Unit-1

BEARINGS

A bearing is a mechanical Element that permits relative motion between two parts, such as the shaft and the housing, with minimum friction.

The functions of the bearing are as follows:

i) The bearing ensures free rotation of the shaft or the axle with minimum friction.

ii) The bearing supports the shaft or the axle and holds it in correct position

iii) The bearing takes up the forces that act on the shaft or the axle and transmits them to the frame or the foundation.

Classification of Bearings:

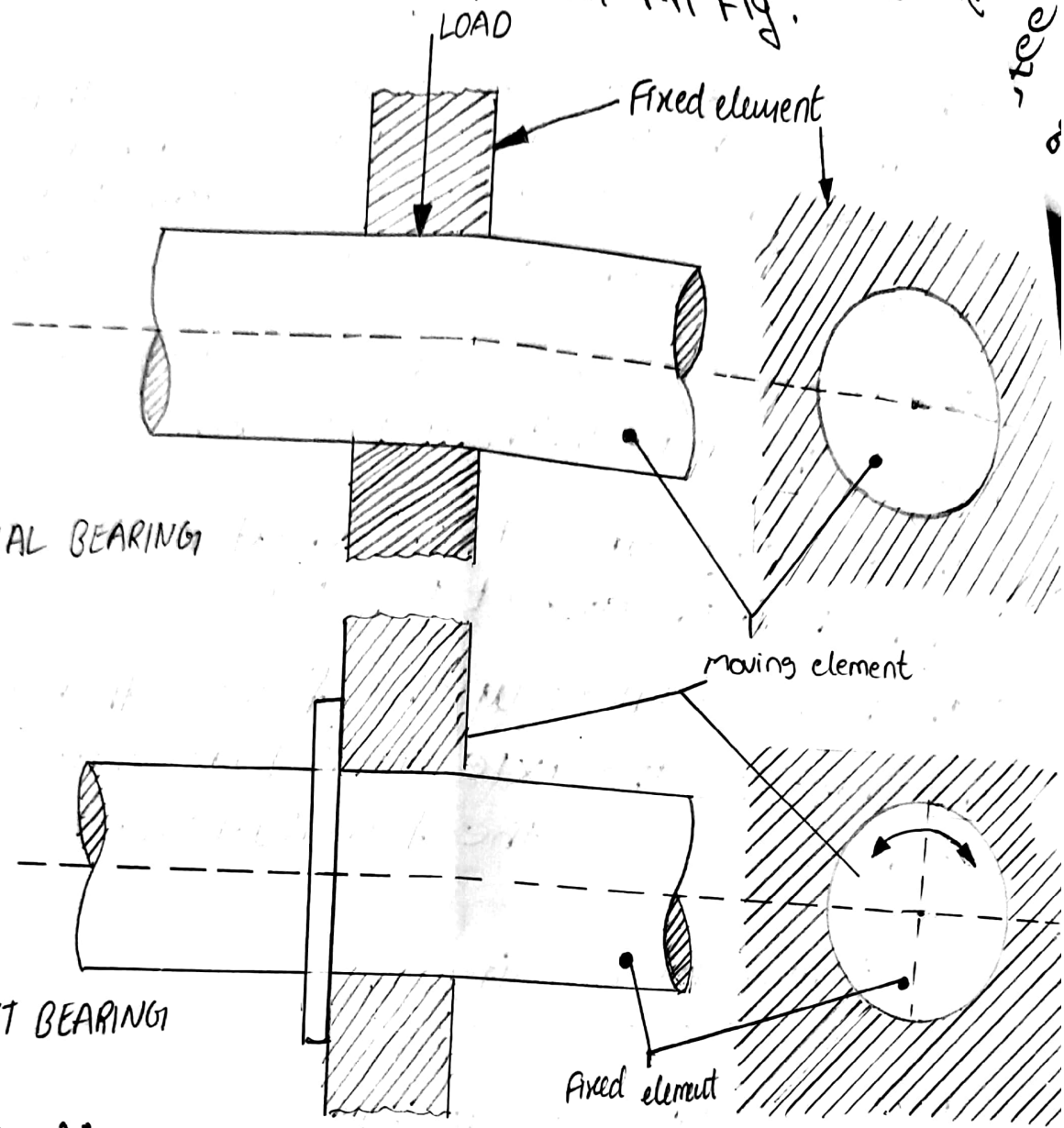
i) Depending upon the direction of load to be supported:

a) Radial bearings
b) Thrust bearings

In radial bearings, the load acts perpendicular to the direction of motion of moving Element as shown Fig

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In thrust bearings, the load acts along the axis of rotation as shown in Fig.



2. Depending upon the nature of contact:

a) Sliding contact bearings
b) Rolling contact bearings

In sliding contact bearings as shown in Figure, the sliding takes place along the surfaces of contact between the moving element and the fixed element. The sliding contact bearings are also known as plain bearings.

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(3)

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

Types of Sliding Contact Bearings:

The sliding contact bearings in which the sliding action is guided in a straight line and carrying radial loads, as shown in Fig. 26(a) may be called slipper or guide bearings such type of bearings are usually found in cross-head of steam engines.

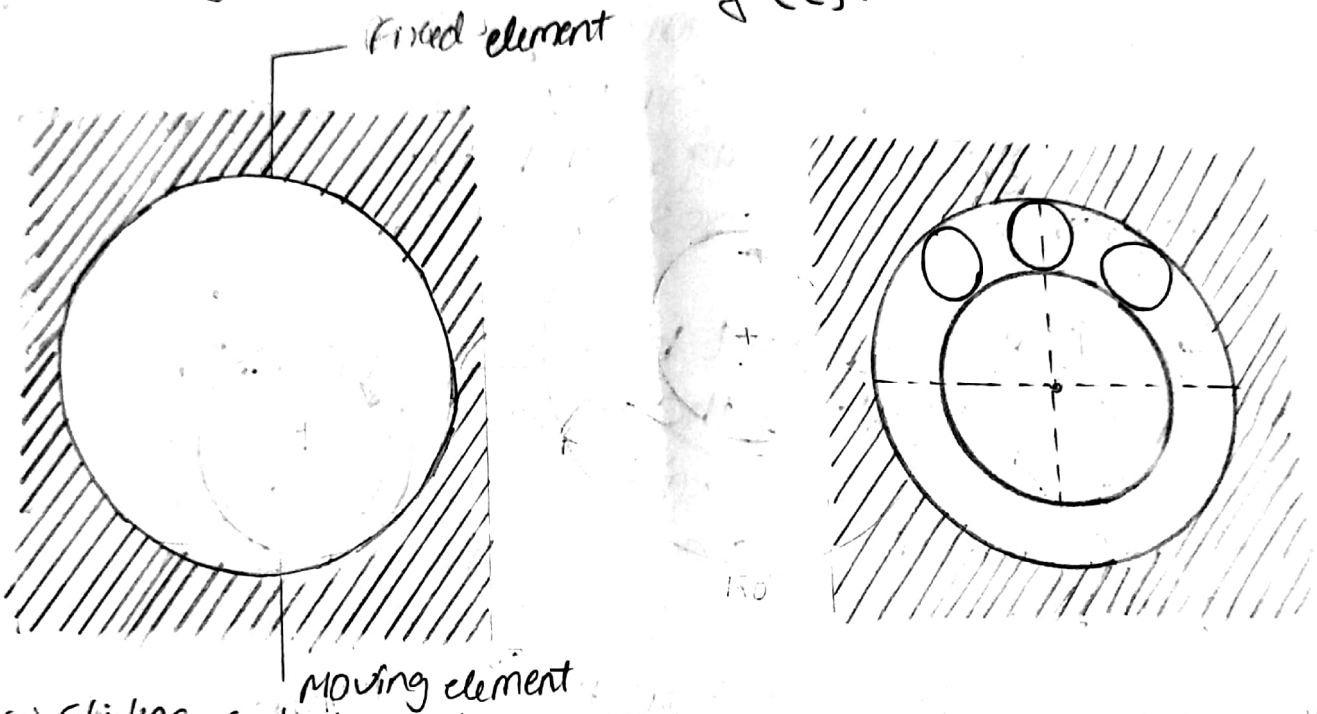
The sliding contact bearings in which the sliding action is along the circumference of a circle or an arc of a circle and carrying radial loads are known as journal or sleeve bearings. When the angle of contact of the bearing with the journal is 360° as shown in Fig (a) then the bearing is called full journal bearing. This type of bearing is commonly used in industrial machinery to accommodate bearing loads in any radial direction.

When the angle of contact of the bearing with the journal is 120°, as shown in Fig (b) then the bearing is said to be partial journal bearing. This type of bearing has less friction than full journal bearing, but it can be used only where the load is always in one direction. The most common application of the partial journal bearings is found in rail road car axles. The full and partial journal bearings

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may be called as clearance bearings because the diameter of the journal is less than that of bearing.

When a partial journal bearing has no clearance i.e. the diameters of the journal and the bearing are equal, then the bearing is called a fitted bearing as shown in Fig (c).



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

The sliding contact bearings, according to the thickness of layer lubricant between the bearing and the journal, may also be classified as follows.

1. Thick film bearings : The thick film bearings are those in which the working surfaces are completely separated from each other by the lubricant. Such type of bearings are also called as hydrodynamic lubricated bearing

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2. Thin film bearings: The thin film bearings are those in which although lubricant is present the working surfaces partially contact each other atleast part of the time. Such type of bearings are also called boundary lubricated bearings.

3. Zero film bearings: The zero film bearings are those which operate without any lubricant present.

4. Hydrostatic or externally pressurized lubricated bearings:

The hydrostatic bearing are those which can support steady loads without any relative motion between journal and the bearing. This is achieved by forcing externally lubricant between the members.

Rolling contact bearings:

For starting conditions and at moderate speeds, the frictional losses in rolling contact bearing are lower than that of an equivalent hydrodynamic journal bearing. This is because the sliding contact is replaced by rolling contact resulting in low coefficient of friction. Therefore, rolling contact bearings are called antifriction bearings.

The type of rolling contact bearing which are frequently used.

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i) Deep Groove Ball bearing : The most frequently used bearing is deep groove ball bearing. It is found in almost all kinds of products in general mechanical engineering. In this type of bearing, the radius of the ball is slightly less than the radii of curvature of the grooves in the races. Kinematically this gives a point of contact between the balls and the races. Therefore the balls and the races may roll freely without any sliding.

ii) Cylindrical Roller Bearing: When maximum load carrying capacity is required in a given space the point contact in ball bearing is replaced by the line contact of roller bearing. It consists of relatively short rollers that are positioned and guided by the cage.

iii) Angular contact Bearing : In angular contact bearing, the grooves in inner and outer races are so shaped, that the line of reaction at the contact between balls and races makes an angle with the axis of the bearing. This reaction has two components radial and axial therefore it can take radial and thrust loads.

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iii) Self-Aligning Bearings: There are two types of self-aligning rolling contact bearings, i.e self-aligning ball bearing and spherical roller bearing

[The principle of self-aligning] - The self-aligning ball bearing consists of two rows of balls, that roll on a common spherical surface in the outer race. In this case the assembly of the shaft, the inner race and the balls with cage can freely roll and adjust itself to the angular misalignment of the shaft.

There is similar arrangement in spherical roller bearing, where balls are replaced by two rows of spherical rollers that run on a common spherical surface in the outer race.

v) Taper Roller Bearing: The taper roller bearing consists of rolling elements in the form of frustum of a cone. They are arranged in such a way that the axes of individual rolling Elements intersect in a common apex point on the axis of the bearing. In kinematics analysis this is the essential requirement for pure rolling motion between conical surfaces.

vi) Thrust Ball Bearing: Thrust ball bearing consists of a row of balls running between two rings - the shaft ring and the housing ring. Thrust ball bearing carry thrust load in only one direction and cannot carry any radial load.

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Static load carrying capacity: Static load is defined as the load acting on the bearing when the shaft is stationary. It produces permanent deformation in balls and races, which increases with increasing load. The permissible static load, therefore, depends upon the permissible magnitude of permanent deformation.

It has been found that a total deformation of 0.0001 of the ball or roller diameter occuring at the most heavily stressed ball and race contact, can be tolerated in practice, without any disturbance like noise or vibrations. The static load carrying capacity of a bearing is therefore defined as the static load which corresponds to a total permanent deformation of balls and races at the most heavily stressed point of contact, equal to 0.0001 of the ball diameter.

Dynamic load carrying capacity: The life of a ball bearing is limited by the fatigue failure at the surfaces of balls and races. The dynamic load carrying capacity of the bearing is therefore based on the fatigue life of the bearing.

The life of an individual ball bearing is define as the number of revolutions (or hours of service at some given constant speed), that the bearing runs before the first evidence of fatigue crack in balls or races.

The rating life of a group of apparently identical ball bearings is defined as the number of revolutions that 90% of the bearings will complete or exceed befor the first evidence of fatigue crack.

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There are number of terms used for this rating life. They are minimum life, catalogue life, L_{10} life or Bio life

The life of an individual ball bearing may be different from rating life. Statistically it can be proved that the life, which 50% of a group of bearings will complete or exceed, is approximately five times the rating or L_{10} life. This means that for majority of bearings the actual life is considerably more than the rated life.

The dynamic load carrying capacity of a bearing is defined as the radial load in radial bearings (or thrust load in thrust bearings) that can be carried for a minimum life of one million revolutions. The minimum life [of one million revolutions] in this definition is the L_{10} life that 90% of the bearings will reach or exceed before fatigue failure.

Equivalent Bearing load: The equivalent dynamic load is defined as the constant radial load in radial bearings (or thrust load in thrust bearings) that if applied to the bearing would give same life as that which the bearing will attain under actual condition of forces. The expressions for the equivalent dynamic load is written as.

$$P = (XVF_r + YF_a)$$

Where

P = Equivalent dynamic load (N)

F_r = radial load (N)

F_a = axial or thrust load (N)

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V = race - rotation factor
S = Service factor
X & Y are radial and thrust factors respectively and their values are given in the manufacturer's catalogues. Assuming V as unity, the general Equation for equivalent dynamic load is given by

$$P = XF_r + YF_a$$

* When the bearing is subjected to pure radial load
i.e $P = F_r$

* When the bearing is subjected to pure thrust load
i.e $P = F_a$

Load Life Relationship:

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

$$L_{10} = \left(\frac{C}{P}\right)^K$$

Where

L_{10} = rated bearing life (in million revolutions)
C = dynamic load capacity (N)
K = 3 (for ball bearings)
= $\frac{10}{3}$ (for roller bearings)

from the above equation

$$C = P(L_{10})^{1/K}$$

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* For all type of ball bearings

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

* For all type of roller bearings

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

The relationship between life in million revolutions and life in working hours is given by

* $$\boxed{L_{10} = \frac{60nL_{10h}}{10^6}}$$

where

L_{10h} = rated bearing (hours)

n = speed of rotation

Selection of Bearing from manufacturer's catalogue

Step 1: Calculate the radial and axial forces acting on the bearing and determine the diameter of the shaft where the bearing is to be fitted.

Step 2: Select the type of bearing for the given application.

Step 3: Determine the values of X & Y, the radial and thrust factors from the catalogue. The values of X & Y factors for single-row deep groove ball bearings are taken from data book. The values depends upon two ratios $\left(\frac{F_a}{F_r}\right)$ and $\left(\frac{F_a}{C_o}\right)$.

C_o = static load capacity

Note: The required data is taken from data book.

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Step 4: Calculate the equivalent dynamic load from the equation.

$$P = XF_r + YF_a$$

Step 5: Make decision about the expected bearing life and express the life L_{10} in million revolutions

Step 6: Calculate the dynamic load capacity from the equation

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

Step 7: Check whether the selected bearing of series 60 has the required dynamic capacity If not select the bearing of the next series and go back to step (3) and continue.

Design for cyclic loads and speeds:

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

i) Radial load 2500 N at 700 rpm for 25% of the time

ii) Radial load 5000 N at 900 rpm for 50% of the time, and

iii) Radial load 1000 N at 450 rpm for the remaining 25% of the time.

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

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

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

* In one revolution, the life consumed is $\left(\frac{1}{L_1}\right)$ or

$$\frac{P_1^3}{C^3} \times \frac{1}{10^6}$$

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

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

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

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

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

$$\frac{N_1 P_1^3}{10^6 C^3} + \frac{N_2 P_2^3}{10^6 C^3} + \cdots + \frac{N_x P_x^3}{10^6 C^3} \quad \cdots\cdots (1)$$

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

$$\frac{N P_e^3}{10^6 C^3} \quad \cdots\cdots (2)$$

where $N = N_1 + N_2 + \cdots + N_x$

Equating expressions (1) & (2)

$$N_1 P_1^3 + N_2 P_2^3 + \cdots + N_x P_x^3 = N P_e^3$$

* $$P_e = \sqrt[3]{\left[\frac{N_1 P_1^3 + N_2 P_2^3 + \cdots}{N_1 + N_2}\right]}$$

(or)

* $$P_e = \sqrt[3]{\frac{\Sigma N P^3}{\Sigma N}}$$

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

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when the loads not vary in steps of constant magnitude, but varies continuously with time the above equation is modified and written as

*

$$P_e = \left[\frac{\int_0^N P^3 dN}{\int_0^N dN} \right]^{1/3}$$

or *

$$P_e = \left[\frac{1}{N} \int P^3 dN \right]^{1/3}$$

Bearing with a probability of survival other than 90%.

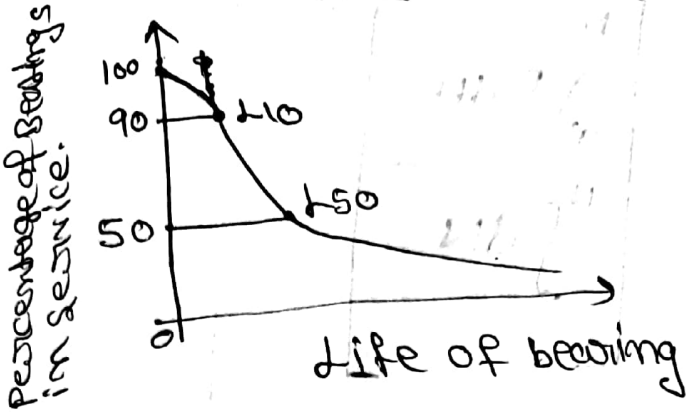
In the definition of rating life, it is mentioned that the rating life is the life that 90% of a group of identical bearings will complete or exceed before fatigue failure. The reliability R is defined as

$$R = \frac{\text{No. of bearings which have successfully completed 1 million revolutions}}{\text{Total number of bearings under test.}}$$

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

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The relationship between bearing life and reliability is given by a statistical curve known as Weibull distribution



For Weibull distribution

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

R = reliablity (in fraction)

L = corresponding life

a, b = constants.

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

* $$\log_e\left(\frac{1}{R}\right) = \left(\frac{L}{a}\right)^b \quad \text{---- (a)}$$

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

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

dividing Eq(a) by Eq(b) we have

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$$\left(\frac{d}{d_{10}}\right) = \left[\frac{\log_e\left(\frac{1}{R}\right)}{\log_e\left(\frac{1}{R_{90}}\right)}\right]^{1/b}$$

where $R_{90} = 0.9$

$a = 6.84$

$b = 1.17$.

Sliding Contact Bearings:

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

i) Liquid lubricants, like mineral or vegetable oils
ii) Semi-solid lubricants, like grease
iii) Solid lubricants, like graphite or molybdenum disulphide

The objectives of lubricant are as follows:

i) to reduce friction;
ii) to reduce or prevent wear;
iii) to carry away heat generated due to friction and
iv) to protect the journal and the bearing from corrosion

The basic modes of lubrication are thick and thin film lubrication. In addition, sometimes a term zero film bearing is used. Zero film bearing is a

bearing that operates without any lubricant i.e without any film of lubricating oil.

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

Thick film lubrication is further divided into two groups - hydrodynamic and hydrostatic lubrication

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

Principle of hydrodynamic lubrication in journal bearing: Initially the shaft is at rest and it sinks to the bottom of the clearance space under the action of load W. The surfaces of the journal and bearing touch during rest. As the journal starts to rotate, it will climb the bearing surface and the speed is further increased, it will force the fluid into wedge-shaped region. Since more and more fluid is forced into the wedge-shaped clearance space, pressure is generated within the system.

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the pressure distribution around the periphery of the journal: Since the pressure is created within the system due to rotation of the shaft, this type of bearing is known as self-acting bearing. The pressure generated in the clearance space supports the External load W. In this case, it is not necessary to supply the lubricant under pressure and the only requirement is sufficient and continuous supply of the lubricant. This mode of lubricant is seen in bearings mounted on engines and centrifugal pumps frequently, a term journal bearing is used.

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

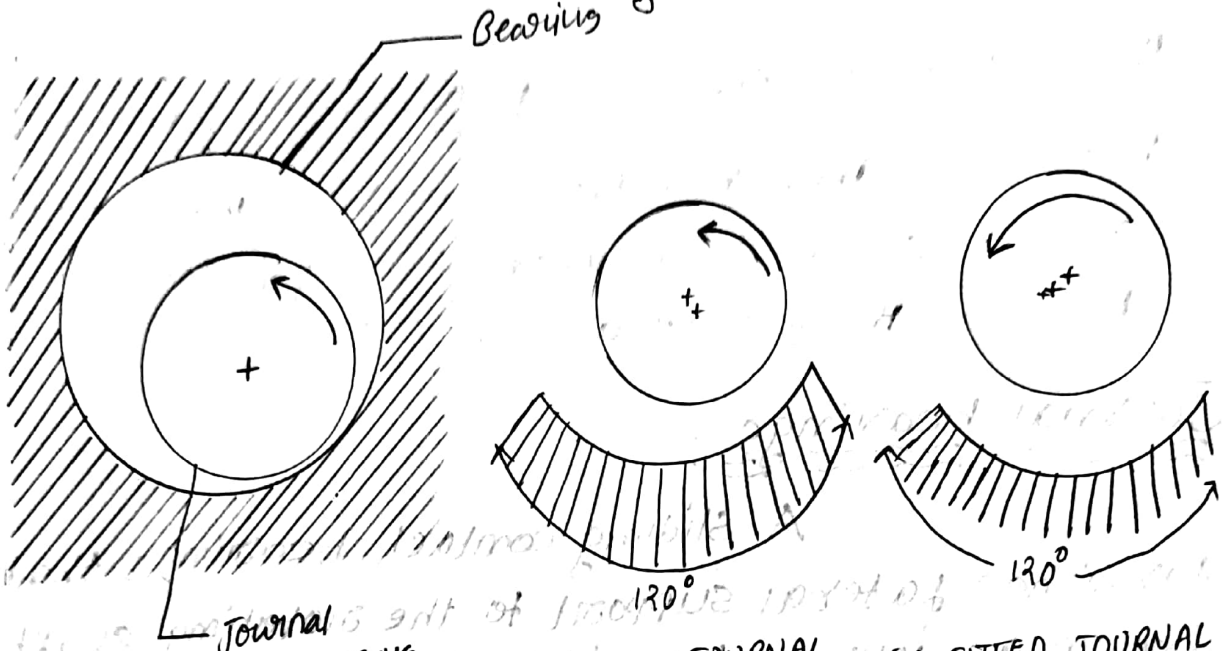
There are two types of hydrodynamic journal bearings namely, full journal bearing and partial bearing.

Full journal bearing: In full journal bearing the angle of contact of the busing with journal is 360°. Full journal bearing can take load in any radial direction. Most of the bearings used in industrial applications are full journal bearing.

Partial bearing: In partial bearings the angle of contanct between the bush and journal is always less than 180°. Most of the partial bearings in practice have 120° angle of contact. These bearings can take load in only one radial direction These bearings are used for rail-road-cars.

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Fitted bearing: When a partial journal bearing has no clearance i.e the diameters of journal and bearing are Equal then the bearing is called a fitted bearing.



(a) FULL JOURNAL BEARING (b) PARTIAL JOURNAL BEARING (c) FITTED JOURNAL BEARING

The sliding contact bearing are also be classified based on thickness of layer of the lubricant between the bearing and the journal. They are

1. Thick film bearings: The thick film bearings are those in which the working surfaces are completely separated from Each other by the lubricant. Such type of bearings are also called hydrodynamic lubricated bearings.

2. Thin film bearings: These bearings are also called boundary lubricated bearings in which lubricant is present, the working surfaces partially contact Eachother atleast part of the time.

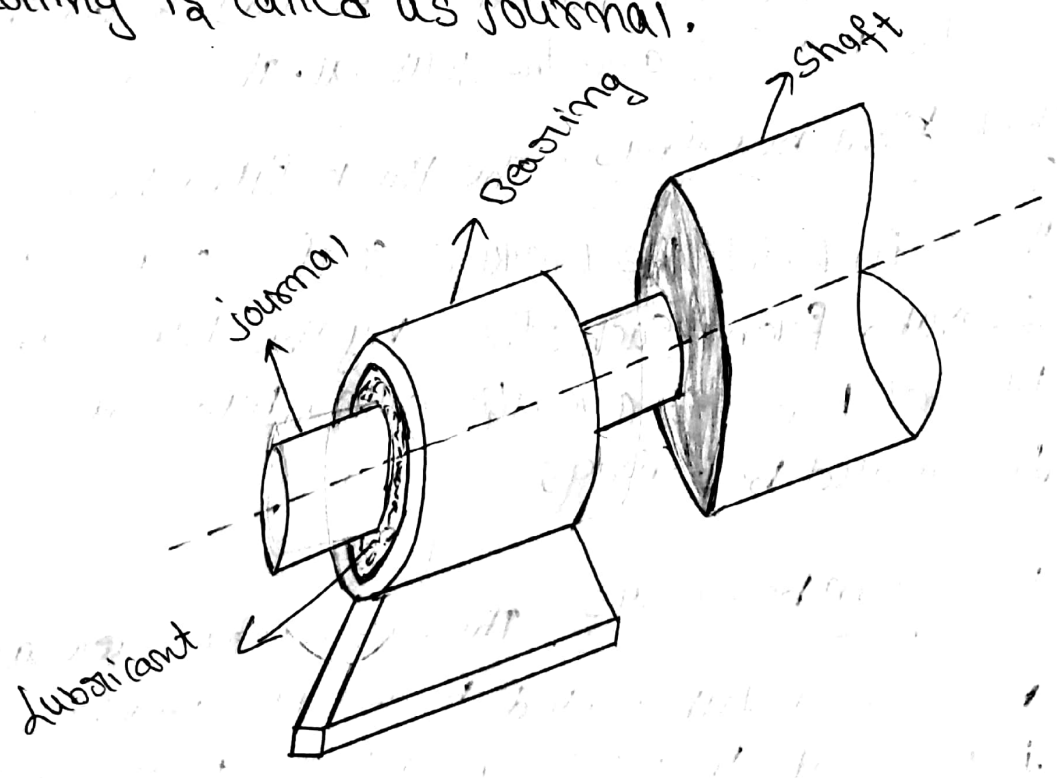
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3. Zero film bearings : The zero film bearings are those which operate without any lubricant present.

4. Hydrostatic or Externally pressurized lubricated bearings : These bearings are those which can support steady loads without any relative motion between the journal and bearing.

Journal bearings :

A sliding contact bearings that furnishes lateral support to the rotating shaft is known as journal bearing. It consists of two main parts, a shaft and a housing. The portion of the shaft inside the housing, also known as bearing is called as journal.

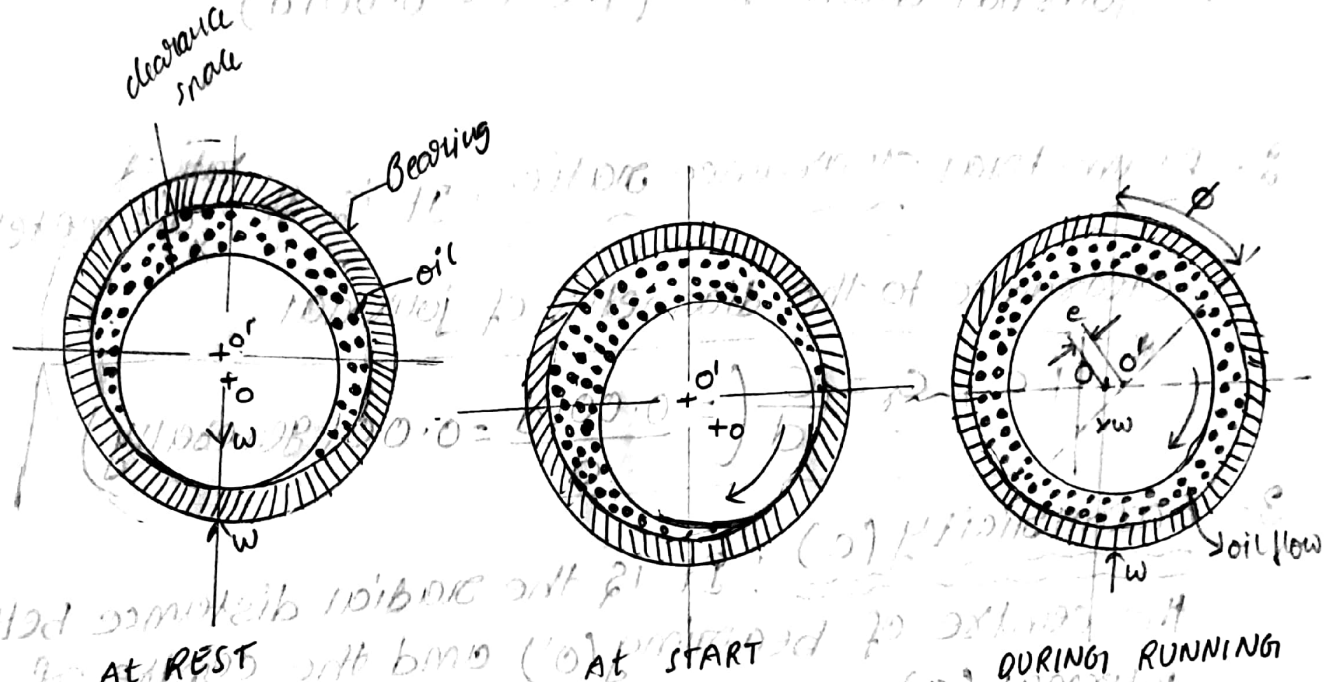


JOURNAL BEARING

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Terminology of Hydrodynamic journal Bearing:

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



Let O' be the centre of bearing
O be the centre of journal
D be the diameter of the bearing
d be the diameter of journal
l be the length of journal

1. Diametral clearance: It is the difference between the diameter of bearing and journal

* $\boxed{c = D - d}$

* Generally c may be assumed as $\frac{1}{1000}$ times of journal diameter (i.e $c = 0.001d$)

2. Diametral clearance ratio: It is the ratio of diametral clearance to the diameter of journal.

* $\boxed{\text{i.e., } C_r = \frac{c}{d} \left(= \frac{0.001d}{d} = 0.001 \text{ generally} \right)}$

3. Eccentricity (e): It is the radial distance between the centre of bearing (O') and the centre of journal (O)

* $\boxed{e = \text{distance } OO' = \frac{c}{2} - h_0}$

4. Minimum film thickness: It is the minimum distance between the bearing and the journal under complete lubrication conditions

* $\boxed{\text{i.e., } h_0 = \frac{c}{2} - e}$ where $\frac{c}{2}$ is called as radial clearance

. Attitude or eccentricity ratio (ε): It is the ratio of the eccentricity to the radial clearance

* $\boxed{\text{i.e } \varepsilon = \frac{e}{(c/2)} = \frac{2e}{c} = \frac{2}{c}\left(\frac{c}{2} - h_0\right) = 1 - \frac{2h_0}{c}}$

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Design procedure for journal bearing:

1. Determine the bearing length by choosing a ratio of l/d from table (i.e from data book)
2. Check the bearing pressure, $p = W/l \cdot d$ from table (i.e from data book)
3. Assume a lubricant from table and its operating temperature which is assumed from 60°C to 80°C
4. Determine the operating value of $\frac{Zn}{P}$ for the assumed bearing temperature and check this value with corresponding values in table (i.e from data book) to determine the possibility of maintaining fluid film operation.
5. Assume a clearance ratio c/d from table
6. Determine the coefficient of friction (μ) by using the proper relation.
7. Determine the heat generated by using the proper relation.
8. Determine the heat dissipated by using the proper relation.
9. Determine the thermal Equilibrium to see that the heat dissipated becomes atleast Equal to heat generated.

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In case the heat generated is more than the heat dissipated then either the bearing is redesigned or it is artificially cooled by water.

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UNIT - 1

HYDRO DYNAMIC JOURNAL BEARINGS

&

BALL AND ROLLER BEARINGS

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(Pb) Design a journal bearing of centrifugal pump by following data.

Speed 900rpm, oil is SAE 10, for which absolute viscosity is 55°C Equal to 0.017 kg/ms Ambient temperature of oil is Equal to 15.5°C. maximum bearing pressure for the pump is 1.5 N/m². Calculate also mass of the lubricating oil required for artificial cooling. if rise of temp of oil be limited to 10°C. Heat dessipation co-efficient is 1232 watt/m²/°C.

Sol

Sol Given data:

load acting on journal (W) = 20000 N

Speed of journal (N) = 900rpm.

oil = SAE10

Absolute (or) dynamic viscosity of oil (μ) = 0.017 kg/ms

Ambient (or) Atmospheric (or) Surrounding (ta) = 15.5°C.

Allowable bearing [(p)] = 1.5 N/mm²

Heat dissipation co-efficient (c) = 1232 w/m²/°C.

difference b/w inlet & outlet temp of oil (ot) = 10°C.

to find:

design of journal bearing of centrifugal pump.

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Step-1: to determine

length of journal (l) = ?

diameter of journal (d) = ?

take $\frac{l}{d}$ ratio from table 19.5

for centrifugal pump

$$\frac{l}{d} = 1-2.0$$

Assume $\frac{l}{d} = 1.6$

$$l = 1.6 \times d$$

Assume $d = 100mm$

$$l = 1.6 \times 100$$

$$l = 160mm$$

Step-2:

Check the developed pressure from Eqn 19.9

$$p = \frac{W}{l \cdot d} \leq (p)$$

$$p = \frac{20000}{160 \times 100} = 1.25\ N/mm^2 \leq (p)$$

Step-3: To find bearing modulus

$$K = \frac{1}{3}\left(\frac{Zn}{P}\right)$$

take $\left(\frac{Zn}{P}\right)$ from the table.

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$$K = \frac{1}{3} \times 28.45$$

$$K = 9.48$$

It operates under hydrodynamic condition

find $\frac{c}{d}$ from table 19.5 for Σ_1

$$\frac{c}{d} = 0.0013.$$

step-4: Based on units consider equation 19.5

$$\mu = \left[\frac{33.25}{10^8} \times \frac{zn}{p} \times \frac{d}{c}\right] + K$$

take $z = 0.011$ kg/ms

$n = 900$

$p = 1.25$ N/mm^2

$k = 0.002$ (from graph 19.2 based on l/t ratio)

$$1/_{cs} = \frac{1}{0.0013}$$

$$\mu = \frac{33.25}{10^8} \times 12.24 \times \left(\frac{1}{0.0013}\right) + 0.002.$$

$$= 0.0053.$$

step-5: Heat generated base on units 19.10

$$Hg = \mu . \omega . V$$

V = Sliding velocity (or) rubbing velocity

$$V = \frac{\pi d n}{60 \times 10^3}$$

$$= \frac{\pi \times 100 \times 900}{60 \times 10^3}$$

$$= 4.71 \text{ m/s}$$

$$Hg = 0.0053 \times 20,000 \times 4.71$$

$$= 499.26 \text{ W}$$

Step-6: Heat dissipated.

$$H_d = C A (t_b - t_a)$$

$$C = 1232 \text{ A/w}^2/^\circ\text{C}$$

$$A = l \times d$$

$$= \frac{160 \times 100}{10^6} = 0.016 \text{ m}^2$$

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

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

$$= 19.75$$

$$H_d = 1232 \times 0.016 \times 19.75$$

$$= 389.312 \text{ W}$$

Compare Hg & Hd

⇒ the amount of artificial cooling required = Hg - Hd

$$= 499.26 - 389.312$$

$$= 109.9848 \text{ w.}$$

⇒ Mass of oil for heat remove by artificial cooling $M = \frac{Hg}{C_p \Delta t} = \frac{109.94}{1900 \times 10} = 0.0057 \text{ kg/s}$

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P6) the load on general bearings is 150kN due to turbine shaft of 300 dia running at 1800 rpm. determine the following.

(a) length of the bearing to the allowable bearing pressure is $1.6 N/mm^2$ and.

(b) Amount of heat to be removed by the lubricant per minute if the bearing temperature is 60° and viscocity of the oil at 60°c is 0.02 kg/ms and bearing clearence is 0.25mm?

Sol Given data.

load acting on journal (W) = 150N

diameter of journal (d) = 300mm

Speed of journal (n) = 1800 rpm.

Diametrol clearence (c) = 0.25mm

Absolute (or) dynamic viscosity of oil $\eta = 0.02$ kg/m-s

bearing pressure (P_b) = $1.6 N/mm^2$

temperature of bearing surface (t_b) = 60°c.

to find:-

length of the bearing & Amount of heat to be removed.

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(i) Step-1 :- length of bearing

$A = l \times d$

$d = 300mm$

$l = ?$

$A = 300\,l$.

Developed bearing pressure $p = \frac{W}{l \cdot d} \leq (p)$ from → (19.9

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

Here $p = 1.6\ N/mm^2$

$W = 150 \times 10^3$.

$d = 300mm$.

$1.6 = \frac{150 \times 10^3}{l \times 300} = 8$

$l = 312.5mm$.

Step-2 :- Amount of heat to be removed.

Heat generated $Hg = \mu wv$ watts.

μ = co-efficient of friction b/w journal & bearing.

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

$= \left[\frac{33.25}{10^8} \times \frac{0.02 \times 1800}{1.6} \times \frac{300}{0.25}\right] + 0.002$.

Here $K = 0.002$ [From graph 19.2 based on l/d ratio

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$\mu = 0.016$

$W = 150 \times 10^3$ N

V = sliding velocity (or) rubbing velocity

$= \frac{\pi d n}{60}$ m/s

$d = 300$ mm

$n = 1800$ rpm.

$= \frac{\pi \times 300 \times 1800}{60 \times 100}$

$= 28.27$ m/s

$\therefore H_g = \mu W V$

$= 0.016 \times 150 \times 10^3 \times 28.27$

$H_g = 67848$ W

(b) A full journal bearing of 50mm dia & 100mm long has a bearing pressure of 1.4 N/mm^2. The speed of journal is 900rpm & the ratio of journal dia to the dia clearance is 1000 the bearing is lubricated with oil whose absolute viscosity at the operating temperature of 75° celcius may taken as 0.01 kg/m sec the room temperature is 35°C. Find the amount of artificial cooling required.

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→ the mass of the lubricated oil required if the difference b/w the outlet & inlet temperature is the oil is 10°C. taken specific heat of the oil has 1850 J/kg/sec?

Sol) Given data:

Diameter of journal (d) = 50 mm

length of journal (l) = 100 mm

bearing pressure (P_b) = 1.4 N/mm²

ratio of journal dia to the dia clearance (d/c) = 1000.

operating temperature t_o = 75°

Atmospheric temperature t_a = 35°

Absolute (or) dynamic viscosity of oil (η) = 0.011 kg/m/sec.

Speed of journal (N) = 900 rpm.

Step-1:

Heat generated during operation

Hg = W.V watts

μ = co-efficient of friction b/w journal & bearing.

$$= \left[\frac{33.25}{10^8} \times \frac{Zn}{P} \times \frac{d}{c}\right] + k. \quad \rightarrow (19.5)$$

k = 0.002. [from graph 19.2 based on l/d ratio].

$$= \left[\frac{33.25}{10^8} \times \frac{0.001 \times 900}{1.4} \times 1000\right] + 0.002.$$

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$= 4.35 \times 10^{-3}$

developed bearing pressure $\rightarrow (19.9)$

$p = \frac{W}{l \times d}$ then.

$W = p \times l \times d$

$= 1.4 \times 50 \times 100$

$= 700.$

$v = \frac{\pi dn}{60}$ m/s

$d = 50$ mm, $n = 900$ rpm.

$= \frac{\pi \times 50 \times 900}{60 \times 1000}$

$= 2.356$ m/s

$H_g = 4.35 \times 10^{-3} \times 700 \times 2.356$

$= 71.74$ W (or) J/s

Step-2: Heat dissipated to the bearing

$H_d = CA (t_b - t_a) \rightarrow (19.12)$

$C = 150$ W/m^2/c [∵ consider 'c' value from (19.12)]

$A = l \times d$

$= 50 \times 100$

$= 5000$ mm^2 $= 0.005$ m^2

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

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$= \frac{1}{2}(75-25)$

$= \frac{1}{2}(40)$

$= 20$

$H_d = 150 \times 0.005 \ (20)$

$= 15\ W$

step-3: Mass of oil for heat removal

$M = \frac{H_g}{C_p \delta t} \rightarrow (19.13)$

$= \frac{71.74}{1850 \times 10}$

$= 3.87 \times 10^{-3}$

$= 0.00387 \ kg/s_{//}$

(b) A 150mm diameter shaft supporting a load of 10kN has a speed of 1500 rpm. the shaft runs in a bearings whose length is 1.5 times the shaft diameter. If the diameter clearence of the bearing is 0.15 mm and the absolute viscosity of the oil at the operating temperature is 0.011 kg/m-s find the power registered in friction (heat generated).

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Sol: Given data

diameter of journal (d) = 150mm

load acting on the journal (w) = 10×10^3 N.

speed of journal (n) = 1500rpm.

diameter clearence (c) = 0.15mm

Absolute (or) dynamic viscosity of oil (η) = 0.011 kg/m-s

length of journal $l = 1.5(d)$.

step-1: length of journal

$l = 1.5(d)$

$= 1.5 \times 150$

$= 225$ mm.

step-2: Developed bearing pressure

$p = \frac{W}{l \cdot d}$ [$\because W = 10 \times 10^3$, $l = 225$mm, $d = 150$mm]

$= \frac{10 \times 10^3}{225 \times 150}$

$= 0.29$ N/mm^2.

step-3: Heat generated during operation

$H_g = \mu w v$

μ: co-efficient of friction b/w journal & bearing

p'

$$\mu = \left[\frac{33.25}{10^8} \times \frac{zn}{p} \times \frac{d}{c}\right] + K$$

$K = 0.002$ [based from graph (19.2) based on l/d ratio]

$$= \left[\frac{33.25}{10^8} \times \frac{0.011 \times 1500}{0.29} \times \frac{150}{0.15}\right] + 0.002$$

$= 0.02$.

$W = 10 \times 10^3$ N.

So'

$V = \frac{\pi dn}{60}$ m/s

$d = 150$ mm, $n = 1500$

$$V = \frac{\pi \times 150 \times 1500}{60 \times 1000}$$

$= 11.78$

$Hg = \mu W V$

$= 0.02 \times 10 \times 10^3 \times 11.78$

$= 2414.9$ W.

Power wasted in friction $= 2414.9$ W.

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5) A 80mm long general bearing supports a load of 2800N on a 50mm dia shaft the bearing has a radial clearence of 0.05mm and the viscosity of the oil is 0.021 kg/m·s at the operating temperature if the bearing is capable of disipiating 80 Joules/s. determine the maximum safe speed.

Sol) Given data.

l = length of journal = 80mm

load acting on the journal (W) = 2800N

diameter of journal (d) = 50mm.

Absolute (or) dynamic viscosity of oil (η) = 0.021 kg/m·s

Heat disipiating (H_d) = 80 J/s

radial clearence $c/d = \frac{0.05}{} = 0.1$

step-1 : Heat generated during operation

developing bearing pressure

$$p = \frac{W}{l \cdot d} \qquad \left[\begin{array}{l} W = 2800 \\ l = 80\text{mm}, d = 50\text{ mm} \end{array}\right]$$

$$= \frac{2800}{80 \times 50} = 0.7 \text{ N/mm}^2$$

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μ = Co-efficient friction b/w journal & bearing.

$$\mu = \left[\frac{33.25}{10^8} \times \frac{zn}{P} \times \frac{d}{c}\right] + k$$

$k = 0.02$ [from graph 19.2 based on l/d ratio]

$$\mu = \left[\frac{33.25}{10^8} \times \frac{0.021 \times n}{0.7} \times \frac{50}{0.1}\right] + 0.002$$

$$= 4.9875 \times 10^{-6} \times n + 0.002$$

$W = 2800\,N$

$$V = \frac{\pi d n}{60}\ m/s$$

$$= \frac{\pi \times 50 \times n}{60 \times 1000}$$

$$= 2.617n \times 10^{-3} n$$

$$Hg = \mu W V$$

$$= [4.9875 \times 10^{-6} \times n + 0.002] \times (2800) \times (2.617 \times 10^{-3} n)$$

$$= 3.65 \times 10^{-5} n^2 + 0.014n.$$

step-2: Heat dissipated by the bearing = Heat generated

$$Hg = Hd.w$$

$$3.65 \times 10^{-5} n^2 + 0.014n = 80.$$

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$= 3.65 \times 10^{-5} n^2 + 0.014n - 80 = 0$

$$n = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$

$$n = \frac{-0.014 \pm \sqrt{(0.014)^2 - 4(3.65 \times 10^{-5})(-80)}}{2(3.65 \times 10^{-5})}$$

$n = 1287.67$ rpm

Equlae's of deep groove ball bearing:

⇒ Dynamic capacity (or) dynamic load rating:

$$C = \left(\frac{L}{L_0}\right)^{1/k} \cdot P$$

Where $k = 3$ for ball bearings

$L_0 = 1$ mr

⇒ Equivalent load:

$$P = (VXF_r + YF_a)S$$

$V = 1.0$ for inner ring rotation and outer ring stationary

$F_r, F_a \rightarrow$ Radial and axial loads

$X, Y \rightarrow$ Radial and axial load factors.

$S \rightarrow$ Service factor.

⇒ Life of bearings in million revolutions:

$$L = (C/P)^k \text{ mr.}$$

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⇒ Life of bearings in hours

$$L_h = \frac{L \times 10^6}{60 \times n} \text{ hours.}$$

⇒ Cubic mean load for periodically variable loading

$$P_m = \left[\frac{P_1^3 n_1 + P_2^3 n_2 + \cdots P_r^3 n_r}{n_1 + n_2 + \cdots n_r}\right]^{1/3}$$ For revolutions varying

$$= \left[\frac{P_1^3 t_1 + P_2^3 t_2 + \cdots P_r^3 t_r}{t_1 + t_2 + \cdots t_r}\right]^{1/3}$$ For time varying.

⇒ mean load for linearly variable loading

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

⇒ probability of survival :-

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

Where L = required life of bearings in mr

L'_{10} = calculated life of selected bearings, for the given load, for 90% survival.

p = probability of survival

$\ln(1/P_{10}) = \ln(1/0.9) = 0.1053$, $b = 1.17$ for a median life $= 5L_{10}$

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P6) Select a single row deep bore bearing for a radial load of 4000N & axial load of 5000N operating with a speed of 160 rpm. for an average life of 5 at 10 hours per day. Assume uniform & steady load?

Sol) Given data:

Radial load (F_r) = 4000N

axial load (F_a) = 5000N

Speed of the bearings (N) = 1600rpm

Assume d = 70mm

step-1: Dynamic capacity (or) dynamic load rating

$$C = \left(\frac{L}{L_{10}}\right)^{1/k} \cdot P \quad \rightarrow (20.1)$$

L_h = life of bearings in hours

$L_h = 5 \times 300 \times 10$

$= 15000$ hours

step-2: Life of bearings in hours

$$L_h = \frac{L \times 10^6}{60 \times n} \quad \rightarrow (20.4)$$

$L_h = 15000$, $n = 1600$ then

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

$$= \frac{15000 \times 60 \times 1600}{10^6}$$

$$L = 1440 \times 10^6$$

step-3 :- Equivalent load

$$P = (VXF_r + YF_a)S$$

V = 1.0 for inner ring rotation and outer ring stationary

S = Service factor (table 20.3)

V = 1, S = 1

Assume :

$$\frac{F_a}{C_0} = 0.025$$

$$e = 0.22 \quad \text{[From table 20.5]}$$

$$\frac{F_a}{F_r} = \frac{5000}{4000} = 1.25$$

$$\frac{f_a}{F_r} > e \quad \text{then}$$

X = 0.56, Y = 2

$$P = (0.56(4000) + 2(5000))(1)$$

$$= 12240$$

From step-1 $\rightarrow$ $C = \left(\frac{1440 \times 10^6}{10^6}\right)^{1/3} \times 12240$

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$C = 138.219 \times 10^3 N$

Based on 'C' value we have to take bearing number & static number from the data book from the table (20.17)

Bearing number = 6321

$C_0 = 140240$., then

$\Rightarrow f_a/c_0 = \frac{5000}{140240} = 0.0356$

$\frac{f_a}{C_0} = 0.04$ [from data book table 20.5]

$e = 0.24$

$\frac{f_a}{f_r} = 1.25 \Rightarrow \left[\frac{5000}{4000} = 1.25\right]$

$\frac{f_a}{f_r} > e. \Rightarrow$ [If $\frac{f_a}{f_r} > e$ then consider X & Y values from table 20.5]

$X = 0.56$

$y = 2 -$

$y = 2 - \frac{(2-1.8)}{(0.04-0.025)} \times (0.04 - 0.0356)$

$= 1.94$

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Equivalent load

$P = (VXF_r + YF_a)S$

$= (0.56 \times 4000 + 1.94 \times 5000)(1)$

$= 11940$

From step-1 $\quad C = \left(\frac{L}{L_0}\right)^{1/K} \cdot (P)$

$= \left(\frac{1440 \times 10^6}{10^6}\right)^{1/3} \times 11940$

$C = 134.831 \times 10^3 N$

$= 134831 N$

Based on 'C' value we have to take bearing number from the data book from the table (20.17)

Bearing Num = 6320

pb) Select a single row deep groove ball bearing for a radial load of 7000N & a thrust load of 2100N the desired life of bearing is 160mr of revolution at 300 rpm? Assume static load of 72000 N. And the diameter of the shaft is 75mm? (11)

Sol Given data.

Radial load $F_r = 7000N$

Axial load $F_a = 2100N$

Speed of the bearings $(n) = 300$ rpm

Life of bearing in million revolutions $L = 160mr$

Static load rating $C_0 = 72000N$

diameter of the shaft $d = 75mm$.

step-1: dynamic capacity (or) dynamic load rating

$$C = \left[\frac{L}{L_{10}}\right]^{1/k} \cdot P \rightarrow (20.1)$$

$L = 160\,mr \Rightarrow 160 \times 10^6$

$L_{10} = 1mr = 10^6$ [from data book (20.1)]

$K = 3$ [taken from (20.1)]

$P = ?$

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Step-II: Equivalent load

$P = [VXF_r + YF_a]S$

V = 1.0 for inner ring rotation and outer ring stationary

F_r, F_a = Radial and axial loads

X, Y = Radial and axial load factors

S = Service factor.

Assume V = 1

S = 1

$\frac{f_a}{C_0} = \frac{\text{Axial load}}{\text{static load rating}}$

$= \frac{2100}{72000} = 0.0291$

$e = 0.22$ → [From data book. table num (20.5)]

$\frac{f_a}{f_r} = \frac{2100}{7000} = 0.3$

$\frac{f_a}{f_r} > e$

$X = 0.56$, $Y = 2$ [From data book table number (20.5)]

ps= Substitute all values in 'P'.

$P = [(1)(0.56)(7000) + 2(2100)] \times (1)$

$= 8120$

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Step-III: Life of bearings in hours

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

$$= \frac{160 \times 10^6}{60 \times 300}$$

$$L_h = 8888.88 \text{ hours.}$$

"Substitute p = 8120 value in step-I"

$$C = \left[\frac{160 \times 10^6}{10^6}\right]^{1/3} \cdot 8120$$

$$= 44082.14 N$$

bearing number = 6213

C_o = ~~35715~~ 38440

Based on 'C' value we taken the bearing number & static number from data ~~book~~ table from the book (20.15)

$$\Rightarrow \frac{f_a}{C_o} = \frac{2100}{35715}$$

$$= 0.058$$

Assume $\frac{f_a}{C_o} = 0.07$

$e = 0.27$

$$\frac{f_a}{f_r} = 0.3.$$

$\frac{fa}{fr} > e$

$X = 0.56$ [taken from data book from table (20.5)]

y = highest y value $-$ $\frac{\text{(difference blw y value)}}{\text{(difference blw fa/co)}} \times$ (data book value $-$ Calculate value)

$y = 2 - \frac{(2-1.6)}{[0.07-0.0291]} \times [0.07-0.058]$

$= 0.4694$

[use the interpolation we get y value.]

$P = [VXFr + YFa]S$

$= [0.56 \times 7000 + 0.4694 \times 2100]$

$= 4905.74$

$C = \left(\frac{L}{L_{10}}\right)^{1/k} \cdot P$

$= \left(\frac{160 \times 10^6}{10^6}\right)^{1/3} \cdot 4905.74$

$C = 26632.45N$

Based on 'c' value we have to take bearing number from the data book from the table (20.15)

∴ Bearing number is 6210 //

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Pb A Single groove ball bearing has a dynamic load capacity of 40,500N and operates of the following work cycle Radial load of 5000N at 500rpm for 25% of the time, Radial load of 10,000N at 700rpm for 50% of the time, Radial load of 7000N at 400rpm for 25% of the time & calculate the expected life of bearing in hrs?

Sol Given data:

Dynamic load capacity (C) = 40,500N.

S.No	load	time	Speed	element time of revolution
1	5000N	0.25	500	125
2	10,000N	0.5	700	350
3	7000N	0.25	400	100
				n = 575

n → speed of the bearings.

Step-1: Cubic mean load for periodically variable loadings

$$P_m = \left[\frac{P_1^3 n_1 + P_2^3 n_2 + P_3^3 n_3}{n_1 + n_2 + n_3}\right]^{1/3}$$ from data book (20.5)

Here P_m = Cubic mean load.

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$$= \left[\frac{(5000)^3(125) + (10,000)^3(350) + (7000)^3(100)}{125+350+100}\right]^{1/3}$$

$= 7855.08N$

$= 8860.02N$

step-II: Life of bearings in million revolutions

$L = \left(\frac{C}{P}\right)^K$ from data book (20.3)

Here $C = 40500N$

$P = 8860.02N$

$K = 3$

Then $L = \left(\frac{40500}{8860.02}\right)^3$

$= 95.51$ - rev.

step-III: Life of bearing in hours

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

L: Life of bearing in million revolutions $= 955/N$

n: Speed of the bearings $= 575$

$L_h = \frac{95.51 \times 10^6}{60 \times 575}$

$L_h = 2768.40$ hrs //

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pb) A single row deep groove ball bearing is subjected to a radial force of 8kN and a thrust force of 3kN. the values of x & y factors are 0.56 and 1.5 respectively. the shaft rotates at 1200 rpm. the diameter of the shaft is 75mm and bearing number is 6315 (C = 112000N) is selected for this application?

(i) Estimate the life of this bearing with 90% reliability

(ii) Estimate the reliability of 20,000 hrs life.

Sol) Given data:

radial load $F_r = 8kN = 8\times10^3 N$

Thrust load $F_a = 3kN = 3\times10^3 N$

Radial load factor (X) = 0.56

thrust load factor (Y) = 1.56

Speed of the bearings (n) = 1200 rpm.

diameter of the shaft (d) = 75mm.

Bearing number = 6315, [C = 112000N).

step-1: Life of bearing in million revolutions

$$L = (C/P)^k \rightarrow (20.1)$$

$$= \left(\frac{112000}{8980}\right)^3 = 1940.10 \text{ mr.}$$

Here $P = (VXF_r + YF_a)S$. → Equivalent load

$= \left[(1)(0.56)(8\times10^3) + (1.5)(3\times10^3)\right]$ (1)

$= 8980\ N.$

Step-II: Estimate the life of bearing.

Life of bearing in hours

$$L_h = \frac{L\times10^6}{60\times n} \rightarrow (20.4)$$

$$= \frac{1940\times10\times10^6}{60\times1200}$$

$$= 26945.83 \text{ hrs.}$$

Step-III: probability of survival.

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

$$\frac{20,000}{26945.8} = \left[\frac{\ln(1/P)}{0.1053}\right]^{1/1.17}$$

$$\ln(1/P) = \left(\frac{20,000}{26945.8}\right)^{1.17} \times 0.1053.$$

$$= 0.07429$$

$$1/P = e^{0.07429}$$

$$P = \frac{1}{e^{0.07429}}$$

$$P = 0.928 \Rightarrow 92.8\%$$

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(b) A single groove deep ball bearing is subjected a 30 sec. 15
M/C cycle following two parts.

	Part -I	Part -II
duration (s)	10	20
radial load (kN)	45	15
axial load (kN)	12.5	6.25
Speed (rpm)	720	1440

the static and dynamic load capacities of ball bearing are 50.68 kN respectively calculate the expected life of bearing in hrs?

Sol) Given data:

dynamic load capacities C = 50, 68 kN

	part-I	part -II
duration (s)	10	20
radial load (kN)(Fr)	45	15
axial load (kN)(Fa)	12.5	6.5
Speed (rpm)	720	1440.

For Part-I

$$\frac{Fa}{C_0} = \frac{12.5}{50} = 0.25$$

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based on $\frac{fa}{Co}$ we have to take the 'e' value from data book from the table. 20.5

$e = 0.37$

$\frac{fa}{fr} = \frac{12.5}{45} = 0.27$

$\frac{fa}{fr} < e$. (based on this step we have to take x, y values from data book from the table 20.5)

$x = 1 \quad y = 0$

Equivalent load $P_1 = (VXF_r + YF_a)S \rightarrow (20.2)$

$= (1 \times 1 \times 45 + 0 \times 12.5)\ (1)$

$P_1 = 45 \times 10^3 N$

FB part-II.

$\frac{fa}{Co} = \frac{6.25}{50} = 0.125$

$e = 0.31$

based on $\frac{fa}{Co}$ we have to take the 'e' value from data book from the table 20.5

$\frac{fa}{fr} = \frac{6.25}{15}$

$= 0.416.$

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$\frac{fa}{fr} > e$ (based on this step we have to take x value. from data book from the table 20.5)

$x = 0.56$

$$Y = 1.6 - \left[\left(\frac{1.6 - 1.4}{0.13 - 0.07}\right) \times (0.13 - 0.125)\right]$$

$$= 1.583.$$

Equivalent load $P_2 = [VXFr_2 + Y Fa_2]S$

$$N_2 = \frac{20}{60} \times 1440 = 480$$

$N_1 + N_2 = 120 + 480 = 600 \rightarrow$

one minute $= 1200 \Rightarrow n = 1200$

Cubic mean load for periodically variable loading

$$P_m = \left[\frac{P_1^3 N_1 + P_2^3 N_2}{N_1 + N_2}\right]^{1/3}$$

$$= \left[\frac{(450 \times 10^3)(120) + (18.293 \times 10^3)(480)}{600}\right]^{1/3}$$

$P_e = 28488.93N.$

Life of bearing in million revolution

$$L = \left(\frac{C}{P}\right)^K$$

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$$= \left(\frac{68 \times 10^3}{28488.93}\right)^3$$

$$L = 13.598 \text{ mrev}$$

Life of bearing in hours

$$L_h = \left[\frac{L \times 10^6}{60 \times n}\right]$$

$$= \frac{13.598 \times 10^6}{60 \times 1200}$$

$$L_h = 188.86 \text{ hrs.}$$

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