BALDOR MOTORS AND DRIVES

BALDOR ELECTRIC COMPANY

MOTION CONTROL APPLICATION NOTES

MN1200

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Drive Technology

The following explores the world of drives, and explains where each is best used.

There are inverter, vector, servo, and now linear drive technologies available. What type of technology should be considered depends upon the application needs.

Inverter Drives

Inverter techniques are used to control an induction motor. These use either six-step techniques or synthesize a sinusoidal waveform. The frequency of the generated waveform controls motor speed.

This control, with a standard AC induction motor, would provide a speed regulation (limited to the slip of the motor) of approximately 1.5-3 percent of base speed. The low-end controllable speed will start about 300 rpm. From there, constant torque can be provided to base speed with constant horsepower to 1.5 times base speed.

Some advantages of inverter drives include low initial cost due to simplicity of motor design, reliability, and ease of use. Design specifications include peak overload capacity of 150-200%, controlled reversing, pre-set speeds, and programmable I/O's to mention a few.

This technology works well for many adjustable-speed applications, such as centrifugal fans, conveyors, pumps, mixers, and packaging equipment.

Vector Drives

Vector control technology primarily uses a PWM synthesized sinusoidal waveform to control motor speed. An induction motor can be controlled by vector control technology. The only requirement is that an appropriate feedback device, such as an encoder, be used.

These controls can provide tighter speed regulation approaching 0.01 percent of set speed. Controllable speed ranges from zero speed to about 5 times base speed are attainable. The constant horsepower range will be about 3.5 times the base speed.

In order to provide the most reliable and efficient package for the application, the vector motor should be designed with a high-efficient winding, an efficient lamination design, and high-temperature insulation materials. The electrical winding must be protected against high dv/dt rates (short rise time pulses) and voltage reflections which will degrade the electrical winding life. Spike resistant wire has been designed for motors controlled by inverters and vectors. The additional protection , which this wire provides, results in longer motor life, reduced down time, and better overall value. All Baldor motors include the above features, designed in as standard.

If the application has high inertial loads, a line regeneration ("line regen") vector control should be considered. "Line regen" controls save energy by returning the energy (power generated by the motor) back to the incoming power line. Additionally, since these designs operate near unity power factor, they result in additional energy savings.

Advantages of vector drive technology includes low initial cost, reliability, capability of rated torque to zero speed, precise speed control, long life, constant horsepower output above rated speed, and such programmable features as controlling acceleration/deceleration time, and tuning of the control.

Vector drives are used in high-performance adjustable-speed applications, machine tool spindles, and industrial test stands just to mention a few applications. Line regen units are used in winders, hoist/crane, presses, HVAC, and other applications. For positioning applications, vector drives are used with existing programmable position controllers.

DC Servo Drives

In the world of servos, there are DC servos and brushless servos. The brushless servos are termed, by manufacturers, as either DC or AC.

The DC servo package takes AC power in, and converts it to DC. The amount of DC output applied to the motor is directly proportional to the desired operating motor speed. The performance provided by DC servos may be considered as “traditional”, and comparable to vector technology when used with a positioning package.

Advantages of DC servos include proven reliability and well known technology (they have been around forever). In positioning applications, and in comparison to vectors, DC servos come in a smaller package size and provide lower inertia, which translates to faster acceleration (gets into position faster). When used within the designed capability range, the DC servo motor, with its brush design, provides a long adequate life for many applications.

DC servos are used in machine tool, factory automation, packaging, woodworking, and many other applications.

Brushless Servo Drives

As indicated, brushless servos are available as either DC brushless or AC brushless. The feedback device determines whether it is considered DC or AC, since this dictates the control scheme. The feedback device may be either Hall sensors, encoders, or resolvers.

With Hall sensor feedback, the three-phase brushless motor is powered by energizing two of the three motor windings at a time. There are six different commutation sections for one mechanical revolution, and within each commutation section, a DC level of power is applied. The amount of DC applied is directly proportional to the desired operating motor speed. Thus the term “DC brushless”.

Encoder feedback is used when position data is required in the application. Some encoders are available with Hall outputs. Again, the Hall signals are used for commutation of the brushless motor.

With resolver feedback, a sinusoidal waveform is applied on the motor windings. Thus the term “AC brushless”. The advantage of this technology is that, for the same torque (compared to “DC brushless”), the “AC brushless” will require less current. Therefore a smaller drive may often be used in the application. This becomes possible since the motor has a three phase sinusoidal winding being powered by a three phase sinusoidal current waveform.

Advantages of brushless technology include higher speed capability, higher torques in a smaller package, much lower inertia (thus much faster acceleration capability), and of course, long reliable maintenance free life in the application. These drives provide good low speed operation down to zero speed. Brushless controls are available with auto-tuning capability.

Brushless drives are used in robotics, packaging, electronic assembly, semiconductor equipment, textile, and any cutting, printing or labeling which may even be performed on a moving web. Many other applications of course exist.

Figure 1 represents a summary of vector and servo drive technology comparisons.

Figure 1.

DRIVE TECHNOLOGY COMPARISON
(1 HP AT 1800 RPM)

	VECTOR	DC SERVO	BRUSHLESS SERVO	
			Std Inertia	Low Inertia
Speed	Traditional	Traditional	Higher Speeds are possible	
Acceleration	Traditional	Fast	Very Fast	Ultra Fast
Typical Motor	7.6″ O. D.	4″ O. D.	4.7″ Sq.	3.5″ Sq.
Response	Traditional	Fast	Very Fast	Ultra Fast
Inertia (lb.-in.-S^2)	0.052	0.024	0.0078	0.0025

Linear Drives

First of all, let's define a "linear" motor. Imagine that a rotary motor is cut open and unrolled. The result is a flat linear motor. The same electromechanical principles apply whether the product is rotary or linear. For example, a permanent magnet DC rotary motor is similar to a permanent magnet linear motor; and an AC induction squirrel cage motor is similar to an induction linear motor. The same electromagnetic force that produces "torque" in a rotary motor also produces "force" in a linear motor. The linear motor utilizes the same controls and programmable position controllers as rotary motors.

Some other similarities of rotary and linear: Rotary "torque" is measured in pound-feet (lb-ft) or pound-inch (lb-in), or newton-meters (N-m); whereas linear "force" is measured in pounds (lb), or newtons (N). Rotary "velocity" is measured in revolution per minute (rpm); whereas linear "velocity" is measured in inches per second (in/sec), or meters/second (m/sec). The "duty cycle" is the same for both technologies; it is the time a motor receives power divided by the total cycle time.

Prior to the invention of linear motors, the only way to produce linear motion was to use either pneumatic or hydraulic cylinders or to translate rotary motion into rotary motion using ball screws or belts and pulleys. Sometimes for some applications, these other linear motion technologies may be a better choice. But each application has to be investigated. As an example, Figure 2 represents a comparison of linear motors vs. ball screws.

Figure 2.

Comparison of Linear Motors vs. Ball Screws

Linear Motors	Ball Screw
Low to Medium Force	High Force
High Speed (to 200 ips)	Low Speed (50 ips)
High acceleration (to 10 g's)	Lower acceleration (2 g's)
Two parts with no contact	Many parts with contact wear point
Lower maintenance	Higher maintenance
No backlash	Longer settling time
Greater accuracy	Lower accuracy

Advantages of linear motors include high acceleration (up to 10 g's), low speed (0.0001 inch/second) and high speed (100 inch/second) capability, small strokes (0.01 inch) and long strokes (excess of 25 feet), and sub-micron position accuracy (with appropriate feedback device). Only one moving part leads to simplicity and high reliability, with no backlash and very high stiffness. The non-contact parts reduce wear leading to long life and reduction in maintenance.

Linear drives are used in semiconductor, material handling, robotics, medical, people movers, and packaging just to indicate a few applications.

Conclusion

For proper sizing and selection, an applications' speed (and speed range), inertia (and inertia matching) and acceleration (relates to overload or peak capacity), and package size (relates to smaller, lighter motor in some technologies), all should be reviewed.

Mechanics

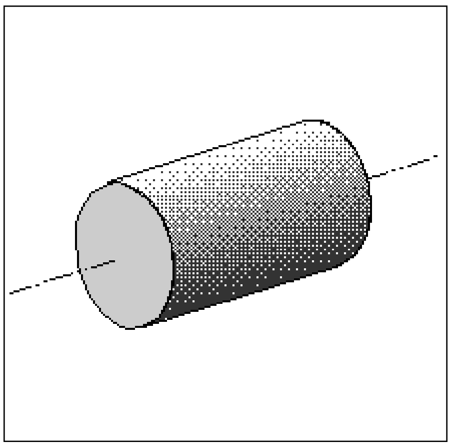
This application note presents formulas for calculating reflected load parameters.

The mechanical systems used in motion control applications can be divided into four basic categories: direct, gear, belt-pulley and leadscrew. Reflecting parameters back to the motor shaft eases the calculations necessary for sizing and selecting.

Cylinder Inertia

The first formula is for calculating the inertia of a cylinder. This is important since many mechanical components can be calculated using this formula.

The inertia of a **solid** cylinder can be calculated if either the weight and radius, or the density and radius and length are known.

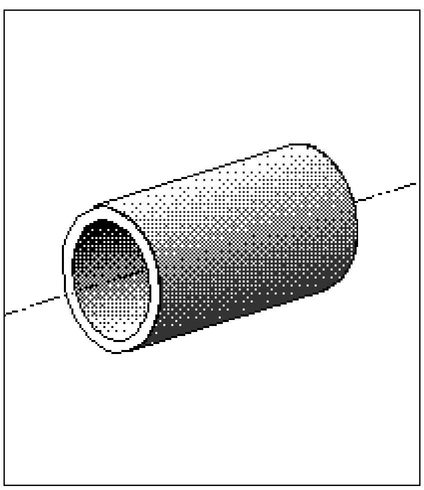


For known weight and radius: $J_L = \frac{1}{2}\frac{W}{g} r^2$

For known density, radius and length: $J_L = \frac{1}{2}\frac{\pi l p r^4}{g}$

Where:
- J_L = inertia (lb-in-s^2)
- W = weight (lb)
- r = radius (in)
- l = length (in)
- p = density of material (lb/in^3)
- g = gravitational constant (386 in/s^2)

The inertia of a **hollow** cylinder can be calculated if the weight and radius are known, or the density and radius and length are known.

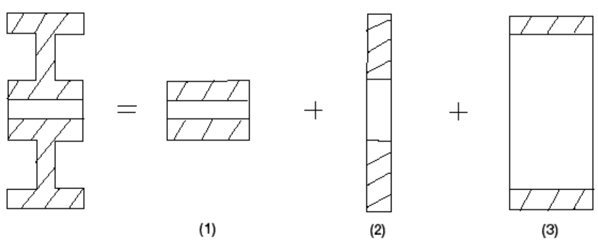


For known weight and radius: $J_L = \frac{1}{2}\frac{W}{g}(r_o^2 + r_i^2)$

For known density, radius and length: $J_L = \frac{\pi}{2}\frac{lp}{g}(r_o^4 - r_i^4)$

Where:
- J_L = inertia (lb-in-s^2)
- W = weight (lb)
- r_o = outside radius (in)
- r_i = inside radius (in)
- l = length (in)
- p = density of material (lb/in^3)
- g = gravitational constant (386 in/s^2)

The inertia of complex concentric rotating parts is calculated by breaking the part up into simple rotating cylinders, calculating their individual inertia, and adding them together.



$J = J_1 + J_2 + J_3$

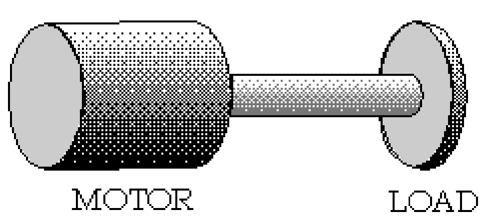
Total

Material Densities

Material	lb/in^3
Aluminum	0.096
Brass	0.300
Bronze	0.295
Copper	0.322
Steel (cold rolled)	0.280
Plastic	0.040

Direct Drive

For direct drive loads, there are no mechanical linkages.



Speed: $W_M = W_L$

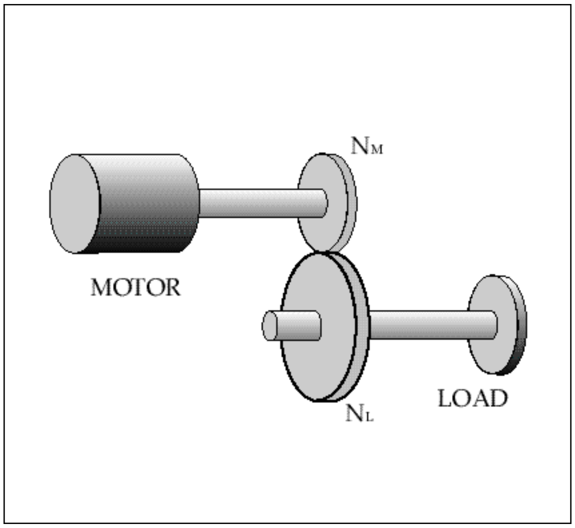
Inertia: $J_T = J_L + J_M$

Torque: $T_L = T'$

Where:
- W_M = motor speed (rpm)
- W_L = load speed (rpm)
- J_T = total system inertia (lb-in-s^2)
- J_L = load inertia (lb-in-s^2)
- J_M = motor inertia (lb-in-s^2)
- T_L = load torque at motor shaft (lb-in)
- T' = load torque (lb-in)

Gear

Loads in a gear application have to be reflected back to the motor shaft by the gear ratio, or the gear ratio squared. The inertia of the gears have to be included in the calculations. Gear inertia may be calculated using the formula for cylinder.



Speed: $W_M = W_L (N_L/N_M)$

Inertia: $J_T = (N_M/N_L)^2 (J_L + J_{NL}) + J_M + J_{NM}$

Torque: $T_L = T' (N_M/N_L)$

Where:
- W_M = motor speed (rpm)
- W_L = load speed (rpm)
- N_M = number of motor gear teeth
- N_L = number of load gear teeth
- T_L = load torque at motor shaft (lb-in)
- T' = load torque (lb-in) - not reflected
- J_T = total system inertia (lb-in-s^2)
- J_L = load inertia (lb-in-s^2)
- J_M = motor inertia (lb-in-s^2)
- J_{NM} = motor gear inertia (lb-in-s^2)
- J_{NL} = load gear inertia (lb-in-s^2)

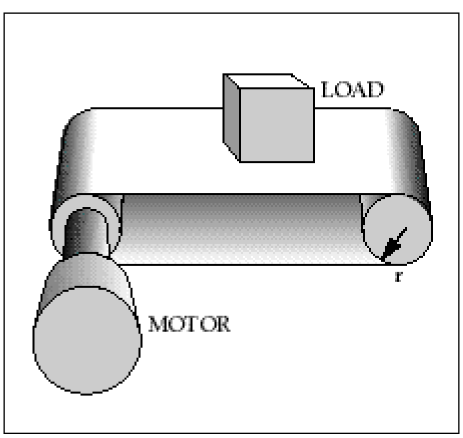
Belt and Pulley

The load parameters have to be reflected back to the motor shaft. Although a belt and pulley arrangement is shown, it can also be a rack and pinion, or timing belt and pulley, or chain and sprocket. The inertia of the pulleys, sprockets, or pinions must be included.

Speed: $W_M = \frac{1}{2\pi} \frac{V_L}{r}$

Inertia: $J_T = \frac{W}{g} r^2 + J_{P1} + J_{P2} + J_M$

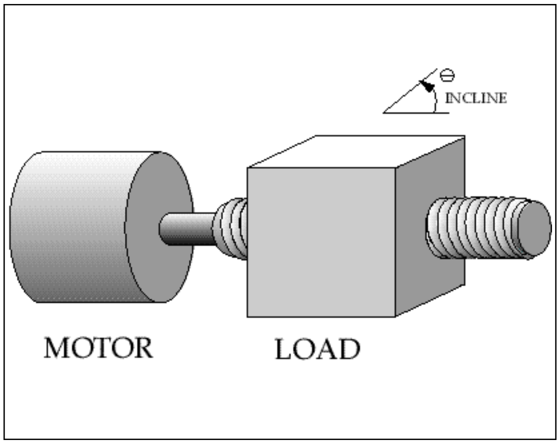
Torque: $T_L = F_F r + F_L r$



Where:
- W_M = motor speed (rpm)
- V_L = linear load speed (inch/min)
- r = pulley radius (in)
- T_L = load torque reflected to motor shaft (lb-in)
- T_F = friction torque (lb-in)
- F_L = load force (lb)
- J_T = total system inertia (lb-in-s^2)
- J_M = motor inertia (lb-in-s^2)
- J_P = pulley inertia (lb-in-s^2)
- W = load weight including belt (lb)
- F_F = frictional force (lb)
- g = gravitational constant (386 in/s^2)

Leadscrew

The load has to be reflected back to the motor shaft. The leadscrew inertia has to be included (can use the formula for inertia of a cylinder). If preloading is used the torque must be included since it may be significant.



Speed: $W_M = V_L P$

Load Torques: $$T_L = \frac{1}{2\pi}\frac{F_L}{pe} + \frac{1}{2\pi}\frac{F_{PF}}{p} \times 0.2$$

Friction Torques: $$T_F = \frac{1}{2\pi}\frac{\mu W \cos\theta + W \sin\theta}{pe}$$

Inertia: $$J_T = \frac{W}{g}\left(\frac{1}{2\pi p}\right)^2 \frac{1}{e} + J_{LS} + J_M$$

Where:

W_M	=	motor speed (rpm)
V_L	=	linear load speed (in/min)
p	=	lead screw pitch (revs/in)
e	=	lead screw efficiency
T_L	=	load torque at motor shaft (lb-in)
T_F	=	friction torque (lb-in)
F_L	=	load force (lb)
F_{PF}	=	preload force (lb)
J_T	=	total system inertia (lb-in-s^2)
J_M	=	motor inertia (lb-in-s^2)
J_{LS}	=	lead screw inertia (lb-in-s^2)
W	=	load weight (lb)
F_F	=	frictional force (lb)
μ	=	coefficient of friction
g	=	gravitational constant (386 in/s^2)

Typical Efficiencies

Type	Efficiency
Ball-nut	0.90
Acme with plastic nut	0.65
Acme with metal nut	0.40

Coefficient of Friction

Material	Coefficient
Steel on steel	0.580
Steel on steel (lubricated	0.150
Teflon on steel	0.040
Ball bushing	0.003

Matched Performance™

The information in this application note explains how to read the speed-torque curves.

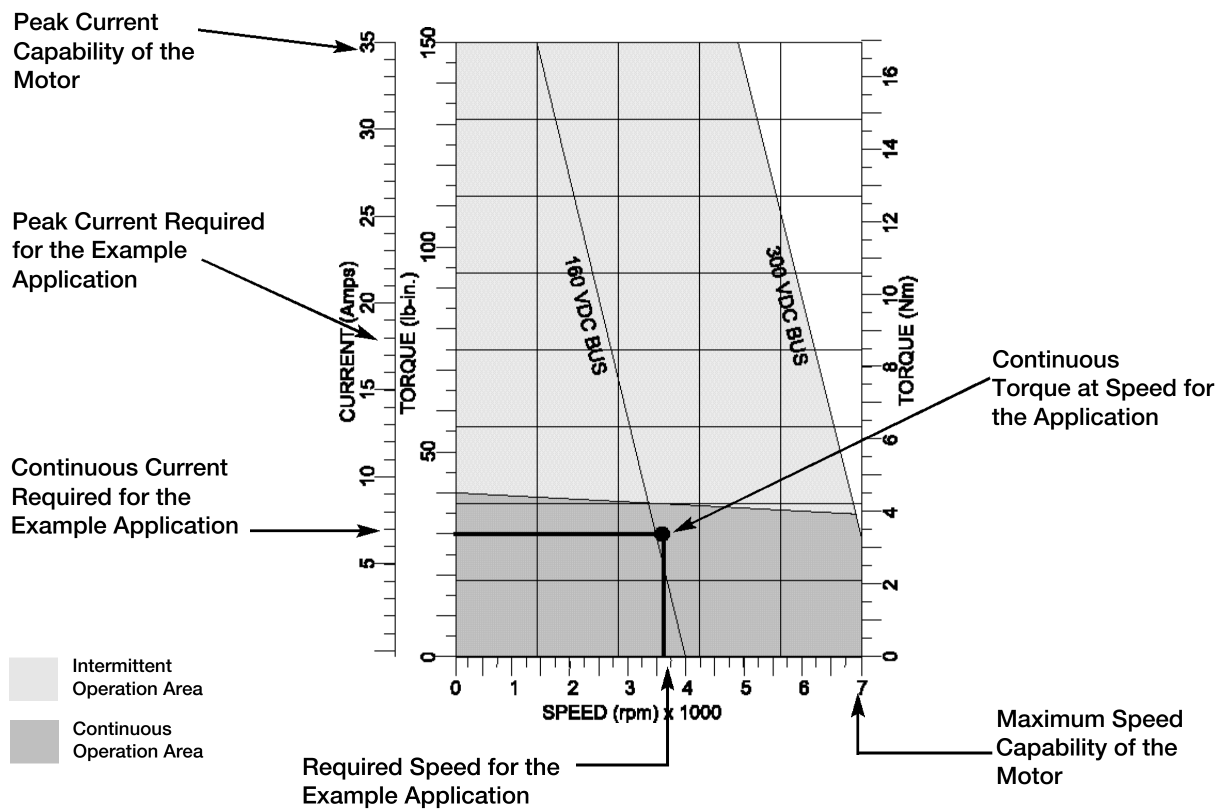
Baldor provides "matched performance™" curves to simplify the process of selecting both a motor and control for a specific application.

Curves

In constant speed applications, motors are defined in terms of horsepower (which is torque at a "base" speed). In positioning applications, the motors normally operate over a wide range - not simply at "base" speed - so they are usually not "rated" at "base" conditions.

The curves display continuous torque (defined as torque which will not overheat the motor), and peak torque (defined as intermittent or acceleration torque).

It is also necessary to know the current and voltage required for the motor to operate, so a control may be selected. The curves have a scale that shows current required for any torque, and voltage required for any speed.



Example

An application requires a continuous torque of 30 lb-in at a speed of 3750 RPM. The peak torque required for acceleration is 80 lb-in.

The curve shows a motor which will work in this application. The bus voltage required is 300 VDC. The continuous and peak currents required are 7 and 18 amps.

A control is selected which can deliver this amount of current or more. The literature would indicate the closest control would deliver 10 amps continuous and 20 amps peak with a 230 VAC input (300 VDC bus).

Data Tables

The motor's voltage constant (back-emf) and torque constant are "cold" values (25°C); the continuous stall torque and current are "hot" values (155°C). The temperature coefficient factor between "cold" and "hot" is 0.95 for rare earth motors, and 0.85 for ferrite.

Basics of Motion

The following will show you how to easily determine the right motor and control for any electromechanical positioning application.

Once the mechanics of the application have been analyzed, and the friction and inertia of the load are known, the next step is to determine the torque levels required. Then, a motor can be selected to deliver the torque and the control sized to power the motor. If friction and inertia are not properly determined, the motion system will either take too long to position the load, it will burn out, or it will be unnecessarily costly.

Motion Control

In a basic motion-control system, Figure 1, the load is coupled through one of several mechanical linkages to the motor.

The motor may be a traditional DC servo motor, a vector motor, or a brushless servo motor.

Motor speed is dictated by the control which takes a low-level incoming command signal and amplifies it to a higher power for moving the motor.

The programmable motion controller is the brain of the motion system. It is programmed to accomplish a specific task for a given application. This controller monitors a feedback signal to control the position of the load. By comparing a pre-programmed, “desired” position with the feedback position, the controller can take action to minimize an error between the actual and desired load positions.

Figure 1. Basic Motion System

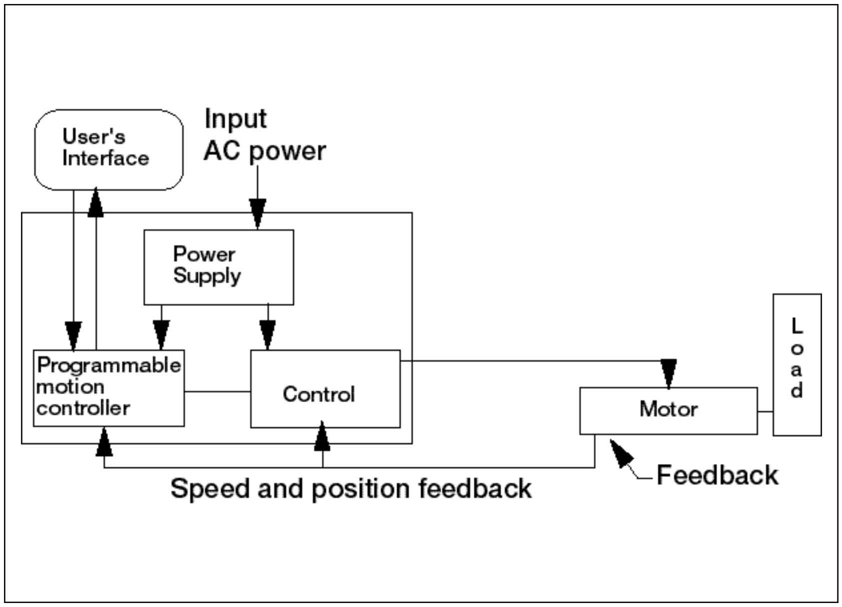
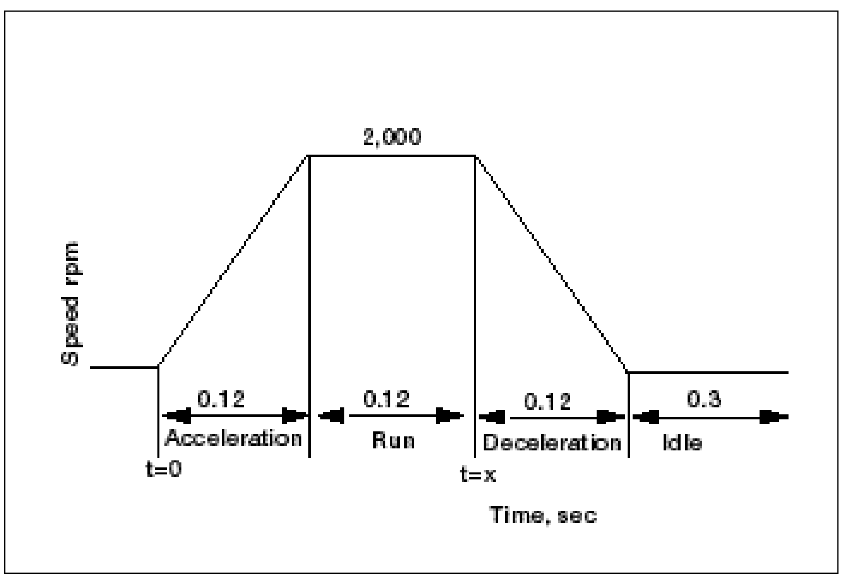


Figure 2. Move Profile



Movement Profile

A movement profile defines the desired acceleration rate, run time, speed, and deceleration rate of the load. For example, suppose with a system at rest (time=0, Figure 2), the programmable motion controller issues a command to start motion. At this instant, the motor has not yet started to move, there is no feedback signal, and the error signal is large.

The control wants to reduce the error signal, and the motor begins to accelerate. As the motor approaches the commanded speed, the error signal is reduced and, in turn, voltage applied to the motor is reduced. As the system stabilizes at running speed, only nominal power (voltage and current) are required to overcome friction and windage. At t=x, the load approaches the desired position and begins to decelerate.

In applications with similar move profiles, most of the input energy is dissipated as heat. Therefore, in such systems, the motor's power dissipation capacity is the limiting factor.

Acceleration Torque

Determining acceleration rate is the first step. For example, with the movement profile of Figure 2, the acceleration rate can be determined from the speed and acceleration time. (Dividing the motor speed rpm by 9.55 converts the speed to radians per second).

$$a_{acc} = \frac{S_m}{9.55 t_{acc}} \qquad (1)$$

$$a_{acc} = \frac{2{,}000}{9.55(0.12)} = 1{,}745.2 \text{ rad/sec}^2$$

The torque required to accelerate the load and overcome mechanical friction is:

$$T_{acc} = J_t(a_{acc}) + T_f \qquad (2)$$

$$= (J_L + J_{LS} + J_m)\, a_{acc} + T_F \qquad (3)$$

Example: Our application, Figure 3, requires moving a load through a leadscrew. The parameters are:

Leadscrew inertia (J_{LS}) = 0.00313 lb-in.-s^2

Motor inertia= 0.0037 lb-in-s^2

Friction torque (F_F) = 0.95 lb-in.

Acceleration torque can be determined by equation 3

T_{acc} = (0.00052 + 0.00313 +

0.0037) 1745.2 + 0.95

= 13.75 lb-in.

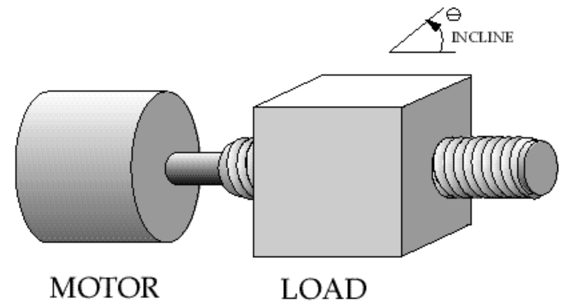
Duty Cycle Torque

In addition to acceleration torque, the motor must be able to provide sufficient torque over the entire duty cycle or movement profile. This includes a certain amount of constant torque during the run phase, and deceleration torque during the stopping phase.

Running torque is equal to friction torque (T_F), in this case, 0.95 lb-in.

During the stopping phase, deceleration torque is:

Figure 3. Example Application



Motor Parameters	Load = 200 lbs
Inertia = 0.0037 lb-in-s^2	Leadscrew
Continuous Stall = 14 lb-in	Inertia = 0.00313 lb-ins^2
Torque Constant (K_T) = 4.8 lb-in/amp	Pitch = 5 revs/inch
Resistance 4.5ohms	Friction = 0.95 lb-in

$$T_{dec} = -J_t(a_{acc}) + T_F \qquad (4)$$

$$= -(0.00052+.0.00313+0.0037)\ 1745.2 + 0.95$$

$$= -11.85 \text{ lb-in.}$$

Now, the root-mean-squared (rms) value of torque required over the movement profile can be calculated:

$$T_{rms} = \sqrt{\frac{T^2_{acc}(t_{acc}) + T^2_{run}(t_{run}) + T^2_{dec}(t_{dec})}{t_{acc}+t_{run}+t_{dec}+t_{idle}}}$$

$$= \sqrt{\frac{(13.75)^2(.12)+(.95)^2(.12)+(11.85)^2(.12)}{.12+.12+.12+.3}}$$

$$= 7.73 \text{ lb-in.}$$

The motor, used for this example, can supply a continuous stall torque of 14 lb-in., which is adequate for the application.

Control Requirements

Determining a suitable control is the next step. The control must be able to supply sufficient accelerating current (I_{acc}), as well as continuous current (I_{rms}) for the application's duty-cycle requirements.

Required acceleration current that must be supplied to the motor is:

$$I_{acc} = \frac{T_{acc}}{K_t} \qquad (6)$$

$$= \frac{13.75}{4.8} = 2.86 \text{ A}$$

Current over the duty cycle, which the control must be able to supply to the motor is:

$$I_{rms} = \frac{T_{rms}}{K_t} \quad (7)$$

$$= \frac{7.73}{4.8} = 1.61 \text{ A}$$

Power Requirements

The control must supply sufficient power for both the acceleration portion of the movement profile, as well as for the duty-cycle requirements. The two aspects of power requirements include (1) power to move the load, P_{del}, and (2) power losses dissipated in the motor, P_{diss}.

Power delivered to move the load is:

$$P_{del} = \frac{T(S_m)(746)}{63{,}025} \quad (8)$$

Power dissipated in the motor is a function of the motor current. Thus, during acceleration, the value depends on the acceleration current (I_{acc}); and while running, it is a function of the rms current (I_{rms}). Therefore, the appropriate value is used for "I" in the following equation.

$$P_{diss} = I^2 (R_m) \quad (9)$$

The sum of these P_{del} and P_{diss} determine total power requirements.

Example: Power required during the acceleration portion of the movement profile can be obtained by substituting in equations 8 and 9:

$$P_{del} = \frac{13.75(2{,}000)}{63025}(746) = 325.5 \text{ W}$$

$$P_{diss} = (2.86)^2(4.5)(1.5) = 55.2 \text{ W}$$

$$P = P_{del} + P_{diss}$$

$$= 325.5 + 55.2 = 380.7 \text{ W}$$

Note: the factor of 1.5 in the P_{diss} calculation is a factor used to make the motor's winding resistance "hot". This is assuming the winding is at 155° C.

Continuous power required for the duty cycle is:

$$P_{del} = \frac{7.73(2{,}000)}{63025}(746) = 182.9 \text{ W}$$

$$P_{diss} = (1.61)^2(4.5)(1.5) = 17.5 \text{ W}$$

$$P = 182.9 + 17.5 = 200.4 \text{ W}$$

In Summary

The control selected must be capable of delivering an acceleration current of 2.86 A, and a continuous current of 1.61 A. The power requirement calls for peak power of 380.7 W and continuous power of 200.4 W.

To aid in selecting, computer software programs are available to perform the iterative calculations necessary to obtain the optimum motor and control.

Is It DC or AC Brushless

The brushless motor can be driven by either a DC control or an AC control. However the torque developed by the package is different.

The torque developed by a brushless motor depends on the control technology used. The same motor can be driven by either a DC control or an AC control scheme. However, the torque developed by the package is different. The following will cover the sine-emf motor when driven by different methods, and results attained.

Brushless Motors

The winding distribution of a brushless motor is sinusoidal, and as shown in Figure 1, the resulting torque generated is a function of the shaft angular position. Thus current into a winding generates a torque which is described by:

$$T = T_{pk} \times \text{Sin (electrical angle)} \qquad (1)$$

$$T = \hat{T} \times \text{Sin} (ø)$$

where T is the instantaneous torque, T_{PK} or $\hat{T}$ is the peak value of torque, and ϕ is the electrical angle of the shaft. The electrical angle is different than the mechanical angle, and these are related by:

$$\text{angle elect} = \frac{N}{2} \times \text{angle mech} \qquad (2)$$

where N is the number of poles.

In a three phase system, the windings are shifted by 120 electrical degrees, and the equations describing torque per winding are:

$$T_r = \hat{T} \times \text{Sin } \phi \qquad (3)$$

$$T_s = \hat{T} \times \text{Sin } (\phi+120°) \qquad (4)$$

$$T_T = \hat{T} \times \text{Sin } (\phi+240°) \qquad (5)$$

Energizing winding R while rotor is at a position of 30 elect degrees (see Figure 2) will result in a torque being developed forcing the shaft to rotate. The shaft will rotate to the 180° (elect) position and stop. However, if when the shaft is at the 150° (elect) position, the current is removed from winding R and applied to winding T, the shaft will continue to rotate. If

Figure 1. Sinusoidal - EMF Motor

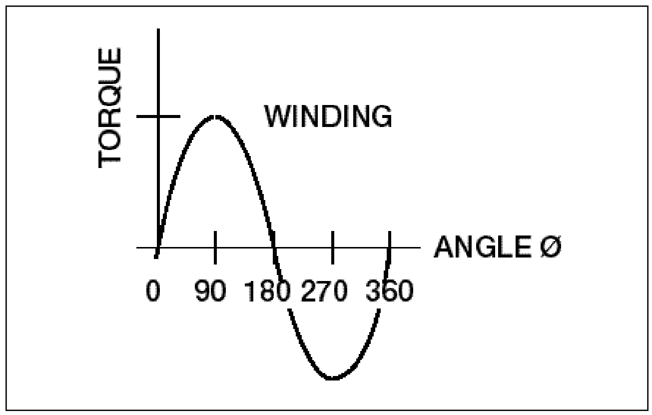
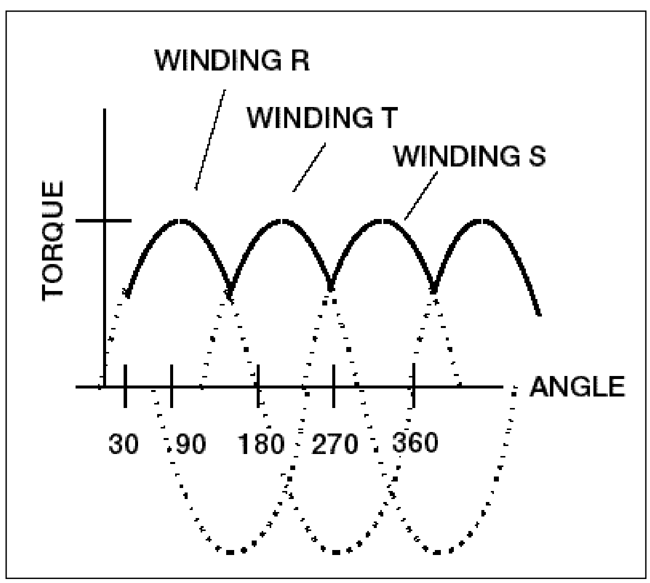


Figure 2. Energizing Sinusoidal EMF Motor



this process is repeated (i.e. current is removed from winding T at 270° and applied to winding S) the shaft will continue to rotate. By continuation of the scheme, rotation is continued.

DC Control

The electrical windings will be energized near the peak of the waveform, to obtain maximum torque from the motor. This corresponds to applying power over 60° as shown in Figure 3.. Since there are six (6) different commutation sections, this commutation scheme is sometimes referred to as a 6-step control.

This yields an "average" back-EMF of:

$$K_{E_{RMS}} = \int_{\frac{\pi}{3}}^{\frac{2\pi}{3}} \frac{\hat{K}_E \sin\phi}{\frac{2\pi}{3} - \frac{\pi}{3}} \qquad (6)$$

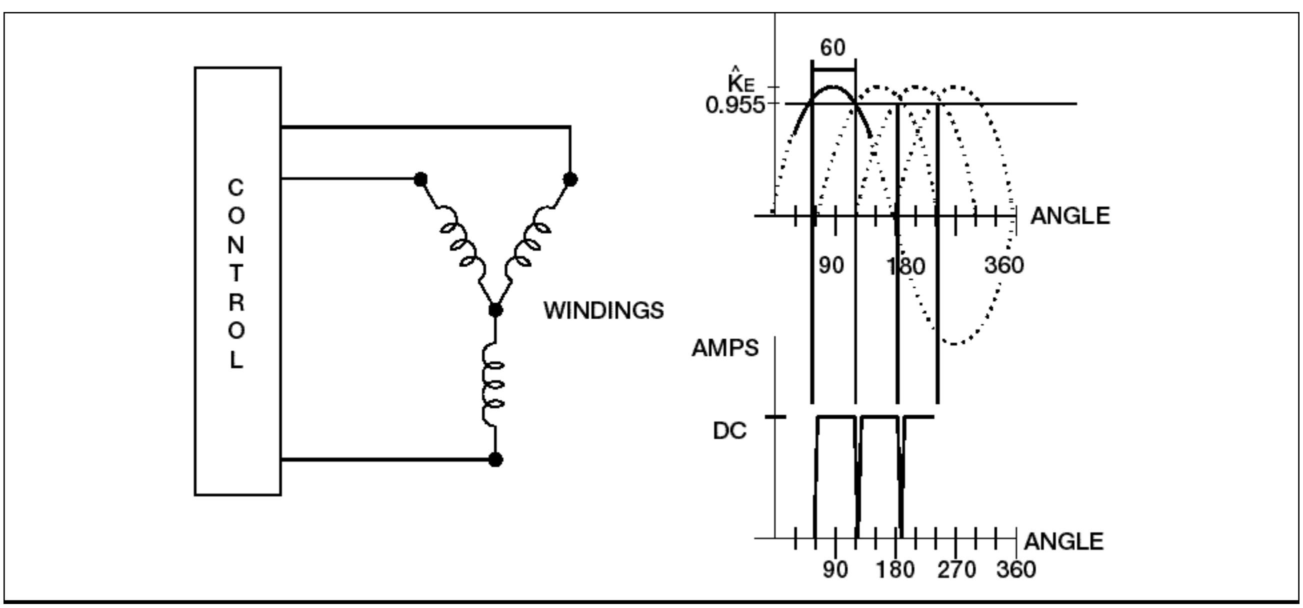
Where K_E is measured phase to phase (ϕ–ϕ)

$$K_{E_{RMS}} = \frac{-3}{2\pi} \hat{K}_E \left(\cos\frac{2\pi}{3} - \cos\frac{\pi}{3}\right) \qquad (7)$$

$$= \frac{3}{\pi} \hat{K}_E$$

$$K_{E_{RMS}} = 0.955\, \hat{K}_E \qquad (8)$$

Figure 3. DC Control



Thus, the expression for torque, with a floating neutral, becomes:

$$T = K_T I \qquad (9)$$

$$T = 0.955\ K_{E\phi\phi}\ I \text{ (Torque in N-m, } K_E \text{ in v/r/s)} \qquad (10)$$

The back-EMF or voltage constant is measured phase to phase, and current is the DC level thru the winding.

With the commutation scheme above, the maximum torque developed occurs at 60°, and is:

$$T_{MAX} = \hat{T}\ (\sin(60)-\sin(300)) \qquad (11)$$

$$= \hat{T} \times 1.73$$

The minimum torque is:

$$T_{MIN} = \hat{T} \times 1.5 \qquad (12)$$

Therefore, theoretical torque ripple in this situation is:

$$\% = \frac{MAX - MIN}{MAX} = \frac{1.73 - 1.5}{1.73} = 13.2\% \qquad (13)$$

Torque ripple depends on the control scheme, and it has to be designed to acceptable application tolerances.

AC Control

The application of a sinusoidal current:

$$I = \hat{I} \times (\text{Sin}\ \phi + \phi \text{ phase}) \qquad (14)$$

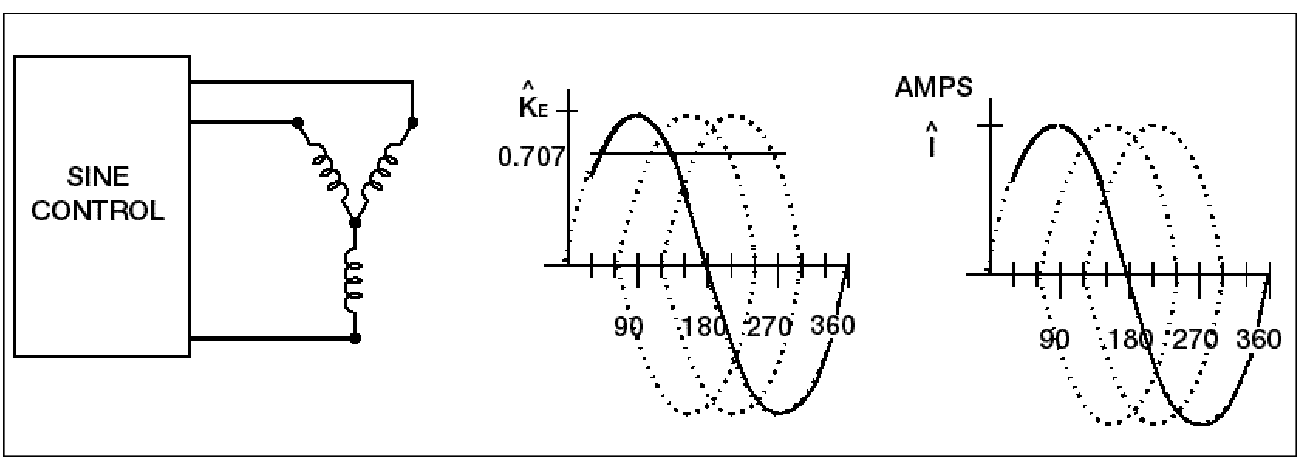
is applied to all three windings (see Figure 4), which are sinusoidal:

$$K_T = \hat{K}_{T\phi}\ \sin \phi \qquad (15)$$

When energizing all three windings, the output torque developed is then equal to the sum of the torques in all three windings:

$$T_M = T_R + T_S + T_T \qquad (16)$$

Figure 4. AC Control



Using these equations we arrive at:

$$T_M = \hat{K}_{T\phi}\,\hat{I}\,[\sin^2 \phi + \sin^2 (\phi + 120) + \sin^2 (\phi + 240)] \quad (17)$$

$$T_M = \hat{K}_{T\phi}\,\hat{I} \times 1.5 \quad (18)$$

With this commutation scheme, there is no difference between the maximum and minimum torque developed. Therefore ideally, there is no torque ripple when employing a sine controller with a sine-EMF motor.

Equation (18) above provides an expression for torque developed in terms of torque constant as measured from phase to neutral. However the neutral is not accessible. Therefore an equivalent phase to phase expression is desired. The equation is developed and is:

$$T = 1.5\, K_{T\,\phi\phi\,RMS}\; I_{RMS} \quad (19)$$

This equation provides a relationship between torque developed, the RMS current (which can be measured), and the phase to phase torque constant of the motor. However K_T cannot be easily measured. The saving factor is that K_E (phase to phase) is very easy to measure. Simply by observing the motor's back-EMF waveform on a scope (when driving the motor by some external means) and measuring that waveform, the value for K_E can be determined. K_E is simply volts divided by KRPM. Then converting from K_E to K_T, equation (19) may be used.

K_T & K_E Relationship

The relationship between the torque constant and voltage constant can be derived as follows:

$$K_{T\phi} = K_{E\phi} \text{ (where } K_T \text{ is N-m/amp and } K_E \text{ is v/r/s)} \quad (20)$$

since in a 3-phase wye connected system:

$$2 \times K_{T\phi} = K_{T\phi\phi} \quad (21)$$

$$\sqrt{3} \times K_{E\,\phi} = K_{E\,\phi\phi} \quad (22)$$

therefore:

$$K_{T\phi\phi} = \frac{2}{\sqrt{3}}\, K_{E\,\phi\phi} \quad (23)$$

$$K_{T\phi\phi} = 1.15473\; K_{E\,\phi\phi} \quad (24)$$

(N-m/amp) (v/r/s)

This is the basic equation for the relationship of torque constant versus voltage constant for a 3 phase motor when driven with a 3 phase excitation. From this the other dimension systems can be derived.

This provides the relationship between torque developed, the RMS current, and the measurable voltage constant of the motor. Note that current and the voltage constant are expressed in RMS terms, i.e. RMS of a sinusoidal waveform. By simply measuring, via a scope, the motor's peak value of K_E the developed torque may now be easily calculated.

Figure 5 summarizes the relationship of a sinusoidal-EMF motor when driven with either a DC drive or an AC drive. By multiplying the peak value of the sine back-EMF times the factor in the table, the "equivalent" or "RMS" value is determined. This RMS value can then be used in calculations and current is calculated from the relationship:

$T = K_T\ I$

where K_T is the torque constant from Figure 5.

Figure 5. Relationship between KE and KT for a brushless motor when driven with a DC control versus an AC control

Voltage constant is measured as V/KRPM peak (phase to phase). The RMS values are determined by multiplying by the value in the table:

		English	Metric
For DC drive	K_E	0.95 V/KRPM	0.009076 v/r/s
	K_T	1.285 oz-in/amp	0.009076 N-m/amp
For Sine drive	K_E	0.707 V/KRPM	0.006754 v/r/s
	K_T	1.656 oz-in/amp	0.011698 N-m/amp

As an example, suppose you measure the sinusoidal waveform as 75 V/KRPM peak.

For a DC drive, the equivalent voltage constant is 71.2 V/KRPM; the torque constant would be 96.3 oz-in/amp; and the current required to develop 180 lb-in of torque is:

$T = K_T I$

$180 = \frac{96.3}{16}\ I$ (Note: Divide 96.3 oz-in/amp by 16 to obtain lb-in/amp)

or $I = 29.9$ amps

For an AC drive, the equivalent voltage constant is 53 V/KRPM; the torque constant would be 124.2 oz-in/amps; and the current required to develop 180 lb-in of torque is:

$T = K_T I$

$180 = \frac{124.2}{16}\ I$

or $I = 23$ amps

Conclusion

Torques developed by the brushless motor depends upon the control technology used. An easy way to determine the type of control is to look at the feedback scheme. The DC control typically uses Hall sensors for feedback, where the AC control typically uses resolvers.

Inertia Matching

Consider matching load and motor inertia. The following presents some points to consider.

System performance depends upon the load and motor coupling, and the ratio selected. These determine response, mechanical resonance, and power dissipated.

Response

Typical system response with “relatively good inertial matching” is shown in Figure 1. As the load to motor mismatch is increased, oscillations occur, and it takes longer to settle in position (Figure 1b).

The fix, to prevent oscillation and overshooting, is to lower the gain and extend the settling time (Figure 1c). This leads to lower accelerations and slower positioning. This approach may not be acceptable for some applications.

Special loop compensation, of course, can be designed. This would allow handling of higher inertial mis-matches. However, this leads to highly custom designs and now standard off the shelf controls cannot be used.

Mechanical Resonance

Analyzing the transfer function of the load, motor shaft, and feedback device, the resultant equation is:

$$f = \frac{1}{2\pi}\sqrt{\frac{(J_L + J_M)\ K}{J_L J_M}}$$

Where J_L is the load inertia, J_M is the motor inertia, and K is the transmission stiffness.

This equation provides the frequency of the mechanical resonance. It points out that the torsional resonance 1) depends upon the load and motor coupling, i.e. the transmission stiffness and 2) the frequency point is lower for high inertia loads.

For the best response, this resonant point should be outside the system bandwidth. It is typical to have the resonant frequency 5-10 times the servo loop bandwidth due to rise time requirements.

The easiest, quickest, and least expensive method is to use gearing, or a larger motor (with more inertia) to improve the inertia ratio.

Power Dissipation

Analyzing the equation for energy dissipation for optimum versus non-optimum ratios, the resultant analysis is plotted in Figure 2. This show the amount of additional system energy dissipated as mis-match increases. It indirectly also reveals the additional current required (by the square root).

The plot reveals that a small deviation from optimum is not critical, however, as the deviation increases, the penalty becomes increasingly severe. As can be seen, the ratio of 1:1 provides for minimal power dissipation. For a mismatch of 2:1 the energy increases by 56%.

System power dissipation is minimized with inertia matching. Inertia ratio ranges, by markets, traditionally are:

1:1 to 3:1 for robotics type applications

4:1 to 7:1 for machine tool type applications

5:1 to 10:1 for other X-Y positioning type applications

Conclusion

Thus recommendations would be: 1) whenever there is a choice between two motor frame sizes, use the larger shorter motor, 2) maintain inertia matching as close as possible, i.e. use a load/motor mismatch of 10:1 as a maximum guide.

Figure 1. Response with various load/motor inertia mis-matches.

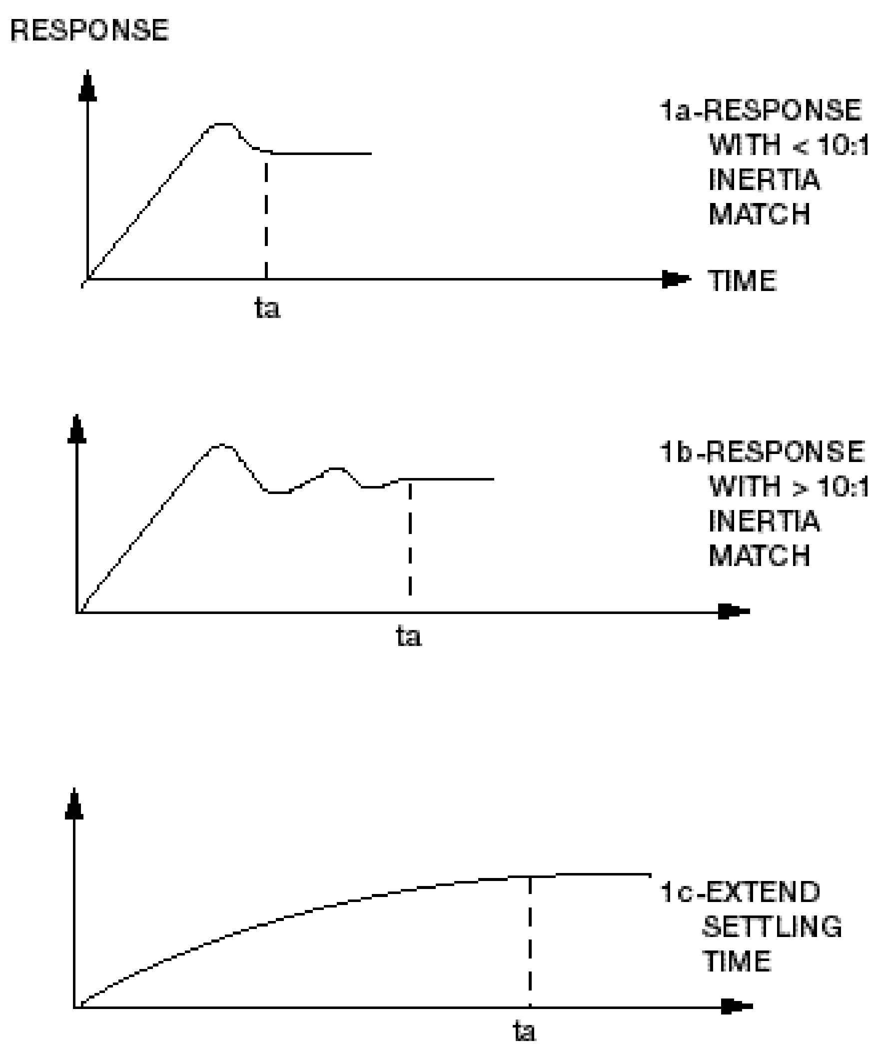
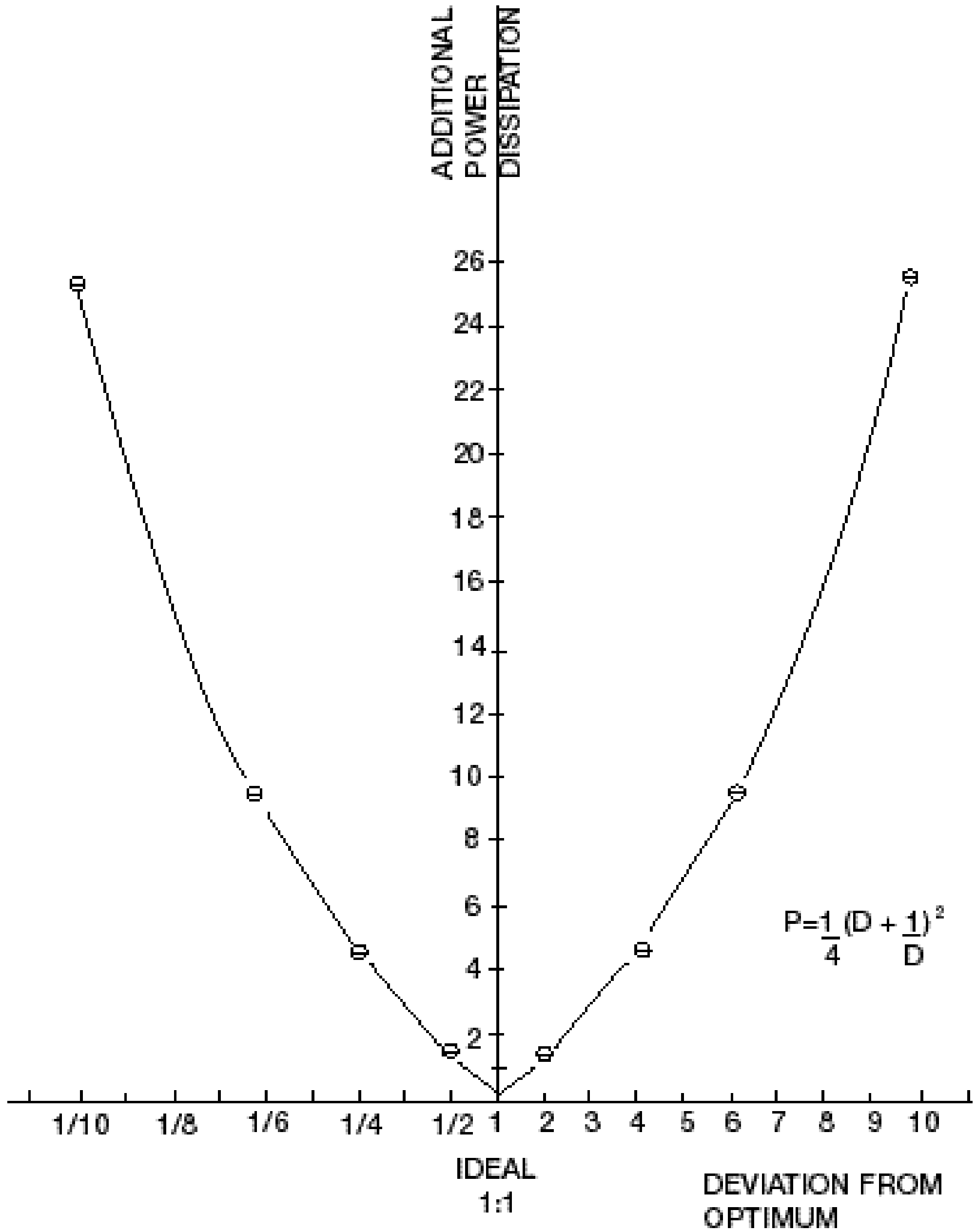


Figure 2. Additional energy dissipation for mis-matched load/motor inertia.



Conversion Tables

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

A \ B	gm-cm^2	oz-in^2	gm-cm-s^2	kg-cm^2	lb-in^2	oz-in-s^2	lb-ft^2	Kg-cm-s^2	lb-in-s^2	lb-ft-s^2 or slug-ft^2
gm-cm^2	1	5.46×10^{-3}	1.01×10^{-3}	10^{-3}	3.417×10^{-4}	1.41×10^{-5}	2.37×10^{-6}	1.01×10^{-6}	8.85×10^{-7}	7.37×10^{-8}
oz-in^2	182.9	1	0.186	0.182	0.0625	2.59×10^{-3}	4.34×10^{-4}	1.86×10^{-4}	1.61×10^{-4}	1.34×10^{-5}
gm-cm-s^2	980.6	5.36	1	0.9806	0.335	1.38×10^{-2}	2.32×10^{-3}	10^{-3}	8.67×10^{-4}	7.23×10^{-5}
Kg-cm^2	1000	5.46	1.019	1	0.3417	1.41×10^{-2}	2.37×10^{-3}	1.019×10^{-3}	8.85×10^{-4}	7.37×10^{-5}
lb-in^2	2.92×10^{3}	16	2.984	2.926	1	4.14×10^{-2}	6.94×10^{-3}	2.98×10^{-3}	2.59×10^{-3}	2.15×10^{-4}
oz-in-s^2	7.06×10^{4}	386.08	72	70.615	24.13	1	0.1675	7.20×10^{-2}	6.25×10^{-2}	5.20×10^{-3}
lb-ft^2	4.21×10^{5}	2304	429.71	421.40	144	5.967	1	0.4297	0.3729	3.10×10^{-2}
Kg-cm-s^2	9.8×10^{5}	5.36×10^{3}	1000	980.66	335.1	13.887	2.327	1	0.8679	7.23×10^{-2}
lb-in-s^2	1.129×10^{6}	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-s^2 or slug-ft^2	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

Conversion Tables

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

A \ B	dyne-cm	gm-cm	oz-in	kg-cm	lb-in	Newton-m	lb-ft	Kg-cm
dyne-cm	1	1.019×10^{-3}	1.416×10^{-5}	1.0197×10^{-6}	8.850×10^{-7}	10^{-7}	7.375×10^{-8}	1.019×10^{-8}
gm-cm	980.65	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	$9.806 \mathrm{X} 10^{5}$	1000	13.877	1	0.8679	$9.806 \mathrm{X} 10^{-2}$	$7.233 \mathrm{X} 10^{-2}$	10^{-2}
lb-in	1.129×10^{6}	1.152×10^{3}	16	1.152	1	0.112	8.333×10^{-2}	1.152×10^{-2}
Newton-m	10^{7}	1.019×10^{4}	141.612	10.197	8.850	1	0.737	0.101
lb-ft	1.355×10^{7}	1.382×10^{4}	192	13.825	12	1.355	1	0.138
Kg-m	9.806×10^{7}	10^{5}	1.388×10^{3}	100	86.796	9.806	7.233	1

Conversion Tables

Multiply	By	To Obtain
Length		
Angstrom units	3.937×10^{-9}	in.
cm	0.3937	in.
ft	0.30480	m
in. (U.S.)	2.5400058	cm
in. (British)	0.9999972	in. (U.S.)
m	10^{10}	Angstrom units
m	3.280833	ft
m	39.37	in.
m	1.09361	yd
m	6.2137×10^{-4}	miles (U.S. statute)
yd	0.91440	m
miles (U.S. statute)	5,280	ft
Area		
cir mils	7.854×10^{-7}	in.^{2}
cm^2	1.07639×10^{-3}	ft^2
cm^2	0.15499969	in.^{2}
ft^2	0.092903	m^2
ft^2	929.0341	cm^2
in.^{2}	6.4516258	cm^2
Volume		
cm^3	3.531445×10^{-5}	ft^3
cm^3	2.6417×10^{-4}	gal (U.S.)
cm^3	0.033814	oz (U.S. fluid)
ft^3 (British)	0.9999916	ft^3
ft^3 (U.S.)	28.31625	L (liter)
m^3	264.17	gal (U.S.)
gal (British)	4,516.086	cm^3
gal (British)	1.20094	gal (U.S.)
gal (U.S.)	0.13368	ft^3 (U.S.)
gal (U.S.)	231	in.^{3}
gal (U.S.)	3.78533	L (liter)
gal (U.S.)	128	oz (U.S. fluid)
oz (U.S. fluid)	29.5737	cm^3
oz (U.S. fluid)	1.80469	in.^{3}
yd^3	0.76456	m^3
yd^3 (British)	0.76455	m^3
Plane Angle		
radian	57.29578	deg
Weight		
Dynes	2.24809×10^{-6}	lb
kg	35.2740	oz (avoirdupois)
kg	2.20462	lb
kg	0.001	tons (metric)
kg	0.0011023	tons (short)
oz (avoirdupois)	28.349527	grams
tons (long)	1,106	kg
tons (long)	2,240	lb
tons (metric)	1,000	kg
tons (metric)	2,204.6	lb
tons (short)	2,000	lb

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