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# Piston Strength Analysis Using FEM

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## ABSTRACT

**This work describes the stress distribution of the piston by using finite element method (FEM). FEM is performed by using computer aided engineering (CAE) software. The main objective of this project is to investigate and analyze the stress distribution of piston at the actual engine condition during combustion process. The parameter used for the simulation is operating gas pressure and material properties of piston. The report describes the mesh optimization by using FEM technique to predict the higher stress and critical region on the component. The piston under study belongs to the two stroke single cylinder engine of SUZUKI Max100 motorcycle. Aluminum is selected as piston material. It is important to locate the critical area of concentrated stress for appropriate modification. Computer aided design (CAD) software PRO-E Wildfire 4.0 is used to model the piston. And static stress analysis and dynamic analysis is performed by using ANSYS 14. Based on stress analysis results the weight optimization of piston is done using ANSYS 14.**

*Keywords:* **Dynamic Analysis, FEM, Optimization, Piston Analysis, Piston Strength Analysis, Static Analysis, Two Stroke**

## I. INTRODUCTION

In internal combustion engine, piston is one of the important components. It reciprocates within the cylinder bore by force produced during the combustion process.

The two main requirements of the piston are as follows:

1- It should contain all the fluids above and below the piston assembly during the cycle.
2- It should transfer the work done during combustion process to the connecting rod with minimal mechanical and thermodynamic losses.

Five main properties of a piston are:

1- Sufficient thermal conductivity
2- Low thermal expansion
3- High hot strength
4- High strength to weight ratio
5- High resistance to surface abrasion

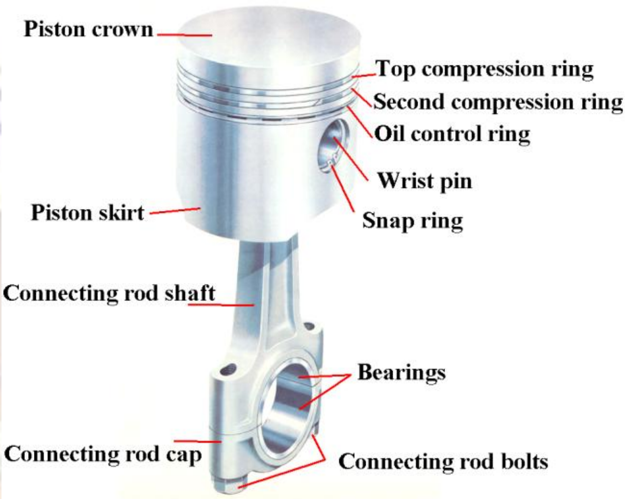



Figure 1.1: Labeled Image of a Piston and Con-Rod.

The piston is the heart of the internal combustion engine and is subjected to loads such as thermal and structural stress. The piston reciprocates within the cylinder. The two extremes of this motion are referred to as Top Dead Center (TDC) and Bottom Dead Center (BDC) shown in Fig. 1.2.

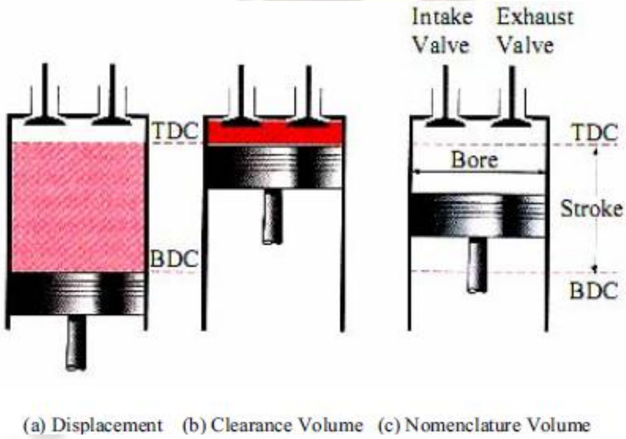



Figure 1.2: Cross Section of a Reciprocating Engine [3]

Top Dead Center is the position of the piston that creates the smallest volume in the cylinder, which is defined as the clearance volume, Vc. This is where combustion takes place in the engine and is also known as the combustion chamber. The Bottom Dead Center is when the piston creates the largest volume in the cylinder [3]. The distance between TDC and BDC is referred to as the stroke, and the volume which the piston displaces during this moment, is called the displacement volume, Vd. The piston is connected to the crankshaft via the connecting rod. The crankshaft converts the linear motion of the piston into rotational motion.

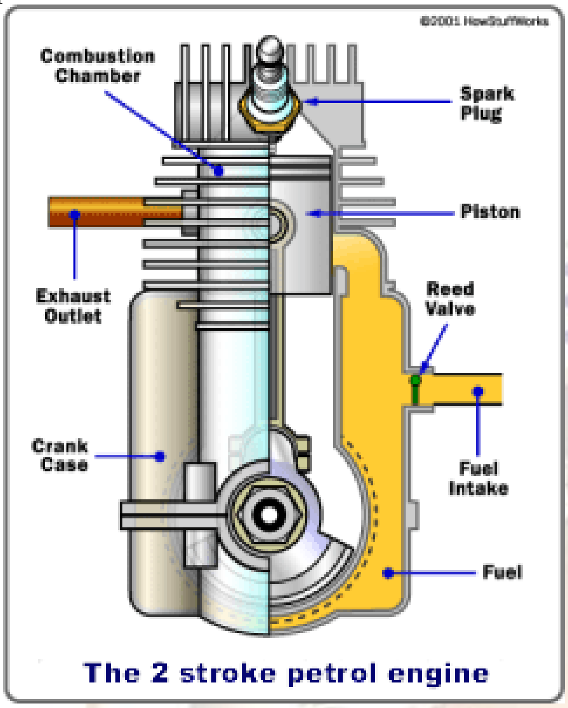



Figure 1.3: Two Stroke Petrol Engine

Finite Element Analysis is a simulation technique which evaluates the behavior of components, equipment and structures for various loading conditions including applied forces, pressures and temperatures. Thus, a complex engineering problem with non-standard shape and geometry can be solved using finite element analysis where a closed form solution is not available. The finite element analysis methods result in the stress distribution, displacements and reaction loads at supports for the model. FEA techniques can be used for mesh optimization, design optimization, material weight minimization, and shape optimization.

## II. PROBLEM STATEMENT

The piston is one of the most critical components of an engine. Therefore, it must be designed to withstand from damage that is caused due to extreme heat and pressure of combustion process. The value of stress that caused the damages can be determined by using FEA. Thus, it can reduce the cost and time due to manufacturing the components and at the same time it can increase the quality of the product.

The objective of this study is:

- To calculate the equivalent (Von Mises) stresses and total deformation by considering the gas load.
- To optimize the piston model for mass reduction.

The piston is implemented in the two stroke single cylinder engine of 100 cc SUZUKI Max100 motorcycle. Aluminum alloy is selected as a piston material. The details of this engine are:

Technical Specifications:

a. Single Cylinder Engine
b. Max Pressure 50 bar
c. Bore 50.0 mm
d. Stroke 50.0 mm
e. Piston displacement 98.2 cc
f. Compression ratio 6.7:1

Performance Parameters:

a. Maximum horsepower @5500 rpm 5.74 KW (7.8 BHP)
b. Max speed Top gear (4th gear) Around 82 km/hr -- 22.777m/s
c. Max torque 9.8 Nm at 5000 rpm
d. Acceleration (0-60 Km/hr) in 9 seconds

(a)


(b)

Figure 2.1: Assembled Engine of SUZUKI Max100 motorcycle

Figure 2.2: Piston of SUZUKI Max100 motorcycle

Table 2.1: Material Properties of Aluminum Alloy

| | |
|---|---|
| Density(Kg/m$^3$) | 2770 |
| Poisson's Ratio | 0.33 |
| Young's Modulus(Pa) | 7.10E+10 |
| Tensile Ultimate Strength(Pa) | 3.10E+08 |
| Tensile Yield Strength(Pa) | 2.80E+08 |
| Compressive Yield Strength(Pa) | 2.80E+08 |

## III. METHODOLOGY

- Theoretical stress calculation
- Create a 3D model of piston for two stroke engine using PRO-E WF 4.0
- Develop a Finite Element Model for mesh optimization of a piston using ANSYS 14.0
- Analyze piston using static stress analysis and dynamic analysis method
- Optimize the model for mass reduction.

### 3.1 Theoretical Stress Calculation

The piston crown is designed for bending by maximum gas forces *Pzmax* as uniformly loaded round plate freely supported by a cylinder.

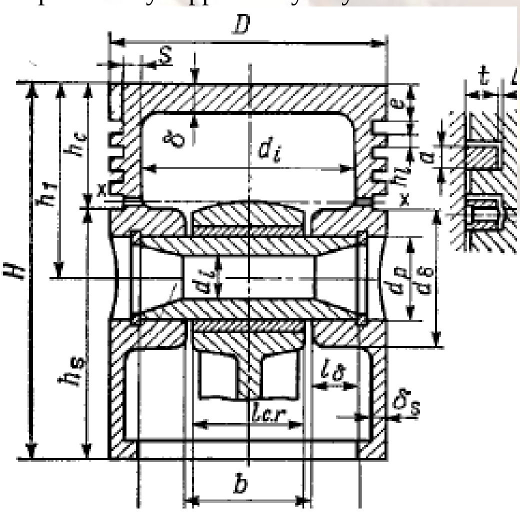



Fig. 3.1: Piston diagram

➢ The stress acting in MPa on piston crown:

$$\sigma_b = M_b/W_b = p_{z\max}\,(r_i/\delta)^2$$

Where

$Mb = (1/3)\ Pzmax\ r_i^3$ is the bending moment, MN m;

$Wb = (1/3)\ r_i\delta^2$ is the moment of resistance to bending of a flat crown, m$^3$;

*Pzmax* = *Pz*, is the maximum combustion pressure, MPa;

$r_i = [D/2 - (s + t + dt)]$ is the crown inner radius, m.

❖ $r_i = D/2-(s+t+ d\ t)$
= 50/2-(4.5+3+0.8)
= 16.7 m
*Pzmax* = 5 Mpa
$\sigma_b = 5 * (16.7/6)^2 = 38.73$ MPa

❖ $r_i = D/2-(s+t+dt)$
= 50/2-(3+3+0.8)
= 18.3 m
*Pzmax* = 5 Mpa
$\sigma_b = 5 * (18.3/4)^2 = 104.6531$ MPa

➢ Cycle time:
As engine rpm = 5000
t = 1/5000 min
t = 0.012 sec

➢ Angular Velocity

$$\omega = \frac{2\pi \cdot \mathrm{RPM}}{60}$$

Angular velocity = 2*3.142*5000/60
= 523.598 rad/sec

➢ Linear Velocity = Angular velocity * radius
= 523.598 * 30.5
= 15.9698 m/s

This Linear Velocity is used as initial condition for dynamic analysis.

### 3.2 Creation of 3D model of piston

3-D model geometry is developed in PRO-E Wildfire 4.0. Dimensions of the piston are taken from the engine model present in the college.

The following is the list of steps that are used to create the required model:

- The base feature is created on three orthogonal datum planes.
- Creating a sketch of piston wall & head section on front plane (with the help of sketcher Option), & then revolving it with respect to vertical axis as a center for rotation i.e. piston wall and head portion is generated.
- Similarly create another sketch of piston pin bore outer dia. on right plane & extrude it symmetrically with the datum plane with 'up to next' option i.e. piston pin bore is partially generated.

- Similarly create another sketch of piston pin bore inner dia. on right plane & extrude it symmetrically with the datum plane with 'up to next' & remove material option i.e. Piston pin bore is fully generated.
- Create another sketch of rectangular cut section on piston skirt on right plane & extrude it symmetrically with the datum plane with 'up to next' & remove material option i.e. rectangular cut section is generated on piston skirt.
- These all features are created on datum planes.
- Apply fillets to all sharp corners using Round tool.

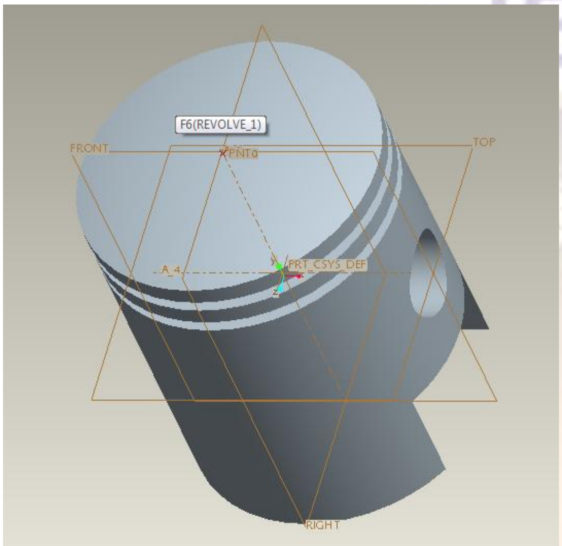



Figure 3.2: Standard Orientation of Piston Model in PRO-E WF4.0

**3.3 Analysis Using Ansys14:**

**3.3.1 Static Analysis:**

a. Frictionless Support at pin bore.

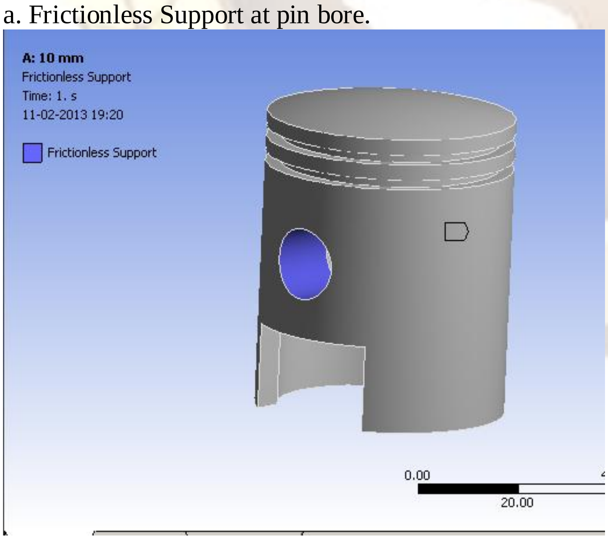



Figure 3.3: Boundary Condition 1

b. Displacement constraint at cylindrical surface

e. Static Analysis Results

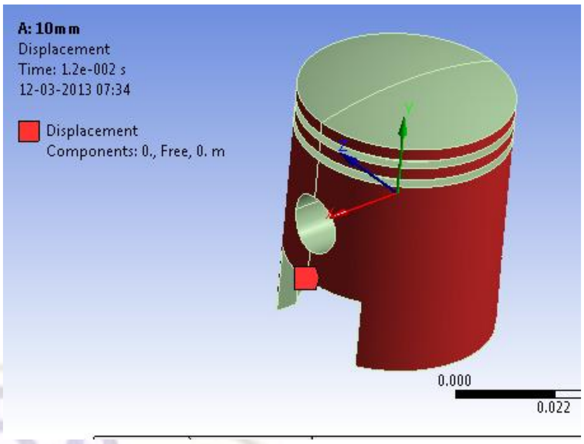



Figure 3.4: Boundary Condition 2

c. Downward force due to gas load acting on piston head.

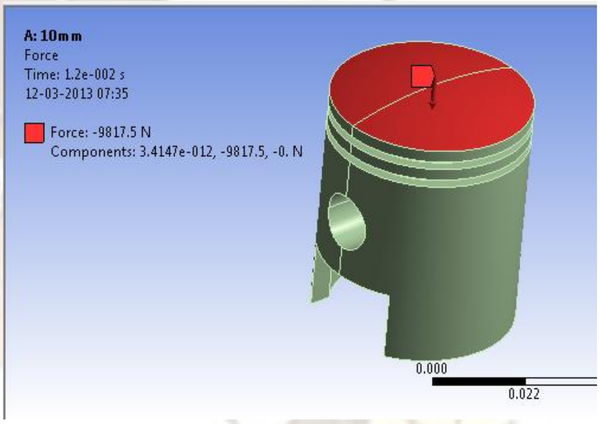



Figure 3.5: Boundary Condition 3

d. Piston model after meshing

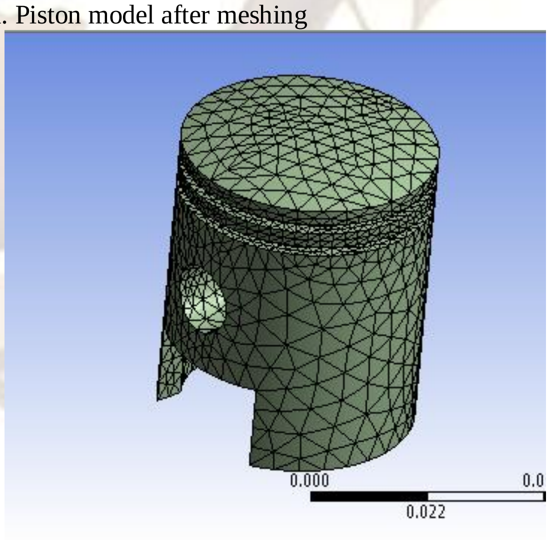



Figure 3.6: Piston model after meshing

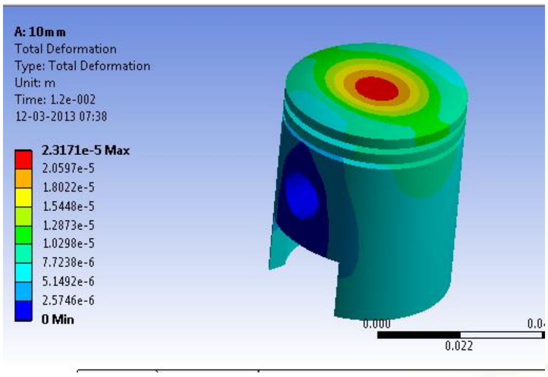



Figure 3.7: Total Deformation

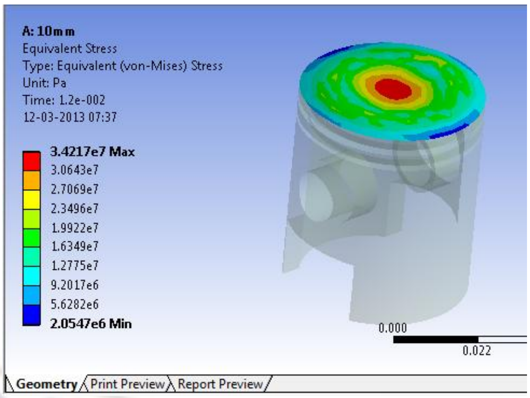
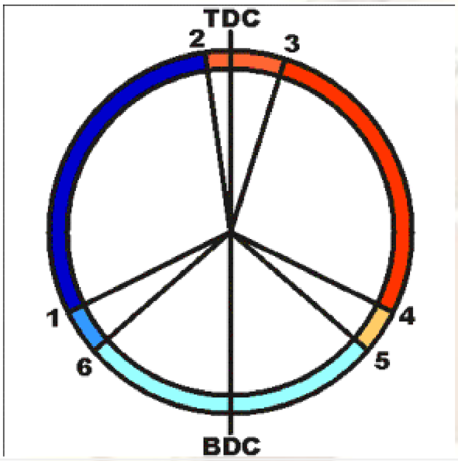



Figure 3.8: Equivalent Stresses

Maximum equivalent (Von Mises) stress = 34.7MPa
Maximum total deformation = $2.43*10^{-5}$m

In the dynamic analysis load is varied with respect to time as it is as per two stroke cycle. Mesh size used is 7.675 and 6.25mm. The two stroke cycle is illustrated on a timing diagram as below:

**3.3.2 Dynamic analysis:**




1 -2 Compression (8-20bar)
2 - 3 Fuel Injection
3 - 4 Power(30-50bar)
4 - 5 Exhaust Blow down(3-5bar)
5 - 6 Scavenging(1.05-1.20bar)
6 - 1 Post Scavenging(0.8-0.9bar)

1. approx 110º BTDC
2. approx 10º BTDC
3. approx 12º ATDC
4. approx 110º ATDC
5. approx 140º ATDC
6. approx 140º BTDC

Figure 3.9: Timing Diagram for Two Stroke Engine

Table 3.1: Time & Force Values

| | STEPS | TIME(sec) | ANGLE(Degrees) | PRESSURE (bar) | FORCE(N) |
|---|---|---|---|---|---|
| 1 | 6 | 0 | 140 BTDC | 0.8106 | 159.1609 |
| 2 | 1 | 0.0009 | 110 BTDC | 0.911925 | 179.056 |
| 3 | 2 | 0.00433 | 10 BTDC | 20 | 3926.99 |
| 4 | TDC | 0.00466 | 180 | 30 | 5890.486 |
| 5 | 3 | 0.005066 | 12 ATDC | 50 | 9817.47699 |
| 6 | 4 | 0.008333 | 110 ATDC | 5 | 981.7476 |
| 7 | 5 | 0.00933 | 140 ATDC | 1.2159 | 238.7414 |
| 8 | BDC | 0.01066 | 0 | 1.06391 | 208.8982 |
| 9 | 6 | 0.012 | 140 BTDC | 0.8106 | 159.1609 |

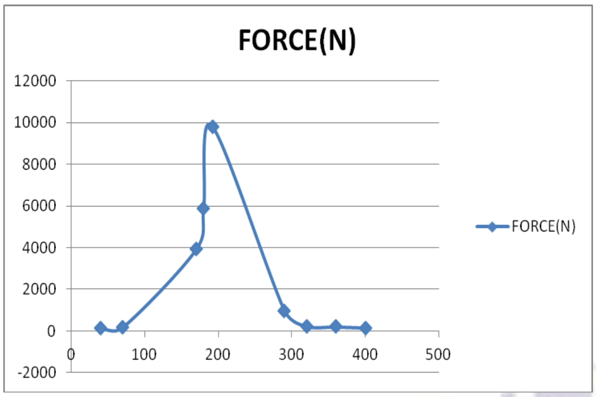



Figure 3.10: Force Variation over Crank Angle

First two boundary conditions, frictionless support at pin bore and displacement constraint at cylindrical surface are same as static analysis. Force varying with respect to cycle time is applied on piston head as one of the preprocessors.

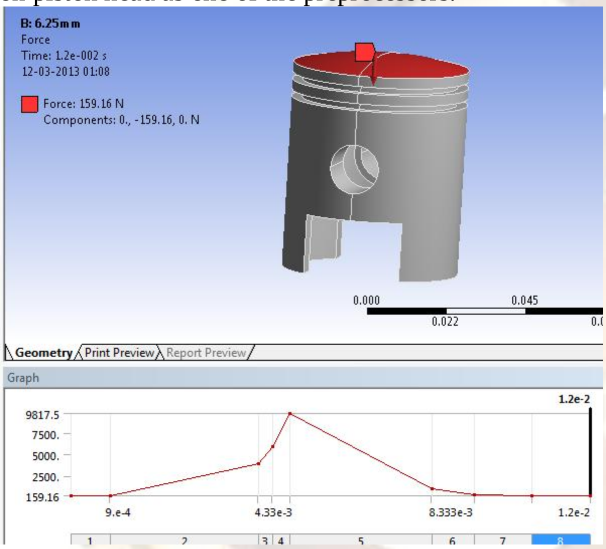



Figure 3.12: Boundary Conditions in Dynamic Analysis

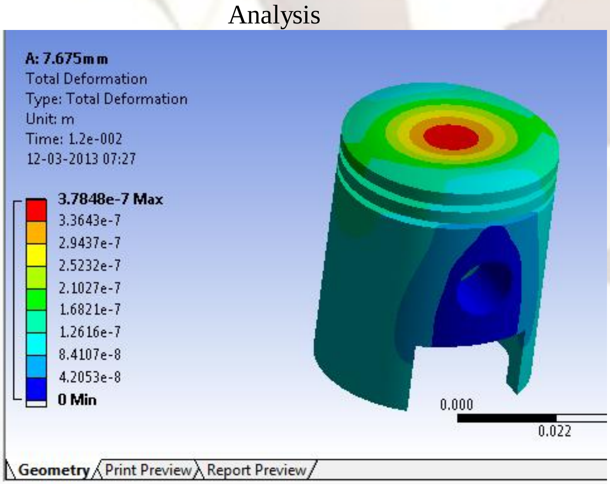



Figure 3.13: Deformation in Dynamic Analysis

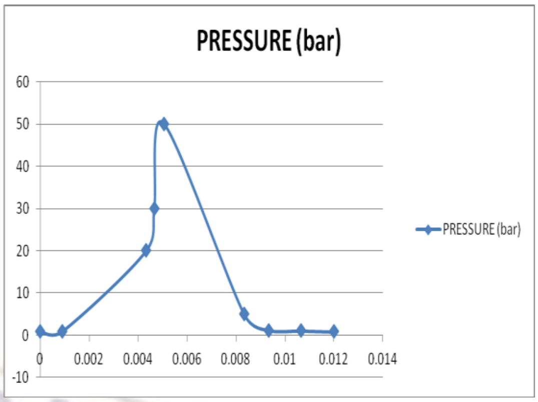



Figure3.11: Pressure Variation over Crank Angle

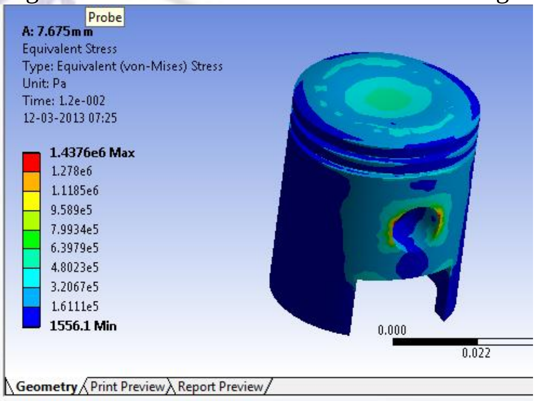



Figure 3.14: Stresses in Dynamic Analysis

Maximum equivalent (Von Mises) stress = 14.9MPa

Maximum total deformation = $3.80*10^{-7}$m

### 3.3.3 Optimization:

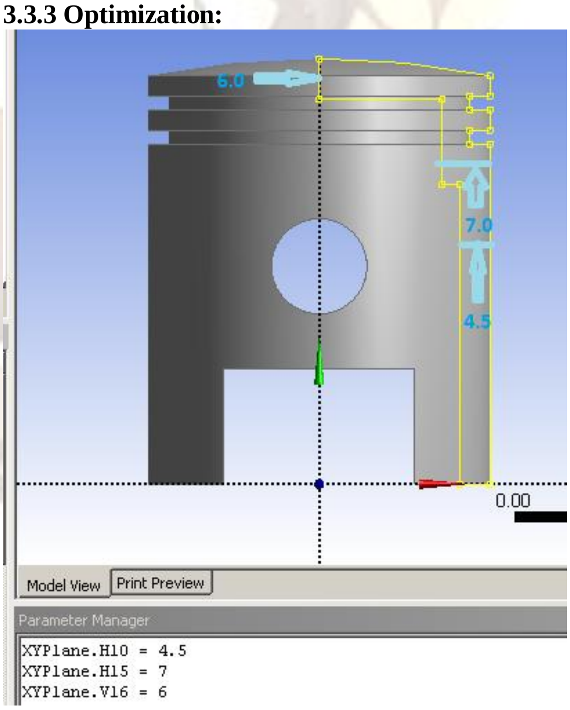



Figure 3.15: Input parameters for optimization

Table 3.2: Optimization Results

able of Schematic C4: Optimization

| | A | B | C | D | E | F |
|---|---|---|---|---|---|---|
| 1 | | P4 - XYPlane.H10 | P5 - XYPlane.H15 | P6 - XYPlane.V16 | P2 - Equivalent Stress Maximum (MPa) | P3 - Solid Mass (kg) |
| 2 | Optimization Domain | | | | | |
| 3 | Lower Bound | 3 | 5.5 | 4 | | |
| 4 | Upper Bound | 4.5 | 7 | 6 | | |
| 5 | Optimization Objectives | | | | | |
| 6 | Objective | No Objective | No Objective | No Objective | Values <= Target | Minimize |
| 7 | Target Value | | | | 124 | |
| 8 | Importance | | | | Higher | Lower |
| 9 | Constraint Handling | | | | As Hard Constraint | |
| 10 | Candidate Points | | | | | |
| 11 | Candidate A | 3.0008 | 5.5008 | 4.001 | 102.99 | 0.090153 |
| 12 | Verification A | | | | 107.26 | 0.090151 |
| 13 | Candidate B | 3.0728 | 5.5711 | 4.3961 | 85.106 | 0.092696 |
| 14 | Candidate C | 3.0968 | 5.5125 | 4.7911 | 74.891 | 0.094022 |

To study the influence of parameters on piston stress levels, number of iterations are run using optimization tool in Ansys. Through these results it was possible to choose the best value for each parameter taking into account the stress levels on the piston and the mass of the piston. The aim is to obtain an assembly as light as possible and with some safety margin.

Factor of safety =Yield point stress / Working or design stress

Automobile industries use factor of safety between 2.0 to 3.0 [8]. As piston is a critical component we are considering Factor of safety as 2.25. For Aluminum alloy, tensile yield strength is 280MPa, Tensile Ultimate strength is 310MPa. And mass of piston is 0.11912Kg.

Working or design stress = 280 / 2.25 = 124MPa

Based on above analysis the maximum stress induced in the piston is 34.7 Mpa, which is less than 124MPa (allowable stress). Hence piston is safe and there is a scope for optimization.

So from the optimization table 3.2 it is clear that the dimension H10 (4.5mm) can be reduced to 3.0mm, dimension H15 (7.0mm) can be reduced to 5.5mm, dimension V16 (6.0mm) can be reduced to 4.0mm. This results in Max equivalent stress of 100.96MPa which is less than allowable stress of 124MPa. & also solid mass is reduced to 0.090151Kg.

So from these results, piston model is modified to new dimensions and static and dynamic analysis is carried out. The results obtained are well below the working stress and mass of piston is also reduced.

## IV. RESULTS AND DISCUSSION

Table 4.1 Analysis Results Obtained Before Optimization

| Mesh Size(mm) | Equivalent Stress (Pa) | Deformation (m) |
|---|---|---|
| Static Analysis | | |
| 10 | 3.42E+07 | 2.32E-05 |
| 9 | 3.40E+07 | 2.32E-05 |
| 8 | 3.39E+07 | 2.33E-05 |
| 7.75 | 3.38E+07 | 2.34E-05 |
| 7.675 | 3.37E+07 | 2.33E-05 |
| 7.5 | 3.38E+07 | 2.34E-05 |
| 7.375 | 3.41E+07 | 2.33E-05 |
| 7.25 | 3.39E+07 | 2.33E-05 |
| 7 | 3.40E+07 | 2.35E-05 |
| 6.75 | 3.40E+07 | 2.34E-05 |
| 6.5 | 3.42E+07 | 2.34E-05 |
| 6.25 | 3.37E+07 | 2.34E-05 |
| 6 | 3.38E+07 | 2.34E-05 |
| 5.5 | 3.39E+07 | 2.34E-05 |
| 5 | 3.40E+07 | 2.35E-05 |
| 4.5 | 3.41E+07 | 2.36E-05 |
| 4 | 3.40E+07 | 2.36E-05 |
| 3 | 3.44E+07 | 2.39E-05 |
| 2 | 3.47E+07 | 2.43E-05 |
| Dynamic Analysis | | |
| 7.765 | 1.44E+06 | 3.78E-07 |
| 6.25 | 1.49E+06 | 3.80E-07 |

Table 4.2 Analysis Results Obtained After Optimization

| Mesh Size(mm) | Equivalent Stress (Pa) | Deformation (m) |
|---|---|---|
| Static Analysis | | |
| 7.675 | 1.04E+08 | 8.01E-05 |
| 6.25 | 1.04E+08 | 8.05E-05 |
| Dynamic Analysis | | |
| 7.765 | 1.69E+06 | 1.30E-06 |
| 6.25 | 1.96E+06 | 1.31E-06 |

From these optimization results it is clear that the piston was originally designed with large factor of safety as previously such sophisticated analysis and optimization tools were not available. But now a days with tools like FEA software one can analyze and optimize the design before manufacturing, resulting in cost and time saving.

## V. CONCLUSION

- The equivalent stress values obtained are well below the permissible value of 124 MPa.
- The stress obtained by theoretical calculation and FEA found to be approximately same.
- From optimization results it is clear that there is a scope for reduction in the thickness of piston skirt, piston crown wall thickness and piston crown thickness.
- Therefore optimization of piston is done and it is found that the mass of optimized piston is 0.090151Kg. Hence percentage reduction in mass compared to non-optimized piston(0.11912Kg) is 24.319 %.
- The static analysis stress results and dynamic analysis stress results obtained are well below the permissible stress value.

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