DHANALAKSHMI COLLEGE OF ENGINEERING

(Dr.VPR Nagar, Manimangalam, Tambaram)
Chennai - 601 301

DEPARTMENT OF MECHANICAL ENGINEERING

III YEAR MECHANICAL - V SEMESTER

GE6503 – DESIGN OF MACHINE ELEMENTS

ACADEMIC YEAR (2017 - 2018) - ODD SEMESTER

UNIT – 5 (STUDY NOTES)

Prepared by:

R. SENDIL KUMAR, AP/MECH

P.V. ELUMALAI, AP/MECH

P. SIVA KUMAR, AP/MECH

Unit 5 - DESIGN OF BEARING AND MISCELLANEOUS ELEMENTS

PART - A

1. What is bearing?
Bearing is a stationery machine element which supports a rotating shafts or axles and confines its motion.
2. Classify the types of bearings.
i. Depending upon the type of load coming upon the shaft:
a. Radial bearing
b. Thrust bearings.
ii. Depending upon the nature of contact:
a. Sliding contact
b. Rolling contact bearings or Antifriction bearings.
3. What are the required properties of bearing materials?
Bearing material should have the following properties.
i. High compressive strength
ii. Low coefficient of friction
iii. High thermal conductivity
iv. High resistance to corrosion
v. Sufficient fatigue strength
vi. It should be soft with a low modulus of elasticity
vii. Bearing materials should not get weld easily to the journal material.

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

5. What are the types of journal bearings depending upon the nature of contact?
1. Full journal bearing
2. Partial bearing
3. Fitted bearing.

6. What are the types of journal bearing depending upon the nature of lubrication?
1. Thick film type
2. Thin film type
3. Hydrostatic bearings
4. Hydrodynamic bearing.

7. What is known as self – acting bearing?
The pressure is created within the system due to rotation of the shaft, this type of bearing is known as self – acting bearing.

8. What is a quill bearing?

Quill bearing are characterized by cylindrical rollers of very small diameter and relatively long. They are also called needle bearings.

9. State the disadvantages of thrust ball bearing.

They are not suitable for high speeds; thrust loads try to shift the plane of rotation of balls.

10. List any four advantages of rolling contact bearings over sliding contact bearings.

Low starting torque can carry combined radial and axial torque. Required less axial space, maintain accurate alignment of shaft.

11. Define anti friction bearing.

The contact between the bearing elements is rolling; this type has very small friction.

12. What is meant by life of anti-friction bearings?

It defined as the life that 90 % of group of identical bearing will complete or exceed before fatigue failure.

13. What is the advantage of Teflon which is used for bearings?

It has low coefficient of friction; it can be used at higher temperature, and chemically inert.

14. What is a Journal bearing? List any two applications. (MAY/JUNE 2013)

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

15. Explain the term Dynamic load carrying capacities of rolling contact bearing.

(NOV/DEC 2012)

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

16. What are the types of thrust ball bearings?

One directional flat race, One directional grooved race, two directional grooved race

17. Classify the roller bearings.

Depending upon the type of rolling element:-ball bearing, roller bearing Depending upon the load to be carried, radial, angular, and thrust bearings.

18. What is load rating?

The load carrying capacity of a rolling element bearing is called load rating.

19. State any points to be considered for selection of bearings.(or) List any six types of bearing materials.

Lead based babbit, tin based babbit, leaded bronze, copper lead alloy, gun metal, phosphor bronze.

20. What is connecting rod?

It is a machine member, used to transmit power from a reciprocating member to rotary one.

21. Why I section used for connecting rod?

It is used due to its lightness and to keep the inertia forces as low as possible. it can also withstand high gas pressure.

22. What are the materials used for C-r

Mild steel and alloy of aluminum —for light duty.

Alloy steels of molybdenum and chromium ----for heavy duty.

23. What are the stresses set up in an IC engine C-rod?

Tensile stress, compressive stress, bending stress due to inertia force.

24. Discuss the forces acting on the connecting rod. Or Under what force the big end bolts and caps are designed. (NOV/DEC 2011) (NOV/DEC 2012)

The combined effect of (i)load on the piston due to the gas pressure and due to inertia of the reciprocating parts, and(ii)the friction of the piston rings, piston, piston rod and cross head.1.inertia of the connecting rod.2.the friction force in the gudgeon and crank pin bearing.

25. State the components of rolling contact bearing.

Outer race, inner race, rolling element, retaining cage.

26. List the basic assumption used in the theory of hydrodynamic lubrication?

(NOV/DEC 2011)

- ❖ The lubricant obeys newton’s law of viscous flow.
- ❖ The pressure is assumed to be constant throughout the film thickness.
- ❖ The lubricant is assumed to be incompressible.
- ❖ The viscosity is assumed to be constant throughout the film thickness.
- ❖ The flow is one dimensional.

27. Classify the sliding contact bearings according to the thickness of layer of the lubricant between the bearing and journal. (MAY/ JUNE 2012)

1. Thick film bearing 2. Thin film bearing 3. Zero film bearing 4.Hydrostatic bearing

28. What are various types of radial ball bearing? (MAY/ JUNE 2012)
1. single row deep groove ball bearing 2.Filling notch bearing 3. Angular contact bearing 4. Double row bearing 5. Self-aligning bearing

29. What do you meant by life of an individual bearing? (MAY/ JUNE 2013)

The life of individual bearing may be defined as the number of revolution which the bearing runs before the first evidence of fatigue develops in the material of one of the rings or any of the rolling elements.

PART - B

Connecting Rod

1. Design a suitable connecting rod for a petrol engine for the following details, diameter of the piston = 100 mm, weight of reciprocating parts per cylinder = 20 N, connecting rod length = 300 mm, compression ratio = 7:1, maximum explosion pressure = 3 N/mm^2, stroke = 140 mm, speed of the engine = 2000 rpm.

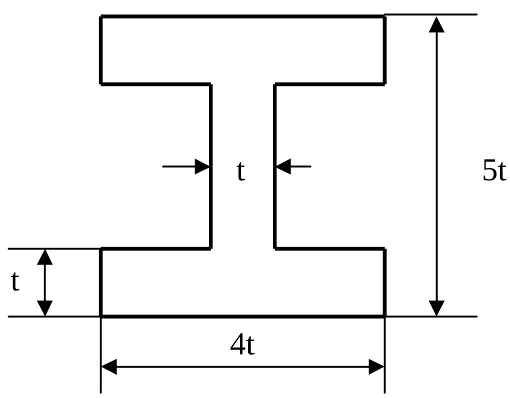
(APR/MAY 2002, NOV/DEC 2011 & MAY/JUNE 2012)

Given:

d = 100 mm, r = 20 N, l = 300 mm, compression ratio = 7:1, p = 3 N/mm^2, stroke = 140 mm, N = 2000 rpm.

Solution:

i. From PSG DDB Pg. No. 7.122,



$a = 11t^2$

$I_{xx} = \frac{419}{12}t^4$

$k_{xx}^2 = 3.18\ t^2.$

ii. Load due to burning of gas (F_G): From PSG DDB Pg. No. 7.122

$$F_G = \frac{\pi d^2}{4} \times p = \frac{\pi \times 100^2}{4} \times 3 = 23561.94N$$

iii. Crippling load (F_{cr}):

F_{cr} = FOS x F_G (Assume FOS = 6)

= 6 x 23561.94 = 141371.67 N

iv. Crippling load by Rankine's formula:

$$F_{cr} = \frac{\sigma_c \times a}{1 + c\left[\frac{l}{k_{xx}}\right]^2}$$

$$141371.67 = \frac{330 \times 11t^2}{1 + \frac{1}{7500}\left[\frac{300}{3.18t^2}\right]^2}$$

$$141371.67 = \frac{330 \times 11t^2}{1 + \frac{3.77}{t^2}} = \frac{3630t^2}{\frac{t^2 + 3.77}{t^2}}$$

$$141371.67 = \frac{3630t^4}{t^2 + 3.77}$$

$$141371.67(t^2 + 3.77) = 3630t^4$$

$$3630t^4 - 141371.67t^2 - 532971.196 = 0$$

$$\div 3630 \Rightarrow t^4 - 38.95t^2 - 146.824 = 0$$

$$t^2 = \frac{-(-38.95) \pm \sqrt{(-38.95)^2 - 4 \times 1 \times (-146.824)}}{2}$$

$$t^2 = \frac{38.95 \pm 45.874}{2} = 42.412$$

$$t = 6.15mm \cong 7mm$$

v. Dimensions of Cross section:

Height of 'I' section = 5t = 5 x 7 = **35 mm**
Width of 'I' section = 4t = 4 x 7 = **28 mm**

vi. Design of small end pin:

$$\frac{L_1}{d_1} = 1.75$$ (Assume P_b = 13 N/mm^2)

$F_G = L_1 \times d_1 \times P_b$

$23561.94 = 1.75d_1 \times d_1 \times 13$

d_1= 32.18 mm = 33 mm

L_1 = 1.75(33) = 57.75 = **58 mm**

vii. Design of Big end pin:

$$\frac{L_2}{d_2} = 1.375$$ (Assume P_b = 8 N/mm^2)

$F_G = L_2 \times d_2 \times P_b$

$23561.94 = 1.375d_2 \times d_2 \times 13$

$\mathbf{d_2 = 46.28\ mm = 47\ mm}$

$\mathbf{L_2} = 1.375(47) = 64.63 = \mathbf{65\ mm}$

viii. Diameter of bolt:

From PSG DDB Pg. No. 7.122,

$$\frac{L_1}{d_1} = 1.75$$

$$F_i = \frac{p}{g}\omega^2 r\left[\cos\theta + \frac{\cos 2\theta}{l/r}\right]$$

$$\omega = \frac{2\pi N}{60} = \frac{2\pi \times 2000}{60} = 209.44 rad/\sec$$

l = 300 mm = 0.3 m

r = radius of crank = stroke length / 2 = $\frac{140}{2}$ = 70 mm = 0.07 m

$$\therefore F_i = \frac{20}{9.81} \times (209.44)^2 \times 0.07 \times \left[1 + \frac{1}{0.3/0.07}\right]$$

$$F_i = 7720.736N$$

W.K.T

$$F_i = n \times \frac{\pi d_c^2}{4} \times \tau$$

$$7720.736 = 4 \times \frac{\pi d_c^2}{4} \times 100$$

$$d_c = 4.95mm \cong 5mm$$

Diameter of bolt $d = \frac{d_c}{0.84} = \frac{5}{0.84} = 5.95 \cong 6mm$

ix. Thickness of big end cap (t_c):

Bending moment $m_c = \frac{F_i \times x}{6}$ $(x=1.5d_2)$

$$= \frac{7220.736 \times 1.5 \times 47}{6}$$

$$= 90718.65 Nmm$$

Modulus, Z $= \frac{bt_c^2}{6}$ $(b=L_2)$

$$\frac{65\times t_c^2}{6} = 10.83t_c^2$$

W.K.T., $\sigma_b = \frac{m_c}{Z}$ (Assume σ_b=120N/mm^2)

$$120 = \frac{90718.65}{10.83t^2}$$

t_c = 8.35 mm

t_c = 8.5 mm

2. Design a journal bearing for a centrifugal pump with the following data:

Diameter of the journal = 150 mm

Load on bearing = 40 kN

Speed of journal = 900 rpm **(NOV/DEC 2007 & MAY/JUNE 2012)**

Given:

D = 150 mm, W = 40 kN, n = 900 rpm, Application = Centrifugal pump

Solution:

i. Diameter of journal is already given in the problem, D = 150 mm

ii. From PSG DDB Pg. No. 7.81, $\frac{L}{D}$ = 1.0 – 2.0

Bearing pressure allowable = 71014 kgf/cm^2,

$$\left(\frac{Zn}{p}\right)_{\min} = 2844.5$$

Take $\frac{L}{D}$ = 1.5, ∴ L = 1.5 D = 1.5 x 150 = 225 mm

iii. Bearing Pressure

P = $\frac{W}{L\times D}$ = $\frac{40\times 10^3}{225\times 150}$ = 1.185 N/mm^2 = 1.185 x 10 kgf/cm^2 = 11.85 kgf/cm^2.

which is less than allowable, so L/D value is acceptable.

iv. From PSG DDB Pg. No. 7.32, **Diameter clearance C = 150 microns**

= 150 x 10^{-3} mm

Clearance ratio, $\frac{C}{D}$ = $\frac{150\times 10^{-3}}{150}$ = 1 x 10^{-3}

v. Selection of lubricating oil.

From PSG DDB Pg. No. 7.31,

$\frac{Zn}{P}$ = 2844.5

Z = $\frac{2844.5\times 11.85}{900}$ = 37.45 = 40 centipoise.

From PSG DDB Pg. No. 7.41, for Z = 40 and temperature = 60° (assume). The suitable lubricating oil is SAE40.

vi. Bearing Characteristics number.

$$\frac{Zn}{P} = \frac{40\times900}{11.85} = 3037.97$$

It is higher than the minimum value given in PSG DDB Pg. No. 7.31.

vii. Calculation of μ.

From PSG DDB Pg. No. 7.34, $\mu = \frac{33.25}{10^{10}}\left(\frac{Zn}{P}\right)\left(\frac{D}{C}\right)+K$

$\frac{Zn}{P} = 3037.97$, $\frac{D}{C} = \frac{1}{1\times10^{-3}}$,

K = 0.0025 (for L/D = 1.5, from PSG DDB Pg. No. 7.34)

$$\mu = \frac{33.25}{10^{10}}\times3037.97\times\frac{1}{1\times10^{-3}}+0.0025$$

$\mu = 0.0126$

H_g and H_d

$H_g = \mu .w .v$ Watts

w in Newton,

$v = \frac{\pi Dn}{60}$ in m/min,

D in meters,

n in rpm

$H_g = 0.0126 \text{ x } 4000 \text{ x } \frac{\pi\times0.15\times900}{60}$ = **3562.56 W**

$$H_d = \frac{(\Delta t+18)^2 L\times D}{k}$$

L in meters

D in meters

K – constant, assume = 0.484 heat dissipation

Δt = temperature of bearing surface

Form ambient temperature

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

t_o = oil temperature, t_a = ambient temperature

$$\Delta t = \frac{1}{2}\left(60^o - 28^o\right) = 16^o C$$

$$H_d = \frac{(16+18)^2 \times 0.225 \times 0.15}{0.484}$$

H_d = 80.61 W

Here $H_g > H_d$ so artificial cooling is required to carry away the excess heat.

Diameter of the bearing D_b = D+C = 150 + 150 x 10^{-3} = **150.15 mm**

Material of Bearing

From PSG DDB Pg. No. 7.30, for pump application material is rubber or moulded plastic laminate.

Summary of Design

Material = Rubber or Moulded plastic laminate
Cooling = Artificial cooling required
Diameter of journal = 150 mm
Length of journal L = 225 mm
Diameter of bearing D_b = 150.15 mm
Diameter of clearance C = 150 microns
Lubricating oil suitable = SAE40
Operating temperature = 60°C
Atmospheric temperature = 28°C

3. Following data is given for a 360° hydro dynamic bearing:
Journal diameter = 100 mm, Radial clearance = 0.12 mm, Radial load =50 kN,
Bearing length = 100 mm, Journal speed = 1440 rpm, Viscosity of lubricant = 16 centipoise.
Calculate: 1. Minimum film thickness
2. Co-efficient of friction
3. Power cost in friction. **(MAY/JUNE 2009)**

Given: D= 100 mm, Radial clearance = 0.12 mm, W = 50kN, L=100 mm, n = 1440 rpm, Z=16 centipoise = 16 x 10^{-3} Ns/m^2.

Solution:

i. Minimum film thickness

W.K.T. Radial clearance = Diameteral clearance /2 = C/2
0.12 = C/2,
C = 0.24 mm

From PSG DDB Pg. No., Sommerfield number s = $\frac{Z'n'}{p}\left(\frac{D}{C}\right)^2$

Z' = viscosity in Ns/m^{2,}
n' = speed of jouranal in rps
p = bearing pressure in N/m^2.

n' = 1440/60 rps

Bearing pressure p = $\frac{W}{L\times D} = \frac{50\times10^3}{(0.1)(0.1)}$

p = 5 x 10^6^ N/mm^2^.

$$s = \frac{16\times10^{-3}\times\left(\frac{1440}{60}\right)}{5\times10^6}\times\left(\frac{100}{0.24}\right)^2$$

s = 0.013

From PSG DDB Pg. No. 7.40, for β = 360°, s = 0.013 and corresponding to L/D =1,

the minimum film thickness variable = $\frac{2h_o}{C}$ **= 0.071**

$$h_o = \frac{0.071\times C}{2} = 8.52 \times 10^{-3} \text{ mm} = 0.00852 \text{ mm}$$

ii. Co-efficient of Friction (μ):

From PSG DDB Pg. No. 7.40, for β = 360°, L/D = 1, s = 0.013

$$\mu\times\frac{D}{C} = 1,\quad \mu = 1\times\frac{C}{D} = \frac{0.24}{100} = 2.4 \times 10^{-3}$$

iii. Power cost due to friction:

H_g = μ w v

$= 2.4 \times 10^{-3} \times 50{,}000 \times \frac{\pi\times100\times1440}{60}$

H_g = 904.8 W

4. Select a bearing for a 40 mm diameter shaft rotates at 400 rpm. Due to a bevel gear mounted in the shaft. The bearing will have to withstand a 5000 N radial load of the bearing thrust load. The life of the bearing expected to be at least 1000 hrs.

Given: d= 40mm, n=400 rpm, F_r = 5000N, F_a = 3000N, L_h = 1000 hrs

Solution:

Select Series 62 and for d = 40 mm, From PSG DDB Pg. No. 4.13, bearing basic design no. SKF 6208. The values of C_o, C are

C_o = 1600 kgf = 1600 x 10 = 16000 N
C = 2280 kgf = 2280 x 10 = 22800 N

i. Equivalent diameter load (P):

$P = (X.F_r + Y.F_a) \times s$

For X and Y values, from PSG DDB Pg. No. 4.4, F_a and e are given

$$\begin{array}{cc} F_a/C_o & e \end{array}$$

$$0.12\left\{\begin{matrix} 0.13 & 0.31 \\ 0.25 & 0.37 \end{matrix}\right\}0.06$$

$$\frac{0.12}{10} = 0.012, \text{and } \frac{0.06}{10} = 0.006$$

For $\frac{F_a}{C_o} = \frac{3000}{16000} = 0.1875 \cong 0.19$, For value of $\frac{F_a}{C_o}\left(0.19\right) = 0.13 + 5(0.012)$

Similarly, For 'e' value = $(5 \times 0.006 + 0.31) = 0.03 + 0.31 = 0.34$

$$\therefore \frac{F_a}{C_o} = 0.19$$

$$e = 0.34$$

$$\frac{F_a}{F_r} = \frac{3000}{5000} = 0.6 > e,$$

So, X value = 0.56, 's' value from PSG DDB Pg. No. 4.2
Y value = 1.2

Therefore, P = [(0.56 x 5000) + (1.2 x 3000)] x 1.2

P = 7680 N

ii. Dynamic Load capacity(C):

From PSG DDB Pg. No. 4.6 (Ball bearing), For 400 rpm and 1000 hrs life

C/P = 2.88

$$\frac{C}{7680} = 2.88$$

C = 2.88 x 7680 = 22118.4 N

This dynamic load is less than the tabulated (allowable) value i.e. 22800 N. So the suitable bearing designation is **SKF 6208.**

5. Select a suitable ball bearing to support the overhung countershaft. The shaft is 60 mm diameter and rotating at 1250 rpm. The bearing is to have 99% reliability corresponding to a life of 4000 hrs. The bearing is subjected to an equivalent radial load of 6000N.

Given:

d = 60 mm, n = 1250 rpm, Reliability = 99% = 0.99 = probability = p, L = 4000 hrs, F_r = 6000N.

Solution:

From PSG DDB Pg. No. 4.2 $\frac{L}{L'10} = \left[\frac{\ln(1/p)}{\ln(1/p_{10})}\right]^{1/b}$

Here, $\ln(1/p_{10}) = 0.1053$, L = 4000 hrs, b = 1.34, p = 0.99

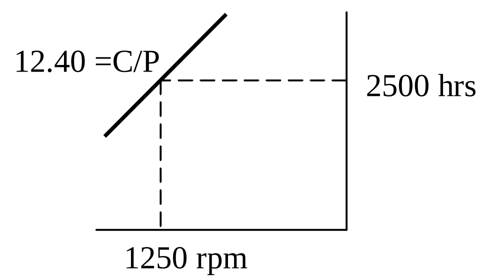
Substitute all value,

$$\frac{4000}{L'10} = (0.09544)^{0.7463}$$

L'10 = 23.093 hrs

From PSG DDB Pg. No. 4.6, For life 23.093 hrs and 1250 rpm,

The $\frac{C}{P}$ value is 12.40



$\frac{C}{P}$ = 12.4

C = 12.4 x F_r

C = 12.4 x 6000

C = 74400N

Select the bearing for C = 74400 N or C = 7440 kgf, and the diameter of the shaft is 60 mm. (From PSG DDB Pg. No. 4.15, series 64)

Result:

SKF 6412 is suitable bearing,

C_o = 7100 kgf,

C = 8450 kgf.

6. A 70mm machine shaft is to supported at ends. If operates continuously for 8 hrs per day ,320 days per year for 8 years the load of speed cycle for one of the hearing are given below,

S.No	Fraction of cycle	Radial load in N	Thrust	Speed rpm	Factors		
					X	y	Z
1.	0.25	3500	1000	600	0.56	1.2	1.5
2.	0.25	3000	1000	800	0.56	1.2	1.5
3.	0.5	4000	2000	900	0.56	1.4	1.5

Select suitable bearing.

Solution:

i. Equivalent load (p)

$P_1=(XFr+YFa)Z$

$=(0.56\times3500+1.2\times1000)1.5$

$=4740w$

$P_2=(0.56\times3000+1.2\times1000)1.5$

$=4320w$

$P_3=(0.56\times4000+1.4\times2000)1.5$

$=7560w$

rom pg 4.2:

$$\text{Cubic mean load Fm} = \left(\frac{p_1^3n_1t_1+p_2^3n_2t_2+p_3^3n_3t_3}{n_1+n_2+n_3}\right)^{1/3}$$

$$= \left(\frac{(4740)^3x600x0.25+(4320)^3x800x0.25+(7560)^3x900x0.5}{600+800+900}\right)$$

Fm = 4618.16 N

W.k.t: equivalent load = cubic mean load (p=fm)

!!) Equivalent speed (N)

$N = n_1t_1+n_2t_2+n_3t_3$

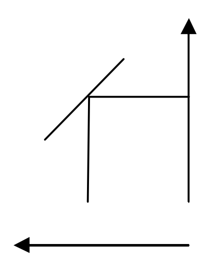
= 600x0.25+800x0.25+900x0.5

N = 800 rpm

!!!) Total life hrs = 8hrs/day , 320day s/yr For F yea
20480 h

= 8x320x8

= 20,480 hrs



800rpm

From pg no 4.6 , For lie 20.480 hrs &800 rpm

C/P= 9.83

C= 9.83x4618.16 = 45396 N

For, C= 45,396 N & d= 70mm

The suitable size of bearing in skf= 6214

7. A single row deep groove ball bearing no: 6002 is subjected to an axial thrust load of 1000N and a radial load of 2200N. find the expected life that 50% of the bearing will complete under this condition.

Given:

Deep groove ball bearing no: 6002

F_a= 1000N

F_a = 2200N

Solution:

From DDB:4.12 :- For bearing no. 6002

C_o = 255kgf = 255*10N

C = 440kgf = 4400N

F_a/c_o=1000/2550 = 0.392

From DDB 4.4 for Fa/co =0.392 the value of e=0.412

Since, f_a/c_o= 1000/2550 = 0.454>e

The radial load factor X= 0.56

Thrust load factor Y= 1.83

Service factor from DDB 4.2, S=1.1to 1.5 , say S=1.3

∴ Equivalent load

(from DD**B**: 4.2) P= $(XF_r+YF_a)S$

= (0.56*2200+1.83*1000)1.3

= 3980.6N

WKT,

L= $(c/p)^b = (4400/3980.6)^3$ =1.35 million revolution

Expected life at 50% reliability(L50) is obtained from

$$L50/L90 = \left(\frac{\ln(^1/R50)}{\text{Ln}(^1/R90)}\right)^{1/b}$$

$$L50/1.35 = \left(\frac{\ln(^1/0.5)}{\text{Ln}(^1/0.9)}\right)^{1/b} = (0.693/0.105)^{0,85} = 4.058$$

L50= 4.958*1.35

L50= 6.69 million rev

8. The load on the journal bearing is 150KN due to turbine of 300mm diameter running at 1800rpm determine the following

(1) Length of the bearing if the allowable bearing pressure is 1.6N/mm^2

(2) Amount of heat to be removed by the lubricant per minute if the bearing temperature is 60^{0}c and viscosity of the oil at 60^{0}c is 0.02kg/m-s and the bearing clearance is 0.25.

(NOV/DEC 2011)

Given:

W=150KN = 150X10^{3}N

D= 300mm=0.3m

N= 1800rpm

P= 1.6N/mm^2

Z=0.02kg/m-s

C= 0.25mm

Solution:

1) **Length of the bearing:**

Let, l = length of bearing(mm)

WKT, projected bearing area

A=l×d=lx300=300lmm^2

And alloeable bearing pressure (P),

$1.6 = \frac{W}{A} = \frac{150x10}{300l} = \frac{500}{l}$

l= 500/1.6 = 312.5mm

2) <u>Amount of heat to be removed by the lubricant:</u>

Wkt, co efficient of friction for the bearing

π= 33/10$^8\left(\frac{ZN}{P}\right)\left(\frac{D}{C}\right)$ + k

= 33/10$^8\left(\frac{0.02X1800)}{1.6}\right)\left(\frac{300}{0.25}\right)$ + 0.002

= 0.009+0.002= 0.011

Rubbing velacity,

V = $\frac{\pi DN}{60}$ = $\left(\frac{\pi\times0.3\times1800}{60}\right)$ = 28.3m/s

∴ Amount of heat to removed by the lubricant.

Q_8 = 0.11x150x10^{3}x28.3

= 46.695 J/s or W

= 46.695kw

9). Design a journal bearing for a centrifugal pump from the following data:

Load on the journal	**= 20000N**
Speed of the journal	**= 900 rpm**
Type of oil is	**= SAE10**
For which absolute viscosity at 55^{o}C	**= 0.017 kg/ms**
Ambient temperature of oil	**= 15.5^{o}C**
Maximum bearing pressure for the pump	**=1.5N/mm^2**

<u>Given:</u>

W = 20000N

N = 900rpm

T_o= 55^{o}C,t_a=15.5^{o}C

Z = 0.017 kg/ms, p = 1.5 N/mm^2

<u>Solution:</u>

1. **<u>To find length of the journal, (l):</u>**

Assume, dia and journal d=100mm

Take l/d =1.6

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

2. **Checking of bearing pressure:**

WKT,

$P = w/ld = 20000/160x100 = 1.25 < 1.5 N/mm^2$ (given)

The value of l&d is safe

3. **Bearing characteristic number (ZN/P):**

ZN/P = 0.017x900/1.25 = 12.24

WKT,

The minimum value of bearing modules 3K = ZN/P
Bearing module at the minimum point of friction:
K = 1/3(ZN/P) = 1/3x28 = 9.33 [ZN/P = 28 from table]
Since the calculated value13.24 is more than 9.33,
Therefore the bearing is operate under hydrodynamic condition.

4. **Clearance ratio (c/d):**

From the table c/d = 0.0013 (for centrifugal pump)

5. **Co-efficient of friction(μ):**

$$\mu = \frac{33}{108}\left(\frac{ZN}{P}\right)\frac{c}{d} + k$$

$$= \frac{33}{108} \text{ x } 12.24 \text{ x } \frac{1}{0.0013} + 0.002$$

$$= 0.0051 \text{ Ans}$$

6. **Heat generated (Q_g):**

$$Q_g = \mu wv$$

$$= \mu w \left(\frac{\pi dN}{60}\right) W$$

$$= 0.0051x20000 \left(\frac{\pi x0.1x900}{60}\right)$$

$$= 480.7 \text{ W}$$

7. **Heat dissipated (Q_d):**

$$Q_d = CA\,(t_b - t_a)$$

$$= cld\,(t_b - t_a)W$$

$$(t_b-t_a) = \frac{1}{2}(t_b-t_a) = \frac{1}{2}(55^o-15.5^o) = 19.75^oC$$

$$Q_d = 1232 \times 0.16 \times 0.1 \times 19.75$$

$$= 389.3\ W \qquad (l,d – \text{in meters})$$

∴ Amount of artificial cooling required

$$= Q_g-Q_d$$

$$= 4807.7 – 389.3$$

$$= 91.4\ W$$

10. A journal bearing is to be designed for a centrifugal pump for the following Data:

Load on the journal = 12kN,
Diameter of the journal = 75mm
Speed (N) = 1440rpm
Atmospheric temperature of the oil =16℃
Operating temperature of the oil = 60℃
Absolute viscosity of oil at 60 ℃ = 0.23 kg/ms
Give the systematic design of the bearing. **(MAY-JUNE-2012)**

Solution: (Solve this problem as per the procedure of previous problem)

11). A single row deep groove ball bearing is subjected to a radial force of 8kN and a thrust force of 3kN. the rotates at 1200rpm the expected life L_{10}^{th} of the bearing is 20000hr the minimum acceptable diameter of having for this application. **(MAY/JUNE 2012)**

Given:

F_n = 8kn

F_a = 3kn

N = 1200rpm

$L_{10}h$ = 20,000

d = 75mm

Solution:

STEP1: X and Y factor

$$X = 0.56\ ,\ Y= 1.5\ ,\ F_r = 8000N$$

$F_a = 3000N$

WKT,

$C = p(L_{10})^{1/3} = (8980)(1440)^{1/3}$

$= 101406.04N$

The shaft of 75mm diameter, bearing no.6315 (c= 112000) is suitable for the above data for this bearing,

$C_o = 72000N$

$\therefore , \left(\frac{Fa}{Fr}\right) = \left(\frac{3000}{8000}\right) = 0.375$

And $\left(\frac{Fa}{Co}\right) = \left(\frac{3000}{72000}\right) = 0.04167$

e = 0.24 (approximately) and $(\frac{Fa}{Fr}) > e$

The value of Y is obtained by liner interpolation .

$Y = 1.8 - (\frac{(1.8-1.6)}{(0.07-0.04)} \text{x} (0.0416 - 0.04) = 1.79$ and $X = 0.56$

STEP 2:

Dynamic load capacity

$P = XFr+YF = 0.56(8000)+ 1.79\ (3000)$

$= 9850N$

$C = p\ (L_{10})^{1/3} = 9850(1440)^{1/3} = 111230.46N$

STEP 3: Selection of bearing

Bearing No.6315 (c= 112000) is suitable for the above application.

Result:

The suitable bearing is 6315

12. Design a connecting rod for an I.C. engine running at 1800 r.p.m. and developing a maximum pressure of 3.15 N/mm^2. The diameter of the piston is 100 mm; mass of the reciprocating parts per cylinder 2.25 kg; length of connecting rod 380 mm; stroke of piston 190 mm and compression ratio 6:1. Take a factor of safety of 6 for the design. Take length to diameter ratio for big end bearing as 1.3 and small end bearing as 2 and the corresponding bearing pressures as 10 N/mm^2 and 15 N/mm^2. The density of material of the rod may be taken as 8000 kg/m^3 and the allowable stress in the bolts as 60 N/mm^2 and in cap as 80 N/mm^2. The rod is to be of I-section for which you can choose your own proportions. Draw a

neat dimensioned sketch showing provision for lubrication. Use Rankine formula for which the numerator constant may be taken as 320 N/mm^2 and the denominator constant 1 / 7500.

Solution. Given : N = 1800 r.p.m. ; p = 3.15 N/mm^2 ; D = 100 mm ; m_R = 2.25 kg ; l = 380 mm = 0.38 m ; Stroke = 190 mm ; *Compression ratio = 6 : 1 ; $F. S.$ = 6.

The connecting rod is designed as discussed below :

1. ***Dimension of I- section of the connecting rod***

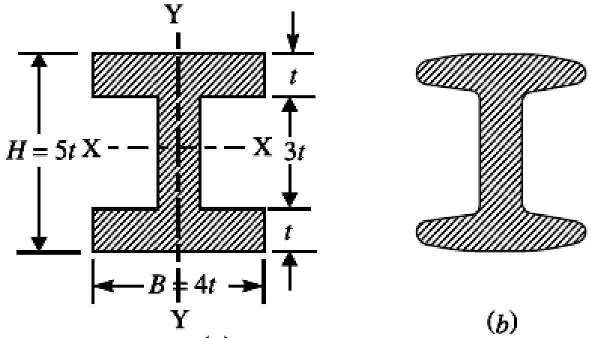
Let us consider an I-section of the connecting rod, as shown in Fig. 32.14 (a), with the following proportions :

Flange and web thickness of the section = t

Width of the section, $B = 4t$

and depth or height of the section,

$$H = 5t$$



(a) (b)

First of all, let us find whether the section chosen is satisfactory or not.

We have already discussed that the connecting rod is considered like both ends hinged for buckling about X-axis and both ends fixed for buckling about Y-axis. The connecting rod should be equally strong in buckling about both the axes. We know that in order to have a connecting rod equally strong about both the axes,

$$I_{xx} = 4\, I_{yy}$$

where I_{xx} = Moment of inertia of the section about X-axis, and

I_{yy} = Moment of inertia of the section about Y-axis.

In actual practice, I_{xx} is kept slightly less than $4\, I_{yy}$. It is usually taken between 3 and 3.5 and the connecting rod is designed for buckling about X-axis.

Now, for the section as shown in Fig. 32.14 (a), area of the section,

$$A = 2\,(4\,t \times t) + 3t \times t = 11\, t^2$$

$$I_{xx} = \frac{1}{12}\left[4t(5t)^3 - 3t \times (3t)^3\right] = \frac{419}{12} t^4$$

and

$$I_{yy} = 2 \times \frac{1}{12} \times t(4t)^3 + \frac{1}{12} \times 3t \times t^3 = \frac{131}{12} t^4$$

$$\therefore \quad \frac{I_{xx}}{I_{yy}} = \frac{419}{12} \times \frac{12}{131} = 3.2$$

Since $\dfrac{I_{xx}}{I_{yy}} = 3.2$, therefore the section chosen in quite satisfactory.

Now let us find the dimensions of this I-section. Since the connecting rod is designed by taking the force on the connecting rod (F_C) equal to the maximum force on the piston (F_L) due to gas pressure, therefore,

$$F_C = F_L = \frac{\pi D^2}{4} \times p = \frac{\pi (100)^2}{4} \times 3.15 = 24\ 740 \text{ N}$$

We know that the connecting rod is designed for buckling about X-axis (*i.e.* in the plane of motion of the connecting rod) assuming both ends hinged. Since a factor of safety is given as 6, therefore the buckling load,

$$W_B = F_C \times F.S. = 24\,740 \times 6 = 148\,440 \text{ N}$$

We know that radius of gyration of the section about X-axis,

$$k_{xx} = \sqrt{\frac{I_{xx}}{A}} = \sqrt{\frac{419\,t^4}{12} \times \frac{1}{11\,t^2}} = 1.78\ t$$

Length of crank,

$$r = \frac{\text{Stroke of piston}}{2} = \frac{190}{2} = 95 \text{ mm}$$

Length of the connecting rod,

$$l = 380 \text{ mm} \qquad \text{...(Given)}$$

$\therefore$ Equivalent length of the connecting rod for both ends hinged,

$$L = l = 380 \text{ mm}$$

Now according to Rankine's formula, we know that buckling load (W_B),

$$148\,440 = \frac{\sigma_c . A}{1 + a\left(\frac{L}{k_{xx}}\right)^2} = \frac{320 \times 11\,t^2}{1 + \frac{1}{7500}\left(\frac{380}{1.78\,t}\right)^2}$$

... (It is given that σ_c = 320 MPa or N/mm^2 and $a = 1 / 7500$)

$$\frac{148\,440}{320} = \frac{11\,t^2}{1 + \frac{6.1}{t^2}} = \frac{11\,t^4}{t^2 + 6.1}$$

$$464\,(t^2 + 6.1) = 11\,t^4$$

or $$t^4 - 42.2\,t^2 - 257.3 = 0$$

$\therefore$ $$t^2 = \frac{42.2 \pm \sqrt{(42.2)^2 + 4 \times 257.3}}{2} = \frac{42.2 \pm 53}{2} = 47.6$$

... (Taking +ve sign)

or $$t = 6.9 \text{ say } 7 \text{ mm}$$

Thus, the dimensions of I-section of the connecting rod are :

Thickness of flange and web of the section

$= t = 7$ mm **Ans.**

Width of the section, $B = 4\,t = 4 \times 7 = 28$ mm **Ans.**

and depth or height of the section,

$H = 5\,t = 5 \times 7 = 35$ mm **Ans.**

These dimensions are at the middle of the connecting rod. The width (B) is kept constant throughout the length of the rod, but the depth (H) varies. The depth near the big end or crank end is kept as $1.1H$ to $1.25H$ and the depth near the small end or piston end is kept as $0.75H$ to $0.9H$. Let us take

Depth near the big end,

$$H_1 = 1.2H = 1.2 \times 35 = 42 \text{ mm}$$

and depth near the small end,

$$H_2 = 0.85H = 0.85 \times 35 = 29.75 \text{ say } 30 \text{ mm}$$

∴ Dimensions of the section near the big end

$$= 42 \text{ mm} \times 28 \text{ mm} \textbf{ Ans.}$$

and dimensions of the section near the small end

$$= 30 \text{ mm} \times 28 \text{ mm} \textbf{ Ans.}$$

Since the connecting rod is manufactured by forging, therefore the sharp corners of *I*-section are rounded off, as shown in Fig. 32.14 (*b*), for easy removal of the section from the dies.

2. *Dimensions of the crankpin or the big end bearing and piston pin or small end bearing*

Let d_c = Diameter of the crankpin or big end bearing,

l_c = length of the crankpin or big end bearing $= 1.3\, d_c$ ...(Given)

p_{bc} = Bearing pressure = 10 N/mm^2 ...(Given)

We know that load on the crankpin or big end bearing

$$= \text{Projected area} \times \text{Bearing pressure}$$

$$= d_c \cdot l_c \cdot p_{bc} = d_c \times 1.3\, d_c \times 10 = 13\, (d_c)^2$$

Since the crankpin or the big end bearing is designed for the maximum gas force (F_L), therefore, equating the load on the crankpin or big end bearing to the maximum gas force, *i.e.*

$$13\, (d_c)^2 = F_L = 24\,740 \text{ N}$$

∴ $(d_c)^2 = 24\,740 / 13 = 1903$ or $d_c = 43.6$ say 44 mm **Ans.**

and $l_c = 1.3\, d_c = 1.3 \times 44 = 57.2$ say 58 mm **Ans.**

The big end has removable precision bearing shells of brass or bronze or steel with a thin lining (1mm or less) of bearing metal such as babbit.

Again, let d_p = Diameter of the piston pin or small end bearing,

l_p = Length of the piston pin or small end bearing $= 2d_p$ ...(Given)

p_{bp} = Bearing pressure = 15 N/mm^2 ..(Given)

We know that the load on the piston pin or small end bearing

$$= \text{Project area} \times \text{Bearing pressure}$$

$$= d_p \cdot l_p \cdot p_{bp} = d_p \times 2\, d_p \times 15 = 30\, (d_p)^2$$

Since the piston pin or the small end bearing is designed for the maximum gas force (F_L), therefore, equating the load on the piston pin or the small end bearing to the maximum gas force,

i.e.

$$30\, (d_p)^2 = 24\,740 \text{ N}$$

∴ $(d_p)^2 = 24\,740 / 30 = 825$ or $d_p = 28.7$ say 29 mm **Ans.**

and $l_p = 2\, d_p = 2 \times 29 = 58$ mm **Ans.**

The small end bearing is usually a phosphor bronze bush of about 3 mm thickness.

3. ***Size of bolts for securing the big end cap***

Let d_{cb} = Core diameter of the bolts,

σ_t = Allowable tensile stress for the material of the bolts

$= 60 \text{ N/mm}^2$ ...(Given)

and n_b = Number of bolts. Generally two bolts are used.

We know that force on the bolts

$$= \frac{\pi}{4}(d_{cb})^2 \sigma_t \times n_b = \frac{\pi}{4}(d_{cb})^2 \; 60 \times 2 = 94.26\,(d_{cb})^2$$

The bolts and the big end cap are subjected to tensile force which corresponds to the inertia force of the reciprocating parts at the top dead centre on the exhaust stroke. We know that inertia force of the reciprocating parts,

$$F_I = m_R \,.\, \omega^2 \,.\, r \left(\cos\theta + \frac{\cos 2\theta}{l/r}\right)$$

We also know that at top dead centre on the exhaust stroke, $\theta = 0$.

$$\therefore \quad F_I = m_R \,.\, \omega^2 \,.\, r\left(1 + \frac{r}{l}\right) = 2.25\left(\frac{2\pi \times 1800}{60}\right)^2 0.095\left(1 + \frac{0.095}{0.38}\right) \text{N}$$

$$= 9490 \text{ N}$$

Equating the inertia force to the force on the bolts, we have

$$9490 = 94.26\,(d_{cb})^2 \quad \text{or} \quad (d_{cb})^2 = 9490 / 94.26 = 100.7$$

$$\therefore \quad d_{cb} = 10.03 \text{ mm}$$

and nominal diameter of the bolt,

$$d_b = \frac{d_{cb}}{0.84} = \frac{10.03}{0.84} = 11.94$$

say 12 mm **Ans.**

4. ***Thickness of the big end cap***

Let t_c = Thickness of the big end cap,

b_c = Width of the big end cap. It is taken equal to the length of the crankpin or big end bearing (l_c)

= 58 mm (calculated above)

σ_b = Allowable bending stress for the material of the cap

$= 80 \text{ N/mm}^2$ ...(Given)

The big end cap is designed as a beam freely supported at the cap bolt centres and loaded by the inertia force at the top dead centre on the exhaust stroke (*i.e.* F_I when $\theta = 0$). Since the load is assumed to act in between the uniformly distributed load and the centrally concentrated load, therefore, maximum bending moment is taken as

$$M_C = \frac{F_I \times x}{6}$$

where x = Distance between the bolt centres

= Dia. of crank pin or big end bearing + 2 × Thickness of bearing liner + Nominal dia. of bolt + Clearance

$= (d_c + 2 \times 3 + d_b + 3)$ mm $= 44 + 6 + 12 + 3 = 65$ mm

∴ Maximum bending moment acting on the cap,

$$M_C = \frac{F_I \times x}{6} = \frac{9490 \times 65}{6} = 102\ 810 \text{ N-mm}$$

Section modulus for the cap

$$Z_C = \frac{b_c\,(t_c)^2}{6} = \frac{58(t_c)^2}{6} = 9.7\ (t_c)^2$$

We know that bending stress (σ_b),

$$80 = \frac{M_C}{Z_C} = \frac{102\ 810}{9.7\ (t_c)^2} = \frac{10\ 600}{(t_c)^2}$$

∴ $(t_c)^2 = 10\ 600 / 80 = 132.5$ or $t_c = 11.5$ mm **Ans.**

Let us now check the design for the induced bending stress due to inertia bending forces on the connecting rod (*i.e.* whipping stress).

We know that mass of the connecting rod per metre length,

$$m_1 = \text{Volume} \times \text{density} = \text{Area} \times \text{length} \times \text{density}$$

$$= A \times l \times \rho = 11t^2 \times l \times \rho \qquad ...(\because A = 11t^2)$$

$$= 11(0.007)^2\,(0.38)\,8000 = 1.64 \text{ kg}$$

$$...[\because \rho = 8\,000 \text{ kg/m}^3 \text{ (given)}]$$

$\therefore$ Maximum bending moment,

$$M_{max} = m\,.\,\omega^2\,.\,r \times \frac{l}{9\sqrt{3}} = m_1\,.\,\omega^2\,.\,r \times \frac{l^2}{9\sqrt{3}} \qquad ...(\because m = m_1 \cdot l)$$

$$= 1.64\left(\frac{2\pi \times 1800}{60}\right)^2 (0.095)\,\frac{(0.38)^2}{9\sqrt{3}} = 51.3 \text{ N-m}$$

$$= 51\,300 \text{ N-mm}$$

and section modulus, $Z_{xx} = \dfrac{I_{xx}}{5t/2} = \dfrac{419\,t^4}{12} \times \dfrac{2}{5\,t} = 13.97\,t^3 = 13.97 \times 7^3 = 4792 \text{ mm}^3$

$\therefore$ Maximum bending stress (induced) due to inertia bending forces or whipping stress,

$$\sigma_{b(max)} = \frac{M_{max}}{Z_{xx}} = \frac{51\,300}{4792} = 10.7 \text{ N/mm}^2$$

Since the maximum bending stress induced is less than the allowable bending stress of 80 N/mm^2, therefore the design is safe.

REVIEW QUESTIONS

2 MARKS

1. List the basic assumption used in the theory of hydrodynamic lubrication?

(NOV/DEC 2011)

2. Classify the sliding contact bearings according to the thickness of layer of the lubricant between the bearing and journal. **(MAY/ JUNE 2012)**

3. What are various types of radial ball bearing? **(MAY/ JUNE 2012)**

4. What do you meant by life of an individual bearing? **(MAY/ JUNE 2013)**

16 MARKS

1. Design of journal bearing for 12 MW, 1000 rpm steam turbine, which is supported by two bearings. Take atmospheric temperature as 16^o C and operating temperature of oil as 60^o C. assume viscosity of oil as 23 Ns/m^2. **(MAY/JUNE 2012)**