BME 552 - Machine Design Lab

Course Outcomes: The student will be able to		Blooms Taxonomy
CO1	Apply the principles of solid mechanics to design various machine Elements subjected to static and fluctuating loads.	K3
CO2	Achieve an expertise in design of Sliding contact bearing in industrial applications.	K2
CO3	Write computer programs and validate it for the design of different machine elements	K4
CO4	Evaluate designed machine elements to check their safety.	K5

A Design of Machine Elements

1. Design a knuckle joint subjected to given tensile load.
2. Design a riveted joint subjected to given eccentric load.
3. Design of shaft subjected to combined constant twisting and bending loads
4. Design a transverse fillet welded joint subjected to given tensile load.
5. Design & select suitable Rolling Contact Bearing for a shaft with given specifications
6. Design a cylinder head of an IC Engine with prescribed parameters.
7. Design of Piston & its parts of an IC Engine

B. Computer Programs for conventional designComputer and Language

Students are required to learn the basics of computer language such as C/C++/MATLAB so that they should be able to write the computer program.

1. Design a pair of Spur Gear with given specifications to determine its various dimensions using Computer Program in C/C++.
2. Design a pair of Helical Gear with given specifications to determine its various dimensions usingComputer Program in C/C++.
3. Design of Sliding Contact Bearing with given specifications & determine its various parameters usingComputer Program in C/C++.

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Department of Mechanical Engineering

Machine Design Practical File

(BME552)

S .N	Design Problem	Page No	Date	Sign

Submitted To: **Submitted By:**

Design 1: Design against Static Load

Example 4.11 The frame of a hydraulic press consisting of two identical steel plates is shown in Fig. 4.37. The maximum force P acting on the frame is 20 kN. The plates are made of steel 45C8 with tensile yield strength of 380 N/mm^2. The factor of safety is 2.5. Determine the plate thickness.

LO 4

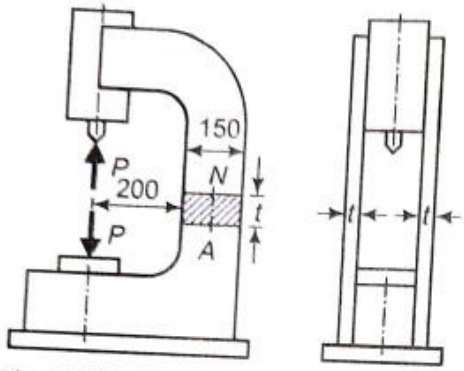

Fig. 4.37 Frame of hydraulic press

Solution **Given** $P = 20$ kN $S_{yt} = 380$ N/mm^2 $(fs) = 2.5$

Step I: Calculation of permissible tensile stress for the plates

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{380}{2.5} = 152 \text{ N/mm}^2 \qquad \text{(i)}$$

Step II: Calculation of direct tensile and bending stresses in each plate

Since the plates are identical, the force acting on each plate is (20/2) or 10 kN. The plates are subjected to direct tensile stress and bending stresses. For each plate, the direct tensile stress is given by,

$$\sigma_t = \frac{P}{A} = \frac{10 \times 10^3}{(150t)} = \frac{66.67}{t} \text{ N/mm}^2 \qquad \text{(a)}$$

The bending stresses are maximum at the inner fibre. For each plate,

$$M_b = P(200 + 75) = 10 \times 10^3 \,(200 + 75) \text{ N-mm}$$

$$I = \left[\frac{1}{12}(t)(150)^3\right] \text{mm}^4$$

$$y = \frac{1}{2}(150) = 75 \text{ mm}$$

$$\sigma_b = \frac{M_b y}{I} = \left[\frac{10 \times 10^3 (200 + 75)(75)}{\left[\frac{1}{12}(t)(150)^3 \right]} \right]$$

$$= \frac{733.33}{t} \text{ N/mm}^2 \qquad \text{(b)}$$

Step III: Calculation of plate thickness

Superimposing direct and bending stresses and equating to permissible stress,

$$\frac{66.67}{t} + \frac{733.33}{t} = 152$$

$$\frac{800}{t} = 152$$

$$t = 5.26 \text{ mm or } 6 \text{ mm} \qquad \text{(Ans)}$$

Design 2: Design against fluctuating Load

M **Example 5.3** A plate made of steel 20C8 (S_{ut} = 440 N/mm^2) in hot-rolled and normalized condition is shown in Fig. 5.28. It is subjected to a completely reversed axial load of 30 kN. The notch sensitivity factor q can be taken as 0.8 and the expected reliability is 90%. The size factor is 0.85. The factor of safety is 2. Determine the plate thickness for infinite life. **LO 6**

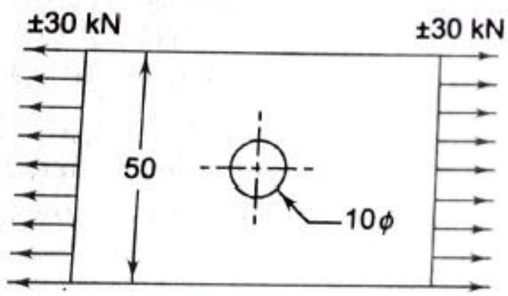

Fig. 5.28

Solution **Given** $P = \pm 30$ kN $S_{ut} = 440$ N/mm^2 $(fs) = 2$ $R = 90\%$ $q = 0.8$ $K_b = 0.85$

Step I: Endurance limit stress for plate

$$S_e' = 0.5\, S_{ut} = 0.5(440) = 220 \text{ N/mm}^2$$

From Fig. 5.24 (hot rolled steel and $S_{ut} = 440$ N/mm^2),

$$K_a = 0.67; \quad K_b = 0.85$$

For 90 % reliability, $K_c = 0.897$

$$\frac{d}{w} = \frac{10}{50} = 0.2$$

From Fig. 5.2, $K_t = 2.51$

From Eq. (5.12), $K_f = 1 + q\,(K_t - 1)$

$$= 1 + 0.8\,(2.51 - 1) = 2.208$$

$$K_d = \frac{1}{K_f} = \frac{1}{2.208} = 0.4529$$

$$S_e = K_a K_b K_c K_d S_e'$$

$$= 0.67\,(0.85)\,(0.897)\,(0.4529)\,(220) = 50.9 \text{ N/mm}$$

For axial load (Eq. 5.30)

$$(S_e)_a = 0.8\, S_e = 0.8(50.9) = 40.72 \text{ N/mm}^2$$

Step II: Permissible stress amplitude

$$\sigma_a = \frac{(S_e)_a}{(fs)} = \frac{40.72}{2} = 20.36 \text{ N/mm}^2 \qquad \text{(a)}$$

Step III: Plate thickness

$$\sigma_a = \frac{P}{(w-d)t} = \frac{(30)(10^3)}{(50-10)t} \text{ N/mm}^2 \qquad \text{(b)}$$

From (a) and (b),

$$20.36 = \frac{(30)(10^3)}{(50-10)t}; \quad t = 36.84 \text{ mm}$$

Design 3: Design of Spur Gear

Example 17.7 It is required to design a pair of spur gears with 20° full-depth involute teeth based on Lewis equation. The velocity factor is to be used to account for dynamic load. The pinion shaft is connected to a 10 kW, 1440 rpm motor. The starting torque of the motor is 150% of the rated torque. The speed reduction is 4: 1. The pinion as well as the gear is made of plain carbon steel 40C8 (S_{ut} = 600 N/mm²). The factor of safety can be taken as 1.5. Design the gears, specify their dimensions and suggest suitable surface hardness for the gears.

Solution Given kW = 10 n = 1440 rpm i = 4
S_{ut} = 600 N/mm² (fs) = 1.5
starting torque = 150% (rated torque)

Step I: Estimation of module based on beam strength

Since both gears are made of the same material, the pinion is weaker than the gear. The minimum number of teeth for 20° pressure angle is 18. Therefore,

$$z_p = 18$$

$$z_g = iz_p = 4(18) = 72$$

$$M_t = \frac{60\times10^6(\text{kW})}{2\pi n_p} = \frac{60\times10^6(10)}{2\pi(1440)}$$

$$= 66314.56 \text{ N-mm}$$

The Lewis form factor is 0.308 for 18 teeth (Table 17.3).

$$Y = 0.308$$

$$C_s = \frac{\text{starting torque}}{\text{rated torque}} = 1.5$$

The velocity factor is unknown at this stage. Assuming a trial value for the pitch line velocity as 5 m/s,

$$C_v = \frac{3}{3+v} = \frac{3}{3+5} = \frac{3}{8}$$

It is assumed that the ratio (b/m) is 10. From Eq. (17.30),

$$m = \left[\frac{60\times10^6}{\pi}\left\{\frac{(\text{kW})C_s(fs)}{z_p n_p C_v\left(\frac{b}{m}\right)\left(\frac{S_{ut}}{3}\right)Y}\right\}\right]^{1/3}$$

$$= \left[\frac{60\times10^6}{\pi}\left\{\frac{(10)(1.5)(1.5)}{(18)(1440)\left(\frac{3}{8}\right)(10)\left(\frac{600}{3}\right)(0.308)}\right\}\right]^{1/3}$$

$$= 4.16 \text{ mm}$$

Step II: Selection of module

The first preference value of the module is 5 mm.

Trial 1 $m = 5$ mm

$$d_p' = mz_p = 5(18) = 90 \text{ mm}$$

$$d_g' = mz_p = 5(72) = 360 \text{ mm}$$

$$b = 10\text{ m} = 10(5) = 50 \text{ mm}$$

Check for design

$$P_t = \frac{2M_t}{d_p'} = \frac{2(66\ 314.56)}{90} = 1473.66 \text{ N}$$

$$v = \frac{\pi\, d_p'\, n_p}{60 \times 10^3} = \frac{\pi(90)(1440)}{60 \times 10^3} = 6.7858 \text{ m/s}$$

$$C_v = \frac{3}{3+v} = \frac{3}{3+6.7858} = 0.3066$$

$$P_{\text{eff}} = \frac{C_s}{C_v} P_t = \frac{1.5(1473.66)}{0.3066} = 7209.69 \text{ N}$$

From Eq.(17.16),

$$S_b = m\, b\, \sigma_b\, Y = 5(50)(200)(0.308) = 15\ 400 \text{ N}$$

$$(fs) = \frac{S_b}{P_{\text{eff}}} = \frac{15\,400}{7209.69} = 2.14$$

The design is satisfactory and the module should be 5 mm.

Step III: Surface hardness for gears

$$Q = \frac{2z_g}{z_g + z_p} = \frac{2(72)}{72+18} = 1.6$$

$$K = 0.16\left(\frac{\text{BHN}}{100}\right)^2$$

$$S_w = bQd_p'\, K = 50(1.6)(90)(0.16)\left(\frac{\text{BHN}}{100}\right)^2$$

$$= 1152\left(\frac{\text{BHN}}{100}\right)^2$$

Since, $S_w = P_{\text{eff}}\,(fs)$

$\therefore$ $1152\left(\frac{\text{BHN}}{100}\right)^2 = 7209.69\ (1.5);$

$\therefore$ BHN = 306.39 or 310

Design 4: Design of Eccentrically Loaded Rivet

Example 8.24 A bracket, attached to a vertical column by means of four identical rivets, is subjected to an eccentric force of 25 kN as shown in Fig. 8.67(a). Determine the diameter of rivets, if the permissible shear stress is 60 N/mm^2. **LO 8**

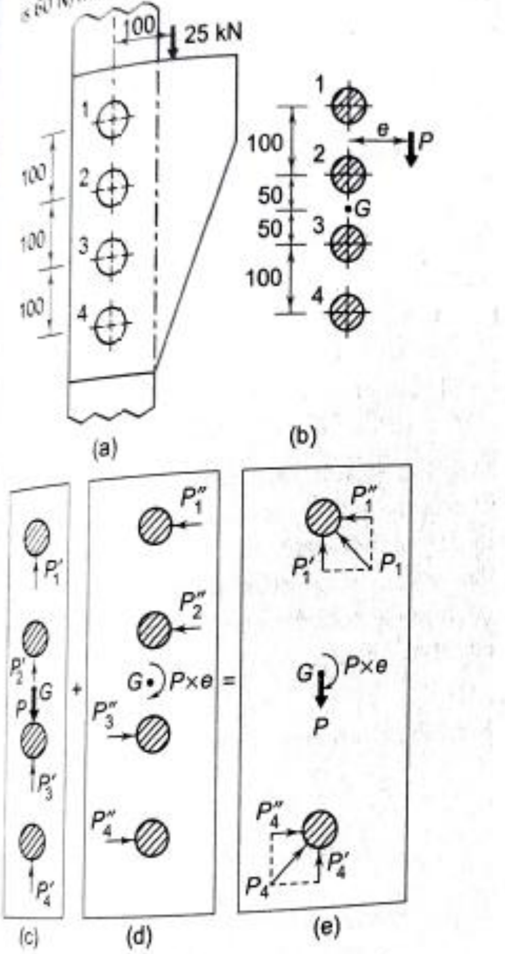

Fig. 8.67

Solution Given $P = 25$ kN $\quad e = 100$ mm

$\tau = 60$ N/mm^2

Step I: Primary shear force

The primary shear force on each rivet is shown in Fig. 8.67(c). It is given by,

$$P_1' = P_2' = P_3' = P_4' = \frac{P}{4} = \frac{25 \times 10^3}{4} = 6250 \text{ N}$$

Step II: Secondary shear force

By symmetry, the center of gravity of four rivets is located midway between the centers of rivets 2 and 3. The radial distances of rivet center from this center of gravity are as follows:

$$r_1 = r_4 = 150 \text{ mm}$$
$$r_2 = r_3 = 50 \text{ mm}$$

From Eq. (8.65),

$$C = \frac{Pe}{\left(r_1^2 + r_2^2 + r_3^2 + r_4^2\right)} = \frac{\left(25 \times 10^3\right)(100)}{\left[2(150)^2 + 2(50)^2\right]} = 50$$

$$P_1'' = P_4'' = Cr_1 = 50(150) = 7500 \text{ N}$$
$$P_2'' = P_3'' = Cr_2 = 50(50) = 2500 \text{ N}$$

Step III: Resultant shear force

Rivets 1 and 4, which are located at farthest distance from the center of gravity, are subjected to maximum shear force. As shown in Fig. 8.67(e), the primary and secondary shear forces acting on rivets 1 or 4 are at right angles to each other. The resultant shear force P_1 is given by,

$$P_1 = \sqrt{(P_1')^2 + (P_1'')^2} = \sqrt{(6250)^2 + (7500)^2}$$
$$= 9762.81 \text{ N}$$

Step IV: Diameter of rivets

Equating the resultant shear force to the shear strength of rivet,

$$P_1 = \frac{\pi}{4} d^2 \tau \quad \text{or} \quad 9762.81 = \frac{\pi}{4} d^2 (60)$$

$$\therefore \qquad d = 14.39 \text{ or } 15 \text{ mm}$$

Design 5: Design of Piston

$\therefore \quad \sigma_b < 84 \text{ N/mm}^2$ (iv)

D **Example 24.10** Design a cast iron piston for a single acting four-stroke diesel engine with following data: LO 3

Cylinder bore = 300 mm
Length of stroke = 450 mm
Speed = 300 rpm
Indicated mean effective pressure = 0.85 MPa
Maximum gas pressure = 5 MPa
Fuel consumption = 0.30 kg per BP per hr
Higher calorific value of fuel = 44 000 kJ/kg

Assume suitable data if required and state the assumptions you make.

Solution Given $D = 300$ mm

$l = 450 \text{ mm} = 0.45 \text{ m} \qquad N = 300$ rpm

$p_m = 0.85 \text{ MPa} = 0.85 \text{ N/mm}^2$

$p_{max} = 5 \text{ MPa} = 5 \text{ N/mm}^2$

$m = 0.3$ kg per BP per hr $\qquad$ HCV = 44 000 kJ/kg

Step I: Piston head or crown

Assumption No.1 The permissible tensile stress for cast iron piston is 40 N/mm^2.

From Eq. 24.16, the thickness of piston head by strength consideration is given by,

$$t_h = D\sqrt{\frac{3}{16}\frac{p_{max.}}{\sigma_b}} = 300\sqrt{\frac{3}{16}\frac{(5)}{(40)}}$$

$$= 45.93 \text{ mm} \qquad \text{(a)}$$

For four-stroke engine,

$$n = \frac{N}{2} = \frac{300}{2} = 150 \text{ strokes/min}$$

$$\text{IP} = \frac{p_m lAn}{60} = \frac{(0.85)(0.45)}{(60)}\left(\frac{\pi(300)^2}{4}\right)(150)$$

$$= 67\,593.33 \text{ W or } 67.59 \text{ kW}$$

Assumption No.2 The mechanical efficiency (η) is 80% or 0.8.

$$\text{BP} = \eta\, IP = 0.8(67.59) = 54.07 \text{ kW}$$

$$m = 0.3 \text{ kg per BP per hr}$$
$$= \left(\frac{0.3}{60 \times 60}\right) \text{ kg per BP per sec}$$
$$= (83.33 \times 10^{-6}) \text{ kg per BP per sec}$$

Assumption No.3 The ratio of heat absorbed by the piston to the total heat developed in the cylinder is 0.05 or 5 %. ($C = 0.05$)

From Eq. 24.19,

$$H = [C \times \text{HCV} \times m \times \text{BP}] \times 10^3$$
$$= [0.05 \times 44000 \times (83.33 \times 10^{-6}) \times 54.07] \times 10^3$$
$$= 9912.44 \text{ W}$$

Assumption No.4 The thermal conductivity factor (k) for cast iron is 46.6 W/m/°C.

Assumption No.5 The temperature difference ($T_c - T_e$) between the centre and the edge of piston head is 220°C

From Eq. 24.18, the thickness of piston head by thermal consideration is given by,

$$t_h = \left[\frac{H}{12.56\,k(T_c - T_e)}\right] \times 10^3$$
$$= \left[\frac{9912.44}{12.56(46.6)(220)}\right] \times 10^3$$
$$t_h = 76.98 \text{ mm} \qquad \text{(b)}$$

From (a) and (b), thermal consideration is the criterion for piston thickness.

$$t_h = 76.98 \text{ or } 77 \text{ mm}$$

Step II: Radial ribs

Since $t_h = 77$ mm $t_h > 6$ mm

Therefore, ribs are required.

Assumption No.6 The number of radial ribs is 4.

From Eq. 24.22,

$$t_R = \left(\frac{t_h}{3}\right) \text{to} \left(\frac{t_h}{2}\right) = \left(\frac{77}{3}\right) \text{to} \left(\frac{77}{2}\right) = 25.67 \text{ to } 38.5$$
$$t_R = 30 \text{ mm}$$

Step III: Requirement of cup

$$\left(\frac{l}{D}\right) = \left(\frac{450}{300}\right) = 1.5$$

Therefore, a cup is required.

Radius of cup $= 0.7\,D = 0.7(300) = 210$ mm

Step IV: Piston rings

Assumption No.7 The allowable radial pressure (p_w) on cylinder wall is 0.035 MPa.

Assumption No.8 The piston rings are made of cast iron and permissible tensile stress is 90 N/mm².

Assumption No.9 The number of compression rings is 3 and there is one oil ring.

From Eq. 24.25,

$$b = D\sqrt{\frac{3p_w}{\sigma_t}} = 300\sqrt{\frac{3(0.035)}{90}} = 10.25 \text{ or } 10.5 \text{ mm}$$

From Eq. 24.26,

$$h = (0.7\,b) \text{ to } b = (0.7 \times 10.5) \text{ to } 10.5$$
$$= 7.35 \text{ to } 10.5 \text{ mm}$$
$$h = 8 \text{ mm}$$
$$z = 3 + 1 = 4$$

Also, $h_{min} = \left(\frac{D}{10z}\right) = \left(\frac{300}{10(4)}\right) = 7.5$ mm

$$h > h_{min}$$

The gap between free ends of piston ring before assembly is given by,

$$G_1 = 3.5\,b \text{ to } 4\,b = 3.5(10.5) \text{ to } 4(10.5)$$
$$= 36.75 \text{ to } 42 \text{ mm}$$
$$G_1 = 40 \text{ mm}$$

The gap between free ends of piston ring after assembly is given by,

$$G_2 = 0.002\,D \text{ to } 0.004\,D$$
$$= 0.002(300) \text{ to } 0.004(300) = 0.6 \text{ to } 1.2 \text{ mm}$$
$$G_2 = 0.8 \text{ mm}$$

From Eq. 24.28, the width of top land is given by,

$$h_1 = (t_h) \text{ to } (1.2\,t_h) = (77) \text{ to } (1.2 \times 77)$$
$$= 77 \text{ to } 92.4 \text{ mm}$$
$$h_1 = 85 \text{ mm}$$

From Eq. 24.29, the width of ring grooves is given by,

$$h_2 = 0.75\,h \text{ to } h = 0.75(8) \text{ to } 8$$
$$= 6 \text{ to } 8 \text{ mm} \quad \therefore \quad h_2 = 7 \text{ mm}$$

Step V: Piston barrel

From Eq. 24.30, the thickness of barrel at the top end is given by

$$t_3 = (0.03\,D + b + 4.9)$$
$$= 0.03(300) + 10.5 + 4.9 = 24.4 \text{ mm or } 25 \text{ mm}$$

From Eq. 24.31, the thickness of barrel at open end is given by,

$$t_4 = (0.25\,t_3) \text{ to } (0.35\,t_3) = (0.25 \times 25) \text{ to } (0.35 \times 25)$$
$$= 6.25 \text{ to } 8.75 \text{ mm}$$
$$t_4 = 7 \text{ mm}$$

Step VI: Piston skirt

Assumption No.10 The allowable bearing pressure (p_b) for skirt portion of piston is 0.45 MPa.

Assumption No.11 The ratio of side thrust on liner to maximum gas load on piston is 0.1. ($\mu = 0.1$)

From Eq. 24.32, the length of skirt is given by,

$$\mu\left(\frac{\pi D^2}{4}\right)p_{max} = p_b D l_s \text{ or } 0.1\left(\frac{\pi (300)^2}{4}\right)(5)$$

$$= (0.45)(300)l_s \quad \therefore \quad l_s = 261.8 \text{ or } 262 \text{ mm}$$

Step VII: Piston length

Refer to Fig. 24.10.

$$\text{Length of ring section} = 4\,h + 3\,h_2 = 4(8) + 3(7)$$

$$= 53 \text{ mm} \qquad \text{[Refer Fig. 24.4]}$$

$$\text{Length of piston} = h_1 + \text{length of ring section} + l_s$$

$$= 85 + 53 + 262 = 400 \text{ mm}$$

$$L = 400 \text{ mm}$$

According to empirical relationship,

$$L = D \text{ to } 1.5\,D = 300 \text{ to } 450 \text{ mm;}$$

$$\therefore \quad D < L < 1.5\,D$$

Step VIII: Piston pin

Assumption No.12 The bearing pressure $(p_b)_1$ at the bush of small end of connecting rod is 30 MPa.

Assumption No.13 The length of piston pin in bush of small end of connecting rod is (0.45 D).

Assumption No.14 The piston pin is made of case hardened alloy steel and the permissible tensile stress is 140 N/mm^2.

$$P = \text{Force on piston} = \left(\frac{\pi D^2}{4}\right)p_{max.}$$

$$= \left(\frac{\pi(300)^2}{4}\right)(5) = 353\,429.17 \text{ N}$$

From Eq. 24.37,

$$\left(\frac{\pi D^2}{4}\right)p_{max.} = (p_b)_1 \times d_o \times l_1 \qquad [l_1 = 0.45D]$$

$$353\,429.17 = 30\,d_o\,(0.45 \times 300);$$

$$\therefore \quad d_o = 87.27 \text{ or } 90 \text{ mm}$$

In order to make the shaft hollow, we will increase the diameter to 120 mm.

$$d_o = 120 \text{ mm}$$

From Eq. 24.39,

$$M_b = \left(\frac{PD}{8}\right) = \frac{(353429.17)(300)}{8}$$

$$= 13\,253.59 \times 10^3 \text{ N-mm}$$

$$I = \frac{\pi(d_o^4 - d_i^4)}{64} = \frac{\pi[(120)^4 - d_i^4]}{64} \text{ mm}^4$$

$$y = \left(\frac{d_o}{2}\right) = \left(\frac{120}{2}\right) = 60 \text{ mm}$$

$$\sigma_b = \frac{M_b y}{I} \quad \text{or} \quad 140 = \frac{(13253.59 \times 10^3)(60)}{\frac{\pi}{64}[(120)^4 - d_i^4]}$$

$$(120)^4 - d_i^4 = 115.71 \times 10^6; \quad \therefore \; d_i = 97.84 \text{ or } 95 \text{ mm}$$

The mean diameter of piston boss is given by,

$$\text{mean diameter of piston boss} = 1.4\,d_o = 1.4(120)$$