UNIT - II

ENGINE PARTS

* PISTON
* CONNECTING ROD
* CYLINDER
* CRANK SHAFT

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UNIT-2

Engine Parts

* Design a cast iron piston for a single acting 4 stroke engine for the following data.

Bore = 100mm, stroke = 125mm, Max gas pressure = 5 N/mm². Indicated mean effective pressure is 0.75 N/mm² and Mechanical efficiency is 80%. Fuel consumption 0.15 kg/break power/hr. Higher-calorific value is 42×10^3 kJ/kg. speed 2000 rpm. Any other data required for the design maybe assumed.

Sol: We need to design a cast iron piston.

Given Data:

Diameter of piston [D] = 100mm

Length of stroke [L] = 125mm

Maximum gas pressure [Max P] = 5 N/mm²

Mean effective pressure = 0.75 N/mm²

Mechanical Efficiency [η_m] = 80% = 0.8

calorific value [C_v] = 42×10^3 kJ/kg

speed of crank [N] = 2000 rpm

Mass of fuel consumed [m] = 0.15 kg/BP/hr

⇒ As, fuel consumed is in hours, we need convert that into seconds,

So, $\frac{0.15}{3600} = 4.16 \times 10^{-5}$ kg/BP/sec

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Step I: Thickness of piston head based on the strength of piston material.

$$t_1 = \sqrt{\frac{3 P_m D^2}{16 \sigma_{tp}}} \text{ mm} \longrightarrow \text{(eq.15.13) (Data Book)}$$

$$= \sqrt{\frac{3 \times 5 \times (100)^2}{16 \times 40}}$$

$$= 15.30 \text{ mm}$$

$$\simeq 16\text{mm (Approximately)}$$

Step II: Thickness of piston head based on heat dissipation

$$t_1 = \frac{1000 H}{12.56 K (T_c - T_e)} \longrightarrow \text{(eq.15.14) (Data Book)}$$

$$\Rightarrow H = 0.05 m \times C_v \times P_I \text{ kW}$$

$$\Rightarrow P_I = \frac{PLAn}{60 \times 10^6} = \frac{0.7 \times \pi/4 \, D^2 \times 125 \times 1000}{60 \times 10^6}$$

$$= 11.450 \text{ kW}$$

$$\Rightarrow P_B = P_I \times \eta_m = 11.45 \times 0.8 = 9.16 \text{ kW}$$

$$H = 0.05 \times 4.16 \times 10^{-5} \times 42 \times 10^3 = 9.16$$

$$= 0.80 \text{ kW}$$

Now, $$t_1 = \frac{1000 H}{12.56 K (T_c - T_e)}$$

$$= \frac{1000 \times 0.80}{12.56 \times 46.6 \times 10^{-3} \times 220} = 6.21 \text{ mm}$$

$$\simeq 7\text{mm [Approx]}$$

[∵ Here, $K = 46.6 \times 10^{-3}$ kW/m/°C for CI
$(T_c - T_e) = 220°C$ for CI]

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Now, we should take the piston head with more thickness

As we compare both thicknesses, thickness of piston head based on the strength of piston material is higher than thickness based on heat dissipation.

$\therefore$ Thickness of piston head $= [t_1] = 16mm$.

Step III : Thickness of Rib $[t_2] = (0.3 \text{ to } 0.5)\, t_1 \rightarrow$ (eq. 15.15) (Data Book)

$= 0.4 \times t_1$

$= 0.4 \times 16$

$= 6.4mm$

$\simeq 7mm$ (Approx)

Step IV : Radial thickness of piston ring

$$[t_3] = D\sqrt{\frac{3P_c}{\sigma_{bn}}} \rightarrow \text{(eq. 15.16) (Data Book)}$$

$$= 100\sqrt{\frac{3 \times 0.25}{90}}$$

$= 2.8\,mm$

$\simeq 3mm$ (Approx)

Step V : Axial thickness of piston ring

$$[t_4] = (0.7 - 1)\, t_3 \rightarrow \text{(eq. 15.17) (Data Book)}$$

$= 0.8 \times 3$

$= 2.4\,mm$

$\simeq 3mm$ (Approx)

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step VI : Radial depth of ring-groove

$b = t_3 + 0.4\,mm \longrightarrow$ (eq. 15.18) (Data Book)

$= 3 + 0.4$

$= 3.4\,mm$

$\simeq 4\,mm$ (Approx)

step VII : Thickness of barrel nearer to piston head

$t_5 = 0.03D + b + 4.5\,mm \longrightarrow$ (eq. 15.19) (Data Book)

$= 0.03(100) + 4 + 4.5$

$= 11.5$

$\simeq 12\,mm$ (Approx)

step VIII : Thickness of barrel at the open end of the piston

$t_6 = (0.25 \text{ to } 0.35)\, t_5 \longrightarrow$ (eq. 15.20) (Data Book)

$= 0.3 \times 12 = 3.45$

$\simeq 4\,mm$ (Approx)

step IX : Length of skirt $L_s = \dfrac{\pi H D P_m}{4 P_s} \longrightarrow$ (eq. 15.21)

$= \dfrac{\pi \times 0.1 \times 100 \times 5}{4 \times 0.5} = 78.53$

$\simeq 79\,mm$

step X : Length of ring section $L_H = i t_4 + (i-1) x_2 \longrightarrow$ (eq. 15.22)

$= 4(3) + (4-1)3$

$= 21\,mm$

$\left[\because \begin{array}{l} i = 4 \\ x_2 \le t_4 \\ x_2 = 3 \end{array}\right]$ from table (15.3)

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Step XI : Length of piston $L_p = L_s + L_H + L_t \longrightarrow$ (eq. 15.23)

[where $L_t = x_1 = t_1$]

$= 79 + 21 + 16$

$= 116\,mm$

Step XII : Diameter of piston pin $d = \dfrac{F_g}{P_b \times l} \longrightarrow$ (eq. 15.25)

$\Rightarrow F_g = \frac{\pi}{4} D^2 \cdot P_m$, $l = 1.5d$, $P_b = 25\,N/mm^2$

$= \frac{\pi}{4} (100)^2 \times 5$

$= 39269.90\,N$

$$d = \frac{F_g}{P_b \times l} = \frac{39269.90}{25 \times 1.5d} = \sqrt{\frac{39269.90}{25 \times 1.5}}$$

$= 32.36\,mm$

Step XIII : Induced bending stress in piston pin

$$\sigma_b = \frac{32M}{\pi d_1^3} < (\sigma_b) \longrightarrow \text{(eq. 15.26)}$$

$$M = \frac{F_g \cdot D}{8} = \frac{39269.90 \times 100}{8} = 490873.75$$

$M = 490873.75$

$$\sigma_b = \frac{32M}{\pi d_1^3} = \frac{32 \times 490873.75}{\pi (32.36)^3} = 139.13\,N/mm^2$$

Induced bending stress in piston pin $\sigma_b = 139.13\,N/mm^2 < (\sigma_b)$

$\therefore$ our model is safe.

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* Design a cast Iron trunk type piston for a single acting 4 stroke engine developing 75 kW / cylinder when running at 600 rpm. The other available data is maximum gas pressure 4.8 N/mm². Indicated Mean effective pressure 0.65 N/mm². Mechanical efficiency 95%. Radius of trunk 110mm. Fuel consumption 0.3 kg/bp/hr. calorific value of fuel is 44×10^3 kJ/kg. Difference of temperature at the centre & edges of the piston head 200°c. Allowable stress for the material of the piston 33.5 Mpa. Allowable stress for the material of piston rings & Gudgeon pin 80 Mpa. Allowable bearing pressure on the piston barrel 0.4 N/mm². Allowable bearing pressure on the gudgeon pin 17 N/mm

Solution: Given Data:

Mass of fuel used [m] = 0.3 kg/bp/hr

We need to convert into seconds

$$m = \frac{0.3}{3600} = 8.33 \times 10^{-5} \text{ kg/bp/sec}$$

Speed of crank [N] = 600 rpm

Maximum gas pressure [P_m] = 4.8 N/mm²

Effective pressure = 0.65 N/mm²

Radius of crank [r] = 110mm

Calorific value of fuel [C] = 44×10^3 kJ/kg

Step (i):

$$P_I = \frac{PLAn}{60 \times 10^6} \text{ kW} \longrightarrow (\text{eq. } 15.14)$$

Stroke Length [L] = 2 × radius

= 2 × 110 = 220 mm

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$$\Rightarrow 75 = \frac{0.65 \times 220 \times A \times 3600}{60 \times 10^6}$$

$$\Rightarrow 75 = \frac{0.65 \times 220 \times \frac{\pi}{4} D^2 \times 3600}{60 \times 10^6} \qquad \because A = \frac{\pi}{4} D^2$$

$$D = \sqrt{\frac{75 \times 60 \times 10^6}{0.65 \times 220 \times 330 \times \pi/4}}$$

$D = 365.45$

$D \simeq 366$ mm (Approx)

step iii: Thickness of piston head based on the strength of piston material

$$t_1 = \sqrt{\frac{3 P_m D^2}{16 \sigma_{tp}}} \text{ mm} \longrightarrow (\text{eq. } 15.13)$$

$$= \sqrt{\frac{3 \times 4.8 \times (366)^2}{16 \times 35}} = 54.9 \text{ mm}$$

$t_1 \simeq 55$ mm (Approx)

step iv: Thickness of piston head based on heat dissipation

$$t_1 = \frac{1000 H}{12.56 K (T_c - T_e)} \text{ mm} \longrightarrow (\text{eq. } 15.14)$$

$H = 0.05 \, m \times C_r \times P_B$ KW

$= 0.05 \times 8.34 \times 10^{-5} \times 44 \times 10^3 \times 71.25$

$= 13.05$ KW

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$$t_1 = \frac{1000\,H}{12.56\,K\,(T_c - T_e)} \text{ mm}$$

$$= \frac{1000 \times 13.05}{12.56 \times 46.6 \times 10^{-3} \times 220}$$

$$= 101.34$$

$t_1 \simeq 102$ mm (Approx)

thickness of piston ring based on heat dissipation greater than thickness of piston head based on strength of piston material.

$\therefore \boxed{t_1 = 102\text{mm}}$

step v: Thickness of rib $t_2 = (0.3 - 0.5)\,t_1 \longrightarrow$ (eq. 15.15)

$t_2 = 0.4 \times 102$

$= 40.8$mm

$\boxed{t_2 \simeq 41\text{mm}}$

step vi: Radial thickness of piston rings

$$t_3 = D\sqrt{\frac{3P_c}{\sigma_{bm}}} \longrightarrow \text{(eq. 15.16)}$$

$$= 366\sqrt{\frac{3(0.025)}{90}}$$

$\boxed{t_3 \simeq 11\text{mm}}$

step vii: Axial thickness of piston rings $t_4 = (0.7 - 1)\,t_3 \longrightarrow$ (eq. 15.17)

$t_4 = 1 \times 11 = 11$mm

$\boxed{t_4 = 11\text{mm}}$

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step viii : Thickness of barrel nearer to piston head

$$t_5 = 0.03D + b + 4.5mm \longrightarrow (eq.\ 15.19)$$

$$b = t_3 + 0.4mm \longrightarrow (eq.\ 15.18)$$

$$b = 11 + 0.4mm$$

$$= 11.4$$

$$b \simeq 12mm \quad \text{Approx}$$

$$t_5 = 0.03\,(366) + 12 + 4.5$$

$$\boxed{t_5 \simeq 28mm}$$

step ix : Thickness of barrel at the open end of the piston

$$t_6 = (0.25 \text{ to } 0.35)\, t_5 \longrightarrow (eq.\ 15.20)$$

$$= 0.25 \times 28$$

$$\boxed{t_6 = 7mm}$$

step x : Length of skirt $L_s = \dfrac{\pi M D P_m}{4 P_s} \longrightarrow (eq.\ 15.21)$

$$L_s = \frac{\pi \times 0.03 \times 366 \times 4.8}{4 \times 0.28}$$

$$\boxed{L_s = 275.95\ mm}$$

step xi : Length of ring section $L_n = i t_4 + (i-1)\, x_2 \longrightarrow (eq.\ 15.22)$

$$= 9\,(11) + (9-1)\,1 \qquad x_2 \le t_4$$

$$= 99mm$$

$$L_n = 99mm$$

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step xiii: Length of piston $L_p = L_s + L_m + L_t \longrightarrow$ (eq. 15.23)

$= 215 + 102 + 99$ where $L_t = x_1 = t_1$

$= 477mm$

$$\boxed{L_p = 477mm}$$

step xiii: Diameter of piston pin $d = \frac{F_g}{P_b \times l} \longrightarrow$ (eq. 15.25)

where $F_g = \frac{\pi}{4} D^2 \cdot P_m$

$= 505002.22$

$$\boxed{d = 140.7 \simeq 141mm}$$

Induced bearing stress in piston pin

$$\sigma_b = \frac{32M}{\pi d_1^3} < (\sigma_b) \longrightarrow \text{(eq. 15.26)}$$

where $M = \frac{F_g \cdot D}{8} = 23103851.57$

$$\sigma_b = \frac{32(23103851.57)}{\pi (141)^3}$$

$$\boxed{\sigma_b = 83.95 \ N/mm^2} < (\sigma_b)$$

$\therefore$ our design is safe

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* Design a connecting rod for an IC engine running at 1800 rpm & developing a maximum pressure of 3.15 N/mm². The diameter of the piston is 100mm. Mass of reciprocating parts for cylinder 2.25 kg. Length of connecting rod 380mm, stroke of piston 190mm. Take a factor of safety is 6 for design. Take length of diameter ratio for big end bearing as 1.3 & small end bearing as 2, & the corresponding bearing pressures as 10 N/mm² & 15 N/mm². the density of material of the rod maybe taken as 8000 kg/m³ & the allowable stress in the bolt as 60 N/mm² & in cap 80 N/mm². The rod is to be I section for which you can choose your own proportions. Use Rankine formula for which the numerator constant maybe taken as 320 N/mm² & denominator constant $\frac{1}{7500}$ N/mm².

Solution: We need to design a connecting rod

Given Data:

Diameter of piston [D] = 100mm

Speed of IC engine [n] = 1800 rpm

Weight of reciprocating parts [W_r] = 2.25 kg

Factor of safety [fs] = 6

length of diameter of bigend $\left[\frac{l_1}{d_1}\right]$ = 1.3

Bearing pressure of big end [P_{b_1}] = 10 N/mm²

length to diameter of small end $\left[\frac{l_2}{d_2}\right]$ = 2

Bearing pressure of small end [P_{b_2}] = 15 N/mm²

Maximum pressure = 3.15 N/mm²

stroke Length [L_s] = 190mm

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Allowable tensile stress of piston rod $[\sigma_t] = 60\ N/mm^2$

Allowable bending stress of cap of big end $[\sigma_{bc}] = 80\ N/mm^2$

Density of connecting rod material $[\rho] = 8000\ kg/m^3$

Allowable compressive stress $[\sigma_c] = 320\ N/mm^2$

Actual length of connecting rod $[L] = 380mm$

<u>step 1:</u> Load due to steam or gas pressure

$$F_g = \frac{\pi}{4} D^2 p \longrightarrow (eq.16.1)$$

$$= \frac{\pi}{4}(100)^2 \times 3.15$$

$$\boxed{F_g = 24740.04\ N}$$

<u>step 2:</u> Design load $F_d = F_g \times f_g \longrightarrow (eq.16.8)$

$$= 24740.04 \times 6$$

$$\boxed{F_d = 148440.25\ N}$$

<u>step 3:</u> Crippling load for connecting rod by using Rankins formula

$$F_{cr} = \frac{\sigma_c \cdot A}{1 + a\left(\frac{L}{K}\right)^2} \geq F_d \longrightarrow (eq.16.4)$$

$$F_{cr} = F_d = 148440.25$$

$\sigma_c = 320\ N/mm^2$ (Given)

$A = 11t^2$ (from fig 16.3)

$a = \frac{1}{7500}$ (given)

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$L = l = 380mm$ (As both ends are hinged)

$K = 1.78t$

$$\Rightarrow 148440.25 = \frac{320 \times 11t^2}{1 + \frac{1}{7500}\left(\frac{380}{1.78t}\right)^2}$$

$$\Rightarrow 148440.25 = \frac{3520t^2}{1 + 1.3 \times 10^{-4}\left(\frac{4557.04}{t^2}\right)}$$

$$\Rightarrow 148440.25 = \frac{3520t^2}{1 + \frac{5.94}{t^2}}$$

$$\Rightarrow 148440.25 = \frac{3520t^2}{t^2 + 5.94}$$

$$(t^2 + 5.94)\,148440.25 = 3520t^2$$

$$3520t^4 - 148440.25t^2 = 881735.02$$

$$t^4 - 42.17t^2 - 250.49 = 0$$

$$t = 6.89$$

$$\boxed{t = 7mm}$$

Width $= 4t$ (from fig 16.3)

$= 4(7) = 28mm$

Length $= 5t$ (from fig 16.3)

$= 5(7)$

$= 35mm$

<u>Dimensions of big end & small end:</u>

<u>Step 4</u>: Diameter of piston pin $= d_1 = \frac{F_g}{P_{b_1} \times l_1}$ $\longrightarrow$ (eq. 16.12)

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$$= \frac{24740.04}{10 \times 1.3 d_1}$$

$$d_1 = \sqrt{\frac{24740.04}{13}}$$

$$\boxed{d_1 \simeq 44mm}$$

step5: Length of piston pin $l_1 = (1.3 \text{ to } 2)\, d_1 \longrightarrow$ (eq. 16.14)

$$l_1 = 1.3 \times 44 = 56.6 \text{ mm}$$

$$\boxed{l_1 = 57mm}$$

step6: Diameter of crank pin $d_2 = \frac{F_g}{P_{b_2} \times l_2} \longrightarrow$ (eq. 16.13)

$$d_2 = \frac{24740.02}{15 \times 2 d_2}$$

$$d_2 = \sqrt{\frac{24740.02}{15 \times 2}}$$

$$\boxed{d_2 = 29mm}$$

step7: Length of crank pin $l_2 = (1 \text{ to } 1.25)\, d_2 \longrightarrow$ (eq. 16.15)

$$l_2 = 2 d_2 = 2(29) = 58$$

$$\boxed{l_2 = 58mm}$$

step8: Inner diameter of small end $D_{is} = d_1 \longrightarrow$ (eq. 16.16)

$$\boxed{D_{is} = 44mm}$$

step9: outer diameter of small end $D_{os} = d_1 + 2t_b + 2t_m \rightarrow$ (eq. 16.17)

$$= 44 + 2(4) + 2(10)$$

$$= 72mm$$

$$\boxed{D_{os} = 72mm}$$

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Step 10: Inner Diameter of big end $D_{ib} = d_2 \longrightarrow$ (eq.16.18)

$$D_{ib} = \boxed{d_2 = 29\,mm}$$

Step 11: outer diameter of big end $D_{ob} = d_2 + 2t_b + 2t_m \longrightarrow$ (eq.16.19)

$$= 29 + 2(4) + 2(10)$$

$$\boxed{D_{ob} = 57\,mm}$$

Step 12: Root diameter of bolt $d_b = 1.2 \times d_c \longrightarrow$ (16.11)

$$d_c = \left(\frac{2F_{im}}{\pi \sigma_t}\right)^{1/2} = \left(\frac{2 \times 787.5}{\pi \times 60}\right)^{1/2}$$

$$d_c = 2.9\,mm$$

$$d_c \simeq 3\,mm$$

$$d_b = 1.2 \times 3$$

$$d_b = 3.6$$

$$\boxed{d_b \simeq 4\,mm}$$

where $F_{im} = \frac{W_n}{g} \times \frac{\omega^2 r}{1000}\left[1 + \frac{1}{(l/r)}\right]$

$$\omega = \frac{2\pi N}{60} = \frac{2\pi \times 1800}{60} = 188.4$$

$$r = \frac{L_s}{2} = \frac{190}{2} = 95\,mm$$

$$W_n = 2.25$$

$$F_{im} = \frac{2.25}{9.81} \times \frac{(188.4)^2 \times 95}{1000}\left[1 + \frac{1}{\frac{380}{95}}\right] = 787.15$$

Step 13: Height of Big end $H_b = D_{ob} + 2d_b + 2t_m \longrightarrow$ (eq.16.20)

$$= 57 + 2(4) + 2(10)$$

$$\boxed{H_b = 85\,mm}$$

Step 14: Thickness of big end cap $t_c = \left[\frac{F_{im} \times l'}{b \cdot \sigma_{bc}}\right]^{1/2} \longrightarrow$ (eq.16.21)

where $l' = d_2 + 2t_b + d_b + 2t_m = 29 + 2(4) + 4 + 2(10) = 61\,mm$

$$b = d_2 - 2t_b = 58 - 2(4) = 50\,mm$$

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$$t_c = \left[\frac{787.56 \times 61}{50 \times 80}\right]^{1/2}$$

$$\boxed{t_c = 12mm}$$

<u>step 15</u>: Induced Maximum bending stress on the connecting rod

$$\sigma_b = \frac{M_{max}}{Z} \longrightarrow (eq.16.19)$$

$$\boxed{\sigma_b = 28\ N/mm^2} \leq (\sigma_{bc})$$

$$M_{max} = \frac{A P \omega^2 r l^2}{9\sqrt{3} \times 10^{12}}\ N\text{-}mm$$

$$= \frac{11(7)^2 \times 8000 \times (188.4)^2 \times 95 \times (380)^2}{9\sqrt{3} \times 10^{12}}$$ [For "A" value refer table 16.3]

$$= 134687.7\ N\text{-}mm$$

Section Modulus $[Z] = 14t^3$ [from table 16.3 in page No: 16.9]

$$= 14(7)^3$$

$$= 14(49)$$

$$\boxed{Z = 4802}$$

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* Determine the dimensions of trisection diesel engine to the particulars. Diameter of the piston is 110mm and stroke length is 140mm. length of connecting rod is 420mm with speed 1500 rpm. & rated rpm of engine is 1600. Mass of reciprocating parts of cylinder is 2.5kg. Maximum gas pressure is 4.5 Mpa and factor of safety is 7. Yield strength and compression strength is equal to 330 Mpa. Determine the whipping stresses.

Solution:

Given Data:

piston Diameter [D] = 110mm

stroke Length [L] = 140mm

Length of connecting rod [l] = 420mm

Speed [N] = 1500 rpm

Rated speed of engine = 1600 rpm

Mass of reciprocating parts / cylinder [m] = 2.5 kg

Maximum gas pressure = 4.5 MPa

Factor of safety [fs] = 7

Yield strength & compression strength = 330 mpa

Step 1: Loss due to gas pressure.

$$F_g = \frac{\pi}{4} D^2 P \longrightarrow (eq.10.1)$$

$$= \frac{\pi}{4}(110)^2 \times 4.5$$

$$= 42764.93 N$$

$$\boxed{F_g = 42764.93 N}$$

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step 2: Design Load

$$F_d = F_g \times f_s \longrightarrow (eq.\ 16.8)$$

$$= 42764.93 \times 7$$

$$= 299354.51$$

$$\boxed{F_d = 299354.51}$$

By using Ranklin's formula we will find crippling load

step 3:

$$F_{cr} = \frac{\sigma_c \cdot A}{1 + a\left(\frac{L}{K}\right)^2} \geq F_d \longrightarrow (eq.\ 16.4)$$

$$\Rightarrow [F_{cr} = F_d]$$

$$299354.51 = \frac{330 \times 11t^2}{1 + \frac{1}{7500}\left[\frac{420}{1.78t}\right]^2}$$ ["$A = 11t^2$" from fig 16.3]

$$299354.51 = \frac{3630t^2}{1 + 1.33 \times 10^{-4}\left(\frac{176400}{3.1684t^2}\right)}$$

$$299354.51 = \frac{3630t^2}{1 + \frac{7.4047}{t^2}}$$

$$299354.51\left(t^2 + 7.4047\right) = 3630t^4$$

$$3630t^4 - 299354.51t^2 = 2216630.34$$

$$\Rightarrow t^4 - 82.64t^2 - 610.64 = 0$$

$$t^2 = \frac{\pm 82.64 \sqrt{(82.46)^4 - 4(-610.64)}}{2}$$

$$\Rightarrow t^2 = 89.29$$

$$\therefore t = 9.44$$

$$\boxed{t \simeq 10mm}$$

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Step 4: Width of connecting rod (w) = 4t → (from fig 16.3)

$= 4t = 4(10)$

$= 40mm$

Height of connecting rod (H) = 5t → (from fig 16.3)

$= 5t = 5(10)$

$= 50mm$

Step 5: Diameter of piston pin $d_1 = \frac{F_g}{P_{b_1} \times l_1}$ → (eq. 16.12)

$d_1 = \frac{42764.93}{12.5 \times 1.5 d_1}$

$= 41.75$

$\boxed{d_1 \simeq 48mm}$

Step 6: length of piston pin $l_1 = (1.5 \text{ to } 2)\, d_1$ → (eq. 16.14)

$= 1.5 \times 48$

$\boxed{l_1 = 72mm}$

Step 7: Diameter of crank pin $(d_2) = \frac{F_g}{P_{b_2} \times l_2}$ → (eq. 16.13)

$= \frac{42764.93}{11 \times d_2}$

$= 62.35$

$\boxed{d_2 \simeq 63mm}$

Length of crank pin $l_2 = (1.0 \text{ to } 1.25)\, d_2$ → (eq. 16.15)

$= 1d_2 = 63$

$\boxed{l_2 = 63mm}$

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Step 8: Inner diameter of small end $D_{is} = d_1$ ⟶ (eq.16.16)

$$\boxed{D_{is} = 48\,mm}$$

outer diameter of small end $D_{os} = d_1 + 2t_b + 2t_m$ ⟶ (eq.16.17)

$$= 48mm + 2(4) + 2(10)$$

$$\left[\because \begin{array}{l} t_b = 2 \text{ to } 5\,mm \\ t_m = 5 \text{ to } 15\,mm \end{array}\right]$$

$$\boxed{D_{os} = 76\,mm}$$

Step 9: Inner diameter of Big end $D_{ib} = d_2$ ⟶ (eq.16.18)

$$\boxed{D_{ib} = 63\,mm}$$

outer diameter of Big end $D_{ob} = d_2 + 2t_b + 2t_m$ ⟶ (eq.16.19)

$$= 63 + 2(4) + 2(10)$$

$$\boxed{D_{ob} = 91\,mm}$$

Step 10: Root diameter of bolt $d_c = \left(\frac{2F_{im}}{\pi \sigma_t}\right)^{1/2}$ ⟶ (eq.16.11)

$$\Rightarrow F_{im} = \frac{W_n}{g} \times \frac{\omega^2 r}{1000}\left[1 + \frac{1}{(l/r)}\right]$$ ⟶ (eq.16.10)

$$= \frac{2.5}{9.81} \times \frac{\left(\frac{2\pi \times 500}{60}\right)^2 \times 70}{1000}\left[1 + \frac{1}{(420/70)}\right] = 511.82$$

$$\Rightarrow \sigma_t = \frac{\sigma_y}{f_s} = \frac{330}{7} = 47.14$$

$$\Rightarrow d_c = \left[\frac{2F_{im}}{\pi \sigma_t}\right]^{1/2} = \left(\frac{2(511.82)}{\pi(47.14)}\right)^{1/2} = 2.629$$

$$\boxed{d_c \simeq 3\,mm}$$

$$d_b = 1.2 \times d_c$$ ⟶ (eq.16.11)

$$= 1.2 \times 3$$

$$\boxed{d_b \simeq 4\,mm}$$

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Step 11: Height of Big end $H_b = D_{ob} + 2d_b + 2t_m$ ⟶ (eq. 16.20)

$= 91 + 2(4) + 2(10)$

$= 119\,mm$

$\boxed{H_b = 119\,mm}$

[∵ The values of d_b & t_m are taken from Data book pg: No: 16.6]

Step 12: Thickness of big-end cap $t_c = \left(\frac{F_{im} \times l'}{b \cdot \sigma_{bc}}\right)^{1/2}$ ⟶ (eq. 16.21)

⇒ Distance between bolt centres of the big end

$l' = d_2 + 2t_b + d_b + 2t_m$

$= 63 + 2(4) + 4 + 2(10)$

⇒ $\boxed{l' = 95\,mm}$

⇒ $b = d_2 - 2t_b = 63 - 2(4) = 63 - 8 = 55$

⇒ $\sigma_{bc} = 100\,N/mm^2$ (Assume)

$$t_c = \left(\frac{F_{im} \times l'}{b \cdot \sigma_{bc}}\right)^{1/2} = \left[\frac{511.82 \times 95}{55 \times 100}\right]^{1/2} = 2.97$$

∴ $\boxed{t_c \simeq 3\,mm}$

Step 13: Whipping stresses $\sigma_b = \frac{M_{max}}{Z}$ ⟶ (eq. 16.9)

$$M_{max} = \frac{A P \omega^2 4 l^2}{9\sqrt{3} \times 10^{12}}\ N\text{-}mm$$

$$= \frac{10^2 (11) \times 7800 \times \left(\frac{2\pi \times 1500}{60}\right)^2 \times 70 \times 420}{9\sqrt{3} \times 10^{12}}$$ [refer table 16.3]

$= 399.27$

$Z = 14t^3 = 14 \times (10)^3 = 14000$

$$\sigma_b = \frac{167695.14}{14000} = 11.978 \le \sigma_{bc}$$ i.e., $100\,N/mm^2$ #

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* Design a crank shaft which is overhung for the following data. Maximum load on the crank pin for maximum torque position is 50KN. Crank radius 200mm. Distance between crankpin centre & nearby bearing centre 300mm. Allowable stress in bending, shear & bearing are 70MPa, 50MPa & 7MPa respectively.

Solution: Given Data

Force transmitted from connecting rod [F] = 50KN
= 50×10^3 N

Distance b/w crankpin centre & nearby bearing centre [x] = 300mm

Radius of crank shaft [r] = 200mm

Allowable stresses:

Bending stress (σ_b) = 70 Mpa

Shear stress (τ) = 50 Mpa

Bearing stress (P_b) = 7 Mpa

Step 1: Design of crank pin

⇒ From bearing stress $P_b = \frac{F}{ld} \leq (P_b)$ ⟶ (eq. 17.3)

$\Rightarrow 7 = \frac{50 \times 10^3}{1.2d \times d}$

$\Rightarrow d = 77.15 \simeq 78mm$ $\boxed{d = 78mm}$

$l = 1.2 \times d = 1.2(78) = 93.6$

$\boxed{l = 94mm}$

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⇒ from plain on shear stress

$$\tau = \frac{4F}{\pi d^2} \leq (\tau) \longrightarrow (eq.17.1)$$

$$= \frac{4 \times 50 \times 10^3}{\pi (78)^2}$$

$$\boxed{\tau = 10.46}$$

$\tau = 10.46$ which is lessthan allowable shear stress i.e., 50Mpa

⇒ From bending stress

$$\sigma_b = \frac{16FJ}{\pi d^3} \leq (\sigma_b) \longrightarrow (eq.17.2)$$

$$= \frac{16 \times 50 \times 10^3 \times 94}{\pi \times 78^3} = 50.44$$

$$\boxed{\sigma_b = 50.44}$$

<u>step 2</u>: Design of crank shaft journal

$$D = (1.25 \text{ to } 1.5)\, d = 1.25 \times 78 = 97.5$$

$$\boxed{D \simeq 98}$$

$$L = 1.25D = 1.25 \times 98 = 122.5$$

$$\boxed{L \simeq 123}$$

As we assume $(\theta + \phi) = 90^\circ$

Radial component of force $F_H = F\cos(\theta+\phi) \longrightarrow (eq.17.14)$

$$F_H = F\cos 90^\circ$$

$$\boxed{F_H = 0}$$

Tangential component of force $F_t = F\sin(\theta+\phi) \longrightarrow (eq.17.15)$

$$F_t = F\sin 90^\circ = F$$

$$\boxed{F_t = 50 \times 10^3 \text{ N}}$$

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Let us assume $D = 140mm$

⇒ Bending stress by tangential component

$$\sigma_{bt} = \frac{32 F_t \cdot x}{\pi D^3} \leq (\sigma_b) \longrightarrow (eq.17.9)$$

$$= \frac{32 \times 50 \times 10^3 \times 300}{\pi \times 140^3}$$

$$= 55.68 < (\sigma_b)$$

$\sigma_{bt} = 55.68$ which is lessthan Allowable bending stress i.e., 70Mpa

⇒ Resultant bending stress

$$\sigma_b = \frac{32 F \cdot x}{\pi D^3} \leq (\sigma_b) \longrightarrow (eq.17.10)$$

$$= \frac{32 \times 50 \times 10^3 \times 300}{\pi \times 140^3}$$

$$= 55.68 < (\sigma_b) = 70 Mpa$$

⇒ Torsional shear stress due to tangential force

$$\tau = \frac{16 \cdot F_t \cdot H}{\pi D^3} \leq (\tau) \longrightarrow (eq.\ 17.11)$$

$$\tau = \frac{16 \times 50 \times 10^3 \times 300}{\pi (140)^3}$$

$$\tau = 18.56 < (\tau = 50 Mpa)$$

⇒ Equivalent bending stress $\sigma_{be} = \frac{1}{2}\left[\sigma_b + \sqrt{\sigma_b^2 + 4\tau^2}\right] \leq (\sigma_b) \rightarrow (eq.17.12)$

$$= \frac{1}{2}\left[55.68 + \sqrt{(55.68)^2 + 4(18.56)^2}\right]$$

$$= 61.29 < (\sigma_b) = 70 Mpa$$

$$\boxed{\sigma_{be} = 61.29 \text{ Mpa}}$$

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⇒ Equivalent shear stress

$$\tau_e = \frac{1}{2}\sqrt{\sigma_b^2 + 4\tau^2} < (\tau) \longrightarrow (eq.17.13)$$

$$= \frac{1}{2}\sqrt{(55.68)^2 + 4(18.56)^2}$$

$$= 33.45 < (\tau = 50 Mpa)$$

$$\boxed{\tau_e = 33.45}$$

Step 3 : Design of crank web

$t = (0.7 \text{ to } 1.0)\, d \longrightarrow$ [from Data Book pg: 17.5]

Width of crank web $w = \frac{a+b}{2} \longrightarrow (eq.17.17)$

$a = 1.5\, d = 1.5 \times 78 = 117 mm$

$b = 1.5\, D = 1.5 \times 140 = 210 mm$

$$w = \frac{a+b}{2} = \frac{117 + 210}{2} = \frac{327}{2} = 163.5$$

$$\boxed{w \simeq 164 mm}$$

⇒ Bending stress By tangential force

$$\sigma_{bt} = \frac{6 F_t . y}{t w^2} \leq (\sigma_b) \longrightarrow (eq.17.6)$$

$$= \frac{6 \times 50 \times 10^3 \times 200}{7 \times (164)^2}$$

$$\sigma_{bt} = 31.41 < (\sigma_b = 70 Mpa)$$

⇒ overall stress acting on the web

$$\sigma = \sigma_a + \sigma_{bn} + \sigma_{bt} \longrightarrow (eq.17.7)$$

$$= 0 + 0 + 31.41$$

$$\boxed{\sigma = 31.41} \leq (\sigma_b)\ 70 Mpa$$

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* A crank shaft of single throw centre crank type is to be designed for a diesel engine. The crank shaft is carrying one fly wheel at one end of each journal. The other specifications are diameter of piston 200mm, stroke of piston 300mm, length of connecting rod 600mm weight of each fly wheel 2KN, distance between flywheel & its nearest journal 150mm, speed of the engine 1000 rpm. Maximum gas pressure is $1 N/mm^2$. Maximum torque is experienced in the crank shaft in the crank angle is 30° from inner dead centre position. Allowable stresses for the crank shaft material are in bending $70 N/mm^2$, in shear $40 N/mm^2$, in bearing $8 N/mm^2$. Design the crank shaft & give a neat sketch.

Solution: Given Data:

Diameter of piston = 200mm

Length of stroke (L_s) = 300mm

Length of connecting rod (L_c) = 600mm

Weight of each fly wheel (W) = 2KN

Distance b/w flywheel & its nearest journal (r) = 150mm

Speed of engine (N) = 1000 rpm

Maximum gas pressure (P_{max}) = $1 N/mm^2$

Crank angle (θ) = 30°

Allowable stresses:

Bending stress (σ_b) = $70 N/mm^2$

Shear stress (τ) = $40 N/mm^2$

Bearing stress (P_b) = $8 N/mm^2$

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Step 1 : Design of crank pin

⇒ From bearing stress $P_b = \frac{F}{ld} \leq (P_b)$ ⟶ (eq.17.3)

Force $F = \frac{\pi}{4} D^2 P$ ⟶ [Not given in the data Book, Write now aside]

$= \frac{\pi}{4} (200)^2 \times 1$

$= 31415.91$

$P_b = \frac{F}{ld}$

$\Rightarrow 8 = \frac{31415.91}{1.2d \times d}$

$\Rightarrow d = 57.80$

$\boxed{d = 58mm}$

$l = 1.2 \times d = 1.2 \times 58 = 69.6$

$\boxed{l \simeq 70mm}$

⇒ From plain or shear stress $\tau = \frac{2F}{\pi d^2} \leq (\tau)$ ⟶ (eq.17.1)

$= \frac{2 \times 31415.91}{\pi \times 58^2}$

$\tau = 5.94 < (\tau = 40 N/m^2)$

⇒ Bending stress $\sigma_b = \frac{8Fx}{\pi d^3} \leq (\sigma_b)$ ⟶ (eq.17.2)

Distance b/w centres of centre crank shaft $x = L + 2t + l$ ⟶ [Not given in the Data Book]

$L = 1.25 D = 1.2 \times 58 = 72.5 \simeq 73$

$D = d = 58$

$t = (0.7 \text{ to } 1.0) d = 0.9 \times 58 = 52.2 \simeq 53$

$x = 73 + 2(53) + 70 = 249 \simeq 250$

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$$\sigma_b = \frac{8F_x}{\pi d^3}$$

$$= \frac{8 \times 31415.91 \times 250}{\pi \times 58^3}$$

$\sigma_b = 102.50 > (\sigma_b = 70)$ [change "d" value to get less value of σ_b]

take $\underline{\underline{d = 70}}$

$L = 1.25 \times 70$	$t = 0.9 \times 70$	$l = 1.2 \times 70$
$= 87.5$	$\boxed{t = 63mm}$	$\boxed{l = 84mm}$
$\boxed{L \simeq 88mm}$		

$$X = l + 2t + L = 88 + 2(63) + 84 = 298$$

$$\sigma_b = \frac{8 \times 31415.91 \times 298}{\pi \times (70)^3} = 69.50$$

$\sigma_b = 69.50 \; (< \sigma_b = 70)$

<u>step 2</u>: Design of crank shaft journal

$D = d = 70mm$

$L = 1.25 \times D = 1.25 \times d = 1.25 \times 70$

$\boxed{L = 88mm}$

$\Rightarrow$ Resultant bending stress $\sigma_b = \frac{32 W.Y}{\pi D^3} \longrightarrow$ (eq. 17.10)

$$= \frac{32 \times 2 \times 10^3 \times 50}{\pi \times 70^3}$$

$= 8.9 \; < (\sigma_b = 70)$

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⇒ Tonsional shear stress due to tangential force

$$\tau = \frac{8 F_n \cdot M}{\pi D^3} \leq (\tau) \longrightarrow (eq.\ 17.11)$$

$$F_n = F\cos(\theta+\phi) \longrightarrow (eq.\ 17.14)$$

$$\frac{L_c}{\sin\theta} = \frac{M}{\sin\phi}$$

$$\frac{600}{\sin 30} = \frac{150}{\sin\phi}$$

$$\sin\phi = \frac{150 \times \cancel{\sin 30}}{600}$$

$$\phi = \sin^{-1}\left[\frac{150 \times \sin 30}{600}\right] = 37.180$$

$$F_n = F\cos(37.180)$$

$$= 31415.9 \times \cos(37.180)$$

$$\underline{F_n = 25030.33}$$

$$F_t = F\sin(\theta+\phi) \longrightarrow (eq.\ 17.15)$$

$$= 31415.91 \times \sin(37.180)$$

$$\underline{F_t = 18985.28}$$

$$\tau = \frac{8 F_n \cdot M}{\pi D^3} = \frac{8 \times 25030.33 \times 150}{\pi \times (70)^3}$$

$$\boxed{\tau = 27.87} < (\tau = 40)$$

⇒ Equivalent bending stress $\sigma_{be} = \frac{1}{2}\left[\sigma_b + \sqrt{\sigma_b^2 + 4\tau^2}\right] \rightarrow (eq.\ 17.12)$

$$= \frac{1}{2}\left[69.50 + \sqrt{69.50^2 + 4(27.87)^2}\right]$$

$$\boxed{\sigma_{be} = 44.54}$$

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Design of crank shaft journal

$D = 65mm$

$$\sigma_b = \frac{32WY}{\pi D^3} \longrightarrow \text{eq.(17.10)}$$

$$= \frac{32 \times 2 \times 10^3 \times 150}{\pi \times (65)^3}$$

$$\boxed{\sigma_b = 11.12}$$

⇒ Equivalent shear stress $\tau_e = \frac{1}{2}\sqrt{\sigma_b^2 + 4\tau^2} \longrightarrow$ (eq.17.13)

$$= \frac{1}{2}\sqrt{(11.12)^2 + 4(34.81)^2}$$

$$= 35.25 < (\tau = 40)$$

Step 3: Design of crank web

$w = 1.5 \times d$

$= 1.5 \times 70$

$= 105$

$t = (0.7 \text{ to } 1.0)d$

$= 0.9 \times 70$

$= 63mm$

⇒ Axial stress by radial force $\sigma_a = \frac{F_R}{2w.t} \leq (\sigma_b) \longrightarrow$ (eq.17.4)

$$= \frac{25030.33}{2 \times 105 \times 63}$$

$$\boxed{\sigma_a = 1.59}$$

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⇒ Bending stress by radial force $\sigma_{bn} = \frac{3F_r[x-(l+t)]}{2wt^2} \leq (\sigma_b) \rightarrow (eq.17.5)$

$$= \frac{3 \times 25030.33\,[298-(84+63)]}{2 \times 105 \times 63^2}$$

$$\boxed{\sigma_{bn} = 13.60}$$

⇒ Bending stress by tangential force $\sigma_{bt} = \frac{6F_t \cdot M}{2tw^2} \leq (\sigma_b) \longrightarrow (eq.17.6)$

$$= \frac{6 \times 18985.28 \times 150}{2 \times 63 \times 105^2}$$

$$\boxed{\sigma_{bt} = 12.30}$$

⇒ overall stress acting on web

$$\sigma = \sigma_a + \sigma_{bn} + \sigma_{bt} \leq (\sigma_b) \longrightarrow (eq.17.7)$$

$$= 1.89 + 13.60 + 12.30$$

$$\boxed{\sigma = 27.791} < (\sigma_b = 90)$$

Hence all the induced stresses are less than all allowable stresses.

∴ our design is safe.

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