BENGAL ENGINEERING & SCIENCE UNIVERSITY, SHIBPUR

B.E. (Mech.) Part- III 6th Semester Examination, 2012

Design of Machine Elements – II (ME-601)

Time: 3 hours Full Marks: 70

Use separate answerscript for each half.
Answer SIX questions, taking THREE from each half.
The questions are of equal value.

FIRST HALF

1. a) Deduce an expression based on uniform wear theory, for the axial force and torque transmitting capacity of a frictional clutch having a single pair of contact surface in terms of the maximum intensity of pressure and geometrical parameters of the disc clutch.

   b) A multiple disc clutch has 5 steel and 4 bronze plates. The clutch is required to transmit 6 kW at 850 rpm. The ratio of inner to outer diameters is 0.7. The coefficient of friction for frictional materials is 0.35. Find the inside and outside diameters and also the axial force required.

2. a) What are the desirable properties of frictional lining materials of clutches?

   b) Mention the main factors to be considered in selecting a type of clutch. Explain any one of them.

   c) An industrial cone clutch is to be designed to transmit 40 kW at 1000 rpm. The design data available is –
   semi cone angle, $\alpha = 12^0$
   allowable normal pressure, p = 0.6 MPa
   coefficient of friction for frictional material(asbestos), $\mu = 0.3$
   Given that the mean diameter of the clutch is 6 times the face width, b. Assuming the uniform wear condition, find –
   (i) the diameters at smaller and larger ends
   (ii) face width
   (iii) axial force required for the engagement of clutch

3. a) With a neat sketch, define (i) pitch cone (ii) pitch cone centre (iii) pitch cone distance (iv) back cone (v) bevel factor of a bevel gear

   b) A pair of straight bevel gears is mounted on shafts which are intersecting at right angles. The number of teeth on the bevel pinion is 24. Teeth are 14.5^0 full depth involute type. The pinion shaft is connected to an electric motor developing 14 kW at 600 rpm. The drive has a speed ratio of 4: 3. The pinion and gear are made of cast steel and heat treated to a surface hardness of 380 BHN. Both bevel pinion and gear are finished to meet the specification of grade 2 for which the deformation factor, λ is 125 kN / m. The Young's modulli for both gears are 210 GPa. Calculation has shown that the module at the larger end of teeth is 4 mm and the face width is 26 mm.

Using Buckingham's equation for dynamic load and wear load, you are requested to ascertain whether the gear parameters, as obtained from the strength consideration, are satisfactory for dynamic and wear load conditions or not.

Determine also the radial and axial forces on the shaft mounting the bevel gear.

4. a) Derive the expression for the pitch cone distance in terms of pitch circle diameters of bevel pinion and gear.

   b) A pair of straight bevel gear is mounted on shafts which are intersecting at right angles. The number of teeth on the pinion and gear are 30 and 45 respectively. The teeth are 20^0 full depth involute type. The pinion shaft is connected to an electric motor developing 12 kW rated power at 900 rpm. The service factor can be taken as 0.85. The pinion and gear are made of C30 steel for which the allowable static working stress, σ_w is 180 MPa and are heat treated to a surface hardness of 360 BHN. The pinion and gear teeth are finished to meet the specification of grade 2. In the initial stage of gear design, assume that the pitch line velocity, V is approximately 6 m / sec.

   Determine the module at the larger diameter, face width and pitch cone distance of gears. Check the design against the dynamic and wear loads. Assume the following data :

   Deformation factor (λ) for grade two accurate teeth = 200 kN / m

   Young's modulus of C30 steel = 210 GPa

   Velocity factor $(C_v) = \dfrac{3}{3+V}$, V is in m / sec.

   Surface endurance limit, $\sigma_{en} = \{2.8 \times (BHN) - 70\}$ MPa

   Load – stress factor, $K_W = \dfrac{\sigma_{en}^{\ 2} \sin\phi . \cos\phi}{1.40}\left(\dfrac{1}{E_p} + \dfrac{1}{E_g}\right)$ Pa

   Lewis form factor, $Y = \pi\left(0.154 - \dfrac{0.912}{Z}\right)$, Z = No. of teeth

5. a) Two co-axial machine rotors has moments of inertia, I_1 and I_2 and running at uniform speeds, ω_1 and ω_2 respectively are engaged by a frictional clutch. If the clutch torque remains constant, show that the total energy lost in the clutch during slipping is given by

$$\frac{1}{2} \cdot \frac{I_1 I_2}{I_1 + I_2} . (\omega_1 + \omega_2)^2$$

   b) Explain the concept of the formatic no. of teeth on bevel gear. What is the relation between the formatic and actual numbers of teeth?

SECOND HALF

ote: *In case you make any assumption, state it clearly. Assume and state any unstated data that may be required to solve problems. Use the following in appropriate form, as and when necessary. Symbols have their usual meaning.*

Lewis form factor: $Y = \pi\left(0.154 - \frac{0.912}{T}\right)$ Dynamic Load: $F_d = F_t + \frac{21V\left(\lambda b\cos^2\psi + F_t\right)\cos\psi}{21V + \sqrt{\lambda b\cos^2\psi + F_t}}$; where V is in m/s and $\lambda = \frac{0.111e}{\frac{1}{E_p} + \frac{1}{E_g}}$; Load stress factor $k_s = \frac{\sigma_{ens}^2}{1.4}\sin\phi_n\left(\frac{1}{E_p} + \frac{1}{E_g}\right)$; where $\sigma_{ens} = \{2.8(BHN) - 70\}$ in MPa

6a) How were the helical teeth of a helical gear conceived and for what reason?

b) State the conditions of meshing of two helical gears.

c) Derive the Lewis equation of a straight tooth spur gear. Now derive the modified form of the Lewis equation for a pair of helical gear teeth transmitting power. State the assumptions and restrictions clearly.

7) Design the module, face width, pitch circle diameters and centre distance of a pair of 20° full depth external straight tooth spur gear to transmit 50 kW from a pinion running at 750 rpm to a gear running at 150 rpm. Go for a compact design and check your design both against dynamic and wear load. In case of failure in any checking, do not repeat calculations but suggest possible remedies to overcome such failure. Forged C30 steel with UTS of 500MPa and CI (grade 35) with UTS of 350MPa may be used as pinion and gear material respectively. Load is steady with no shock and the service condition is normal.

The gears are cut with class 2 accuracy with the tooth action error as 0.03mm. Surfaces are so heat treated so that the gears have an average BHN of 250. Take E_p = 210 GPa and E_g = 105 GPa. Standard modules (first choice) in mm are: 1, 1,125, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16, 20. Velocity factor may be taken as $\frac{3}{3+V}$; V being the pitch line velocity in m/s.

8a) An 18 teeth helical pinion rotating at 2000 rpm transmits 7.5kW to a 45 teeth gear. Normal module, normal pressure angle and the helix angle are 6mm, 20° and 23° respectively. Determine the tangential, radial and axial components of the tooth force between the meshing teeth. Also calculate the minimum face width required for the gears.

b) Two helical gears are used in a speed reducer to transmit 75kW at a pinion speed of 1200 rpm. The speed ratio is 3:1. Determine the normal module, face width, number of teeth in each gear for a compact design based on strength. The pinion steel has the UTS of 550MPa while that of the gear is 420MPa. For both the gears the modulus of elasticity is 210GPa. Normal pressure angle and helix angle are 20° and 23° respectively. The design is to be checked against dynamic load with a permissible tooth action error as 0.02 mm. The velocity factor may be taken as $\frac{15}{15+V}$, V being the pitch line velocity in m/s. Load is steady with no shock and service condition is normal. Checking against wear is not necessary.

9a) Derive the necessary and sufficient conditions for a function f(**X**) to be minimum, where **X** is a vector of design variables $X = (x_1, x_2, \ldots, x_n)^T$. There is no constraint.

b) What is an active constraint?

c) The deflection of a rectangular beam with a fixed length is inversely proportional to the width and cube of the depth. Find the cross-sectional dimensions (width, depth) of the beam for its minimum deflection. The beam is cut from a cylindrical log of radius r. Formulate the optimization problem, express it in the standard form and solve by the Lagrange Multiplier method.

10a) Write the Kuhn-Tucker necessary conditions for minimizing a function f(**X**), subject to an equality constraint $h(X) = 0$ and an inequality constraint $g(X) \le 0$. **X** is the vector of design variables $X = (x_1, x_2, \ldots, x_n)^T$. State the restriction on the solution, if any. No derivation is necessary.

b) Obtain values of x_1, x_2 that minimizes $f(x_1, x_2) = (x_1 - 1)^2 + (x_2 - 1)^2$, subject to $x_1 + x_2 - 4 \le 0$ and $2 - x_1 \le 0$.