5.2 BRAKES

5.2.1 Introduction

A brake is a device by means of which artificial frictional resistance is applied to a moving machine member, in order to retard or stop the motion of a machine. In the process of performing this function, the brake absorbs either kinetic energy of the moving member or potential energy given up by objects being lowered by hoists, elevators etc. The energy absorbed by brakes is dissipated in the form of heat. This heat is dissipated in the surrounding air (or water which is circulated through the passages in the brake drum) so that excessive heating of the brake lining does not take place. The design or capacity of a brake depends upon the following factors :

1. The unit pressure between the braking surfaces,
2. The coefficient of friction between the braking surfaces,
3. The peripheral velocity of the brake drum,
4.The projected area of the friction surfaces, and
5. The ability of the brake to dissipate heat equivalent to the energy being absorbed.

The major functional difference between a clutch and a brake is that a clutch is used to keep the driving and driven member moving together, whereas brakes are used to stop a moving member or to control its speed.

5.22 Energy Absorbed by a Brake

The energy absorbed by a brake depends upon the type of motion of the moving body. The motion of a body may be either pure translation or pure rotation or a combination of both translation and rotation. The energy corresponding to these motions is kinetic energy. Let us consider these motions as follows :

1. ***When the motion of the body is pure translation*****.** Consider a body of mass (*m*) moving with a velocity $v1$ m / s. Let its velocity is reduced to $v2$ m / s by applying the brake. Therefore, the change in kinetic energy of the translating body or kinetic energy of translation,

$$E_1 = \frac{1}{2} m \left[(v_1)^2 - (v_2)^2 \right]$$

This energy must be absorbed by the brake. If the moving body is stopped after applying the brakes, then $v2 = 0$, and

$$E_1 = \frac{1}{2} m (v_1)^2$$

2. ***When the motion of the body is pure rotation*****.** Consider a body of mass moment of inertia *I* (about a given axis) is rotating about that axis with an angular velocity □1 rad / s. Let its angular velocity is reduced to ω2 rad / s after applying the brake. Therefore, the change in kinetic energy of the rotating body or kinetic energy of rotation,

$$E_2 = \frac{1}{2} I \left[(\omega_1)^2 - (\omega_2)^2 \right]$$

This energy must be absorbed by the brake. If the rotating body is stopped after applying the ωbrakes, then □ω2 = 0, and

$$E_2 = \frac{1}{2} I (\omega_1)^2$$

3. *When the motion of the body is a combination of translation and rotation*. Consider a body having both linear and angular motions, *e.g.* in the locomotive driving wheels and wheels of a moving car. In such cases, the total kinetic energy of the body is equal to the sum of the kinetic energies of translation and rotation.

∴ Total kinetic energy to be absorbed by the brake,

$$E = E_1 + E_2$$

Sometimes, the brake has to absorb the potential energy given up by objects being lowered by hoists, elevators etc. Consider a body of mass m is being lowered from a height h_1 to h_2 by applying the brake. Therefore the change in potential energy,

$$E_3 = m.g\,(h_1 - h_2)$$

If v_1 and v_2 m / s are the velocities of the mass before and after the brake is applied, then the change in potential energy is given by

$$E_3 = m.g\left(\frac{v_1 + v_2}{2}\right) t = m.g.v.t$$

$$v = \text{Mean velocity} = \frac{v_1 + v_2}{2}, \text{ and}$$

Thus, the total energy to be absorbed by the brake,

$$E = E_1 + E_2 + E_3$$

Let F_t = Tangential braking force or frictional force acting tangentially at the contact surface of the brake drum,

d = Diameter of the brake drum,

N_1 = Speed of the brake drum before the brake is applied,

N_2 = Speed of the brake drum after the brake is applied, and

N = Mean speed of the brake drum =

$$\frac{N_1 + N_2}{2}$$

We know that the work done by the braking or frictional force in time t seconds

$$= F_t \times \pi\, d\, N \times t$$

Since the total energy to be absorbed by the brake must be equal to the wordone by the frictional force, therefore

$$E = F_t \times \pi\, d\, N \times t \qquad \text{or} \qquad F_t = \frac{E}{\pi\, d\, N.t}\ .$$

The magnitude of F_t depends upon the final velocity (v_2) and on the braking time (t). Its value is maximum when $v_2 = 0$, *i.e.* when the load comes to rest finally.

We know that the torque which must be absorbed by the brake,

$$T = F_t \times r = F_t \times \frac{d}{2}$$

where r = Radius of the brake drum.

5.23 Heat to be Dissipated during Braking

The energy absorbed by the brake and transformed into heat must be dissipated to the surrounding air in order to avoid excessive temperature rise of the brake lining. The *temperature rise depends upon the mass of the brake drum, the braking time and the heat dissipation capacity of the brake. The highest permissible temperatures recommended for different brake lining materials are given as follows :

1. For leather, fibre and wood facing = 65 – 70°C

2. For asbestos and metal surfaces that are slightly lubricated = 90 – 105°C

3. For automobile brakes with asbestos block lining = 180 – 225°C

Since the energy absorbed (or heat generated) and the rate of wear of the brake lining at a particular speed are dependent on the normal pressure between the braking surfaces, therefore it is an important factor in the design of brakes. The permissible normal pressure between the braking surfaces depends upon the material of the brake lining, the coefficient of friction and the maximum rate at
which the energy is to be absorbed. The energy absorbed or the heat generated is given by

$$E = H_g = \mu.R_N.v = \mu.p.A.v \text{ (in J/s or watts)}$$

where = Coefficient of friction,

R_N = Normal force acting at the contact surfaces, in newtons,
p = Normal pressure between the braking surfaces in N/m^2,
A = Projected area of the contact surfaces in m^2, and
v = Peripheral velocity of the brake drum in m/s.

The heat generated may also be obtained by considering the amount of kinetic or potential energies which is being absorbed. In other words,

$H_g = E_K + E_P$

where E_K = Total kinetic energy absorbed, and
E_P = Total potential energy absorbed.

The heat dissipated (H_d) may be estimated by

$H_d = C (t_1 – t_2) A_r$ ...***(ii)***

where C = Heat dissipation factor or coefficient of heat transfer in $W/m^2/°C$
$t_1 – t_2$ = Temperature difference between the exposed radiating surface and the surrounding air in °C, and
A_r = Area of radiating surface in m^2.

The value of C may be of the order of 29.5 $W/m^2/°C$ for a temperature difference of 40°C and increase up to 44 $W/m^2/°C$ for a temperature difference of 200°C. The expressions for the heat dissipated are quite approximate and should serve only as an indication of the capacity of the brake to dissipate heat. The exact performance of the brake should be determined by test. It has been found that 10 to 25 per cent of the heat generated is immediately dissipated to the surrounding air while the remaining heat is absorbed by the brake drum causing its temperature to rise. The rise in temperature of the brake drum is given by

$$\Delta t = \frac{H_g}{m.c} \qquad ...(iii)$$

where Δt = Temperature rise of the brake drum in °C,

Hg = Heat generated by the brake in joules,
m = Mass of the brake drum in kg, and
c = Specific heat for the material of the brake drum in J/kg °C.

In brakes, it is very difficult to precisely calculate the temperature rise. In preliminary design analysis, the product *p.v* is considered in place of temperature rise. The experience has also shown that if the product *p.v* is high, the rate of wear of brake lining will be high and the brake life will be low. Thus the value of *p.v* should be lower than the upper limit value for the brake lining to have reasonable wear life. The following table shows the recommended values of *p.v* as suggested by various designers for different types of service.

5.24 Materials for Brake Lining

The material used for the brake lining should have the following characteristics :

1. It should have high coefficient of friction with minimum fading. In other words, the coefficient of friction should remain constant over the entire surface with change in temperature.
2. It should have low wear rate.
3. It should have high heat resistance.
4. It should have high heat dissipation capacity.
5. It should have low coefficient of thermal expansion.
6. It should have adequate mechanical strength.
7. It should not be affected by moisture and oil.

5.25 Types of Brakes

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

1. Hydraulic brakes *e.g.* pumps or hydrodynamic brake and fluid agitator,
2. Electric brakes *e.g.* generators and eddy current brakes, and
3. Mechanical brakes.

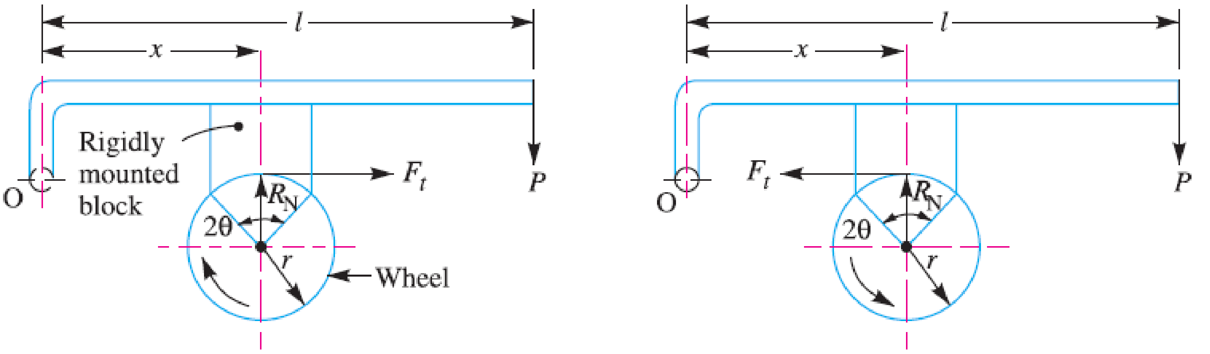
The hydraulic and electric brakes cannot bring the member to rest and are mostly used where large amounts of energy are to be transformed while the brake is retarding the load such as in laboratory dynamometers, high way trucks and electric locomotives. These brakes are also used for retarding or controlling the speed of a vehicle for down-hill travel. The mechanical brakes, according to the direction of acting force, may be divided into the following two groups :

(*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 element, 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. Since we are concerned with only mechanical brakes, therefore, these are discussed in detail, in the following pages.

5.26 Single Block or Shoe Brake

A single block or shoe brake is shown in Fig. 25.1. 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



(*a*) Clockwise rotation of brake wheel. (*b*) Anticlockwise rotation of brake wheel.

the rim of the wheel. This type of a brake is commonly used on railway trains and tram cars. The friction between the block and the wheel causes a tangential braking force to act on the wheel, which retard the rotation of the wheel. The block is pressed against the wheel by a force applied to one end of a lever to which the block is rigidly fixed as shown in Fig. 25.1. The other end of the lever is pivoted on a fixed fulcrum *O*.

Let *P* = Force applied at the end of the lever,
*R*N = Normal force pressing the brake block on the wheel,
r = Radius of the wheel,
2□□= Angle of contact surface of the block,
□□= Coefficient of friction, and
Ft = Tangential braking force or the frictional force acting at the contact surface of
the block and the wheel.

If 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. In such cases, tangential braking force on the wheel,

$$F_t = \mu . R_N$$

and the braking torque, $T_B = F_t . r = \mu\, R_N . r$

Let us now consider the following three cases :

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 as shown in Fig. 25.1 (*a*), then for equilibrium, taking moments about the fulcrum *O*, we have

$$R_N \times x = P \times l \qquad \text{or} \qquad R_N = \frac{P \times l}{x}$$

$\therefore$ Braking torque, $$T_B = \mu.R_N.r = \mu \times \frac{P.l}{x} \times r = \frac{\mu.P.l.r}{x}$$

It may be noted that when the brake wheel rotates anticlockwise as shown in Fig. 25.1 (*b*), then the braking torque is same, *i.e.*

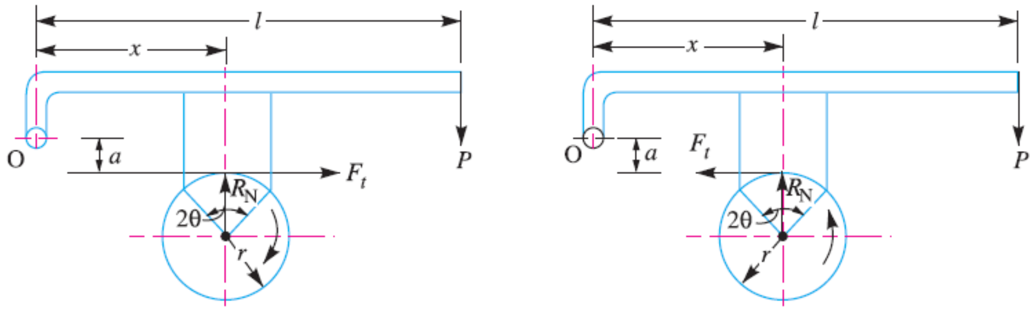
$$T_B = \mu.R_N.r = \frac{\mu.P.l.r}{x}$$

Case 2. When the line of action of the tangential braking force (F_t) passes through a distance '*a*' below the fulcrum *O*, and the brake wheel rotates clockwise as shown in Fig. 25.2 (*a*), then for equilibrium, taking moments about the fulcrum *O*,

$$R_N \times x + F_t \times a = P.l$$

or $$R_N \times x + \mu\, R_N \times a = P.l \qquad \text{or} \qquad R_N = \frac{P.l}{x + \mu.a}$$

and braking torque, $$T_B = \mu\; R_N.r = \frac{\mu.P.l.r}{x + \mu.a}$$



(*a*) Clockwise rotation of brake wheel. (*b*) Anticlockwise rotation of brake wheel.

Fig. 25.2. Single block brake. Line of action of F_t passes below the fulcrum.

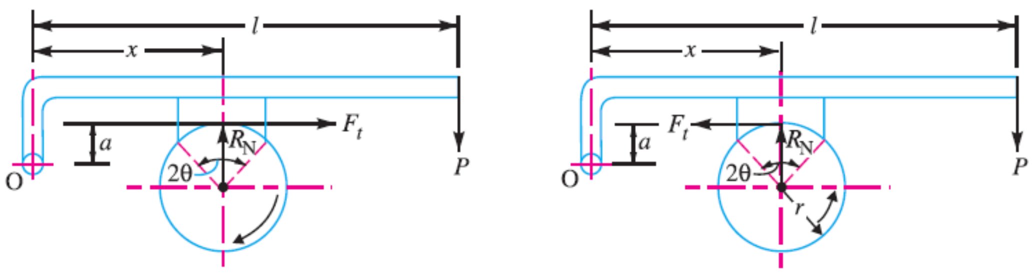
When the brake wheel rotates anticlockwise, as shown in Fig. 25.2 (*b*), then for equilibrium,

$$R_N.x = P.l + F_t.a = P.l + \mu.R_N.a \qquad \text{...}(i)$$

or $$R_N\,(x - \mu.a) = P.l \qquad \text{or} \qquad R_N = \frac{P.l}{x - \mu.a}$$

and braking torque, $$T_B = \mu.R_N.r = \frac{\mu.P.l.r}{x - \mu.a}$$

Case 3. When the line of action of the tangential braking force passes through a distance '*a*' above the fulcrum, and the brake wheel rotates clockwise as shown in Fig. 25.3 (*a*), then for equilibrium, taking moments about the fulcrum *O*, we have



(*a*) Clockwise rotation of brake wheel. (*b*) Anticlockwise rotation of brake wheel.

Fig. 25.3. Single block brake. Line of action of F_t passes above the fulcrum.

$$R_N.x = P.l + F_t.a = P.l + \mu.R_N.a \quad ...(ii)$$

or $$R_N(x - \mu.a) = P.l \quad \text{or} \quad R_N = \frac{P.l}{x - \mu.a}$$

and braking torque, $$T_B = \mu.R_N.r = \frac{\mu.P.l.r}{x - \mu.a}$$

When the brake wheel rotates anticlockwise as shown in Fig. 25.3 (*b*), then for equilibrium, taking moments about the fulcrum *O*, we have

$$R_N \times x + F_t \times a = P.l$$

or $$R_N \times x + \mu.R_N \times a = P.l \quad \text{or} \quad R_N = \frac{P.l}{x + \mu.a}$$

and braking torque, $$T_B = \mu.R_N.r = \frac{\mu.P.l.r}{x + \mu.a}$$

Notes: 1. From above we see that when the brake wheel rotates anticlockwise in case 2 [Fig. 25.2 (*b*)] and when it rotates clockwise in case 3 [Fig. 25.3 (*a*)], the equations (*i*) and (*ii*) are same, *i.e.*

5.27 Pivoted Block or Shoe Brake

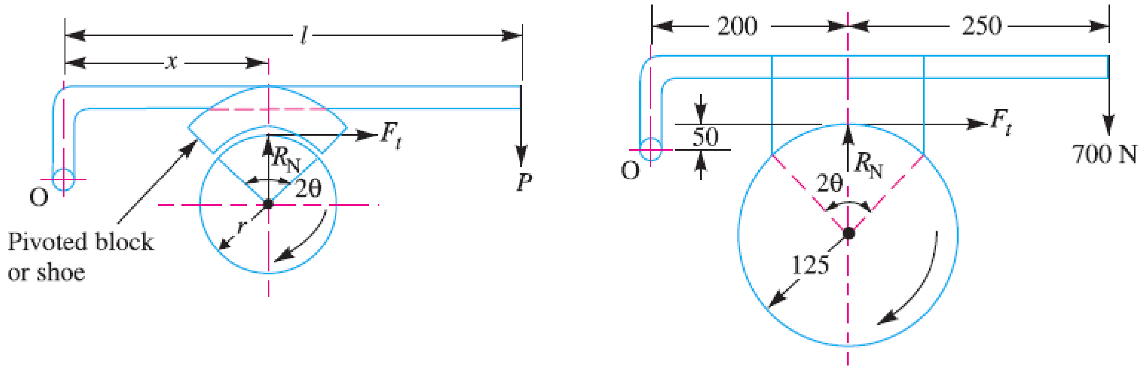
We have discussed in the previous article that when the angle of contact is less than 60°, then it may be assumed that the normal pressure between the block and the wheel is uniform. But when the angle of contact is greater than 60°, then the unit pressure normal to the surface of contact is less at the ends than at the centre. In such cases, the block or shoe is pivoted to the lever as shown in Fig. 25.4, instead of being rigidly attached to the lever. This gives uniform wear of the brake lining in the direction of the applied force. The braking torque for a pivoted block or shoe brake (*i.e.* when 2□□> 60°) is given by

$$T_B = F_t \times r = \mu'.R_N.r$$

where $$\mu' = \text{Equivalent coefficient of friction} = \frac{4\mu \sin\theta}{2\theta + \sin 2\theta}, \text{ and}$$

$$\mu = \text{Actual coefficient of friction.}$$

These brakes have more life and may provide a higher braking torque.



5.28 Double Block or Shoe Brake

When a single block brake is applied to a rolling wheel, and additional load is thrown on the shaft bearings due to the normal force (*RN*). This produces bending of the shaft. In order to overcome this drawback, a double block or shoe brake as shown in Fig. 25.10, is used. It consists of two brake blocks applied at the opposite ends of a diameter of the wheel which eliminate or reduces the unbalanced force on the shaft. The brake is set

by a spring which pulls the upper ends of the brake arms together. When a force P is applied to the bell crank lever, the spring is compressed and the brake is released. This type of brake is often used on electric cranes and the force P is produced by an electromagnet or solenoid. When the current is switched off, there is no force on the bell crank lever and the brake is engaged automatically due to the spring force and thus there will be no downward movement of the load. In a double block brake, the braking action is doubled
by the use of two blocks and the two blocks may be operated practically by the same force which will operate one. In case of double block or shoe brake, the braking torque is given by

$T_B = (F_{t1} + F_{t2})\, r$

where F_{t1} and F_{t2} are the braking forces on the two blocks.

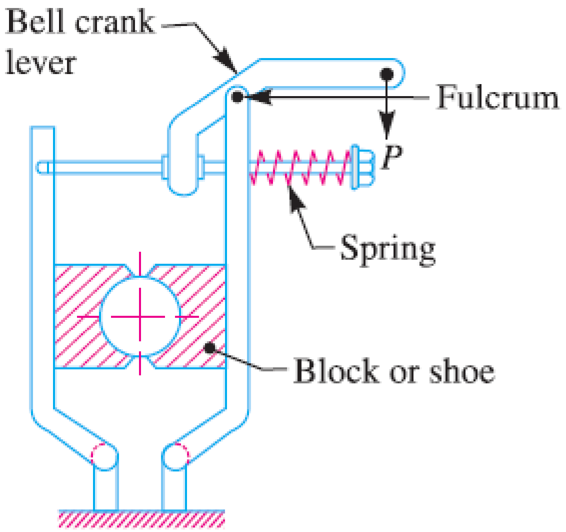
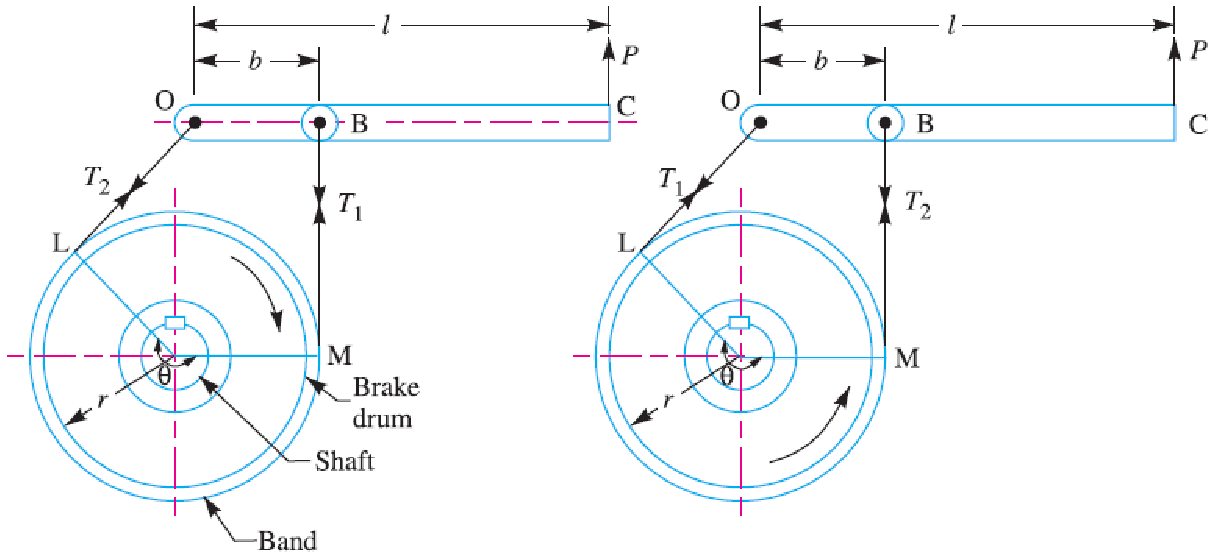


Fig. 25.10. Double block or shoe brake.

5.29 Simple Band Brake

A band brake consists of a flexible band of leather, one or more ropes, or a steel lined with friction material, which embraces a part of the circumference of the drum. A band brake, as shown in Fig. 25.14, is called a ***simple band brake*** in which one end of the band is attached to a fixed pin or fulcrum of the lever while the other end is attached to the lever at a distance b from the fulcrum.

When a force P is applied to the lever at C, the lever turns about the fulcrum pin O and tightens the band on the drum and hence the brakes are applied. The friction between the band and the drum provides the braking force. The force P on the lever at C may be determined as discussed below :



(*a*) Clockwise rotation of drum. (*b*) Anticlockwise rotation of drum.

Let $T1$ = Tension in the tight side of the band,
$T2$ = Tension in the slack side of the band,
□□= Angle of lap (or embrace) of the band on the drum,
□□= Coefficient of friction between the band and the drum,
r = Radius of the drum,
t = Thickness of the band, and
re = Effective radius of the drum = $r + t / 2$.
We know that limiting ratio of the tensions is given by the relation,

$$\frac{T_1}{T_2} = e^{\mu.\theta} \quad \text{or} \quad 2.3 \log\left(\frac{T_1}{T_2}\right) = \mu.\theta$$

and braking force on the drum

$= T1 - T2$

∴□Braking torque on the drum,
$TB = (T1 - T2)\, r$ ...(Neglecting thickness of band)
$= (T1 - T2)\, re$ ...(Considering thickness of band)

Now considering the equilibrium of the lever *OBC*. It may be noted that when the drum rotates in the clockwise direction as shown in Fig. 25.14 (*a*), the end of the band attached to the fulcrum *O* will be slack with tension *T*2 and end of the band attached to *B* will be tight with tension *T*1. On the other hand, when the drum rotates in the anticlockwise direction as shown in Fig. 25.14 (*b*), the tensions in the band will reverse, *i.e*. the end of the band attached to the fulcrum *O* will be tight with tension *T*1 and the end of the band attached to *B* will be slack with tension *T*2. Now taking moments about the fulcrum *O*, we have

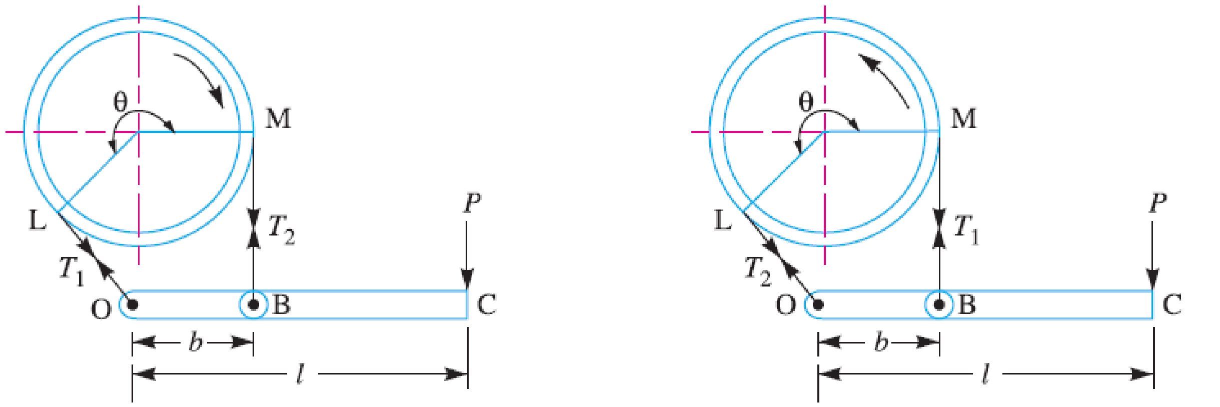
$P.l = T1.b$ ...(for clockwise rotation of the drum) and $P.l = T2.b$ ...(for anticlockwise rotation of the drum)

where l = Length of the lever from the fulcrum (OC), and

b = Perpendicular distance from O to the line of action of $T1$ or $T2$.

Notes: 1. When the brake band is attached to the lever, as shown in Fig. 25.14 (***a***) and (*b*), then the force (P) must act in the upward direction in order to tighten the band on the drum.

2. Sometimes the brake band is attached to the lever as shown in Fig. 25.15 (*a*) and (*b*), then the force (P) must act in the downward direction in order to tighten the band. In this case, for clockwise rotation of the drum, the end of the band attached to the fulcrum O will be tight with tension $T1$ and band of the band attached to B will be slack with tension $T2$. The tensions $T1$ and $T2$ will reverse for anticlockwise rotation of the drum.



(*a*) Clockwise rotation of drum. (*b*) Anticlockwise rotation of drum.

3. If the permissible tensile stress (□*t*) for the material of the band is known, then maximum tension in the band is given by $T_1 = \sigma_t \times w \times t$ where w = Width of the band, and

t = Thickness of the band.

4. The width of band (w) should not exceed 150 mm for drum diameter (d) greater than 1 metre and 100 mm for drum diameter less than 1 metre. The band thickness (t) may also be obtained by using the empirical relation *i.e.* $t = 0.005\ d$ For brakes of hand operated winches, the steel bands of the following sizes are usually used :

Width of band (w) in mm	25 – 40	40 – 60	80	100	140 – 200
Thickness of band (t) in mm	3	3 – 4	4 – 6	4 – 7	6 – 10