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

Time : 4 Hours   Max. Marks : 100

Instructions : *1) Answer **any three** questions from **each** Section.*
*2) Answers to the **two** Sections should be written in **separate** books.*
*3) **Neat** diagrams must be drawn **wherever** necessary.*
*4) Black figures to the **right** indicate **full** marks.*
*5) Use of electronic pocket calculator is **allowed**.*
*6) Assume suitable data, if **necessary**.*

SECTION – I

Unit – I

1. a) Discuss "Load-Life" Relationship for rolling contact bearings. 3

b) A shaft is supported on two bearings A and B which are 250 mm apart. A gear is attached at a distance of 100 mm on the right from left hand side bearing A. Weight of a pulley is 100 N which is attached at an overhung of 150 mm on the right of right hand side bearing B. Horizontal belt tensions are 498 N and 166 N respectively. Horizontal tangential force component for the gear is 497 N which is directed same as belt tensions. Vertically downward radial force component for the gear is 181 N. The load factor for application is 2.5 and expected life of bearings is 8000 hours. If the shaft speed is 720 rpm find dynamic load capacity for bearings A and B so that they can be selected from manufacturer's catalogue. 13

OR

2. a) Discuss preloading of rolling contact bearings. 3

b) An equivalent radial load on a bearing varies continuously from 0 to 20 kN in a sinusoidal manner. Determine the dynamic load rating at 90% reliability, if the bearing is to have a life of 20 million revolutions at a reliability of 99%. Assume shaft speed as 1000 rpm.

Use Life Reliability relationship as

$$\frac{L}{L_{10}} = \left[9.491 \log_e\left(\frac{1}{R}\right)\right]^{\frac{1}{1.17}}.$$ 13

Unit – II

3. a) Discuss different type of clutches. 3

b) For a single plate clutch consisting only one pair of contacting surfaces, Derive a relation for optimum ratio of inner diameter of friction disk to outer diameter of friction disc, which will yield maximum torque transmitting capacity.

Assuming maximum torque transmitting capacity for a clutch, find its inner and outer diameter of friction lining using following information :

Number of pairs of contacting surfaces : 1

Maximum torque transmitted : 120 Nm

Load factor : 1.5

Permissible intensity of pressure : 350 kPa

Coefficient of friction : 0.35

Assume uniform wear theory. 13

OR

4. a) Discuss energy absorbed by brakes. 3

b) A double pivoted shoe brake is as shown in Fig. 1. Find torque, that can be sustained by the brake when the drum is rotating in clockwise rotation.

Assume coefficient of friction between the shoe and the drum as 0.28. 13

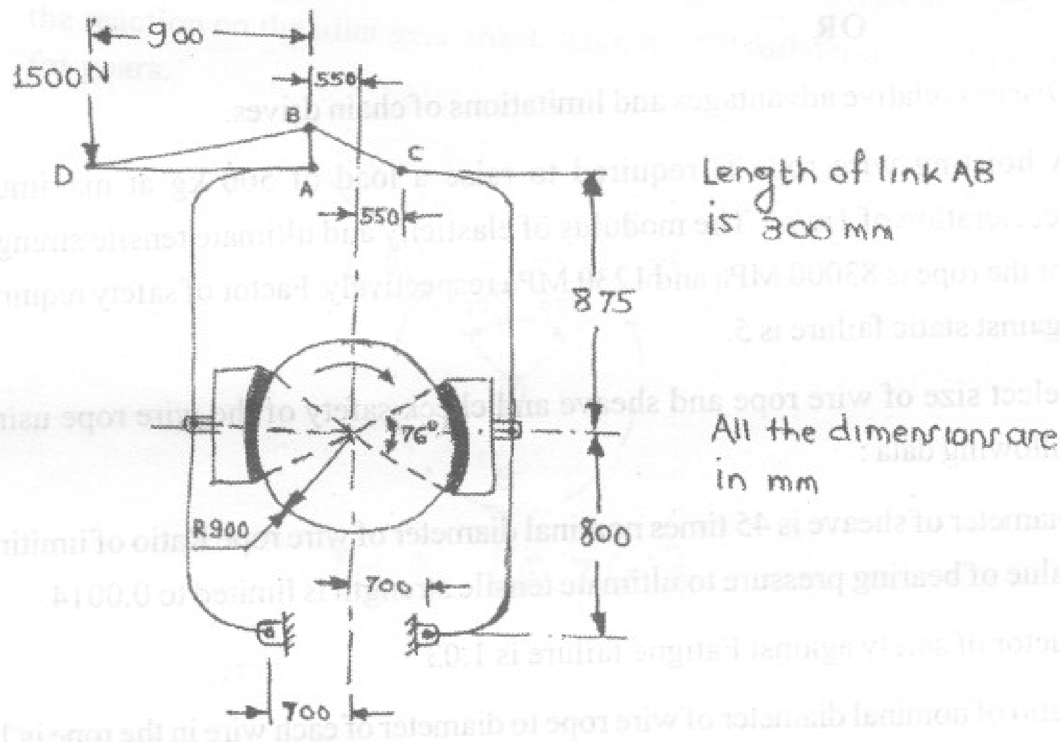

Fig. 1 [Q.4 (b)]

Unit – III

5. a) Discuss different belt-tightening (tensioning) methods. 3

b) Derive a relation for optimum velocity of a belt for maximum power transmission capacity in turms of total tension in tight side of the belt and mass per unit length of the belt. 6

c) A single V-belt is used to transmit power from a grooved pulley of pitch diameter 200 mm running at 1500 rpm to a flat pulley running at 500 rpm. The central distance between the pulleys is 1m. Mass of 1 meter length of the belt is 300 gm. The coefficient of friction between the pulley and the belt is 0.25. For the belt allowable tension is 800 N. Assuming groove angle for the grooved pulley as 38°. Find power capacity of belt and Initial tension required in the belt. 9

OR

6. a) Discuss relative advantages and limitations of chain drives. 3

b) A hoisting wire rope is required to raise a load of 500 kg at maximum acceleration of $1 m/s^2$. The modulus of elasticity and ultimate tensile strength for the rope is 83000 MPa and 1230 MPa respectively. Factor of safety required against static failure is 5.

Select size of wire rope and sheave and check safety of the wire rope using following data :

Diameter of sheave is 45 times nominal diameter of wire rope Ratio of limiting value of bearing pressure to ultimate tensile strength is limited to 0.0014.

Factor of safety against Fatigue failure is 1.03

Ratio of nominal diameter of wire rope to diameter of each wire in the rope is 16

Cross-sectional area of wires in the rope is $0.404d^2$ where d is nominal diameter of wire rope.

Breaking strength of wire rope is 54000N. 15

SECTION – II

Unit – IV

7. a) Discuss standard system of Gear tooth. 6

b) Explain why Involute profile is preferred over cycloidal profile for gear tooth. 2

c) The pitch circles for a train of spur gears are shown in Fig. 2. Gear 'A' receives 3.5 kW at 700 rpm through its shaft and rotates in the clockwise direction. Gear 'B' is the idler gear while gear 'C' is the driven gear. The number of teeth on gear 'A', 'B' and 'C' are 30, 60, 40 respectively. While the module is 5 mm calculate the torque on each gear shaft and the components of gear tooth forces. Draw a free body diagram of forces and determine the reaction on the idler gear shaft. Assume 20° full depth involute system for gears. 9

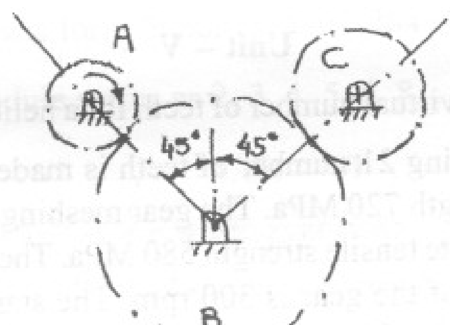

Fig. 2 [Q.7 (c)]

OR

8\. a) Discuss lubrication of gears. 3

b) A pinion having 22 number of teeth is to mesh with a gear having 60 number of teeth. Both the pinion and the gear are made up of steel having ultimate tensile strength 600 MPa and 300 MPa respectively. The pinion is connected to a 10 kW, 1440 rpm three phase induction motor. Design the gear pair and specify the surface hardness required on gear teeth using following data :

Starting torque of the motor is 30% greater than the rated torque.

Face width is ten times the module

Load distribution factor is 1.2

Deformation factor is 80 N/mm

Factor of safety 1.75

Velocity factor is given by $\frac{6}{6+V}$

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

Buckingham's equation for dynamic load is

$$P_d = \frac{21V\,(bc + P_{t\,max})}{21V + \sqrt{bc + P_{t\,max}}}$$

Standard modules are 3, 4, 5, 6, 8, 10, 12, ... 14

Unit – V

9. a) Derive a relation for virtual number of teeth for a helical gear. 3

b) A helical pinion having 21 number of teeth is made of carbon steel having ultimate tensile strength 720 MPa. The gear meshing with the pinion is made of steel having ultimate tensile strength 580 MPa. The pinion transmits 10 kW at 1000 rpm. Speed of the gear is 300 rpm. The starting torque is 125% of rated torque. Factor of safety required is 1.25. The face width is 10 times the normal module. Helix angle is 25°. The gear and the pinion are having surface hardness 300 BHN and 350 BHN respectively. Design the gear pair using Bukingham's equation for Dynamic load. Assume Deformation factor 'C' for gear pair as 11500 e N/mm

Use $K_V = \frac{5.6}{5.6 + \sqrt{V}}$

$$e = 8.0 + 0.63\left[M_n + 0.25\sqrt{d}\right]$$

$$Y' = 0.484 - \frac{2.87}{Z'}$$

$$P_d = \frac{21V\,(bc\cos^2\psi + P_{t\,max})}{21V + \sqrt{bc\cos^2\psi + P_{t\,max}}}\cos\psi$$

Standard values of normal module ..., 3, 4, 5, 6, 8, 10,... 14

OR

10. a) Explain Formative number of teeth for bevel gears. 3

b) A straight bevel pinion having 25 number of teeth is made of alloy steel having ultimate tensile strength of 840 MPa. The pinion has to mesh with a gear having 60 number of teeth and made of alloy steel having ultimate strength of 690 MPa. The pinion is connected to a 8.5 kW, 1440 rpm electric motor whose starting torque is 125% of rated torque. The factor of safety assumed is 1.1. The pinion and the gear are hardened to 450 and 400 BHN respectively. The deformation factor is given by 11000 e... N/mm where e is given by $5+0.4\phi$. The tolerance factor ϕ is given by $m+0.25\sqrt{d}$. Axes of the pinion and the gear are at right angles. Design the gear pair assuming velocity factor as $\frac{6}{6+V}$ and Lewis form factor as $Y=0.484-\frac{2.87}{Z}$

Use standard module series as 2, 3, 4, 5, 6, 8, ... 14

Unit – VI

11. a) Explain Force analysis for a worm gear drive. 6

b) A worm transmits 3 kW at 1440 rpm to a gear having 60 number of teeth. The PCD for the triple start worm is 90 mm. Module for the gear is 4 mm. The worm is having right hand helix, it is above the gear and it rotates in anticlockwise sense looking from right. Calculate force components for both the pair members. Decide direction of forces and find efficiency of drive. 10

OR

12. a) Discuss the thermal considerations in worm gear drive. 3

b) A worm gear pair 2/40/10/4 is having phosphor bronze gear with ultimate tensile strength 300 MPa. The worm is made of steel with ultimate strength 740 MPa. The coefficient of friction between the worm and the worm gear is 0.03 and normal pressure angle 20°. The worm gear wear factor is 0.9 N/mm^2. The overall heat transfer coefficient for the gear box is 18 $W/m^2{}^\circ C$. The permissible temperature rise for lubricating oil is 50°C. The worm rotates at 720 rpm and service factor is 1.5.

Determine input power rating based on beam strength, wear strength and thermal consideration.

Assume effective surface area of gear box as 0.8 m^2 and factor of safety as 1.5. 13