

Pneumatic and Hydraulic Clutches and Brakes



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.

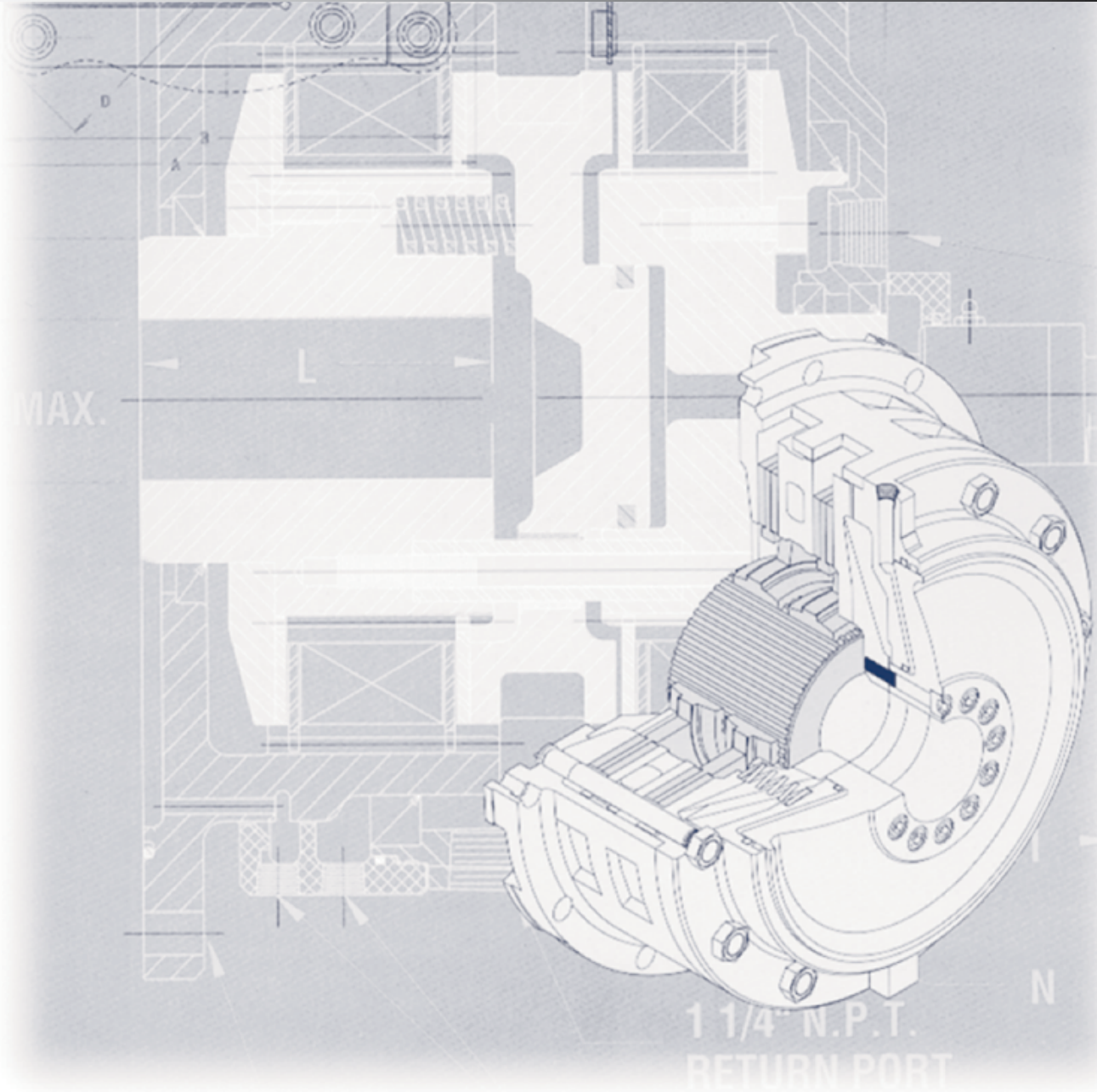
This warranty shall not apply to any product that has been repaired or altered without the manufacturer's knowledge and consent. This warranty also does not cover damage or defects resulting from neglect, improper installation, improper maintenance or operation, or accidents.

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



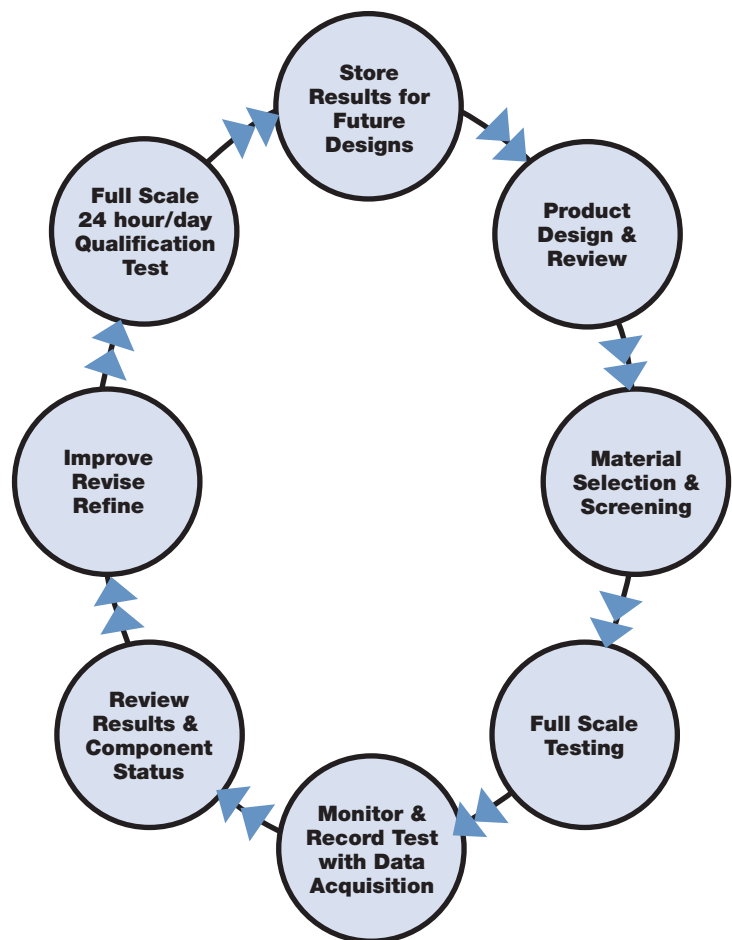
The Company

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



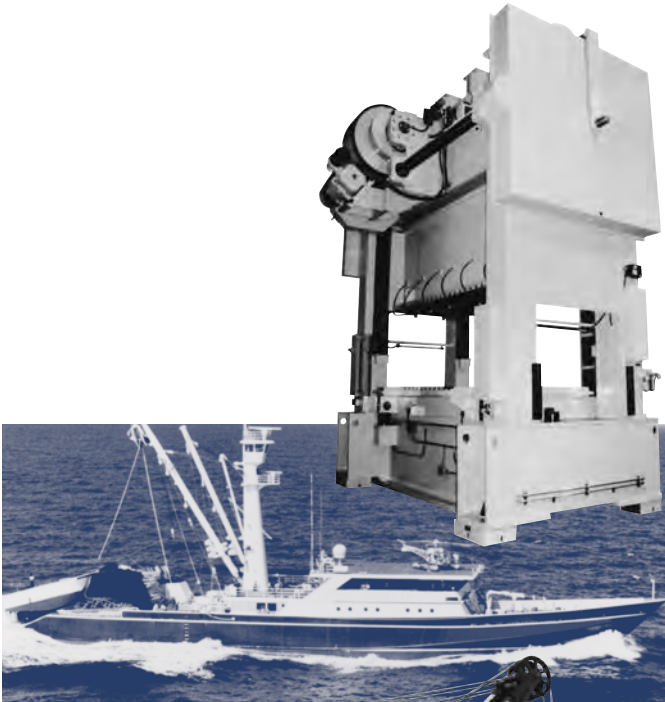
MODEL CBH



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.

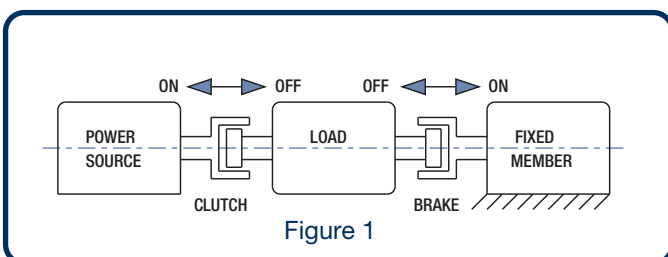
If the **DATA APPLICATION** forms listed in Section XV do not seem suitable for your application, contact our Engineering Department for additional assistance.

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:

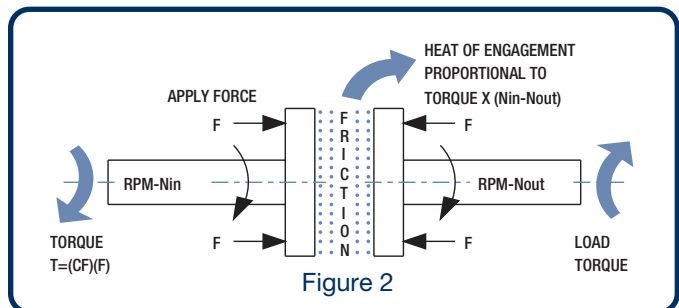
1. Transmits the normal and peak load torques with a suitable service factor.

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.

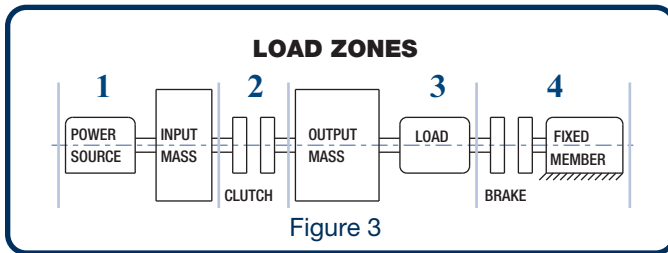


During engagement, the magnitude of the clutch or brake torque multiplied by the speed differential over the engagement time period is proportional to power. This manifests itself as **HEAT**. The clutch or brake unit must successfully dissipate this heat in accordance with the frequency of engagement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

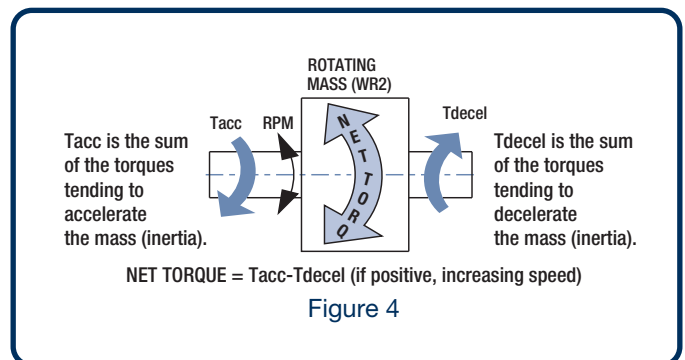
IV. CLUTCHES & BRAKES – DYNAMICS OF MASS MOTION

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
 α 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-ft²). These are familiar and easily calculated or given values.

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

Where: α is the acceleration rate (rad/sec²)
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-ft²)
g is the conversion constant (32.2 ft/sec²)

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.

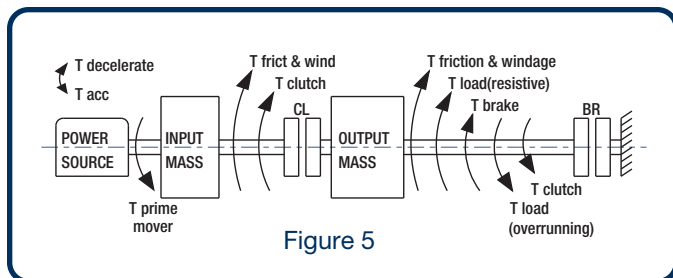


Figure 5

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.

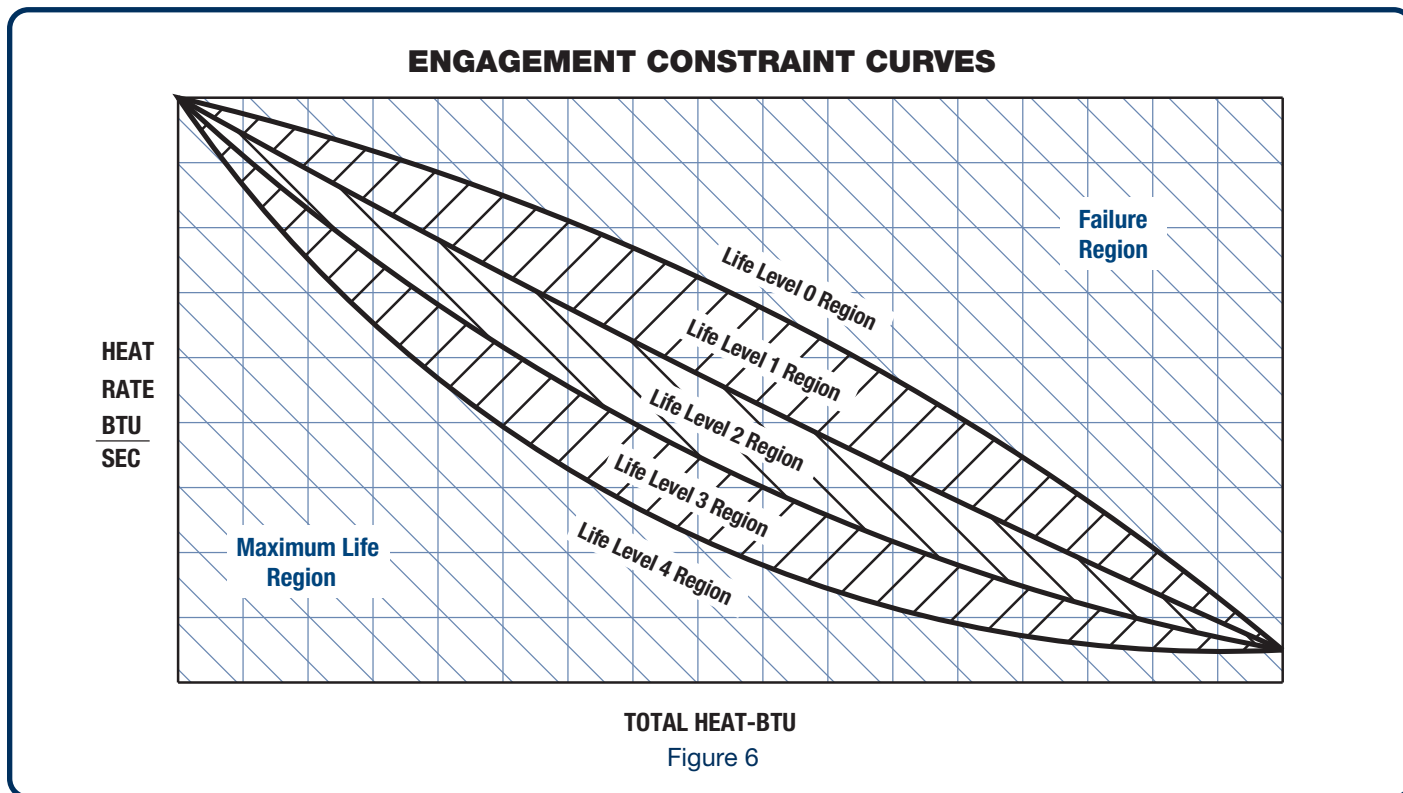
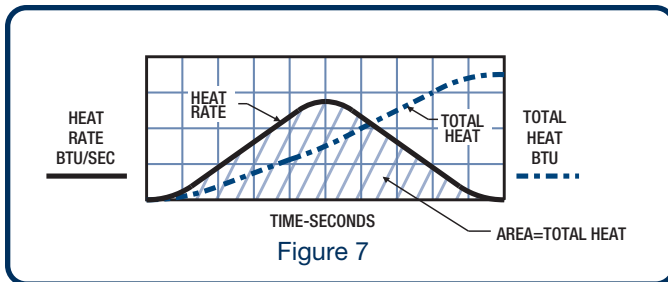


Figure 6

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.

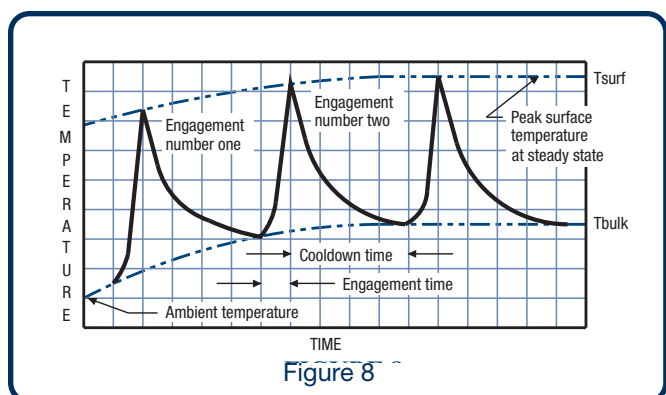


FIGURE 8 shows a typical time vs. temperature chart that could occur in an industrial application. The rise in temperature followed by the exponential decay of temperature is classic for clutches and brakes. The primary aim of Industrial Clutch Products is to utilize the latest technology, materials, 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

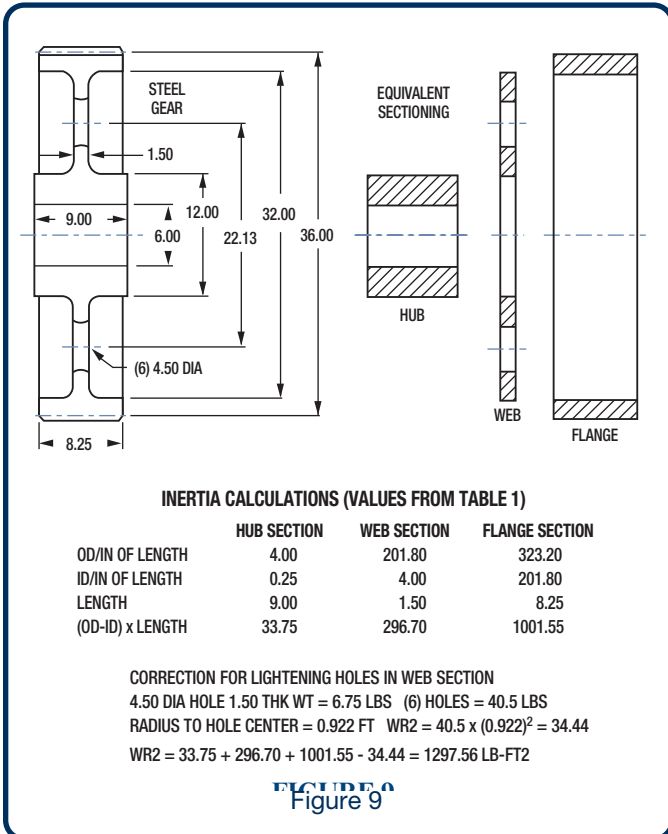
Reducing a drive train to an equivalent system, such as shown in FIGURE 3, requires an understanding of how inertia is calculated and how it may be reflected to different locations within the drive train.

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.



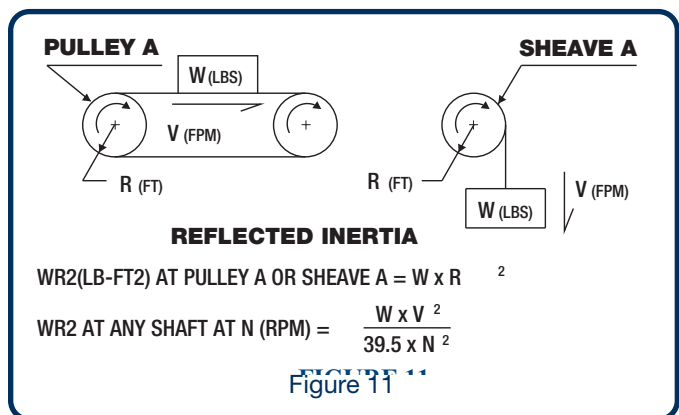
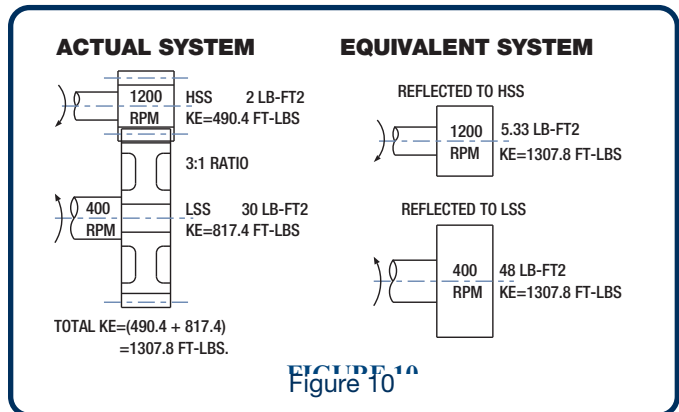
B. Reflecting inertia

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)
 WR² is inertia (lb-ft²)
 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

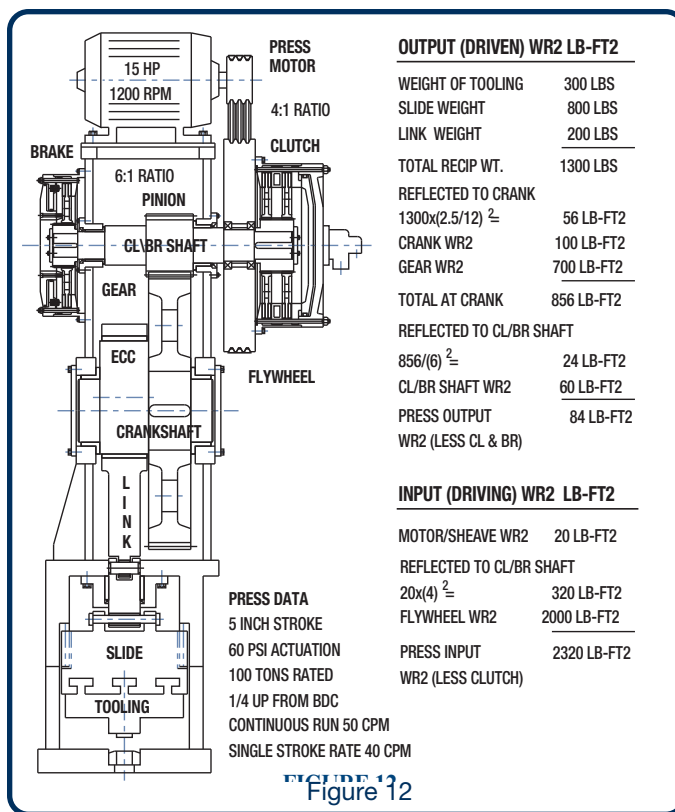
SERVICE FACTOR CATEGORIES

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.



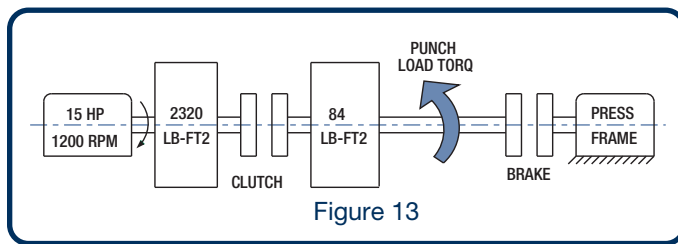
XIII. APPLICATION EXAMPLES

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. (i.e., 50% penetration is 1/2 material thickness and SF = 0.5)

Application Examples

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

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

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

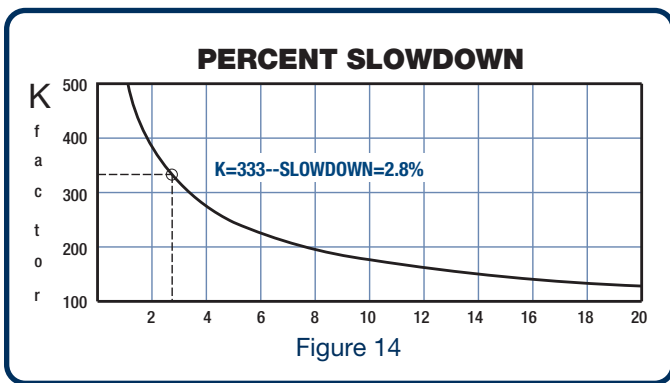


FIGURE 14 may be used to obtain the press slowdown when punching by selecting the appropriate K factor.

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

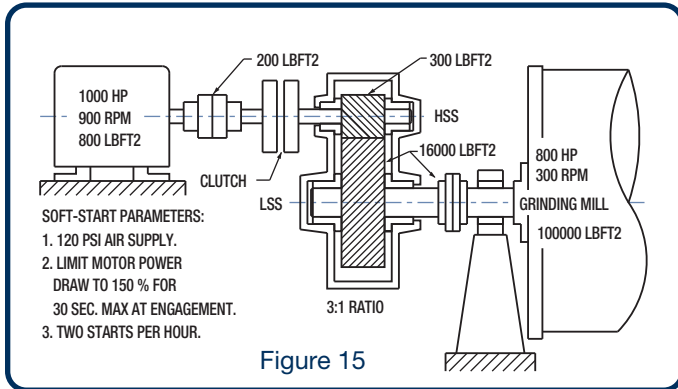


Figure 15

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

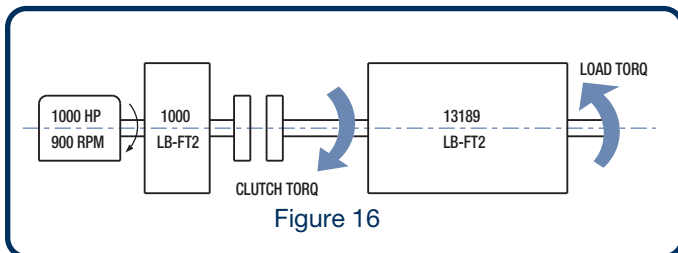


Figure 16

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:

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

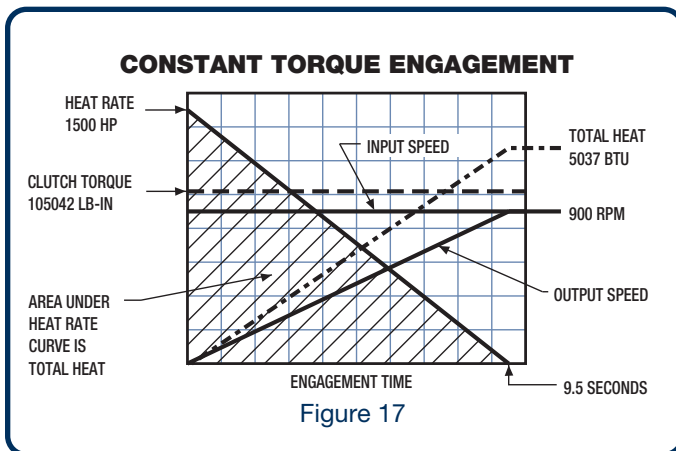
TOTAL HEAT is equal to the area under the heat rate curve for any engagement. Here the heat rate curve is triangular in shape with the maximum heat rate occurring at time t=0 and diminishing to zero at time t=9.5 seconds. The area of a triangle is 1/2(base;time)x(height;heat rate) so, in this case, the 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:

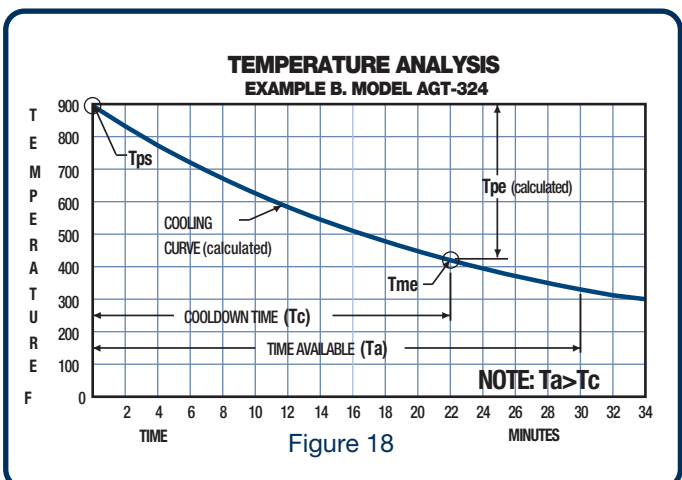
1. Surface temperature rise for various heat rates, engagement times, and interface combinations.
2. The effectiveness of the thermal mass for bulk temperature rise.
3. The time to reach a 90% uniform plate temperature after an engagement. (initial soak temperature)
4. Convective heat transfer coefficients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

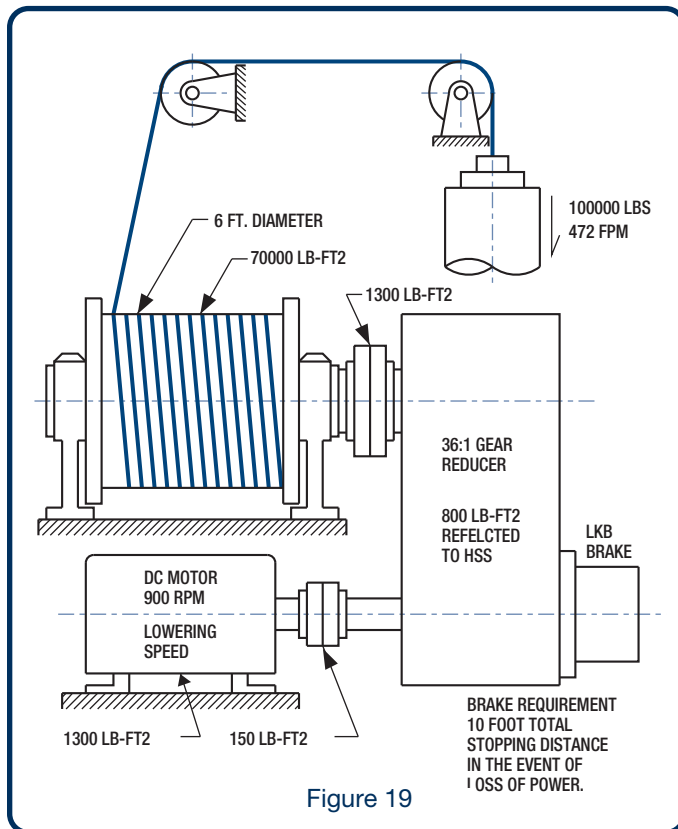


exceeds the actual cooldown time and the Model AGT-324 is suitable for this duty cycle.

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

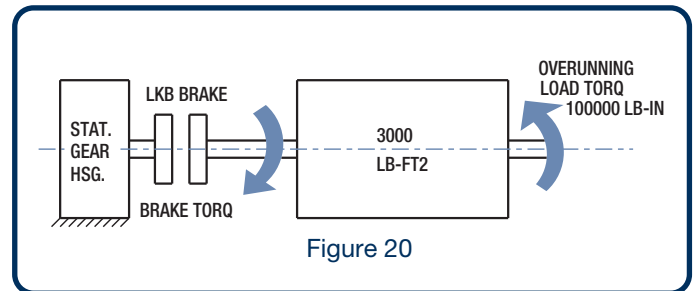
FIGURE 19 describes a geared-hoist arrangement such as may be found on a drilling apparatus or crane. This application incorporates a DC motor which uses regenerative braking to lower the load at a controlled rate.

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.

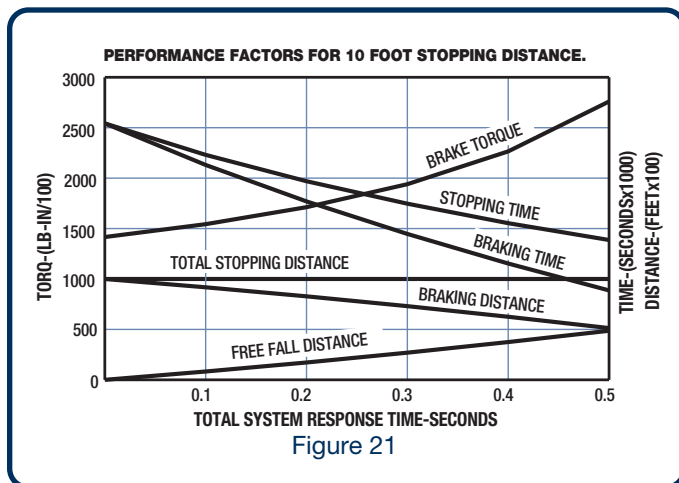


Figure 21

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.

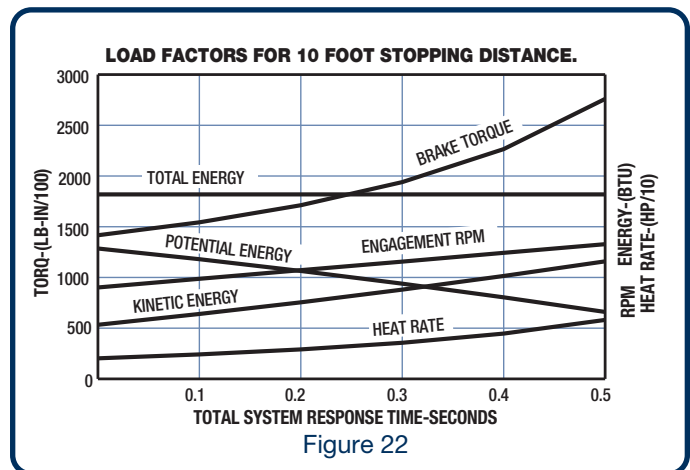


Figure 22

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).

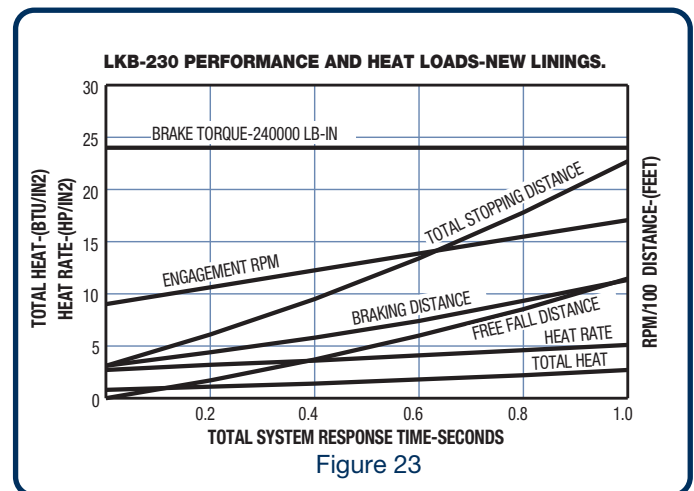
The above can be summarized for a 0.3 second system response time:

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

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

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

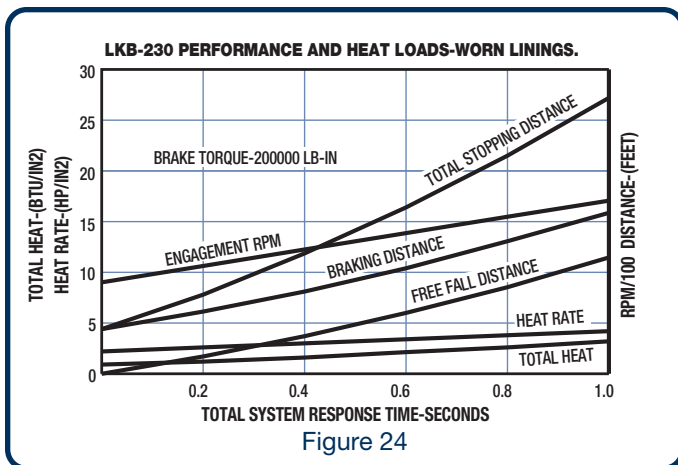
Brake Factors	Performance Effect
Brake Torque Capacity with new and worn linings	<ul style="list-style-type: none"> ■ Stopping Distance and Time ■ Brake Heat Loads ■ Drive Component Stress Levels
Brake Heat Capacity	<ul style="list-style-type: none"> ■ Frequency of Engagement ■ Lining Life
Brake Response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

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-FT²) – 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-FT²) 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 10⁴ 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.³

WR2 PER INCH – STEEL @ .283 LB/IN³

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

Service and Conversion Factors

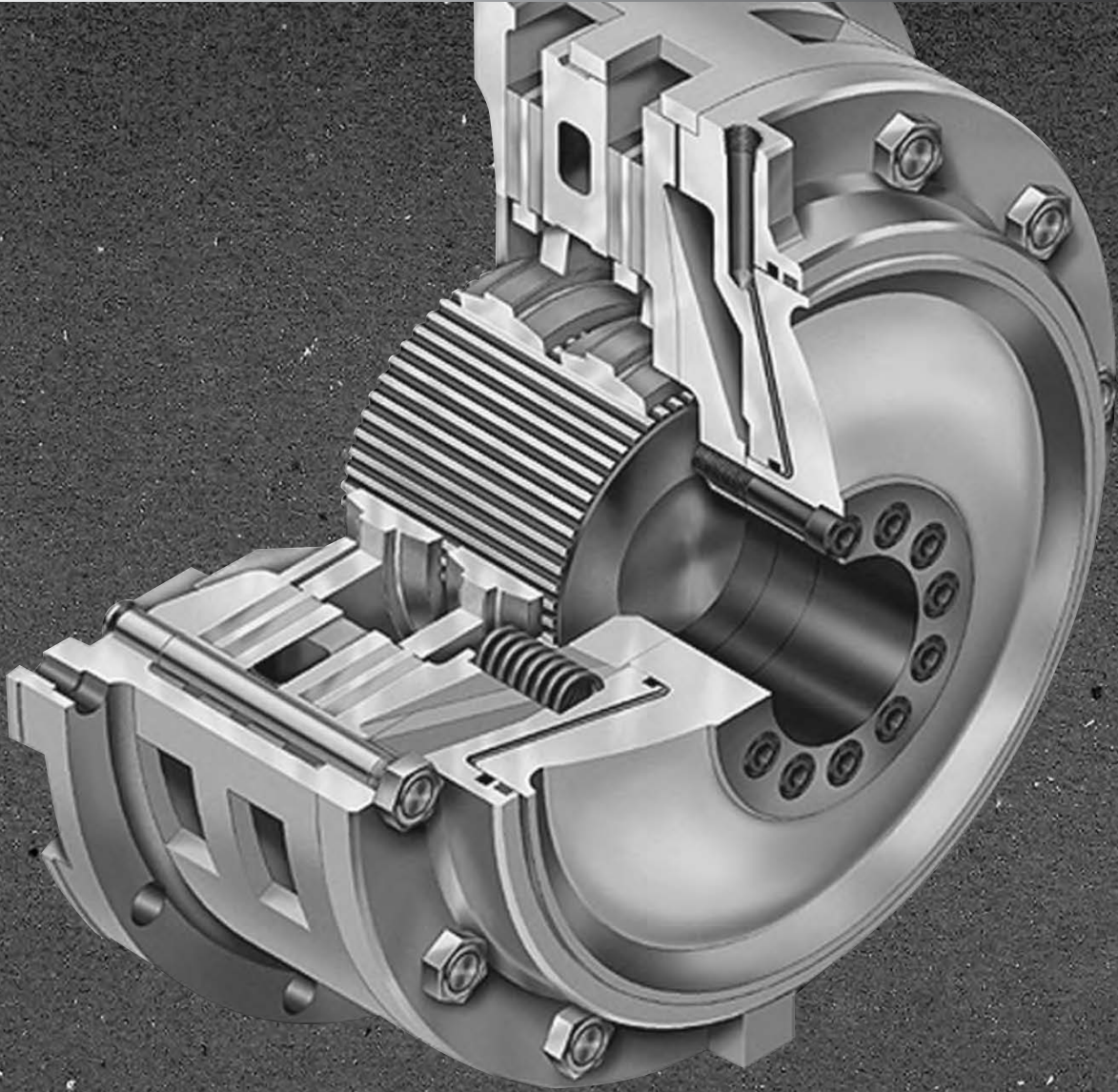
TABLE 3 – STATIC TORQUE SERVICE FACTORS (Refer to section XII for use)

Industry	Machinery	Model (Service Factor)	
LOGGING & LUMBER	CHIPPERS	AGT(3.5)	
	DE-BARKERS	AGT(3.0)	
	YARDER DRUMS	AGT(2.5) HBA(1.4)	
MARINE	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)	
MATERIAL HANDLING	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)	
METAL-FORMING	PRESSES, SHEARS, AND PRESS BRAKES	AGT(2.5) LK,LKT(2.0) CBA(1.3) LKB(1.3) CBH(1.3)	
	MILLS	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)	
MINING (SHOVELS & CRANES)	HOIST, DRAG, CROWD	LKB(1.4) HBS(1.2)	
	SWING	LKB(1.3) HBS(1.1)	
	PROPEL	LK(1.7) HC(1.5)	
PAPER	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)	
PETROLEUM (DRILLING & SERVICING)	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)	
PUMPING	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

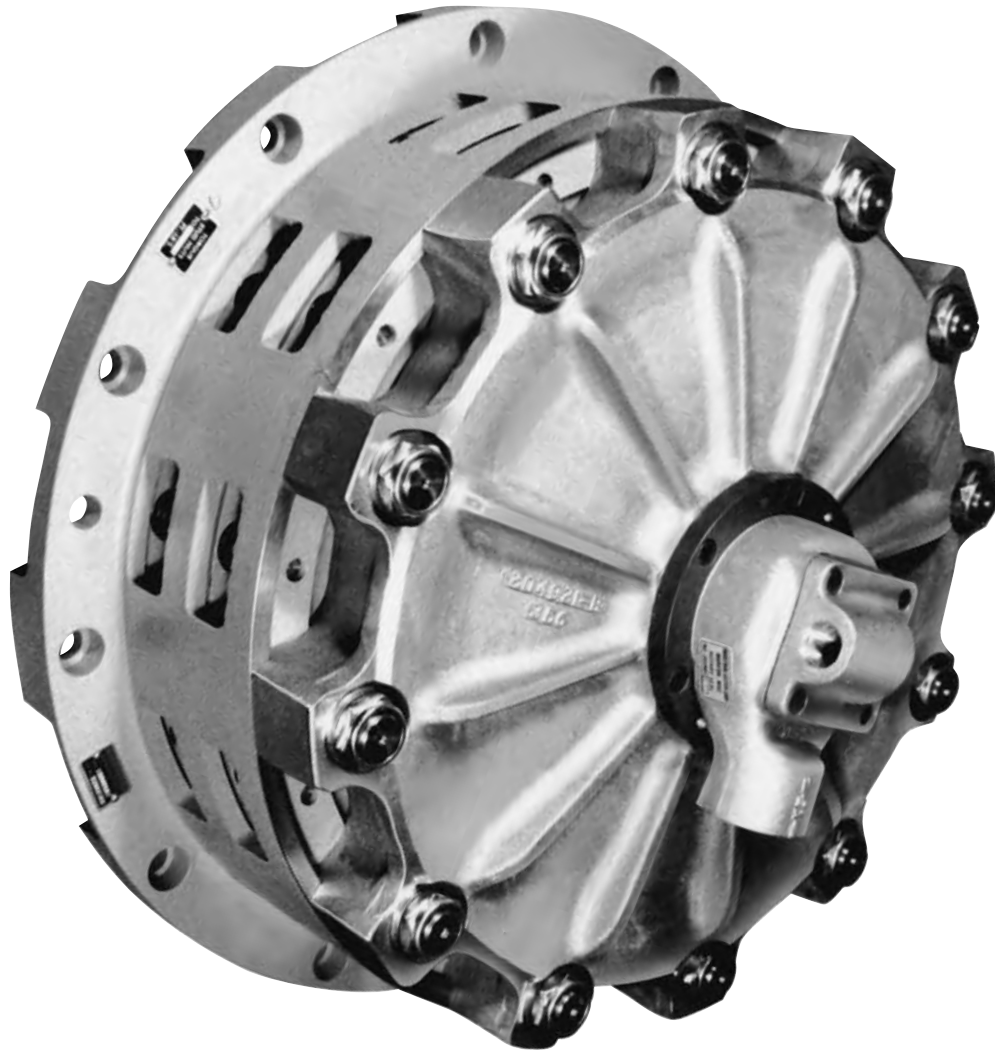
TO CONVERT...	INTO...	MULTIPLY BY ...
ATMOSPHERES	LBS/IN214.7
ATMOSPHERES	KG/CM21.0333
BARS	ATMOSPHERES0.987
BARS	LBS/IN214.5
BTU	FT-LBS778.3
BTU	JOULES1055
BTU/MIN	HP0.02356
BTU/MIN	KW0.01757
CENTIMETERS (CM)	INCHES0.3937
DEGREES	RADIANS0.01745
FEET (FT)	INCHES12
FEET	METERS0.3048
FT-LBS	BTU0.001285
GALLONS (US)	IN3231
GALLONS (IMP)	IN3277.42
GALLONS (IMP)	GALLONS (US)1.2
HORSEPOWER (HP)	BTU/MIN42.44
HP	FT-LBS/MIN33000
HP	FT-LBS/SEC550
HP	KW0.7457
HP	WATTS745.7
JOULES	BTU0.000948
KILOGRAMS (KG)	LBS2.2046
KG/CM2	LBS/IN214.2248
KG-M2 (GD2)	LB-FT2 (WR2)5.93
KG-M2 (WR2)	LB-FT2 (WR2)23.73
KG-M	LB-IN86.8
KILOWATTS (KW)	HP1.341
KW	BTU/MIN56.92
LB-FT2 (GD2)	LB-FT2 (WR2)4
LB-FT	NM1.35575
LB-IN	NM0.11302
LBS/IN2	BARS0.06897
LBS/IN2	KG/CM20.0703
LITERS	GALLONS (US)0.2642
LITERS	GALLONS (IMP)0.2201
METERS (M)	FEET3.281
MILLIMETERS (MM)	INCHES0.03937
NEWTONS (N)	LBS0.2248
NM	LB-IN8.848
NM	LB-FT0.7376
POUNDS (LBS)	KG0.4536
RADIANS	DEGREES57.296
TONS (METRIC)	TONS (SHORT)1.103
WATTS	HP0.001341

Dry Friction Clutches & Brakes



Air set / spring set / clutches & brakes

 **Industrial Clutch**[®]
Altra Industrial Motion



MODEL LK

Air Set Clutches & Brakes

LK Clutch or Brake Design Features

Self-contained separator springs for positive disengagements are integral with each plate and cannot fall out upon assembly or disassembly.

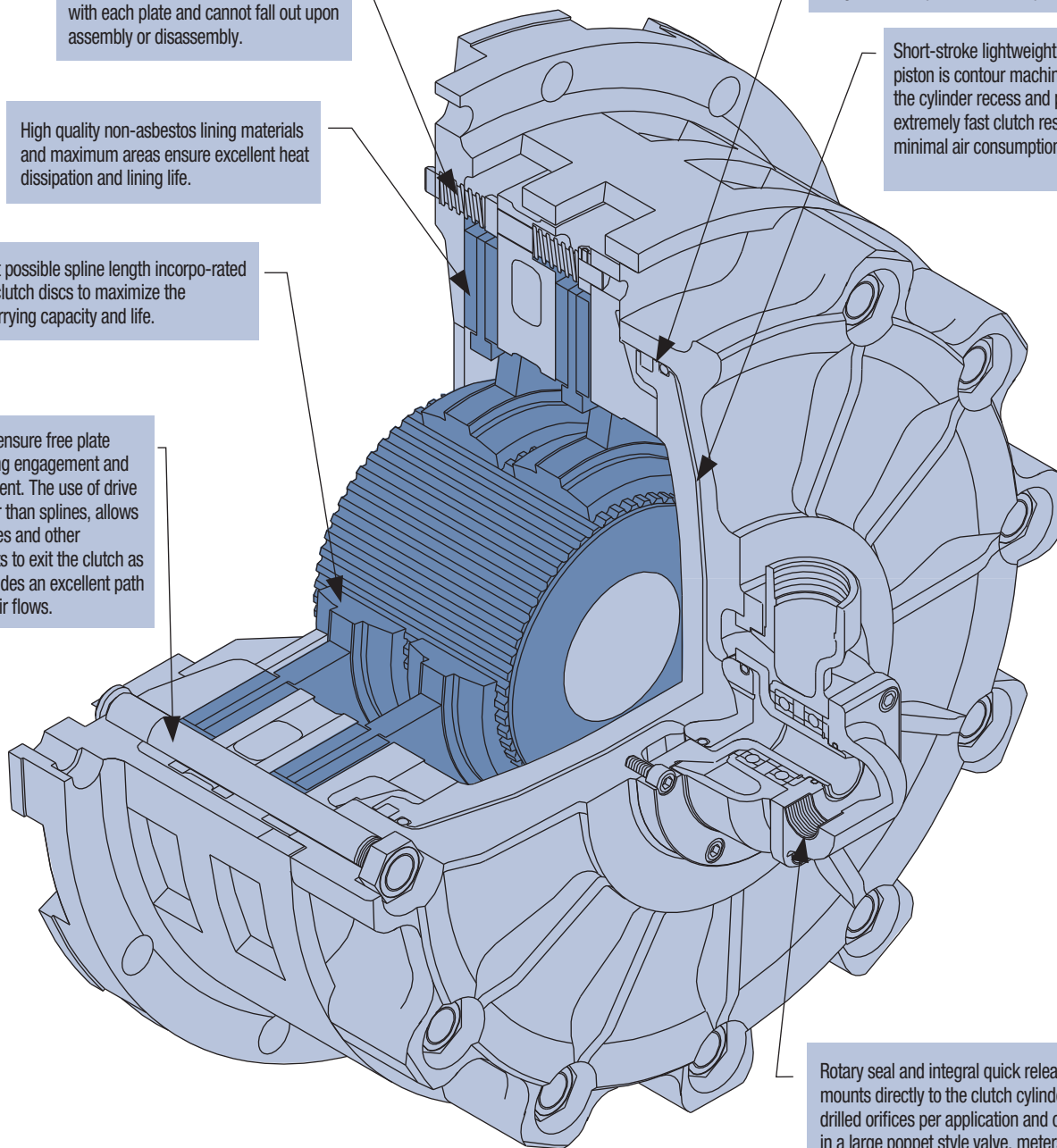
High quality non-asbestos lining materials and maximum areas ensure excellent heat dissipation and lining life.

Longest possible spline length incorporated on the clutch discs to maximize the load-carrying capacity and life.

Drive studs ensure free plate motion during engagement and disengagement. The use of drive studs, rather than splines, allows wear particles and other contaminants to exit the clutch as well as provides an excellent path for cooling air flows.

Long life, low replacement cost piston seals.

Short-stroke lightweight aluminum piston is contour machined to match the cylinder recess and provides for extremely fast clutch response with minimal air consumption.



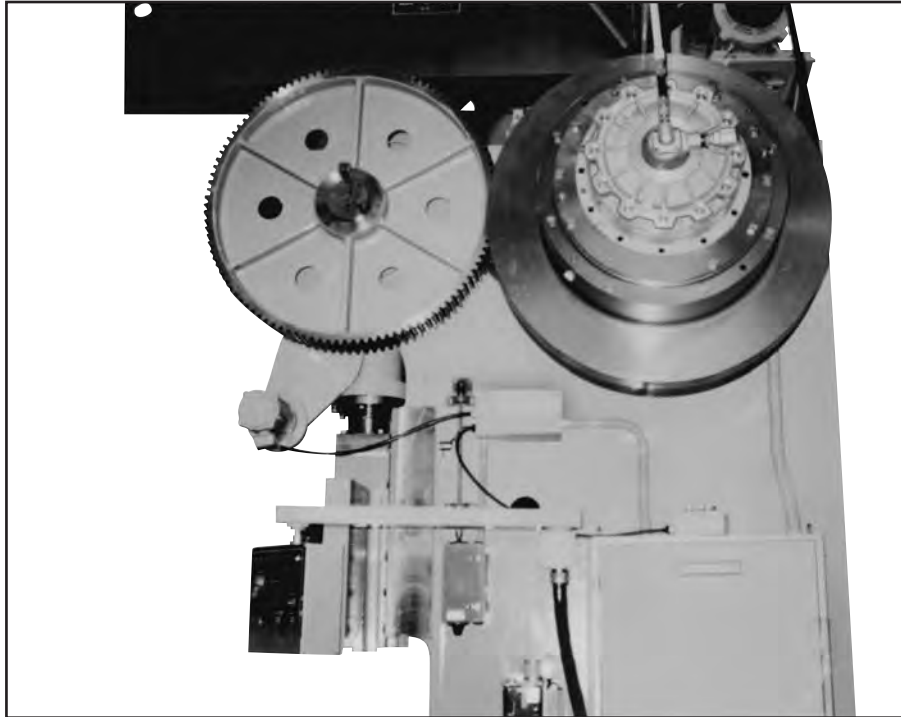
Rotary seal and integral quick release valve mounts directly to the clutch cylinder. Custom drilled orifices per application and clutch size in a large poppet style valve, meters the inlet air to provide smooth, shockless engagements and allows for immediate release upon command.

Clutch designed with total attention given to maximizing ventilation for cool running operation. Benefits include: longer lining life; stable friction coefficients; and lower maintenance costs.

All materials selected and field proven to provide the best available resistance to wear and thermal distress.

Precision machining of all components ensures positive interchangeability of all mating parts.

LK Clutch or Brake Design Features



**LK-217 Clutch
installed on
200-ton press**

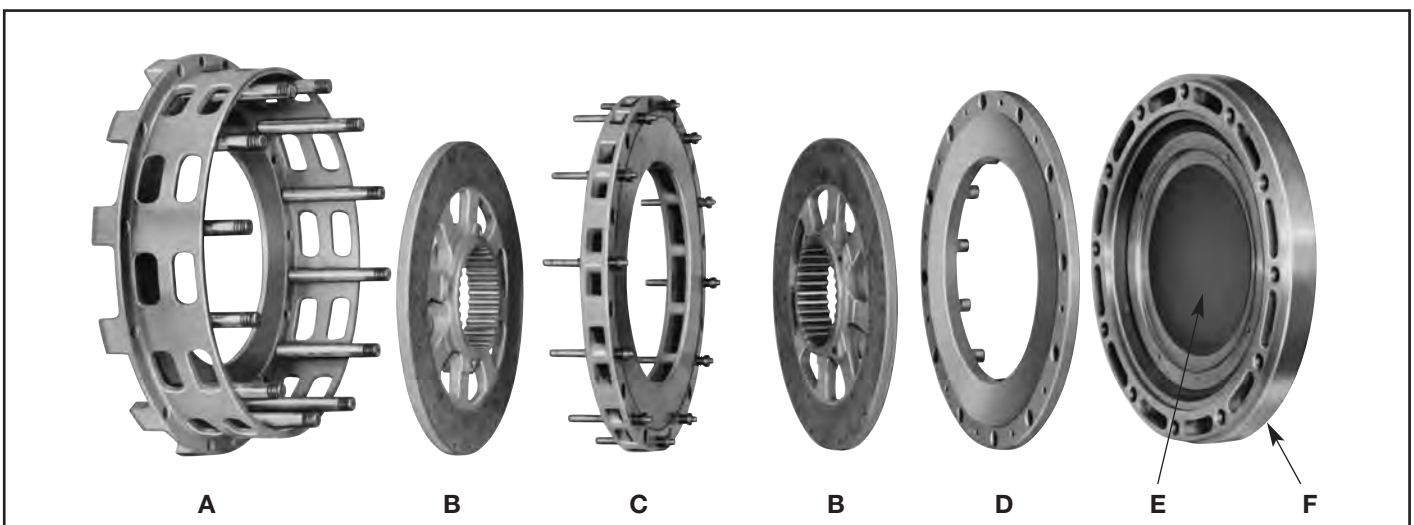
Photo courtesy of Kansai Iron
and Sunny Ltd.

Low Inertia LK Clutch or Brake Description

The Model LK clutch and brake has evolved over many years as a perfect choice for applications requiring rugged and efficient clutching and braking of tough and unusual type loads. The design incorporates an extremely low driven inertia, highest torque capacity per envelope size, and the quickest response with minimal air consumption of any clutch or brake available. It is particularly suited to those applications requiring frequent cycling and long life such as is found in metalforming or indexing applications. The components are arranged simply and efficiently.

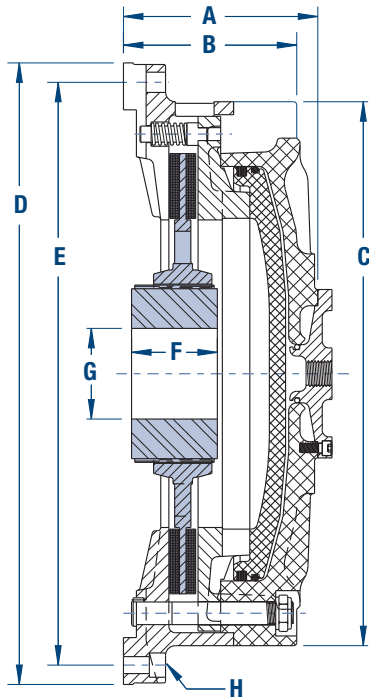
- A. Drive ring with integral drive studs.
- B. Lined friction discs with internal drive splines.
- C. Driving center plate with integral separating springs.
- D. Driving pressure plate with integral separating springs.
- E. Actuating piston and seals.
- F. Actuating cylinder with integral rotary seal and QRV.
- G. Externally splined inner hub (not shown).

Whether the application is in metalforming, oilfield machinery, cranes, or shovels, the LK low inertia design is an ideal choice for end of shaft mounting arrangements.

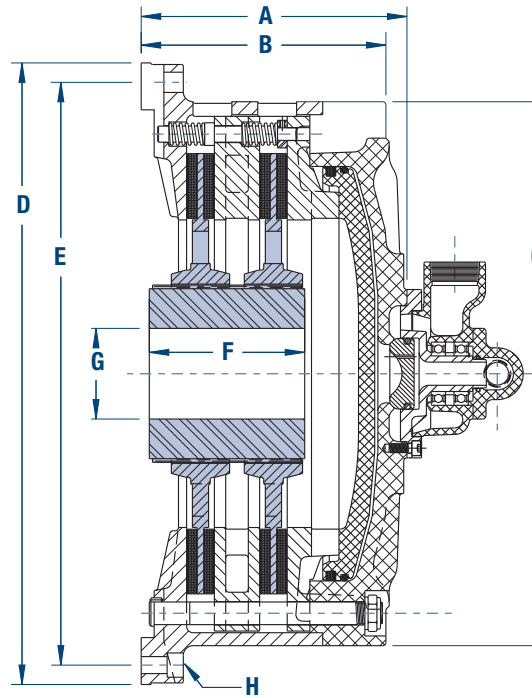


Model LK

**SINGLE DISC CLUTCH
WITH ROTARY SEAL ADAPTOR PLATE**



**DOUBLE DISC CLUTCH
WITH ROTARY SEAL AND QRV ASSEMBLY**



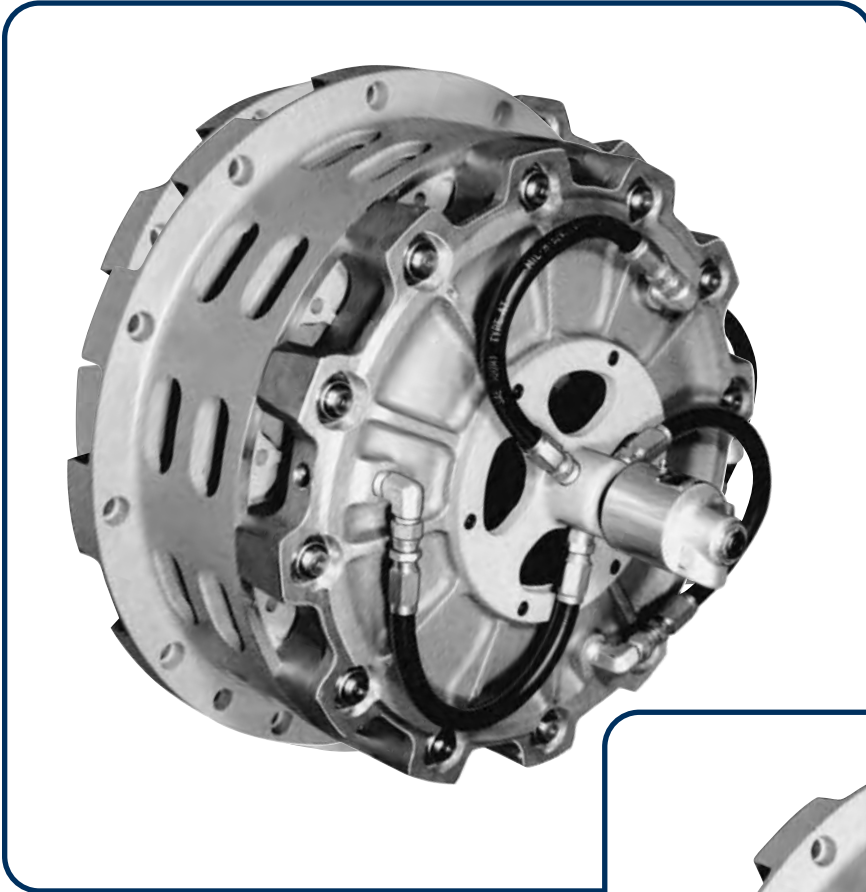
DIMENSIONAL DATA

All dimensions in inches

LK Model	Static Torque @ 100 PSIG (lb.-in.)	Dynamic Torque @ 100 PSIG (lb.-in.)	Weight Outer (lbs.)	Weight Inner (lbs.)	Total Weight (lbs.)	WR ² Outer (lb.-ft. ²)	WR ² Inner (lb.-ft. ²)	A	B	C	D	E	F	G Min.-Max.	H No.-Size
LK-108	9,890	8,570	35	7	42	4.55	.20	4.75	3.75	10.50	12.25	11.250	2.25	1-1/8 - 2	8 - 3/8
LK-208	19,780	17,140	49	13	62	6.96	.37	6.38	5.38	10.50	12.25	11.250	3.44	1-1/8 - 2	8 - 3/8
LK-110	17,070	14,800	47	10	57	9.30	.60	4.91	4.25	12.13	14.25	13.000	2.25	1-1/8 - 2	8 - 3/8
LK-210	34,140	29,590	66	12	78	12.17	1.15	6.72	6.06	12.13	14.25	13.000	4.13	1-1/8 - 2	8 - 3/8
LK-111	27,730	24,030	65	13	78	15.74	.98	5.63	4.69	14.25	16.50	15.250	2.75	1-1/8 - 2-3/4	8 - 1/2
LK-211	55,460	48,065	84	28	112	20.29	1.95	7.25	6.31	14.25	16.50	15.250	4.38	1-3/4 - 2-3/4	8 - 1/2
LK-113	48,375	41,925	111	35	146	44.88	2.06	6.63	5.81	16.63	18.88	17.625	3.75	2-1/2 - 3-3/4	8 - 5/8
LK-213	96,750	83,850	140	52	192	57.70	3.77	8.44	7.63	16.63	18.88	17.625	4.50	2 - 3-3/4	8 - 5/8
LK-117	86,400	74,880	185	58	243	96.61	9.35	7.50	6.69	21.00	24.00	22.500	3.31	2-3/8 - 4-1/2	12 - 5/8
LK-217	172,800	149,760	275	111	386	141.06	18.32	10.25	9.44	21.00	24.00	22.500	6.00	2-5/16 - 4-1/2	12 - 5/8
LK-118	106,340	92,160	246	63	309	162.63	10.88	7.68	6.31	22.63	26.25	24.500	3.31	2-3/8 - 4-1/2	12 - 3/4
LK-218	212,680	184,320	353	121	474	223.26	21.50	10.50	9.13	22.63	26.25	24.500	6.00	2-5/16 - 4-1/2	12 - 3/4
LK-121	168,800	146,300	232	72	304	189.40	18.35	7.63	6.63	25.00	28.25	26,750	3.31	2-3/8 - 4-1/2	12 - 3/4
LK-221	337,600	292,590	348	139	487	271.44	36.32	10.19	9.19	25.00	28.25	26,750	6.00	2-5/16 - 4-1/2	12 - 3/4
LK-321	506,400	438,880	363	207	570	353.48	54.33	12.75	11.75	25.00	28.25	26,750	8.75	2-7/16 - 4-1/2	12 - 3/4
LK-125	292,000	253,070	388	131	519	418.0	38.8	9.19	7.50	29.75	32.50	31,000	5.00	3-3/4 - 5-3/8	12 - 3/4
LK-225	584,000	506,130	540	226	766	580.8	74.9	12.38	10.69	29.75	32.50	31,000	7.25	3-3/4 - 5-3/8	12 - 3/4
LK-325	876,000	759,200	634	331	965	745.0	111.8	15.53	13.84	29.75	32.50	31,000	10.25	3-3/4 - 5-3/4	12 - 3/4
LK-130	514,880	446,230	789	211	1,000	1,050.0	90	12.88	9.75	35.50	39.00	37,000	5.00	4-5/8 - 7-1/2	12 - 7/8
LK-230	1,029,760	892,460	1,018	356	1,374	1,384.0	172	16.13	13.00	35.50	39.00	37,000	7.13	4-5/8 - 7-1/2	12 - 7/8
LK-330	1,544,600	1,338,650	1,226	526	1,752	1,722.0	257	19.38	16.25	35.50	39.00	37,000	10.38	4-5/8 - 7-1/2	12 - 5/8
LK-235	1,640,400	1,421,680	1,071	517	1,588	2,053.0	327	13.75	10.48	40.00	39.25	36,375	7.13	5-1/8 - 9-5/8	24 - 1-1/32
LK-335	2,460,600	2,132,520	1,168	732	1,900	2,446.0	483	16.19	12.91	40.00	39.25	36,375	9.56	5-1/8 - 9-5/8	24 - 1-1/32
LK-242	2,851,750	2,471,515	2,749	1,015	3,764	7,473.0	954	18.27	16.58	48.75	48.00	44,000	8.50	6-1/2 - 12	24 - 1-3/4
LK-342	4,277,625	3,707,275	3,286	1,497	4,783	9,145.0	1,422	22.13	20.44	48.75	48.00	44,000	12.31	6-1/2 - 12	24 - 1-3/4
LK-248	4,333,200	3,755,440	3,849	1,420	5,269	13,013.0	1,688	20.45	18.77	54.75	54.00	50,000	9.50	8 - 14	24 - 1-3/4
LK-348	6,499,800	5,633,160	4,628	2,110	6,738	15,954.0	2,522	24.93	23.25	54.75	54.00	50,000	14.00	8 - 14	24 - 1-3/4
LK-448	8,666,400	7,510,880	5,407	2,800	8,207	18,895.0	3,357	29.41	27.73	54.75	54.00	50,000	18.50	8 - 14	24 - 1-3/4

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.) Consult factory or refer to application information when selecting units.



MODEL LKT

Air Set Clutches & Brakes

LKT Clutch or Brake Design Features

Self-contained separator springs for positive disengagements are integral with each plate and cannot fall out upon assembly or disassembly.

High quality non-asbestos lining materials and maximum areas ensure excellent heat dissipation and lining life.

Longest possible spline length incorporated on the clutch discs to maximize the load-carrying capacity and life.

Drive studs ensure free plate motion during engagement and disengagement. The use of drive studs, rather than splines, allows wear particles and other contaminants to exit the clutch as well as provides an excellent path for cooling air flows.

Long life, low replacement cost piston seals.

Short-stroke lightweight aluminum piston is contour machined to match the cylinder recess and provides for extremely fast clutch response with minimal air consumption.

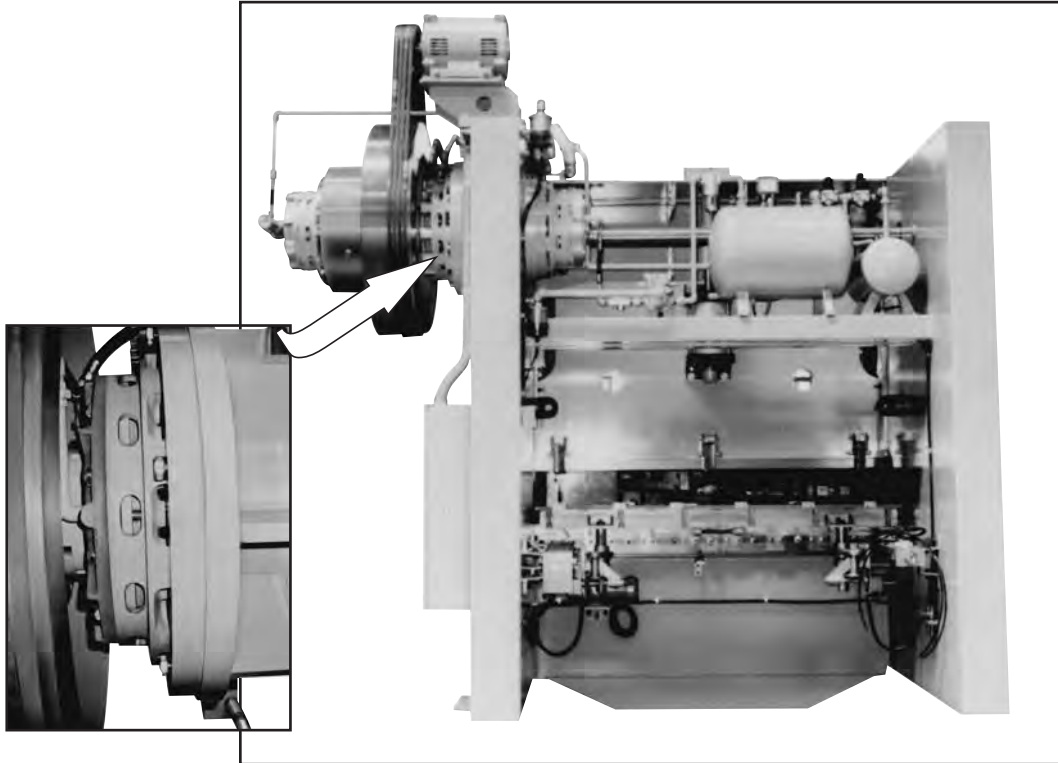
Ventilated air inlet spider mounts directly to cylinder for efficient actuating air flow into and out of unit.

Unit has been designed with total attention given to maximizing ventilation for cool-running operation. Benefits include: longer lining life; stable friction coefficients; and lower maintenance costs.

All materials selected and field proven to provide the best available resistance to wear and thermal distress.

Precision machining of all components ensures positive interchangeability of all mating parts.

LKT Clutch or Brake Design Features



**LK-121-T
Brake installed
on 200-ton,
2-speed press
brake**

Photo courtesy of Kansai Iron and Sunny Ltd.

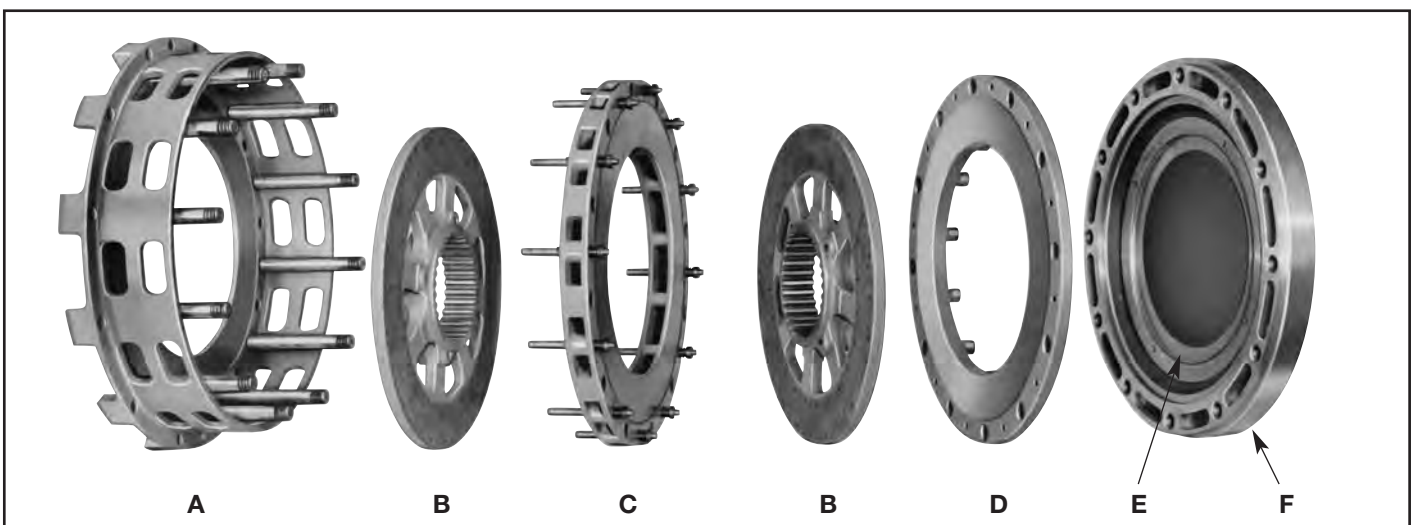
Low Inertia LKT Clutch or Brake Description

The Model LKT clutch or brake contains identical features to the Model LK units with the exception being the cylinder piston arrangement. Whereas the Model LK unit has a full cylinder covering its entire outboard end, the Model LKT has a doughnut type cylinder which allows it to be mounted in a through-shaft arrangement as an air set brake. The design incorporates an extremely low driven inertia and the quickest response with minimal air consumption of any clutch or brake available. It is particularly suited to those applications requiring frequent cycling and long life such as is found in metalforming or indexing applications.

The components are arranged simply and efficiently.

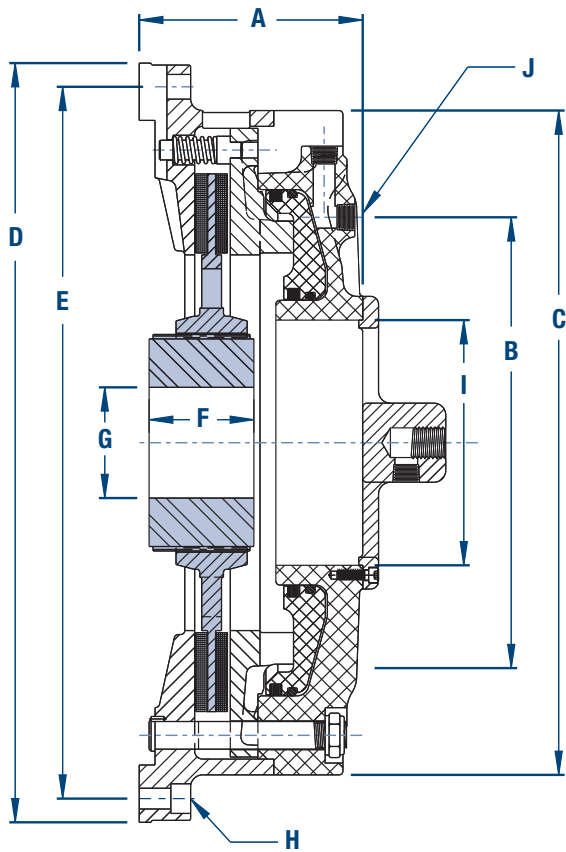
- A. Drive ring with integral drive studs.
- B. Lined friction discs with internal drive splines.
- C. Driving center plate with integral separating springs.
- D. Driving pressure plate with integral separating springs.
- E. Actuating piston and seals.
- F. Actuating cylinder with integral air inlet spider.
- G. Externally splined inner hub (not shown).

Whether the application is in metalforming, oilfield machinery, cranes, shovels, paper machinery or marine equipment, the LKT low inertia design is an ideal choice for through shaft mounting arrangements as an air set brake or end of shaft mounting as a clutch.

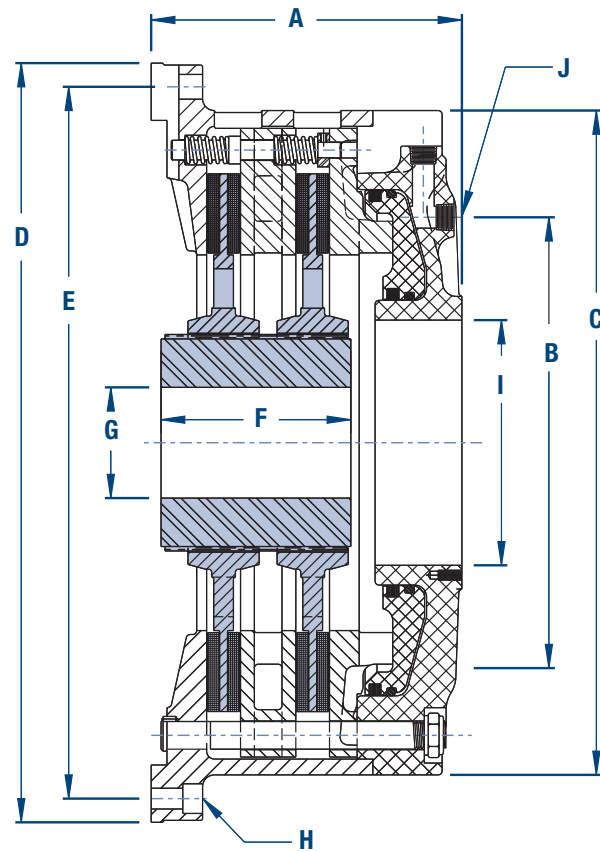


Model LKT

**SINGLE DISC CLUTCH
WITH ROTARY SEAL ADAPTOR PLATE**



DOUBLE DISC BRAKE



DIMENSIONAL DATA

All dimensions in inches

LKT Model	Static Torque @ 100 PSIG (lb.-in.)	Dynamic Torque @ 100 PSIG (lb.-in.)	Weight Outer (lbs.)	Weight Inner (lbs.)	Total Weight (lbs.)	WR ² Outer (lb.-ft. ²)	WR ² Inner (lb.-ft. ²)	A	B	C	D	E	F	G Min.-Max.	H No.-Size	I	J No.-Size
LKT-113	26,140	22,655	111	35	146	44.88	2.06	5.75	10.75	16.63	18.88	17.625	3.75	2-1/4 - 3-1/4	(8) - 5/8	7.00	(2) - 1
LKT-213	52,280	45,310	140	52	192	57.70	3.77	7.56	10.75	16.63	18.88	17.625	4.50	2-1/4 - 3-1/4	(8) - 5/8	7.00	(2) - 1/2
LKT-117	60,650	52,565	185	58	243	96.61	9.35	7.06	14.28	21.00	24.00	22.500	3.31	2-3/8 - 4-1/4	(12) - 5/8	7.75	(3) - 1/2
LKT-217	121,300	105,125	275	111	386	141.06	18.32	9.81	14.25	21.00	24.00	22.500	6.00	2-7/16 - 4-1/4	(12) - 5/8	7.75	(3) - 1/2
LKT-121	119,965	103,970	232	72	304	189.40	18.35	6.72	16.00	25.00	28.25	26.750	3.31	2-3/4 - 4-1/2	(12) - 3/4	9.75	(2) - 3/4
LKT-221	239,930	207,940	348	139	487	271.44	36.32	9.22	16.00	25.00	28.25	26.750	6.00	2-3/4 - 4-1/2	(12) - 3/4	9.75	(2) - 3/4
LKT-125	270,960	234,830	388	131	519	418.0	38.80	9.19	16.50	29.75	32.50	31.000	5.00	3-3/4 - 5-3/8	(12) - 3/4	6.88	(2) - 1
LKT-225	541,915	469,660	540	226	766	580.8	74.90	12.38	16.50	29.75	32.50	31.000	7.25	3-3/4 - 5-3/8	(12) - 3/4	6.88	(2) - 1
LKT-130	387,640	335,955	789	211	1,000	1050.0	90.00	12.88	20.25	35.50	39.00	37.000	5.00	4-5/8 - 7-1/2	(12) - 7/8	9.75	(2) - 1
LKT-230	775,285	671,910	1,018	356	1,374	1384.0	172.00	16.13	20.25	35.50	39.00	37.000	7.13	4-5/8 - 7-1/2	(12) - 7/8	9.75	(2) - 1

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.

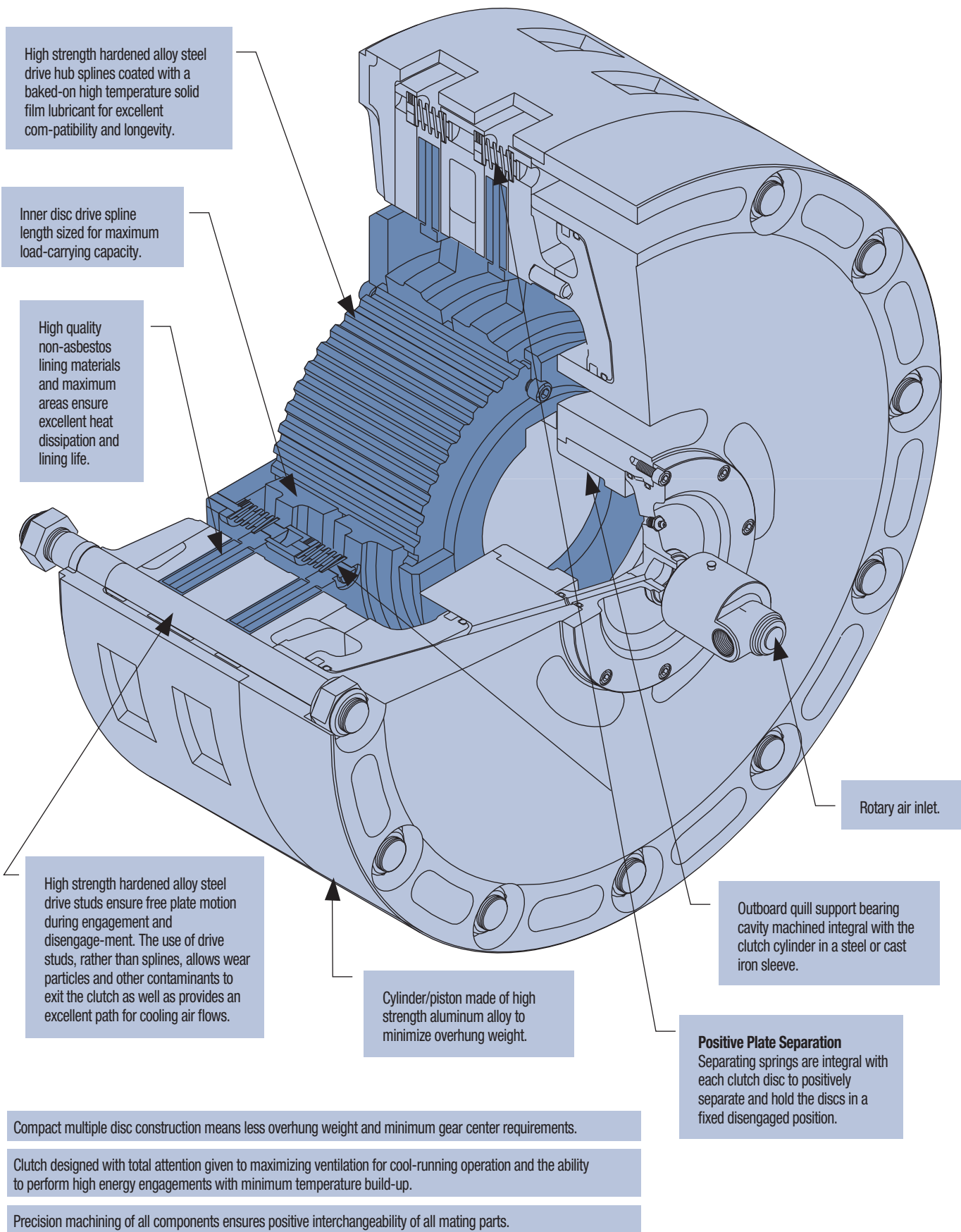
4.) Consult factory or refer to application information when selecting units.



MODEL LKM

Air Set Marine Clutches

LKM Clutch Design Features



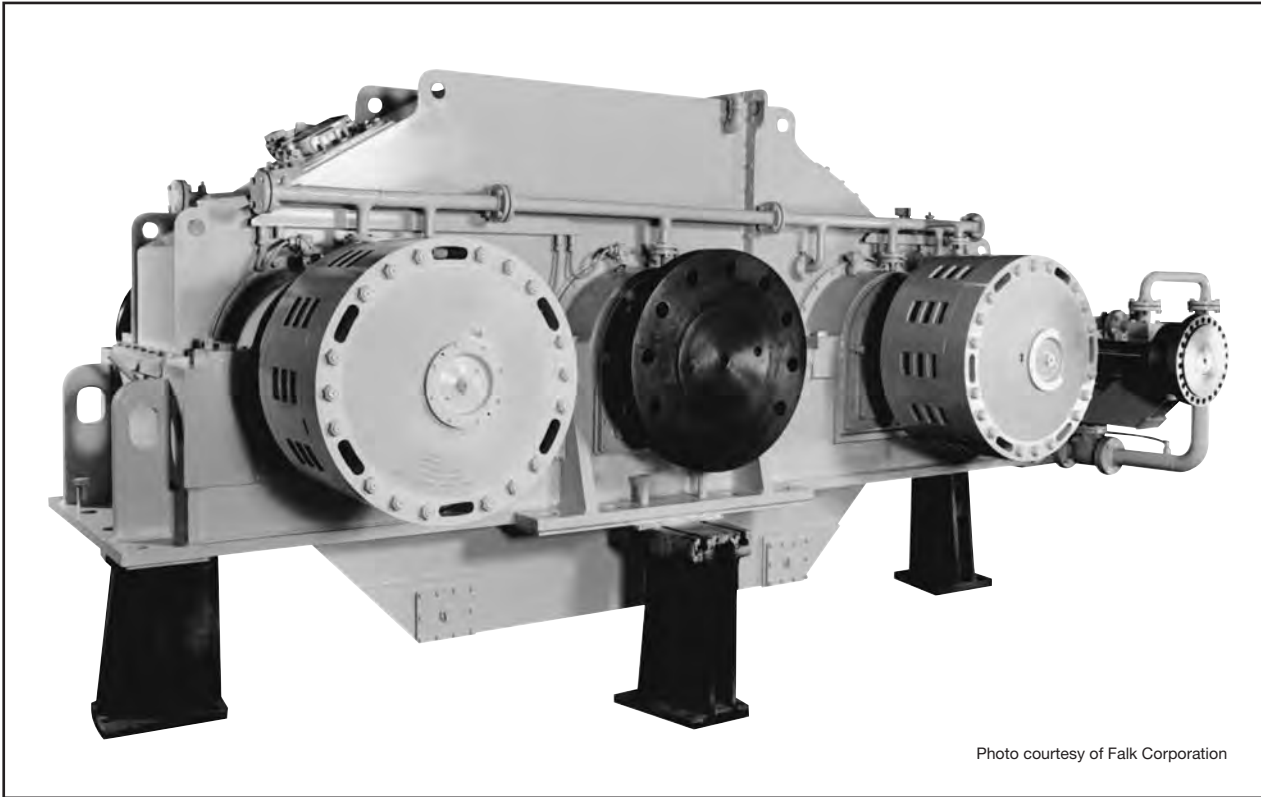


Photo courtesy of Falk Corporation

Model LKM-330-B clutches, each rated 5250 HP @ 500 RPM connect dual engine inputs to a single propeller on a Falk Model 55 x 20 DMA1-S marine gear.

LKM AIR SET MARINE CLUTCH DEVELOPMENT HISTORY

The Model LKM marine clutch incorporates many of the features found in our Model LK clutch series with some notable exceptions. Cyclic applications, where the LK style finds its most frequent use, contain duty cycles which bear little resemblance to marine main propulsion service. Marine clutches are required to make infrequent, yet in some instances, severe engagements from a heat standpoint. The clutches are also called upon to run for extended periods in the engaged position (ahead clutches) and for extended periods in the disengaged position (reverse clutches).

The Model LK clutches were rugged enough to handle the load duty cycle from a torque and heat standpoint but required modification to their separation systems in order to be suitable for marine service. Industrial Clutch Products was the first to recognize the importance of positive plate separation for marine service and all clutches installed in marine service contained this feature. Disc clutches for marine service without positive plate separation have experienced a sad approval rating from users.

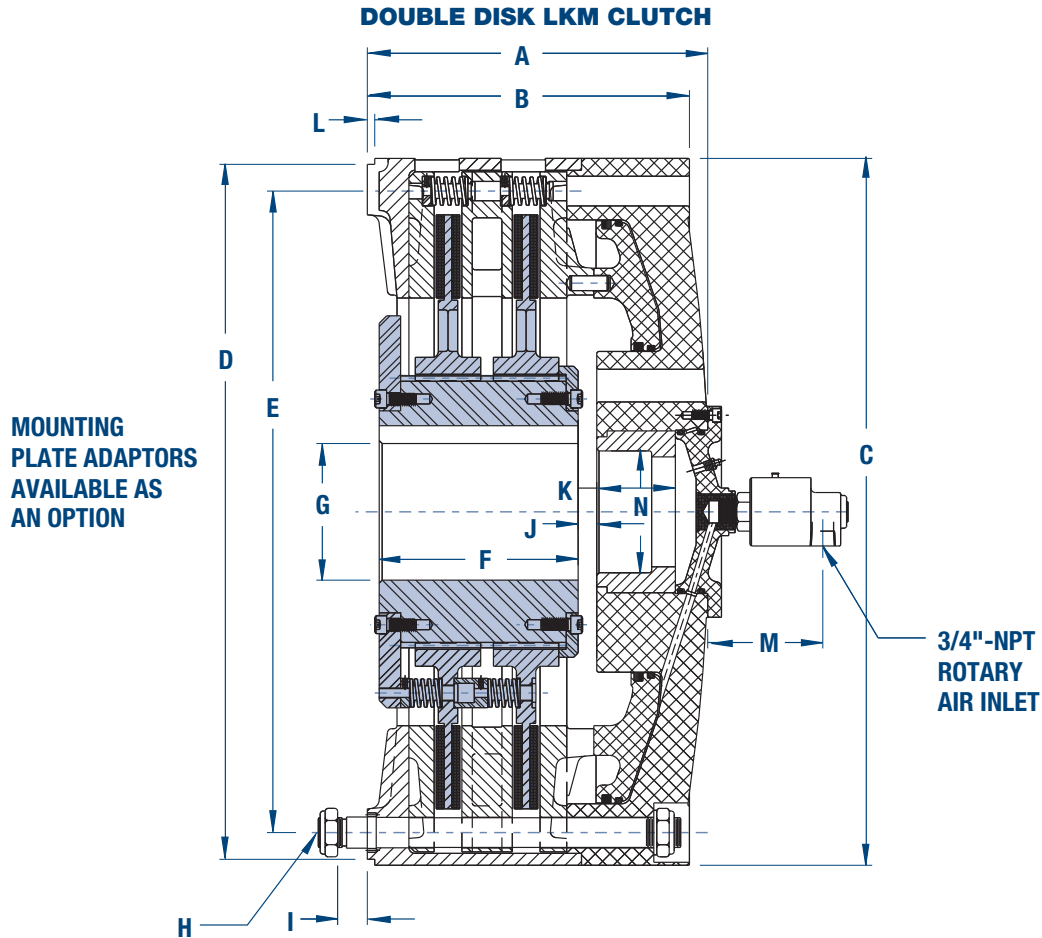
This is particularly true in forward/reverse service where, strangely enough, the unloaded reverse clutch gives the greatest degree of trouble. Because of the gearing arrangement, the outer members of this clutch rotate in

one direction while the inner member rotates in the other. This gives rise to the term “twice relative rotation.” The cant of the gear to which these clutches are mounted in order to align with the propeller shaft as well as vessel motion due to wave action causes the clutch plates, if not positively separated, to kiss and wear or burn-out. Exactly the same result would occur if one were going down the road in an automobile at a high rate of speed with the brake pedal lightly depressed.

To eliminate this, Industrial Clutch Products designed and patented positive plate separation which has accrued over 30 years of field experience in marine main propulsion service. The separating system is self-adjusting for wear and no adjustments are ever required by the user during regular use or during routine maintenance. No plate contact can occur with this unique positive plate separation system and the many advantages of the disc clutch can be fully realized for marine service.

The incorporation of positive plate separation, a bearing cavity for the outboard quill bearing, and modifications made to increase the load carrying capacity on a continual basis changed the time-proven LK design into the Model LKM marine clutch.

Model LKM



DIMENSIONAL DATA

All dimensions in inches

LKM Model	A	B	C	D	E	F	G Min.-Max.	H No.-Size	I	J	K	L	M	N Min.-Max.
LKM-213	8.69	7.86	16.63	16.25	14.65	5.25	2-7/8 - 3-1/2	8-15/16	1	.44	1.50	.18	4	3-1/2 - 4-3/8
LKM-313	10.50	9.69	16.63	16.25	14.63	7.06	2-7/8 - 3-1/2	8-15/16	1	.44	1.50	.18	4	3-1/2 - 4-3/8
LKM-218	11.31	10.31	22.63	21.50	20.00	7.75	2-7/8 - 4-1/8	12 - 1-1/16	1	.56	1.38	.18	4	4-1/2 - 5-9/16
LKM-318	14.19	13.19	22.63	21.50	20.00	10.63	2-7/8 - 4-1/8	12 - 1-1/16	1	.56	1.38	.18	4	4-1/2 - 5-9/16
LKM-221	11.06	10.44	25.00	24.50	22.50	6.81	3-3/8 - 5-1/2	12 - 1-1/16	1	.31	2.19	.18	4	4-1/2 - 5-15/16
LKM-321	13.63	13.00	25.00	24.50	22.50	9.38	3-3/8 - 5-1/2	12 - 1-1/16	1	.31	2.19	.18	4	4-1/2 - 5-15/16
LKM-225	14.38	13.63	29.75	29.25	27.00	8.50	4-5/8 - 7-1/2	12 - 1-5/16	1-1/4	.78	3.31	.18	4	4-1/2 - 6
LKM-325	17.63	16.86	29.75	29.25	27.00	11.75	4-5/8 - 7-1/2	12 - 1-5/16	1-1/4	.78	3.31	.18	4	4-1/2 - 6
LKM-230	15.63	13.25	35.50	35.00	32.00	9.00	5-1/4 - 9-5/8	12 - 1-1/4	2	1.06	3.06	.25	4-5/8	5 - 8
LKM-330	18.94	16.56	35.50	35.00	32.00	12.25	5-1/4 - 9-5/8	12 - 1-1/4	2	1.06	3.06	.25	4-5/8	5 - 8
LKM-235	15.63	13.38	41.50	40.75	36.88	7.69	8 - 12	24 - 1-1/4	2	1.13	3.88	.25	3-7/8	6 - 9-1/16
LKM-335	18.06	15.81	41.50	40.75	36.88	10.13	8 - 12	24 - 1-1/4	2	1.13	3.88	.25	3-7/8	6 - 9-1/16
LKM-242	18.27	16.58	48.75	48.00	44.00	10.75	8 - 12	24 - 1-3/4	3	1.00	4.00	.25	3-7/8	6-1/2 - 9-1/16
LKM-342	22.13	20.44	48.75	48.00	44.00	14.63	8 - 12	24 - 1-3/4	3	1.00	4.00	.25	3-7/8	6-1/2 - 9-1/16
LKM-248	21.02	18.77	54.75	54.00	50.00	11.56	9-1/8 - 14	24 - 1-3/4	3	1.38	5.00	.25	2-3/4	10 - 14.50
LKM-348	25.50	23.25	54.75	54.00	50.00	16.00	9-1/8 - 14	24 - 1-3/4	3	1.38	5.00	.25	2-3/4	10 - 14.50

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.) Consult factory or refer to application information when selecting

units.

OPERATIONAL DATA

LKM Model	Static Torque @ 125 PSIG (lb.-in.)	Dynamic Torque @ 125 PSIG (lb.-in.)	Weight Outer (lbs.)	Weight Inner (lbs.)	Total Weight (lbs.)	WR ² Outer (lb.-ft. ²)	WR ² Inner (lb.-ft. ²)	Maximum Speed (RPM)
LKM-213	94,600	82,000	155	65	220	50	7	2100
LKM-313	142,000	123,000	220	90	310	70	9	2100
LKM-218	200,000	173,333	440	135	575	255	24	1600
LKM-318	300,000	260,000	480	200	680	284	35	1600
LKM-221	382,000	331,000	430	200	630	320	50	1400
LKM-321	572,000	445,733	525	285	810	396	71	1400
LKM-225	510,000	442,000	730	360	1,090	775	110	1200
LKM-325	765,000	663,000	850	451	1,301	836	158	1200
LKM-230	1,000,000	866,666	1,095	475	1,570	1,456	222	1000
LKM-330	1,500,000	1,300,000	1,290	710	2,000	1,894	320	1000
LKM-235	1,455,000	1,261,000	1,925	675	2,600	3,091	472	850
LKM-335	2,183,000	1,892,000	2,088	1,012	3,100	4,580	720	850
LKM-242	2,716,000	2,354,000	2,770	1,098	3,868	7,495	990	700
LKM-342	4,074,600	3,531,320	3,305	1,565	4,870	9,148	1,458	700
LKM-248	2,466,600	3,004,386	3,942	1,535	5,477	13,048	1,752	650
LKM-348	5,200,000	4,506,666	4,721	2,205	6,926	15,989	2,584	650

SELECTION CHART – Marine Engine Horsepower (S.F. 2.0 @ 125 PSIG)

Clutch RPM	250	500	750	1000	1250	1500	1750	2000	2500	3000	3500	4000	4500	5000	6000	7000	8000	9000	10000	12000	14000	16000	18000	20000	Clutch RPM		
150	318	321 225	325 230	230	330 235	330 235	330	335 242	335 242	242	342 248	342 248	342	348	348	-	-	-	-	-	-	-	-	-	-	150	
200	218	221	321 225	325 230	230	230	330 225	330 225	335 242	335 242	242	242	342 248	342 248	348	348	348	-	-	-	-	-	-	-	-	200	
250	313 218	318 221	321 221	321	325 230	325 230	230	330 225	330 225	335 242	335 242	335 242	242	242	342 248	342	342	348	348	-	-	-	-	-	-	250	
300	313 218	318 221	321 221	321 225	321	325 230	325 230	230	330 235	330 235	330	335 242	335 242	335 242	242	342 248	342 248	342	348	348	-	-	-	-	-	300	
350	313 218	318 221	318 221	321 225	321	325 230	325 230	230	330 235	330 235	330 235	335 242	335 242	335 242	242	342 248	342 248	342	342	348	-	-	-	-	-	350	
400	213	218	318 221	321 221	321 225	321	325 230	230	230	330 235	330 235	330 235	335 242	335 242	242	342 248	342 248	342	342	348	348	-	-	-	-	400	
450	213	313 218	318 221	318 221	321 225	321	321	321	325 230	230	230	330 235	330 235	330 235	335 242	335 242	242	242	342 248	342 248	342	348	348	-	-	450	
500	213	313 218	318 218	318 221	321 221	221	321	321	325 230	230	230	330 235	330 235	330 235	335 242	335 242	242	242	342 248	342 248	342	348	348	342	348	500	
550	213	313 218	318 218	318 221	321 221	221	321	321	325 225	225	321	325 230	230	230	330 235	330 235	335 242	335 242	335 242	242	342 248	342 248	342	348	348	550	
600	213	313 218	318 218	318 221	321 221	221	221	321	321	325 225	225	325 230	230	230	330 235	330 235	330	335 242	335 242	335 242	242	342 248	342 248	342	348	600	
650	213	313 218	318 218	318 221	318 221	221	221	321	321	325 225	225	325 230	230	230	330 235	330 235	335 242	335 242	335 242	242	342 248	342 248	342	348	342	650	
700	213	213	313 218	318 218	318 221	221	221	321	321	325 225	225	325 230	230	230	330 235	330 235	330 235	335 242	335 242	335 242	242	242	342	342	342	700	
750	213	213	313 218	318 218	318 221	221	221	321	321	325 225	225	325 230	230	230	330 235	330 235	335	335	335	335	-	-	-	-	-	750	
800	213	213	313 218	318 218	318 221	221	221	321	321	325 225	225	325 230	230	230	330 235	330 235	330 235	335	335	335	-	-	-	-	-	800	
850	213	213	313 218	318 218	318 221	221	221	321	321	325 225	225	325 230	230	230	330 235	330 235	330 235	330	330	330	-	-	-	-	-	850	
900	213	213	313 218	313 218	318 218	218	218	318 221	318 221	221	221	321 225	321 225	321	325 230	325 230	230	230	330	330	330	-	-	-	-	-	900
1000	213	213	213	313 218	318 218	218	218	318 221	318 221	221	221	321 225	321 225	321	325 230	325 230	230	-	-	-	-	-	-	-	-	-	1000
1100	213	213	213	313 218	318 218	218	218	318 221	318 221	221	221	321 225	321 225	321	321	325	-	-	-	-	-	-	-	-	-	-	1100
1200	213	213	213	313 218	313 218	218	218	318 221	318 221	221	221	321 225	321 225	321	321	325	325	-	-	-	-	-	-	-	-	-	1200
1400	213	213	213	313 218	313 218	218	218	318 221	318 221	221	221	321 221	321 221	321	321	321	-	-	-	-	-	-	-	-	-	-	1400
1600	213	213	213	313 218	313 218	218	218	313 218	313 218	218	218	318	-	-	-	-	-	-	-	-	-	-	-	-	-	-	1600
1800	213	213	213	213	213	313	313	313	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	1800	
2000	213	213	213	213	213	213	313	313	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	2000	

Model LKM

LKM Marine Clutch Application Information

The selection chart on the previous page is suitable for forward/reverse main propulsion service at 125 PSI actuation.

For controllable pitch propellers, dredge pumps, winches, and fire pumps use the following service factors.

Machinery	Service Factor
Main Propulsion Cont. Pitch Prop.	1.8
Winches, Centrifugal Fire Pumps	1.65
Centrifugal Dredge Pumps	2.5
Main Propulsion Continuous Slip from Engine Idle	Consult Factory

Example: Centrifugal dredge pump Disconnect clutch application.

Input Parameters

The engine is rated 3000 HP and the shaft the clutch is to be mounted on is rotating at 900 RPM. The available air pressure is 150 PSIG.

$$\text{Working Torque} = \frac{\text{HP} \times 63025}{\text{RPM}} = \frac{3000 \times 63025}{900} = 210083 \text{ LBIN}$$

$$\text{Minimum Static Torque} = \text{Work Torque} \times \text{Service Factor} \\ = 210083 \times 2.5 = 525208 \text{ LBIN}$$

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

$$\text{Minimum Catalogue Rating} = 525208 \times \frac{125}{150} = 437674 \text{ LBIN}$$

The Model LKM-225-B (rated at 510,000 LB-IN) is selected from the static torque column shown in the operational data chart on the previous page. This unit is the nearest size that meets or exceeds the value of 437,674 LB-IN required.

The allowable speed for the Model LKM-225-B is 1200 RPM and exceeds the operating speed of 900 RPM. The selection is appropriate from a torque and speed standpoint.

Dredge pumps can draw extraneous matter into the pump which may suddenly stall the pump impeller. This places a huge shock on the drive system and the clutch is called upon to act as a fuse in these instances. A review of the magnitude of the actual applied service factor is always required to be certain the drive system components can withstand these overloads. It may be necessary to reduce the clutch service factor which increases the likelihood of clutch damage but saves a more costly drive system repair.

The preferred way to reduce the service factor is to adjust the operating air pressure accordingly and maintain the clutch size as originally selected.

The following procedure may be used to calculate the air pressure required for any desired service factor. In this example, the recommended service factor was 2.5. We selected a Model LKM-225-B clutch since it was the closest unit that met or exceeded the calculated torque requirement of 437,674 LB-IN. Note that the actual service factor is 2.91.

$$\text{Tact} = \text{Actual Static Capacity} = \text{Cat. Stat. Rating} \times \frac{\text{Supply Press}}{\text{Rated Press}}$$

$$\text{Tact} = 510000 \times \frac{150}{125} = 612000 \text{ LBIN}$$

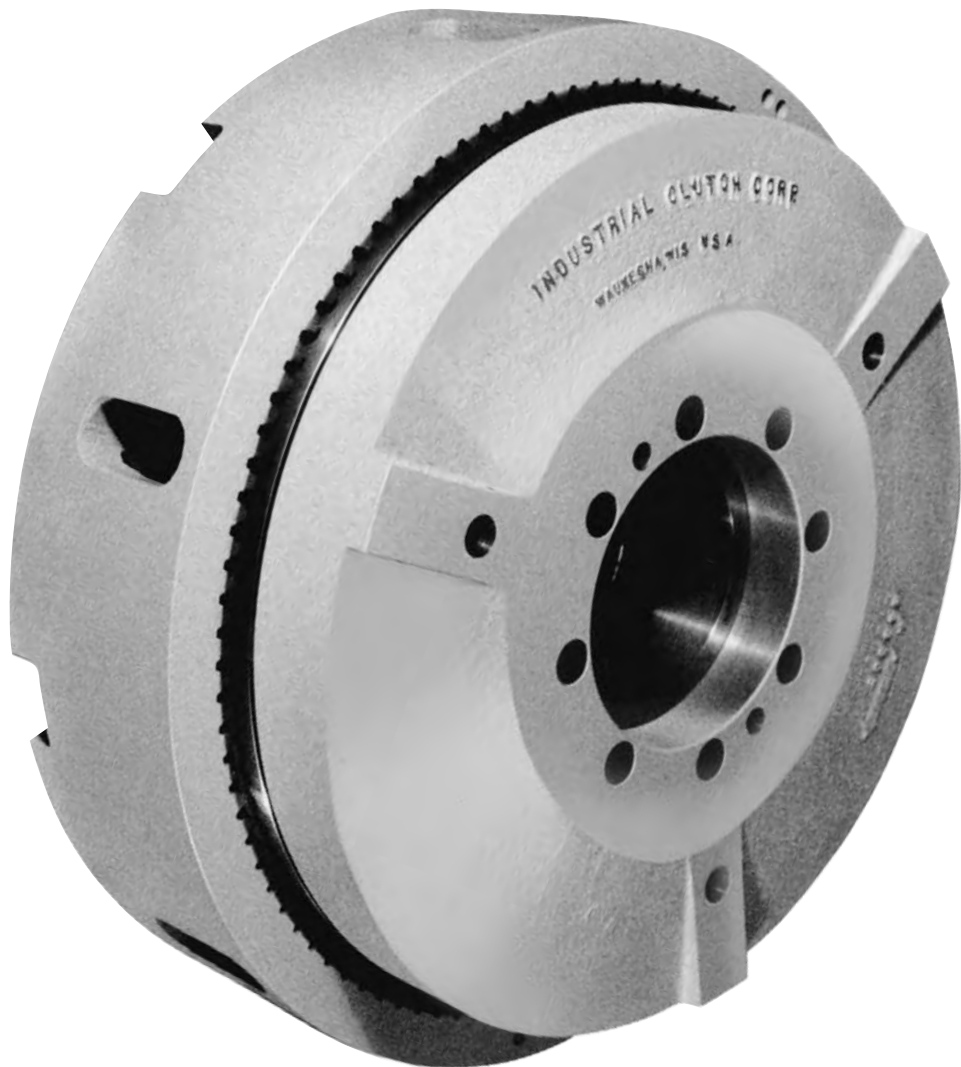
$$\text{Actual Serv. Fact.} = \frac{\text{Tact}}{\text{Working Torq}} = \frac{612000}{210083} = 2.91$$

If, by way of example, it was decided that a service factor of 2.0 was required, the supply air pressure should be set at 103 PSI.

$$\text{Operating Pressure} = \frac{125 \times \text{Reqd. Serv. Fact.} \times \text{Work Torq}}{\text{Cat. Stat. Torq. Rating}}$$

$$\text{Operating Pressure} = \frac{125 \times 2.0 \times 210083}{510000} = 103 \text{ PSIG}$$

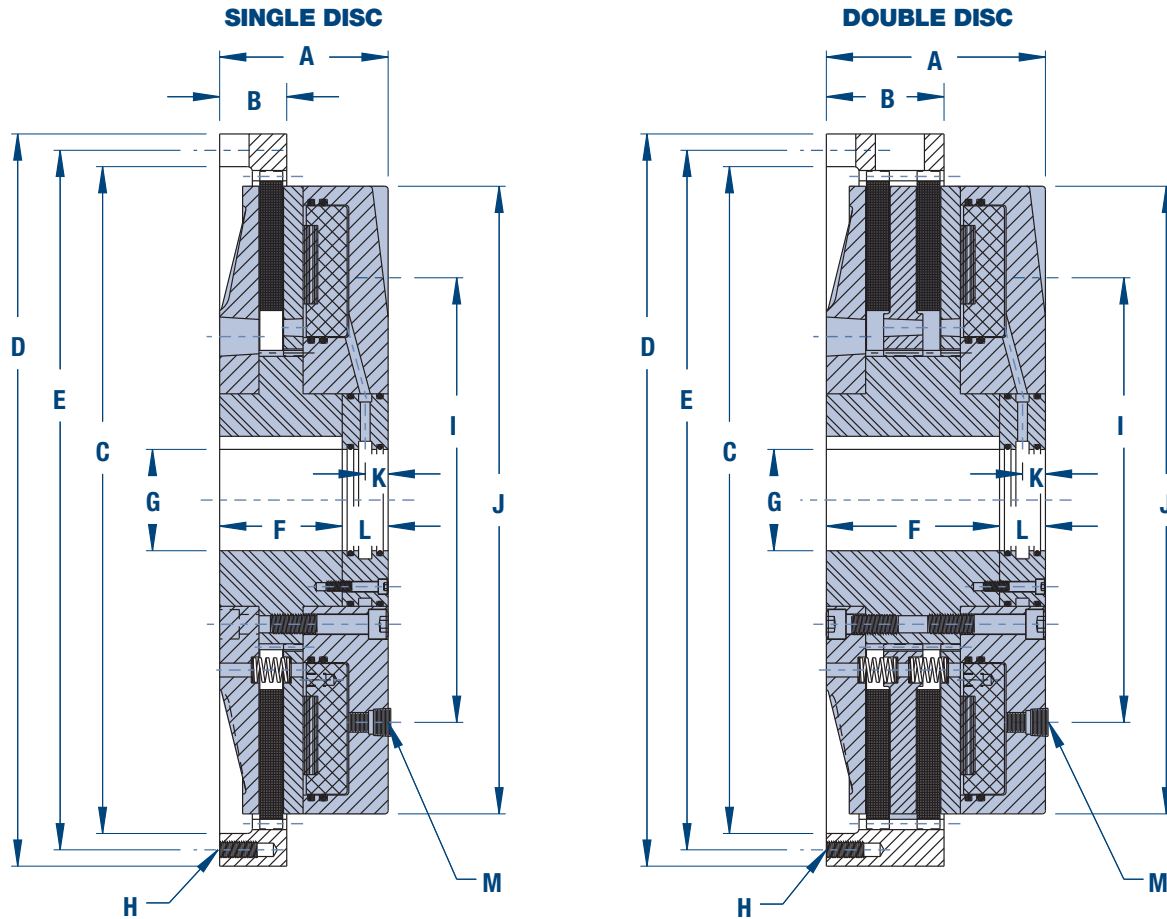
To summarize, a Model LKM-225-B operating at 103 PSIG will provide a 2.0 service factor for this application.



MODEL AGT

Air Set Clutches & Brakes

Model AGT



DIMENSIONAL DATA

All dimensions in inches

AGT Model	Static	Dynamic	Max. Speed (RPM)	Wt. Outer (lbs.)	Wt. Inner (lbs.)	Wt. Total (lbs.)	WR ² Outer (lb.-ft ²)	WR ² Inner (lb.-ft ²)	A	B	C	D	E	F	G Min.-Max.	H No.-Size	I	J	K	L	M
	Torque @ 100 PSIG (lb.-in.)	Torque @ 100 PSIG (lb.-in.)																			
AGT-115	38,300	33,200	2,000	40	130	170	15	40	4.56	1.75	16.000	18	17	2.94	2 - 3/4	(6) 5/8 - 11	11	15	0.81	1.63	1/2 NPT
AGT-215	76,600	66,400	2,000	60	160	220	25	50	5.88	3.00	16.000	18	17	4.25	2 - 3/4	(6) 5/8 - 11	11	15	0.81	1.63	1/2 NPT
AGT-315	115,000	99,600	2,000	78	192	270	35	60	7.19	4.38	16.000	18	17	5.56	2 - 3/4	(6) 5/8 - 11	11	15	0.81	1.63	1/2 NPT
AGT-118	71,500	62,000	1,750	45	185	230	30	50	5.06	1.88	19.500	22	20-3/4	3.31	2-1/2 - 4-1/2	(6) 5/8 - 11	15	18	0.88	1.75	1/2 NPT
AGT-218	143,000	124,000	1,750	63	304	367	45	82	6.63	3.38	19.500	22	20-3/4	4.88	2-1/2 - 4-1/2	(6) 5/8 - 11	15	18	0.88	1.75	1/2 NPT
AGT-318	214,600	186,000	1,750	80	420	500	55	113	8.19	5.00	19.500	22	20-3/4	6.44	2-1/2 - 4-1/2	(6) 5/8 - 11	15	18	0.88	1.75	1/2 NPT
AGT-121	103,400	89,600	1,400	82	320	402	69	119	5.63	2.13	22.500	25	23-3/4	3.88	3 - 6	(6) 5/8 - 11	16	21	0.88	1.75	1/2 NPT
AGT-221	206,800	179,200	1,400	124	435	559	105	143	7.25	3.75	22.500	25	23-3/4	5.50	3 - 6	(6) 5/8 - 11	16	21	0.88	1.75	1/2 NPT
AGT-321	310,200	268,800	1,400	210	550	760	194	205	8.88	5.38	22.500	25	23-3/4	7.13	3 - 6	(6) 5/8 - 11	16	21	0.88	1.75	1/2 NPT
AGT-124	159,700	138,400	1,100	117	433	550	132	199	6.44	2.63	25.500	28	26-3/4	4.69	3 - 6	(12) 5/8 - 11	17	24	0.88	1.75	3/4 NPT
AGT-224	319,400	276,800	1,100	196	582	778	217	263	8.38	4.50	25.500	28	26-3/4	6.63	3 - 6	(12) 5/8 - 11	17	24	0.88	1.75	3/4 NPT
AGT-324	479,100	415,200	1,100	290	734	1024	321	327	10.31	6.50	25.500	28	26-3/4	8.56	3 - 6	(12) 5/8 - 11	17	24	0.88	1.75	3/4 NPT
AGT-130	316,300	274,400	1,000	180	695	875	260	620	7.88	3.25	31.500	34	32-3/4	5.88	5-1/2 - 8-1/2	(12) 5/8 - 11	22	30	1.00	2.00	3/4 NPT
AGT-230	632,600	548,800	1,000	270	1080	1350	450	800	10.19	5.63	31.500	34	32-3/4	8.19	5-1/2 - 8-1/2	(12) 5/8 - 11	22	30	1.00	2.00	3/4 NPT
AGT-330	949,900	823,200	1,000	375	2075	2450	550	1475	12.56	8.00	31.500	34	32-3/4	10.56	5-1/2 - 8-1/2	(12) 5/8 - 11	22	30	1.00	2.00	3/4 NPT
AGT-136	598,300	475,200	800	275	1700	1975	625	1400	8.63	3.50	38.500	41	39-3/4	6.38	7 - 10	(18) 5/8 - 11	27	36	1.13	2.25	3/4 NPT
AGT-236	1,096,600	950,400	800	415	2300	2715	945	1900	11.19	6.13	38.500	41	39-3/4	8.94	7 - 10	(18) 5/8 - 11	27	36	1.13	2.25	3/4 NPT
AGT-336	1,644,900	1,425,600	800	550	3050	3600	1250	2525	13.81	8.75	38.500	41	39-3/4	11.56	7 - 10	(18) 5/8 - 11	27	36	1.13	2.25	3/4 NPT

NOTES: 1.) Use certified drawing dimensions only for final layouts.
2.) DXF and IGES files available upon request.

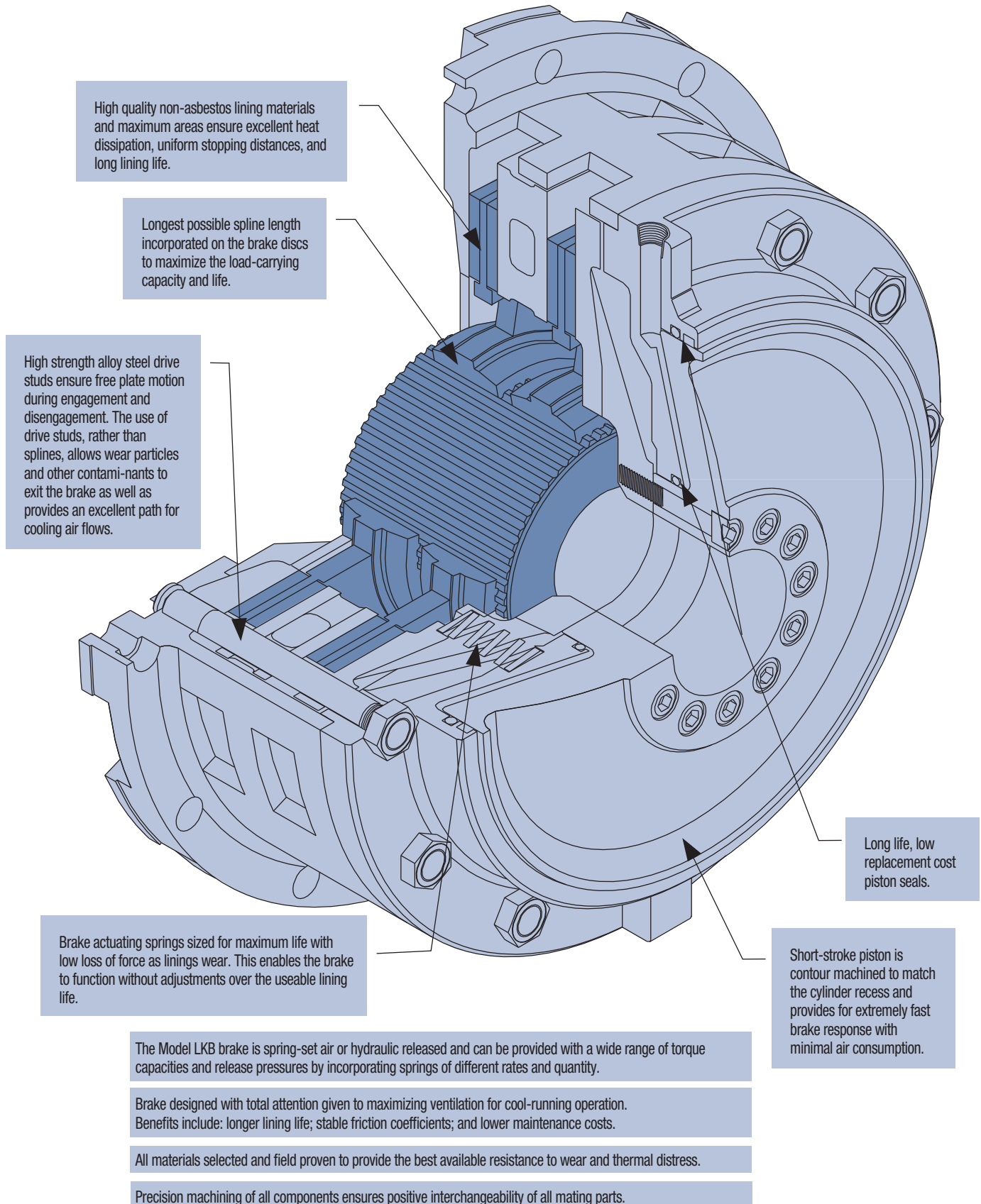
3.) Dimensions subject to change without notice.

4.) Consult factory or refer to application information when selecting units.



MODEL LKB
*Spring Set -
Air Released Brakes*

LKB Spring-Set Air or Hydraulic Released Brake Design Features



Model LKB-221 Brakes installed on a series of 250-ton presses

Photo courtesy of CNB



Low Inertia LKB Brake Description

The Model LKB spring-set brakes incorporate all of the design features of the LK style clutch and have been in existence equally as long. They are the perfect companion to the LK clutch if the application requires a clutch and a brake since both units are extremely fast in response, have low air consumption, and contribute the lowest possible inertia to the system that must be started and stopped.

The separate clutch-brake combination has been applied in the metal forming industry on presses, press-brakes, and shears, for over forty years and are long-living, easy to maintain and require no adjustment for wear. The units rugged simplicity and fast-acting repeatable response makes them an ideal choice for such applications.

Being ruggedly designed, and capable of dissipating large thermal loads, the Model LKB brakes are also used frequently as primary working brakes, safety brakes, and holding brakes for a multitude of loads found in many areas of the heavy industrial marketplace. Cranes, shovels, draglines, drilling equipment, oilfield machinery,

conveyors, and mining equipment are but a few of such applications.

The Model LKB brakes are available with many options. Torque capacities can be changed through the use of springs with different rates and quantities. The unit may be provided with chambers suitable for pneumatic or hydraulic release. Totally enclosed housings for adverse environments and limit switch monitoring of brake release and wear are available. Units modified to operate vertically may also be provided. The list of modifications that have been applied to the Model LKB brakes is long and the above are but a few. Industrial Clutch Products can assist in providing the most suitable unit for your application.

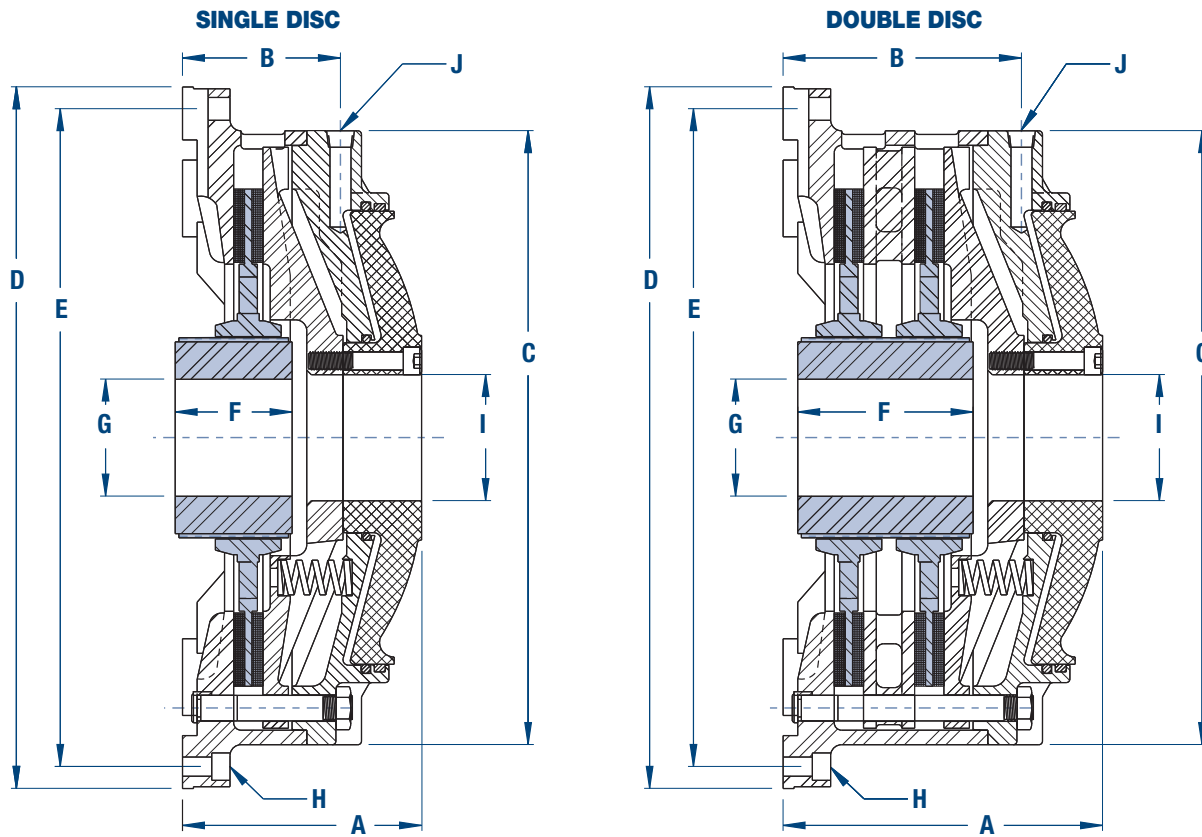
Modifications of the original and well designed LKB brakes have expanded this product's usefulness to many areas of industry. The complete range of sizes covers loads and speeds common to metalforming, mining, material handling, oilfields, and the marine marketplaces.



Several sizes of LKB brakes are used on the hoist, crowd, swing and propel functions for this 2.5 million lb. working weight electric mining shovel.

Photo courtesy of Bucyrus International

Model LKB



DIMENSIONAL DATA

All dimensions in inches

LKB Model	Static Torque 60 PSI Release (lb.-in.)	Dynamic Torque 60 PSI Release (lb.-in.)	Weight Outer (lbs.)	Weight Inner (lbs.)	Weight Total (lbs.)	WR ² Inner (lb.-ft ²)	A	B	C	D	E	F	G Min. - Max.	H No.-Size	I	J
LKB-108	5,190	4,500	42	7	49	.20	5.06	3.19	10.50	12.25	11.25	2.00	1-1/8 - 2	8 - 3/8	2.13	1/2
LKB-208	9,565	8,290	61	13	74	.37	6.69	4.81	10.50	12.25	11.25	4.13	1-1/8 - 2	8 - 3/8	2.13	1/2
LKB-111	12,080	10,470	77	13	90	.980	5.69	4.06	14.25	16.50	15.25	2.75	1-1/8 - 2-3/4	8 - 1/2	3.13	1/2
LKB-211	24,160	20,940	97	28	125	1.950	7.28	5.81	14.25	16.50	15.25	4.38	1-3/4 - 2-3/4	8 - 1/2	3.13	1/2
LKB-113	16,865	14,615	125	35	160	2.06	6.63	4.56	16.63	18.88	17.63	4.00	2-1/4 - 3-1/4	8 - 5/8	4.63	1/2
LKB-213	35,785	31,015	168	52	220	3.77	8.44	6.44	16.63	18.88	17.63	4.50	2 - 3-3/4	8 - 5/8	4.63	1/2
LKB-117	37,230	32,265	272	58	330	9.78	8.31	5.44	21.00	24.00	22.50	4.00	2-1/4 - 4-1/2	12 - 5/8	4.00	1/2
LKB-217	74,460	64,530	324	111	435	18.32	11.13	8.13	21.00	24.00	22.50	6.00	2-5/16 - 4-1/2	12 - 5/8	4.00	1/2
LKB-121	72,860	63,145	318	72	390	18.55	8.56	5.31	25.00	28.25	26.75	3.63	2-3/4 - 4-1/2	12 - 3/4	6.00	3/4
LKB-221	145,720	126,290	436	139	575	36.32	11.13	8.31	25.00	28.25	26.75	6.00	2-5/16 - 4-1/2	12 - 3/4	6.00	3/4
LKB-125	114,160	98,940	589	131	720	38.78	10.50	6.25	29.75	32.50	31.00	5.00	3-3/4 - 5-3/4	12 - 3/4	7.25	3/4
LKB-225	228,323	197,880	708	226	934	74.91	13.75	9.50	29.75	32.50	31.00	7.25	3-3/4 - 5-3/4	12 - 3/4	7.25	3/4
LKB-130	210,510	182,440	1089	211	1300	90.00	12.88	7.25	35.50	39.00	37.00	5.00	4-5/8 - 7-1/2	12 - 7/8	8.25	3/4
LKB-230	378,460	328,000	1314	356	1670	172.00	16.13	10.50	35.50	39.00	37.00	7.13	4-5/8 - 7-1/2	12 - 7/8	8.25	3/4
LKB-135	274,660	238,040	1100	350	1450	181.00	10.69	6.81	40.00	39.25	36.38	6.00	5-1/8 - 9-5/8	24 - 1-1/32	17.00	3/4
LKB-235	519,090	449,880	1183	517	1700	327.00	13.13	9.25	40.00	39.25	36.38	7.13	5-1/8 - 9-5/8	24 - 1-1/32	17.00	3/4
LKB-142	482,450	418,120	3038	611	3649	509.00	13.70	10.73	48.75	48.00	44.00	6.00	6-1/2 - 12	24 - 1-3/4	18.75	1
LKB-242	906,760	785,860	3574	1015	4589	954.00	17.46	14.40	48.75	48.00	44.00	8.50	6-1/2 - 12	24 - 1-3/4	18.75	1
LKB-148	730,380	633,000	4225	894	5119	923.00	14.76	11.26	54.75	54.00	50.00	7.00	8 - 14	24 - 1-3/4	21.75	1
LKB-248	1,360,930	1,179,470	5004	1420	6424	1688.00	19.15	15.65	54.75	54.00	50.00	9.50	8 - 14	24 - 1-3/4	21.75	1

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.) The torques shown are at 60 PSI release and may be varied by the use of different springs and spring combinations. For further technical data and recommendations please consult our Engineering Department.

5.) Consult factory or refer to application information when selecting units.

6.) Hydraulic release also available. Consult factory.

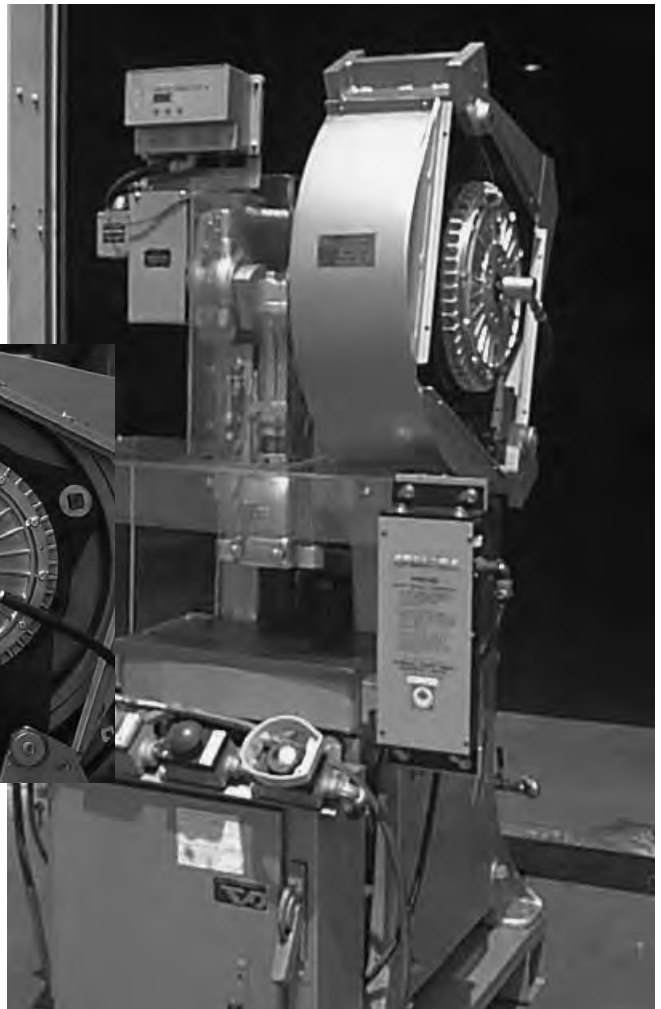
INDUSTRIAL CLUTCH



Combination Pneumatic Clutch/Brake



- The new Industrial Clutch Combination Clutch/Brake is a dry running unit with a pneumatically actuated clutch and a spring set brake.
- Available in various sizes from two different series.
- Low inertia design with five mounting styles.
- Used in OEM applications and ideal for press conversions.



Designation:

0 - 100 - 13 5 - 15 - 4
 Model ICCB Series Mounting Style Size Spring Complement

100-120 Series (Standard)

Size	Torque N-m		Volume cm ³		Inertia kg-m ²		Max. Bore mm	Max. Speed rpm
	Clutch@5.5 bar	Brake	New	Worn	Internal	External		
7	260	180	70	100	0.02	0.008	35	2750
9	550	350	130	170	0.058	0.025	45	2250
12	1050	700	230	290	0.188	0.063	65	1750
15	2150	1500	460	560	0.55	0.2	80	1400
18	4250	3000	820	1150	1.5	0.575	95	1200
19	5150	3500	1000	1370	2.175	0.825	105	1100
21	6300	4500	1180	1710	3.125	1.175	110	1000
23	8500	6000	2150	3150	5	1.85	125	900
26	12,900	8500	2220	3210	9.25	3.75	145	800
30	17,000	12,000	2650	3850	15.75	6.5	160	700
35	26,000	18,000	3750	5700	28.75	11.75	180	600

100-130 Series (Low-Inertia, High-Torque)

Size	Torque N-m		Volume cm ³		Inertia kg-m ²		Max. Bore mm	Max. Speed rpm
	Clutch@5.5 bar	Brake	New	Worn	Internal	External		
15	2600	2100	450	640	0.52	0.2	90	1500
17	3500	2800	530	760	0.84	0.33	95	1400
18	4900	3900	760	1140	1.41	0.5	105	1250
21	7700	6100	1210	1850	2.94	1.08	125	1100
23	10,000	8100	1590	2350	5	1.7	145	1000
26	15,000	11,900	2370	3570	8.8	3.34	160	850
30	21,500	17,400	3040	4580	15.5	5.78	180	750

Metric Specifications

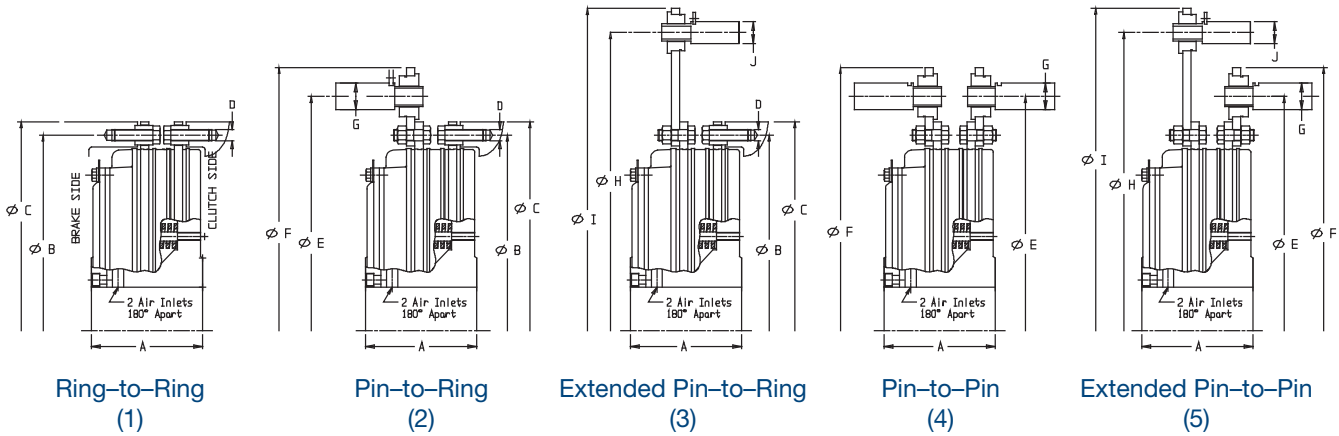
100-120 Series (Standard)

Size	Torque in-lb		Volume in ³		Inertia lb-ft ²		Max. Bore in	Max. Speed rpm
	Clutch@80 psi	Brake	New	Worn	Internal	External		
7	2300	1500	4.27	6.1	0.47	0.19	1.378	2750
9	4800	3000	7.93	10.37	1.4	0.6	1.772	2250
12	9200	6100	14.04	17.7	4.5	1.5	2.559	1750
15	19,000	13,000	28.07	34.17	13.1	4.7	3.15	1400
18	37,000	26,000	50.04	70.18	35.6	13.6	3.74	1200
19	45,000	30,000	61.02	83.6	51.6	19.6	4.134	1100
21	55,000	39,000	72.01	104.35	74.2	27.9	4.331	1000
23	76,000	53,000	131.2	192.22	118.7	43.9	4.921	900
26	113,000	75,000	135.47	195.87	219.5	89	5.709	800
30	150,000	106,000	161.71	234.94	373.8	154.2	6.299	700
35	230,000	159,000	228.84	347.84	682.2	278.8	7.087	600

100-130 Series (Low-Inertia, High-Torque)

Size	Torque in-lb		Volume in ³		Inertia lb-ft ²		Max. Bore in	Max. Speed rpm
	Clutch@80 psi	Brake	New	Worn	Internal	External		
15	22,500	18,200	27.5	39	12.3	4.7	3.543	1500
17	30,400	24,300	32.3	46.4	19.9	7.8	3.740	1400
18	42,500	33,800	46.4	69.6	33.5	11.9	4.134	1250
21	66,800	52,900	74	113	69.8	25.6	4.921	1100
23	86,800	70,300	97	143	119	40	5.709	1000
26	130,200	103,300	145	218	209	79	6.299	850
30	186,600	151,000	186	279	368	137	7.087	750

Available Mounting Arrangements



ALL DIMENSIONS IN MILLIMETERS

100-120 Series (Standard)

Size	A	B	C	D	E	F	G	H	I	J
7	58	205	220	M5	250	282	14	325	357	14
9	58	255	275	M6	315	360	22	410	442	14
12	82	325	347	M8	390	435	22	490	522	14
15	100	408	435	M10	495	560	30	635	680	22
18	125	500	535	M14	610	695	40	790	855	30
19	140	536	570	M14	645	730	40	830	895	30
21	145	584	620	M14	695	780	40	885	950	30
23	160	640	680	M16	770	870	45	990	1075	40
26	185	725	775	M20	880	1000	55	1135	1235	45
30	203	755	810	M24	970	1090	55	1235	1335	45
35	255	945	1000	M24	1140	1285	65	1450	1570	55

100-130 Series (Low-Inertia, High-Torque)

Size	A	B	C	D	E	F	G	H	I	J
15	112	408	435	M10	495	560	30	635	680	22
17	125	450	482	M12	550	620	32	710	775	30
18	140	500	535	M14	610	695	40	790	855	30
21	160	584	620	M14	695	780	40	885	950	30
23	185	640	680	M16	770	870	45	990	1075	40
26	205	725	775	M20	880	1000	55	1135	1235	45
30	230	810	865	M24	970	1090	55	1235	1335	45

Oil Immersed Clutches & Brakes



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

 **Industrial Clutch**[®]
Altra Industrial Motion

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.

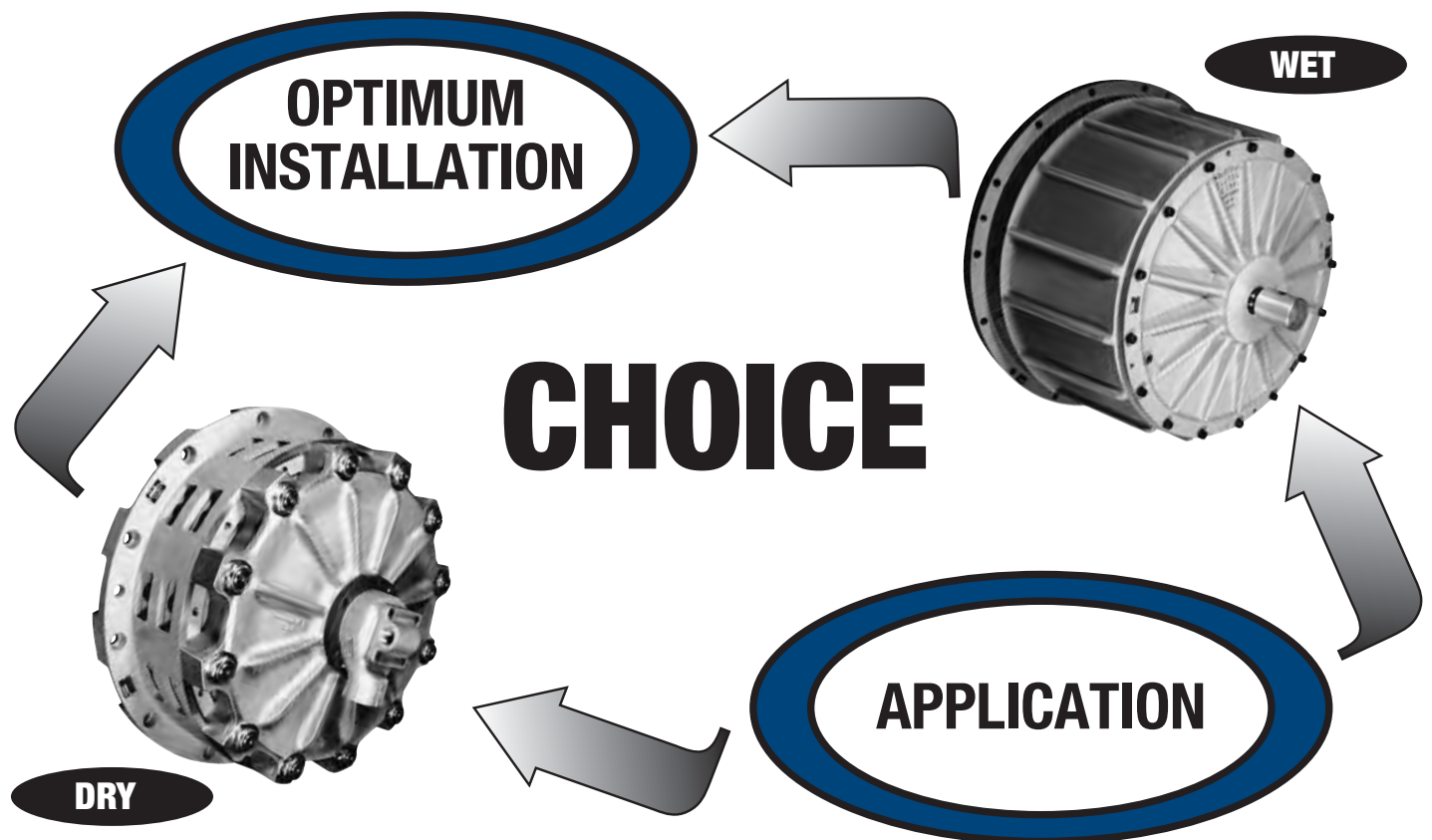
The wet units have repeatedly demonstrated their superior ability to operate with greater longevity than their dry counterparts by a factor of five to ten times, even though in many cases, the loads have increased significantly.

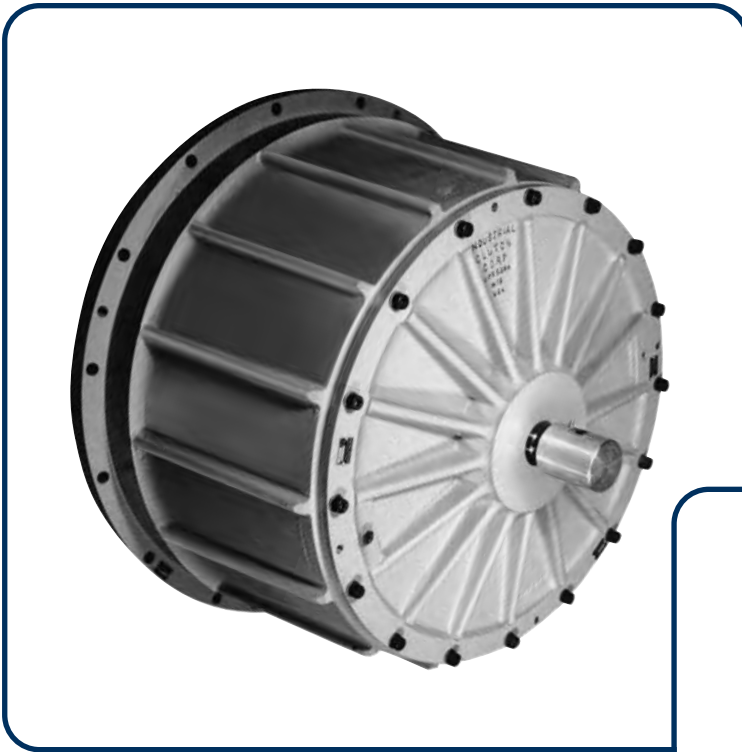
Even so, wet units are not applicable for all applications. It is not possible to compete with the simplicity, ruggedness, and economy our dry units offer for a majority of the marketplace.

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 CBA

*Air or Hydraulically
Actuated Combination
Clutch-Brakes*

Air or Hydraulically Actuated Combination Clutch-Brakes

Performance Characteristics

The Model CBA combination clutch-brake is an oil immersed unit utilizing a self-contained cooling oil-pumping system and capable of being actuated with either air or hydraulic pressure.

These units are capable of running at higher speeds and greater cyclic frequencies without exhibiting the wear magnitudes common to clutches and brakes operating dry under the same conditions.

Reliability and longevity are the most important features these units have to offer and find usage in those applications requiring the ultimate in clutch and brake performance.

Over fifteen years of field experience shows intervals between maintenance and machine up-time to increase from five to ten times when oil immersed units are used as direct replacements for those operating dry.

Industrial Clutch Products has designed the Model CBA unit to be compatible and complementary to their dry unit products. All of the desirable features found in our standard line of dry products were carefully evaluated for incorporation into the Model CBA combination clutch-brake. The user may expect the lowest possible driven inertia; fastest speed of response; and the same high quality ruggedness contained throughout our product line.

The Model CBA unit was designed and developed as part of our commitment to provide wide ranging and usefully products for the heavy industrial marketplace. As a result, the user has another level of performance from which to choose.

Advantages

- Combination clutch-brake eliminates the possibility of clutch-brake fight
- Increase in lining life
- Increase in reliability
- Impervious to adverse operating environments
- Low driven inertia
- Fast speed of response
- High cyclic rates
- High operating speeds

Usages

Original Equipment

Machinery which requires unusually high performance levels, operating speeds, or cyclic rates as well as precise stopping accuracy.

Conversions

Machinery which is being upgraded to increase production rates, reliability, or capacity.

Model CBA Combination Clutch-Brake Description

Model CBA

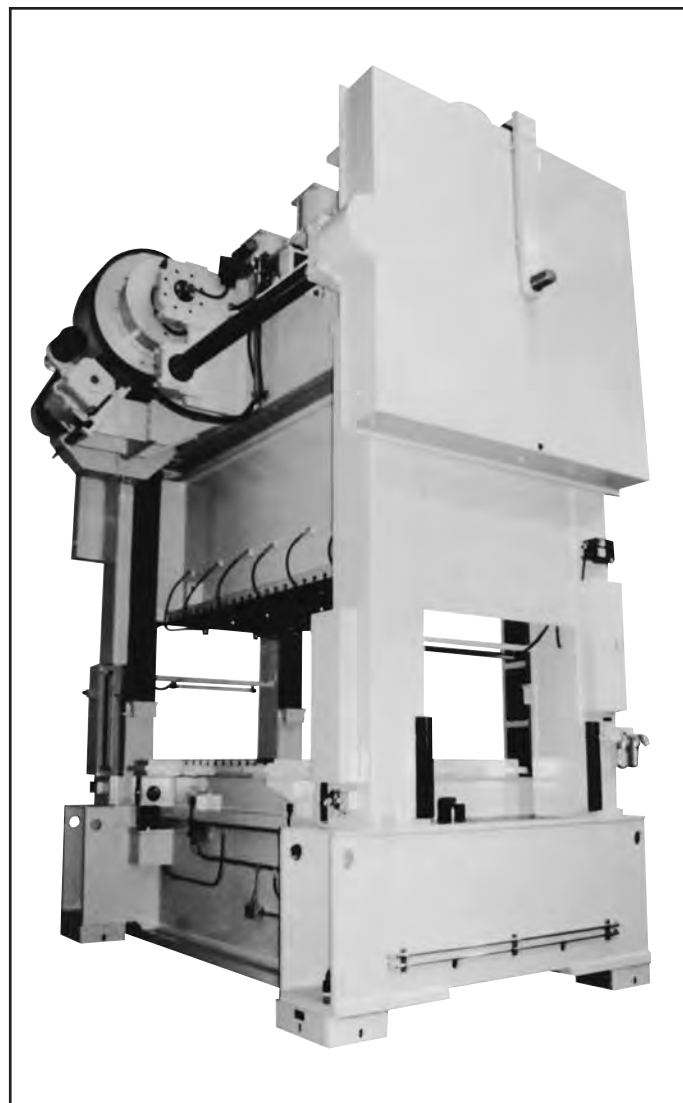
The Model CBA combination clutch-brake is arranged so that it may be mounted in an end of shaft or thru shaft configuration. The clutch driving members attach to the flywheel or input power source. The stationary brake members are attached to the machine frame by the use of a quill member which also provides a path for moving the cooling oil into and out of the unit if the application requires external cooling. Refer to Figure 1 and Figure 2 for a typical arrangement of an outboard (CBA) and thru-shaft (CBAT) mount respectively.

The unit may be provided with either pneumatic or hydraulic actuation chambers as well as a variety of brake springs to accommodate the clutch and brake torque requirements. Due to the many types of machinery configurations, it is often more convenient to house the clutch-brake unit within a user provided mounting cavity.

The Model CBA contains its own circulating oil-pumping system which pumps oil through the disc packs from either direction of rotation anytime the input member rotates. Refer to Figure 3. If external cooling is required due to the speed and cyclic rate, the internal pump output is directed from the outer housing cavity through the quill/brake support member and external cooler where the engagement heat is removed. Cooled oil then returns through the quill/brake support member into the disk pack where engagement heat is transferred to the oil. This heated oil is then pumped back to the cooler for heat removal.

If external cooling is not required, the circulating pump simply moves the oil internally from the outer housing cavity through the disk packs and back to the housing cavity where the engagement heat is transferred through the rotating housing into the surrounding atmosphere.

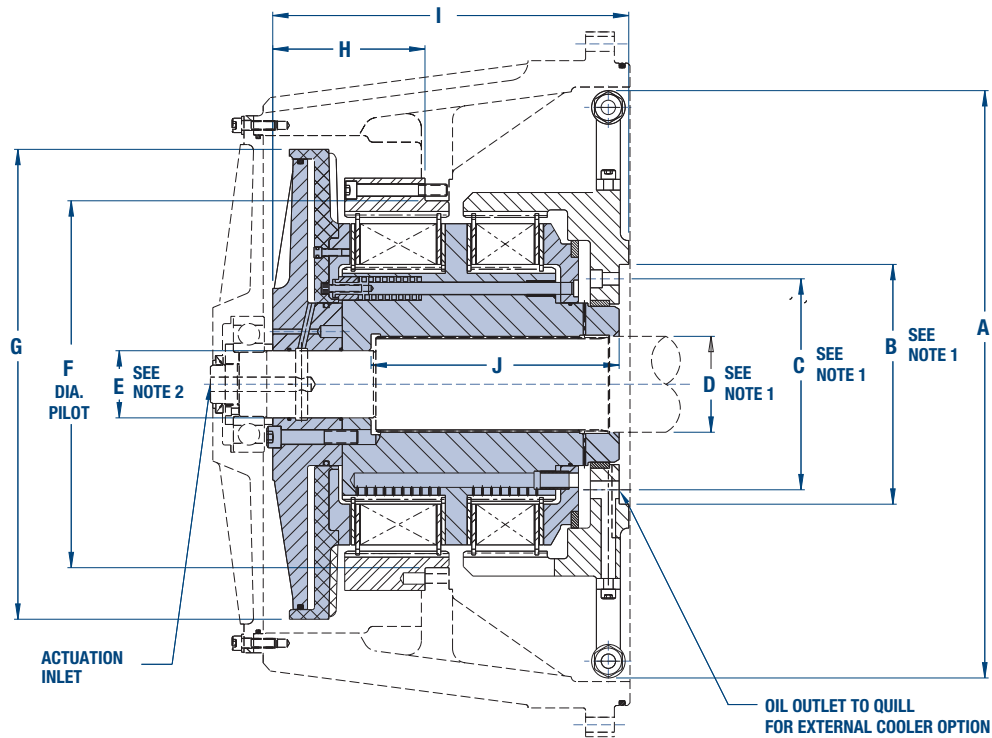
The ability of the CBA combination clutch-brake to continuously and effectively transfer large quantities of energy in this manner is the key factor to its longevity and reliability. In addition, the torque transmitting members, being lubricated by this same oil, are well suited to carry the heavy duty industrial type loads these models encounter.



Model CBAT-32580 Combination Clutch-Brake installed on a PTC 600-ton 144 x 60 transfer press.

Photo courtesy of Press Technology Corp.

Model CBA



DIMENSIONAL DATA

All dimensions in inches

CBA Model	A	B	C	D	E	F	G	H	I	J
CBA-3008	16.88	5.906	5.250	3.625	2.000	11.50	13.75	3.83	9.58	5.68
CBA-4512	16.88	5.906	5.250	3.625	2.000	11.50	13.75	5.21	10.96	7.00
CBA-6015	16.88	5.906	5.250	3.625	2.000	11.50	13.75	6.58	12.33	8.31
CBA-10025	21.63	8.661	7.625	5.000	3.000	15.250	18.25	5.51	12.90	7.65
CBA-14035	21.63	8.661	7.625	5.000	3.000	15.250	18.25	6.20	14.31	9.06
CBA-18045	21.63	8.661	7.625	5.000	3.000	15.250	18.25	6.90	15.72	10.47
CBA-24060	30.63	13.750	12.250	7.000	3.500	19.125	24.50	5.45	14.82	9.37
CBA-32580	30.63	13.750	12.250	7.000	3.500	19.125	24.50	6.06	16.69	11.24
CBA-400100	30.63	13.750	12.250	7.000	3.500	19.125	24.50	7.94	18.56	12.86
CBA-550150	34.13	13.500	12.000	9.250	5.625	34.750	30.50	8.87	22.88	16.01
CBA-650175	34.13	13.500	12.000	9.250	5.625	34.750	30.50	8.87	25.48	18.61
CBA-800200	34.13	13.500	12.000	9.250	5.625	34.750	30.50	8.87	28.08	21.21

NOTES: 1.) Industrial Clutch Prod. will supply shaft end detail and quill support drilling and machining details. Dimensions may be modified to suit customer requirements.
2.) End of shaft only. For thru-shaft designs, consult engineering.

3.) Use certified drawing dimensions only for final layouts.
4.) DXF and IGES files available upon request.
5.) Dimensions subject to change without notice.

OPERATIONAL DATA

CBA Model	Static Clutch Torque (lb.-in.)	Dynamic Brake Torque (lb.-in.)	Actuation Volume (in. ³)	Weight Outer (lbs.)	Weight Inner (lbs.)	Weight Stationary (lbs.)	WR ² Outer (lb.-ft. ²)	WR ² Inner (lb.-ft. ²)	Maximum Speed (RPM)
CBA-3008	30,000	8,000	24	30	103	43	6.5	9.5	1,200
CBA-4512	45,000	12,000	29	41	118	46	8.3	10.6	1,200
CBA-6015	60,000	15,000	33	48	132	52	9.5	11.6	1,200
CBA-10025	100,000	25,000	100	56	253	137	21	44	1,000
CBA-14035	140,000	35,000	100	70	288	151	26	49	1,000
CBA-18045	180,000	45,000	100	101	322	166	39	54	1,000
CBA-24060	240,000	60,000	175	101	635	248	55	204	800
CBA-32580	325,000	80,000	175	148	699	272	83	219	800
CBA-400100	400,000	100,000	175	212	763	296	123	235	800
CBA-550150	550,000	150,000	415	494	978	445	648	495	800
CBA-650175	650,000	175,000	444	549	1093	500	688	537	800
CBA-800200	800,000	200,000	473	604	1208	555	728	697	800

NOTES: 1.) Operating pressure: 80 PSIG air.

2.) Torque capacities can be modified. Higher torques may be obtained using hydraulic actuation. Consult engineering.

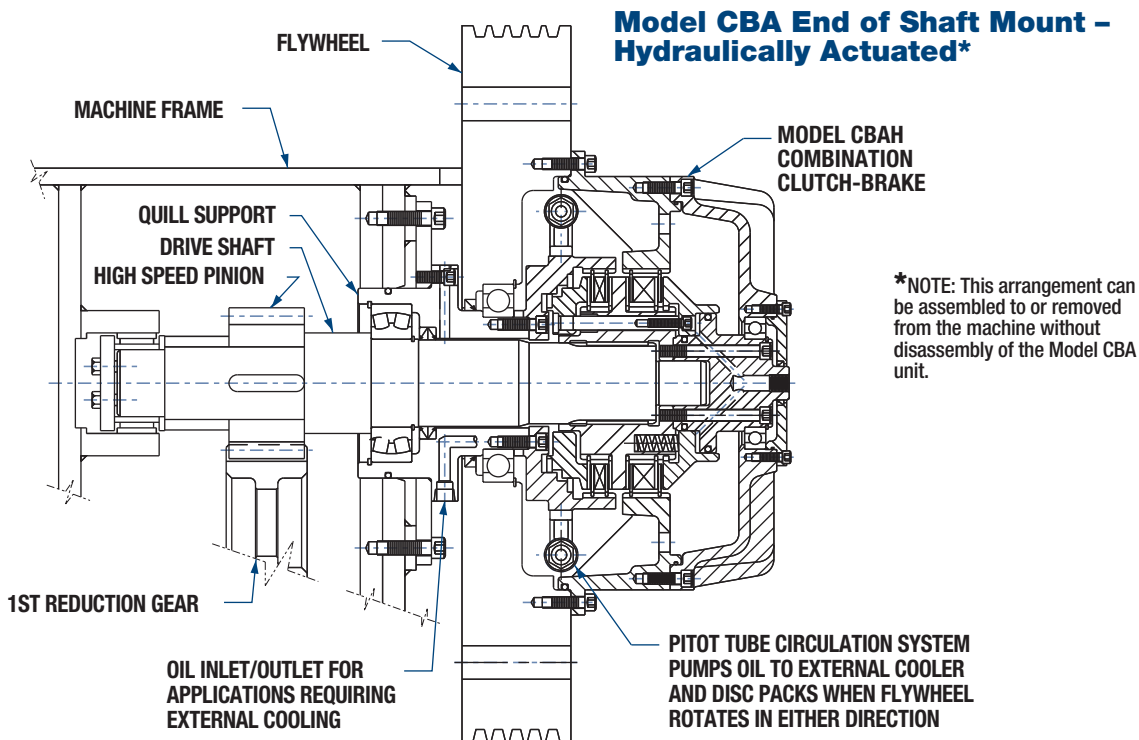


Figure 1

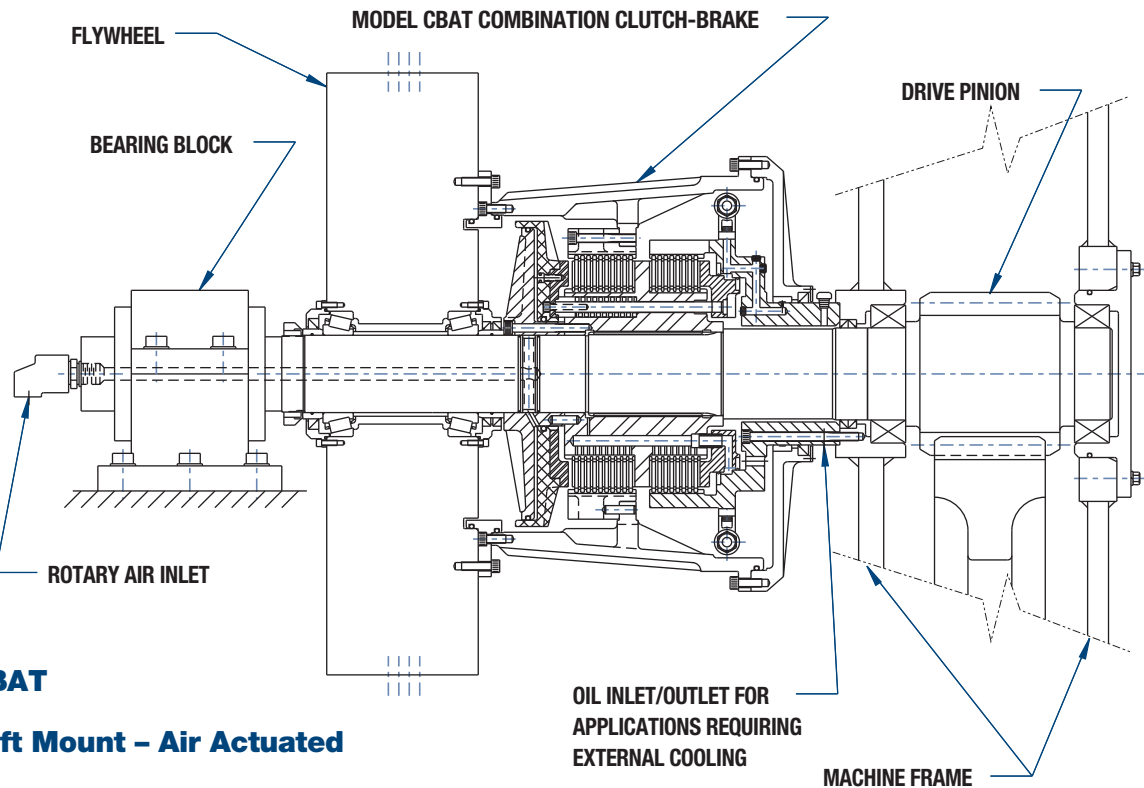


Figure 2

MODEL CBA COMBINATION CLUTCH-BRAKE

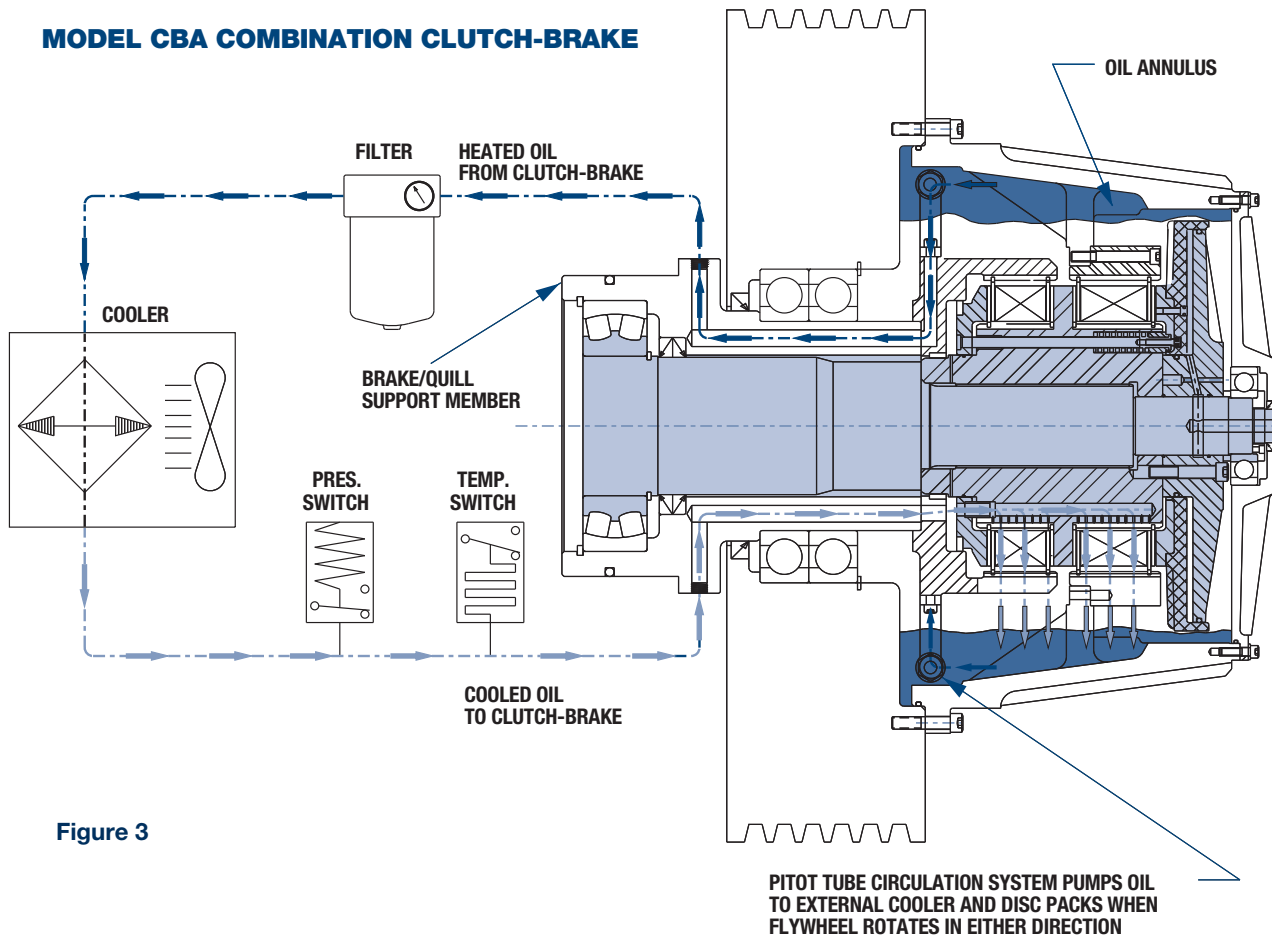


Figure 3

EXTERNAL COOLING DESCRIPTION

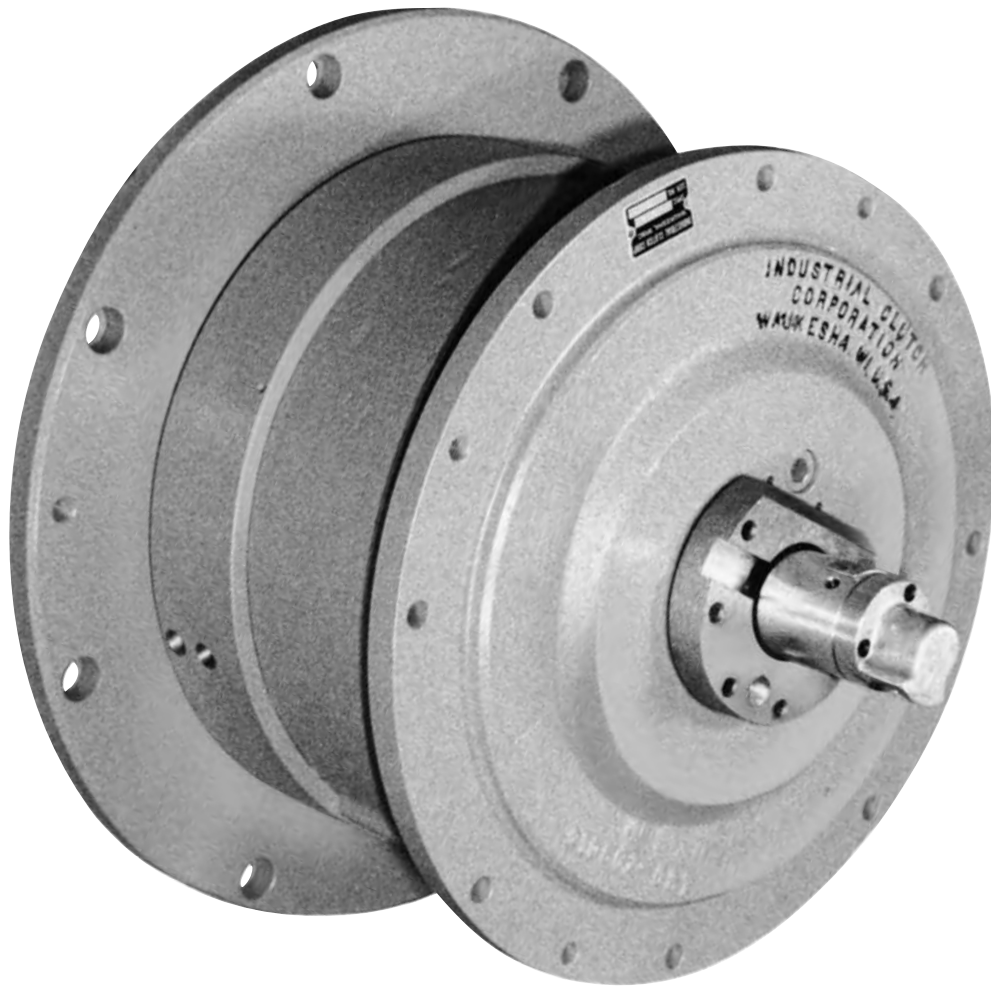
For those applications requiring external cooling, the pitot tubes scoop the oil from the rotating column of oil within the Model CBA's housing. This oil contains the engagement heat. The oil is then pumped through passages drilled in the quill support member to an external cooler where the engagement heat is removed.

Cooled oil returns through a different set of passages drilled in the quill support member and is introduced through a series of orifices directly into the center of the clutch and brake disc packs. This oil provides the cooling for the disc packs and removes the heat of engagement. It then returns to the rotating column of oil where it is again removed by the pitot tube pumping system and delivered to the cooler to repeat the cycle.

The pitot tube pumping system is bi-directional and will pump oil from either direction of rotation. Being a centrifugal type pumping system, its output is related to the speed of rotation. This closely matches the cooling oil requirements of the disc packs since engagement energy is also related to the speed of rotation in the same manner.

This system is completely self-contained and requires no additional energy source other than housing rotation. If the housing rotates, oil is delivered in the correct quantity to the disc packs.

Low pressure switches, (indicating low oil level), as well as hi-temperature switches, and a filter are normally included as shown to complete the cooling lube system.



MODEL CBH

*Hydraulically Actuated
Combination Clutch-Brakes*

HYDRAULICALLY ACTUATED COMBINATION CLUTCH BRAKES

PERFORMANCE CHARACTERISTICS

The Model CBH combination clutch-brakes are oil immersed units with a spring-set brake and hydraulically-set clutch. They are designed for extremely high cyclic rates for those applications that require a high degree of stopping accuracy.

Hydraulic actuation and precise control of piston stroke contributes to an extremely fast response time. Actuation volumes are very small and the fluid's virtually incompressible characteristics means that piston motion times can be as low as 5-10 milliseconds. This, in turn, allows for more shaft motion time since the unit's hydraulic response time is so short. Cyclic rates in excess of 300 cpm are not uncommon in some applications.

Such high cyclic rates induce a large amount of shock into the mechanical components of the unit and all members are designed to withstand these loads. Oil films help dampen these shock loads and the units run extremely quiet. Noise reductions of 25-30 dBA are common when compared to dry units running at the same speeds and loads.

Wear characteristics of the Model CBH units are excellent. As an example, a Model CBH unit was endurance tested for 120 million cycles with no appreciable lining or mechanical wear. A dry unit was tested at the same load level and found to require re-line every 10 million cycles and complete unit replacement every 30 million cycles. There are many industrial applications which require longevities of this magnitude and the Model CBH has exhibited outstanding performance in these instances.

ADVANTAGES

- Combination clutch-brake eliminates the possibility of clutch-brake fight
- Hydraulic actuation for fastest possible response time
- Impervious to adverse operating environments
- Wear virtually non-existent
- Capable of high cyclic rates
- Extreme stopping accuracy
- Quiet operation

Usages

Original Equipment

Machinery which cycle frequently and requires the ultimate performance with respect to life, reliability, and accuracy.

Conversions

Machinery which is being upgraded to increase production or reliability.



Strippit® Model 1250M
30 metric ton turret punch press uses a
Model CBH-4010 clutch-brake

Photo courtesy of Strippit, Inc., a unit of IDEX Corp.

MODEL CBH COMBINATION CLUTCH-BRAKE DESCRIPTION

The Model CBH combination clutch-brake is arranged so it may be mounted in an end of shaft configuration. The unit is a self-contained, oil cooled, hydraulically actuated combination clutch-brake designed for maximum capacity in a minimum space envelope.

The unit contains a spring-set, hydraulically released brake while the clutch is hydraulically-set and spring released. The clutch-brake action is effected using a shuttling piston actuator which moves axially to engage either the clutch or brake. With this arrangement, it is not possible to engage the clutch and brake simultaneously so the possibility of clutch or brake overlap does not exist.

The Model CBH unit uses only the best quality, time proven materials within its construction and has been designed to withstand the rigors of high cyclic applications with precision machined clearances to provide the fastest possible response.

The input flywheel, gear, or sheave is bolted to the clutch driving member. The clutch-brake's inner or driven members can be keyed or splined to the machine's output shaft for maximum driving torque capacity and ease of assembly.

The stationary brake member is affixed to the machine frame through the use of a brake mounting plate provided by others.

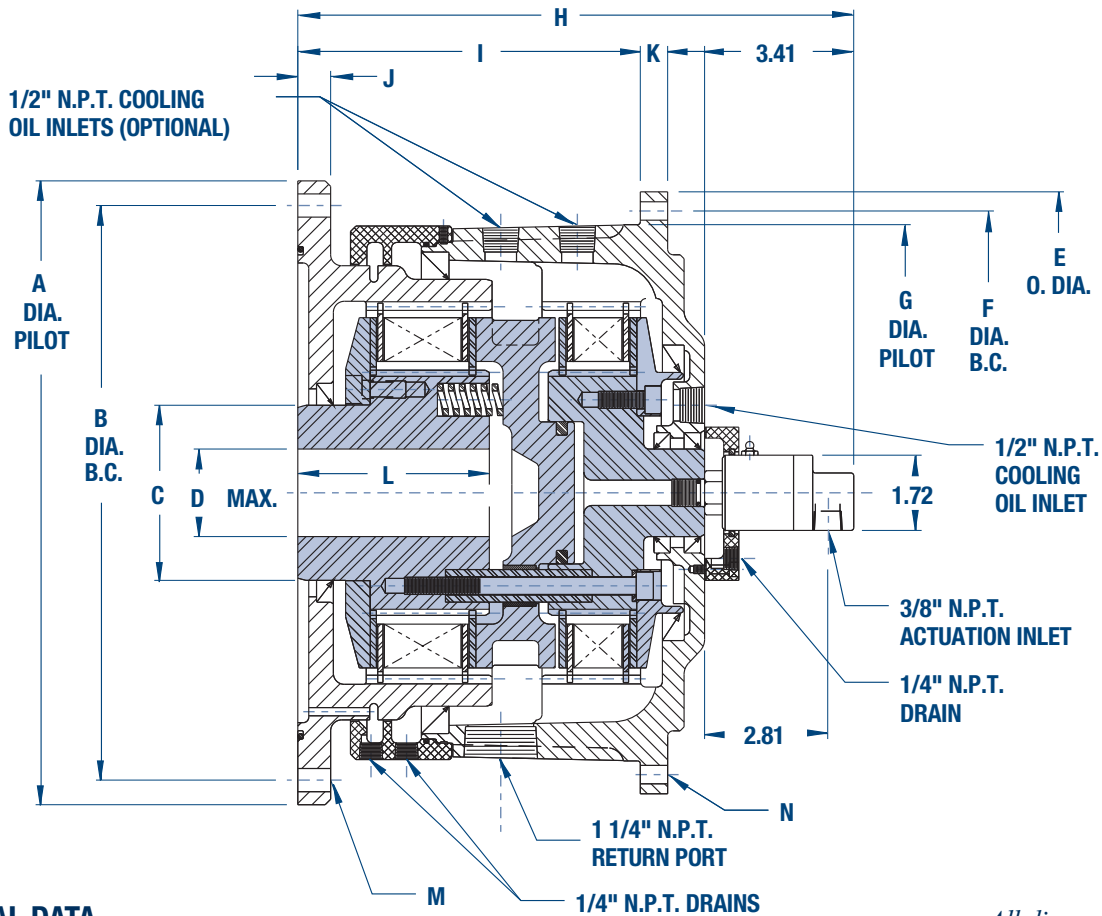
The stationary housing provides a sealed cavity for the oil that cools the clutch-brake disc packs. If external cooling is required, this housing provides a means of oil entry and exit.

The cylinder/piston actuating mechanism has been integrated within the brake hub thereby eliminating the need for cross-drilling actuation oil entry holes in the user's shaft and the possibility of leakage at the hub/shaft interface.

The Model CBH combination clutch-brake is available with high-cyclic rate actuation valve assemblies as well as combination cooling/actuation hydraulic power units.

Model CBH

HYDRAULICALLY ACTUATED COMBINATION CLUTCH/BRAKE



DIMENSIONAL DATA

All dimensions in inches

CBH Model	A	B	C	D	E	F	G	H	I	J	K	L	M	N
3010	16.498	15.25	4.00	2.938	13.75	12.88	12.249	12.71	7.79	0.75	0.63	4.38	(8) .53	(8) .42
4010	16.498	15.25	4.00	2.938	13.75	12.88	12.249	12.71	7.79	0.75	0.63	4.38	(8) .53	(8) .42
5010	16.498	15.25	4.00	2.938	13.75	12.88	12.249	13.00	8.08	0.75	0.63	4.67	(8) .53	(8) .42
5012	16.498	15.25	4.00	2.938	13.75	12.88	12.249	13.32	8.40	0.75	0.63	4.67	(8) .53	(8) .42
6012	16.498	15.25	4.50	3.500	15.88	15.00	14.374	12.56	7.52	0.75	0.63	4.11	(8) .53	(8) .42
7015	16.498	15.25	4.50	3.500	15.88	15.00	14.374	13.20	8.16	0.75	0.63	4.11	(8) .53	(8) .42
8015	16.498	15.25	4.50	3.500	15.88	15.00	14.374	13.52	8.48	0.75	0.63	4.43	(8) .53	(8) .42
9020	16.498	15.25	4.50	3.500	15.88	15.00	14.374	14.16	9.12	0.75	0.63	4.75	(8) .53	(8) .42

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

CBH Model	Clutch Torque (lb.-in.)	Brake Torque (lb.-in.)	Actuation Volume (in. ³)	Weight Outer (lbs.)	Weight Inner (lbs.)	Weight Stationary (lbs.)	Weight Total (lbs.)	WR ² - Outer (lb.-ft. ²)	WR ² - Inner (lb.-ft. ²)	Maximum Speed (RPM)
3010	30,000	10,000	0.75	44.0	63.1	74.9	182	7.06	3.35	800
4010	40,000	10,000	0.75	46.0	69.1	74.9	190	7.24	3.64	800
5010	50,000	10,000	0.75	47.0	72.0	74.9	194	7.33	4.03	800
5012	50,000	12,000	0.75	47.0	75.1	75.9	198	7.33	4.17	800
6012	60,000	12,000	1.13	69.0	87.0	106.8	263	15.52	5.85	675
7015	70,000	15,000	1.13	70.3	95.4	106.8	273	15.80	6.40	675
8015	80,000	15,000	1.13	71.6	99.6	106.8	278	16.10	6.70	675
9020	90,000	20,000	1.13	72.9	108.3	108.1	289	16.30	7.30	675

NOTES: 1.) Operating pressure: 900 PSIG
2.) Torque capacities can be modified. Consult engineering.

TURRET PUNCH PRESS APPLICATION

Turret punch presses are one example of where the Model CBH combination clutch-brakes have been successfully applied on both original equipment as well as conversion replacement for dry clutch and brakes on existing machines.

Turret punch presses are ideal candidates for the operating features the Model CBH units can offer.

- Longevity
- Reliability
- Quiet Operation
- Stopping Accuracy
- Repeatable Operation
- Greatly Reduced Maintenance
- Increased Machine Productivity
- Eliminates Wear Particles
- Eliminates Air Lubricator Mist
- Eliminates Air Supply Requirements

Industrial Clutch Products has been building clutches and brakes for turret punch presses for over 30 years and can state without reservation that the Model CBH combinations clutch-brake provides the user of this equipment an extremely high degree of reliability and longevity.

CONVERSIONS

The Model CBH is ideally suited for conversion of older turret punch presses to state-of-the-art oil immersed clutch-brakes.

Virtually all of the turret punch presses in the field today can be converted within a minimal timeframe and budget. Paybacks are rapid and in most older installations the machine productivity can be increased by over 50%.

The process involves obtaining a conversion kit provided by Industrial Clutch Products (see photos). The existing clutch-brake or separate clutch and brake is removed and a mounting bracket is fabricated to attach the Model CBH to the press frame. Crankshaft mounting diameters and flywheel mounting bolt circles are preserved in the majority of cases as the Model CBH can be modified to mount directly to, or through the use of an adapter, directly to these surfaces. All former clutch and brake air equipment is stripped from the press frame as it is no longer required. The valve assembly is mounted and hoses and power are run to the hydraulic power unit. The control is modified to provide a single valve signal for both the clutch and brake function. The unit is then ready for operation. (Refer to schematic-next page.)

Industrial Clutch Products will be pleased to provide conversion assistance to ensure a successful installation.

*Model CBH
Clutch-Brake*



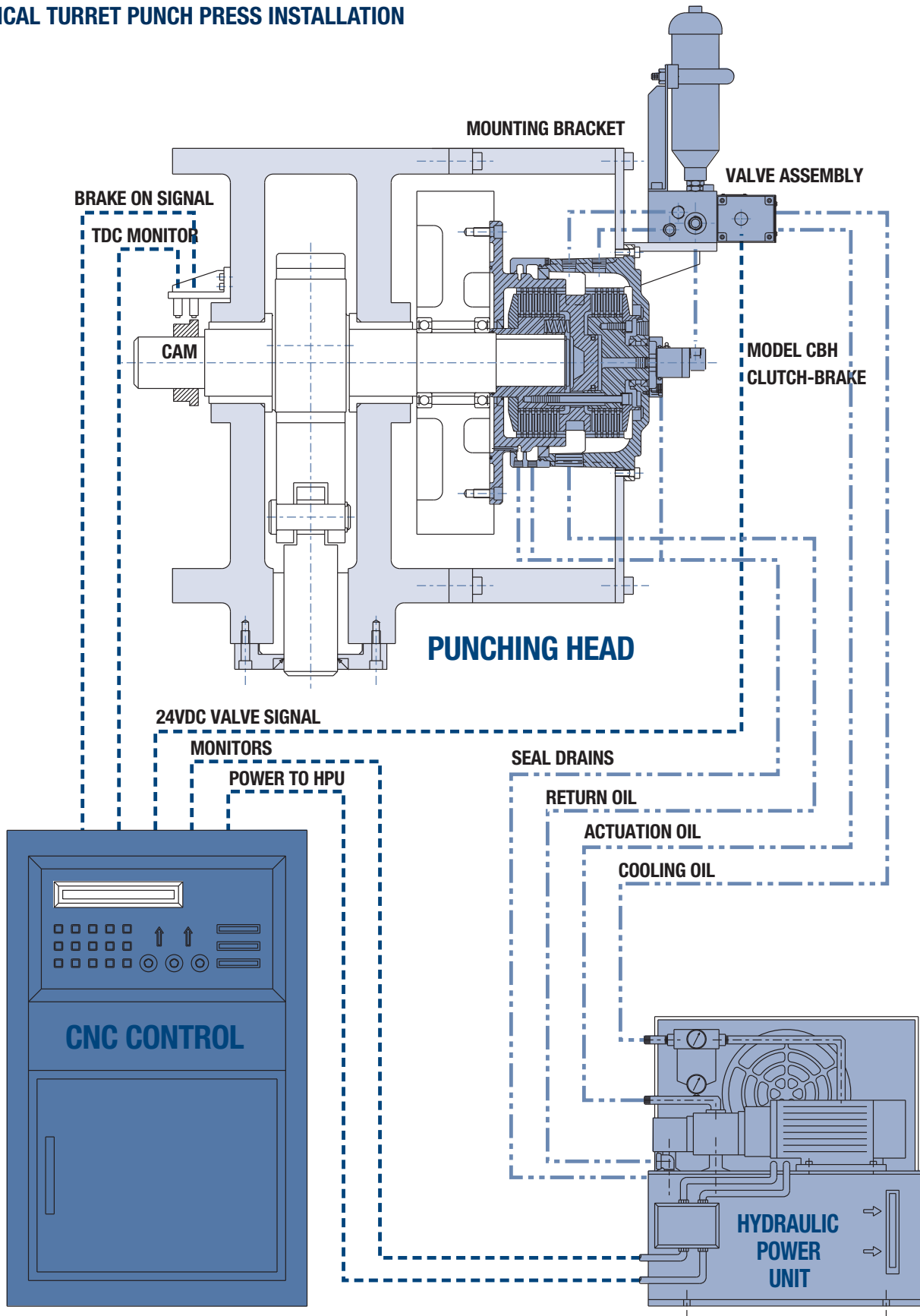
*Actuation
Valving*



*Hydraulic
Power Unit*

Model CBH

TYPICAL TURRET PUNCH PRESS INSTALLATION





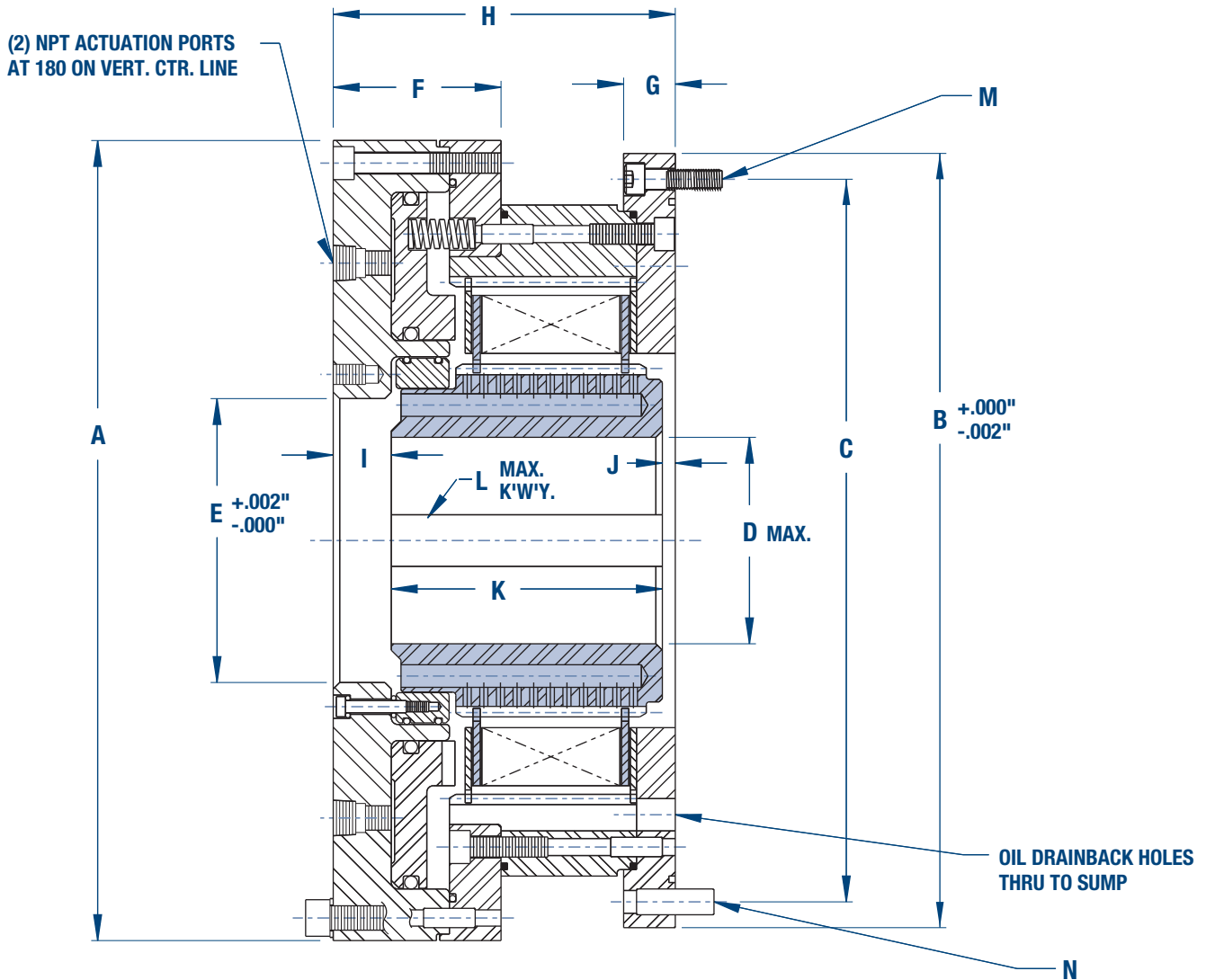
MODEL HBA

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. ³)	Weight Outer (lbs.)	Weight Inner (lbs.)	WR ² Inner (lb.-ft. ²)	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

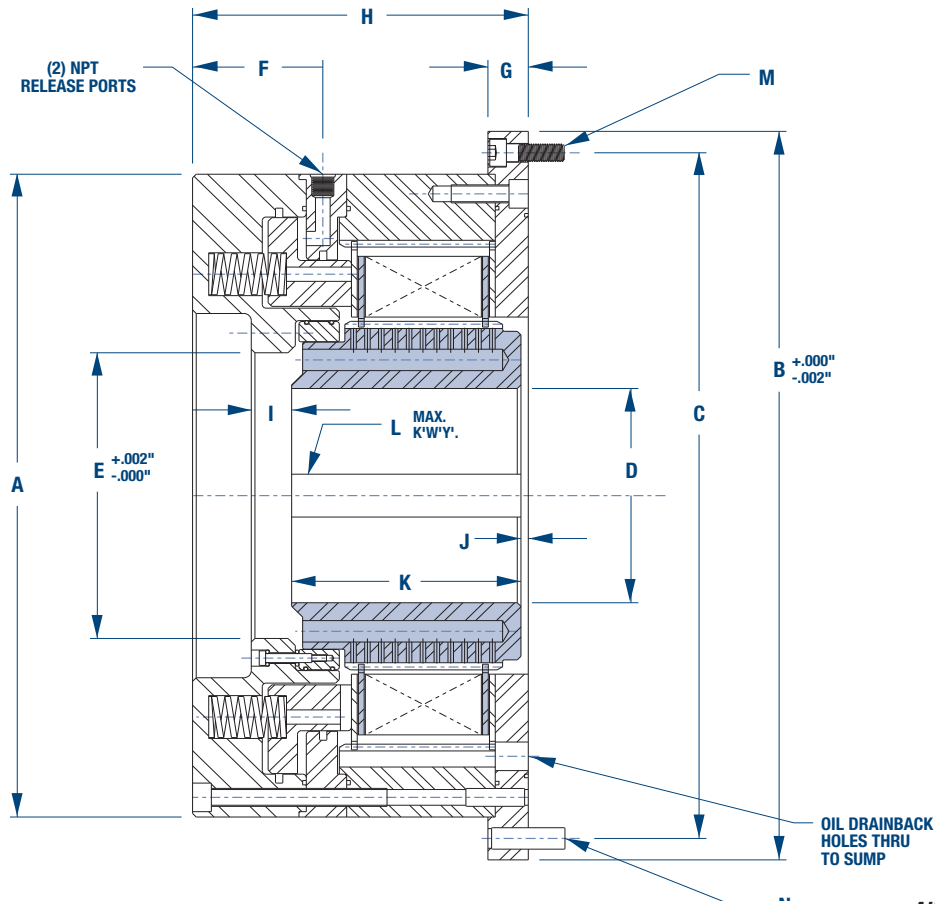
The Model HBS brake is identical to the Model HBA brake except the unit is designed to be spring-set and hydraulically or pneumatically released. This spring-set actuation feature makes it suitable for those applications requiring the presence of full braking torque in the event of loss of actuation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

Model HBS



DIMENSIONAL DATA

All dimensions in inches

Brake Model	A	B	C	D	E	F	G	H	I	J	K	L	M	N
HBS-8	11.25	12.875	12.000	3.000	4.000	2.75	0.84	7.25	0.75	0.19	4.94	0.75	(8) - M8	(4) - .375
HBS-10	14.25	15.000	14.000	4.000	5.500	3.38	1.00	8.13	1.13	0.25	5.25	1.00	(8) - M10	(4) - .500
HBS-13	17.50	19.750	18.750	6.000	8.750	3.69	1.13	9.00	1.19	0.13	6.00	1.50	(8) - M12	(4) - .500
HBS-15	20.50	23.500	22.000	6.500	8.750	4.31	1.38	11.31	1.25	0.25	7.81	1.50	(8) - M16	(4) - .625
HBS-16	22.50	25.500	24.000	7.500	10.000	4.44	1.38	11.44	1.38	0.25	7.81	1.50	(8) - M16	(4) - .750
HBS-18	25.50	28.750	27.000	8.000	11.000	4.88	1.75	14.38	1.56	0.50	9.78	1.50	(8) - M20	(4) - 1.00
HBS-20	27.50	31.000	29.250	9.000	12.000	5.31	1.88	14.88	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. ³)	Weight Outer (lbs.)	Weight Inner (lbs.)	WR ² Inner (lb.-ft. ²)	Maximum RPM
HBS-8	37,000	26,500	6.8	135	29.5	1.43	4,000
HBS-10	63,000	45,000	12.3	205	48.2	3.48	3,400
HBS-13	168,000	120,000	24.1	370	106.0	15.75	2,400
HBS-15	245,000	175,000	32.0	650	152.0	28.68	2,150
HBS-16	350,000	250,000	42.2	815	197.0	45.30	1,900
HBS-18	455,000	325,000	62.0	1250	308.0	84.80	1,750
HBS-20	605,000	435,000	86.0	1495	356.0	119.40	1,600

NOTES: 1.) Release pressure: 400 PSIG
2.) Torque capacities can be modified. Consult engineering.
3.) Consult factory for service factors required per application.

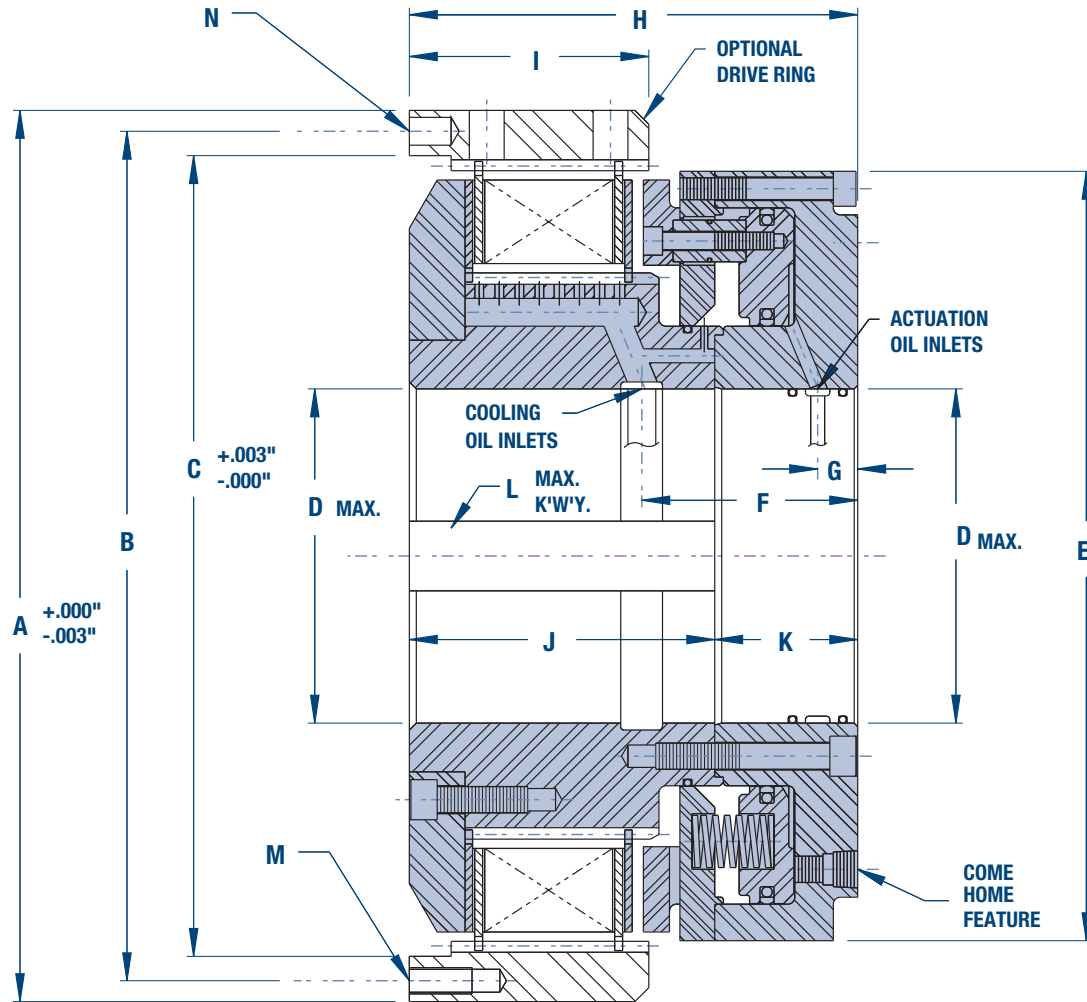
4.) Consult factory for oil flow requirements per application and shaft oil hole diameters.



MODEL HC

Air or Hydraulically Actuated Clutches

Model HC



DIMENSIONAL DATA

All dimensions in inches

Clutch Model	A	B	C	D	E	F	G	H	I	J	K	L	M	N
HC-8-6	10.124	9.4375	8.750	3.000	8.31	3.50	0.51	6.35	2.68	3.81	2.57	0.75	(12) - M10	(4) - .375
HC-8-8	10.124	9.4375	8.750	3.000	8.31	3.50	0.51	6.95	3.28	4.40	2.57	0.75	(12) - M10	(4) - .375
HC-8-10	10.124	9.4375	8.750	3.000	8.31	3.50	0.51	7.55	3.88	5.00	2.57	0.75	(12) - M10	(4) - .375
HC-10-6	11.749	11.063	10.375	4.000	9.81	3.63	0.56	6.54	2.97	4.24	2.38	1.00	(12) - M10	(4) - .500
HC-10-8	11.749	11.063	10.375	4.000	9.81	3.63	0.56	7.25	3.61	4.88	2.38	1.00	(12) - M10	(4) - .500
HC-10-10	11.749	11.063	10.375	4.000	9.81	3.63	0.56	7.89	4.25	5.52	2.38	1.00	(12) - M10	(4) - .500
HC-13-6	15.998	15.250	14.375	6.000	13.81	4.63	0.72	8.09	3.58	4.77	3.32	1.25	(12) - M12	(4) - .500
HC-13-8	15.998	15.250	14.375	6.000	13.81	4.63	0.72	8.80	4.30	5.48	3.32	1.25	(12) - M12	(4) - .500
HC-13-10	15.998	15.250	14.375	6.000	13.81	4.63	0.72	9.52	5.02	6.20	3.32	1.25	(12) - M12	(4) - .500
HC-15-6	18.373	17.375	16.375	6.500	15.75	4.43	0.78	9.06	4.37	5.74	3.31	1.50	(12) - M16	(4) - .625
HC-15-8	18.373	17.375	16.375	6.500	15.75	4.43	0.78	10.00	5.31	6.68	3.31	1.50	(12) - M16	(4) - .625
HC-15-10	18.373	17.375	16.375	6.500	15.75	4.43	0.78	10.94	6.25	7.62	3.31	1.50	(12) - M16	(4) - .625
HC-16-6	19.998	19.000	18.000	7.500	17.25	5.11	1.38	9.29	4.37	5.75	3.54	1.50	(12) - M16	(4) - .625
HC-16-8	19.998	19.000	18.000	7.500	17.25	5.11	1.38	10.23	5.31	6.69	3.54	1.50	(12) - M16	(4) - .625
HC-16-10	19.998	19.000	18.000	7.500	17.25	5.11	1.38	11.17	6.25	7.63	3.54	1.50	(12) - M16	(4) - .625
HC-18-6	21.998	20.750	19.500	8.000	18.63	5.50	1.50	10.96	5.82	7.13	3.81	1.50	(12) - M20	(4) - .750
HC-18-8	21.998	20.750	19.500	8.000	18.63	5.50	1.50	12.23	7.10	8.41	3.81	1.50	(12) - M20	(4) - .750
HC-18-10	21.998	20.750	19.500	8.000	18.63	5.50	1.50	13.51	8.38	9.69	3.81	1.50	(12) - M20	(4) - .750
HC-20-6	24.998	23.750	22.500	9.000	20.63	6.50	1.75	12.04	6.19	7.66	4.37	1.50	(12) - M20	(4) - .750
HC-20-8	24.998	23.750	22.500	9.000	20.63	6.50	1.75	13.32	7.47	8.94	4.37	1.50	(12) - M20	(4) - .750
HC-20-10	24.998	23.750	22.500	9.000	20.63	6.50	1.75	14.60	8.75	10.22	4.37	1.50	(12) - M20	(4) - .750

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

Clutch Model	Static Torque (lb.-in.)	Dynamic Torque (lb.-in.)	Act. Vol. (in. ³)	Weight Outer (lbs.)	Weight Inner (lbs.)	WR ² Outer (lb.-ft. ²)	WR ² Inner (lb.-ft. ²)	Maximum RPM
HC-8-6	55000	39285	4.37	5.8	60.6	0.54	3.79	2600
HC-8-8	70000	50000	4.37	7.7	65.3	0.72	4.04	2600
HC-8-10	85000	60715	4.37	9.7	70.0	0.90	4.29	2600
HC-10-6	75000	53570	7.3	7.8	83.7	1.03	7.50	2200
HC-10-8	100000	71430	7.3	10.4	90.7	1.38	8.06	2200
HC-10-10	125000	89285	7.3	13.0	97.6	1.72	8.60	2200
HC-13-6	200000	142860	14.0	16.3	195.7	4.3	35.9	1700
HC-13-8	275000	196430	14.0	21.7	212.1	5.8	38.4	1700
HC-13-10	350000	250000	14.0	27.1	227.4	7.2	40.8	1700
HC-15-6	325000	232140	19.0	34.0	291.0	11.0	67.0	1450
HC-15-8	425000	303570	19.0	45.0	316.0	15.0	72.0	1450
HC-15-10	525000	375000	19.0	57.0	341.0	19.0	77.0	1450
HC-16-6	450000	321430	24.0	40.0	357.0	16.0	103.0	1300
HC-16-8	600000	428570	24.0	54.0	387.0	22.0	110.0	1300
HC-16-10	750000	535715	24.0	67.0	417.0	27.0	118.0	1300
HC-18-6	570000	407140	31.0	63.0	494.0	30.0	164.0	1200
HC-18-8	700000	500000	31.0	84.0	544.0	40.0	179.0	1200
HC-18-10	950000	678570	31.0	105.0	594.0	50.0	194.0	1200
HC-20-6	765000	546430	51.0	95.0	671.0	58.0	279.0	1100
HC-20-8	1020000	728570	51.0	127.0	729.0	78.0	300.0	1100
HC-20-10	1275000	910710	51.0	158.0	786.0	97.0	321.0	1100

NOTES: 1.) Operating pressure: 350 PSIG

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.

5.) For static engagement applications dry lining units are available. Torque ratings are 3 times that shown. Consult Engineering

MODEL HC CLUTCH DESCRIPTION

The Model HC clutches are designed to be used in either an end of shaft or through shaft mounting configuration. Their compact size makes these units ideal for incorporating within a gear housing. Multiple speed transmissions use a variety of these units to effect fixed mesh speed changes or they may be used as a stand alone device for disconnect service.

The **Model** clutches may be provided with an internally splined outer drive ring for bolting to the user's driving or driven member, or **Model HC** clutches may be provided without this drive ring to allow the user to spline the inside diameter of their driving or driven member which conserves radial space and provides for design compactness.

All units incorporate provisions for forced oil cooling allowing high energy engagements to be made without causing thermal distress within the disc pack. Cooling oil is introduced from an external supply through axial holes provided in the shaft.

Actuation pressure may come from either a pneumatic or hydraulic source. This pressure may be introduced through an axial hole provided at the shaft centerline, or, in the case of end of shaft mounts, it may be introduced through a separate manifold.

All torque transmitting members are designed for continuous heavy duty industrial service. Hubs and drive rings are fully hardened and manufactured from high quality alloy steel forgings. Disc pack cores are designed using extra-heavy plate thicknesses to maximize torque and heat capacity.

Friction materials and grooving patterns are designed to provide for high thermal and smooth engagement performance. All of the Model HC clutches contain friction materials which allow for controlled slip operation in those applications requiring this feature. Slip operation is described on following pages.

Marine main propulsion forward/reverse or disconnect service, winch disconnects, conveyor or mill soft-starts, and multi-speed transmissions are but a few of the applications for the Model HC oil immersed clutches.

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