CHAPTER 30
CLUTCHES AND BRAKES

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GLOSSARY OF SYMBOLS

a	Vehicle deceleration, ft/s^2 (m/s^2)
a	Location of shoe pivot, in (m)
a	Lever arm for larger band force, in (m)
A	Area, in^2 (m^2)
b	Percentage of grade
b	Width of band, shoe, or web, in (m)
c	Lever arm for smaller band force, in (m)
C	Center of pressure
C	Specific heat, Btu/(lbm · °F) [J/(kg · °C)]
d	Inside disk diameter, in (m)
D	Outside disk diameter, in (m)
D	Pitch diameter of gear, in (m)
D_{max}	Maximum roll diameter, in (m)
e	Radius to center of circular brake pad, in (m)
E	Total energy dissipated, ft · lb or Btu (J)
f	Coefficient of friction
f_v	Ventilation factor

F	Actuating force, lb (N)
F_D	Prime-mover factor
F^ℓ	Actuating force on leading shoe, lb (N)
F_L	Load factor
F_n	Normal force, lb (N)
F_S	Starting factor
F_t	Tension force on web, lb (N)
F^t	Actuating force on trailing shoe, lb (N)
g	Local acceleration of gravity, ft/s^2 (m/s^2)
g_c	Gravitational constant, 32.174 lbm · ft/(lb · s^2) [1 kg · m/(N · s^2)]
h	Overall heat transfer coefficient, Btu/(in^2 · s · °F) [W/(m^2 · °C)]
h_c	Convection heat transfer coefficient, Btu/(in^2 · s · °F) [W/(m^2 · °C)]
h_r	Radiation heat transfer coefficient, Btu/(in^2 · s · °F) [W/(m^2 · °C)]
H_{av}	Average rate of heat dissipation, lb · ft/s or Btu/s (W)
I	Mass moment of inertia, lb · in · s^2 (kg · m^2)
I_L	Mass moment of inertia on load side, lb · in · s^2 (kg · m^2)
I_P	Mass moment of inertia on prime-mover side, lb · in · s^2 (kg · m^2)
K_S	Service factor
ℓ	Moment arm of actuating force (drum brake); length of actuating lever in a band brake, in (m)
m	Mass, lbm (kg)
M_a	Moment of actuating force, lb · in (N · m)
M_f	Moment of resultant friction force, lb · in (N · m)
M_n	Moment of resultant normal force, lb · in (N · m)
n	Shaft speed, r/s (Hz) or r/min
N	Number of pairs of friction surfaces in disk clutches or brakes
N	Number of shoes in centrifugal clutch
p	Normal pressure, psi (MPa) or r/min
p_{av}	Average contact pressure, psi (MPa)
p_h	Hydraulic pressure, psi (MPa)
p_{max}	Maximum contact pressure, psi (MPa)
p^ℓ_{max}	Maximum contact pressure on leading shoe, psi (MPa)
p^t_{max}	Maximum contact pressure on trailing shoe, psi (MPa)
P	Resultant normal force between drum and shoe, lb (N)
P	Power, Btu/s or hp (kW)
P_x	Component of normal force in x direction, lb (N)
P_y	Component of normal force in y direction, lb (N)
P_1	Larger band tension, lb (N)
P_2	Smaller band tension, lb (N)
q	Rate of energy dissipation during clutch slip, Btu/s (W)

Q	Actuating force (band brake), lb (N)
r	Brake drum radius, in (m)
r	Radius to point on disk, in (m)
r_f	Radius to center of pressure, in (m)
R	Tire-rolling radius, in (m)
R	Reaction force (drum brake), lb (N)
R	Radius to rim of centrifugal brake, in (m)
R_e	Effective friction radius, in (m)
R_i	Inside radius, in (m)
R_o	Outside radius, in (m)
s	Total stopping distance, ft (m)
S	Initial tension, lb (n)
S	Stops per hour
t	Web thickness, mils (mm)
t_d	Combined delay time for driver reaction and brake system reaction, s
t_s	Total stopping time, s
T	Torque; nominal torque, lb · ft (N · m)
T_a	Temperature of surrounding air, °F (°C)
T_d	Disk temperature, °F (°C)
ΔT	Temperature rise, °F (°C)
T_{des}	Design torque, lb · ft (N · m)
T_L	Load torque, lb · ft (N · m)
T_{max}	Maximum torque, lb · ft (N · m)
T_P	Prime mover torque, lb · ft (N · m)
V	Rubbing velocity, ft/s (m/s)
V_o	Initial velocity, ft/s (m/s)
V_f	Final velocity, ft/s (m/s)
V_w	Web velocity, ft/s (m/s)
w	Web tension per unit thickness and unit width, lb/(mil · in) [N/(mm · m)]
W	Vehicle weight, lb (N)
α	Cone angle, deg
δ	Multiplier for circular disk brake pads
ε	Angular position of actuation force, deg
θ	Angle of wrap, deg
θ	Angular position, deg
θ_1	Starting position of brake shoe lining, deg
θ_2	Ending position of brake shoe lining, deg
ω	Shaft speed, rad/s
ω_e	Engagement speed, rad/s
ω_o	Initial shaft speed, rad/s

ω_f Final shaft speed, rad/s

Ω_L Initial shaft speed on load side of clutch, rad/s

Ω_P Initial shaft speed on prime-mover side of clutch, rad/s

This chapter begins with an introduction to brakes and clutches, the various types and their applications. The problem of energy dissipation and temperature rise is discussed along with the proper selection of friction materials. Design methods are presented for almost every type of brake and clutch. A discussion of the actuation problems of brakes and clutches, including electromagnetic devices, is also presented.

30.1 TYPES, USES, ADVANTAGES, AND CHARACTERISTICS

30.1.1 Types of Clutches

The characteristic use of a clutch is to connect two shafts rotating at different speeds and bring the output shaft up to the speed of the input shaft smoothly and gradually.

Classifying clutches is done by distinguishing (1) the physical principle used to transmit torque from one member to another and (2) the means by which the members are engaged or by which their relative speed is controlled. Here, we classify clutches as follows:

1. Engagement or actuation method
   a. Mechanical
   b. Pneumatic
   c. Hydraulic
   d. Electrical
   e. Automatic
2. Basic operating principle
   a. Positive contact
      (1) Square jaw
      (2) Spiral jaw
      (3) Toothed
   b. Friction
      (1) Axial
      (2) Radial
      (3) Cone
   c. Overrunning
      (1) Roller
      (2) Sprag
      (3) Wrap-spring
   d. Magnetic
      (1) Magnetic particle
      (2) Hysteresis
      (3) Eddy current
   e. Fluid coupling
      (1) Dry fluid
      (2) Hydraulic

Coupling Methods. *Positive-contact clutches* have interlocking engaging surfaces to form a rigid mechanical junction. Three types of positive-contact clutches are shown in Fig. 30.1.

Frictional clutches are used most frequently. Two opposing surfaces are forced into firm frictional contact. Figures 30.2, 30.3, and 30.4 show *axial, radial,* and *cone* types.

Overrunning clutches are used when two members are to run freely relative to each other in one direction but are to lock in the other. Roller, sprag, and wrap-spring types are shown in Fig. 30.5. In the *roller-ramp clutch* (Fig. 30.5*a*), the members are locked together when the rollers (or balls) ride on a race with a slight cam profile. Eccentric cams are pinched between concentric races in the *sprag-type clutches* (Fig. 30.5*b*). And in the basic *wrap-spring clutch* (Fig. 30.5*c*), the spring's inside diameter is slightly smaller than the outside diameters of the input and output hubs. When the spring is forced over the two hubs, rotation of the input hub in the

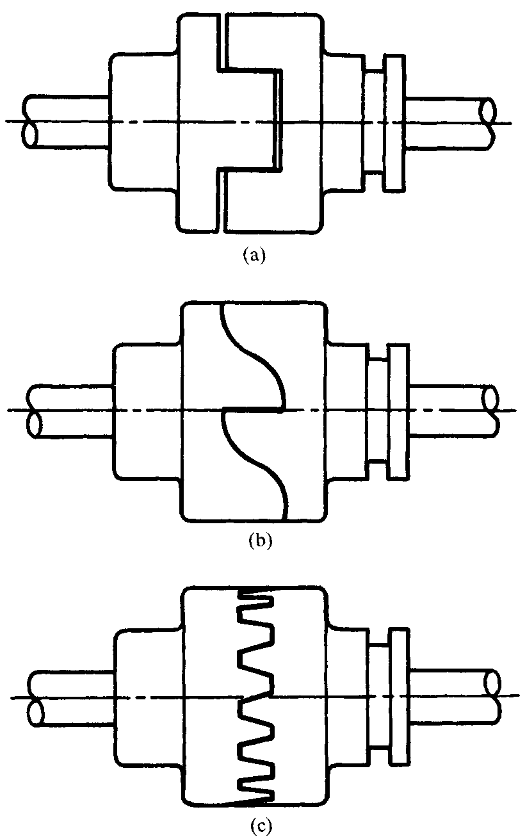


FIGURE 30.1 Positive-contact clutches. (*a*) Square-jaw, the square teeth lock into recesses in the facing plate; (*b*) spiral-jaw, the sloping teeth allow smoother engagement and one-way drive; (*c*) toothed-clutch, engagement is made by the radial teeth.

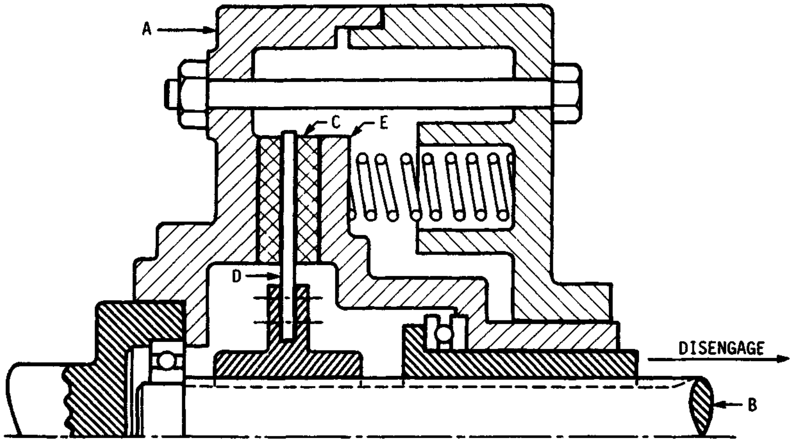


FIGURE 30.2 Schematic drawing of an axial clutch; *A,* driving member; *B,* driven shaft; *C,* friction plates; *D,* driven plate; *E,* pressure plate.

drive direction causes the spring to tighten down on the hubs. Torque is then transmitted. But rotation in the opposite direction opens the spring, and no torque is transmitted.

A *magnetic clutch* (Sec. 30.8) uses magnetic forces to couple the rotating members or to provide the actuating force for a friction clutch.

Fluid couplings may make use of a hydraulic oil or a quantity of heat-treated steel shot. In the *dry-fluid coupling,* torque is developed when the steel shot is thrown centrifugally to the outside housing (keyed to the input shaft) as the input

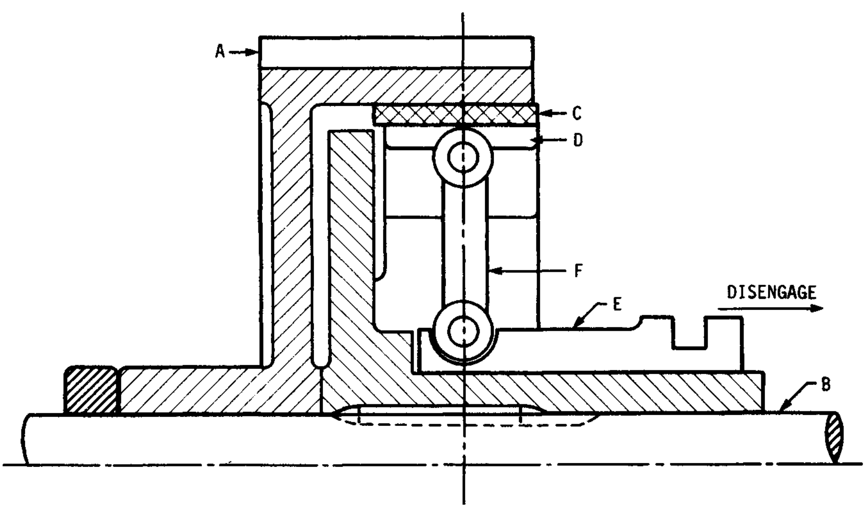


FIGURE 30.3 Schematic drawing of a radial clutch built within a gear; *A,* gear, the driving member; *B,* driven shaft; *C,* friction plate; *D,* pressure plate; *E,* movable sleeve; *F,* toggle link. This type of clutch can also be made within a V-belt sheave.

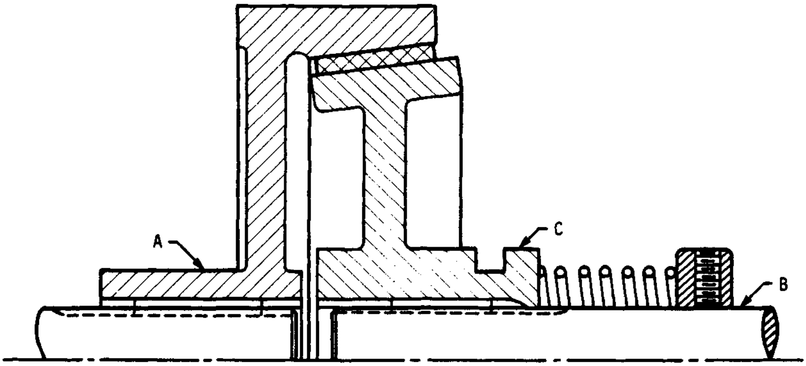


FIGURE 30.4 Schematic drawing of a cone clutch; *A,* driving member; *B,* driven shaft; *C,* movable sleeve.

shaft begins to rotate. At the design speed the shot is solidly packed, and the housing and rotor lock together.

Control Methods. *Mechanical control* is achieved by linkages or by balls or rollers working over cams or wedges. The actuating force can be supplied manually or by solenoid, electric motor, air cylinder, or hydraulic ram.

Electrical control of friction or tooth clutches often involves engaging the clutch electrically and releasing it by spring force. Thus the clutch is *fail-safe:* If power fails, the clutch is disengaged automatically. But where shafts are coupled for much longer periods than they are uncoupled, the opposite arrangement may be used: spring force to engage, electromagnetic force to disengage.

Pneumatic, or *hydraulic, control* is accomplished in several ways. Actuating pistons may be used either to move the actuating linkage or to directly apply a normal force between frictional surfaces. In other designs an inflatable tube or bladder is used to apply the engagement force. Such designs permit close control of torque level.

Automatic control of clutches implies that they react to predetermined conditions rather than simply respond to an external command. Hydraulic couplings and eddy-current clutches both have torque regulated by the slip. *Centrifugal clutches* (Fig. 30.6) use speed to control torque.

30.1.2 Selecting Clutches

A starting point is a selection table constructed by Proctor [30.5] and reproduced here as Table 30.1. Four additional tables in Proctor's article also are useful for preliminary decisions. Designers will have to consult the manufacturers before making final decisions.

30.1.3 Types of Brakes

Physically, brakes and clutches are often nearly indistinguishable. If two shafts initially at different speeds are connected by a device to bring them to the same speed,

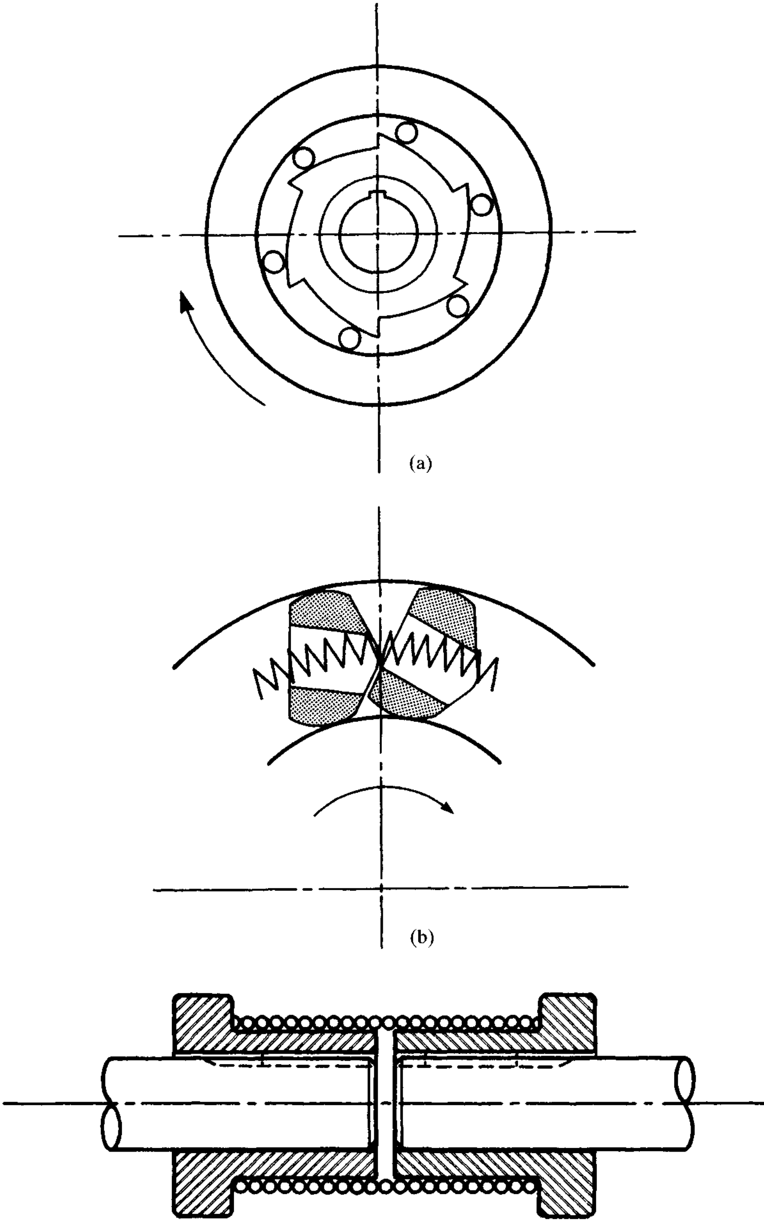


FIGURE 30.5 Overrunning clutches. (*a*) Roller-ramp clutch; springs are often used between the rollers and the stops. (*b*) Portion of a Formsprag clutch. Rockers or *sprags,* acting as cams, are pushed outward by garter springs at both ends of the prismatic sprags. (*c*) Torsion spring winds up when the clutch is in "drive" and grips both hubs. Larger-torque loads can be carried by making the springs of rectangular-section wire.

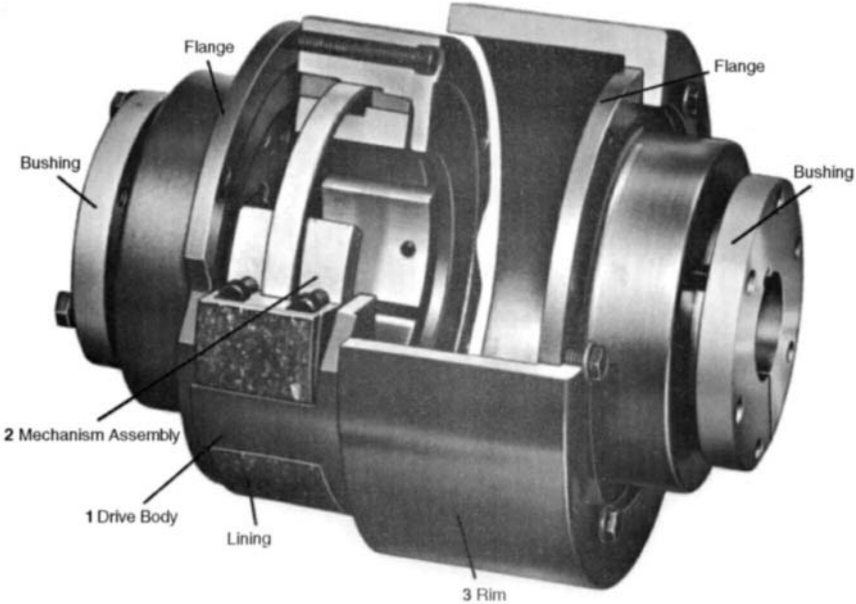


FIGURE 30.6 Centrifugal clutch. *(BLM Automatic Clutch Limited.)*

it is a clutch. If one member is fixed and the torque is used to slow down or stop the rotating member, the device is a brake. A classification scheme for brakes is presented in Fig. 30.7.

Brake Configuration. *Band brakes* can be made *simple* (not self-energizing) or *differential* (self-energizing). In designing a differential band brake (Fig. 30.19), care must be taken to ensure that the brake is not self-locking.

Short-shoe brakes have been used for hoists. Centrifugal brakes employ speed as the actuating signal for short-shoe internal-block brakes and are used in a wide variety of applications.

Drum brakes (Fig. 30.8) are used principally for vehicles, although seldom on the front axles of passenger cars. On the rear axles, drum brakes supply high braking torque for a given hydraulic pressure because one or both of the long shoes can be made self-energizing.

For a *leading shoe,* the friction moment exerted on the shoe by the drum assists in actuating the shoe. The friction moment on a *trailing shoe* opposes the actuating moment. Thus a leading shoe is *self-energizing,* but a trailing shoe is *self-deenergizing.*

The *leading-shoe trailing-shoe* design (Fig. 30.8) provides good braking torque in forward or reverse. The *two-leading-shoe* design has an even higher braking torque in forward, but a much lower braking capacity in reverse. Very high braking torque is available from the *duo-serve* design. Here the friction force on the "leading shoe" assists in actuating the "trailing shoe."

One difficulty with drum brakes is instability. If a brake's output is not sensitive to small changes in the coefficient of friction, the brake is *stable.* But if small changes

TABLE 30.1 Selecting the Right Clutch

Load characteristic or clutch function	Type of clutch: General utility	Centrifugal and fluid self-actuating	Continuous slip: Automatic	Continuous slip: Variable	Overrunning or freewheeling
1. No-load start					
a. Manual or externally controlled	✓	...	...	...	✓
b. Automatic	...	✓			
2. Smooth load pickup					
a. Normal load	✓	✓	...	...	✓
b. High-inertia load	...	✓	...	✓	
c. High breakaway load (more than 100% running torque)	...	✓	...	✓	
d. Automatic delayed pickup	...	✓	...	✓	
e. Extended acceleration	...	✓	...	✓	
f. Auxiliary starter	...	...	...	...	✓
3. Running operation					
a. Normal load (no slip at full load, full speed)	✓	✓	✓		
b. Control variable-torque load	...	...	...	✓	
c. Control constant-torque load	...	...	✓	✓	
d. Control constant-tension load	...	...	...	✓	
4. Overload protection and stopping					
a. General protection: transient and infrequent overloads	✓	✓			
b. Limit speed (prevent runaway load)	...	✓	...	...	✓
c. Limit torque	...	✓	✓	...	✓
d. Automatic overload release	✓	...	...	...	✓
e. Dynamic braking	...	✓	...	...	✓
f. Backstopping	...	✓	...	...	✓
5. Intermittent operation					
a. On-off, with driver at speed	✓	...	...	...	✓
b. Inching and jogging	✓				
c. Indexing and load positioning	...	...	...	...	✓
6. Dual-drive and standby operation	...	✓	...	...	✓

SOURCE: Ref. [30.5]

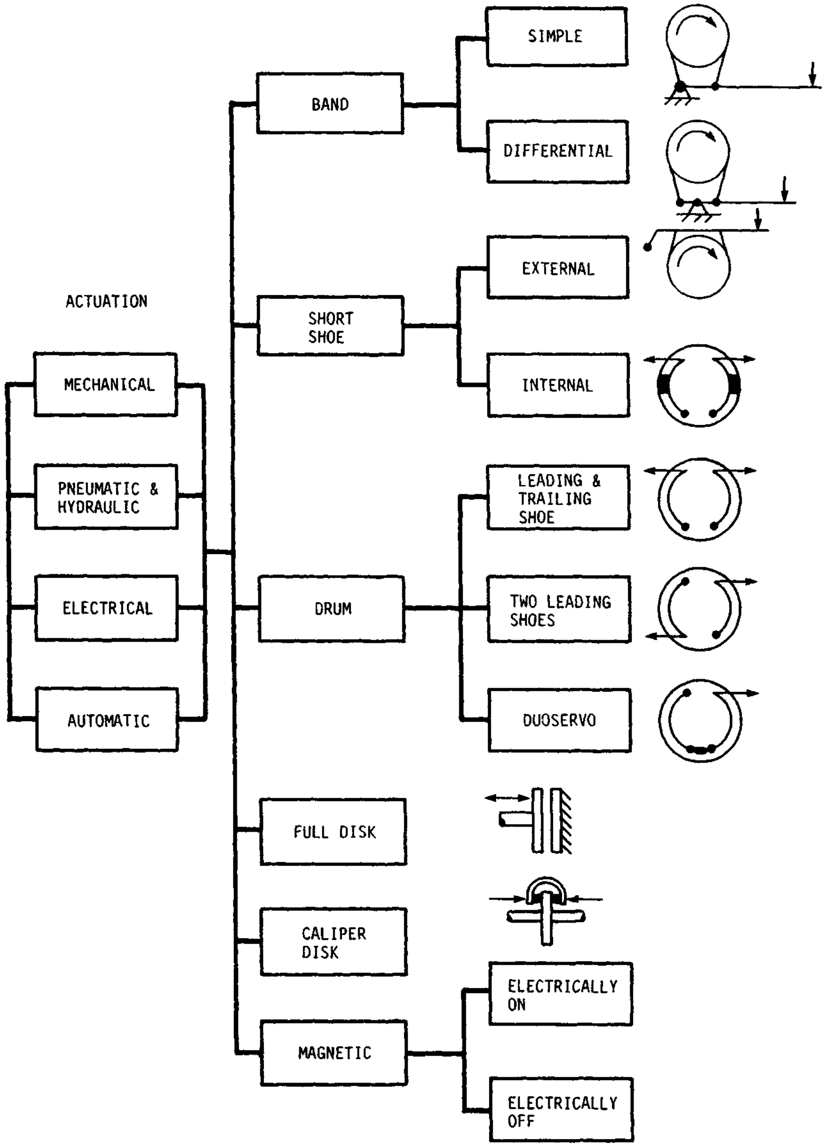


FIGURE 30.7 Classification of brakes.

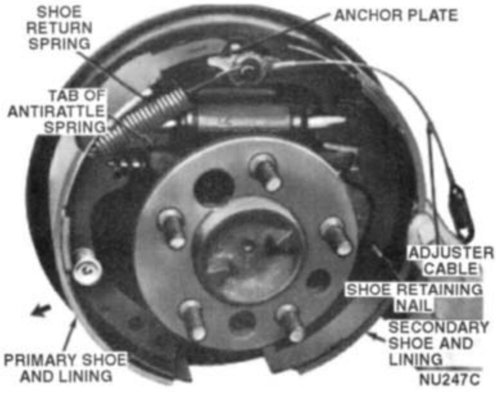


FIGURE 30.8 Drum brake. *(Chrysler Corporation.)*

in the friction coefficient cause large changes in brake torque, the brake is *unstable.* It will tend to grab and squeal if the coefficient of friction increases. But if the coefficient decreases (say, because of a temperature increase), there will be a noticeable drop in capacity.

Full disk brakes are used principally for industrial machinery. They are very much like full disk clutches in construction. Indeed, they are found in *clutch-brake combination drives* where both members of the drive are full disk in construction.

Caliper disk brakes (Fig. 30.9) are now familiar components of vehicles, but they find applications in industrial equipment as well. The Chrysler brake shown uses a floating caliper. In this design, an automatic mechanism to adjust for pad wear can be incorporated easily.

Generally caliper disk brakes are not self-energizing, although they can be. An advantage of the non-self-energizing disk brake is its great stability; it is relatively insensitive to changes in the coefficient of friction.

Brake Actuation. Four principal actuation methods are shown in the classification chart of Fig. 30.7: mechanical, pneumatic (or hydraulic), electric, and automatic. Sometimes the methods are combined.

The drum brake of Fig. 30.8 and the disk brake of Fig. 30.9 are both *hydraulically operated.* Both are intended for vehicles. In industrial applications, air is often the actuating fluid. The air-tube configuration in Fig. 30.10 can be used for a *pneumatically operated* clutch or brake.

Electrically operated brakes most commonly use electromagnetic forces to actuate a full-disk-friction brake. However, a number of other designs are found (see Sec. 30.8).

Automatically operated brakes are used for both transportation and industrial equipment. Sometimes manual operation is overridden by automatic actuation. Truck brakes are available with spring actuators that engage if the air pressure is lost. The air brake, as originally conceived by George Westinghouse for railroad applications, was of this *fail-safe* design. Electrically, hydraulically, and pneumatically operated brakes can all be designed for automatic operation. Antiskid brakes for automobiles and trucks superimpose on the usual manual control an automatic control that releases braking pressure if lockup and skidding are imminent.

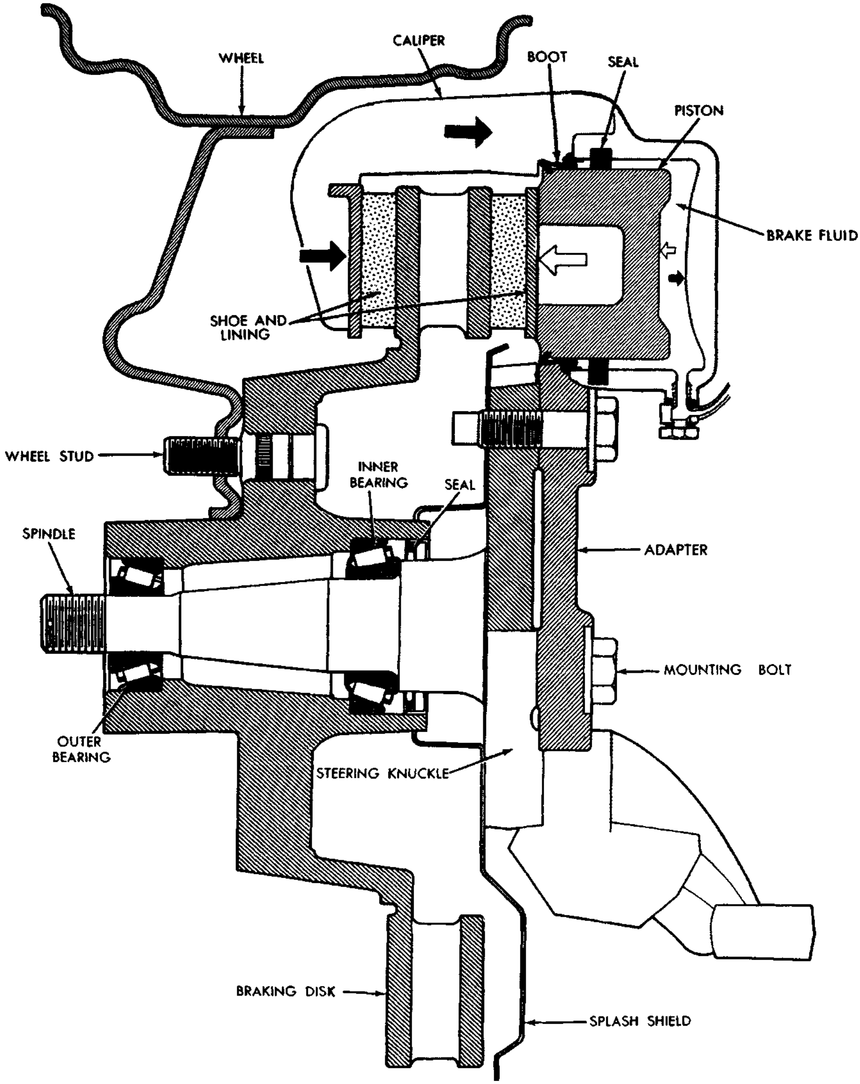


FIGURE 30.9 Automotive disk brake. *(Chrysler Corporation.)*

30.1.4 Selecting a Brake

To help narrow the choice of a brake, Table 30.2 has been provided. Some general indicators are given for performance requirements and environmental conditions. Typical applications are listed as well. The *brake factor* is the ratio of the frictional braking force developed to the actuating force.

Note that temperature considerations have been omitted from the environmental conditions in Table 30.2. For high temperatures, the capacity of all brakes listed is limited by the type of friction material. The performance of all brakes listed is con-

FIGURE 30.10 A pneumatically actuated brake using an expandable tube. *(Eaton Corporation.)*

sidered *good* for low temperatures, but ice buildup must be avoided.

30.2 TORQUE AND ENERGY CONSIDERATIONS

In selecting or designing a clutch, the torque requirement, energy dissipation, and temperature rise are the principal factors to be considered. The torque requirement and energy dissipation are covered in this section. Estimating temperature rise is the subject of Sec. 30.3.

30.2.1 Torque Requirement: Clutches

The torque requirement of a clutch will have to be substantially greater than the nominal torque it is transmitting in order to accelerate the load. The character of the prime-mover output torque and of the load torque also influence the designer's selection of torque capacity.

Gagne [30.4] recommended the following technique for calculating clutch capacity for design purposes. Calculate the design torque as a multiple of the nominal torque T:

$$T_{\mathrm{des}} = K_S T \tag{30.1}$$

where K_S = service factor taking into account the load inertia, the character of the prime mover's output torque, and the character of the load torque. The service factor K_S is

$$K_S = (F_S^2 + F_D^2 + F_L^2 - 2)^{1/2} \tag{30.2}$$

where F_S, F_D, and F_L are the starting, prime-mover, and load factors, respectively. Recommended values for these factors are given in Tables 30.3 to 30.5. Note that if each factor is unity, the service factor is unity also.

But the service factor K_S will usually be greater than unity. Indeed, an old rule of thumb was that the clutch should be designed for a torque capacity at least twice the nominal torque.

Example 1. A multicylinder diesel engine is used to drive an electric generator in a hospital's emergency-power facility. What service factor should be used?

Solution. From Table 30.3, a reasonable selection of values for the torque factors is $F_S = 2$, $F_D = 1.5$, and $F_L = 1.0$. The corresponding service factor is

$$K_S = [2^2 + (1.5)^2 + (1.0)^2 - 2]^{1/2} = 2.29$$

TABLE 30.2 Selecting the Right Brake

Type of brake	Performance requirements: Maximum operating temperature	Performance requirements: Brake factor	Performance requirements: Stability	Environmental conditions: Wet and humid	Environmental conditions: Dust and dirt	Typical applications
Differential band brake	Low	Very high	Very low	Unstable but still effective	Good	Winches, hoists, excavators, tractors, etc.
External drum brake (leading-trailing edge)	Low	Moderate	Moderate	Unstable if humid; poor performance if wet	Good	Mills, elevators, winders
Internal drum brake (leading-trailing edge)	Higher than external brake	Moderate	Moderate	Unstable if humid; completely ineffective if wet	Very good if sealed	Vehicles (rear axles on passenger cars)
Internal drum brake (two leading shoes)	Higher than external brake	High	Low	Unstable if humid; completely ineffective if wet	Very good if sealed	Vehicles (rear axles on passenger cars)
Internal drum brake (duo-servo)	Low	Very high	Low	Unstable if humid; completely ineffective if wet	Very good if sealed	Vehicles (rear axles on passenger cars)
Caliper disk brake	High	Low	High	Good	Poor; should be shielded	Vehicles and industrial machinery
Full disk brake	High	Low	High	Good	Poor; should be shielded	Machine tools and other industrial machinery

SOURCE: Ref. [30.6].

TABLE 30.3 Suggested Values of Torque Starting Factor F_s for Friction Clutches

Type of load	F_S
Free start; no load	1.0
Average inertia load	2.0
High inertia load	3.0

SOURCE: Ref. [30.4].

TABLE 30.4 Suggested Values of Torque Drive (Prime-Mover) Factor F_D for Friction Clutches

Type of drive	F_D
Nonpulsating, such as three-phase motors	1.0
Moderate pulsation—single-phase motors, multicylinder engines, etc.	1.5
Severe pulsation, such as a single-cylinder gas engine	2.0

SOURCE: Ref. [30.4].

TABLE 30.5 Suggested Values of Torque Load Factor F_L for Friction Clutches

Type of load	F_L
Nonpulsating—blowers, centrifugal pumps, generators under steady load, etc.	1.0
Moderate shock, such as a multicylinder pump	1.5–1.75
Severe shock—crane, shovel, single-cylinder compressor, rock crusher, etc.	2.0–3.0

SOURCE: Ref. [30.4].

30.2.2 Equivalent Inertias

Two shafts geared together and rotating at different speeds are shown in Fig. 30.11*a*. The inertias I_1 and I_2 are each assumed to include the corresponding shaft and gear. For design calculations, it is necessary to have an equivalent inertia for the whole system referred to a single shaft. Figure 30.11*b* and *c* shows this. In each case an equivalent inertia has been added to the shaft. So I_2' is the equivalent inertia on shaft 1 of shaft 2 and its hardware. Similarly, I_1' is the equivalent inertia on shaft 2 of shaft 1 and its hardware.

A simple way to find the equivalent inertia is to equate the kinetic energies of the actual and equivalent inertias. Thus, to find the equivalent inertia I_2' referred to shaft 1 (Fig. 30.11*b*), we write

$$\frac{1}{2}I_2'\omega_1^2 = \frac{1}{2}I_2\omega_2^2$$

Thus

$$I_2' = \left(\frac{\omega_2}{\omega_1}\right)^2 I_2 \tag{30.3}$$

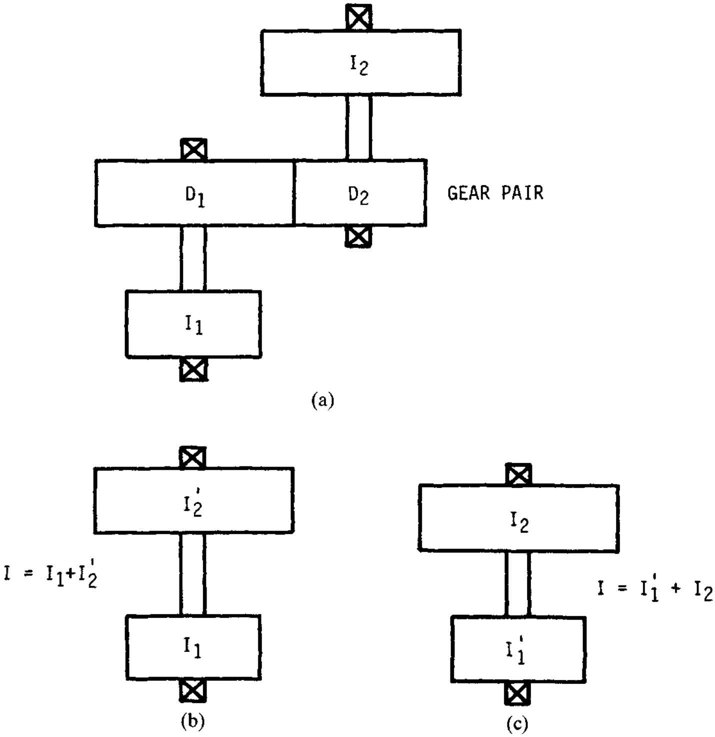


FIGURE 30.11 Equivalent inertia. I_1, I_2 = inertia of input and output shafts, respectively; I' = equivalent inertia. (*a*) Original configuration; (*b*) equivalent system referred to input shaft; (*c*) equivalent system referred to output shaft. For a more extensive treatment of equivalent inertias, see "Suggested Reading" list, Mischke.

Similarly,

$$I_1' = \left(\frac{\omega_1}{\omega_2}\right)^2 I_1 \tag{30.4}$$

In general,

$$I_i' = \left(\frac{\omega_i}{\omega_j}\right)^2 I_i = \left(\frac{n_i}{n_j}\right)^2 I_i \tag{30.5}$$

where the equivalent inertia of the *i*th shaft is referred to the *j*th shaft. Equation (30.5) can be used to reduce a machine with several shafts connected by gears or flexible connectors to a single equivalent shaft.

Example 2. For the two-shaft machine in Fig. 30.11*a*, the inertias are I_1 = 2.88 pound-inch-square seconds (lb · in · s^2) [0.3254 kilogram-square meters (kg · m^2)] and I_2 = 0.884 lb · in · s^2 (0.09988 kg · m^2). The pitch diameters of the gears are D_1 = 4 in [0.102 meter (m)] and D_2 = 7 in (0.178 m). What is the equivalent inertia of shaft 2 referred to shaft 1?

Solution. Equation (30.3) can be used once the speed ratio ω_2/ω_1 is known. For spur or helical gears,

$$\frac{\omega_2}{\omega_1}=\frac{D_1}{D_2}=\frac{4}{7}=0.5714$$

Thus

$$I_2'=(0.5714)^2(0.884)=0.2887 \text{ lb}\cdot\text{in}\cdot\text{s}^2\ (0.0326 \text{ kg}\cdot\text{m}^2)$$

30.2.3 Torque Requirement: Brakes

Industrial Brakes. The torque to bring a rotating machine from an initial speed ω_o to a lower one ω_f (perhaps to rest) in a slowdown time of t_S is

$$T=\frac{I(\omega_o-\omega_f)}{t_S} \tag{30.6}$$

Vehicle Brakes. The braking torque to stop a vehicle of weight W at a deceleration rate a on a grade of b percent can be estimated as

$$T=\frac{WR}{g}\left(\frac{a}{g}+\frac{b}{100}\right) \tag{30.7}$$

Here R is the tire-rolling radius. This is a conservative approach; both tire-rolling resistance and air resistance have been neglected. Of course, this torque capacity T must be allocated to the several brakes in a rational way (for example, in proportion to the weight of the vehicle supported by the corresponding wheel during a panic stop).

For parking-brake capacity, simply set $a=0$ in Eq. (30.7).

The required acceleration rate a can be determined by setting either a total stopping time t_S or a total stopping distance s:

$$a=\frac{V_o}{t_S-t_d} \tag{30.8}$$

$$a=\frac{V_o^2}{2(s-V_o t_d)} \tag{30.9}$$

In these equations, t_d is the combined delay time (about 1 s for a passenger car) for driver reaction and brake system reaction.

30.2.4 Energy Dissipation: Clutches

A simple model of a clutch connecting a prime mover and a load is shown in Fig. 30.12. The clutch capacity is T, the driving torque provided by the prime mover is T_P, and the load torque is T_L. The inertias I_P and I_L include all rotating masses on their respective sides of the clutch.

If the two sides of the clutch are initially rotating at Ω_P and Ω_L radians per second (rad/s) when the clutch is actuated, the duration of the slip period is

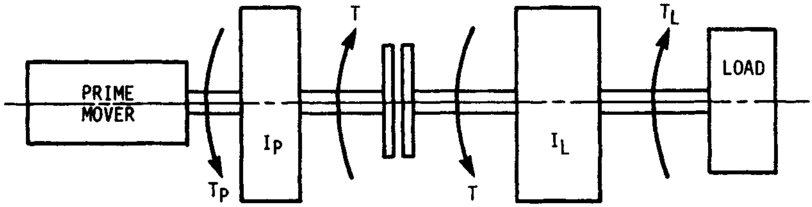


FIGURE 30.12 Abstract model of a machine using a clutch.

$$t_S = \frac{I_P I_L(\Omega_P - \Omega_L)}{T(I_P + I_L) - (I_L T_P + I_P T_L)} \tag{30.10}$$

The rate at which energy is dissipated during the slip period is, at t s from the beginning,

$$q(t) = T\left[\frac{I_L T_P + I_P T_L - T(I_P + I_L)t}{I_P I_L + \Omega_P - \Omega_L}\right] \tag{30.11}$$

And the total energy dissipated in one actuation operation is

$$E = \frac{T(\Omega_P - \Omega_L)^2 I_P I_L}{2T(I_P + I_L) - (I_L T_P + I_P T_L)} \tag{30.12}$$

30.2.5 Energy Dissipation: Brakes

Vehicle Brakes. When a vehicle of weight W is slowed from an initial velocity V_o to a final velocity V_f, the heat energy that the brakes must dissipate is equal to the change of kinetic energy E:

$$E = \frac{W}{2g}(V_o^2 - V_f^2) \tag{30.13}$$

In dealing with individual brakes, let W be that portion of the vehicle's weight for which the brake is responsible.

Example 3. A sports car weighing 3185 lb [14.2 kilonewtons (kN)] has 62 percent of its weight on the front axle during an emergency stop. What energy must each of the front-wheel brakes dissipate in braking from 55 miles per hour (mph) [88 kilometers per hour (km/h)] to rest? Local acceleration of gravity is 32.17 feet per second (ft/s) (9.81 m/s).

Solution. Each front brake is responsible for a weight of

$$W = 0.5(0.62)(3185) = 987 \text{ lb } (4.39 \text{ kN})$$

The initial velocity is

$$V_o = 55\left(\frac{88}{60}\right) = 80.7 \text{ ft/s } (24.6 \text{ m/s})$$

Finally, the energy to be dissipated is

$$E = \frac{W}{2g}(V_o^2 - V_f^2) = \frac{987}{2(32.17)}[(80.7)^2 - 0^2]$$

$$= 99\,900 \text{ lb} \cdot \text{ft } (135.5 \text{ kN} \cdot \text{m})$$

$$= 128.4 \text{ Btu } [135.5 \text{ kilojoules (kJ)}]$$

Industrial Brakes. The approach is the same as for vehicular brakes. The heat energy the brake must dissipate equals the change in kinetic energy of the rotating machine:

$$E = \frac{I}{2}(\omega_o^2 - \omega_f^2) \tag{30.14}$$

where, with n in rev/min,

$$\omega = \frac{2\pi n}{60} \tag{30.15}$$

In many industrial applications, the brakes are applied frequently. The average rate of heat dissipation is, for S stops per hour,

$$H_{av} = \frac{ES}{3600} \tag{30.16}$$

Tensioning Applications. In tensioning applications, a continuous application of the brake is required, for example, in unwinding a roll of aluminum foil. The maximum torque occurs at the maximum roll diameter D_{max}. It is

$$T_{max} = \frac{D_{max} F_t}{2} \tag{30.17}$$

The tension F_t is, for material width b and thickness t,

$$F_t = wtb \tag{30.18}$$

Typical values of tension per unit width and per unit thickness for a few materials are given in Table 30.6.

TABLE 30.6 Tension Data for Typical Materials

Material	Unit tension, lb/mil per inch of web width
Aluminum foil	1.00
Cellophane	0.75
Mylar	0.60
Polystyrene	1.00

SOURCE: The Carlson Company, Inc., Wichita, Kansas.

The rate at which heat is generated by the brake friction is

$$H_{av} = F_t V_w \tag{30.19}$$

Example 4. A printing press is to print on Mylar 0.002 in [0.051 millimeter (mm)] thick. The web velocity is 4000 ft/min (20.3 m/s). The maximum roll diameter is 55 in (1.4 m). The web is 54 in wide. Find the necessary braking torque and the rate at which heat is generated by braking.

Solution. For Mylar the unit tension is 0.60 lb/mil per inch (379.2 kN/mm per meter). So the web tension is

$$F = wtb = 0.60(2)(54) = 64.8 \text{ lb } (288 \text{ N})$$

The maximum brake torque is

$$T_{max} = \frac{D_{max}F_t}{2} = \frac{55(64.8)}{2} = 1782 \text{ in} \cdot \text{lb } (201 \text{ N} \cdot \text{m})$$

The rate at which the brake must dissipate heat is, by Eq. (30.19),

$$H_{av} = F_t V_w$$

The web velocity is

$$V_w = \frac{4000}{60} = 66.67 \text{ ft/s } (20.3 \text{ m/s})$$

So H_{av} is

$$H_{av} = 64.8(66.67) = 4320 \text{ ft} \cdot \text{lb/s [5855 watts (W)]}$$
$$= 5.55 \text{ Btu/s}$$

30.3 TEMPERATURE CONSIDERATIONS

30.3.1 Intermittent Operation: Clutches and Brakes

The temperature rise can be estimated as

$$\Delta T = \frac{E}{Cm} \tag{30.20}$$

where m [pounds mass (lbm) or kilograms (kg)] = mass of the parts adjacent to the friction surfaces. The specific heat C for steel or cast iron is about 0.12 Btu/(lbm · °F) [500 J/(kg · °C)].

30.3.2 Frequent Operation: Caliper Disk Brakes

The average rate at which heat must be dissipated can be calculated by Eq. (30.16). The disk is capable of dissipating heat by a combination of convection and radiation.

And the convection-heat transfer is sensitive to the velocity of air moving over the disk. The rate at which the disk can dissipate heat is

$$H_{\text{diss}} = hA(T_d - T_a) \tag{30.21}$$

The overall heat transfer coefficient h is

$$h = h_r + f_v h_c \tag{30.22}$$

The heat transfer coefficients for radiation h_r and convection h_c are plotted in Fig. 30.13 against the temperature rise of the disk above the surrounding air. The ventilation factor f_v is plotted against the velocity of the moving air in Fig. 30.14.

Example 5. An industrial caliper brake is used 19 times per hour on average to stop a machine with a rotating inertia of $I = 328$ lb · in · s^2 (37.06 kg · m^2) from a speed of 315 rev/min. The mean air velocity over the disk will be 30 ft/s (9.14 m/s). What minimum exposed area on the disk is needed to limit the disk's temperature rise to 200°F (111°C)?

Solution. From Figs. 30.13 and 30.14, $h_r = 3.1 \times 10^{-6}$ Btu/(in^2 · s · °F), $h_c = 2.0 \times 10^{-6}$ Btu/(in^2 · s · °F), and $f_v = 5.25$. The overall heat transfer coefficient is

$$h = 3.1 \times 10^{-6} + 5.25(2.0 \times 10^{-6}) = 13.6 \times 10^{-6} \text{ Btu/(in}^2 \cdot \text{s} \cdot \text{°F)}$$

The energy the brake must dissipate per stop is, by Eq. (30.14),

$$E = \frac{I}{2}(\omega_o^2 - \omega_f^2)$$

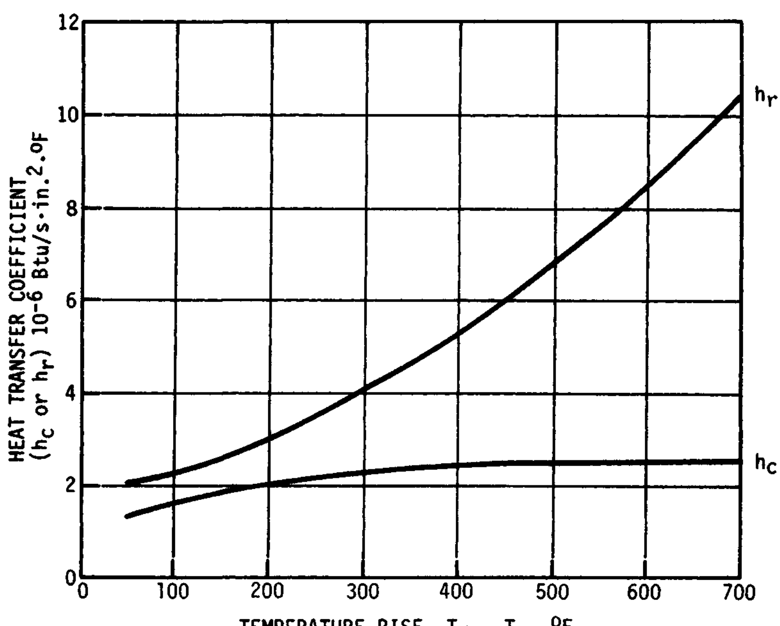


FIGURE 30.13 Heat transfer coefficients in still air. *(Tol-o-matic.)*

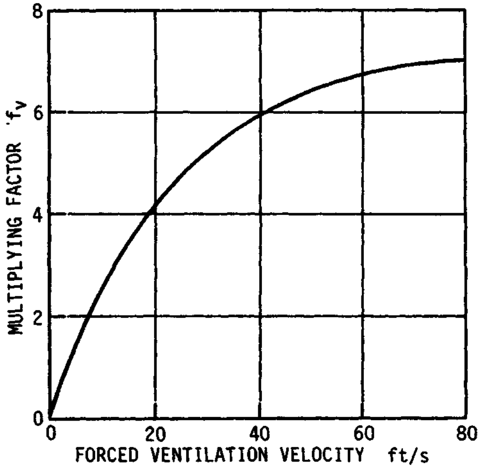


FIGURE 30.14 Ventilation factors. *(Tol-o-matic.)*

Here

$$\omega_o = \frac{2\pi}{60}(315) = 33 \text{ rad/s} \qquad \omega_f = 0$$

Therefore,

$$E = \frac{328}{2}(33^2) = 178.6 \times 10^3 \text{ lb} \cdot \text{in}$$

$$= 19.1 \text{ Btu } (20.150 \text{ kJ})$$

The average rate of energy generation is found by Eq. (30.16):

$$H_{av} = \frac{ES}{3600} = \frac{19.1(19)}{3600} = 0.101 \text{ Btu/s } (107 \text{ W})$$

The disk area needed can be calculated by using Eq. (30.21) and setting $H_{av} = H_{diss} =$ 0.101 Btu/s. Thus

$$A = \frac{H_{diss}}{h(T_d - T_a)} = \frac{0.101}{(13.6 \times 10^{-6})(200)} = 37.1 \text{ in}^2 \ (0.0239 \text{ m}^2)$$

30.4 FRICTION MATERIALS

To help in preliminary design, Tables 30.7 and 30.8 have been compiled, principally from data supplied by British and U.S. manufacturers of friction materials. Although these data are representative, they are hardly exhaustive. And they should be used

TABLE 30.7 Characteristics of Friction Materials for Brakes and Clutches

Material	Friction coefficient μ	Maximum pressure p_{max}, psi	Maximum temperature		Maximum velocity V_{max}, ft/min	Applications
			Instantaneous, °F	Continuous, °F		
Cermet	0.32	150	1500	750		Brakes and clutches
Sintered metal (dry)	0.29–0.33	300–400	930–1020	570–660	3600	Clutches and caliper disk brakes
Sintered metal (wet)	0.06–0.08	500	930	570	3600	Clutches
Rigid molded asbestos (dry)	0.35–0.41	100	660–750	350	3600	Drum brakes and clutches
Rigid molded asbestos (wet)	0.06	300	660	350	3600	Industrial clutches
Rigid molded asbestos pads	0.31–0.49	750	930–1380	440–660	4800	Disk brakes
Rigid molded nonasbestos	0.33–0.63	100–150		500–750	4800–7500	Clutches and brakes
Semirigid molded asbestos	0.37–0.41	100	660	300	3600	Clutches and brakes
Flexible molded asbestos	0.39–0.45	100	660–750	300–350	3600	Clutches and brakes
Wound asbestos yarn and wire	0.38	100	660	300	3600	Vehicle clutches
Woven asbestos yarn and wire	0.38	100	500	260	3600	Industrial clutches and brakes
Woven cotton	0.47	100	230	170	3600	Industrial clutches and brakes
Resilient paper (wet)	0.09–0.15	400	300		$PV < 500\,000$ psi·ft/min	Clutches and transmission bands

SOURCES: Ferodo Ltd., Chapel-en-le-frith, England; Scan-pac, Mequon, Wisc.; Raybestos, New York, N.Y. and Stratford, Conn.; Gatke Corp., Chicago, Ill.; General Metals Powder Co., Akron, Ohio; D. A. B. Industries, Troy, Mich.; Friction Products Co., Medina, Ohio.

TABLE 30.8 Area of Friction Material Required for a Given Average Braking Power

Duty cycle	Typical applications	Ratio of area to average braking power, $in^2/(Btu/s)$		
		Band and drum brakes	Plate disk brakes	Caliper disk brakes
Infrequent	Emergency brakes	0.85	2.8	0.28
Intermittent	Elevators, cranes, and winches	2.8	7.1	0.70
Heavy-duty	Excavators, presses	5.6–6.9	13.6	1.41

SOURCES: Refs. [30.6], Sec. A51, and [30.7].

for preliminary design estimates only. A friction materials manufacturer should be consulted both to learn of additional options and to get more authoritative data.

Although Table 30.7 lists maximum recommended values for contact pressure and rubbing velocity, it is not very likely that you can go the limit on both parameters at once. And a careful distinction must be made between the maximum temperature permissible for a short time and the safe temperature level for continuous operation. The temperature limit for continuous operation is much lower than that for a brief temperature peak.

Preliminary design of brakes is aided by calculating the lining area needed for the average rate at which energy has to be dissipated by the brakes (*braking power*). Table 30.8 lists values that are typical of modern design practice. Again, after using these data to make some preliminary design estimates, you will need to contact the manufacturers of the friction materials before making final design decisions.

30.5 TORQUE AND FORCE ANALYSIS OF RIM CLUTCHES AND BRAKES

30.5.1 Long-Shoe Rim Brake

One shoe of an internal expanding rim brake is shown in Fig. 30.15. Usually there is a second shoe as well. The shoe is pivoted about the fixed point A. It is actuated by a force F which can be provided in a number of ways: mechanically, hydraulically, pneumatically, electromagnetically, or by some combination of these.

The forces on the shoe include the actuating force F, a reaction force R at the pivot, the distributed normal force, and the distributed friction force, the latter two exerted by the drum on the shoe.

For purposes of analysis, the distributed normal and frictional forces on the shoe can be replaced by a resultant normal force P and a resultant frictional force fP. Use of these fictional concentrated forces simplifies the analysis. There is one odd consequence, however. The resultant frictional force fP has to be regarded as acting beyond the surface of the shoe at some point C, the *center of pressure*. Figure 30.16 shows the shoe subjected to this equivalent force system.

Pressure Distribution along Lining. A first step in developing an equation for the torque capacity of the shoe is to adopt a model for the pressure distribution along

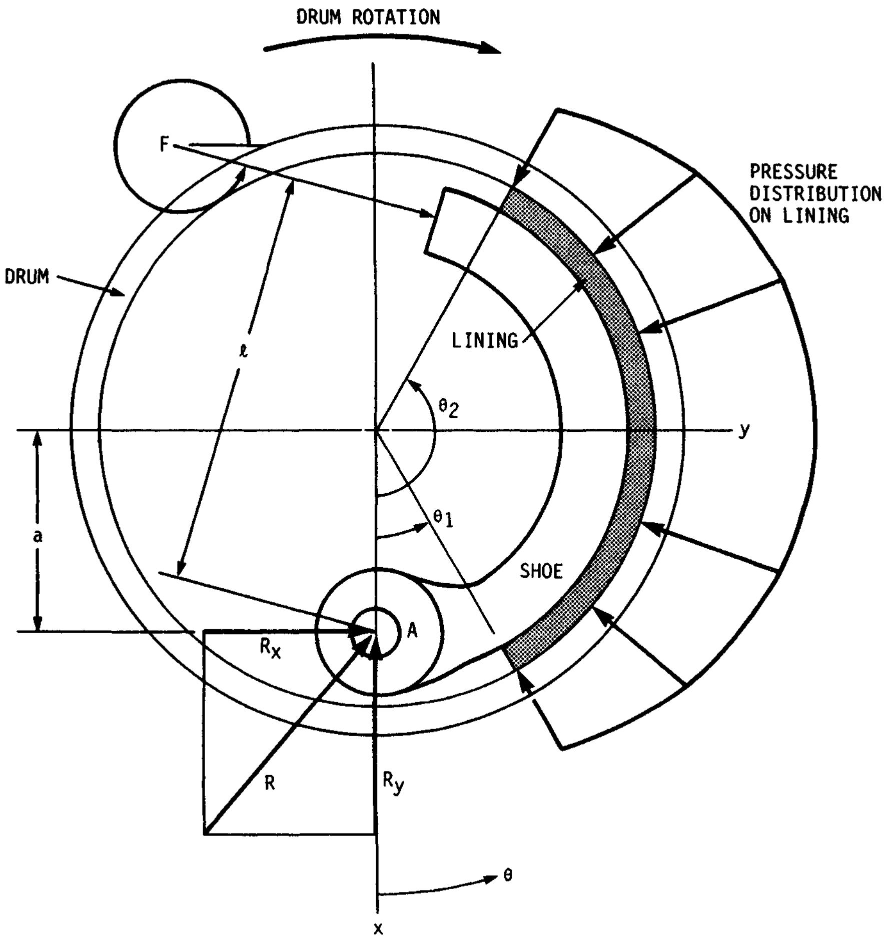


FIGURE 30.15 Forces on an internal long shoe.

the lining of the shoe. Assuming that the lining is elastic and of uniform thickness, but that the drum is perfectly rigid, the expression for the normal pressure p on the lining at any angular position θ is

$$p = \frac{p_{\max} \sin \theta}{(\sin \theta)_{\max}} \tag{30.23}$$

In the usual case, the shoe straddles the $\theta = 90°$ position and $(\sin \theta)_{\max} = 1$.

The variation of contact pressure along the lining is significant. To assume a uniform pressure is to oversimplify. For example, for a shoe extending from $\theta = 30°$ to $\theta = 165°$, the pressure varies from $0.5p_{\max}$ at 30° to $p_{\max}$ at 90°, to $0.26p_{\max}$ at 165°.

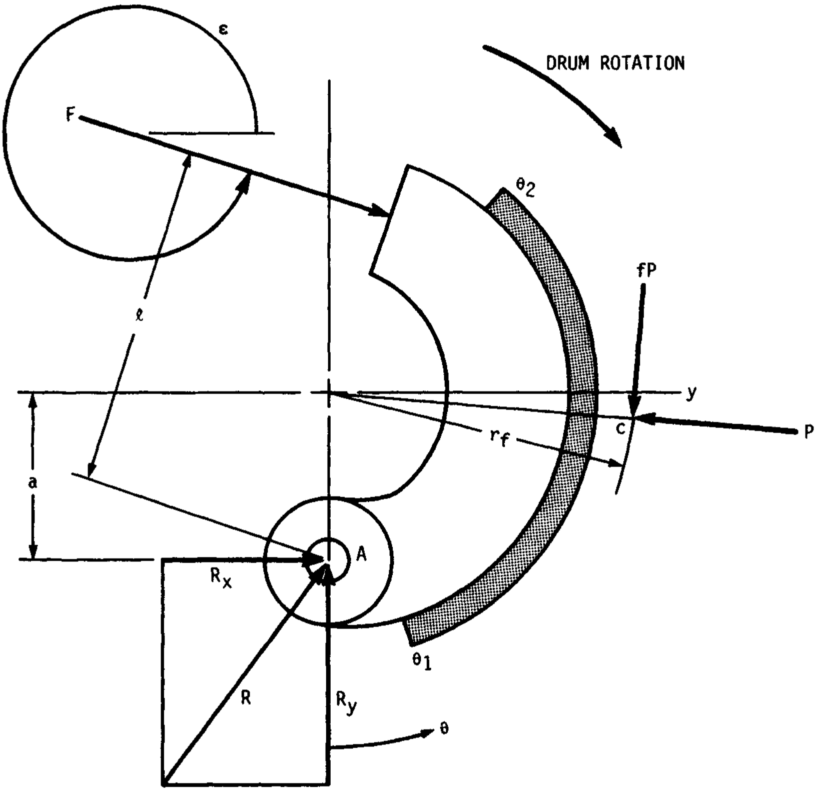


FIGURE 30.16 Equivalent force system on a long internal shoe.

Torque Capacity. With a model for contact pressure variation in hand, the friction torque exerted by the shoe on the drum can be found by a simple integration:

$$T = \int_{\theta_1}^{\theta_2} fpbr^2 \, d\theta \tag{30.24}$$

where b = width of lining. Substituting for p from Eq. (30.23) and integrating, we get

$$T = \frac{p_{\max}}{(\sin\theta)_{\max}} fbr^2(\cos\theta_1 - \cos\theta_2) \tag{30.25}$$

Normal Force. To determine the actuating force and the pin reaction, it is necessary first to find the normal force P. The components of P are

$$P_x = \int_{\theta_1}^{\theta_2} pbr \, d\theta \sin\theta \tag{30.26}$$

and

$$P_y = \int_{\theta_1}^{\theta_2} pbr \, d\theta \cos\theta \tag{30.27}$$

Again, substituting for p from Eq. (30.23) and integrating, we find

$$P_x = \frac{brp_{\max}}{2(\sin\theta)_{\max}}(\theta_2 - \theta_1 + \sin\theta_1\cos\theta_1 - \sin\theta_2\cos\theta_2) \tag{30.28}$$

and

$$P_y = \frac{brp_{\max}}{2(\sin\theta)_{\max}}(\cos^2\theta_1 - \cos^2\theta_2) \tag{30.29}$$

The resultant normal force P has the magnitude

$$P = (P_x^2 + P_y^2)^{1/2} \tag{30.30}$$

and is located at the angle θ_p, where

$$\theta_p = \tan^{-1}\frac{P_x}{P_y} \tag{30.31}$$

Effective Friction Radius. The location r_f of the center of pressure C is found by equating the moment of a concentrated frictional force fP to the torque capacity T:

$$T = fPr_f \qquad \text{or} \qquad r_f = \frac{T}{fP} \tag{30.32}$$

Brake-Shoe Moments. The last basic task is to find a relation among actuating force F, normal force P, and the equivalent friction force fP. The moments about the pivot point A are

$$M_a - M_n + M_f = 0 \tag{30.33}$$

where

$$M_a = F\ell \tag{30.34}$$

$$M_n = Pa\sin\theta_p \tag{30.35}$$

$$M_f = P(r_f - a\cos\theta_p) \tag{30.36}$$

Self-energizing Shoes. The brake shoe in Fig. 30.16 is said to be *self-energizing,* for the frictional force fP exerts a clockwise moment about point A, thus assisting the actuating force. On vehicle brakes, this would also be called a *leading shoe.* Suppose a second shoe, a *trailing shoe,* were placed to the left of the one shown in Fig. 30.16. For this shoe, the frictional force would exert a counterclockwise moment and oppose the action of the actuating force. Equation (30.33) can be written in a form general enough to apply to both shoes and to external shoes as well:

$$M_a - M_n \pm M_f = 0 \tag{30.37}$$

Burr [30.2], p. 84, proposes this simple rule for using Eq. (30.37): "If to seat a shoe more firmly against the drum it would have to be rotated in the same sense as the drum's rotation, use the positive sign for the M_f term. Otherwise, use the negative sign."

Pin Reaction. At this point in the analysis, the designer should sketch a free-body diagram of each shoe, showing the components of the actuating force F, the normal force P, and the friction force fP. Then the components of the pin reaction can be found by setting to zero the sum of the force components in each direction (x and y).

Design. The design challenge is to produce a brake with a required torque capacity T. A scale layout will suggest tentative values for the dimensions $\theta_1, \theta_2, a, \ell$, and ε. From the lining manufacturer we learn the upper limit on maximum contact pressure $p_{\max}$ and the expected range for values of the frictional coefficient f. The designer must then determine values for lining width b and the actuating force F for each shoe.

Since the friction force assists in seating the shoe for a self-energizing shoe but opposes the actuating moment for a self-deenergizing shoe, a much larger actuating force would be needed to provide as large a contact pressure for a trailing shoe as for a leading shoe. Or if, as is often the case, the same actuating force is used for each shoe, a smaller contact pressure and a smaller torque capacity are achieved for the trailing shoe.

The lining manufacturer will usually specify a likely range of values for the coefficient of friction. It is wise to use a low value in estimating the torque capacity of the shoe.

In checking the design, make sure that a self-energizing shoe is not, in fact, self-locking. For a *self-locking* shoe, the required actuating force is zero or negative. That is, the lightest touch would cause the brake to seize. A brake is self-locking when

$$M_n \leq M_f \tag{30.38}$$

As a design rule, make sure that self-locking could occur only if the coefficient of friction were 25 to 50 percent higher than the maximum value cited by the lining manufacturer.

Example 6. Figure 30.17 shows a preliminary layout of an automotive brake with one leading shoe and one trailing shoe. The contact pressure on the lining shall not exceed 1000 kilopascals (kPa). The lining manufacturer lists the coefficient of friction as 0.34 ± 0.02. The brake must be able to provide a braking torque of 550 N · m. Two basic design decisions have already been made: The same actuating force is used on each shoe, and the lining width is the same for each.

Check dimension a to make sure that self-locking will not occur. Determine the lining width b, the actuating force F, and the maximum contact pressure $p_{\max}$ for each shoe.

Solution. One way to proceed is to express the braking torque T, the normal moment M_n, and the frictional moment M_f in terms of lining width b and maximum contact pressure $p_{\max}$. Then the design can be completed by equating the actuating force for the two shoes, setting the sum of the braking torques to 550 N · m, and selecting the lining width b so that the maximum contact pressure is within bounds.

1. The dimension a is

$$a = (83^2 + 25^2)^{1/2} = 86.7 \text{ mm} = 0.0867 \text{ m}$$

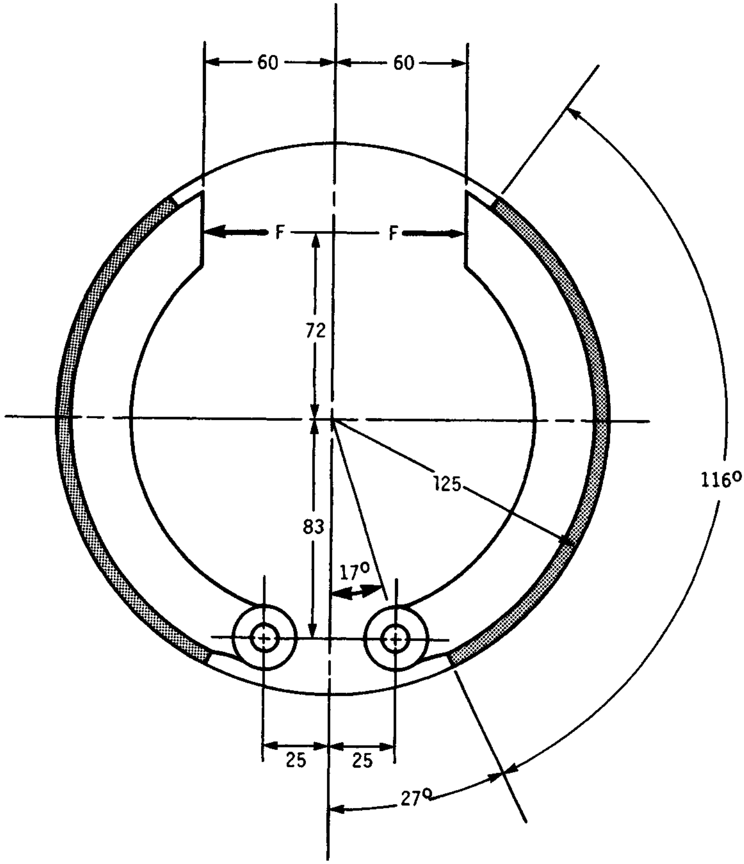


FIGURE 30.17 A leading-shoe trailing-shoe automotive brake.

2. For either shoe the braking torque can be written in terms of the coefficient of friction, the lining width, and the maximum contact pressure. From Eq. (30.25),

$$T = \frac{p_{\max}}{(\sin\theta)_{\max}} fbr^2(\cos\theta_1 - \cos\theta_2)$$

$$= \frac{p_{\max}}{1} fb(0.125)^2(\cos 10° - \cos 126°)$$

$$= 0.024\,57 fbp_{\max}$$

The numerical values of T are different for the two shoes, since $p_{\max}$ differs.

3. Now the normal force P must be calculated. Equations (30.28), (30.29), (30.30), and (30.31) are used:

$$P_x = \frac{brp_{\max}}{2(\sin\theta)_{\max}}(\theta_2 - \theta_1 + \sin\theta_1\cos\theta_1 - \sin\theta_2\cos\theta_2)$$

$$= \frac{b(0.125)p_{\max}}{2(1)}\left(\frac{126° - 10°}{57.296°} + \sin 10°\cos 10° - \sin 126°\cos 126°\right)$$

$$= 0.1671bp_{\max}$$

$$P_y = \frac{brp_{\max}}{2(\sin\theta)_{\max}}(\cos^2\theta_1 - \cos^2\theta_2)$$

$$= \frac{b(0.125)p_{\max}}{2(1)}(\cos^2 10° - \cos^2 126°) = 0.0390bp_{\max}$$

$$P = (P_x^2 + P_y^2)^{1/2} = 0.1716bp_{\max}$$

$$\theta_p = \tan^{-1}\frac{P_x}{P_y} = \tan^{-1}\frac{0.1671}{0.0390} = 76.85°$$

4. The effective friction radius r_f is, by Eq. (30.32),

$$r_f = \frac{T}{fP} = \frac{0.024\,57fbp_{\max}}{f(0.1716bp_{\max})} = 0.1432\text{ m} = 143.2\text{ mm}$$

5. The moments about pivot point A are found by Eqs. (30.34) to (30.36):

$$M_a = F\ell = F(0.072 + 0.083) = 0.155F$$

$$M_n = Pa\sin\theta_p = 0.1716bp_{\max}(0.0867)(\sin 76.85°) = 0.014\,49bp_{\max}$$

$$M_f = fP(r_f - a\cos\theta_p)$$

$$= f(0.1716bp_{\max})(0.1432 - 0.0867\cos 76.85°)$$

$$= 0.021\,19fbp_{\max}$$

6. For the leading (self-energizing) shoe, the proper form of Eq. (30.37) is

$$M_a - M_n + M_f = 0$$

Therefore

$$0.155F^l - 0.014\,49bp_{\max}^l + 0.021\,19fbp_{\max}^l = 0$$

The superscript l has been used to designate the leading shoe.

7. For the trailing shoe, Eq. (30.37) has the form

$$M_a - M_n - M_f = 0$$

Thus

$$0.155F^t - 0.014\,49bp_{\max}^t - 0.021\,19fbp_{\max}^t = 0$$

8. Since the same actuating force is used for each shoe, $F^l = F^t$. After substituting from the two moment equations, we obtain

$$\frac{0.014\,49bp^l_{\max} - 0.021\,19fbp^l_{\max}}{0.155} = \frac{0.014\,49bp^t_{\max} + 0.021\,19fbp^t_{\max}}{0.155}$$

After cancellation and substitution of $f = 0.32$ (the most pessimistic assumption),

$$p^l_{\max} = 2.759p^t_{\max}$$

9. The torque capacities of the shoe must sum to 550 N · m:

$$T^l + T^t = 550$$

$$(0.024\,57)(0.32)b(p^l_{\max} + p^t_{\max}) = 550$$

Then

$$b(p^l_{\max} + p^t_{\max}) = 69.95 \text{ kN} \cdot \text{m}$$

10. Enough information has been accumulated to begin to specify the design. Since $p^l_{\max} = 1000$ kPa, $p^t_{\max} = 1000/2.759 = 362.4$ kPa. Now the distance b can be found. Since

$$b(p^l_{\max} + p^t_{\max}) = 69.95$$

we have

$$b = \frac{69.95 \times 10^3}{(1000 + 362.4)(10^3)} = 0.0513 \text{ m, or about 51 mm}$$

11. The actuating force F ($F^l = F^t$) can be found from either shoe's moment equation. For the leading shoe,

$$0.155F^l - 0.014\,49bp^l_{\max} + 0.021\,19fbp^l_{\max} = 0$$

Substituting values of b, f, and $p^l_{\max}$ gives $F = 2.55$ kN. Thus $F = F^l = F^t = 2.55$ kN.

12. Now we determine whether the leading shoe is self-locking or too close to self-locking for safety. The moment equation used above in step 11 is used again, but this time with an f value 50 percent higher than the maximum value cited by the lining manufacturer. Use

$$f = 1.5(0.34 + 0.02) = 0.54$$

Then by substituting into the moment equation, the corresponding value of F can be found:

$$0.155F^l - (0.014\,49)(0.0513)(1000) + (0.021\,19)(0.54)(0.0513)(1000) = 0$$

Solving gives $F^l = F^t = 1.001$ kN. Since a large positive force is required to activate the leading shoe even for this very high coefficient of friction, the brake is in no danger of self-locking.

30.5.2 Centrifugal Clutches

The simple centrifugal clutch shown in Fig. 30.18 has a number of shoes which can move radially and against the drum as the input shaft speed increases. A garter spring regulates the engagement speed. At engagement speed the weights contact the drum's inner surface and begin to drive it, and the attached pulley, by means of friction to bring it up to speed.

Design Equations. The normal force between each shoe and the inner circumference of the drum is principally due to centrifugal force. However, the garter spring exerts some inward force. The net normal force [30.1] is

$$F_n = \frac{mr\omega^2}{g_c} - 2S\cos\left(90° - \frac{180°}{N_s}\right) \tag{30.39}$$

Engagement occurs at the shaft speed ω_e when $F = 0$. The proper initial tension S for the garter spring is found by setting $\omega = \omega_e$ and $F = 0$ in Eq. (30.39). The engagement speed is selected by the designer (for example, about 70 percent of running speed). The required value for S is

$$S = \frac{mr\omega_e^2}{2g_c \cos(90° - 180°/N_s)} \tag{30.40}$$

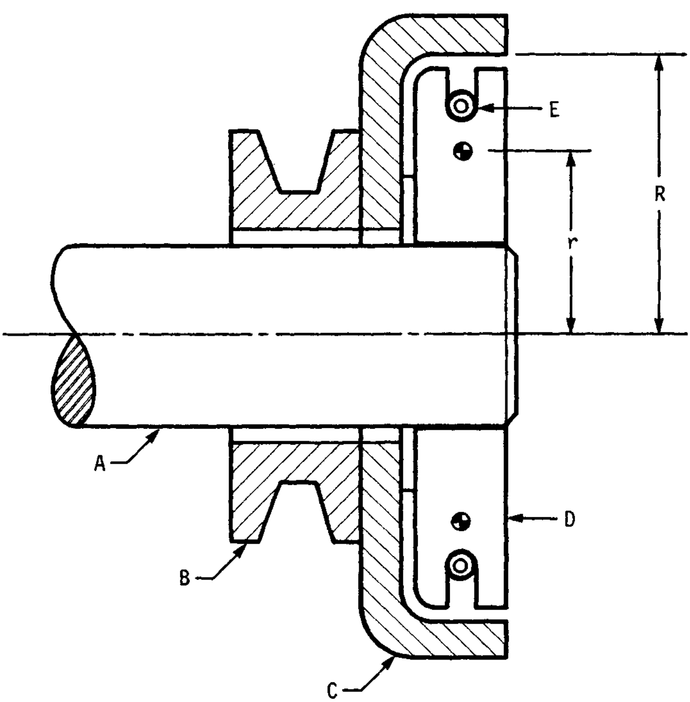


FIGURE 30.18 Free-shoe centrifugal clutch with garter spring to regulate engagement speed. *A*, input shaft; *B*, output pulley; *C*, drum; *D*, weights; *E*, garter spring.

The torque capacity at any shaft speed ω is

$$T = fF_n RN_s \tag{30.41}$$

Example 7. A centrifugal clutch is being considered for an application where the running speed is 3000 rev/min and engagement is to begin at 1000 rev/min. It is planned to use four shoes, each with a mass of 140 grams (g). A coefficient of friction $f = 0.3$ can be achieved. The inner diameter of the drum is 75 mm, and the radius R to the center of gravity of each shoe is 25 mm. (1) What should be the initial tension of the garter spring? (2) What is the normal force on each shoe at running speed? (3) What is the torque capacity at running speed? (4) What is the power capacity at running speed if a service factor of 2.25 is required?

Solution.

1. The initial tension of the garter spring is found from Eq. (30.40):

$$S = \frac{0.140(0.025)[1000(2\pi/60)]^2}{2(1)\cos(90° - 180°/4)} = 27.1 \text{ N}$$

2. The normal force on each shoe at 3000 rev/min is, by Eq. (30.39),

$$F_n = \frac{(0.140)(0.025)[(2\pi/60)(3000)]^2}{1} - 2(27.1)\cos(90° - 180°/4)$$

$$= 307.1 \text{ N}$$

3. The torque capacity at running speed is

$$T = fF_n RN_s = (0.3)(307.1)\left(\frac{0.075}{2}\right)(4) = 13.8 \text{ N} \cdot \text{m}$$

4. The power capacity uncorrected for service factor is

$$P = \frac{Tn}{9550} = \frac{13.8(3000)}{9550} = 4.34 \text{ kW}$$

Correcting for service factor, we see the power rating is

$$P_{\text{rating}} = \frac{P}{K_S} = \frac{4.34}{2.25} = 1.93 \text{ kW}$$

30.6 BAND AND CONE BRAKES AND CLUTCHES

30.6.1 Band Brakes

A typical design for a band brake is shown in Fig. 30.19. A flexible metal band lined with a friction material contacts the drum over an angle of wrap θ and exerts a braking torque T on the drum. This particular design is self-energizing, since the moment exerted on the lever by force P_1 assists in actuating the brake.

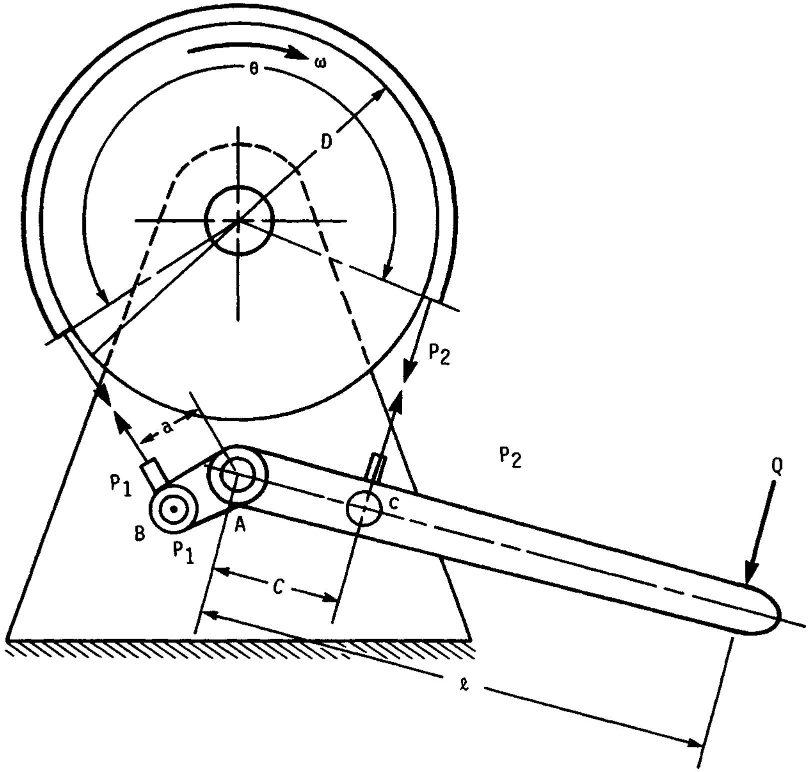


FIGURE 30.19 Forces on a band brake; b = bandwidth.

Four basic relationships are needed for analysis or design. For a band wrapped around a drum, the ratio of tensions is

$$\frac{P_1}{P_2} = e^{f\theta} \tag{30.42}$$

where the notation is indicated in Fig. 30.19. The net torque exerted on the drum by the band is

$$T = (P_1 - P_2\left(\frac{D}{2}\right) \tag{30.43}$$

The maximum contact pressure between the band and the drum occurs at the more taut P_1 end of the band:

$$p_{\max} = \frac{2P_1}{bD} \tag{30.44}$$

Finally, it is necessary to sum moments on the lever about pivot A to get the relationship involving the actuating force Q:

$$Q\ell - P_2c + P_1a = 0 \tag{30.45}$$

After substituting for P_1 and P_2 in terms of p_{max}, f, and θ, we get the following equation for actuating force:

$$Q = \frac{bDp_{max}}{2\ell}\left(\frac{c}{e^{f\theta}} - a\right) \tag{30.46}$$

For p_{max} you should use the value suggested by the lining manufacturer.

This brake could be self-locking if the designer were to get careless. The actuating force Q should always be positive. If it were zero or negative, the slightest touch on the lever would cause the brake to lock abruptly. The expression in the parentheses in Eq. (30.46) must always be positive. Thus, as a rule, design the brake so that

$$a < \frac{c}{e^{f\theta}} \tag{30.47}$$

In checking for self-locking, use a value for f that is 25 to 50 percent greater than the maximum value cited by the lining manufacturer.

Example 8. A band brake like that in Fig. 30.19 is needed to exert a braking torque of 3100 lb · in (350 N · m) on a drum with 10-in (0.254-m) diameter. The actuating force Q (exerted by the operator's foot) should not have to exceed 25 lb (111 N). Limit the maximum contact pressure to 60 pounds per square inch (psi) [0.414 megapascal (MPa)]. The value for f is 0.31 ± 0.03. (1) Make sure the brake will not be self-locking for an f value 30 percent above the maximum value. (2) Calculate the bandwidth b to limit the contact pressure. (3) Find the length ℓ for the operating lever. (4) For the same actuating force, what is the braking torque if the drum's rotation is reversed?

Solution. A scale layout indicates that $a = 1$ in (0.0254 m) and $\theta = 200°$ are feasible values when the lever's pivot point A is placed directly below the center of the drum. Then $c = 5$ in (0.127 m), corresponding to the drum's radius.

1. First make sure that the brake will not be self-locking. Use $f = 1.3(0.31 + 0.03) = 0.442$ and $\theta = 200°/57.296 = 3.491$ rad. Then, from Eq. (30.47),

$$a < \frac{5}{0.442(3.491)} \qquad \text{or} \qquad a < 1.069 \text{ in}$$

So the dimension $a = 1$ in will do nicely.

2. Now select a bandwidth b so that p_{max} does not exceed 60 psi (0.414 MPa). By Eqs. (30.42) and (30.43),

$$\frac{P_1}{P_2} = e^{0.28(3.491)} = 2.658$$

$$3100 = (P_1 - P_2)\left(\frac{10}{2}\right)$$

Solving these for P_1 and P_2 gives $P_1 = 994$ lb (4420 N) and $P_2 = 374$ lb (1664 N). So, by Eq. (30.44), the bandwidth is

$$b = \frac{2P_1}{p_{max}D} = \frac{2(994)}{60(10)} = 3.31 \text{ in (say 3.5 in)}$$

3. Next the moment arm length ℓ for the actuating force Q must be found from Eq. (30.45):

$$\ell = \frac{P_2 c - P_1 a}{Q} = \frac{374(5) - 994(1)}{25} = 35 \text{ in}$$

4. Finally, the torque capacity of the brake for the same actuating force, but with the drum rotating counterclockwise, can be evaluated. Since the drum turns counterclockwise, the braking torque must be clockwise. Thus, forces P_1 and P_2 are interchanged. The larger force P_1 is applied at point *C*, and the force P_2 is applied at point *B*. Equation (30.45), suitably rewritten, is

$$P_1 c - P_2 a - Q\ell = 0$$

But Eqs. (30.42) and (30.43) do not need to be changed:

$$P_1(5) - P_2(1) - 25(35) = 0$$

or

$$5P_1 - P_2 = 875$$

$$\frac{P_1}{P_2} = e^{f\theta} = 2.658$$

Then $P_1 = 189$ lb and $P_2 = 70$ lb. And the net braking torque is

$$T = (P_1 - P_2)\left(\frac{D}{2}\right) = (189 - 70)\left(\frac{10}{2}\right) = 595 \text{ lb} \cdot \text{in}$$

This is considerably less than the 3100-lb · in capacity for clockwise rotation of the drum.

30.6.2 Cone Brakes and Clutches

Two mating cones kept in contact by an axial force can be used as a clutch, as in Fig. 30.20, or as a brake, as in Fig. 30.21. A small cone angle α produces a wedging action, and a large torque capacity is achieved for a small actuating force. But if the cone angle is too small, it becomes difficult to disengage the cones. A cone angle of 10° to 15° is a reasonable compromise.

Basic Relationships. A cone is shown schematically in Fig. 30.22 with three elementary quantities indicated: area dA, normal contact force dP, and actuating force dF. From these it follows that the elementary torque dT is

$$dT = fr\, dP = frp \frac{2\pi r\, dr}{\sin \alpha} = \frac{2\pi f p r^2\, dr}{\sin \alpha} \tag{30.48}$$

Similarly, the elementary actuating force dF is

$$dF = dP \sin \alpha = 2\pi pr\, dr \tag{30.49}$$

The actuating force F and the torque capacity T are found by integrating in Eqs. (30.49) and (30.48) from the inside radius $d/2$ to the outside radius $D/2$:

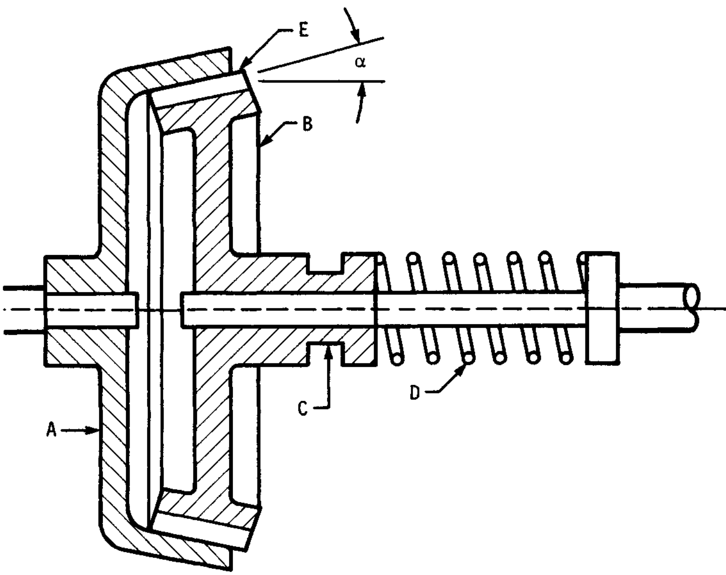


FIGURE 30.20 Cone clutch; *A,* cup; *B,* cone; *C,* shifting groove; *D,* spring; *E,* friction lining; α = cone angle.

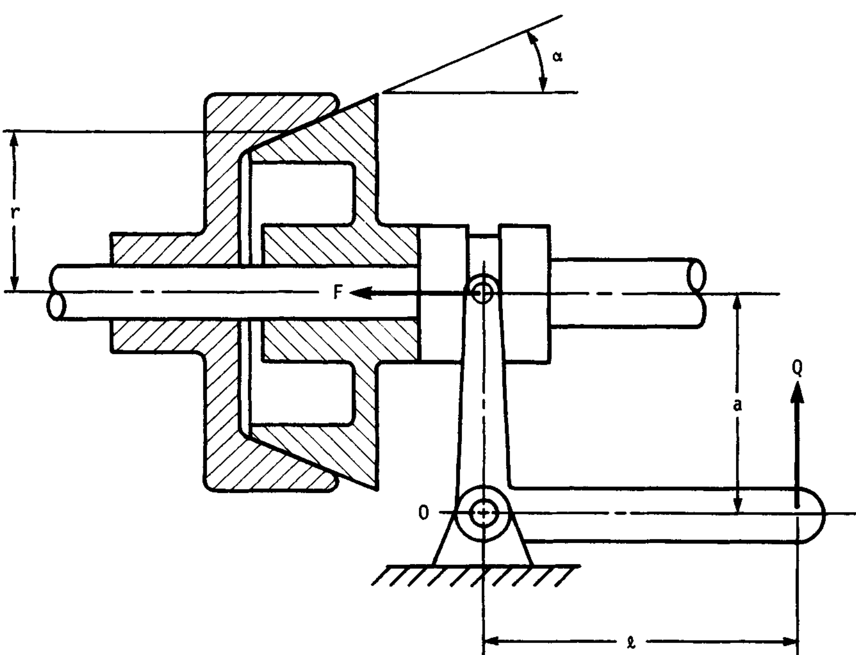


FIGURE 30.21 Cone brake; α = cone angle.

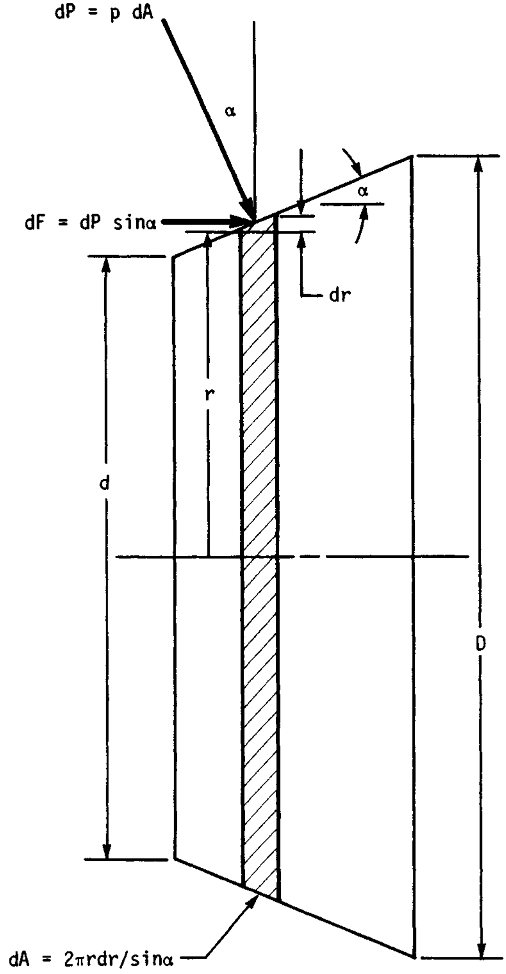


FIGURE 30.22 Elementary quantities on conical surface.

$$F = 2\pi \int_{d/2}^{D/2} pr\,dr \tag{30.50}$$

$$T = \frac{2\pi}{\sin\alpha} \int_{d/2}^{D/2} fpr^2\,dr \tag{30.51}$$

Before these integrations can be carried out, assumptions have to be made about the way in which contact pressure p and friction coefficient f vary across the active face of the cone. In what follows, the variations of f with pressure and rubbing velocity have been neglected. Only the variation in contact pressure p is used.

Contact-Pressure Distribution. When the friction surfaces are new, the pressure is fairly uniform across the clutch face. But after an initial wear-in period, the pressure accommodates itself to a uniform rate of wear. We assume that the wear rate is proportional to the frictional work per unit area, that is, to fpV. If the variations in f are neglected, then the wear rate is proportional simply to pV, the product of contact pressure and rubbing velocity. We can write

$$pV = 2\pi r\omega p = \text{constant} \tag{30.52}$$

Thus across the face the product pr is constant, implying that p_{max} occurs at the inner radius $d/2$. In general,

$$p = \frac{p_{max} d}{2r} \tag{30.53}$$

A typical pressure distribution is shown in Fig. 30.23.

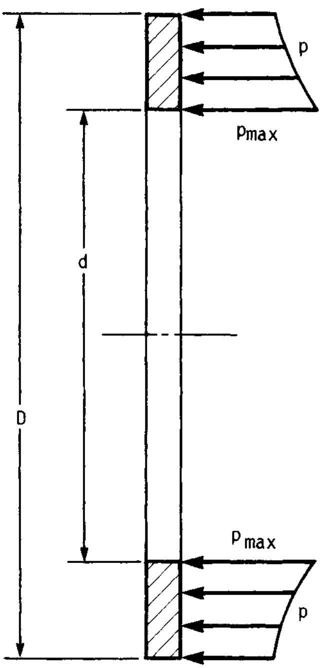


FIGURE 30.23 Contact-pressure distribution on face of cone after wear-in period (for constant f).

Torque Capacity. By substituting for p from Eq. (30.53) in Eq. (30.51) and carrying out the integration, we find the torque capacity T:

$$T = \frac{\pi f p_{max} d}{8 \sin \alpha}(D^2 - d^2) \tag{30.54}$$

Actuating Force. Equation (30.50) can be integrated to yield

$$F = \frac{\pi p_{max} d}{2}(D - d) \tag{30.55}$$

The last two equations can be combined to produce the useful result

$$T = \frac{Ff}{4 \sin \alpha}(D + d) \tag{30.56}$$

The last equation indicates that for the uniform-wear assumption, the mean friction radius is simply the average radius.

Moment Equation for Brake Lever. The axial actuating force for the cone brake shown in Fig. 30.21 can be found by summing moments on the lever about the pivot point O and solving for F. Thus

$$F = \frac{Q\ell}{a} \tag{30.57}$$

30.7 DISK CLUTCHES AND BRAKES

30.7.1 Multidisk Clutches and Brakes

The multidisk clutch in Fig. 30.24 is intended for wet operation using an oil coolant. Similar clutches are built for dry operation. Disk brakes are similar in construction to the disk clutch. In either case, an axial force is applied to the flat surfaces of the elements to produce tangential frictional forces. Typically not more than the outer 40 percent of the radius is used. The ratio of inside to outside diameters may be as high as 0.80/1.

Contact-Pressure Distribution. The reasoning used to establish pressure distribution on the annular clutch or brake plate is the same as that used for cone clutches and brakes. After an initial wear-in period, the pressure distribution accommodates itself to a constant rate of wear across the active portion of the disk (Fig. 30.25). Equations (30.52) and (30.53) apply here also.

Axial Actuating Force. For a given set of dimensions and a permissible contact pressure, the corresponding actuating force is given by Eq. (30.55), which was developed for cone clutches and brakes.

Torque Capacity. For this pressure distribution, the torque capacity is

$$T = \frac{1}{8}[\pi f p_{\max} d(D^2 - d^2)N_p] \tag{30.58}$$

Although the torque equation could be derived independently, it can also be derived directly from Eq. (30.54) by setting the cone angle $\alpha = 90°$ and inserting N_p, the num-

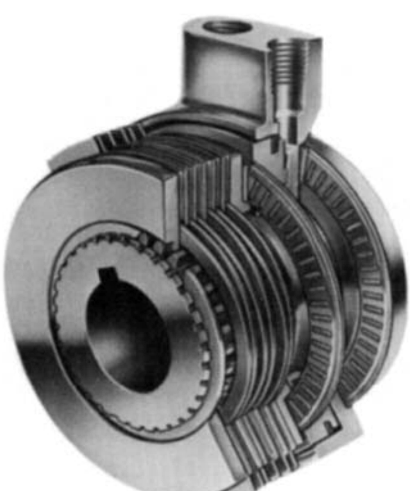


FIGURE 30.24 An oil-actuated multiple-disk clutch for enclosed operation in an oil spray or bath. *(Twin Disc, Inc.)*

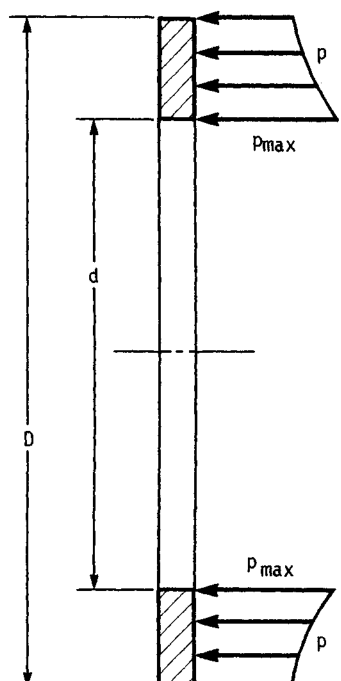


FIGURE 30.25 A friction member of a multiple-disk clutch or brake with the pressure distribution for uniform wear.

ber of pairs of contacting disk faces. Torque capacity can also be expressed in terms of the actuating force F:

$$T = \frac{FfN_p}{4}(D + d) \qquad (30.59)$$

30.7.2 Caliper Disk Brakes

The automotive caliper disk brake shown in Fig. 30.9 is hydraulically operated. Two pads are pressed against opposite sides of the brake disk to provide a braking torque. The principle of operation is shown schematically in Fig. 30.26*a*.

Usually each pad is nearly the annular shape, illustrated in Fig. 30.26*b*, but occasionally a circular pad ("puck" or "button") is used (Fig. 30.26*c*).

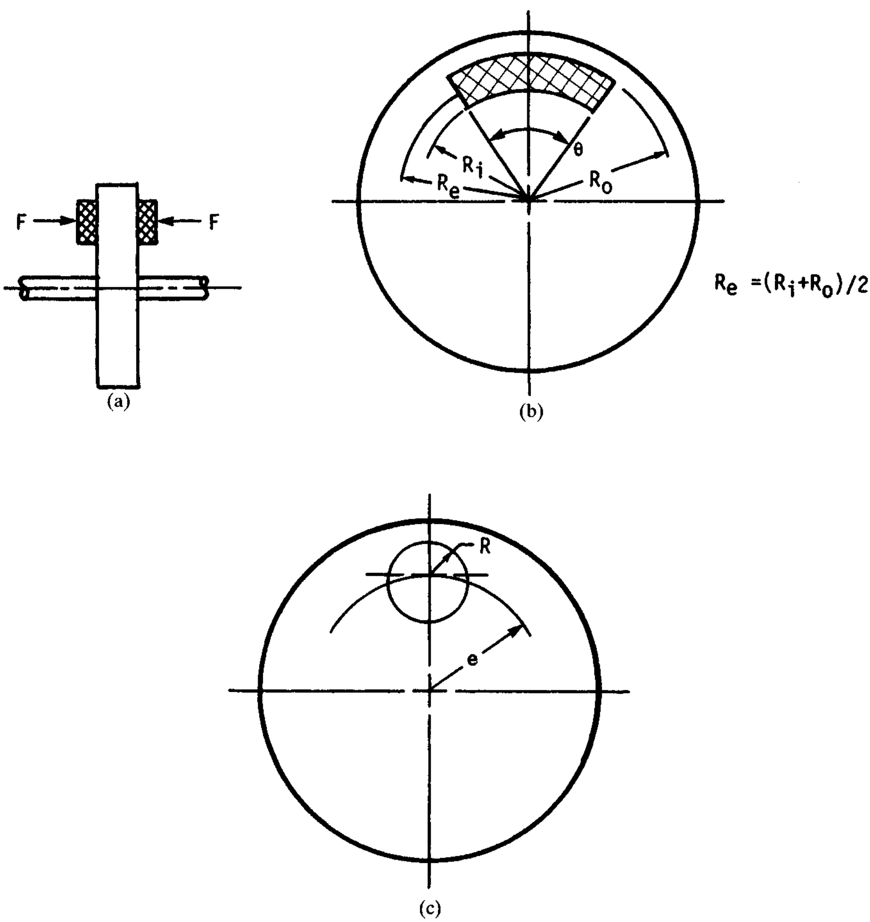


FIGURE 30.26 Caliper disk brake. (*a*) Principle of operation; (*b*) annular pad; (*c*) circular pad.

Torque Capacity. The torque capacity *per pad* is

$$T = fFR_e \tag{30.60}$$

If two pads are used, then the torque capacity is double the value calculated by Eq. (30.60).

This torque equation is quite simple. A friction force fF acting at an effective friction radius R_e produces a braking torque T. The practical issues are (1) the value to use for the effective friction radius R_e and (2) the maximum contact pressure p_{max} developed.

Actuating Force for Annular Pad. Brake designers often assume that the contact pressure does not vary very much over the annular pad. But you may prefer to use the uniform-wear approach as more realistic. Both approaches are given here.

The following two equations are used to calculate the actuating force for the constant-contact-pressure and the uniform-wear approaches, respectively:

$$F = p_{av}\theta\frac{R_o^2 - R_i^2}{2} \tag{30.61}$$

$$F = p_{max}\theta R_i(R_o - R_i) \tag{30.62}$$

The notation is shown in Fig. 30.26*a* and *b*.

The relation between average and maximum contact pressures for the uniform-wear approach is

$$\frac{p_{av}}{p_{max}} = \frac{2R_i/R_o}{1 + R_i/R_o} \tag{30.63}$$

In the limit, the average and maximum pressures become equal as the inner radius approaches the outer one. For $R_i/R_o = 0.60$, $p_{av} = 0.75p_{max}$; but for $R_i/R_o = 0.80$, $p_{av} = 0.89p_{max}$.

The uniform-wear assumption is the more conservative approach. For a given actuating force, it implies a smaller torque capacity.

Effective Friction Radius for Annular Pad. If the contact pressure is assumed to be constant over the pad, the effective friction radius is

$$R_e = \frac{2}{3}\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2} \tag{30.64}$$

When the uniform-wear assumption is made, the effective friction radius is simply

$$R_e = \frac{R_i + R_o}{2} \tag{30.65}$$

Circular Pads. Fazekas [30.3] has derived the basic equations for circular pads. The effective friction radius is

$$R_e = \delta e \tag{30.66}$$

where e = radius from the center of the disk to the center of the pad and δ = multiplier found in Table 30.9. Also tabulated in Table 30.9 is the ratio of maximum to average contact pressure.

TABLE 30.9 Design Factors for Caliper Disk Brakes with Circular Pads

R/e	$\delta = R_e/e$	p_{max}/p_{av}
0	1.000	1.000
0.1	0.983	1.093
0.2	0.969	1.212
0.3	0.957	1.367
0.4	0.947	1.578
0.5	0.938	1.875

SOURCE: Ref. [30.3].

The actuating force on each pad is

$$F = \pi R^2 p_{av} \tag{30.67}$$

The torque capacity per pad is found by using Eq. (30.60) after F and R_e have been calculated.

Example 9. A sports car requires disk brakes for the front wheels. It has been decided to use two annular pads per wheel with $R_i = 3.875$ in, $R_o = 5.5$ in, and $\theta = 108°$. The friction material supplier guarantees a coefficient of friction of at least $f = 0.37$. Each pad is actuated by two hydraulic cylinders, each 1.5 in in diameter. Each front-wheel brake provides a braking torque capacity of 13×10^3 lb · in. What hydraulic pressure is needed at the wheel cylinders? What are the average and the maximum contact pressures? Assume uniform wear.

Solution

1. The torque capacity per pad has to be $(13 \times 10^3)/2 = 6500$ lb · in. Given uniform wear, the effective friction radius is, by Eq. (30.65),

$$R_e = \frac{R_i + R_o}{2} = \frac{3.875 + 5.5}{2} = 4.69 \text{ in}$$

The corresponding actuating force is, by Eq. (30.60),

$$T = \frac{T}{fR_e} = \frac{6500}{0.37(4.69)} = 3750 \text{ lb per pad}$$

or

$$F = 1875 \text{ lb per wheel cylinder}$$

The hydraulic pressure at the wheel cylinder has to be

$$p_h = \frac{F}{A_p} = \frac{4F}{\pi d_p^2} = \frac{4(1875)}{\pi(1.5)^2} = 1060 \text{ psi}$$

2. Equation (30.62) can be used to find the maximum contact pressure:

$$p_{max} = \frac{F}{\theta R_i(R_o - R_i)} = \frac{3750}{(108°/57.296)(3.875)(5.5 - 3.875)} = 316 \text{ psi}$$

For finding the average contact pressure, use Eq. (30.63):

$$p_{av} = p_{max} \frac{2R_i/R_o}{1 + R_i/R_o} = 316 \frac{2(3.875)/5.5}{1 + 3.875/5.5} = 261 \text{ psi}$$

30.8 ELECTROMAGNETIC TYPES

Electromagnetic forces are used in a variety of ways to couple two sides of a clutch or a brake. The use of electrically generated forces implies relatively easy means of automatic control.

30.8.1 Magnetically Actuated Friction Clutches and Brakes

The most common use of electromagnetic forces is to provide the actuating force for a friction brake or clutch. The configuration shown in Fig. 30.27 can be used either as a clutch or as a brake. When power is applied to the coil, the magnet, faced with a friction material, attracts the armature. Torque can be varied by using a potentiometer. If one member is fixed, the device functions as a brake; otherwise, it is a clutch.

The combination clutch-brake in Fig. 30.28 uses a single solenoid coil. When the coil is deenergized, the clutch is disengaged and springs prevent the release of the brake. When the coil is energized, the clutch is engaged and the brake is released. An arrangement of opposing mechanical springs ensures that the input-side clutch is fully engaged for a brief time before the brake is released. This is done by making the springs assisting clutch engagement weaker than those resisting brake release.

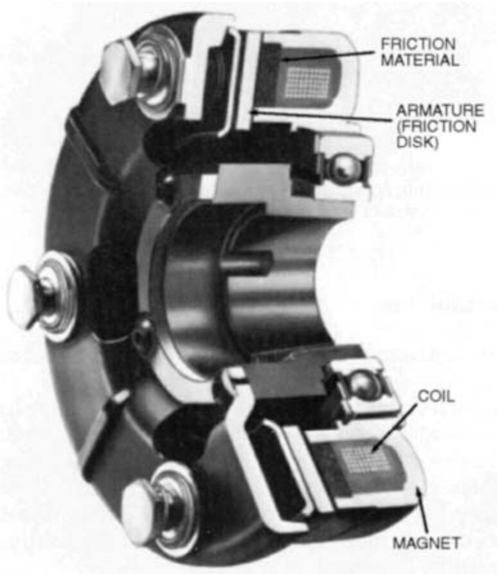


FIGURE 30.27 Electromagnetic friction clutches and brakes. *(Warner Electric Brake and Clutch Co.)*

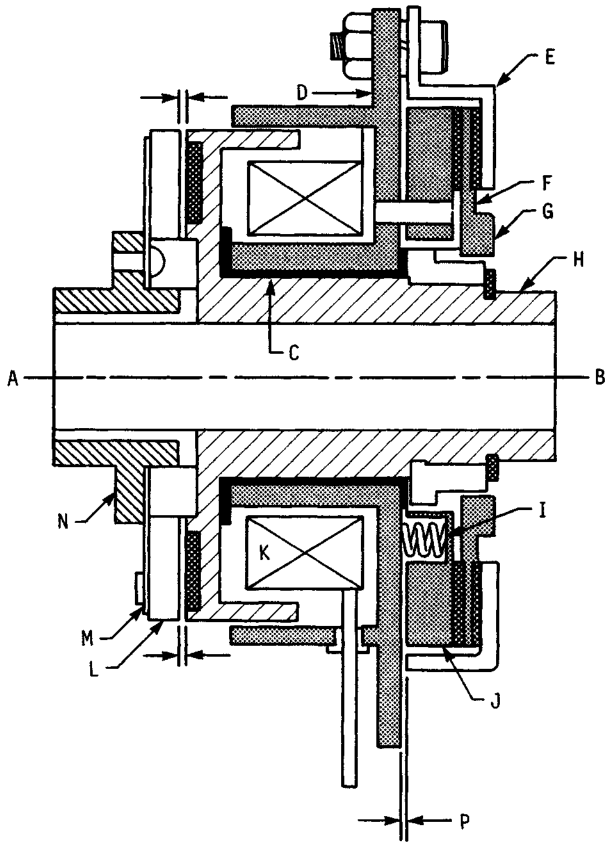


FIGURE 30.28 Clutch-brake transmission. *A,* input; *B,* output; *C,* field coil bearing; *D,* field coil assembly; *E,* pressure cup; *F,* brake plate; *G,* hub spling; *H,* rotor assembly; *I,* brake spring; *J,* brake armature; *K,* field coil; *L,* clutch armature; *M,* clutch spring; *N,* drive plate; *P,* air gap. *(Electroid Corporation.)*

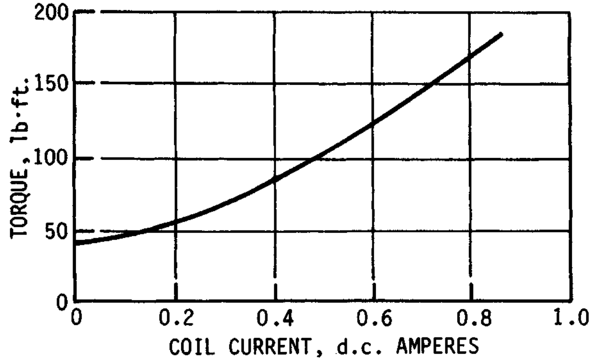
30.8.2 Magnetic Clutches

The operating characteristics of three types of magnetic clutches are shown in Fig. 30.29.

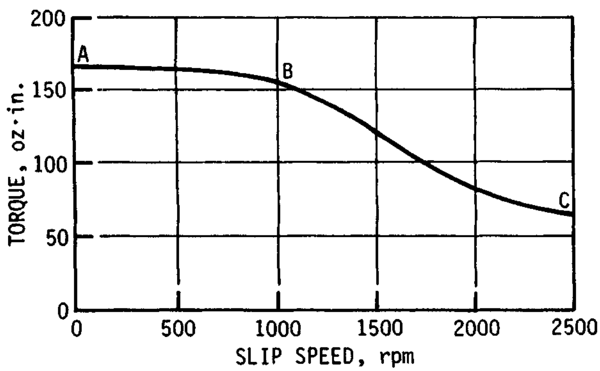
Magnetic-particle clutches (Fig. 30.29*a*) use an iron powder mixed with a lubricant to partially fill the annular gap between members. When a direct-current (dc) coil induces a magnetic field, the iron particles form chains and provide the means to transmit torque. There is a nearly linear relation between coil current and torque.

Hysteresis clutches (Fig. 30.29*b*) directly couple the two members as long as the load does not exceed the torque rating. They can also slip continuously to maintain a constant-torque output independent of speed.

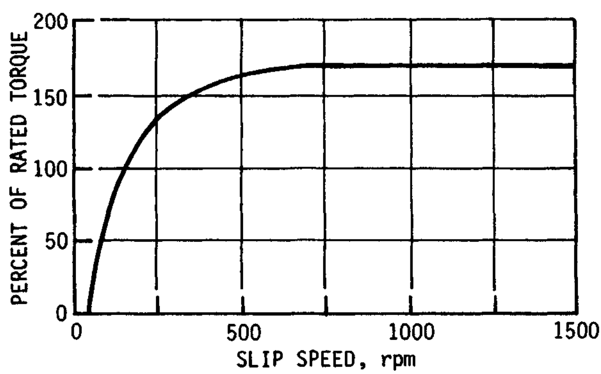
The *eddy-current clutch* (Fig. 30.29*c*) is rather like the hysteresis clutch in construction. Torque is developed if there is slip. The torque is associated with dissipation of eddy currents in the rotor ring's electric resistance.



(a)



(b)



(c)

FIGURE 30.29 Torque characteristics of magnetic clutches. (*a*) Magnetic-particle clutch has a characteristic that is independent of slip and increases almost linearly with coil current. (*b*) Hysteresis clutch exhibits almost a constant torque out to the thermal, which begins at *B;* temperature then limits the torque capacity from *B* to *C.* (*c*) Eddy-current clutch exhibits a constant-torque characteristic at rated slip speed. *(From Ref. [30.5].)*

30.8.3 Dynamic Braking

When it is necessary to bring a motor-driven load from operating speed to rest in less than the normal coasting time, braking is necessary. However, for motors, the braking can be done by purely electrical means through dynamic braking. Electric braking, or *dynamic braking,* is done by altering the connections to the motor. It may be done with or without the aid of an external power source. Dynamic braking is available for fractional-horsepower motors. The designer should keep this option in mind.

30.9 ACTUATION PROBLEMS

30.9.1 General

Table 30.10 lists the characteristics of four basic brake actuation methods. Note that many brakes are made to operate on the fail-safe principle. This means that the brake is applied by using strong springs and that the method of actuation releases or holds off the brake. Thus, a reduction, say, in hydraulic pressure would cause the brake to be applied.

30.9.2 Brake Actuation Systems for Vehicles

The hydraulic system shown schematically in Fig. 30.30 is suitable for a passenger vehicle using disk brakes on the front axles and drum brakes on the rear. The pedal force, multiplied by the leverage ratio, is applied to the *master cylinder* to produce a

TABLE 30.10 Characteristics of Various Methods of Actuating Band, Drum, and Disk Brakes

Actuation method	Advantages	Disadvantages	Possible difficulties
Mechanical	Robust; simple; manual operation gives good control	Large leverage needed	Friction losses at pins and pivots
Pneumatic	Large forces available	Compressed air supply needed; brake chambers may be bulky; slow response time	Length of stroke (particularly if diaphragm type)
Hydraulic†	Compact; large forces available; quick response and good control	Special fluid needed; temperatures must not be high enough to vaporize fluid	Seals
Electric‡	Suitable for automatic control; quick response	On/off operation	Air gap

†Used for spot-type disk brakes.
‡Used for spot- and plate-type disk brakes.
SOURCE: Ref. [30.6].

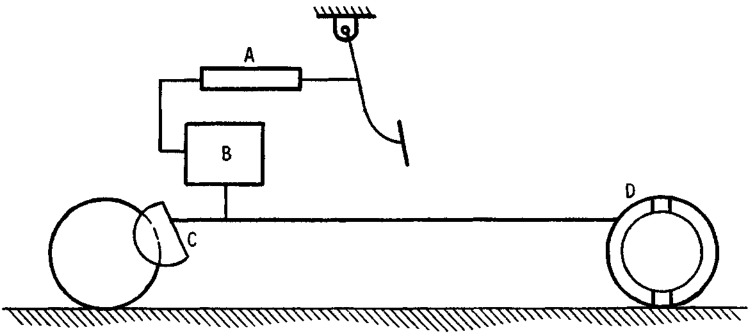


FIGURE 30.30 Hydraulic system with indirect servo. *A,* master cylinder; *B,* slave cylinder and servo; *C,* disk brake; *D,* leading-trailing shoe brake.

hydraulic pressure. This hydraulic pressure can itself be multiplied by a *power-brake assist unit* which uses either the intake manifold's vacuum or a positive hydraulic pressure from the power-steering pump to create a pressure differential.

In a *split system,* the master cylinder has two pistons and two reservoirs, one for the front-wheel brakes, the other for the rear. When the primary piston is pushed forward, the secondary piston is pushed forward as well by the primary piston spring and the buildup of pressure between the two pistons. Thus hydraulic pressure is built up in both systems. The springs return the pistons when the brake-pedal force is removed.

With the split system, one set of brakes can function even if the hydraulic system for the other set of brakes is damaged. For example, if the hydraulic system of the front brakes fails, no hydraulic pressure is built up in the front-brake system, and the secondary piston continues to move until its nose butts against the end of the cylinder. The primary piston continues to move and build up pressure between the primary and secondary pistons to actuate the rear brakes.

Most hydraulic brake systems are equipped with automatic valves:

1. *Pressure-differential valve* to turn on a warning light if either of the hydraulic systems (front or rear) fails.
2. *Proportioning valve* to improve the braking balance between the front and rear brakes and prevent skidding resulting from the rear brakes locking up before the front brakes.
3. *Metering valve* to delay the flow of brake fluid to the front-brake calipers until the system pressure has risen sufficiently. The motives are to overcome the tension of the retracting springs at the rear-brake shoes, expand the shoes, and supply the rear brakes before the front brakes. The metering valve prevents front-brake lockup during light braking on slippery or icy roads.

All three functions may be merged into a *combination valve* mounted near the master cylinder.

A *vacuum-brake booster* functions in three modes: released, applied, and holding. When the brake pedal is released, the engine intake manifold pulls air from the front shell through a check valve. There is a vacuum on both sides of the diaphragm, and the pressures are equal. The diaphragm is held to the rear by its spring. No force is exerted on the master cylinder by the push rod.

When the brake pedal is depressed, the valve rod pushes the valve plunger forward to close the vacuum port and open the atmospheric port. With atmospheric pressure on the rear side of the diaphragm and a vacuum at its front side, the diaphragm moves forward and pushes against the push rod. This is the *applied position.*

To provide gradual braking when needed (*holding position*), a position between the released and the applied positions is provided. The driver has control over the degree of braking.

30.9.3 Antiskid Brakes

In an antiskid braking system, the brakes are normally under manual control; but if wheel lockup is imminent, the antiskid system takes corrective action.

Whenever a rear wheel starts to lock, a wheel sensor detects an abrupt deceleration. A computer then causes the pressure in the rear braking system to decrease slightly, allowing the wheels to accelerate. When the wheel speed approaches its normal level for the vehicle's speed, the wheel cylinder pressure is restored. The antiskid system goes into action repeatedly to prevent wheel lockup and skidding until the vehicle speed drops to about 5 miles per hour (mph).

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