Drum Brakes

Among the various types of devices to be studied, based on their practical use, the discussion will be limited to "Drum brakes" of the following types which are mainly used in automotive vehicles and cranes and elevators.

Drum Brake Types:

• Rim types with internal expanding shoes

• Rim types with external contracting shoes

Internal expanding Shoe

The rim type internal expanding shoe is widely used for braking systems in automotive applications and is generally referred as internal shoe drum brake. The basic approach applied for its analysis is known as long-rigid shoe brake analysis.

Long –rigid Shoe Analysis

• A schematic sketch of a single shoe located inside a rotating drum with relevant notations, is shown in the figure below. In this analysis, the pressure at any point is assumed to be proportional to the vertical distance from the hinge pin, the vertical distance from the hinge pin, which in this case is proportional to sine of the angle and thus,

$$p \propto d \sin\theta \propto \sin\theta$$

Since the distance d is constant, the normal pressure at any point is just proportional to sinΘ. Call this constant of proportionality as K

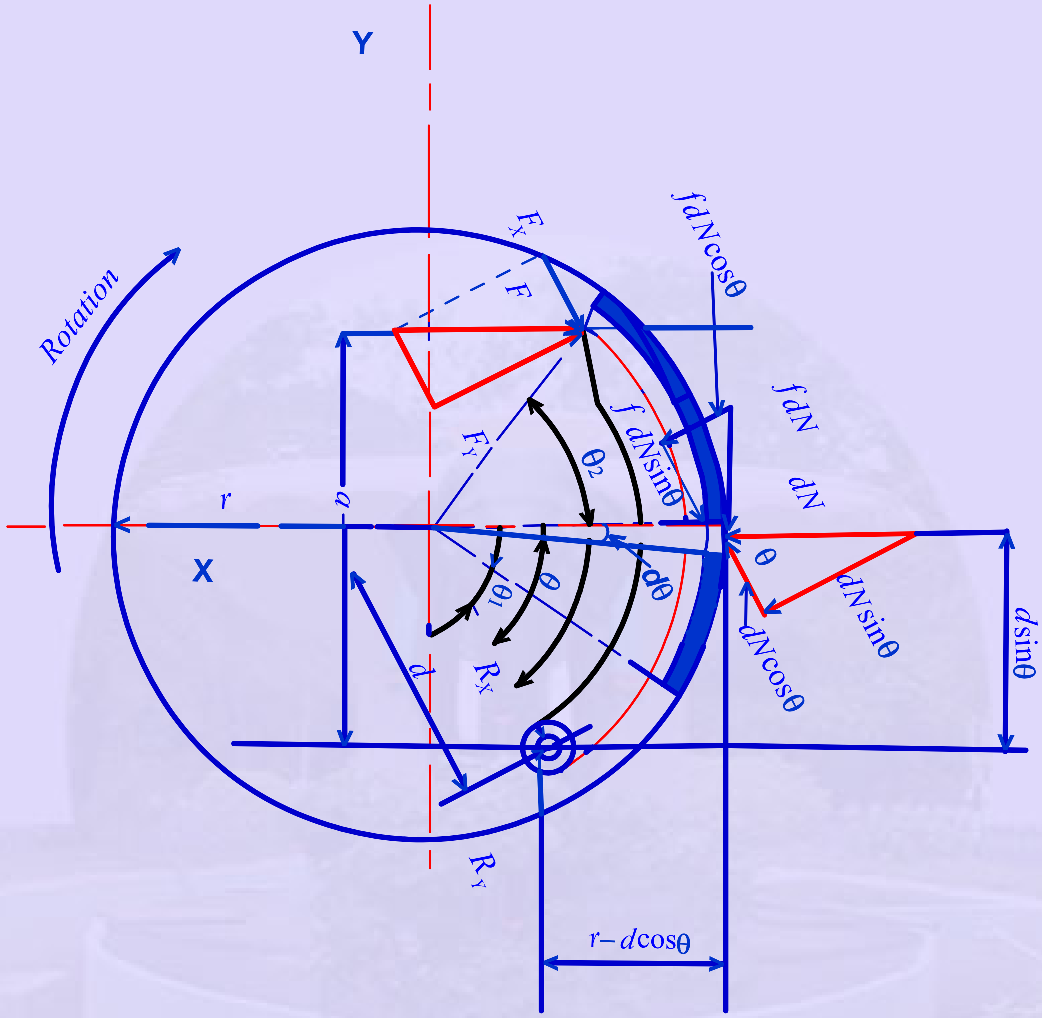


Figure 3.1.4

Thus $$p = K \sin\theta$$

It the maximum allowable pressure for the lining material is p_{max} *then the constant K can be defined as*

$$K = \frac{p}{\sin\theta} = \frac{p_{max}}{\sin\theta_{max}}$$

$$p = \frac{p_{max}}{\sin\theta_{max}}\sin\theta$$

• The normal force dN is computed as the product of pressure and area and the frictional force as the product of normal force and frictional coefficient i.e. f dN.• By integrating these over the shoe length in terms of its angle the braking torque T, and other brake parameters are computed.

To determine the actuating force F, the moment equilibrium about the pivot point is applied. For this we need to determine the moment of the normal force M_N and moment of the frictional force about the pivot point. Moment of the normal force is equal to the normal force times its moment arm about the pivot point. From the figure it is clear that the moment arm in this case is equal to d sin Θ where d is the distance between the drum center and pivot center

$$M_N = \int_{\theta_2}^{\theta_1} p.b.r.d\theta.d\sin\theta = \int_{\theta_2}^{\theta_1} b.p.r.d.\sin\theta.d\theta$$

$$= \int_{\theta_2}^{\theta_1} b.d.r.\frac{p_{max}}{\sin\theta}\sin^2\theta d\theta$$

$$M_N = \frac{p_{max} b.d.r}{\sin\theta_a}\left[\frac{1}{2}\left(\theta_2 - \theta_1\right) - \frac{1}{4}(\sin 2\theta_2 - \sin 2\theta_1)\right]$$

On similar lines the moment of friction force is computed

$$M_F = \int_{\theta_2}^{\theta_1} f.p.b.r.d\theta\left(r - d\sin\theta\right)$$

$$= \int_{\theta_2}^{\theta_1} f.b.r.\frac{p_{max}}{\sin\theta_{max}}\sin\theta\left(r - d\sin\theta\right)d\theta$$

$$M_f = \frac{f.p_{max}.b.r}{\sin\theta_a}\left[-r\left(\cos\theta_2 - \cos\theta_1\right) - \frac{d}{2}\left(\sin^2\theta_2 - \sin^2\theta_1\right)\right]$$

The actuating force F is determined by the summation of the moments of normal and frictional forces about the hinge pin and equating it to zero.

Summing the moment about point O gives

$$F = \frac{M_N \pm M_f}{c}$$

where,

• M_N and M_f are the moment of the normal and frictional forces respectively, about the shoe pivot point.

The sign depends upon the direction of drum rotation,

(- sign for self energizing and + sign for non self energizing shoe)*Where the lower sigh is for a self energizing shoe and the upper one for a self deenergizing shoe.*

The reaction forces are determined by applying force summation and equilibrium

$$R_x = \int dN.\cos\theta + \int dF.\sin\theta$$

$$= \int_{\theta_1}^{\theta_2} b.r.p\cos\theta d\theta + f\int_{\theta_1}^{\theta_2} b.r.p\sin\theta d\theta$$

$$= \int_{\theta_1}^{\theta_2} b.r.\frac{p_{max}}{\sin\theta_{max}}\sin\theta\cos\theta d\theta + f\int_{\theta_1}^{\theta_2} b.r.\frac{p_{max}}{\sin\theta_{max}}\sin^2\theta d\theta$$

$$= \frac{p_{max}.b\,r}{\sin\theta_{max}}\left(\frac{1}{2}\left[\frac{1}{2}(\theta_2 - \theta_1) - \frac{1}{4}\left(\sin 2\theta_2 - \sin 2\theta_1\right)\right] \pm f\,\frac{1}{2}\left(\sin^2\theta_2 - \sin^2\theta_1\right)\right)$$

The equations can be simplified and put as

$$R_x = \frac{p_a br}{\sin\theta_a}(A \mp fB)$$

$$R_y = \frac{p_a br}{\sin\theta_a}(B \pm fA) - F_x$$

Where

$$A = \frac{1}{2}\left(\sin^2\theta_2 - \sin^2\theta_1\right)$$

$$B = \frac{1}{2}\left[\frac{1}{2}(\theta_2 - \theta_1) - \frac{1}{4}\left(\sin 2\theta_2 - \sin 2\theta_1\right)\right]$$

The braking torque T on the drum by the shoe is of the frictional forces f.dN times the radius of the drum and resulting equation is,

$$T = \int_{\theta_1}^{\theta_1} f.b.p.r.d\theta.r$$

$$= \int_{\theta_1}^{\theta_1} fbr^2 \frac{p_{max}}{\sin\theta_{max}} \sin\theta d\theta$$

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

T

Double Shoe Brakes Twin Shoe Brakes

Behavior of a single shoe has been discussed at length. Two such shoes are combined into a complete practical brake unit, two being used to cover maximum area and to minimize the unbalanced forces on the drum, shaft and bearings.

- *If both the shoes are arranged such that both are leading shoes in which self energizing are prevailing, then all the other parameters will remain same and the total braking torque on the drum will be twice the value obtained in the analysis.*

- *However in most practical applications the shoes are arranged such that one will be leading and the other will be trailing for a given direction of drum rotation*
- *If the direction of drum rotation changes then the leading shoe will become trailing and vice versa.*
- *Thus this type of arrangement will be equally effective for either direction of drum rotation. Further the shoes can be operated upon using a single cam or hydraulic cylinder thus provide for ease of operation*

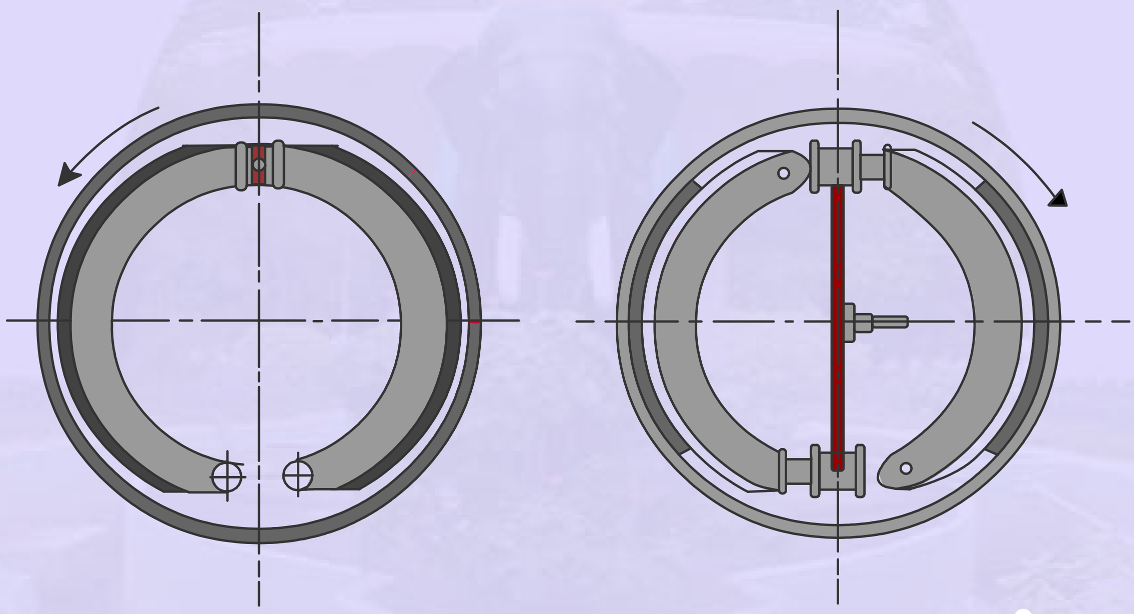


Figure 3.1.5

However the total braking torque will not be the twice the value of a single shoe, if the same normal force is applied or created at the point of force application on both the brake shoes which is the normal practice as they are actuated using a common cam or hydraulic cylinder.

• *This is because the effective contact pressure (force) on the trailing shoe will not be the same, as the moment of the friction force opposes the normal force, there by reducing its actual value as in most applications the same normal force is applied or created at the point of force application on the brake shoe as noted above*

• *Consequently we may write the actual or effective pressure prevailing on a trailing shoe*

$$p_a' = p_a \cdot \left[\frac{F.a}{(M_n + M_f)} \right]$$

Resulting equation for the braking torque

$$T_B = f.w.r^2 . \frac{p_a}{\sin\theta_a} (\cos\theta_1 - \cos\theta_2)(p_a + p_a')$$

Some pictorial illustrations of the automotive drum brakes are presented below

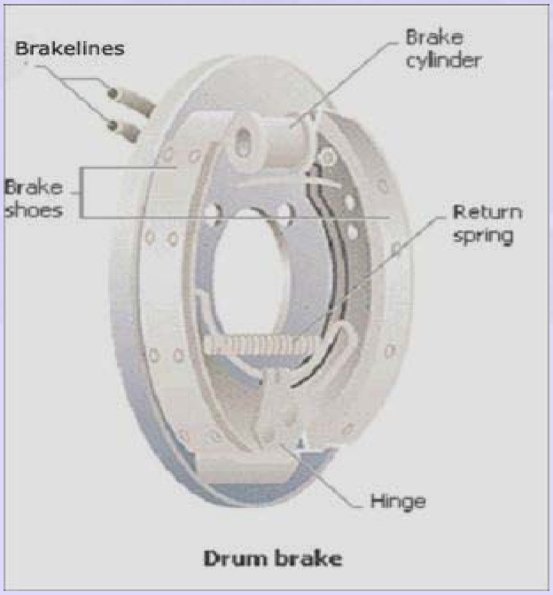


Figure 3.1.6

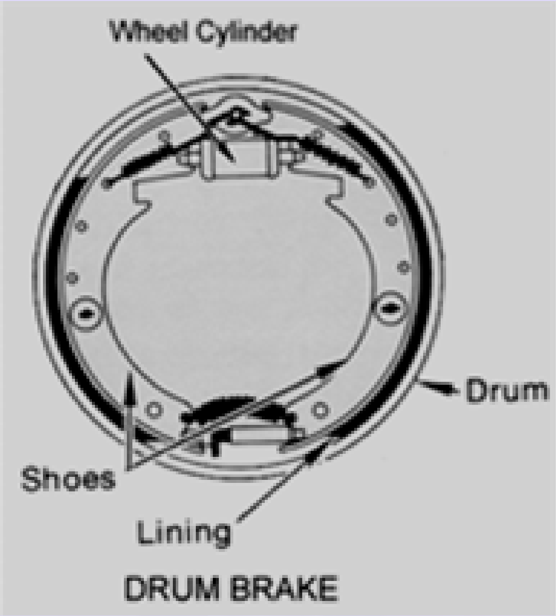
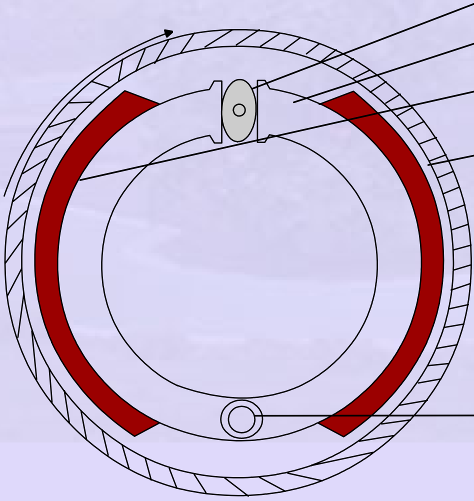


Figure 3.1.7



The anatomy of the single leading shoe drum Brake

Animation

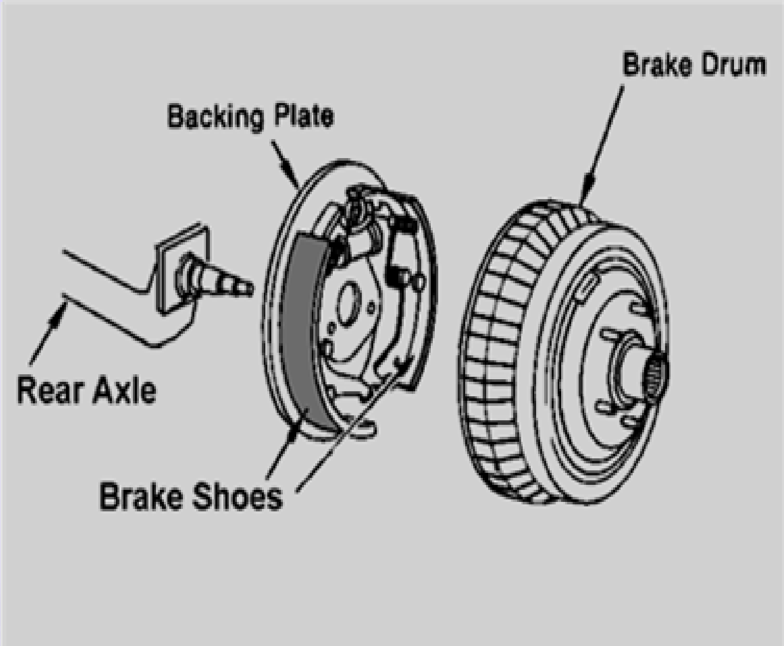


Figure 3.1.9

External Contracting Shoe

• The same analysis can be extended to a drum brake with external contracting type of shoes, typically used in elevators and cranes.

A schematic sketch of as single shoe located external to the rotating drum is with all relevant notations is shown in the figure below.

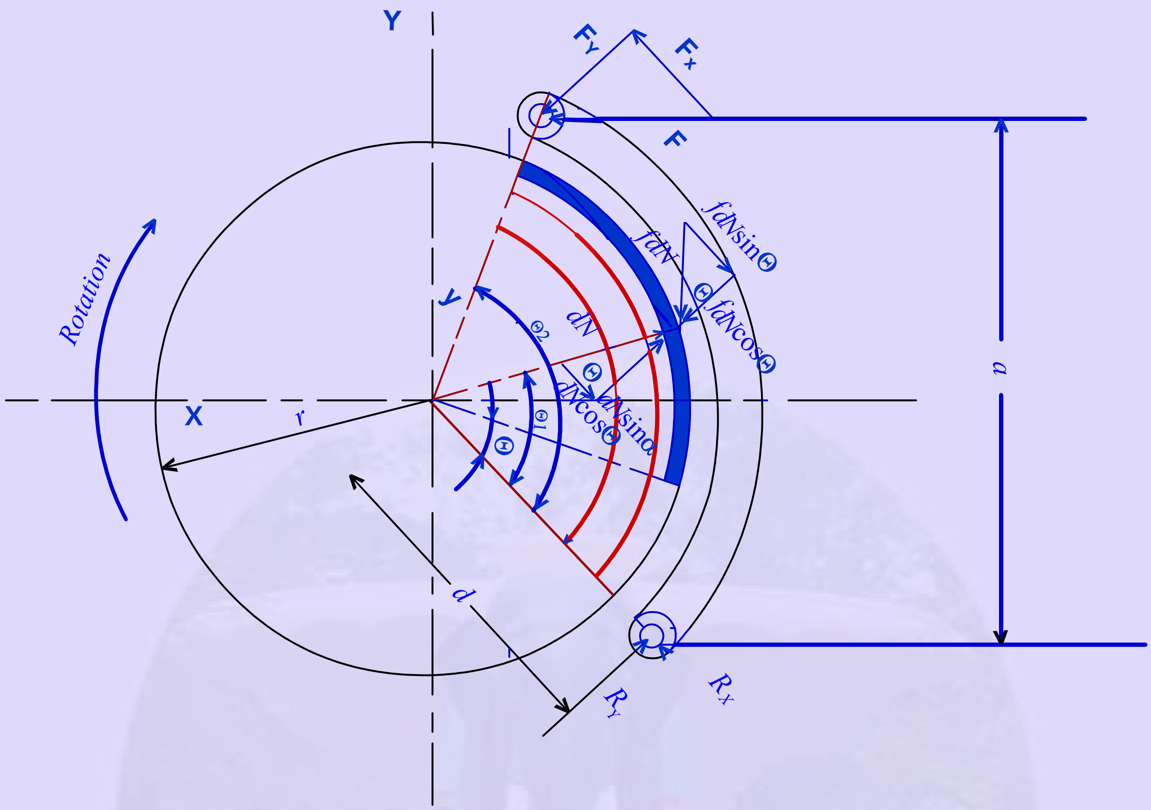


Figure 3.1.11

- Corresponding contact geometry is shown in the figure
- The resulting equations for moment of normal and frictional force as well as the actuating force and braking torque are same as seen earlier.
- For convenience they are reproduced here again

$$T = \frac{fbp_a r^2 \left(\cos\theta_1 - \cos\theta_2\right)}{\sin\theta_a}$$

$$F = \frac{M_N \pm M_f}{c}$$

$$M_N = \frac{p_a bra}{\sin\theta_a}\left[\frac{1}{2}\left(\theta_2 - \theta_1\right) - \frac{1}{4}(\sin 2\theta_2 - \sin 2\theta_1)\right]$$

$$M_f = \frac{f p_a b r}{\sin\theta_a}\left[r\left(\cos\theta_1 - \cos\theta_2\right) - \frac{d}{2}(\sin^2\theta_2 - \sin^2\theta_1)\right]$$

TWIN SHOE BRAKES

As noted earlier for the internal expanding shoes, for the double shoe brake the braking torque for one leading and one trailing shoe acted upon a common cam or actuating force the torque equation developed earlier can be applied.

i.e $$T_B = f.w.r^2.\frac{p_a}{\sin\theta_a}(\cos\theta_1 - \cos\theta_2)(p_a + p_a')$$