Use Rankine formula for which the numerator constant may be taken as 320 N/ mm^2 and the denominator constant 1 / 7500.

Q.5 i. Explain self locking in brakes. **2**

ii. A band and block brake having 12 blocks, each of which subtends an angle of 16°at the centre, is applied to a rotating drum with a diameter of 600 mm. the blocks are 75 mm thick. The drum and the flywheel mounted on the same shaft have a mass of 1800 Kg and have a combined radius of gyration of 600 mm. the two ends of band are attached to pins on the opposite sides of the brake fulcrum at distances of 40 mm and 150 mm from it. If a force of 250 N is applied on the lever at a distance of 900 mm from the fulcrum, find the **8**
(a) Maximum braking torque
(b) Angular retardation of the drum
(c) Time taken by the system to be stationary from the rated speed of 300 rpm.

OR iii. Explain the working of internal expanding brake with neat sketch. **8**

Q.6 i. What are various causes of clutch plate failure? **2**

ii. A centrifugal clutch is to be designed to transmit 15 KW at 900 r.p.m. The shoes are four in number. The speed at which the engagement begins is ¾th of the running speed. The inside radius of the pulley rim is 150 mm. Te shoes are lined with ferrodo for which the coefficient of friction may be taken as 0.25. Determine: **8**
(a) Mass of the shoes, and (b) Size of the shoes.

OR iii. A plate clutch has three discs on the driving shaft and two discs on the driven shaft. The outer and inner diameter of contact surfaces are 240 mm and 120 mm respectively. Assuming uniform pressure and coeff of friction =0.3, find the total spring load pressing the plates together to transmit 25 KW at 1575 rpm. **8**
If there are 6 springs each of stiffness 13N/mm and each of contact surfaces has worn away by 1.25 mm, find the maximum power that can be transmitted, assuming uniform wear

******

Total No. of Questions: 6 *Total No. of Printed Pages:4*

Enrollment No......................................

Faculty of Engineering
End Sem (Odd) Examination Dec-2018
AU3CO13/ME3CO12 Machine Design-II

Programme: B.Tech. Branch/Specialisation: AU/ME

Duration: 3 Hrs. **Maximum Marks: 60**

Note: (i) All questions are compulsory. Internal choices, if any, are indicated. Answers of Q.1 (MCQs) should be written in full instead of only a, b, c or d.
(ii) Use of Design data book is permitted.
(iii) Assume suitable data, if required, giving reason.

Q.1 i. Which of the following are functions of bearings? **1**
(a) Ensure free rotation of shaft with minimum friction
(b) Holding shaft in a correct position
(c) Transmit the force of the shaft to the frame
(d) All of these

ii. A_______ bearing supports the load acting along the axis of the shaft. **1**
(a) Thrust (b) Radial (c) Longitudinal (d) Transversal

iii. The gears used to connect non-parallel and non-intersecting shafts is **1**
(a) Straight bevel gears (b) Spiral bevel gears
(c) Spiral gears (d) Double helical gears

iv. Bevel gears used for connecting intersecting shafts at 90^0 and having speed ratio 1:1 is known as **1**
(a) Bevel gears (b) Beveloid gears
(c) Mitre gears (d) None of these

v. A petrol engine has compression ratio....... **1**
(a) 6 to 10 (b) 10 to 15 (c) 15 to 25 (d) 25 to 40

vi. Supercharging......the power developed by the engine **1**
(a) Has no effect on (b) Increases
(c) Decreases (d) None of these

vii. The following is not a drum brake **1**
(a) External contracting brake (b) Internal expanding brake
(c) Disc brake (d) All of these

viii. The mechanical brakes are operated by means of **1**
(a) Levers (b) Bell cranks (c) Cams (d) All of these

ix. In Disc clutch, the clutch disc acts as a **1**
(a) Driving member (b) Driven member
(c) Neutral member (d) Any of these

x. Clutch and friction linings are ___ to the clutch plate **1**
(a) Riveted (b) Welded (c) Welted (d) Any of these

Q.2 i. Explain utility of McKee Curve. **2**

ii. Design a journal bearing for a centrifugal pump from the following data: **8**
Load of journal = 20 KN,
Speed of journal = 900 rpm,
Absolute viscosity of oil at 55 ºC =0.017 kg/ms,
Ambient temp = 15.5 ºC
Maximum bearing pressure for the pump =1.5 MPa
Calculate also the mass of lubricating oil required for artificial cooling, if rise in temp of oil be limited to 10 ºC. Take heat dissipation coefficient as 1232 W/m^2 ºC.

OR iii. A 306 radial ball bearing with inner ring rotation has a 10 sec work cycle as follows: **8**

	For 2 sec	For 8 sec
Radial load	4 KN	3 KN
Axial load	2 KN	Zero
Speed	900 rpm	1200 rpm
Nature of load	Light shock load (K_s =1.5)	Steady load (K_s =1)

If the basic dynamic capacity of bearing is 24.25 KN. Determine expected average life of bearing.
Take - Radial load factor = 0.56, Axial load factor = 1.43

Q.3 i. What are various ways in which gear fails? **2**

ii. Design spur gear drive with the following data: **8**
Maximum power to be transmitted = 22.5 KW,
Velocity ratio =2:1
Speed of pinion = 200 r.p.m.
Distance between centres of gear = 600 mm

Tooth profile = 20º stub involute.
Static stress for both gear = 60 MPa
Deformation factor or Dynamic factor = 80
Material combination factor for wear = 1.4
Also check the design for dynamic and wear load.

OR iii. A helical cast steel gear with 30° helix angle has to transmit 35 KW at 1500 r.p.m. If the gear has 24 teeth, determine the necessary module, pitch diameter and face width for 20° full depth teeth. The static stress for cast steel may be taken as 56 MPa. The width of face may be taken as 3 times the normal pitch. The tooth factor for 20° full depth involute gear may be taken as 0.154 - (0.912 /T_E) ,Where T_E represents the equivalent no. of teeth. **8**

Q.4 i. Define Piston slap. **2**

ii. Design a cast iron piston for a single acting 4-S engine for the following data : **8**
Cylinder bore = 100 mm
Stroke = 125 mm
Maximum gas pressure = 5 N / mm^2
Indicated mean effective pressure Pm = 0.75 N / mm^2
Mechanical efficiency = 80 %
Speed = 2000 r.p.m
Fuel consumption = 0.15 kg / stroke power per hour
HCV = 42 X 10^3 KJ / kg
Any other data required for the design may be assumed.

OR iii. Design a connecting rod for an I.C. engine running at 1800 r.p.m. and developing a maximum pressure of 3.15 N / mm^2. The diameter of the piston is 100 mm; mass of the reciprocating parts per cylinder 2.25 kg; length of connecting rod 380 mm; stroke of piston 190 mm and compression ratio 6: 1. Take a factor of safety of 6 for the design. Take length to diameter ratio for big end bearing as 1.3 and small end bearing as 2 and the corresponding bearing pressures as 10 N / mm^2 and 15 N / mm^2. The density of material of the rod may be taken as 8000 kg / m^3 and the allowable stress in the bolts as 60 N / mm^2 and in cap as 80 N / mm^2. The rod is to be of I- section for which you can choose your own properties. **8**

SCHEME & SOLUTION OF MACHINE DESIGN II
(AU3CO13 / ME3CO12)

Q.1 i) (d) ; vi) (b)

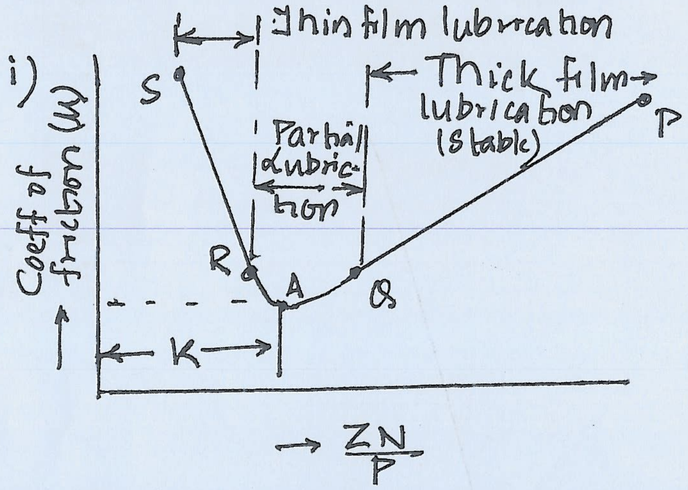
ii) (a) ; vii) (c)

iii) (c) ; viii) (d)

iv) (c) ; ix) (b)

v) (a) ; x) (a)

Q.2 i) (1)



Explaination of Fig. (1)

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ii) $p = \frac{W}{ld} = 1.25$ (Assume: d = 100 mm & l/d = 1.6) 1 ~~(2)~~

$\frac{ZN}{P} = \frac{0.017 \times 900}{1.25} = 12.24$ - - - - - - - (1)

$3K = \frac{ZN}{P}$ or, $K = \frac{ZN}{3p} = 4.33$ - - - - (1)

(K → Bearing Modulus)

As $\frac{ZN}{P} > K$, Design is safe

From table, for pumps, $c/d = 0.0013$ -- (1)

Now, $\mu = \frac{33}{10^8}\left[\frac{ZN}{P}\right]\left[\frac{d}{c}\right] + k$ (K = 0.002)

$= 0.0031 + 0.002 = 0.0051$ -- (1)

Heat generated $Q_g = \mu W V = \mu W\left(\frac{\pi D N}{60}\right)$
$= 480.7 W$ — (1)

Heat dissipated $Q_d = CA(t_b - t_a) = C.L.d(t_b - t_a)$

As $(t_b - t_a) = \frac{1}{2}(t_o - t_a) = \frac{1}{2}(55 - 15.5) = 19.75°$

$\therefore Q_d = 389.3 W$ — (1) — (1)

Mass of Lub. Oil for cooling $= m = (Q_g - Q_d)/S\Delta t = (91.4)/(1900)(10) = 0.0048$ kg/s

where, $S \to$ Sp.ht. of Oil $= 1900$ J/kg°c

Q. 2 (iii) From Data book Mahadevan (P. 388), we get $C_0 = 14220N$ for 6306 bearing
(Static load Capacity)

$X \to$ Radial force factor $= 0.56$ } Given
$Y \to$ Axial force factor $= 1.43$ }

For the first part of 2 secs

finding $P_1 = (XF_r + YF_a)K_s$

$= [0.56(4) + 1.43(2)]1.5$

$= \cancel{7.65} \; 7.65 KN$

$\therefore N_1 = \frac{2}{60} \times 900 = 30$ rotations — (2)

Similarly,

for the second part of 8 secs

finding $P_2 = 0.56(3) + 1.43(0)$

$= 1.68 KN$

$\therefore N_2 = \frac{8}{60} \times 1200 = 160$ rotations — (2)

$\therefore n = N_1 + N_2 = 190$ rotations

$\therefore$ 190 rotations completed in 10 sec. cycle

$\therefore$ 1140 rotations will be completed in 1 min.

$\therefore P_c = \sqrt[3]{\frac{N_1 P_1^3 + N_2 P_2^3}{(N_1 + N_2)}} = \sqrt[3]{\frac{30(7.65)^3 + 160(1.63)^3}{190}}$

$$= \sqrt[3]{\cancel{\frac{3979.53 + 692.91}{190}}} \quad \sqrt[3]{\frac{13430.9 + 692.91}{190}}$$

$$= \sqrt[3]{\cancel{\frac{4672.4}{190}}} = \sqrt[3]{\cancel{24.6}} = \cancel{2.90\,kN} = 4.2\,KN \quad \text{--- (2)}$$

$$= \cancel{2900\,N}.$$

$$\therefore L_{10} = \left[\frac{C}{P_c}\right]^3 = \left[\frac{24.25}{4.2\cancel{00}}\right]^3 = \cancel{58471} \quad \text{--- (1)}$$

$$= (5.77)^3 = 192.48 \text{ million rev.}$$

$$\therefore (L_{10})_h = \frac{L_{10} \times 10^6}{60n} = \frac{192 \times 10^6}{60 \times 190} = 16877 \text{ hrs.} \quad \text{--- (1)}$$

Q3 i) Various ways of gear touth failure are-

a) Abrasive wear b) Corrosion wear -- (0.5x4)

c) Initial Pitting d) Destructive Pilting

~~e) Scouring~~ Description of each in 2-3 lines

ii) Assume width of gear touth 'b' = 10 m

By Centre Distance formula,

$$C = D_P/2 + D_a/2 = \frac{D_P}{2} + \frac{2D_P}{2}$$

$$= 1.5 D_P$$

[Here Pinion is half the dia. of Gear as given in numerical V.R - = 2:1 & pinion is the driver. & it will be designed as it is weaker]

$$\therefore D_P = 400\,mm, \quad D_a = 800\,mm$$

$$V = 4.2\,m/s$$

As $V < 12\,m/s$, $C_V = \frac{3}{3+V} = 0.417.$ —— (1)

$$\therefore T_P = 400/m$$

$\therefore$ form factor for pinion $Y_P = 0.175 - 0.0021m$ --(1)

Tangential load $W_T = \frac{P}{v} \times C_s = \frac{22500}{4.2} \times 1$ P-4

$= 5357 N$

By Lewis equation

$5357 = \sigma_{wp} b.\pi.m.y_p = (\sigma_{op} \times C_v) b.\pi.m y_p$

$= (60 \times 0.417) 10m \times \pi m (0.175 - 0.0021m)$

$= 0.137.6m^2 - 1.65 m^3$

$\therefore m = 6.65$ say $m = 8$ mm — (2)

$\therefore b = 10m = 80$ mm

$T_P = 50$ — (1)

$T_G = 100$

Dynamic load.

$$W_D = W_T + \frac{21v(bC + W_T)}{21V + \sqrt{bC + W_T}}$$

$= 10630$ N — (1)

$\therefore y_p = 0.175 - 0.0021m = 0.1582$

Taking $\sigma_e = 84 N/mm^2$ from table for C.I.

Endurance ~~load~~ strength 'Ws' $= \sigma_e . b . \pi . m . y_p = 26722 N$ — (1)

$\therefore$ Ratio factor $Q = \frac{2VR}{VR+1} = \frac{2 \times 2}{2+1} = 1.33$

Wear load $= W_w = D_p . b Q . K = 400 \times 80 \times 1.33 \times 1.4$

$= 59584$ N — (1)

As both 'W_s' & 'W_w' are greater than W_D, design is safe

Q 3 iii) Torque 'T' = 223×10^3 N.mm --- (1)

$T_E = \frac{T_G}{\cos^3\alpha} = 37$

$y' = 0.154 - \frac{0.912}{T_E} = 0.129$ -- (1)

$\therefore W_T = \frac{T}{D_G/2} = \frac{18600}{m}$

$v = \pi D_G . N/60 = 1.88 m (m/s)$

$\therefore C_v = \frac{15}{15+v} = \frac{15}{(15+1.88m)}$ --- (1)

$\therefore W_T = (\sigma_0 \times C_v) b . \pi . m y' = \frac{2780 m^2}{15+1.885m}$

By Trial & Error, $= \frac{18600}{m}$

$m = 5.5$ say 6 --- (2)

$\therefore D_G = 144$ mm --- (1)

$b = 3 P_N = 3 P_c \cos\alpha = 3 \pi m \cos\alpha$

$= 50$ mm --- (1)

End thrust or axial load on the gear

$W_A = W_T \tan\alpha = 1790$ N --- (1)

Q.4 i) Due to piston clearance more than required piston movement is not smooth. (2)

ii) Piston Head thickness by Strength Criteria

$t_H = \sqrt{\frac{3PD^2}{16\sigma_t}} = \sqrt{\frac{3 \times 5 (100)^2}{16(38)}} = 15.7 \simeq 16$

Assume $\sigma_t = 38$ MPa for C.I.

--- (1)

Piston Head by Heat Dissipation Criteria - P-C

$n = N/2$ for 4S engine
$= 1000$

$\therefore A = \frac{\pi}{4}(D)^2 = 7855\ mm^2$

$\therefore$ I.P. $= \frac{P_{mL} \cdot A N}{60} = 12270\ W$ — ①

$\therefore$ B.P. $= 0.8 \times 12.27 = 9.8\ kW$

Heat through Piston Head.

$H = C \times HCV \times m \times B.P$ where $C = 5\% = 0.05$

$= 860\ W$

$\therefore t_H = \frac{H}{12.56k(T_C - T_E)} =$ (Take $k = 46.6\ W/m/°C$, $T_C - T_E = 220°C$)

$\therefore t_H = 0.0067\ m = 6.7\ mm$ — ①

Cup Dia. $= 0.7D = 70\ mm$ as $L/D = 1.25$

2 Ribs are 4 in no., thickness of ribs $t_R = \frac{16}{3}$ to $\frac{16}{2}$
$= 5.33$ to 8
$\therefore t_R = 7\ mm.$ — ①

3 Piston Rings

Thickness $t_1 = D\sqrt{\frac{3P_W}{\sigma_t}} = 100\sqrt{\frac{3 \times 0.035}{90}}$
$= 3.4\ mm.$ — ①

Axial thickness $= t_2 = 3\ mm$

4) Piston Barrel.

Piston Wall thickness $t_4 = 3.4\ mm$ — ①

5) Piston Pin : d_0 = Outer dia. of pin
l_1 = length of pin in bush of small end

Load on Pin $= 25 \times d_0 \times 0.45 \times 100 = 1125 d_0\ N$

~~Lo~~ Max. load on Piston = 39275 N

$= \frac{\pi}{4}(D)^2 \times p$

$\therefore d_o = 35\,mm$ — (1)

$d_i = 0.6 \times 35 = 21\,mm$

Checking the pin in bending.

$M = \frac{P.D}{8} = 491 \times 10^3$ N-mm

$\therefore \sigma_b = 134\ N/mm^2$ — (1)

As Induced < permissible, design is safe
(140 N/mm²)

Q4 iii) -- Refer Photocopy or Printout

Q.5 i) Diagram of brake — (1)

Equation of braking force — (1)

ii) Refer Printout

iii) Diagram of Brake — (4)

Working of brake — (4)

Q6 i) 4 points — (4×0.5)

Q6.ii) i) Mass of Shoes

Running speed = ω = 94.26 rad/s

Speed at engagement = $\frac{3}{4}(94.26) = 70.7$ rad/s — (1)

Centrifugal at each shoe = $P_c = m\omega^2 r$

$= 1066\,m$ — (1)

Inward force ~~by~~ on each shoe by each spring, centrifugal force at engagement speed ω_1. $P_s = m(\omega_1)^2 r = 600\,m$ — (1)

$T = \frac{P \times 60}{2\pi N} = 159$ N-m.

Also

$T = \mu(P_c - P_s) R n = 70\,m$ — (1)

$\therefore m = 159/70 = 2.27$ kg

2. Size of Shoes

l = Contact length

b = Width of the shoes

$l = \theta . R = \frac{\pi}{3} \times 150 = 157 mm$ — (1)

$A = l . b = 157 b \; mm^2$

Assume 'p' exerted on shoe = $0.1 N/mm^2$, so force with which shoe presses against rim

$A . p = 157 b \times 0.1 = 15.7 b N$ -- (1)

Force with which shoe presses against rim

$= P_c - P_s = 1066m - 600m = 466m$

$= 1058 N$ --- (1)

$\therefore b = 1058 / 15.7 = 67.4 mm$ — — (1)

Q6 iii) Torque by clutch =

$n_1 = 3, n_2 = 2, n_s = 4$

$= \frac{P \times 60}{2 \pi N} = 151.5 \; \cancel{N} \; N.m.$

—— (1)

Mean Radius of contact surface for uniform forces.

$R = \frac{2}{3} \left[\frac{(r_1)^3 - (r_2)^3}{(r_1)^2 - (r_2)^2} \right] = 93.3 mm$ — (1)

Also. $T = n \mu W R$, we get,

$W = 1353 N$ —— (2)

Max. Power transmitted

No. of Spring = 6

Wear on each surface = 1.25 mm.

$\therefore$ Total wear = $8 \times 1.25 = 10 mm$ --- (1)

Reduction in Spring force = Total Wear × Stiffness/Spring × No. of Spring.

$= 0.01 \times 13 \times 10^3 \times 6 = 780 N$

$\therefore$ New load $= 1353 - 780 = 573 N$ --- (1)

$\therefore$ Mean radius $R = \frac{r_1 + r_2}{2} = 90 mm$ & $T' = n \mu W R = 62 N.m$ — (1)

$\therefore$ Power transmitted = 10227 W — (1)

(Q.5 ii) Given $2\theta = 16°$,

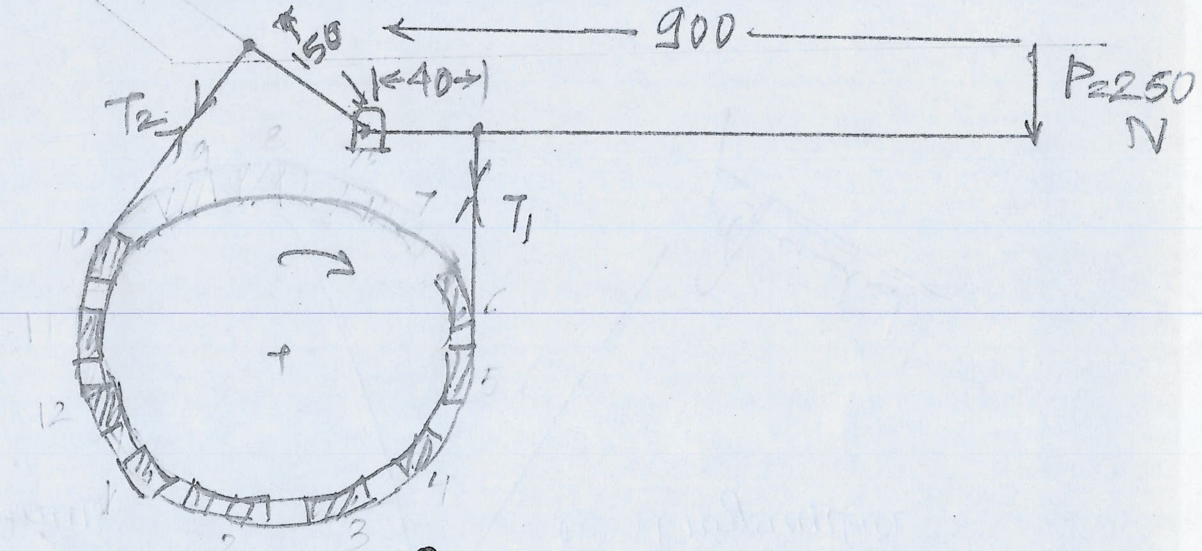
$n = 12$, $d_b = 600mm$; $r_b = 300$

k_g = radius of gyration of drum & flywheel = 600 mm

m_b = 1800kg, Mass of drum

$\mu = 0.3$ (~~Not~~ Assumed, not given numerical)

$t = 75$ mm = shoe thickness



$$\frac{T_1}{T_2} = \left[\frac{1+\mu\tan\theta}{1-\mu\tan\theta}\right]^n = \left[\frac{1+(0.3)\tan 6}{1-(0.3)\tan 6}\right]^{12}$$

$$= \left[\frac{1+0.3(0.105)}{1-0.3(0.105)}\right]^{12} = \left[\frac{1.0315}{0.9685}\right]^{12} = [1.065]^{12}$$

$$= 2.13 \quad \text{—— (1)}$$

Also, $T_1 \times 40 + P \times 900 = T_2 \times 150$

or, $40T_1 + 225000 = 150T_2$

or, $150T_2 - 40T_1 = 225000$

Now, $150T_2 - 40(2.13T_2) = 225000$

$\therefore (150 - 85.2)T_2 = 225000$

or $T_2 = 3472.2$ N.

$\therefore T_1 = 7395.8$ N —— (1)

i) Braking torque $= (T_1 - T_2)\, r_b' = (3923.6).(300 + \frac{t}{2})$

$= 3923.6.(300+37.5) = 1324215$ N.mm

$= 1324.2$ N.m —— (2)

b) Ang. retardation of drum

By eq. of motion for rotation

ω_1 = Initial speed =

$$\omega_2 = \omega_1 + \alpha t \quad ; \quad \omega_2 = \text{Final Speed} = 0$$

We know B. Torque = T = I. α

$$I = m_b k_g^2 = 1800 \times (0.6)^2 = 648 \text{ kg.m}^2$$

$\therefore$ Ang. retard. 'α' $= \dfrac{1324.2}{648} = 2.04 \text{ rad/s}^2$ --- (2)

(c) Time taken to rest

Now, $\omega_1 = \dfrac{2\pi(300)}{60} = 10\pi = 31.4 \text{ rad/s}$

$\omega_2 = 0$

$\therefore$ $t = \dfrac{-31.4}{2.04} = 15.4$ secs. --- (2)

(c)

—x—

Q. 4 (vii)

Q. 4 (iii) Design a connecting rod for an I.C. engine running at 1800 r.p.m. and developing a maximum pressure of 3.15 N/mm2. The diameter of the piston is 100 mm ; mass of the reciprocating parts per cylinder 2.25 kg; length of connecting rod 380 mm; stroke of piston 190 mm and compression ratio 6 : 1. Take a factor of safety of 6 for the design. Take length to diameter ratio for bearing as 1.3 and small end bearing as 2 and the corresponding bearing pressures as 10 N/mm2 and 15 N/mm^2. The density of material of the rod may be taken as 8000 kg/m3 and the allowable stress in the bolts as 60 N/mm^2 and in cap as 80 N/mm^2. The rod is to be of I-section for which you can choose your own proportions. . Use Rankine formula for which the numerator constant may be taken as 320 N/mm2 and the denominator constant 1 / 7500.

* We know that the maximum bending moment for a simply or freely supported beam with a uniformly distributed load of F_1 over a length x between the supports (In this case, x is the distance between the cap bolt centres) is $\frac{F_1 \times x}{8}$. When the load F_1 is assumed to act at the centre of the freely supported beam, then the maximum bending moment is $\frac{F_1 \times x}{4}$. Thus the maximum bending moment in between these two bending moments $\left(i.e. \frac{F_1 \times x}{8} \text{ and } \frac{F_1 \times x}{4}\right)$ is $\frac{F_1 \times x}{6}$.

In actual practice, I_{xx} is kept slightly less than 4 I_{yy}. It is usually taken between 3 and 3.5 and the connecting rod is designed for buckling about X-axis.

Now, for the section as shown in Fig. 32.14 (a), area of the section,

$$A = 2(4t \times t) + 3t \times t = 11t^2$$

$$I_{xx} = \frac{1}{12}\left[4t(5t)^3 - 3t \times (3t)^3\right] = \frac{419}{12}t^4$$

and

$$I_{yy} = 2 \times \frac{1}{12} \times t(4t)^3 + \frac{1}{12} \times 3t \times t^3 = \frac{131}{12}t^4$$

$$\therefore \quad \frac{I_{xx}}{I_{yy}} = \frac{419}{12} \times \frac{12}{131} = 3.2$$

(1)

Solution. Given : $N = 1800$ r.p.m. ; $p = 3.15$ N/mm^2 ; $D = 100$ mm ; $m_R = 2.25$ kg ; $l = 380$ mm $= 0.38$ m ; Stroke = 190 mm ; *Compression ratio = 6 : 1 ; F. S. = 6.

The connecting rod is designed as discussed below :

1. *Dimension of I- section of the connecting rod*

Let us consider an I-section of the connecting rod, as shown in Fig. 32.14 (a), with the following proportions :

Flange and web thickness of the section = t

Width of the section, $B = 4t$

and depth or height of the section,

$$H = 5t$$

First of all, let us find whether the section chosen is satisfactory or not.

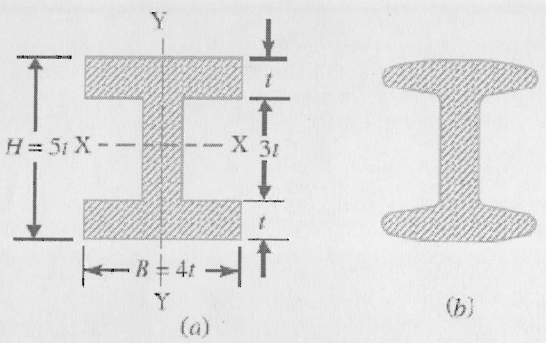


Fig. 32.14

Now let us find the dimensions of this *I*-section. Since the connecting rod is designed by taking the force on the connecting rod (F_C) equal to the maximum force on the piston (F_L) due to gas pressure, therefore,

$$F_C = F_L = \frac{\pi D^2}{4} \times p = \frac{\pi (100)^2}{4} \times 3.15 = 24\ 740 \text{ N}$$

1

We know that the connecting rod is designed for buckling about *X*-axis (*i.e.* in the plane of motion of the connecting rod) assuming both ends hinged. Since a factor of safety is given as 6, therefore the buckling load,

$$W_B = F_C \times F.S. = 24\ 740 \times 6 = 148\ 440 \text{ N}$$

We know that radius of gyration of the section about *X*-axis,

$$k_{xx} = \sqrt{\frac{I_{xx}}{A}} = \sqrt{\frac{419 t^4}{12} \times \frac{1}{11 t^2}} = 1.78\ t$$

Length of crank,

$$r = \frac{\text{Stroke of piston}}{2} = \frac{190}{2} = 95 \text{ mm}$$

Length of the connecting rod,

$$l = 380 \text{ mm} \qquad \text{...(Given)}$$

∴ Equivalent length of the connecting rod for both ends hinged,

$$L = l = 380 \text{ mm}$$

Now according to Rankine's formula, we know that buckling load (W_B),

$$148\ 440 = \frac{\sigma_c . A}{1 + a\left(\frac{L}{k_{xx}}\right)^2} = \frac{320 \times 11\ t^2}{1 + \frac{1}{7500}\left(\frac{380}{1.78\ t}\right)^2}$$

... (It is given that σ_c = 320 MPa or N/mm² and $a = 1 / 7500$)

$$\frac{148\ 440}{320} = \frac{11\ t^2}{1 + \frac{6.1}{t^2}} = \frac{11\ t^4}{t^2 + 6.1}$$

$$464\ (t^2 + 6.1) = 11\ t^4$$

or $$t^4 - 42.2\ t^2 - 257.3 = 0$$

∴ $$t^2 = \frac{42.2 \pm \sqrt{(42.2)^2 + 4 \times 257.3}}{2} = \frac{42.2 \pm 53}{2} = 47.6$$

... (Taking +ve sign)

or $$t = 6.9 \text{ say } 7 \text{ mm}$$

2

Thus, the dimensions of *I*-section of the connecting rod are :

Thickness of flange and web of the section

$$= t = 7 \text{ mm } \textbf{Ans.}$$

Width of the section, $B = 4\ t = 4 \times 7 = 28$ mm **Ans.**

and depth or height of the section,

$$H = 5\ t = 5 \times 7 = 35 \text{ mm } \textbf{Ans.}$$

These dimensions are at the middle of the connecting rod. The width (B) is kept constant throughout the length of the rod, but the depth (H) varies. The depth near the big end or crank end is kept as $1.1H$ to $1.25H$ and the depth near the small end or piston end or piston end is kept as $0.75H$ to $0.9H$. Let us take

Depth near the big end,

$$H_1 = 1.2H = 1.2 \times 35 = 42 \text{ mm}$$

and depth near the small end,

$$H_2 = 0.85H = 0.85 \times 35 = 29.75 \text{ say } 30 \text{ mm}$$

$\therefore$ Dimensions of the section near the big end

$$= 42 \text{ mm} \times 28 \text{ mm } \textbf{Ans.}$$

and dimensions of the section near the small end

$$= 30 \text{ mm} \times 28 \text{ mm } \textbf{Ans.}$$

1

Since the connecting rod is manufactured by forging, therefore the sharp corners of I-section are rounded off, as shown in Fig. 32.14 (b), for easy removal of the section from the dies.

2. *Dimensions of the crankpin or the big end bearing and piston pin or small end bearing*

Let d_c = Diameter of the crankpin or big end bearing,

l_c = length of the crankpin or big end bearing $= 1.3\, d_c$ ...(Given)

p_{bc} = Bearing pressure = 10 N/mm^2 ...(Given)

We know that load on the crankpin or big end bearing

$$= \text{Projected area} \times \text{Bearing pressure}$$

$$= d_c . l_c \cdot p_{bc} = d_c \times 1.3\, d_c \times 10 = 13\, (d_c)^2$$

Since the crankpin or the big end bearing is designed for the maximum gas force (F_L), therefore, equating the load on the crankpin or big end bearing to the maximum gas force, *i.e.*

$$13\, (d_c)^2 = F_L = 24\,740 \text{ N}$$

$\therefore$ $(d_c)^2 = 24\,740 / 13 = 1903$ or $d_c = 43.6$ say 44 mm **Ans.**

and $l_c = 1.3\, d_c = 1.3 \times 44 = 57.2$ say 58 mm **Ans.**

The big end has removable precision bearing shells of brass or bronze or steel with a thin lining (1mm or less) of bearing metal such as babbit.

Again, let d_p = Diameter of the piston pin or small end bearing,

l_p = Length of the piston pin or small end bearing $= 2d_p$ ...(Given)

p_{bp} = Bearing pressure = 15 N/mm^2 ..(Given)

We know that the load on the piston pin or small end bearing

$$= \text{Project area} \times \text{Bearing pressure}$$

$$= d_p \cdot l_p \cdot p_{bp} = d_p \times 2\, d_p \times 15 = 30\, (d_p)^2$$

Since the piston pin or the small end bearing is designed for the maximum gas force (F_L), therefore, equating the load on the piston pin or the small end bearing to the maximum gas force, 1

i.e.

$$30\, (d_p)^2 = 24\,740 \text{ N}$$

$\therefore$ $(d_p)^2 = 24\,740 / 30 = 825$ or $d_p = 28.7$ say 29 mm **Ans.**

and $l_p = 2\, d_p = 2 \times 29 = 58$ mm **Ans.**

The small end bearing is usually a phosphor bronze bush of about 3 mm thickness.

3. Size of bolts for securing the big end cap

Let d_{cb} = Core diameter of the bolts,

σ_t = Allowable tensile stress for the material of the bolts

$= 60\ \text{N/mm}^2$ ...(Given)

and n_b = Number of bolts. Generally two bolts are used.

We know that force on the bolts

$$= \frac{\pi}{4}(d_{cb})^2\,\sigma_t \times n_b = \frac{\pi}{4}(d_{cb})^2\ 60 \times 2 = 94.26\ (d_{cb})^2$$

The bolts and the big end cap are subjected to tensile force which corresponds to the inertia force of the reciprocating parts at the top dead centre on the exhaust stroke. We know that inertia force of the reciprocating parts,

1

$$F_I = m_R \cdot \omega^2 \cdot r\left(\cos\theta + \frac{\cos 2\theta}{l/r}\right)$$

We also know that at top dead centre on the exhaust stroke, $\theta = 0$.

$$\therefore \quad F_I = m_R \cdot \omega^2 \cdot r\left(1 + \frac{r}{l}\right) = 2.25\left(\frac{2\pi \times 1800}{60}\right)^2 0.095\left(1 + \frac{0.095}{0.38}\right)\text{N}$$

$$= 9490\ \text{N}$$

Equating the inertia force to the force on the bolts, we have

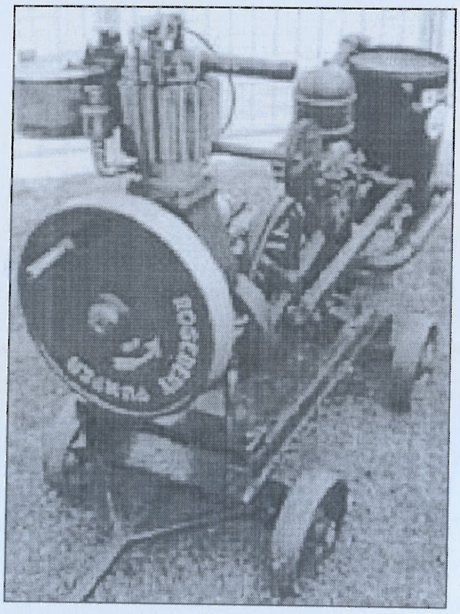
$$9490 = 94.26\ (d_{cb})^2 \quad \text{or} \quad (d_{cb})^2 = 9490 / 94.26 = 100.7$$

$$\therefore \quad d_{cb} = 10.03\ \text{mm}$$

and nominal diameter of the bolt,

$$d_b = \frac{d_{cb}}{0.84} = \frac{10.03}{0.84} = 11.94$$

say 12 mm **Ans.**



4. Thickness of the big end cap

Let t_c = Thickness of the big end cap,

b_c = Width of the big end cap. It is taken equal to the length of the crankpin or big end bearing (l_c)

= 58 mm (calculated above)

σ_b = Allowable bending stress for the material of the cap

$= 80\ \text{N/mm}^2$ ...(Given)

The big end cap is designed as a beam freely supported at the cap bolt centres and loaded by the inertia force at the top dead centre on the exhaust stroke (*i.e.* F_I when $\theta = 0$). Since the load is assumed to act in between the uniformly distributed load and the centrally concentrated load, therefore, maximum bending moment is taken as

$$M_C = \frac{F_I \times x}{6}$$

where x = Distance between the bolt centres

= Dia. of crank pin or big end bearing + 2 × Thickness of bearing liner + Nominal dia. of bolt + Clearance

$= (d_c + 2 \times 3 + d_b + 3)$ mm $= 44 + 6 + 12 + 3 = 65$ mm

∴ Maximum bending moment acting on the cap,

$$M_C = \frac{F_I \times x}{6} = \frac{9490 \times 65}{6} = 102\ 810 \text{ N-mm}$$

Section modulus for the cap

$$Z_C = \frac{b_c (t_c)^2}{6} = \frac{58 (t_c)^2}{6} = 9.7\ (t_c)^2$$

We know that bending stress (σ_b),

$$80 = \frac{M_C}{Z_C} = \frac{102\ 810}{9.7\ (t_c)^2} = \frac{10\ 600}{(t_c)^2}$$

∴ $(t_c)^2 = 10\ 600 / 80 = 132.5$ or $t_c = 11.5$ mm **Ans.**

Let us now check the design for the induced bending stress due to inertia bending forces on the connecting rod (*i.e.* whipping stress).

We know that mass of the connecting rod per metre length,

$$m_1 = \text{Volume} \times \text{density} = \text{Area} \times \text{length} \times \text{density}$$

$$= A \times l \times \rho = 11t^2 \times l \times \rho \qquad \text{...}(\because A = 11t^2)$$

$$= 11(0.007)^2\ (0.38)\ 8000 = 1.64 \text{ kg}$$

...[∵ $\rho = 8\ 000$ kg/m^3 (given)]

∴ Maximum bending moment,

$$M_{max} = m\ .\ \omega^2\ .\ r \times \frac{l}{9\sqrt{3}} = m_1\ .\ \omega^2\ .\ r \times \frac{l^2}{9\sqrt{3}} \qquad \text{...}(\because m = m_1 \cdot l)$$

$$= 1.64 \left(\frac{2\pi \times 1800}{60}\right)^2 (0.095)\ \frac{(0.38)^2}{9\sqrt{3}} = 51.3 \text{ N-m}$$

$$= 51\ 300 \text{ N-mm}$$

and section modulus,

$$Z_{xx} = \frac{I_{xx}}{5t/2} = \frac{419\ t^4}{12} \times \frac{2}{5\ t} = 13.97\ t^3 = 13.97 \times 7^3 = 4792 \text{ mm}^3$$

∴ Maximum bending stress (induced) due to inertia bending forces or whipping stress,

$$\sigma_{b(max)} = \frac{M_{max}}{Z_{xx}} = \frac{51\ 300}{4792} = 10.7 \text{ N/mm}^2$$

1

Since the maximum bending stress induced is less than the allowable bending stress of 80 N/mm^2, therefore the design is safe.