Sliding Contact Bearings

Classification of Bearings

1. **<u>According to the direction of load to be supported.</u>**

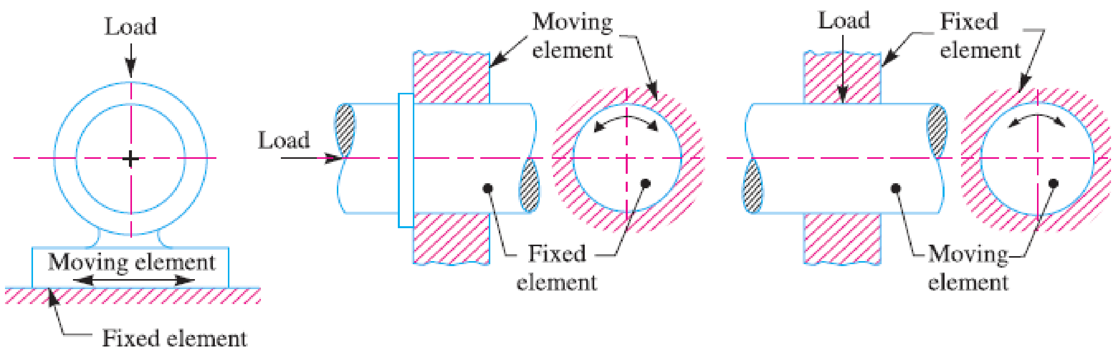
The bearings under this group are classified as:

(*a*) Radial bearings

(*b*) Thrust bearings.

In ***radial bearings***, the load acts perpendicular to the direction of motion of the moving element as shown in the Figure 1 (a & b).

In ***thrust bearings,*** the load acts along the axis of rotation as shown in (*c*).



(*a*) Radial bearing. (*b*) Radial bearing. (*c*) Thrust bearing.

Figure 1

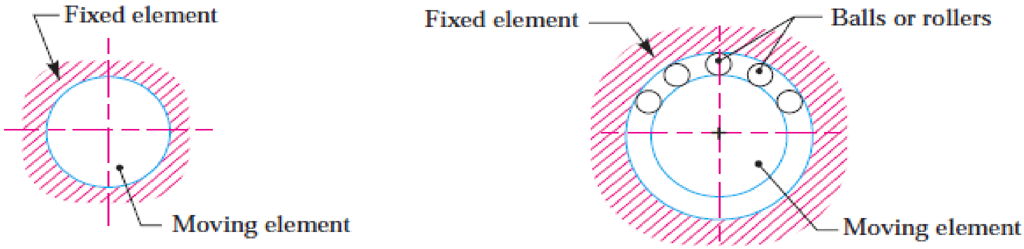
2. **<u>According to the nature of contact.</u>**

The bearings under this group are classified as:

(*a*) Sliding contact bearings,

(*b*) Rolling contact bearings.

In ***sliding contact bearings,*** as shown in the following Figure 2. (*a*), the sliding takes place along the surfaces of contact between the moving element and the fixed element. The sliding contact bearings are also known as ***plain bearings.***



(*a*) Sliding contact bearing. (*b*) Rolling contact bearings.

Figure 2

In ***rolling contact bearings,*** as shown in figure 2 (*b*), the steel balls or rollers, are interposed between the moving and fixed elements. The balls offer rolling friction at two points for each ball or roller.

Types of Sliding Contact Bearings

The sliding contact bearings in which the sliding action is guided in a straight line and carrying radial loads, as shown in figure 3 (*a*), may be called ***slipper*** or ***guide bearings.*** Such type of bearings are usually found in cross-head of steam engines.

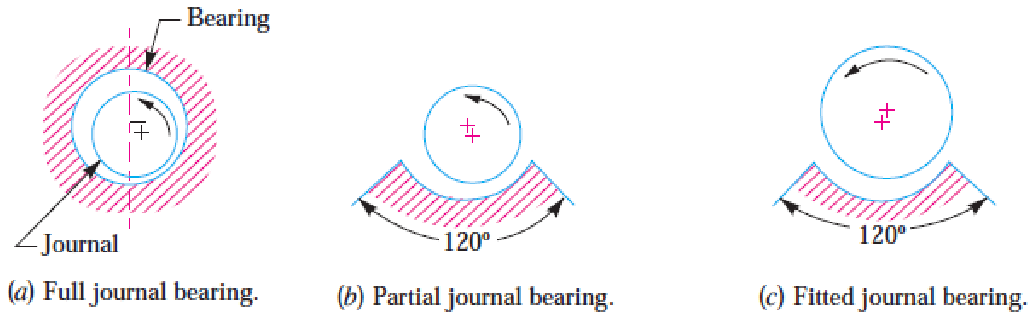


Figure 3

The sliding contact bearings in which the sliding action is along the circumference of a circle or an arc of a circle and carrying radial loads are known as ***journal*** or ***sleeve bearings.*** When the angle of contact of the bearing with the journal is 360° as shown in the above figure (*a*), then the bearing is called a ***full journal bearing.*** This type of bearing is commonly used in industrial machinery to accommodate bearing loads in any radial direction.

When the angle of contact of the bearing with the journal is 120°, as shown in figure 3 (*b*), then the bearing is said to be ***partial journal bearing*.** This type of bearing has less friction than full journal bearing, but it can be used only where the load is always in one direction. The most common application of the partial journal bearings is found in rail road car axles. The full and partial journal bearings may be called as ***clearance bearings*** because the diameter of the journal is less than that of bearing.

When a partial journal bearing has no clearance *i.e*. the diameters of the journal and bearing are equal, then the bearing is called a ***fitted bearing,*** as shown in figure 3 (*c*).

The sliding contact bearings, according to the thickness of layer of the lubricant between the bearing and the journal, may also be classified as follows:

1. <u>Thick film bearings</u>. The thick film bearings are those in which the working surfaces are completely separated from each other by the lubricant. Such type of bearings are also called as ***hydrodynamic lubricated bearings.***

2. <u>Thin film bearings</u>. The thin film bearings are those in which, although lubricant is present, the working surfaces partially contact each other atleast part of the time. Such type of bearings are also called ***boundary lubricated bearings.***

3. <u>Zero film bearings.</u> The zero film bearings are those which operate without any lubricant present.

4*<u>. Hydrostatic or externally pressurized lubricated bearings</u>*. The hydrostatic bearings are those which can support steady loads without any relative motion between the journal and the bearing. This is achieved by forcing externally pressurized lubricant between the members.

<u>Hydrodynamic Lubricated Bearings</u>

We have already discussed that in hydrodynamic lubricated bearings, there is a thick film of lubricant between the journal and the bearing. A little consideration will show that when the bearing is supplied with sufficient lubricant, a pressure is build up in the clearance space when the journal is rotating about an axis that is eccentric with the bearing axis. The load can be supported by this fluid pressure without any actual contact between the journal and bearing. The load carrying ability of a hydrodynamic bearing arises simply because a viscous fluid resists being pushed around. Under the proper conditions, this resistance to motion will develop a pressure distribution in the lubricant film that can support a useful load. The load supporting pressure in hydrodynamic bearings arises from either:

1. The flow of a viscous fluid in a converging channel (known as ***wedge film lubrication***).

2. The resistance of a viscous fluid to being squeezed out from between approaching surfaces (known as ***squeeze film lubrication***).

Assumptions in Hydrodynamic Lubricated Bearings

The following are the basic assumptions used in the theory of hydrodynamic lubricated bearings:

1. The lubricant obeys Newton's law of viscous flow.

2. The pressure is assumed to be constant throughout the film thickness.

3. The lubricant is assumed to be incompressible.

4. The viscosity is assumed to be constant throughout the film.

5. The flow is one dimensional, *i.e.* the side leakage is neglected.

Important Factors for the Formation of Thick Oil Film in Hydrodynamic Lubricated Bearings

According to Reynolds, the following factors are essential for the formation of a thick film of oil in hydrodynamic lubricated bearings:

1. A continuous supply of oil.

2. A relative motion between the two surfaces in a direction approximately tangential to the surfaces.

3. The ability of one of the surfaces to take up a small inclination to the other surface in the direction of the relative motion.

4. The line of action of resultant oil pressure must coincide with the line of action of the external load between the surfaces.

Wedge Film Journal Bearings

The load carrying ability of a wedge-film journal bearing results when the journal and/or the bearing rotates relative to the load. The most common case is that of a steady load, a fixed (nonrotating) bearing and a rotating journal. Fig.4 (*a*) shows a journal at rest with metal to metal contact at *A* on the line of action of the supported load. When the journal rotates slowly in the anticlockwise direction, as shown in Fig.4 (*b*), the point of contact will move to *B*, so that the angle *AOB* is the angle of sliding friction of the surfaces in contact at *B*. In the absence of a lubricant, there will be dry metal to metal friction. If a lubricant is present in the clearance space of the bearing and journal, then a thin absorbed film of the lubricant may partly separate the surface, but a continuous fluid film completely separating the surfaces will not exist because of slow speed.

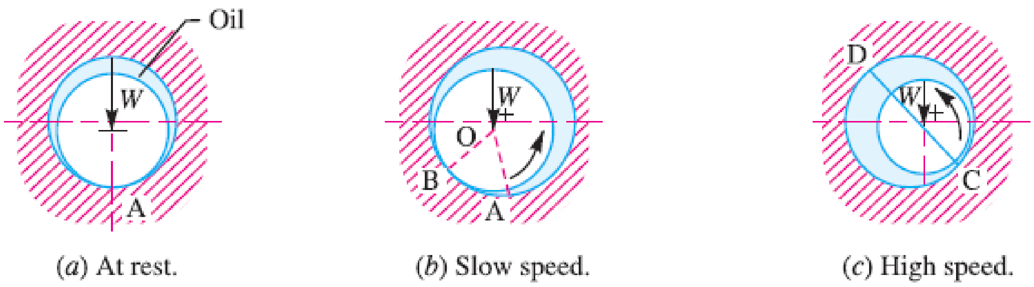


Figure 4

When the speed of the journal is increased, a continuous fluid film is established as in (*c*). The center of the journal has moved so that the minimum film thickness is at *C*. It may be noted that from *D* to *C* in the direction of motion, the film is continually narrowing and hence is a converging film. The curved converging film may be considered as a wedge shaped film of a slipper bearing wrapped around the journal. A little consideration will show that from *C* to *D* in the direction of rotation, as shown in (*c*), the film is diverging and cannot give rise to a positive pressure or a supporting action. The following Figure 5 shows the two views of the bearing shown in (*c*) above, with the variation of pressure in the converging film. Actually, because of side leakage, the angle of contact on which pressure acts is less than 180°.

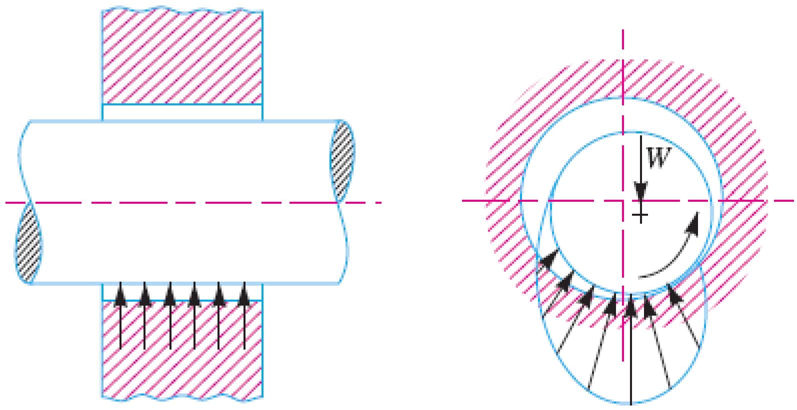


Figure 5

<u>Lubricants</u>

The lubricants are used in bearings to reduce friction between the rubbing surfaces and to carry away the heat generated by friction. It also protects the bearing against corrosion. All lubricants are classified into the following three groups:

1. Liquid, **2.** Semi-liquid, and **3.** Solid.

The ***liquid lubricants*** usually used in bearings are mineral oils and synthetic oils. The mineral oils are most commonly used because of their cheapness and stability. The liquid lubricants are usually preferred where they may be retained. A grease is a ***semi-liquid lubricant*** having higher viscosity than oils. The greases are employed where slow speed and heavy pressure exist and where oil drip from the bearing is undesirable. The ***solid lubricants*** are useful in reducing friction where oil films cannot be maintained because of pressures or temperatures. They should be softer than materials being lubricated. A graphite is the most common of the solid lubricants either alone or mixed with oil or grease.

Properties of Lubricants

1. *Viscosity*. It is the measure of degree of fluidity of a liquid. It is a physical property by virtue of which an oil is able to form, retain and offer resistance to shearing a buffer film-under heat and pressure. The greater the heat and pressure, the greater viscosity is required of a lubricant to prevent thinning and squeezing out of the film. The fundamental meaning of viscosity may be understood by considering a flat plate moving under a force *P* parallel to a stationary plate, the two plates being separated by a thin film of a fluid lubricant of thickness *h*, as shown in Fig.6. The particles of the lubricant adhere strongly to the moving and stationary plates. The motion is accompanied by a linear slip or shear between the particles throughout the entire height (*h*) of the film thickness. If *A* is the area of the plate in contact with the lubricant, then the unit shear stress is given by

$$\boldsymbol{\tau = \frac{P}{A}}$$

According to Newton's law of viscous flow, the magnitude of this shear stress varies directly with the velocity gradient (dV / dy). It is assumed that

(*a*) The lubricant completely fills the space between the two surfaces,
(*b*) The velocity of the lubricant at each surface is same as that of the surface, and
(*c*) Any flow of the lubricant perpendicular to the velocity of the plate is negligible.

$$\boldsymbol{\tau = \frac{P}{A} \propto \frac{dV}{dy} \quad or \quad \tau = Z \times \frac{dV}{dy}}$$

Where *Z* is a constant of proportionality and is known as ***absolute viscosity*** (or simply viscosity) of the lubricant.

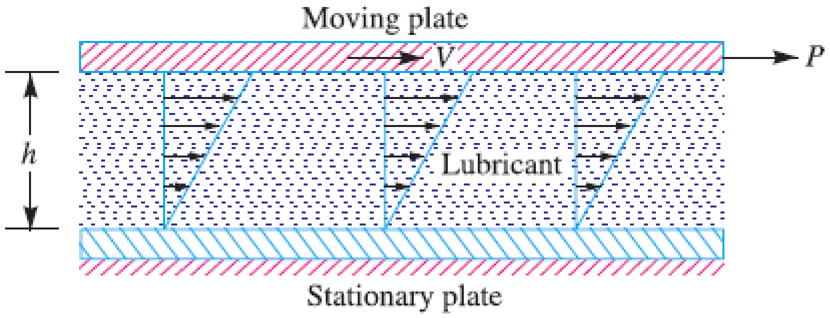


Figure 6

When the thickness of the fluid lubricant is small which is the case for bearings, then the velocity gradient is very nearly constant as shown in Fig.6, so that

$$\frac{dV}{dy} = \frac{V}{y} = \frac{V}{h}$$

$$\therefore \quad \tau = Z \times \frac{V}{h} \qquad or \qquad Z = \tau \times \frac{h}{V}$$

When τ is in N/m^2, h is in meters and V is in m/s, then the unit of absolute viscosity is given by

$$Z = \tau \times \frac{h}{V} = \frac{\text{N}}{\text{m}^2} \times \frac{\text{m}}{\text{m/s}} = \text{N-s/m}^2$$

However, the common practice is to express the absolute viscosity in mass units, such that

$$1 \text{ N-s/m}^2 = \frac{1\text{kg-m}}{s^2} \times \frac{\text{s}}{\text{m}^2} = 1 \text{ kg/m-s}$$

Thus the unit of absolute viscosity in S.I. units is kg / m-s.

The viscosity of the lubricant is measured by Saybolt universal viscometer. It determines the time required for a standard volume of oil at a certain temperature to flow under a certain head through a tube of standard diameter and length. The time so determined in seconds is the Saybolt universal viscosity. In order to convert Saybolt universal viscosity in seconds to absolute viscosity (in kg / m-s), the following formula may be used:

$$Z = \text{Sp. gr. of oil} \left(0.000\,22\,S - \frac{0.18}{S}\right) \text{kg/m-s}$$

where Z = Absolute viscosity at temperature t in kg / m-s, and

S = Saybolt universal viscosity in seconds.

2. Oiliness. It is a joint property of the lubricant and the bearing surfaces in contact. It is a measure of the lubricating qualities under boundary conditions where base metal to metal is prevented only by absorbed film. There is no absolute measure of oiliness.

3. *Density.* This property has no relation to lubricating value but is useful in changing the kinematic viscosity to absolute viscosity.

Mathematically Absolute viscosity = $\rho \times$ Kinematic viscosity (in m^2/s)

Where ρ = Density of the lubricating oil.

The density of most of the oils at 15.5°C varies from 860 to 950 kg / m^3 (the average value may be taken as 900 kg / m^3). The density at any other temperature (t) may be obtained from the following relation, *i.e*.

$$\rho_t = \rho_{15.5} - 0.000\,657\,t$$

Where $\rho_{15.5}$ = Density of oil at 15.5° C.

4. *Viscosity index.* The term viscosity index is used to denote the degree of variation of viscosity with temperature.

5. *Flash point.* It is the lowest temperature at which an oil gives off sufficient vapor to support a momentary flash without actually setting fire to the oil when a flame is brought within 6 mm at the surface of the oil.

6. *Fire point.* It is the temperature at which an oil gives off sufficient vapor to burn it continuously when ignited.

7. *Pour point or freezing point.* It is the temperature at which an oil will cease to flow when cooled.

A hydrodynamic journal bearing is shown in Fig.7, in which *O* is the center of the journal and *O* ‘is the center of the bearing.

Let *D* = Diameter of the bearing,

d = Diameter of the journal,

And *l* = Length of the bearing.

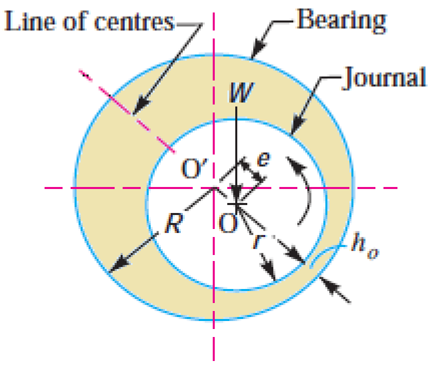


Figure 7

The following terms used in hydrodynamic journal bearing are important from the subject point of view:

1. *Diametral clearance.* It is the difference between the diameters of the bearing and the journal. Mathematically, diametral clearance,

$c = D - d$

Note: The diametral clearance (c) in a bearing should be small enough to produce the necessary velocity gradient, so that the pressure built up will support the load. Also the small clearance has the advantage of decreasing side leakage. However, the allowance must be made for manufacturing tolerances in the journal and bushing. A commonly used clearance in industrial machines is 0.025 mm per cm of journal diameter.

2. *Radial clearance.* It is the difference between the radii of the bearing and the journal. Mathematically, radial clearance,

$$c_1 = R - r = \frac{D-d}{2} = \frac{c}{2}$$

3. *Diametral clearance ratio.* It is the ratio of the diametral clearance to the diameter of the journal. Mathematically, diametral clearance ratio

$$= \frac{c}{d} = \frac{D-d}{d}$$

4. *Eccentricity.* It is the radial distance between the center (O) of the bearing and the displaced center (O') of the bearing under load. It is denoted by e.

5. *Minimum oil film thickness.* It is the minimum distance between the bearing and the journal, under complete lubrication condition. It is denoted by h_0 and occurs at the line of centers as shown in Fig.7. Its value may be assumed as $c / 4$.

6. *Attitude or eccentricity ratio.* It is the ratio of the eccentricity to the radial clearance. Mathematically, attitude or eccentricity ratio,

$$\varepsilon = \frac{e}{c_1} = \frac{c_1 - h_o}{c_1} = 1 - \frac{h_o}{c_1} = 1 - \frac{2h_o}{c}$$

7. Short and long bearing. If the ratio of the length to the diameter of the journal (*i.e.* l / d) is less than 1, then the bearing is said to be **short bearing.** On the other hand, if l / d is greater than 1, then the bearing is known as ***long bearing.***

Notes:

1. When the length of the journal (l) is equal to the diameter of the journal (d), then the bearing is called ***square bearing.***

2. Because of the side leakage of the lubricant from the bearing, the pressure in the film is atmospheric at the ends of the bearing. The average pressure will be higher for a long bearing than for a short or square bearing. Therefore, from the stand point of side leakage, a bearing with a large l / d ratio is preferable. However, space requirements, manufacturing, tolerances and shaft deflections are better met with a short bearing. The value of l / d may be taken as 1 to 2 for general industrial machinery. In crank shaft bearings, the l / d ratio is frequently less than 1.

<u>Bearing Characteristic Number and Bearing Modulus for Journal Bearings</u>

The coefficient of friction in design of bearings is of great importance, because it affords a means for determining the loss of power due to bearing friction. It has been shown by experiments that the coefficient of friction for a full lubricated journal bearing is a function of three variables, *i.e.*

(i) $\frac{ZN}{p}$ (ii) $\frac{d}{c}$ (iii) $\frac{l}{d}$

Therefore the coefficient of friction may be expressed as

$$\mu = \varphi\left(\frac{ZN}{p}, \frac{d}{c}, \frac{l}{d}\right)$$

Where μ = Coefficient of friction,

φ = A functional relationship,

Z = Absolute viscosity of the lubricant, in kg / m-s,

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

p = Bearing pressure on the projected bearing area in N/mm2,
= Load on the journal ÷ $l \times d$
d = Diameter of the journal,
l = Length of the bearing, and
c = Diametral clearance.
The factor ZN / p is termed as ***bearing characteristic number*** and is a dimensionless number.

Coefficient of Friction for Journal Bearings

In order to determine the coefficient of friction for well lubricated full journal bearings, the following empirical relation established by McKee based on the experimental data, may be used.

Coefficient of friction,

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + k$$ (When Z is in kg / m-s and p is in N / mm^2)

Where Z, N, p, d and c have usual meanings as discussed in previous article, and
k = Factor to correct for end leakage. It depends upon the ratio of length to the diameter of the bearing (*i.e.* l / d).
= 0.002 for l / d ratios of 0.75 to 2.8.
The operating values of ZN / p should be compared with values given in special Tables to ensure safe margin between operating conditions and the point of film breakdown.

Table 26.3. Design values for journal bearings.

Machinery	*Bearing*	*Maximum bearing pressure (p) in N/mm²*	*Operating values*			
			Absolute Viscosity (Z) in kg/m-s	*ZN/p Z in kg/m-s p in N/mm²*	$\frac{c}{d}$	$\frac{l}{d}$
Automobile and air-craft engines	Main Crank pin Wrist pin	5.6 – 12 10.5 – 24.5 16 – 35	0.007 0.008 0.008	2.1 1.4 1.12	—	0.8 – 1.8 0.7 – 1.4 1.5 – 2.2
Four stroke-Gas and oil engines	Main Crank pin Wrist pin	5 – 8.5 9.8 – 12.6 12.6 – 15.4	0.02 0.04 0.065	2.8 1.4 0.7	0.001	0.6 – 2 0.6 – 1.5 1.5 – 2
Two stroke-Gas and oil engines	Main Crank pin Wrist pin	3.5 – 5.6 7 – 10.5 8.4 – 12.6	0.02 0.04 0.065	3.5 1.8 1.4	0.001	0.6 – 2 0.6 – 1.5 1.5 – 2
Marine steam engines	Main Crank pin Wrist pin	3.5 4.2 10.5	0.03 0.04 0.05	2.8 2.1 1.4	0.001	0.7 – 1.5 0.7 – 1.2 1.2 – 1.7
Stationary, slow speed steam engines	Main Crank pin Wrist pin	2.8 10.5 12.6	0.06 0.08 0.06	2.8 0.84 0.7	0.001	1 – 2 0.9 – 1.3 1.2 – 1.5
Stationary, high speed steam engine	Main Crank pin Wrist pin	1.75 4.2 12.6	0.015 0.030 0.025	3.5 0.84 0.7	0.001	1.5 – 3 0.9 – 1.5 13 – 1.7
Reciprocating pumps and compressors	Main Crank pin Wrist pin	1.75 4.2 7.0	0.03 0.05 0.08	4.2 2.8 1.4	0.001	1 – 2.2 0.9 – 1.7 1.5 – 2.0
Steam locomotives	Driving axle Crank pin Wrist pin	3.85 14 28	0.10 0.04 0.03	4.2 0.7 0.7	0.001	1.6 – 1.8 0.7 – 1.1 0.8 – 1.3
Railway cars	Axle	3.5	0.1	7	0.001	1.8 – 2
Steam turbines	Main	0.7 – 2	0.002 – 0.016	14	0.001	1 – 2
Generators, motors, centrifugal pumps	Rotor	0.7 – 1.4	0.025	28	0.0013	1 – 2
Transmission shafts	Light, fixed Self -aligning Heavy	0.175 1.05 1.05	0.025- 0.060	7 2.1 2.1	0.001	2 – 3 2.5 – 4 2– 3
Machine tools	Main	2.1	0.04	0.14	0.001	1–4
Punching and shearing machines	Main Crank pin	28 56	0.10	—	0.001	1–2
Rolling Mills	Main	21	0.05	1.4	0.0015	1–1.5

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,

$$p = \frac{ZN}{4.75\times10^6}\left(\frac{d}{c}\right)^2\left(\frac{l}{d+l}\right)\ N/mm^2$$ ...(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,

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

For design purposes, its value is taken as follows:

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

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

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 = $p\ (l \times d)$,

V = Rubbing velocity in m/s $= \frac{\pi dN}{60}$

d is in meters, 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,

$$Q_d = C.A\ (t_b - t_a)\ \frac{J}{s}\ or\ W$$

Where *C* = Heat dissipation coefficient in $W/m^2/°C$,

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 420 $W/m^2/°C$
For well ventilated bearings = 490 to 1400 $W/m^2/°C$

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,

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

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,

$Q_t = m \cdot S \cdot t$ 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.