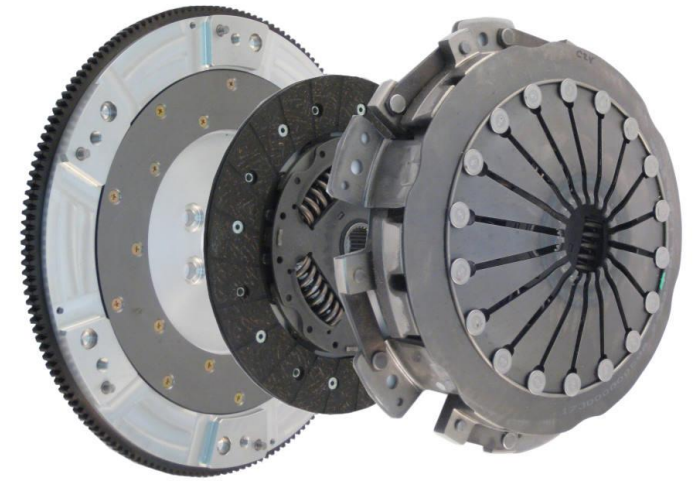


DESIGN OF MACHINE ELEMENTS- II



Module-I

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RSET
RAJAGIRI SCHOOL OF
ENGINEERING & TECHNOLOGY



Syllabus

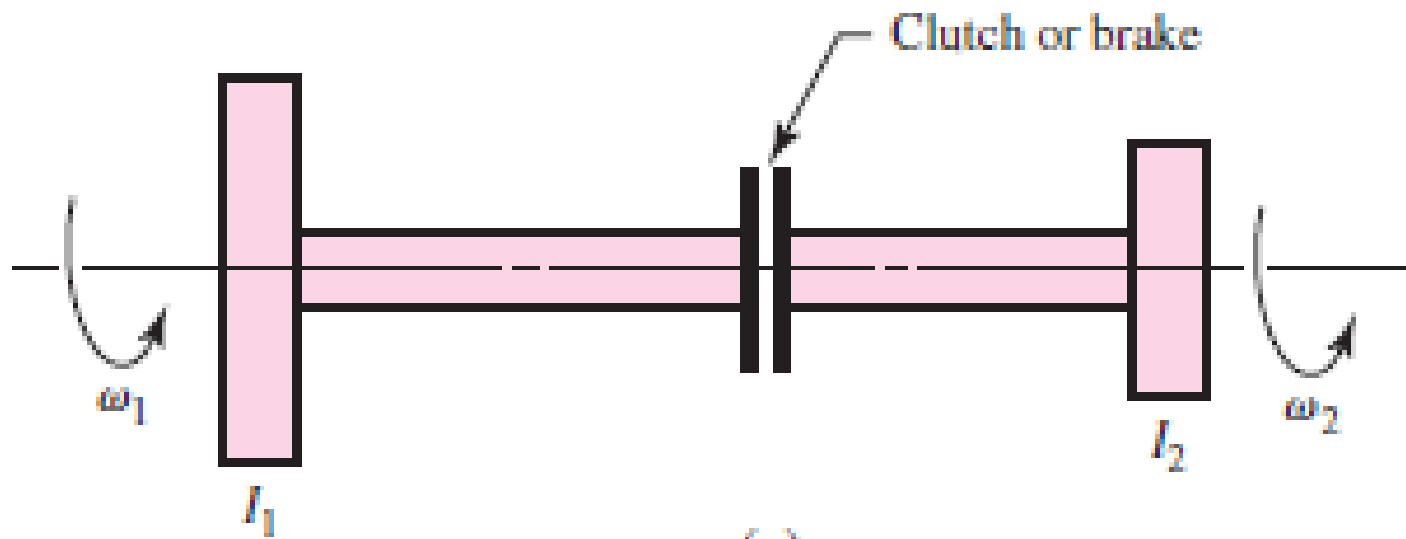
Module	Topics
1	<p>Clutches - friction clutches, design considerations, multiple disc clutches, cone clutch, centrifugal clutch. (2)</p> <p>Brakes- Block brake, band brake, band and block brake, internal expanding shoe brake (3)</p>
2	<p>Rolling contact bearing- Design of bearings, Types, Selection of a bearing type, bearing life, static and dynamic load capacity, axial and radial loads, selection of bearings, dynamic equivalent load, (4)</p> <p>Sliding contact bearing- lubrication, lubricants, viscosity, Journal bearings, hydrodynamic theory, Sommerfield number, design considerations, heat balance, bearing housing and mountings (4)</p>
3	<p>Gears- classification, Gear nomenclature, Tooth profiles, Materials of gears, Law of gearing (review only), virtual or formative number of teeth, gear tooth failures, Beam strength, Lewis equation, Buckingham's equation for dynamic load, wear load, endurance strength of tooth, surface durability, heat dissipation – lubrication of gears – Merits and demerits of each type of gears. (3)</p> <p>Design of spur gear (3)</p>

Syllabus

Module	Topics
4	Design of helical gear (2) Design of bevel gear (2) Design of worm & worm wheel (3)
5	Design of flat belt - materials for belts, slip of the belts, creep, centrifugal tension (3) Design of V-belt drives , Advantages and limitations of V-belt drive (3) Selection of roller chains , power rating of roller chains, galling of roller chains, polygonal action, silent chain. (3)
6	Connecting rod – material, connecting rod shank, small end, big end, connecting rod bolts, inertia bending stress, piston (5) Pressure vessels , thin cylinders, Thick cylinder equation, open and closed cylinders (2)

Clutches

Clutch is a **mechanical device**, which is used to **connect/disconnect** the **source of power** from the remaining parts of the power transmission system at the will of the operator



Dynamic representation of a clutch/brake (Mechanical Engineering Design:

Richard G Budynas and J. Keith)

Clutches

Initial Condition

- Driving member rotating and driven member at rest

Final Condition

- Both members rotate at same speed with no relative motion

Brakes

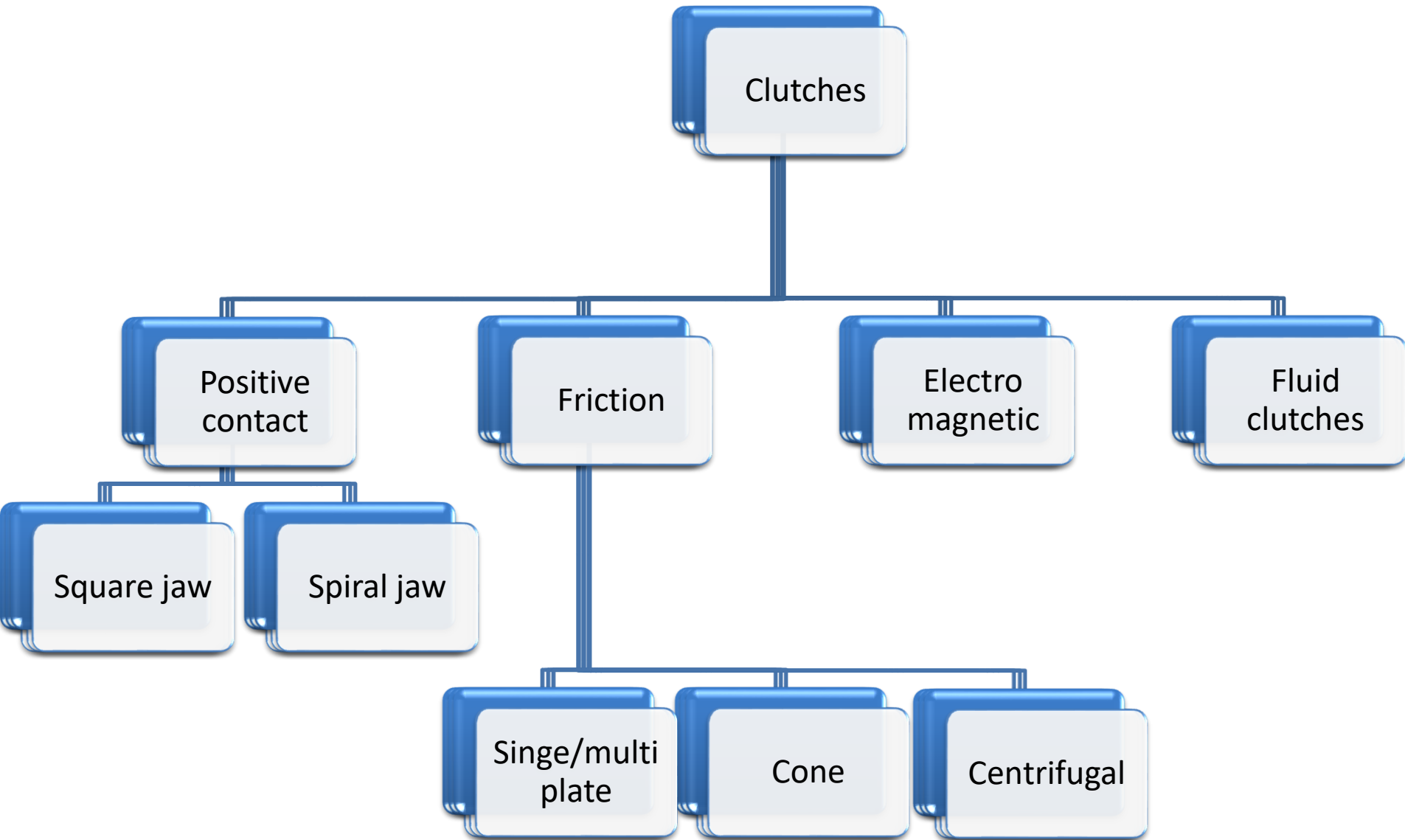
Initial Condition

- One member (brake drum) rotating and braking member (brake shoe) is stationary

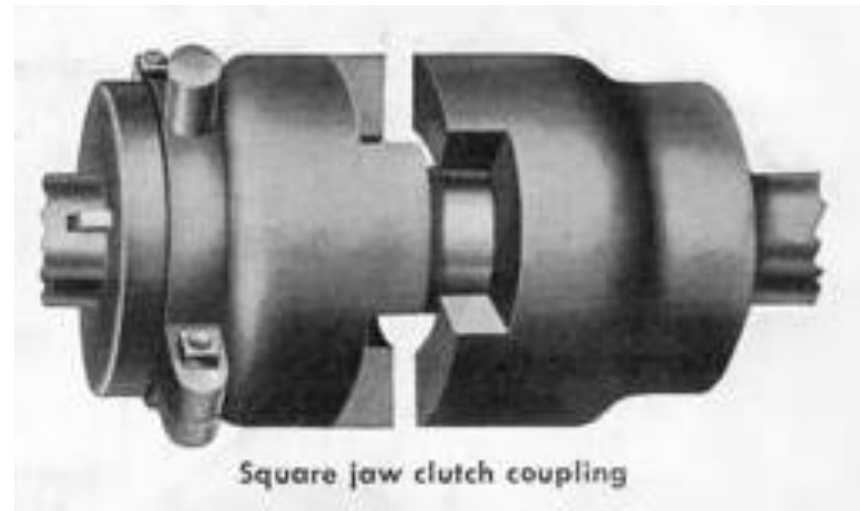
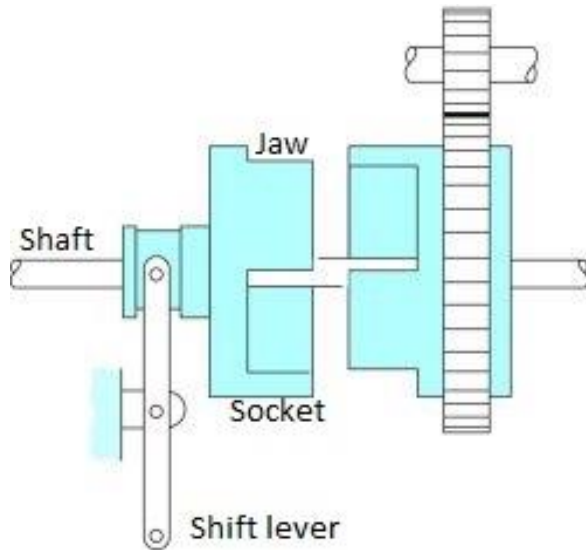
Final Condition

- Both members are at rest with no relative motion

Classifications



Positive Contact Clutches



- Two halves carrying projections or halves
- One half is fixed and the other can move along the axis
- Jaws of moving half enter into socket of mating half
- Eg. Square jaw, spiral jaw

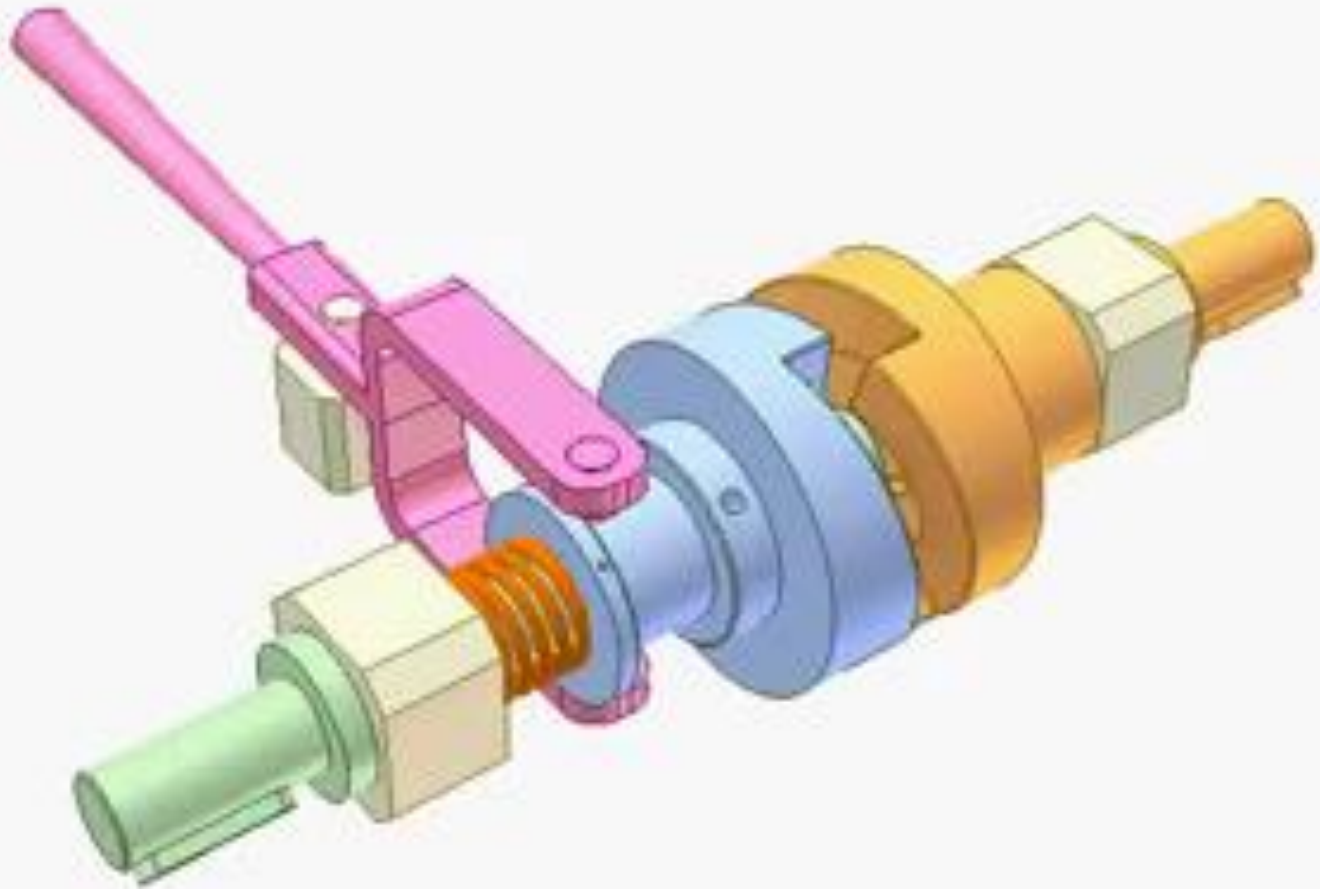
Advantages

- No slip and positive engagement
- No heat during engagement/disengagement

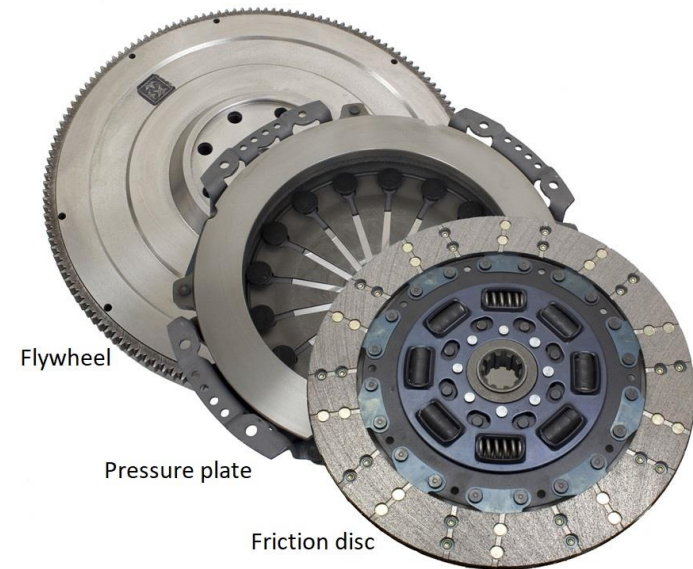
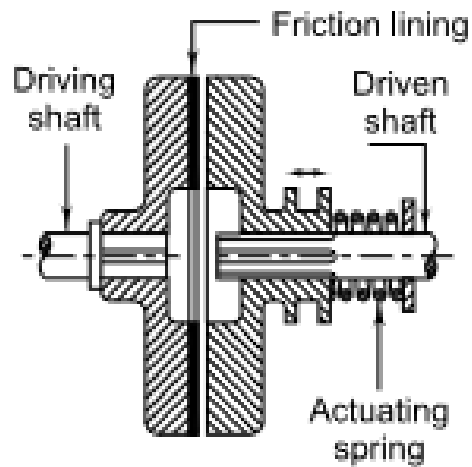
Drawbacks

- Engagement only when stationary or rotate at very low speed
- High speed engagement results in shocks

Positive Contact Clutches

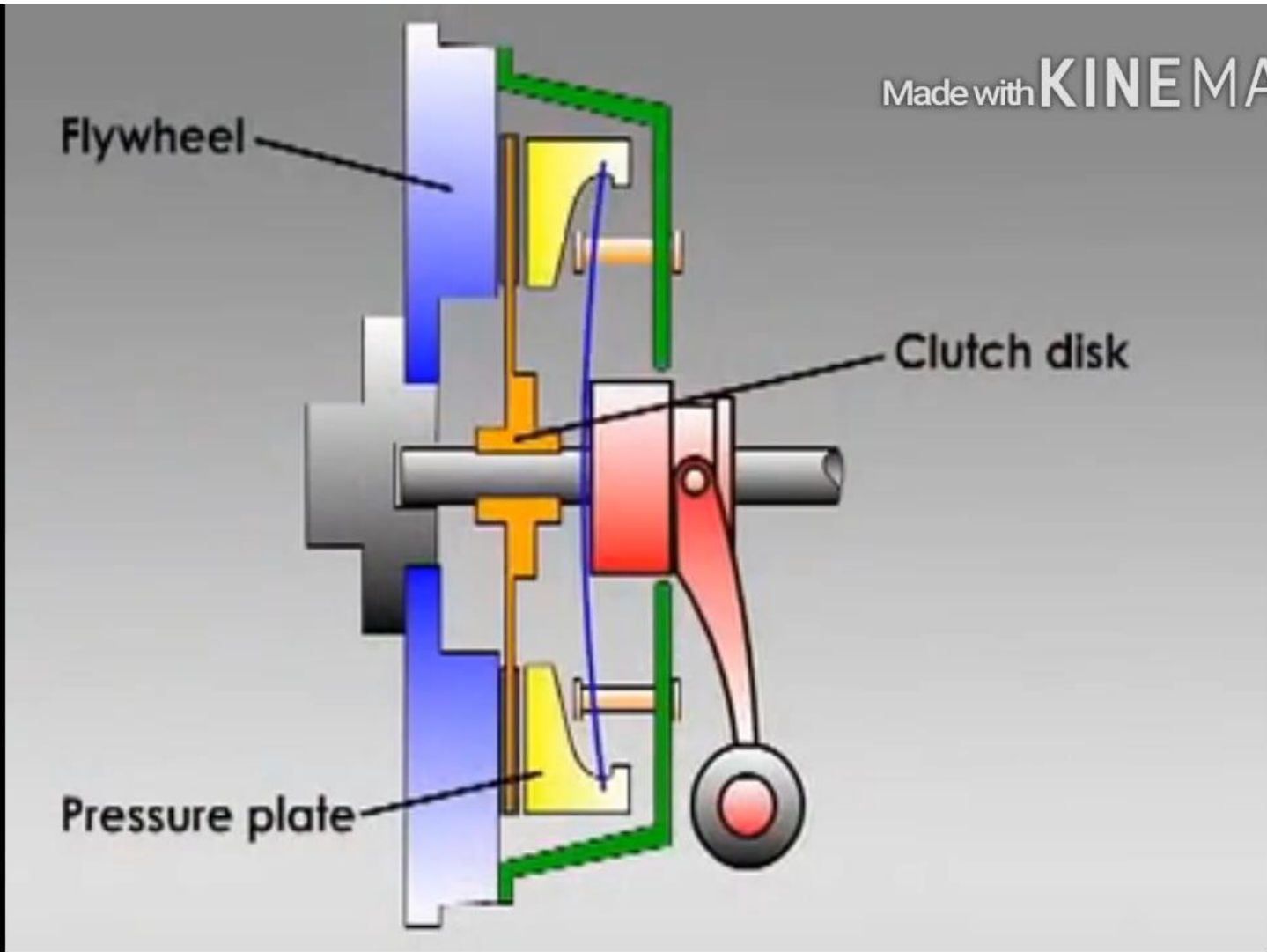


Friction Clutch: Single plate



- Consists of **two flanges**, one rigidly **keyed** to driving shaft and one connected to driven shaft with **splines**
- **Actuating force** is given by **spring**
- **Power transmission** between driving and driven flange is **through friction**
- **Fork** inserted on the collar of the driven flange **for axial movement** of driven flange

Friction Clutch: Single plate



Friction Clutch: Single plate

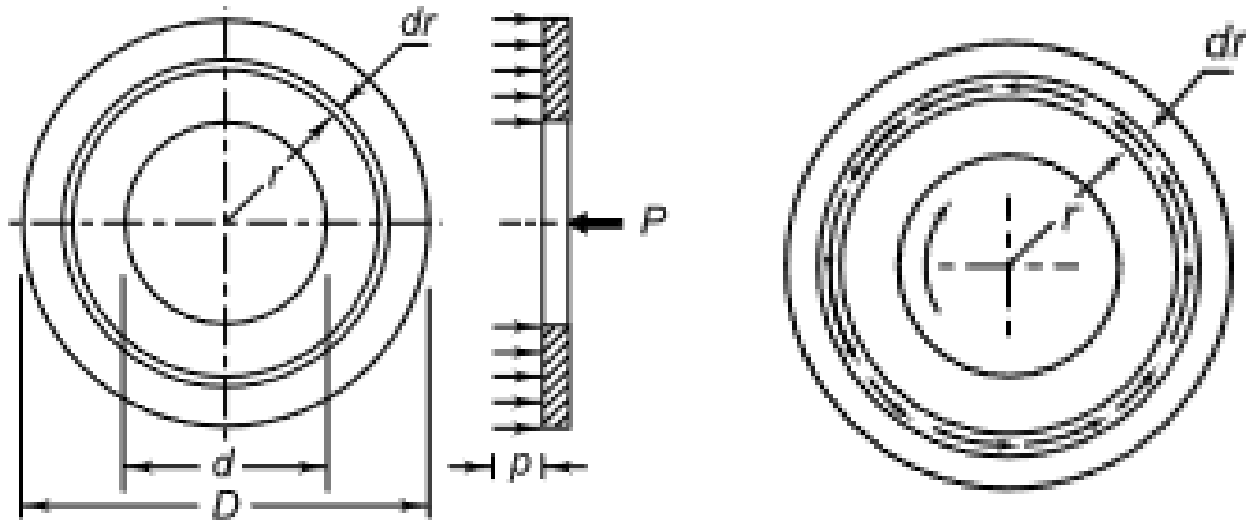
Advantages

- **Smooth** engagement
- Slip only during engagement
- Acts as **safety** device

Design Considerations

- Selection of **type** of clutch
- Selection of **friction material** of contact surfaces
- Designing for sufficient **torque capacity**
- **Smooth** engagement/disengagement
- Provision for **holding contact surface** without any external assistance
- **Low weight** of **rotating parts** to reduce inertia forces
- Provision for taking or **compensating wear**
- Provision for **carrying away the heat generated**

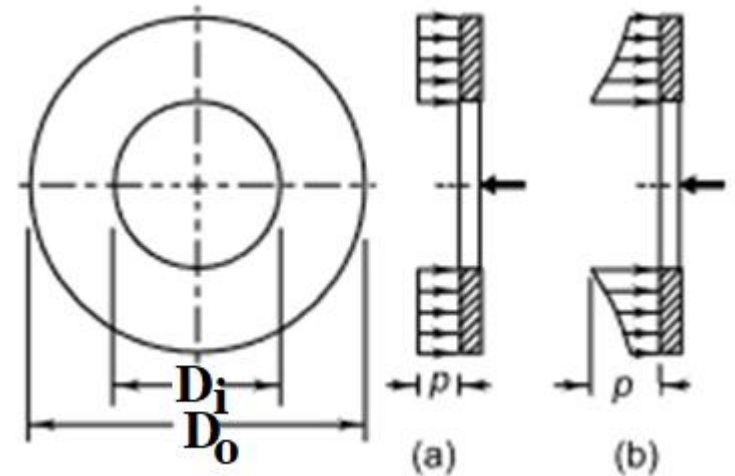
Single plate clutch: Torque transmitting capacity



D	: Outer diameter of friction disc (mm)	:	D_o
d	: Inner diameter of friction disc (mm)	:	D_i
p	: Intensity of pressure at radius r (N/mm^2)	:	p
P	: Operating axial force (N)	:	F_{ap}
M_t	: Torque transmitted (N-mm)	:	T
μ	: Friction coefficient	:	μ
r	: radius of the element dr from the centre of the disc	:	r

Single plate clutch: Torque transmitting capacity

- Total axial load, $F_a = 2\pi \int_{D_i/2}^{D_o/2} p r dr$
- Total torque, $T = 2\pi\mu \int_{D_i/2}^{D_o/2} p r^2 dr$



Using uniform pressure theory (New clutch)

- Total axial load, $F_{ap} = \pi p \frac{(D_o^2 - D_i^2)}{4}$ eqn 13.9a
- Total torque, $T = \frac{1}{2} \mu F_{ap} D_{mp}$: $D_{mp} = \frac{2(D_o^3 - D_i^3)}{3(D_o^2 - D_i^2)}$ eqn 13.9b

Using uniform wear theory (Worn out clutch)

- Total axial load, $F_{aw} = \pi p_{max} D_i \frac{(D_o - D_i)}{2}$ eqn 13.9d
- Total torque, $T = \frac{1}{2} \mu F_{aw} D_{mw}$: $D_{mw} = \frac{1}{2} (D_o + D_i)$ eqn 13.9e

Comparison: Uniform Pressure & Wear Theory

Total torque, $T = \frac{1}{2} \mu F_a D_m$:

- $D_{mp} = \frac{2(D_o^3 - D_i^3)}{3(D_o^2 - D_i^2)}$ **Using uniform pressure theory**
- $D_{mw} = \frac{1}{2} (D_o + D_i)$ **Using uniform wear theory**

D_o	D_i	$\frac{2(D_o^3 - D_i^3)}{3(D_o^2 - D_i^2)}$	$\frac{1}{2} (D_o + D_i)$	% difference
140	80	112.72	110	2.4
200	100	155.56	150	3.5
200	180	190.18	190	0.08

Single plate clutch

Conclusions

- Uniform **pressure theory** applicable for **new friction lining**
- Uniform **wear theory** applicable for **worn-out friction lining**
- **Friction radius** for new clutch is slightly **greater** than worn-out clutches
- **Torque transmitting** capacity of new clutch **slightly higher**
- **Major life** of friction lining comes under **uniform wear** criterion
- Logical and **safer** to use **uniform wear theory** in design of clutches

Factors Considered for design

- Service factor: To compensate for the starting torque
- Location of clutch: Low torque capacity requirement
- Coefficient of friction:

Problem 1.1

A plate clutch consists of one pair of contacting surfaces. The inner and outer diameters of the friction disk are 100 and 200 mm respectively. The coefficient of friction is 0.1 and the permissible intensity of pressure is 1 N/mm^2 . Assuming uniform wear theory, calculate the power transmitting capacity of the clutch at 750 rpm. Calculate the power transmitting capacity of the clutch using uniform pressure theory.

Problem 1.2

An automotive clutch consists of two pairs of contacting surfaces with an asbestos friction lining. The torque transmitting capacity of the clutch is 550 N m. The coefficient of friction is 0.25 and the permissible intensity of pressure is 0.5 N/mm². Due to space limitations, the outer diameter of the friction disk is fixed as 250 mm. Using uniform wear theory, calculate

- The inner diameter of the friction disk: and
- The spring force required to keep the clutch in an engaged position

Problem 1.3

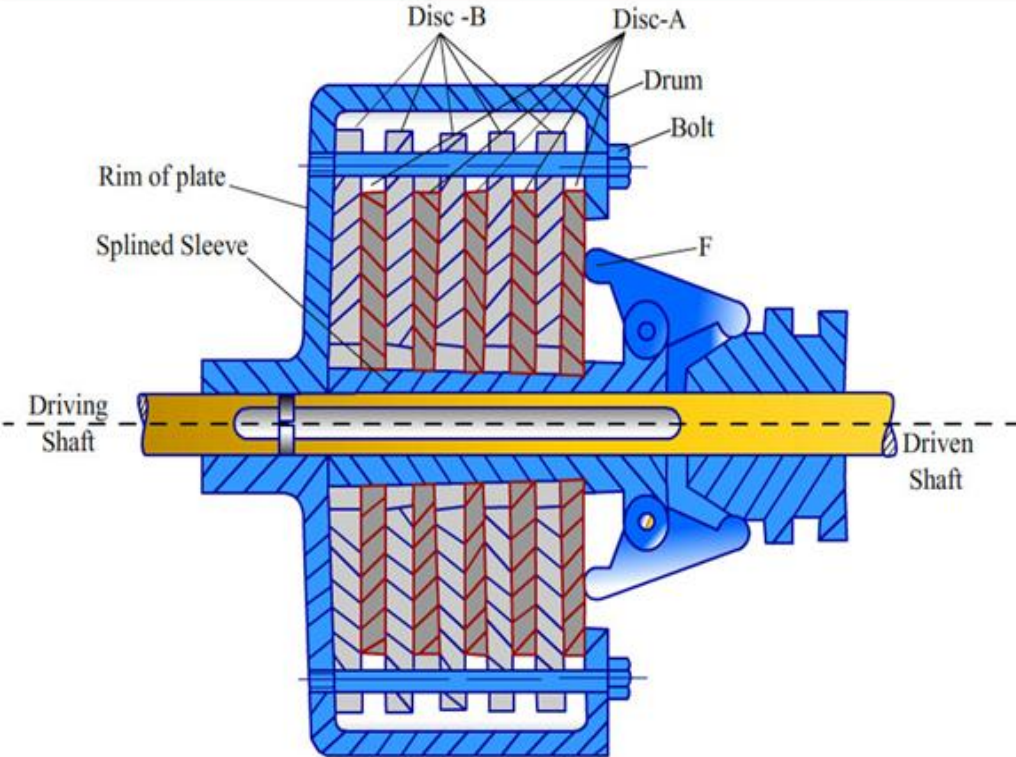
An automotive plate clutch consists of two pairs of contacting surfaces with asbestos friction lining. The maximum engine torque is 250 N-m. The coefficient of friction is 0.35. The inner and outer diameters of friction lining are 175 and 250 mm respectively. The clamping force is provided by nine springs, each compressed by 5 mm to give a force of 800 N, when the clutch is new.

- What is the factor of safety with respect to slippage when the clutch is brand new?
- What is the factor of safety with respect to slippage after initial wear has occurred?
- How much wear of friction lining can take place before the clutch will slip?

Problem 1.4

A single plate clutch consists of one pair of contacting surfaces. Because of space limitations, the outer diameter of the friction disk is fixed as D_2 . The permissible intensity of pressure is p and the coefficient of friction, μ . Assuming uniform wear theory, plot the variation of the torque transmitting capacity against the ratio of diameters (D_1/D_2). Show that the torque transmitting capacity of the clutch is maximum when (D_1/D_2) is equal to 0.577

Multi-Disk Clutches



Disc A:

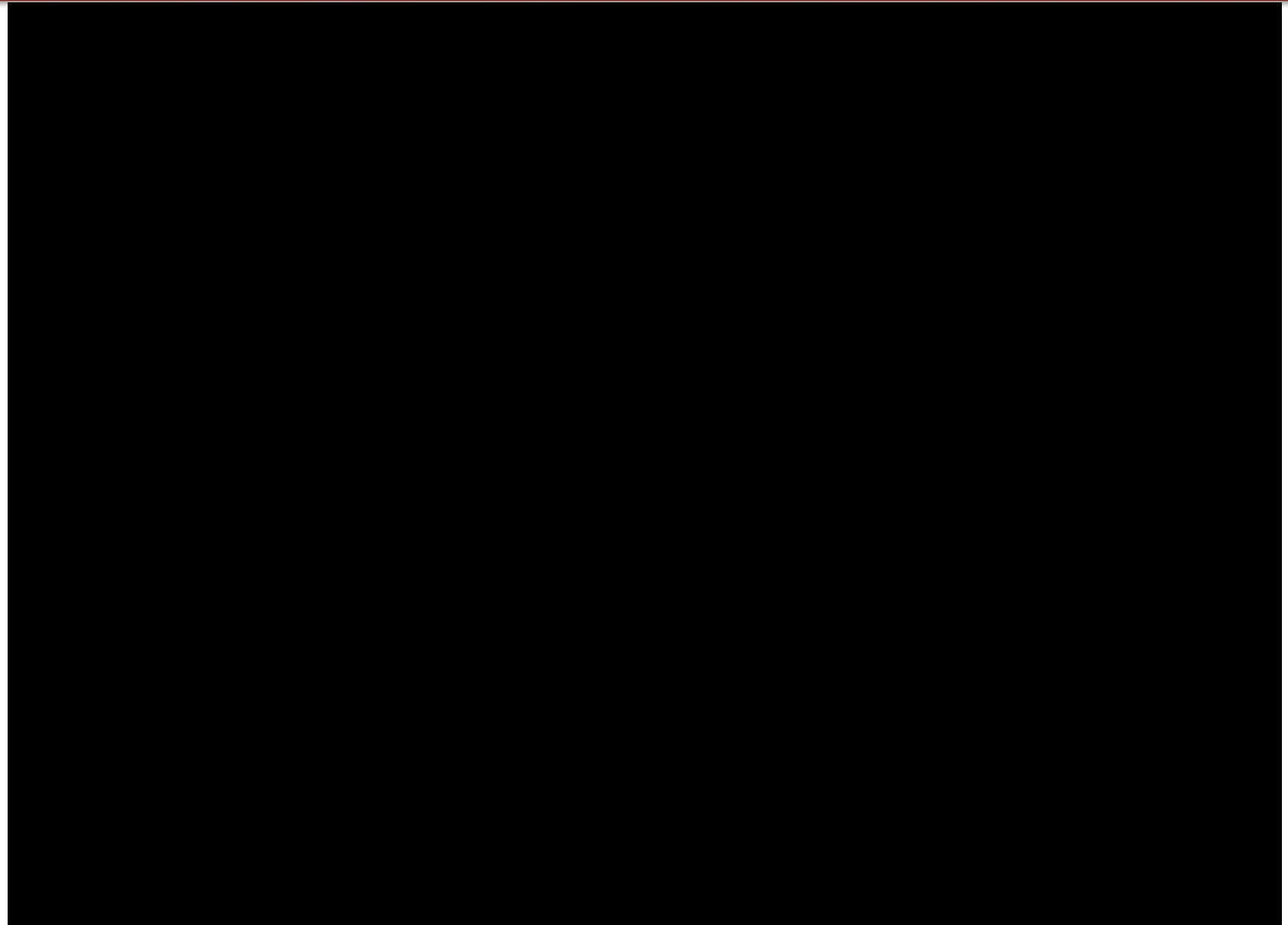
- Hardened steel
- Connected to driven shaft using splines
- Free to move in axial direction

Disc B:

- Made of Bronze
- Connected by four bolts through holes
- Bolts are rigidly fixed to a rotating drum keyed to driving shaft
- Axial force required are provided by means of springs



Multi-Disk Clutches



Governing Equations

- Total torque, $M_t = \frac{1}{2} i \mu F_a D_m$:

$$D_m = \frac{2(D_2^3 - D_1^3)}{3(D_2^2 - D_1^2)} \quad (\text{New clutch})$$

$$D_m = \frac{1}{2} (D_1 + D_2) \quad (\text{worn-out clutch})$$

- Number of friction surface, $i = i_1 + i_2 - 1$ (eqn 19-93)
- Number of driving disks, $i_1 = \frac{i}{2}$ (eqn 19-94)
- Number of driven disks, $i_2 = \frac{i}{2} + 1$ (eqn 19-95)

Difference Single Plate & Multi Plate

Parameters	Single plate	Multi plate
Contacting surface	Maximum of two	Can be more than two
Torque transmitting capacity	Less	More (decreases with cooling oil)
Heat generation	Less	More
Space Requirement	More	Less
Cooling oil	Not required	Required

Difference Wet and Dry clutch

Parameters	Wet clutch	Dry clutch
Coefficient of friction	Less	More
Torque capacity	Less	High
Moisture contamination	Not a serious issue	Proper sealing required
Heat dissipation	Lubricating oil	Difficult
Wear rate	Less	More
Clutch engagement	Smooth	Rough
Surface area	Reduced (grooving's)	More

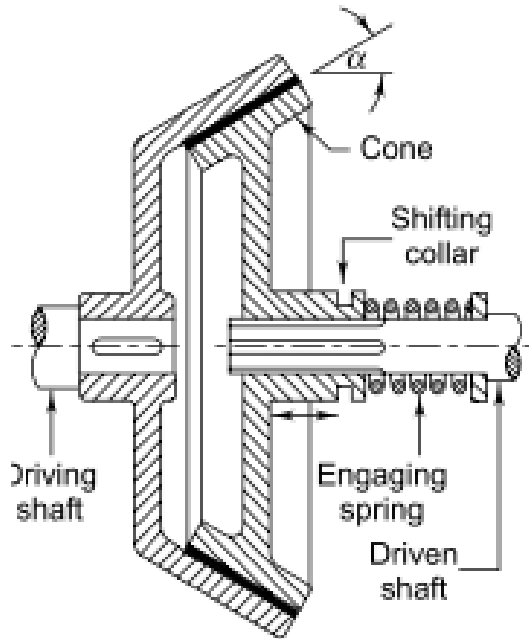
Problem 1.5

An oil immersed multi-disk clutch with cork sheet as the friction material is used on a scooter engine. The torque transmitted by the clutch is 10 N-m. The coefficient of friction between the cork sheet and the steel plate in the wet condition is 0.2. The permissible pressure on the cork sheet is 0.1 N/mm^2 . The inner and outer diameters of the friction lining are 65 and 95 mm, respectively. There are radial slots, on the friction surface for the circulation of the coolant, which reduces the effective friction area. To account for these slots, the number of contacting surfaces can be increased by 5%. Assuming uniform wear theory, calculate the required number of contacting surfaces.

Problem 1.6

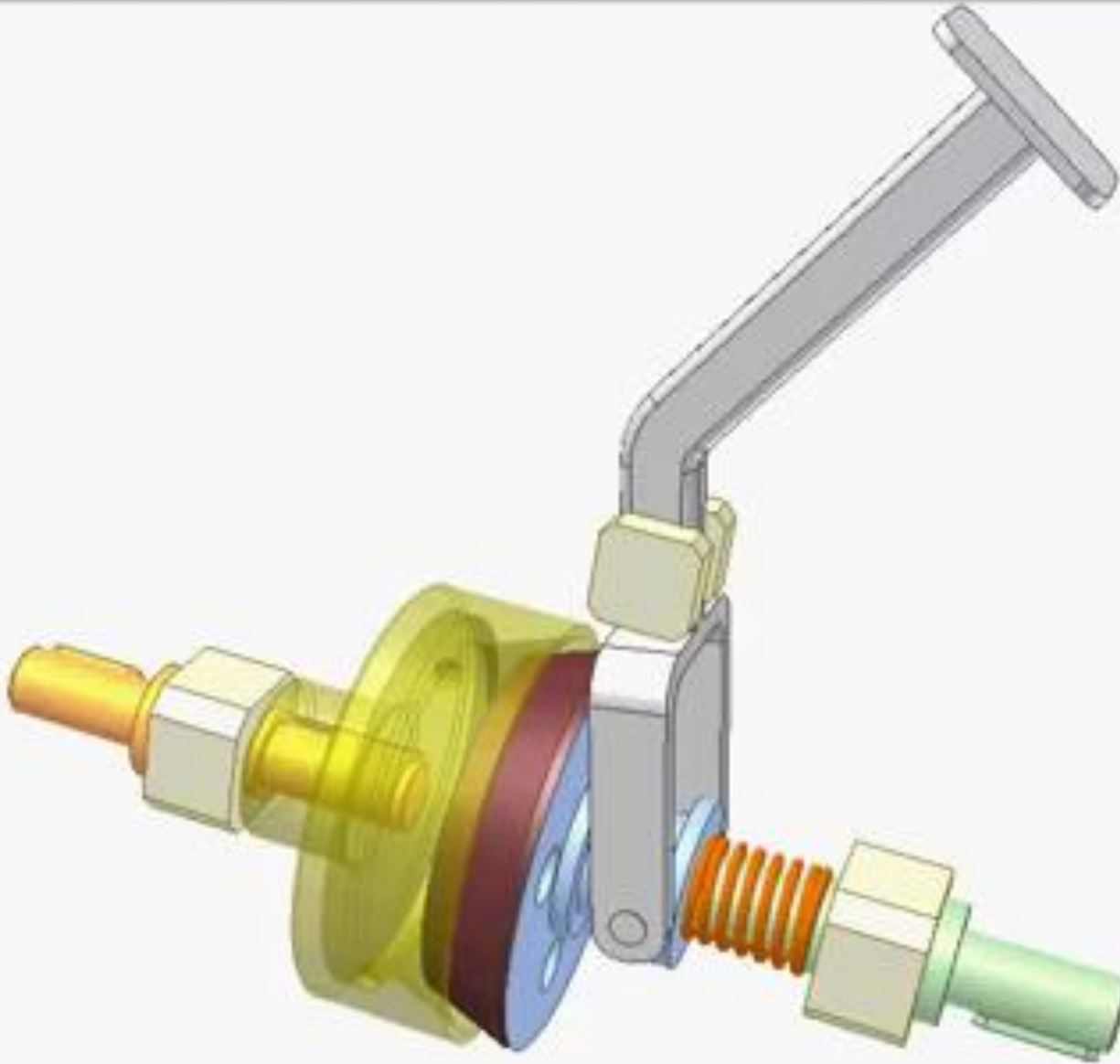
An oil immersed multi-disk clutch with moulded asbestos on one side and steel disks on the other, is used in an application. The torque transmitted by the clutch is 75 N-m. The coefficient of friction between the asbestos lining and the steel plate in the wet condition is 0.1. The permissible intensity of pressure on the asbestos lining is 500 kPa. The outer diameter of the friction lining is kept as 100 mm due to the space limitation. Assuming uniform wear theory, calculate the inside diameter of the disks, the required number of disks and the clamping force.

Cone Clutches

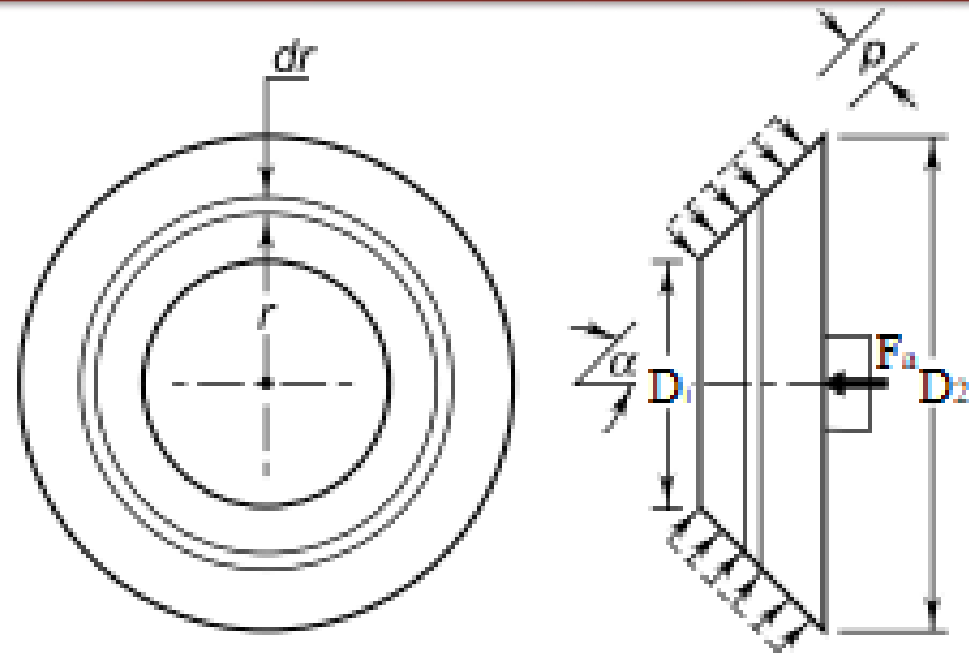


- Consists of inner and outer conical surfaces
- Outer cone keyed to driving shaft and inner cone connected to driven shaft using splines
- Wedge action causes considerable friction force with a small engaging force
- Recommended semi cone angle – 12.5°
- Strict co-axiality required

Cone Clutches



Cone Clutches: Governing Equations



Total axial load, $F_a = \pi p \frac{(D_2^2 - D_1^2)}{4}$ (Eqn. 19-86b)

$$F_a = \pi p D_1 \frac{(D_2 - D_1)}{2}$$
 (Eqn. 19-83)

- Total torque, $M_t = \frac{\mu F_a D_m}{2 \sin \alpha}$ (Eqn. 19-78)

$$D_m = \frac{2(D_2^3 - D_1^3)}{3(D_2^2 - D_1^2)}$$
 (New clutch)

$$D_m = \frac{1}{2}(D_1 + D_2)$$
 (worn-out clutch)

Observations: Torque Equation

- Torque capacity is inversely proportional to $\sin \alpha$
- $\alpha < \phi$, can cause self engagement
- For $M_t = \text{constant}$, $F_a \propto \sin \alpha$, small axial force

for decrease in semi cone angle

- $$\frac{(M_t)_{\text{cone}}}{(M_t)_{\text{plate}}} = \frac{1}{\sin \alpha} = \frac{1}{\sin 12.5} = 4.62$$

Friction materials: Desirable Properties

- High coefficient of friction
- Constant coefficient of friction for a wide range of temperature
- Good thermal conductivity
- Unaffected by environmental conditions
- Resistance to abrasive and adhesive wear
- Good resilience to provide good distribution of pressure at contacting surface

Friction materials: Types

Woven



Asbestos fibre woven around brass, copper or zinc wires and impregnated with rubber or asphalt

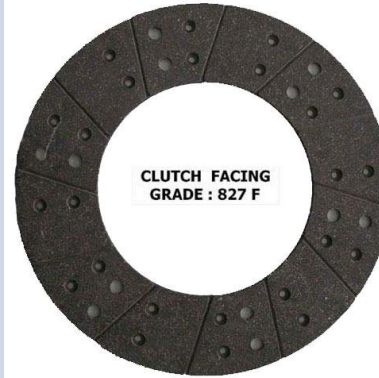
Flexible

Higher friction coefficient

Cheap cost

Higher wear rate

Moulded



Wet mixture of brass chips and asbestos poured into mould. Heated and pressed for curing time

Rigid

Less

Higher cost

Less

Friction materials: Four Basic Ingredients

- **Fibers:** Rigidity and strength (Steel wool or aramid replacing asbestos)
- **Filler:** Fills space between fibers and extend lining life (barytes, clay and calcium carbonate
For metallic lining: fine powder is used as filler)
- **Binder:** Glue that holds lining ingredients together (phenylformaldehyde)
- **Friction modifiers:** Control abrasive properties of lining (Brass and zinc)

Problem 1.7

A cone clutch with asbestos friction lining transmits 30 kW power at 500 rpm. The coefficient of friction is 0.2 and the permissible intensity of pressure is 0.35 N/mm^2 . The semi-cone angle is 12.5° . The outer diameter is fixed as 300 mm from space limitations. Assuming uniform wear theory, Calculate:

- The inner diameter
- The face width of the friction lining
- The force required to engage the clutch

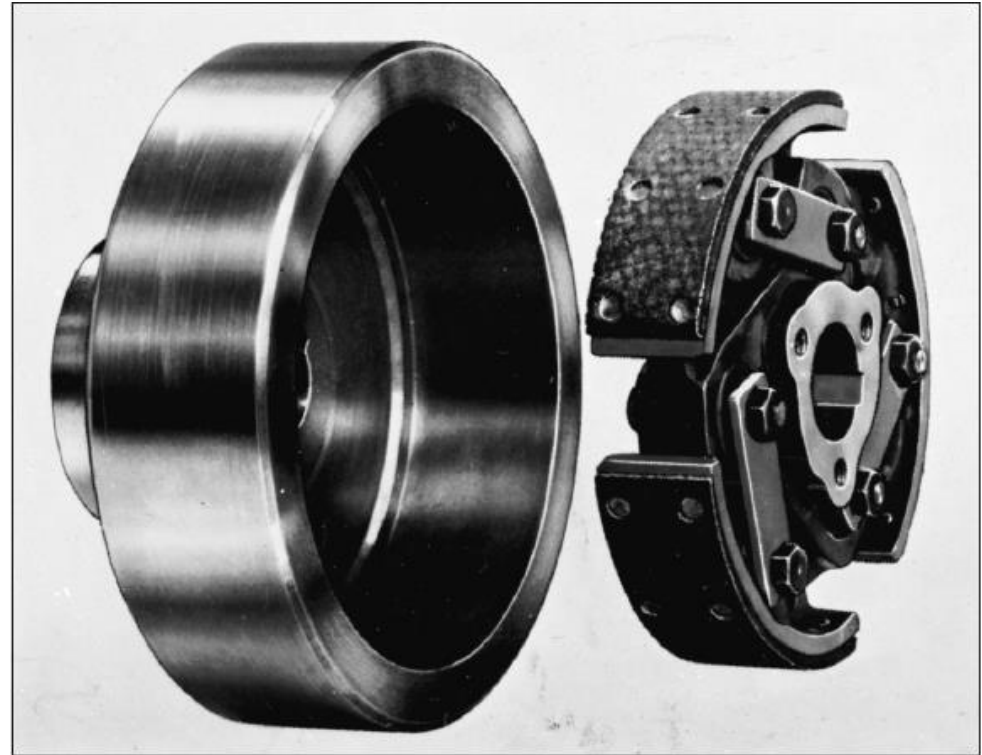
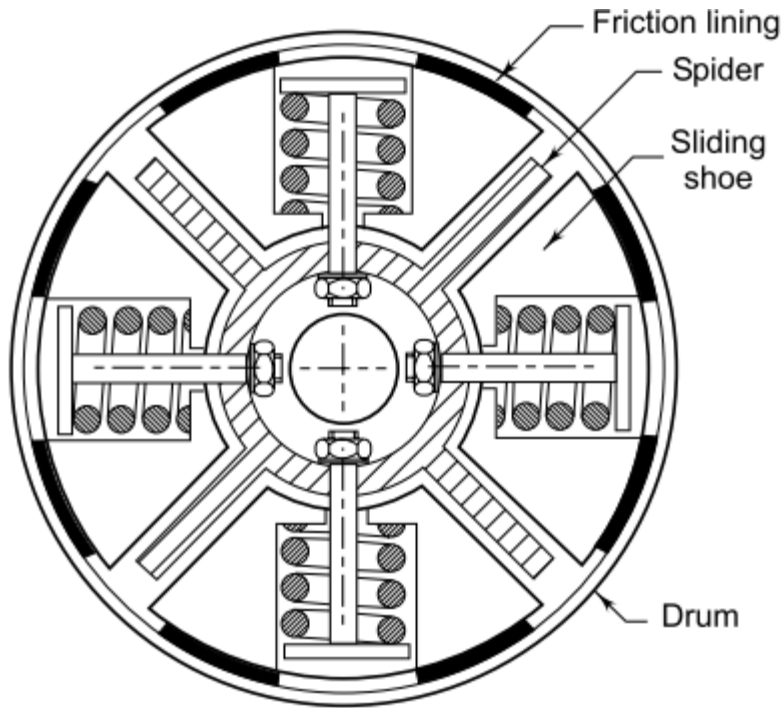
Problem 1.8

A cone clutch is used to connect an electric motor running at 1440 rpm with a machine which is stationary. The machine is equivalent to a rotor of 150 kg mass and radius of gyration as 250 mm. The machine has to be brought to the full speed of 1440 rpm from stationary condition in 40s. The semi-cone angle is 12.5° . The mean radius of the clutch is twice the face width. The coefficient of friction is 0.2 and the normal intensity of pressure between contacting surfaces should not exceed 0.1N/mm^2 .

Assuming uniform wear criterion, calculate:

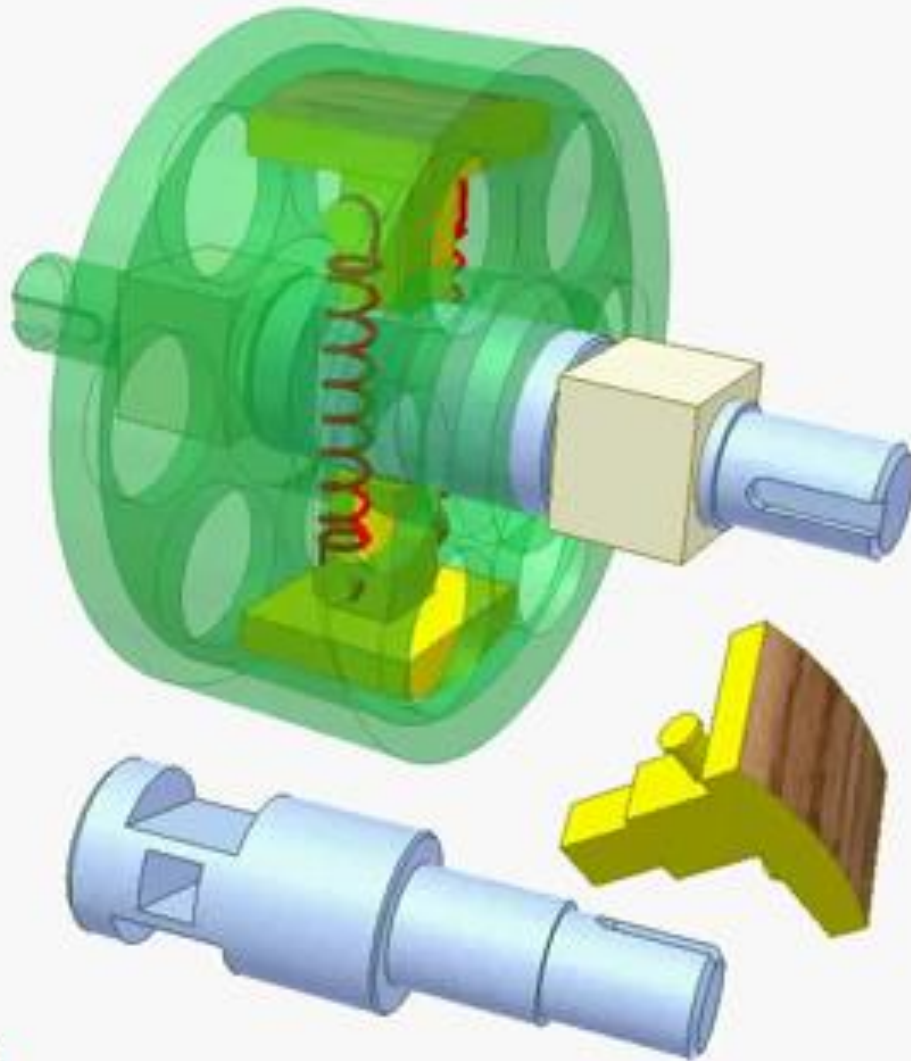
- The inner and outer diameter
- The face width of friction lining
- The force required to engage the clutch
- The amount of heat generated during each engagement of clutch

Centrifugal clutches

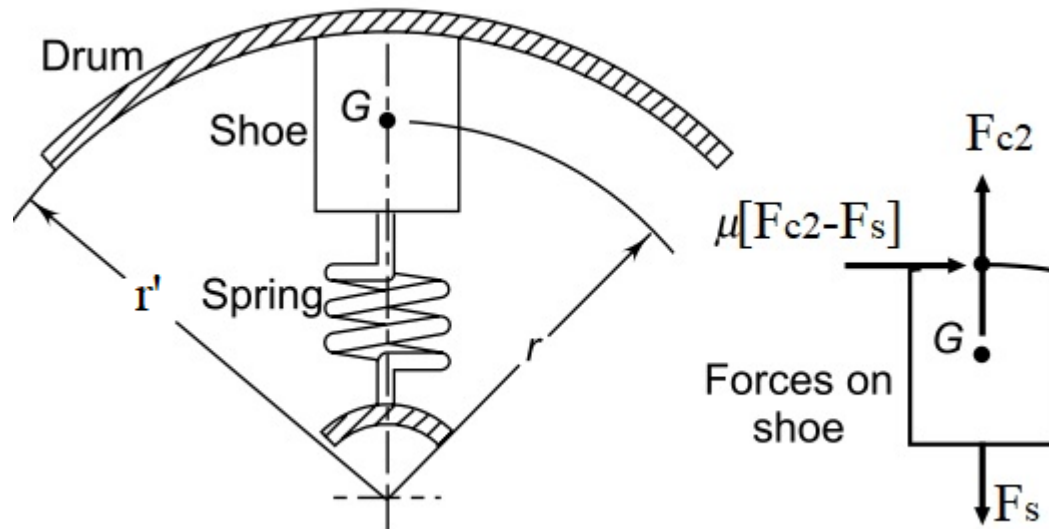


- Engages the load after a particular speed
- Helps in warm-up and acceleration without load
- Centrifugal force increases with speed which causes engagement of clutch

Centrifugal clutches: Animation



Centrifugal clutches



Notations used:

r' = inner radius of the drum (mm)

r = radius of the centre of gravity of shoe in engaged position (mm)

w = weight of each shoe (N)

F_c = Centrifugal force (N)

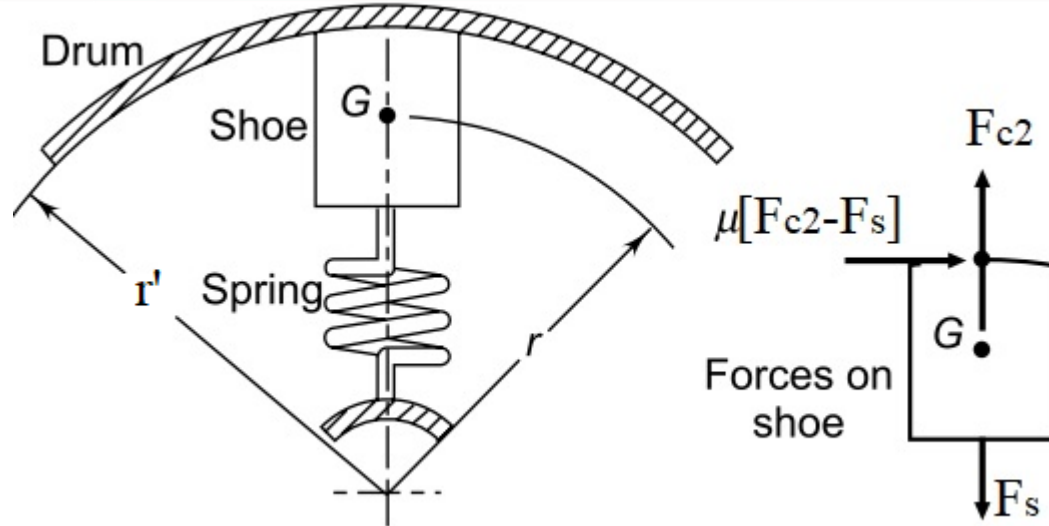
F_s = Spring force (N)

i = number of shoes

ω_2 = running speed (rad/s)

ω_1 = speed at which engagement starts (rad/s)

Centrifugal clutches: Governing Equations



Force during clutch engagement, $F_{c1} = \frac{w}{1000g} \omega_1^2 r$ Eqn. 19-114

Outside radial force, $F_{c2} = \frac{w}{1000g} \omega_2^2 r$ Eqn. 19-115

Spring force, $F_s = \frac{w}{1000g} \omega_1^2 r$

Net force on drum, $F_c = \frac{w}{1000g} (\omega_2^2 - \omega_1^2) r$ Eqn. 19-116a,b

Friction force, $F_f = \mu \frac{w}{1000g} (\omega_2^2 - \omega_1^2) r$

Friction torque, $M_t = \mu \frac{w}{1000g} (\omega_2^2 - \omega_1^2) r r'$ Eqn. 19-118

Problem 1.9

A centrifugal clutch, transmitting 20kW at 750 rpm consists of four shoes. The clutch is to be engaged at 500 rpm. The inner radius of the drum is 165 mm. The radius of the centre of gravity of the shoes is 140 mm, when the clutch is engaged. The coefficient of friction is 0.3, while the permissible pressure on friction lining is 0.1 N/mm². Calculate:

- The mass of each shoe
- The dimensions of friction lining

Problem 1.10

A centrifugal clutch consists of four shoes, each having a mass of 1.5 kg. In the engaged position, the radius to the centre of gravity of each shoe is 110 mm, while the inner radius of the drum is 140 mm. The coefficient of friction is 0.3. The pre-load in the spring is adjusted in such a way that the spring force at the beginning of engagement is 700 N. The running speed is 1440 rpm. Calculate:

- The speed at which the engagement begins
- The power transmitted by the clutch at 1440 rpm

Problem 1.11

Design a single plate automobile clutch to transmit a maximum torque of 250 N-m at 2000 rpm. The outside diameter of the clutch is 250 mm and the clutch is engaged at 55 km/h. Find

- The inner diameter
- The number of revolutions of the clutch slip during engagement and
- Heat to be dissipated by the clutch for each engagement

Following additional data available are

- Engine torque during engagement = 100 N-m
- Mass of automobile = 1500 kg
- Diameter of the automobile wheel = 0.7
- Moment of inertia of combined engine parts, flywheel and input side of the clutch = 1 kg-m^2
- Gear reduction ratio at differential = 5
- Torque at rear wheels available for accelerating automobile = 175 Nm
- Coefficient of friction of clutch material = 0.3
- Permissible pressure = 0.13 N/mm^2

Problem 1.12

A single dry plate clutch is to be designed to transmit 7.5 KW at 900 rpm. Find

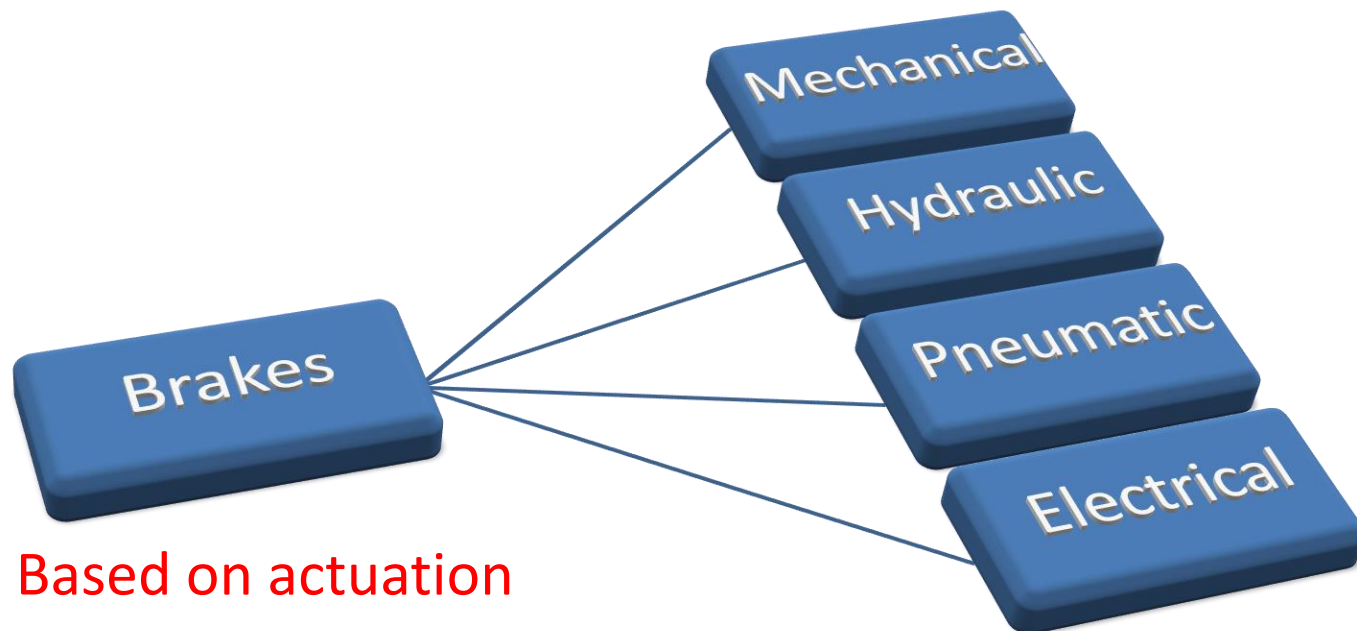
- Diameter of the shaft, Allowable Shear stress of shaft, $\tau_1 = 40 \text{ N/mm}^2$
- Mean radius and face width of friction lining assuming ratio of mean radius to the face width as 4
- Outer and inner radii of the clutch plate
- Dimensions of the spring assuming no of springs are 6 and spring index=6. The allowable shear stress of the spring wire may be taken as 420 MPa.
- Assume missing data

Brakes

A mechanical device, which is used to absorb the energy possessed by a moving system or mechanism by means of friction.

Energy absorbed can be:

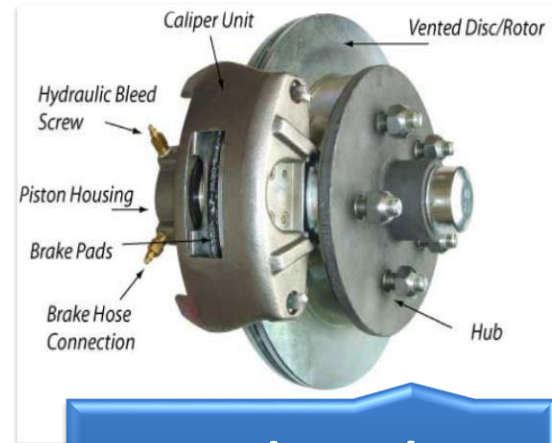
- **Potential energy**- Hoists and elevators
- **Kinetic energy**-Moving vehicle



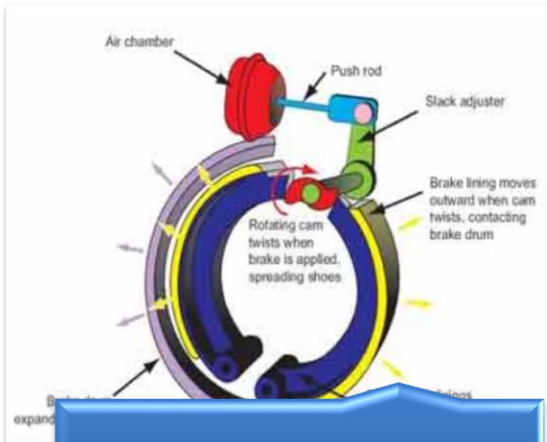
Brakes: Types



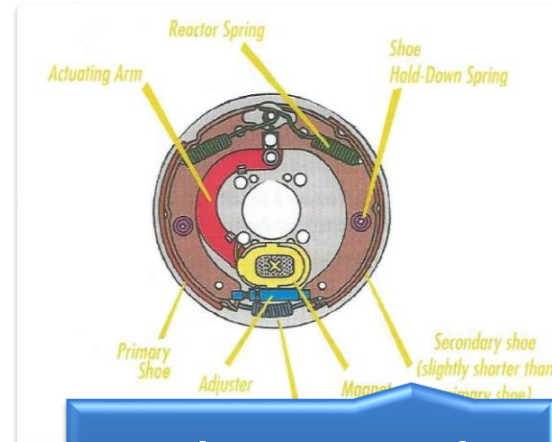
Mechanical



Hydraulic



Pneumatic



Electrical

Brakes: Energy Equations

For a mechanical system moving with a mass, m and velocity v_1 slowing down to v_2 during the period of braking

$$K.E = \frac{1}{2}m(v_1^2 - v_2^2) \quad (\text{Eqn 19:135a})$$

For a rotating body,

$$(K.E)_r = \frac{1}{2}I(\omega_1^2 - \omega_2^2) \quad (\text{Eqn 19:136})$$

In hoists

$$P.E = mgh$$

Total energy, E

$$E = K.E + (K.E)_r + P.E$$

Total energy = work done

$$E = M_t\theta$$

Problem 1.13

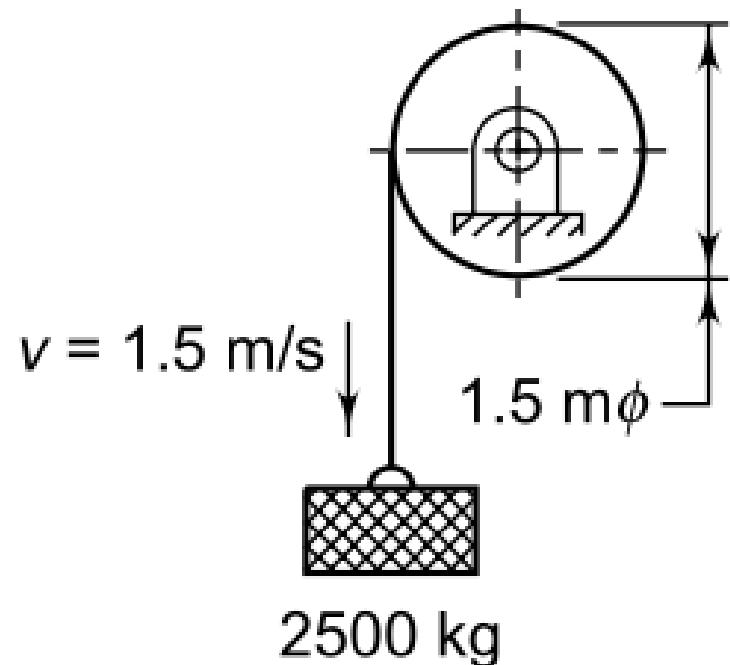
A four wheeled automobile car has a total mass of 1000 kg. The moment of inertia of each wheel about a transverse axis through its centre of gravity is 0.5 kg-m^2 . The rolling radius of the wheel is 0.35 m. The rotating and reciprocating parts of the engine and the transmission system are equivalent to a moment of inertia of 2.5 kg-m^2 , which rotates at 5 times the road-wheel speed. The car is travelling at a speed of 100 km/h on a plane road. When the brakes are applied, the car decelerates at 0.5 g . The are brakes on all four wheels. Calculate:

- The energy absorbed by each brake
- The torque capacity of each brake

Problem 1.14

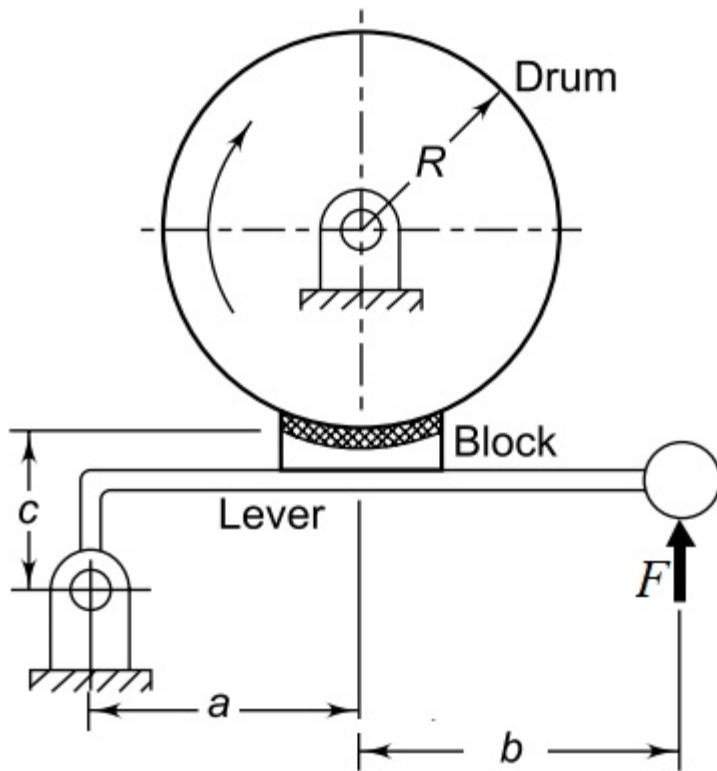
A mass of 2500 kg is lowered at a velocity of 1.5 m/s from the drum as shown in Figure. The mass of the drum is 50 kg and its radius of gyration can be taken as 0.7 m. On applying the brake, the mass is brought to rest in a distance of 0.5 m. Calculate:

- The energy absorbed by the brake
- The torque capacity of the brake

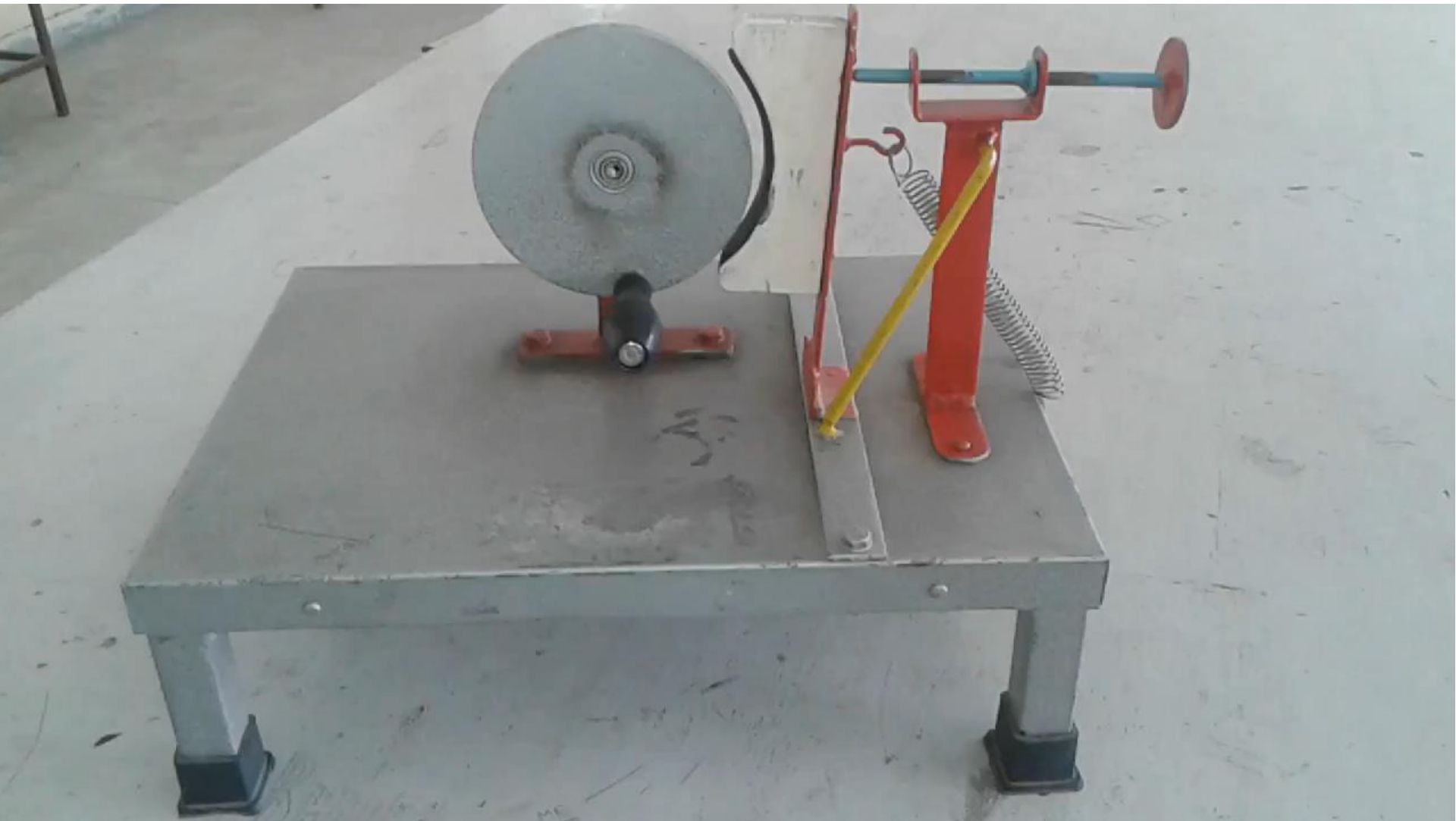


Block Brake: Short Shoe

- **Block brake:** A simple block pressed against rotating drum
- **Friction** between block and brake drum **causes retardation**
- Commonly employed in **railway wagons**
- **Angle of contact** between the block and brake drum is usually **small**
- Intensity of **pressure is uniform**



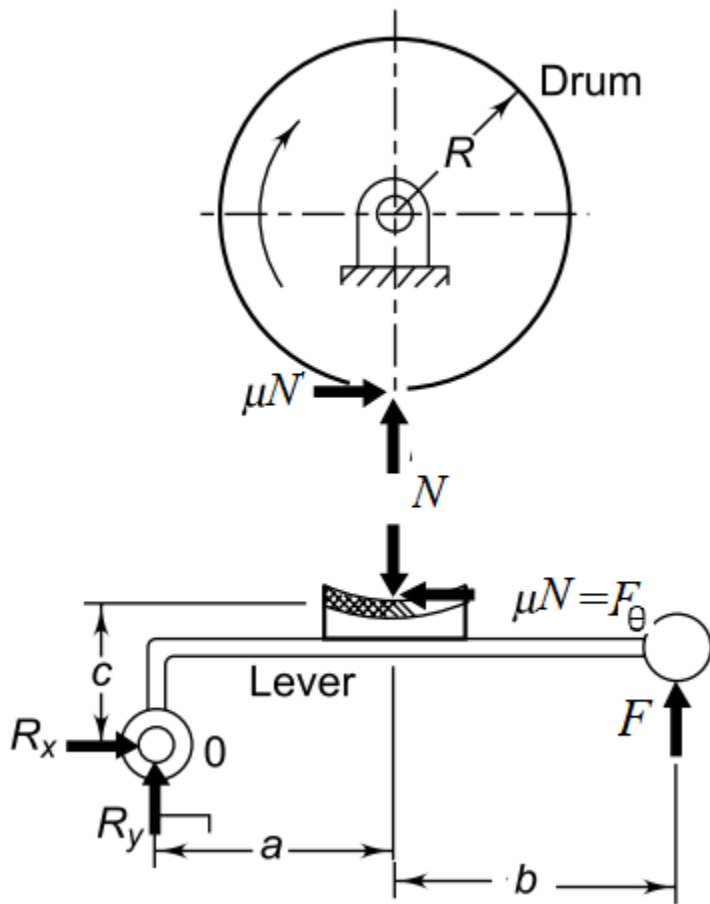
Block Brakes: Short Shoe



Brake Capacity

- Pressure between braking surfaces
- Contacting area of braking surface
- Radius of brake drum
- Coefficient of friction
- Ability of brake to dissipate heat

Block Brake: Short Shoe



$$F = \frac{F_{\theta} a}{a + b} \left(\frac{1}{\mu} - \frac{c}{a} \right)$$

(Eqn:19-145)

Assumptions:

- Block is rigidly attached to the lever
- Angle of contact small, resulting in uniform pressure distribution
- Brake drum rotating in clockwise direction

Notations:

- $M_t =$ braking torque
- $R =$ Radius of the drum
- $\mu N = F_{\theta} =$ Frictional force
- $p =$ permissible pressure intensity
- $l =$ length of the block
- $w =$ width of the block

$$M_t = \mu N R$$

$$N = p l w$$

Optimum width

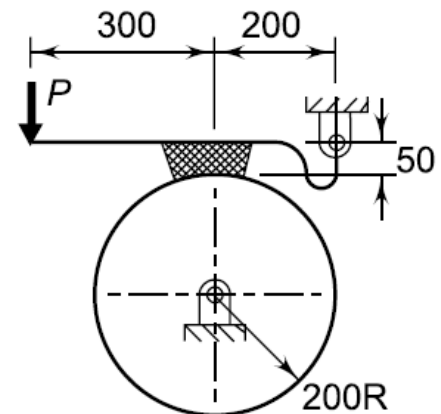
$$\frac{1}{4} \text{ drum diameter} < w < \frac{1}{2} \text{ drum diameter}$$

Problem 1.15

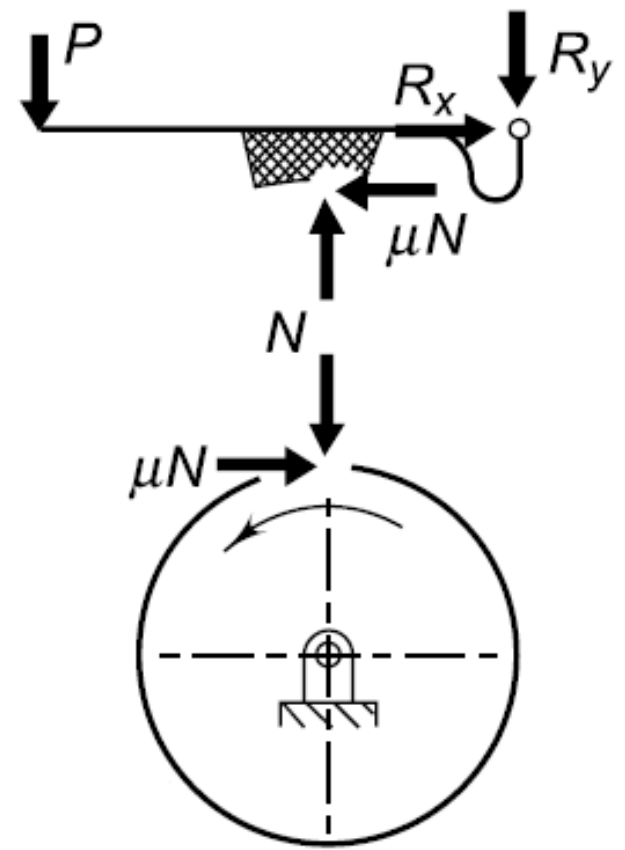
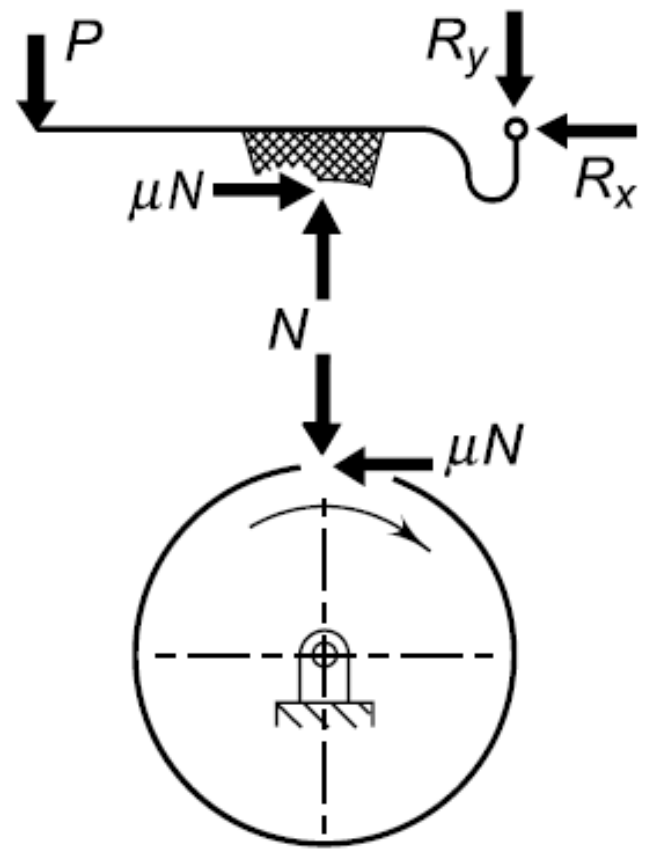
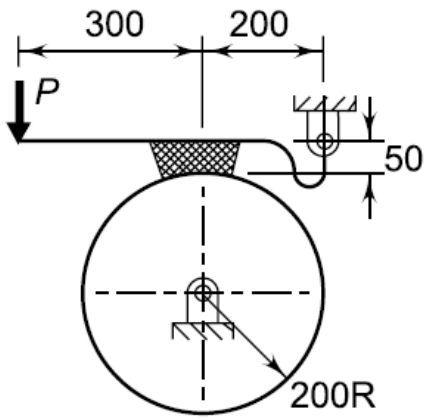
A single block brake with a torque capacity of 250 N-m is shown in Figure. The brake drum rotates at 100 rpm and the coefficient of friction is 0.35. Calculate:

- The actuating force and the hinge-pin reaction for clockwise rotation of the drum:
- The actuating force and hinge-pin reaction for anti clockwise rotation of the drum
- The rate of heat generated during the braking action
- The dimensions of the block, if the intensity of pressure between the block and brake drum is 1 N/mm^2 . The length of the block is twice its width.

State whether the brake is self-locking

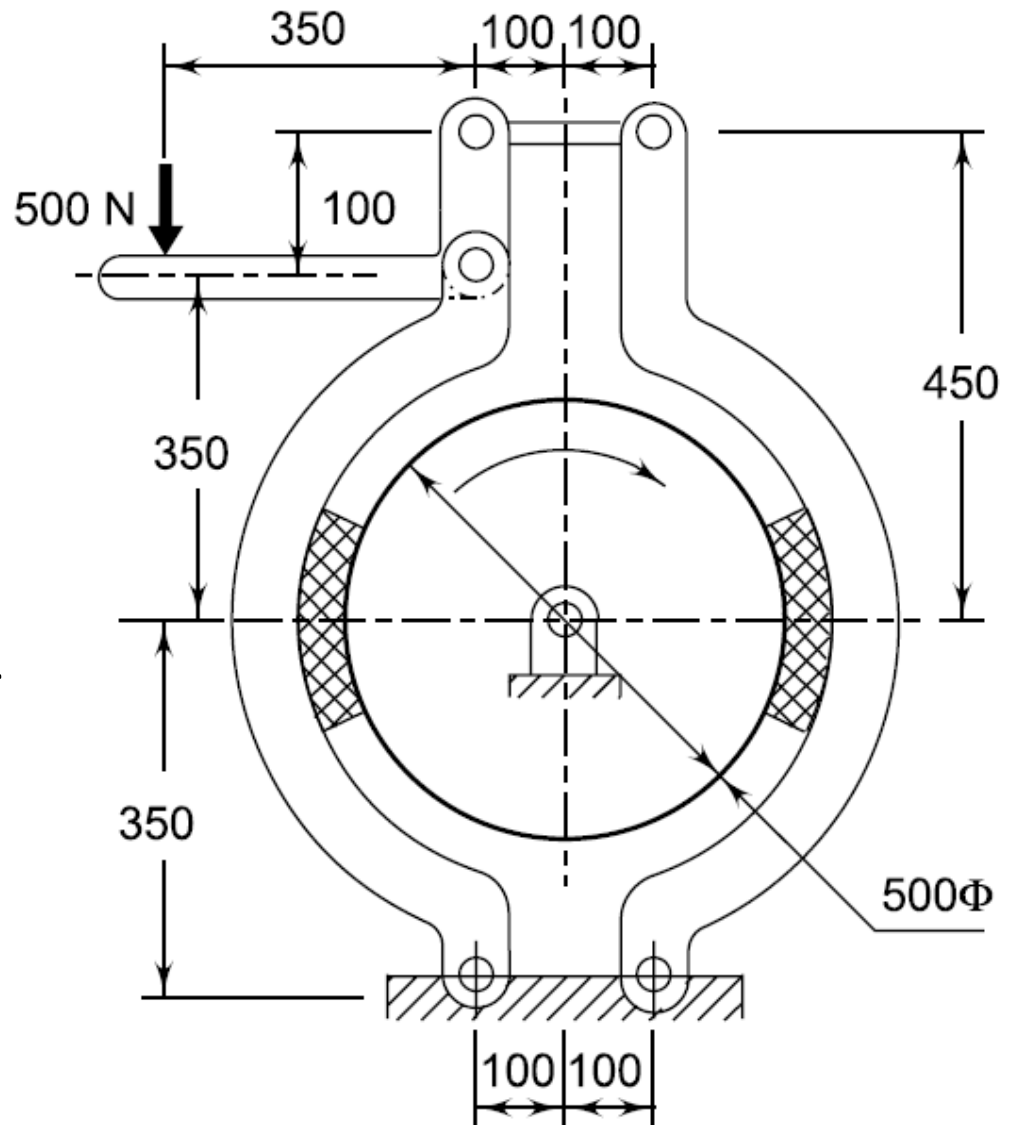


Problem 1.15

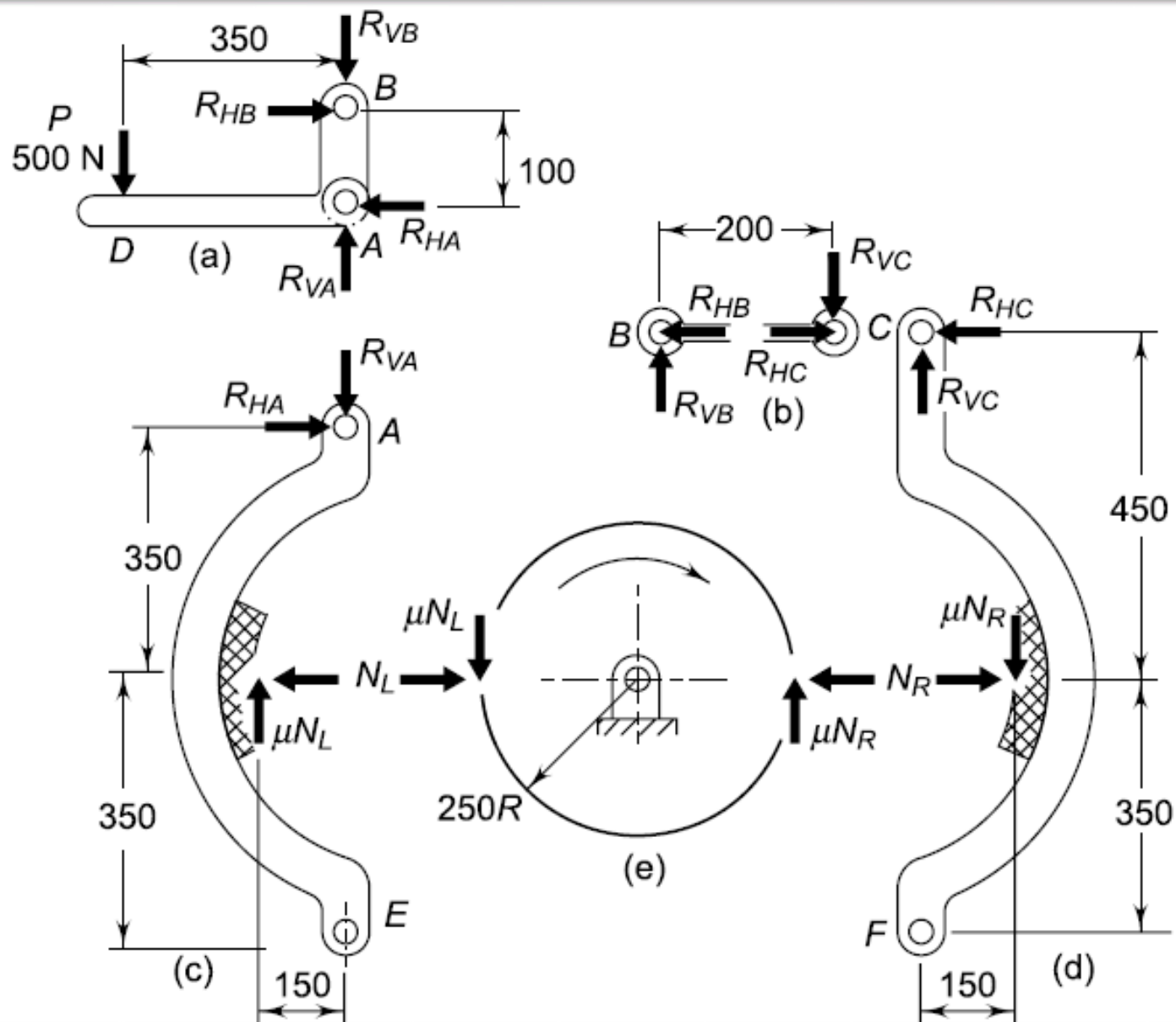


Problem 1.16

A double brake is shown in Figure. The brake drum rotates in clockwise direction and the actuating force is 500 N. The coefficient of friction between the blocks and the drum is 0.35. Calculate the torque absorbing capacity of the brake.



Problem 1.16



Problem 1.17

A block brake with a short shoe is shown in Figure. It is to be designed so that the product pv is limited to 2.

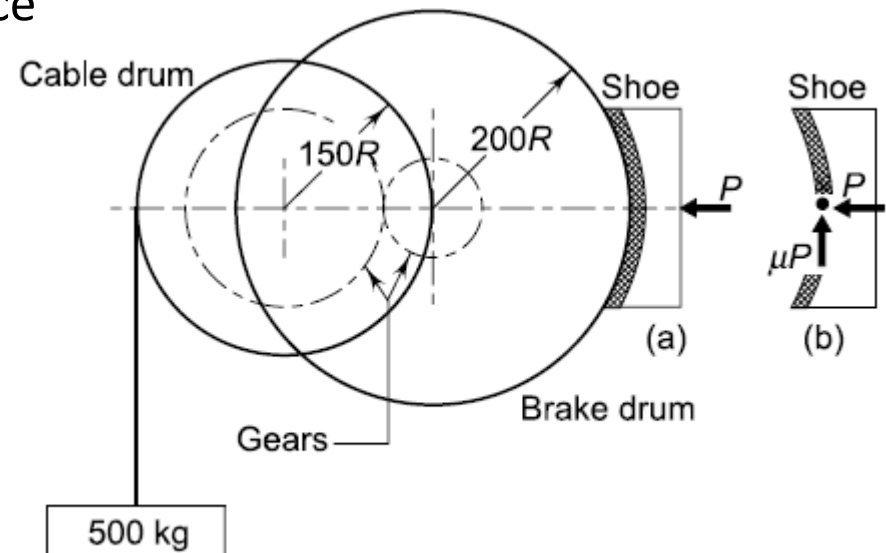
Where,

p = normal pressure between friction lining and the brake drum (N/mm²)

v = peripheral velocity of brake drum (m/s)

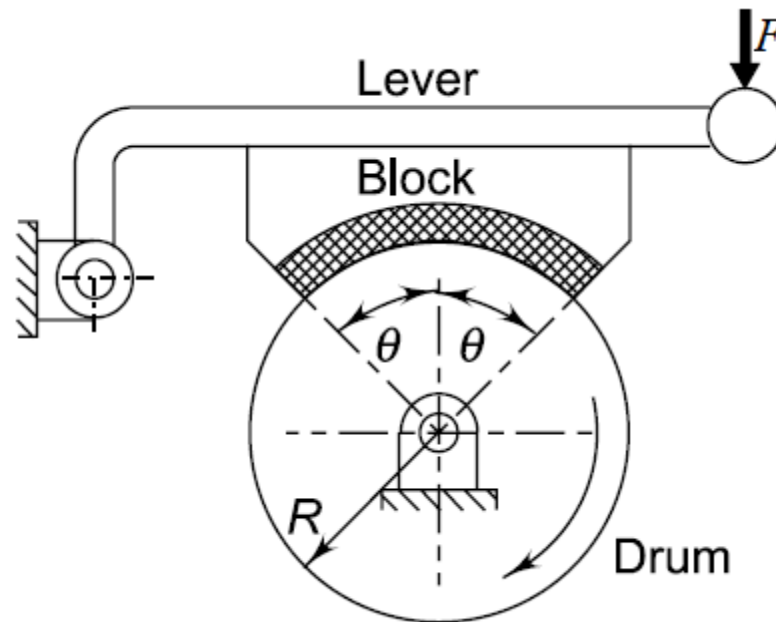
The coefficient of friction between the brake drum and the friction lining is 0.2. The cable drum is connected to the brake by means of a pair of spur gears. The brake drum rotates four times as fast as the cable drum. The permissible intensity of pressure on friction lining is 1N/mm². Calculate:

- (i) The magnitude of the brake shoe force
- (ii) The area of friction lining
- (iii) The uniform velocity at which the mass can be lowered



Block Brake: Long Shoe

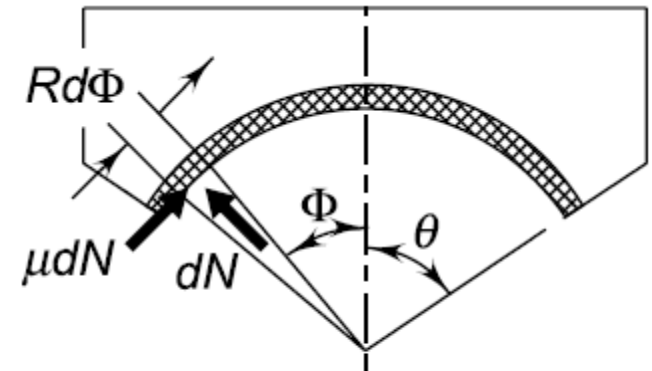
- Short Shoe: Angle of contact less than 45°
- Assumed normal reaction (N) and frictional force (μN) are concentrated at the midpoint of the shoe
- Not applicable for long shoe



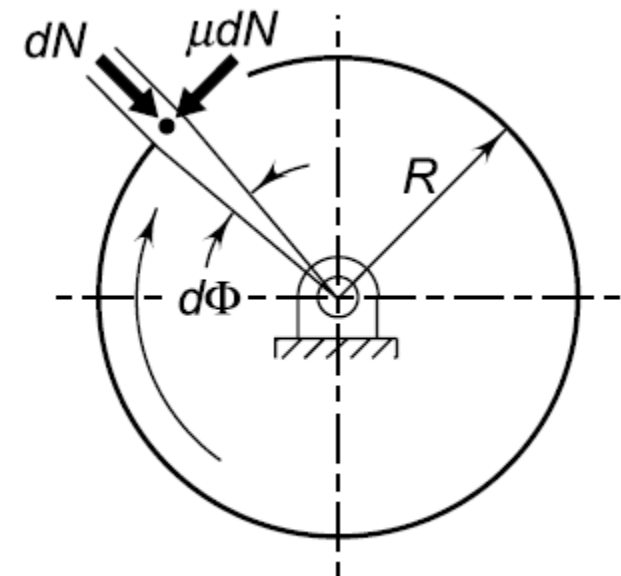
Block Brake: Long Shoe

Notations

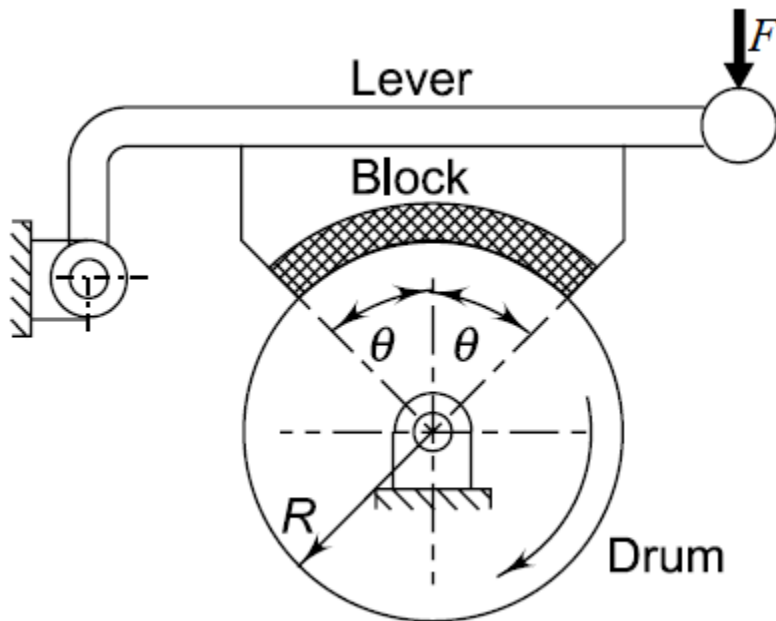
- θ - Semi block angle
- $d\phi$ - Element of friction lining
- ϕ –Location of friction lining element
- $Rd\phi w$ - Area of the element
- w - width of the friction lining parallel to the axis
- dN -Normal reaction on the element



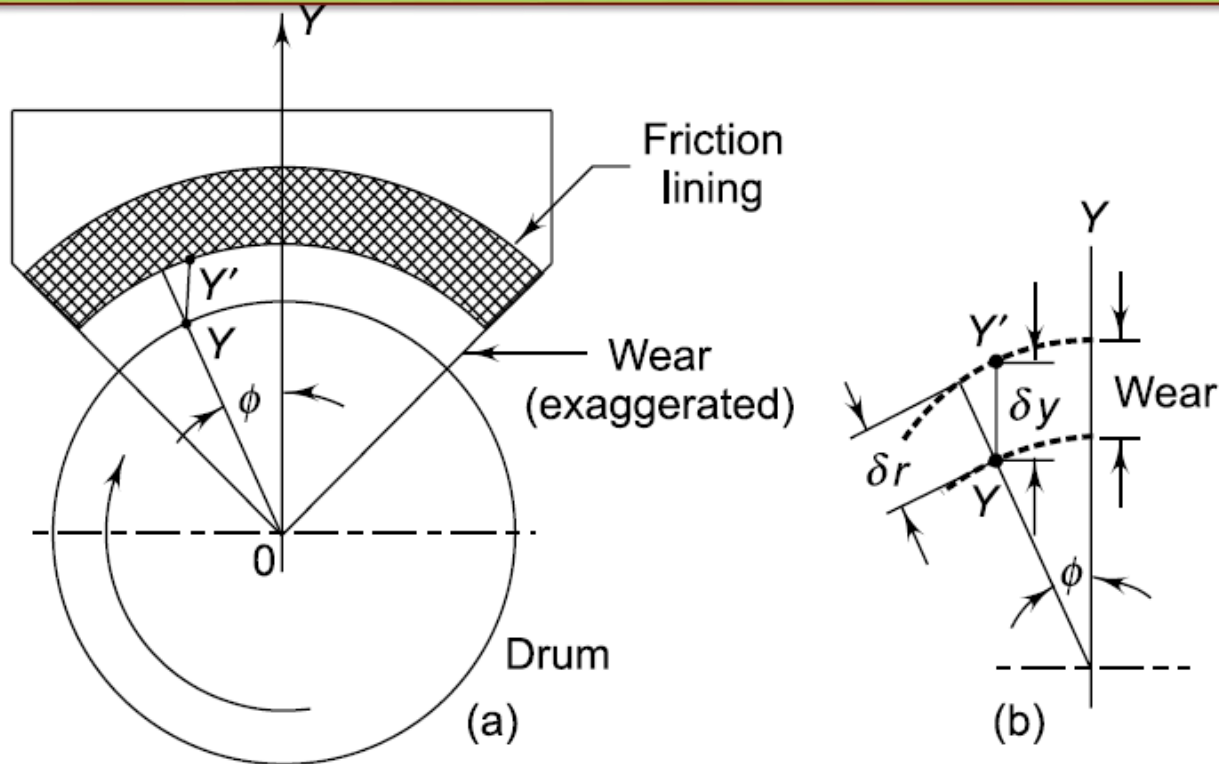
(a) Forces on lining



(b) Forces on drum



Block Brake: Long Shoe



Assumptions:

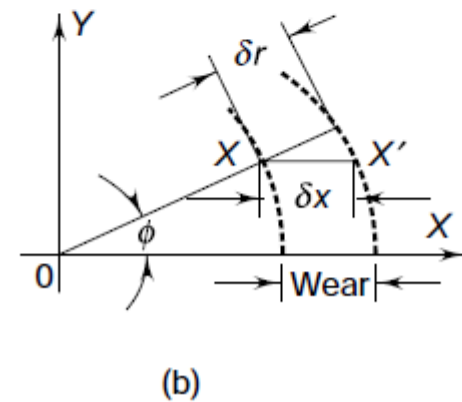
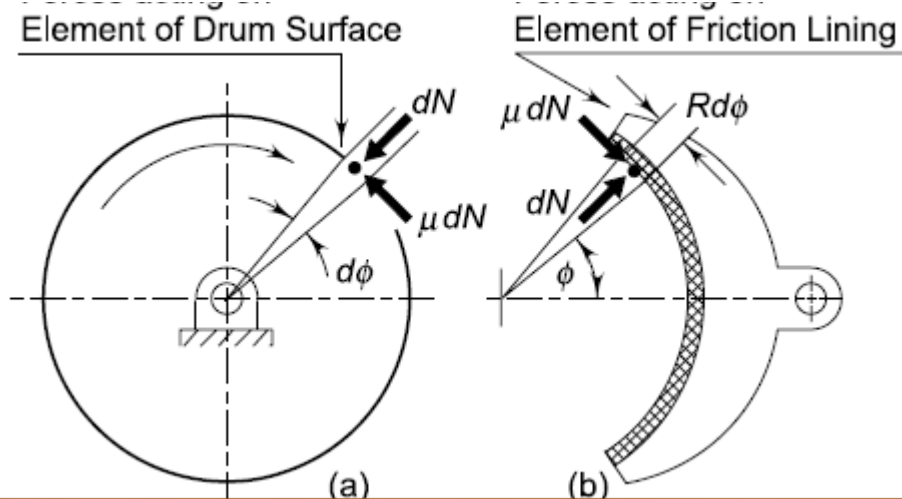
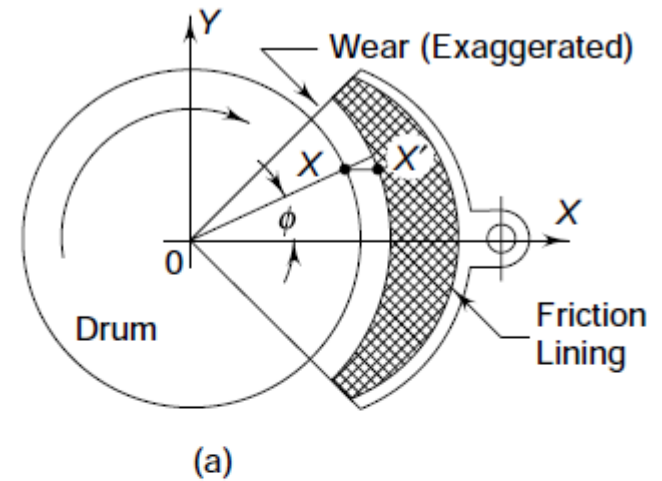
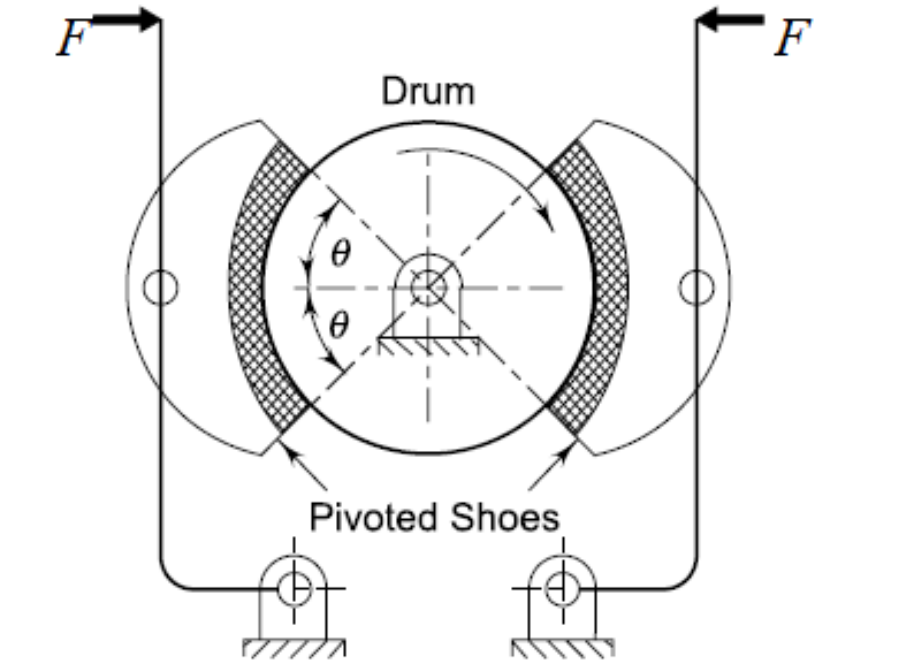
- Wear occurs at the brake shoe
- $\delta y = \text{wear in } y \text{ direction (constant)}$
- $\delta r = \text{wear in radial direction}$

$$M_t = \mu NR \left(\frac{4 \sin \theta}{2\theta + \sin 2\theta} \right)$$

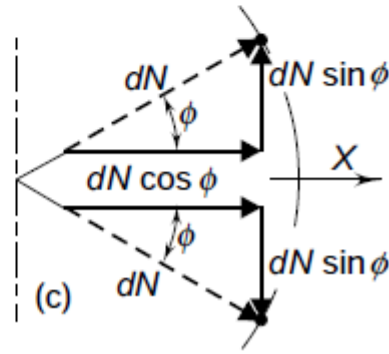
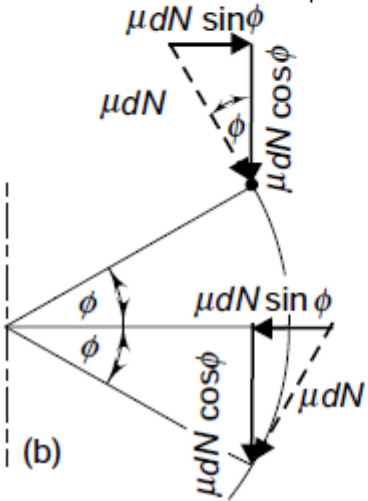
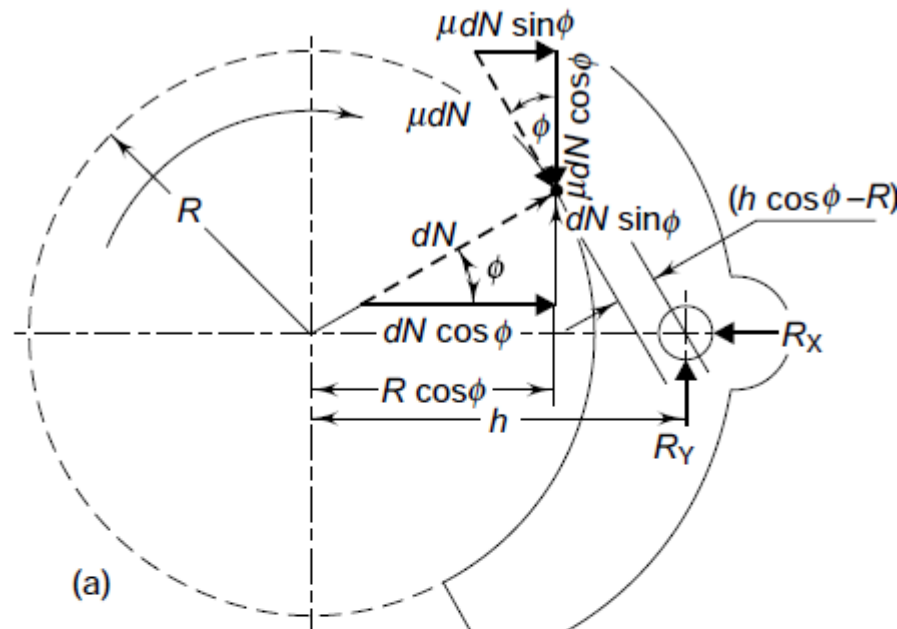
$$M_t = \mu F \frac{D}{2} \left(\frac{4 \sin \theta}{2\theta + \sin 2\theta} \right)$$

(Eqn 19-141)

Block Brake: Pivoted Long Shoe



Pivoted Long Shoe: Force Diagram



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

$$M_t = 2\mu R^2 w p_{max} \sin \theta$$

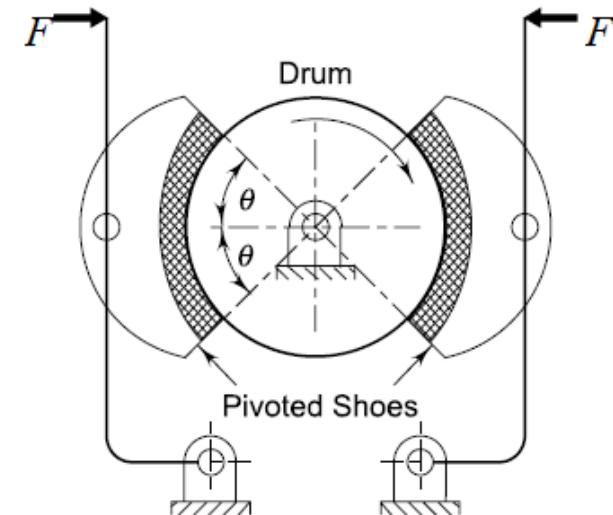
$$R_x = \frac{1}{2} R w p_{max} (2\theta + \sin 2\theta)$$

$$R_y = \frac{1}{2} \mu R w p_{max} (2\theta + \sin 2\theta)$$

Problem 1.18

A pivoted double-block brake, similar to that in Figure has two shoes, which subtend an angle of 100° . The diameter of the brake drum is 500 mm and the width of the friction lining is 100 mm. The coefficient of friction is 0.2 and the maximum intensity of pressure between the lining and the brake drum is 0.5 N/mm^2 . The pivot of each shoe is located in such a manner that the moment of the frictional force on the shoe is zero. Calculate:

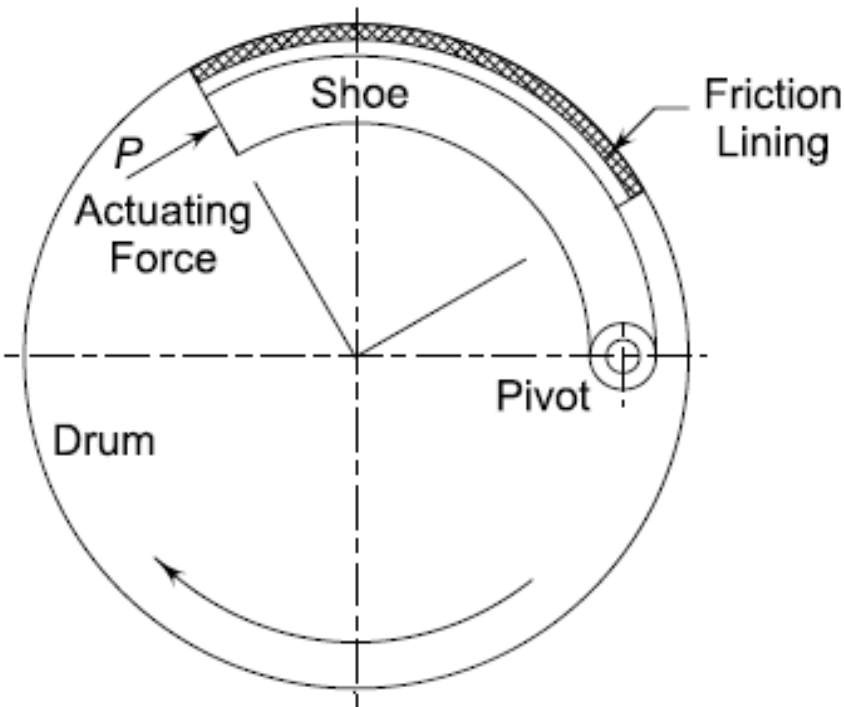
- The distance of the pivot from the axis of the brake drum
- The torque capacity of each shoe
- The reactions at the pivot



Internal Expanding Brake: Animation



Internal Expanding Brake

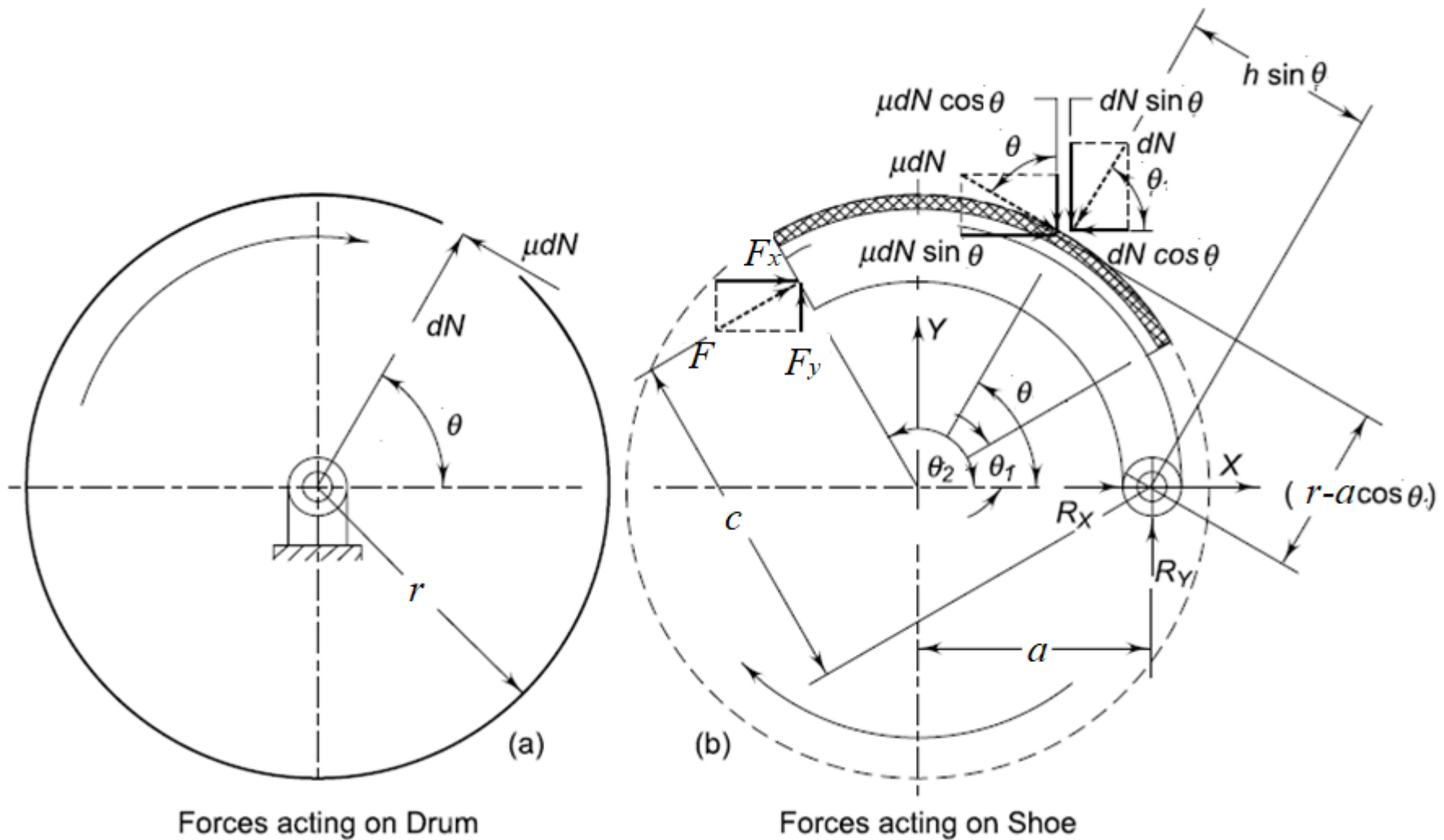


- A shoe pivoted at one end and subject to an actuating force
- Friction lining fixed on the shoe
- Complete assembly of shoe, lining and pivot is placed inside the brake drum
- Internal shoe brakes with two symmetrical shoes are used on all automobiles

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 shoes are rigid
- The centrifugal force acting on the shoe is negligible
- Coefficient of friction is constant

Internal Expanding Brake: Force Diagram



Internal Expanding Brake: Final Equations

Moment due to friction force about pivot

$$M_{t\mu} = \frac{\mu p_a r b}{\sin \theta_a} \int_{\theta_1}^{\theta_2} \sin \theta (r - a \cos \theta) d\theta = \frac{\mu p_a r b [4r(\cos \theta_1 - \cos \theta_2) - a(\cos 2\theta_1 - \cos 2\theta_2)]}{4 \sin \theta_a} \quad \text{Eqn 19-174a}$$

Moment due to normal force about pivot

$$M_{tn} = \frac{p_a r b a}{\sin \theta_a} \int_{\theta_1}^{\theta_2} \sin^2 \theta d\theta = \frac{p_a r b a [2(\theta_2 - \theta_1) - (\sin 2\theta_2 - \sin 2\theta_1)]}{4 \sin \theta_a} \quad \text{Eqn 19-175}$$

Frictional torque

$$M_t = \frac{\mu r^2 p_a b (\cos \theta_1 - \cos \theta_2)}{\sin \theta_a} \quad \text{Eqn 19-177}$$

For the clockwise rotation of the brake drum

$$P/F = \frac{M_{tn} - M_{t\mu}}{c} \quad \text{Eqn 19-176}$$

For the anti-clockwise rotation of the brake drum

$$P/F = \frac{M_{tn} + M_{t\mu}}{c} \quad \text{Eqn 19-180}$$

Internal Expanding Brake: Force Diagram

For forward motion of the vehicle

$$P/F = \frac{M_{tn} - M_{t\mu}}{c}$$

For backward motion of the vehicle

$$P/F = \frac{M_{tn} + M_{t\mu}}{c}$$

Advantages

- Simple construction
- More reliable
- Little maintenance
- Small braking force can produce a large braking torque
- Protection against foreign particles

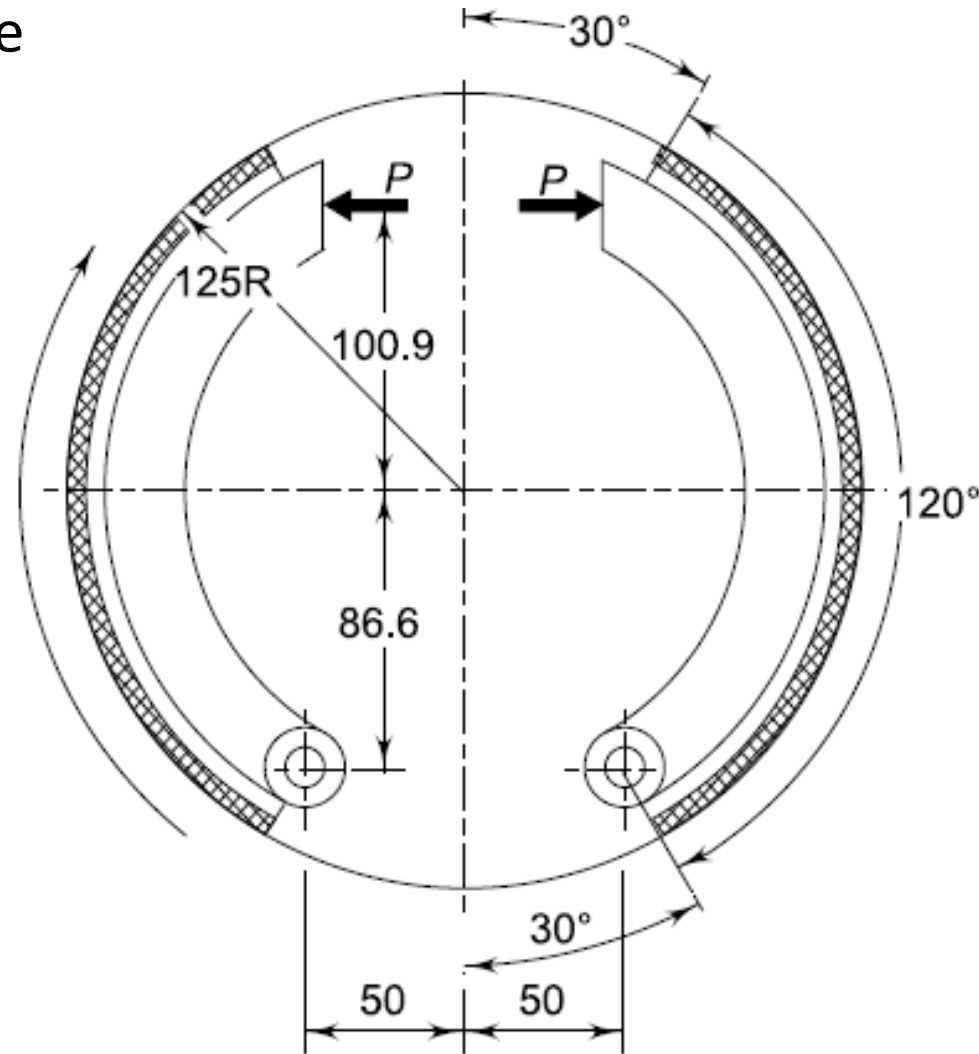
Disadvantages

- Relatively poor heat dissipating capacity
- Can become self locking due to wear if improperly designed

Problem 1.19

An internal expanding double shoe brake is shown in Figure. The face width of 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:

- The actuating force: F/P
- The torque absorbing capacity

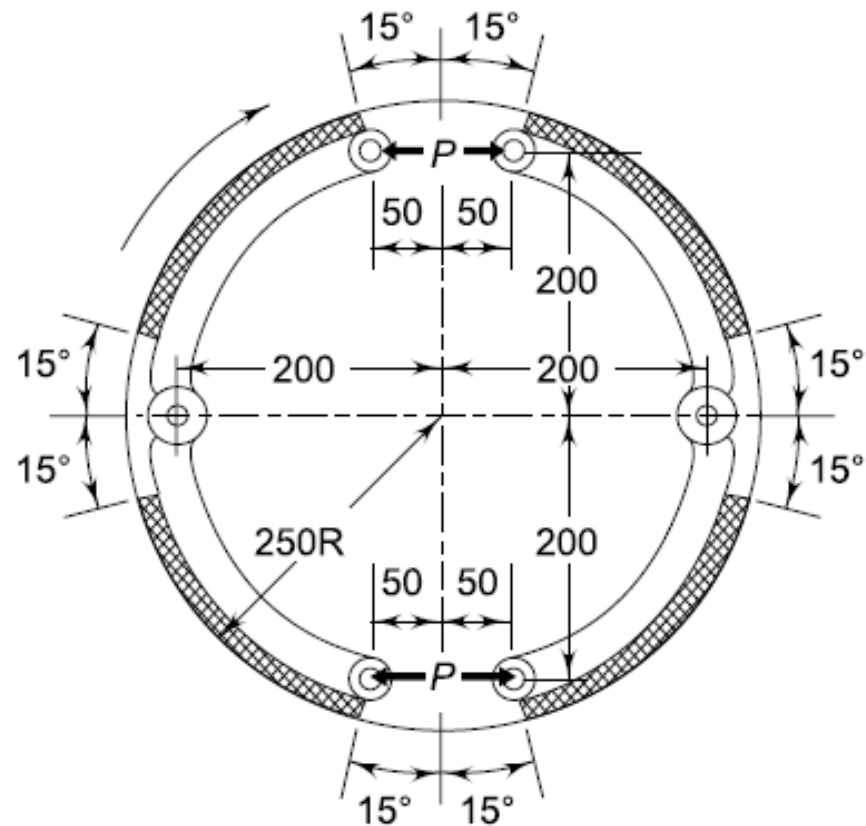


Problem 1.20

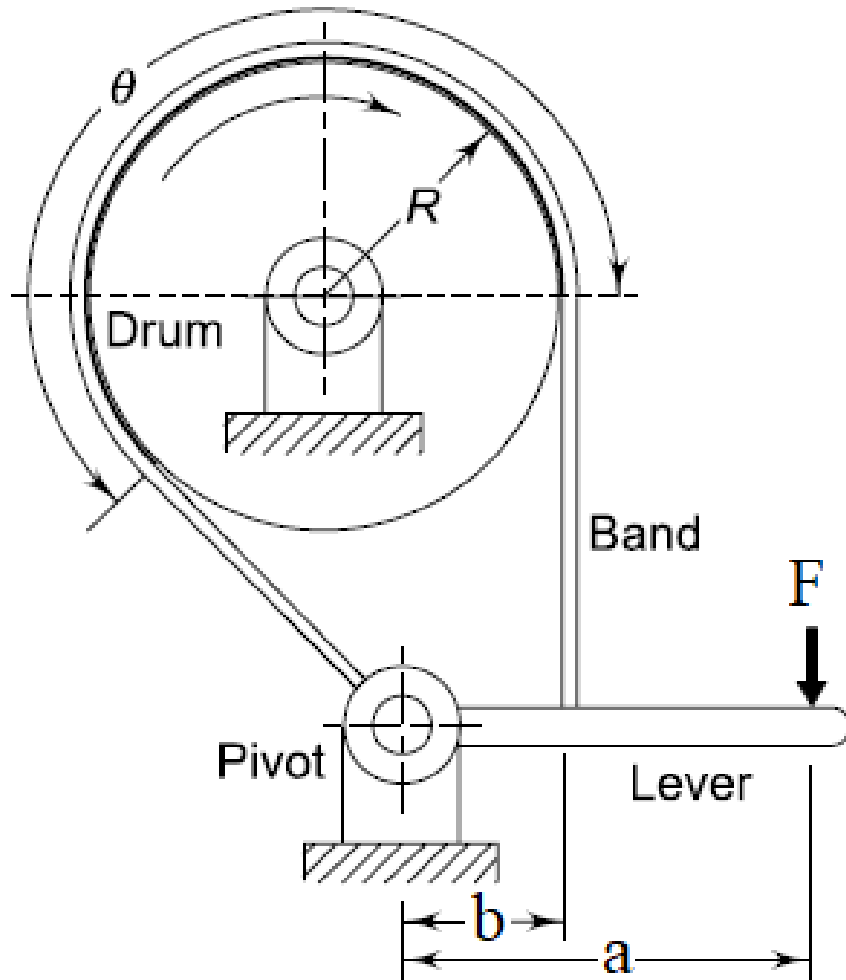
An internal expanding brake with four identical shoes is shown in Figure. The actuating hinge pin support a pair of shoes. The actuating mechanism is designed in such a way that it produces the same force P/F on each of the four shoes. The face width of the friction lining is 50 mm and the maximum intensity of normal pressure is limited to 1N/mm^2 . The coefficient of friction is 0.30.

Calculate:

- The actuating force
- The torque absorbing capacity of the brake



Simple Band Brakes

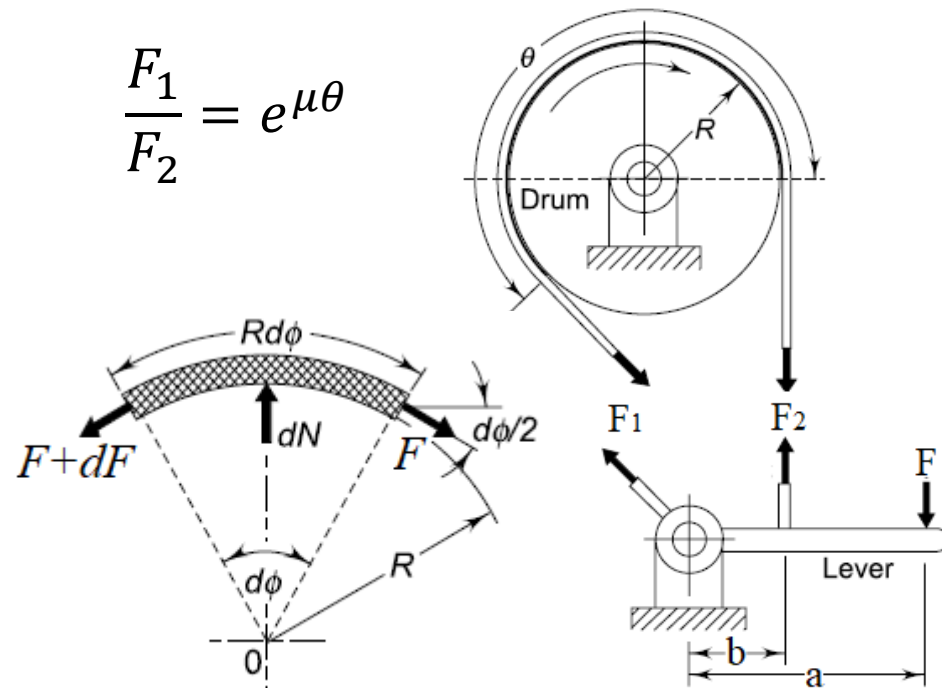


$F_1 =$ tension on tight side
 $F_2 =$ tension on loose side
 $\mu =$ coefficient of friction
 $\theta =$ angle of wrap

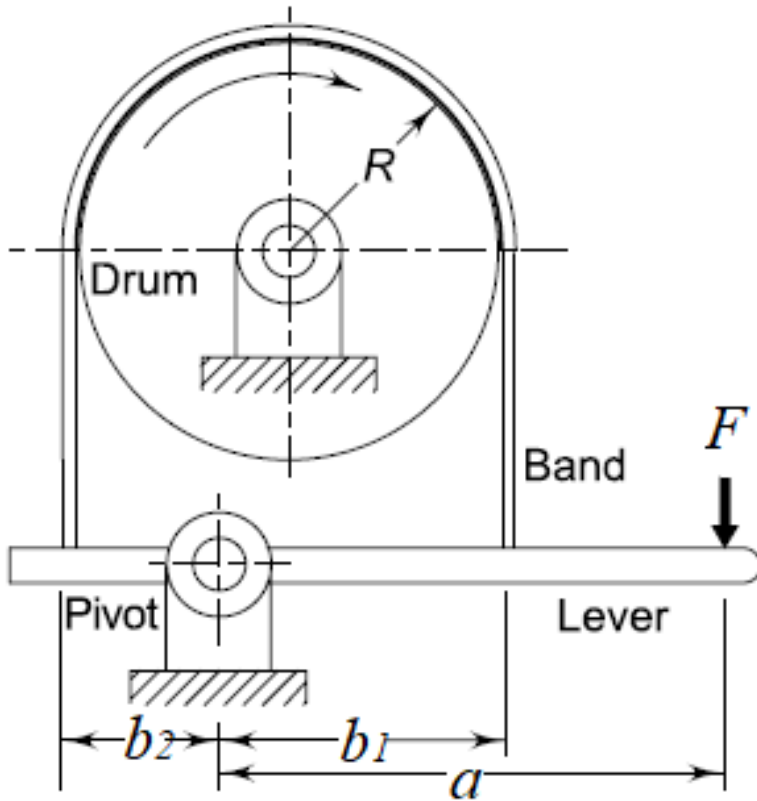
The torque absorbing capacity

$$M_t = (F_1 - F_2)R$$

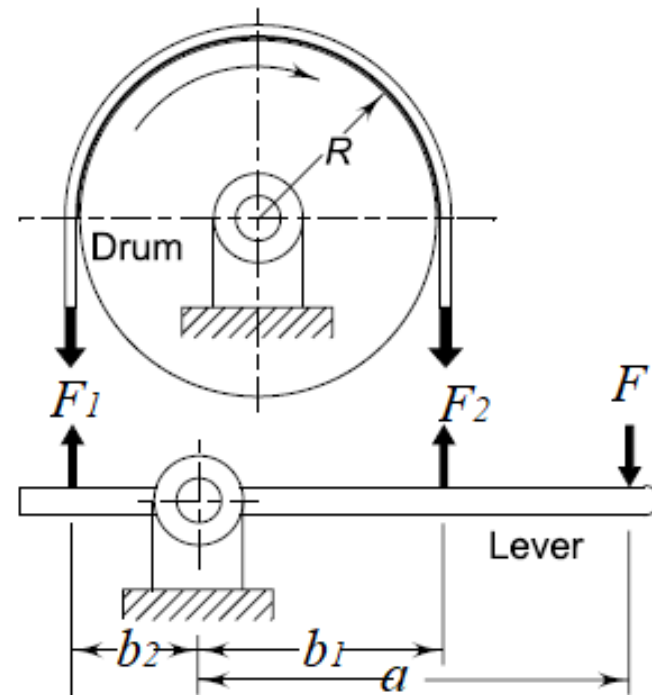
$$\frac{F_1}{F_2} = e^{\mu\theta}$$



Differential Band Brakes



F_1 = tension on tight side
 F_2 = tension on loose side
 μ = coefficient of friction
 θ = angle of wrap



Problem 1.21

A differential band brake is shown in Figure. 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:

- The tensions in the band
- The actuating force
- The torque capacity of the brake
- Find out whether the brake is self locking

