UNIT 1

BEARINGS

Course objectives:

- To apply principles of design to mechanical power transmission elements like bearings and to design appropriate bearing

Course Out comes:

Student will be able to:

- To gain the knowledge on bearings and Select suitable bearings and its constituents from manufacturers catalogues under given loading conditions

BEARINGS

A bearing is machine part, which support a moving element and confines its motion. The supporting member is usually designated as bearing and the supporting member may be journal. Since there is a relative motion between the bearing and the moving element, a certain amount of power must be absorbed in overcoming friction, and if the surface actually touches, there will be a rapid wear.

1.2.1 Classification: Bearings are classified as follows:

1. **Depending upon the nature of contact between the working surfaces:-**

   a) Sliding contact bearings
   b) Rolling contact bearings.

a) SLIDING BEARINGS:

- Hydrodynamically lubricated bearings
- Bearings with boundary lubrication
- Bearings with Extreme boundary lubrication.
- Bearings with Hydrostatic lubrication.

b) ROLLING ELEMENT BEARINGS:

- Ball bearings
- Roller bearings
- Needle roller bearings

1. **Based on the nature of the load supported:**

- Radial bearings - Journal bearings
- Thrust bearings
  - Plane thrust bearings
  - Thrust bearings with fixed shoes
  - Thrust bearings with Pivoted shoes
- Bearings for combined Axial and Radial loads.

JOURNAL BEARING:

It is one, which forms the sleeve around the shaft and supports a bearing at right angles to the axis of the bearing. The portion of the shaft resting on the sleeve is called the journal. **Example of journal bearings are**- Solid bearing, Bushed bearing and Pedestal bearing.

Solid bearing:

A cylindrical hole formed in a cast iron machine member to receive the shaft which makes a running fit is the simplest type of solid journal bearing. Its rectangular base plate has two holes drilled in it for bolting down the bearing in its position as shown in the figure1.1. An oil hole is provided at the top to lubricate the bearing. There is no means of adjustment for wear and the shaft must be introduced into the bearing endwise. It is therefore used for shafts, which carry light loads and rotate at moderate speeds.

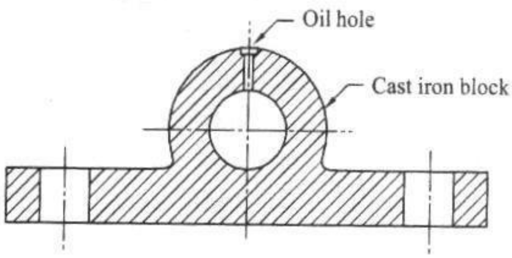


Fig. 7.1 Solid bearing

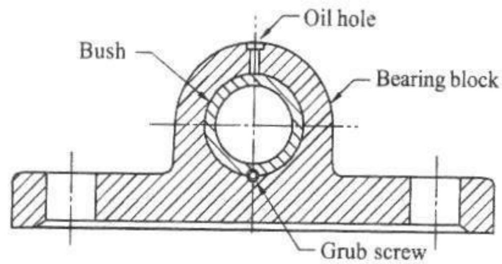


Fig. 7.2 Bushed bearing

Bushed bearing:

It consists of mainly two parts, the cast iron block and bush; the bush is made of soft material such as brass, bronze or gunmetal. The bush is pressed inside the bore in the cast iron block and is prevented from rotating or sliding by means of grub- screw as shown if the figure 1.2. When the bush gets worn out it can be easily replaced. Elongated holes in the base are provided for lateral adjustment.

Pedestal bearing:

It is also called Plummer block. Figure 1.3 shows half sectional front view of the Plummer block. It consists of cast iron pedestal, phosphor bronze bushes or steps made in two halves and cast iron cap. A cap by means of two square headed bolts holds the halves of the steps together. The steps are provided with collars on either side in order to prevent its axial movement. The snug in the bottom step, which fits into the corresponding hole in the body, prevents the rotation of the steps along with the shaft. This type of bearing can be placed anywhere along the shaft length.

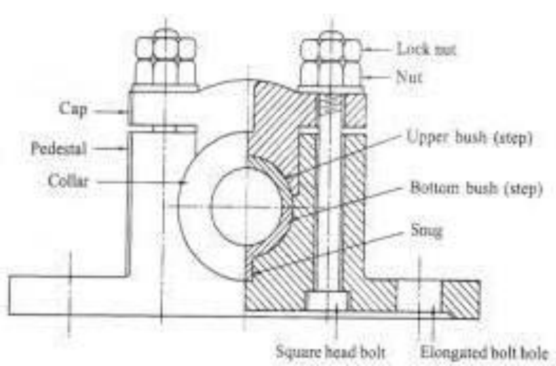


Fig 1.3: Pedestal Bearing

Thrust bearing:

It is used to guide or support the shaft, which is subjected to a load along the axis of the shaft. Since a thrust bearing operates without a clearance between the conjugate parts, an adequate supply of oil to the rubbing surfaces is extremely important. Bearings designed to carry heavy thrust loads may be broadly classified in to two groups-

FOOT STEP BEARING, AND COLLAR BEARING

Footstep bearing: Footstep bearings are used to support the lower end of the vertical shafts. A simple form of such bearing is shown in fig 1.4. It consists of cast iron block into which a gunmetal bush is fitted. The bush is prevented from rotating by the snug provided at its neck. The shaft rests on a concave hardened steel disc. This disc is prevented from rotating along with the shaft by means of pin provided at the bottom.

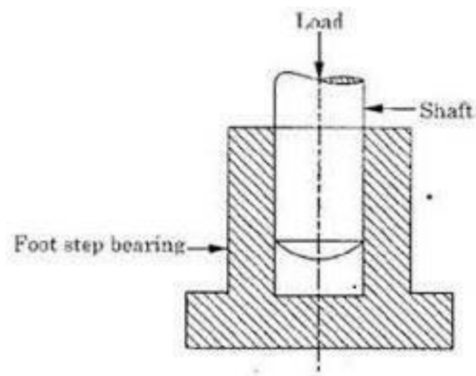


Fig 1.4: Foot step Bearing

Collar bearing:

The simple type of thrust bearing for horizontal shafts consists of one or more collars cut integral with the shaft as shown in fig.1.5. These collars engage with corresponding bearing surfaces in the thrust block. This type of bearing is used if the load would be too great for a step bearing, or if a thrust must be taken at some distance from the end of the shaft. Such bearings may be oiled by reservoirs at the top of the bearings.

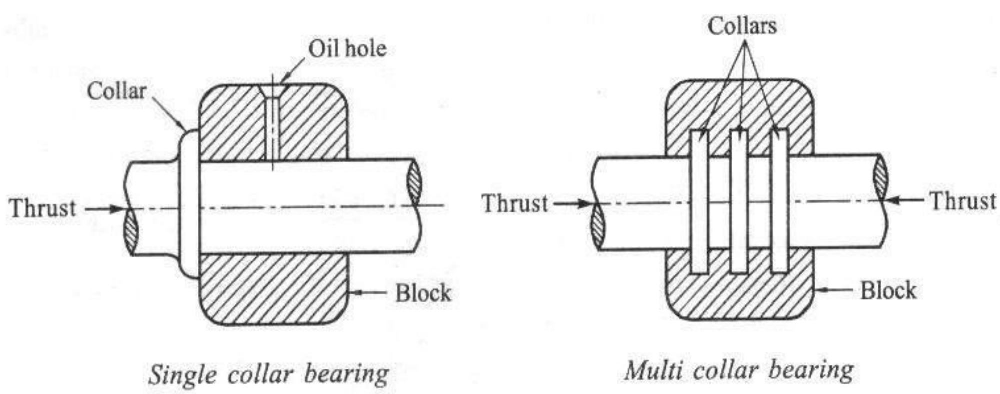
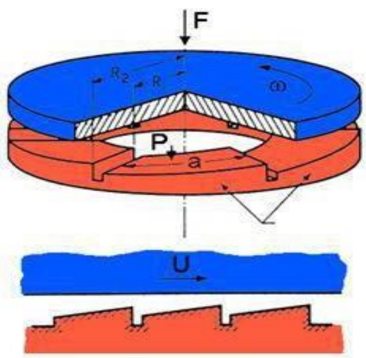


Fig 1.5: Collar bearings

Thrust bearings of fixed inclination pad and pivoted pad variety are shown in figure 1.6 a & b. These are used for carrying axial loads as shown in the diagram. These bearings operate on hydrodynamic principle.



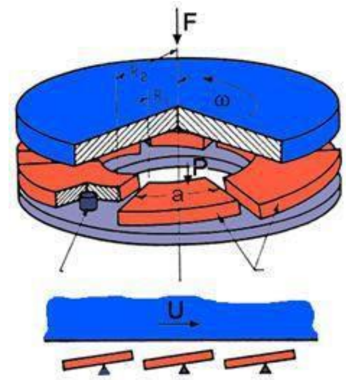


Fig 1.6(a): Fixed-incline-pads thrust bearing   Fig 1.6(b): Pivoted-pads thrust bearing

Rolling contact bearings:

The bearings in which the rolling elements are included are referred to as rolling contact bearings. Since the rolling friction is very less compared to the sliding friction, such bearings are known as anti friction bearings.

Ball bearings:

It consists of an inner ring which is mounted on the shaft and an outer ring which is carried by the housing. The inner ring is grooved on the outer surface called inner race and the outer ring is grooved on its inner surface called outer race. In between the inner and outer race there are number of steel balls. A cage pressed steel completes the assembly and provides the means of equally spacing and holding the balls in place as shown in the figure 1.7. Radial ball bearings are used to carry mainly radial loads, but they can also carry axial loads.

Cylindrical roller bearings

The simplest form of a cylindrical roller bearing is shown in fig 1.8. It consists of an inner race, an outer race, and set of roller with a retainer. Due to the line contact between the roller and the raceways, the roller bearing can carry heavy radial loads.

Tapered roller bearings:

In tapered roller bearings shown in the fig. 1.9, the rollers and the races are all truncated cones having a common apex on the shaft centre to assure true rolling contact. The tapered roller bearing can carry heavy radial and axial loads. Such bearings are mounted in pairs so that the two bearings are opposing each other’s thrust.

ADVANTAGES OF SLIDING CONTACT BEARINGS:

- They can be operated at high speeds.
- They can carry heavy radial loads.
- They have the ability to withstand shock and vibration loads.
- Noiseless operation.

Disadvantages:

- High friction losses during staring.
- More length of the bearing.
- Excessive consumption of the lubricant and high maintenance.

ADVANTAGES ROLLING CONTACT BEARINGS:

- Low starting and less running friction.
- It can carry both radial as well as thrust loads.
- Momentary over loads can be carried without failure.
- Shaft alignment is more accurate than in the sliding bearings.

Disadvantages:

- More noisy at high speeds.
- Low resistance to shock loads.
- High initial cost.
- Finite life due to eventual failure by fatigue

SOLID FRICTION

1. Resistance force for sliding
   - Static coefficient of friction
   - Kinetic coefficient of friction
2. Causes
   - Surface roughness (asperities)
   - Adhesion (bonding between dissimilar materials)
3. Factors influencing friction
   - Sliding friction depends on the normal force and frictional coefficient, independent of the sliding speed and contact area
4. Effect of Friction
   - Frictional heat (burns out the bearings)
   - Wear (loss of material due to cutting action of opposing motion)
5. Engineers control friction
   - Increase friction when needed (using rougher surfaces)
   - Reduce friction when not needed (lubrication)

The coefficients of friction for different material combinations under different conditions are given in table 1.1.

TABLE 1.1

COEFFICIENTS OF FRICTION

Material	μ
Perfectly clean metals in vacuum	Seizure μ > 5
Clean metals in air	0.8-2
Clean metals in wet air	0.5-1.5
Steel on dry bearing metals (e.g. lead, bronze)	0.1-0.5
Steel on ceramics	0.1-0.5
Ceramics on ceramics (e.g. carbides on carbides)	0.05-0.5
Polymers on polymers	0.05-1.0
Metals and ceramics on polymers (PE, PTFE, PVC)	0.04-0.5
Boundary lubrication of metals	0.05-0.2
High-temperature lubricants (MoS_2, graphite)	0.05-0.2
Hydrodynamic lubrication	0.001-0.005

LUBRICATION:

Prevention of metal to metal contact by means of an intervening layer of fluid or fluid like material.

Types of sliding lubrication:

- Sliding with Fluid film lubrication.
- Sliding with Boundary lubrication.
- Sliding with Extreme boundary lubrication.
- Sliding with clean surfaces.

HYDRODYNAMIC / THICK FILM LUBRICATION / FLUID FILM LUBRICATION

Metal to Metal contact is prevented. This is shown in figure 1.10. Friction in the bearing is due to oil film friction only.Viscosity of the lubricant plays a vital role in the power loss, temperature rise & flow through of the lubricant through the bearing.The principle operation is the Hydrodynamic theory.This lubrication can exist under moderately loaded bearings running at sufficiently high speeds.

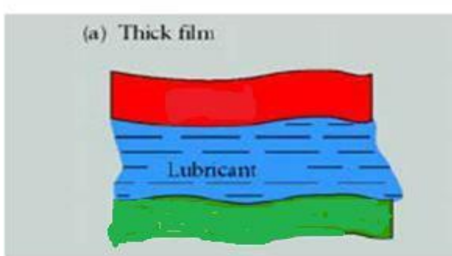


Fig 1.10: Thick Film Lubrication

BOUDARY LUBRICATION (THIN FILM LUBRICATION)

During starting and stopping, when the velocity is too low, the oil film is not capable of supporting the load. There will be metal to metal contact at some spots as shown in figure 1.11. Boundary lubrication exists also in a bearing if the load becomes too high or if the viscosity of the lubricant is too low. Mechanical and chemical properties of the bearing surfaces and the lubricants play a vital role.

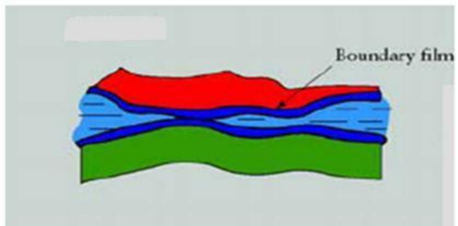


Fig 1.11: Boundary Lubrication

Oiliness of lubricant becomes an important property in boundary lubrication. Anti oxidants and Anti-corrosives are added to lubricants to improve their performance. Additives are added to improve the viscosity index of the lubricants.

Oiliness Agents

- Increase the oil film's resistance to rupture, usually made from oils of animals or vegetables.
- The molecules of these oiliness agents have strong affinity for petroleum oil and for metal surfaces that are not easily dislodged.
- Oiliness and lubricity (another term for oiliness), not related to viscosity, manifest itself under boundary lubrication; reduce friction by preventing the oil film breakdown.

Anti-Wear Agents

Mild EP additives protect against wear under moderate loads for boundary lubrications Anti-wear agents react chemically with the metal to form a protective coating that reduces friction, also called as anti-scuff additives.

Extreme boundary lubrication

Under certain conditions of temperature and load, the boundary film breaks leading to direct metal to metal contact as shown in figure 1.12. Seizure of the metallic surfaces and destruction of one or both surfaces begins. Strong intermolecular forces at the point of contact results in tearing of metallic particles. "Plowing" of softer surfaces by surface irregularities of the harder surfaces. Bearing material properties become significant. Proper bearing materials should be selected.

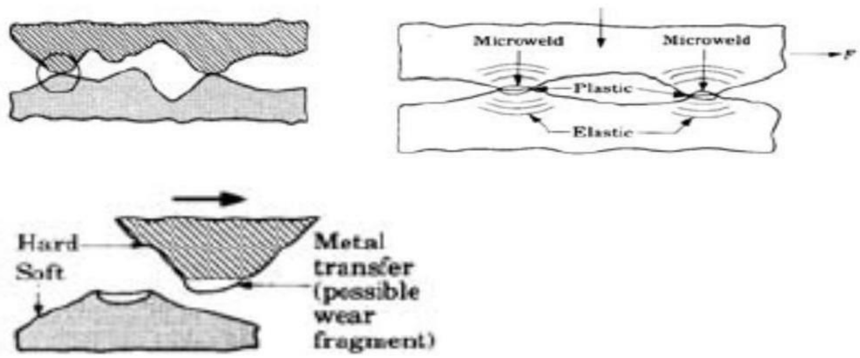


Fig 1.12: Extreme Boundary Lubrication

Extreme-Pressure Agents

Scoring and pitting of metal surfaces might occur as a result of this case, seizure is the primarily concern. Additives are derivatives of sulphur, phosphorous, or chlorine. These additives prevent the welding of mating surfaces under extreme loads and temperatures.

Stick-Slip Lubrication

A special case of boundary lubrication when a slow or reciprocating action exists. This action is destructive to the full fluid film. Additives are added to prevent this phenomenon causing more drag force when the part is in motion relative to static friction. This prevents jumping ahead phenomenon.

Solid film lubrication

When bearings must be operated at extreme temperatures, a solid film lubricant such as graphite or molybdenum di-sulphide must be used because the ordinary mineral oils are not satisfactory at elevated temperatures. Much research is currently being carried out in an effort to find composite bearing materials with low wear rates as well as small frictional coefficients.

1.4.5. Hydrostatic lubrication

Hydrostatic lubrication is obtained by introducing the lubricant, which is sometimes air or water, into the load-bearing area at a pressure high enough to separate the surfaces with a relatively thick film of lubricant. So, unlike hydrodynamic lubrication, this kind of lubrication does not require motion of one surface relative to another. Useful in designing bearings where the velocities are small or zero and where the frictional resistance is to be an absolute minimum.

1.4.6 Elasto Hydrodynamic lubrication

Elasto-hydrodynamic *lubrication* is the phenomenon that occurs when a lubricant is introduced between surfaces that are in rolling contact, such as mating gears or rolling bearings. The mathematical explanation requires the Hertzian theory of contact stress and fluid mechanics.

Newton's Law of Viscous Flow

In Fig. 1.13 let a plate *A* be moving with a velocity *U* on a film of lubricant of thickness *h*. Imagine the film to be composed of a series of horizontal layers and the force *F* causing these layers to deform or slide on one another just like a deck of cards. The layers in contact with the moving plate are assumed to have a velocity *U*; those in contact with the stationary surface are assumed to have a zero velocity. Intermediate layers have velocities that depend upon their distances *y* from the stationary surface.

Newton's viscous effect states that the shear stress in the fluid is proportional to the rate of change of velocity with respect to *y*.

$$\text{Thus } T = F/A = Z\,(du/dy).$$

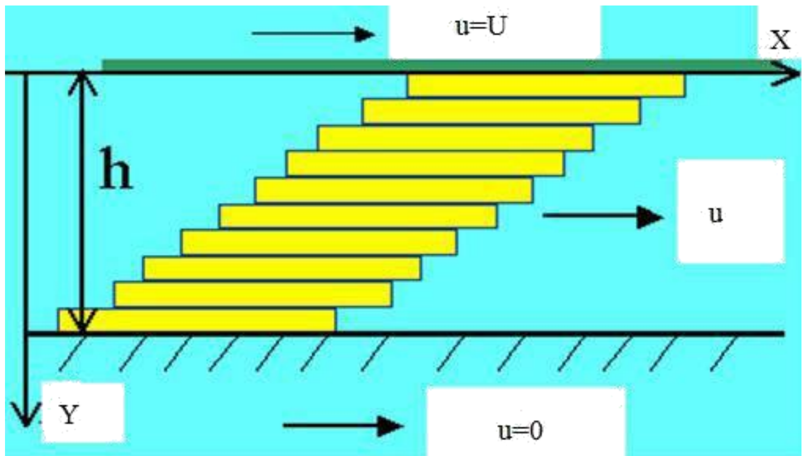


Fig 1.13: Viscous flow

where Z is the constant of proportionality and defines *absolute viscosity,* also called *dynamic viscosity*. The derivative *du*/*dy* is the rate of change of velocity with distance and may be called the rate of shear, or the velocity gradient. The viscosity Z is thus a measure of the internal frictional resistance of the fluid.

For most lubricating fluids, the rate of shear is constant, and *du*/*dy* = *U*/*h*. Fluids exhibiting this characteristic are known as a Newtonian fluids.

$$\text{Therefore } \tau = F/A = Z\,(U/h).$$

The absolute viscosity is measured by the pascal-second (Pa · s) in SI; this is the same as a Newton-second per square meter.
The poise is the CGS unit of dynamic or absolute viscosity, and its unit is the dyne second per square centimeter (dyn · s/cm2). It has been customary to use the centipoises (cP) in analysis, because its value is more convenient. The conversion from cgs units to SI units is as follows:

$$\mathbf{Z\ (Pa \cdot s) = (10)^{-3}\ \mathit{Z}\ (cP)}$$

Kinematic Viscosity is the ratio of the absolute Viscosity to the density of the lubricant.

$$\mathbf{Z_k = Z\ /}\ p$$

The ASTM standard method for determining viscosity uses an instrument called the Saybolt Universal Viscosimeter. The method consists of measuring the time in seconds for 60 mL of lubricant at a specified temperature to run through a tube 17.6 micron in diameter and 12.25 mm long. The result is called the *kinematic viscosity,* and in the past the unit of the square centimeter per second has been used. One square centimetre per second is defined as a **stoke.**

The kinematic viscosity based upon seconds Saybolt, also called *Saybolt Universal viscosity*

(SUV) in seconds, is given by:

$$\mathbf{Zk = (0.22t - 180/t)}$$

where Z_k is in centistokes (cSt) and *t* is the number of seconds Saybolt.

Viscosity -Temperature relation

Viscous resistance of lubricating oil is due to intermolecular forces. As the temperature increases, the oil expands and the molecules move further apart decreasing the intermolecular forces. Therefore the viscosity of the lubricating oil decreases with temperature as shown in the figure.1.14. If speed increases, the oil's temperature increases and viscosity drops, thus making it better suited for the new condition. An oil with high viscosity creates higher temperature and this in turn reduces viscosity. This, however, generates an equilibrium condition that is not optimum. Thus, selection of the correct viscosity oil for the bearings is essential.

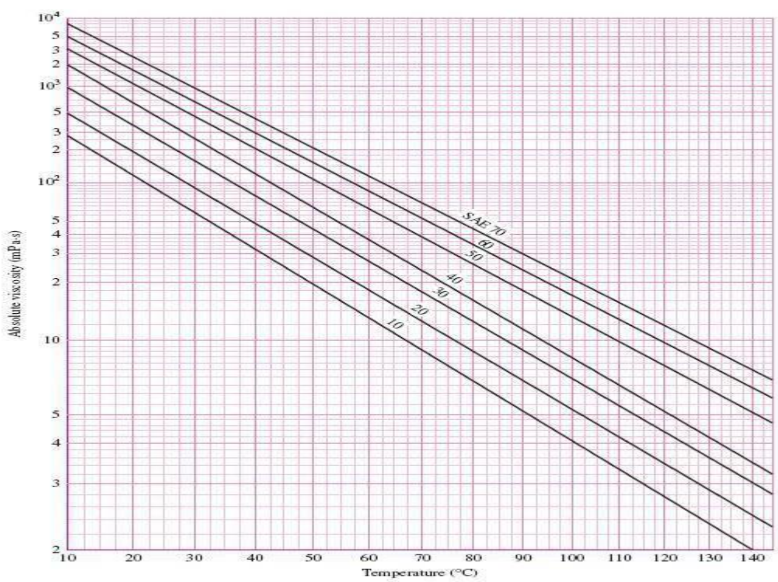


Fig.1.14 Viscosity temperature relationship

Viscosity index of a lubricating oil

Viscosity Index (V.I) is value representing the degree for which the oil viscosity changes with temperature. If this variation is small with temperature, the oil is said to have a high viscosity index. The oil is compared with two standard oils, one having a V.I. of 100 and the other Zero. A viscosity Index of 90 indicates that the oil with this value thins out less rapidly than an oil with V.I. of 50.

Types of lubricants

- Vegetable or Animal oils like Castor oil, Rapeseed oil, palm oil, Olive oil etc.
- Animal oils like lard oil, tallow oil, whale oil, etc.
- Mineral oils-petroleum based- Paraffinic and Naphthenic based oils

Properties of lubricants

- Availability in wide range of viscosities.
- High Viscosity index.
- Should be chemically stable with bearing material at all temperatures encountered.
- Oil should have sufficient specific heat to carry away heat without abnormal rise in temperature.
- Reasonable cost.

Selection Guide for Lubricants

The viscosity of lubricating oil is decisively for the right thickness of the lubricating film (approx. 3-30µm) under consideration of the type of lubricant supply

Low sliding speed	High Viscosity
High sliding speed	Low viscosity
High bearing clearance	High Viscosity
High load (Bearing pressures)	Higher Viscosity

Bearing materials

Relative **softness** (to absorb foreign particles), reasonable strength, **machinability** (to maintain tolerances), **lubricity**, **temperature and corrosion resistance**, and in some cases, **porosity** (to absorb lubricant) are some of the important properties for a bearing material.

A bearing element should be *less than one-third as hard* as the material running against it in order to provide **embedability** of abrasive particles.

A bearing material should have high compression strength to withstand high pressures without distortion and should have good fatigue strength to avoid failure due to pitting. e.g. in Connecting rod bearings, Crank shaft bearings, etc. A bearing material should have conformability. Soft bearing material has *conformability.* Slight misalignments of bearings can be self-correcting if plastic flow occurs easily in the bearing metal. Clearly there is a compromise between load-bearing ability and conformability.

In bearings operating at high temperatures, possibility of oxidation of lubricating oils leading to formation of corrosive acids is there. The bearing material should be **corrosion**

resistant. Bearing material should have easy **availability and low cost.** The bearing material should be soft to allow the dirt particles to get embedded in the bearing lining and avoid further trouble. This property is known as **Embeddability.**

Different Bearing Materials

- **Babbitt or White metal --** usually used as a lining of about 0.5mm thick bonded to bronze, steel or cast iron.
  - Lead based & Tin based Babbitt's are available.
  - Excellent conformability and embeddability
  - Good corrosion resistance.
  - Poor fatigue strength

- **Copper Based alloys** - most common alloys are copper tin, copper lead, phosphor bronze: harder and stronger than white metal: can be used **un-backed as a solid bearing.**
- **Aluminum based alloys** - running properties not as good as copper based alloys but cheaper.

  - **Ptfe -** suitable in very light applications
  - **Sintered bronze** - Sintered bronze is a porous material which can be impregnated with oil, graphite or Ptfe. Not suitable for heavily loaded applications but useful where lubrication is inconvenient.

  - **Nylon** - similar to Ptfe but slightly harder: used only in very light applications.

Triple-layer composite bearing material consists of 3 bonded layers: steel backing, sintered porous tin bronze interlayer and anti-wear surface as shown in figure 1.15. High load capacities and low friction rates, and are oil free and anti-wear.

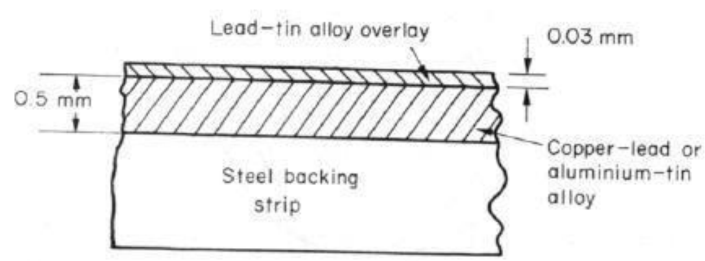


Fig.1.15 Tri-metal Bearing

If oil supply fails, frictional heating will rapidly increase the bearing temperature, normally lead to metal-to-metal contact and eventual seizure. Soft bearing material (low melting point) will be able to shear and may also melt locally. **Protects the journal** from severe surface damage, and helps to avoid component breakages (sudden locking of mating surfaces).

Petroff's Equation for lightly Loaded Bearings

The phenomenon of bearing friction was first explained by Petroff on the assumption that the shaft is concentric. This can happen when the radial load acting on the bearing is zero or very small, speed of the journal is very high and the viscosity of the lubricant is very high. Under these conditions, the eccentricity of the bearing (the offset between journal center and bearing center) is very small and the bearing could be treated as a concentric bearing as shown in figure 1.16

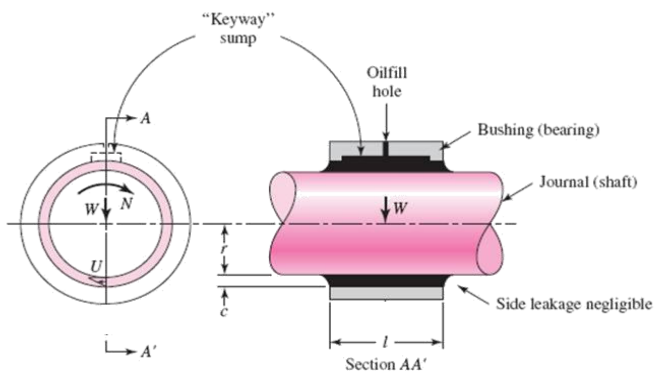


Fig.1.16 Concentric Bearing

Let us now consider a shaft rotating in a guide bearing. It is assumed that the bearing carries a very small load, that the clearance space is completely filled with oil, and that leakage is negligible (Fig. 7.16). Let the radius of the shaft be r, and the length of the bearing by l.
If the shaft rotates at N′ rev/s, then its surface velocity is $U = 2mr N'$. Since the shearing stress in the lubricant is equal to the velocity gradient times the viscosity,

$$\tau = Z\,U/h = 2mrN'Z/c$$

where the radial clearance c has been substituted for the distance **h.**

F= Frictional force= $\tau A = (2mrN'Z/c)\,(2mrl) = (4m2r^2lZN'/c)$

Frictional torque= $Fr = (4m2r^3lZN'/c)$

The coefficient of friction in a bearing is the ratio of the frictional force F to the Radial load W on the bearing. $f = F/W = (4m2r^3lZN'/cW)$

The unit bearing pressure in a bearing is given by p= W/2rL= Load/ Projected Area of the Bearing. Or **W= 2prL**

Substituting this in equation for f and simplifying

$$f = 2m^2\,(ZN'/p)\,(r/c)$$

This is the Petroff's equation for the coefficient of Friction in Lightly Loaded bearings.

Example on lightly loaded bearings

E1. A full journal bearing has the following specifications:

- **Journal Diameter:46 mm**
- **Bearing length: 66 mm**
- **Radial clearance to radius ratio: 0.0015**
- **speed : 2800 r/min**
- **Radial load: 820 N.**
- **Viscosity of the lubricant at the operating temperature:8.4 cP**

Considering the bearing as a lightly loaded bearing, Determine (a) the friction torque (b) Coefficient of friction under given operating conditions and (c) power loss in the bearing.

Solution:

Since the bearing is assumed to be a lightly loaded bearing, Petroff's equation for the coefficient of friction can be used.

$$\mathbf{f = 2m^2 (ZN'/p)\ (r/c)}$$

N = 2800/60=46.66 r/sec.

$Z = 8.4\ cP = 8.4 \times 10^{-3}\ Pa.sec$

r = 46/2 =23 mm = 0.023 m

P= w/2rL= 820/ 2X0.023X0.066= 270092 Pa.

Substituting all these values in the equation for f,**f = 0.019**

T=Frictional torque: Frictional force x Radius of the Journal

= (f W) r

= 0.019 x 820 x 0.023

= **0.358 N-m**

= 0.358 x 46.66/ 1000

= **0.016 kW**

HYDRODYNMIC JOURNAL BEARINGS

The film pressure is created by the moving surface itself pulling the lubricant into a wedge-shaped zone at a velocity sufficiently high to create the pressure necessary to separate the surfaces against the load on the bearing.

One type occurs when the rate of shear across the oil film is a constant value and the line representing the velocity distribution is a straight line. In the other type the velocity distribution is represented by a curved line, so that the rate of shear in different layers across the oil film is different. The first type takes place in the case of two parallel surfaces having a relative motion parallel to each other as shown in Fig.1.19.

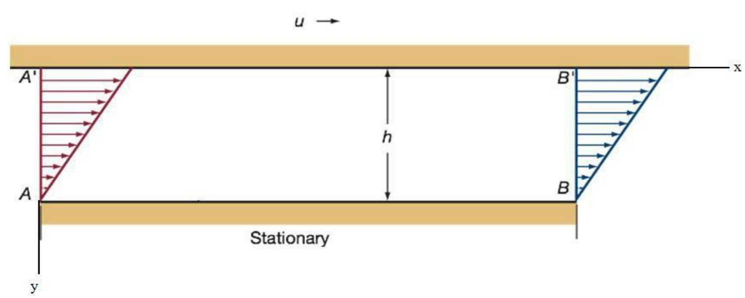


Fig 1.19: Velocity profiles in a parallel-surface slider bearing.

There is no pressure development in this film. This film cannot support an external Load. The second type of velocity distribution across the oil film occurs if pressure exists in the film. This pressure may be developed because of the change of volume between the surfaces so that a lubricant is squeezed out from between the surfaces and the viscous resistance of flow builds up the pressure in the film as shown in Fig 1.20 or the pressure may be developed by other means that do not depend upon the motion of the surfaces or it may develop due to the combination of factors. What is important to note here is the fact that pressure in the oil film is always present if the velocity distribution across the oil film is represented by a curved line

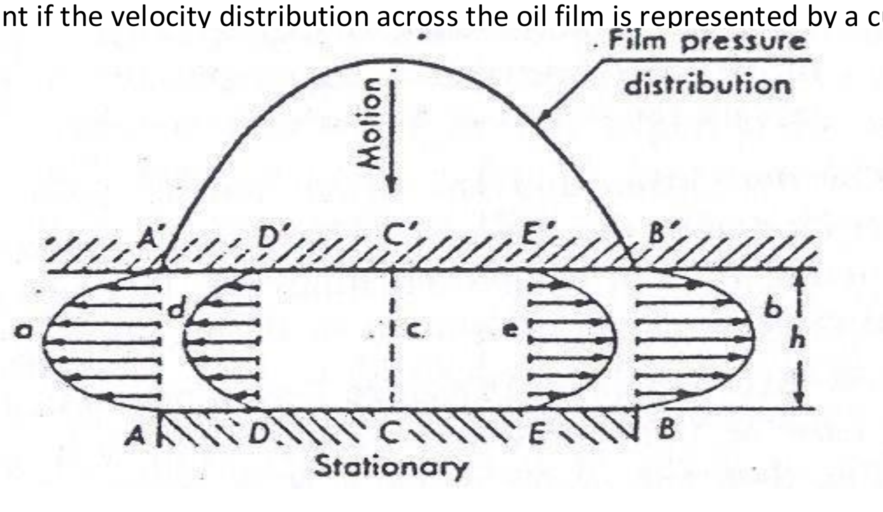


Fig 1.20: Flow between two parallel surface

Plate AB is stationary while A′ B′ is moving perpendicular to AB.

Note that the velocity distribution is ***<u>Curvilinear</u>***. This is a ***<u>pressure induced flow.</u>***

This film can support an External load.

Hydrodynamic film formation

Consider now the case of two non parallel planes in which one is stationary while the other is in motion with a constant velocity in the direction shown in Fig 1.21. Now consider the flow of lubricant through the rectangular areas in section AA’ and BB’ having a width equal to unity in a direction perpendicular to the paper.

The volume of the lubricant that the surface A’B’ tends to carry into the space between the surfaces AB and A’B’ through section AA’ during unit time is AC’A’. The volume of the lubricant that this surface tends to discharge from space through section BB’ during the same period of time is BD’B’. Because the distance AA’ is greater than BB’ the volume AC’A’ is greater than volume BC’B’ by a volume AEC’. Assuming that the fluid is incompressible and that there is no flow in the direction perpendicular to the motion, the actual volume of oil carried into the space must be equal to the discharge from this space. Therefore the excess volume of oil is carried into these space is squeezed out through the section AA’ and BB’ producing a constatnt pressure – induced flow through these sections.

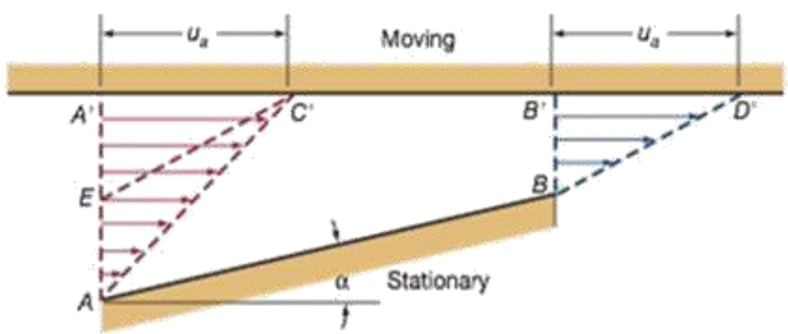


Fig.1.21 Velocity distribution only due to moving plate

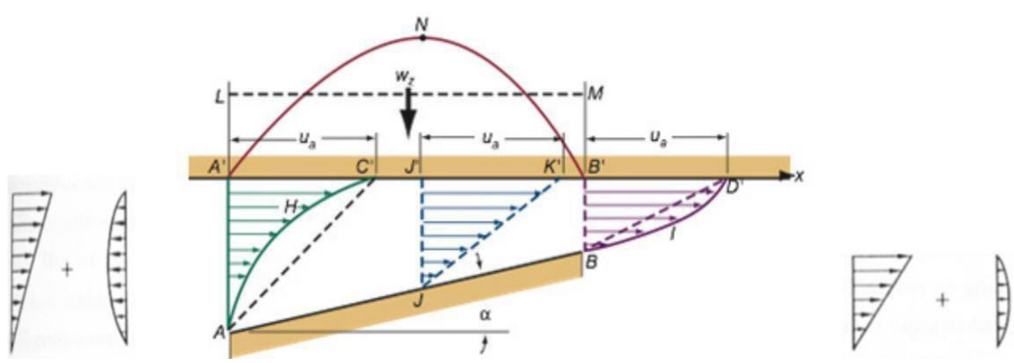


Fig.1.22 Resultant Velocity Distribution

The actual velocity distribution in section AA’ and BB’ is the result of the combined flow of lubricant due to viscous drag and due to pressure –induced flow. The resultant velocity distributions across these sections are as shown in Fig 1.22.

The curve A’NB’ shows the general character of the pressure distribution in the oil film and the line LM shows the mean pressure in the oil film. Because of the pressure developed in the oil film the, plane A’B’ is able to support the vertical load W applied to this plane, preventing metal to metal contact between the surfaces AB and A’B’. This load is equal to the product of projected area of the surface AB and mean pressure in the oil film.

Conditions to form hydrodynamic lubrication

There must be a wedge-shaped space between two relative moving plates;

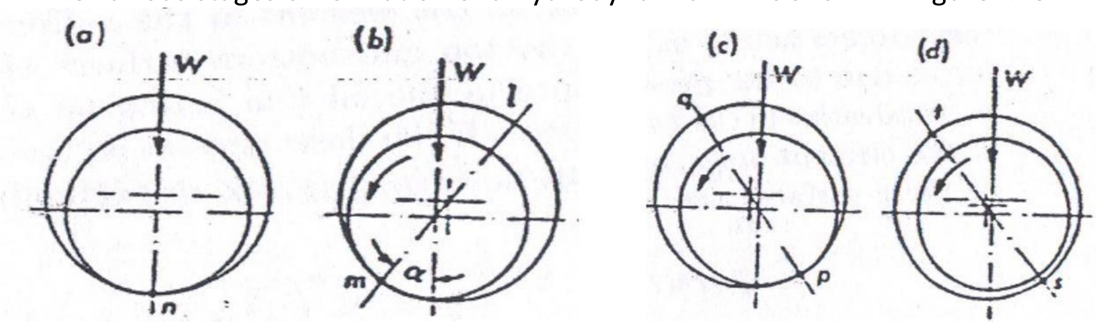
There must be a relative sliding velocity between two plates, and the lubricant must flow from big entrance to small exit in the direction of the moving plate;

The lubricant should have sufficient viscosity, and the supply of the lubricant is abundant.

Formation of oil film in a Journal bearing

Imagine a journal bearing with a downward load on the shaft that is initially at rest and then brought up to operating speed. At rest (or at slow shaft speeds), the journal will contact the lower face of the bearing as shown in the figure 1.23. This condition is known as boundary lubrication and considerable wear can occur. As shaft speed increases, oil dragged around by the shaft penetrates the gap between the shaft and the bearing so that the shaft begins to “float” on a film of oil. This is the transition region and is known as thin-film lubrication. The journal may occasionally contact the bearing particularly when shock radial load occur. Moderate wear may occur at these times. At high speed, the oil film thickness increases until there comes a point where the journal does not contact the bearing at all. This is known as thick film lubrication and no wear occurs because there is no contact between the journal and the bearing.

The various stages of formation of a hydrodynamic film is shown in figure1.23.



a) Journal at rest

b) Journal position during starting

c) Journal position after increase in speed

d) Journal position Under operating conditions

Pressure distribution around an idealised journal bearing

A typical pressure distribution around the journal in a hydrodynamic bearing is as shown in the Fig. 1.24.

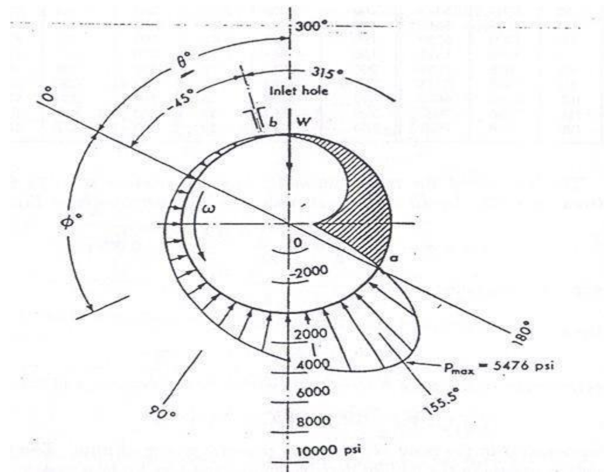
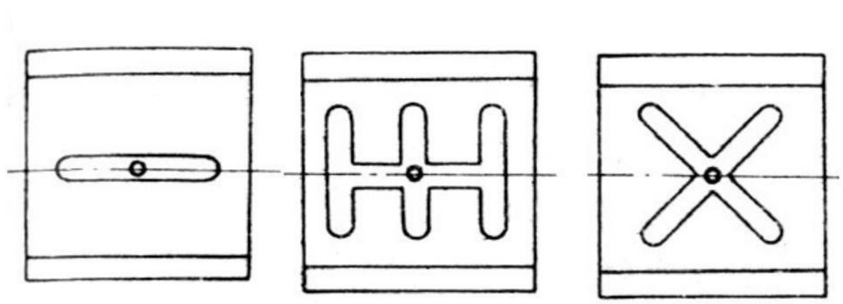


Fig.1.24: Bearing pressure distribution in a journal bearing

Typical oil groove patterns



Some typical groove patterns are shown in the above figure. In general, the lubricant may be brought in from the end of the bushing, through the shaft, or through the bushing. The flow may be intermittent or continuous. The preferred practice is to bring the oil in at the center of the bushing so that it will flow out both ends, thus increasing the flow and cooling action.

Thermal aspects of bearing design

Heat is generated in the bearing due to the viscosity of the oil. The frictional heat is converted into heat, which increases the temperature of the lubricant. Some of the lubricant that entersthe bearing emerges as a side flow, which carries away some of the heat. The balance of the lubricant flows through the load-bearing zone and carries away the balance of the heat generated. In determining the viscosity to be used we shall employ a temperature that is the average of the inlet and outlet temperatures, or

$$T_{av}=(T_i+T)/2$$

where= (T_i + T) is the inlet temperature and T is the temperature rise of the lubricant from inlet to outlet. The viscosity used in the analysis must correspond to T_{av}.

Self contained bearings:

These bearings are called ***self contained*** bearings because the lubricant sump is within the bearing housing and the lubricant is cooled within the housing. These bearings are described as *pillow-block* or *pedestal* bearings. They find use on fans, blowers, pumps, and motors, for example. Integral to design considerations for these bearings is dissipating heat from the bearing housing to the surroundings at the same rate that enthalpy is being generated within the fluid film.

Heat dissipated based on the projected area of the bearing:

Heat dissipated from the bearing, J/S $\quad H_D= CA\ (t_B-t_A)$

Where C= Heat dissipation coefficient from data hand book

Another formula to determine the heat dissipated from the bearing $H_D= ld\ (T+18)^2/ K_3$

Where K_3= 0.2674x 10^6 for bearings of heavy construction and well ventilated = 0.4743x10^6 for bearings of light construction in still air air

$T= t_B - t_A$

Where,

t_B = Bearing surface temperature

t_A = Ambient temperature

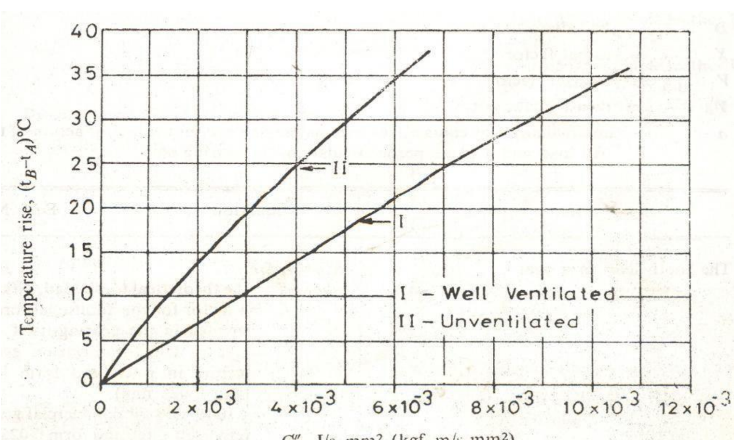


Fig. 15.16 Coefficient, C″

For good performance the following factors should be considered.

Surface finish of the shaft (journal): This should be a fine ground finish and preferably lapped.

Surface hardness of the shaft: It is recommended that the shaft be made of steel containing at least 0.35-0.45% carbon. For heavy duty applications shaft should be hardened.

Grade of the lubricant: In general, the higher the viscosity of the lubricant the longer the life. However the higher the viscosity the greater the friction, so high viscosity lubricants should only be used with high loads. In high load applications, bearing life may be extended by cutting a grease groove into the bearing so grease can be pumped in to the groove.

Heat dissipation: Friction generates heat and causes rise in temperature of the bearing and lubricant. In turn, this causes a reduction in the viscosity of the lubricating oil and could result in higher wear. Therefore the housing should be designed with heat dissipation in mind. For example, a bearing mounted in a Bakelite housing will not dissipate heat as readily as one mounted in an aluminium housing.

Shock loads: Because of their oil-cushioned operation, sliding bearings are capable of operating successfully under conditions of moderate radial shock loads. However excessive prolonged radial shock loads are likely to increase metal to metal contact and reduce bearing life. Large out of balance forces in rotating members will also reduce bearing life.

Clearance: The bearings are usually a light press fit in the housing. A shouldered tool is usually used in arbour press. There should be a running clearance between the journal and the bush. A general rule of thumb is to use a clearance of 1/1000 of the diameter of the journal.

Length to diameter ratio (l/d ratio): A good rule of thumb is that the ratio should lie in the range 0.5-1.5. If the ratio is too small, the bearing pressure will be too high and it will be difficult to retain lubricant and to prevent side leakage. If the ratio is too high, the friction will be high and the assembly misalignment could cause metal to metal contact.

Examples on journal bearing design

1. **Following data are given for a 360º hydrodynamic bearing: Radial load=3.2 kN**
   Journal speed= 1490
   r.p.m. Journal
   diameter=50 mm Bearing
   length=50mm Radial
   clearance=0.05 mm
   Viscosity of the lubricant= 25 cP

   Assuming that the total heat generated in the bearing is carried by the total oil flow in the bearing, calculate:
   - **Power lost in friction;**
   - **The coefficient of friction;**
   - **Minimum oil film thickness**
   - **Flow requirement in 1/min; and**
   - **Temperature rise.**

   Solution:

   P= W/Ld = 3.2x1000/ (50x50) =1.28 MPa.= 1.28×10^6

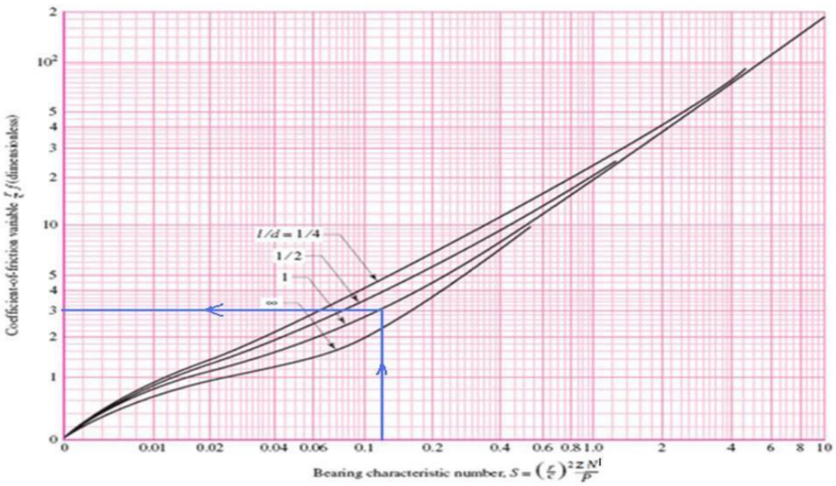
   Pa Sommerfeld number = S= (ZN′/p) $(r/c)^2$

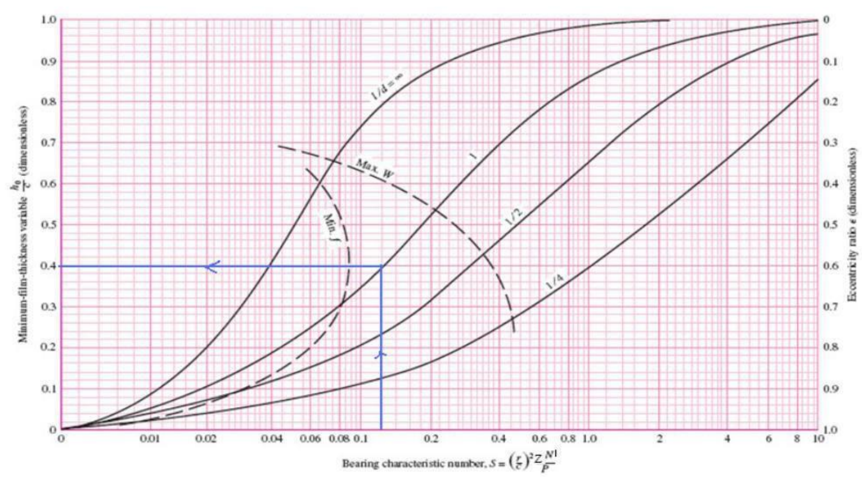
   r/c =25/0.0.05 = 500
   Z= 25 cP = 25×10^{-3} Pa.sec
   = 1490/60= 24.833 r/sec. Substituting the above values, we get
   S=0.121
   For S= 0.121 & L/d=1,
   Friction variable from the graph= (r/c) f= 3.22
   Minimum film thickness variable= h_o /c =0.4
   Flow variable= Q/rcN′L= 4.33

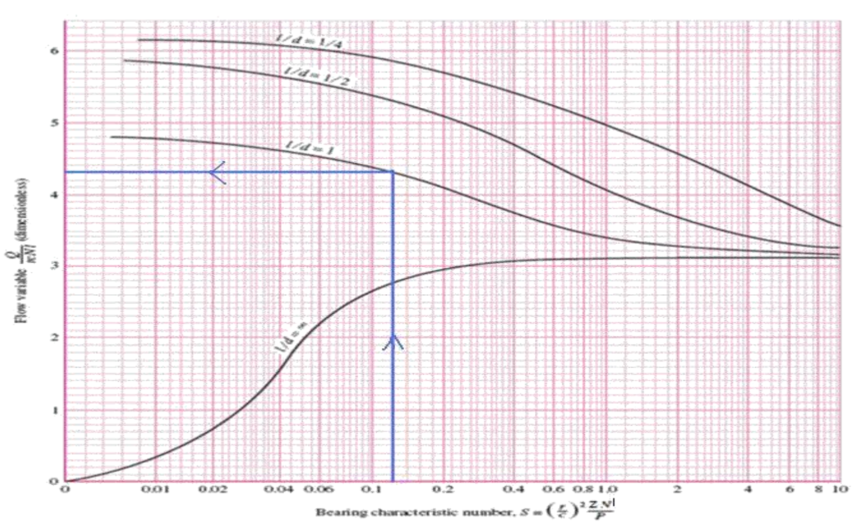
   f = 3.22x0.05/25= 0.0064
   Frictional torque= T= fWr = 0.0064x3200x 0.025
   = 0.512 N-m
   Power loss in the Bearing= 2m N′ T/ 1000 kW
   = 0.080 kW

   h_o = 0.4x 0.05= 0.02 mm

   Q/r c N^{l} L= 4.33 from which we get, Q= 6720.5 mm^3 / sec.


Coefficient-of-friction variable $\frac{r}{c}f$ (dimensionless)
Bearing characteristic number, $S = \left(\frac{r}{c}\right)^2 \frac{ZN'}{P}$
$l/d = 1/4$
1/2
1
∞


Minimum-film-thickness variable $\frac{h_0}{c}$ (dimensionless)
Eccentricity ratio ε (dimensionless)
Bearing characteristic number, $S = \left(\frac{r}{c}\right)^2 \frac{ZN'}{P}$
$l/d = \infty$
1
1/2
1/4
Max. W
Min. f


Flow variable $\frac{Q}{rcNl}$ (dimensionless)
Bearing characteristic number, $S = \left(\frac{r}{c}\right)^2 \frac{ZN'}{P}$
$l/d = 1/4$
$l/d = 1/2$
$l/d = 1$
$l/d = \infty$

Example E2:

A 50 mm diameter hardened and ground steel journal rotates at 1440 r/min in a lathe turned bronze bushing which is 50 mm long. For hydrodynamic lubrication, the minimum oil film thickness should be five times the sum of surface roughness of journal bearing. The data about machining methods are given below:

	Machining method	surface Roughness(c.l.a)
Shaft	grinding	1.6 micron
Bearing	turning/boring	0.8 micron

The class of fit is H8d8 and the viscosity of the lubricant is 18 cP. Determine the maximum radial load that the journal can carry and still operate under hydrodynamic conditions.

Solution:

Min. film thickness = h_o= 5 [0.8+1.6] = 12 micron = 0.012 mm For H8 d8 fit, referring to table of tolerances,

Ø50 H8 = Min. hole limit = 50.000 mm

Max. hole limit = 50.039 mm

Mean hole diameter= 50.0195 mm

Ø 50 d8 = Max. shaft size = 50- 0.080= 49.920 mm Min. shaft size = 50- 0.119= 49.881 mm

Mean shaft diameter= 49.9005 mm.

Assuming that the process tolerance is centered,

Diametral clearance= 50.0195- 49.9005= 0.119 mm

Radial clearance= 0.119/2= 0.0595 mm

h_o /c = 0.012/ 0.0595 = 0.2

L/d = 50/50= 1

From the graph, Sommerfeld number= 0.045

$S= (ZN'/p)\,(r/c)^2 = 0.045$

r/c= 25/0.0595= 420.19

Z= 18 cP= 18×10^{-3} Pa.sec

N'= 1440/60= 24 r/sec

From the above equation, Bearing pressure can be calculated.

p= 1.71×10^{6} Pa = 1.71 MPa.

The load that the bearing can carry:

W= pLd = 1.71x 50x 50= 4275 N

Example E3:

The following data are given for a full hydrodynamic journal bearing:

Radial load=25kN

Journal speed=900 r/min.

Unit bearing pressure= 2.5

MPa (l/d) ratio= 1:1

Viscosity of the lubricant=20cP Class of fit=H7e7

Calculate: 1.Dimensions of bearing

2. Minimum film thickness and

3. Requirement of oil flow

Solution:

$N' = 900/60 = 15$ r/sec

$P=W/Ld$

$2.5= 25000/Ld=25000/d^2$

As L=d.

d= 100 mm & L=100 mm

For H7 e7 fit, referring to table of tolerances,

Ø100 H7 = Min. hole limit = 100.000 mm

Max. hole limit = 100.035 mm

Mean hole diameter= 100.0175 mm

Ø 100 e7 = Max. shaft size = 100- 0.072= 99.928 mm Min. shaft size = 100- 0.107= 99.893 mm

Mean shaft diameter= 99.9105 mm

Assuming that the process tolerance is centered, Diametral clearance= 100-0175- 99.9105= 0.107 mm Radial clearance= 0.107/2= 0.0525mm

Assume r/c = 1000 for general bearing applications.

C= r/1000=50/1000 = 0.05 mm.

$Z= 20$ cP$= 20\times10^{-3}$ Pa.sec

$N^{l} = 15$ r/sec

$P= 2.5$ MPa$= 2.5 \times10^{6}$ Pa

$S= (ZN'/p)(r/c)^2 = 0.12$

For L/d=1 & S=0.12, Minimum Film thickness variable= $h_o/c = 0.4$

$h_o = 0.4\times 0.05= 0.02$ mm

Example E4:

A journal bearing has to support a load of 6000N at a speed of 450 r/min. The diameter of the journal is 100 mm and the length is 150mm.The temperature of the bearing surface is limited to 50 ºC and the ambient temperature is 32 ºC. Select a suitable oil to suit the above conditions.

Solution:

N^l = 450/60 =7.5 r/sec, W=6000 N, L=150mm, d=100 mm, t_A = 32 ºC, t_B = 50 ºC.
Assume that all the heat generated is dissipated by the bearing.

Use the Mckee's Equation for the determination of coefficient of friction.

f=Coefficient of friction= K_a (ZN^l /p) (r/c) 10^{-10} +f

p= W/Ld= 6000/100x150 = 0.4 MPa.
K_a = 0.195x 10^6 for a full bearing
f = 0.002
r/c= 1000 assumed
U= 2mrN^l = 2x3.14x 50x7.5= 2335 mm/sec= 2.335 m/sec.
f = 0.195x 10^6 x (Z * 7.5 / 0.4) x 1000 x 10^{-10} +0.002
f = 0.365Z+0.002

Heat generated= f *W*U
Heat generated= (0.365Z+ 0.002)x6000x2.335

Heat dissipated from a bearing surface is given by:

H_D= ld $(T+18)^2$/ K_3

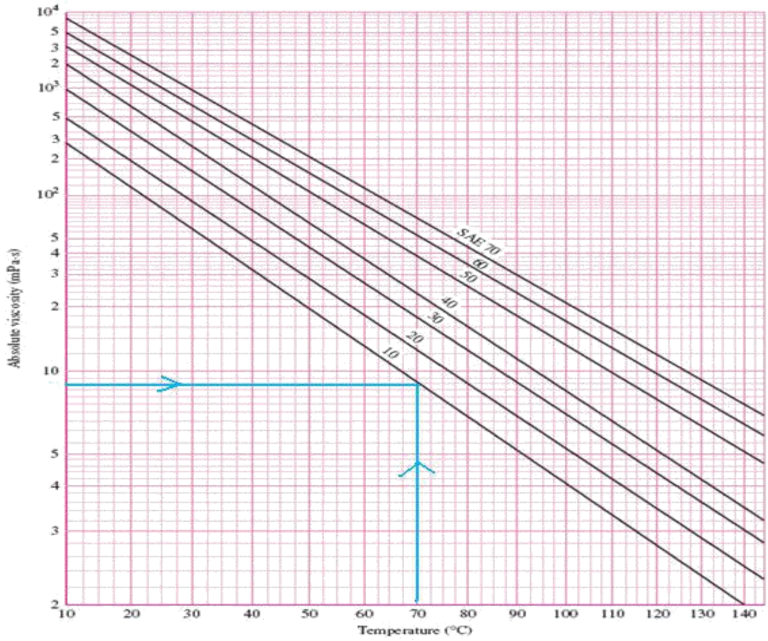
Where K_3= 0.2674x 10^6 for bearings of heavy construction and well ventilated = 0.4743x10^6 for bearings of light construction in still air air

T= t_B - t_A = 50-32 =18ºC
H_D = 150x100$(18+18)^2$ / 0.2674x10^6 =72.7 Watt

H_D = Hg for a self contained bearing.

72.7 = (0.365Z+ 0.002)x6000x2.335
Z= 0.0087 Pa.Sec.

Relation between oil temp, Amb. temp, & Bearing surface temperature is given by
$t_B - t_A$= ½ (t_O- t_A)
t_O = oil temperature= 68 ºC

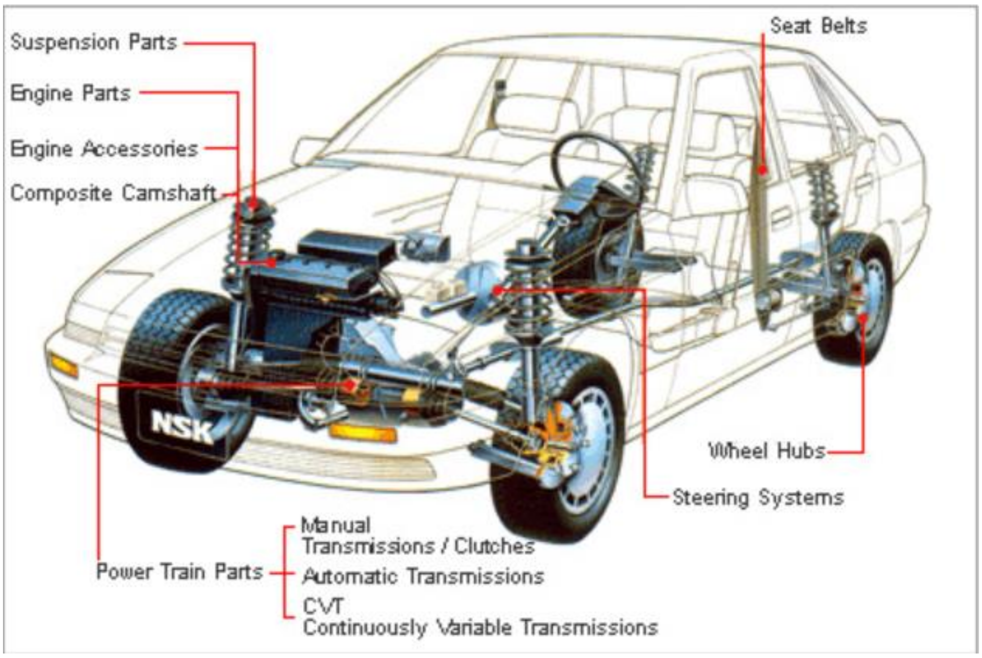


Select SAE 10 Oil for this application

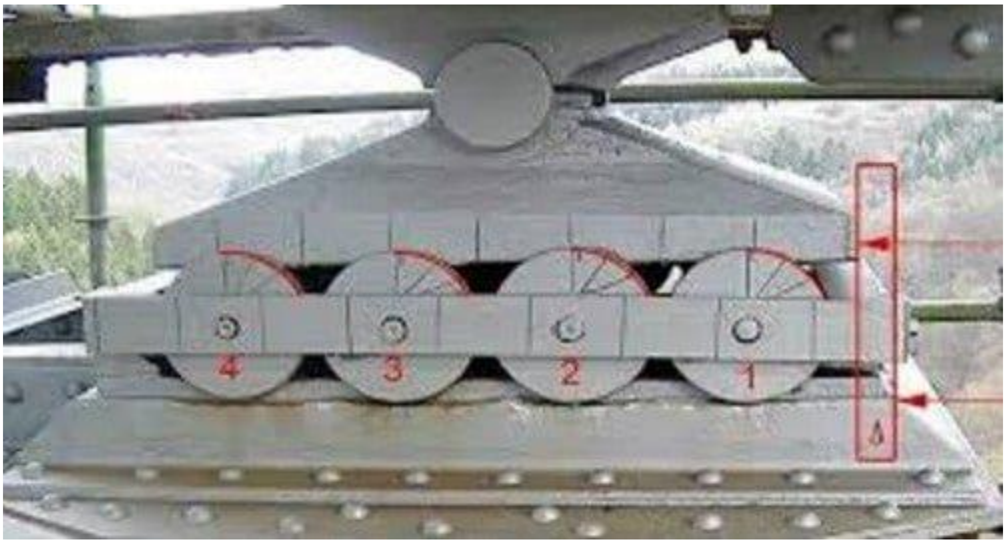
INDUSTRIAL APPLICATIONS

1. Power transmission in industries

2. Automobile



3. Bearings in bridge structure



TUTORIAL QUESTIONS

UNIT 1

1. Design a bearing and journal to support a load of 4500N at 600 rev/min using a hardened steel journal and a bronze backed Babbitt bearing. The bearing is lubricated by oil rings. Take room temperature as 21^0 C and the oil temperature as 80^0 C.
2. A ball bearing operates on the following work cycle:

Element No	Radial load (N)	Speed (R.P.M)	Element time (%)
1.	3000	720	30
2.	7000	1440	40
3.	5000	900	30

The dynamic load capacity of the bearing is 16600N. Calculate The average speed of rotation; b) The equivalent radial load c) the bearing life.

3. The following data is given for a 360^0 hydrodynamic bearing:

Journal diameter = 100 mm ; Bearing length = 100
mm Radial load = 50 kN ; Journal speed = 1440
rpm Radial clearance = 0.12 mm ; Viscosity of lubricant = 16 cp

Calculate (a) minimum film thickness (b) coefficient of friction and (c) power lost in friction.

4. a) Define dynamic load carrying capacity of rolling contact bearing.

b) The radial load acting on a ball bearing is 2500 N for the first 5 revolutions and reduces to 1500 N for the next ten revolutions. The load variation then repeats itself. The expected life of the bearing is 20 million revolutions. Determine the dynamic load carrying capacity of the bearing.

5. A bearing for an axial flow compressor is to carry a radial load of 2500 N and thrust of 1500 N. The service imposes light shock and the bearing will be in use for 40 hours/week in 5 years. The speed of the shaft is 1000 rpm. Select suitable ball bearing for the purpose and give the required tolerances on the shaft and the housing. Diameter of the shaft is 50 mm.

ASSIGNMENT QUESTIONS

1. A rolling contact bearing is subjected to the following work cycle: (a) Radial load of 6000 N at 150 rpm for 25 % of the time; (b) Radial load of 7500 N at 600 rpm for 20 % of the time; and (c) Radial load of 2000 N at 300 rpm for 55 % of the time . The inner ring rotates and loads are steady. Select a bearing for an expected average life of 2500 hours.
2. Select a ball bearing to carry satisfactorily a 65KN radial load together with 10 KN of thrust load. The journal supported by the bearing rotates at 1400 rpm for an estimated 0.1 million hours of life. The journal diameter is 100 mm.
3. A 80 mm long journal bearing supports a load of 2800 N on a 50 mm diameter shaft. The bearing has a radial clearance of 0.05 mm and the viscosity of the oil is 0.021 kg / m-s at the operating temperature. If the bearing is capable of dissipating 80 J/s, determine the maximum safe speed.
4. A 100 mm long and 60 mm diameter journal bearing supports a load of 2500 N at 600 rpm. If the room temperature is 20^0C , what should be the viscosity of oil to limit the bearing surface temperature to 60^0C ? The diametral clearance is 0.06 mm and the energy dissipation coefficient based on projected area of bearing is 210 W/ $m^2/^0$C.

BEARINGS

UNIT 1

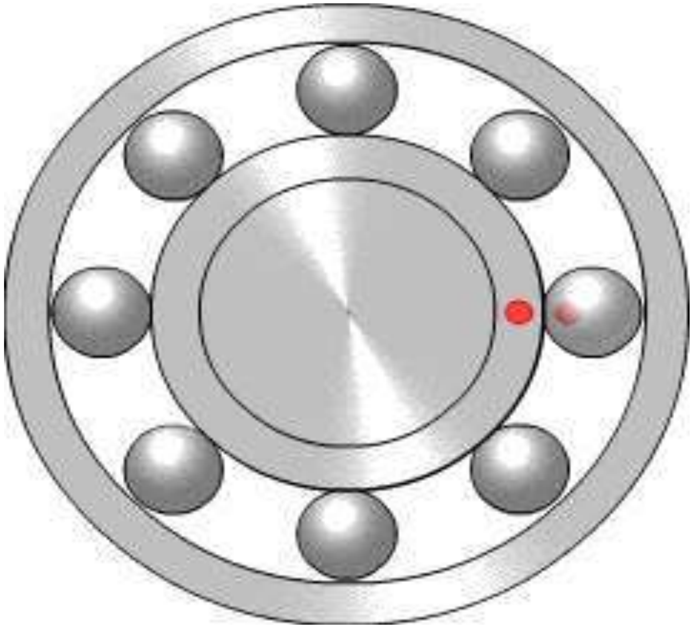
MRCET CAMPUS

UGC Autonomous

DEPARTMENT OF MECHANICAL ENGINEERING

BEARINGS

- A bearing is a machine element that constrains relative motion between moving parts to only the desired motion



APPLICATIONS

- Today ball and roller bearings are used in many applications which include a rotating component.
- Examples include ultra high speed bearings in dental drills, Aerospace Bearings in the Mars Rover, gearbox and wheel bearings on automobiles, flexure bearings in optical alignment systems and bicycle wheel hubs.

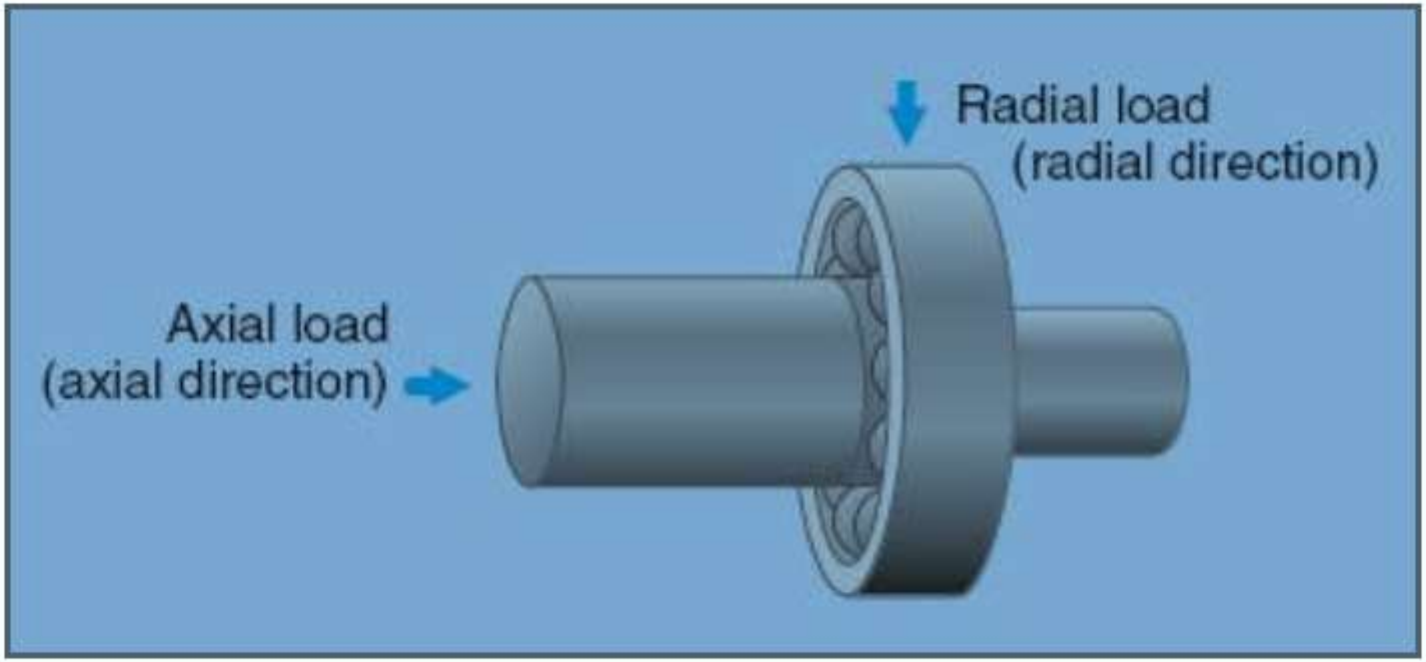
FUNCTIONS OF BEARING

- The main function of Bearing is rotating shaft is to transmit power from one end of the line to the other.
- It needs a good support to ensure stability and frictionless rotation. The support for the shaft is known as **“bearing”.**
- The shaft has a “running fit” in a bearing. All bearing are provided some lubrication arrangement to reduced friction between shaft and bearing.

CLASSIFICATION

- **PLAIN BEARING/SLIDER BEARING:** In which the rotating shaft has a sliding contact with the bearing which is held stationary .
- Due to large contact area friction between mating parts is high requiring greater lubrication.
- **ROLLING OR ANTI-FRICTION BEARING:** Due to less contact area rolling friction is much lesser than the sliding friction , hence these bearings are also known as antifriction bearing.

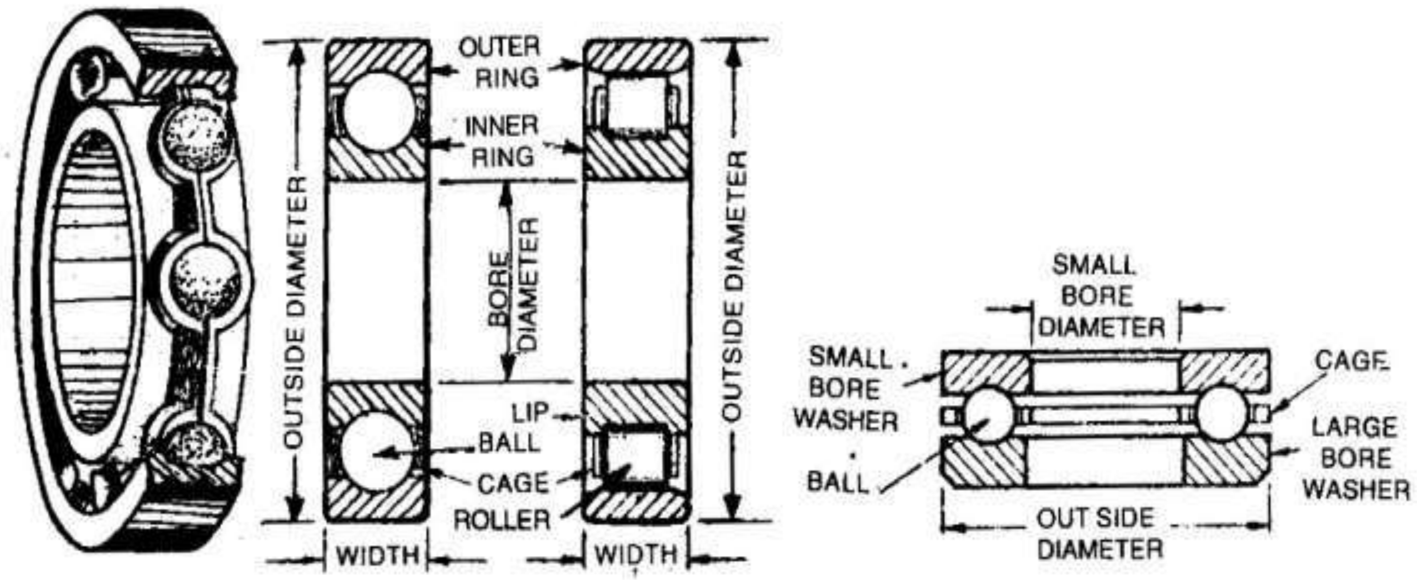
LOAD DIRECTION



BALL & ROLLER BEARINGS

- Frictional resistance considerably less than in plain bearings
- Rotating – non-rotating pairs separated by balls or rollers
- Ball or rollers has rolling contact and sliding friction is eliminated and replaced by much lower rolling friction.
- In plain bearing the starting resistance is much larger than the running resistance due to absence of oil film.
- In ball and rolling bearings the initial resistance to motion is only slightly more than their resistance to continuous running.

BALL & ROLLER BEARINGS



(*a*) Ball bearing (*b*) Rolled bearing (*c*) Thrust ball bearing

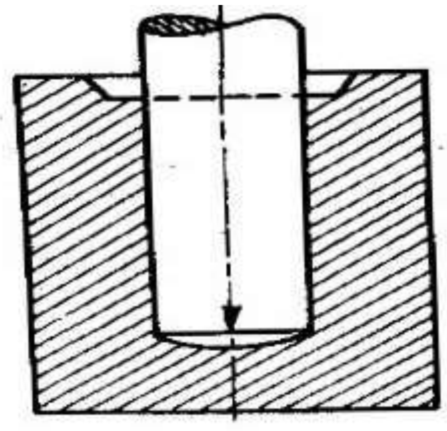
Ball and roller bearing

TYPES OF ROLLING BEARINGS

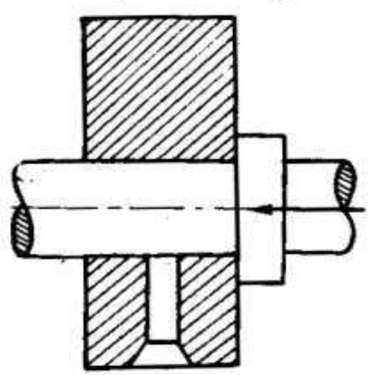
- **SINGLE ROW DEEP GROOVE BALL BEARING:** Incorporating a deep hardened raceway which makes them suitable for radial and axial loads in either direction, provided the radial loads are greater than the axial loads.
- **SINGLE ROW ROLLER BEARINGS:** Roller bearing have a greater load-carrying capacity than ball bearing of equivalent size as they make line contact rather than point contact with their rings. e axial loads.

CLASSIFICATION OF THE SLIDING CONTACT BEARING

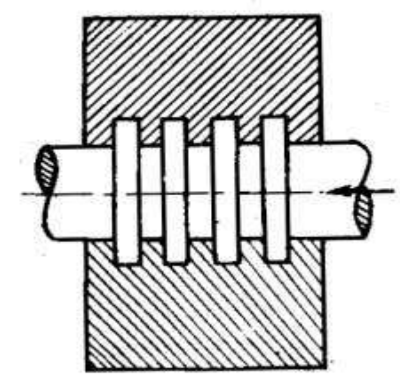
- JOURNAL BEARING
- FOOTSTEP BEARING
- COLLAR THRUST BEARING



(*b*) **Foot-step bearing**



(*c*) Thrust bearing

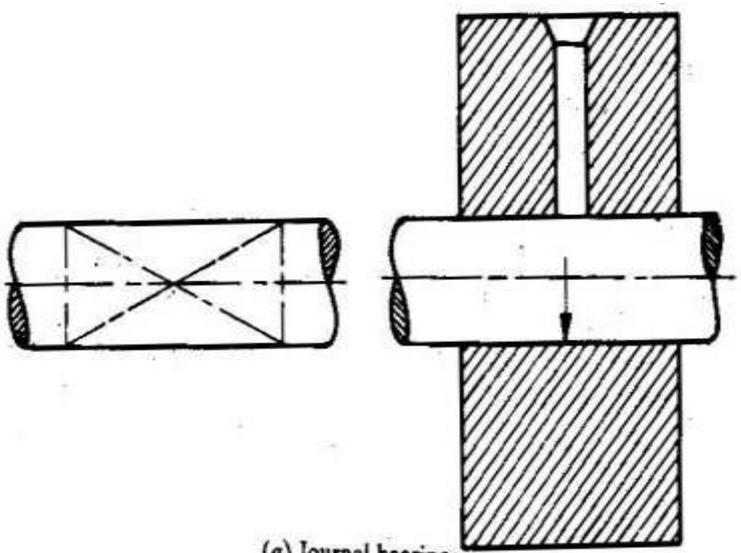


(*d*) Thrust bearing, multiple collar

Kinds of bearings

JOURNAL BEARING

- Journal bearing – in this the bearing pressure is exerted at right angles to the axis of the axis of the shaft. The portion of the shaft lying within the bearing in known as journal.
- Shaft are generally made of mild steel.



(a) Journal bearing