Engine Design

Chapter 05: IC Engine Component Design-The Piston

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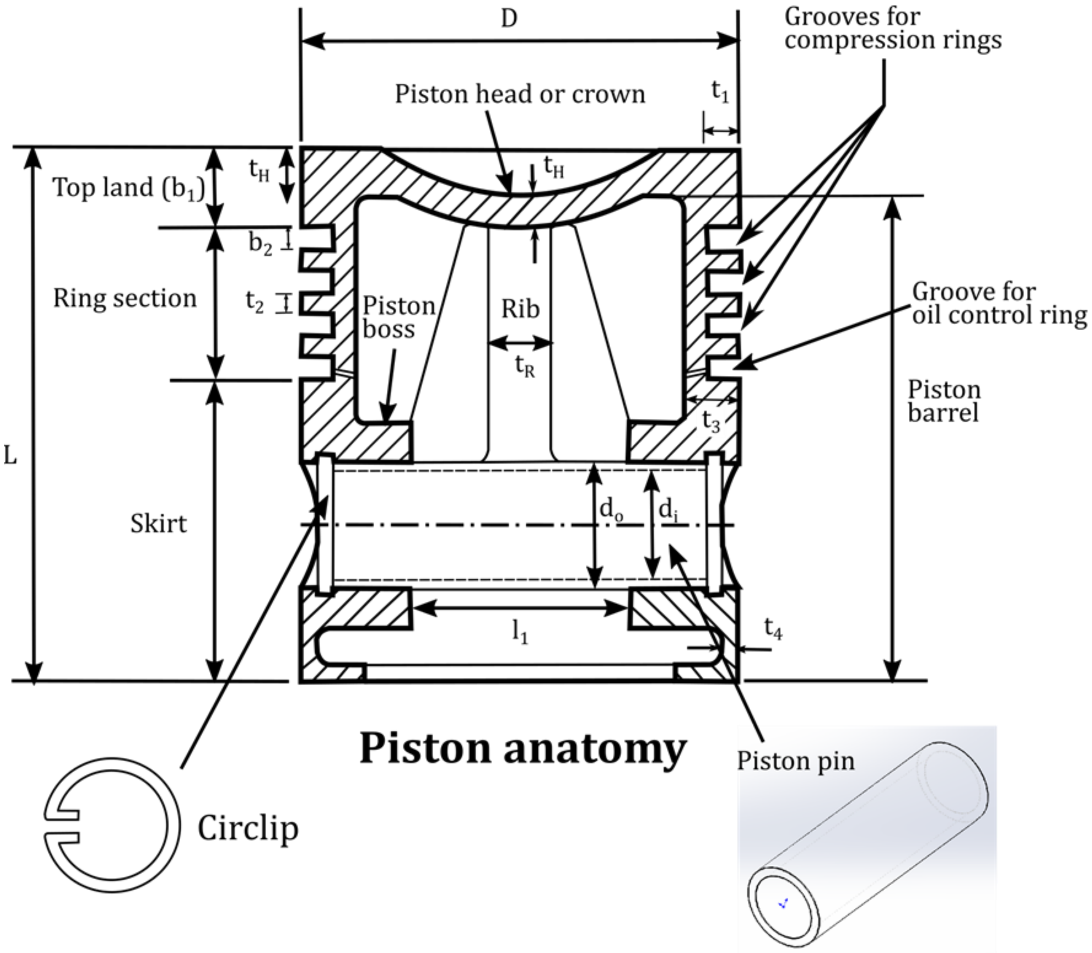
The Piston

- It should have:
  - Enormous strength (pressures & forces).
  - Minimum mass (inertia forces).
  - Form effective gas and oil sealing.
  - Provide bearing area (prevent wear).
  - Disperse the heat quickly.
  - High speed reciprocation (without noise).
  - Rigid construction (distortions).

Piston material

- Cast iron
- Cast aluminium
- Forged aluminium
- Cast steel
- Forged steel

Piston anatomy



t_1 = The radial thickness

t_2 = The minimum axial thickness of ring

t_3 = Thickness of piston barrel

t_4 = Piston wall thickness

b_1 = Width of the top land

b_2 = Width of the ring land

t_R = The thickness of the ribs

t_H = The thickness of piston head

d_o = Outer diameter

d_i = Inner diameter

L = Length

D = Diameter of cylinder bore

l_1 = Length of bearing

Design of piston head / crown

- The thickness of the piston head by strength consideration is,

$$t_h = \sqrt{\frac{3p_{max}.D^2}{16\sigma_b}}$$

D = cylinder bore (mm) = 81mm

$$p_{max} = \text{maximum gas pressure (mm)}$$
$$= \ 4 \text{ MPa (assume average)}$$

For aluminum alloy,

$$\sigma_b = permissible\ bending\ stress\ (\frac{\text{N}}{\text{mm}^2})$$
$$= 50 \text{ N/mm}^2 \text{ (assume)}$$

Design of piston head / crown

Therefore,

$$t_H = \sqrt{\frac{3p_{max}.D^2}{16\sigma_b}} = \sqrt{\frac{3\ x\ 4\ x\ 81^2}{16\ x\ 50}} = 9.92\ \text{mm}$$

Design of piston head / crown

- The thickness of the piston head by thermal consideration is,

$$t_h = \frac{H}{12.56k(T_c - T_e)}$$

H = the amount of heat conducted through the piston head (W)

= $C * HCV * m * BP * 10^3$

C = the ratio of heat absorbed by the piston to the total heat developed in the cylinder = 0.05

HCV = higher calorific value of fuel (kJ/kg)

For petrol = $47 * 10^3 \text{kJ/kg}$

m = mass of fuel used per brake power per second (kg/kW/s)

= (density x volume x speed)/BP

= $\frac{0.00013 \text{ x } 719.7 \text{ x } 250}{52.16} = 1.19 \text{ x } 10^{-4} \frac{\text{kg}}{\text{kW}}/\text{s}$

Design of piston head / crown

- BP = brake power of the engine per cylinder (kW) = 52.16kW

  k = thermal conductivity factor W/m/°C

  For aluminium alloy = 175 W/m/°C

  $T_c = \text{Temperature at the center of piston head (°C)}$

  $T_e = \text{Temperature at the edge of the piston head (°C)}$

  $For\ aluminium\ alloy, (T_c - T_e) = 75°\text{C}$

Therefore, $H = 0.05 * 47 * 1.19 * 10^{-4}\, x\, 52.16\, x\, 10^3 = 14586.54\ \text{W}$

$$t_h = \frac{14586.54}{12.56 * 175 * 75} = 0.0884\ \text{m} = 88.48\ \text{mm}$$

Ribs

- Since $t_h = 88.48\text{mm} > 6mm$

  therefore, ribs are required

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

$$t_R = \text{thickness of the ribs}$$

$$= \frac{88.48}{3}\ to\ \frac{88.48}{2} =$$

29.49 mm to 44.24 mm

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

Piston cup

Since *l*/D = 90/81 < 1.5,

Piston cup is required and the radius is,

0.7D = 0.7 x 81 = 56.7 mm

Piston rings

- $b = D\sqrt{\frac{3p_w}{\sigma_t}}$

b = radial width of ring (mm)

$p_w =$ allowable radial pressure on cylinder $wall$ $\left(\frac{\text{N}}{\text{mm}^2}\right)$

$= 0.035$ MPa (assumed)

$\sigma_t =$ permissible tensile stress for the ring material $\left(\frac{\text{N}}{\text{mm}^2}\right)$

$= 90$ MPa (assumed)

$$b = D\sqrt{\frac{3p_w}{\sigma_t}} = 81 \text{ x } \sqrt{\frac{3 \text{ X } 0.035}{90}} = 2.77 \text{ mm}$$

Piston rings

- $t = 0.7b \ to \ b$

  t = axial thickness of piston ring (mm)

  = 0.7 x 2.77 to 2.77 = 1.94 mm to 2.77 mm = 2.5 mm

- $t_2 = \frac{D}{10z}$

  $t_2 = \text{minimum axial thickness of piston ring (mm)}$

  Assume the number of compression ring is 3 and one oil ring,

  z = number of rings = 3+1 = 4

  $t_2 = \frac{D}{10z} = \frac{81}{10 * 4} = 2\text{mm} \qquad \text{t} > t_2$

Piston rings

- The gap between free end of the piston ring before assembly is,

$G_1 = 3.5b\ to\ 4b = 3.5 \text{ x } 2.77 \text{ to } 4 \text{ x } 2.77 = 9.70 \text{ mm to} 11.08 \text{ mm}$

= 10 mm

The gap between the free end of the piston ring after assembly is,

$G_2 = 0.002D\ to\ 0.004D = 0.002 \text{ x } 81\ to\ 0.004 \text{ x } 81$

= 0.16 mm to 0.32 mm = 0.2 mm

The width of the top land is,

$b_1 = t_h\ to\ 1.2t_h = 88.48\ to\ 1.2 \text{ x } 88.48 = 88.48 \text{ mm}\ to\ 106.18 \text{ mm}$

= 100 mm

The width of the ring grooves is,

$b_2 = 0.75h\ to\ h = 0.75 \text{ x } 2.5\ to\ 2.5 = 1.9 \text{ mm to } 2.5 \text{ mm}$

Piston barrel

The thickness of the barrel at the top end is,

$$t_3 = 0.03D + b + 4.9$$

$$= (0.03 \text{ x } 81) + \ 2.77 + 4.9 = 10.1 \text{ mm}$$

The thickness of the barrel at the open end is,

$$t_4 = 0.25\ t_3\ to\ 0.35\ t_3$$

$$= 0.25 \text{ x } 10.1\ to\ 0.35 \text{ x } 10.1$$

$$= 2.53 \text{ mm}\ to\ 3.54 \text{ mm}$$

$$= 3 \text{ mm}$$

Piston skirt

Maximum gas force on piston head = $\frac{\pi D}{4} p_{max}$

Side thrust = $\mu \left(\frac{\pi D^2}{4} p_{max} \right) = \ p_b D l_s$

μ = coefficient constant of friction = 0.1

$$p_b = \text{allowable bearing pressure} \left(MPa\ or\ \frac{N}{mm}^{2} \right)$$

$$= 0.45\ \text{MPa(assume)}$$

$$l_s = \text{length of skirt (mm)}$$

Therefore,

$$0.1 \left(\frac{\pi\, \text{x} 81^2}{4}\ \text{x}\ 4 \right) = \ 0.45\ \text{x}\ 81 l_s$$

$$l_s = \ 56.5\ \text{mm}$$

Piston length

Length of ring section = $4h + 3h_2 = 4(2.5) + 3(2)$

= 16 mm

Length of piston = h_1+length of ring section $+ l_s$

= 100 + 16 + 56.5 = 172.5 mm = 173 mm

According to empirical relationship,

L = D to 1.5D = 81 to 122 mm

$\therefore$ D< L >1.5D so it is out of limit

Piston pin

$$P = Force\ on\ piston$$

= $\frac{\pi D^2}{4} p_{max} = \frac{\pi(81^2)}{4}$ x $4 = 20611.99\ N$

Assume:

Assume the piston pin is made of heat treated alloy steel and the permissible tensile stress is $140\ \frac{\text{N}}{\text{mm}^2}$

The bearing pressure $(p_b)_1$ at the bush of the small end of the connecting rod is 30 MPa.

Bearing area = d_0 x l_1

$$\frac{\pi D^2}{4} p_{max} = (p_b)_1 \text{ x } d_0 \text{ x } l_1 \quad (\text{Given } l_1 / d_0 = 1.5)$$

$$20{,}611.99 = 30 d_0 \, (1.5 d_0\,)$$

$$(d^0)^2 = 458.04$$

$$d_0 = 21.40 \text{ mm}$$

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