GANPAT UNIVERSITY
B. Tech. Sem. - VIII Mechatronics Engineering
2MC804 Design of Mechanical Systems
(CBCS) Regular Examination April - June 2015

[Time: 3 Hour] **[Total Marks: 70]**

Instructions:

(1) Attempt all questions.
(2) Assume suitable data if necessary.
(3) Figures to the right indicate full marks.
(4) Only scientific calculator is allowed.
(5) Use of design data book is strictly restricted.

SECTION - I

Que. 1 Attempt the followings.

(a) Define basic static capacity and rating life of bearing. Derive equation for load-life relationship for rolling contact bearing. [4]

(b) A ball bearing used in truck carries radial load of 5 KN and the expected life for 90% of the bearings is 8000 hr. Calculate the dynamic load carrying capacity of the bearing. [4]

(c) A set of taper roller bearing having face to face arrangement in which F_{r1} = 800 N, F_{r2} = 1000 N and F_a = 50 N. The axial force acting towards left to right. Assume bearing pairs to be 32007X, having C = 40200 N, e = 0.46 and y = 1.3 and K_a = 1. Calculate the equivalent load for the given bearing. [4]

OR

Que. 1 Attempt the followings.

(a) Average life of bearing is 60 million revolutions and it is running at 1000 rpm. Find average life of bearing in hours. Aldo find equivalent dynamic load if the bearing selected from manufacturer's catalog of number 6205. [4]

(b) A shaft supported by single row deep groove ball bearings which are rotates at 1200 rpm. Bearing is subjected to a radial force of 8000 N and as thrust force of 3000 N. The expected life of the bearing is 20000 hrs. The minimum acceptable diameter of the shaft is 75 mm. Select a suitable ball bearing for this application. [8]

Que. 2 Attempt the followings.

(a) Derive equation for equivalent dynamic load for bearing under cyclic loads with neat sketch. [4]

(b) A single row deep groove ball bearing is subjected to the work cycles as given below. If the desired rating life of the bearing is 15000 hours, select the bearing from the manufacturer's catalog. What is the average speed of the bearing? [8]

Sr. No.	Fraction of cycle	Radial Load 'F_r' KN	Thrust Load 'F_a' KN	Radial Factor 'X'	Thrust Factor 'Y'	Race Rotating	Service Factor	Speed rpm
1	1/10	1.5	0.25	1.0	0.0	Inner	1.2	400
2	1/5	1.0	0.75	0.56	2.0	Outer	1.8	500
3	3/5	5.0	1.1	0.56	2.0	Inner	1.5	600
4	remaining	1.0	---	1.0	0.0	Outer	2.0	800

OR

Que. 2 Attempt the followings.

(a) Two identical ball bearings A and B are used in two different applications. The load on the bearing B is half of that on bearing A. The remaining conditions are identical. What will be the expected life of the bearing B as compared to the life of bearing A? [4]

(b) A shaft with centrally mounted helical pinion is supported by deep-groove ball bearings at [8] both ends. The center distance between the bearings is 100 mm. The shaft transmits 5 kW power at 3000 RPM. The pitch circle diameter of the pinion is 80 mm. The normal pressure angle and helix angle are 20° and 19° respectively. The expected life of the bearings is 8000 hours with a reliability of 95%. Calculate the dynamic basic capacity of the bearing which takes up the axial thrust, so that it can be selected from the manufacturer's catalogue based on a reliability of 90%. Take shock load factor as 1.25, radial factor of 0.56 and thrust facture is equal to 1.2.

Que. 3

(a) Attempt the followings (Any one). [4]

(i) Classify the sliding contact bearing. Explain the working of hydrostatic bearing with neat sketch and write their applications, advantages and limitations.

(ii) Explain the frictional power loss in hydrostatic step bearing and prove that it is inversely proportional to the optimum oil film thickness, start the derivation with Newton's equation of viscosity.

(b) The following data is given for a hydrostatic thrust bearing: [7]
Shaft speed = 720 rpm, supply pressure = 5 MPa, shaft diameter = 400 mm, recess diameter = 250 mm, film thickness = 0.15 mm, viscosity of lubricant = 30 cP, specific heat of lubricant = 1.76 KJ/kg°C, and specific gravity of lubricant = 0.86, calculate: (i) load carrying capacity of the bearing, (ii) flow required in lit/min, (iii) frictional power loss, (iv) pumping power loss, and (v) temperature rise.
Assume that the total power loss in the bearing is converted into frictional heat.

SECTION-II

Que. 4 Attempt the followings.

(a) Explain Johnson's method of optimum design with different three forms of equations. [4]

(b) Design a tensile bar of length, L = 200 mm to carry a tensile load of 5 KN for minimum cost, out of the following materials. Take factor of safety as 3. **(Ref. Fig. A.)** [8]

Material	**Mass density ρ, (kg/m^3)**	**Material cost Per Unit Mass c, (Rs/kg)**	**Yield Strength S_{yt}, (N/mm^2)**
Steel	7500	14	400
Aluminium Alloy	2800	66	150
Titanium Alloy	4500	1100	800
Magnesium Alloy	1800	75	100

OR

Que. 4 Attempt the followings.

(a) Define normal specifications. Explain the procedural step of optimum design for normal specifications. [4]

(b) A thin spherical pressure vessel is subjected to an internal pressure of 4 N/mm^2. The mass of the empty vessel should not exceed 125 kg. If the factor of safety is 3.0, design the pressure vessel with the objective of maximizing the gas storage capacity, out of the following materials: [8]

Material	**Mass density ρ, (kg/m^3)**	**Ultimate Tensile Strength S_{ut}, (N/mm^2)**
Low Alloy Steel-15 Cr90 Mo55	7800	14
Aluminium Alloy-74530	2800	66
Copper Alloy- Cu Ni31 Mn 1 Fe	8400	1100

Que. 5 Attempt following.

(a) It is required to design a pair of spur gear with 20^0 full depth involute teeth consisting of a 20 teeth pinion meshing with a 50 teeth gear. The pinion shaft is connected to a 10 kW, 1440 rpm electric motor. The starting torque of the motor can be taken as 150% of the rated torque. The material of the pinion is plain carbon steel Fe410 (S_{ut} = 410 N/mm^2), while the gear is made of grey cast iron FG250 (S_{ut} = 250 N/mm^2). The factor of safety is 1.5. Design the gear based on Lewis equation and using velocity factor to account for the dynamic load. [8]

Z	Y	Z	Y	Z	Y	Z	Y	Z	Y	Z	Y
16	0.295	20	0.320	24	0.337	28	0.352	33	0.367	40	0.389
17	0.302	21	0.326	25	0.340	29	0.355	35	0.373	45	0.399
18	0.308	22	0.330	26	0.344	30	0.358	37	0.380	50	0.408
19	0.314	23	0.333	27	0.438	32	0.364	39	0.386		

(b) Explain factor affecting on pitting failure. [3]

OR

Que. 5 Attempt following.

(a) It is required to design a pair of spur gear with 20^0 full depth involute teeth based on Lewis equation. The velocity factor is to be used account for dynamic load. The pinion shaft is connected to a 10 kW, 1440 rpm motor. The starting torque of the motor is 150% of the rated torque. The speed reduction is 4:1. The pinion as well as the gear are made of plain carbon steel 40C8 (S_{ut} = 600 N/mm^2). The factor of safety can be taken as 1.5. Design the gears, specify their dimensions and suggest suitable surface hardness for the gears. [11]

Que. 6 Attempt following.

(a) A planetary gear train is shown in Fig.(B) . The sun gear A rotates in a clockwise direction and transmit 7.5 KW power at 1440 rpm to the gear train. The number of teeth on sun gear A, planet gear B and the fixed ring C are 40, 80 and 180 respectively. The module is 4 mm and the pressure angle 20^0 . Draw a free body diagram of forces acting on each gear and calculate the torque that the arm D can deliver to its output shaft. [10]

(b) Evaluate Parallel and Crossed helical gear. [2]

Table 1: Equivalent loads with taper roller bearing mountings

Arrangement	Load Case	Equivalent axial loads
Back to Back OR Face to Face	$F_{r1} \geq F_{r2}$ $F_a \geq 0$	$F_{a1} = I = F_{r1}/2y$ $F_{a2} = F_a + I$
	$F_{r1} < F_{r2}$ $F_a \geq 0.5(F_{r2} - F_{r1})$	$F_{a1} = I = F_{r1}/2y$ $F_{a2} = F_a + I$
	$F_{r1} < F_{r2}$ $F_a < 0.5(F_{r2} - F_{r1})$	$F_{a2} = I = F_{r2}/2y$ $F_{a1} = I - F_a$
Back to Back OR Face to Face	$F_{r1} < F_{r2}$ $F_a \geq 0$	$F_{a2} = I = F_{r2}/2y$ $F_{a1} = F_a + I$
	$F_{r1} > F_{r2}$ $F_a \geq 0.5(F_{r1} - F_{r2})$	$F_{a2} = I = F_{r2}/2y$ $F_{a1} = F_a + I$
	$F_{r1} > F_{r2}$ $F_a < 0.5(F_{r1} - F_{r2})$	$F_{a1} = I = F_{r1}/2y$ $F_{a2} = I - F_a$

Table 2: Radial and thrust factors for single –row deep groove ball bearing

F_a/C_0	F_a/C_0		$(F_a/VF_r) \geq e$		e
	X	Y	X	Y	
0.025	1	0	0.56	2.0	0.22
0.04	1	0	0.56	1.8	0.24
0.07	1	0	0.56	1.6	0.27
0.13	1	0	0.56	1.4	0.31
0.25	1	0	0.56	1.2	0.37
0.5	1	0	0.56	1.0	0.44

Table 3: Dimensions and basic capacities of single-row deep-groove ball bearings.

Bearing No.	Principal Dimensions			Basic Capacity	
	Bore 'd' mm	Outside Diameter 'D' mm	Width 'B' mm	Static 'Co' KN	Dynamic 'C' KN
6005	25	47	12	6.55	11.20
6205	25	52	15	7.80	14.00
6305	25	62	17	11.60	22.50
6405	25	80	21	19.30	35.80
6011	55	90	18	21.20	28.10
6211	55	100	21	29.00	43.60
6311	55	120	29	45.00	71.50
6411	55	140	33	62.00	99.50
6014	70	110	20	31.00	37.70
6214	70	125	24	45.00	60.50
6314	70	150	35	68.00	104.00
6414	70	180	42	104.00	143.00

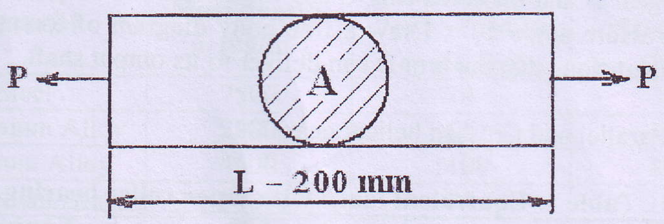

Fig. (A)

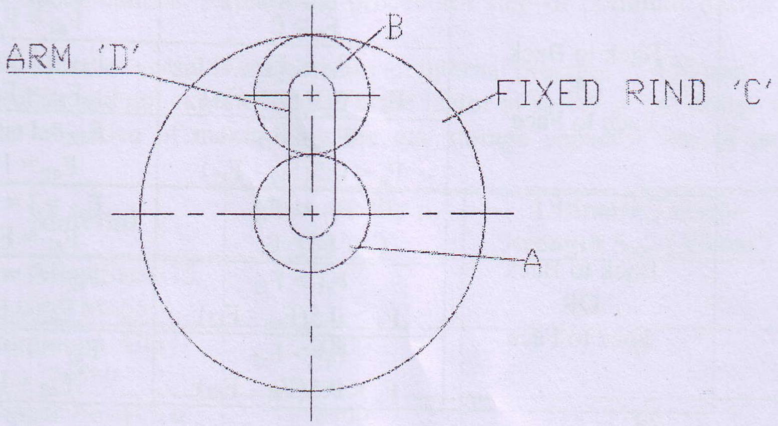

Fig. (B)

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