Critical Pressure of the Journal Bearing

The pressure at which the oil film breaks down so that metal to metal contact begins, is known as critical pressure or the minimum operating pressure of the bearing. It may be obtained by the following empirical relation, i .e .

Critical pressure or minimum operating pressure,

$$p = \frac{ZN}{4.75 \times 10^6} \left(\frac{d}{c}\right)^2 \left(\frac{l}{d+l}\right) \text{ N/mm}^2 \qquad \text{...(when Z is in kg / m-s)}$$

Sommerfeld Number

The Sommerfeld number is also a dimensionless parameter used extensively in the design of journal bearings. Mathematically,

$$\text{Sommerfeld number} = \frac{ZN}{p}\left(\frac{d}{c}\right)^2$$

For design purposes, its value is taken as follows :

$$\frac{ZN}{p}\left(\frac{d}{c}\right)^2 = 14.3 \times 10^6 \qquad \text{... (when } Z \text{ is in kg / m-s and } p \text{ is in N / mm}^2\text{)}$$

Heat Generated in a Journal Bearing

The heat generated in a bearing is due to the fluid friction and friction of the parts having relative motion. Mathematically, heat generated in a bearing,

$Q_g = \mu . W.V$ N-m/s or J/s or watts ... (i)

where μ = Coefficient of friction,

W = Load on the bearing in N,

= Pressure on the bearing in N/mm^2 × Projected area of the bearing in mm ^{2} = p (l × d),

V = Rubbing velocity in m/s $= \frac{\pi d.N}{60}$, d is in metres, and

N = Speed of the journal in r.p.m.

After the thermal equilibrium has been reached, heat will be dissipated at the outer surface of the bearing at the same rate at which it is generated in the oil film. The amount of heat dissipated will depend upon the temperature difference, size and mass of the radiating surface and on the amount of air flowing around the bearing. However, for the convenience in

bearing design, the actual heat dissipating area may be expressed in terms of the projected area of the journal.

Heat dissipated by the bearing,

$$Q_d = C.A\,(t_b - t_a) \text{ J/s or W} \qquad \ldots (\because\ 1 \text{ J/s} = 1 \text{ W}) \quad \ldots(ii)$$

where C = Heat dissipation coefficient in W/m²/°C,

A = Projected area of the bearing in $m^2 = l \times d$,

t_b = Temperature of the bearing surface in °C, and

t_a = Temperature of the surrounding air in °C.

The value of C have been determined experimentally by O. Lasche. The values depend upon the type of bearing, its ventilation and the temperature difference. The average values of C (in W/m² /°C), for journal bearings may be taken as follows :

For unventilated bearings (Still air) = 140 to 42 0 W/m²/°C

For well ventilated bearings = 490 to 1400 W/m²/°C

It has been shown by experiments that the temperature of the bearing (t_b) is approximately mid-way between the temperature of the oil film (t_0) and the temperature of the outside air (t_a). In other words,

$$t_b - t_a = \frac{1}{2}\,(t_0 - t_a)$$

Notes : 1. For well designed bearing, the temperature of the oil film should not be more than 60°C, otherwise the viscosity of the oil decreases rapidly and the operation of the bearing is found to suffer. The temperature of the oil film is often called as the ***operating temperature*** of the bearing.

2. In case the temperature of the oil film is higher, then the bearing is cooled by circulating water through coils built in the bearing.

3. The mass of the oil to remove the heat generated at the bearing may be obtained by equating the heat generated to the heat taken away by the oil. We know that the heat taken away by the oil,

$$Q_t = m.S.t \text{ J/s or watts}$$

where m = Mass of the oil in kg / s,

S = Specific heat of the oil. Its value may be taken as 1840 to 2100 J / kg / °C,

t = Difference between outlet and inlet temperature of the oil in °C.

Design Procedure for Journal Bearing

The following procedure may be adopted in designing journal bearings, when the bearing load, the diameter and the speed of the shaft are known.

1. Determine the bearing length by choosing a ratio of l / d from Table 26.3.

2. Check the bearing pressure, p = W / l.d from Table 26.3 for probable satisfactory value.

3. Assume a lubricant from Table 26.2 and its operating temperature (t_0). This temperature should be between 26.5°C and 60°C with 82°C as a maximum for high temperature installations such as steam turbines.

4. Determine the operating value of ZN / p for the assumed bearing temperature and check this value with corresponding values in Table 26.3, to determine the possibility of maintaining fluid film operation.

5. Assume a clearance ratio c / d from Table 26.3.

6. Determine the coefficient of friction (?) by using the relation as discussed in Art. 26.15.

7. Determine the heat generated by using the relation as discussed in Art. 26.18.

8. Determine the heat dissipated by using the relation as discussed in Art. 26.18.

9. Determine the thermal equilibrium to see that the heat dissipated becomes at least equal to the heat generated. In case the heat generated is more than the heat dissipated then either the bearing is redesigned or it is artificially cooled by water.

Example) Design a journal bearing for a centrifugal pump from the following data : Load on the journal = 2 0 000 N; Speed of the journal = 900 r.p.m.; Type of oil is SAE 10, for which the absolute viscosity at 55°C = 0.017 kg / m-s; Ambient temperature of oil = 15.5°C ; Maximum bearing pressure for the pump = 1.5 N / mm ^{2}. Calculate also mass of the lubricating oil required for artificial cooling, if rise of temperature of oil be limited to 10°C. Heat dissipation coefficient = 12 32 W/m^2/°C.

Solution.

T he journal bearing is designed as discussed in the following steps :

1. First of all, let us find the length of the journal (l). Assume the diameter of the journal (d)

as 100 mm. From Table 26.3, we find that the ratio of l / d for centrifugal pumps varies from 1 to 2.

Let us take l / d = 1.6.

$$\therefore \quad l = 1.6\,d = 1.6 \times 100 = 160 \text{ mm} \quad \textbf{Ans.}$$

2. We know that bearing pressure,

$$p = \frac{W}{l.d} = \frac{20\ 000}{160 \times 100} = 1.25$$

Since the given bearing pressure for the pump is 1.5 N/mm2, therefore the above value of p is safe and hence the dimensions of l and d are safe.

3.

$$\frac{Z.N}{p} = \frac{0.017 \times 900}{1.25} = 12.24$$

From Table 26.3, we find that the operating value of

$$\frac{Z.N}{p} = 28$$

We have discussed in Art. 26.14, that the minimum value of the bearing modulus at which the oil film will break is given by

$$3\ K = \frac{ZN}{p}$$

∴ Bearing modulus at the minimum point of friction,

$$K = \frac{1}{3}\left(\frac{Z.N}{p}\right) = \frac{1}{3} \times 28 = 9.33$$

Since the calculated value of bearing characteristic number $\left(\frac{Z.N}{p} = 12.24\right)$ is more than 9.33,

therefore the bearing will operate under hydrodynamic conditions.

4. From Table 26.3, we find that for centrifugal pumps, the clearance ratio (c/d) = 0.0013

5. We know that coefficient of friction,

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + k = \frac{33}{10^8} \times 12.24 \times \frac{1}{0.0013} + 0.002$$

$$= 0.0031 + 0.002 = 0.0051 \qquad \text{... [From Art. 26.13, } k = 0.002]$$

6. Heat generated,

$$Q_g = \mu\ W\ V = \mu\ W\left(\frac{\pi d.N}{60}\right) \text{W} \qquad \ldots\left(\because V = \frac{\pi d.N}{60}\right)$$

$$= 0.0051 \times 20000\left(\frac{\pi \times 0.1 \times 900}{60}\right) = 480.7\ \text{W}$$

... (d is taken in metres)

7. Heat dissipated,

$$Q_d = C.A\ (t_b - t_a) = C.l.d\ (t_b - t_a)\ \text{W} \qquad \ldots\ (\because A = l \times d)$$

We know that

$$(t_b - t_a) = \frac{1}{2}(t_0 - t_a) = \frac{1}{2}(55° - 15.5°) = 19.75°\text{C}$$

$$\therefore \quad Q_d = 1232 \times 0.16 \times 0.1 \times 19.75 = 389.3 \text{ W}$$

... (l and d are taken in metres)

We see that the heat generated is greater than the heat dissipated which indicates that the bearing is warming up. Therefore, either the bearing should be redesigned by taking t_0 = 63°C or the bearing should be cooled artificially. We know that the amount of artificial cooling required = Heat generated – Heat dissipated = $Q_g - Q_d$

= 480.7 – 389.3 = 91.4 W

Mass of lubricating oil required for artificial cooling

Let m = Mass of the lubricating oil required for artificial cooling in kg / s.

We know that the heat taken away by the oil, Q_t = m.S.t = m × 1900 × 10 = 19 000 m W

... [∵ Specific heat of oil (S) = 1840 to 2100 J/kg/°C]

Equating this to the amount of artificial cooling required, we have

$$19\,000\, m = 91.4$$

$$\therefore \quad m = 91.4 / 19\,000 = 0.0048 \text{ kg / s} = 0.288 \text{ kg / min}$$ **Ans.**

Solid Journal Bearing

A solid bearing, as shown in Fig. 26.9, is the simplest form of journal bearing. It is simply a block of cast iron with a hole for a shaft providing running fit. The lower portion of the block is extended to form a base plate or sole with two holes to receive bolts for fastening it to the frame. An oil hole is drilled at the top for lubrication. The main disadvantages of this bearing are

1. There is no provision for adjustment in case of wear, and

2. The shaft must be passed into the bearing axially, i.e . endwise.

Since there is no provision for wear adjustment, therefore this type of bearing is used when the shaft speed is not very high and the shaft carries light loads only.

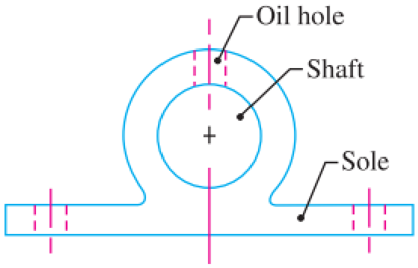

Fig. 26.9. Solid journal bearing.

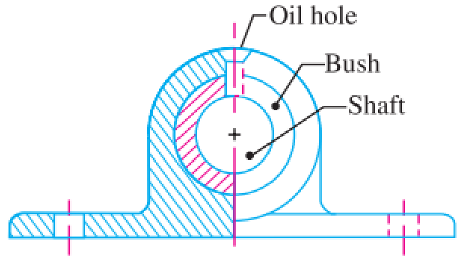

Fig. 26.10. Bushed bearing.

Bushed Bearing

A bushed bearing, as shown in Fig. 26.10, is an improved solid bearing in which a bush of brass or gun metal is provided. The outside of the bush is a driving fit in the hole of the casting whereas the inside is a running fit for the shaft. When the bush gets worn out, it can be easily replaced. In small bearings, the frictional force itself holds the bush in position, but for shafts transmitting high power, grub screws are used for the prevention of rotation and sliding of the bush.

Design of Bearing Caps and Bolts

When a split bearing is used, the bearing cap is tightened on the top. The load is usually carried by the bearing and not the cap, but in some cases e.g . split connecting rod ends in double acting steam engines, a considerable load comes on the cap of the bearing. Therefore, the cap and the holding down bolts must be designed for full load.

The cap is generally regarded as a simply supported beam, supported by holding down bolts and loaded at the centre as shown in Fig. 26.13.

Let W = Load supported at the centre,

α = Distance between centers of holding down bolts,

l = Length of the bearing, and

t = Thickness of the cap.

We know that maximum bending moment at the centre,

M = W.a /4

and the section modulus of the cap,

Z = l. t^2 /6

∴ Bending stress,

$$\sigma_b = \frac{M}{Z} = \frac{W.a}{4} \times \frac{6}{l.t^2} = \frac{3W.a}{2l.t^2}$$

and

$$t = \sqrt{\frac{3\,W.a}{2\sigma_b.l}}$$

Note : When an oil hole is provided in the cap, then the diameter of the hole should be subtracted from the length of the bearing.

The cap of the bearing should also be investigated for the stiffness. We know that for a simply supported beam loaded at the centre, the deflection,

$$\delta = \frac{W.a^3}{48\,E.I} = \frac{W.a^3}{48\,E \times \frac{l.t^3}{12}} = \frac{W.a^3}{4\,E.l.t^3} \qquad \ldots\left(\because I = \frac{l.t^3}{12}\right)$$

$$\therefore \qquad t = 0.63\,a\left[\frac{W}{E.l.\delta}\right]^{1/3}$$

The deflection of the cap should be limited to about 0.025 mm.

In order to design the holding down bolts, the load on each bolt is taken 33% higher than the normal load on each bolt. In other words, load on each bolt is taken $\frac{4\,W}{3n}$, where n is the number of bolts used for holding down the cap.

Let d_c = Core diameter of the bolt, and

σ_t = Tensile stress for the material of the bolt.

$$\therefore \qquad \frac{\pi}{4}(d_c)^2\,\sigma_t = \frac{4}{3} \times \frac{W}{n}$$

From this expression, the core diameter (d_c) may be calculated. After finding the core diameter,

the size of the bolt is fixed.