ROUP OF COLLEGES

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Department:Mechanical

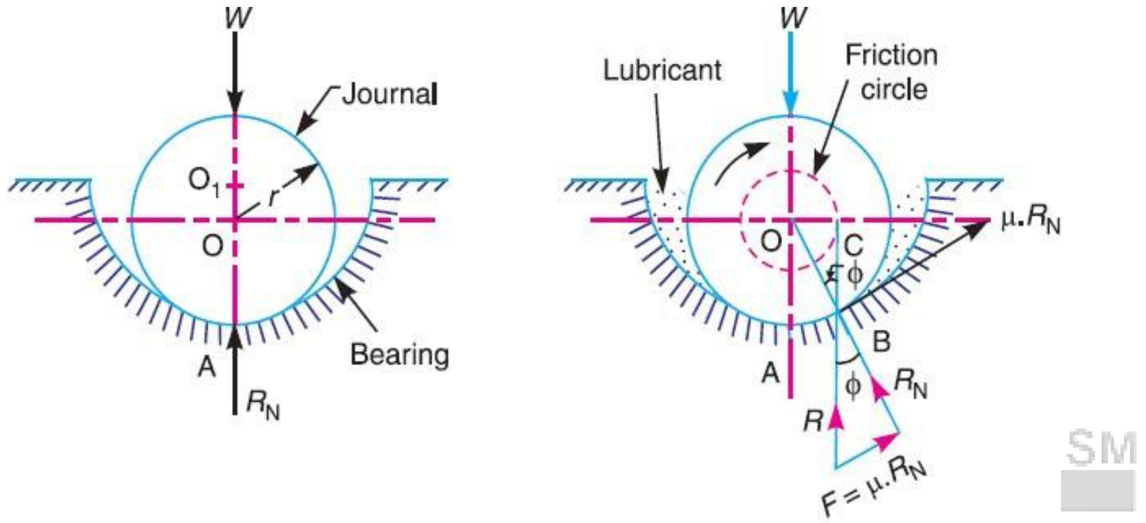
Subject:Theory of Machine

Unit:4

Topic: Pivots and Collars

Friction in Journal Bearing-Friction Circle

A journal bearing forms a turning pair as shown in Fig(*a*). The fixed outer element of a turning pair is called a ***bearing*** and that portion of the inner element (*i.e.* shaft) which fits in the bearing is called a ***journal.*** The journal is slightly less in diameter than the bearing, in order to permit the free movement of the journal in a bearing



(*a*) (*b*)

When the bearing is not lubricated (or the journal is stationary), then there is a line contact between the two elements as shown in Fig (a). The load W on the journal and normal reaction R_N (equal to W) of the bearing acts through the centre. The reaction RN acts vertically upwards at point A. This point A is known as **seat** or **point of pressure.**

Now consider a shaft rotating inside a bearing in clockwise direction as shown in Fig(*b*). The lubricant between the journal and bearing forms a thin layer which gives rise to a greasy friction. Therefore, the reaction *R* does not act vertically upward, but acts at another point of pressure *B*. This is due to the fact that when shaft rotates, a frictional force $F = \mu R_N$ acts at the circumference of the shaft which has a tendency to rotate the shaft in opposite direction of motion and this shifts the point *A* to point *B*.

In order that the rotation may be maintained, there must be a couple rotating the shaft.

Let ø= Angle between *R* (resultant of *F* and *R*N) and R_N,

μ = Coefficient of friction between the journal and bearing,

T = Frictional torque in N-m, and

r = Radius of the shaft in metres.

For uniform motion, the resultant force acting on the shaft must be zero and the resultant turning moment on the shaft must be zero. In other words,

R = *W*, and *T* = *W* × *OC* = *W* × *OB* sinϕ_= *W*.*r*sin_ϕ

Since ø is very small, therefore substituting sinϕ= tan ϕ_

T = *W*.*r* tan ø= μ .*W*.*r* ....(μ = tanø)

If the shaft rotates with angular velocity _ rad/s, then power wasted in friction,

P = *T*_= *T* × 2_*N*/60 watts

Where *N* = Speed of the shaft in r.p.m.

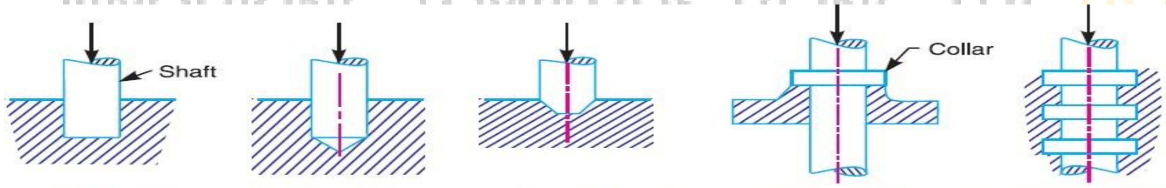
Friction of Pivot and Collar Bearing

The rotating shafts are frequently subjected to axial thrust. The bearing surfaces such as pivot and collar bearings are used to take this axial thrust of the rotating shaft. The propeller shafts of ships, the shafts of steam turbines, and vertical machine shafts are examples of shafts which carry an axial thrust.

The bearing surfaces placed at the end of a shaft to take the axial thrust are known as pivots. The pivot may have a flat surface or conical surface as shown in Fig. 10.16 (a) and (b) respectively. When the cone is truncated, it is then known as truncated or trapezoidal pivot as shown in Fig (c).

The collar may have flat bearing surface or conical bearing surface, but the flat surface is most commonly used. There may be a single collar, as shown in Fig (*d*) or several collars along the length of a shaft, as shown in Fig(*e*) in order to reduce the intensity of pressure.

In



(*a*) Flat pivot. (*b*) Conical pivot. (*c*) Truncated pivot. (*d*) Single flat collar. (*e*) Multiple flat collar.

In modern practice, ball and roller thrust bearings are used when power is being transmitted and when thrusts are large as in case of propeller shafts of ships.

A little consideration will show that in a new bearing, the contact between the shaft and bearing may be good over the whole surface. In other words, we can say that the pressure over the rubbing surfaces is uniformly distributed. But when the bearing becomes old, all parts of the rubbing surface will not move with the same velocity, because the velocity of rubbing surface increases with the distance from the axis of the bearing. This

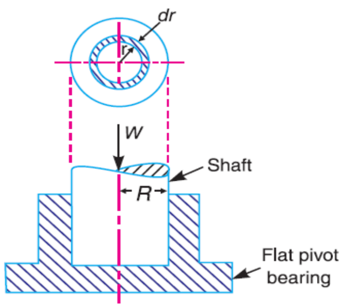
means that wear may be different at different radii and this causes to alter the distribution of pressure. Hence, in the study of friction of bearings, it is assumed hat

1. The pressure is uniformly distributed throughout the bearing surface, and

2. The wear is uniform throughout the bearing surface.

Flat Pivot Bearing

When a vertical shaft rotates in a flat pivot bearing (known as **foot step bearing**), as shown in Fig., the sliding friction will be along the surface of contact between the shaft and the bearing.



Let *W* =Load transmitted over the bearing surface,

R =Radius of bearing surface,

p =Intensity of pressure per unit area of bearing Surface between rubbing surfaces, and

μ =Coefficient of friction.

We will consider the following two cases:

1. When there is a uniform pressure

2. When there is a uniform wear

1. Considering uniform pressure

When the pressure is uniformly distributed over the bearing area, then

$$p = \frac{W}{\pi R^2}$$

Consider a ring of radius *r* and thickness *dr* of the bearing area.

Area of bearing surface, *A* = 2_*r.dr*

Load transmitted to the ring,

_*W* = *p* × *A* = *p* × 2 _ *r.dr* ....**(*i*)**

Frictional resistance to sliding on the ring acting tangentially at radius *r*,

Fr= μ ._*W* = μ *p* × 2_ *r.dr* = 2_ μ .*p.r.dr*

Frictional torque on the ring,

Tr = *Fr* × *r*= 2_ μ *p r.dr* × *r* = 2 _ μ *p r*2 *dr* ....**(*ii*)**

Integrating this equation within the limits from 0 to *R* for the total frictional torque on the pivot bearing.

$$\therefore \text{ Total frictional torque, } T = \int_0^R 2\pi\mu\, p r^2\, dr = 2\pi\mu\, p \int_0^R r^2\, dr$$

$$= 2\pi\mu\, p \left[\frac{r^3}{3}\right]_0^R = 2\pi\mu\, p \times \frac{R^3}{3} = \frac{2}{3} \times \pi\mu . p . R^3$$

$$= \frac{2}{3} \times \pi\mu \times \frac{W}{\pi R^2} \times R^3 = \frac{2}{3} \times \mu . W . R$$

When the shaft rotates at ω rad/s, then power lost in friction,

$$P = T.\omega = T \times 2\pi N/60 \qquad \ldots($$

$$N = \text{Speed of shaft in r.p.m.}$$

2. Considering uniform wear

We have already discussed that the rate of wear depends upon the intensity of pressure (p) and the velocity of rubbing surfaces (*v*). It is assumed that the rate of wear is proportional to the product of intensity of pressure and the velocity of rubbing surfaces (*i.e. p.v.*). Since the velocity of rubbing surfaces increases with the distance (*i.e.* radius *r*) from the axis of the bearing, therefore for uniform Wear

$$p.r = C \text{ (a constant)} \quad \text{or} \quad p = C/r$$

ınd the load transmitted to the ring,

$$\delta W = p \times 2\pi r.dr \qquad \ldots\text{[From equation } (i)\text{]}$$

$$= \frac{C}{r} \times 2\pi r.dr = 2\pi C.dr$$

∴ Total load transmitted to the bearing

$$W = \int_0^R 2\pi\, C.dr = 2\pi\, C\,[r]_0^R = 2\pi C.R \quad \text{or} \quad C = \frac{W}{2\pi R}$$

We know that frictional torque acting on the ring,

$$T_r = 2\pi\mu\, p r^2\, dr = 2\pi\mu \times \frac{C}{r} \times r^2\, dr \qquad \ldots\left(\because\ p = \frac{C}{r}\right)$$

$$= 2\pi\, \mu.C.r\ dr \qquad \ldots(iii)$$

∴ Total frictional torque on the bearing,

$$T = \int_0^R 2\pi\, \mu.C.r.dr = 2\pi\mu.C\left[\frac{r^2}{2}\right]_0^R$$

$$= 2\pi\mu.C \times \frac{R^2}{2} = \pi\mu.C.R^2$$

$$= \pi\mu \times \frac{W}{2\pi R} \times R^2 = \frac{1}{2} \times \mu.W.R \qquad \ldots\left(\because\ C = \frac{W}{2\pi R}\right)$$

PROBLEMS

Example 1. A conical pivot supports a load of 20 kN, the cone angle is 120º and the intensity of normal ressure is not to exceed 0.3 N/mm2. The external diameter is twice the internal diameter. Find the outer and inner radii of the bearing surface. If the shaft rotates at 200 r.p.m. and the coefficient of friction is 0.1, find the power absorbed in friction. Assume uniform pressure.

Solution. Given : *W* = 20 kN = 20 × 103 N ; 2 _= 120º or _= 60º ; *pn*= 0.3 N/mm2 ;*N* =

200 r.p.m. or _= 2_× 200/60 = 20.95 rad/s ; μ = 0.1

Outer and inner radii of the bearing surface.

Let *r*1 and *r*2 = Outer and inner radii of the bearing surface, in mm.

Since the external diameter is twice the internal diameter, therefore

*r*1 = 2 *r*2

We know that intensity of normal pressure (p_n),

$$0.3 = \frac{W}{\pi[(r_1)^2 - (r_2)^2]} = \frac{20 \times 10^3}{\pi[(2r_2)^2 - (r_2)^2]} = \frac{2.12 \times 10^3}{(r_2)^2}$$

$$(r_2)^2 = 2.12 \times 10^3 / 0.3 = 7.07 \times 10^3 \quad \text{or} \quad r_2 = 84 \text{ mm} \quad \textbf{Ans.}$$

$$r_1 = 2\, r_2 = 2 \times 84 = 168 \text{ mm} \quad \textbf{Ans.}$$

Power absorbed in friction

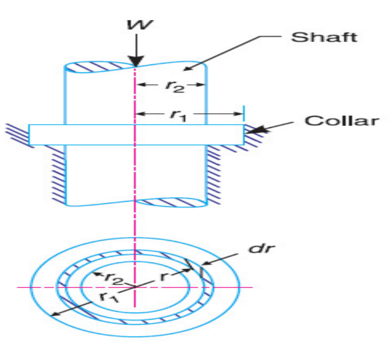
We know that total frictional torque (assuming uniform pressure),

Power absorbed in friction

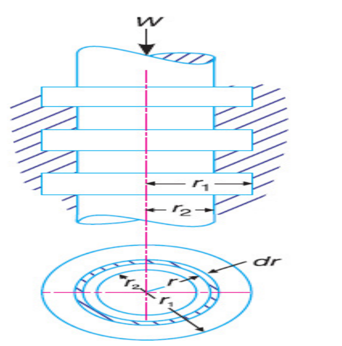
P = *T*.ω_ = 301.76 × 20.95 = 6322 W = 6.322 kW

Flat Collar Bearing

Consider We have already discussed that collar bearings are used to take the axial thrust of the rotating hafts. There may be a single collar or multiple collar bearings as shown in Fig.(*a*) and (*b*)respectively. The collar bearings are also known as ***thrust bearings.*** The friction in the collar bearings may be found as discussed below :



(*a*) Single collar bearing



(*b*) Multiple collar bearing.

Consider a single flat collar bearing supporting a shaft as shown in Fig(*a*).

Let *r*1 = External radius of the collar,

*r*2 = Internal radius of the collar.

Area of the bearing surface,

$$A = \pi\,[(r_1)^2 - (r_2)^2]$$

1. Considering uniform pressure

When the pressure is uniformly distributed over the bearing surface, then the intensity of pressure

$$p = \frac{W}{A} = \frac{W}{\pi[r_1)^2 - (r_2)^2]} \qquad \text{...}(i)$$

the frictional torque on the ring of radius *r* and thickness *dr*,

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

Integrating this equation within the limits from *r*2 to *r*1 for the total frictional torque on the collar.

Total frictional torque

$$T = \int_{r_2}^{r_1} 2\pi\mu.p.r^2.dr = 2\pi\mu.p\left[\frac{r_3}{3}\right]_{r_2}^{r_1} = 2\pi\mu.p\left[\frac{(r_1)^3 - (r_2)^3}{3}\right]$$

Substituting the value of *p* from equation **(*i*),**

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

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

2. Considering uniform wear

the load transmitted on the ring, considering uniform wear is,

$$\delta W = p_r.2\pi r.dr = \frac{C}{r} \times 2\pi r.dr = 2\pi C.dr$$

Total load transmitted to the collar,

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

$$C = \frac{W}{2\pi(r_1 - r_2)} \qquad \text{...}(ii)$$

We also know that frictional torque on the ring,

$T_r = \mu\, \delta W\, r = \mu\, 2\pi C r dr$

Total frictional Torque on the bearing from r1 to r2

Substituting the value of *C* from equation (*ii*),

$$T = \pi\mu \times \frac{W}{2\pi(r_1 - r_2)}[(r_1)^2 - (r_2)^2] = \frac{1}{2} \times \mu.W(r_1 + r_2)$$

PROBLEMS

Example 1. A thrust shaft of a ship has 6collars of 600 mm external diameter and 300 mm internal diameter. The total thrust from the propeller is 100 kN. If the coefficient of friction is 0.12 and speed of the engine90 r.p.m., find the power absorbed in friction at the thrust block, assuming
l. uniform pressure
2. uniform wear.
Solution. Given : *n* = 6 ; *d*1 = 600 mm or *r*1 = 300 mm ; *d*2 = 300 mm or *r*2 = 150 mm ; *W* = 100 kN= 100 × 103 N ; μ = 0.12 ; *N* = 90 r.p.m. or _= 2 _× 90/60 = 9.426 rad/s

1. Power absorbed in friction, assuming uniform pressure

We know that total frictional torque transmitted,

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

$$= \frac{2}{3} \times 0.12 \times 100 \times 10^3\left[\frac{(300)^3 - (150)^3}{(300)^2 - (150)^2}\right] = 2800 \times 10^3 \text{ N-mm}$$

$$= 2800 \text{ N-m}$$

Power absorbed in friction,

P = *T*_= 2800 × 9.426 = 26 400 W = 26.4 kW

2. Power absorbed in friction assuming uniform wear

We know that total frictional torque transmitted,

$$T = \frac{1}{2} \times \mu.W(r_1 + r_2) = \frac{1}{2} \times 0.12 \times 100 \times 10^3\ (300 + 150) \text{ N-mm}$$

$$= 2700 \times 10^3 \text{ N-mm} = 2700 \text{ N-m}$$

Power absorbed in friction,

$P = T.\ \omega = 2700 \times 9.426 = 25\ 450\ \text{W} = 25.45\ \text{kW}$