CHAPTER

32

Internal Combustion Engine Parts

32.1 Introduction

As the name implies, the internal combustion engines (briefly written as I. C. engines) are those engines in which the combustion of fuel takes place inside the engine cylinder. The I.C. engines use either petrol or diesel as their fuel. In petrol engines (also called ***spark ignition engines*** or ***S.I engines***), the correct proportion of air and petrol is mixed in the carburettor and fed to engine cylinder where it is ignited by means of a spark produced at the spark plug. In diesel engines (also called ***compression ignition engines*** or ***C.I engines***), only air is supplied to the engine cylinder during suction stroke and it is compressed to a very high pressure, thereby raising its temperature from 600°C to 1000°C. The desired quantity of fuel (diesel) is now injected into the engine cylinder in the form of a very fine spray and gets ignited when comes in contact with the hot air.

The operating cycle of an I.C. engine may be completed either by the two strokes or four strokes of the piston. Thus, an engine which requires two strokes of the piston or one complete revolution of the crankshaft to complete the cycle, is known as ***two stroke engine.*** An engine which requires four strokes of the piston or two complete revolutions of the crankshaft to complete the cycle, is known as ***four stroke engine.***

The two stroke petrol engines are generally employed in very light vehicles such as scooters, motor cycles and three wheelers. The two stroke diesel engines are generally employed in marine propulsion.

The four stroke petrol engines are generally employed in light vehicles such as cars, jeeps and also in aeroplanes. The four stroke diesel engines are generally employed in heavy duty vehicles such as buses, trucks, tractors, diesel locomotive and in the earth moving machinery.

32.2 Principal Parts of an Engine

The principal parts of an I.C engine, as shown in Fig. 32.1 are as follows :

1. Cylinder and cylinder liner, 2. Piston, piston rings and piston pin or gudgeon pin, 3. Connecting rod with small and big end bearing, 4. Crank, crankshaft and crank pin, and 5. Valve gear mechanism.

The design of the above mentioned principal parts are discussed, in detail, in the following pages.

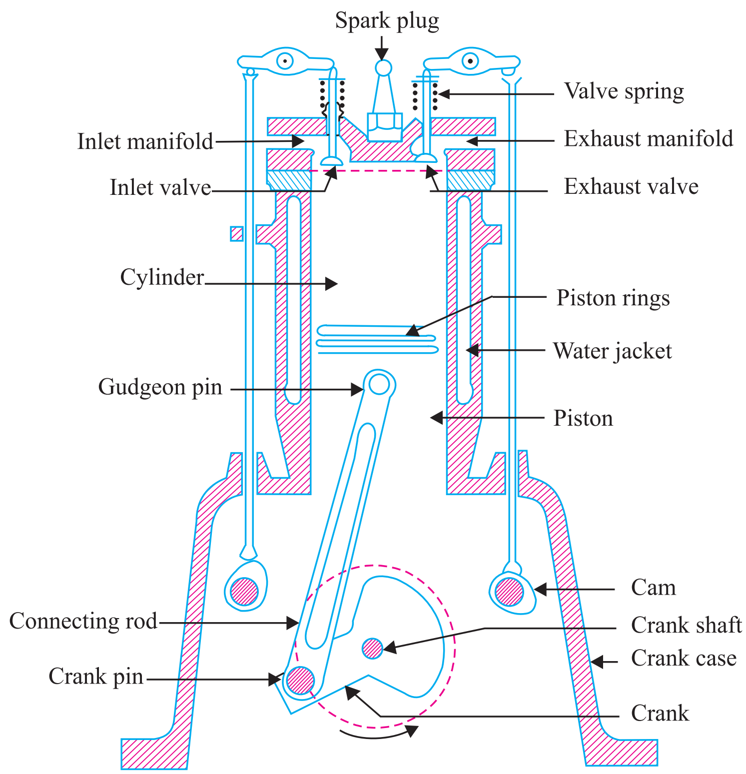
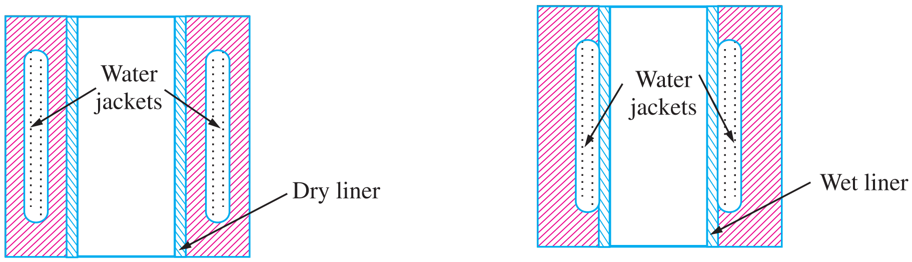


Fig. 32.1. Internal combustion engine parts.

32.3 Cylinder and Cylinder Liner

The function of a cylinder is to retain the working fluid and to guide the piston. The cylinders are usually made of cast iron or cast steel. Since the cylinder has to withstand high temperature due to the combustion of fuel, therefore, some arrangement must be provided to cool the cylinder. The single cylinder engines (such as scooters and motorcycles) are generally air cooled. They are provided with fins around the cylinder. The multi-cylinder engines (such as of cars) are provided with water jackets around the cylinders to cool it. In smaller engines. the cylinder, water jacket and the frame are made as one piece, but for all the larger engines, these parts are manufactured separately. The cylinders are provided with cylinder liners so that in case of wear, they can be easily replaced. The cylinder liners are of the following two types :

1. Dry liner, and **2.** Wet liner.



(*a*) Dry liner. (*b*) Wet liner.

Fig. 32.2. Dry and wet liner.

A cylinder liner which does not have any direct contact with the engine cooling water, is known as ***dry liner,*** as shown in Fig. 32.2 (*a*). A cylinder liner which have its outer surface in direct contact with the engine cooling water, is known as ***wet liner,*** as shown in Fig. 32.2 (*b*).

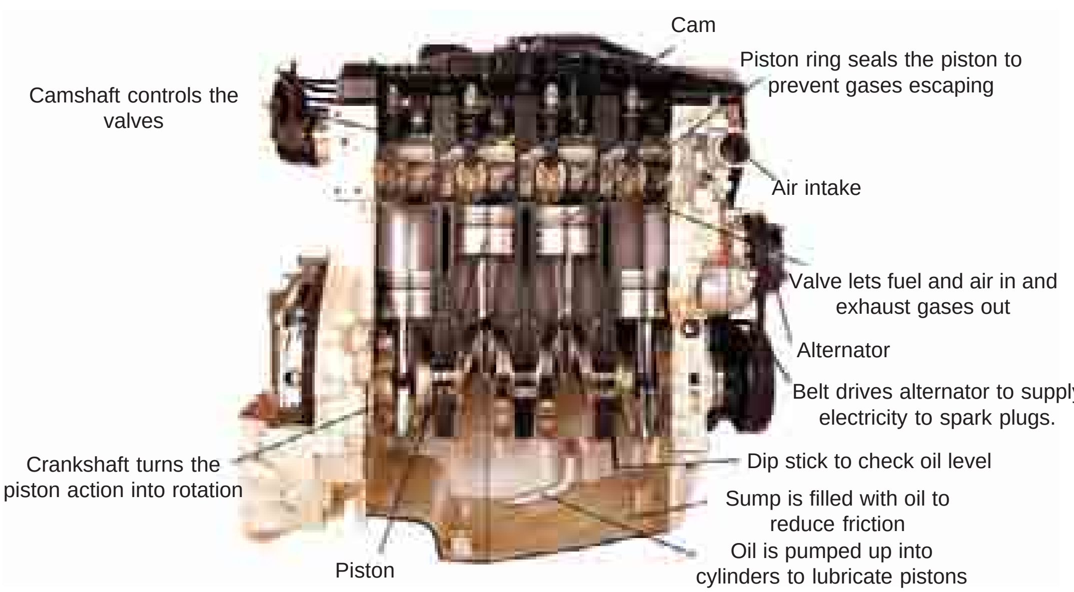
The cylinder liners are made from good quality close grained cast iron (*i.e.* pearlitic cast iron), nickel cast iron, nickel chromium cast iron. In some cases, nickel chromium cast steel with molybdenum may be used. The inner surface of the liner should be properly heat-treated in order to obtain a hard surface to reduce wear.

32.4 Design of a Cylinder

In designing a cylinder for an I. C. engine, it is required to determine the following values :

1. Thickness of the cylinder wall. The cylinder wall is subjected to gas pressure and the piston side thrust. The gas pressure produces the following two types of stresses :

(a) Longitudinal stress, and ***(b)*** Circumferential stress.



The above picture shows crankshaft, pistons and cylinder of a 4-stroke petrol engine.

Since these two stressess act at right angles to each other, therefore, the net stress in each direction is reduced.

The piston side thrust tends to bend the cylinder wall, but the stress in the wall due to side thrust is very small and hence it may be neglected.

Let D_0 = Outside diameter of the cylinder in mm,

D = Inside diameter of the cylinder in mm,

p = Maximum pressure inside the engine cylinder in N/mm^2,

t = Thickness of the cylinder wall in mm, and

$1/m$ = Poisson's ratio. It is usually taken as 0.25.

The apparent longitudinal stress is given by

$$\sigma_l = \frac{\text{Force}}{\text{Area}} = \frac{\frac{\pi}{4} \times D^2 \times p}{\frac{\pi}{4}[(D_0)^2 - D^2]} = \frac{D^2 \,.\, p}{(D_0)^2 - D^2}$$

and the apparent circumferential stresss is given by

$$\sigma_c = \frac{\text{Force}}{\text{Area}} = \frac{D \times l \times p}{2t \times l} = \frac{D \times p}{2t}$$

... (where l is the length of the cylinder and area is the projected area)

$\therefore$ Net longitudinal stress $= \sigma_l - \dfrac{\sigma_c}{m}$

and net circumferential stress $= \sigma_c - \dfrac{\sigma_l}{m}$

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

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

where p = Maximum pressure inside the cylinder in N/mm^2,

D = Inside diameter of the cylinder or cylinder bore in mm,

σ_c = Permissible circumferential or hoop stress for the cylinder material in MPa or N/mm^2. Its value may be taken from 35 MPa to 100 MPa depending upon the size and material of the cylinder.

C = Allowance for reboring.

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

Table 32.1. Allowance for reboring for I. C. engine cylinders.

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

The thickness of the cylinder wall usually varies from 4.5 mm to 25 mm or more depending upon the size of the cylinder. The thickness of the cylinder wall (t) may also be obtained from the following empirical relation, *i.e.*

$$t = 0.045\, D + 1.6 \text{ mm}$$

The other empirical relations are as follows :

Thickness of the dry liner

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

Thickness of the water jacket wall

= $0.032\ D + 1.6$ mm or $t / 3\ m$ for bigger cylinders and $3t/4$ for smaller cylinders

Water space between the outer cylinder wall and inner jacket wall

= 10 mm for a 75 mm cylinder to 75 mm for a 750 mm cylinder or $0.08\ D + 6.5$ mm

2. *Bore and length of the cylinder.* The bore (*i.e.* inner diameter) and length of the cylinder may be determined as discussed below :

Let p_m = Indicated mean effective pressure in N/mm^2,

D = Cylinder bore in mm,

A = Cross-sectional area of the cylinder in mm^2,

= $\pi D^2/4$

l = Length of stroke in metres,

N = Speed of the engine in r.p.m., and

n = Number of working strokes per min

= N, for two stroke engine

= $N/2$, for four stroke engine.

We know that the power produced inside the engine cylinder, *i.e.* indicated power,

$$I.P. = \frac{p_m \times l \times A \times n}{60} \text{ watts}$$

From this expression, the bore (D) and length of stroke (l) is determined. The length of stroke is generally taken as 1.25 D to 2D.

Since there is a clearance on both sides of the cylinder, therefore length of the cylinder is taken as 15 percent greater than the length of stroke. In other words,

Length of the cylinder, $L = 1.15 \times$ Length of stroke $= 1.15\ l$

Notes : (*a*) If the power developed at the crankshaft, *i.e.* brake power (*B. P.*) and the mechanical efficiency (η_m) of the engine is known, then

$$I.P. = \frac{B.P.}{\eta_m}$$

(*b*) The maximum gas pressure (p) may be taken as 9 to 10 times the mean effective pressure (p_m).

3. *Cylinder flange and studs.* The cylinders are cast integral with the upper half of the crankcase or they are attached to the crankcase by means of a flange with studs or bolts and nuts. The cylinder flange is integral with the cylinder and should be made thicker than the cylinder wall. The flange thickness should be taken as 1.2 t to 1.4 t, where t is the thickness of cylinder wall.

The diameter of the studs or bolts may be obtained by equating the gas load due to the maximum pressure in the cylinder to the resisting force offered by all the studs or bolts. Mathematically,

$$\frac{\pi}{4} \times D^2 . p = n_s \times \frac{\pi}{4} (d_c)^2 \sigma_t$$

where D = Cylinder bore in mm,

p = Maximum pressure in N/mm^2,

n_s = Number of studs. It may be taken as $0.01\ D + 4$ to $0.02\ D + 4$

d_c = Core or minor diameter, *i.e.* diameter at the root of the thread in mm,

σ_t = Allowable tensile stress for the material of studs or bolts in MPa or N/mm^2. It may be taken as 35 to 70 MPa.

The nominal or major diameter of the stud or bolt (d) usually lies between 0.75 t_f to t_f, where t_f is the thickness of flange. In no case, a stud or bolt less than 16 mm diameter should be used.

The distance of the flange from the centre of the hole for the stud or bolt should not be less than d + 6 mm and not more than 1.5 d, where d is the nominal diameter of the stud or bolt.

In order to make a leak proof joint, the pitch of the studs or bolts should lie between $19\sqrt{d}$ to $28.5\sqrt{d}$, where d is in mm.

4. *Cylinder head.* Usually, a separate cylinder head or cover is provided with most of the engines. It is, usually, made of box type section of considerable depth to accommodate ports for air and gas passages, inlet valve, exhaust valve and spark plug (in case of petrol engines) or atomiser at the centre of the cover (in case of diesel engines).

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

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

where D = Cylinder bore in mm,

p = Maximum pressure inside the cylinder in N/mm^2,

σ_c = Allowable circumferential stress in MPa or N/mm^2. It may be taken as 30 to 50 MPa, and

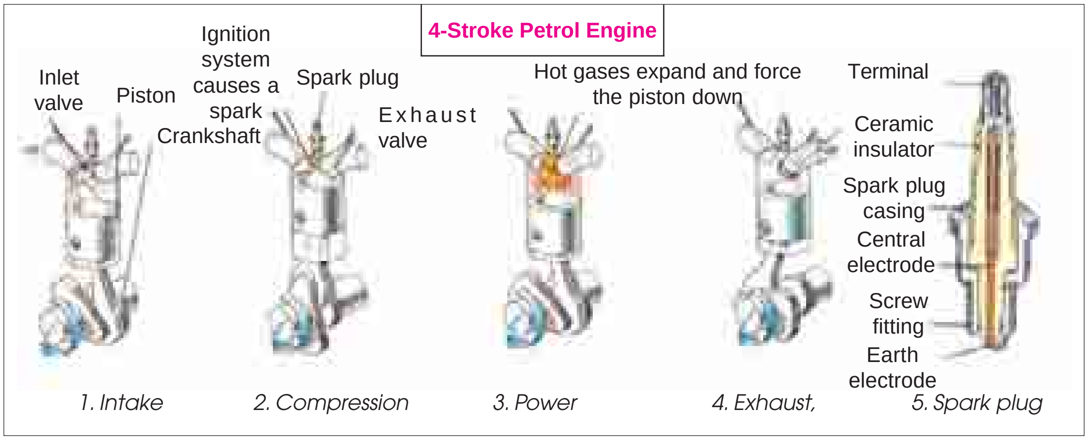
C = Constant whose value is taken as 0.1.

The studs or bolts are screwed up tightly alongwith a metal gasket or asbestos packing to provide a leak proof joint between the cylinder and cylinder head. The tightness of the joint also depends upon the pitch of the bolts or studs, which should lie between $19\sqrt{d}$ to $28.5\sqrt{d}$. The pitch circle diameter (D_p) is usually taken as D + 3d. The studs or bolts are designed in the same way as discussed above.

Example 32.1. *A four stroke diesel engine has the following specifications :*

Brake power = 5 kW ; Speed = 1200 r.p.m. ; Indicated mean effective pressure = 0.35 N / mm^2 ; Mechanical efficiency = 80 %.

Determine : 1. bore and length of the cylinder ; 2. thickness of the cylinder head ; and 3. size of studs for the cylinder head.



Solution. Given: $B.P.$ = 5kW = 5000 W ; N = 1200 r.p.m. or $n = N / 2 = 600$; $p_m = 0.35$ N/mm^2; $\eta_m = 80\% = 0.8$

1. Bore and length of cylinder

Let D = Bore of the cylinder in mm,

A = Cross-sectional area of the cylinder = $\frac{\pi}{4} \times D^2$ mm^2

l = Length of the stroke in m.

= 1.5 D mm = 1.5 D / 1000 m ....(Assume)

We know that the indicated power,

$$I.P = B.P. / \eta_m = 5000 / 0.8 = 6250 \text{ W}$$

We also know that the indicated power ($I.P.$),

$$6250 = \frac{p_m . l . A . n}{60} = \frac{0.35 \times 1.5D \times \pi D^2 \times 600}{60 \times 1000 \times 4} = 4.12 \times 10^{-3} D^3$$

...($\because$ For four stroke engine, $n = N/2$)

$\therefore$ $D^3 = 6250 / 4.12 \times 10^{-3} = 1517 \times 10^3$ or $D = 115$ mm **Ans.**

and $l = 1.5\ D = 1.5 \times 115 = 172.5$ mm

Taking a clearance on both sides of the cylinder equal to 15% of the stroke, therefore length of the cylinder,

$L = 1.15\ l = 1.15 \times 172.5 = 198$ say 200 mm **Ans.**

2. Thickness of the cylinder head

Since the maximum pressure (p) in the engine cylinder is taken as 9 to 10 times the mean effective pressure (p_m), therefore let us take

$$p = 9\ p_m = 9 \times 0.35 = 3.15 \text{ N/mm}^2$$

We know that thickness of the cyclinder head,

$$t_h = D\sqrt{\frac{C.p}{\sigma_t}} = 115\sqrt{\frac{0.1 \times 3.15}{42}} = 9.96 \text{ say } 10 \text{ mm } \textbf{Ans.}$$

...(Taking $C = 0.1$ and $\sigma_t = 42$ MPa = 42 N/mm^2)

3. Size of studs for the cylinder head

Let d = Nominal diameter of the stud in mm,

d_c = Core diameter of the stud in mm. It is usually taken as 0.84 d.

σ_t = Tensile stress for the material of the stud which is usually nickel steel.

n_s = Number of studs.

We know that the force acting on the cylinder head (or on the studs)

$$= \frac{\pi}{4} \times D^2 \times p = \frac{\pi}{4}(115)^2 3.15 = 32\ 702 \text{ N} \quad ...(i)$$

The number of studs (n_s) are usually taken between 0.01 D + 4 ($i.e.$ 0.01 × 115 + 4 = 5.15) and 0.02 D + 4 ($i.e.$ 0.02 × 115 + 4 = 6.3). Let us take $n_s = 6$.

We know that resisting force offered by all the studs

$$= n_s \times \frac{\pi}{4}(d_c)^2\ \sigma_t = 6 \times \frac{\pi}{4}(0.84d)^2\ 65 = 216\ d^2 \text{N} \quad ...(ii)$$

...(Taking σ_t = 65 MPa = 65 N/mm^2)

From equations ***(i)*** and ***(ii)***,

$$d^2 = 32\ 702 / 216 = 151 \text{ or } d = 12.3 \text{ say } 14 \text{ mm}$$

The pitch circle diameter of the studs (D_p) is taken $D + 3d$.

$\therefore$ $D_p = 115 + 3 \times 14 = 157$ mm

We know that pitch of the studs

$$= \frac{\pi \times D_p}{n_s} = \frac{\pi \times 157}{6} = 82.2 \text{ mm}$$

We know that for a leak-proof joint, the pitch of the studs should lie between $19\sqrt{d}$ to $28.5\sqrt{d}$, where d is the nominal diameter of the stud.

$\therefore$ Minimum pitch of the studs

$$= 19\sqrt{d} = 19\sqrt{14} = 71.1 \text{ mm}$$

and maximum pitch of the studs

$$= 28.5\sqrt{d} = 28.5\sqrt{14} = 106.6 \text{ mm}$$

Since the pitch of the studs obtained above (*i.e.* 82.2 mm) lies within 71.1 mm and 106.6 mm, therefore, size of the stud (d) calculated above is satisfactory.

$\therefore$ $d = 14$ mm **Ans.**

32.5 Piston

The piston is a disc which reciprocates within a cylinder. It is either moved by the fluid or it moves the fluid which enters the cylinder. The main function of the piston of an internal combustion engine is to receive the impulse from the expanding gas and to transmit the energy to the crankshaft through the connecting rod. The piston must also disperse a large amount of heat from the combustion chamber to the cylinder walls.

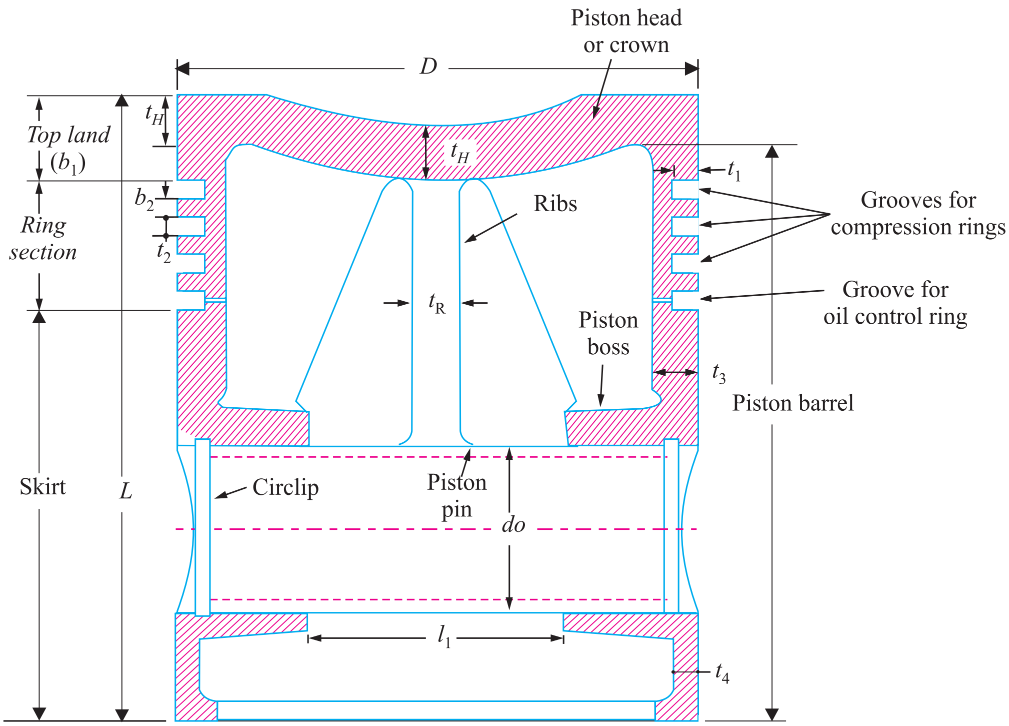


Fig. 32.3. Piston for I.C. engines (Trunk type).

The piston of internal combustion engines are usually of trunk type as shown in Fig. 32.3. Such pistons are open at one end and consists of the following parts :

1. *Head or crown.* The piston head or crown may be flat, convex or concave depending upon the design of combustion chamber. It withstands the pressure of gas in the cylinder.

2. *Piston rings.* The piston rings are used to seal the cyliner in order to prevent leakage of the gas past the piston.

3. *Skirt.* The skirt acts as a bearing for the side thrust of the connecting rod on the walls of cylinder.

4. *Piston pin.* It is also called ***gudgeon pin*** or ***wrist pin.*** It is used to connect the piston to the connecting rod.

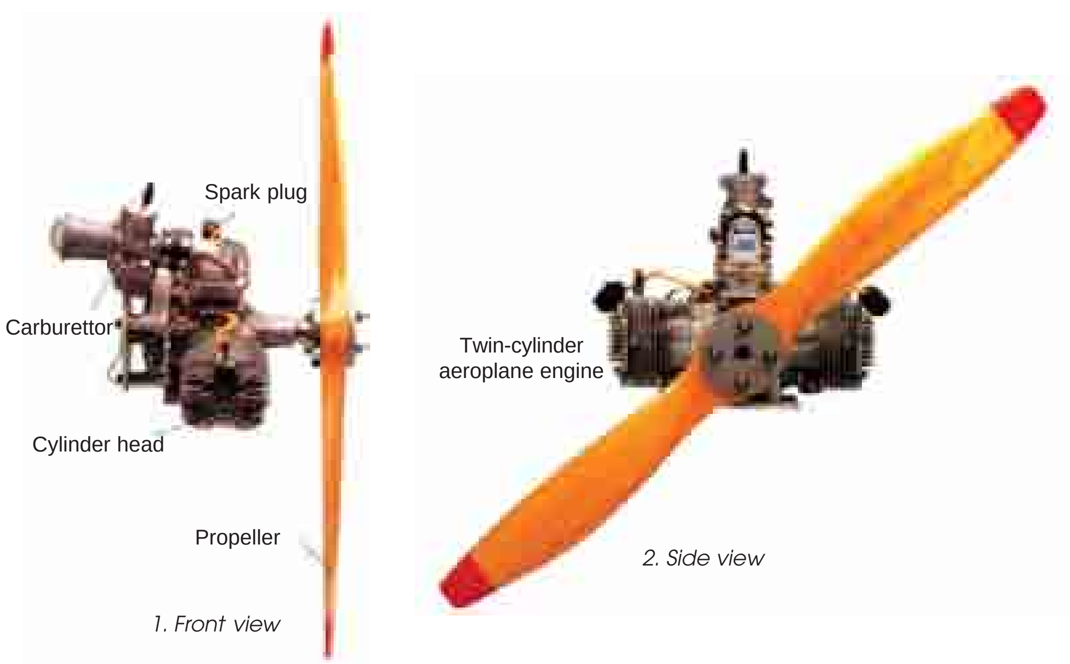
32.6 Design Considerations for a Piston

In designing a piston for I.C. engine, the following points should be taken into consideration :

1. It should have enormous strength to withstand the high gas pressure and inertia forces.
2. It should have minimum mass to minimise the inertia forces.
3. It should form an effective gas and oil sealing of the cylinder.
4. It should provide sufficient bearing area to prevent undue wear.
5. It should disprese the heat of combustion quickly to the cylinder walls.
6. It should have high speed reciprocation without noise.
7. It should be of sufficient rigid construction to withstand thermal and mechanical distortion.
8. It should have sufficient support for the piston pin.

32.7 Material for Pistons

The most commonly used materials for pistons of I.C. engines are cast iron, cast aluminium, forged aluminium, cast steel and forged steel. The cast iron pistons are used for moderately rated



Twin cylinder airplane engine of 1930s.

engines with piston speeds below 6 m / s and aluminium alloy pistons are used for highly rated engines running at higher piston sppeds. It may be noted that

1. Since the *coefficient of thermal expansion for aluminium is about 2.5 times that of cast iron, therefore, a greater clearance must be provided between the piston and the cylinder wall (than with cast iron piston) in order to prevent siezing of the piston when engine runs continuously under heavy loads. But if excessive clearance is allowed, then the piston will develop ***'piston slap'*** while it is cold and this tendency increases with wear. The less clearance between the piston and the cylinder wall will lead to siezing of piston.

2. Since the aluminium alloys used for pistons have high **heat conductivity (nearly four times that of cast iron), therefore, these pistons ensure high rate of heat transfer and thus keeps down the maximum temperature difference between the centre and edges of the piston head or crown.

Notes: (*a*) For a cast iron piston, the temperature at the centre of the piston head (T_C) is about 425°C to 450°C under full load conditions and the temperature at the edges of the piston head (T_E) is about 200°C to 225°C.

(*b*) For aluminium alloy pistons, T_C is about 260°C to 290°C and T_E is about 185°C to 215°C.

3. Since the aluminium alloys are about ***three times lighter than cast iron, therfore, its mechanical strength is good at low temperatures, but they lose their strength (about 50%) at temperatures above 325°C. Sometimes, the pistons of aluminium alloys are coated with aluminium oxide by an electrical method.

32.8 Piston Head or Crown

The piston head or crown is designed keeping in view the following two main considerations, *i.e.*

1. It should have adequate strength to withstand the straining action due to pressure of explosion inside the engine cylinder, and

2. It should dissipate the heat of combustion to the cylinder walls as quickly as possible.

On the basis of first consideration of straining action, the thickness of the piston head is determined by treating it as a flat circular plate of uniform thickness, fixed at the outer edges and subjected to a uniformly distributed load due to the gas pressure over the entire cross-section.

The thickness of the piston head (t_H), according to Grashoff's formula is given by

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

where p = Maximum gas pressure or explosion pressure in N/mm^2,

D = Cylinder bore or outside diameter of the piston in mm, and

σ_t = Permissible bending (tensile) stress for the material of the piston in MPa or N/mm^2. It may be taken as 35 to 40 MPa for grey cast iron, 50 to 90 MPa for nickel cast iron and aluminium alloy and 60 to 100 MPa for forged steel.

On the basis of second consideration of heat transfer, the thickness of the piston head should be such that the heat absorbed by the piston due combustion of fuel is quickly transferred to the cylinder walls. Treating the piston head as a flat ciucular plate, its thickness is given by

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

* The coefficient of thermal expansion for aluminium is 0.24×10^{-6} m / °C and for cast iron it is 0.1×10^{-6} m / °C.

** The heat conductivity for aluminium is 174.75 W/m/°C and for cast iron it is 46.6 W/m /°C.

*** The density of aluminium is 2700 kg / m^3 and for cast iron it is 7200 kg / m^3.

where H = Heat flowing through the piston head in kJ/s or watts,

k = Heat conductivity factor in W/m/°C. Its value is 46.6 W/m/°C for grey cast iron, 51.25 W/m/°C for steel and 174.75 W/m/°C for aluminium alloys.

T_C = Temperture at the centre of the piston head in °C, and

T_E = Temperature at the edges of the piston head in °C.

The temperature difference $(T_C - T_E)$ may be taken as 220°C for cast iron and 75°C for aluminium.

The heat flowing through the positon head (H) may be determined by the following expression, *i.e.*,

$$H = C \times HCV \times m \times B.P. \text{ (in kW)}$$

where C = Constant representing that portion of the heat supplied to the engine which is absorbed by the piston. Its value is usually taken as 0.05.

HCV = Higher calorific value of the fuel in kJ/kg. It may be taken as 45×10^3 kJ/kg for diesel and 47×10^3 kJ/ kg for petrol,

m = Mass of the fuel used in kg per brake power per second, and

$B.P.$ = Brake power of the engine per cylinder

Notes : 1. The thickness of the piston head (t_H) is calculated by using equations ***(i)*** and ***(ii)*** and larger of the two values obtained should be adopted.

2. When t_H is 6 mm or less, then no ribs are required to strengthen the piston head against gas loads. But when t_H is greater then 6 mm, then a suitable number of ribs at the centre line of the boss extending around the skirt should be provided to distribute the side thrust from the connecting rod and thus to prevent distortion of the skirt. The thickness of the ribs may be takes as $t_H / 3$ to $t_H / 2$.

3. For engines having length of stroke to cylinder bore (L / D) ratio upto 1.5, a cup is provided in the top of the piston head with a radius equal to 0.7 D. This is done to provide a space for combustion chamber.

32.9 Piston Rings

The piston rings are used to impart the necessary radial pressure to maintain the seal between the piston and the cylinder bore. These are usually made of grey cast iron or alloy cast iron because of their good wearing properties and also they retain spring characteristics even at high temperatures. The piston rings are of the following two types :

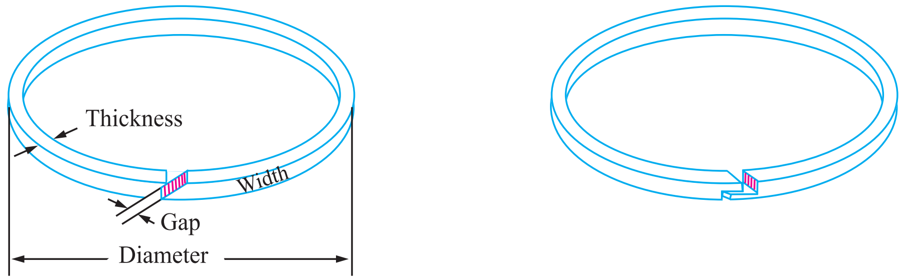
1. Compression rings or pressure rings, and

2. Oil control rings or oil scraper.

The ***compression rings*** *or* ***pressure rings*** are inserted in the grooves at the top portion of the piston and may be three to seven in number. These rings also transfer heat from the piston to the cylinder liner and absorb some part of the piston fluctuation due to the side thrust.

The ***oil control rings*** or ***oil scrapers*** are provided below the compression rings. These rings provide proper lubrication to the liner by allowing sufficient oil to move up during upward stroke and at the same time scraps the lubricating oil from the surface of the liner in order to minimise the flow of the oil to the combustion chamber.

The compression rings are usually made of rectangular cross-section and the diameter of the ring is slightly larger than the cylinder bore. A part of the ring is cut- off in order to permit it to go into the cylinder against the liner wall. The diagonal cut or step cut ends, as shown in Fig. 32.4 (a) and (b) respectively, may be used. The gap between the ends should be sufficiently large when the ring is put cold so that even at the highest temperature, the ends do not touch each other when the ring expands, otherwise there might be buckling of the ring.



(*a*) Diagonal cut.

(*b*) Step cut.

Fig. 32.4. Piston rings.

The radial thickness (t_1) of the ring may be obtained by considering the radial pressure between the cylinder wall and the ring. From bending stress consideration in the ring, the radial thickness is given by

$$t_1 = D\sqrt{\frac{3p_w}{\sigma_t}}$$

where D = Cylinder bore in mm,

p_w = Pressure of gas on the cylinder wall in N/mm^2. Its value is limited from 0.025 N/mm^2 to 0.042 N/mm^2, and

σ_t = Allowable bending (tensile) stress in MPa. Its value may be taken from 85 MPa to 110 MPa for cast iron rings.

The axial thickness (t_2) of the rings may be taken as 0.7 t_1 to t_1.

The minimum axial thickness (t_2) may also be obtained from the following empirical relation:

$$t_2 = \frac{D}{10\,n_R}$$

where n_R = Number of rings.

The width of the top land (*i.e.* the distance from the top of the piston to the first ring groove) is made larger than other ring lands to protect the top ring from high temperature conditions existing at the top of the piston,

∴ Width of top land,

$$b_1 = t_H \text{ to } 1.2\,t_H$$

The width of other ring lands (*i.e.* the distance between the ring grooves) in the piston may be made equal to or slightly less than the axial thickness of the ring (t_2).

∴ Width of other ring lands,

$$b_2 = 0.75\,t_2 \text{ to } t_2$$

The depth of the ring grooves should be more than the depth of the ring so that the ring does not take any piston side thrust.

The gap between the free ends of the ring is given by 3.5 t_1 to 4 t_1. The gap, when the ring is in the cylinder, should be 0.002 D to 0.004 D.

32.10 Piston Barrel

It is a cylindrical portion of the piston. The maximum thickness (t_3) of the piston barrel may be obtained from the following empirical relation :

$$t_3 = 0.03\,D + b + 4.5 \text{ mm}$$

where b = Radial depth of piston ring groove which is taken as 0.4 mm larger than the radial thickness of the piston ring (t_1)

$= t_1 + 0.4$ mm

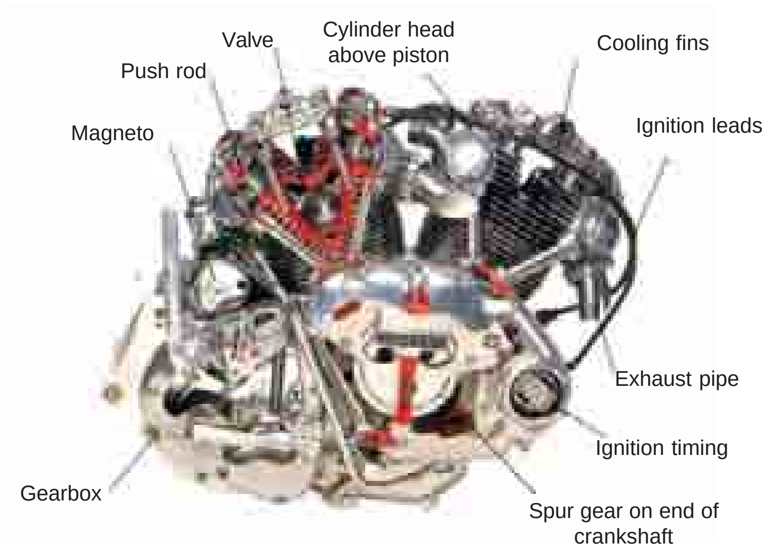
Thus, the above relation may be written as

$$t_3 = 0.03\,D + t_1 + 4.9 \text{ mm}$$

The piston wall thickness (t_4) towards the open end is decreased and should be taken as 0.25 t_3 to 0.35 t_3.

32.11 Piston Skirt

The portion of the piston below the ring section is known as ***piston skirt.*** In acts as a bearing for the side thrust of the connecting rod. The length of the piston skirt should be such that the bearing pressure on the piston barrel due to the side thrust does not exceed 0.25 N/mm^2 of the projected area for low speed engines and 0.5 N/mm^2 for high speed engines. It may be noted that the maximum thrust will be during the expansion stroke. The side thrust (R) on the cylinder liner is usually taken as 1/10 of the maximum gas load on the piston.



1000 cc twin -cylinder motorcycle engine.

We know that maximum gas load on the piston,

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

∴ Maximum side thrust on the cylinder,

$$R = P/10 = 0.1\,p \times \frac{\pi D^2}{4} \qquad ...(i)$$

where p = Maximum gas pressure in N/mm^2, and

D = Cylinder bore in mm.

The side thrust (R) is also given by

R = Bearing pressure × Projected bearing area of the piston skirt

$= p_b \times D \times l$

where l = Length of the piston skirt in mm. ...(*ii*)

From equations **(*i*)** and **(*ii*)**, the length of the piston skirt (l) is determined. In actual practice, the length of the piston skirt is taken as 0.65 to 0.8 times the cylinder bore. Now the total length of the piston (L) is given by

$$L = \text{Length of skirt} + \text{Length of ring section} + \text{Top land}$$

The length of the piston usually varies between D and 1.5 D. It may be noted that a longer piston provides better bearing surface for quiet running of the engine, but it should not be made unnecessarily long as it will increase its own mass and thus the inertia forces.

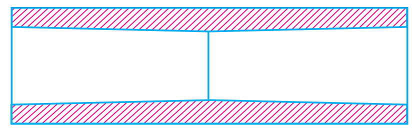


Fig.32.5. Piston pin.

32.12 Piston Pin

The piston pin (also called gudgeon pin or wrist pin) is used to connect the piston and the connecting rod. It is usually made hollow and tapered on the inside, the smallest inside diameter being at the centre of the pin, as shown in Fig. 32.5. The piston pin passes through the bosses provided on the inside of the piston skirt and the bush of the small end of the connecting rod. The centre of piston pin should be 0.02 D to 0.04 D above the centre of the skirt, in order to off-set the turning effect of the friction and to obtain uniform distribution of pressure between the piston and the cylinder liner.

The material used for the piston pin is usually case hardened steel alloy containing nickel, chromium, molybdenum or vanadium having tensile strength from 710 MPa to 910 MPa.

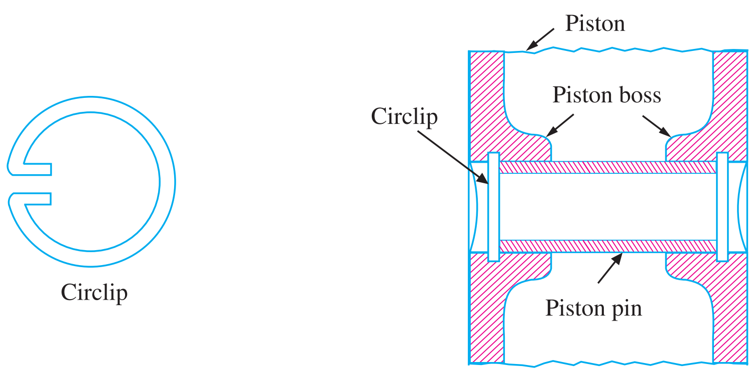


Fig. 32.6. Full floating type piston pin.

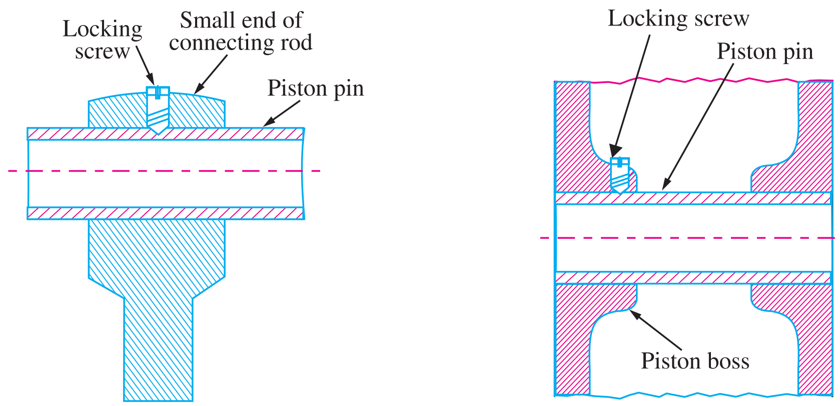
The connection between the piston pin and the small end of the connecting rod may be made either ***full floating type*** or ***semi-floating type.*** In the full floating type, the piston pin is free to turn both in the *piston bosses and the bush of the small end of the connecting rod. The end movements of the piston pin should be secured by means of spring circlips, as shown in Fig. 32.6, in order to prevent the pin from touching and scoring the cylinder liner.

In the semi-floating type, the piston pin is either free to turn in the piston bosses and rigidly secured to the small end of the connecting rod, or it is free to turn in the bush of the small end of the connecting rod and is rigidly secured in the piston bosses by means of a screw, as shown in Fig. 32.7

The piston pin should be designed for the maximum gas load or the inertia force of the piston, whichever is larger. The bearing area of the piston pin should be about equally divided between the piston pin bosses and the connecting rod bushing. Thus, the length of the pin in the connecting rod bushing will be about 0.45 of the cylinder bore or piston diameter (D), allowing for the end clearance

* The mean diameter of the piston bosses is made 1.4 d_0 for cast iron pistons and 1.5 d_0 for aluminium pistons, where d_0 is the outside diameter of the piston pin. The piston bosses are usually tapered, increasing the diameter towards the piston wall.

of the pin etc. The outside diameter of the piston pin (d_0) is determined by equating the load on the piston due to gas pressure (p) and the load on the piston pin due to bearing pressure (p_{b1}) at the small end of the connecting rod bushing.



(*a*) Piston pin secured to the small end of the connecting rod. (*b*) Piston pin secured to the boss of the piston.

Fig. 32.7. Semi-floating type piston pin.

Let d_0 = Outside diameter of the piston pin in mm

l_1 = Length of the piston pin in the bush of the small end of the connecting rod in mm. Its value is usually taken as 0.45 *D*.

p_{b1} = Bearing pressure at the small end of the connecting rod bushing in N/mm². Its value for the bronze bushing may be taken as 25 N/mm².

We know that load on the piston due to gas pressure or gas load

$$= \frac{\pi D^2}{4} \times p \qquad \text{..(i)}$$

and load on the piston pin due to bearing pressure or bearing load

$$= \text{Bearing pressure} \times \text{Bearing area} = p_{b1} \times d_0 \times l_1 \qquad \text{...(ii)}$$

From equations (*i*) and (*ii*), the outside diameter of the piston pin (d_0) may be obtained.

The piston pin may be checked in bending by assuming the gas load to be uniformly distributed over the length l_1 with supports at the centre of the bosses at the two ends. From Fig. 32.8, we find that the length between the supports,

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

Now maximum bending moment at the centre of the pin,

$$M = \frac{P}{2} \times \frac{l_2}{2} - \frac{P}{l_1} \times \frac{l_1}{2} \times \frac{l_1}{4}$$

$$= \frac{P}{2} \times \frac{l_2}{2} - \frac{P}{2} \times \frac{l_1}{4}$$

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

$$= \frac{P.\, l_1}{8} + \frac{P.D}{8} - \frac{P.\, l_1}{8} = \frac{P.D}{8} \qquad \text{...(iii)}$$

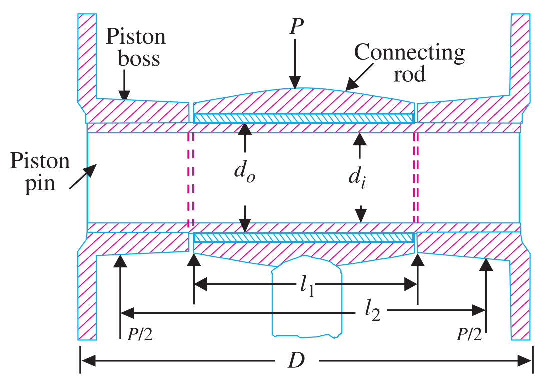


Fig. 32.8

We have already discussed that the piston pin is made hollow. Let d_0 and d_i be the outside and inside diameters of the piston pin. We know that the section modulus,

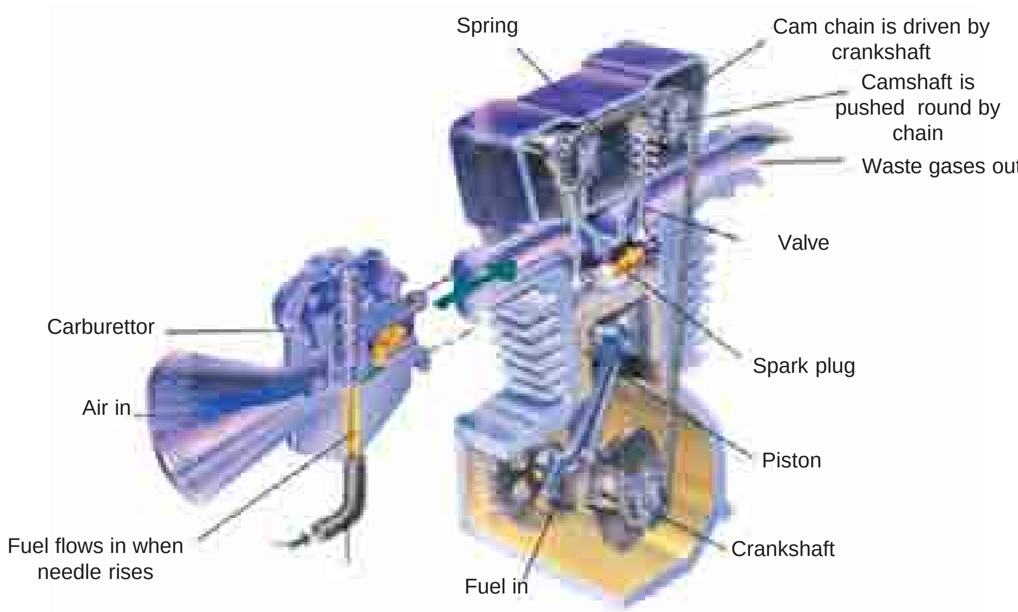
$$Z = \frac{\pi}{32}\left[\frac{(d_0)^4 - (d_i)^4}{d_0}\right]$$

We know that maximum bending moment,

$$M = Z \times \sigma_b = \frac{\pi}{32}\left[\frac{(d_0)^4 - (d_i)^4}{d_0}\right]\sigma_b$$

where σ_b = Allowable bending stress for the material of the piston pin. It is usually taken as 84 MPa for case hardened carbon steel and 140 MPa for heat treated alloy steel.

Assuming $d_i = 0.6\ d_0$, the induced bending stress in the piston pin may be checked.



Another view of a single cylinder 4-stroke petrol engine.

Example 32.2. *Design a cast iron piston for a single acting four stroke engine for the following data:*

Cylinder bore = 100 mm ; Stroke = 125 mm ; Maximum gas pressure = 5 N/mm² ; Indicated mean effective pressure = 0.75 N/mm² ; Mechanical efficiency = 80% ; Fuel consumption = 0.15 kg per brake power per hour ; Higher calorific value of fuel = 42 × 10³ kJ/kg ; Speed = 2000 r.p.m.

Any other data required for the design may be assumed.

Solution. Given : D = 100 mm ; L = 125 mm = 0.125 m ; p = 5 N/mm² ; p_m = 0.75 N/mm²; η_m = 80% = 0.8 ; m = 0.15 kg / BP / h = 41.7 × 10^{-6} kg / BP / s; HCV = 42 × 10^3 kJ / kg ; N = 2000 r.p.m.

The dimensions for various components of the piston are determined as follows :

1. *Piston head or crown*

The thickness of the piston head or crown is determined on the basis of strength as well as on the basis of heat dissipation and the larger of the two values is adopted.

We know that the thickness of piston head on the basis of strength,

$$t_H = \sqrt{\frac{3p.D^2}{16\,\sigma_t}} = \sqrt{\frac{3 \times 5(100)^2}{16 \times 38}} = 15.7 \text{ say } 16 \text{ mm}$$

...(Taking σ_t for cast iron = 38 MPa = 38 N/mm^2)

Since the engine is a four stroke engine, therefore, the number of working strokes per minute,

$$n = N / 2 = 2000 / 2 = 1000$$

and cross-sectional area of the cylinder,

$$A = \frac{\pi\ D^2}{4} = \frac{\pi\ (100)^2}{4} = 7855 \text{ mm}^2$$

We know that indicated power,

$$IP = \frac{p_m.L.A.n}{60} = \frac{0.75 \times 0.125 \times 7855 \times 1000}{60} = 12\ 270 \text{ W}$$

$$= 12.27 \text{ kW}$$

$\therefore$ Brake power, $BP = IP \times \eta_m = 12.27 \times 0.8 = 9.8$ kW ...($\because$ $\eta_m = BP / IP$)

We know that the heat flowing through the piston head,

$$H = C \times HCV \times m \times BP$$

$$= 0.05 \times 42 \times 10^3 \times 41.7 \times 10^{-6} \times 9.8 = 0.86 \text{ kW} = 860 \text{ W}$$

....(Taking C = 0.05)

$\therefore$ Thickness of the piston head on the basis of heat dissipation,

$$t_H = \frac{H}{12.56\,k\,(T_C - T_E)} = \frac{860}{12.56 \times 46.6 \times 220} = 0.0067 \text{ m} = 6.7 \text{ mm}$$

...($\because$ For cast iron , k = 46.6 W/m/°C, and $T_C - T_E$ = 220°C)

Taking the larger of the two values, we shall adopt

t_H = 16 mm **Ans.**

Since the ratio of L / D is 1.25, therefore a cup in the top of the piston head with a radius equal to 0.7 D (*i.e.* 70 mm) is provided.

2. *Radial ribs*

The radial ribs may be four in number. The thickness of the ribs varies from t_H / 3 to t_H / 2.

$\therefore$ Thickness of the ribs, t_R = 16 / 3 to 16 / 2 = 5.33 to 8 mm

Let us adopt t_R = 7 mm **Ans.**

3. *Piston rings*

Let us assume that there are total four rings (*i.e.* n_r = 4) out of which three are compression rings and one is an oil ring.

We know that the radial thickness of the piston rings,

$$t_1 = D\sqrt{\frac{3p_w}{\sigma_t}} = 100\sqrt{\frac{3 \times 0.035}{90}} = 3.4 \text{ mm}$$

...(Taking p_w = 0.035 N/mm^2, and σ_t = 90 MPa)

and axial thickness of the piston rings

$$t_2 = 0.7\ t_1 \text{ to } t_1 = 0.7 \times 3.4 \text{ to } 3.4 \text{ mm} = 2.38 \text{ to } 3.4 \text{ mm}$$

Let us adopt t_2 = 3 mm

We also know that the minimum axial thickness of the pistion ring,

$$t_2 = \frac{D}{10\ n_r} = \frac{100}{10 \times 4} = 2.5\ \text{mm}$$

Thus the axial thickness of the piston ring as already calculated (*i.e.* $t_2 = 3$ mm)is satisfactory. **Ans.**

The distance from the top of the piston to the first ring groove, *i.e.* the width of the top land,

$$b_1 = t_H \text{ to } 1.2\ t_H = 16 \text{ to } 1.2 \times 16 \text{ mm} = 16 \text{ to } 19.2 \text{ mm}$$

and width of other ring lands,

$$b_2 = 0.75\ t_2 \text{ to } t_2 = 0.75 \times 3 \text{ to } 3 \text{ mm} = 2.25 \text{ to } 3 \text{ mm}$$

Let us adopt $b_1 = 18$ mm ; and $b_2 = 2.5$ mm **Ans.**

We know that the gap between the free ends of the ring,

$$G_1 = 3.5\ t_1 \text{ to } 4\ t_1 = 3.5 \times 3.4 \text{ to } 4 \times 3.4 \text{ mm} = 11.9 \text{ to } 13.6 \text{ mm}$$

and the gap when the ring is in the cylinder,

$$G_2 = 0.002\ D \text{ to } 0.004\ D = 0.002 \times 100 \text{ to } 0.004 \times 100 \text{ mm}$$
$$= 0.2 \text{ to } 0.4 \text{ mm}$$

Let us adopt $G_1 = 12.8$ mm ; and $G_2 = 0.3$ mm **Ans.**

4. *Piston barrel*

Since the radial depth of the piston ring grooves (b) is about 0.4 mm more than the radial thickness of the piston rings (t_1), therefore,

$$b = t_1 + 0.4 = 3.4 + 0.4 = 3.8 \text{ mm}$$

We know that the maximum thickness of barrel,

$$t_3 = 0.03\ D + b + 4.5 \text{ mm} = 0.03 \times 100 + 3.8 + 4.5 = 11.3 \text{ mm}$$

and piston wall thickness towards the open end,

$$t_4 = 0.25\ t_3 \text{ to } 0.35\ t_3 = 0.25 \times 11.3 \text{ to } 0.35 \times 11.3 = 2.8 \text{ to } 3.9 \text{ mm}$$

Let us adopt $t_4 = 3.4$ mm

5. *Piston skirt*

Let l = Length of the skirt in mm.

We know that the maximum side thrust on the cylinder due to gas pressure (p),

$$R = \mu \times \frac{\pi D^2}{4} \times p = 0.1 \times \frac{\pi (100)^2}{4} \times 5 = 3928 \text{ N}$$

...(Taking $\mu = 0.1$)

We also know that the side thrust due to bearing pressure on the piston barrel (p_b),

$$R = p_b \times D \times l = 0.45 \times 100 \times l = 45\ l \text{ N}$$

...(Taking $p_b = 0.45$ N/mm^2)

From above, we find that

$$45\ l = 3928 \text{ or } l = 3928 / 45 = 87.3 \text{ say } 90 \text{ mm}$$ **Ans.**

∴ Total length of the piston ,

$$L = \text{Length of the skirt} + \text{Length of the ring section} + \text{Top land}$$
$$= l + (4\ t_2 + 3b_2) + b_1$$
$$= 90 + (4 \times 3 + 3 \times 3) + 18 = 129 \text{ say } 130 \text{ mm}$$ **Ans.**

6. *Piston pin*

Let d_0 = Outside diameter of the pin in mm,

l_1 = Length of pin in the bush of the small end of the connecting rod in mm, and

p_{b1} = Bearing pressure at the small end of the connecting rod bushing in N/mm^2. It value for bronze bushing is taken as 25 N/mm^2.

We know that load on the pin due to bearing pressure

$$= \text{Bearing pressure} \times \text{Bearing area} = p_{b1} \times d_0 \times l_1$$

$$= 25 \times d_0 \times 0.45 \times 100 = 1125\, d_0 \text{ N} \qquad \text{...(Taking } l_1 = 0.45\, D)$$

We also know that maximum load on the piston due to gas pressure or maximum gas load

$$= \frac{\pi\, D^2}{4} \times p = \frac{\pi\,(100)^2}{4} \times 5 = 39\ 275 \text{ N}$$

From above, we find that

$$1125\, d_0 = 39\ 275 \quad \text{or} \quad d_0 = 39\ 275 / 1125 = 34.9 \text{ say } 35 \text{ mm}$$ **Ans.**

The inside diameter of the pin (d_i) is usually taken as 0.6 d_0.

$\therefore$ $d_i = 0.6 \times 35 = 21$ mm **Ans.**

Let the piston pin be made of heat treated alloy steel for which the bending stress (σ_b)may be taken as 140 MPa. Now let us check the induced bending stress in the pin.

We know that maximum bending moment at the centre of the pin,

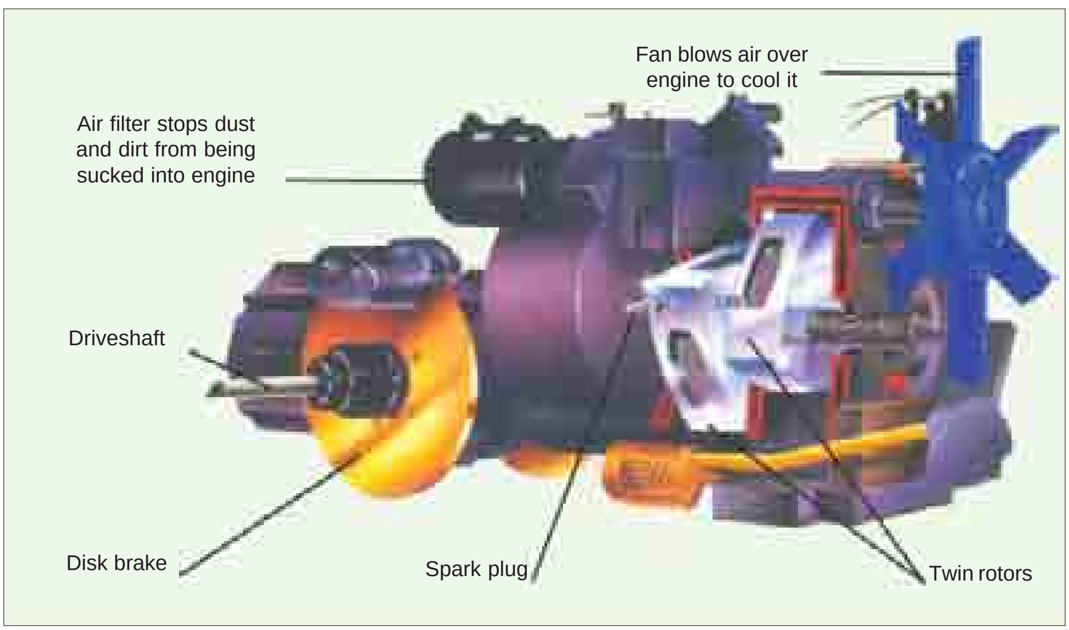
$$M = \frac{P.D}{8} = \frac{39\,275 \times 100}{8} = 491 \times 10^3 \text{ N-mm}$$

We also know that maximum bending moment (M),

$$491 \times 10^3 = \frac{\pi}{32}\left[\frac{(d_0)^4 - (d_i)^4}{d_0}\right]\sigma_b = \frac{\pi}{32}\left[\frac{(35)^4 - (21)^4}{35}\right]\sigma_b = 3664\,\sigma_b$$

$$\therefore \quad \sigma_b = 491 \times 10^3 / 3664 = 134 \text{ N/mm}^2 \text{ or MPa}$$

Since the induced bending stress in the pin is less than the permissible value of 140 MPa (*i.e.* 140 N/mm^2), therefore, the dimensions for the pin as calculated above (*i.e.* d_0 = 35 mm and d_i = 21 mm) are satisfactory.



German engineer Fleix Wankel (1902-88) built a rotary engine in 1957. A triangular piston turns inside a chamber through the combustion cycle.