Machine Design

Course No. MEC3110

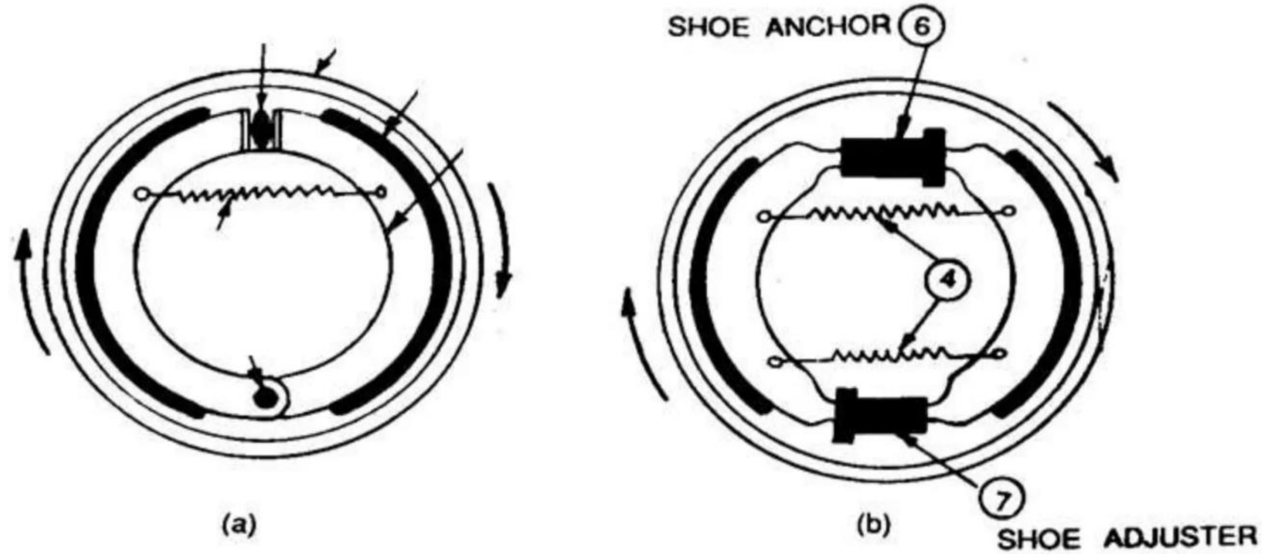
Drum Brakes

- The first proper brakes on cars were the drum brakes, invented by French manufacturer Louis Renault in 1902.
- A drum brake consists of a drum, a housing connected to the wheel, brake shoes, which are fitted in the housing and a master cylinder that is connected to the brake pedal.
- When the brake pedal is pushed the master cylinder causes the **brake shoes** to rub against the inside of the rotating drum, creating friction and ultimately slowing down the wheel to a complete stop.
- Drum brakes are used on many mass produced cars and on the rear wheels on more premium car models.

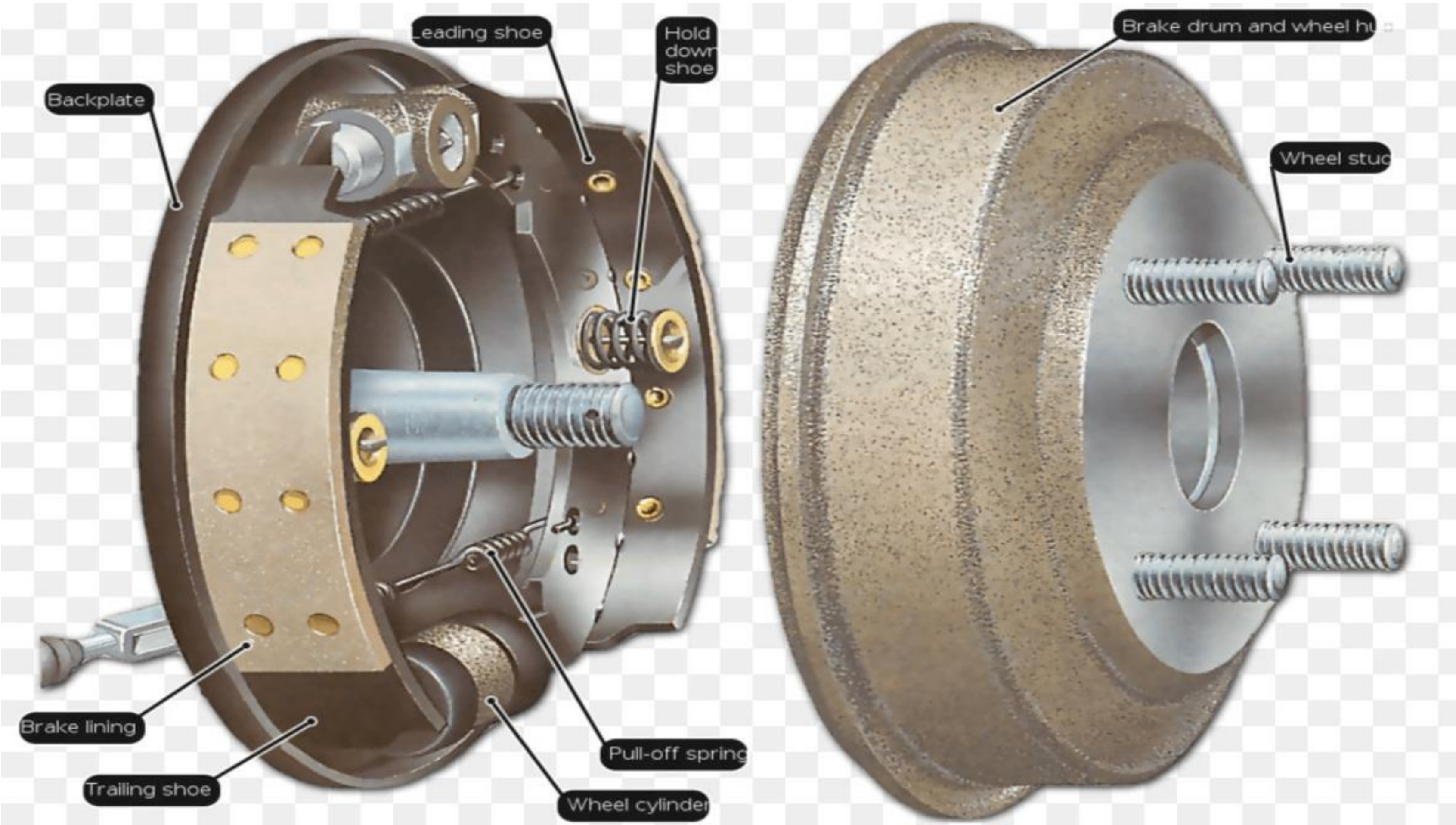
Drum Brakes

The main components of drum brakes are

1. Brake drum
2. Back plate
3. Brake shoes
4. Brake Liners
5. Retaining Springs
6. Cam
7. Brake Linkages



Drum Brakes


Leading shoe
Hold down shoe
Backplate
Brake lining
Trailing shoe
Pull-off spring
Wheel cylinder
Brake drum and wheel hu
Wheel stud

Drum Brakes (Construction)

- In this system the wheel is attached to drum.

- There are brake shoes used to contact the rotating drum for braking operation.

- The shoes provide lining on their outer surface.

- The cam is used to lift the brake shoes at one end, other end is connected by some method so as to make as the brake sleeve come into contact in the brake drum.

- The retaining spring is provided for bringing the brake shoes back to its original position, after releasing the brake pedal.

- All these parts are fitted in the back plate and enclosed with brake drum.

Drum Brakes (Working)

- When the pedal is pressed the cam moves the shoes outwards through linkages, there by coming in frictional contact with the rotating drum.

- As soon as the brake pedal is released the retaining springs help the brake shoes to brought back and release the brakes.

Drum Brakes

Advantages

- **Durability:** Since drum brakes have an increased friction contact area, they last longer.
- **Low Cost:** Drum brakes are cheaper to manufacture, hence they are more widely used on almost all types of vehicles.
- **Low Input Force:** Some drum brakes require low input force to get them activated, meaning less push on the brake pedal, this can be done through various means like hydraulic pressure.
- **Low maintenance:** Due to better corrosion resistance, as they are inside of a housing, they are slightly easier to maintain.

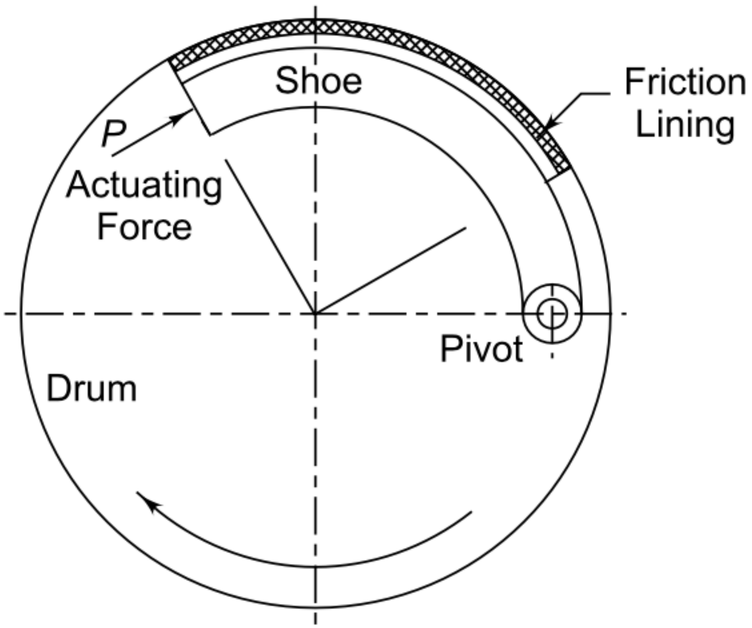
Drum Brakes

Disadvantages

- **Overheat:** Drum brakes tend to overheat in high braking conditions like going down a hill or frequent high speed braking, as they don't have an outlet to cool themselves down. This causes the brake shoes to glaze or smoothen out and produce no braking action.
- **Pushing The Brake Pedal Farther:** When hard braking occurs, the drum expands slightly due to a phenomenon called thermal expansion. So, to induce braking the driver must press the brake pedal farther.
- **Grabbing:** Grabbing in a drum brake occurs when the brake shoe becomes wet or is slightly rusty. This causes it to offer more braking force than required and stays that way for a brief moment of time. When a grab occurs, the tyres skid and keep on skidding even when the pedal is released. This results in a loss of control of the car and is dangerous for the driver, especially at high speeds.
- **Complex construction:** A drum brake has many components, requiring an expert mechanic for repairs.

Drum Brakes Design

- It consists of a shoe, which is pivoted at one end and subjected to an actuating force *P* at the other end.
- A friction lining is fixed on the shoe and the complete assembly of shoe, lining and pivot is placed inside the brake drum.
- Internal shoe brakes, with two symmetrical shoes, are used on all automobile vehicles.
- The actuating force is usually provided by means of a hydraulic cylinder or a cam mechanism.



Internal Expanding Brake

Drum Brakes Design

The analysis of the internal shoe brake is based on the following assumptions:

- The intensity of normal pressure between the friction lining and the brake drum at any point is proportional to its vertical distance from the pivot.
- The brake drum and the shoe are rigid.
- The centrifugal force acting on the shoe is negligible.
- The coefficient of friction is constant.

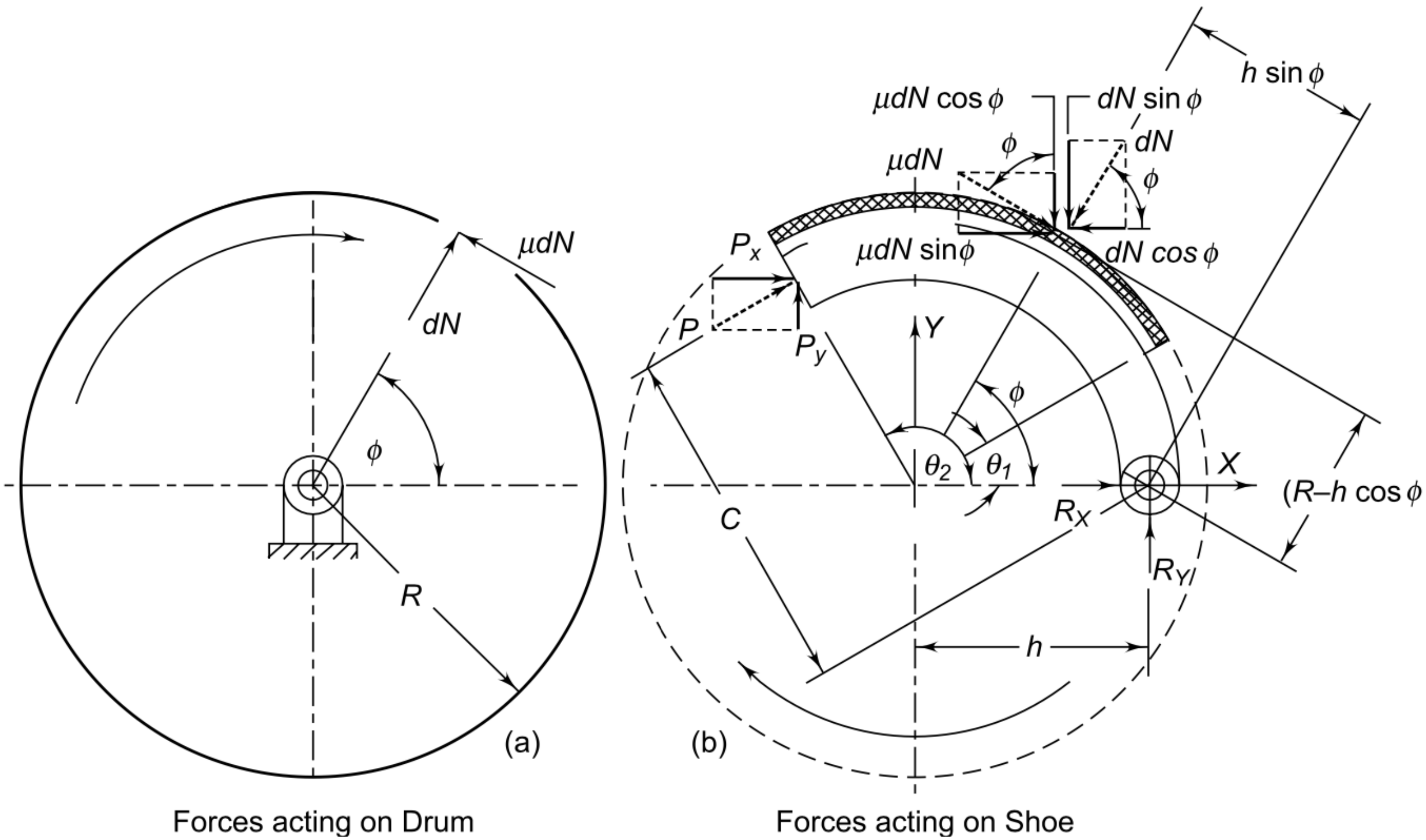
Drum Brakes Design

- The free-body diagram of forces acting on an element on the surface of the drum and the surface of friction lining.
- It should be noted that angles $\emptyset$, θ_1, θ_2 begin with a line drawn from the centre of the drum to the centre of the pivot of the shoe.
- The friction material begins at an angle θ_1 from this line and ends at an angle θ_2.

Consider an elemental area on the friction lining located at an angle $\emptyset$ and subtending an angle $d\emptyset$. The elemental area will be ($Rd\emptyset\ w$) where w is the width of the friction lining parallel to the axis of the brake drum. If p is the intensity of normal pressure on this elemental area, the normal reaction dN is given by,

$$dN = pR\ wd\emptyset$$

Drum Brakes Design



Free-body Diagram of Forces

Drum Brakes Design

As mentioned in the first assumption, the normal pressure p is proportional to the vertical distance ($R \sin \phi$) of the element from the pivot. Therefore,

$$p \propto \sin \phi \quad \text{or} \quad p = C_1 \sin \phi \qquad \text{(b)}$$

Assuming $p = p_{max.}$ when $\phi = \phi_{max.}$ we have,

$$p_{max.} = C_1 \sin \phi_{max.} \qquad \text{(c)}$$

From Eqs (b) and (c),

$$p = \frac{p_{max.} \sin \phi}{\sin \phi_{max.}} \qquad (12.16)$$

It can be seen from Eq. (b) that

$\phi_{max.} = 90°$ when $\theta_2 > 90°$

$\phi_{max.} = \theta_2$ when $\theta_2 < 90°$

Substituting Eq. (12.16) in Eq. (a),

$$dN = \frac{p_{max.} Rw}{\sin \phi_{max.}} \sin \phi \, d\phi \qquad \text{(d)}$$

The moment M_f of the frictional force (μdN) about the pivot point is given by

$$M_f = \int \mu dN (R - h \cos \phi)$$

Substituting Eq. (d),

$$M_f = \frac{\mu p_{max.} Rw}{\sin \phi_{max.}} \int_{\theta_1}^{\theta_2} \sin \phi (R - h \cos \phi) d\phi \qquad (12.17)$$

The moment M_n of the normal force (dN) about the pivot point is given by

$$M_n = \int dN (h \sin \phi)$$

Substituting Eq. (d),

$$M_n = \frac{p_{max.} Rwh}{\sin \phi_{max.}} \int_{\theta_1}^{\theta_2} \sin^2 \phi \, d\phi \qquad (12.18)$$

$$\int_{\theta_1}^{\theta_2} \sin \phi (R - h \cos \phi) d\phi$$

$$= R \int_{\theta_1}^{\theta_2} \sin \phi \, d\phi - h \int_{\theta_1}^{\theta_2} \left(\frac{\sin 2\phi}{2} \right) d\phi$$

$$= R[-\cos \phi]_{\theta_1}^{\theta_2} - h \left[-\frac{\cos 2\phi}{4} \right]_{\theta_1}^{\theta_2}$$

$$= \frac{1}{4} [4R(\cos \theta_1 - \cos \theta_2) - h(\cos 2\theta_1 - \cos 2\theta_2)]$$

From Eq. (12.17)

$$M_f = \frac{\mu p_{max.} Rw [4R(\cos \theta_1 - \cos \theta_2) - h(\cos 2\theta_1 - \cos 2\theta_2)]}{4 \sin \phi_{max.}} \qquad (12.19)$$

Drum Brakes Design

From Eq. (12.17)

$$M_f = \frac{\mu p_{max.} Rw[4R(\cos\theta_1 - \cos\theta_2) - h(\cos 2\theta_1 - \cos 2\theta_2)]}{4 \sin \phi_{max.}} \quad (12.19)$$

Similarly,

$$\int_{\theta_1}^{\theta_2} \sin^2\phi \, d\phi = \int_{\theta_1}^{\theta_2} \left(\frac{1-\cos 2\phi}{2}\right) d\phi = \left[\frac{\phi}{2} - \frac{\sin 2\phi}{4}\right]_{\theta_1}^{\theta_2}$$

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

From Eq. (12.18),

$$M_n = \frac{p_{max.} Rwh[2(\theta_2 - \theta_1) - (\sin 2\theta_2 - \sin 2\theta_1)]}{4 \sin \phi_{max.}} \quad (12.20)$$

Referring to Fig. 12.17, the elemental torque due to frictional force (μdN) is (μdNR). Therefore,

$$M_t = \int \mu \, dN R$$

Substituting Eq. (d),

$$M_t = \frac{\mu R^2 p_{max.} w}{\sin \phi_{max.}} \int_{\theta_1}^{\theta_2} \sin\phi \, d\phi$$

or $$M_t = \frac{\mu R^2 p_{max.} w(\cos\theta_1 - \cos\theta_2)}{\sin \phi_{max.}} \quad (12.21)$$

Considering the forces acting on the shoe and taking moments about the pivot,

$$P \times C + M_f - M_n = 0$$

The couple ($P \times C$) is clockwise. The couple due to (dN), i.e., M_n is anticlockwise, while the couple due to ($\mu \, dN$), i.e., M_f is clockwise. Therefore,

$$P = \frac{M_n - M_f}{C} \quad (12.22)$$

The above equation is derived for the clockwise rotation of the brake drum. The direction of the frictional force (μdN) is reversed for the anticlockwise rotation of the brake drum. In that case, the couple due to (μdN), i.e. M_f will be anticlockwise and Eq. (12.22) will be written as

$$P = \frac{M_n + M_f}{C} \quad (12.23)$$

Drum Brakes Design

Let us assume that the vehicle is moving in the forward direction for clockwise rotation of the brake drum. The vehicle will be travelling in 'reverse' for anti-clockwise rotation of brake drum. The following observations are made:

(i) When the vehicle is moving forward (clockwise rotation of the brake drum), the couple due to actuating force ($P \times C$) and the couple due to friction force ($\mu\ dN$), i.e. M_f are both clockwise. Therefore, friction force helps to reduce the actuating force and consequently the force on the brake pedal is reduced. This is self-energizing effect [$P = (M_n - M_f)/C$].

(ii) When the vehicle is moving in 'reverse' (anti-clockwise rotation of the brake drum), the couple due to actuating force ($P \times C$) and the couple due to friction force ($\mu\ dN$), i.e., M_f are opposite. Therefore, friction force tends to increase the actuating force and consequently, the force on the brake pedal is increased [$P = (M_n + M_f)/C$].

(iii) Therefore, braking action when traveling in 'reverse' is not as effective as when traveling 'forward'.

Drum Brakes Design

Example 12.8 *An automotive type internal-expanding double-shoe brake is shown in Fig. 12.18. The face width of the friction lining is 40 mm and the maximum intensity of normal pressure is limited to 1 N/mm². The coefficient of friction is 0.32. The angle θ_1 can be assumed to be zero. Calculate:*

(i) the actuating force P; and

(ii) the torque-absorbing capacity of the brake.

Solution

Given $w = 40$ mm $\quad \mu = 0.32 \quad p_{max.} = 1$ N/mm²
$R = 125$ mm

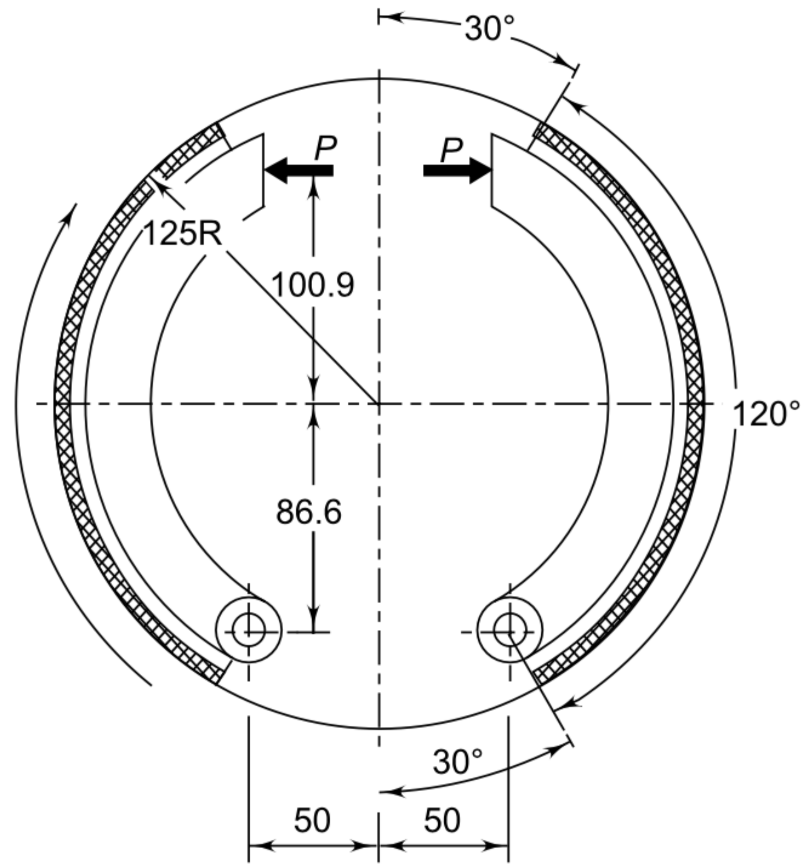
Step I *Actuating force*

It is assumed that the maximum normal pressure will occur between the lining on the right-hand shoe and the brake drum. For the right-hand shoe,

$\theta_1 = 0 \qquad \theta_2 = 120° \qquad \phi_{max.} = 90° \qquad \sin\phi_{max.} = 1$

The distance h of the pivot from the axis of the brake drum is given by

$$h = \sqrt{86.6^2 + 50^2} = 100 \text{ mm}$$



Automotive Double Shoe Brake

Drum Brakes Design

$$M_f = \frac{\mu p_{max.} Rw[4R(\cos\theta_1 - \cos\theta_2) - h(\cos 2\theta_1 - \cos 2\theta_2)]}{4\sin\phi_{max.}}$$

$$= \frac{0.32(1)(125)(40)[4(125)(1-\cos 120°) - 100(1-\cos 240°)]}{4(1)}$$

$= 240\ 000$ N-mm

From Eq. (12.20),

$$M_n = \frac{p_{max.} Rwh[2(\theta_2 - \theta_1) - (\sin 2\theta_2 - \sin 2\theta_1)]}{4\sin\phi_{max.}}$$

$$= \frac{1(125)(40)(100)\left[2\left(\frac{120\pi}{180}\right) - \sin(240°)\right]}{4(1)}$$

$= 631\ 851.95$ N-mm

From Eq. (12.22),

$$P = \frac{M_n - M_f}{C}$$

$$= \frac{631\ 851.95 - 240\ 000}{100.9 + 86.6} = 2089.88 \text{ N} \qquad \text{(i)}$$

Step II *Torque-absorbing capacity*

From Eq. (12.21), the torque $(M_t)_R$ for the right-hand shoe is given by

$$(M_t)_R = \frac{\mu R^2 p_{max.} w(\cos\theta_1 - \cos\theta_2)}{\sin\phi_{max.}}$$

$$= \frac{0.32(125)^2(1)(40)(1-\cos 120°)}{1}$$

$= 300\ 000$ N-mm

The maximum intensity of pressure for the left-hand shoe is unknown. For identical shoes, it can be seen from the expressions of M_n and M_f, that both are proportional to $(p_{max.})$. For left-hand shoe, the maximum intensity of pressure is taken as $(p'_{max.})$. Therefore, for the left-hand shoe,

$$M'_f = \frac{240\ 000\, p'_{max.}}{p_{max.}} = \frac{(240\ 000)\, p'_{max.}}{(1)}$$

$$= 240\ 000\, p'_{max.}$$

Drum Brakes Design

Similarly,

$$M_n' = \frac{631\ 851.95\, p'_{\text{max.}}}{p_{\text{max.}}} = \frac{(631\ 851.95)\, p'_{\text{max.}}}{(1)}$$

$$= 631\ 851.95\, p'_{\text{max.}}$$

For the left-hand shoe,

$$P = \frac{M_n' + M_f'}{C}$$

or $$2089.88 = \frac{(240\ 000 + 631\ 851.95)\, p'_{\text{max.}}}{(100.9 + 86.6)}$$

$\therefore$ $$p_{\text{max.}} = 0.45 \text{ N/mm}^2$$

Since the shoes are identical,

$$(M_t)_L = 300\,000 \left(\frac{p'_{\text{max.}}}{p_{\text{max.}}} \right) = 300\,000 \left(\frac{0.45}{1} \right)$$

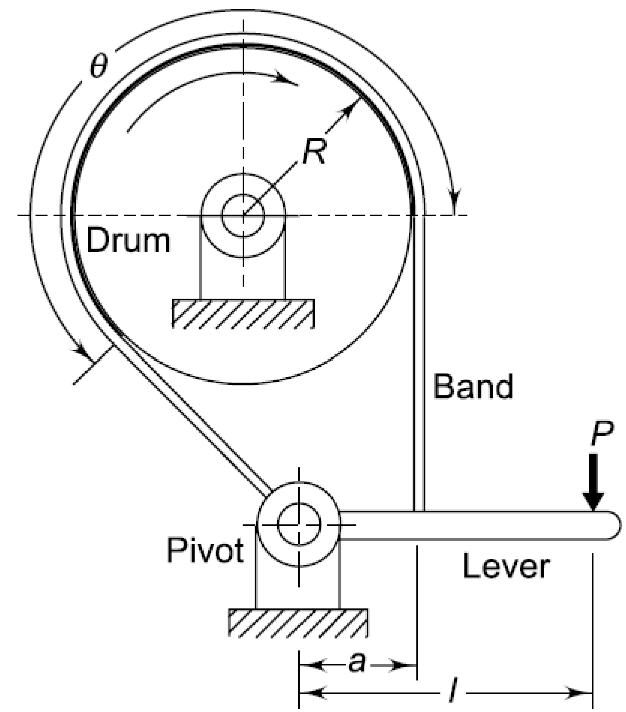
$$= 135\,000 \text{ N-mm}$$

The total torque-absorbing capacity of the brake is given by,

$M_t = 300\ 000 + 135\ 000 = 435\ 000$ N-mm
or 435 N-m (ii)

Band Brakes

- It consists of a flexible steel strip lined with friction material, which is pressed against the rotating brake drum.
- If one end of the steel band passes through the fulcrum of the actuating lever, the brake is called the **simple band brake.**



The ratio of band tensions is given by,

$$\frac{P_1}{P_2} = e^{\mu\theta}$$

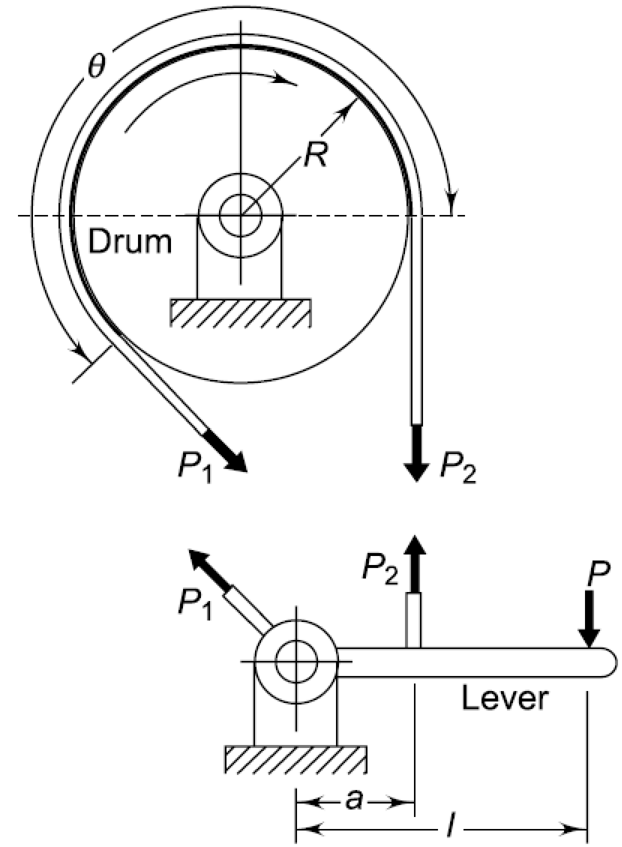
Band Brakes

where,

P_1 = tension on the tight side of the band (N)

P_2 = tension on the loose side of the band (N)

μ = coefficient of friction between the friction lining and the brake drum



θ = angle of wrap (rad)

The torque M_t absorbed by the brake is given by,

$$M_t = (P_1 - P_2)R$$

where,

M_t = torque capacity of the brake (N-mm)

R = radius of the brake drum (mm)

Band Brakes

Advantages

Band brake offers the following advantages:

(i) Band brake has simple construction. It has small number of parts. These features reduce the cost of band brake.

(ii) Most equipment manufacturers can easily produce band brake without requiring specialized facilities like foundry or forging shop. The friction lining is the only part which must be purchased from outside agencies.

(iii) Band brake is more reliable due to small number of parts.

(iv) Band brake requires little maintenance.

Disadvantages

The disadvantages of band brake are as follows:

(i) The heat dissipation capacity of a band brake is poor.

(ii) The wear of friction lining is uneven from one end to the other.

Band brakes are used in applications like bucket conveyors, hoists and chain saws. They are more popular as back-stop devices.

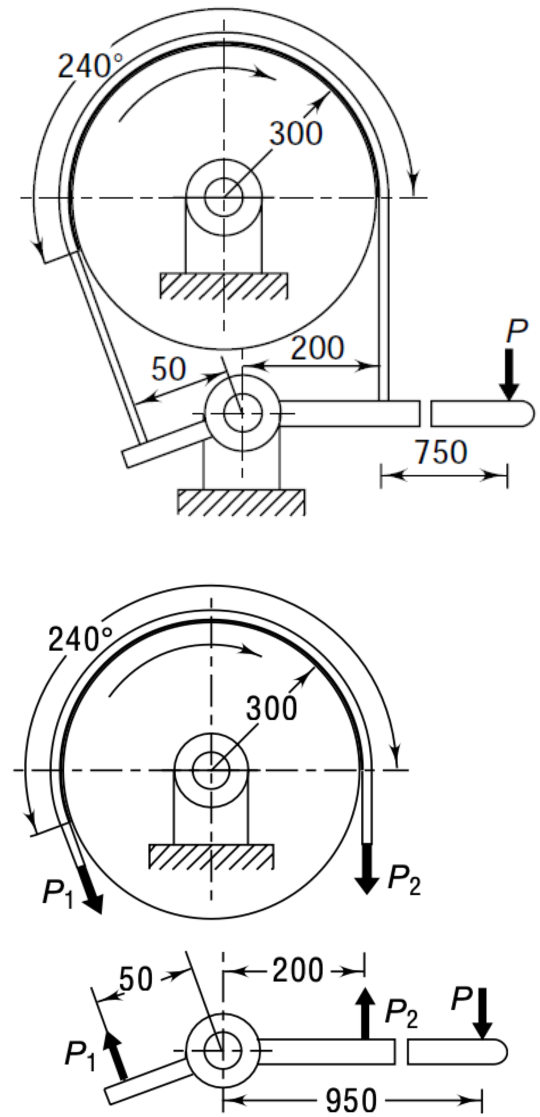
Materials for Brake Lining

1. Low wear rate
2. High Heat Resistance
3. High Heat Dissipation capacity.
4. Low coefficient of thermal expansion.
5. It should have adequate mechanical strength.
6. It should not be affected by moisture and air.
7. It should have high coefficient of friction with minimum fading.

Band Brakes

Example 12.10 *A differential band brake is shown in Fig. 12.24(a). The width and the thickness of the steel band are 100 mm and 3 mm respectively and the maximum tensile stress in the band is 50 N/mm^2. The coefficient of friction between the friction lining and the brake drum is 0.25. Calculate:*

(i) the tensions in the band;
(ii) the actuating force; and
(iii) the torque capacity of the brake.
Find out whether the brake is self-locking.



(b)

Band Brakes

Solution

Given $w = 100$ mm $t = 3$ mm $\mu = 0.25$

$\sigma_t = 50$ N/mm^2 $R = 300$ mm = 0.3 m

Step I *Tensions in band*

The maximum tension in the band is P_1.

$$P_1 = \sigma_t wt = 50(100)3 = 15\,000 \text{ N}$$

$$\frac{P_1}{P_2} = e^{\mu\theta} = e^{\left\{\frac{(0.25 \times 240)\pi}{180}\right\}} = 2.85$$

$$\therefore \quad P_2 = \frac{P_1}{2.85} = \frac{15\,000}{2.85} = 5263 \text{ N} \qquad \text{(i)}$$

Step II *Actuating force*

The free-body diagram of forces acting on the band and the actuating lever is shown in Fig. 12.24(b). Taking moment of forces about the fulcrum,

$$P_1(50) + P(950) - P_2(200) = 0$$

$$(15\,000)(50) + P(950) - (5263)(200) = 0$$

$$\text{or} \quad P = 318.5 \text{ N} \qquad \text{(ii)}$$

Step III *Torque capacity of brake*

$$M_t = (P_1 - P_2)R = (15\,000 - 5263)(0.3)$$

$$= 2921.1 \text{ N-m} \qquad \text{(iii)}$$

Step IV *Self-locking property*

Since $\left(\frac{a}{b}\right) = \frac{200}{50} = 4$ and $e^{\mu\theta} = 2.85$

$$\therefore \quad \left(\frac{a}{b}\right) > e^{\mu\theta}$$

The brake is not self-locking.