CH # 16

Clutches, Brakes, Couplings and Flywheels

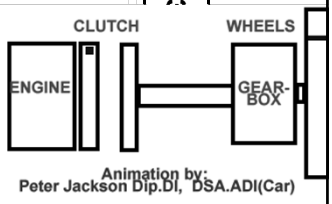
ME-305 Mechanical Engineering Design II

Introduction

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

1 ⇨



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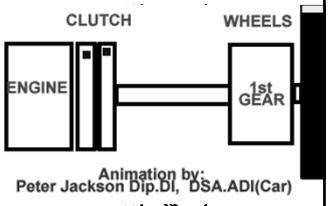

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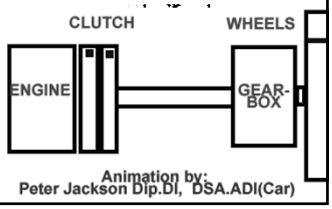

Introduction…

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.

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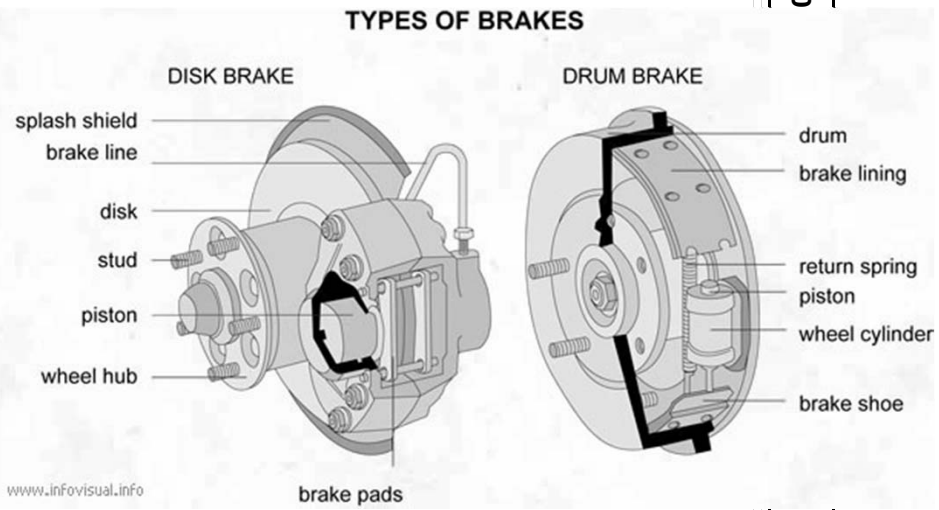
Introduction…



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Introduction…

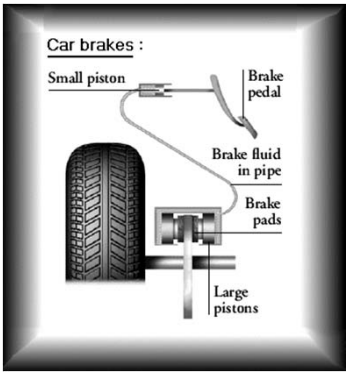
Types

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

Introduction…

Parameters to analyze

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



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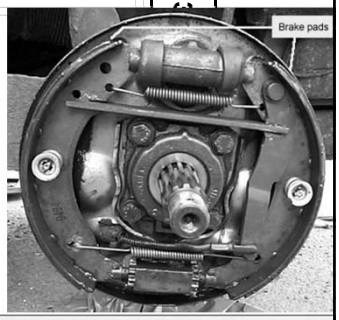
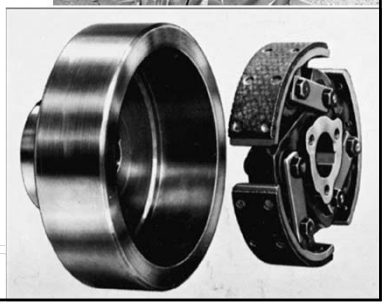
16-2 Internal expanding rim clutches and brakes

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

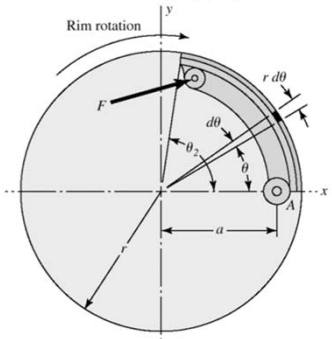




16-2 Internal expanding rim clutches and brakes...

- 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 $\mathrm{d}\theta$
- It can then be shown that the elemental normal force is then given by;

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

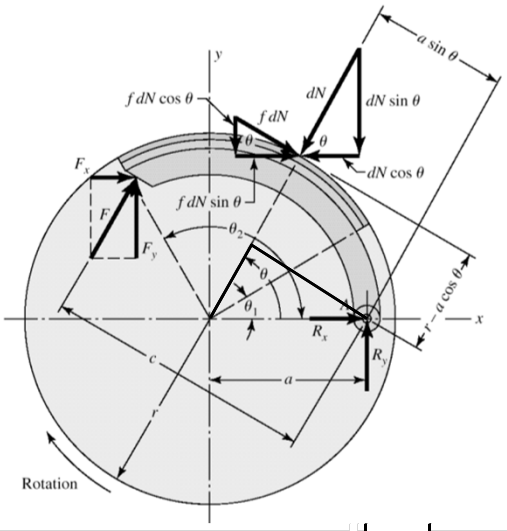


16-2 Internal expanding rim clutches and brakes...

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

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

$$M_f = \frac{f p_a b r}{4\sin\theta_a} [4r(\cos\theta_1 - \cos\theta_2) - a(\cos 2\theta_1 - \cos 2\theta_2)]$$

- 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:

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

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

16-2 Internal expanding rim clutches and brakes…

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

- If θ_1 = 0°

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

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

- 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

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16-2 Internal expanding rim clutches and brakes…

- 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

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

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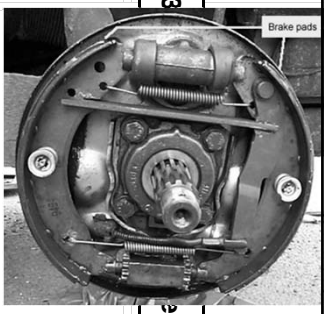
To Remember

- 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

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.



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)*.

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

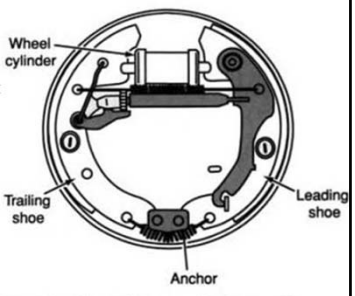


Figure 1. A typical nonservo brake assembly.

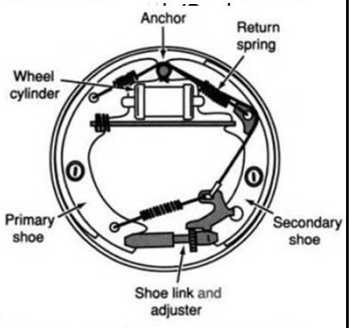
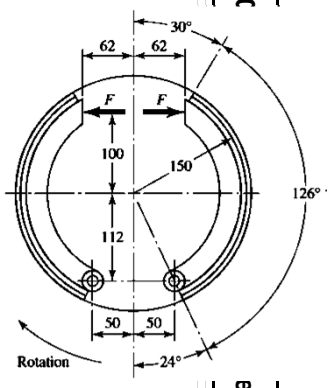


Figure 2. A typical duo-servo brake assembly.

Example 16-2

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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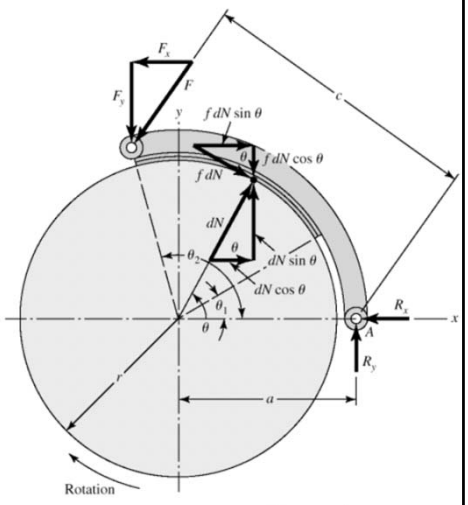
16-3 External Contracting Rim Clutches and Brakes

- 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:

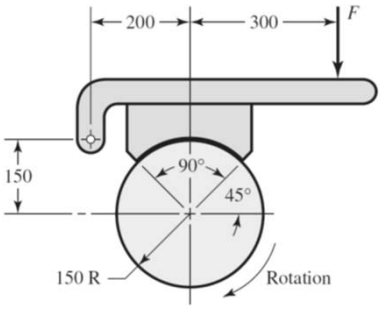
$$R_x = \frac{p_a br}{\sin\theta_a}(A + fB) - F_x, \quad R_y = \frac{p_a br}{\sin\theta_a}(fA - B) + F_y \quad \text{(For CW)}$$

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

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

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.



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Symmetrically located long shoe brake and clutch

Linear Sliding Wear

- Wear is denoted by *w* (in mm)
- Consider a block moving against a plate with pressure *p*, on area *A* and f_s
- We can write;
  Volume removed due to wear = work done by force by displacing the block →(1)
- Volume removed = $w \times A$ (mm^3)
- Work done by force = $f_s \times P \times A$ (N)
- Block displacement = $V \times t$ (mm.s)
- (1) becomes

$$w = KPVt \text{ (mm)}$$

- Where *K* is the proportionality constant depends on Material and units are $\left(\frac{cm^3.min}{kg.m.h}\right) = \left(\frac{mm^2}{N}\right)$.
- *K* is determined experimentally by the manufacturers.

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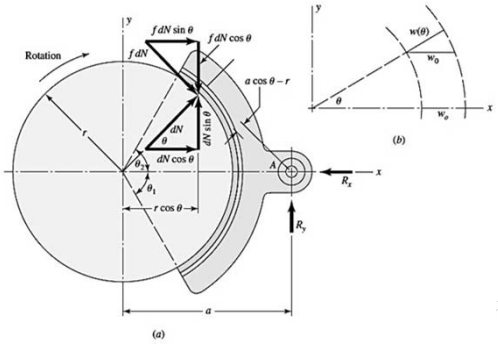
Symmetrically located long shoe brake and clutch…

- For constant wear in the shoe, we can write;

$$p(\theta) = \frac{w(\theta)}{KVt}$$

- Force is applied along x-axis, so maximum wear w_o is along x-axis. We can write;

$$w(\theta) = w_0 cos\theta$$



Symmetrically located long shoe brake and clutch…

- and

$$p(\theta) = \frac{w_0 cos\theta}{KVt}$$

- Since all surface see same w_0 and speed for same time so $\frac{w_0}{KVt}$ is constant. We write;

$$p(\theta) = \text{constant} \times cos\theta$$

$$p(\theta) = p_a cos\theta$$

- p_a is maximum pressure.
- We know that;

$$dN = pbrd\theta$$

- Put for p, we have

$$dN = p_a cos\theta . brd\theta$$

Symmetrically located long shoe brake and clutch…

- We need to Fix pivot point such that to get $M_f = 0$. Zero M_f ensure that;
  a) R_y creates a uniform wear
  b) The pressure distribution is cosinusoidal
- We can write (see Fig);

$$M_f = 2\int_0^{\theta_2=\theta_1} (fdN)(a.cos\theta - r) = 0$$

- Put for *dN* to get;

$$a = \frac{4rsin\theta_2}{2\theta_2 + sin2\theta_2}$$

- If "*a*" is not correct, M_f dislocates and non-uniform wear, cause the shoe life low.
- The reactions are;

$$R_x = \frac{p_a br}{2}(2\theta_2 + sin2\theta_2)$$

$$R_y = \frac{p_a brf}{2}(2\theta_2 + sin2\theta_2)$$

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Symmetrically located long shoe brake and clutch…

- Because of symmetry

$$R_x = -N \text{ and } R_y = -fN$$

- The torque capacity is

$$T = afN$$

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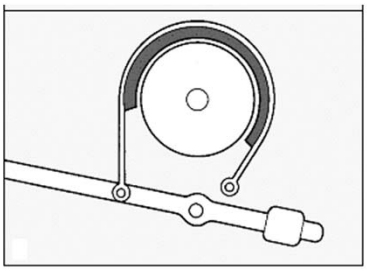
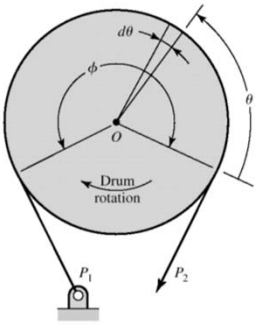
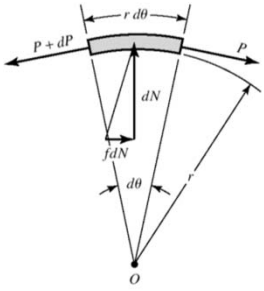
16-4 Band type clutches and brakes

- 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

$$dP-fPd\theta=0\rightarrow(3)$$

Solve (3) by integrating from P_1 to P_2

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

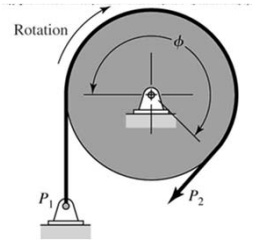
$$p_a = \frac{2P_1}{bD}$$

Note that P_1 and P_2 are forces in "N" and p is pressure in "Pa"

Problem 16-11

The maximum band interface pressure on the brake shown in the figure is 620 kPa. 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.

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



16-6 Disk Brakes

- No fundamental difference between disk clutch and disk brake
- Analysis procedure is the same
- In drum brake, a small change in “cof” due to temperature or moisture causes a large difference in pedal force
- 30% reduction is found in “cof” due to temp. and moist.
- No self-energization in disk brake and has minimum changes to “cof”.
- Disk brakes are designed based on;
  - Uniform pressure theory
  - Uniform wear theory

16-6 Disk Brakes…

Uniform Pressure

- The force is calculated as:

$$F = \int_{\theta_1}^{\theta_2} \int_{r_i}^{r_o} p r dr d\theta$$

$$F = \frac{1}{2}(\theta_1 - \theta_2) p_a (r_o^2 - r_i^2)$$

- The torque is calculated as:

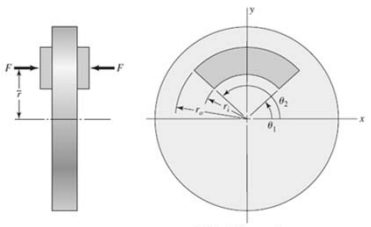
$$T = \frac{1}{3}(\theta_2 - \theta_1) f p_a (r_o^3 - r_i^3)$$

- The equivalent radius is given by:

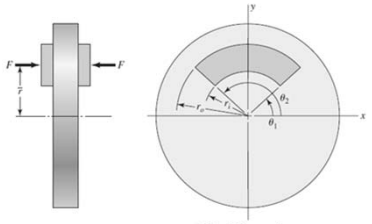
$$r_e = \frac{T}{fF} = \frac{2}{3}\frac{(r_o^3 - r_i^3)}{(r_o^2 - r_i^2)}$$

- The locating coordinate $\bar{r}$ of the actuating force is:

$$\bar{r} = \frac{M_x}{F} = \frac{2}{3}\frac{(r_o^3 - r_i^3)}{(r_o^2 - r_i^2)}\frac{(cos\theta_1 - \cos\theta_2)}{(\theta_2 - \theta_1)}$$



16-6 Disk Brakes…



Uniform wear

- The governing axial wear equation is:

$$\varpi = f_1 f_2 KPVt$$

- The force is calculated as;

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

- The torque is calculated as;

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

- The equivalent radius is given by:

$$r_e = \frac{r_0 + r_i}{2}$$

- The locating coordinate $\bar{r}$ of the actuating force is:

$$\bar{r} = \frac{(r_0 + r_i)}{2} \frac{(cos\theta_1 - \cos\theta_2)}{(\theta_2 - \theta_1)}$$

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Example 16-3

Two annular pads, r_i =98 mm, r_o = 140 mm, subtend an angle of 108°, have a coefficient of friction of 0.37, and are actuated by a pair of hydraulic cylinders 38 mm in diameter. The torque requirement is 1470 N-m. For uniform wear

(*a*) Find the largest normal pressure p_a .
(*b*) Estimate the actuating force *F*.
(*c*) Find the equivalent radius r_e and force location $\bar{r}$.
(*d*) Estimate the required hydraulic pressure.

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Example 16-3...

Solution:

(a)

$$p_a = \frac{2T}{(\theta_2 - \theta_1) f r_i \left(r_o^2 - r_i^2\right)} = 2.15 \text{ MPa}$$

(b)

$$F = (\theta_2 - \theta_1) p_a r_i (r_o - r_i) = 16.7 \text{ kN}$$

(c)

$$r_e = \frac{r_o + r_i}{2} = 119 \text{ mm}$$

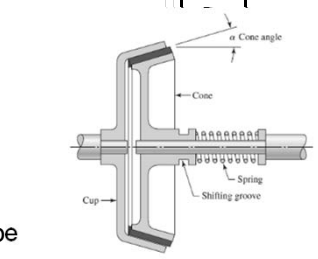
$$\bar{r} = \frac{(r_0 + r_i)}{2} \frac{(cos\theta_1 - cos\theta_2)}{(\theta_2 - \theta_1)} = 102 \text{ mm}$$

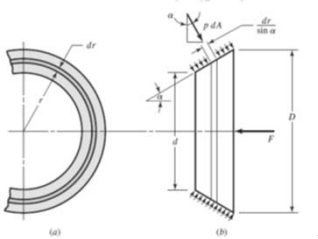
(d)

$$P_{hydraulic} = \frac{F}{A_{Piston}} = \frac{4F}{\pi d^2} = 14.7 \text{ MPa}$$

16-7 Cone Clutches and Brakes

- A cone clutch consists of a *cup* keyed or splined to one of the shafts.
- The cone angle α, the diameter and the face width of the cone are important design parameters.
- A good compromise cone angle is between 10 and 15°.
- The operating force *F* and the torque *T* can be determined using the Uniform Wear and Uniform Pressure assumptions.





16-7 Cone Clutches and Brakes…

ME-3

Uniform Wear

- The actuating force *F* (on an elemental area d*A*) is given by

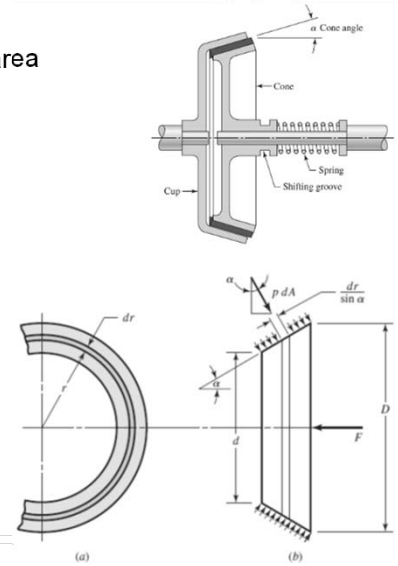
$$F = \int pdA\sin\alpha \rightarrow (1)$$

- It can be shown that $p = p_a \frac{d}{2r}$, and $dA = \frac{(2\pi r dr)}{\sin\alpha}$, (1) becomes

$$F = \int_{d/2}^{D/2} \left(p_a \frac{d}{2r}\right)\left(\frac{2\pi r dr}{\sin\alpha}\right)(\sin\alpha)$$

$$F = \pi p_a d \int_{d/2}^{D/2} dr$$

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



16-7 Cone Clutches and Brakes…

ME-3

- The Torque *T* caused by the differential friction force $fpdA$ is given by

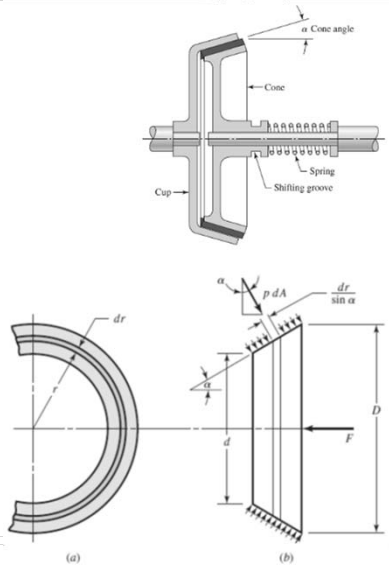
$$T = \int_{d/2}^{D/2} (rf)\left(p_a \frac{d}{2r}\right)\left(\frac{2\pi r dr}{\sin\alpha}\right)$$

$$T = \frac{\pi f p_a d}{sin\alpha} \int_{d/2}^{D/2} r dr$$

$$T = \frac{\pi f p_a d}{8\sin\alpha}(D^2 - d^2)$$

- The torque can also be written as

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



16-7 Cone Clutches and Brakes…

Uniform pressure

- For uniform pressure, $p = p_a$
- The actuating force *F* is given by

$$F = \int p_a dA \sin\alpha$$

$$F = \int_{d/2}^{D/2} (p_a)\left(\frac{2\pi r dr}{\sin\alpha}\right)(\sin\alpha)$$

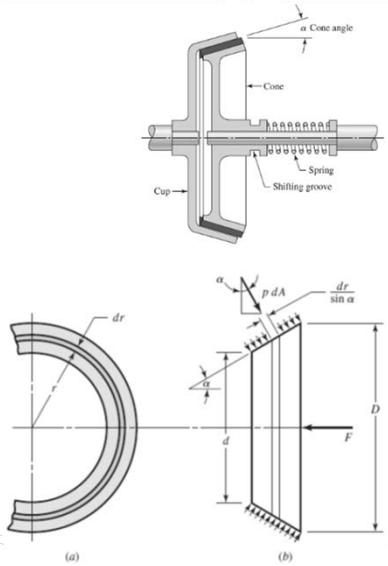
$$F = \frac{\pi p_a}{4}(D^2 - d^2)$$

- The torque *T* is

$$T = \int_{d/2}^{D/2} (rfp_a)\left(\frac{2\pi r dr}{\sin\alpha}\right)$$

$$T = \frac{\pi f p_a}{12\sin\alpha}(D^3 - d^3)$$

$$T = \frac{Ff}{3\sin\alpha}\left(\frac{D^3 - d^3}{D^2 - d^2}\right)$$



16-10 Friction Materials

- A brake or friction clutch should have the following lining material characteristics to a degree that is dependent on the severity of service
  - High and reproducible coefficient of friction
  - Imperviousness to environmental conditions, such as moisture
  - The ability to withstand high temperatures, together with good thermal conductivity, as well as high specific heat capacity
  - Good resiliency
  - High resistance to wear
  - Compatible with environment
  - Flexibility

16-10 Friction Materials…

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.

Duty Cycle	Typical Applications	Ratio of Area to Average Braking Power, $(10^{-6})m^2/(joules/s)$		
		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

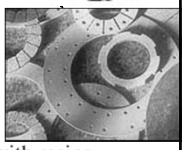
16-10 Friction Materials…

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…

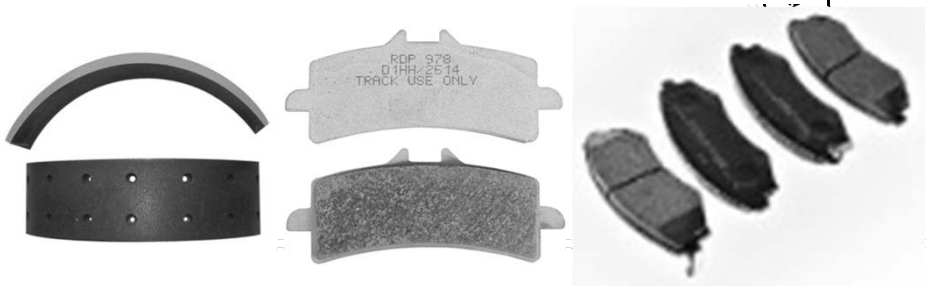
ME-3

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

- 16-1 to 16-7
- 16-11 to 16-16
- 16-19

9th Edition

ME-305 Mechanical Engineering Design II