Design, Analysis and Optimization of a Four Stroke Diesel Engine Piston

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Abstract-In this project the design, analysis and optimization of a piston, which is lighter, stronger and to manufacture with minimum cost, of a four stroke diesel engine is considered. The analysis is carried out for the stress distribution in various parts of the piston, taking into consideration the gas pressure and thermal loads with structural parameter variation. For the finite element analysis of the piston, first 3D modelling is done using CATIA V5 and then this model is imported to ANSYS 11.0 for analysis and optimization. After optimization it is observed that there is variation in the Von Mises stress and deflections. However, all the stress and deflection values are well within the permissible limits.

1. INTRODUCTION

Automobile components are in great demand these days because of increased use of automobiles. The increased demand is due to improved performance and reduced cost of these components. Engineers should develop critical components in shortest possible time to minimize launch time for new products. This necessitates understanding of new technologies and quick absorption in the development of new components.

A piston is a moving or reciprocating part contained in cylinders and is made gas tight by piston rings. The main function of the piston is to transmit the force due to gas pressure inside the cylinder to the crankshaft through the connecting rod. It compresses the gas during the compression stroke and seals the inside portion of the cylinder from the crankcase by means of piston rings. It also takes the side thrust resulting from the obliquity of the connecting rod. Another important function is that it dissipates large amounts of heat from the combustion chamber to the cylinder wall.

A piston possesses the following characteristics like;

- Sufficient strength to withstand force due to combustion of fuel and the inertia forces due to the reciprocating parts
- It should have sufficient rigidity to withstand thermal and mechanical distortions; it should have adequate capacity to dissipate heat and should have minimum weight so as to reduce inertia forces.
- It should form an efficient seal to prevent leakage of lubricating oil.
- It should have sufficient bearing area to take side thrust.

1.1 Trunk type pistons

Trunk pistons (Fig 1.1) are long, relative to their diameter. They act as both piston and also as a cylindrical crosshead. Trunk pistons have been a common design of piston since the early days of the reciprocating internal combustion engine. They are found in both petrol and diesel engines, although high speed engines have now adopted the lighter weight slipper pistons.

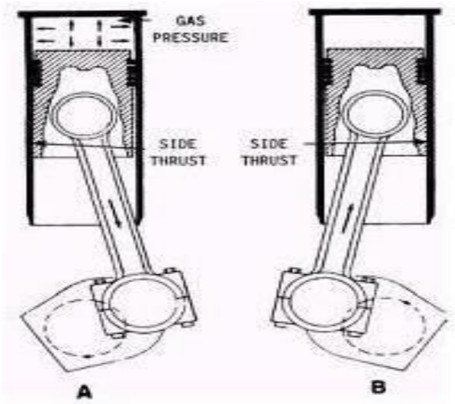


Fig 1.1 Trunk type piston

1.2 Materials of Piston

Pistons are basically made of aluminum alloys for light weight and excellent thermal conductivity. The different types of materials used are discussed below.

Cast aluminum pistons: They are best suited for stock engines. They have a lower price. They are more brittle and hence they are not suitable for high performance applications.

Hypereutectic aluminum pistons: They are best suited for 600 HP or 700 HP aspirated engines. They have a moderate price. It is similar to cast piston but with a little amount of silicon added (approximately 16%) which is a much stronger version than a cast piston. And the engine also becomes harder and is less susceptible to scuffing. Because of the high silicon content it is a less ductile alloy. It is best suited for normally aspirated engines rather than engines with nitrous applications or boost.

Forged aluminum pistons: This material is best for engines producing 1000-1200 HP. They are relatively of higher price. They are the strongest pistons on the market. They can be used for extreme duty racing applications. They allow for the most extreme usage conditions, but longevity is eventually compromised after countless heat cycles.

1.3 Piston Rings

Piston rings seal the combustion chamber, transferring heat to the cylinder wall and controlling oil consumption. A piston ring seals the combustion chamber through inherent and applied pressure. A piston ring must provide a predictable and positive radial fit between the cylinder wall and the running surface of the piston ring for an efficient seal. There are three kinds of rings in the piston, and they are the compression ring, the wiper ring and the oil ring.

1.4. Stresses in Piston

As one of the major moving parts in the power transmitting assembly, the piston must be so designed that it can withstand the extreme heat and pressure of combustion. Pistons must also be light enough to keep inertial loads on related parts to a minimum. The piston also aids in sealing the cylinder to prevent the escape of combustion gases. It also transmits heat to the cooling oil and some of the heat through the piston rings to the cylinder wall. Damages in a piston may have different origins like mechanical stresses, thermal stresses; wear mechanisms, temperature, degradation, oxidation mechanisms, etc. Piston damages are attributed to wear and lubrication sources, but fatigue is responsible for a significant number of piston damages. Principal stresses, minimum principal stresses, Von Mises stresses, total deflection occurred during working condition, etc are evaluated in the project. If these stresses exceed the designed values, failure of piston may take place. The stresses due to combustion are considered to avoid the failure of the piston. Intensity of thermal and structural stresses should be reduced to have safe allowable limits.

1.5. Thermal Analysis of Piston

Piston as one of the most important parts of the engine, the working conditions are harsh, because it is exposed to the influence of the thermal load in the work process. As the most critical part of the engine, and the working conditions of the piston are possible to greatly affect the life and performance of the engine, so it is particularly important to carry out the thermal analysis to the engine piston.

Burning of the high pressure gas products high temperature, which makes piston, expands in order that its interior produces thermal stress and thermal deformation. The thermal deformation and mechanical deformation will cause piston cracks.

When Pistons are operating, they directly touch the high temperature gases and their transient temperature can reach more than 2500 K and then generates power. The piston is heated seriously and its heat transfer coefficient is low and hence its heat dissipation condition is poor, so the piston temperature can reach 600 ~ 700 K approximately and the temperature distributes unevenly. On the basis of these conditions, the thermal analysis for the piston is done.

In this work, a piston of a 4 stroke single cylinder diesel engine is designed and analyzed for transient thermal and static structural stresses. The static structural stresses applied in this analysis are used to determine the deformations, equivalent stresses and strains, and heat flux at different sections of the piston at different thermal loads. Static structural analysis is used at static loading conditions, and is used to determine deformations, strains, stresses and reaction forces.

In this finite element analysis is done by using the Finite Element (FEA) software package ANSYS, version 11.0. Finite Element Analysis is a numerical method of deconstructing a complex system into very small pieces, of user designated size called elements. The software implements equations that govern the behaviour of those elements and solves them, creating a comprehensive explanation of how the loads are acting on the component.

2. DESIGN OF PISTON

The piston used in this project is a trunk type piston. The design of a piston consists of the following parts: Piston head, Piston rings, Piston barrel, Piston skirt and Piston pin.

The detailed design procedure of the above said parts is discussed below:

The Piston head being used in this design is the flat type piston head. The criterion for this design is strength. The piston head is fixed at the outer edge and is subjected to uniformly distributed gas pressure. According to Grashoff's formula the thickness of the piston head is given by,

$$t_h = D\sqrt{[(3\times p_{max})\div(16\times\sigma_b)]} \qquad (3.1)$$

Since the material considered for the piston is aluminium alloy. Then permissible bending stress for aluminium alloy is assumed to be between 50 and 90 N/mm^2 and the maximum gas pressure is taken between 4 to 5 MPa. An empirical relation for the thickness of the piston head is given as,

$$t_h = 0.032\,D + 1.5\text{ mm} \qquad (3.2)$$

Piston rings in IC engines are of two kinds and the rings to be designed are (i) Compression rings and (ii) Oil rings. The compression ring is used to maintain a seal between the cylinder wall and piston and prevent leakage of the fluids (fig 1.5).The Oil rings (fig 1.7) are normally placed below the compression rings and they provide proper lubrication of the cylinder liner and they reduce frictional losses.

Piston rings are usually made of grey cast iron and in some cases alloy cast iron. Grey cast iron has excellent wear resistance and also retains the spring characteristics at high temperatures. The number of compression rings in automobile and aircraft engines are usually 3 to 4. In stationary diesel engines 5 to 7 rings are used. The number of oil rings usually varies from 1 to 3. The compression rings have a rectangular cross-section. The radial thickness of the rings is given by,

$$t_1 = D \sqrt{[(3\times p_w)\div\sigma_t]} \qquad (3.3)$$

The radial wall pressure is taken between 0.025 to 0.042 MPa. The permissible tensile stress for cast iron is taken from 85 to 110 N/mm^2. The axial thickness of the piston ring is given by,

$$t_2 = (0.7\ t_1) \text{ to } t_1 \qquad (3.4)$$

The limit for the axial thickness is given by,

$$t_{2min} = (D/10\times n_r) \qquad (3.5)$$

The distance from the top of the piston to the first ring groove (b_1) is called the top land. And it is given as,

$$b_1 = (t_h) \text{ to } (1.2t_h) \qquad (3.6)$$

The distance between two ring grooves is called the width of ring land (b_2). And it is given by,

$$b_2 = 0.75t_2 \text{ to } t_2 \qquad (3.7)$$

Piston barrel is the cylindrical portion of the piston below the piston head is called the piston barrel. The thickness of the piston barrel at the top end and lower end are given by,

$$t_3 = (0.03D + t_1 + 4.9) \text{ mm} \qquad (3.8)$$

$$t_4 = (0.25\ t_3) \text{ to } (0.35t_3) \qquad (3.9)$$

Piston skirt is the cylindrical portion between the last oil ring and the lower end of the piston barrel is called the piston skirt. The piston skirt acts as a bearing surface for the side thrust. The length of the piston skirt is limited to 0.25MPa. The empirical relation for the length of the skirt is given as,

$$l_s = (0.65D) \text{ to } (0.8D) \qquad (3.10)$$

The total length of the piston is taken as the sum of the length of the top land, length of the ring section and the length of the skirt. The length of the piston is also given by the following empirical relation,

$$L = D \text{ to } 1.5D \qquad (3.11)$$

The function of the Piston pin is to connect the piston to the connecting rod. Here it acts as a fixed support while analyzing the stresses in the piston.

3. FINITE ELEMENT ANALYSIS OF PISTON

3.1. Modelling of Piston

The modelling of piston is done using CATIA, V5. CATIA (Computer Aided Three-dimensional Interactive Application) is one of the most comprehensive, widely used engineering tools used in industries by thousands of companies around the world. CATIA in one seat provides a consistent design package. It contains the main CATIA Mechanical, Shape Design and Styling domains.

It provides the tools for 3D part and assembly design, generation of production drawings, and creates wireframe construction elements and advanced surfaces. To assist with initial innovative thought process, a 'clay modelling' style tool is also provided. The range of CATIA capabilities allows it to be applied in a wide variety of industries, such as aerospace, automotive, industrial machinery, electrical, electronics, shipbuilding, plant design.

The sequence of steps followed in CATIA for generating piston models are given below:

- Start → Mechanical Design → Part Design.
- Select the Sketcher option from the tool bar on the right hand side of the page. Now we enter into the workbench.
- In the workbench using the line tools draw a 2D half cross section of the piston. And draw an axis to which the piston can be rotated 3 dimensionally.
- Exit the workbench and Select the shaft option from the tool bar.
- This shaft option generates a 3D image of the piston around the axis drawn in the workbench.
- A hole is created in the piston by pocketing.

4. STRESS ANALYSIS OF PISTON USING ANSYS

Finite Element Analysis is used to find the solution of a complicated problem by replacing it with simpler element. Since the actual problem is replaced by a simpler one it will be able to find only an approximate solution rather than the exact solution. In F.E.A it is possible to improve or refine the approximate solution.

In F.E.A, the actual body of matter like solid, liquid and gas is represented as an assembly of sub divisions called finite elements. These elements are considered to be interconnected in specified joints called nodes. These nodes usually lay on the element boundaries.

4.2.1 Step by step procedure of F.E.A

Step 1: Discretization of component/structure is the first step in the F.E.A is to divide the component or object into sub divide called elements. Hence the structure is to be modelled with suitable elements. The model type and size of the elements are divided.

Step 2: Selection of proper interpolation of displacement function describes the actual variation of field variables inside a finite element can be approximated by a simple function. These approximate functions are called displacement functions and are defined in terms of values of the field variables at nodes. In general interpolation function is taken in the form of polynomial.

Step 3: Derivation of element stiffness matrices and load vectors, From the assumed displacement function the stiffness matrix and load vector are to be derived using equilibrium equations.

Step 4: Assembly of element equations to obtain the overall equilibrium equations, the structure is composed of several finite elements. Hence the individual element stiffness matrices and load vectors are to be assembled in a suitable manner and the overall equilibrium equations are formulated.

Step 5: Solutions for unknown nodal displacements, the overall equilibrium equations have to be modified to account for the boundary conditions of the problem. After incorporating the boundary conditions the equilibrium equations solved for the nodal displacements.

Step 6: Computation of element stresses and strains, is done from the known nodal displacements the element stresses and strains can be computed by using necessary equations of solid mechanics.

5. NUMERICAL ANALYSIS

This section completely deals with piston dimensions and its details, CATIA diagrams and ANSYS diagrams of piston. The piston is designed according to the procedure and specifications which are given in Ref. [7]. The calculations done are detailed below:

(i) The thickness of the piston head is given by

$t_h = D\sqrt{[(3\times p_{max})\div(16\times\sigma_b)]}$

$t_h = 87.5\sqrt{[(3\times 5.4)\div(16\times 90)]}$

$= 9.28$ mm

Where, p_{max} is considered to be 54 bars

D = 87.5mm

σ_b= 90 MPa

(ii) Radial thickness of ring (t_1)

$t_1 = D\sqrt{[(3\times p_w)\div\sigma_t]}$

$= 87.5\sqrt{[(3\times 0.042)\div 90]}$

$= 3.273$mm

Where, p_w is considered as 0.042 N/mm^2

σ_t is taken as 90 MPa

(iii) Axial thickness of the ring (t_2)

$t_2 = (0.7\ t_1)$ to t_1

Considering t_2 to be minimum it is taken as 3mm

(iv) Width of the top land (b_1)

$b_1 = (t_h)$ to $(1.2t_h)$

b_1 is considered as 11.136mm

(v) Width of other lands (b_2)

$b_2 = 0.75t_2$ to t_2

$= 2.4$ mm

(vi) Maximum thickness of Barrel at top and bottom end is given by

$t_3 = (0.03D + t_1 + 4.9)$ mm

$= (0.03* 87.5 + 3.673 + 4.9)$

$= 10.798$mm

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

$= 3.5$mm

(vii) Length of the piston is calculated as 95mm

The previous chapters dealt with the design of piston. The dimensions for the piston calculated above are shown in table 1.

Table 1: Details of piston dimensions:

Dimensions	Size in mm
Bore diameter of cylinder D	87.5
Total length of the piston L	95

Thickness of piston head t_H	9.28
Radial thickness of ring t_1	3.27
Axial thickness of rings t_2	3
Thickness of piston barrel at top end t_3	10.798
Thickness of piston barrel at lower end t_4	3.5
Width of top land b_1	11.136
Width of bottom ring land b_2	2.4

The above dimensions are used for modelling of piston in CATIA V5. With the use of these formulae 5 piston models were generated with varying dimensions [6] and are listed in table 5.2. Table 5.3 gives detailed properties of the material used (Aluminium Alloy).

Table 5.2: Dimensions of Piston Models (in mm)

Piston Model/ Dimensions(mm)	Model 1	Model 2	Model 3	Model 4	Model 5
t_H	9.28	9.28	9.28	9.28	9.28
t_1	2.27	2.77	3.27	3.77	4.27
t_2	2	2.5	3	3.5	4
t_3	9.798	10.298	10.798	11.298	11.798
t_4	2.5	3	3.5	4	4.5
b_1	10.136	10.636	11.136	11.636	12.136
b_2	1.4	1.9	2.4	2.9	3.4

Table 5.3: Properties of piston material

S. No	Name of the property	Value
1	Thermal Conductivity	174.15 W/ mK
2	Specific Heat	0.13 J/ kgK
3	Young's Modulus	71e3 MPa
4	Poisson's ratio	0.33
5	Density	2.77e-6 kg/ mm^3

5.1 Piston Models Generated in CATIA

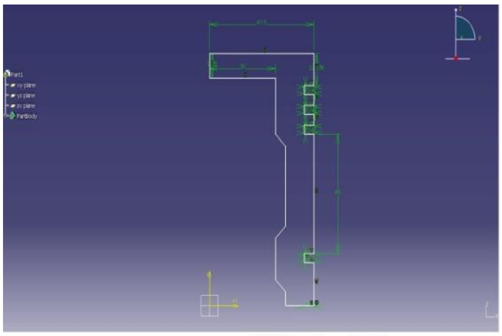


Fig 5.1 2D piston model generated in CATIA

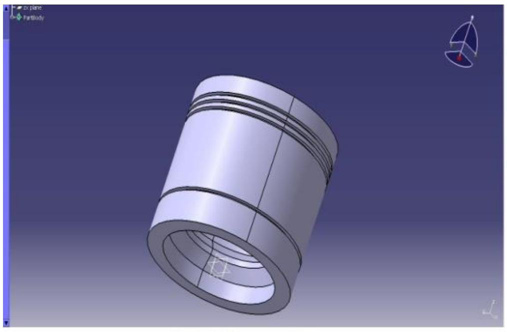


Fig 5.3 3D piston model generated in CATIA

Piston model after 3D modelling is done

The modelling of piston models given in Table 5.2 is done in CATIA. Fig 5.1provides a sample of 2D piston model of Model 1 whereas fig 5.2 and fig 5.3 represent the 3D model of the same piston.

5.4 Analysis in ANSYS

The CATIA models are imported into ANSYS. The imported models are then meshed into finite elements of user designated size (fig 5.4). The thermal and static structural loads are then applied to the models and solved. The solutions give the deformations, equivalent stresses and strains at different cross sections of the piston models.

5.4.1 Meshed models in ANSYS

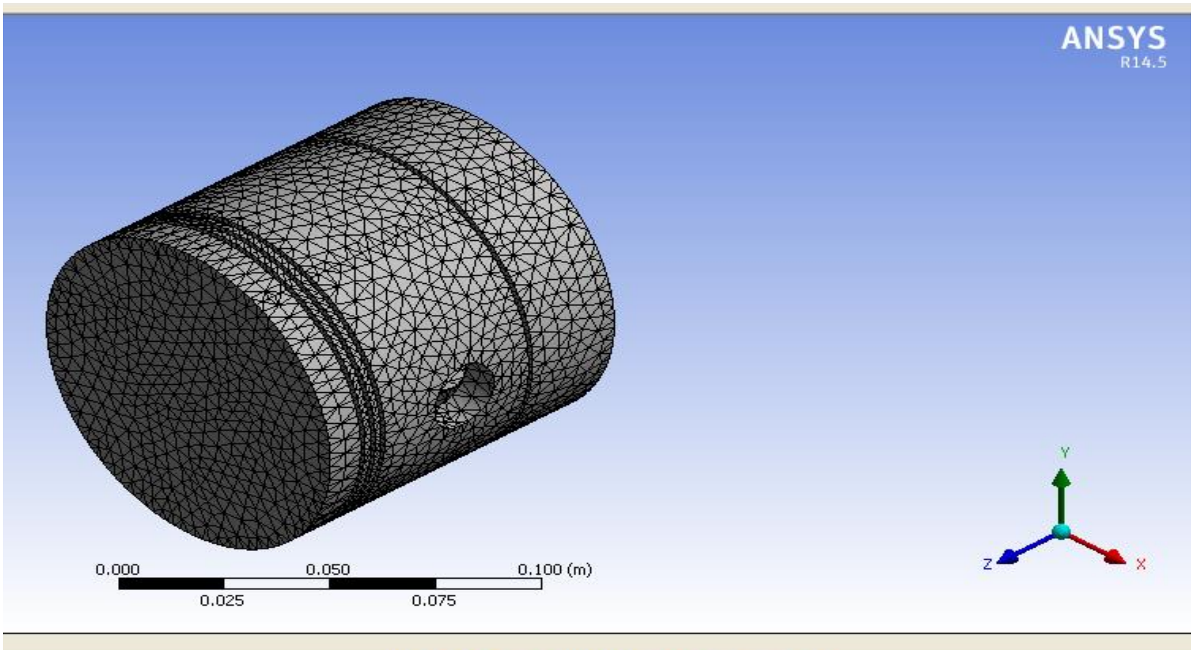


Fig 5.4(a) Meshed piston model

5.4.2. Solutions after applying transient thermal load

Figures 5.5 and 5.6 show the effect of application of thermal loads on the piston and figure 5.7 shows the total heat flux. The static structural loads are applied after the application of thermal loads (fig 5.8). All the figures are sample figures out of total 35 related figures.

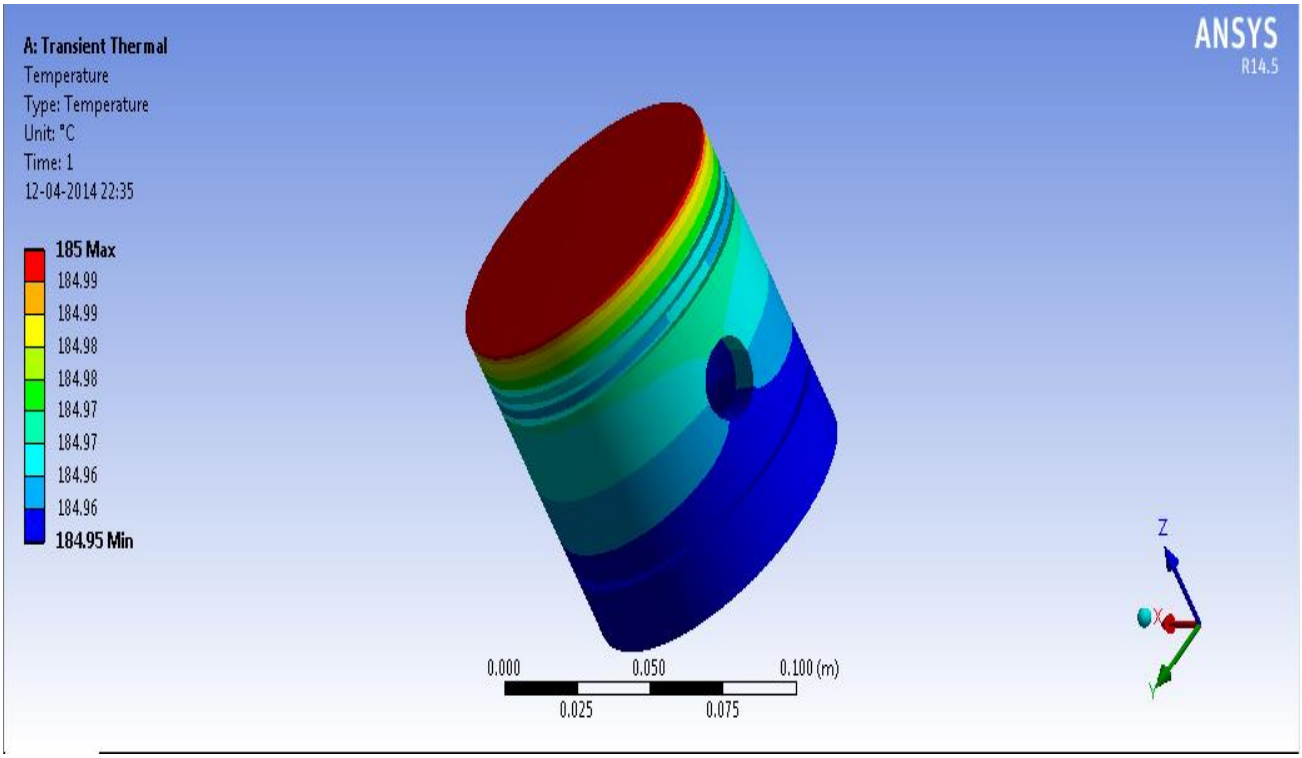


Fig 5.5 Temperature load

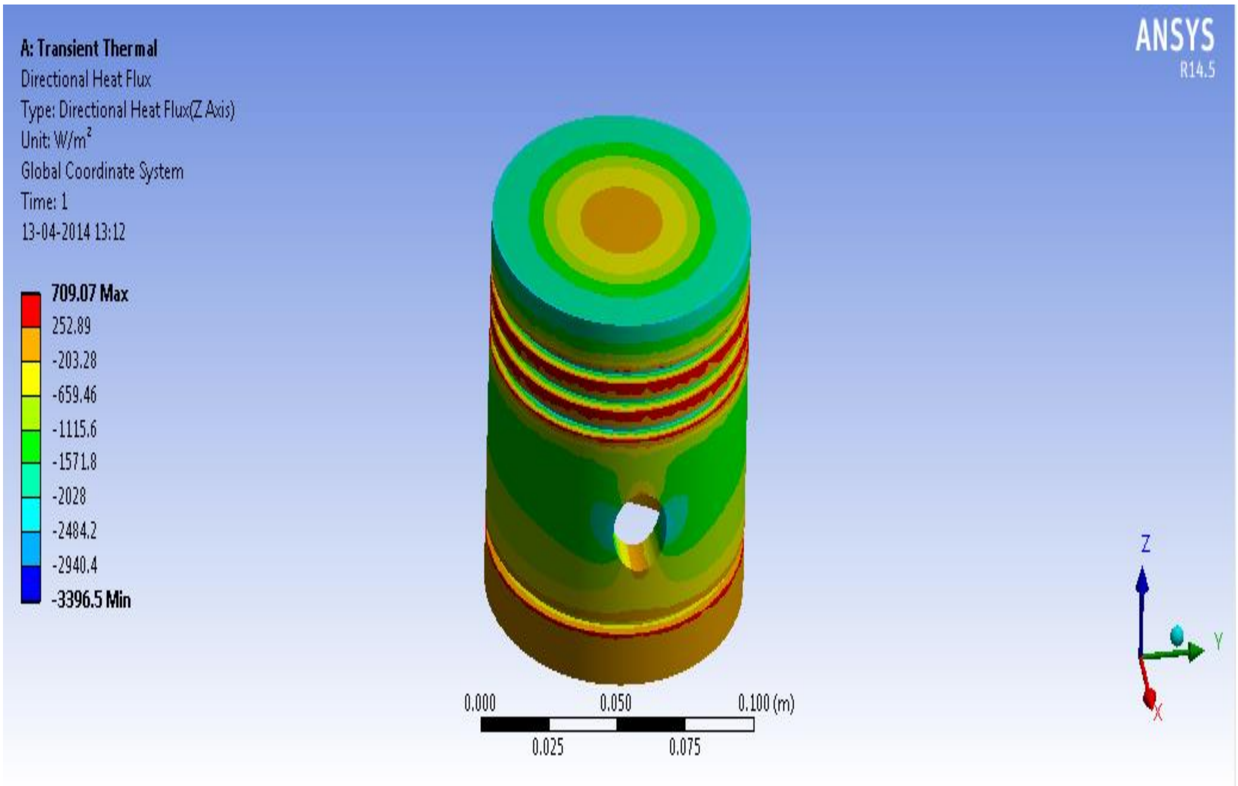


Fig 5.6 Directional heat flux (Z-axis)

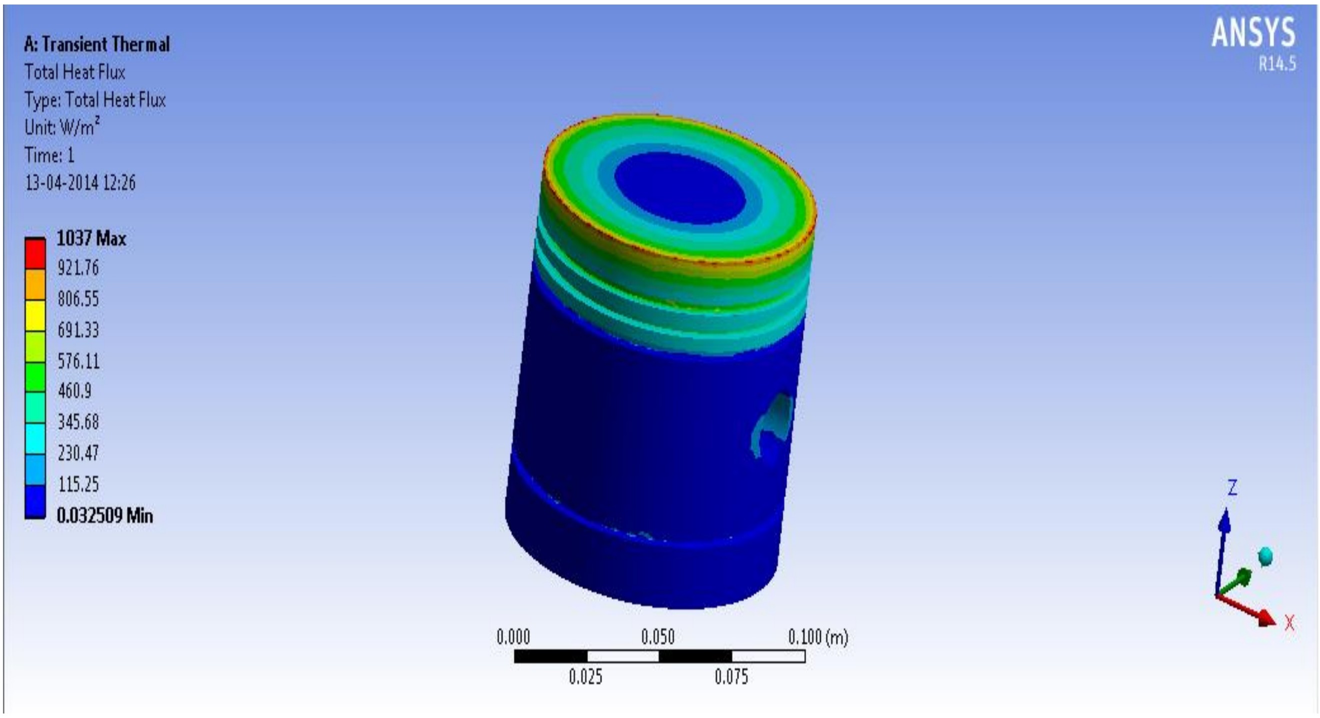
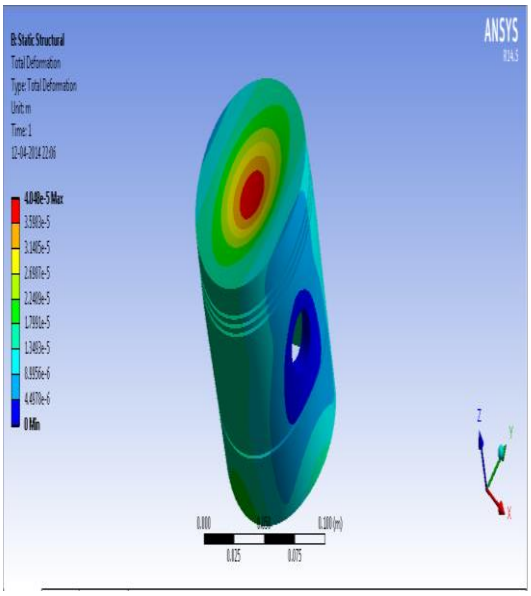


Fig 5.7 Total heat flux

5.4.3 Solutions to application of structural loads after the application of thermal load



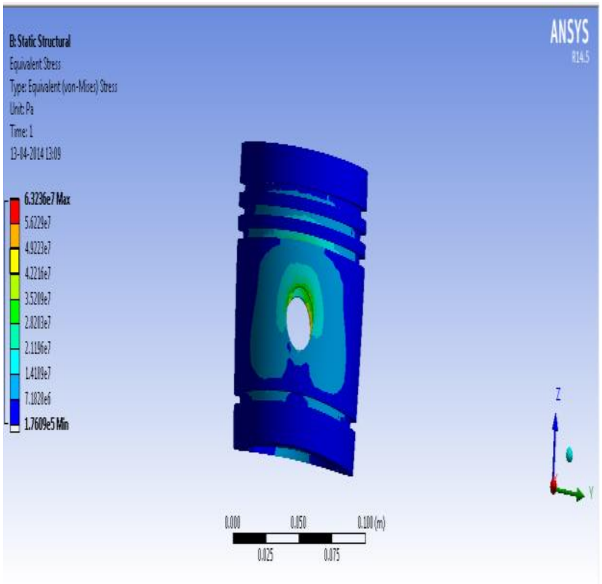


Fig 5.8(a) Total deformation Fig 5.8(b) Equivalent stress (Von Mises)

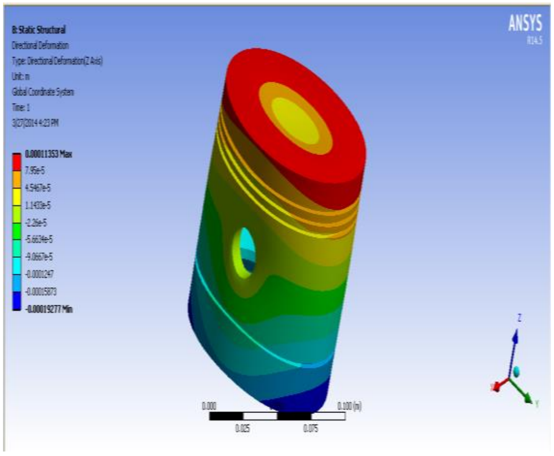


Fig 5.8(c) Directional deformation

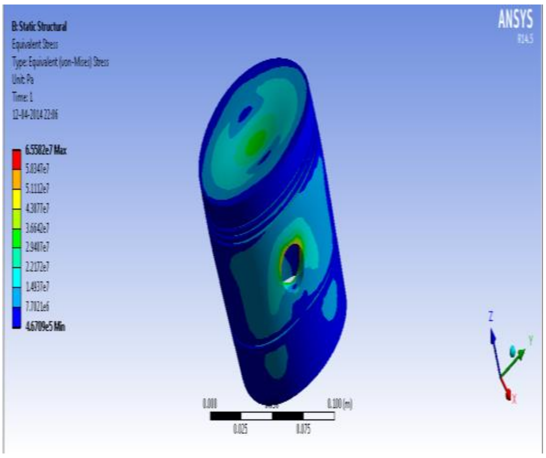


Fig 5.8(d) Equivalent stress

6. DISCUSSION OF RESULTS

Analysis of different piston models is done using finite element analysis software, ANSYS. Model geometry is created and meshing of the components is done. Thermal and structural loads are applied to the meshed piston models in ANSYS, and the solutions are obtained. The solutions obtained for total heat flux, Von Mises stresses, and directional deformation, equivalent strain and volume are tabulated (fig 5.1 to fig 5.8) in table 6.1.

Table 6.1. Results obtained after application of stresses

Piston models	Total deformation (mm)	Directional Deformation (Z-axis)	Equivalent stress (MPa)	Mesh Elements Nodes	Volume (m^3)
1	0.04048	0.016317	65.58	32032 55131	3.1976e^{-4}
2	0.038185	0.01291	61.8	29914 51625	3.1549e^{-4}
3	0.037607	0.010645	58.06	30231 52013	3.2295e^{-4}
4	0.036327	0.010701	55.8	29394 50934	3.439e^{-4}
5	0.036808	0.0094035	63.23	29961 51726	3.471 e^{-4}

Plots of table 6.1 are shown in Fig 6.2(a) to fig 6.2(d). From fig 6.2, the piston Model 2 which has all the parameters considered, are smaller values than the other models. The comprehensive results of piston model 1 for fig 6.2

is a higher side; however they are well within the limits. Therefore the piston (model 1) is considered of optimum dimensions.

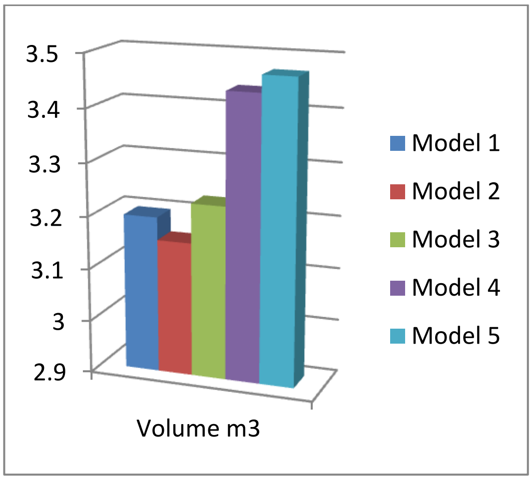


Fig 6.2(a)

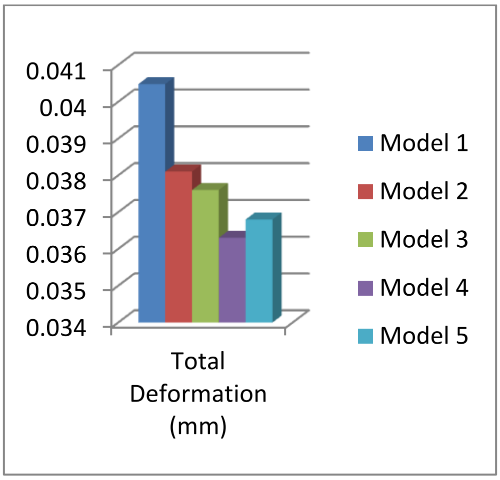


Fig 6.2(b)

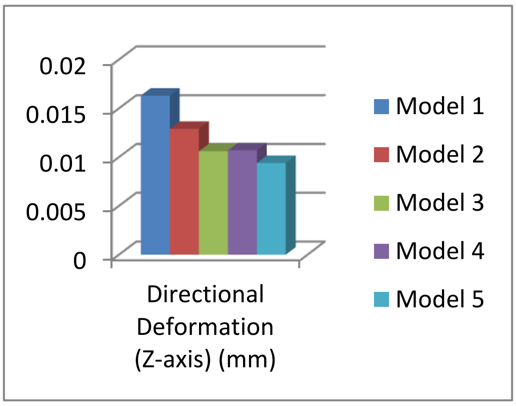


Fig 6.2(c)

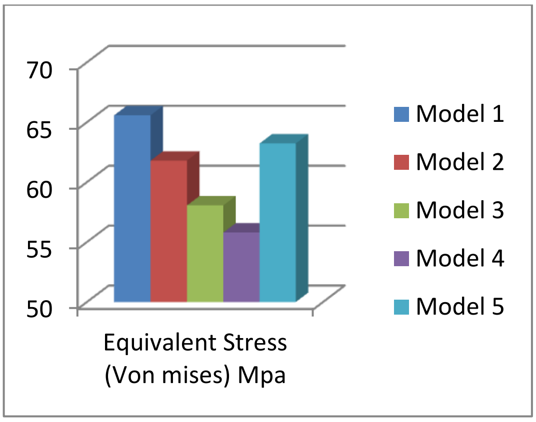


Fig 6.2(d)

Model 3 is the model with the dimensions of the 4 stroke single cylinder diesel engine considered for analysis. The Von Mises stresses of this piston model are lower than the other 4 models. Comparing the stresses of the other 4 models to this model, the second model is considered to be optimal within the considered range of Von Mises stresses is of minimum volume.

The following are the dimensions for the optimal piston model:

- Thickness of the piston head: 9.28 mm
- Radial thickness of ring : 2.77 mm
- Axial thickness of ring : 2.5 mm
- Width of top land : 10.636 mm
- Width of other lands : 1.9 mm
- Thickness of barrel from top end and lower end : 10.298 mm & 3 mm

- Total length of piston : 95 mm

7. CONCLUSIONS AND FUTURE WORK

7.1 Conclusions

The present work is mainly, design, analysis and optimization of a four stroke diesel engine piston. A trunk type of piston is considered here. 3D modelling of the piston is done using CATIA V5 and then this model is imported to ANSYS 11.0 for analysis and optimization. About 5 models of pistons are designed analyzed and model 2 piston is considered an optimized piston.

The radial thickness of the piston is affected more as it is very less in dimension, and the temperature and heat flow are high to this value. Also the volume of the piston is decreased resulting in more deformation at the piston crown and an increase in the value of the Von Mises stresses from 58.06 MPa to 61.8 MPa. The permissible limit to the Von Mises stresses is 90 MPa so the piston with these considerations can withstand easily the deflection due to pressure applied and the temperature loads.

7.2 Future Work

As the piston is exhibiting optimum Von Mises stresses, even though the volume is reduced to a great extent, this in turn reduces the inertial force. This procedure can be extended to different types of pistons, connecting rods, crankshafts of any engine part to reduce the volume as well as the inertial force. ANSYS 11.0 is used in the present work for the analysis of the piston models. The piston models can also be analyzed using NISA, NASTRAN, or any other high performance F.E.A software and the performance of these software can be estimated, relatively. Here, the piston crown is considered to be flat in the analysis carried out. But the original shape of the piston crown i.e., spherical groove may also be considered for analysis.

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