Total No. of Questions :12]

SEAT No. :

# P2814

[Total No. of Pages :8

## [4958]-116
## T.E. (Mechanical Engineering)
## MACHINE DESIGN - II
## (2008 Course) (Semester - II) (302047)

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

***Instructions to the candidates:***

1) ***Answer three questions from section I and three questions from section II.***
2) ***Answers to the two sections should be written in separate answer- books.***
3) ***Figures to the right side indicate full marks.***
4) ***Neat diagrams must be drawn wherever necessary.***
5) ***Use of Electronic Pocket Calculator is allowed.***
6) ***Assume suitable data, if necessary, and mention it clearly.***

## SECTION-I

***Q1)*** a) A single row deep groove ball bearing operated with following work cycle. If the expected life of bearing is 13,000 hours with reliability of 90%. Calculate the dynamic load rating of the bearing and determine reliability of a system consisting of four such bearings. The work cycle is **[12]**

| Element no | Element time | Fr (kN) | Fa (kN) | Radial factor | Thrust factor | Race Rotating | CS | Speed (r.p.m.) |
|---|---|---|---|---|---|---|---|---|
| 1 | 30% | 5 | 1.5 | 0.56 | 1.1 | Inner | 1.25 | 960 |
| 2 | 40% | 3.7 | 0.73 | 0.56 | 1.3 | Outer | 1.4 | 1440 |
| 3 | 30% | - | - | - | - | Outer | - | 720 |

b) Explain the mounting and preloading of a taper roller bearing with appropriate sketch. **[6]**

OR

*P.T.O.*

***Q2)*** a) A transmission shaft is supported by two deep grove ball bearings at two ends. The centre distance between the bearings is 160 mm. A load of 300 N acts vertically downwards at 60 mm distance from the left hand bearing whereas a load of 550 N acts horizontally at 50 mm distance from right had bearing.

Shaft speed is 3000 rpm and expected life of bearing 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. **[11]**

b) Derive Stribeck's equation for static capacity of a rolling contact bearing. **[7]**

***Q3)*** Following data is given for a 360$^{o}$ hydrodynamic bearing. **[16]**

Radial load = 10kN,

Journal speed = 1440 rpm,

Unit pressure in bearing = 1000 KPa,

Clearance ratio (r/c) = 800

$l/d = 1$

Viscosity of lubricant = 30 mPa-S.

Assuming that total heat generated in the bearing is carried by the total oil flow in the bearing. Calculate

a) Dimensions of bearings

b) Coefficient of friction

c) Power lost in friction

d) Total flow of oil

e) Side leakage

f) Temperature rise

g) Average temperature (inlet temperature is 40$^{o}$C)

h) Find maximum pressure ($P_{max}$)

Use the data given in table No.1.

OR

*Q4)* a) Explain significance of following variables in connection with hydrodynamic bearing. **[8]**

i) l/d ratio

ii) unit bearing pressure

iii) radial clearance

iv) Minimum oil film thickness

b) Derive Petroff's equation for hydrodynamic bearing. **[8]**

*Q5)* A cantilever beam of circular cross section is fixed at the end and is subjected to completely reversed load of $\pm$ 100 N at the free end. The force is perpendicular to the axis of beam. The distance between the fixed and the free end is 400 mm. There is no stress concentration in the beam. The beam is made of steel with an ultimate tensile strength of 1300 MPa. The surface finish factor for the beam is 0.87 and size factor is 0.85 respectively. The reliability factor is 0.868. Determine the diameter of the beam for a life of 47500 cycles. **[16]**

OR

*Q6)* a) Explain stress concentration, its causes. What are methods of reducing stress concentration? Explain with neat sketch. **[8]**

b) Explain modified Goodman diagram for **[8]**

i) Fluctuating axial or bending stresses

ii) Fluctuating torsional shear stresses.

## SECTION-II

*Q7)* a) A double block brack with an identical pivoted shoes is to be used for braking torque capacity of 1kN/m the diameter of brake drum is 400 mm & angle of wrap for each shoe is $120^{o}$. The coefficient of friction is 0.3 and the permissible intensity of pressure is 0.8 $N/mm^2$. The pivot of each shoe is located in such a way that moment of frictional force on the shoe about pivot is zero. Calculate, **[8]**

i) Distance of pivot from the axis of brake drum.

ii) The width of friction lining parallel to axis of drum.

b) Explain **[8]**

i) Parameters to be considered for the selection of the friction lining for brakes

ii) Self energizing and self locking of brakes.

OR

***Q8)*** a) Discuss the properties of friction lining materials. Draw a labeled sketch of Multi-plate clutch. **[6]**

b) Design a centrifugal clutch for the following data: **[10]**

- Power to be transmitted = 15kW
- Running speed = 720 rpm
- Engagement speed = 540 rpm
- No. of shoes = 4
- Inner radius of the drum = 162.5 mm
- The radius of C.G. of the shoe = 150 mm, when the clutch is engaged
- The coefficient of friction is 0.25
- Permissible pressure on friction lining is 0.21 N/mm$^2$

Calculate

i) The mass of each shoe.

ii) The dimensions of friction lining. Assume shoe subtends an angle of 60$^o$ at the center of the spider.

***Q9)*** a) A spur gear pair transmitting 5kW power from an electric motor running at 1440 rpm to a machine running at 480 rpm. **[12]**

- Consider the following data
- No. of teeth on pinion = 18

- Center distance = 216 mm
- Face width = 10  x module mm.
- Allowable bending stress for pinion and gear = 160 N/mm$^2$
- Surface hardness for gear pair = 300 B.H.N.
- Tooth system = 20$^o$ full depth involute.
- Deformation factor  = 11500 e. N/mm

The gears are machined to the specifications of grade 6,

e = 8+0.63 Φ µm  Where,  Φ= m+0.25 $\sqrt{d}$

Lewis form factor $Y = 0484 - \frac{2.87}{Z}$

Assuming dynamic load is accounted by the Buckingham's equations.

$$F_d = \frac{21V(bC + Ft\max)}{21V + \sqrt{bC + Ft\max}}$$

Calculate:

i) The factor of safety against bending failure and

ii) The factor of safety against pitting failure.

b) What are the assumptions in the analysis of beam strength of spur gear tooth. Write beam strength equation only. **[4]**

OR

***Q10)*** a) A pair of helical gear consist of 24 teeth pinion rotating at 5000 rpm and supplying 2.5 kW power to a gear. The speed reduction is 4:1. The normal pressure angle and helix angle are 20$^o$ and 23$^o$ respectively. Both the gears are made of hardened steel ($S_{ut}$ = 750 N/mm$^2$). The service factor, factor of safety and load concentration factor are 1.5, 2.0 and 1.0 respectively. The gears are finish as per grade -4. **[12]**

i) In initial stage of gear design assume velocity factor accounts dynamic load and face width is 10 X module and assume pitch line velocity V = 10 m/s. for estimating normal module.

ii) Select first preference module and calculate dimensions of gears.

iii) Determine the dynamic load by Buckingham's equation also calculate factor of safety in bending.

iv) Specify the surface hardness at factor of safety 2.0

Use following data:

- Lewis form factor $Y' = 0.484 - \frac{2.87}{Z'}$
- For grade 4 ; e= 3.20 + 0.25 ($m_n + 0.25\sqrt{d}$ )
- Buckingham's equation $P_d = \frac{21V(bC.\cos^2\psi + P_{tmax})\cos\psi}{21V + \sqrt{bC.\cos^2\psi + P_{tmax}}}$
- Velocity factor $C_v = \frac{5.6}{5.6 + \sqrt{V}}$

First preference module (mm)-1.0, 1.25, 1.5, 2.0, 2.5, 3, 4, 5, 6, 8, 10, 12, 16 and 20.

b) Draw the neat sketch showing force analysis of helical gear pair, and state these forces. **[4]**

***Q11)*** a) Explain with neat sketch hypoid gear and spiral bevel gear. **[6]**

b) A straight bevel pinion having 21 teeth to be made of alloy steel ($S_{ut}$ = 800 N/mm$^2$) is to mesh with a gear to be made of plain carbon steel ($S_{ut}$ = 720 N/mm$^2$). The axes of pinion and gear intersects at right angle. Gear pair is to transmit 20kW power from a spindle running at 500 rpm to a machine running at 300 rpm. The starting torque of the motor is 110% of the rated torque. The factor of safety required is 1.5. The tooth system is 20$^o$ full depth involute. The gears are cut to meet the specifications grade 6. Hardness of gear pair is 350 BHN. The deformation factor C is 10900 e. N/mm. Design the gear pair. Use velocity factor for preliminary estimation and Buckingham's equation for precise estimation of dynamic load **[12]**

Use following data,

- Lewis form factor $Y = 0484 - \frac{2.87}{z'}$

- $K_v = \frac{6}{6+V}$

- $F_d = \frac{21V(bC + Ft\max)}{21V + \sqrt{bC + Ft\max}}$

- For grade 6, e = 8+0.63 Φ μm  Where,  Φ= m+0.25 $\sqrt{d}$

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

OR

***Q12)*** a) State materials used for worm & worm gear. Derive an expression for the efficiency of worm gear pair. **[6]**

b) A double start worm made of case hardened alloy steel 16Ni80Cr60 ($S_{ut}$ = 700 N/mm$^2$) is to mesh with worm gear to be made of phosphor bronze ($S_{ut}$ = 240 N/mm$^2$). The gear pair is required to transmit 5kW power from an electric motor running at 1500rpm to a machining running at 75rpm. The service factor is 1.25, while the factor of safety required is 2.0. The face width of worm gear is 0.73 times the pitch circle diameter of worm. The worm gear factor is 0.685 N/mm$^2$, while the diametrical quotient is 10. The normal pressure angle is 14.5$^o$. If the coefficient of friction between worm and worm gear teeth is 0.03, design the gear pair and find the power lost. Would you recommend a fan for the gear box? Assume the permissible temperature rise is 50$^o$C. **[12]**

Use following data,

- Lewis form factor -- $Y = 0.39 - \frac{2.15}{Z_G}$

- Velocity factor, $C_v = \frac{6}{6+V_G}$

- Area of housing, A = 1.14 x 10$^{-4}$ x (a)$^{1.7}$ m$^2$, where a = centre distance in mm.

**Table No·1** *Dimensionless performance parameters for full journal bearing with side flow*

| $\left(\frac{l}{d}\right)$ | $\varepsilon$ | $\left(\frac{h_o}{c}\right)$ | $S$ | $\phi$ | $\left(\frac{r}{c}\right)f$ | $\left(\frac{Q}{rcn_s l}\right)$ | $\left(\frac{Q_s}{Q}\right)$ | $\left(\frac{p}{p_{max}}\right)$ |
|---|---|---|---|---|---|---|---|---|
| ∞ | 0 | 1.0 | ∞ | (70.92) | ∞ | $\pi$ | 0 | — |
| | 0.1 | 0.9 | 0.240 | 69.10 | 4.80 | 3.03 | 0 | 0.826 |
| | 0.2 | 0.8 | 0.123 | 67.26 | 2.57 | 2.83 | 0 | 0.814 |
| | 0.4 | 0.6 | 0.0626 | 61.94 | 1.52 | 2.26 | 0 | 0.764 |
| | 0.6 | 0.4 | 0.0389 | 54.31 | 1.20 | 1.56 | 0 | 0.667 |
| | 0.8 | 0.2 | 0.021 | 42.22 | 0.961 | 0.760 | 0 | 0.495 |
| | 0.9 | 0.1 | 0.0115 | 31.62 | 0.756 | 0.411 | 0 | 0.358 |
| | 0.97 | 0.03 | — | — | — | — | 0 | — |
| | 1.0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 1 | 0 | 1.0 | ∞ | (85) | ∞ | $\pi$ | 0 | — |
| | 0.1 | 0.9 | 1.33 | 79.5 | 26.4 | 3.37 | 0.150 | 0.540 |
| | 0.2 | 0.8 | 0.631 | 74.02 | 12.8 | 3.59 | 0.280 | 0.529 |
| | 0.4 | 0.6 | 0.264 | 63.10 | 5.79 | 3.99 | 0.497 | 0.484 |
| | 0.6 | 0.4 | 0.121 | 50.58 | 3.22 | 4.33 | 0.680 | 0.415 |
| | 0.8 | 0.2 | 0.0446 | 36.24 | 1.70 | 4.62 | 0.842 | 0.313 |
| | 0.9 | 0.1 | 0.0188 | 26.45 | 1.05 | 4.74 | 0.919 | 0.247 |
| | 0.97 | 0.03 | 0.00474 | 15.47 | 0.514 | 4.82 | 0.973 | 0.152 |
| | 1.0 | 0 | 0 | 0 | 0 | 0 | 1.0 | 0 |
| $\left(\frac{1}{2}\right)$ | 0 | 1.0 | ∞ | (88.5) | ∞ | $\pi$ | 0 | — |
| | 0.1 | 0.9 | 4.31 | 81.62 | 85.6 | 3.43 | 0.173 | 0.523 |
| | 0.2 | 0.8 | 2.03 | 74.94 | 40.9 | 3.72 | 0.318 | 0.506 |
| | 0.4 | 0.6 | 0.779 | 61.45 | 17.0 | 4.29 | 0.552 | 0.441 |
| | 0.6 | 0.4 | 0.319 | 48.14 | 8.10 | 4.85 | 0.730 | 0.365 |
| | 0.8 | 0.2 | 0.0923 | 33.31 | 3.26 | 5.41 | 0.874 | 0.267 |
| | 0.9 | 0.1 | 0.0313 | 23.66 | 1.60 | 5.69 | 0.939 | 0.206 |
| | 0.97 | 0.03 | 0.00609 | 13.75 | 0.610 | 5.88 | 0.980 | 0.126 |
| | 1.0 | 0 | 0 | 0 | 0 | — | 1.0 | 0 |
| $\left(\frac{1}{4}\right)$ | 0 | 1.0 | ∞ | (89.5) | ∞ | $\pi$ | 0 | – |
| | 0.1 | 0.9 | 16.2 | 82.31 | 322.0 | 3.45 | 0.180 | 0.515 |
| | 0.2 | 0.8 | 7.57 | 75.18 | 153.0 | 3.76 | 0.330 | 0.489 |
| | 0.4 | 0.6 | 2.83 | 60.86 | 61.1 | 4.37 | 0.567 | 0.415 |
| | 0.6 | 0.4 | 1.07 | 46.72 | 26.7 | 4.99 | 0.746 | 0.334 |
| | 0.8 | 0.2 | 0.261 | 31.04 | 8.8 | 5.60 | 0.884 | 0.240 |
| | 0.9 | 0.1 | 0.0736 | 21.85 | 3.50 | 5.91 | 0.945 | 0.180 |
| | 0.97 | 0.03 | 0.0101 | 12.22 | 0.922 | 6.12 | 0.984 | 0.108 |
| | 1.0 | 0 | 0 | 0 | 0 | – | 1.0 | 0 |

Table No-I for Q. 3

EEE