JOURNAL BEARINGS

Course Contents

3.1 Introduction

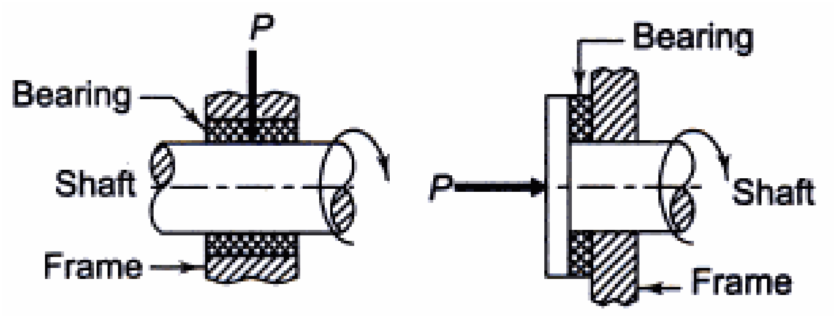
Bearing is a mechanical element that permits relative motion between two parts, such as the shaft and the housing, with minimum friction. The functions of the bearing are as follows:

(i) The bearing ensures free rotation of the shaft or the axle with minimum friction.
(ii) The bearing supports the shaft or the axle and holds it in the correct position.
(iii) The bearing takes up the forces that act on the shaft or the axle and transmits them to the frame or the foundation.

3.2 Classification of Bearings

Bearings are classified in different ways.

1. Depending upon the direction of force that act on them, bearings are classified into two categories – radial and thrust bearings, as shown in Fig. 3.1.

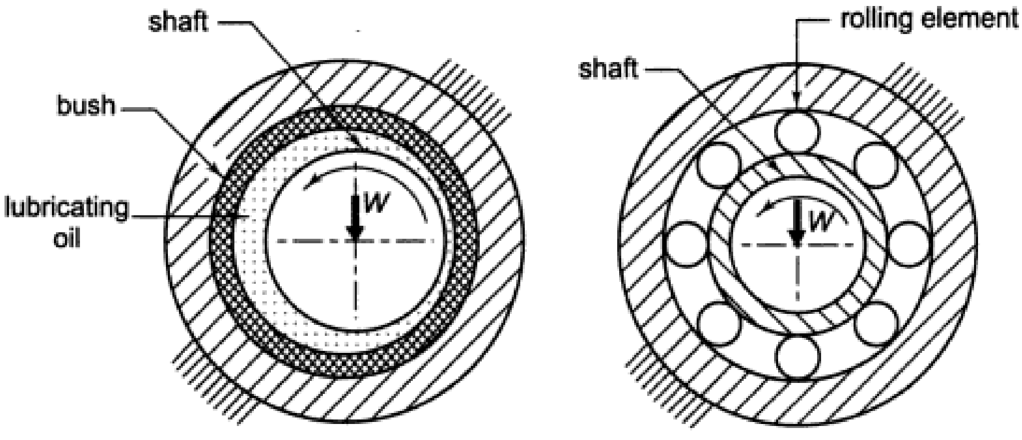


(a) Radial Bearing *(b) Thrust Bearing*

Fig. 3.1

– A radial bearing supports the load, which is perpendicular to the axis of the shaft.
– A thrust bearing supports the load, which acts along the axis of the shaft.

2. Depending upon the type of friction, bearings are classified into two main groups – sliding contact bearings and rolling contact bearings, as shown in Fig. 3.2.



(a) Sliding Contact Bearing *(b) Rolling Contact Bearing*

Fig. 3.2

– Sliding contact bearings are also called plain bearings, journal bearings or sleeve bearings. In this case, the surface of the shaft slides over the surface of the bush resulting in friction and wear. In order to reduce the friction, these two surfaces are separated by a film of lubricating oil. The bush is made of special bearing material like white metal or bronze.
– Rolling contact bearings are also called antifriction bearings. Rolling elements, such as balls or rollers, are introduced between the surfaces that are in relative motion. In this type of bearing, sliding friction is replaced by rolling friction.

Sliding contact bearings are used in the following applications:

(i) Crankshaft bearings in petrol and diesel engines
(ii) Centrifugal pumps
(iii) Large size electric motors
(iv) Steam and gas turbines and
(v) Concrete mixers, rope conveyors and marine installations.

Rolling contact bearings are used in the following applications:

(i) Machine tool spindles
(ii) Automobile front and rear axles
(iii) Gear boxes
(iv) Small size electric motors and
(v) Rope sheaves, crane hooks and hoisting drums.

3.3 Sliding Contact Bearing (Journal Bearing)

Lubrication is the science of reducing friction by application of a suitable substance called lubricant, between the rubbing surfaces of bodies having relative motion. The lubricants are classified into following three groups:

(a) Liquid lubricants like mineral or vegetable oils
(b) Semi – solid lubricants like grease
(c) Solid lubricants like graphite or molybdenum disulphite.

The objectives of lubrication are as follows:

(a) To reduce friction
(b) To reduce or prevent wear
(c) To carry away heat generated due to friction
(d) To protect the journal and the bearing from the corrosion.

3.4 The basic modes of lubrication

(i) Thick film lubrication
(ii) Thin film lubrication
(iii) Zero film bearing – is a bearing which operates without any lubricants, i.e., without any film of lubricating oil.

- **Thick film lubrication** describes a condition of lubrication, where two surfaces of the bearing in relative motion are completely separated by a film of fluid.
- Since there is no contact between the surfaces, the properties of surface, like surface finish, have little or no influence on the performance of the bearing. The resistance to relative motion arises from the viscous resistance of the fluid. Therefore, the viscosity of the lubricant affects the performance of the bearing.
- Thick film lubrication is further divided into two groups: hydrodynamic and hydrostatic lubrication.

- **Hydrodynamic lubrication** is defined as a system of lubrication in which the load supporting fluid film is created by the shape and relative motion of the sliding surfaces.

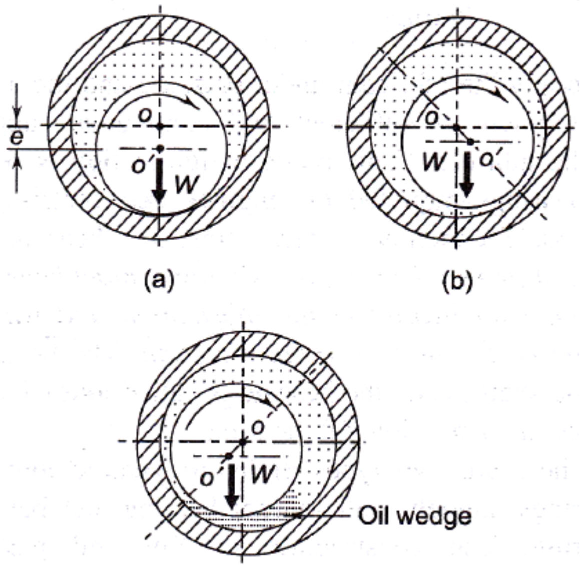


Fig. 3.3 Hydrodynamic Lubrication (a) Journal at Rest (b) Journal Starts to Rotate (c) Journal at Full Speed

- The principle of hydrodynamic lubrication in journal bearings is shown in Fig. 3.3.Initially, the shaft is at rest (a) and it sinks to the bottom of the clearance space under the action of load W.
- The surfaces of the journal and bearing touch during 'rest'. As the journal starts to rotate, it climbs the bearing surface (b) and as the speed is further increased it forces the fluid into the wedge – shaped region (c).
- Since more and more fluid is forced into the wedge-shaped clearance space, pressure is generated with in the system.

- **Pressure distribution in hydrodynamic bearing**

- Since the pressure is created within the system due to rotation of the shaft, this type of bearing is known as self-acting bearing. The pressure generated in the clearance space supports the external load (W).

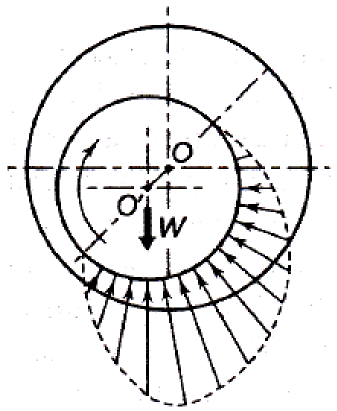


Fig. 3.4 Pressure Distribution in Hydrodynamic Bearing

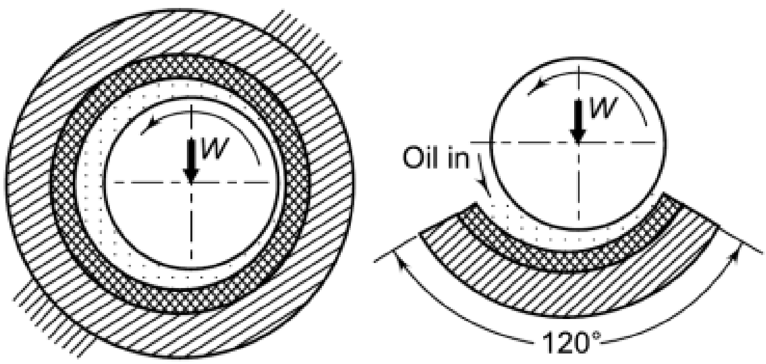
– In this case, it is not necessary to supply the lubricant under pressure and the only requirement is sufficient and continuous supply of the lubricant. This mode of lubrication is seen in bearings mounted on engines and centrifugal pumps. Frequently, a term 'journal' bearing issued.

– A journal bearing is a sliding contact bearing working on hydrodynamic lubrication and which supports the load in radial direction. The portion of the shaft inside the bearing is called journal and hence the name 'journal' bearing.

- **Types of hydrodynamic journal bearings**

– There are two types of hydrodynamic journal bearings, namely, full journal bearing and partial bearing. The construction of full and partial bearings is illustrated in Fig. 3.5.

– In full journal bearing, the angle of contact of the bushing with the journal is 360°. Full journal bearing can take loads in any radial direction. Most of the bearings used in industrial applications are full journal bearings.



(a) Full Bearing *(b) Partial Bearing*

Fig. 3.5 Full and Partial Bearings

– In partial bearings, the angle of contact between the bush and the journal is always less than 180°. Most of the partial bearings in practice have 120° angle of contact. Partial bearing can take loads in only one radial direction. Partial bearings are used in railroad-cars.

– The advantages of partial bearings compared to full journal bearing are as follows:

(i) Partial bearing is simple in construction.

(ii) It is easy to supply lubricating oil to the partial bearing.

(iii) The frictional loss in partial bearing is less. Therefore, temperature rise is low.

– There are two terms with reference to full and partial bearings, namely, 'clearance' bearing and 'fitted' bearing.

– A clearance bearing is a bearing in which the radius of the journal is less than the radius of the bearing. Therefore, there is a clearance space between the journal and the bearing. Most of the journal bearings are of this type.

– A fitted bearing is a bearing in which the radius of the journal and the bearing are equal. Obviously, fitted bearing must be partial bearing and the journal must run eccentric with respect to the bearing in order to provide space for lubricating oil.

- **Hydrostatic lubrication** is defined as a system of lubrication in which the load supporting fluid film, separating the two surfaces is created by an external source, like a pump, supplying sufficient fluid under pressure.

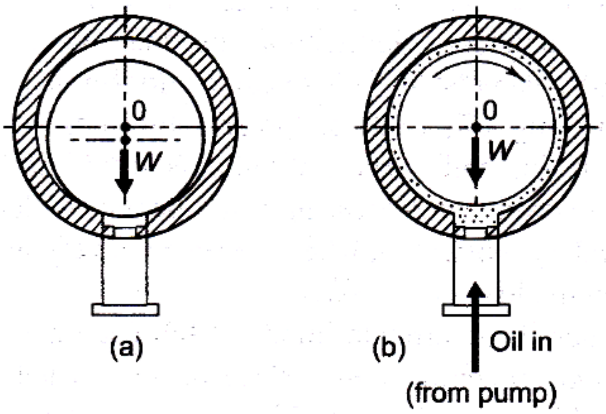


Fig. 3.6 Hydrostatic Lubrication (a) Journal at Rest
(b) Journal at Full Speed

– Since the lubricant is supplied under pressure, this type of bearing is called externally pressurized bearing. The principle of hydrostatic lubrication in journal bearing is illustrated in Fig. 3.6. Initially, the shaft rests on the bearing surface [Fig.3.6(a)].

– As the pump starts, high pressure fluid is admitted in the clearance space, forcing the surfaces of the bearing and journal to separate out [Fig. 3.6(b)]. Hydrostatic bearings are used on vertical turbo generators, centrifuges and ball mills.

– Compared with hydrostatic bearings, hydro dynamic bearings are simple in construction, easy to maintain and lower in initial as well as maintenance cost.

– Hydrostatic bearings, although costly, offer the following advantages:

(i) high load carrying capacity even at low speeds;

(ii) no starting friction; and

(iii) no rubbing action at any operating speed or load.

- **Thin film lubrication**, which is also called boundary lubrication, is defined as a condition of lubrication where the lubricant film is relatively thin and there is partial metal to metal contact. This mode of lubrication is seen in door hinges and machine tool slides. The conditions resulting in boundary lubrication are excessive load, insufficient surface area or oil supply, low speed and misalignment.

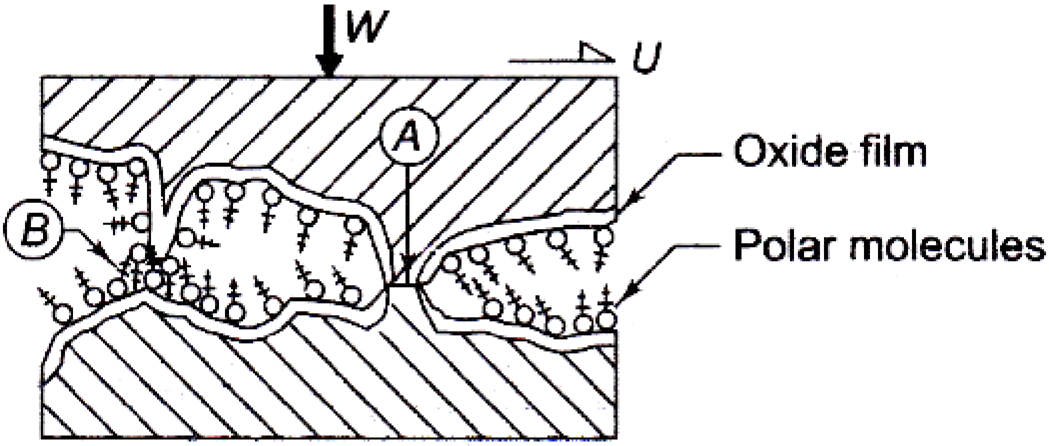


Fig. 3.7 Boundary Lubrication: (a) Metal to Metal Contact (b) Cluster of Molecules

– The mechanism of boundary lubrication is shown in Fig. 3.7. There are certain fatty acids which contain polar molecules.

– Molecules in which there is a permanent separation of positive and negative charges are called polar molecules. Their polarity has a tendency to orient and stick to the surface in a particular fashion.

– The clusters of polar molecules, cohering to one another and adhering to the surface, form a compact film which prevents metal to metal contact as is seen in the region B. This results in partial lubrication.

– There is also a zone (region A) where metal to metal contact takes place, junctions are formed at high spots and shearing takes place due to relative motion. The performance of bearing under boundary lubrication depends upon two factors, namely, the chemical composition of the lubricating oil, such as polar molecules (at the region B), and surface roughness (at region A).

– The hydrodynamic bearing also operates under the boundary lubrication when the speed is very low or when the load is excessive.

- There is a particular mode of lubrication known as elasto hydrodynamic lubrication. When the fluid film pressure is high and the surfaces to be separated are not sufficiently rigid, there is elastic deformation of the contacting surfaces. This elastic deflection is useful in the formation of the fluid film in certain cases. Since the hydrodynamic film is developed due to elastic deflection of the parts, this mode of lubrication is called elasto hydrodynamic lubrication. This type of lubrication occurs in gears, cams and rolling contact bearings.

3.5 Properties of lubricants

- **Viscosity** is defined as the internal fictional resistance offered by a fluid to change its shape or relative motion of its parts.
- **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.
- **Viscosity Index:** The rate of change of viscosity with respect to temperature is indicated by a number called Viscosity Index.
- **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.
- **Fire point:** It is the temperature at which an oil gives off sufficient vapour to burn it continuously when ignited.
- **Pour point or freezing point:** It is the lowest temperature at which the oil can flow.

3.6 Terms used in hydrodynamic journal bearing

O – centre of journal

O' – centre of bearing

D – diameter of bearing

d – diameter of journal

l – length of bearing

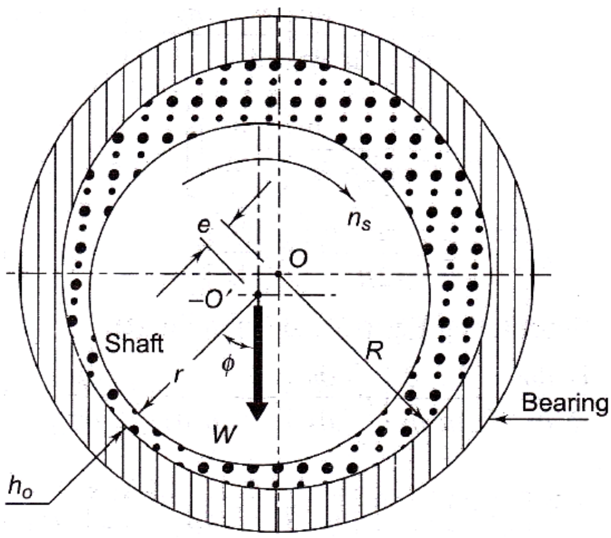


Fig. 3.8

- **Diametral clearance (c_1)** – It is difference between the diameters of the bearing and journal.

$$c_1 = D - d$$

- **Radial clearance (c)** – It is difference between radii of the bearing and journal.

$$c = R - r$$

- **Diametral clearance ratio –** It is the ratio of the diametral clearance to the diameter of the journal. Mathematically, Diametral clearance ratio = c_1 / d

- **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.
- **Minimum oil film thickness –** It is the minimum distance between the bearing and the journal, under complete lubrication condition. It is denoted by h_O and occurs at the line of centres. Its value may be assumed as c / 4.
- **Attitude or eccentricity ratio** – It is the ratio of the eccentricity to the radial clearance. Mathematically, attitude or eccentricity ratio,

$$\varepsilon = e / c$$

- **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.

3.7 Mckee's Investigation

- In hydrodynamic bearings, initially the journal is at rest. There is no relative motion and no hydrodynamic film. Therefore, there is metal to metal contact between the surfaces of the journal and the bearing.
- As the journal starts to rotate, it takes some time for the hydrodynamic film to build sufficient pressure in the, clearance space. During this period, there is partial metal to metal contact and a partial lubricant film. This is thin film lubrication.
- As the speed is increased, more and more lubricant is forced into the wedge – shaped clearance space and, sufficient pressure is built up, separating the surfaces of the journal and the bearing. This is thick film lubrication. Therefore, there is a transition from thin film lubrication to thick film lubrication as the speed increases.

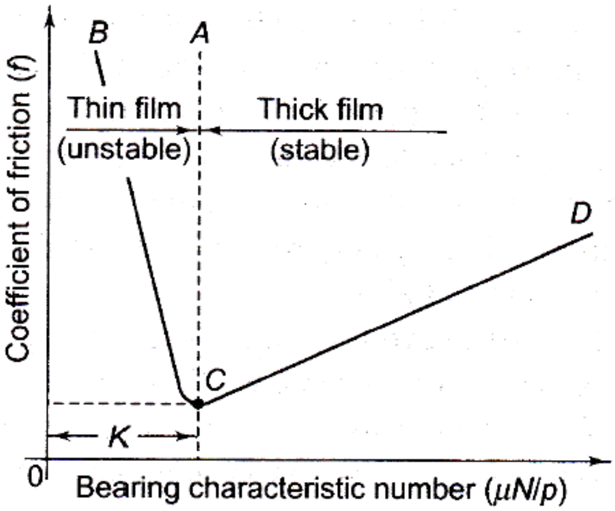


Fig. 3.9 μN/p Curve

- The transition, from thin film lubrication to thick film hydrodynamic lubrication can be better visualized by means of a curve called μN/p curve. This curve is shown in Fig. 3.9. The μN/p curve is an experimental curve developed by McKeebrothers. A bearing characteristic number is a dimensionless group of parameters given by,

– Bearing characteristic number $= \frac{\mu N}{p}$

where, μ= absolute viscosity of the lubricant
N = speed of the journal
p = unit bearing pressure (load per unit of projected area of bearing)

– The bearing characteristic number is plotted on the abscissa. The coefficient of friction f is plotted on the ordinate. The coefficient of friction f is the ratio of tangential frictional force to the radial load acting on the bearing. As seen in Fig. 3.9, there are two distinct parts of the curve – BC and CD.

(i) In the region BC, there is partial metal to metal contact and partial patches of lubricant. This is the condition of thin film or boundary lubrication.

(ii) In the region CD, there is relatively thick film of lubricant and hydrodynamic lubrication takes place.

(iii) AC is the dividing line between these two modes of lubrication. The region to the left of the line AC is the thin film zone while the region to the right of the line AC is the thick film zone.

(iv) It is observed that the coefficient of friction is minimum at C or at the transition between these two modes. The value of the bearing characteristic number corresponding to this minimum coefficient is called the bearing modulus. It is denoted by K in the figure.

– The bearing should not be operated near the critical value K at the point C. A slight drop in the speed (N) or a slight increase in the load (p) will reduce the value of μN/p resulting in boundary lubrication. The guidelines for hydrodynamic lubrication are as follows:

(i) In order to avoid seizure, the operating value of the bearing characteristic number (μN/p) should be at least 5 to 6 times that when the coefficient of friction is minimum. (5K to 6K or 5 to 6 times the bearing modulus).

(ii) If the bearing is subjected to fluctuating loads or impact conditions, the operating value of the bearing characteristic number (μN/p) should be at least 15 times that when the coefficient of friction is minimum. (15K or 15 times the bearing modulus).

– It is observed from the (μN/p) curve that when viscosity of the lubricant is very low, the value of (μN/p) parameter will be low and boundary lubrication will result. Therefore, if the viscosity of the lubricant is very low then the lubricant will not separate the surfaces of the journal and the bearing and metal to metal contact will occur resulting in excessive wear at the contacting surfaces.

– The (μN/p) curve is important because it defines the stability of hydrodynamic journal bearing and helps to visualize the transition from boundary lubrication to thick film lubrication.

3.8 Thrust bearing

– A thrust bearing is used to guide or support the shaft which is subjected to a load along the axis of the shaft. Such types 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.

- **Foot step or pivot bearings**

– A simple type of footstep bearing suitable for a slow running and lightly loaded shaft as shown in Fig. 3.10. 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 uniform. It may be noted that the wear is maximum at the outer radius and zero at the centre.

– 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.

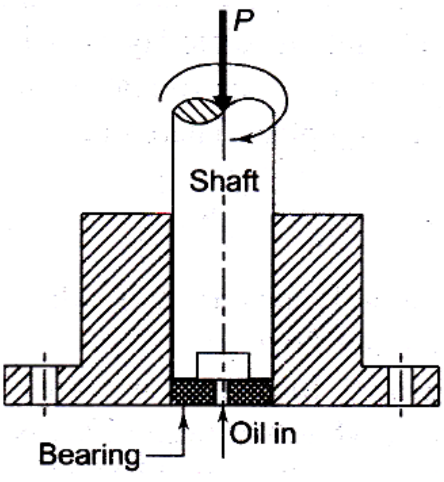


Fig. 3.10 Footstep bearing

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

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

Total frictional torque, $T = \frac{2}{3}\mu WR$

Power lost in friction, $P = \frac{2\pi NT}{60}$

For counter boring shaft,

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

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

- **Collar bearings**

– 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. 3.11. 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.

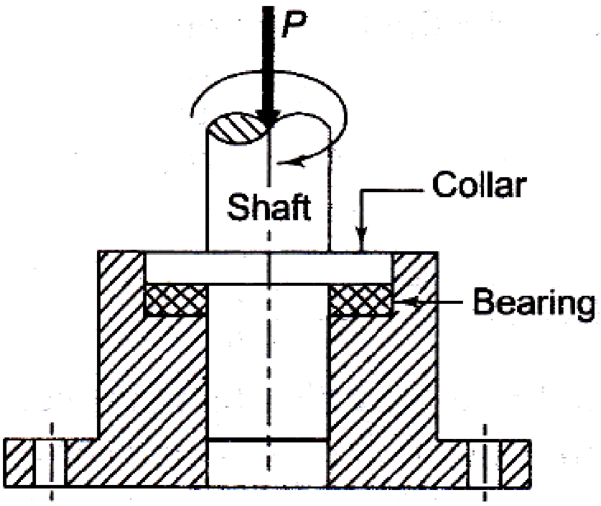


Fig. 3.11 Collar bearing

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, betweenrubbing 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,

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

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

Power lost in friction, $P = \frac{2\pi NT}{60}$

3.9 Bearing Materials

The desirable properties of a good bearing material are as follows:

(i) When metal to metal contact occurs, the bearing material should not damage the surface of the journal. It should not stick or weld to the journal surface.

(ii) It should have high compressive strength to withstand high pressures without distortion.

(iii) In certain applications like connecting rods or crankshafts, bearings are subjected to fluctuating stresses. The bearing material, in these applications, should have sufficient endurance strength to avoid failure due to pitting.

(iv) The bearing material should have the ability to yield and adopt its shape to that of the journal. This property is called conformability. When the load is applied, the journal is deflected resulting in contact at the edges. A conformable material adjusts its shape under these circumstances.

(v) The dirt particles in lubricating oil tend to jam in the clearance space and, if hard, may cut scratches on the surfaces of the journal and bearing. The bearing material should be soft to allow these particles to get embedded in the lining and avoid further trouble. This property of the bearing material is called embeddability.

(vi) In applications like engine bearings, the excessive temperature causes oxidation of lubricating oils and forms corrosive acids. The bearing material should have sufficient corrosion resistance under these conditions.

(vii) The bearing material should have reasonable cost and should be easily available in the market.

- The most popular bearing material is babbitt. Due to its silvery appearance, babbitt is called 'white' metal. There are two varieties of babbitts lead-base and tin-base, depending upon the major alloying element.
- They are used in the form of a strip or thin lining-about 0.5 mm thick-bonded to steel shells. Babbitts have excellent conformability and embeddability.
- Tin-base babbitts have better corrosion resistance and can be easily bonded to steel shells. High cost and shortage of tin are their main limitations.
- Babbitts, whether lead-base or tin-base, are inherently weak and their strength decreases rapidly with increasing temperature. Their use is further restricted due to poor fatigue strength.

- The other bearing materials are bronze, copper – lead, aluminium alloys and plastics. Compared with babbitts, bronze is cheaper, stronger and can withstand high pressures. It has got excellent casting and machining characteristics.
- Bronze bearing is made as a single solid unit. The main drawback of bronze bearing is its tendency to stick to the surface of the journal at high temperatures. Copper – lead bearings (70% Cu and 30%Pb) are used in the form of a thin lining like white metal.
- They have more hardness and fatigue strength and are used in heavy duty applications at high temperatures. Tin – aluminium alloys have higher fatigue strength and they retain their strength even at high temperatures. They are used in engine bearings.

- There are certain non – metallic bearings like graphite, plastics (Teflon) and rubber. For high temperature applications, conventional bearings with lubricating oils cannot be used.
- In such cases, bearings made of pure carbon (graphite) are employed. Teflon has an extremely low coefficient of friction and requires no external lubricant like an oil.
- They are particularly useful where the bearing is located at an inaccessible position or where the lubricating oil is likely to cause contamination such as bearings for food processing machines. Rubber is used as bearing material in marine applications.

3.10 Bearing Failure

Fatigue failures are not common in journal bearings unlike ball bearings. The failures in journal bearings are mainly associated with insufficient lubricant, contamination of lubricant and faulty assembly. The principal types of bearing failure are as follows:

(a) **Abrasive Wear:** Abrasive wear on the surface of the bearing is a common type of bearing failure. It is in the form of scratches in the direction of motion often with embedded particles. Abrasive wear occurs when the lubricating oil is contaminated with dust, foreign particles, rust or spatter. Proper enclosures for the bearing and the housing, cleanliness of lubricating oil and use of high viscosity oil are some of the remedies against this type of wear.

(b) **Wiping of Bearing Surface**: When the rotating journal touches the bearing, excessive rubbing occurs resulting in melting and smearing of the surface of the bearing. This type of failure is in the form of surface melting and flow of bearing material. The main causes for this type of wear are inadequate clearance, excessive transient load and insufficient oil supply. The remedy is to keep these factors under control.

(c) **Corrosion**: The corrosion of bearing surface is caused by the chemical attack of reactive agents that are present in the lubricating oil. These oxidation products corrode materials such as lead, copper, cadmium and zinc. Lead reacts rapidly with all oxidation agents. The remedy is to use oxidation inhibitors as additive in the lubricating oil.

(d) **Distortion**: Misalignment and incorrect type of fit are the major sources of difficulties in journal bearings. When the fit is too tight, bore distortion occurs. When foreign particles are trapped between the bearing and the housing during the assembly, local bore distortion occurs. Correct selections of the fit and proper assembly procedure are the remedies against this type of wear.

Example 3.1: The following data is given for a 360° hydrodynamic bearing:

Radial load	= 3.2 KN
Journal speed	= 1490 rpm
Journal diameter	= 50 mm
Bearings length	= 50 mm
Radial clearance	= 0.05 mm
Viscosity of lubricant	= 25 cp

Assume that total heat generated in bearing is carried by total oil flow in bearing.

Calculate (i) co-efficient of friction

(ii) power lost in friction

(iii) minimum oil film thickness

(iv) flow requirement in liters/min

$$\left(\frac{r}{c}\right)f = 3.22 \qquad \frac{h_o}{c} = 0.4 \qquad \frac{Q}{rcnl} = 4.33$$

Solution: W = 3.2 KN

n = 1490 rpm

d = 50 mm

l = 50 mm

c = 0.05 mm

μ = 25 cp

Co-efficient of friction (*f*)

$$\left(\frac{r}{c}\right)f = 3.22$$

$$\left(\frac{25}{0.05}\right)f = 3.22$$

$$f = 0.00644$$

Power lost in friction (P)

P = *fVW*

$$P = 0.00644 \times \frac{\pi dn}{60} \times W$$

$$P = 0.00644 \times \frac{\pi \times 50 \times 10^{-3} \times 1490}{60} \times 3.2 \times 10^{3}$$

P = 80.388 W

Minimum oil film thickness

ho = 0.4 c

= 0.4 (0.05)

= 0.02 mm

Flow requirement

Q = 4.33 r c n l

= 4.33 x 25 x 0.05 x 1490 x 50

= 403231.25 mm^3/min

= 403231.25 x 10^{-3} cm^3/min

= 403231.25 x 10^{-3} x 10^{-3} lit/min (1000cc = 1 liter)

= 0.403 lit/min

Example 3.2: The following data is given for a 360° hydrodynamic bearing:

Radial load	= 10 KN
Journal speed	= 1440 rpm
Unit bearing pressure	= 1000 KPa
Clearance ratio (r/c)	= 800
Viscosity of lubricant	= 30 mPa.s

Assume that total heat generated in bearing is carried by total oil flow in bearing.

Calculate (i) dimensions of bearing

(ii) co-efficient of friction

(iii) power lost in friction

(iv) total flow of oil

(iv) side leakage

$\left(\frac{r}{c}\right)f = 9.55$ $\frac{Q}{rcnl} = 3.78$ $\frac{Q_s}{Q} = 0.38$

Solution: W = 10 KN

n = 1440 rpm

p = 1000 KPa = 1 MPa

r/c = 800

μ = 30 mPa.s = 30 x 10^{-3}Pa.s

= 30 x 10^{-9}MPa.s

= 30 x 10^{-9} N.s/mm^2

Assume *l*/*d* = 1

$$p = \frac{W}{ld} = \frac{W}{d^2}$$

$$1 = \frac{10 \times 10^3}{d^2}$$

d = 100 mm

l = *d* = 100 mm

Co-efficient of friction (*f*)

$$\left(\frac{r}{c}\right)f = 3.22$$

$$(800)f = 3.22$$

$f = 0.0119$

Power lost in friction (P)

P = fVW

$$P = f \times \frac{\pi dn}{60} \times W$$

$$P = 0.0119 \times \frac{\pi \times 0.1 \times 1440}{60} \times 10 \times 10^3$$

P = 897.24 W

Total flow of oil

Q = 3.78 r c n l

$$= 3.78 \times 50 \times \left(\frac{50}{800}\right) \times 1440 \times 100$$

= 1701000 mm^3/min

Side leakage

$$\frac{Q_s}{Q} = 0.38$$

Q_S = 1701000 x 0.38

= 646380 mm^3/min

Example 3.3: 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 in diameter. The thrust on the shaft is 200 KN and speed is 75 rpm.

Find: (i) Numbers of collars required

(ii) Power lost in friction and

(iii) Heat generated at the bearing in KJ/min.

Take μ = 0.05 and bearing pressure = 0.3 N/mm^2.

Solution: d = 300 mm

W = 200 KN

n = 75 rpm

μ = f = 0.05

p = 0.3 N/mm^2

Assume outer diameter of the collar D is taken as 1.5d

D = 1.5 d = 1.5 x 300

= 450 mm

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

$$0.3 = \frac{200 \times 10^3}{n \times \pi (225^2 - 150^2)}$$

No. of collar, n = 7.5

≈ 8

Total frictional torque, $T = \frac{2}{3} f\, W \left(\frac{R^3 - r^3}{R^2 - r^2} \right)$

$$T = \frac{2}{3} \times 0.05 \times 200 \times 10^3 \left(\frac{225^3 - 150^3}{225^2 - 150^2} \right)$$

= 1900 x 10^3 N.mm

= 1900 N.m

Power lost in friction, $P = \frac{2\,\pi\, n\, T}{60}$

$$P = \frac{2 \times \pi \times 75 \times 1900}{60}$$

P = 14.92 kW

Heat generated at the bearing = Power lost in friction

= 14.92 kW or KJ/s

= 14.92 x 60

= 895.2 KJ/min.

Example 3.4: A foot step bearing supports a shaft of 100 mm diameter which is counter-bored at the end with a hole diameter of 50 mm. The bearing pressure is limited to 1 N/mm^2. The speed of shaft is 100 rpm. Assume μ = 0.015.

Find (i) the load to be supported

(ii) the power lost in friction

(iii) the heat generated in bearing

Solution: D = 100 mm

d = 50 mm

p = 1 N/mm^2

n = 100 rpm

μ = *f* = 0.015

The load to be supported

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

$$1 = \frac{W}{\pi (50^2 - 25^2)}$$

W = 5890.48 N

Total frictional torque, $T = \frac{2}{3} f\, W \left(\frac{R^3 - r^3}{R^2 - r^2} \right)$

$$T = \frac{2}{3} \times 0.015 \times 5890.48 \left(\frac{50^3 - 25^3}{50^2 - 25^2} \right)$$

= 3436.11 N.mm

= 3.436N.m

Power lost in friction, $P = \frac{2\, \pi\, n\, T}{60}$

$$P = \frac{2 \times \pi \times 100 \times 3.436}{60}$$

P = 35.98 W

Heat generated at the bearing = Power lost in friction

= 35.98W.

Example 3.5: Following data is given for a 360° hydrodynamic bearing:

Length to diameter ratio	= 1
Journal speed	= 1350 rpm
Journal diameter	= 100 mm
Diametral clearance	= 100 µm
External load	= 9 KN
The value of minimum film thickness variable	= 0.3
S	= 0.0828

Find viscosity of oil that need be used.

Solution: l/d = 1

n = 1350 rpm

d = 100 mm

Diametral clearance = 100 µm

W = 9 KN

$\frac{h_o}{c} = 0.3$

Sommerfeld number = 0.0828

$$p = \frac{W}{ld} = \frac{9000}{100 \times 100}$$

p = 0.9 N/mm^2

Radial clearance, $c = \frac{diametral\ clearance}{2} = \frac{100}{2}$

c = 50 µm

= 50 x 10^{-3} mm

Sommerfeld number, $S = \left(\frac{r}{c}\right)^2 \frac{\mu\, n_s}{p}$

$$0.0828 = \left(\frac{50}{50 \times 10^{-3}}\right)^2 \frac{\mu \times 1350}{0.9 \times 60}$$

$\mu = 3.31 \times 10^{-9}$ N.s/mm^2

= 3.31 cP (Centi Poise)

Example 3.6: The following data refer to 360^0 hydrodynamic journal bearing:

Journal speed = 900 rpm
End leakage factor = 0.002
Journal diameter = 50 mm
Bearing length = 100 mm
Clearance ratio (c/d) = 0.001
Bearing pressure = 1.4 N/mm^2
Absolute Viscosity of lubricant = 0.011 kg/m-sec at 75^0C operating temperature
Room temperature = 35^0 C
Inlet temperature of the oil = 10^0C
Specific heat of the oil = 1850 J/kg/^{0}C
Heat dissipation Coefficient = 280W/ m^2/^{0}C

Calculate: (i) the amount of artificial cooling required
(ii) the mass of the lubricating oil required.

Solution:

n = 900 rpm — k = 0.002
d = 50 mm — l = 100 mm
c/d = 0.001 — p = 1.4 N/mm^2
μ = 0.011Kg/m.s — $T_O = 75°$ C
C_P = S =1850 J/kg/^{0}C — $T_a = 35°$ C
C = 280W/ m^2/^{0}C — T = 10° C

Coefficient of friction, $f = \frac{33}{10^8}\left(\frac{\mu n}{p}\right)\left(\frac{d}{c}\right) + k$

$$f = \frac{33}{10^8}\left(\frac{0.011 \times 900}{1.4}\right)(1000) + 0.002$$

$f = 0.0043$

$$p = \frac{W}{ld}$$

Load on bearing, W = 1.4 x 100 x 50

= 7000 N

Velocity, $v = \frac{\pi\, d\, n}{60} = \frac{\pi \times 50 \times 900 \times 10^{-3}}{60}$

= 2.356 m/s

Heat generated, H_g = f v W

= 0.0043 x 2.356 x 7000

= 70.91 W

$$(T_b - T_a) = \frac{1}{2}(T_O - T_a) = \frac{1}{2}(75 - 35)$$

= 20° C

Heat dissipated, H_d = C.A (T_b – T_a) = C.l.d (T_b – T_a)

= 280 x 0.1 x 0.05 x 20

= 28 W

Amount of artificial cooling required = H_g – H_d

= 70.91 – 28

= 42.91 W

Mass of the lubricating oil required

Heat taken away by the oil, H_t = m. Cp. T

H_g = H_t = m. Cp. T

42.91 = m x 1850 x 10

m = 0.0023 kg/sec

Example 3.7: Design a journal bearing from the following data:

Radial load	= 20 KN
diameter of journal	=100 mm
Speed of journal	=900 r.p.m.
oil SAE 10 with viscosity at 55^0 C	= 0.017 Kg/m-sec
ambient temperature	=15.5^0 C
maximum bearing pressure	= 1.5 MPa
permissible rise in oil temperature	= 10^0 C
heat dissipation coefficient	= 1232 $W/m^2/^0C$
L/D ratio	= 1.6
Design parameter µN/p	= 28
clearanceratio(c/d)	= 0.0013
specific heat of oil	= 1900 $J/kg/^0C$

Solution: W = 20 KN   d = 100 mm

n = 900 rpm   k = 0.002 (Assume)

l/d = 1.6   µN/p = 28 mm

c/d = 0.0013   p = 1.5 N/mm^2

μ = 0.017 Kg/m.s T_O = 55° C

C_P = S =1900J/kg/^{0}C T_a = 15.5° C

C = 1232 W/ m^2/^{0}C T = 10° C

Amount of artificial cooling required

Coefficient of friction, $f = \frac{33}{10^8}\left(\frac{\mu n}{p}\right)\left(\frac{d}{c}\right) + k$

$$f = \frac{33}{10^8}(28)\left(\frac{1}{0.0013}\right) + 0.002$$

f = 0.0091

Velocity, $v = \frac{\pi\, d\, n}{60} = \frac{\pi \times 50 \times 900 \times 10^{-3}}{60}$

= 2.356 m/s

Heat generated, H_g = f v W

$$f \times \frac{\pi\, d\, n}{60} \times W = 0.0091 \times \frac{\pi \times 0.1 \times 900}{60} \times 20 \times 10^3$$

$$= 0.0091 \times \frac{\pi \times 0.1 \times 900}{60} \times 20 \times 10^3$$

= 857.65 W

$$(T_b - T_a) = \frac{1}{2}(T_O - T_a) = \frac{1}{2}(55 - 15.5)$$

= 19.75° C

Heat dissipated, H_d = C.A (T_b – T_a) = C.l.d (T_b – T_a)

= 1232 x 0.16 x 0.1 x 19.75

= 389.3 W

Amount of artificial cooling required = H_g – H_d

= 857.65 – 389.3

= 468.35 W

Mass of the lubricating oil required

Heat taken away by the oil, H_t = m. Cp. T

468.35 = m x 1900 x 10

m = 0.024 kg/sec

Example 3.8: A 75 mm diameter full journal bearing runs at 400 r.p.m. It is 75 mm long and is subjected to a radial load of 2500 N. The bearing is lubricated with SAE 30 oil with the viscosity 16.5×10^{-3} kg/m-s flows into the bearing at a temperature of 75°C. The radial clearance is 0.03 mm.

Determine:(1) Sommerfeld number
(2) Minimum film thickness
(3) Attitude
(4) Power loss in friction
(5) Heat loss
(6) Side flow & Total flow of lubricants
(7) Temperature rise

Solution: d = 75 mm N = 400 rpm
l = 75 mm W = 2500 N
$\mu = 16.5\times10^{-3}$ kg/m-s To = 75°C
c = 0.03 mm

Bearing pressure, $p = \frac{W}{l \times d} = \frac{2500}{75\times75}$ = 0.44 N/mm^2

Sommerfeld number, $S = \left(\frac{r}{c}\right)^2\left(\frac{\mu N_s}{p}\right) = \left(\frac{37.5}{0.03}\right)^2\left(\frac{16.5\times10^{-3}\times400}{0.44\times60}\right)$

$S = 0.39 \times 10^6$

For value l/d = 1 and full bearing at S = 0.39

Attitude or eccentricity ratio, ε = 0.27

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

$$0.27 = 1 - \frac{h_o}{0.03}$$

h_O = 0.0219 mm

for *ε = 0.27,* $f\left(\frac{r}{c}\right) = 10.35$

Co-efficient of friction, $f = 10.35\left(\frac{0.03}{37.5}\right)$

= 0.00828

Power loss in friction, $P = fvW = f\left(\frac{\pi dN}{60}\right)W$

$= 0.00828\left(\frac{\pi\times0.075\times400}{60}\right)2500$

= 32.51 W

Heat loss =Power loss in friction = 32.51 W

For ε = 0.27, $\frac{4Q}{dcN_s l} = 3.73$, $\frac{Q_s}{Q} = 0.3906$

$$\frac{4Q}{dcN_s l} = 3.73$$

$$Q = \frac{3.73 \times 0.075 \times 0.03 \times 10^{-3} \times 400 \times 0.075}{4 \times 60}$$

$= 1.049 \times 10^{-6}\ m^3/s$

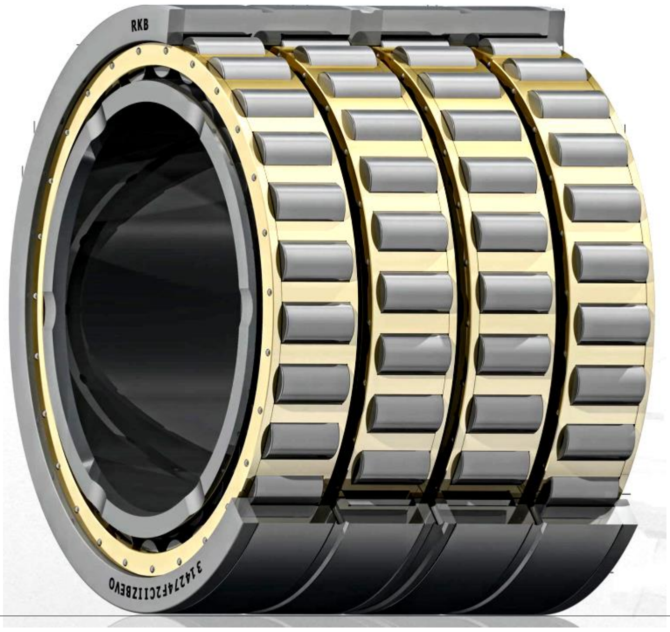
$$\frac{Q_s}{Q} = 0.3906$$

$Q_S = 4.097 \times 10^{-7}\ m^3/s$

For $\varepsilon = 0.27$, $\rho c' = 142 \times 10^4 N/m^2\,^\circ C$, $\frac{\rho c' \Delta T}{p} = 42.37$

$$\Delta T = \frac{42.37 \times 0.44 \times 10^6}{142 \times 10^4} = 13.13^\circ C$$

ROLLING CONTACT BEARINGS



4.1 Introduction

– Rolling contact bearings are also called antifriction bearings or simply ball bearings. Rolling elements, such as balls or rollers, are introduced between surfaces that are in relative motion. In this type of bearing, sliding friction is replaced by rolling friction.

4.2 Advantages and Disadvantages of Rolling Contact Bearings over Sliding Contact Bearings

The following are some 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.

4.3 Types of Rolling Contact Bearings:

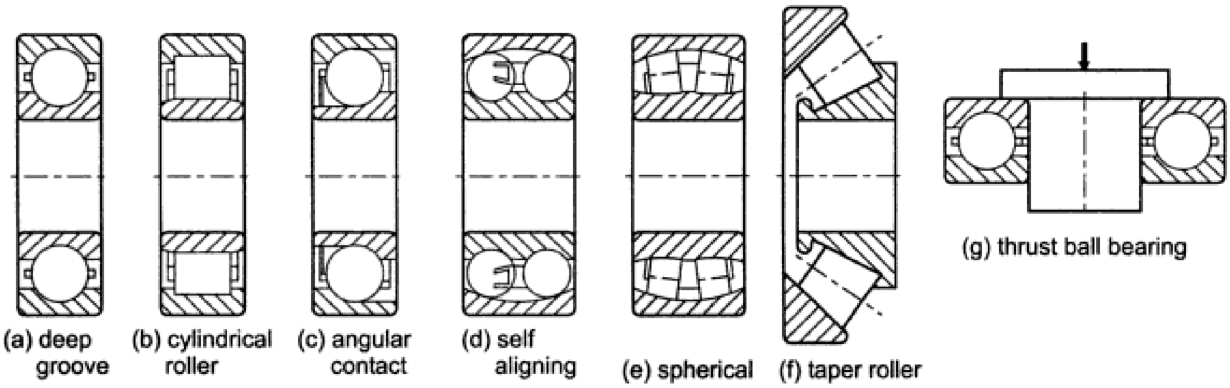


Fig. 4.1 Types of Rolling Contact Bearings

There are main two types of rolling contact bearing: Ball bearing and Roller bearing.

- **Types of Ball bearing**

– Single row deep groove ball bearing
– Filling notch bearing
– Angular contact bearing
– Double row bearing

– Self – aligning bearing
– Thrust ball bearing

- **Types of Roller bearing**

– Cylindrical roller bearing
– Spherical roller bearing
– Needle roller bearing
– Taper roller bearing

4.4 Parts of rolling contact bearing

- **Inner race** – which is pressed on to shaft.
- **Outer race** – which fits in bearing housing.
- **Rolling element** – balls, rollers which roll over the race track.
- **Cage** – for separating the balls so that they do not touch each other to reduce wear and noise.
- **Seal** is provided to avoid enter the dirt and foreign particle in bearing.

4.5 Selection of Bearing-Type

The selection of the type of bearing in a particular application depends upon the requirement of the application and the characteristics of different types of bearings. The guidelines for selecting a proper type of bearing are as follows:

(i) For low and medium radial loads, ball bearings are used, whereas for heavy loads and large shaft diameter so roller bearings are selected.

(ii) Self-aligning ball bearings and spherical roller bearings are used in applications where a misalignment between the axes of the shaft and housing is likely to exist.

(iii) Thrust ball bearings are used for medium thrust loads whereas for heavy thrust loads, cylindrical roller thrust bearings are recommended. Double acting thrust bearings can carry the thrust load in either direction.

(iv) Deep groove ball bearings, angular contact bearings and spherical roller bearings are suitable in applications where the load acting on the bearing consists of two components – radial and thrust.

(v) The maximum permissible speed of the shaft depends upon the temperature rise in the bearing. For high speed applications, deep groove ball bearings, angular contact bearings and cylindrical roller bearings are recommended.

(vi) Rigidity controls the selection of bearings in certain applications like machine tool spindles. Double row cylindrical roller bearings or taper roller bearings are used under these conditions. The line of contact in these bearings, as compared with the point of contact in ball bearings, improves the rigidity of the system.

(vii) Noise becomes the criterion of selection in applications like household appliances. For such applications, deep groove ball bearings are recommended.

4.6 Static Load Carrying Capacity

- Static load is defined as the load acting on the bearing when the shaft is stationary. It produces permanent deformation in balls and races, which increases with increasing load.
- The permissible static load, therefore, depends upon the permissible magnitude of permanent deformation.
- From past experience, it has been found that a total permanent deformation of 0.0001 of the ball or roller diameter occurring at the most heavily stressed ball and race contact, can be tolerated in practice, without any disturbance like noise or vibrations.
- The static load carrying capacity of a bearing is defined as the static load which corresponds to a total permanent deformation of balls and races, at the most heavily stressed point of contact, equal to 0.0001 of the ball diameter.
- Formulae are given in standards for calculating the static load carrying capacity of different types of bearings. However, while selecting the bearings, it is not necessary to use these formulae.
- The values of static load carrying capacities are directly given in the manufacturer's catalogues, which are based on the above formulae. Where conditions of friction, noise and smoothness are not critical, a much higher permanent deformation can be tolerated and consequently static loads up to four times the static load carrying capacity may be permissible.
- On the other hand, where extreme smoothness of operation is desired, a smaller permanent deformation is permitted.

4.7 Dynamic Load Carrying Capacity

- The life of a ball bearing is limited by the fatigue failure at the surfaces of balls and races. The dynamic load carrying capacity, of the bearing is, therefore, based on the fatigue life of the bearing.
- The life of an individual ball bearing is defined as the number of revolutions (or hours of service at some given constant speed), which the bearing runs before the first evidence of fatigue crack in balls or races.
- Since the life of a single bearing is difficult to predict, it is necessary to define the life in terms of the statistical average performance of a group of bearings.
- Bearings are rated on one of the two criteria – the average life of a group of bearings or the life, which 90% of the bearings will reach or exceed. The second criterion is widely used in bearing industry.
- The rating life of a group of apparently identical ball bearings is defined as the number of revolutions that 90% of the bearings will complete or exceed before the first evidence of fatigue crack. There are a number of terms used for this rating life.

They are minimum life, catalogue life, L_{10} or B_{10} life. These terms are synonyms fits for rating life.

- The life of an individual ball bearing may be different from rating life. Statistically, it can be proved that the life, which 50% of a group of bearings will complete or exceed, is approximately five times the rating or L_{10} life. This means that for the majority of bearings, the actual life is considerably more than the rated life.
- The dynamic load carrying capacity of a bearing is defined as the radial load in radial bearings (or thrust load in thrust bearings) that can he carried for a minimum life of one million revolutions.
- The minimum life in this definition is the L_{10} life, which90% of the bearings will reach or exceed before fatigue failure. The dynamic load carrying capacity is based on the assumption that the inner race is rotating while the outer race is stationary. The formulae for calculating the dynamic load capacity for different types of, bearings are given in standards.

4.8 Equivalent Bearing Load

- In actual applications, the force acting on the bearing has two components – radial and thrust. It is therefore necessary to convert the two components acting on the bearing into a single hypothetical load, fulfilling the conditions applied to the dynamic load carrying capacity. Then the hypothetical load can be compared with the dynamic load capacity.
- The equivalent dynamic load is defined as the constant radial load in radial bearings (or thrust load in thrust bearings), which if applied to the bearing would give same life as that which the bearing will attain under actual condition of forces. The expression for the equivalent dynamic load is written as,

$$P = X\,V\,F_r + Y\,F_a$$

where, P = equivalent dynamic load (N)

F_r = radial load (N)

F_a = axial or thrust load (N)

V = race-rotation factor

X and Y are radial and thrust factors respectively and their values are given in the manufacturer's catalogues.

- The race-rotation factor depends upon whether the inner race is rotating or the outer race. The value of V is 1when the inner race rotates while the outer race is held stationary in the housing. The value of V is 1.2 when the outer race rotates with respect to the load, while the inner race remains stationary.
- In most of the applications, the inner race rotates and the outer race is fixed in the housing. Assuming V as unity, the general equation for equivalent dynamic load is given by,

$$P = X\,F_r + Y\,F_a$$

– When the bearing is subjected to pure radial load F_r,

$$P = F_r$$

– When the bearing is subjected to pure load F_a,

$$P = F_a$$

4.9 Load-Life Relationship

The relationship between the dynamic load carrying capacity, the equivalent dynamic load, and the bearing life is given by.

$$L_{10} = \left(\frac{C}{P}\right)^p \qquad \text{..............(a)}$$

where, L_{10} = rated bearing life (in million revolutions)

C = dynamic load capacity (N), and

p = 3 (for ball bearings)

p =10/3 (for roller bearings)

Rearranging Eq. (a),

$C = P\,(L_{10})^{1/p}$

For all types of ball bearings,

$C = P\,(L_{10})^{1/3}$

For all types of roller bearings,

$C = P\,(L_{10})^{0.3}$

The relationship between life in revolutions and life in working hours is

$$L_{10} = \frac{60\ n\ L_{10h}}{10^6}$$

where, L_{10h} = rated bearing life (hours)

n = speed of rotation (rpm)

4.10 Design for Cyclic Loads and Speeds

In certain applications, ball bearings are subjected to cyclic loads and speeds. As an example, consider a ball bearing operating under the following conditions:

(a) radial load 2500 N at 700 rpm for 25% of the time,
(b) radial load 5000 N at 900 rpm for 50% of the time, and
(c) radial load 1000 N at 750 rpm for remaining 25% of the time.

Under these circumstances, it is necessary to consider the complete work cycle while finding out the dynamic load capacity of the bearing. The procedure consists of dividing the work cycle into a number of elements, during which the operating conditions of load and speed are constant.

Suppose that the work cycle is divided into x elements. Let $P_1 P_2, \ldots P_x$ be the loads and $n_1, n_2, \ldots, n_x$ be the speeds during these elements. During the first element, the life L_1 corresponding to load P_1, is given by

$$L_1 = \left(\frac{C}{P_1}\right)^3 \times 10^6 \text{ rev.}$$

In one revolution, the life consumed is $\left(\frac{1}{L_1}\right)$ or $\left(\frac{P_1^{\,3}}{C^3} \times \frac{1}{10^6}\right)$.

Let us assume that the first element consists of N_1 revolutions. Therefore, the life consumed by the first element is given by,

$$\frac{N_1 P_1^{\,3}}{10^6 C^3}$$

Similarly, the life consumed by the second element is given by

$$\frac{N_2 P_2^{\,3}}{10^6 C^3}$$

Adding these expressions, the life consumed by the complete work cycle is given by

$$\frac{N_1 P_1^{\,3}}{10^6 C^3} + \frac{N_2 P_2^{\,3}}{10^6 C^3} + \ldots.. + \frac{N_x P_x^{\,3}}{10^6 C^3} \qquad \ldots\ldots\ldots\ldots\ldots(a)$$

If $P_e$is the equivalent load for the complete work cycle, the life consumed by the work cycle is given by.

$$\frac{N P_e^{\,3}}{10^6 C^3} \qquad \ldots\ldots\ldots\ldots\ldots(b)$$

Where, $N = N_1 + N_2 + \ldots\ldots. + N_x$

Equating expressions (a) and (b),

$$N_1 P_1^{\,3} + N_2 P_2^{\,3} + \ldots. + N_x P_x^{\,3} = N P_e^{\,3}$$

$$P_e = \sqrt[3]{\frac{N_1 P_1^{\,3} + N_2 P_2^{\,3} + \ldots.}{N_1 + N_2 + \ldots.}}$$

The above equation is used for calculating the dynamic load capacity of bearing.

In case of bearings, where there is a combined radial and axial load, it should be first converted into equivalent dynamic load before the above computations are carried out.

4.11 Bearing with a probability of survival other than 90 percent

In the definition of rating life, it is mentioned that the rating life is the life that 90% of a group of identical bearings will complete or exceed before fatigue failure. The reliability R is defined as,

$$R = \frac{\text{No. of bearing which have successfully completed L million revolution}}{\text{Total number of bearings under test}}$$

Therefore, reliability of bearings selected from the manufacturer's catalogue is 0.9 or 90%.

In certain applications, where there is risk to human life, it becomes necessary to select a bearing having a reliability of more than 90%. Fig. 4.2 shows the distribution of bearing failures. The relationship between bearing life and reliability is given by a statistical curve known as Weibull distribution.

For Wiebull distribution,

$$R = e^{-(L/a)^b}$$

Where R is the reliability (in friction), L is the corresponding life and a and b are constants.

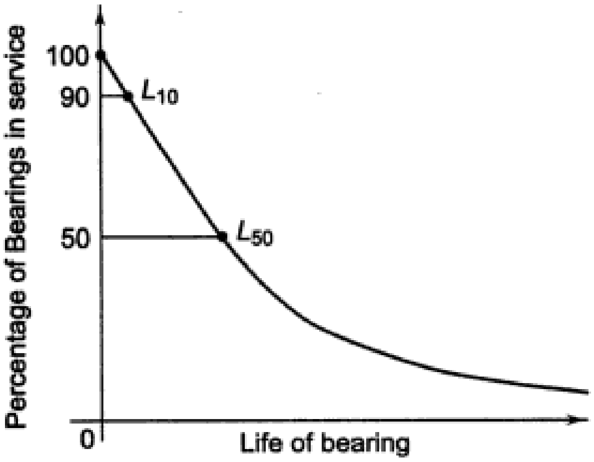


Fig. 4.2

Rearranging the above equation,

$$\left(\frac{1}{R}\right) = e^{(L/a)^b}$$

$$\log_e\left(\frac{1}{R}\right) = \left(\frac{L}{a}\right)^b \qquad \text{.............(a)}$$

If L_{10} is the life corresponding to a reliability of 90% or R_{90}, then,

$$\log_e\left(\frac{1}{R_{90}}\right) = \left(\frac{L_{10}}{a}\right)^b \qquad \text{.............(b)}$$

Dividing Eq. (a) by (b),

$$\left(\frac{L}{L_{10}}\right) = \left[\frac{\log_e\left(\frac{1}{R}\right)}{\log_e\left(\frac{1}{R_{90}}\right)}\right]^{1/b} \qquad \text{..............(c)}$$

where $R_{90} = 0.9$

The values of a and b are

a = 6.84 and b = 1.17

These values are obtained from the condition,

$L_{50} = 5L_{10}$

where L_{50} is the median life median life or life which 50% of the bearings will complete or exceed before fatigue failure. Equation (c) is used for selecting the bearing when the reliability is other than 90%.

In a system, if there are a number of bearings, the individual reliability of each bearing should be fairly high. If there are N bearings in the system, each having the same reliability R then the reliability of the complete system is given by,

$$R_S = (R)^N$$

Where, R_S indicates the probability of one out of N bearings failing during its lifetime.

4.12 Bearing Failure – Causes and Remedies

- There are two basic types of bearing failure breakage of parts like races or cage and the surface destruction. The fracture in the outer race of the ball bearing occurs due to overload.
- When the bearing is misaligned, the load acting on some balls or rollers sharply increases and may even crush them. The failure of the cage is caused due to the centrifugal force acting on the balls.
- The complete breakage of the parts of the ball bearing can be avoided by selecting the correct ball bearing, adjusting the alignment between the axes of the shaft and the housing and operating within permissible speeds.

- In general, the failure of antifriction bearing occurs not due to breakage of parts but due to damage of working surfaces of their parts. The principal types of surface wear are as follows:

(i) **Abrasive Wear:** Abrasive wear occurs when the bearing is made to operate in an environment contaminated with dust, foreign particles, rust or spatter. Remedies against this type of wear are provision of oil seals, increasing surface hardness and use of high viscosity oils. The thick lubricating film developed by these oils allows fine particles to pass without scratching.

(ii) **Corrosive Wear:** The corrosion of the surfaces of bearing parts is caused by the entry of water or moisture in the bearing. It is also caused due to corrosive elements present in the Extreme Pressure (EP) additives that are added in the lubricating oils. These elements attack the surfaces of the bearing, resulting in fine wear uniformly distributed over the entire surface. Remedies against this type of wear are, providing complete enclosure for the bearing free from external contamination, selecting proper additives and replacing the lubricating oil at regular intervals.

(iii) **Pitting:** Pitting is the main cause of the failure of antifriction bearings. Pitting is a surface fatigue failure which occurs when the load on the bearing part exceeds the surface endurance strength of the material. This type of failure is characterised by pits, which continue to grow resulting in complete destruction of the bearing surfaces. Pitting depends upon the magnitude of Hertz' contact stress and the number of stress cycles. The surface endurance strength can be improved by increasing the surface hardness.

(iv) **Scoring:** Excessive surface pressure, high surface speed and inadequate supply of lubricant result in breakdown of the lubricant film. This results in excessive frictional heat and overheating at the contacting surfaces. Scoring is a stick-slip phenomenon, in which alternate welding and shearing takes place rapidly at high spots. Here, the rate of wear is faster. Scoring can be avoided by selecting the parameters, such as surface speed, surface pressure and the flow of lubricant in such a way that the resulting temperature at the contacting surfaces is within permissible limits.

4.13 Lubrication of Rolling Contact Bearings

The purpose of lubrication in antifriction bearing is to reduce the friction between balls and races. The other objectives are dissipation of frictional heat prevention of corrosion and protection of the bearing from dirt and other foreign particles. There are two types of lubricants – oil and grease. Compared with grease, oil offers the following advantages:

(i) It is more effective in carrying frictional heat.
(ii) It feeds more easily into contact areas of the bearing under load.
(iii) It is more effective in flushing out dirt, corrosion and foreign particles from the bearing.

The advantages offered by grease lubricated bearings are simple housing design, less maintenance cost, better sealing against rust and less possibility of leakage. The guidelines for selecting the lubricant are as follows:

(i) When the temperature is less than 100°C, grease is suitable, while lubricating oils are preferred for applications where the temperature exceeds 100°C.

(ii) When the product of bore (in mm) x speed(in rpm) is below 200000, grease is suitable. For higher values, lubricating oils are recommended.

(iii) Grease is suitable for low and moderate loads, while lubricating oils are used for heavy duty applications.

(iv) If there is a central lubricating system, which is required for the lubrication of other parts, the same lubricating oil is used for bearings, e.g., gearboxes.

The choice of lubricating oil is necessary for high speed, heavy load applications, while in the remaining majority of applications; grease offers the simplest and cheapest mode of lubrication.

Example 4.1: Design a self-aligning ball bearing for a radial load of 7000 N and a thrust load of 2100 N. The desired life of the bearing is 160 mr at 300 rpm. Assume uniform and steady load. The value of X and Y factors are 0.65 and 3.5 respectively. The outer ring rotates.

Solution: F_r = 7000 N
F_a = 2100 N
L = 160 mr
N = 300 rpm
S = 1
X = 0.65
Y = 3.5
V = 1.2

Equivalent bearing load, P_e = S (X V F_r + Y F_a)
= 1 (0.65 x 1.2 x 7000 + 3.5 x 2100)
= 12810 N

Life of bearing, $L = \left(\dfrac{C}{P_e}\right)^p$

$$160 = \left(\frac{C}{12810}\right)^3$$

C = 69543.38 N

Example 4.2: It is required to select a ball bearing suitable for 50 mm diameter shaft rotating at 1500 rpm. The radial and thrust loads at the bearing are 4500 N and 1600 N respectively. The value of X and Y factors are 0.56 and 1.2 respectively. Select a proper ball bearing from following table for rotating life of 22500 hr. The inner ring rotates and service factor is 1.

Bearing No.	6010	6210	6310	6410
C (N)	21600	35100	61800	87100

Solution: d = 50 mm
N = 1500 rpm
F_r = 4500 N
F_a = 1600 N
X = 0.56
Y = 1.2
L_h = 22500 hr
V = 1

$$L = \frac{L_h \times 60 \times N}{10^6}$$

$$L = \frac{22500 \times 60 \times 1500}{10^6}$$

L = 2025 mr

Equivalent bearing load, $P_e = S\,(X\,V\,F_r + Y\,F_a)$

$= 1\,(0.56 \times 1 \times 4500 + 1.2 \times 1600)$

$= 4440$ N

$$\text{Life of bearing, } L = \left(\frac{C}{P_e}\right)^p$$

$$2025 = \left(\frac{C}{4440}\right)^3$$

C = 56172.6 N

From table, C = 61800 N

Bearing No. = 6310

Example 4.3: A single - row deep groove ball bearing No. 6002 is subjected to an axial thrust of 1000 N and a radial load of 2200 N. Find the expected life that 50% of the bearings will complete under this condition. [Static load capacity C_O: 2500 N, Dynamic Load Capacity C: 5590N]

Solution: $F_r = 2200$ N, $F_a = 1000$ N

$C_O = 2500$ N, C = 5590 N

Assume, X = 0.56

Y = 1.08

S = 1

V = 1

Equivalent bearing load, $P_e = S\,(X\,V\,F_r + Y\,F_a)$

$= 1\,(0.56 \times 1 \times 2200 + 1.08 \times 1000)$

$= 2312$ N

$$\text{Life of bearing, } L = \left(\frac{C}{P_e}\right)^p$$

$$L = \left(\frac{5590}{2312}\right)^3$$

L = 14.13 mr

$L_{50} = 5\,L_{10}$

$= 5(14.13)$

$= 70.65$ mr

Example 4.4: Single row deep groove ball bearing 6010 is subjected to an axial trust of 1200N and radial load 2400 N. Find the expected life that 50% of the bearing will complete under this condition.

C_O= 13200 N, C = 21600 N.

Fa / Co	Fa / Fr> e		e	Fa / Fr< e	
	X	Y		X	Y
0.07	0.56	1.6	0.27	1	0
0.13	0.56	1.4	0.31	1	0

Solution: $F_a = 1200$ N
$F_r = 2400$ N
$C_O = 13200$ N
C = 21600 N

$$\frac{F_a}{C_O} = \frac{1200}{13200} = 0.09$$

Related e = 0.31

$$\frac{F_a}{F_r} = \frac{1200}{2400} = 0.5$$

So, $F_a / F_r > e$

From table, X = 0.56 and Y = 1.4
Take S = 1 and V = 1

Equivalent bearing load, $P_e = S\,(X\,V\,F_r + Y\,F_a)$
= 1 (0.56 x 1 x 2400 + 1.4 x 1200)
= 3024 N

Life of bearing, $L = \left(\frac{C}{P_e}\right)^p$

$$L = \left(\frac{21600}{3024}\right)^3$$

L = 364.43mr

$L_{50} = 5\,L_{10}$
= 5(364.43)
= 1822.157mr

Example 4.5: For SKF 6207 bearing is to operate on following work cycle.

- Radial load of 6307 N at 200 rpm for 25% of time
- Radial load of 9080 N at 600 rpm for 20% of time
- Radial load of 3638 N at 400 rpm for 55% of time

The inner ring rotates. The loads are steady. Find expected average life of this bearing in hours if C = 25500 N.

Element No.	Load (N)	Element time	Speed (rpm)	Element speed (revolution)
1	6307	0.25	200	50
2	9080	0.2	600	120
3	3638	0.55	400	220

Equivalent bearing load, $P_e = \sqrt[3]{\frac{P_1^3 N_1 + P_2^3 N_2 + P_3^3 N_3}{N_1 + N_2 + N_3}}$

$$P_e = \sqrt[3]{\frac{(6307)^3(50) + (9080)^3(120) + (3638)^3(220)}{50 + 120 + 220}}$$

P_e= 6616.578 N

Life of bearing, $L = \left(\frac{C}{P_e}\right)^p$

$$L = \left(\frac{25500}{6616.578}\right)^3$$

= 57.24 mr

$$L_h = \frac{L \times 10^6}{60 \times N} = \frac{57.24 \times 10^6}{60 \times 390}$$

Rating life = 2446.27 hr

Average life = 5 x 2446.27

= 12231.35 hr

Example 4.6: A ball bearing is operating on a work cycle consisting of three parts:

- a radial load of 3000 N at 1440 rpm for one quarter cycle,
- a radial load of 5000 N at 720 rpm for one half cycle and
- a radial load of 2500 N at 1440 rpm for remaining cycle.

The expected life of the bearing is 10000 hours. Calculate the dynamic load carrying capacity of the bearing.

Element No.	Load (N)	Element time	Speed (rpm)	Element speed (revolution)
1	3000	0.25	1440	360
2	5000	0.5	720	360
3	2500	0.25	1440	360

Equivalent bearing load, $P_e = \sqrt[3]{\frac{P_1^3 N_1 + P_2^3 N_2 + P_3^3 N_3}{N_1 + N_2 + N_3}}$

$$P_e = \sqrt[3]{\frac{(3000)^3(360) + (5000)^3(360) + (2500)^3(360)}{360 + 360 + 360}}$$

= 3823 N

$$L = \frac{L_h \times 60 \times N}{10^6}$$

$$L = \frac{10000 \times 60 \times 1080}{10^6}$$

L = 648mr

Life of bearing, $L = \left(\dfrac{C}{P_e}\right)^p$

$$648 = \left(\frac{C}{3823}\right)^3$$

C = 33082.3 N

Example 4.7: The following data refers to ball bearing work cycle:

Sr. no	Radial load (N)	Axial load (N)	Radial factor	Thrust factor	% time	Service factor	Speed (r.p.m.)
1	4000	800	1	0	30 %	1.25	900
2	8000	3000	0.56	2	40 %	1	600
3	-	-	-	-	30 %	-	600

Calculate the dynamic load rating of the bearing, if the expected bearing life is 10000 hrs with reliability of 95 %.

$P_{e1} = S\,(X\,V\,F_r + Y\,F_a)$
$= 1.25\,(1 \times 1 \times 4000 + 0 \times 800)$
$= 5000$ N

$P_{e2} = S\,(X\,V\,F_r + Y\,F_a)$
$= 1\,(0.56 \times 1 \times 8000 + 2 \times 3000)$
$= 10480$ N

$P_{e3} = 0$

Element No.	Load (N)	Element time	Speed (rpm)	Element speed (revolution)
1	5000	0.3	900	270
2	10480	0.4	600	240
3	0	0.3	600	180

Equivalent bearing load, $P_e = \sqrt[3]{\dfrac{P_1^3 N_1 + P_2^3 N_2 + P_3^3 N_3}{N_1 + N_2 + N_3}}$

$$P_e = \sqrt[3]{\frac{(5000)^3(270) + (10480)^3(240) + 0}{270 + 240 + 180}}$$

= 7658.9 N

$$L_{95} = \frac{L_{95h} \times 60 \times N}{10^6}$$

$$L_{95} = \frac{10000 \times 60 \times 690}{10^6}$$

L_{95} = 414mr

$$\frac{L_{95}}{L_{10}} = \left[\frac{\ln(1/R_{95})}{\ln(1/R_{90})}\right]^{1/b}$$

$$\frac{414}{L_{10}} = \left[\frac{\ln(1/0.95)}{\ln(1/0.9)}\right]^{1/1.17}$$

$$\frac{414}{L_{10}} = \left[\frac{0.0513}{0.105}\right]^{0.8547}$$

L_{10} = 763.6 mr

$$\text{Life of bearing, } L = \left(\frac{C}{P_e}\right)^p$$

$$763.6 = \left(\frac{C}{7658.9}\right)^3$$

C = 69849 N

Example 4.7: A ball bearing, subjected to a radial load of 5 KN, is expected to have a life of 8000 hrs at 1450 rpm with a reliability of 99%. Calculate the dynamic load capacity of bearing, so that it can be selected from manufacturer's catalogue based on a reliability of 90%.

Solution: F_r = 5 KN

L_{99h} = 8000 hr

n = 1450 rpm

$$L_{99} = \frac{L_{99h} \times 60 \times N}{10^6}$$

$$L_{95} = \frac{8000 \times 60 \times 1450}{10^6}$$

L_{95} = 696 mr

$$\frac{L_{99}}{L_{10}} = \left[\frac{\ln(1/R_{99})}{\ln(1/R_{90})}\right]^{1/b}$$

$$\frac{696}{L_{10}} = \left[\frac{\ln(1/0.99)}{\ln(1/0.9)}\right]^{1/1.17}$$

$$\frac{696}{L_{10}} = \left[\frac{0.01005}{0.10536}\right]^{0.8547}$$

L_{10} = 5186mr

$$\text{Life of bearing, } L = \left(\frac{C}{P_e}\right)^p$$

$$5186 = \left(\frac{C}{5000}\right)^3$$

C = 86546 N

Example 4.8: A single row deep groove ball bearing is subjected to a radial load of 8000 N and a thrust load of 3000 N. The values of X and Y factors are 0.56 and 1.5 respectively. The shaft speed is 1200 r.p.m. and diameter of shaft is 75 mm. The bearing selected for this application is No.6315 (C = 112000 N).Find the life of the bearing with 90 % reliability and estimate the reliability for 20000 hr life.

Solution: F_r = 8000 N, X = 0.56

F_a = 3000 N, Y = 1.5

N = 1200 rpm

d = 75 mm

C = 112000 N

P_{e1} = S (X V F_r + Y F_a)

= 1 (0.56 x 1 x 8000 + 1.5 x 3000)

= 8980 N

$$\text{Life of bearing, } L_{10} = \left(\frac{C}{P_e}\right)^p$$

$$L_{10} = \left(\frac{112000}{8980}\right)^3$$

= 1940.10 mr

$$L_{10} = \frac{L_{10h} \times 60 \times N}{10^6}$$

$$1940.10 = \frac{L_{10h} \times 60 \times 1200}{10^6}$$

L_{10h} = 26945.83 hr

$$L = \frac{L_h \times 60 \times N}{10^6}$$

$$L = \frac{20000 \times 60 \times 1200}{10^6}$$

L = 1440 mr

$$\frac{L}{L_{10}} = \left[\frac{\ln(1/R)}{\ln(1/R_{90})}\right]^{1/b}$$

$$\frac{1440}{1940.1} = \left[\frac{\ln(1/R)}{\ln(1/0.9)}\right]^{1/1.17}$$

$$0.742 = \left[\frac{\ln(1/R)}{\ln(1.11)}\right]^{0.8547}$$

$$0.7053 = \left[\frac{\ln(1/R)}{\ln(1.11)}\right]$$

$$\ln(1/R) = 0.7053 \times \ln(1.11)$$

$$\ln(1) - \ln(R) = 0.0736$$

$$0 - \ln(R) = 0.0736$$

$$\ln(R) = -0.0736$$

$$R = e^{-0.0736}$$

$$R = 0.929$$

Reliability is 92.9 %