Reg. No. : | | | | | | | | | | |

T 3357

B.E./B.Tech. DEGREE EXAMINATION, APRIL/MAY 2008.

Sixth Semester

Mechanical Engineering

ME 1352 — DESIGN OF TRANSMISSION SYSTEMS

(Regulation 2004)

Time : Three hours Maximum : 100 marks

Usage of Approved Design Data Book is permitted.

Assumptions and assumed data have to be stated clearly.

PSG Design Data Book may be permitted.

Answer ALL questions.

PART A — ($10 \times 2 = 20$ marks)

1. Give the relationship of ratio of tensions in a V–belt drive.

2. Define maximum tension in a belt.

3. What is backlash in gears?

4. What is the advantage of helical gear over spur gear?

5. State the use of bevel gears.

6. State the advantage of worm gear drive in weight lifting machines.

7. List six standard speeds starting from 18 rpm with a step ratio 1.4.

8. Sketch the kinematic layout of gears for 3 speeds between two shafts.

9. State the advantage of cam over other reciprocating mechanisms.

10. How the "uniform rate of wear" assumption is valid for clutches?

PART B — (5 × 16 = 80 marks)

11. (a) Design a V–belt drive to transmit 50kW at 1440 rpm from an electric motor to a textile machine running 24 hours a day. The speed of the machine shaft is 480 rpm. (16)

Or

(b) A blower is to run at 600 rpm. Power to the blower is available from a motor rated 8kW at 1500 rpm. Design a chain drive for the system if the centre distance is to be 800 mm. (16)

12. (a) A motor shaft rotating at 1500 rpm has to transmit 15 kW to a low speed shaft with a speed reduction of 3:1. Assume starting torque to be 25% higher than the running torque. The teeth are 20° involute with 25 teeth on the pinion. Both the pinion and gear are made of C45 steel. Design a spur gear drive to suit the above conditions and check for compressive and bending stresses and plastic deformation. Also sketch the spur gear drive. (16)

Or

(b) A helical gear with 30° helix angle has to transmit 35 kW at 1500 rpm. with a speed reduction ratio 2.5. If the pinion has 24 teeth, determine the necessary module, pitch diameter and face width for 20° full depth teeth. Assume 15Ni 2Cr 1 Mo15 material for both pinion and wheel. (16)

13. (a) A 25 kW motor running at 1200 rpm drives a compressor at 780 rpm through a 90° bevel gearing arrangement. The pinion has 30 teeth. The pressure angle of the teeth is 20°. Both the pinion and gear are made of heat treated cast iron grade 35. Determine the cone distance, average module and face width of the gears. (16)

Or

(b) A 2 kW power is applied to a worm shaft at 720 rpm. The worm is of quadruple start type with 50 mm as pitch circle diameter. The worm gear has 40 teeth with 5 mm module. The pressure angle in the diametral plane is 20°. Determine (i) the lead angle of the worm, (ii) velocity ratio, and (iii) centre distance. Also, calculate efficiency of the worm gear drive, and power lost in friction. (16)

14. (a) Sketch the arrangements of a six speed gear box. The minimum and maximum speeds required are around 460 and 1400 rpm. Drive speed is 1440 rpm. Construct speed diagram of the gear box and obtain various reduction ratios. Use standard output speeds and standard step ratio. Calculate number of teeth in each gear and verify whether the actual output speeds are within ±2% of standard speeds. (16)

Or

(b) Design the layout of a 12 speed gear box for a milling machine having an output of speeds ranging from 180 to 2000 rpm. Power is applied to the gear box from a 6 kW induction motor at 1440 rpm. Choose standard step ratio and construct the speed diagram. Decide upon the various reduction ratios and number of teeth on each gear wheel. Sketch the arrangement of the gear box. (16)

15. (a) A multi–disk clutch consists of five steel plates and four bronze plates. The inner and outer diameters of friction disks are 75 mm and 150 mm respectively. The coefficient of friction is 0.1 and the intensity of pressure is limited to 0.3 N/mm^2. Assuming the uniform wear theory, calculate (i) the required operating force, and (ii) power transmitting capacity at 750 rpm. (16)

Or

(b) An automotive type internal–expanding double–shoe brake is shown in figure 15 b. The face width of the friction lining is 40 mm and the intensity of normal pressure is limited to 1 N/mm^2. The coefficient of friction is 0.32. The angle θ_1 can be assumed to be zero. Calculate (i) the actuating force P, and (ii) the torque–absorbing capacity of the brake. (16)

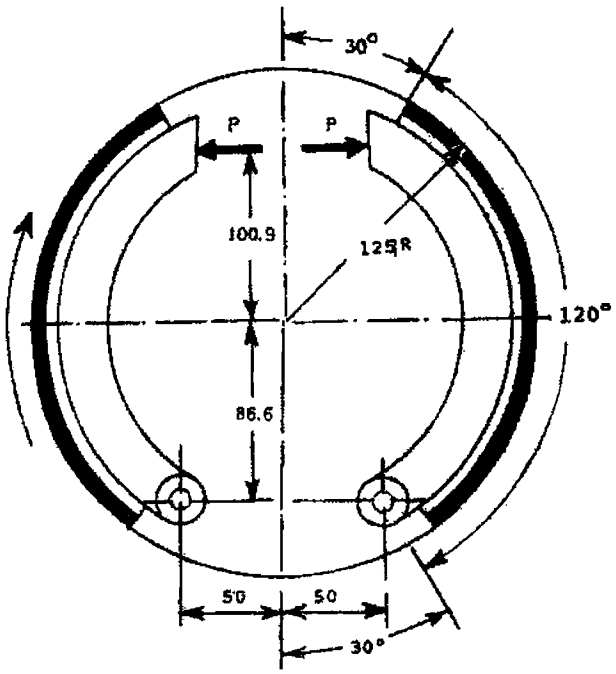

Figure 15 b.