CH # 16

Clutches, Brakes, Couplings and Flywheels

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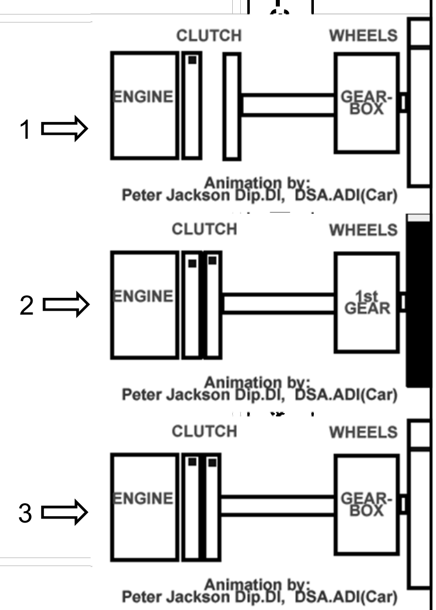
Introduction

ME-3

Clutch;

- Clutch is a device that transfers power
- The thee animations show
  1. Clutch is not engaged
  2. Clutch is partially engaged
  3. Clutch is fully engaged



Introduction

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Brake;

- A brake is a device for slowing or stopping the motion
- to keep it from starting to move again.
- The kinetic energy lost by the moving part is usually translated to heat by friction.

Introduction

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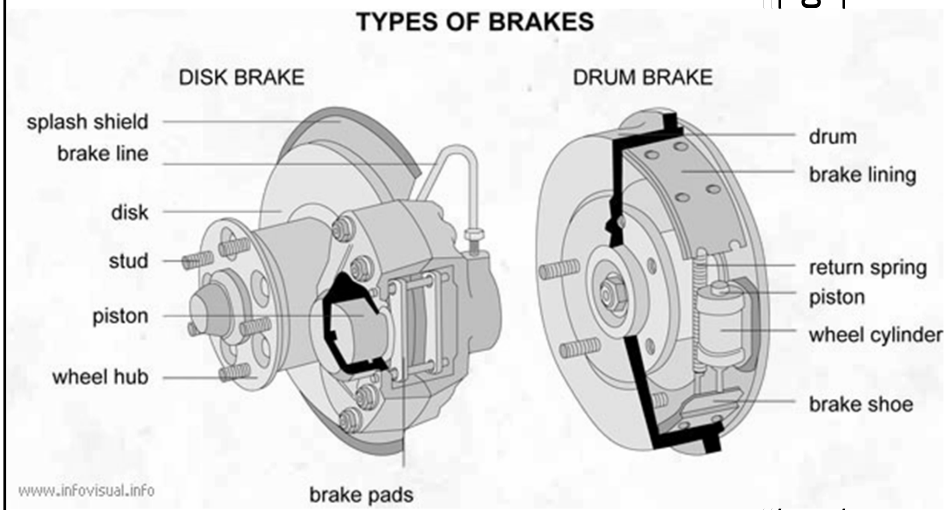

Introduction

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Types

- Rim types with internal expanding shoes
- Rim types with external contracting shoes
- Band type
- Disk or axial type
- Cone type
- Miscellaneous types

Introduction

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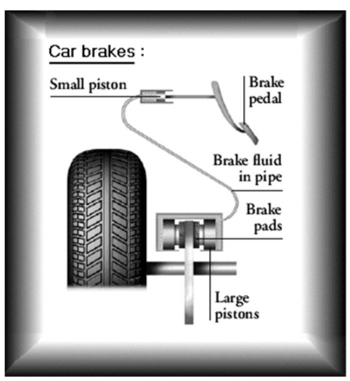
Parameters to analyze

- The actuating force
- The torque required
- The energy loss
- The temperature rise



16-2 Internal expanding rim clutches and brakes

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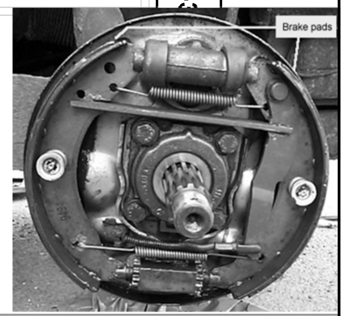
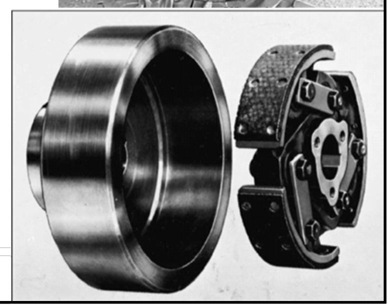
- Internal expanding clutch and brake is shown in figure
- The sub-types are;
  - Expanding-ring
  - Centrifugal
  - Magnetic
  - Hydraulic and
  - Pneumatic





16-2 Internal expanding rim clutches and brakes

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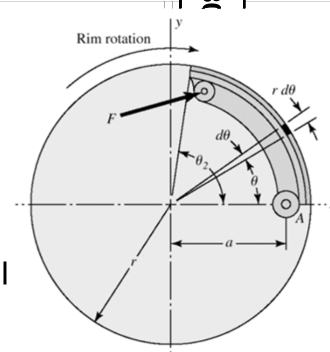
- Consider a pressure *P* acting on the element area located at θ.
- Consider the maximum pressures is p_a which is located at θ_a.
- The shoe deforms by an angle $d\theta$
- It can then be shown that the elemental normal force is then given by;

$$dN = \frac{p_a r b}{sin\theta_a} sin\theta d\theta$$



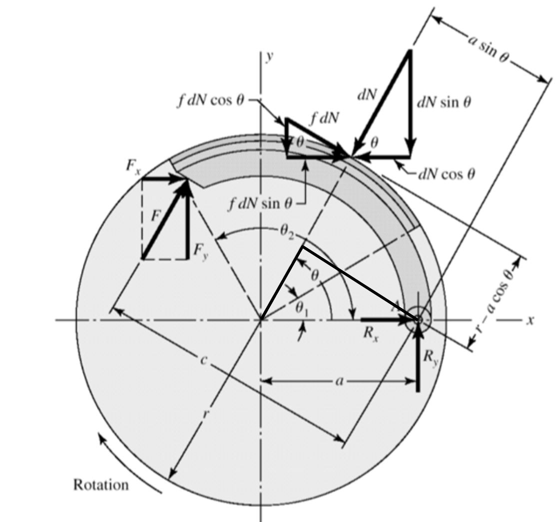
Design II

16-2 Internal expanding rim clutches and brakes

Forces.

We are interested in…

- Actuating force F
- Torque T
- Pin reactions R_x & R_y



16-2 Internal expanding rim clutches and brakes

- Taking moment of the frictional force about pivot point

$$M_f = \int_{\theta_1}^{\theta_2} f dN (r - a cos\theta)$$

- Where

$$dN = \frac{p_a r b}{sin\theta_a} sin\theta d\theta$$

- Then

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

$$M_f = \frac{f p_a b r}{4 sin\theta_a} [4r(cos\theta_1 - cos\theta_2) - a(cos2\theta_1 - cos2\theta_2)]$$

- And similarly the moment of the normal force is

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

Note:

$$sin\theta cos\theta = \frac{sin2\theta}{2}$$

$$sin^2\theta = \frac{1 - cos2\theta}{2}$$

16-2 Internal expanding rim clutches and brakes

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$$M_N = \frac{p_a bra}{4sin\theta_a}[2(\theta_2 - \theta_1) - (sin2\theta_2 - sin2\theta_1)]$$

- If $\theta_1 = 0^o$

$$M_f = \frac{fp_a br}{sin\theta_a}\left(r - rcos\theta_2 - \frac{a}{2}sin^2\theta_2\right)$$

$$M_N = \frac{p_a bra}{sin\theta_a}\left(\frac{\theta_2}{2} - \frac{1}{4}sin2\theta_2\right)$$

- The actuating force acting on the shoe is (taking moment about the pivot point);

$$F \times C + M_f - M_N = 0$$

$$F = \frac{M_N - M_f}{C}$$

$$F = \frac{M_N + M_f}{C}$$ If drum is rotating in reverse

16-2 Internal expanding rim clutches and brakes

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- And Torque is due to the friction force (tangential component) and is given by;

$$T = \int fdN \times r = \frac{fp_a br^2}{sin\theta_a}\int_{\theta_1}^{\theta_2} sin\theta d\theta$$

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

- Reactions are calculated by summing forces in *x* and *y* directions (for c.w and for c.c.w):

$$R_x = \frac{p_a br}{\sin\theta_a}(A \mp fB) - F_x, \quad R_y = \frac{p_a br}{\sin\theta_a}(B \pm fA) - F_y$$

- Where

$$A = \left(\frac{1}{2}\sin^2\theta\right)_{\theta_1}^{\theta_2} \text{ and } B = \left(\frac{\theta}{2} - \frac{1}{4}\sin 2\theta\right)_{\theta_1}^{\theta_2}$$

To Remember

- The governing equations are valid when
  - The coordinate system has its origin at the center of the drum. The +ve x-axis is taken through the hinge pin. The +ve y-axis is always in the direction of the shoe.
  - The pressure is proportional to the distance from the pivot point.
  - The centrifugal effects are neglected
  - The shoe is rigid
  - The coefficient of friction is not changing

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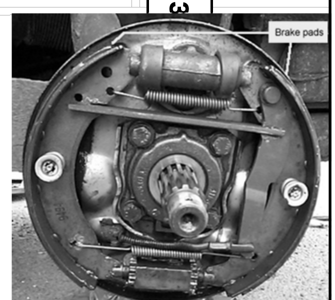
Self-energizing action

- Drum brakes have a natural "self-applying" characteristic, known as "self-energizing.
- The rotation of the drum can drag the shoe into the friction surface, causing the brake to bite harder, which increases the force holding them together.
- This effect occurs on one shoe.
- Less actuating force is then required
- The forces are different on each brake shoe resulting in one shoe wearing faster.



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Self-energizing action

- Also self-energization can be obtained from the equation

$$F = \frac{M_N - M_f}{C}$$

- If $M_N = M_f$ then zero actuating force is required
- Due to the self-energizing action, the brake mechanism and brake shoes should be properly designed *(Serve and Non-serve brake system)*.

Video

- More information on this topic: TechOne: Automotive Brakes by Jack Erjavec, Cengage Learning, 01-Sep.-2003

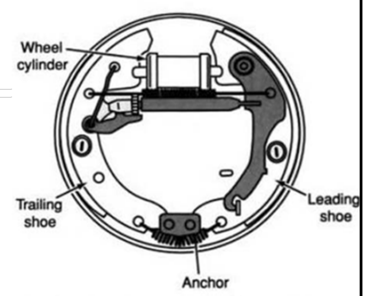

Figure 1. A typical nonservo brake assembly.

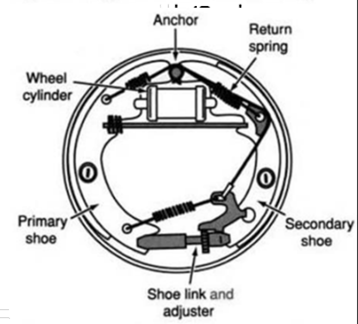

Figure 2. A typical duo-servo brake assembly.

Example 16-2

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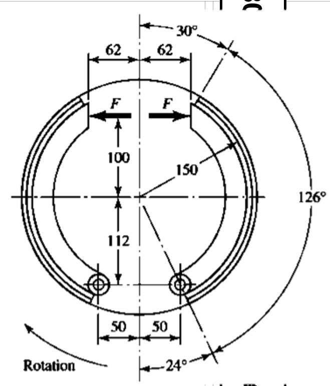
The brake shown in Fig. 16–8 is 300 mm in diameter and is actuated by a mechanism that exerts the same force *F* on each shoe. The shoes are identical and have a face width of 32 mm. The lining is a molded asbestos having a coefficient of friction of 0.32 and a pressure limitation of 1000 kPa. Estimate the maximum

a) Actuating force *F*
b) Braking capacity
c) Hinge-pin reactions.



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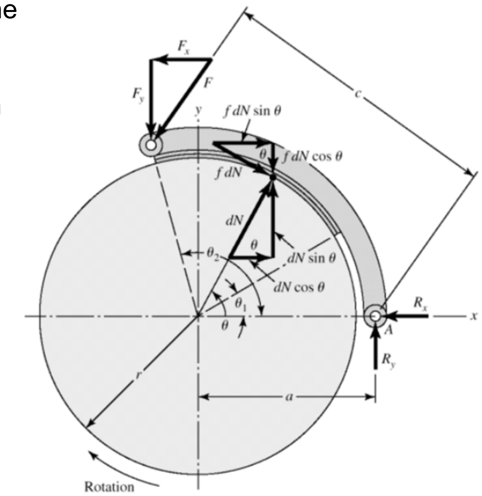
16-3 External Contracting Rim Clutches and Brakes

- The moments of the forces are the same as in case of internal contracting brakes and clutches
- The only change is the change in direction of the normal force *dN*.
- Thus

$$F = \frac{M_N + M_f}{c} \text{ (For c.w. rotation)}$$

$$F = \frac{M_N - M_f}{c} \text{ (For c.c.w. rotation)}$$

- M_N and M_f are the same as calculated in section 16-2.



16-3 External Contracting Rim Clutches and Brakes

- Reactions forces on the pin are;

$$R_x = \frac{p_a br}{\sin\theta_a}(A + fB) - F_x, \quad R_y = \frac{p_a br}{\sin\theta_a}(fA - B) + F_y \quad \text{(For CW)}$$

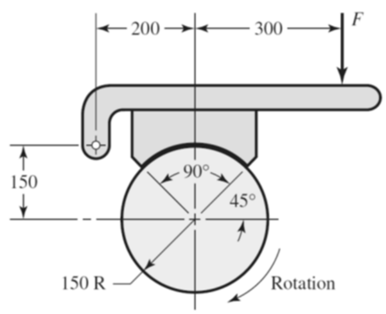
$$R_x = \frac{p_a br}{\sin\theta_a}(A - fB) - F_x, \quad R_y = \frac{p_a br}{\sin\theta_a}(-fA - B) + F_y \quad \text{(For CCW)}$$

- A and B are the same as calculated in section 16-2.

c

Problem 16-5

The block-type hand brake shown in the figure has a face width of 30 mm and a mean coefficient of friction of 0.25. For an estimated actuating force of 400 N, find the maximum pressure on the shoe and find the braking torque.



16-4 Band type clutches and brakes

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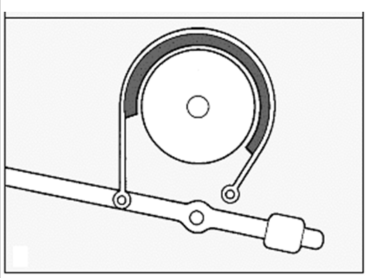
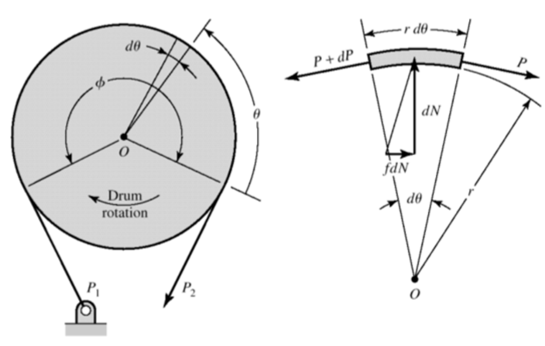
- Band brakes are used in power excavators and in hoisting and other machinery.
- Because of friction and rotation of the drum, P_2 (force in N) is less than P_1.
- Consider a small element of length $d\theta$.





16-4 Band type clutches and brakes

Vertical forces

$$(P+dP)\sin\frac{d\theta}{2}+P\sin\frac{d\theta}{2}-dN=0$$

Ignore dP

$$2P\sin\frac{d\theta}{2}=dN$$

For small angle, $\sin\frac{d\theta}{2}=\frac{d\theta}{2}$

$$dN=Pd\theta \rightarrow (1)$$

Horizontal forces

$$(P+dP)\cos\frac{d\theta}{2}-P\cos\frac{d\theta}{2}-fdN=0$$

For small angle, $\cos\frac{d\theta}{2}=1$

$$P+dp-P-fdN=0$$

$$dP-fdN=0\rightarrow(2)$$

From (1) in (2) yeilds

$$dP-fPd\theta=0\rightarrow(2)$$

Solve (2) by integrating from P_1 to P_2

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16-4 Band type clutches and brakes

$$\int_{P_2}^{P_1}\frac{dP}{P}-f\int_0^{\phi}d\theta=0$$

$$\ln\frac{P_1}{P_2}=f\phi$$

$$\frac{P_1}{P_2}=e^{f\phi}$$

And Torque is, $T=(P_1-P_2)\frac{D}{2}$

To find out the maximum pressure p_a,

The normal force is

$$dN=pbrd\theta$$

Put in (1) to get

$$Pd\theta=pbrd\theta \Rightarrow p=\frac{P}{br}=\frac{2P}{bD}$$

And p_a is then

$$p_a=\frac{2P_1}{bD}$$

Note that P_1 and P_2 are forces in “N” and *p* is band the pressure in “Pa”

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Problem 16-11

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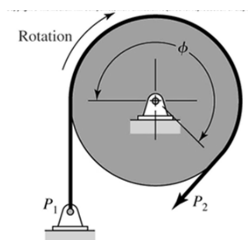
The maximum band interface pressure on the brake shown in the figure is 620kPa. Use a 350 mm-diameter drum, a band width of 25 mm, a coefficient of friction of 0.3, and an angle-of-wrap of 270°. Find the band tensions and the torque capacity.

- $p_a = \frac{2P_1}{bD}$
- $\frac{P_1}{P_2} = e^{f\emptyset}$



16-10 Friction Materials

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Table 16–2

Area of Friction Material Required for a Given Average Braking Power *Sources:* M. J. Neale, *The Tribology Handbook,* Butterworth, London, 1973; *Friction Materials for Engineers,* Ferodo Ltd., Chapel-en-le-frith, England, 1968.

		Ratio of Area to Average Braking Power, $(10^{-6})m^2/(joules/s)$		
Duty Cycle	**Typical Applications**	**Band and Drum Brakes**	**Plate Disk Brakes**	**Caliper Disk Brakes**
Infrequent	Emergency brakes	52	171	17.1
Intermittent	Elevators, cranes, and winches	171	434	43
Heavy-duty	Excavators, presses	342-422	832	86

16-10 Friction Materials

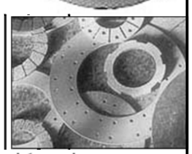
ME-3

Table 16–3

Characteristics of Friction Materials for Brakes and Clutches *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.

Material	Friction Coefficient f	Maximum Pressure p_{max}, MPa	Maximum Temperature Instantaneous, °C	Maximum Temperature Continuous, °C	Maximum Velocity V_{max}, m/s	Applications
Cermet	0.32	1.0	815	400		Brakes and clutches
Sintered metal (dry)	0.29–0.33	2.1-2.8	500-550	300-350	18	Clutches and caliper disk brakes
Sintered metal (wet)	0.06–0.08	3.4	500	300	18	Clutches
Rigid molded asbestos (dry)	0.35–0.41	0.7	350-400	180	18	Drum brakes and clutches
Rigid molded asbestos (wet)	0.06	2.1	350	180	18	Industrial clutches
Rigid molded asbestos pads	0.31–0.49	5.2	500-750	230-350	24	Disk brakes
Rigid molded nonasbestos	0.33–0.63	0.7-1.0		260-400	24-38	Clutches and brakes
Semirigid molded asbestos	0.37–0.41	0.7	350	150	18	Clutches and brakes
Flexible molded asbestos	0.39–0.45	0.7	350-400	150-180	18	Clutches and brakes
Wound asbestos yarn and wire	0.38	0.7	350	150	18	Vehicle clutches
Woven asbestos yarn and wire	0.38	0.7	260	130	18	Industrial clutches and brakes
Woven cotton	0.47	0.7	110	75	18	Industrial clutches and brakes
Resilient paper (wet)	0.09–0.15	2.8	150		PV < 18 MPa. m/s	Clutches and transmission bands

16-10 Friction Materials



- Manufacture of friction material is highly specialized process
- Consult the manufacturers for catalogue, handbook or specs
- Select the material based on your design requirements as well as the standard sizes available

The *woven-cotton lining* is produced as a fabric belt that is impregnated with resins and polymerized. It is used mostly in heavy machinery and is usually supplied in rolls up to 50 ft in length. Thicknesses available range from $\frac{1}{8}$ to 1 in, in widths up to about 12 in.

A *woven-asbestos lining* is made in a similar manner to the cotton lining and may also contain metal particles. It is not quite as flexible as the cotton lining and comes in a smaller range of sizes. Along with the cotton lining, the asbestos lining was widely used as a brake material in heavy machinery.

16-10 Friction Materials

ME-3

Molded-asbestos linings contain asbestos fiber and friction modifiers; a thermoset polymer is used, with heat, to form a rigid or semirigid molding. The principal use was in drum brakes.

Molded-asbestos pads are similar to molded linings but have no flexibility; they were used for both clutches and brakes.

Sintered-metal pads are made of a mixture of copper and/or iron particles with friction modifiers, molded under high pressure and then heated to a high temperature to fuse the material. These pads are used in both brakes and clutches for heavy-duty applications.

Cermet pads are similar to the sintered-metal pads and have a substantial ceramic content.

Problems from Chapter 16

- Problem No. 1 to 16

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