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RN-7461

B. E. IV (Sem. VII) (Mech.) Examination

May / June – 2010

Machine Design - II

Time : **3** Hours] [Total Marks : **100**

Instructions :

(1)

નીચે દર્શાવેલ ☛ નિશાનીવાળી વિગતો ઉત્તરવહી પર અવશ્ય લખવી.
Fillup strictly the details of ☛ signs on your answer book.

Name of the Examination :

☛ **B. E. 4 (Sem. 7) (Mech.)**

Name of the Subject :

☛ **Machine Design-2**

☛ Subject Code No. : **7 4 6 1** ☛ Section No. (1, 2,.....) : **1&2**

Seat No. :

Student's Signature

(2) Figures to the right indicate full marks.
(3) Draw figures wherever necessary.
(4) Assume suitable data, if required.
(5) Use of standard design databook is permissible.

SECTION - I

1 (a) Answer the following : (any five) **10**

(i) How will you decide the power transmission capacity of a newly installed clutch? Give justification.

(ii) Compare helical gears with spur gear.

(iii) Write briefly about 20^{o} FD involute gear tooth profile. State its advantages compared to other tooth profiles.

(iv) State and explain the factors affecting selection of a suitable clutch.

(v) State and explain the materials used for gears for high power transmission capacities.

(vi) How will you designate the worm gears?

(vii) State the different uses of gear boxes.

(b) A nine speed gear box is used to connect a motor running at 720 r.p.m. through a belt drive. The gear box is to have a minimum speed of 31.5 r.p.m. and a maximum speed of 500 r.p.m. using standard spindle speeds : **10**

(i) Draw the structure and speed diagram for the arrangement.
(ii) Draw the gearbox layout.
(iii) Determine the number of teeth on each gear of the gear box.

2 (a) Design a spur gear pair to transmit 15 kW power from a motor shaft running at 1440 r.p.m. to a low speed shaft running at 500 r.p.m. The pinion has 25 no. of teeth. A 20° FD involute tooth profile is used. Both pinion and gear are made of cast iron having design stress of 55 MPa. **12**

(b) Why is the heat balance (thermal consideration) an important aspects in the design of worm gears? Explain this aspect in detail. **3**

OR

2 (a) Design a pair of helical gears to transmit 30 kW power at a speed reduction ratio of 4:1. The input shaft rotates at 2000 r.p.m. The value of helix and normal pressure angles are 25° and 20° respectively. The no. of teeth on pinion is 30. Both pinion and gear are made of steel having the following data: **12**

		Permissible Stress	Hardness (BHN)
(i)	Pinion	55 MPa	340
(ii)	Gear	40 MPa	300

(b) Explain the dynamic load consideration in case of gears in detail. **3**

3 (a) Design a multiplate clutch to transmit 15 kW power at 1000 r.p.m. Due to space limitation, the outer diameter of the clutch plate is limited to 150 mm. The maximum permissible pressure is 0.35 MPa and take inner radius equals to 0.6 times outer radius. Design the clutch considering 25% overload. **15**

(b) Design a cone clutch to transmit 10 kW power at 1000 r.p.m. The outer cone is of C.I. and forms the part of an I.C. Engine flywheel. The overall dimensions restricts the mean diameter of the cone to 300 mm. The semi cone angle is 15°. The inner cone is positioned by means of a centrally placed helical spring. The allowable shear stress for the shaft material is 45 MPa. The allowable pressure intensity is 0.09 MPa and the coefficient of friction is 0.2. Design the clutch considering 25% more torque (overload).

OR

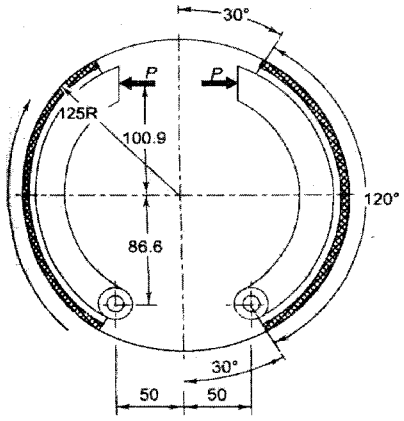
3 (a) Design a single dry disc clutch (having both sides effective) to transmit 12.5 kW power at 1000 r.p.m. The intensity of pressure is 0.06 MPa and $\mu = 0.25$. The ratio of mean radius to radial face width is 4. Assume uniform wear criteria for the design. The design shear stress is 45 MPa for the shaft. **15**

(b) The following data refers to a centrifugal clutch :
Mass of each shoe = 4 kg
No. of shoes = 4
Inner diameter of the drum = 320 mm
Distance of shoe C.G. from the spider centre = 135 mm
Coefficient of friction between shoes and drum = 0.25
Running speed = 1000 r.p.m.
Shoe engagemnt speed = 750 r.p.m.
Determine :
(i) Power transmission capacity of the clutch
(ii) Size of each shoe, if p = 0.1 MPa.

SECTION - II

4 (a) Answer the following : (any five) **10**
(i) What is the difference between stresses in thin and thick cylinders?
(ii) Discuss the basic principle of working of brakes.
(iii) Define : "Reliability" and "MTBF". Also explain it.
(iv) Explain the importance of service factor in belt design. Explain this.
(v) Draw a neat sketch of pulley. Why crown is provided to it?
(vi) Explain hydrodynamic lubrication with the help of neat sketches.

(b) An automotive type internal expanding double-shoe brake is shown in figure. The face width of the friction lining is 40 mm and the maximum intensity of normal pressure is limited to 1 N/mm^2. The coefficient of friction is 0.32. The angle $\varnothing_1$ can be assumed to be zero. Calculate: (i) the actuating force P and (ii) the torque-absorbing capacity of the brake. **10**



5 A hydraulic press has a maximum capacity of 1000 kN. A piston diameter is 250 mm. calculate the wall thickness if the cylinder is made of material for which the permissible strength may be taken as 80 MPa. This material may be as a brittle material. **5**

OR

5 (a) If a device has a failure rate 2×10^6 failures per hour. What is its reliability in an operating period of 500 hours? If there are 2000 items in the test, how many failures are expected? Assume a constant failure rate. Also calculate the MTBF. **5**

(b) A flat belt pulley 900 mm diameter is driven by V-belts from a 250 mm diameter V-pulley, running at 1600 r.p.m. the drive transmits 15 kW. **10**

(i) The center distance between the pulleys = 1 m
(ii) Angle of groove of V-pulley = 40^o
(iii) Coefficient of friction between belt and pulley = 0.2
(iv) Density of belt material = 1100 kg/m^3 and
(v) Allowable stress = 2.1 MPa.

Determine the number of belts (C-size) each having a cross-sectional area of 230 mm^2.

OR

(b) Derive the equation for the length of belt for cross-belt drive. **10**

6 (a) Derive the pressure distribution in hydrostatic bearing and derive the equation for flow requirement for hydrostatic step bearing. **7**

OR

(a) Derive Stribeck's Equation. **7**

(b) A single row deep-groove ball bearing is subjected to a radial force of 8 kN and a thrust force of 3 kN. The shaft rotates at 1200 r.p.m. the expected life L_{10h} of the bearing is 20,000 hr. the minimum acceptable diameter of the shaft is 75 mm. selected a suitable ball bearing for this application. **8**
