Airflex®

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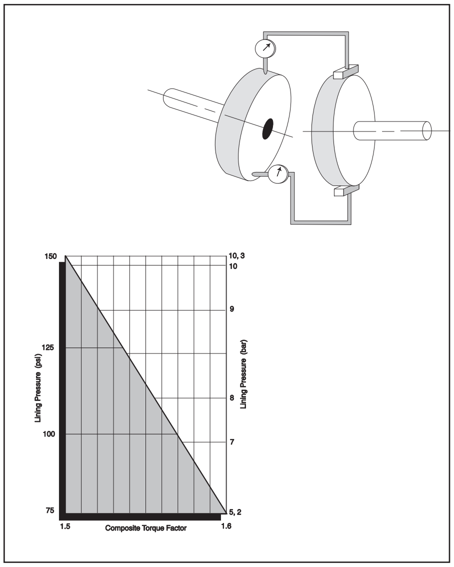
Technical Information Y



Clutches and Brakes

A **clutch** is a device which transfers energy from one rotating shaft to another in order to perform some useful work.

In the simplest terms, a clutch can be thought of as a starting device because that is what happens when a clutch is engaged. But, more importantly, while engaged it is transferring energy. The clutch takes energy from a power source such as an electric motor or engine and transfers it to where it is required. The power source is referred to as the **prime mover** because it is the primary source of energy. A clutch is engaged for one reason — to perform some useful work.

A clutch consists of two halves: a driving half and a driven half. The **driving half** is attached to the power source and rotates with it. The **driven half** is attached to the shaft requiring the energy and is started with each engagement. In addition, the clutch must have some means of engaging and disengaging the two halves.

A **brake** on the other hand, is a device which absorbs energy that is stored in rotating and linear moving components and/or prevents an energy transfer to them.

Again, in simple terms, a brake can be thought of as a stopping device because that is what happens when a brake is engaged. Like a clutch, a brake also consists of two halves: a driven half and a stationary half. The driven half is attached to the rotating and linear moving bodies from which energy must be removed. The stationary half is reacted so that it cannot move. The brake also has a means of engaging and disengaging the two halves.

Clutch and brake engagement is made by connecting the two halves. Since one half is rotating and the other is stationary, the halves will slide or slip relative to each other.

When there is no relative motion between clutch halves, the clutch is **locked** and maximum energy is transferred to the work shaft. If a relative speed differential is allowed to exist between each half, only partial energy is transferred, and the device is referred to as a **slip clutch**.

When there is no relative motion between brake halves, the brake is **set** and maximum energy has been removed from the driven components and/or prevents an energy transfer to them. The latter situation is referred to as a **holding brake**. If a relative speed differential is allowed to exist between each half, only partial energy is absorbed, and the device is referred to as a **drag or tension brake**.

The connection between clutch and brake halves for Airflex products is dependent upon a frictional couple. The product of the resulting frictional force and the distance to its axis of rotation determines the **torque capacity** of the clutch or brake. The torque magnitude determines the amount of energy that can be transferred by the clutch or absorbed by the brake.

Heat is generated whenever torque is transmitted and a speed differential exists between the clutch and brake halves. The ability to absorb and dissipate this heat determines the **thermal capacity** of the clutch and brake.

A clutch or barke must not only have sufficient torque capacity to transfer the required energy, but must also have sufficient thermal capacity to handle the heat generated due to slippage.

Clutch Functions

Depending upon the operating characteristics of the prime mover and the energy demand of the work shaft there are several ways in which the clutch can function to control the energy transfer. These include:

Coupling or disconnect function - This function permits the prime mover; e.g., synchronous motor or diesel engine, to obtain operating speed and/or temperature before being coupled or connected to the work shaft. When energy is not required at the work shaft and it is not desirable to stop the prime mover, the clutch provides the disconnect between the two. This type of application is usually associated with long periods of clutch engagement, as opposed to one in which several engagements per minute are required.

Starting function - This function permits controlled acceleration of the work shaft with minimal torsional shock when accelerating delicate or breakable materials. For high inertia starts, it permits the prime mover to run continuously at efficient speeds.

Direction or speed change function - Multi-speeds and direction changes in many machines are accomplished with gear boxes or gear trains equipped with clutches. Generally, one clutch is required for each speed and for changing the direction of rotation.

Cyclic function - This function requires that the work shaft be started very frequently while the prime mover runs continuously. A brake is usually required to stop the work shaft in order to obtain the cyclic rate.

Continuous slip function - This function requires that the driven side of the clutch rotate at a speed slower than the driver side. An application example is a rewind stand where material must be wound into a roll under constant tension. By controlling clutch torque, material tension and speed is regulated.

Overload protection function - This function limits the maximum torque which can be transmitted to prevent damage to drive components. If the maximum torque is exceeded, the clutch will slip. To avoid thermal damage, the clutch should not be allowed to slip for an excessive length of time.

In any given application, the clutch can provide one or a number of these functions. It is important to identify all the functions the clutch will be subjected to in the clutch selection process.

Brake Functions

Brake functions closely parallel those of clutches, except they absorb and dissipate the energy of the driven shaft. These functions include:

Holding function - In this function the brake prevents an energy transfer to the driven shaft by holding it stationary. Holding may be required either prior to or after shaft rotation and is usually required to maintain or hold a position or location of a driven component.

Stopping function - This function provides a means to stop the driven shaft or machine in a controlled manner by limiting its coasting time and/or distance. The brake torque determines how quickly the stop occurs.

Emergency stop function - This function rapidly stops all moving components to protect the operator and/or equipment.

Cyclic function - This function works in conjunction with a clutch and permits frequent starting and stopping of the driven shaft while allowing the prime mover to run continuously.

Controlled slip function - This function provides a retarding torque to the driven shaft. There is constant relative movement between the driven shaft and the stationary half of the brake.

In any given application, the brake can provide one or a number of these functions. It is important to identify all the functions the brake will be subjected to in the brake selection process.

Friction

Friction may be defined as the resistive force occurring between two surfaces as they slide or tend to slide across each other. The contacting surfaces can be thought of as consisting of peaks and valleys which mesh together. The resistive force is developed by the effort to slide the peaks on each surface out of the meshing valleys. When the surfaces slide across each other, the peaks and valleys are not as free to mesh as when the surfaces are stationary. This is the reason frictional resistance decreases when sliding occurs.

It is apparent that the frictional resistive force is proportional to the normal force holding the surfaces in contact with each other. The constant of proportionability is called the **dynamic coefficient of friction** when sliding occurs, and the **static coefficient of friction** when sliding does not occur.

Friction Couples

Airflex clutches and brakes are of the frictional type, that is, they rely upon a frictional force occurring between two surfaces to develop the required torque. The torque is called **dynamic torque** when slippage occurs between the surfaces and **static torque** when no slippage occurs. Usually the two surfaces are of dissimilar materials. The combination of the two materials used is referred to as a **friction couple** and their contacting surfaces as **interfaces**.

When the friction couple operates within a fluid, it is referred to as **wet operation**. **Dry operation** does not depend upon the presence of fluid.

Depending upon application, it may be desirable to have a large or small differential between the static and dynamic torques. For instance, when engagement is made at rest (no slippage between interfaces), as would occur for a clutch-coupling or a holding brake, the static torque should be more dominant. If the clutch or brake is required to slip continuously, as in a tensioning application, very little differential is desirable to avoid a **stick-slip** condition.

Friction Material

Friction material is a specially formulated composition intended to provide a specific stable coefficient of friction over a wide range of operating temperature. Beyond the operating range, the coefficient drops drastically, resulting in loss of frictional force. This condition is referred to as **lining fade**.

Some desirable friction material properties are that they have good wear life, be non-aggressive to the surface they interface with, and have sufficient strength so they can be attached to other components. The material is basically a replaceable lining to which wear can be confined; hence, the description **friction lining**.

The majority of Airflex products utilize **organic linings**. Their composition consists of three types of ingredients - fillers, fibers and binders. Fillers, in addition to providing bulk and assisting in material processing, are used as augmenting agents to affect the coefficient of friction. Fibers are used for reinforcement. Binders bond all the ingredients together.

Airflex magnetic and hydraulic products utilize **sintered metal lining**. This friction material is produced from a mixture of powdered metals and non-metals by pressure and fusion. Its primary advantage is being able to withstand high thermal stresses and operating pressures. To assist in heat dissipation, and because of its aggressiveness on its mating surface, it is intended for wet operation.

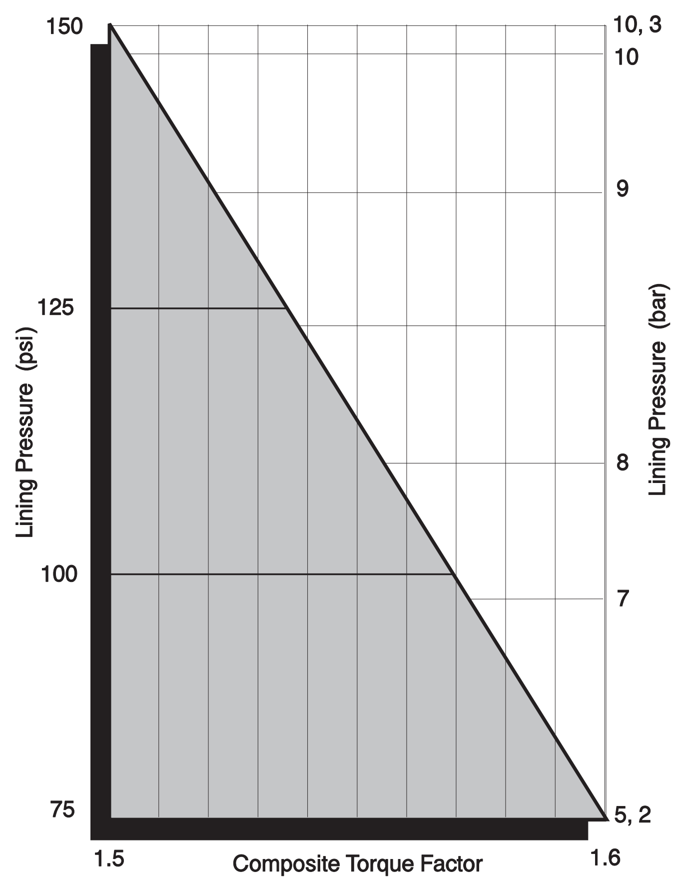
Friction linings offered by Airflex fall into three categories which are descriptive of their coefficient of friction. Due to the nature of the products and their suitability to specific applications, linings in each category are not available for all product lines or models. The three categories are:

Standard linings - These are linings furnished as standard material on Airflex products which permits them to operate within their published capacities.

Lo-co linings - These are linings which have a coefficient of friction approximately 65% of the standard linings. They are formulated for coefficient stability and wear characteristics and are used primarily for continuous slip applications.

Hi-co linings - These are linings which have a static coefficient of friction higher than the standard lining. They are used in applications requiring large locked-up torques with little or no slip between interfaces.

Two common types of hi-co linings furnished by Airflex are neoprene rubber and cork composite. The cork material is pressure sensitive. When this material is used, published torque ratings are increased by the factor obtained from the graph shown.



Drum and Disc Materials

Gray iron is used as an interface in the majority of Airflex clutch and brake applications. Other materials used for structural or thermal reasons include ductile iron, carbon steel and copper alloys.

For applications where little or no thermal energy is generated due to slippage, material selection is based upon their mechanical properties and cost. Cast gray iron castings are inexpensive and patterns are available for all standard drums and disks. Castings of ductile iron provides additional strength and ductility that may be needed in high speed applications and those that must endure shock or impact loads. Plain carbon steel fabrications can provide the additional strength and ductility without the need for a casting pattern. Fabrications may offer a price advantage when only one or two parts are involved and may be more readily obtainable than cast parts. The use of copper alloys is generally limited to applications requiring high thermal conductivity and where high first cost is not prohibitive.

In applications where a significant amount of heat is generated, the thermal properties of the materials are more significant. Their pertinent properties are shown in the table, below.

Copper alloys have thermal properties which are best exploited by using forced-convection - usually, water cooling; however, special non-aggressive friction material is required to provide an acceptable wear rate of the alloy.

The thermal conductivity of gray iron combined with its low modulus of elasticity results in lower temperature and stresses at its sliding surface.

Ductile iron has a lower thermal conductivity and higher modulus of elasticity. This results in higher surface temperatures, lining fade and accelerated wear under extreme conditions. Surface stresses are higher and prone to heat checking.

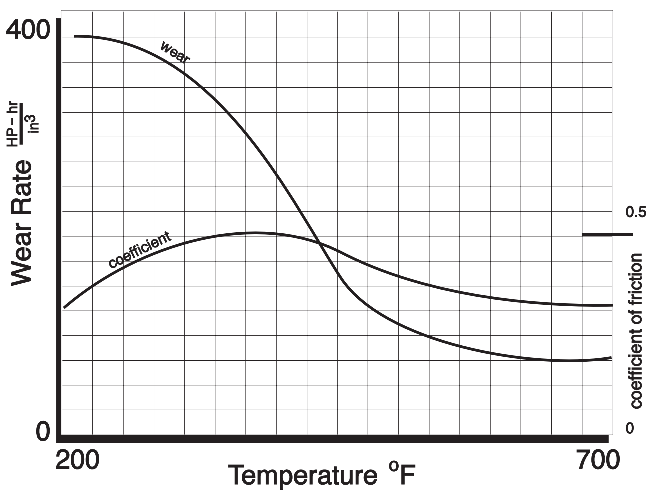
Carbon steel has a thermal conductivity similar to that of gray iron and a modulus of elasticity similar to ductile iron. Therefore, it will yield low surface temperature, but will be somewhat prone to heat checking.

Material	Thermal Conductivity k		Heat Capacity c		Coefficient of Thermal Expansion α		Modulus of Elasticity E		Density ρ	
	English	SI	English	SI	English	SI	English	SI	English	SI
	$\frac{BTU}{hr \cdot ft \cdot {}^0F}$	$\frac{W}{m \cdot k}$	$\frac{BTU}{lb \cdot {}^0F}$	$\frac{J}{kg \cdot K}$	$\frac{1}{{}^0F}$	$\frac{1}{{}^0C}$	psi	$\frac{N}{m^2}$	$\frac{lb}{in^3}$	$\frac{kg}{m^3}$
Gray Iron	28	48	0.13	544	6.7E-06	12,1E-06	16E+06	11E+10	0.26	7197
Ductile Iron	17	29	0.13	544	6.5E-06	11,7E-06	25E+06	17E+10	0.26	7197
Carbon Steel	29	50	0.11	460	6.4E-06	11,5E-06	29E+06	20E+10	0.28	7750
Copper Alloys	182	315	0.09	376	9.5E-06	17,1E-06	16.5E+06	11E+10	0.32	8858

Thermal Considerations

High operating temperatures are detrimental to clutch and brake performance. At elevated temperatures torque capacities are reduced and high thermal stresses cause heat checking and warping. If the units are intended for wet operation (where sliding interfaces are exposed to a coolant), the fluid may oxidize and decompose. Because of these temperature effects, a clutch or brake must be carefully selected to handle the required thermal loads without overheating.

For friction type clutches and brakes to accelerate or decelerate machine components smoothly, the friction interfaces must slip until they attain identical speeds. Slippage generates heat which must be dissipated.



Typically, friction materials have insulating properties which confine the heat to its sliding surface. Surface temperatures often exceed 1000°F (540°C). Prolonged exposure causes the friction material binder to break down at the surface, resulting in loss of coefficient of friction (lining fade). The clutch or brake can no longer develop its rated torque. Lining wear increases rapidly. The graph illustrates characteristic temperature effects on lining coefficient and wear.

The friction material interface is chosen for its ability to absorb (**specific heat capacity**) and conduct (**thermal conductivity**) heat away from the sliding interface. Absorbed heat is a function of the interface mass (**heat sink**). For a given amount of heat, the larger the heat sink, the lower the resulting temperature rise. The ability to absorb large quantities of heat is important in high inertia applications, as well as other high energy, low cyclic rate applications.

The rate at which heat is absorbed is a function of the interface area. If heat is generated faster than it can be conducted into the heat sink, surface temperatures will rise rapidly. The ability to absorb heat at high rates is important in high energy applications where starting and stopping times are short.

Heat dissipation occurs through radiation and convection, and depends upon the relative flow over convecting surfaces, heat radiating surface areas and the difference between ambient temperature and clutch and brake temperatures. Over a long time span, clutch or brake temperature rises until the rate of heat dissipation equals the rate of heat input. If the heat dissipation surface is too small and/or the flow is too low to accommodate the rate of heat input, clutch and brake operating temperatures will be high. Clutches and brakes should be designed so air can circulate freely; vanes are often added to encourage flow and fins are added to increase radiating surface area. The ability to dissipate large quantities of heat is important in high cycle and constant tensioning applications.

For a given application, operating temperatures are lower with increased clutch or brake size because of greater frictional area, larger heat radiating surfaces and more heat sink capacity. When size must remain small, but thermal load is large, then external cooling must be applied. Water, oil and forced air cooling are common.

Forced air convection is the simplest way to improve heat dissipation. Blowers or fans are used to create air flow and shrouding is usually needed to direct the flow across critical surfaces. Forced air cooling is not always practical as large volumes of air must be circulated to reduce the operating temperature. Shrouding, duct work and the blower require considerable space and give the equipment a bulky appearance. Noise and vibration may be objectionable.

Water cooling is a more efficient means of dissipating heat. When a clutch or brake is water cooled, materials which could not otherwise be employed can now be used for the friction interfaces. Using an interface with a high thermal conductivity will rapidly conduct heat away from the sliding surface to its underside which is exposed to circulating water. Because almost all the heat flow is into the water, friction materials which would be rejected because of their tendency to fade at elevated temperatures become practical.

Wear

Wear can best be explained by the adhesion theory. According to the theory, when two nominally smooth surfaces are pressed together, very high stresses exist over small regions or junctions where high spots touch each other. These stresses lead to local plastic deformation, which, in turn, produces the real area of contact.

A strong adhesion or bond takes place at the junctions. Friction force then arises from the need to break the junctions in shear. From the wear standpoint, it is desirable to have friction interfaces which limit the strength of the bonds formed. Wear results whenever a junction, in being broken, parts along some line other than the original interface. These wear particles then become embedded into one of the interfaces, usually the friction lining, and tend to score or dig a groove into the other interface.

Since these particles have little mass, they instantly absorb a tremendous amount of heat during clutch or brake engagement and fuse together. Eventually, these particles form a mass large enough to prevent lining contact.

Wear is influenced by a combination of factors which include, but not limited to:

Contacting materials - It is desirable to have a friction couple which limits the strength of the adhesion or bond formed between the interface.

Micro-structure constituents - Strength of the metallic interface is affected by the amount and disposition of graphite. Graphite counteracts thermal shock and reduces scoring by lubricating the sliding surfaces.

Surface hardness - Wear rate decreases as material hardness increases. Hardness is increased by alloying or heat treatment. However, thermal conductivity is reduced and increased wear may occur because of higher resulting operating temperatures.

Pressure - Wear appears to increase almost proportionally with pressure.

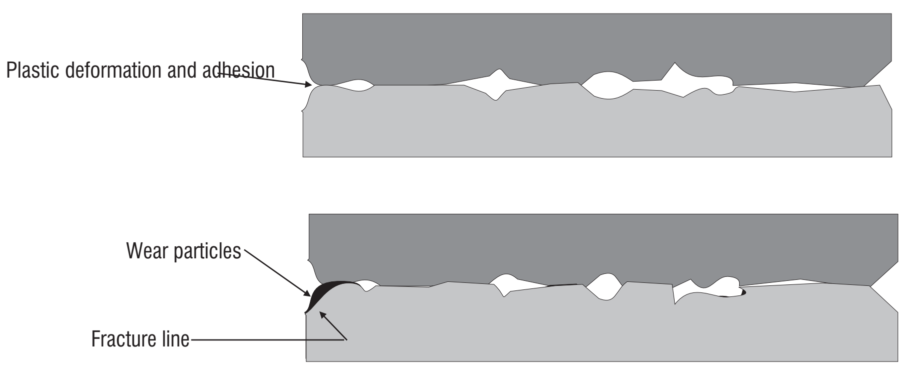
Temperature - High temperatures break down friction material binders and cause heat checking on the metallic interface.

Surface finish - Generally, the rougher the surface, the higher the wear rate. However, very smooth surfaces cannot store protective contaminants that may tend to lubricate the sliding surfaces.

Contaminants - Contaminants can be beneficial, like lubricants; or harmful, like abrasives.

Slipping speed - In general, wear rate increases with higher speeds.

As discussed above, contributors to wear are not only mechanical, but also metallurgical and thermal in nature. A combination of any or all of these factors make wear life difficult to predict.



Squeal

Squeal, like wear, is influenced by several factors which make it difficult to predict and/or eliminate. Squeal can result from thermal distortion of the friction interfaces. Temperature and humidity can affect the coefficient of friction and cause a squeal condition.

If squeal does occur, the installation should be inspected to determine if supporting structures have sufficient rigidity and that all bolts and nuts have been sufficiently torqued. The friction couple should be inspected to determine if the interfaces are in full contact, and, if not, be allowed to wear in.

In most cases, squeal can be eliminated by using a friction material having a lower coefficient of friction and increasing interface pressure. Changing lining flexibility by grooving or slotting may also help. Dampening clutch and brake components by the addition of weight or springs, where feasible, will reduce vibration that may be contributing to the squealing condition.

Mounting Arrangements

For all applications it is highly desirable to mount the smaller inertia clutch and/or brake components on the driven shaft and the larger inertia components on the driving shaft. This component arrangement minimizes the thermal load on the clutch or brake. It is essential for cyclic applications. The cyclic thermal graphs which appear in the catalog are based upon this arrangement.

For those applications where it is not practicable to mount the components in the preferred manner, close attention must be given to the clutch or brake thermal load. Oversize units may be necessary to handle the requirement.

Some components are designed to circulate cooling air through the clutch or brake. These components should be mounted on the shaft which rotates continuously or has the longer period of rotation. The open end of clutch or brake drums should be exposed to the atmosphere to avoid hot air pockets. Ventilating holes should be provided so cooling air can circulate freely.

Gap arrangements for clutch couplings are preferred over the close mounted arrangements. The gap between shaft ends allows clutch removal without disturbing the driving or driven components. When applications are close mounted or mounted between shaft support bearings, space should be provided to permit axial movement of components in order to access the clutch or brake for maintenance. If space does not exist, splitable units should be considered. Certain sizes of clutch and brake elements are available with this feature.

Bearing mounted clutch arrangements are available having provisions for attaching sheaves or sprockets. These arrangements can be mounted on shaft extensions or between shaft support bearings.

Whatever arrangement is employed, mounting shafts must be of sufficient diameter to accommodate the torsional and bending stresses associated with each installation.

Component Shaft Mounting

To accommodate the shock, transient and vibratory torques associated with any power transmission drive, it is recommended that all shaft mounted components be affixed with a key and an interference (shrink) fit. The interference fit minimizes, if not eliminates, fretting corrosion, key and keyway damage, which would result with a loose fit. In addition, some components rely on the interference fit to provide a bore to shaft seal for the actuating media.

Product Alignment

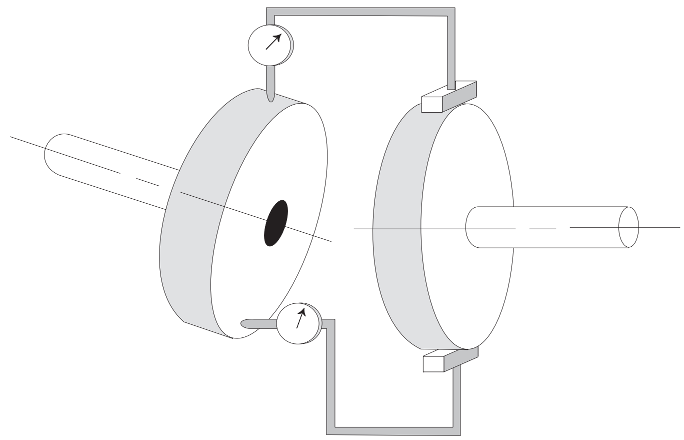
Good shaft and component alignment is highly desirable to minimize unnecessary mechanical stresses in couplings, clutches, brakes, shafts, bearings and their supporting structure. Perfect alignment, which for all practical purposes is unattainable, can only occur if the dynamic axes of rotation are concentric throughout their entire lengths.

Misalignment is completely described by measurements of parallel and angular misalignment. Since both types of misalignment occur in the horizontal and vertical planes, top-to-bottom and side-to-side measurements must be taken.

Parallel misalignment, also referred to as offset or run out, occurs when the axes of rotation are parallel to each other. It is measured by attaching a dial indicator to one shaft; rotating the shaft with the indicator about the outside diameter of the other shaft and noting indicator readings.

Angular misalignment, also referred to as gap, occurs when the axes of rotation intersect each other. Either a micrometer or dial indicator can be used to take gap measurements as shown in the figure.

Parallel and angular measurements are then used to determine the necessary adjustments. Initial shaft alignment should be done as accurately as possible. Foundation movement, frame deflection and expansion can change the initial alignment to the point where full misalignment capacity of the product is required. Although shaft speed generally dictates misalignment tolerances, the values shown in the table have been found acceptable for the Airflex product lines. The angular indicator reading should not exceed the product of the table value and the diameter at which the indicator readings were taken.



Product	Parallel		Angular	
	in	mm	in/in of dia. ❶	mm/mm of dia. ❶
CB, CM, EB, ER	0.015	0,38	0.0007	0,0007
CS, CSA, CTF, DBA, DBB, DC, DP, E, HB3, HC3, VE, VC, WC, WCB, WCS	0.010	0,25	0.0005	0,0005
CH	0.005	0,13	0.0003	0,0003
AR, AS, SB, SC	0.003	0,08	0.0003	0,0003

Note:

❶ Diameter at which measurements are taken.

Balancing

Unbalance exists when a body's mass is not equally distributed about its axis of rotation. The unbalance results in centrifugal forces acting on the body's support bearings and structure and induces forced vibrations which affects the wear and life of all machine components.

Balancing of a body attempts to redistribute its mass about the axis of rotation by mass removal or addition. Because all bodies are both statically and dynamically unbalanced, mass corrections should be made. If the body diameter is greater than twice the distance between the planes in which corrections can be made, single plane correction is usually sufficient for most installations.

Since the mass corrections are never perfect, some residual unbalance will always exist. Residual unbalance is measured in ounce-inches, gram-inches or gram-millimeters. Its value is the product of an amount of weight or mass and its distance from the axis of rotation. The value of the weight and mass or distance can vary as long as the product of the two does not exceed the required residual unbalance.

ISO1940 addresses the balance quality of rotating rigid bodies. Balance quality grades have been established which permit a classification of the quality requirements; however, their recommendations are not intended to serve as acceptance specifications. Permissible values of residual unbalance can only be determined experimentally by observing how the amount of unbalance affects the vibration, running smoothness and operation of the machine.

Some Airflex products are routinely balanced prior to shipment. The quality and method of balance is indicated in the table. If a finer balance quality is necessary, a balancing charge is required. Products not shown are only balanced when specified and are subject to a balancing charge.

Components balanced to an acceptable value, when combined (element and spider or drum and hub), can result in an assembly having unacceptable residual unbalance. The unbalance occurs due to the fit and geometrical tolerances and the mass addition of connecting elements (bolts and keys). If individually balanced components are unacceptable, the assembled components can be balanced as a unit. The position of the assembled components relative to each other must be match marked and assembled with shoulder bolts so that they cannot be reassembled in a different position.

Product	Number of Balancing Planes	Correction by Mass	ISO Balance Quality and Grade
12 thru 45 CB Single Elements ❶	Single	Addition	16
12 thru 45 CB Dual Elements ❶	Two	Addition	16
12 thru 45 CB Spiders ❷	Single	Removal	16
CB Sheave Clutches ❶	Single	Addition	16
CM Elements ❶	Single	Addition	16
CM Drums	Single	Removal	16
VC Single Narrow Elements ❶	Single	Addition	16
VC Dual Narrrow Elements ❶	Two	Addition	16
14 thru 52 VC Single Wide Elements ❶	Two	Addition	24
60 and 66 VC Single Wide Elements ❶	Single	Addition	24
14 thru 42 VC Dual Wide Elements ❶	Two	Addition	24
VC Spiders ❷	Single	Removal	16
VC Ventilated Adapters	Single	Removal	16
E and EB Vaned Drums	Single	Removal	16
DBA Solid Discs	Single	Removal	6.3
FSPA Combination Drums	Single	Removal	40

Notes:

❶ Clutch elements have moving parts, which make them non-rigid bodies. The grades indicated are nominal. Actual values can be above or below these grades.

❷ Components ordered with unfinished bores are not balanced.

Pneumatic and Hydraulic Control Devices

There are numerous devices available which, when incorporated into a control system, enables the clutch and/or brake to perform the required function. The most common devices are discussed below:

Pressure regulator - This device is used to establish the maximum pressure of the actuating media. Since torque is dependent upon media pressure, it follows that the regulator setting establishes the torque capacity of the clutch or brake.

Check valve - This device allows media flow in one direction only. Its purpose is to prevent reverse media flow should a pressure drop or failure occur in the pressure source.

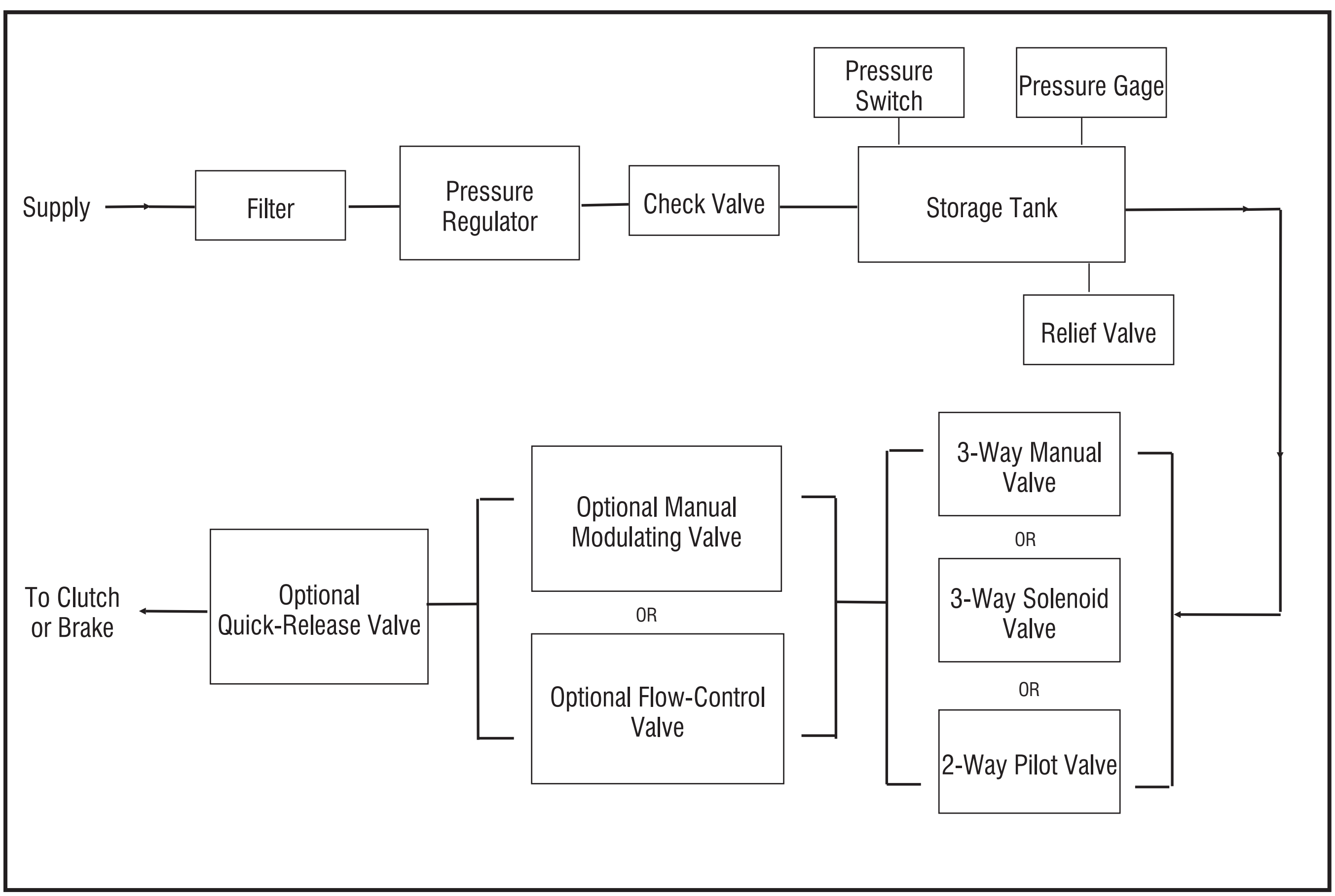
Pressure switch - This device senses media pressure and makes or breaks electrical contacts once a predetermined pressure is reached. It acts as a safety device ensuring that sufficient pressure is or is not available for clutch or brake actuation.

On/off/exhaust valve - This device is used to permit or prevent passage of the actuating media. It can be activated by an electric solenoid, a pressure-activated pilot or manually by an operator.

Flow control valve - This device is used to control the flow rate of the actuating media. The rate of flow determines the rate of pressure build-up and hence the rate of clutch or brake engagement. The valve setting is done manually. Once established, it need only be changed when changes occur in the operating conditions. These valves usually have controlled flow in one direction, and free flow in the opposite direction.

Pressure modulating valve - This device provides a means to continually adjust the clutch or brake pressure within a range determined by the minimum and maximum operating pressure of the valve. Adjustment is usually done manually by movement of a lever. It permits operator control of the acceleration or deceleration of changing loads, for instance, when starting or stopping a fully loaded, partially loaded or unloaded conveyor.

Quick release valve - This device provides a convenient port, close to a pressurized chamber, for quickly releasing the actuating media. It improves operation by reducing exhaust time and overlap between clutch and brake engagements.



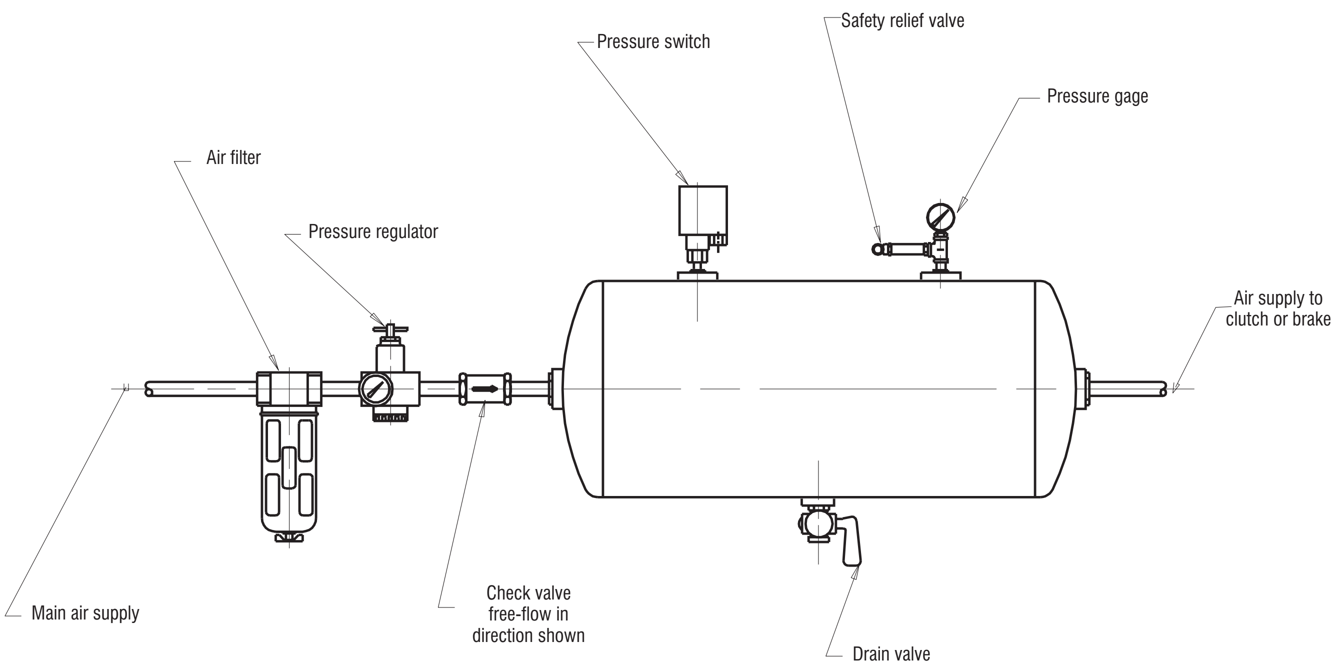
Air Supply

Section Y contains information pertaining to air compressor capacity. Refer to this section to size a new compressor or to determine if an existing compressor has ample capacity to handle new requirements.

To ensure that sufficient pressurized air is available (essential for high cyclic applications), it is recommended that an air receiver tank be installed as close as possible to the clutch and/or brake location. For very infrequent engagements, tank volume should be approximately 10 times the clutch and brake engagement volume. For high cyclic engagements, tank volume should be approximately 20 times the engagement volume.

A typical air receiver tank and suggested accessories are shown in the diagram. The air filter removes compressor and air line contaminates, the pressure regulator establishes the maximum air receiver pressure, and the check valve prevents air flow back to the air supply source. The relief valve ensures that the maximum allowable working tank pressure is not accidentally exceeded. The pressure switch determines if sufficient pressure is available and if not, prevents operation through its electrical interlocks. Water condensation can be removed from the tank through the drain valve. A pressure gauge at the out port of the receiver provides visual confirmation of the supply pressure to the clutch and/or brake.

Use full size valves and piping, with a minimum number of bends and fittings, between the air receiver tank and the clutch or brake. Do not use long lengths of flexible hose. Pipe size should be consistent with the recommended rotorseal size for a clutch application or the tube inlet valve for a brake application. Optimum sizing of the circuit for rapid response may require experimentation on the application.



Product Elastomers

The following specification applies to the rubber actuating tubes used in the CB, CM, E, EB, ER, VC and VE elements and the diaphragms used in DBA elements.

Elastomer: Inner & outer tube cover neoprene compound, type SC classification as covered by SAE10R4 Specifications for Automotive Elastomer Compounds. Cord plies have natural rubber coating.

Durometer: 60 æ 5 Shore A. Tube should be removed from service when it reaches a durometer reading of 72 Shore A or shows cracks from brittleness.

Temperature:

Heat - for continuous service 200ºF (82ºC to 93ºC); for intermittent service up to 250ºF (121ºC). Over 250ºF (121ºC) the elastomer hardens and rapidly loses resilience.

Cold - little change in performance characteristics down to -10ºF (-23ºC). Below this range, the elastomer stiffens and becomes brittle at -40ºF (-40ºC).

Resistance to Oils and Grease: Aniline point must be 200ºF (93ºC) or higher. Limited to non-aromatic hydrocarbons.

Resistance to Chemicals: Little change in properties when exposed to alkalies, dilute mineral acids or inorganic salt solutions. Acid and salt solutions of a highly oxidizing nature will cause surface deterioration and loss of strength.

An aliphatic hydro-carbon solvent can be used to clean grease and oil from the tubes. Kerosene is a good example of this type of solvent. After cleaning, any residue can be removed by wiping the tube with a clean cloth soaked in isopropyl alcohol.

Compounds used in other products are:

- Polyurethane - for the diaphragm or plunger seal used in the quick release valves.
- Buna N - for the CS, CSA and CTE brake pistons.

Selection Procedure

Analytical Method

The analytical procedure for determining the clutch and brake requirements is the same for any given application and consists of making the following determinations:

- Required Torque
- Thermal Load
- Mounting and Space Envelope
- Operating Environment
- Industrial and Federal Regulations

Formulas for calculating the torque and thermal load are given elsewhere in this section. Regardless of the application the operating data required to perform the calculations is basically the same and includes the torque to perform the necessary machine work, total inertia (including clutch and brake components) which must be started or stopped, acceleration or deceleration time, speeds of all rotating and linear moving components, clutch or brake shaft speed and frequency of clutch or brake engagement.

A service factor SF is generally applied to the required torque M to determine clutch torque M_c or brake torque M_b. This is done for two reasons: to have sufficient torque to accommodate any unpredictable transient torques or lining fade, and to compensate for the accuracy of the data. Typically, service factors range from a minimum value of 1.3 to a maximum value of 2. A factor in the low range is used when the drive and data is accurately defined. Values in the higher range are used when there are several unknowns.

A thermal load occurs whenever clutch or brake slippage occurs. During starting or stopping the thermal load W_t is basically the energy W required adjusted to reflect the work done W_w, if any, by the machine during the period of acceleration or deceleration. If slippage occurs during machine operation, the thermal load is determined by the product of clutch or brake torque and the angle through which slippage occurs. The total energy and the rate at which it must be absorbed and/or dissipated must be within the clutch or brake capacity.

A clutch or brake is then selected which has equal or larger torque and thermal capacities. The unit's dimensional and physical properties are then reviewed for drive compatibility. It must fit within a space which is free of all interference when the machine is both running and stopped. Mounting components and other machine components such as shafts and bearings must be able to handle the unit's weight and the stresses involved during operation. Operating speeds must not exceed the unit's limitation.

The clutch or brake location should have sufficient ventilation to dissipate the thermal load. The location must take into account the safety of all operating personnel. All rotating equipment must be well guarded. Actuating means and external cooling, when required, must be readily accessible, as well as an effective actuation route and/or cooling circuits to them.

Unusual or harsh environmental conditions can affect the success of an application. External protection must be provided if the conditions cannot be avoided. These conditions include high or low ambient temperatures, saltwater spray, heavy dust concentration and exposure to oil.

And, finally, all laws and regulations and industrial standards must be considered when making the final selection.

Service Factor Method

While it is preferable to use the detailed analytical method to evaluate applications, there is a simplistic alternative procedure which uses service factors. This procedure sizes clutches and brakes by multiplying the power rating of the prime mover by a factor. This factor is an experience based number developed for the specific type of application and is intended to compensate for the thermal loads and other considerations normally encountered. Values range from a minimum value of 1.5 to a maximum value of 5. Selection by service factor is used extensively where machines and power trains are standard. It is not to be used for flywheel driven machines, which are dependent upon flywheel slowdown for its power requirements.

The product of the power rating and the service factor is referred to as the **design hp**. Its value is used to calculate torque at the clutch or brake shaft.

Care must be used when selection by service factor is employed. Problems can occur when the application is not typical of the standard, or when the prime mover is not properly sized or sized on criteria other than torque.

These notes apply to the following Service Factor Tables:

❶ If no service factor is shown, refer to Section X, Power Presses, Brakes and Shears.

❷ If no service factor is shown, refer to Section X, Engines Clutches.

❸ Refer to Section X, Marine Drives.

❹ Refer to Section X, Grinding Mills.

❺ If no service factor is shown, refer to Section X, Well Drilling.

❻ If no service factor is shown, refer to Section X, Paper Machine Drives.

❼ Selection dependent upon thermal capacity.

❽ Torsional analysis required.

❾ Refer to Section X, Tensioning, Winding and Unwinding.

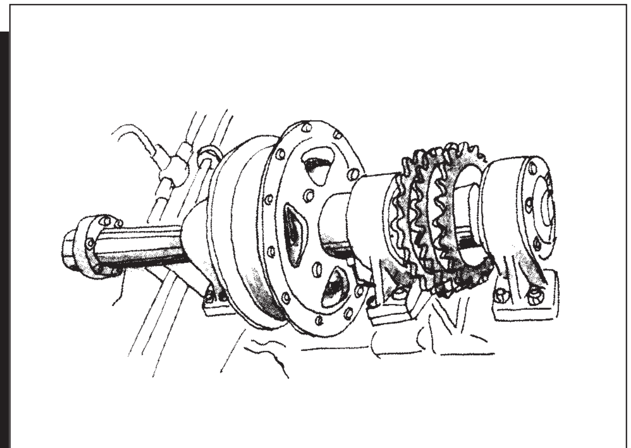
Industry	Machine or Equipment	Service Factor
Agricultural	Crop Spraying	1.5
	Grain Elevator	2.5
	Irrigation	1.8
	Sugar Refining	2.5
Amusement	All types of amusement ride drives.	2.5
Breweries	Bottle Washers	1.8
	Conveyors	2.0
	Labeling	1.8
	Uncasers	2.0
Can Making ❶	Bodymaker	
	Cap Machine	
	Cupper	
	End Press	
	Necker-Flanger	.0
	Seamer	2.0
	Shell Press	
	Strip Feed Press	
	Tab Press	
Ceramics & Clay	Block Splitter	2.5
	Brick Press	2.5
	Brick Stacker	2.5
	Extruder	2.5
	Kiln	2.5
	Oven	2.0
	Pug Mill	2.5
Cement	See Mining	
Chemical	Agitators	1.5
	Centrifuge	3.0
	Clarifiers	1.5
	Compressors	2.0
	Hammer Mill	3.0
	Kilns	2.5
	Mixers	2.5
	Pumps	2.0
Construction	Air Compressor	2.0
	Blast Hole Drilling	2.5
	Capstans	1.5
	Concrete Trawlers	2.0
	Conveyors	2.0
	Engines (Power Take-Off)	❷
	Excavating	2.5
	Floor Sanding, Polishing & Buffing Machines	1.5
	Hoists	1.5
	Hydraulic Pump Drive	1.8

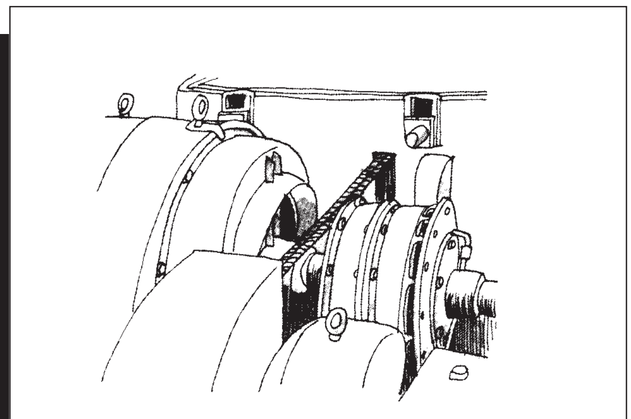
Industry	Machine or Equipment	Service Factor
Construction, Cont'd.	Insulation Shear	❶
	Locomotive Crane	2.5
	Overhead Crane	2.5
	Power Line Stringing	2.0
	Pumps	1.8
	Tunnel Boring	3.0
	Vibratory Soil Compactor	2.0
	Winches	1.5
Dynamometer	Absorber	❼
	Holding Brake	1.3
Engines	Damper	❽
	Generator Set	❷
	Power Take-Off	❷
	Torsional Coupling	❽
Fishing	Hoists & Winches	2.0
	Propulsion	❸
Glass	Edging Decks	
2.0		
	Fiberglass Winders	2.0
	Glass Sand Mill	3.0
	Molding	2.0
	Shears	❶
Laundry	Bar Soap Extruders	2.0
	Extractors	2.5
	Washers	2.5
Leather	Blade Grinder	1.8
	Die Cutting	❶
	Embossers	❶
	Tanning Mills	3.0
Logging	Skidders	3.0
	Yarders	❼
Lumber	Band Saw	2.5
	Breakdown Hoist	1.8
	Carriage Drives	❼
	Conveyors	2.0
	Plywood Stacking	2.0
	Setworks	2.0
	Stackers	2.0
	Veneer Clipping	2.0
	Veneer Lathe	2.0

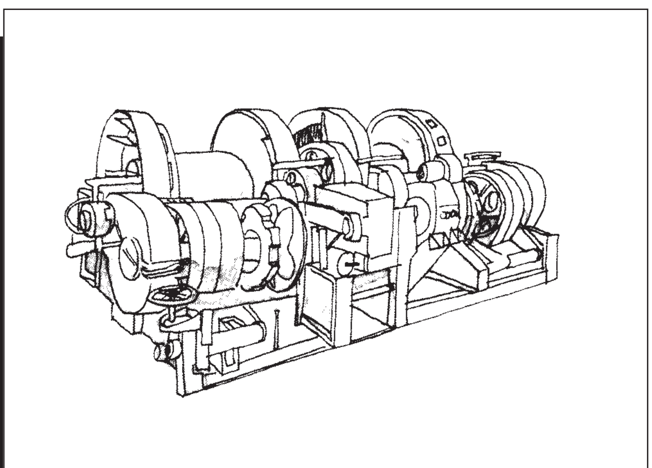
Airflex® Service Factor Table

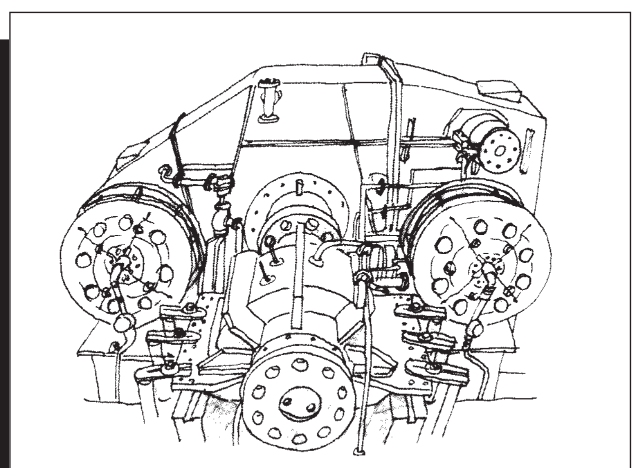
Industry	Machine or Equipment	Service Factor
Marine	Anchor Winch & Windlass	❼
	Bow Thruster	2.0
	Deck Machinery	2.0
	Dredges	2.5
	Generator	1.8
	Main Propulsion	❸
	Pipe Laying Equipment	2.5
	Power Take-off	1.8
	Propeller Shaft Brake	❸
	Pumps	1.8
	Radar & Aerial Systems	2.0
Material Handling	Bucket Elevator	2.5
	Conveyors	2.0
	Cranes & Hoists	2.0
Metalworking ❶	Alligator Shear	
	Car Shredders	❼
	Coining Press	
	Drawbenches	
	Expanders	
	Flywheel Brakes	❼
	Forging Presses	
	Headers & Upsetters	
	Machine Tools	2.5
	Multi-Slide	2.0
	Powder Metal Presses	
	Press Brakes	
	Rebar Shear	
	Rewind Stand	❾
	Roll Forming	2.0
	Roller Leveler	2.0
	Shears	
	Slitters	2.0
	Spring Coiling	
	Stamping, Punching & Forming Presses	
	Unwind Stands	❾
	Wire Cage	2.0
Mining & Cement	Blast Hole Drill	2.5
	Conveyors	2.0
	Crushers	3.0
	Dragline	3.0
	Drilling	2.5
	Elevators	2.5
	Grinding Mills	❹
	Hammer Mills	❼
	Kilns	2.5
	Locomotives	2.5
	Pulverizers	2.5
	Shovels	2.0
	Shuttle Cars	2.0
	Ventilating Fan	2.5
Miscellaneous	Clamping Device	1.3
	Lifting Device	2.0
Paper ❻	Calendar	
	Chippers	❼
	Converters	❼
	Conveyors	2.0
	Core Expanders	
	Couch	
	Debarkers	2.5
	Dryer	
	Presses	
	Pulpers	2.5
	Reel	
	Rewind Stand	❾
	Slitters	2.0
	Unwind Stand	❾
	Woodyard Machinery	2.5
	Yankee Dryer	
Printing	Book Binder	1.8
	Paper Shear	❶
	Presses	2.0
Rubber	Calendars	2.5
	Clipper Press	2.0
	Mills	2.5
	Mixer	2.5
	Tire Builders	2.0
Steel	Accumulator	❾
	Conveyors	2.0
	Heat Treating Furnace	2.0
	Rewind Stand	❾
	Rolling Mills	2.0
	Rollover	2.0
	Sand Mullers	2.5
	Screwdowns	2.5
	Tube Mills	2.5
	Unwind Stands	❾
	Wire Drawing	2.5
Test Benches & Stands ❼	Car/Truck Dynamometer	
	Dynamometer	
	Engines	
	Gear Boxes	
Textile	Beaming Machines	2.0
	Rag Cleaning Mill	2.0
	Rag Cutting	2.0
	Warping Machines	2.0
Transportation	Airport Ramp	2.0
	Locomotive Compressor	3.0

Industry	Machine or Equipment	Service Factor
	Locomotive Fan	2.5
	Plane Ground Support	2.0
Turbines	Starter Drive	3.0
	Water	2.5
	Windmill	❼
Well Drilling ❺ (Gas, Oil & Water)	Cat Head	1.5
	Compound	
	Construction Barge	2.0
	Drawworks	
	Inertia Brake	
	Offshore Pipe Laying	2.5
	Pumps	
	Rotary Table	
	Sand Reel	
	Semi-Submersible Anchor	❼









Units for wk^2 are lb·ft^2. Unless specified in the description column, all units are per the Table of Units and Measures.

Quantity				Formulas	
	Symbol	**No.**	**Description**	**English System**	**SI System**
Torque	**M**	**1**	Total torque required. M_f is positive for clutching, negative for braking.	$M_{(a\ or\ d)}$ æ M_f æ M_w	$M_{(a\ or\ d)}$ æ M_f æ M_w
	$M_{(a\ or\ d)}$	**2**	Torque required to accelerate or decelerate an inertia up to speed within a given time.	$\frac{\cdot}{25.58 \cdot t_{(aord)}}$	$\frac{\cdot}{9{,}55 \cdot t_{(aord)}}$
	M_b	**3**	Required brake torque.	M • SF	M • SF
	M_c	**4**	Required clutch torque.	M • SF	M • SF
	M_f	**5**	Frictional torque.	Estimated or measured.	Estimated or measured.
	M_p	**6**	Torque of the prime mover.	$\frac{\cdot}{n}$	$\frac{\cdot}{n}$
	M_t	**7**	Torque which will develop a torsional shearing stress in a solid circular shaft of diameter D.	$\frac{\tau \cdot \pi \cdot}{16}$	$\frac{\tau \cdot \pi \cdot}{16}$
		8	Torque which will develop a torsional shearing stress in a hollow circular shaft of outside diameter D and inside diameter d.	$\frac{\tau \cdot \pi \cdot}{16 \cdot D}$	$\frac{\tau \cdot \pi \cdot}{16 \cdot D}$
	M_w	**9**	Torque required to perform the necessary work and to overcome machine forces i.e., springs, cams, etc.	Measured or calculated.	Measured or calculated.
		10	Torque required to perform the necessary work during an angular displacement.	$\cdot \frac{}{\theta}$	$\frac{}{\theta}$
Work or Energy	**W**	**11**	Total energy	$W_f + W_1 + W_p + W_r + W_w$	$W_f + W_1 + W_p + W_r + W_w$
		12	Work done by torque M during angular displacement q	$\frac{\cdot \theta}{}$	$\cdot \theta$
	W_f	**13**	Frictional work resulting from linear and rotational movement.	$\cdot \frac{\cdot \theta}{}$	$\cdot \quad \cdot \theta$
	W_1	**14**	Kinetic energy of linear moving mass.	$\frac{\cdot}{}$	$\cdot \quad \cdot$
	W_0	**15**	Energy output of prime mover during time t.	$\cdot \quad \cdot$	$\cdot \quad \cdot$
	W_p	**16**	Potential energy of mass displaced a vertical distance h ft (m).	$\cdot$	$\cdot \quad \cdot$
	W_r	**17**	Kinetic energy of a rotating inertia.	$\frac{\cdot}{}$	$\frac{\cdot}{}$
	W_t	**18**	Thermal energy which clutch must absorb when used to change direction of a rotation from forward to reverse.	$\frac{\cdot}{}$	$\frac{\cdot}{}$
		19	Thermal energy which clutch must absorb when used to change direction of linear speed from forward to reverse.	$\frac{\cdot}{}$	$\cdot \quad \cdot$
	W_w	**20**	Energy required to perform the necessary work and to overcome machine forces i.e., springs, cams, etc.	Measured or calculated.	Measured or calculated.

Units for wk² are lb·ft². Unless specified in the description column, all units are per the Table of Units and Measures.

Quantity				Formulas	
	Symbol	No.	Description	English System	SI System
Power	P	21	Total power		
		22	Power required to develop the necessary torque.		
	P_c	23	Cyclic thermal power		
	P_D	24	Design power		
	P_p	25	Power of prime mover	Given	Given
	P_t	26	Thermal power to be absorbed.		
Time	$t_{(a\ or\ d)}$	27	Time required to constantly accelerate or decelerate an inertia.		
		28	Time required to constantly accelerate or decelerate to or from a given velocity.		
		29	Time required to traverse a given distance with constant acceleration or deceleration.		
	t_L	30	Lag time from signal to when a reaction occurs.	Measured or estimated.	Measured or estimated.
Angle	q	31	Total angle traversed.	$q_{(a\ or\ d)} + q_L$	$q_{(a\ or\ d)} + q_L$
	$q_{(a\ or\ d)}$	32	Angle traversed with constant acceleration or deceleration.		
	q_L	33	Angle traversed during lag time.		
Velocity	v	34	Final velocity after being accelerated or decelerated from an initial velocity v_o.	±	±
		35	Final velocity after being accelerated or decelerated from an initial velocity v_o.	±	±
Distance	s	36	Distance traveled at constant velocity.		
		37	Distance traveled while accelerating from an initial velocity v_o.	±	±
Heat	J	38	Quantity of heat absorbed which results in a temperature rise DT.	Δ	Δ
rpm	n	39	Revolutions per minute which will produce a peripheral velocity at diameter D inches (mm).		

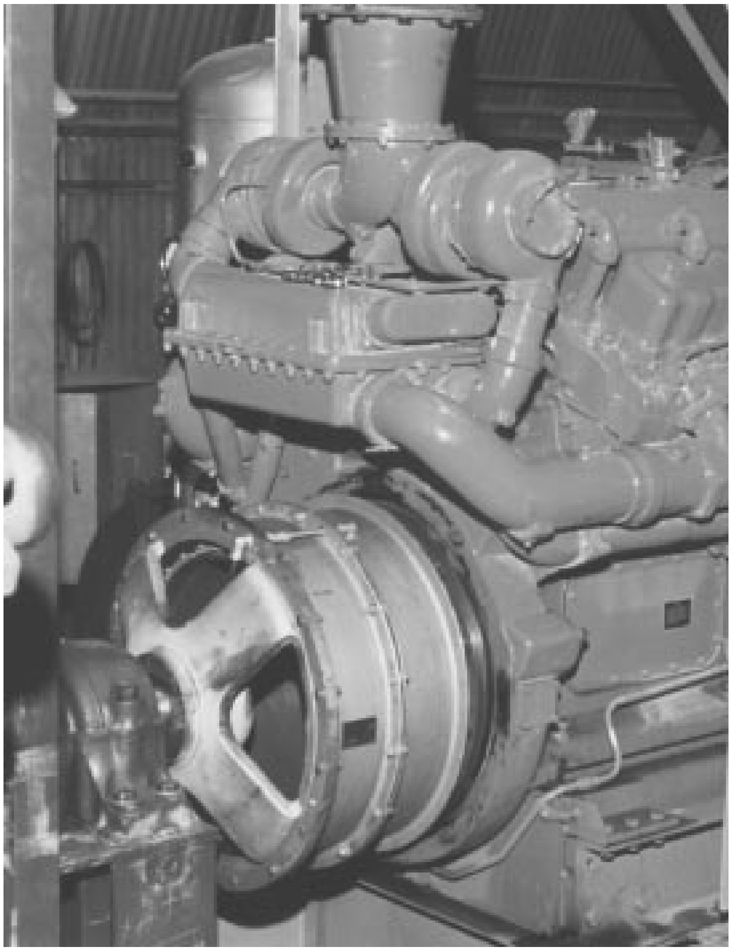
Measure			SI Units		English Units		Conversion English Units to SI Units
	Quantity	Symbol	Unit	Symbol	Unit	Symbol	
Space and Time	Acceleration, Linear	a	———	—	———	fps^2	3.048 E-01 · fps^2 = —
	Acceleration, Angular	a	———	——	———	——	—
	Angle	q	radian	rad	degree	deg	1.745 E-02 · deg = rad
	Area	A	$meter^2$	m^2	$inch^2$	in^2	6.451 E-04 · $in^2 = m^2$
	Length	l	meter	m	inch	in	2.540 E-02 · in = m
					foot	ft	3.048 E -01 · ft = m
	Time	t	second	s	second	sec	—
					minute	min	6.000 E-01 · min = s
					hour	hr	3.6 E+03 · hr = s
	Velocity, Linear	v	———	—	———	fpm	5.080 E-03 · fpm = —
	Velocity, Angular	w	———	——	———	——	—
	Volume	V	$meter^3$	m3	$inch^3$	in^3	1.639 E-05 · $in^3 = m^3$
					gallon	gal	3.785 E-03 · gal = m^3
Periodic	Frequency	f	Hertz	Hz	Hertz	Hz	—
	Frequency, Rotational	n	$minute^{-1}$	min^{-1}	—————	rpm	—
Mechanics	Density	r	————	——	———	——	2.786 E+04 · —— ——
	Energy or Work	W	joule	J	foot•pound	ft·lb	1.356 E+00 · ft lb = J
	Force	F	Newton	N	pound	lb	4.448 E+00 · lb = N
	Mass	m	kilogram	kg	————— ———	slug or lb_m	4.531 E-01 · Wt *(lb)* = kg 1.459 E-02 · slug = kg
	Moment of Inertia	J	kilogram·$meter^2$	·	lb·ft·sec^2	I ——	1.356 E+00 · I = kg·m^2 4.214 E-02 · Wk^2 (lb·ft^2) = kg·m^2
	Power	P	kilowatt	kW	horsepower	hp	7.457 E-01 · hp = kW
	Pressure	p	bar	bar	———	psi	6.895 E-02 · psi = bar
	Stress	t	———	——	———	psi	6.895 E+03 · psi = ——
	Torque	M	Newton·meter	N·m	pound·inch	lb·in	1.129 E-01 · lb·in = N·m
	Viscosity, Kinematic	n	———	——	———	——	9.290 E-02 · —— ——

Measure			SI Units		English Units		Conversion English Units to SI Units
	Quantity	Symbol	Unit	Symbol	Unit	Symbol	
Heat	Specific Heat Capacity	c					4.184 E+03 · $\frac{BTU}{lb \cdot {}^oF} = \frac{J}{kg \cdot K}$
Heat	Temperature	t	Celsius	°C	Fahrenheit	°F	5.556 E-0 · 1 (ºF-32) = ºC
Heat	Thermal Conductivity	k					1.731 E+00 · $\frac{BTU}{hr \cdot ft \cdot {}^oF} = \frac{W}{m \cdot K}$
Heat	Thermal Expansion	a					1.800 E+00 · $\frac{1}{{}^oF} = \frac{1}{{}^oC}$
Heat	Quantity of Heat	J	joule	J	British Thermal Unit	BTU	1.055 E+03 · BTU = J
Electrical	Current	I	ampere	A	Ampere	i	—
Electrical	Potential	V	volt	V	volt	v	—

Multiples of SI Units

The prefixes given in the table are used to form names and symbols of multiples of the SI units.

Factor	Prefix	Symbol
E+06	mega	M
E+03	kilo	k
E+02	hecto	h
E+01	deca	da
E-01	deci	d
E-02	centi	c
E-03	milli	m
E-06	micro	µ



Typical CB engine mounted clutch applications. At left, a dual 24CB500 clutch installed on a 1200 HP (895 kW) @ 1200 rpm engine mud pump drive. Lower application uses a 20CB500 clutch in a power take-off arrangement.