SLIDING CONTACT BEARINGS

17.1. BEARINGS - An Introduction :

One of the major problems faced by Engineers when developing the working characteristics of machine elements has been how to utilize the useful energy received from various sources in the optimum way. Among the various factors like type of load, environmental conditions, material properties, labour skill which are affecting the working characteristics of machine parts, the surface roughness and friction act as predominant factors. *The friction is defined as the resisting force that is developed between the two relatively moving surfaces.*

For some machine elements, the friction is used as an eminent property without which they can not be operated and for some other machine elements this friction should be an avoidable one. For the first category, the brakes and friction clutches are considered as good examples and the bearing is a notable example for the second category because in clutches, the friction between the clutch discs is used as an useful medium for transmitting power from the engine to the wheels through propeller shaft and in brakes, the friction is used to slow down the speed of the brake-drum when the vehicle is to be stopped, whereas in bearings the friction should be reduced to a maximum level in-order to help for smooth running of vehicle.

The bearing is a machine member which supports a moving part and confines its motion. This motion may be rotational or linear. The bearings employed in vehicles, generators, pumps, turbines etc. are used for rotational motion whereas in Co-ordinate measuring machine, the bearing is used for linear motion. The selection of the bearing type depends on quite a few design and service factors.

17.2. CLASSIFICATION OF BEARINGS :

On the basis of friction caused by relative movement of the bearing surfaces, the bearings are classified as,

1. Sliding contact bearings or Plain bearings.
2. Rolling contact bearings or Antifriction bearings.

Depending upon the direction of load applied, they may be grouped into

1. Radial bearings or Circumferentially loaded bearings.
2. Thrust bearings or Axially loaded bearings.

The detailed classification has been given in the table 17.1

17.3. SLIDING CONTACT BEARINGS, (Parts and types) :

The sliding contact bearing consists of two main parts namely shaft and sleeve. The shaft rotates inside the sleeve. (also called as bearing itself). For avoiding friction, a lubricant is supplied between the mating surfaces. This lubricant carries away the heat generated during operation and reduces its wear and maintains the operating temperature to a safety level.

These sliding contact bearings are generally classified on the basis of usage as plain journal bearing, thrust bearing, spherical bearing, pivot bearing. Depending upon the lubricant applied, they may classified into;

a) **Zero film bearings:** in which no lubricant is present and hence metal to metal contact will occur and so they are not generally preferred.

b) **Thin film bearings:** in which the lubricant is very thin due to its low viscosity and hence the lubricant is no longer able to separate the moving surfaces, partial metal to metal contact can occur. This type of lubrication is also called as ***"Boundary lubrication"***.

c) **Thick film bearings:** in which high viscous lubricant is supplied which separates the rotating surfaces and hence metal to metal contact may be prevented. This type of lubrication is called ***"Hydrodynamic lubrication"*** and most preferable one.

d) **Externally pressurised lubricated bearings:** in which the lubricant, which may sometimes be air or water, is supplied with a sufficient pressure to separate the rotating surfaces as similar to thick film lubrication. This kind is called as ***"Hydro-static lubrication"***.

There is one more sliding contact bearing in which a slider slides on a flat surface as employed in Co-ordinate measuring machine.

In this chapter, the design of hydro-dynamic lubricated bearings, particularly journal bearings, has been mainly focused.

17.4. JOURNAL BEARINGS :

A sliding contact bearing that furnishes lateral support to the rotating shaft is known as journal bearing. It consists of two main parts, a shaft and a housing, as shown in figure 17.1. The portion of the shaft inside the housing, also known as bearing, is called as journal. In most of the applications, the journal rotates while the bearing is stationary. However, there are some applications where the journal is stationary and the bearing rotates and even somewhere both the journal and bearing rotate.

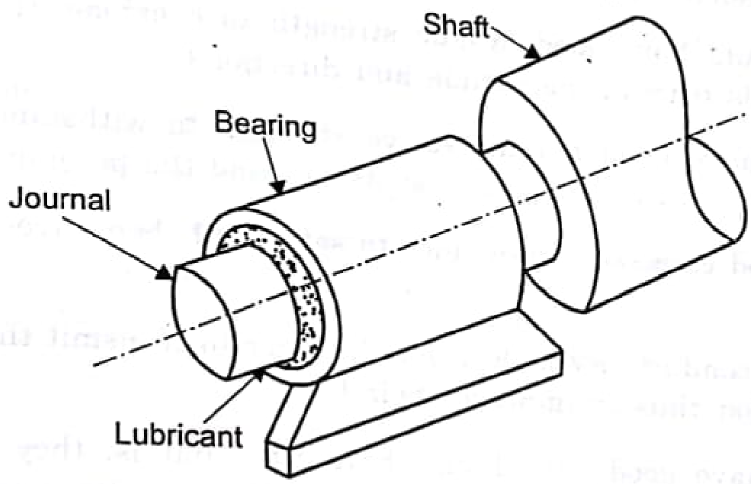
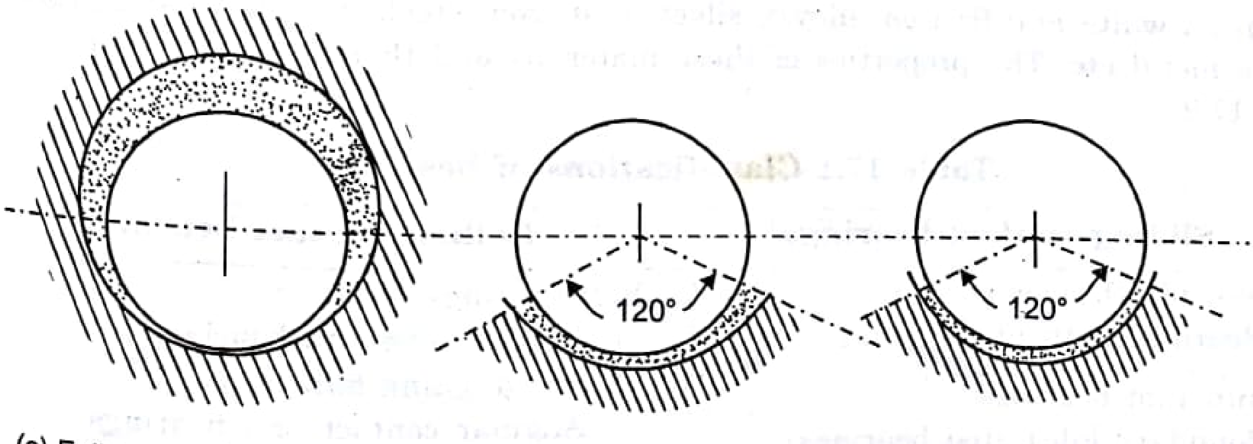


Fig. 17.1: Journal Bearing

This journal bearing may be classified as ***"full journal bearing"*** and ***"partial journal bearing"*** depending upon whether the journal is fully or partially covered by the bearing as shown in fig. 17.2. That is, for the full journal bearing, the angle of contact is 360° whereas the angle of contact for partial journal bearing is less



(a) Full journal Bearing (b) Partial Journal Bearing (c) Fitted Journal Bearing

Fig. 17.2: Types of Journal Bearings

than 360° and especially in between 90° to 180° and also 120° is the preferred angle of contact. A partial journal bearing having zero clearance is known as a *"fitted bearing"*. Zero clearance simply means that the radii of the journal and bearing are equal.

17.5. BEARING – MATERIALS :

Eventhough the sliding contact bearing is having sufficient lubrication which separates the rotating parts completely, the materials used for making journal and bearing should have some essential characteristics for efficient functioning.

1. Materials should have good fatigue strength to overcome the failure due to change of loads both in magnitude and directions.
2. They should have enough compressive strength to withstand the maximum pressure developed by radial loads so as to avoid the permanent deformation.
3. They need good corrossion resistance to safeguard themselves from corrossive atmospheres.
4. Good thermal conductivity is desirable in order to transmit the heat produced during operation thus to improve their life.
5. They should have good embedding character. That is, they may be able to absorb foreign materials like dust, grit etc which results scoring of materials which is to be prevented.
6. The sliding contact bearing materials should be the easily available and may be cheaper.

Some of the important types of materials employed for sliding contact bearings are aluminium alloys, copper alloys, babbit alloys (ie, alloys of tin, lead and antimony known as white anti-friction alloys), silver, cast iron, steel, various plastics, rubber, porous metal etc. The properties of these materials and their uses are given in the table 17.2.

Table 17.1 Classifications of bearings

Sliding contact bearings	Rolling contact bearings
1. Zero film bearings (Bearings without lubricant) 2. Thin film bearings (Boundary lubricated bearings)	1. Ball bearings a) Deep groove ball bearings b) Self-aligning ball bearings c) Angular contact ball bearings i) Single row angular contact ii) Double row angular contact d) Thrust bearings i) Single thrust and ii) Double thrust

Sliding contact bearings	Rolling contact bearings
3. Thick film bearings Hydro - dynamic bearings) a) Full journal bearings b) Partial journal bearing	2. Roller bearings a) Cylindrical roller bearings b) Spherical roller bearings c) Taper roller bearings d) Needle roller bearings e) Thrust roller bearings
4. Externally pressurised bearings (Hydro-static bearings) a) Oil journal bearing b) Air journal bearings	
5. Pivot bearings	
6. Collar bearings	

Table 17.2: Materials and Applications for Journal bearing.

Materials	Characteristics	Limitations	Applications
Wood	Self lubricating, low cost, long life.	Light loads at high speeds, under 65°C	Conveyors
Cast Iron	Low friction, low cost	Not over 3.5 N/mm^2 and 40 m/min.	Cam shafts
Steel	Low cost	Light loads, 45 m/min	Guides
Bronze bushing	Low cost, simple construction	Loads upto 21 N/mm^2. Speeds upto 270 m/min.	All equipments.
Heavy babbit liner on steel or cast iron	Long life, low friction, must have good lubrication.	Steady loads under 7 N/mm^2 to 14 N/mm^2	Motor, turbine shafting.
Light liner on steel or bronze backing	Heavy duty, general purpose, good for dynamic loads.	About 35 N/mm^2 and 1200 m/min	Gas and Diesel engines, compressors.
Rubber	Low friction, resists abrasion, shock absorbent, long life.	About 0.55 N/mm^2 needs water lubrication	Marine propellers, pumps, turbines
Carbon graphite	No lubrication needed, light duty applications.	Under 455°C, 4 N/mm^2 at low speeds.	Electric motors, meters, conveyors

Materials	Characteristics	Limitations	Applications
Moulded plastic laminate	Low friction, stronger than babbit when water lubricated	About 120 m/min and 17.5 N/mm^2 must be well cooled.	Pumps, propellers.
Moulded plastic	Low friction, clean.	Low loads if used at high speeds.	Dairy, textile and food machinery.
Sintered bearings, steel bronze	Self lubricating	Low loads, high speeds.	All inaccessible places, typewriter, electric fans, instruments, piston rings.

17.6. LUBRICANTS AND THEIR PROPERTIES :

We have already known that, the bearings are the elements used for controlled and smooth running of the journal which requires nil friction to be present in between the relatively moving surfaces. The most effective way of reducing friction and wear is to prevent the contact surfaces. This is achieved by introducing a lubricant in between the contact surfaces which separates the contact surfaces by its film thickness and hence energy loss due to friction may be reduced and at the same time the heat generated during operation may also be transmitted away and thus the operating temperature may be properly maintained which improves the life of the bearing.

Lubricants are generally divided into three groups with respect to their usage and availability, ***1) Liquid, 2) Semi-solid and 3) Solid***. The most commonly used liquid lubricants are ***mineral*** and ***synthetic oils***. These oils are designated by a concern known as ***"Society of Automotive Engineers"*** as SAE 10, SAE 20, SAE 30 ..... and so on. When the grade number increases, its viscosity also increases. That is, higher viscous liquids are referred by higher grade numbers. For example SAE 40 oil is thicker than SAE 30 oil and so on. The liquid lubricants are mainly used in automobiles, aircrafts, transmission systems, turbines, refrigerating machines and all machine tools like lathe, milling, drilling machine etc.

Grease is the notable semi-solid lubricant. Sometimes the liquid lubricant may evaporate at high temperatures. For such applications, grease is used. A good example for the usage of grease is the bearings of fan, wet grinder and many parts of automobiles.

Solid lubricants are employed to reduce wear in those situations where a conventional lubricant (oil or grease) cannot be used like in high vacuum space-craft or at very high temperatures. They can be applied to bearing surfaces by means of an adhesive such as resin or they may be introduced in a powder form between the two rubbing metals at low loads, in which case they adhere to the metal surface, eventually forming a fairly coherent covering. Some of the solid lubricants are ***graphite, molybdenum disulphide, lead monoxide*** etc.

The properties of lubricants are improved by adding suitable chemicals called additives containing sulphur, chlorine and phospherous.

The following are the some of the important properties of the lubricants.

(i) Viscosity : It describes the internal friction of fluids, the property to resist the shear (or movement) of one layer of fluid with respect to the other. In other words, viscosity may be defined as the measure of fluidity of a liquid. i.e., viscosity is the measure of fluid's resistance to shear. This viscosity can be expressed in two ways such as ***absolute or dynamic viscosity and kinematic viscosity.***

The viscosity varies inversely with temperature and directly with pressure, both in a non linear fashion. According to Newton's law of viscous flow, for a specific temperature and pressure, the force required to move a layer of viscous fluid from its adjacent layer is directly proportional to area of contact, velocity of layer and inversely proportional to the distance between these adjacent layers.

i.e., $F \propto \frac{Av}{h}$

(or) $F = Z.A.\frac{v}{h}$

where

F = Force resisting the mutual shear of two adjacent layers of fluid (or) force required to move one layer from its adjacent layer in a laminar flow as shown in figure 17.3

A = Area of contact between the layers.

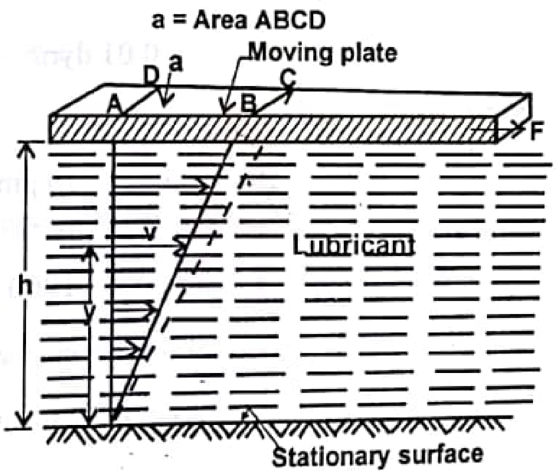


Fig. 17.3

h = Distance between the layers.

v = Velocity of relative shear of the layers ie; relative velocity of adjacent layers.

Z = Proportionality factor (Dynamic factor of viscosity)

If we take $F = 1$ N, $A = 1\ m^2$; $h = 1$ m, $v = 1$ m/s, the dynamic viscosity according to SI unit system is obtained as

$$Z = \frac{F\,h}{A\,v} = \frac{1\,N \times 1\,m}{1\,m^2 \times 1\,m/s} = 1\,N - s/m^2$$

That is, one unit of dynamic viscosity is the force of 1 N required to move a layer of fluid 1 m^2 in area at a velocity rate of 1 m/s relative to an immobile layer of the same size located at a distance of 1 m from the first layer.

Sometimes the absolute viscosity is represented in a different way as

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

In MKS units, the dynamic (or absolute) viscosity is denoted as $Z = 1\ kgf - sec/m^2$. Generally the viscosity may be expressed in CGS units as poise, named after the French physician Poisewille.

ie, $1\ poise = 1\ dyne - sec/cm^2$

$$= 1\ gm\ \frac{cm}{sec^2} \times \frac{sec}{cm^2} = 1\ gm/cm - sec$$

In practice, a further smaller unit, the centipoise is frequently used.

ie, $1\ centipoise\ (cp) = \frac{1}{100}\ poise$

$$= 0.01\ dyne - sec/cm^2$$

Note :

$$\frac{1\ kg}{m - s} = \frac{1000\ gm}{100\ cm - sec} = \frac{10\ gm}{cm - sec} = 10\ poise = 1000\ cp$$

ie, 1 kg/m – s = 1000 centipoises

The ratio of dynamic viscosity (Z) of a fluid to its density (ρ) at the same temperature is referred as kinematic viscosity (γ) ie $\gamma = \frac{Z}{\rho}$

The unit of kinematic viscosity in CGS system is cm^2/sec and is called as stokes.

ie, $1\ stoke = \frac{1\ gm}{cm - sec} \times \frac{cm^3}{gm} = 1\ cm^2/sec$

(a) Viscosity Index :

Usually the viscosity of oil decreases with increase of temperature. This variation of viscosity with temperature is defined as viscosity index.

(b) Viscosity measuring device :

The Saybolt universal viscometer is perhaps the most widely used viscosity measuring instrument. It utilises the equation for flow through a capillary tube to measure the kinematic viscosity of the lubricating oil. The instrument consists of a standard 60 cm^3 receiving flask and a reservoir surrounded by a constant temperature bath. The oil sample to be tested is placed in the reservoir, and, when the bath and reservoir reach the desired test temperature, a stopper at the bottom is removed. The time required for 60 cm^3 of the oil to flow through a capillary tube is recorded. This time, measured in seconds, has become a standard unit of viscosity known as Saybolt Universal Seconds or simply SUS.

An empirical formula may be adopted to convert SUS to absolute viscosity and is given by

$$Z = \rho_t \left(0.22\, S - \frac{180}{S} \right)$$

where Z = Absolute viscosity in centipoise at test temperature (t°C)

S = Saybolt Universal Seconds

ρ_t = Specific gravity of oil at test temperature (t°C) expressed in gm/cm^3

The specific gravity (ie, density) of the lubricating oil at test temperature (t°C) is obtained as

$$\rho_t = \rho_{15.5} - 0.6 \times 10^{-6}\, t$$

where

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

= 0.86 to 0.95 gm/cm^3 (or) 0.9 gm/cm^3 (approx)

(or)

In S.I. unit system,

$$Z = \rho_t \left(0.00022\, S - \frac{0.18}{S} \right)$$

Here

Z = Viscosity in kg/m – s at t°C

S = Saybolt universal Seconds

ρ_t = Density in kg/m^3 at t°C

and $\rho_t = \rho_{15.5} - 0.6 \times 10^{-3}\, t$

where

$\rho_{15.5}$ = Density of oil at 15.5°C in kg/m^3

= 860 to 950 kg/m^3 (Average value of 900 kg/m^3 may be taken)

The specific gravity of different types of oil are shown in table 17.3.

Table 17.3: Specific Gravity at 15°C

S.No.	Type of oil	Specific gravity at 15°C
1.	Light oil for light service high speeds	0.875
2.	Turbine oil (oil rings) for light service and high speeds	0.890
3.	Extra-light motor oil for ring-oiled bearings, transmission shafting, small generators, motors, and high-speed engines.	0.935
4.	S.A.E. 20 – light transmission oil for gears	0.925
5.	S.A.E. 40-medium transmission oil for large generators, motors, steam turbines, high-speed gears, heavy motor oil	0.930
6.	Airplane 100 G-light cylinder oil	0.890
7.	S.A.E. 110-light steam cylinder oil; heavy duty gears	0.930
8.	Medium cylinder oil; slow speed worm gears	0.910
9.	S.A.E. 160-heavy cylinder oil, heavy-duty slow-speed gearing	0.935
10.	Heavy steam cylinder oil	0.930

(ii) Pour Point (or) Freezing Point

It is the temperature at which an oil will cease to flow when cooled.

(iii) 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 6mm at the surface of oil.

(iv) Fire Point :

It is the temperature at which an oil gives off sufficient vapour to burn it continuously when ignited.

17.7. TERMINOLOGY OF HYDRO-DYNAMIC JOURNAL BEARING :

For a hydrodynamic journal bearing, the relative positions of journal and bearing with respect to various operating conditions are shown in figure 17.4.

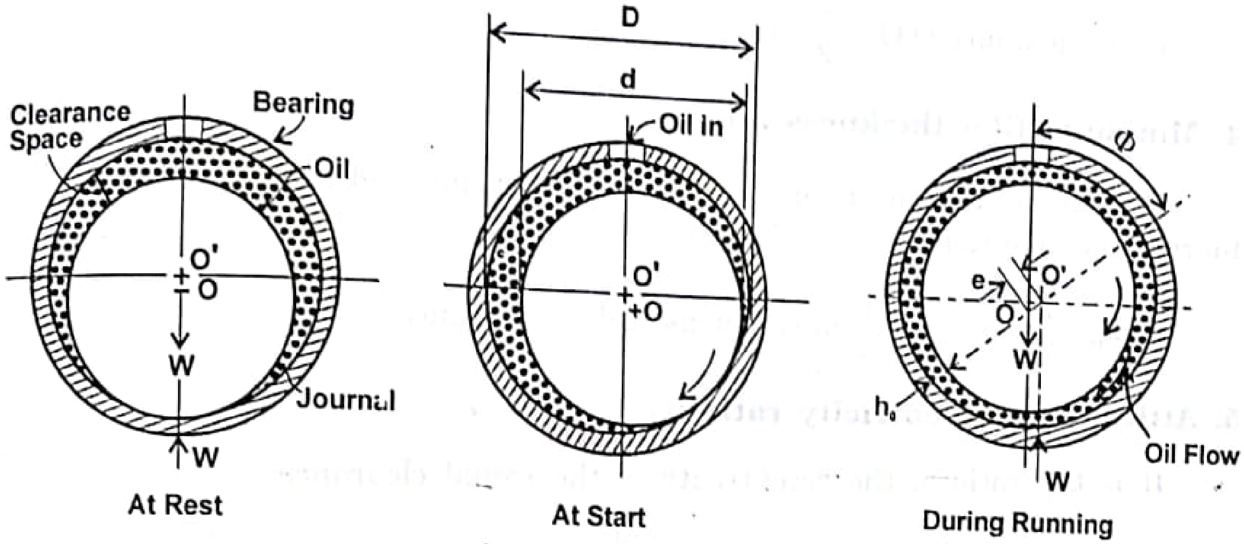


Fig. 17.4: Mechanism of hydro-dynamic lubrication

From the position of shaft running at high speed, some specific terms can be defined as follows.

Let O′ be the centre of bearing

O be the centre of journal

D be the diameter of bearing

d be the diameter of journal

l be the length of journal

1. Diametral clearance (c) :

It is the difference between the diameters of bearing and journal

$$\text{ie, } c = D - d$$

Generally c may be assumed as $\frac{1}{1000}$ times of journal diameter

$$(\text{ie, } c = 0.001\, d)$$

2. Diametral clearance ratio :

It is the ratio of diametral clearance to the diameter of the journal.

$$\text{ie, } c_r = \frac{c}{d}\left(= \frac{0.001\, d}{d} = 0.001 \text{ generally}\right)$$

3. Eccentricity (e) :

It is the radial distance between the centre of bearing (O′) and the centre of journal (O)

$$\text{ie, } e = \text{distance } O\,O' = \frac{c}{2} - h_0$$

4. Minimum film thickness : (h_o)

It is the minimum distance between the bearing and the journal under complete lubrication conditions.

$$\text{ie, } h_o = \frac{c}{2} - e \text{ where } \frac{c}{2} \text{ is called as radial clearance.}$$

5. Attitude or eccentricity ratio (ε) :

It is the ratio of the eccentricity to the radial clearance,

$$\text{ie, } \varepsilon = \frac{e}{(c/2)} = \frac{2e}{c} = \frac{2}{c}\left(\frac{c}{2} - h_o\right) = 1 - \frac{2h_o}{c}$$

6. Film thickness at any angle θ, $h = \frac{c}{2}(\cos\theta + 1)$

7. Square bearing :

If the length of journal (l) is equal to its diameter (d), it is called as square bearing.

$$\text{ie, } l = d \text{ (or) } \frac{l}{d} = 1 \text{ for square bearing}$$

8. **Long bearing** : Here, length > diameter ie, $\frac{l}{d} > 1$

9. **Short bearing :** For this kind, length < diameter ie, $\frac{l}{d} < 1$

17.8. DESIGN OF HYDRO-DYNAMIC JOURNAL BEARING :

The function of the hydro-dynamic journal bearing depends on the pressure developed in the lubricant supplied between the bearing and journal. If the viscosity of the lubricant is very thin, then metal to metal contact will occur or sticking will be the result if it is more thick. Thus for good design, the selection of lubricant should be proper and the heat generated during operation must be transmitted away in order to keep the bearing in the optimum working temperature. Otherwise the bearing will not be saved from abnormal heat produced which results even sometimes melting of bearing elements.

The main criterian for the design of journal bearing is the power loss due to heat generated which in turn depends on bearing friction. It has been found by experiments that, the coefficient of friction for journal bearing is a function of three variables,

(1) $\frac{Zn}{p}$, (2) $\frac{d}{c}$ and (3) $\frac{l}{d}$

ie, $\mu = f\left(\frac{Zn}{p}, \frac{d}{c}, \frac{l}{d}\right)$

where μ = Coefficient of friction between the journal and bearing.

Z = Viscosity of lubricant

n = Speed of the journal

p = Pressure developed on the bearing over projected area

= $\frac{W}{l.d}$ (where W = Load applied on the bearing)

l, d = Length and diameter of journal respectively.

c = Diametral clearance.

The variable Zn/p is termed as ***'Bearing characteristic number'*** and is a dimensionless number. The variation of coefficient of friction with the operating values of bearing characteristic number (Zn/p) as found out by McKee brothers in an actual test of friction is shown in fig. 17.5.

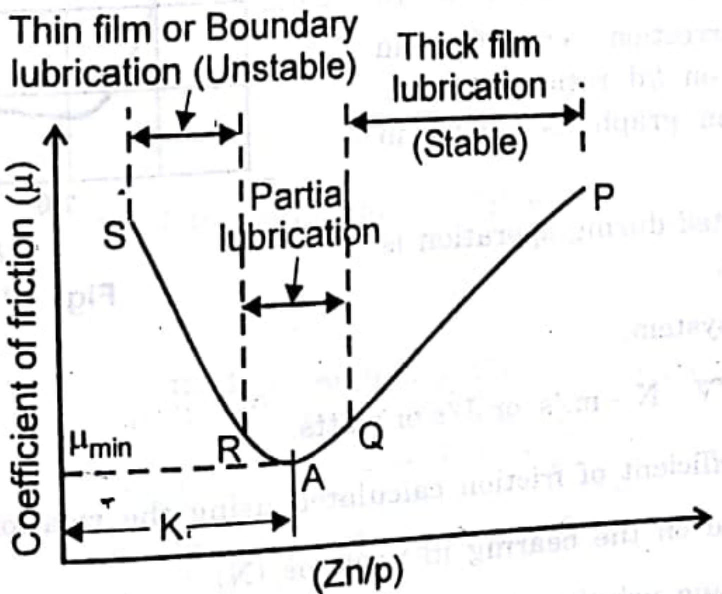


Fig. 17.5: Variation of coefficient of friction with Zn/p.

In this figure the part of the curve PQ, which is almost straight line, represents the region of thick film lubrication where stable operating conditions may be obtained.

Here μ varies directly with Zn/p. The curved line QR represents the region of partial lubrication and hence partial metal to metal contact may occur in this region. The portion RS represents the region of thin film or boundary lubrication. Here the lubrication will not be adequate. The coefficient of friction μ is minimum at the point A in the curve. The value of bearing characteristic number at the lowest friction (i.e., at the point A) is specified as ***'Bearing modulus'*** and is denoted as K.

The bearing should not be operated at this value of bearing modulus, because a slight decrease in speed or slight increase in pressure will break the oil film and make the journal to operate with metal to metal contact, resulting high friction, wear and heat. Hence the bearing should be designed for a value of atleast three times of bearing modulus (K) for safety. If the bearing is to be designed for large fluctuations of load with heavy impact, then the value of Zn/p = 15 K may be adopted.

By doing a number of tests, McKee brothers found a relation between the friction coefficient and other parameters for a hydro-dynamic (ie, thick film lubricated as shown by the line PQ) journal bearing, which is given by

$$\mu = \left[\frac{33.25}{10^8} \times \frac{Zn}{p} \times \frac{d}{c}\right] + k \qquad \text{....... (A)}$$

when Z is in kg/m-s and p is in N/mm^2

$$\text{(or) } \mu = \left[\frac{33.25}{10^{10}} \times \frac{Zn}{p} \times \frac{d}{c}\right] + k \qquad \text{...... (B)}$$

when Z is in centipoises and p is in kgf/cm^2 and k = correction factor for end leakage. It depends on l/d ratio. Its value can be obtained from graph as shown in figure 17.6

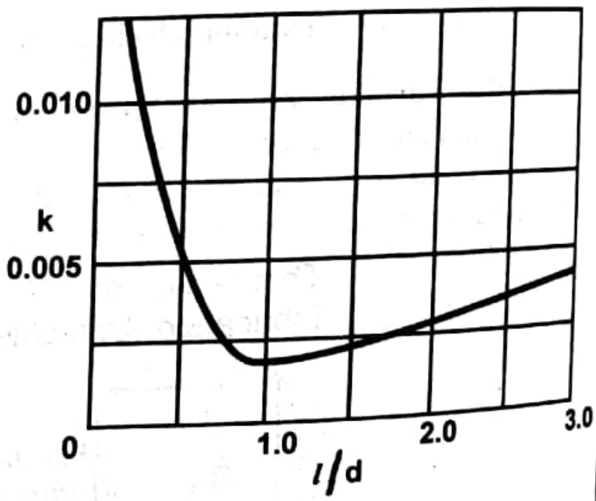


Fig. 17.6

The heat generated during operation is given by the relation,

(i) According to S.I. system,

$$H_g = \mu WV \quad N-m/s \text{ or } J/s \text{ or watts.}$$

where μ = Coefficient of friction calculated using the relation (A)

W = Load on the bearing in newtons (N)

V = Sliding velocity of journal in m/s

$= \frac{\pi d n}{60}$ where d is in metres and n is in rpm.

(ii) According to M.K.S. system,

$$H_g = \frac{\mu WV}{J} \text{ k cal/min ; (or) } \mu WV \quad \text{kgf} - \text{m/min}$$

where W = Load in kgf

V = Velocity in m/min

J = Joule's coefficient $= 427$ kgf – m/k cal

μ = Coefficient of friction obtained from the relation (B)

Heat dissipated by the bearing is given by

$H_d = CA\,(t_b - t_a)$ watts (according to S.I. system)

$= CA\,(t_b - t_a)$ kcal/min (according to M.K.S. system)

where C = Heat dissipation coefficient expressed in

$W/m^2/°C$ (or $kcal/min/cm^2/°C$)

$= 140$ to $420\ W/m^2/°C$ (or 2×10^{-4} to 6×10^{-4} $kcal/min/cm^2/°C$)

for unventilated bearings (Still air)

$= 490$ to $1400\ W/m^2/°C$ (or 7×10^{-4} to 20×10^{-4} $kcal/min/cm^2/°C$)

for well ventilated bearings.

A = Projected area of the bearing in $m^2 = (l \times d)$

t_b = Temperature of the bearing surface in °C

and t_a = Temperature of the surrounding air in °C

[Note :

$$1\ kcal/min/cm^2/°C = 427\ kgf - m/min/cm^2/°C$$

$$= 427 \times 9.81\ N - m/min/cm^2/°C$$

$$= \frac{4200}{60} \times 10^4\ N - m/s/m^2/°C$$

$$= 7 \times 10^5\ W/m^2/°C]$$

For safe and optimum design, the value of heat dissipated should be more than heat generated.

The mass of the oil required to remove the heat generated can be determined by equating the heat generated to the heat taken away by the oil.

ie, $H_g = H_t = m\, c_p\, \Delta t$ J/s or watts from which $m = \dfrac{H_g}{c_p\, \Delta t}$

where m = Mass of the oil in kg/s

c_p = Specific heat of the oil and its value may be taken as 1840 to 2100 J/kg/° C

Δt = Difference between the outlet and the inlet temperatures of the oil in °C.

In M.K.S. units;

$$H_t = m\, c_p\, \Delta t \text{ kcal/min}$$

where m is in kg/min; c_p is in kcal/kg/°C and its value is taken from 0.44 to 0.49 kcal/kg/°C.

17.9. SOMMERFELD NUMBER : (S)

Sometimes, one of the important dimensionless parameters called Sommerfeld number will help us to design for the hydro-dynamic journal bearing, and is given by

$$S = \frac{Z\, n'}{p}\left(\frac{d}{c}\right)^2$$

where S = Sommerfeld number.

Z = Absolute viscosity in $N - s/m^2$

p = Bearing pressure in N/m^2

n′ = Revolutions per second

(or) **in MKS units**

Z = Absolute viscosity in $kgf\text{–}sec/m^2$

p = Bearing pressure in kgf/m^2

n′ = Revolutions per second.

Sommerfeld values may be selected from the table 17.4. From the selected value of S, Z can be calculated and then coefficient of friction may be determined.

17.10. PETROFF'S EQUATION :

Similar to McKee brothers, Mr. Petroff also derived an equation for the coefficient of friction relating the parameters of journal. The derivation of Petroff's equation can be explained as follows.

Considering a journal of diameter d and length l rotates at n′ revolutions per second inside a bearing as shown in figure 17.7.

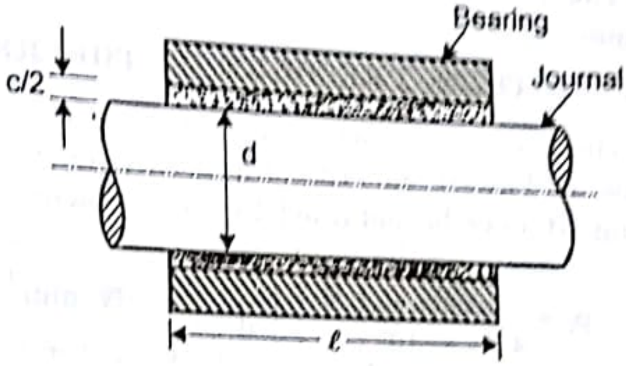


Fig. 17.7

According to Newton's law of viscous flow, the force required to rotate the journal is given by

$$F = \frac{Z\,A\,v}{h}$$

Here,

Z = Dynamic viscosity

A = Surface area of contact $= \pi\, d\, l$

v = Surface velocity of journal $= \pi\, d\, n'$

h = Radial clearance $= c/2$

Substituting, we get

$$F = Z \cdot \pi\, d\, l\, .\, \pi\, d\, n' \frac{2}{c}$$

The friction torque, T = F.r

$$\text{i.e., } T = Z\, .\, \pi\, d\, l \cdot \pi\, dn' \cdot \frac{2}{c} \cdot \frac{d}{2} = \pi^2 \frac{Zn'}{c} \cdot d^3\, l \quad .... \text{(A)}$$

Suppose W is the load acting on the journal, then the friction torque is given by

$$T = \mu\, W\, r = \mu\, p\, l\, d \cdot \frac{d}{2} = \mu\, p\, l\, \frac{d^2}{2} \quad .... \text{(B)}$$

Equating (A) and (B) we get

$$\left[\begin{array}{l} p = \text{Bearing pressure} \\ l\, d = \text{Projected area} \\ W = p \cdot l\, d \end{array}\right]$$

$$\mu\, p\, l\, \frac{d^2}{2} = \pi^2 \frac{Zn'}{c}\, d^3\, l$$

$$\mu = 2\,\pi^2 \left(\frac{Zn'}{p}\right)\left(\frac{d}{c}\right) \quad .... \text{(C)}$$

The equation (C) is the Petroff's equation or Petroff's law.

The Petroff's equation and McKee's equations are employed for lightly loaded bearings.

17.11. CRITICAL PRESSURE OF THE JOURNAL BEARING :

The pressure at which the oil film breaks down so that metal to metal contact begins, is known as critical pressure or the minimum operating pressure of the bearing. It may be obtained by the following empirical relation,

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

(or) $$p_c = \frac{Zn}{475 \times 10^6}\left(\frac{d}{c}\right)^2\left(\frac{l}{d+l}\right) \ \text{kgf/cm}^2 \quad \text{.. (when Z is in centipoise)}$$

Note:

For easy reference of design data from some data handbooks, page numbers of these databooks where the above said data are available have also been presented in this text book. For example, at any place of this book, if the matters like table 17.5 (PSG 7.31, JDB 19.13) are seen, then they refer that the data available in table 17.5 of this text book are also available in the page 7.31 of PSG data book and in the page 19.13 of Jalal databook.

17.12 DESIGN STEPS FOR HYDRODYNAMIC JOURNAL BEARING :

When designing a hydro-dynamic journal bearing, we should have clear idea about what parameters to be determined Some of such parameters are length of journal, diameter of the journal, diametral clearance, material and also proper lubricant. The following steps can be generally adopted in designing journal bearing.

1. Calculate the diameter of journal (d) from power, torque and stress relationships such as

$$T = \frac{60\,P}{2\,\pi\,N} \quad \& \quad d = \left[\frac{16\,T}{\pi\,S_s}\right]^{1/3}$$

The recommended speeds of some machineries are given in table 17.6.

2. Find the length of journal, choosing an appropriate (l/d) ratio from the table 17.5, (PSG 7.31, JDB 19.13)

3. Now evaluate the bearing pressure p using the relation $p = \frac{W}{l.d}$ (where W = applied load) and check this pressure with allowable pressure, given in the same table. If the pressure will not be within the range, then change $\frac{l}{d}$ ratio suitably.

4. Consider the operating temperature properly. Commonly it may be assumed from 60° to 90°C.
5. Choose the diametral clearance (c) from the table 17.5
6. Select the value of 'Bearing characteristic number' $\frac{Zn}{p}$ from the same table and from that parameter determine Z and find corresponding lubricant, from the graphs shown in figure 17.8, (PSG 7.41, JDB 19.26).
7. Calculate the coefficient of friction (μ) using appropriate relation
8. Then determine the heat generated (H_g) and Heat dissipated (H_d)

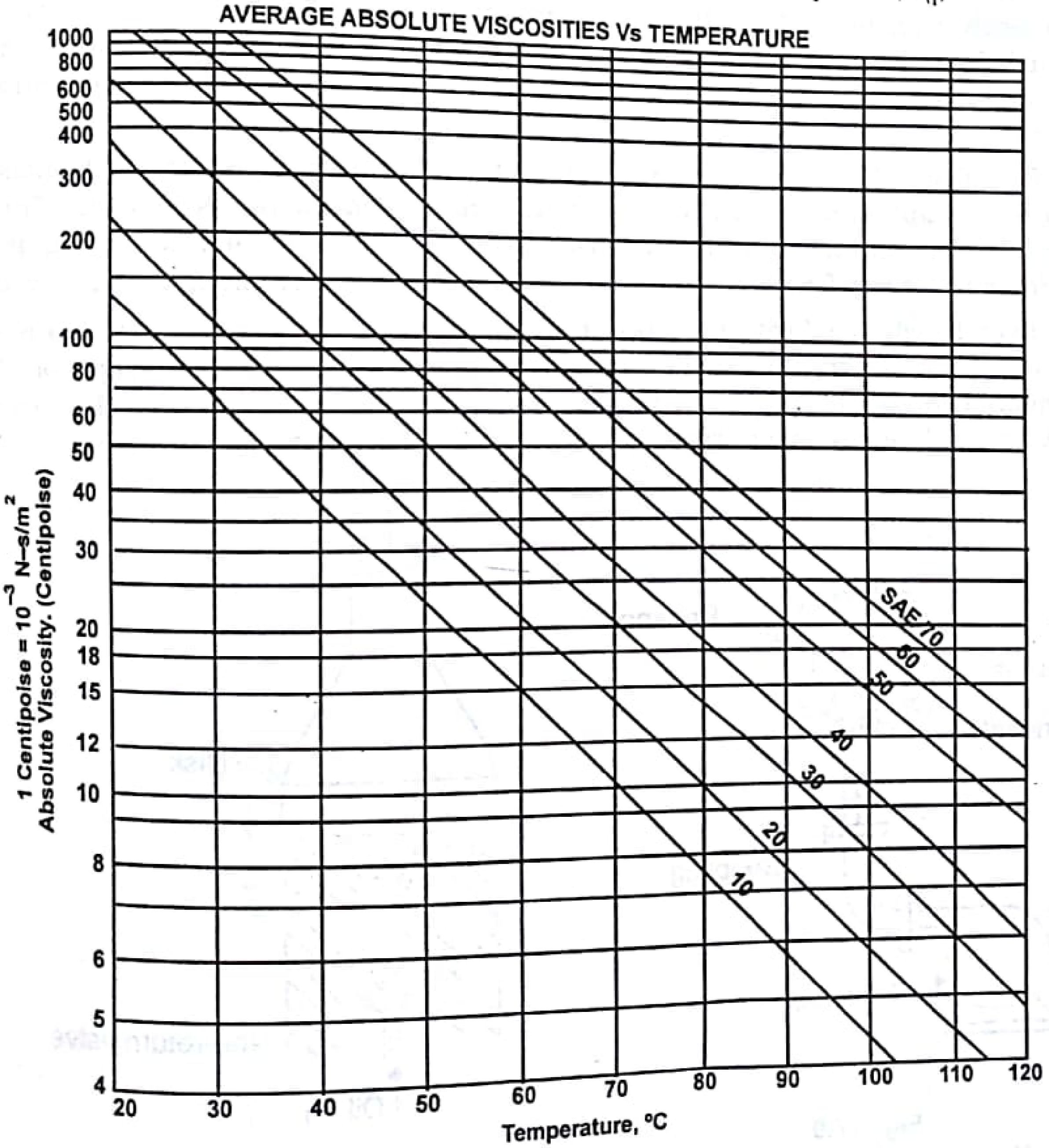


Fig. 17.8

9. If the generated heat is more than the dissipated heat, provide artificial cooling arrangements

10. Decide proper bearing material and other required dimensions for the journal bearing.

Note:

Sometimes the Sommerfeld number (S) may also by used to find out the value of Z and from that, μ can be calculated.

17.13. HYDROSTATIC BEARINGS :

The hydrodynamic lubrication is possible only when the shaft rotates at speeds sufficiently high to establish the thick film, otherwise boundary lubrication will be created. In hydrostatic bearings, the oil (i.e., lubricant) is supplied, during starting or slow running conditions, at such a point and at such a pressure that the journal is lifted up by the oil film that is formed (Fig. 17.9).

The figure 17.10 shows the hydrostatic thrust bearing in which the lubricant from a constant displacement pump is forced into a central recess and then flows outwards between the bearing surfaces, developing pressure and separation and returning to a sump for recirculation. By this way, the load carrying force is produced.

Eventhough the hydrostatic bearings are superior to hydrodynamic bearings because of not occurring the boundary lubrication due to the external supply of oil, the main drawback of hydrostatic bearings is that they always depend on the proper function of oil supply pump which is not required for hydrodynamic bearings.

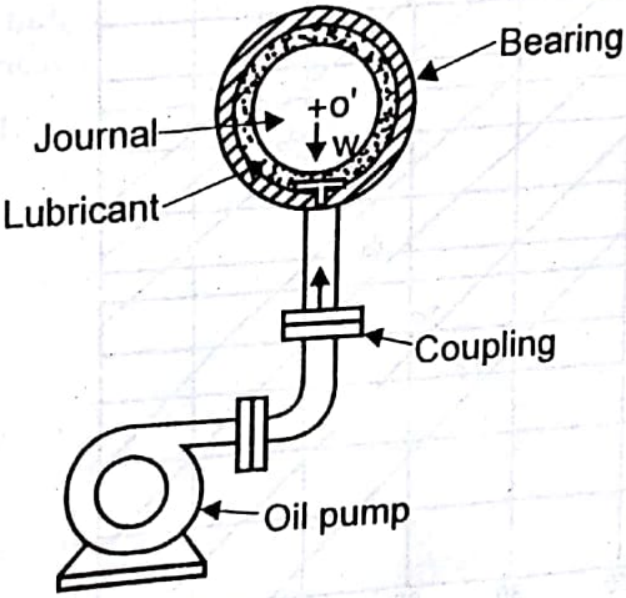


Fig. 17.9
Hydrostatic journal bearing

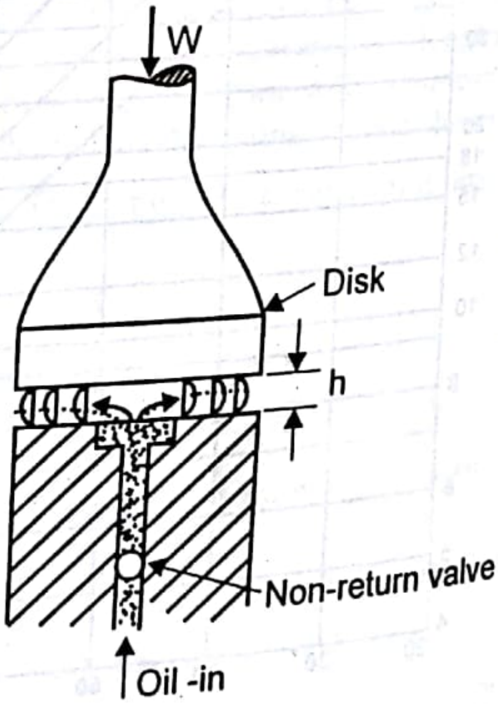


Fig. 17.10
Hydrostatic thrust bearing

Table 17.4(a): Dimensionless performance parameters for full journal bearings with side flow

$\frac{l}{d}$	ε	$\frac{2h_0}{c}$	S	Φ	$\mu\frac{d}{c}$	$\frac{4q}{dcn\,l}$	$\frac{q_s}{q}$	$\frac{\rho c' \Delta t_o}{p}$	$\frac{P}{P_{max}}$
∞	0	1.0	∞	70.92	∞	π	0	∞	–
	0.1	0.9	0.240	69.10	4.80	3.03	0	19.9	0.826
	0.2	0.8	0.213	67.26	2.57	2.83	0	11.4	0.814
	0.4	0.8	0.213	67.94	1.52	2.26	0	8.47	0.764
	0.6	0.4	0.0389	54.31	1.20	1.56	0	9.73	0.667
	0.8	0.2	0.021	42.22	0.961	0.760	0	15.9	0.495
	0.9	0.1	0.0115	31.62	0.756	0.411	0	23.1	0.358
	0.97	0.03	–	–	–	–	0	–	–
	1.0	0	0	0	0	0	0	∞	0
1	0	1.0	∞	(85)	∞	π	0	∞	–
	0.1	0.9	1.33	79.5	25.4	3.37	0.150	106	0.540
	0.2	0.8	0.631	74.02	12.8	3.59	0.280	52.1	0.529
	0.4	0.6	0.264	63.10	5.79	3.99	0.497	24.3	0.484
	0.6	0.4	0.121	50.58	3.22	4.33	0.680	14.2	0.415
	0.8	0.2	0.0446	36.24	1.70	4.62	0.842	8.00	0.313
	0.9	0.1	0.0188	26.45	1.05	4.74	0.919	5.16	0.247
	0.97	0.03	0.00474	15.47	0.514	4.82	0.973	2.61	0.247
	1.0	0	0	0	0	–	1.0	0	0
0.5	0	1.0	∞	(88.5)	π	0	∞	∞	–
	0.1	0.9	4.31	81.62	85.6	3.43	0.173	343.0	0.523

$\frac{l}{d}$	ε	$\frac{2h_0}{c}$	S	ϕ	$\mu\frac{d}{c}$	$\frac{4q}{dcn'l}$	$\frac{q_s}{q}$	$\frac{\rho c' \Delta t_o}{p}$	$\frac{P}{P_{max}}$
	0.2	0.8	2.03	74.94	40.9	3.72	0.318	164.0	0.506
	0.4	0.6	0.779	61.45	17.0	4.29	0.552	68.6	0.441
	0.6	0.4	0.319	48.14	8.10	4.85	0.730	33.0	0.365
	0.8	0.2	0.0923	33.31	3.26	5.41	0.874	13.4	0.267
	0.9	0.1	0.0313	23.66	1.60	5.69	0.939	6.66	0.206
	0.97	0.03	0.00609	13.75	0.61	5.88	0.980	2.56	0.126
	1.0	0	0	0	0	–	1.0	0	0
0.25	0.0	1.0	∞	(89.5)	π	0	0	∞	–
	0.1	0.9	16.2	82.31	322.0	3.45	0.180	1287.0	0.515
	0.2	0.8	7.57	75.18	153.0	3.76	0.330	611.0	0.489
	0.4	0.6	2.83	60.86	61.1	4.37	0.567	245.0	0.415
	0.6	0.4	1.07	46.72	269.7	4.99	0.746	107.0	0.334
	0.8	0.2	0.261	31.04	8.80	5.60	0.884	35.4	0.240
	0.9	0.1	0.0736	21.85	3.50	5.91	0.945	14.1	0.180
	0.97	0.03	0.0101	12.22	0.922	6.12	0.984	3.73	0.108
	1.0	0	0	0	0	–	1.0	0	0

q, mm^3/sec; ρ, density of the oil $= 900\ kg/m^3$ c′ Specific heat of the oil $= 1840$ to $2100\ J/kg°C$ $\rho c' = 142 \times 10^4\ N/m^2°C$

Values of $\frac{2h_o}{c}$

l/d ratio	∞	1	0.5	0.25
For min. friction	0.6	0.3	0.12	0.03
For max. load	0.66	0.53	0.43	0.27

Table 17.4(b): Dimensionless performance parameters for 120° bearing, centrally loaded, with side flow

$\frac{l}{d}$	ε	$\frac{2h_o}{c}$	S	ϕ	$\mu\frac{d}{c}$	$\frac{4q}{dcn'L}$	$\frac{q_s}{q}$	$\frac{\rho c' \Delta t_o}{P}$	$\frac{P}{P_{max}}$
∞	0	1.0	∞	90.0	∞	π	0	∞	–
	0.1	0.9007	0.877	66.69	6.02	3.02	0	25.1	0.610
	0.2	0.8	0.431	52.60	3.26	2.75	0	14.9	0.599
	0.4	0.6	0.181	39.02	1.78	2.13	0	10.5	0.566
	0.6	0.4	0.0845	32.67	1.21	1.47	0	10.3	0.509
	0.8	0.2	0.0328	26.80	0.853	0.759	0	14.1	0.405
	0.9	0.1	0.0147	21.51	0.653	0.388	0	21.2	0.311
	0.97	0.03	0.00406	13.86	0.399	0.118	0	42.4	0.199
	1.0	0	0	0	0	0	0	∞	0
1	0	1.0	∞	90.0	∞	π	0	∞	–
	0.1	0.9024	2.14	72.43	14.5	3.20	0.0876	59.5	0.421
	0.2	0.8	1.01	58.25	7.44	3.11	0.157	32.6	0.420
	0.4	0.6	0.385	43.98	3.60	2.75	0.272	19.0	0.396
	0.6	0.4	0.162	35.65	2.16	2.24	0.384	15.0	0.356
	0.8	0.2	0.0531	27.42	1.27	1.57	0.535	13.9	0.290
	0.9	0.1	0.0208	21.29	0.855	1.11	0.657	14.4	0.233
	0.97	0.03	0.00498	13.49	0.461	0.694	0.812	14.0	0.162
	1.0	0	0	0	0	–	1.0	0	0
0.5	0	1.0	∞	90.0	∞	π	0	–	–
	0.1	0.9034	5.42	74.99	36.6	3.29	0.124	149.0	0.431

$\frac{l}{d}$	ε	$\frac{2h_o}{c}$	S	ϕ	$\mu\frac{d}{c}$	$\frac{4q}{dcn'L}$	$\frac{q_s}{q}$	$\frac{\rho c' \Delta t_o}{P}$	$\frac{F}{P_{max}}$
	0.2	0.8003	2.51	63.38	18.1	3.32	0.225	77.2	0.424
	0.4	0.6	0.914	48.07	8.20	3.15	0.385	40.5	0.389
	0.6	0.4	0.354	38.50	4.43	2.80	0.530	27.0	0.336
	0.8	0.2	0.0973	28.02	2.17	2.18	0.684	19.0	0.261
	0.9	0.1	0.0324	21.02	1.24	1.70	0.787	15.1	0.203
	0.97	0.03	0.00631	13.00	0.550	1.19	0.899	10.6	0.136
	1.0	0	0	0	0	–	1.0	0	0
0.25	0	1.0	∞	90.0	∞	π	0	∞	–
	0.1	0.9044	18.4	76.97	124.0	3.34	0.143	502.0	0.456
	0.2	0.8011	8.45	65.97	60.4	3.44	0.260	254.0	0.438
	0.4	0.6	3.04	51.23	26.6	3.42	0.442	125.0	0.389
	0.6	0.4	1.12	40.42	13.5	3.20	0.599	75.8	0.321
	0.8	0.2	0.268	28.38	5.65	2.67	0.753	42.7	0.237
	0.9	0.1	0.0743	20.55	2.63	2.21	0.846	25.9	0.178
	0.97	0.03	0.0105	12.11	0.832	1.69	0.931	11.6	0.112
	1.0	0	0	0	0	–	1.0	0	0

Values of $\frac{2h_o}{c}$,

l/d ratio	∞	1	0.5	0.25
for min friction	0.5	0.4	0.28	0.06
for max load	0.53	0.46	0.38	0.26

Table 17.5: Design values for Journal bearing

Machinery	Bearing	Maximum bearing pressure		Absolute viscosity (Z)		Operating values (Z n/p)		$\frac{c}{d}$	$\frac{l}{d}$ ratio
		N/mm^2	kgf/cm^2	kg/m – s	Centipoise	Z in kg/m-s p in N/mm^2	Z in centipoise p in kgf/cm^2		
Stationary High	Main	1.75	17.5	0.015	15	3.56	356	0.001	1.5-3.0
Speed Steam	Crank Pin	4.2	42.0	0.030	30	0.850	85		0.9-1.5
Engines	Wrist Pin	12.6	126.0	0.025	25	0.71	71		1.3-1.7
Gas and Oil	Main	4.9-8.4	49-84			2.85	285		0.6-2.0
Engines	Crank Pin	10.8-12.6	108-126	0.02-0.065	20-65	1.42	142	0.001	0.6-1.5
(Four Stroke)	Wrist Pin	12.5-15.4	125-154			0.71	71		1.5-2.0
Gas and Oil	Main	3.5-12.5	35-125			3.56	356		0.6-2.0
Engines	Crank Pin	7-10.5	70-105	0.02-0.065	20-65	1.71	171	0.001	0.6-1.5
(Two Stroke)	Wrist Pin	8.4-12.5	84-125			1.42	142		1.5-2.2
Aircraft and	Main	5.6-12	56-120	0.008		2.13	213		0.8-1.8
Automobile	Crank Pin	10.5-24.5	105-245	0.008	8	1.42	142	0.001	0.7-1.4
Engines	Wrist Pin	16-35	160-350	0.008		1.14	114		1.5-2.2
Reciprocating	Main	1.75	17.5			4.27	427		1.0-2.2
Compressors and	Crank Pin	4.2	42.0	0.03-0.08	30-80	2.85	285	0.001	0.9-1.7
Pumps	Wrist Pin	7	70.0			1.42	142		1.5-2.0
Centrifugal Pumps Motor and Generators	Rotor	0.7-1.4	7.14	0.025	25	28.45	2845	0.0013	1.0-2.0
Machine Tools	Main	2.1	21.0	0.04	40	0.140	14	0.001	1.0-4.0
Steam Turbines	Main	0.7-2	7-20	0.002-0.016	2-16	14.22	1422	0.001	1.0-2.0
Railway Cars	Axle	3.5	35	0.1	100	7.11	7.11	0.001	1.9
Marine Steam	Main	3.5	35	0.03	30	2.85	285		0.7-1.5
Engines	Crank Pin	4.2	42	0.04	40	2.13	213	0.001	0.7-1.2
	Wrist Pin	10.5	105	0.03	30	1.42	142		1.2-1.7
Transmissions	Light, Fixed	0.18	1.8	0.025	25	14.22	1422	0.001	2-3
Gyroscopes	Rotors	6	60	0.03	30	7.82	782	0.001	–
Shafting	Self Aligning	1.1	11	0.06	60	4.27	427	0.001	2.5-4
	Heavy	1.1	11	0.06	60	4.27	427		2-3
Cotton Mills	Spindle	0.007	0.07	0.002	2	14.24	1424	0.001	–
Punching and	Main	28	280	0.1	100	–	–		1-2
Shearing Machines	Crank Pin	56	560	0.1	100	–	–	0.001	1-2
Rolling Mills	Main	21	210	0.05	50	1.42	142	0.0015	1-1.5

Table 17.6: Recommended Speeds of Some Machineries, rev./min

Location of bearing	Speed, rev/min
Automobile crank shaft	900 to 14000
Aeronautic engine crank shaft	1800 to 2000
Stationary gas-engine main	250 to 800
-do- crank pin	250 to 800
-do- cross head	250 to 800
Diesel engine main	60 to 160
-do- crank pin	60 to 160
Marine steam engine main	180
-do- crank pin	180
Stationary slow speed main	40 to 80
-do- crank pin	40 to 80
-do- cross head	40 to 80
Cotton mill spindle	8000 to 12000
Stationary high speed main	360
-do- crank pin	360
-do- cross head	360
Locomotive drive wheel	250
-do- crank pin	250
-do- cross head	250
Marine steam turbine	2000
Stationary steam turbine	2000
De Laval 5 kW steam turbine	30000
Railway car axle	300
Generator and motor	150 to 500
Rolling mill main	60
Gyroscope	800 to 1500

17.14: SOLVED PROBLEMS :

Problem 17.1 :

Design a journal bearing for a steam turbine, whose shaft is supported on two bearings one at each side of the turbine, and is coupled with a generator for power production. The weight of turbine with shaft is measured as 40 kN and the shaft rotates at 1500 rpm. Diameter of the shaft is 100 mm.

(Madras University)

Solution: (Refer figure 17.11)

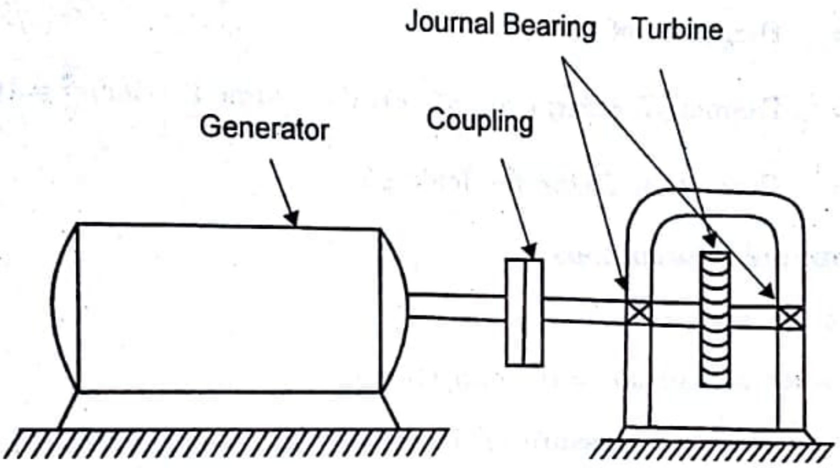


Fig. 17.11

Given:

Weight of turbine, acting on two bearings $= 40\ \text{kN} = 40000\ \text{N}$.

ie, Load acting on one bearing $W = 20000\ \text{N}$

Speed of the shaft (journal) $n = 1500\ \text{rpm}$

Diameter of the shaft (journal) $d = 100\ \text{mm}$

The journal bearing is designed based on two main data such as

1. Heat generation
2. Heat dissipation

Heat generated over the bearing surface during operation

$$= \mu\, W V\ \text{N-m/s (or) watts}$$

where μ = Coefficient of friction

W = Load acting on the bearing in newtons

V = Sliding velocity (or) Rubbing velocity in m/s

Now $\mu = \left(\frac{33.25}{10^8} \times \frac{Z\,n}{p} \times \frac{d}{c}\right) + k$

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

p = Pressure developed in the bearing over the projected area in N/mm^2

d = Diameter of the journal

c = Diametral clearance $= 0.001\,d$ (assumed) $\left[\text{(or)}\ \frac{d}{c} = 1000\right]$

k = Correction factor for leakage

To find out journal dimensions :

Let l = Length of journal

d = Diameter of journal $= 100$ mm (Given)

The developed bearing pressure on journal bearing is given by,

$$p = \frac{\text{Load}}{\text{Projected area}} = \frac{W}{l \cdot d}$$

Assuming $l/d = 1.2$ (For steam turbine from table 17.5) (PSGDB 7.31, JDB 19.14)

$$l = 1.2\,d = 1.2 \times 100 = 120 \text{ mm.}$$

Now, pressure developed, $p = \frac{20{,}000}{120 \times 100} = 1.67\ N/mm^2$

This pressure is within the safe range (0.7 to 2 N/mm^2) (Table 17.5) (PSG 7.31, JDB 19.14)

Let the operating temperature of bearing, $t_b = 75°C$

The atmospheric temperature, $t_a = 30°C$

To findout the viscosity of lubricant (Z):

Now take $\left(\frac{Z\,n}{p}\right)_{min}$ **= 14.22 (from table 17.5)**

Hence, $Z_{min} = 14.22 \times \frac{p}{n} = \frac{14.22 \times 1.67}{1500} = 0.016$ kg/m – s

Since the minimum viscosity required for the lubricant is 0.016 kg/m-s, the next standard lubricant ie, SAE 30 oil will be selected whose actual viscosity is 0.017 kg/m-s at 75°C. (Refer fig. 17.8) (PSG 7.41, JDB 19.26)

Now, Heat generated, $H_g = \mu WV$

$$\text{where } \mu = \left(\frac{33.25}{10^8} \times \frac{Zn}{p} \times \frac{d}{c}\right) + k$$

$$= \left[\frac{33.25}{10^8} \times \frac{0.017 \times 1500}{1.67} \times 1000\right] + 0.002 \qquad \left(k = 0.002 \text{ for } \frac{l}{d} = 1.2\right)$$

(From fig. 17.6) (PSG 7.34, JDB 19.24)

$$= (0.005 + 0.002) = 0.007$$

$$\text{Sliding velocity, } V = \frac{\pi dn}{60 \times 1000} = \frac{\pi \times 100 \times 1500}{60 \times 1000} = 7.85 \text{ m/sec}$$

Hence, Heat generated, $H_g = \mu WV$ watts

$$= 0.007 \times 20000 \times 7.85 = 1099 \text{ watts}$$

Heat dissipated, $H_d = CA(t_b - t_a)$ watts

where C = Heat dissipation coefficient

= 1000 W/m^2/°C (assumed)

A = Projected area in m^2 = $l \times d$

$$= \frac{100 \times 120}{1000 \times 1000} = (0.1 \times 0.12) \text{ m}^2$$

Now, Heat dissipated, $H_d = 1000 \times 0.1 \times 0.12 \times (75 - 30) = 540$ watts.

Since the generated heat is more than the dissipated heat, artificial cooling arrangements must be provided for proper functioning and also to limit the rise of temperature. This cooling arrangements can be done by providing cooling fans or by circulated water around the bearing.

$$\text{Now, Diameter of bearing, } D = d + c = d + \frac{d}{1000} = d(1 + 0.001)$$

$$= 100(1.001) = 100.1 \text{ mm}$$

Specifications for Hydro-dynamic full journal bearing :

1. Diameter of the journal (d) = 100 mm
2. Length of the journal l = 120 mm
3. Diameter of bearing (D) = 100.1 mm

4. Diameter clearance (c) = 100 microns (= 0.1 mm)
5. Lubricant selected = SAE 30 oil
6. Operating temperature = 75°C
7. Atmospheric temperature = 30°C
8. Material selected = Bronze bushing (or) Heavy babbit liner on steel or Cast iron.

Problem 17.2: (In M.K.S Units)

Design a journal bearing for a generator to the following specifications.

Load on the journal = 1200 kgf

Diameter of the journal = 75 mm

Speed of the journal = 1400 rpm **(Madras University)**

Solution :

Given :

Load, W = 1200 kgf

Diameter, d = 75 mm = 7.5 cm

Speed, n = 1400 rpm

To find journal dimensions:

For the generator, the $\frac{l}{d}$ ratio is 1.0 to 2.0 (Table 17.5) (PSG 7.31) (JDB 19.14)

Let us select $\frac{l}{d} = 1.5$

ie, Length of journal, $l = 1.5 \times d = 1.5 \times 7.5 = 11.25$ cm

Developed bearing pressure $p = \frac{W}{l \times d} = \frac{1200}{7.5 \times 11.25} = 14.22 \text{ kgf/cm}^2$

Since the safe pressure range is 7 to 14 kgf/cm^2. (Table 17.5),

let us increase the length as 130 mm, so that

$$l/d = \frac{130}{75} = 1.73$$

Now pressure developed over the projected area, (ie, bearing pressure)

$$p = \frac{1200}{13 \times 7.5} = 12.3 \text{ kgf/cm}^2$$

Now the bearing pressure is within the safe range.

To find the viscosity of oil:

To find out the value of Z, (the viscosity), from the table 17.5 it is known that, the minimum value of

$$\frac{Zn}{p} = 2845$$

$$\therefore Z_{min} = \frac{2845 \times 12.3}{1400} = 25 \text{ centipoise}$$

Minimum viscosity required = 25 cp.

Let us assume the operating temperature = 70°C

and atmospheric temperature, = 25°C

Hence the lubricant oil selected is SAE 40 oil, which is having the viscosity of 27 cp at 70°C. (Refer figure 17.8) (PSG 7.41, JDB 19.26)

$$\text{Heat generated, } H_g = \frac{\mu WV}{J} \text{ kcal/minute}$$

$$\text{where } \mu = \text{Coefficient of friction} = \left[\frac{33.25}{10^{10}} \times \frac{Z_n}{p} \times \frac{d}{c}\right] + k \text{ (PSG 7.34, JDB 19.3)}$$

Assume the diametral clearance, $c = 0.001$ d.

$$\text{Then, } \mu = \left(\frac{33.25}{10^{10}} \times \frac{27 \times 1400}{12.3} \times 1000\right) + 0.0025 = 0.013 \qquad (\because k = 0.0025 \text{ for } l/d = 1.73)$$

(from figure 17.6) (PSG 7.34, JDB 19.24)

$$\text{Sliding velocity} \quad V = \pi dn \quad \text{m/min}$$

$$= \pi \times \frac{7.5}{100} \times 1400 = 330 \text{ m/min}$$

$$J = 427 \text{ kgf} - \text{m/k cal}$$

$$\text{Hence, Heat generated,} \quad H_g = \frac{\mu WV}{J}$$

$$= \frac{0.013 \times 1200 \times 330}{427} = 12 \text{ k cal/min}$$

Heat dissipated, $H_d = CA\,(t_b - t_a)$ k cal/min

where C = Heat dissipation coefficient

= 15×10^{-4} kcal/min/cm^2/°C for well ventilated conditions (assume)

A = Projected area in cm^2

= $7.5 \times 13 = 97.5$ cm^2

$(t_b - t_a)$ = operating temperature - atmospheric temperature

= $70 - 25 = 45$°C

$\therefore$ H_d = $15 \times 10^{-4} \times 97.5 \times 45 = 6.58$ k cal/min

Since the dissipated heat in less than the generated heat, artificial cooling must be arranged by providing cooling fans and so on.

Diameter of bearing, $D = d + c = 75 + (0.001 \times 75)$

$= 75.075$ mm

Specifications :

1.	Type of bearing	=	Hydro dynamic full journal bearing.
2.	Length of journal	=	130 mm
3.	Diameter of journal	=	75 mm
4.	Diameter of bearing	=	75.075 mm
5.	Lubricant selected	=	SAE 40 oil
6.	Operating temperature	=	70°C
7.	Atmospheric temperature	=	25°C
8.	Material selected	=	Bronze bushing.

Problem : 17.3

The load on a journal bearing is 150 kN due to a turbine shaft of 250 mm diameter running at 1800 rpm. Determine the following

(i) Length of bearing if the allowable bearing pressure 1.6 MPa.

(ii) Amount of heat to be removed by the lubricant per minute if the bearing temperature is 60°C and the viscosity of the oil at 60°C is 20 centipoises and the bearing clearance is 0.25 mm.

(Madras University, April 2000)

Solution:

Given:

Load on the journal,	$W = 150\ kN = 150 \times 10^3\ N$
Diameter of journal,	$d = 250\ mm.$
Speed of journal,	$n = 1800\ rpm.$
Bearing pressure,	$p = 1.6\ MPa = 1.6 \times 10^6\ N/m^2 = 1.6\ N/mm^2$
Operating temperature	$= 60°C$
Viscosity of oil,	$Z = 20\ cp = 20 \times 10^{-3}\ N-s/m^2$
Bearing clearance,	$c = 0.25\ mm.$

(i) To find length of bearing:

Let l = Length of bearing.

The bearing pressure over the projected area is given by,

$$\boxed{p = \frac{W}{l \cdot d}}$$

$$\text{(or)}\ l = \frac{W}{p.d} = \frac{150 \times 10^3}{1.6 \times 250} = \mathbf{375\ mm\ (Answer)}$$

(ii) Amount of heat to be removed:

The heat generated in bearing due to turbine shaft rotation is given by

$H_g = \mu\ WV\ N-m/min.$ (or) J/min.

where,

μ = Coefficient of friction,

$$= \left(\frac{33.25}{10^8} \times \frac{Zn}{p} \times \frac{d}{c}\right) + k$$

$$= \left(\frac{33.25}{10^8} \times \frac{0.020 \times 1800}{1.6} \times \frac{250}{0.25}\right) + 0.002$$

$$= 0.0075 + 0.002 = 0.0095$$

[For $\frac{l}{d} = \frac{375}{250} = 1.5$

k is approximately equal to 0.002]

(From Fig. 17.6), (PSG 7.34, JDB 19.24)

V = Sliding velocity

$$= \frac{\pi\ dn}{1000}\ m/min = \frac{\pi \times 250 \times 1800}{1000} = 1414\ m/min.$$

Hence, $H_g = 0.0095 \times 150 \times 10^3 \times 1414 = 2015 \times 10^3\ N-m/min$

$= 2015\ kN-m/min$ (i.e., 2015 kJ/min.)

This generated heat should be removed by the lubricant in order to safeguard the turbine from over heating.

Hence, the amount of heat to be removed $= 2015$ **kJ/min (Answer)**

Problem : 17.4

The load on a 100 mm full hydro-dynamic journal bearing is 9000 N. Speed of the journal is 320 rpm.

Let $l/d = 1$, $c/d = 0.0011$. The operating temperature $= 65°C$ and minimum oil film thickness $= 0.022$ mm

(i) Select an oil that will closely accord with the stated conditions. For the selected oil, determine (ii) the friction loss (iii) the hydro dynamic oil flow through the bearing, (iv) the amount of leakage (v) the temperature rise of oil passes through the bearing and (vi) maximum oil pressure.

(M.K. University, November 1996)
(Anna University, Oct. 2003)

Solution :

Given :

Load on the bearing, $W = 9000$ N

Diameter of the journal, $d = 100$ mm

Speed of the journal, $n = 320$ rpm

$l/d = 1, \quad c/d = 0.0011$

Minimum oil film thickness, $h_0 = 0.022$ mm

Operating temperature $= 65°C$

Therefore, the length of journal, $l = d = 100$ mm

Diametral clearance, $c = 0.0011\, d$

$= 0.0011 \times 100 = 0.11$ mm

Sommerfeld number, $S = \dfrac{Z\, n'}{p} (d/c)^2$

where Z = Absolute viscosity in $N - s/m^2$ or $kg/m - s$

n' = Revolutions per second (rps)

p = Bearing pressure, in N/m^2

Now, pressure developed in the bearing, over the projected area, (i.e., bearing pressure)

$$P = \frac{W}{l \times d} = \frac{9000}{\frac{100}{1000} \times \frac{100}{1000}} = \frac{9000}{0.1 \times 0.1} = 9 \times 10^5 \text{ N/m}^2$$

$$n' = \frac{320}{60} = 5.33 \text{ rps}$$

$$\text{Now } \frac{2\,h_o}{c} = \frac{2 \times 0.022}{0.11} = \frac{0.044}{0.11} = 0.4$$

For $\frac{2h_o}{c} = 0.4$, Sommerfeld number, S = 0.121 (from the table 17.4 (a)) (PSG 7.36, JDB 19.16)

$$\text{Also } S = \frac{Z\,n'}{p}\left(\frac{d}{c}\right)^2$$

i.e.,
$$0.121 = \frac{Z \times 5.33}{9 \times 10^5}\left(\frac{1}{0.0011}\right)^2$$

$$Z = \frac{0.121 \times 9 \times 10^5 \times (0.0011)^2}{5.33} = 0.0247 \text{ N} - \text{s/m}^2 = 0.025 \text{ kg/m} - \text{s}$$

(i) The next standard oil, **SAE 30 oil** will closely accord with the stated conditions, whose viscosity is 0.0255 kg/m – s (≈ 0.025 kg/m – s)

Hence S = 0.121

Then, $\mu \frac{d}{c} = 3.22$ (for S = 0.121) [Table 17.4(a) PSG 7.36, JDB 19.16]

$$\therefore \mu = 3.22 \times \frac{c}{d} = 3.22 \times 0.0011 = 0.0035$$

i.e., Coefficient of friction, $\mu = 0.0035$

Frictional force : $\mu W = 0.0035 \times 9000 = 31.5$ N

Velocity of the journal; $v = \frac{\pi\, d\, n}{60 \times 1000} = \frac{\pi \times 100 \times 320}{60 \times 1000} = 1.68$ m/s

(ii) ∴ Power loss due to friction, $P_f = \mu\, W \cdot v = 31.5 \times 1.68$

$$= 52.92 \text{ N} - \text{m/s} = 53 \text{ watts}$$

(iii) Total oil flow :

From the Table 17.4(a) $\frac{4\,q}{d\,c\,n'\,l} = 4.33$ (for S = 0.121)

∴ Total oil flow through the bearing,

Scanned by CamScanner

$$q = \frac{4.33 \times d\, c\, n'\, l}{4}$$

$$= \frac{4.33 \times 0.1 \times 0.11 \times 10^{-3} \times 5.33 \times 0.1}{4} = 6.35 \times 10^{-6} \text{ m}^3/\text{s}$$

(iv) Side leakage :

Now $\frac{q_s}{q} = 0.68$ (for S = 0.121)

SIde leakage, $q_s = 0.68 \times q = 0.68 \times 6.35 \times 10^{-6} = 4.32 \times 10^{-6} \text{ m}^3/\text{s}$

(v) Temperature rise :

From table (17.4(a)) $\frac{\rho\, c'\, \Delta t_o}{p} = 14.2$

$$\Delta t_o = \frac{14.2 \times p}{\rho\, c'} = \frac{14.2 \times 9 \times 10^5}{142 \times 10^4} = 9°C$$

Rice in temperature $= 9°C$

(vi) Maximum pressure :

From the table 17.4(a) $\frac{p}{p_{max}} = 0.415$

$$\therefore p_{max} = \frac{p}{0.415} = \frac{9 \times 10^5}{0.415} = 21.7 \times 10^5 \text{ N/m}^2$$

Maximum oil pressure $= 21.7 \times 10^5 \text{ N/m}^2$

Problem 17.5:

A sleeve bearing is 10 mm in diameter and 10 mm long. SAE 10 oil at an inlet temperature 50°C is used to lubricate the bearing. The radial clearance is 0.0076 mm. If the journal speed is 3600 rpm and the radial load on the bearing is 68 N find the temperature rise of the lubricant and the minimum film thickness.

Solution:

Load on the journal, $W = 68$ N

Diameter of journal, $d = 10$ mm

Length of journal, $l = 10$ mm.

Type of oil = SAE 10

Operating temperature $= 50°C$

Radial clearance $= 0.0076$ mm.

Speed of journal $= 3600$ rpm.

Now,

$\frac{l}{d} = \frac{10}{10} = 1$ and diameter clearance $c = 2 \times 0.0076 = 0.0152$ mm

Sommerfeld number is given by

$$S = \frac{Zn'}{p}\left(\frac{d}{c}\right)^2 \quad \text{.... (A)}$$

where Z = Absolute viscosity of oil at 50°C

= 23×10^{-3} N – s/m^2 (Fig. 17.8) (PSG 7.41, JDB 19.26)

n′ = Journal speed in rps $= \frac{3600}{60} = 60$ rps

p = Pressure developed over projected area.

$$= \frac{W}{l \cdot d} = \frac{68}{10 \times 10} = 0.68 \text{ N/mm}^2 = 0.68 \times 10^6 \text{ N/m}^2$$

Substituting the above values in the equation (A), we get,

$$S = \frac{23 \times 10^{-3} \times 60}{0.68 \times 10^6}\left(\frac{10 \times 10^{-3}}{0.0152 \times 10^{-3}}\right)^2 = 0.878$$

To findout the temperature rise:

Assume the bearing as full journal bearing.

Also $\frac{l}{d} = 1$.

From the table 17.4(a), for $\frac{l}{d} = 1$ and $S = 0.878$

we have,

$$\frac{\rho\, c'\, \Delta t_0}{p} = 71.15 \text{ (By interpolation)}$$

$\therefore \Delta t_0$ = Temperature rise

$$= 71.15 \times \frac{p}{\rho\, c'} = \frac{71.15 \times 0.68 \times 10^6}{142 \times 10^4} = 34°C$$

($\rho\, c' = 142 \times 10^4$ N/m^2 °C, table 17.4(a))

To findout minimum film thickness:

From table 17.4(a), for $\frac{l}{d} = 1$ and $S = 0.878$, we have,

$$\frac{2h_0}{c} = 0.84 \text{ (By interpolation)}$$

$\therefore h_0$ = Minimum film thickness

$$= \frac{c}{2} \times 0.84 = \frac{0.0152}{2} \times 0.84 = \mathbf{0.0064\ mm.}$$

17.15. SHORT QUESTIONS AND ANSWERS :

1. What is a bearing?

Bearing is a machine member, used to support the axles and power transmitting shafts, directs the motion of shafts and also reduce friction between the contact surfaces, while carrying the load.

2. How are bearings classified?

Bearings are classified based on

(a) Nature of contact between bearing surfaces as

(i) Sliding contact bearings or plain bearings.

(ii) Rolling contact bearings or Anti-friction bearings.

(b) Direction of load applied as

(i) Radial bearings or circumferentially loaded bearings.

(ii) Thrust bearings or axially loaded bearings.

3. Specify the types of sliding contact bearings.

(i) Zero film bearings (Bearings without lubricant)

(ii) Thin film bearings (Boundary lubricated bearings)

(iii) Thick film bearings (Hydro-dynamic bearings)

(iv) Externally pressurised bearings (Hydro-static bearings)

(v) Pivot bearings

(vi) Collar bearings.

4. What is meant by Journal bearing?

A sliding contact bearing that supports load in a radial direction is called as journal bearing. It consists of two main parts such as shaft and a sleeve or bearing. The portion of the shaft inside the bearing is termed as journal as shown in figure 17.1.

5. **Distinguish between full journal bearing and partial journal bearing.**

In full journal bearing, the shaft (i.e., journal) is fully covered by the sleeve (i.e; bearing) whereas in partial journal bearing, the shaft is partly covered by the sleeve as shown in figure 17.2. Usually the full journal bearing is employed for comparatively smaller size machines than partial journal bearing using machines.

6. **Name few materials for sliding contact bearings.**

Aluminium alloys, copper alloys, babbit alloys, silver, cast iron, steel etc.

7. **What is a babbit alloy?**

Babbit alloy is the alloy of tin, lead, copper and antimony.

8. **Express the composition of gun-metal.**

Gun metal is the alloy of copper, tin and zinc in the following proportions as
Copper - 88%
Tin - 10%
Zinc - 2%

9. **What are the desirable properties of bearing materials?**

(1) Good fatigue strength to overcome failure due to change of loads.
(2) Enough compressive strength to withstand maximum radial loads.
(3) Good resistance to corrosive atmosphere.
(4) Good thermal conductivity to transfer the heat produced during operation.

10. **Fill in the blanks of the following**

(a) The preferred angle of contact for partial journal bearing is ....
(b) The rolling contact bearings are otherwise called as ....

Answers : (a) 120°; (b) Anti-friction bearings.

11. **Specify some applications of sliding contact bearings.**

Sliding contact bearing find applications in gas & diesel engines, pumps, compressors, turbines, conveyors and so on.

12. **Fill in the blanks of the following.**

(a) The journal bearing whose journal radius and bearing radius are equal is called as ....

(b) In sliding contact bearings, the friction is reduced by ....

Answers : (a) Fitted bearing. (b) Lubricants.

CHAPTER – 18

ROLLING CONTACT BEARINGS

18.1. INTRODUCTION :

The advent of automobiles and many high speed machineries make very much use of another type of bearings, known as rolling contact bearings. Since the friction produced in these bearings is very low or almost negligible, these bearings are also called as ***"Anti-friction bearings"***. They differ from sliding contact bearings in their structure and usage. The sliding contact bearings are made integral with parent machine, whereas the rolling bearings are made separately and they can be assembled with machine and disassembled from it whenever needed. The comparison of sliding and rolling bearings is given in table 18.1

18.2. COMPONENTS OF ROLLING BEARINGS :

The rolling bearing consists of four main components *(1) the inner ring, (2) outer ring, (3) the balls or rollers and (4) the retainers or separators* as shown in figure 18.1. The inner ring is force-fitted with machine shaft and the outer ring is force-fitted with machine housing. The shaft rotates because of relative rotations of balls or rollers. The retainer is used to prevent the balls or rollers exiting from bearing rings during operation.

Some of the rolling bearings are available with

1. shields to prevent dirt from entering and to retain grease
2. shields and seals to contain lubricant for self-lubrication and
3. snap rings and flanges that provide for simple bearing containment.

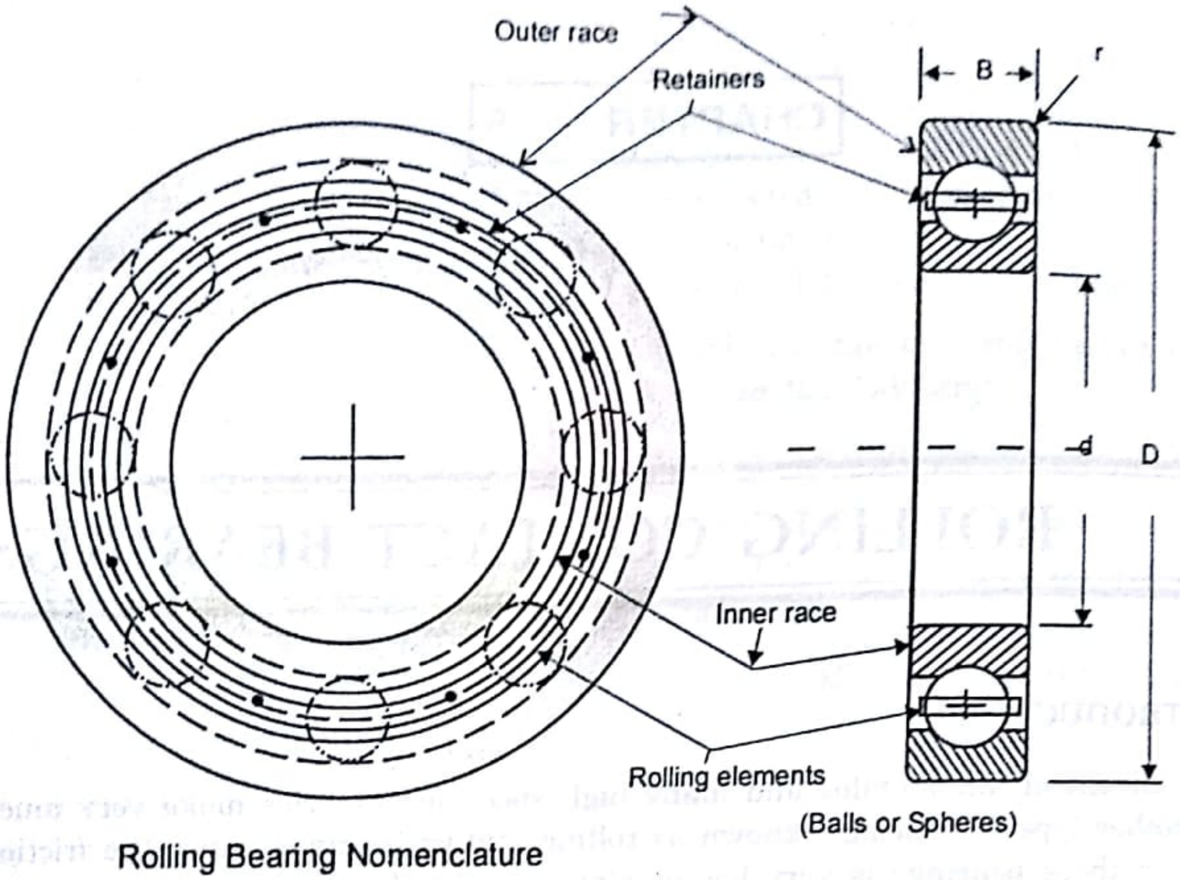


Fig. 18.1

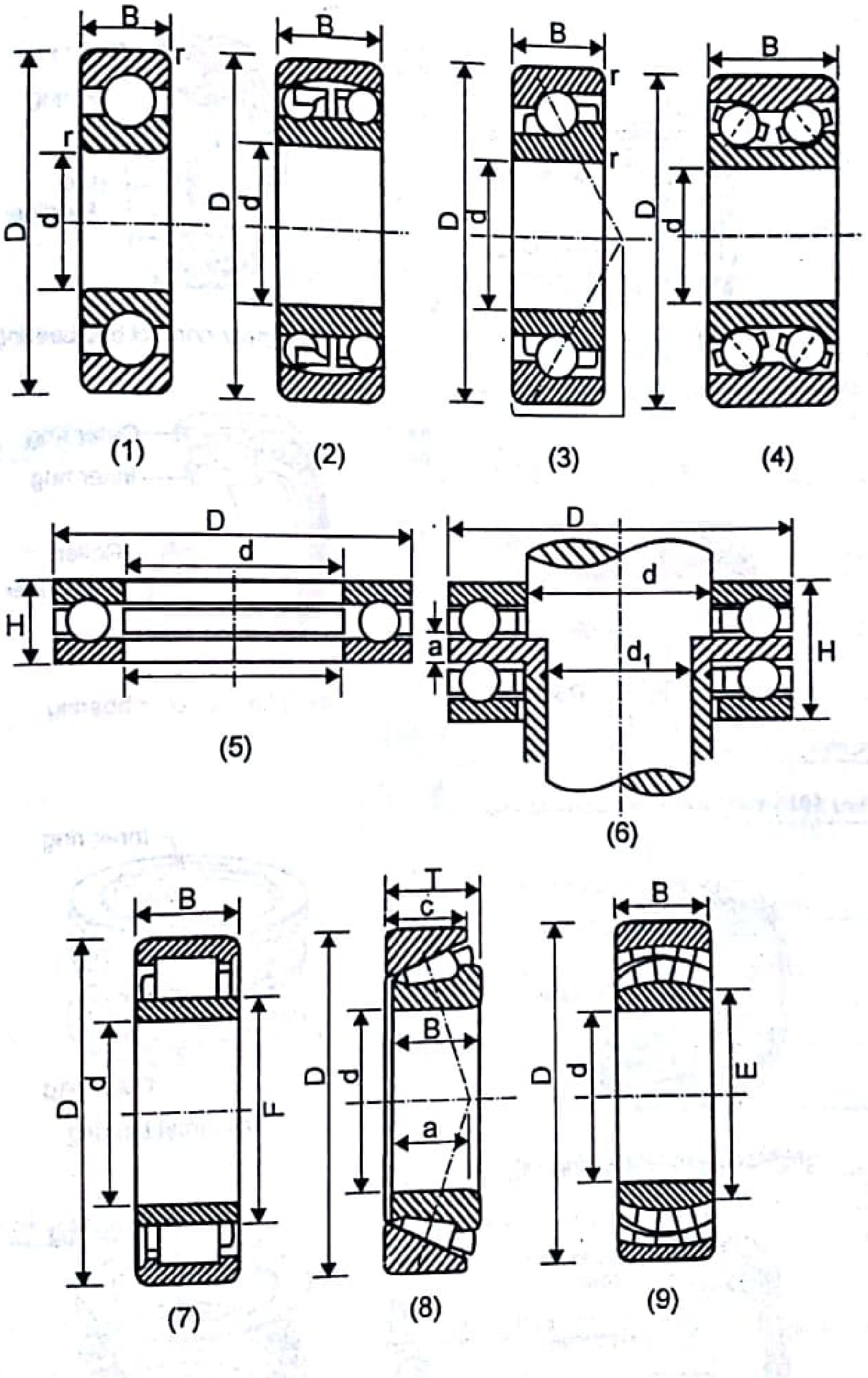
18.3. CLASSIFICATION OF ROLLING CONTACT BEARINGS :

The rolling contact bearings are classified into two major groups with respect to their structure as

1. Ball bearings.
2. Roller bearings.

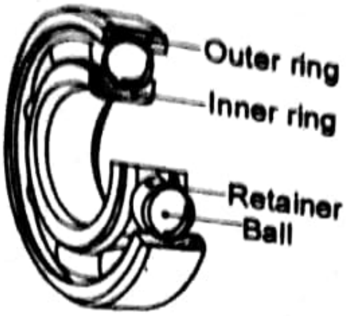
Basically the structure of ball and roller bearings are similar except that whether the rolling elements between the inner ring and outer ring are balls or rollers. Also these ball bearings are made into many types such as deep groove or ***Conrad-type*** ball bearings, angular contact ball bearings, thrust ball bearings and so on. Similarly the roller bearings are made of taper roller bearings, cylindrical roller bearings, thrust roller bearings and so on. The schematic diagrams for various rolling bearings are shown in figure 18.2(a), (b) and (c)

Both types of bearings can carry radial loads and axial loads separately or in combined form. Generally the ball bearings are used for light loads and the roller bearings are usually employed for heavier loads. Also, in the case of ball bearings, the nature of contact is the point contact and hence the friction produced is very low compared to roller bearings where the nature of contact is the line contact which produce more friction.

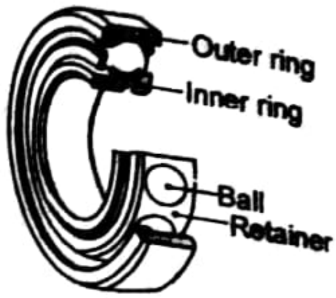


1. Deep groove ball bearing
2. Self aligning ball bearing
3. Single row angular contact ball bearing
4. Double row angular contact ball bearing
5. Single thrust ball bearing
6. Double thrust ball bearing
7. Cylindrical roller bearing
8. Taper roller bearing
9. Spherical roller bearing

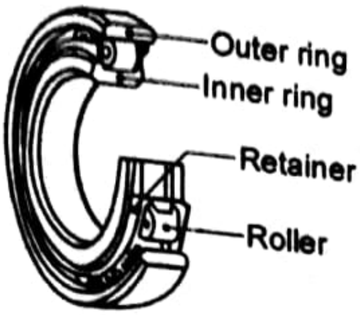
Fig. 18.2(a): Types of Rolling Contact Bearings.



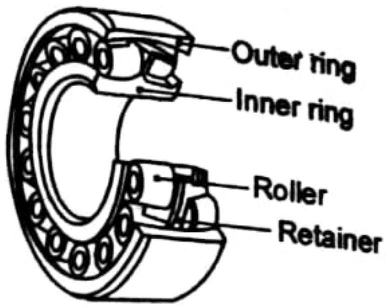
(a) Deep-groove (Conrad) ball bearing



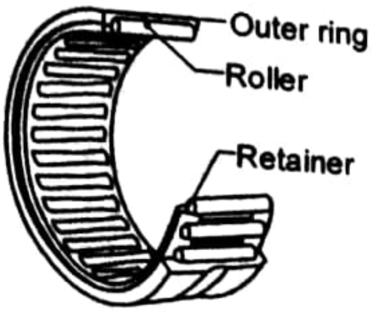
(b) Angular-contact ball bearing



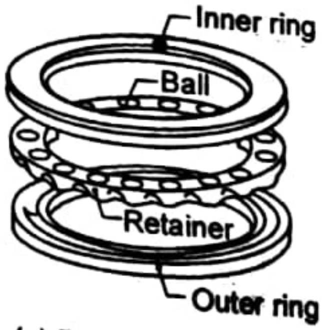
(a) Cylindrical roller bearing



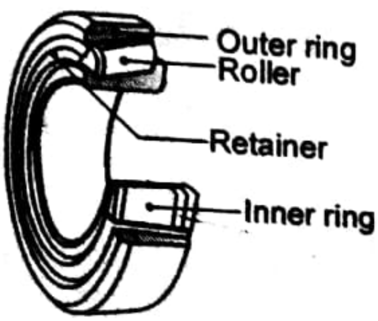
(d) Spherical roller bearing



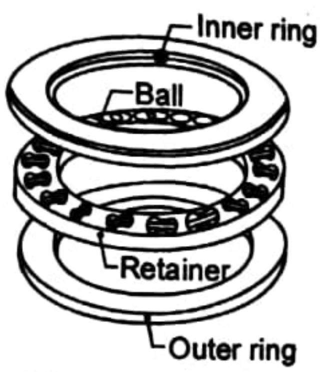
(b) Needle roller bearing



(a) Ball thrust bearing



(c) Tapered roller bearing



(b) Roller thrust bearing

Fig. 18.2(b)

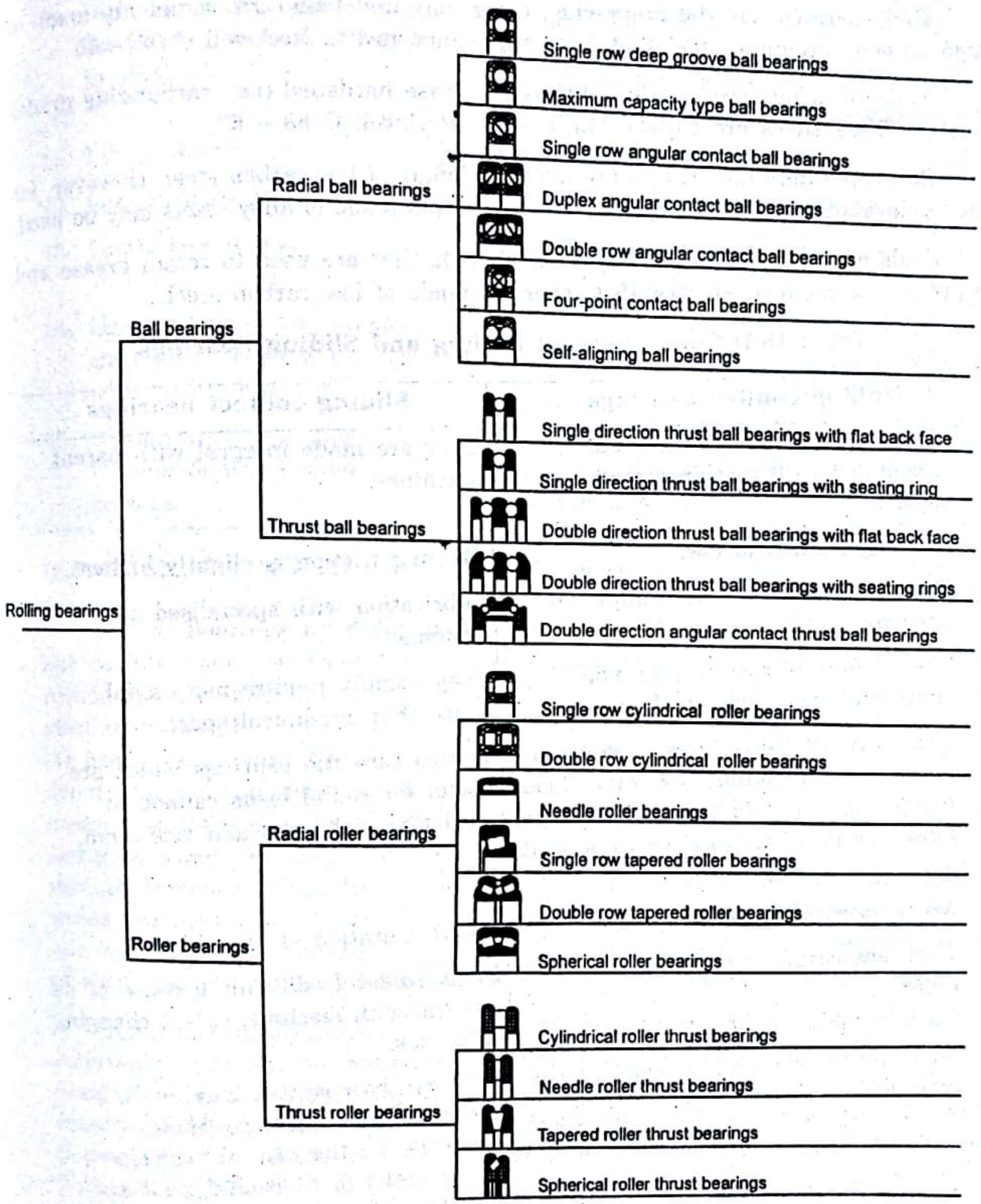


Fig. 18.2(c): Classification of Rolling Element Bearings

18.4. MATERIALS USED FOR ROLLING BEARINGS :

Ball bearings (i.e, the inner ring, outer ring and balls) are commonly made of high carbon chromium steel that is through hardened to Rockwell C 58 – 65.

Roller bearings are usually fabricated of case hardened (i.e., carburizing grade) steels. These steels are surface hardened to Rockwell C 58 – 63.

Ball separators (i.e., cages) are normally made of low carbon steel. However, for use of elevated temperature upto 400°C, separators made of alloy steels may be used.

Seals are made of low carbon steel. Shields that are used to retain grease and to prevent chips, dirt, etc are also generally made of low carbon steel.

Table 18.1: Comparison of Rolling and Sliding Bearings

Rolling contact bearings	Sliding contact bearings
1. These are made separately and assembled with machines whenever needed.	They are made integral with parent machines.
2. Starting friction is low.	Starting friction is slightly higher.
3. Ease of lubrication with simple oil systems	Lubrication with specialised oil systems.
4. They generally require less axial space and more diametral space.	They usually require more axial space than diametral space.
5. Generally all the rolling bearings will support both radial and axial loads except some specific bearings which are most suitable for radial or axial loads only.	In this case the bearings which are useful for radial loads cannot be used for axial loads and vice-versa.
6. Noisy operation at high speed.	Silent operation at any speed.
7. They are readily replaceable as stock items.	Replacement is difficult because of its integral with machine, except changing of bushes.
8. They can be operated at any inclined position.	They can be operated only at horizontal or vertical positions. Generally all the journal bearings are operated in horizontal positions and thrust bearings are operated at vertical positions.
9. Initial cost is usually higher.	Initial cost is slightly lower.

Rolling contact bearings	Sliding contact bearings
10. Dirt, metal chip which enter inside the bearing will reduce its life due to abrasion.	Sliding contact bearings will not suffer by these materials because these foreign materials will be washed away during operation.
11. Life is finite.	Life is almost infinite.
12. Less capacity to withstand shock.	High capacity to withstand shock.
13. Power loss is low.	Power loss is usually higher due to friction.
14. Usage is more. i.e., majority of high class machine tools are operated with rolling bearings.	Here the usage is limited, i.e., they may be used in those places where rolling bearings cannot be used.
15. Work accuracy is more. This is due to coincidence of shaft and bearing centres.	Here accuracy is poor, because of non-coincidence of shaft and bearing centres.

18.5. DESIGNATING THE ROLLING BEARINGS :

Rolling Bearings are designated in different methods by different manufacturers, but mainly based on their load carrying capacities. According to SAE (Society of Automotive Engineers), bearings are grouped into (1) Very light series whose number start with 100 (2) light series which start with 200, (3) medium series with 300 and (4) heavy series with 400. In any bearing number, the last two digits are used to denote the bore diameter in multiples of 5 mm. Thus the bearing 315 signifies a medium series bearing of 75mm bore. Similarly the bearing 415 signifies the heavy series of same (75 mm) bore, but its load carrying capacity is more. Generally, medium series bearings have load carrying capacity upto 40% greater than light series bearings of same bore, but occupy more axial and radial space on the shaft and in the frame. Similarly the heavy series bearings have a load carrying capacity of about 20 to 30% greater than medium series bearings.

According to Anti-Friction Bearing Manufacturers Association (AFBMA) and International Standards Organisation (I.S.O) and also Indian standards, bearings are designated by their bore diameter followed by type of bearing and type of duty. For example, for the bearing represented by 30 BC 02, the letter BC represents the type of bearing. The prefix 30 indicates its bore diameter in mm and the suffix 02 indicates that it belongs to light duty.

According to SKF (Svenska Kullager Fabriken) ball bearing company, bearings are designated by different series for different groups of bearings. For example, series started with following numbers are,

60, 62, 63, 64	–	for deep groove ball bearings
12, 13, 22, 23	–	for self-aligning ball bearings
72, 73, 32, 33	–	for angular contact ball bearings
512, 513, 522, 523, 532	–	for thrust ball bearings
222, 230, 231, 240, 241	–	for spherical roller bearings
302, 322, 323	–	for taper roller bearings.
NU2, NU22, NN 30 K, NNU	–	for cylindrical roller bearings
NA49, NA69, RNA 49, NKXR	–	for needle bearings

In SKF bearings also, the last two digits of any bearing number represent the bore diameter in multiples of 5 mm except the first four numbers of any series.

For example :

SKF Designation	:	6200	6201	6202	6203	6204	6205
Bore diameter in mm	:	10	12	15	17	20	25

18.6. FACTORS INFLUENCING THE SELECTION OF BEARINGS :

Each type of bearing has its own characteristic properties which make it particularly suitable for certain applications. When selecting the bearings, several factors must be considered.

(i) Space availability :

The bore diameter, axial and radial spaces are some of the eminent parameters which are to be considered when selecting the bearings. Deep groove ball bearings are normally selected for small diameter shafts whereas cylindrical roller bearings and spherical roller bearings are generally preferred for large diameter. If radial space is limited, bearings with low sectional height like needle roller bearings may be chosen. Where axial space is limited narrow bearings are preferred.

(ii) Load :

The magnitude of load usually decides the size of bearing. Generally roller bearings can carry heavier loads than ball bearings having the same external dimensions. Deep groove ball bearings, cylindrical roller bearings are especially employed for radial loads. Thrust ball bearings, thrust roller bearing are usually used for axial loading. Other bearings like angular contact bearing, taper roller bearing, can carry radial loads as well as axial loads separately or combinedly.

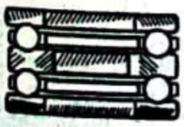
(iii) Speed :

For very high speed applications, generally deep groove ball bearings may be preferred.

(iv) Misalignment :

Self-aligning ball bearings may be used where any angular displacement of the shaft is to be adjusted. Cylindrical roller bearings can adjust any small axial misalignment. The different types of bearings for different types of applications are given in table 18.2.

Table 18.2: Applications of Ball and Roller Bearings

Figure	Types of bearings	Application
	Deep groove ball bearing	For considerable thrust load apart from radial load-high speed.
	Self aligning ball bearing	Minor angular displacements of shafts will not affect.
	Single row angular contact ball bearing	For heavy axial loads
	Double row angular contact ball bearing	For radial loads with heavy thrust in both directions
	Spherical roller bearing	High carrying capacity, self aligning-for heavy radial loads with considerable axial load in both directions
	Cylindrical roller bearing	For heavy radial loads at high speeds-permit slight axial displacement.
	Taper roller bearing	For combined radial and axial loads
	Single thrust ball bearing	For axial load in one direction only
	Double thrust ball bearing	For axial loads acting on both directions
	Spherical roller thrust bearing	For heavy axial loads-high speed-self aligning

18.7. SELECTION OF BEARING SIZE :

The size of a bearing for any application may be decided on the basis of its load carrying capacity in relation to the loads to be carried and the requirements regarding life and reliability. Generally the basic static load rating and basic dynamic load rating are used to express the load carrying capacities of any bearing.

(i) **Basic static load rating : (symbolized by C_o)** is used in calculations when the bearings are to rotate at very slow speeds, or to be stationary under load during certain periods. It must also be taken into account when heavy shock loads at short duration acting on a rotating (dynamically stressed) bearing. The value of C_o depends on the bearing material, the number of rows of rolling elements, the number of rolling elements per row, the bearing contact angle, and the ball or roller diameter. Normally the basic static load rating has little influence in the selection of rolling bearing.

(ii) **Basic dynamic load rating (C)** also known as *specific dynamic capacity* is used for calculations involving dynamically stressed bearings, i.e., a bearing which is rotating under load. It expresses a bearing load which will give a basic rating life of 1 million (i.e., 10^6) revolutions. The basic dynamic load rating for different materials, symbolized by C, depends on the same factors which determine C_0, except for additional parameter concerning the load geometry. The dynamic load rating C enters directly into the process of selecting a bearing.

(iii) **The life** of a rolling bearing is defined as the number of revolutions (or the number of operating hours at a given constant speed) which the bearing is capable of enduring before the first evidence of fatigue, that is developed in the material of either rings or rolling elements. It is, however, evident from both laboratory tests and practical experience that seemingly identical bearings operating under identical conditions have different lives (i.e., number of hours or revolutions). All information presented on dynamic load rating is based on the life that 90% of sufficiently large group of apparently identical bearings can be expected to attain or exceed. This is called as the ***basic rating life (or nominal life)***. It may also be referred as L_{10} life and this is the minimum life. The majority of bearings attain a much longer life than this nominal life which may be known as ***"the median life"***, and it is approximately equal to five times the nominal life. (i.e., median life $= 5\,L_{10}$). The ***average life*** of bearing is defined as the summation of all bearing lives in series of life tests divided by the number of life tests. This average life is different from median life. Some manufacturers define average life of a bearing as approximately equal to median life, i.e,e, 5 times the nominal life.

18.8. INFLUENCE OF OPERATING TEMPERATURE ON BEARING MATERIAL :

At elevated temperature, the hardness of the bearing materials is reduced and thus the dynamic load carrying capacity is also reduced as a consequence. The reduction in dynamic load carrying capacity at different temperature is taken into account for determining the rating life at that elevated temperature. The following table 18.3 shows the decrease of load capacity with temperatures.

Table 18.3

Bearing temperature (in °C)	:	125	150	175	200	225	250
Decrease in capacity (in %)	:	5	10	15	25	35	40

18.9. SELECTION OF BEARINGS FOR STEADY LOADING :

The size of bearing required is judged by the magnitude and nature of applied load, life and reliability. The bearing load is composed of weights involved, forces derived from power transmitted and additional forces based on method of operation etc.

Different manufacturers recommend different methods for the selection of bearing in accordance with load rating. According to AFBMA, the bearings are selected based on dynamic load rating C. By conducting number of tests, the dynamic load capacity can be determined as,

$$C = \left(\frac{L}{L_{10}} \right)^{\frac{1}{k}} \times P \qquad \text{.... (18.1)}$$

where

C = Basic dynamic load rating

L = Life of the bearing required in million revolutions

L_{10} = Life of bearing for 90% survival at 1 million revolutions (or 10% failure)

P = Equivalent load

k = Exponent

= 3 for ball bearings

= 10/3 for roller bearings

The equivalent load (P) is defined according to AFBMA as that constant radial load, which, if applied to a bearing with rotating inner ring and stationary outer ring, would give the same life as that which the bearing would attain under the actual condition of load and rotation. In many applications, bearings have to carry a load composed of axial and radial components. In addition, they are sometimes required to operate with a rotating outer and a stationary inner ring. It then becomes necessary to express these conditions by an equivalent load satisfying the above definition. The expression used to define the equivalent load is given by

$$P = (V X F_r + Y F_a) S \quad \text{.... (18.2)}$$

where P = Equivalent load

F_r = Radial load

F_a = Axial load

X = Radial load factor (Table 18.4)

Y = Axial load factor (Table 18.4)

V = Race rotation factor

= 1.0 for inner ring rotation and outer ring stationary.

= 1.2 for inner ring stationary and outer ring rotation.

= 1.0 for inner ring or outer ring rotation in the case of self-aligning ball bearing.

S = Service factor which is given in table 18.5

The equation (18.1) can be specified in the following form also in order to show the relationship between the life of bearing and load capacity as

$$L = \left(\frac{C}{P}\right)^k \text{ million revolutions} \quad \text{.... (18.3)}$$

Table 18.4: Equivalent Bearing Load

Type of bearing	Series (SKF)		$\frac{F_a}{F_r} \le e$ X	$\frac{F_a}{F_r} \le e$ Y	$\frac{F_a}{F_r} > e$ X	$\frac{F_a}{F_r} > e$ Y	e
Deep groove ball bearing	Series 60, 62, 63, 64	$\frac{F_a}{C_0} = 0.025$	1	0	0.56	2	0.22
		= 0.04	1	0	0.56	1.8	0.24
		= 0.07	1	0	0.56	1.6	0.27
		= 0.13	1	0	0.56	1.4	0.31
		= 0.25	1	0	0.56	1.2	0.37
		= 0.5	1	0	0.56	1.0	0.44
Angular contact ball bearing	72B, 73B, 32, 33		1	0	0.35	0.57	1.14
			1	0.73	0.62	1.17	0.86
Self aligning ball bearing	2200– 2204		1	1.3	0.65	2	0.5
	05– 07		1	1.7	0.65	2.6	0.37
	08– 09		1	2	0.65	3.1	0.31
	10– 13		1	2.3	0.65	3.5	0.28
	14– 20		1	2.4	0.65	3.8	0.26
	21– 22		1	2.3	0.65	3.5	0.28
	2301		1	1	0.65	1.6	0.63
	2302– 2304		1	1.2	0.65	1.9	0.52
	05– 10		1	1.5	0.65	2.3	0.43
	11– 18		1	1.6	0.65	2.5	0.39
Spherical roller bearing	22205C–22207C		1	2.1	0.67	3.1	0.32
	08C–09C		1	2.5	0.67	3.7	0.27
	10C–20C		1	2.9	0.67	4.4	0.23
	22C–44C		1	2.6	0.67	3.9	0.26
Taper roller bearing	32206–208		1	0	0.4	1.6	0.37
	09– 22		1	0	0.4	1.45	0.41
	24– 30		1	0	0.4	1.35	0.44

Table 18.5: Service Factor (S)

Machineries	Service factor (S)
For bearings carrying geared shafts	
Rotary machines with no impact	1.1 - 1.5
Reciprocating machines	1.3 - 1.9
Machines with pronounced impact, hammer mill, presses	1.6 - 4
For electric motors:	
Stationary machines	1.5 - 2
Traction motors	1.5 - 2.5
For belt and chain drives:	
Chain drives	1.5 - 1.7
V-belt drives	2.0 - 2.5
Leather belt drives	2.5 - 3.5
Fabric belt drives	2.0 - 3.0
Steel belt drives	3.0 - 4.0

Note: The following values of service factor may also be used for the lack of specifications about the machineries.

(i) Constant or steady load – 1.0

(ii) Light shocks – 1.5

(iii) Medium shocks – 2.0

(iv) Heavy shocks – 2.5

If a bearing is required to operate for a particular life and the selected bearing is having more life than requirement, then, the probability of selected bearing surviving the required life is given by expression as

$$\frac{L}{L'_{10}} = \left[\frac{ln(1/p)}{ln(1/p_{10})}\right]^{1/b} \quad \text{.... (18.4)}$$

where,

L = Required life in million revolutions

L'_{10} = Calculated life of selected bearing for given load at 90% survival in million revolutions.

$ln\ (1/p_{10}) = (1/0.9) = 0.1053$

b = 1.17 for a median life $= 5\ L_{10}$

= 1.34 for a median life $= 4.08\ L_{10}$ (for Deep groove ball bearing)

p = Required probability of selected bearing.

For system having "x" bearings, each having the probability as p, then the probability of survival of $p_{system} = p^x$

18.10. SELECTION PROCEDURE FOR ROLLING BEARINGS :

1. Determine the radial and axial forces (i.e., loads) accurately from the working conditions.
2. Select the type of bearing such as ball or roller bearing etc. from load conditions (Table 18.2).
3. Calculate the working life of bearing (or select from the table 18.6) (PSG 4.5, JDB 20.6)
4. Find out the dynamic capacity C using

$$C = \left(\frac{L}{L_{10}}\right)^{\frac{1}{k}} \cdot P \qquad \text{(PSG 4.2, JDB 20.2)}$$

where L = Life required in million revolutions

$$= \frac{\text{Life in hours} \times \text{r.p.m} \times 60}{10^6}$$

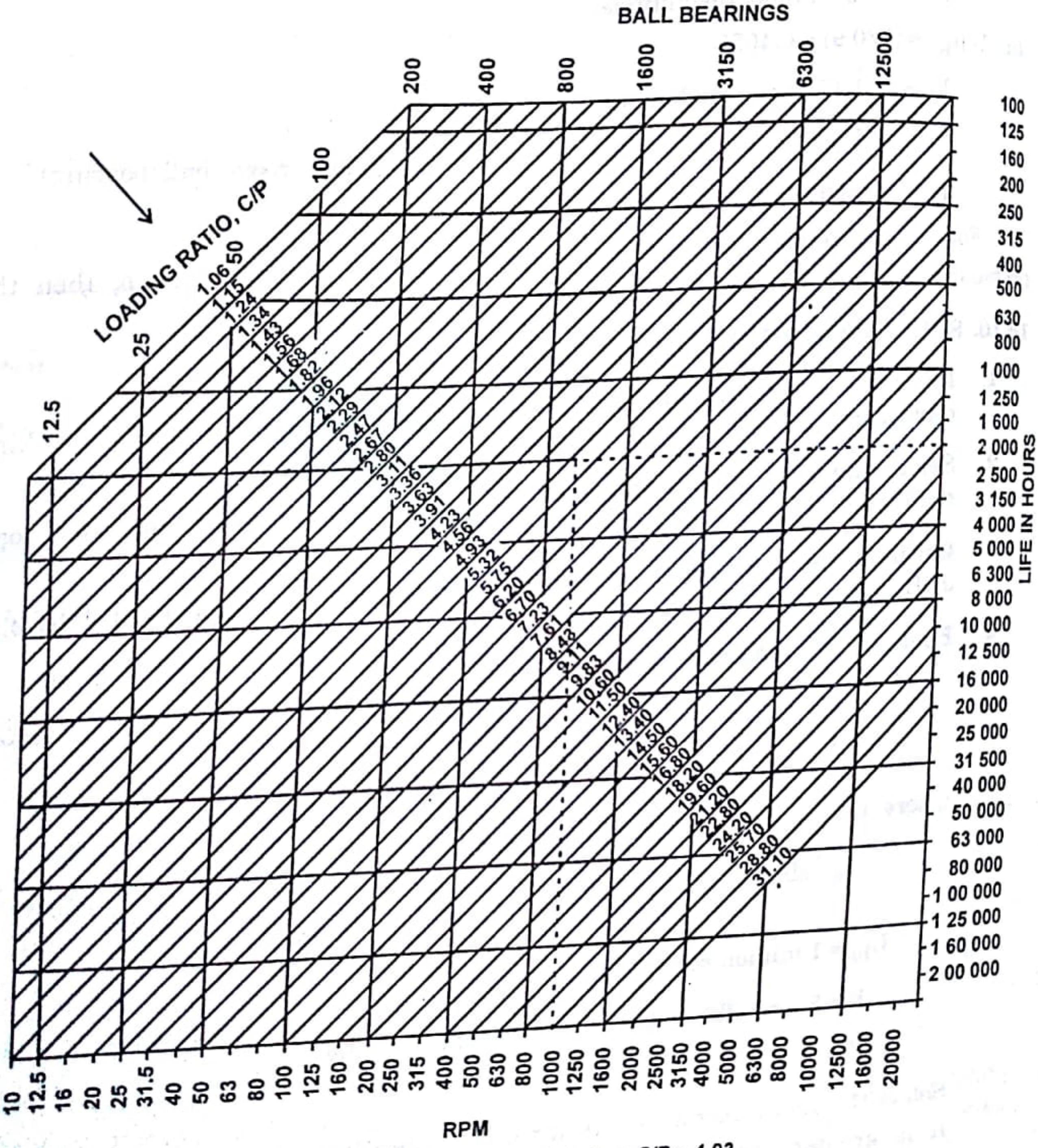
L_{10} = 1 million revolutions

k = 3 for ball bearings; 10/3 for roller bearings

$P = (V\,X\,F_r + Y\,F_a)\,S$

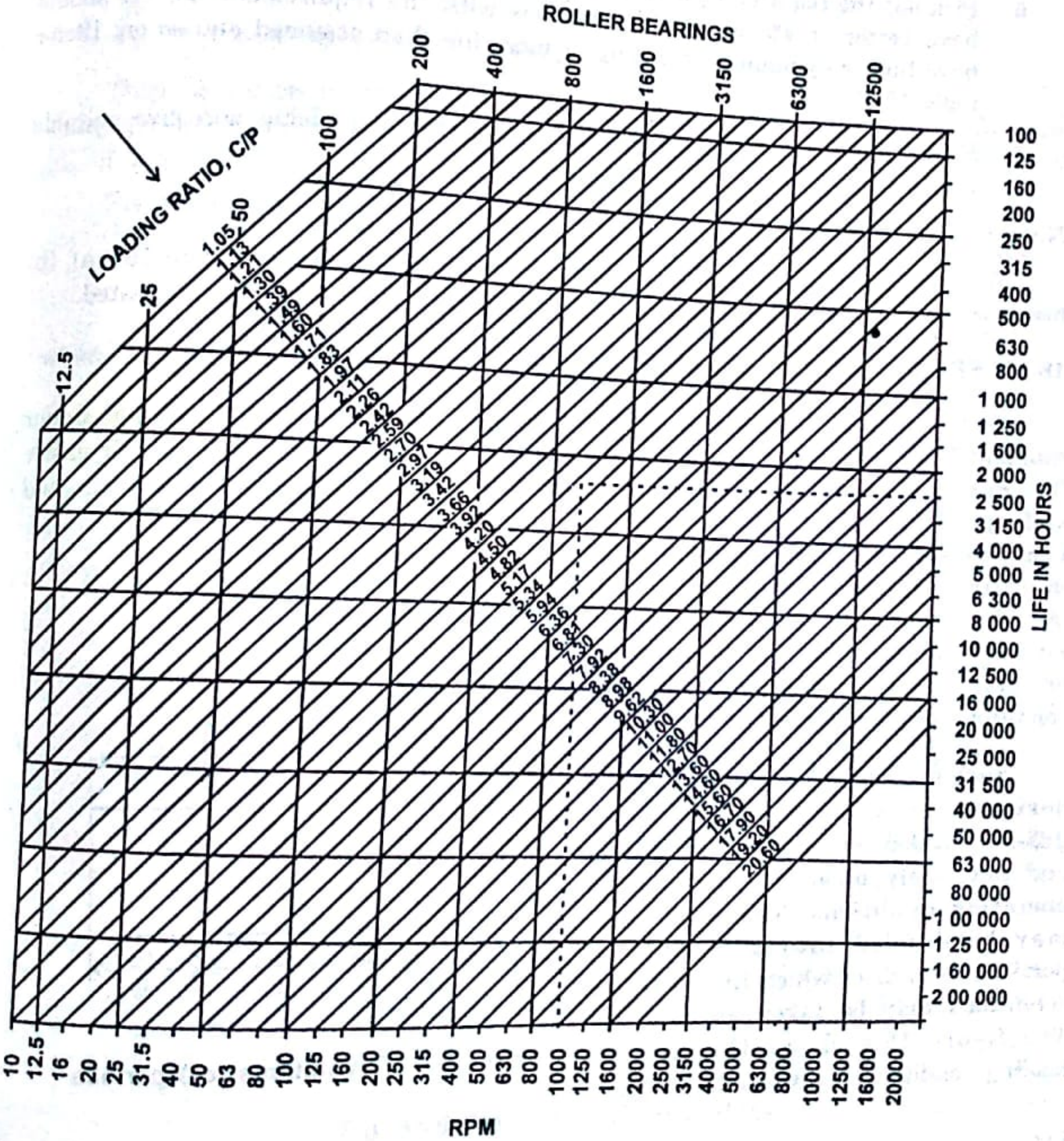
Select the radial factor (X), axial factor (Y) based on the ratio F_a/F_r, whether it is greater than or less than the limiting value "e" by using the table 18.4.The factor 'e' defines a minimum ratio between the axial and radial forces below which the axial force can be ignored (i.e., set to zero) in equation 18.2.

Assume other factors like V and S based on the requirements.



Example: At 1000 RPM and a life of 2000 Hrs, C/P = 4.93

Fig. 18.3(a)



Example: At 1000 RPM and a life of 2500 Hrs, C/P = 4.50

Fig. 18.3(b)

5. If the bearing is to be operated at higher temperatures, then determine the dynamic capacity suitably.
6. Pick out the required bearing from the manufacturer's catalogue which should have better or atleast equal properties with the requirement, i.e., it should have higher dynamic load rating or more life than assumed and so on. [Refer table 18.7].
7. Evaluate the other proportions of bearing and its housing and give suitable tolerance.

Note :

We may select C/P ratio directly from the charts as shown in figure 18.3(a), (b) based on the life and speed of bearing and from that also C can be calculated.

18.11. SELECTION OF BEARING FOR VARIABLE LOADING :

So far what we have learnt are the methods for the selection of bearings under constant loading at given period. ie., the bearings acted by same load in their entire life. But practically, the rolling bearings are frequently operated under variable load and speed conditions. This is due to many causes like power fluctuations as in electrical machineries or requirement of different cutting forces for different kinds of materials as in machine tools, or running with loading and unloading conditions as in automobiles and so on.

Such a variable loaded bearings are designed by considering all these different loaded conditions of work cycle and not solely upon the most severe operating conditions. The work cycle may be divided into a number of portions in each of which the operating conditions may be taken as constant. The figure 18.4 shows the variable loading conditions for variable speed.

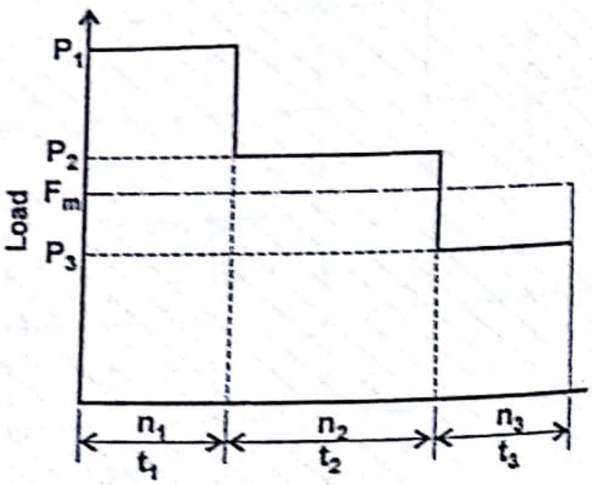


Fig. 18.4: Revolutions (or) periods

Let

P_1 = Constant load during n_1 revolutions (or) during the period of time t_1

P_2 = Constant load during n_2 revolutions (or) during the period of time t_2.

P_3 = Constant load at n_3 (or) t_3

. . .

. . .

. . .

P_n = Constant load at n_n (or) t_n

Then the approximate value of the equivalent mean load ***(i.e., cubic mean load)*** for the whole cycle of operation, which will have the same influence on the life of bearing as the fluctuating load, is obtained as

For variable speed, the cubic mean load

$$P_m = \left[\frac{P_1^3 n_1 + P_2^3 n_2 + P_3^3 n_3 + ... + P_n^3 n_n}{\Sigma n} \right]^{1/3}$$

where $\Sigma n = n_1 + n_2 + n_3 + ... + n_n$

And for variable time

$$P_m = \left[\frac{P_1^3 t_1 + P_2^3 t_2 + P_3^3 t_3 + ... + P_n^3 t_n}{\Sigma t} \right]^{1/3}$$

where $\Sigma t = t_1 + t_2 + t_3 + ... + t_n$

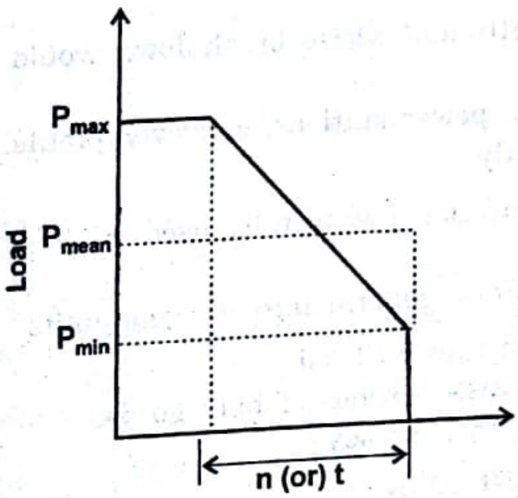


Fig. 18.5: Revolutions (or) periods

If the load is fluctuating linearly over a certain period as shown in figure 18.5 the mean load for this period may be obtained as

$$P_{mean} = \frac{2\,P_{max} + P_{min}}{3}$$

18.12 SELECTION PROCEDURE FOR THE BEARINGS UNDER VARIABLE LOADING :

1. Note the different loads acting at different speeds or at different periods accurately from the working conditions.
2. Calculate the number of revolutions per unit time or portion of period per unit revolution.
3. Find out the equivalent mean load assuming time or speed constant.
4. If the load is acting linearly over a certain period, then calculate the mean load for that period before calculating equivalent mean load.
5. -Then determine the basic dynamic capacity, and the life of bearings as mentioned in the previous article 18.10

Table 18.6: LIFE OF BEARINGS

Class of Machines	Bearing Life Working Hours
Instruments and apparatus that are used only rarely e.g. Demonstration apparatus, sliding door mechanism	500
Machines used for short periods or intermittently and whose break-down would not have serious consequences e.g. Hand tools, lifting table, agricultural machines, domestic appliances	4000 to 8000
Machines working intermittently and whose break-down would have series consequences e.g. Auxiliary machines in, power stations, conveyor plants, lifts, m/c tools used infrequently	8000 to 12000
Machines for use 8 hrs/day and not always fully used e.g. Stationary electric motors, general purpose gear units	12000 to 20000
Machines for use 8hrs/day and fully utilised e.g. Machines for engg. industry-cranes of bulk goods, ventilating fans, counter shaft in gear box.	20000 to 30000
Machines for continuous use 24hrs/day e.g. Separators, compressors, pumps, mine hoists, stationary electric machines, on-board naval vessel machines.	40000 to 60000
Machines required to work with a high degree of reliability for 24 hrs/day e.g. Pulp and paper making machinery, public power-plants, mine pumps, water works, on-board merchant ship machines	100000 to 200000

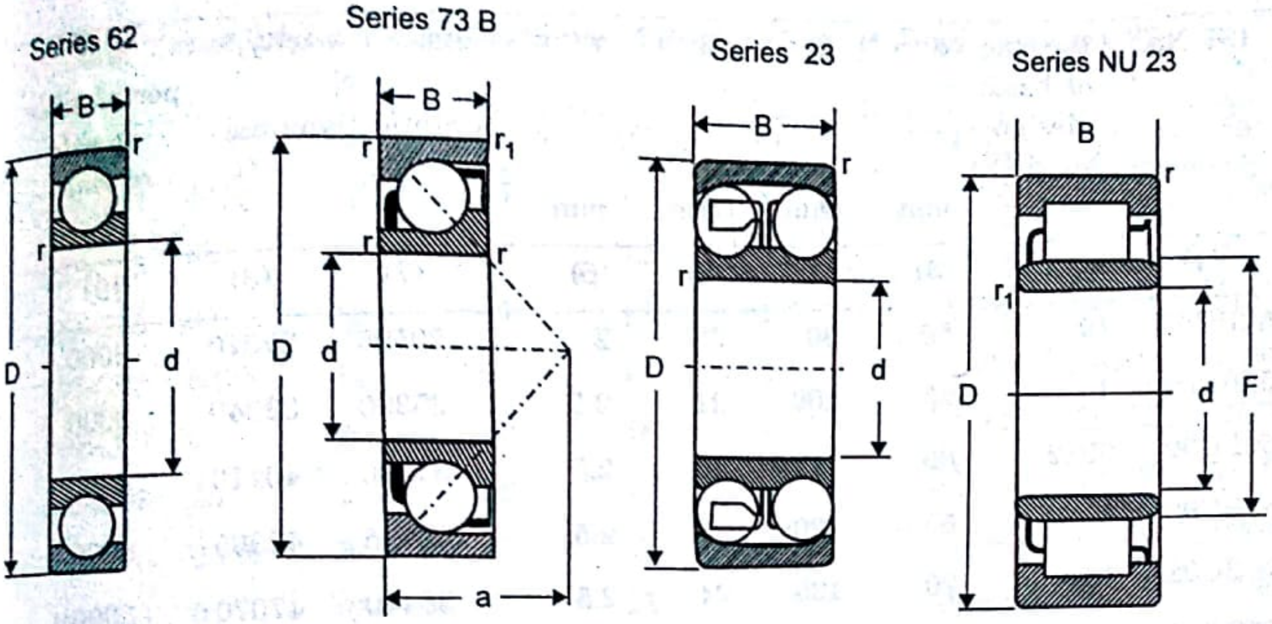


Fig. 18.6

Table 18.7(a): Deep Groove Ball Bearings (Series 62)

ISI No.	Bearing of basic design No.(SKF)	d mm	D mm	B mm	r mm	Basic capacity,* N Static C_0	Dynamic C	Max. permissible speed rev/min
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
10BC02	6200	10	30	9	1	2160	3925	20000
12BC02	01	12	32	10	1	2940	5250	20000
15BC02	02	15	35	11	1	3430	5980	16000
17BC02	6203	17	40	12	1	4315	7355	16000
20BC02	04	20	47	14	1.5	6375	9805	16000
25BC02	05	25	52	15	1.5	6965	10690	13000
30BC02	6206	30	62	16	1.5	9805	14710	13000
35BC02	07	35	72	17	2	13536	19615	10000
40BC02	08	40	80	18	2	15495	22165	10000
45BC02	6208	45	85	19	2	17750	24910	8000

ISI No.	Bearing of basic design No.(SKF)	d mm	D mm	B mm	r mm	Basic capacity,* N Static C_0	Dynamic C	Max. permissible speed rev/min
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
50BC02	10	50	90	20	2	20595	27070	8000
55BC02	11	55	100	21	2.5	25300	33340	8000
60BC02	6212	60	110	22	2.5	31580	40210	6000
65BC02	13	65	120	23	2.5	35715	42905	6000
70BC02	14	70	125	24	2.5	38440	47070	5000
75BC02	6215	75	130	25	2.5	41385	50600	5000
80BC02	16	80	140	26	3	44130	55505	5000
85BCO2	17	85	150	28	3	53450	63745	4000
90BC02	6218	90	160	30	3	60800	74040	4000
95BC02	19	95	170	32	3.5	71100	82870	4000
100BCO2	20	100	180	34	3.5	79925	97145	3000
105BC02	6221	105	190	35	3.5	90710	101500	3000
110BC02	22	110	200	38	3.5	101010	108850	3000
120BC02	24	120	215	40	3.5	101010	110815	3000
	6226	130	230	40	4	112780	120130	2500
	28	140	250	42	4	127680	126510	2500
	30	150	270	45	4	139745	135330	2500
	32	160	290	48	4	151020	140235	2000
	6234	170	310	52	5	184365	159850	2000
	36	180	320	52	5	200060	169165	1600
	38	190	340	55	5	235360	196130	1600
	40	200	360	5	58	259880	205940	1600

Table 18.7(b): Deep Groove Ball Bearings (Contd.) (Series 63)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
10BC03	6300	10	35	11	1	3570	6080	16000
12BC03	01	12	37	12	1.5	4220	7550	16000
15BC03	02	15	42	13	1.5	5100	8580	16000
17BC03	6303	17	47	14	1.5	6080	10300	13000
20BC03	04	20	52	15	2	7550	12260	13000
25BC03	05	25	62	17	2	10150	15985	10000
30BC03	6306	30	72	19	2	14220	20990	10000
35BC03	07	35	80	21	2.5	16970	25300	8000
40BC03	08	40	90	23	2.5	20990	31380	8000
45BC03	6309	45	100	25	2.5	29225	40700	8000
50BC03	10	50	110	27	3	34720	47070	6000
55BCO3	11	55	120	29	3	41190	53940	6000
60BC03	6312	60	130	31	3.5	47070	62270	5000
65BCO3	13	65	140	33	3.5	53450	71100	5000
70BC03	14	70	150	35	3.5	60800	79925	5000
75BC03	6315	75	160	37	3.5	71100	89280	4000
80BC03	16	80	170	39	3.5	78450	94140	4000
85BC03	17	85	180	51	4	85810	101500	4000
90BC03	6318	90	190	43	4	96110	107870	3000
95BC03	19	95	200	45	4	107870	117680	3000
100BC03	20	100	215	47	4	129450	135330	3000
105BC03	6321	105	225	49	4	140240	140235	2500
110BC03	22	110	240	50	4	160730	152980	2500
120BC03	24	120	260	55	4	162790	157890	2500
	6326	130	280	58	5	191230	173580	2500
	28	140	300	62	5	215750	196130	2000
	30	150	320	65	5	249090	210840	2000

Table 18.7(c): Self-Aligning Ball Bearings (Series 22)

Bearing with cylindrical bore No.(SKF)	Bearing with taper bore No. (SKF)	d mm	D mm	B mm	r mm	Basic capacity,* N Static C_0	Basic capacity,* N Dynamic C	Max. permissible Speed, rev/min
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
2200		10	30	14	1	1670	5540	20000
01		12	32	14	1	1960	5640	20000
02		15	35	14	1	2110	5740	16000
2203		17	40	16	1	2745	7500	16000
04	2204K	20	47	18	1.5	3825	9610	16000
05	05 K	25	52	18	1.5	4120	9610	13000
2206	2206 K	30	62	20	1.5	5390	11770	13000
07	07 K	35	72	23	2	7845	16180	10000
08	08 K	40	80	23	2	8825	16920	10000
2209	2209 K	45	85	23	2	9810	17140	8000
10	10 K	50	90	23	2	10490	17410	8000
11	11 K	55	100	25	2.5	12455	20200	8000
2212	2212 K	60	110	28	2.5	15300	25810	6000
13	13 K	65	120	31	2.5	19610	33340	6000
14	–	70	125	31	2.5	21080	34080	5000
2215	2215 K	75	130	31	2.5	21575	34080	5000
16	16 K	80	140	33	3	24520	37850	5000
17	17 K	85	150	36	3	29030	44620	4000
2218	2218 K	90	160	40	3	35550	53450	4000
19	19 K	95	170	43	3.5	42170	63740	4000
20	20 K	100	180	46	3.5	49030	76000	3000
2221	–	105	190	50	3.5	53840	82870	3000
22	222K	110	200	53	3.5	62270	96600	3000

Table 18.7(d): Single Row Angular Contact Ball Bearings (Series 72 B)

ISI No.	Bearing No. (SKF)	d mm	D mm	B mm	r mm	r1 mm	a mm	Basic capacity,* N Static C_0	Dynamic C	Max permissible speed rev/min
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)
15BA02	7202 B	15	35	11	1	0.5	16	3680	6080	13000
17BA02	03 B	17	40	12	1	0.8	18	4460	7700	13000
20BA02	04 B	20	47	14	1.5	0.8	21	6370	10150	10000
25BA02	7205 B	25	52	15	1.5	0.8	24	7700	11280	10000
30BA02	06 B	30	62	16	1.5	0.8	27	10790	15400	10000
35BA02	07 B	35	72	17	2	1	31	14710	20590	8000
40BA02	7208 B	40	80	18	2	1	34	18490	24520	8000
45BA02	7209 B	45	85	19	2	1	37	21180	27655	6000
50BA02	10 B	50	90	20	2	1	39	23140	28440	6000
55BA02	7211 B	55	100	21	2.5	1.2	43	29175	36285	6000
60BA02	12 B	60	110	22	2.5	1.2	47	36285	42900	5000
65BA02	13 B	65	120	23	2.5	1.2	50	42410	49030	5000
70BA02	7214 B	70	150	28	3	1.5	53	44620	52470	5000
75BA02	15 B	75	130	25	2.5	1.2	56	49030	54430	4000
80BA02	16 B	80	140	26	3	1.5	59	55650	60800	4000
85BA02	7217 B	85	150	28	3	1.5	64	63740	69630	4000
90BA02	18 B	90	160	30	3	1.5	67	75760	81400	4000
95BA02	19 B	95	170	32	3.5	2	71	85810	92670	3000
100BA02	7220 B	100	180	34	3.5	2	76	90710	98070	3000
105BA02	21 B	105	190	36	3.5	2	80	101500	106890	2500
110BA02	22 B	110	220	38	3.5	2	84	112780	117689	2500

Table 18.7(e): Single Row Angular Contact Ball Bearings (Contd.) (Series 73 B)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)
17BA03	7303 B	17	47	14	1.5	0.8	21	7110	11283	10000
20BA03	04 B	20	52	15	2	1	23	8140	13580	10000
25BA03	05 B	25	62	17	2	1	27	12260	18930	10000
30BA03	7306 B	30	72	19	2	1	31	15840	24025	8000
35BA03	07 B	35	80	21	2.5	1.2	35	20005	28050	8000
40BAO3	08 B	40	90	23	2.5	1.2	39	24910	34715	6000
45BA03	7309 B	45	100	25	2.5	1.2	43	33340	44620	6000
50BA03	10 B	50	110	27	3	1.5	47	40210	51485	6000
55BA03	11 B	55	120	29	3	1.5	52	46580	59820	5000
60BA03	7312 B	60	130	31	3.5	2	55	53450	69630	5000
65BA03	13 B	65	140	33	3.5	2	60	61290	78450	5000
70BA03	14 B	70	150	35	3.5	2	64	72570	87280	4000
75BA03	7315 B	75	160	37	3.5	2	68	80415	96600	4000
80BA03	16 B	80	170	39	3.5	2	72	89240	102970	4000
85BA03	17 B	85	180	41	4	2	76	98070	11150	3000
90BA03	7318 B	90	190	43	4	2	80	111500	120130	3000
95BA03	19 B	95	200	45	4	2	84	122580	129250	2500
100BA03	20 B	100	215	47	4	2	90	149060	144650	2500
105BA03	7321 B	105	225	49	4	2	94	160090	153230	2500
110BA03	22 B	110	240	50	4	2	99	188780	168920	2500

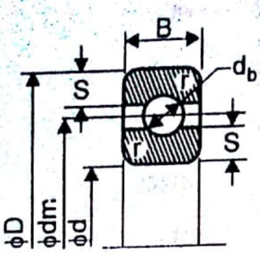
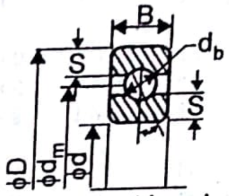
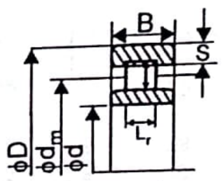
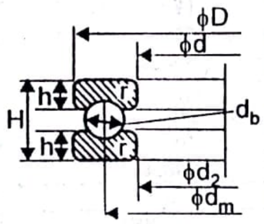
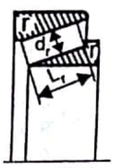
Table 18.7(f): Double Row Angular Contact Ball Bearings (Series 33 A)

Bearing No. (SKF)	d mm	D mm	B mm	r mm	Basic capacity,* N Static C_0	Dynamic C	Max. permissible speed, rev/min
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
3302 A	15	42	19.0	1.5	9070	13730	10000
03 A	17	47	22.2	1.5	12650	18930	8000
04 A	20	52	22.2	2	13730	18930	8000
3305 A	25	62	25.4	2	19615	26085	6000
06 A	30	72	30.2	2	27165	35300	6000
07 A	35	80	34.9	2.5	35600	43640	5000
3308 A	40	90	36.5	2.5	44620	53450	5000
09 A	45	110	39.7	2.5	54430	62270	4000
10 A	50	110	44.4	3	72570	80170	4000
3311 A	55	120	49.2	3	78450	85910	4000
12 A	60	130	54.0	3.5	94530	98070	3000
13 A	65	140	58.7	3.5	108950	115720	3000
3314 A	70	150	63.5	3.5	12615	135820	3000
15 A	75	160	68.3	3.5	138080	140235	2500
16 A	80	170	86.3	3.5	153965	157890	2500
3317 A	85	180	73.0	4	173580	173580	2500
18 A	90	190	73.0	4	205940	200150	2500

Table 18.7(g): Cylindrical Roller Bearings (Series NU. 22)

Bearing No. (SKF)	d mm	D mm	B mm	r mm	r1 mm	F mm	Basic capacity, N* Static C_0	Dynamic C	Max. permissible Speed, rev/min
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
NU 2205	25	52	18	1.5	1	32	11030	15790	13000
2206	30	62	20	1.5	1	38.5	16970	23140	13000
2207	35	72	23	2	1	43.8	27655	35600	10000
NU 2208	40	80	23	2	2	50	32750	40700	10000
2209	45	85	23	2	2	55	35600	43640	8000
2210	50	90	23	2	2	60.4	38540	45500	8000
NU 2211	55	100	25	2.5	2	66.5	45500	52560	8000
2212	60	110	28	2.5	2.5	73.5	59820	696300	6000
2213	65	120	31	2.5	2.5	79.6	74040	81640	6000
NU 2214	70	125	31	2.5	2.5	84.5	78450	85910	5000
2215	75	130	31	2.5	2.5	88.5	84730	96350	5000
2216	80	140	33	3	3	95.3	98070	109340	5000
NU 2217	85	150	36	3	3	101.8	117680	127000	4000
2218	90	160	40	3	3	107	135820	140235	4000
2219	95	170	43	3.5	3.5	113.5	160340	173580	4000
NU 2220	100	180	46	3.5	3.5	120	184855	196130	3000
2222	110	200	53	3.5	3.5	132.5	226435	254170	3000
2224	120	215	58	3.5	3.5	143.5	271645	280470	3000
NU 2226	130	230	64	4	4	156	306700	296650	2500
2228	140	250	68	4	4	169	378540	362850	2500
2230	150	270	73	4	4	182	436400	423650	2500

Table 18.8: Geometric Relations: (General proportions for Anti friction Bearings)

 Deep groove ball bearing	$d_m = \frac{D+d}{2}$ $d_b = 0.3\,(D-d)$ $s \approx 0.15\,(D-d) = \frac{d_b}{2}$ $r_1 = 0.5r$ $r = 0.515\,d_b$ $z \approx 2.9\left(\frac{D+d}{D-d}\right)$ d_m, mean diameter; s, radial thickness of races; d_b, diameter of ball; z, no. of balls	 Angular contact bearing	$d_m = \frac{D+d}{2}$ $d_b = 0.3\,(D-d)$ $s = 0.15\,(D-d)$ $z = 4.2\left(\frac{D+d}{D-d}\right)$ $\beta = 25°$
		 Roller bearing	$d_m = \frac{D+d}{2}$, $d_r = 0.25\,(D-d)$; $L_r = 1$ or $1.5d$; $s = 0.15\,(D-d)$
 Thrust bearing	$d_m = \frac{D+d}{2}$; $d_b \approx 0.5H$; $h \approx 0.3\,H$; $r_1 = 0.5r$; $z \approx 3.56\left(\frac{D+d}{D-d}\right)$	 Taper roller bearing	$d_r = 0.25\,(D-d)$ $L_r = 1.6\,d_r$

18.13. SOLVED PROBLEMS :

Problem 18.1:

A bearing for an axial flow compressor is used to support a radial load of 2500 N and thrust load of 1500 N. The service required for the bearing is 5 years at the rate of 40 hours per week. The speed of the shaft is 1000 rpm. Select suitable ball bearing for the purpose. Diameter of the shaft is 50 mm.

(Madras University)

Solution :

Given	Radial load, F_r	=	2500 N
	Axial load, F_a	=	1500 N
	Life	=	5 years at the rate of 40 hours/week
	Speed , n	=	1000 rpm.
	Diameter of shaft	=	50 mm

Since the bearing is subjected to radial and axial loads, angular contact ball bearing is preferred than deep groove ball bearing. Let us select single row angular contact ball bearing.

Generally the selection of bearing is based on its life, load applied on it, service conditions etc. Here the selection of bearing is based on its "Dynamic Load Capacity" which includes all the above parameters like life, load etc.

Now, Dynamic load capacity of the bearing is given by the expression as

$$C = \left(\frac{L}{L_{10}}\right)^{\frac{1}{k}} \cdot P = \left(\frac{L}{L_{10}}\right)^{1/k} \times (V\,X\,F_r + Y\,F_a)\,S$$

where
- P = Equivalent load
- V = Race rotation factor
  = 1 (Assuming inner ring rotates and outer ring stationary)
- X = Radial factor
- F_r = Radial load = 2500 N
- F_a = Axial load = 1500 N
- k = Exponent = 3 (for ball bearing)
- S = Service factor = 1.5 (for compressor)

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To find out the values of "X" and "Y", compare the ratio of $\left(\frac{F_a}{F_r}\right)$ to the values of "e"

For the available single row angular contact ball bearings 72B and 73B, the "e" value = 1.14 (from table 18.4) (PSG 4.4, JDB 20.8)

$$\frac{F_a}{F_r} = \frac{1500}{2500} = 0.6 \le e\,(1.14)$$

Hence X = 1, Y = 0 (from table 18.4)

$$\therefore P = (1 \times 1 \times 2500 + 0 \times 1500)\ 1.5 = 3750\ \text{N}$$

L = Life in million revolutions

$$= \frac{\text{Life in hours} \times n \times 60}{10^6} = \frac{(5 \times 52 \times 40) \times (1000 \times 60)}{10^6}$$

$$= \frac{10400 \times 1000 \times 60}{10^6} = 624 \text{ million revolutions}$$

L_{10} = 1 million revolutions.

$$\therefore C = \left(\frac{624}{1}\right)^{1/3} \times 3750 = 32050\ \text{N}$$

For the shaft diameter of 50 mm, either 7210 B or 7310 B SKF ball bearing (from tables 18.7(d), (e)) may be selected. The 7210B bearing is having the dynamic capacity of 28440 N and 7310B is having the dynamic capacity of 51485 N. Since the selected bearing should have higher value of dynamic load capacity than the calculated value (32050 N due to given conditions), **7310B SKF** Ball bearing may be selected for the purpose.

Another method of finding out the bearing using graph as shown in figure 18.3 (PSG 4.6, JDB 20.11)

Calculate the equivalent Load P using

$$P = (V\,X\,F_r + Y\,F_a)\,S = 3750\ \text{N (As before)}$$

Find the life in hours as

Life in hours $= 5 \times 52 \times 40 = 10400$ hours

Then using the graph, (Fig. 18.3) find $\frac{C}{P}$ ratio for the given life and rpm as

$\frac{C}{P} = 8.56$ (for 10400 hours and 1000 rpm)

$\therefore C = 8.56 \times 3750 = 32100$ N

Hence SKF 7310 B Angular Contact Ball Bearing may be selected.

Problem 18.2:

Select a suitable deep groove ball bearing for supporting a radial load of 10 KN and an axial load of 3 KN for a life of 4000 hours at 800 rpm. Select from series 63. Calculate the expected life of the selected bearing.

(M.K. University)

Solution :

Radial load, $F_r = 10$ KN $= 10000$ N

Axial load, $F_a = 3$KN $= 3000$ N

Life required $= 4000$ hours

Speed, $n = 800$ rpm

Required bearing should be of SKF63 series. Since the desired life is 4000 hours, the selected bearing should have more life than than or atleast equal to 4000 hours. Hence we may select any one of the bearings from SKF63 series arbitrarily and compare its life with the required life.

Let us select SKF 6312 bearing, For this bearing the static capacity, $C_0 = 47070$ N

and the dynamic capacity, $C = 62270$ N (Table 18.8(b)), (JDB 20.17)

Now $\frac{F_a}{C_0} = \frac{3000}{47070} = 0.06$ and corresponding 'e' value $= 0.26$ (From table 18.4)

(PSG 4.4, JDB 20.8)

and $\frac{F_a}{F_r} = \frac{3000}{10000} = 0.3$

Since $\frac{F_a}{F_r} > e$, the radial and axial load factors are, $X = 0.56$, $Y = 1.65$ (From table 18.4)

The equivalent load, $P = (V X F_r + Y F_a) S$

Let V = 1 (Assume inner ring rotation)

S = 1.5

Then $P = (1 \times 0.56 \times 10000 + 1.65 \times 3000)\ 1.5 = 15825$ N

Life of this bearing,

$$L = \left(\frac{C}{P}\right)^k = \left(\frac{62270}{15825}\right)^3 = 61 \text{ million revolution}$$

$$= \frac{61 \times 10^6}{60 \times n} = \frac{61 \times 10^6}{60 \times 800} = 1270 \text{ hours}$$

Since the life of this bearing is less than our requirement, let us select a higher capacity bearing.

Now let the bearing may be SKF 6316 bearing.

For this bearing, $C_0 = 78450$ N and $C = 94140$ N

Now $\frac{F_a}{C_0} = \frac{3000}{78450} = 0.04$ and the corresponding $e = 0.24$

Since $\frac{F_a}{F_r} = 0.3 > e$, $X = 0.56$ and $Y = 1.8$

$\therefore P = (0.56 \times 10000 + 1.8 \times 3000)\ 1.5 = 16500$ N

Now life, $L = \left(\frac{C}{P}\right)^k = \left(\frac{94140}{16500}\right)^3 = 185.73$ m.r.

$$= \frac{185.73 \times 10^6}{60 \times 800} = 3870 \text{ hours}$$

Now also, the life of selected bearing is little bit lower, let us try for the life of next bearing, i.e., SKF 6317 bearing. For this bearing.

$C_0 = 85810$ N and $C = 101500$ N

Now $\frac{F_a}{C_0} = \frac{3000}{85810} = 0.035$ and $e = 0.235$

Also $\frac{F_a}{F_r} = 0.3 > e$ and hence $X = 0.56$, $Y = 1.85$

Then $P = (0.56 \times 10000 + 1.85 \times 3000)\ 1.5 = 16725$ N

$$\text{Life, } L = \left(\frac{C}{P}\right)^k = \left(\frac{101500}{16725}\right)^3 = 223.5 \text{ m.r}$$

$$= \frac{223.5 \times 10^6}{60 \times 800} = 4656 \text{ hours}$$

Since the life of **SKF 6317** bearing is more than required life, this bearing is selected for the purpose.

Problem 18.3:

The radial reaction on a bearing is 9000 N. It also carries a thrust of 5000 N. The speed of shaft is 1000 rpm. The outer ring stationary. Expected average life of bearing is about 25000 hours. The load on the bearing is smooth, the service is 8 hours/day.

1. **Select a suitable roller bearing.**
2. **What is the rated 90% life of selected bearing.**
3. **Compute the probability of the selected bearing surviving 25000 hours.** **(Anna University)**

Solution :

Radial load, $F_r = 9000$ N

Axial (Thrust) load, $F_a = 5000$ N

Speed, $n = 1000$ rpm

Expected average life $= 25000$ hours

Since the bearing is to support a high radial and axial loads, taper roller bearings are preferred which are denoted by SKF 322 series. Also a suitable taper roller bearing is selected by trial and error method similar to previous problem. That is, by selecting a particular bearing, its life is compared with the required life. If it is not suitable, then another bearing is selected and so on.

It is given that, expected average life $= 25000$ hours.

$\therefore$ Nominal (or rated) life $= \frac{25000}{5} = 5000$ hours. ($\because$ Average life $= 5 \times$ nominal life)

Now, let us consider 32212 bearing. (Refer table 18.7(h)), (JDB 20.29)

For this bearing $C_0 = 75710$ N, $C = 78450$ N and $e = 0.41$

$\frac{F_a}{F_r} = \frac{5000}{9000} = 0.55 > e$ and hence $X = 0.4$, $Y = 1.45$ (From table 18.4) (PSG 4.4)

Now $P = (V X F_r + Y F_a) S$

Assume $V = 1$ and $S = 1.2$ for smooth operation and inner ring rotation.

Then $P = (0.4 \times 9000 + 1.45 \times 5000)\ 1.2 = 13020$ N

$$\text{Now life } L = \left(\frac{C}{P}\right)^k = \left(\frac{78450}{13020}\right)^{10/3} = 398 \text{ m.r}$$

$$= \frac{398 \times 10^6}{60 \times n} = \frac{398 \times 10^6}{60 \times 1000} = 6634 \text{ hours}$$

Let us try for a slightly lower capacity bearing, since the required life is around 5000 hours.

Select 32211 bearing

For this bearing, $C_0 = 61050$, $C = 65115$ and $e = 0.41$

$$\text{Since } \frac{F_a}{F_r} = \frac{5000}{9000} = 0.55 > e, \quad X = 0.4, Y = 1.45$$

$\therefore P = (0.4 \times 9000 + 1.45 \times 5000)\ 1.2 = 13020$ N

$$\text{Now, life, } L = \left(\frac{C}{P}\right)^k = \left(\frac{65115}{13020}\right)^{\frac{10}{3}} = 214 \text{ m.r.}$$

$$= \frac{214 \times 10^6}{60 \times 1000} = 3565 \text{ hours}$$

Since this life is less than required life, the previous 32212 taper roller bearing can be selected.

Its rated 90% life is known as L_{10} life or nominal life which is nothing but what we have calculated before.

i.e., Rated 90% life $= 6634$ hours.

The expected life of selected bearing = Rated Life × 5

$= 6634 \times 5 = 33170$ hours.

Probability of selected bearing surviving 25000 hours :

We know that the relationship between the lives and the probabilities of survival is given by the expression.

$$\frac{L}{L'_{10}} = \left[\frac{l_n(1/p)}{l_n(1/p_{10})}\right]^{1/b} \qquad \text{(PSG4.2, JDB 20.3)} \qquad \text{.....(A)}$$

where L = Required life in million revolution

$$= \frac{25000 \times 60 \times n}{10^6} = \frac{25000 \times 60 \times 1000}{10^6} = 1500 \text{ m.r}$$

L'_{10} = Calculated life of the selected bearing at 90% survival in million revolution.

$$= \frac{33170 \times 60 \times 1000}{10^6} = 1990 \text{ m.r.}$$

b = 1.17 for a median life 5 L_{10}

$$l_n\left(\frac{1}{p_{10}}\right) = l_n\left(\frac{1}{0.9}\right) = 0.1053$$

Now substituting the above values in the expression (A) we get,

$$\frac{1500}{1990} = \left[\frac{l_n\left(\frac{1}{p}\right)}{0.1053}\right]^{\left(\frac{1}{1.17}\right)}$$

$$\therefore l_n\left(\frac{1}{p}\right) = \left(\frac{1500}{1990}\right)^{1.17} \times 0.1053 = 0.076$$

$$\frac{1}{p} = e^{0.076} = 1.08, \quad \therefore p = \frac{1}{1.08} = 0.93$$

Probability of survival for the selected bearing = 93%

Problem 18.4:

A 6207 radial bearing is to operate in the following work cycle.

Radial load of 4500 N at 150 rpm for 30% of time

Radial load of 6750 N at 600 rpm for 10% of time

Radial load of 2250 N at 300 rpm for 60% of time

The inner-ring rotates, loads are steady, what is the expected average life of the bearing. (Madras University)

SOLUTION :

Since the loads are variable with time, we should find out cubic mean load to have an equivalent effect produced by the above variable loads on the bearing.

To evaluate the cubic mean load, the following method is adopted.

Consider the total period of operation as 1 minute. For this period of time, the number of revolutions and the corresponding operating loads are tabulated as follows.

Load (P) in N	Cycle time ratio	Speed in rpm.	Number of revolutions
(1)	(2)	(3)	(4) = (2) × (3)
4500 (P_1)	0.3 = (30%)	150	45 (n_1)
6750 (P_2)	0.1 = (10%)	600	60 (n_2)
2250 (P_3)	0.6 = (60%)	300	180 (n_3)
			285 (Σ n)

Cubic mean load can be obtained by using the relation as

$$P_m = \left[\frac{P_1^3 n_1 + P_2^3 n_2 + P_3^3 n_3}{\Sigma n}\right]^{1/3} \quad \text{(PSG 4.2, JDB 20.3)}$$

$$= \left[\frac{(4500^3 \times 45) + (6750^3 \times 60) + (2250^3 \times 180)}{285}\right]^{1/3} = 4420 \text{ N} = (P)$$

Dynamic capacity (C) of 6207 bearing is 19615 N (Refer table 18.7(a))

Then, rated L_{10} life of bearing

$$L = \left(\frac{C}{P}\right)^k = \left(\frac{19615}{4420}\right)^3 = 87.4 \text{ million revolutions.}$$

$$\text{Life in hours} = \frac{\text{Life in million revolutions} \times 10^6}{60 \times n}$$

$$= \frac{87.4 \times 10^6}{60 \times 285} = 5110 \text{ hours}$$

Expected life of bearing $= 5 \times \text{Rated life} = 5 \times 5110 = 25550$ hours (Answer)

Problem 18.5:

For 6307 ball bearing, the load varies as follows.

	Radial load N	Axial load N	Cycle Time ratio	Speed in rpm
1.	6000	3000	0.5	400
2.	7500	–	0.3	650
3.	4000	1000	0.2	900

The inner – ring rotates, loads are steady. Find the expected average life of the bearing.

(Bangalore University)

Solution :

First the equivalent loads are determined for the first, second the third cycle time ratios.

For 6307 ball bearing $C_0 = 16970$ N and $C = 25300$ N

To find equivalent load for first operating period : (P_1)

$$\frac{F_{a1}}{C_0} = \frac{3000}{16970} = 0.18 \text{ and corresponding "e" value } = 0.33$$

(Table 18.4) (By interpolation)

$$\text{And } \frac{F_{a1}}{F_{r1}} = \frac{3000}{6000} = 0.5 > e \text{ and hence } X = 0.56,\ Y = 1.3$$

Now the equivalent load, $P_1 = (V X F_{r1} + Y F_{a1})\, S$

Assuming inner ring rotation, and smooth loading, take $V = 1$ and $S = 1.2$

Then $P_1 = [\,(0.56 \times 6000) + (1.3 \times 3000)\,]\ 1.2 = 8712$ N.

To find second operating equivalent load P_2 :

Since $F_{a2} = 0$, We get

$$P_2 = F_{r2} \times S = 7500 \times 1.2 = 9000 \text{ N}$$

To find third operating equivalent load P_3 :

$$\frac{F_{a3}}{F_{r3}} = \frac{1000}{4000} = 0.25 \text{ and } \frac{F_{a3}}{C_0} = \frac{1000}{16970} = 0.06 \text{ and } e = 0.26$$

$$\therefore \frac{F_{a3}}{F_{r3}} < e,\ X = 1, \text{ and } Y = 0$$

$\therefore P_3 = [\,(1 \times 4000) + (0 \times 1000)\,]\ 1.2 = 4800$ N.

All the equivalent loads and the corresponding operating periods are tabulated as follows in order to find out cubic mean load.

Load (P) in "N" 1.	Cycle time ratio 2.	Speed in rpm. 3.	No. of revolution for 1 minute cycle $4 = 2 \times 3$
8712 (P_1)	0.5	400	200 (n_1)
9000 (P_2)	0.3	650	195 (n_2)
4800 (P_3)	0.2	900	180 (n_3)
		Total	575 (Σ n)

Now, the cubic mean load

$$P_m = \left[\frac{P_1^3 n_1 + P_2^3 n_2 + P_3^3 n_3}{n_1 + n_2 + n_3}\right]^{1/3}$$

$$= \left[\frac{(8712^3 \times 200) + (9000^3 \times 195) + (4800^3 \times 180)}{575}\right]^{1/3} = 8000 \text{ N}$$

Rated life of bearing,

$$L = \left(\frac{C}{P}\right)^k \text{ million revolutions}$$

$$= \left(\frac{25300}{8000}\right)^3 = 31.63 \text{ m.r} \quad (\text{k for ball bearing} = 3)$$

$$= \frac{31.63 \times 10^6}{60 \times \text{rpm}} = \frac{31.63 \times 10^6}{60 \times 575} = 917 \text{ hours}$$

Expected average life of bearing

$= 5 \times$ Rated life $= 5 \times 917 = 4585$ hours (Answer)

Problem 18.6:

The operating schedule of a ball bearing is as follows.

Radial load of 1650 N at 2000 rpm for 5% of life time,

Radial load of 1140 N at 3300 rpm for 15% of time,

Radial load of 560 N at 1750 rpm for 35% of time and

Radial load of 445 N at 2200 rpm for 45% of the time.

The inner ring rotates and the loads are steady. The life is to be 10 years at 2 hours per day operation. Select a suitable ball bearing.

(Cochin University)

Solution:

The required life of the bearing in hours

$= 2 \times 300 \times 10 = 6000$ hours

(Assuming 300 working days per year)

Consider the period of operation as 1 minute. For this period the various acting loads and the number of revolutions are given in the following table.

Load (P) in N (1)	Cycle time ratio (2)	Speed in rpm (3)	Number of revolutions for 1 minute cycle. $4 = 2 \times 3$
1650 (P_1)	0.05 = (5%)	2000	100 (n_1)
1140 (P_2)	0.15 = (15%)	3300	495 (n_2)
560 (P_3)	0.35 = (35%)	1750	613 (n_3)
445 (P_4)	0.45 = (45%)	2200	990 (n_4)
			2198 (Σ n)

Now the cubic mean load

$$P_m = \left[\frac{P_1^3 n_1 + P_2^3 n_2 + P_3^3 n_3 + P_4^3 n_4}{\Sigma n}\right]^{1/3}$$

$$= \left[\frac{(1650^3 \times 100) + (1140^3 \times 495) + (560^3 \times 612.5) + (445^3 \times 990)}{2198}\right]^{1/3}$$

$$= 856 \text{ N} = (P)$$

Since the required bearing is a ball bearing, from graph shown in figure 18.3(a), the loading ratio $\left(\frac{C}{P}\right)$ for the life of 6000 hours and the speed of 2198 rpm is given by

$$\frac{C}{P} = 9.5$$

$\therefore C = 9.5 \times P = 9.5 \times 856 = 8132$ N

Since the required dynamic capacity is 8132 N, we should select a bearing having more capacity than required.

Since the diameter of shaft for fitting the bearing is not specified in the given problem, we can select either SKF 6006 bearing whose C = 10400 N or SKF 6204 whose C = 9805 N or SKF 6302 whose C = 8580 N

Let the bearing selected may be **6302** bearing C Value = 8580 N

The actual life of this bearing, $L = \left(\frac{C}{P}\right)^k = \left(\frac{8580}{856}\right)^3$ m.r = 1007 m.r

$$= \frac{1007 \times 10^6}{60 \times 2197.5} = \textbf{7640 hours (Answer)}$$

Problem 18.7 :

The radial load on a roller bearing varies as follows. A load of 50 KN is acting 20% of time at 500 rpm and a load of 40 KN is acting 50% of time at 600 rpm. In the remaining time, the load is varying from 40 KN to 10KN linearly at 700 rpm. Select a roller bearing from NU22 series for a life of atleast 4000 hours. The operating temperature is 175°C. (I.I.T, Bombay)

Solution :

The bearing is to be selected from NU22 series, ie; cylindrical roller bearing.

Life required ≥ 4000 hours.

Operating temperature of bearing $= 175°C$

The loading diagram is shown in figure 18.7.

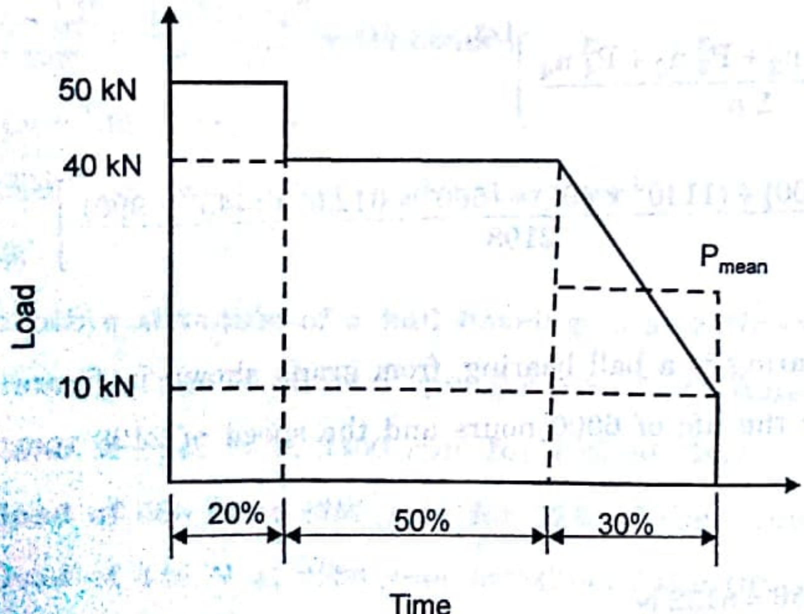

Fig. 18.7

For first 20% time, a steady load of 50 KN is acting. For the second 50% time, a steady load of 40 KN is acting. But in the remaining 30% of time, the load decreased from 40 KN to 10 KN linearly. Hence an equivalent steady load for the third period (i.e., for 30% of time) is obtained as

$$P_{mean} = \frac{P_{min} + 2\,P_{max}}{3} = \frac{10 + (2 \times 40)}{3} = 30 \text{ KN}$$

Now as usual, the loads and cycle time ratios are tabulated for 1 minute time of operation.

Load (P) in KN	Cycle time ratio	Speed in rpm	Number of revolutions for 1 minute cycle.
(1)	(2)	(3)	(4) = (2) × (3)
50 (P_1)	0.2 = (20%)	500	100 (n_1)
40 (P_2)	0.5 = (50%)	600	300 (n_2)
30 (P_3)	0.3 = (30%)	700	210 (n_3)
			610 (Σ n)

$$\text{Cubic mean load, } P_m = \left[\frac{P_1^3 n_1 + P_2^3 n_2 + P_3^3 n_3}{\Sigma n}\right]^{1/3}$$

$$= \left[\frac{(50^3 \times 100) + (40^3 \times 300) + (30^3 \times 210)}{610}\right]^{1/3}$$

$$= 39.5 \text{ KN} = 39500 \text{ N}$$

The loading ratio $\frac{C}{P}$ required for 4000 hours and 610 rpm is obtained from figure 18.3(b) (PSG 4.7, JDB 20.12), as

$$\frac{C}{P} = 4.5$$

$\therefore C = 4.5 \times P = 4.5 \times 39500 = 177750$ N

Since the bearing is operating at 175°C, the dynamic capacity needed for 175°C = 177750 N

Also since the decrease in dynamic capacity at 175°C is 15%, the dynamic capacity needed at room temperature $= \frac{177750}{0.85} = 209120$ N

Hence select NU2222 bearing whose C value is 254170 N which is more than minimum requirement. (From table 18.7(g), JDB 20.27)

The life of NU2222 bearing,

$$L = \left(\frac{C}{P}\right)^k = \left(\frac{254170 \times 0.85}{39500}\right)^{10/3} = (5.47)^{10/3} = 288 \text{ m.r}$$

$$= \frac{288 \times 10^6}{60 \times \text{rpm}} = \frac{288 \times 10^6}{60 \times 610} = 7870 \text{ hours} \qquad (\because k \text{ for roller bearing is } 10/3)$$

(or) From graph shown in figure 18.3 (b), life for 610 rpm and $\frac{C}{P}$ ratio of 5.47 is approximately 8000 hours.

Problem 18.8:

Select a suitable Conrad-type ball bearing for the following data. The radial load is 7500 N and axial load is 4500 N. The shaft speed is 2000 rpm and the L_{10} life required is 4.9×10^8 revolutions. The inner ring of the bearing rotates. **(Anna University, May 2004)**

Solution:

Given:

Radial load,	$F_r = 7500$ N
Axial load,	$F_a = 4500$ N
Speed of shaft,	$n = 2000$ rpm
L_{10} life required,	$L = 4.9 \times 10^8$ revolutions.

The selected bearing should be 'Conrad-type ball bearing' which is nothing but 'Deep-groove ball bearing.

Now, let us select the bearing from SKF 63 series. Consider SKF 6318 bearing.

For this bearing, $C_0 = 96110$ N; $C = 107870$ N

Now $\frac{F_a}{C_0} = \frac{4500}{96110} = 0.05$ and corresponding e value $= 0.25$

and $\frac{F_a}{F_r} = \frac{4500}{7500} = 0.6 > e\ (= 0.25)$

Hence $X = 0.56$; $Y = 1.74$ (From table 18.4)

The equivalent load; $P = (V X F_r + Y F_a) S$

Here $V = 1$ ($\because$ Inner ring rotates)

$S = 1.2$ (Assume as service factor)

Then, $P = (1 \times 0.56 \times 7500 + 1.74 \times 4500)\ 1.2 = 14436$ N

Life of this bearing,

$$L = \left(\frac{C}{P}\right)^k = \left(\frac{107870}{14436}\right)^3 = 417.2 \text{ m.r} = 4.17 \times 10^8 \text{ revolutions}$$

Since this bearing has life less than the requirement (i.e., 4.9×10^8 revolutions), let us select a higher capacity bearing.

Consider SKF 6319 bearing.

For this bearing, $C_0 = 107870$ N ; $C = 117680$ N

Now, $\frac{F_a}{C_0} = \frac{4500}{107870} = 0.04$ and corresponding $e = 0.24$

Since $\frac{F_a}{F_r} = 0.6 > e$; choose $X = 0.56$ and $Y = 1.8$ (From table 18.4)

$P = (1 \times 0.56 \times 7500 + 1.8 \times 4500)\ 1.2 = 14760$ N

Life of bearing, $L = \left(\frac{C}{P}\right)^k = \left(\frac{117680}{14760}\right)^3 = 506.8$ m.r

$= 5.068 \times 10^8$ revolutions

Since this bearing has more life than required life, SKF 6319 ball bearing may be selected.

18.14. SHORT QUESTIONS AND ANSWERS :

1. **Name a few applications of rolling contact bearings.**

   Rolling contact bearings are employed in automobiles, agricultural machineries, fans, motors, machine tools, air-crafts and so on.

2. **Why are rolling contact bearing called as anti-friction bearings?**

   Since the friction produced in rolling contact bearings is very low and almost negligible, these bearings are known as "Anti-friction bearings".

3. **In what ways, rolling contact bearings are differed from sliding contact bearings?**

   (a) The sliding contact bearings are made integral with parent machine whereas the rolling contact bearings are made separately and they can be assembled with the machine whenever needed.

   (b) The sliding contact bearings require more axial space and high maintenance cost due to specialised lubricating oil systems whereas the rolling contact bearings require less axial space and low maintenance cost due to simple oil systems.

4. **Specify the materials by which the rolling contact bearings are made?**

   Rolling contact bearings are made of high carbon chromium steel that is through hardened to Rockwell C 58 – 63.

5. How are rolling bearings designated?

Rolling bearings are designated indifferent methods by different manufacturers and committee members, but mainly based on their load carrying capacities.

According to Society of Automotive Engineers, (SAE), the bearings are classified as light series bearings which are started by the number 200, medium series bearings starting with the number 300 and heavy series bearings with 400 mm, in which the last two digits represent the bore diameter in multiples of 5 mm for example 312 represent medium duty bering whose bore diameter is 60 mm.

According to Anti-Friction Bearing Manufacturing Association (AFBMA), International standard organisation (ISO) and Indian Standards (IS) bearings are designated by their bore diameter followed by the type of bearings and type of duty. For example, 30 BC 02 represents the bearing of 30 mm bore diameter, BC type, light duty.

According to SKF (SKEFKO) company, bearings are designated by different numbers of their own method such as 62, 64, 72 B, NU 22, NA 49 and so on. For example, 6210 represents deep groove ball bearing whose bore diameter is 50 mm. i.e. here also, the last two digits represent the bore diameter in multiples of 5.

6. What factors should be considered when selecting rolling bearings?

a) Space availability
b) Type and amount of load,
c) Speed
d) Alignment
e) Environmental conditions.

7. What are the main components of rolling bearings?

The rolling bearings consist of four components such as (i) outer ring, (ii) inner ring, (iii) balls or rollers and (iv) retainers etc.

8. What is an anti-friction bearing?

A bearing, in which the friction developed between the relative rotating members such as shaft and sleeve is very low or almost nil is termed as anti-friction bearing. Since the above property exists in rolling contact bearings, they are called as anti-friction bearings.

9. Specify the merits of rolling contact bearings over sliding contact bearings?

1. Rolling contact bearings are made separately and hence they can be assembled or disassembled whenever required, but this facility is not provided in sliding contact bearings.
2. Less axial space, low starting frictions, ease of lubrication, easy replacement, operation at any inclined position, low power loss etc. are some other merits of rolling contact bearings.

10. Suggest one suitable material for the following giving the reasons.
a) Balls in ball bearing.
b) Ball bearing races.
c) Cages in ball bearings.

(a) & (b) : Balls and races (inner ring and outer ring) of ball bearings are made of high carbon chromium steel in order to have high compressive strength.

(c) Cages (i.e., separators) are normally made of low carbon steel (i.e., mild steel) to have flexible character so that they can be made into very thin strips.

11. State True or false.

(a) For high speed and light loading, ball bearings are preferred.

(b) For heavy axial and thrust loading, taper roller bearing are employed.

Answer : (a) True. (b) True

12. Name some of the companies where rolling bearings are manufactured?

a) SKEFKO India Ltd.
b) TATA Bearings.
c) Bi-metal Bearings.

13. Briefly explain about (a) Nominal life and (b) average life of the rolling bearings.

The nominal life of a rolling bearing is defined as the number of revolutions which the bearing is capable of enduring before the first evidence of fatigue, that is developed in the bearing material of either rings or rolling elements under dynamic load rating.

The average life of bearing is defined as the summation of all bearing lives in a series of life tests and is divided by the number of life tests. Usually this average life is approximately equal to five times the nominal life.

14. Indicate the influence of operating temperature on rolling bearing materials.

At elevated temperatures, the hardness of the bearing materials is reduced and thus their dynamic load carrying capacity is also reduced.

15. Write short notes on

(a) Basic static load rating

(b) Basic dynamic load rating

(a) Basic static load rating is the load acting on a non-rotating bearing under which a permanent deformation should appear in balls or raceways.

(b) Basic dynamic load rating is the actual or real load acting on the rolling contact bearing during running conditions. It is equivalent to a constant stationary load which a group of apparently identical ball bearing with stationary outer ring can endure for a rating life of one million revolutions of the innerring.

16. Define the equivalent load.

Equivalent load is defined as that constant stationary radial load,which, is applied to a bearing with rotating inner ring and stationary outer ring, would give the same life as that which the bearing would attain under the actual condition of load and rotation. This equivalent load can be expressed as

$P = (V X F_r + Y F_a) S$

where P = Equivalent load, V = Race rotation factor

F_r = Radial load, F_a = Axial load

X = Radial load factor, Y = Axial load factor

S = Service factor

18.15. REVIEW QUESTIONS AND EXERCISE PROBLEMS:

1. In what respects rolling contact bearings are preferred to sliding contact bearings?
2. Briefly explain the following.

   Ball bearings.

   Roller bearings.
3. Lay down the procedure of selection of antifriction bearings for a particular application.

4. Write short notes on

   (i) Static load rating

   (ii) Dynamic load rating

   (iii) Rated and average life or bearing.

   (iv) Influence of temperature on bearing capacity.

   (v) Selection of bearing based on variable loading.
5. Give a classification of Rolling contact bearings.
6. How are rolling bearings designated?
7. A ball bearing for a drilling machine spindle of 40 mm diameter is rotating at 3000 rpm. It is subjected to a radial load of 200 kg. and an axial thrust of 75 kg. It is to work at 45 hours per week for one year. Select and specify a suitable ball bearing.

   Select a suitable spherical roller bearing from SKF series 222C to support a radial load of 4 KN and an axial load of 2 KN. Minimum life required is 10,000 hours at 1000 rpm. For the selected bearing find.

   (a) The expected life under the given loads.

   (b) The equivalent load that can be supported for a life of 10000 hours.

   The load that can be supported with a probability of survival of 95% with 10,000 hours
8. Select a suitable antifriction bearing for the following requirements :

   Radial load on the bearing = 5000 N.

   Thrust load = 3000 N

   Speed of the shaft = 1000 rpm

   Expected life = 10,000 hours
9. The radial reaction on a bearing is 7000 N; it also carires a thrust of 5000 N; shaft rotates at 1500 rpm; outer ring stationary; smooth load 8 hrs/day service, say 15000 hours.

   Select a deep groove ball bearing.

   What is the rated 90% life of the selected bearing? (M.K. University)

10. From 6308 series, the load varies as follows.

Radial lond (kg)	Axial load (kg)	Cycle time ratio	Speed in rpm
600	300	0.5	400
750	–	0.25	630
400	100	0.25	1000

Find the expecting life of bearing. (Madras University)

11. A load on a radial bearing varies as follows. A load of 5000 kgf is acting 20% of time at 100 rpm. and load 4000 kgf is acting 50% of time at 50 rpm. In the resting percentage time at 200 rpm, the load varies from 4000 to 1000 kgf linearly. Select a roller bearing from NU 23 series for a life of at-least 4000 hours. The environment temperature is 150°C.

12. Select a suitable Conrad-type deep-groove ball bearing for the following data. The radial load is 7500 N and axial load is 4500 N. The shaft speed is 2000 rpm and the L_{10} life required is 7.2×10^8 revolutions. The inner ring of the bearing rotates.