Design of Three Cylinder In-line Petrol Engine

An engine Suitable for a small hatch back

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Abstract : Now-a-days automobile became a necessity to common man. Most of the light vehicles are run by petrol engines. When compared with the Diesel engines Petrol engines are cheaper, light in weight, more powerful but high running cost. Though the running cost of the petrol engines is more most of the people prefer petrol engines, because due to ease in maintenance, smooth in operation and smoke free. In this paper an effort is made to Design a Petrol engine which is suitable for small car applications. All most all components of Petrol engine such as Cylinder, Piston, Connecting rod, Crank shaft etc. were designed using scientific principles as well as empirical relations and tested for their strengths against allowable stresses. Finally the results were tested with the common specifications of the engines available in the market for their validation.

***Index Terms* – Engine Components, small car, hatch back model, Piston, Crankshaft.**

I. INTRODUCTION

Small car is a dream to fulfill the dream of common man. To fulfill this many car manufacturers introduced many models prestigiously such as Maruti '800', DCM Daewoo 'Matiz', Hyundai 'Santro', Tata 'nano' etc. Later they became premium due to customs, taxes and so many reasons. In this project an effort is made to design a small car engine which is cheaper to manufacture and to inculcate the design practice for young researchers.

INTRODUCTION

A	: cross section Area of the cylinder
bsfc	: Brake specific fuel consumption
Di	: Inside diameter of piston
d_{ps}	:outside diameter and d_i =Inside diameter of piston pin
D	: Bore of the cylinder
D_o	: Out side diameter of the cylinder
h	:Width of the ring
I_{xx}	: Moment of Inertia about X-axis
l_p	:Length of piston pin
k	: Heat conducting of piston material
L	: Strake Length
p_m	: Mean effective pressure
M	: Bending moment
P	:Gas force
P_{gas}	: maximum gas force
q	: heat flow from the gases
R	: Side thrust
R max	: Maximum side thrust
t_h	: Head thickness
t	: Thickness of piston crown
t_{rad}	: Radial thickness of the piston rings
t_l	:thickness of the cylinder block wall
t_2	:Thickness of the cylinder flange
t_3	: Thickness of piston crown wall
V_S	: Stroke volume
Z	: section modulus
σ	: compressible strength
σ_θ	: Hoop stress
σ_{th}	: Theoretical stress
σ_b	:Bending stress,
μ	: Poisson's ratio
σ_L	: Longitudinal stress
σ_b	: Bending stress
σ_{cr}	:Critical stress
τ	: shear stress

Lighter vehicles are equipped with petrol engines due to their better qualities when compared with other alternatives such as Diesel engines. Petrol, or gasoline, is a liquid mixture created form crude oil. It is made up of hydrocarbons and iso-octane. It is a fuel most commonly used in internal combustion engines.

- Relatively concentrated and can travel many hundred km with one full tank of petrol
- It is highly available
- It is fairly cheap
- It is not difficult to make - it just has to be distilled and no waste is produced
- It is easy to carry around
- It is fairly safe to store

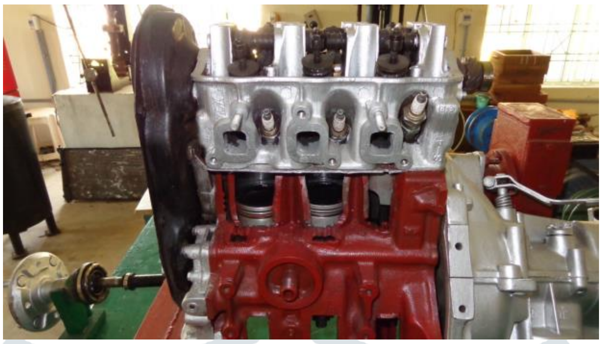

Fig. 1 Cut section of 3Cylinder 4 Stroke Petrol engine

DIFFERENCES BETWEEN PETROL AND DIESEL ENGINES

Agenda	**Petrol Engine**	**Diesel Engine**
Number of piston rings	2 to 3	3 to 5
Type of piston	Deflector or Flat crown type is used	Combustion chamber type is used
Weight	Light	Heavy
Cycle of operation	Otto	Diesel
Cause of ignition	Spark plug	High temperature caused by high compression of air
Compression ratio	6.5 to 10.5	15 to 22
Compression pressure	About 10 Kgf/cm^2	30 – 40 Kgf/cm^2
Speed	Higher rpm	Lower rpm

II. METHODOLOGY

In this project an effort is made to design all most all components of Petrol engine like Cylinder, Piston, Connecting rod, Crank shaft etc. using scientific principles and empirical relations and tested for their strengths against their allowable stresses. Finally we tested the results with the common specifications of the engines available in the market for their validation. The results obtained are found satisfactory.

Also selected a small car engine and sectioned it on slotting machine as shown in Fig.1. Again that is assembled and rotated by an electric motor for demonstration of its working. To rotate the engine at very slow speed we designed and manufactured a high reduction gear box. For this purpose we have chosen worm and worm wheel gear train, because it gives very high speed reduction. We cut the worm on Lathe machine using thread chasing mechanism. The outside diameter of the worm is 25mm and pitch 25 mm. we cut the worm wheel on Milling machine using gear tooth profile cutter. The worm wheel consists of 20 number of teeth. So we got 1:20 speed reduction. That is the worm has to rotate 20 times for one revolution of worm wheel. Finally we connected the assembled cut section of the engine to electrical motor of 500 rpm through reduction gear box. If we switch on the electrical motor that rotates the engine at very slow speed about 25 rpm.

Standard proportions of piston

Dimension	Petrol engine	Diesel engine
Piston crown thickness (t)	(0.05 – 0.1) D	(0.12 – 0.2) D
Piston height (H)	(0.8 – 1.3) D	(1.0 – 1.7) D
Height of piston (top part) (h_1)	(0.45 – 0.75) D	(0.6 – 1.0) D
Skirt length (h_2)	(0.6 – 0.8) D	(0.7 – 1.1) D
Top land (t_1)	(0.06 – 0.12) D	(0.11 – 0.2)D
Radial clearance of ring (Δt)	(0.7 – 0.95) mm	(0.7 -0.95) mm
Length of pin (l_p)	(0.78 – 0.93) D	(0.8 – 0.93) D

III. DESIGN PROCEDURE

Important components of engine are Cylinder block, head,piston, connectingrod and carnak shaft. Their design procedure is given below.

3.1 Design of Cylinder

Data Assumed:

1. Brake Power = 35HP = 26.1KW
2. Speed = 2500rpm
3. Indicated mean effective pressure= 0.55MPa
4. Mechnical efficiency = 85%

Mechanical efficiency $= \dfrac{BrakePower}{Indicatedepower}$

Thus, Indicated Power $= \dfrac{BrakePower}{Mechanicalefficiency}$

$$= [26.1 \div 0.85] = 30.7kw$$

Indicated Power $= \dfrac{p_m LAN}{60}$

p_m = Mean effective pressure = 0.55 KPa

L = Strake Length = 1.05 D

D = Bore of the cylinder

There fore, $30700 = 550 \times 1.05D \times \dfrac{\pi}{4} D^2 \times \dfrac{2500/2}{60}$

Then, D = 68.5 mm

L = 72mm

The maximum explosion pressure P_{max} $= 8 \times P_m$

$= 8 \times 550$

$= 4400$ KN/m^2

For east Iron, Allowable compressible strength σ = 60 N/mm^2

And Poisson's ratio μ = 0.21

The thickness of the cylinder with reboring allowance is given by

$$t = \frac{P_{max} \times .D}{2\sigma} + C$$

C=Allowance for reboring =1.5

$$t = \frac{4.4 \times 68.5}{2 \times 60} + 1.5$$

=4.91 mm say 6mm

(i) The hoop stress produced in the cylinder

$$\sigma_\theta = \frac{P_{\max}.D}{2t} = \frac{4.4 \times 68.5}{2 \times 6} = 25.1 N / mm^2$$

(ii) Thermal stress $= \sigma_{th} = \left[\frac{E.\alpha.\Delta I}{2(1-m)}\right]$

E = 100 KN/mm^2

α = 11 × 10^{-6} mm/^{0}c

ΔI = 120^{0}C

$$\sigma_{th} = \left[\frac{1\times10^5 \times 11\times10^{-6} \times 120}{2(1-0.21)}\right]$$

= 83.5 N/mm^2

Total stress, $\sigma_\theta + \sigma_{th}$ = 25.1 + 83.5

= 108.6 N/mm^2

The stress is with in the permissible tensile strength of cast iron (130 N/mm^2)

(iii) Longitudinal tensile stress

$$\sigma_\theta = \frac{P_{\max}.D}{4t} = \frac{4.4 \times 68.5}{4 \times 6} = 12.5 N / mm^2$$

(iv) the side thrust in the cylinder

R = θ sin Ø = I tan Ø

Assuming the for small obliquity of the connecting rod the maximum side thrust is 10% of the gas force,

i.e. 0.1 × Area × pressure

$$R = 0.1 \times \frac{\pi}{4} \times 68.5^2 \times 4.4$$

There fore R_{max} = 1621.5 N

Length of the cylinder= 1.25 × stroke length

= 1.25 × 72

= 90 mm

(v)Bending stress induced in the cylinder

$$\sigma_b = \frac{M}{Z}$$

where M = Bending moment

$= R_{max} \times \frac{ab}{a+b}$

a, b are position of piston fram T.D.C. and B.D.C.

a = 50 mm

b = 70 mm

$$M = 1621.5 \times \frac{50 \times 70}{50 + 70} = 47293.75 N - mm$$

Z : section modulus

$$Z = \frac{\pi}{32}\left[\frac{D_o^4 - D^4}{D_o}\right]$$

Where D_o = D + 2t = 68.5 + 2 × 6

= 80.5 mm

$$Z = \frac{\pi}{32}\left[\frac{80.5^4 - 68.5^4}{80.5}\right]$$

= 24362.5 mm^3

∴ Bending Stress $\sigma_b = \frac{M}{Z} = \frac{47293.75}{24362.5} = 1.94 N/mm^2$

Total tensile stress = $\sigma_L + \sigma_b$

= 12.5 + 1.94 = 14.44 N/mm^2

Which is less than the allowable strength. Hence the design is satisfactory.

OTHER DIMENSIONS

(i) Thickness of the cylinder block wall

t_l = 0.045 D + 2

= (0.045 × 68.5) + 2

= 5.08 mm = say 6mm

(ii) Thickness of the cylinder flange

t_2 = 1.3 t_l = 1.3×6 = 8 mm

Thickness of the cylinder head

$$t_h = D\left[\frac{c.p_{max}}{\sigma}\right]^{0.5}$$ where c = a constant = 0.162

$$= 68.5\left[\frac{0.162 \times 4.4}{60}\right]^{0.5}$$

= 5.49 mm = Say 6 mm

3.2 Design of Piston:

Data Assumed

Piston Material = Aluminium

Piston Dia meter = 68.5mm

Length of strake = 72 mm

Mean effective pressure (P_m) = 0.55 N/ mm^2

1. Thickness of piston crown

$$t = \sqrt{\frac{3 \times p_{max} \times D^2}{16 \times \sigma_t}}$$

σ_t = Tensile stress for Aluminium may vary from 50 MPa to 90 MPa

$$t = \sqrt{\frac{3 \times 4.4 \times 68.5^2}{16 \times 75}}$$

= 7.18 mm = say 8 mm

2. Thickness of piston crown

$$t = \frac{H}{12.56 \times K(Tc - Te)} \text{ in mm}$$

K = Thermal conductivity factor in W / m / ^{0}C

= 174.5 W / m / ^{0}C for aluminium

Tc = Temperature at the center of the piston head in ^{0}C

Te = Temperature at the edges of the piston head in ^{0}C

Tc - Te = 75 ^{0}C for alluminium

H = (C × HCV × m × BP) × 10^3 in watts

C = constant representing the heat supplied to the engine which is absorbed by the piston

C = 0.05

HCV = High calorific value of fuel

= 47 × 10^3 KJ / kg

m = mass of fuel in kg per brake power per second

$$= \left[\frac{0.2\times 0.3}{60\times 60}\right] \quad kg / KW / sec$$

$$= 1.6\times 10^{-5} \quad kg / KW / sec$$

BP = brake power of engine per cylinder

$$H = (C \times HCV \times m \times BP) \times 10^{3}$$

$$H = (0.05 \times 47 \times 10^{3} \times 1.6\times 10^{-5} \times 26.1) \times 10^{3}$$

= 981 Watts

$$t = \frac{H}{12.56\times K(Tc - Te)}$$

$$= \frac{981}{12.56\times 174.75\times 75}$$

= 5.9 mm = 6 mm

Thus the acceptable value of the thickness of the crown is 8 mm

3.3 Piston Rings:

Radial thickness of the piston rings

$$t_{rad} = D\sqrt{\frac{3.P_{rad}}{\sigma_t}}$$

where, P_{rad} = Radial pressure on the rings

= 0.025 N/mm^2 to 0.042 N /mm^2

σ = Allowable stress CI = 85 N/mm^2

$$\text{Thus } t_{rad} = 99\sqrt{\frac{3\times 0.025}{85}} = 2.4 say 4mm$$

Width of the ring, h = (0.7 to 1) t_{rad}

= 0.8 × 0.4 = 3.2 mm

$$\text{Number of rings (i)} = \frac{D}{10h} = \frac{68.5}{10\times 3.2} = 2.1 say 3$$

Let us adopt 3 compression rings and one oil ring.

Distance between the first ring grove and top surface

i.e, Top land (t_1) = 0.08 to 0.2 D

= 0.08 × 68.5 = 5.4 mm say 6 mm

Thickness of piston crown wall (t_3) = 0.05 to 0.1 D

= 0.1× 68.5

= 6.85mm say 7 mm

Piston inner diameter Di = D - 2 (t_3 + t_{rad} + Δt)

Where Δt = Radial clearance of ring

= 0.7 to 1.1 mm

= 0.8 mm

D_i = 68.5 - 2 (7 + 4 + 0.8)

= 44.9 mm, say 45 mm

Thickness of the skirt wall

t_2 = 2 to 5 mm, say 5 mm

3.4 Piston pin :

Load on piston due to gas pressure $= \frac{\pi}{4}(D)^2 \times P_{max}$

$= \frac{\pi}{4}(68.5)^2 \times 4.4$

$= 16215$ N

Load on piston due to bearing pressure $= P_{b1} \times d_0 \times l_1$

P_{b1} = bearing pressure at the bushing of small end of connecting rod.

= 25 MPa

l_1 = Length of piston pin in the bush of the small end of the connecting rod.

= 0.45 D

= 0.45 × 68.5

= 30.8 mm say 31 mm

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

Length lp = 0.9D = 61.65mm = 62mm

Force on piston = resisting force

$$\frac{\pi}{4}(D)^2 \times P_{max} = P_{b1} \times d_0 \times l_1$$

$$16215 = 25 \times d_0 \times 31$$

d_0 = 20.9 mm say 21 mm

d_i = 0.6 × d_0

= 12.6 mm say 13 mm

We know that $\sigma_b = \frac{M}{Z}$

$$M = \frac{Pmax \times D}{8}$$

$$= \frac{16215 \times 68.5}{8}$$

M = 138840 N-mm

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

$$= \frac{\pi}{32}\left[\frac{21^4 - 13^4}{21}\right]$$

Z = 775.6 mm^3

$$\sigma_b = \frac{M}{Z}$$

$$\sigma_b = \frac{138840}{775.}$$

σ_b = 179 MPa

Inducted shear stress, $\tau = \frac{2Pmax}{\pi(d_0^2 - d_i^2)}$

$$= \frac{2 \times 16215}{\pi(21^2 - 13^2)}$$

= 38 N/mm^2

Which is in permissible limit.

3.5 Connecting Rod

Data :

Diameter of piston = 68.5mm

Mass of reallocating parts = 2kg

Length of connecting rod = 325 mm

Stroke = 72mm

Speed = 2500rpm

Compression ratio = 12 : 1

Marcimum explosion plessure p_{max} = 3.5 M pa

I – section is the most suitable section

Width of flange, B = 4t

Height of I-section, M = 5t

Where, t : Web thickness

The maximum gas force,

$$P_{gas} = \frac{\pi}{4} D^2 \times P_{\max}$$

$$= \frac{\pi}{4} (68.5)^2 \times 3.5$$

= 12898 N

According to the Ranking formula

$$F_{cr} = \frac{\sigma_{cr} A}{1 + a\left[\frac{l}{k}\right]^2}$$

Taking factor of safety = 5

F_{cr} = P_{gas} × 5 = 12898 × 5 = 64490 N

Assuming σ_{cr} = 460 N mm^2

$$a = \frac{1}{6250}$$ (for both ends hinged)

I_{xx} = A. K^2_{xx}

A = 4.t^2

Thus,

$$134709 = \frac{460 \times 10^6 \times 11t^2}{1 + \frac{1}{6250}\left[\frac{0.325^2}{3.18t^2}\right]}$$

t = 5.6 mm Say 6 mm

3.6 Crank Shaft :

Data :

Maximum explosion pressure = 3.5 N/mm^2
Engine Speed = 2500 rpm
Brake Power = 26.1 KW
Maximum Force on bearing = 150 KN
Maximum force on the crank pin,

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

$$150 \times 10^3 = \frac{\pi}{4} D^2 \times 3.5$$

Diameter of bearing D = 233 mm

$\frac{l}{d}$ ratio for overlung crank shaft = 1 .1

There fore length of bearing = 256mm

IV. RESULTS AND DISCUSSION

The Table 1 indicates the proposed materials and manufacturing methods of various engine components.

Table 1 MATERIALS AND MANUFACTURING METHODS

S.No.	COMPONENT	MATERIAL	MANUFACTURING METHOD
1.	Cylinder Block	Cast Iron	Casting
2.	Cylinder Head	Aluminium	Casting
3.	Piston	Aluminium	Casting
4.	Piston Rings	Cast Iron	Casting
5.	Gudgeon Pin	Medium carbon steel	Turning
6.	Connecting Rod	Cast Steel	Casting
7.	Crank Shaft	Cast Steel	Casting
8.	Inlet valve	NiChrome Steel	Forging
9.	Out let Valve	SiChrome Steel	Forging

4.2 ENGINE SPECIFICATIONS:

Brake Power = 35HP = 26.1 KW
Speed = 2500rpm
Indicated mean effective pressure = 0.55MPa
Mechnical efficiency = 85%
Indicated power = 30.7 KW

4.3 CYLINDER DIMENSIONS:

MATERIAL : Cast Iron
Cylinder Bore = 68.5 mm
Stroke Length = 72mm
Thickness of cylinder with reboring allowance = 6mm
Length of the Cylinder = 90mm
Thickness of cylinder block wall = 6mm
Thickness of the cylinder flange = 8mm
Thickness of the cylinder head =6 mm

4.4 PISTON DIMENSIONS:

MATERIAL: Aluminium
Diameter = 68.5mm
Thickness of piston crown = 8mm
Number of piston rings = 3
Radial thickness = 4mm
Width of ring = 3.2mm
Piston pin out side diameter = 21mm
Inside diameter = 13mm
Length = 62mm

4.5 CONNECTING ROD:

Thickness of web = 6mm
Small end bearing diameter = 21mm
Big end bearing diameter = 235mm

4.6 CRANK SHAFT:

Diameter of bearing = 233mm
Length of bearing = 256mm

IV. CONCLUSIONS

Finally calculated various dimensions of all most all components of a Petrol engine such as Cylinder, Piston, Connecting rod, Crank shaft etc. using scientific principles and empirical relations and also tested the strengths of these components against their allowable stresses. The results with the common specifications of the engines were compared with the engines available in the market for their validation. The results obtained are found satisfactory.

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