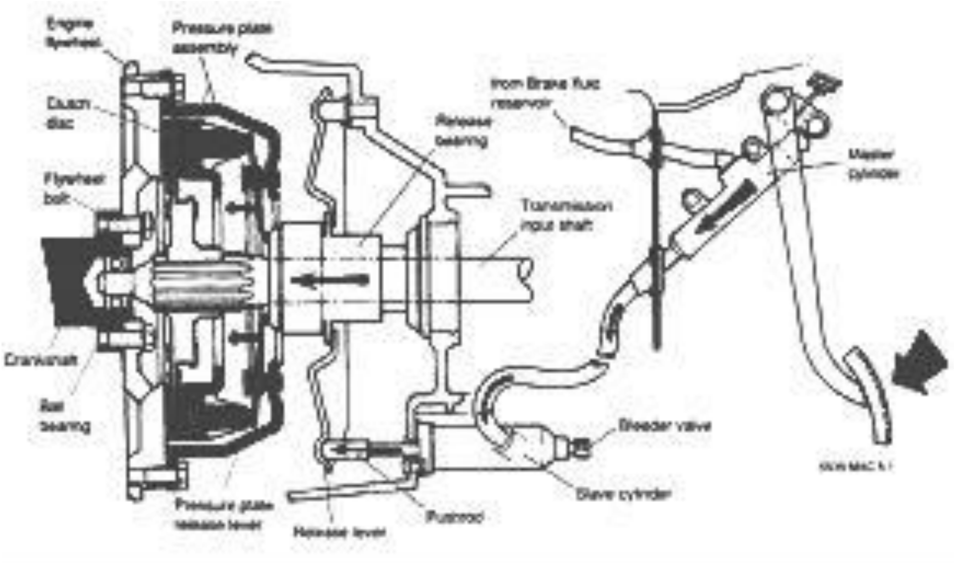
LECTURE SIXTEEN & SEVENTEEN

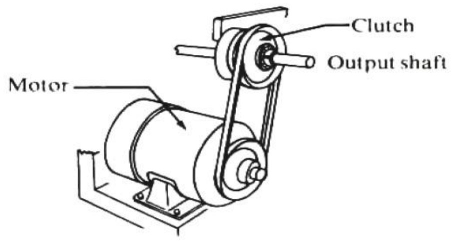
MOTION CONTROL "CLUTCHES AND BRAKES"



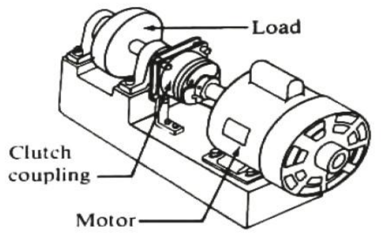
Reference: "Machine Elements in Mechanical Design" 4th Edition in SI units, By: Robert L. Mott, Chapter 22.

Introduction

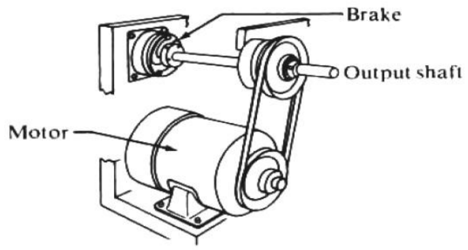
- A brake is a device used to bring a moving system to rest, to slow its speed or to control its speed to a certain value under varying condition.
- A clutch is a device used to connect or disconnect a driven component from the prime mover of the system.
- See section 22-2 (Page 833) and figure 22-1:



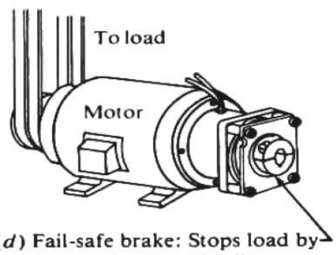
(*a*) Clutch: Transmits rotary motion to a parallel shaft only when coil is energized, by using sheaves, sprockets, gears, or timing pulleys.



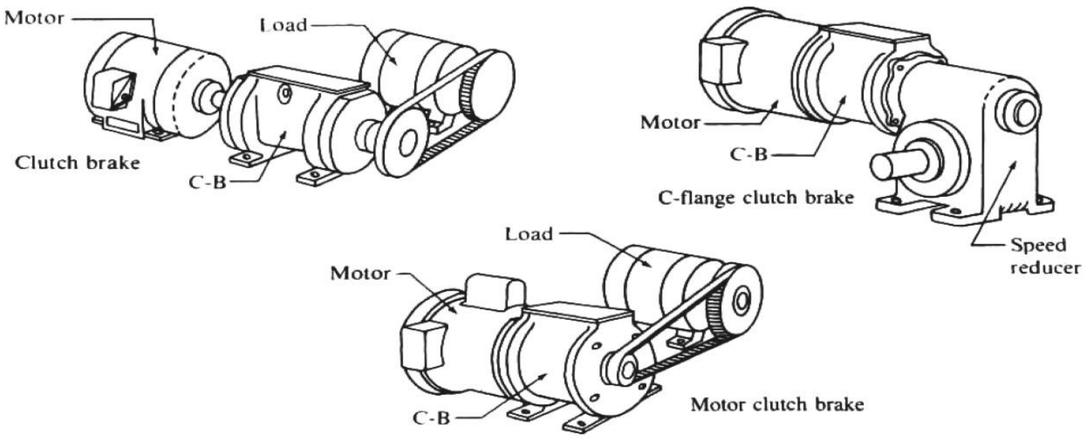
(*b*) Clutch coupling: Transmits rotary motion to an in-line shaft only when coil is energized. Split shaft applications.



(*c*) Brake: Stops (brakes) load when coil is energized. Panel shows clutch imparting rotary motion to the output (load) shaft while brake is de-energized. Conversely, de-energizing clutch coil and energizing brake coil causes load to stop.



(*d*) Fail-safe brake: Stops load by de-energization of coil; power off—brake on.

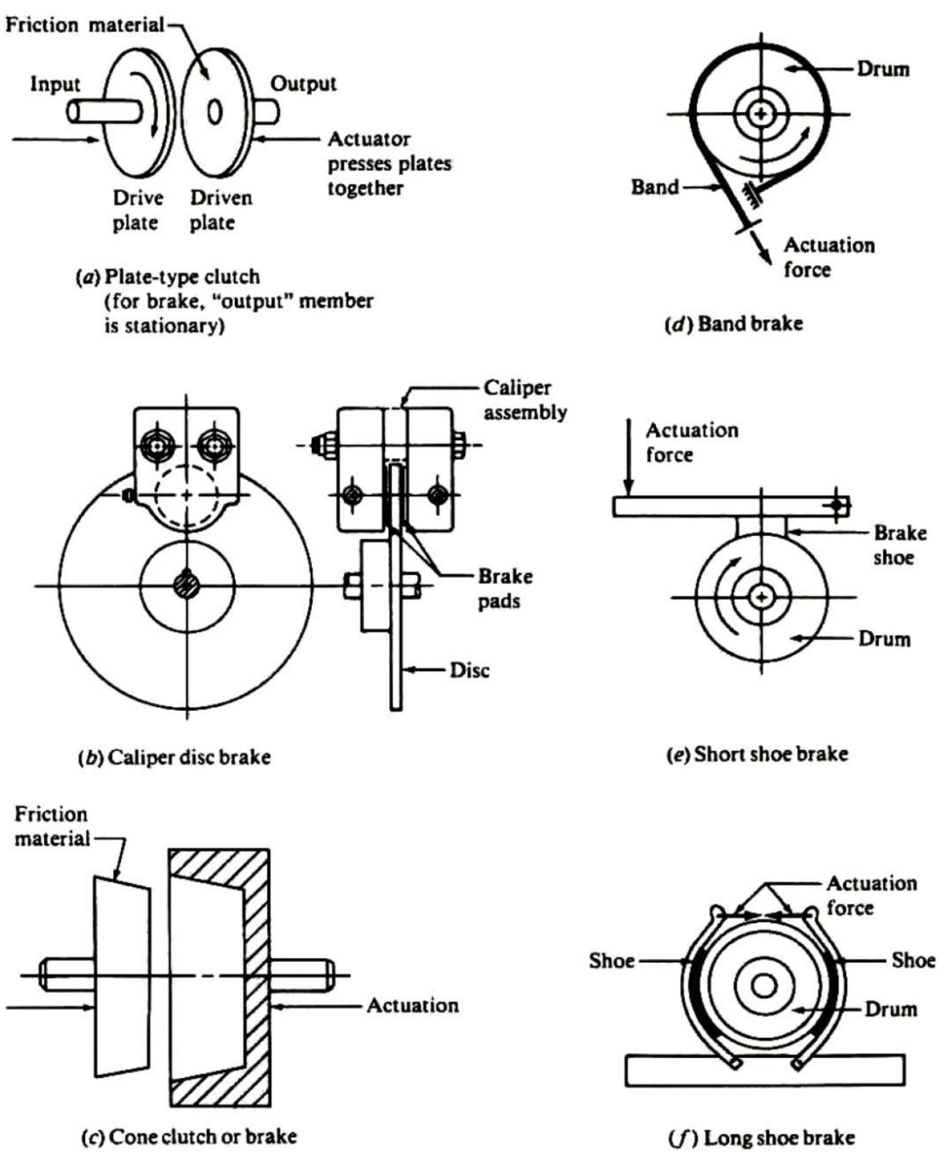


(*e*) Three types of mountings for clutch-brake modules: *Clutch brake* combines functions of clutch and brake in a complete preassembled package with input and output shafts. *C-flange clutch brake* performs same function but for use between NEMA "C" flange motor and speed reducer. *Motor clutch brake* is a preassembled module for mounting to a NEMA "C" flange motor and presenting an output shaft for connection to load.

FIGURE 22–1 Typical applications of clutches and brakes (Electroid Company, Springfield, NJ)

- Types of friction clutches & brakes (section 22-2 , page 835):

FIGURE 22–2 Types of friction clutches and brakes [(b), Tol-O-Matic, Hamel, MN]

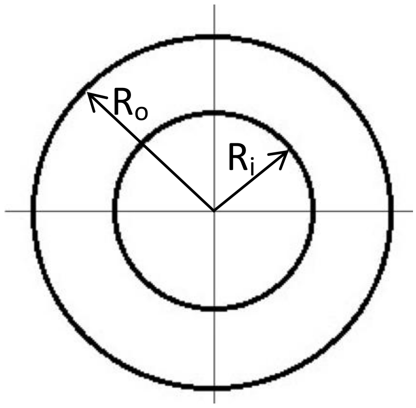


- Friction material and coefficient of friction (table 22-2, page 850):

TABLE 22–2 Coefficients of friction

Friction material	Dynamic friction coefficient		Pressure range	
	Dry	In oil	(psi)	(kPa)
Molded compounds	0.25–0.45	0.06–0.10	150–300	1035–2070
Woven materials	0.25–0.45	0.08–0.10	50–100	345–690
Sintered metal	0.15–0.45	0.05–0.08	150–300	1035–2070
Cork	0.30–0.50	0.15–0.25	8–15	55–100
Wood	0.20–0.45	0.12–0.16	50–90	345–620
Cast iron	0.15–0.25	0.03–0.06	100–250	690–1725
Paper-based		0.10–0.15		
Graphite/resin		0.10–0.14		

Plate-type clutch or brake (section 22-11, Page 851)



$$\boldsymbol{R_m = \frac{R_o + R_i}{2}}$$

$\boldsymbol{T_f = f * N * R_m}$ **………. (22-8) (page 851)**

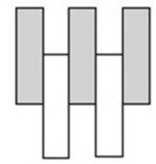
Where:

T_f : Friction torque
f : Coefficient of friction
N : Axial force which pressed plates together

Notes:

1. For multiple disc clutch or brake let (n) be the number of pairs of contact surfaces:

$$\boldsymbol{T_f = n * f * N * R_m}$$



2. If there are (n_{1d}) No. of discs on driving shafts and (n_{2d}) No. of discs on driven shaft then No. of pairs of contacts surface are:

$$\boldsymbol{n_p = n_{1d} + n_{2d} - 1}$$

3. A reasonable for the ratio (R_o/R_i) is from (1.2 – 2.5).

4. The required torque capacity is usually expressed as:

$$\boldsymbol{T_f = \frac{C * P_f * K}{\omega}} \quad \dots\dots (\mathbf{22-1})$$

Where:

C : Conversion factor for units
P_f: Power
K : Service factor base application
K = 1 for average condition; 1.5 for moderate duty; 3 for heavy duty
C = 1 if P: in watt and n: in rad/sec and T: in N.m

5. Judge the suitability of wear rating (WR)

$$\boldsymbol{WR = \frac{P_f}{A}} \dots\dots (\mathbf{22-9})$$

$$\boldsymbol{P_f = T_f * \omega} \dots\dots (\mathbf{22-10})$$

WR = 0.046 W/mm^2 for frequent application (conservation rating)
= 0.116 W/mm^2 for average service
= 0.46 W/mm^2 for infrequently use

Example Problem 22-6:

Compute the dimensions of an annular plate-type brake to produce a braking torque of 33.9 N.m. Springs will provide a normal force of 1423.4 N between the friction surfaces. The coefficient of friction is 0.25. The brake will be used in average industrial service, stopping a load from 78.5 rad/sec.

Solution:

1- Compute the required mean radius from equation (22-8).

$$R_m = \frac{T_f}{fN} = \frac{33.9\ N.m}{(0.25)*(1423.4N)} = 95.25\ mm$$

2- Specify a desired ratio of R_o/R_i and solve for the dimensions. A reasonable value for the ratio is approximately 1.50. The range can be from 1.2 to about 2.5, at the designer's choice. Using 1.50, $R_o = 1.50R_i$ and

$$R_m = \frac{R_o + R_i}{2} = \frac{1.5R_i + R_i}{2} = 1.25R_i$$

Then,

$$R_i = \frac{R_m}{1.25} = \frac{95.25\ mm}{1.25} = 76.2\ mm$$

$$R_o = 1.5R_i = 1.5(76.2) = 114.3\ mm$$

3- Compute the area of the friction surface:

$$A = \pi({R_o}^2 - {R_i}^2) = \pi((114.3)^2 - (76.2)^2) = 0.02278\ m^2$$

4- Compute the frictional power absorbed:

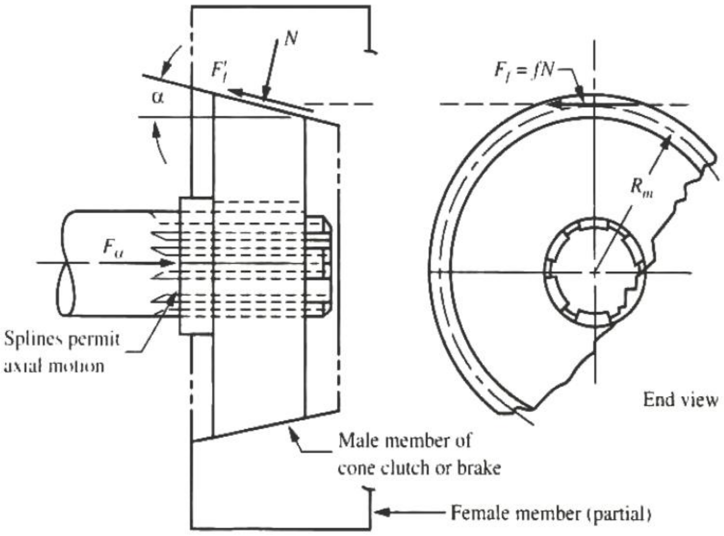
$$P_f = T_f * \omega = (33.9\ N.m)\left(78.5\frac{rad}{\text{sec}}\right) = 2661\ W$$

5- Compute the wear ratio:

$$WR = \frac{P_f}{A} = \frac{2661\ W}{22.7*10^3\ mm^2} = 0.116\ W/mm^2$$

6- Judge the suitability of *WR.* If *WR* is too high, return to Step 2 and increase the ratio. If *WR* is too low, decrease the ratio. In this example, WR is acceptable.

Cone clutch or Brake (section 22-13), Page 854



$$T_f = F_f * R_m = f * N * R_m \quad \text{...... (22-8)}$$

In addition to tangential friction force, there is a friction force along surface of cone = $F_f' = f N$

$$F_a = N \sin\alpha + F_{f'} \cos\alpha$$

$$F_a = N\,(\sin\alpha + f\cos\alpha)$$

$$N = \frac{F_a}{\sin\alpha + f\cos\alpha} \quad \ldots\ldots\ldots(22-12)$$

$$T_f = \frac{f * R_m * F_a}{\sin\alpha + f\cos\alpha} \quad \ldots\ldots\ldots(22-13)$$

Example Problem 22-7:

Compute the axial force required for a cone brake if it is to exert a braking torque of 67.8 N.m. The mean radius of the cone is 127mm Use f = 0.25. Try cone angles of 10°, 12°, and 15°.

Solution: We can solve Equation (22-13) for the axial force F_a:

$$F_a = \frac{T_f(\sin\alpha + f\cos\alpha)}{f * R_m} = \frac{(67.8\ \mathrm{N.m})(\sin\alpha + 0.25\cos\alpha)}{(0.25)(0.127)m}$$

$$F_a = 2135(\sin\alpha + 0.25\cos\alpha)\ N$$

Then the values of (F_a) as a function of the cone angle are as follows:

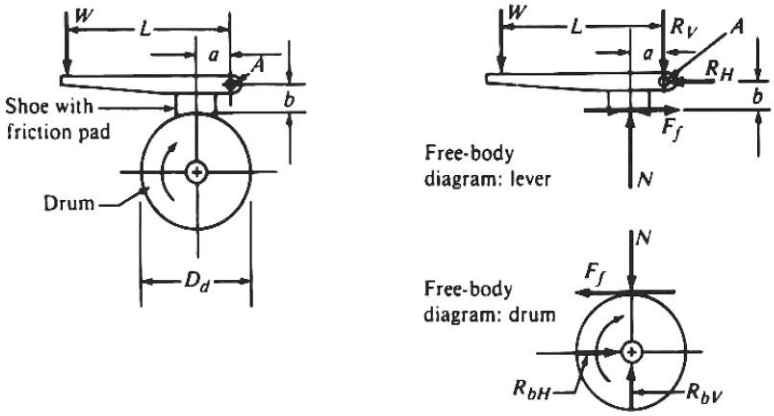
For α = 10°
F_a =898.5 N

For α = 12°
F_a =965.2 N

For α = 15°
F_a =1067.5 N

Short shoe drum brakes (section 22-14), Page 855:

$\sum M_A = 0$



$$\mathbf{0 = W * L - N * a + F_f * b}$$

But note that: $\boldsymbol{F_f = f * N}$

Or: $\boldsymbol{N = \frac{F_f}{f}}$ then,

$$\mathbf{0 = WL - \frac{F_f\, a}{f} + F_f b = WL - F_f\left(\left(\frac{a}{f}\right) - b\right)}$$

Solving for W gives:

$$\boldsymbol{W = \frac{F_f\left(\frac{a}{f} - b\right)}{L}} \quad \ldots\ldots\ldots\ldots (22-15)$$

Solving for F_f gives:

$$\boldsymbol{F_f = \frac{WL}{\left(\frac{a}{f} - b\right)}} \quad \ldots\ldots\ldots\ldots\ldots (22-16)$$

$$\boldsymbol{T_f = F_f * \frac{D_d}{2} = \frac{W * L * D_d}{2\left(\frac{a}{f} - b\right)}} \quad \ldots\ldots\ldots\ldots (22-17)$$

Note 1 : for alternate position of pivot, see part (b) & (c) on fig. (22-17) Page 856
Note 2 : see example 22-8 (Page 857)
Note 3 : if the angle of contact is greater than 45°, in such case $T_f = F_f' * D_d/2$
Where f' : Equivalent coefficient of friction

$$\boldsymbol{f' = \frac{4f\,\sin\theta}{2\theta + \sin 2\theta}} \quad ; \quad \textbf{where } \boldsymbol{2\theta} \textbf{ : angle of contact}$$

Example Problem 22-8:

Compute the actuation force required for the short shoe drum brake of the figure above to produce a friction torque of 67.8 N.m. Use a drum diameter of 254 mm, a = 76.2 mm, and L =381 mm. Consider values of (f) of 0.25, 0.50, and 0.75, and different points of location of pivot A such that b ranges from 0 to 152.4 mm.

Solution:

The required friction force can be found from Equation (22-17):

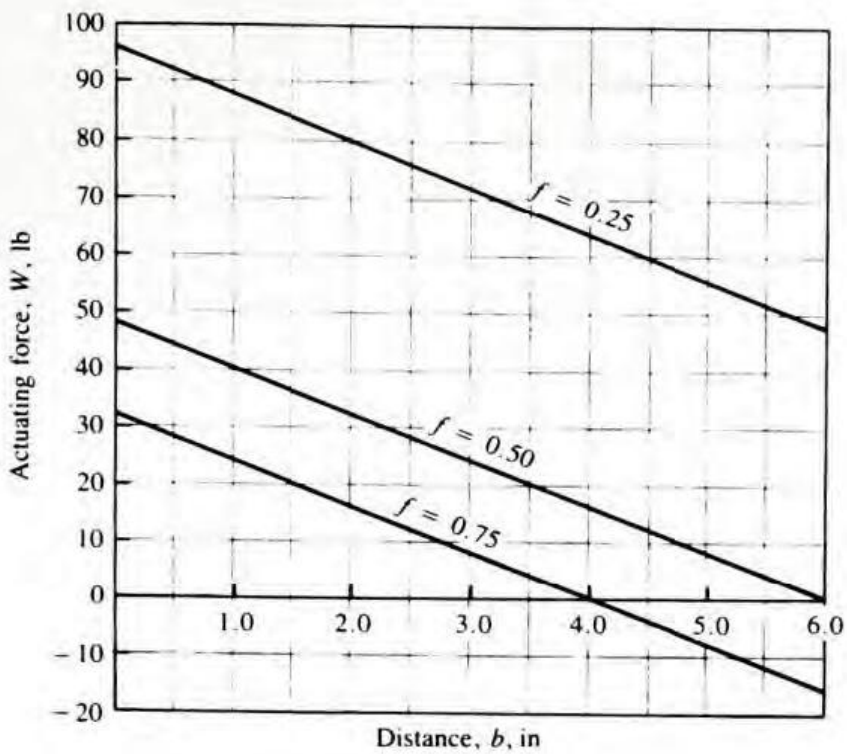
$$\boldsymbol{F_f = \frac{2T_f}{D_d} = \frac{(2)(67.8\ N.m)}{0.254\ m} = 534\ N}$$

In Equation (22-15), we can substitute for a, L, and F_f.

$$\boldsymbol{W = \frac{F_f\left(\frac{a}{f} - b\right)}{L} = \frac{534\ N * \left(\frac{76.2\ mm}{f} - b\right)}{381\ mm} = 1.402 N/mm\left(\frac{76.2\ mm}{f} - b\right) N}$$

We can substitute the varying values of f and b into this last equation to compute the data for the curves of Figure 22-18, showing the actuating force versus the distance b for different values of (f) , Note that for some combinations, the value of W is *negative.* This means that the brake is *self-actuating* and that an upward force on the lever would be required to release the brake.

FIGURE 22–18
Results: actuating load force vs. distance b

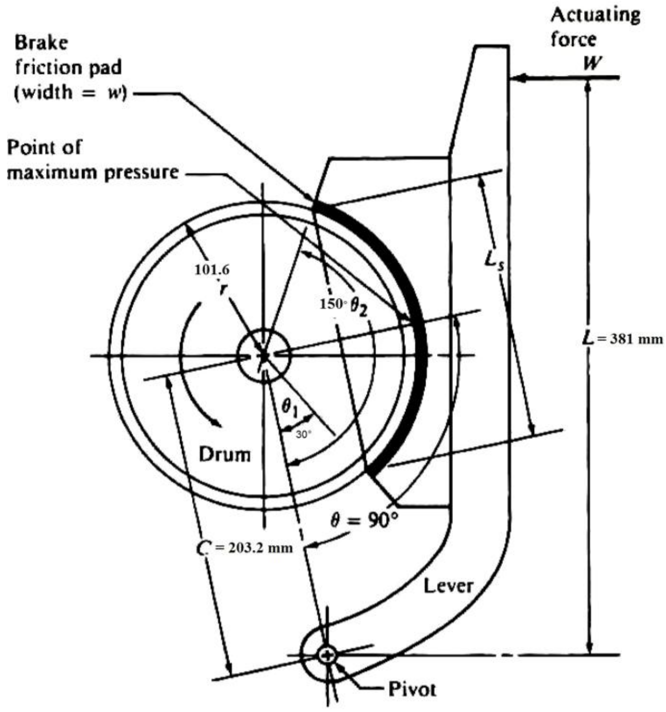


Example:

f = 0.25, P_{max} = 496.6 KPa

T_f = 84.75 N.m , ω= 12.564 rad/sec

Design shoe drum brake?



$$f' = \frac{4f\ sin\theta}{2\theta + sin2\theta} = \frac{4*0.25\ sin60}{120*\pi/180 + sin120}$$

f' = 0.293

$T_f = F_f * r$

$F_f = 84750/101.6 = 834.2\ N$

$F_f = f' * N$

$N = 834.2 / 0.293 = 2847\ N$

$W * L + F_f * r = N * C \longrightarrow W = (2847*203.2 - 834.2*101.6) / 381 = 1296\ N$

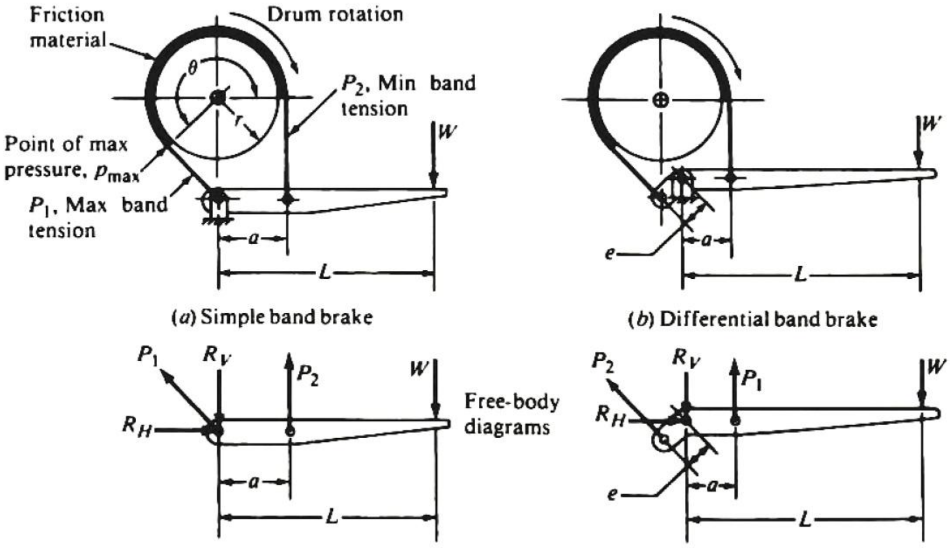
$A = (2r\ sin\ \theta) * w$

$P_{max} = N/A = 2847 / (2*101.6*w*sin60) \longrightarrow w = 32.6\ mm$

Note: you can use equations from 22-18 to 22-24 for solving the example (page 858).

Band brakes (section 22-15), (Page 860):

FIGURE 22–20
Band brake design



(*a*) Simple band brake
(*b*) Differential band brake

$$T_f = (P_1 - P_2) * r \quad \ldots\ldots\ldots (22-25), page\ 860$$

$$\frac{P_1}{P_2} = e^{f\theta} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (22-26)$$

$$P_1 = P_{max} * r * w \quad \ldots\ldots\ldots (22-27)$$

$$W = \frac{P_2 * a}{L} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (22-28)\ for\ fig.(a)$$

$$W = \frac{P_2 * a - P_1 * e}{L} \ldots\ldots\ldots\ldots (22-28)\ for\ fig.(b)$$

Example Problem (22-10):

Design a band brake to exert a braking torque of 81.36 N.m, while slowing the drum from120 rpm (12.564 rad/sec).

Solution

1- Select a material and specify a design value for the maximum pressure. A woven friction material is desirable to facilitate the conformance to the cylindrical drum shape. Let's use p_{max} = 172.4*10^3 N/m^2 and a design value of *f* = 0.25. See Section 22-10.
2- Specify trial geometry: r, θ, w: For this problem, let's try r = 0.152 m, θ =225°, and w=0.051 m. Note that 225°= 3.93 rad.
3- Compute the maximum band tension. P_1, from equation (22-27)

$$P_1 = P_{max} * r * w = (172 * 10^3\ N/m^2)(0.152\ m)(0.051\ m) = 1333\ N$$

4- Compute tension P_2 from Equation (22-26):

$$\boldsymbol{P_2 = \frac{P_1}{e^{f\theta}} = \frac{1333\ \text{N}}{e^{(0.25*3.93)}} = 498\ N}$$

5- Compute the friction torque, $\mathbf{T}_f$:

$$\boldsymbol{T_f = (P_1 - P_2) * r = (1333\ N - 498\ N)(152.4\ mm) = 127\ N.m}$$

Note: Repeat Steps 2-5 until you achieve a satisfactory geometry and friction torque. Let's try a smaller design, say, r =0.127 m:

$$\boldsymbol{P_1 = (172.4 * 10^3\ N/m^2)(0.127\ m)(0.051\ m) = 1110\ N}$$

$$\boldsymbol{P_2 = \frac{1110\ \text{N}}{e^{(0.25*3.93)}} = 417\ N}$$

$$\boldsymbol{T_f = (1110\ N - 417\ N)(0.127\ m) = 88\ N.m} \quad \text{(Okay)}$$

6- Specify the geometry of the lever, and compute the required actuation force. Let's use a = 0.127 m and L = 0.381 m. Then:

$$\boldsymbol{W = \frac{P_2 * a}{L} = (417\ N) * \frac{0.127\ m}{0.381\ m} = 139\ N}$$

7- Compute the average wear ratio from WR= P_f/A:

$$\boldsymbol{A = 2\pi r w\left(\frac{\theta}{360}\right) = 2\pi * (0.127\ m) * (0.051\ m)\left(\frac{225}{360}\right) = 0.025\ m^2}$$

$$\boldsymbol{P_f = T_f * \omega = (88\ N.m)(12.564\ rad/sec) = 1110\ W}$$

$$\boldsymbol{WR = \frac{P_f}{A} = \frac{1110\ W}{0.025\ m^2} = 44.4\frac{W}{m^2}}$$

This should be conservative for average service.

Tedata

Program : MDESIGN	User :	Customer :
Version : 1.1.2	Date : 19.12.2013	Proj. Nr :

Plate-Type Clutch or Brake

Input data:

Plate-Type Clutch or Brake

Braking torque	T =	33.9	N.m
Normal force	N =	1423.424	N
Coefficient of friction	f =	0.25	
Rotational speed	n =	750	rpm
Reasonable value for the ratio Ro/Ri	rv =	1.5	

Results

Required mean radius	Rm	=	95.250	mm
Inside radius	Ri	=	76.200	mm
Outside radius	Ro	=	114.300	mm
Area of the friction surface	A	=	22796.182	mm^2
Frictional power absorbed	Pf	=	2.663	kW
Wear ratio	WR	=	0.101	hp/in^2

Tedata

Program : MDESIGN	User :	Customer :
Version : 1.1.2	Date : 19.12.2013	Proj. Nr :

Cone Clutch or Brake

Input data:

Cone Clutch or Brake

Braking torque	T =	5.65	N.m
Mean radius	Rm =	127	mm
Coefficient of friction	f =	0.25	
Cone angle	α =	10	°

Results

Axial force required for a cone brake	Fa	=	74.703	N

Tedata

Program : MDESIGN	User :	Customer :
Version : 1.1.2	Date : 19.12.2013	Proj. Nr :

Short Shoe Drum Brakes

Input data:

Short Shoe Drum Brakes

Friction torque	T = 5.65	N.m
Drum diameter	D = 254	mm
Pivot to pin centerline dimension	a = 76.2	mm
Pivot to contact dimension	b = 114.3	mm
Length	L = 381	mm
Coefficient of friction	f = 0.25	

Results

Friction force	F	=	44.482	N
Actuation force required for the short shoe drum brake	W	=	22.241	N

Tedata

Program : MDESIGN	User :	Customer :
Version : 1.1.2	Date : 19.12.2013	Proj. Nr :

Long Shoe Drum Brake

Input data:

Long Shoe Drum Brake

Friction torque	T =	84.75	N.m
Rotational speed	n =	120	rpm
Approximate maximum pressure	Pmax =	0.517107	N/mm²
Radius	r =	101.6	mm
Parameter	C =	203.2	mm
Length	L =	381	mm
Angle	θ1 =	30	°
Angle	θ2 =	150	°
Coefficient of friction	f =	0.25	

Results

Required width	w	=	36.662	mm
Let's specify	w	=	38.100	mm
Maximum pressure	pmax	=	0.498	N/mm²
Moment of the normal force	Mn	=	579.419	N.m
Moment of the friction force	Mf	=	84.750	N.m
Required actuation force	W	=	1298.162	N
Frictional power	Pf	=	1.065	kW
Projected area	A	=	6703.037	mm²
Wear ratio	WR	=	0.137	hp/in²

Tedata

Program : MDESIGN	User :	Customer :
Version : 1.1.2	Date : 19.12.2013	Proj. Nr :

Band Brakes

Input data:

Band Brakes

Braking torque	T =	81.36	N.m
Rotationall speed	n =	120	rpm
Maximum pressure	pmax =	0.172369	N/mm²
Coefficient of friction	f =	0.25	
Trial geometry			
Radius	r =	152.4	mm
Angle	θ =	225	°
Band width	w =	50.8	mm
Band distance	a =	127	mm
Length	L =	381	mm

Results

Maximum band tension	P1	=	1334.460	N
Tension	P2	=	499.963	N
Friction torque	Tf	=	127.195	N.m
Required actuation force	W	=	166.654	N
Friction area	A	=	30394.909	mm²
Friction power	Pf	=	1.599	kW
Average wear ratio	WR	=	0.045	hp/in²