PIETS's College of Engineering, Pune [COEP]

(An Autonomous Institute of Government of Maharashtra)

END SEMESTER EXAMINATION

(PE – 304) KINEMATICS AND DYNAMICS OF MACHINES

Programme :- T. Y. B. Tech. (Production Sandwich)
Duration :- 4 Hrs.
Max. Marks :- 50

Date :- 30/04/2013
Year :- 2012 – 13
Semester :- II

Instructions:-

1. Q. No. 1 is compulsory.
2. Attempt only five questions.
3. Figures to the right indicate full marks.
4. Draw neat sketches wherever required.
5. Use of non-programmable calculators is allowed.
6. Assume suitable data wherever necessary.

Q.1. (A) A gear shaft bearing is subjected to a load of 15 kN in the radial direction and 5 kN in the axial direction. The expected life of the radial deep groove ball bearing is 2000 hrs with the shaft running at 500 rpm. Select a bearing from the sizes. Assume that the inner race rotates and the load factor is 1.1. **(5)**

Bearing Number	Dynamic load carrying capacity, C (N)	Static load carrying capacity, C_o (N)
6010	21600	13200
6210	35100	19600
6310	61800	36000
6410	87100	52000

(F_a/C_o)	$(F_a/F_r) \le e$		$(F_a/F_r) > e$		e
	X	Y	X	Y	
0.025	1	0	0.56	2.0	0.22
0.040	1	0	0.56	1.8	0.24
0.070	1	0	0.56	1.6	0.27
0.130	1	0	0.56	1.4	0.31
0.250	1	0	0.56	1.2	0.37
0.500	1	0	0.56	1.0	0.44

(B) (i) Two 20^0 involute spur gears have a module of 10 mm. The addendum is one module. The larger gear has 50 teeth and the pinion 13 teeth. Does the interference occur? If it occurs, to what value should the pressure angle be changed to eliminate interference? **(5)**

(ii) A follower is operated by a uniformly rotating cam. The follower is raised through a distance of 40 mm in 120^0 rotation of the cam, remains at rest for the next 30^0 and is lowered during further 120^0 rotation of the cam. The raising of the follower takes place with cycloidal motion and the lowering with uniform acceleration and deceleration. However, the uniform acceleration is $3/4^{th}$ of the uniform deceleration. Draw the displacement diagram for the cam which will be further helpful for construction of the cam profile.

Q.2. (A) A punching press is required to punch 40 mm diameter holes in a plate of 15 mm thickness at the rate of 30 holes per minute. It requires 6 N-m of energy per mm^2 of sheared area. If the punching takes 1/10 of a second and the r.p.m. of the flywheel varies from 160 to 140, determine the mass of the flywheel having radius of gyration of 1m. **(5)**

(B) In an in-line six cylinder engine working on two-stroke cycle, the cylinder centre lines are spaced at 600 mm. In the end view, the cranks are 60^0 apart and in the order 1-4-5-2-3-6. The stroke of each piston is 400 mm and the connecting rod length is 1 m. The mass of the reciprocating parts is 200 kg per cylinder and that of rotating parts 100 kg per crank. The engine rotates at 300 r.p.m. Examine the engine for the balance of primary and secondary forces and couples. Find the maximum unbalanced forces and couples. **(5)**

Q.3. (A) Write a short note of any one absorption type dynamometer. **(5)**

(B) Derive an expression for dynamic deflection of steady state vibratory motion of a mechanical system excited by rotating eccentric mass. **(5)**
A machine of 100 kg mass has a 20 kg rotor with the eccentricity of 0.5 mm. The machine is mounted on springs having stiffness 85 x 10^3 N/m and the damping ratio 0.02. The machine vibrates in vertical plane and operates at 600 rpm. Find out the dynamic amplitude of the machine.

Q.4. (A) A worm gear box with an effective surface area of 1.5 m^2 is operating in still air with a heat transfer coefficient of 15 $W/m^2\ ^0C$. The temperature rise of the lubricating oil above the atmospheric temperature is limited to 50^0C. The worm gears are designated as 1/30/10/8. The worm shaft is rotating at 1440 r.p.m. and the normal pressure angle is 20^0. Calculate the power transmitting capacity based on the thermal considerations. **(5)**
Use the following relation for coefficient of friction:
$\mu = 0.0765/(V_s + 0.4)^{(1/2)}$ where, V_s = sliding velocity, m/s

(B) Derive an expression for transmissibility and show that transmissibility is lower at higher operating speeds. **(5)**
A machine having mass of 1000 kg is mounted on four springs, having net effective stiffness of 1.96 x 10^6 N/m. The machine is subjected to a harmonic external force of amplitude 490 N with frequency 180 r.p.m. Find out the amplitude of motion of the machine and maximum force transmitted to the foundation during vibrations.

Q.5. (A) Answer the following questions: **(5)**
(i) Explain clearly the terms 'static balancing' and 'dynamic balancing'. State the necessary conditions to achieve them.
(ii) What is the importance of Petroff's equation? State the assumptions.
(iii) What is 'bearing characteristic number' and 'bearing modulus' as applied to the journal bearing?
(iv) Discuss any four guidelines/parameters/factors/features to be considered while selecting a proper type of rolling contact bearing.
(v) Define bearing life related to rolling contact bearings.

(B) A pair of straight bevel gears with 20^0 full-depth involute teeth, manufactured by generation, consists of 24 teeth bevel pinion rotating at 2880 r.p.m. meshing with 60 teeth bevel gear. The axes of pinion and gear intersect at right angle. The module at the large end of the teeth is 4 mm, while the face width is 32 mm. The pinion is made of alloy steel, while gear is made of plain carbon steel for which the permissible bending stresses are 200 N/mm^2 and 150 N/mm^2 respectively. The pinion and gear are to be case hardened to 350 BHN and 300 BHN respectively. Determine: **(5)**
(i) the beam strength, and
(ii) the wear strength

$$Y = 0.484 - \left(\frac{2.87}{Z'}\right) \qquad K = 0.16\left(\frac{BHN}{100}\right)^2$$

Q.6. (A) The following data is given for a hydrostatic step bearing: **(5)**
Thrust load = 400 KN, Shaft speed = 700 rpm, Shaft diameter = 480 mm, Recess diameter = 240 mm, Oil film thickness = 0.15 mm, Absolute viscosity of lubricant = 29 X 10^{-9} N-s/mm^2, Specific Heat of lubricant = 1.76 KJ/kg^{0}C, Specific gravity of lubricant = 0.86. Calculate:
(i) supply pressure,
(ii) flow requirement in l/min,
(iii) frictional power loss,
(iv) pumping power loss,
(v) temperature rise.
Assume that the total power loss is converted into frictional heat.

(B) The following data is given for a 20^0 full-depth involute steel helical gear pair transmitting 150 kW power form a shaft rotating at 1440 r.p.m. to another parallel shaft rotating at 360 r.p.m. The centre distance is *approximately* 435 mm. **(5)**

Helix Angle	=	24^0	Face Width	=	14 m_n
Number of Teeth on Pinion	=	20	Service Factor	=	1.53
Permissible Bending Stress for Pinion Material	=	152 N/mm^2	Combined teeth error	=	0.0406 mm
Permissible Bending Stress for Gear Material	=	125 N/mm^2	Deformation Factor	=	11600 e N/mm

Pinion is weaker than gear in bending and the maximum tangential load that the pinion tooth can transmit without bending failure is 47163.3 N. Select the standard normal module and determines the exact centre distance. Assuming the dynamic load is accounted by the Buckingham's Equation, calculate:
(i) The factor of safety against bending failure; and
(ii) The surface hardness, if the factor of safety against pitting failure is 1.5.

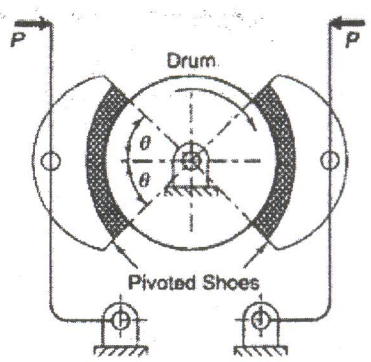
Buckingham's Equation for Dynamic Load,

$$P_d = \frac{21\,V\,(bC\cos^2\psi + P_t)\cos\psi}{21\,V + \sqrt{bC\cos^2\psi + P_t}} \qquad \text{and} \qquad K = 0.16\left(\frac{BHN}{100}\right)^2$$

The standard normal modules in mm recommended by ISO are:

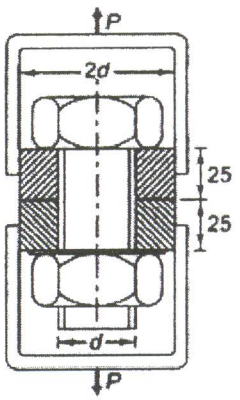
First Choice	1.0, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16, 20, 25, 32, 40 and 50
Second Choice	1.125, 1.375, 1.75, 2.25, 2.75, 3.5, 4.5, 5.5, 7, 9, 11, 14, 18, 22, 28, 36 and 45.

Q.7. (A) A pivoted block brake, as shown in figure, has two shoes, which subtend an angle (2θ) of 100^0. The diameter of the brake drum is 500 mm and the width of the friction lining is 100 mm. The coefficient of friction is 0.2 and the maximum intensity of pressure between the lining and the brake drum is 0.5 N/mm^2. The pivot of each shoe is located in such a manner that the moment of the frictional force on the shoe is zero. Calculate: **(5)**
(i) the distance of the pivot from the axis of the brake drum
(ii) the torque capacity of each shoe
(iii) the reactions at the pivot



Pivoted Double Block Brake

(B) The assembly of two circular plates clamped together by means of a bolt, as shown in figure, is subjected to a variable force P varying from 0 to 10 kN. The bolt is made of plain carbon steel 45C8 (S_{ut} = 630 N/mm^2, S_{yt} = 380 N/mm^2, and E = 207000 N/mm^2). The two circular plates are made of aluminium (E = 71000 N/mm^2). The fatigue stress concentration factor is 2.2 and the expected reliability is 90% (Reliability factor = 0.897). The initial preload in the bolt is 5 kN. Determine the size of the bolt if the factor of safety is 2. The surface finish factor is incorporated in the fatigue stress concentration factor. The size factor is 1 for axial load. **(5)**



Basic dimensions for ISO metric screw threads (fine series)

Designation	*Nominal or major dia d/D (mm)*	*Pitch (p) (mm)*	*Pitch diameter d_p/D_p (mm)*	*Minor diameter*		*Tensile stress area (mm^2)*
				d_c (mm)	D_c (mm)	
M 6 × 1	6	1.00	5.350	4.773	4.917	20.1
M 6 × 0.75	6	0.75	5.513	5.080	5.188	22.0
M 8 × 1.25	8	1.25	7.188	6.466	6.647	36.6
M 8 × 1	8	1.00	7.350	6.773	6.917	39.2
M 10 × 1.25	10	1.25	9.188	8.466	8.647	61.2
M 10 × 1	10	1.00	9.350	8.773	8.917	64.5
M 12 × 1.5	12	1.50	11.026	10.160	10.376	88.1
M 12 × 1.25	12	1.25	11.188	10.466	10.647	92.1
M 16 × 1.5	16	1.50	15.026	14.160	14.376	167
M 16 × 1	16	1.00	15.350	14.773	14.917	178
M 20 × 2	20	2.00	18.701	17.546	17.835	258
M 20 × 1.5	20	1.50	19.026	18.160	18.376	272
M 24 × 2	24	2.00	22.701	21.546	21.835	384
M 24 × 1.5	24	1.50	23.026	22.160	22.376	401
M 30 × 3	30	3.00	28.051	26.319	26.752	581
M 30 × 2	30	2.00	28.701	27.546	27.835	621
M 36 × 3	36	3.00	34.051	32.319	32.752	865
M 36 × 2	36	2.00	34.701	33.546	33.835	915
M 42 × 4	42	4.00	39.402	37.093	37.670	1150
M 42 × 3	42	3.00	40.051	38.319	38.752	1210
M 48 × 4	48	4.00	45.402	43.093	43.670	1540
M 48 × 3	48	3.00	46.051	44.319	44.752	1600