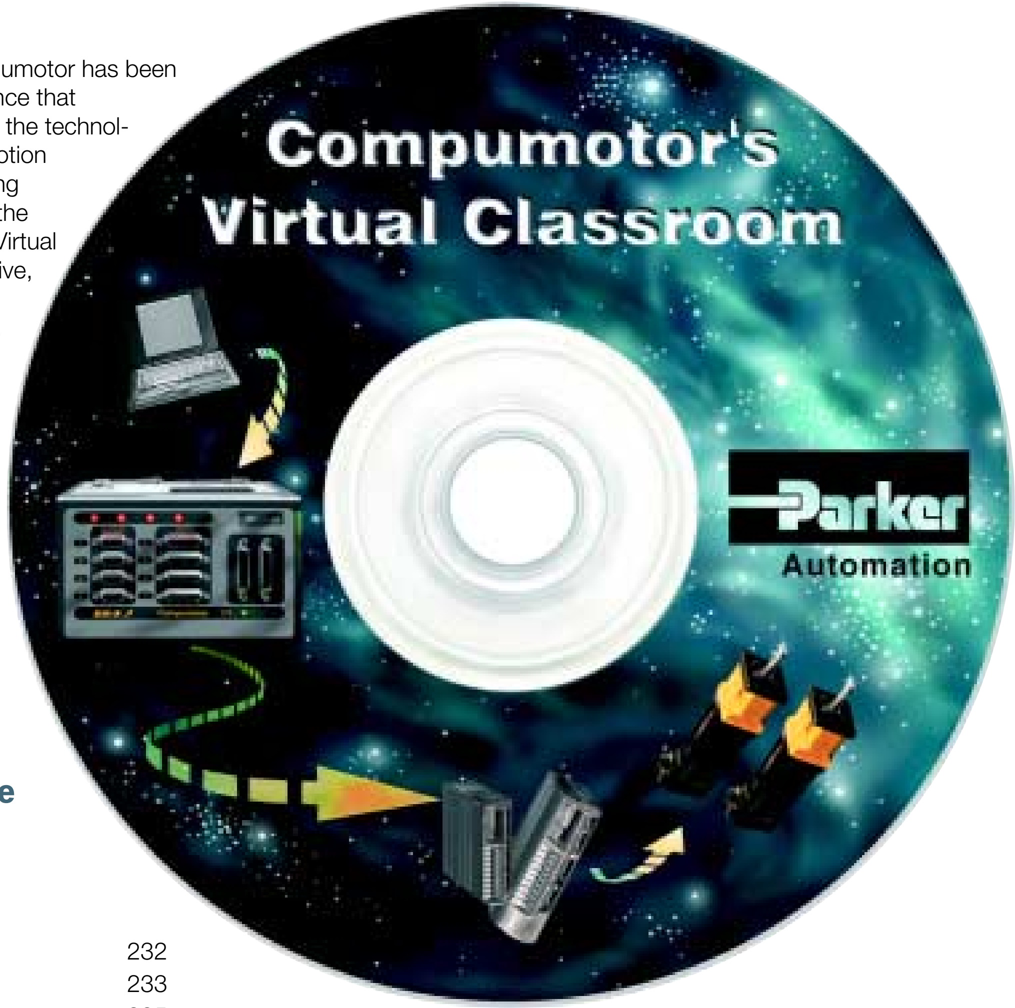
Engineering Reference Guide



Over the past twenty years, Compumotor has been developing an Engineering Reference that compiles important information on the technology and practical application of motion control. Compumotor's Engineering Reference was the foundation for the Virtual Classroom CD ROM. The Virtual Classroom CD-ROM is an interactive, multi-media tool that makes the Engineering Reference come alive. To request your free copy, go to www.compumotor.com.

For your convenience, we are providing an abbreviated version of the Engineering Reference that concentrates on the Sizing and Selection of a motion system. In the future, the complete Engineering Reference will be printed as a separate text to allow greater use and distribution to colleges and universities. Check the website for availability.

Engineering Reference Table of Contents:

Motor Sizing and Selection Process

Before you can select your motors drives and controls, you must define the mechanical system and determine the performance requirements of each axis of motion. The information in this section will help you:

1. Calculate move profile for each axis of motion.
2. Calculate total inertia for each axis of motion
3. Calculate torque required for acceleration for each axis of motion
4. Calculate max motor speed for each axis of motion
5. Review the application considerations to determine special

Sizing Step 1: Move Profile

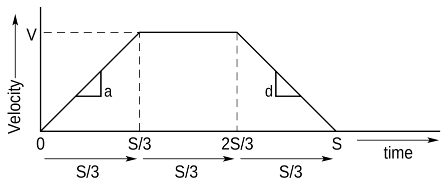
Before calculating torque requirements of an application, you need to know the velocities and accelerations needed. For those positioning applications where only a distance (X) and a time (S) to move that distance are known, the trapezoidal motion profile and formulas given below are a good starting point for determining your requirements. If velocity and acceleration parameters are already known, you can proceed to one of the specific application examples on the following pages.

Move distance X in time S.

Assume that:

1. Distance X/4 is moved in time S/3 (Acceleration)
2. Distance X/2 is moved in time S/3 (Run)
3. Distance X/4 is moved in time S/3 (Deceleration)

The graph would appear as follows:



The acceleration (a), velocity (v) and deceleration (d) may be calculated in terms of the knowns, X and S.

$$a = -d = \frac{2X}{t^2} = \frac{2\left(\frac{X}{4}\right)}{\left(\frac{S}{3}\right)^2} = \frac{\frac{X}{2} \times 9}{S^2} = \frac{4.5X}{S^2}$$

$$v = at = \frac{4.5X}{S^2} \times \frac{S}{3} = \frac{1.5X}{S}$$

Example

You need to move 6" in 2 seconds

$$a = -d = \frac{4.5\ (6\ \text{inches})}{(2\ \text{seconds})^2} = 6.75\ \frac{\text{inches}}{\text{second}^2}$$

$$v = \frac{1.5\ (6\ \text{inches})}{(2\ \text{seconds})} = 4.5\ \frac{\text{inches}}{\text{second}}$$

Common Move Profile Considerations

Distance: ______________ Inches of Travel ________________________ revolutions of motor

Move Time: ____ seconds

Accuracy: ____ arcminutes, degrees or inches

Repeatability: ____ arcseconds, degrees or inches

Duty Cycle

on tme: ____ seconds

off time: ____ seconds

Cycle Rate: ____ sec. min. hour

Sizing Step 2 and Step 3: Inertia and Torque Calculations

Leadscrew Drives

Leadscrews convert rotary motion to linear motion and come in a wide variety of configurations. Screws are available with different lengths, diameters, and thread pitches. Nuts range from the simple plastic variety to precision ground versions with recirculating ball bearings that can achieve very high accuracy.

The combination of microstepping and a quality leadscrew provides exceptional positioning resolution for many applications. A typical 10-pitch (10 threads per inch) screw attached to a 25,000 step/rev. motor provides a linear resolution of 0.000004" (4 millionths, or approximately 0.1 micron) per step. A flexible coupling should be used between the leadscrew and the motor to provide some damping. The coupling will also prevent excessive motor bearing loading due to any misalignment.

Leadscrew Application Data

Inertia of Leadscrews per Inch

Diameter In.	Steel	Brass	Alum.	
0.25	0.0017	0.0018	0.0006	oz-in^2
0.50	0.0275	0.0295	0.0094	oz-in^2
0.75	0.1392	0.1491	0.0478	oz-in^2
1.00	0.4398	0.4712	0.1512	oz-in^2
1.25	1.0738	1.1505	0.3691	oz-in^2
1.50	2.2266	2.3857	0.7654	oz-in^2
1.75	4.1251	4.4197	1.4180	oz-in^2
2.00	7.0372	7.5399	2.4190	oz-in^2
2.25	11.2723	12.0774	3.8748	oz-in^2
2.50	17.1807	18.4079	5.9059	oz-in^2

Diameter In.	Steel	Brass	Alum.	
2.75	25.1543	26.9510	8.6468	oz-in^2
3.00	35.6259	38.1707	12.2464	oz-in^2
3.25	49.0699	52.5749	16.8678	oz-in^2
3.50	66.0015	70.7159	22.6880	oz-in^2
3.75	86.9774	93.1901	29.8985	oz-in^2
4.00	112.5956	120.6381	38.7047	oz-in^2
4.25	143.4951	153.7448	49.3264	oz-in^2
4.50	180.3564	193.2390	61.9975	oz-in^2
4.75	223.9009	239.8939	76.9659	oz-in^2
5.00	274.8916	294.5267	94.4940	oz-in^2

Coefficients of Static Friction Materials

(Dry Contact Unless Noted)	μ_s
Steel on Steel	0.58
Steel on Steel (lubricated)	0.15
Aluminum on Steel	0.45
Copper on Steel	0.22
Brass on Steel	0.19
PTFE	0.04

Leadscrew Efficiencies Type	Efficiency (%) High	Median	Low
Ball-nut	95	90	85
Acme with metal nut*	55	40	35
Acme with plastic nut	85	65	50

*Since metallic nuts usually require a viscous lubricant, the coefficient of friction is both speed and temperature dependent.

Leadscrew Drives

Vertical or Horizontal Application:

ST	Screw type, ball or acme	ST = ______	
e	Efficiency of screw	e = ______	%
μ_S	Friction coefficient	μ_S = ______	
L	Length ofscrew	L = ______	inches
D	Diameter of screw	D = ______	inches
p	Pitch	p = ______	threads/inch
W	Weight of load	W = ______	lbs.
F	Breakaway force	F = ______	ounces
Directly coupled to the motor?		yes/no ______	
If yes, CT – Coupling type		______	
If no, belt & pulley or gears		______	
Radius of pulley or gear		______	inches
Gear: Number of teeth – Gear 1		______	
Number of teeth – Gear 2		______	
Weight of pulley or gear		______	ounces
Weight of belt		______	ounces

Leadscrew Formulas

The torque required to drive load W using a leadscrew with pitch (p) and efficiency (e) has the following components:

$$T_{Total} = T_{Friction} + T_{Acceleration}$$

$$T_{Friction} = \frac{F}{2\pi pe}$$

Where:

F = frictional force in ounces
p = pitch in revs/in
e = leadscrew efficiency

$F = \mu_s W$ for horizontal surfaces where μ_s = coefficient of static friction and W is the weight of the load. This friction component is often called “breakaway”.

Dynamic Friction: $F = \mu_D W$ is the coefficient to use for friction during a move profile. However, torque calculations for acceleration should use the worst case friction coefficient, μ_s.

$$T_{Accel} = \frac{1}{g} \left(J_{Load} + J_{Leadscrew} + J_{Motor}\right) \frac{\omega}{t}$$

$$\omega = 2\pi pv$$

$$J_{Load} = \frac{W}{(2\pi p)^2} \; ; \; J_{Leadscrew} = \frac{\pi L \rho R^4}{2}$$

Where:

T = torque, oz-in
ω = angular velocity, radians/sec
t = time, seconds
v = linear velocity, in/sec
L = length, inches
R = radius, inches
ρ = density, ounces/in^3
g = gravity constant, 386 in/sec^2

The formula for load inertia converts linear inertia into the rotational equivalent as reflected to the motor shaft by the leadscrew.

Problem

Find the torque required to accelerate a 200-lb steel load sliding on a steel table to 2 inches per second in 100 milliseconds using a 5 thread/inch steel leadscrew 36 inches long and 1.5 inches in diameter. Assume that the leadscrew has an Acme thread and uses a plastic nut. Motor inertia is given as 6.56 oz-in^2. In this example, we assume a horizontally oriented leadscrew where the force of gravity is perpendicular to the direction of motion. In non-horizontal orientations, leadscrews will transmit varying degrees of influence from gravity to the motor, depending on the angle of inclination. Compumotor Sizing Software automatically calculates these torques using vector analysis.

1. Calculate the torque required to overcome friction. The coefficient of static friction for steel-to-steel lubricant contact is 0.15. The median value of efficiency for an Acme thread and plastic nut is 0.65. Therefore:

$$F = \mu_s W = 0.15\,(200\text{ lb})\,\frac{(16\text{ oz})}{\text{lb}} = 480\text{ oz}$$

$$T_{Friction} = \frac{F}{2\pi pe} = \frac{480\text{ oz}}{\frac{2\pi}{\text{rev}} \times \frac{5\text{ rev}}{\text{in}} \times 0.65} = 23.51\text{ oz-in}$$

2. Compute the rotational inertia of the load and the rotational inertia of the leadscrew:

$$J_{Load} = \frac{W}{(2\pi p)^2} = \frac{200\ lb}{\frac{(2\pi 5)^2}{in^2}} \times \frac{16\ oz}{lb} = 3.24\ oz\text{-}in^2$$

$$J_{Leadscrew} = \frac{\pi L \rho R^4}{2} = \frac{\pi}{2}(36\ in)(4.48\ \frac{oz}{in^3})(0.75\ in)^4 = 80.16\ oz\text{-}in^2$$

3. The torque required to accelerate the load may now be computed since the motor inertia was given:

$$T_{Accel} = \frac{1}{g}\left(\frac{1}{e}J_{Load} + J_{Leadscrew} + J_{Motor}\right)\frac{\omega}{t}$$

$$\omega = 2\pi\left(\frac{5}{in}\right)\left(\frac{2\ in}{sec}\right) = \frac{20\pi}{sec}$$

$$= \frac{1}{386\ in/sec^2}[4.99 + 80.16 + 6.56(oz\text{-}in^2)]\ \frac{\frac{20\pi}{sec}}{0.1sec}$$

$$= 149\ oz\text{-}in$$

$$T_{Total} = T_{Friction} + T_{Accel}$$

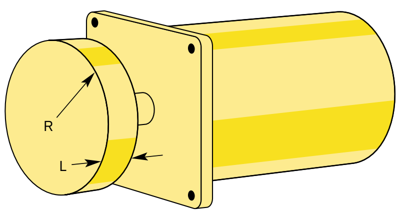
$$T_{Total} = 23.51\ oz\text{-}in + 149\ oz\text{-}in = 172.51\ oz\text{-}in$$

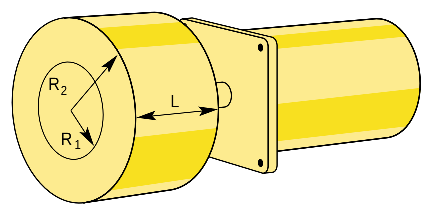
Directly Driven Loads

There are many applications where the motion being controlled is rotary and the low-speed smoothness and high resolution of a Compumotor system can be used to eliminate gear trains or other mechanical linkages. In direct drive applications, a motor is typically connected to the load through a flexible or compliant coupling. This coupling provides a small amount of damping and helps correct for any mechanical misalignment.

Direct drive is attractive when mechanical simplicity is desirable and the load being driven is of moderate inertia.

Direct Drive Formulas





R – Radius
R(1) – Inner radius
R(2) – Outer radius
L – Length
W – Weight of disc
ρ – Density/Material
g – Gravity constant

R = ______________ inches
R(1) = ______________ inches
R(2) = ______________ inches
L = ______________ inches
W = ______________ ounces
ρ = ______________ ounces/inch³
g = 386 in/sec²

Solid Cylinder (oz-in²)

Inertia: $J_{Load} = \frac{WR^2}{2}$

Where weight and radius are known

Inertia (oz-in²) $J_{Load} = \frac{\pi L \rho R^4}{2}$

Where ρ, the material density is known

Weight $W = \pi L \rho R^2$

Inertia may be calculated knowing either the weight and radius of the solid cylinder (W and R) or its density, radius and length (ρ, R and L.)

The torque required to accelerate any load is:

T (oz-in) = Ja

$a = \frac{\omega_2 - \omega_1}{t}$ = 2π (accel.) for Accel. in rps²

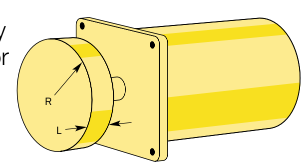
Where:

a = angular acceleration, radians/sec²
ω_2 = final velocity, radians/sec
ω_1 = initial velocity, radians/sec
t = time for velocity change, seconds
J = inertia in units of oz-in²

The angular acceleration equals the time rate of change of the angular velocity. For loads accelerated from zero, $\omega_1 = 0$ and $a = \frac{\omega}{t}$

$$T_{Total} = \frac{1}{g}(J_{Load} + J_{Motor})\frac{\omega}{t}$$

T_{Total} represents the torque the motor must deliver. The gravity constant (g) in the denominator represents acceleration due to gravity (386 in/sec²) and converts inertia from units of oz-in² to oz-in-sec².



Hollow Cylinder

$$J_{Load} = \frac{W}{2}(R^2_1 + R^2_2)$$

Where W, the weight, is known

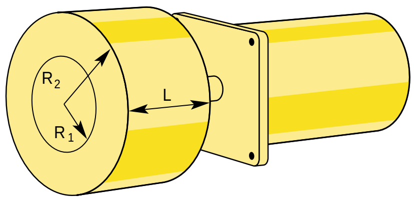
or

$$J_{Load} = \frac{\pi L \rho}{2}(R^4_2 - R^4_1)$$

Where r, the density, is known

$$W = \pi L \rho (R^2_2 - R^2_1)$$

$$T = \frac{1}{g}(J_{Load} + J_{Motor})\frac{\omega}{t}$$



Problem

Calculate the motor torque required to accelerate a solid cylinder of aluminum 5" in radius and 0.25" thick from rest to 2.1 radians/sec (0.33 revs/sec) in 0.25 seconds. First, calculate J_{Load} using the density for aluminum of 1.54 oz/in^3.

$$J_{Load} = \frac{\pi L \rho R^4}{2} = \frac{\pi \times 0.25 \times 1.54 \times 5^4}{2} = 378 \text{ oz-in}^2$$

Assume the rotor inertia of the motor you will use is 37.8 oz-in^2.

$$T_{Total} = \frac{1}{g}(J_{Load} + J_{Motor}) \times \frac{\omega}{t}$$

$$= \frac{1}{386} \times (378 + 37.8) \times \frac{2.1}{0.25}$$

$$= 9.05 \text{ oz-in}$$

Gear Drives

Traditional gear drives are more commonly used with step motors. The fine resolution of a microstepping motor can make gearing unnecessary in many applications. Gears generally have undesirable efficiency, wear characteristics, backlash, and can be noisy.

Gears are useful, however, when very large inertias must be moved because the inertia of the load reflected back to the motor through the gearing is divided by the square of the gear ratio.

In this manner, large inertial loads can be moved while maintaining a good load-inertia to rotor-inertia ratio (less than 10:1).

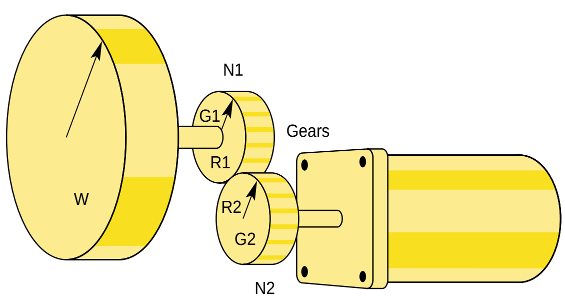
Gear Driven Loads

R – Radius
R(1) – Radius gear #1
R(2) – Radius gear #2
N(1) – Number of teeth G#1
N(2) – Number of teeth G#2
G – Gear ratio $\frac{N(1)}{N(2)}$
W – Weight of load
W(1) – Weight G#1
W(2) – Weight G#2
L – Length L =
F – Friction F =
BT – Breakaway torque

R = ______ inches
R(1) = ______ inches
R(2) = ______ inches
N(1) = ______
N(2) = ______
G = ______

W = ______ ounces
W(1) = ______ ounces
W(2) = ______ ounces
inches______
______

BT = ______ ounce/inches



Gear Drive Formulas

$$J_{Load} = \frac{W_{Load}}{2} R^2_{Load} \left(\frac{N_{Gear\,2}}{N_{Gear\,1}}\right)^2$$

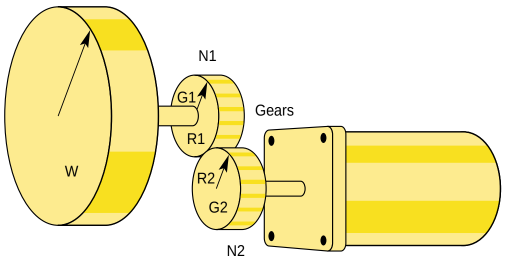
or

$$J_{Load} = \frac{\pi L_{Load} \rho_{Load}}{2} R^4_{Load} \left(\frac{N_{Gear\,2}}{N_{Gear\,1}}\right)^2$$

$$J_{Gear1} = \frac{W_{Gear1}}{2} R^2_{Gear1} \left(\frac{N_{Gear\,2}}{N_{Gear\,1}}\right)^2$$

$$J_{Gear2} = \frac{W_{Gear2}}{2} R^2_{Gear2}$$

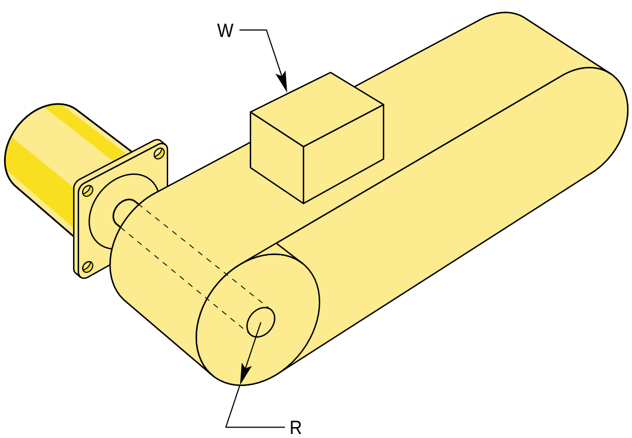
$$T_{Total} = \frac{1}{g} \left(J_{Load} + J_{Gear1} + J_{Gear2} + J_{Motor}\right) \frac{\omega}{t}$$



Where:

J = inertia, oz-in (gm-cm²) "as seen by the motor"
T = torque, oz-in (gm-cm)
W = weight, oz (gm)
R = radius, in. (cm)
N = number of gear teeth (constant)
L = length, in (cm)
ρ = density, oz/in³ (gm/cm³)
ω = angular velocity, radians/sec @ motor shaft
t = time, seconds
g = gravity constant, 386 in/sec²

Tangential Drives



R– Radius
W – Weight (include weight of belt or chain)
W(P) – Weight of pulley or material
F – Breakaway force
V – Linear velocity
CT – Coupling type
SL – Side load

R = ______________________ inches
W = ______________________ ounces
W(P) ______________________ ounces
F = ______________________ ounces
V = ______________________ inches/sec
CT = ______________________
SL = ______________________

Tangential Drive Formulas

$$T_{Total} = T_{Load} + T_{Pulley} + T_{Belt} + T_{Motor} + T_{Friction}$$

$$T_{Total} = \frac{1}{g}(J_{Load} + J_{Pulley} + J_{Belt} + J_{Motor})\frac{\omega}{t} + T_{Friction}$$

$$J_{Load} = W_L R^2$$

$$J_{Pulley} = \frac{W_p R^2}{2}$$

Remember to multiply by 2 if there are two pulleys.

$$J_{Belt} = W_B R^2$$

$$T_{Friction} = FR$$

$$\omega = \frac{V}{R}$$

Where:

- T = torque, oz-in (gm-cm)
- ω = angular velocity, radians/sec
- t = time, seconds
- W_L = weight of the load, oz
- W_P = pulley weight, oz
- W_B = belt or rack weight, oz
- F = frictional force, oz (gm)
- R = radius, in (cm)
- V = linear velocity
- g = gravity constant, 386 in/sec^2
- ρ = density, oz/in^3

Problem

What torque is required to accelerate a 5-lb load to a velocity of 20 inches per second in 10 milliseconds using a flat timing belt? The motor drives a 2-inch diameter steel pulley 1/2-inch wide. The timing belt weighs 12 oz. Load static friction is 30 ozs. Motor rotor inertia is 10.24 oz-in.^{2}

$$J_{Load} = W_L R^2 = 5\text{ lb} \times 16\ \frac{\text{oz}}{\text{lb}} \times (1\text{ in})^2 = 80\text{ oz-in}^2$$

$$J_{Pulley} = \frac{2(\pi L \rho R^4)}{2} = \pi \times 0.5\text{ in} \times (4.48\text{ oz/in}^3)(1\text{ in})^4$$

$$= 7.04\text{ oz-in}^2$$

$$J_{Belt} = W_B R^2 = 12\text{ oz }(1\text{ in})^2 = 12\text{ oz-in}^2$$

$$T_{Friction} = F \times R = 30\text{ oz} \times 1\text{ in} = 30\text{ oz-in}$$

$$\omega = \frac{V}{R} = 20\frac{\text{in}}{\text{sec}} \times \frac{1\text{ rad}}{1\text{ in}} = 20\ \frac{\text{rad}}{\text{sec}}$$

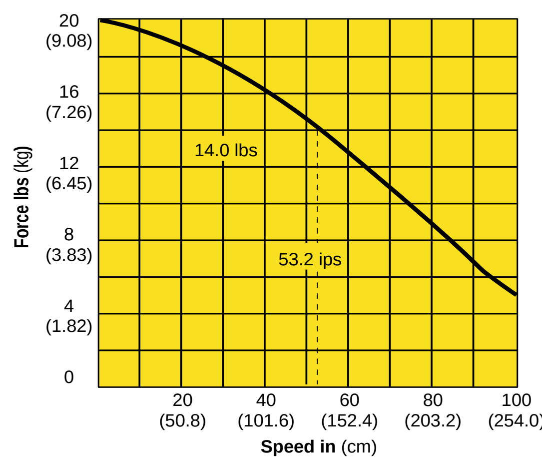
$$T_{Total} = \frac{1}{386}(80 + 7.04 + 12 + 10.24)\frac{20}{.01} + 30$$

$$T_{Total} = 596.2\text{ oz-in}$$

Sizing Step 4: Motor/Drive Selection

Based on Continuous Torque Requirements

Having calculated the torque requirements for an application, you can select the motor/drive suited to your needs. Microstepping motor systems (Gemini, ZETA Series OEM750 Series) have speed/torque curves based on continuous duty operation. To choose a motor, simply plot total torque vs. velocity on the speed/torque curve. This point should fall under the curve and allow approximately a 50% margin for safety. A ZETA106-178 and a ZETA83-135 curve are shown here. *Note: When selecting a ZETA or Gemini product, a 50% torque margin is not required.*



Example

Assume the following results from load calculations:

F =	25 oz-in	Friction torque
A =	175 oz-in	Acceleration torque
T =	200 oz-in	Total torque
V =	15 rev/sec	Maximum velocity

The ZETA83-135 has approximately 250 oz-in available at V max (25% more than required). The Zeta106-178 has 375 oz-in available, an 88% margin.

In this case, we would select the Zeta106-178 motor/drive to assure a sufficient torque margin to allow for changing load conditions.

Motor/Drive Selection

Based on peak torque requirements

Servo-based motor/drives have two speed/torque curves: one for continuous duty operation and another for intermittent duty. A servo system can be selected according to the total torque and maximum velocity indicated by the continuous duty curve. However, by calculating the root mean square (RMS) torque based on your duty cycle, you may be able to take advantage of the higher peak torque available in the intermittent duty range.

$$T_{RMS} = \sqrt{\frac{\sum Ti^2\, ti}{\sum ti}}$$

Where:

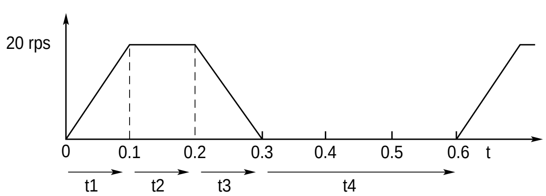
- Ti is the torque required over the time interval ti
- $\sum$ means "the sum of"

Example

Assume the following results from your load calculations.

T_F	= 25 oz-in	Friction Torque
T_A	= 775 oz-in	Acceleration Torque
T_T	= 800 oz-in	Total Torque
V_{max}	= 20 rps	Maximum Velocity

Motion Profile



Duty Cycle

Index 4 revs in 0.3 seconds, dwell 0.3 seconds then repeat.

If you look at the Zeta106-178 speed/torque curve, you'll see that the requirements fall outside the curve.

T_1 = Torque reqired to accelerate the load from zero speed to maximum speed ($T_F + T_A$)

T_2 = Torque required to keep the motor moving once it reaches max speed (T_F)

T_3 = Torque required to decelerate from max speed to a stop (T_A - T_F)

T_4 = Torque required while motor is sitting still at zero speed (Ø)

t_1 = Time spent accelerating the load

t_2 = Time spent while motor is turning at constant speed

t_3 = Time spent decelerating the load

t_4 = Time spent while motor is at rest

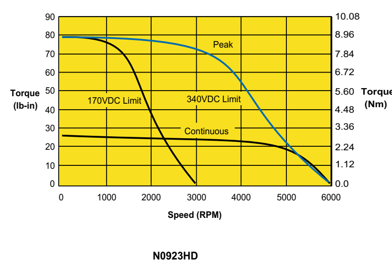
$$T_{RMS} = \sqrt{\frac{T_1{}^2 t_1 + T_2{}^2\, t_2 + T_3{}^2\, t_3 + T_4{}^2\, t_4}{t_1 + t_2 + t_3 + t_4}}$$

$$= \sqrt{\frac{(800)^2(.1) + (25)^2(.1) + (750)^2(.1) + (0)^2(.3)}{(.1) + (.1) + (.1) + (.3)}}$$

T_{RMS} = 447 oz. in.

Now plot T_{RMS} and T_T vs. T_{max} on the speed/torque curve.

The drawing below resembles the speed/torque curve for the NeoMetric 922 motor.

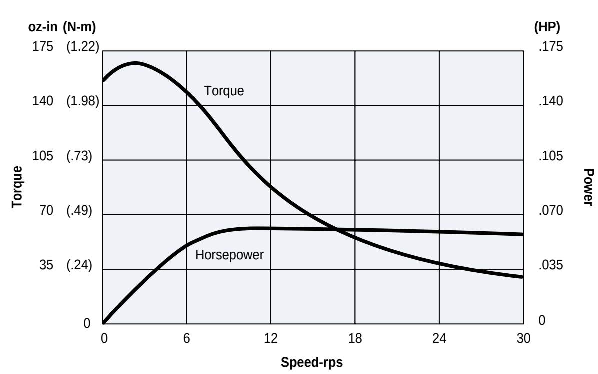


The NeoMetric 922 motor will meet the requirements. RMS torque falls within the continuous duty cycle and total torque vs. velocity falls within the intermittent range.

How to Use a Step Motor Horsepower Curve

Horsepower (HP) gives an indication of the motor's top usable speed. The peak or "hump" in a horsepower curve indicates a speed that gives maximum power. Choosing a speed beyond the peak of the HP curve results in no more power: the power attained at higher speeds is also attainable at a lower speed. Unless the speed is required for the application, there is little benefit to going beyond the peak as motor wear is faster at higher speeds.

Applications requiring the most power the motor can generate, not the most torque, should use a motor speed that is *just* below the peak of the HP curve.



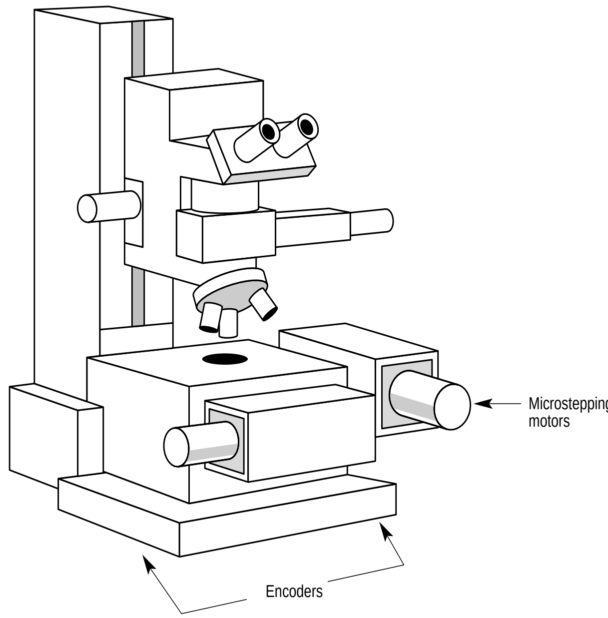
Parker Automation

Parker Hannifin Corporation
Compumotor Division

System Calculations

Microscope Positioning

Application Type: X/Y Point to Point
Motion: Linear
Description: A medical research lab automated their visual inspection process. Each specimen has an origin imprinted on the slide with all other positions referenced from that point. The system uses a HMI for data input from the operator, and determines the next data point based on previous readings. Each data point must be accurate to within 0.1 microns.



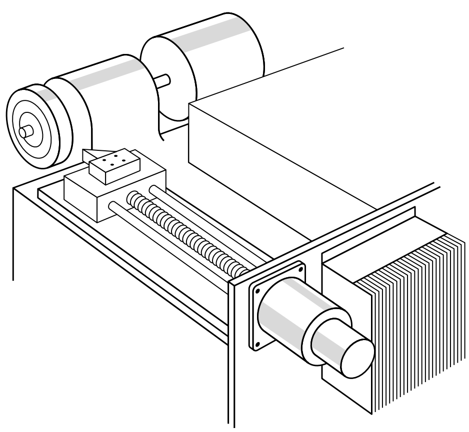
Machine Objectives

- Sub-micron positioning
- Specimen to remain still during inspection
- Low-speed smoothness (delicate equipment)
- Fast communications to the HMI

Motion Control Requirements

- High resolution
- Stepper (zero speed stability)
- Microstepping
- High speed interface

Compumotor Solution: Microstepping motors and drives, in conjunction with a precision ground 40 pitch leadscrew table, provide a means of sub-micron positioning with zero speed stability. To provide full X, Y, Z micro-scope control, it is necessary to use the 6K4 Controller which utilizes high speed ethernet communications to the HMI.



Other Leadscrew Drive Applications

- XY Plotters
- Facsimile transmission
- Tool bit positioning
- Cut-to-length machinery
- Back gauging
- Microscope drives
- Coil winders
- Slides
- Pick-and-Place machines
- Articulated arms

Precision Grinder

A bearing manufacturer replaced a bearing race finishing machine. The old machine utilized a two-stage grinding arrangement where one motor and gearbox provided a rough cut and a second motor with a higher ratio gearbox performed the finishing cut. The designer simplified the mechanics and eliminated one motor. He used a single leadscrew and exploited the wide speed range available with servos to perform both cuts. This was accomplished by moving a cutting tool mounted on the end of the leadscrew into the workpiece at two velocities; an initial velocity for the rough cut and a much reduced final velocity for the finish cut.

The torque required to accelerate the load and overcome the inertia of the load and the rotational inertia of the leadscrew varies, but does not exceed 80 oz-in. The torque necessary to overcome friction was measured with a torque wrench and found to be 20 oz-in. A servo motor with 144 oz-in of continuous torque was selected and provided adequate torque margin.

This grinder is controlled by a programmable controller (PC) and theenvironment requires that the electronics withstand a 60° C environment. A Gemini GV6 drive/controller provides the necessary velocities and accelerations. The speed change in the middle of the grinding operation is signaled to the PC via a limit switch at which time the PC programs the new velocity into the servo controller.

Application Considerations

Load characteristics, performance requirements, and coupling techniques need to be understood before the designer can select the best motor/drive for the job. While not a difficult process, several factors need to be considered for an optimum solution. A good designer will adjust the characteristics of the elements under his control –including the motor/drive and the mechanical transmission type (gears, lead screws, etc.) – to meet the performance requirements. Some important parameters are listed below.

Torque

Rotational force (ounce-inches or pound-inches) defined as a linear force (ounces) multiplied by a radius (inches). When selecting a motor/drive, the torque capacity of the motor must exceed the load. The torque a motor can provide may vary with its speed. Individual speed/torque curves should be consulted by the designer for each application.

Inertia

An object's inertia is a measure of its resistance to change in velocity. The larger the inertial load, the longer it takes a motor to accelerate or decelerate that load. However, the speed at which a motor rotates is independent of inertia. For rotary motion, inertia is proportional to the mass of the object being moved times the square of its distance from the axis of rotation.

Friction

All mechanical systems exhibit some frictional force, and this should be taken into account when sizing the motor, as the motor must provide torque to overcome any system friction. A small amount of friction is desirable since it can reduce settling time and improve performance.

Torque-to-Inertia Ratio

This number is defined as a motor's rated torque divided by its rotor inertia. This ratio is a measure of how quickly a motor can accelerate and decelerate its own mass. Motors with similar ratings can have different torque-to-inertia ratios as a result of varying construction.

Load Inertia-to-Rotor Inertia Ratio

For a high performance, relatively fast system, load inertia reflected to the motor should generally not exceed the motor inertia by more than 10 times. Load inertias in excess of 10 times the rotor inertia can cause unstable system behavior and inefficent power usage.

Torque Margin

Whenever possible, a motor/drive that can provide more motor torque than the application requires should be specified. This torque margin accommodates mechanical wear, lubricant hardening, and other unexpected friction. Resonance effects, while dramatically reduced with Compumotor's microstepping systems, can cause a stepper motor's torque to be slightly reduced at some speeds. Selecting a motor/drive that provides at least 50% margin for steppers, and 20% for servos, above the minimum needed torque is good practice.

Velocity

Because available torque varies with velocity, motor/drives must be selected with the required torque at the velocities needed by the application. In some cases, a change in the type of mechanical transmission used is needed to achieve the required performance.

Resolution

The positioning resolution required by the application will have an effect on the type of transmission used and the motor resolution. For instance, a leadscrew with 4 revolutions per inch and a 25,000-step-per-revolution motor/drive would give 100,000 steps per inch. Each step would then be 0.00001 inches.

Duty Cycle

Servo motors can produce peak torque for *short* time intervals as long as the RMS or average torque is within the motor's continuous duty rating. To take advantage of this feature, the application torque requirements over various time intervals need to be examined so RMS torque can be calculated.

Solving Duty Cycle Limitation Problems

Operating a motor beyond its recommended duty cycle results in excessive heat in the motor and drive. This can destroy the motor and drive package. The duty cycle may be increased by providing active cooling to the drive and the motor. A fan directed across the motor and another directed across the drive's heatsink will result in increased duty cycle capability.

Note: Motors will run at case temperatures up to 100°C (212°F)—temperatures hot enough to burn individuals who touch the motors.

To Improve Duty Cycle:

- Use a motor large enough for the application
- Mount the drive with heatsink fins running vertically
- Fan cool the motor
- Fan cool the drive
- Put the drive into REMOTE POWER SHUTDOWN when it isn't moving, or reduce current (Steppers Only)
- Reduce the peak current to the motor (if possible)

Application Considerations (continued)

Accuracy

An accuracy specification defines the maximum error in achieving a desired position. Some types of accuracy are affected by the application. For example, repeatability will change with the friction and inertia of the system the motor is driving.

Accuracy in a rotary motor is usually defined in terms of arc minutes or arc seconds (the terms arc second and arc minute are equivalent to second and minute, respectively). There are 1,296,000 seconds of arc in a circle. For example, an arc second represents 0.00291 inches of movement on a circle with a 50-foot radius. This is equivalent to about the width of a human hair.

Stepper Accuracy

There are several types of performance listed under Compumotor's motor specifications: repeatability, accuracy, relative accuracy, and hysteresis.

Repeatability

The motor's ability to return to the same position from the same direction. Usually tested by moving the motor one revolution, it also applies to linear step motors moving to the same place from the same direction. This measurement is made with the motor unloaded, so that bearing friction is the prominent load factor. It is also necessary to ensure the motor is moving to the repeat position from a distance of at least one motor pole. This compensates for the motor's hysteresis. A motor pole in a Compumotor is 1/50 of a revolution.

Accuracy

Also referred to as absolute accuracy, this specification defines the quality of the motor's mechanical construction. The error cancels itself over 360° of rotation, and is typically distributed in a sinusoidal fashion. This means the error will gradually increase, decrease to zero, increase in the opposite direction and finally decrease again upon reaching 360° of rotation. Absolute accuracy causes the size of microsteps to vary somewhat because the full motor steps that must be traversed by a fixed number of microsteps varies. The steps can be over or undersized by about 4.5% as a result of absolute accuracy errors.

Relative Accuracy

Also referred to as step-to-step accuracy, this specification tells how microsteps can change in size. In a perfect system, microsteps would all be exactly the same size, but drive characteristics and the absolute accuracy of the motor cause the steps to expand and contract by an amount up to the relative accuracy figure. The error is not cumulative.

Hysteresis

The motor's tendency to resist a change in direction. This is a magnetic characteristic of the motor, it is not due to friction or other external factors. The motor must develop torque to overcome hysteresis when it reverses direction. In reversing direction, a one revolution move will show hysteresis by moving the full distance less the hysteresis figure.

Servo & Closed-Loop Stepper Accuracy

Repeatability, accuracy and relative accuracy in servos and closed-loop stepper systems relate as much to their feedback mechanisms as they do to the inherent characteristics of the motor and drive.

Servos

Compumotor servos use either encoders or resolver feedback to determine their resolution and position. It is essentially the resolution of the device reading the feedback position that determines the highest possible accuracy in the system. The positional accuracy is determined by the drive's ability to move the motor to the position indicated by the resolver or encoder. Changes in friction, inertia, or tuning parameters will adversely affect the accuracy of the system.

Closed-Loop Steppers

Compumotor closed-loop stepper systems use an encoder to provide feedback for the control loop. The encoder resolution determines the system's accuracy. When enabled, the drive/controller attempts to position the motor within the specified deadband from the encoder. Typically, this means the motor will be positioned to within one encoder step. To do this satisfactorily, the motor must have more resolution than the encoder. If the step size of the motor is equal to or larger than the step size of the encoder, the motor will be unable to maintain a position and may become unstable. Compumotor recommends a minimum of a 4:1 ratio. In a system with adequate motor-to-encoder resolution, the motor is able to maintain one encoder step of accuracy with great dependability. This is a continuous process that will respond to outside events that disturb the motor's position.

Technology Comparison Summary

A comparison of stepper and servo motor technologies

Stepper motors, DC brush servos and brushless servos each have their respective benefits and drawbacks. No single motor technology is ideal in every application, despite what some manufacturers may claim. This section reviews the relative merits of each technology and lists the application types most appropriate to each.

Stepper motor benefits

- Lowest-cost solution
- A stepper motor will always offer the cheapest solution. If a stepper will do the job, use it.

Rugged and Reliable

Steppers are mechanically very simple and apart from the bearings (like in servos) there is nothing to deteriorate or fail.

No Maintenance

There are no brushes or other wearing parts requiring periodic checking or replacement.

Industry-standard ranges (Nema or metric)

Steppers are produced to standard flange and shaft sizes so finding a second source is not a problem.

Few environmental constraints

A stepper may be used in just about any environment, including in a vacuum. Special magnets may be needed if there are very large magnetic fields around, e.g. in evaporation chambers. Watch heat dissipation in a vacuum (there is no convection cooling).

Inherently failsafe

There are no conceivable faults within the drive to cause the motor to run away. Since current must be continually switched for continuous rotation most faults cause the motor to stop rotating. A brush motor is internally-commutated and can run away if continuous current is applied. A brushless servo relies on the feedback signal. If the signal is damaged, or absent the motor will run away.

Not easily demagnetized by excessive current

Owing to the perpendicular planes of the permanent magnet and alternating flux paths stepper motors will more often melt the windings before demagnetizing the permanent magnet, as would happen in a brushed motor.

Inherently stable at standstill

With DC flowing in the winding, the rotor will remain completely stationary. There is no tendency to jitter around an encoder or resolver position. This is useful in applications using vision systems.

Can be stalled indefinitely without damage

There is no increase in motor current as a result of a stall or jam as in a servo system. There is no risk of overdriving a stepper system due to large loads, or high speeds.

High continuous torque in relation to size

Compared with brushed servos of the same size, a stepper can produce greater continuous torque at low speeds.

Only 4 leads required

This minimizes the installed cost, particularly important in applications where connections are expensive (e.g. vacuum chambers).

Stepper motor drawbacks

Ringing, resonance and poor low speed smoothness

These are criticisms generally leveled at full-step drives. These problems may be almost wholly overcome by the use of a higher-resolution drive.

Undetected position loss in open loop

This should only occur under overload conditions and in many applications it causes few problems. When position lost must not go undetected, a check encoder may be fitted which then results in a very secure system. The encoder is not needed for positioning, only for confirmation. If a positioning encoder is desired a servo system should be used.

Uses full current at standstill

Since current is needed to produce holding torque, this increases motor heating at standstill.

Noisy at high speeds

The 50-pole rotor has a magnetic frequency of 2.5 kHz at 3000 rpm. Magneto-striction causes a correspondingly high-pitched sound.

Excessive iron losses at high speed

Again due to the high pole count, hysteresis and eddy current losses are higher than in a servo. A stepper is therefore not recommended for continuous operation at speeds approximately above 2000 rpm.

Brush Servo benefits

Low Cost

Brush servo motors are well developed an are inexpensive to produce.

Smooth rotation at low speeds

Brush motors are available which are specially designed for low speed smoothness with a large number of commutator segments. Brushed motors are the smoothest of the three discussed motor technologies.

Low cost drive

A DC brush drive can be made very economically since only a single bridge circuit is required.

No power used at standstill

With no static loads on the motor, no current is required to hold position.

Technology Comparison Summary (continued)

High peak torque available
In intermittent duty applications, particularly when positioning mainly-inertial loads, the motor can be overdriven beyond its continuous rating.

Flat speed-torque curve
Gives optimum performance with easily generated linear acceleration ramps.

Wide variety of types available
Brush motors are produced in many styles including very low inertia types for high dynamic applications.

High speed attainable
Brush servos are typically good for speeds up to 5000 rpm.

Brush servo drawbacks

Brush Maintenance
Not necessarily a problem if the motor is easily accessible, but a nuisance if the motor is not. Brushes also create dust as they wear; therefore limiting their use in clean rooms, and other environments where brush dust is not acceptable.

Problems in hazardous environments or a vacuum
Arcing at the brushes is fundamental to their operation.

Commutator limitations
Arduous duty cycles promote wear, and the mechanical commutation limits top speed. Very short repetitive moves, less than one revolution of the motor, may wear part of the commutator.

Poor thermal performance
All the heat is generated in the rotor, from which the thermal path to the outer casing is very inefficient.

Can be demagnetized
Excessive current can result in partial demagnetization of the motor.

Increased Installed cost
The installed cost of a servo system is higher than that of a stepper due to the requirement for feedback components.

Brushless servo benefits

Maintenance free
The lack of a commutator and brush system eliminates the need for periodic maintenance.

Good thermal performance
All the heat is generated in the stator where it can be efficiently coupled to the outside casing.

Very high speeds possible
There is no mechanical commutator to impose a speed limit, small motors are typically rated at up to 12,000 rpm.

Virtually no environment constraints
Due to the absence of brush gear, a brushless servo can be used in almost any environment. For high temperature operation, the use of a resolver feedback avoids any electronics buried in the motor.

Brushless servo drawbacks

Higher motor cost
This is largely due to the use of rare earth magnets

Drive more complex and costly
Six state, or trapezoidal drives, are not much more expensive than DC brush drives, but the higher performance sine wave drive can cost several times that of the DC brush drive.

Which technology to use?
The following section gives some idea of the applications that are particularly appropriate for each motor type, together with certain applications which are best avoided. It should be stressed that there is a wide range of applications which can be equally well met by more than one motor type, and the choice will often be dictated by customer preference, previous experience or compatibility with existing equipment.

With the increased requirement for intelligent drives, the real cost differential between brush and brushless servo systems is diminishing. In the majority of new applications the choice is therefore between stepper and brushless servo.

Cost conscious applications are always worth attempting with a stepper, as it will generally be hard to beat on cost. This is particularly true when the dynamic requirements are not severe, such as “setting” type applications like periodic adjustments on printing machines.

High Torque, low speed, continuous duty applications are appropriate for direct drive servos and frequently also for stepper motors. At low speeds the stepper is very efficient in terms of torque output relative to both size and input power. A typical example would be a metering pump for accurate flow control.

High torque, high speed, continuous duty applications suit a servo motor, and in fact, a stepper should be avoided in such applications because the high speed losses can lead to excessive motor heating. A DC motor can deliver greater continuous shaft power at high speeds than a stepper of the same frame size.

Short, rapid repetitive moves may demand the use of a servo if there are high dynamic requirements. However the stepper will offer a more economic solution when the requirements are more modest.

Positioning applications where the load is primarily inertia rather than friction are efficiently handled by a servo. The ability to overdrive a servo motor in intermittent duty allows a smaller motor to be used where the main torque demand only occurs during acceleration an deceleration.

Very arduous applications with a high dynamic duty cycle or requiring very high speeds will normally require a brushless servo.

Low speed, high smoothness applications are appropriate for a microstep system or a direct drive servo.

Applications in hazardous environments cannot normally use a brush servo. Depending on the demands of the load, use a stepper or brushless servo. Bear in mind that heat dissipation can present a problem in a vacuum.

Glossary of Terms

Absolute Positioning
Refers to a motion control system employing position feedback devices (absolute encoders) to maintain a given mechanical location.

Absolute Programming
A positioning coordinate referenced wherein all positions are specified relative to some reference, or "zero" position. This is different from incremental programming, where distances are specified relative to the current position.

AC Servo
A general term referring to a motor drive that generates sinusoidal shaped motor currents in a brushless motor wound as to generate sinusoidal back EMF.

Acceleration
The change in velocity as a function of time. Acceleration usually refers to increasing velocity and deceleration describes decreasing velocity.

Accuracy
A measure of the difference between expected position and actual position of a motor or mechanical system. Motor accuracy is usually specified as an angle representing the maximum deviation from expected position.

Ambient Temperature
The temperature of the cooling medium, usually air, immediately surrounding the motor or another device.
ASCII
American Standard Code for Information Interchange. This code assigns a number series of electrical signals to each numeral and letter of the alphabet. In this manner, information can be transmitted between machines as a series of binary numbers.

Bandwidth
A measure of system response. It is the frequency range that a control system can follow.

BCD
Binary Coded Decimal is an encoding technique used to describe the numbers 0 through 9 with four digital (on or off) signal lines. Popular in machine tool equipment, BCD interfaces are now giving way to interfaces requiring fewer wires, such as RS-232C.

Bit
Abbreviation of Binary Digit, the smallest unit of memory equal to 1 or 0.

Back EMF
The voltage produced across a winding of a motor due to the winding turns being cut by a magnetic field while the motor is operating. This voltage is directly proportional to rotor velocity and is opposite in polarity to the applied voltage. Sometimes referred to as counter EMF.

Block Diagram
A simplified schematic representing components and signal flow through a system.

Brushless DC Servo
A general term referring to a motor drive that generates trapezoidal shaped motor currents in a motor wound as to generate trapezoidal Back EMF.

Byte
A group of 8 bits treated as a whole, with 256 possible combinations of ones and zeros, each combination representing a unique piece of information.

Commutation
The switching sequence of drive voltage into motor phase windings necessary to assure continuous motor rotation. A brushed motor relies upon brush/bar contact to mechanically switch the windings. A brushless motor requires a device that senses rotor rotational position, feeds that information to a drive that determines the next switching sequence.

Closed Loop
A broadly applied term relating to any system where the output is measured and compared to the input. The output is then adjusted to reach the desired condition. In motion control, the term describes a system wherein a velocity or position (or both) transducer is used to generate correction signals by comparison to desired parameters.

Critical Damping
A system is critically damped when the response to a step change in desired velocity or position is achieved in the minimum possible time with little or no overshoot.

Crossover Frequency
The frequency at which the gain intercepts the 0 dB point on a Bode plot (used in reference to the open-loop gain plot).

Daisy-Chain
A term used to describe the linking of several RS-232C devices in sequence such that a single data stream flows through one device and on to the next. Daisy-chained devices usually are distinguished by device addresses, which serve to indicate the desired destination for data in the stream.

Damping
An indication of the rate of decay of a signal to its steady state value. Related to settling time.

Damping Ratio
Ratio of actual damping to critical damping. Less than one is an underdamped system and greater than one is an overdamped system.

Dead Band
A range of input signals for which there is no system response.

Decibel
A logarithmic measurement of gain. If G is a system's gain (ratio of output to input), then 20 log G = gain in decibels (dB).

Detent Torque
The minimal torque present in an unenergized motor. The detent torque of a step motor is typically about 1% of its static energized torque.

Direct Drive Servo
A high-torque, low-speed servo motor with a high resolution encoder or resolver intended for direct connection to the load without going through a gearbox.

Duty Cycle
For a repetitive cycle, the ratio of on time to total cycle time.

$$\textbf{Duty cycle} = \frac{\text{On Time}}{(\text{On Time} + \text{Off Time})}$$

Efficiency
The ratio of power output to power input.

Electrical Time Constant
The ratio of armature inductance to armature resistance.
Encoder
A device that translates mechanical motion into electronic signals used for monitoring position or velocity.

Glossary of Terms (continued)

Ethernet
High speed computer interface used in a majority of Network systems today.

Fiedbus
A high speed serial interfaces used to comunicate between devices.

Form Factor
A term used to describe wether a system is a stand alone, bus based or PLC based.

Friction
A resistance to motion. Friction can be constant with varying speed (Coulomb friction) or proportional to speed (viscous friction).

Gain
The ratio of system output signal to system input signal.

Holding Torque
Sometimes called static torque, it specifies the maximum external force or torque that can be applied to a stopped, energized motor without causing the rotor to rotate continuously.

Home
A reference position in a motion control system derived from a mechanical datum or switch. Often designated as the "zero" position.

Hysteresis
The difference in response of a system to an increasing or a decreasing input signal.

IEEE-488
A digital data communications standard popular in instrumentation electronics. This parallel interface is also known as GPIB, or General Purpose Interface Bus.

Incremental Motion
A motion control term that describes a device that produces one step of motion for each step command (usually a pulse) received.

Incremental Programming
A coordinate system where positions or distances are specified relative to the current position.

Inertia
A measure of an object's resistance to a change in velocity. The larger an object's inertia, the larger the torque that is required to accelerate or decelerate it. Inertia is a function of an object's mass and its shape.

Inertial Match
For most efficient operation, the system coupling ratio should be selected so that the reflected inertia of the load is equal to the rotor inertia of the motor.

Indexer
See PMC.

I/O
Abbreviation of input/output. Refers to input signals from switches or sensors and output signals to relays, solenoids etc.

Limits
Properly designed motion control systems have sensors called limits that alert the control electronics that the physical end of travel is being approached and that motion should stop.

Logic Ground
An electrical potential to which all control signals in a particular system are referenced.

Mid-range Instability
Designates the condition resulting from energizing a motor at a multiple of its natural frequency (usually the third orders condition). Torque loss and oscillation can occur in underdamped open-loop systems.

Microstepping
An electronic control technique that proportions the current in a step motor's windings to provide additional intermediate positions between poles. Produces smooth rotation over a wide speed range and high positional resolution.

Multitasking
The ability to run two or more programs at the same time.

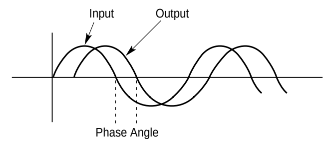
Open Collector
A term used to describe a signal output that is performed with a transistor. An open collector output acts like a switch closure with one end of the switch at ground potential and the other end of the switch accessible.

Open Loop
Refers to a motion control system where no external sensors are used to provide position or velocity correction signals.

Opto-isolated
A method of sending a signal from one piece of equipment to another without the usual requirement of common ground potentials. The signal is transmitted optically with a light source (usually a Light Emitting Diode) and a light sensor (usually a photosensitive transistor). These optical components provide electrical isolation.

Parallel
Refers to a data communication format wherein many signal lines are used to communicate more than one piece of data at the same time.

Phase Angle
The angle at which the steady state input signal to a system leads the output signal.



Phase Margin
The difference between 180° and the phase angle of a system at its crossover frequency.

PLC
Programmable logic controller; a machine controller that activates relays and other I/O units from a stored program. Additional modules support motion control and other functions.

PMC
Programmable motion controller, primarily designed for single- or multi-axis motion control with I/O as an auxiliary function.

Pulse Rate
The frequency of the step pulses applied to a motor driver. The pulse rate multiplied by the resolution of the motor/drive combination (in steps per revolution) yields the rotational speed in revolutions per second.

PWM
Pulse Width Modulation. A method of controlling the average current in a motors phase windings by varying the on-time (duty cycle) of transistor switches.

Ramping
The acceleration and deceleration of a motor. May also refer to the change in frequency of the applied step pulse train.

Glossary of Terms (continued)

Rated Torque
The torque producing capacity of a motor at a given sped. This is the maximum torque the motor can deliver to a load and is usually specified with a torque/speed curve.

Regeneration
Usually refers to a circuit in a drive amplifier that accepts and drains energy produced by a rotating motor either during deceleration or free-wheel shutdown.

Registration Move
Changing the predefined move profile that is being executed, to a different predefined move profile following receipt of an input or interrupt.

Repeatability
The degree to which the positioning accuracy for a given move performanced repetitively can be duplicated.

Resolution
The smallest positioning increment that can be achieved. Frequently defined as the number of steps required for a motor's shaft to rotate one complete revolution.

Resolver
A feedback device with a construction similar to a motor's construction (stator and rotor). Provides velocity and position information to a drive's microprocessor or DSP to electronically commutate the motor.

Resonance
Designates the condition resulting from energizing a motor at a frequency at or close to the motor's natural frequency. Lower resolution, open-loop systems will exhibit large oscillations from minimal input.

Ringing
Oscillation of a system following a sudden change in state.

RMS Torque
For an intermittent duty cycle application, the RMS Torque is equal to the steady- state torque that would produce the same amount of motor heating over long periods of time.

$$T_{RMS} = \sqrt{\frac{\Sigma(Ti^2\ ti)}{\Sigma\ ti}}$$

Where: Ti = Torque during interval i
ti = Time of interval i

RS-232C
A data communications standard that encodes a string of information on a single line in a time sequential format. The standard specifies the proper voltage and time requirements so that different manufacturers devices are compatible.

Servo
A system consisting of several devices which continuously monitor actual information (position, velocity), compares those values to desired outcome and makes necessary corrections to minimize that difference.

Static Torque
The maximum torque available at zero speed.

Step Angle
The angle the shaft rotates upon receipt of a single step command.

Stiffness
The ability to resist movement induced by an applied torque. Is often specified as a torque displacement curve, indicating the amount a motor shaft will rotate upon application of a known external force when stopped.

Synchronism
A motor rotating at a speed correctly corresponding to the applied step pulse frequency is said to be in synchronism. Load torques in excess of the motor's capacity (rated torque) will cause a loss of synchronism. The condition is not damaging to a step motor.

Torque
Force tending to produce rotation.

Useful Technical Conversion Data

Rotary Inertia Conversion Table

Don't confuse mass-inertia with weight-inertia: $\text{mass inertia} = \dfrac{\text{wt. inertia}}{g}$

To convert from A to B, multiply by entry in Table.

A \ B	kg-m²	kg-cm²	g-cm²	kg-m-sec²	kg-cm-sec²	g-cm-sec²	oz-in²	oz-in-s²	lb-in⁻²	lb-in-s²	lb-ft²	lb-ft-s² (slug-ft⁻²)
kg-m²	1	10^{4}	10^{7}	0.101972	10.1972	1.01972-10^{4}	5.46744-10^{4}	1.41612-10^{2}	3.41716-10^{3}	8.85073	23.7303	0.737556
kg-cm²	10^{-4}	1	10^{3}	1.01972-10^{-5}	1.01972-10^{-3}	1.01972	5.46744	1.41612-10^{-2}	0.341716	8.85073-10^{-4}	2.37303-10^{-3}	7.37556-10^{-5}
g-cm²	10^{-7}	10^{-3}	1	1.01972-10^{-8}	1.01972-10^{-6}	1.01972-10^{-3}	5.46744-10^{-3}	1.41612-10^{-5}	3.41716-10^{-4}	8.85073-10^{-7}	2.37303-10^{-6}	7.37556-10^{-8}
kg-m-s²	9.80665	9.80665-10^{4}	9.80665-10^{7}	1	10^{2}	10^{5}	5.36173-10^{5}	1.38874-10^{3}	3.35109-10^{4}	86.796	2.32714-10^{2}	7.23295
kg-cm-s²	9.80665-10^{-2}	9.80665-10^{2}	9.80665-10^{5}	10^{-2}	1	10^{3}	5.36173-10^{3}	13.8874	3.35109-10^{2}	0.86796	2.32714	7.23295-10^{-2}
g-cm-s²	9.80665-10^{-5}	0.980665	9.80665-10^{2}	10^{-5}	10^{-3}	1	5.36173	1.38874-10^{-2}	0.335109	8.6796-10^{-4}	2.32714-10^{-3}	7.23295-10^{-5}
oz-in²	1.82901-10^{-5}	0.182901	1.82901-10^{2}	1.86507-10^{-6}	1.86507-10^{-4}	0.186507	1	2.59001-10^{-3}	6.25001-10^{-2}	1.618801-10^{-4}	4.34029-10^{-4}	1.349-10^{-5}
oz-in-s²	7.06154-10^{-3}	70.6154	7.06154-10^{4}	7.20077-10^{-4}	7.20077-10^{-2}	72.0077	3.86085-10^{2}	1	24.13044	6.24998-10^{-2}	0.167572	5.20828-10^{-3}
lb-in²	2.92641-10^{-4}	2.92641	2.92641-10^{3}	2.98411-10^{-5}	2.98411-10^{-3}	2.98411	16	4.14415-10^{-2}	1	2.59009-10^{-3}	6.94445-10^{-3}	2.15839-10^{-4}
lb-in-s²	0.112985	1.12985-10^{3}	1.12985-10^{6}	1.15213-10^{-2}	1.15213	1.15213-10^{3}	6.17739-10^{3}	16	3.86087-10^{2}	1	2.681176	8.33327-10^{-2}
lb-ft²	4.21403-10^{-2}	4.21403-10^{2}	4.21403-10^{5}	4.29711-10^{-3}	0.429711	4.29711-10^{2}	2.304-10^{3}	5.96758	144	0.372973	1	3.10808-10^{-2}
lb-ft-s² (slug ft²)	1.35583	1.35583-10^{4}	1.35583-10^{7}	0.138256	13.8256	1.38256-10^{4}	7.41292-10^{4}	192	4.63308-10^{3}	12	32.1742	1

Torque Conversion Table

To convert from A to B, multiply by entry in Table.

A \ B	N-m	N-cm	dyn-cm	kg-m	kg-cm	g-cm	oz-in	ft-lbs	in-lbs
N-m	1	10^{2}	10^{7}	0.101972	10.1972	1.01972-10^{4}	141.612	0.737561	8.92172
N-cm	10^{-2}	1	10^{5}	1.01972-10^{-3}	0.101972^{-3}	1.01972-10^{2}	1.41612	7.37561-10^{-3}	8.92172-10^{-2}
dyn-cm	10^{-7}	10^{-5}	1	1.01972-10^{-8}	1.01972-10^{-6}	1.01972-10^{-3}	1.41612-10^{-5}	7.37561-10^{-8}	8.92172-10^{-7}
kg-m	9.80665	9.80665-10^{2}	9.80665-10^{7}	1	10^{2}	10^{5}	1.38874-10^{3}	7.233	87.4922
kg-cm	9.80665-10^{-2}	9.80665	9.80665-10^{5}	10^{-2}	1	10^{3}	13.8874	7.233-10^{-2}	.874922
g-cm	9.80665-10^{-5}	9.80665-10^{-3}	9.80665-10^{2}	10^{-5}	10^{-3}	1	1.38874-10^{-2}	7.233-10^{-5}	8.74992-10^{-4}
oz-in	7.06155-10^{-3}	0.706155	7.06155-10^{4}	7.20078-10^{-4}	7.20078-10^{-2}	72,0078	1	5.20832-10^{-3}	6.30012-10^{-2}
ft-lbs	1.35582	1.35582-10^{2}	1.35582-10^{7}	0.138255	13.8255	1.38255-10^{4}	192	1	12.0962
in-lbs	0.112086	11.2086	1.12086-10^{6}	1.14296-10^{-2}	1.14296	1.15827-10^{3}	15.8727	8.26703-10^{-2}	1

Force Conversion

1 lb_f = 4.45 N
1N = 0.225 lb
1 kg = 2.2 lb
1 kg = 9.8 N

Lengths

1 in = 25.4 mm
1 m = 39.37 in (~40 in)
1 mm = 0.03937 in

Densities of Common Materials

Material	oz/in³	gm/cm³
Aluminum (cast or hard-drawn)	1.54	2.66
Brass (cast or rolled 60% CU; 40% Zn)	4.80	8.30
Bronze (cast, 90% CU; 10% Sn)	4.72	8.17
Copper (cast or hand-drawn)	5.15	8.91
Plastic	0.64	1.11
Steel (hot or cold rolled, 0.2 or 0.8% carbon)	4.48	7.75
Hard Wood	0.46	0.80
Soft Wood	0.28	0.48

Calculate Horsepower

$$\text{Horsepower} = \frac{\text{Torque x Speed}}{16{,}800}$$

Torque = oz-in

Speed = revolutions per second

* The horsepower calculation uses the torque available at the specified speed

1 Horsepower = 746 watts

Most tables give densities in lb/ft³. To convert to oz/in³ *divide* this value by 108. To convert lb/ft³ to gm/cm³ *divide* by 62.5. The conversion from oz/in³ to gm/cm³ is performed by multiplying oz/in³ by 1.73.

Reference: *Elements of Strength of Materials*, S. Timoshinko and D.H. Young, pp. 342-343.

Application Examples with 6000 Based Software

Position-Based Following

Position following is a standard feature on all 6K and 6000 series controllers. Position following is commonly used in applications such as:

- Electronic gearbox
- Flying Cutoff
- Random Timing Infeed
- Web Processing
- Product Spacing
- Cut-to-Length

Positioning following can include continuous, preset, and registration-like moves in which the velocity is replaced with a ratio.

Position Following allows for these capabilities and more:

- The slave may follow in either direction and change ratio while moving.
- Phase shifts are allowed during motion.
- Ratio changes or new moves may be dependent on master position or based on receipt of a trigger input.
- A slave axis may perform following moves or normal time-based moves in the same application because following can be enabled and disabled at will.

In position following, acceleration ramps between ratios are dependent upon a specified master distance. Product cycles can be easily specified with the “master cycle concept.”

Continuous Process Automation

In any continuous process, throughput can often be increased if familiar motion functions are done on moving targets as opposed to stopping the process. For example, a conveyor belt carries trays of parts which are to be unloaded. If a controller could detect and track the motion of the tray and then perform pick and place operations on those parts without stopping the process, overall efficiency would be dramatically increased.

The Position Following feature of the 6K and 6000 controllers has the capability to solve continuous automation process applications. Controller capability requirements range from simple concepts, such as electronic gearbox, track ball and slave feed-to-length, to complex changes of ratio based on master position. Common applications include packaging, printing, continuous cut-to-length, and coil winding.

The following examples represent common applications in packaging and printing. Each application demonstrates the advance and retard capabilities or the use of the Periodic Master/Slave Synchronization features of the 6K 6000 controllers. The latter feature is important for applications in which periodic operations must occur in intervals which are not perfectly repeatable. For these, the master and slave must be resynchronized every cycle.

The features of the 6K and 6000 controller’s following package that are used here are not specific to these applications. They can be combined in a variety of ways to solve almost any application which requires coordination and synchronization of multiple axes. Even if you do not recognize your exact requirements in these examples, there is a very good chance that your synchronization application could be solved using these features.

Electronic Gearbox

Two or more axes are “electronically” geared together to maintain an exact relationship between the different axes and allow any mechanical gearing or linkages to be removed. Mechanical backlash and wear are eliminated.

The Following Applications are Highlighted:

Random Timing Infeed

Randomly spaced product on a conveyor is adjusted so that it is placed precisely on another conveyor for a later process. See following description.

Product Spacing

A method of spacing out product that may have been cut in a flying cutoff process without stopping the product. Production times are reduced as the product does not have to be stopped to be separated.

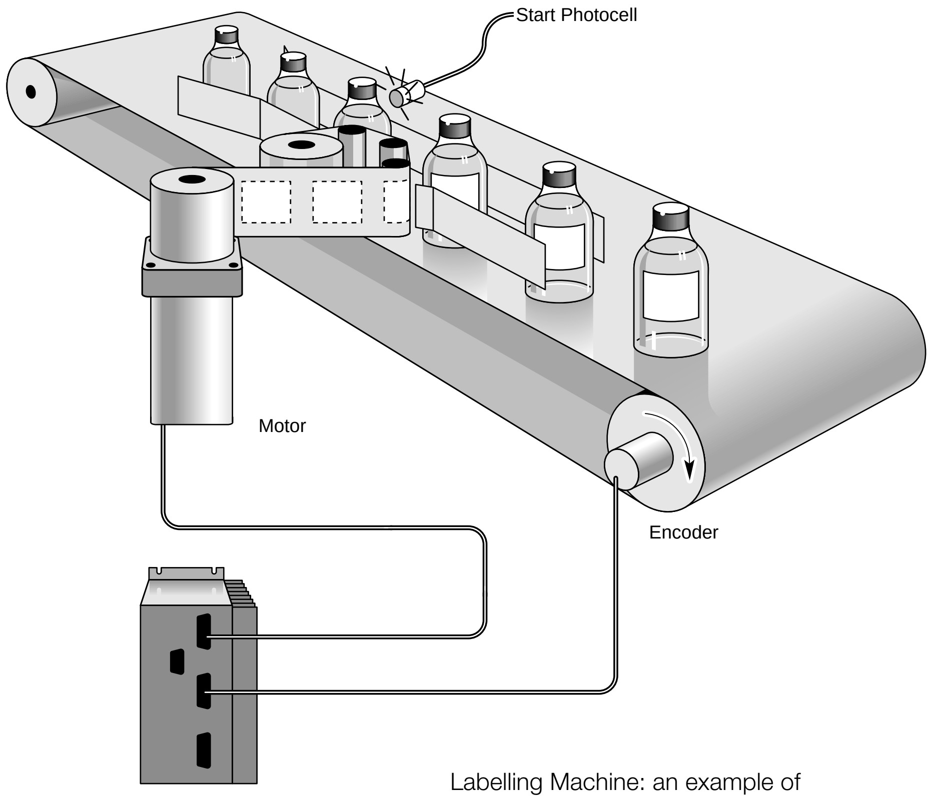
Web Processing

Any process that must be performed on a continuous web of product can be accomplished on the fly with following. See following example on next page.

Flying Cutoff (cut to length)

A continuous bar sheet or other extrusion of material is cut to specific lengths while the material is in motion. Production times are reduced as product does not have to be stopped to be cut.

Random Timing Infeed Application Highlight



Labelling Machine: an example of following application.

Application Description

Bottles on a conveyor run through a labelling mechanism that applies a label to the bottle. The spacing of the bottles on the conveyor is not regulated and the conveyor can slow down, speed up, or stop at any time.

Machine Requirements:

- Accurately apply labels to bottles in motion
- Allow for variable conveyor speed
- Allow for inconsistent distance between bottles
- Pull label web through dispenser
- Smooth, consistent labelling at all speeds

Motion Control Requirements:

- Synchronization to conveyor axis
- Electronic gearbox function
- Registration control
- High torque to overcome high friction
- High resolution
- Open-loop stepper if possible

Application Solution

A motion controller that can accept input from an encoder mounted to the conveyor and reference all of the speeds and distances of the label roll to the encoder is required for this application. A servo system may be required to provide the torque and speed to overcome the friction of the dispensing head and the inertia of the large roll of labels. A photosensor connected to a programmable input on the controller monitors the bottles' positions on the conveyor. The controller commands the label motor to accelerate to line speed by the time the first edge of the label contacts the bottle. The label motor moves at line speed until the complete label is applied, and then decelerates to a stop and waits for the next bottle.

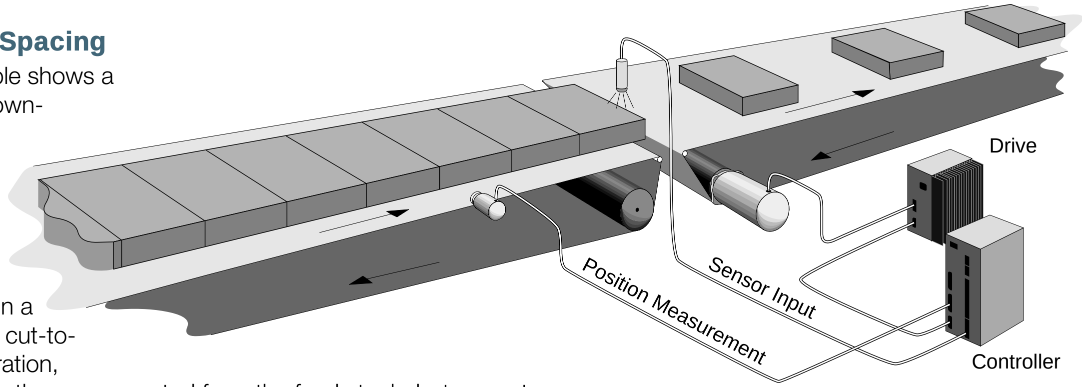
Product Solutions:

Controller	Motor
6K controller and Gemini GV drive	NO704FR

*The Apex 6152 or Zeta 6108 single axis drive/controlles have also been used in these types of applications.

Product Spacing Application Highlight

Product Spacing

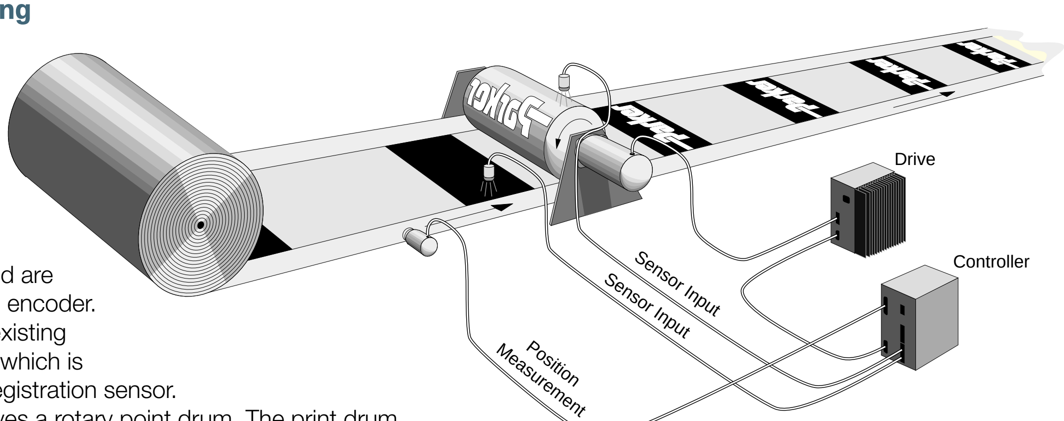


This example shows a possible down-stream operation from a continuous cut-to-length operation. In a continuous cut-to-length operation, product lengths are separated from the feed stock, but are not spaced apart from each other. In order to wrap the individual products, they must be spaced at specific intervals.

The master conveyor moves the product away from the cutting operation where newly cut product is spaced close together. The slave conveyor (moving at a velocity proportional to the master) takes the correctly spaced products to a wrapping station. A sensor at the entrance to the slave conveyor is wired into the controller. The front edge of the product is detected by the sensor as it moves off the master conveyor. The controller waits for the entire product length to leave the master conveyor before it superimposes a distance-shift (i.e., a short fast advance) to introduce uniform product spacing on the slave conveyor. This advances each product a specific distance ahead of the next product as they move onto the second (or following) conveyor.

Web Processing Application Highlight

Web Processing



Rotary printing on a moving web is a typical web processing application. In this example, the web is the master, and its position and speed are measured with an encoder. The web has an existing repeating pattern which is detected with a registration sensor. The slave axis drives a rotary point drum. The print drum has a raised and inked portion which will lay down the new pattern on top of the existing pattern. During the print segment of its rotation, the print drum surface must match the position and speed of the existing print on the web in order to print without smearing, and maintain the proper registration. This requires that the surface speeds match at a 1:1 ratio during the actual print portion. The length of the existing pattern on the web, however, will generally not match the circumference of the drum. This requires that the print drum change to a different ratio during the non-print portion of the cycle. In other words, it requires the construction of a periodic ratio profile. Such a profile allows the drum to match the web speed during printing, yet rotate to correct position at the start of the next web pattern. Registration errors could arise if the web slips, stretches, or the original pattern is not perfectly regular. This problem is solved by using the registration sensor. Each new edge of each repeated web pattern is detected and the drum position is noted automatically. If the drum is not in the required position at the instant the web pattern is detected, a corrective advance or retard is superimposed on the regular cycle. This allows continuous perfect print alignment, even if the existing pattern is slightly irregular. Because the control of the print drum is based on the measurement of the web positions and speed, registration is maintained regardless of web speed changes.

Random Timing Application Highlight

Random Timing Infeed

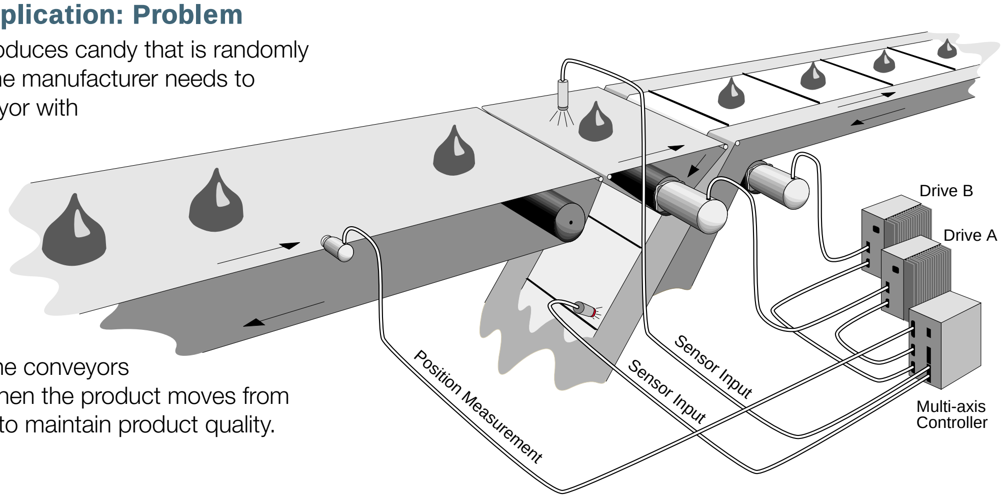
Random timing infeed refers to operations in which a product, at a particular point in a process, enters a conveyor with non-repeatable, or random timing, yet must leave the conveyor with perfect spacing. Typically, there is an infeed conveyor on which products are randomly spaced, a short conveyor on which the correction is made, and an exit conveyor with dividers on which the product must be placed. Line speed is critical when the product moves from one conveyor to another.

Two sensors, one located on the junction between the infeed conveyor and correction conveyor, and the other sensor on the exit conveyor must have a fixed phase relationship. The first sensor senses the product and the other sensor detects the locations of the dividers. The phase relationship is simply the distance the correction conveyor has traveled between the activation of the sensor on the exit conveyor and the activation of the product sensor.

The goal of the correction conveyor is to correct for variations in this phase relationship. Both the correction and exit conveyors are normally moving, or slaving, at a 1:1 relationship with the infeed conveyor, or master axis. When the product enters the correction conveyor, the 6000 controller can determine how much correction is needed by looking a the phase relationship between sensors. After the product has completely left the infeed conveyor, the 6000 controller adds an advance to the correction conveyor so that the calculated adjustment is made in time for the product to be placed on the exit conveyor. The controller can also be programmed to adapt to changes in mechanical alignment of the conveyors.

Random Timing Infeed Application: Problem

A food products manufacturer produces candy that is randomly spaced on an infeed conveyor. The manufacturer needs to place this candy on an exit conveyor with perfect spacing so that the candy can be wrapped and packaged accurately. The candy must be transferred and spaced properly without stopping the conveyor so that the throughput of the process can be maximized. The line speed of the infeed conveyor fluctuates, but the line speed of the conveyors must match each other exactly when the product moves from one conveyor to another in order to maintain product quality.



I/O Requirements

- 2 high-speed sensor inputs
- encoder feedback from the infeed conveyor

Solution

The solution of this application uses the Following and Random Timing Infeed functionality of the 6000 controller to control a short correction conveyor as well as the exit conveyor onto which the product is placed. Two sensors, one located on the junction between the infeed conveyor and the correction conveyor and the other on the exit conveyor, are placed with a fixed phase relationship. The first sensor recognizes the product and the other sensor detects the locations of the dividers between which the product must be placed. The purpose of the correction conveyor is to correct for variations in this phase relationship. The phase relationship is simply the distance the correction conveyor has traveled between activation of the exit conveyor sensor and activation of the product sensor. Both the correction and exit conveyors are moving, or slaving, at a 1:1 relationship with the infeed conveyor, or master axis. The conveyors are following the feedback from the encoder on the infeed conveyor.

When the product enters the correction conveyor, the 6000 controller can determine how much correction is needed by looking at the phase relationship between sensors. After the product has completely left the infeed conveyor, the correction conveyor makes the required shift and then matches the speed of the exit conveyor for a smooth transfer. The result is a quality product that is perfectly spaced on the exit conveyor.

Contouring (Circular Interpolation) Application Highlight

Random Timing Infeed

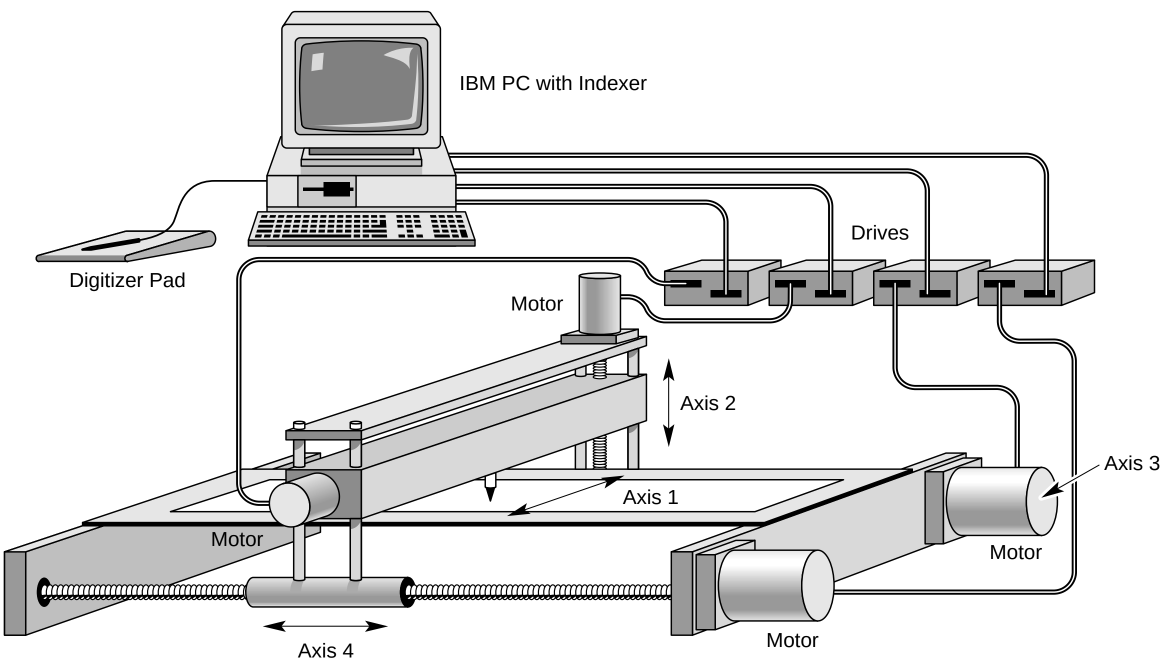
Contouring makes it easy to follow a two-dimensional path consisting of multiple line and arc segments. Circular interpolation is useful for making arcs and circles, especially when a constant path velocity must be maintained along a two-dimensional path. Dispensing and engraving applications often need contouring with constant path velocity.

In addition, a tangential axis is available to keep an angular position which changes linearly with the path direction. The tangential axis would be used in applications that require a work piece or tool to remain tangent or perpendicular to the path direction. Typical examples that requires a tangential axis as a knife pointing into the cut or a welding head staying normal to the weld.

A proportional axis may also be used to keep a position proportional to the distance traveled along the path described by X and Y. This allows helical interpolation. Contouring allows users to create complex move profiles on multiple axes. The easy programming language enables the user to generate arcs by only specifying the endpoint, radius or origin of the arc, and the direction of travel. Contouring also provides the user with the ability to control I/O while in the contouring mode.

Coordinated systems allow the assignment of an arbitrary X-Y position as a reference position for subsequent absolute end point specifications.

Example of Contouring Application—Engraving Machine



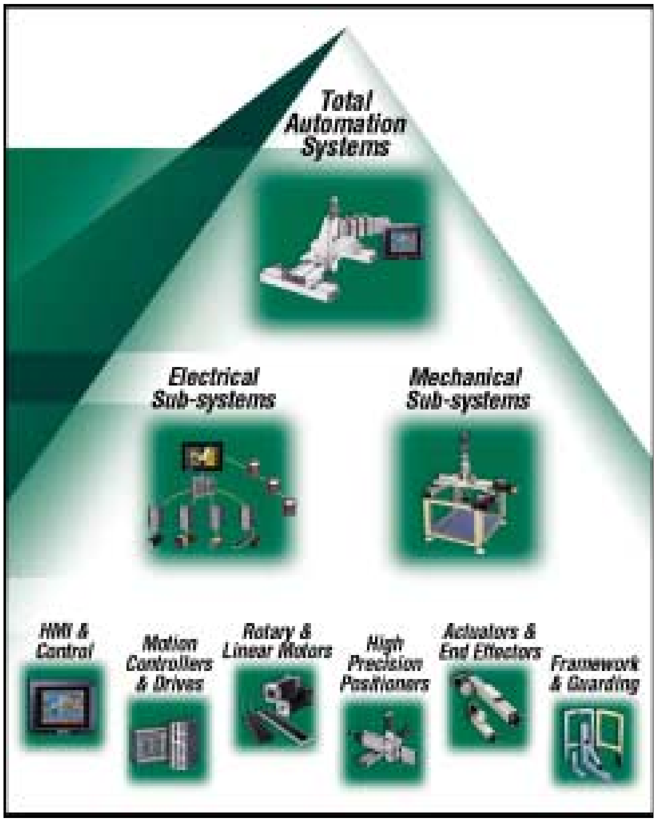
Parker Hannifin's Automation Group Offers A Total Solution

As a member of the Automation Group of Parker Hannifin, Compumotor's solution approach broadens with complimentary product and technology offerings from other Parker divisions. The Automation Group is comprised of four divisions including CTC, Daedal, AAD, and Compumotor offering the widest array of products in the motion control industry.

Each division has quality products manufactured and produced at their respective locations. Parker's engineers and management work together to tightly couple products to provide seamless integration for your next application.

- **CTC**--As the newest member of the Automation Group, CTC bundles a tightly integrated Human Machine Interface and Soft Control solution with an open PC hardware platform. Now there is a single source that provides affordable integration of factory-hardened PC workstations with the industry's leading HMI and Control software.
- **Daedal**--Provides unmatched precision and dependablity for for high speed performance for positioning tables and positioning systems. Offers a full spectrum of bearing technologies, drive systems, and positioner designs that allow you to match table performance to your design requirements and budget.
- **AAD**--A leading producer of pneumatic, hydraulic, and electromechanical products. Products include ParFrame aluminum framing systems, electric cylinders, pnematic positioning systems, and hydraulic and pneumatic rotary actuators.
- **Compumotor**--Offers a full lline of rugged multi-axis motion controllers, servo and stepper drive and drive/ controllers, and servo motors.

Parker's solution provides the "perfect fit" for End Users, Machine Builders, OEMs and System Integrators alike. Select the Parker solution for your next automation application and you'll soon discover the strength of Parker's Automation Group.



Leading Systems Integrator Finds Success with Automation Group

"... The system we designed and built was a total Parker solution."

Hughes Automation has designed a system which fully uses not only products from Compumotor, but also products from other Parker divisions comprising the Automation Group. Built for a large motor manufacturer, the application involved taking an engine rocker cover and dispensing sealant using an XYZ positioner.

A Compumotor 6K4 Controller, along with CompuCAM software, was used for the motion control dispensing unit. The drive systems included three Compumotor TQ10-EHS drives coupled with SM brushless servo motors. Positioning tables, 406XR and 404XR, from Parker Daedal were incorporated into the system as well. Parker CTC's P2 HMI interfaces rounded out the sytem. In trying to keep the system a full Parker solution, Hughes Automation used ParFRAME from Parker AAD as the structural guarding and framing of the system.

As a leading Systems Integrator in the Southeast, Hughes continues to look for innovative control packages to use in their systems. According to Matt Haddad, mechancial systems engineer with Hughe's Custom Machine Group, "The 6K4 allowed us to create complex pre-compiled move profiles for our X-Y-Z dispensing system. The soft operating system (6000 Series Command Language) made the programming the controller and interfacing it with a Parker CTC display very user friendly. Plus, the 6K4 small package size made it extremely modular, and it easily mounted inside the control cabinet on DIN rail. In fact, the system we designed and built was a total Parker Solution."

The application requires that an operator manually place a rocker lever cover into the system. Once the cover is in position, the operator starts the machine cycle by engaging an opto-touch switch. A CAD drawing of the rocker lever cover is converted to 6K motion commands using CompuCAM software. The 6K controls two axes of motor-driven positioning tables to accurately dispense a sealant on the edge of the rocker lever cover. At the end of the cycle, the dispensing head returns to a home position.

Once the cycle is complete, the operator manually removes the rocker lever cover and installs it on the engine in production. The operator has completed control over all system diagnostics, as well as other variable changes, through the system's CTC panel. Because the dispensing system offers the manufacturer the ability to automate its capabilities, and because the machine was delivered on time to the customer, work is now under way for the delivery of similar dispensing machines to other divisions within the motor manufacturer.