"Seller" and/or "Stearns" refers to Rexnord Industries, LLC (which sells products and services under the Stearns brand) for the entirety of this catalog, warranty, products, and services.

The performance of Stearns brakes, clutches, clutch-brake combinations, solenoids, and controls depends upon the proper application of the product, adequate run in, installation and maintenance procedures, and reasonable care in operation.

All torque values listed in our bulletins are nominal and are subject to the variations normally associated with friction devices. The purchaser should take into consideration all variables shown in the applicable specification sheets. Although our application engineers are available for consultation, final selection and performance assurance on the purchaser's machine is the responsibility of the purchaser. Careful purchaser selection, adequate testing at time of installation, operation and maintenance of all products of the seller are required to obtain effective performance.

Stearns warrants to its purchasers that all its products will be free from defects in material and workmanship at the time of shipment to the purchaser for a period of one (1) year from the date of shipment. All warranty claims must be submitted in writing to Stearns within the warranty period, or shall be deemed waived. As to products or parts thereof which Stearns finds to have been defective at the time of shipment, its sole responsibility hereunder shall be to repair, correct or replace (whichever Stearns deems advisable) such defective products or parts without charge, FOB Stearns factory. In the alternative, Stearns may, at its option, either before or after attempting a different remedy, refund the purchase price upon return of the product or parts.

This warranty shall not apply to any product which has been subjected to misuse: misapplication: neglect (including but not limited to improper maintenance and storage); accident: improper installation; modification (including but not limited to use of other than genuine Stearns replacement parts or attachments); adjustment; or repair.

THE FOREGOING IS IN LIEU OF ALL OTHER WARRANTIES, WHETHER EXPRESSED, IMPLIED, OR STATUTORY, INCLUDING THAT OF MERCHANTABILITY AND OF FITNESS FOR A PARTICULAR PURPOSE, AND OF ANY OTHER OBLIGATION OR LIABILITY ON OUR PART OF ANY KIND OR NATURE WHATSOEVER.

No Stearns representative has any authority to waive, alter, vary or add to the terms hereof without prior approval in writing, to our purchaser, signed by an officer of the seller.

Stearns liability for its products, whether for breach of contract, negligence, strict liability in tort, or otherwise, shall be limited to the repair, correction, or replacement of the products or parts thereof, or to the refund of the purchase price of such products or parts. Stearns will not be liable for any other injury, loss, damage or expense, whether direct or consequential, including but not limited to loss of use, income, profit or production, or increased cost of operation, or spoilage of or damage to material, arising in connection with the sale, installation, use of, inability to use, or the repair or replacement of, or late delivery of, Stearns products.

Any cause of action for breach of the foregoing warranty must be brought within one (1) year from the date the alleged breach occurs.

Note on Special Applications:

Stearns products are designed for standard industrial and commercial applications. Operating requirements, environments and required tolerances such as in nuclear and aircraft applications may be beyond the commercial standards of the Stearns Divisions products. Stearns will assume absolutely no responsibility for the use of and/ or resale of Stearns products for such applications unless approved in writing in advance by Stearns.

View the most up-to-date terms and conditions at www. regalrexnord.com/terms-and-conditions-of-sale.

Introduction to Armature Actuated Brakes (AAB)

Stearns armature actuated brakes (AAB) are spring-set, electrically released friction devices that develop holding and braking torque in the absence of electrical power. This type of brake can decelerate and hold a rotational load or can be ordered to provide a holding function only, where the motor is used as the dynamic brake.

Available sizes range from 3 lb-in up to 300 lb-ft static torque, in a variety of mounting options. A short hub is available for face mounting or to provide for maximum space efficiency. Features include Class H magnet wire coils, corrosion resistance and optional manual release lever. Custom designs and modifications are possible; consult the factory for more information.

Operating Principle

A hub which is attached to the shaft supports the rotatable friction disc. Brake torque is developed when springs apply a clamping force between the brake armature, friction disc and pressure plate. When electrical power is applied, the armature is pulled by the electromagnetic force in the magnet body assembly which overcomes spring action allowing the friction disc to rotate freely. When electrical power is interrupted, the electromagnetic force is removed and the pressure spring will mechanically force the armature plate to clamp the friction disc between itself and the pressure plate, thereby torque is developed.

Disengaged Condition (power on)

Armature Actuated Brakes Product Overview

Series 321 & Series 322 Totally-Enclosed Non-Ventilated (TENV)

Sizes from 3 to 72 lb-in (.34 to 8.14 Nm) of torque.

Series 321 & Series 322 NEMA C-Face TENV

Direct mounting to 48C and 56C motors.

Sizes from 1.5 to 25 lb-ft (2 to 34 Nm) of

torque.

Series 331 & 333

High performance brake for metric/IEC applications.

Direct drop-in for Kebco, Lenze and Binder. Sizes ranging from 3 to 300 lb-ft, (4 to 400 Nm) of torque.

(47-400 Nm).

AAB Modifications Descriptions & Availability

Armature Actuated Brakes (AAB) Torque Selection

Select the proper torque rating based on horsepower and rpm (speed at the clutch or brake) using the torque selection chart below. Based on 1.4 service factor.

For other service factors and speeds, use the formulas shown below.

Formula for TABLE 1 Formula for TABLE 2 $\overline{)}$

P = Horsepower, hp

N = Shaft speed at brake, rpm

SF = Service Factor

63,025 = Constant

 $T =$ Static torque, lb-ft. P = Horsepower, hp N = Shaft speed at brake, rpm SF = Service Factor 5,252 = Constant T = $\frac{63,025 \times P}{N}$ x SF T = $\frac{5,252 \times P}{N}$ x SF

Caution: Do not use Table 1 to select brakes for overhauling or high inertial loads, or where a stop in specified time or distance is required. For these applications the total inertia of the load and power transmission system must be determined to make a brake selection. Refer to sections on torque and thermal ratings and determination.

NOTE: Series 310 and 311 for holding applications only.

TABLE 1

Series 320, 321 & 322 Static Torque in lb-in. *(Nm)*

TABLE 2

Series 333, 350 & 360 Static Torque in lb-ft. *(Nm)*

* Exceeds maximum speed rating.

Series 310 Mounting Face: Servo Motor

Static Torque: 10 through 350 lb-in (1.13 to 39.5 Nm) *Installation & Service Instructions:* P/N 8-078-888-00

Features

- Spring activated and DC voltage released
- High torque, compact size
- Splined hub for quiet dependable operation
- Corrosion resistant finishes
- Low inertia rotating parts
- Holding applications only
- Standard voltages 24 and 90 Vdc; 115 and 230 Vac
- Available voltages 12, 36, 48 and 180 Vdc
- Coil insulation: Class 180(H)

Option A, Long Hub

Provision for (4) B diameter mounting screws on K bolt circle

Series 310 Continued Mounting Face: Servo Motor

Engineering Specifications

Ordering Information

Voltages – **TABLE 2** Hub bore and Keyseat – **TABLE 1** Character **Options Available by size** 0 standard brake ALL H Quiet operation, 1.79, 2.0, 2.
plastic carrier disc 3.35^{*}, 4.25^{*} 1.79, 2.0, 2.87, *With carrier disc option, size 3.35 torque is 50 lb-in holding, and size 4.25 torque is 100 lb-in holding

0 None **Numeral Enclosure**

3= Reduced Torque

Table 2: Coil Voltage 310-2401-0H-XA

*These voltages are non-standard.

Table 3: Options 310-24010-0H-JX

*Sizes 3.35, 4.25 and 5.0 have a pass-through hub.

NOTE: Non-standard bore sizes available, contact factory.

Series 311 Mounting Face: Servo Motor

Static Torque: 75 through 400 lb-in (8.5 to 45.2 Nm) *Installation & Service Instructions:* P/N 8-078-888-00

Features

- Spring activated and DC voltage released
- Splined hub for quiet dependable operation
- Corrosion resistant finishes
- Low inertia rotating parts
- Holding applications only
- Standard voltages 24 and 90 Vdc
- Available voltages 12, 36, 48 and 180 Vdc
- Coil insulation: Class 180(H)

Dimensions in Inches (*mm***)**

NOTE: Mounting bolt circles, mounting hole thread sizes, and quantity of mounting holes can be changed to meet your requirements. Please contact factory to request mounting dimensions other than those shown here.

Series 311 Continued Mounting Face: Servo Motor

Engineering Specifications

Ordering Information

Table 1: Hub Bores 311-64010-XX-JA

Table 2: Coil Voltage 311-6401-0H-XA

*These voltages are non-standard.

Table 3: Options 311-64010-0H-JX

*Pass-through hub.

NOTE: Non-standard bore sizes available, contact factory. *Keyseats made to ANSI B17.1 standard.

Series 320 Mounting Face: Small Gear Motor

Static Torque: 3 through 50 lb-in / (.34 to 5.65 Nm)

Installation & Service Instructions: P/N 8-078-889-00

Features

- UR and CUR recognized insulation system, E-125303 and sizes 1.8, 2.8 brakes with internal power supply File E-71115
- Class B temperature rise with Class H mag wire
- Holding (H) or dynamic (D) stopping applications
- Corrosion resistant

Options

- "Double D" friction discs are available in 3-7 lb-in dynamic and 3-15 lb-in holding brakes. DD shafts fit the brake directly without a brake hub, no shaft keyway cost and simplify assembly.
- Maintained manual release
- Mounting plates
- AC rectifiers internal or external in-line
- Band seal (not available for 1.2 size)

D

ਾ Shaft

OD Disc

.25 + .05/-.00 $.30 + .13/-.00$ **Brake** A B .075 max. .075 max.

E .3135 /.3115 .105 /.103 .376 /.374 .126 /.124 .052 .063 5/16 3/8 $\begin{array}{|c|c|c|c|c|}\n\hline\n\text{Shaff} & \text{C} & \text{D}\n\end{array}$ **Size**

Dimensions in Inches (mm)

1.2 1.8

Size

*Size 2.8 can be pressure plate mounted using the long hub. The F**`** dimension shown for size 2.8 is for pressure plate mount using the long hub. **No thru bore with manual release option.

Series 320 Continued Mounting Face: Small Gear Motor

 $\mathbb D$ For holding applications only. $\mathcal D$ Set screws located 120° from keyway.

Table 1: 320-44010-XX-JD

Engineering Specifications

(Double "D" Bores)

NOTE: Can be used up to 15 lb-in for holding.

NOTE: 65°C maximum ambient temperature for all external in-line rectifiers.

Table 3: Options 320-44010-0H-JX

*Short hub not required for size 2.8 pressure plate mount.

NOTE: Final part number may change due to specifications or options selected or other product design considerations. A number such as a 2, 3, 4 etc., in the 12th position is used to designate a unique brake (custom) and can only be assigned by Stearns Design Engineering Department.

Modifications are available - see AAB Modification Section.

 $\textcircled{1}$ Set screws located 120° from keyway. $@$ Hubs are provided without keyway. *Keyseats made to ANSI B17.1 standard.

Series 321 & Series 322 Totally-Enclosed Non-Ventilated (TENV)

Mounting Face: Small Gear Motor

Static Torque: 3 lb-in through 72 lb-in (.34 to 8.14 Nm)

IP Rating: 321 Series IP 42; 322 Series IP 54

Installation & Service Instructions: P/N 8-078-901-00

Features

• UR and CUR recognized insulation system - File E-125303; sizes 1.8 and 2.8 brakes and 48C and 56C motor frame brakes with internal power supply File E-71115

- Class B temperature rise with Class H mag wire
- Pressure plate mount
- Direct-acting, quiet operation

Options

- "Double D" friction discs are available in 3-7 lb-in dynamic and 3-15 lb-in holding brakes. DD shafts fit the brake directly without a brake hub, no shaft keyway cost and simplify assembly.
- Maintained manual release lever, or non-maintained pull release
- Through-shaft
- AC rectifiers internal or external in-line (availability depends on size)

Size 2.8 with thrushaft and manual release option.

Size 1.8 flange mount with manual release option.

Engineering Specifications

Set screws located 120°** from keyway.

Series 321 & Series 322 TENV Continued

Small Gear Motor

*Also, the location of the external leads for the 322 series is on the backside of the magnet body - not on the side (as shown in the above drawings) as with the 321 series.

Series 321 & Series 322 TENV Options

Small Gear Motor

THROUGH-SHAFT

when the brake is manually released.

Series 321 & Series 322 TENV Continued Small Gear Motor

**N/A = Not available.

***Mounting bolt circle inside (less than) the outside diameter of magnet body. (BC dimensions shown on AAB 320 Series page).

Ordering Information

Modifications are available - see AAB Modification Section.

Stearns Design Engineering Department.

Series 321 & Series 322 Totally-Enclosed Non-Ventilated (TENV) Mounting Face: NEMA 48 & 56C

Features

- IP Rating: Series 321 is IP 42; Series 322 is IP 54
- Out-of-box torque no burnishing required
- Class B temperature rise with class H mag wire
- Brake housing integrated with mag body creating a heat exchanger that keeps the brake coil cool
- Field replaceable coil
- Splined hub and friction disc
- High cycling

Options

- Internal rectifier
- Quick-set rectifiers for fast response time even when wired directly across motor
- Maintained manual release
- Brake release indicator switch

- Single point torque adjustment- to 50% of nameplate torque
- Through-shaft
- Conduit box
- Quiet armature actuations
- Carrier ring friction disc

Series 321 & Series 322 TENV Continued NEMA Frame

Engineering Specifications

Note: Available as standard or with manual release.

Ordering Information

Other voltages available.

NOTE: Final part number may change due to specifications or options selected or other product design considerations. A number such as a 2, 3, 4 etc., in the 12th position is used to designate a unique brake (custom) and can only be assigned by Stearns Design Engineering Department.

Modifications are available - see AAB Modification Section.

Direct replacement for European brakes: Kebco, Lenze & Binder*

Features

- Direct acting
- Torque 3 through 300 lb-ft (4 to 400 Nm)
- UL recognized Class H coil insulation system to US standards (UR)
- Spring-set and DC voltage released AC rectifiers optional
- Series 333 torque adjustable
- Pre-adjusted air gap for easy assembly
- Corrosion resistance
- Spline hub for quiet dependable operation
- Metric and US customary bore sizes

Product Overview

331 Series

Basic brake without the torque adjust option.

Available in torque ratings from 3 to 300 lb-ft (4 to 400 Nm).

Manual release optional, can be provided with non-maintained release lever or maintained release bolts.

Metric mount; also can be ordered with C-face adapter or as the C-face enclosed version, and as severe duty.

333 Series

Static torque from 3 to 300 lb-ft, with nine different sizes ranging from 72mm bolt circle up to 278 mm bolt circle.

Torque can be adjusted down to approximately 50% of the nameplate torque rating.

Shown here with optional nonmaintained manual release lever; other options include boot (band seal), end cap plug, through-shaft seal, and many more listed in the AAB modification section.

33X Series with NEMA C-Face Adapter

Series 331 or Series 333 can be provided with a C-face adapter for motor frames from 48C through 404/405TC, TSC, UC, USC.

All other available modifications for the 33X Series can be ordered for this brake.

Options

- See AC rectifiers section for specifications (full and half wave)
- Band seal (boot)
- Tach/encoder mounting
- Manual release non-maintained or maintained
- Shaft seal
- Mounting flange
- Electronic brake release indicator switch

33X Series with NEMA C-Face Adapter & Enclosed Housing

Series 33X with a C-face adapter and a brake housing. Order as an IP 43 rating with or without external manual release:

33B Series for brake without torque adjust 33C Series for brake with torque adjust

OR IP 54 rating with the option of internal maintained manual release:

33H Series for brake without torque adjust 33J Series for brake with torque adjust

Also Available.....

330 Series

Magnet body is not machined for a manual release option. See ordering information for the 33X Series brakes.

33X Severe Duty

Any of the 33X Series can be ordered as severe duty, appropriate for high-cycle rate applications. See ordering information for the 33X series brakes.

332 Series

Basic brake with the torque adjust option, and the magnet body not machined for the manual release option. See ordering information for the 33X series brakes.

*Kebco is a Registered Trademark of Kebco, Inc. or its affiliates. Lenze is a Registered Trademark of Lenze Power Transmission or its affiliates. Binder is a Registered Trademark of Kendrion or its affiliates.

Series 331 (Without Torque Adjust) & Series 333 (Torque Adjustable) Mounting Face: IEC/Metric 72, 90, 112, 132, 145, 170, 196 & 278

Dimensions: Sizes 72 thru 170

* Without keyway pilot bore. **170 33X-64010** 2.559

¹⁴⁵ 33X-54010 2.165

45

55

65

¹³² 33X-44010 1.772

** Keyway to DIN 6885/3 - standard metric keyway DIN 6885/1.

62 6.6

2.913 *74*

3.307 *84*

2.441 0.261 0.261

0.354 *9*

6.6

9

0.354 *9*

0.354 0.354 *9*

5.709 *145*

6.299 *160*

7.283 *185*

1.200 *31*

1.575 *40*

1.338 *34*

0.924 *23.47*

1.200 *30.48*

1.042 *26.47*

0.866 *22*

1.142 *29*

0.984 *25*

3.35 *85.1*

3.95 *100.3*

4.69 *119.1*

5.47 *138.9*

6.90 *175.3*

7.73 *196.3*

.950 *24.13*

1.25 *31.9*

1.25 *31.9*

17.5 *444.5*

17.5 *444.5*

23.6 *600*

1 Size 132: "h2" and "h3" dimensions, contact factory. May vary .09 inch. For verification of manual release dimensions, contact factory.

Engineering Specifications: Sizes 72 thru 170

Series 331 (Without Torque Adjust) & Series 333 (Torque Adjustable) Continued IEC/Metric Frame

Dimensions: Sizes 196 thru 278

* Without keyway pilot bore.

** Keyway to DIN 6885/3 - standard metric keyway DIN 6885/1.

Engineering Specifications: Sizes 196 thru 278

Series 331 & Series 333 Mounting Face: NEMA 48 & 56C, NEMA 145, 182 & 256TC, NEMA 324, 326, 365, 404 & 405TC, TSC, UC & USC

Dimensions

*On these sizes, the brake diameter is larger than the adapter mounting/bolt circle. ***On each side of the 12:00 position (the location of second hole is not shown on

**L min. & max: size 196 = .187 min. & .479 max. size 230/278 = .340 min. & .730 max.

above drawing).

 \triangle Frame shaft size may require derate of a larger brake. Confirm hub bore.

Engineering Specifications

Series 331 & Series 333 Mounting Face: NEMA 182 & 256TC with Brake Housing

Outline Drawing for IP 43

IP 43 Dimensions

For sizes 196 through 278 with C-face and housing, see Series 350 or Series 360.

Outline Drawing for IP 54 (No manual release/optional internal manual release)

IP 54 Dimensions

*For NEMA 48C and 56C mounting, see AAB 321 and 322 NEMA C-face section.

Series 331 & Series 333 Continued IEC/Metric & NEMA Frames

Ordering Information

Group "3" Armature Acting Brake $-333-44110 -0$ $-$ K_C

Table 1 - Hub Bores 333-14010-XX-EA

196 230 278

7 8 9

NOTE: See dimensions for C-face brake motor systems section for recommended minimum bore size by torque.

6 | Std torque with air gap shim

5 Size 90 Only; 8.34 lb-ft holding applications only

Table 2 - STD Coil Voltage 333-14010-14-XA

4 IP54 rating

*In-line rectifier. Consult factory for pricing.

**In-line quickset rectifier. Consult factory for pricing.

- Contact factory for non-standard coils.
- Add the following for non-std. coil voltage.

• Sizes 72 through 112 = Consult factory for pricing.

• Sizes 132 through 170 = Consult factory for pricing.

- Sizes 196 through 278 = Consult factory for pricing.
- For separate AC rectifiers, see AC rectifier section.

Table 3 - Options 333-14010-14-EX

Modifications are available - see AAB Modification section.

*Standard U.S. Keyseats made to ANSI B17.1 standard. Standard metric Keyseat DIN 6885/1.

Series 350 Pressure Plate Mounted Mounting Face: NEMA 182, 284 & 324TC, 256, 286 & 405TSC

Brake with IP 56 Aluminum Cover | Brake showing space heater

- Torque: 75 300 lb-ft (102 to 400 Nm)
- Installation Instructions & Parts List: P/N 8-078-895-00
- Universal mounting
- Class 180(H) insulation
- Maintained manual release
- Corrosion resistance (stainless steel external hardware)
- IP 56 protection (available in ductile cast iron or aluminum cover)

and release indicator location

- ABS and CE certification
- Simple wear adjustment with access hole for air gap inspection
- Metric and US Customary bore sizes
- C-Face mounting various adapter plates available for 182TC through 405TSC frame mounting
- Splined hub for quiet dependable operation

Features Options

- See AC rectifier section
- Tach/encoder mounting
- Space heater 115, 230 or 460 Vac
- Thru-shaft
- Conduit box- specify F1 or F2 location (F1 location shown)

Dimensions

*Key to DIN 6885/3p9-standard metric keyway DIN 6885/1.

D1** & A** for aluminum cover. D1*** & A*** for ductile iron cover.

Component Materials:

- Adapter plate steel (zinc plate)
- Splined hub steel (zinc plate)
- Splined carrier aluminum
- Armature steel (zinc plate)
- Magnet body steel (zinc plate)
- Hardware steel (corrosion resistant plating or stainless)
- Cover: Size 196 182T thru 286TS NEMA - Aluminum (anodized) (additional paint optional) Size 196 - 324T thru 405TS NEMA - Cast Iron (primed) (additional paint optional)
- Size 230 284T thru 405TS NEMA Cast Iron (primed) (additional paint optional)
	- Size 278 324T thru 405TS NEMA Cast Iron (primed) (additional paint optional)

Series 350 Pressure Plate Mounted Continued

NEMA Frame

Specifications

(Direct acting with a DC Coil) **Mounting Design**

Pressure Plate Mount

Numeral Design

 Ordering Information *Part number example:* 355- 7 4 4JCOMEA Group "3" Armature Acting Brake

Options **Table 3**

Voltages - **Table 2**

Hub bore and keyset - **Table 1**

*Contact factory for pricing on these mounting options.

Table 3 - Additional Options

NOTE: Final part number may change due to specifications or options selected or other product design considerations. A number such as a 2, 3, 4 etc., in the 12th position is used to designate a unique brake (custom) and can only be assigned by Stearns Design Engineering Department.

Modifications are available - see AAB Modification Section.

300

Table 1 - Hub Bores

9

Numeral/

NOTE: See Dimensions for C-Face AC Brake Motor Systems for recommended minimum bore sizes by torque.

278

Numeral IP Rating 4 IP56 (standard) H IP66 conduit box* with terminal strip M **IP66** conduit box*

*Specify F1 or F2 location for conduit box modification in quote request.

Table 2 - Coil Voltage

For AC rectifiers, see AC rectifier section.

*Standard U.S. keyseats made to ANSI B17.1 standard. Standard metric keyseat DIN 6885/1. **Keyseat to DIN 6885/3.

Series 360 Magnet Body Mounted

Mounting Face: IEC 170, 196, 230 & 278

Features

- IP 56 rating
- Installation Instructions & Parts List: P/N 8-078-898-00
- Universal mounting
- Internal maintained manual release
- ABS and CE Certification
- Brake gaskets are captive (o-ring), so parts are not lost during maintenance
- Stainless steel nameplate (exterior)
- Modular brake assembly. Install and remove brake without having to readjust air gaps.

HÉAD C
USE MI

P SCREWS
HEX ALLEN **WRENCH**

size 170 & 196: (3) MB SOCKET HEAD CAP SCREWS
USE M6 HEX ALLEN WRENCH

• Class 180(H) insulation

Options

- See AC rectifier section
- Tach/encoder mounting
- Space heater • Electronic brake release indicator
- Electronic wear indicator
-
- Thru-shaft
- External non-maintained/maintained manual release
- IP 66 conduit box mounted on adapter plate. Wiring is not disturbed when brake housing is removed.

Dimensions

O RING

THRU HOLES

 (4) HUB

Note: Dimensions for estimating purposes only.

Component Materials for 361-X Series:

• Adapter plate - steel (zinc phosphate, prime & paint)

• Splined hub - steel (zinc plate)

- Armature steel (normalized)
- Pressure Plate steel (normalized)
- Magnet body steel (zinc plated)
- Housing ductile iron (primed & painted):

(3) M26x1.5 PLUGS & SEALS
ON O.D. FOR AIR GAP INSPEC

• Hardware - steel (corrosion resistant plated or stainless)

PRODUCT
VAMEPLATE

LIFTING EYERN 1

Series 360 Magnet Body Mounted Continued IEC Frame

Specifications

Ordering Information

Table 1 - Hub Bores

NOTE: See dimensions for C-face AC brake motor systems for recommended minimum bore sizes by torque.

Table 2 - Coil Voltage

Other voltages available - consult factory. For AC rectifiers, see AC rectifier section.

*Contact factory for pricing on these mounting options.

Table 3 - Additional Options

NOTE: Final part number may change due to specifications or options selected or other product design considerations. A number such as a 2, 3, 4 etc., in the 12th position is used to designate a unique brake (custom) and can only be assigned by Stearns Design Engineering Department.

Modifications are available - see AAB Modification Section.

*Standard U.S. keyseats made to ANSI B17.1 standard. Metric keyseats to DIN 6885/1.

AC Rectifiers for Use with Armature Actuated Brakes

For brake response times with and without AC rectifiers, see technical data section.

Product Overview

Provides coil turn off nearly as fast as DC side switching. Includes line filter for AC drive applications or whenever electrical filtering is required to protect the rectifier from high-frequency electrical line pulses. Must be switched on/off by a switch in an AC lead of the TOR-AC. Maximum operating voltage +10% of nominal, frequency 50/60 Hz. Maximum ambient temperature range is -40**°**C to 65**°**C.

A rectifier that provides a quick brake response time even when the rectifier is permanently wired across the windings of an AC motor. The QuickSet Rectifier detects the decaying, motor generated voltage that occurs when power is removed from the motor circuit, and interrupts brake coil current in response. QuickSet Rectifiers can be specified full wave or half wave. Operating voltage is ±10% of nominal, frequency 50/60 Hz. Maximum ambient temperature range is -40**°**C to 65**°**C. **QuickSet**

QuickSet/QuickRelease

A rectifier that provides a timed, full wave rectified "over-excitation" brake release function, followed by continuous, half wave rectified brake released "holding" function, when used in conjunction with an appropriate brake coil voltage rating.

USED AS WATTSAVER: Provides a timed, full wave rectified brake release function, followed by continuous, half wave rectified brake released "wattsaver" function, when used in conjunction with an appropriate brake coil voltage rating. The Wattsaver serves to reduce the electrical power consumption and dissipation of the brake in the released state. Operating

voltage is ±10% of nominal, frequency 50/60 Hz. Maximum ambient temperature varies by part number - see information by part number on following pages. **Input**

Output

Bi-Phase Rectifiers

A rectifier that is typically used in single phase, reversing, permanent

split capacitor (PSC) motor applications. A single phase, reversing, PSC motor typically has two windings of equivalent resistance. The winding which serves as the main winding is connected directly across the power line, the winding which serves as the auxiliary winding is connected in series with a run capacitor across the power line. The direction of rotation is reversed by interchanging the function of the two windings. The Bi-Phase Rectifier provides the same voltage to the brake coil regardless of the direction of rotation of the motor. The Bi-Phase Rectifier has five leads and comes in standard response and QuickSet versions. Bi-Phase Rectifiers are application specific. Please contact factory for more information.

AC Rectifiers Selection

AC Rectifiers Selection

L.

AC Rectifiers Continued

Rectifier Dimensions

Terminal location or connection may differ from sketch Flange or Tape Mount

Wiring Diagrams/Switching

Use DC side switching with the following Rectifiers ONLY: 4-1-20291-01K, 4-1-20292-01K, 4-1-20292-03K, 4-1-20493-01K, 4-1-20491-01K

Armature Actuated Brake Modifications

Series 320, 321 & 322

Armature Actuated Brake Modifications

Series 333, 350 & 360

Armature Actuated Brake Modifications

Technical Data SAB Motor Frame Adapter Dimensions Selection

To select an adapter for a specific brake, refer to the motor frame adapter tables as shown in the brake series sections of this catalog. After selecting the adapter stock number, refer to the Tables below for dimensions.

All adapters are constructed with an opening for internal lead wire connection, corresponding to the NEMA standard location for the motor frame size.

Screws for mounting adapter to motor must be provided by customer. Socket head cap screws are supplied for mounting brake to adapter.

Dimensions for estimating only. For installation purposes, request certified prints.

* 1/2-13 flat head screws are supplied with adapter.

** When adding an adapter to a hazardous location brake, refer to the "mounting requirements" on the product page for the recommended brake series for accommodating adapters.

Kits include the foot mounting bracket and hardware to fit the BF mounting holes.

Brake Series	Torque	Foot Mounting Kit Number		Dimensions in Inches (Dimensions in Millimeters)															Wgt		
			Α	AJ	AK	B	BB	No.	BF Thd.	C	D	E	FA	FB	G	H		K	L	M No.	lbs.
56,000	$1.5 - 25$	5-55-5023-00	7.00 (177.80)	5.88 (149.22)	4.499 4.498 114.275 $\frac{1}{114.249}$	2.38 (60.32)	.12 (3.18)	$\overline{2}$	$3/8 - 16$	6.50 (165.10)	3.50 (88.90)	2.88 (73.02)	1.50 (38.10)	۰	.38 (9.52)	.41 (10.32)	1.50 (38.10)	.50 (12.70)	2.50 (63.50)	$\overline{2}$	4.5
87,000	$6 - 125$	5-55-7021-00	8.62 (219.08)	7.25 (184.15)	8.499 8.498 215.875 215.849	3.00 (76.20)	.25 (6.35)	$\overline{4}$	$1/2 - 13$	8.62 (218.95)	5.00 (127.00)	3.56 (90.49)	2.00 (50.80)	$\overline{}$.38 (9.52)	.53 (13.49)	1.62 (41.28)	.56 (14.29)	5.75 (146.05)	$\overline{2}$	$\overline{7}$
81,000	125-230	5-55-2022-00	15.50 (393.70)	11.00 (279.40)	12.499 12.498 (317.475) 317.449	7.00 (177.80)	.25 (6.35)	$\overline{4}$	$5/8 - 11$	13.25 (336.55)	8.50 (215.90)	6.88 (174.62)	2.00 (50.80)	4.00 (101.60)	.62 (15.88)	.69 (17.46)	3.00 (76.20)	.88 (22.22)	9.00 (228.60)	Δ	40
82,000	125-550																				
86,000	$500 -$ 1000	5-55-6021-00	18.25 (463.55)	14.00 (355.60)	16.000 15.995 406.400 406.273	8.00 (203.20)	.22 (5.56)	4	$5/8 - 11$	17.00 (431.80)	10.88 (276.22)	6.38 (161.92)	3.38 (85.72)	3.00 (76.20)	1.00 (25.40)	.81 (20.64)	4.12 (104.78)	1.22 (30.96)	8.50 (215.90)	4	75

Dimensions for estimating only. For installation purposes, request certified prints.

Dimensions for C-Face Brake Motor Systems

Brakes Externally Wired to Motor

C-face motor with double shaft extension.

Stearns disc brakes are designed to mount on standard C-face motors having the same dimensions and tolerances on the accessory end as on the drive end. They also mount on foot mounting brackets and machine mounting faces having the same mounting dimensions and tolerances. Some motor accessory end

Drive End Dimensions (Inches)

Tolerances (Inches)

AK Dimension, Face Runout, Permissible Eccentricity of Mounting Rabbet

Width of Shaft Extension Keyseats

SOURCE: ANSI/NEMA Standards Publication No. MG 1-1987; Part 4 and Part 11.

Shaft Extension Diameters

Shaft Runout

Dimensions for C-Face AC Brake Motor Systems Continued

Accessory End

143TFC to 184TFC Frames, Inclusive 213TFC to 326TFC Frames, Inclusive

Dimensions (Inches)

NOTE: Standards have not been developed for the shaft extenison diameter and length, and keyseat dimensions.

Tolerances* (Inches)

FAK Dimension, Face Runout, Permissible Eccentricity of Mounting Rabbet

Part 4 and Part 11. * Tolerance requirement on 56,X00 and 87,000 Series brake kits is .015

T.I.R. (total indicated runout shaft to motor register face).

Stearns Recommended Minimum Shaft Diameter by Torque

Minimum recommended shaft size considers a keyed C1045 steel shaft under *dynamic* use in a typical spring set brake application.

Shaft Runout

0.002 0.003 0.3750 to 1.625, inclusive Over 1.625 to 6.500, inclusive **Shaft Runout Shaft Diameter**

Maximum Permissible

SOURCE: ANSI/NEMA Standards Publication No. MG 1-1987;

Set & Release Times

The models listed below were tested for typical set and release times. Times listed below are defined as follows:

T1 = Total set time to 80% of rated static torque T2 = Release time, measured as the time from when the power is applied to the brake to the time that the solenoid plunger or armature is fully seated.

NOTE: Times will vary with the motor used, and brakes tested with factory-set air gap. The times shown should be used as a guide only.

AAB Series 310/311/320/321 Times in Milliseconds

SAB T1/T2 Time in Milliseconds

Brake and motor are switched separately. All brakes tested in horizontal position. Coil is energized for >24 hours before testing. Ambient temperature 70°F at time of test.

AAB Series 333 Times in Milliseconds

Conversions

English-Metric Conversion Factors

Multiply the base unit by the factor shown to obtain the desired conversion.

English-English Conversion Factors for Thermal Capacity

Decimal Equivalents of Fractions

Application Engineering

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.

Since Stearns SABs 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:

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:

$$
T_{\rm S} = \frac{T_{\rm d}}{0.8}
$$

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:

$$
T_{\rm S} = \frac{5,252 \times P}{N} \times SF
$$

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 (SF)

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 \times 5252) / rpm with nonoverhauling 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 (2.0 for holding duty only) 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
\nMotor speed (N) - 1,750 rpm
\nService factor (SF) - 1.4
\n
$$
T = \frac{5,252 \times P}{N} \times SF
$$
\n
$$
= \frac{5,252 \times 5}{1,750} \times 1.4
$$
\n
$$
T = 21 lb-fit
$$

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.

$$
T_s = F \times R \times SF
$$

$$
= 5 \times 2 \times 1.4
$$

$$
T_s = 14 \text{ lb-fit}
$$

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-ft²

Wk² = Inertia of brake, lb-ft²

- Wk²_M = Inertia of motor rotor, lb-ft²
- Wk_i^2 = Equivalent inertia of load reflected to brake shaft. lb-ft²

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:

Equivalent Wk² = Wk² $\left(\frac{N_L}{N_S}\right)^2$ Where.

Equivalent Wk $_{B}^{2}$ = Inertia of rotating load reflected to brake shaft, lb-ft² $Wk_i²$ = Inertia of rotating load, lb-ft² N_i = Shaft speed at load, rpm $N_B = Shaff speed$ at brake, rpm

Horizontal Linear Motion

Equivalent Wk_w⁼ W $\left(\frac{V}{2\pi N}\right)^{2}$

Where.

Equivalent Wk $_{w}^{2}$ =Equivalent inertia of linear moving load reflected to brake shaft, lb-ft² W = Weight of linear moving load, lb $V = Linear velocity$ of load, ft/min

 $N_B = Shaff speed$ at brake, rpm

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

$$
T_{d} = \frac{Wk_{\rm T}^2 \times N_{\rm B}}{308 \times t}
$$

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

> $Wk^2 = \text{Total inertia reflected}$ to brake, $Ib-ft^2$

 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-fit

$$
T_d = \text{System dynamic torque, lb-fit}
$$

Examples 3, 4, 5 and 6 illustrate how brake torque is determined for nonoverhauling 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 $_{h}^{2}$) - 0.075 lb-ft² Flywheel inertia (Wk_F^2) - 4 lb-ft² Brake inertia (Wk3) - 0.042 lb-ft² Required stopping time (t) - 1 sec

First determine the total inertia to be stopped,

 $Wk_f^2 = Wk_M^2 + Wk_{FW}^2 + Wk_B^2$ $= 0.075 + 4 + 0.042$ Wk_{τ}^2 = 4.117 lb-ft²

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

$$
T_{d} = \frac{Wk_{T}^{2} \times N_{BM}}{308 \times t}
$$

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

$$
T_{d} = 23.4 \text{ lb-fit}
$$

Converting T_d to static torque

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

= $\frac{23.4}{0.8}$

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

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,

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

=
$$
\frac{Wk_f^2 \times N_{BM}}{308 \times (0.8 T_s)}
$$

=
$$
\frac{4.117 \times 1,750}{308 \times (0.8 \times 35)}
$$

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 (Wk2) - 0.075 lb-ft<sup>2</sup>
Gear reduction (GR) - 20:1
Gear reducer inertia at high
speed shaft (Wk_{GR}^2) - 0.025 lb-ft<sup>2</sup>
Flywheel inertia (Wk_{FW}^2) - 20 lb-ft<sup>2</sup>
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 rpm

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

$$
Wk_6^2 = Wk_{FW}^2 \left(\frac{N_{FW}}{N_M}\right)^2
$$

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

 $Wk_6^2 = 0.05$ lb-ft²

Determining the total Wk²,

$$
Wk_{\uparrow}^{2} = Wk_{M}^{2} + Wk_{GR}^{2} + Wk_{B}^{2}
$$

= 0.075 + 0.025 + 0.05

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

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

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

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

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

Converting dynamic torque to static torque,

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

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

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

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,

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

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

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

t = 0.18 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_0) - 32 rpm

Weight of load (W) - 30 lb

Convevor pulley and belt inertia $(Wk_0^2) - 4.0$ lb-ft²

Conveyor pulley diameter (d_n) - 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 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.

 $Wk_T^2 = Wk_W^2 + Wk_F^2$ $= 7.5 + 4.0$ $Wk_1^2 = 11.5$ lb-ft²

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

$$
T_{d} = \frac{Wk_{1}^{2} \times N_{p}}{308 \times t}
$$

$$
T_{d} = \frac{11.5 \times 32}{308 \times 0.25}
$$

$$
T_{d} = 4.8 \text{ lb-fit}
$$

Converting dynamic torque to static torque,

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

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

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

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

 $W_T = 2,100$ lb

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

$$
N_{\rm B} = \frac{V}{2\pi r}
$$

= $\frac{180}{2 \times \pi \times 0.75}$
N_B = 38.2 rom

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

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

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

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

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

$$
T_{d} = \frac{Wk_{1}^{2} \times N_{B}}{308 \times t}
$$

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

$$
T_{d} = 73 \text{ lb-fit}
$$

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,

$$
T_d = \frac{W_T^{\vee}}{gt} \times I
$$

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
- q = Gravitational acceleration constant, 32.2 ft/sec²
- $t =$ Desired stopping time, sec
- $r =$ Length of the moment arm (wheel radius), ft

or, for this example,

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

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

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,

$$
T = \frac{W_T^V}{gT_d} \times r
$$

=
$$
\frac{W_T^V}{g \times (0.8 \times T_s)} \times r
$$

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

t = 1.8 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-fit

Wk $\frac{2}{3}$ = Total inertia reflected to brake. Ib-ft²

- N_B = Shaft speed at brake, rpm. Consider motor slip when descendina.
	- $t =$ Desired stopping time, **SAC**

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

$$
T_{\circ} = W \times R \times \frac{N_{L}}{N_{B}}
$$

- Where, T_0 = 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_0 = Overhauling dynamic torque of load reflected to brake shaft. Ib-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-fit

$$
T_t = \text{System dynamic torque, lb-fit}
$$

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 (Wk2) - 5 lb-ft² Required stopping time (t) -1 sec

First, determine brakemotor shaft speed (N_B) .

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

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

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

Then, determine the equivalent inertia of the overhauling load.

$$
Wk_1^2 = W\left(\frac{V}{2\pi N_B}\right)^2
$$

= 100\left(\frac{667}{2\pi \times 849}\right)^2
Wk_1^2 = 1.56 lb-ft²

Therefore, the total inertia at the brake is,

$$
Wk_1^2 = Wk_0^2 + Wk_1^2
$$

= 5 + 1.56
 $Wk_1^2 = 6.56$ lb-fit²

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

$$
T_{d} = Wk_{1}^{2} \times N_{B}
$$

= $\frac{6.56 \times 850}{308 \times 1}$
 $T_{d} = 18.1 \text{ lb-fit}$

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

$$
T_0 = W \times R
$$

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

$$
T_0 = 12.5 \text{ lb-fit}
$$

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

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

Dynamic torque is then converted to static torque.

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

= $\frac{30.6}{0.8}$
T_s = 38.3 lb-fit

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,

$$
T_{s} = \frac{T_{t}}{0.8}
$$

$$
= \frac{T_{d} + T_{0}}{0.8}
$$

or,

$$
T_{d} = 0.8T_{s} - T_{0}
$$

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

$$
T_{d} = 27.5 \text{ lb-ft}
$$

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 (WK2) - 0.65 lb-ft² Gear reduction (GR) - 300:1 Drum diameter (D) - 1.58 ft Weight of load (W) - 4,940 lb Drum inertia (WK²) - 600 lb-ft² Required stopping time (t) - 0.5 sec

First, calculate all inertial loads reflected to the brake motor shaft.

The rotational speed of the drum is,

$$
ND = \frac{NM}{GR}
$$

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

$$
ND = 3.83 rpm
$$

 \mathbb{R}^2

From this, the cable speed can be determined.

$$
V = ND x πD= 3.83 x π x 1.58V = 19.0 ft/min
$$

The equivalent inertia of the load reflected to the brake motor shaft is,

$$
Wk_{\tilde{1}}^{2} = W \left(\frac{V}{2\pi N_{\text{BM}}}\right)^{2}
$$

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

$$
Wk_{\tilde{1}}^{2} = 0.034 \text{ lb-ft}^{2}
$$

The equivalent inertia of the drum at the brake motor shaft speed is,

$$
Wk_{d}^{2} = Wk_{b}^{2} \left(\frac{N_{b}}{N_{BM}}\right)^{2}
$$

$$
= 600 \left(\frac{3.83}{1.150}\right)^{2}
$$

Finally, the total inertia the brake will retard is,

 $Wk_1^2 = Wk_1^2 + Wk_1^2 + Wk_2^2$ $Wk_{\text{T}}^{\text{d}} = 0.0067 \text{ lb-ft}^2$ Wk $_{7}^{2}$ = 0.691 lb-ft²

The dynamic torque required to decelerate the total inertia is,

$$
T_{d} = \frac{Wk_{1}^{2} \times N_{BM}}{308 \times t}
$$

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

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

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

$$
T_0 = W \times R = W \times \frac{1}{2}D
$$

= 4,940 x $\frac{1.58}{2}$

$$
T_0 = 3,903
$$
 lb-fit

Which reflected to the brake motor shaft becomes,

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

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

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

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

$$
T_t = 18.16
$$
 lb-fit

Dynamic torque is then converted to static torque.

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

= $\frac{18.16}{0.8}$
 $T_s = 22.7$ lb-fit

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 $(P) - 7\frac{1}{2}$ hp Speed (N_M) - 1,165 rpm Rotor inertia (WK2) - 1.4 lb-ft²

Gear reducer data:

Reduction (G_R) - 110:1 Inertia at input shaft $(Wk_R²) - 0.2$ ib-ft²

Drum data Diameter (D_n) - 1.5 ft Inertia (Wk $_{0}^{2}$) - 75 lb-ft²

Pulley data Diameter $(D_P) - 1.5$ ft Inertia (Wk $_{p}^{2}$) - 20 lb-ft² Bucket weight (W_B) - 700 lb Maximum weight of load $(W_L) - 4,000$ 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 brake motor shaft. The rotational speed of the drum is:

$$
N_{D} = \frac{N_{M}}{GR}
$$

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

$$
N_{D} = 10.6
$$
 rpm

From this the cable speed can be determined:

$$
V = ND \times \pi DD
$$

$$
= 10.6 \times \pi \times 1.5
$$

$$
V = 50
$$
 ft/min

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

$$
Wk_1^2 = W\left(\frac{V}{2\pi N_M}\right)^2
$$

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

$$
Wk_1^2 = 0.219 \text{ lb-ft}^2
$$

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 brake motor shaft is,

Wkβ = Wkβ(
$$
\frac{N_D}{N_M}
$$
)² = Wkβ($\frac{1}{GR}$)
= 20 x ($\frac{1}{110}$)²
Wkβ = 0.0017 lb-ft²

The inertia of the drum reflected to the brake motor shaft is,

$$
Wk_{d}^{2} = Wk_{0}^{2} \frac{(N_{D})^{2}}{(N_{M})^{2}} = Wk_{D}^{2} \frac{1}{GR} \frac{1}{(R)^{2}}
$$

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

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

The total inertia to be stopped is, $Wk_1^2 = Wk_1^2 + Wk_0^2 + Wk_4^2 + Wk_6^2 + Wk_9^2$

 $= 0.219 + 0.0017 + 0.0062 + 0.2 + 1.4$

 $Wk_f^2 = 1.827$ lb-ft

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

$$
T_{d} = \frac{Wk_{T}^{2} \times N_{M}}{308 \times T_{d}}
$$

$$
T_{d} = \frac{1.827 \times 1,165}{308 \times 1}
$$

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_TsinB is the force necessary to retard downward motion of the loaded hoist bucket.

$$
T_0 = W_T \sin B \times R_0
$$

= W_T \sin B \times \frac{1}{2} D_0
= 4,700 \times \sin 52.7^\circ \times \frac{1}{2}(1.5)
= 4,700 \times 0.7955 \times 0.75
T_0 = 2,804 lb-fit

Which reflected to the brake motor shaft becomes,

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

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

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

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

$$
T_t = T_d + T_m
$$

$$
= 6.9 + 25.5
$$

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

Converting to static torque,

 $T_s = \frac{T_t}{0.8}$ $=\frac{32.4}{0.8}$ $T_s = 40.5$ 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 = $\frac{T_{t}}{0.8}$
= $\frac{T_{d} + T_{m}}{0.8}$
or, T_d = 0.8T_s - T_m
= (0.8)(50) - 25.5
T_d = 14.5 lb-ft
therefore,
 $t = \frac{1.827 \times 1,165}{308 \times 14.5}$
t = 0.48 sec

See section on stopping time.

Stopping Time & 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:

$$
T_{d} = \frac{Wk_{f}^{2} \times N_{B}}{308 \times t}
$$

or,
$$
t = \frac{Wk_{f}^{2} \times N_{B}}{308 \times (0.8 \times T_{s})}
$$

Where, $t =$ Stopping time, sec

 Wk_1^2 = Total inertia reflected to brake, lb-ft²

 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 x 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:

Revolutions to stop = $\frac{\text{t} \times \text{N}_\text{B}}{120}$ 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/sec²

 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.

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 *(-7°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.

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:

 $KE_{I} = \frac{W_{V}^{2}}{2g}$

- KE_I = 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/sec²

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

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

- Where, KE_r = Kinetic energy of linear load, lb-ft
	- Wk_f^2 = Inertia of the rotating load reflected to brake shaft, lb-ft²
	- 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:

$$
PE = W_s
$$

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 stopping an overhauling load is:

$$
E_T = KE_T + KE_r + PE
$$

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:

$$
KE_{I} = \frac{W_v^2}{2g}
$$

= $\frac{4,940 \times 0.317^2}{2 \times 32.2}$
KE_I = 7.71 ft-lb

Next, calculate the kinetic energy for the rotational load. The motor inertia is 0.65 lb-ft2 and the drum inertia reflected to the brake shaft speed is 0.0067 lb-ft2. The total rotational inertia at the brake motor shaft is,

$$
Wk_f^2 = Wk_M^2 + Wk_d^2
$$

= 0.65 + 0.0067

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

And the kinetic energy of the rotating components is,

$$
KE_r = \frac{Wk_f^2 \times N_0^2}{5,875}
$$

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

KE_T = 147.8 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 dynamic torque.

$$
T_t = T_s \times 0.8
$$

$$
= 25 \times 0.8
$$

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

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,

$$
T_d = T_t - T_m
$$

$$
= 20 - 13
$$

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

The stopping time resulting from this dynamic torque is,

$$
t = \frac{Wk_{\text{TX}}^2 N_{\text{M}}}{308 \times T_{\text{d}}}
$$

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

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

Where,
$$
Wk_T^2 = 0.690
$$
 lb-fit² 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,

> $s = \frac{1}{2}$ vt $=$ $\frac{1}{2}$ x 0.317 x 0.369 $s = 0.059$ lb-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,

$$
PE = W_s
$$

= 4,940 x 0.059
 $PE = 291$ ft-lb

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

$$
E_T = KE_T + KE_r + PE
$$

= 7.71 + 147.8 + 291

$$
E_T = 447
$$
 lb-fit

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 (ftlb/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:

$$
TC = \frac{Wk_1^2 \times N_8^2 \times n}{3.2 \times 10^6}
$$

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

- Wk_f^2 = Total system inertia reflected to brake, lb-ft²
	- 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.

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

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

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

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-ft2. The shaft speed of the brake motor is 1,800 rpm. Therefore, the required thermal capacity is,

$$
TC = \frac{Wk_1^2 \times N_{01}^2 \times n}{3.2 \times 10^6}
$$

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

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

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,

 $n_{\text{max}} = \frac{TC_{\text{rated}} \times (3.2 \times 10^6)}{Wk_T^2 \times N_M^2}$ $=\frac{9 \times (3.2 \times 10^6)}{0.15 \times 1,800^2}$ n_{max} = 59 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

Please see Super-Mod® dimensional data section.

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 is 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. 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. 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 *(-7°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.

Formulas

The following formulas cover the basic calculations used in brake application engineering.

