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CHAPTER

26

Sliding Contact Bearings

26.1 Introduction

A bearing is a machine element which support another moving machine element (known as journal). It permits a relative motion between the contact surfaces of the members, while carrying the load. A little consideration will show that due to the relative motion between the contact surfaces, a certain amount of power is wasted in overcoming frictional resistance and if the rubbing surfaces are in direct contact, there will be rapid wear. In order to reduce frictional resistance and wear and in some cases to carry away the heat generated, a layer of fluid (known as lubricant) may be provided. The lubricant used to separate the journal and bearing is usually a mineral oil refined from petroleum, but vegetable oils, silicon oils, greases etc., may be used.

26.2 Classification of Bearings

Though the bearings may be classified in many ways, yet the following are important from the subject point of view:

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Roller Bearing

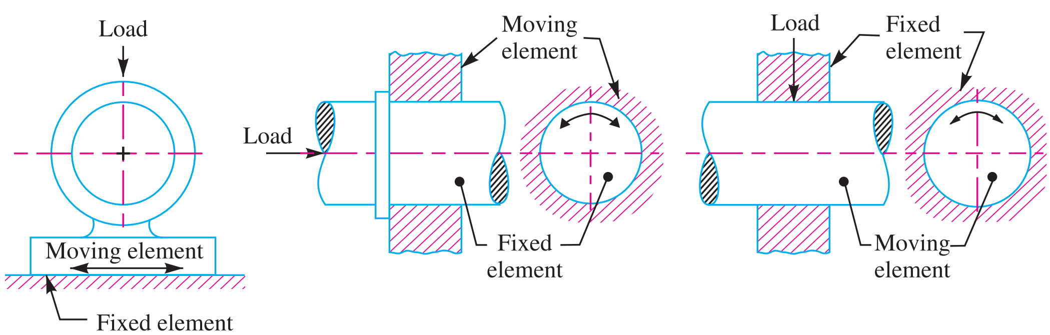
1. *Depending upon the direction of load to be supported.* The bearings under this group are classified as:

(*a*) Radial bearings, and **(*b*)** Thrust bearings.

In ***radial bearings***, the load acts perpendicular to the direction of motion of the moving element as shown in Fig. 26.1 (*a*) and (*b*).

In ***thrust bearings,*** the load acts along the axis of rotation as shown in Fig. 26.1 (*c*).

Note : These bearings may move in either of the directions as shown in Fig. 26.1.



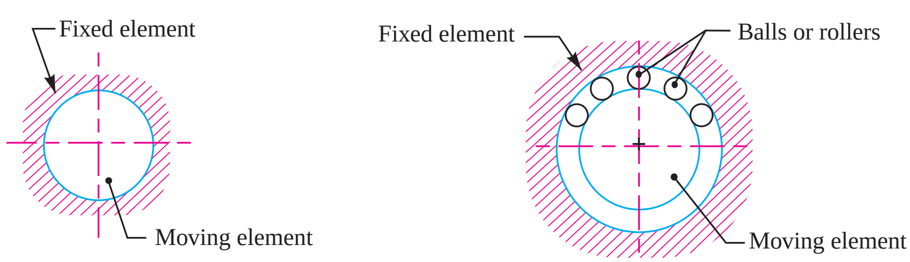
(*a*) Radial bearing. (*b*) Radial bearing. (*c*) Thrust bearing.

Fig. 26.1. Radial and thrust bearings.

2. *Depending upon the nature of contact.* The bearings under this group are classified as :

(*a*) Sliding contact bearings, and **(*b*)** Rolling contact bearings.

In ***sliding contact bearings,*** as shown in Fig. 26.2 (*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.

Fig. 26.2. Sliding and rolling contact bearings.

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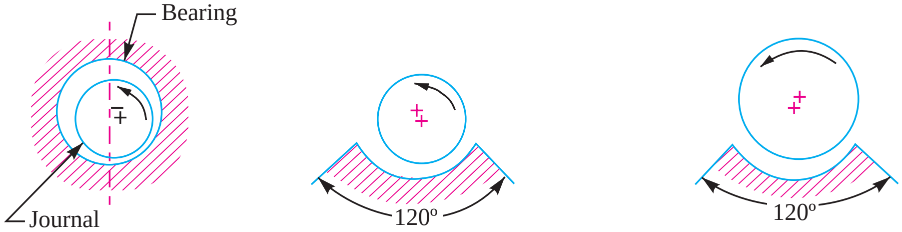
In ***rolling contact bearings,*** as shown in Fig. 26.2 (*b*), the steel balls or rollers, are interposed between the moving and fixed elements. The balls offer rolling friction at two points for each ball or roller.

26.3 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 Fig. 26.1 (*a*), may be called ***slipper*** or ***guide bearings.*** Such type of bearings are usually found in cross-head of steam engines.



(*a*) Full journal bearing. (*b*) Partial journal bearing. (*c*) Fitted journal bearing.

Fig. 26.3. Journal or sleeve bearings.

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 Fig. 26.3 (*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 Fig. 26.3 (*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.

Sliding contact bearings are used in steam engines

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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 Fig. 26.3 (*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. ***Thick film bearings.*** 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. ***Thin film bearings.*** 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. ***Zero film bearings.*** The zero film bearings are those which operate without any lubricant present.
4. ***Hydrostatic or externally pressurized lubricated bearings.*** 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.

26.4 Hydrodynamic Lubricated Bearings

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

Hydrodynamic Lubricated Bearings

1. the flow of a viscous fluid in a converging channel (known as ***wedge film lubrication***), or
2. the resistance of a viscous fluid to being squeezed out from between approaching surfaces (known as ***squeeze film lubrication***).

26.5 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.

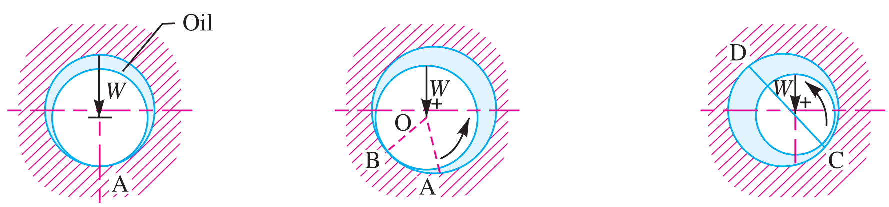
26.6 Important Factors for the Formation of Thick Oil Film in Hydrodynamic Lubricated Bearings

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.

26.7 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 (non-rotating) bearing and a rotating journal. Fig. 26.4 (*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. 26.4 (*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.



(*a*) At rest. (*b*) Slow speed. (*c*) High speed.

Fig. 26.4. Wedge film journal bearing.

When the speed of the journal is increased, a continuous fluid film is established as in Fig. 26.4 (*c*). The centre 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 Fig. 26.4 (*c*), the film is diverging and cannot give rise to a positive pressure or a supporting action.

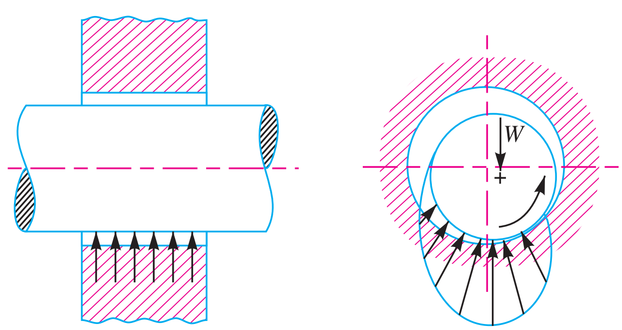


Fig. 26.5. Variation of pressure in the converging film.

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Fig. 26.5 shows the two views of the bearing shown in Fig. 26.4 (*c*), 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°.

26.8 Squeeze Film Journal Bearing

We have seen in the previous article that in a wedge film journal bearing, the bearing carries a steady load and the journal rotates relative to the bearing. But in certain cases, the bearings oscillate or rotate so slowly that the wedge film cannot provide a satisfactory film thickness. If the load is uniform or varying in magnitude while acting in a constant direction, this becomes a thin film or possibly a zero film problem. But if the load reverses its direction, the squeeze film may develop sufficient capacity to carry the dynamic loads without contact between the journal and the bearing. Such bearings are known as ***squeeze film journal bearing.***

Journal bearing

26.9 Properties of Sliding Contact Bearing Materials

When the journal and the bearings are having proper lubrication *i.e.* there is a film of clean, non-corrosive lubricant in between, separating the two surfaces in contact, the only requirement of the bearing material is that they should have sufficient strength and rigidity. However, the conditions under which bearings must operate in service are generally far from ideal and thus the other properties as discussed below must be considered in selecting the best material.

1. ***Compressive strength.*** The maximum bearing pressure is considerably greater than the average pressure obtained by dividing the load to the projected area. Therefore the bearing material should have high compressive strength to withstand this maximum pressure so as to prevent extrusion or other permanent deformation of the bearing.

2. ***Fatigue strength.*** The bearing material should have sufficient fatigue strength so that it can withstand repeated loads without developing surface fatigue cracks. It is of major importance in aircraft and automotive engines.

3. ***Comformability.*** It is the ability of the bearing material to accommodate shaft deflections and bearing inaccuracies by plastic deformation (or creep) without excessive wear and heating.

4. ***Embeddability.*** It is the ability of bearing material to accommodate (or embed) small particles of dust, grit etc., without scoring the material of the journal.

5. ***Bondability.*** Many high capacity bearings are made by bonding one or more thin layers of a bearing material to a high strength steel shell. Thus, the strength of the bond *i.e.* bondability is an important consideration in selecting bearing material.

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6. *Corrosion resistance.* The bearing material should not corrode away under the action of lubricating oil. This property is of particular importance in internal-combustion engines where the same oil is used to lubricate the cylinder walls and bearings. In the cylinder, the lubricating oil comes into contact with hot cylinder walls and may oxidise and collect carbon deposits from the walls.

7. *Thermal conductivity.* The bearing material should be of high thermal conductivity so as to permit the rapid removal of the heat generated by friction.

8. *Thermal expansion.* The bearing material should be of low coefficient of thermal expansion, so that when the bearing operates over a wide range of temperature, there is no undue change in the clearance.

Marine bearings

All these properties as discussed above are, however, difficult to find in any particular bearing material. The various materials are used in practice, depending upon the requirement of the actual service conditions. The choice of material for any application must represent a compromise. The following table shows the comparison of some of the properties of more common metallic bearing materials.

Table 26.1. Properties of metallic bearing materials.

Bearing material	*Fatigue strength*	*Comformability*	*Embeddability*	*Anti scoring*	*Corrosion resistance*	*Thermal conductivity*
Tin base babbit	Poor	Good	Excellent	Excellent	Excellent	Poor
Lead base babbit	Poor to fair	Good	Good	Good to excellent	Fair to good	Poor
Lead bronze	Fair	Poor	Poor	Poor	Good	Fair
Copper lead	Fair	Poor	Poor to fair	Poor to fair	Poor to fair	Fair to good
Aluminium	Good	Poor to fair	Poor	Good	Excellent	Fair
Silver	Excellent	Almost none	Poor	Poor	Excellent	Excellent
Silver lead deposited	Excellent	Excellent	Poor	Fair to good	Excellent	Excellent

26.10 Materials used for Sliding Contact Bearings

The materials commonly used for sliding contact bearings are discussed below :

1. *Babbit metal.* The tin base and lead base babbits are widely used as a bearing material, because they satisfy most requirements for general applications. The babbits are recommended where the maximum bearing pressure (on projected area) is not over 7 to 14 N/mm^2. When applied in

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automobiles, the babbit is generally used as a thin layer, 0.05 mm to 0.15 mm thick, bonded to an insert or steel shell. The composition of the babbit metals is as follows :

***Tin base babbits* :** Tin 90% ; Copper 4.5% ; Antimony 5% ; Lead 0.5%.

***Lead base babbits* :** Lead 84% ; Tin 6% ; Anitmony 9.5% ; Copper 0.5%.

2. *Bronzes.* The bronzes (alloys of copper, tin and zinc) are generally used in the form of machined bushes pressed into the shell. The bush may be in one or two pieces. The bronzes commonly used for bearing material are gun metal and phosphor bronzes.

The ***gun metal*** (Copper 88% ; Tin 10% ; Zinc 2%) is used for high grade bearings subjected to high pressures (not more than 10 N/mm^2 of projected area) and high speeds.

The ***phosphor bronze*** (Copper 80% ; Tin 10% ; Lead 9% ; Phosphorus 1%) is used for bearings subjected to very high pressures (not more than 14 N/mm^2 of projected area) and speeds.

3. *Cast iron.* The cast iron bearings are usually used with steel journals. Such type of bearings are fairly successful where lubrication is adequate and the pressure is limited to 3.5 N/mm^2 and speed to 40 metres per minute.

4. *Silver.* The silver and silver lead bearings are mostly used in aircraft engines where the fatigue strength is the most important consideration.

5. *Non-metallic bearings.* The various non-metallic bearings are made of carbon-graphite, rubber, wood and plastics. The **carbon-graphite bearings** are self lubricating, dimensionally stable over a wide range of operating conditions, chemically inert and can operate at higher temperatures than other bearings. Such type of bearings are used in food processing and other equipment where contamination by oil or grease must be prohibited. These bearings are also used in applications where the shaft speed is too low to maintain a hydrodynamic oil film.

The ***soft rubber bearings*** are used with water or other low viscosity lubricants, particularly where sand or other large particles are present. In addition to the high degree of embeddability and comformability, the rubber bearings are excellent for absorbing shock loads and vibrations. The rubber bearings are used mainly on marine propeller shafts, hydraulic turbines and pumps.

The ***wood bearings*** are used in many applications where low cost, cleanliness, inattention to lubrication and anti-seizing are important.

Industrial bearings.

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The commonly used plastic material for bearings is ***Nylon*** and ***Teflon.*** These materials have many characteristics desirable in bearing materials and both can be used dry *i.e.* as a zero film bearing. The Nylon is stronger, harder and more resistant to abrasive wear. It is used for applications in which these properties are important *e.g.* elevator bearings, cams in telephone dials etc. The Teflon is rapidly replacing Nylon as a wear surface or liner for journal and other sliding bearings because of the following properties:

1. It has lower coefficient of friction, about 0.04 (dry) as compared to 0.15 for Nylon.
2. It can be used at higher temperatures up to about 315°C as compared to 120°C for Nylon.
3. It is dimensionally stable because it does not absorb moisture, and
4. It is practically chemically inert.

26.11 Lubricants

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.

Wherever moving and rotating parts are present proper lubrication is essential to protect the moving parts from wear and tear and reduce friction.

26.12 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. 26.6. 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

$$\tau = P / A$$

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.

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

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

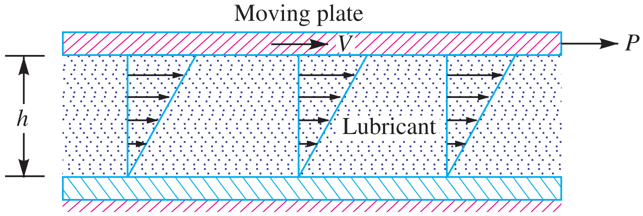

Fig. 26.6. Viscosity.

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. 26.6, so that

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

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

When τ is in N/m^2, h is in metres 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

$$1 \text{ N-s / m}^2 = \frac{1 \text{kg-m}}{s^2} \times \frac{\text{s}}{\text{m}^2} = 1 \text{ kg / m-s} \qquad \ldots (\because\ 1 \text{ N} = 1 \text{ kg-m / s}^2)$$

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

The viscocity 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:

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

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

S = Saybolt universal viscosity in seconds.

The variation of absolute viscosity with temperature for commonly used lubricating oils is shown in Table 26.2 on the next page.

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.

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Table 26.2. Absolute viscosity of commonly used lubricating oils.

S. No.	Type of oil	Absolute viscosity in kg / m-s at temperature in °C											
		30	*35*	*40*	*45*	*50*	*55*	*60*	*65*	*70*	*75*	*80*	*90*
1.	SAE 10	0.05	0.036	0.027	0.0245	0.021	0.017	0.014	0.012	0.011	0.009	0.008	0.005
2.	SAE 20	0.069	0.055	0.042	0.034	0.027	0.023	0.020	0.017	0.014	0.011	0.010	0.0075
3.	SAE 30	0.13	0.10	0.078	0.057	0.048	0.040	0.034	0.027	0.022	0.019	0.016	0.010
4.	SAE 40	0.21	0.17	0.12	0.096	0.78	0.06	0.046	0.04	0.034	0.027	0.022	0.013
5.	SAE 50	0.30	0.25	0.20	0.17	0.12	0.09	0.076	0.06	0.05	0.038	0.034	0.020
6.	SAE 60	0.45	0.32	0.27	0.20	0.16	0.12	0.09	0.072	0.057	0.046	0.040	0.025
7.	SAE 70	1.0	0.69	0.45	0.31	0.21	0.165	0.12	0.087	0.067	0.052	0.043	0.033

Note : We see from the above table that the viscosity of oil decreases when its temperature increases.

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

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

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 vapour 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 vapour 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.

26.13 Terms used in Hydrodynamic Journal Bearing

A hydrodynamic journal bearing is shown in Fig. 26.7, in which O is the centre of the journal and O' is the centre of the bearing.

Let D = Diameter of the bearing,

d = Diameter of the journal, and

l = Length of the bearing.

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

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

$$c = D - d$$

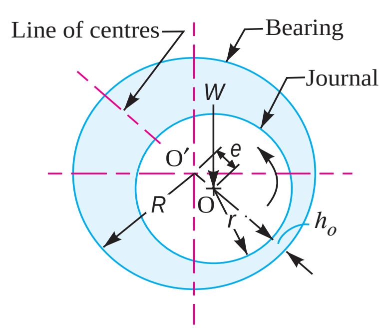

Fig. 26.7. Hydrodynamic journal bearing.

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

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

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4. *Eccentricity.* It is the radial distance between the centre (O) of the bearing and the displaced centre (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 centres as shown in Fig. 26.7. 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,

$$\varepsilon = \frac{e}{c_1} = \frac{c_1 - h_0}{c_1} = 1 - \frac{h_0}{c_1} = 1 - \frac{2h_0}{c} \quad \ldots (\because\ c_1 = c / 2)$$

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.***

Axle bearings

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.

26.14 Bearing Characteristic Number and Bearing Modulus for Journal Bearings

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*) $\dfrac{ZN}{p}$; (*ii*) $\dfrac{d}{c}$; and (*iii*) $\dfrac{l}{d}$

Therefore the coefficient of friction may be expressed as

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

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/mm^2,
= 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. The variation of coefficient of friction with the operating values of bearing characteristic number (ZN / p) as obtained by McKee brothers (S.A. McKee and T.R. McKee) in an actual test of friction is shown in Fig. 26.8. The factor ZN/p helps to predict the performance of a bearing.

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The part of the curve *PQ* represents the region of thick film lubrication. Between *Q* and *R*, the viscosity (*Z*) or the speed (*N*) are so low, or the pressure (*p*) is so great that their combination *ZN / p* will reduce the film thickness so that partial metal to metal contact will result. The thin film or boundary lubrication or imperfect lubrication exists between *R* and *S* on the curve. This is the region where the viscosity of the lubricant ceases to be a measure of friction characteristics but the oiliness of the lubricant is effective in preventing complete metal to metal contact and seizure of the parts.

Clutch bearing

It may be noted that the part *PQ* of the curve represents stable operating conditions, since from any point of stability, a decrease in viscosity (*Z*) will reduce *ZN / p*. This will result in a decrease in coefficient of friction (μ) followed by a lowering of bearing temperature that will raise the viscosity (*Z*).

From Fig. 26.8, we see that the minimum amount of friction occurs at *A* and at this point the value of *ZN / p* is known as ***bearing modulus*** which is denoted by *K*. The bearing should not be operated at this value of bearing modulus, because a slight decrease in speed or slight increase in pressure will break the oil film and make the journal to operate with metal to metal contact. This will result in high friction, wear and heating. In order to prevent such conditions, the bearing should be designed for a value of *ZN / p* at least three times the minimum value of bearing modulus (*K*). If the bearing is subjected to large fluctuations of load and heavy impacts, the value of *ZN / p* = 15 *K* may be used.

From above, it is concluded that when the value of *ZN / p* is greater than *K*, then the bearing will operate with thick film lubrication or under hydrodynamic conditions. On the other hand, when the value of *ZN / p* is less than *K*, then the oil film will rupture and there is a metal to metal contact.

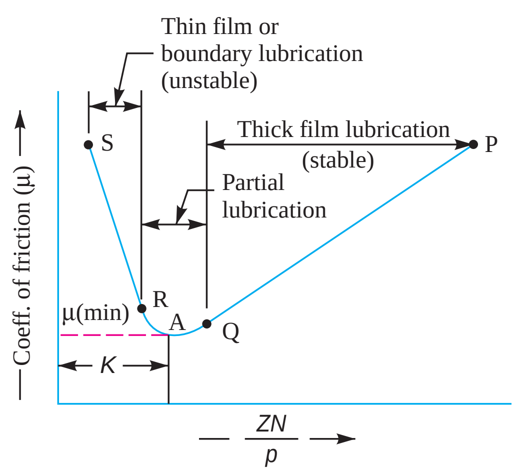

Fig. 26.8. Variation of coefficient of friction with *ZN*/*p*.

26.15 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.

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*Coefficient of friction,

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

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 Table 26.3 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

* This is the equation of a straight line portion in the region of thick film lubrication (*i.e.* line *PQ*) as shown in Fig. 26.8.

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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*	$\frac{c}{d}$	$\frac{l}{d}$
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

26.16 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,

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

26.17 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 :

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

26.18 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,

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

where μ = Coefficient of friction,

W = Load on the bearing in N,

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= 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,

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

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/m^2/°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,

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

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.

26.19 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.

Journal bearings are used in helicopters, primarily in the main rotor axis and in the landing gear for fixed wing aircraft.

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 (μ) 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 atleast 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 26.1. *Design a journal bearing for a centrifugal pump from the following data :*

Load on the journal = 20 000 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 / mm².

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 = 1232 W/m²/°C.

Solution. Given : W = 20 000 N ; N = 900 r.p.m. ; t_0 = 55°C ; Z = 0.017 kg/m-s ; t_a = 15.5°C ; p = 1.5 N/mm² ; t = 10°C ; C = 1232 W/m²/°C

The 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,

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

$$= 0.0051 \times 20\,000\left(\frac{\pi \times 0.1 \times 900}{60}\right) = 480.7 \text{ W}$$

... (d is taken in metres)

7. Heat dissipated,

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

We know that

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

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

... (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

$$= \text{Heat generated} - \text{Heat dissipated} = Q_g - Q_d$$

$$= 480.7 - 389.3 = 91.4 \text{ W}$$

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,

$$Q_t = m.S.t = m \times 1900 \times 10 = 19\,000\, m \text{ W}$$

... [∵ 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.**

Example 26.2. *The load on the journal bearing is 150 kN due to turbine shaft of 300 mm diameter running at 1800 r.p.m. Determine the following :*

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1. Length of the bearing if the allowable bearing pressure is 1.6 N/mm², and

2. Amount of heat to be removed by the lubricant per minute if the bearing temperature is 60°C and viscosity of the oil at 60°C is 0.02 kg/m-s and the bearing clearance is 0.25 mm.

Axle bearing

Solution. Given : $W = 150$ kN $= 150 \times 10^3$ N ; d = 300 mm = 0.3 m ; N = 1800 r.p.m. ; $p = 1.6$ N/mm² ; Z = 0.02 kg / m-s ; c = 0.25 mm

1. Length of the bearing

Let l = Length of the bearing in mm.

We know that projected bearing area,

$$A = l \times d = l \times 300 = 300\, l \text{ mm}^2$$

and allowable bearing pressure (p),

$$1.6 = \frac{W}{A} = \frac{150 \times 10^3}{300\, l} = \frac{500}{l}$$

$\therefore$ $l = 500 / 1.6 = 312.5$ mm **Ans.**

2. Amount of heat to be removed by the lubricant

We know that coefficient of friction for the bearing,

$$\mu = \frac{33}{10^8}\left(\frac{Z.N}{p}\right)\left(\frac{d}{c}\right) + k = \frac{33}{10^8}\left(\frac{0.02 \times 1800}{1.6}\right)\left(\frac{300}{0.25}\right) + 0.002$$

$$= 0.009 + 0.002 = 0.011$$

Rubbing velocity,

$$V = \frac{\pi d.N}{60} = \frac{\pi \times 0.3 \times 1800}{60} = 28.3 \text{ m/s}$$

$\therefore$ Amount of heat to be removed by the lubricant,

$$Q_g = \mu.W.V = 0.011 \times 150 \times 10^3 \times 28.3 = 46\,695 \text{ J/s or W}$$

$$= 46.695 \text{ kW} \quad \textbf{Ans.} \qquad \text{... (1 J/s = 1 W)}$$

Example 26.3. *A full journal bearing of 50 mm diameter and 100 mm long has a bearing pressure of 1.4 N/mm². The speed of the journal is 900 r.p.m. 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/m-s. The room temperature is 35°C. Find : 1. The amount of artificial cooling required, and 2. 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.*

Solution. Given : d = 50 mm = 0.05 m ; l = 100 mm = 0.1 m; p = 1.4 N/mm² ; N = 900 r.p.m. ; d / c = 1000 ; Z = 0.011 kg / m-s ; t_0 = 75°C ; t_a = 35°C ; t = 10°C ; S = 1850 J/kg / °C

1. Amount of artificial cooling required

We know that the coefficient of friction,

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + k = \frac{33}{10^8}\left(\frac{0.011 \times 900}{1.4}\right)(1000) + 0.002$$

$$= 0.002\,33 + 0.002 = 0.004\,33$$

Load on the bearing,

$$W = p \times d.l = 1.4 \times 50 \times 100 = 7000 \text{ N}$$

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and rubbing velocity,

$$V = \frac{\pi d.N}{60} = \frac{\pi \times 0.05 \times 900}{60} = 2.36 \text{ m/s}$$

∴ Heat generated,

$$Q_g = \mu.W.V = 0.004\,33 \times 7000 \times 2.36 = 71.5 \text{ J/s}$$

Let t_b = Temperature of the bearing surface.

We know that

$$(t_b - t_a) = \frac{1}{2}(t_0 - t_a) = \frac{1}{2}(75 - 35) = 20°\text{C}$$

Since the value of heat dissipation coefficient (C) for unventilated bearing varies from 140 to 420 W/m^2/°C, therefore let us take

$$C = 280 \text{ W/m}^2/° \text{ C}$$

We know that heat dissipated,

$$Q_d = C.A\,(t_b - t_a) = C.l.d\,(t_b - t_a)$$
$$= 280 \times 0.05 \times 0.1 \times 20 = 28 \text{ W} = 28 \text{ J/s}$$

∴ Amount of artificial cooling required

$$= \text{Heat generated} - \text{Heat dissipated} = Q_g - Q_d$$
$$= 71.5 - 28 = 43.5 \text{ J/s or W}$$ **Ans.**

2. Mass of the lubricating oil required

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

We know that heat taken away by the oil,

$$Q_t = m.S.t = m \times 1850 \times 10 = 18\,500\, m \text{ J/s}$$

Since the heat generated at the bearing is taken away by the lubricating oil, therefore equating

$$Q_g = Q_t \quad \text{or} \quad 71.5 = 18\,500\, m$$

∴ $m = 71.5 / 18\,500 = 0.003\,86$ kg / s = 0.23 kg / min **Ans.**

Example 26.4. *A 150 mm diameter shaft supporting a load of 10 kN has a speed of 1500 r.p.m. The shaft runs in a bearing whose length is 1.5 times the shaft diameter. If the diametral clearance of the bearing is 0.15 mm and the absolute viscosity of the oil at the operating temperature is 0.011 kg/m-s, find the power wasted in friction.*

Solution. Given : d = 150 mm = 0.15 m ; W = 10 kN = 10 000 N ; N = 1500 r.p.m. ; l = 1.5 d ; c = 0.15 mm ; Z = 0.011 kg/m-s

We know that length of bearing,

$$l = 1.5\, d = 1.5 \times 150 = 225 \text{ mm}$$

∴ Bearing pressure,

$$p = \frac{W}{A} = \frac{W}{l.d} = \frac{10\,000}{225 \times 150} = 0.296 \text{ N/mm}^2$$

We know that coefficient of friction,

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + k = \frac{33}{10^8}\left(\frac{0.011 \times 1500}{0.296}\right)\left(\frac{150}{0.15}\right) + 0.002$$
$$= 0.018 + 0.002 = 0.02$$

and rubbing velocity,

$$V = \frac{\pi d.N}{60} = \frac{\pi \times 0.15 \times 1500}{60} = 11.78 \text{ m/s}$$

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We know that heat generated due to friction,

$$Q_g = \mu.W.V = 0.02 \times 10\,000 \times 11.78 = 2356 \text{ W}$$

$\therefore$ Power wasted in friction

$$= Q_g = 2356 \text{ W} = 2.356 \text{ kW} \quad \textbf{Ans.}$$

Example 26.5. *A 80 mm long journal bearing supports a load of 2800 N on a 50 mm diameter shaft. The bearing has a radial clearance of 0.05 mm and the viscosity of the oil is 0.021 kg / m-s at the operating temperature. If the bearing is capable of dissipating 80 J/s, determine the maximum safe speed.*

Solution. Given : l = 80 mm ; W = 2800 N ; d = 50 mm ; = 0.05 m ; c / 2 = 0.05 mm or c = 0.1 mm ; Z = 0.021 kg/m-s ; Q_d = 80 J/s

Let N = Maximum safe speed in r.p.m.

We know that bearing pressure,

$$p = \frac{W}{l.d} = \frac{2800}{80 \times 50} = 0.7 \text{ N/mm}^2$$

and coefficient of friction,

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + 0.002 = \frac{33}{10^8}\left(\frac{0.021\,N}{0.7}\right)\left(\frac{50}{0.1}\right) + 0.002$$

$$= \frac{495\,N}{10^8} + 0.002$$

Front hub-assembly bearing

$\therefore$ Heat generated,

$$Q_g = \mu.W.V = \mu.W\left(\frac{\pi\,d\,N}{60}\right) \text{J/s}$$

$$= \left(\frac{495\,N}{10^8} + 0.002\right) 2800 \left(\frac{\pi \times 0.05\,N}{60}\right)$$

$$= \frac{3628\,N^2}{10^8} + 0.014\,66\,N$$

Equating the heat generated to the heat dissipated, we have

$$\frac{3628\,N^2}{10^8} + 0.014\,66\,N = 80$$

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or $\quad N^2 + 404\,N - 2.2 \times 10^6 = 0$

$$\therefore \qquad N = \frac{-404 \pm \sqrt{(404)^2 + 4 \times 2.2 \times 10^6}}{2}$$

$$= \frac{-404 \pm 2994}{2} = 1295 \text{ r.p.m} \quad \textbf{Ans.} \qquad \text{... (Taking +ve sign)}$$

Example 26.6. *A journal bearing 60 mm is diameter and 90 mm long runs at 450 r.p.m. The oil used for hydrodynamic lubrication has absolute viscosity of 0.06 kg / m-s. If the diametral clearance is 0.1 mm, find the safe load on the bearing.*

Solution. Given : d = 60 mm = 0.06 m ; l = 90 mm = 0.09 m ; N = 450 r.p.m. ; Z = 0.06 kg / m-s ; c = 0.1 mm

First of all, let us find the bearing pressure (p) by using Sommerfeld number. We know that

$$\frac{Z\,N}{p}\left(\frac{d}{c}\right)^2 = 14.3 \times 10^6$$

$$\frac{0.06 \times 450}{p}\left(\frac{60}{0.1}\right)^2 = 14.3 \times 10^6 \quad \text{or} \quad \frac{9.72 \times 10^6}{p} = 14.3 \times 10^6$$

$$\therefore \qquad p = 9.72 \times 10^6 / 14.3 \times 10^6 = 0.68 \text{ N/mm}^2$$

We know that safe load on the bearing,

$$W = p.A = p.l.d = 0.68 \times 90 \times 60 = 3672 \text{ N} \quad \textbf{Ans.}$$

26.20 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

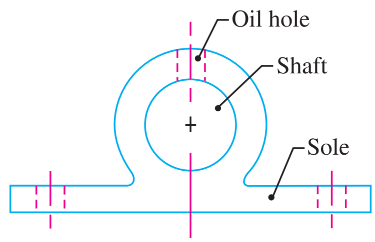

Fig. 26.9. Solid journal bearing.

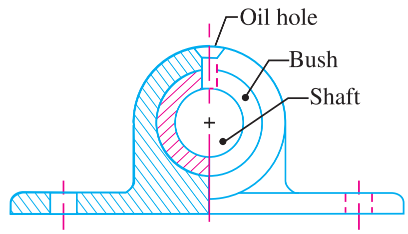

Fig. 26.10. Bushed bearing.

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.

26.21 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.

Bronze bushed bearing assemblies

26.22 Split Bearing or Plummer Block

A split-bearing is used for shafts running at high speeds and carrying heavy loads. A split-bearing, as shown in Fig. 26.11, consists of a cast iron base (also called block or pedestal), gunmetal or phosphor bronze brasses, bushes or steps made in two-halves and a cast iron cap. The two halves of the brasses are held together by a cap or cover by means of mild steel bolts and nuts. Sometimes thin shims are introduced between the cap and the base to provide an adjustment for wear. When the bottom wears out, one or two shims are removed and then the cap is tightened by means of bolts.

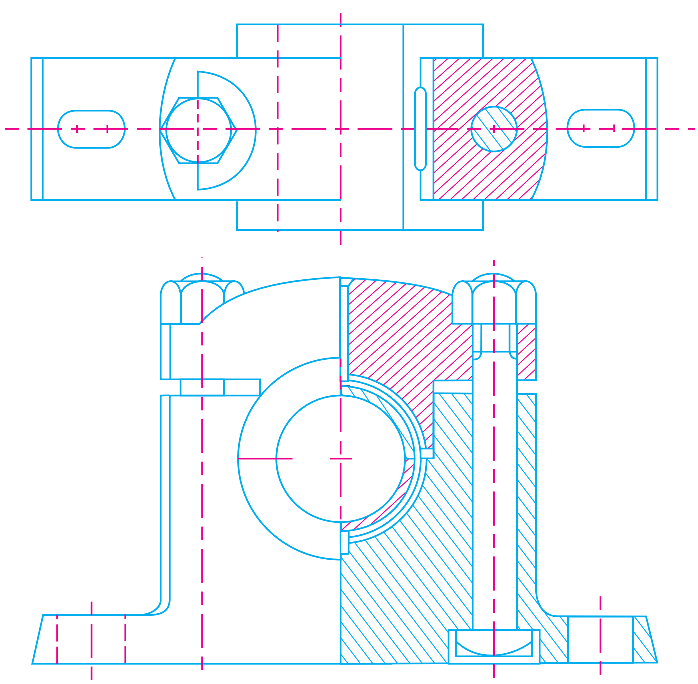

Fig. 26.11. Split bearing or plummer block.

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The brasses are provided with collars or flanges on either side in order to prevent its axial movement. To prevent its rotation along with the shaft, the following four methods are usually used in practice.

1. The sungs are provided at the sides as shown in Fig. 26.12 (*a*).
2. A sung is provided at the top, which fits inside the cap as shown in Fig. 26.12 (*b*). The oil hole is drilled through the sung.
3. The steps are made rectangular on the outside and they are made to fit inside a corresponding hole, as shown in Fig. 26.12 (*c*).
4. The steps are made octagonal on the outside and they are made to fit inside a corresponding hole, as shown in Fig. 26.12 (*d*).

The split bearing must be lubricated properly.

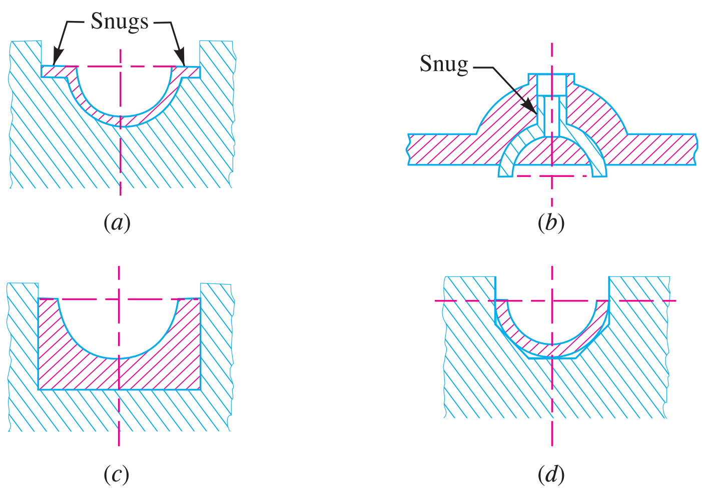

Fig. 26.12. Methods of preventing rotation of brasses.

26.23 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.

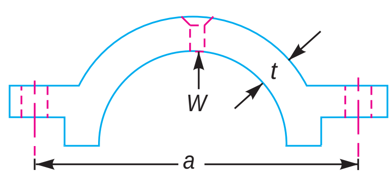

Fig. 26.13. Bearing cap.

Let W = Load supported at the centre,

α = Distance between centres 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,

$$Z = l.t^2 / 6$$

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∴ 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,

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

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

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.

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

From this expression, the core diameter (d_c) may be calculated. After finding the core diameter, the size of the bolt is fixed.

A self-locking nut used in bearing assemblies.

26.24 Oil Groves

The oil grooves are cut into the plain bearing surfaces to assist in the distribution of the oil between the rubbing surfaces. It prevents squeezing of the oil film from heavily loaded low speed journals and bearings. The tendency to squeeze out oil is greater in low speed than in high speed bearings, because the oil has greater wedging action at high speeds. At low speeds, the journal rests upon a given area of oil film for a longer period of time, tending to squeeze out the oil over the area of greatest pressure. The grooves function as oil reservoirs which holds and distributes the oil especially during starting or at very low speeds. The oil grooves are cut at right angles to the line of the load. The circumferential and diagonal grooves should be avoided, if possible. The effectiveness of the oil grooves is greatly enhanced if the edges of grooves are chamfered. The shallow and narrow grooves with chamfered edges distributes the oil more evenly. A chamfered edge should always be provided at the parting line of the bearing.

Example 26.7. *A wall bracket supports a plummer block for 80 mm diameter shaft. The length of bearing is 120 mm. The cap of bearing is fastened by means of four bolts, two on each side of the shaft. The cap is to withstand a load of 16.5 kN. The distance between the centre lines of the bolts is*

150 mm. Determine the thickness of the bearing cap and the diameter of the bolts. Assume safe stresses in tension for the material of the cap, which is cast iron, as 15 MPa and for bolts as 35 MPa. Also check the deflection of the bearing cap taking E = 110 kN / mm².

Solution : Given : $d = 80$ mm ; $l = 120$ mm ; $n = 4$; $W = 16.5$ kN $= 16.5 \times 10^3$ N ; $a = 150$ mm ; $\sigma_b = 15$ MPa $= 15$ N/mm^2; $\sigma_t = 35$ MPa $= 35$ N/mm^2 ; $E = 110$ kN/mm^2 $= 110 \times 10^3$ N/mm^2

Thickness of the bearing cap

We know that thickness of the bearing cap,

$$t = \sqrt{\frac{3\,W.a}{2\sigma_b.l}} = \sqrt{\frac{3 \times 16.5 \times 10^3 \times 150}{2 \times 15 \times 120}} = \sqrt{2062.5}$$

$$= 45.4 \text{ say } 46 \text{ mm } \textbf{Ans.}$$

Diameter of the bolts

Let $\quad d_c$ = Core diameter of the bolts.

We know that

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

or

$$\frac{\pi}{4}(d_c)^2\,35 = \frac{4}{3} \times \frac{16.5 \times 10^3}{4} = 5.5 \times 10^3$$

$$\therefore \quad (d_c)^2 = \frac{5.5 \times 10^3 \times 4}{\pi \times 35} = 200 \quad \text{or} \quad d_c = 14.2 \text{ mm } \textbf{Ans.}$$

Deflection of the cap

We know that deflection of the cap,

$$\delta = \frac{W.a^3}{4\,E.l.t^3} = \frac{16.5 \times 10^3\,(150)^3}{4 \times 110 \times 10^3 \times 120\,(46)^3} = 0.0108 \text{ mm } \textbf{Ans.}$$

Since the limited value of the deflection is 0.025 mm, therefore the above value of deflection is within limits.

26.25 Thrust Bearings

A thrust bearing is used to guide or support the shaft which is subjected to a load along the axis of the shaft. Such type of bearings are mainly used in turbines and propeller shafts. The thrust bearings are of the following two types :

1. Foot step or pivot bearings, and **2.** Collar bearings.

In a ***foot step*** or ***pivot bearing,*** the loaded shaft is vertical and the end of the shaft rests within the bearing. In case of ***collar bearing,*** the shaft continues through the bearing. The shaft may be vertical or horizontal with single collar or many collars. We shall now discuss the design aspects of these bearings in the following articles.

Footstep bearing

26.26 Footstep or Pivot Bearings

A simple type of footstep bearing, suitable for a slow running and lightly loaded shaft, is shown in Fig. 26.14. If the shaft is not of steel, its end must be fitted with a steel face. The shaft is guided in a gunmetal bush, pressed into the pedestal and prevented from turning by means of a pin.

Since the wear is proportional to the velocity of the rubbing surface, which (*i.e.* rubbing velocity) increases with the distance from the axis (*i.e.* radius) of the bearing, therefore the wear will be different at different radii. Due to this wear, the distribution of pressure over the bearing surface is not

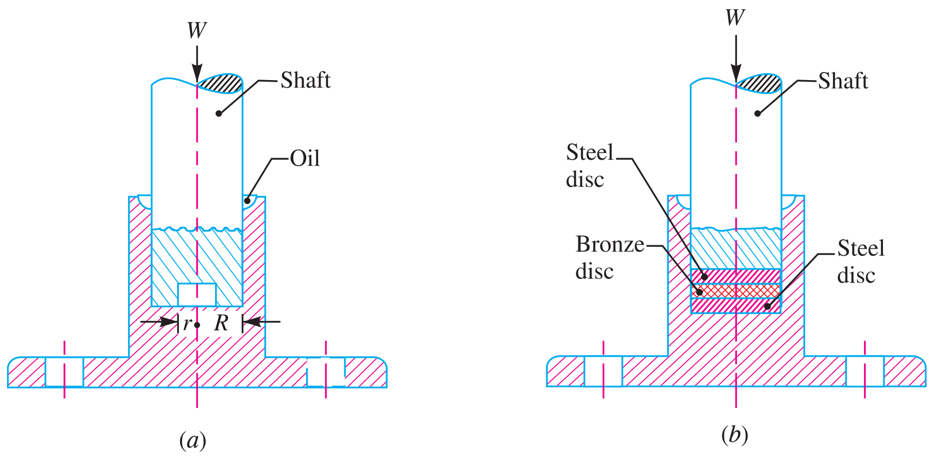

Fig. 26.14. Footstep or pivot bearings.

uniform. It may be noted that the wear is maximum at the outer radius and zero at the centre. In order to compensate for end wear, the following two methods are employed.

1. The shaft is counter-bored at the end, as shown in Fig. 26.14 (*a*).

2. The shaft is supported on a pile of discs. It is usual practice to provide alternate discs of different materials such as steel and bronze, as shown in Fig. 26.14 (*b*), so that the next disc comes into play, if one disc seizes due to improper lubrication.

It may be noted that a footstep bearing is difficult to lubricate as the oil is being thrown outwards from the centre by centrifugal force.

In designing, it is assumed that the pressure is uniformly distributed throughout the bearing surface.

Let W = Load transmitted over the bearing surface,

R = Radius of the bearing surface (or shaft),

A = Cross-sectional area of the bearing surface,

p = Bearing pressure per unit area of the bearing surface between rubbing surfaces,

μ = Coefficient of friction, and

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

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

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

and the total frictional torque,

$$T = \frac{2}{3}\mu.W.R$$

∴ Power lost in friction,

$$P = \frac{2\pi N.T}{60} \text{ watts} \qquad \text{... (} T \text{ being in N-m)}$$

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Notes : 1. When the counter-boring of the shaft is considered, then the bearing pressure,

$$p = \frac{W}{\pi(R^2 - r^2)}, \quad \text{where } r = \text{Radius of counter-bore,}$$

and the total frictional torque,

$$T = \frac{2}{3}\mu.W\left(\frac{R^3 - r^3}{R^2 - r^2}\right)$$

2. The allowable bearing pressure (p) for the footstep bearings may be taken as follows :

(a) For rubbing speeds (V) from 15 to 60 m/min, the bearing pressure should be such that $p.V \leq 42$, when p is in N/mm^2 and V in m/min.

(b) For rubbing speeds over 60 m/min., the pressure should not exceed 0.7 N/mm^2.

(c) For intermittent service, the bearing pressure may be taken as 10.5 N/mm^2.

(d) For very slow speeds, the bearing pressure may be taken as high as 14 N/mm^2.

3. The coefficient of friction for the footstep bearing may be taken as 0.015.

26.27 Collar Bearings

We have already discussed that in a collar bearing, the shaft continues through the bearing. The shaft may be vertical or horizontal, with single collar or many collars. A simple multicollar bearing for horizontal shaft is shown in Fig. 26.15. The collars are either integral parts of the shaft or rigidly fastened to it. The outer diameter of the collar is usually taken as 1.4 to 1.8 times the inner diameter of the collar (*i.e.* diameter of the shaft). The thickness of the collar is kept as one-sixth diameter of the shaft and clearance between collars as one-third diameter of the shaft. In designing collar bearings, it is assumed that the pressure is uniformly distributed over the bearing surface.

Collar bearings

Let
W = Load transmitted over the bearing surface,
n = Number of collars,
R = Outer radius of the collar,
r = Inner radius of the collar,
A = Cross-sectional area of the bearing surface = $n\,\pi\,(R^2 - r^2)$,
p = Bearing pressure per unit area of the bearing surface, between rubbing surfaces,
μ = Coefficient of friction, and
N = Speed of the shaft in r.p.m.

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

$$p = \frac{W}{A} = \frac{W}{n.\pi\,(R^2 - r^2)}$$

and the total frictional torque,

$$T = \frac{2}{3}\mu.\,W\left(\frac{R^3 - r^3}{R^2 - r^2}\right)$$

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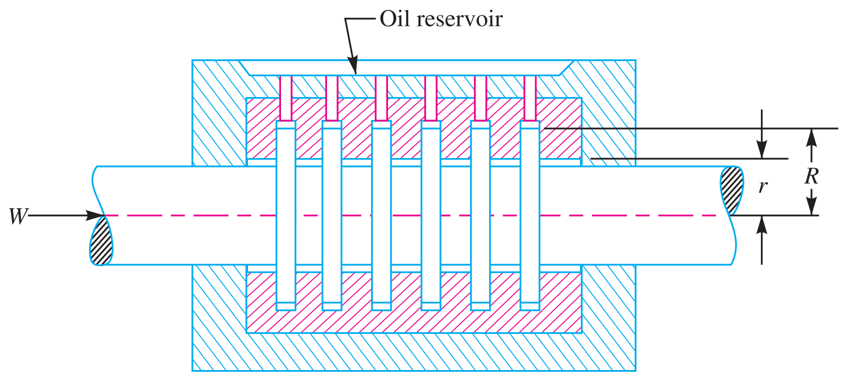

Fig. 26.15. Collar bearing.

∴ Power lost in friction,

$$P = \frac{2\pi\ N.T}{60} \text{ watts} \qquad \text{... (when } T \text{ is in N-m)}$$

Notes : 1. The coefficient of friction for the collar bearings may be taken as 0.03 to 0.05.

2. The bearing pressure for a single collar and water cooled multi-collared bearings may be taken same as for footstep bearings.

Example 26.8. *A footstep bearing supports a shaft of 150 mm diameter which is counter-bored at the end with a hole diameter of 50 mm. If the bearing pressure is limited to 0.8 N/mm² and the speed is 100 r.p.m.; find : 1. The load to be supported; 2. The power lost in friction; and 3. The heat generated at the bearing.*

Assume coefficient of friction = 0.015.

Solution. Given : D = 150 mm or R = 75 mm ; d = 50 mm or r = 25 mm ; p = 0.8 N/mm^2 ; N = 100 r.p.m. ; μ = 0.015

1. Load to be supported

Let W = Load to be supported.

Assuming that the pressure is uniformly distributed over the bearing surface, therefore bearing pressure (p),

$$0.8 = \frac{W}{\pi\,(R^2 - r^2)} = \frac{W}{\pi\,[(75)^2 - (25)^2]} = \frac{W}{15\,710}$$

∴ $W = 0.8 \times 15\,710 = 12\,568$ N **Ans.**

2. Power lost in friction

We know that total frictional torque,

$$T = \frac{2}{3}\,\mu.W\left(\frac{R^3 - r^3}{R^2 - r^2}\right)$$

$$= \frac{2}{3} \times 0.015 \times 12568 \left[\frac{(75)^3 - (25)^3}{(75)^2 - (25)^2}\right] \text{ N-mm}$$

$$= 125.68 \times 81.25 = 10\,212 \text{ N-mm} = 10.212 \text{ N-m}$$

∴ Power lost in friction,

$$P = \frac{2\ \pi\ N\ T}{60} = \frac{2\ \pi \times 100 \times 10.212}{60} = 107 \text{ W} = 0.107 \text{ kW} \quad \textbf{Ans.}$$

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3. *Heat generated at the bearing*

We know that heat generated at the bearing

= Power lost in friction = 0.107 kW or kJ / s

= 0.107 × 60 = 6.42 kJ/min **Ans.**

Example 26.9. *The thrust of propeller shaft is absorbed by 6 collars. The rubbing surfaces of these collars have outer diameter 300 mm and inner diameter 200 mm. If the shaft runs at 120 r.p.m., the bearing pressure amounts to 0.4 N/mm². The coefficient of friction may be taken as 0.05. Assuming that the pressure is uniformly distributed, determine the power absorbed by the collars.*

Solution. Given : n = 6 ; D = 300 mm or R = 150 mm ; d = 200 mm or r = 100 mm ; N = 120 r.p.m. ; p = 0.4 N/mm^2 ; μ = 0.05

First of all, let us find the thrust on the shaft (W). Since the pressure is uniformly distributed over the bearing surface, therefore bearing pressure (p),

$$0.4 = \frac{W}{n\,\pi\,(R^2 - r^2)} = \frac{W}{6\pi\,[(150)^2 - (100)^2]} = \frac{W}{235\,650}$$

$$\therefore \quad W = 0.4 \times 235\,650 = 94\,260 \text{ N}$$

We know that total frictional torque,

$$T = \frac{2}{3}\,\mu.W\left(\frac{R^3 - r^3}{R^2 - r^2}\right) = \frac{2}{3} \times 0.05 \times 94260\left[\frac{(150)^3 - (100)^3}{(150)^2 - (100)^2}\right] \text{ N-mm}$$

$$= 597\,000 \text{ N-mm} = 597 \text{ N-m}$$

∴ Power absorbed by the collars,

$$P = \frac{2\pi.N.T}{60} = \frac{2\,\pi \times 120 \times 597}{60} = 7503 \text{ W} = 7.503 \text{ kW} \quad \textbf{Ans.}$$

Example 26.10. *The thrust of propeller shaft in a marine engine is taken up by a number of collars integral with the shaft which is 300 mm is diameter. The thrust on the shaft is 200 kN and the speed is 75 r.p.m. Taking μ constant and equal to 0.05 and assuming the bearing pressure as uniform and equal to 0.3 N/mm², find : 1. Number of collars required, 2. Power lost in friction, and 3. Heat generated at the bearing in kJ/min.*

Industrial bearings.

Solution. Given : d = 300 mm or r = 150 mm ; W = 200 kN = 200 × 10^3 N ; N = 75 r.p.m. ; μ = 0.05 ; p = 0.3 N/mm^2

1. *Number of collars required*

Let $\qquad n$ = Number of collars required.

Since the outer diameter of the collar (D) is taken as 1.4 to 1.8 times the diameter of shaft (d), therefore let us take

$$D = 1.4\,d = 1.4 \times 300 = 420 \text{ mm} \quad \text{or} \quad R = 210 \text{ mm}$$

We know that the bearing pressure (p),

$$0.3 = \frac{W}{n\,\pi\,(R^2 - r^2)} = \frac{200 \times 10^3}{n\,\pi\,[(210)^2 - (150)^2]} = \frac{2.947}{n}$$

$$\therefore \quad n = 2.947 / 0.3 = 9.8 \text{ say } 10 \quad \textbf{Ans.}$$

2. ***Power lost in friction***

We know that total frictional torque,

$$T = \frac{2}{3}\mu W\left(\frac{R^3 - r^3}{R^2 - r^2}\right) = \frac{2}{3}\times 0.05\times 200\times 10^3\left[\frac{(210)^3 - (150)^3}{(210)^2 - (150)^2}\right] \text{ N-mm}$$

$= 1817 \times 10^3$ N-mm = 1817 N-m

∴ Power lost in friction,

$$P = \frac{2\,\pi N.T}{60} = \frac{2\,\pi\times 75\times 1817}{60} = 14\ 270 \text{ W} = 14.27 \text{ kW} \quad \textbf{Ans.}$$

3. ***Heat generated at the bearing***

We know that heat generated at the bearing

= Power lost in friction = 14.27 kW or kJ/s

= 14.27 × 60 = 856.2 kJ/min **Ans.**

EXERCISES

1. The main bearing of a steam engine is 100 mm in diameter and 175 mm long. The bearing supports a load of 28 kN at 250 r.p.m. If the ratio of the diametral clearance to the diameter is 0.001 and the absolute viscosity of the lubricating oil is 0.015 kg/m-s, find : 1. The coefficient of friction ; and 2. The heat generated at the bearing due to friction.

   [Ans. 0.002 77 ; 101.5 J/s]

2. A journal bearing is proposed for a steam engine. The load on the journal is 3 kN, diameter 50 mm, length 75 mm, speed 1600 r.p.m., diametral clearance 0.001 mm, ambient temperature 15.5°C. Oil SAE 10 is used and the film temperature is 60°C. Determine the heat generated and heat dissipated.

   Take absolute viscosity of SAE10 at 60°C = 0.014 kg/m-s. **[Ans. 141.3 J/s ; 25 J/s]**

3. A 100 mm long and 60 mm diameter journal bearing supports a load of 2500 N at 600 r.p.m. If the room temperature is 20°C, what should be the viscosity of oil to limit the bearing surface temperature to 60°C? The diametral clearance is 0.06 mm and the energy dissipation coefficient based on projected area of bearing is 210 W/m^2/°C. **[Ans. 0.0183 kg/m-s]**

4. A tentative design of a journal bearing results in a diameter of 75 mm and a length of 125 mm for supporting a load of 20 kN. The shaft runs at 1000 r.p.m. The bearing surface temperature is not to exceed 75°C in a room temperature of 35°C. The oil used has an absolute viscosity of 0.01 kg/m-s at the operating temperature. Determine the amount of artificial cooling required in watts. Assume d/c = 1000. **[Ans. 146 W]**

5. A journal bearing is to be designed for a centrifugal pump for the following data :

   Load on the journal = 12 kN ; Diameter of the journal = 75 mm ; Speed = 1440 r.p.m ; Atmospheric temperature of the oil = 16°C ; Operating temperature of the oil = 60°C; Absolute viscosity of oil at 60°C = 0.023 kg/m-s.

   Give a systematic design of the bearing.

6. Design a journal bearing for a centrifugal pump running at 1440 r.p.m. The diameter of the journal is 100 mm and load on each bearing is 20 kN. The factor ZN/p may be taken as 28 for centrifugal pump bearings. The bearing is running at 75°C temperature and the atmosphere temperaturic is 30°C. The energy dissipation coefficient is 875 W/m^2/°C. Take diametral clearance as 0.1 mm.

7. Design a suitable journal bearing for a centrifugal pump from the following available data :

   Load on the bearing = 13.5 kN; Diameter of the journal = 80 mm; Speed = 1440 r.p.m.; Bearing characterisitic number at the working temperature (75°C) = 30 ; Permissible bearing pressure intensity

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= 0.7 N/mm^2 to 1.4 N/mm^2; Average atmospheric temperature = 30°C.

Calculate the cooling requirements, if any.

8. A journal bearing with a diameter of 200 mm and length 150 mm carries a load of 20 kN, when the journal speed is 150 r.p.m. The diametral clearance ratio is 0.0015.

If possible, the bearing is to operate at 35°C ambient temperature without external cooling with a maximum oil temperature of 90°C. If external cooling is required, it is to be as little as possible to minimise the required oil flow rate and heat exchanger size.

1. What type of oil do you recommend ?
2. Will the bearing operate without external cooling?
3. If the bearing operates without external cooling, determine the operating oil temperature?
4. If the bearing operates with external cooling, determine the amount of oil in kg/min required to carry away the excess heat generated over heat dissipated, when the oil temperature rises from 85°C to 90°C, when passing through the bearing.

QUESTIONS

1. What are journal bearings? Give a classification of these bearings.

2. What is meant by hydrodynamic lubrication?

3. List the basic assumptions used in the theory of hydrodynamic lubrication.

4. Explain wedge film and squeeze film journal bearings.

5. Enumerate the factors that influence most the formation and maintenance of the thick oil film in hydrodynamic bearings.

6. Make sketches to show the pressure distribution in a journal bearing with thick film lubrication in axial and along the circumference.

7. List the important physical characteristics of a good bearing material.

8. What are the commonly used materials for sliding contact bearings?

9. Write short note on the lubricants used in sliding contact bearings.

10. Explain the following terms as applied to journal bearings :

(*a*) Bearing characteristic number ; and (*b*) Bearing modulus.

11. What are the various terms used in journal bearings analysis and design? Give their definitions in brief.

12. Explain with reference to a neat plot the importance of the bearing characteristic curve.

13. What is the procedure followed in designing a journal bearing?

14. Explain with sketches the working of different types of thrust bearing.

OBJECTIVE TYPE QUESTIONS

1. In a full journal bearing, the angle of contact of the bearing with the journal is

(*a*) 120° (*b*) 180°

(*c*) 270° (*d*) 360°

2. A sliding bearing which can support steady loads without any relative motion between the journal and the bearing is called

(*a*) zero film bearing (*b*) boundary lubricated bearing

(*c*) hydrodynamic lubricated bearing (*d*) hydrostatic lubricated bearing

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3. In a boundary lubricated bearing, there is a .... of lubricant between the journal and the bearing.
   (*a*) thick film (*b*) thin film
4. When a shaft rotates in anticlockwise direction at slow speed in a bearing, then it will
   (*a*) have contact at the lowest point of bearing
   (*b*) move towards right of the bearing making metal to metal contact
   (*c*) move towards left of the bearing making metal to metal contact
   (*d*) move towards right of the bearing making no metal to metal contact
5. The property of a bearing material which has the ability to accommodate small particles of dust, grit etc., without scoring the material of the journal, is called
   (*a*) bondability (*b*) embeddability
   (*c*) comformability (*d*) fatigue strength
6. Teflon is used for bearings because of
   (*a*) low coefficient of friction (*b*) better heat dissipation
   (*c*) smaller space consideration (*d*) all of these
7. When the bearing is subjected to large fluctuations of load and heavy impacts, the bearing characteristic number should be .... the bearing modulus.
   (*a*) 5 times (*b*) 10 times
   (*c*) 15 times (*d*) 20 times
8. When the length of the journal is equal to the diameter of the journal, then the bearing is said to be a
   (*a*) short bearing (*b*) long bearing
   (*c*) medium bearing (*d*) square bearing
9. If Z = Absolute viscosity of the lubricant in kg/m-s, N = Speed of the journal in r.p.m., and p = Bearing pressure in N/mm^2, then the bearing characteristic number is
   (*a*) $\frac{Z\ N}{p}$ (*b*) $\frac{Z\ p}{N}$
   (*c*) $\frac{Z}{p\ N}$ (*d*) $\frac{p\ N}{Z}$
10. In thrust bearings, the load acts
   (*a*) along the axis of rotation (*b*) parallel to the axis of rotation
   (*c*) perpendicular to the axis of rotation (*d*) in any direction

ANSWERS

1. (*d*)	**2.** (*d*)	**3.** (*b*)	**4.** (*c*)	**5.** (*b*)
6. (*a*)	**7.** (*c*)	**8.** (*d*)	**9.** (*a*)	**10.** (*a*)

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