Design, Optimization and Fatigue Life Estimation of Diesel Engine Piston

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Abstract---Engine pistons are one of the most complex components among all automotive and other industry field components. The engine can be called the heart of a vehicle and the piston may be considered the most important part of an engine. There are lots of research works proposing, for engine pistons, new geometries, materials and manufacturing techniques and this evolution has undergone with a continuous improvement over the last decades and required thorough examination of the smallest details. Notwithstanding all these studies, there are a huge number of damaged pistons. Damage mechanisms have different origins and are mainly wear, temperature, and fatigue related & cannot be repaired easily. The fatigue related piston damages play a dominant role mainly due to thermal and mechanical fatigue, either at room or at high temperature. The paper describes the Design, Optimization (in terms of Stress & Weight Reduction) and Fatigue Life Estimation of four stroke diesel engine piston. The main objective is to investigate the Stress Distribution, Temperature Variation, deformation & Fatigue Life estimation. In this work the FEA technique to predict the higher stress and critical region on the component. The optimization will be carried out to reduce the stress concentration on the piston using ANSYS. Fatigue life of the piston will be predicted using constant amplitude S-N data for the material used.

Keywords: Stresses, Deformation, Finite element method, Fatigue life, I.C Engine

I. INTRODUCTION

Internal combustion engines are those engines in which the combustion of the 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 carburetor and fed to engine cylinder where it is ignited by means of spark produced by the spark plug.

In Diesel engines also called compression ignition engines or C.I engines, only air is supplied to the engine cylinder during the suction stroke and it is compressed to a very high pressure, thereby raising the temperature from 600 degree centigrade to 1000. The desired quantity of fuel is now injected into the engine cylinder in the form of very fine spray and gets ignited when it 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.

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

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, tractors, diesel locomotives and in earth moving machinery [1].

Some notations Used are

- D= Diameter of Piston = 100mm
- Speed = n = 1000rpm
- Brake Power = 50Bhp
- Mechanical Efficiency = 0.8
- Indicated Power = IP = 45937.5Watts
- P_m = Mean Effective Pressure
- T_h = Thickness of the head
- P = Maximum load on Piston
- R = Side Thrust on the Piston
- t_1 = Radial thickness of rings
- t_2 = Axial Thickness of rings
- t_3 = Thickness of the barrel
- P_b = Bearing pressure = 1.65N
- d_0 = Outside diameter of piston pin
- d_i = Inner diameter of Piston Pin
- p_w =Pressure of gas on the cylinder wall= 0.042 N/mm^2.
- p = Working bearing pressure = 14.7MPa

A. *Parts Of I.C Engine:*

The principle parts of I.C engines, as shown in Fig 1 are as follows:

- Cylinder and cylinder liner
- Piston, Piston rings and piston pin
- Connecting rod with small and big end bearing
- Crank, crankshaft and crank pin and
- Valve gear mechanism.

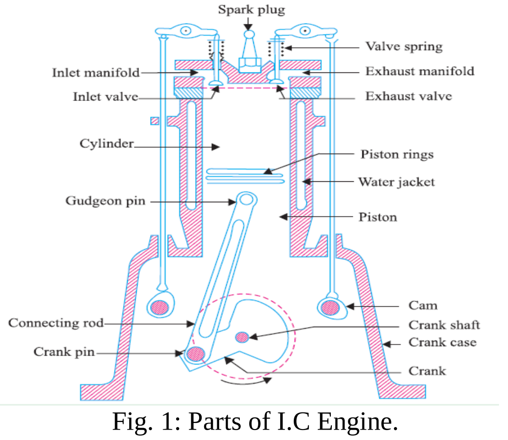

Fig. 1: Parts of I.C Engine.

B. Piston:

Piston is a disc which reciprocates within the 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.

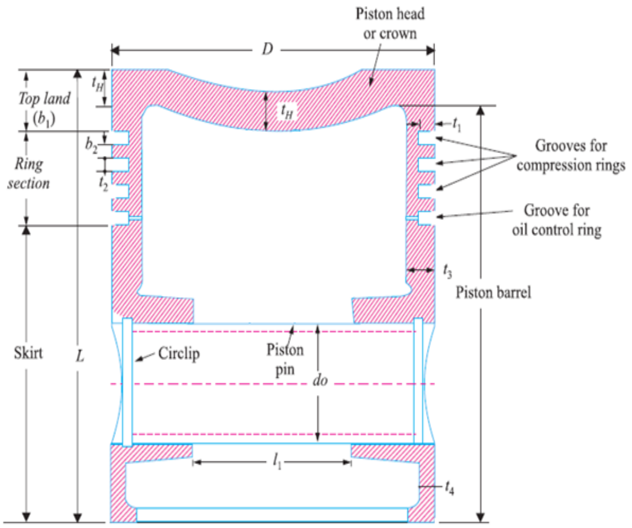

Fig 2: Parts of Piston.

The piston of internal combustion engines are as shown in figure 2. The piston consists of the following parts:

- Head Or Crown
- Piston Ring
- Skirt
- Piston Pin

C. Fatigue Loading:

Fatigue loading is primarily the type of loading which causes cyclic variations in the applied stress or strain on a component. Thus any variable loading is basically a fatigue loading.

D. Fatigue Failure Mechanism:

Fatigue failure begins with a small crack; the initial crack may be so minute and cannot be detected. The crack usually develops at a point of localized stress concentration like discontinuity in the material, such as a change in cross section, a keyway or a hole. Once a crack is initiated, the stress concentration effect become greater and the crack propagates. Consequently the stressed area decreases in size, the stress increase in magnitude and the crack propagates more rapidly. Until finally, the remaining area is unable to sustain the load and the component fails suddenly. Thus fatigue loading results in sudden, unwarned failure.

E. Fatigue Failure Stages:

Thus three stages are involved in fatigue failure namely

-Crack initiation

-Crack propagation

-Fracture

1) *Crack Initiation:* Crack are generally originated from a geometrical discontinuity or metallurgical stress raiser like sites of inclusions

2) *Crack Propagation:* As a result of the local stress concentrations at these locations, the induced stress goes above the yield strength (in normal ductile materials) and cyclic plastic straining results due to cyclic variations in the stresses. On a macro scale the average value of the induced stress might still be below the yield strength of the material.

This further increases the stress levels and the process continues, propagating the cracks across the grains or along the grain boundaries, slowly increasing the crack size.

As the size of the crack increases the cross sectional area resisting the applied stress decreases and reaches a thresh hold level at which it is insufficient to resist the applied stress.

3) *Final Fracture:* As the area becomes too insufficient to resist the induced stresses any further a sudden fracture results in the component [4].

II. DESIGN OF DIESEL ENGINE PISTON

The following design considerations are essential in the design of the piston,

- It should have enormous strength to withstand the high pressure gas and inertia forces.
- It should have minimum mass to minimize the inertia forces.
- It should form an effective gas and oil sealing of cylinder.
- It should provide sufficient bearing area to prevent the undue wear.
- It should disperse the heat of combustion quickly to the cylinder wall.
- It should have high speed reciprocation without noise.
- It should be of sufficient rigid construction to withstand thermal and mechanical distortion.
- It should have sufficient support for the piston pin [1].

Properties of Aluminium alloy are:

- Young Modulus = $70000 N/mm^2$
- Poisson Ratio = 0.35
- Thermal Conductivity = 237W/m/° K
- Linear Coefficient of expansion = $23.1\, e^{-6}$/° K
- Density=$2700 kg/m^3$ [2][3]

Assuming the Brake Power as 50BHP and the diameter of the piston as 100mm and rpm as 1000rpm for 4 stroke engine the mean effective is calculated using the Relation

$D^2= (1000*4\, IP) / ((\pi/4)*P_m*n)$

$P_m = 4 \times 1000 \times 45937.5/(\pi/4 \times 100^2 \times 1000)$

$= 2.34 N/mm^2$

Thickness of piston head using pressure

$T_h=\sqrt{(3P_m \times D^2/ (16 \times \sigma))}$ (Grashoff's formula)

$T_h = 6.05$ mm

Design of Ring Section

Radial thickness of rings,

$t_1=\sqrt{(D^2 \times 3P_w/\sigma)}$ =3.24mm

Width of top land, $b_1=T_h$ to $1.2 \times T_h$ =7.26mm

Minimum axial thickness of rings t_2

$t_2=D / (10 \times n_r)$

Where 3 is the number of rings

t_2= 3.33mm

Width of other ring land, $b_2= 0.75\, t_2 = t_2$ =3.33mm

Maximum thickness of the barrel t_3

$t_3= (0.03 \times D) + t_1+ 4.9$mm =11.14mm

Design of Piston Skirt

Max load on piston,

$P = (\pi/4) \times D^2 \times P_{max} = 165375$ N
Max side thrust on cylinder,
$R = P/10 = 16537.5$ N
The side thrust is given by,
Side thrust R = Bearing Pressure x Projected area of the piston skirt $= P_b \times D \times L_s$
L_s = Side Thrust / ($P_b \times D$) = 100.25mm
Total length of the piston in mm is given by,
Sum of Length of skirt (L_s) + Length of ring section + Top Land (b_1) $= L_s + (3xt_2 + b_2) + b_1 = 124.16$mm.
Design of piston pin
Inner diameter of piston pin,
$d_i = \sqrt{(\text{max load}/(p \times i)} = 27.39$ mm
Where Maximum load is the side thrust in N
i is length to diameter ratio = 124.14/100 = 1.25
Outer diameter of piston pin, $d_i = 0.6 \times d_o$
$d_o = 45.64$mm
Length of the piston pin in the bush of the small end of the connecting rod in mm is given by

$$L = 0.45 \times D = 0.45 \times 100 = 45\text{mm [1]}$$

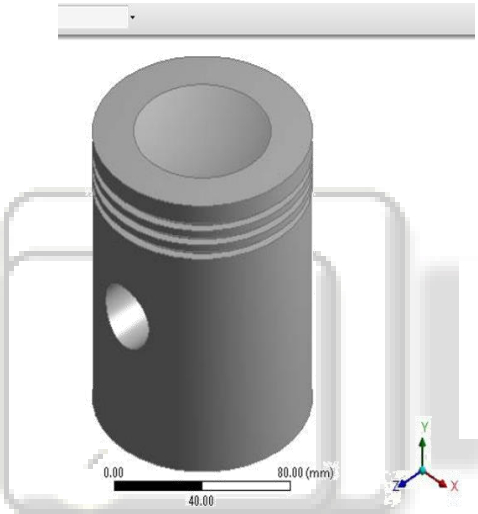

Fig. 3: Piston Model

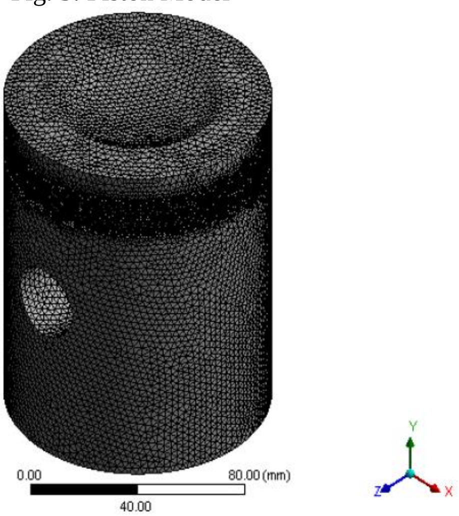

Fig. 4: Tetrahedral Mesh.

Boundary Conditions are

- Maximum Temperature at the top of the piston Head & Piston Rings
- Frictionless Support at the piston pin area
- Maximum Pressure at the head of the Piston.

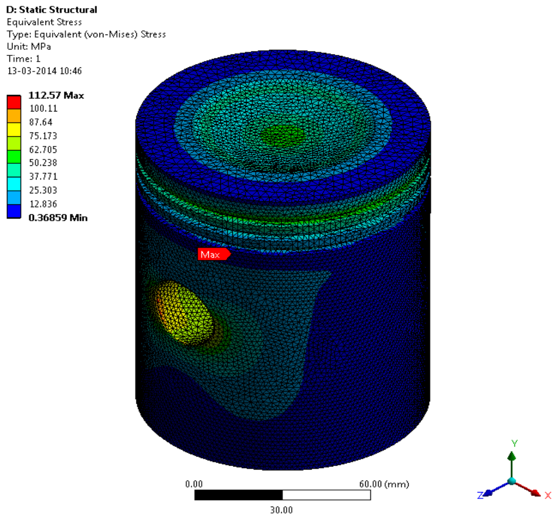

Fig. 5: Stress Distribution.

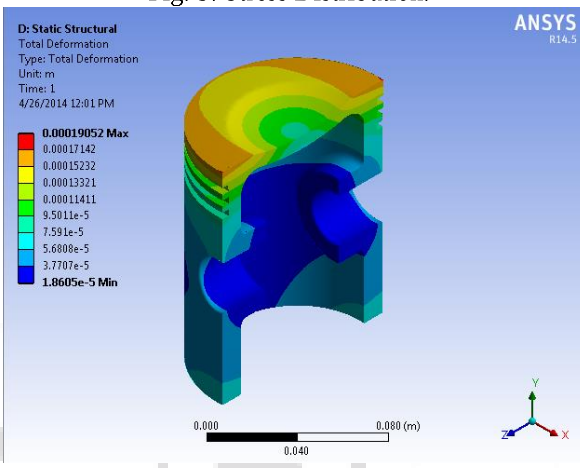

Fig 6: Deformation.

Analytically the stresses & deformation are calculated as shown below,
Deformation = Coefficient of Thermal Expansion x Change in Temperature (T_C-T_E) x Length
= 0.0000237 x 75 x 124.14 = 0.00035
Equivalent Stresses = Young's Modulus x Thermal Expansion
= 70000 x 0.0000237 x 75
= 123.14MPa

The variation of the stresses is determined by varying the Fillet radius and the diameter of the Piston pin diameter and are shown in Figure 7 and Figure 8 respectively.

Table. 1: Variation of Stresses with the variation of Fillet radius.

Table of Design Points

	A	B	C	D	E	F
1	Name	P1 - fillet	P2 - Tem... Max...	P3 - Equivalent Stress Maximum	P4 - Total Deformation Maximum	P5 - Geom... Mass
2	Units		C	Pa	m	kg
3	DP 9	0.5	275	1.076E+08	0.00025601	1.2458
4	Current	1	275	1.1257E+08	0.00019052	1.2456
5	DP 1	1.5	275	1.0857E+08	0.00017816	1.2453
6	DP 2	2	275	1.1292E+08	0.00022889	1.2448
7	DP 3	2.5	275	1.1803E+08	0.000195	1.2442
8	DP 4	3	275	1.1356E+08	0.00019082	1.2435
9	DP 5	3.5	275	1.3188E+08	0.00026564	1.2425
10	DP 6	4	275	1.2871E+08	0.0001696	1.2414
11	DP 7	4.5	275	1.2918E+08	0.00018447	1.2401
12	DP 8	5	275	1.3688E+08	0.00019962	1.2387

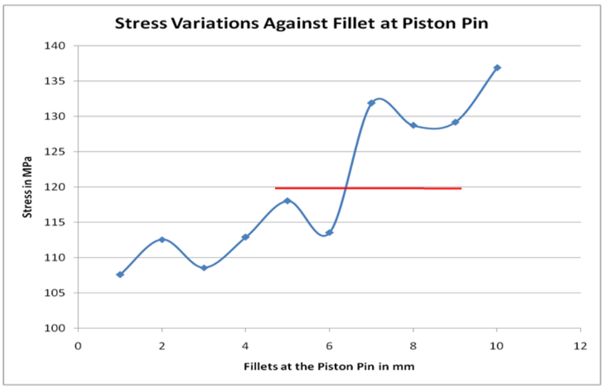

Fig. 7: Variation of Stresses with the variation of Fillet radius.

Table. 2: Variation of Stresses with the variation of Pin Diameter.

Table of Design Points

	A	B	C	D	E	F	G
1	Name	P1 - fillet	P6 - PD	P2 - Tempera... Maximum	P3 - Equivalent Stress Maximum	P4 - Total Deforma... Maximum	P5 - Geo... Mass
2	Units			C	Pa	m	kg
3	DP 9	1	25	275	1.0984E+08	0.00016968	1.2602
4	Current	1	27.39	275	1.1257E+08	0.00019052	1.2456
5	DP 1	1	30	275	1.0418E+08	0.00017183	1.2284
6	DP 3	1	35	275	1.0341E+08	0.00017965	1.1914

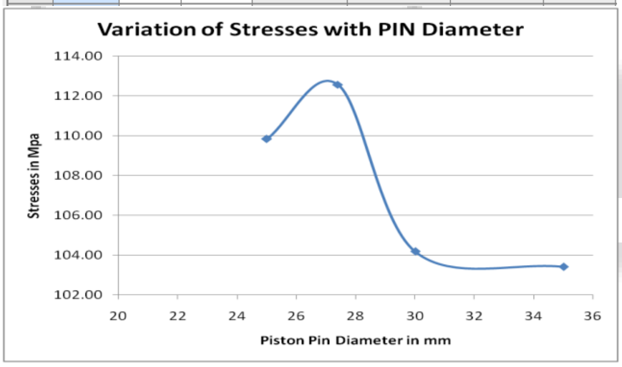

Fig. 8: Variation of Stresses with the variation of Pin Diameter.

III. PISTON OPTIMIZATION

Piston is optimized by providing the cut out in the stress free region at bottom skirt of piston.

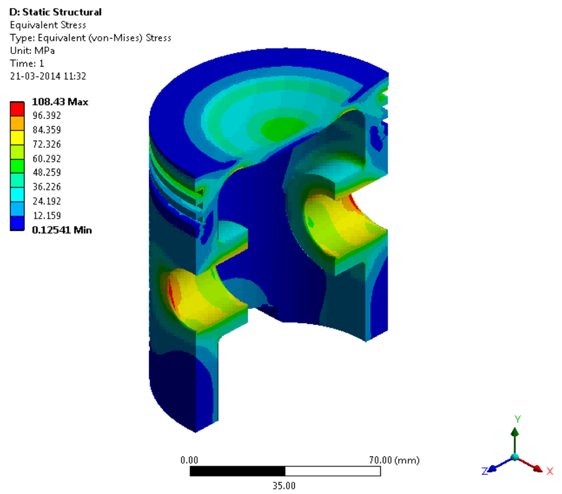

Fig. 9: Stress Distribution of optimized Piston.

Variation of Stresses with the variation of the fillet radius is shown in figure 10.

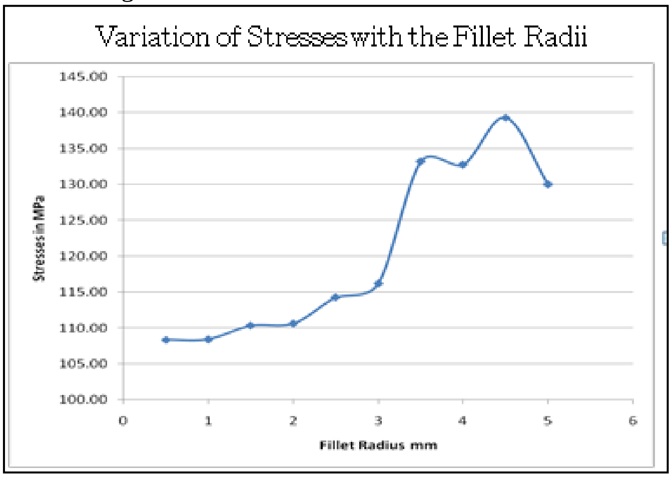

Fig. 10: Variation of Stresses with the variation of Fillet Radius.

IV. RESULTS

Theoretical, Stresses induced in the designed piston and Optimized piston are tabulated in table 3 below,

Table 3: Comparison of Results

	Theoretical Results	Designed Piston	Optimized Piston
Maximum Stress in Mpa	123.14	112.57	108.43
Deformation in mm	0.35	0.19	0.17
Weight in Kg	NA	1.25	1.22

V. FATIGUE LIFE ESTIMATION

Stress Amplitude $\sigma = \frac{(\sigma_{max}-\sigma_{min})}{2}$

The different correction factors are

- Size factor = C_{size} = $1.189(d)^{0.097} = 0.76$
- Load correction factor = 0.7
- Surface Correction factor = C_{Surf} = $A(S_{ut})^{b}$= $272\ (310)^{-0.995}$ =0.903
- Temperature factor = C_{temp} = 1
- Reliability Factor = C_R=1

Correction Factor = (Size factor x Load factor x Surface factor x Temperature Factor x Reliability Factor).

Correction Factor = 0.4808 [5].

Table 4: Number of Cycles for different Stress Levels from Ansys

Sl. No	Number of Cycles	Stress (MPa)
1	4.2×10^{6}	108
2	9×10^{4}	160
3	2.5×10^{4}	190
4	6.8×10^{3}	220

Table. 5: Number of Cycles for different Stress Level from S-N curve considering Stress Correction Factor.

SL.NO	MAX STRESS (MPa)	MIN STRESS (Mpa)	STRESS AMPLITUDE (MPa)	STRESS CONSIDERING CORRECTION FACTOR (MPa)	NF VALUE FROM S-N CURVE
1	108	0	59	122.9	7×10^{7}
2	160	0	80	166.6	6×10^{6}
3	190	0	95	197.9	7×10^{4}
4	220	0	120	250	2.5×10^{4}

Table. 6: Number from Cycles from S-N curve and FEA

Sl. No	Number of Cycles From Ansys	Number of cycle from S-N curve	Damage Fraction
1	4.2×10^6	7×10^7	0.06
2	9×10^4	6×10^6	0.02
3	2.5×10^4	7×10^4	0.4
4	6.8×10^3	2.5×10^4	0.27

The FEA cycles and the cycles from S-N curve are tabulated in table 6 and the damage fraction is determined.

Total Damage Fraction

$D = D_1 + D_2 + D_3 + D_4 = 0.06 + 0.02 + 0.4 + 0.27$

$D = 0.75 < 1$ and hence the crack is not initiated [4].

VI. CONCLUSION

In order to improve the piston performance, the piston is optimized to reduce the stress levels and also the weight of the piston. For the optimized piston the Fatigue Life is estimated and damage fraction is determined, and the values suggest that the design is safe.

VII. ACKNOWLEDGEMENT

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