UNIT-V

BEARINGS

PART - A

1. **How are bearings classified? (Dec 2010 & MAY 2014)**

    Depending upon the type of contact

    a. Rolling element bearing
    b. Sliding contact bearing

    Depending upon the type of rolling element

    a. Ball bearing
    b. Roller bearing

2. **What is a journal bearing? List any two applications. (Nov 2006, May 2013 & May 2014)**

    A journal bearing is a sliding contact bearing which gives lateral support to the rotating shaft.

3. **Classify the sliding contact bearings according to the thickness of layer of the lubricant between the bearing and the journal.(May 2012)**

    a. Thick film type
    b. Thin film type
    c. Hydrostatic bearings
    d. Hydrodynamic bearings

4. **For a journal bearing, the maximum operating temperature must be less than 80^{o}C. why?(Dec 2010)**

    Temperature rise will result in the reduction of the viscosity of the oil used in the bearing. It would lead to metal to metal contact therby affecting the bearing performance and life.

5. **What is known as self acting bearing? (NOV 2007)**

    The pressure is created within the system due to rotation of the shaft known as self acting bearing.**In hydro dynamic bearing, what are factors which influence the formation of wedge fluid film? (Nov 2014)**

1. The contact surfaces must meet at a slight angle to allow the formation of the lubricant wedge.
2. The fluid viscosity must be high to maintain an adequate film thickness to separate the contacting surfaces at operating speeds.

6. **Explain the term dynamic load carrying capacities of rolling contact bearing. (Dec 2012)**

   The basic dynamic load rating is defined as the constant stationary radial load (in case of radial ball or roller bearings) or constant axial load (in case of thrust ball or roller bearings) which a group of apparently identical bearings with stationary outer ring can endure for a rating life of one million revolutions (which is equivalent to 500 hours of operation at 33.3 r.p.m.) with only 10 per cent failure.

7. **What are the types of radial ball bearing? (May 2012)**
   1. Deep groove ball bearing
   2. Self angular ball bearing
   3. Angular contact ball bearing
   4. Filing notch bearing.

8. **What is the application of thrust bearing? (Dec 2010)**

   Thrust bearings are used to support axial loads imposed on the rotaing elements.

9. **What is meant by life of anti-friction bearing? (Apr 2008 & May 2013)**

   For an individual rolling bearing, the number of revolutions which one of the bearing rings makes in relation to the other rings under the prevailing working conditions before the first evidence of fatigue develops in the material of one of the rings or rolling elements.

10. **List any four advantages to rolling contact bearings over sliding contact bearings. (Apr 2009)**

    1. Starting friction is low
    2. Lubrication is simple
    3. It requires less axial space

11. **State the disadvantages of thrust ball bearing. (Apr 2009)**

    1. High initial cost
    2. Less capacity to withstand shock
    3. Noisy operation at very high speed.
    4. Life is finite

12. **Define static capacity of bearing. (Nov 2014)**

    It is defined as load acting on a non rotating bearing under which permanent deformation is 0.0001 times the ball or roller diameter.

13. **Explain the term dynamic load carrying capacities of rolling contact bearing. (Nov 2004)**

    It is defined as the radial load in radial bearings that can be carried for a minimum life of one million revolutions.

14. **What are the modes of failure of rolling contact bearing? (Dec 2010)**

1. Flaking or surface fatigue, 2. Peeling, 3. Scoring, 4. Fretting, 5.Creep

15. What are anti friction bearings? (April 2017)

Advantage of a rolling contact bearing over a sliding bearing that it has a low starting friction. Due to this low friction offered by rolling contact bearings, these are called antifriction bearings.

UNIT – V

BEARINGS

1. Design a journal bearing for a Centrifugal pump from the following data: Load on the journal = 20,000 N; Speed of the journal = 900 r.p.m, Type of oil is SAE 10, for which the absolute Viscosity at 55°c = 0.017 kg/m-s; Ambient temperature of oil = 15.5°c; Maximum bearing pressure for the pump = 1.5 N/mm²; Calculate the mass of the lubricating oil required for artificial Cooling, if rise of temperature of oil be limited to 10°c. Heat dissipation Coefficient is 1232 W/m²/°c. [NOV/DEC 2013]

GIVEN DATA:

$W = 20{,}000\,N$; $N = 900$ r.p.m; SAE 10 oil; $z = 0.017$ kg/ms

$\Delta t = 10°c$; $t_0 = 55°c$; $P_{bmax} = P_b = 1.5\,N/mm^2$;

$K = 1232\,w/m^2/°c$.

TO FIND:
1. Length of Journal, L
2. Mass of Lubricating oil, m
3. Diameter of Bearing, D_b.

SOLUTION:

I Length of Journal, L:

From DDB P.NO: 7.31 Corresponding to Centrifugal pump $\frac{L}{D} = 1$ to 2:

We take $\frac{L}{D} = 1.6$

Assume $D = 100$ mm

$L = 1.6 \times D$
$= 1.6 \times 100$
$L = 160$ mm

II Developed Bearing Pressure, p:

$$p = \frac{W}{L.D} = \frac{20,000}{160 \times 100} = 1.25\ N/mm^2$$

In DDB. P.No: 7.31 Corresponding to Centrifugal pump, Bearing Pressure, p lies between 7-14 kgf/cm^2 that is 0.7 to 1.4 N/mm^2.

Calculated Pressure, $p = 1.25\ N/mm^2$ lies in this range.

∴ Pressure, p is Safe and dimensions L&D are Safe.

III Bearing Characteristic Number, $\frac{ZN}{p}$:

From DDB. P.No: 7.31 Corresponding to Centrifugal pump $\frac{ZN}{p} = 2844.5$

Note: We have to take first two digit from DDB Value

$$\therefore \frac{ZN}{p} = 28.$$

$$\frac{ZN}{p_{cal}} = \frac{0.017 \times 900}{1.25} = 12.24.$$

Minimum value of Bearing modulus at which oil film will break is given by

$$3K = \frac{ZN}{p}$$

$$K = \frac{1}{3} \times \frac{ZN}{p}$$

$$= \frac{1}{3} \times 28$$

$K = 9.33$

Calculated bearing characteristic number 12.24 is greater than 9.33.

∴ Bearing will operate under hydrodynamic (or) with thick lubrication Condition.

<u>Note 1:</u> Clearance Ratio for Railway Cars, Steam Turbines, Transmission Shafts, machine Tools, punching and shearing machines is 0.001.

<u>Note 2:</u> Clearance Ratio for Generators, motors and Centrifugal pump is 0.0013.

<u>Note 3:</u> Clearance ratio for Rolling mills is 0.0015

iv Clearance Ratio, $\frac{c}{D}$:

For Centrifugal pump, $\frac{c}{D} = 0.0013$

v Heat Generated, H_g:

$$H_g = \mu \cdot V \cdot W$$

DDB. P.NO: 7.34

McKees Equation

$$\mu = \frac{33.25}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{D}{c}\right) + k.$$

From DDB.P.NO: 7.34 graph Corresponding to $\frac{L}{D} = \frac{160}{100} = 1.6$, the value of k is 0.0025

$$\therefore \mu = \frac{33.25}{10^8} \times 12.24 \times \left(\frac{1}{0.0013}\right) + 0.0025$$

$$= 0.0051$$

v = Rubbing Velocity.

$$= \frac{\pi D N}{60}, \text{ where diameter in metres.}$$

$$= \frac{\pi \times 0.1 \times 900}{60}$$

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

$H_g = \mu . v . W$

$= 0.0051 \times 4.71 \times 20{,}000$

$H_g = 480.7\ W$

VI Heat Dissipated, H_d:

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

$= C \times L \times D \times \frac{1}{2}(t_o - t_a)$

$= 1232 \times 0.16 \times 0.1 \times \frac{1}{2}(55 - 15.5)$

$H_d = 389.3 W$

VII Amount of Artificial Cooling Required:

$= H_g - H_d$

$= 480.7 - 389.3$

$= 91.4 W$

VIII Mass of Lubricating Oil require for Artificial Cooling, m:

S = Specific heat of oil

$= 1840$ to $2100\ J/kg/°c$

We take, $S = 1900\ J/kg/°c$.

$H_t = H_g - H_d = m . s . \Delta t$

$91.4 = m \times 1900 \times 10$

$\therefore m = 0.0048\ kg/s$

$= 0.0048 \times 60$

$\boxed{m = 0.288\ kg/min}$.

IX Diameter of Bearing, D_b:

$$D_b = D + c$$

$$\frac{c}{D} = 0.0013$$

$$c = 0.0013 \times D$$
$$= 0.0013 \times 100$$
$$= 0.13\ mm$$
$$= \text{Diametral Clearance}$$

$$D_b = 100 + 0.13 = 100.13\ mm$$

SPECIFICATIONS:

1. Diameter of Journal, $D = 100mm$
2. Length of Journal, $L = 160mm$
3. Diameter of Bearing, $D_b = 100.13mm$
4. Diametral Clearance, $c = 0.13mm$
5. Lubricant = SAE 10 Oil.
6. Operating Temperature, $t_o = 55°c$.
7. Atmospheric Temperature, $t_a = 15.5°c$

2. A 150mm diameter Shaft Supporting a load of 10kN has a speed of 1500 r.p.m. The shaft runs in a bearing whose length is 1.5 times the shaft diameter. If the diametral clearance of the bearing is 0.15mm and the absolute Viscosity of the oil at the operating temperature is 0.011 kg/m-s. Find the power wasted in friction and increase in the oil temperature. [Nov/DEC 2014]

GIVEN DATA:

$D = 150mm$; $N = 1500 r.p.m$; $C = 0.15mm$

$W = 10 kN = 10\times10^3 N$; $L = 1.5D$; $\therefore \frac{L}{D} = 1.5$; $Z = 0.011 kg/m\text{-}s$.

TO FIND: (1) Power lost in friction, H_g.

(2) Increase in oil temperature, Δt

I. Length of the Journal, L:

$$L = 1.5 \times D = 1.5 \times 150 = 225mm$$

II Coefficient of friction, μ and Rubbing Velocity, V:

$$p = \frac{W}{L\times D} = \frac{10{,}000}{225\times150} = 0.296 N/mm^2$$

From DDB. P.No: 7.34 Corresponding to $\frac{L}{D}$, the value of k is 0.002.

$$\mu = \frac{33.25}{10^8}\left(\frac{ZN}{p}\right)\times\frac{D}{C} + k.$$

$$= \frac{33.25}{10^8}\left(\frac{0.011\times1500}{0.296}\right)\times\left(\frac{150}{0.15}\right) + 0.002$$

$$\mu = 0.02$$

$$V = \frac{\pi DN}{60} = \frac{\pi\times0.15\times1500}{60} = 11.78 m/s.$$

III Heat Generated Due to Friction (H_g):

H_g = Power Wasted in Friction

$$= \mu . V . W = 0.02\times11.78\times10\times10^3$$

$$H_g = 2356 W$$

III Increase in Oil Temperature, Δt:

$$H_g = H_d$$
$$= C \times L \times D \frac{1}{2}(t_o - t_a)$$

$$2356 = 1400 \times 0.225 \times 0.15 \times \frac{1}{2}(\Delta t)$$

$$\boxed{\Delta t = 98°C}$$

3. Select a single row deep groove ball bearing for a radial load of 4000N and an axial load of 5000N, operating at a Speed of 1600 r.p.m for an average life of 5 years at 10 hours per day. Select from series 64. [MAY/JUNE 2013]

GIVEN DATA:

Radial Load, $F_r = 4000N$

Axial Load, $F_a = 5000N$

$N = 1600 r.p.m$

Average Life = 5 years at hours per day.

TO FIND: ① Life of Bearing in Revolutions

② Dynamic Capacity

③ Selection of Bearing.

SOLUTION:

I Life of Bearing in Revolutions, L:

L_H = Life of bearing in working hours.

= Working years × Working days × Working Hours.

$= 5 \times 300 \times 10$

$L_H = 15,000$ hours

$L = L_H \times N \times 60$

$= 15{,}000 \times 1600 \times 60$

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

II Dynamic Equivalent Load, P.

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

Where X = Radial Load factor.

Y = Axial Load Factor.

S = Rotational Service factor.

DDB . P.No: 4.4

Assume $\frac{F_a}{C_o} = 0.5$; where C_o = Static Load Capacity.

In tabulation available in DDB. P.No: 4.4 Corresponding to deep groove ball bearing and $\frac{F_a}{C_o}$, the value of e is 0.44.

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

$\frac{F_a}{F_r} > e$

$1.25 > 0.44$

∴ In DDB. P.No: 4.4 tabulation Corresponding to deep groove ball bearing & $\frac{F_a}{F_r} > e$, we have to take the value of 'X' and 'Y'

$X = 0.56$ and $Y = 1$

For most of the bearings Rotational Service factor, $S = 1$

$$P = [(0.56 \times 4000) + (1 \times 5000)] \times 1$$

$$= 7240N$$

III Dynamic Load Rating (or) Dynamic Capacity, C:

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

Where L_{10} = Life of bearing for 90% Survival at 1 million revolutions.

= One million revolutions.

$= 10^6$ revolutions

$K = 3$ [For Ball Bearings]

$$C = \left[\frac{1440 \times 10^6}{10^6}\right]^{1/3} \times 7240$$

$$= 81760N$$

$$C = 8176 \text{ kg}$$

IV Selection of Bearing: DDB. P.NO: 4.6

From graph Corresponding to 1600 r.p.m in X-axis and L_H = 15,000 hrs in Y-axis, the Loading ratio $\frac{C}{P} = 11.5$

$C = 11.5 \times P = 11.5 \times 7240$

$= 83260N$

$C = 8326 kg$

Take highest value of 'C'

$\therefore C = 83260N$

$= 8326 kg$

Standard deep groove ball bearings Specification available in DDB. P.NO: 4.12, 4.13, 4.14 & 4.15

From DDB. P.NO: 4.15 deep groove ball bearing having SKF 6412 deep groove ball bearing having C = 8450 kg may be Selected.

4. Select a Suitable Conrad type ball bearing for the following data. The radial load is 7500N and axial load is 4500N. The Shaft Speed is 2000 r.p.m and the L_{10} Life required is 4.9×10^6 revolutions. The inner ring of the bearing rotates. [MAY/JUNE 2012]

GIVEN DATA:

$F_r = 7500N$	$L = 4.9 \times 10^6$ revolutions
$F_a = 4500N$	Inner ring rotates. $\therefore V = 1$
$N = 2000$ r.p.m	Conrad ball bearing is 'Deep groove Ball Bearing'.

To FIND: Selection of type of Bearing.

SOLUTION:

I Dynamic Equivalent Load, P:

Let us select SKF 6318 bearing from DDB.

For SKF 6318, $C_o = 96110N$; $C = 107870N$.

$$\frac{F_a}{C_o} = \frac{4500}{96110} = 0.05$$

Corresponding value of 'e' from DDB P.No: 4.4 is 0.25

$$\frac{F_a}{F_r} = \frac{4500}{7500} = 0.6$$

$= 0.6$

$> e = 0.25$

In DDB. P.NO: 4.4 tabulation Corresponding to deep groove ball bearing and $\frac{F_a}{F_r} > e$, the value of 'x' and 'Y' are 0.56 and 1.74.

$$P = (VxF_r + YF_a) s$$

$$= (1 \times 0.56 \times 7500 + 1.74 \times 4500)$$

$$\boxed{P = 14436N}$$

II Selection of Bearing:

$$L = \left(\frac{C}{P}\right)^k = \left(\frac{107870}{14436}\right)^3 = 417.2 \text{ million revolutions}$$

$$= 4.17 \times 10^8 \text{ rev}$$

SKF 6318 bearing has life 4.17×10^8 rev. Which is less than the requirement (4.9×10^8 rev).

∴ Select next higher Capacity bearing

SKF 6319 bearing.

$C_0 = 107870N$ and $C = 117680N$.

$$\frac{F_a}{C_0} = \frac{4500}{107870} = 0.04.$$

Corresponding Value of $e = 0.24$

$$\frac{F_a}{F_r} = \frac{4500}{7500} = 0.6$$

$$= 0.6 > e:$$

From DDB. P.NO: 4.4, $X = 0.56$ and $Y = 1.8$.

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

$$= 14760N.$$

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

$$= 5.068 \times 10^8 \text{ revolutions}$$

$$> 4.9 \times 10^8 \text{ revolutions.}$$

∴ **SKF 6319 ball bearing may be Selected.**

5. For 6307 ball bearing the load Varies as follows

Radial Load (N)	Axial Load (N)	Cycle Time Ratio	Speed in r.p.m.
6000	3000	0.5	400
7500	—	0.3	650
4000	1000	0.2	900

The inner ring rotates, loads are Steady Find the expected average Life of the bearing [NOV/DEC 2014]

GIVEN DATA:

$F_{r1} = 6000N$; $F_{r2} = 7500N$; $F_{r3} = 4000N$

$F_{a1} = 3000N$; $F_{a2} = 0N$; $F_{a3} = 1000N$

$N_1 = 400 r.p.m$; $N_2 = 650 r.p.m$; $N_3 = 900 r.p.m.$

TO FIND: (1) L_H = Average Life of Bearing

(I) $\frac{F_{a1}}{C_o} = \frac{3000}{17600} = 0.17$

From DDB. P.NO: 4.4. Corresponding to Deep groove ball bearing the value of 'e' is 0.33

$$\frac{F_{a1}}{F_{r1}} = \frac{3000}{6000} = 0.5$$

$$\frac{F_{a1}}{F_{r1}} = 0.5 > e$$

∴ From DDB. P.NO: 4.4, $X = 0.56$ and $Y = 1.3$

Dynamic Equivalent Load $= P_1$

$P_1 = (X F_{r1} + Y F_{a1}) \times S$

$= [(0.56 \times 6000) + (1.3 \times 3000)] \times 1$

$= 7260 N$

(II) $\frac{F_{a2}}{C_o} = 0$

$P_2 = (X F_{r2} + Y F_{a2}) \times S$

where $F_{a2} = 0$ and $X = 1$

$P_2 = (1 \times 7500) \times 1 = 7500 N$

(III) $\frac{F_{a3}}{C_o} = \frac{1000}{17,600} = 0.056 \simeq 0.06$

DDB. P.NO: 4.4. Corresponding to Deep gloove ball bearing the value of e is 0.26

$\frac{F_{a3}}{F_{r3}} = \frac{1000}{4000} = 0.25$

$\frac{F_{a3}}{F_{r3}} = 0.25 < e$

$\therefore X = 1$ & $Y = 0$.

$P_3 = [(X F_{r3}) + (Y F_{a3})] \times S$

$= [(1 \times 4000) + (0 \times 1000)] \times 1$

$= 4000 N$

Load in N ①	Cycle Time Ratio ②	Speed in r.p.m ③	No. of revolutions for 1 Minute cycle ④ = ② × ③
$P_1 = 7260N$	0.5	400	$n_1 = 200$
$P_2 = 7500N$	0.3	650	$n_2 = 195$
$P_3 = 4000N$	0.2	900	$n_3 = 180$
			$\Sigma n = 575$

IV Cubic Mean Load, P_m:

DDB. P.No: 4.2

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

$$= \left[\frac{(7260^3 \times 200) + (7500^3 \times 195) + (4000^3 \times 180)}{575}\right]^{1/3}$$

$$= 6665.97N \simeq 6666N$$

$$P_m = P$$

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

V Life of Bearing:

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

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

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

$$L = \left(\frac{26000}{6666}\right) \times 10^6$$

$$= 59.33 \times 10^6 \text{ rev.}$$

$$L = L_H \times N \times 60$$

$$L_H = \frac{L}{N \times 60}$$

$$= \frac{59.33 \times 10^6}{575 \times 60}$$

$= 1719.9$ hours.

Expected Average Life of Bearing

= 5 years × Rated Life

$= 5 \times 1719.9$

$= 8598.55$ hours

Life of Bearing = 8598.55 hours