Application Engineering

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Introduction

Information and guidelines provided in the application section are intended for general selection and application of spring set brakes. Unusual operating environments, loading or other undefined factors may affect the proper application of the product. Stearns application services are available to assist in proper selection or to review applications where the specifier may have questions.

A spring set brake is used to stop and hold a rotating shaft. Generally the brake is mounted to an electric motor, but can also be mounted to gear reducers, hoists, machinery or utilize a foot mount kit.

The brake should be located on the high speed shaft of a power transmission system. This permits a brake with the lowest possible torque to be selected for the system.

Spring set disc brakes use friction to stop (dynamic torque) and hold (static torque) a load. Energy of the motor rotor and moving load is converted to thermal energy (heat) in the brake during deceleration. The brakes are power released, spring applied. No electrical current is required to maintain the spring set condition.

The system designer will need to consider the mount surface and match the brake to the load and application. Factors include: brake torque, stopping time, deceleration rate, load weight and speed, location and environment. Brake thermal ratings, electrical requirements and environmental factors are discussed in separate sections.

Electrical Considerations

Solenoid actuated brakes (SAB) are available with standard motor voltages, frequencies and Class B or H coil insulation. Most models can be furnished with either single or dual voltage coils. Coils in most models are field replaceable.

Inrush and holding amperage information is published for the common coil voltages and factory available for other voltages or frequencies. Amperage information for specific coil sizes is provided for selection of wire size and circuit protection at brake installation. Fixed voltage - 50/60 Hz dual frequency coils are available in many models.

All SAB AC coils are single phase and can be wired to either single or three phase motors without modifications. All solenoid coils have a voltage range of +/- 10% of the rated nameplate voltage at the rated frequency. Instantaneous rated voltage must be supplied to the coil to insure proper solenoid pull in and

maximum coil cycle rate. The plunger rapidly seats in the solenoid and the amperage requirements drops to a holding amperage value.

Instantaneous voltage must be supplied to the coil to insure proper solenoid pull-in and maximum coil cycle rate.

Because Stearns Solenoid Actuated Brakes (SAB's) require low current to maintain the brake in the released position, the response time to set the brake *can* be affected by EMF voltages generated by the motor windings. It may be necessary to isolate the brake coil from the motor winding.

The solenoid coil cycle rate limits the engagements per minute of a static or holding duty brake. Brake thermal performance, discussed in another section, limits engagements per minute in dynamic applications.

Class B insulation is standard in most SAB models, class H coil insulation is optional and is recommended for environments above 104°F (40°C), or rapid cycling applications.

Armature actuated brakes (AAB) are available in standard DC voltages. Available AC rectification is listed in the catalog section. Wattage information is provided in the catalog pages. Unlike solenoid actuated brakes, armature actuated brakes do not have inrush amperage. Coil and armature reaction time and resulting torque response time information is available. Like SAB, mechanical reaction time depends on typical application factors including load, speed and position.

Electrical response time and profiles are unique to the SAB and AAB. Reaction time requirements should be considered when selecting or interchanging brakes.

All Stearns brake coils are rated for continuous duty and can be energized continually without overheating. The coil heating effect is greatest at coil engagement due to engaging, pull in or inrush amperage.

Temperature limits as established by UL controls standards are:

Class A insulation	221°F (105°C)
Class B insulation	266°F (130°C)
Class H insulation	356°F (180°C).

Types of Applications

In order to simplify the selection of a disc brake, loads can be classified into two categories, non-overhauling and overhauling.

Loads are classified as non overhauling, if (1) no components of the connected equipment or external material undergo a change of height, such as would occur in hoisting, elevating or lowering a load, and (2) there is only rotary motion in a horizontal plane. For example, a loaded conveyor operating in a horizontal plane would be typical of a non-overhauling load.

If the same conveyor were transporting material to a lower level, it would be classified as an overhauling load. The external material or load undergoes a change in height, with the weight of the load attempting to force the conveyor to run faster than its design speed or to overhaul.

Non-overhauling loads require braking torque only to stop the load and will remain at rest due to system friction. Overhauling loads, such as a crane hoist, have two torque requirements. The first requirement is the braking torque required to *stop* the load, and the second requirement is the torque required to *hold* the load at rest. The sum of these requirements is considered when selecting a brake for an overhauling load.

Alignment

Requirements per NEMA:

Permissible ECCENTRICITY of mounting rabbet (AK dimension):

42C to 286TC frames inclusive is 0.004" total indicator reading. 324TC to 505TC frames inclusive is 0.007" total indicator reading.

Face Runout:

42C to 286TC frames inclusive is 0.004" total indicator reading.

If a customer furnishes a face on the machine for brake mounting, the same tolerances apply. Floor mounted brakes must be carefully aligned within 0.005" for concentricity and angular alignment. Use of dowels to insure permanent alignment is recommended.

In offset brake mount locations such as fan covers, cowls or jack shafting, proper mount rigidity and bearing support must be provided. Spring set frictional brakes characteristically have a rapid stop during torque application which may affect the mount surface or contribute to shaft deflection.

Printed installation information is published and available on all Stearns spring set brakes.

Determining Brake Torque

Torque ratings

Brake torque ratings are normally expressed as nominal static torque. That is, the torque required to begin rotation of the brake from a static,engaged condition. This value is to be distinguished from dynamic torque, which is the retarding torque required to stop a linear, rotating or overhauling load. As a general rule, a brake's dynamic torque is approximately 80% of the static torque rating of the brake for stopping time up to one second. Longer stopping time will produce additional brake heat and possible fading (reduction) of dynamic torque. The required dynamic torque must be converted to a static torque value before selecting a brake, using the relationship:

$$T_s = \frac{T_d}{0.8}$$

Where, T_s = Static torque, lb-ft

T_d = Dynamic torque, lb-ft

0.8 = Constant (derating factor)

All Stearns brakes are factory burnished and adjusted to produce no less than rated nominal static torque. Burnishing is the initial wear-in and mating of the rotating friction discs with the stationary metallic friction surfaces of the brake.

Although brakes are factory burnished and adjusted, variations in torque may occur if components are mixed when disassembling and reassembling the brake during installation. Further burnishing may be necessary after installation. Friction material will burnish under normal load conditions. Brakes used as holding only duty require friction material burnishing at or before installation to insure adequate torque.

When friction discs are replaced, the brake must be burnished again in order to produce its rated holding torque.

System Friction

The friction and rolling resistance in a power transmission system is usually neglected when selecting a brake. With the use of anti-friction bearings in the system, friction and rolling resistance is usually low enough to neglect. Friction within the system will assist the brake in stopping the load. If it is desired to consider it, subtract the frictional torque from the braking torque necessary to decelerate and stop the load. Friction and rolling resistance are neglected in the examples presented in this guide.

Non-overhauling Loads

There are two methods for determining brake torque for non-overhauling loads. The first method is to size the brake to the torque of the motor. The second is to select a brake on the basis of the total system or load inertia to be stopped.

Selecting Brake Torque from the Motor Data

Motor full-load torque based or nameplate horsepower and speed can be used to select a brake. This is the most common method of selecting a brake torque rating due to its simplicity. This method is normally used for simple rotary and linear inertial loads. Brake torque is usually expressed as a percent of the full load torque of the motor. Generally this figure is not less than 100% of the motor's full load torque. Often a larger service factor is considered. Refer to *Selection of Service Factor.*

The required brake torque may be calculated from the formula:

$$T_s = \frac{5{,}252 \times P}{N} \times SF$$

Where, T_s = Static brake torque, lb-ft

P = Motor horsepower, hp

N = Motor full load speed, rpm

SF = Service factor

5,252 = Constant

Match the brake torque to the hp used in the application. When an oversized motor hp has been selected, brake torque based on the motor hp may be excessive for the actual end use.

Nameplate torque represents a nominal static torque. Torque will vary based on combinations of factors including cycle rate, environment, wear, disc burnish and flatness. Spring set brakes provide a rapid stop and hold and are generally not used in repeat positioning applications.

Selection of Service Factor

A service factor is applied to the basic drive torque calculation. The SF compensates for any tolerance variation, data inaccuracy, unplanned transient torque and potential variations of the friction disc.

When using the basic equation: T= (hp x 5252) / rpm with non-overhauling loads, a service factor of 1.2 to 1.4 is typical. Overhauling loads with unknown factors such as reductions may use a service factor of 1.4 to 1.8.

Spring set brakes combined with variable frequency drives use service factors ranging from 1.0 to 2.0 depending on the system design. These holding duty brakes must be wired to a separate dedicated power supply.

Occasionally, a brake with a torque rating less than the motor full load torque or with a service factor less than 1.0 is selected. These holding or soft stop applications must be evaluated by the end user or system designer to insure adequate sizing and thermal capacity.

Typically a brake rated 125% of the motor full load torque, or with a 1.25 service factor, provides a stop in approximately the same time as that required for the motor to accelerate the load to full load speed.

Occasionally a motor is oversized or undersized for the load or application. In these situations, the load inertia and desired stopping time calculations should be used rather than relying on the service factor method alone.

Service factor selection can be based on motor performance curves. Motor rotor and load inertia should be considered in this selection process. Depending on the motor design (NEMA A, B, C and D), rpm and horsepower, the maximum torque is either the starting or breakdown torque. A NEMA design B, 3 phase, squirrel cage design motor at breakdown torque produces a minimum of 250% the full load torque. A service factor of 2.5 would be selected. Typical service factors depending on NEMA motor design are: NEMA design A or B: 1.75 to 3.0, NEMA design C: 1.75 to 3.0 and NEMA design D: not less than 2.75.

A brake with an excessive service factor may result in system component damage, an unreasonably rapid stop or loss of load control. A SF above 2.0 is not recommended without evaluation by the end user or system designer.

Example 1: Select brake torque from motor horsepower and speed.

Given: Motor power (P) - 5 hp

Motor speed (N) - 1,750 rpm

Service factor (SF) - 1.4

$$T = \frac{5{,}252 \times P}{N} \times SF$$

$$= \frac{5{,}252 \times 5}{1{,}750} \times 1.4$$

$$T = 21 \text{ lb-ft}$$

A brake having a standard rating of 25 lb-ft nominal static torque would be selected.

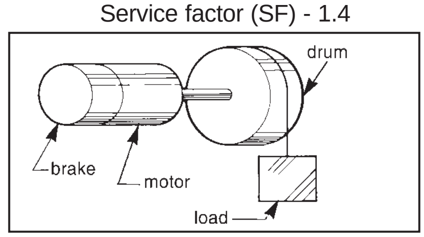
Example 2 illustrates selection of a brake to provide proper static torque to hold a load if dynamic braking were used to stop the load.

Example 2: Select a brake to hold a load in position after some other method, such as dynamic braking of the motor, has stopped all rotation.

Given: Weight of load (W) - 5 lb

Drum radius (R) - 2 ft

Service factor (SF) - 1.4



The static holding torque is determined by the weight of the load applied at the drum radius. A service factor is applied to ensure sufficient holding torque is available in the brake.

$$T_S = F \times R \times SF$$

$$= 5 \times 2 \times 14$$

$$T_S = 14 \text{ lb-ft}$$

Sizing the Brake to the Inertial Load

For applications where the load data is known, where high inertial loads exist, or where a stop in a specified time or distance is required, the brake should be selected on the basis of the total inertia to be retarded. The total system inertia, reflected to the brake shaft speed, would be:

$$Wk_T^2 = Wk_B^2 + Wk_M^2 + Wk_L^2$$

Where, Wk_T^2 = Total inertia reflected to the brake, lb-ft^2

Wk_B^2 = Inertia of brake, lb-ft^2

Wk_M^2 = Inertia of motor rotor, lb-ft^2

Wk_L^2 = Equivalent inertia of load reflected to brake shaft, lb-ft^2

Other significant system inertias, including speed reducers, shafting, pulleys and drums, should also be considered in determining the total inertia the brake would stop.

If any component in the system has a rotational speed different than the rotational speed of the brake, or any linear moving loads are present, such as a conveyor load, their equivalent inertia in terms of rotary inertia at the brake rotational speed must be determined. The following formulas are applicable:

Rotary motion:

$$\text{Equivalent } Wk_B^2 = Wk_L^2 \left(\frac{N_L}{N_B}\right)^2$$

Where,

Equivalent Wk_B^2 = Inertia of rotating load reflected to brake shaft, lb-ft^2

Wk_L^2 = Inertia of rotating load, lb-ft^2

N_L = Shaft speed at load, rpm

N_B = Shaft speed at brake, rpm

Horizontal Linear Motion

$$\text{Equivalent } Wk_W^2 = W\left(\frac{V}{2\pi N_B}\right)^2$$

Where,

Equivalent Wk_W^2 = Equivalent inertia of linear moving load reflected to brake shaft, lb-ft^2

W= Weight of linear moving load, lb

V= Linear velocity of load, ft/min

N_B = Shaft speed at brake, rpm

Once the total system inertia is calculated, the required average dynamic braking torque can be calculated using the formula:

$$T_d = \frac{Wk_T^2 \times N_B}{308 \times t}$$

Where, T_d = Average dynamic braking torque, lb-ft

Wk_T^2 = Total inertia reflected to brake, lb-ft^2

N_B = Shaft speed at brake, rpm

t = Desired stopping time, sec

308 = Constant

The calculated dynamic torque is converted to the static torque rating using the relationship:

$$T_S = \frac{T_d}{0.8}$$

Where, T_S = brake static torque, lb-ft

T_d = System dymanic torque, lb-ft

Examples 3, 4, 5 and 6 illustrate how brake torque is determined for non-overhauling loads where rotary or horizontal linear motion is to be stopped.

Example 3: Select a brake to stop a rotating flywheel in a specified time.

Given: Motor speed (N_M) - 1,750 rpm

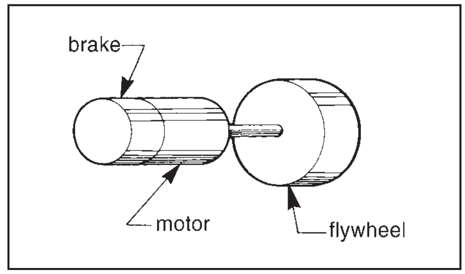
Motor inertia (Wk_M^2) - 0.075 lb-ft^2

Flywheel inertia (Wk_{FW}^2) - 4 lb-ft^2

Brake inertia (Wk_B^2) - 0.042 lb-ft^2

Required stopping time (t) - 1 sec.

First determine the total inertia to be stopped,



$$Wk_T^2 = Wk_M^2 + Wk_{FW}^2 + Wk_B^2$$

$$= 0.075 + 4 + 0.042$$

$$Wk_T^2 = 4.117 \text{ lb-ft}^2$$

The dynamic braking torque required to stop the total inertia in 1 second is,

$$T_d = \frac{Wk_T^2 \times N_{BM}}{308 \times t}$$

$$= \frac{4.117 \times 1{,}750}{308 \times 1}$$

$$T_d = 23.4 \text{ lb-ft}$$

Converting T_d to static torque

$$T_S = \frac{T_d}{0.8}$$

$$= \frac{23.4}{0.8}$$

$$T_S = 29.3 \text{ lb-ft}$$

A brake having a standard static torque rating of 35 lb-ft would be selected. Since a brake with more torque than necessary to stop the flywheel in 1 second is selected, the stopping time would be,

$$t = \frac{Wk_T^2 \times N_{BM}}{308 \times T_d}$$

$$= \frac{Wk_T^2 \times N_{BM}}{308 \times (0.8 \times T_S}$$

$$\frac{4.117 \text{ x } 1{,}750}{308 \text{ x } (0.8 \text{ x } 35)}$$

t = 0.84 sec.

See section on *Stopping Time* and *Thermal Information.*

Example 4: Select a brake to stop a rotating flywheel, driven through a gear reducer, in a specified time.

Given: Motor speed (N_M) – 1,800 rpm

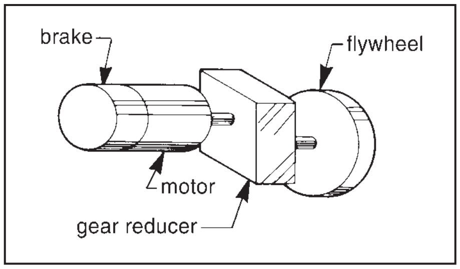
Motor inertia (Wk_M^2) - 0.075 lb-ft^2

Gear reduction (GR) - 20:1

Gear reducer inertia at high speed shaft (Wk_{GR}^2) - 0.025 lb-ft^2

Flywheel inertia (Wk_{FW}^2) - 20 lb-ft^2

Required stopping time (t) - 0.25 sec.



First determine rotating speed of flywheel (N_{FW})

$$N_{FW} = \frac{N_{BM}}{GR}$$

$$= \frac{1{,}800}{20}$$

$$N_{FW} = 90 \text{ rpm}$$

Next, the inertia of the flywheel must be reflected back to the motor brake shaft.

$$Wk_b^2 = Wk_{FW}^2\left(\frac{N_{FW}}{N_M}\right)^2$$

$$= 20\left(\frac{90}{1{,}800}\right)^2$$

$$Wk_b^2 = 0.05 \text{ lb-ft}^2$$

Determining the total WK^2,

$$WK_T^2 = WK_M^2 + WK_{GR}^2 + WK_b^2$$

$$= 0.075 + 0.025 + 0.05$$

$$Wk_T^2 = 0.15 \text{ lb-ft}^2$$

The required dynamic torque to stop the flywheel in 0.25 seconds can now be determined.

$$T_d = \frac{Wk_T^2 \times N_{BM}}{308 \times t}$$

$$T_d = \frac{0.15 \times 1{,}800}{308 \times 0.25}$$

$$T_d = 3.5 \text{ lb-ft}$$

Converting dynamic torque to static torque,

$$T_s = \frac{T_d}{0.8}$$

$$= \frac{3.5}{0.8}$$

$$T_s = 4.4 \text{ lb-ft}$$

A brake having a standard static torque rating of 6 lb-ft would be selected. Since a brake with more torque than necessary to stop the flywheel in 0.25 seconds is selected, the stopping time would be,

$$t = \frac{Wk_T^2 \times N_M}{308 \times T_d}$$

$$= \frac{Wk_T^2 \times N_M}{308 \times (0.8 \times T_s)}$$

$$= \frac{0.15 \times 1{,}800}{308 \times (0.8 \times 6)}$$

$$t = 0.18 \text{ sec.}$$

See section on *Stopping Time* and *Thermal Information*.

Example 5: Select a brake to stop a load on a horizontal belt conveyor in a specified time.

Given:

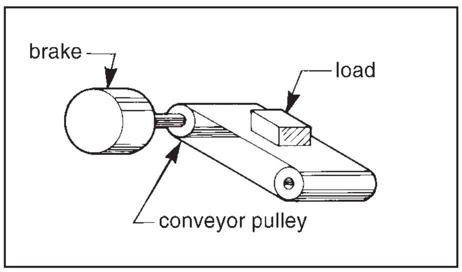
Conveyor pulley speed (N_p) - 32 rpm

Weight of load (W) - 30 lb

Conveyor pulley and belt inertia (Wk_p^2) - 4.0 lb-ft^2

Conveyor pulley diameter (d_p) - 1 ft

Required stopping time (t) - 0.25 sec.



First, convert the rotational pulley speed to linear belt speed (V_B).

$$V_B = \pi d_p N_p$$

$$= \pi \times 1 \times 32$$

$$V_B = 100.5 \text{ ft/min}$$

Next, determine inertia of load.

$$Wk_W^2 = W\left(\frac{V_B}{2\pi \times N_p}\right)^2$$

$$= 30\left(\frac{100.5}{2\pi \times 32}\right)^2$$

$$Wk_W^2 = 7.5 \text{ ft-lb}^2$$

Then, determine total inertial load

$$Wk_T^2 = Wk_W^2 + Wk_p^2$$

$$= 7.5 + 4.0$$

$$Wk_T^2 = 115 \text{ lb-ft}^2$$

The required dynamic torque to stop the conveyor load in 0.25 seconds can now be determined.

$$T_d = \frac{Wk_T^2 \times N_p}{308 \times t}$$

$$T_d = \frac{115 \times 32}{308 \times 0.25}$$

$$T_d = 4.8 \text{ lb-ft}$$

Converting dynamic torque to static torque,

$$T_s = \frac{T_d}{0.8}$$

$$= \frac{4.8}{0.8}$$

$$T_s = 6 \text{ lb-ft}$$

A brake having a standard static torque rating of 6 lb-ft would be selected. See *Thermal Information.*

Example 6: Select a brake to stop a trolley crane and its load in a specified time. Brake mounted on wheel axle.

Given:

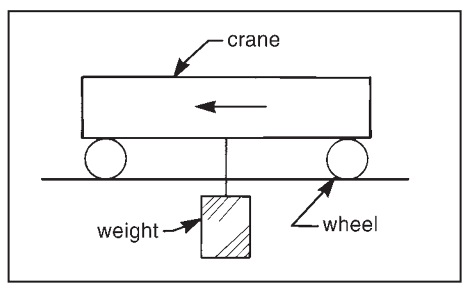
Weight of crane (W_C) - 2,000 lb

Weight of load (W_L) - 100 lb

Trolley velocity (v) - 3 ft/sec or 180 ft/min

Radius of trolley wheel (r) - 0.75 ft

Required stopping time (t) - 2 sec



The dynamic braking torque required to stop the trolley crane and load can be determined by one of two methods. The first method is to determine the equivalent inertia of the linearly moving crane and load, then calculate the dynamic braking torque. The second method is to determine the dynamic braking torque directly.

Using the first method, the total weight to be stopped is determined first.

$$W_T = W_L + W_C$$

$$= 100 + 2{,}000$$

$$W_T = 2{,}100 \text{ lb}$$

Next, the rotational speed of the axle (N_B) is calculated.

$$N_B = \frac{V}{2\pi r}$$

$$= \frac{180}{2 \times \pi \times 0.75}$$

$$N_B = 38.2 \text{ rpm}$$

Then, the equivalent inertia of the linearly moving crane and load is determined.

$$Wk_T^2 = W_T \left(\frac{V}{2\pi N_B}\right)^2$$

$$= 2{,}100 \left(\frac{180}{2\pi \times 38.2}\right)^2$$

$$Wk_T^2 = 1{,}181 \text{ lb - ft}^2$$

Finally, the dynamic braking torque required to stop the total inertia in 2 seconds is,

$$T_d = \frac{Wk_T^2 \times N_B}{308 \times t}$$

$$= \frac{1{,}181 \times 38.2}{308 \times 2}$$

$$T_d = 73 \text{ lb - ft}$$

Using the second method, the dynamic braking torque required to stop the crane and load in 2 seconds can be calculated directly using the formula,

$$T_d = \frac{W_T v}{gt} \times r$$

Where, T_d = Average dynamic braking torque, lb-ft

W_T = Total weight of linear moving load, lb

v = Linear velocity of load, ft/sec

g = Gravitational acceleration constant, 32.2 ft/sec^2

t = Desired stopping time, sec

r = Length of the moment arm (wheel radius), ft

or, for this example,

$$T_d = \frac{2{,}100 \times 3}{32.2 \times 2} \times .75$$

$$T_d = 73 \text{ lb-ft}$$

For both methods above, the required dynamic braking torque is converted to static torque,

$$T_s = \frac{T_d}{0.8}$$

$$= \frac{73}{0.8}$$

$$T_s = 91 \text{ lb-ft}$$

A smaller brake could be mounted on the high speed shaft in place of the higher torque on the low speed shaft.

A brake having a standard static torque rating of 105 lb-ft is selected. Since a brake with more torque than necessary to stop the load in 2 seconds is selected, the stopping time would be,

$$t = \frac{W_T v}{g T_d} \times r$$

$$= \frac{W_T v}{g \times (0.8 \times T_s} \times r$$

$$= \frac{2{,}100 \times 3}{32.2 \times (0.8 \times 105)} \times 0.75$$

$$t = 1.8 \text{ sec}$$

See section on *Stopping Time* and cycle rates, *Thermal Selection.* Stops should be under 2 seconds. Longer stops require application test.

Overhauling Loads

Applications with a descending load, such as power lowered crane, hoist or elevator loads, require a brake with sufficient torque to both *stop* the load, and *hold* it at rest. Overhauling loads having been brought to rest still invite motion of the load due to the effect of gravity. Therefore, brake torque must be larger than the overhauling torque in order to stop and hold the load. If brake torque is equal to or less than the overhauling torque, there is no net torque available for stopping a descending load.

First, the total system inertia reflected to the brake shaft speed must be calculated.

Second, the average dynamic torque required to decelerate the descending load in the required time is calculated with the formula:

$$T_d = \frac{Wk_T^2 \times N_B}{308 \times t}$$

Where, T_d = Average dynamic braking torque, lb-ft

Wk_T^2 = Total inertia reflected to brake, lb-ft^2

N_B = Shaft speed at brake, rpm. Consider motor slip when descending.

t = Desired stopping time,sec

Third, the overhauling torque reflected to the brake shaft is determined by the formula:

$$T_o = W \times R \times \frac{N_L}{N_B}$$

Where, T_o = Overhauling dynamic torque of load reflected to brake shaft, lb-ft

W = Weight of overhauling load, lb

R = Radius of hoist or elevator drum, ft

N_L = Rotating speed of drum, rpm

N_B = Rotating speed at brake, rpm

Or alternately, the dynamic torque to overcome the overhauling load can be calculated with the formula:

$$T_o = \frac{0.158 \times W \times V}{N_B}$$

Where, T_o = Overhauling dynamic torque of load reflected to brake shaft, lb-ft

W = Weight of overhauling load, lb

V = Linear velocity of descending load, ft/min

N_B = Shaft speed at brake, rpm

0.158 = Constant

Next, the total dynamic torque required to stop and hold the overhauling load is the sum of the two calculated dynamic torques:

$$T_t = T_d + T_o$$

Finally, the dynamic torque must be converted to static brake torque to select a brake:

$$T_s = \frac{T_d}{0.8}$$

Where, T_s = Brake static torque, lb-ft

T_t = System dynamic torque, lb-ft

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If the total inertia of the system and overhauling load cannot be accurately determined, a brake rated at 180% the motor full load torque should be selected. Refer to *Selection of Service Factor.* The motor starting torque may permit a heavier than rated load to be lifted; the brake must stop the load when descending.

Examples 7, 8 and 9 illustrate how brake torque would be determined for overhauling loads. In these examples brakes are selected using the system data rather than sizing them to the motor. Refer to the section on *Thermal Calculations* to determine cycle rate.

Consider motor slip in calculation. An 1800 rpm motor with 10% slip would operate at 1,620 rpm when the load is ascending and 1,980 rpm when descending. Motor rpm, armature inertia and load position will affect stop time. Brakes on overhauling loads should be wired through a dedicated relay.

Example 7: Select a brake to stop an overhauling load in a specified time.

Given: Cable speed (V) - 667 ft/min

Weight of load (W) - 100 lb

Drum diameter (D) - 0.25 ft

Drum inertia (Wk_D^2) - 5 lb - ft^2

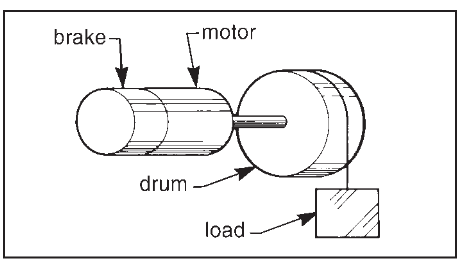
Required stopping time (t) - 1 sec

First, determine brakemotor shaft speed (N_B).

$$N_B = \frac{V}{\pi D}$$

$$= \frac{667}{\pi \times 0.25}$$

$$N_B = 849 \text{ rpm}$$



Then, determine the equivalent inertia of the overhauling load.

$$Wk_l^2 = W\left(\frac{V}{2\pi N_B}\right)^2$$

$$= 100\left(\frac{667}{2\pi \times 849}\right)^2$$

$$Wk_l^2 = 156 \text{ lb-ft}^2$$

Therefore, the total inertia at the brake is,

$$Wk_T^2 = Wk_D^2 + Wk_l^2$$

$$= 5 + 156$$

$$Wk_T^2 = 6.56 \text{ lb-ft}^2$$

Now, the dynamic torque required to decelerate the load and drum in the required time is calculated.

$$T_d = \frac{Wk_T^2 \times N_B}{308 \times t}$$

$$= \frac{6.56 \times 850}{308 \times 1}$$

$$T_d = 18.1 \text{ lb-ft}$$

Next, calculate the dynamic torque required to overcome the overhauling load.

$$T_o = W \times R$$

$$= 100 \times \frac{0.25}{2}$$

$$T_o = 12.5 \text{ lb-ft}$$

The total dynamic torque to stop and hold the overhauling load is the sum of the two calculated dynamic torques.

$$T_t = T_d + T_o$$

$$= 18.1 + 12.5$$

$$T_t = 30.6 \text{ lb-ft}$$

Dynamic torque is then converted to static torque.

$$T_s = \frac{T_t}{0.8}$$

$$= \frac{30.6}{0.8}$$

$$T_s = 38.3 \text{ lb-ft}$$

A brake having a standard torque rating of 50 lb-ft is selected based on expected stop time. Since a brake with more torque than necessary to stop the load in 1 second is selected, the stopping time would be,

$$t = \frac{Wk_T^2 \times N}{308 \times T_d}$$

where, $$T_s = \frac{T_t}{0.8}$$

$$= \frac{T_d + T_o}{0.8}$$

or, $$T_d = 0.8\,T_s - T_o$$

$$= (0.8)\,(50) - 12.5$$

$$T_d = 27.5 \text{ lb-ft}$$

therefore, $$t = \frac{6.56 \times 850}{308 \times 27.5}$$

$$t = 0.7 \text{ sec}$$

Wire the brake through a dedicated relay on overhauling loads where stop time or distance is critical. See section on *Stopping time*.

Example 8: Select a brake to stop an overhauling load driven through gear reducer in a specified time.

Given: Motor speed (N_M) - 1,150 rpm

Motor inertia (Wk_M^2) - 0.65 lb-ft^2

Gear reduction (GR) - 300:1

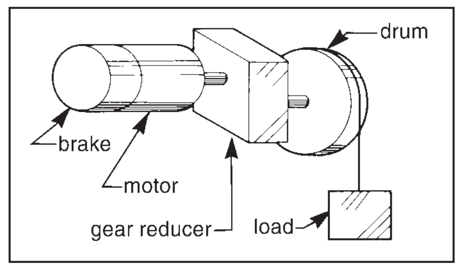
Drum diameter (D) - 1.58 ft

Weight of load (W) - 4,940 lb

Drum inertia (Wk_D^2) - 600 lb-ft^2

Required stopping time (t) - 0.5 sec

First, calculate all inertial loads reflected to the brakemotor shaft.



The rotational speed of the drum is,

$$N_D = \frac{N_M}{GR}$$

$$= \frac{1{,}150}{300}$$

$$N_D = 3.83 \text{ rpm}$$

From this, the cable speed can be determined.

$$V = N_D \times \pi D$$

$$= 3.83 \times \pi \times 158$$

$$V = 19.0 \text{ ft / min}$$

The equivalent inertia of the load reflected to the brakemotor shaft is,

$$Wk_l^2 = W\left(\frac{V}{2\pi N_{BM}}\right)^2$$

$$= 4{,}940\left(\frac{19.0}{2\pi\ 1{,}150}\right)^2$$

$$Wk_l^2 = 0.034 \text{ lb-ft}^2$$

The equivalent inertia of the drum at the brakemotor shaft speed is,

$$Wk_d^2 = Wk_D^2\left(\frac{N_D}{N_{BM}}\right)^2$$

$$= 600\left(\frac{3.83}{1{,}150}\right)^2$$

Finally, the total inertia the brake will retard is,

$$Wk_T^2 = Wk_M^2 + Wk_l^2 + Wk_d^2$$

$$Wk_T^d = .0067\ \text{lb-ft}^2$$

$$Wk_T^2 = 0.691\ \text{lb-ft}^2$$

The dynamic torque required to decelerate the total inertia is,

$$T_d = \frac{Wk_T^2 \times N_{BM}}{308 \times t}$$

$$= \frac{0.691 \times 1{,}150}{308 \times 0.5}$$

$$T_d = 5.16\ \text{lb-ft}^2$$

Now, calculate the dynamic torque to overcome the overhauling load.

$$T_o = W \times R = W \times \tfrac{1}{2}D$$

$$= 4{,}940 \times \frac{158}{2}$$

$$T_o = 3{,}903\ \text{lb-ft}$$

Which reflected to the brakemotor shaft becomes,

$$T_m = \frac{T_o}{GR}$$

$$= \frac{3{,}903}{300}$$

$$T_m = 13.0\ \text{lb-ft}$$

Then, the total dynamic torque to stop and hold the overhauling load is the sum of the two calculated dynamic torques.

$$T_t = T_d + T_m$$

$$= 5.16 + 13.0$$

$$T_t = 18.16\ \text{lb-ft}$$

Dynamic torque is then converted to static torque.

$$T_s = \frac{T_t}{0.8}$$

$$= \frac{18.16}{0.8}$$

$$T_s = 22.7\ \text{lb-ft}$$

A brake having a standard torque rating of 25 lb-ft is selected.

Example 9: Select a brake to stop and hold a load on an inclined plane (skip hoist).

Given: **Motor data**

Power (P) - 7 ½ hp

Speed (N_M) - 1,165 rpm

Rotor inertia (Wk_M^2) - 1.4 lb-ft^2

Gear reducer data

Reduction (G_R) - 110:1

Inertia at input shaft (Wk_R^2) - 0.2 lb-ft^2

Drum Data

Diameter (D_D) - 1.5 ft

Inertia (Wk_D^2) - 75 lb-ft^2

Pulley data

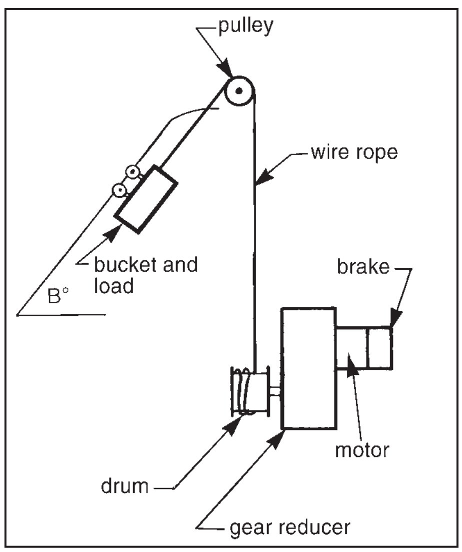
Diameter (D_P) - 1.5 ft

Inertia (Wk_P^2) - 20 lb-ft^2

Bucket weight (W_B) - 700 lb

Maximum weight of load (W_L) - 4000 lb

Slope of track (B) -52.7°



Required stopping time (t) -1 sec

The bucket is full when ascending the track and is empty when descending. When selecting a brake the most severe condition would be a fully loaded bucket backed down the hoist track. In normal operation the descending bucket would be empty. In this example, the brake is selected for the most severe condition.

The total torque to stop and hold the bucket and load when descending is the sum of (a) the torque to decelerate the total inertia and (b) the torque required to hold the loaded bucket.

First, calculate all inertial loads reflected to the brakemotor shaft. The rotational speed of the drum is

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$$N_D = \frac{N_M}{GR}$$

$$= \frac{1{,}165}{110}$$

$$N_D = 10.6\ \text{rpm}$$

From this the cable speed can be determined

$$V = N_D \times \pi D_D$$

$$= 10.6 \times \pi \times 15$$

$$V = 50\ \text{ft/min}$$

The equivalent inertia of the loaded bucket reflected to the brakemotor shaft is,

$$Wk_l^2 = W\left(\frac{V}{2\pi \times N_M}\right)^2$$

$$= 4{,}700\left(\frac{50}{2\pi \times 1{,}165}\right)^2$$

$$Wk_l^2 = 0.219\ \text{lb-ft}^2$$

Next, the inertia of the pulley and drum are reflected to the brake motor shaft speed so the total inertia at the brake can be determined.

Since the diameters of the pulley and drum are the same, 1.5 ft, their rotational speeds would be the same, 10.6 rpm.

The inertia of the pulley reflected to the brakemotor shaft is,

$$Wk_p^2 = Wk_P^2\left(\frac{N_D}{N_M}\right)^2 = Wk_P^2\left(\frac{1}{GR}\right)^2$$

$$= 20 \times \left(\frac{1}{110}\right)^2$$

$$Wk_p^2 = 0.0017\ \text{lb-ft}^2$$

The inertia of the drum reflected to the brakemotor shaft is,

$$Wk_d^2 = Wk_D^2\left(\frac{N_D}{N_M}\right)^2 = Wk_D^2\left(\frac{1}{GR}\right)^2$$

$$= 75 \times \left(\frac{1}{110}\right)^2$$

$$Wk_d^2 = 0.0062\ \text{lb-ft}^2$$

The total inertia to be stopped is,

$$Wk_T^2 = Wk_l^2 + Wk_p^2 + Wk_d^2 + Wk_R^2 + Wk_M^2$$

$$= 0.219 + 0.0017 + 0.0062 + 0.2 + 14$$

$$Wk_T^2 = 1827\ \text{lb-ft}$$

Then, the dynamic torque required to bring the descending bucket and load to rest is,

$$T_d = \frac{Wk_T^2 \times N_M}{308 \times T_d}$$

$$T_d = \frac{1827 \times 1{,}165}{308 \times 1}$$

The additional dynamic torque required to hold the overhauling load would be determined by the unbalanced component of the force acting along the plane of the hoist track, $W_T \sin B$, and the length of the moment arm which is the drum radius (R_D). $W_T \sin B$ is the force necessary to retard downward motion of the loaded hoist bucket.

$$T_o = W_T \sin B \times R_D$$

$$= W_T \sin B \times \tfrac{1}{2} D_D$$

$$= 4{,}700 \times \sin 52.7° \times \tfrac{1}{2}(1.5)$$

$$= 4{,}700 \times 0.7955 \times 0.75$$

$$T_o = 2{,}804 \text{ lb-ft}$$

Which reflected to the brakemotor shaft becomes,

$$T_m = \frac{T_o}{GR}$$

$$= \frac{2{,}804}{110}$$

$$T_m = 25.5 \text{ lb-ft}$$

Then, the total dynamic torque to stop and hold the descending bucket and load is the sum of the two calculated dynamic torques.

$$T_t = T_d + T_m$$

$$= 6.9 + 25.5$$

$$T_t = 32.4 \text{ lb-ft}$$

Converting to static torque,

$$T_s = \frac{T_t}{0.8}$$

$$= \frac{32.4}{0.8}$$

$$T_s = 40.5 \text{ lb-ft}$$

A brake having a standard torque rating of 50 lb-ft is selected. Since a brake with more torque than necessary to stop the load in 1 second is selected, the stopping time would be,

$$t = \frac{W_T^2 \times N_M}{308 \times T_d}$$

Where, $T_s = \dfrac{T_t}{0.8}$

$$= \frac{T_d + T_m}{0.8}$$

or, $T_d = 0.8\, T_s - T_m$

$$= (0.8)\,(50) - 25.5$$

$$T_d = 14.5 \text{ lb - ft}$$

therefore,

$$t = \frac{1827 \times 1165}{308 \times 14.5}$$

$$t = 0.48 \text{ sec}$$

See section on *Stopping time.*

Stopping Time and Deceleration Rate

In the formulas used to determine dynamic torque, stopping time or "t" in seconds is a desired or assumed value selected on the requirements of the application. For optimum brake performance, a stopping or braking time of 1 second or less is desirable. Stop times between 2 and 3 seconds require test. *A brake of insufficient torque rating will lengthen the stopping time. This may result in overheating of the brake to a point where torque falls appreciably. The friction material could carbonize, glaze, or fail.*

After determining the braking torque required by a system, it may be necessary to recalculate the stopping time based on the actual brake size selected to insure that stopping time falls within the 0 to 2 second range. Any formula, where the stopping time is a variable, may be rewritten to solve for the new stopping time. For instance, the dynamic torque equation may be transposed as follows:

$$T_d = \frac{Wk_T^2 \times N_B}{308 \times t}$$

or, $$t = \frac{Wk_T^2 \times N_B}{308 \times (0.8 \times T_s)}$$

Where, t = Stopping time, sec

Wk_T^2 = Total inertia reflected to brake, lb-ft^2

N_B = Shaft speed at brake, rpm

T_s = Nominal static torque rating of brake, lb-ft

T_d = Dynamic braking torque ($0.8 \times T_s$), lb-ft

0.8 = Constant (derating factor)

308 = Constant

Brakes are rated in static torque. This value is converted to dynamic torque, as done in the above equation, when stopping time is calculated. That is,

$$T_d = 0.8 \times T_s$$

where, T_d = Dynamic braking torque, lb - ft

T_s = Nominal static torque rating of brake, lb-ft

The approximate number of revolutions the brake shaft makes when stopping is:

$$\text{Revolutions to stop} = \frac{t \times N_B}{120}$$

Where, t = Stopping time, sec

N_B = Shaft speed at brake, rpm

120 = Constant

The average rate of deceleration when braking a linearly moving load to rest can be calculated using the stopping time determined by the above formula and the initial linear velocity of the load.

$$a = -\frac{V_i}{t}$$

Where, a = Deceleration, ft/sec^2

V_i = Initial linear velocity of load, ft / sec

t = Stopping time, sec

RPM Considerations

The maximum allowable rotational speed of the brake should not be exceeded in braking. Maximum brake rpm as listed in the catalog is intended to limit stopping time to 2 seconds or less and insure friction disc stability. Brakes are not dynamically balanced because of the low brake inertia.

Determining Required Thermal Capacity

Thermal Ratings

When a brake stops a load, it converts mechanical energy to thermal energy or heat. The heat is absorbed by components of the brake. This heat is then dissipated by the brake. The ability of a given brake to absorb and dissipate heat without exceeding temperature limitations is known as thermal capacity.

There are two categories of thermal capacity for a brake. The first is the *maximum* energy the brake can absorb in one stop, generally referred to as a "crash" or "emergency" stop. The second is the heat dissipation capability of the brake when it is cycled frequently. *To achieve optimum brake performance, the thermal rating should not be exceeded. They are specified for a predetermined maximum temperature rise of the brake friction material.*

The ability of a brake to absorb and dissipate heat is determined by many factors, including the design of the brake, the ambient temperature, brake enclosure, position of the brake, the surface that the brake is mounted to, and the altitude.

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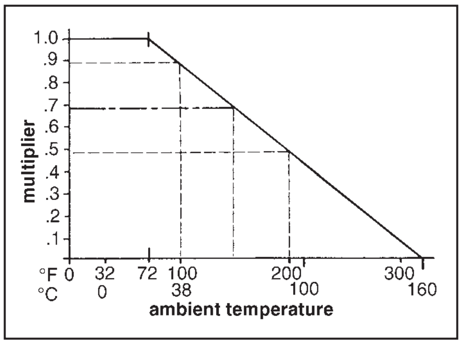
The rating for a given brake is the maximum allowable. Longer brake life results when the brake has more thermal capacity than a power transmission requires. Much shorter life or brake failure will result when the thermal capacity rating is exceeded. Ratings are determined at an ambient temperature of 72°F *(22°C)*, with the brake in a horizontal position, with a stopping time of 1 second or less, and with no external heat source such as a motor.

Ambient temperature will limit the thermal capacity of a brake. Temperatures above 72°F *(22°C)* require derating of the thermal capacity rating. For example, at 150°F, thermal capacity is reduced approximately 30% (see *Derating Thermal Capacity Chart*).

CHART: Derating Thermal Capacity



A temperature range of 20°F *(0°C)* to 104°F *(40°C)* is acceptable in most brake applications. Above 104°F also consider Class H coil insulation.

Thermal capacity ratings are determined with enclosures on the brake. Other customer furnished covers or cowls may affect a brake's thermal capacity. The effect on thermal capacity should be evaluated. In some cases, thermal capacity may be increased by use of air or liquid cooling. However, provisions must be made to prevent contaminating the brake internally.

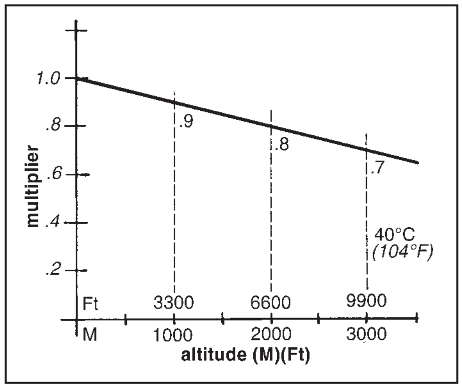
Brakes with brass stationary discs are derated 25%.

The mounting position of a brake will also affect thermal capacity. The specified ratings are for brakes mounted in a horizontal position with the solenoid plunger above the solenoid. For brakes mounted in a vertical position, or 15° or more from horizontal, the thermal capacity decreases due to friction disc drag. Brakes are modified for vertical operation to minimize the drag. 2- and 3- disc brakes are derated 25%, 4-disc brakes are derated 33%. 4- and 5-disc brakes are not recommended for vertical use.

Thermal capacity ratings are established without external sources of heat increasing the brake temperature. The surface that a brake is mounted to, such as an electric motor or gear reducer, will limit the heat dissipation capability or thermal capacity of a brake. These sources of heat should be evaluated when determining the thermal requirements of the system for which the brake is selected.

High altitudes may also affect a brake's thermal capacity. Stearns brakes will operate to 10,000 ft above sea level at 72°F *(22°C)* ambient temperature. At 104°F *(40°C)* ambient temperature, altitude and temperature adjustments occur. Refer to *NEMA MG1-1993 Section 14 for additional information.*

CHART: Altitude & Thermal Capacity



Maximum Energy Absorption

The thermal capacity of a brake is limited by the maximum energy it can absorb in one stop. This factor is important when stopping extremely high inertial loads at infrequent intervals. Such use of a brake requires extensive cooling time before it can be operated again.

The energy a brake is required to absorb in one stop by a given power transmission system is determined by the formulas below. *The calculated energy of the system should not exceed the maximum kinetic energy rating of the brake. System energy exceeding the brake's maximum rating may result in overheating of the brake to a point where torque falls appreciably. The friction material of the brake could glaze, carbonize or fail.*

In the case of linear loads, the energy that the brake must absorb is kinetic energy. It is determined by the formula:

$$KE_l = \frac{Wv^2}{2g}$$

KE_l = Kinetic energy of linear moving load, lb-ft

W = Weight of load, lb

v = Linear velocity of load, ft/sec

g = Gravitational acceleration constant, 32.2 ft/sec^2

In the case of rotational loads, the energy that the brake must absorb is also kinetic energy. It is determined by the formula:

$$KE_r = \frac{Wk_r^2 \times N_B^2}{5875}$$

Where, KE_r = Kinetic energy of linear load, lb-ft

Wk_T^2 = Inertia of the rotating load reflected to brake shaft, lb-ft^2

N_B = Shaft speed at brake, rpm

5875 = Constant

In the case of overhauling loads, both the kinetic energy of the linear and rotating loads and the potential energy transformed into kinetic energy by the change in height or position must be considered when determining the total energy that the brake must absorb. The potential energy transformed to kinetic energy is determined by the formula:

$$PE = Ws$$

Where, PE = Change in potential energy, ft-lb

W = Weight of overhauling load, lb

s = Distance load travels, ft

Thus, the total energy to be absorbed by a brake stoping an overhauling load is:

$$E_T = KE_l + KE_r + PE$$

Example 10 illustrates how energy absorption for Example 8 would be determined for one stop.

Example 10: Determine the total energy absorbed by a brake in one stop.

In Example 8, the calculation for total energy to be absorbed would be as follows.

First, calculate the kinetic energy of the linear load. The load weight was 4,940 lb and the velocity is 19 ft/min or 0.317 ft/sec. The kinetic energy is:

$$KE_l = \frac{Wv^2}{2g}$$

$$= \frac{4{,}940 \times 0.317^2}{2 \times 32.2}$$

$$KE_l = 7.71 \text{ ft-lb}$$

Next, calculate the kinetic energy for the rotational load. The motor inertia is 0.65 lb-ft^2 and the drum inertia reflected to the brake shaft speed is 0.0067 lb-ft^2. The total rotational inertia at the brakemotor shaft is,

$$Wk_r^2 = Wk_M^2 + Wk_d^2$$

$$= 0.65 + 0.0067$$

$$Wk_r^2 = 0.6567 \text{ lb-ft}^2$$

And the kinetic energy of the rotating components is,

$$KE_r = \frac{Wk_r^2 \times N_B^2}{5{,}875}$$

$$= \frac{0.6567 \times 1{,}150^2}{5{,}875}$$

$$KE_l = 147.8 \text{ ft-lb}$$

Now, calculate the potential energy converted to kinetic energy due to the change in position of the load while descending. A descending load is the most severe case since potential energy is transformed to kinetic energy that the brake must absorb. A 25 lb-ft brake was selected in Example 8. The 25 lb-ft static torque rating is converted to dymanic torque,

$$T_t = T_s \times 0.8$$

$$= 25 \times 0.8$$

$$T_t = 20 \text{ lb-ft}$$

Of this torque, 13.0 lb-ft is required to overcome the overhauling load as determined in Example 8. The dynamic torque available to decelerate the load is,

$$T_d = T_t - T_m$$

$$= 20 - 13$$

$$T_d = 7 \text{ lb-ft}$$

The stopping time resulting from this dynamic torque is,

$$t = \frac{Wk_T^2 \times N_M}{308 \times T_d}$$

$$= \frac{0.691 \times 1{,}150}{308 \times 7}$$

$$t = 0.369 \text{ sec}$$

Where $Wk_T^2 = 0.690$ lb-ft^2 is the total inertia the brake is to retard as determined in Example 8. With the load traveling at 19.0 ft/min or 0.317 ft/sec, the distance it will travel is,

$$s = \tfrac{1}{2} vt$$

$$= \tfrac{1}{2} \times 0.317 \times 0.369$$

$$s = 0.059 \text{ ft}$$

Wire the brake through a dedicated relay on overhauling loads where stop time or distance is critical. The potential energy transformed to kinetic energy in this distance would be,

$$PE = Ws$$

$$= 4{,}940 \times 0.059$$

$$PE = 291 \text{ ft-lb}$$

Thus, the total energy to be absorbed by the brake would be,

$$E_T = KE_l + KE_r + PE$$

$$= 7.71 + 147.8 + 291$$

$$E_T = 447 \text{ lb-ft}$$

The 25 lb-ft brake selected in Example 8 should be capable of absorbing 447 ft-lb of energy. The brake's maximum kinetic energy absorption rating should exceed this value.

Motor slip and test loads (150% of load) should be considered both in sizing and thermal calculations.

Brakes overheated in testing will require inspection before using in the standard application.

Heat dissipation in cyclic applications

In general, a brake will repetitively stop a load at the duty cycle that a standard electric motor can repetitively start the load. A brake's thermal capacity is based upon the heat it can absorb and dissipate while cycling. The thermal capacity ratings for brakes are listed in the specification tables for specific brake models.

The energy that a brake is required to absorb and dissipate by a given power transmission system is determined from the total inertia of the load and system, the rotating or linear speed of the load, and the number of times the load is to be stopped in a given time period. The rate of energy dissipation is expressed in horsepower seconds per minute (hp-sec/min). Other common units for energy rates, such as foot pounds per second (ft-lb/sec), can be converted to hp-sec/min using the conversion factors given in the *Technical Data* section.

Refer to the Thermal Capacity Chart for use above 104°F *(40°C)* ambient temperature.

For applications demanding optimum brake performance, such as high inertial loads and frequent stops, the rate of energy dissipation required by the system is determined using the following formulas. *The calculated rate of energy dissipation should not exceed the thermal capacity of the brake. Thermal dissipation requirements exceeding the brake's rating may result in overheating of the brake to a point where torque falls appreciably. The friction material of the brake could glaze, carbonize or fail.*

For rotating or linear loads, the rate at which a brake is required to absorb and dissipate heat when frequently cycled is determined by the relationship:

$$TC = \frac{Wk_T^2 \times N_B^2 \times n}{3.2 \times 10^6}$$

Where, TC = Thermal capacity required for rotating or linear loads hp-sec/min

Wk_T^2 = Total system inertia reflected to brake, lb-ft^2

N_B = Shaft speed at brake, rpm

n= Number of stops per minute, not less than 1

3.2×10^6 = Constant

The rotating speed enters the formula as a squared function. Therefore, thermal requirements are of particular significance in systems where the brake will be operated at high speeds.

For overhauling loads, the rate at which a brake is required to absorb and dissipate heat when frequently cycled is determined by the relationship:

$$TC = \frac{E_T \times n}{550}$$

Where, TC = Thermal capacity required for overhauling loads, hp-sec/min

E_T = Total energy brake absorbs, ft-lb

n= Number of stops per minute, not less than 1

550 = Constant

Example 11 illustrates how the required thermal capacity would be determined for Example 4.

Example 11: Determine the thermal capacity required to stop a rotating load frequently.

Referring back to Example 4, the flywheel will be stopped 20 times per minute. The required thermal capacity of the 6 lb-ft brake selected in this example is determined as follows.

The total inertial load the brake is to retard is 0.15 lb-ft^2. The shaft speed of the brake motor is 1,800 rpm. Therefore, the required thermal capacity is,

$$TC = \frac{Wk_T^2 \times N_M^2 \times n}{3.2 \times 10^6}$$

$$= \frac{0.15 \times 1{,}800^2 \times 20}{3.2 \times 10^6}$$

$$TC = 3.0 \text{ hp-sec/min}$$

The 6 lb-ft brake selected in Example 4 should have a thermal capacity rating equal to or greater than 3.0 hp-sec/min.

A brake with greater thermal capacity will result in greater wear life.

If productivity is to be improved in Example 4 by increasing the cycle rate, the maximum number of stops per minute is determined by the rated thermal capacity of the brake. If the 6 lb-ft brake selected in Example 4 has rated thermal capacity of 9 hp-sec/min, the maximum permissible stops per minute would be determined by transposing the above formula to,

$$n_{max} = \frac{TC_{rated} \times (3.2 \times 10^6)}{Wk_T^2 \times N_M^2}$$

$$= \frac{9 \times (3.2 \times 10^6)}{0.15 \times 1{,}800^2}$$

$$n_{max} = 59 \text{ stops/min}$$

So, the brake could be operated up to 36 times per minute without exceeding its ability to absorb and dissipate the heat generated by the frequent stops and meet the maximum solenoid cycle rating. *Cycle rate cannot exceed the solenoid cycle rate appearing in the catalog.*

Electrical Considerations

Stearns spring-set disc brakes are available with standard NEMA motor voltages, frequencies and insulation classes. Most models can be furnished with either single or dual voltage AC coils. Both AC and DC brake solenoid coils are available on most models. Solenoid coils with special voltages and frequencies are also available. All AC coils are single-phase and can be wired to either single or three-phase motors without modification.

AC and DC brake solenoid coils have a voltage range of ±10% of the rated nameplate voltage at rated frequency.

The inrush and holding current for specific coil sizes is given on the specification pages for each series to help select wire size and circuit protection for installing a brake. The standard voltage ratings are also listed.

Class B insulation is standard for solenoid coils in most brake models. Class H insulation is also available. Maximum coil temperatures are:

Class A insulation, 221°F *(105°C)*

Class B insulation, 266°F *(130°C)*

Class H insulation, 356°F *(180°C)*

These temperature limits are established by UL control standards including a maximum ambient temperature of 104°F *(40°C)*. If ambient temperature is above 104°F *(40°C)*, or there is rapid cycling, or high inertial loads, a Class H insulated coil is recommended.

Stearns Brake solenoid coils are rated for continuous duty and can be energized continuously without overheating. When cycled, the coil heating effects are greatest due to inrush current and the cycle rate must not exceed the maximum rating given in the specification tables, in the catalog, for specific brake models.

Environmental Considerations

Brakes with standard open enclosures when mounted on NEMA C-face motors are drip-proof, except where a manual release lever has a clearance opening in the housing. The standard enclosure is commonly used on open, drip-proof and enclosed motors operating indoors or in protected outdoor environments.

NEMA 4, IP 54 enclosures are available on most brake models and are commonly used for outdoor installations, or where there are moist, abrasive or dusty environments. Standard and severe duty NEMA 4 enclosures are available in some brake series.

Brakes of various styles and materials for above or below deck on ships and dockside installation are available. The materials are usually specified by the ship designers or Navy specification MIL-B-16392C. Brakes are also available to meet MIL-E-17807B for shipboard weapon and cargo elevators. Refer to *Marine, Maritime and Navy Catalog* pages.

Brakes Listed by Underwriters Laboratories, Inc. and certified by Canadian Standards Association are available for use in hazardous locations, including Class I, Groups C and D; and Class II, Groups E, F and G. *Motor-mounted*, hazardous-location electric disc brakes are listed only when mounted to a Listed hazardous-location motor of the same Class and Group at the motor manufacturer's facility, and where the combination has been accepted by UL or CSA. This procedure completes the hazardous duty assembly of the brake. However, foot-mounted hazardous-location disc brakes that are Listed are also available for coupling to a motor, and may be installed by anyone.

Hazardous-location brakes are *not* gasketed unless indicated in the brake description. The enclosure prevents flame propagation to the outside atmosphere through controlled clearances. Protection from weather and washdowns must be provided. If the brake is used in a high humidity or low temperature environment, internal electric heaters should be used.

Standard ambient temperature range for brake operation is from 20°F *(0°C)* to 104°F *(40°C)*. Refer to *Thermal Ratings* section for brake operation at higher ambient temperatures. Heaters may be available for brake operation at low ambient temperatures and high humidity environments. Ductile iron construction and heaters are recommended for prolonged cold climate use.

Conclusion

The spring-set, electrically released disc brake is an important accessory to electric motors used in cycling and holding operations. It is available in a wide variety of enclosures. In most applications, a brake requires no additional wiring, controls or auxiliary electrical equipment. It is simple to maintain since the replaceable items, the friction discs, can be easily changed.

Many spring-set motor brakes are equipped with features such as simple wear adjustment to provide optimum friction disc life, visual wear indicator, torque adjustment and manual release. Featured on some types of brakes is automatic adjustment to compensate for friction disc wear. This feature eliminates the need for periodic adjustment and is advantageous in remote or inaccessible locations. Not all of the brakes on the market provide all of these features, but there are many Stearns motor brakes offering these features.

Care should be exercised in properly selecting a brake giving due consideration to torque as well as environment and thermal requirements. On applications where all the pertinent information is not available, selection must be based on previous experience of the designer and user, as well as the brake manufacturer, and should be confirmed by tests under actual operating conditions. If the brake is selected with reasonable allowances made for extremes in operating conditions, it will perform its task with little attention or maintenance.