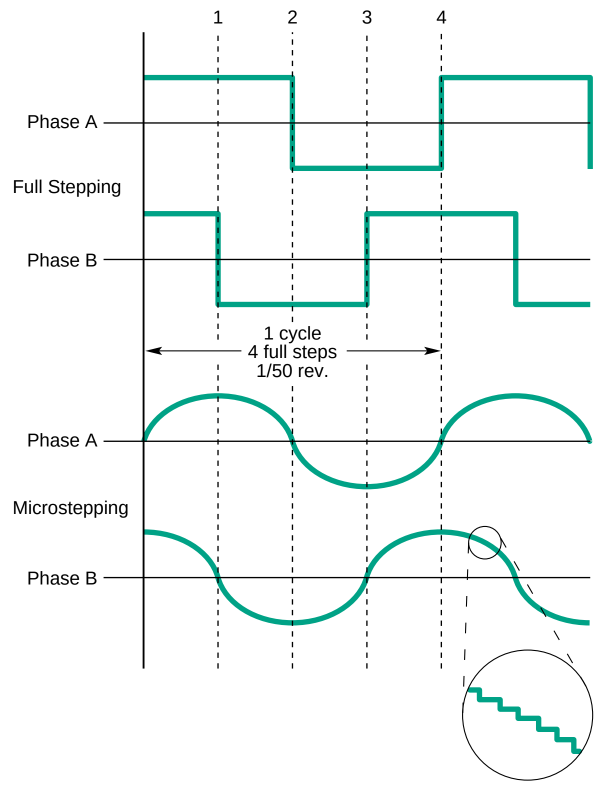
A typical "full-step" system will achieve 200 steps per revolution with a 50-tooth hybrid step motor by following a simple four-state sequence of positive and negative winding currents. After each four steps an adjacent rotor tooth will be lined up with a given stator tooth. Fifty cycles through the sequence results in a complete revolution, with the original rotor and stator teeth lined up. The direction you proceed through the sequence determines the motor direction.

Microstepping is a technique that increases resolution by controlling both the direction and amplitude of current in each winding. Instead of a square wave with four steps, we sequence through what looks like a sine and cosine waveform. The cycle is divided into as many as 1000 steps. The result is a positioning resolution of 50,000 steps or more per revolution. Microstepping improves low speed smoothness and minimizes the low speed resonance effects common in full step systems.

IDC microstepping drives employ unique anti-resonance circuitry which damps oscillations and improves performance and throughput.

Step motors can operate open loop in most applications and provide excellent repeatability (±5 arc sec, ± 0.0005" with actuators, submicron with precision tables). An encoder can be used if closed loop operation is desired. This may be important when a jam or stall condition must be detected. An encoder attached directly to the load can improve the overall accuracy of the system as well.





Common Applications

- Short, rapid moves
- Accurate positioning systems

Advantages

- High acceleration, duty cycle
- Excellent repeatability
- No maintenance (brushless)
- High torque at rest, moderate speeds
- Mechanically stiff with no vibration at rest
- Lower cost than brushless servo

Disadvantages

- Inefficient at high speeds
- Excessive loads can stall motor

Introduction to Motion Control Technology

Brushless Servo Systems

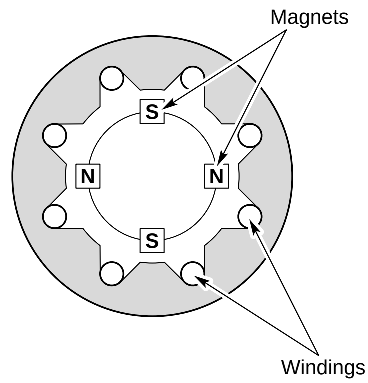
IDC brushless servo systems use a three-phase brushless servo motor with an incremental encoder for position feedback.

Brushless servo motors are constructed with the windings in the stator or outer portion of the motor and have permanent magnets attached to the rotor. The location of the windings next to the finned surface of the motor provides a shorter path for heat generated in the windings to escape. This improved thermal efficiency when compared to a dc motor results in higher power ratings for a similar sized motor. Less rotor inertia and no mechanical commutator limits result in higher speed and acceleration capabilities.

The servo controller and drive use the encoder feedback signal to continuously adjust the motor torque so that the desired position is maintained. This is referred to as a closed loop servo system. The electronics required to operate a brushless motor and “close the loop” are therefore more complex and expensive than microstepping or dc motor controls.

Unlike DC motors, which use a mechanical commutator to deliver current to the appropriate torque producing windings, the brushless motor requires a feedback device to sense the position of the permanent magnet rotor in relation to the motor’s windings. The servo drive must maintain the correct relationship between the magnetic vector created by the windings and the rotor position to produce torque.

IDC brushless servo drives employ proprietary vector torque control and advanced servo algorithms to provide unmatched closed loop servo performance (see page K-32 for a more thorough discussion).



Common Applications

- Higher speeds and accelerations
- Changing loads
- Clamping, pressing

Advantages

- Closed loop control
- Highest torque at high speeds
- No maintenance (brushless)
- Efficient operation

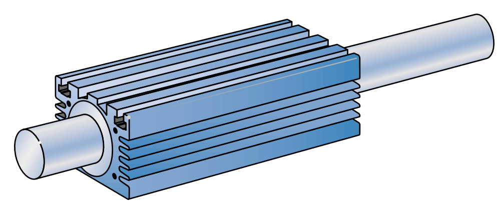
Disadvantages

- Higher cost

Linear Brushless Servo Motors

A special type of brushless servo motor, IDC direct drive linear motors use an innovative design that houses the permanent magnets in a cylindrical thrust rod. The motor’s windings reside in the thrust block which surrounds the rod. A linear incremental encoder is used for closed loop position feedback along with hall sensors for commutation.

All of the electromagnetic force is utilized to produce thrust directly, eliminating the need for rotary to linear conversion mechanisms. This eliminates backlash, lead error and other mechanical system innacuracies.



Common Applications

- Highest linear speeds and accelerations
- Repeatable linear positioning

Advantages

- Direct production of thrust
- High reliability
- Efficient operation

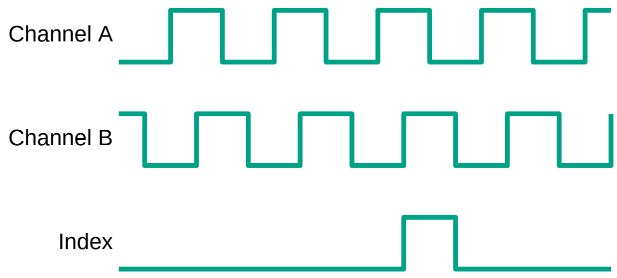
Disadvantages

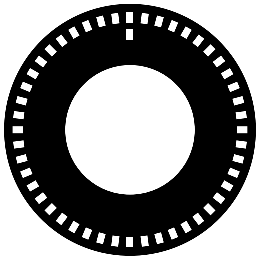
- Higher cost

Introduction to Feedback Devices

Incremental encoder

Optical encoders generate an output signal as the encoder disk rotates. The disk passes between a light source and photo detectors. The number of lines on the disk determines the number of on-off cycles of each output during a revolution. The detectors are arranged such that the two output signals are in "quadrature" (shifted in phase 90°). A once per revolution index channel is usually provided for establishing a home or zero position upon power up.

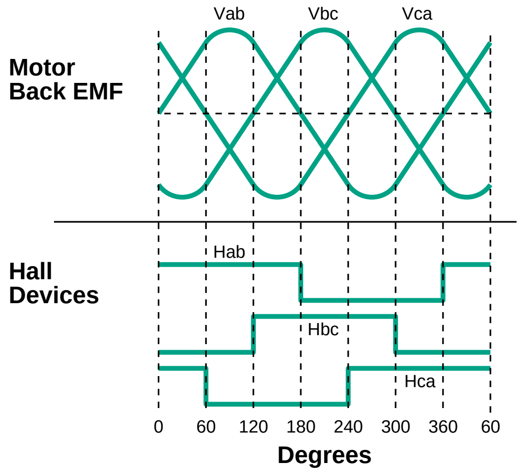


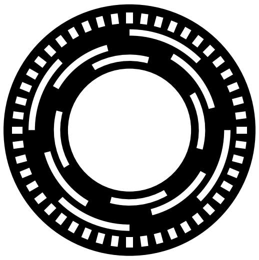


Electronic circuitry in the motion controller counts the pulses from the encoder to determine the position of the motor. The quadrature signal allows the controller to count up when rotating in one direction and down in the other, thus always remembering the absolute position of the system as long as power is applied. There are four unique transitions per cycle and controllers will generally count each transition. The number of counts per revolution is often referred to as the "post quadrature" resolution. For example, an encoder with 1000 lines will generate 4000 counts per revolution "post quadrature".

Incremental encoder with commutation channels

The process of steering current through the appropriate motor windings in order to produce output torque is called commutation. In brush motors, commutation is performed electro-mechanically via brushes and the commutator. In brushless motors, commutation is performed by electronically steering the current to the appropriate winding. To do this, the rotor position must be determined. A circuit board containing the Hall devices is aligned with a magnet on the rotor, so that the relationship shown between the Hall outputs and the motor back EMF can be established.





When a brushless motor is used in a servo application requiring position feedback, room must be made for both the commutation circuit board as well as the encoder. This generally adds both cost and complexity to the motor package. Encoders are now available with integrated commutation outputs equivalent to those produced by Hall devices. The result is a more compact system, reduced alignment time and superior switching accuracy, due to the much lower hysteresis of the encoder when compared to a Hall device. Adding additional data tracks to the encoder disk provides the commutation outputs.

Increasing Step Motor Performance

How *Anti-Resonance* Increases Step Motor Performance and Throughputs

The goal in step motor/drive selection is to identify the most economical motor/drive package that will reliably perform all the moves required in a given application. This can be a challenging task, since using conventional step motor systems is not always a straightforward procedure. Step motors can lose synchronization (stall), which causes a loss of positional accuracy resulting in an unsuccessful move. Step motor systems will stall when the torque demand for the move being executed, plus the torque lost in overcoming vibration, exceeds the available torque from the motor. One of the major causes of step motor stalling is a phenomenon called Resonance.

In conventional step motor systems, the increased torque demand on the motor due to Resonance decreases the performance of the motor throughout the motor's speed range due to a reduction in useable torque to accelerate the load. This reduction in useable torque can either severely limit the system's performance, leading to reduced throughput, or prevent the stepper system from working at all. In Microstepping systems, Resonance is especially troublesome in the motor's speed range of 10-15 rps where it is commonly called Mid Range Instability. IDC's microstepping drives eliminate the problem of Resonance via proprietary Anti-Resonance circuitry, and thereby give the user the maximum amount of torque possible throughout the motor's speed range. To illustrate the negative impact of resonance on conventional step motor performance, and to show how much better the same system will work with an IDC microstepping drive, a look at the causes and effects of resonance on underdamped microstepping systems is in order.

Step Motor Dynamics

Motor torque is delivered to the step motor's rotor when the rotor lags behind or leads the commanded position (see **Figure 4**). The maximum amount of torque the motor will exert on the rotor occurs when the rotor is displaced 1.8° from the commanded position (Torque equals T_m). Any deflection past 1.8° from the commanded position will cause a reduction in applied motor torque from the max torque T_m (see **Figure 4**). When the rotor is deflected from the commanded position, a restoring torque acts on the rotor to move it back to the commanded position (equilibrium) much like a spring. The torque seen by the rotor at any time is a sinusoidal function of its displacement from the commanded position. In **Figure 4** a 1.8° full step (dashed line) is applied to a rotor which is located at the previous commanded position. The instantaneous torque seen by the rotor when the full step is applied is T_m. Since the torque seen by the rotor is a function of position, the 1.8° step command has caused a step change in applied torque to the rotor that will cause the motor to ring about the new commanded position (**See Figure 6**).

Resonance

Resonance is troublesome to step motor users due to the increased torque demand caused by motor vibration. This vibration causes the difference between the commanded and actual position of the rotor to vary in an undesirable manner. The difference between the commanded and actual position is called the position error. It is this position error that dictates the amount of torque that is applied to the load. Resonant behavior

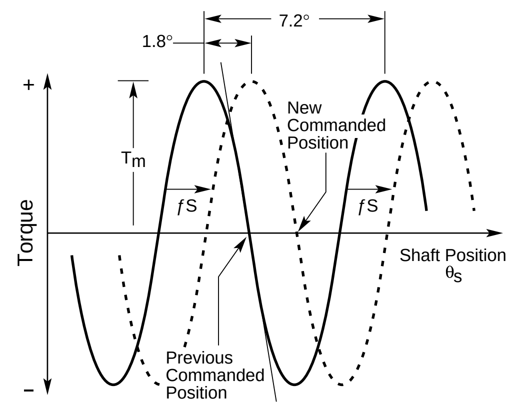


Figure 4. Operating detent of a step motor.

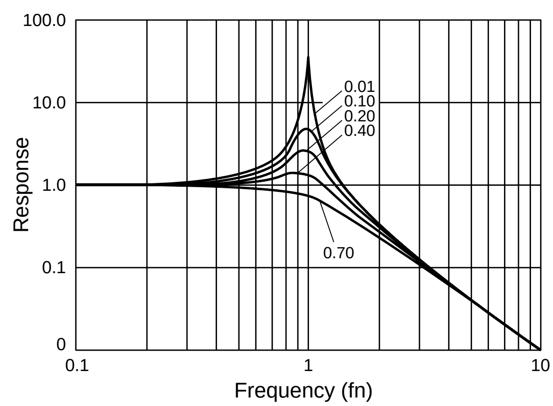


Figure 5. Frequency response of step motor system. Higher Values of ζ lead to reduced shaft vibration.

in a poorly damped step motor system increases the likelihood of the position error exceeding 1.8° (point of maximum motor torque). Once the rotor has slipped past 1.8° there is a decrease in torque applied to the load which can quickly lead to a motor stall if the required move needs the motor torque that was lost. This torque loss due to shaft vibration is why we look to increase the amount of damping present in the system. Minimizing shaft vibration leads to optimal torque utilization.

The amount of damping in a system can be described by the system's damping ratio ζ (Zeta). The higher the value of ζ the more the system is damped. A typical value of ζ for a conventional underdamped step motor system is 0.02. A step motor system with the optimal amount of damping has a ζ value of approximately 0.7. The effect of increasing the damping ratio is shown in **Figure 5**. A typical underdamped step motor system magnifies disturbances (vibration) around its natural frequency by a factor of 50. It is this vibration magnification which often causes step motors to stall. An optimally damped step motor system (like the *NextStep*™) with a damping ratio of 0.7 shows no such vibration amplification. This well behaved system can fully utilize a motor's torque to do real work in your application, hence higher acceleration rates are possible and greater headroom is provided for varying loads.

To illustrate how increasing the damping of a step motor system reduces ringing and improves settling time, it is useful to look at the response of a step motor to a unit step input for different values of ζ. This is shown in **Figure 6**. A system with a damping ratio (ζ) of 0.7 settles to the commanded position promptly, while a conventional step motor system with a damping ratio of 0.02 rings far off to the right of the graph. Reducing a system's settling time increases your system's throughput, allowing a user to make more parts per minute.

A step motor system that lacks adequate damping not only has ringing when a position step is applied, but also has significant ringing when the motor is commanded to follow a given velocity profile. A popular velocity profile used in most applications is the trapezoidal velocity profile (accel-slew-decel). A typical step motor system's response to a trapezoidal velocity profile is shown in **Figure 7.** Notice the ringing in the motor's velocity both at the higher speed and after the ramp. Compare this to the velocity tracking found in an optimally damped system like the *NextStep*™ (**Figure 8**). The superb velocity tracking and quick settling found in the IDC microstepping drives allows a user to make very aggressive moves without having excessive ringing at speed.

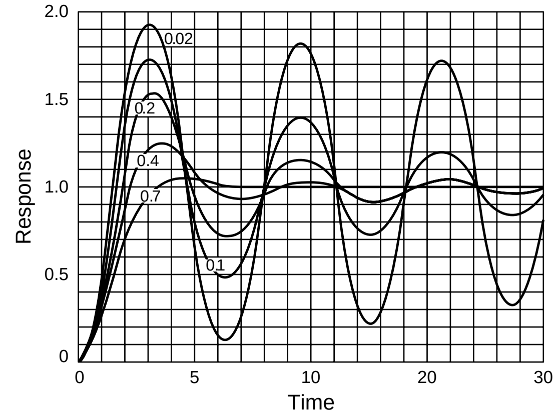


Figure 6. Ringing transients following a unit step. Increasing ζ results in more rapid settling.

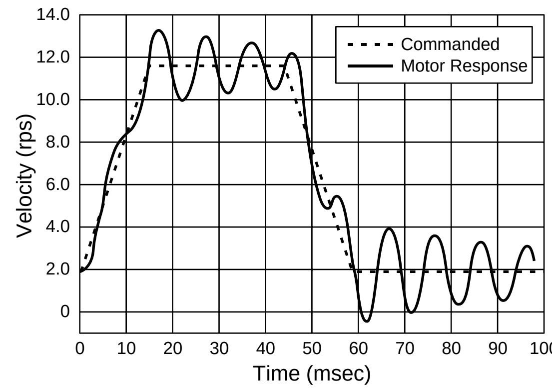


Figure 7. Typical step motor system's response to a trapezoidal velocity profile. Lack of damping leads to vibration at the top and bottom of the profile.

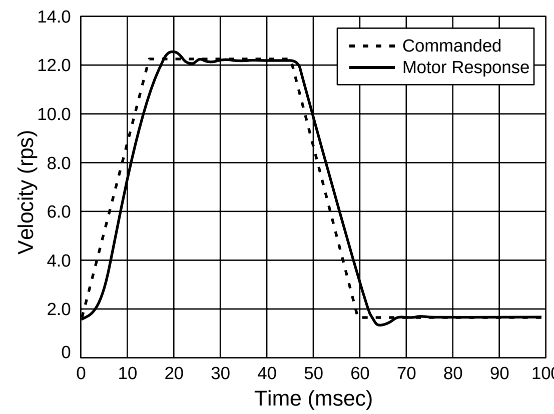


Figure 8. *NextStep*™ system response to a trapezoidal velocity profile. Increasing the damping leads to better velocity tacking and smoother overall motion.

Rotary System Selection

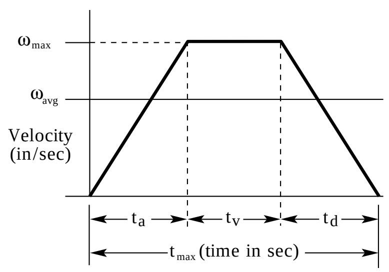
This section provides useful information for calculating your application's mechanical requirements, and selecting the proper motor and drive to meet your needs. To insure the proper motor/drive system is selected, follow these steps:

1. Sketch the **move profile** and calculate acceleration, deceleration and maximum velocity required to make the desired move.
2. Select **mechanical drive mechanism** to be used and calculate inertia, friction and load torque using formulas for the mechanical drive mechanism.
3. Determine **peak and continuous (RMS) torque** requirements for the application.
4. **Select a system** – choose the appropriate motor and drive combination that meets all of the application requirements.

1. Move Profile

Refer to the Move Profile section on page K-10 to determine your peak velocity and acceleration. Rotary distance units should be radians. Time units are seconds. **NOTE: 1 rev = 2π radians.**

Example: We need to move a total distance of 10 revolutions in 1 second using a 1/3, 1/3, 1/3 trapezoidal move profile. What is the distance (d_{tot}), peak velocity (ω_{max}) and acceleration (α) required to make the move.



$$\text{Total distance, } d_{tot} = 10 \text{ rev} \cdot 2\pi \frac{\text{rad}}{\text{rev}} = 63 \text{ rad}$$

$$\text{Max velocity, } \omega_{max} = \frac{1.5 d_{tot}}{t_{tot}} = \frac{1.5 \cdot 63}{1} = 95 \frac{\text{rad}}{\text{sec}}$$

$$\text{Acceleration, } \alpha = \frac{4.5 d_{tot}}{(t_{tot})^2} = \frac{4.5 \cdot 63}{12} = 284 \frac{\text{rad}}{\text{sec}^2}$$

2. Mechanical Drive Mechanisms

The system equations on the following page will help you calculate the reflected inertia (J), reflected applied loads (T), motor speed (ω), and acceleration (α), based on the move requirements that were determined in step one.

3. Peak Torque and RMS Torque Requirements

To find the peak and RMS torque required by the motor to make the move successfully, the Reflected Torque, T_{RL}, is added to the Torque required to accelerate (or decelerate) the load. T_{RL} includes all external forces, such as gravity, friction, and applied forces. The equations for peak torque and RMS torque required are:

$$T_{PEAK} = T_{RL} + \left[\frac{J_T \alpha}{e} \right]$$

$$T_{RMS} = \sqrt{\frac{T_A^2 t_A + T_{RL}^2 t_R + T_D^2 t_D}{t_c}}$$

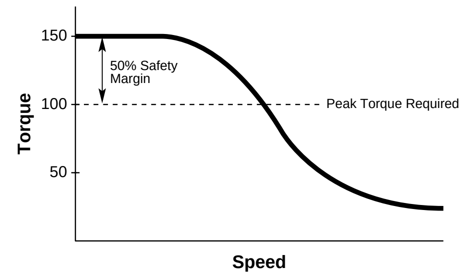
4. Selecting a System

Once the above torques have been calculated, a safety factor needs to be added. The safety margin varies with the motor type desired.

$$\text{Safety Margin} = \left(\frac{(\text{Torque Avail}) - (\text{Torque Req.})}{(\text{Torque Req.})} \right) \text{x } 100$$

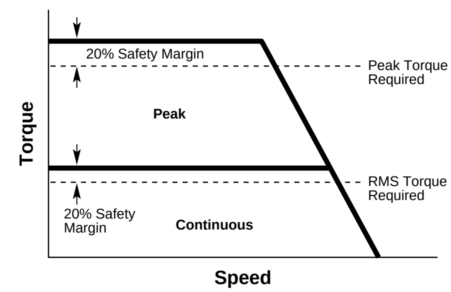
Stepper Systems

When selecting a step motor system, you should allow a torque safety margin of 50% above your calculated peak torque requirements at the peak speed required by your application.



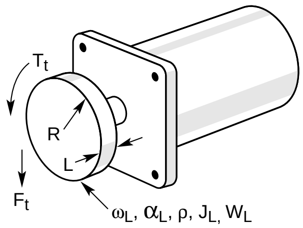
Servo Systems

For servo systems, a torque safety margin of at least 20% is recommended. The peak torque required by the application must fall within the peak torque rating of the motor at the peak speed. You must also calculate the RMS torque based on your application's duty cycle. The RMS torque must fall within the continuous area of the speed torque curve at the peak speed of the application.

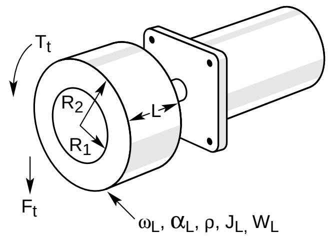


Rotary System Selection

Inertia Solid Cylinder

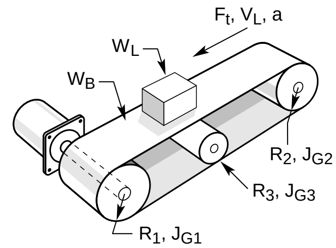


Inertia Hollow Cylinder



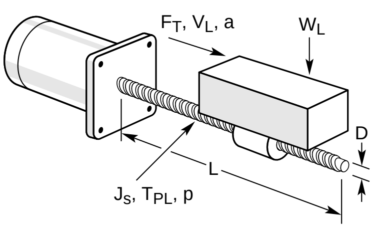
Tangential

(Conveyor, Rack and Pinion, etc.)

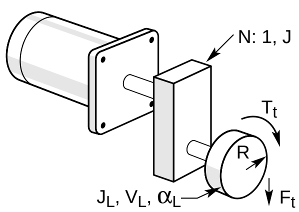


	Inertia Solid Cylinder	Inertia Hollow Cylinder	Tangential
ω	$\omega = \omega_L$	$\omega = \omega_L$	$\omega = \frac{V_L}{R_1}$
α	$\alpha = \alpha_L$	$\alpha = \alpha_L$	$\alpha = \frac{a}{R_1}$
J_T	$W_L = \pi L \rho R^2$ $J_T = J_L + J_{Motor}$ $J_L = \frac{1}{2}\frac{W_L}{g} \bullet R^2$	$W_L = \pi L \rho [R_2^2 - R_1^2]$ $J_T = J_L + J_{Motor}$ $J_L = \frac{1}{2}\frac{W_L}{g}[R_2^2 + R_1^2]$	$J_{RL} = \frac{W_L + W_B}{g} \bullet R_1^2$ $J_T = J_{Motor} + J_{RL} + J_{G1} + J_{G2}\left(\frac{R_1}{R_2}\right)^2 + J_{G3}\left(\frac{R_1}{R_3}\right)^2$
T_{RL}	$F_t = F_{Friction} + F_{Applied} + F_{Gravity}$ $T_{RL} = F_t \bullet R + T_t$	$F_t = F_{Friction} + F_{Applied} + F_{Gravity}$ $T_{RL} = F_t \bullet R_2 + T_t$	$F_t = F_{Friction} + F_{Applied} + F_{Gravity}$ $T_{RL} = \frac{F_t \bullet R_1}{e}$

Leadscrew



Gear Box



	Leadscrew	Gear Box
ω	$\omega = 2\pi p V_L$	$\omega = \frac{V_L}{N}$
α	$\alpha = 2\pi p a$	$\alpha = \frac{\alpha_L}{N}$
J	$J_S \approx 0.0012\ LD^4$ (for steel) $J_{RL} = \frac{W_L}{g}\left[\frac{1}{2\pi p}\right]^2$ $J_T = J_{RL} + J_S + J_{Motor}$	$J_{RL} = \frac{J_L}{N^2}$ $J_T = J_{RL} + J_{Reducer} + J_{Motor}$
T	$F_t = F_{Friction} + F_{Applied} + F_{Gravity}$ $T_{RL} = \frac{F_t}{2\pi \bullet p \bullet e} + T_{PL}$	$F_t = F_{Friction} + F_{Applied} + F_{Gravity}$ $T_{RL} = \frac{(F_t \bullet R) + T_t}{N \bullet e}$

Gear Reducers

Follow these guidelines when selecting an IDC precision gearhead:

1. Be sure your application's peak torque is less than the maximum momentary torque rating of the gearhead. Be sure to multiply the motor's torque by the gearhead efficiency and gear ratio when determining output torque from the reducer.

2. Be sure your application's RMS torque is less than the rated continuous torque of the gearhead.

Units

a = acceleration rate (rad/sec²)
α = rotary acceleration (rad/sec²)
D = diameter
e = efficiency of mechanism
F_t = total load force including friction, gravity, or other external forces (oz)
g = gravity = 386 in/sec²
J = rotary inertia (oz-in-sec²)
J_T = total inertia seen by motor (oz-in-sec²)
J_{RL} = reflected load inertia (oz-in-sec²)
L = length (in)
ρ = material density (oz/in³)
p = pitch of screw (rev/in)
R = radius (in)
t_A = acceleration time (sec)
T_A = acceleration torque (oz-in)
t_D = deceleration time (sec)
T_D = deceleration torque (oz-in)
T_{PL} = preload torque of leadscrew (oz-in)
t_R = running time (sec)
T_R = running torque (oz-in)
T_{RL} = reflected load torque due to friction, gravity, or other external forces (oz-in)
V_L = linear velocity of load (in/sec)
ω = rotary velocity (rad/sec)
W = weight (oz)

Rotary System Selection

EXAMPLE 1–STEPPER: Calculate the motor torque required to accelerate a solid cylinder of aluminum 5" in radius and 0.40" in length from rest to 10 rev/sec in 0.3 seconds. Assume no friction is present and there are no applied forces or gravity forces opposing the motor's rotation.

1

Calculate maximum velocity and acceleration rates.

$$\omega = 10\,\frac{\text{rev}}{\text{sec}} \bullet 2\pi\,\frac{\text{rad}}{\text{rev}} = 63\,\frac{\text{rad}}{\text{sec}}$$

$$\alpha = \frac{\omega_{\text{final}} - \omega_{\text{initial}}}{t_A} = \frac{63 - 0}{0.3} = 210\,\frac{\text{rad}}{\text{sec}^2}$$

2

Mechanical drive mechanism selected as direct drive.

3

Calculate inertia (assume no friction or load torque).
Hint: The density of aluminum can be found on page K-47.

$$W = \rho \pi L_S R^2 = 3.14 \bullet 0.40 \bullet 1.57 \bullet 5.0^2 = 49.3 \text{ oz}$$

$$J_L = \frac{1}{2} \bullet \frac{w}{g} \bullet R^2 = \frac{1}{2} \bullet \frac{49.3}{386} \bullet 5^2 = 1.6 \text{ oz-in sec}^2$$

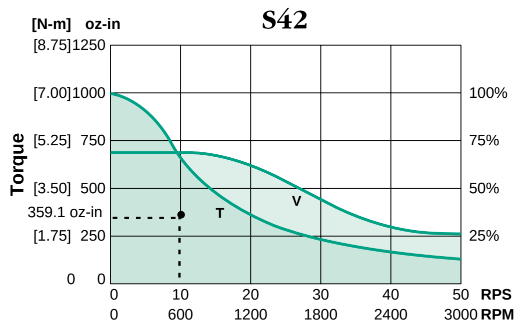
$$J_{Total} = J_{Load} + J_{Motor} \quad \left(\text{Assume motor is S42, rotor inertia is 0.114 oz-in-sec}^2\right)$$

$$= 1.6 + 0.114 = 1.71 \text{ oz-in-sec}^2$$

4

Determine peak and continuous torque requirements.

$$T_A = T_{RL} + \left[\frac{J_T \alpha}{e}\right] = 0 + \frac{1.71 \bullet 210}{1.0} = 359.1 \text{ oz-in}$$



Plotting the required torque on the S42 speed torque curve shows sufficient torque is present with a safety margin of 80%.

EXAMPLE 2–SERVO: Select a motor which will move a 500 lb load 10 inches in 1 second using a 2 pitch leadscrew 24 inches in length, and 1 inch in diameter. The load is externally supported by plastic bushings sliding on steel supports. There is a 125 lb applied force. There will be 1.0 second dwell between moves. The orientation of the leadscrew is horizontal.

1

Calculate distance the motor must rotate, acceleration, deceleration and maximum velocity using a 1/3, 1/3, 1/3 trapezoidal move profile.

$$D = 2\pi \bullet p \bullet d = 2\pi\,\frac{\text{rad}}{\text{rev}} \bullet 2\,\frac{\text{rev}}{\text{in}} \bullet 10 \text{ in} = 126 \text{ rad}$$

$$\omega = \frac{1.5D}{T} = \frac{1.5 \bullet 126}{1.0} = 189\,\frac{\text{rad}}{\text{sec}}$$

$$\alpha = \frac{4.5D}{T} = \frac{4.5 \bullet 126}{1.0^2} = 567\,\frac{\text{rad}}{\text{sec}^2}$$

$$t_A = t_D = t_R = t/_3 = 0.33 \text{ sec.}$$

2

Select mechanical drive mechanism as a leadscrew.

3

Calculate inertia, friction and load torque. A B32 motor was chosen to see if it will meet the requirements of the application. The friction coefficient of plastic on steel is 0.2.

$$J_{RL} = \frac{W_L}{g}\left[\frac{1}{2\pi p}\right]^2 = \frac{500 \text{ lb} \bullet 16\,\frac{\text{oz}}{\text{lb}}}{386 \text{ in/sec}^2}\left[\frac{1}{2\pi \bullet 2\,\frac{\text{rev}}{\text{in}}}\right]^2 = 0.131 \text{ oz-in sec}^2$$

$$J_S = 0.0012 \bullet L_S \bullet D^4 = 0.0012 \bullet 24 \bullet 1^4 = 0.0288 \text{ oz-in sec}^2$$

$$J_{Total} = J_{Motor} + J_{RL} + J_S = 0.016 + 0.131 + 0.029 = 0.176 \text{ oz-in sec}^2$$

$$F_t = W_L \bullet \mu_S + F_{Applied} = 500 \text{ lbs} \bullet 16\,\frac{\text{oz}}{\text{lb}} \bullet 0.2 + 125 \text{ lbs x } 16\,\frac{\text{oz}}{\text{lb}}$$

$$+ 2000 = 3600 \text{ oz}$$

$$T_{RL} = \frac{F_t}{2\pi p e} + T_{PL} = \frac{3600}{2\pi \bullet 2 \bullet 0.9} + 0 = 318 \text{ oz-in}$$

4

Determine peak and continuous torque requirements.

$$T_A = T_{RL} + \frac{J_T \alpha}{e} = 318 + \frac{0.176 \bullet 567}{0.9} = 429 \text{ oz-in}$$

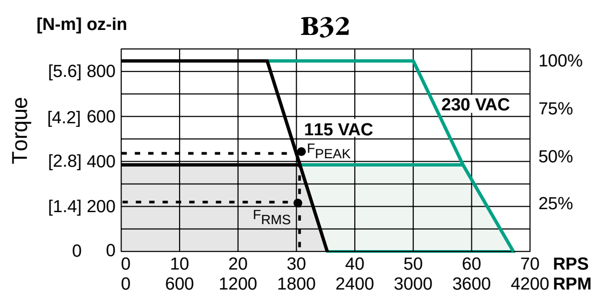
$$T_D = T_{RL} - \frac{J_T \alpha}{e} = 318 - \frac{0.176 \bullet 567}{0.9} = 207 \text{ oz-in}$$

If the dwell time between moves is 1.0 second, the total cycle time becomes 2.0 seconds.

$$T_{RMS} = \sqrt{\frac{T_A^2 t_A + T_R^2 t_R + T_D^2 t_D}{t_c}}$$

$$= \sqrt{\frac{(429)^2 \bullet 0.33 + (318)^2 \bullet 0.33 + (207)^2 \bullet 0.33}{2.0}}$$

$$= 233 \text{ oz-in}$$



The maximum torque required by the application is 429 oz-in. This falls within the speed/torque curve of the B32 motor, but it falls in the peak region. In order to have adequate torque margin in this application the B8000 Series drive used with the B32 motor would need to be run off of 230 VAC. Assuming the B8000 Series drive is run off of 230 VAC, the peak torque safety margin is 98.1%

Interfacing with IDC Controls

IDC controls are designed to provide the optimum combination of cost and performance for a variety of applications All of our controls are designed for use with IDC actuators, providing the fastest implementation time in the industry.

Drives

IDC drives provide the power to the motor and control it's torque, velocity or position output. They are designed to accept industry standard signals from motion controllers.

Microstepping systems control the position of a step motor and provide resolutions up to 50,400 steps per revolution. These drives accept a digital step and direction signal and move the motor one microstep for each pulse received. Varying the frequency of incoming pulses controls velocity.

Brushless servo drives are available to control torque, velocity or position of a brushless servomotor. These drives accept digital step and direction or analog command signals.

When to use: Use drives when your application cannot be solved with one of IDC's packaged motion controllers.

Limit Switch Controls

Limit switch controls combine the functionality of a DC variable speed drive with a very simple position control that uses limit switches to provide position feedback.

Manually adjustable (actuator or load mounted) limit switches sense the load position and signal the control to reverse direction, change speed or stop the motor. Inputs are also provided for jog or run modes.

Limit switch controls often interface with PLCs or industrial PCs and are our most cost effective solution for applications requiring simple position control.

When to use: Use limit switch controls when your positioning requirements can be met with limit switches and when positions, loads and speeds don't change frequently.

Edge Guide Controls

IDC edge guide controls are a special purpose type of limit switch control used to maintain the position of the edge of a moving web of material.

The control uses feedback from sensors that can detect the edge of the material and move an actuator when adjustment is required. Two different correction speeds can be defined based on the amount of error.

When to use: When you want to control a web edge based on input from simple on-off sensors.

Analog Position Controls

Analog position controls combine our drive electronics with an analog absolute position controller. They control the absolute position of an actuator in direct proportion to an analog input signal. An IDC actuator equipped with a linear potentiometer provides position feedback to the controller.

Analog position commands often come from a PLC, industrial computer, sensor, data acquisition system, or a manually operated joystick. Typical applications include controlling flow or mixing of materials, web edge guiding, and remote manual positioning.

When to use: Use analog position controls when you need to control absolute position relative to an analog command signal, when you cannot home the system or when you need to know the position at power up.

Programmable Motion Controls

IDC's Smart Drives combining one or more of our drive products packaged together with a powerful programmable positioning control.

A single package combines a one or two axis motion controller, drive(s), an AC input power supply, support for OPTO22 modules, an optional detachable front panel and RS232 communications interface.

They use IDC's powerful and intuitive *IDeal* programming language. This language has been called "simply sophisticated" by some due to its combination of advanced capabilities in an easy to understand format. You'll have your system up in running in a minimum amount of time.

When to use: Use programmable Smart Drives when you need a flexible, powerful and easy to use motion control system.

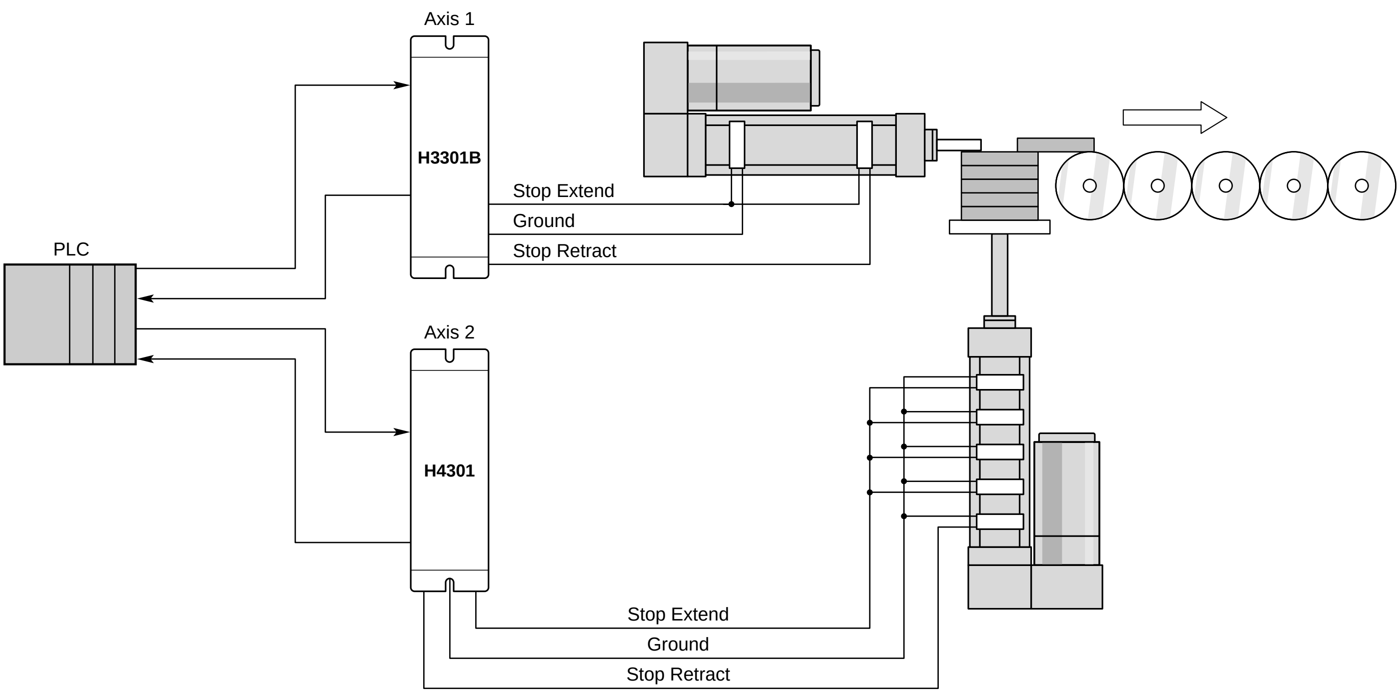
Interfacing with IDC Controls

One of the most important considerations when selecting a linear motion system is how it will interface to your PLC, your machine, your operators and the rest of your plant. In many applications more than one type of interface is employed. For example, you could use

RS-232C for programming, dedicated "program select" inputs for program execution, digital I/O for sensors, and a keypad/display for entering run-time operation variables. The following examples show common interface techniques using IDC controls.

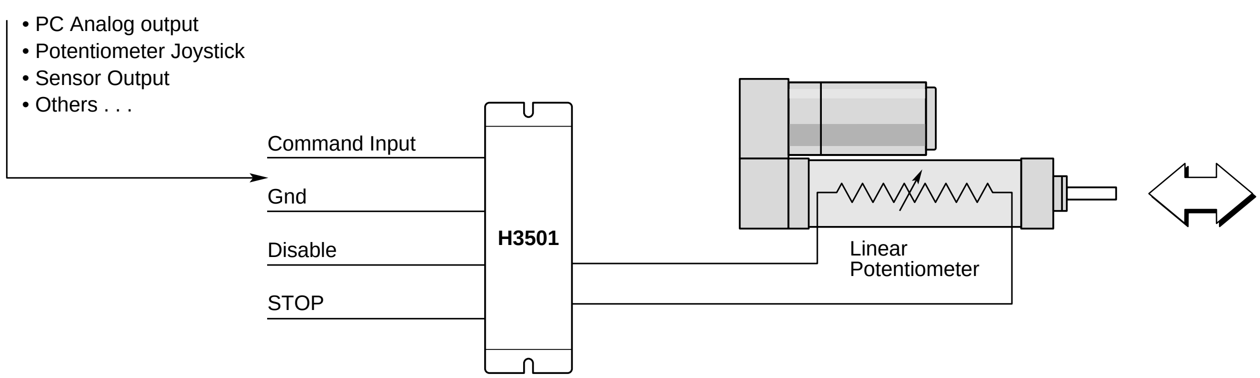
Limit Switch Control – PLC, Using Discrete I/O

Industrial Devices' H3301B, H4301 and D2000 Series Controls allow simple positioning control using extend, retract, and stop inputs. Limit switches (reed or hall effect) can be mounted to the cylinder sides and wired back to the control to signal a stop position, change in speed or direction, or to activate an output.



Analog Position Control, Using an Analog Input

Industrial Devices' H3501, H4501, B8501 Controls can accept analog position signals (4-20ma or 0-10V) from a variety of sources. IDC actuators are available with linear potentiometer position feedback. The actuator position is proportional to the input signal. Adjustments allow the user to scale and offset the stroke and tune the move profile to suit the application. We also offer drives with analog velocity or torque inputs for use with servo controllers.

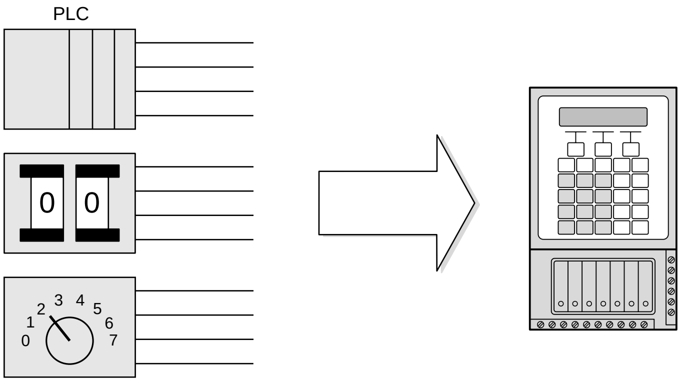


Industrial Devices Corporation
707-789-1000 • 800-747-0064 • E-mail: info@idcmotion.com

Interfacing with IDC Controls

Remote Program Control – PLC, Using Binary or BCD Inputs

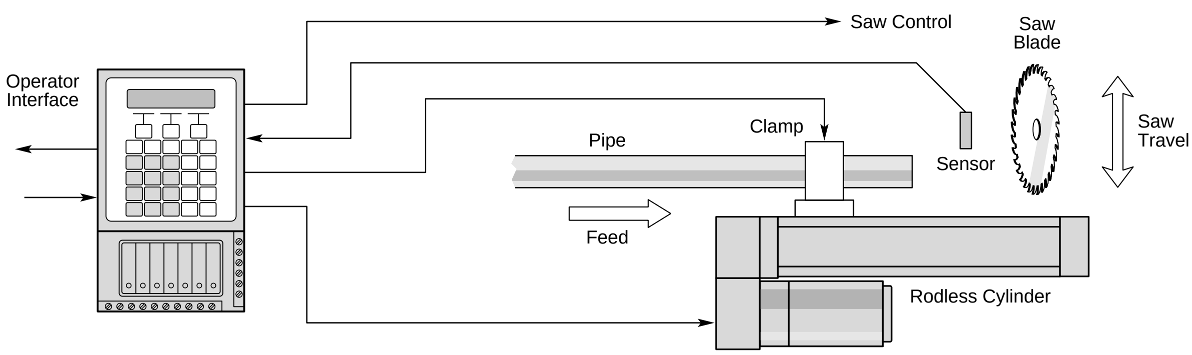
Industrial Devices' controls have the capability of remote program execution. A program is selected using binary (or BCD) coded digital inputs with a PLC, computer, or simple switches (thumbwheels or rotary).



Program Number	Inputs 3	2	1
0	1	1	1
1	1	1	0
2	1	0	1
3	1	0	0
4	0	1	1
5	0	1	0
6	0	0	1
7	0	0	0

Stand-Alone Machine Control

An Industrial Devices' keypad control can function as a basic control system for your machine. Our *IDEAL*™ motion programming language includes functions for I/O control, time delays, variable input, conditional branching, and interrupts. With 24 optically isolated I/O, and a built in operator interface with programmable function keys, an IDC keypad control can replace a small PLC in many cases. To further increase the interfacing possibilities, 8 I/O ports are OPTO 22 compatible. By selecting the appropriate OPTO 22 modules the user can output or input AC, DC or analog signals.



Applications

Application 1: Packaging

Customer's Objectives

In a manufacturing line, the maximum throughput of the line is determined by the slowest component in the line. In this case, the customers throughput was limited to 200 bottles per minute by the packaging equipment.

To improve the throughput of the line, (which was capable of a maximum of 1000 bottles per minute, without the packaging bottle neck), the customer wanted to divert the line to 5 packaging machines, each capable of 200 bottles per minute, thus increasing his throughput to 1000 bottles per minute.

The customer needed to do several product changeovers (bottle size, quantity per case, etc.) per day, and wanted to use just one control for all of the machine functions including

I/O and intelligent control. In addition he wanted to make the machine easy to use, by making product changeovers with just a turn of a rotary switch.

IDC Solution

Control	B8961
Actuator	EC Series

The EC series cylinder High thrust and duty cycle capability coupled with B8961's high level programming language proved to be the perfect solution for this customers application. The conditional branching math and counter functions allowed the control to know when to shift lanes, and the ability to skip a lane if a bottle jam had been detected by remote photo eyes. The Flexible Opto 22 I/O allowed the customer to control all of the inputs and outputs without needing any additional interface modules.

Inputs

1-5	Jam Sensors
6	Bottle Counter Input
7-8	Ends of travel
9	Start Button
10	Estop
11	Reset
12-15	BCD Program Select from thumbwheel switch

Outputs

1-5	"Pack" Signal to machine
6	"Run" light
7	"Error" light
8	Fault Output to main system

Why IDC?

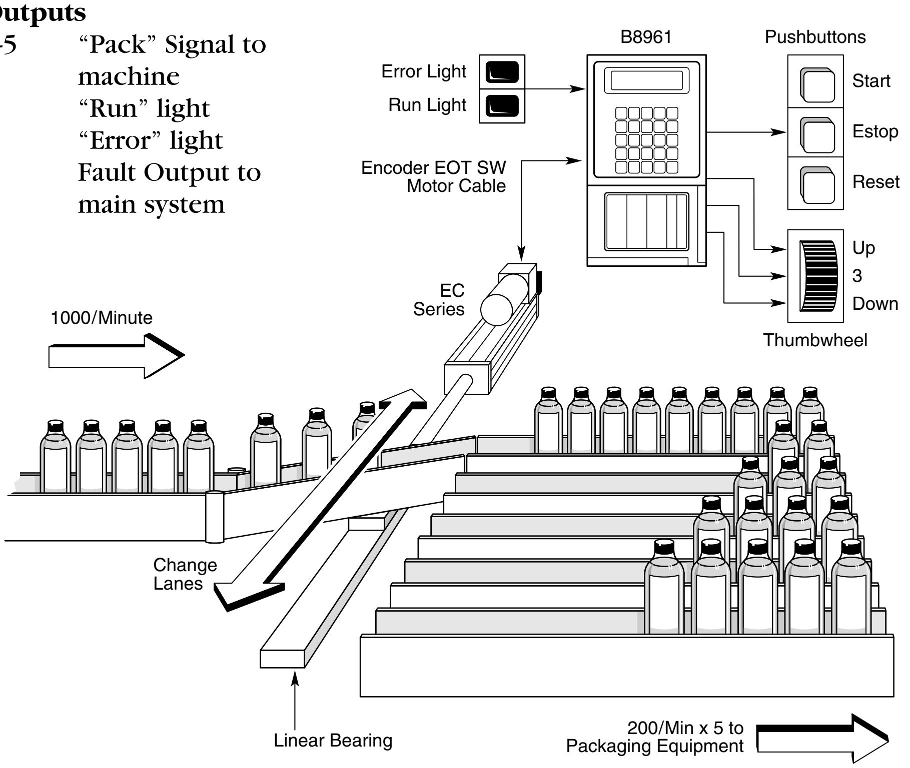
This customer had considered using pneumatics or building the system from components and using another manufacturers control.

Compared to the pneumatic solution, the IDC system offered much simpler set up, (verses moving the hard stops on the pneumatic cylinder), which cuts production downtime significantly. The cost savings realized from the additional production, (as well as lower maintenance costs) quickly overcame the initial saving of the pneumatic system.

Purchasing a complete system from IDC reduced vendors, came with a 1 year warranty, and was a lower total cost. In addition his I/O requirements could not be met by other control manufacturers without additional components.

Lane Changer Requirements

Load	200 lbs
Speed	10 in/sec
Duty Cycle	70%
Repeatability	0.010"
Programmability	8 different cominations of bottle size and package quantity
Interface	Thumbwheel Switch and RS232 Interface for programmaing



Applications

Application 2: Cable Manufacturing

Customer's Objectives

A cable manufacture wanted to build a wire winding machine for less than the cost of off-the-shelf general purpose winders.

Most general-purpose machines use complex and expensive master-slave position following controls. Since this customer's products were established and their winding requirements were predictable and consistent, a less sophisticated control solution would be sufficient.

The wire would be fed at a constant speed from an extruder, and the wire would be guided onto the spool by a linear positioning device. To get a uniform wind, the traverse speed of the guide needs to decrease as the circumference of the wire wrapped on the spool increases, and as the rotational speed of the spool slows down.

IDC Solution

Control	H3301B
Actuator	R3H105B-18-SC-MF3-Q
Accessories	RPS-1 Limit switches (for end positions)

The customer found a simple, cost effective solution in our H3301B Limit Switch Control and a small PLC with analog input/output capability. The H3301B drives an R3H Series Rodless actuator between limit switches for the traverse motion.

The PLC calculates the diameter of the roll based on a speed signal from a tachometer attached to the spool, and adjusts the traverse velocity (through the H3301B's remote speed input) as the spool speed changes.

H3301B Inputs

1. Analog speed, from PLC
2. Extend, from PLC and switch
3. Retract, from switch
4. Stop Extend, from switch
5. Stop Retract, from switch

H3301B Outputs

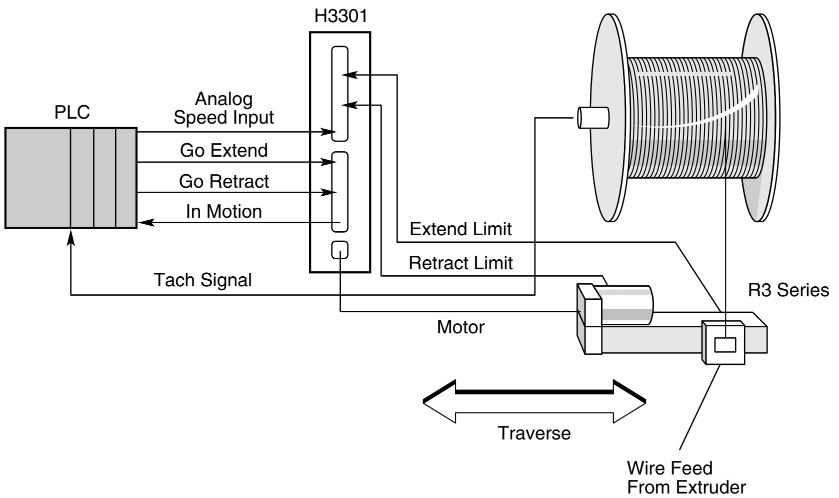
1. In Motion, to PLC
2. Fault/Current Limit, to PLC

Why IDC?

The unique features of the H3301B control make it an ideal solution for this application. Other controls and actuators were considered, however the unnecessary complexity and cost made the H3301B an excellent choice.

Winding Control Requirements

Load	50 lbs
Speed	15 in/sec max
Duty Cycle	100%
Repeatability	±0.1 inches
Programmability	Automatic extend/retract sequencing
Interface	Discrete I/O to PLC



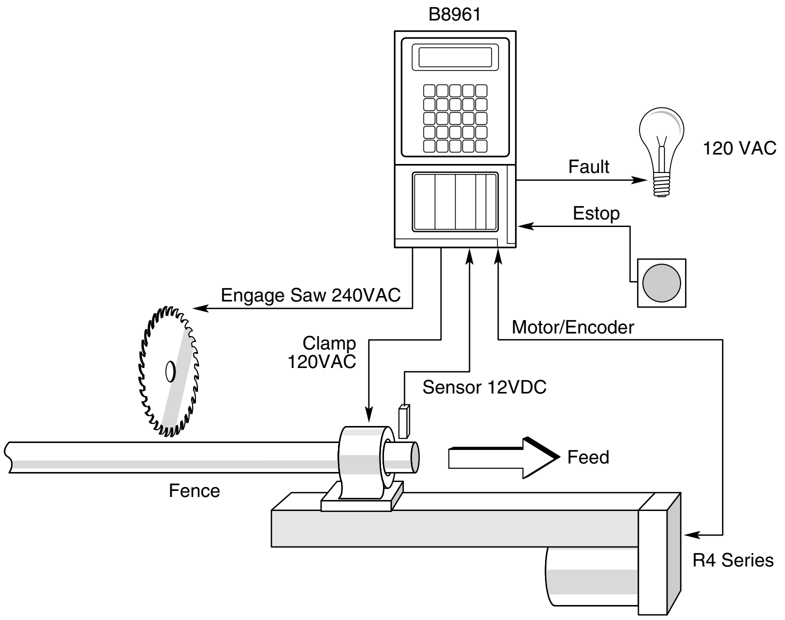
Applications

Application 3: Fence Making

Customer's Objectives

A manufacturer of fence making equipment received a special request from their customer for a machine which cuts fencing sections to length from 0 to 120 inches, with as many as 150 product (length) changes per day.

Although the company already has fence cutting equipment, the set up time to change lengths (changing stops on a hydraulic cylinder) took more than five minutes. A five minute set up was unacceptable to this customer.



Feed to Length Requirements

Load	150 lbs moving, 800 lbs holding
Speed	3.0 inches/sec
Duty Cycle	40%
Repeatability	±0.01 inches
Programmability	Prompt operator for position in inches and quantity
Interface	Operator interface

IDC Solution

Control	B8961 Servo Control
Actuator	R4 Series
Accessories	RP1 home sensor
	RP2 end-of-travel

The customer mounted the fence gripper to the carriage of our 120 inch R4H Series rodless cylinder. To make a fully automated 0-120 inch cut to length system.

The B8961 display prompts the operator to enter the length of the fence in inches, as well as the number of sections to cut. The Home offset command allows the user to enter in the actual length of fence.

B8961 Inputs

1. Material present, from sensor (12 VDC)
2. Emergency stop, from push-button (12 VDC)

B8961 Outputs

1. Grip fencing, to air valve (120 VAC)
2. Engage saw, to cutter (240 VAC)
3. Fault/Current limit, to yellow light (120 VAC)

Why IDC?

The customer chose our RH Series actuator because of its compact size and excellent carriage moment load capacity.

The B8961 control was chosen for its high performance servo response, integral operator interface, ease of programming, and I/O capability, which eliminated the need for a PLC.

Applications

Application 4: Sawing

Customer's Objectives

A manufacturer of large, custom size masonry flooring material wanted to automate the cutting width of their saw. Several set-up changes occurred per day, each one taking 10 to 15 minutes. The operator would set the backstop to a new position, make a sample cut, measure the results and then make further fine adjustments. The goal was to reduce set up time to a few seconds and reduce scrap by 90%.

The eight foot long backstop has to be held perpendicular to the saw blade. Because of this, the customer wanted actuators to position each end of the backstop and at least 800 lbs. of holding capacity during cutting.

IDC Solution

Control	S6962 two axis micro-stepping smart drive
Actuators	EC Series
Accessories	RPS-1 home sensor RPS-2 end-of-travel sensors

The S6962-2 axis microstepping smart drive has the ability to synchronize two actuators and independently home them to ensure orthogonality. The S6962 stepper drive provides the high current to operate the two EC Series cylinders. The keypad/display prompts the operator to enter the width and provides a signal to begin cutting.

6962 Inputs

1. Move (push-button)
2. Inhibit motion (while cutting)
3. Operator Interface (for position)

6962 Outputs

1. In position
2. Fault

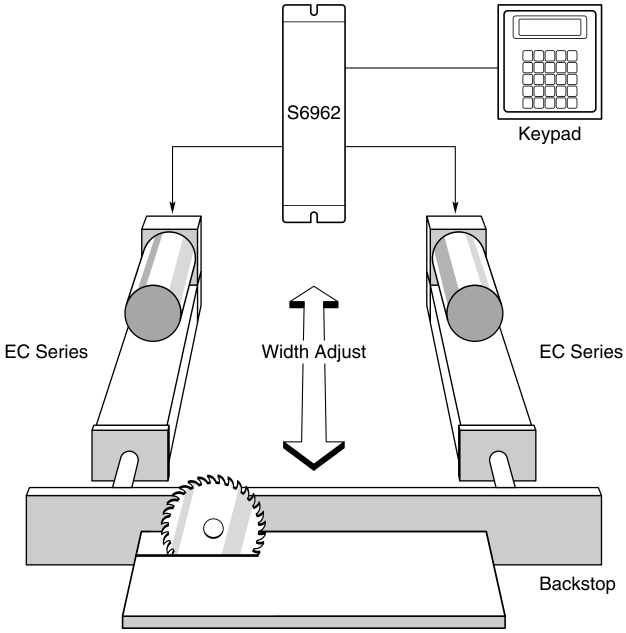
Why IDC?

The high repeatability of the system (±0.0005) provided the consistent high quality product the customer desired. The programmability and operator interface capability of the S6962 provided the rapid set-up needed. The cylinder was provided with the -PB option which protects the cylinder from cutting fluid spray and masonry particles.

Prior to being contacted by an Industrial Devices distributor, the customer had considered building their own. After reviewing the steps required to build, and seeing what IDC had to offer, the customer found the IDC solution to be more simple, and more cost-effective.

Back-stop Adjustment Requirements

Load	400 lbs moving, 800 lbs holding
Speed	1 in/sec
Duty Cycle	10%
Repeatability	±0.0005 inches
Programmability	Synchronize two actuators
Interface	RS-232C programming

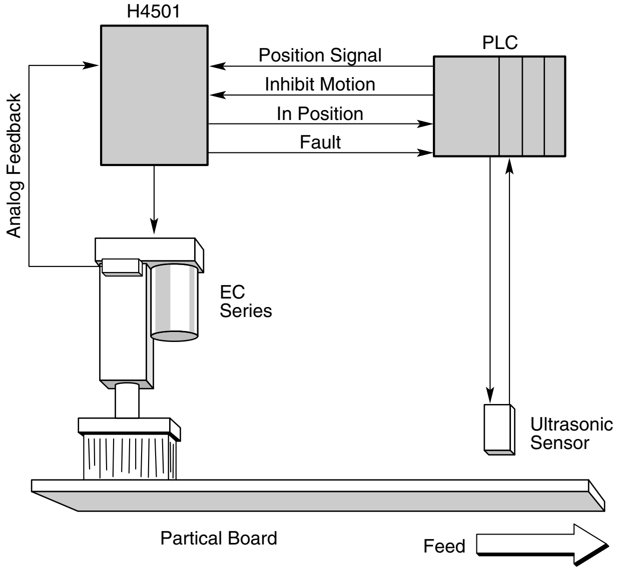


Applications

Customer's Objectives

A particle board manufacturer needed an inexpensive way to automatically adjust the height of a scrubbing deck, based on the thickness of the material passing underneath. The rotating wire brushes on the scrub deck were used to smooth the surface of the wood.

The company received several quotes for expensive position following servo controls which were capable of much greater accuracy than they needed (0.025" inches). They also wanted the solution to take full advantage of the PLC being used to control the machine.



Height Control Requirements

Load	1000 lbs, vertical
Speed	0.3 inches/sec
Duty Cycle	30%
Repeatability	±0.025 inches
Control	Analog position command from PLC
Interface	4-20 mA signal

IDC Solution

Control	H4501-C
Actuator	EC Series

Brush height control was easily realized using a H4501-C Control which accepts a 4-20 mA analog (position) output from the PLC. The PLC takes the height measurement from an ultrasonic sensor, and provides a corresponding position signal to the H4501-C Control.

H4501 Inputs

1. 4-20 mA position input, from PLC
2. Inhibit, from PLC

H4501 Outputs

1. In position, to PLC
2. Fault/Stall to PLC

Why IDC?

The H4501 control provides a simple, low cost alternative to the more expensive servo positioning systems.

Application 6: Valve Positioning

Customer's Objectives

A power company has had many problems with a mechanical control system which manipulates a large hydraulic valve. This valve regulates the flow of water through a Hydro-Electric power generator. As part of their plant modernization program, the customer wished to replace the old valve control system with a more reliable and more accurate system.

Valve Control Requirements

Load	1,200 lbs.
Speed	1.8 in/sec
Duty Cycle	50%
Repeatability	±0.001 inches
Programmability	Start-up Sequence Shutdown Sequence Run Sequence Emergency Shutoff
Interface	Stand-alone programming,

IDC Solution

Control	B8961 Servo Control
Actuators	EC Series
Accessories	RP2 end-of-travel sensors

The customer's computer commanded the B8961/EC actuator to the proper valve position using discrete outputs. Since there was a large amount of electrical noise present, a self-contained control system was desired which did not rely on RS-232C or analog communication from the master computer. The EC's cylinder's high speed and thrust ensure rapid response to changes in signal. Furthermore, in the event of system problems, an input was triggered, which runs and emergency shutdown procedure (program #98)

B8961 Inputs

1. Start, from computer

Using program select lines from computer:

1. Coarse Increment
2. Fine Increment
3. Coarse Decrement
4. Fine Decrement

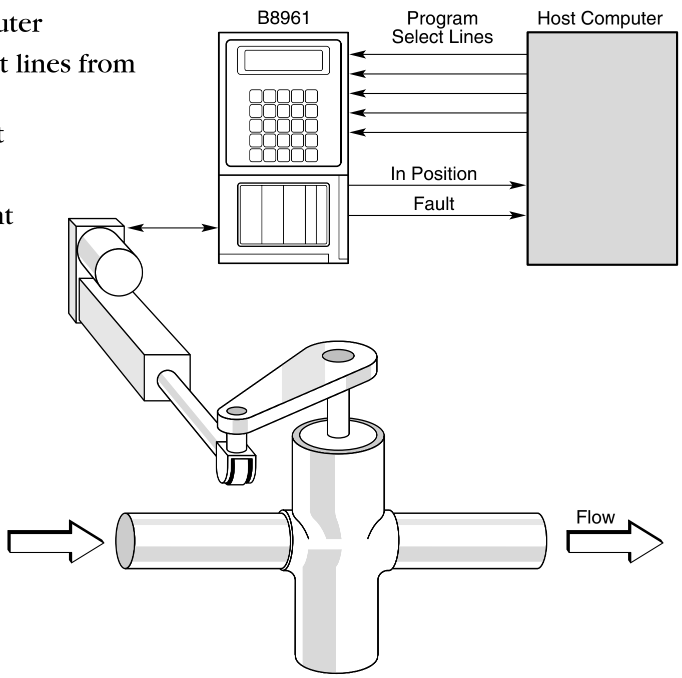
B8961 Outputs

1. In position
2. Fault

Why IDC?

High precision hydraulics were also considered. Because of the need to have an orderly shutdown in the event of hydraulic system failure, a separate power unit would have been needed. Furthermore, 0.001 inch repeatability proved to be difficult and costly to achieve.

The B8961's easy-to-use, yet powerful *IDEAL*™ motion programming, extensive I/O capability, and compact drive/control/power supply package encouraged this customer to choose this solution.



Glossary

Absolute Move
A move referenced from a fixed absolute zero position.

Acceleration
The change in velocity as a function of time, going from a lower speed to a higher speed.

Accuracy
An absolute measurement defining the difference between expected and actual position.

Acme Screw
A leadscrew which uses a threaded screw design with sliding surfaces between the screw and nut.

Actuator (Electric Cylinder)
A self contained leadscrew system which converts rotary motion (from a motor) to linear motion.

ASCII (American Standard Code for Information Interchange)
A code which assigns a number to each numeral and letter of the alphabet. This allows information to be transmitted between machines as a series of binary numbers.

Backdrive
Tendency of a cylinder to creep out of its set position due to an applied load or force.

Backlash
The amount of play (lost motion) between a set of moveable parts when changing the direction of travel. Typically seen in drive trains, leadscrews, & bearings.

Ball Screw
A leadscrew which uses a ball nut which houses one or more circuits of recirculating steel balls which roll between the nut and screw.

Baud Rate
The number of binary bits transmitted per second on a serial communication link such as RS232C.

BCD (Binary Coded Decimal)
A binary numbering system in which the decimal digits 0 to 9 are represented by a 4 bit binary number.

Binary
Numbering system in which the base is two, each number being expressed in the powers of two, by 0 or 1.

Bearing
A support device which allows a smooth, low friction motion between two surfaces loaded against each other.

Bushing
A cylindrical metal sleeve inserted into a machine part to reduce friction between moving parts.

Closed Loop
A positioning system which employs feedback information to regulate the output response.

Cogging
Motor torque variations which occur at low speeds due to a weak magnetic field.

Critical Speed
Rotational speed of a leadscrew at which vibrations (whipping) will occur.

Current
The flow of charge through a conductor.

Cycle
One complete extension and retraction of a cylinder.

Deceleration
The change in velocity as a function of time, going from a higher speed to a lower speed.

Drive Ratio
The ratio of motor revolutions per leadscrew revolution.

Drive Train
The arrangement by which the motor is coupled to the leadscrew. Typically provided by gears, timing belt/pulley or direct coupling.

Duty Cycle
The ratio of Motor on time and total cycle time within a given cycle of operation.

$$\text{Duty Cycle } (\%) = \frac{\text{Motor ON Time}}{\text{Total Cycle Time}} \times 100\%$$

Dwell Time
Time within a move cycle where no motion occurs

EEPROM (Electrically Erasable Programmable Read Only Memory)
Non-volatile data storage chip.

Efficiency
Ratio of output power vs. input power.

EMI (Electromagnetic Interference)
Electrical disturbances which interfere with proper transmission of electrical signals, also known as "Electrical noise".

Encoder
An electromechanical device which produces discrete electrical pulses directly related to the angular position of the input shaft, providing high resolution feedback data on position, velocity, and direction.

Force
The action of one body on another which tends to change the state of motion of that body. Typically described in terms of magnitude, direction, and point of application.

Friction
The resistance to motion of two surfaces that touch.

Helical Gear
Gears with teeth that spiral around the gear.

Incremental Move
A move referenced from the current set position.

Inertia
Property of an object that resists a change in motion. It is dependent on the mass and shape of the object. The greater an object's mass, the greater its inertia, and the more force is necessary to accelerate and decelerate.

Lead
The linear distance a nut on a leadscrew will travel with one revolution of the leadscrew.

Glossary

Leadscrew
Device which converts rotary motion to linear motion.

Mass
The quantity of matter that an object contains.

Microprocessor
A device that incorporates many or all functions of a computer in a single integrated circuit. Used to perform calculations and logic required to do motion or process control.

Moment (Load)
Rotational forces applied to a linear axis, typically expressed as yaw, pitch, and roll.

Motion Profile
A method of describing a move operation in terms of time, position, and velocity. Typically velocity is characterized as a function of time or distance which results in a triangular or trapezoidal profile.

Motor
A device which converts electrical energy into mechanical energy.

Non-Volatile Memory
Memory that does not lose information on loss of power.

Open Collector (NPN)
An output signal which is provided by a transistor where the “open collector output” acts like a switch closure to ground when activated.

Open Loop
A positioning system which does not employ feedback information.

Optically Coupled
An interface circuit that transmits a signal with no direct electrical connection except for the logic ground.

Optical Isolation
An interface circuit that transmits a signal with no direct electrical connection.

Overshoot
The amount by which a parameter being controlled exceeds the desired value. Typically referring to velocity or position in servo systems.

PID (Proportional, Integral, and Derivative)
Refers to a group of gain parameters used for tuning or optimizing the response of a closed loop positioning system.

Pitch
The number of revolutions a leadscrew must turn for the nut to travel one inch (single start only).

PLC (Programmable Logic Controller)
A programmable device which utilizes “ladder” logic to control a bank of inputs and outputs which are interfaced to external devices.

Power
How much work is done in a specific amount of time.

PWM (Pulse Width Modulation)
A type of adjustable frequency drive output where the drive’s output voltage is always a constant amplitude and by “chopping” (pulse width modulating) the average output power is controlled.

RAM (Random Access Memory)
A memory chip that can be read from and written to. Used as a medium for temporary information storage. Data is lost after power loss.

Repeatability
The ability of a positioning system to return to an exact location during operation (from the same direction with the same load and speed).

Resistance
The opposition to the flow of charge through a conductor.

Resolution
The smallest positioning increment achievable. In digitally programmed systems it is the smallest specifiable positioning increment.

Resonance
Oscillatory behavior in a mechanical body when operated or subjected to a periodic force occurring at its natural frequency.

ROM (Read Only Memory)
A memory chip that can be read but not altered.

RS232C
A method of Serial Communication where data is encoded and transmitted on a single line in a sequential time format.

Servo Motor
A motor which is used in closed loop systems where feedback is used to control motor velocity, position, or torque.

Spur Gear
Gears with teeth straight and parallel to the axis of rotation.

Stepper Motor
Motor which translates electrical pulses into precise mechanical movements. Through appropriate drive circuitry, controlling the rate and quantity of pulses will control the motor’s velocity and position.

Thrust
The measurement of linear force.

Torque
A measure of angular force which produces rotational motion.

TTL (Transistor-Transistor-Logic)
Refers to a family of integrated circuit devices used for control. Typically use voltage levels from 5-12 VDC.

Velocity (Speed)
The change in position as a function of time.

Voltage
Difference in electrical potential between two points.

Weight
Force of gravity acting on a body. Determined by multiplying the mass of the object by the acceleration due to gravity.

Conversion Tables

Torque

B / A	dyne-cm	gm-cm	oz-in	kg-cm	lb-in	N-m	lb-ft	kg-m
dyne-cm	1	1.019×10^{-2}	1.416×10^{-5}	1.0197×10^{-6}	8.850×10^{-7}	10^{-7}	7.375×10^{-6}	1.019×10^{-6}
gm-cm	980.665	1	1.388×10^{-2}	10^{-3}	8.679×10^{-4}	9.806×10^{-5}	7.233×10^{-5}	10^{-5}
oz-in	7.061×10^{4}	72.007	1	7.200×10^{-2}	6.25×10^{-2}	7.061×10^{-3}	5.208×10^{-3}	7.200×10^{-4}
kg-cm	9.806×10^{5}	1000	13.877	1	.8679	9.806×10^{-2}	7.233×10^{-2}	10^{-2}
lb-in	1.129×10^{6}	1.152×10^{3}	16	1.152	1	.112	8.333×10^{-2}	1.152×10^{-2}
N-m	10^{7}	1.019×10^{4}	141.612	10.197	8.850	1	.737	.102
lb-ft	1.355×10^{7}	1.382×10^{4}	192	13.825	12	1.355	1	.138
kg-m	9.806×10^{7}	10^{5}	1.388×10^{3}	100	86.796	9.806	7.233	1

Inertia (Rotary)

B / A	gm-cm²	oz-in²	gm-cm-s²	kg-cm²	lb-in²	oz-in-s²	lb-ft²	kg-cm-s²	lb-in-s²	lb-ft-s² or slug-ft-s²
gm-cm²	1	5.46×10^{-2}	1.01×10^{-3}	10^{-3}	3.417×10^{-4}	1.41×10^{-5}	2.37×10^{-6}	1.01×10^{-4}	8.85×10^{-7}	7.37×10^{-4}
oz-in²	182.9	1	.186	.182	.0625	2.59×10^{-3}	4.34×10^{-4}	1.86×10^{-4}	1.61×10^{-4}	1.34×10^{-5}
gm-cm-s²	980.6	5.36	1	.9806	.335	1.38×10^{-2}	2.32×10^{-3}	10^{-3}	8.67×10^{-4}	7.23×10^{-5}
kg-cm²	1000	5.46	1.019	1	.3417	1.41×10^{-2}	2.37×10^{-3}	1.019×10^{-3}	8.85×10^{-4}	7.37×10^{-5}
lb-in²	2.92×10^{3}	16	2.984	2.925	1	4.14×10^{-2}	6.94×10^{-3}	2.96×10^{-3}	2.59×10^{-3}	2.15×10^{-4}
oz-in-s²	7.06×10^{4}	386.08	72.0	70.615	24.13	1	.1675	7.20×10^{-2}	6.25×10^{-2}	5.20×10^{-3}
lb-ft²	4.21×10^{5}	2304	429.71	421.40	144	5.967	1	.4297	.3729	3.10×10^{-2}
kg-cm-s²	9.8×10^{5}	5.36×10^{3}	1000	980.66	335.1	13.887	2.327	1	.8679	7.23×10^{-2}
lb-in-s²	1.129×10^{4}	6.177×10^{3}	1.152×10^{3}	1.129×10^{3}	386.08	16	2.681	1.152	1	8.33×10^{-2}
lb-ft-s² or slug-ft²	1.355×10^{7}	7.41×10^{4}	1.38×10^{4}	1.35×10^{4}	4.63×10^{3}	192	32.17	13.825	12	1

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Conversion Tables

Angular Velocity

A \ B	deg/s	rad/s	rpm	rps
deg/s	1	**1.75×10^{-2}**	.167	**2.78×10^{-3}**
rad/s	57.3	**1**	9.55	**.159**
rpm	6	**.105**	1	**1.67×10^{-2}**
rps	360	**6.28**	60	**1**

Linear Velocity

A \ B	in/min	ft/min	in/sec	ft/sec	mm/sec	m/sec
in/min	1	**.0833**	.0167	**1.39×10^{-3}**	0.42	**4.2×10^{-4}**
ft/min	12	**1**	.2	**.0167**	5.08	**5.08×10^{-3}**
in/sec	60	**5**	1	**.083**	25.4	**.0254**
ft/sec	720	**60**	12	**1**	304.8	**.3048**
cm/sec	23.62	**1.97**	.3937	**.0328**	10	**0.01**
m	2362.2	**196.9**	39.37	**3.281**	1000	**1**

Abbreviated Terms

C	=	Celsius
cm	=	centimeter
F	=	Fahrenheit
ft	=	foot
g	=	gravity
gm	=	gram
gm(f)	=	gram force
HP	=	Horse Power
in	=	inch
kg	=	kilogram
kg(f)	=	kilogram force
KW	=	Kilowatt
lb(f)	=	pound force
lb(m)	=	pound mass
min	=	minute
mm	=	millimeter
m	=	meter
N	=	Newton
oz(f)	=	ounce force
oz(m)	=	ounce mass
rad	=	radians
rpm	=	revs per minute
rps	=	revs per second
s	=	seconds

Metric Prefixes

Name	Abbreviation		Multiple
Giga	G	10^{9}	1,000,000,000
Mega	M	10^{6}	1,000,000
Kilo	k	10^{3}	1,000
Hecto	h	10^{2}	100
deka	da	10^{1}	10
—	—	10^{0}	1
deci	d	10^{-1}	.1
centi	c	10^{-2}	.01
milli	m	10^{-3}	.001
micro	µ	10^{-6}	.000001
nano	n	10^{-9}	.000000001

Conversion Tables

(To convert from A to B, multiply by entry in table)

Length

A \ B	in	ft	micron (µm)	mm	cm	m
in	1	**0.0833**	2.54×10^{4}	**25.4**	2.54	**0.0254**
ft	12	**1**	3.048×10^{5}	**304.8**	30.48	**0.3048**
micron(µm)	3.937×10^{-7}	**3.281×10^{-6}**	1	**0.001**	1.0×10^{-4}	**1.0×10^{-6}**
mm	0.03937	**0.00328**	1000	**1**	0.1	**0.001**
cm	0.3937	**0.03281**	1.0×10^{4}	**10**	1	**0.01**
m	39.37	**3.281**	1.0×10^{6}	**1000**	100	**1**

Mass

A \ B	gm	kg	slug	lb(m)	oz(m)
gm	1	**.001**	6.852×10^{-5}	**2.205×10^{-3}**	.03527
kg	1000	**1**	6.852×10^{-2}	**2.205**	35.274
slug	14590	**14.59**	1	**32.2**	514.72
lb(m)	453.6	**.45359**	.0311	**1**	16
oz(m)	28.35	**.02835**	1.94×10^{-3}	**.0625**	1

Force

A \ B	lb(f)	N	dyne	oz(f)	kg(f)	gm(f)
lb(f)	1	**4.4482**	4.448×10^{5}	**16**	.45359	**453.6**
N	.22481	**1**	100.000	**3.5967**	.10197	——
dyne	2.248×10^{-6}	**.00001**	1	**3.59×10^{-5}**	——	**980.6**
oz(f)	.0625	**.27801**	2.78×10^{4}	**1**	.02835	**28.35**
kg(f)	2.205	**9.80665**	__	**35.274**	1	**1000**
gm(f)	2.205×10^{-3}	——	1.02×10^{-3}	**.03527**	.001	**1**

Note: lb(f) = 1slug x 1 ft/s² N = 1kg x 1 m/s² dyne = 1gm x 1 cm/s²

Power

A \ B	Watts	KW	HP(english)	HP(metric)	ft-lb/s	in-lb/s
Watts	1	**1×10^{-3}**	1.34×10^{-3}	**1.36×10^{-3}**	.74	**8.88**
KW	1000	**1**	1.34	**1.36**	738	**8880**
HP(english)	746	**.746**	1	**1.01**	550	**6600**
HP(metric)	736	**.736**	.986	**1**	543	**6516**
ft-lb/s	1.35	**1.36×10^{-3}**	1.82×10^{-3}	**1.84×10^{-3}**	1	**12**
in-lb/s	.113	**1.13×10^{-4}**	1.52×10^{-4}	**1.53×10^{-4}**	8.3×10^{-2}	**1**

NEMA and Material Specifications

Material Densities

	oz/in³	lb/in³	gm/cm³
Aluminum	1.57	.098	2.72
Brass	4.96	.31	8.6
Bronze	4.72	.295	8.17
Copper	5.15	.322	8.91
Plastic	.64	.04	1.11
Steel	4.48	.28	7.75
Hard Wood	.46	.029	.8
Soft Wood	.28	.018	.48

Mechanism Efficiencies

Acme Screw (Bronze Nut)	0.4
Acme Screw (Plastic Nut)	0.5
Ball Screw	0.9
Helical Gear	0.7
Spur Gear	0.6
Timing Belt/Pulley	0.9

Friction Coefficients

(Sliding)	μ_s
Steel on Steel	0.58
Steel on Steel (Greased)	0.15
Aluminum on Steel	0.45
Copper on Steel	0.36
Brass on Steel	0.40
Plastic on Steel	0.2
Linear Bearings	0.001

Temperature

°F = (1.8 x °C) + 32
°C = .555 (°F - 32)

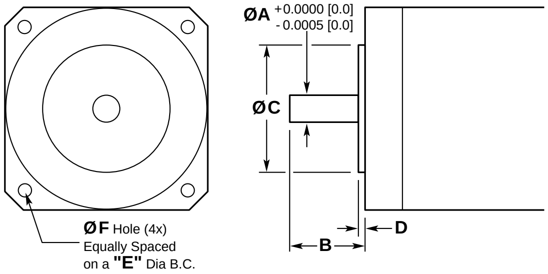
Gravity

(Acceleration Constant)
g = 386 in/s² = 32.2 ft/s² = 9.8 m/s²

NEMA Standard Motor Dimensions

Dimension (in)	NEMA 23	NEMA 34	NEMA 42
"A" Motor Shaft Diameter	0.250	0.375	0.625
"B" Motor Shaft Length*	0.810	1.250	1.380
"C" Pilot Diameter	1.500	2.875	2.186
"D" Pilot Length*	0.062	0.062	0.062
"E" Mounting Bolt Circle	2.625	3.875	4.950
"F" Bolt Hole Size	0.195	0.218	0.218

* These dimensions can be less than value indicated.



Selection Worksheet

For selection assistance, fax, to your local IDC Distributor or directly to IDC

Prepared By	Prepared For
Name ______________________	Name ______________________
Company ______________________	Company ______________________
Phone ______________________	Phone ______________________
Fax ______________________	Fax ______________________
Email ______________________	E-mail ______________________
Address ______________________	Address ______________________

User's primary business ____

____

Type of machine IDC product to be used on ____

____

Current IDC user? Yes ☐ No ☐

Project Time Frame	Volume Requirements
Proposal ____/____/____	Next 12 months: ______________________
Build prototype ____/____/____	Year 2: ______________________
In production ____/____/____	Year 3: ______________________

Action Required

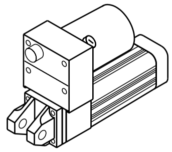
☐ Demo ☐ Price quotation

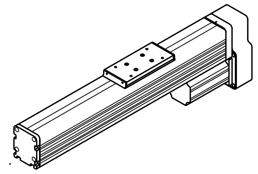
☐ Recommend product ☐ Call me to discuss

Please include drawings, comments or additional information on separate pages.

Linear Actuator Selection Data

☐ Electric Cylinder or ☐ Rodless Actuator





Loads

Payload

Weight ____________ lbs

☐ Payload Externally Supported, by ________________ (rails, etc.)

Hold Position:
☐ After move ☐ Power off

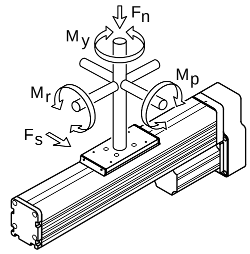
Carriage Loads
(Rodless only)

M_p ____________ in-lbs

M_r ____________ in-lbs

M_y ____________ in-lbs

Side Load ________ lbs



Orientation

☐ Vertical

☐ Horizontal

☐ Inclined_____ °
(angle from horizontal plane)

Motion

Travel

Stroke Length Required _____ in
(= usable travel distance + min. 2 inches for limit switches)

Shortest Move ____________ in

Max. Avaliable Stroke Length (in)
Electric Cylinders:

Speed (WCM=Worst-Case Move)

WCM Distance ____________ in

Time for WCM ____________ sec

or

Max. Speed ____________ in/sec

Min. Speed ____________ in/sec

Complete Move Profile Chart (see p. K-55)

Precision

Repeatability ____________ in

Accuracy ____________ in

Max. Backlash ____________ in

Resolution ____________ in

Straightness/Flatness _______ in

Thrust Calculation (See Engineering Section in IDC catalog for assistance)

Thrust

Thrust = $\text{Force}_{\text{ACCELERATED MASS}}$ + $\text{Force}_{\text{FRICTION}}$ + $\text{Force}_{\text{GRAVITY}}$ + $\text{Force}_{\text{EXTERNAL}}$

_____ lbs = ____________ lbs + _______ lbs + _______ lbs + _______ lbs

Duty Cycle/Life

Duty Cycle

Total Cycle Time __________ sec. Extend/Retract Cycles per day ___

Sum of Move Times _______ sec. Move Distance per cycle ________

Complete Move Profile Chart (see page K-55)

Required Life

Units: ☐ Inches ☐ Meters ☐ Cycles ☐ Months ☐ Years

Minimum Life ____________

Maintenance/Lube Interval ___

Environment

Operating Temperature

☐ Normal 32-140°F [0-60°C]

☐ High Temp.__________ °F / °C

☐ Low Temp.__________ °F / °C

Contaminants (Check all that apply)

Solid: ____________________

☐ non-abrasive ☐ coarse chips

☐ abrasive ☐ fine dust

Liquid: ____________________

☐ Dripping ☐ Non-corrosive

☐ Mist / Spray ☐ Corrosive

☐ Splashing

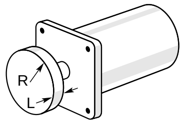
☐ High Pressure

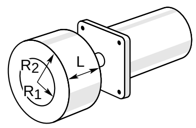
Conditions

☐ Washdown ☐ Outdoor ☐ Vacuum ☐ Cleanroom

Motor Selection Data

Direct Drive System





Radius (R): ____ in Weight of Cylinder ____ oz

Inner Radius (R1) ____ in Length (L) ____ in

Outer Radius (R2) ____ in

Density of Material ____ oz/in^3

Type of Material ____

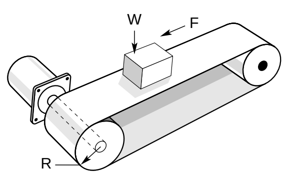
Will a gearbox be used? Yes / No / Not Sure

Orientation

☐ Horiz ☐ Vertical ☐ Incline: ____

Distance from Cylinder CL to Motor Face ____ in

Tangential Drive System



Radius (R) ____ in

Efficiency of belt or chain ____

Weight of load plus belt or chain (W) ____ lbf

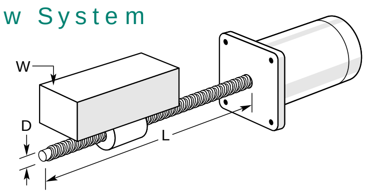
Weight of Pulleys ____ lbf Friction (F) ____ lbf

Will a gearbox be used? Yes / No / Not Sure

Belt Tension ____ lbf

Will pulleys be supported by external bearings? Yes / No

Leadscrew System



Orientation

☐ Horiz ☐ Vertical ☐ Incline: ____

☐ Ball Screw ☐ Acme Screw ☐ Other ____

Efficiency of Screw ____

Pitch of Screw ____ revs/in

Length of Screw (L) ____ in

Diameter of Screw (D) ____ in

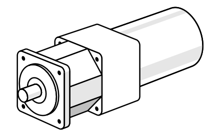
Weight of Load (W) ____ lbf

Running Friction Coefficient (Load/Surface) ____

Breakaway Force ____ lbf

Will a gearbox be used? Yes / No / Not Sure

Gearhead / Geartrain



Gear Ratio ____

Gearhead Inertia (reflected to pinion) ____ oz-in-sec^2

Efficiency ____ %

Radius of Pinion ____ in Radius of Driven Gear ____ in

Weight of Pinion ____ oz Weight of Driven Gear ____ oz

Radial load on output shaft ____ lbf

Distance of radial load from gearhead face ____ in

Axial load on output shaft ____ lbf

Move Requirements

Move Distance ____ inches ____ revs of motor

Move Time ____ secs

Required Motor Peak Speed ____ rev/s

Required Accel Time ____ secs

Required Decel Time ____ secs

Minimum Motor Speed ____ rev/s

Accuracy ____ arcminutes, degrees, or inches

Repeatability ____ arcminutes, degrees, or inches

Duty Cycle ____ (%) $\left(\frac{\text{Time in Motion}}{\text{Total Cycle Time}}\right)$

Cycle Time ____ secs

Maximum Continuous Time in Motion ____ secs

Cabling Requirements

Length of Motor / Encoder Cable Required ____ ft

Will cable be moving in application? Yes / No

Environment

Operating Temperature

☐ Normal 32-140°F [0-60°C]

☐ High Temp. ____ °F / °C

☐ Low Temp. ____ °F / °C

Contaminants (Check all that apply)

Solid: ____

☐ non-abrasive ☐ coarse chips

☐ abrasive ☐ fine dust

Liquid: ____

☐ dripping ☐ non-corrosive ☐ splashing

☐ mist / spray ☐ corrosive ☐ high pressure

Conditions

☐ Washdown ☐ Outdoor

☐ Vacuum ☐ Cleanroom

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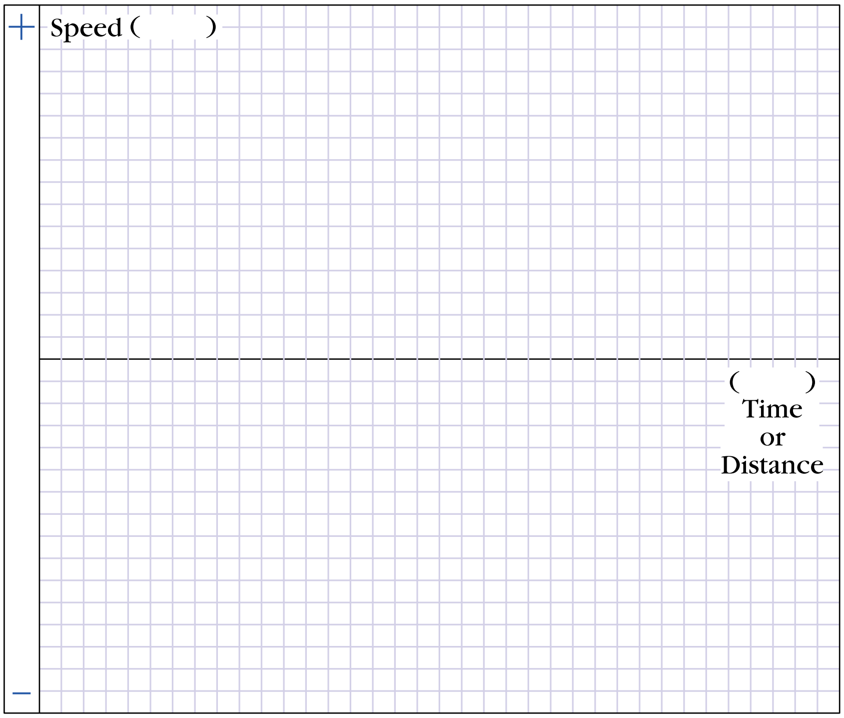
Motion Control Data

Motion Profile

Graph your most demanding cycle, include accel/decel, velocity and dwell times. You may also want to indicate load variations and I/O changes during the cycle. Label axes with proper scale and units.



Control Method

☐ Programmable ☐ External Control Signal

☐ Manual Jog ☐ Digital (Step & Direction) ☐ Analog Velocity

☐ Limit Switches ☐ Analog Torque ☐ Analog Position

Description of Application

Motor Type Preferred

☐ Servo ☐ Stepper

☐ Other ____________________

Axes of Motion

☐ Single ☐ Multiple # ________

☐ Synchronized

Interface

Host ☐ PLC ☐ Computer

☐ Analog I/O ☐ RS232

☐ Digital I/O Control

☐ Other ______________

Operator

☐ Keypad/LCD Display

☐ Pushbuttons

☐ Potentiometer/Joystick

☐ Thumbwheels

Supply Voltage

☐ 110 AC ☐ 220 AC

☐ Other ____________________

Feedback Required

☐ Encoder ☐ Linear Potentiometer

☐ Other ____________________

Input Functions

Output Functions