Affirmative Abrasion: Advancements and Innovations in Industrial Braking Systems

The selection of brakes for industrial cranes is a highly analytical process that includes a great deal of consideration involving many factors. This article presents the different types of brakes and the variables involved in selecting brakes that enhance overhead crane operation and provide reliable braking performance.

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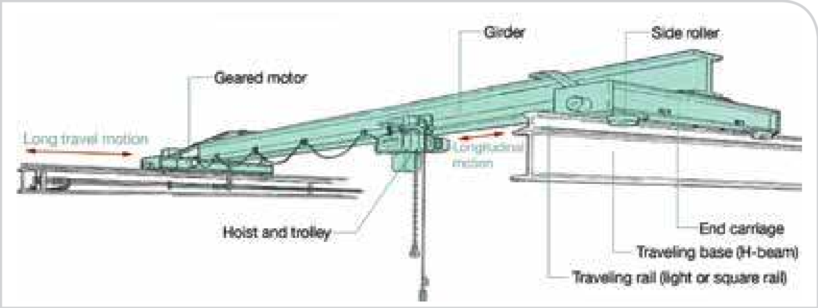
This article examines key elements in the engineering mechanics and component specifications that affect overhead crane brake selection, with the intent of empowering readers to make appropriate product selection. The employment of an optimal brake for overhead crane applications yields desirable and positive results: increased safety, greater efficiency, improved uptime and reduced maintenance. Conversely, improper brake selection can be catastrophic. After considerable field experience, substantive consultations with subject matter experts and extensive professional research, the author highlights a set of critical variables to aid professionals in choosing the most suitable brake for any given overhead crane application. When carefully considered and applied, these variables will enhance crane operation and provide reliable braking performance.

Industrial Crane Brakes: The Fundamentals

In iron- and steelmaking applications, industrial cranes perform three main motions: (1) forward movement along the x-axis, which is referred to as bridge motion; (2) lateral/left-right movement along the y-axis, which is referred to as trolley motion; and (3) up/down movement along the z-axis, which is referred to as hoist motion.

Historically, the most typical brakes found in U.S. iron- and steelmaking plants are DC magnetic drum brakes, known as "clapper brakes." To set the brake, force is applied to the brake drum by brake pads, which are

Figure 1



Electric overhead traveling (EOT) crane motions.

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bonded or riveted to the brake shoes. Like every brake used in overhead crane applications, DC magnetic drum brakes are fail-safe: a spring releases to "set" or apply the brake when power is interrupted. This effectively prevents loads from moving or falling during power failures. The brake is released through the DC-powered magnetic coil, which is either shunt- or series-wound. The brake is set when force is applied to the brake drum, which reduces or stops shaft rotation. In this way, a ladle loaded with molten steel can be prevented from falling vertically or making unsafe bridge or trolley movements.

The Association of Iron and Steel Engineers (AISE) established standard brake drum diameters: 8, 10, 13, 16, 23 and 30 inches.[1] Although the majority of cranes currently in use conform to AISE standards, there are some general-purpose brakes that utilize other sizes, such as 14- and 18-inch drum diameters, which were common before the standard was established. Because electrical overhead traveling (EOT) cranes range in carrying capacity from 25 to over 400 tons, the type of brakes needed to safely control a given load can vary significantly, according to the following factors:

- Understanding the numbers involved in calculating mechanical braking torque.
- The real versus nominal coefficient of friction.
- The specific type of braking application.
- Burnishing of the linings.
- The type of brake being used.
- Additional safety considerations that may require secondary emergency brakes to be installed.

Understanding the Numbers

One of the primary purposes of this paper is to stress the importance of accurately specifying the appropriate brake for a given application. To accomplish this objective, it is necessary to have an understanding of the variables involved in the braking process and analyze the effects of their interaction.

A critical variable involved in calculating the specifications of a given braking application is torque. Torque can be defined as "a measurement of the propensity of a given force to cause the object upon which it acts to twist about a certain axis. The torque is simply the product of the magnitude of the applied force and the length of the lever arm."[2] For any given industrial braking application, there will be a specific torque requirement to which the brake must conform in order to meet the demands of the crane on which it is utilized. There are fundamental torque formulas critical for specifying the optimal brake for a given application. Electrical (full-motor torque) and mechanical braking torque are the two principal torque values used for these calculations.

Figure 2

AC thruster and DC magnetic drum brakes, spring-set and fail-safe.

Motor torque is an electrical formula that refers to the maximum load the motor can produce. The following is the formula for motor torque, which uses the horsepower and rpm of the motor being used and is expressed in foot-pounds (ft-lb):

$$\text{Torque} = (\text{hp} \times 5{,}250)/\text{rpm} \qquad \text{(Eq. 1)}$$

The mechanical torque produced by the brake must be sufficient to overcome full-motor torque, and a service factor must also be considered. According to *AIST Technical Report No. 6,* "Brake sizes shall be as recommended by the brake manufacturer for the service, but in no case shall the summation of all brake ratings in percent of hoist full load hoisting torque at the points of brake application be less than the following:

1. 150% when only one brake is used.
2. 150% when multiple brakes are used and the hoist is not used to handle hot metal; failure of any one brake shall not reduce total braking torque below 100%.
3. 175% for hoists handling hot metal; failure of any one brake shall not reduce total braking torque below 125%.

For example, if two brakes are used, each must be rated 100% of the total full load hoisting torque (125% each for hot metal). If three brakes are used, each must be rated 50% (62.5% each for hot metal). If four brakes are used, each must be rated 37.5% (43.75% each for hot metal). In each of these cases, the failure of one brake does not cause the remaining braking torque to fall below the required minimum."[3]

As will be further delineated, it is often prudent to use a higher service factor due to variables that can potentially decrease the anticipated "nominal" torque value and thus precipitate the need for a larger "cushion" to prevent an underestimation of torque. The following is the formula for calculating mechanical braking torque, which is expressed in foot-pounds:

$$\text{Mechanical braking torque} = AF \times 0.42(D–F)/24 \quad \text{(Eq. 2)}$$

where

AF = applied force,
0.42 = standard (nominal) coefficient of friction,
D = disc/drum diameter and
F = face of the caliper/shoe.

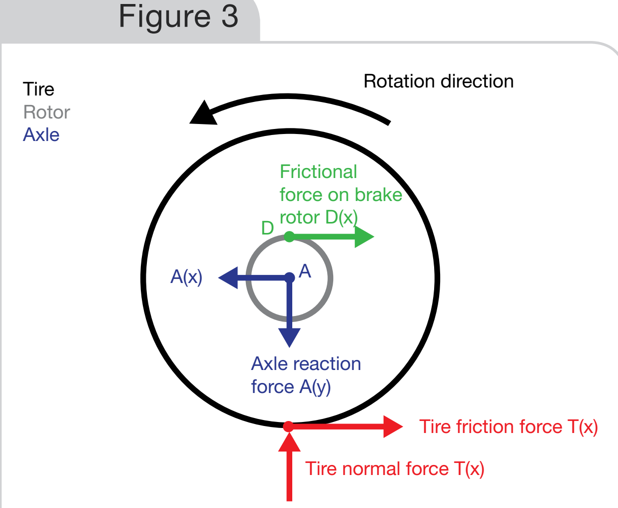
Applied force is the first variable used in calculating mechanical braking torque, a linear force denoted in pounds (lbs.). Force comes from the mechanism of the brake, which is the compression spring plus the mechanical advantage (lever ratio). In other words, applied force is the product of force times distance and is calculated by multiplying the spring force times the mechanical ratio (which is determined by the brake design). Mechanical braking torque measures the maximum torque value that can be generated by the brake, and it is a calculation of the rotational (torque) value that is yielded by the amount of applied force it has. Fundamentally, torque is a rotational force and applied force is a linear force. For fail-safe, spring-applied brakes that are used in iron- and steelmaking facilities, applied force is the force with which the shoe/pad will contact the brake drum/disc as applied by a spring. Therefore, mechanical braking torque is a measure of the maximum torque value that can be produced by the brake.

It can be seen in the mechanical braking torque formula that there are several variables involved in converting applied force into torque. D is a measure of the disc or drum diameter, and F is a measure of the face of the caliper/brake shoe. As will be further explained, the friction coefficient is a vital component of the torque calculation and specifying the brake as a whole. While 0.42 is a common friction coefficient for linings in the context of industrial braking applications, friction linings can be furnished in coefficients that are much higher and much lower. The higher the coefficient of friction, the more applied force is converted into braking force. Hypothetically, if 100% of applied force could be converted into braking force, then a 1.00 coefficient of friction would be achieved.

Friction: More Art Than Science

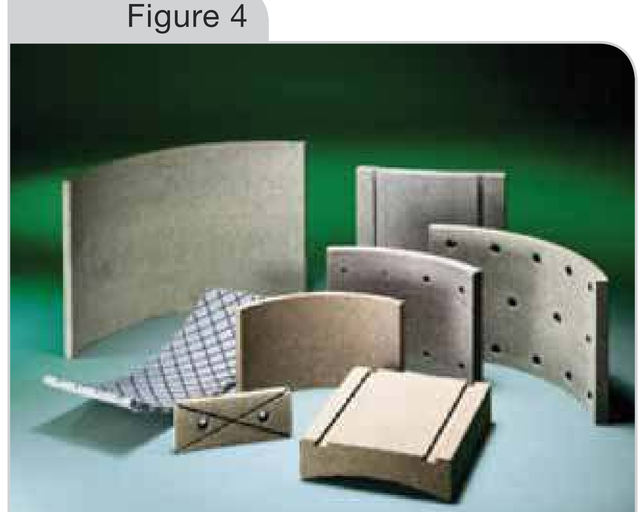
Choosing the best brake for a given application is more art than science, and there is no algorithm to calculate the precise coefficients. Too often, the assumption is made that brake size is the pre-eminent variable in determining the amount of torque yields. It is true that the size of a brake is proportionate to the amount of torque that will be generated: the larger the brake, the greater the torque. However, several other variables are extremely important in determining the torque value of a brake, and brake size is not necessarily the paramount determinant for braking torque. When first specifying a brake for

Figure 3



Visual depiction of braking forces.

Figure 4



Non-asbestos rigid molded friction linings can be manufactured to various friction coefficients.

an application, there will be an ideal torque value that, with other things being equal, will be achieved. This will be referred to as the "nominal" torque value. However, torque is never a precise science, and knowing the "real" torque value provided in an application could be considered a "subjective equilibrium." There are many different factors to be considered, and achieving a desired torque value is sometimes elusive and subject to many different factors. Of these factors, especially important but often underestimated is the coefficient of the friction linings/pads. As previously stated, 0.42 is an estimated standard value that is commonly used for braking torque formulas. The coefficient of friction between two surfaces in contact is equal to the force required to overcome the friction divided by the reaction force between the two surfaces.[4] The formula used to calculate the coefficient of friction is:

$$\mu = F/R \tag{Eq. 3}$$

where

μ = coefficient of friction,
F = force required to overcome the friction and
R = reaction force between the two surfaces.

While a friction coefficient value of 0.42 is a relatively good estimate for current metallic impregnated material, this information does not exist in published standards. Neither AIST, National Electrical Manufacturers Association (NEMA) nor American National Standards Institute (ANSI) standards contain any reference to friction coefficient or a definition of the term *burnishing*, which is another term that relates to friction coefficient and is prevalently used in this paper.[5] The 0.42 coefficient of friction value is an ideal standard which could be considered the "nominal" friction coefficient. This nominal friction coefficient makes a number of assumptions, including but not limited to the brake being used in a dynamic application, the brake being operated in clean environmental conditions, and whether or not the friction linings are adequately burnished, among others. It is possible that this "real" friction value will be achieved in an application. However, to get a more realistic and accurate prediction of the friction value that will be achieved, a more intricate and strategic approach must be taken in which the "real" coefficient of friction is the true value that is achieved once all factors are considered based on the coefficient of friction of the linings actually being used, among other factors.

This number is subject to fluctuations. For instance, in an industrial environment generating a substantial amount of dirt and grime, the friction linings are likely to become contaminated and the friction coefficient will be effectively reduced. For example, friction linings that are 0.42 in a clean environment could become 0.35 after contamination. Consequently, in an application that is subject to high amounts of contamination, it may be wise to increase the service factor to compensate for the environmentally reduced friction values. For drum brakes, the material used in manufacturing the brake drum can also affect the real coefficient of friction. With the use of a ductile iron wheel or qt100 disc, the friction coefficient will not be reduced. However, certain end users may have a preference for using a stainless steel brake drum, and if this is the case, the friction coefficient can potentially decrease as much as 20%, which would consequently de-rate the mechanical braking torque.

Another indispensable constituent in determining the real coefficient of friction is the type of brake application itself. The two principal braking applications we must consider to determine the proper friction coefficient are dynamic versus static. Service brakes are primary brakes that are in continuous use, whereas secondary parking brakes are used as a backup. Both primary and secondary brakes can be used in either dynamic or static (otherwise known as "holding") applications. Dynamic braking is the process of reducing the speed of any rotating machine.[6] In the case of EOT crane braking systems, dynamic braking occurs when brakes are used to stop the load, which is achieved by stopping the rotating shaft from turning. The secondary brakes are considered holding brakes, since they do not stop the load, but instead hold the load.

According to the International Society of Automation, the holding brake's principal use is to safely keep a load in place in situations where the power either is turned off or fails. Holding brakes are available in two common configurations: dynamic stopping and static holding. Dynamic stopping brakes are commonly used as cycling brakes, which take the wear and tear of constant on/off engagements while the shaft is rotating and still provide long life. Static holding brakes are for simple load-holding applications. Understanding the difference between dynamic stopping and static holding applications is significant because a different brake design is required for each respective scenario; ergo, using the wrong brake can face potentially negative consequences. To elaborate, the International Society of Automation says, "A static holding brake that's incorrectly applied in a frequent cycling application will wear out and fail quickly because it isn't designed for wearing applications. Similarly, a unit designed as a dynamic brake needs engagements at speed to maintain its full torque rating. A dynamic brake that's used solely as a holding brake may experience torque degradation, reducing

Figure 5

Thruster disc brakes in holding application.

its performance over time due to a loss of burnish."[7] It is imperative to understand that the coefficient of friction is different in a static holding versus a dynamic stopping application. A dynamic stopping application will generally have a higher friction coefficient than its static counterpart. In a static application, the braking starts from a stationary position; the lining will not achieve full contact on the disc/drum because there will be high points/ridges on the linings. In a dynamic stopping application, the high points will be diminished due to the more active braking application, and the enhanced grip and contact with the disc/drum yields a greater coefficient of friction. As a result of this use differentiation, it is suggested that the friction coefficient for a static application is to be de-rated approximately 20% from its nominal value. The same application from a stopping position would yield 0.4, but would yield 0.5 in a dynamic application.

Brake Fade — In a dynamic braking application, there is the propensity for the phenomenon of brake fade to occur, and this is an important concept to factor into brake selection. To elaborate on the previous paragraph, the primary difference between static and dynamic braking is that dynamic braking applications encompass a repetitive stopping of a rotating machine, and thus the brake has to absorb the kinetic energy that is built up by inertial loads. As a result, the brake must transfer this kinetic energy, which results in heat buildup and wear on the surfaces of the rotating components. The brakes function by converting the kinetic energy of the rotating shaft into thermal energy during deceleration, which produces a substantial amount of heat that must then be transferred into the surroundings and into the airstream. Conversely, with static holding applications, all rotating components come to a rest and the brake simply holds the load. In these applications, no heat builds up and there is very little wear.[8]

Brake/pad fade occurs when the temperature at the interface between the pad and the disc exceeds the thermal capacity of the pad. One result of brake fade is the formation of resin components on the linings, commonly referred to as "glazing," which effectively reduces the integrity of the linings and thus lowers the coefficient of friction of the linings. This is typically indicated by a telltale odor and/or smoke. Fluid boiling and vaporization are other results of brake fade which have adverse effects on friction coefficient and ultimately the ability for the brake to stop. Gas bubbles are formed when fluid boils on the friction materials and ultimately degrade the ability for the friction material to interface with the disc/drum. This process is incremental with several warning signs.[9]

Since brake fade occurs as a result of continuous braking and the heat that ensues, decreasing the stopping/braking time is the principal means to mitigate the risk of brake fade. Stopping time is directly proportionate to the amount of heat generated during the friction occurrence. In other words, if stopping time is decreased from four seconds to two seconds, heat generation is reduced by 50%. Less heat means less pad/shoe wear. One way to reduce stopping time is by increasing the torque of the brake. As discussed, the variables that can be manipulated to create variation in the torque formulae are disc diameter, friction coefficient and applied force of the brake. Using a larger disc, using friction linings/pads with a higher coefficient of friction, or using a brake with a greater amount of applied force will yield a higher torque value. Contrary to popular belief, increasing the pad area will not increase the brake torque. However, increasing the pad area will decrease pad wear and mitigate the risk of brake fade; larger pad area means greater surface area to dissipate heat. Theoretically, the same amount of torque can be achieved by using either an extremely large pad or an extremely small one. However, the amount of heat being generated in a high-torque application would be far too great for the smaller pad to dissipate, increasing the danger that it could burn up very quickly.[10] There are also safety features modern brakes offer that mitigate the risks of brake fade. Self-adjusting mechanisms are ideal because they compensate for lining wear and, as a result, torque does not decrease due to lining wear. Lining wear indicators are helpful, too; they provide a signal when lining thickness reaches a critical threshold and indicate that linings must be changed.

Pursuant to the concept of static versus dynamic braking, another critical component in calculating the real coefficient of friction value is the concept of burnishing. One definition of burnishing of brake linings holds that "burnishing is a method of conditioning the top layer of the friction compound during the manufacturing process, removing the need for the bedding-in. This effect is achieved by process equipment that subjects the friction surface, in a controlled environment, to a short burst of heat exposure in the temperature range where green fade occurs. Burnishing is therefore a method of preconditioning the brake lining, rendering it ready for full service upon installation. The user is in a position to install and drive without having to follow complicated bedding-in procedures, although a certain amount of 'self-bedding-in' is still present."[11] While some original equipment manufacturers attempt to burnish friction linings during the manufacturing process, it is virtually impossible to achieve complete burnishing until the brake is installed into its true application due to application-specific variables that cannot be accounted for at the manufacturing facility. These variables include, but are not limited to:

Figure 6

(a) (b)

Example of friction lining burnishing: before (a) and after (b).

1. Height of the brake: if a brake is offset and not perfectly centered, the pads will not contact as designed. If the brake is too high or skewed to one end, there will be unbalanced contact and the pads will not achieve optimal contact.
2. Brake mounting: if the brake is not centered and exactly aligned, the brake drum will not be centered as the manufacturer intended; subsequently, the linings will not contact the drum at the optimal angle and provide expected performance.
3. Concentricity of the brake drum (or flatness of the disc): if the brake drum is perfectly round (or if the disc is perfectly flat), then there will be no difference between burnishing at the manufacturing site versus at the live application. However, if the drum being used is not perfectly round, then the linings will contact the drum imprecisely and fail to deliver desired performance.

Once friction linings become burnished, the coefficient of friction will increase by eliminating the high/uneven points on the surface of the linings which yields greater contact and therefore greater friction. Unburnished linings will have high points. When the brake is applied, these high points will contact the brake wheel but the thinner points will achieve no contact. If burnished perfectly, linings will achieve 100% contact. In light of this information, it can be implied that, when first put into service, linings will yield a lower amount of torque due to having less contact with the mating surface, such as the drum, potentially causing more slip. After friction linings are thoroughly burnished, the high points of the linings are eliminated; they wear off and effectively become much smoother and achieve close to 100% contact with the mating surface, and more contact means greater torque.

Size Does Not Necessarily Matter

At first glance, it may seem that bigger brakes provide better braking. Actually, a smaller-diameter brake can generate as much torque as a larger one by simply increasing the coefficient of friction linings applications and/or enlarging the disc diameter the calipers mate to. The following is an empirical evaluation of friction and disc diameter:

Scenario One — Using a 100-hp motor with 1,800 rpm on the low-speed side:

- Full-motor torque: (5,250 x 100)/1,800 = 292.
- Using service factor of 1.5: 292 x 1.5 = 438.
- Using 10-to-1 gearbox ratio: 438 x 10 = 4,380 ft-lbs of full-motor torque.

To determine the appropriate brake to accommodate this torque requirement, calculate the static torque of the brake. Using a 28-inch disc with 3.4-inch pad width (manufacturer specification) in conjunction with a brake with applied force of 8,520 lbs. and 0.4 coefficient of friction: 8,520 x 0.4(28-3.4)/24 = 3,493.20 ft-lbs.

That torque is insufficient: 3,493.20 < 4,380.00.

Figure 7

Spring-set thruster drum brake.

Scenario Two — Using the same disc diameter but with a larger brake which furnishes applied force of 12,100 ft-lbs:

$$12{,}100 \times 0.4(28–4.7)/24 = 4{,}698.83 \text{ ft-lbs}$$

This brake would provide sufficient torque for the application.

Under these circumstances, the smaller brake will not suffice. However, using the identical setup as Scenario One but using friction pads with a 0.6 coefficient of friction and increasing the disc diameter to 31.5 inches provides an applied force of 8,520 lbs. x 0.6(31.5–3.4)/24 = 5,985.3 ft-lbs.

This brake is more than capable of handling this torque value. In fact, the disc diameter can be decreased to 24 inches and still generate 4,387.80 ft-lbs of torque, which is more than sufficient for the given application. This solution has the added advantage of requiring less clearance than the 28-inch disc and provides a degree of cost savings as well.

The implication of these scenarios is critical: if a torque calculation for a given brake uses the nominal coefficient of friction (0.42) without taking other variables into consideration, there is the potential for the brake to underperform. A 0.05 decrease in friction coefficient may not necessarily yield a substantive difference, but in some cases it could reduce the torque value to the point of not being capable of carrying the load, and thus the results would become catastrophic. Additionally, stopping power varies as a result of the application. For instance, is it desirable to have an immediate, "hard" stop or a gradual, smoother stop? The standard 0.42 friction coefficient may not necessarily yield the desired results for a given application. Consider a comparison of a hoist versus trolley application. For a trolley application, the stop can potentially become too "grabby" with too much friction, which causes a hard stop and can wobble the load and cause its instability. This goes back to the concept of "subjective equilibrium": the objective for bridge and trolley application is to achieve sufficient torque to stop the brake, but having too much torque is a liability. Therefore, using a medium friction coefficient is ideal to avoid a situation where the linings are too grabby. For a hoist application, a hard stop is not a liability, and it may actually be considered an asset, since the objective is to get as much torque as possible to stop the load from falling. In these instances, a higher coefficient of friction may be desirable.

Another significant implication of the potential torque fluctuation is the concept of service factor. As previously stated, there are standard service factors provided by the Construction Management Association of America (CMAA), Occupational Safety and Health Administration (OSHA), and AISE which are a percentage of full-motor torque. Ostensibly, these standard service factors can be used to calculate a mechanical braking torque value that is more than sufficient to handle the load in the given application. However, being that there are several variables that contribute to the "real" mechanical braking torque value that can be inherent or extrinsic to the braking system, these service factors may not be sufficient to provide the safest and most accurate full-motor torque value for a given application. For instance, if a 1.25 service factor were used as a product of full-motor torque but extrinsic variables such as environmental (i.e., high degree of contamination), and intrinsic/application-specific variables (i.e., assuming higher coefficient of friction despite static application with lack of burnishing) are not taken into consideration, it is quite possible that the 1.25 service factor may be quickly downgraded to 1.00 or lower. If this were the case, the risk of failure would increase because the "security blanket" that a safety factor allows would be eliminated. Due to the magnitude of service factors in the safety of an industrial braking system, it is imperative that the parties involved in specifying the braking system perform an application-specific analysis to take all factors into consideration to use the best and most realistic service factor to ensure the utmost safety and reliability.

Not All Brakes Are Created Equal

As mentioned in the introduction, DC magnetic drum brakes (magnet brakes) have historically been the brake of choice for iron- and steelmaking plants. Magnet brakes possess some excellent features: they are "cleaner" to operate because they do not use hydraulic fluids. Hydraulic brakes are subject to leakage and can be messy, which also poses both a safety hazard and an environmental issue for nuclear plants and other specialized industries.

A drawback to magnetically actuated brakes is that the magnet stroke is much smaller than that of an electrohydraulic thruster actuator. The air gap of a magnet brake is subject to contamination; if dust or oil infiltrates the magnet, the functionality of the magnet coil is compromised, which threatens the performance of the brake. While dust gaitors have been developed for magnet brakes to mollify the potential for contamination, most dust gaitors do not encompass the entire circumference of the coil housing and thus are not 100% effective. Thruster brakes are far less subject to potential contamination, and their preventive maintenance is minimal.

Another drawback of magnet brakes is that they are not adjustable in a practical way because there is no adjustable torque tube/scale on them. Magnet brakes are adjusted by means of a torque screw, but it is difficult to regulate resulting torque values, creating potentially negative consequences. Certain improved brakes have eliminated this problem by utilizing an adjustable and scalable torque tube. The main benefit of having an adjustable torque tube/scale is that it allows the torque to be properly adjusted due to changing conditions by providing the end user with a far more precise and transparent means of adjusting the torque of a brake. As previously discussed, it is sometimes prudent to use a higher service factor due to application-specific variables and environmental conditions that can potentially yield a lower real torque value than initially anticipated. Therefore, if a higher service factor is used and the torque is increased, the end user can subsequently adjust the torque accordingly while the brake is in service and the variables and conditions present themselves. In other words, having the sliding torque scale calculates a precise torque value when initially specifying the brake and allows the end user to adjust the brake on-site. Adjusting the torque is a critical procedure and should be conducted by a site engineer in consultation with the brake representative.

Another unique feature that is offered by thruster brakes is a lowering valve whose primary purpose is to slow down the application of the brake in a gradual "ramp-down" nature, creating a smoother stop, which is especially ideal in bridge and trolley applications. Magnet brakes are typically off-on and thus do not offer this feature. Lining wear indicators and mechanical limit switches are other devices that can be added to thruster brakes that enhance their safety and performance.

Figure 8

Dual caliper magnetic disc brake.

The Case for Disc Brakes — Disc brakes have become an optimal brake choice for many reasons. In contrast to magnet drum brakes which use only one size of brake drum, disc brakes can be used with a relatively broad range of disc diameters, which can dramatically increase or decrease the brake's generated torque. A smaller-diameter brake can be more economical and reduce space allocations for braking mechanisms. Additionally, disc brakes contain fewer moving parts and require less maintenance. Disc brake friction linings grip to a flat surface. Therefore, less burnishing is typically required to achieve full contact with the disc than its drum brake counterpart, whose linings grip onto a curved surface.

Balanced Load Disc Brakes: Greater Contact and Increased Safety — Disc brakes typically consist of a brake on only one side of the disc. This produces an unbalanced load because the brake has a tendency to pull to one side. Balanced load disc brakes contact the disc in symmetrical and opposite points. Interfacing with the disc at multiple and balanced contact points yields an even brake load. The second caliper prevents any lateral shifting of the shaft because it is now locked in the center. As a result, these balanced load disc brakes provide enhanced safety and performance. With the traditional disc brake, it is inevitable that forces will be applied to the shaft. One implication of the lateral shifting that is caused by asymmetrical contact in traditional disc brakes is that the shaft will have loads that compel it to "twist." Twisting potentially can cause the shaft to break. Another drawback of

Figure 9

Hydraulic release rail clamp.

an unbalanced load disc brake is a variance in torque between the upwardly rotating side of the disc and the downwardly rotating side. Braking torque is typically around 7% greater with downward disc rotation and 10% less with upward disc rotation. Balanced load disc brakes eliminate these problems and dramatically reduce the potential for failure. When a brake interfaces with the disc at symmetrical contact points, there are no vibrational issues and no external loads on the shaft, thus reducing the implied risk of a shaft bending and possibly breaking. With a more consistent/balanced load, it does not matter on which side of the disc the brake is installed because, regardless of location, there will always be an even amount of braking torque. This is a benefit to safety due to the lack of torque fluctuation and ultimately allowing for greater control and predictability over the actual torque value that will be generated. Also, a more consistent load will yield braking that is smoother and more seamless. Balanced load disc brakes are manufactured to AISE standards and can be used as drop-in replacements to most braking systems used on EOT cranes and other applications.

Storm Brakes: Protecting Your Crane From The Elements — Storm brakes are designed to ensure safety in the event of high winds, storms, earthquakes, or other environmental forces that can adversely impact the function and integrity of an outdoor crane. There are different types of storm brakes, and the most commonly used are referred to as rail clamps. Rail clamps are spring-applied, with friction pads clamping along the sides of the rail, essentially locking down the crane. Rail clamps will protect the crane from upward and downward forces as well as crosswinds. Rail brakes are another type of storm brake that function by leveraging the weight of the crane and pushing down at the top of the rail. Rail brakes will protect from downward forces and crosswinds, but they do not completely protect the crane from upward forces.

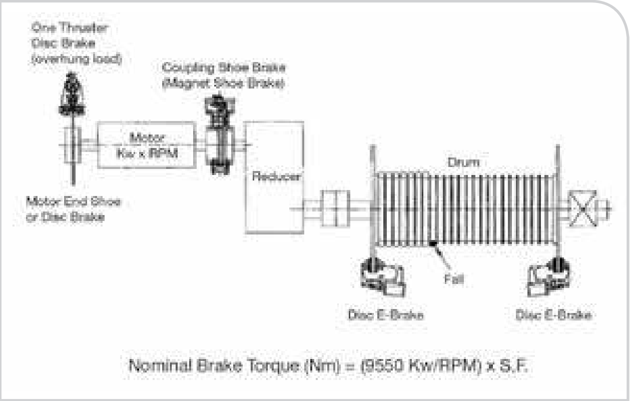
Storm brakes are fail-safe, spring-applied and hydraulically released. Rail clamps can provide a wide range of holding forces, from 1,000 to 150,000 lbs. Specifying a storm brake system for an outdoor crane requires substantial analysis, and selecting the appropriate storm brakes is essential to ensure that the crane will be secure and protected in the event of a potentially damaging natural force.

There are multiple variables and factors involved when specifying a storm brake system. For instance, the maximum wind velocity is required, along with determining the surface area of the crane and type of rail being used for the friction pads. Separate or integral power units can be packaged with the brakes, depending on end-user preference as well as distance between brakes. Having a thorough understanding of the crane and its surroundings is imperative, as is properly adjusting and maintaining the storm brake system. The reality is that the occurrence of a runaway crane is a viable threat, and when it does occur, the effects are catastrophic and extremely costly.

Emergency Brakes: The Optimal Safety for EOT Braking Systems — There are many ways to improve the safety of EOT cranes, including the use of fail-safe brakes, upgrading safety factors and integrating secondary brakes. Despite the palpable increase in safety that is created by these features, there are still extraordinary circumstances that ultimately preclude these enhanced safety features from preventing major problems. This involves the rare but potentially catastrophic occurrence of a gearbox or load-side problem. For example, if a coupling or gearbox malfunctions, or if the shaft breaks, both the primary and secondary brakes will be unable to prevent the load from falling. In other words, the low-speed side of the crane poses the greatest risk and, regardless of whether the primary and secondary/parking brakes are functioning properly on this side of the crane, these kinds of failures will render the brakes incapable of preventing a load from falling and causing disastrous consequences. Emergency brakes are the solution to this problem.

The primary and secondary brakes have pads that mate to rotating drums/discs connected to the shaft. Therefore, if the shaft breaks, then the entire braking system is fatally compromised and rendered incapable of preventing the load from falling. The preventive measure to this rare but potentially catastrophic risk is to implement a braking system with a rotating disc that connects to the drum of the crane. Provided that this braking system possesses sufficient torque to handle the entire maximum load of the crane, the load will be stopped even in the event of a low-speed side failure such as a shaft breaking or a gearbox or coupling failure. While these emergency braking systems are currently uncommon in U.S. iron- and steelmaking facilities, they are prevalent in other industries and should certainly be considered for certain applications such as main hoist ladle cranes where the costs associated with failure are especially severe and life-threatening.

Figure 10



Typical hoist brake setup.

The Proactive Approach to Safety — How does one justify designing and implementing an emergency braking system for a given application? The question one must ask is, "How critical is it to bear the costs associated with implementing an emergency braking system for a given application?" This is a subjective matter that requires a diligent cost-benefit analysis. For instance, a scrap crane may not merit an emergency braking system, but a main hoist crane carrying hot metal would be a compelling candidate. In the latter application, the costs of equipment, downtime, and raw materials and finished product must be factored; furthermore, there is the incalculable risk of human injury or fatality. While emergency braking systems are not inexpensive by any means, they are certainly far less costly than the aforementioned costs associated with failure and the potentially high costs of facility shutdown while cleanup and repair operations are conducted. Therefore, it is imperative that, for any given application, all risks be carefully considered through both a quantitative and qualitative analysis.

As previously stated, emergency braking systems are commonly implemented in other industries and applications. For example, in the mining industry, where heavy pieces of equipment are being lifted, and in industries where there are applications that involve sinking winches, sag mills or conveyor systems (for instance, on downhill conveyors where there are falling rocks), there are often secondary/emergency brakes on the low-speed side. In applications where there is a motor driving the gearbox, the low speed goes to the pulley shaft. With a horizontal conveyor, emergency brakes are not always used. However, the bigger and more downward sloping the application is, the greater the likelihood of implementing an emergency braking system. In the nuclear industry, low-speed brakes are often in the specifications written by the customer, which means that these braking systems are preferred by end users who perceive their applications as high-risk in the event of an emergency.[12] There are currently no rules or regulations in the United States that make emergency braking systems mandatory for any given EOT crane braking application. *AIST Technical Report No. 6* states, "Hoists may also have an additional redundant hoist braking system comprised of a rope drum flange caliper-type disc braking system. This would be used in addition to the motor speed shaft braking system…"[3] Europe, however, takes a more conservative and earnest stance on this matter as European standard EN 14492-2 was implemented, which cites that emergency braking systems on EOT cranes are mandatory under specific circumstances. In his article "Breaking the Taboo," Andrew Pimblett, managing director of Street Crane, says, "In Europe the crane design standards have recently become far more prescriptive with regard to ensuring the safety of personnel in certain steel mill crane applications. For example, cranes for 'lifting and transporting hot molten masses' must now be designed in such a way that in the event of a structural component failing in the kinetic chain the load is prevented from falling. This makes it mandatory to fit emergency brakes that act on the hoist drum in the event of a failure in the hoist transmission. Mandatory 'safety categories' have also been introduced for crane control systems involving electronics."[13] It seems

Figure 11

(a) (b) (c)

Emergency caliper brake systems: air (a), magnetic (b) and hydraulic released (c).

prudent for the United States to consider making emergency brakes a requirement as well for relevant applications. The amount of increased safety when adding caliper-style brakes on the drum is significant, and their benefits are incapable of being quantified. The question ultimately becomes, "Should emergency braking systems for cranes lifting hot metal be optional and subject to the discretion of the end user?" When it comes to the rare but catastrophic consequences that may ensue when a shaft breaks or other type of low-speed side failure occurs, installing emergency brakes as a means to prevent the load from falling is unequivocally the right choice because strict adherence to safety is a fundamental principle of any iron- and steelmaking facility.

Example of Emergency Braking System Used for 500-Ton Main Hoist Ladle Crane — A main hoist ladle crane with a 500-ton maximum capacity is a workhorse in the iron- and steelmaking industry. In this application, the emergency braking system will attach to a disc that is connected to the drum. The objective of emergency braking systems is to prevent a load from falling if the primary brakes fail or another failure occurs, such as the shaft breaking, which would thus render the primary and secondary/holding brakes useless since their rotating discs or drums connect to the shaft. Therefore, the emergency brake must be capable of preventing the entire load from falling, which means its torque must be greater than or equal to the entire full-motor torque of the crane. For instance, take a 265-hp motor with 400 rpm and a 65-to-1 gearbox ratio. The required torque value would be equal to:

Full-motor torque = (265 hp x 5,250)/400 rpm = approximately 3,480 ft-lbs torque

3,480 ft-lbs torque x 1.5 (service factor) x 65 (gearbox ratio) = 339,300 ft-lbs torque

With two motors per system, the total required torque is equal to:

339,300 x 2 = 678,600 ft-lbs torque

A torque value of 678,600 ft-lbs requires a powerful braking system. Under the circumstances, certain types of brakes, such as magnetically and thruster-released, will not provide sufficient torque for this application. Hydraulic disc brakes, however, will provide the required braking torque and thus are often the preferred brake for applications with extremely demanding torque requirements. Using a hydraulic caliper disc brake that has 200,500 lb. of applied force, a 0.4 coefficient of friction and a 112-inch disc, the braking torque amounts to 349,248 ft-lbs Multiplying this value by two (two brakes per system), the torque yield is 698,496 ft-lbs, which will be more than sufficient to accommodate the 678,500 ft-lbs of full-motor torque. These brakes would be actuated by a hydraulic power unit, which would furnish the pressure needed to release these brakes (emergency brakes are spring-applied and fail-safe).

Specifying an emergency braking system can be an arduous task that requires a substantive analysis. Several factors must be considered, including real estate, drum clearance and required braking torque, among others. The initial costs incurred to design and manufacture braking systems are high in comparison to primary service brakes, holding brakes and variable frequency drives. Albeit a subjective statement, it seems fair to assert that the overall benefits of emergency braking systems far outweigh the costs. Many other industrial applications, such as conveyors and sag mills in mining, utilize emergency brakes because there is too great a risk posed if a braking system is not in place to accommodate the entire load. For ladle cranes at iron- and steelmaking facilities, it is difficult to quantify the likelihood of a falling load occurring. However, the catastrophic consequences of brake failure are so severe that it is essential that facility managers make the investment in saving machinery, production time, resources and, most importantly, human lives. Therefore, prudence suggests that emergency brakes whose benefits far exceed their costs are a necessity rather than a luxury in iron- and steelmaking facilities.

Conclusions

Industrial braking systems are used on virtually every crane in iron- and steelmaking facilities, and there is a multitude of brakes from which to choose. At a cursory glance, a brake is just a brake, and its function is to stop something from moving. Specific brakes are chosen for different reasons. Decision making can be predicated on psychological biases, such as using the same brake for many years and not wanting to change due to one being familiar with the mechanics and functionality of a particular brake. Sometimes a brake is chosen based on a "package deal," which includes multiple items or a complete crane and thus the crane manufacturer's brake of choice is used, and other times cost is the most important consideration for brake choice. So what is the optimal solution for an industrial braking system to be used on an EOT crane? The reality is there is no one-size-fits-all product for overhead crane brakes. The brake selection process is highly analytical and includes a great deal of consideration involving multiple factors. While critical numbers such as braking torque and applied force are scientific in nature, choosing a brake is more art than science and relies on a diligent and scrupulous approach to weighing several factors. For instance, at face value, a brake may be known to yield a specific torque value. This torque value is based on certain assumptions and ideals, such as a normal and clean operating environment and linings/pads that have a specific coefficient of friction. However, there are myriad variables to consider which affect the braking torque that will actually be achieved in a certain environment, and a thorough analysis in specifying a brake will contribute to a more successful braking system. Brakes can take on many different forms, such as drum, disc, magnet, thruster, hydraulic and pneumatic. Oftentimes, there are multiple brakes that can suit a specific application. There is certainly no perfect brake; every brake has its advantages and disadvantages. Hydraulic disc brakes are ideal because they can accommodate very high torque requirements, but they also utilize hydraulic fluid, which can be messy and potentially create issues pertaining to leakage or environmental concerns. Magnet brakes are ideal because of their simplicity and cleanliness, and thruster brakes can allow a smoother braking and often have adjustable torque scales. Disc brakes have fewer moving parts, which may mean less maintenance, and greater contact between the disc and the pad is achieved through having a flat surface. However, disc brakes can generate an uneven load as opposed to their drum brake counterpart. Situational factors must be taken into consideration. For instance, if hydraulic fluid is already accessible, it may be prudent to use hydraulic disc brakes.

A proactive approach in specifying an EOT crane braking system is the best route to take. Weighing every pertinent variable involved, becoming educated on the various types of brakes to choose from and understanding their strengths and weaknesses, and scrupulously analyzing the factors involved will make for a more successful choice. A long-run approach to choosing an industrial braking system will equate to less downtime, lower maintenance costs and heightened safety. The world of industrial braking systems continues to evolve and innovate, and the successful design and implementation of a new braking system can be achieved in large part by keeping safety as a paramount factor, having an open mind, and being sensitive to the inherent and extrinsic factors that affect the braking environment.

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