

# Pneumatic and Hydraulic Clutches and Brakes



 **Industrial**<sup>®</sup>  
Clutch

*An Altra Industrial Motion Company*

## Warranty Policy

Industrial Clutch warrants all new products leaving the factory against defects in workmanship and material for a period of two (2) years after shipment with the exception of rotary unions, control valves and hydraulic power units which are warranted for a period of one (1) year after shipment. Adjustment under this warranty will be made after verification of defect of the part or product by a factory representative. This warranty does not cover replacement or refurbishment of normal wear items unless such items are deemed by a factory representative to have flawed material or workmanship.

Liability under the warranty shall extend only to the replacement or correction of any defective part or product as determined by Industrial Clutch, and extends only to the initial purchaser and is not transferable or assignable without the prior consent of Industrial Clutch.

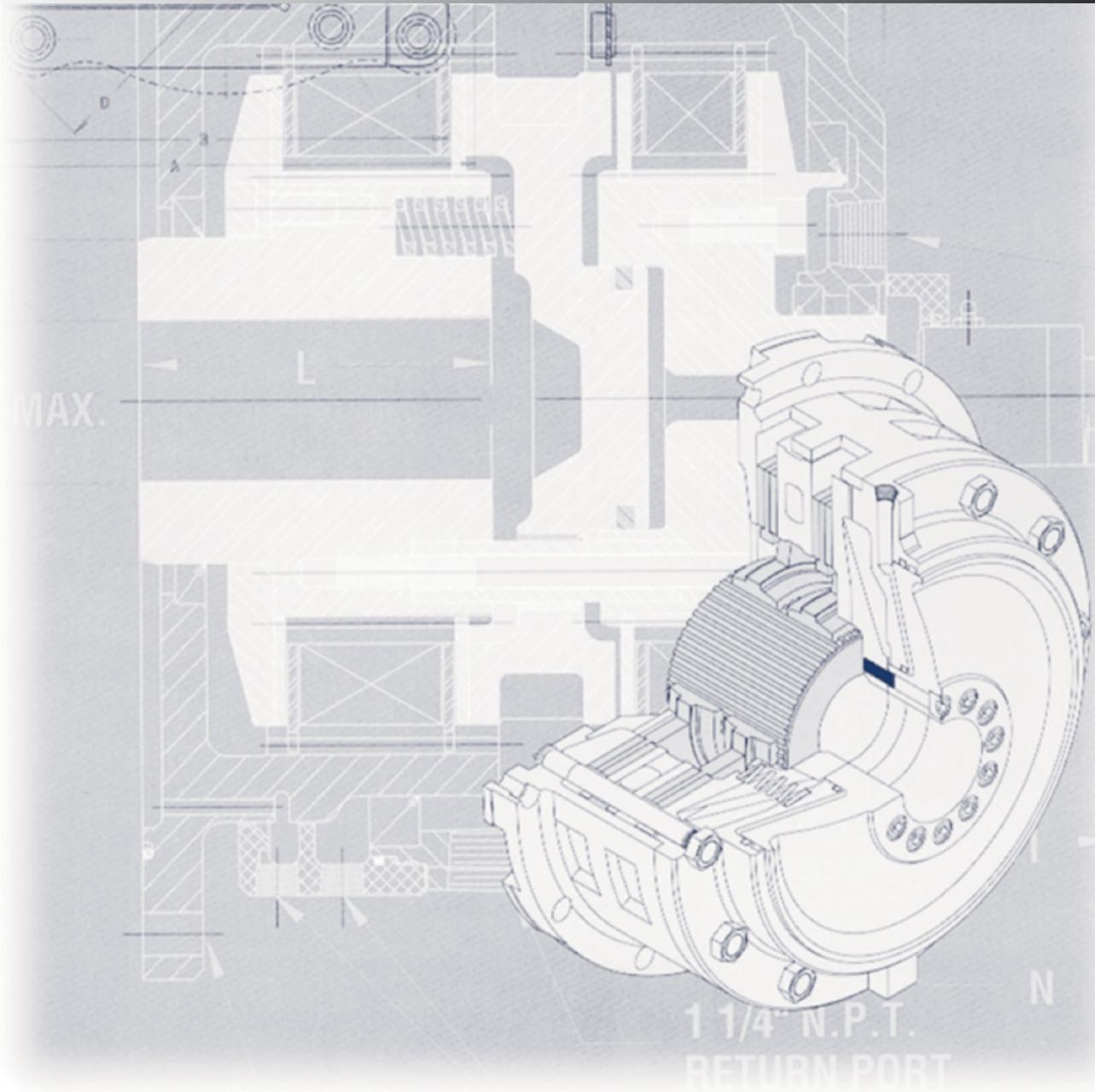
This warranty shall not apply to any product that has been repaired or altered without the manufacturer's knowledge and consent. This warranty also does not cover damage or defects resulting from neglect, improper installation, improper maintenance or operation, or accidents.

This warranty is made in lieu of all other warranties, expressed or implied, including but not limited to warranties of merchantability or fitness for particular purpose, and there are no other warranties that extend beyond this expressed warranty. Industrial Clutch's obligation under this warranty is limited to repair or replacement of the defective part or product and in no event shall Industrial Clutch be liable for consequential, indirect, or incidental damages of any kind incurred by reason of the manufacture, sale or use of any defective product.

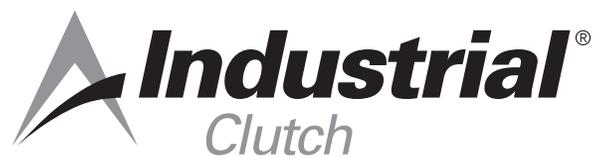
Industrial Clutch reserves the right to discontinue models or to change specifications at any time without notice. No discontinuance or change shall create any liability on the part of Industrial Clutch in respect to its products in the hands of customers or products on order not incorporating such changes even though delivered after any such change.

***Revised – September 2005***

Design Engineering



Principles, examples, and applications



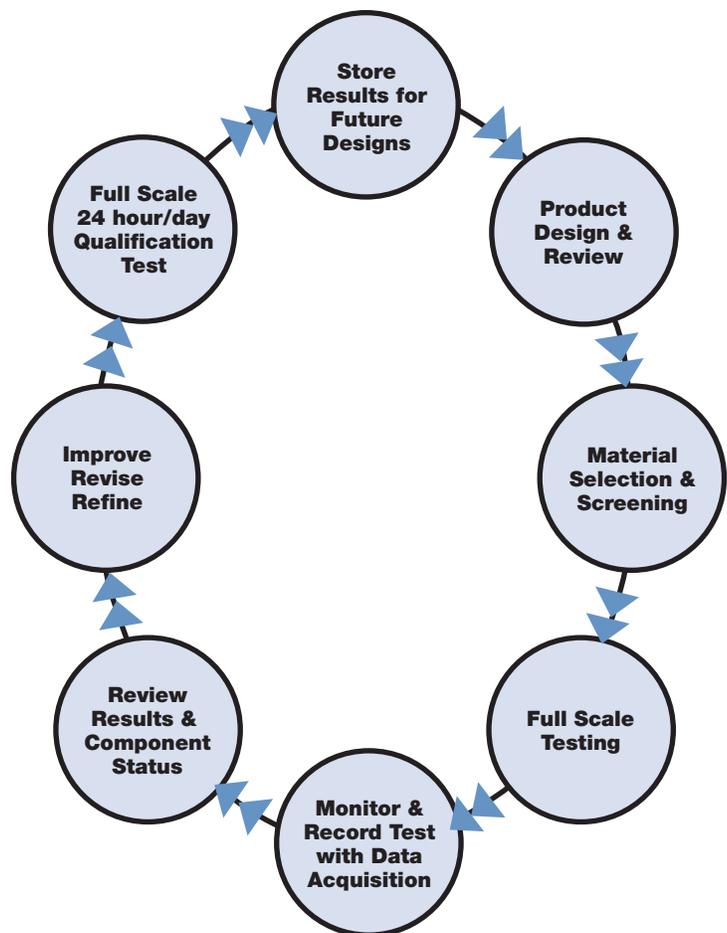
# The Company

Industrial Clutch Products designs and manufactures a complete line of dry and oil-immersed clutches and brakes, ranging in size from 5 to over 48 inches in diameter, with torque capacities exceeding eight million pound-inches. We also specialize in hydraulics and controls for use with our own clutches and brakes, or as stand-alone systems, utilizing the latest advancements in microprocessor-based controls.

We pride ourselves in our ability to respond to customers needs for standard, modified, and custom designs for unique or volume requirements. We are dedicated to ongoing research & development programs and full-scale testing, thus eliminating using the customer's installation as a test site. We are also a company who believes in the importance of strong supplier interaction. It is our suppliers who provide the quality components that allow our products to live up to the industry's most demanding applications.

Our end goal is to utilize our 75-plus years of experience and expertise to our customers' benefit in providing products that perform beyond expectations at competitive prices.

## Packaged Products Benefits



## Clutches



MODEL LK



MODEL AGT



MODEL LKT



MODEL HCD



MODEL LKM

## Brakes



MODEL LKT



MODEL LKB



MODEL LKBH



MODEL HBA/HBS



MODEL BD



MODEL LKV



MODEL LK



MODEL RB/RBS

## Clutch/Brakes



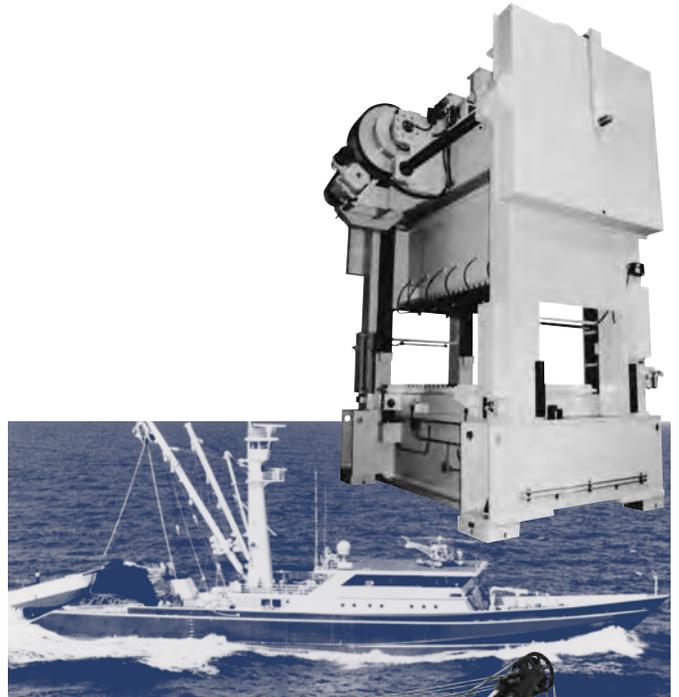
MODEL CBH



MODEL CBAG



MODEL CBA



# Clutch & Brake Design Principles

## I. CLUTCHES & BRAKES – SELECTION METHOD

Optimum sizing of heavy duty clutches and brakes requires a thorough understanding of the drive system parameters. The machine designer specifies the machine's function and performance levels. Industrial Clutch Products is aware of the capabilities of their products in many diverse applications based upon past experience and a sound testing and product development program. Pooling of this knowledge is an excellent way to ensure a successful installation will result.

Industrial Clutch Products encourages the use of **DATA APPLICATION** forms such as shown in Section XV to assist in providing the customer with the best possible selection. The **DATA APPLICATION** forms must be filled out completely to be effective. The information contained on these forms allow selections to be made quickly and accurately utilizing computer programs designed to define the torque and heat loads that will be present.

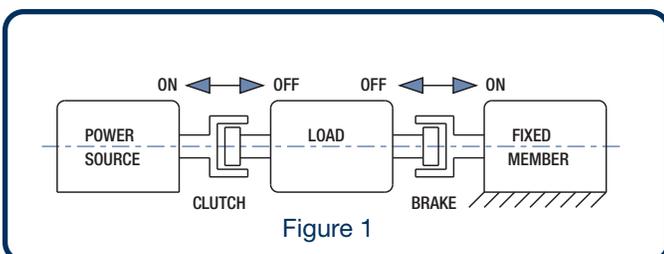
If the **DATA APPLICATION** forms listed in Section XV do not seem suitable for your application, contact our Engineering Department for additional assistance.

The machinery to which these products are applied, are complex, demanding, safety orientated, and can stand little, if any, unscheduled downtime. Industrial Clutch Products products will provide outstanding performance when selected by this time proven method.

## II. CLUTCHES & BRAKES – DESCRIPTION AND FUNCTION

A clutch is a device that is installed between a power source and a load so that the power source can be connected or disconnected from the load upon command.

A brake is a device that is installed between a fixed member and a load so that the load can be disconnected or connected to the fixed member upon command. Refer to the schematic shown in FIGURE 1.



A clutch or a brake performs two basic functions:

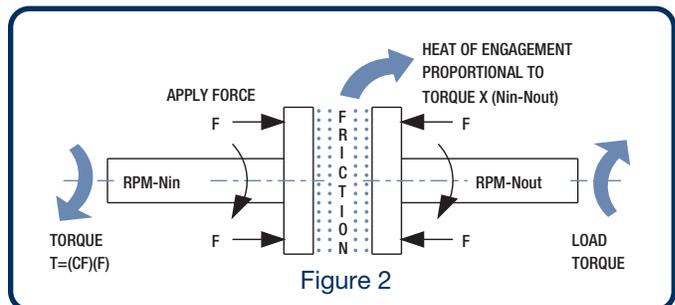
1. Transmits the normal and peak load torques with a suitable service factor.

2. Dissipates the heat generated during the engagement sequence.

Maximizing system reliability, minimizing engagement shock loads, and providing a clutch or brake unit that exhibits suitable durability must be the goals of every installation.

All clutches and brakes manufactured by Industrial Clutch Products transmit **TORQUE** by friction. Once the geometry of any unit is fixed, its torque transmitting characteristic is determined by the magnitude of the force applied to the friction surface and the coefficient of friction at this surface.

All clutches and brakes have a speed differential between their input and output members when they are disengaged. When engaged, there is no speed differential since they are "locked-up" and performing one of their primary functions of transmitting load torque. Therefore, to go from a disengaged condition to one of being engaged, a clutch or brake must go through a slipping phase until "lock-up" occurs. Note that the unit must transmit torque to or from the load during this slipping phase or no change in load motion will occur.

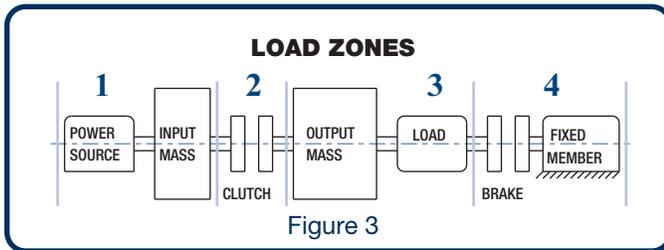


During engagement, the magnitude of the clutch or brake torque multiplied by the speed differential over the engagement time period is proportional to power. This manifests itself as **HEAT**. The clutch or brake unit must successfully dissipate this heat in accordance with the frequency of engagement the application requires. Refer to the schematic shown in FIGURE 2.

## III. CLUTCHES & BRAKES – SYSTEM MASS EFFECTS

Any drive system employing clutches and/or brakes can be simplified into a schematic such as shown in FIGURE 3.

A clutch or a brake simply transmits torque in accordance with its geometry, apply force, and coefficient of friction. It can be difficult to comprehend how important this is to the drive train stress levels and clutch or brake heat loads unless the system mass effects are considered.



Every system is composed of an input and output mass coupled together with a clutch, or grounded to some fixed member, with a brake. If the mass is in motion, it contains energy which will be changed during the clutch or brake engagement. In a like manner, a mass that is at rest requires energy to place it in motion. **Any mass whose motion is to be changed, will require a transfer of energy.** The clutch or brake is the member that controls the rate of energy transfer by varying the force on its frictional couple during engagement.

Large input masses or high input speeds, can yield huge sources of energy for the clutch to draw upon, particularly if the output mass is relatively small. In these cases, a system exists with a high input to output mass ratio. Conversely it is possible to have a system with a small mass ratio where the output mass is much greater than the input mass. Any number of combinations are possible and this mass ratio must be defined for the purpose of establishing both the system stress levels and the clutch or brake heat loads.

The potential exists for extremely large clutching and braking loads to be imposed upon the drive system due to these mass effects. These loads can be, and often are, several magnitudes greater than the work loads performed by the machine. It is imperative that the clutch-brake units and the drive train components be properly sized to withstand the loads generated during engagement.

The clutch-brake apply force must be appropriately controlled so as to not only limit the mechanical stresses to the system, but also the thermal stresses imposed upon the clutch or brake unit.

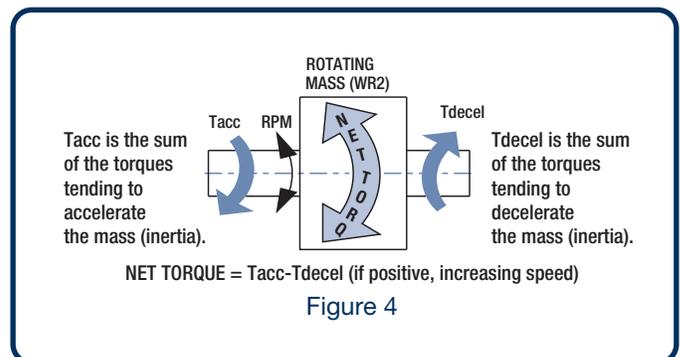
## IV. CLUTCHES & BRAKES – DYNAMICS OF MASS MOTION

In Section III., FIGURE 3 showed an equivalent drive system schematic that is suitable for analysis of any clutch or brake torque and heat load. There are four distinct **LOAD ZONES** shown in FIGURE 3 which separately and as a whole will affect system performance.

1. Input
2. Clutch
3. Output
4. Brake

It is necessary to understand the factors involved that cause a mass system speed change to occur. This will allow an accurate calculation of the clutch or brake engagement heat loads.

The input and output segments of the simplified model each contain mass and various loads tending to change its motion. It is only necessary to discuss a single mass system, such as shown in FIGURE 4, to understand any multiple mass system.



The mass system motion is governed by the following mathematical equation:

$$T = I \times \alpha$$

Where: T is the **NET** torque acting on the mass  
I is the magnitude of the mass  
 $\alpha$  is the acceleration rate of the mass

The above equation can take many forms due to the various systems of measures in use. The (ft-lb-sec) units of measure will be used here.

In clutch and brake system analysis, it is common to refer to speed (N in rpm) and inertia (WR2 in lb-ft<sup>2</sup>). These are familiar and easily calculated or given values.

Note that N relates to  $\alpha$ :  $\alpha = dw/dt = k(dN)/dt$

Where:  $\alpha$  is the acceleration rate (rad/sec<sup>2</sup>)  
dw is the change in velocity (rad/sec)  
dt is the change in time (sec)  
dN is the change in rpm (rpm)  
k is the conversion constant (1/9.549)

and that WR2 relates to I:  $I = WR2/g$

Where: I is the mass moment of inertia  
WR2 is the inertia (lb-ft<sup>2</sup>)  
g is the conversion constant (32.2 ft/sec<sup>2</sup>)

It is important to recognize that it is the **NET** torque acting upon a mass that changes its speed. The magnitude of the **NET** torque is the summation of torques which could come from any number of sources. (Refer to FIGURE 5.)

# Clutch & Brake Design Principles

On an input mass, the prime mover torque is positive since it tends to sustain motion. Friction and windage torques tend to retard motion and are negative. Clutch torque acting on an input mass is negative because it tends to “slowdown” the input mass **during engagement**.

Alternately, on an output mass, the clutch torque, **during engagement**, is positive since it is tending to accelerate the output mass. Work load torque is negative if it is a resistive type load, however, could be positive if the load is overrunning such as is seen on a downhill conveyor or a falling type load. Brake torque, friction and windage torques are all negative.

In summary, the torque tending to accelerate or decelerate a mass is simply a summation of all of the extraneous torques acting on the mass during any particular time period.

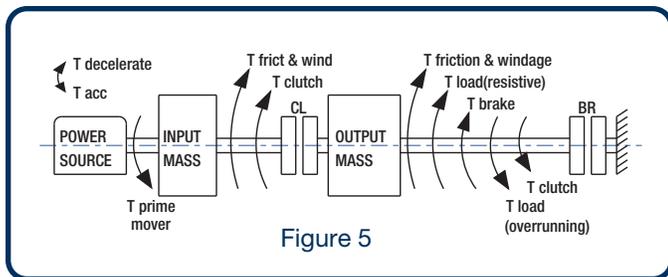


Figure 5

## V. CLUTCHES & BRAKES – HEAT LOADS

Section IV. explained the effects NET torque has on a rotating mass. To establish net torque, it is necessary to define the torque versus speed characteristics of each contributing component.

Prime movers (power sources) and work loads have fairly well defined torque versus speed characteristics. Clutches or brakes also have definable torque versus speed characteristics within the bounds of the many variables affecting the coefficient of friction.

Knowing these torques and summing their effect into **NET TORQUES** acting on the input and output masses, allows the change in speed of each mass to be calculated over a predetermined time interval. Then, knowing the speed of each mass system at intervals of time during the engagement sequence, allows a determination of the differential speed between the unit’s input and output member at any instant of time.

This is paramount to defining the clutch or brake heat load since the unit’s **TORQUE** at any time “t” multiplied by the **SPEED DIFFERENTIAL** at time “t” is the rate of heat generation. ( $q=T\omega$ ) This is commonly called **HEAT RATE** and is a very important factor in clutch and brake applications. Of equal importance, is the **TOTAL HEAT** generated during an engagement.

## ENGAGEMENT CONSTRAINT CURVES

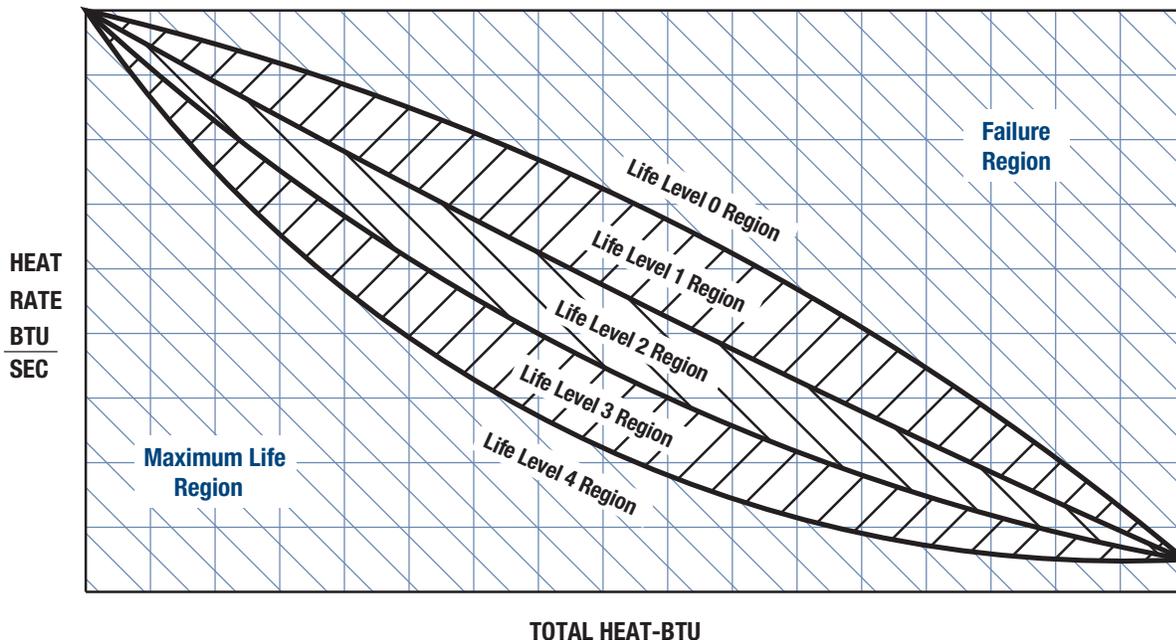
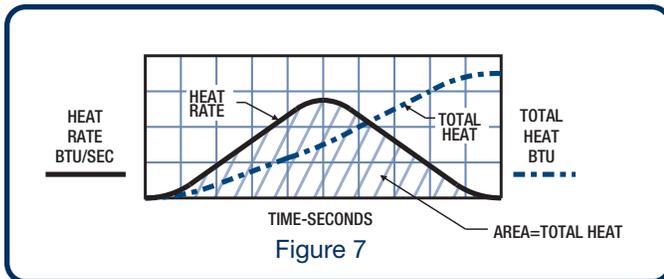


Figure 6

This is arrived at by summing the heat rate over a time interval. ( $Q = \sum T \omega t$ )

FIGURE 6 describes a **HEAT RATE vs. TOTAL HEAT** engagement constraint curve that is typically used by clutch and brake application engineers. The design of the unit will cause the curves to take on different shapes and operating limits, but all designs have envelopes within which successful engagements can be made.

FIGURE 7 shows a typical heat rate and total heat curve generated due to a hypothetical clutch or brake engagement. Note that the area under the heat rate curve is total heat. Engagement profile modifications will change the shape of these curves. It is often possible to achieve heat rate and total heat profiles that can be better tolerated by the system as well as the clutch or brake unit.



## VI. CLUTCHES & BRAKES – TORQUE CAPACITY

There are many configurations of clutches and brakes. The styles manufactured by Industrial Clutch Products are single or multiple disc units that may be actuated by pneumatics, hydraulics, or by mechanical means. These units are further divided into two categories. Those that may be operated:

- 1) “**DRY**” – in atmospheric conditions; or 2) “**WET**” – in oil immersed conditions.

The basic torque capacity of these units may be expressed by the following formula:

$$T = F \times CF \times N \times R$$

Where: T is the torque capacity (lb.-in.)

F is the apply force (lbs.)

CF is the coefficient of friction

N is the number of friction surfaces

R is the disc mean radius (in.)

Of the above factors, all but two, are constant for any specific clutch or brake since the physical size is fixed during the design phase.

The two variable factors are:

1. The apply force which is a function of the operating mechanism.

2. The coefficient of friction which is a function of many variables:

- a) Temperature .....(Dry & Wet)
- b) Humidity .....(Dry)
- c) Velocity .....(Dry & Wet)
- d) Unit pressure .....(Dry & Wet)
- e) Material composition .....(Dry & Wet)
- f) Material cure levels .....(Dry & Wet)
- g) Material wear depths .....(Dry)
- h) Surface roughness .....(Dry & Wet)
- i) Contamination .....(Dry)
- j) Groove patterns .....(Wet)
- k) Tribology(oil factors).....(Wet)

It is apparent that many factors can affect the torque capacity of a clutch or brake. Industrial Clutch Products, utilizing its sixty years of experience and a sound product design, development, and testing program, strives to define all of the factors which affect clutch and brake performance.

## VII. CLUTCHES & BRAKES – HEAT CAPACITY

In Section V. the principles of **HEAT RATE** and **TOTAL HEAT** were explained. The importance of establishing an accurate **NET TORQUE** on the input and output mass was also discussed.

**NET TORQUE, which is the summation of all torques acting on a mass, controls the MOTION (speed) of the mass during the engagement sequence. NET TORQUE is not clutch or brake torque. Clutch or brake torque is only one component of several that contribute to NET TORQUE. CLUTCH OR BRAKE TORQUE, multiplied by the DIFFERENTIAL SPEED between the unit’s input and output determines the HEAT RATE. Multiplying HEAT RATE by the time period over which it acts, yields TOTAL HEAT. HEAT RATE and TOTAL HEAT determine the heat load that the clutch or brake unit must dissipate.**

Therefore, it can be summarized:

CLUTCHES AND BRAKES MUST HAVE THE RELATIVE MOTION BETWEEN THEIR INPUT AND OUTPUT MEMBERS DEFINED WHEN ACCELERATING OR DECELERATING A LOAD. TO DO SO, REQUIRES AN ACCURATE DETERMINATION OF THE SPEED OF THE MASSES CONNECTED TO THE CLUTCH OR BRAKE DURING ENGAGEMENT.

# Clutch & Brakes Design Principles

Industrial grade heavy-duty clutches and brakes encounter a wide variety of loads:

1. Low energy -infrequent engagements.
2. High energy-infrequent engagements.
3. Low energy -frequent engagements.
4. High energy-frequent engagements.

The load duty cycle, coupled with the unit's ability to dissipate heat, defines the operating temperature range. Temperatures must be maintained within acceptable limits if the unit is to exhibit consistent torque capacity and repeatable stopping times, angles, or distances. Of equal importance, is the ability of the clutch or brake unit to resist wear and this is highly dependent upon the operating temperature.

The importance of HEAT RATE and TOTAL HEAT cannot be overstressed. Any clutch or brake unit must function within the confines of its geometry, heat transfer laws, metallurgy of the opposing friction surfaces, and chemistry of the linings.

HEAT RATE is used to predict the temperature at the friction surface interface. This is commonly called SURFACE TEMPERATURE and is an extremely important factor in clutch and brake design. The metals and friction materials used in clutches and brakes have definable heat transfer capabilities. When the materials at the friction interface cannot remove the heat as rapidly as it is generated, the surface temperature rises until the temperature difference is high enough to drive the heat through the materials. In extreme cases, when the heat cannot be removed rapidly enough, surface temperatures rise to the point where the materials melt or ablate. This excess heat is then removed by providing the energy required to cause the material to change phase. This is a totally unacceptable method of heat removal in most cases, but can arise, when an unexpected load is encountered in an industrial application.

Excessive surface temperatures will cause high thermal gradients (temperature differences) to exist throughout the material's thickness. This causes a phenomenon to occur which is called HEAT CHECKING.

Heat checking is minute cracks which appear at the friction surface interface. Repeated high temperature engagements will cause these cracks to enlarge. After some period of time, the cracks grow and cause surface irregularities which can be felt with one's thumbnail. This "shaves" the linings and causes accelerated lining wear. Finally, if the cracks continue to progress, they will propagate into larger cracks which can split the plates and render the clutch or brake useless.

Controlling SURFACE TEMPERATURES within acceptable limits is dependent upon knowing the peak HEAT RATES

and the heat conducting and radiating capabilities of the clutch or brake unit.

Once the peak HEAT RATES are established, it is necessary to determine the magnitude of TOTAL HEAT that will be generated during a single engagement. Heat transfer laws are used to establish the temperature rise. The rise in temperature of the clutch or brake plates is dependent upon their mass. This rise in temperature is commonly referred to as the BULK TEMPERATURE of the unit.

The ability of the unit to convectively and radiantly remove this heat is dependent upon the ventilating characteristics incorporated within the clutch or brake design, as well as other factors, such as rotational speed, ambient temperatures, and cleanliness of the heat transfer surfaces.

BULK TEMPERATURE is important because it is the determining factor in the allowable frequency of engagement. Repeated engagements will cause the BULK TEMPERATURE of the unit to continue to rise until some steady state condition is reached where the temperature difference of the surrounding atmosphere and the unit's BULK TEMPERATURE is sufficiently high enough to remove the TOTAL HEAT entering the clutch or brake.

The SURFACE TEMPERATURE rise is fairly constant for each engagement within the practical confines of acceptable operating temperature ranges. By knowing the SURFACE TEMPERATURE rise and the steady state BULK TEMPERATURE, it is possible to predict the PEAK SURFACE TEMPERATURE by adding the steady state bulk temperature to the surface temperature rise per engagement.

Industrial Clutch Products can calculate the actual operating temperatures of their units when **the proper input data is provided**. This is accomplished by utilizing computer models which have been previously verified or, in some cases, modified by actual tests to generate accurate information for the unit involved.

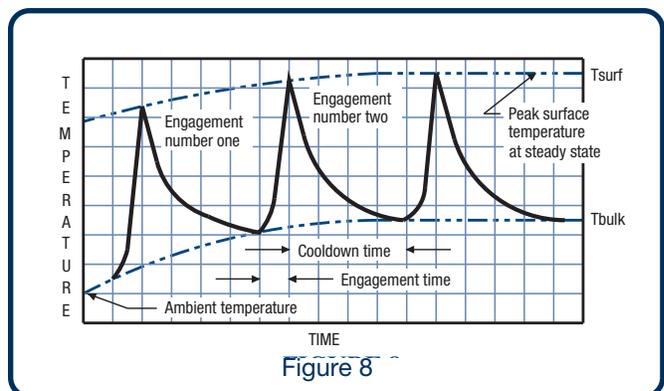


FIGURE 8 shows a typical time vs. temperature chart that could occur in an industrial application. The rise in temperature followed by the exponential decay of temperature is classic for clutches and brakes. The primary aim of Industrial Clutch Products is to utilize the latest technology, materials, and past experience to achieve operating limits in excess of previous designs and competitive products. This is accomplished by testing, documenting, and more testing.

## VIII. AIR-HYDRAULIC & CONTROL SYSTEMS

Most applications require some consideration be given to the control system to ensure that the drive will operate as specified. This is critical in those applications requiring specific starting or stopping times, distances, angles, or those that must operate within specified system engagement shock and power draw limits.

### FACTORS AFFECTING THE CLUTCH/BRAKE UNIT PERFORMANCE

- Valve Flow Characteristics
- Valve Electrical Delay
- Valve Location
- Quick Release Valve Characteristics
- Relay Electrical Delay
- Plumbing – Line Size, Length, and Fittings
- Environment – Cleanliness, Temperature, and Humidity
- Clutch/Brake Pneumatic or Hydraulic Response
- Clutch/Brake Chamber Fill and Exhaust Times
- Adequacy of Supply Pressure

**DO NOT IGNORE CONTROL SYSTEM RESPONSE.**

## IX. TYPES OF LOADS

Virtually all loads can be categorized into one of three types with respect to their torque requirement vs. speed.

**VARIABLE TORQUE:** Centrifugal fans, blowers, pumps, and compressors. Some impeller type mixers and axial flow compressors.

**CONSTANT TORQUE:** Reciprocating and lobe type compressors. Extruders, conveyors, ball and rod mills. Tension stands, most positive displacement pumps and drum type mixers.

**CONSTANT HP:** Wire drawing machinery, grinders, and wind-up stands. Some let-off reels, chippers, de-barkers, and mixers.

As mentioned in Section V, it is important to define the load's torque vs. speed characteristic during clutch or brake engagement. If the machine is to be started unloaded, these factors do not contribute to the clutch or brake heat load. Here the problem reduces to one of the available energy of the input mass and prime mover, versus the load breakaway torque and the energy the output mass requires to place it in motion. The clutch controls the transfer of energy during the engagement sequence.

However, if the machine is to be started in a loaded condition, defining the load's torque vs. speed response is extremely important. This is generally referred to as the **load's superimposed work torque** during the engagement sequence.

For any given clutch torque, the addition of a superimposed work torque, lengthens the start or stop time. This longer slip period increases the TOTAL HEAT of engagement. Increasing the clutch torque to compensate for the superimposed work torque will shorten the engagement time. This is feasible only if an acceptable clutch HEAT RATE is not exceeded and, if the input power system can provide the necessary energy to accelerate the output system within acceptable design stress levels.

The purpose of any clutch or brake is to control the start/stop motion of the system within a definable range. This requires a balancing of several factors of which the type of load, its breakaway torque characteristics, and allowable accelerating limits are but a few.

## X. CALCULATION OF SYSTEM INERTIAS

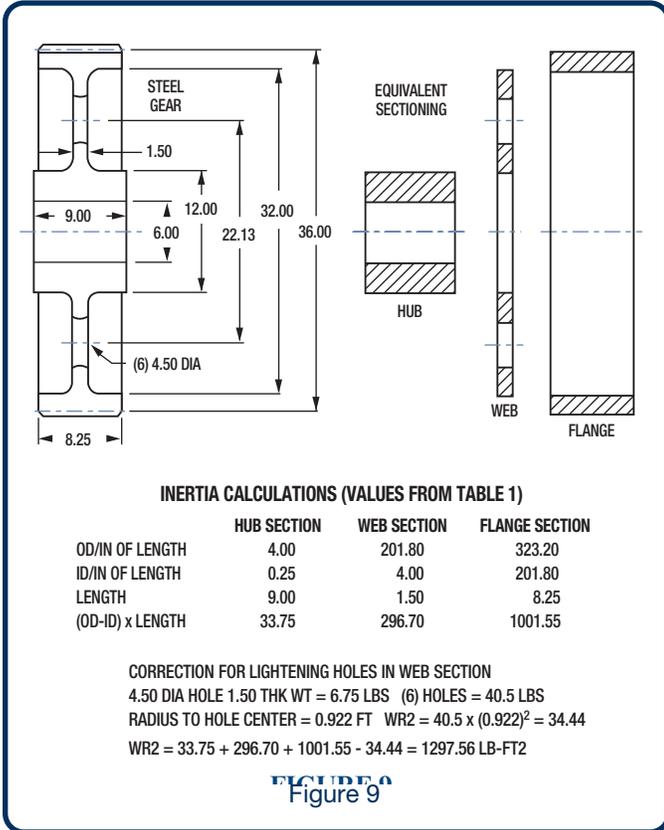
Reducing a drive train to an equivalent system, such as shown in FIGURE 3, requires an understanding of how inertia is calculated and how it may be reflected to different locations within the drive train.

### A. Inertia calculations

The calculation of inertias in a rotating system is a simple process but can be time consuming in a complicated multi-mass system. The usual procedure is to break each component into several cylindrical shapes for which inertia values can be readily tabulated or calculated. This is outlined in FIGURE 9.

# Clutch & Brake Design Principles

Table 1 may be used to obtain the inertia values of circular shapes.



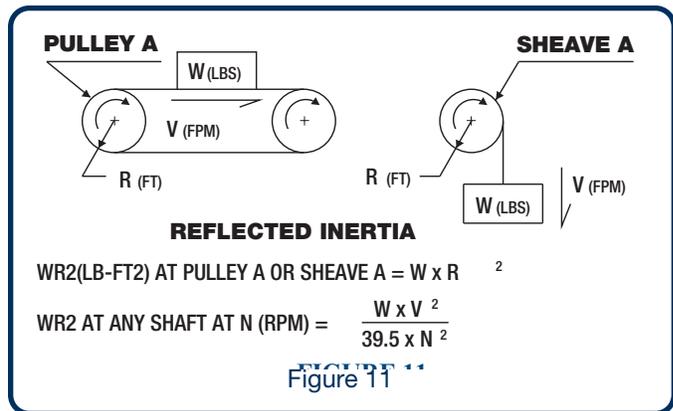
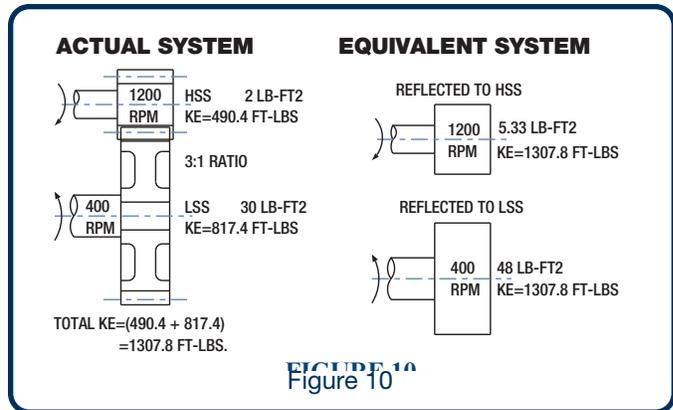
## B. Reflecting inertia

The rules of reflecting inertia in a rotating system are easily understood if it is recognized that any rotating mass contains energy proportional to the magnitude of the inertia and its rotational speed squared.

$$KE = \frac{1}{2} MV^2 = \frac{1}{2} \frac{W}{g} (R\omega)^2 = \frac{1}{2} \frac{WR^2}{32.2} \left(\frac{2\pi N}{60}\right)^2 = \frac{WR^2 N^2}{5872.6}$$

Where: KE is the mass energy (ft-lbs)  
 WR<sup>2</sup> is inertia (lb-ft<sup>2</sup>)  
 N is speed (rpm)

In order to move inertias to different locations within the drive train, it is necessary to preserve the level of energy the mass contained at its original location. For this to happen, the magnitude of the inertia must change in accordance with the speed ratio squared between the shaft the inertia is being moved from and the shaft the inertia is being moved to. FIGURE 10 shows this principle and the rules involved are listed below.



## RULES OF REFLECTING INERTIA

### HELPFUL AIDS

- Reflect inertias to the shaft where the clutch or brake is located.
- Linear inertias, such as found on conveyors, sheaves, or drums, can be reflected as shown in FIGURE 11.
- When reflecting from a low speed shaft to a higher speed shaft:  
***DIVIDE BY THE RATIO SQUARED***
- When reflecting from a high speed shaft to a lower speed shaft:  
***MULTIPLY BY THE RATIO SQUARED***

## XI. TORQUE CAPACITY RATINGS

Torque capacity of a clutch or brake unit is easily misunderstood due to the manner in which the industry has decided to assign its terminology. Identical units may be assigned several torque capacities. They all make sense if an explanation is given as to what the listed capacity entails. If this is not done, service factors are heaped upon service factors, or not included at all.

Industrial Clutch Products lists the STATIC AND DYNAMIC TORQUE CAPACITY for their units. These terms and others are defined below to allow the user to relate to their meaning and use when selecting a unit. This is no guarantee that an evaluation of competitive units on an equal basis can be made, since each manufacturer uses its own rating method.

### STATIC OR SLIP TORQUE CAPACITY

This is the theoretical calculated torque capacity that an engaged unit could transmit without slippage. It is based upon an appropriate coefficient of friction for the lining in use at a specified actuation pressure.

Note there is no service factor in this rating and provides the base to which service factors may be applied. Another term synonymous with STATIC TORQUE is SLIP TORQUE which comes from the fact that this is the level of torque that causes the unit to just start to slip.

The STATIC TORQUE capacity listed for brakes, allows an evaluation of the holding capacity of the unit after engagement. Brake applications which have stringent holding requirements must use a suitable service factor applied to the STATIC TORQUE listed.

### WORKING TORQUE CAPACITY

Working torque is dictated by the application and relates to the amount of work being performed by the machine on a regular basis. The unit must be selected based on the maximum torque that could be imposed. This may or may not be equivalent to the prime mover full load torque.

WORKING TORQUE, multiplied by a suitable service factor, is STATIC TORQUE. Any time WORKING TORQUE is listed in a manufacturer's catalogue, a service factor has already been applied. Unfortunately, these service factors are not universal, and in some cases, are not given. A more appropriate base from which to start is STATIC TORQUE.

Industrial Clutch Products has removed all references to working torque within their ratings to help eliminate the confusion and sizing difficulties that result. We use service

factors based on the application duty and multiply the WORKING TORQUE by this service factor to obtain the minimum required STATIC TORQUE capacity.

***SELECT NO UNIT BASED ON STATIC TORQUE WITHOUT HAVING FIRST APPLIED THE PROPER SERVICE FACTOR.***

### DYNAMIC TORQUE CAPACITY

This is the torque the unit transmits during the engagement sequence. At this time, there is relative (dynamic) motion between the input and output members.

DYNAMIC TORQUE capacity is a function of the many variables affecting coefficient of friction as listed in Section VI. However, within certain confines, a conservative value of dynamic coefficient of friction can be selected and used with adequate confidence for most applications.

Practically all materials exhibit a different static and dynamic coefficient of friction. This gives rise to the term STATIC TO DYNAMIC RATIO and is critical in some applications such as tensioning drives or those which require controlled slip. For these types of drives, a material selection which has a STATIC TO DYNAMIC RATIO of one or less is desirable to eliminate a phenomenon known as stick-slip. Stick-slip can be very troublesome as the relative speed between the input and output members approaches zero. A high ratio means the coefficient of friction changes radically at this point and smooth speed or tension control is impossible in this range. Industrial Clutch Products lists the DYNAMIC TORQUE capacity for its units to allow calculations to be made for starting and stopping times or angles.

## XII. SERVICE FACTORS

SERVICE FACTORS (SF) are applied to the LOAD TORQUE (LT) to determine the Clutch or Brake's STATIC TORQUE CAPACITY (STC).

$$STC = (LT) (SF)$$

### SERVICE FACTOR VALUES

Service Factor Values are determined by operating conditions:

- DUTY CYCLE
- POWER SOURCE
- LOAD CHARACTERISTICS
- INERTIA RATIOS
- OPERATING SPEED
- OPERATING ENVIRONMENT

# Clutch & Brake Design Principles

## ■ DUTY CYCLE

**LIGHT** – INFREQUENT ENGAGEMENTS; STEADY POWER SOURCES; UNIFORM LOADS

**NORMAL** – FREQUENT ENGAGEMENTS; STEADY POWER SOURCES; NON-UNIFORM LOADS; LOW INERTIA RATIOS; MEDIUM SPEEDS

**HEAVY** – FREQUENT ENGAGEMENTS; PULSATING POWER SOURCES; HEAVY SHOCK LOADS; HIGH INERTIA RATIOS; HIGH SPEED; UNFRIENDLY ENVIRONMENT

## ■ POWER SOURCE

**UNIFORM** – ELECTRIC or HYDRAULIC MOTORS; GAS or STEAM TURBINES

**PULSATING** – INTERNAL COMBUSTION ENGINES

## ■ LOAD CHARACTERISTICS

**UNIFORM** – STEADY POWER DRAW

**MEDIUM SHOCK** – OCCASIONAL POWER SURGES AND OVERLOADS

**HEAVY SHOCK** – REGULAR AND HIGH FLUCTUATIONS IN POWER DRAW AND OVERLOADS

## ■ INERTIA RATIOS (WR2 INPUT/WR2 OUTPUT)

**LOW** – (1) or less

**MEDIUM** – (1) to (9)

**HIGH** – (10) or more

**NOTE:** As Inertia Ratios increase, the effect of the Power Source during engagement becomes less of a factor.

## ■ OPERATING SPEED

**LOW** – 400 RPM or less

**MEDIUM** – 400 to 1000 RPM

**HIGH** – 1000 RPM & up

## ■ OPERATING ENVIRONMENT

**FRIENDLY** – CLEAN; NO EXPOSURE TO WEATHER; AMBIENT TEMPERATURES LESS THAN 100 DEGREES F.

**UNFRIENDLY** – TEMPERATURES OVER 100 DEGREES F; EXPOSURE TO EXCESSIVE DUST, DIRT, OR CORROSIVE MATERIALS; EXPOSURE TO THE ELEMENTS: RAIN, SNOW, SLEET, HIGH HUMIDITY, OR SALT LADEN AIR; EXPOSURE TO SURROUNDING MACHINERY CONTAMINANTS: OIL; GREASE; EXCESSIVE OIL LADEN AIR

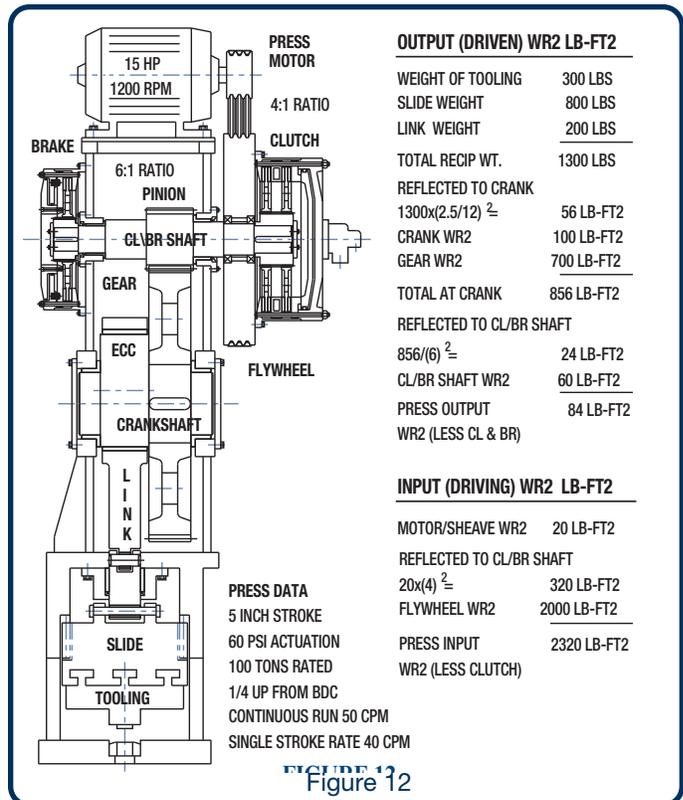
## SERVICE FACTOR CATEGORIES

It is apparent that many variables affect the category into which a drive system should be placed in order to select the Service Factor. Generically classifying machinery into categories is possible, however, it is necessary to be aware of all of the operating conditions to establish adequate Service Factors. For example, a system comprised of elements that qualify it for light duty might be exposed to a totally unfriendly environment that requires a heavy duty service factor be applied for satisfactory operation and longevity.

**TABLE 3** describes some of the more common Service Factors utilized when selecting Industrial Clutch Products' clutches and brakes.

### IMPORTANT

*Clutches and Brakes must not only transmit TORQUE, they must dissipate the HEAT of engagement. The simple application of a TORQUE service factor will NOT necessarily ensure satisfactory performance. Unit size does increase with larger Service Factors, and, as a result, the heat capacity increases. The final test however, is the application's HEAT LOAD vs. the unit's HEAT CAPACITY. PLEASE REFER ALL USER SELECTIONS TO INDUSTRIAL CLUTCH CORPORATION FOR REVIEW AND FINAL APPROVAL.*



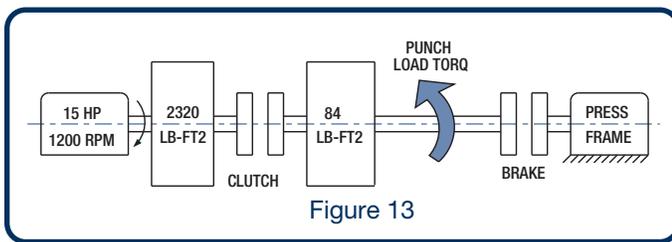
## XIII. APPLICATION EXAMPLES

The following examples represent a cross-section of applications that demonstrate the principles outlined in Sections I through XII. Table 1 (WR2 values), Table 2 (equations) and Table 3 (service factors) are also used through-out these examples.

Rather than just tabulate the equations and fill in the blanks, the equations are introduced in logical sequence with appropriate explanations of their use. This leans toward a textbook approach which should help the user gain a greater insight into the application and those factors which we incorporate within our designs to allow for an efficient drive system.

### A. PUNCH PRESS

FIGURE 12 describes the press layout and pertinent data required to obtain the simplified schematic shown in FIGURE 13.



The input and output inertia values shown in FIGURE 13 are for the press components. To obtain the total system inertia, the clutch and brake components that rotate with these masses must be included. This requires that the unit's preliminary size be calculated.

The clutch working torque can be calculated using the following equations.

$$\text{Crankshaft Torque} = 100 \times 2000 \sqrt{.25(5-.25)} = 217945 \text{ LBIN}$$

$$\text{Clutchshaft Torque} = \frac{\text{Crank Torq}}{\text{Gear Ratio}} = \frac{217945}{6} = 36324 \text{ LBIN}$$

The clutch minimum static torque is obtained by multiplying the working torque by a suitable service factor.

Table 3 lists the service factor for punch presses at 2.0.

The minimum static torque **catalogue rating** would be:

$$\text{Clutch Static Torq} = \text{Working Torq} \times \text{Serv Fact} \times \frac{\text{Rated Press}}{\text{Supply Press}}$$

$$\text{Clutch Static Torq} = 36324 \times 2.0 \times \frac{100}{60} = 121080 \text{ LBIN}$$

The preliminary unit selected will be a model LK-217 clutch with a static torque capacity of 172800 lb-in as this is the smallest unit that meets or exceeds the calculated torque requirement.

In press applications, the brake is generally one-half the size of the clutch. A reasonable selection would be the Model LKB-117 spring-set, air-released brake. This allows us to calculate a preliminary total system inertia. Selections can be modified later if the engagement heat, and stopping times or angles are excessive.

MACHINE INPUT INERTIA	2320 LB-FT2
LK-217 DRIVING INERTIA	141 LB-FT2
TOTAL INPUT INERTIA	2461 LB-FT2
MACHINE OUTPUT INERTIA	84 LB-FT2
LK-217 DRIVEN INERTIA	18.3 LB-FT2
LKB-117 ROTATING INERTIA	9.8 LB-FT2
TOTAL OUTPUT INERTIA	112.1 LB-FT2
INERTIA RATIO	= 2461/112.1 = 22:1

$$\text{ENGAGED INERTIA} = 2461 + 112.1 = 2573.1 \text{ LB-FT2}$$

The **INERTIA RATIO** can be used to calculate the flywheel slowdown due to the clutch engagement. This is important to press motor life since excessive slowdown on a repetitive basis will cause overheating.

$$\% \text{ Engagement Slowdown} = \left( 1 - \sqrt{1 - \frac{2 \times \text{WR}^2 \text{ Output}}{\text{WR}^2 \text{ Input}}} \right) 100$$

$$\% \text{ Engagement Slowdown} = \left( 1 - \sqrt{1 - \frac{2 \times 112.1}{2461}} \right) 100 = 4.6\%$$

The **ENGAGED INERTIA** can be used to calculate the slowdown while performing the work (punching). Here it is assumed the motor has restored the rotating system to full RPM prior to punching. This may or may not be the case, however to accurately calculate recovery, requires the use of a computer. Our programs are designed to incorporate the effect of motor power through-out the cycle so this data is available. For the purpose of estimating, the following procedure can be used.

$$\text{Punching Energy} = \text{Tonnage} \times 2000 \times \frac{\text{Thickness}}{12} \times \text{Shear Factor}$$

**NOTE:** Shear factor is the percentage of penetration the punch makes in the material to the fracture point. (ie., 50% penetration is 1/2 material thickness and SF = 0.5)

# Application Examples

$$\text{Punching Energy} = 100 \times 2000 \times \frac{.25}{12} \times .5 = 2083 \text{ FT LBS}$$

$$\text{Slowdown K factor} = \frac{\text{RPM}}{\sqrt{\frac{\text{Punch Energy}}{\text{Engaged WR}^2}}}$$

$$\text{Slowdown K factor} = \frac{300}{\sqrt{\frac{2083}{2573.1}}} = 333$$

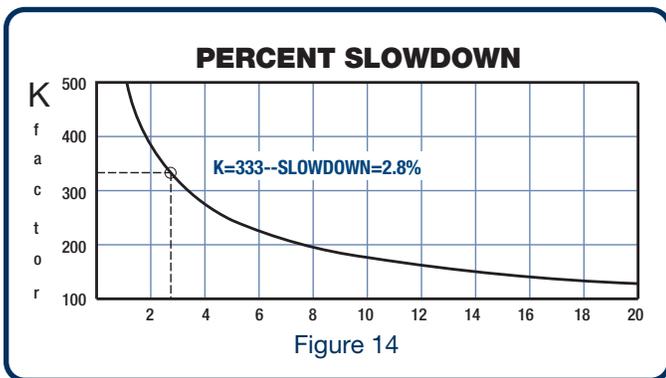


FIGURE 14 may be used to obtain the press slowdown when punching by selecting the appropriate K factor.

The following equations may be used to calculate the heat of engagement and the total heat **EACH** unit must dissipate. For pure inertial acceleration of a mass started from rest, the heat energy per cycle for the clutch, is equal to the final energy contained in the mass after acceleration.

Decelerating a mass to rest with a brake, requires that the energy the mass contains be dissipated by the brake. Therefore, for pure inertial engagements such as found in press applications, the amount of energy the clutch and brake each must dissipate is equal. The magnitude of the energy per engagement is equal to the energy contained in the engaged output mass.

$$\text{Engagement Heat} = \frac{\text{WR}^2 \times \text{RPM}^2}{5872.6} = \frac{112.1 \times 300^2}{5872.6} = 1718 \frac{\text{FT LBS}}{\text{CYCLE}}$$

$$\text{Total Heat} = \frac{\text{FT LBS}}{\text{CYCLE}} \times \frac{\text{CYCLES}}{\text{MIN}} = 1718 \times 40 = 68700 \frac{\text{FT LBS}}{\text{MIN}}$$

The units selected for this application are a LK-217 clutch and a LKB-117 brake. Note the units are identical in size with the exception that the brake is a single plate unit and the clutch is a two plate unit. The brake has half the area of

the clutch yet must dissipate the same amount of heat. The clutch rotates and its heat dissipating capacity is further enhanced when compared to the brake members which are stationary.

In the majority of press applications, the primary screening criteria for the clutch, is the torque capacity required for punching; and for the brake, it is the heat capacity required for start-stop cycling.

The service factors for this application are as follows:

$$\text{Clutch Service Factor @ Oper. Press} = \frac{\text{Clutch Static Torq}}{\text{Required Torq}} = \frac{103680}{36324} = 2.86:1$$

$$\text{Brake Service Factor} = \frac{\text{Heat Capacity}}{\text{Total Heat}} = \frac{79800}{67800} = 1.18:1$$

These factors are acceptable since they exceed minimum values established by Industrial Clutch Products.

MINIMUM CLUTCH TORQ SERVICE FACTOR = 2.0

MINIMUM BRAKE HEAT SERVICE FACTOR = 1.0

**NOTE:** These factors and heat capacities are strictly applicable for press applications and vary widely for other types of duty. It is not feasible to publish data for all of the duties our products are exposed to, so we request that the user refer applications to Industrial Clutch Products.

The use of service factors allow for load variables that have not been considered, yet are present to some degree, in every application. A few are listed below:

- Adverse environmental conditions
- Unreliable air supplies
- Press friction
- Drive efficiency
- Overload forces
- Counter-balance forces
- Die stripping forces

Since the engagement energy and the punching energy are now known, a calculation of the cycling power the motor must provide can be made.

$$\text{Cycle Power} = (\text{Punch Energy} + 2 \times \text{Engage Heat}) \frac{\text{Cycles}}{\text{Min} \times 33000}$$

$$\text{Cycle Power} = (2083 + 2 \times 1718) \frac{40}{33000} = 6.69 \text{ HP}$$

It is interesting to note, that of the 6.69 HP required to cycle the press at 40 CPM with a typical 100 ton punching

requirement, 4.17 HP must be provided by the motor to accommodate the energy required to start-stop the output inertia. 25% of this start-stop energy is required by the clutch and brake driven inertia and 75% by the press components.

It is important that the press drive and clutch-brake components be sized to minimize the output inertia in order to reduce the motor power requirements, the clutch and brake heat loads, and the brake torque requirements.

The LK-LKB separate clutch brake units contain very low inertia values when compared to combination clutch-brake units and for this reason will cycle faster and with greater longevity. **DRIVEN INERTIA IS AN IMPORTANT CONSIDERATION.**

Brake heat capacity, as mentioned above, is the primary screening criteria in press applications. Equally important is the ability of the brake to stop the press accurately and reliably within an acceptable stopping angle/distance and time. All of the spring-set air released Model LKB brakes can be provided with variable numbers of springs with different rates to establish the torque required.

The torque required to stop the press is determined by specifying a stopping angle or time and solving for the torque using the equations that follow.

$$\text{Torque} = \frac{WR^2 \times \text{RPM}^2 \times .11736}{\text{Crank Angle} \times \text{Ratio}} = \frac{WR^2 \times \text{RPM} \times .03912}{\text{Time}}$$

The above equations use the angle at the crankshaft while the ratio is that between the brake shaft and crankshaft.

For this application assume a stopping angle of 15 degrees is acceptable and solve for the brake torque.

$$\text{Torque} = \frac{112.1 \times 300^2 \times .11736}{15 \times 6} = 13156 \text{ LBIN}$$

The dynamic torque rating for the Model LKB-117 is 32265 lb-in at 60 psi release pressure. Note that release pressure is a function of the springs used in the unit.

Since 13156 lb-in is all that is required, one could select a unit which uses a lower release pressure. This will aid in preventing clutch-brake fight when cycling the press and is commonly done. We do not list all of the available torques and release pressures available. The upper limit of 60 psi as listed, is the highest practical range for press applications in order to minimize clutch-brake fight. Consult our Engineering Department to obtain a unit size for torque capacities above or below those listed.

The user should be aware of the accuracy of these equations in predicting actual stopping times and distances. The equations are accurate provided full unit torque is exhibited through-out the engagement period.

Note the stopping time in this application:

$$\text{Time} = \frac{112.1 \times 300 \times .03912}{13156} = .100 \text{ SEC}$$

This is a very short stopping time and not unusual in most press applications. Because the time period is so small, the pneumatic response of the brake has a bearing on the results of the calculations. A finite time (approximately .050 sec for the LKB-117) is required to exhaust (set) the brake. While the brake is exhausting, the torque of the brake is changing from zero at the instant the plates contact up to the rated level when the chamber pressure decays to zero.

Rather than having a constant brake torque imposed on the system, a variable brake torque as a function of the pressure in the brake chamber is in evidence for the duration of the exhaust period. This increases the actual stopping angle and time when compared to the constant torque calculations made above.

It is difficult to incorporate the effects of pneumatic response without using the iterative powers of a computer. Our programs will do this for the user. As a rough rule of thumb, you may expect stopping angles and times to increase two to three times in actual use when using the chamber volumes incorporated in our designs. Competitive units which do not have these low volume chambers will require factors larger than this. **ACTUATION VOLUMES ARE AN IMPORTANT CONSIDERATION.**

## PRESS APPLICATION SUMMARY:

The following factors are important when selecting a clutch and brake for a press application:

Clutch & Brake Factors	Performance Effect
Clutch Torque Capacity	■ Press capacity
Brake Heat Capacity	■ Stopping accuracy
	■ Lining life
Clutch and Brake driven inertias	■ Brake heat
	■ Stopping angle
	■ Stopping time
	■ Motor power
	■ Inertia ratios
	■ Slowdown factors
Clutch and Brake actuation volumes	■ Stopping angle
	■ Stopping time
	■ Air consumption

# Application Examples

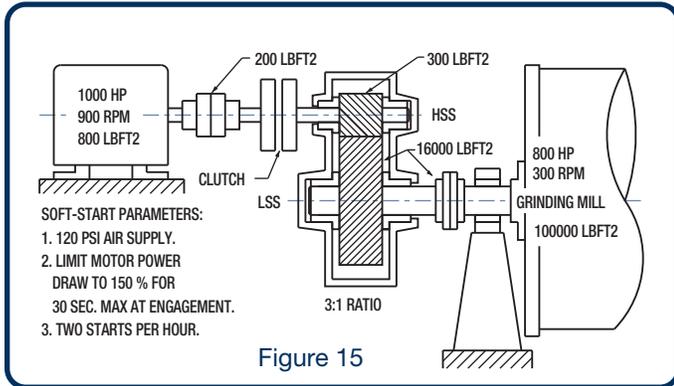


Figure 15

## B. CONTROLLED SOFT-START APPLICATION

Figure 15 describes a typical geared soft-start application such as may be found in a rod or ball mill. This application has a constant superimposed work torque present during the engagement sequence. Breakaway torque will not be considered here since the magnitude of the breakaway torque will be less than the work torque load. This is not the case when a variable torque work load is present and

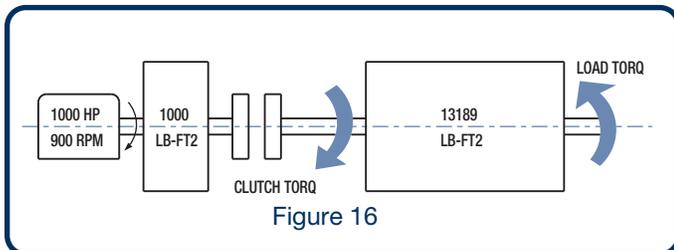


Figure 16

breakaway torque becomes an important consideration.

This drivetrain can be reduced to the two mass system shown in FIGURE 16, by following the procedure listed below:

### OUTPUT WR2

Low Speed Shaft WR2 = 16000 + 100000 = 116000 LBFT2  
 Refl. to High Speed Shaft = 116000/(3 x 3) = 12889 LBFT2  
 High Speed Shaft Output WR2 = 300 + 12889 = 13189 LBFT2

### INPUT WR2

Motor and Coupling WR2 = 800 + 200 = 1000 LBFT2

**INERTIA RATIO** = 1000/13189 = 0.076 : 1

This drive-train can be classified as a low inertia ratio system which implies that the transfer of energy from the input mass to the output mass will be negligible during the engagement sequence. The prime mover will have to supply virtually all of the energy to accelerate the output mass as well as provide the torque necessary to perform the work.

The specifications for this application require that the prime mover not be loaded over 150% for a time period not to exceed 30 seconds. The clutch allows the motor to reach its no load speed in an unloaded condition and then work on a more efficient portion of the motor's torque vs. speed curve during the engagement. While inrush current on an unloaded motor is high, it is nowhere near that which would be seen if the motor were to be started loaded. In fact, the pull-up torque of the motor may not be sufficient to start the load before motor breakdown torque is reached and overheating or stalling results. Finally, in some instances, the entire electrical operating costs of a site is based upon the highest peak power draw that occurred during any billing period so a soft-start clutch makes economical sense.

Because of the huge output mass, the inertia of the clutch components can be neglected without appreciably affecting the results of the calculations. Heat rate and total heat can be calculated without pre-selecting a clutch.

Calculations for this application can be performed as follows:

$$\text{Allowable Motor Torq} = \frac{1.5 \times 1000 \times 63025}{900} = 105042 \text{ LBIN}$$

$$\text{Allowable Clutch Torq During Engagement} = 105042 \text{ LBIN}$$

$$\text{Superimposed Load Torq} = \frac{800 \times 63025}{300 \times 3} = 56022 \text{ LBIN}$$

$$\text{Net Accelerating Torq} = 105042 - 56022 = 49020 \text{ LBIN}$$

$$\text{Acceleration Time} = \frac{13189 \times 900 \times .03912}{49020} = 9.5 \text{ SEC}$$

$$\text{Heat Rate} = \frac{\text{Clutch Torq} \times \Delta \text{RPM}}{63025}$$

In this application, the clutch torque is constant during the engagement sequence so the **PEAK HEAT RATE** occurs when a maximum differential speed exists between the input and output members of the clutch. Maximum differential speed is at the instant of initial engagement (900 rpm). Peak heat rate, under these circumstances, is calculated as follows:

$$\text{Peak Heat Rate} = \frac{105042 \times 900}{63025} = 1500 \text{ HP}$$

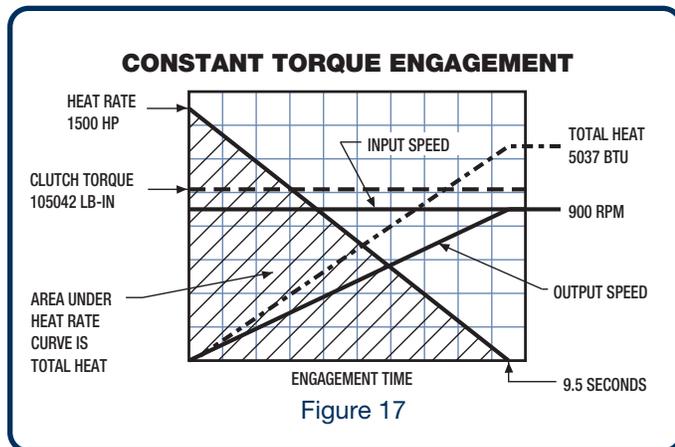
**TOTAL HEAT** is equal to the area under the heat rate curve for any engagement. Here the heat rate curve is triangular in shape with the maximum heat rate occurring at time t=0 and diminishing to zero at time t=9.5 seconds. The area of a triangle is 1/2(base;time)x(height;heat rate) so, in this case, the total heat is easy to calculate. This would not be the case if the clutch torque was varying with time or the load torque was varying with rpm. Here, an irregularly shaped heat rate curve would result. Our computer programs handle this by breaking the area under the curve into many rectangles and summing the area of each rectangle into a total heat figure.

The equation that follows may appear to not resemble the simple area equation, however, it is developed directly from this equation with modifications incorporated for the appropriate units of measure.

$$\text{Total Heat} = \frac{\text{Clutch Torq} \times \text{DRPM} \times \text{DTime}}{178305}$$

$$\text{Total Heat} = \frac{105042 \times 900 \times 9.5}{178305} = 5037 \text{ BTU}$$

The results are shown graphically in FIGURE 17.



In preparation for unit selection, a summary of the drive parameters may be listed:

FULL LOAD MOTOR TORQUE:	70028 LB-IN
GRINDING MILL LOAD TORQ:	56022 LB-IN
ENGAGEMENT CLUTCH TORQ:	105042 LB-IN
ENGAGEMENT TIME:	9.5 SEC.
OPERATIONAL SPEED:	900 RPM
AIR SUPPLY:	120 PSI
HEAT LOAD:	5037 BTU @ 1500 HP

The clutch selection is based upon:

1. Torque capacity and service factor.
2. Heat capacity and life.
3. Rotational speed capacity.

**1. Torque capacity and service factor.** For this application, torque capacity will be based upon the motor power available since the power is dedicated solely to the load and not split in any other manner. Users find ways to maximize machine performance to utilize all available power. Overloads may also occur and the clutch will be expected to pull through these conditions without slippage.

$$\text{Working Torq} = \frac{\text{HP} \times 63025}{\text{RPM}} = \frac{1000 \times 63025}{900} = 70028 \text{ LBIN}$$

The service factor for this application from Table 3 is 5.0 and the minimum static torque will be:

$$\text{Minimum Static Torque} = 70028 \times 5.0 = 350140 \text{ LBIN}$$

The catalogue static torque ratings are likely calculated at some actuation pressure other than the application's available supply pressure. A correction for this must be made so that the catalogue ratings column can be used directly.

This application will use the Model AGT Series of clutches due to the mounting configuration required. These units are rated at 100 psi and the supply is 120 psi. A correction is required before entering these rating columns and selecting a unit.

$$\text{Minimum Catalogue Rating} = \text{Static Torq Req'd} \times \frac{\text{Rated Press}}{\text{Supply Press}}$$

$$\text{Minimum Catalogue Rating} = 350140 \times \frac{100}{120} = 291780 \text{ LBIN}$$

The Model AGT-224 is selected on a torque basis since it has a static torque capacity of 319400 lb-in which exceeds the value of 291780 lb-in required.

**2. Heat capacity and life.** The selected unit must have sufficient heat capacity to withstand the heat loads and exhibit an acceptable life. The effective lining area of the clutch is one of the criteria used to investigate the suitability of the unit for the imposed heat loads. The Model AGT-224 has an area of 1152 sq.in. The heat loads divided by this area are commonly referred to as the unit's SPECIFIC HEAT LOADS. These values are then used with engagement constraint curves such as shown in FIGURE 6 to estimate the life factors.

$$\text{Specific Heat Loads} = \frac{5037}{1152} @ \frac{1500}{1152} = 4.37 \frac{\text{BTU}}{\text{INCH}^2} @ 1.3 \frac{\text{HP}}{\text{INCH}^2}$$

# Applications Examples

Specific heat loads of this magnitude will place the performance in a limited life region of 2000-3000 engagements. This may well be acceptable, however, life may be extended by utilizing an AGT-324 which contains 50% more area and might be a desirable alternate. This reduces the specific heat loads accordingly.

$$\text{Specific Heat Loads} = \frac{5037}{1728} @ \frac{1500}{1728} = 2.92 \frac{\text{BTU}}{\text{INCH}^2} @ 0.87 \frac{\text{HP}}{\text{INCH}^2}$$

This heat load level will extend the number of engagements into the 6000-7000 range and may be worth the additional initial expense to reduce the long term maintenance costs.

While this may seem like a limited lining live, it must be remembered that most mills start infrequently and may run continuously for days, weeks, or even months. This could translate into years of service between maintenance if the unit is consistently exposed to these levels of heat loads.

The unit we would select for this application would be the Model AGT-324. The torque service factor is 8.21 and greatly exceeds the recommended service factor of 5.0. This is not unusual and results from the heat loads dictating the unit size.

The clutch pressure during engagement must be controlled to not exceed the allowable motor torque.

$$\text{Eng. Press.} = \frac{\text{ENG. TORQ.}}{\text{DYN. TORQ.}} \times \text{Rated Press.} = \frac{105042}{415200} \times 100 = 25 \text{ PSI}$$

The application calls for two engagements per hour which means the unit must cool sufficiently between engagements so as not to exceed peak surface temperatures which would cause undue distress to occur at the friction interfaces.

Determining these temperature levels is not a straightforward application of theory since "cookbook" thermodynamic formulas are rarely applicable. Accurate temperature calculations can result if the unit has been previously tested over a wide range of thermal loads. This allows the determination of factors that are then used to modify heat transfer coefficients found in classical thermodynamic equations.

Modification factors are investigated in order to establish the following for any particular unit in question:

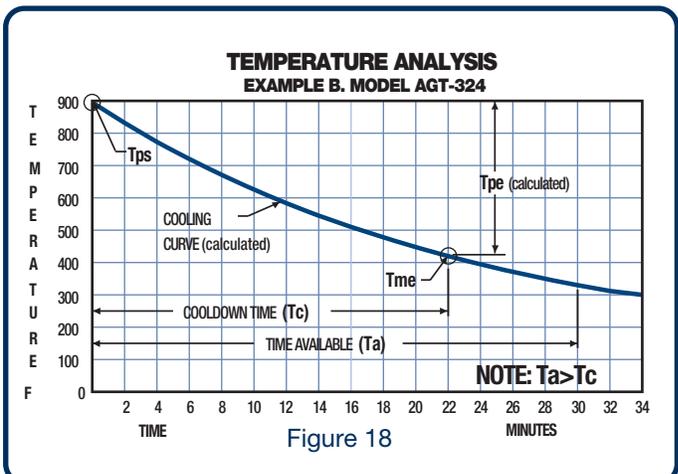
1. Surface temperature rise for various heat rates, engagement times, and interface combinations.
2. The effectiveness of the thermal mass for bulk temperature rise.
3. The time to reach a 90% uniform plate temperature after an engagement. (initial soak temperature)
4. Convective heat transfer coefficients during heating and cooling.
5. The effect of temperature on the modifying factors.

The above approach lets us customize the unit's thermal characteristics to the calculated heat loads with a high degree of certainty that the results will be accurate. The outline of the basic analysis for this application is listed below. The duty cycle calls for an engagement at 30 minute intervals. The effects of uniform soak temperature and bulk temperature rise can be ignored here since these are not important factors at a cyclic rate such as this.

1. Establish the maximum allowable peak surface temperature. **(Tps)** Note: This is a function of the interface (lining/opposing surface) in use.
2. Calculate the peak surface temperature per engagement. **(Tpe)**
3. Calculate the maximum allowable engagement temperature. **(Tme=Tps-Tpe)**
4. Calculate the time to cool from the maximum allowable peak surface temperature to the maximum allowable engagement temperature **(Tc)**. Here it is necessary to know the units' convective heat transfer coefficients under the worst case condition at the maximum ambient temperature encountered to be certain that the unit will cool down sufficiently in the time interval available **(Ta)**.

It is difficult to provide all of the information necessary to allow a user to select a unit based on thermal capacity. The data that would have to be tabulated is lengthy, subject to change as material advancements are made, and may require interpolation to match some applications. The user can tentatively select unit size but must fill out the DATA APPLICATION forms to ensure that the result will be an installation that performs as required.

The results of performing this analysis on the Model AGT-324 clutch for this application are shown graphically in FIGURE 18. These calculations were performed on a computer utilizing modified equations resulting from thermal tests. FIGURE 18 shows that the time available for cooling

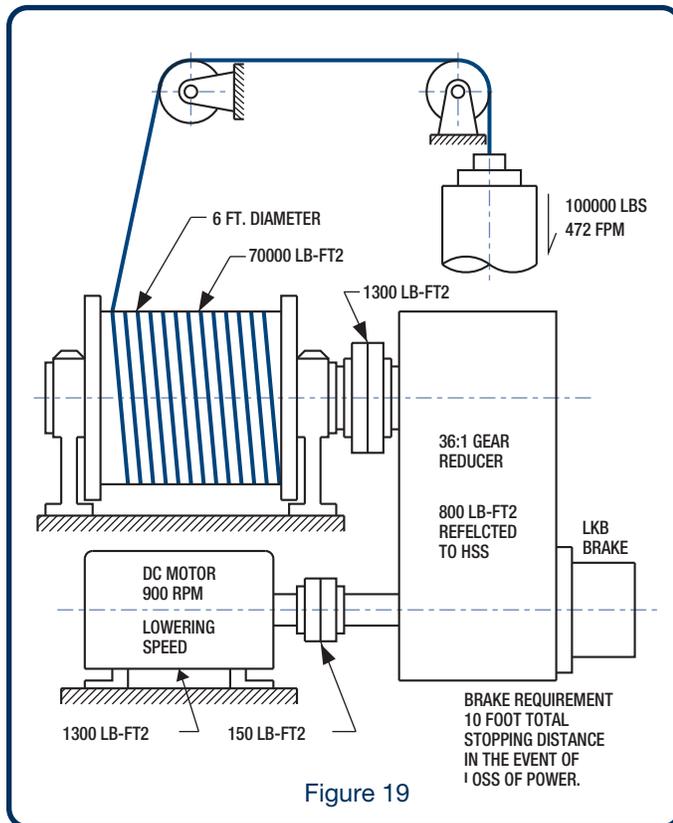


exceeds the actual cooldown time and the Model AGT-324 is suitable for this duty cycle.

**3. Rotational Speed Capacity.** The Model AGT-324 speed capacity of 1100 RPM exceeds the system operating speed of 900 RPM.

## CONTROLLED SOFT-START APPLICATION SUMMARY

Clutch Factors	Performance Factors
Clutch Torque Capacity	■ Drive Capacity
Clutch Heat Capacity	■ Frequency of Engagement
	■ Lining Life
Engagement Pressure	■ Clutch Heat Capacity
	■ Drive Component Stress Levels



### C. Hoist Safety Brake Application

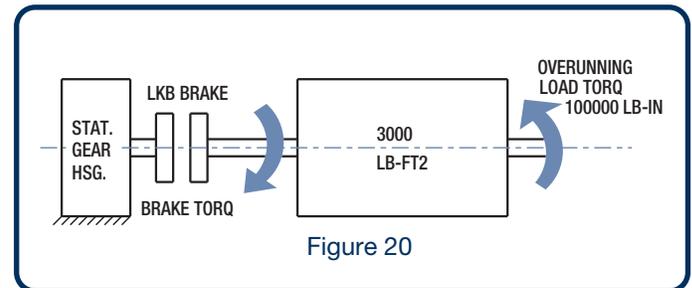
FIGURE 19 describes a geared-hoist arrangement such as may be found on a drilling apparatus or crane. This application incorporates a DC motor which uses regenerative braking to lower the load at a controlled rate.

In the event of loss of power, the DC motor's ability to control the load is lost and a safety brake is set to prevent load runaway. The brakes commonly used in this type of application are spring-set and pneumatically or hydraulically released. For this example, a Model LKB spring-set air released brake will be selected.

The drive specifications call for a maximum stopping distance of 10 feet in the event of loss of power to prevent damage to surrounding structures. Frequency of engagement is based strictly upon the quality and reliability of the machine's power supply since the brake only sets upon loss of machine power. Brake cooldown times will not be investigated in this example. An analysis similar to that made in Example B would be required to define the allowable number of engagements per hour.

This application has an overrunning type load which accelerates the drive system until the brake sets and stops the load. Several important concepts can be demonstrated with this example.

After a power loss, the drive train can be reduced to the single mass system shown in FIGURE 20. The DC motor is not producing any retarding torque since a power loss has occurred. The only torques acting on the system are the load torque (accelerating) and the brake torque (decelerating) the system mass.



**SYSTEM INERTIA** (Reflected to the brake shaft.)

$$\text{Linear WR2} = \frac{(100000)(472^2)}{(39.5)(900^2)} = 695 \text{ LBFT}^2$$

$$\text{Low Speed Shaft WR2} = \frac{70000 + 1300}{36^2} = 55 \text{ LBFT}^2$$

$$\text{Reflected Gear Reducer WR2} = 800 \text{ LBFT}^2$$

$$\text{High Speed Shaft WR2} = 1300 + 150 = 1450 \text{ LBFT}^2$$

$$\text{Total System WR2 (less Brake)} = 3000 \text{ LBFT}^2$$

# Applications Examples

## ACCELERATING LOAD TORQUE (At the brake shaft.)

$$\text{Load Torq} = \frac{\text{Load} \times \text{Drum Radius}}{\text{Gear Ratio}} = \frac{100000 \times 36}{36} = 100000 \text{ LBIN}$$

Net System Torque = Load Torque-Brake Torque

Note the accelerating load torque is a constant superimposed WORK TORQUE that must be overcome by the brake torque before mass deceleration can occur. The NET SYSTEM TORQUE acting on the system mass can be either positive (accelerating) or negative (decelerating) depending upon the magnitudes of the accelerating load torque and the brake torque at any instant of time.

## SYSTEM RESPONSE TIME EFFECTS

The equations in Table 2 are used later in this example to select the proper model LKB brake. First, however, we can use these same equations to generate graphs which can be used to describe how the system performance and brake loads are affected if total system response time is considered.

Total system response time is time required for the brake to develop full torque after a loss of electrical power. This time is composed of several factors.

1. Brake valve relay time.
2. Brake valve shift time.
3. Brake pneumatic or hydraulic response time.

Note that item 3) is a function of the piping arrangement, fittings and elbows; valve flow characteristics; as well as the piping and brake chamber volumes that must be exhausted.

Figure 21 describes the calculated performance for this application as the system response time varies from zero to one-half second. The level of brake torque is adjusted in order to maintain the required 10 foot stopping distance.

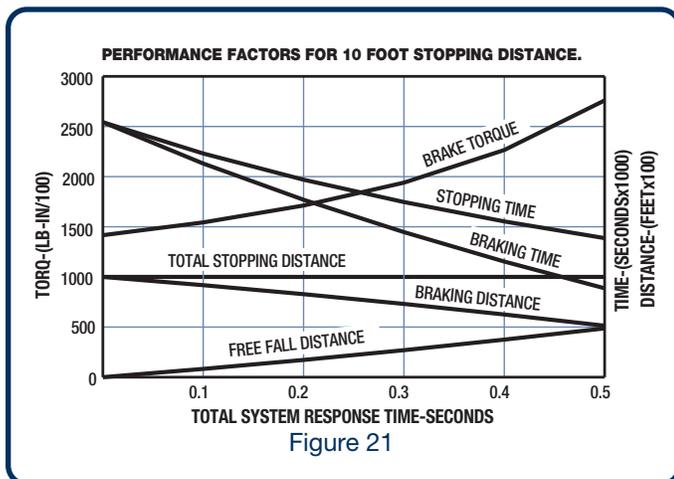


Figure 21

If the system response were zero, meaning the brake sets instantly upon loss of power, the brake torque to stop the load within 10 feet would be 142,000 lb-in. However, if the system response time were one-half second, the brake torque required would be 272,000 lb-in. This is due to the accelerating load torque running unchecked for one-half second which uses up approximately 5 ft. of the available 10 ft. stopping distance during free fall. As a result, the brake has only 5 feet left to stop the load and the brake torque must be increased accordingly.

Therefore, knowledge of the system response time is critical to sizing the brake. Note also that the mechanical components of the drive train must be designed to withstand the braking loads that increase dramatically as the system response time increases.

Figure 22 shows the brake heat load factors as response time increases over this same time period. In this case, the TOTAL ENERGY the brake must dissipate is constant since we are adjusting the brake torque to stop the load within a fixed distance of 10 feet.

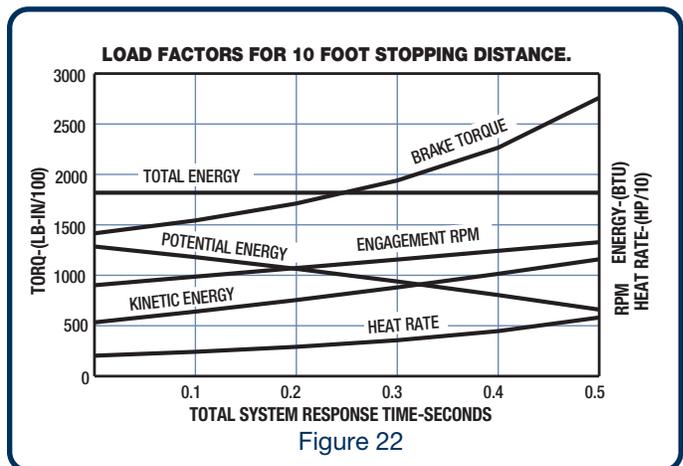


Figure 22

A more subtle brake heat load characteristic becomes evident when the HEAT RATE (the product of the brake torque and the engagement rpm) is considered. Both of these factors increase with increasing system response time. At zero response, the heat rate is 2000 HP, while at one-half second, the heat rate climbs to over 5800 HP. Referring back to Figure 6, it can be noted that as the heat rate increases, the total heat a unit can safely dissipate is diminished.

It can be concluded that the system response time has a great bearing on brake sizing and performance. A one-half second response would not be unusual for a system of this size and must be taken into consideration. This is true for any overrunning type work load that continues to be in

existence after a power loss occurs. The effect of this load must be negated as fast as sound mechanical and control system design will allow in order for the brake to be sized efficiently.

**SYSTEM RESPONSE TIMES IN OVERRUNNING LOAD APPLICATIONS ARE IMPORTANT AND CANNOT BE IGNORED.**

## UNIT SELECTION

A summary of previous calculations can be listed:

- System Mass (less brake) = 3000 LBFT2
- Accelerating load torque = 100000 LBIN
- Initial speed of mass = 900 RPM

The allowable stopping distance is given and a system response time is assumed:

- Allowable stopping distance = 10 feet
- System response time = 0.3 seconds

The following equations are used to calculate the information required to size the brake:

$$\text{Free Fall Speed Change} = \Delta N = \frac{(100000)(0.3)}{(3000)(0.03912)} = 256 \text{ RPM}$$

$$\text{Brake Set Speed} = \text{Initial RPM} + \Delta \text{RPM} = 900 + 256 = 1156 \text{ RPM}$$

$$\text{Average RPM for Free Fall Distance} = \frac{900 + 1156}{2} = 1028 \text{ RPM}$$

$$\text{Free Fall Distance} = \frac{(0.262)(6)(12)(1028)(0.3)}{(36)(60)} = 2.7 \text{ FT}$$

$$\text{Braking Distance} = 10 - 2.7 = 7.3 \text{ FT}$$

$$\text{@ 7.3 FT Drum Payout} = \frac{(360)(7.3)(36)}{(\pi)(6)} = 5019 \text{ Degrees @ Brake}$$

$$\text{Decelerating Torq Req'd} = \frac{(3000)(1156^2)(0.11736)}{(5019)} = 93743 \text{ LBIN}$$

$$\text{Brake Torque} = 100000 + 93743 = 193743 \text{ LBIN}$$

$$\text{Heat Rate} = \frac{(193743)(1156)}{(63025)} = 3554 \text{ HP}$$

$$\text{TE} = \text{KE} + \text{PE} = \frac{(3000)(1156^2)}{(5872.6)(778)} + \frac{(100000)(7.3)}{(778)} = 1816 \text{ BTU}$$

Note that the TOTAL ENERGY the brake must dissipate is the sum of the mass rotational energy (KINETIC ENERGY) and the load positional energy change during braking (POTENTIAL ENERGY).

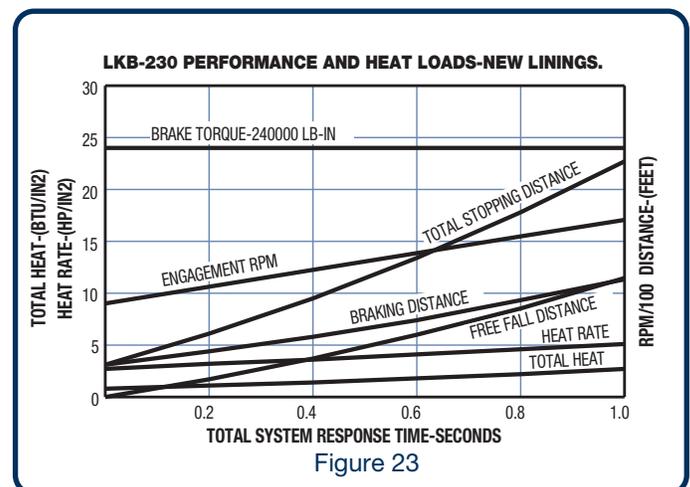
The above can be summarized for a 0.3 second system response time:

$$\text{BRAKE TORQUE REQUIRED} = 193741 \text{ LBIN}$$

$$\text{BRAKE HEAT LOAD} = 1816 \text{ BTU @ 3554 HP}$$

The Model LKB spring-set, air-released brake chosen for this application must develop the torque calculated within 0.3 seconds after loss of power occurs in order to meet the 10 ft stopping distance as well as to have the heat load shown imposed. Once a spring-set brake is selected, the torque of the brake is fixed since it is a function of the spring force used to apply the brake. If the actual response is longer, we do not have the luxury of simply increasing the brake torque to maintain a 10 ft. stopping distance as was done in Figures 21 and 22. The result is that the STOPPING DISTANCE, the TOTAL HEAT, and the HEAT RATE all increase if the response time is greater than 0.3 seconds.

Figure 23 displays the effect of system response time varying from zero to 1.0 seconds on a brake unit having a fixed torque capacity. It should be noted that very little change in response time causes a great deal of change in the stopping distance and brake heat loads.



Another important fact that must be accounted for is the loss of brake torque that occurs as the linings wear. The springs exhibit forces that are directly related to their compressed length. As the linings wear, the spring's compressed lengths change and less actuating force is available. All spring-set units contain this characteristic to some degree. The magnitude of the difference between the NEW and WORN torque is controllable by shimming or using other techniques as the linings wear. It is important to know this difference and size the unit so its worn torque meets the maximum stopping distance allowed.

# Applications Examples

For this example, we'll use a 20% difference between new and worn torque and calculate the results.

WORN BRAKE TORQUE REQUIRED  
= (AS CALCULATED) = 193743 LBIN

NEW BRAKE TORQUE REQUIRED  
= (1.2)(193743) = 232492 LBIN

A unit will be selected that comes closest to meeting these values using available spring sizes.

We will select a Model LKB-230 brake with a new dynamic brake torque of 240000 lb-in and a worn brake torque of 200000 lb-in. The system inertia can now be modified to include the selected brake's inertia. A check of the unit's chamber response time is made to assure that a 0.3 second system response time is achievable provided sound practice is followed in valve and piping selections.

To conclude the procedure, calculations are redone using the selected unit's new and worn torque values in the same manner as before with the results listed below.

BRAKE MODEL .....LKB-230  
SYSTEM INERTIA .....3172 LBFT2  
NEW BRAKE TORQUE .....240000 LBIN  
WORN BRAKE TORQUE .....200000 LBIN  
STOPPING DISTANCE NEW .....7.7 FT.  
STOPPING DISTANCE WORN .....9.8 FT.  
HEAT LOAD NEW .....1.21 BTU/IN2 @ 3.44 HP/IN2  
HEAT LOAD WORN .....1.42 BTU/IN2 @ 2.87 HP/IN2  
MAX ALLOWABLE SPEED .....1600 RPM  
ENGAGEMENT SPEED .....1156 RPM  
CHAMBER RESPONSE TIME .....0.22 SECONDS  
SYSTEM RESPONSE TIME .....0.3 SECONDS

Figures 23 and 24 display the selected unit's calculated heat loads and stopping distances in the new and worn condition

plotted against a variable system response time. The importance of maintaining system response times and brake torques within the original design boundaries cannot be overstressed as is clearly shown by these Figures.

In summary, the machine designer will have to be certain the drive train components can withstand braking loads of this magnitude (240000 lb-in) since it can be noted the normal load torque (100000 lb-in) for the system is much less. It would be prudent to consider what might be done to reduce the brake torque requirement. Factors that will help lower brake torque requirements are: faster system response times; slower lowering speeds; and/or increased stopping distances.

## HOIST SAFETY BRAKE APPLICATION SUMMARY

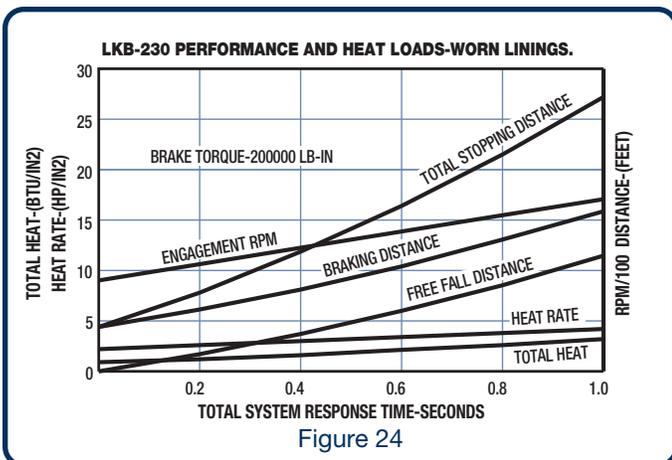
Brake Factors	Performance Effect
Brake Torque Capacity with new and worn linings	<ul style="list-style-type: none"> <li>■ Stopping Distance and Time</li> <li>■ Brake Heat Loads</li> <li>■ Drive Component Stress Levels</li> </ul>
Brake Heat Capacity	<ul style="list-style-type: none"> <li>■ Frequency of Engagement</li> <li>■ Lining Life</li> </ul>
Brake Response Time with new and worn linings	<ul style="list-style-type: none"> <li>■ Stopping Distance and Time</li> <li>■ Brake Heat Loads</li> </ul>

## XIV. MODIFICATIONS

Many applications require some modifications to standard units in order to meet the drive system requirements. Industrial Clutch Products has performed many such modifications to accommodate our customers as well as designed custom units where a standard would not suffice.

Some of these modifications are listed to assist in evaluating their suitability for other applications.

- Higher or lower friction coefficient linings.
- Organic or metallic linings. (Totally non-asbestos.)
- Additional friction plates for design compactness.
- Shims for wear adjustment.
- Wear adjustment features without the use of shims.
- Larger or greater number of ports for increased response.
- Special chamber designs for air and hydraulic operation.
- Air or hydraulic boost chambers on spring-set Models.
- Oversized hub bores.
- Special certified materials for high speed operation.
- Special dynamic balances for high speed operation.



- Special inner and outer member separation features for positive running clearances in high speed operation.
- Special separation feature for vertical operation.
- Special designs for forced air cooling.
- Totally enclosed construction for weather tightness.
- Low temperature operation seals.
- Incorporation of caliper and drum friction surfaces for auxiliary brakes.

**MODIFICATIONS ARE AN IMPORTANT PART OF OUR COMMITMENT TO PROVIDE THE USER WITH A UNIT THAT WILL SATISFY UNIQUE DESIGN REQUIREMENTS.**

## XV. APPLICATION DATA TO BE SUBMITTED TO INDUSTRIAL CLUTCH PRODUCTS FOR UNIT SELECTION.

Please provide the information requested on the forms in this section to allow Industrial Clutch Products to select the proper unit for your application.

Alternately, if the FORMS do not seem suitable, please provide the following:

1. The drive train sketch or schematic.
2. The prime mover power and torque vs. speed characteristics.
3. The work load power and torque vs. speed characteristics.
4. The frictional and windage torques vs. speed characteristics, if available.
5. The load side breakaway torque.
6. The operating duty cycle including the start/stop accuracy and any limitations of the start/stop times/distances the application may require.
7. The drive system safety requirements.
8. Calculate all inertias and reduce the drive system to the four load zone schematic shown in Figure 3. **Note:** Industrial Clutch Products will be pleased to perform these calculations if the proper dimensional data is provided.
9. Submit all data to Industrial Clutch Products.

## DATA APPLICATION FORMS

### PRESS APPLICATION DATA

Press Rated \_\_\_\_\_ Tons @ \_\_\_\_\_ UP BDC  
 Crankshaft Stroke \_\_\_\_\_ Inches  
 Geared \_\_\_\_\_ Non-geared \_\_\_\_\_  
 Wt. Ram & Tooling \_\_\_\_\_ lbs.  
 Motor: \_\_\_\_\_ HP @ \_\_\_\_\_ RPM NEMA \_\_\_\_\_  
 Eddy Current Coupling Used? \_\_\_\_\_  
 Max. Full Load RPM \_\_\_\_\_ Min \_\_\_\_\_  
 Other Type VSD: \_\_\_\_\_  
 Speed Range: \_\_\_\_\_  
 RPM: CL \_\_\_\_\_ BR \_\_\_\_\_ FW \_\_\_\_\_ Crank \_\_\_\_\_  
 Time (Sec): Start \_\_\_\_\_ Stop \_\_\_\_\_  
 Angle (Deg): Start \_\_\_\_\_ Stop \_\_\_\_\_  
 Rated Single Strokes/Min \_\_\_\_\_  
 WR2 (LB-FT2) Ref. to CL/BR. Shaft  
 Input \_\_\_\_\_ Output \_\_\_\_\_  
 Available Air Pressure \_\_\_\_\_ PSIG

Please Provide Sketch of:  
 A. Drive System  
 B. Control System

Other Specifications/Conditions: \_\_\_\_\_  
 \_\_\_\_\_  
 \_\_\_\_\_  
 \_\_\_\_\_

# Data Application Forms

## CONVEYOR APPLICATION DATA

Type:  Downhill  Uphill  Horizontal  
 Horizontal Length \_\_\_\_\_ Ft  
 Vertical Height/Drop \_\_\_\_\_ Ft  
 Tons/Hr \_\_\_\_\_  Metric  Short  
 Prime Mover Type: \_\_\_\_\_  
 AC Motor HP \_\_\_\_\_ @ \_\_\_\_\_ RPM NEMA \_\_\_\_\_  
 Other: VFAC \_\_\_\_\_ ECC \_\_\_\_\_ DC \_\_\_\_\_  
 Speed Range: \_\_\_\_\_ : 1  
 Motor/Pulley Ratio: \_\_\_\_\_ : 1  
 Is this a  Start Load or  Stop Load?  
 Duty Cycle \_\_\_\_\_ Eng. Per \_\_\_\_\_  
 Max Allow Eng. Torque \_\_\_\_\_ LB-FT  
 Max Breakaway Torque \_\_\_\_\_ LB-FT  
 Belt: \_\_\_\_\_ Lbs/Ft \_\_\_\_\_ Inches Wide  
 Total Belt Length: \_\_\_\_\_ Feet  
 Belt Velocity: \_\_\_\_\_ Ft./Min  
 Pulleys: Dia \_\_\_\_\_ In. Wt \_\_\_\_\_ Lbs ea.  
 Quantity: \_\_\_\_\_  
 Idlers: Dia \_\_\_\_\_ In. Wt \_\_\_\_\_ Lbs ea.  
 Quantity: \_\_\_\_\_  
 Clutch Location: \_\_\_\_\_ RPM Shaft  
 Shaft Diameter: \_\_\_\_\_ Inches  
 Quantity: \_\_\_\_\_  
 Brake Location: \_\_\_\_\_ RPM Shaft  
 Shaft Diameter: \_\_\_\_\_ Inches  
 Quantity: \_\_\_\_\_  
 Time (Sec): \_\_\_\_\_ Start \_\_\_\_\_ Stop  
 WR2 (LB-FT<sup>2</sup>) – System Components  
 Motor \_\_\_\_\_ @ \_\_\_\_\_ RPM  
 Motor Coupling \_\_\_\_\_ @ \_\_\_\_\_ RPM  
 Reducer \_\_\_\_\_ @ \_\_\_\_\_ RPM  
 Lbs. Coupling \_\_\_\_\_ @ \_\_\_\_\_ RPM  
 Pulley \_\_\_\_\_ @ \_\_\_\_\_ RPM  
 Idlers \_\_\_\_\_ @ \_\_\_\_\_ RPM  
 Available Air Pressure \_\_\_\_\_ PSIG  
 Please Provide Sketch of:  
 A. Drive System  
 B. Control System  
 Other Specifications/Conditions: \_\_\_\_\_  
 \_\_\_\_\_  
 \_\_\_\_\_

## CONTROLLED START/STOP/DISCONNECT APPLICATION DATA

Application \_\_\_\_\_  
 Prime Mover Type: \_\_\_\_\_  
 PM HP \_\_\_\_\_ @ \_\_\_\_\_ F.L. RPM  
 PM HP \_\_\_\_\_ @ \_\_\_\_\_ Engage RPM  
 Type of Load:  
 HP = K \_\_\_\_\_ Torq = K \_\_\_\_\_ HP = Var \_\_\_\_\_  
 Load BHP \_\_\_\_\_ @ \_\_\_\_\_ Max RPM  
 Load BHP \_\_\_\_\_ @ \_\_\_\_\_ Engage RPM  
 Is this a  Start Load or  Stop Load?  
 Does Unit Engage Under Load?  Yes  No  
 Duty Cycle \_\_\_\_\_ Eng. Per \_\_\_\_\_  
 Clutch or Brake Shaft RPM \_\_\_\_\_  
 Clutch or Brake Shaft Dia. \_\_\_\_\_  
 Time (Sec): Start \_\_\_\_\_ Stop \_\_\_\_\_  
 Angle (Deg): Start \_\_\_\_\_ Stop \_\_\_\_\_  
 Max. Allow Eng. Torque \_\_\_\_\_ LB-FT  
 Max. Breakaway Torque \_\_\_\_\_ LB-FT  
 WR2 (LB-FT<sup>2</sup>) Ref. to Clutch or Brake Shaft  
 Input \_\_\_\_\_ Output \_\_\_\_\_  
 Available Air Pressure \_\_\_\_\_ PSIG  
 Please Provide Sketch of:  
 A. Drive System B. Control System  
 Other Specifications/Conditions: \_\_\_\_\_  
 \_\_\_\_\_  
 \_\_\_\_\_

## ADDITIONAL DATA REQUIRED ALL APPLICATIONS

Customer: \_\_\_\_\_  
 Address: \_\_\_\_\_  
 Contact: \_\_\_\_\_  
 Phone No.: \_\_\_\_\_  
 Fax No.: \_\_\_\_\_  
 Machine Model: \_\_\_\_\_  
 Reference No.: \_\_\_\_\_  
 No. Units This Inquiry: \_\_\_\_\_  
 Response Required By: \_\_\_\_\_  
 Delivery Required By: \_\_\_\_\_  
 Please furnish sketches requested and list any other specifications or conditions for the application.

**TABLE 1 – INERTIAS OF STEEL DISCS**

To determine the WR2 of a given shaft or disc, multiply the WR2 given below, by the length of shaft, or thickness of disc, in inches. To determine inertias of solids of greater diameter than shown below, multiply the nearest tenth of the diameter by 10<sup>4</sup> or move decimal point 4 places to the right

and multiply by length as above. For hollow shafts, subtract WR2 of inside diameter from WR2 of outside diameter and again multiply by length. For materials other than steel, multiply tabular values by the ratio of the materials specific weight to steel, i.e.: (p matl/.283) (WR2steel) (tabular value) = WR2 matl where p matl = lbs./in.<sup>3</sup>

**WR2 PER INCH – STEEL @ .283 LB/IN<sup>3</sup>**

Diameter (Inches)	WR2 (lb. Ft. <sup>2</sup> )	Diameter (Inches)	WR2 (lb. Ft. <sup>2</sup> )	Diameter (Inches)	WR2 (lb. Ft. <sup>2</sup> )	Diameter (Inches)	WR2 (lb. Ft. <sup>2</sup> )
3/4	.00006	7-3/4	0.699	14-1/2	8.54	40	492.8
1	.0002	8	0.791	14-3/4	9.15	41	543.9
1-1/4	.0005	8-1/4	0.895	15	9.75	42	598.8
1-1/2	.001	8-1/2	1.00	16	12.61	43	658.1
1-3/4	.002	8-3/4	1.13	17	16.07	44	721.4
2	.003	9	1.27	18	20.21	45	789.3
2-1/4	.005	9-1/4	1.41	19	25.08	46	861.8
2-1/2	.008	9-1/2	1.55	20	30.79	47	939.3
2-3/4	.011	9-3/4	1.75	21	37.43	48	1021.8
3	.016	10	1.93	22	45.09	49	1109.6
3-1/4	.022	10-1/4	2.13	23	53.87	50	1203.1
3-1/2	0.029	10-1/2	2.35	24	63.86	51	1302.2
3-3/4	0.038	10-3/4	2.58	25	75.19	52	1407.4
4	0.049	11	2.83	26	87.96	53	1518.8
4-1/4	0.063	11-1/4	3.09	27	102.30	54	1636.7
4-1/2	0.079	11-1/2	3.38	28	118.31	55	1761.4
5	0.120	11-3/4	3.68	29	136.14	56	1898.1
5-1/4	.146	12	4.00	30	155.92	57	2031.9
5-1/2	0.177	12-1/4	4.35	31	177.77	58	2178.3
5-3/4	.211	12-1/2	4.72	32	201.8	59	2332.5
6	0.250	12-3/4	5.11	33	228.2	60	2494.7
6-1/4	0.296	13	5.58	34	257.2	66	3652.5
6-1/2	0.345	13-1/4	5.96	35	288.8	72	5172
6-3/4	0.402	13-1/2	6.42	36	323.2	78	7125
7	0.464	13-3/4	6.91	37	360.7	84	9584
7-1/4	0.535	14	7.42	38	401.3	90	12629
7-1/2	0.611	14-1/4	7.97	39	445.3	96	16349

# Equations

**TABLE 2 – EQUATIONS**

$$T_1 = \frac{(HP) (63025)}{(N)}$$

$$T_2 = (F) (R)$$

$$T_3 = (L) (2000) \sqrt{(H) (S - H)}$$

$$t = \frac{(WR^2) (N) (.03912)}{(T)}$$

$$\theta = \frac{(WR^2) (N^2) (.11736)}{(T)}$$

$$\theta = (3) (N) (t)$$

$$KE = \frac{(WR^2) (N^2)}{(5872.6)}$$

$$PE = (W) (d)$$

$$V = (.262) (D) (N)$$

$$WR_1^2 = WR_2^2 \left( \frac{N_2}{N_1} \right)^2$$

$$WR_1^2 = \frac{(W) (V^2)}{(39.5) (N_1)}$$

$$\%SD = \left[ 1 - \sqrt{\left( 1 - \frac{2 \times WR_{2OUTPUT}^2}{WR_{1INPUT}^2} \right)} \right] 100$$

$$KF = \frac{(N)}{\sqrt{\left( \frac{CE}{WR_{ENGAGED}^2} \right)}}$$

$$SF = \frac{FPD}{MT}$$

$$CE = (L) (166.7) (FPD)$$

$$CH = (KE) (CPM)$$

$$CHP = CE + (2 \times CH)$$

$$HR = \frac{(T_4) (N_I - N_O)}{(63025)}$$

$$TH = \frac{(T_4) (N_I - N_O) (ts)}{178305}$$

Symbol	Description	Units
<b>T1</b>	PRIME MOVER OR LOAD TORQUE	LB-IN
<b>T2</b>	DRUM, PULLEY, OR SHEAVE TORQUE	LB-IN
<b>T3</b>	PRESS CRANKSHAFT TORQUE	LB-IN
<b>T4</b>	CLUTCH OR BRAKE TORQUE	LB-IN
<b>HP</b>	POWER	HORSEPOWER
<b>N</b>	SPEED	RPM
<b>N1</b>	SPEED AT LOCATION ONE	RPM
<b>N2</b>	SPEED AT LOCATION TWO	RPM
<b>NI</b>	INPUT SPEED	RPM
<b>NO</b>	OUTPUT SPEED	RPM
<b>F</b>	LINE OR BELT PULL	LBS
<b>R</b>	DRUM, PULLEY, OR SHEAVE RADIUS	INCH
<b>L</b>	PRESS CAPACITY	TONS
<b>H</b>	PRESS RATED DISTANCE FROM BOTTOM	INCH
<b>S</b>	PRESS STROKE	INCH
<b>WR2</b>	INERTIA	LB-FT <sup>2</sup>
<b>WR21</b>	INERTIA AT LOCATION ONE	LB-FT <sup>2</sup>
<b>WR22</b>	INERTIA AT LOCATION TWO	LB-FT <sup>2</sup>
<b>θ</b>	START OR STOP ANGLE	DEGREES
<b>t</b>	START OF STOP TIME	SECONDS
<b>ts</b>	SLIP PERIOD	SECONDS
<b>KE</b>	ROTATIONAL ENERGY	FL-LBS
<b>PE</b>	POTENTIAL ENERGY	FT-LBS
<b>V</b>	VELOCITY	FT/MIN
<b>D</b>	DRUM, PULLEY, SHEAVE, OR DISC DIA.	INCH
<b>W</b>	WEIGHT OF MOVING COMPONENTS	LBS
<b>SD</b>	TWO MASS ENGAGEMENT SLOWDOWN	%
<b>KF</b>	TWO MASS SLOWDOWN FACTOR	FIGURE 14
<b>SF</b>	SHEAR FACTOR	NONE
<b>FPD</b>	MATERIAL FRACTURE POINT DEPTH	INCH
<b>MT</b>	MATERIAL THICKNESS	INCH
<b>CE</b>	CUT ENERGY	FT-LBS
<b>CH</b>	INERTIA CYCLING HEAT	FT-LBS
<b>CPM</b>	HIT RATE	CYCLES/MIN
<b>CHP</b>	PRESS CYCLING POWER	HORSEPOWER
<b>HR</b>	CLUTCH OR BRAKE HEAT RATE	HORSEPOWER
<b>TH</b>	CLUTCH OR BRAKE TOTAL HEAT	BTU

# Service and Conversion Factors

**TABLE 3 – STATIC TORQUE SERVICE FACTORS** (Refer to section XII for use)

Industry	Machinery	Model (Service Factor)	
<b>LOGGING &amp; LUMBER</b>	CHIPPERS	AGT(3.5)	
	DE-BARKERS	AGT(3.0)	
	YARDER DRUMS	AGT(2.5) HBA(1.4)	
<b>MARINE</b>	MAIN PROPULSION (CP)	LKM(2.0) HC(1.8)	
	MAIN PROPULSION (F/R)	LKM(2.5) HC(2.0)	
	SHAFT BRAKE	LK(0.7) HBA(0.9)	
	WINCHES	LK,LKM(2.0) HC(1.5)	
<b>MATERIAL HANDLING</b>	CONVEYORS UNIFORMLY LOADED	AGT(1.8) LK, LKB(1.5)	
	RECIPROCATING	AGT(2.2) LK(2.0) LKB(1.8)	
	VIBRATORY	AGT(3.0) LK(2.5) LKB(2.0)	
<b>METAL-FORMING</b>	PRESSES, SHEARS, AND PRESS BRAKES	AGT(2.5) LK,LKT(2.0) CBA(1.3) LKB(1.3) CBH(1.3)	
	<b>MILLS</b>	AUTOGENOUS	AGT(6.0) LK(5.0) HC(4.0)
		BALL/ROD/GRINDING	AGT(5.0) LK(4.0) HC(3.0)
HAMMER/CRUSHING		AGT(6.0) LK(5.0) HC(4.0)	
ROLLING		AGT(3.0) LK(2.5) HC(2.0)	
RUBBER		AGT(3.0) LK(2.5) HC(2.0)	
SUGAR		AGT(3.0) LK(2.5) HC(2.0)	
<b>MINING (SHOVELS &amp; CRANES)</b>	HOIST, DRAG, CROWD	LKB(1.4) HBS(1.2)	
	SWING	LKB(1.3) HBS(1.1)	
	PROPEL	LK(1.7) HC(1.5)	
<b>PAPER</b>	CALENDAR	AGT(5.0) LK(4.5) HC(4.0)	
	DRYER	AGT(6.0) LK(5.5) HC(5.0)	
	FOURDRINIER	AGT(5.0) LK(4.5) HC(4.0)	
	PRESS	AGT(5.0) LK(4.5) HC(4.0)	
	REEL	AGT(3.0) LK(2.5) HC(2.0)	
<b>PETROLEUM (DRILLING &amp; SERVICING)</b>	CAT HEAD	AGT(4.5) LK/LKT(4.0) HC(3.5)	
	HOIST DRUMS	AGT(3.5) LK/LKT(2.8) LKB(2.5)	
	PUMPS — CENTRIFUGAL	AGT(2.5) LK/LKT(2.0) HC(1.8)	
	MUD/SLUSH SAND REEL	AGT(5.0) LK/LKT(4.5) HC(4.0) AGT(3.0) LK/LKT(2.5) HC(2.0)	
<b>PUMPING</b>	PUMPS/COMPRESSORS		
	AXIAL	AGT(3.0) LK/LKT(2.5) HC(2.0)	
	CENTRIFUGAL RECIPROCATING	AGT(2.5) LK/LKT(2.0) HC(1.8) AGT(4.0) LK/LKT(3.0) HC(2.5)	

**TABLE 4 – CONVERSION FACTORS**

TO CONVERT...	INTO...	MULTIPLY BY ...
ATMOSPHERES	LBS/IN2	.....14.7
ATMOSPHERES	KG/CM2	.....1.0333
BARS	ATMOSPHERES	.....0.987
BARS	LBS/IN2	.....14.5
BTU	FT-LBS	.....778.3
BTU	JOULES	.....1055
BTU/MIN	HP	.....0.02356
BTU/MIN	KW	.....0.01757
CENTIMETERS (CM)	INCHES	.....0.3937
DEGREES	RADIANS	.....0.01745
FEET (FT)	INCHES	.....12
FEET	METERS	.....0.3048
FT-LBS	BTU	.....0.001285
GALLONS (US)	IN3	.....231
GALLONS (IMP)	IN3	.....277.42
GALLONS (IMP)	GALLONS (US)	.....1.2
HORSEPOWER (HP)	BTU/MIN	.....42.44
HP	FT-LBS/MIN	.....33000
HP	FT-LBS/SEC	.....550
HP	KW	.....0.7457
HP	WATTS	.....745.7
JOULES	BTU	.....0.000948
KILOGRAMS (KG)	LBS	.....2.2046
KG/CM2	LBS/IN2	.....14.2248
KG-M2 (GD2)	LB-FT2 (WR2)	.....5.93
KG-M2 (WR2)	LB-FT2 (WR2)	.....23.73
KG-M	LB-IN	.....86.8
KILOWATTS (KW)	HP	.....1.341
KW	BTU/MIN	.....56.92
LB-FT2 (GD2)	LB-FT2 (WR2)	.....4
LB-FT	NM	.....1.35575
LB-IN	NM	.....0.11302
LBS/IN2	BARS	.....0.06897
LBS/IN2	KG/CM2	.....0.0703
LITERS	GALLONS (US)	.....0.2642
LITERS	GALLONS (IMP)	.....0.2201
METERS (M)	FEET	.....3.281
MILLIMETERS (MM)	INCHES	.....0.03937
NEWTONS (N)	LBS	.....0.2248
NM	LB-IN	.....8.848
NM	LB-FT	.....0.7376
POUNDS (LBS)	KG	.....0.4536
RADIANS	DEGREES	.....57.296
TONS (METRIC)	TONS (SHORT)	.....1.103
WATTS	HP	.....0.001341

## Oil Immersed Clutches & Brakes



**Pneumatic / Hydraulic, Actuated Clutches, Brakes, Controls & Hydraulics**



# Oil Immersed Clutches & Brakes

Oil immersed clutches and brakes have many desirable operating characteristics due to the manner in which engagements and the removal of engagement heat are accomplished.

## ENGAGEMENT SEQUENCE

The method by which friction is generated in an oil clutch or brake is threefold. At the point of initial engagement, where a high relative speed exists, the oil film between the rotating discs shears as the film thickness is compressed. The shearing of this oil film manifests itself as a frictional couple and assists in transmitting load torque. This occurs during what is referred to as the **full fluid film mode**.

As the engagement sequence continues, the oil film starts to break down and friction is now generated partially by the shearing of the oil film and partially by direct contact of the mating parts. This occurs during the **mixed film mode** of the engagement cycle.

Finally, as the engagement nears completion, the oil film collapses completely and friction is generated by coulomb or static friction where the plates are in complete contact. This phase of the cycle is called the **static film mode** and completes the engagement sequence.

## ENGAGEMENT HEAT

The heat of engagement is transmitted to both the oil and to the mating members of an oil immersed clutch or brake. The proportional amount of heat transmitted into the oil and into the mating members is a function of the engagement slip time. Shorter engagements require that more of the heat be stored in the mating members. Longer engagements allow the heat to be effectively transferred to the oil and transmitted away from the working surfaces of the disc pack.

Oil immersed units may be splash cooled or force cooled by introducing a continuous flow of oil into the center of the disc pack. Industrial Clutch Products manufactures units for heavy-duty service and all of our units contain the force cooled feature.

Grooving patterns on the friction disc surfaces allow this oil to be distributed in the most effective manner for cooling during engagement as well as provides a path for continuous cooling oil flows through-out the disc pack after the disc pack is engaged.

In this manner, large amounts of engagement heat can be carried away by the circulating oil for removal either through the housing to the surrounding atmosphere, or to an external cooler.

## SPEED OF ENGAGEMENT

Oil units can engage at higher speeds and not exhibit the wear magnitudes or heat checking found in dry clutches and brakes. The explanations for this are found in the oil film shearing portion of the engagement cycle where the highest heat rates are generated.

## DISC PACK DESIGN ARRANGEMENT

Oil clutches and brakes contain more working surfaces than dry units. This is due to the friction coefficients for the wet units being much lower than those in dry clutches and brakes. The solution for design compactness is to stack more discs in wet units and allow the forced oil cooling to evenly remove the heat generated during engagement from each working surface.

## DISC PACK PLATE DESIGN

The discs in oil clutches and brakes are generally much thinner than those found in dry units. There are two reasons for this. One is that with the larger number of discs and oil cooling, more volume exists to absorb the heat of engagement. The second is that the same oil that cools the disc's working surfaces also lubricates the splines that carry the torque loads which increases their capacity significantly.

We carry this one step further and design our disc packs with extra-heavy plate thicknesses since our units must function in the most demanding applications. Each plate size has been designed with **torque** and **heat** capacity in mind since it is not possible to ignore the types of loads these units encounter.

## TORQUE TO INERTIA RATIOS

Extremely attractive compactness can be achieved in a properly designed oil clutch or brake. The torque capacity of any disc style clutch or brake is directly proportional to the disc diameter; the coefficient of friction; the number of discs; and the apply force (unit pressure) on the linings.

As was stated, the coefficient of friction is lower in an oil unit than in a dry unit. Increasing the number of equally sized discs, in the same proportion as the friction ratio change, would yield a disc pack of identical torque capacity. However, it is also possible to increase the apply force (unit pressure) on a wet disc pack which allows us to further increase the torque capacity without increasing the disc diameter. We can therefore carry higher torque capacities within a wet disc pack when compared to a dry pack of equal diameters.

# Oil Immersed Clutches & Brakes

With the greater quantity of discs found in a wet disc pack, the disc pack may be longer, however, the torque capacity is higher. Inertia increases with the fourth power of diameter and only directly with length so it is possible to achieve higher torque to inertia ratios with wet units than with those operating dry.

High torque to inertia ratios are desirable in any application requiring compactness, speed of response, and thermally efficient cyclic operation.

## WET VS. DRY

Industrial Clutch Products manufactures a complete line of both wet and dry units.

There are many applications where dry units have been used successfully and, because of machine performance upgrades or re-designs, we have applied wet units.

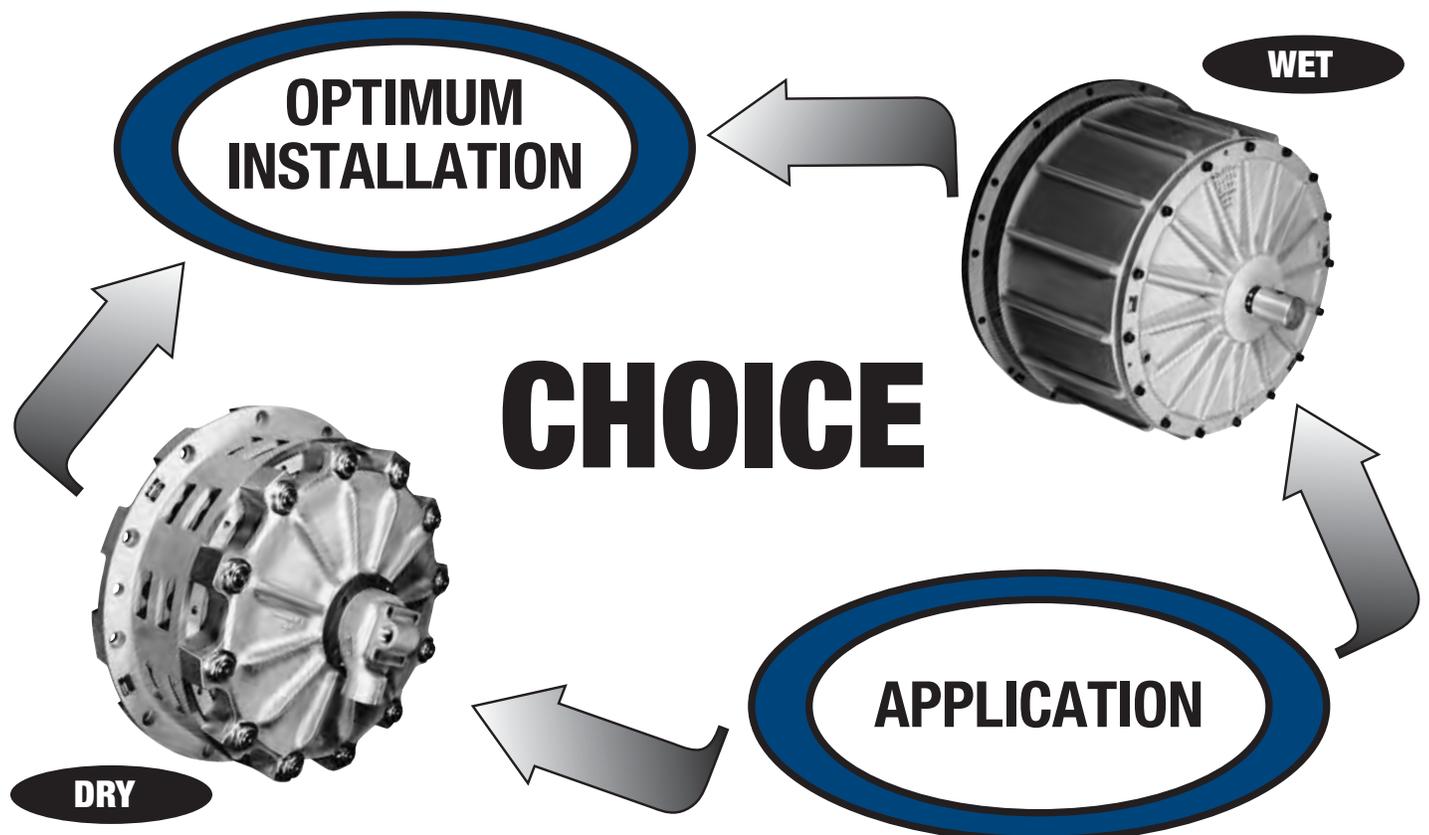
The wet units have repeatedly demonstrated their superior ability to operate with greater longevity than their dry counterparts by a factor of five to ten times, even though in many cases, the loads have increased significantly.

Even so, wet units are not applicable for all applications. It is not possible to compete with the simplicity, ruggedness, and economy our dry units offer for a majority of the marketplace.

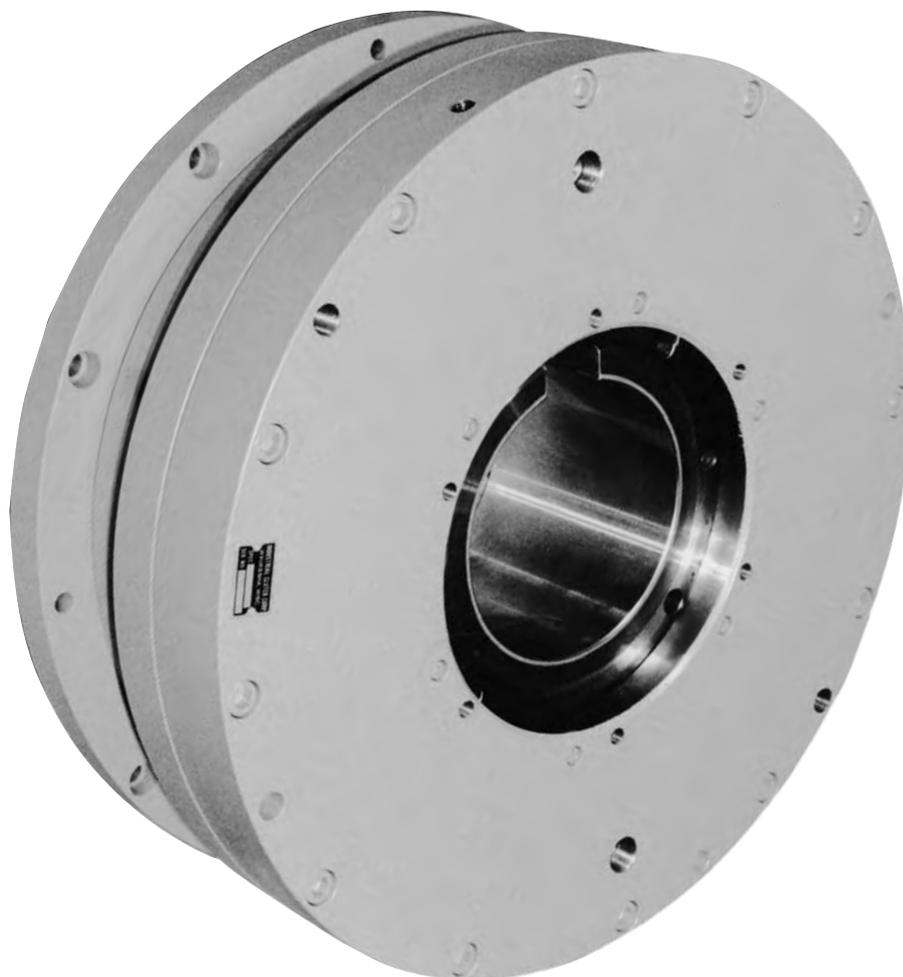
The oil-immersed clutches and brakes offer another viable choice when unusually tough and demanding applications are encountered.

Some of these applications have evolved to the point where dry units are now obsolete since users are accustomed to the performance demonstrated by oil-immersed clutches and brakes.

Industrial Clutch Products has designed their oil-immersed clutches and brakes to complement their dry units. In doing so, we are able to offer a wide range of choice to the builder and user of today's most sophisticated equipment.







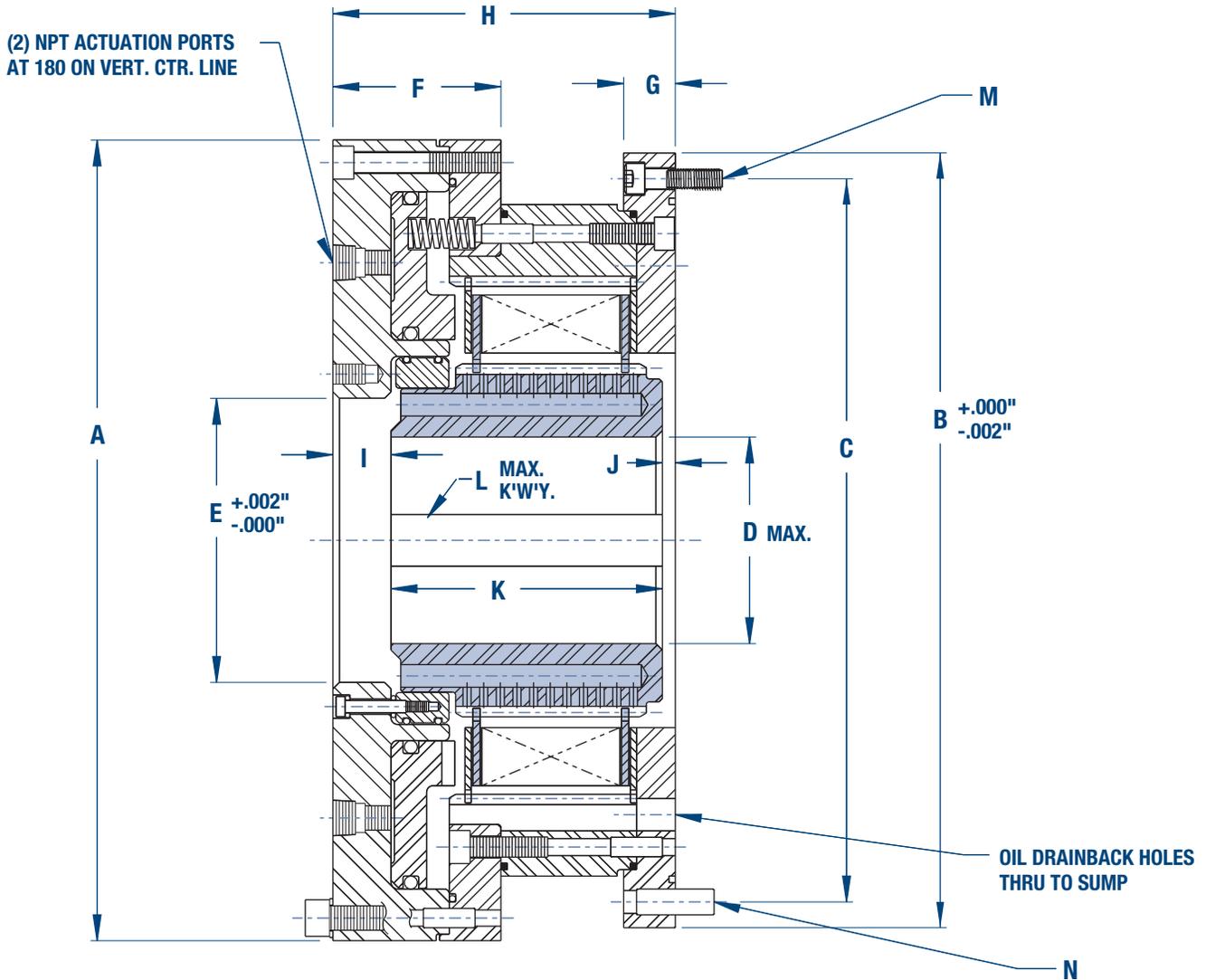
**MODEL HBA**

*Air or Hydraulically Actuated Brakes*

**MODEL HBS**

*Spring-Set Air or Hydraulically Released*

# Model HBA



## DIMENSIONAL DATA

*All dimensions in inches*

Brake Model	A	B	C	D	E	F	G	H	I	J	K	L	M	N
HBA-8	13.25	12.875	12.000	3.000	4.000	2.63	0.84	5.88	0.75	0.19	4.94	0.75	(8) - M8	(4) - .375
HBA-10	15.50	15.000	14.000	4.000	5.500	3.25	1.00	6.63	1.13	0.25	5.25	1.00	(8) - M10	(4) - .500
HBA-13	21.25	19.750	18.750	6.000	8.750	3.50	1.13	7.31	1.19	0.13	6.00	1.50	(8) - M12	(4) - .500
HBA-15	24.50	23.500	22.000	6.500	8.750	4.13	1.38	9.31	1.25	0.25	7.81	1.50	(8) - M16	(4) - .625
HBA-16	27.50	25.500	24.000	7.500	10.000	4.25	1.38	9.44	1.38	0.25	7.81	1.50	(8) - M16	(4) - .750
HBA-18	30.25	28.750	27.000	8.000	11.000	4.69	1.75	11.84	1.56	0.50	9.78	1.50	(8) - M20	(4) - 1.00
HBA-20	32.25	31.000	29.250	9.000	12.000	5.06	1.88	12.28	1.63	0.63	10.03	1.50	(8) - M20	(4) - 1.00

NOTES: 1.) Use certified drawing dimensions only for final layouts.

2.) DXF and IGES files available upon request.

3.) Dimensions subject to change without notice.

4.) All threaded fasteners are metric.

## OPERATIONAL DATA

Brake Model	Static Torque (lb.-in.)	Dynamic Torque (lb.-in.)	Act. Vol. (in. <sup>3</sup> )	Weight Outer (lbs.)	Weight Inner (lbs.)	WR <sup>2</sup> Inner (lb.-ft. <sup>2</sup> )	Maximum RPM
HBA-8	63,000	45,000	10.6	148.6	29.5	1.43	4,000
HBA-10	105,000	75,000	19.2	227.7	48.2	3.48	3,400
HBA-13	280,000	200,000	37.7	411.0	106.0	15.75	2,400
HBA-15	420,000	300,000	50.0	719.0	152.0	28.68	2,150
HBA-16	595,000	425,000	66.0	900.0	197.0	45.30	1,900
HBA-18	770,000	550,000	97.0	1,385.0	308.0	84.80	1,750
HBA-20	1,035,000	740,000	135.0	1,655.0	356.0	119.40	1,600

**NOTES:** 1.) Operating pressure: 100 PSIG Hydraulic actuation available  
 2.) Torque capacities can be modified. Consult engineering.  
 3.) Consult factory for service factors required per application.

4.) Consult factory for oil flow requirements per application and shaft oil hole diameters.

## MODEL HBA BRAKE DESCRIPTION

The Model HBA brake is an oil immersed, air or hydraulically set brake designed to be mounted on a drive train shaft extension. In the majority of installations, the Model HBA brake utilizes the system's lubricating oil for cooling.

Cooling oil is fed into the brake at the cylinder end of the unit where it is directed into the oil distribution holes incorporated within the rotating hub for dispersal through-out the disc pack.

Unique friction plate grooving patterns, as well as the forced oil cooling feature, allow the Model HBA brake to dissipate the heat generated by stopping the high energy loads common to today's heavy duty industrial equipment.

All torque transmitting members are designed to withstand the rigors of the loads imposed. The unit incorporates high quality, time proven materials selected to provide strength, durability, and reliable performance.

The Model HBA housing contains the stationary plates of the disc pack. This housing is bolted to the stationary machine member and provides the grounding path for stopping the rotating components of the drive train when the disc pack is engaged. The housing also serves as a containment for the cooling oil and provides a path for the oil to return to the machine reservoir.

## MODEL HBS BRAKE DESCRIPTION

The Model HBS brake is identical to the Model HBA brake except the unit is designed to be spring-set and hydraulically or pneumatically released. This spring-set actuation feature makes it suitable for those applications requiring the presence of full braking torque in the event of loss of actuation pressure or machine power.

## USAGES

- Any type of industrial equipment that requires braking of high speed-high energy loads with maximum reliability and longevity.
- Those applications that must operate in adverse environments due to the Model HBA and HBS brake's totally enclosed features.
- Tensioning control systems.

Propeller shaft brakes, winch holding brakes, mining safety brakes (Model HBS), conveyors, lumber and paper making equipment are some examples of where these brakes can be applied. They are particularly suited to mounting directly to gear cases. This allows sharing of the lubrication and cooling system as well as provides system compactness and a professional design appearance.