CBCS SCHEME

USN

18ME62

Sixth Semester B.E. Degree Examination, Jan./Feb. 2023
Design of Machine Elements – II

Time: 3 hrs. Max. Marks: 100

Note: 1. Answer any FIVE full questions, choosing ONE full question from each module.
2. Use of design data hand book is permitted.
3. Missing data if any, may suitably be assumed.

Module-1

1 a. A railway Wagon moving at a velocity of 1.5 m/s is brought to rest by a bumper consisting of two helical springs arranged in parallel. The mass of the Wagon is 15,000 kg. The springs are compressed by 150 mm in bringing the Wagon to rest. The spring index can be taken as 6. The springs are made of oil-hardened and tempered steel wire with ultimate tensile strength of 1250 N/mm^2 and modulus of rigidity of 81,370 N/mm^2. The permissible shear stress for the spring wire can be taken as 50% of the ultimate tensile strength. The springs should have square and ground ends. Design the spring. **(10 Marks)**

b. Discuss the significance of nipping of leaf springs with appropriate sketch. **(04 Marks)**

c. A semi-elliptic leaf spring used for automobile suspension consists of three extra-full length leaves and 15 graduated length leaves including the master leaf. The centre-to-centre distance between two eyes of the spring is 1 m. The maximum force that can act on the spring is 75 kN. For each leaf, the ratio of width to thickness is 9 : 1. The modulus of elasticity of the leaf material is 207 GPa. The leaves are pre-stressed in such a way that when the force is maximum, the stresses induced in all leaves are same and equal to 450 N/mm^2. Determine (i) the width and thickness of leave ; (ii) the initial nip (iii) the initial pre-load required to close the gap 'C'. **(06 Marks)**

OR

2 a. Describe the phenomenon of creep and slip in the belt drive. **(04 Marks)**

b. It is required to select a V-belt drive from a normal torque motor of 5 kW capacity, which runs at 1440 rpm to a light duty compressor running at 970 rpm. The compressor runs for 24 hours per day. Space is available for a centre distance of about 500 mm. Assume that the pitch diameter of the driving pulley is 150 mm. Design the V-belt. **(08 Marks)**

c. It is required to select a 6×19 wire rope with 1569 as tensile designation for a hoist on the basis of long life. The weight of the hoist along with the material is 5 KN. It is to be raised from a depth of 100 m. The maximum speed of 5 m/s is attained in 5 seconds. Determine the size of wire rope and the sleave diameter for long life on the basis of the fatigue as failure criterion. Take 0.5 kg/m as mass per unit length of the wire rope. 70 KN as the breaking strength of the wire rope. What is the factor of safety of this wire rope under static conditions? Take the dimensionless quantity $\frac{P}{S_{ut}} = 0.0015$ for long fatigue life. **(08 Marks)**

Module-2

3 a. Describe gear tooth failure modes. **(04 Marks)**

b. It is required to design a pair of spur gears with 20° full-depth involute teeth based on the Lewis equation. The velocity factor is to be used to account for dynamic load. The pinion shaft is connected to a 10 KW, 1440 rpm motor. The starting torque of the motor is 150% of the rated torque. The speed reduction is 4 : 1. The pinion as well as the gear is made of plain carbon steel with an allowable static stress of 200 N/mm^2. Design the gears, specify their dimensions and suggest suitable surface hardness for the gears. Take a f.o.s 1.5 for beam strength. The minimum number of teeth on pinion is 18. Endurance limit for checking the beam strength of the teeth is 259 N/mm^2. Take face width to module ratio as 10. Assume carefully cut gears (class II). **(16 Marks)**

OR

4 a. Obtain Lewis equation for the beam strength of a spur gear tooth. **(04 Marks)**

b. A pair of helical gears with a 23° helix angle is to transmit 2.5 kW at 10,000 rpm of pinion. The velocity ratio is 4 to 1. Both pinion and gear are to be made of hardened steel with an allowable stress $\sigma_d = 100\,\text{MPa}$. The gears are 20° stub and the pinion to have 24 teeth. Determine minimum diameter of the gear that may be used and the required BHN. Take wear and lubrication factor as 1.15. Ratio of face width to normal module as 10. **(16 Marks)**

Module-3

5 a. Describe formative number of teeth for a bevel gear. **(02 Marks)**

b. A pair of right angle bevel gears is to be used to transmit 9 kW. The number of teeth an pinion is 21 and on the gear is 60. The material of the pinion is steel with allowable static stress of 85 MPa and that of the gear is C.I with 55 MPa. The pinion rotates at 1200 rpm and the gear at 420 rpm. The tooth profile is $14\frac{1}{2}^{\circ}$ (14.5 degree) composite. The teeth are to be generated. Take $C_S = 1.5$, b = 10 m. The gears are expected to be precission cut. Determine the required module and diameters of the gears. Design for strength using the Lewis equation and check for wear, considering the effect of overhanging. Suggest suitable surface hardness for the gear pair. **(18 Marks)**

OR

6 a. List any four applications of worm gears. **(02 Marks)**

b. A pair of worm and worm wheel is designated as, 1/30/10/10. The input speed of the worm is 1200 rpm, The worm wheel is made of centrifugally cast, phosphor bronze and the worm is made of case-hardened carbon steel. Determine the power transmitting capacity based on, (i) the beam strength (ii) wear strength

Bending stress factor for worm = 28.2
and worm wheel = 7
Speed factor for strength of worm = 0.25 and
For worm wheel = 0.48
Speed factor for wear of worm = 0.112
and for worm wheel = 0.26
Surface stress factor for worm = 4.93
and for worm wheel = 1.55
zone factor = 1.143 **(18 Marks)**

Module-4

7 a. Explain any six desirable properties of a good friction material used in clutches. **(06 Marks)**

b. A multi-disk clutch consists of five steel plates and four bronze plates. The inner and outer diameters of the friction disks are 75 and 150 mm respectively. The coefficient of friction is 0.1 and the intensity of pressure on friction lining is limited to 0.3 N/mm^2. Assuming uniform wear theory, calculate (i) the required force to engage the clutch, and (ii) Power transmitting capacity at 750 rpm. **(06 Marks)**

c. A cone clutch with asbestor friction lining transmits 30 kW power at 500 rpm. The coefficient of friction is 0.2 and the permissible intensity of pressure is 0.35 N/mm^2. The semi cone angle is 12.5°. The outer diameter is fixed as 300 mm from space limitations. Assuming uniform wear theory, calculate ;

(i) The inner diameter.
(ii) The face width of the friction lining
(iii) The force required to engage the clutch. **(08 Marks)**

OR

8 a. A single block brake with a torque capacity of 250 Nm is shown in Fig.Q8 (a). The brake drum rotates at 100 rpm and the co-efficient of friction is 0.35. Calculate

(i) The actuating force and hinge-pin reaction for clockwise rotation of the drum.

(ii) The actuating force and hinge-pin reaction for anticlockwise rotation of the drum.

(iii) The dimensions of the block, if the intensity of pressure between the block and brake drum is 1 N/mm^2. The length of the block is twice its width.

State whether the brake is self locking

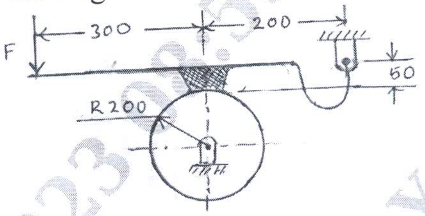

Fig. Q8 (a) **(12 Marks)**

b. A differential band brake is shown in Fig. Q8 (b). The width and thickness of the steel band are 100 mm and 3 mm respectively and the maximum tensile stress in the band is 50 N/mm^2. The coefficient of friction between the friction lining and the brake drum is 0.25. Calculate (i) the tensions in the band (ii) the actuating force (iii) the torque capacity of the brake. Find out whether the brake is self-locking

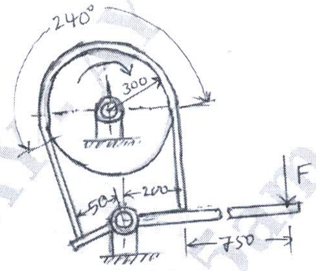

Fig. Q8 (b) **(08 Marks)**

Module-5

9 a. Obtain Petroff's equation for co-efficient of friction. Mention two assumptions. **(06 Marks)**

b. A 75 mm long full journal bearing of diameter 75 mm supports a load of 12 kN on a journal turning at 1800 rpm. Assuming a r/c ratio of 1000, and an oil of viscosity 0.01 kg/ms at the operating temperature. Determine the coefficient of friction by using (i) the McKee equation, (ii) the Raimondi and Boyd curve (iii) also determine the amount of heat generated using the coefficient of friction as calculated by the McKee equation, and (iv) determine the probable surface temperature of the bearing, using the following equation and assuming that the heat generated in all dissipated in still air at 20°C.

$$H_d = \frac{(\Delta T + 18)^2}{0.484} LD \times 10^{-6}$$ **(14 Marks)**

OR

10 a. Describe (i) Static load carrying capacity and (ii) Dynamic load carrying capacity with respect to anti-friction bearings. **(04 Marks)**

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 rpm. The expected life L_{10h} of the bearing is 20,000h. The minimum acceptable diameter of the shaft is 75 mm. Select a suitable ball bearing for this application. **(10 Marks)**

c. A single row deep groove ball bearing is subjected to an axial thrust of 1000 N and a radial load of 2200 N. Find the expected life that 50% of the bearings will complete under this condition. Take C_O = 2500 N and C = 5590 N. **(06 Marks)**

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