Design of Diesel Engine Components

Power plant suitable to TATA nano car

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Abstract : The major problem is its low performance petrol engine. I wish that by replacing the petrol engine with high performance diesel engine the car can decrease the running cost and also can compete with most of the small cars in the market.The objective of the project is to design two cylinder, Compression Ignition, water cooled engine that can enhance the performance of the Nano car.In this project various components of Diesel engine like, Cylinder, Piston, Piston pins, Piston pin, Connecting rod and Crankshaft are designed from theoretical design procedures.

Index Terms – Nano Car, Diesel Engine, Engine components, Engine design .

I. INTRODUCTION

As the name implies, the IC engines are those engines in which the combustion of fuel takes place inside the engine cylinder. The IC engines use either petrol or diesel as fuel. In Diesel engines (also known as CI engines), only air is supplied to the engine cylinder during suction stroke and it is compressed to a very high compression ratio and there by raises its temparature from 600^{0}C to 1000^{0}C. The fixed 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 IC engine may be completed either by the two strokes or four strokes of the piston. Thus, an engine which requires 2 strokes of the piston or one complete revolution of the crankshaft to complete cycle, is known as two stroke engine. An engine which requires to strokes of the piston or two complete revolutions of the crankshaft to complete the cycle, is known as 4-stroke engines. The 4- stroke diesel engines are generally employed in heavy duty vehicles such as buses, trucks, tractors, diesel locomotives and in earth moving machinery. But now a days the utilization of the diesel engines increasing with the modern technology introduced from the modern metal manufacturing technology in producing light weight metals and friction free materials. The injection system is also made simpler by distribution pump system which uses a rotary pump instead of a reciprocating pump as in earlier days Diesel engines. With these advancements the diesel engines became more robust, high torque and high speed engines and competing with petrol engines. Now-a-days the utilization of diesel engines increased in luxury cars and also sports cars. Recently the JCB Company (England based company) tested a sports car name Diesel max broke down the Formula-1 car record.

II. DESIGN

Components of an engine are designed manually by using mathematical relations of design data book and by knowing the loads acting on the engine components and material properties of a materials used for the components.

2.1 Design of cylinder

Data Assumed:

1. Brake Power = 36HP =26.8 KW
2. Speed = 3000rpm
3. Indicated mean effective pressure = 0.7MPa
4. Mechnical efficiency = 80%
5. Number of Cylinders = 2

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

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

$$= \left[\frac{26.8}{0.8}\right] = 33.5KW$$

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

p_m = Mean effective pressure = 700 KPa
L = Strake Length = 1.25 D
D = Bore of the cylinder
Let the number of cylinders be 2, then IP of each cylinder is 16.75 KW

There fore, 16,750 = 750 × 1.25D × $\frac{\pi}{4} D^2 \times \frac{3000/2}{60}$

Then, D= 99 mm, L= 124mm

The maximum explosion pressure P_{max} = 8 × P_m = 8 × 700= 5600 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

$$t = \left[\left[\frac{\sigma + (1-M)p_{max}}{\sigma - (1+M)p_{max}}\right]^{0.5} - 1\right]$$

$$t = \left[\left[\frac{60 + (1-0.21)5.6}{60 - (1+0.21)5.6}\right]^{0.5} - 1\right]$$

= 4.95 mm Say 6mm

(i) The hoop stress produced in the cylinder, $\sigma_\theta = \frac{P_{max}.D}{2t} = \frac{5.6 \times 99}{2 \times 6} = 46.2 N/mm^2$

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

E = N/mm^2, α = 11 × 10^{-6} mm/^{0}c, ΔI = 120^{0}C

$$\sigma_{th} = \left[\frac{1 \times 10^5 \times 11 \times 10^{-6} \times 120}{2(1-0.21)}\right] = 83.5 \text{ N/mm}^2$$

Total stress, $\sigma_\theta + \sigma_{th}$ = 46.2 + 83.5 =129.7 N/mm^2

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

(iii) Longitudinal tensile stress $\sigma_\theta = \frac{P_{max}.D}{4t} = \frac{5.6 \times 99}{4 \times 6} = 23.1 N/mm^2$

(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

Therefore R_{max} =4310 N

Length of the cylinder= 1.25 × stroke length = 1.25 × 124 = 155 mm

(v)Bending stress induced in the cylinder, ' σ_b' = $\frac{M}{Z}$

where M = Bending moment = R_{max} × $\frac{ab}{a+b}$

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

a = 50 mm, b = 70 mm

$$M = 430 \times \frac{50 \times 70}{50 + 70} = 1257083 N - mm$$

Z : section modulus

Z = $\frac{\pi}{32}\left[\frac{D_o^h - D^h}{D_o}\right]$, Where D_o = D + 2t = 99 + 2 × 6 = 111 mm

Z = $\frac{\pi}{32}\left[\frac{114^4 - 99^4}{111}\right]$ = 49,306 mm^3

$\therefore$ Bending Stress $\sigma_b = \frac{M}{Z} = \frac{1257083}{49306} = 2.55 N/mm^2$

Total tensile stress = $\sigma_L + \sigma_b$ = 23.1 + 2.55 = 25.61 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 × 99) + 2
= 6.5 mm = say 7mm

(ii) Thickness of the cylinder flange, t_2 = 1.3 t_l = 9 mm

Thickness of the cylinder head

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

$$= 99\left[\frac{0.162 \times 5.6}{60}\right]^{0.5}$$

= 12.17 mm Say 13 mm

2.2 Design of Piston:

Data Assumed

Piston Material = Cast Iron
Piston Dia meter = 99mm
Length of strake = 124 mm
L/r ratio = 4
Mean effective pressure (P_m) = 0.7 N/ mm^2
Bsfc : Brake specific fuel consumption = 0.26 kg / KWh
Heat anducted through the piston crown = 10% of heat generated during embustion
Ev: colorfic value of the fuel = 42 MJ / Kg

1. Thickness of piston crown on the basis of strength

t = $CD\sqrt{\frac{P_{max}}{\sigma_t}}$

C = 0.43

D = 99

P_{max} = 8 × mep

= 8 × 0.7 = 5.6 N/mm^2

σ_t = 50 N/mm^2 for C.I.

Hence, t = 0.43 × $99\sqrt{\frac{5.6}{50}}$

= 14.25 mm, say 15 mm

2. Thickness of piston crown on the basis of heart dissipation

t = $\frac{D^2 q}{1600k(T_c - T_e)}$

where q = heat flow from the gases J/S – m^2
k = Heat conducting of piston material = 460 J/S m^2 . C/mm length of CI
T_c - T_e = 220 for CI
Fuel exsumption = bsfc × BP = 0.26 × 13.4 = 3.484 ks /hr
Heat supplied = fuel consumption × C.V

$$= \frac{3.484 \times 42 \times 10^6}{3600} = 40646 J/Sec$$

Heat conducted through crown = 10% of heat supplied
= 0.1 × 40646
= 4064.6 J/sec

Heat flow rate (q) = $\frac{Heatconducted}{Crosssectionarea} = \frac{4064.6}{\frac{\pi}{4}\times(0.099)^2} = 528kj/s-m^2$

Thickness of the Crown (t)= $\frac{D^2.q}{1600k(T_c - T_e)} = \frac{(99)^2 \times 528000}{1600\times 460\times 220}$ = 32 mm

Thus the acceptable value of the thickness of the crown is 32 mm

2.3 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 for 4- s Diesel engine
σ = Allowable stress CI = 85 N/mm^2

Thus t_{rad} = $99\sqrt{\frac{3\times 0.025}{85}} = 2.94 say 4mm$

Width of the ring, h = (0.7 to 1) t_{rad}
= 0.8 × 0.4 = 3.2 mm

Number of rings (i) = $\frac{D}{10h} = \frac{99}{10\times 3.2} = 3$

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 × 99 = 8 mm

Thickness of piston crown wall (t_3) = 0.05 to 0.1 D
= 10mm

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 = 99-1 (10 + 4 + 0.8)
= 69.4 mm, say 70mm

Thickness of the skirt wall
t_2 = 2 to 5 mm, say 5 mm

2.4 Piston pin :

Inducted bearing pressure P = $\frac{P}{d_{ps}\times l_s}$

i.e. P = $P_{br} \times d_{ps} \times l_s$
Gas force P = P_{max} × Area

= 5.6 × $\frac{\pi}{4}(99)^2$

= 43107 N

Assuming $\frac{l_s}{d_{ps}}$ = 1.5

Gas pressure = 20 N/mm^2

43107 = $20 \times 1.5\,(d_{ps})^2$

There fore, outside diameter of piston pin

d_{ps} = 38 mm

Inside diameter, d_i = $0.6 \times d_{ps}$ = 23 mm

Length, l_p = 0.9D = 90mm

Distance between base ends, b = length of small end of connection rod + clearance

= $1.5\, d_{ps}$ + h

= $1.5 \times 38 + 4$

= 61 mm

Bending moment M $= \frac{pb}{4} - \frac{p}{2}\left[\frac{b}{2} + \frac{l_p - b}{6}\right]$

$= \frac{43107 \times 61}{4} - \frac{43107}{2}\left[\frac{61}{2} + \frac{90-61}{6}\right]$

= -104175 N-mm

Section Modulus, Z = $\frac{\pi}{32}\left[\frac{38^4 - 23^4}{38}\right]$

= 4.664 mm^3

Bending stress, σ_b = $\frac{M}{Z}$ = -22.33 N/mm^2

This is the permissible value

The piston pin may fail in double shear

Inducted shear stress, $\tau = \frac{2P}{\pi\left(d_{ps}{}^2 - d_i{}^2\right)}$

$= \frac{2 \times 43107}{\pi\left(38^2 - 23^2\right)}$

= 30 N/mm^2

Which is in permissible limit.

2.5 Connecting Rod:

Data :

Diameter of piston = 99mm

Mass of reciprocating parts = 2kg

Length of connecting rod = 325 mm

Stroke = 124mm

Speed = 3000rpm

Compression ratio = 20 : 1

Maximum explosion pressure , 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} = $\frac{\pi}{4} D^2 \times P_{max}$

= $\frac{\pi}{4}(99)^2 \times P_{max}$

= 26942 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} = $P_{gas} \times 5 = 26942 \times 5 = 134709$ 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,

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

t = 5.6 mm Say 6 mm

2.6 Crank Shaft :

Data :

Maximum explosion pressure = 3.5 N/mm^2

Engine Speed = 300 rpm

Brake Power = 13.5 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 = 235 mm

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

There fore length of bearing = 260 mm

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

III. RESULTS AND DISCUSSION

3.1 ENGINE SPECIFICATIONS:

2 Cylinder 4-Stroke Compression Ignition Water Cooled Engine

Brake Power	= 36HP = 26.8 KW
Speed	= 3000rpm
Indicated mean effective pressure	= 0.7MPa
Mechnical efficiency	= 80%
Indicated power	= 33.5 KW

3.2 CYLINDER DIMENSIONS:

MATERIAL : Cast Iron

Cylinder Bore	= 99 mm
Stroke Length	= 124mm
Thickness of cylinder with reboring allowance	= 6mm
Length of the Cylinder	= 155mm
Thickness of cylinder block wall	= 7mm
Thickness of the cylinder flange	= 9mm
Thickness of the cylinder head	= 13mm

3.3 PISTON DEMENSIONS :

MATERIAL: Cast Iron

Diameter	= 99mm
Thickness of piston crown	= 32mm
Number of piston rings	= 3
Radial thickness	= 4mm
Width of ring	= 3.2mm
Piston pin out side diameter	= 38mm
Inside diameter	= 23mm
Length	= 90mm

3.4 CONNECTING ROD:

MATERIAL: Forged Steel

Thickness of web	= 6mm
Small end bearing diameter	= 38mm
Big end bearing diameter	= 235mm

3.5 CRANK SHAFT:

MATERIAL: Cast Steel

Diameter of bearing	=235mm
Length of bearing	= 260mm

IV. CONCLUSIONS

The main components of the Diesel engine which is intended to replace the existing petrol engine of TATA nano car are designed theoretically. I hope this Diesel engine can be the better alternative to the existing petrol engine and this Diesel engine can perform equivalently tothe petrol engine and gives better mileage.

Brake Power	= 26.8 KW (36HP)
Speed	= 3000rpm
Indicated mean effective pressure	= 0.7MPa
Mechnical efficiency	= 80%
Indicated power	= 33.5 KW

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