dia. of Flange & PCD

Unit - I

* **Materials for Piston Rings :-**

requirements of material :-

1. wear resistance
2. Spring characteristics at high temperature

mtrl:

1. Grey C.I — It has good wear resistance & spring characteristics at high temp.

2. Alloy C.I

Some times Piston rings are chromium plated to reduce wear.

* **Piston Pin**

characteristics of mtrl :-

1. It should have good bearing strength
2. — " — " — good bending strength.
3. Good wear resistance

mtrl :-
It is made of 1. Carbon steel
or
2. alloy steel.

It is hardened and ground to reduce wear while turning inside phosphorus bronze bush.

material for Connecting rod

- It is made ~~up of on~~ by the drop forging process and outter surfaces are left unfinished
- The most I.C Engine have conventional 2-piece Connecting rod.
- The whole rod is forged in one piece the bearing cap is cut off, faced & bolted in place for final machining of the big end.
- The small end is general made of Solid eye. ~~the~~ & then machined.
- C.R is most heavily stressed part in I.C. Engine subjected to either gas force & Inertia forces continuously
- Mtrl. used are
  - Medium Carbon Steel — Contains 0.35 to 0.45%. carbon
    or
  - alloy steels. — It include (1) nickel chromium
    (2) Chromium Molybdenum steel.

- Medium Carbon steel are used in C.R of industrial application I.C Engine.
- Alloy Steel are used for Automobile & aero I.C. Engine connecting rod, ~~&~~

The

material of Crankshaft

- Crankshaft should have sufficient strength to withstand bending & twisting moments.

- It should have sufficient rigidity to keep the lateral and angular deflection within permissible limits.

- It is subjected to fluctuating stresses

- They are made ~~up of~~ by drop forging process

- mtls used:
  1. Plain carbon steels — 40C8, 45C8, and 50C4.
  2. Alloy steel. — nickel chromium steels such as. 16 Ni3Cr2, 35 Ni5Cr2, and 40Ni10 Cr3 Mo6..

materials of valve:-

- In slow speed engines, valve has composite construction with C.I head and steel stem.

- In high speed engines one piece construction is used & valves are forged

- Exhaust valves are subjected to temp. 1900°c to 2200°c.

* materials for cylinder liners

desirable properties for cylinders & cylinder liners

(1) Strong ~~eng~~ enough to withstand high gas pressure ~~of~~ during combustion of fuel.

(2) It should withstand thermal stresses

(3) wear resistance should be good.

(4) Good surface finish, to reduce friction & wear.

(5) Corrosion resistant (As water jacket are provided)

mtrls :-

(1) Grey C.I with homogeneous and closed grained structure — commonly used.

(2) For heavy duty, Nickel C.I.
Chromium C.I

(3) Sometimes cast steel and alluminium alloy are used for cylinders.

— they are made by centrifugal castings

* ~~Piston~~ materials for Piston

Requirement :- 1) high strength to withstand gas pressure
2) high rigidity to withstand thermal & mechanical distortions
3) Good heat conductivity
4) light weight to reduce Inertia forces
5) wear resistance, bearing resistance

materials:-

commonly used materials : Cast iron, cast steel, forged steel cast alluminium alloys, forged alluminium alloys.

C.I	Alluminium alloy.
1) high strength. 2) high wear strength.	1) high thermal conductivity 2) light wt. 3) Coefficient thermal expansion (high)

The properties of valve mtrl are:

1. It should be heat resistant
2. good thermal conductivity
3. Corrosion resistant
4. wear resistant . surface
5. Shock ~~resistan~~ resistant

— Inlet valves are subjected to less temperature comparitevely to exhaust temp.

Inlet valves are made of :- Nickel-chromium steel.

exhaust values are made of :- heat resistant Silicon - Chromium.

- For heavy duty engines valves are made of - chromium - vanadium steel.

- Valves are heat treated & surface hardness is in range of 250 to 300 HB.

* material for Rocker arm

- In order to reduce wt. & Inertia rocker-arm are made of I-section.
- They are made of Grey-C.I., malleable Cast Iron or Cast steel.

* <u>valve spring</u>

- valve spring is made of oil-hardened and tempered valve spring wire of Grade - VW.

* <u>Push rod</u>

- ~~Push~~ Push rod are made ~~up~~ of bright drawn steel tubes with 4% Carbon or duralumin tubes.

* <u>Flywheel</u>

C.I and Cast steel.

Unit-I

DESIGN OF ENGINE COMPONENTS

* Cylinders and Cylinder Liner

Functions:-

1. To retain working fluid. (i.e. Air Fuel mixture)
2. To guide the piston

— very high temperature in cylinder, hence cooling required

Cooling :-

1. Air cooling — Small Engine (Fins) Scooters, motorcycle
2. water cooled → water Jacket.

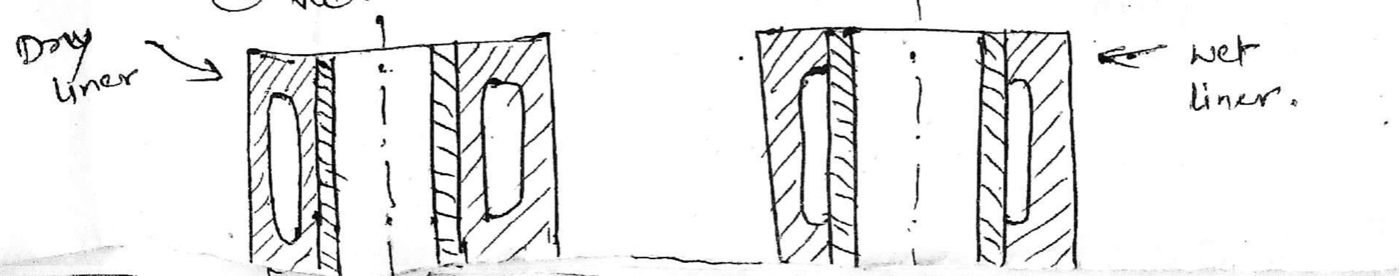
— Small engine, cylinder liner is one piece casting
- large Engine, seperate liner is used.

Advantages of Seperate Liner

1. Replacement of liner is easy after wornout.
2. Instead of using better material for whole assembly only cylinder liner mtl. can be used of better grade
3. Cylinder liner allows longitudinal expansion.

— Types of liners:

1. Dry liner. — doesn't have any contact with cooling water
2. wet liner — has contact with outer surface.



* Bore & length of Cylinder

$$\boxed{\eta = \frac{B.P}{I.P}}$$

and

$$\boxed{I.P = \frac{P_m L A n K}{60}}$$

where

P_m = mean effective Pressure

l = length of stroke

A = Area of C/s = $\frac{\pi}{4} D^2$

no. of working strokes per min, $n = N$ — For 2-stroke

$n = \frac{N}{2}$ — For 4-stroke

K = No. of Cylinders.

D = Bore Dia. of Cylinder.

$$\boxed{\frac{l}{D} = 1.25 \text{ to } 2}$$

If not given take $\frac{l}{D} = 1.5$

— The length of Cylinder is more than length of stroke. There is clearance on both sides. The total clearence on two sides is taken as 15% of stroke length

~~length of stroke~~

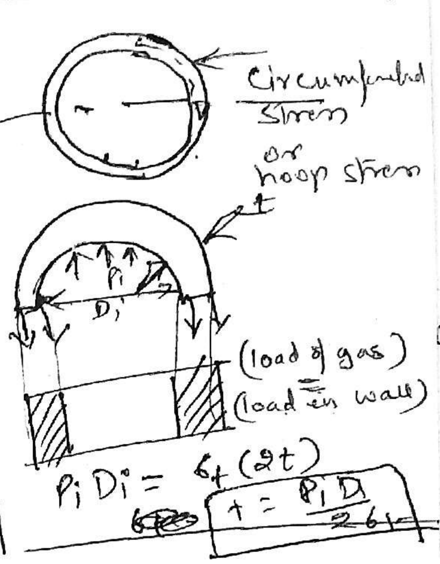
$$\boxed{\text{length of Cylinder } (L) = 1.15\, l.}$$

* Thickness of Cylinder wall

The engine Cylinder or Cylinder liner is treated as thin cylinder

The thickness of thin cylinder is given by, (circumferential or hoop stress)

$$\boxed{t = \frac{P_{max} D}{2\sigma_c} + C}$$



where,

t = thickness of Cylinder wall

P_{max} = maximum gas pressure (N/mm²)

D = Inner Bore diameter (mm)

σ_c = Circumferential or hoop stress

C = Reboring allowance (mm)

Note

① If P_{max} not given than $\boxed{P_{max} = 10(P_m)}$ — assumed.

② $\boxed{\sigma_c = \sigma_t = \frac{S_{ut}}{f_s}}$

if σ_c & f_s not given,

take σ_c = 35 to 100 N/mm²

③ Reboring allowance:-

Reboring is done to compensate the wear on inner wall of cylinder. hence to make Surface smooth, Reboring is to be done

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

* Empirical relations

(i) thickness of cylinder

$$\boxed{t = 0.045D + 1.6} \quad (mm)$$

(ii) thickness of dry liner

$$= 0.03D \text{ to } 0.035D$$

(iii) thickness of water jacket wall

$$= \left(\frac{1}{3}\right)t \text{ to } \left(\frac{3}{4}\right)t$$

or

$$= 0.032D + 1.6 \text{ mm}$$

(iv) Water space between outter cylinder wall & Inner Jacket wall = ~~9 mm for 75mm Cylinder~~

= (9mm ~~For~~ For 75mm cylinder bore)

to

(75mm for 750 mm Cylinder bore)

or

$$= 0.08D \text{ to } 6.5 \text{ mm}$$

(v) Thickness of Cylinder Flange

$$= 1.2t \text{ to } 1.4t.$$

or

$$= 1.25d \text{ to } 1.5d$$

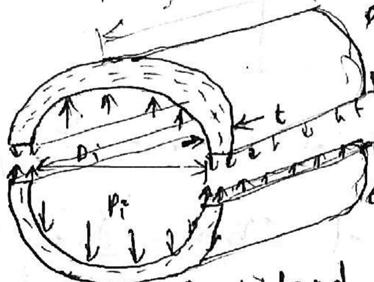
where d = nomial dia. of bolt.

(vi) Radial distance between outer dia. of Flange & PCD of studs (bolts)

$= (d+6)$ to $(1.5d)$ mm.

* stresses in cylinder wall :-

a) Circumferential stress (Hoop stress)



P_i = Internal Pressure of cylinder

D_i = Internal diameter " — " —

t = thickness of ~~vessels~~ cylinder

σ_t = Hoop stress.

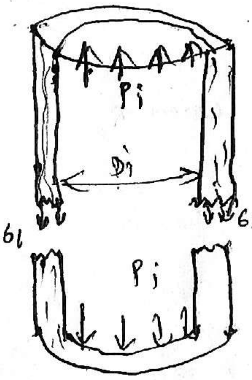
Circumferential load inside cylinder = circum. load on vessel sheets.

$$(P_i \times [D_i \cdot l]) = \sigma_t \cdot (2t) \cdot l$$

$$\boxed{\sigma_t = \frac{P_i D_i}{2t}}$$

b) Longitudinal stress :-

σ_l = Longitudinal stress.



Long. load inside cylinder = longitudinal load on vessel sheets

$$P_i \cdot \left(\frac{\pi}{4} D_i^2\right) = \sigma_l (\pi D_i \cdot t).$$

$$\boxed{\sigma_l = \frac{P_i D_i}{4t}}$$

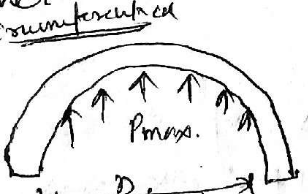
– So comparing both stresses Hoop stress is more severe

(a) Apparent stresses (for engine cylinders)

Two principal stresses in cylinder → ① Circumferential (Hoop) ② longitudinal.

It is assumed stresses are uniformly distributed over wall thickness.

a) circumferential

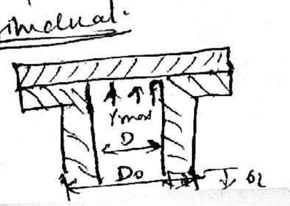


$$\boxed{\sigma_c = \frac{P_{max} \cdot D}{2t}}$$

$$[P_{max} \cdot D] = [\sigma_c \cdot 2t]$$

$$\sigma_c = \frac{P_{max} \cdot D}{2t}$$

b) longitudinal.



$$\left[P_{max} \cdot \frac{\pi}{4} D^2\right] = \left[\sigma_l \cdot \frac{\pi}{4}(D_o^2 - D^2)\right]$$

$$\boxed{\sigma_l = \frac{P_{max} \cdot D^2}{(D_o^2 - D^2)}}$$

(b) Net stresses :-

σ_c & σ_l act Right angle to each other. There fore Net stresses are reduced in Particular directions.

$$\boxed{(\sigma_c)_{net} = \sigma_c - \mu \cdot \sigma_l}$$

$$\boxed{(\sigma_l)_{net} = \sigma_l - \mu \sigma_c}$$

where,

σ_c = apparent circumferential stress = $\frac{P_{max} D}{2t}$.

σ_l = apparent longitudinal Stress = $\frac{P_{max} D^2}{(D_o^2 - D^2)}$

$(\sigma_c)_{net}$ = Net circumferential stress.

$(\sigma_l)_{net}$ = Net longitudinal stress

μ = Poission's ratio = 0.25. (for C.I.)

Cylinder Head :-

The cylinder head or cover accomodates following Parts :-

1. Inlet & exhaust valves.
2. Air & gas ports.
3. Spark plug - Petrol Engine
   atomizer - Diesel Engine

Due to this the head becomes Complicated.

Shape :- The box type section with ~~consi~~ Considerable thickness is used for Cylinder head.

Calculating various dimensions of the actual cylinder head is difficult ~~excess~~ exercise.

The cylinder head is assumed a circular plate and its thickness is given by

$$\boxed{t_h = D\sqrt{\frac{KP_{max}}{\sigma_c}}}$$

where,

σ_c = allowable circumferential stress

$K = 0.162$

σ_c = ~~30~~ 30 to 50 N/mm² — if (σ_c and f_t not given.)

* Design of Studs (bolts) For Cylinder head

Studs used to make assembly of cylinders, cylinder head & gasket.

Studs are subjected to tensile stress due to gas pressure.

Parameters to design:-

1. No. of studs (Z)

$$Z_{min} = 0.01D + 4$$

$$Z_{max} = 0.02D + 4$$

2. Diameter of stud

Gas force on cylinder cover $= \left(\frac{\pi}{4} D^2\right) P_{max}$

Resisting force offered by stud $= Z\left(\frac{\pi d_c^2}{4}\right) \cdot \sigma_t$

Equating both

$$\left(\frac{\pi}{4} D^2\right) P_{max} = z\left(\frac{\pi}{4} d_c^2\right) \cdot \sigma_t$$

(i) studs are made of steel. here

$$\sigma_t = \frac{S_{yt}}{f_s}$$

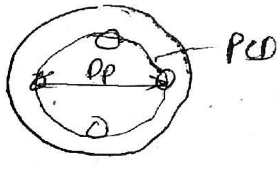
if S_{yt} & f_s not given

$$\sigma_t = 35 \text{ to } 70 \text{ N/mm}^2$$

$$\boxed{d = \frac{d_c}{0.8}}$$

(ii) Pitch of studs

P.C.D of studs, $\boxed{D_p = D + 3d}$



$$\boxed{\text{pitch of stud} = \frac{\pi D_p}{z}}$$

In order to be leak proof joint, the pitch must be,

minimum pitch $= 19\sqrt{d}$

Maximum pitch $= 28.5\sqrt{d}$

The ~~foll~~ Analysis is Elementary only

other Parameter required for analysis of stud.

1. Stiffness of gasket
2. — " — " — Flanges of cylinder
3. — " — " — " — of cover
4. initial preload.

25.) 4-stroke diesel Engine.

$B.P = 3.75\,kW$

$N = 1000\,rpm$

$P_m = 0.35\,MPa$

$\eta_{mech} = 80\%$

Bore, $D = ?$

length of cylinder $L = ?$

$$\eta_{mech} = \frac{B.P}{I.P.}$$

$$0.8 = \frac{3.75}{I.P}$$

$$\boxed{I.P = 4.68\,kW}$$

$$I.P = \frac{P_m L A N}{60}$$

$\cancel{4.68 \times 10^3 = (0.35 \times 10^6)(L)(\frac{\pi}{4}}$ Assume $\frac{l}{D} = 1.5$

Since length of stroke $l = 1.5D$

$$(4.68 \times 10^3) = \frac{(0.35 \times 10^6)(1.5D)(\frac{\pi}{4}D^2)(1000)}{60 \times 2}$$

$$561600 = 412334036\,D^3$$

$$0.00136 = D^3$$

$$\boxed{D = 0.110\,m}$$

$$\boxed{D = 110\,mm.}$$

$$\frac{l}{D} = 1.5$$

length of stroke, $\boxed{l = 166.27\,mm}$

length of cylinder $L = 1.15\,l$

$$\boxed{L = 191.21\,mm}$$

V.B. Bhandari
25.2

4-stroke Diesel Engine.

$D = 150mm$

$P_{max} = 3.5$ Mpa.

Cylinder matl = Grey C.I FG200 ($S_{ut} = 200 N/mm^2$)

$f_s = 5$

$\mu = 0.25$

thickness of cylinder wall, $t = ?$

apparent stress = ?

net circumferential stress = ?

longitudinal stress = ?

A (i) thickness

For cylinder wall,

$$\sigma_t = \frac{S_{ut}}{f_s} = \frac{200}{5} = 40 N/mm^2.$$

from table

for $D = 150mm$, $C = 4mm$.

$$t = \frac{P_{max} D}{2\sigma_c} + C$$

$$= \frac{(3.5)(150)}{2(40)} + 4$$

$$= 10.56 mm$$

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

(ii) Apparent Stresses.

(1) Circumferential stress,

$$\sigma_c = \frac{P_{max} D}{2t}$$

$$= \frac{(3.5)(150)}{2(12)} = 21.88 N/mm^2$$

since

$(\sigma_c < 40\,N/mm^2)$

(2) Longitudinal stress

$D_o = D + 2t$

$= 150 + 2(12) = 174\,mm$

$\sigma_l = \frac{P_{max} D^2}{(D_o^2 - D^2)}$

$= \frac{(3.5)(150)^2}{(174^2) - (150^2)}$

$\sigma_l = 10.13\,N/mm^2$

~~$\sigma_l < 40\,N/mm^2$~~

(III) Net stress

(1) Circumferential

$(\sigma_c)_{net} = \sigma_c - \mu\sigma_l$

$= 21.88 - \{(0.25)(10.13)\}$

$\boxed{(\sigma_c)_{net} = 19.35\,N/mm^2}$

(2) longitudinal stress

$(\sigma_l)_{net} = \sigma_l - \mu\sigma_c$

$= (10.13) - (0.25 \times 21.88)$

$\boxed{(\sigma_l)_{net} = 4.66\,N/mm^2}$

4-stroke Diesel Engine.

2.5.3. $D = 150$ mm.

$P_{max} = 3.5$ mpa

Cylinder head mtrl. C I FG (200)

$S_{ut} = 200 N/mm^2$.

$f_s = 5$.

$t_h = ?$

Studs,

Fe E (250) ($S_{yt} = 250 N/mm^2$), $f_s = 5$

(1) $z = ?$

(2) $d = ?$

(3) P.C.D = ? & Pitch = ?

(I) thickness of cylinder head

$$\sigma_t = \frac{S_{ut}}{f_s}$$

$$\sigma_t = \frac{200}{5} = 40 N/mm^2$$

$\sigma_t = \sigma_c =$ Allowable circumferential stress

$$t_h = D\sqrt{\frac{K p_{max}}{\sigma_c}}$$

$$= 150\sqrt{\frac{(0.162)(3.5)}{40}}$$

$t_h = 17.85$ mm

$$\boxed{t_h = 18 \text{ mm}}$$

(II) No. of studs.

$z_{min} = 0.01 D + 4$

$= 0.01 (150) + 4 = 5.5$

$$Z_{max} = 0.02D + 4$$
$$= 0.02(150) + 4$$
$$Z_{max} = 7$$

So take $\boxed{Z = 6 \text{ studs}}$

(iii) Nominal dia. of studs

$$P_{max}\left(\frac{\pi}{4}D^2\right) = \sigma_t \cdot Z \cdot \frac{\pi}{4} d_c^2.$$

but $\sigma_t = \frac{S_{yt}}{f_s} = \frac{250}{5} = 50 \text{ N/mm}^2$

$$(3.5)\left[\cancel{\frac{\pi}{4}}(150)^2\right] = (50)(6)\left(\cancel{\frac{\pi}{4}}\right)d_c^2$$

$$3.5(150)^2 = 300 \times d_c^2.$$

$$d_c^2 = 262.5$$

$$\boxed{d_c = 16.20 \text{ mm}}$$

$$d = \frac{d_c}{0.8}$$

$$= \frac{16.20}{0.8}$$

$$d = 20.25 \text{ mm}$$

$$\boxed{d = 20 \text{ mm}}$$

(iv) Pitch of stud

D.C.D of studs (D_p)

$= D + 3d$

$= 150 + 3(20)$

$\boxed{D_p = 210 \text{ mm}}$

Pitch of studs $= \dfrac{\pi D_p}{z}$

$= \dfrac{\pi (210)}{(6)}$

$\boxed{\text{Pitch of studs} = 109.96 \text{ mm}}$

Minimum pitch $= 19\sqrt{d} = 19\sqrt{20} = 84.97$ mm

Maximum pitch $= 28.5\sqrt{d} = 28.5\sqrt{20}$

$= 127.46$ mm.

it is in limit.

25.4 4-stroke diesel Engine.

B.P = 7.5 kW

N = 1400 rpm.

$P_m = 0.35$ mpa.

$\eta_{mech} = 80\%$

$P_{max} = 3.5$ mpa.

Cylinder liner & head,

mtl: Grey C.I FG(260) $S_{ut} = 260$ N/mm^2

$\mu = 0.25$.

Studs, Plain carbon steel 40C8 ($S_{yt} = 380$ N/mm^2)

$f_s = 6$.

~~Req~~ (1) $D = ?$, $L = ?$

(2) $t = ?$

(3) $t_n = ?$

(4) $d = ?$, $z = ?$, $D_p = ?$, Pitch $= ?$

~~A~~

$\eta_{mech} = \frac{B.P}{I.P}$

$0.80 = \frac{7.5}{I.P}$

$\boxed{I.P = 9.375 \text{ kW}}$

let $\frac{l}{D} = 1.5$

$I.P = \frac{P_m l A N}{60}$

$9.375 \times 10^3 = \frac{(0.35 \times 10^6)(1.5D)(\frac{\pi}{4}D^2)(\frac{1400}{2})}{60}$

$562500 = 288633825 \, D^3$

$D^3 = 0.0019488$

$D = \cancel{0.04413 \text{ m}} \; 0.1249 \text{ m}$

$\boxed{D = \cancel{44.13 \text{ mm}} \; 125 \text{ mm}}$

$l = 1.5D$

$\boxed{l = 187.5 \text{ mm}}$

Length of cylinder $= L = 1.15 l$

$L = 215.46 \text{ mm}$

$\boxed{L = 216\ mm}$

(ii) Thickness of liner,

$$t = \frac{P_{max} D}{2\sigma_c} + C$$

$$\therefore \sigma_c = \frac{S_{ut}}{f_s} = \frac{260}{6} = 43.33\ N/mm^2$$

$$t = \frac{(3.5)(125)}{2(43.33)} + 3.2$$

$\boxed{t = 8.24\ mm}$ or $\underline{10\ mm}$

(iii) thickness of cylinder head

$$t_h = D\sqrt{\frac{K P_{max}}{\sigma_c}}$$

$$= (125)\sqrt{\frac{(0.162)(3.5)}{43.33}}$$

$$= 14.29\ mm$$

$\boxed{t_h = 15\ mm}$

Apparent stress

(I) circumferential.

$$\sigma_c = \frac{P_{max} D}{2\sigma_c} = \frac{(3.5)(125)}{2(43.33)}$$

$\sigma_c = 21.88\ N/mm^2$

$\sigma_c < 43.33\ N/mm^2$

Longitudinal

$$\sigma_l = \frac{P_{max} D^2}{(D_o^2 - D^2)}$$

$$= \frac{(3.5)(125)^2}{[(125+20)]^2 + (125)^2}$$

$$\boxed{\sigma_l = 10.13\ N/mm^2}$$

Net stress

$(\sigma_c)_{net} = \sigma_c - \mu \sigma_l$

$= 21.88 - [(0.25)(10.13)]$

$\boxed{\sigma_{c\,net} = 19.35\ N/mm^2}$

$(\sigma_l)_{net} = \sigma_l - \mu \sigma_c$

$= 10.13 - [(0.25)(21.88)]$

$\boxed{(\sigma_l)_{net} = 4.66\ N/mm^2}$

(iv) No. of studs

$Z_{min} = 0.01D + 4$

$= 0.01(125) + 4 = 5.25$

$Z_{max} = 0.02D + 4$

$= 0.02(125) + 4$

$= 6.5$

$\boxed{Z = 6}$

(v) Nominal dia of studs

$$\sigma_t = \frac{380}{6} = 63.33 \text{ N/mm}^2$$

$$P_{max} \cdot \left(\frac{\pi}{4} D^2\right) = z \cdot \left(\frac{\pi}{4} d_c^2\right) \cdot \sigma_t$$

$$(3.5)\left[\frac{\pi}{4}(125)^2\right] = (6)\left[\frac{\pi}{4}(d_c^2)\right](63.33)$$

$$d_c^2 = 143.92$$

$$\boxed{d_c = 12 \text{mm}}$$

$$d = \frac{d_c}{0.8} = 15 \text{mm}$$

$\boxed{d = 15 \text{mm}}$ — nominal dia.

(vi) P.C.D of studs $(D_p) = D + 3d$

$$= 125 + 3(15) = 170 \text{mm}.$$

Pitch of stud $= \frac{\pi D_p}{z} = \frac{\pi(170)}{6} = 89.01 \text{mm}$

Limits

Pitch minimum $= 19\sqrt{d} = 19\sqrt{15} = 73.59 \text{mm}$

Pitch max $= 28.5\sqrt{d} = 28.5\sqrt{15} = 110.38 \text{mm}$

Here its in limit

* Piston

The Piston is a reciprocating part of IC Engine that performs a number of functions. The main functions of the piston are as follows;

(i) It Transmits force due to gas to the crankshaft through connecting Rod.

(ii) It compresses gas.

(iii) It seals inside portion of cylinder from the crankcase by means of piston rings.

(iv) It takes side thrust resulting form obliquity of connecting rod.

(v) It dissipates large amount of heat from the combustion chamber to cylinder wall.

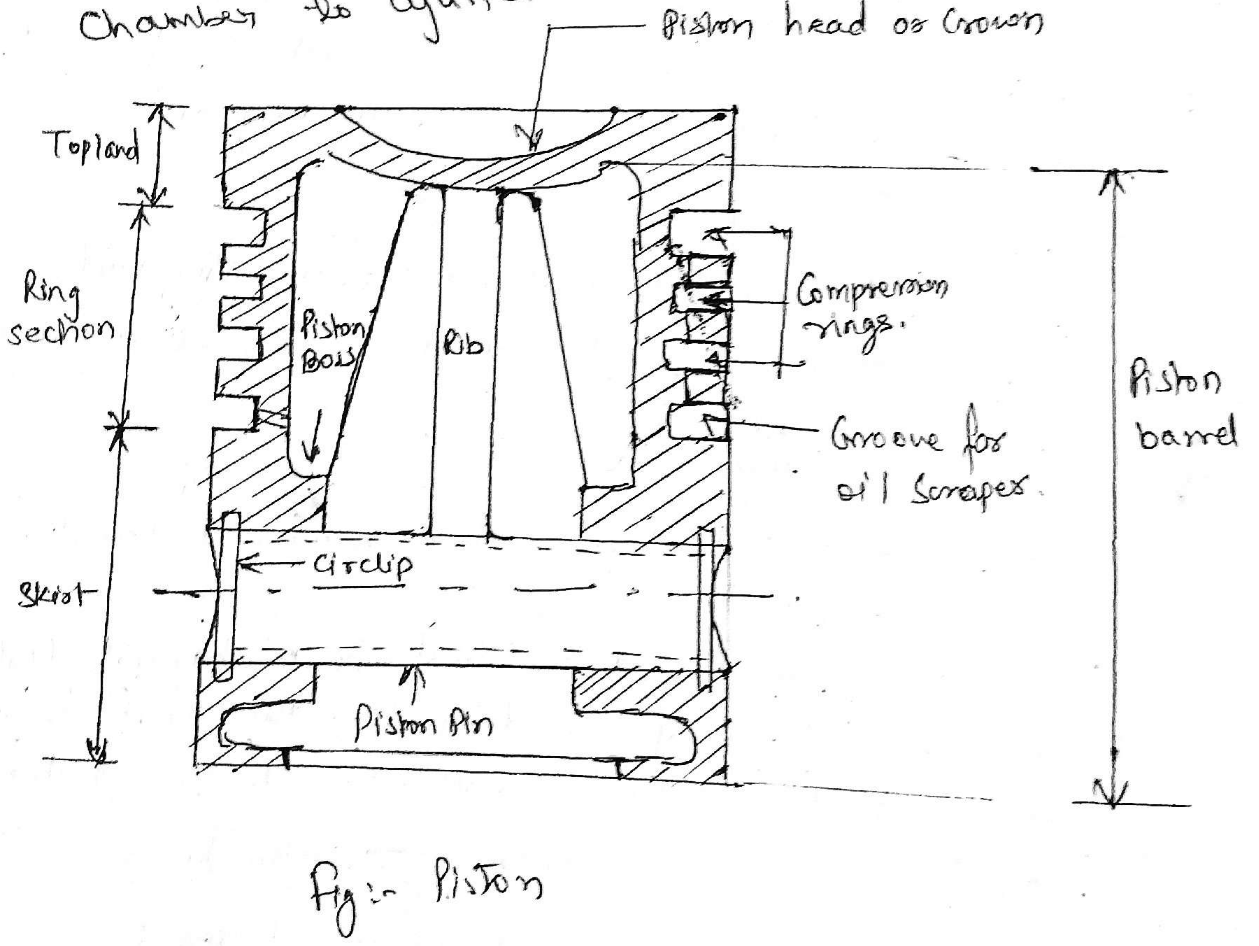


Fig:- Piston

... of ... Shells

Trunk type of Piston is used in I.C. Engine

* Parts:-

1. Piston Head or Crown:-
   - Top portion of piston which withstand gas pressure
   - Flat, concave or convex shapes.

2. Piston rings:
   - They act as seal & prevent leakage of gas.
   - Also called compression rings.

3. Oil Scrapper Ring:
   - It prevent leakage of lubricating oil into combustion chamber.

4. Piston Skirt:
   - lower part of piston below piston rings.
   - It acts as bearing surface for side thrust ~~ext~~ exerted by connecting rod.

5. Piston Pin:-
   - connects piston to connecting rod.
   - also called gudgeon pin or wrist pin.

* Design requirements:-

1. Strength to withstand force of combustion & Inertia forces of reciprocating parts
2. Strength to withstand thermal & mechanical distortions.
3. Adequate Capacity to dissipate the heat from the crown to the cylinder wall through the piston ring & skirt.
4. Minimum weight to reduce Inertia forces
5. Sufficient leakage — To avoid passing of gases / To avoid Passing of lubricating oil
6. It Should have Sufficient bearing area to take side Thrust & prevent undue wear.

(7) It should have noiseless operation
(8) It should provide adequate support for piston pin, which connects small end of connecting rod.

Piston materials :- requirements :- (1) high strength (2) resistance to thermal & mechanical distortion (3) High Heat transfer rate

Commonly used materials :- Cast Iron, Cast steel, forged steel, Cast alluminium alloys & forged aluminium alloy.

Advantages of Alluminium alloy over Cast Iron :-

(i) Thermal Conductivity of ~~all~~ aluminium alloy is approximately three times that of C.I.

Therefore ~~alloy~~ alluminium alloy piston has less variation in temp. from crown to piston rings.

(ii) $\rho_{\text{alluminum alloy}} \approx \frac{1}{3} \rho_{CI}$

hence alluminium alloy are light wt.

Advantage of C.I over alluminium alloy :-

(i) C.I have high strength compared to Alluminum alloy,

temp. ↑ strength ↓ — alluminium alloy

hence due to high strength thin section of C.I can be made

(ii) wear Strength of C.I is more.

(iii) $\left(\text{Coeff. of thermal expansion}\right)_{\text{alluminum}} \approx 2 \left(\text{Coeff. of thermal expansion}\right)_{C.I}$

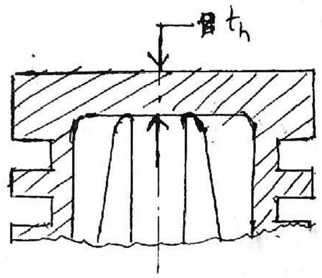
Therefore alluminium alloy piston needs more Clearance between Cylinder walls & piston rings.

C.I piston — used for moderate speed below 6m/s	Alluminum alloy piston are used for high speed above 6m/s.

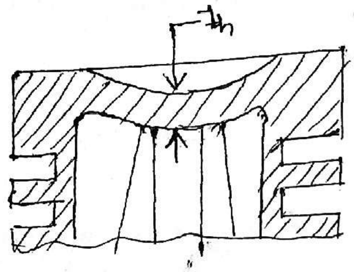
Thickness of Piston Head

Piston heads [Selection of head is based upon the required volume for combustion chamber and types of valves]

Flat type



Cup type



There are two criteria for calculating Piston heads:-

- strength
- thermal (heat dissipation)

① On strength Basis

— Piston head is treated as flat circular plate of uniform thickness fixed at outer edge & subjected to uniformly distributed gas pressure (P_{max}) over entire Surface area.

— According to grashoff's formula,

thickness of piston head (t_h)

$$\boxed{t_h = D\sqrt{\frac{3}{16}\frac{P_{max}}{\sigma_b}}}$$

where

t_h = thickness of piston head (mm)

D = Diameter of bore (mm)

P_{max} = explosion pressure or max. gas pressure (Mpa)

σ_b = Permissible bending Stress (N/mm^2)

Note :-

① bending stress (σ_b) is allowable tensile stress (σ_t)

since ~~bearing mater~~ piston head material is brittle

$$\boxed{\sigma_b = \sigma_t = \frac{S_{ut}}{f_s}}$$ — As head mtrl. is brittle.

② If S_{ut} & f_s are not specified.

$(\sigma_b)_{allow.}$ for C.I = 35 to 40 N/mm^2

$(\sigma_b)_{allow.}$ for Al. Alloy = 50 to 90 N/mm^2.

③ P_{max} may be 8 Mpa.

- Average value of P_{max} is 4 to 5 Mpa.

or

- Empirical formula recommended by Held & Favary.

for piston ~~the~~ head thickness.

$$\boxed{t_h = 0.032D + 1.5 \text{ mm}}$$

② On basis of heat dissipation

Piston head absorbs heat during combustion of fuel $\xrightarrow{\text{transmit}}$ Cylinder wall

In order to transmit heat quickly, it should have sufficient thickness.

on basis of heat dissipation, thickness of Piston head.

$$\boxed{t_h = \left[\frac{H}{12.56K(T_c - T_e)}\right] \times 10^3}$$

t_h = thickness of piston head ~~(mm)~~ (mm)

H = Amount of heat conducted through piston head (W)

K = thermal conductivity factor (W/m/°C)

T_c = temp. at the center of piston head (°C)

T_e = " " edges " " (°C)

* Notes:-

(1) values of k,

For grey C-I, $k = 46.6$ W/m/°C

for Alluminum alloy, $k = 175$ W/m/°C

(2) values of $(T_c - T_e)$

For grey C-I, $(T_c - T_e) = 220°C$

for Alluminum alloy, $(T_c - T_e) = 75°C$.

(3) Amount ~~head~~ heat conducted through piston head

$$H = [C \times HCV \times m \times BP] \times 10^3$$

where

HCV = higher calorific value of fuel (KJ/Kg)

HCV values,

For diesel, $HCV = 44 \times 10^3$ KJ/Kg

for Petrol, $HCV = 47 \times 10^3$ KJ/Kg.

$BSFC_{(m)} = \frac{\dot{m}_f}{B.P.}$

Energy of fuel,
$H = C \times [\dot{m}_f \cdot C_v]$

So in the empirical relation m is BSFC.

m = mass of fuel used per brake power per second
(Kg / KW / s)

Average Consumption of fuel in diesel Engine is 0.24 to 0.30 kg / kw / h.

$$m = \frac{0.24 \text{ to } 0.30}{60 \times 60} \text{ kg / kw / s.}$$

BP = Brake power of engine per cylinder (kw)

C = ratio of heat absorbed by the piston to the total heat developed in the cylinder.
= 5% or 0.05.

* **Piston Ribs & Cup :-**

The piston head is provided with number of ribs for following ~~reason~~ reasons :-

1. Ribs Strengthen piston head against gas pressure.
Ribs Strengthen piston head & distortion of piston head.
↑ rigidity of head & ↓ distortion of piston head.

2. transmit large portion of heat from piston head to piston rings, Hence reduces ~~to~~ large temp. difference between centre & edges.

3. The side thrust Created by obliquity of connecting rod is transmitted to the piston at piston pin.

Side thrust due to Obliquity of connecting rod → Piston Pin → Piston → Skirt → Cylinder wall

The stiffening ribs provided at Centre of boss and extending around skirt, distributes the side thrust more uniformly and prevents distortion of the skirt.

Guidelines for ribs :-

① When th thickness of Piston

$t_h \leq 6mm$ — no ~~and~~ ribs required.

$t_h > 6mm$ — (Provide ribs)

② No. of ribs = 4 to 6.

③ thickness of ribs is given by.

$$t_R = \left(\frac{t_h}{3}\right) \text{ to } \left(\frac{t_h}{2}\right)$$

where, t_R = thickness of ribs.

t_h = —"— of Piston head.

* Cup {Piston cup}

A cup provides additional space for combustion of fuel.

- Provision of cup at top on piston head depends upon volume of combustion chamber.

- ~~volume~~

- Arrangement of valves - if Inlet & exhaust valves open & close at angle Near TDC then there is possibility of either strike the piston top.

cup provided depends upon the

① volume

② Arrangement of valves.

Guideline for cup :-

(1) when $\frac{l}{D} \leq 1.5$ then cup is at top.

(2) when $\frac{l}{D} > 1.5$ — No cup required.

(3) $R_C = 0.7D$

where,

R_C = Radius of cup

D = Bore Diameter.

Piston Rings:-

Piston rings

- Compression ring
- Scraper rings.

Compression ring

- The main function is to provide seal between cylinder wall & piston
- Prevents leakage of ~~gas~~ Combustion gases from combustion chamber.
- Transfer heat from piston head to cylinder wall.
- It absorbs fluctuations in side thrust.

Scraper rings.

- They are also called oil control rings.
- They provide proper lubrication & reduces frictional losses.
- at upward stroke, oil scrapper ring provide sufficient lubricating.
- During downward stroke, it removes ~~the~~ excess oil. and sends back to crankcase.
- It prevents leakage of oil to combustion chamber.

* Guidelines for design of Piston rings :-

(1) Material :- Grey Cast Iron — Commonly used
alloy Cast Iron — Sometimes used

(i) It has excellent wear resistance
(ii) It retains spring characteristics at high temp

(ii) **No. of Piston rings:-**

a) There is no Specific rules for No. of ~~rings~~ of Compression rings.

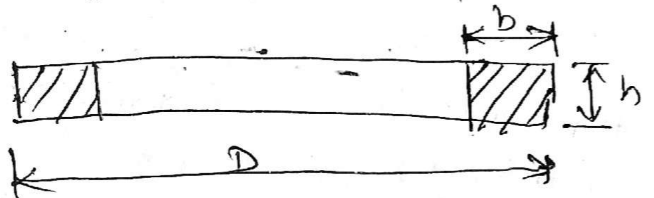
b) In automobile Engine — 3 to 4 compression rings

c) In stationary Diesel Engine — 5 to 7 compression rings

d) The oil Scrapper ring - 1 to 3.

(iii) **Dimensions of Cross - Section:**

Ring has rectangular Cross-section



$$\boxed{b = D\sqrt{\frac{3P_W}{\sigma_t}}}$$

where,

b = Radial width. of ring (mm)

P_W = allowable radial pressure on cylinder wall. (N/mm²)

σ_t = Permissible tensile stress of ring material (N/mm²)

(1) P_W = 0.025 Mpa to 0.042 Mpa.

(2) Permissible σ_t = 85 to 110 N/mm².

(3) h = 0.7b to b , $(h)_{min} = \left(\frac{D}{10Z}\right)$ Z → no of rings.

— It is prefered to provide more No. of ~~rings~~ thin rings than a Small No. of thick rings.

Advantages of more Number of rings :-

(a) Reduce frictional loss & wear of surface.

(b) It is gives better Sealing action.

(c) Thin rings occupy less piston length.

(d) Provide better heat transfer from piston top to the cylinder.

(iv) Gap between free Ends :-

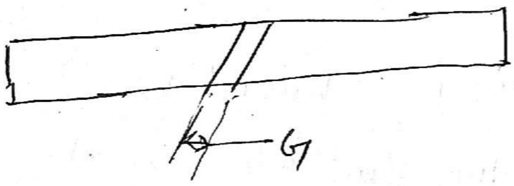
(a) $(\text{Diameter})_{ring} > (\text{Diameter})_{piston\ bore}$.

(b) Ring is cut slightly ~~diagno~~ diagonally.

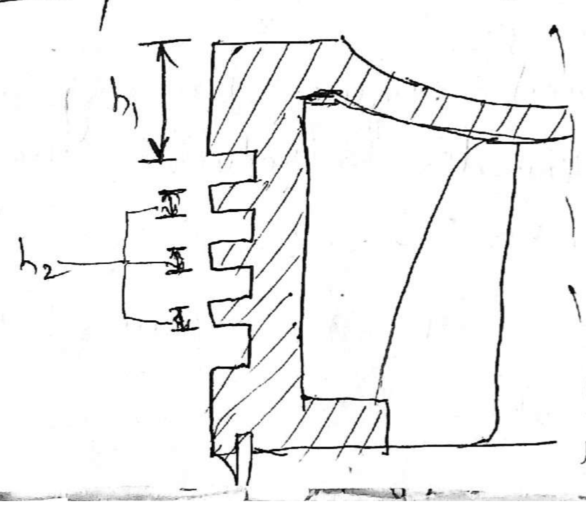
Gap between free ends of rings is as follows,

$G = 3.5b$ to $4b$ (before assembly)

$G = 0.002D$ to $0.004D$ (After Assembly in cylinder)



(v) Width of Top land & Ring Land :-



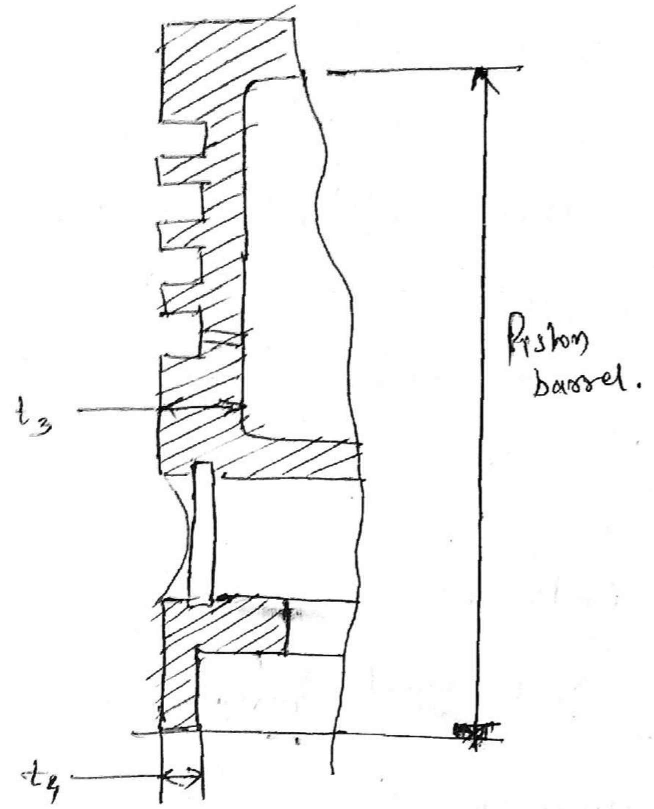
— Distance from top of piston to ring grove is called top Land (h_1).

$$\boxed{h_1 = t_h \text{ to } (1.2 t_h)}$$

— Distance between two Consecutive grooves is called as width of ring lands.

$$\boxed{h_2 = 0.75h \text{ to } h}$$

* Piston Barrel



Thickness of Piston barell at top land,

$$\boxed{t_3 = (0.03D + b + 4.9)}$$

where
b = radial width of ring.

Thickness of piston barrel at lower or open end.

$$\boxed{t_4 = (0.25\, t_3) \text{ to } 0.35\, t_3}$$

* Piston Skirt :-

- The cylindrical Portion ~~between~~ of Piston between the last scrapper ring & the opened end is called the piston skirt.

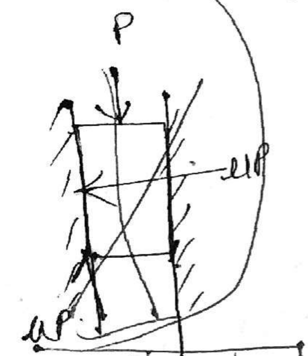
- It acts as the bearing surface for side thrust.

- The length of skirt should be such that the bearing pressure due to side thrust is restricted to 0.25 Mpa on the projected area.

- In high speed engine, bearing pressure upto 0.5 Mpa is allowed to reduced the wt. of reciprocating piston

- Maximum side thrust occurs during expansion stroke.

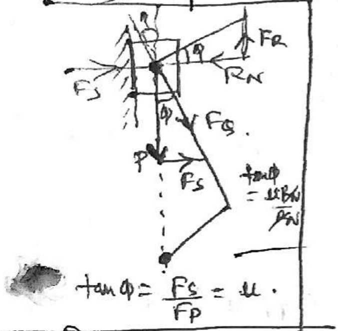
- Maximum gas force on Piston head $= \left(\frac{\pi D^2}{4}\right) P_{max}$.



$$\boxed{\text{Side thrust} = \mu \left(\frac{\pi D^2}{4}\right) P_{max}} \quad \text{——} (1)$$

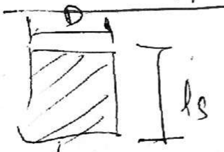
where,

μ = Coeff. of friction = 0.1



It is also given by,

$$\boxed{\text{Side thrust} = P_b D l_s} \quad \text{——} (2)$$



Projected area.

where,

P_b = allowable bearing pressure (Mpa)

l_s = length of skirt (mm)

— Equating equtn (1) & (2).

$$\mu\left(\frac{\pi}{4} D^2\right) P_{max} = P_b \cdot D l_s.$$

$$\boxed{0.1 \left(\frac{\pi}{4} D^2\right) P_{max} = P_b \cdot D l_s}$$

— The length of skirt is as follow:-

$$\boxed{l_s = (0.65 D) \text{ to } (0.8 D)}$$

- The total length of piston is given by,

L = Top land + length of ring section + length of skirt.

Empirical relation.

length of Piston. $\boxed{L = D \text{ to } (1.5D)}$

* Piston Pin

- It is also called gudgeon pin or wrist pin.
- It connects the piston to the connecting rod.
- It is made of hollow circular cross-section to reduce its weight.
- It is often tapered at inside & smallest diameter is at the centre of pin.
- The piston pin passes through bosses which are inserted in the pin & bearing bush inside ~~the~~ the small end of connecting rod.
- The end movement of piston pin is restricted by circlips.
- ~~There~~ There are two types of connections between ~~pin~~ piston pin & small end of connecting rod.

  - Full floating → Piston pin is free to turn both in bush ~~and~~ as well as boss.
  - Semi-floating → It is either free to turn in piston boss only or Either free to turn in bush of connecting rod.

There are two Criteria for design of piston Pin :-

(1) Bearing Consideration

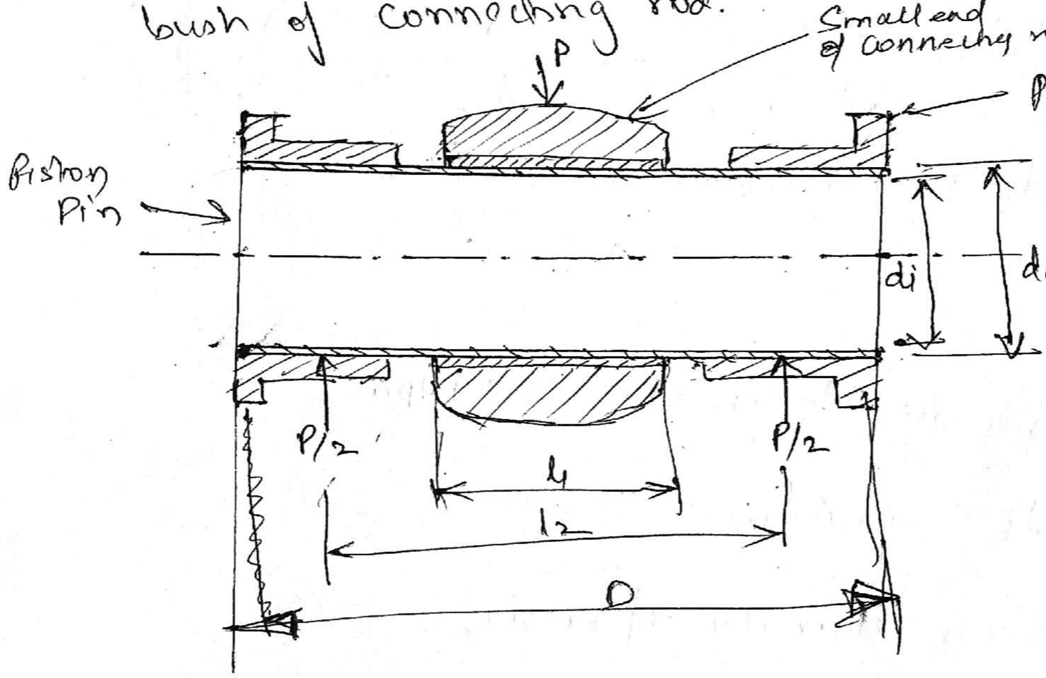
(2) Bending Consideration

Boss → It is a protruding feature on a workpiece. Boss is a pocket on Piston

Bush :- Bush is a mechanical fixing between two Possibly moving parts.

(1) Bearing Consideration

Since ~~pis~~ piston pin is partly in Contact with piston bosses and partly with bush of connecting rod.



— It is assumed that the length of pin in connecting rod bush is 45% of Piston Diameter.

$$\boxed{l_1 = 0.45D}$$

— The outer diameter of ~~pin~~ piston pin (d_o) is determined by equating the force acting on the pin and the resisting bearing force offered by piston pin.

Force on Piston $= P_{max} \cdot \left(\frac{\pi}{4} D^2\right)$ —— ①

Resisting force $= (P_b)_1 \times d_o \times l_1$ —— ②

so equating both ① & ②

$$P_{max}\left(\frac{\pi}{4} D^2\right) = (P_b)_1 \times d_o \times l_1$$

where,

$(P_b)_1$ = bearing pressure at the bush of small end connecting rod.

(d_o) = outer diameter of pin.

Note :-

① $(P_b)_1$ is taken as 25Mpa

② $d_i = 0.6 d_o$

③ Mean diameter of bosses,

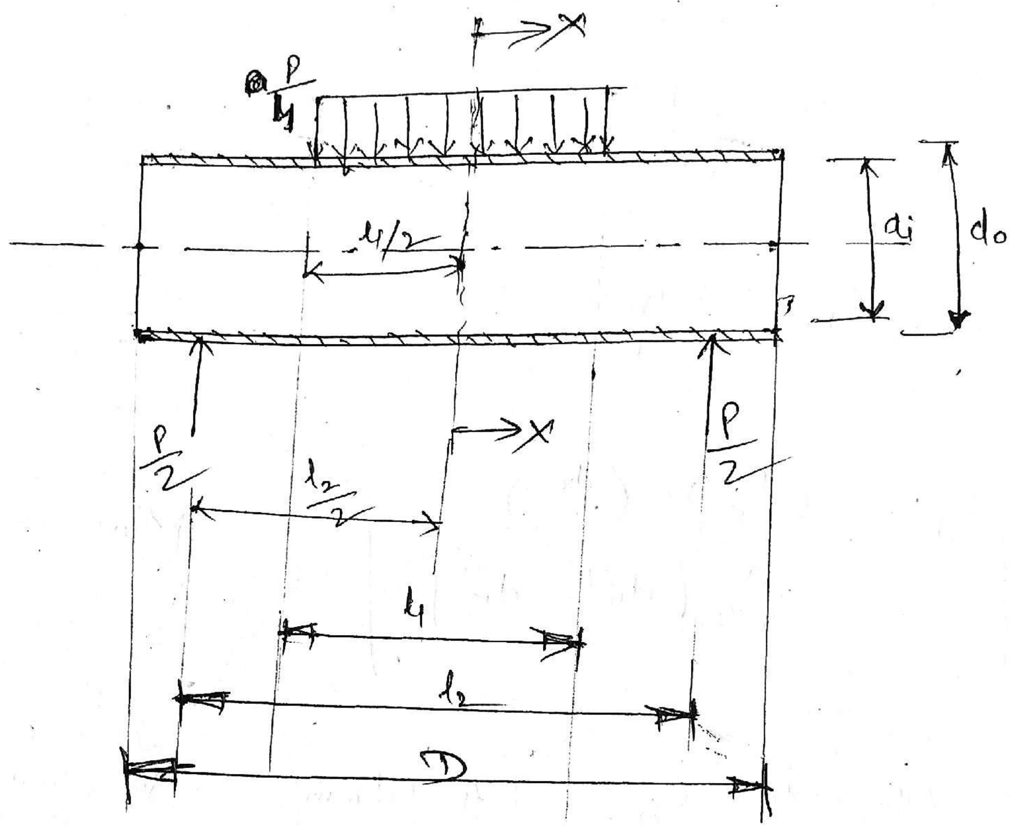
if Grey C.I

Mean diameter $= 1.4 d_o$

if Alluminium alloy,

Mean diameter $= 1.5 d_o$

(2) Bending Consideration



B.M at section XX i.e @ center of Pin.

$$M_b = \left\{\left(\frac{P}{2}\right)\times\left(\frac{l_2}{2}\right)\right\} - \left\{\left(\frac{P}{l_1}\times\frac{l_1}{2}\right)\times\left(\frac{l_1}{4}\right)\right\}$$

$$M_b = \left\{\frac{Pl_2}{4}\right\} - \left\{\frac{Pl_1}{8}\right\}$$

$$l_2 = \frac{D}{2} + \frac{l_1}{2}$$ —— Average value.

$$l_2 = \frac{D + l_1}{2}$$

$$M_b = \left\{\frac{P}{4}\cdot\frac{(D+l_1)}{2}\right\} - \left\{\frac{Pl_1}{8}\right\}$$

$$M_b = \frac{PD}{8} + \frac{Pl_1}{8} - \frac{Pl_1}{8}$$

$$\boxed{M_b = \frac{PD}{8}}$$

$$I = \frac{\pi}{64}\left[d_o^4 - d_i^4\right]$$

$$y = \left(\frac{d_o}{2}\right)$$

$$\sigma_b = \frac{M_b y}{I}$$

$$\boxed{\sigma_b = \frac{\left(\frac{PD}{8}\right)\cdot\left(\frac{d_o}{2}\right)}{\frac{\pi}{64}\left[d_o^4 - d_i^4\right]}}$$ — Maximum bending stress in piston pin

Note → Allowable $\sigma_b = 84\ N/mm^2$ — For case hardened Carbon steel

$\sigma_b = 140\ N/mm^2$ — For heat treated alloy steel.

(3) Piston clearence

— Clearence between piston & cylinder liner is provided accordingly to thermal expansion & distortion under load

— If clearence is insufficient — Piston seizure occurs

— If clearance is more — Piston slap occurs. hence more Noise

Magnitude of Piston clearance depending upon piston Dia & engine type is = 0.0375 to 0.1875 mm.

$$\text{In} \begin{pmatrix} \text{Piston of} \\ \text{Alluminium alloy} \\ \text{Clearence} \end{pmatrix} = 2 \begin{pmatrix} \text{C.I piston} \\ \text{Clearence} \end{pmatrix}$$

- The clearence is less when proper cooling is provided. hence less expansion

V.B Bhandari Problems

25.6 . 4- stroke diesel Engine.

$D = 250mm$

$L = 300mm$

$N = 600rpm$

P_m (Indicated mean eff. pr.) = 0.6 Mpa.

$\eta_{mech} = 80\%$

$P_{max} = 4Mpa$

Fuel consumption = 0.25 Kg | kwh.

$C_f = 44000 kJ/kg$

Assume 5% heat transmitted to piston

Piston mtrl. Grey C.I.

$S_{ut} = 200 N/mm^2$

$K = 46.6 \ W/m/°C$

$f_s = 5$.

$T_c - T_e = 220°C$.

(1) $t_h = ?$ — by strength consider

(2) $t_h = ?$ — by thermal consid.

(3) which criteria decides Piston head thickness.

(4) ~~—~~ ribs required?

(5) No. of ribs & thickness of ribs.

(6) Cup is required?

(7) if so, radius of cup?

Ans (i) Thickness of piston head by strength Consideration.

Permissible tensile strength.

$$\sigma_t = \frac{S_{ut}}{fs} = \frac{200}{5} = 40\ N/mm^2$$

$$\sigma_b = 40\ N/mm^2$$

$$t_h = D\sqrt{\frac{3}{16}\,\frac{P_{max}}{\sigma_b}}$$

$$t_h = 250\sqrt{\frac{3}{16}\times\frac{4}{40}}$$

$$t_h = 34.23 \text{ or } 35\ mm$$

$$\boxed{t_h = 35\ mm}$$ by strength

1m = 10^3 mm

1mm = 10^{-3} m

(ii) Thickness by thermal ~~conductivity~~ consideration,

$$I.P = \frac{P_m\,L\,A\,n\,K}{60\times10^3}$$

1MPa = 1 N/mm²

$$= \frac{(0.6\times10^6)\times(0.300)\times\left(\frac{\pi}{4}(0.25)^2\right)\times\left(\frac{600}{2}\right)(1)}{60\times10^3}$$

$$\boxed{I.P = 44.18\ Kw}$$

$$\eta_{mech.} = \frac{B.P}{IP}$$

$$0.80 = \frac{B.P}{44.18}$$

$$\boxed{B.P = 35.34 \text{ kW}}$$

$$BSFC = \frac{\dot{m}_f}{B.P}$$

$$\frac{0.25}{60 \times 60} = \frac{\dot{m}_f}{35.34}$$

$$\frac{kg}{kW\,h} = \frac{kg/s}{kJ/s}$$

$$\dot{m}_f = \frac{0.25 \times 35.34}{60 \times 60}$$

$$\boxed{\dot{m}_f = 0.002454 \text{ kg/s.}}$$

$$\frac{kg}{s} \times \frac{kJ}{kg} \times K$$

Energy of fuel,

$$H = C \times \dot{m}_f \cdot C_v$$

$$= 0.05 \times 0.002454 \times 44000 \times 10^3$$

$$\boxed{H = 5398.82 \text{ W}}$$

$$t_h = \left[\frac{H}{12.56\,K\,(T_c - T_e)}\right] 10^3$$

$$= \left[\frac{5398.82}{12.56 \times 46.6\,(220)}\right] \times 10^3$$

$$t_h = 41.92 \text{ mm} \quad \text{or} \quad 42 \text{ mm}.$$

$\boxed{t_h = 42\text{ mm}}$

(iii) Criteria to decide

from both i.e by strength consideration

$\boxed{t_h = 35\text{ mm}}$

and by thermal consideration

$\boxed{t_h = 42\text{ mm}}$

hence we will select maximum value for safety.

$\boxed{t_h = 42\text{ mm}}$

(iv). Requirement of ribs.

As $t_h > 6\text{mm}$ hence ribs required

(v). No. of ribs.

It is assumed 4 no. of ribs.

thickness of ribs,

$$t_R = \frac{t_h}{3} \text{ to } \frac{t_h}{2}$$

$$= \frac{42}{3} \text{ to } \frac{42}{2}$$

$$t_R = 14 \text{ to } 21$$

$\boxed{t_R = 18\text{ mm}}$

(vi) Requirement of cup

$$\frac{l}{D} = \frac{300}{250} = 1.2$$

if $\left(\frac{l}{D}\right) \leq 1.5$ — ~~cup~~ hence Cup required.

(vii) Radius of cup

radius of cup = 0.7 D

= 0.7 × 250

$\boxed{\text{radius of cup} = 175 \text{ mm}}$

Q5-7

4-Stroke diesel Engine, Grey C.I. ~~piston piston~~ Piston

$D = 250 mm$

$\sigma_t = 100 N/mm^2$

P_w ~~max~~ = 0.03 Mpa — Allowable radial pressure on cylinder wall

$t_h = 42 mm$

No. of piston rings = 4

(i) Radial width of Piston rings.

(ii) Axial thickness of Piston rings

(iii) Gap between free end of Piston rings before assembly.

(iv) Gap between — " — " — " — after — " —

(v) width of Top land

(vi) width of ring groove

(vii) thickness of Piston barrel

(viii) thickness of barrel at open end.

A (i) Radial width of piston rings:-

$$b = D\sqrt{\frac{3P_w}{\sigma_t}}$$

$$= 250\sqrt{\frac{3\times0.03}{100}}$$

$$\boxed{b = 7.5\,mm}$$

(ii) Axial thickness of Piston ring

$h = 0.7b$ to b

$= 0.7(7.5)$ to 7.5

$h = 5.25$ to 7.5

$$\boxed{h = 6.375\,mm}$$

Also,

$$h_{min} = \left(\frac{D}{10z}\right)$$

$$= \left(\frac{250}{10\times4}\right)$$

$$\boxed{h_{min} = 6.25\,mm}$$

$h > h_{min}$

$$\boxed{h = 6.375\,mm} \text{ or } 7\,mm$$

(iii) Gap between free ends of Piston rings before assembly

$G = 3.5b$ to $4b$

$= 3.5(7.5)$ to $4(7.5)$

$= 26.25$ to 30

$$\boxed{G = 28.125\,mm}$$

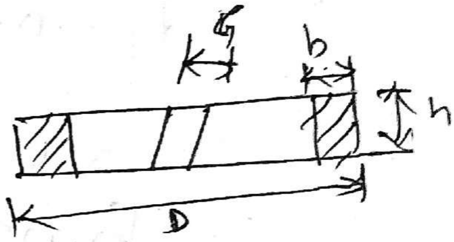
$$\boxed{G_1 = 28\,mm}$$

Gap between free ends of piston rings after assembly

$$G_2 = 0.002D \text{ to } 0.004D$$

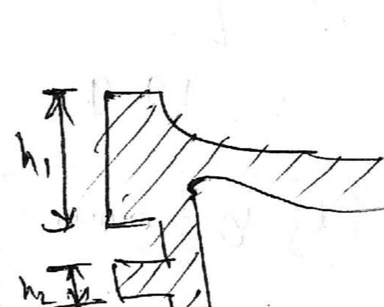
$$= 0.002(250) \text{ to } 0.004(250)$$

$$= 0.5 \text{ to } 1\,mm.$$



$$\boxed{G_2 = 0.75\,mm}$$

(V) width of top land



$$h_1 = t_h \text{ to } (1.2\,t_h)$$

$$= 42 \text{ to } [1.2(42)]$$

$$= 42 \text{ to } 50.4$$

$$= 46.2$$

$$\boxed{h_1 = 46\,mm}$$

~~(VI) width of ring grooves~~

~~$h_2 = 0.75h$ to h~~

~~$= 0.7$~~

(VI) width of ring grooves

$$h_2 = 0.75h \text{ to } h$$

$$= 0.75(7) \text{ to } (7)$$

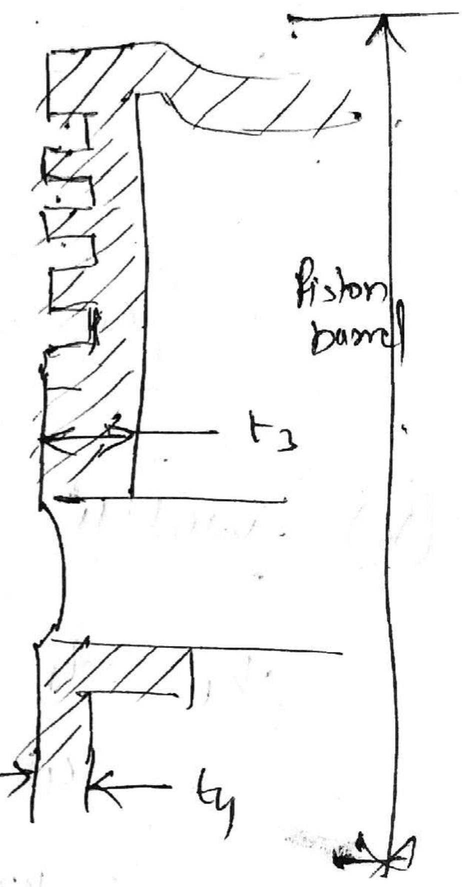
$= 287.35\,mm$

$h_2 = 5.25$ to 7

$h_2 = 6.125$ mm

$\boxed{h_2 = 6 \text{ mm}}$

(vii) Thickness of piston barrel.



$t_3 = 0.03D + b + 4.9$

$= \cancel{0.03D}$

$= [0.03(250) + 7.5 + 4.9]$

$= 19.9$

$\boxed{t_3 \approx 20 \text{ mm}}$

$t_4 = 0.25 t_3$ to $0.35 t_3$

$= 0.25(20)$ to $0.35(20)$

$= 5$ to 7

$\boxed{t_4 = 6 \text{ mm}}$

25.

$D = 250mm$

$P_{max} = 4Mpa$

$P_b = 0.4Mpa$

$$\frac{\text{Side thrust}}{\text{Maximum load on piston}} = 0.1$$

$$\frac{\text{Side thrust}}{P_{max} \cdot \frac{\pi}{4}(D^2)} = 0.1$$

hence $\mu = 0.1$

$$\text{Side thrust} = (0.1)\left(P_{max} \frac{\pi}{4} D^2\right)$$

$h_1 = 45mm$

$h_2 = 6mm$

$Z = 4$

$h = 7mm$

length of skirt (l_s) = ?

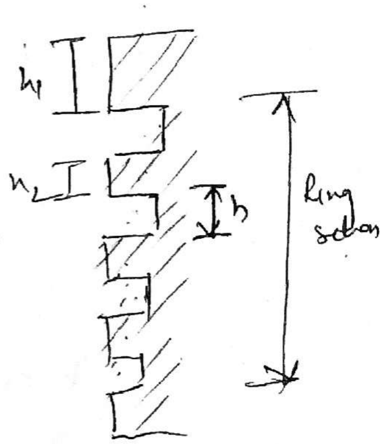
length of piston (L) = ?

Side thrust = Bearing load

$$(0.1)(P_{max})\left(\frac{\pi}{4}D^2\right) = P_b \, l_s \cdot D$$

$$(0.1)(4)\left[\frac{\pi}{4}(250)^2\right] = (0.4)(250) \cdot l_s$$

$$\boxed{l_s = 196.35mm}$$



length of ring section $= 3h_2 + 4h$

$= 3(6) + 4(7)$

$= 18 + 28$

$$\boxed{\text{length of ring section} = 46mm}$$

Total length of Piston (L)

$L = h_1 + (\text{length of ring}) + l_s$

$h = 5.25$ to 7

$\boxed{L = 288\,mm}$

According to empirical ~~relations~~ relationship.

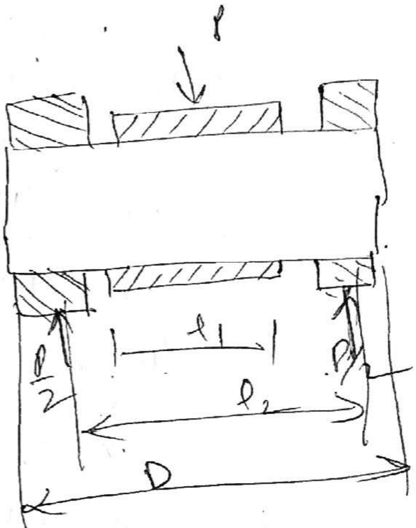
$L = D$ to $1.5D$

$= 250$ to 375

~~275 mm~~

$\boxed{\therefore D < L > 1.5D}$ hence satisfies.

259. 4-stroke diesel engine



$D = 250\,mm$

$P_{max} = 4\,Mpa$

$(P_b)_{pin} = 15\,mpa$

$l_1 = 0.45D$

$\frac{d_i}{d_o} = 0.6$

$(d_m)_{boss} = 1.4 \times d_o$

$(\sigma_b)_{pin} = 84\,N/mm^2$

(i) $d_o = ?$

(ii) $d_i = ?$

(iii) $(d_m)_{boss} = ?$

(iv) Check bending stress.

Ans

Force on Piston, $= P_{max}\left(\frac{\pi}{4}D^2\right)$

Force on Piston $= (4)\left[\frac{\pi}{4}(250)^2\right]$

Force on Piston $= 196349.54\,N$

Force on piston = bearing load on pin eay

$196349.54 = (P_b)_{pin} \cdot l_1\, d_o$

$196349.54 = (15) \cdot (0.45D)\, d_o$

$\boxed{d_o = 116.36}$ or $\boxed{118\,mm}$

(ii) $d_i = 0.6(d_o)$

$\boxed{d_i = 69.82}$ or $\boxed{d_i = 70 \text{ mm}}$

(iii) $(d_m)_{boss} = 1.4 d_o$

$= 1.4(116.36)$

$\boxed{(d_m)_{boss} = 162.91}$ $\boxed{(d_m)_{boss} = 165 \text{ mm}}$

(iv) check for bending stress.

$M_b = \frac{PD}{8}$

$= (196349.54) \cdot \frac{(250)}{8}$

$\boxed{M_b = 6135.92 \times 10^3 \text{ N-mm}}$

$I = \frac{\pi}{64}(d_o^4 - d_i^4)$

$= \frac{\pi}{64}((118)^4 - (70)^4) = 8338.36 \times 10^3$

$\boxed{I = (8338.36 \times 10^3) \text{ mm}^4}$

$y = \frac{d_o}{2} = \frac{118}{2} = 59 \text{ mm}$

$\boxed{y = 59 \text{ mm}}$

$(\sigma_b)_{design} = \frac{M_b y}{I}$

$= \frac{6135.92 \times 10^3 \times 59}{8338.36 \times 10^3} = 43.42 \text{ N/mm}^2$

$(\sigma_b)_{design} < (\sigma_b)_{\cancel{design}\ allowed}$ i.e 84 N/mm^2 here Safe.

25.10 4-stroke diesel Engine Mtrl. C.I ($\sigma_b = \sigma_t = 40 N/mm^2$)

$D = 200 mm$

$L = 450 mm$

$N = 300 rpm$

$P_m = 0.85 MPa$

$P_{max} = 5 MPa$

$BSFC = 0.30 kg/kwh$

$C_v = 44000$ ~~kg~~ kJ/kg

Assumption $\sigma_b = 40 N/mm^2$

$\eta_{mech} = 0.8$

5% heat absorbed by Piston

let $K = 46.6 W/m/°C$

$(T_c - T_e) = 220°C$

No. of ribs = 4

allowable radial pr. on cylinder walls $= 0.035 MPa$

Compression ring = 3

Oil ring = 1

σ_t of rings = $90 N/mm^2$

allowable bearing Pressure for Skirt Portion of Piston = 0.45 MPa

A (1) Piston head

t_h by strength consideration

$$t_h = D\sqrt{\frac{3}{16}\frac{P_{max}}{\sigma_b}}$$

$$= 300\sqrt{\frac{3}{16}\left(\frac{5}{40}\right)}$$

$$\boxed{t_h = 45.93 mm}$$

By thermal consideration

$$I.P = \frac{P_m LAN}{60 \times 10^3}$$

$$= \frac{(0.85\times10^6)(0.45)\left[\frac{\pi}{4}(0.3)^2\right]\left[\frac{300}{2}\right]}{60\times10^3}$$

$$\boxed{IP = 67.59 kW}$$

~~$BSFC = \frac{\dot{m}_f}{BP}$~~

$$\eta_{mech} = \frac{B.P}{I.P}$$

$$0.8 = \frac{B.P}{67.59}$$

$$\boxed{B.P = 54.07 kW}$$

$$BSFC = \frac{\dot{m}_f}{B.P}$$

$$0.30 = \frac{\dot{m}_f}{54.07}$$

$$\boxed{\dot{m}_f = 16.22 kg/hr}$$

Total Heat Energy generated by fuel.

$$Q = \dot{m}_f \cdot C_v$$

$$Q = 16.22 \times 44000$$

$$\boxed{Q = 713785.55 kJ/hr}$$

Design of Piston ...

$$Q = \frac{713785.55 \times 10^3}{3600}$$

$$\boxed{Q = 198273.76 \text{ W}}$$

Heat absorbed by piston is assumed 5%.

$$H = 0.05 \times Q$$

$$\boxed{H = 9913.68 \text{ W}}$$

$$t_h = \left[\frac{H}{12.56 K (T_c - T_e)}\right] \times 10^3$$

$$t_h = \left[\frac{9913.68}{12.56 (46.6)(220)}\right] \times 10^3$$

$$\boxed{t_h = 76.98 \text{ mm}}$$

so from both thermal & strength considerations.

$t_h = 76.98$ is selected.

So $\boxed{t_h = 77 \text{ mm}}$

(2) Radial Ribs :-

since $t_h = 77$mm so $t_h > 6$mm hence ribs are required.

Take number of ribs = 4.

$$t_R = \left(\frac{t_h}{3}\right) \text{ to } \left(\frac{t_h}{2}\right)$$

$$= \left(\frac{77}{3}\right) \text{ to } \left(\frac{77}{2}\right)$$

$$= 25.67 \text{ to } 38.5$$

$$\boxed{t_R = 30 \text{ mm}}$$

③ Requirement of Cup.

$$\frac{l}{D} = \frac{450}{300} = 1.5$$

$$\therefore \frac{l}{D} \leq 1.5 \quad \text{hence cup } \cancel{\text{requ}} \text{ required}$$

$$\text{Radius of cup} = 0.7D$$
$$= 0.7(300)$$

$$\boxed{\text{Radius of cup} = 210 \text{ mm}}$$

④ Piston rings:-

$P_W = 0.035$ Mpa

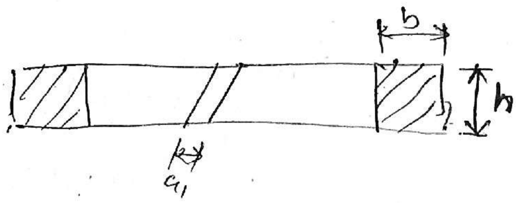
Compression rings = 3

oil rings = 1

Total rings = 4.

rings mad of C.I allowable $\sigma_t = 90 N/mm^2$

$$b = D\sqrt{\frac{3P_W}{\sigma_t}}$$



$$b = 300\sqrt{\frac{3(0.035)}{90}}$$

$$b = 10.25 \text{ mm}$$

$$\boxed{b = \cancel{10 \text{ mm}}\ 10.5 \text{ mm}}$$

$$h = (0.7b) \text{ to } b$$
$$= (0.7 \times 10.5) \text{ to } 10.5$$
$$= 7.35 \text{ to } 10.5$$

$$\boxed{h = 8 \text{ mm}}$$

$$\boxed{z = 4} \text{ — Total rings.}$$

$$h_{min} = \left(\frac{D}{10z}\right)$$

$$= \left(\frac{300}{10 \times 4}\right) \Rightarrow 7.5 \text{ mm.}$$

1.. 8 mm

Gap between Piston rings.

before of assembly

$G_1 = 3.5b$ to $4b$

$= 3.5(10.5)$ to $4(10.5)$

$= 36.75$ to 42 mm

$\boxed{G_1 = 40\text{ mm}}$ — Gap before assembly

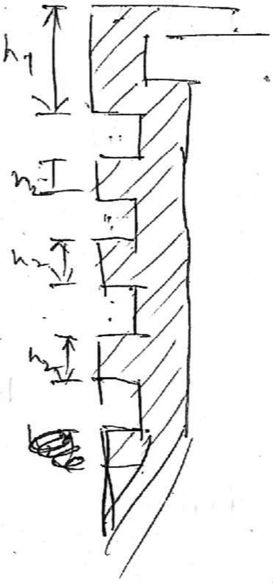
Gap after assembly

$G_2 = 0.002D$ to $0.004D$

$= 0.002(300)$ to $0.004(300)$

$= 0.6$ to 1.2

$\boxed{G_2 = 0.8\text{ mm}}$



width of top land

$h_1 = t_h$ to $1.2t_h$

$= 77$ to (1.2×77)

$= 77$ to 92.4

$\boxed{h_1 = 85\text{mm}}$

$h_2 = 0.75h$ to h

$= 0.75(8)$ to 8

$= 6$ to 8

$\boxed{h_2 = 7\text{ mm}}$

(5) Piston barrel

let allowable bearing pressure for skirt $= 0.45$ MPa

let $\mu = 0.1$

$\mu(P_{max})\left(\frac{\pi}{4}D^2\right) = P_b \cdot l_s \cdot D$

$(0.1)(5)\left(\frac{\pi}{4}(300)^2\right) = (0.45)(l_s)(300)$

$l_s = 261.8$ or $\boxed{l_s = 262\text{mm}}$

(7) Piston length

length of ring section $= 4h + 3h_2$

$= 4(8) + 3(7)$

$= 32 + 21$

$$\boxed{\text{length of ring section} = 53\,mm}$$

length of piston $= h_1 + (\text{length of ring section}) + l_3$

$= \overset{85}{\cancel{}} + 53 + 262$

Length of skirt $\boxed{L = 400\,mm}$

By Empirical relation

$L = D$ to $1.5D$

$= 300$ to $1.5(300)$

$= 300$ to 450

$D < L > 1.5D$

(8) Piston Pin :