Brake with a pivoted long shoe

When the shoe is rigidly fixed to the lever, the tendency of the frictional force ($f.F_n$) is to unseat the block with respect to the lever. This is eliminated in the case of pivoted or hinged shoe brake and it also provides some additional advantages.

Long Hinged Shoe

In a hinged shoe brake - the shoes are not rigidly fixed but hinged or pivoted to the posts. The hinged shoe is connected to the actuating post by the hinge, G, which introduces another degree of freedom - so the shoe tends to assume an optimum position in which the pressure distribution over it is less peaked than in a rigid shoe.

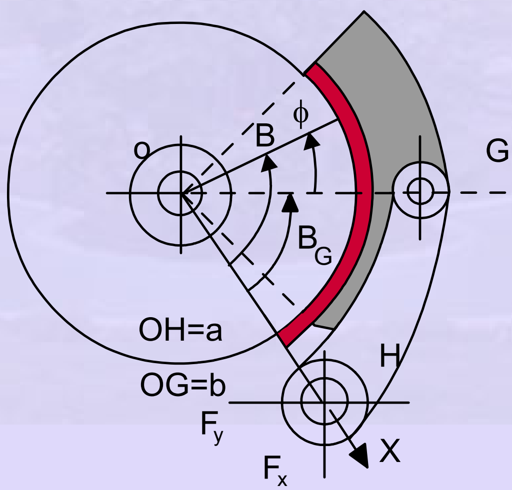


Figure 3.1.12

As wear proceeds the extra degree of freedom allows the linings to conform more closely to the drum than would be the case to rigid shoes. This permits the linings to act more effectively and also reduces the need for wear adjustment.

The extra expense of providing another hinge is thus justified on the grounds of more uniform lining wear and consequently a longer life. This is the main advantage of the pivoted shoe brake

This is possible only if the shoe is in equilibrium.

For equilibrium of the shoe moments of the forces about the hinge pin should balance -

i.e $\Sigma M_G = T + F_x b_y - F_y b_x = 0$ where $b_x = b.\cos\theta_G$

$b_y = b.\sin\theta_G$

This needs that the resultant moment due to the frictional force (and due to the normal force) about the pivot point should be zero, so that no rotation of the shoe will occur about the pivot point. To facilitate this location of the pivot is to be selected carefully.

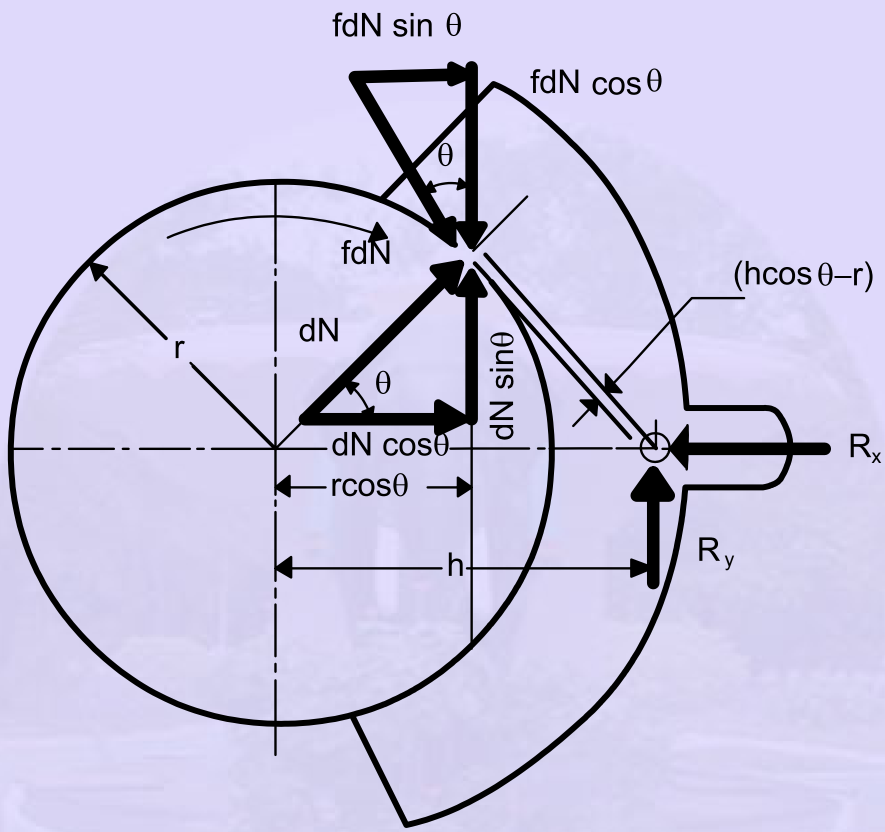
The actuating force P is applied to the post HG so the shoe itself is subject to actual and ideal contacts only - the (ideal) at pin G and the actual as distributed contact with the drum.

The location is in such a way that the moment of frictional force (and the normal force) about the pivot is zero. i.e the actual distributed contact leads to the ideal (concentrated)contact at the hinge or pivot.

i.e the actual distributed contact leads to the ideal contact at the hinge or pivot Further it is desirable to minimize the effect of pin reaction for which the shoe pivot and post pivot points are made con current.

Let us now look how this can be met, satisfying the conditions set above and consequently the derive the equations relating the location of the pivot from the center of the drum

A schematic sketch of a single shoe is shown in the figure



Force acting on shoe

Figure 3.1.13

An element of friction lining located at an angle Θ and subtending to a small angle $|d\theta|$ is shown in figure. The area if the element is $(r.d\theta.b)$, where b is the width of the friction lining parallel to the axis of the brake drum. If the intensity of pressure at the element is p, the normal reaction dN on the element is given by

$$dN = (rd\theta b)p \qquad (a)$$

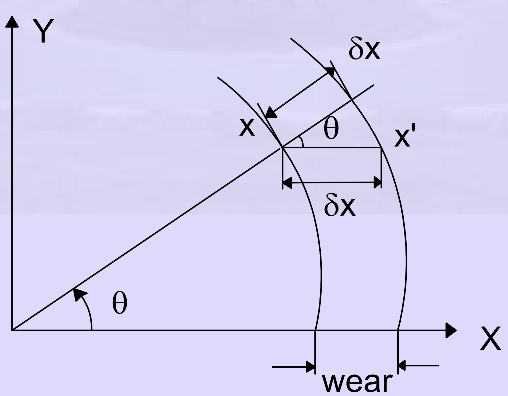
Distribution of pressure

If the shoe is long then the pressure will not be uniform

We need to determine the distribution of pressure along the lining; the pressure distribution should be conducive for maintaining a uniform wear

Since the brake drum is made of a hard material like cast iron or steel, the wear occurs on the friction lining, which is attached to the shoe. As shown in fig the lining need to retain the cylindrical shape of the brake drum when wear occurs. After the radial wear has take place, a point such as X' moves to X in order to maintain contact on the lining with the brake drum. In figure δx is the wear in the X direction and δr is the wear in the radial direction. If it is assumed that the shoe is constrained to move towards the brake drum to compensate to wear, δx should be constant because it need to be same for all points. Therefore,

$$\delta x = \frac{\delta r}{\cos\theta} = \text{constant} \qquad (b)$$



wear of friction lining

Figure 3.1.14

The radial wear δr is proportional to the work done by the frictional force. The work done by the frictional force depends upon the frictional force (fdN) and the rubbing velocity. Since the rubbing velocity is constant for all points on friction lining,

$$\delta r \propto fdN$$

Or $\delta r \propto (frd\theta bp)$

Therefore $\delta r \propto p$ (c)

From the expression (b) and (c)

$$\frac{p}{\cos\theta} = \text{constant} \quad \text{or } p = C_1 \cos\theta \qquad \text{(d)}$$

Where C_1 is the constant of proportionality. The pressure is maximum when $\theta = 0$.

Substituting,

$$p_{max} = C_1 \qquad \text{(e)}$$

From Eqs (d) and (e),

$$p = p_{max} \cos\theta$$

Substituting this value in Eq. (a

$$dN = (rd\theta b)p_{max} \cos\theta \qquad \text{(f)}$$

The forces acting on the element of the friction lining are shown in figure. The distance h of the pivot is selected in such a manner that the moment of frictional force about it is zero.

Therefore, $M_f == 0$

dM_f=f.dN moment arm

moment arm in this case = $(h\cos\theta - r)$

$$M_f = \int_{-\theta}^{\theta} f dN(h\cos\theta - r) = 0$$

$$M_f = \int_{-\theta}^{\theta} f p_{max} r d\theta \cos\theta (h - r\cos\theta)$$

Substituting dN from Eq. (f),

$$\int_{0}^{\theta}\left(h\cos^2\theta - r\cos\theta\right)d\theta = 0$$

$$\text{or} \quad h\int_{0}^{\theta}\left[\frac{1+\cos 2\theta}{2}\right]d\theta - r\int_{0}^{\theta}\cos\theta d\theta = 0$$

$$\text{or} \quad h\left[\frac{\phi + \frac{1}{2}\sin 2\phi}{2}\right]_{0}^{\theta} - R\left(\sin\phi\right)_{0}^{\theta} = 0$$

$$h = \frac{4R\sin\theta}{2\theta + \sin 2\theta}$$

The elemental torque of frictional force $|fdN|$ about the axis of brake drum is $|fdNR|$. Therefore

$$T_B = 2\int_{-\theta}^{\theta} f dNr$$

Substituting the value of dN from Eq.(f)

$$T_B = 2fr^2 b p_{max}\int_{-\theta}^{\theta}\cos\theta d\theta$$

$$T_B = 2fr^2 b p_{max}\sin\theta$$

The reaction R_X can be determined by considering two components $|(dN\cos\theta)|$ and $|(fdN\sin\theta)|$.

Due to symmetry, the other two vertical components of the force balances

i.e

$$\int f dN \sin\theta = 0$$
$$\int dN \ \sin\theta = 0$$

Therefore,

$$R_x = \int_{-\theta}^{\theta} dN \cos\theta$$
$$= rbp_{max} \int_{-\theta}^{\theta} \cos^2\theta d\theta$$
$$= 2rbp_{max}\left[\frac{2\theta + \sin 2\theta}{4}\right]$$
$$\text{or} \quad R_x = \frac{1}{2} rbp_{max}(2\theta + \sin 2\theta)$$

Note that R_x is also = F_n

The reaction R_y can be determined by considering two components $\left|(dN \sin\theta)\right|$ and $\left|(f dN \cos\theta)\right|$

Due to symmetry,

$$\int dN \sin\theta = 0$$

Therefore,

$$R_y = \int_{-\theta}^{\theta} f dN \cos\theta$$

$$= frbp_{max} \int_{-\theta}^{\theta} \cos^2\theta d\theta$$

$$\text{or} \quad R_y = \frac{1}{2} frbp_{max}(2\theta + \sin 2\theta)$$

As noted earlier,

$$T_B = 2fr^2 bp_{max} \sin\theta$$

Rewriting it,

$$T_B = frbp_{max} \frac{2\theta + \sin\theta}{2} \cdot \frac{4r\sin\theta}{2\theta + \sin\theta}$$

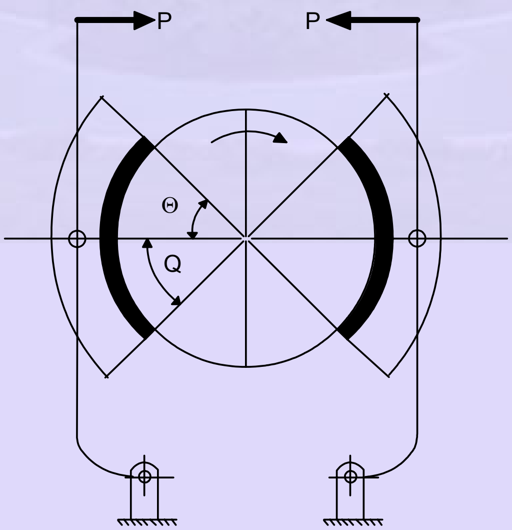
$$= fF_n h$$

DOUBLE BRAKE SHOE

A double block brake with two symmetrical and pivoted shoes is show in figure.

If the same magnitude of actuating forces are acted upon the posts, then

$$T_B = f.(F_{n1} + F_{n2}).h = 2f.F_n.h$$



Pivoted double block brake

Figure 3.1.15

Pivoted shoe brakes are mainly used in hoists and cranes. Their applications are limited because of the physical problem in locating pivot so close to the drum surface.