T.E. (Mechanical) (Semester – II) Examination, 2010
TRANSMISSION SYSTEM DESIGN
(2003 Course)

Time : 4 Hours Max. Marks : 100

Instructions *: 1) Answer* ***three*** *questions from Section* ***I*** *and* ***three*** *questions from Section* ***II****.*

2) Answers to the ***two*** *Sections should be written in* ***separate books.***

3) ***Neat*** *diagrams must be drawn* ***wherever*** *necessary.*

4) ***Use*** *of calculator is* ***allowed****.*

5) Assume suitable data, ***if*** *necessary.*

SECTION – I

Unit – I

1. a) Discuss equivalent dynamic load and load-life relationship for rolling contact bearings. **4**

b) Discuss different types of rolling contact bearings. **2**

c) A shaft running at 500 rpm is supported on two identical deep groove ball bearing which are 1m apart. The shaft carriers two flat belt pulleys at a distance of 200 mm and 800 mm from the left hand bearings. For both the pulleys angle of wrap is 180° and diameter 500 mm. The two belt directions are perpendicular to each other. Maximum belt tension in any belt is 2500 N. The ratio of belt tensions is 2.2. Load factor is 1.75 and expected life of bearing is 65000 hours with a reliability of 95%.

Determine the required basic dynamic capacity of bearing so that it can be selected from a standard manufacturer's catalogue.

Use reliability-life

relationship as $\frac{L}{L_{10}} = \left[9.4912 \log_e \left(\frac{1}{R} \right) \right]^{\frac{1}{1.17}}$. **10**

OR

2. a) Discuss different methods of mounting the bearing. 4

b) Discuss reliability of rolling contact bearing. 2

c) A single row deep groove ball bearing operates with the following work cycle.

If expected life of the bearing is 28000 hours with a reliability of 80%, calculate the basic dynamic load rating capacity of bearing so that it can be selected from a standard manufacturer's catalogue. If there are such four bearings in the system, what is the probability of the system survival for 28000 hours ?

Element time %	Radial Load kN	Axial Load kN	Radial Factor	Axial Factor	Race Rotating	Service Factor	Speed rpm
30	2.8	1.2	0.56	1.4	Inner	1.75	900
40	5.3	1	0.56	1.6	Outer	1.35	1200
Remaining	NIL	NIL	-	-	Outer	-	1500

Use reliability-life relationship as $\frac{L}{L_{10}} = \left[9.4912 \log_e \left(\frac{1}{R} \right) \right]^{\frac{1}{1.17}}$. 10

Unit – II

3. a) Derive equation for torque capacity of a single plate clutch considering uniform pressure theory and uniform wear theory. 4

b) A cone clutch is used to transmit 20 kW at 1500 rpm. Assuming uniform pressure conditions. Determine :

i) Dimensions of friction surfaces

ii) Axial force required to engage the clutch

iii) Operating force.

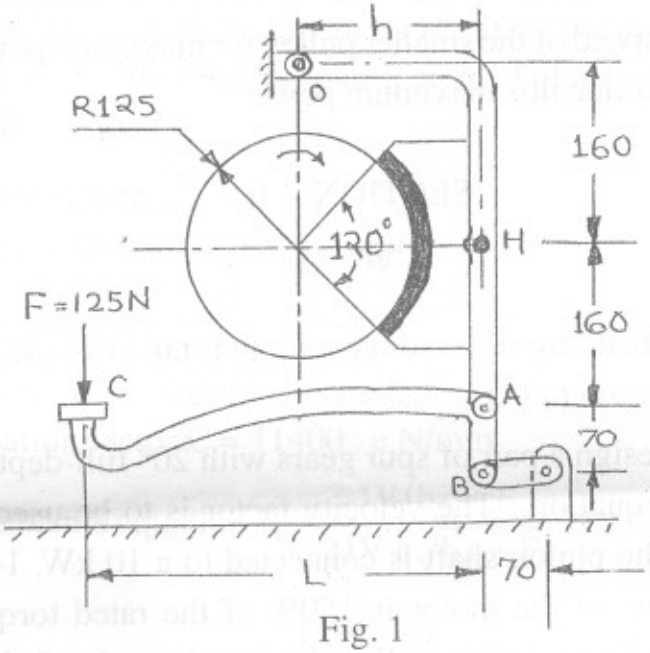
Use the following information :

- Coefficient of friction for friction material 0.23
- Permissible pressure on clutch face 330 kN/m^2
- Semi-cone angle 12.5°
- Mean radius of friction surface is twice the face width. 12

OR

4. a) Discuss the classification of Brakes. 3

b) A single block shoe brake is operated through linkages as shown in figure 1. The shoe is pivoted to the arm at H. The dimension 'h' is so selected as to get moment of frictional force about the pivot equal to zero. Actuating force is 125 N. Coefficient of friction and permissible intensity of pressure between the friction lining and brake drum are 0.33 and 0.5 N/mm^2 respectively. Determine the lever length 'L' and width of the brake shoe parallel to axis of drum if braking torque capacity is 120 N-m. 13



[All the dimensions are in mm]

Fig. 1

Unit – III

5. a) Derive the relation and condition of maximum power transmitted by flat belt in terms of initial Tension Ti. 5

b) A V-belt drive is used for speed reduction between two shafts. Groove angle for the pulley is 40°. Mass of belt is 0.75 kg/m. The coefficient of friction is 0.1025. Assuming angle of wrap as 210°, plot a graph of power transmitted by the belt as a function of belt speed. Assume allowable tension in the belt as 900 N.

Determine optimum belt speed and maximum power transmission also find speed of belt at which no power will be transmitted. 13

OR

6. a) Discuss polygonal effect in chains. 3

   b) Discuss construction and lay of wire ropes. 3

   c) Two parallel shafts are to be connected by an open flat belt. The diameter of pulleys are 400 mm and 800 mm and they are 1m apart. The initial tension in the belt when it is stationary is 2 kN. The mass of the belt is 2 kg/m. The coefficient of friction between the belt and the pulley is 0.3. Calculate the power transmitted if the smaller pulley rotates at 1000 rpm.

   Also suggest the speed of the smaller pulley for maximum power transmission by the belt. Determine this maximum power. 12

SECTION – II

Unit – I

7. a) Why permissible bending stress for gear tooth is taken as one third of ultimate tensile strength ? 6

   b) It is required to design a pair of spur gears with 20° full-depth involute teeth based on Lewis equation. The velocity factor is to be used to 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 is made of plain carbon steel 40 C_8 (σ_{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.

   Use following data :

   - Lewis form factor, $Y = 0.484 - \frac{2.87}{Z}$
   - Velocity factor $Cv = \frac{3}{3+V}$
   - Load stress factor, $K = 0.16 \left(\frac{BHN}{100}\right)^2 N/mm^2$.
   - Assume number of teeth on pinion 18. 10

   OR

8. It is required to design a spur gear speed reducer for a compressor running at 250 rpm driven by a 7.5 kW, 1000 rpm electric motor. The centre distance between the axes of the gear shafts should be exactly 250 mm. The starting torque of the motor can be assumed to be 150% of the rated torque. The gears are made of carbon steel $50C_4$ (σ_{ut} = 700 N/mm^2). The pressure angle is 20°. The factor of safety is 2 for preliminary design based on the use of velocity factor. **16**
   i) Design the gears and specify their dimensions
   ii) Assume that the gears are manufactured to meet the requirements of Grade-6 and calculate the dynamic load by using Buckingham's equation
   iii) Calculate the effective load and factor of safety against bending failure.
   iv) Using the same factor of safety against pitting failure, specify suitable surface hardness for the gears.

   Use following data :
   - Lewis form factor $Y = 0.484 - \frac{2.87}{Z}$
   - Velocity factor $Cv = \frac{3}{3+V}$, load stress factor $K = 0.16\left(\frac{BHN}{100}\right)^2$ N/mm^2.
   - For grade – 6 e = 8.00+0.63 (m+0.25$\sqrt{D}$) microns.
   - Deformation factor C = 11400 . e N/mm
   - Buckingham's equation $F_d = \frac{21V(b.c + F_t)}{21V + \sqrt{b.c + F_t}}$ N.

Unit – II

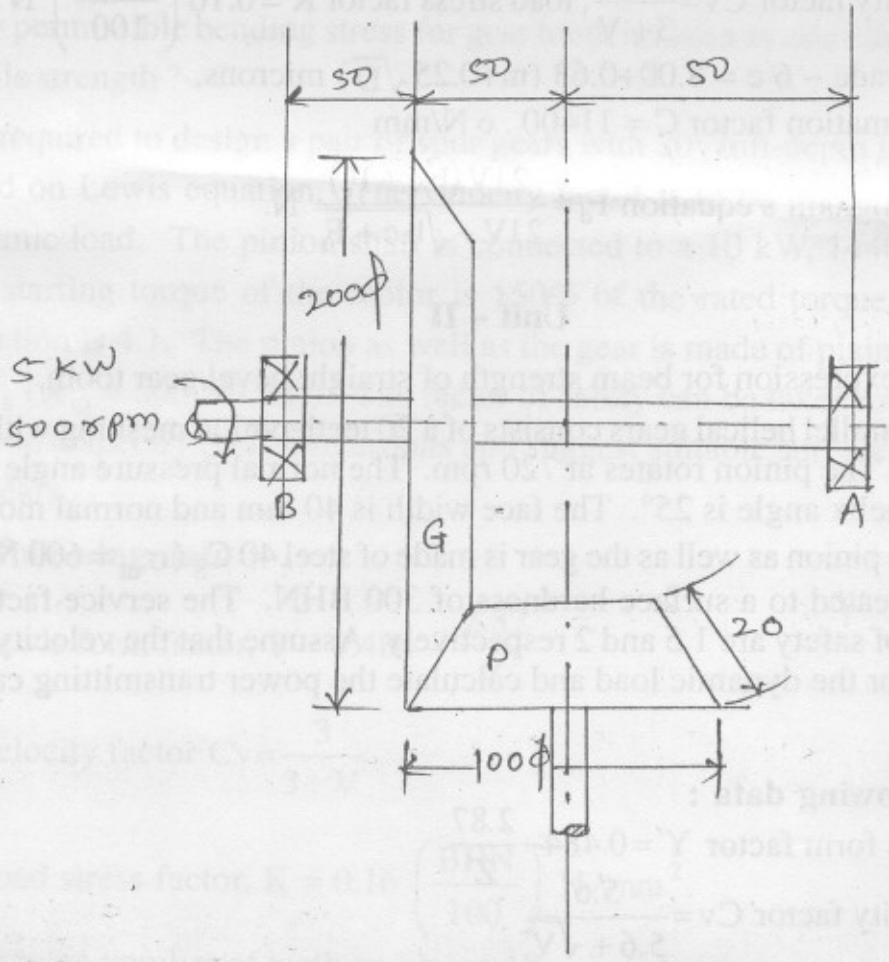
9. a) Derive an expression for beam strength of straight bevel gear tooth. **8**
   b) A pair of parallel helical gears consists of a 20 teeth pinion meshing with a 100 teeth gear. The pinion rotates at 720 rpm. The normal pressure angle is 20°, while the helix angle is 25°. The face width is 40 mm and normal module is 4 mm. The pinion as well as the gear is made of steel $40C_8$ (σ_{ut} = 600 N/mm^2) and heat treated to a surface hardness of 300 BHN. The service factor and the factor of safety are 1.5 and 2 respectively. Assume that the velocity factor accounts for the dynamic load and calculate the power transmitting capacity of gears.

   Use following data :
   - Lewis form factor $Y' = 0.484 - \frac{2.87}{Z'}$
   - Velocity factor $Cv = \frac{5.6}{5.6 + \sqrt{V}}$
   - Load stress factor $K = 0.16\left(\frac{BHN}{100}\right)^2$ N/mm^2. **10**

   OR

10. a) What is formative number of teeth in helical gear ? Derive the expression for the same with sketch. 6

b) The dimensions of a pair of bevel gears are given in fig. The gear 'G' delivers 5 kW power at 500 rpm to the output shaft. The bearings A and B are mounted on the output shaft in such a way that the bearing B can take radial as well as thrust load, while the bearing A can only take radial load. Determine the reactions at the two bearings. 12



Unit – III

11. A double start worm made of case hardened alloy steel 16 Ni 80 Cr 60 ($\sigma_{ut} = 700$ N/mm^2) is to mesh with worm gear to be made of phosphor bronze ($\sigma_{ut} = 240$ N/mm^2). The gear pair is required to transmit 5 kW power from an electric motor running at 1500 rpm to a machine running at 75 rpm. The service factor is 1.25, while the factor of safety required is 2.0. The face width of the worm gear is 0.73 times the pitch circle diameter of worm. The worm gear wear factor is 0.685 N/mm^2, while the diametral quotient is 10. The normal pressure angle is 14.5°. 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 tan for the gear box ? **16**

Use following data :

- Lewis form factor $Y = 0.39 - \frac{2.15}{Z}$
- Velocity factor $Cv = \frac{6}{6+V}$
- Area of housing $A = 1.14 \times 10^{-4} \times (a)^{1.7}$ m^2 where a - centre distance.
- $\Delta T = 50°C$.

OR

12. a) Derive an expressions for effective face width of the worm wheel and length of the root of the worm wheel teeth. **8**

b) A pair of worm gears is designated as 1/30/10/08.

Calculate :

i) Centre distance

ii) Speed reduction

iii) Dimensions of the worm and

iv) Dimensions of the worm wheel. **8**
