GANPAT UNIVERSITY

B. Tech. Sem.-VII Mechanical Engineering

Regular Examination Nov./Dec. - 2012

ME702 - Design of Mechanical Systems

[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.

SECTION - I

Que. 1

(A) Draw the sectional line diagram for different types of roller bearings and explain them with their loading capacity. [4]

(B) In a particular application, the radial load acting on a ball bearing is 5 KN and the expected life for 90% of the bearings is 8000 hr. Calculate the rating life of the bearing, when the shaft rotates at 1450 rpm. If same bearing is used for other application with 95% reliability then determine the life of bearing using statistical distribution. [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 = 650 N. 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

A 45 mm shaft carrying two pulleys is as shown in fig. (A). Pulley A receives power anc transmits to pulley B at 1440 rpm. The ratio of tensions in the belt is 2:1 and the maximun tension in each belt is limited to 3 KN. Select a radial deep groove ball bearing with 45 mr bore diameter, such that the life of the bearing is 8000 hrs. Assume a load factor of 2. [12]

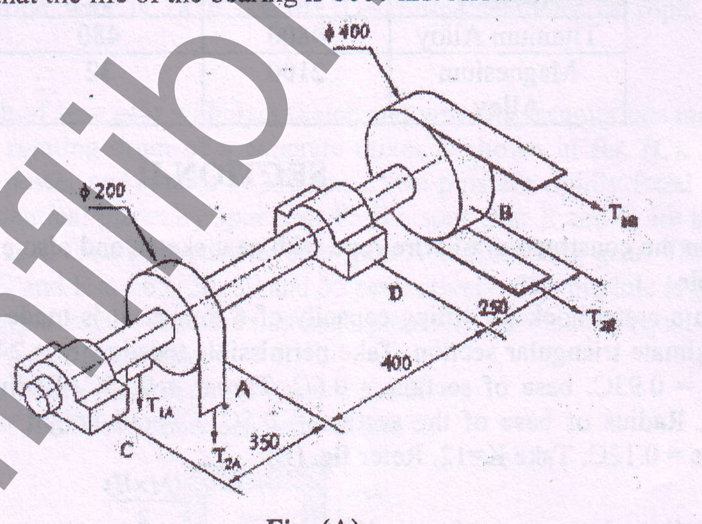

Fig. (A)

Que. 2

(A) Explain optimum oil film thickness in the context of sliding contact bearing. [4]

(B) The following data is given for the hydrostatic step bearing: [8]
Thrust load = 450 KN, Shaft speed = 750 rpm, Shaft radius = 200 mm, Recess diameter = 250 mm, Specific gravity of lubricant = 0.86, Specific heat of lubricant = 2 KJ/Kg.°C, Viscosity of lubricant = 30 cP, Calculate: (i) The optimum oil film thickness for minimum power loss (ii) Frictional power loss (iii) pumping power loss; and (iv) Temperature rise. Assume that the total power loss is converted into frictional heat.

OR

(P. T. O)

Que. 2

(A) Explain Johnson's method of optimum design with different three forms of equations and also write the basic steps in method of optimum design. **[4]**

(B) The bore of a cylinder of the four stroke diesel engine is 150 mm. The maximum gas pressure inside the cylinder is limited to 3.5 MPa. The cylinder head is made of grey cast iron FG 200 (σ_{ut} = 200 N/mm^2) and the factor of safety is 5, Determine the thickness of the cylinder head. **[8]**
Studs are used to fix the cylinder head to the cylinder and obtained a leak proof joint. They are made of steel FeE 250 (σ_{yt} = 250 N/mm^2) and the factor of safety is 5. Calculate:
(i) Number of studs, (ii) Nominal diameter of studs, and (iii) Pitch of studs.

Que. 3

(A) **Attempt the followings (Any one).** **[4]**
(1) Explain boundary lubrication, mixed lubrication and hydrodynamic lubrication with neat sketch.
(2) Enlist the main functions of the piston. Explain the construction of trunk type piston with neat sketch.

(B) In a light weight equipment, a shaft is transmitting a torque of 900 Nm and is to have a rigidity of 90 Nm/degree. Assume a factor of safety of 1.5 based on yield stress. Design the shaft with minimum weight. What will be the change in design for minimum cost? Assume maximum shear stress theory of failure. Use the following data for the materials. **[7]**

Material	Mass density (kg/m^3)	Material cost (Rs/N weight)	Yield strength (MPa)
Steel Alloy	8500	16	130
Al-Alloy	3000	32	50
Titanium Alloy	4800	480	90
Magnesium Alloy	2100	32	20

SECTION-II

Que. 4

(A) Explain the construction of wire rope with neat sketch and also explain its designation with example. **[4]**

(B) Design a crane hook for lifting capacity of 6 tonnes. It is made from forged steel and has approximate triangular section. Take permissible tensile stress 245 N/mm^2 for forged steel. Depth = 0.93C, base of section = 0.6C, Throat = 0.75C, Radius of curvature of hook = 1.25C, Radius of base of the section = 0.5C, Overall height of hook = 2.75C, Radii at corners = 0.12C. Take K=12. Refer **fig. (B)**. **[8]**

Take radius of curvature of neutral axis $= \dfrac{\frac{(M\times H)}{2}}{\left[\left(\frac{M\times r_o}{H}\right) \log_e \frac{r_o}{r_1} - M\right]}$

Maximum stress induced at the intrados of section due to bending,

$$(\sigma_t)_{ri} = \frac{W \times R \times h_1}{A \times e \times r_i}$$

Maximum compressive stress at extrados,

$$(\sigma_c)_{ro} = \frac{W \times R \times h_2}{A \times e \times r_o}$$

(P.T.O)

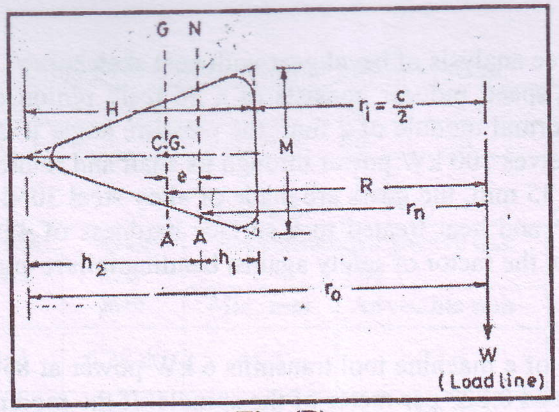

Fig. (B)

OR

Que. 4

(A) Explain design of hook with neat sketch. **[4]**

(B) In an oil drilling rig a 6 × 37 wire rope with pulley tackle is used for providing a lifting capacity of 10 tonnes. The tackle has three pulleys at the top and same at the bottom. The ultimate tensile strength of the rope is 1600 MPa and Er = 1.05×10^5 MPa. Design the rope giving a neat sketch of the hoisting arrangement. Assume 10% overload. Take C = 1.05, factor of safety as 4, wire diameter = 0.0045d **[4]**

(C) A wire rope used to lift the cage of a vertical mine hoist 410 m deep is 6 × 19. The weight of cage is 1200 kg and it has to lift 3000 kg of ore at a speed of 16 m/s which is to be attained in 10 seconds. Take $D_{min}/d = 75$. Assume factor of safety of 5 and E = 8.1×10^4 N/mm^2. Weight of rope/metre = $0.0034d^2$ kg. Cross-sectional area of rope = $0.38d^2$. Ultimate stress for rope wire is 1820 N/mm^2. Determine diameter of rope. Take d_w = 0.0063d m. **[4]**

Que. 5

(A) Explain beam strength of spur gear with neat sketch and write its assumptions made. **[4]**

(B) The gearbox for the rotating drum of a concrete mixer is shown in **fig. (C)**. The mixing drum receives 5 kW power and rotates at 250 rpm. Two pins are rigidly fixed to the drum and each carries an identical planetary spur gear F. The spur gear E and C are integral with the shaft and rotates at the same speed. The spur gear A is fixed ring gear. The number of teeth on gears A, B, C and E are 65, 20, 80 and 35 respectively. The module is 5 mm for all gears. The pressure angle is 20°. Assume that each planetary gear shares equal part of load and neglect frictional losses. Calculate: **[7]**

(i) Components of tooth force between gears E and F.

(ii) Components of tooth force between gears B and C.

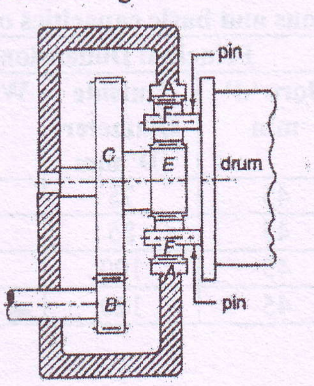

Fig. (C)

OR

(P.T.O)

Que. 5

(A) Explain the force analysis of bevel gear with neat sketch. **[4]**

(B) A herringbone speed reducer consists of a 26 teeth pinion driving a 104 teeth gear. The gears have a normal module of 2 mm. the pressure angle is 20° and the helix angle is 25°. The pinion receives 100 kW power through its shaft and rotates at 3600 rpm. The face width of each half is 35 mm. the gears are made of alloy steel 30Ni4Cr1 having ultimate strength = 1500 N/mm^2 and heat treated to a surface hardness of 450 BHN. The service factor is 1.25. Determine the factor of safety against bending failure and against pitting failure. **[7]**

Que. 6 **[12]**

(A) A steel spindle of a machine tool transmits 6 kW power at 850 rpm. The angular deflection should not exceed 0.22° per meter of the spindle. If the modulus of rigidity for the material of the spindle is 84 GPa. Find the diameter of the spindle.

(B) Enlist atleast six different types of gear tooth failure and explain any two in brief

(C) A 21 teeth straight bevel pinion rotating at 720 rpm transmits 10 KW power to a 40 teeth bevel gear. The module is 6 mm and the pressure angle is 20°. The face width is 45 mm. If the shaft angle is 90°, determine the components of force acting on meshing teeth.

Table 1: Equivalent loads with taper roller bearing mountings

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

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
6009	45	75	16	14.60	20.80
6209	45	85	19	21.60	33.20
6309	45	100	25	31.50	52.70
6409	45	120	29	45.00	76.10

Table 4: Standard bed diameter of hook

Std. bed dia. (mm)	75, 80, 85, 90, 95, 100, 105, 108, 110

(P.T.O).

Table 5: No. of bands for pulley system

No. of bands	1	2	3	4	5	6	7	8	9	10
D_{min}/d	16	20	23	25	26.5	28	30	31	32	33

Table 6

Rope	Ultimate strength N	Wire dia. d_w mm	Area (A) mm	Recommended sheave dia. (D)		Applications
				Min. mm	Advisable mm	
6 x 7	480 d^2	0.11 d	0.38 d^2	45 d	72 d	Mines, hailage, Trameways
6 x 19	510 d^2	0.07 d	0.4 d^2	30 d	45 d	Cargo cranes
6 x 37	480 d^2	0.045 d	0.4 d^2	18 d	27 d	Cranes
8 x 19	440 d^2	0.05 d	0.35 d^2	21 d	31 d	Extra flexible hoisting rope

Table 7: Values of the Lewis form factor Y for 20 full depth Involute system

z	Y	z	Y	z	Y
15	0.289	27	0.348	55	0.415
16	0.295	28	0.352	60	0.421
17	0.302	29	0.355	65	0.425
18	0.308	30	0.358	70	0.429
19	0.314	32	0.364	75	0.433
20	0.320	33	0.367	80	0.436
21	0.326	35	0.373	90	0.442
22	0.330	37	0.380	100	0.446
23	0.333	39	0.386	150	0.458
24	0.337	40	0.389	200	0.463
25	0.340	45	0.399	300	0.471
26	0.344	50	0.408	Rack	0.484

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