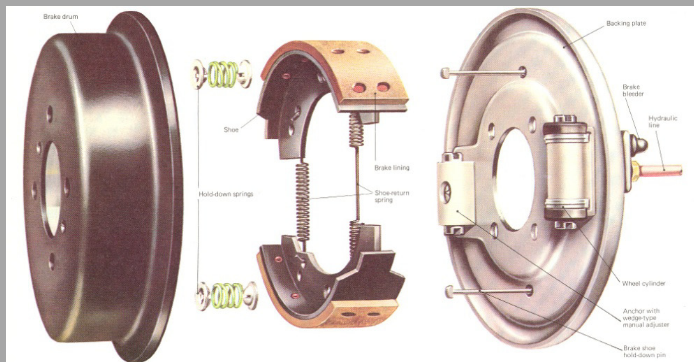
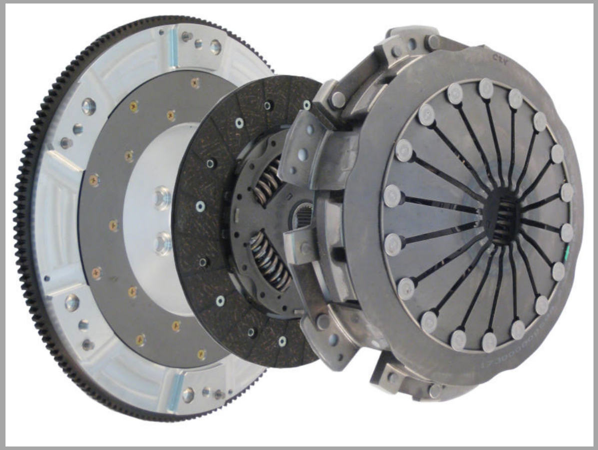
Clutches and Brakes

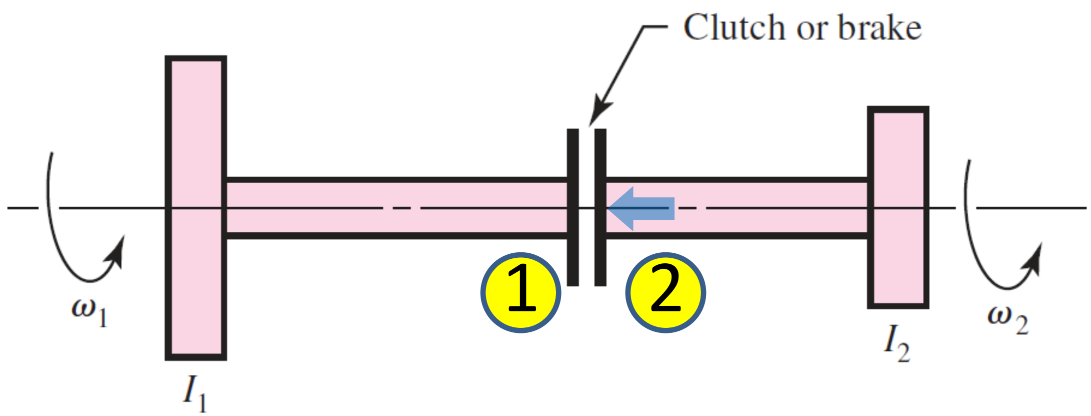




2103320 Des Mach Elem
Mech. Eng. Department
Chulalongkorn University

Introduction (1)

Clutch and brake are the machine elements used in transmission and rotation control. Here only clutch and brake using the concept of friction between surfaces are focused.



Engage disk 2 with disk 1,

- Both of disks are brought to the same speed by the effect of friction. (Clutch)
- Speed of disk 1 is decreased. (brake)

Clutch $\omega_1' = \omega_2 \neq 0$

Brake $\omega_1' < \omega_1 \quad \omega_2 = 0$

Introduction (2)

To design clutch or brake, the followings must be considered

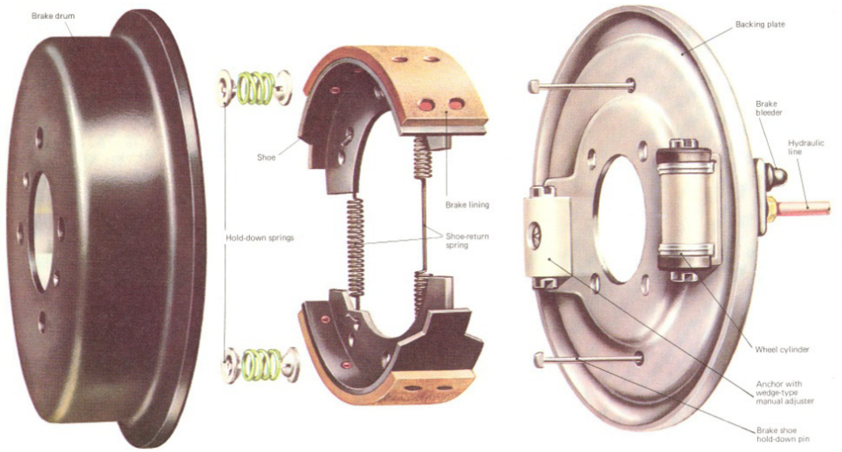
1. The actuating force: The force used to press clutch or brake to engage
2. The torque transmitted
3. The energy loss from the slippage between two surface
4. The temperature rise due to the energy loss

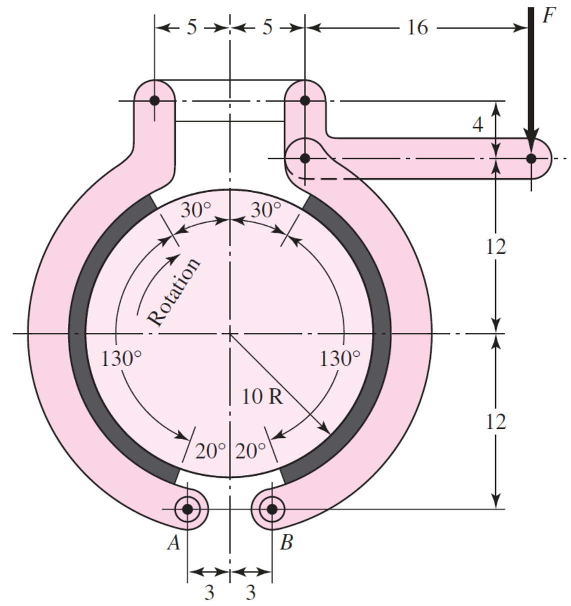
Types of clutch and brake

1. **Rim types with internal expanding shoes**
2. Rim types with external contracting shoes
3. Band types
4. **Disk or axial types**
5. Cone types
6. Miscellaneous types

Types of clutch and brake (1)

Rim types with internal expanding shoes



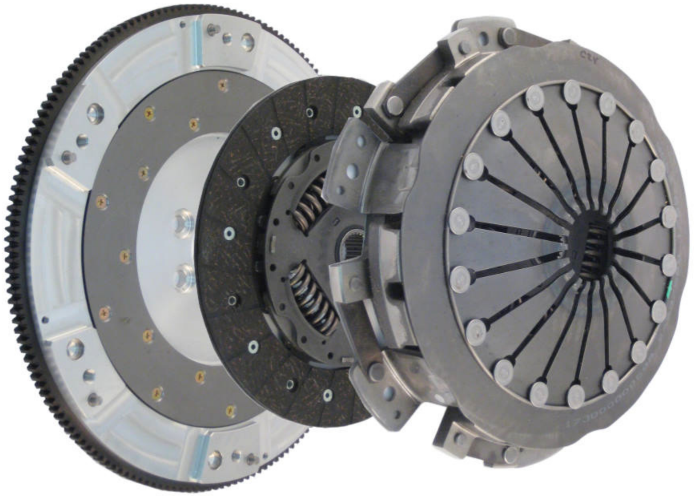


Rim types with external contracting shoes

Band types

Types of clutch and brake (2)

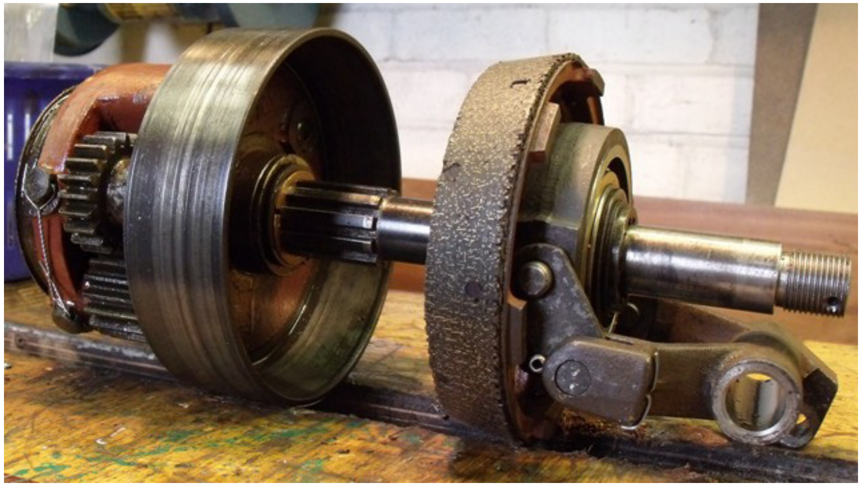
Disk or axial types



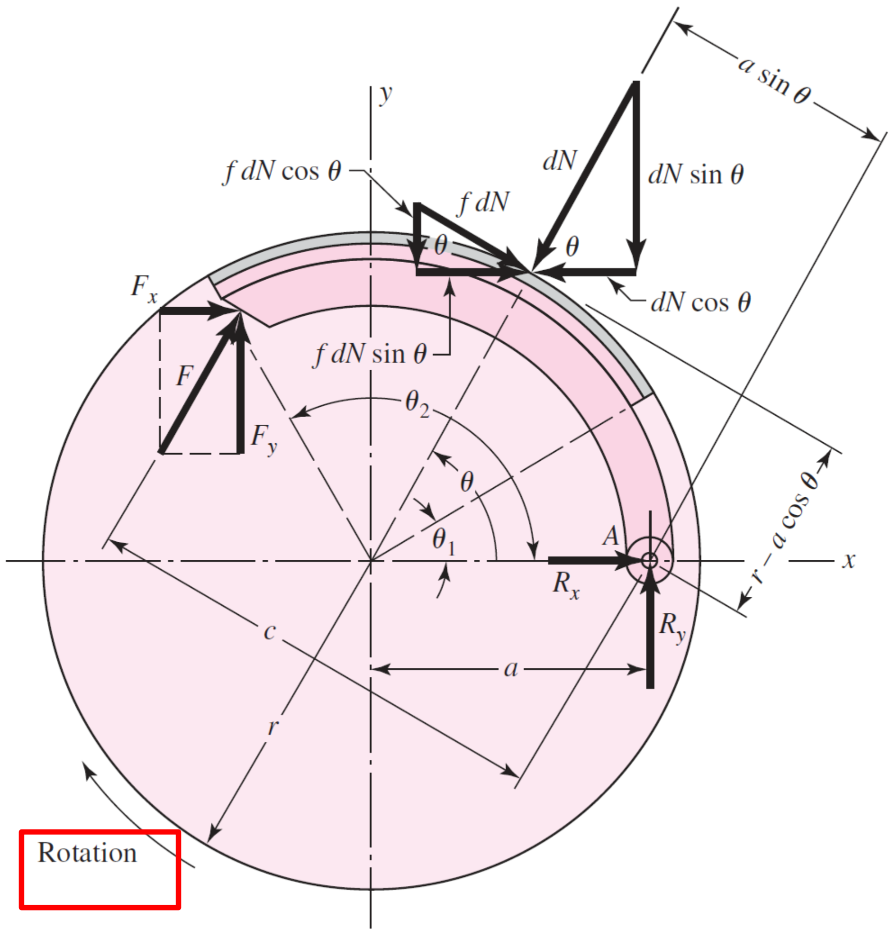
Clutch in automobile

Disk brake & brake pad

Cone types



Internal Expanding Rim type (1)



Pressure distribution along the internal shoe

$$p = \frac{p_a}{\sin\theta_a}\sin\theta$$

a : Maximum pressure

- long shoe, $\theta > 90°$, Max, pressure is occurring at $90°$
- short shoe, $\theta < 90°$, Max, pressure is occurring at the end of the shoe θ_2

p_a depend on material and relate closely to the friction coefficient

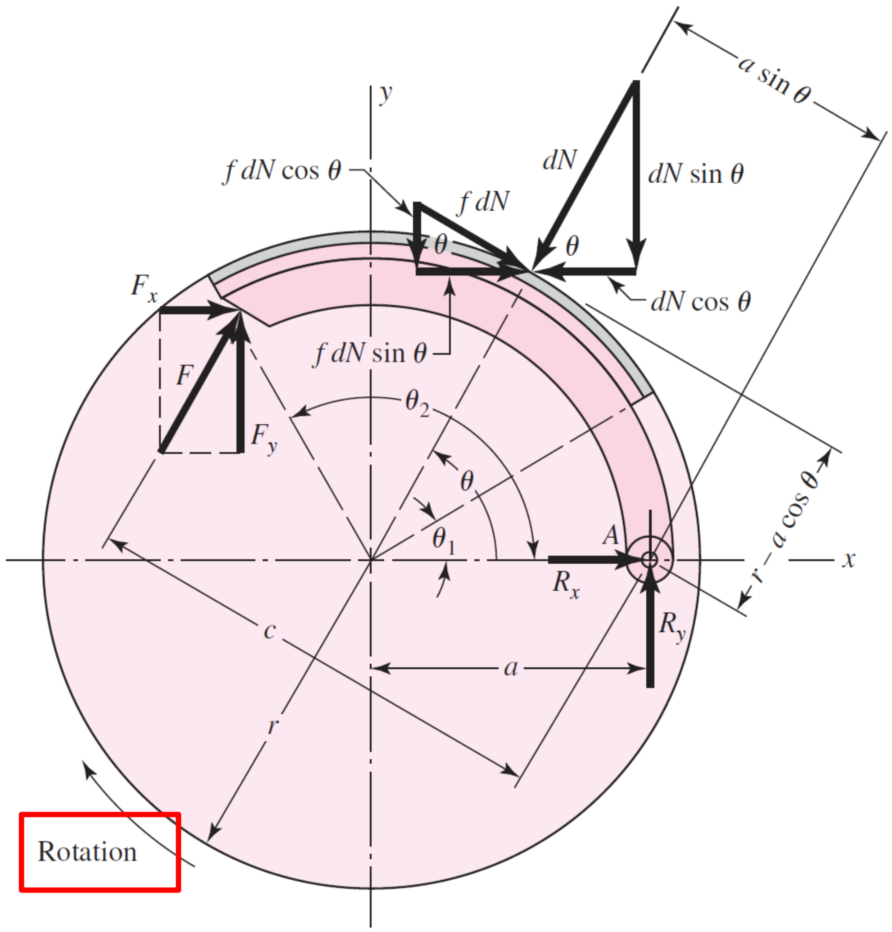
Normal force $dN = pbr\, d\theta = \dfrac{p_a br \sin\theta\, d\theta}{\sin\theta_a}$

b is the face width of the friction material

x-axis: center to the hinge-pin

y-axis: normal to x-axis and point to brake shoe

Internal Expanding Rim type (2)



Moment about point A (Hinge pin)

$$M_f = \int f\, dN\, (r - a\cos\theta)$$

$$= \frac{f p_a b r}{\sin\theta_a} \int_{\theta_1}^{\theta_2} \sin\theta \cdot (r - a\cos\theta)\, d\theta$$

$$M_N = \int dN\, (a \sin\theta)$$

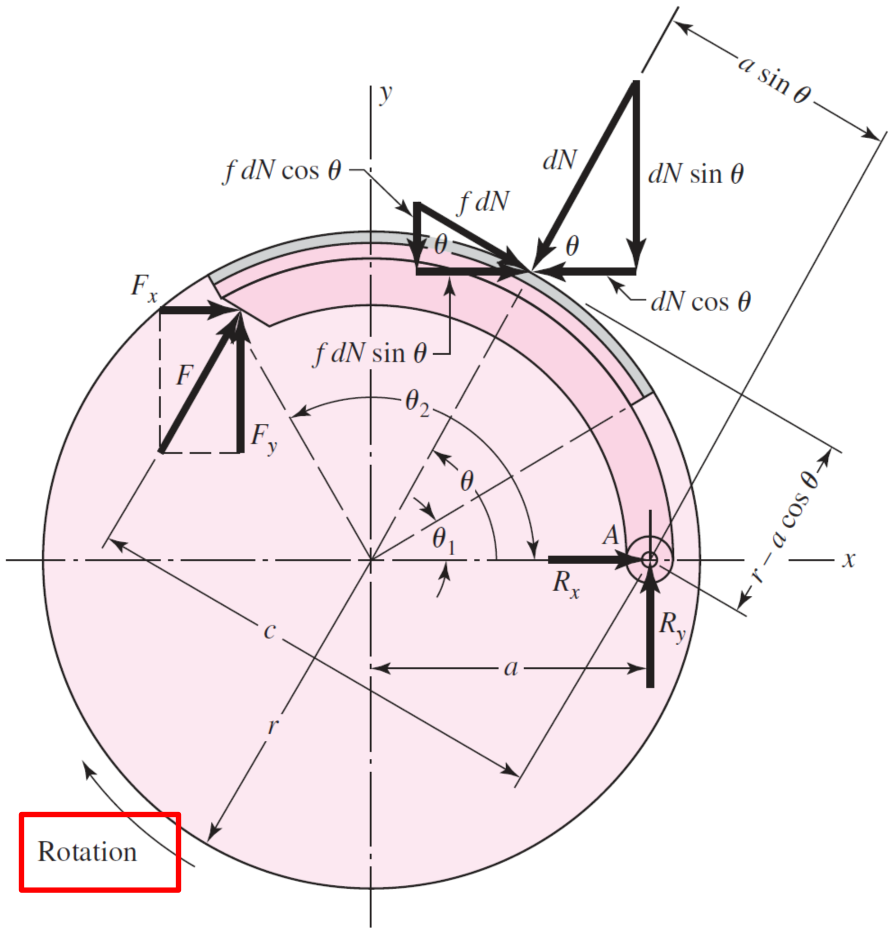
$$= \frac{p_a b r a}{\sin\theta_a} \int_{\theta_1}^{\theta_2} \sin^2\theta\, d\theta$$

The actuating force, F

$$\left[\sum M_A = 0\right] \Rightarrow F = \frac{M_N - M_f}{c}$$

If $M_N = M_f$, self-locking is obtained, and no actuating force is required to force a brake pad to engage with a drum. This effect is also called **the self-energizing effect**.

Internal Expanding Rim type (3)



The torque applied to the drum by the brake shoe = The sum of torque from the frictional force about the center of the drum

$$T=\int fr\;dN=\frac{fp_a br^2}{\sin\theta_a}\int_{\theta_1}^{\theta_2}\sin\theta\;d\theta$$

$$T=\frac{fp_a br^2(\cos\theta_1-\cos\theta_2)}{\sin\theta_a}$$

Hinge-pin reactions

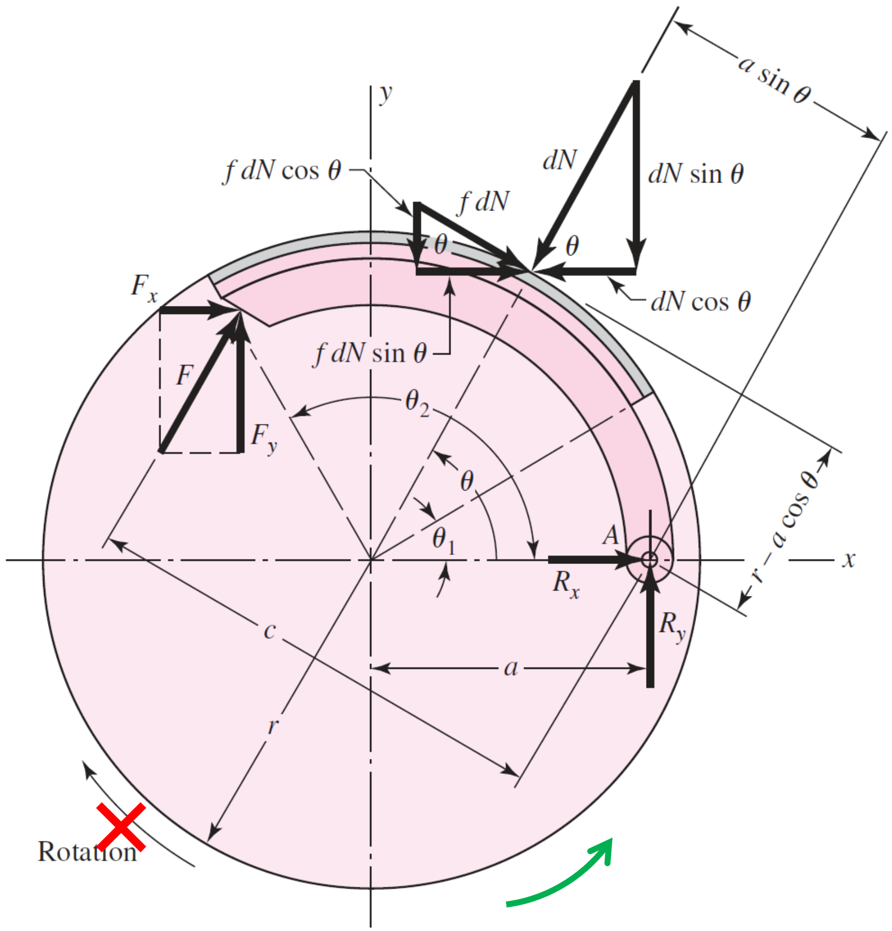
$$\left[\sum F_x=0\right]$$

$$\Rightarrow\quad R_x=\int dN\cos\theta-\int f\;dN\sin\theta-F_x$$

$$\left[\sum F_y=0\right]$$

$$\Rightarrow\quad R_y=\int dN\sin\theta+\int f\;dN\cos\theta-F_y$$

Internal Expanding Rim type (4)



If the rotation of the drum is reversed

The actuating force, F

$$\left[\sum M_A = 0\right] \Rightarrow F = \frac{M_N + M_f}{c}$$

Self-energizing effect is lost in this case. More actuating force is required to generate the same braking torque

The calculation of the hinge-pin reaction is the same. Only the direction of friction is different.

Example

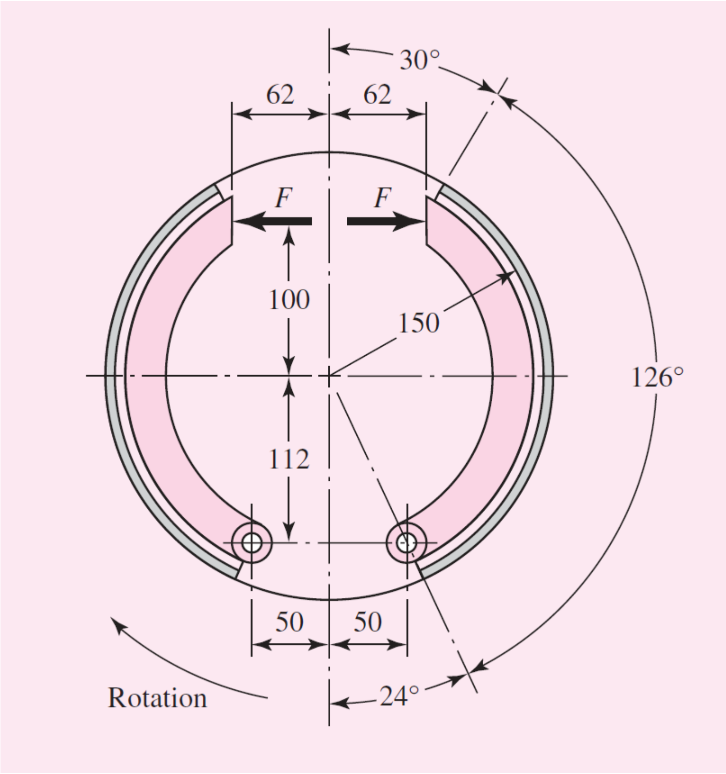
The brake shown in the figure is 300 mm in diameter and is actuated by a mechanism that exerts the same force F on each shoe. The shoes are identical and have a face width of 32 mm. The lining is a molded asbestos having a coefficient of friction of 0.32 and a pressure limitation of 1000 kPa. Estimate the maximum

(a) Actuating force F

(b) Braking capacity

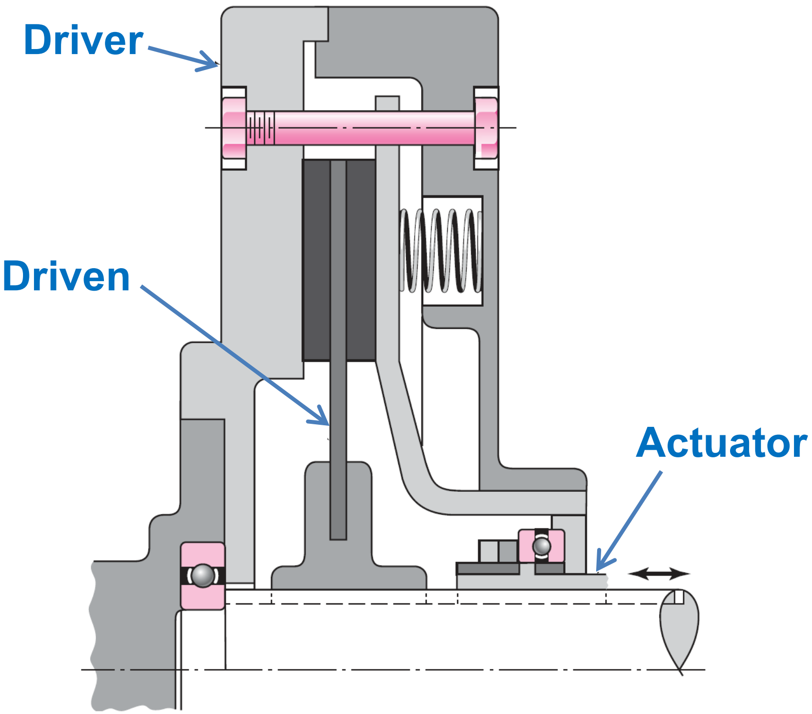
(c) Hinge-pin reactions

[Ex.16-2 Shigley's Mechanical Engineering Design 9th Edition. Richard G. Budynas and J. Keith Nisbett]



Frictional-Contact Axial Clutches

1. Do not have the centrifugal effect
2. Large frictional area and can be installed in a small place
3. Effective heat-dissipation surfaces
4. Slippage is possible, hence the impact force during engagement is low. Engagement at high speed is probable.



Wear of the clutch disk

1. For a new clutch, clutch disks are rigid
2. The greatest amount of wear will occur in the outer areas. (uniform pressure but the outer areas are more slippage)
3. After a certain amount of wear has taken place, the pressure distribution is not uniform, but the wear is uniform.

Assumption used in calculation

1. **Uniform wear**
2. **Uniform pressure**

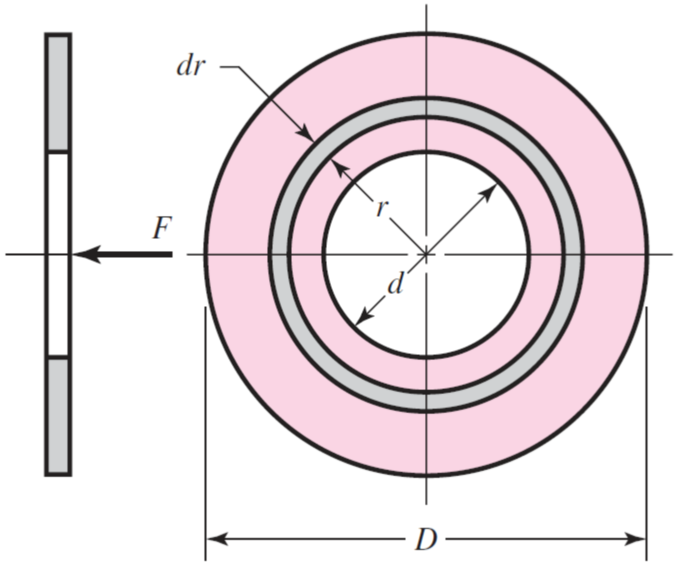
Uniform Wear

For uniform wear

$$pr\omega = (\text{constant})$$

$$pr = p_a r_i = p_a \frac{d}{2}$$

- p : Contact pressure
- p_a : Max. contact pressure
- r : radius
- ω : angular velocity



The actuating force, F

$$F = \int_{d/2}^{D/2} 2\pi pr\, dr = \pi p_a d \int_{d/2}^{D/2} dr = \frac{\pi p_a d}{2}(D - d)$$

Torque, T

$$T = \int_{d/2}^{D/2} 2\pi fpr^2\, dr = \pi f p_a d \int_{d/2}^{D/2} r dr = \frac{\pi f p_a d}{8}(D^2 - d^2)$$

$$T = \frac{Ff}{4}(D + d)$$

Uniform Pressure

For p = p_a = constant

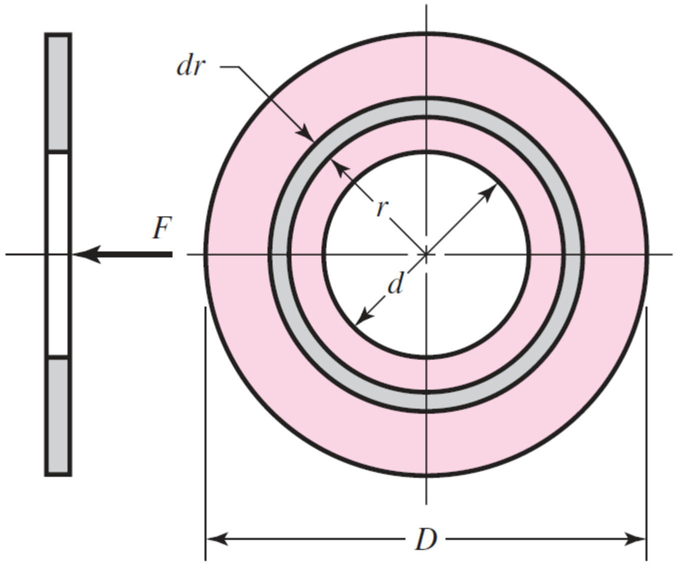
Total actuating force, F

$$F = \frac{\pi p_a}{4}(D^2 - d^2)$$

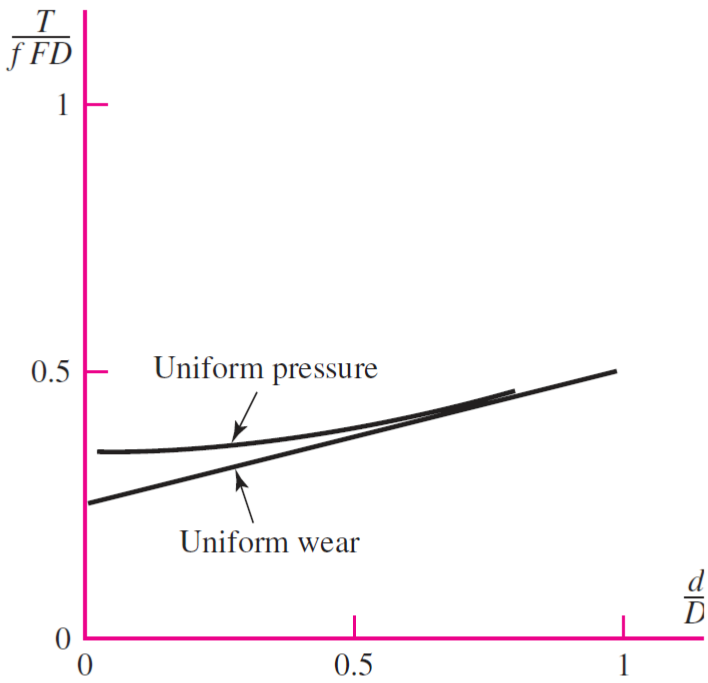
Torque, T

$$T = 2\pi f p_a \int_{d/2}^{D/2} r^2 dr = \frac{\pi f p_a}{12}(D^3 - d^3)$$

$$T = \frac{Ff}{3}\frac{D^3 - d^3}{D^2 - d^2}$$



Uniform Wear : Uniform Pressure



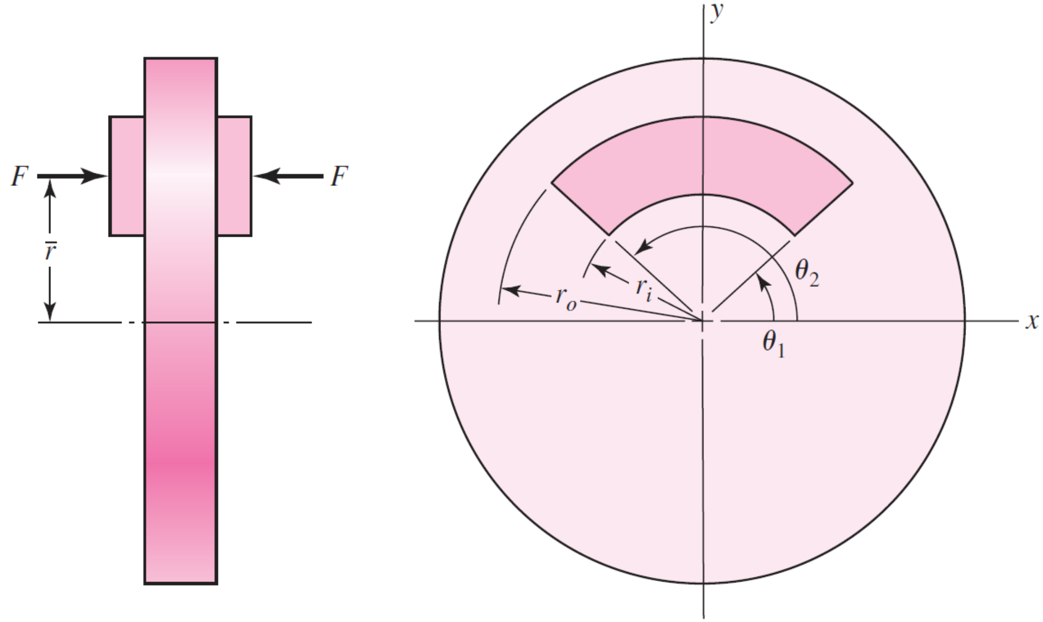
1. Transmitted torque calculated from uniform wear assumption is always lower than torque calculated from uniform pressure assumption.
2. For d/D > 0.6, both method give almost the same result.
3. Uniform wear assumption is frequently used, because it is safer than the uniform pressure assumption.
4. Many of disks are probable used to increase the transmitted torque.

Disk Brakes (1)

Calculation can be done by the same method as clutch calculation

$$F = \int_{\theta_1}^{\theta_2}\int_{r_i}^{r_o} pr\,dr\,d\theta = (\theta_2 - \theta_1)\int_{r_i}^{r_o} pr\,dr$$

$$T = \int_{\theta_1}^{\theta_2}\int_{r_i}^{r_o} f\,pr^2\,dr\,d\theta = (\theta_2 - \theta_1) f\int_{r_i}^{r_o} pr^2\,dr$$



Equivalent radius, r_e ⇨ $fFr_e = T$

$$r_e = \frac{T}{fF} = \frac{\int_{r_i}^{r_o} pr^2\,dr}{\int_{r_i}^{r_o} pr\,dr}$$

Position to center of force, $F\bar{r} = M_x$

$$M_x = F\bar{r} = \int_{\theta_1}^{\theta_2}\int_{r_i}^{r_o} pr(r\sin\theta)\,dr\,d\theta$$

$$\bar{r} = \frac{M_x}{F} = \frac{(\cos\theta_1 - \cos\theta_2)}{\theta_2 - \theta_1} r_e$$

Disk Brakes (2)

Uniform wear

By uniform wear assumption

$$pr = p_a r_i = p_a \frac{d}{2}$$

Actuating force, F

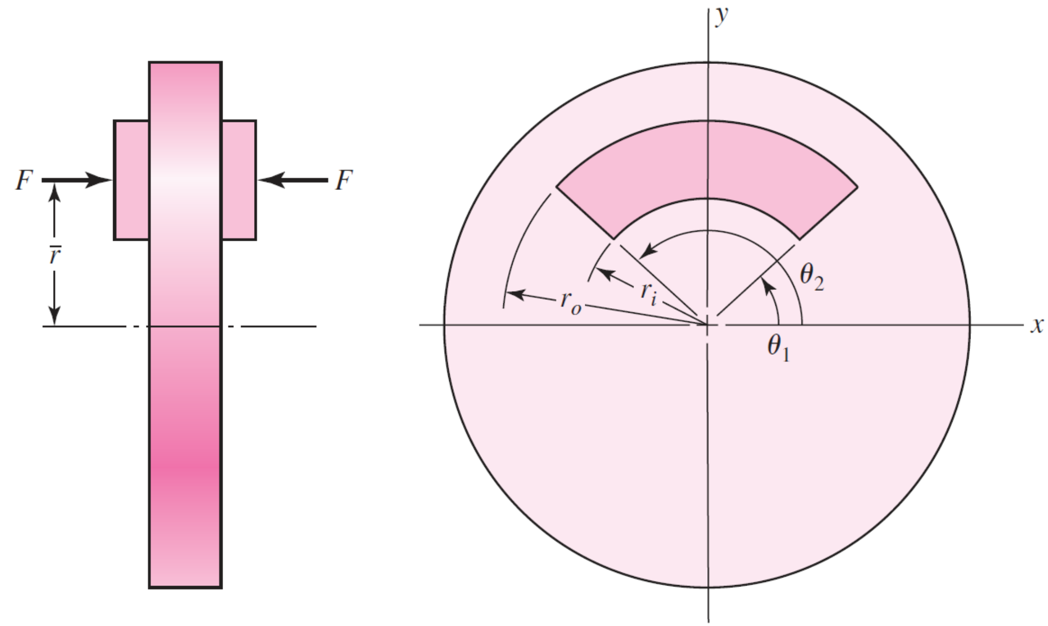
$$F = \int_{\theta_1}^{\theta_2}\int_{r_i}^{r_o} pr\,dr\,d\theta = (\theta_2 - \theta_1)\int_{r_i}^{r_o} pr\,dr$$

$$\Rightarrow \quad F = (\theta_2 - \theta_1) p_a r_i (r_o - r_i)$$

Transmitted torque, T

$$T = \int_{\theta_1}^{\theta_2}\int_{r_i}^{r_o} f p r^2\,dr\,d\theta = (\theta_2 - \theta_1) f \int_{r_i}^{r_o} pr^2\,dr$$

$$\Rightarrow \quad T = \frac{1}{2}(\theta_2 - \theta_1) f p_a r_i (r_o^2 - r_i^2)$$



Equivalent radius, r_e

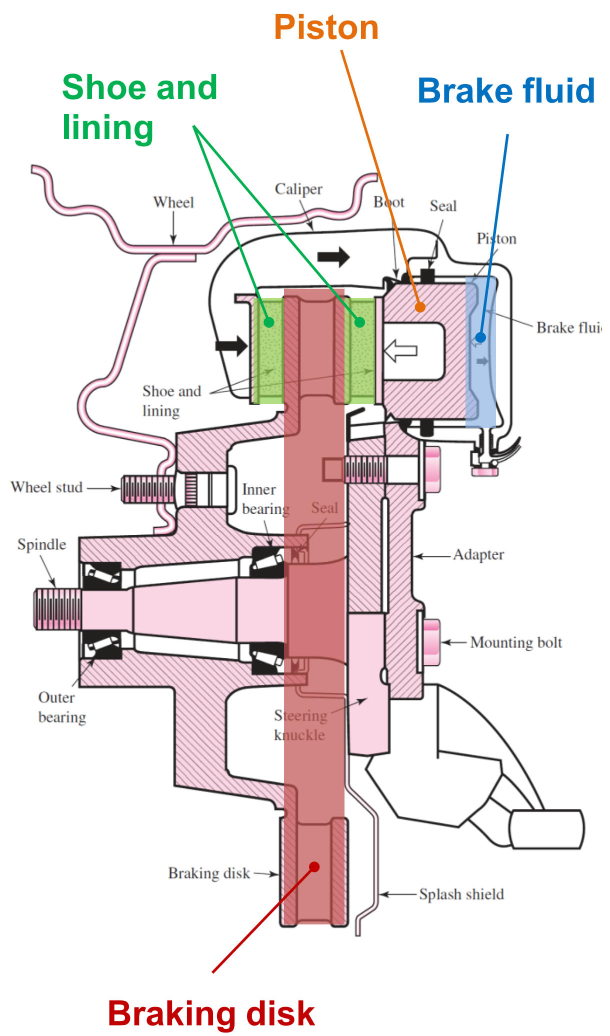
$$r_e = \frac{r_o + r_i}{2}$$

Position to center of force

$$\bar{r} = \frac{\cos\theta_1 - \cos\theta_2}{\theta_2 - \theta_1} \cdot \frac{r_o - r_i}{2}$$

Drum Brakes : Disk Brakes

Drum brakes	Disk brakes
• Can be designed for self-energization. The braking effort can be reduced.	• Has no self-energization.
• Slight change in coef. of friction (from temperature) will cause a large change in the pedal force required for braking. • More force is required to obtain the same braking torque	• Not so susceptible to change in the coefficient of friction • Floating caliper brake design help to compensate for wear and ensure a fairly constant pressure over the area of the friction pads



Example

Two annular pads, r_i = 3.875 in, r_o = 5.50 in, subtend an angle of 108°, have a coefficient of friction of 0.37, and are actuated by a pair of hydraulic cylinders 1.5 in in diameter. The torque requirement is 13,000 lbf·in. For uniform wear

(a) Find the largest normal pressure p_a.

(b) Estimate the actuating force F.

(c) Find the equivalent radius re and force location.

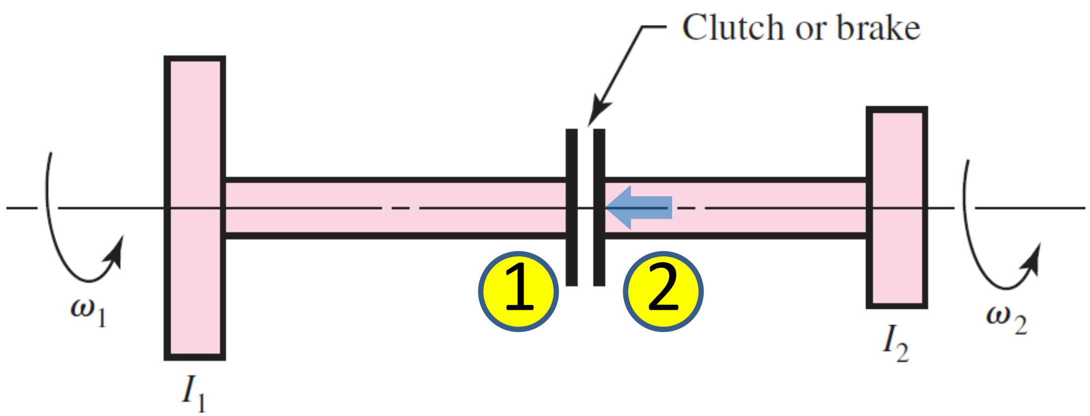
(d) Estimate the required hydraulic pressure

[Ex.16-3 Shigley's Mechanical Engineering Design 9th Edition. Richard G. Budynas and J. Keith Nisbett]

Energy Considerations (1)

Capability of a clutch or brake is limited by 2 factors,

1. The characteristics of material (coeff. of friction, safe normal pressure) ➡ Transmitted torque
2. Ability to dissipate heat ➡ Temperature-rise problem



Assumption:

Two shafts are rigid and that the clutch torque is constant

$$\Rightarrow \begin{cases} I_1\ddot{\theta}_1 = -T \\ I_2\ddot{\theta}_2 = T \end{cases}$$

	I_1	I_2
$t = 0$	ω_1	ω_2
$t = t_f$ Operation is completed	ω_f	ω_f

Transmitted torque, T depends on shape, coef. of friction, max. pressure

Energy Considerations (2)

$$I_1\ddot{\theta}_1 = -T \quad \Rightarrow \quad \dot{\theta}_1 = -\frac{T}{I_1}t + \omega_1$$

$$I_2\ddot{\theta}_2 = T \quad \Rightarrow \quad \dot{\theta}_2 = \frac{T}{I_2}t + \omega_2$$

Relative velocity

$$\dot{\theta} = \dot{\theta}_1 - \dot{\theta}_2 = \left(-\frac{T}{I_1}t + \omega_1\right) - \left(\frac{T}{I_2}t + \omega_2\right)$$

$$= \omega_1 - \omega_2 - T\left(\frac{I_1 + I_2}{I_1 I_2}\right)t$$

Time required for the entire operation t_f

$$\dot{\theta}_1 = \dot{\theta}_2$$
$$\dot{\theta} = 0$$

$$\Rightarrow \quad t_f = \frac{I_1 I_2(\omega_1 - \omega_2)}{T(I_1 + I_2)}$$

The rate of energy-dissipation

Sliding loss

$$u = T\dot{\theta} = T\left[\omega_1 - \omega_2 - T\left(\frac{I_1 + I_2}{I_1 I_2}\right)t\right]$$

Total energy-dissipation

$$E = \int_0^{t_f} u\,dt = T\int_0^{t_f}\left[\omega_1 - \omega_2 - T\left(\frac{I_1 + I_2}{I_1 I_2}\right)t\right]dt$$

$$E = \frac{I_1 I_2(\omega_1 - \omega_2)^2}{2(I_1 + I_2)}$$

Energy Considerations (3)

Total energy dissipated can also be calculated by using energy method

	I_1	I_2
$t = 0$	ω_1	ω_2
$t = t_f$ Operation is completed	ω_f	ω_f

$$E_0 = \frac{1}{2} I_1 \omega_1^2 + \frac{1}{2} I_2 \omega_2^2$$

$$E_f = \frac{1}{2} I_1 \omega_f^2 + \frac{1}{2} I_2 \omega_f^2$$

Total energy-dissipation

$$E = E_0 - E_f = \frac{1}{2} I_1 (\omega_1^2 - \omega_f^2) + \frac{1}{2} I_2 (\omega_2^2 - \omega_f^2)$$

$$E = \frac{I_1 I_2 (\omega_1 - \omega_2)^2}{2(I_1 + I_2)}$$

$$\dot{\theta}_1 = -\frac{T}{I_1} t + \omega_1 \Rightarrow \omega_f = -\frac{T}{I_1} t_f + \omega_1$$

$$\dot{\theta}_2 = \frac{T}{I_2} t + \omega_2 \Rightarrow \omega_f = \frac{T}{I_2} t_f + \omega_2$$

$$\omega_1 - \omega_2 = T t_f \left(\frac{I_1 + I_2}{I_1 I_2} \right)$$

Energy Considerations (Brake)

For brake, only drums or disks rotate,

brake pads are stationary parts

	I_1	$I_2=0$
$t = 0$	ω_1	0
$t = t_f$ Operation is completed	ω_f	0

$$E_0 = \frac{1}{2} I_1 \omega_1^2$$

$$E_f = \frac{1}{2} I_1 \omega_f^2$$

Total energy-dissipation

$$E = E_0 - E_f = \frac{1}{2} I_1 \omega_1^2 - \frac{1}{2} I_1 \omega_f^2$$

$$E = \frac{1}{2} I_1 \left(\omega_1^2 - \omega_f^2 \right)$$

Temperature Rise

Total energy dissipated is changed to be thermal energy and lead to temperature rise.

$$E = mC_p\Delta T$$

m : mass of clutch or brake parts, kg

C_p : specific heat capacity

500 J/(kg-°C) for steel or cast iron

ΔT : temperature rise, °C

Heat from energy dissipated is transmitted to environment, hence the temperature is decrease.

$$\frac{T - T_\infty}{T_1 - T_\infty} = \exp\left(-\frac{\bar{h}_{CR} A}{W C_p} t\right)$$

T : temperature at time t

T_1 : initial temperature

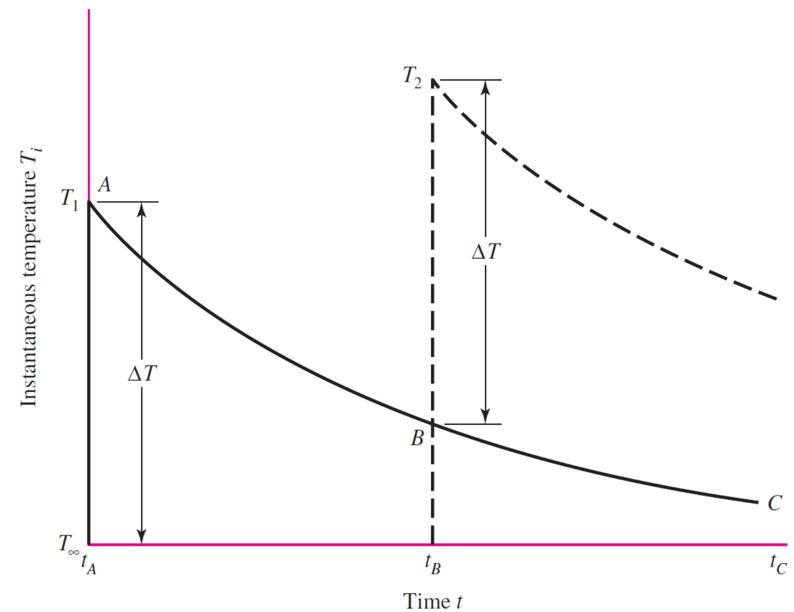
T_∞ : environment temperature

$\bar{h}_{CR}$: overall coef. of heat transf.

A : lateral surface area

W : mass of the object

C_p : specific heat capacity



The effect of clutching or braking operation on temperature.

The temperature rise may be different for each operation.

Friction Materials (1)

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

	Woven Lining	Molded Lining	Rigid Block
Compressive strength, kpsi	10–15	10–18	10–15
Compressive strength, MPa	70–100	70–125	70–100
Tensile strength, kpsi	2.5–3	4–5	3–4
Tensile strength, MPa	17–21	27–35	21–27
Max. temperature, °F	400–500	500	750
Max. temperature, °C	200–260	260	400
Max. speed, ft/min	7500	5000	7500
Max. speed, m/s	38	25	38
Max. pressure, psi	50–100	100	150
Max. pressure, kPa	340–690	690	1000
Frictional coefficient, mean	0.45	0.47	0.40–45

Some Properties of Brake Linings

Friction Materials (2)

Friction Materials for Clutches

Material	Friction Coefficient		Max. Temperature		Max. Pressure	
	Wet	Dry	°F	°C	psi	kPa
Cast iron on cast iron	0.05	0.15–0.20	600	320	150–250	1000–1750
Powdered metal* on cast iron	0.05–0.1	0.1–0.4	1000	540	150	1000
Powdered metal* on hard steel	0.05–0.1	0.1–0.3	1000	540	300	2100
Wood on steel or cast iron	0.16	0.2–0.35	300	150	60–90	400–620
Leather on steel or cast iron	0.12	0.3–0.5	200	100	10–40	70–280
Cork on steel or cast iron	0.15–0.25	0.3–0.5	200	100	8–14	50–100
Felt on steel or cast iron	0.18	0.22	280	140	5–10	35–70
Woven asbestos* on steel or cast iron	0.1–0.2	0.3–0.6	350–500	175–260	50–100	350–700
Molded asbestos* on steel or cast iron	0.08–0.12	0.2–0.5	500	260	50–150	350–1000
Impregnated asbestos* on steel or cast iron	0.12	0.32	500–750	260–400	150	1000
Carbon graphite on steel	0.05–0.1	0.25	700–1000	370–540	300	2100

*The friction coefficient can be maintained with ±5 percent for specific materials in this group.

Friction Materials (3)

Characteristics of Friction Materials for Brakes and Clutches

Material	Friction Coefficient f	Maximum Pressure p_{max}, psi	Maximum Temperature Instantaneous, °F	Maximum Temperature Continuous, °F	Maximum Velocity V_{max}, ft/min	Applications
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