BEARING DESIGN

Subject: Machine Component Design

Subject Code: ME 602

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INTRODUCTION :

In rolling contact bearings, the contact between the bearing surfaces is rolling instead of sliding as in sliding contact bearings. We have already discussed that the ordinary sliding bearing starts from rest with practically metal-to-metal contact and has a high coefficient of friction. It is an outstanding advantage of a rolling contact bearing over a sliding bearing that it has a low starting friction.

TYPES OF ROLLING CONTACT BEARINGS

Following are the two types of rolling contact bearings:

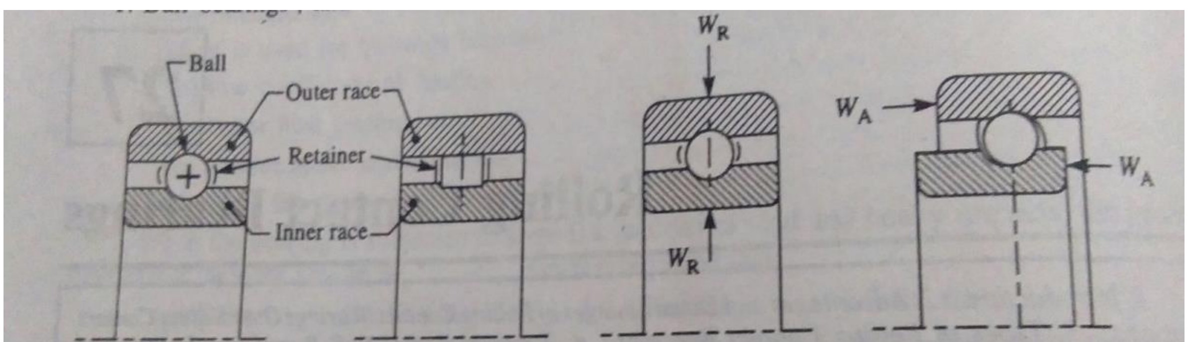
1. Ball bearings:
2. Roller bearings.

The ball and roller bearings consist of an inner race which is mounted on the shaft or journal and an outer race which is carried by the housing or casing. In between the inner and outer race, number of balls or rollers are used and these are held at proper distances by retainers so that they do not touch each other. The retainers are thin strips and used for the balls have been properly spaced.

The ball bearings are used for light loads and the roller bearings are used for heavier loads.

The rolling contact bearings, depending upon the load to be carried, are classified as :

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



(a) Ball bearing. (b) Roller bearing. (a) Radial ball bearing. (b) Thrust ball bearing.

ADVANTAGES AND DISADVANTAGES OF ROLLING CONTACT BEARINGS OVER

SLIDING CONTACT BEARINGS:

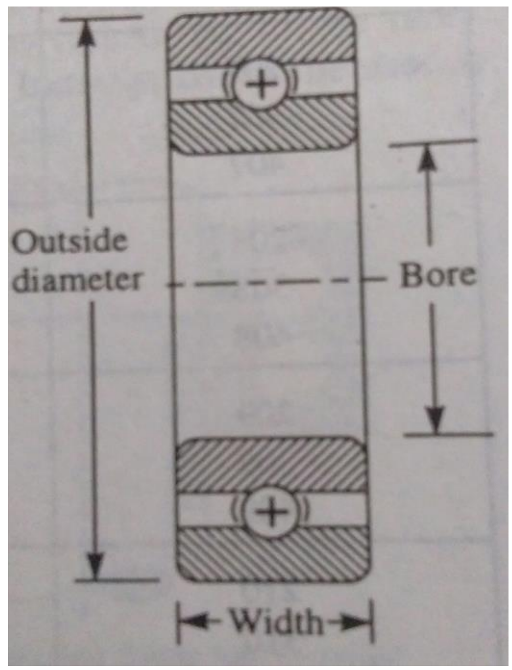
ADVANTAGES

1. Low starting and running friction except at very high speeds.
2. Ability to withstand momentary shock loads.
3. Accuracy of shaft alignment.
4. Low cost of maintenance, as no lubrication is required while in service.
5. Small overall dimensions.
6. Reliability of service.
7. Easy to mount and erect.
8. Cleanliness.

DISADVANTAGES

1. More noisy at very high speeds.
2. Low resistance to shock loading.
3. More initial cost.
4. Design of bearing housing complicated.

STANDARD DIMENSIONS AND DESIGNATIONS OF BALL BEARINGS:



The dimensions that have been standardised on an international basis are shown in Fig. These dimensions are a function of the bearing bore and the series of bearing. The standard dimensions are given in millimetres. There is no standard for the size and number of steel balls. The bearings are designated by a number. In general, the number consists of atleast three digits. Additional digits or letters are used to indicate special features e.g. deep groove, filling notch etc.

The last three digits give the series and the bore of the bearing. The last two digits from 04 onwards, when multiplied by 5, give the bore diameter in millimetres. The third from the last digit designates the series of the bearing.

The most common ball bearings are available in four series as follows :

1. Extra light (100), 2. Light (200), 3. Medium (300), 4. Heavy (400)

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BASIC STATIC LOAD RATING OF ROLLING CONTACT BEARINGS

The load carried by a non-rotating bearing is called a static load. The basic static load rating is defined as the static radial load (in case of radial ball or roller bearings) or axial load (in case of thrust ball or roller bearings) which corresponds to a total permanent deformation of the ball (or roller) and race, at the most heavily stressed contact, equal to 0.0001 times the ball (or roller) diameter. In single row angular contact ball bearings, the basic static load relates to the radial component of the load, which causes a purely radial displacement of the bearing rings in relation to each other.According to IS : 3823–1984, the basic static load rating (C0) in newtons for ball and roller bearings may be obtained as discussed below :

1. **For radial ball bearings, the basic static radial load rating (C0) is given by**

$$C_0 = f_0 . i . Z . D^2 \cos \alpha$$

Where,

i = Number of rows of balls in any one bearing,

Z = Number of ball per row,

D = Diameter of balls, in mm,

α = Nominal angle of contact i.e. the nominal angle between the line of action of the ball load and a plane perpendicular to the axis of bearing, and f_0 = A factor depending upon the type of bearing. The value of factor (f_0) for bearings made of hardened steel are taken as follows :

f_0 = 3.33, for self-aligning ball bearings

= 12.3, for radial contact and angular contact groove ball bearings.

2. For radial roller bearings, the basic static radial load rating is given by

$$C0 = f_0 . i . Z . le . D \cos \alpha$$

Where,

i = Number of rows of rollers in the bearing,

Z = Number of rollers per row,

le = Effective length of contact between one roller and that ring

D = Diameter of roller in mm. It is the mean diameter in case of tapered rollers,

α = Nominal angle of contact. It is the angle between the line of action of the roller resultant load and a plane perpendicular to the axis of the bearing, and

f_0 = 21.6, for bearings made of hardened steel.

3 For thrust ball bearings,

The basic static axial load rating is given by

$$\mathbf{C0 = f_0.Z.D^2} \sin \alpha$$

where

Z = Number of balls carrying thrust in one direction, and

f_0 = 49, for bearings made of hardened steel.

4 For thrust roller bearings,

The basic static axial load rating is given by

$$\mathbf{C0 = f_0.Z.le .D.\sin \alpha}$$

where Z = Number of rollers carrying thrust in one direction, and

f_0 = 98.1, for bearings made of hardened steel.

STATIC EQUIVALENT LOAD FOR ROLLING CONTACT BEARINGS

The static equivalent load may be defined as the static radial load (in case of radial ball or roller bearings) or axial load (in case of thrust ball or roller bearings) which, if applied, would cause the same total permanent deformation at the most heavily stressed ball (or roller) and race contact as that which occurs under the actual conditions of loading. The static equivalent radial load (W_{0R}) for radial or roller bearings under combined radial and axial or thrust loads is given by the greater magnitude of those obtained by the following two equations, i.e.

$$\mathbf{W_{0R} = X_0.W_R + Y_0.W_A ;}$$

X_0 = Radial load factor, and

Y_0 = Axial or thrust load factor.

According to IS : 3824 – 1984, the values of X0 and Y0 for different bearings are given in the **databook table**

S. No.	Type of bearing	Single row bearing		Double row bearing	
		X_0	Y_0	X_0	Y_0
1.	Radial contact groove ball bearings.	0.60	0.50	0.60	0.50
2.	Self aligning ball or roller bearings and tapered roller bearing.	0.50	0.22 cot θ	1	0.44 cot θ
3.	Angular contact groove bearings :				
	α = 15°	0.50	0.46	1	0.92
	α = 20°	0.50	0.42	1	0.84
	α = 25°	0.50	0.38	1	0.76
	α = 30°	0.50	0.33	1	0.66
	α = 35°	0.50	0.29	1	0.58
	α = 40°	0.50	0.26	1	0.52
	α = 45°	0.50	0.22	1	0.44

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LIFE OF A BEARING

The life of an individual ball (or roller) bearing may be defined as the number of revolutions (or hours at some given constant speed) which the bearing runs before the first evidence of fatigue develops in the material of one of the rings or any of the rolling elements. The rating life of a group of apparently identical ball or roller bearings is defined as the number of revolutions (or hours at some given constant speed) that 90 per cent of a group of bearings will complete or exceed before the first evidence of fatigue develops.

The term minimum life is also used to denote the rating life. It has been found that the life which 50 per cent of a group of bearings will complete or exceed is approximately 5 times the life which 90 per cent of the bearings will complete or exceed. In other words, we may say that the average life of a bearing is 5 times the rating life (or minimum life).

It may be noted that the longest life of a single bearing is seldom longer than the 4 times the average life and the maximum life of a single bearing is about 30 to 50 times the minimum life.

DYNAMIC LOAD RATING OF ROLLING CONTACT BEARINGS

The basic dynamic load rating is defined as the constant stationary radial load (in case of radial ball or roller bearings) or constant axial load (in case of thrust ball or roller bearings) which a group of apparently identical bearings with stationary outer ring can endure for a rating life of one million revolutions with only 10 per cent failure. The basic dynamic load rating (C) in newtons for ball and roller bearings may be obtained as discussed below :

1. According to IS: 3824 (Part 1)– 1983, the basic dynamic radial load rating for radial and angular contact ball bearings, except the filling slot type, with balls not larger than 25.4 mm in diameter, is given by

$$\mathbf{C = f_c (i \cos \alpha)^{0.7} Z^{2/3} . ^{D1.8}}$$

and for balls larger than 25.4 mm in diameter,

$$\mathbf{C = 3.647\ f_c (i \cos \alpha)^{0.7} Z^{2/3} . D^{1.4}}$$

DYNAMIC EQUIVALENT LOAD FOR ROLLING CONTACT BEARINGS

The dynamic equivalent load may be defined as the constant stationary radial load (in case of radial ball or roller bearings) or axial load (in case of thrust ball or roller bearings) which, if applied to a bearing with rotating inner ring and stationary outer ring, would give the same life as that which the bearing will attain under the actual conditions of load and rotation.

The dynamic equivalent radial load (W) for radial and angular contact bearings, except the filling slot types, under combined constant radial load (W_R) and constant axial or thrust load (W_A) is given by

$$\mathbf{W = X \cdot V \cdot W_R + Y \cdot W_A}$$

where V = A rotation factor, = 1, for all types of bearings when the inner race is rotating,
= 1, for self-aligning bearings when inner race is stationary,
= 1.2, for all types of bearings except self-aligning, when inner race is stationary.

The values of radial load factor (X) and axial or thrust load factor (Y) for the dynamically loaded bearings may be taken from **Data Table:**

DYNAMIC LOAD RATING FOR ROLLING CONTACT BEARINGS UNDER VARIABLE LOADS

The approximate rating (or service) life of ball or roller bearings is based on the fundamental equation,

$$\mathbf{L = (C / W)^k \times 10^6 \quad revolutions}$$

Or $$\mathbf{C = W (L/10^6)^{1/k}}$$

Where

L = Rating life,

C = Basic dynamic load rating,

W = Equivalent dynamic load, and

k = 3, for ball bearings, = 10/3, for roller bearings.

The relationship between the life in revolutions (L) and the life in working hours (LH) is given by

$$\mathbf{L = 60\ N \cdot L_H} \qquad \text{revolutions}$$

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MATERIALS AND MANUFACTURE OF BALL AND ROLLER BEARINGS

Since the rolling elements and the races are subjected to high local stresses of varying magnitude with each revolution of the bearing, therefore the material of the rolling element (i.e. steel) should be of high quality.

The balls are generally made of high carbon chromium steel. The material of both the balls and races are heat treated to give extra hardness and toughness. The balls are manufactured by hot forging on hammers from steel rods. They are then heattreated, ground and polished. The races are also formed by forging and then heat-treated, ground and polished.

LUBRICATION OF BALL AND ROLLER BEARINGS

The ball and roller bearings are lubricated for the following purposes :

1. To reduce friction and wear between the sliding parts of the bearing,
2. To prevent rusting or corrosion of the bearing surfaces,
3. To protect the bearing surfaces from water, dirt etc., and
4. To dissipate the heat. In general, oil or light grease is used for lubricating ball and roller bearings. Only pure mineral oil or a calcium-base grease should be used. If there is a possibility of moisture contact, then potassium or sodium-base greases may be used.

SOLVE QUESTIONS

1. A single row angular contact ball bearing number 310 is used for an axial flow compressor. The bearing is to carry a radial load of 2500 N and an axial or thrust load of 1500 N. Assuming light shock load, determine the rating life of the bearing. Solution.

Given : $W_R = 2500$ N ; $W_A = 1500$ N

From data Table we find that for single row angular contact ball bearing, the values of radial factor (X) and thrust factor (Y)

For WA / WR = 1500 / 2500 = 0.6 are X = 1 and Y = 0

Since the rotational factor (V) for most of the bearings is 1,

therefore dynamic equivalent load,

$$W = X.V.W_R + Y.W_A$$

$$= 1 \times 1 \times 2500 + 0 \times 1500$$

= 2500 N 1016

we find that for light shock load, the service factor (KS) is 1.5.

Therefore the design dynamic equivalent load should be taken as

W = 2500 × 1.5 = 3750 N From Table

we find that for a single row angular contact ball bearing number 310,

the basic dynamic capacity, C = 53 kN = 53 000 N

We know that rating life of the bearing in revolutions,

$$L = (C / W)^k \times 10^6 \text{ revolutions}$$

$$= 2823 \times 10^6 \text{ rev Ans. ...}$$

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(PART B)

SLIDING CONTACT BEARING

If the rubbing surfaces of a bearing are in direct contact without any rolling elements, 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.

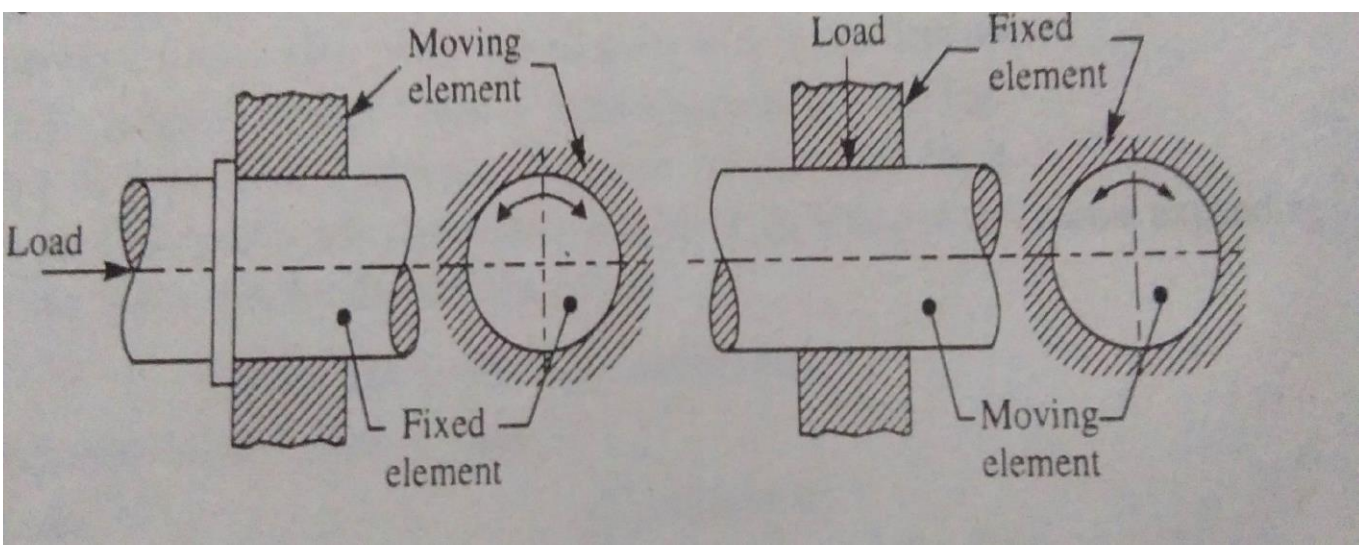
The lubricant used to separate the journal and bearing is usually a mineral oil refined from petroleum, but vegetable oils, silicon oils, greases etc., may be used.

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.) may be called slipper or guide bearings. Such type of bearings are usually found in cross-head of steam engine.

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° 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°, 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, 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.

HYDRODYNAMIC LUBRICATED BEARINGS

In hydrodynamic lubricated bearings, there is a thick film of lubricant between the journal and the bearing. 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), or

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

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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.

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. 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.

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.

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 Marine bearings, 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/mm2 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/mm2 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.

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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 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 $\tau = P/A$

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 m2/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 / m3 (the average value may be taken as 900 kg / m3). 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.

TERMS USED IN HYDRODYNAMIC JOURNAL BEARING A hydrodynamic journal bearing is shown in Fig. in which O is the centre of the journal and O′ is the centre of the bearing.

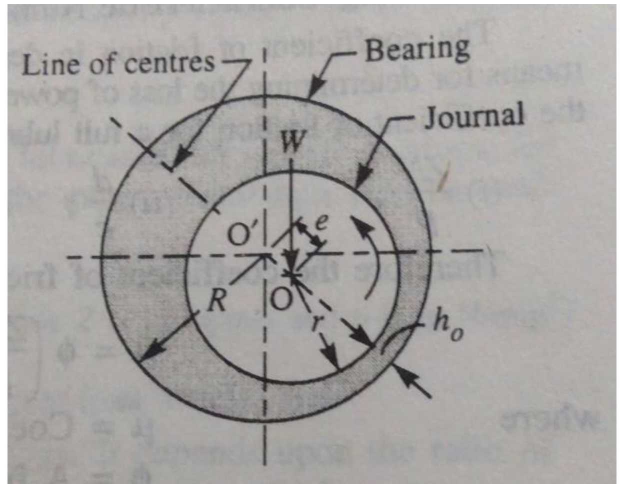

Fig. 26.7. Hydrodynamic journal bearing.

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,

$$c = D - d$$

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. 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,

$$c_1 = (D-d)/2 = R - r = c/2$$

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

$$= c/d = (D-d)/d$$

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. 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, $\varepsilon = e/c_1 = 1 - h_o/c_1$

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
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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) ; ZN /p (ii) d/ c and (iii) l /d

Therefore the coefficient of friction may be expressed as

$$\mu = \varphi \left[ZN/p,\ d/c\ ,\ l/d \right]$$

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,

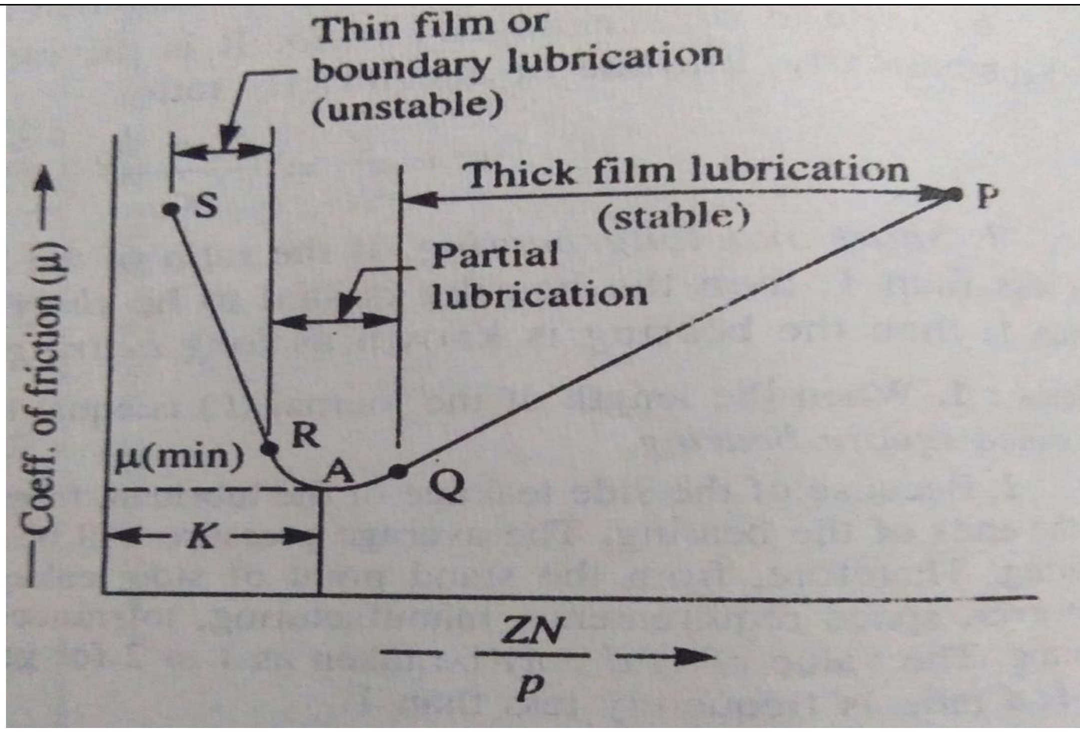
= Load on the journal $\div$ l $\times$ d

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. The factor ZN/p helps to predict the performance of a bearing.



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,

$\mu = (33/10^{8})\,(ZN/p)\,(d/c) + k$ (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 Table of design values of journal bearing to ensure safe margin between operating conditions and the point of film breakdown.

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,

$$p = (ZN/P)(d/c)^2$$

SOMMERFELD NUMBER

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

$$\text{Sommerfeld number} = (ZN/4.75 \times 10^6)(d/c)^2$$

For design purposes, its value is taken as follows : 14.3×10^6

(when Z is in kg / m-s and p is in N / mm2)

HEAT GENERATED IN A JOURNAL BEARING

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

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

where μ = Coefficient of friction,

W = Load on the bearing in N,

DESIGN PROCEDURE FOR JOURNAL BEARING

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

1. Determine the bearing length by choosing a ratio of l/d from Table
2. Check the bearing pressure, p = W / l.d from Table 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, to determine the possibility of maintaining fluid film operation.
5. Assume a clearance ratio c/d from Table
6. Determine the coefficient of friction (μ) by using the relation
7. Determine the heat generated by using the relation as discussed
8. Determine the heat dissipated by using the relation as discussed
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

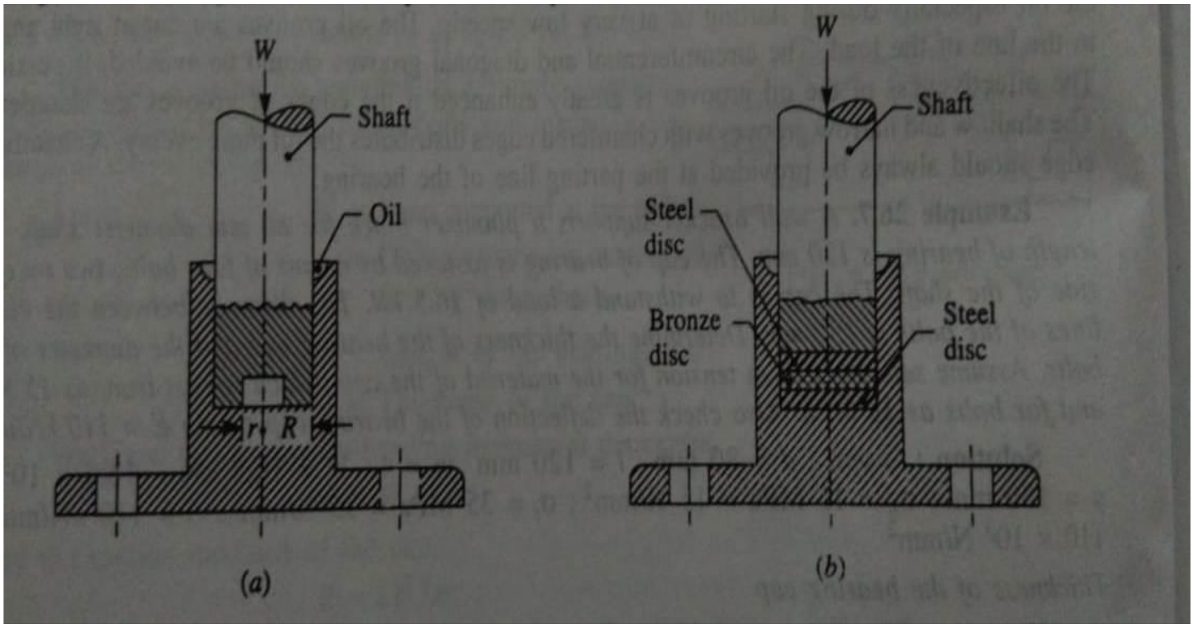
UNIT – 5
BEARING BESIGN
Unit-05 B/Lecture-09

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.



Footstep or Pivot Bearings

A simple type of footstep bearing, suitable for a slow running and lightly loaded shaft, is shown in Fig. 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 increases with the distance from the axis (i.e. radius) of the bearing, therefore the wear will be different at different radii. 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.
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. (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

$$p = W/A = W/\pi R^2$$

and the total frictional torque,

$$T = (2/3)\mu W R$$

∴ Power lost in friction, $P = (2 \pi N T)/60$ ... (T being in N-m)