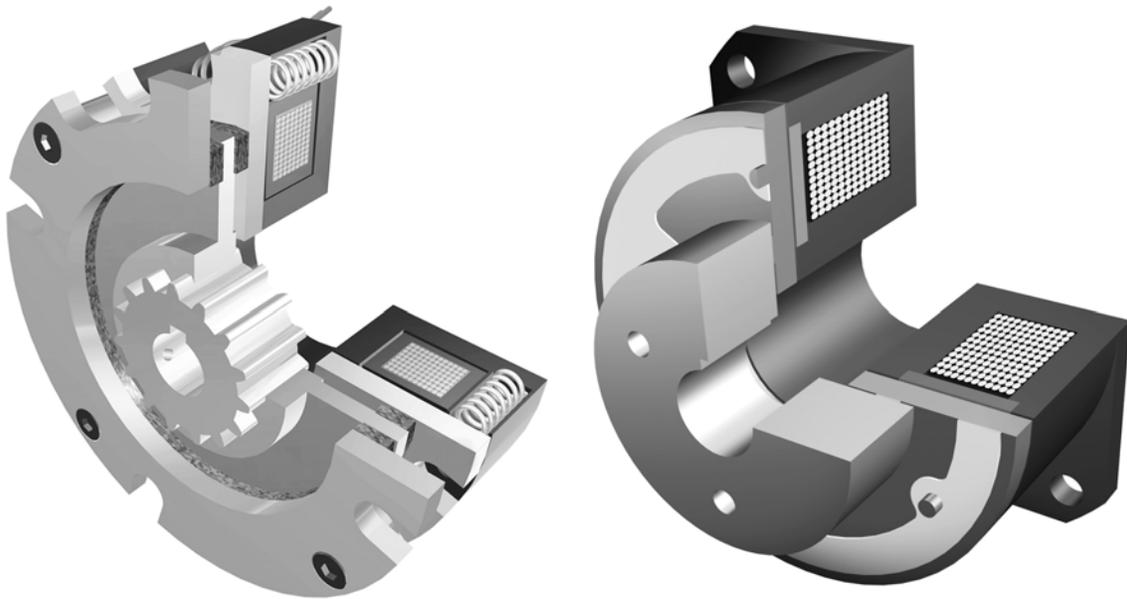


How to Size and Apply Dynamic Spring-Set Brakes

Obtaining high reliability, repeatability, accuracy, and long life from dynamic spring-set brakes depend upon a few simple, but critical force and energy calculations.

By John Pieri

Dynamic spring-set brakes come in two basic types. The first holds the load with power applied to the electric brake solenoid and the other type holds with the power disconnected. Power-on brakes secure the load when the specified voltage energizes the solenoid, but unfortunately lose their grip during an inadvertent power failure. Power-off brakes, on the other hand are safe by default. Either intentional or inadvertent power loss engages the brake to hold the load by virtue of a mechanical spring. In addition, the latter brake is the safest and the most preferred type to use in machines where the load moves vertically.



Power-off brake (left) and power-on brake.

Engaging and disengaging the brake takes a finite amount of time, so the equations that compute the time delay determine the final position at which a machine member will either arrive or depart, or the time for a shaft to start or stop moving when commanded by an electrical signal. The two choices available in the brake selection process determine whether the machine stops in a specified time or within a specified amount of travel.

Worst-case selection parameters

Consider the following list of parameters whether the brake is a power-on or power-off type, and before determining the brake inertia, brake torque, and operating time delays.

Then select a family of brakes from the manufacturers catalog based on the following principal parameters:

- Brake mode: power on or power off
- Temperature range
- Voltage/current range for the brake solenoid
- Allowable solenoid temperature rise
- Duty cycle – continuous or intermittent
- Mechanical brake mounting restrictions or preferences
- Stopping operations/requirements: emergency dynamic only, static hold only or dynamic stop every cycle (total number of dynamic stops required by the application)
- Maximum speed and direction of rotation
- Mounting position of brake: vertical or horizontal
- System drag or friction torque
- System inertia
- Allowable deceleration time and allowable number of shaft revolutions after issuing a stop command (overshoot)
- Cycle-to-cycle stopping tolerance and variation over life

Power-off operation

Consider the power-off mode first. The brake engages and stops the load within a few milliseconds after the brake solenoid receives the command to release or de-energize. Braking component makers usually supply a graph of the solenoid current versus the time delays. The amount of time delay between de-energizing the solenoid and stopping the load depends on the total system inertia, which is composed of the load inertia and the brake inertia.

From Figure 2:

$$T_s = T_e + T_d \quad \text{EQN 1}$$

Where: T_s = time to stop, sec.
 T_e = energize time, sec
 T_d = deceleration time, sec

The deceleration rate, $d\omega/dt$ (rad/sec²), depends upon the total torque required to stop the load and the system inertia:

$$d\omega/dt = T_t/J_s \quad \text{EQN 2}$$

Where: T_t = total torque, lb-in.
 J_s = system inertia = $J_l + J_b$ (load inertia + brake inertia), lb-in.-sec²

The brake-engage time is typically different from the release time.

Stop within specified time

First, estimate the torque required to stop the system inertia within an interval that is half the time that the servomotor can safely hold the load without assistance. At this point in

the analysis only the load inertia is known, so a rule of thumb is to add about 25% to the load inertia to estimate the brake rotor inertia, that is, $J_s = 1.25J_l$. This information helps determine a candidate brake. After calculating the brake inertia, the supplier's data sheet provides the response time specifications for that brake.

The equation for the estimated torque T_t is:

$$T_t = 0.1047J_s(d\omega/dt) \quad \text{EQN 3}$$

Where: 0.1047 is a factor that converts rad/sec to rpm
 J_s = system inertia, $1.25J_l$, lb-in.-sec²

When the system has substantial drag, T_d , it aids in deceleration, so it subtracts from the brake torque rating, T_b :

$$T_b = T_t - T_d \quad \text{EQN 4}$$

On the other hand, when an overrunning torque develops during deceleration, the brake torque rating increases:

$$T_b = T_t + T_d \quad \text{EQN 5}$$

Next, select a suitable brake from the catalog, and run through the calculations again using the actual load torque and the specified brake rotor inertia for a new system inertia value. Verify that the brake selected can handle the system torque. If it is unsatisfactory, select the next higher size. When it proves to be satisfactory, calculate the deceleration time, dt , using the new total system inertia:

$$dt = 0.1047J_s(d\omega/T_t). \quad \text{EQN 6}$$

The deceleration time plus the brake engage time must be less than (or equal to) the desired stopping time.

The time required for the brake to engage depends on the time that the solenoid's magnetic field decays. That, in turn, depends on the type of arc suppression circuit used for the solenoid. Usually, the solenoid comes with an ordinary silicon diode or no protection at all. Unfortunately, both of these circuit configurations have a relatively long time constant. When the circuit comes with a single silicon suppression diode, add a Zener diode in series with it that has a voltage rating twice the solenoid voltage rating. In other words, when a 24-V coil is used, select a Zener diode with a 48-V rating. Other arc suppression circuits are composed of resistors, capacitors, or a MOV (metal oxide varistor). Resistor and capacitor networks also have relatively long time constants, but often a single MOV will be sufficient. In any event, the best policy is to consult with the brake manufacturer for the optimal suppression circuit for a specific application.

Sometimes a technique known as “coil over excitation” is employed which makes a clutch or brake engage faster and have greatly improved starting and stopping accuracy. A pulsed voltage, higher than the normal steady-state value, is applied momentarily to the coil, which decreases the magnetizing time while increasing the solenoid pull force. This over-voltage condition combined with high force springs increases the units torque, also reduces the start/stop times and friction face wear normally caused by slippage during a slower engagement time. Although an over-excitation circuit can reduce the start time significantly, the proper coil suppression circuit must be used to prevent damage to the system.

Stop within a specified distance

The second option is selecting a brake to stop the load within a specified distance. First, find a brake that fits all the other selection criteria given above, and then calculate the total travel allowed after the command signal is issued to stop the load. This includes the travel while the armature engages, plus the travel during load deceleration. The total travel, S, given in the manufacturer’s graph of Figure 3 is:

$$S = [(t_4-t_3)+(t_5-t_4)/2]\omega/60 \quad \text{EQN 7}$$

When the travel does not meet specifications, add a Zener diode as described above, or select another brake assembly and repeat the calculations, with and without the Zener diode arc-suppression circuit. After the travel is acceptable, calculate the energy absorption.

Energy absorption

The above calculations address only a single cycle. Now verify that the brake can dissipate the kinetic energy that it absorbs per cycle for as long as it takes to perform the entire operation. In addition, repeated cycling builds up heat, which the solenoid must be able to withstand continuously.

First, calculate the per cycle energy absorption, Eb:

$$E_b = 4.6(J\omega^2)10^{-4}\text{ft-lb/cycle} \quad \text{EQN 8}$$

When the friction drag is significant compared to brake torque, modify the energy calculation by the ratio of brake torque to total torque:

$$E_b = [T_b/T_t + T_d][4.6(J\omega^2)10^{-4}\text{ft-lb/cycle}] \quad \text{EQN 9}$$

When braking at a relatively rapid rate, multiply Eb per cycle by the cycle rate, N:

$$E_{bmin} = [T_b/T_t + T_d][4.6(J\omega^2)10^{-4}(N)\text{ft-lb/min}] \quad \text{EQN 10}$$

Compare the calculated values of energy per cycle and energy per minute with the values given in the product data sheets. The values should be equal to or less than the catalog ratings to ensure that the brakes survive over a long lifetime. Most manufacturer’s

provide wear life data, which help determine the expected life, considering the total number of cycles expected. This is usually estimated by the equation:

$$T_n = T_e/T_c$$

Where: T_n = maintenance-free life, cycles
 T_e = total allowable energy absorption, ft-lb
 T_c = calculated energy absorbed per cycle, ft-lb/cycle

Final Proofing

All solenoid-operated brake systems designed for dynamic braking also apply to static-holding systems. The wear factors for friction materials are more critical for the dynamic mode, so in all cases, the brake component manufacturer should verify the final selection. Not all relevant information and data (including some engineering units of measure) that might be critical to each unique installation is available in catalog data sheets. For example, some information regarding friction materials are proprietary and the life data given in catalog sheets and graphs are computed and collected under optimum laboratory testing conditions. This is because not all real-world installation details can be anticipated and evaluated. The manufacturer's calculations may show that your selection proves to have a much greater life expectancy than initially expected, but on the other hand, it may show that the next higher rating is the optimal brake assembly for your machine. In most cases the component manufacturer does not charge for this service, so in the end, it is less expensive and shortens project time by obtaining the verification.

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