20ME602

Hall Ticket Number:

III/IV B.Tech (Regular) DEGREE EXAMINATION

July/August, 2023 **Mechanical Engineering**

Sixth Semester **(SCHEME)** **Design of Transmission Elements**

Time: Three Hours **Maximum:** 70 Marks

Answer question 1 compulsory. **(14X1 = 14Marks)**

Answer one question from each unit. **(4X14=56 Marks)**

			CO	BL	M
1	a)	What type of stresses is induced in shafts?	CO1	L2	1M
	b)	How the shaft is designed when it is subjected to twisting moment only?	CO1	L2	1M
	c)	How are the keys classified? State their applications.	CO1	L1	1M
	d)	How does the working of a clamp coupling differ from that of a muff coupling? Explain	CO1	L2	1M
	e)	What are rolling contact bearings?	CO2	L1	1M
	f)	Write the various types of lubrications.	CO2	L1	1M
	g)	Write the various properties of lubricants.	CO2	L1	1M
	h)	How does the function of a brake differ from that of a clutch	CO3	L2	1M
	i)	What are the advantages of V-belt drive over flat belt drive?	CO3	L2	1M
	j)	What is meant by a clutch?	CO3	L1	1M
	k)	What is the main function of a flywheel in an engine?	CO4	L1	1M
	l)	Discuss the various types of stresses induced in a flywheel rim.	CO4	L2	1M
	m)	Define a gear?	CO4	L1	1M
	n)	Explain how the gears are classified? Name the various types of gears.	CO4	L2	1M
		Unit-I			
2	a)	Write the applications of split muff couplings?	CO1	L1	4M
	b)	Design a muff coupling to connect two shafts transmitting 40KW at 120rpm. The permissible shear and crushing stress for the shaft and key material (mild steel) are 30Mpa and 80Mpa respectively. The material of muff is cast Iron with permissible shear stress of 15Mpa. Assume that the maximum torque transmitted is 25 percent greater than mean torque.	CO1	L4	10M
		(OR)			
3	a)	A 10kW power is transmitted at 800 rpm, from a motor shaft, through a key, to a machine shaft by a means of a pulley and a belt. Design the key. Take the allowable shear stress and crushing stress are 45MPa and 100Mpa.	CO1	L4	6M
	b)	A shaft is required to transfer 43kW of power at 600rpm. The outside diameter must not exceed 50mm and the maximum shear stress is not to exceed 70N/mm2. Find out the dimensions of hollow and solid shaft, which would meet these requirements.	CO1	L3	8M
		Unit-II			
4		A shaft, 150mm in diameter, rotates in a bearing at 2000rpm. The length of the bearing is 1.4 times its diameter. The bearing pressure is 1N/mm2, and the coefficient of friction at the bearing surface is 0.005. Determine the power loss in friction. The temperature of the bearing is entirely controlled by the flow of oil through the bearing. If the difference between the outlet and inlet temperatures is 15 ^{0}C, determine the quantity of coolant oil required, if the specific heat of the oil is 1900 J/kg/ ^{o} C.	CO2	L3	14M
		(OR)			
5	a)	State any four advantages of Rolling contact bearings over sliding contact bearings.	CO2	L2	4M
	b)	A rolling contact bearing is subjected to the following work cycle: (a) Radial load of 6000N at 150 r.p.m for 25% of the time; (b) Radial load of 7500N at 600 r.p.m for 20% of the time; and (c) Radial load of 2000N at 300 r.p.m for 55% of the time. The inner ring rotates and loads are steady .Select a bearing for an expected average life of 2500 hours.	CO2	L4	10M

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Unit-III

6		A V- belt is to transmit 20 kW from a 250 mm pitch diameter sheave to a 900mm diameter pulley. The centre distance between the two shafts is 1000 mm. The groove angle is 40 degree and the coefficient of friction between the belt and the sheave and also between the belt and the pulley is 0.2. The cross-section of the belt is 40 mm wide at the top, 20 mm wide at the bottom and 25 mm deep. The density of the belt is 1000 kg/m3 and the allowable tension per belt is 1000 N. Find the number of belts required.	CO3	L3	14M
		(OR)			
7	a)	A multiple disc clutch, steel on bronze, is to transmit 4.5 kW at 750 r.p.m. The inner radius of the contact is 40 mm and outer radius of the contact is 70 mm. The clutch operates in oil with an expected coefficient of 0.1. The average allowable pressure is 0.35 N/mm2. Find : 1. The total number of steel and bronze discs; 2. the actual axial force required and 3. the actual maximum pressure.	CO3	L3	10M
	b)	Discuss the different types of brakes giving atleast one practical application for each.	CO3	L2	4M
		Unit-IV			
8		The intercepted areas between the output torque curve and the mean resistance line of a turning moment diagram for a multicylinder engine, taken in order from one end are as follows: – 35, + 410, – 285, + 325, – 335, + 260, – 365, + 285, – 260 mm2. The diagram has been drawn to a scale of 1 mm = 70 N-m and 1 mm = 4.5°. The engine speed is 900 r.p.m. and the fluctuation in speed is not to exceed 2% of the mean speed. Find the mass and cross-section of the flywheel rim having 650 mm mean diameter. The density of the material of the flywheel may be taken as 7200 kg / m3. The rim is rectangular with the width 2 times the thickness. Neglect effect of arms, etc.	CO4	L3	14M
		(OR)			
9		A pair of straight teeth spur gears, having 20° involute full depth teeth is to transmit 12 kW at 300 r.p.m. of the pinion. The speed ratio is 3: 1. The allowable static stresses for gear of cast iron and pinion of steel are 60 MPa and 105 MPa respectively. Assume the following: Number of teeth of pinion = 16; Face width = 14 times module; Determine the module, face width and pitch diameter of gears. Check the gears for wear; given σ_{es} = 600 MPa; E_P = 200 kN/mm2 and E_G = 100 kN/mm2.	CO4	L4	14M

			CO	BL	1M
1	a)	What type of stresses is induced in shafts? The following stresses are induced in the shafts. Shear stresses due to the transmission of torque (due to torsional load). Bending stresses (tensile or compressive) due to the forces acting upon the machine elements like gears and pulleys as well as the self weight of the shaft.	CO1	L2	1M
	b)	How the shaft is designed when it is subjected to twisting moment only? When the shaft is subjected to a twisting moment (or torque) only, then the diameter of the shaft may be obtained by using the torsion equation. $$\frac{T}{J} = \frac{\tau}{r}$$	CO1	L2	1M
	c)	How are the keys classified? State their applications. As per the applications, different types of keys in machine design are: Rectangular sunk key:- it is used in heavy-duty applications and for preventing rotation of gear and pulley on the shaft. Saddle key:- Use in case of light duty and low power transmission	CO1	L1	1M
	d)	How does the working of a clamp coupling differ from that of a muff coupling? Explain The main difference between clamp coupling and muff coupling are as follows: In muff coupling, torque is transmitted by shear resistance of keys. On the other hand, torque is transmitted partly by means of friction between the sleeve halves and the shaft and partly by shear resistance of key is case of clamp coupling.	CO1	L2	1M
	e)	What are rolling contact bearings? The term rolling contact bearings refers to the wide variety of bearings that use spherical balls or some other type of roller between the stationary and the moving elements. • The most common type of bearing supports a rotating shaft, resisting purely radial loads or a combination of radial and axial (thrust) loads.	CO2	L1	1M
	f)	Write the various types of lubrications. The lubrication methods available for bearings on a machine tool include grease lubrication, oil mist lubrication, air-oil lubrication, and jet lubrication. Each method has unique advantages. Therefore, a lubricating system should be selected that best suits the lubrication requirements.	CO2	L1	1M
	g)	Write the various properties of lubricants. Functions of lubrication : To lubricate each part of the bearing, and to reduce friction and wear To carry away heat generated inside bearing due to friction and other causes To cover rolling contact surface with the proper oil film in order to prolong bearing fatigue life To prevent corrosion and contamination by dirt	CO2	L1	1M
	h)	How does the function of a brake differ from that of a clutch A clutch is a transmission and control device that provides for energy transfer from the driver to the driven shaft. A brake is a transmission and control device that stops a moving load, regulates movement, or holds a load at rest by transforming kinetic energy into heat.	CO3	L2	1M
	i)	What are the advantages of V-belt drive over flat belt drive? Following are the advantages of the V-belt drive over flat belt drive: a) The V-belt drive gives compactness due to the small distance between the centres of pulleys. b) The drive is positive, because the slip between the belt and the pulley groove is negligible.	CO3	L2	1M

	j)	What is meant by a clutch? A clutch is a mechanical device that allows the output shaft to be disconnected from the rotating input shaft. The clutch's input shaft is typically attached to a motor, while the clutch's output shaft is connected to the mechanism that does the work.	CO3	L1	1M
	k)	What is the main function of a flywheel in an engine? A flywheel is a circular-shaped device used in an IC engine that stores the energy produced by the engine (during power stroke) in the form of kinetic energy and provides this energy to the engine during the other three strokes.	CO4	L1	1M
	l)	Discuss the various types of stresses induced in a flywheel rim. The rim is subjected to tensile stress due to P and bending stress due to M. Under the action of centrifugal force, the tendency of the rim is to fly outward which is prevented due to the tensile force P1 acting in each spoke. The spokes of the flywheel are subjected to tensile stress.	CO4	L2	1M
	m)	Define a gear? Gear is a toothed wheel that engages with another toothed wheel or with a rack in order to change the speed or direction of transmitted motion	CO4	L1	1M
	n)	Explain how the gears are classified? Name the various types of gears. Gears are classified into 3 categories; parallel axes gears, intersecting axes gears, and nonparallel and nonintersecting axes gears. Spur gears and helical gears are parallel axes gears. Bevel gears are intersecting axes gears. Screw or crossed helical, worm gear and hypoid gears belong to the third category.	CO4	L2	1M

Unit-I

2	a)	Write the applications of split muff couplings? In split muff coupling, the sleeve or muff is not a single different part instead it is split into 2 Applications of split muff couplings are Split muff couplings are used for medium to heavy duty load with moderate speed. Any automobile with 4 wheel chassis and above.	CO1	L1	4M
	b)	Design a muff coupling to connect two shafts transmitting 40KW at 120rpm. The permissible shear and crushing stress for the shaft and key material (mild steel) are 30Mpa and 80Mpa respectively. The material of muff is cast Iron with permissible shear stress of 15Mpa. Assume that the maximum torque transmitted is 25 percent greater than mean torque.	CO1	L4	10M

2 (b). Power Transmitted $P = 40\,kW$ at ~~1000~~ 120 rpm

$\tau_{allow} = 30\,MPa$ $\sigma_{c\,allow} = 80\,MPa$ (shaft and keys)

$\tau_{allow} = 15\,MPa$ (Muff)

$M_t =$ ~~100~~ 25% extra $(M_t)_{mean}$

mean Torque $M_t = \dfrac{60 \times 10^6 \times Power}{2\pi N}$

$= \dfrac{60 \times 10^6 \times 40}{2\pi \times 120} =$

$M_t = 3{,}184{,}713.31\ N\text{-}mm$

Design Torque $= 1.25 \times (M_t)_{mean}$

$= 3{,}980{,}891.72\ N\text{-}mm$ — 2M

Dia of shaft:

$(\tau_{ind})_{shaft} = \dfrac{16 M_t}{\pi d^3}$

$30 = \dfrac{16 \times 3980891.72}{\pi d^3}$

dia of shaft $d = 87.77\ mm$

$\boxed{d \simeq 88\ mm}$ — 2M

design of sleeve:

outer dia of the sleeve $D = 2d + 13$

$= 2 \times 88 + 13$

$D = 189\ mm$

Length of the sleeve $L = 3.5d$

$= 308\ mm$

Check for shear stress in sleeve.

$\tau_{sleeve} = \dfrac{M_t \cdot r}{J}$ $r = D/2$

$= 94.5\ mm$

$J = \dfrac{\pi(D^4 - d^4)}{32} = \dfrac{\pi(189^4 - 88^4)}{32}$

$J = 119.32 \times 10^6\ mm^4$

$\tau_{sleeve} = \dfrac{3{,}980{,}891.72 \times 94.5}{119.32 \times 10^6}$

$= 3.15\ MPa < 15\ MPa$.

$\therefore$ sleeve is safe — 2M

Design of key:

from data handbook for $d = 88$

key size $b \times h = 25 \times 14$

Length of the key in each shaft

$l = L/2 = \dfrac{308}{2} = 154\ mm$ — 2M

check for stress in key:

shear stress in key $\tau = \dfrac{2M_t}{dbl}$

$= \dfrac{2 \times 3980891.72}{88 \times 25 \times 154}$

$\tau_{ind} = 23.5\ MPa$

$< \tau_{allow}$

Crushing stress in key

$\sigma_{c\,key} = \dfrac{4M_t}{dhl} = \dfrac{4 \times 3980891.72}{88 \times 14 \times 154}$

$= 83.92\ MPa$

$> \sigma_{c\,allow}$

change the key dimension $\boxed{28 \times 16}$

$\sigma_{c\,ind} = 73.43 < \sigma_{c\,allow}$

$\therefore$ key is safe — 2M

(OR)

3	a)	A 10kW power is transmitted at 800 rpm, from a motor shaft, through a key, to a machine shaft by a means of a pulley and a belt. Design the key. Take the allowable shear stress and crushing stress are 45MPa and 100Mpa.	CO1	L4	6M
	b)	A shaft is required to transfer 43kW of power at 600rpm. The outside diameter must not exceed 50mm and the maximum shear stress is not to exceed 70N/mm2. Find out the dimensions of hollow and solid shaft, which would meet these requirements.	CO1	L3	8M

3(a). Power $P = 10$ kW ; Speed $N = 800$ rpm

$\tau_{allow} = 45$ MPa ; $\sigma_{c\,allow} = 100$ MPa

dia of the shaft using shear stress

Torque Transmitted M_t

$$M_t = \frac{60 \times 10^6 \times P}{2\pi N} = \frac{60\times10^6\times10}{2\times\pi\times800}$$

$$M_t = 119426.75 \text{ N-mm}$$

Shear stress $\tau = \frac{16 M_t}{\pi d^3}$

$$45 = \frac{16\times119426.75}{3.14\times d^3}$$

dia of shaft $d = 23.8$ mm

std dia of the shaft

$d = 30$ mm

from data handbook for $d = 30$ mm

std dimension of key are

$\boxed{b \times h = 10 \times 8}$ mm^2 — 4M

~~depth t = 5mm~~

check for the stress in key

$$\tau = \frac{2M_t}{dbl} = \frac{2\times119436.75}{30\times10\times45}$$

$l = 1.5d$
$= 45$

$\tau_{actual} = 17.69 < \tau_{allow}$

check for Crushing stress

$$\sigma_c = \frac{4M_t}{dhl}$$
$$= \frac{4\times119436.75}{30\times8\times45}$$

$\sigma_{c\,ind} = 44.23 < \sigma_{c\,allow}$

$\therefore$ key is safe dimension are

$b\times h\times l = 10\times8\times45$ — 2M
sunk key

3(b) Power = 43 kW Speed $N = 600$ rpm

$D = 50$ mm ; $\tau_{allow} = 70$ MPa.

Torque Transmitted

$$M_t = \frac{60\times10^6\times P}{2\pi N}$$
$$= \frac{60\times10^6\times43}{2\times3.14\times600}$$
$$M_t = 684713.3 \text{ N-mm}$$

Torsional shear stress $\tau = \frac{M_t \cdot r}{J}$.

for solid shaft $r = R = \frac{50}{2} = 25$

$$J = \frac{\pi D^4}{32} = \frac{\pi\times50^4}{32}$$
$$J = 613281.25 \text{ mm}^4$$

$$\therefore \tau = \frac{684713.3\times25}{613281.25} =$$
$$\tau = 27.9 \text{ MPa} < \tau_{allow}.$$

if D is unknown

$$\tau_{allow} = \frac{16 M_t}{\pi D^3}.$$

$$70 = \frac{16\times684713.3}{\pi D^3}$$

solid shaft $D = 36.8 < 50$ mm

$\boxed{D = 38 \text{ mm}}$ — 4M

for hollow shaft

$$\tau_{allow} = \frac{16 M_t}{\pi(D^3 - d^3)}$$

$D = 50$

$$70 = \frac{16\times684713.3}{\pi(50^3 - d^3)}$$

$d = 42.20$ mm

$\boxed{D = 50 ;\ d = 42 \text{ mm}}$ — 4M

Unit-II

4	A shaft, 150mm in diameter, rotates in a bearing at 2000rpm. The length of the bearing is 1.4 times its diameter. The bearing pressure is 1N/mm2, and the coefficient of friction at the bearing surface is 0.005. Determine the power loss in friction. The temperature of the bearing is entirely controlled by the flow of oil through the bearing. If the difference between the outlet and inlet temperatures is 15 ^{0}C, determine the quantity of coolant oil required, if the specific heat of the oil is 1900 J/kg/ ^{o} C.	CO2	L3	14M

4. dia of the shaft $d = 150$ mm; speed $N = 2000$ rpm; length of the bearing $l = 1.4 \times d = 210$ mm

bearing pressure $p = 1\ N/mm^2$ Coeff of friction $\mu = 0.005$ Power loss = ??

$\Delta T = 15°C$ $C_p = 1900\ J/kg/°C$

bearing pressure $p = \frac{W}{l \times d}$

load on the journal $W = p \times l \times d$

$= 1 \times 150 \times 210$

$W = 31500\ N$

Assuming shaft is used for power transmission for bearing pressure of $1\ N/mm^2$

Absolute viscosity $(Z) \doteq 0.060\ kg/m\text{-}s$

$= 0.006\ N/m\text{-}s$

$c/d = 0.001$

Critical pressure $p = \frac{ZN}{4.75 \times 10^6}\left(\frac{d}{c}\right)^2\left(\frac{l}{d+l}\right)$

$p = \frac{0.080 \times 2000}{4.75 \times 10^6}\left[\left(\frac{1}{0.001}\right)^2\left(\frac{210}{150+210}\right)\right]$

$= \cancel{1473\ N/m^2}\ 1.473\ N/mm^2$

$\frac{ZN}{p} = \frac{0.0060 \times 2000}{1} = 12$

heat generated

$Q_g = \mu . W . V$

$= 0.005 \times 31500 \times \frac{\pi \times 150 \times 2000}{60}$ 15.7

$V = \frac{\pi \times 0.15 \times 2000}{60}$
$= 15.7\ m/s$

$Q_g = 2472.75\ W$

heat dissipated

$Q_d = C_p . A . \Delta T$

$= 1900 \times l \times d \times \Delta T$

$= 1900 \times 210 \times 150 \times 15$

$Q_d = 897.75\ W$

Artificial Cooling required

$= $ Heat generated $-$ Heat dis

$= 2472.75 - 897.75$

$Q_t = 1575\ W$

mass of oil required for artificial coo

$Q_t = m\, C_p \times \Delta T$

$1575 = m \times 1900 \times 15$

$m = 0.055\ kg/s$

$\boxed{m = 3.3\ kg/min}$

Power wasted in friction

$P_{loss} = Q_g = 2472.7$

$P_{loss} = 2.472\ kW$

(OR)

5 a) State any four advantages of Rolling contact bearings over sliding contact bearings. CO2 L2 4M

The advantages of rolling contact bearings over sliding contact bearings are as follows.

- Except at very high speeds, little starting and running friction.
- Capacity to absorb transient shock loads.
- Shaft alignment precision.
- Low maintenance costs because no lubrication is required while in use.
- The size is compact.

b) A rolling contact bearing is subjected to the following work cycle: CO2 L4 10M
(a) Radial load of 6000N at 150 r.p.m for 25% of the time;
(b) Radial load of 7500N at 600 r.p.m for 20% of the time; and
(c) Radial load of 2000N at 300 r.p.m for 55% of the time.
The inner ring rotates and loads are steady .Select a bearing for an expected average life of 2500 hours.

5 (b). Rolling Contact bearing (Steady load)
Radial load of 6000N at 150 rpm for 25% of time
Radial load of 7500N at 600 rpm for 20% of the time
Radial load of 2000N at 300 rpm for 55% of the time
inner ring rotates select a bearing for an expected avg life of 2500 hr.

Dynamic equivalent load factor $W = [X \cdot V \cdot W_R + Y \cdot W_A] K_s$.

X - radial load factor = 1
Y - Axial load factor = 1.5
V = rotation factor = 1
K_s = Service factor for steady load = 1

$\therefore W_1 = 6000N;\ W_2 = 7500N;\ W_3 = 2000N$

Life of the bearing in revolution
$L = 60 \cdot N \cdot L_H = 60 \cdot N \times 2500$
$= 15 \times 10^4 \cdot N$ rev. —(2M)

Life of the bearing for $\frac{1}{4}$th (or) of sch
$L_1 = \frac{1}{4} \times 15 \times 10^4 \times N_1 = \frac{1}{4} \times 15 \times 10^4 \times 1500$
$L_1 = 56.25 \times 10^6$ rev

Life of bearing for $\frac{1}{5}$ (20%) of the time
$L_2 = \frac{1}{5} \times 15 \times 10^4 \times N_2 = \frac{1}{5} \times 15 \times 10^4 \times 6000$
$L_2 = 18 \times 10^6$ rev

Life of bearing for $55 \times \frac{1}{100}$ of time
$L_3 = \frac{55}{100} \times 15 \times 10^4 \times N_3 = \frac{55}{100} \times 15 \times 10^4 \times 300$
$L_3 = 24.75 \times 10^6$ rev. —(2M)

equivalent dynamic load
$$W = \left[\frac{L_1 W_1^{3.33} + L_2 W_2^{3.33} + L_3 W_3^{3.33}}{L_1 + L_2 + L_3}\right]^{\frac{1}{k}}$$

k = 3 for ball brg
k = $\frac{10}{3}$ for roller brg

$$W = \left[\frac{56.25 \times 10^6 \times 6000^{3.33} + 18 \times 10^6 \times 7500^{3.33} + 24.75 \times 10^6 \times 2000^{3.33}}{(56.25 + 18 + 24.75) \times 10^6}\right]^{\frac{1}{3.33}}$$

$$= \left[\frac{3.612 \times 10^{20}}{99 \times 10^6}\right]^{\frac{1}{3.33}}$$

$= [3.65 \times 10^{12}]^{\frac{1}{3.33}}$

$W = 5921.8\ N$ —(2M)

Total life in revolution
$L = L_1 + L_2 + L_3 = 99 \times 10^6$ rev

Dynamic load rating $C = W \left[\frac{L}{10^6}\right]^{\frac{1}{k}}$

$C = 5921.8 \left[\frac{99 \times 10^6}{10^6}\right]^{\frac{1}{3.33}}$

$= 23,536.56\ N = 23.53\ kN$ —(2M)

from std data hand book
Bearing 6306 with shaft dia 30 D×B 72×19 mm for 28100N can be selected. —(2M)

Unit-III

6 A V- belt is to transmit 20 kW from a 250 mm pitch diameter sheave to a 900mm diameter pulley. The centre distance between the two shafts is 1000 mm. The groove angle is 40 degree and the coefficient of friction between the belt and the sheave and also between the belt and the pulley is 0.2. The cross-section of the belt is 40 mm wide at the top, 20 mm wide at the bottom and 25 mm deep. The density of the belt is 1000 kg/m3 and the allowable tension per belt is 1000 N. Find the number of belts required. CO3 L3 14M

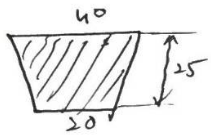
6. V-belt Power $P = 20\,kW$ ~~Speed~~ pully dia $D = 900\,mm$

pitch circle diameter $d = 250\,mm$; Centre distance $C = 1000\,mm$; groove angle $2\alpha = 40°$ $\mu = 0.2$

c/s belt is $w_{top} = 40\,mm$ $w_{bot} = 20\,mm$ height $h = 25\,mm$ density $\rho = 1000\,kg/m^3$;

allowable Tension/belt $= 1000\,N$.

area of V belt $A = \frac{1}{2} h (b_1 + b_2)$



$= \frac{1}{2} 25 (40 + 20)$

$= 750\,mm^2$

$A = 0.75 \times 10^{-3}\,m^2$

mass/m $= \rho \times A \times l$

$= 1000 \times 0.75 \times 10^{-6} \times 1$

$m = 0.75\,kg/m$

angle of Contact $\theta = 180 - 2\sin^{-1}\left(\frac{D-d}{2C}\right)$

$= 180 - 2\sin^{-1}\left[\frac{900-250}{2 \times 1000}\right]$

$= 142.06°$

$\theta = 2.478\,rad$ — (2M)

Tension ratio $\frac{P_1 - mv^2}{P_2 - mv^2} = e^{\mu\theta/\sin\alpha}$

$\frac{P_1 - 0.75v^2}{P_2 - 0.75v^2} = e^{0.2 \times 2.478/\sin 20}$

$= 4.259$

for max belt force $v = 0$

$\frac{P_1}{P_2} = 4.259$

$P_1 = 1000\,N$ (max Tension)

$P_2 = 234.79\,N$ — (2M)

Initial Tension $P_i = \frac{1}{2}(P_1 + P_2)$

$= \frac{1}{2}(1000 + 234.79)$ — (2M)

$P_i = 617.39\,N$

for max Power Transmission velocity $v = \sqrt{\frac{P_i}{3m}}$

$v = \sqrt{\frac{617.39}{3 \times 0.75}}$

$v = 16.56\,m/sec$ — (2M)

$P_1 + P_2 = 2P_i$

$P_1 + P_2 = 2 \times 617.39$

$P_1 + P_2 = 1234.78$ — (1)

$\frac{P_1 - 0.75 \times 16.56^2}{P_2 - 0.75 \times 16.56^2} = 4.259$

$\frac{P_1 - 205.67}{P_2 - 205.67} = 4.259$

$P_1 - 205.67 = 4.259P_2 - 875.97$

$P_1 = 4.259P_2 - 1081.6$ — (2)

Solving eq (1) & (2)

$4.259P_2 - 1081.6 + P_2 = 1234.78$

$5.259P_2 = 2316.38$

Tension on slack side: $\boxed{P_2 = 440.46\,N}$

Tension on Tight side $\boxed{P_1 = 794.32\,N}$ — (2M)

Power Transmitted by single belt

$P/belt = (P_1 - P_2)v$

$= (794.32 - 440.46)\,16.56$

$P/belt = 5.859\,kW$

no of belt required $n = \frac{\text{Total power}}{\text{Power/belt}}$

$= \frac{20}{5.859}$

$n = 3.41$

no of belts required $\boxed{n \simeq 4}$ — (4M)

(OR)

7 a) A multiple disc clutch, steel on bronze, is to transmit 4.5 kW at 750 r.p.m. The inner radius of the contact is 40 mm and outer radius of the contact is 70 mm. The clutch operates in oil with an expected coefficient of 0.1. The average allowable pressure is 0.35 N/mm2. Find : 1. The total number of steel and bronze discs; 2. the actual axial force required and 3. the actual maximum pressure. CO3 L3 10M

7(a). Multi disk clutch Power $P = 4.5\ kW$ speed $N = 750$ rpm:

Inner radius $r = 40$ mm outer radius $R = 70$ mm: Coefficient of friction μ: 0.

pressure $p_{allow} = 0.35\ N/mm^2$

Torque Transmitted $M_t = \frac{60 \times 10^6 \times P}{2\pi N} = \frac{60 \times 10^6 \times 4.5}{2\pi \times 750} = 57324.8$ N.

Torque Transmitted for Multi-Disk clutch.

$$M_t = \frac{\mu P z}{4}(D+d) \quad \text{(for uniform wear)}$$

where P is axial force required

$$P = \frac{\pi p_a \cdot d}{2}(D-d)$$

z – no of pairs of contact

$$= \frac{\pi \times 0.35 \times 80}{2}(140-80)$$

$$\boxed{P = 2637.6\ N} \quad \text{— 4M}$$

$$\therefore M_t = \frac{\mu P z}{4}(D+d) = \frac{0.1 \times 2637.6 \times z}{4}(140+80)$$

$$57324.8 = 14506.8 \cdot z$$

$\therefore$ ~~z~~ no of pairs of friction lines $z = 3.95$
$= 4$

n_1 = no of disk of driving shaft (steel)
n_2 = no of disks of driven shaft (bronze) $n_1 + n_2 = z + 1$

$n_1 + n_2 = 4 + 1$
$n_1 + n_2 = 5$
$3 + 2 = 5$

$\therefore$ no of steel disks $n_1 = 3$
no of bronze disk $n_2 = 2$ — 4M

actual max pressure p_{max} is occurring at $d/D = 0.577$

$$M_t = \frac{\pi \mu \cdot p_{max} \cdot d \cdot z\,(D^2 - d^2)}{8} = \frac{\pi \mu\, p_{max}\, d\, D^2 z\,(1-(d/D)^2)}{8}$$

$$57324.8 = \frac{3.14 \times 0.1 \times p_{max} \times 80 \times 140^2 \times 5\,(1-0.577^2)}{8}$$

$$\boxed{p_{max} = 0.44\ N/mm^2} \quad \text{— 2M}$$

b) Discuss the different types of brakes giving atleast one practical application for each. CO3 L2 4M

Types of Brake:

- Mechanical Brake System
- Hydraulic Brake System
- Pneumatic Brake System
- Electromagnetic Brake System
- Servo Brake System
- Electrical brake system
- Disc Brake System
- Drum Brake System
- Emergency Brake
- Anti-lock Braking system
- Service Brakes or Foot oriented Brakes and
- Hand Brake System

DISC BRAKE

The disk brake may be a device for slowing or stopping the rotation of a wheel while it's in motion. A brake disc is usually made from forged iron, but in some cases, it is often made from composite materials like carbon-carbon or ceramic matrix-reinforced composites. This is connected to the wheel axle.

To stop the wheel, the friction material within the sort of restraint (mounted during a device called a brake caliper) is forced hydraulically, pneumatically, or electromagnetically against both sides of the disc. Friction causes the disc and the connected wheel to slow down or stop. The brakes convert motion into heat, and if they overheat, they subside effectively, a phenomenon referred to as brake fade.

DRUM BRAKES

A drum brake is a brake in which friction is caused by a series of pads or pads pressing against a rotating drum-shaped part called a brake drum. This term generally means a brake in which the shoes press on the inner surface of the drum. When the shoes press on the outside of the drum, it is generally called a buckle brake.

When the drum is squeezed between two shoes, similar to a conventional disc brake, it is sometimes called a "caliper drum brake", although such brakes are relatively rare. A related type of brake uses a flexible strap or "band" that wraps around the outside of a drum, called a band brake.

Band Brakes

Band brakes consist of a flexible band made of steel or other material that wraps around a drum. The band is connected to a lever or pedal, which causes it to tighten around the drum when pressed. The friction between the band and the drum slows down or stops the wheel's rotation. Band brakes are commonly found on bicycles, motorcycles, and some industrial machinery.

Unit-IV

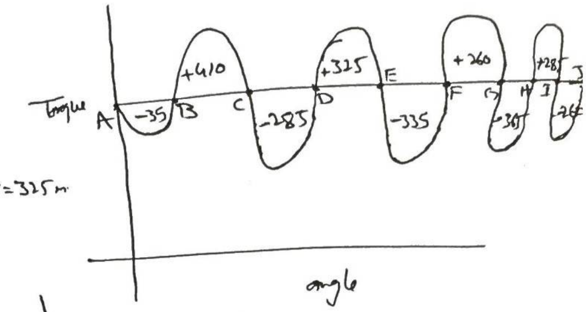
8	The intercepted areas between the output torque curve and the mean resistance line of a turning moment diagram for a multicylinder engine, taken in order from one end are as follows: – 35, + 410, – 285, + 325, – 335, + 260, – 365, + 285, – 260 mm2. The diagram has been drawn to a scale of 1 mm = 70 N-m and 1 mm = 4.5°. The engine speed is 900 r.p.m. and the fluctuation in speed is not to exceed 2% of the mean speed. Find the mass and cross-section of the flywheel rim having 650 mm mean diameter. The density of the material of the flywheel may be taken as 7200 kg / m3. The rim is rectangular with the width 2 times the thickness. Neglect effect of arms, etc.	CO4	L3	14M

8.

Scale
1mm = 70 N-m
1mm = 4.5°

engine speed $N = 900$ rpm
$C_s = 2\% = 0.02$

rim mean dia $D = 650$ mm $= R = 325$ mm
density $\rho = 7200$ kg/m³
rim c/s. $b = 2t$, t



angular velocity $\omega = \frac{2\pi N}{60}$

$= \frac{2\times3.14\times900}{60}$

$\omega = 94.2$ rad/sec —(2M)

Area of Torque diagram given Energy.

1mm² area $= 70 \times 4.5\times\pi/180$
$= 5.495$ N-m —(2M)

Let Energy at A = E
Energy at B = E−35
Energy at C = E−35+410 = E+375
Energy at D = E+375−285 = E+90
E = E+90+325 = E+415
F = E+415−335 = E+80
G = E+80+260 = E+340
H = E+340−365 = E−25
I = E−25+285 = E+260
J = E+260−260 = E

max Energy $E_{max} = E+415$
min Energy $E_{min} = E-35$

max fluctuation of Energy $\Delta E = E_{max}-E_{min}$
$= E+415-(E-35)$
$\Delta E = 450$ mm² (4M)

in Term of N-m
$\Delta E = 450 \times 5.495$
$\Delta E = 2472.75$ N-m

max fluctuation of Energy

$\Delta E = mR^2\omega^2 \cdot C_s$

$2472.75 = m \times 0.325^2 \times 94.2^2 \times 0.02$

mass of flywheel $\boxed{m = 131.91\ kg}$ (2M)

Let Rim c/s. $b \times t$
Area $= b\times t = 2t \times t = 2t^2$ mm²

mass = Volume × density
$m = 2\pi R \times A \times \rho$
$131.91 = 2\pi \times 0.325 \times 2t^2 \times 7200$
$t = 0.067$ m $= 67$ mm
$\therefore b = 2\times t = 134$ mm

Thickness of the Rim $t = 67$ mm
width of the Rim $b = 134$ mm
(4M)

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(OR)

9	A pair of straight teeth spur gears, having 20° involute full depth teeth is to transmit 12 kW at 300 r.p.m. of the pinion. The speed ratio is 3: 1. The allowable static stresses for gear of cast iron and pinion of steel are 60 MPa and 105 MPa respectively. Assume the following: Number of teeth of pinion = 16; Face width = 14 times module; Determine the module, face width and pitch diameter of gears. Check the gears for wear; given σes = 600 MPa; E_P = 200 kN/mm2 and E_G = 100 kN/mm2.	CO4	L4	14M

9. 20° involute power Transmitted $P = 12$ kW ; speed $N = 300$ r/m

speed ratio $V.R = 3:1 = \frac{N_P}{N_G} = \frac{3}{1}$; $\sigma_{o_G} = 60$ MPa ; $\sigma_{o_P} = 105$ MPa.

$N_P = 16$; face width $b = 14m$; $\sigma_{es} = 600$ MPa ; $E_P = 200$ kN/mm² $= 200\times10^3$ N/mm² ; $E_G = 100$ kN/mm² $= 100\times10^3$

let m = module in mm

D_P = Pitch circle dia of pinion in mm

pitch line velocity $v = \frac{\pi D_P N_P}{60}$

$= \frac{\pi \times m T_P N}{60}$

$= \frac{\pi \times m \times 16 \times 300}{60} = 251m$ m/s

$v = 0.251m$ m/s (2M)

Assume steady load condition 8-10 hrs of service per day. The service factor (C_s) for this $C_s = 1$

Tangential tooth load

$W_T = \frac{P}{v} \times C_s = \frac{12\times10^3}{0.251m} \times 1$

$W_T = \frac{47.8\times10^3}{m}$ N (2M)

velocity factor $C_v = \frac{4.5}{4.5+v} = \frac{4.5}{4.5+0.251m}$

Tooth form factor for pinion

$y_P = 0.154 - \frac{0.912}{N_P}$

$= 0.154 - \frac{0.912}{16} = 0.097$

for gear $y_G = 0.154 - \frac{0.912}{N_G}$

$= 0.154 - \frac{0.912}{3\times16} = 0.135$

$\sigma_{bp} \times y_P = 105 \times 0.097 = 10.185$

$\sigma_{bG} \cdot y_G = 60 \times 0.135 = 8.1$

$\sigma_{bp} y_P > \sigma_{bG} y_G$ (Gear is weak). (2M)

Tangential load for gear

$W_T = \sigma_{bG} \cdot b \pi \cdot m y_G$

$= \sigma_{oG} C_v \, b \pi m \cdot y_G$

$\frac{47.8\times10^3}{m} = 60 \times \left[\frac{4.5}{4.5+0.251m}\right] 14m \times \pi m \times 0.135$

$\frac{47.8\times10^3}{m} = \frac{1603.4m^2}{4.5+0.251m}$

$4.5 + 0.251m = 0.0335m^3$

solving this eq by trial and error

$m = 5.6$ say 6 mm (4M)

face width $b = 14m = 84$ mm

pitch dia of pinion $D_P = m \cdot N_P = 6\times16 = 96$ mm

pitch dia of gear $D_G = m \cdot N_G = 6 \times 48 = 288$ mm

Checking the gears for wear

$Q = \frac{2 \times V.R}{V.R+1} = \frac{2\times3}{3+1} = 1.5$

load stress factor $K = \frac{\sigma_{es}^2 \sin\phi}{1.4}\left[\frac{1}{E_P} + \frac{1}{E_G}\right]$

$= \frac{600^2 \sin 20}{1.4}\left[\frac{1}{200\times10^3} + \frac{1}{100\times10^3}\right]$

$= 0.44 + 0.88 = 1.32$ N/mm²

max (or) limiting load for wear

$W_w = D_P \cdot b \cdot Q \cdot K$

$= 96 \times 84 \times 1.5 \times 1.32$

$= 15967$ N (2M)

Tangential load on the tooth

$W_T = \frac{47.8\times10^3}{m}$

$= \frac{47.8\times10^3}{6} = 7967$ N

$W_w >> W_T$ ∴ design is satisfactory (2M)

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