Total No. of Questions : 12]

SEAT No. :

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[5153]-16
T.E. (Mechanical Engineering)
MACHINE DESIGN - II
(2008 Course) (Semester - II) (302047)

Time : 4 Hours] *[Max. Marks : 100*

Instructions to the candidates:

1) ***Answers to the two sections should be written in separate answer books.***
2) ***Answer any three questions from each section***
3) ***Neat diagrams must be drawn wherever necessary.***
4) ***Figures to the right indicate full marks.***
5) ***Use of calculator is allowed.***
6) ***Assume suitable data, if necessary.***

SECTION - I

Q1) An intermediate shaft of a two stage co-axial gear box, shown in Fig.1, receives 10KW power at 288 rpm through right hand helical gear and transmits it to the output shaft through the spur pinion. The pitch circle diameters of helical gear and spur pinion are 450 mm and 108 mm respectively. The helix angle and normal pressure angle for helical gear are 23° and 20° respectively. The pressure angle for the spur pinion is 20°. The diameters of shaft at bearings A and B are 50 mm and 60 mm respectively. The load factor is 1.8 and the expected rating life of the bearings is 25000 hours. Select the deep groove ball bearings at A and B. The bearings are mounted such that bearing at A takes the thrust load. **[16]**

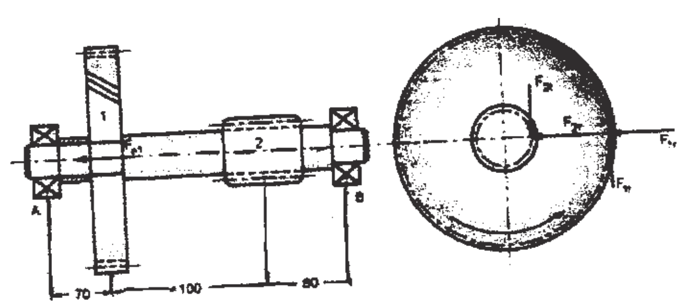

Fig.1

Basic Capacities of Single Row Deep Groove Ball Bearings

Bearing Number	Basic Capacity	
	Static C_0 KN	Dynamic C KN
6010	16.00	21.60
6210	23.20	35.10
6312	52.00	81.90

Radial & Thrust Factors for Single Row Deep Groove Ball Bearings

F_a/C_0	$(F_a/V\ F_r) \leq e$		$(F_a/V\ F_r) > e$		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

OR

Q2) A single-row deep groove ball bearing is subjected to following work cycle **[16]**

Fraction of cycle	Radial Load 'Fr' kN	Thrust Load 'Fa' kN	Radial factor 'X'	Thrust Factor 'Y'	Race Rotating	Service Factor	Speed RPM
1/10	1.5	0.25	1.0	0	inner	1.2	400
1/5	1.0	0.75	0.56	2.0	outer	1.8	500
3/5	5.0	1.1	0.56	2.0	inner	1.5	600
Remaining	1.0	-	1.0	0	outer	2.0	800

If desired rating life of bearing is 15,000 Hrs. Select bearing from following data.

Bearing No.	6011	6211	6311	6411
Dynamic capacity 'C' kN	28.1	43.6	71.5	99.5

Q3) Design a full hydrodynamic journal bearing with the following specification for machine tool application: **[18]**

Journal diameter = 75 mm

Radial load = 10 KN

Journal Speed = 1440 rpm

Minimum oil film thickness = 25.5 microns

Permissible unit bearing pressure = 2 N/mm^2

Inlet temperature = 40°C

Radial clearance = 0.001 (r) mm

Mass density of lubricant = 860 Kg/m^3

Specific heat capacity = 1.76 KJ/kg°k

Bearing material = Babbit

Determine the length of the bearing and select suitable oil for this application

l/d	h_0/c	ε	S	(r/c) f	$Q/rcn_s l$	Q_s/Q	P_{max}/p
1	0.4	0.6	0.121	3.22	4.33	0.680	2.409
1	0.6	0.4	0.264	5.79	3.99	0.497	2.066

OR

Q4) The following data is given for 360° hydrodynamic bearing: **[18]**

- Journal diameter = 100 mm
- Bearing length = 50 mm
- Journal Speed = 1500 r.p.m
- Minimum oil-film thickness = 15 microns
- Viscosity of lubricants = 30 CP
- Specific gravity of lubricant = 0.86
- Specific heat of lubricant = 2.09 KJ/kg°C
- Fit between the journal and bearing is normal running fit H_7e_7.

Calculate:

a) The load carrying capacity of bearing;

b) The coefficient of friction;

c) The power lost in friction;

d) The total flow rate of the lubricant;

e) The side leakage; and

f) The temperature rise.

Diameter, mm	Tolerances, mm	
	H_7	e_7
100	+0.035 +0.00	–0.072 –0.107

Dimensionless Parameters for full journal bearings.

$\frac{l}{d}$	$\frac{h_0}{c}$	ε	S	$\left(\frac{r}{c}\right)f$	$\frac{Q}{r c n_s l}$	$\frac{Q_s}{Q}$	$\frac{P_{max}}{p}$
1/2	0.2	0.8	0.0923	3.26	5.41	0.874	3.745
1/2	0.4	0.6	0.319	8.10	4.85	0.730	2.739

Q5) A machine component is subjected to a completely reversed bending stresses cycle consisting of following parts” **[16]**

$\pm$300 MPa for 30% of time

$\pm$275 MPa for 25% of time

$\pm$400 MPa for 10% of time

$\pm$325 MPa for 25% of time

No load for remaining cycle

The material properties are the follows: UTS = 1200 MPa, YTS = 400 MPa, Corrected endurance strength = 128.0916 MPa. Take factor of safety as 1. Determine the life of the component and derive the expression you use.

OR

Q6) A stepped shaft is subjected to a uniform torque of 200 Nm, and a completely reversed bending moment of 550 Nm at the step. The shaft is made of cold drawn steel with ultimate tensile strength of 650 N/mm^2 and yield strength of 380 N/mm^2. The theoretical stress concentration factor for bending and torsion are 2 and 1.6 respectively. The other factors are

as follows:

Notch sensitivity = 0.96, Size factor = 0.85, Reliability factor = 0.868, Surface finish factor = 0.9. If the factor of safety is 1.5, determine the diameter of the shaft corresponding to the expected life of 15000 cycles and also for infinite life. **[16]**

SECTION - II

Q7) A single plate clutch with a single pair of contacting surfaces has an inner and outer radii of friction surface as 50 mm and 100 mm respectively. The coefficient of friction between the surfaces is 0.3. The normal intensity of pressure at any radius r is given by, $p = C_1 + C_2/r$, where C_1 and C_2 are constants. The normal intensity of pressure at inner radius is 1/3 times more than that at the outer radius. If the axial force is 4500 N, determine the torque transmitting capacity of the clutch. Derive an expression, you use. **[16]**

OR

Q8) A four wheeler has a total mass of 900 Kg. The mass moment of inertia of each wheel about an transverse axis through its centre of gravity is 0.5 Kg-m^2. The rolling radius of wheel is 0.35 m. The rotating and reciprocating parts of the engine and the transmission system are equivalent to a mass moment of inertia of 2.2 Kg-m^2 rotating at 5 times the speed of the wheel. The car is travelling at a speed of 80 Km/hr on a plane road. When the brakes are applied on all four wheels, the car decelerates at 0.4 g. Determine: **[16]**

a) the energy absorbed by each brake

b) the torque capacity of the brake

Q9) A spur gear pair is to be used to transmit 20 KW power from an electric motor running at 1440 rpm to the machine tool expected to run exactly at 600 rpm. The pinion and gear are to be made of alloy steel (S_{ult} = 800 N/mm^2) and plain carbon steel (S_{ult} = 700 N/mm^2) respectively. The service factor and factor of safety are 1.5 and 1.35 respectively. The face width is 12 times module for which load distribution factor is 1.4. The tooth system is 20° full depth involute. The gears are to be machined to meet the specifications of grade 7. The pinion and gear are to be case hardened to 400 BHN and 350 BHN respectively. Design the gear pair by using the velocity factor and Buckingham's equation for dynamic load. Use the **[18]**

following data:

Velocity factor Kv = 6/6 + V

Load stress factor K = 0.16 (BHN/ 100)^2 N/mm^2

Lewis form factor Y = 0.484 - 2.87/Z

For Grade 7 , $e = 11.0 + 0.9 (m + 0.25\sqrt{d})$

Deformation factor $C = 0.111e(Ep.Eg/Ep + Eg)$ N/mm

Modulus of elasticity for pinion $Ep = 207 \times 10^3$ N/mm^2

Modulus of elasticity for gear $Eg = 207 \times 10^3$ N/mm^2

Buckingham's equation $F_d = 21V(bC + Ftmax) / 21V + \sqrt{bC + Ft\max}$ N

$Ftmax = Ka.Km.Ft$

Standard module in mm - 1,1.25,1.5,2.0,2.5,3.0,4.0,5.0,6.0,8.0,10.0,12,16

OR

Q10) a) A pair of helical gears consists of 20 teeth pinion meshing with 100 teeth gear. The pinion rotates at 720 rpm. The normal pressure angle is 20°, while the helix angle is 25°. The face width is 40 mm and normal module is 4 mm. The pinion is made of plain carbon steel 55C8 (S_{ut} = 720 N/mm^2) while the gear is made of plain carbon steel 40C8 (S_{ut} = 580 N/mm^2). The pinion and gear are heat treated to a surface hardness of 350 BHN and 300 BHN respectively. The service factor and factor of safety are 1.5 and 2.0 respectively. Assuming the velocity factor accounts for dynamic load, calculate the power transmitting capacity of helical gear pair. Use following data:

Velocity factor, $Kv = 5.6 /5.6 + \sqrt{V}$. **[12]**

b) Derive the relation for virtual number of teeth for a helical gear. **[6]**

Q11) A straight bevel gear pair is to be used to transmit 25 K W power from an electric motor rotating at 1500 rpm to a machine required to rotate exactly at 600 rpm. The axes of the pinion and gear intersect at right angles. The pinion and gear are to be made of plain carbon steel 55C8 (S_{ut} = 720 N/mm^2). The service factor and factor of safety are 1.25 and 1.75 respectively. The tooth system is 20° full depth involute. The gears are to be manufactured to meet the specifications of grade 6. The pinion and gear are to be case hardened to 420 BHN and 400 BHN respectively. Design the gear pair by using the velocity factor, $Kv = (6/6+V)$ and the Buckingham's equation for dynamic load. Take

Y_p' = 0.3166. Machining grade 6 : e = 8.0 + 0.63 $\left(m + 0.25\sqrt{2rm}\right)$

Buckingham's equation F_d = 21V (bC + Ftmax)/21V + $\sqrt{bC + Ft\max}$ N**[16]**

OR

Q12) 1/54/10/5 worm gear pair consists of worm made of case hardened carbon steel 10C4 and worm gear made of centrifugally cast phosphor bronze having permissible bending stress of 80 N/mm^2. The wear load factor for worm gear is 0.55 N/mm^2. The tooth system is 200 full depth involute, while the face width of the worm gear is 0.75 times the pitch circle diameter of worm. The coefficient of friction between worm and worm gear teeth is 0.04. The application factor and factor of safety are 1.25 and 1.5 respectively. The external surface area of the housing is 0.6 m^2. The overall heat transfer coefficient is 18 W/m^2 °C. The permissible temperature rise for the lubricant oil is 50°C. If the worm rotates at 1000 rpm, **[16]**

determine:

a) The beam strength of worm gear

b) The wear strength of worm gear

c) The maximum static load the worm gear can take, and

d) The maximum input power the worm can take. Use the following data:

Lewis form factor, Y:

z	20	30	40	50	60
Y	0.3204	0.3581	0.3890	0.4084	0.4210

Velocity factor, K_v = 6/6 + V

x x x