CMR INSTITUTE OF TECHNOLOGY

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Internal Assessment Test II May 2017

SUBJECT: Design of Machine Elements-II								Code:	10ME62
Date:	08/05/2017	Duration:	90 min	Max. Marks:	50	Sem:	06	Branch:	MECH

Note: Answer any two questions.

Use of Design Data Hand Book is permitted.

Q. No.		Question	Marks	OBE MAP CO	OBE MAP RBT
		PART – A			
1		A pair of spur gears with 20^0 full depth involute teeth consists of 20 teeth pinion meshing with 41 teeth gear. The module is 3mm while the face width is 40mm. The material for the pinion as well as gear is Steel with an ultimate tensile strength of 600 MN/m^2. The gears are heat treated to a surface hardness of 400 BHN. The pinion rotates at 140 rpm and the service factor for the application is 1.75. Assume that the velocity factor accounting for the dynamic load and factor of safety is 1.5. Determine the rated power that the gears can transmit.	[25]	CO6	L3
2		A pair of helical gears is to transmit 15KW. The teeth are 20^0 stub in diametral plane and have a helix angle of 45^0. The pinion has 80 mm pitch diameter and operates at 10,000 rpm. The gear has 320 mm pitch diameter. If the gears are made of cast steel (σ_0=100 MPa), determine suitable module and face width. The pinion is heat treated to a Brinell of 300 and gear has a Brinell hardness of 200. Check the design for dynamic and wear loads.	[25]	CO6	L3
3	A	An industrial cone clutch has a minor diameter of 180 mm and a major diameter of 200mm. The half cone angle is 15^0. The material for friction lining has a coefficient of friction of 0.35 and an allowable bearing pressure of 0.6 MPa. Determine i) Axial force capacity ii) Torque Capacity iii) Power transmitted at 750 rpm. iv) Axial force for engaging cone clutch.	[10]	CO3	L3
	B	A multiple disc clutch Steel on Bronze is to transmit 4 KW at 750 rpm. The inner radius of contact is 40mm and outer radius of contact is 70mm. The clutch operates in oil with an expected coefficient of friction of 0.1. The average allowable pressure is 350KN/m^2 maximum.(Low design pressures are used to provide for sufficient size to give good heat dissipation capacity.) Assume uniform wear. 1) How many total discs of Steel and Bronze are required? 2) What is the average pressure? 3) What axial force is required? 4) What is the actual Maximum pressure?	[15]	CO3	L3

2016-17 (even Semester)

Solutions for IAT 2 question paper

Sem: VI Staff: RPR.

Subject: Design of Machine Elements II

Subject code: 10ME62 Date of exam: 08.05.2017

Q 1 A pair of spur gears with 20° full depth involute teeth consists of 20 teeth pinion with 41 teeth gear. The module is 3mm while the face width is 40mm. The material for pinion as well as gear is steel with an ultimate tensile strength of 600MN/m². The gears are heat treated to a surface hardness of 400BHN. The pinion rotates at 140 RPM and the service factor for the application is 1.75. Assume that the velocity factor accounting for the dynamic load and factor of safety is 1.5. Determine the rated power that the gears can transmit.

Ans: data

$\alpha = 20°$ FDI

$z_1 = 20$

$z_2 = 41$

$m = 3mm$

$b = 40mm$

$\sigma_{u1} = \sigma_{u2} = 600\,MPa = 600\,N/mm^2$

BHN for pinion = 400

BHN for gear = 400

$n_1 = 400\,RPM$

$C_s = 1.75$

$C_v = 1.5$

$FS = 1.5$

$\sigma_{b1} = \sigma_{b2} = \frac{\sigma_u}{FS} = \frac{600}{1.5} = 400\,N/mm^2$

to find

1) Rated power (N).

1) Rated power as per beam Strength

$$F_t = \frac{C_v \cdot \sigma_b \cdot b \cdot Y \cdot m}{1.5}$$

here $C_v = 1.5$

$\sigma_{b1} = 400\ N/mm^2$

$b = 40\ mm$

$m = 3\ mm$.

Since both pinion & gear are made of same material, pinion is weaker.

$$Y_1 = \pi\left[0.154 - \frac{0.912}{20}\right]$$

$$= 0.340.$$

$$\therefore F_t = \frac{1.5 \times 400 \times 40 \times 340 \times 3}{1.5}$$

$$= 16,320\ N.$$

Now, $$F_t = \frac{9550 \times 10^3 \times N}{n_1} \times \frac{C_s}{r_1}.$$

taking $C_s = 1.5$,

$d_1 = mZ_1$

$= 3 \times 20$

$= 60\ mm$

$\therefore r_1 = 30\ mm$

$$16320 = \frac{9550 \times 10^3 \times N}{140} \times \frac{1.5}{30}$$

$\Rightarrow N = 4.784\ KW.$

2) Rated power as per endurance strength

Endurance strength of pinion (F_f) $= \sigma_{ef} \cdot b \cdot y_1 \cdot m$.

σ_{ef} for steel (BHN = 400) $= 689.6$ MPa.
$= 689.6\ N/mm^2$
(T23.33/P23.76).

$\therefore F_f = 689.6 \times 40 \times 0.340 \times 3$
$= 28,135.68$ N.

For safe design $F_f \geq F_d$.

consider $F_f = F_d$ for safe power.

$$F_d = F_t + \frac{21v(F_t + bc)}{21v + \sqrt{F_t + bc}}.$$

$$v = \frac{\pi d_1 n_1}{60,000} \text{ m/sec} = \frac{\pi \times 60 \times 140}{60,000}$$
$$= 0.439 \text{ m/sec}.$$

For $v = 0.439$ m/sec, allow error$(f) = 0.15$ mm.

For steel & steel & 20° FDI system,

For $f = 0.125$ mm, $c = 1450$ N/mm

For $f = 0.15$ mm, $c = ?$

$$\frac{0.15}{0.125} \times 1450$$
$= 1740$ N/mm.

$$28135 = F_t + \frac{(21 \times 0.439)[F_t + (40 \times 1740)]}{[(21 \times 0.439) + \sqrt{F_t + (40 \times 1740)}]}$$

Solving $F_t = 3087.56$ N.

Now $F_t = \dfrac{9550 \times 10^3 \times N_2}{n_1} \times \dfrac{C_s}{r_1}$

$$3087.56 = \frac{9550 \times 10^3 \times N_2}{140} \times \frac{1.5}{30}$$

$\Rightarrow N_2 = 0.905$ KW.

Selecting the lower of N_1 & N_2

safe power = 0.905 KW.

Q2 A pair of helical gears is to transmit 15 KW. The teeth are 20° stub in diametral plane and have a helix angle of 45°. The pinion has 80mm pitch diameter and operates at 10,000 rpm. The gear has 320mm pitch diameter. If the gears are made of cast steel (σ_0 = 100MPa), determine suitable module and face width. The pinion is heat treated to a brinell of 300 and gear has a Brinell hardness of 200. Check the design for dynamic & wear loads.

Ans: data

$P = 15$ KW.

$\alpha_t = 20°$ stub

$\beta = 45°$

$d_1 = 80$mm.

$n_1 = 10{,}000$ rpm.

$d_2 = 320$mm.

$i = \dfrac{320}{80} = 4.$

$n_2 = \dfrac{10{,}000}{4} = 2500$ rpm.

Materials : Pinion : cast Steel $\sigma_{01} = 100MPa$ (5)

gear : cast Steel $\sigma_{02} = 100MPa$.

BHN for Pinion = 300
BHN for gear = 200

1. Design tangential tooth load (F_t)

$$F_t = \frac{9550 \times 10^3 \times P}{n} \times \frac{C_s}{z}$$

$$= \frac{9550 \times 10^3 \times 15}{10,000} \times \frac{1.5}{40} \quad (\text{taking } C_s = 1.5)$$

$$= 537 N.$$

2. module (m_t)

Since pressure angle is measured in diametral plane (or) transverse plane, m_t should be a standard value.

$$\therefore F_t = \frac{K_v \cdot \sigma_0 \cdot b \cdot y \cdot m_t \cos\beta}{C_w} \quad 23.286/P23.54$$

Since both pinion & gear are made of same material, pinion is weaker.

$$V_m = \frac{\pi d_1 n_1}{60,000} = 41.88 m/sec.$$

$$\therefore K_v = \frac{5.6}{5.6 + \sqrt{V_m}} = 0.46 \quad (\text{as } V_m > 20 m/sec)$$

(29.290a/ P23.57)

Assume $b = 10 m_t$.

$$Z_{v1} = \frac{Z_1}{\cos^3\beta} = \frac{d_1}{m_t} \cdot \frac{1}{\cos^3\beta} = \frac{80}{m_t} \cdot \frac{1}{\cos^3 45^\circ}$$

$$= \left(\frac{226}{m_t}\right)$$

$$Z_{v2} = \frac{Z_2}{\cos^3\beta} = \frac{d_2}{m_t} \cdot \frac{1}{\cos^3\beta} = \frac{320}{m_t} \cdot \frac{1}{\cos^3 45^\circ}$$

$$= \left(\frac{905}{m_t}\right).$$

$$\therefore Y_1 = \pi\left[0.17 - \frac{0.95}{\left(\frac{226}{m_t}\right)}\right]$$

$$= (0.534 - 0.013\,m_t).$$

$$Y_2 = \pi\left[0.17 - \frac{0.95}{\left(\frac{905}{m_t}\right)}\right]$$

$$= 0.534 - (3.29\times10^{-3})m_t$$

assuming scant lubrication, $f_{es} = 1.25$.

Applying Lewis eq^n to pinion,

$$537 = \frac{0.46 \times 100 \times 10m_t \times (0.534 - 0.013 m_t) \times m_t \times \cos 45^\circ}{1.25}$$

Solving for m_t, $m_t = 40.98\,mm$, 2.01 mm, 1.09 mm

$\therefore$ select $m_t = 2.5\,mm$

3. No. of teeth (Z_1, Z_2)

$$Z_{v1} = \frac{226}{2.5} = 90.4$$

but $Z_{v1} = \frac{Z_1}{\cos^3\beta} \Rightarrow Z_1 = 31.96$ say 32.

$\therefore z_2 = z_1 \times 4$
$= 32 \times 4$
$= 128.$

4. Face width (b)

$b = 10\, m_t$
$= 10 \times 2.5$
$= 25\, mm.$

but $b_{min} = \frac{1.15 \pi m_t}{\tan \beta}$
$= 9.03\, mm.$

$\therefore$ Select $b = 25\, mm.$

5. gear teeth proportions

$m_n = m_t \cos \beta$
$= 2.5 \cos 45^\circ$
$= 1.77\, mm.$

Addendum $h_a = 0.8\, m_n$
$= 0.8 \times 1.77$
$= 1.416\, mm$

Dedundum $(h_f) = 1 m_n$
$= 1.77\, mm.$

Working depth $(h') = 1.6 m_n$
$= 2.83\, mm.$

Total depth $(h) = 1.8\, m_n$
$= 3.186\, mm.$

Clearance $(c) = 0.2 m_n$
$= 0.354\, mm.$

⑧

6. Check for dynamic load (F_d).

$$F_d = F_t + \frac{21v\,(F_t + bC\cos^2\beta)\cos\beta}{21v + \sqrt{(F_t + bC\cos^2\beta)}}$$

For $v_m = 41.88$ m/sec, $f = 0.01$mm. (Fig 23.35a, p 23.35)

$\therefore C = 150.1$ N/mm (T 23.32).

$$\therefore F_d = 537 + \frac{21\times 41.88\,[537 + 17\times150.1\times\cos^2 45^\circ]\cos 45^\circ}{21\times 41.88 + \sqrt{537 + (17\times 150.1 \times \cos^2 45^\circ)}}$$

$= 2153$ N.

Endurance strength $F_f = \sigma_f \cdot b \cdot y \cdot m_t \cos\beta$.

σ_f for cast steel $= 290$MPa. (T 23.33, P 23.76)

$\therefore F_f = 290 \times 17 \times 0.501 \times 2.5 \cos 45^\circ$

$= 6420$N.

Since $F_f > F_d$, the design is safe.

7. Check for wear

Wear load $F_w = \dfrac{d_1 \cdot b \cdot Q \cdot K}{\cos^2\beta}$.

$$Q = \frac{2z_2}{z_1+z_2} = \frac{2\times128}{(32+128)} = 1.6.$$

$K = 0.9095$ (For BHN $= 300, 200$) (T 23.37B)

$$\therefore F_W = \frac{80 \times 25 \times 1.6 \times 0.9035}{\cos^2 45^\circ}$$

$$= 5780\ N.$$

Since $F_W > F_d$, the design is safe.

3 A An industrial cone clutch has a minor diameter of 180 mm and a major diameter of 200 mm. The half cone angle is 15°. The material for friction lining has a co-eff. of friction of 0.35 and an allowable bearing pressure of 0.6 MPa. Determine

i) Axial force capacity
ii) Torque capacity
iii) Power transmitted at 750 rpm.
iv) axial force for engaging cone clutch.

Ans: data

$D_1 = 180$ mm
$D_2 = 200$ mm
$\Rightarrow D_m = \frac{D_1 + D_2}{2}$ (for uniform wear)
$= 190$ mm.

$\alpha = 15^\circ$

$\mu = 0.35$

$P = 0.6\ N/mm^2$

to find

i) F_a

ii) M_t

iii) N at $n = 750$ rpm.

iv) F_a'.

(i) F_a

$$F_a = \pi D_m \cdot P \cdot b \sin\alpha$$

here $D_m = D_1 + b\sin\alpha$.

$$190 = 180 + b\sin 15^\circ$$

$$\Rightarrow b = 38.63\,mm.$$

$$\therefore F_a = \pi \times 190 \times 0.6 \times 38.63 \times \sin 15^\circ$$

$$= 3580.76\,N.$$

ii) M_t

$$M_t = \frac{\mu F_a \cdot D_m}{2\sin\alpha}$$

$$= \frac{0.35 \times 3580.76 \times 190}{2\sin 15^\circ}$$

$$= 460 \times 10^3\ N\text{-}mm.$$

iii) N at n = 750 RPM.

$$M_t = 9550 \times 10^3 \times \frac{N}{n}.$$

$$460 \times 10^3 = 9550 \times 10^3 \times \frac{N}{750}$$

$$\Rightarrow N = 36.12\,KW.$$

(iv) F_a'

$$F_n = \frac{F_a}{\sin\alpha}$$

$$= 13835\ N.$$

$$F_a' = F_n(\sin\alpha + \mu\cos\alpha).$$

$$= 13835(\sin 15^\circ + 0.35\cos 15^\circ)$$

$$= \underline{8.25\,KN}$$

3B. A multiple disc clutch steel on Bronze is (11) to transmit 4KW at 750RPM. The inner radius of contact is 40mm and the outer radius of contact is 70mm. The clutch operates in oil with an expected co-eff. of friction of 0.1. The average allowable pressure is 350 KN/m² maximum. (Low design pressures are used to provide for sufficient size to give good heat dissipation capacity.)

Assume uniform wear.

Ans: data

$N = 4KW$.

$n = 750 RPM$

$R_1 = 40 mm$

$\Rightarrow D_1 = 80 mm$.

$R_2 = 70 mm$

$\Rightarrow D_2 = 140 mm$

$\mu = 0.1$

$(P_m)_{allow} = 350 \; KN/m^2$

$= 0.35 MN/m^2$

$= 0.35 MPa$.

to find

1) i_1, i_2

2) P_m

3) F_a

4) P_{max}.

data

$N = 4 KW$

$n = 750 RPM$

$R_1 = 40 mm$

$\therefore D_1 = 80 mm$

$R_2 = 70 mm$

$\Rightarrow D_2 = 140 mm$

$\mu = 0.1$

$(P_m)_{allow} = 350 KN/m^2$
$= 350 KPa$
$= 0.35 MPa$
$= 0.35 N/mm^2$

$$D_m = \frac{D_1 + D_2}{2} = \frac{80 + 140}{2} = 110 mm$$

to find

1) i_1, i_2

2) P_m

3) F_a

4) P_{max}

1) i_1, i_2

$$F_a = \frac{1}{2} \pi P_m D_m (D_2 - D_1)$$

$$M_t = \frac{1}{2} \mu F_a \cdot D_m \cdot i$$

$$= \frac{1}{2} \mu \left[\frac{1}{2} \pi P_m D_m (D_2 - D_1) \right] \left(\frac{D_1 + D_2}{2} \right) \times i$$

$$= \frac{\pi \mu P_m D_m (D_2^2 - D_1^2) \times i}{8}$$

$$M_t = 9550 \times 10^3 \times \frac{N}{n}.$$

$$= 9550 \times 10^3 \times \frac{4}{750}$$

$$= 50.93 \times 10^3 \text{ N-mm}.$$

$$50.93 \times 10^3 = \frac{\pi \times 0.1 \times 0.35}{16} \times 110 \left(140^2 - 80^2\right) i$$

$$\Rightarrow i = 2.55.$$

Selecting i to be even,

Select $i = 4$.

$\therefore i_1 = \frac{i}{2} = 2$ (No. of driver plates)

$i_2 = \frac{i}{2} + 1 = 3$. (No. of driven plates).

3) $\underline{F_a}$: Using 4 pairs of contact surfaces, a reduced axial force can be used

$$M_t = \frac{1}{2} \mu F_a D_m i$$

$$50.93 \times 10^3 = \frac{1}{2} \times 0.1 \times F_a \times 110 \times 4$$

$$\Rightarrow F_a = 2315 \text{ N}.$$

2) $\underline{P_m}$

$$F_a = \frac{1}{2} \pi P_m D_m (D_2 - D_1).$$

$$2315 = \frac{1}{2} \pi P_m \times 110 (140 - 80)$$

$$\Rightarrow P_m = 0.223 \text{ N/mm}^2.$$

4) $\underline{P_{max}}$

$$P_{max} D_1 = P_m D_m.$$

$$P_{max} \times 80 = 0.223 \times 110$$

$$\Rightarrow P_{max} = 0.307\,N/mm^2.$$