DHANALAKSHMI COLLEGE OF ENGINEERING

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Chennai - 601 301

DEPARTMENT OF MECHANICAL ENGINEERING

III YEAR MECHANICAL - VI SEMESTER

ME 6601 – DESIGN OF TRANSMISSION SYSTEMS

EVEN SEMESTER

UNIT - V

STUDY NOTES

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UNIT V

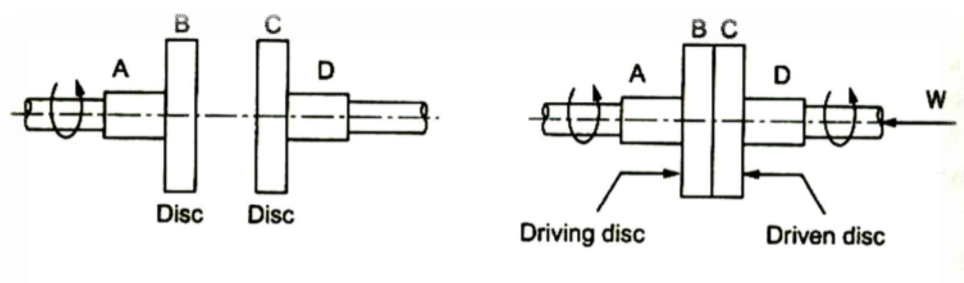
DESIGN OF CAM CLUTCHES AND BRAKES

Cam Design: Types-pressure angle and under cutting base circle determination-forces and surface stresses.

Design of plate clutches –axial clutches-cone clutches-internal expanding rim clutches – internal and external shoe brakes.

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 between them. The friction between the two surfaces depends upon the area of the surfaces, pressure applied upon them and coefficient of friction of the surface materials. The principles of friction clutch is understood with the help of the following fig. The two surfaces can be separated and brought into contact when required. One surface in considered as driving member and the other as driven member. The driving member is kept rotating. When the driven member 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 principles behind the operation of the clutch.



(a) Before engagement (b) After engagement

CLASSIFICATION OF CLUTCHES

The clutches are classified in two ways:

1. **Based on the engagement or actuation method used*:**
   a) Mechanics,
   b) Pneumatic,
   c) Hydraulic,
   d) Electrical and
   e) Automatic.
   f)
2. ***Based on the basic operating principle used**:***
   a) Positive contact clutches

- Square jaw
- Spiral jaw
- Toothed

b) Frictional clutches
- Axial
- Radial
- Cone

c) Overrunning clutches
- Roller
- Sprag
- Wrap – spring

d) Magnetic clutches
- Magnetic particles
- Hysteresis
- Eddy current

e) Fluid coupling
- Dry fluid
- Hydraulic

SINGLE PLATE CLUTCH

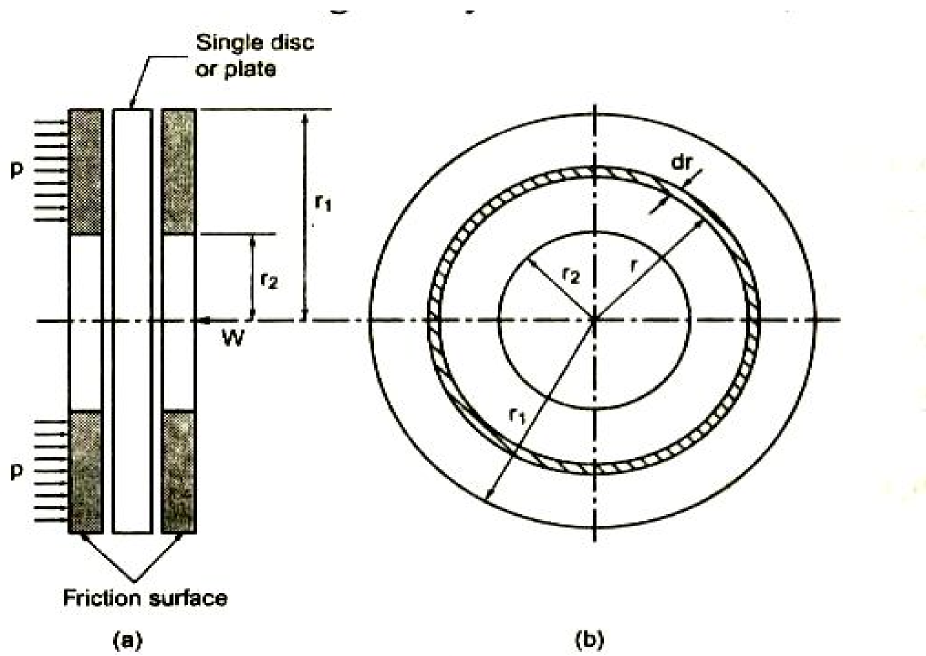
This type of clutch is mostly used in motor vehicles. Is consists of one clutch plate, clutch shaft, clutch spring, pressure plate, friction lining and bearing. The flywheel is mounted on the engine crankshaft and rotes with it the pressure plate is bolted to the flywheel is mounted on the engine crankshaft and rotates with it. The pressure plate is bolted to the flywheel through clutch springs. The friction linings are on both sides of the clutch plate. Figure shows the arrangement of single plate clutch.

Operation: When the clutch is engaged, the clutch plate is gripped between the flywheel and the pressure plates. Due to friction, the clutch plate and shaft revolves. When the clutch pedal is pressed, the pressure plate moves back against the force of the springs, and the clutch plate becomes free between the flywheel and the pressure plate.

Thus the flywheel remain rotating as long as the engine is running and the clutch shaft speed reduces slowly and finally it stops rotating.

Design of a single plate clutch (Torque transmitted by the single plate clutch)

Consider two friction surfaces held together by an axial thrust W, as shown in figure



Let T = Torque transmitted by the clutch,

P = Intensity of axial pressure acting on contact surfaces,

r_1 = External radius of friction surface,

r_2 = Internal radius of friction surface, and

μ = Coefficient of friction

Consider an elementary ring of radius r and thickness dr, as shown in figure
Area of the element ring = $2\pi r$. dr

Normal or axial force on the ring, $2\pi r$. dr and the frictional force on the ring acting tangentially

at radius r is given by

$$F_r = \mu \cdot \delta W = \mu p \times 2\pi r \cdot dr$$

$\therefore$ Frictional torque acting on the ring, $T_r = F_r \times r$

$$T_r = \mu p \times 2\pi r \cdot dr \times r$$

$$= 2\pi\mu \, p \, r^2 \, dr$$

The design of friction clutch is done based on any one of the following assumptions :

(i) When there is a uniform pressure, and

(ii) When there is a uniform wear.

(i) Considering uniform pressure :

Area of the friction surface, $A = \pi (r_1^2 - r_2^2)$

Uniform intensity of pressure (p) is given by

$$p = \frac{W}{A} = \frac{W}{\pi (r_1^2 - r_2^2)} \quad \ldots (10.1)$$

Total frictional torque acting on the friction surface or on the clutch is obtained by integrating the equation of the frictional torque on the elementary ring within the limits from r_2 to r_1.

$$\therefore \quad T = \int_{r_2}^{r_1} 2\pi \mu p r^2 \, dr = 2\pi \mu p \left[\frac{r^3}{3} \right]_{r_2}^{r_1} = 2\pi \mu p \left[\frac{r_1^3 - r_2^3}{3} \right]$$

Substituting the value of p from equation (10.1),

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

or

$$T = \frac{2}{3} \times \mu W \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2} \right] = \mu WR \quad \ldots (10.2)$$

where R = Mean radius of friction surface

$$R = \frac{2}{3} \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2} \right] \quad \ldots (10.3)$$

(ii) Considering uniform wear : For uniform wear, the intensity of pressure varies inversely with the distance. Therefore,

$$p \cdot r = \text{constant} = C \quad \text{or} \quad p = \frac{C}{r}$$

So $\boxed{p_1 \cdot r_1 = p_2 \cdot r_2 = C}$

where p_1 and p_2 are intensities of pressure at radii r_1 and r_2 respectively.

We know that normal or axial force on the elementary ring,

$$\delta W = p\, 2\pi\, r\, dr = 2\pi\,(p \cdot r)\, dr = 2\pi\, C\, dr \qquad [\because p = \frac{C}{r}]$$

$\therefore$ Total force acting on the friction surface,

$$W = \int_{r_2}^{r_1} 2\pi\, C\, dr = 2\pi\, C\, [\, r\,]_{r_2}^{r_1} = 2\pi C\,(r_1 - r_2)$$

or

$$\boxed{C = \frac{W}{2\pi\,(r_1 - r_2)}} \qquad \ldots (10.4)$$

We know that the frictional torque acting on the elementary ring,

$$T_r = 2\pi\mu\, p r^2\, dr = 2\pi\mu \times \frac{C}{r} \times r^2 \cdot dr \qquad [\because p = \frac{C}{r}]$$

$$= 2\pi\, \mu \cdot C \cdot r \cdot dr$$

$\therefore$ Total frictional torque on the friction surface,

$$T = \int_{r_2}^{r_1} 2\pi\, \mu \cdot C \cdot r \cdot dr = 2\pi\, \mu\, C\left[\frac{r^2}{2}\right]_{r_2}^{r_1} = 2\pi\mu\, C\left[\frac{r_1^2 - r_2^2}{2}\right]$$

$$T = \pi\, \mu\, C\, [\, r_1^2 - r_2^2\,]$$

Substituting the value of C from equation (10.4),

$$T = \pi\, \mu \times \frac{W}{2\pi\,(r_1 - r_2)} \times (\, r_1^2 - r_2^2\,)$$

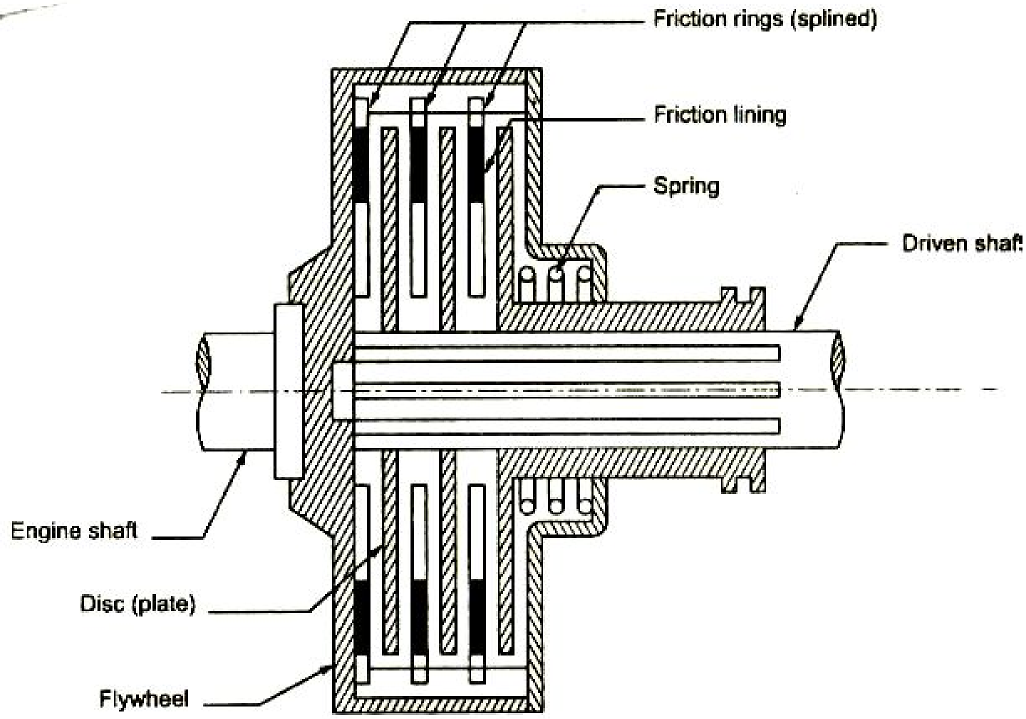
$$\boxed{T = \frac{1}{2} \times \mu \cdot W\,(r_1 + r_2) = \mu \cdot W \cdot R} \qquad \ldots (10.5)$$

where $\quad R$ = Mean radius of the friction surface

or

$$\boxed{R = \frac{r_1 + r_2}{2}} \qquad \ldots (10.6)$$

MULTIPLATE OR MULTIPLE DISC CLUTCH



A multiplate clutch is used when large amount of torque is to be transmitted. In a multiplate clutch, the number of frictional linings and the metal plates are increased which increases the capacity of the clutch to transmit torque, as shown in Fig.10.4. The multiplate clutch works in the same way as the single plate clutch by operating the clutch pedal. They are extensively used in motor cars, machine tools, *etc.*

10.7.1. Design of a Multiplate Clutch (Torque transmitted on Multiplate Clutch)

Let n_1 = Number of discs on the driving shaft, and

n_2 = Number of discs on the driven shaft.

∴ Number of pair of contact surfaces,

$$n = n_1 + n_2 - 1$$

Then, total frictional torque acting on the clutch is given by

$$\boxed{T = n \cdot \mu \cdot W \cdot R}$$

where R = Mean radius of the friction surfaces

$$R = \frac{2}{3}\left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right] \quad \text{[For uniform pressure]}$$

$$R = \frac{r_1 + r_2}{2} \quad \text{[For uniform wear]}$$

Example 10.1 *An automotive single plate clutch consists of two pairs of contacting surfaces. The inner and outer radii of friction plate are 120 mm and 250 mm respectively. The coefficient of friction is 0.25 and the total axial force is 15 kN. Calculate the power transmitting capacity of the clutch plate at 500 r.p.m. using*

(i) Uniform wear theory, and

(ii) Uniform pressure theory.

Given Data : $n = 2$; $r_1 = 250$ mm $= 0.25$ m ; $r_2 = 120$ mm $= 0.12$ m ;

$\mu = 0.25$; $W = 15$ kN $= 15 \times 10^3$ N ; $N = 500$ r.p.m.

☺ ***Solution : (i) Using uniform wear theory :*** Torque transmitted on clutch is given by

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

$$= 2 \times 0.25 \times 15 \times 10^3 \frac{(0.25 + 0.12)}{2} = 1387.5 \text{ N-m}$$

$$\therefore \text{ Power transmitted } = \frac{2\pi NT}{60} = \frac{2\pi \times 500 \times 1387.5}{60} = \mathbf{72.65 \text{ kW}} \quad \textbf{Ans.}$$

(ii) Using uniform pressure theory : Torque transmitted on clutch is given by

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

$$= 2 \times 0.25 \times 15 \times 10^3 \times \frac{2}{3} \times \left[\frac{(0.25)^3 - (0.12)^3}{(0.25)^2 - (0.12)^2} \right] = 1444.6 \text{ N-m}$$

$$\therefore \text{ Power transmitted } = \frac{2\pi N T}{60} = \frac{2\pi \times 500 \times 1444.6}{60} = \mathbf{75.64 \text{ kW}} \quad \textbf{Ans.}$$

Example 10.10 *A multiplate disc clutch transmits 55 kW of power at 1800 r.p.m. 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 Data : $P = 55\text{ kW} = 55 \times 10^3\text{ W}$; $N = 1800$ r.p.m.; $\mu = 0.1$;

$$p_{max} = 160\text{ kN/m}^2 = 160 \times 10^3\text{ N/m}^2 ;$$

$$r_2 = 80\text{ mm} = 80 \times 10^{-3}\text{ m} ; \quad r_2 = 0.7\, r_1 \quad \text{or} \quad \frac{r_2}{r_1} = 0.7.$$

To find : Number of plates needed to transmit the required torque.

☺ ***Solution :*** $r_2 = 0.7\, r_1$ or $\frac{r_2}{r_1} = 0.7$

6

or $$r_1 = \frac{r_2}{0.7} = \frac{80 \times 10^{-3}}{0.7} = 0.1143\text{ m}$$

Assuming uniform wear, axial force exerted is given by

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

We know that the maximum intensity of pressure (p_{max}) is at the inner radius (r_2).

$\therefore$ $$p_{max} \cdot r_2 = C \quad \text{or} \quad C = 160 \times 10^3 \times 80 \times 10^{-3} = 12800\text{ N/m}$$

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

$$= 2\pi \times 12800\,(0.1143 - 0.08) = 2758.57\text{ N}$$

Torque transmitted by a single friction surface is given by

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

$$= 0.1 \times 2758.57 \times \frac{(0.1143 + 0.08)}{2}$$

$\therefore$ Torque required per surface, $T = 26.8$ N-m

The total torque required can be calculated as given below.

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

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

or Total torque required, $T = 291.78$ N-m

$$\boxed{\text{Number of friction surfaces required} = \frac{\text{Total torque required}}{\text{Torque required per surface}}} \quad \dots (10.10)$$

$$= \frac{291.78}{26.8} = 10.887 \approx 11$$

$$\boxed{\text{Total number of plates} = \text{Number of pairs of contact surface} + 1} \quad \dots (10.11)$$

$$= 11 + 1 = 12 \text{ surfaces}$$

10.9.1. Design of a Cone Clutch (Torque transmitted on the cone clutch)

Consider a pair of friction surface as shown in Fig.10.5(b). Fig.10.5(c) shows a small elemental ring of radius r and thickness dr.

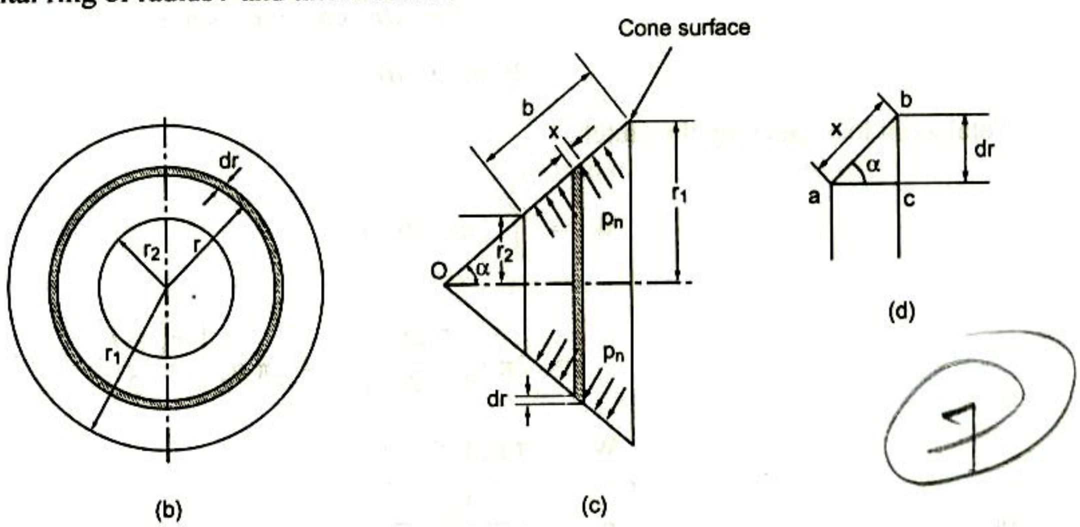


Fig. 10.5.

Let p_n = Normal intensity of pressure on friction surface,

α = Semi-angle of cone or face angle of the cone or the angle of the friction surface with the axis of the clutch,

r_1 = Outer radius of friction surface,

r_2 = Inner radius of friction surface,

R = Mean radius of the friction surface $= \frac{r_1 + r_2}{2}$,

μ = Coefficient of friction between contact surfaces, and

b = Face width of clutch plate.

Let x is length of ring of friction surface. Then from the geometry of Fig.10.5(d),

$$dr = x \cdot \sin\alpha \quad \text{or} \quad x = dr \cdot \text{cosec}\,\alpha$$

Area of the ring, $A = 2\pi r \cdot x = 2\pi r \cdot dr \cdot \text{cosec}\,\alpha$

The design of conical clutch is done based on any one of the following assumptions.

(i) When there is a uniform pressure, and

(ii) When there is a uniform wear.

(i) Considering uniform pressure :

Normal load on the ring, δW_n = Normal pressure × Area of ring

$$\delta W_n = p_n \times 2\pi r \cdot dr \cdot \text{cosec}\,\alpha$$

Axial load acting on the ring, δW = Horizontal component of δW_n

$$= \delta W_n \times \sin\alpha$$

$$= p_n \times 2\pi r \cdot dr \cdot \text{cosec}\,\alpha \times \sin\alpha$$

$$= 2\pi p_n \cdot r \cdot dr$$

Total axial load taken by the clutch,

$$W = \int_{r_2}^{r_1} 2\pi r \cdot dr \cdot p_n$$

$$= 2\pi p_n \left[\frac{r^2}{2}\right]_{r_2}^{r_1} = 2\pi p_n \left[\frac{r_1^2 - r_2^2}{2}\right]$$

$$W = \pi p_n [r_1^2 - r_2^2]$$

or

$$\boxed{p_n = \frac{W}{\pi [r_1^2 - r_2^2]}} \quad \ldots (10$$

The frictional force on the ring acting tangentially at radius r,

$$F_r = \mu \cdot \delta W_n = \mu \cdot p_n \times 2\pi r \cdot \text{cosec}\,\alpha \cdot dr$$

$\therefore$ Frictional torque acting on the ring,

$$T_r = F_r \times r = 2\pi\mu \cdot p_n \cdot \text{cosec}\,\alpha \cdot r^2\, dr$$

Then, total frictional torque $$T = \int_{r_2}^{r_1} 2\pi\mu \cdot p_n \cdot \operatorname{cosec}\alpha \cdot r^2 \cdot dr$$

$$= 2\pi\mu \cdot p_n \cdot \operatorname{cosec}\alpha \left[\frac{r^3}{3}\right]_{r_2}^{r_1}$$

$$= 2\pi\mu \cdot p_n \cdot \operatorname{cosec}\alpha \left[\frac{r_1^3 - r_2^3}{3}\right]$$

Substituting the value of p_n from equation (10.12), we get

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

or $$\boxed{T = \frac{2}{3}\mu W \cdot \operatorname{cosec}\alpha \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right]} \quad \ldots (10.13)$$

or $$\boxed{T = \mu \cdot W \cdot R \operatorname{cosec}\alpha}$$

where $$R = \frac{2}{3}\left[\frac{r_1{}^3 - r_2{}^3}{r_1{}^2 - r_2{}^2}\right] = \text{Mean radius of the friction plate}$$

(ii) Considering uniform wear : Let us assume p is the intensity of pressure at radius r.

We know that $$p \cdot r = C = \text{constant} \quad \text{or} \quad p = \frac{C}{r}$$

The normal load acting on the ring $= \delta W_n$

$= \text{Normal pressure} \times \text{Area of ring}$

$$\delta W_n = p_r \cdot 2\pi r \cdot dr \cdot \operatorname{cosec}\alpha$$

Axial load acting on the ring, $$\delta W = \delta W_n \cdot \sin\alpha$$

$$= p_r \cdot 2\pi r \cdot dr \cdot \operatorname{cosec}\alpha \cdot \sin\alpha = p_r\, 2\pi r \cdot dr$$

$$= \frac{C}{r} \times 2\pi r \cdot dr = 2\pi\, C \cdot dr \qquad [\because p_r = \frac{C}{r}]$$

∴ Total axial load transmitted to the clutch,

$$W = \int_{r_2}^{r_1} 2\pi C \cdot dr = 2\pi\, C\,[\,r\,]_{r_2}^{r_1} = 2\pi C\,(r_1 - r_2)$$

or $$\boxed{C = \frac{W}{2\pi\,(r_1 - r_2)}} \quad \ldots (10.14)$$

The frictional force acting on the ring is given by

$$F_r = \mu \cdot \delta W_n = \mu \cdot p_r \cdot 2\pi r \cdot dr \cdot \operatorname{cosec}\alpha$$

and frictional torque acting on the ring is given by

$$T = F_r \times r = \mu \cdot p_r \cdot 2\pi r \cdot dr \cdot \operatorname{cosec} \alpha \times r$$

$$= \mu \times \frac{C}{r} \times 2\pi r^2 \cdot dr \cdot \operatorname{cosec} \alpha$$

$$= 2\pi\mu \cdot C \operatorname{cosec} \alpha \times r \cdot dr$$

$\therefore$ Total frictional torque acting on the clutch,

$$T = \int_{r_2}^{r_1} 2\pi\mu \cdot C \cdot \operatorname{cosec} \alpha \cdot r \cdot dr \quad = 2\pi\mu \cdot C \cdot \operatorname{cosec} \alpha \left[\frac{r^2}{2}\right]_{r_2}^{r_1}$$

$$= 2\pi\mu \cdot C \cdot \operatorname{cosec} \alpha \left[\frac{r_1^2 - r_2^2}{2}\right]$$

Substituting the value of C from equation (10.14), we get

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

$$\therefore \quad \boxed{T = \mu \cdot W \cdot \operatorname{cosec} \alpha \left[\frac{r_1 + r_2}{2}\right]} \quad \ldots$$

or $\boxed{T = \mu \cdot W \cdot R \cdot \operatorname{cosec} \alpha}$

where $R = \frac{r_1 + r_2}{2}$ = Mean radius of the friction plate

Example 10.17 ***A leather faced conical friction clutch has a cone angle of 30°. The intensity of pressure between the contact surface is not to exceed 6 × 10⁴ N/m² and the breadth of the conical surface is not to be greater than 1/3 of the mean radius if μ = 0.20 and the clutch transmits 37 kW at 2000 r.p.m. Find the dimensions of contact surface.***

Given Data : $\alpha = 30°$; $p_n = 6 \times 10^4$ N/m²;

$b = \frac{R}{3}$; $\mu = 0.2$;

$P = 37$ kW $= 37 \times 10^3$ W;

$N = 2000$ r.p.m.

To find : Dimensions of contact surface (r_1 and r_2).

☺ ***Solution :*** Power transmitted, $P = \frac{2\pi N T}{60}$

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

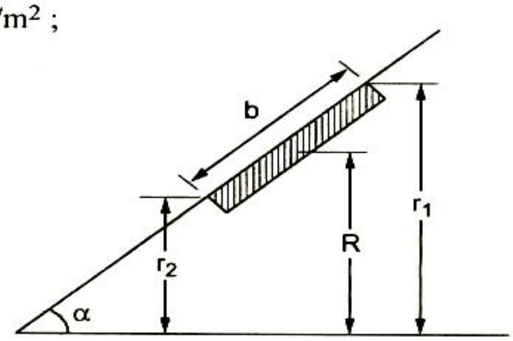


Fig. 10.6.

or $T = 176.66$ N-m

Assuming service factor, $k_s = 2.5$,

Design torque, $[T] = 176.66 \times 2.5 = 441.65$ N-m

Torque transmitted is also given by, $T = 2\pi\mu\, p_n \cdot R^2 \cdot b$

$$441.65 = 2\pi \times 0.2 \times 6 \times 10^4 \times R^2 \left(\frac{R}{3}\right) = 25132.74\ R^3$$

$$R = 0.25998 \text{ m or } 259.98 \text{ mm}$$

Face width is given by, $b = \dfrac{R}{3} = \dfrac{0.25998}{3}$

$$= 0.08666 \text{ m or } 86.66 \text{ mm}$$

From Fig.10.6, we find that $\dfrac{r_1 - r_2}{b} = \sin\alpha$

$$r_1 - r_2 = b \cdot \sin\alpha = 0.08666 \times \sin 15°$$

or $r_1 - r_2 = 0.02242$ m … (i)

and Mean radius, $R = \dfrac{r_1 + r_2}{2} = 0.25998$ m

or $r_1 + r_2 = 0.5199$ m … (ii)

Solving equations (i) and (ii), we get

Outer radius of contact surface, r_1 = **0.2711 m or 271.1 mm** and

Inner radius of contact surface, r_2 = **0.2487 m or 248.7 m Ans.**

CLASSFICATION OF BRAKES

A classification scheme for brake is presented in Fig.11.1.

From our subject point of view, the following are the main types of mechanical brakes :

1. Block or shoe brake,

   (i) Single block brake, and (ii) Double block brake

2. Band brake,

   (i) Simple band brake, and (ii) Differential band brake

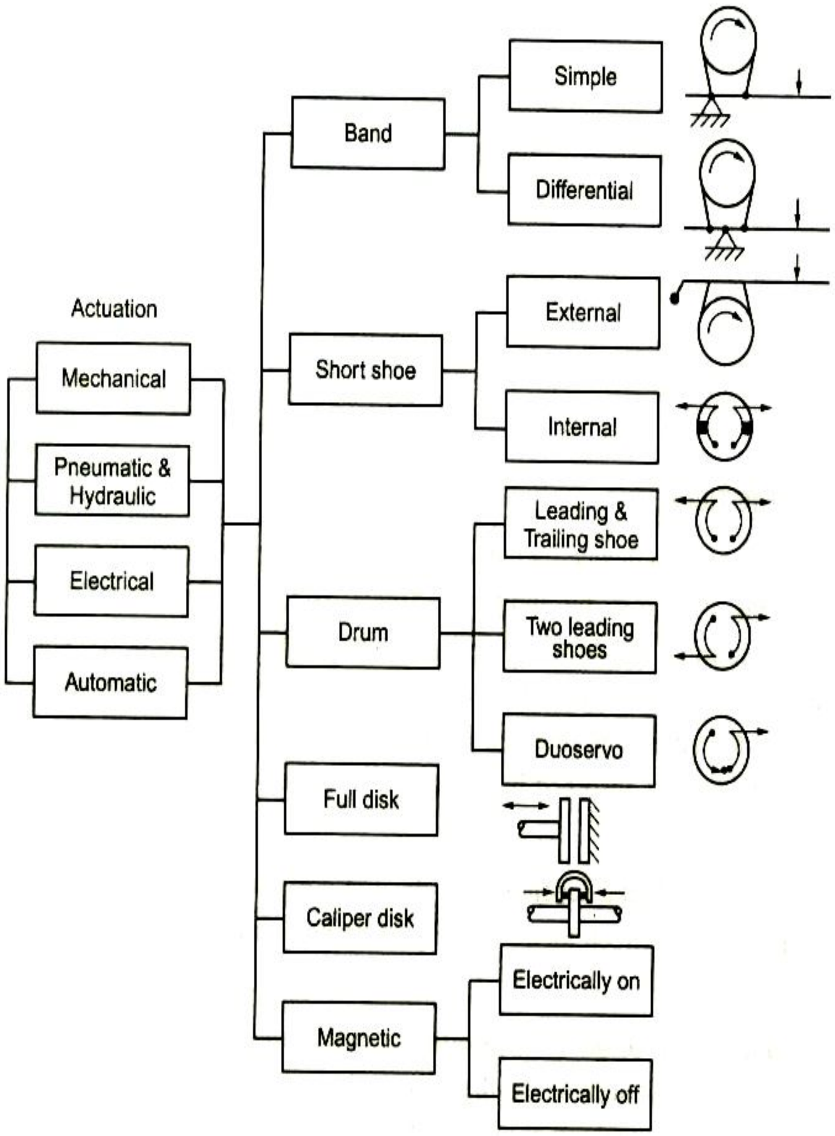


Fig. 11.1. Classification of brakes

3. Band and block brake,
4. Internal expanding shoe brake, and
5. External contracting shoe brake.

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

(a) ***Radial brakes :*** In radial brakes, the force acts radially on the drum.

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

(b) ***Axial brakes :*** In axial brakes, the force acts axially on the drum.

Examples : Cone brakes and disc brakes.

11.5. SINGLE BLOCK OR SHOE BRAKE

A single block or shoe brake is shown in Fig.11.2. The friction between the block and the brake drum causes the retarding of the drum. This type of brake is commonly used on railway trains and tram cars.

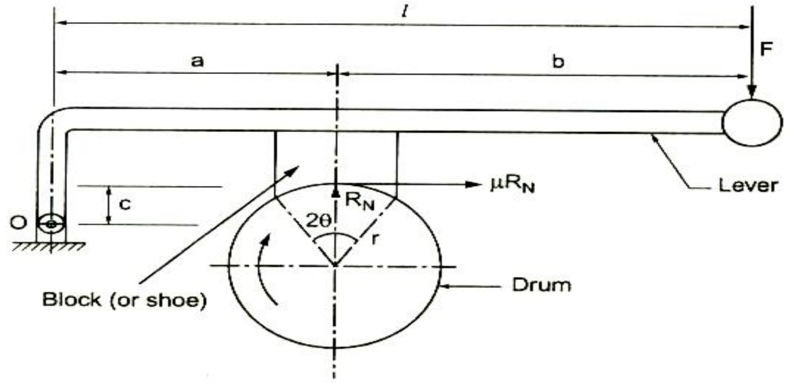


Fig. 11.2. Clockwise rotation of brake drum

The block is pressed against the drum by a force (F) applied on one end of a lever. The other end of the lever is pivoted on a fixed fulcrum O.

Let 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, and

T_B = Braking torque.

11.5.1. When the Rotation of the Drum is Clockwise

Fig.11.2, shows the clockwise rotation of brake drum.

Braking torque on the drum is given by

$$T_B = \mu R_N \cdot r \quad \text{... (i)}$$

Taking moments about pivot O,

$$F \cdot l + \mu R_N \cdot c = R_N \cdot a$$

$$F \cdot l - R_N \cdot a + \mu R_N \cdot c = 0 \quad \text{... (ii)}$$

$$F = \frac{R_N (a - \mu c)}{l} \quad \text{... (iii)}$$

or

$$R_N = F \cdot \frac{l}{a - \mu c} \quad \text{... (iv)}$$

Substituting R_N value from equation (iv) in equation (i), we have

$$\boxed{T_B = \mu \cdot \frac{F \cdot l \cdot r}{a - \mu c}} \quad \text{... (11.1)}$$

1. Self-locking Brake

In equation (iii), the value of F is zero when $a = \mu c$. It means that when the force required to apply the brake is zero, then the brake is applied automatically. ***When the frictional force is sufficient enough to apply the brake with no external force, then the brake is said to be self-locking brake.***

11.6. 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 sides of the drum, as shown in Fig.11.7. This also doubles the braking torque. The double shoes on the drum reduce the unbalanced force on the shaft. The blocks or shoes are held on the drum by means of spring force.

Let S = Spring force required to set the blocks on the drum,

r = Radius of drum,

R_{N1} and μR_{N1} = Normal reaction and the braking force on the left hand side shoe, and

R_{N2} and μR_{N1} = Normal reaction and the braking force on the right hand side shoe.

The drum is rotating in the clockwise direction. Taking moments about the fulcrum O_1, we get

$$S \cdot b + \mu R_{N1}\left[r - \frac{c}{2}\right] = R_{N1} \cdot a \qquad \ldots \text{(i)}$$

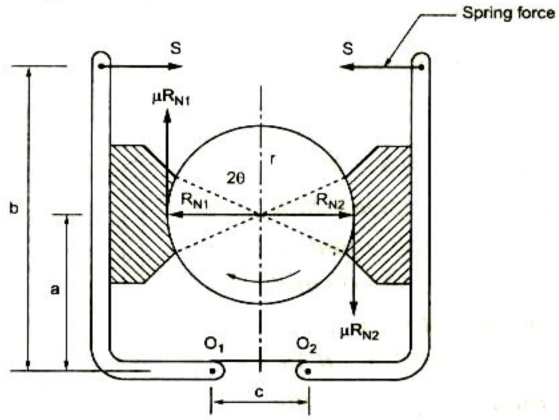


Fig. 11.7. Double shoe brake

and taking moments about fulcrum O_2, we get

$$S \cdot b - \mu R_{N2}\left[r - \frac{c}{2}\right] = R_{N2} \cdot a \qquad \ldots \text{(ii)}$$

In case of double block or shoe brake, the braking torque is given by

$$T_B = (\mu R_{N1} + \mu R_{N2}) r = \mu r (R_{N1} + R_{N2}) \quad \dots (11.6)$$

The values of R_{N1} and R_{N2} can be obtained from equations (i) and (ii). This may be substituted in equation (11.6) to get T_B.

Let b = Width of brake shoe

Then, projected bearing area of one shoe is given by

$$A = 2rb \sin\theta \quad \dots (11.7)$$

Bearing pressure on the lining material is given by

$$p = \frac{R_N}{A} \quad \dots (11.8)$$

where R_N = Maximum normal load on any shoe.

Note ***1. Total energy to be absorbed*** by a brake is given by

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

$$= \frac{1}{2} m (v_1^2 - v_2^2) + W \times x + \frac{1}{2} I w^2 \quad \dots (11.9)$$

2. Braking torque in terms of total energy absorbed by a brake is given by

$$T_B = \frac{60 \times E_T}{\pi \times N_1 \times t} \quad \dots (11.10)$$

where N_1 = Initial speed of brake drum, and

t = Time of application of brake.

Example 11.5 ***The layout of a double block brake is shown in Fig.11.9. The brake is rated at 250 N-m at 650 r.p.m. The drum diameter is 250 mm. 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 ?

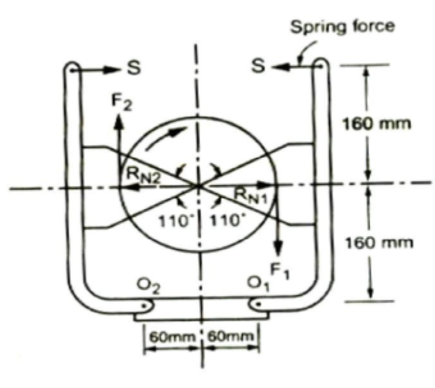


Fig. 11.9.

Given Data : T_B = 250 N-m ; N = 650 r.p.m; d = 250 mm or r = 125 mm ;

μ = 0.3 ; pv = 1000 ; $2\theta = 110° = 110 \times \frac{\pi}{180} = 1.92$

☺ ***Solution :*** Since angle of contact (2θ) is greater than 40°, therefore the equivalent coefficient of friction is given by

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

(i) Spring force (S) required : Consider right hand side brake shoe :

Taking moments about fulcrum O_1, we get

$$S\,(160 + 160) = (R_{N1} \times 160) + F_1\,(125 - 60)$$

$$320\,S = F_1 \left[\frac{160}{0.344} + 65 \right] \qquad \ldots \left[\because R_{N1} = \frac{F_1}{\mu'} \right]$$

or $$F_1 = 0.604\,S$$

Consider left hand side brake shoe :

Taking moments about fulcrum O_2, we get

$$S\,(160 + 160) = (R_{N2} \times 160) - F_2\,(125 - 60)$$

$$320\,S = F_2 \left[\frac{160}{0.344} - 65 \right] \qquad \ldots \left[\because R_{N2} = \frac{F_2}{\mu'} \right]$$

or $$F_2 = 0.8\,S$$

Braking torque is given by

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

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

or Spring force, S = **1424.5 N Ans.**

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

Let b = Width of the brake shoes in mm

We know that the projected bearing area for one shoe,

$$A = 2\,r\,b \sin\theta \qquad \ldots \text{[From equation (11.7)]}$$

$$= 2 \times 125 \times b \times \sin 55° = 204.79\,b \text{ mm}^2$$

∴ Normal force on the right hand side of the shoe,

$$R_{N1} = \frac{F_1}{\mu'} = \frac{0.604\,S}{\mu'} = \frac{0.604 \times 1424.5}{0.344} = 2501.16 \text{ N}$$

and normal force on the left hand side of the shoe,

$$R_{N2} = \frac{F_2}{\mu'} = \frac{0.8\,S}{\mu'} = \frac{0.8 \times 1424.5}{0.344} = 3312.79 \text{ N}$$

Since $R_{N2} > R_{N1}$, therefore R_{N2} will be used in calculating the maximum bearing pressure.

We know that the bearing pressure on the lining material,

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

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

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

Given that, $pv = 1000$ (kPa) m/s $= 1000 \times 10^3$ N/m-s

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

or Width of block shoe, b = **137.66 mm Ans.**

(iii) Wear ratio : We know that the wear of block shoe depends upon the friction force.

$\therefore \quad$ Wear ratio $= \frac{F_1}{F_2}$

$$\frac{F_1}{F_2} = \frac{0.604\ S}{0.8\ S} = \mathbf{0.755}$$ **Ans.**

As $F_2 > F_1$, left hand side shoe will have greater wear. **Ans.**

11.6.1. Design Procedure for Block Brake

1. Calculate the total energy absorbed by the brake.

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

2. Calculate the torque capacity (or the braking torque) by using the relation

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

where N_1 = Initial speed of brake drum, and

t = Time of application of brake.

3. Calculate the initial braking power by using the relation

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

4. Select (or assume) the brake drum diameter.
5. Select the suitable brake drum and block shoe materials. For the chosen materia consulting Table 11.1, the coefficient of friction is obtained.
6. Consulting Table 11.2, calculate the induced bearing pressure (p).

Table 11.2. Limiting values of pv (from data book, page no. 7.130)

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	3.0

7. Calculate the projected area of the shoe by using the relation, $A = \frac{R_N}{p}$

8. Finally calculate the breadth and width of the shoe by using the relation

   Projected area of the shoe, A = Breadth × Width

Example 11.6 ***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 (i.e., 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 Data : 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.

☺ ***Solution :***

1. Calculation of the total energy absorbed by the brake :

The various sources of energy to be absorbed are :

(a) Kinetic energy of translation $= \frac{1}{2} mv^2 = \frac{1}{2} m(v_1^2 - v_2^2)$

where v = Velocity at the time of applying the brake, and

v_1 and v_2 = Initial and final velocities of the load.

(b) Potential energy = Weight × Vertical distance = $W \times x$

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

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

Neglecting the kinetic energy of the drum,

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

Initial velocity of load, v_1 = 1.15 m/s ... [Given]

and final velocity of load, v_2 = 0

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

$$= \mathbf{149.283\ kN\text{-}m}$$ **Ans.** ☜

2. Calculation of braking torque (or torque capacity) :

Braking torque in terms of energy absorbed is given by

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

where N_1 = Initial speed of brake drum, and

t = Time of application of brake.

Given that, the distance travelled by the load, $x = 3.25$ m

But we know that, $x = \frac{1}{2}(v_1 + v_2)\,t$

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

or Time of application of brake, $t = 5.652$ sec

Initial speed of brake drum, $N_1 = \frac{60 \times v_1}{\pi D}$ $\left[\because v_1 = \frac{\pi D N_1}{60}\right]$

$$= \frac{60 \times 1.15}{\pi \times 1.25} = 17.57 \text{ r.p.m.}$$

$\therefore$ Braking torque, $T_B = \frac{60 \times 149.283 \times 10^3}{\pi \times 17.57 \times 5.652}$ = **28.71 kN-m Ans.**

3. Calculation of initial braking power :

We know that, Braking power, $P = \frac{2\pi N_1 T_B}{60}$

$$= \frac{2\pi \times 17.57 \times 28.71 \times 10^3}{60} = \textbf{52.82 kW}$$ **Ans.**

4. Selection of brake drum diameter : Assume a brake drum diameter = 1.5 m

5. Selection of brake drum and block shoe material : A cast iron brake drum and sintered metal block shoe may be chosen, from Table 11.1.

From Table 11.1, safe value of coefficient of friction, $\mu = 0.15$.

6. Selection of induced bearing pressure : From Table 11.2, for continuous service, poor heat condition,

pv = 1.05 (MPa) m/s is selected.

$\therefore$ $pv \le 1.05$ (MPa) m/s

or $p \le \frac{1.05}{1.15} \le 0.913$ MPa

But from Table 11.1, $p_{max} = 2.8$ MPa

Therefore let us use, bearing pressure $p = 2.5$ MPa.

7. Calculation of projected area of the shoe :

We know that induced bearing pressure, $p = \frac{R_N}{A}$

where R_N = Normal force, and

A = Projected area of the shoe.

To find R_N : Assume equal friction force on each shoe.

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

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

or Friction force, $F = 19140$ N

$\therefore$ Normal reaction, $R_N = \frac{F}{\mu} = \frac{19140}{0.15} = 127.6$ kN

Therefore, projected area of the shoe, $A = \frac{R_N}{p} = \frac{127.6 \times 10^3}{2.5 \times 10^6} = 0.051\ m^2$

8. Calculation of breadth and width of the shoe : Assuming breadth (b) of the block shoe is twice its width (w).

$\therefore$ Projected area of the shoe, A = Breadth × Width $= 2\,w \times w = 2\,w^2$

$$0.051 = 2\,w^2$$

or Width, $W = 0.15968$ m or **159.68 mm Ans.**

and Breadth, $b = 2\,w = 2 \times 159.68 =$ **319.37 mm Ans.**

11.7.2. Design Procedure for Band Brakes

1. ***Calculate the braking torque*** required from the data given.
2. If not given, ***select the suitable diameter (D) of the brake drum,*** consulting Table 11.3.

Table 11.3. Dimensions of brake drum (from data book, page no. 7.98)

Power of the motor, kW	Brake drum diameter, mm	Brake drum width, mm
7.36	160	50
11.04	200	65
14.72	250	80
25.76	320	100
44.16	400	125
73.6	500	160
110.4	630	200
184	800	250

3. ***Determine the tight and slack side tensions.***
4. ***Calculate the thickness (t) of band.*** Take thickness of band as 0.005 × Diameter of brake drum.
5. ***Calculate the band width (w)*** based on the induced tensile stress (σ_t). Use the following relations :

$$\text{Induced tensile stress, } \sigma_t = \frac{T_1}{w \times t} \quad \ldots (11.15)$$

where T_1 = Tight side tension in the band,

w = Width of the band, and

t = Thickness of the band = 0.005 D.

Permissible tensile stress for steel band, $[\sigma_t] = 50$ to $80\ \text{N/mm}^2$

Using $\frac{T_1}{w\,t} \leq [\sigma_t]$, width of band is obtained.

6. ***Check for bearing pressure :*** Calculate the maximum bearing pressure between band and drum using the relation

$$p_{max} = \frac{T_1}{w \cdot r} \quad \ldots (11.16)$$

where r = Radius of the drum

Now compare the calculated bearing pressure p with the safe permissible bearing pressure $[p]$ obtained from Table 11.4. If $p < [p]$, then the design is safe and satisfactory.

7. ***Calculate the force*** to be applied at the end of the lever.

Table 11.4. Safe bearing pressure in band brakes (from data book, page no. 7.98)

Type of brake	Materials of the rubbing surfaces			
	Steel band on C.I. or steel drum	Asbestos brake band on steel or C.I. drum	Rolled, press formed and shaped friction material on metal drum	Wood on C.I. drum
Holding	1.5	0.6	0.8	0.6
Lowering	1.0	0.3	0.4	0.4

Example 11.7 ***A simple band brake is operated by a lever of length 500 mm long. The brake drum has a diameter of 500 mm and the brake band embraces 5/8 of the circumference. One end of the band is attached to the fulcrum of the lever while the other is attached to a pin on the lever 100 mm from the fulcrum. If the effort applied to the end of the lever is 2000 N and the coefficient of friction is 0.25, then design the simple band brake.***

Given Data : a = 500 mm = 0.5 m; d = 500 mm or r = 250 mm = 0.25 m ; b = 100 mm = 0.1 m ; F = 2000 N ; μ = 0.25.

To find : Design the simple band brake.

☺ ***Solution :*** Considering anticlockwise rotation of drum, the arrangement of simple band brake is shown in Fig.11.11(b).

$$\text{Angle of contact, } \theta = \frac{5}{8} \times 2\pi = 3.927 \text{ rad}$$

$$\text{Tension ratio is given by, } \frac{T_1}{T_2} = e^{\mu\theta} = e^{0.25 \times 3.927} = 2.669$$

or $$T_1 = 2.669\, T_2 \quad \ldots \text{(i)}$$

Taking moments about the fulcrum O, we get

$$F \cdot a = T_2 \cdot b$$

$$2000 \times 0.5 = T_2 \times 0.1$$

or $$T_2 = 10000 \text{ N}$$

From equation (i), $$T_1 = 2.669\, T_2 = 2.669 \times 10000 = 26690 \text{ N}$$

1. Braking torque :

$$\text{Braking torque, } T_B = (T_1 - T_2)\, r$$

$$= (26690 - 10000) \times 0.25 = \mathbf{4172.5 \text{ N-m}}$$

2. Brake drum diameter : d = 500 mm … (Given)

3. Tight and slack side tensions : Tight side tension, T_1 = **26690 N**

and Slack side tension, T_2 = **10000 N** … (already calculated)

4. Thickness of band : t = 0.005 × Brake drum diameter

= 0.005 × 500 = **2.5 mm**

5. ***Band width (w) :***

$$\text{Induced tensile stress, } \sigma_t = \frac{T_1}{w \times t}$$

$$\sigma_t = \frac{26690}{w \times 2.5} \leq [\sigma_t]$$

$$\frac{26690}{w \times 2.5} \leq 50 \text{ N/mm}^2 \quad [\because [\sigma_t] = 50 \text{ N/mm}^2 \text{ is assumed}]$$

or Width of band, w = 213.52 mm ≈ **215 mm**

6. Check for bearing pressure : Maximum bearing pressure between band and drum is given by

11.8. BAND AND BLOCK BRAKE

Obviously, this arrangement is a combination of both the band and the block brakes, as shown in Fig.11.19. The band is lined with a number of wooden blocks, each of which is in contact with the rim of the brake drum. When the brake is applied, the blocks are pressed against the drum. The advantage of using wooden blocks is that they provide higher coefficient of friction and they can be easily and economically replaced after being worn out.

Let T_n = Tension in the band on tight side,

T_0 = Tension in the band on slack side,

T_1 = Tension in band between the first and second block,

T_2 = Tension in band between second and third block,

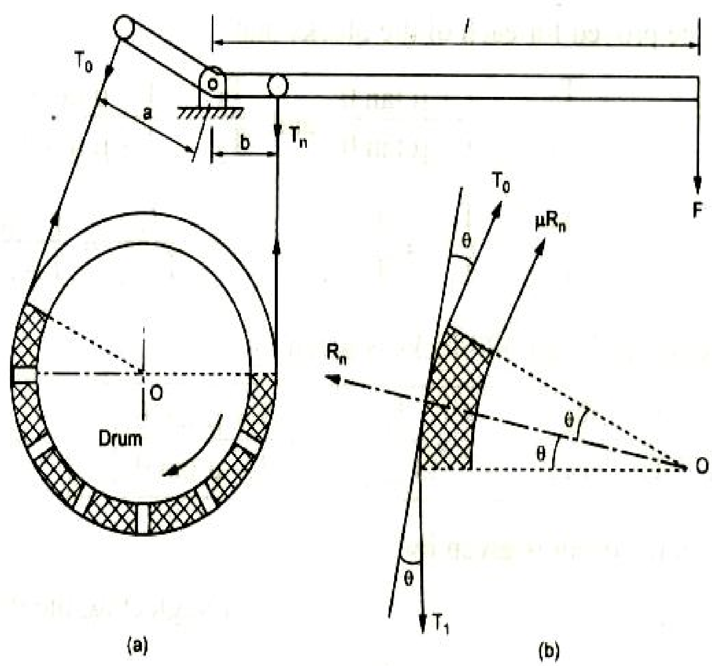
T_3 = Tension in band between third and fourth blocks and so on,

n = Number of wooden blocks,

μ = Coefficient of friction between the block and the drum,

2θ = Angle subtended by each block at the drum centre, and

R_N = Normal reaction on the block.



Consider one of the blocks (say first block) as shown in Fig.11.19(b).

This is in equilibrium under the action of the following forces.

1. Tension in the band on tight side, T_n
2. Tension in the band between first and second block, T_1
3. Normal reaction of the drum R_N of the block, and
4. The frictional force, μR_N.

Resolving the forces radially, we get

$$(T_1 + T_0)\sin\theta = R_N \quad \text{... (i)}$$

Resolving the forces tangentially, we get

$$(T_1 - T_0)\cos\theta = \mu R_N \quad \text{... (ii)}$$

Dividing equation (i) by (ii), we get

$$\mu \tan\theta = \frac{T_1 - T_0}{T_1 + T_0}$$

$$\frac{1 + \mu\tan\theta}{1 - \mu\tan\theta} = \frac{1 + \left(\dfrac{T_1 - T_0}{T_1 + T_0}\right)}{1 - \left(\dfrac{T_1 - T_0}{T_1 + T_0}\right)} = \frac{2\,T_1}{2\,T_0} = \frac{T_1}{T_0}$$

or

$$\frac{T_1}{T_0} = \frac{1 + \mu\tan\theta}{1 - \mu\tan\theta}$$

Similarly, it can be proved for each of the blocks that

$$\frac{T_2}{T_1} = \frac{1 + \mu\tan\theta}{1 - \mu\tan\theta} \quad \text{and} \quad \frac{T_3}{T_2} = \frac{1 + \mu\tan\theta}{1 - \mu\tan\theta}$$

Therefore,

$$\frac{T_1}{T_0} = \frac{T_2}{T_1} = \frac{T_3}{T_2} = \ldots\ldots = \frac{T_n}{T_{n-1}} = \frac{1 + \mu\tan\theta}{1 - \mu\tan\theta}$$

So the ratio of tensions for all '*n*' blocks is given by

$$\boxed{\frac{T_n}{T_0} = \frac{T_1}{T_0} \times \frac{T_2}{T_1} \times \frac{T_3}{T_2} \times \ldots\ldots \times \frac{T_n}{T_{n-1}} = \left[\frac{1 + \mu\tan\theta}{1 - \mu\tan\theta}\right]^n} \quad \text{... (11.17)}$$

Braking torque on the drum is given by

$$T_B = (T_1 - T_2)\,r \quad \text{[Neglecting the thickness of the belt]}$$

$$T_B = (T_1 - T_2)\left(\frac{d + 2t}{2}\right) \quad \text{[Considering the thickness of the belt]}$$

Example 11.15 *In the band and block brake shown in Fig.11.20, the band is lined with 12 blocks each of which subtends an angle of 15° at the centre of the rotating drum. The thickness of the block is 75 mm and the diameter of the drum is 850 mm. If, when the brake is in action, the greatest and least tensions in the brake trap are T_1 and T_2, show that*

$$\frac{T_1}{T_2} = \left[\frac{1+\mu \tan 7.5^\circ}{1-\mu \tan 7.5^\circ}\right]^{12}$$

where μ is the coefficient of friction for the blocks. Find the least force required at 'C' for the blocks to absorb 225 kW at 240 r.p.m. The coefficient of friction between the band and blocks is 0.40.

Given Data :

$n = 12$;

$2\theta = 15^\circ$ or $\theta = 7.5^\circ$;

$t = 75$ mm $= 75 \times 10^{-3}$ m;

$d = 850$ mm $= 0.85$ m;

$P = 225$ kW $= 225 \times 10^3$ W;

$N = 240$ r.p.m.;

$\mu = 0.4$;

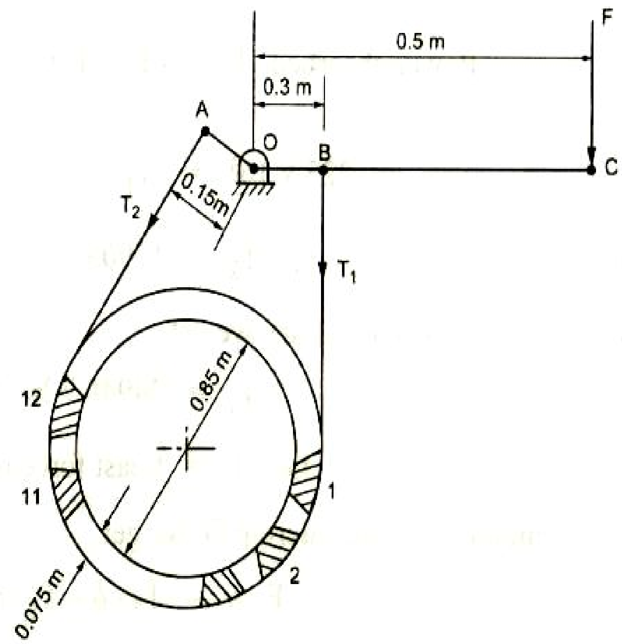


Fig. 11.20.

$$OA = c = 150 \text{ mm} = 0.15 \text{ m};$$

$$OB = b = 30 \text{ mm} = 30 \times 10^{-3} \text{ m};$$

$$OC = a = 500 \text{ mm} = 0.5 \text{ m}$$

☺ ***Solution : (i) Ratio between the greatest and least tensions :***

Since OA > OB, so force at C must act downwards. Also the drum rotates clockwise, s the band attached to A will experience slack side having tension T_2 (least tension) and th band attached to B will be tight side having tension T_1 (greatest tension).

For derivation, refer article 11.8.

So,
$$\frac{T_1}{T_2} = \left[\frac{1+\mu \tan 7.5}{1-\mu \tan 7.5}\right]^{12} \quad \textbf{Ans.}$$

(ii) Least force required at C : Effective diameter of the drum,

$$D = d + 2t = 0.85 + 2 \times 0.075 = 1 \text{ m}$$

We know that the tension ratio,

$$\frac{T_1}{T_2} = \left[\frac{1+\mu \tan \theta}{1-\mu \tan \theta}\right]^n$$

$$\frac{T_1}{T_2} = \left[\frac{1+0.4 \tan 7.5}{1-0.4 \tan 7.5}\right]^{12} = \left[\frac{1.05266}{0.9473}\right]^{12}$$

$$= 3.5449$$

or
$$T_1 = 3.5449\, T_2 \qquad \ldots (i)$$

$$\text{Power absorbed, } P = (T_1 - T_2) \times \frac{\pi D N}{60}$$

$$225 \times 10^3 = (T_1 - T_2) \times \frac{\pi \times 1 \times 240}{60}$$

or
$$T_1 - T_2 = 17905 \qquad \ldots (ii)$$

Solving equations (i) and (ii), we get

$$T_1 = 24940.7 \text{ N} \quad \text{and} \quad T_2 = 7035.70 \text{ N}$$

Let F = Least force required at C

Taking moments about fulcrum O, we get

$$F \cdot a = T_2 \cdot b - T_1 \cdot c$$

$$F \times 0.5 = 7035.7 \times 0.15 - 24940.7 \times 0.03$$

$$\therefore \quad F = \mathbf{614.3 \text{ N}} \quad \textbf{Ans.}$$

PART – A

1. **State about the profile of cam that gives no jerk and mention how jerk is eliminated. (M/J 2012)**

   4-5-6-7 polynomial cam profile gives zero jerks. Profile smoothing techniques can remove the excessive jerks in a cam profile.

2. **Why is it necessary to dissipate the heat generated during clutch operation? (M/J 2012)**

   When clutch engages, most of the work done will be liberated as heat at the interface. Consequently the temperature of the rubbing surface will increase. This increased temperature may destroy the clutch. So heat dissipation is necessary in clutches.

3. **What is self-locking in a brake? (N/D 2011)**

   When a frictional force is sufficient enough to apply the brake with no external force, then the brake is said to be self-locking brake.

4. **What are the factors upon which the torque capacity of a clutch depends? (N/D 2011)**

   Torque capacity of a clutch depends on

   i. Number of pair of contact surfaces,
   ii. Coefficient of friction,
   iii. Axial thrust exerted by the spring, and
   iv. Mean radius of friction surface.

5. **When do we use multiple disk clutches? (A/M 2010)**

   A multiple clutch is used when large amount of torque is to be transmitted. In a multi plate clutch, the number frictional linings and the metal plates are increased which increases the capacity of the clutch to transmit torque.

6. **What is the disadvantage of block brake with one short shoe? What is the remedy? (A/M 2010)**

   If only one block is used for braking, then there will be side thrust on the bearing of wheels shaft. This drawback can be removed by providing two blocks on the two sides of the drum. This also doubles the braking torque.

7. Under what condition of a clutch, uniform rate of wear assumption is more valid?

(M/J 2009)

If the clutch is an old clutch, then uniform rate of wear assumption is more valid.

8. Name four profiles normally used in cams. (M/J 2009)

The four profiles normally used in cams are

i. Uniform velocity,

ii. Simple harmonic motion,

iii. Uniform acceleration and retardation and

iv. Cycloidal motion.

9. How the 'uniform rate of wear' assumption is valid for clutches? (A/M 2008)

In clutches, the value of normal pressure, axial load for the given clutch is limited by the rate of wear that can be tolerated in the brake linings. Moreover, the assumption of uniform wear rate gives a lower calculated clutch capacity than the assumption of uniform pressure. Hence clutches are usually designed on the basis of uniform wear.

10. What are the significances of pressure angle in cam design? (N/D 2007)

The pressure angle is very important in cam design as it represents steepness of the cam profile. If the pressure angle is too large, a reciprocating follower will jam in its bearing.

UNIVERSITY QUESTIONS

1. An automobile single plate clutch consists of two pairs of contacting surfaces. The inner outer radii of friction plate are 120 mm and 250 mm respectively. The coefficient of friction is 0.25 and the total axial force is 15 kN. Calculate the power transmitting capacity of the clutch plate at 500 rpm using
   (i) Uniform wear theory and
   (ii) Uniform pressure theory. **(M/J 2013)**
2. Describe with the help of a neat sketch the design procedure of an internal expanding shoe brake. Also deduce the expression for the braking torque.
   (M/J 2013)
3. Design a cam for operating the exhaust valve of an oil engine. It is required to give equal uniform acceleration and retardation during opening and closing of the valve, each of which corresponding to 60º of cam rotation. The valve should remain in the fully open position for 20º of cam rotation. The lift of the valve is 32 mm and the least radius of the cam is 50 mm, the follower is provided with a roller of 30 mm diameter and its line of stroke passes through the axis of the cam. **(M/J 2012)**
4. 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

between the plates is not to exceed 0.07 N/mm^2. 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.

(M/J 2012)

5. The displacement function of a cam-follower mechanism is given by y(θ) = 100 (1-cosθ) mm, 0≤θ≤2π, where y is the follower displacement and θ is the cam rotation. The cam speed is 1000 rpm. The spring constant is 20 N/mm and the spring has an initial compression of 10 mm, when the roller follower is in its lowest position. The weight of the mass to be moved including the follower is 10 N, length of the follower outside the guide A= 40 mm, length of the guide B = 100 mm, R_b = 50 mm, R_r = 10 mm and the coefficient of friction between the guide and the follower μ = 0.05. Computer normal force and the cam shaft torque when t cam has rotated 60 degrees.
6. 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.m^2. The inner and outer radii of the friction discs are 50 mm and 100 mm respectively. Assuming uniform pressure of 0.1 N/mm^2 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.
7. 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/mm^2. Design the clutch. Take μ = 0.3, n_1=3, n_2=2.
8. 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 weights 45 kN and moves downwards with a velocity of 1.15 m/s. The pitch diameter of the hoist drum is 1.25 m. The hoist must be stopped with in a distance of 3.25 m. The kinetic energy of the drum may be neglected.

Reference books:

1. Machine design (volume –II), Design of Transmission Systems, S.Md.Jalaludeen
2. Machine design – R.S. Khurmi & J.K. Gupta
3. Design of transmission systems – T.J. Prabhu
4. Design of transmission systems – V. Jayakumar