MACHINE DESIGN – II
(MECH 3201)

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

Figures out of the right margin indicate full marks.

Candidates are required to answer Group A and any 5 (five) from Group B to E, taking at least one from each group.

Candidates are required to give answer in their own words as far as practicable.

Group – A
(Multiple Choice Type Questions)

1. Choose the correct alternative for the following: **10 × 1 = 10**

(i) In Lewis equation, gear tooth is considered as
(a) simply supported beam (b) cantilever beam
(c) curved beam (d) Over hung beam.

(ii) Which of the following type of gears provide maximum velocity ratio?
(a) spur gears (b) bevel gears
(c) worm gears (d) helical gears.

(iii) Buckingham's equation of gear tooth is based on
(a) maximum crushing stress in gear tooth
(b) maximum bending stress in gear tooth
(c) maximum shear stress in gear tooth
(d) maximum contact stress in gear tooth.

(iv) Two bevel gears with 25 and 50 teeth are in mesh with each other. The pitch angle of pinion is
(a) $\tan^{-1} 0.5$ (b) $\tan^{-1} 2$ (c) $\sin^{-1} 0.5$ (d) $\sin^{-1} 2$

(v) A pair of worm gears is designated as (1/30/10/8). The center distance between the worm and worm wheel is
(a) 160 mm (b) 30 mm (c) 80 mm (d) 96 mm

(vi) The thickness of high-pressure oil and gas pipes is determined by
(a) Lame's equation (b) Clavarino's equation
(c) Birnie's equation (d) Barlow's equation.

(vii) The friction moment in a clutch with uniform wear as compared to friction moment with uniform pressure is
(a) more (b) equal (c) less (d) more or less depends on speed.

(viii) When the frictional force helps to apply the brake, the brake is said to be
(a) partially self energizing (b) self locking (c) back stop (d) self acting.

(ix) $1 N\text{-}s/mm^2 = ?$
(a) 10^2 P (b) 10^6 P (c) 10^7P (d) 10^9P

(x) Boundary lubricated bearing is
(a) thick film bearing (b) thin film bearing
(c) hydrodynamic bearing (d) hydrostatic bearing.

Group – B

2. (a) What are the advantages of involute teeth gears?

(b) A train of spur gears is shown in Fig.1. Gear 1 is the driving gear and transmits 5 kW power at 720 rpm. The number of teeth on gears 1, 2, 3 and 4 are 20, 50, 30 and 60 respectively. The module for all gears is 4 mm. The gears have a 20° full-depth involute profile. Calculate & show direction with proper sketch, the tangential and radial components of the tooth force between (i) Gears 1 and 2 and (ii) Gears 3 and 4.

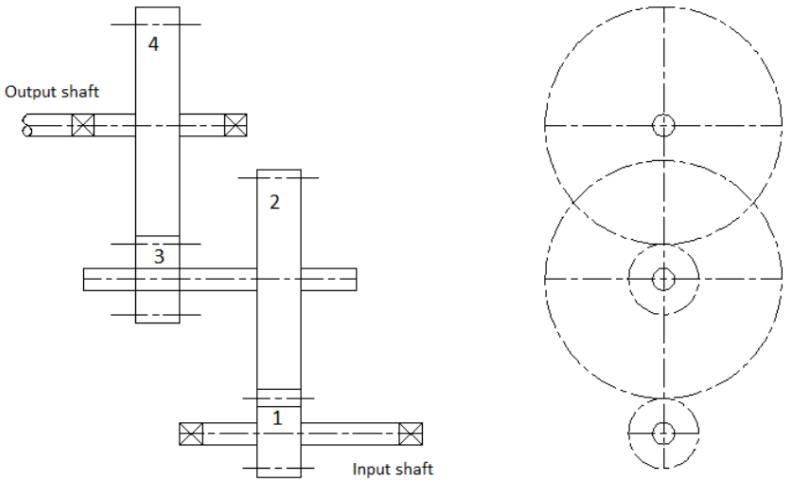


Fig.1

4 + (4 + 4) = 12

3. (a) Why is the pinion weaker than the gear made of same material?

(b) A pair of spur gears consists of a 24 teeth pinion, rotating at 1000 rpm and transmitting power to a 48 teeth gear. The module is 6 mm, while the face width is 60 mm. Both gears are made of steel with an ultimate tensile strength of 450 N/mm2. They are heat treated to a surface hardness of 250 BHN. Assume that velocity factor accounts for the dynamic load. Calculate (i) beam strength; (ii) wear strength; and (iii) the rated power that the gears can transmit, if service factor and the factor of safety are 1.5 and 2, respectively.

3 + (4 + 3 + 2) = 12

Group – C

4. The following data is given for a pair of parallel helical gears made of steel: power transmitted = 20 kW, speed of pinion = 720 rpm, number of teeth on pinion = 35,

number of teeth on gear = 70, centre distance = 285 mm, normal module = 5 mm, face width = 50 mm, normal pressure angle = 20°, ultimate tensile strength = 600 N/mm2, surface hardness = 300 BHN, grade of machining = Gr. 6, service factor = 1.25.
Calculate (i) the helix angle; (ii) the beam strength; (iii) the wear strength; (iv) the static load; (v) the dynamic load by Buckingham's equation; (vi) the effective load; (vii) the effective factor of safety against bending failure; and (viii) the effective factor of safety against pitting failure.

12

5. (a) A pair of straight bevel gears has a velocity ratio of 2:1. The pitch circle diameter of the pinion is 80 mm at the large end of the tooth. 5 kW power is supplied to the pinion, which rotates at 800 rpm. The face width is 40 mm and the pressure angle is 20°. Calculate & show the direction with proper sketch, the tangential, radial and axial components of the resultant tooth force acting on the pinion.

(b) A pair of worm gears is designated as 2/54/10/5. Explain.

8 + 4 = 12

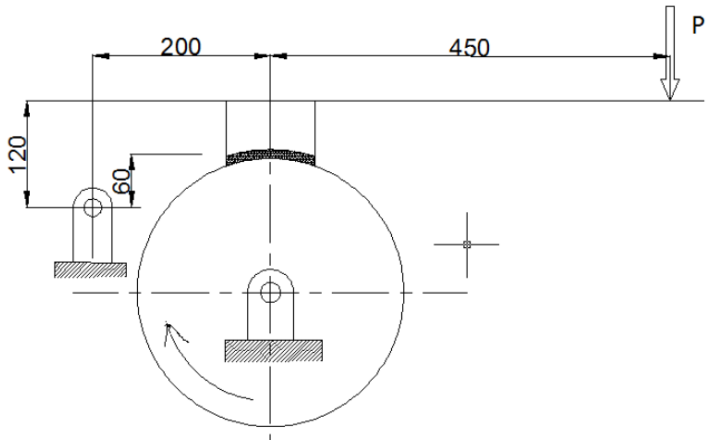
Group – D

6. (a) What are the types of stresses in closed end thick cylinders? Show the distribution of stresses in a thick cylinder under internal pressure with proper sketch.

(b) A hydraulic cylinder with closed ends is subjected to an internal pressure of 15 MPa. The inner and outer diameters of the cylinder are 200 mm and 240 mm respectively. The cylinder material is cast iron FG 300. Determine the factor of safety used in design. If the cylinder pressure is further increased by 50%, what will be the factor of safety?

(2 + 3) + 7 = 12

7. (a) A single block brake with a torque capacity of 15 N-m is shown in Fig.2. The coefficient of friction is 0.3 and the maximum pressure on the brake lining is 1 N/mm2. The width of the block is equal to its length and the drum radius is 150 mm. Calculate (i) the actuating force; (ii) the dimensions of the block; (iii) the resultant hinge-pin reaction.



All dimensions are in mm
Fig.2

(b) A multi-disk clutch consists of two steel disks with one bronze disk. The inner and outer diameters of the contacting surfaces are 200 and 250 mm respectively. The coefficient of friction is 0.1 and the maximum pressure between the contacting surfaces is limited to 0.4 N/mm2. Assuming uniform wear theory, calculate the required force to engage the clutch and the power transmitting capacity at 720 rpm.

(2 + 2 + 2) + (3 + 3) = 12

Group – E

8. (a) Compare rolling and sliding contact bearing with load characteristics curve. Write down Petroff's equation.

(b) The following data is given for a full hydrodynamic bearing used for electric motor: radial load= 1200N, journal speed= 1440 rpm, journal diameter= 50mm, static load on the bearing= 350N. The value of surface roughness of the journal and the bearing are 2 and 1 micron respectively. The minimum oil film thickness should be 6 times the sum of surface roughness of the journal and the bearings. Determine i) length of the bearing and ii) minimum oil film thickness. Consider the range of permissible bearing pressure in the application of an electric motor is from 0.7 to 1.5 N/mm^2.

(4 + 2) + 6 = 12

9. (a) What is L_{10} life and L_{50} life? State the difference between dynamic load carrying capacity and equivalent bearing load for ball bearing.

(b) A ball bearing with a dynamic load capacity of 22.8 kN is subjected to a radial load of 10 kN. Calculate i) the expected life in million revolutions that 90% of the bearing will reach; ii) the corresponding life in hours, if the shaft is rotating at 1450 rpm and iii) the life that 50% of the bearings will complete or exceed before fatigue failure.

(3 + 3) + 6 = 12

Refer the empirical relationships & tables given below for appropriate data wherever suitable.

Table 1: Values of Lewis form factor Y for 20° full-depth involute system

z	Y	z	Y	z	Y
15	0.289	27	0.348	55	0.415
16	0.295	28	0.352	60	0.421
17	0.302	29	0.355	65	0.425
18	0.308	30	0.358	70	0.429
19	0.314	32	0.364	75	0.433
20	0.320	33	0.367	80	0.436
21	0.326	35	0.373	90	0.442
22	0.330	37	0.380	100	0.446
23	0.333	39	0.386	150	0.458
24	0.337	40	0.389	200	0.463
25	0.340	45	0.399	300	0.471
26	0.344	50	0.408	Rack	0.484

Ratio factor for external gears $Q = 2z_g/(z_g + z_p)$
Load stress factor $K = 0.16 (BHN/100)^2$

The values of velocity factor are as follows:

(i) For v < 10 m/s
$C_v = 3/(3+v)$

(ii) For v < 20 m/s
$C_v = 6/(6+v)$

(iii) For v > 20 m/s
$C_v = 5.6/ (5.6+\sqrt{v})$

Table 2: Tolerances on the adjacent pitch

Grade	e (microns)
6	8.00 + 0.63 φ

φ = m + 0.25 √d' where φ = tolerance factor, m = module (mm), d' = pitch circle diameter

Table 3: Values of deformation factor C (N/mm^2)

Materials		14.5° full depth teeth	20° full depth teeth	20° stub teeth
Pinion material	Gear material			
Grey CI	Grey CI	5500	5700	5900
Steel	Grey CI	7600	7900	8100
Steel	Steel	11000	11400	11900

Buckingham equation for dynamic load in helical gears
$P_d = \{21v(Ceb \cos^2\psi + P_t) \cos\psi\} / \{21v + \sqrt{(Ceb \cos^2 \psi + P_t)}\}$

Department & Section	Submission Link
ME	https://classroom.google.com/c/MzAwMzcwNDY4MTE0/a/MzU3NTUwMjgyOTc2/details