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FINITE ELEMENT ANALYSIS OF A CNG CONVERTED DIESEL ENGINE PISTON FOR OPTIMUM DIMENSIONS

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ABSTRACT

This paper implies finite element analysis of piston groove over the piston face for optimum dimension of the groove. Diesel engine to CNG engine conversion procedure involves changing the compression ratio (CR) of the engine. Diesel engine runs with high compression ratio (CR= 16 to 24) while petrol or gas engine runs with lower compression ratio [CR= 9 to 12].Typically, this can be accomplished by increasing the clearance volume (V_c) of the engine by making a groove on the piston top. The present study entails such analysis by Finite Element Method (FEM) on the piston using a commercial FEA package. Displacement and stress analyses are performed on the piston for different combinations of groove depth and groove radius for the same amount of load. The optimum depth-radius combination is determined for a particular compression ratio (CR=11) and obtained stress is compared with that of a typical commercially available CNG converted piston. It is found that the optimum converted piston is about 29% safer than the commercially available one. Similar analyses are performed for CR=9, 10 & 12. For each case, optimum dimension of the piston groove geometry is determined.

Keywords: Compression Ratio, Finite Element Method, Stress Analysis.

1. INTRODUCTION

It is general practice to convert large diesel engine into gas engine run by CNG due to higher price of diesel fuel as compared to compressed natural gas (CNG). The converted engine saves typically more than 50% of the fuel cost. To convert diesel engine to CNG engine, the compression ratio (CR) of the engine needs to be changed. Diesel engine runs with higher compression ratio (CR= 16 to 24) while petrol or gas engines run with lower compression ratio (9 to 12) [1]. To accomplish this, groove is generally made on the top of the piston face. But the problem is to optimize the dimension of the groove. The optimized dimension of the groove offers the minimum stress in the piston under the pressure applied on the top face of the piston. In order to optimize the groove dimension of the piston, finite element analysis [2] is carried out on the piston using a FEA package.

Aluminium alloys are generally used in piston manufacturing. There are many reasons for using aluminium alloys in pistons for gasoline and diesel engines: low weight, high thermal conductivity, simplicity of production, high reliability and very good recycling properties. Basically two common types of pistons are available- cast piston and forged piston. Common alloys used in forged pistons are Al-4032 and Al-2618 alloy. Generally, Al-2024 alloy is used for cast piston. To calculate the fatigue safety factor a forged piston of Al-4032 alloy was considered. It has an ultimate tensile strength of 380 MPa and yield strength of 317 MPa. This alloy contains the following chemical composition: Si=12.2%, Cu=0.9%, Mg=1.05%, Ni = 0.9%, Al = Balance [3]. The piston material used for the finite element analysis comprises the modulus of elasticity, E = 70 GPa, density, ρ = 2700Kg/m^3 and Poisson ratio, σ = 0.35.

2. FINITE ELEMENT ANALYSIS

A CNG converted diesel engine piston was collected from an engine and 3-D model of the piston was generated using ANSYS. The 3-D model is shown in figure 1.

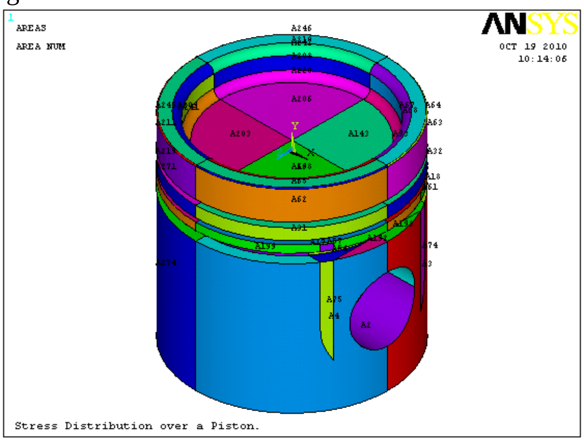


Fig 1. Model (unmeshed area view)

The groove dimensions are kept as variable parameter in the ANSYS script. There are many dimensional features in the inner side of the piston where gudgeon pin is located. For simplification, not all the features were considered in modeling of the piston. Skirt-off in the bottom part of the piston was also dispensed with the model as the stress level is insignificant in that region.

The piston model was meshed with 3-D quadratic element SOLID92. This element is generally used for irregular shaped body. Mesh sensitivity analysis [4] was performed to select the optimum mesh size. The optimum mesh size is selected based on 3 mesh sensitivity curves - 1) Maximum deformation, 2) Maximum tensile stress, 3) Maximum compressive stress vs no. of elements. Only mesh sensitivity curve 1 is shown in figure 2. Further analyses over the piston are performed based on the optimum mesh size. The meshed view of the piston model is shown in figure 3 with 70,000 mesh elements.

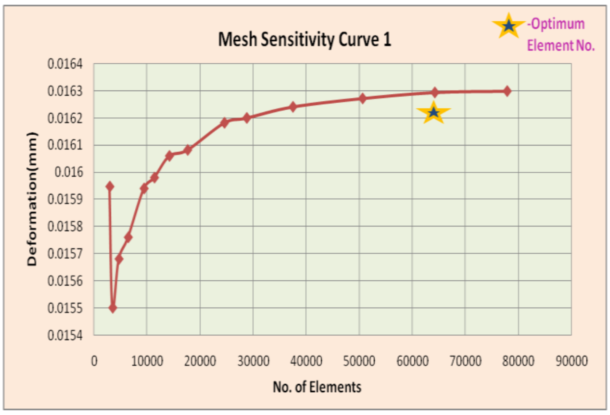


Fig 2. Variation of maximum deformation with No. of elements

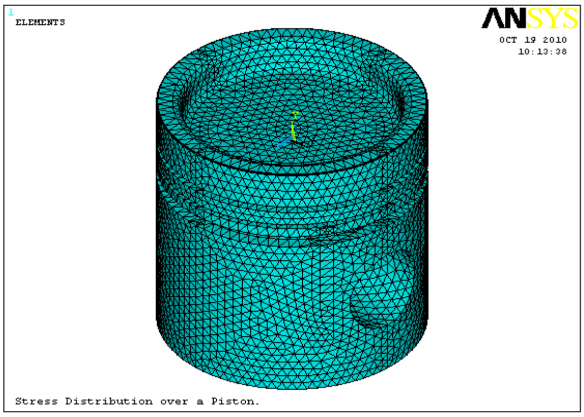


Fig 3. Meshed view of the piston model

For design purposes always critical conditions need to be taken into consideration. The peak pressure developed inside the combustion chamber of the engine during combustion was taken as the design load for the analysis. Uniform surface pressure was applied on the top surfaces of the piston. A typical piston is shown in figure 4. The gudgeon pin supports the piston against the peak pressure of the combustion chamber of the engine. There is a semi-cylindrical area contact between the piston and the gudgeon pin. This pin is tightly coupled with the piston and the connecting rod so that at static condition the piston could not move in either direction. Moreover, the gudgeon pin is locked in the piston by two snap rings on both sides of the pin.

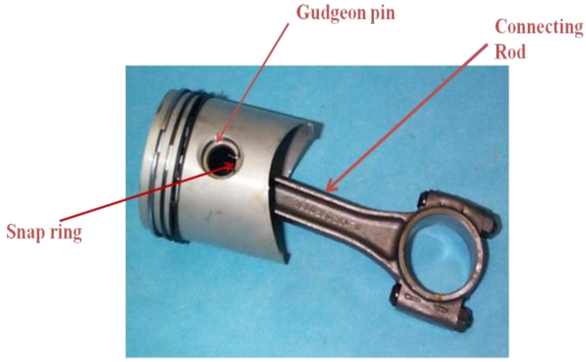


Fig 4. A piston with connecting rod

Considering all those things, fixed support is considered on the pin and piston contact area i.e. displacements Ux=0, Uy=0 and Uz=0 in the contact area. For simplicity of analysis, we assumed no peripheral constraint although there are some constraints at the peripheral side by the compression rings in between piston and cylinder. Typically, in a diesel engine peak pressure is around 60 bar (6 MPa) for a compression ratio 18. Figure 5 indicates the loading and boundary condition of the piston model.

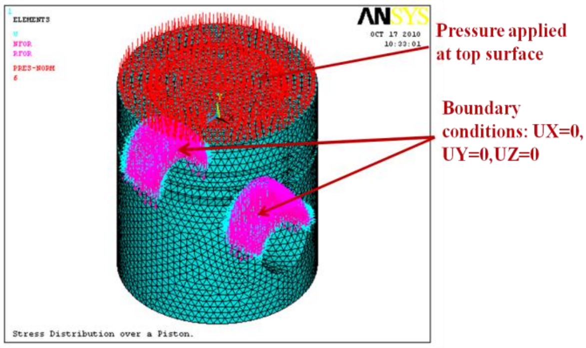


Fig 5. Piston model under load and constraint

Several methods of solving the system of simultaneous equations are available in the ANSYS program: sparse direct solution, Preconditioned Conjugate Gradient (PCG) solution, Jacobi Conjugate Gradient (JCG) solution, Incomplete Cholesky Conjugate Gradient (ICCG) solution, Quasi-Minimal Residual (QMR) solution, frontal direct solution, and an automatic iterative solver option (ITER) [5].

This problem contains around 70,000 elements and it is a three degrees of freedom system. So, sparse direct solver is used for the analysis because it is based on direct elimination of equations and it is a shared-memory parallel solver. It provides robustness and speed of the solution which is required for the analyses.

This procedure is repeated for various combinations of groove dimensions. For a particular compression ratio clearance volume of the engine cylinder is fixed. To attain this clearance volume groove radius and groove depth can be changed in a number of ways. For each case maximum stress developed in the piston is recorded from the analysis using ANSYS.

The obtained results are verified in a number of ways as this problem did not have any analytical solution. Verification consists of qualitative and quantitative methods. Solving processes are verified by performing a simple cantilever beam problem [6]. Qualitative verification includes:

(1) Displacement U_X, U_Y, U_Z should be zero at the support. From ANSYS, it is found that U_X=0, U_Y=0, U_Z=0 at the support. Figure 6 gives the illustration of the displacement distribution found from the analysis using ANSYS.

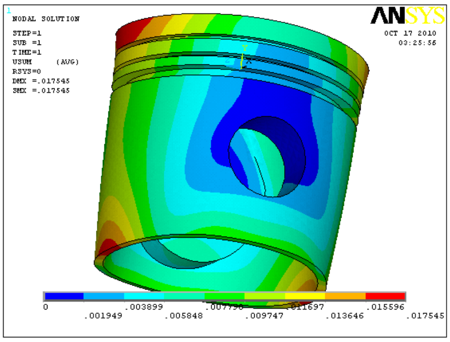


Fig 6. Displacement contour plot.

(2) Shear stress at the top surface should be zero [7]. From ANSYS, it is found that at top surface shear stress is almost zero i.e. value is negligible. Figure 7 gives the illustration of the shear stress distribution found from the analysis using ANSYS.

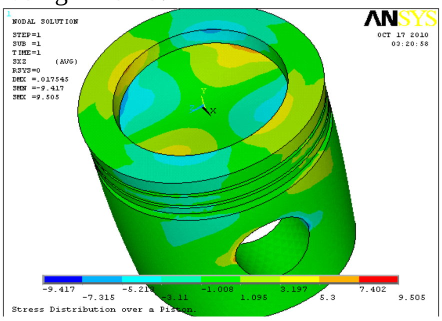


Fig 7. Shear stress contour plot.

(3) Displacements in the Y direction at all nodes should be negative [8]. The values of displacements obtained from the ANSYS are also negative. Figure 8 gives the illustration of the displacement distribution found from the analysis using ANSYS.

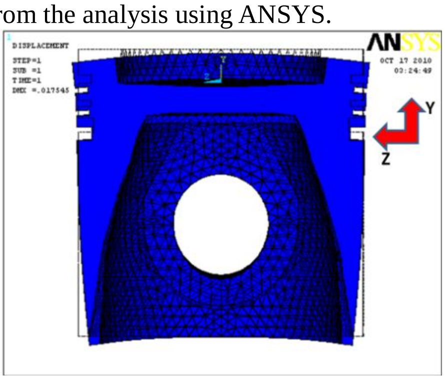


Fig 8. Displacement plot (capped view)

(4) After applying load on the piston, the nodal displacements of the piston should be symmetrical because supports are considered only at the piston pin. Figure 9 illustrates the symmetrical displacement distribution.

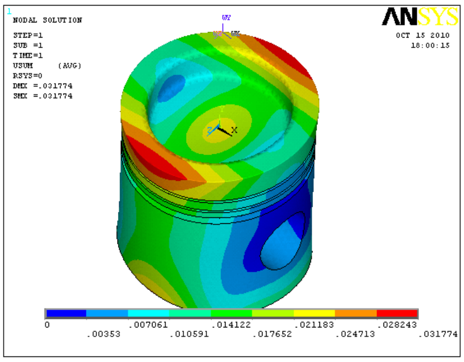


Fig 9. Symmetrical displacement distribution.

(5) Number of equations should be three times the number of nodes as it is a three degrees of freedom system. From the analysis it is also verified. For compression ratio 11 and optimum depth-

No. of nodes	No. of equations
104576	313728

Quantitative verification was performed by using a universal testing machine of Strength of Materials Laboratory of Mechanical Department, BUET. Dial gage was used for the displacement measurement. Figure 10 illustrates the experimental setup.

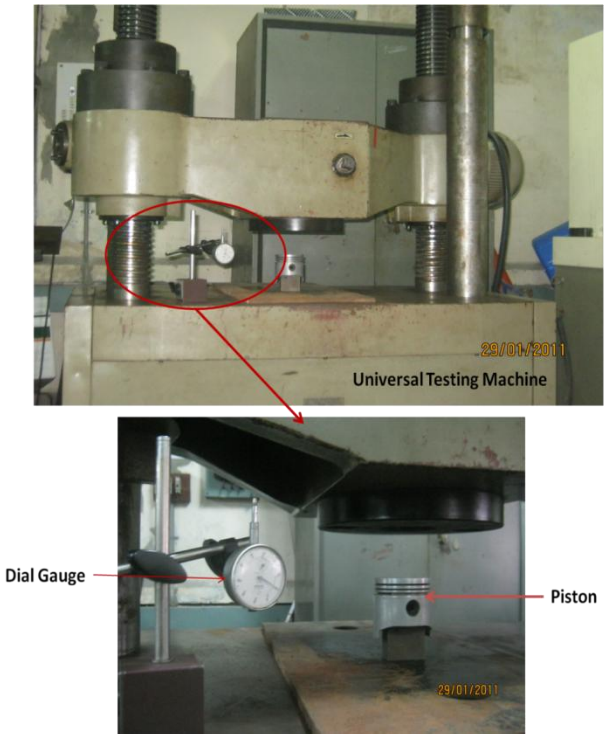


Fig 10. Experimental set up

A car engine piston with flat top surface was collected from the local market. The dimension of the piston was measured. The piston was modeled, meshed and boundary conditions are applied using ANSYS. By solving, the stresses and displacements at different nodes of the piston model was obtained. To compare the ANSYS results with experimental results the piston was tested in universal testing machine applying load and boundary conditions in similar manner in the ANSYS model. The differences between two results were found to be within 7.5% for different loads. Only displacement U_y of ANSYS result was compared with the experimental result.

3. RESULT AND DISCUSSION

The stress analyses are carried out following the two theories-a) Maximum shear stress theory and b) von Mises Hencky theory. The maximum-shear-stress theory predicts that yielding begins whenever the maximum shear stress in any element equals or exceeds the maximum shear stress in a tension test specimen of the same material when that specimen begins to yield. The MSS theory is also referred to as the Tresca or Guest theory. [9]

For the general state of stress, the maximum-shear-stress theory predicts yielding when-

$$\tau_{max} = \frac{\sigma 1 - \sigma 3}{2} \geq \frac{Sy}{2} \quad \text{or,} \quad \sigma 1 - \sigma 3 \geq Sy \qquad (1)$$

In ANSYS, σ_1-σ_2 is termed as stress intensity.

Von Mises Hencky theory or distortion-energy theory predicts that yielding occurs when the distortion strain energy per unit volume reaches or exceeds the distortion strain energy per unit volume for yield in simple tension or compression of the same material. [9]

For the general state of stress this theory predicts yielding when-

$$\left[\frac{(\sigma 1 - \sigma 2)^2 + (\sigma 2 - \sigma 3)^2 + (\sigma 3 - \sigma 1)^2}{2}\right]^{1/2} \geq Sy \qquad (2)$$

Using xyz components of three-dimensional stress, the von Mises stress can be written as

$$\sigma' = \frac{1}{\sqrt{2}}\left[\, (\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau^2_{xy} + \tau^2_{yz} + \tau^2_{zx})^2 \,\right]^{1/2} \qquad (3)$$

Maximum shear stress theory gives more conservative result than von Mises Hencky theory.

For a particular compression ratio (CR=11), stress analyses are carried out over the collected converted piston and maximum stress developed in the piston is determined for a particular groove dimension (h/r ratio). After that the groove dimensions are changed and each time maximum stress developed in the piston is recorded for the same CR. Load is kept same (60 bar) for each case. Although the peak pressure developed for CR=11 is not 60 bar, it is used only for comparison purpose. It eases the comparison between the original uncut piston and the converted cut piston. The results obtained from ANSYS are presented in table 1. Stress intensity, von Mises stress and deformation vary from node to node of the piston model. Only maximum value of each item is presented in table1.

Fatigue safety factor is calculated based on von Mises stress by using Modified Goodman equation [10]

$$\frac{\sigma a}{se} + \frac{\sigma m}{sut} = \frac{1}{n} \qquad (4)$$

Where, $\sigma a = \frac{\sigma max - \sigma min}{2}$

$\sigma m = \frac{\sigma max + \sigma min}{2}$

Table 1: Fatigue safety factors for different h/r ratios

Load = 60 bar and CR = 11(Converted Piston)				
Groove Depth h(mm)	Stress Intensity (MPa)	Von Mises Stress (MPa)	Deformation (mm)	Fatigue Safety Factor*
10	137.745	119.8	0.08743	0.881
11.4	97.363	86.289	0.046144	1.223
12.45	84.267	76.847	0.040901	1.371
13.67	84.3	76.959	0.037466	1.372
15.06	89.944	81.99	0.03463	1.288
17.61	98.18	91.623	0.031774	1.152
18.5	107.029	98.956	0.031067	1.067
20.86	112.942	102.621	0.035189	1.064
22.5	157.032	148.625	0.054137	0.71
23.54	203.479	185.122	0.069975	0.57
Load = 60 bar and CR = 18(Original Diesel Piston)				
uncut	71.68	62.208	0.020429	1.679

The maximum von Mises stress is found to be about 99 MPa for the collected CNG converted piston. After that, changing the groove dimension (changing the h/r ratio) over the piston face similar analysis is executed. Figure 6 epitomizes the variation of maximum stress intensity and von Mises stress with depth of groove so as to the radius of the groove.

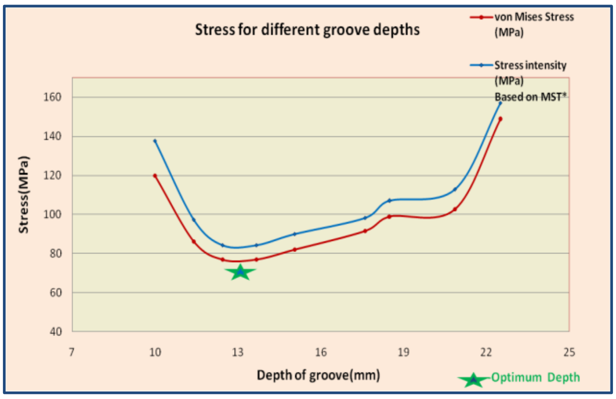
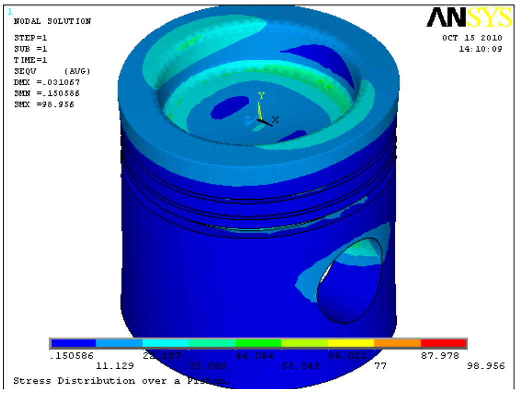


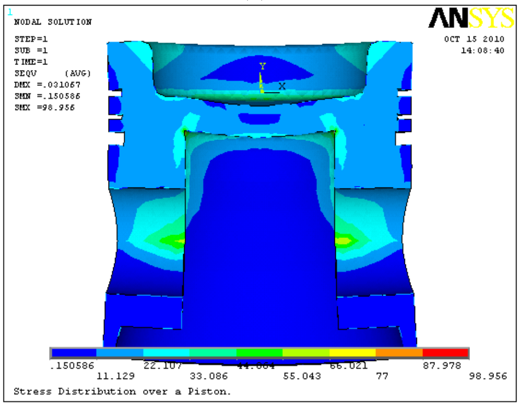
Fig 6. Stress vs depth of groove plot

The star indicated point exemplifies the optimum groove dimension as it confers the minimum stress developing situation.

Deformation and stress analyses were performed for each combination of groove dimension (h/r ratio). Analyses for commercially available converted piston and optimum converted piston are presented in this paper. von Mises stress distribution for collected converted piston is presented in figure 5(a & b).



(a)



(b)

Fig 5. von Mises stress distribution for collected CNG converted piston a) 3-D solid view b) capped view

The von Mises stress distribution for optimum CNG converted piston is represented in figure 7. This figure is also incarnate with most critical zone of the piston.

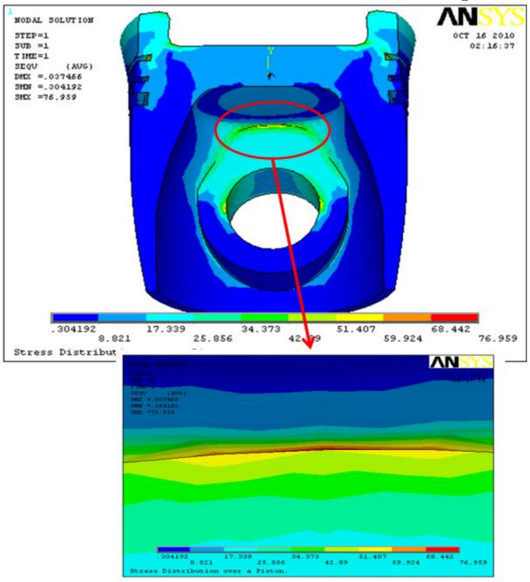


Fig 7. Stress distribution over the piston with most critical zone (cut view)

It is obtained that maximum von Mises stress for optimum converted piston is about 77 MPa which is around 22% lower than that of commercially converted piston.

Often machine members are found to have failed under the action of repeated or fluctuating stresses; yet the careful analysis reveals that the actual maximum stresses were below the ultimate strength of the material and quite even frequently below the yield strength. The most distinguishing characteristics of the failures is such the stresses have been repeated a very large number of times. Such failure is called fatigue failure [10]. And in running condition of an engine, the piston is subjected to fluctuating load from peak pressure during power stroke to almost zero pressure during the suction stroke. Moreover, high temperature is a common fact for the piston of an engine. Hence, fatigue safety factor is premeditated for both cases to anticipate the failure criteria. It is observed that fatigue safety factor is 1.067 for commercially converted piston where as optimum converted piston bears a fatigue safety factor of 1.372 for 60 bar load. Fatigue safety factor is about 28.6% higher in case of optimum CNG converted piston. As per study of diesel engine, common material used in diesel engine piston and temperature variation within the piston, 2200 rpm is selected as the optimum engine speed and maximum temperature of 250°C on the piston face [1]. Although there is a temperature variation all over the piston but fatigue safety factor is calculated assuming the maximum temperature.

Table 2: Safety factors for converted pistons

Items	Collected converted piston (CR=11)	Optimum converted piston (CR=11)	Original piston(CR=18)
Load	45 bar	45 bar	60 bar
Static safety factor	2.479	3.197	3.183
Fatigue safety factor	1.33	1.717	1.679

Static and fatigue safety factors for collected and optimum CNG converted pistons as well as original piston are tabulated in table 2. Note that, safety factor decreases by 20.7% for commercially converted piston as compared to the original piston. It indicates that after CNG conversion the piston becomes weaker than the original piston. If the piston is optimistically cut then the safety factor would be increased by 2.2% as compared to the original one.

Analysis of mass moment of inertia is also performed on both pistons to observe whether there is any significant change of dynamic behavior between two pistons. For compression ratio 11, center of mass and mass moment of inertia of the piston with different groove depth is presented in table 3. The collected CNG converted piston has the groove depth of 18.5 mm and after stress analysis the optimum piston groove depth was found to be 13.67 mm.

It is perceived from the table 3 that center of mass shifts when groove depth of the piston changes. But this change is within a fraction of millimeter. At the same time mass moment of inertia differs due to the variation of center of mass. But this variation is within 3% as compared to the collected one. Therefore, it can be avowed that there would be no significant change of dynamic behavior if the collected piston is replaced by the optimum piston in CNG conversion process.

Table 3: Center of Mass and Mass moment of inertia

Compression Ratio=11						
Depth (mm)	Center of Mass (mm)			Mass Moment of inertia (Kg-mm^2)		
h	X_c	Y_c	Z_c	I_{xx}	I_{yy}	I_{zz}
18.5	7.23E-6	-25.142	5.43E-6	1905	2057	1865
13.67	-1.26E-5	-25.583	-1.06E-5	1848	2009	1808

4. CONCLUSIONS

Finite Element Analysis of groove geometry (h/r ratio) is significant in terms of stress concentration in modified piston. Study of the particular piston revealed the following optimum dimensions:

(1) For CR=9, a through flat cut of 10.69 mm should be made with a typical depth of the groove of 7.93 mm and radius of 36.1 mm.

(2) For CR=10, a through flat cut of 7.186 mm should be made with a typical depth of the groove of 8.84 mm and radius of 36.1 mm.

(3) For CR=11, the typical depth of the groove should be 13.67 mm and radius is 42 mm.
For CR=12, the typical depth of the groove should be 13.15 mm and radius is 37.8 mm.

(4) This optimum dimensions of the converted piston that can be applied in diesel to CNG conversion process. This can enhance the performance and longevity of the converted CNG engine.

There is no significant change of mass moment of inertia in case of optimum CNG converted piston as compared to the commercially CNG converted piston.

Analysis of stress distribution of optimum CNG converted piston implies that in maximum portion of the piston there is a stress level in the range of about 20-45 MPa for CR=11 where as only a few region experience higher stress in the range of 70-77 MPa. Thus, by taking care of these critical zones the stress level could significantly be reduced.

Local reinforcement on the critical zones may reduce the maximum stress development in the piston. At the same time changing the fillet radius of the piston groove, gudgeon pin side region and bottom inner region, stress analysis can be performed to determine the optimum fillet radius of these critical zones.

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6. NOMENCLATURE

Symbol	Meaning	Unit
S_y	Yield Strength	(MPa)
σ	Stress	(MPa)
τ	Shear stress	(MPa)
h	Groove depth	(mm)
r	Groove radius	(mm)
CR	Compression ratio	-
σ_a	Stress amplitude	(MPa)
σ_m	Mean stress	(MPa)

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