Difference between dynamic and static coefficient of friction

Dr. Eng. Römer

Drum brakes or disc brakes (see figure 2 and 1) are well known worldwide. The simple and effective functionality of these brakes ensures a high level of safety at a favorable price-performance ratio if the brakes have been correctly dimensioned.

The actuation of such brakes is mainly done by an electro hydraulic thruster according to the German industry standard DIN 15430. Unfortunately, this simple brake system has only one function-related defect, especially when used for hoists in cranes:

The dead time

Due to the functioning of the electro-hydraulic lifting device (thruster), you have to calculate with a dead time (reaction time) for emergency braking. The dead time describes the period from the moment of actuating the mechanical friction brake until its effectiveness.

Effectiveness of the brake means that only at that precise moment when the brake torque is effective, the brake shoes rub on the brake drum or brake disc. The produced mechanical energy is converted by friction in the friction layer into heat. This created friction energy in the friction layer changes the physicochemical properties of the friction pairing, especially the brake lining itself.

As a consequence and provided that certain safety limits are observed (contact pressure, temperature of the friction layer, sliding velocity) a certain friction coefficient, generally known as μ_{dyn} will result.

Of course, not all maximum values may occur simultaneously, so that the specific friction energy for organically bound brake linings of $\dot{q} = 400 - 450\ W / cm^2$ should not be exceeded.

Exceeding the individual limits or even more is not allowed and will inevitably lead to a significant reduction of the friction coefficient. A failure of the brake could be the result.

The earlier a hoist brake works the smaller the over speed will be, i. e. the rotational speed at the beginning of braking.

Using the example of a speed course of a hoist with a load during the lowering you will recognize the significance and influence of dead time on the braking process.

At the time when the emergency stop braking or emergency braking should apply 1, the motor will be normally turned of (disconnected from the power supply). From this point until the effectiveness of the mechanical braking 2, the lifting capacity accelerates the speed of the hoist.

The longer the dead time, the higher the over-speed and thus the speed at the start of braking will be.

The period from releasing the brake until standstill is called braking time.

If it was possible to shorten the dead time and thus minimise overspeed, the brake time and energy could be reduced as well."

So far, braking was always triggered by having the thruster off, as well as the lift motor. Due to the coasting of the impeller with the rotor of the thruster motor, time passes until the application of the brake. Usually, 300-500 ms pass.

If an emergency stop has been carried out, a local overload of the brake lining happens especially if the brake shoes show a bad contact pattern.

This creates a local burning of the brake lining and friction carbon will be the result.

Furthermore, an abrasive brake shoe, due to non-equal lifting, could cause a thermal overload, before a braking operation is performed. Thus, an equal lifting of the brake shoes is absolutely necessary for a good functionality of the friction brake.

An example will be used to describe the catastrophic effects of a grinding brake shoe when a friction brake is lifted.

Assumption:

1.) The slight grinding stress of a brake lining would cause a braking torque of 100 Nm.
2.) The speed of the brake drum or brake disk would be 1500 rpm.
3.) The friction energy will result to:

Figure 1: An example disc brake.

Figure 2: An example drum brake.

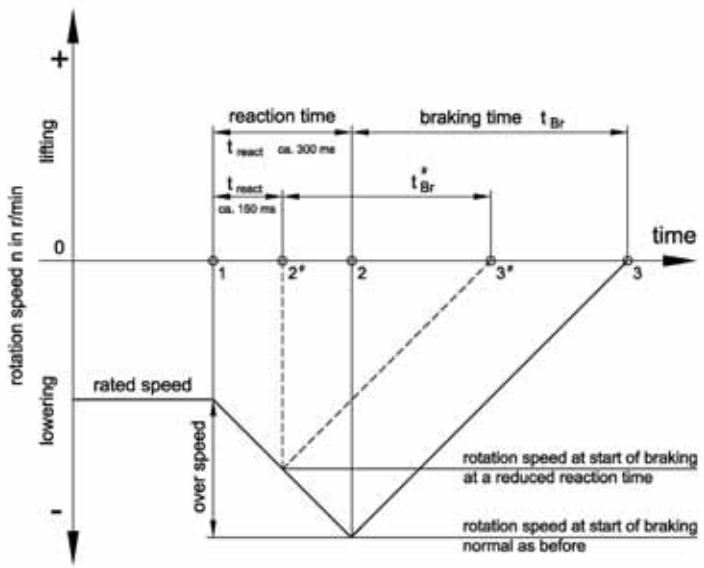

Figure 3: Diagram of dead time impact on braking process.

$P_{fric} = T_{grind} \times \omega$
P_{fric} = 15,7 kW = Heating power for a family house during winter (-20°C)

At this energetic load the grinding brake lining wears extremely quick (within a few days only) so that the residual stroke of the thruster comes to zero. The brake spring can no longer support itself on the brake shoes and the brake torque is no longer available.

Using a synchronous releasing mechanism and a correct assembly of the brake guarantee an equal lifting gap at the brake shoes.

High-tech friction brake

Through a special control of the electro-hydraulic thruster, the dead time of 300 - 500 ms will be reduced now to **150 ms only**, independent of the load acting on the thruster.

The speed development in a lifting mechanism is clearly shown in figure 3.

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Difference between dynamic and static coefficient of friction

In Mechanics you hear talking about static friction coefficient > coefficient of sliding friction. This statement applies to the use of natural materials only.

The organically bound brake lining is a specially developed material, which generates an acceptable higher friction coefficient of normally μ_{stat} = 0.4, even at higher sliding speeds, higher temperatures and large surface pressure. A stop brake always generates heat by friction. For this type of braking the dynamic medium friction coefficient μ_{stat} = 0.35 to 0.4, indicated by friction manufacturers, applies.

However, frequency-controlled drives have lately been used during the recent years to an increasing extent. As a consequence the friction brake operates only as a holding brake. In these cases no friction will be produced so that the static friction coefficient will be considerably lower than the dynamic friction coefficient

$\mu_{stat} = 0.5 \text{ to } 0.6 \bullet \mu_{dyn}$

This low static coefficient of friction is unfortunately frequently ignored when calculating a holding brake or it will be mistaken with the dynamic coefficient of friction, so that, under certain circumstances, the load is possible to slip if the brake has been calculated to small.

A secure holding torque - but how to realize?

The calculation of a friction brake regarding its holding function during standstill could be described as follows:

Required holding torque = $T_{Mn} \bullet 2.5$
T_{Mn} = nominal torque of the hoist motor

For such a calculation you have to consider that after each emergency stop or shut down braking with lowering full or partial load, several simple brakings without load have to follow in order to remove possible friction carbon in the friction layer or on top of the brake drum or brake disk respectively.

In addition it is possible that a 2nd brake spring, which only switched on at zero speed, compensates the smaller static friction by a higher contact pressure.

Monitoring systems for friction brakes

Changes in the brake behavior of mechanical friction brakes could be detected early for stop-brakes as the braking distance was getting more and more longer. However, holding brakes hold or not. An early detection of this complicated situation is very difficult. For this reason it is advisable to monitor and control the brake constantly regarding its functionality.

A limit switch should monitor the position of a brake sending the signal to the crane operator:

Limit switch 1: "Brake open"
Limit switch 2: "Brake closed"
Limit switch 3: "Readjust brake":

Particularly Limit switch 3 "Readjust brake" has proven to be especially important and appropriate.

There is nothing worse for a spring applied friction brake, than the missing of the required residual stroke of the thruster. At the moment, the spring force of the brake spring cannot be transmitted to the brake shoe and the brake is without any function. Such a situation can be avoided if the limit switch "Readjust brake" signals the need of service.

When using older brakes it can happen that the pin joints are getting stiff. The result is that the brake applies and lifts more slowly.

If now the movement speed of the lever, to which the brake spring and also the thruster are connected, is getting slower, you could measure the changed speed in conjunction with a linear device and compare the set value by using a PLC control system.

Now if the actual value is considerably different from the set value, the crane operator gets an respective signal.

With todays storage technology it is very easy to store the measured values and to notice a time related deviation of certain set values.

An extension of such a surveillance system would allow to transmit signals via Internet to client-oriented service networks.

Few additional features are required to control the temperature of the thruster, speed of the brake drum during an emergency brake or the number and frequency of dynamic brakings. These values could be sent to authorized service points by data communication, too. Data communication by multi-channel systems or by radio do not pose a particular problem today.

Automated systems like cranes cannot be operated safely anylonger, if the operating systems are not controlled an surveilled sufficiently.

One major dammage could exceed the costs by multiples of a high tec brake system which could save lives, time and money.

A danger detected is a danger less!

ABOUT THE AUTHOR

Dr. Eng. Römer studied engineering. In 1982 he gained his doctorate at Bochum's University, Chair of Machine Parts and Materials Handling.

As Owner and General Manager of Römer Fördertechnik, he has almost 40 years expertise in industrial brake technology.

ABOUT THE COMPANY

Römer Fördertechnik GmbH, based in the German city of Wetter, is specialised in the production of safety components and systems used in the fields of material handling and motion. Since being founded by Dr. Eng. Roland Römer in 1982, the company has grown to cover more than 8400 m² and four plants. All Römer products are distributed wordwide and have the highest level of safety technology in the market. Professional and competent consultants, quality products and constant availability of standard parts are cornerstones of Römer Fördertechnik.

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