(15)

00041

The Open University of Sri Lanka
Faculty of Engineering Technology

Study Programme	:Bachelor of Technology Honours in Engineering
Name of the Examination	: Final Examination
Course Code and Title	:Vehicle Dynamics and Design of Automotive Components MEX6232/ DMX6532
Academic Year	: 2017/18
Date	: 13th February 2019
Time	:.09:30- 12:30
Duration	: 3 hours

Instructions

This question paper consists of seven (07) questions. You are required to answer only five (05) questions. All questions carry equal marks

---

Question 01

A frequent failure of the cylinder head of a four stroke four-cylinder diesel engine with a bore diameter of 122mm, stroke of 183 mm and cylinder head thickness of 10mm prompted an engineer to investigate the theoretical strength of the cylinder head. He tested the engine on an engine dynamometer and revealed that the engine developing maximum of 80 kW at 3500 r.p.m. He subsequently obtained the Mechanical efficiency through a Morse test, which is 80 %. He tested a specimen taken from an identical cylinder head and revealed that the allowable stress of the material as 60 N/mm^2.
Engine design guidelines stipulate that the design maximum design pressure has to be 15 times the Indicated Men Effective Pressure.

I. Determine the indicated mean effective pressure of the engine.
II. Assess theoretically whether the thickness of the cylinder head is adequate or not.

Question 02

A car with curb weight of 1200kg has been designed to carry a 500 kg load including the driver. The un-laden (when not loaded) ground clearance is 190 mm with requiring to maintain a minimum 160 mm under fully laden (loaded with 500kg load) condition. In order to maintain good ride-comfort, the damped natural frequency has to be maintained in the range of 70 cycles per minute to 120 cycles per minute. To provide good adhesion between road and tyres the damping ratio has to be 0.7. Determine the appropriate stiffness of the spring and the damping coefficient for the damper.

Question 03

A vehicle of mass m with the wheel track a and center of gravity located at height h from ground level is travelling on a banked track with radius of r as shown in figure Q3.

a) Draw the free body diagram for the vehicle.
b) Starting from fundamentals derive an expression to express the overturning speed in terms of r, a, h, and θ.

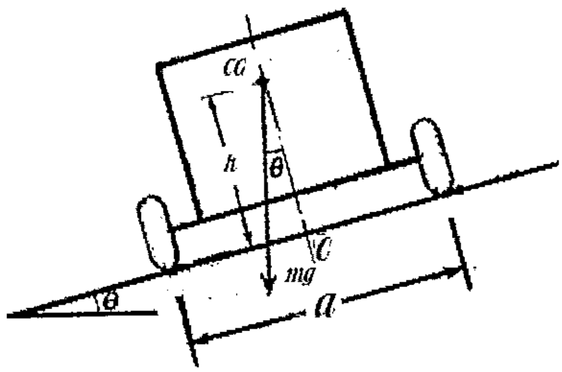

Figure Q3

Question 04

Starting from fundamentals show that the inertia force (F_i) produced by the reciprocating parts of a crank mechanism is given by

$$F_i = mr\omega^2\left[\cos\theta + \left(\frac{r}{L}\right)\cos 2\theta\right]$$

where m - mass of reciprocating parts
r - crank radius
L - Length of the connecting rod
ω - angular speed of the crank shaft
θ - angular displacement of the crank shaft from the outer dead centre position.

The following data are available for a vertical single cylinder engine and its crank mechanism.

Length of the connecting rod	-	800 mm
Stroke length	-	340 mm
Cylinder bore diameter	-	160 mm
Effective mass of reciprocating parts	-	120 kg
Speed of the engine	-	2800 rev/min
Maximum gas pressure on the piston	-	1.3 MPa

Determine the effective turning moment on the crankshaft when the crank shaft has turned through an angle of 45° from the top dead centre position. Neglect the effect of gravitational forces of the reciprocating parts.

Question 05

Show that the maximum torque (T) transmitted through a plate clutch having n pairs of contacting surfaces under uniform pressure conditions is given by

$$T = \frac{2}{3}\mu n P\left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right]$$

where μ - coefficient of friction between the plate surfaces
P - axial force
r_1 - outer radius of the plate
r_2 - inner radius of the plate

A single plate clutch with one pair of contact surface transmits 75 kW from an engine running at 1500 rev/min. Assuming that the pressure distribution is limited to 0.6 N/mm^2 and that the outer radius of the clutch plate is three times bigger than the inner radius, determine the outer and inner radii of the clutch plate. Assume that the coefficient of friction between the contact surfaces is 0.25.

Question 06

The following information is available for a truck and its motion.

Mass of the truck	- 2000 kg
Maximum gradient which the truck will have to negotiate	- 10 degrees
Maximum Velocity at the maximum grade	- 40 km/h
Gear engaged while driving at the maximum gradient	- 1st gear
Coefficient of rolling resistance	- 0.016
Air resistance in Newton	- 1/2 $\rho C d A V^2$

Where,	
ρ - Density of air	- 1.293 kg/m^3
Cd - Aerodynamic drag coefficient	- 0.33
A - Total projected frontal area	- 3 m^2
V - Velocity of the vehicle in m/s	
Transmission efficiency in the first gear	- 70%
Effective diameter of tyres	- 850 mm
Gear reduction ratio in the differential drive unit	- 4

Determine the following:

a) The minimum power available at the engine speed of 2400 rpm on first gear.
b) The maximum speed of the vehicle in the top gear on a level road at the engine speed of 2000 rpm. Assume the transmission efficiency on top gear as 90% .
c) The gear ratio on the top gear for the condition given in (b) above.

Question 07

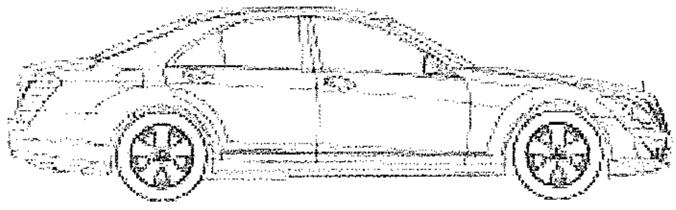

Figure Q7

a) A sketch of a typical sedan car is shown in figureQ7. Sketch air streamlines assuming. that the vehicle is moving in forward direction.
b) In the above sketch show pressure distribution over and under the body structure of the automobile.
c) Briefly explain boundary layer with respect to aerodynamics.
d) By means of a sketch,explain wake in fluid dynamics and how the wake can occur with an automobile.

THE OPEN UNIVERSITY OF SRI LANKA
MEX6232/MEX6332 - VEHICLE DYNAMICS AND DESIGN OF AUTOMOTIVE COMPONENTS
Data Sheet

1. Wear Tooth Load for gears

$W_w = D_p . b. Q. K$

where W_w = Maximum or limiting load for wear in newtons.
D_p = Pitch circle diameter of the pinion in mm,
b = Face width of the pinion in mm,
Q = Ratio factor

$$= \frac{2 \times V.R}{V.R + 1} = \frac{2T_G}{T_G + T_P}, \text{ for external gears}$$

$$= \frac{2 \times V.R}{V.R - 1} = \frac{2T_G}{T_G + T_P}, \text{ for internal gears.}$$

V.R. = Velocity ratio = T_G / T_p,
K = Load-stress factor (also known as material combination factor) in N/mm^2.

2. The thickness of a cylinder wall

The thickness of a cylinder wall (t) is usually obtained by using a thin cylindrical formula, i.e.

$$t = \frac{p \times D}{2\sigma_c} + C$$

where
p = Maximum pressure inside the cylinder in N/mm^2,
D = Inside diameter of the cylinder or cylinder bore in mm,
σ_c = Permissible circumferential or hoop stress for the cylinder material in MPa or N/mm^2.
C = Allowance for re-boring.

The allowance for re-boring (C) depending upon the cylinder bore (D) for I. C. engines is given in the following table:

D (mm)	75	100	150	200	250	300	350	400	450	500
C (mm)	1.5	2.4	4.0	6.3	8.0	9.5	11.0	12.5	12.5	12.5

3. Design of Cylinder Head

The cylinder head may be approximately taken as a flat circular plate whose thickness (t_h) may be determined from the following relation:

$$t_h = D\sqrt{\frac{C.p}{\sigma_c}}$$

where

D – Cylinder bore in mm,
p - Maximum pressure inside the cylinder in N/mm^2,
σ_c - Allowable circumferential stress in MPa or N/mm^2. It may be taken as 30 to 50 MPa,
C - Constant whose value is taken as 0.1.

4. Design of the Piston Crown

The thickness of the piston head for strength
The thickness of the piston head (t_H), according to Grashoff'sformula is given by

$$t_H = \sqrt{\frac{3p.D^2}{16\sigma_t}}\text{(in mm)}$$

Where,

p = Maximum gas pressure or explosion pressure in N/mm^2,
D = Cylinder bore or outside diameter of the piston in mm, and
σ_t = Permissible tensile stress for the material of the piston in MPa or N/mm^2

Design of piston head for heat transfer
On the basis of second consideration of heat transfer

$$t_H = \frac{H}{12.56k(T_C - T_E)}\text{(in mm)}$$

5. Design Against fatigue

Gerber method

$$\frac{1}{F.S} = \left(\frac{\sigma_m}{\sigma_u}\right)^2 F.S. + \frac{\sigma_v \times K_f}{\sigma_e}$$

The Goodman criteria

$$\frac{1}{F.S} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_v \times K_f}{\sigma_e}$$

The Soderberg criteria

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v \times K_f}{\sigma_e}$$

where F.S. = Factor of safety,
σ_m = Mean stress.
σ_u = Ultimate stress,
σ_v = Variable stress.
σ_e = Endurance limit for reversed loading,
K_f = Fatigue stress concentration faction

6. Vibration

$$\omega_d = \omega_n \sqrt{1-\zeta^2}$$

$$\omega_n = \sqrt{\frac{k}{m}}, \quad \zeta = \frac{c}{2m\omega_n}$$

$$c_c = \sqrt{4Mk}$$

$$x(t) = Ae^{-\zeta\omega_n t} \sin(\omega_d t + \phi_d)$$

$$\zeta = \frac{C}{C_c} = \frac{C}{\sqrt[2]{km}}$$

ζ - damping ratio

- End -