then

$$R+S=\frac{W}{g}\omega^2 r$$

where W is the weight of each shoe and r is the radial distance of the centre of gravity of each shoe from the axis.

At the commencement of engagement $R=0$ and the angular velocity of rotation is

$$\omega=0.75\times 750\frac{2\pi}{60}=\frac{75\pi}{4}, \text{rad s}^{-1}$$

so that

$$S=W\left(\frac{75\pi}{4}\right)^2\times 0.126=436.7W.$$

At a speed of 750 r.p.m.,

$$R+S=W(78.5)^2\times 0.126=776.4W$$

so

$$R=776.4W-S=776.4W-436.7W=339.7W.$$

The couple due to each shoe

$$=fRa, \text{ very nearly,}$$
$$=0.25\times 339.7W\times 0.150=12.74W, \text{Nm}$$

$$\text{power transmitted}=\frac{4\times 12.74W\times 2\pi\times 750}{60}=30\,000\text{ Watts}$$

and finally,

$$W=\frac{30\,000\times 60}{4\times 12.74\times 2\pi\times 750}=7.5\text{ kg}.$$

4.9. Boundary lubricated sliding bearings

Under boundary lubrication conditions the surfaces are considered to be technically dry or only slightly lubricated, so that the resistance to relative motion is due to the interaction between the highest asperities covered by the boundary film. Then, frictional force $F=fR$, where f is the kinetic coefficient of friction. The magnitude of the friction couple retarding the motion of the journal is determined by the assumed geometric conditions of the bearing surface.

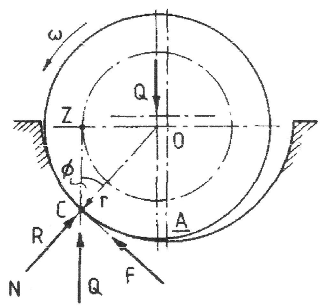

Figure 4.25

Case A. Journal rotating in a loosely fitting bush

Figure 4.25 represents a cross-section of a journal supporting a load Q at the centre of the section. When the journal is at rest the resultant from pressure will be represented by the point A on the line of action of the load Q, i.e. contact is then along a line through A perpendicular to the plane of the section. When rotating commences, we may regard the journal as mounting the bush until the line of contact reaches a position C, where slipping occurs at a rate which exactly neutralizes the rolling action. The resultant reaction at C must be parallel to the line of action of Q at 0, and the two forces will constitute a couple of moment $Q \times OZ$ retarding the motion of the journal. Further, Q at C must act at an angle ϕ to the common normal CN and, if r is the radius of the journal

$$OZ = r \sin \phi$$

hence,

$$\text{Friction couple} = Qr \sin \phi \tag{4.59}$$

The circle drawn with radius $OZ = r \sin \phi$ is known as the friction circle for the bearing.

Case B. Journal rotating in a closely fitting bush

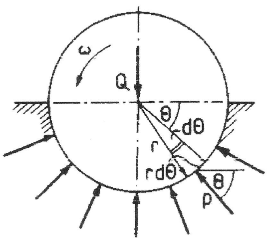

Figure 4.26

A closely fitted bearing may be defined as one having a uniform distribution of radial pressure over the complete area of the lower part of the bush (Fig. 4.26). Let

p = the radial pressure per unit area of the bearing surface,
Q = the vertical load on the journal,
l = the length of the bearing surface.

Then,

$$Q = \int_0^\pi plr \, d\Theta \sin \Theta = plr \int_0^\pi \sin \Theta \, d\Theta$$

$$= 2plr \tag{4.60}$$

$$\text{friction couple} = \int_0^\pi fplr \, d\Theta r = fplr^2 \int_0^\pi d\Theta = \pi fplr^2 \tag{4.61}$$

and substituting for Q,

$$\text{friction couple} = \tfrac{1}{2}\pi frQ. \tag{4.62}$$

For the purpose of comparison take case A as the standard, and assume boundary conditions of lubrication $f = 0.1$, so that

$$f = \tan \phi = \sin \phi \qquad \text{very nearly}$$

and

$$Qr \sin \phi = frQ \qquad \text{very nearly.}$$

In general, we may then express the friction couple in the form $f'rQ$, where f' is defined as the virtual coefficient of friction, and for the closely fitting bush

$$\text{friction couple} = \tfrac{1}{2}\pi frQ = f'rQ$$

and

$$\text{virtual coefficient of friction, } f' = \tfrac{1}{2}\pi f = 1.57f.$$

Case C. Journal rotating in a bush under ideal conditions of wear

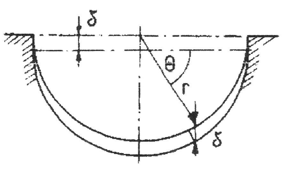

Figure 4.27

Let us be assumed that the journal remains circular and unworn and that, after the running-in process, any further wear in the bush reduces the metal in such a way that vertical descent is uniform at all angles. The volume of metal worn away at different angles is proportional to the energy expanded in overcoming friction, so that the pressure will vary over the bearing surface. For vertical displacement, δ, the thickness worn away at angle Θ is $\delta \sin \Theta$, where δ is constant (Fig. 4.27).

Hence, since frictional resistance per unit area is proportional to the intensity of normal pressure p, and the relative velocity of sliding over the circle of radius r is constant, it follows that:

$$p = k \sin \Theta \tag{4.63}$$

where k is a constant,

$$\text{vertical load, } Q = lr \int_0^{\pi} p \sin \Theta \, d\Theta$$

$$= klr \int_0^{\pi} \sin^2 \Theta \, d\Theta = \tfrac{1}{2}\pi klr \tag{4.64}$$

$$\text{friction couple} = flr^2 \int_0^{\pi} p \, d\Theta$$

$$= fklr^2 \int_0^{\pi} \sin \Theta \, d\Theta = 2fklr^2. \tag{4.65}$$

Hence

$$\text{virtual coefficient} = \frac{\text{friction couple}}{Qr}$$

i.e.

$$f' = \frac{2fklr^2}{\tfrac{1}{2}\pi klr^2} = 1.275f.$$

Summarizing the results of the above three cases

$$\begin{aligned} \text{virtual coefficient, } f' &= f \text{ in a loose bearing,} \\ &= 1.57f \text{ in a new well-fitted bearing,} \\ &= 1.275f \text{ in a well-worn bearing.} \end{aligned}$$

4.9.1. Axially loaded bearings

Figure 4.28 shows a thrust block or pivot designed on the principle of uniform displacement outlined in case C. In other words, we have the case of a journal rotating in a bush under ideal conditions of wear.