UNIT

LUBRICATION AND BEARINGS

7.1 LUBRICATION

Friction is the resistance to relative motion between the two surfaces in contact. The function of lubricant is to reduce the friction. Any substance placed between any two rubbing surfaces, which reduces friction is called lubricant.

7.2 TYPES OF LUBRICANT

Lubricant may be classified into three groups.

(i) Liquid (ii) Semi liquid and (iii) Solid.

Liquid lubricants are usually preferred where they have to be be retained. Example for liquid lubricants are mineral oil, animal and vegetable oil, synthetic oil etc.

Mineral oil, is the most commonly used liquid lubricant because of its cheapness.

Greases are semiliquid lubricants of higher viscosity than oils. Grease is used where slow and heavy pressure exists.

Solid lubricants are useful in reducing friction where oil films cannot be maintained because of pressure or temperature.

Most commonly used solid lubricant is graphite, other solid lubricants are wax, mica, french chalk, soap stone, talc, etc.

7.3 PROPERTIES OF LUBRICANT

The important properties of lubricant are

i) **Viscosity :** Viscosity is the measure of degree of fluidity of a liquid. For more heat and pressure the greater viscosity is required. Viscosity is important since the load carrying capacity is proportional to viscosity. It is also defined as the property of a fluid by virtue of which it offers resistance to shear.

The viscosity of a fluid, known as absolute viscosity or dynamic viscosity is defined as the ratio of intensity of shear stress to the rate of deformation. If v is the uniform velocity, h is the thickness of a layer of a fluid and τ is the intensity of shear stress, then Absolute viscosity $\eta = \frac{\tau}{\frac{v}{h}} = \frac{\tau h}{v}$

The unit of absolute viscosity in SI system is N-sec/m^2 or Pas

Kinematic viscosity is defined as the ratio of absolute viscosity to mass density of fluid.

$\therefore$ kinematic viscosity $\upsilon = \frac{\eta}{\rho}$. The unit of kinematic viscosity in SI system is m^2/sec. The relation between absolute viscosity in centripoise and specific viscosity in seconds obtained from say bolt viscometer is

$$\eta_1 = \gamma_t \left(0.22 t_s - \frac{180}{t_s}\right) \qquad \text{---- 24.7}$$

where η = Absolute viscosity in centipoise

γ_t = Specific gravity of the lubricant at temperature t° C

$= \gamma_{15.5} - 0.000637\,(t - 15.5)$

Values of specific gravity of oils at 15.5° C are given in Table 24.1 (DDHB)

Now, absolute viscosity $\eta = 10^{-3}\,\eta_1$ where η in Pas ---- 24.2 a

ii) **Oiliness :** Oiliness refer to wettability, surface tension and slipperiness. It is the ability of an oil to maintain an unbroken lubricating film between the rubbing surfaces.

iii) **Flash point :** Flash point is the lowest temperature at which an oil gives off sufficient vapour to give a momentary flash on introduction of a flame. A good lubricant should have the flash point above the operating temperature.

iv) **Fire point :** It is the lowest temperature at which an oil gives off sufficient vapour to burn continuously when ignited.

v) **Pour point :** It is the lowest temperature at which an oil ceases to flow when cooled.

vi) **Cloud point :** It is the temperature at which an oil becomes cloudy in appearance when cooled.

7.4 SELECTION OF LUBRICANT

The important factors that influence the choice of lubricant are:

i) The relative speed between members.

ii) The temperature range under which the lubricant must operate.

iii) The intensity of pressure between the mating surfaces.

iv) The atmosphere in which the bearing will operate.

v) The method of applying lubricant.

vi) The life required.

vii) The frictional losses allowed.

7.5 PURPOSE OF LUBRICATION

The purpose of lubrication is

i) To reduce friction between the moving surfaces of machine parts.

ii) To cool parts by carrying away the heat generated due to friction.

iii) To clean parts by washing away decomposition of carbon and metal particles caused by wear.

iv) To cushion the parts against vibration and impact.

v) To reduce wear and thereby increasing the life of bearing.

vi) To provide protection against corrosion.

vii) To seal the space between the piston and cylinder.

7.6 REQUIREMENT OF GOOD LUBRICANT

i) It must have sufficient viscosity.

ii) It should have physical stability with regard to temperature and pressure.

iii) High flash and fire point.

iv) Chemical stability against oxidation.

v) Resistance to emulsion.

vi) Non-volatile.

vii) Free from corrosion.

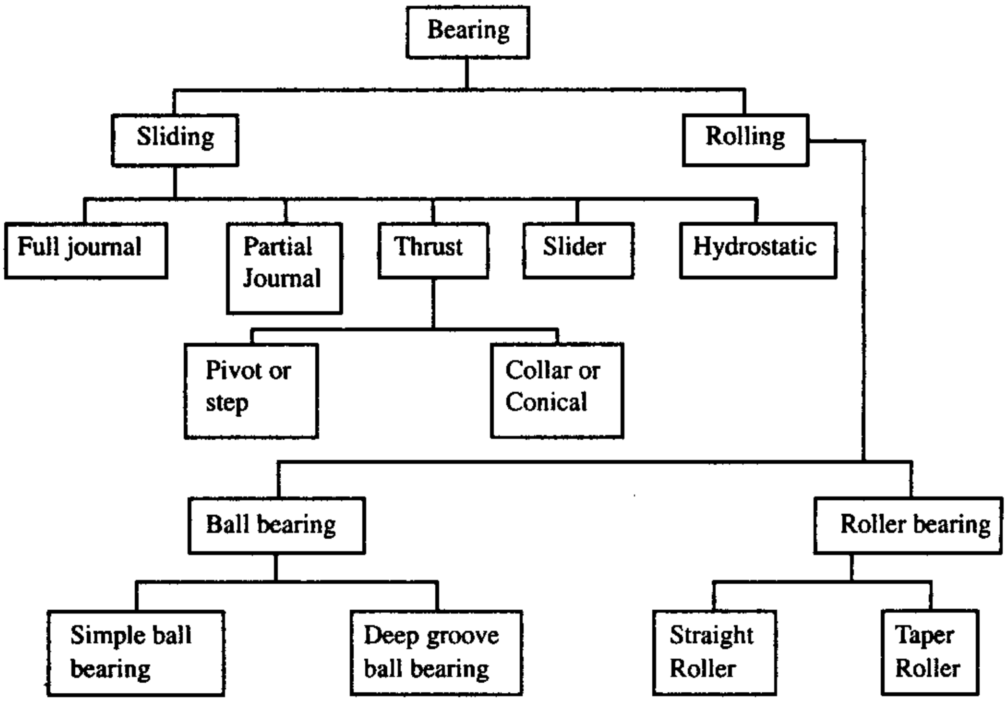
viii) Easy fluidity at low temperature.

7.7 BEARINGS

A bearing is a machine member, which supports, guide and restrain moving elements.

All machineries are provided with supports for rotating shafts. This supporting device is called bearing. It is a stationary member which carries the load. The portion of the shaft supported by the bearing is known as journal. The common applications of bearings are shafting in workshops, spindles of machine tools such as lathe, drilling, milling machine, axles of automobiles etc.

7.8 TYPES OF BEARINGS



7.9 CLASSIFICATION OF BEARINGS

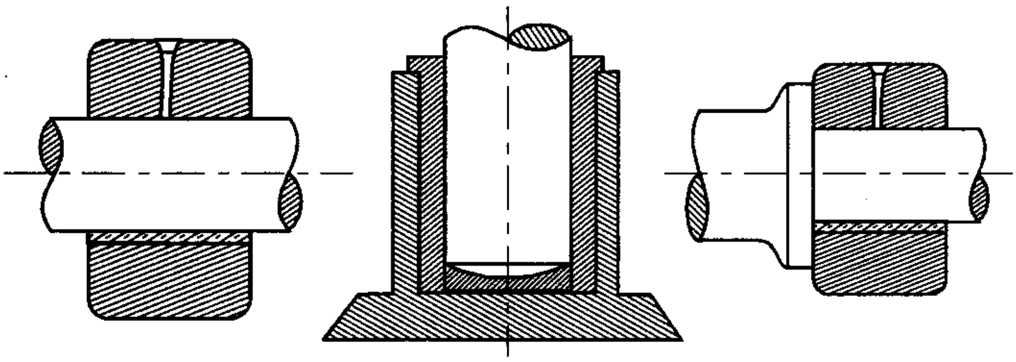
They may be classified as follows :

i) Depending upon the direction of load to be supported, it is classified as

a) Radial bearing b) Foot step or pivot bearing c) Thrust or collar bearing.

ii) Depending upon the nature of contact between the working surfaces, it is classified as

a) Sliding contact bearing b) Rolling contact bearing.



a) Radial or journal bearing ***b) Foot-step bearing or pivot bearing*** ***c) Thrust collar bearing bearing***

Fig. 7.1

If the relative motion between two machine parts is of rotation and the pressure on the bearing is perpendicular to the axis of the shaft, the bearing is known as a journal bearing. The part which is enclosed by and rubs against the other is called the journal and the part which encloses the journal is called bearing.

If the pressure is parallel to the axis of the shaft, the bearing is called a thrust bearing. In a thrust bearing if the shaft terminates at the bearing surface the bearing is a pivot bearing. If the shaft extends through and beyond the bearing, the bearing is known as a collar bearing. The common examples of collar thrust bearings are bearings of propeller shaft, shaft carrying worm and bevel gears. Spindles of drill presses etc.

A slider bearing is a bearing in which the surfaces are parallel or nearly parallel to each other and the relative motion is one of translation. A journal bearing of infinite radius would be a slider bearing.

In a hydrostatic bearing, lubricant is supplied at a high pressure to a pocket or pockets in the bearing which lifts the shaft.

In rolling bearing, the surfaces are in rolling contact in contrast to sliding contact as in sliding bearings. If the supported, member runs on cylindrical or conical rollers, the bearing is known as roller bearing. If hardened steel balls are used in place of rollers, the bearing is termed as ball bearing.

Sliding bearings are classified as

i) Thick film bearings in which the surfaces are completely separated from each other by the lubricant.

ii) Thin film or boundary lubricated on bearings in which inspite of lubricant presence, surfaces partially contact each other at certain amount of time.

iii) Zero film bearings which operate without any lubricant.

Lubrication by a pressurized fluid film may be divided into three groups

i) **Hydrostatic Lubrication :** In this the fluid film pressure is obtained by supplying the lubricant at high pressure through a set of holes in the bearing shell positioned so that force exerted by the pressurized lubricant supports the loaded journal at all times.

ii) **Hydrodynamic Lubrication :** In this the fluid film pressure is generated only by the rotation of the journal, the journal taking up a position in the bearing so that its rotation is able to produce a continuous film of lubricant where there is sufficient change of pressure to produce a force which will support the journal load.

The advantage of hydrostatic over hydro dynamic lubrication is in starting and stopping. The advantage of hydrodynamic bearing is that, it doesn't require external equipment to supply high pressure lubricant. The pressure is developed when journal runs at high speed. These two methods of lubrication can be combined so that a bearing can operate hydrostatically when starting and stopping and can act hydrodynamically during running.

iii) **Elasto-hydro dynamic lubrication :** In which the elastic deformation of the parts must be taken into account as well as the increase in viscosity of the lubricant due to high pressure. This small elastic flattening of the parts together with the increase in viscosity, provides a film, although very thin, that is much thicker than that would prevail with completely rigid parts.

7.10 HYDRODYNAMIC THEORY OF LUBRICATION

Perfect lubrication (thick film) is lubrication that maintains a complete film of lubricant between the surfaces of the shaft and bearing.

Boundary lubrication (thin film) is the type where metal to metal contact can sometimes occur with no lubricant, there is a continuous abrasive action between the two materials, which causes additional wear and generation of heat.

The action of lubricant in a plain journal bearing according to hydro dynamic theory is briefly explained as below:

When the journal is at rest, it comes in contact with the bearing at its lowest point P leaving a crescent shaped space above, which is filled with lubricant as shown in Fig 7.2.

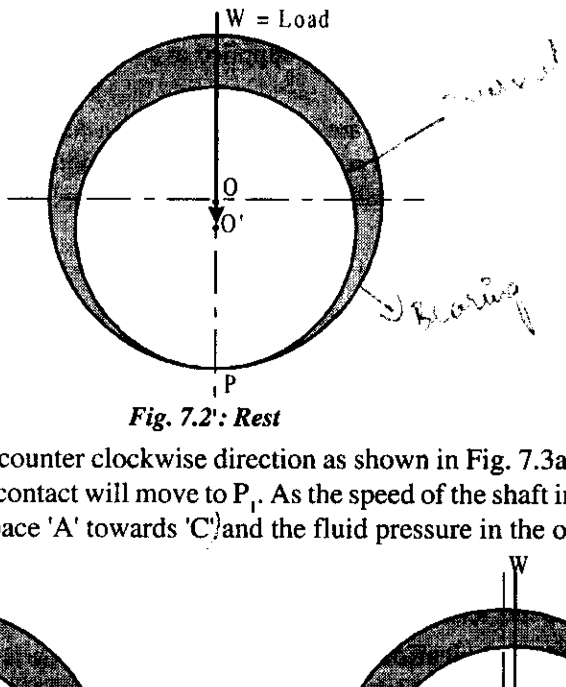


Fig. 7.2': Rest

When the shaft rotates in counter clockwise direction as shown in Fig. 7.3a, it tends to move above slightly and the point of contact will move to P_1. As the speed of the shaft increases it carries the oil with it from the wide space 'A' towards 'C' and the fluid pressure in the oil rises.

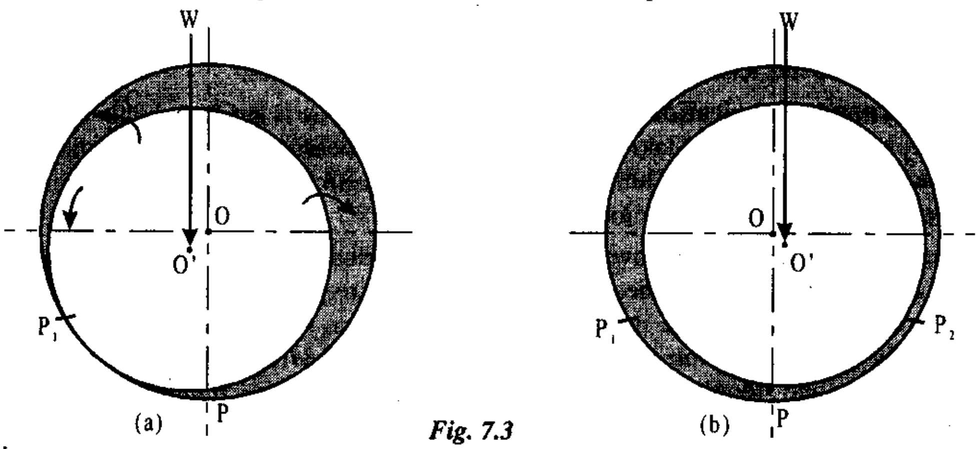


Fig. 7.3

When the speed is higher, the pressure is sufficient to carry the weight of journal. Then there is no metal-to-metal contact, but the shaft is supported by the oil, being nearest to the bearing surface at a point P_2 on the off side. (Fig. 7.3b). The point of maximum fluid pressure and position of the shaft will be as shown in Fig 7.4.

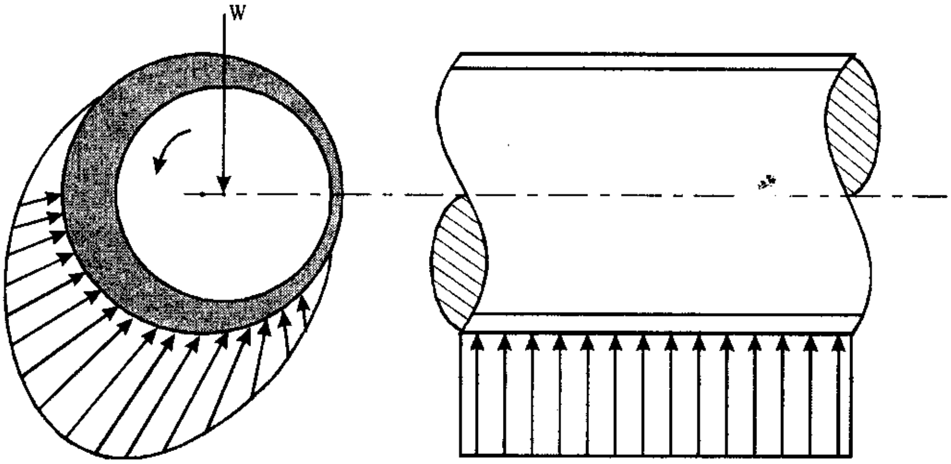


Fig. 7.4

Under the above conditions the fluid friction in the lubricant is substituted for sliding friction between the journal and the bearing. Fig 7.4 shows the ideal variation of pressure in the converging film in radial and axial directions.

Figure 7.5 shows the variations of friction with speed of rotation.

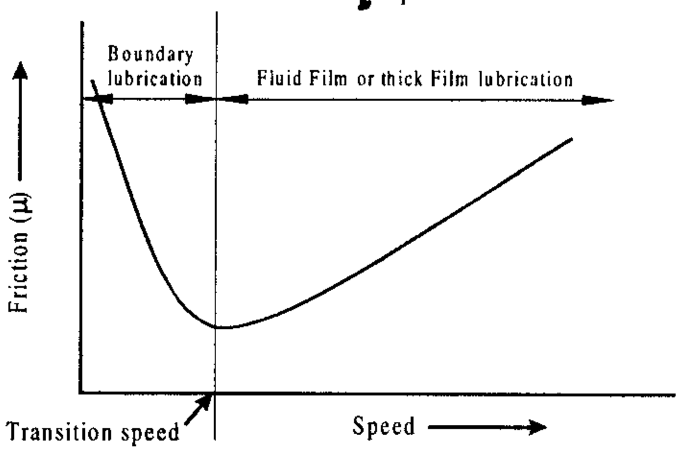


Fig. 7.5

If the speed of rotation is less than a critical value, depending on the load, lubrication and surface finish, complete separation of the surfaces by a film is not possible and some of the load is taken by the contact of asperities on the surfaces. This condition is called "boundary lubrication".

7.11 BEARING CHARACTERISTIC NUMBER AND BEARING MODULUS FOR JOURNAL BEARING

The coefficient of friction in design of bearings is of great importance. Various investigators, have shown that in journal, coefficient of friction is a function of at least three dimensionless parameters. $\frac{\eta n'}{P}$, $\frac{d}{c}$, and $\frac{L}{d}$

Where η = Absolute viscosity, Pas

n' = Speed in r.p.s.

P = Bearing pressure on projected area

$= \frac{W}{L.d}$, N/m^2

d = Diameter of journal, m

c = Diametral clearance, m

L = Length of bearing, m

The parameter $\frac{\eta n'}{P}$ is called bearing characteristic number and is a dimensionless number.

The variation of coefficient of friction with the operating values of bearing characteristic number is as shown in Fig 7.6.

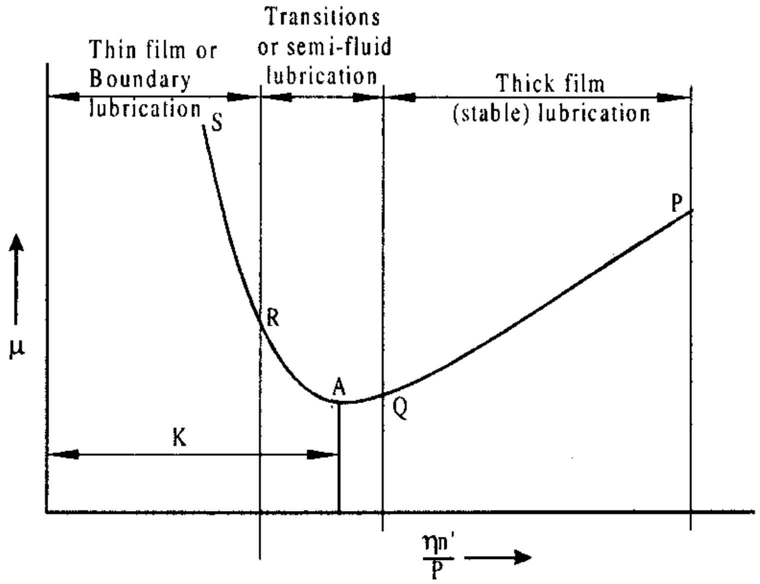


Fig. 7.6

The part of the curve PQ represents thick film lubrication. Between Q and R, the viscosity (η) or speed (n') are low or the pressure (P) is high, that their combination (η n'/P) will reduce the film thickness, so that partial metal to metal contact will take place. Therefore thin film or boundary or imperfect lubrication exists between R and S on the curve.

For any bearing there is a combination of η, n' and P that results in *minimum friction* indicated by 'K' on the curve. Values of $\frac{\eta n'}{P}$ greater than 'K' indicate that the bearing may operate with complete film lubrication.

(At values less than 'K' the oil film shall break down resulting metal to metal contact and consequent higher friction and wear. The value of $\frac{\eta n'}{P}$ where the oil film ruptures or break down is called *Bearing Modulus.*)

When the bearing is operated at or near this value, slight decrease in speed or increase in pressure may be accompanied by large increase in friction, heat and wear. To prevent such conditions the bearing should operate at values of $\frac{\eta n'}{P}$ atleast three times the minimum value of K and if the bearings is subjected to large fluctuations of load and heavy impacts, values as high as 15 K may be used.

7.12 SOMMERFELD NUMBER : (S)

Sommerfeld number (S) is another dimensionless parameter used extensively in lubrication analysis. Based on hydrodynamic theory, it can be shown that the sommerfeld number is a function of attitude only. It may be plotted against the quantity $\mu\left(\frac{d}{c}\right)$ which is also a function of attitude only and the coefficient of journal friction can be obtained. The sommerfeld number is

$$S = \left(\frac{\eta n'}{P}\right)\left(\frac{d}{c}\right)^2 = \frac{(\eta n')}{P}\left(\frac{1}{\psi^2}\right) \quad \text{where} \quad \psi = \frac{c}{d}$$

One of the main factors that Petroff's equation fails to take into account is the eccentricity of the bearing under load. The Sommerfeld number when plotted against $\mu\left(\frac{d}{c}\right)$ in accordance with hydro dynamic theory takes this eccentricity into account. The centre of the journal when under load is not concentric with the bearing but moves approximately along a semi circular arc of diameter $\frac{c}{2}$. This results in the establishment of a minimum film thickness $h_o(h_{min})$ as shown in the Fig 7.7. The distance between the bearing centre and the shaft centre is called the eccentricity and denoted by 'e'.

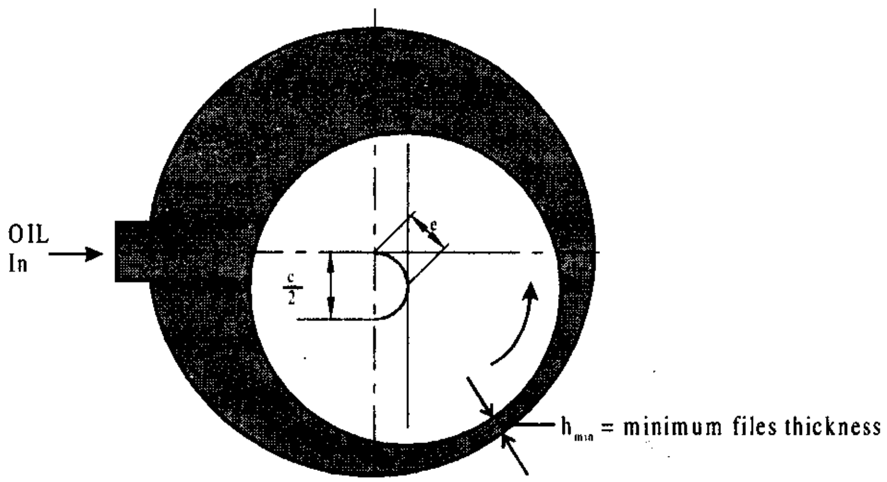


Fig. 7.7

The ratio of this eccentricity to the radial clearance is called the attitude or eccentricity ratio.

$$\text{Attitude} = \varepsilon = \frac{2e}{c} = 1 - \frac{2h_{min}}{c}$$

Coefficient of friction may be determined from Fig 7.8, where coefficient of friction variable is plotted against sommerfeld number.

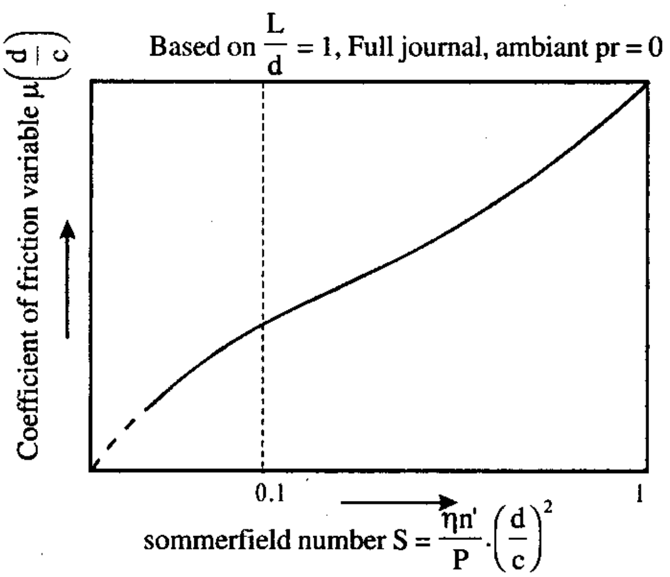
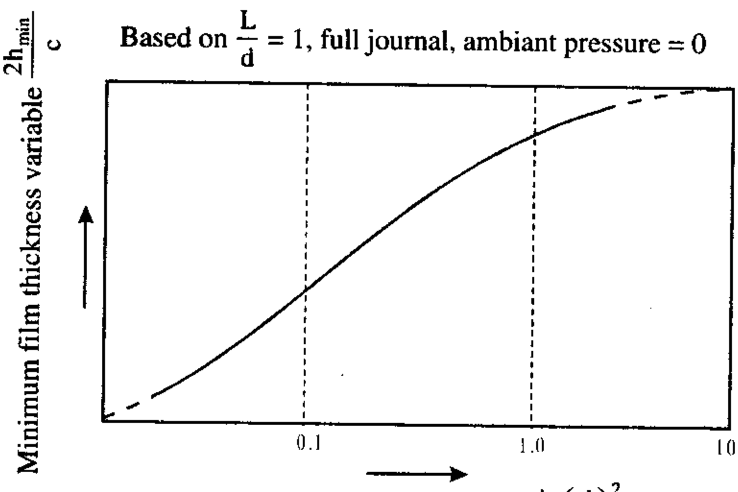


Fig. 7.8

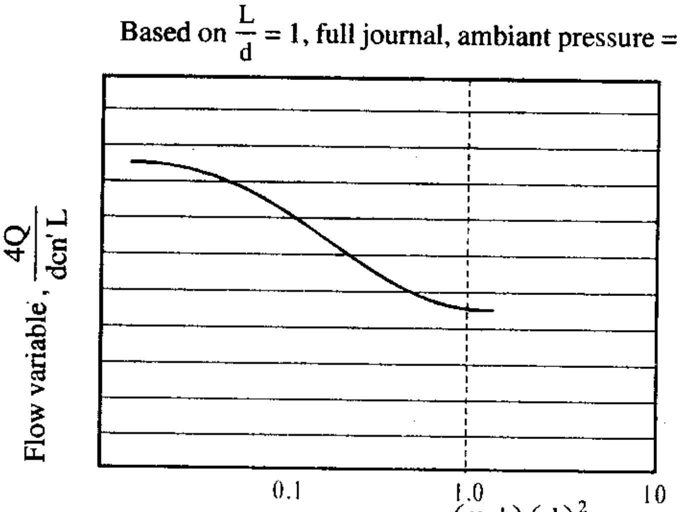
Minimum film thickness variable may be determined from Fig 7.9 where minimum film thickness variable $\frac{2h_o}{c}$ is plotted against the sommerfeld number.



Sommerfeld number $S = \frac{\eta n'}{P} \cdot \left(\frac{d}{c}\right)^2$

Fig. 7.9

Oil flow through the bearing due to pumping action of the shaft may be determined from Fig 7.10 where the flow variable $\frac{4Q}{dcn'L}$ is plotted against Sommerfeld number.



Sommerfeld number, $S = \left(\frac{\eta n'}{P}\right)\left(\frac{d}{c}\right)^2$

Fig. 7.10

7.13 TERMS USED IN HYDRODYNAMIC JOURNAL BEARING

i) **Diametral Clearance :** It is the difference between the diameter of journal or shaft and the inside diameter of the bearing in which it rotates

$\therefore \quad c = D - d$ where c = Diametral clearance
D = Diameter of Bearing
d = Diameter of journal or shaft

ii) **Radial Clearance :** Radial clearance is one half the diametral clearance, i.e., it is the difference between the radii of the bearing and the journal

$$\therefore \quad \text{Radial clearance } c_r = R - r = \frac{D-d}{2} = \frac{c}{2}$$

iii) **Diametral Clearance Ratio (ψ) :** It is the ratio of the diametral clearance to the diameter of the journal.

$$\therefore \quad \text{Diameter clearance ratio } \psi = \frac{c}{d}$$

iv) **Eccentricity (e) :** It is the radial distance between the centre 'O' of the bearing and the displaced centre O' of the bearing under load.

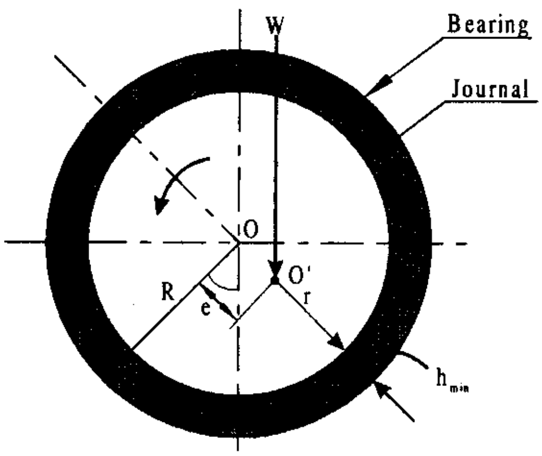


Fig. 7.11

v) **Minimum oil film thickness :** It is the minimum distance between the bearing and the journal, under complete lubrication condition. It is denoted by 'h_{min}' and occur at the line of centres as shown in Fig 7.11.

vi) **Attitude or eccentricity ratio (ε) :** It is the ratio of the eccentricity to the radial clearance.

$$\therefore \quad \varepsilon = \frac{e}{c_r} = \frac{c_r - h_{min}}{c_r} = 1 - \frac{h_{min}}{c_r} = 1 - \frac{2h_{min}}{c} \qquad \left(\because c_r = \frac{c}{2}\right)$$

vii) **Short and long bearing :** If the ratio of the length to the diameter of the journal is less than 1, then the bearing is said to be short bearing.

If the ratio of the length to the diameter of the journal is greater than 1, then the bearing is said to be long bearing.

viii) **Square Bearing :** When the length of the journal is equal to the diameter of the journal, then the bearing is called square bearing.

Note :

Because of the side leakage of the lubricant, the pressure in the film is atmospheric at the ends of the bearing.

7.14 PETROFF'S EQUATION

Consider a vertical shaft rotating in a guide bearing. It is assumed that the bearing.

i) Carries a very small load.
ii) The clearance c is completely filled with oil.
iii) The end leakage is completely negligible.
iv) The oil used is of high viscosity.
v) The journal revolves at very high speeds.
vi) The bearing run concentrically.

Let, d = diameter of journal or shaft

c = Diametral clearance

n' = Speed of shaft or journal in r.p.s

$= \frac{n}{60}$

L = Length of bearing

$\psi = \frac{c}{d}$ = diametral clearance ratio

η = Viscosity of oil in Pas

v = Velocity $= \frac{\pi dn}{60} = \pi dn'$ m/sec.

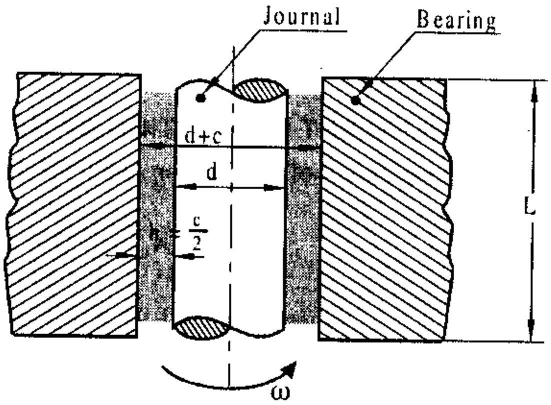


Fig. 7.12

Shear stress $\tau = \eta.\frac{v}{h} = \eta.\frac{\pi dn'}{\left(\frac{c}{2}\right)} = \frac{2\pi d\eta.n'}{c}$

Surface area $A = \pi dL$

$\therefore$ Force $F = \tau A = \frac{(2\pi d\eta.n')}{c}(\pi dL) = \frac{2\pi^2 d^2 n'\eta.L}{c}$

Torque M_t = Force × radius $= F.\frac{d}{2} = \frac{2\pi^2 d^2 n'\eta.L}{c}\frac{d}{2}$

$$\therefore M_t = \frac{\pi^2 d^2 n'\eta.L}{\left(\frac{c}{d}\right)} = \frac{\pi^2 d^2 n'\eta.L}{\psi} \quad \text{---- (1)}$$

But, $M_t = (\mu W)\dfrac{d}{2}$ where μW = Friction force

$W = P.A = P.L.d = \text{Load}$

P = Bearing pressure in N/m^2

$$\therefore \quad \mathbf{M_t = \mu.(PLd).\frac{d}{2}} \qquad \text{---- (2)}$$

Equating (1) and (2)

$$\mu\,(PLd).\frac{d}{2} = \frac{\pi^2 d^2 n' \eta . L}{\psi}$$

$$\therefore \quad \text{Coefficient of friction} \quad \mu = 2\,\pi^2\left(\frac{\eta . n'}{P}\right).\frac{1}{\Psi}$$

This equation is called Petroff's equation.

7.15 COEFFICIENT OF FRICTION BY MCKEE

The coefficient of friction for a well lubricated journal bearing is

$$\mu = K_a\left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi}\right) 10^{-10} + \Delta\mu \qquad \text{---- 24.22}$$

where K_a = 1.95×10^{11} for full journal bearing

= $5.4 \times 10^{8}\,\beta$ for partial bearing

β = Attitude angle

η = Absolute viscosity in Pas

$\Delta\mu$ = Factor to correct for end leakage = 0.002

ψ = Diametral clearance ratio = $\dfrac{c}{d}$

n' = Speed in rps = $\dfrac{n}{60}$

P = Bearing pressure.

7.16 BEARING MATERIALS

A good bearing material must have the following properties :

1. The material should have good antiweld and anti scoring properties.
2. High thermal conductivity.
3. High fatigue strength.
4. Good machinability.
5. Good corrosion resistance.

6. Low cost.
7. Soft enough to absorb foreign materials.

Some of the bearing materials used are white metal alloy, copper base materials, cadmium base alloys, Aluminium base alloys, silver, cast iron, steel, porous metals etc.

7.17 PROCEDURAL STEPS FOR THE DESIGN OF JOURNAL BEARING

If the load is perpendicular to the axis of shaft, then the support is known as journal bearing and that portion of the shaft resting in the bearing is called journal.

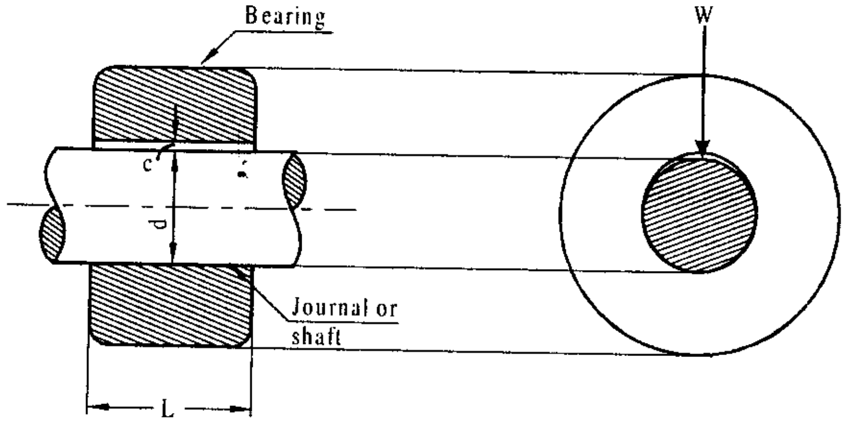


Fig. 7.13

Let
d = diameter of shaft or journal
c = diametral clearance
L = Length of bearing
ψ = Diametral clearance ratio = $\frac{c}{d}$
η = Absolute viscosity in Pas
n = Speed in rpm
n' = Speed in rps = $\frac{n}{60}$
P = Bearing pressure in N/m² = $\frac{W}{Ld}$
W = Applied load
t_o = Operating temperature in °C
t_b = Bearing temperature in °C
t_a = Ambient or room temperature

μ = Coefficient of friction

h_{min} = Minimum film thickness

v = Velocity = $\frac{\pi dn}{60}$, m/sec

S" = Bearing modulus = $\frac{\eta n'}{P}$

β = Angle of bearing = 360° for full journal bearing

Q = Quantity of oil flow, m^3/sec

N_μ = Power lost in friction

1. **Select the recommended design value for the given type of bearing and machine or machinery from Table 24.2 (DDHB)**

   i.e. Maximum pressure P

   Diametral clearance ratio $\psi = \frac{c}{d}$

   Ratio $\frac{L}{d}$

   Viscosity η in Pas

   Bearing modulus S" in SI units

2. **Design**

   Select the allowable pressure as 50 to 60% of the maximum pressure

   i.e. $P = \frac{W}{Ld}$ where Ld = Projected area of the bearing

   ∴ Length of bearing 'L' (Select whole number or round the values)

   Diameter of journal 'd'

   After rounding off the values of L and d, find once again the pressure. It gives the actual pressure.

3. **Selection of oil**

   Select η from first step and assume the operating temperature between 60° and 70° C if it is not given.

   Now using Fig 24.2 or 24.2 b (DDHB) for the given η and t_o select the oil.

   For the selected oil, using the same figure note down the actual value of η.

4. **Check for the oil**

   Bearing modulus $S_{in}" = \frac{n' \eta_{act}}{P_{act}}$

   If $S"_{in} > S"_{recom}$ then thick film lubrication is possible. If thick film lubrication is possible, then the selected oil is suitable, otherwise select the next higher oil and check again.

5. Heat generated

Heat generated $H_g = \mu\,(PLd)\,v$, J/s or Watts ---- 24.72 a

where $v = \frac{\pi dn}{60}$; $P = P_{act}$

$$\mu = K_a\left(\frac{\eta n'}{P}\right)\frac{1}{\psi} \times 10^{-10} + \Delta\mu \qquad \text{---- 24.22}$$

where $K_a = 1.95 \times 10^{11}$ for $\beta = 360°$; $\Delta\mu = 0.002$

The heat generated can also be found by knowing the temperature rise of lubricant oil which is used to carry away heat generated in the bearing $H_g = \gamma C_{SP}\, Q\Delta T$ ---- 24.72 c

where γ = Weight per unit volume of lubricant in N/m^2

C_{SP} = Specific heat of lubricant in J/NK

Q = Flow rate of lubricant in m^3/sec

ΔT = Difference between outlet and inlet temperature of the lubricant in °K

6. Heat dissipated

Heat dissipated by self contained bearings $H_d = CA\,(t_b - t_a)$ Watts ---- 24.77

where C = Combined coefficient of radiation and convection

$= 11.36 \times 10^{-3}\ kW/m^2K = 11.36\ W/m^2 K$

A = Effective surface area of bearing housing = 25 dL in m^2

t_b = Temperature of bearing °C

t_a = Ambient or temperature of surrounding air °C

$$t_b - t_a \approx \frac{t_o - t_a}{2}$$

Also according to Lasche's equation for heat radiating capacity of bearing

$$H_d = \frac{(\Delta T + 18)^2 (Ld)}{K'} \text{ Watts}$$

K' = 0.475 for bearings of light construction in still air

= 0.273 for bearings of heavy construction in well ventilated.

Heat dissipating capacity of bearing based on projected area of bearing

$H_d = 697.8\,k\,(t_b - t_a)\,Ld$ Watts ---- 24.79

where $k\,(t_b - t_a)$ are taken from Fig 24.40 a (DDHB)

7. Minimum film thickness

Minimum oil film thickness variable $\delta = \frac{2h_{min}}{c} = 1 - \epsilon$ ---- 24.33

where ϵ = Attitude = $\frac{2e}{c}$ ---- 24.30

c = Diametral clearance

h_{min} = Minimum film thickness

Bearing characteristic number $S = \frac{\eta n'}{P} \cdot \frac{1}{\psi^2}$

Figure 24.13 a (Page 24.33) shows oil film thickness variable δ depends upon S and $\frac{L}{d}$

For side leakage Fig 24.26 (DDHB) shows the relationship between load correction factor C_w, $\frac{B}{L}$ and δ.

8. Flow rate

Figure 24.37 (DDHB) shows flow variable depends upon bearing characteristic number S and $\frac{L}{d}$

$$\text{Flow variable } \lambda_Q = \frac{4Q\psi}{c^2 n' L} \quad \text{---- 24.70}$$

Where Q = Quantity of oil flow without side leakage

$$\text{Oil flow through a bearing with side leakage } Q = \frac{\lambda_Q C^2 n' L C_Q}{4\psi} \quad \text{---- 24.71 b}$$

Where C_Q = Flow correction factor taken from figure 24.38

9. Power loss due to friction

$$N_\mu = \frac{\mu W v}{1000} \text{ kW}$$

Example : 7.1

A 75 mm long full journal bearing of diameter 75 mm supports a load of 10 kN. The speed of the journal is 1200 rpm. The absolute viscosity of the oil is 10×10^{-3} Pas and diametral clearance ratio is 0.001. Determine the coefficient of friction by using (i) Petroff's equation (ii) Mckee's equation and (iii) Raimondi and Boyd curve.

Data :

L = 75 mm; d = 75 mm; $W = 10 \text{ kN} = 10^4$ N; n = 1200 rpm;

$\eta = 10 \times 10^{-3}$ Pas; $\psi = 0.001$

Solution :

i) By using Petroff's equation

$$\text{Coefficient of friction } \mu = 2\pi^2 \left(\frac{\eta n'}{P}\right) \cdot \frac{1}{\psi} \quad \text{---- 24.21}$$

$$\text{Bearing pressure } P = \frac{W}{Ld} = \frac{10^4}{(0.075)(0.075)} = 1.778 \times 10^6$$

Speed in rps $n' = \frac{n}{60} = \frac{1200}{60} = 20$

$$\therefore \quad \mu = 2\pi^2\left(\frac{10\times10^{-3}\times20}{1.778\times10^6}\right)\left(\frac{1}{0.001}\right) = 2.22\times10^{-3}$$

ii) By using Mckee's equation

Coefficient of friction $\mu = K_a\left(\frac{\eta n'}{P}\right)\frac{1}{\psi}\times10^{-10} + \Delta\mu$ ---- 24.22

where $K_a = 1.95\times10^{11}$ for full journal bearing

$\Delta\mu = 0.002$

$$\therefore \quad \mu = (1.95\times10^{11})\left(\frac{10\times10^{-3}\times20}{1.778\times10^6}\right)\left(\frac{1}{0.001}\right)\times10^{-10} + 0.002 = 4.2\times10^{-3}$$

ii) By using Raimondi and Boyd curve

Length of bearing in the direction of motion $B = \frac{\pi d\beta^\circ}{360}$ ---- 24.19

$$= \frac{\pi\times0.075\times360}{360} = \pi\times0.075$$

$$\therefore \quad \text{Ratio } \frac{B}{L} = \frac{\pi\times0.075}{0.075} = \pi = 3.14$$

Sommerfeld or Bearing characteristic number $S = \frac{\eta n'}{P}\cdot\frac{1}{\psi^2}$

$$= \left(\frac{10\times10^{-3}\times20}{1.778\times10^6}\right)\left(\frac{1}{0.001}\right)^2 = 0.1125$$

From Fig 24.19 (DDHB) for $S = 0.1125$ and $\frac{B}{L} = 3.14$

Coefficient of friction variable $\frac{\mu}{\psi} = 4.5$

$\therefore$ Coefficient of friction $\mu = 4.5\,\psi = 4.5\times10^{-3}$

PA *Example : 7.2*

Determine the power loss for a Petroff bearing 100 mm in diameter and 150 mm long. The radial clearance is 0.05 mm. Speed of the journal is 1000 rpm. The lubricating oil is SAE 10 and bearing operating temperature is 60° C.

Data :

$d = 100$ mm $= 0.1$ m; $L = 150$ mm $= 0.15$ m;

$c_r = 0.05$ mm $= 0.05\times10^{-3}$ m; $n = 1000$ rpm

Automobile oil SAE 20; $t_o = 60°$ C

Solution :

From Table 24.1 for SAE 10 the number specified is B.

Now from Fig 24.2 (DDHB) for $t_o = 60°$ and B-oil

Absolute viscosity $\eta = 14 \text{ cP} = 14 \times 10^{-3}$ Pas

[Figure 24.2 b (DDHB) can also be used to find absolute viscosity η]

Speed in rps $n' = \frac{n}{60} = \frac{1000}{60} = 16.667$ m/sec

Diametral clearance $c = 2 \times$ radial clearance $= 2 \times 0.05 \times 10^{-3}$

$= 0.1 \times 10^{-3}$ m

$\therefore$ Diametral clearance ratio $\psi = \frac{c}{d} = \frac{0.1\times10^{-3}}{0.1} = 10^{-3}$

$\therefore$ According to Petroff's equation coefficient of friction

$$\mu = 2\pi^2\left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi}\right) = 2\pi^2\left(\frac{14\times10^{-3}\times16.667}{P}\right)\left(\frac{1}{10^{-3}}\right) = \frac{4605.9}{P}$$

$\therefore$ Power loss due to friction $N_\mu = \frac{\mu W v}{1000}$ kW

$$= \frac{\mu(PLd)\left(\frac{\pi dn}{60}\right)}{1000}$$

$$= \frac{\left(\frac{4605.9}{P}\right)(P\times0.15\times0.1)\left(\frac{\pi\times0.1\times1000}{60}\right)}{1000}$$

$= 0.36175 \text{ kW} = 361.75$ Watts

Example : 7.3

The viscosity of an oil is 110 saybolt seconds at 50° C and the specific gravity is 0.8894 at 15.5° C. Determine the absolute viscosity at the bearing operating temperature of 80° C.

Data :

$$\gamma_{15.5} = 0.8894; \quad t_o = 80° \text{ C.}$$

Solution :

Specific weight at the operating temperature

$$\gamma_t = \gamma_{15.5} - 0.000637\,(t - 15.5) \quad \text{---- 24.10 a}$$

$= 0.8894 - 0.000637\,(80 - 15.5)$

$= 0.8483135$ N/m³

Saybolt centipoise $\eta_t = \gamma_t\left(0.22t - \frac{180}{t}\right)$ ---- 24.7

Where t Saybolt seconds at 80° C.

From Fig 24.3 (DDHB) for 110 saybolt seconds at 50° C, the oil selected is B.

Now from the same figure for B-oil at 80° C,

Saybolt seconds $t = 56.5$

$$\therefore \eta_1 = 0.8483135\left[0.22\times 56.5-\frac{180}{56.5}\right] = 7.842 \text{ cP}$$

$\therefore$ Absolute viscosity $\eta = 7.842 \times 10^{-3}$ Pas or

2nd Method

From Fig 24.3 (DDHB) for 110 Saybolt seconds at 50° C, the oil selected is B

Now from Fig 24.2 (DDHB) for B-oil at 80° C

Absolute viscosity $\eta = 8 \text{ cP} = 8 \times 10^{-3}$ Pas.

Example : 14

A 75 mm long full journal bearing of diameter 75 mm supports a radial load of 12 kN at the shaft speed of 1800 rev/min. Assume ratio of diameter to the diametral clearance as 1000. The viscosity of oil is 0.01 Pas at the operating temperature. Determine the following.

a) Sommerfeld number

b) The coefficient of friction based on Mckee equation

c) Amount of heat generated. **VTU, July/August 2004**

Data :

$$L = 75 \text{ mm} = 0.075 \text{ m};\ d = 75 \text{ mm} = 0.075 \text{ m};$$

$$W = 12 \text{ kN} = 12\times 10^3 \text{ N};\ n = 1800 \text{ rpm},\ \frac{d}{c} = 1000\ \therefore\ \psi = 10^{-3};\ \eta = 0.01 \text{ Pas}$$

Full journal

Solution :

a) Sommerfeld number

$$\text{Sommerfeld number } S = \left(\frac{\eta n'}{P}\right)\cdot\frac{1}{\psi^2}$$

$$\text{Bearing pressure } P = \frac{W}{Ld} = \frac{12\times 10^3}{0.075\times 0.075} = 2.133\times 10^6 \text{ N/m}^2$$

$$\text{Speed in rps } n' = \frac{n}{60} = \frac{1800}{60} = 30$$

$$\text{i.e. } S = \left(\frac{0.01\times 30}{2.133\times 10^6}\right)\left(\frac{1}{10^{-3}}\right)^2 = 0.140625$$

$\therefore$ Sommerfeld number $S = 0.140625$

b) Coefficient of friction by Mckee equation

$$\text{Coefficient of friction } \mu = K_a\left(\frac{\eta n'}{P}\right)\frac{1}{\psi}\times 10^{-10} + \Delta\mu \qquad \text{---- 24.22}$$

where $K_a = 1.95 \times 10^{11}$ for full journal bearing

$\Delta\mu = 0.002$

$$\therefore \mu = (1.95\times 10^{11})\left(\frac{0.01\times 30}{2.133\times 10^6}\right)\left(\frac{1}{10^{-3}}\right)\times 10^{-10} + 0.002 = 4.7422\times 10^{-3}$$

c) Heat generated

Heat generated $H_g = \mu\,(PLd)\,v$ ---- 24.72 a

$$= (4.7422 \times 10^{-3})\,(2.133 \times 10^{6} \times 0.075 \times 0.075)\left(\frac{\pi \times 0.075 \times 1800}{60}\right)$$

$$= 402.246 \text{ watts}$$

Example : 7.5

A lightly loaded journal bearing has a load of 1 kN. The oil used is SAE 60 and mean effective temperature of operation is 40° C. The journal has a diameter of 50 mm and the bearing has a diameter of 50.5 mm. The speed of journal is 15000 rpm. The $\frac{L}{d}$ ratio is limited to 1.2. Determine the coefficient of friction and power loss in friction. **VTU, January/February 2004**

Data :

$$W = 1 \text{ kN} = 1000 \text{ N}; \text{ Oil-SAE 60}; \; t_o = 40° \text{ c}; \; d = 50 \text{ mm} = 0.05 \text{ m};$$

$$D = 50.5 \text{ mm} = 0.0505 \text{ m}; \; n = 15000 \text{ rpm}; \; \frac{L}{d} = 1.2$$

Solution :

$$\frac{L}{d} = 1.2; \text{ i.e. } \frac{L}{0.05} = 1.2$$

$\therefore$ Length of bearing $L = 0.06$ m

From Table 24.1 (DDHB) for SAE 60 oil, select the oil number H.

Now from Fig 24.2 (DDHB) for oil H and $t_o = 40°$ C

Absolute viscosity $\eta = 210$ cP $= 210 \times 10^{-3}$ Pas

[Figure 24.2 b (DDHB) can also be used to find η]

Diametral clearance $c = D - d = 0.0505 - 0.05 = 0.0005$

Diametral clearance ratio $\psi = \frac{c}{d} = \frac{0.0005}{0.05} = 0.01$

Speed of journal in rps $n' = \frac{n}{60} = \frac{15000}{60} = 250$

Bearing pressure $P = \frac{W}{Ld} = \frac{1000}{(0.06)(0.05)} = 0.333 \times 10^{6}$ N/m²

Coefficient of friction by Mckee's equation

$$\mu = K_a\left(\frac{\eta n'}{P}\right)\frac{1}{\psi} \times 10^{-10} + \Delta\mu$$ ---- 24.22

Assume full journal $\therefore$ $K_a = 1.95 \times 10^{11}$; $\Delta\mu = 0.002$

$$\therefore \; \mu = (1.95 \times 10^{11})\left(\frac{210 \times 10^{-3} \times 250}{0.333 \times 10^{6}}\right)\left(\frac{1}{0.01}\right) \times 10^{-10} + 0.002$$

$$= 0.309125$$

Power loss in friction $N_\mu = \dfrac{\mu W v}{1000}$ kW

$$= \frac{(0.309125)(1000)\left(\dfrac{\pi \times 0.05 \times 15000}{60}\right)}{1000}$$

$= 12.14$ kW

Or

Since the journal is lightly load and it revolves at very high speed, Petroff's equation can also be used for determining the coefficient of friction.

According to Petroff's equation $\mu = 2\pi^2\left(\dfrac{\eta n'}{P}\right)\dfrac{1}{\Psi}$ ---- 24.21

$$= 2\pi^2\left(\frac{210\times10^{-3}\times250}{0.333\times10^{6}}\right)\left(\frac{1}{0.01}\right) = 0.3108925 \approx 0.3109$$

i.e., $\mu = 0.3109$

Power loss due to friction $N_\mu = \dfrac{\mu W v}{1000}$ kW

$$= \frac{(0.3109)(1000)\left(\dfrac{\pi \times 0.05 \times 15000}{60}\right)}{1000} = 12.2087\text{ kW} \approx 12.21\text{ kW}$$

Example : 7.6

SAE 20 oil is used to lubricate a hydrodynamic journal bearing of diameter 75 mm and length 75 mm, oil enters at 40° C. The journal rotates at 1200 rpm. The diametral clearance is 75 μm (0.075 mm). Assume operating temperature of the oil as 53° C and determine

a) Magnitude and location of the minimum film thickness

b) Power loss

c) Oil flow through the bearing

d) Side leakage **VTU, February 2002**

Data :

Oil SAE 20; d = 75 mm = 0.075 m; L = 75 mm = 0.075 m;
t_i = 40° C; n = 1200 rpm; c = 0.075 mm = 0.075 × 10^{-3} m; t_o = 53° C

Solution :

Diametral clearance ratio $\psi = \dfrac{c}{d} = \dfrac{0.075}{75} = 0.001$

From Table 24.1 (DDHB) for SAE 20, select oil No E

Now from Fig 24.2 (DDHB) for E-oil and $t_o = 53°$

Absolute viscosity $\eta = 32$ cP $= 32 \times 10^{-3}$ Pas

[Figure 24.2 b (DDHB) can also be used to find η]

Speed in rps $n' = \frac{1200}{60} = 20$

Velocity $v = \frac{\pi dn}{60} = \frac{\pi \times 0.075 \times 1200}{60} = 4.7124$ m/sec

a) Magnitude and location of the minimum film thickness

Length in the direction of motion $B = \pi d \frac{\beta^\circ}{360^\circ}$ ---- 24.19

$$= \pi \times 0.075 \times \frac{360}{360} = \pi \times 0.075$$

$$\therefore \frac{\text{Length in the direction of motion}}{\text{Length in direction perpendicular to motion}} = \frac{B}{L} = \frac{\pi \times 0.075}{0.075} = 3.142$$

From Fig 24.24 (DDHB) for $\frac{B}{L} = 3.142$, load leakage factor $C_w = 0.086$

Now from Fig 24.26 (DDHB) for $\frac{B}{L} = 3.142$ and $C_w = 0.086$

Minimum film thickness variable $\delta = 0.8$

Also $\delta = \frac{2h_{min}}{c}$ ---- 24.33

i.e., $0.8 = \frac{2 \times h_{min}}{0.075 \times 10^{-3}}$

$\therefore$ Minimum film thickness $h_{min} = 3 \times 10^{-5}$ m

From figure 24.13 a (Page 24.33) (DDHB) for $\delta = 0.8$, $\frac{L}{d} = \frac{0.075}{0.075} = 1$ and assuming full journal bearing

Bearing characteristics number or Sommerfeld number

$S = 0.65$

From Fig 24.11 a (Page 24.30) (DDHB) for $S = 0.65$ and $\frac{L}{d} = 1$

Position of minimum film thickness $\phi = 74^\circ$

Attitude $\epsilon = 1 - \delta = 1 - 0.8 = 0.2$ ---- 24.33

Also $\epsilon = \frac{2e}{c}$

i.e., $0.2 = \frac{2e}{0.075 \times 10^{-3}}$

$\therefore$ Eccentricity $e = 7.5 \times 10^{-6}$ m

Hence minimum film thickness $h_{min} = 3 \times 10^{-5}$ m

Location or position of minimum film thickness $\phi = 74^\circ$

Eccentricity $e = 7.5 \times 10^{-6}$ m

b) Power loss

Power loss due to friction $N_\mu = \mu W v = \mu\,(PLd)\,v$ Watts

Sommerfeld number $S = \dfrac{\eta n'}{P} \cdot \dfrac{1}{\psi^2}$ ---- 24.39

$$\text{i.e., } 0.65 = \left(\frac{32 \times 10^{-3} \times 20}{P}\right)\left(\frac{1}{0.001}\right)^2$$

$\therefore$ Bearing pressure $P = 0.9846 \times 10^6$ N/m^2

According to Mckee's equation, coefficient of friction

$$\mu = K_a \left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi}\right)(10^{-10}) + \Delta\mu \quad \text{---- 24.22}$$

$K_a = 1.95 \times 10^{11}$ for full journal; $\Delta\mu = 0.002$

$$\therefore \ \mu = (1.95 \times 10^{11}) \left(\frac{32 \times 10^{-3} \times 20}{0.9846 \times 10^6}\right)\left(\frac{1}{0.001}\right) 10^{-10} + 0.002 = 0.014675$$

$\therefore$ Power loss $N_\mu = (0.014675)\,(0.9846 \times 10^6 \times 0.075 \times 0.075)\,(4.7124) = 383$ Watts

c) Oil flow through bearing

From Fig 24.37 for $S = 0.65, \beta = 360°$ and $\dfrac{L}{d} = 1$

Flow variable $\lambda_Q = 3.55$

From Fig 24.38 (DDHB) for $\dfrac{B}{L} = 3.142$ and $\delta = 0.8$

Flow correction factor $C_Q = 1.3$

$\therefore$ Oil flow through a bearing with side leakage

$$Q = \frac{\lambda_Q c^2 n' L}{4\psi} \cdot C_Q = \frac{3.55 \times \left(0.075 \times 10^{-3}\right)^2 (20)(0.075)}{4 \times 0.001} \times 1.3$$

$$= 9.735 \times 10^{-6} \text{ m}^3/\text{sec}$$

d) Side leakage

From Figure 24.39 (DDHB) for $S = 0.65$ and $\dfrac{L}{d} = 1$

Oil flow ratio $\dfrac{Q_S}{Q} = 0.26$

$\therefore$ $Q_s = 0.26\,Q = 0.26 \times 9.735 \times 10^{-6} = 2.531 \times 10^{-6}$ m^3/sec

= Side flow of lubricant

Example : 7.7

The oiliness curve for a 75 mm × 150 mm long bearing happens to be a straight line passing through points $\dfrac{\eta n'}{P} = 191 \times 10^{-9}$ and $\mu = 0.002$ and another point $\dfrac{\eta n'}{P} = 956 \times 10^{-9}$ and $\mu = 0.0065$. The load

supported by the bearing is 6 kN and the speed of the journal 1200 rpm. Calculate the friction loss in kW at the bearing, if the oil film has a temperature of 80° C and the viscosity of the lubricant is 8.7 cP at the operating temperature. Assume $\psi = 0.001$ **VTU, July/August 2003**

Data :

$$d = 75 \text{ mm} = 0.075 \text{ m}; \; L = 150 \text{ mm} = 0.15 \text{ m}; \; \mu = 0.002 \text{ when } \frac{\eta n'}{P} = 191 \times 10^{-9}; \; \mu = 0.0065$$

$$\text{when } \frac{\eta n'}{P} = 956 \times 10^{-9}; \; W = 6 \text{ kN} = 6000 \text{ N}; \; n = 1200 \text{ rpm}; \; t_o = 80° \text{ C};$$

$$\psi = 0.001; \; \eta = 8.7 \text{ cP} = 8.7 \times 10^{-3} \text{ Pas}$$

Solution :

$$\text{Bearing pressure } P = \frac{W}{Ld} = \frac{6000}{(0.15)(0.075)} = 0.533 \times 10^6 \text{ N/m}^2$$

$$\text{Speed in rps } n' = \frac{1200}{60} = 20$$

$$\text{Bearing modulus } S'' = \frac{\eta n'}{P} = \frac{8.7 \times 10^{-3} \times 20}{0.533 \times 10^6} = 326.25 \times 10^{-9}$$

$$\text{Velocity } v = \frac{\pi d n}{60} = \frac{\pi \times 0.075 \times 1200}{60} = 4.712 \text{ m/sec}$$

Since the points joining bearing modulus are straight line, by linear interpolation

$$\frac{x}{0.0065 - 0.002} = \frac{326.25 \times 10^{-9} - 191 \times 10^{-9}}{956 \times 10^{-9} - 191 \times 10^{-9}}$$

$\therefore \; x = 7.956 \times 10^{-4}$

$\therefore$ For $\frac{\eta n'}{P} = 326.25 \times 10^{-9}$, coefficient of friction

$\mu = 0.002 + 7.956 \times 10^{-4} = 2.7956 \times 10^{-3}$

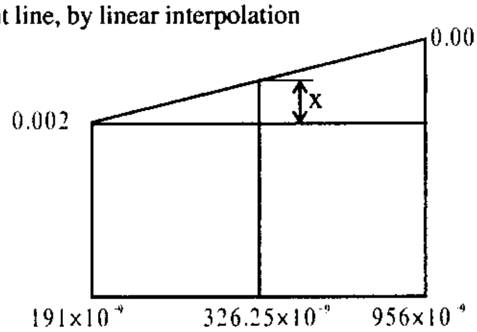


Fig.

$$\therefore \text{ Power loss due to friction } N_\mu = \frac{\mu W v}{1000} \text{ kW}$$

$$= \frac{(2.7956 \times 10^{-3})(6000)(4.712)}{1000} = 0.079 \text{ kW}$$

Example : 7.8

A turbine shaft 60 mm in diameter rotates at a speed of 10000 rpm. The load on each bearing is estimated at 2 kN and the length of bearing is 80 mm. Taking radial clearance as 0.05 mm and SAE-20 oil for lubrication determine the coefficient of friction, power loss, minimum film thickness and the oil flow rate. The temperature of the bearing is not to exceed 50° C. **VTU, July/August 2002**

Data :

$$d = 60 \text{ mm} = 0.06 \text{ m}; \; n = 10000 \text{ rpm}; \; W = 2 \text{ kN} = 2000 \text{ N}; \; L = 80 \text{ mm} = 0.08 \text{ m};$$

$$c_r = 0.05 \text{ mm} = 0.05 \times 10^{-3} \text{ m}; \; \text{Oil-SAE 20}; \; t_b = 50° \text{ C}$$

Solution :

Diametral clearance $c = 2c_r = 2 \times 0.05 \times 10^{-3} = 0.1 \times 10^{-3}$ m

Diametral clearance ratio $\psi = \frac{c}{d} = \frac{0.1 \times 10^{-3}}{0.06} = 1.667 \times 10^{-3}$

Assume ambient temperature $t_a = 25°$ C

Now $t_b - t_a \approx \frac{10000}{60}$

i.e., $50 - 25 \approx \frac{t_o - 25}{2}$

$\therefore$ Operating temperature $t_o = 75°$ C

Speed in rps $n' = \frac{n}{60} = \frac{10000}{60} = 166.667.$

From Table 24.1 (DDHB) for SAE 20 oil, select oil No : E

From Fig 24.2 (DDHB) for oil-B and $t_o = 75°$ C

Absolute viscosity $\eta = 14$ cP $= 14 \times 10^{-3}$ Pas

[Figure 24.2b (DDHB) can also be used to find η]

Bearing pressure $P = \frac{W}{Ld} = \frac{2000}{(0.08)(0.06)} = 0.4167 \times 10^6$ N/m^2

i) Coefficient of friction

Coefficient friction according to Mckee's equation

$$\mu = K_a\left(\frac{\eta n'}{P}\right)\frac{1}{\psi} \times 10^{-10} + \Delta\mu \qquad \text{---- 24.22}$$

Assume full journal $\therefore$ $K_a = 1.95 \times 10^{11}$; $\Delta\mu = 0.002$

$$\therefore \mu = (1.95 \times 10^{11})\left(\frac{14 \times 10^{-3} \times 166.667}{0.4167 \times 10^6}\right)\left(\frac{1}{1.667 \times 10^{-3}}\right)(10^{-10}) + 0.002$$

$$= 0.0675$$

ii) Power loss

Power loss due to friction $N_\mu = \left(\frac{\mu W v}{1000}\right)$ kW

$$= \frac{(0.0675)(2000)\left(\frac{\pi \times 0.06 \times 10000}{60}\right)}{1000} = 4.242 \text{ kW}$$

iii) Minimum film thickness (considering side leakage)

Sommerfeld number $S = \frac{(\eta n')}{P}\frac{1}{\psi} = \left(\frac{14 \times 10^{-3} \times 166.667}{0.4167 \times 10^6}\right)\left(\frac{1}{1.667 \times 10^{-3}}\right)^2 = 2.015$

Length in the direction of motion $B = \pi d \times \frac{\beta}{360} = \pi \times 0.06 \times \frac{360°}{360} = \pi \times 0.06$

$$\therefore \frac{\text{Length in the direction of motion}}{\text{Length in direction perpendicular to motion}} = \frac{B}{L} = \frac{\pi \times 0.06}{0.08} = 2.3562$$

From Fig 24.24 (DDHB) for $\frac{B}{L} = 2.3562$, Load leakage correction factor $C_w = 0.175$

From Fig 24.26 (DDHB) for $\frac{B}{L} = 2.3562$ and $C_w = 0.175$

Minimum film thickness variable $\delta = 0.6$

$$\text{Also } \delta = \frac{2h_{min}}{c} \qquad \text{---- 24.33}$$

$$\text{i.e., } 0.6 = \frac{2h_{min}}{0.1 \times 10^{-3}}$$

$\therefore$ Minimum film thickness $h_{min} = 3 \times 10^{-5}$ m

iv) Oil flow rate considering side leakage

Bearing characteristic number $S = \frac{\eta n'}{P} \cdot \frac{1}{\psi^2} = 2.015$

From Fig 24.37 (DDHB) for $S = 2.015, \beta = 360°$ and $\frac{L}{d} = \frac{0.08}{0.06} = 1.33$

Flow variable $\lambda_Q = 3.25$

From Fig 24.38 (DDHB) for $\frac{B}{L} = 2.3562$ and $\delta = 0.6$

Flow correction factor $C_Q = 1.45$

$\therefore$ Oil flow through a bearing with side leakage

$$Q = \frac{\lambda_Q c^2 n' L}{4\psi} C_Q$$

$$= \frac{3.25 \times (0.1 \times 10^{-3})^2 \times 166.667 \times 0.08}{4 \times 1.667 \times 10^{-3}} \times 1.45$$

$$= 9.423 \times 10^{-5} \text{ m}^3\text{/sec.}$$

Example : 7.9

A full journal bearing having diameter of 50 mm and 100 mm long has a bearing pressure of 1.2 N/mm². The speed of the journal is 1000 rpm. The bearing is lubricated at 75° C [bearing surface temperature] having viscosity of 0.011 Pas. The room temperature is 35° C. Take the minimum film thickness as $\frac{1}{4}$ of diametral clearance. The specific heat of oil is 1850 J/kg° C. The ratio of journal diameter to diametral clearance is 1000. Calculate

(i) Load which can be supported by bearing

(ii) Power lost due to friction

(iii) Attitude of bearing and eccentricity

(iv) The amount of artificial cooling required. **VTU, August 2001**

Data :

$$d = 50\text{ mm} = 0.05\text{ m}; L = 100\text{ mm} = 0.10\text{m}; P = 1.2\text{ N/mm}^2 = 1.2 \times 10^6\text{ N/m}^2$$

$$n = 1000\text{ rpm}; t_o = 75°C; \eta = 0.011\text{ Pas}; t_a = 35°C$$

$$h_{min} = \frac{c}{4}; C_p = 1850\text{ J/kg}°C = 188.6\text{ J/N}°C; \frac{d}{c} = 1000$$

Solution :

$$\text{Diametral clearance ratio } \psi = \frac{c}{d} = \frac{1}{1000} = 10^{-3}$$

$$\text{Clearance } c = 10^{-3}d = 10^{-3} \times 0.05 = 5 \times 10^{-5}\text{ m}$$

$$\text{Minimum film thickness } h_{min} = \frac{c}{4} = \frac{5\times10^{-5}}{4} = 1.25 \times 10^{-5}\text{ m}$$

i. Load which can be supported by bearing

$$\text{Bearing pressure } P = \frac{W}{Ld}$$

$$\text{i.e., } 1.2 \times 10^6 = \frac{W}{(0.1)(0.05)}$$

$$\therefore \text{ Radial Load on the bearing } W = 6000\text{ N}$$

ii. Power Lost due to friction

$$\text{Velocity } v = \frac{\pi dn}{60} = \frac{\pi\times0.05\times1000}{60} = 2.618\text{ m/sec}$$

Coefficient of friction according to Mckee's equation

$$\mu = K_a\left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi}\right)\times 10^{-10} + \Delta\mu \qquad \text{----24.22}$$

$$\text{For full journal } K_a = 1.95\times10^{11}; \Delta\mu = 0.002$$

$$\therefore \mu = (1.95 \times 10^{11})\left(\frac{0.011\times\frac{1000}{60}}{1.2\times10^6}\right)\left(\frac{1}{10^{-3}}\right)(10^{-10}) + 0.002$$

$$= 4.98 \times 10^{-3}$$

$$\therefore \text{ Power lost due to friction } N_\mu = \mu Wv \text{ Watts} = (4.98 \times 10^{-3})(6000)(2.618) = 78.2\text{ Watts}$$

iii. Attitude of bearing and eccentricity

$$\text{Minimum film thickness variable } \delta = \frac{2h_{min}}{c} = 1 - \varepsilon \qquad \text{---- 24.33}$$

$$\therefore \delta = \frac{2\times1.25\times10^{-5}}{5\times10^{-5}} = 0.5$$

$$\text{Now, } 0.5 = 1 - \varepsilon$$

$$\therefore \text{ Attitude } \varepsilon = 0.5$$

$$\text{Also } \varepsilon = \frac{2e}{c}$$

i.e., $0.5 = \frac{2 \times e}{5 \times 10^{-5}}$; $\therefore$ Eccentricity $e = 1.25 \times 10^{-5}$ m

iv. Amount of artificial cooling required

$$\text{Sommerfeld Number } S = \frac{\eta n'}{P} \cdot \frac{1}{\psi^2} \qquad \text{---- 24.39}$$

$$= \left(\frac{0.011 \times \frac{1000}{60}}{1.2 \times 10^6} \right) \left(\frac{1}{10^{-3}} \right)^2 = 0.1528$$

From Fig. 24.37 (DDHB) for $S = 0.1528$, $\beta = 360°$ and $\frac{L}{d} = \infty$

Flow variable $\lambda_Q = 2.9$

$$\text{Length in the direction of motion } B = \pi d \times \frac{\beta}{360} \qquad \text{---- 24.19}$$

$$= \pi \times 0.05 \times \frac{360}{360} = \pi \times 0.05$$

$$\therefore \frac{\text{Length in the direction of motion}}{\text{Length in direction perpendicular to motion}} = \frac{B}{L} = \frac{\pi \times 0.05}{0.1} = 1.57$$

From Fig. 24.38 (DDHB) for $\frac{B}{L} = 1.57$ and $\delta = 0.5$

Flow correction factor $C_Q = 1.5$

$\therefore$ Oil flow through the bearing with side leakage

$$Q = \frac{\lambda_Q c^2 n^1 L}{4\psi} . C_Q \qquad \text{---- 24.71b}$$

$$= \frac{2.9 \times (5 \times 10^{-5})^2 \left(\frac{1000}{60} \right) (0.1)(1.5)}{4 \times 10^{-3}}$$

$= 4.53125 \times 10^{-6}$ m³/sec

Temperature rise $\Delta T = t_h - t_a = 75 - 35 = 40°C$

Heat generated $H_g = \gamma C_{SP} Q \Delta T$ 24.72c

where γ = Weight per unit volume of lubricant

$= 8.83$ kN/m³ $= 8830$ N/m³

$\therefore H_g = (8830)(188.6)(4.53125 \times 10^{-6})(40)$

$= 301.84$ Watts

According to Lasche's equation, Heat dissipated $H_d = \frac{(\Delta T + 18)^2 (LD)}{K'}$

$$\text{where } K' = 0.475 \text{ for bearings in still air (assume)}$$

$$\therefore H_d = \frac{(40+18)^2(0.1\times0.05)}{0.475} = 35.41 \text{ Watts}$$

Since $H_d < H_g$, artificial cooling is necessary.

$$\text{Amount of heat to be removed} = H_g - H_d = 301.84 - 35.41 = 266.43 \text{ Watts}$$

Example : 7.10

A Full journal bearing of 50 mm diameter, 75 mm long supports a radial load of 1000 N. The speed of the shaft is 600 rpm. The surface temperature of bearing is limited to 60°C and the room temperature is 30°C. Determine the viscosity of the oil if the bearing is well ventilated and no artificial cooling is to be used. The ratio of journal diameter to diametral clearance is 1000

Data :

$$\mathbf{d = 50\ mm = 0.05\ m;\ L = 75\ mm = 0.075m;\ W = 1000\ N}$$

$$\mathbf{n = 1000\ rpm;\ t_b = 60°C;\ t_a = 30°C;\ \frac{d}{c} = 1000}$$

Solution :

$$\text{Clearance } c = \frac{d}{1000} = \frac{0.05}{1000} = 5\times10^{-5}\text{ m}$$

$$\text{Diametral clearance ratio } \psi = \frac{c}{d} = \frac{1}{1000} = 10^{-3}$$

$$\text{Rise in temperature } \Delta T = t_b - t_a = 60 - 30 = 30°C$$

$$\text{Speed in rps } n' = \frac{n}{60} = \frac{600}{60} = 10$$

$$\text{Bearing pressure } P = \frac{W}{Ld} = \frac{1000}{0.05\times0.075} = 0.267\times10^6\text{ N/m}^2$$

Coefficient of friction according to Mckee's equation

$$\mu = K_a\left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi}\right)\times10^{-10} + \Delta\mu \qquad \text{---- 24.22}$$

$$\text{For full journal } K_a = 1.95\times10^{11}\text{ ; } \Delta\mu = 0.002$$

$$\therefore \mu = (1.95\times10^{11})\left(\frac{\eta\times10}{0.267\times10^6}\right)\left(\frac{1}{10^{-3}}\right)(10^{-10}) + 0.002$$

$$= 0.73125\eta + 0.002$$

$$\therefore \text{ Heat generated } H_g = \mu\text{ (PLD) v} \qquad \text{---- 24.72a}$$

$$= \mu W v = (0.73125\eta + 0.002)\ 1000\times\frac{\pi\times0.05\times1000}{60}$$

$$= 1914.408\eta + 5.236$$

According to Lasche's equation

$$\text{Heat dissipated } H_d = \frac{(\Delta T+18)^2 LD}{K'}$$

where K' = 0.273 for well ventilated

$$\therefore\ H_d = \frac{(30+18)^2(0.075)(0.05)}{0.273} = 31.6484 \text{ Watts.}$$

Since artificial cooling is not required atleast $H_g = H_d$

i.e., $1914.408\eta + 5.236 = 31.6484$

$\therefore$ Absolute viscosity $\eta = 13.8 \times 10^{-3}$ Pas.

Example : 7.11

A 75 mm diameter full journal bearing supports a radial load of 3500 N. The bearing is 75 mm long and the shaft operates at 400 rpm. Assume a permissible minimum film thickness of 0.02 mm and a normal running fit for the bearing bore. Using Raimondi and Boyd curves determine (i) Absolute viscosity of the oil, (ii) Coefficient of friction, (iii) Heat generated, (iv) Amount of oil pumped through bearing, (iv) Amount of end leakage (vi) Temperature rise of the oil flowing through the bearing

Data :

d = 75 mm = 0.075 m; W = 3500 N; L = 75 mm = 0.075 m ; n = 400 rpm;

h_{min} = 0.02 mm = 2×10^{-5} m, Normal running fit

Solution :

From Table 11.5 (DDHB. Vol. I) for normal running fit, combination of shaft and hole is H_7f_7 (select normal)

From Table 11.12 (Vol. I) (DDHB) for f_7

Shaft size = 75^{-30}_{-60}

$\therefore$ Maximum size of shaft = 74.97 mm

Minimum size of shaft = 74.94 mm

From Table 11.13 (DDHB. Vol I) for H_7

Hole size = 75^{+30}_{+0}

$\therefore$ Maximum size of hole = 75.03 mm

Minimum size of shaft = 75 mm

$\therefore$ Maximum clearance c = 75.03 – 74.94 = 0.09 mm = 9×10^{-5} m

i. Absolute viscosity of oil

$$\text{Maximum film thickness variable } \delta = \frac{2h_{min}}{c} \quad \text{---- 24.33}$$

$$= \frac{2 \times 2 \times 10^{-5}}{9 \times 10^{-5}} = 0.444$$

From Fig. 24.13a (DDHB) for $\delta = 0.444$ and $\frac{L}{D} = \frac{0.075}{0.075} = 1$

Sommerfeld number S = 0.143

$$\text{Bearing pressure } P = \frac{W}{Ld} = \frac{3500}{(0.075)(0.075)} = 0.622 \times 10^6 \text{ N/m}^2$$

$$\text{Diametral clearance ratio } \psi = \frac{c}{d} = \frac{9\times10^{-5}}{0.075} = 1.2\times10^{-3}$$

Now Bearing characteristic number or Sommerfeld number

$$S = \frac{\eta n'}{P}\left(\frac{1}{\psi^2}\right)$$

$$\text{i.e., } 0.143 = \left(\frac{\eta\times\frac{400}{60}}{0.622\times10^6}\right)\left(\frac{1}{1.2\times10^{-3}}\right)^2$$

$$\therefore \text{ Absolute viscosity } \eta = 19.2\times10^{-3} \text{ Pas.}$$

ii. Coefficient of friction

$$\text{Length in the direction of motion B} = \pi d\times\frac{\beta}{360} \quad \text{----24.19}$$

$$= \pi\times0.075\times\frac{360}{360} = \pi\times0.075$$

$$\therefore \frac{\text{Length in the direction of motion}}{\text{Length in direction perpendicular to motion}} = \frac{B}{L} = \frac{\pi\times0.075}{0.075} = 3.14$$

From Fig. 24.19 (DDHB) for $\frac{B}{L} = 3.14$, $S = 0.143$ and Full journal i.e., $\beta = 360°$

$$\text{Coefficient of friction variable } \frac{\mu}{\psi} = 5.2$$

$$\therefore \text{ Coefficient of friction } \mu = 5.2\,\psi = 5.2\times1.2\times10^{-3} = 6.24\times10^{-3}$$

iii. Heat generated

$$\text{Heat generated } H_g = \mu\,(PLd)\,v \quad \text{---- 24.72a}$$

$$= (6.24\times10^{-3})(0.622\times10^6\times0.075\times0.075)\left(\frac{\pi\times0.075\times400}{60}\right)$$

$$= 34.31 \text{ Watts}$$

iv. Amount of oil pumped through the bearing

From Fig. 24.37 (DDHB) for $S = 0.143$, $\beta = 360°$ and $\frac{L}{d} = 1$

$$\text{Flow variable } \lambda_Q = 4.223$$

From Fig. 24.38for $\frac{B}{L} = 3.14$ and $\delta = 0.444$

$$\text{Flow correction factor } C_Q = 1.85$$

$\therefore$ Oil flow through the bearing with side leakage

$$Q = \frac{\lambda_Q c^2 n' L}{4\psi} C_Q = \frac{4.223(9\times10^{-5})^2\left(\frac{400}{60}\right)(0.075)(1.85)}{4\times1.2\times10^{-3}}$$

$$= 6.592 \times 10^{-6} \text{ m}^3/\text{sec.}$$

v. Amount of end leakage

From Fig. 24.39 (DDHB) for S = 0.143 and $\frac{L}{d} = 1$

$$\text{Oil flow ratio } \frac{Q_S}{Q} = 0.63$$

$$\therefore Q_S = 0.63\,Q = 0.63 \times 6.592 \times 10^{-6} = 4.153 \times 10^{-6} = \text{Side flow of lubricant}$$

vi. Temperature rise of the oil

From Fig. 24.40 (DDHB) for S = 0.143, $\beta = 360°$ and $\frac{L}{d} = 1$

Temperature rise of the oil film variable $\lambda T = 16$

$$\text{Now } \lambda T = \frac{\gamma C_{SP} \Delta T}{P} \text{ where}$$

γ = Weight per unit volume of the lubricant = 8.83 kN/m^3 = 8830 N/m^3

C_{SP} = Specific heat of the lubricant = 0.19 kJ/NK = 190 J/NK

P = Pressure and ΔT = temprise in °K or °C

$$\therefore 16 = \frac{8830\times190\times\Delta T}{0.622\times10^6}$$

$\therefore$ Temperature rise ΔT = 5.932° K or °C

Example : 7.12

Determine the dimensions of bearing and journal to support a load of 1000N at 450 rpm using hardened steel journal and bronze backed babbitt bearing. The oil used has a specific gravity of 0.9 at 15.5°C and a viscosity of 9 centistoke at 82°C which may be taken as the limiting temperature for the oil. Allows a clearance of 0.03 mm per cm diameter. Also find the rate of heat generated.

Data :

W = 1000 N; n = 450 rpm; $\gamma_{15.5} = 0.9$

Viscosity = 9 centistroke at 82°C; t_o = 82°C

$$c = \frac{0.03\text{mm}}{\text{cm diameter}} = 0.003 \text{ mm per mm diameter}$$

Journal material - Hardened steel

Bearing material - Bronze backed babbitt

Solution :

$$\text{Diametral clearance ratio } \psi = \frac{c}{d} = 0.003$$

Specific weight at the operating temperature

$$\gamma_t = \gamma_{15.5} - 0.000637\,(t - 15.5) \qquad \text{----24.10a}$$

$$= 0.9 - 0.000637\,[82 - 15.5] = 0.8576395$$

From Fig. 24.3 (DDHB) for kinematic viscosity = 9 centistroke and $t_o = 82°C$

The selected oil is B and the corresponding Saybolt seconds = 55

$$\therefore \text{ Saybolt centipoise } \eta_1 = \gamma_t\left(0.22t - \frac{180}{t}\right) \qquad \text{---- 24.7}$$

$$= 0.8576395\left[0.22 \times 55 - \frac{180}{55}\right] = 7.57$$

$$\therefore \text{ Absolute viscosity } \eta = 7.57 \times 10^{-3} \text{ Pas}$$

Or

$$\text{Kinematic viscosity } \upsilon = 10^{-6}\,\upsilon_K \text{ where } \upsilon_K \text{ in centistroke} \qquad \text{---- 24.8b}$$

$$\therefore \upsilon = 9 \times 10^{-6}$$

$$\text{Also kinematic viscosity } \upsilon = \frac{\eta g}{\gamma} \times 10^{-4} \qquad \text{---- 246b}$$

$$\text{i.e., } 9 \times 10^{-6} = \frac{\eta \times 9.8066}{0.8576395} \times 10^{-4}$$

$$\therefore \text{ Absolute viscosity } \eta = 7.83 \times 10^{-3} \text{ Pas}$$

OR

From Fig. 24.2 (DDHB) for B-oil and $t_o = 82°C$

$$\text{Absolute viscosity } \eta = 7.6\,cP = 7.6 \times 10^{-3} \text{ Pas}$$

$$\text{Take } \eta = 7.83 \times 10^{-3} \text{ Pas}$$

From Table 24.7 (DDHB) for Hardened steel shaft and Babbitt bearing, bearing modulus

$$S'' = \frac{\eta n'}{P} = 48.5 \times 10^{-9}$$

i. Dimensions of bearing and journal

For thick film lubrication $S''_{Cal} \geq S''_{recom}$

$$\text{i.e., } \frac{\eta n'}{P} \geq 48.5 \times 10^{-9}$$

$$\therefore \frac{7.83 \times 10^{-3} \times \frac{450}{60}}{P} \geq 48.5 \times 10^{-9}$$

$$\therefore \frac{0.058725}{P} \geq 48.5 \times 10^{-9}$$

$$\text{i.e., } 1.2108 \times 10^{6} \geq P$$

$$\text{i.e., } 1.2108 \times 10^{6} \geq \frac{W}{Ld}$$

$$\text{i.e., } 1.2108 \times 10^6 \geq \frac{1000}{d^2} \text{ [Assume L = d]}$$

$$\therefore d \geq 0.02874 \text{ m.}$$

Hence take diameter of journal d = 0.03m = 30 mm

∴ Length of bearing L = 0.03m = 30 mm

Diameter of bearing D = d + c = 30 + 0.003 × 30 = 30.09 mm

ii. Heat generated

$$\text{Heat generated } H_g = \mu (PLd) v \quad \text{---- 24.72a}$$

$$= \mu W v$$

$$\text{According to Mckee's equation } \mu = K_a \left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi}\right) 10^{-10} + \Delta\mu \quad \text{---- 24.22}$$

Assume full journal ∴ $K_a = 1.95 \times 10^{11}$; $\Delta\mu = 0.002$

$$\text{Bearing pressure } P = \frac{W}{Ld} = \frac{1000}{(0.03)(0.03)} = 1.111 \times 10^6 \text{ N/m}^2$$

$$\therefore \mu = (1.95 \times 10^{11}) \left(\frac{7.83 \times 10^{-3} \times \frac{450}{60}}{1.111 \times 10^6}\right)\left(\frac{1}{0.003}\right)(10^{-10}) + 0.002$$

$$= 2.3435 \times 10^{-3}$$

$$\text{Velocity } v = \frac{\pi d n}{60} = \frac{\pi \times 0.03 \times 450}{60} = 0.70686$$

$$\therefore \text{Heat generated } H_g = (2.3435 \times 10^{-3})(1000)(0.70686)$$

$$= 1.6565 \text{ Watts.}$$

Example : 7.13

Determine the dimensions of bearing and journal to support a load of 7.5 kN at 1000 rpm. The journal is made of hardened steel and the bearing is of Babbitt material abundance of oil is supplied by oil rings. The oil viscosity is 300 Saybolt seconds at 40°C and specific gravity is 0.915 at 15.5°C. The operating temperature of the oil is 75°C allows a clearance of 0.001 mm per mm diameter. Also find the minimum film thickness and oil flow without side leakage.

Data :

W = 7500 N; n = 1000 rpm

Viscosity = 300 Saybolt seconds at 40°C

t_0 = 75° C; c = 0.001 mm per mm diameter

$\gamma_{15.5}$ = 0.915

Journal material - Hardened steel

Bearing material - Babbitt

Solution :

Diametral clearance ratio $\psi = \frac{c}{d} = 0.001$

Specific weight at the operating temperature

$$\gamma_t = \gamma_{15.5} - 0.000637\,(t - 15.5) \qquad \text{---- 24.10a}$$

$$= 0.915 - 0.000637\,[75 - 15.5] = 0.8770985$$

$$\text{Saybolt centipoise } \eta_1 = \gamma_t\left(0.22t - \frac{180}{t}\right) \text{ where t saybolt seconds at 75°C.} \qquad \text{---- 24.7}$$

From Fig. 24.3 (DDHB) for 300 saybolt seconds at 40°C, the oil selected is E. Now from the same figure for E-oil at 75°C, saybolt t = 80

$$\therefore\ \eta_1 = 0.8770985\left(0.22\times 80 - \frac{180}{80}\right) = 13.5 \text{ cP}$$

$\therefore$ Absolute viscosity $\eta = 13.5 \times 10^{-3}$ Pas

Or From Fig. 24.2 (DDHB) for E-oil and $t_o = 75°C$

Absolute viscosity $\eta = 13.5 \text{ cP} = 13.5 \times 10^{-3}$ Pas.

i. Dimensions of bearing and journal

From Table 24.7 (DDHB) for hardened steel shaft and Babbitt bearing, Bearing modulus $S'' = \frac{\eta n'}{P} = 48.5 \times 10^{-9}$

For thick film lubrication

$$S''_{Cal} \geq S''_{recom}$$

$$\text{i.e., } \frac{\eta n'}{P} \geq 48.5 \times 10^{-9}$$

$$\text{i.e., } \frac{13.5\times 10^{-3}\times\left(\frac{1000}{60}\right)}{P} \geq 48.5 \times 10^{-9}$$

$$\text{i.e., } 4.64 \times 10^{6} \geq P$$

$$\text{i.e., } 4.64 \times 10^{6} \geq \frac{W}{Ld}$$

$$\text{i.e., } 4.64 \times 10^{6} \geq \frac{7500}{d^2} \text{ [Assume L = d]}$$

$$\therefore\ d \geq 0.040 \text{ m.}$$

$\therefore$ Diameter of journal d = 0.04m = 40 mm

$\therefore$ Length of bearing L = 0.04m = 40 mm

Diameter of bearing D = $d + c = 40 + 0.001 \times 40 = 40.04$ mm

ii. Minimum film thickness

$$\text{Sommerfeld number } S = \frac{\eta n'}{P}\left(\frac{1}{\psi^2}\right)$$

$$\text{Bearing pressure } P = \frac{W}{Ld} = \frac{7500}{(0.04)(0.04)} = 4.6875 \times 10^6 \text{ N/m}^2$$

$$\therefore\ S = \left(\frac{13.5\times10^{-3}\times\frac{1000}{60}}{4.6875\times10^{6}}\right)\left(\frac{1}{0.001}\right)^2 = 0.048$$

From Fig. 24.13a (page 24.33) (DDHB) for S = 0.048, $\frac{L}{d} = \frac{0.04}{0.04} = 1$

$$\text{Minimum film thickness variable } \delta = 0.24$$

$$\text{Also } \delta = \frac{2h_{min}}{c}$$

$$\text{i.e., } 0.24 = \frac{2h_{min}}{0.001\times40}$$

$$\therefore \text{ Minimum film thickness } h_{min} = 4.8 \times 10^{-3} \text{ mm}$$

iii. Oil flow without side leakage

From Fig. 24.37 (DDHB) for S = 0.048, β = 360° and $\frac{L}{d} = 1$

$$\text{Flow variable } \lambda_Q = 4.56$$

∴ Oil flow through the bearing without side leakage

$$Q = \frac{\lambda_Q c^2 n' L}{4\psi} = \frac{(4.56)(4\times10^{-5})^2\left(\frac{1000}{60}\right)(0.04)}{4\times0.001}$$

$$= 1.216 \times 10^{-6} \text{ m}^3/\text{sec}$$

Example : 7.14

A Partial self contained 120° centrally **loaded bearing has the following specifications; journal diameter = 90 mm; Length of bearing = 125 mm; speed = 400 rpm; Viscosity of lubricant = 0.04 Pas. Assuming a clearance of 1.39 mm per meter of diameter, determine the following using graphs**

(i) Load carrying capacity of the bearing corresponding to a minimum oil film thickness of 0.00625 mm

(ii) Coefficient of friction

(iii) Maximum pressure in the oil film.

VTU, July/Aug 2003

Data :

β = **120°; d = 90 mm = 0.09 ; L = 125 mm = 0.125 m**

n = 400 rpm ; η **= 0.04 Pas**

$$c = 1.39 \text{ mm per meter diameter} = \frac{1.39}{1000} \times 90 = 0.125 \text{ mm} = 0.1251 \times 10^{-3} \text{ m}$$

$$h_{min} = 0.00625 \text{ mm} = 6.25 \times 10^{-6} \text{ m}$$

Solution :

$$\text{Minimum film thickness variable } \delta = \frac{2h_{min}}{c} = 1 - \varepsilon \qquad \text{---- 24.33}$$

$$\therefore \frac{2 \times 6.25 \times 10^{-6}}{0.1251 \times 10^{-3}} = 1 - \varepsilon$$

$$\therefore \text{Attitude } \varepsilon = 0.9$$

$$\text{Diametral clearance ratio } \psi = \frac{c}{d} = \frac{0.1251 \times 10^{-3}}{0.09} = 1.39 \times 10^{-3}$$

$$\text{Length of bearing in the direction of motion } B = \pi d \times \frac{\beta}{360} \qquad \text{---- 24.19}$$

$$= \pi \times 0.09 \times \frac{120}{360} = 0.09425$$

$$\therefore \frac{\text{Length in the direction of motion}}{\text{Length in direction perpendicular to motion}} = \frac{B}{L} = \frac{0.09425}{0.125} = 0.754$$

i. Load carrying capacity of the bearing

From Fig. 24.30 (DDHB) for $\varepsilon = 0.9$, $\beta = 120°$ and $\frac{B}{L} = 0.754$

$$\text{Sommerfeld number } S = 0.02$$

$$\text{Now, } S = \frac{\eta n'}{P} \cdot \frac{1}{\psi^2}$$

$$\text{i.e., } 0.02 = \frac{\left(0.04 \times \frac{400}{60}\right)}{P} \left(\frac{1}{1.39 \times 10^{-3}}\right)^2$$

$$\therefore \text{Bearing pressure } P = 6.9 \times 10^6 \text{ N/m}^2$$

$$\text{Also } P = \frac{W}{A} \text{ where } A = dL \sin\frac{\beta}{2}$$

$$\therefore 6.9 \times 10^6 = \frac{W}{(0.09)(0.125)\sin\left(\frac{120}{2}\right)}$$

$$\therefore \text{Load } W = 67225 \text{ N} = 67.225 \text{ kN}$$

ii. Coefficient of friction

From Fig. 24.17 (DDHB) for $S = 0.02$, $\frac{B}{L} = 0.754$ and $\beta = 120°$

$$\text{Coefficient of friction variable } \frac{\mu}{\psi} = 0.825$$

$$\therefore \text{Coefficient of friction } \mu = 0.825\,\psi = 0.825 \times 1.39 \times 10^{-3} = 1.14675 \times 10^{-3}$$

iii. Maximum pressure

From Fig. 24.27 (DDHB) for $\frac{B}{L}$ = 0.754 and $\varepsilon = 0.9$

Pressure ratio $\frac{P_{max}}{P} = 4.25$

$\therefore$ Maximum pressure $P_{max} = 4.25\ P = 4.25 \times 6.9 \times 10^6 = 29.325 \times 10^6\ N/m^2$

Example : 7.15

Design the main bearing for a stationary slow speed steam engine for the following data. Journal diameter = 200 mm; Maximum Load on the piston = 80 kN. Engine speed = 200 rpm

Data :

$$W = 80\ kN = 80 \times 10^3\ N;\ d = 200\ mm = 0.2\ m;\ n = 200\ rpm$$

Solution :

i. Select the recommended values for the stationary slow speed steam engine - main bearing from Table 24.2 (DDHB)

Maximum pressure $P_{max} = 2.75\ MPa = 2.75 \times 10^6\ N/m^2$

Diameter clearance ratio $\psi = \frac{c}{d} < 0.001$

Ratio $\frac{L}{d}$ = 1 to 2

Absolute viscosity $\eta = 60 \times 10^{-3}$ Pas

Bearing modulus $S'' = \frac{\eta n'}{P} = 48.4 \times 10^{-9}$

ii. Design

Select the allowable pressure $P = 2\ MPa = 2 \times 10^6\ N/m^2$

Also $P = \frac{W}{Ld}$

i.e., $2 \times 10^6 = \frac{80 \times 10^3}{L \times 0.2}$

$\therefore$ Length of bearing $L = 0.2\ m = 200\ mm$

Ratio $\frac{L}{d} = \frac{200}{200} = 1$. It is within the recommended value.

iii. Selection of oil

Assume the operating temperature $t_o = 65°C$

From Fig. 24.2 (DDHB) for $\eta = 60 \times 10^{-3}$ Pas and $t_o = 65°C$

The selected oil is H

Actual value of $\eta = 60 \times 10^{-3}$ Pas

From Table 24.1 (DDHB), H - oil is SAE 60

Check for the oil

$$\text{Bearing modulus } S''_{cal} = \frac{\eta n'}{P} = \frac{60\times10^{-3}\times\frac{200}{60}}{2\times10^{6}} = 100\times10^{-9}$$

Since $S''_{cal} > S''_{recom}$ (i.e., 48.4×10^{-9}), thick film lubrication is possible. Hence the selected oil is suitable.

iv. Heat generated

$$\text{Heat generated } H_g = \mu\,(PLd)\,v \qquad \text{---- 24.72a}$$
$$= \mu W v$$

$$\text{From Mckee's equation } \mu = K_a\left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi}\right)10^{-10} + \Delta\mu \qquad \text{---- 24.22}$$

Assume full journal $\therefore$ $K_a = 1.95 \times 10^{11}$, $\Delta\mu = 0.002$

$$\therefore \mu = (1.95\times10^{11})(100\times10^{-9})\left(\frac{1}{0.0009}\right)10^{-10} + 0.002 = 4.167\times10^{-3}$$

[take $\psi = 0.0009$ since $\psi < 0.001$]

$$\therefore H_g = (4.167\times10^{-3})(80\times10^{3})\left(\frac{\pi\times0.2\times200}{60}\right) = 698.132 \text{ Watts}$$

v. Heat dissipated

$$\text{Heat dissipated } H_d = CA\,(t_b - t_a) \qquad \text{---- 24.77}$$

where $C = 11.36 \times 10^{-3}$ kW/m²K $= 11.36$ W/m²K

$A = 25\,dL = 25 \times 0.2 \times 0.2 = 1$ m²

Assume ambient temperature $t_a = 25°C$

$$\therefore t_b - t_a \approx \frac{t_0 - t_a}{2}$$
$$\approx \frac{65-25}{2} = 20°C \text{ or } 20°K$$

$$\therefore H_d = (11.36)(1)(20) = 227.2 \text{ Watts}$$

$\therefore$ Amount of heat to be removed $= H_g - H_d = 698.132 - 227.2 = 470.932$ Watts

Hence artificial cooling is required.

vi. Minimum film thickness and its location

$$\text{Sommerfeld number } S = \left(\frac{\eta n'}{P}\right)\frac{1}{\psi^2} \qquad \text{---- 24.39}$$

$$= (100\times10^{-9})\left(\frac{1}{0.0009}\right)^2 = 0.1235$$

From Fig. 24.13a (page 24.33) (DDHB) for $S = 0.1235$ and $\frac{L}{d} = 1$

Minimum film thickness variable $\delta = 0.4$ for full journal

$$\text{Also } \delta = \frac{2h_{min}}{c} \quad \text{where } c = \text{diametral clearance} =$$

$$\psi d = 0.0009 \times 0.2 = 1.8 \times 10^{-4}$$

$$\text{i.e., } 0.4 = \frac{2h_{min}}{1.8 \times 10^{-4}}$$

∴ Minimum film thickness of oil $h_{min} = 3.6 \times 10^{-5}$ m

From Fig. 24.11a (Page 24.30) (DDHB) for S = 0.1235 and $\frac{L}{d} = 1$

Position of minimum film thickness $\phi = 52°$.

vii. Oil flow rate and end leakage

From Fig. 24.37 (DDHB) for S = 0.1235 and $\frac{L}{d} = 1$

Flow variable λ_Q = 4.3, for full journal bearing

$$\text{Length in the direction of motion } B = \frac{\pi d \times \beta°}{360} = \frac{\pi \times 0.2 \times 360}{360} = \pi \times 0.2$$

$$\therefore \frac{\text{Length in the direction of motion}}{\text{Length in direction perpendicular to motion}} = \frac{B}{L} = \frac{\pi \times 0.2}{0.2} = 3.14$$

From Fig. 24.38 (DDHB) for $\frac{B}{L}$ = 3.14 and $\delta = 0.4$

Flow correction factor C_Q = 2

∴ Considering side leakage, oil flow through bearing

$$Q = \frac{\lambda_Q c^2 n' L}{4\psi} . C_Q$$

$$= \frac{4.3 \times \left(1.8 \times 10^{-4}\right)^2 \left(\frac{200}{60}\right)(0.2)(2)}{4 \times 0.0009}$$

$$= 5.16 \times 10^{-5} \text{ m}^3\text{/sec}$$

From Fig. 24.39 (DDHB) for S = 0.1235 and $\frac{L}{d} = 1$

$$\text{Oil flow ratio } \frac{Q_s}{Q} = 0.66$$

∴ Amount of end leakage Q_s = $0.66Q = 0.66 \times 5.16 \times 10^{-5} = 3.4056 \times 10^{-5}$ m³/sec

viii. Power loss due to friction

Power loss due to friction N_μ = μWv = 698.132 Watts [Refer step 4]

Example : 7.16

Design the main bearing of a steam turbine that runs at 1800 rpm. The load on the bearings is estimated to be 2500 N.

Data :

Steam turbine main bearing, n = 1800 rpm, W = 2500 N

Solution :

i. Select the recommended values for the Steam turbine - Main bearing from Table 24.2 (DDHB)

Maximum pressure P = 0.69 to 1.87 MPa

Diametral clearance ratio ψ = 0.001

$$\text{Ratio } \frac{L}{d} = 1 \text{ to } 2$$

Absolute viscosity η = 2 to 16×10^{-3} Pas

Bearing modulus S" = 241.8×10^{-9}

ii. Design

Select the allowable pressure P = 1 MPa = 1×10^6 N/m^2

$$\frac{L}{d} = 1$$

$$\text{Also } P = \frac{W}{Ld}$$

$$\text{i.e., } 1 \times 10^6 = \frac{2500}{d^2} \quad (\because L = d)$$

$\therefore$ Diameter of journal d = 0.05 m = 50 mm

Length of bearing L = 0.05 m = 50 mm

$$P_{act} = \frac{2500}{(0.05)(0.05)} = 1 \times 10^6 \text{ N/m}^2$$

iii. Selection of oil

Assume the operating temperature t_o = 70°C and take $\eta = 16 \times 10^{-3}$ Pas

From Fig. 24.2 (DDHB) for η = 16×10^{-3} Pas and t_o = 70°C,

The selected oil is 'E'

Actual value of η = 16×10^{-3} Pas

From Table 24.1 (DDHB), E - oil is SAE 20

Check for the oil

$$\text{Bearing modulus } S''_{cal} = \frac{\eta n'}{P} = \left(\frac{16 \times 10^{-3} \times \frac{1800}{60}}{1 \times 10^6} \right) = 480 \times 10^{-9}$$

Since $S''_{cal} > S''_{recom}$ (i.e., 241.8×10^{-9}), thick film lubrication is possible. Hence the selected oil is suitable.

iv. Heat generated

$$\text{Heat generated } H_g = \mu\,(PLd)\,v \qquad \text{---- 24.72a}$$

$$\text{From Mckee's equation } \mu = K_a \left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi}\right) 10^{-10} + \Delta\mu \qquad \text{---- 24.22}$$

Assume full journal $\therefore$ K_a = 1.95×10^{11}, $\Delta\mu = 0.002$

$$\therefore \mu = (1.95 \times 10^{11})(480 \times 10^{-9})\left(\frac{1}{0.001}\right) 10^{-10} + 0.002 = 0.01136$$

$$\therefore H_g = (0.01136)(2500)\left(\frac{\pi \times 0.05 \times 1800}{60}\right) = 133.832 \text{ Watts}$$

v. Heat dissipated

$$\text{Heat dissipated } H_d = CA(t_b - t_a) \quad \text{---- 24.77}$$

$$\text{where } C = 11.36 \times 10^{-3} \text{ kW/m}^2\text{K} = 11.36 \text{ W/m}^2\text{K}$$

$$A = 25\, dL = 25 \times 0.05 \times 0.05 = 0.0625 \text{ m}^2$$

Assume ambient temperature t_a = 25°C

$$\therefore t_b - t_a \approx \frac{t_o - t_a}{2}$$

$$\approx \frac{70-25}{2} \approx 22.5°\text{C or } 22.5°\text{K}$$

$$\therefore H_d = (11.36)(0.0625)(22.5) = 15.975 \text{ Watts}$$

$\therefore$ Amount of heat to be removed = $H_g - H_d$ = 133.832 – 15.975 = 117.857 Watts

Hence artificial cooling is required.

vi. Minimum film thickness and its location

$$\text{Sommerfeld number } S = \left(\frac{\eta n'}{P}\right)\frac{1}{\psi^2} \quad \text{---- 24.39}$$

$$= (480 \times 10^{-9})\left(\frac{1}{0.001}\right)^2 = 0.48$$

From Fig. 24.13a (page 24.33) (DDHB) for S = 0.48 and $\frac{L}{d} = 1$

Minimum film thickness variable δ = 0.74, for full journal bearing

$$\text{Also } \delta = \frac{2h_{min}}{c} \text{ where } c = \psi d = 0.001 \times 0.05 = 5 \times 10^{-5}$$

$$\text{i.e., } 0.74 = 2 \times \frac{h_{min}}{5 \times 10^{-5}}$$

$\therefore$ Minimum film thickness $h_{min} = 1.85 \times 10^{-5}$ m

From Fig. 24.11a (Page 24.30) (DDHB) for S = 0.48 and $\frac{L}{d} = 1$

Position of minimum film thickness ϕ = 72°.

vii. Oil flow rate and end leakage

From Fig. 24.37 (DDHB) for S = 0.48 and $\frac{L}{d} = 1$

Flow variable λ_Q = 3.6, for full journal bearing

Length in the direction of motion B $= \frac{\pi d \times \beta^\circ}{360} = \frac{\pi \times 0.05 \times 360}{360} = \pi \times 0.05$

$\therefore \frac{\text{Length in the direction of motion}}{\text{Length in direction perpendicular to motion}} = \frac{B}{L} = \frac{\pi \times 0.05}{0.05} = 3.14$

From Fig. 24.38 (DDHB) for $\frac{B}{L}$ = 3.14 and $\delta = 0.74$

Flow correction factor C_Q = 1.25

$\therefore$ Considering side leakage, oil flow through bearing

$$Q = \frac{\lambda_Q c^2 n' L}{4\psi} . C_Q$$

$$= \frac{3.6 \times (5 \times 10^{-5})^2 \left(\frac{1800}{60}\right)(0.05)(1.25)}{4 \times 0.001}$$

$$= 4.22 \times 10^{-6} \text{ m}^3/\text{sec}$$

From Fig. 24.39 (DDHB) for S = 0.48 and $\frac{L}{d} = 1$

Oil flow ratio $\frac{Q_s}{Q}$ = 0.34

$\therefore$ Amount of end leakage Q_s = $0.34Q = 0.34 \times 4.22 \times 10^{-6} = 1.4348 \times 10^{-6}$ m³/sec

viii. Power loss due to friction

Power loss due to friction N_μ = $\mu W v$ = 133.832 Watts [Refer step 4]

Example : 7.17

A 50 mm diameter full journal bearing supports a load of 2500 N. The bearing is 50 mm long and the shaft operates at 300 rpm. The radial clearance is 0.05 mm. The bearing is lubricated with SAE 110 oil and the operating temperature is 80°C. Determine (i) Somerfeld number (ii) Cofficient of friction (iii) Minimum film thickness (iv) Attitude (v) Heat generated (vi) Flow of lubricant into the bearing (vii) amount of end leakage (viii) Maximum pressure on the bearing and (ix) Temperature rise of the lubricant. Using Raimondi and Boyd's curves

Data :

Full journal; d = 50 mm = 0.05m ; L = 50 mm = 0.05 m

W = 2500 N; n = 300 rpm; c_r = 0.05 mm = 0.05×10^{-3} m

Oil - SAE 110; t_o = 80°C

Solution :

Ratio $\frac{L}{d} = \frac{0.05}{0.05} = 1$

Bearing pressure $P = \frac{W}{Ld} = \frac{2500}{(0.05)(0.05)} = 1 \times 10^6$ N/m²

Speed in rps $n' = \frac{n}{60} = \frac{300}{60} = 5$

diametral clearance $c = 2c_r = 2 \times 0.05 \times 10^{-3} = 10^{-4}$ m

diametral clearance ratio $\psi = \frac{c}{d} = \frac{10^{-4}}{0.05} = 2 \times 10^{-3}$

From Table 24.1 (DDHB) for SAE 110, the oil number is I.

From Fig. 24.2 (DDHB) for I-oil and $t_o = 80°C$

Absolute viscosity $\eta = 55$ cP $= 55 \times 10^{-3}$ Pas

i. Sommerfeld Number

$$\text{Sommerfeld number } S = \frac{\eta n'}{P} \frac{1}{\psi^2} \qquad \text{---- 24.39}$$

$$= \left(\frac{55 \times 10^{-3} \times 5}{1 \times 10^{6}}\right)\left(\frac{1}{2 \times 10^{-3}}\right) = 0.06875$$

ii. Coefficient of friction

$$\text{Length in the direction of motion } B = \pi d \times \frac{\beta°}{360°}$$

$$= \frac{\pi \times 0.05}{360} \times 360 = \pi \times 0.05 \qquad \text{---- 24.39}$$

$$\therefore \frac{\text{Length in the direction of motion}}{\text{Length in direction perpendicular to motion}} = \frac{B}{L} = \frac{\pi \times 0.05}{0.05} = 3.14$$

From Fig. 24.19 (DDHB) for $S = 0.06875$, $\frac{B}{L} = 3.14$ and $\beta = 360°$

Coefficient of friction variable $\frac{\mu}{\psi} = 3.4$

$\therefore$ Coefficient of friction $\mu = 3.4\,\psi = 3.4 \times 2 \times 10^{-3} = 6.8 \times 10^{-3}$

iii. Minimum film thickness

From Fig. 24.13a (Page 24.33) (DDHB) for $S = 0.06875$ and $\frac{L}{d} = 1$

Minimum film thickness variable $\delta = 0.26$

Also $\delta = \frac{2h_{min}}{c}$

i.e., $0.26 = \frac{2 \times h_{min}}{10^{-4}}$

$\therefore$ Minimum film thickness $h_{min} = 1.3 \times 10^{-5}$ m

iv. Attitude

From Fig. 24.13a (DDHB) for $S = 0.06875$, $\frac{L}{d} = 1$ and full journal bearing

Attitude $\varepsilon = 0.74$

v. Heat generated

$$\text{Heat generated } H_g = \mu W v = (6.8 \times 10^{-3})\,(2500)\left(\frac{\pi \times 0.05 \times 300}{60}\right)$$

$$= 13.352 \text{ Watts.}$$

vi. Flow of lubricant into the bearing

$$\text{From Fig. 24.37 (DDHB) for } S = 0.06875 \text{ and } \frac{L}{d} = 1$$

$$\text{Flow variable } \lambda_Q = 4.45$$

$$\text{From Fig. 24.38 (DDHB) for } \frac{B}{L} = 3.14 \text{ and } \delta = 0.26$$

$$\text{Flow correction factor } C_Q = 3$$

Considering side leakage oil flow through bearing

$$Q = \frac{\lambda_Q c^2 n' L}{4\psi} . C_Q = \frac{(4.45)(10^{-4})^2 (5)(0.05)(3)}{4 \times 2 \times 10^{-3}}$$

$$= 4.172 \times 10^{-6} \text{ m}^3/\text{sec}$$

$$\text{Without side leakage } Q = \frac{\lambda_Q c^2 n' L}{4\psi} = 1.39 \times 10^{-6} \text{ m}^3/\text{sec}$$

vii. Amount of end leakage

$$\text{From Fig. 24.39 (DDHB) for } S = 0.06875 \text{ and } \frac{L}{d} = 1$$

$$\text{Oil flow ratio } \frac{Q_S}{Q} = 0.8$$

$$\therefore \text{ Amount of end leakage } Q_S = 0.8 \times 4.172 \times 10^{-6} = 3.3376 \times 10^{-6} \text{ m}^3/\text{sec}$$

$$\text{Amount of end leakage without sideflow } Q_S = 0.8 \times 1.39 \times 10^{-6} = 1.112 \times 10^{-6} \text{ m}^3/\text{sec}$$

viii. Maximum pressure on the bearing

$$\text{From Fig. 24.15a (page 24.18) (DDHB) for } S = 0.06875 \text{ and } \frac{L}{d} = 1$$

$$\frac{P}{P_{max}} = 0.35$$

$$\therefore \text{ Maximum pressure } P_{max} = \frac{P}{0.35} = \frac{1 \times 10^6}{0.35} = 2.86 \times 10^6 \text{ N/m}^2$$

ix. Temperature rise of the libricant

$$\text{From Fig. 24.40 (DDHB) for } S = 0.06875 \text{ and } \frac{L}{d} = 1 \text{ and } \beta = 360°$$

$$\text{Temperature rise of the lubricant film variable } \lambda_T = 10$$

$$\text{Also } \lambda_T = \frac{\gamma C_p \Delta T}{P}$$

Where γ = weight per unit volume of lubricant

= $8.83\ kN/m^3 = 8830\ N/m^3$

= Specific heat of lubricant

= $0.19\ kJ/NK = 190\ J/NK$

$$\therefore\ 10 = \frac{8830 \times 190 \times \Delta T}{1 \times 10^6}$$

$\therefore$ Temperature rise ΔT = 5.96 °C or 5.96° K

Example : 7.18

Solve, the Example 7.17 using Raimondi and Boyd's Table

Data :

Full journal; d = 50 mm = 0.05m ; L = 0.05 m

W = 2500 N; n = 300 rpm; c_r = 0.05 mm = 0.05 × 10^{-3} m

Oil - SAE 110; t_o = 80°C

Solution :

From Example 7.17 $\frac{L}{d} = 1$

Bearing pressure P = $1 \times 10^6\ N/m^2$

speed in rps n' = 5

diametral clearance c = 10^{-4} m

diametral clearance ratio ψ = 2×10^{-3} m

Oil No: I, Absolute viscosity η = 55×10^{-3} Pas

i. Sommerfeld Number

$$\text{Sommerfeld number } S = \frac{\eta n'}{P}\frac{1}{\psi^2} \quad \text{---- 24.39}$$

$$= \left(\frac{55 \times 10^{-3} \times 5}{1 \times 10^6}\right)\left(\frac{1}{2 \times 10^{-3}}\right) = 0.06875$$

ii. Coefficient of friction

From Table 24.11 (DDHB) Coefficient of friction variable $\frac{\mu}{\psi}$ for $\frac{L}{d} = 1$ and full journal bearing

By Lagrangian interpolation find the value of y for any intermediate value of x

$$\text{i.e., } y = \frac{x - x_2}{x_1 - x_2} y_1 + \frac{x - x_1}{x_2 - x_1} y_2$$

$\therefore$ The value of $\frac{\mu}{\psi}$ at S = 0.06875 is,

S	$\frac{\mu}{\psi}$
$x_1 = 0.121$	$y_1 = 3.22$
$x_2 = 0.0446$	$y_2 = 1.7$

$$\frac{\mu}{\psi} = \frac{0.06875-0.0446}{0.121-0.0446} \times 3.22 + \frac{0.06875-0.121}{0.0446-0.121} \times 1.7$$

$$= 0.3161 \times 3.22 + 0.6839 \times 1.7 = 2.18$$

∴ Coefficient of friction

$$\mu = 2.18\,\psi = 2.18 \times 2 \times 10^{-3} = 4.36 \times 10^{-3}$$

iii. Minimum film thickness

From Table. 24.11a (DDHB) Minimum film thickness variable δ

S	δ
$x_1 = 0.121$	$y_1 = 0.4$
$x_2 = 0.0446$	$y_2 = 0.2$

For $\frac{L}{d}$ = 1 and full journal bearing

By Lagrangian interpolation, the value of δ at

S = 0.06875 is

$$\delta = \frac{0.06875-0.0446}{0.121-0.0446} \times 0.4 + \frac{0.06875-0.121}{0.0446-0.121} \times 0.2$$

$$= 0.26322$$

$$\text{Also } \delta = \frac{2 \times h_{min}}{c}; \text{ i.e., } 0.26322 = \frac{2 \times h_{min}}{10^{-4}}$$

∴ Minimum film thickness $h_{min} = 1.3161 \times 10^{-5}$ m

iv. Attitude

S	ε
$x_1 = 0.121$	$y_1 = 0.6$
$x_2 = 0.0446$	$y_2 = 0.8$

From Fig. 24.11 (DDHB) attitude for $\frac{L}{d}$ = 1 and full journal bearing

By Lagrangian interpolation, the value of ε at S = 0.06875 is

$$\varepsilon = \frac{0.06875-0.0446}{0.121-0.0446} \times 0.6 \; \frac{0.06875-0.121}{0.0446-0.121} \times 0.8$$

$$= 0.73678$$

v. Heat generated

$$\text{Heat generated } H_g = \mu W v = (4.36 \times 10^{-3})(2500)\left(\frac{\pi \times 0.05 \times 300}{60}\right)$$

$$= 8.561 \text{ Watts.}$$

vi. Flow of lubricant into the bearing

From Table 24.11 (DDHB) flow variable

S	$\frac{4Q}{\psi d^2 n' L}$
$x_1 = 0.121$	$y_1 = 4.33$
$x_2 = 0.0446$	$y_2 = 4.62$

$\frac{L}{d}$ = 1 and full journal

By Lagrangian interpolation, the value of flow variable at

S = 0.06875 is,

$$\frac{4Q}{\psi d^2 n' L} = \frac{0.06875-0.0446}{0.121-0.0446} \times 4.33 + \frac{0.06875-0.121}{0.0446-0.121} \times 4.62 = 4.544963$$

$$\text{i.e., } \frac{4Q}{(2\times10^{-3})(0.05)^2(5)(0.05)} = 4.544963$$

∴ Oil flow through bearing Q = 1.42×10^{-6} m^3/sec.

vii. Amount of end leakage

From Fig. 24.11 (DDHB) Flow ratio

$\frac{Q_S}{Q}$ for $\frac{L}{d}$ = 1 and full journal

S	$\frac{Q_S}{Q}$
$x_1 = 0.121$	$y_1 = 0.68$
$x_2 = 0.0446$	$y_2 = 0.842$

By Lagrangian interpolation, the value of flow ratio $\frac{Q_S}{Q}$ at

S = 0.06875 is,

$$\frac{Q_S}{Q} = \frac{0.06875-0.0446}{0.121-0.0446}\times0.68+\frac{0.06875-0.121}{0.0446-0.121}\times0.842=0.7907918$$

∴ Amount of end leakage

$$Q_S = 0.7907918\times1.42\times10^{-6}=1.123\times10^{-6}\text{ m}^3\text{/sec}$$

viii. Maximum pressure on the bearing

S	$\frac{P}{P_{max}}$
$x_1 = 0.121$	$y_1 = 0.415$
$x_2 = 0.0446$	$y_2 = 0.313$

From Table 24.11 (DDHB) pressure ratio $\frac{P}{P_{max}}$ for $\frac{L}{d}$ = 1 and full journal

By Lagrangian interpolation, the value of pressure ratio $\frac{P}{P_{max}}$ at

S = 0.06875 is,

$$\frac{P}{P_{max}} = \frac{0.06875-0.0446}{0.121-0.0446}\times0.415+\frac{0.06875-0.121}{0.0446-0.121}\times0.313=0.3452422$$

∴ Maximum pressure

$$P_{max} = \frac{P}{0.3452422} = \frac{1\times10^6}{0.3452422} = 2.8965\times10^6\text{ N/m}^2$$

ix. Temperature rise of the libricant

From Table 24.11 (DDHB) temperature rise of the lubricant film variable $\frac{\gamma C_{SP}\Delta T}{P}$ for

$\frac{L}{d}$ = 1 and full journal bearing

S	$\frac{\gamma C_{SP}\Delta T}{P}$
$x_1 = 0.121$	$y_1 = 14.2$
$x_2 = 0.0446$	$y_2 = 8$

By Lagrangian interpolation the value of temperature rise of the lubricant film variable at

S = 0.06875 is,

$$\frac{\gamma C_{SP}\Delta T}{P} = \frac{0.06875-0.0446}{0.121-0.0446}\times14.2+\frac{0.06875-0.121}{0.0446-0.121}\times8=0.95982$$

i.e., $\frac{8830 \times 190 \times \Delta T}{1 \times 10^6} = 9.95982$

[For the values of γ and C_{sp} Refer page 24.37 (DDHB) if not given]

$\therefore$ Temperature rise ΔT = 5.9366 °C or 5.9366° K

Example : 7.19

A partial self contained 120° centrally loaded bearing has the following specifications :

Diameter of journal = 50 mm

Length of bearing = 150 mm

Diametral clearance = 0.125 mm

Speed = 400 rpm

Minimum film thickness = 0.005 mm

Determine (i) Sommerfeld number (ii) Coefficient of friction, by using Raimondi and Boyd's table.

Data :

$$\beta = 120°;\ d = 50 \text{ mm} = 0.05 \text{ m};\ L = 150 \text{ mm} = 0.15 \text{ m};\ c = 0.125 \text{ mm} = 0.125 \times 10^{-3} \text{ m};$$

$$n = 400 \text{ rpm};\ h_{min} = 0.005 \text{ mm} = 0.005 \times 10^{-3} = 5 \times 10^{-6} \text{ m}$$

Solution :

$$\text{Ratio } \frac{L}{d} = \frac{150}{50} = 3$$

Minimum film thickness variable $\delta = \frac{2h_{min}}{c} = \frac{2 \times 5 \times 10^{-6}}{0.125 \times 10^{-3}} = 0.08$

Diametral clearance ratio $\psi = \frac{c}{d} = \frac{0.125 \times 10^{-3}}{0.05} = 2.5 \times 10^{-3}$

i) **Sommerfeld number**

From Table 24.13 (DDHB) Sommerfeld number for $\frac{L}{d} = \alpha$ and 120° journal bearing.

By Lagrangian interpolations the value of Sommerfeld number at $\delta = 0.08$ is,

δ	S
$x_1 = 0.1$	$y_1 = 0.0147$
$x_2 = 0.03$	$y_2 = 0.00406$

$$S = \frac{0.08 - 0.03}{0.1 - 0.03} \times 0.0147 + \frac{0.08 - 0.1}{0.03 - 0.1} \times 0.00406$$

$$= 0.714286 \times 0.0147 + 0.285714 \times 0.00406 = 0.01166$$

$\therefore$ For $\delta = 0.08$, Sommerfeld number $S = 0.01166$

ii) Coefficient of friction

From table 24.13 coefficient of friction variable $\frac{\mu}{\psi}$

for $\frac{L}{d} = \alpha$ and 120° journal bearing

δ	$\frac{\mu}{\psi}$
$x_1 = 0.1$	$y_1 = 0.653$
$x_2 = 0.03$	$y_2 = 0.399$

By Lagrangian interpolation the value of coefficient of friction variable $\frac{\mu}{\psi}$ at $\delta = 0.08$ is

$$\frac{\mu}{\psi} = \frac{0.08-0.03}{0.1-0.03} \times 0.653 + \frac{0.08-0.1}{0.03-0.1} \times 0.399 = 0.58$$

$\therefore$ Coefficient of friction $\mu = 0.58\,\psi = 0.58 \times 2.5 \times 10^{-3} = 1.45 \times 10^{-3}$

Note :

If $\frac{L}{d} > 1$ then use the values of $\frac{L}{d} = \alpha$

Example : 7.20

A 75 mm journal bearing of diameter 75 mm supports a load of 15 kN. The ratio of $\frac{d}{c} = 1000$ and the viscosity of oil is 25×10^{-3} Pas. The heat generated in the bearing is 442 Watts. Determine the maximum speed of the journal using Makee's equation.

Data :

$$L = 75 \text{ mm} = 0.075 \text{ m};\ d = 75 \text{ mm} = 0.075 \text{ m};\ W = 15 \text{ kN} = 15 \times 10^3 \text{ N};\ \frac{d}{c} = 1000;$$

$$\eta = 25 \times 10^{-3};\ H_g = 442 \text{ Watts.}$$

Solution :

Diametral clearance ratio $\psi = \frac{c}{d} = \frac{1}{1000} = 10^{-3}$

Heat generated $H_g = \mu\,(PLd)\,v = \mu\,Wv$ ---- 24.72 a

$$\text{i.e.,}\quad 442 = \mu \times 15 \times 10^3 \times \frac{\pi \times 0.075 \times n}{60}$$

$$\therefore\ \mu = \frac{7.5}{n}$$

From Mckee's equation $\mu = K_a \left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi}\right) 10^{-10} + \Delta\mu$ ---- 24.22

$$\text{Bearing pressure } P = \frac{W}{Ld} = \frac{15\times10^3}{0.075\times0.075} = 2.667\times10^6 \text{ N/mm}^2$$

Assume full journal $\therefore$ $K_a = 1.95\times10^{11}$

$$\Delta\mu = 0.002 \text{ for } \frac{L}{d} \text{ ranging from 0.75 to 2.8}$$

$$\therefore \frac{7.5}{n} = (1.95\times10^{11})\left(\frac{25\times10^{-3}\times\frac{n}{60}}{2.667\times10^6}\right)\left(\frac{1}{10^{-3}}\right)10^{-10} + 0.002$$

$$= 3.0465\times10^{-6}\,n + 0.002$$

$$\text{i.e., } 7.5 = 3.0465\times10^{-6}\,n^2 + 0.002\,n$$

$$\text{i.e., } n^2 + 656.5\,n - 2.462\times10^6 = 0$$

$$\therefore n = \frac{-656.5\pm\sqrt{656.5^2+4\times1\times2.462\times10^6}}{2}$$

$\therefore$ Speed $n = 1275$ rpm

Example : 7.21

A oil ring full journal bearing is to operate in still air. The bearing diameter is 75 mm and the length is 75 mm. The bearing is subjected to a load of 5 kN and is rotating at 500 rev/min, radial clearance is 0.0625 mm, the oil is SAE 30 and the ambient air temperature is 20° C. Determine the equilibrium temperature and the viscosity of the oil.

Data :

d = 75 mm = 0.075 m; L = 75 mm = 0.075 m; W = 5 kN = 5000 N; n = 500 rpm;

c_r = 0.0625 mm; oil-SAE 30; t_a = 20° C.

Solution :

Diametral clearance $c = 2c_r = 2\times0.0625 = 0.125 \text{ mm} = 1.25\times10^{-4}$ m

$$\text{Diametral clearance ratio } \psi = \frac{c}{d} = \frac{1.25\times10^{-4}}{0.075} = 1.667\times10^{-3}$$

$$\text{Velocity } v = \frac{\pi dn}{60} = \frac{\pi\times0.075\times500}{60} = 1.9635 \text{ m/sec}$$

$$\text{Bearing pressure } P = \frac{W}{Ld} = \frac{5000}{(0.075)(0.075)} = 0.889\times10^6 \text{ N/m}^2$$

From Mckee's equation coefficient of friction

$$\mu = K_a\left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi}\right)10^{-10} + \Delta\mu \qquad \text{---- 24.22}$$

For full journal $K_a = 1.95\times10^{11}$

$$\Delta\mu = 0.002 \text{ for } \frac{L}{d} \text{ ranging from 0.75 to 2.8}$$

$$\therefore \quad \mu = (1.95 \times 10^{11}) \left(\frac{\eta \times \frac{500}{60}}{0.889 \times 10^6} \right) \left(\frac{1}{1.667 \times 10^{-3}} \right) 10^{-10} + 0.002$$

$$\therefore \quad \mu = 0.10967\,\eta + 0.002$$

From Table 24.1 (DDHB) for SAE 30, the oil number is F

Trail : 1

Take operating temperature $t_o = 70°$ C

From Fig 24.2 (DDHB) for oil-F and $t_o = 70°$ C

Absolute viscosity $\eta = 22.5$ cP $= 22.5 \times 10^{-3}$ Pas

$\therefore \quad \mu = 0.10967 \times 22.5 \times 10^{3} + 0.002 = 4.4676 \times 10^{-3}$

Heat generated $H_g = \mu W v = 4.4676 \times 10^{-3} \times 5000 \times 1.9635 = 43.86$ Watts ---- 24.72 a

Heat dissipated $H_d = CA\,(t_b - t_a)$ ---- 24.77

where $C = 9.6 \times 10^{-3}$ kW/m^2 K $= 9.6$ W/m^2 °K ($\because$ still air)

$A = 25\,dL = 25 \times 0.075 \times 0.075 = 0.140625$ m^2

$$t_b - t_a \approx \frac{t_o - t_a}{2} = \frac{70 - 20}{2} = 25° \text{ C}$$

$\therefore \quad H_d = 9.6 \times 0.140625 \times 25 = 33.75$ Watts

Trail : 2

Assume operating temperature $t_o = 80°$ C

From Fig 24.2 (DDHB) for oil-F and $t_o = 80°$ C

Absolute viscosity $\eta = 16.5$ cP $= 16.5 \times 10^{-3}$ Pas

$\therefore \quad \mu = 0.10967 \times 16.5 \times 10^{-3} + 0.002 = 3.81 \times 10^{-3}$

Heat generated $H_g = \mu W v$ ---- 24.72 a

$= (3.81 \times 10^{-3})\,(5000)\,(1.9635) = 37.4$ Watts

$$t_b - t_a = \frac{t_o - t_a}{2} = \frac{80 - 20}{2} = 30° \text{ C}$$

$\therefore$ Heat dissipated $H_d = (9.6)\,(0.140625)\,(30) = 40.5$ Watts

Trail : 3

Assume $t_o = 90°$ C

From Fig. 24.2 (DDHB) for oil-F and $t_o = 90°$ C

Absolute viscosity $\eta = 12.5$ cP $= 12.5 \times 10^{-3}$ Pas

$\therefore \quad \mu = 0.10967 \times 12.5 \times 10^{-3} + 0.002 = 3.371 \times 10^{-3}$

$\therefore \quad H_g = (3.371 \times 10^{-3})\,(5000)\,(1.9635) = 33.1$ Watts.

$$t_b - t_a = \frac{t_o - t_a}{2} = \frac{90 - 20}{2} = 35° \text{ C.}$$

$\therefore \quad H_d = (9.6)(0.140625)(35) = 47.25$ Watts.

The heat generated and heat dissipated are plotted versus oil temperature as shown in figure 7.14.

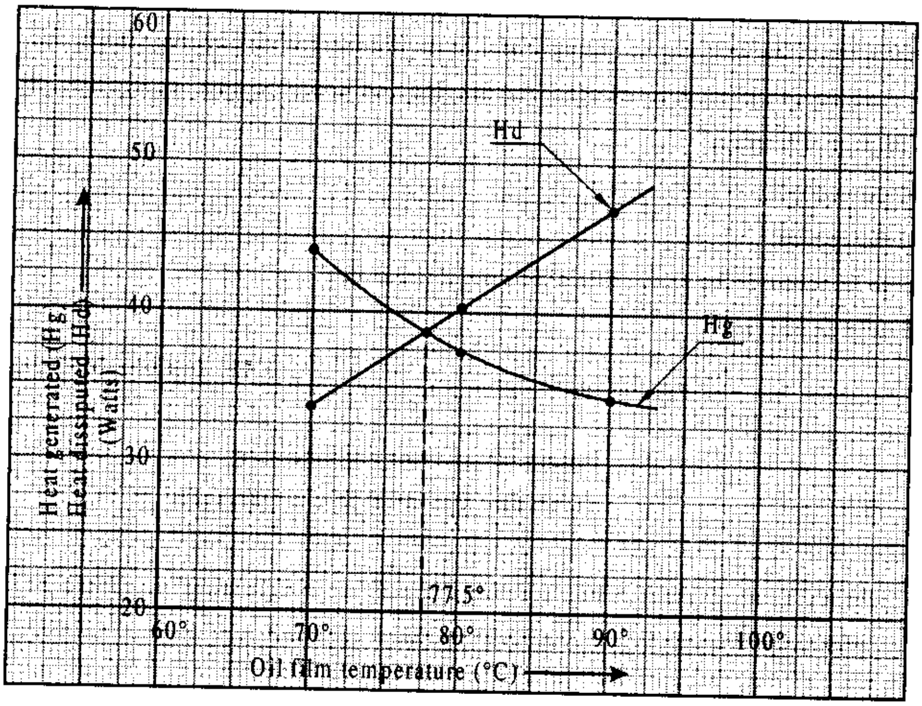


Fig. 7.14

From graph the equilibrium temperature is 77.5° C

From Fig 24.2 (DDHB) for F-oil and $t_o = 77.5°$ C

Absolute viscosity $\eta = 18$ cP $= 18 \times 10^{-3}$ Pas

Example : 7.22

A journal bearing is required to be designed for a rotary compressor for operation at a speed of 1500 rpm. The bearing is to sustain a load of 4500 N and the diameter of the main shaft is 50 mm. Determine

(i) Dimensions of the bearing i.e., Length and inner diameter of the bearing bush.

(ii) The viscosity of the oil to be used for the bearing and hence suggest appropriate oil.

(iii) Coefficient of friction.

(iv) Heat generated

(v) Heat dissipated

(vi) Heat to be removed by the artificial cooling if necessary

(vii) Sommerfeld number

(VTU, Jan/Feb 2005, Jan/Feb 2006)

Data :

Rotary compressor main bearing

d = 50mm; W = 4500 N; n = 1500 rpm

Solution :

Select the recommended values for the rotary compressior main bearing from Table 24.2 (DDHB)

$$\text{Maximum Pressure } P_{max} = 1.66 \text{ MPa} = 1.66 \times 10^6 \text{ N/m}^2$$

$$\text{Diametral clearance ratio } \psi = \frac{c}{d} < 0.001$$

$$\text{Ratio } \frac{L}{d} = 1 \text{ to } 2.2$$

$$\text{Absolute viscosity } \eta = 30 \times 10^{-3} \text{ Pa s}$$

$$\text{Bearing modulus } S^* = 72.5 \times 10^{-9}$$

(i) Dimensions of the bearing

Trial:1

$$\text{Let } \frac{L}{d} = 1$$

$$\therefore \; L = d = 50\text{mm} = 50 \times 10^{-3} \text{ m}$$

$$\therefore \; \text{Operating pressure } P = \frac{W}{Ld} = \frac{4500}{50\times10^{-3}\times50\times10^{-3}} = 1.8 \times 10^6 \frac{N}{m^2} > 1.66 \times 10^6 \frac{N}{m^2}$$

Working or operating pressure 1.8×10^6 N/m^2 is greater than the Permissible maximum pressure 1.66×10^6 N/m^2, the selected $\frac{L}{d}$ ratio is not suitable.

Trial:2

$$\text{Let } \frac{L}{d} = 1.5$$

$\therefore$ Length of bearing L = 1.5d = 1.5 × 50 = 75mm = 75 × 10^{-3} m.

$$\therefore \text{Operating maximum pressure } P = \frac{W}{Ld} = \frac{4500}{75\times10^{-3}\times50\times10^{-3}} = 1.2\times10^{6}\,N/m^2 < 1.66\times10^{6}\,N/m^2$$

Since the Operating maximum pressure 1.2 × 10^6 N/m² is less than the permissible maximum pressure 1.66 × 10^6 N/m², the selected $\frac{L}{D}$ ratio is suitable.

$\therefore$ Length of bearing L = 75 mm = 75 × 10^{-3} m

Since $\psi < 0.001$, take $\psi = \frac{c}{d} = 0.009$

Diametral clearance c = ψd = 0.0009 × 50 = 0.045 mm

Diameter of bearing D = d + c = 50 + 0.045 = 50.045 mm = 50.045 × 10^{-3} m

Hence Length of bearing L = 75 mm = 75 × 10^{-3} m

Diameter of bearing D = 50.045 mm = 50.045 × 10^{-3} m

Working or Operating maximum Pressure P = 1.2 × 10^6 N/m²

(ii) Viscosity of the oil

Assume operating temperature t_o = 55°C

From Fig 24.2 (DDHB) for η = 30 × 10^{-3} Pas and t_o = 55°C

The oil selected is E

Actual value of η = 30 × 10^{-3} Pas.

From Table 24.1 (DDHB), E - oil is Automobile oil, SAE 20 of specific gravity 0.9254 at 15.5°C

Check for the Oil

$$\text{Bearing modules } S''_{cal} = \frac{\eta n'}{P} = \frac{30\times10^{-3}\times\frac{1500}{60}}{1.2\times10^{6}} = 625\times10^{-9}$$

Since $S''_{cal} > S''_{recom}$ (ie 72.5 ×10^{-9}), thick film lubrication is possible. Hence the selected oil is suitable.

$\therefore$ Absolute viscosity of the oil used is 30 × 10^{-3} Pas and the suggested lubricating oil is SAE 20 of specific gravity 0.9254 at 15.5°C i.e., E-oil

(iii) Coefficient of friction

$$\text{From Mckee's equation } \mu = K_a\left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi}\right)10^{-10} + \Delta\mu \qquad \text{---- 24.22}$$

Assume full journal i.e., β = 360°

$\therefore K_a$ = 1.95 × 10^{11}, $\Delta\mu$ = 0.002

$$\mu = (1.95 \times 10^{11})(625 \times 10^{-9})\left(\frac{1}{0.0009}\right)10^{-10} + 0.002$$

$\therefore$ Coefficient of friction $\mu = 0.015542$

(iv) Heat generated

$$\text{Heat generated } H_g = \mu (PLd)\, v = \mu W v \quad \text{---- 24.72a}$$

$$= (0.015542)(4500)\left(\frac{\pi \times 50 \times 10^{-3} \times 1500}{60}\right)$$

$$= 274.65 \text{ Watts}$$

(v) Heat dissipated

$$\text{Heat dissipated } H_d = CA\,(t_h - t_a) \quad \text{---- 24.77}$$

$$\text{where } C = 11.36 \times 10^{-3} \text{ kW/m}^2\text{ K} = 11.36 \text{ W/m}^2\text{ K}$$

$$A = 25\, dL = 25 \times 50 \times 10^{-3} \times 75 \times 10^{-3} = 0.09\,375 \text{ m}^2$$

Assume ambient temperature t_a = 25°C

$$\therefore t_h - t_a \cong \frac{t_0 - t_a}{2} \cong \frac{55 - 25}{2} \cong 15°\text{C or } 15°\text{ K}$$

$\therefore$ Heat dissipated Hd = (11.36) (0.09375) (15) = 15.975 Watts

(vi)

Heat to be removed by artificial cooling = $H_g - H_d$ = 274.65 – 15.975 = 258.675 Watts

(vii) Sommerfeld number

$$\text{Sommerfeld number } S = \left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi^2}\right) = (625 \times 10^{-9})\left(\frac{1}{0.0009}\right)^2 = 0.7716.$$

Example : 7.23

Design a journal bearing for a centrifugal pump running at 1200 rpm. Diameter of journal is 100mm and load on bearing is 15 kN. Take L/d = 1.5, bearing temperature 50°C and ambient temperature 30°C. Find whether artificial cooling is required. (VTU, Dec'08/Jan'09, June/July 08)

Data :

n = 1200 rpm; d = 100 mm; W = 15 kN = 15000 N ;

L/d = 1.5 ; t_a = 30° ; t_h = 50°

Solution :

$$t_h - t_a \cong \frac{t_o - t_a}{2}$$

$$50 - 30 \cong \frac{t_o - 30}{2}$$

$\therefore$ Operating temperature t_o = 70°C

(i) Select the recommended values for centrifugal pump rotor journal bearing from Table 24.2 (DDHB)

Maximum pressure P = 0.69 to 1.37 Mpa

Diametral clearance ratio ψ = 0.0013

Ratio $\frac{L}{d}$ = 1.5 (given)

Absolute viscosity η = 25×10^{-3} Pas

Bearing modulus S" = 483.5×10^{-9}

(ii) Design

Select the allowable pressure $P = 1$ Mpa $= 1 \times 10^6$ N/m^2

$$\text{Also } P = \frac{W}{Ld}$$

$$\text{i.e., } 1 \times 10^6 = \frac{15\times10^3}{(1.5d)(d)} \quad (\because L = 1.5\,d)$$

$\therefore$ Diameter of journal d = 0.1 m = 100 mm

Length of bearing L = 1.5d = 1.5 × 0.1 = 0.15m = 150 mm

$$P_{act} = \frac{15\times10^3}{(0.15)(0.1)} = 1 \times 10^6 \text{ N/m}^2$$

(iii) Selection of oil

From Fig 24.2 (DDHB) for η = 25×10^{-3} Pas and $t_o = 70°C$,

The selected oil is 'G'

$\therefore$ Actual value of η = 31×10^{-3} Pas

From Table 24.1 (DDHB), G - oil is SAE 40

Check for the oil

$$\text{Bearing modulus } S''_{cal} = \frac{\eta.n'}{P} = \frac{31\times10^{-3}\times\frac{1200}{60}}{1\times10^6} = 620 \times 10^{-9}$$

As $S''_{cal} > S''_{reco}$ (i.e., 483.5×10^{-9}), thick film lubrication is possible. Hence the selected oil is suitable.

(iv) Heat generated

Heat generated H_g = μ (PLd) v ----- 24.72 a

From Mckee's equation $\mu = K_a\left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi}\right)10^{-10} + \Delta\mu$ ----- 24.22

Assume full journal $\therefore K_a = 1.95 \times 10^{11}$, $\Delta\mu = 0.002$

$$\therefore \mu = 1.95 \times 10^{11} \times 620 \times 10^{-9}\left(\frac{1}{0.0013}\right)10^{-10} + 0.002 = 0.0113$$

$$\therefore H_g = (0.0113)(1 \times 10^6 \times 0.15 \times 0.1) \times \left(\frac{\pi\times0.1\times1200}{60}\right) = 1065 \text{ Watts}$$

(v) Heat dissipated

Heat dissipated $H_d = CA\,(t_b - t_a)$ ----- 24.77

where $C = 11.36 \times 10^{-3}$ kW/m^{2}K = 11.36 W/m^{2}K

$A = 25\,dL = 25 \times 0.1 \times 0.15 = 0.375$ m^2

$\therefore\; H_d = (11.36)\,(0.375)\,(50 - 30) = 85.2$ Watts

Amount of heat to be removed $= H_g - H_d = 1065 - 85.2 = 979.8$ Watts

Hence artificial cooling is required.

(vi) Minimum film thickness and its location

$$\text{Sommerfeld number } S = \left(\frac{\eta n'}{P}\right)\cdot\frac{1}{\psi^2} = 620 \times 10^{-9} \times \left(\frac{1}{0.0013}\right)^2 = 0.367$$

From Fig 24.13a (Page 24.33) (DDHB) for $S = 0.367$ and $\frac{L}{d} = 1.5$

Minimum film thickness variable $\delta = 0.935$, for full journal bearing

$$\text{Also } \delta = \frac{2h_{min}}{c} \quad \text{where } c = \psi d = 0.0013 \times 0.1 = 1.3 \times 10^{-4}$$

$$\text{i.e., } 0.935 = \frac{2 \times h_{min}}{1.3 \times 10^{-4}}$$

$\therefore$ Minimum film thickness $h_{min} = 6.0775 \times 10^{-5}$ m

From Fig 24.11a (Page 24.30) (DDHB) for $S = 0.367$ and $\frac{L}{d} = 1.5$

Position of minimum film thickness $\phi = 70°$

(vii) Oil flow rate and end leakage

From Fig 24.37 (DDHB) for $S = 0.367$ and $\frac{L}{d} = 1.5$

Flow variable $\lambda_Q = 3.1$, for full journal bearing

$$\text{Length in the direction of motion } B = \frac{\pi d \times \beta°}{360} = \frac{\pi \times 0.1 \times 360}{360} = 0.314$$

$$\therefore \frac{\text{Length of bearing in the direction of motion}}{\text{Length in direction perpendicular to motion}} = \frac{B}{L} = \frac{0.314}{0.15} = 2.0944$$

From Fig. 24.38 (DDHB) for $\frac{B}{L} = 2.0944$ and $\delta = 0.935$

Flow correction factor $C_Q = 1$

$$\text{. Considering side leakage, oil flow through bearing } Q = \frac{\lambda_Q c^2 n' L}{4\psi}.C_Q$$

$$= \frac{3.1 \times \left(1.3 \times 10^{-4}\right)^2 \times \left(\frac{1200}{60}\right)(0.15)(1)}{4 \times 0.0013} = 3.0225 \times 10^{-5} \text{ m}^3\text{/sec}$$

From Fig. 24.39 (DDHB) for S = 0.367 and $\frac{L}{d} = 1.5$

$$\text{Oil flow ratio } \frac{Q_s}{Q} = 0$$

∴ Amount of end leakage $Q_s = 0$

(viii) Power loss due to friction

$$\text{Power loss due to friction } N_\mu = \mu W v = 0.0113 \times 15 \times 10^3 \times \frac{\pi \times 0.1 \times 1200}{60} = 1065 \text{ Watts}$$

Example : 7.24

Design a full journal bearing subjected to 6000 N at 1000 rpm of the journal. The journal is of hardened steel and the bearing is of babbit metal. The bearing is operating with SAE40 oil at 70° C and the ambient temperature is 30°C. Also determine the amount of artificial cooling required.

(VTU, Dec 07/Jan'08)

Data :

W = 6000 N; n = 1000 rpm; Journal – Hardened steel, Bearing – Babbit metal; Oil – SAE 40, t_o = 70° C, t_a = 30° C

Solution :

$$t_b - t_a \cong \frac{t_o - t_a}{2}$$

$$\text{ie., } t_b - 30 \cong \frac{70 - 30}{2}$$

∴ Bearing temperature t_b = 50° C

From Table 24.7 (DDHB) for hardened steel shaft and Babbitt bearing,

$$\text{Bearing modulus } S'' = \frac{\eta n'}{P} = 48.5 \times 10^{-9}$$

From Table 24.1 (DDHB) for SAE.40, the oil selected is G-oil

From Fig. 24.2 (DDHB) for G-oil and t_o = 70° C

Absolute viscosity $\eta = 31 \times 10^{-3}$ Pas

(i) Design of journal and bearing

For thick film lubrication, $S''_{cal} \geq S''_{recom}$

$$\text{ie., } \frac{\eta n'}{P} \geq 48.5 \times 10^{-9}$$

$$\text{ie., } \frac{31 \times 10^{-3} \times \frac{1000}{60}}{P} \geq 48.5 \times 10^{-9}$$

$$\text{ie., } 10.653 \times 10^6 \geq P$$

$$\geq \frac{W}{Ld}$$

$$\geq \frac{6000}{d^2} \text{ (Assume L = d)}$$

ie., $d^2 \geq 5.63 \times 10^{-4}$

$\therefore$ d ≥ 0.0237 m

take, Diameter of journal d = 0.025 m = 25 mm

$\therefore$ Length of bearing L = 0.025 m = 25 mm

Assume diametral clearance ratio $\psi = 0.001$

$$\text{Also } \psi = \frac{c}{d}$$

$$\text{ie., } 0.001 = \frac{c}{0.025}$$

$\therefore$ Diametral clearance c = 2.5×10^{-5} m

Diameter of bearing D = d + c = $0.025 + 2.5 \times 10^{-5}$

= 25.025×10^{-3} m = 25.025 mm

(iii) Amount of artificial cooling

Heat generated Hg = μ (PLd) v = μWv ----- 24.72 a

According to Mckee's equation $\mu = K_a \left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi}\right) 10^{-10} + \Delta\mu$ ----- 24.22

For full journal bearing $K_a = 1.95 \times 10^{11}$; $\Delta\mu = 0.002$

$$\text{Bearing pressure } P = \frac{W}{Ld} = \frac{6000}{(0.025)(0.025)} = 9.6 \times 10^6 \text{ N/m}^2$$

$$\therefore \mu = 1.95 \times 10^{11} \times \frac{\left(31 \times 10^{-3} \times \frac{1000}{60}\right)}{9.6 \times 10^6}\left(\frac{1}{0.001}\right) 10^{-10} + 0.002$$

$$= 3.05 \times 10^{-3}$$

$$\therefore H_d = (3.05 \times 10^{-3})(6000)\left(\frac{\pi \times 0.025 \times 1000}{60}\right) = 23.955 \text{ Watts}$$

ie., Heat generated H_g = 23.955 Watts

Heat dissipated $H_d = CA(t_b - t_a)$ ----- 24.77

where C = 11.36×10^{-3} kW/m²K = 11.36 W/m² K

A = 25 dL = $25 \times 0.025 \times 0.025 = 0.015625$ m²

$\therefore H_d = 11.36 \times 0.015625 (50 - 30) = 3.55$ Watts.

$\therefore$ Amount of heat to be removed = $H_g - H_d = 23.955 - 3.55 = 20.405$ Watts

Hence artificial cooling is required

7.18 THRUST BEARING

Force acting along the axis of the shaft is known as thrust. A pivot is a bearing provided at the end of the shaft as shown in Fig. 7.15. A collar is a bearing surface of revolution provided at any place along the length of the shaft as shown in Fig 7.16. Thrust bearings are used in propeller shafts of ships, air planes, steam turbines etc.

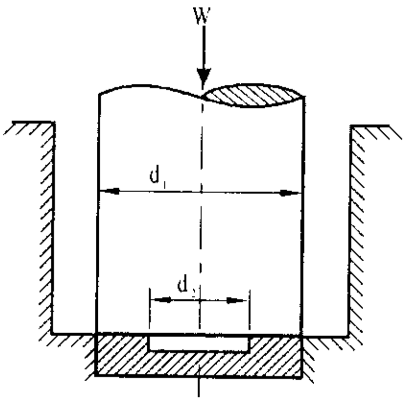


Fig. 7.15

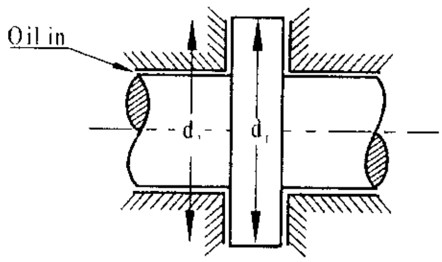


Fig. 7.16

Flat Pivot (Fig. 7.15)

Let d_1 = diameter of shaft

d_2 = diameter of the hole in the thrust disc

d_m = mean diameter

μ = coefficient of friction

$N\mu$ = Friction power in kW

n = Speed in rpm

M_t = Frictional torque

W = Axial load

P = Intensity of normal pressure

For uniform wear theory

Total axial load $W = \frac{1}{2}\pi P\, d_2 (d_1 - d_2)$ ---- Vol-I 19.83

Mean diameter $d_m = \frac{d_1 + d_2}{2}$ ---- Vol.I 19.85 b

Frictional torque $M_t = \mu W \left(\frac{d_1 + d_2}{4}\right)$ ---- 24.114

Frictional power $N\mu = \frac{M_t . n}{9550}$ Where M_t in Nm

For uniform pressure theory

Total axial load $W = P\pi\left(\frac{d_1^2 - d_2^2}{4}\right)$ ---- 24.112

Mean diameter $d_m = \frac{2}{3}\left(\frac{d_1^3 - d_2^3}{d_1^2 - d_2^2}\right)$ ---- Vol.I 19.85a

Frictional torque $M_t = \frac{1}{3}\mu W\left(\frac{d_1^3 - d_2^3}{d_1^2 - d_2^2}\right)$ ---- 24.113

Frictional power $N_\mu = \frac{M_t.n}{9550}$ Where M_t in Nm

Conical pivot

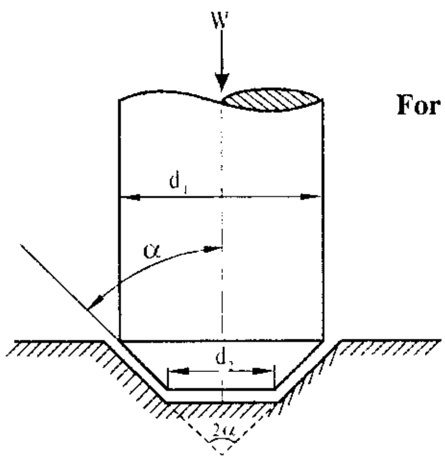


Fig. 7.17

Let 2α = Cone angle of pivot

α = half cone angle

For uniform wear theory

Total axial load $W = \frac{1}{2}\pi P\, d_2\,(d_1 - d_2)$

Frictional torque $M_t = \frac{\mu W}{\sin\alpha}\left(\frac{d_1 + d_2}{4}\right)$ ---- 24.117

Mean diameter $d_m = \frac{d_1 + d_2}{2}$

For uniform pressure theory

Total axial load $W = P\pi\left(\frac{d_1^2 - d_2^2}{4}\right)$ ---- 24.112

Mean diameter $d_m = \frac{2}{3}\left(\frac{d_1^3 - d_2^3}{d_1^2 - d_2^2}\right)$

Frictional torque $M_t = \frac{1}{3}\frac{\mu W}{\sin\alpha}\left(\frac{d_1^3 - d_2^3}{d_1^2 - d_2^2}\right)$ ---- 24.116

Collar bearing theory (Fig. 7.16)

For uniform wear theory

$$\text{Total axial load} \quad W = \frac{1}{2}\pi P\, d_2\,(d_1 - d_2) \times i$$

where i = number of collars

$$\text{Total frictional torque } M_t = \mu W \left(\frac{d_1 + d_2}{4}\right) \qquad \text{---- 24.123b}$$

$$\text{Frictional power} \quad N_\mu = \frac{M_t . n}{9550} \quad \text{where } M_t \text{ in Nm}$$

$$\text{Coefficient of friction } \mu = 83.8\,\frac{v^{0.5}}{P^{0.67}} \text{ where } v = \text{Rubbing speed} = \frac{\pi d_m n}{60{,}000},\ \text{m/sec} \qquad \text{---- 24.125}$$

For uniform pressure theory

Average intensity of pressure per collar

$$P = \frac{W}{0.784\left(d_1^{\,2} - d_2^{\,2}\right) \times i} \qquad \text{---- 24.120}$$

Total frictional torque

$$M_t = \frac{1}{3}\,\mu W \left(\frac{d_1^{\,3} - d_2^{\,3}}{d_1^{\,2} - d_2^{\,2}}\right) \qquad \text{---- 24.122}$$

$$\text{Total axial load} \quad W = P\pi \left(\frac{d_1^{\,2} - d_2^{\,2}}{4}\right) \times i$$

where i = number of collars

Example 7.25

Design a multi collar thrust bearing for a propeller shaft of a 400 kW marine oil engine. The engine makes 300 rpm. The propeller has a pitch of 2.5 m and slip is 30%. The permissible bearing pressure is 0.5 N/mm². Assume uniform pressure theory.

Data : **N = 400 kW ; Slip = 30%**

n = 300 kW; P = 0.5 N/mm²

pitch p = 2.5 m ; Uniform pressure condition

Solution :

(i) Velocity

$$v = \frac{l \times n \times (1 - S)}{60} \quad \text{where } l = \text{lead} = \text{pitch} \times \text{number of starts}$$

Assume single start $\therefore\ l = p = 2.5$ m

$$= \frac{2.5 \times 300(1 - 0.3)}{60} = 8.75 \text{ m/sec}$$

(ii) Load

$$N = \frac{W \times v}{1000}$$

$$\text{i.e.,}\quad 400 = \frac{W \times 8.75}{1000}$$

$$\therefore \text{ Total load } W = 45714.3 \text{ N}$$

(iii) Diameter of collar

$$\text{diameter of shaft } d_2 = \sqrt[3]{\frac{16M_t}{\pi\tau_s}}$$

where τ_s = Allowable shear stress on the propeller shaft material. Assume $\tau_s = 60$ N/mm^2

$$\text{Total torque } M_t = 9550 \times 1000 \times \frac{N}{n} = 9550 \times 1000 \times \frac{400}{300}$$

$$= 12.7333 \times 10^6 \text{ Nmm}$$

$$\therefore \text{ Diameter of shaft } d_2 = \sqrt[3]{\frac{12.7333 \times 10^6 \times 16}{\pi \times 60}} = 102.625 \text{ mm}$$

$$\text{take } d_2 = 110 \text{ mm}$$

$$\text{Outer diameter of collar } d_1 = 1.5\, d_2 = 1.5 \times 110 = 165 \text{ mm}$$

(iv) Number of collars

For uniform pressure theory

$$\text{Total axial load } W = P\pi \left(\frac{d_1^2 - d_2^2}{4}\right) \times i$$

$$\text{i.e., } 45714.3 = 0.5 \times \pi \left(\frac{165^2 - 110^2}{4}\right) \times i$$

$$\therefore \text{ i.e., } i = 7.7$$

$$\therefore \text{ Number of collars } i = 8$$

(v) Frictional torque

$$\text{Total frictional torque for uniform intensity of Pressure } M_t = \frac{1}{3}\mu W \left(\frac{d_1^3 - d_2^3}{d_1^2 - d_2^2}\right) \quad \text{---- 24.122}$$

$$\mu = 83.8 \frac{v^{0.5}}{P^{0.67}} \quad \text{---- 24.126}$$

$$\text{Actual intensity of pressure } P = \frac{W}{\frac{\pi}{4}\left(d_1^2 - d_2^2\right)i} = \frac{45714.3}{\frac{\pi}{4}\left(165^2 - 110^2\right)\times 8}$$

$$= 0.481035 \text{ N/mm}^2 = 0.481035 \times 10^6 \frac{\text{N}}{\text{m}^2}$$

$$\text{Mean diameter } d_m = \frac{2}{3}\left(\frac{d_1^3 - d_2^3}{d_1^2 - d_2^2}\right) = \frac{2}{3}\left(\frac{165^3 - 110^3}{165^2 - 110^2}\right) = 139.33 \text{ mm}$$

$$\therefore \text{ Rubbing speed } v = \frac{\pi d_m . n}{60{,}000} = \frac{\pi \times 139.33 \times 300}{60{,}000} = 2.1886 \text{ m/sec}$$

$$\therefore \mu = \frac{83.8(2.1886)^{0.5}}{\left(0.481035 \times 10^6\right)^{0.67}} = 0.019332$$

$$\therefore M_t = \frac{1}{3} \times 0.019332 \times 45714.3 \times \left(\frac{165^3 - 110^3}{165^2 - 110^2}\right) = 61566.71 \text{ N mm}$$

(vii) Power loss due to friction

$$N_\mu = \frac{M_t \times n}{9550 \times 1000} \text{ kW} = \frac{61566.71 \times 300}{9550 \times 1000} = 1.934 \text{ kW}$$

Example 7.26

A multicollar thrust bearing for a marine engine propeller shaft to transmit 300 kW at 120 rpm. The pitch of the propeller shaft is 2.5 m and the slip is 20%. Permissible pressure is 0.6 N/mm². Determine the number of collars and power loss due to friction. Assume uniform wear theory.

Data : **N = 300 kW ; n = 120 rpm ; p = 2.5 m = pitch**

Slip = 20%; P = 0.6 N/mm² ; Uniform wear theory

Solution :

(i) Number of collars

$$\text{Velocity } v = \frac{l \times n \times (1 - S)}{60} \quad \text{where } l = \text{lead} = \text{pitch} \times \text{number of starts}$$

Assume single start thread $\therefore l = p = 2.5$ m

$$\therefore v = \frac{2.5 \times 120(1 - 0.2)}{60} = 4 \text{ m/sec}$$

$$\text{Power } N = \frac{W \times v}{1000} \text{ kW}$$

$$\therefore \text{ Total maximum load } W = \frac{N \times 1000}{v} = \frac{300 \times 1000}{4} = 75000 \text{ N}$$

Diameter of shaft or inner diameter of collar

$$d_2 = \sqrt[3]{\frac{16M_t}{\pi\tau_s}}$$

Where τ_s = allowable shear stress on the propeller shaft material. Assume τ_s = 60 N/mm^2

$$M_t = 9550 \times 1000 \times \frac{N}{n} = 9550 \times 1000 \times \frac{300}{120} = 23875000 \text{ Nmm}$$

$$\therefore\ d_2 = \sqrt[3]{\frac{16 \times 23875000}{\pi \times 60}} = 126.55 \text{ mm}$$

Take inner diameter of collar d_2 = 130 mm

∴ Outer diameter of collar d_1 = 1.5 × 130 = 195 mm

For uniform wear theory,

Total axial load $W = \frac{1}{2}\pi P d_2 (d_1 - d_2)\, i$

$$\text{i.e., } 75000 = \frac{1}{2}\pi \times 0.6 \times 130\,(195 - 130)\, i$$

$$i = 9.417$$

$$\therefore \text{ Number of collars } i = 10$$

(ii) Power loss due to friction

Total frictional torque for uniform rate of wear

$$M_t = \mu W \left(\frac{d_1 + d_2}{4}\right) \qquad \text{----24.123b}$$

For uniform wear, mean diameter $d_m = \frac{d_1 + d_2}{2} = \frac{195 + 130}{2} = 162.5$ mm

$$\therefore \text{ Rubbing speed } v = \frac{\pi d_m n}{60000} = \frac{\pi \times 162.5 \times 120}{60000} = 1.021 \text{ m/sec}$$

$$\text{Actual intensity of pressure } P = \frac{2W}{\pi d_2 (d_1 - d_2) i}$$

$$= \frac{2 \times 75000}{\pi \times 130(195 - 130) \times 10}$$

$$= 0.565 \text{ N/mm}^2 = 0.565 \times 10^6 \text{ N/m}^2$$

$$\text{Coefficient of friction } \mu = 83.8 \frac{v^{0.5}}{P^{0.67}} \qquad \text{----24.125}$$

$$= \frac{83.8(1.021)^{0.5}}{(0.565 \times 10^6)^{0.67}} = 0.011854$$

$$\therefore \quad M_t = (0.011854)(75{,}000)\left(\frac{195+130}{4}\right) = 72238.6\,\text{Nmm}$$

$\therefore$ Power loss due to friction $N_\mu = \dfrac{M_t \times n}{9550 \times 1000}$ kW where M_t in Nmm

$$= \frac{72238.6 \times 120}{9550 \times 1000} = 0.908\ \text{kW}$$

Example 7.27

Determine the main dimensions and powerloss of a multicollar thrust bearing for a propeller shaft of 450 kW marine oil engine. The engine makes 250 rpm. The shaft diameter is 150 mm and the speed of the ship is 5 m/sec.

Data : $N = 450$ kW ; $n = 120$ rpm ; $d_2 = 150$ mm

$v = 5$ m/sec

Solution :

(i) Load

$$\text{Power } N = \frac{W \times V}{1000}$$

$$\therefore \quad \text{Total axial load } W = \frac{450 \times 1000}{5} = 90000\ \text{N} = 90\ \text{kN}$$

(ii) Outer diameter of collar $d_1 = 1.5\ d_2 = 1.5 \times 150 = 225$ mm

(iii) Number of collars

Assume uniform wear theory

$$\therefore \quad \text{Total axial load } W = \frac{1}{2}\pi P\, d_2 (d_1 - d_2)\, i$$

$$\text{Assume pressure } P = 0.5\ \text{MPa}$$

$$\text{i.e., } 90{,}000 = \frac{1}{2} \times \pi \times 0.5 \times 150\,(225 - 150) \times i$$

$$\text{i.e., } i = 10.186$$

$$\therefore \quad \text{Number of collars } i = 11$$

(iv) Frictional torque

Total frictional torque for uniform rate of wear $M_t = \mu W \left(\dfrac{d_1 + d_2}{4}\right)$ ---- 24.123b

For uniform wear, mean diameter $d_m = \dfrac{d_1 + d_2}{2} = \dfrac{225 + 150}{2} = 187.5$mm

$\therefore$ Rubbing speed $v = \frac{\pi d_m n}{60000} = \frac{\pi \times 187.5 \times 250}{60,000} = 2.4544$ m/sec

Actual intensity of pressure $P = \frac{2W}{\pi d_2 (d_1 - d_2) i}$

$$= \frac{2 \times 90000}{\pi \times 150 \times (225 - 150) \times 11} = 0.463 \text{ N/mm}^2$$

$$= 0.463 \times 10^6 \text{ N/m}^2$$

$\therefore$ Coefficient of friction $\mu = 83.8 \frac{v^{0.5}}{P^{0.67}}$ ---- 24.125

$$= \frac{83.8(2.4544)^{0.5}}{(0.463 \times 10^6)^{0.67}} = 0.021$$

$$\therefore M_t = (0.021)(90000)\left(\frac{225+150}{4}\right) = 177187.5 \text{ Nmm}$$

Power loss due to friction $N_\mu = \frac{M_t \times n}{9550 \times 1000}$ kW Where M_t in Nmm

$$= \frac{177187.5 \times 250}{9550 \times 1000} = 4.64 \text{ kW}$$

REVIEW QUESTIONS

1. What is Sommerfeld number? What is its application in designing Hydrodynamic journal Bearings? Explain at least four dimensionless parameters, which depend upon the Sommerfeld number as plotted by Raimondi and Boyd. **VTU, July/August 2004**
2. Derive Petroff's equation for coefficient of friction for Hydrodynamic bearing. **VTU, February 2002, August 2001**
3. List the different forms of lubrication and bearing materials. **VTU, August 2001**
4. Explain the significance of the bearing characteristic number in the design of sliding contact bearing. **VTU, July/August 2002**
5. Explain the following :
   i) Viscosity ii) Newtonion fluid iii) Fluidity iv) Oil film thickness
   VTU, January/February 2004
6. What are the factors affecting viscosity. Explain. **VTU, January/February 2004**
7. Explain the mechanism of hydrodynamic lubrication in journal bearing. **BU, August/September 2001**

8. What are the limitations in Petroff's law. **BU, August 1997**
9. What are collar bearings? Explain briefly. **BU, March/April 1999**
10. What do you understand by,
    a) Minimum oil film thickness and
    b) Coefficient of friction in bearing. **BU, August/September 2001**
11. Write note one :
    i) Bearing modulus ii) Bearing characteristic number **BU, December 2003**
12. Explain the following :
    i) Hydrostatic lubrication ii) Boundary lubrication iii) Thick film lubrication
    iv) Journal bearing v) Thrust bearing.

EXERCISES

1. A journal bearing is to be designed for the main bearings of a four stroke oil engine to sustain a load of 50 kN for a shaft diameter 50 mm. The engine runs at a speed of 1500 rpm. Design the bearings completely.
2. Design the bearing for a centrifugal pump for the following data.
   Load on the rotor = 30 kN; speed = 1000 rpm.
3. Design a bearing and journal to support a load of 4500 N at 600 rpm using a hardened steel journal and a bronze backed babbit bearing. The bearings is lubricated by oil rings. The ambient temperature is 20° C and maximum oil temperature is 60° C.
4. A journal bearing is required for the following conditions
   Radial force 4000 N
   Speed 600 rpm
   Diameter of shaft 60 mm
   Ambient temperature 23° C and the place is unventilated. Design the details of the bearing for satisfactory working and find whether the bearing require artificial cooling.
5. The thrust for propeller shaft in a marine engine is taken up by a number of collars integrated with shafts, which is 300 mm in diameter. The thrust on the shaft is 200 kN and the speed is 75 rpm. Taking μ as constant and equal to 0.05 and assuming the bearing pressure as uniform and equal to 0.3 N/mm^2 find
   i) Number of collars required
   ii) Power lost in the friction
   iii) Heat generated at the bearing.
6. A 50 mm diameter hardened and ground steel journal rotates at 1440 rpm in a lathe turned bronze bushing which is 50 mm long. For hydrodynamic lubrication the minimum oil film thickness should be 0.012 mm. The class of fit is $H_8\ d_8$ and the viscosity of lubricant is 18×10^{-3} Pas. Determine the maximum radial load that the journal can carry and still operate under hydrodynamic conditions.

7. The main bearings of a steam turbine operates at a speed of 1500 rpm and the diameter is 40 mm. The load on the bearing is estimated to be 5 kN. Assume a clearance ratio of 0.001, length to diameter ratio of 1.5 and the bearing to be well ventilated. The operating temperature of the oil film is 50° C and the viscosity of oil is 0.02 Pas. Determine :
   i) Whether fluid film lubication can be expected
   ii) Whether artificial cooling is necessary and if so the amount of heat to be removed by artificial cooling.
   iii) The amount of oil flow.
8. a) Explain with sketches theory of hydrodynamic lubrication.
   b) A journal bearing is required to be designed for a rotary compressor for operation at a speed of 1500 rpm. The bearing is to sustain a load of 4500 N and the diameter of the main shaft is 50 mm. Determine
      i) Length and inner diameter of the bearing bush
      ii) Viscosity of an oil to be used as a lubricant and hence suggest a lubricating oil
      iii) The coefficient of friction
      iv) Heat generated
      v) Heat dissipating capacity
      vi) Amount of heat to be removed by artificial cooling
      vii) Sommerfeld number. **VTU, January/February 2005**
9. a) Derive Petroff's equation for coefficient of friction of a lightly loaded journal bearing.
   b) It is required to design a main bearing of a four stroke oil engine to sustain a load of 6 kN over a shaft of diameter 50mm. The operating speed of the shaft is 1000 rpm and the operating temperature is 50° C. Determine
      i) Dimensions of the bearing.
      ii) The viscosity of the oil to be used for the bearing and hence suggest appropriate oil.
      iii) Coefficient of friction
      iv) Heat generated
      v) Heat dissipated
      vi) Heat to be removed by the artificial cooling if necessary
      vii) Sommerfeld number. **VTU, January/February 2006**
10. a. Derive the petroff's equation for frictional power loss of a lightly loaded journal rotating at high speed concentric to the bearing.
    b. Determine the power loss in a bearing, the diameter of journal is 60 mm and length 80mm. The diametral clearance is 0.12 mm. The bearing supports a load of 5000 N and the journal rotates at a speed of 2500 rpm. The kinematic viscosity and the specific gravity of the oil used at the operating temperature of the bearing are 50 centi-stokes and 0.9, respectively. **VTU, July 2006**

11. a) Explain the Bearing modulus.

    b) Derive Petroff's equation for a lightly loaded bearing.

    c) A 75mm long full journal bearing of diameter 75mm supports a radial load of 12kN at the shaft speed of 1800 rpm. Assume ratio of diameter to the diametral clearance as 1000. the viscosity of oil is 0.01 Pas at the operating temperature. Determine

    i) Sommerfeld number.

    ii) Coefficient of friction based on Mckee's equation.

    iii) Amount of heat generated **VTU, Dec. 06/ Jan. 2007**

12. a) Explain the significance of the bearing characteristic number in the design of sliding contact bearings.

    b) A full journal bearing of 60 mm diameter and 100 mm long has a bearing pressure of 1.4 N/mm^2. The speed of the journal is 800 rpm and the ratio of journal diameter to the diametral clearance is 1000. The bearing is lubricated with oil whose absolute viscosity at the operating temperature of 75°C may be taken as 0.011 kg/ms. The room temperature is 30°C. Find: i) The amount of artificial cooling required, and ii) The mass of the lubricating oil required, if the difference between the outlet and inlet temperature of the oil is 10°C. Take specific heat of the oil as 1850 J/kg/°C. **VTU, July 2007**

13. a) Explain with sketch theory of hydrodynamic lubrication

    b) Design a full journal bearing subjected to 6000 N at 1000 rpm of the journal. The journal is of hardened steel and the bearing is of babbit metal. The bearing is operating with SAE 40 oil at 70°C and the ambient temperature is 30°C. Also determine the amount of artificial cooling required. **VTU, Dec. 07/ Jan. 2008**

14. a) What is bearing modulus? Explain the significance of bearing modulus in the design of bearing.

    b) Design a journal bearing for a centrifugal pump from the following data:

    Load on the journal = 10kN

    Speed of the journal = 900 rpm

    Ambient temperature = 15°C. **VTU, June/July 2008**

15. a) Explain with a neat sketch, the importance of bearing characteristic number in design of journal bearing.

    b) Design a journal bearing for a centrifugal pump running at 1200 rpm. Diameter of journal is 100 mm and load on bearing is 15 kN. Take l/d = 1.5, bearing temperature 50° and ambient temperature 30°. Find whether artificial cooling is required.

    VTU, Dec. 08/ Jan. 2009

UNIT 8

Design of Belts, Ropes and Chains

8.1 INTRODUCTION

Power is transmitted from the prime mover to a machine by means of intermediate mechanism called drives. This intermediate mechanism known as drives may be belt or chain or gears. Belt is used to transmit motion from one shaft to another shaft with the help of pulleys preferably if the centre distance is long. It is not a positive drive since there is slip in belt drive. Three types of belt drives are commonly used. They are (i) Flat belt drive (ii) V-belt drive (iii) Rope or circular belt drive.

8.2 FLAT BELT DRIVE

When the distance between two pulleys is around 10 meters and moderate power is required then flat belt drive is preferred. This may be arranged in two ways (i) open belt drive (ii) Cross belt drive. When the direction of rotation of both the pulleys are required in the same direction, then we can use open belt drive; if direction of rotation of pulleys are required in opposite direction then cross belt drive is used. The pulley which drives the belt is known as driver and the pulley which follows driver is known as driven or follower.

8.2.1 Merits and Demerits of Feat belt drive

Merits :

(i) Simplicity, low cost, smoothness of operation, ability to absorb shocks, flexibility and efficiency at high speeds.

(ii) Protect the driven mechanism against breakage in case of sudden overloads owing to belt slipping.

(iii) Simplicity of care, low maintenance and service.

(iv) Possibility to transmit power over a moderately long distance.

Demerits :

(i) It is not a positive drive.

(ii) Comparatively large size.