

There are many factors to take into account when determining what type of brake to use on any particular machine. For example, what is the function of the brakes? Are there any external loads? Are there any time delays?

The primary consideration should be to answer the question:

“What do you want the brake to do?”

The application will often fall into one of three main types:

- **Holding Brakes**

In such static applications a suitable factor should always be used to compensate for the possibility of foreign matter contaminating the disc surface, loss of conditioning of the brake pad surface, and to provide an appropriate safety margin, depending on pad material, statutory or other requirements.

- **Dynamic and Emergency Brakes**

In a dynamic application the brake must stop all of the moving parts of the machine. Often this function is asked for in conjunction with the holding requirements as an

emergency braking function. Basic calculations should, in these cases, include an assessment of operating temperature, power dissipation (to ensure that the pads are maintained in good condition), and expected pad life.

- **Tension (Continuous Duty) Brakes**

This application arises when the brake is called upon to provide a continuous torque or tension on material passing through the machine, for example on tension reels. Special considerations are required to ensure adequate pad life.

Braking calculations are fundamental to good brake selection. The following calculations are intended only as a guide for those wishing to make their own selection. For a more detailed analysis of a proposed braking system, Twiflex Application Engineers can assist in providing details of all expected operating characteristics using specialist software.

Definition of Braking Terms

Clamping Force (F_n) is the force pressing each brake pad against the disc.

Braking Force (F_b) is the tangential friction force acting between the brake pads and disc.

$$F_b = 2 \cdot \mu \cdot F_n$$

Where: μ is the coefficient of friction between the pad and the disc (a nominal value of 0.4 is assumed for fully conditioned pads of standard material).

Braking Torque (T_b) is the moment of braking force about the center of rotation.

$$T_b = F_b \cdot r_e$$

Where r_e is the effective disc radius.

Calculated braking torques for the range of Twiflex brake calipers are shown in the brochure for a range of standard disc sizes.

Braking Calculations

Definition of Symbols and Units

It is important, when making calculations, that a consistent set of units is used.

Symbols and units

ω_m	Maximum disc speed	[rad/sec]
ω	Deceleration during braking	[rad/sec ²]
J	Total inertia referred to braked shaft	[kgm ²]
m	External load	[kg]
T_B	Total braking torque	[Nm]
T_L	Load torque (out-of-balance torque)	[Nm]

T_J	Inertia Torque	[Nm]
T_F	Friction torque	[Nm]
t_b	Braking time	[sec]
t_d	Delay time for brake signal	[sec]
t_s	Overall stopping time	[sec]
g	Acceleration due to gravity	[m/sec ²]

Basis of Brake Calculation

The fundamentals of braking calculations are to ensure that:

- There is sufficient torque to stop and hold the machine at rest
- For dynamic stops, the power dissipation is acceptable for the area of the brake pads being used
- The operating temperature of the brake disc is controlled, to avoid brake fade and reduced performance

The basic data needed to make an initial calculation are any out-of-balance forces acting on the machine, and the total inertia of the moving parts of the machine.

The total braking torque is defined as follows:

$$T_B = T_J + T_L - T_F$$

Generally the effects of friction can be ignored for a conservative estimate of the required braking torque.

It is important, in all cases of dynamic braking, that the inertia of all of the moving parts is referred to the braked shaft when calculating **T_J** (see below).

Brake Calculation

Basic braking calculations are derived from simple mechanics. Both the effects of out of balance loads and the torque needed to stop the inertia in the desired way need to be calculated.

Load torque **T_L**

Where the machine has any out-of-balance load applied the effective out-of-balance torque must be calculated. In the case of a suspended load, for example this is simply given by

$$T_L = m \cdot g \cdot r \text{ [Nm]}$$

where **r** is the radius at which the load acts (based on drum diameter, reel diameter etc.) The brake will need to overcome this load before it can start to slow down the machine; (it is assumed that the out of balance opposes the action of the brake). If the load is at rest, the static brake torque will prevent the load from moving. In practice a safety factor should be used in the case where the brake is called upon only to hold this load and is only infrequently used in a dynamic manner. In these cases a service factor of 2 is recommended to allow for external environmental conditions, i.e. the brake should be rated to give twice this calculated value.

With some types of machines, such as downhill conveyors or escalators, a component of the load acts in the direction of motion. In these cases the following equation should be used:

$$T_L = m \cdot g \cdot r \cdot \sin \theta \text{ [Nm]}$$

Where θ is the angle of the motion to the horizontal. In cases where the load does not produce a direct torque on the braked shaft (eg. there is a gearbox between the loaded shaft and the brake) the load torque should be calculated accordingly.

In cases of dynamic braking, where the load torque is not constant, for example on wind turbines where the aerodynamic load varies with the speed of the machine, then the equation for T_L becomes complex and it is recommended that selection should be referred to Twiflex.

Inertia torque T_J (dynamic braking only)

Having overcome the external forces additional braking torque is required to stop all the moving parts of the machine. This additional torque is calculated as follows:

$$T_J = J \cdot \dot{\omega} \text{ [Nm]}$$

In this equation the deceleration required can readily be calculated from the braking time.

$$\dot{\omega} = \omega / t_b$$

In some special cases more precise calculation of speed and time may be required.

The moment of inertia of the system (J) should represent all moving parts. For example, in a hoist calculation allowance should be made to include the motor, brake disc, gearbox, winding drum, ropes, load etc. again with a suitable allowance for any mechanical advantage at various parts of the system.

The effective inertia of the load is given simply as:

$$J = m \cdot r^2 \text{ [kgm}^2\text{]}$$

For the winding drum the following well established formula can be used for an initial estimate (can also be used for flat discs):

$$J = \pi \rho l (D^4 - d^4) / 32 \text{ [kgm}^2\text{]}$$

Where l is the length of the drum,

D is the outside diameter

d is the inside diameter and ρ is the density of the material (7840 kg/m³ for steel)

The inertia of the motor should also be included in the calculations. These values can be estimated but should ideally be obtained directly from the manufacturer.

Note: All inertia must be referred to the brake shaft.

For example, if the brake is positioned on the low speed shaft of a machine (gearbox output shaft) then all calculated inertia values (motor, couplings, etc.) must be referred to this shaft.

When referring inertia J_B from one shaft with speed ω_B to another shaft with speed ω_A use:

$$J_A = J_B (\omega_B / \omega_A)^2$$

Special requirements

In certain cases there are other direct considerations to be applied when making braking calculations.

In mine hoists the service factor for holding is defined, as are allowable levels of deceleration, especially when manriding.

For escalators, allowable stopping distances or even rates of deceleration may be defined in local standards.

Twiflex is able to offer assistance with calculations in such special cases.

Braking Calculations

Other Considerations

To optimise braking performance there are other considerations which must be taken into account.

Rubbing speed

For high-speed applications a further consideration is the linear speed at which the disc is passing between the pads when the brake operates. This is important and is usually measured at the effective radius.

Thus the rubbing speed is given as:

$$\text{Rubbing speed} = \omega_m \cdot r_e \text{ (m/s)}$$

In general a maximum speed of 30m/s is recommended for standard brake pad materials. Above this value it is likely that the effective coefficient of friction will be reduced leading to a reduction in braking performance. For certain applications where the rubbing speed is particularly high (up to 100m/s) special pads manufactured from sintered materials are available. Refer to Twiflex for details.

Operating temperatures

During a dynamic application of a brake the energy of the machine will be converted to heat, generated between the pad and the disc. It is the temperature of the disc surface that is normally used to assess the brake performance. Failure to take account of the peak temperature can lead to a reduced braking performance due to the onset of brake fade. With standard brake pads a peak temperature of 250°C has been found to be acceptable, although in certain cases higher temperatures might be permitted. In the case where sintered pads are used peak temperatures in excess of 600°C are possible. The operating temperature also determines pad wear. The higher the temperature, the greater the wear rate of the pads. These calculations require a detailed analysis, carried out by the application engineers at Twiflex, using dedicated software.

Power dissipation

In order to provide a consistent controlled performance of a brake it is also important to check the power dissipated during a stop. This affects the condition of the brake pads.

To calculate the power dissipation it is necessary to calculate the total energy absorbed during the stop, estimated as follows:

$$\text{Kinetic energy (KE)} = J \cdot \omega_m^2 / 2 \text{ (Joules)}$$

In the case of external loads further allowance should also be made for the change in potential energy of the system. For example in the case of a hoist this is determined by how far the load drops during the braking cycle. The mean power dissipated is therefore given by:

$$\text{Mean power dissipation} = \text{KE} / t_b \text{ (Watts)}$$

This is usually converted to what is termed the Mean Specific Power Dissipation (kW/cm²) for evaluation of pad performance, i.e. the mean power dissipation is divided by the pad area.

A value of 0.7kW/cm² has been shown to be acceptable for emergency stops of around 10 seconds duration, providing that the pads have been well bedded in and conditioned. Higher values can be used for shorter stopping times.

For tensioning duties the value of the mean specific power dissipation is more typically around 0.06 kW/cm². Failure to observe this basic selection criteria may result in poor braking performance and limited pad life.

Twiflex application engineers will be only too happy to assist in the selection process to meet your application requirements.