

Total No. of Questions : 12]

P754

SEAT No. :

[Total No. of Pages : 8

[4263] - 216

**T.E. (Mechanical)**

**MACHINE DESIGN - II**

(2008 Pattern) (Semester - II)

**Time : 4 Hours**

**[Max. Marks : 100]**

**Instructions to the candidates:**

- 1) Answer any 3 questions from each section.
- 2) Answers to the two sections should be written in separate book.
- 3) Neat diagrams must be drawn wherever necessary.
- 4) Figures to the right indicate full marks.
- 5) Use of logarithmic tables slide rule, Mollier charts, electronic pocket calculator and steam tables is allowed.
- 6) Assume suitable data, if necessary.

**SECTION - I**

**Unit-I**

- Q1)** a) What is the objective of preloading? Explain mounting and preloading of a taper roller bearing with appropriate sketch. [6]
- b) A ball bearing carries a radial load of 400N at 1760 rpm for 40% time, 600N at 880rpm for 30% time, 200N at 1000rpm for 10% time and no load at 1500rpm for remaining period of the cycle. If the expected life of the bearing is 10,000 hours with 95% reliability, calculate [10]
- i) Basic dynamic load capacity of the bearing.
  - ii) Average speed of bearing operation.

Use following relation for reliability analysis

$$\frac{L}{L_{10}} = \left[ 9.4911 \ln \left( \frac{1}{R} \right) \right]^{\frac{1}{1.17}}$$



**OR**

**P.T.O**

- Q2)** a) A transmission shaft is supported by two deep groove ball bearings at two ends. The center distance between the two bearings is 160mm. A load of 300N acts vertically downward at 60mm distance from the left hand bearing whereas a load of 550N acts horizontally at 50mm distance from the right hand bearing. Shaft speed is 3000 rpm and expected life of the bearings is 7000 hours with a reliability of 95%. It is intended to use same bearing at both ends of the shaft. Calculate dynamic load rating of the bearing so that it can be selected from manufacturer's catalogue. Use above relation for reliability analysis. [12]
- b) Discuss equivalent dynamic load and load life relationship for rolling contact bearings. [4]

### Unit-II

- Q3)** a) Explain the mechanism of pressure development in oil film and draw radial & axial pressure distribution for hydrodynamic journal bearing. [4]
- b) The following data refers to a  $360^\circ$  hydrodynamic journal bearing [12]
- Radial load = 10kN
  - Journal speed = 1450 rpm
  - $\ell/d$  ratio = 1
  - Eccentricity = 15 microns
  - Radial clearance = 20 microns
  - Bearing length = 50mm
  - Specific gravity of lubricant = 0.86
  - Specific heat of lubricant =  $2.09 \text{ kJ/kg}^\circ\text{C}$  Calculate :
    - 1) the minimum oil film thickness
    - 2) the coefficient of friction
    - 3) the power loss in friction
    - 4) the viscosity of lubricant in CP
    - 5) the total flow rate of lubricant in lit/min
    - 6) the side leakage
    - 7) the average temperature, if make up oil is supplied at  $30^\circ\text{C}$

(Refer Table 3)

OR

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- Q4) a)** Derive from First principal Reynold's equation with usual notation.  

$$\frac{\partial}{\partial x}(h^3 \frac{\partial p}{\partial x}) + \frac{\partial}{\partial y}(h^3 \frac{\partial p}{\partial y}) = 6\mu u \frac{\partial h}{\partial x} \quad [6]$$
- b)** A 50mm diameter hardened and ground steel journal rotates at 1440 rpm in a lathe turned bronze bushing which is 50mm long. For hydrodynamic lubrication, the minimum oil film thickness should be five times the sum of surface roughness (clearance values) of journal and bearing. The data about machining methods is as follows (Table 1). [10]

Table 1

Elements	Machining methods	Clearance values
Shaft	Grinding	1.6 microns
Bearing	Turning/Boaring	0.8 microns

The class of fit is H8d8 and the viscosity of the lubricant is 18 cP. Determine the maximum radial load that the journal can carry and still operate under hydrodynamic conditions. Also, calculate quantity of lubricating oil required.

(Refer Table 3)

### Unit-III

- Q5) a)** Draw the following diagrams and write their equations [6]
- Goodman diagram
  - Soderberg diagram
- b)** A spherical pressure vessel, with 500mm inner diameter is welded from steel plates. The welded joints are sufficiently strong and do not weaken the vessel. The plates are made from cold drawn steel 20C8 ( $S_{ut} = 440 \text{ MPa}$  and  $S_{yt} = 242 \text{ MPa}$ ). The vessel is subjected to internal pressure, which varies from 0 to 6 N/mm<sup>2</sup>. The expected reliability is 50 % and the factor of safety is 3.5. The size factor and surface finish factor is 0.85 and 0.82 respectively. The vessel is expected to withstand infinite number of stress cycles. Calculate the thickness of the plates. [12]

OR

- Q6)** a) What is endurance strength of material? [4]
- b) A transmission shaft having an ultimate tensile strength of 600MPa and yield strength of 380MPa is subjected to a fluctuating torque of 200 Nm anticlockwise and 800Nm clockwise. The factor of safety is 2 and the expected reliability is 50%. The surface factor is 0.8 and the size factor is 0.85. Assuming that there is no stress concentration, determine the diameter of the shaft for infinite life. Assume the distortion energy theory of failure. Also determine the diameter of the shaft, if only the fluctuating torque is change to 10Nm anticlockwise to 800Nm clockwise. [14]

## SECTION - II

### Unit-IV

- Q7)** a) What is self energizing and self locking block brake? [4]
- b) Figure 1 shows the arrangement and dimensions of a pivoted block brake with a face width of 50 mm. The coefficient of friction and the permissible intensity of pressure between the friction lining and the brake drum are 0.25 and 0.55 N/mm<sup>2</sup> respectively. If the pivot of the shoe is located such that the moment of frictional force on shoe about the pivot is zero. Calculate [12]
- 1) The braking torque capacity
  - 2) The actuating torque
  - 3) The heat generated, if the speed of the brake drum is 100 rpm and the brake is applied for 5 sec to bring the drum to the rest.



Figure 1

**OR**

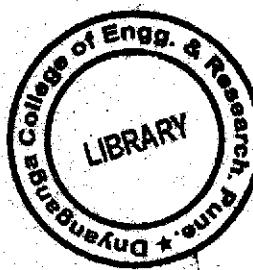
**Q8) a)** An automotive type clutch of heavy vehicle has inside and outside diameters of 150 and 225mm respectively. Clamping force is provided by nine springs, each compressed 6.5 mm to give a force of 625 N when the clutch is new. The moulded friction material provides a conservatively estimated coefficient of friction of 0.35. The maximum engine torque is 280 N-m. [12]

- i) What is the safety factor with respect to slippage of a brand new clutch?
  - ii) What is the safety factor after initial wear has occurred?
  - iii) How much wear of the friction material can take place before the clutch will slip?
- b)** Why heat dissipation is necessary in clutches? [4]

**Unit-V**

**Q9) a)** An electric hoist is being designed for lifting capacity of 30kN at 80m/min. The rope drum diameter will be approximately 640mm and it will be driven by an electric motor running at 1440 rpm through a pairs of spur gears only. The pinion and gear are made of plain carbon steel 55 C8 ( $S_{ut}$  - 720 N/mm<sup>2</sup>). The tooth system is 20° full depth involute and number of teeth on pinion as minimum as possible. The service factor and factor of safety are taken as 1.25 and 1.5 respectively. The face width 12 times module. Suggest suitable number of stages for the reduction drive. The velocity ratio in each stage should not exceed 6:1. Design the first stage gear pair by using velocity factor. Also, suggest suitable capacity of electric motor. Use following data:[12]

- Velocity factor  $C_v = \frac{3}{3+V}$
- Lewis form factor  $Y = 0.484 - \frac{2.87}{Z}$



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• Load stress factor  $K = 0.16 \left[ \frac{BHN}{100} \right]^2 \frac{N}{mm^2}$

• Stage efficiency = 95%

Standard module in mm 1,1.25,2,3,4,5,6,8,10,12,16

Standard kW Rating of electric motors 5,10,15, 20,25,30, 35, 40, 45, 50,60

- b) Define beam and wear strength of spur gear.

[4]

OR

**Q10)** A belt conveyor system is to be driven by a 30kW, 720rpm electric motor through a helical gear pair running at 225 rpm. A pinion having 14 teeth made of alloy steel ( $S_{ut} = 800 \text{ N/mm}^2$ ) is meshed with gear made of plain carbon steel ( $S_{ut} = 720 \text{ N/mm}^2$ ). The application and load concentration factor is 1.3 and 1.1 respectively, while the factor of safety is 2. The face width is 10 times normal module and tooth system is 20°full depths involute. The helix angle is 25°. The gears are to be machined to meet the specification of grade 7. The deformation factor for gear pair is 11000 e. N/mm. Design the gear pair by using velocity factor and Buckingham's equation for dynamic load. Also suggest the surface hardness for gear pair. Use following data Standard module in mm=1,1.25,2,3,4,5,6,8,10,12,14

[16]

$$\text{Velocity factor } Cv = \frac{5.6}{5.6 + \sqrt{V}}$$

$$\text{Dynamic load } P_d = \frac{21V(bC \cos^2 \psi + P_{t\max}) \cos \psi}{21V + \sqrt{bC \cos^2 \psi + P_{t\max}}} N$$

$$\text{Lewis form factor } Y' = 0.484 - \frac{2.87}{Z'}$$

$$\text{Load stress factor } K = 0.16 \left[ \frac{BHN}{100} \right]^2 \frac{N}{mm^2}$$

$$\text{for grade 7 } e = 11.0 + 0.9(m + 0.25\sqrt{d}) \mu m$$

Unit-VI

**Q11)** A centrifugal pump submerged in a well is driven at 600 rpm by a 25kW motor running at 1500 rpm through a pair of straight bevel gears. The bevel pinion and gear are made of alloy steel with ultimate tensile strength 720 N/mm<sup>2</sup>. The axis of pinion and gear intersect at right angle. The starting torque is 125% to the rated torque. The factor of safety required is 1.75. The tooth system is 20° full depth involute. The gears are to be cut to meet the specification of grade 6. The pinion and gear are case hardened to 420 BHN and 400 BHN respectively. The deformation factor is 11488e N/mm. Design the gear pair. [18]

Use following data: Standard module in mm = 1, 1.25, 1.5, 2.0, 3, 4, 5, 6, 8, 10 and 12.

$$\text{Lewis form factor } Y = 0.484 - \frac{2.87}{Z'}$$

$$\text{Load stress factor } K = 0.16 \left[ \frac{\text{BHN}}{100} \right]^2 \frac{N}{mm^2}$$

$$\text{For grade 7 } e = 8.0 + 0.63(m + 0.25\sqrt{r_m}) \mu m$$

$$\text{Velocity factor } Cv = \frac{6}{6+V}$$

$$\text{Dynamic load } P_d = \frac{21V(bC + P_{t\max})}{21V + \sqrt{bC + P_{t\max}}} N$$

OR

**Q12)** A pair of worm and worm gear is used to drive an elevator cage for the following specifications shown in table 2. [18]

S.N.	Particulars	Worm	Worm gear	Remark
1	Pressure angle	20°	20°	Normal plane
2	Axial pitch	18.85mm	—	—
3	Pitch circle diameter	48mm	192mm	—
4	Lead	18.85mm	—	—
5	Effective width	—	36mm	—
6	Speed	3500 rpm	—	—
7	Permissible bending stress	—	90N/mm <sup>2</sup>	—
8	Wear factor	—	0.83N/mm <sup>2</sup>	—

- Overall heat transfer coefficient without fan =  $16 \text{ W/m}^2\text{C}$
  - Overall heat transfer coefficient with fan =  $15.2 + 8.25 \times 10^{-3} \times n_w \text{ W/m}^2\text{C}$   
Where  $n_w$  is speed of worm in rpm
  - Effective area of housing =  $9 \times 10^{-5} \times (a)^{1.88} \text{ m}^2$   
(where  $a$  = centre distance in mm)
  - Frictional losses in bearing = 4.5% of total input power  
Take coefficient of friction between worm and worm gear teeth = 0.025
- Determine
- 1) Dimensions of worm gear pair
  - 2) Input power rating on strength basis
  - 3) Temperature rise of lubricating oil with or without fan. Is fan necessary?

Give comment.

Use following equations

$$\text{Lewis form factor } Y' = 0.484 - \frac{2.87}{Z'}$$

$$\text{Velocity factor } Cv = \frac{6}{64W \cdot sppuonline.com}$$

$$\text{Efficiency } \eta = \frac{\tan \lambda}{\tan(\phi_v + \lambda)}$$

Where  $\lambda$  = Lead angle of worm

Table 3

$\ell/d$	$h_o/c$	E	S	$(r/c)f$	$Q/\text{rcn}_s \ell$	$Q_s/Q$	$P_{\max}/P$
1	0.0	1.0	0	0	----	1	----
	0.03	0.97	0.00474	0.514	4.82	0.973	6.579
	0.1	0.9	0.0188	1.05	4.74	0.919	4.048
	0.2	0.8	0.0446	1.70	4.62	0.842	3.195
	0.4	0.6	0.121	3.22	4.33	0.680	2.409
	0.6	0.4	0.264	5.79	3.99	0.497	2.066
	0.8	0.2	0.631	12.8	3.59	0.280	1.890
	0.9	0.1	1.33	26.4	3.37	0.150	1.852
	1.0	0.0	$\infty$	$\infty$	3.142	0	----

