BENGAL ENGINEERING AND SCIENCE UNIVERSITY, SHIBPUR
B.E. (Mech.) 7^{TH} SEMESTER FINAL EXAMINATION, 2013

TRIBO-DESIGN OF MACHINE ELEMENTS (ME – 701)

Full Marks: 70 **Time: 3 hrs**

Use separate answer script for each half.
Answer SIX questions, taking THREE from each half.
All questions carry equal marks.

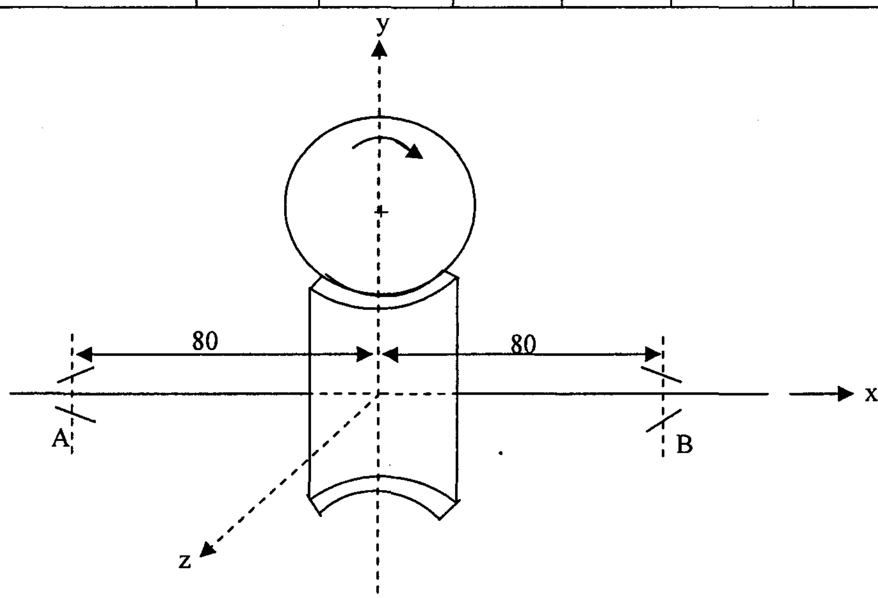
FIRST HALF

1. A 10 kW power at 1000 r.p.m. is supplied to the worm shaft as shown in the figure below. The worm gear drive is designated as 2/40/10/5. The worm has left hand threads and pressure angle is 20^0. The worm wheel is mounted between two tapered roller bearings A and B. Find the speed of worm wheel and its direction of rotation. Draw two figures showing (i) the relative sliding velocity of worm shaft with respect to worm wheel and (ii) resultant tangential force, radial force and axial force acting on the worm wheel.

The gearbox for the worm gears has an effective surface area of 1.5 m^2. A fan is mounted on the worm shaft to circulate air over the surface of the fins. The overall coefficient of heat transfer can be taken as 25W/m^2/°C. The permissible temperature rise of the lubricating oil above the atmospheric temperature is 50°C.
Calculate the power transmitting capacity based on thermal considerations.

The following data of coefficient of friction for given rubbing speed may be used:

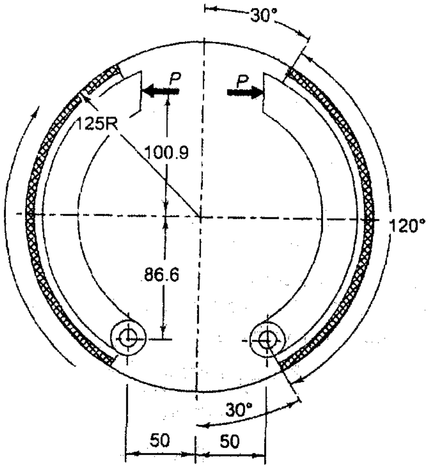
Rubbing Speed (m/s)	0.5	1.0	2.5	2.7	3.0	5.0	10
Coefficient of friction	0.065	0.055	0.048	0.042	0.038	0.032	0.025



(All dimensions are in mm)

2. In the given problem in Q. No.1 for power transmission of 10 kW, draw a complete free body diagram of the shaft and find the reactions at the two tapered roller bearings at A & B. Following AFBMA and Timken's prescription, find out the Required Radial Ratings of the two tapered roller bearings. Given that the desired life of bearings, application factor, the ratio of basic radial rating to basic thrust rating are 12000hrs, 1.35 and 1.5 respectively.

3. An automotive type internal expanding double-shoe brake is shown in the figure below. The face width of the friction lining is 50 mm and the allowable intensity of normal pressure is limited to 1.2 MPa. The coefficient of friction of lining material with the drum is 0.33. State clearly all the assumptions and find the distribution of contact pressures on the frictional lining materials. Calculate from the first principle (i) the actuating force P, (ii) torque absorbing capacity of the brake & (iii) the reactions at both the pivots. Assume that the friction lining starts immediately from the pivot location and ends at a position making an angle of contact of 120^0 with the centre of the drum as shown in the figure.



(All dimensions are in mm)

4. (a) State the appropriate mechanical properties needed in selecting worm and worm wheel materials and cite examples.

   (b) Stating all the assumptions, derive the Stribeck's Equation for static load capacity of a ball bearing.

   (c) From the hertzian contact theory, write the expressions of contact area, maximum hertzian contact stress and maximum subsurface shear stress developed for contact of a pair of teeth of pinion & gear (spur), having pitch circle radii "R_p" & "R_g" in mm respectively. The driver pinion, having pressure angle φ, transmits power of "x" kW with "n" rpm. The modulus of elasticity and poisson's ratio of both pinion and gear materials are "E" GPa & "υ" respectively.

   (d) What is the meaning of the Greek word 'Tribos'? State the sources of friction and different wear mechanisms. Explain the mechanism of spalling failure of rolling contact bearings.

SECOND HALF

5. (a) Establish the expressions for load capacity and frictional power loss of a circular step thrust bearing.
   (b) The following data is provided for a circular step thrust bearing of a vertical turbo generator:

Inner diameter	= 80 mm
Outer diameter	= 300 mm
Film thickness	= 0.05 mm
Absolute viscosity of oil	= 0.04 Pas
Speed of the runner	= 1200 rpm
Recess pressure	= 3.5 MPa

   Determine –
   (i) load capacity (ii) frictional power loss

6. (a) Discuss the various dimensionless design parameters of a line contact.
   (b) Using Hertzian contact pressure distribution, determine the load capacity for an unlubricated point contact.
   (c) Derive the expression for the radius, 'a' of the developed contact zone from a point contact between two spheres of radii, R_1 and R_2 under the action of a load, W. Assume the dry point contact.
   If two spheres are made of steel having E = 210 GPa and Poisson's ratio, $\sigma = 0.3$, determine the value of the developed contact zone radius, 'a' using the following basic equation of deflection at r = a

$$w = a\left(\frac{1-\sigma^2}{4E}\right)\pi p_{max}$$ with the usual notations.

7. (a) Define the bearing modulus of a lubricated journal bearing.
   (b) Discuss the composition and applications of Tin-based babbit as bearing material.
   (c) A journal of diameter of 120 mm operating at 1000 rpm is supported by means of a 180 mm long full bearing which is subjected to a radial load of 38 kN. It is desirable to limit the bearing temperature to 62° C and the surrounding (inlet) temperature is 30° C. If the average energy dissipation coefficient is 880 W / m^2 – ° C, determine the followings:
      (i) Operating temperature of oil, t_o using Cameron's equation
      (ii) Absolute viscosity of oil at t_o
      (iii) Coefficient of friction, μ using McKee's equation

(iv) Heat generated

(v) Heat dissipated. Comment on the possibility of artificial cooling.

Use Vogel's equation for viscosity – temperature variation with the following data:

Absolute viscosity of oil, η at 30° C = 20×10^{-3} Pas

Absolute viscosity of oil, η at 90° C = 5×10^{-3} Pas

Take D / C = 1000 and leakage factor, $K_f = 0.002$

8. (a) What is the advantage of a pressure – fed oil grooved journal bearing in application?

(b) Deduce the expression for total oil flow from both ends of a pressure – fed journal bearing having a circumferential oil groove operating with an eccentricity ratio of ε_0.

(c) A journal bearing is fed by SAE – 30 oil at 50° C and 250 kPa supply pressure. The radial clearance is 0.05 mm. The journal rotates at 40 rev / sec, supporting a bearing load of 5 kN. The SAE – 30 oil has a viscosity coefficient of 0.03 Pas corresponding to a temperature of 50° C. The mass density of oil is 860 kg / m^3 and specific heat, C_H is 1760 J / kg – ° C.

Given that length of bearing = 60 mm and diameter of journal = 50 mm.

Find –

(i) total oil flow rate from bearing

(ii) operating Sommerfeld number at inlet condition

(iii) temperature rise of oil assuming that entire heat generated due to power loss in bearing is carried out by oil.

The following data may be consulted to find out μ(R/C) corresponding to Sommerfeld number, S.

S	μ(R/C)
0.14	2.14
0.16	2.33
0.18	2.61
0.20	2.93

The above data is referred to L / D = 1.2.