UNIT-V

DESIGN OF CAM, CLUTCHES AND BRAKES

Cam design: Types, pressure angle and under cutting - Base circle determination, forces and surfaces stresses.

Design of plate clutches - Axial clutches - cone clutches - Internal expanding rim clutches - Internal and external shoe brakes.

Introduction:-

=> Clutch is a transmission device which is used to engage and disengage the power from the engine to the rest of the system.

=> The clutch is located in b/w the engine and the transmission system. When the clutch is engaged to the engine, power is transmitted to the wheels.

=> If it is disengaged, the power is not transmitted to the rest of the system even through engine is running and hence the vehicle stops.

⇒ Therefore for coupling the engine smoothly to the power transmission during starting from rest and gear shifting, clutch is used.

FUNCTIONS OF THE CLUTCH:-

→ To connect and disconnect the shafts at will.

→ To start (or) stop a machine (or a rotating element) without starting and stopping the prime mover.

→ To maintain constant speed, torque and power.

→ To reduce shocks transmitted between m/c shafts

→ For automatic disconnect, quick start and stop, gradual starts, and non-reversing and over running functions.

PRINCIPLES OF OPERATION OF THE CLUTCH:-

⇒ The clutch works on the principle of friction.

⇒ when two friction surfaces are brought in contact with each other and pressed, they are united due to the friction b/w them.

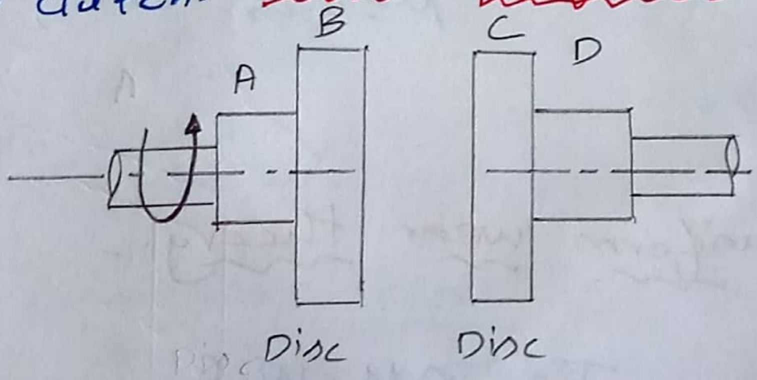
=) The friction b/w the two surfaces depends upon the area of the surfaces pressure applied upon them and coefficient of friction of the surface materials.

=) one surface is considered as driving member and the other as driven member.

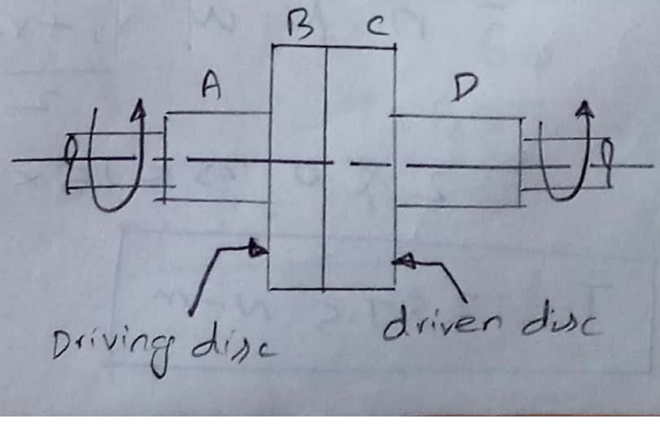
=) The driving member is kept rotating. When the driven member is brought in contact to the driving member, it also starts rotating.

=) when the driven member is separated from the driving member it does not revolve. This is the principle behind the operation of the clutch.

Before engagement



After engagement



JAYA
Pg - 10.11
Ex: 10.1

M/J 2013

Problems on single plate clutch:-

An automotive single plate clutch consists of two pairs of contacting surfaces. The inner and outer radii of friction plate are 120mm and 250mm respectively. The coefficient of friction is 0.25 and the total axial force is 15kN. Calculate the power transmitting capacity of the clutch plate at 500 rpm using

(i) uniform wear theory

(ii) uniform pressure theory.

given data:-

$r_2 = 120\,mm = 0.12\,m$

$n = 2$, $r_1 = 250\,mm = 0.25\,m$

$\mu = 0.25$, $W = 15\,kN = 15 \times 10^3\,N$

$N = 500\,rpm.$

i) uniform wear theory:-

$$T = n\mu \cdot W \cdot R \qquad \therefore R = \left[\frac{r_1 + r_2}{2}\right]$$

$$= n \cdot \mu \cdot W \cdot \frac{r_1 + r_2}{2}$$

$$= 2 \times 0.25 \times 15 \times 10^3 \left[\frac{0.12 + 0.25}{2}\right]$$

$$\boxed{T = 1387.5\ N\text{-}m}$$

$$P = \frac{2\pi N T}{60} = \frac{2\pi \times 500 \times 1387.5}{60}$$

$$P = 72\,649 \text{ W} = 72.65 \text{ kW}$$

$$\boxed{P = 72.65 \text{ kW}}$$

(ii) Uniform pressure theory:-

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

$$= n.\mu.W.\frac{2}{3}\left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right] \qquad R = \frac{2}{3}\left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right]$$

$$= \cancel{2 \times 0.25 \times 15 \times 10^3 \times \frac{2}{3}\left[\frac{(0.12)^3 - (0.25)^3}{(0.12)^2 - (0.25)^2}\right]}$$

$$T = 2 \times 0.25 \times 15 \times 10^3 \times \frac{2}{3}\left[\frac{(0.25)^3 - (0.12)^3}{(0.25)^2 - (0.12)^2}\right]$$

$$\boxed{T = 1444.6 \text{ N-m}}$$

$$\text{Power transmitted} = \frac{2\pi N T}{60} \qquad \frac{\text{N-m}}{\text{s}} = \text{W}$$

$$= \frac{2\pi \times 500 \times 1444.6}{60}$$

$$= 75639.08 \text{ W}$$

$$\boxed{P = 75.64 \text{ kW}}$$

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M/J 2012

A dry single plate clutch is to be designed to transmit 112 kW at 2000 rpm. The outer radius of the friction plate is 1.25 times the inner radius. The intensity of pressure b/w the plates is not to exceed 0.07 N/mm². The coefficient of friction may be assumed equal to 0.3. The helical springs required by this clutch to provide axial force necessary to engage the clutch are 8. If each spring has stiffness equal to 40 N/mm. Determine the dimensions of the friction plate and initial compression in the springs.

Given:-

$P = 112\,kW = 112 \times 10^3\,W$, $N^* = 2000\,rpm$

$r_1 = 1.25\,r_2$, $P = 0.07\,N/mm^2$, $\mu = 0.3$

Number of springs $= 8$, $\frac{\text{stiffness}}{\text{spring}} = 40\,N/mm$

$n = 2$

To find:-

(i) Dimensions of the friction plate

(ii) Initial compression in the springs

Solution:-

(i) To find the dimensions of the friction plate :

Let r_1 and r_2 be the outer and inner radii of the friction plate respectively. Since the outer radius 1.25 times more than inner radius. Therefore

$$r_1 = 1.25 r_2$$

For uniform conditions, $p \cdot r = C$ [a constant]

Since the intensity of pressure is maximum at the inner radius (r_2),

$\therefore \quad p \cdot r_2 = C$ (or) $C = 0.07 \times r_2$ N/mm

and axial load acting on the friction plate,

$$W = 2\pi C (r_1 - r_2)$$

$$= 2\pi \times 0.07 r_2 \times (1.25 r_2 - r_2)$$

$$= 2\pi \times 0.07 r_2 \times (1.25 - 1) r_2$$

$$\boxed{W = 0.11 r_2^2 \text{ N}}$$

W.K.T that the mean radius of the friction plate, for uniform ~~pressure~~ wears,

$$R = \frac{r_1 + r_2}{2} = \frac{1.25r_2 + r_2}{2} = 1.125\,r_2$$

∴ Torque transmitted

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

$$534.76 \times 10^3 = 2 \times 0.3 \times 0.11\,r_2^2 \times 1.125\,r_2$$

$$r_2^3 = \frac{534.76 \times 10^3}{2 \times 0.3 \times 0.11 \times 1.125}$$

$$r_2 = \sqrt[3]{\frac{(534.76 \times 10^3)}{(2 \times 0.3 \times 0.11 \times 1.125)}}$$

$$\boxed{r_2 = 193\,mm}$$

$$P = \frac{2\pi NT}{60}$$

$$112 \times 10^3 = \frac{2\pi NT}{60}$$

$$T = \frac{P \times 60}{2\pi N} = \frac{112 \times 10^3 \times 60}{2\pi \times 2000}$$

$$\boxed{T = 534.76\ N\text{-}m}$$

$$534.76 \times 10^3\ N\text{-}mm$$

W.K.T

$$r_1 = 1.25\,r_2$$

$$r_1 = 1.25 \times 193$$

$$\boxed{r_1 = 241.25\,mm}$$

(ii) **To find the initial compression in the springs:**

W.K.T total stiffness of the springs.

S = stiffness per spring × no. of springs

$S = 40 \times 8 = 320$ N/mm.

Axial force required to engage the clutch,

$W = 0.11\,(r_2)^2 = 0.11 \times (193)^2$

$\boxed{W = 4097\text{ N}}$

Initial compression in the springs $= W/S$

$= \dfrac{4097}{320}$

$= 12.8$ mm

N/D 2011

A single plate clutch, both sides being effective, is required to connect a machine shaft to a driver shaft which runs at 500 rpm. The moment of inertia of the rotating parts of the machine is 1 kg·m². The inner and outer radii of the friction discs are 50mm and 100mm respectively. Assuming uniform pressure of 0.1 N/mm² and coefficient of friction of 0.25, determine the time taken for the machine to reach full speed when the clutch is suddenly engaged. Also determine the power transmitted by the clutch, the energy dissipated during clutch slip and the energy supplied to the machine during engagement.

given:-

$N = 500$ rpm $\quad I = 1\ kg.m^2$, $n = 2$

$r_1 = 100$ mm $= 0.1$ m, $r_2 = 50$ mm $= 0.05$ m $\quad \mu = 0.25$

$P_{max} = 0.1\ N/mm^2$

To find:-

(i) The time to attain the full speed by the machine (with uniform wear)

Since the intensity of pressure (p) is maximum at the inner radius r_2, therefore

$P_{max} \cdot r_2 = C$

$C = 0.1 \times 50 \quad N/mm^2 \times mm$

N/mm.

$C = 0.1 \times 50 \times 10^{3}$

$\boxed{C = 5000\ N/m}$

$1m = 1000$ 10^{-3} m

Axial thrust exerted

$$W = 2\pi C (r_1 - r_2)$$

$$= 2\pi \times 5000 (0.1 - 0.05)$$

$$\boxed{W = 1570.79\,N}$$

Torque transmitted

$$T = n.\mu.\,W R \qquad \therefore R = \left[\frac{r_1 + r_2}{2}\right]$$

$$= n.\mu.W.\left[\frac{r_1 + r_2}{2}\right]$$

$$T = 2 \times 0.25 \times 1570.79.\left[\frac{0.1 + 0.05}{2}\right]$$

$$\boxed{T = 58.90\ N-m}$$

W.K.T Power transmitted

$$P = \frac{2\pi N T}{60} = \frac{2\pi \times 500 \times 58.90}{60}$$

$$\boxed{P = 3084\,W}$$

$T = I\alpha$, where α is angular acceleration

Also $58.90 = 1 \times \alpha$

$\frac{N-m}{kg.m^2}$

$\frac{N}{kg}$

$$\alpha = 58.90\ rad/sec^2$$

we also know that $\alpha = \frac{\omega}{t} = 58.90$

$$\left(\frac{2\pi n}{60}\right) \times \frac{1}{t} = 58.90 \quad \frac{rad}{sec} \times \frac{1}{t} = \frac{rad}{sec^2}$$

$$t = \frac{rad}{sec} \times \frac{sec^2}{rad}$$

$$\omega = \frac{rad}{sec}$$

$$\frac{2\pi \times 500}{60 \times t} = 58.90 \quad (or) \quad t = \frac{2\pi \times 500}{60 \times 58.90}$$

$$\boxed{t = 0.89 sec}$$

Thus the full speed is attained by the machine in 0.89 seconds.

(ii) The energy lost in slipping of the clutch:

Angle turned by the driving shaft $\theta_1 = \omega t = \frac{2\pi n}{60} \times t$

$$= \frac{2\pi \times 500}{60} \times 0.89$$

$$\boxed{\theta_1 = 46.60 \text{ rad.}}$$

and angle turned by the driven shaft

$$\theta_2 = \omega_0 t + \frac{1}{2} \alpha t^2$$

$$= 0 + \frac{1}{2} \times 58.90 \times (0.89)^2$$

$$\boxed{\theta_2 = 23.33 \text{ rad}}$$

Energy loss in friction $= T(\theta_1 - \theta_2)$

$= 58.90 \times (46.60 - 23.33)$

$= 1370.60$ N-m

(iii) Intensity of pressure, if the condition is uniform pressure:

$$\text{Intensity of pressure}, p = \frac{W}{\pi(r_1^2 - r_2^2)}$$

$$= \frac{1570.79}{\pi((0.1)^2 - (0.05)^2)}$$

$= 66666.~~4~~$ N/m²

$$\boxed{p = 66.67 \text{ KN/m}^2}$$

(iv) Ratio of power transmitted with uniform wear to that with uniform pressure:

$$\left.\begin{array}{l}\text{Power transmitted with} \\ \text{uniform wear}\end{array}\right\} = 3084 \text{ W}$$

$$\left.\begin{array}{l}\text{Torque transmitted with} \\ \text{uniform pressure}\end{array}\right\} = n \cdot \mu \cdot W \cdot \frac{2}{3}\left[\frac{r_1^3 - r_2^3}{r_p^2 - r_2^2}\right]$$

$$= 2 \times 0.25 \times 1570 \times \frac{2}{3}\left[\frac{(0.1)^3 - (0.05)^3}{(0.1)^2 - (0.05)^2}\right]$$

$$\boxed{T = 61.09 \text{ N-m}}$$

Power transmitted with uniform pressure is given by

$$P = \frac{2\pi N T}{60}$$

$$= \frac{2\pi \times 500 \times 61.09}{60}$$

$$\boxed{P = 3199\ W}$$

$$\frac{\text{power transmitted with uniform wear}}{\text{power transmitted with uniform pressure}}$$

$$= \frac{3084}{3199} = 0.96$$

JAYA Pg. no. 10.19
EX: 10.10

A multiplate disc clutch transmits 55 kW of power at 1800 rpm. coefficient of friction for the friction surfaces is 0.1. Axial intensity at pressure is not to exceed 160 KN/m². The internal radius is 80 mm and is 0.7 times the external radius. Find the number of plates needed to transmit the required torque.

given:-

$P = 55kW = 55\times10^3 W \quad N = 1800 rpm$

$\mu = 0.1, \; P_{max} = 160\; kN/m^2 = 160\times10^3 N/m^2$

$r_2 = 80mm = 80\times10^{-3}m, \; r_2 = 0.7 r_1$

$$\frac{r_2}{r_1} = 0.7$$

1mm = 1000 m
m = $\frac{1}{1000}$ mm
m = multi $\times (\frac{1}{1000})$
mm = multi 1000

To find:-

Number of plates needed to transmit the required torque.

solution:-

$$r_2 = 0.7 r_1 \quad (or) \quad \frac{r_2}{r_1} = 0.7$$

$$r_1 = \frac{r_2}{0.7}$$

$$= \frac{80\times10^{-3}}{0.7}$$

$$\boxed{r_1 = 0.1143\, m}$$

Assuming uniform wear, axial force exerted is given by

$$W = 2\pi C(r_1 - r_2)$$

W.K.T, the maximum intensity of pressure (P_{max}) is at the inner radius (r_2)

$$P_{max} \cdot r_2 = C \quad (or) \quad C = 160\times10^3 \times 80\times10^{-3}$$

$$\boxed{C = 12800\; N/m}$$

$W = 2\pi C(r_1 - r_2)$

$= 2\pi \times 12800 \times (0.1143 - 0.08)$

$$\boxed{W = \cancel{2758}.57\,N \quad 2750.52}$$

Torque transmitted by a single friction surface is given by

$$T = \mu \cdot W \cdot \frac{(r_1 + r_2)}{2}$$

$$= 0.1 \times \cancel{2758.5}7 \times \left(\frac{0.1143 + 0.08}{2}\right)$$

2750.52

$$\boxed{T = \cancel{26.8}\,N\text{-}m \quad T = 26.70\,N\text{-}m}$$

The total torque required can be calculated as given below.

Power $P = \frac{2\pi N T}{60}$

$$T = \frac{P \times 60}{2\pi N} = \frac{55 \times 10^3 \times 60}{2\pi \times 1800}$$

Total torque required $\boxed{T = 291.78\ N\text{-}m}$

Number of friction surface required $\Big\}$ = $\frac{\text{Total torque required}}{\text{~~Total~~ Torque required per surface}}$.

$$= \frac{29\cancel{1}.78}{26.8} = \cancel{10.887}^{\,10.92} \approx 11$$

Total number of plates

= Number of pairs of contact surface + 1

= 11 + 1 = 12 surfaces.

Hence, there will be 12 total plates, in which six plate each driving and driven shafts having

A/M 2011 A multi-disc clutch has three discs on the driving shaft and two on the driven shaft is to be designed for a machine tool, driven by an electric motor of 22 kW running at 1440 rpm. The inside diameter of the contact surface is 130 mm. The maximum pressure between the surfaces is limited to 0.1 N/mm². Design the clutch. Take $\mu = 0.3$, $n_1 = 3$, $n_2 = 2$.

given:-

$P = 22\,kW = 22 \times 10^3\,W$

$N = 1440\,rpm$

$d_2 = 130\,mm$ (or) $r_2 = 65\,mm$, $P_{max} = 0.1\,N/mm^2$

$= 0.65\,m$ $\quad = 0.1 \times 10^6\,N/m^2$

To find:-

Design the clutch (i.e. determine the outside diameter of disc, total number of discs, and clamping, etc).

Solution:- Assume uniform wear

1. outside diameter of disc(d_1):

W.K.T, the torque transmitted

$$T = \frac{P \times 60}{2\pi N} = \frac{22 \times 10^3 \times 60}{2\pi \times 1440} = 145.89 \text{ N-m}$$

$$\boxed{T = 145.89 \text{ N-m}}$$

Design torque, $[T] = T \times K_s$

where service factor, $K_s = K_1 + K_2 + K_3 + K_4$

From 7.90, $K_1 = 0.5$ (for electric motor)

From PSGDB 7.91, $K_2 = 1.25$ (for m/c tools)

From PSGDB 7.91, $K_3 = 0.38$ (for 1440rpm)

From PSGDB 7.91, $K_4 = 0.75$ (Assuming 32 engagements/shift)

$$K_s = 0.5 * 1.25 * 0.38 * 0.75 = 2.88$$

$$\boxed{K_s = 2.88}$$

Design torque $[T] = 145.89 \times 2.88$

$= 420.16$ N-m

W.K.T, maximum intensity of pressure (P_{max}) is at the inner radius (r_2).

Therefore $P_{max} \cdot r_2 = C$ (or) $C = 0.1 \times 10^6 \times 65 \times 10^{-3}$

$= 6500 \ N/m$.

For uniform wear, axial force exerted is given by

$$W = 2\pi C (r_1 - r_2)$$

$$= 2\pi \times 6500 (r_1 - 65\times10^{-3})$$

$$= 40840.7 (r_1 - 0.065) \quad -① $$

W.K.T

number of pairs of contact surfaces,

$$n = n_1 + n_2 - 1 = 3 + 2 - 1 = 4$$

Torque transmitted $[T] = n . \mu . w . \left(\frac{r_1 + r_2}{2}\right)$

$$420.16 = 4 \times 0.3 \times 40840.7 (r_1 - 0.065) \times \frac{(r_1 + 0.065)}{2}$$

$$= 2 \times 0.3 \times 40840.7 (r_1^2 + \cancel{0.065 r_1} - \cancel{0.065 r_1} - 4.225\times10^{-3})$$

$$= 24504.42 (r_1^2 - 4.225\times10^{-3})$$

$$r_1 = 0.14618 m = 146.18 mm$$

$$\boxed{r_1 = 146.18 \ mm}$$

$\therefore$ outer diameter of the disc $\boxed{d_1 = 292.37 mm}$

2) Total number of disc :-

Total number of disc = No. of pairs of contact surface+1

= n+1

= 4+1 = 5.

3) Clamping force (or axial force exerted)

~~total~~

substituting the value of r_1 in eqn ①, we get

Axial force exerted $W = 40840.7(0.14618 - 0.065)$

$$\boxed{W = 3315.45\ N}$$

Problem on cone clutch

JAYA
EX: 10.17
Pg. no. 10.28

A leather faced conical friction clutch has a cone angle of 30°. The intensity of pressure b/w the contact surface is not to exceed $6\times10^4\ N/m^2$ and the breadth of the conical surface is not to be greater than 1/3 of the mean radius if $\mu = 0.20$ and the clutch transmits 37kW at 2000 rpm. Find the dimensions of contact surface.

Given:-

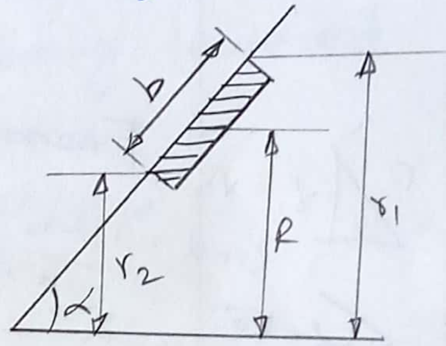
$2\alpha = 30°$ $\quad p_n = 6 \times 10^4 \ N/m^2$

$b = R/3$; $\mu = 0.2$, $P = 37 \ kW = 37 \times 10^3 \ W$

$N = 2000$ rpm.

To find:-

Dimensions of contact surface (r_1 and r_2).



Solution:-

Power transmitted, $P = \frac{2\pi N T}{60}$

$$T = \frac{P \times 60}{2\pi \times N}$$

$$= \frac{37 \times 10^3 \times 60}{2\pi \times 2000}$$

$$\boxed{T = 176.66 \ N\text{-}m}$$

Assuming service factor $K_s = 2.5$

Design torque $[T] = T \times K_s$

$= 176.66 \times 2.5$

$= 441.65 \ N\text{-}m$

Torque transmitted is also given by

$$T = 2\pi \mu p_n \cdot R^2 \cdot b$$

$$441.65 = 2\pi \times 0.2 \times 6 \times 10^4 \times R^2 \times (R/3)$$

$$= 25132.74 R^3$$

$R = 0.25998\,m$ (or) $259.98\,mm$

$$\boxed{R = 259.98\,mm}$$

Face width is given by $b = R/3 = \frac{0.25998}{3}$

$= 0.08666\,m$

(or)

$$\boxed{b = 86.66\,mm}$$

$\sin\alpha = \frac{r_1 - r_2}{b}$

$2\alpha = 30^\circ$

$\alpha = 15^\circ$

From, fig. we find that $\frac{r_1 - r_2}{b} = \sin\alpha$

$r_1 - r_2 = b \cdot \sin\alpha$

$r_1 - r_2 = 0.08666 \times \sin 15^\circ$

$r_1 - r_2 = 0.02242\,m$ — (1)

mean radius, $R = \frac{r_1 + r_2}{2} = 0.25998$

$r_1 + r_2 = 0.5199\,m$ — (2)

Solving eqn (1) & (2), we get

outer radius of contact surface

$r_1 = 0.2711$ (or) $271.1\,mm$

Inner radius of contact surface

$r_2 = 0.2487\,m$ (or) $248.7\,m$

BRAKES

Brake is a device which is used to bring to rest (or) slow down a moving body. Safe operation of vehicles demands dependable brakes. Brake is required to absorb the kinetic energy of the moving parts (or) the potential energy of the objects being lowered by hoist when the rate of descent is controlled. The absorbed energy is dissipated in the form of heat.

clutch vs brake

The functional difference between a clutch and a brake is that a clutch connects two moving members of a machine, whereas a brake connects a moving members to a stationary member. That is, if any one of the moving member of a clutch is fixed, then the device becomes a brake.

classification of brakes: [Main types of mechanical brakes]

1. Block (or) Shoe brake
   i) single block brake ii) Double block brake
2. Band brake
   i) simple band brake ii) Differential band brake.
3. Band and block brake.
4. Internal expanding Shoe brake

5. External contracting brakes.

The mechanical brakes, according to the direction of active force, may be divided into the following two groups.

i) Radial brakes:-

In radial brakes, the force acts radially on the drum

Examples: Band brakes, block brakes, and internal expanding brakes.

ii) Axial brakes:-

In axial brakes, the force acts axially on the drum.

Examples:- cone brakes and disc brakes.

Brake Lining materials.

The required qualities of a good brake lining materials are.

=> A high and uniform coefficient of friction

=> The ability to withstand high temperatures, together with high heat dissipation capacity.

=> Adequate mechanical and thermal strengths.

=> High resistance to wear

=> Resistance against environmental conditions, such as moisure and oil.

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Types of brake linings

Three basic types of lining are:

1) **organic linings:** organic linings are generally compounded of six basic ingredients.

i) Asbestos:
For heat resistance and high coefficient of friction.

ii) Friction modifier:-
To give desired friction qualities.
Example: The oil of cashew nut shells.

iii) Fillers: To control noise. Example: Rubber chips.

iv) curing agents:
To produce the desired chemical reactions during manufacture.

v) other materials:
For improved overall braking performance. Example: powdered lead, brass chips, and aluminium powders.

vi) Binders:
For holding the ingredients together. Example: Phenolic resins.

2) **Semi-metallic linings:-**
These linings substitute Iron, steel, and graphite for part of the asbestos and organic components of an organic lining.

3) metallic linings:-

These materials are obtained from metallic powder by the process of powder metallurgy in the form of 0.25 to 6.0 mm thick straps. These straps can be pressed onto steel discs on one (or) both sides.

The two types of sintered metal friction materials used are:

i) Bronze-based

ii) Iron-based.

properties of brake lining materials

Material	μ	Allowable pressure (P_{max}) Mpa	Max. Temp. °C
wood on metal	0.25	0.48	65
metal on metal	0.25	1.4	315
Leather on metal	0.35	0.17	65
Asbestos on metal in oil.	0.40	0.34	260
power metal lining on C.I in oil	0.15	2.8	260

BLOCK OR SHOE BRAKE

In this type of brake, one (or) more blocks (or) shoes are pressed against the rim surface of a brake drum. The blocks are made of wood, asbestos in rubber compound, cast iron (or) of any other suitable metal. The material of the block (or) shoe is softer than that of the drum. The blocks are pressed against the rim by the application of a force through suitable leverage (or) brake hanger.

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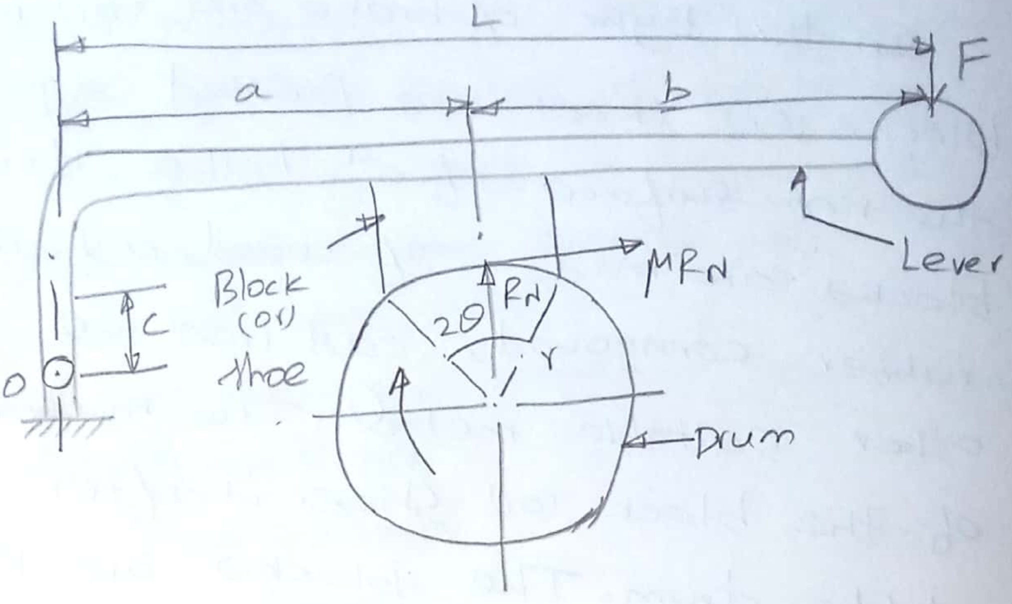
SINGLE BLOCK [OR] SHOE BRAKE



r = Radius of drum

R_N = Normal reaction of the block

F = Force applied at lever end

μ = coefficient of friction

μR_N = Frictional force

T_B = Braking torque

When the Rotation of the Drum is clockwise

Braking torque on the drum

$$T_B = \mu R_N \cdot r \quad \text{—— (1)}$$

Taking moments about pivot O

$$F \times l + \mu R_N \times c - R_N \times a = 0$$

$$F \times l = R_N \times a - \mu R_N \times c$$

$$F \times l = R_N(a - \mu c)$$

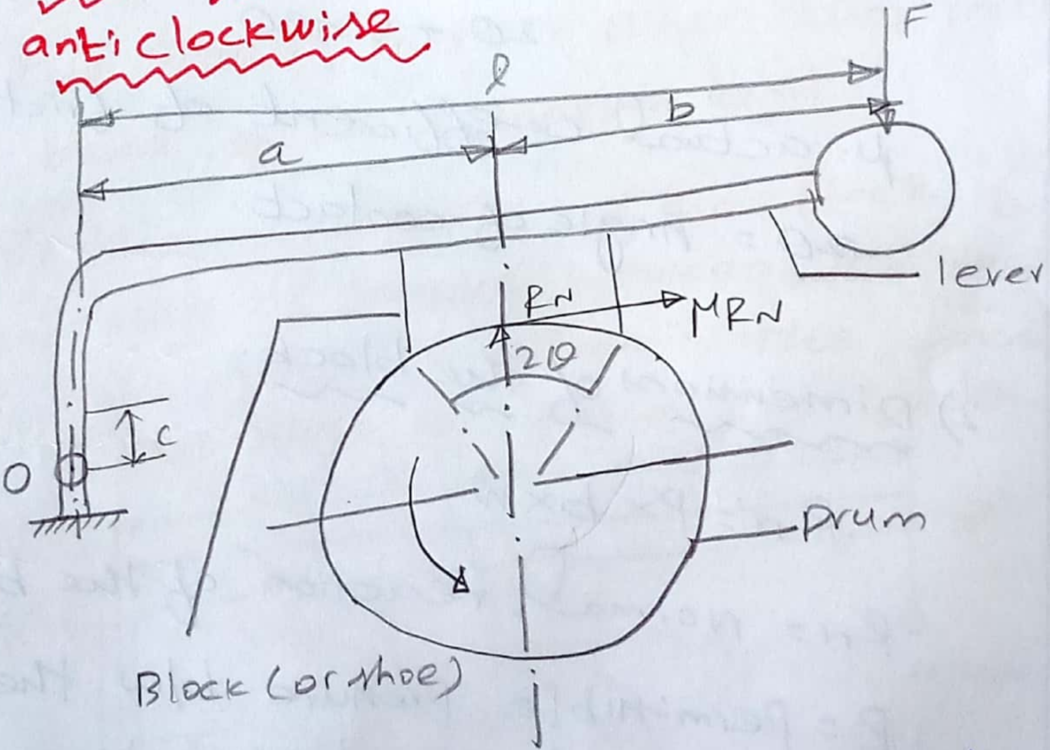
$$\boxed{F = \frac{R_N(a - \mu c)}{l}}$$

$$\boxed{R_N = \frac{F \times l}{(a - \mu c)}} \quad \text{--- (2)}$$

sub (2) in (1)

$$T_B = \mu \times \frac{F \times l \times r}{(a - \mu c)}$$

When the rotation of the Drum is anticlockwise



Consider the anticlockwise rotation of brake drum as shown in fig.

$$R_N = \frac{F \times l}{a + \mu c}$$

Braking torque $= T_B = \dfrac{\mu \times F \times l \times r}{a + \mu c}$

Note:-

1) pivoted Block (or) shoe Brake
($2\theta > 40°$)

Equivalent coefficient of friction

$$\mu' = \frac{4\mu \sin\theta}{2\theta + \sin 2\theta}$$

μ: actual co-efficient of friction

2θ = Angle of contact.

2) Dimensions of the block:

$R_N = P \times b \times w$

R_N = Normal reaction of the block

P = permissible pressure b/w the block and the brake drum.

w = width of the block.

3) Rate of heat generated:

Rate of heat generated $= \{$Frictional force$\} \times$ Avg. velocity

Rated of heat generated = $\mu \cdot R_N \times v$

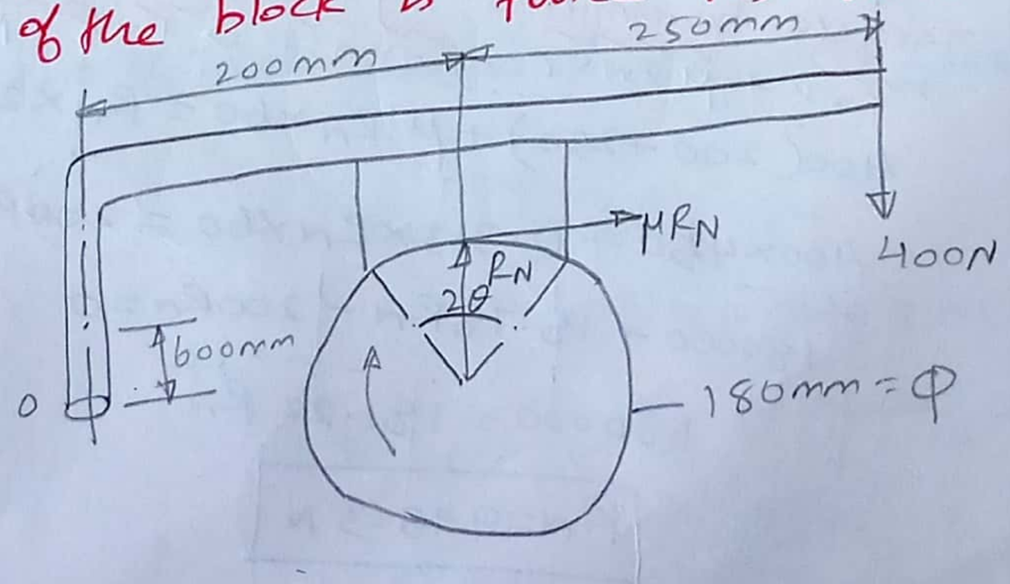
JAYA Ex: 11.1 Pg - 11.6

A single block brake is shown in Fig. The diameter of the drum is 180mm and the angle of contact is 60°. If the operating force of 400N is applied at the end of a lever and the coefficient of friction between the drum and the lining is 0.30, determine

(i) the torque that may be transmitted by the block brake.

(ii) the rate of heat generated during the braking action, when the initial brake speed is 300rpm and

(iii) the dimensions of the block if the intensity of pressure between the block and brake drum is $1 N/mm^2$. The breadth of the block is twice its width.



Given:-

$d = 180mm$ (or) $r = 90mm = 90\times10^{-3}m$

$F = 400N$, $2\theta = 60^\circ = 60\times\pi/180 = \pi/3\ rad$, $\mu = 0.30$

$N = 300rpm$, $b = 2W$

To find:-

(i) Torque transmitted by the block shoe.

(ii) Rate of heat generated during braking

(iii) Dimensions of block.

Solution:-

$2\theta > 40^\circ$, find equivalent coefficient of friction.

$$\mu' = \frac{4\mu\sin\theta}{2\theta+\sin2\theta} = \frac{4\times0.30\times\sin30^\circ}{\pi/3+\sin60^\circ}$$

$$\boxed{\mu' = 0.313}$$

$l = 200+250$
$= 450$

(i) Torque transmitted by the block shoe (T):

$F\times l + \mu R_N\times c - R_N\times a = 0 \Rightarrow F\times l + \mu R_N\times c = R_N\times a$

$F\times l + \mu R_N\times c$

$400(200+250) + \mu' R_N\times60 = R_N\times200$

$400\times450 + 0.313\times R_N\times60 = 200R_N$

$180000 + 18.78R_N - 200R_N = 0$

$180000 = 181.22\ R_N$

$$\boxed{R_N = 993.3\ N}$$

Braking force $= \mu' R_N = 0.313 \times 993.3$

$= 310.9\,N$

Braking torque $= \mu' R_N \times r = 310.9 \times 0.09$

$$\boxed{T_B = 27.98\ N\text{-}m}$$

(ii) Rate of heat generated during braking

Initial velocity of the drum $v_1 = \frac{\pi d N}{60}$

$= \frac{\pi \times 0.018 \times 300}{60}$

$= 2.827\ m/s$

Final velocity of the drum $v_2 = 0$

Avg. velocity of the drum $= v = \frac{v_1 + v_2}{2}$

$= \frac{2.827 + 0}{2}$

$= 1.414\ m/s$

Rate of heat generated = Frictional force × Avg. velocity

$= \mu R_N \times v$

$= 0.3 \times 993.3 \times 1.414$

$= 421.36\ N\text{-}m/s$ (or) W

$= 421.36\,W$

(iii) Diamensions of the brake shoe: -

$$R_N = P \times b \times W$$

$$993.3 = 1 \times 2W \times W = 2W^2$$

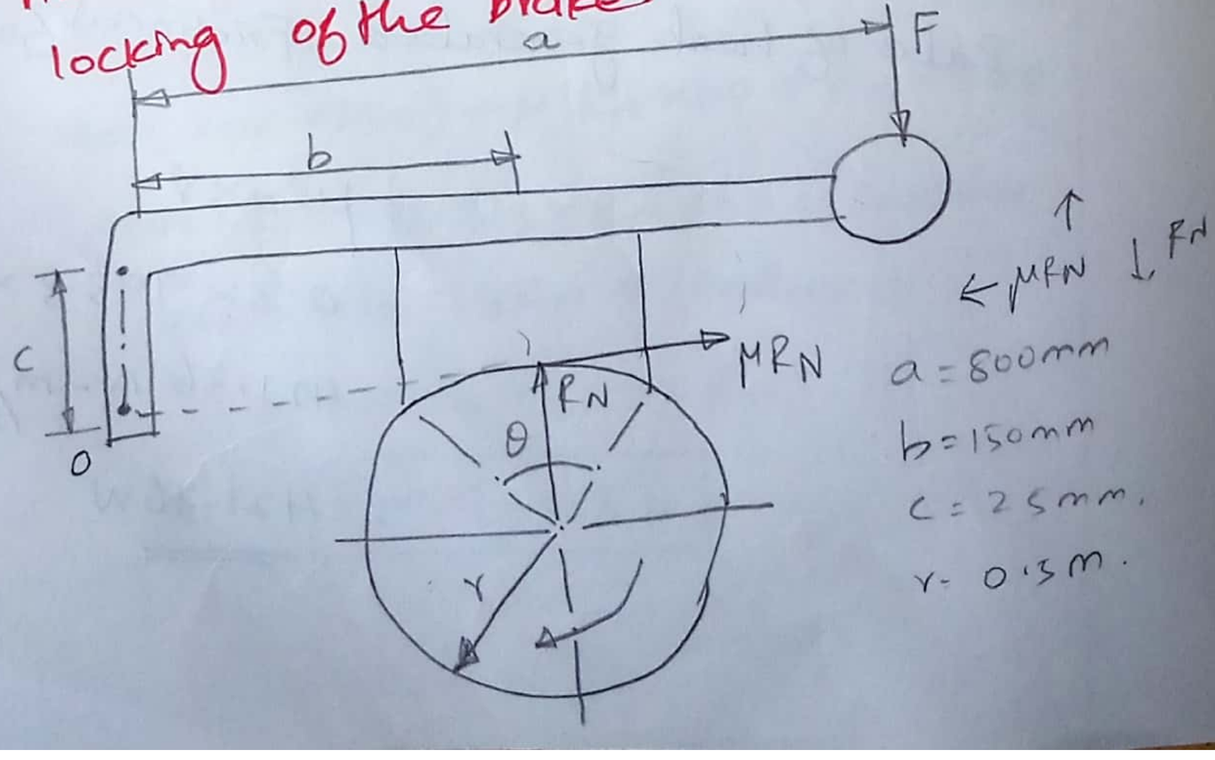
$$\boxed{W = 22.285 \, mm}$$

$$b = 2W = 2 \times 22.285 = 44.57 \, mm$$

$$\boxed{b = 44.57 \, mm}$$

JAYA
Ex: 11.2
Pg - 11.8

The diameter of the brake drum of a single block shown in fig. is 1m. It sustains 240N-m of torque at 400rpm. The coefficient of friction is 0.32. Determine the required force to be applied when the rotation of the drum is (a) clockwise, (b) counter clockwise, and the angle of contact (i) 35° and (ii) 100°. Given that a = 800mm, b = 150mm and c = 25mm. Also the new values of 'c' for self locking of the brake.



a = 800mm

b = 150mm

c = 25mm.

r = 0.5m.

Given:-

$d = 1m$ (or) $r = 0.5m$

$T_B = 240 N\text{-}m$, $N = 400 rpm$, $\mu = 0.32$

$a = 800 mm$ $b = 150 mm = 0.15 m$ $c = 25 mm$
$= 0.025 m.$

$a = 0.8 m$

Solution:-

$T_B = \mu R_N \cdot r$

$240 = 0.32 \times R_N \times 0.50$

$\boxed{R_N = 1500 N}$

(i) when angle of contact, $2\theta = 35°$

(a) Rotation of drum clockwise:

Taking moments about o, we get

$F \cdot a + \mu R_N \times c - R_N \times b = 0$

$F \times a = -\mu R_N \times c + R_N \times b$

$F \times a = R_N \times b - \mu R_N \times c$

$F \times 0.8 = \cancel{0.32 \times 1500 \times 0.025 + 1500 \times 0.15}$ $1500 \times 0.15 - (0.32 \times 1500 \times 0.025)$

$\boxed{F = \cancel{296.25 N}}$ $\boxed{F = 213 N}$

(b) Rotation of drum counter clockwise:-

Taking moments about o, we get.

$R_N \times b - F \times a - \mu R_N \times c = 0$

$R_N \times b = F \times a + \mu R_N \times c$

$F \times a = R_N \times b + \mu R_N \times c$

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$1500 \times 0.15 = F \times 0.80 + 0.32 \times 1500 \times 0.025$

$\boxed{F = 266.25\,N}$

(c) New value of 'c' for self-locking of the brake:

F=0

$R_N \cdot b = \mu R_N \times c$

$b = \mu c$

$c = b/\mu = \frac{0.15}{0.32} = 0.469m = 469mm$

$\boxed{c = 469mm}$

(ii) when angle of contact $2\theta = 100°$

$2\theta = 100°$, to find equivalent friction

$\mu' = \frac{4\mu \sin\theta}{2\theta + \sin 2\theta} = \frac{4 \times 0.32 \times \sin 50°}{100 \times \pi/180 + \sin 100°}$

$\boxed{\mu' = 0.359}$

Braking torque $T_B = \mu' R_N \times r$

$240 = 0.359 \times R_N \times 0.50$

$\boxed{R_N = 1337N}$

(a) Rotation of drum clockwise

$F \times a = R_N \times b + \mu' R_N \times c$

$F \times 0.80 = 1337 \times 0.15 + 0.359 \times 1337 \times 0.025$

$\boxed{F = 265.7 N}$

(b) Rotation of drum counter clockwise

$F \times a = R_N \times b - \mu' R_N \times c$

$F \times 0.80 = 1337 \times 0.15 - 0.359 \times 1337 \times 0.025$

$\boxed{F = 235.7 N}$

(c) New value of 'c' for self-locking of the brake:

For self locking, the externally applied force F must be zero. This is possible for counter clockwise rotation of the drum.

$F \times a = R_N \times b - \mu' R_N \times c$

$0 = R_N \times b - \mu' R_N \times c$

$c = b/\mu = \frac{0.15}{0.359} = 0.417 m$

$\boxed{c = 417 mm}$

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DOUBLE BLOCK [OR] DOUBLE SHOE BRAKE

⇒ If only one block is used for braking, then there will be side thrust on the bearing of wheel shaft.

⇒ This drawback can be removed by providing two blocks on the two side of the drum.

Let S = spring force ~~applied~~ required to set the blocks on the drum.

r = Radius of drum.

R_{N_1} and μR_{N_1} = Normal reaction and the braking force on the left side.

R_{N_2} and μR_{N_2} = Normal reaction and the braking force on the right side.

The drum is rotating in the clockwise direction

Taking moment about the fulcrum O_1

$$S \times b + \mu R_{N_1}\left[r - c/2\right] - R_{N_1} \times a = 0$$

$$S \times b + \mu R_{N_1}\left[r - c/2\right] = R_{N_1} \times a$$

Taking moment about the fulcrum O_2

$$-S \times b + \mu R_{N_2}\left[r - c/2\right] + R_{N_2} \times a = 0$$

$$S \times b - \mu R_{N_2}\left[r - c/2\right] - R_{N_2} \times a = 0$$

$$S \times b - \mu R_{N_2}[r - c/2] = R_{N_2} \times a.$$

Double shoe brake, braking torque

$$T_B = (\mu R_{N_1} + \mu R_{N_2}) r = \mu r (R_{N_1} + R_{N_2}$$

projected bearing area of one shoe is given by

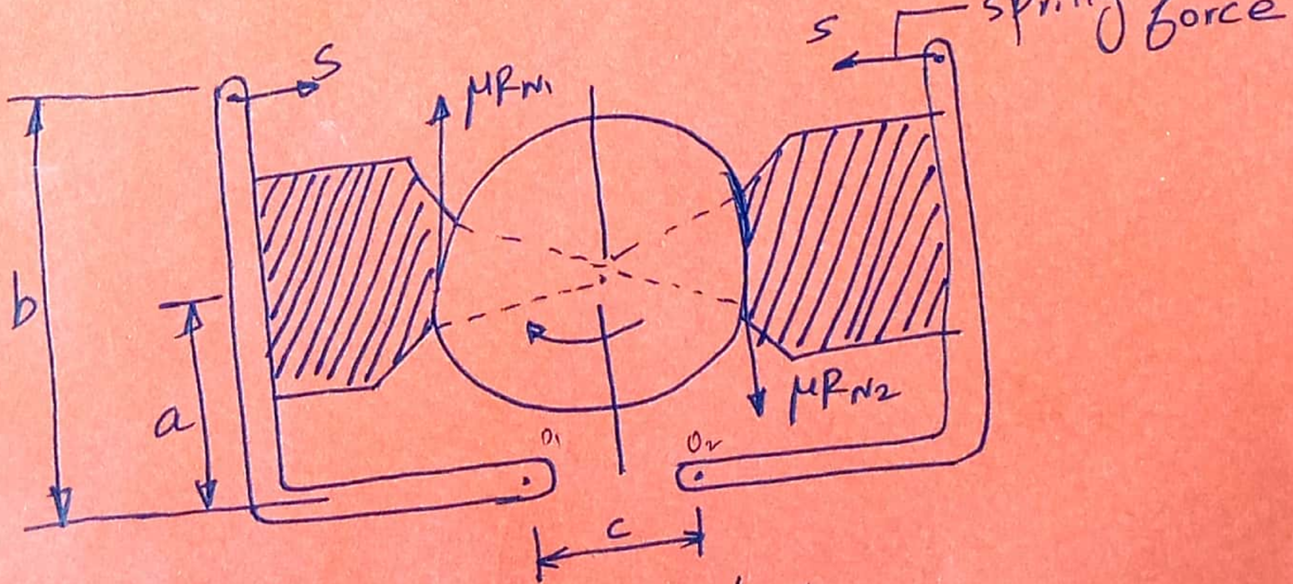
$$A = 2 r b \sin\theta$$

Bearing pressure

$$p = \frac{R_N}{A}$$

b = width of braking shoe

R_N = max. normal load.



Double shoe brake

Note:-

1) total energy to be absorbed

E_T = change in K.E of load + change in P.E of load + change in KE of all other rotating parts

$$= \frac{1}{2} m (v_1^2 - v_2^2) + W \times x + \frac{1}{2} I \omega^2.$$

2) Braking torque in terms of total energy absorbed

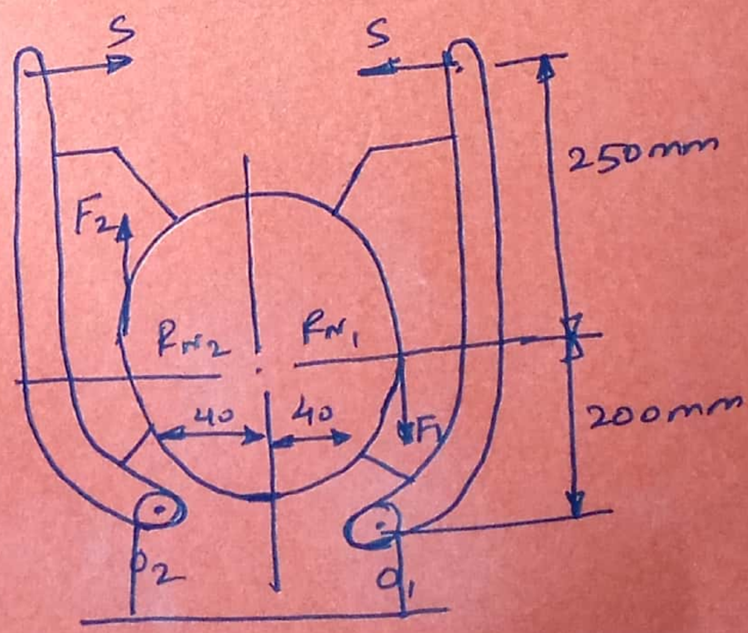
$$T_B = \frac{60 \times E_T}{\pi \times N_1 \times t}$$

N_1 = Initial speed of brake drum

t = Time of application of brake.

Ex: 11.4
Pg - 11.12

The block brake shown in fig., is set by a spring that produces a force S on each arch equal to 3500N. The wheel diameter is 350mm and the angle of contact for each block is 120°. Take coefficient of friction as 0.35, determine (i) the maximum torque that the brake is capable of absorbing, and (ii) the width of the brake shoes, if the bearing pressure on the lining material is not to exceed 0.3 N/mm².



Given:-

$S = 3500N \quad d = 350mm \quad (or) \quad r = 175mm$
$= 0.175m,$

$2\theta = 120° = 120 \times \pi/180° = 2.094$ rad, $\mu = 0.35$

$P = 0.3 N/mm^2$

To find:-

(i) Torque absorbed by the brake

(ii) width of the brake shoe.

Solution:-

$2\theta > 40°$, find equivalent coefficient friction

$$\mu' = \frac{4\mu \sin\theta}{2\theta + \sin 2\theta} = \frac{4 \times 0.35 \times \sin 60°}{2.094 + \sin 120°}$$

$$\boxed{\mu' = 0.409}$$

(i) Torque absorbed by the brake (T_B)

consider the left hand side brake shoe,

$S(25+20) = R_{N_2} \times 20 - F_2(17.5-4)$

$455 = R_{N_2} \times 20 - F_2 \times 13.5$

$[\because F_2 = \mu^2 R_{N_2}]$

$= \left[\frac{20}{0.409} - 13.5\right] F_2 \qquad R_{N_2} = \frac{F_2}{\mu'} = \frac{F_2}{0.409}$

$45 \times 3500 = 35.4 F_2$

$\boxed{F_2 = 4449.2N}$

Consider right hand side brake shoe:

Taking moments about fulcrum O, we get

$$S(20+25) = R_{N_1} \times 20 + F_1(17.5-4)$$

$$45S = F_1\left[\frac{20}{0.409} + 13.5\right] \quad \left[\because F_1 = \mu' R_{N_1} \quad R_{N_1} = \frac{F_1}{\mu'} = \frac{F_1}{0.409}\right]$$

$$45 \times 3500 = 62.39 F_1$$

$$\boxed{F_1 = 2524.1N}$$

Braking torque T_B is given by, $T_B = (F_1 + F_2) r$

$$= (2524.1 + 4449.2) \times 0.175$$

$$\boxed{T_B = 1220.32 N\text{-}m}$$

(ii) width of the brake shoe (b):

b = width of the brake shoes in mm

$$A = 2rb\sin\theta$$

$$= 2 \times 125 \times b \times \sin 55^\circ = 204.79b \ mm^2$$

$$\boxed{A = 204.79b \ mm^2}$$

Normal force on the left hand side of the shoe.

$$R_{N_1} = \frac{F_1}{\mu'} = \frac{0.6\cancel{04}S}{\mu'} = \frac{0.604 \times 1\cancel{42}4.5}{0.344}$$

$$\boxed{R_{N_1} = 2501.16 N}$$

Normal force on the left hand side of the shoe

$$R_{N_2} = \frac{F_2}{\mu'} = \frac{0.8S}{\mu'} = \frac{0.8 \times 1424.5}{0.344} = 3312.79N$$

$\boxed{R_{N2} = 3312.79N}$ $R_{N2} > R_{N1}$, R_{N2} is used in calculating the max bearing pressure

W.K.T, the bearing pressure on the lining material,

$$P = \frac{R_{N2}}{A} = \frac{3312.79}{204.79b} = \frac{16.18}{b} N/mm^2 = \frac{16.18}{b} \times 10^6 N/m^2$$

and rubbing velocity

$$v = \frac{\pi d N}{60} = \frac{\pi \times 0.25 \times 650}{60} = 8.51 m/s$$

$$PV = \frac{16.18}{b} \times 10^6 \times 8.51 = \frac{1.376 \times 10^8}{b} N/m\text{-}s$$

$$PV = 1000 \text{ (kPa) m/s} = 1000 \times 10^3 N/m\text{-}s$$

$$1000 \times 10^3 = \frac{1.376 \times 10^8}{b}$$

(or) width of block shoe

$$\boxed{b = 137.66mm}$$

(iii) wear ratio: W.K.T the wear of block shoe depends upon the friction force.

$$\text{wear ratio} = \frac{F_1}{F_2}$$

$$\frac{F_1}{F_2} = \frac{0.6045}{0.85} = \underline{0.755}$$

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JAYA
Ex: 11.5
Pg - 11.14

The layout of a double block brake is shown in fig. The brake is rated at 250N-m at 650rpm. The drum diameter is 250mm. Assuming coefficient of friction to be 0.3 and for conditions of service a PV value of 1000 (KPa) m/s may be assumed. Determine.

(a) spring force 'S' required to set the brake, and

(b) width of shoes

which shoe will have greater rate of wear and what will be the ratio of rates of wear of the two shoes?

Given:-

$T_B = 250 N\text{-}m$

$N = 650 rpm$

$d = 250 mm$

$r = 125 mm$

$\mu = 0.3$

$PV = 1000$

$2\theta = 110^\circ = 110 \times \pi/180$

$= 1.92 rad$

solution:-

$$\mu' = \frac{4\mu \sin\theta}{2\theta + \sin 2\theta} = \frac{4 \times 0.3 \times \sin 55^\circ}{1.92 + \sin 110^\circ} = 0.344$$

(i) Spring force (S) required:

Consider right hand side brake shoe.

Taking moments about fulcrum O_1, we get

$$-S(160+160) + (R_{N_1} \times 160) + F_1(125-60) = 0$$

$\times(-)$

$$S(160+160) - (R_{N_1} \times 160) - F_1(65) = 0$$

$$320S = 160 R_{N_1} + 65F_1$$

$F_1 = \mu' R_{N_1}$

$$= 160 \times \frac{F_1}{0.344} + 65F_1$$

$R_{N_1} = \frac{F_1}{\mu'}$

$$= F_1\left(\frac{160}{0.344} + 65\right)$$

$$\boxed{0.604S = F_1} \quad -(1)$$

Consider left hand side brake shoe

Taking moments about fulcrum O_2,

$$S(160+160) + F_2(125-60) - (R_{N_2} \times 160) = 0$$

$$320S + 65F_2 - 160 R_{N_2} = 0$$

$$320S = 160R_{N_2} - 65F_2$$

$F_2 = \mu' R_{N_2}$

$$= 160 \times \frac{F_2}{0.344} - 65F_2$$

$R_{N_2} = F_2 / \mu'$

$$\boxed{0.8S = F_2} \quad -(2)$$

Braking torque is given by

$$T_B = (F_1 + F_2) r$$

$$250 = (0.604S + 0.8S) \times 0.125$$

spring force $\boxed{S = 1424.5N}$

(ii) <u>width of the brake shoe (b)</u>

$$A = 2rb\sin\theta$$

$$= 2 \times 125 \times b \times \sin 55^\circ$$

$$A = 204.79b \, mm^2$$

Normal reaction on the right hand side of the shoe

$$R_{N1} = \frac{F_1}{\mu'} = \frac{0.604S}{0.344} = \frac{0.604 \times 1424.5}{0.344}$$

$$\boxed{R_{N1} = 2501.16N}$$

Normal reaction on the Left hand side of the shoe

$$R_{N2} = \frac{F_2}{\mu'} = \frac{0.8S}{0.344} = \frac{0.8 \times 1424.5}{0.344}$$

$$\boxed{R_{N2} = 3312.79N}$$

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since $R_{r_2} > R_{r_1}$ ∴ R_{r_2} will be used to calculating the maximum bearing pressure.

bearing pressure

$$P = \frac{R_{r_2}}{A} = \frac{3312.79}{204.796} = \frac{16.18}{b} \, N/mm^2$$

$$= \frac{16.18}{b} \times 10^6 \, N/m^2$$

rubbing velocity $V = \frac{\pi d N}{60} = \frac{\pi \times 0.25 \times 650}{60}$

$$\boxed{V = 8.51 m/s}$$

$$PV = \frac{16.18}{b} \times 10^6 \times 8.51$$

$$= \frac{1.376 \times 10^8}{b} \, N/m\text{-}s$$

given that $PV = 1000 \, (kPa) \, m/s$

$$= 1000 \times 10^3 \, N/m\text{-}s$$

$$1000 \times 10^3 = \frac{1.376 \times 10^8}{b}$$

$$\boxed{b = 137.66 mm}$$

(iii) wear ratio :-

$$\text{wear ratio} = \frac{F_1}{F_2}$$

$$\frac{F_1}{F_2} = \frac{0.6045}{0.85} = 0.755.$$

As $F_2 > F_1$, Left hand side will have greater wear.

Design procedure for Block Brake

1) calculate the total energy absorbed by the brake

$$E_T = \frac{1}{2} m v^2 + \frac{1}{2} I \omega^2 + W \times x$$

2) calculate the torque capacity (or braking torque) by using the relation

$$T_B = \frac{60 E_T}{\pi \times N_1 \times t}$$

N_1 = Initial speed

t = Time of application of brake.

3) cal. the initial braking power by using the relation.

$$P = \frac{2\pi N_1 T_B}{60}$$

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4) select (or assume) the brake drum diameter

5) select the suitable brake drum and block shoe materials. For the chosen materials consulting Table-1 ~~[illegible]~~, the coefficient of friction is obtained. [PSGDB: 7.128]

6) consulting Table, cal. the induced bearing pressure

operating conditions	pv (mpa) m/s		
continuous service, poor heat dissipation	1.05		
intermittent service poor heat dissipation	2.1		
continuous service good heat dissipation as in oil bath	3.01		

~~consulting to [illegible]. the induced bearing pressure(p))~~

7) cal. the projected area of the shoe by using the relation $A = R_N / P$.

8) Finally calculate the breath and width of the shoe by using the relation projected area of the shoe.

A = Breath x width.

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JAYA
Ex: 11.b
Pg - 11.17

Determine the capacity and the main dimensions of a double block brake for the following data:
The brake sheave is mounted on the drum shaft. The hoist with its load weighs 45 kN and moves downwards with a velocity of 1.15 mps (1.15 m/s). The pitch diameter of the hoist drum is 1.25 m. The hoist must be stopped within a distance of 3.25 m. The kinetic energy of the drum may be neglected.

Given:-

Load = 45 KN, $V = 1.15 m/s$, $D = 1.25 m$

$x = 3.25 m$.

To find:- capacity and main dimensions of a double block brake.

(1) Calculation of the total energy absorbed by the brake

a) kinetic energy $= \frac{1}{2} m v^2 = \frac{1}{2} m(v_1^2 - v_2^2)$

b) potential energy = weight × vertical distance

$= W \times x$

c) Kinetic energy of rotation $= \frac{1}{2} I\omega^2$

Total energy $E_T = \frac{1}{2} m(v_1^2 - v_2^2) + W \times x$
$+ \frac{1}{2} I\omega^2$

neg. the kinetic energy of the drum

$E_T = \frac{1}{2} m(v_1^2 - v_2^2) + W \times x$

Initial velocity $v_1 = 1.15 m/s$

Final " $v_2 = 0$

$E_T = \frac{1}{2} \times \frac{45000}{9.81} (1.15^2 - 0^2) + (45000 \times 3.25)$

$\boxed{E_T = 149.283 \; KN\text{-}m}$

2) calculation of braking torque (or torque capacity)

$$T_B = \frac{60 \times E_T}{\pi \times N_1 \times t}$$

Distance travelled by the load $x = 3.25 m$

$x = \frac{1}{2}(v_1 + v_2) t$

$3.25 = \frac{1}{2}(1.15 + 0) t$

$\boxed{t = 5.652 sec}$

Initial speed of brake drum

$$N_1 = \frac{60 \times V_1}{\pi D} \qquad \left[V_1 = \frac{\pi D N_1}{60} \right.$$

$$= \frac{60 \times 1.15}{\pi \times 1.25} = 17.57 \text{ rpm}.$$

$$\boxed{N_1 = 17.57 \text{ rpm}}$$

Braking torque $T_B = \frac{60 \times 149.283 \times 10^3}{\pi \times 17.57 \times 5.652}$

$$\boxed{T_B = 28.71 \text{ kN-m}}$$

3) calculation of initial braking power

Braking power $P = \frac{2\pi N_1 T_B}{60}$

$$P = \frac{2\pi \times 17.57 \times 28.71 \times 10^3}{60}$$

$$\boxed{P = 52.82 \text{ KW}}$$

4) selection of brake drum diameter:-

Assume a brake drum diameter = 1.5 m

5) selection of brake drum and block shoe materials.

brake drum - cast Iron

block shoe - sintered metal may be chosen.

6) selection of induced bearing pressure:-

~~From PSGDB: T.129~~

Assume $PV = 1.05$ (Mpa) m/s is selected

$PV \le 1.05$ (Mpa) m/s

$$P \le \frac{1.05}{1.15} \le 0.913 \text{ Mpa}$$

From Table, PSGDB T.129 $P_{max} = 2.8$ Mpa

For conventional purpose $P = 2.5$ Mpa may be used.

7) calculation of projected area of the shoe

$$P = \frac{R_N}{A}$$

To find R_N: Assume equal frictional force on each shoe

$$\boxed{\text{Braking torque} = F \times D/2 \times 2}$$

$$28.71 \times 10^3 = F \times \frac{1.5}{2} \times 2$$

Friction force $F = 19140$ N

Normal reaction $R_N = F/\mu$

$$= \frac{19140}{0.15}$$

$$= 127.6 \text{ KN}$$

Projected area of shoe $A = \frac{R_N}{P} = \frac{127.6\times10^3}{2.5\times10^6}$

$$\boxed{A = 0.051 m^2}$$

8) Calculation of breadth and width of the shoe:

Assume $b = 2W$

Projected area of the shoe

$A = \text{Breadth} \times \text{width}$

$= b \times W = 2W^2$

$0.051 = 2W^2$

$W = 0.15968 m$

$$\boxed{W = 159.68 mm}$$

$b = 2W = 2\times159.68 = 319.37 mm$

$$\boxed{b = 319.37 mm}$$