MMME2044 Introduction to Brakes and Clutches

- Brakes and Clutches transfer the momentum/energy/force between two components capable of relative motion
- Normally we are considering rotating systems
- The mechanism for this transfer is friction
- Contacting surfaces generate heat due to friction and must meet certain requirements to perform effectively
- For brakes one component of the system is fixed

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Key principles – Properties of frictional materials

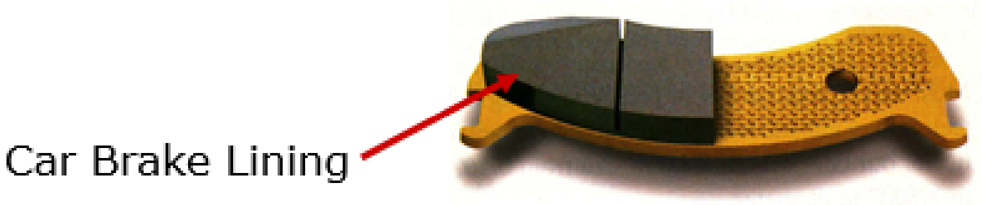
Frictional material for brakes and clutches:

- High coefficient of friction
- Unaffected by moisture or heat
- Good heat conductivity
- High contact pressures (p_{max})
- High resistance to wear and scoring

Lubrication can be used to assist with heat transfer

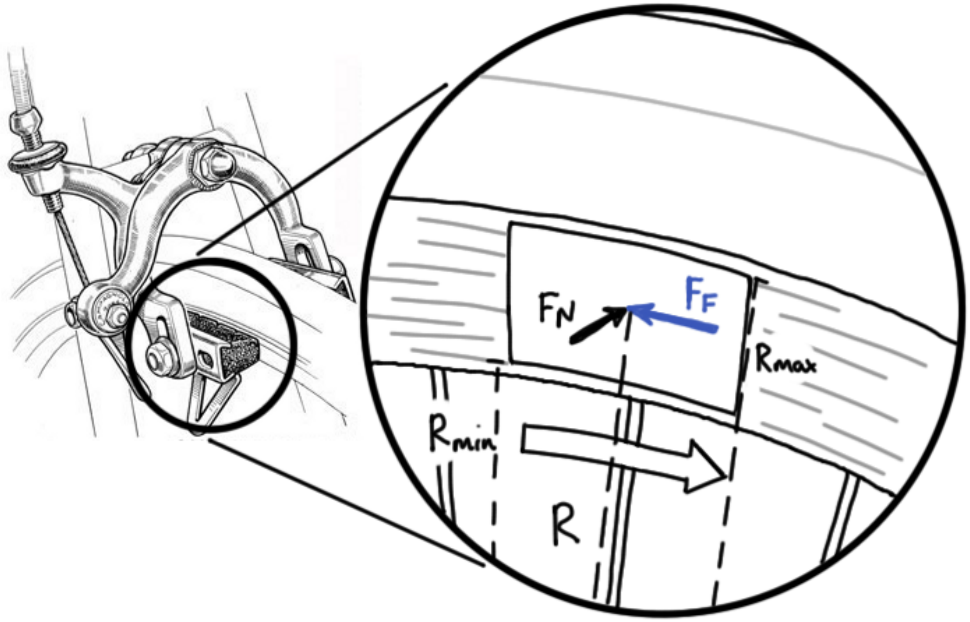
Moulded friction material (lining):

aramid fibres + metallic particles + phenolic resin



Rim brake

The common principle for all arrangements of brakes and clutches is that they rely on friction to transfer momentum/energy/force.
Perhaps the simplest example is the bicycle rim brake



Force is applied to the brake pad by the calliper

The force is normal to the surface

The amount of force is limited by the maximum rated pressure of the brake pad material, P_{max}

$P = F_N/\text{Area}$

Rim brake

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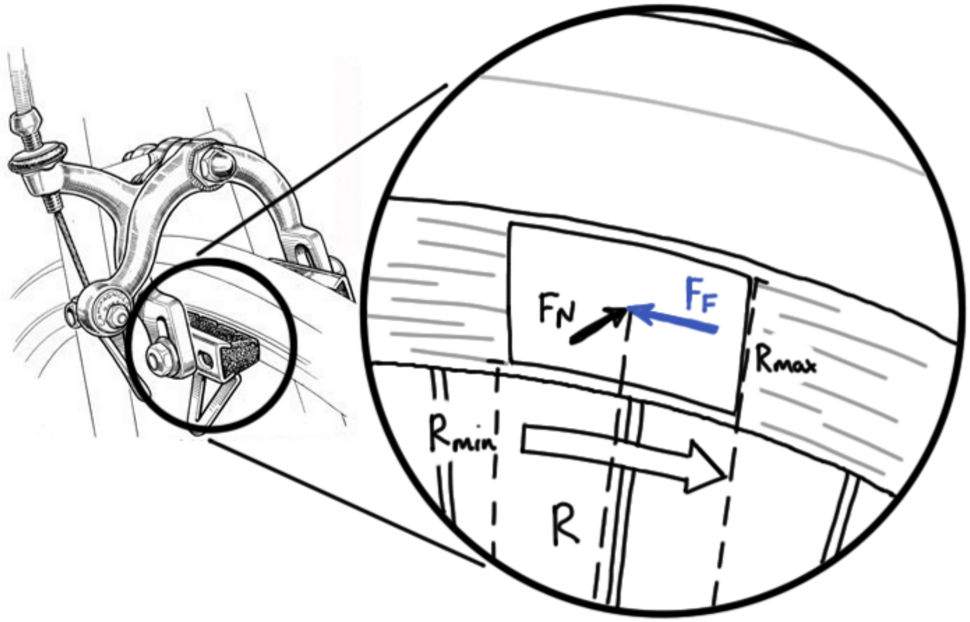


Normal force is distributed evenly

Friction force is therefore distributed evenly

Radius to the centre is large

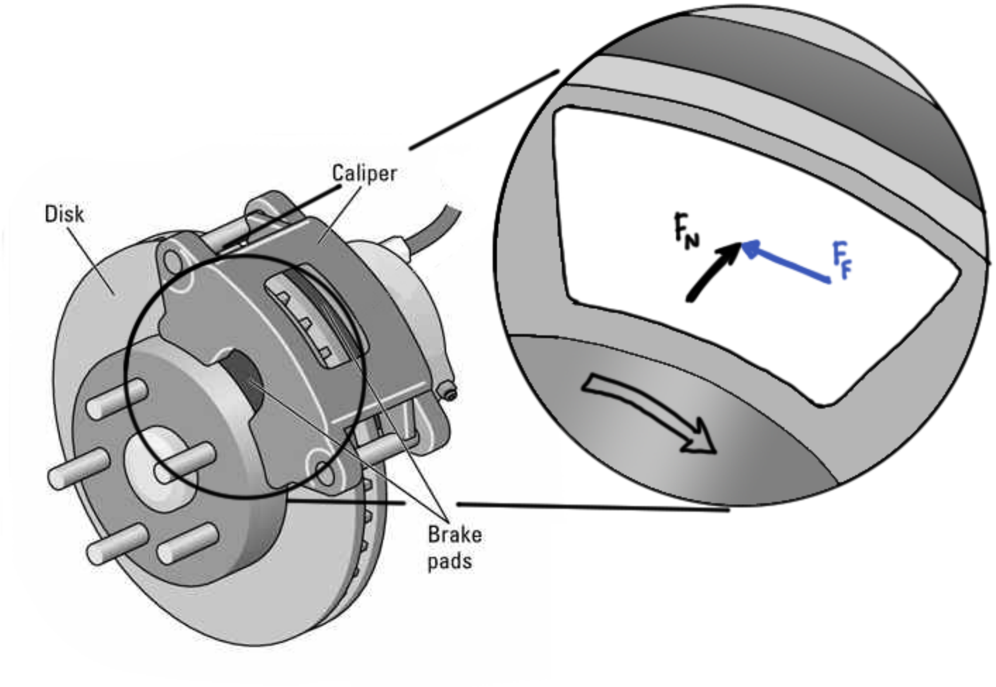
$r_{min} \approx r_{max}$

Torque, $T = F_F.r = \mu.F_N.r$

Torque is limited by P_{max} & geometry

A simple disc brake/clutch

The same principles apply when the geometry of the frictional surfaces change, as in the example of the disc brake/clutch



Normal force is distributed evenly

Friction force is distributed evenly

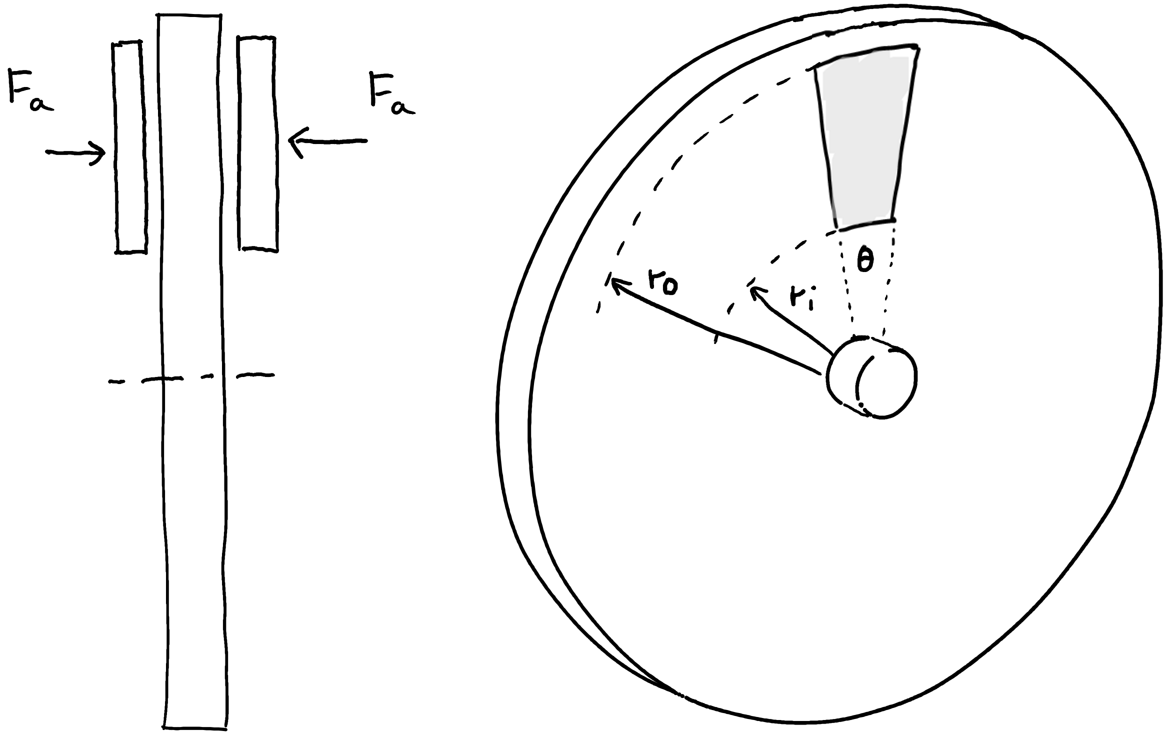
But radius is no longer large

$r_{min} << r_{max}$

Torque, $T = \int F_F.r\ dr = \int \mu F_N.r\ dr$

F_N is still limited by P_{max}

Disc brake simple pad geometry



Area of contact = $\theta \cdot r_i \cdot (r_o - r_i)$

$F_a = p_{max} \cdot N \cdot Area$

Area of Annulus sector $\frac{\theta}{2}(r_o^2 - r_i^2)$

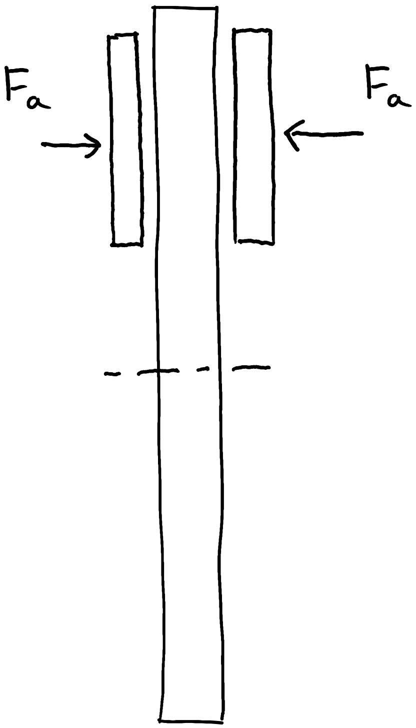
When θ is small we only need to integrate over radius

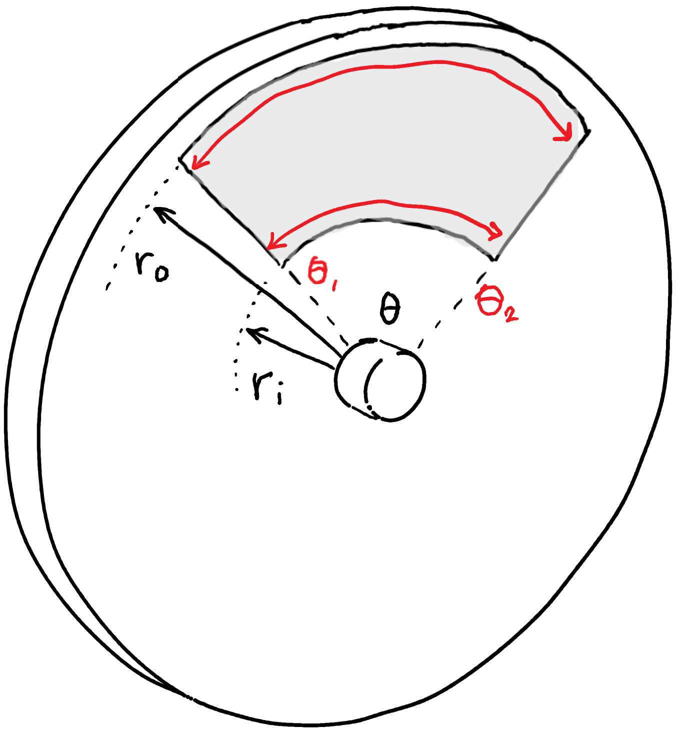
$$T = \int_{r_i}^{r_o} N.\mu.Fa.r \, dr$$

$$T = N \cdot \mu \cdot F_a \left(\frac{r_i + r_o}{2} \right)$$

N is the number of frictional contacts. In this case N = 2

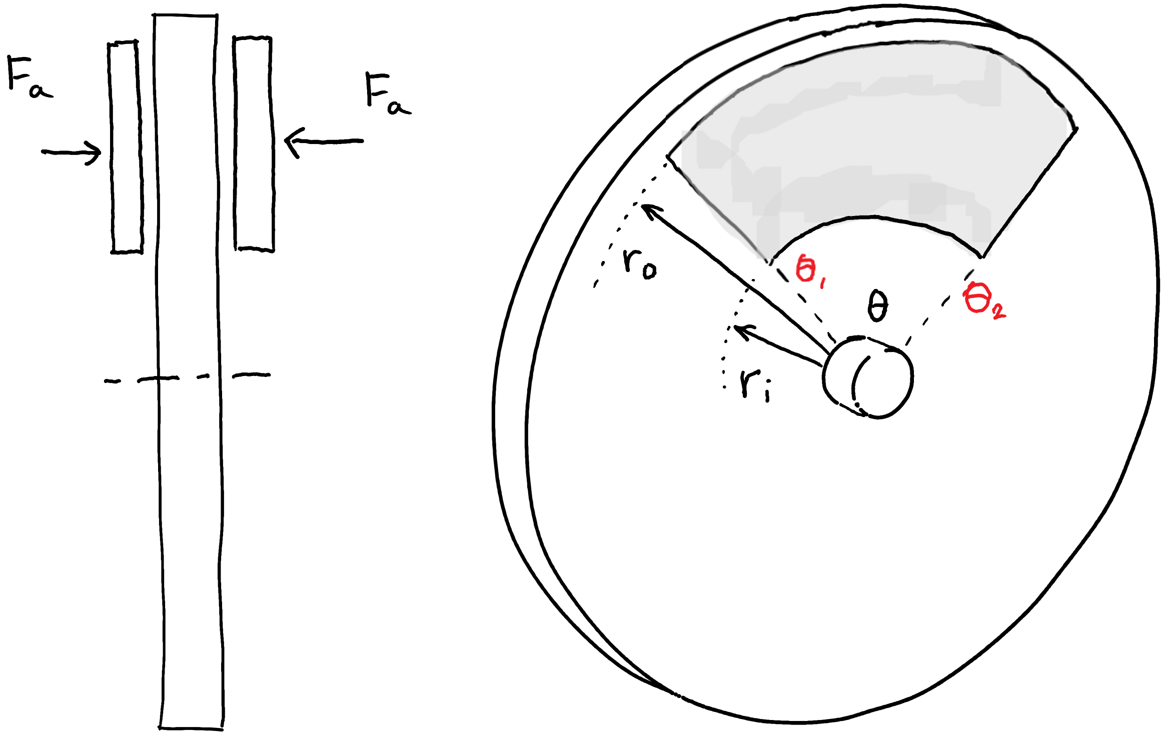
Disc brake wide pad geometry





As θ becomes large we have more area at a longer moment arm (as radius increases) and it becomes necessary to perform a double integration.

Disc brake wide pad geometry continued



Assuming pressure is uniform

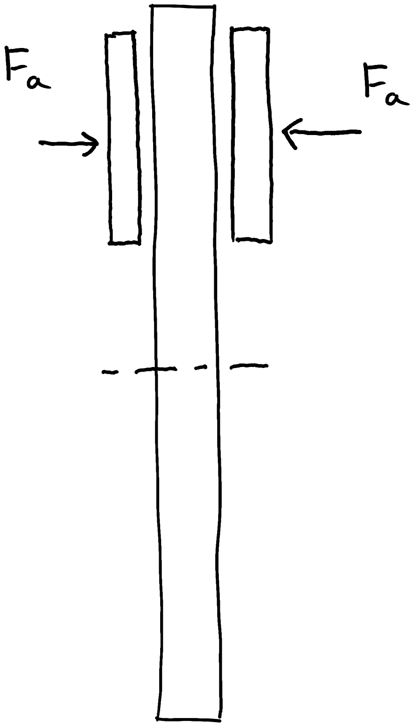
$$T = \int_{\theta_1}^{\theta_2} \int_{r_i}^{r_o} N.\mu.P.r \; dr\,d\theta$$

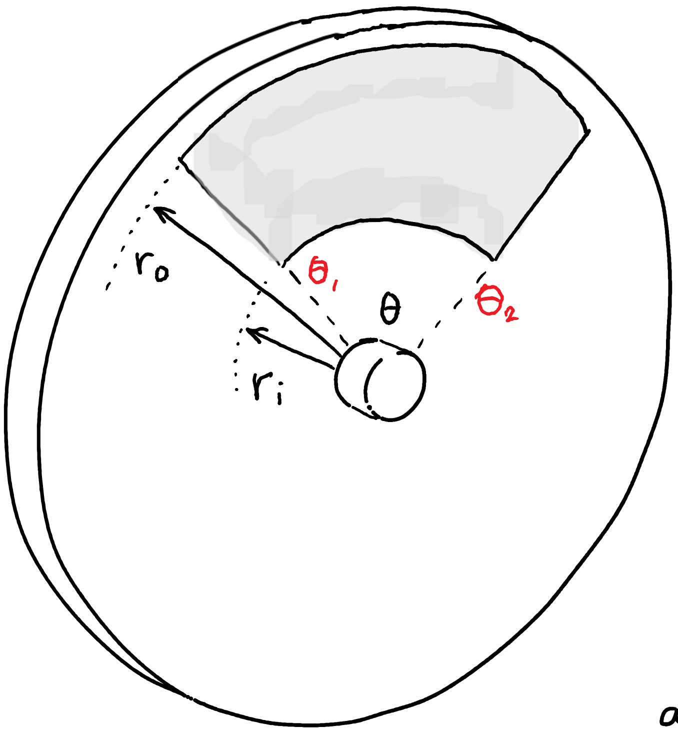
$$T = \tfrac{1}{3}(\theta_2 - \theta_1).N.\mu.P.(r_o^3 - r_i^3)$$

$$F_a = \tfrac{1}{2}(\theta_2 - \theta_1).N.P.(r_o^2 - r_i^2)$$

This is likely to be the case when the brake is new and has no wear.

Disc brake wide pad geometry continued



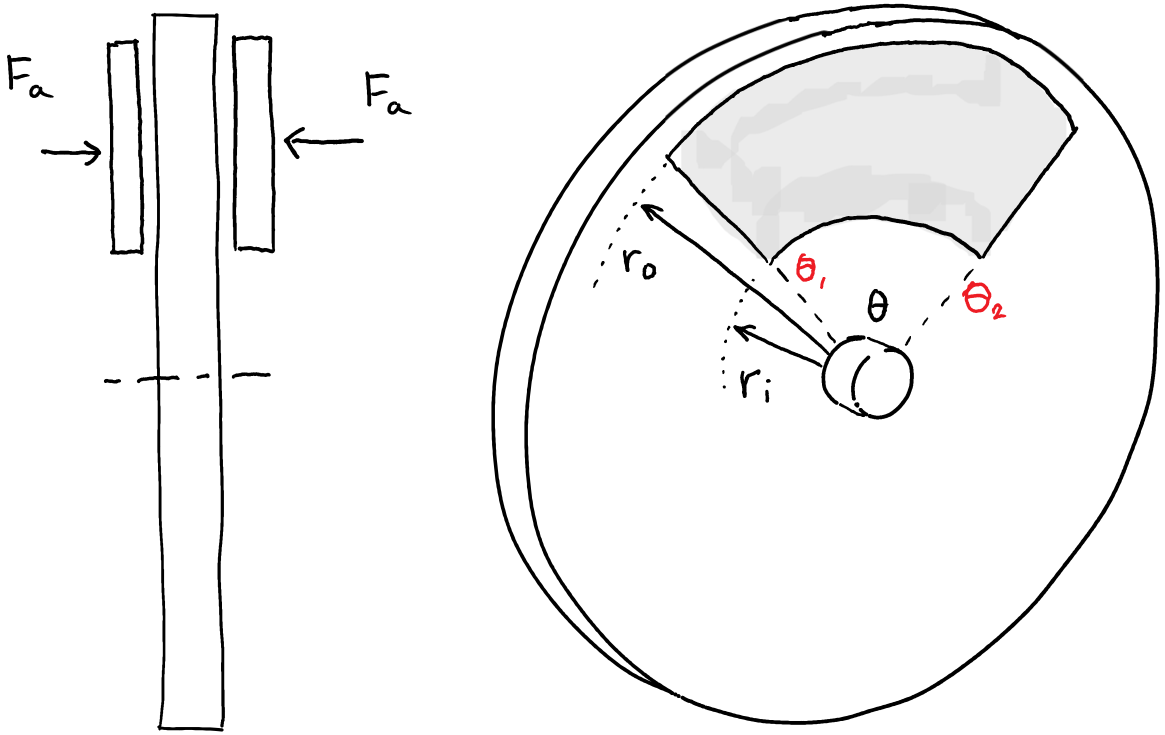


We know from our study of bearings that wear rate is proportional to contact force and sliding distance.

$$\text{Wear factor} = \frac{\text{volume } [mm^3]}{\text{force } [N] \,.\, \text{distance } [m]}$$

∴ we would expect higher wear at the outer edge of pad and disc.

Disc brake wide pad geometry continued



Assuming pressure varies with wear.

$P.r$ becomes a constant and our integral becomes

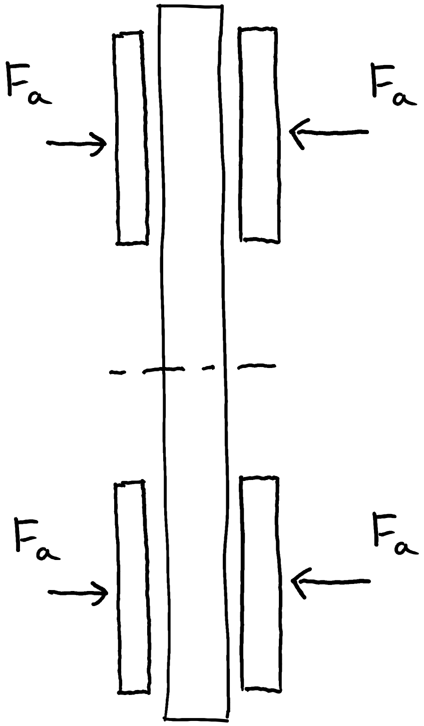
$$T = \frac{1}{2}(\theta_2 - \theta_1).N.\mu P.r_i.(r_o^2 - r_i^2)$$

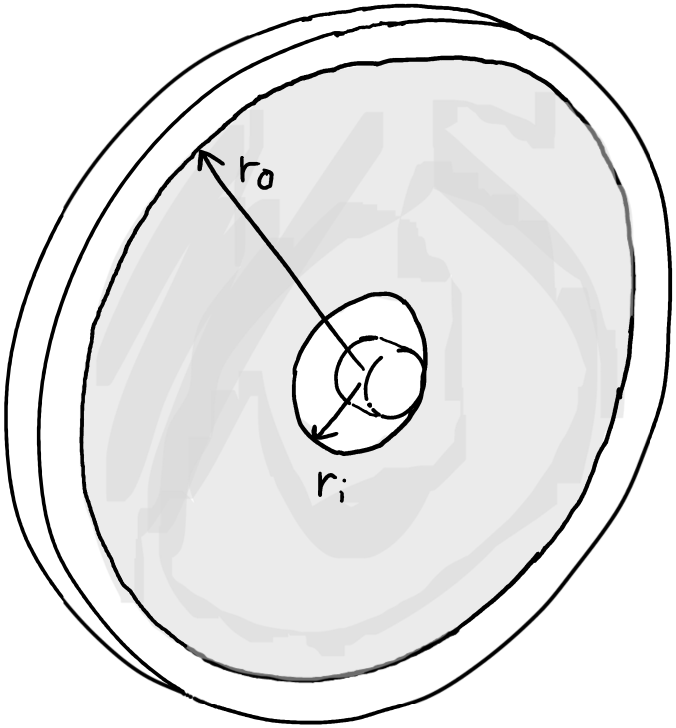
$$F_a = (\theta_2 - \theta_1).N.P.r_i.(r_o - r_i)$$

This is likely to be the performance when the brake or clutch is in normal operation.

Axial plate clutch

In the special case $\theta = 2\pi$





$$\text{Area} = N.\pi(r_o^2 - r_i^2)$$

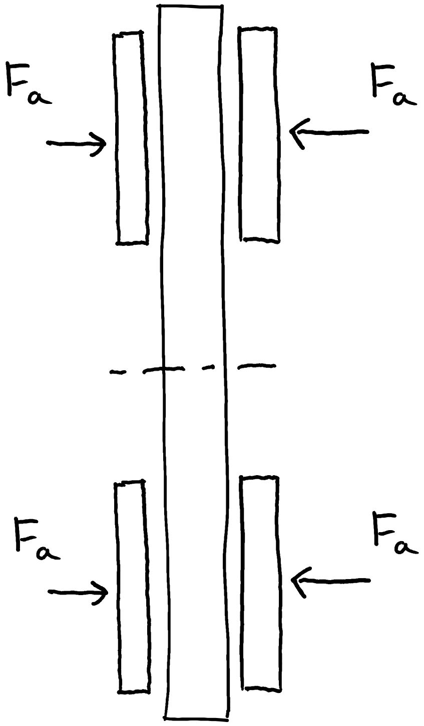
$$F_a = N.P.\pi(r_o^2 - r_i^2) \text{ for}$$

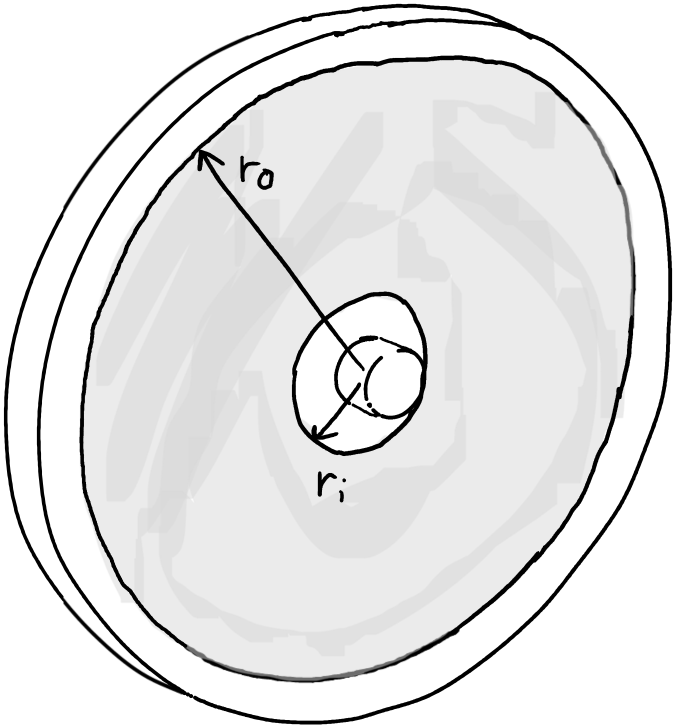
constant pressure scenarios.

$$T = \int_{r_i}^{r_o} 2\pi.\mu.N.P.r^2\,dr$$

$$T = \frac{2}{3}\pi.\mu.N.P(r_o^3 - r_i^3)$$

Axial plate clutch





In the worn condition where

$P.r$ is a constant

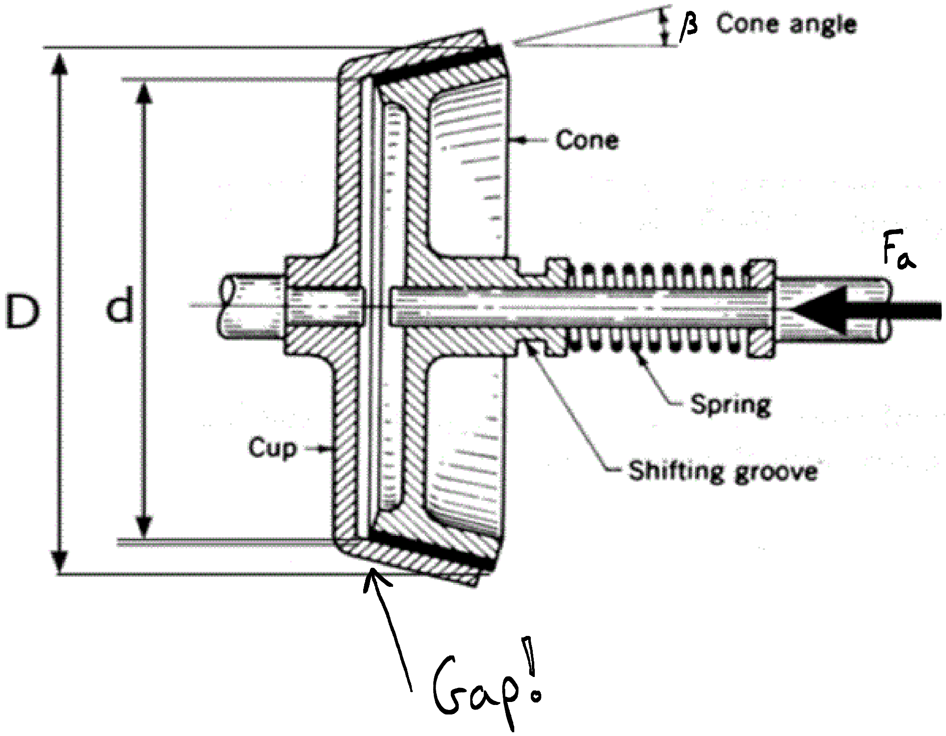
$$T = \frac{\pi.\mu.N.P.2.r_i(2r_o^2-2r_i^2)}{8}$$

$$F_a = \pi.N.P.r_i.(2r_o-2r_i)$$

This will be the most likely performance in long-term operation.

Designers must carefully consider which set of calculations best matches their application conditions.

Cone clutch



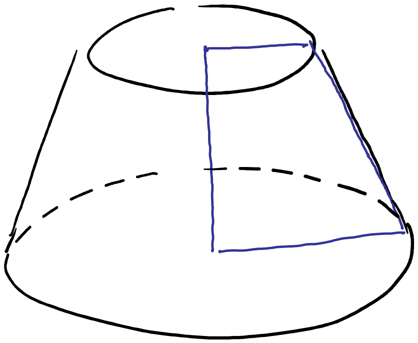
High torque transfer in a small volume.

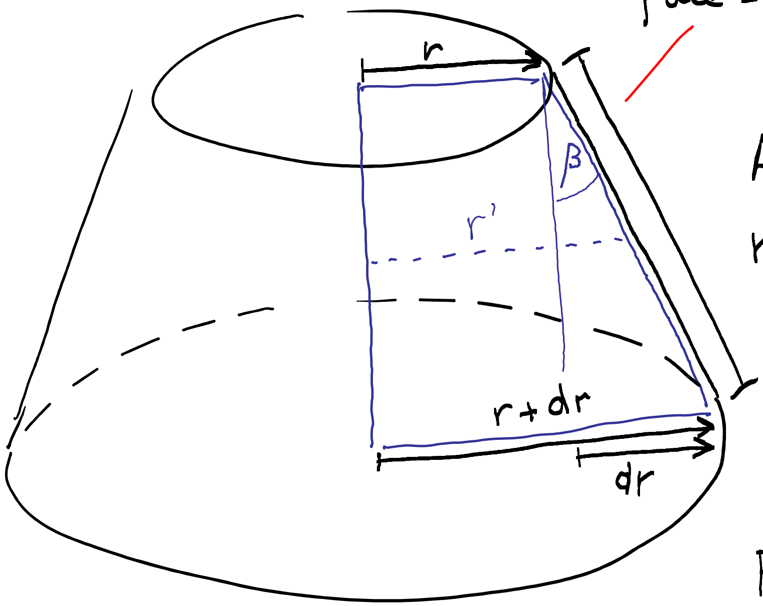
Cone acts like a wedge, multiplying the axial force for cone angles < 45°.

Can have issues with release.

Same principle, normal force over a frictional contact area.

Truncated cone
or
conical frustum.





Length of cone face = $\dfrac{dr}{\sin\beta}$ [1]

Average radius

$$r' = \frac{r + (r + dr)}{2}$$

Circumference
$= 2\pi r'$

For short cones
Circumference
$\approx 2\pi r$

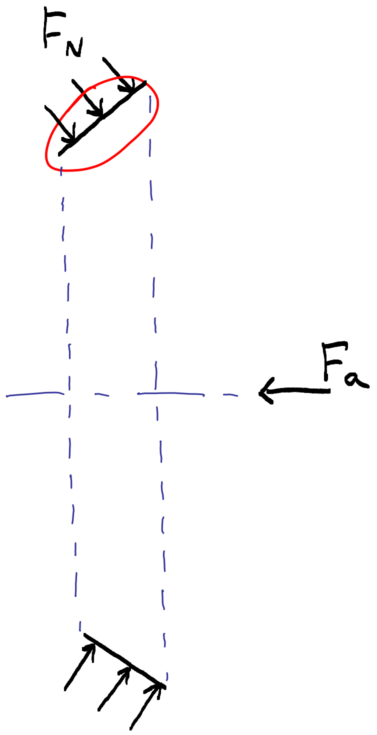
Area is face length times circumference.

$$\text{Area} = \frac{2\pi r . dr}{\sin\beta} \quad [2]$$

When the cone clutch is new and there is no wear, we can assume that contact pressure is uniform and ∴ so is the normal force.

$$F_N = \frac{p . 2\pi r . dr}{\sin\beta} \quad [3]$$

Uniform pressure means uniform normal force



$$F_N = \frac{p \cdot 2\pi r \cdot dr}{\sin\beta} \quad [3]$$

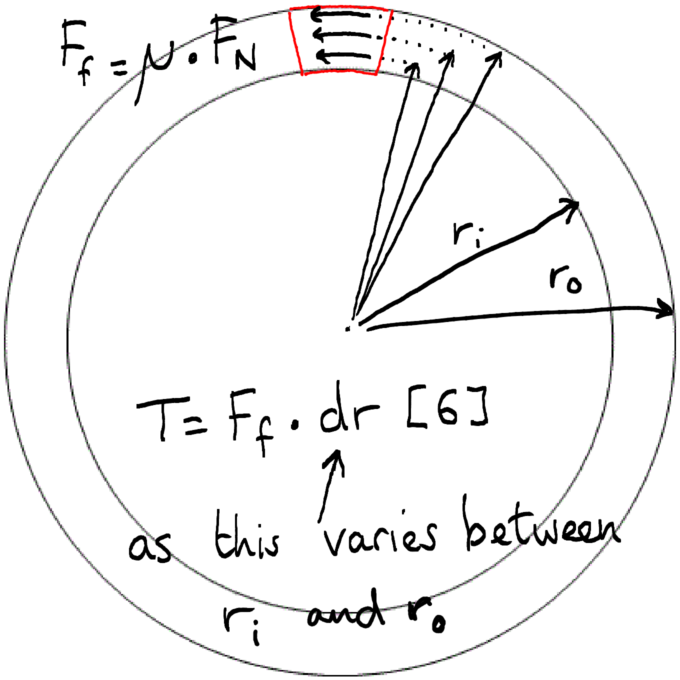
$$F_f = \frac{\mu \cdot p \cdot 2\pi r \cdot dr}{\sin\beta} \quad [5]$$

also $F_a = F_N \cdot \sin\beta$

$$\therefore F_a = \frac{p \cdot 2\pi r \cdot dr}{\sin\beta} \cdot \sin\beta$$

$$F_a = p \cdot 2\pi r \cdot dr \quad [4]$$

Uniform normal force means that at any element the friction force is the same.



$F_f = \mu \cdot F_N$

$$T = F_f \cdot dr \quad [6]$$

as this varies between r_i and r_o

Expressing [4] and [6] as integrals across that range we get.

$$T = \frac{\mu . 2\pi}{\sin\beta} \int_{r_i}^{r_o} p . r^2 . dr \quad [7]$$

$$F_a = 2\pi \int_{r_i}^{r_o} p . r . dr \quad [8]$$

We can now make one of two assumptions.

1) The clutch has perfect geometry and p is constant.

[8] becomes

$$F_a = \pi . p (r_o^2 - r_i^2) \quad [9]$$

[7] becomes

$$T = \frac{\mu . 2\pi . p}{3 . \sin\beta} (r_o^3 - r_i^3) \quad [10]$$

rearranging [9] for p we get

$$p = F_a / \pi . (r_o^2 - r_i^2)$$

substituting into [10]

$$T = \frac{2 \mu F_a (r_o^3 - r_i^3)}{3 \sin\beta (r_o^2 - r_i^2)} \quad [11]$$

$$F_a = \frac{3 . \sin\beta (r_o^2 - r_i^2) . T}{2 . \mu . (r_o^3 - r_i^3)} \quad [12]$$

This approach only applies when there is no wear.

2) Our alternative assumption is that wear has occured. Because the outer edge of the cone has larger circumference, it slides further and wears faster.

The result is that (p.r) is now constant.

i.e. the inner edge has higher pressure (from minimal wear) but a smaller radius. The outer edge has less pressure (due to wear) but a larger radius.

Now our integrals become:

$$F_a = 2\pi . p . r_i (r_o + r_i) \quad [13]$$

$$T = \frac{\mu . \pi . p . r_i (r_o^2 - r_i^2)}{\sin\beta} \quad [14]$$

rearrange

$$p = F_a / 2\pi r_i (r_o + r_i) \quad [15]$$

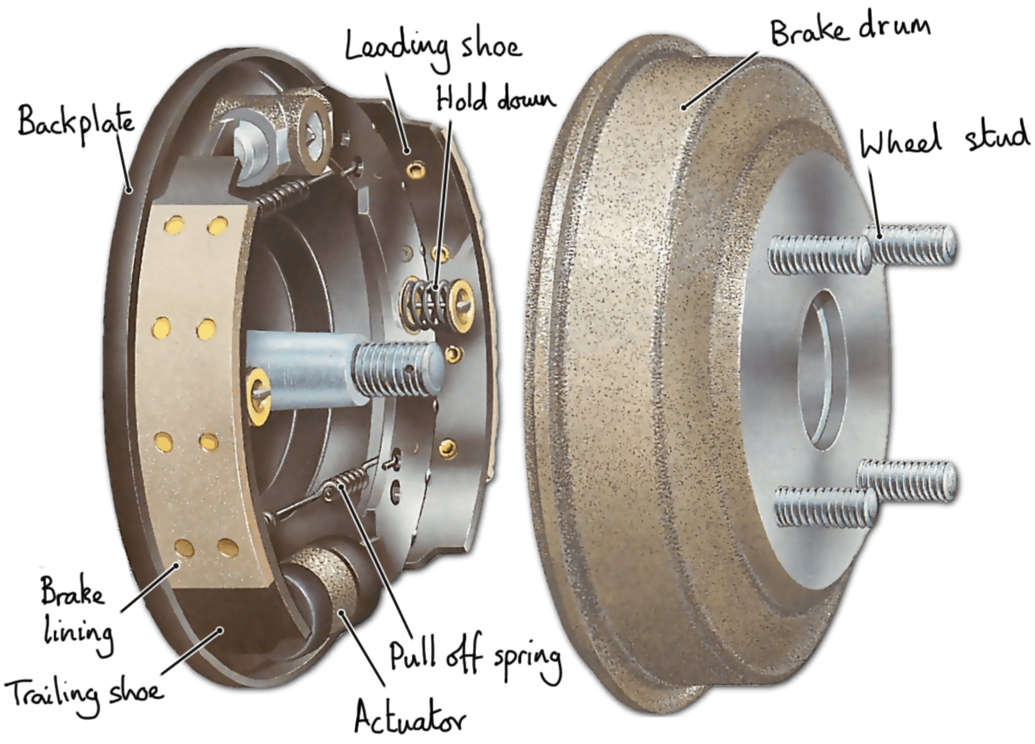
Substitute into [14]

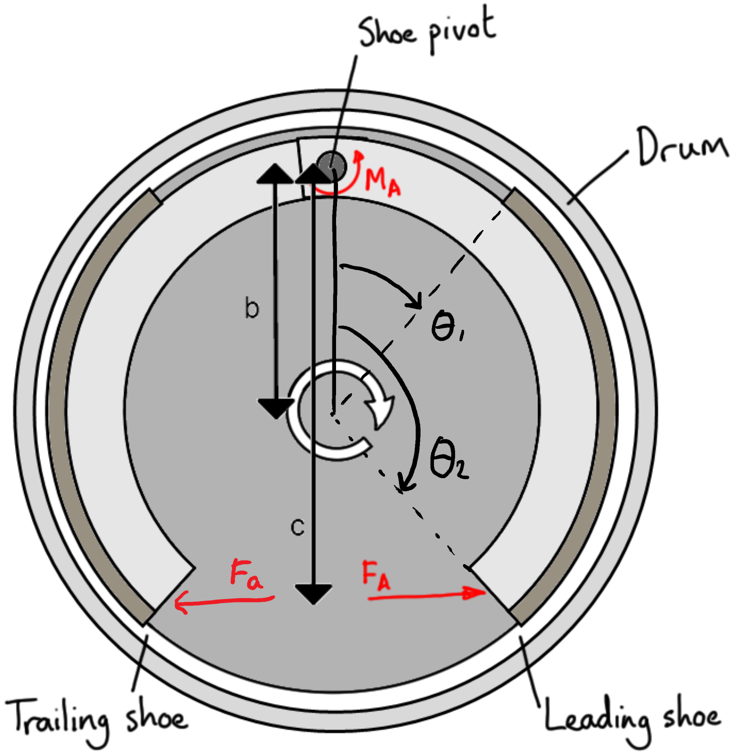
$$T = \frac{\mu F_a . (r_o + r_i)}{2 \sin\beta} \quad [16]$$

$$F_a = \frac{2 \sin\beta}{\mu (r_o + r_i)} . T \quad [17]$$

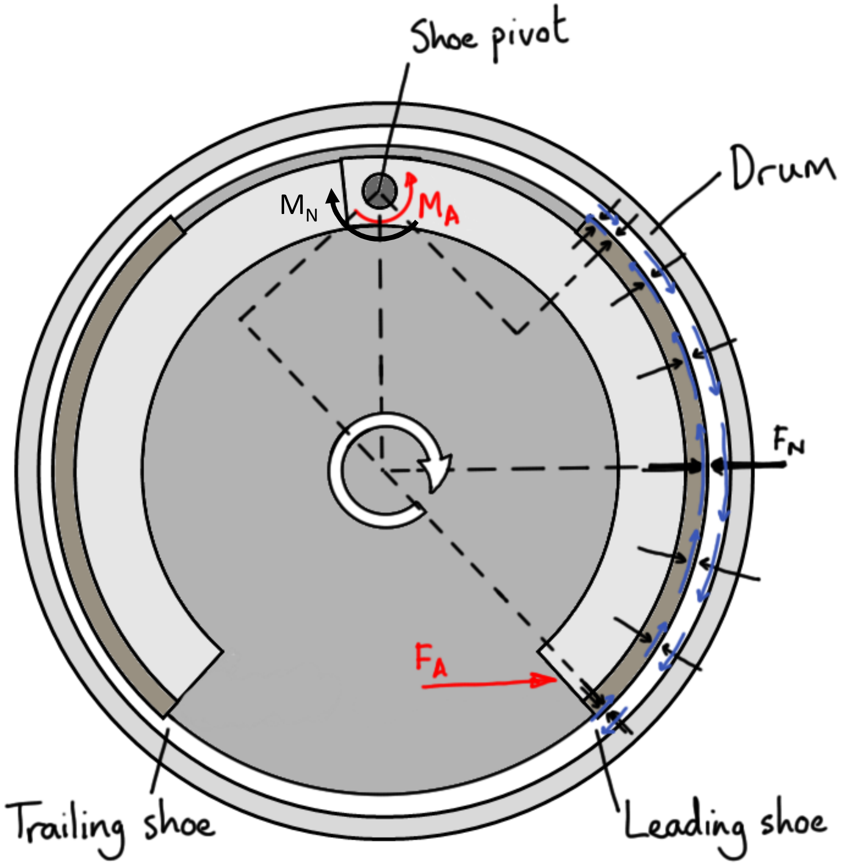
Key principles – Drum brakes/clutches

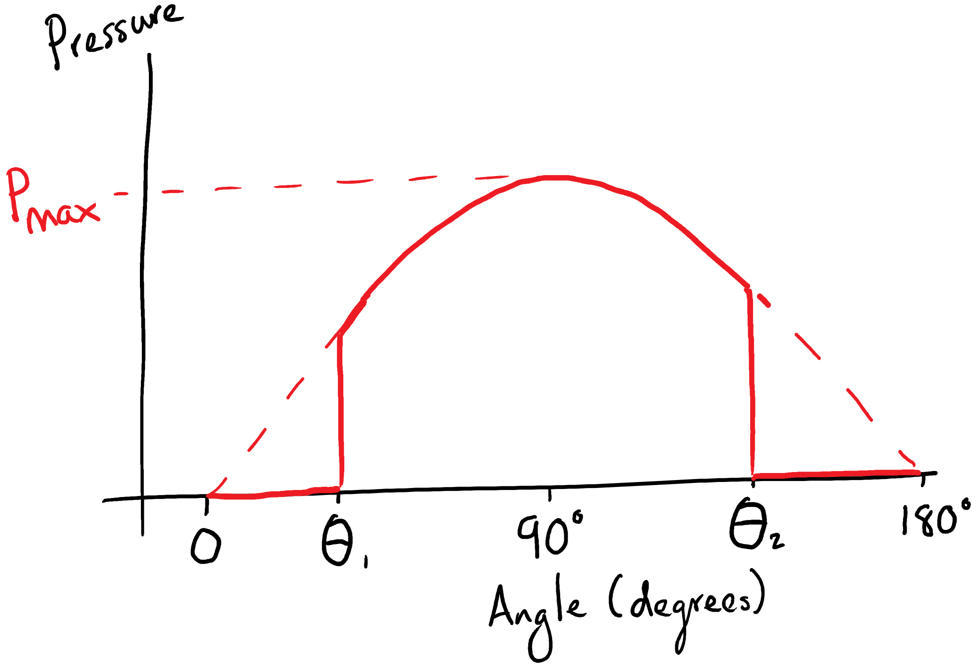
For drum geometries the frictional contact surface is at constant radius but varying normal force



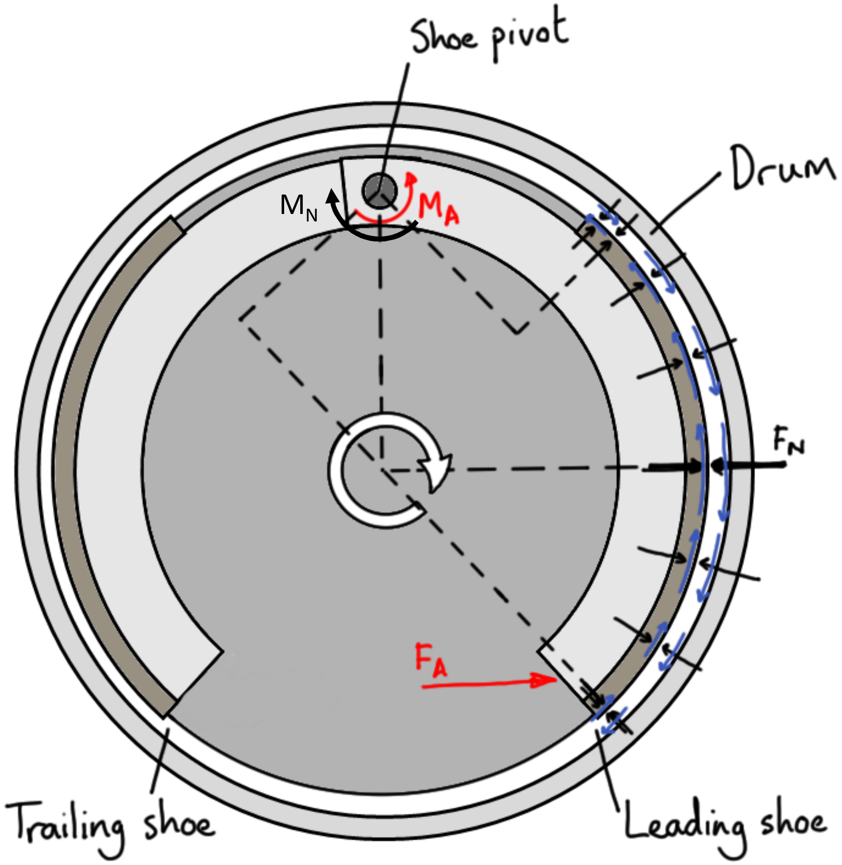


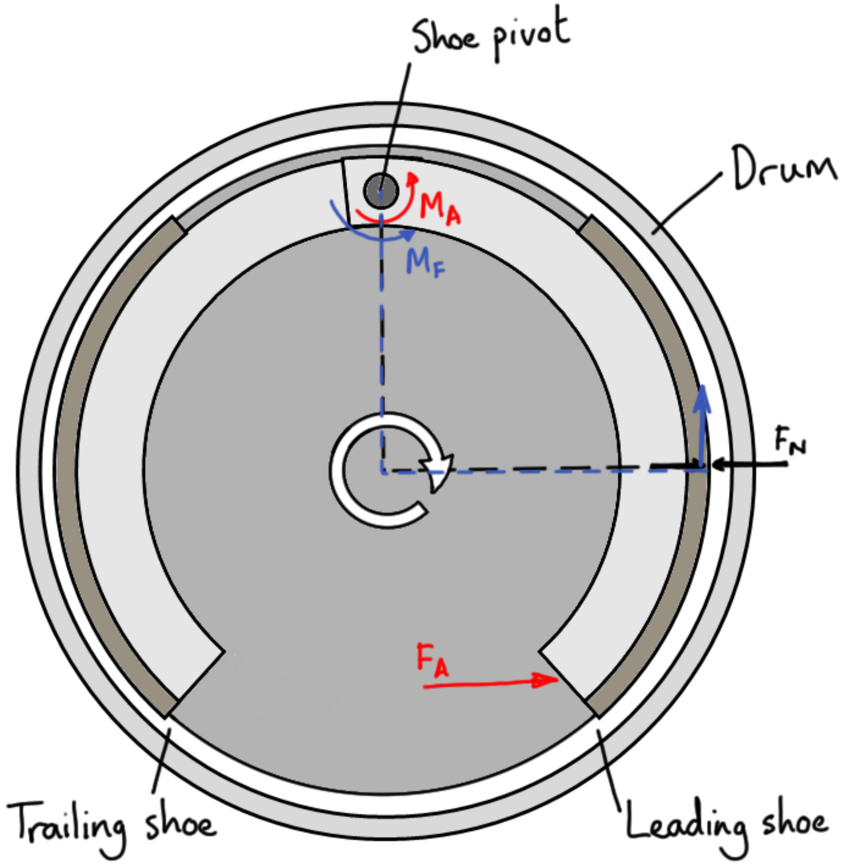
Key principles – Drum brakes/clutches





Key principles – Drum brakes/clutches





Leading shoe: braking torque capacity

Determine braking torque applied by <u>leading shoe</u>

max pressure felt by lining

(must be ≤ to suppliers spec.)

normal forces

$$T_L = \int r \times \mu \, dF_n = \mu \, w r^2 \, \frac{p_{\max}}{\sin \theta_{\max}} (\cos \theta_1 - \cos \theta_2)$$

drum radius

angle at which max pressure occurs

w = width
b = the distance from the pivot to the centre of rotation
r = the radius at which the shoe contacts the drum, with respect to the centre of rotation
c = the distance from the pivot to the brake actuators

Leading shoe: normal moment

Determine the normal moment from the leading shoe

normal forces

$$M_n = \int l_{\perp\ pivot} \cdot dF_n \quad = \frac{wrbp_{\max}}{\sin \theta_{\max}} \left[\frac{1}{2}(\theta_2 - \theta_1) - \frac{1}{4}\left(\sin 2\theta_2 - \sin 2\theta_1\right) \right]$$

perp. distance to pivot

w = width
b = the distance from the pivot to the centre of rotation
r = the radius at which the shoe contacts the drum, with respect to the centre of rotation
c = the distance from the pivot to the brake actuators

Leading shoe: frictional moment

Determine the frictional moment from the leading shoe

$$M_f = \int l_{\perp\,pivot} \cdot \overset{\substack{\text{normal forces}\\ \searrow}}{\mu\, dF_n} = \frac{\mu w r p_{\max}}{(\sin\theta)_{\max}}\left[r(\cos\theta_1 - \cos\theta_2) + \frac{b}{4}(\cos 2\theta_2 - \cos 2\theta_1)\right]$$

$$\underset{\text{perp. distance to pivot}}{\nearrow}$$

w = width
b = the distance from the pivot to the centre of rotation
r = the radius at which the shoe contacts the drum, with respect to the centre of rotation
c = the distance from the pivot to the brake actuators

Leading shoe - Sum moments to zero (equilibrium)

$$0 = cF_a + M_f - M_n$$

Sum moments about the shoe pivot

additive for leading shoe

$$F_a = \frac{M_n - M_f}{c}$$

"Self Locking"

if $M_n \leq M_f$

then $F_a \leq 0$

lining will "grab" and lock to drum on contact

w = width
b = the distance from the pivot to the centre of rotation
r = the radius at which the shoe contacts the drum, with respect to the centre of rotation
c = the distance from the pivot to the brake actuators

Trailing shoe - Sum moments to zero (equilibrium)

$$0 = cF_a - M_f - M_n$$

Sum moments about the shoe pivot

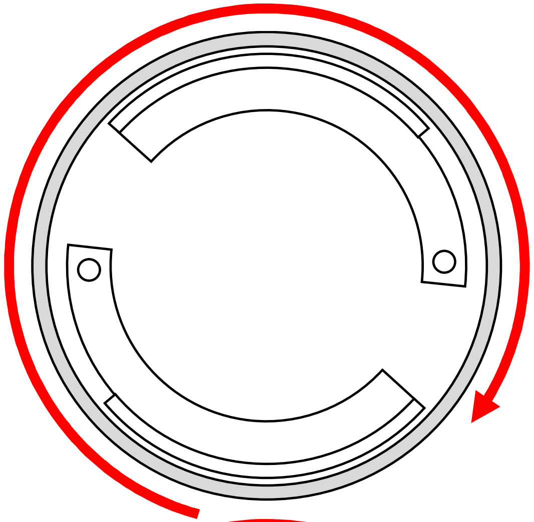
Subtractive for trailing shoe

$$F_a = \frac{M_n + M_f}{c}$$

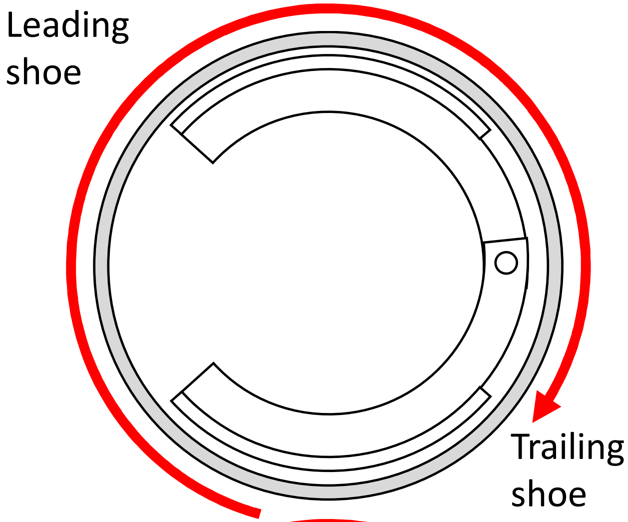
Trailing shoes are always de-energising

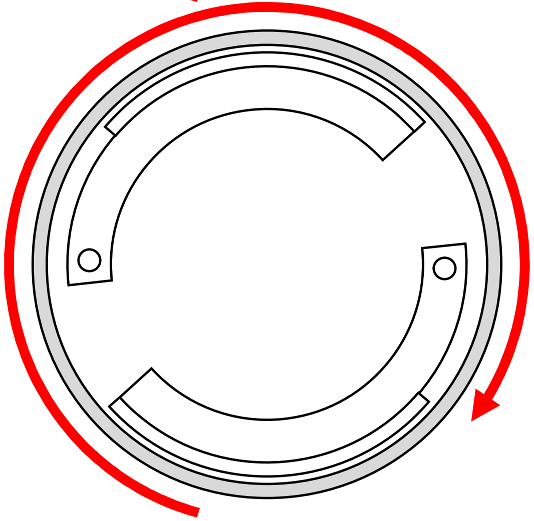
If your design has both leading and trailing shoes you will normally design for the leading shoe first as the self-energising effect will mean this shoe always has the higher maximum pressure and will exceed material pressure limit at lower actuation force

w = width
b = the distance from the pivot to the centre of rotation
r = the radius at which the shoe contacts the drum, with respect to the centre of rotation
c = the distance from the pivot to the brake actuators

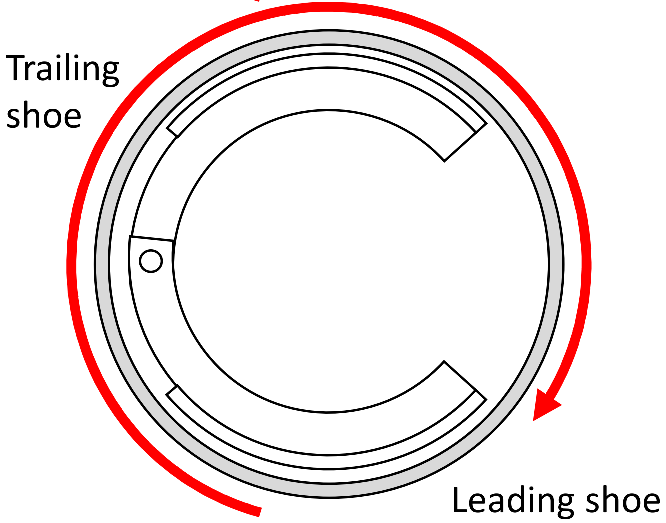


Leading – Leading brake
High breaking force
Brakes are self engaging
Will remain engaged even
if the actuation force is
removed.



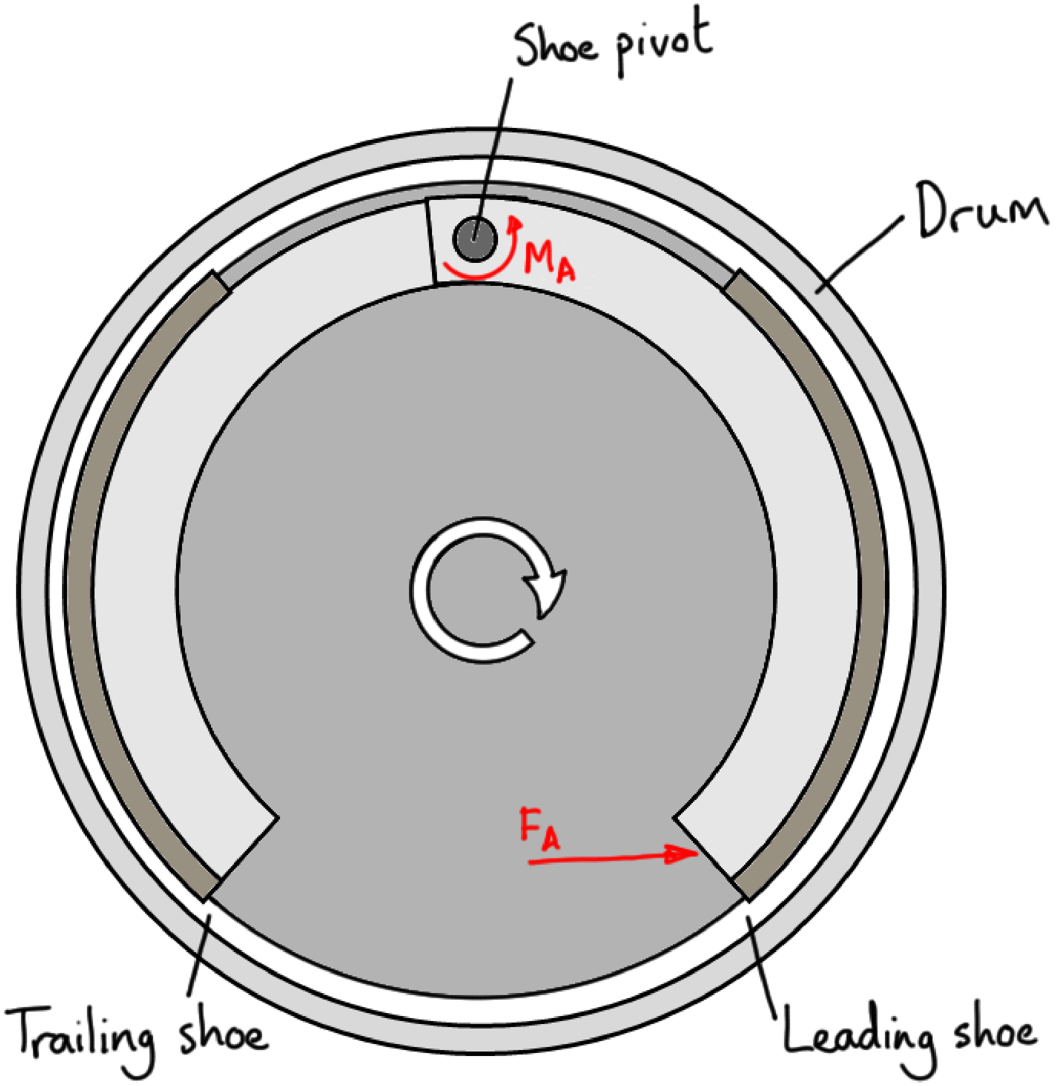


Trailing - Trailing brake
Low breaking force
Brakes self releasing
Actuation force must be
maintained to maintain
brake torque.



Summary

- Brakes and clutches both use friction to bring rotating systems to the same velocity
- The frictional materials for contacts must:
  - Have high and uniform coefficient of friction
  - Be unaffected by moisture and heat
  - Have good heat conductivity
  - Withstand high pressures
  - Withstand wear and scoring
- Torque is limited by P_{max} and the contact area
- Disc systems have varying radius and nominally constant pressure
- Drum systems have nominally constant radius but varying pressure
- Wear can cause the performance of the systems to degrade. Performance loss is greater for disc/plate/cone systems due to greater wear at larger radius
- Long shoe systems must consider leading/trailing orientation self-energise/de-energise


Shoe pivot
Drum
M_A
F_A
Trailing shoe
Leading shoe