UNIT 1

CLUTCHES, BRAKES AND DYNAMOMETER

Friction Clutches

A friction clutch has its principal application in the transmission of power of shafts and machines which must be started and stopped frequently. Its application is also found in cases in which power is to be delivered to machines partially or fully loaded. The force of friction is used to start the driven shaft from rest and gradually brings it up to the proper speed without excessive slipping of the friction surfaces. In automobiles, friction clutch is used to connect the engine to the driven shaft. In operating such a clutch, care should be taken so that the friction surfaces engage easily and gradually brings the driven shaft up to proper speed. The proper alignment of the bearing must be maintained and it should be located as close to the clutch as possible.

1. The contact surfaces should develop a frictional force that may pick up and hold the load with reasonably low pressure between the contact surfaces.
2. The heat of friction should be rapidly dissipated and tendency to grab should be at a minimum.
3. The surfaces should be backed by a material stiff enough to ensure a reasonably uniform distribution of pressure.
4. The friction clutches of the following types are important from the subject point of view :

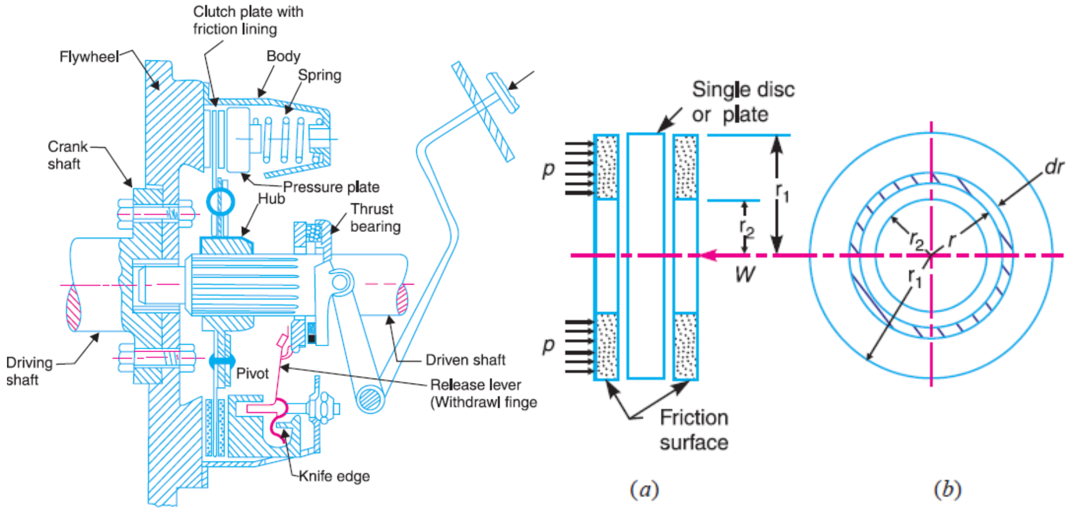
   1. Disc or plate clutches (single disc or multiple disc clutch),

   2. Cone clutches, and

   3. Centrifugal clutches.

Single Disc or Plate Clutch

Consists of a clutch plate whose both sides are faced with a friction material (usually of Ferrodo). It is mounted on the hub which is free to move axially along the splines of the driven shaft. The pressure plate is mounted inside the clutch body which is bolted to the flywheel.



Now consider two friction surfaces, maintained in contact by an axial thrust W, Consider an elementary ring of radius r and thickness dr. *We know that area of contact surface or friction surface, = 2 π r.dr. Normal or axial force on the ring, δW = Pressure × Area = p × 2 π r.dr. and the frictional force on the ring acting tangentially at radius r, Frictional torque acting on the ring,*

We shall now consider the following two cases :

***1.** When there is a uniform pressure, and*

***2.** When there is a uniform wear.*

Multiple Disc Clutch

may be used when a large torque is to be transmitted. The inside discs (usually of steel) are fastened to the driven shaft to permit axial motion. (except for the last disc). The outside discs (usually of bronze) are held by bolts and are fastened to the housing which is keyed to the driving shaft. The multiple disc clutches are extensively used in motor cars, machine tools etc.

∴ Number of pairs of contact surfaces,

$$n = n_1 + n_2 - 1$$

and total frictional torque acting on the friction surfaces or on the clutch,

$$T = n.\mu.W.R$$

where R = Mean radius of the friction surfaces

$$= \frac{2}{3}\left[\frac{(r_1)^3 - (r_2)^3}{(r_1)^2 - (r_2)^2}\right] \qquad \text{...(For uniform pressure)}$$

$$= \frac{r_1 + r_2}{2} \qquad \text{...(For uniform wear)}$$

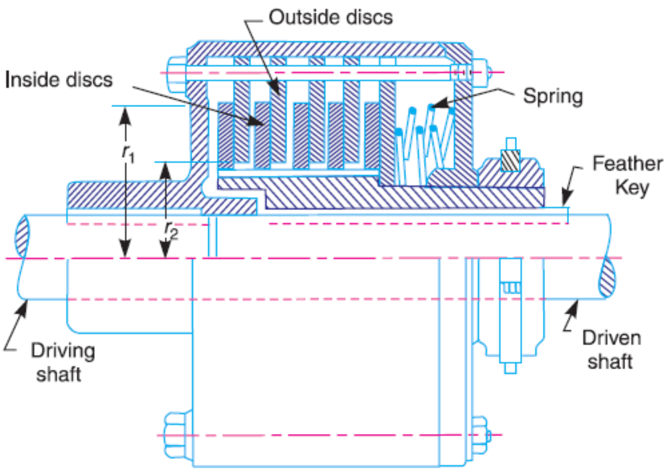


plate clutch has three discs on the shaft and 2 discs on the driven shaft, providing four pairs of contact surfaces. The outside diameter surfaces is 240 mm and dia inside 120 mm. Assuming UP and μ = 0.3; find the total spring load pressing the plates together to transmit 25 kW at 1575 r.p.m. For six springs with constant 13 kN/m and worn away 1.25 mm, find power transmitted.

$$25 \times 10^3 = T.\omega = T \times 165T$$

$$= \frac{25 \times 10^3}{165} = 151.5$$

$$MR = \frac{2}{3}\left[\frac{{r_1}^3 - {r_2}^3}{{r_1}^2 - {r_2}^2}\right]\frac{2}{3}\left[\frac{120^3 - 60^3}{120^2 - 60^2}\right]$$

$$= 93.3$$

$$Torque 151.5 = n.\mu.W.R$$

$$= 0.3W.4.0.09333$$

$$= 0.112WW$$

$$= 1353\ N.$$

$$No\ of\ Spring\ = 6$$

$$Surface = 8$$

$$Wear\ Contact\ = 1.25\ mm$$

$$Total\ Weak = \ 8 \times 1.25\ = 0.01\ m$$

$$Spring\ Contact\ = \ 13 \times 10^3 \frac{N}{m}$$

$$R = \frac{r_1 + r_2}{2} = \frac{120 + 60}{2} = 90$$

$T = n.\mu.W.R = 4.0.3.573.0.09 = 62$

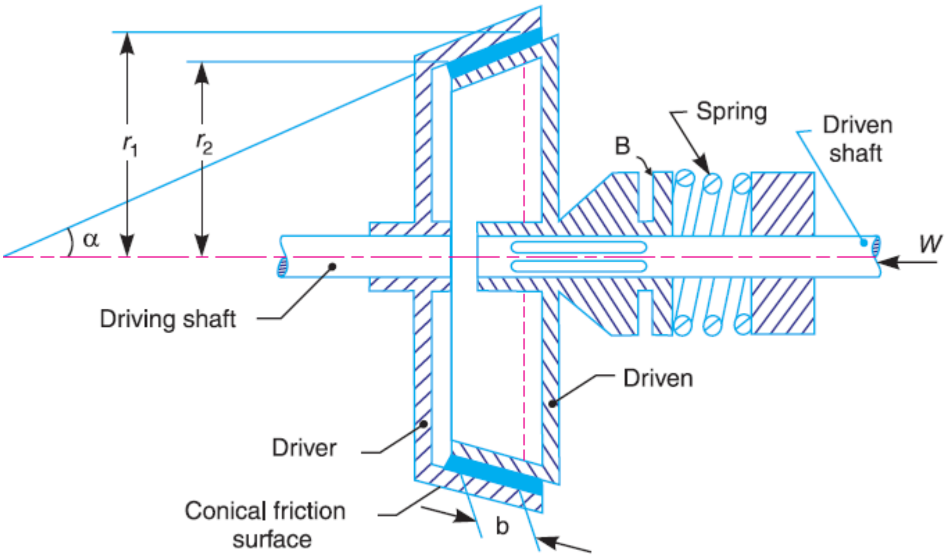
$P = T.\omega = 62.155P = 10.23\ kW$

Cone Clutch

was extensively used in automobiles but now-a-days it has been replaced completely by the disc clutch. Consider a small ring of radius *r* and thickness *dr*, as shown Let *dl* is length of ring of the friction surface, such that We shall consider the following two cases :

1. When there is a uniform pressure, and

2. When there is a uniform wear



1. ***Considering uniform pressure***

We know that normal load acting on the ring, δW_n = Normal pressure × Area of ring

Total axial load transmitted to the clutch or the axial spring force required,

$$p_n = \frac{W}{\pi[(r_1)^2 - (r_2)^2]}$$

Total frictional torque,

$$T = 2\pi\mu \times \frac{W}{\pi[(r_1)^2 - (r_2)^2]} \times \text{cosec}\ \alpha \left[\frac{(r_1)^3 - (r_2)^3}{3}\right]$$

$$= \frac{2}{3} \times \mu.W.\text{cosec}\ \alpha \left[\frac{(r_1)^3 - (r_2)^3}{(r_1)^2 - (r_2)^2}\right]$$

uniform wear

let *pr* be the normal intensity of pressure at a distance *r* from the axis of the clutch. We know that, in case of uniform wear, the intensity of pressure varies inversely with the distance.

We know that the normal load acting on the ring,

$$\delta W_n = \text{Normal pressure} \times \text{Area of ring} = p_r \times 2\pi r.dr \operatorname{cosec} \alpha$$

The contact surfaces in a cone clutch have an effective diameter of 75 mm. The semi-angle of the cone is 15º. The coefficient of friction is 0.3. Find the torque required to produce slipping of the clutch if an axial force applied is 180 N. This clutch is employed to connect motor running at 1000 r.p.m. The flywheel has a mass of 13.5 kg and k is 150 mm. Compute time to reach full speed and also slipping energy loss.

Centrifugal Clutch

The centrifugal clutches are usually incorporated into the motor pulleys. It consists of a number of shoes on the inside of a rim of the pulley. The outer surface of the shoes are covered with a friction material. These shoes, which can move radially in guides, are held against the boss (or spider) on the driving shaft by means of springs.

1. Mass of the shoes

We know that the centrifugal force acting on each shoe at the running speed, and the inward force on each shoe exerted by the spring at the speed at which engagement begins to take place and total frictional torque transmitted.

$T = \mu (P_c - P_s) R \times n = n.F.R$

2. Size of the shoes

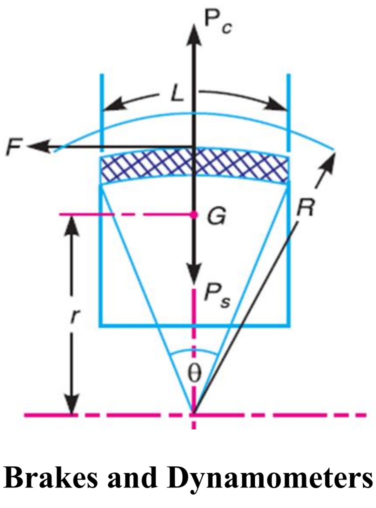
Let l = Contact length of the shoes,

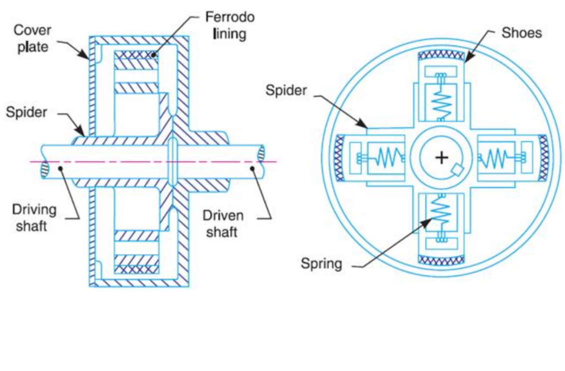
b = Width of the shoes,

R = Contact radius of the shoes. It is same as the inside radius of the rim of the pulley.

θ = Angle subtended by the shoes at the centre of the spider in radians.

p = Intensity of pressure exerted on the shoe. In order to ensure reasonable life, the intensity of pressure may be taken as 0.1 N/mm2.





Brakes and Dynamometers

The capacity of a brake depends upon the following factors

- Unit pressure between the braking surfaces,
- Coefficient of friction between the braking surfaces,
- Peripheral velocity of the brake drum,
- Projected area of the friction surfaces, and

Types of Brakes

The brakes, according to the means used for transforming the energy by the braking elements, are classified as :

1. Hydraulic brakes

2. Electric brakes

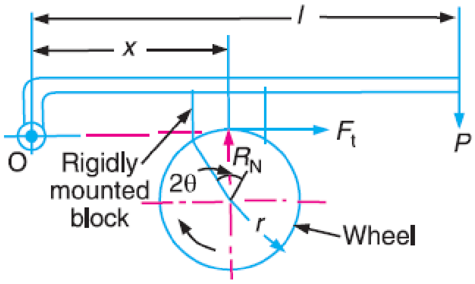
3. Mechanical brakes

(*a*) ***Radial brakes.*** In these brakes, the force acting on the brake drum is in radial direction. The radial brakes may be sub-divided into ***external brakes*** and ***internal brakes.*** According to the shape of the friction elements, these brakes may be ***block*** or ***shoe brakes*** and ***band brakes.***

(*b*) ***Axial brakes.*** In these brakes, the force acting on the brake drum is in axial direction. The axial brakes may be disc brakes and cone brakes. The analysis of these brakes is similar to clutches

Single Block or Shoe Brake

It consists of a block or shoe which is pressed against the rim of a revolving brake wheel drum. The block is made of a softer material than the rim of the wheel. This type of a brake is commonly used on railway trains and tram cars



Let

P = Force applied at the end of the lever,

RN= Normal force pressing the brake block on the wheel,

r = Radius of the wheel,

2θ = Angle of contact surface of the block,

μ = Coefficient of friction, and

Ft = Tangential braking force or the frictional force acting at the contact surface of the block and the wheel

Case 1. When the line of action of tangential braking force (Ft) passes through the fulcrum O of the lever, and the brake wheel rotates clockwise

From this we see that the moment of frictional force (μ.RN.a) adds to the moment of force ($P.l$). In other words, the frictional force helps to apply the brake. Such type of brakes are said to be ***self energizing brakes.*** When the frictional force is great enough to apply the brake with no external force, then the brake is said to **be *self-locking brake.*** From the above expression, we see that if $x \leq \mu . a$, then P will be negative or equal to zero. This means no external force is needed to apply the brake and hence the brake is self locking. Therefore the condition for the brake to be self locking is

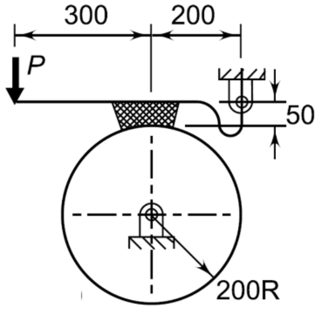
The self locking brake is used only in back-stop applications. The brake should be self energizing and not the self locking. In order to avoid self locking and to prevent the brake from grabbing, x is kept greater than $\mu\ .\ a.$ If Ab is the projected bearing area of the block or shoe, then the bearing pressure on the shoe,

pb = RN / Ab, We know that Ab = Width of shoe × Projected length of shoe = $w(2r\ sin\theta)$ When a single block or shoe brake is applied to a rolling wheel, an additional load is thrown on the shaft bearings due to heavy normal force (RN) and produces bending of the shaft.

Pivoted Block or Shoe Brake

when the angle of contact is less than 60°, then it may be assumed that the normal pressure between the block and the wheel is uniform. when the angle of contact is greater than 60°, then the unit pressure normal to the surface of contact is less at the ends than at the center. In such cases, the block or shoe is pivoted to the lever, These brakes have more life and may provide a higher braking torque.

Question. A brake acts on the 3/4th of a drum of 450 mm dia. The provides a torque of 225 N-m. 1 end of the band fixed to fulcrum lever and 2 end to a pin 100 mm from the fulcrum. A force is applied at 500 mm from the fulcrum and m= 0.25, find the Force during the rotation in the (a) anticlockwise and clockwise direction.



Answers: ***(a) anticlockwise***

We know that angle of wrap $\boldsymbol{\theta} = \frac{3}{4}$ th of circumference

$$= \frac{\mathbf{3}}{\mathbf{4}} \times 360 = 270 \times \frac{\pi}{180} = 4.713\, rad$$

$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu.\theta = 0.25 \times 4.713 = 1.178$$

$$\log\left(\frac{T_1}{T_2}\right) = \frac{1.178}{2.3} = 0.5123\ (or)$$

$$\frac{T_1}{T_2} = 3.253$$

we know that of braking torque $(\mathrm{T_B})$,

$$225 = (\mathrm{T_1} - \mathrm{T_2})\mathrm{r} = (\mathrm{T_1} - \mathrm{T_2})0.255$$

$$\mathrm{T_1} - \mathrm{T_2} = \frac{225}{0.225} = 1000N$$

From equations

$$\mathrm{T_1} = 1444\mathrm{N}; \mathrm{T_{2=}}444\mathrm{N}$$

Now taking moments about the fulcrum O ,we have

$$p \times \mathrm{l} = \mathrm{T_2}.b\ \text{(or)}$$

$$p \times 0.5 = 444 \times 0.1 = 44.4$$

$$p = \frac{44.4}{0.5} = 88.8\ \mathrm{N}$$

(b) clockwise direction

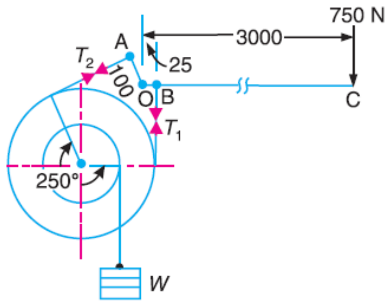
Moment about O

$$p \times \mathrm{l} = \mathrm{T_1}.b\ \text{(or)}$$

$$p \times 0.5 = 1444 \times 0.1 = 144.4$$

$$p = \frac{144.4}{0.5} = 288.8\ \mathrm{N}$$

A load W and rope wound round a barrel 450 mm dia. A differential band brake acts on a drum 800 mm dia. The two ends of the bands at distances of 25 mm and 100 mm from the fulcrum. The angle of lap is 250° and m 0.25. Evaluate the max, load W a force of 750 N is at a distance of 3000 mm?



Given: $D = 450mm \ or \ R = 225mm$

$$d = 800mm \ or \ r = 400mm$$

$$OB = 25mm$$

$$OA = 100mm$$

$$\theta = 250 = 250\left(\frac{\pi}{180}\right) = 4.364rad$$

$$\mu = 0.25 \mathrm{p} = 750\mathrm{N}$$

$$\mathrm{l} = \mathrm{OC} = 3000\mathrm{mm}$$

Since OA is greater than OB, therefore the operating force P = 750 N will act downwards.

First of all, let us consider that the drum rotates in clockwise direction

We know that when the drum rotates in clockwise direction, the end of band attached to will be slack with tension T2 and the end of the band attached to B will be tight with tension T1.

$$2.3log\left(\frac{T_1}{T_2}\right) = \mu.\theta$$

$$=0.25\times4.364=1.091$$

$$\log\left(\frac{T_1}{T_2}\right) = \frac{1.091}{2.3}$$

$$= 0.4743 \ (or)$$

$$\frac{T_1}{T_2} = 2.98$$

$$\mathrm{T_1} = 2.98 \ \mathrm{T_2}$$

Now taking moments about the fulcrum O,

$$750\times3000 + T1\times25 = T2\times100$$

or

$$T2\times100-2.98 \ T2\times25=2250\times 10^3N$$

$$22.5T_2 = 2250 \times 10^3N$$

$$T_2 = 2250 \times \frac{10^3}{25.5}$$

$$= 2.98 \times 88 \times 10^3$$

$$= 262 \times 10^3N$$

$$T_B = (T_1 - T_2)r$$

$$= (262 \times 10^3 - 88 \times 10^3)400$$

$$= 69.6 \times 10^6\mathrm{N - mm}\ldots\ldots\ldots \ (1)$$

The torque

Ans.

$$T_W = W.R = W\times225 = 225 \text{ kN-mm}$$

from equations (1) and (2).

$$W=69.6\times\frac{10^6}{225} = 309 \times 10^3\mathrm{N}=309 \ \mathrm{KN}$$

the end of the band attached to A will be tight with tension T1 and end of the band attached to B will be slack with tension T2,

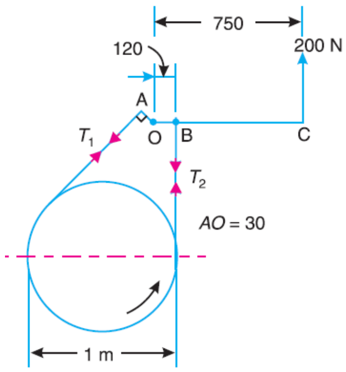
$$\frac{T_1}{T_2} = 2.98(\text{or})\ \mathrm{T_1} = \ 2.98\mathrm{T_2}$$

taking moments about O,

$$750 \times 300 + \mathrm{T_2} \times 25 = \mathrm{T_1} \times 100 \ldots\ldots (\mathrm{T_1} = 2.98\mathrm{T_2})$$

$$2.98\mathrm{T_2} \times 100 - \mathrm{T_2} \times 25 = 2250 \times 10^3$$
$$273\mathrm{T_2} = 2250 \times \frac{10^3}{273} = 8242N$$
$$T_1 = 2.98$$
$$T_2 = 2.98 \times 8242 = 24561N$$
$$T_B = (T_1 \times T_2)r$$
$$= (24561 - 8242)400$$
$$= 6.53 \times 10^6 N - mm$$
$$W = 6.53 \times \frac{10^6}{225}$$
$$= 29 \times 10^3$$
$$= 29KN$$

brake, having 14 blocks each of which subtends an angle of 15 degree at the center, is applied to a drum of 1000 mm effective dia. The mass of 2000 kg and a gyration of 500 mm. The radius 30 mm and 120 mm from the fulcrum. If a force of 200 N is applied at a distance of 750 mm from the fulcrum,
find: 1. maximum braking torque, 2. angular retardation of the drum, and 3. time taken by the system to come to rest from the rated speed of 360 r.p.m. Friction coefficient as 0.25.



All dimensions in mm

Given

$$\mathrm{n} = 142,\ \theta = 15,\ \theta = 7.5,\ \mathrm{d} = 1\mathrm{m\ or}\ ,\ r = 0.5\mathrm{mm} = 2000\mathrm{kg},\ \mathrm{k} = 500\mathrm{mm} = 0.5\mathrm{m},\ \mathrm{p} = 200\mathrm{N},\ \mathrm{N} = 360\mathrm{rpm},\ \mathrm{l} = 750\mathrm{mm},\ \mu = 0.25$$

$$\frac{T_1}{T_2} = \left(\frac{1+\mu\tan(\theta)}{1-\mu\tan(\theta)}\right)^{14} = \left(\frac{1+0.25\tan(7.5)}{1-0.25\tan(7.5)}\right)^{14} = \left(\frac{1+0.25\times 0.1317}{1-0.25\times 0.1317}\right)^{14} = (1.068)^{14} = 2.512$$

$$T_1 = 8440N$$

$$T_2 == 3360N$$

$$T_B = (T_1 - T_2)r = (8440 - 3360)0.5 = 2540\text{N} - \text{m}$$

$$2540 = I\alpha = m.k^2.\alpha = 2000(0.5)^2\alpha = 500\alpha$$

$$\alpha = \frac{2540}{500} = 5.08 rad/s^2,$$

$$\omega_1 = 2\pi \times \frac{360}{60} = \frac{37.7 rad}{s}$$

$$\omega_2 = 04,$$

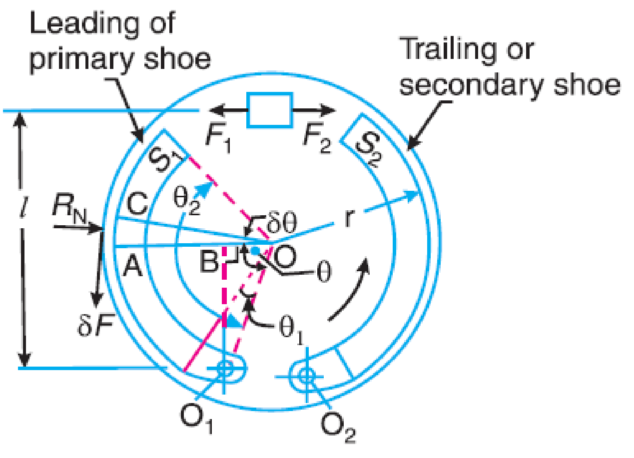
$$\omega_2 = \omega_1 - \alpha.t, t$$

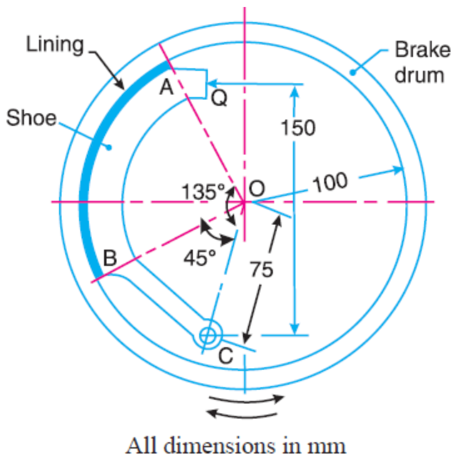
$$= \frac{\omega_1}{\alpha} = \frac{37.7}{5.08}$$

$$= 7.42$$

Numerical Problem:

The distance 'CO' is 75 mm, O being the center of the drum. The internal radius of the brake drum is 100 mm. The friction lining extends over an arc AB, such that the angle AOC is 135° and angle BOC is 45°. The coefficient of friction may be taken as 0.4 and the braking torque required is 21 N-m. Calculate the force Q required to operate the brake when ***1.*** *The drum rotates clockwise, and* ***2.*** *The drum rotates anticlockwise.*





$$OC = 75mm$$

$$r = 100mm \quad \theta_2 = 135 = 135 \times \frac{\pi}{180} = 2.356\frac{rad}{s}$$

$$\theta_1 = 45 = 45 \times \frac{\pi}{180} = \frac{0.786 rad}{s}$$

$$l = 150mm$$

$$\mu = 0.4$$

$$T_B = 21 = 21 \times 10^3 N - m$$

$$21 \times 10^3 = \mu . p_1 . b . r^2 \left(\cos(\theta_1) - \cos(\theta_2)\right)$$

$$= 0.4 \times p_1 \times b(100)^2 (\cos(45) - \cos(135))$$

$$= 5656 p_1 . b p_1 . b$$

$$= \frac{21 \times 10^3}{5656}$$

$$= 3.7$$

$$M_N = \frac{1}{2} p_1 . b . r . OC \left[(\theta_1 - \theta_2) + \frac{1}{2}\left(\sin(2\theta_1) - \sin(2\theta_2)\right)\right]$$

$$= \frac{1}{2} \times 3.7 \times 100 \times 75 \left[(2.356 - 0.786) + \frac{1}{2}\left(\sin(90) - \sin(270)\right)\right]$$

$$= 13.875(1.57 + 1) = 35660 N - mm$$

$$M_F = \mu . p_1 . b . r [r(\cos(\theta_1) - \cos(\theta_2) + \frac{OC}{4}(\cos(2\theta_1) - \cos(2\theta_2)]$$

$$= 0.4 \times 3.7 \times 100[100(\cos(45) - \cos(135) + \frac{75}{4}(\cos(270) - \cos(90)]$$

$$= 148 \times 141.4 = 20930 N - mm$$

$$Q \times 150 = M_N + M_F = 35660 + 20930 = 56590 Q = \frac{56590}{150} = 377 N$$

$$Q \times 150 = M_N - M_F = 35660 - 20930 = 14730 Q = \frac{14730}{150} = 98.2 N \backslash$$

$$Given\ N_A = 600 rmp$$

$$D_A = D_B = D_C = 300 mm = 0.3 m$$

$$W \times 750 = 2T_1 \times 300 - 2T_2 \times 300 = 600(T_1 - T_2)$$

$$T_1 - T_2 = W \times \frac{750}{600} = 1.25 W N$$

$$4500 = \frac{(T_1 - T_2)\pi D_A N_A}{60} = \frac{1.25 W \times \pi \times 0.3 \times 600}{60} = 11.78 W$$

$$W = \frac{45000}{11.78} = 382 N$$

Dynamometer

A dynamometer is a brake but in addition it has a device to measure the **frictional resistance**. Knowing the frictional resistance, we may obtain the **torque transmitted and hence the power of the engine.**

Types of Dynamometers

Following are the two types of dynamometers, used for measuring the brake power of an engine

1. Absorption dynamometers, absorption dynamometers, the entire energy or power produced by the engine is absorbed by the friction resistances of the brake and is transformed into heat, during the process of measurement.
2. Transmission dynamometers- the energy is not wasted in friction but is used for doing work. The energy or power produced by the engine is transmitted through the dynamometer to some other machines where the power developed is suitably measured.

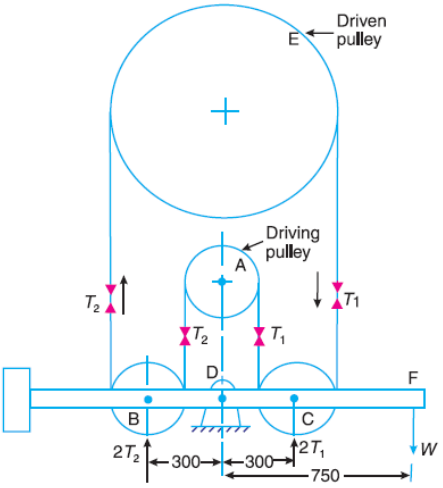
Classification of Absorption Dynamometers

1. Prony type Brake
2. Rope Type

Classification of Transmission Dynamometers

1. Epicyclic-train Dynamometer
2. Belt Transmission Dynamometer-Froude or Thorneycroft Transmission Dynamometer

The dynamometer are shown in Fig. A is the driving pulley at 600 r.p.m. E is the driven pulley and all portions of the belt between the pulleys are vertical. A, B and C are each 300 mm diameter. The length DF is 750 mm. Find : 1. the value of the weight W to maintain the beam in a horizontal position when 4.5 kW is being transmitted, and 2. the value of W, when the belt just begins to slip on pulley A. The coefficient of friction being 0.2 and maximum tension in the belt 1.5 kN.



$$Given\ N_A = 600rmp$$

$$D_A = D_B = D_C = 300mm = 0.3m$$

$$W \times 750 = 2T_1 \times 300 - 2T_2 \times 300 = 600(T_1 - T_2)$$

$$T_1 - T_2 = W \times \frac{750}{600} = 1.25WN$$

$$4500 = \frac{(T_1 - T_2)\pi D_A N_A}{60} = \frac{1.25W \times \pi \times 0.3 \times 600}{60} = 11.78W$$

$$W = \frac{45000}{11.78} = 382N$$

$$Given$$

$$\mu = 0.2$$

$$T_1 = 1.5KN = 1500N$$

$$2.3log\frac{T_1}{T_2} = \mu.\theta = 0.2 \times \pi = 0.6284$$

$$log\frac{T_1}{T_2} = \frac{0.6284}{2.3} = 0.2732$$

$$\frac{T_1}{T_2} = 1.876$$

$$T_2 = \frac{T_1}{1.876} = \frac{1500}{1.876} = 800N$$

$$\mathrm{W} \times 750 = 2T_1 \times 300 - 2T_2 \times 300 = 2 \times 1500 \times 300 - 2 \times 800 \times 300 = 420 \times 10^3$$

$$= \frac{420 \times 10^3}{750} = 560\mathrm{N}$$