Scientific Journal of Impact Factor (SJIF): 4.14

e-ISSN (O) : 2348-4470
p-ISSN (P) : 2348-6406

International Journal of Advance Engineering and Research Development

Volume 3, Issue 9, September -2016

Design, Analysis and Optimization of Four Stroke S.I. Engine Piston using Finite Element Analysis in ANSYS software

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Abstract- *The aim of this paper is to design, analysis and optimization of four stroke S.I. engine piston, which is strong and lightweight using finite element analysis with the help of ANSYS Software. Solid Model of piston has been made using ANSYS 16.2 Geometric module and Thermo-Mechanical (Static Structural Analysis + Steady-State Thermal Analysis) analysis is done to analyze stresses, total deformation and factor of safety distribution in various parts of the piston to know the effect due to gas pressure and thermal variations using ANSYS 16.2.*

Piston optimized using Response Surface Optimization module. The thickness of piston barrel is reduced by 52.28%, the thickness of the piston crown head increased by 9.41%, the width of top land increased by 3.81%, axial thickness of the ring is increased by 2.38% and radial thickness of the ring reduced by 5.31%, resultant mass of the piston reduced by 26.07% and it's factor of safety increased by 3.072%.

Key Words: *S.I. engine piston; weight optimization; Thermo-mechanical analysis; Response surface optimization; CAE and CAD;*

I. INTRODUCTION

In the cylinder of an engine, the energy bound up in the fuel is converted into heat and pressure during the expansion stroke. The heat and pressure values increase considerably within a short period of time. The piston, as the moving part of the combustion chamber, has the task of converting part of this released energy into mechanical work. The basic structure of the piston is a hollow cylinder, closed on one side, with the segments piston crown with ring belt, piston pin boss, and skirt. The piston crown transfers the compression forces resulting from the combustion of the fuel-air mixture via the piston pin boss, the piston pin, and the connecting rod, to the crankshaft.

The most important tasks that the piston must fulfill are transmission of power from and to the working gas, sealing off the working chamber, linear guiding of the connecting rod and heat dissipation. [1]A piston should have adaptability in operating conditions, simultaneous running smoothness, low weight with sufficient shape stability, low pollutant emissions values and lowest possible friction losses inside the engine for operating smoothly.

On the basis of this piston designed according to procedure and specifications, which are given in standard machine design and data books. Solid Model of piston has been made using ANSYS 16.2 Geometric module. Thermo-Mechanical (Static Structural Analysis + Steady-State Thermal Analysis) analysis is done for Piston. Piston optimized using Response Surface Optimization module. Piston is designed for TVS scooty Pep+ four stroke S.I. engine configuration.

Nomenclature

b_1	Width of the topland
b_2	Width of the other land
D	Cylinder bore
L	Piston length
t_1	Radial thickness of the ring
t_2	Axial thickness of the ring
t_3	Maximum thickness of barrel
t_H	The piston crown Thickness

II. LITERATURE REVIEW

Heinz K. Junker, in this book, MAHLE experts share their broad-based, extensive technical knowledge of pistons, including layout, design, and testing. They write detailed information on everything to do with pistons: their function, requirements, types, and design guidelines. They describe simulation of operational strength using finite element analysis, and piston materials, cooling, and component testing. Engine testing, as well as for validating new simulation programs and systematically compiling design specifications. [1]

Ch.Venkata Rajam et al, they designed, analyzed and optimized to piston which is stronger, lighter-weight with minimum cost and with less manufacturing time. In their paper they analyzed stress distribution in the various parts of the piston to know the stresses due to the gas pressure and thermal variations using with Ansys. The Piston of an engine is designed, analyzed and optimized by using graphics software. The CATIA V5R16, CAD software for performing the design phase and ANSYS 11.0 for analysis and optimization phases are used. They reduced the volume of the piston by 24%, the thickness of barrel is reduced by 31%, width of other ring lands of the piston is reduced by 25%, von-mises stress is increased by 16% and deflection is increased after optimization. But all the parameters are well within design consideration. [2]

Ekrem Buyukkaya et al, in their paper performed thermal analyse on a conventional (uncoated) diesel piston, made of aluminum silicon alloy and steel. And then, thermal analyse are performed on pistons, coated with MgO–ZrO2 material by using ANSYS. From the obtained results, the maximum temperature value of the coated piston was shown at the piston's combustion bowl lip. Therefore, this area must be coated oversensitivity. The maximum surface temperature of the coated piston with material which has low thermal conductivity is improved approximately 48% for the AlSi alloy and 35% for the steel. The maximum surface temperature of the base metal of the coating piston is 261 °C for AlSi and 326 °C for steel, and also find out by using of ceramic coating, strength and deformation of the materials are improved. [3]

Muhammet Cerit in his paper determined the temperature and the stress distributions in a partial ceramic coated spark ignition engine's piston. Effects of coating thickness and width on temperature and stress distributions were investigated including comparisons with results from an uncoated piston. It is observed that the coating surface temperature increase with increasing the thickness in a decreasing rate. Surface temperature of the piston with 0.4 mm coating thickness was increased up to 82 °C. The normal stress on the coated surface decreases with coating thickness, up to approximately 1 mm for which the value of stress is the minimum. However, it rises when coating thickness exceeds 1 mm. As for bond coat surface, increasing coating thickness, the normal stress decreases steadily and the maximum shear stress rises in a decreasing rate. The optimum coating thickness was found to be near 1 mm under the given conditions. [4]

Xiqun Lu et al, inverse heat transfer method is employed to conduct thermal numerical analysis on a 4-ring articulated piston of marine diesel engine and determine the coefficient of heat transfer at each interface in the thermal system. The secondary motion of piston and piston ring, and the lubrication oil film has been considered in estimating the coefficient of heat transfer values. They manufactured metal plugs were installed in the head of an articulated piston and the piston skirt to measure the temperature distribution of them. A Series of thermal couples were used for cylinder temperature measurement. The boundary condition for numerical simulation is verified with experiment result and applied to predict the temperature distribution of a new piston design which had small change of piston head profile and one less ring scheme. [5]

III. DESIGN OF S.I. ENGINE PISTON

The piston is designed according to the procedure and specification which are given in machine design and data reference books. [6]

3.1 Configuration of engine

In this study single cylinder, 4 stroke, air cooled, SOHC, TVS Scooty Pep+ bike engine configuration considers for parametric design of piston. Engine Configuration shown in Table 1. [7]

Table 1. TVS Scooty Pep+ Engine Configuration

Cylinder bore, D	51 mm
Stroke, L	43 mm
Piston displacement	87.8 cc
Compression ratio	10.1:1
Maximum power in KW, (IP)	3.68@6500 rpm
Maximum torque in Nm	5.80@4000 rpm
Maximum speed	60 km/hr

3.2 Design considerations for piston

In designing a piston for an engine, the following points should be taken into consideration. [6]

(a) It should have enormous strength to withstand the high pressure.
(b) It should have minimum weight to withstand the inertia forces.
(c) It should form effective oil sealing in the cylinder.
(d) It should provide sufficient bearing area to prevent undue wear.
(e) It should have high speed reciprocation without noise.
(f) It should be of sufficient rigid construction to withstand thermal and mechanical distortions.
(g) It should have sufficient support for the piston pin.

3.3 The Piston dimensions calculation

The piston is designed according to the procedure and specification which are given in machine design and data reference books. [6]

3.3.1 Thickness of the piston (t_H)

The piston head or crown is designed according to the following two main considerations,

(a) It should have adequate strength to withstand the straining action due to pressure of explosion inside the engine cylinder, and
(b) It should dissipate the heat of combustion to the cylinder walls as quickly as possible.

On the basis of first consideration of straining action, the thickness of the piston head is determined by treating it as a flat circular plate of uniform thickness, fixed at the outer edges and subjected to a uniformly distributed load due to the gas pressure over the entire cross-section.

The piston thickness of the Piston head calculated by the following Grashoff's formula,

$$t_H = \sqrt{\frac{3pD^2}{16\sigma_t}} \qquad (1)$$

t_H = 5.6973 mm

Where

- Maximum pressure in N/mm², P = 6 N/mm²,
- Cylinder bore outside diameter of the piston in mm, D = 50.958 mm,
- Material is a particular grade of AL-Si alloy whose yield tensile strength is 285 Mpa and F.O.S. is 2.25.
- Permissible tensile stress for the material in N/mm², σ_t = 125 in N/mm²

On the basis of second consideration of heat transfer, the thickness of the piston head should be like that the heat absorbed by the piston due combustion of fuel is quickly transferred to the cylinder walls, Treating the piston head as a flat circular plate, its thickness is given by

$$t_H = \frac{H}{12.56k(t_c - t_e)} \qquad (2)$$

t_H = 5.195 mm

Where

- Heat flowing through the piston head in kJ/s or KW,

$$H = C \times HCV \times m \times B.P. \qquad (3)$$

= 2.4295 KW

- Heat conductivity in W/m/°C, K = 175 W/m/°C for
- The temperature difference ($T_C - T_E$) =75°C for aluminium alloy
- Constant that portion of the heat supplied to the engine that is absorbed by the piston, C = 0.05
- Higher calorific value of the fuel in kJ/kg, HCV = 47 × 103 kJ/ kg for petrol,
- Mass of the fuel used in kg per brake power per second, m = 0.15 KJ/Break/Hr
- Break Power in KW,

$$B.P. = 2\pi NT/60 \qquad (4)$$

= 2.4295 KW

3.3.2 Radial thickness of ring (t_1)

The radial thickness (t_1) of the ring is obtained by considering the radial pressure between the cylinder wall and ring, from bending stress consideration in ring. The radial thickness is given by

$$t_1 = D\sqrt{\frac{3p_w}{\sigma_t}} \qquad (5)$$

$t_1 = 1.7247$ mm

Where

- Cylinder bore in mm, D = 51 mm
- Pressure of fuel on cylinder wall in N/mm², Pw = 0.025 to 0.042 N/mm².
- For present material, $\sigma_t = 110$Mpa

3.3.3 Axial thickness of ring (t_2)

The thickness of the rings may be taken as

$$t_2 = 0.7t_1 \text{ to } t_1$$

Let $t_2 = 1.2073$ mm

Minimum axial thickness, $t_2 = D/(10 \times n_r) = 1.6986$ mm

Where

- Number of rings but the reference number, $n_r = 3$

3.3.4 Width of the top land (b_1)

The width of the top land varies from,

$$b_1 = t_H \text{ to } 1.2\, t_H$$

$B_1 = 6.234$ mm

3.3.5 Width of other land (b_2)

Width of other ring lands varies from,

$$b_2 = 0.75t_2 \text{ to } t_2,$$

$b_2 = 3$ mm

3.3.6 Maximum thickness of barrel (t_3)

Maximum thickness of barrel can be calculated by

$$t_3 = (0.03 \times D) + b + 4.5 \text{ mm} = 5.1534 \text{ mm}$$

Where

- b = Radial depth of piston ring groove

Table 2. calculated dimensions of piston

Dimensions	Designed Dimensions Values (mm)
Cylinder bore, D	50.957
Pistol length, L	40.5
The piston crown Thickness, t_H	5.6973
Maximum thickness of barrel, t_3	8.1534
Width of the topland, b_1	6.234
Width of the other land, b_2	3
Axial thickness of the ring, t_2	1.6
Radial thickness of the ring, t_1	1.7247

IV. FINITE ELEMENT ANALYSIS

The objective of FEA is to investigate stresses and problem area experienced by piston. Aluminium alloy is material of piston for the FEA, since this piston is also used at optimization. Thermo-mechanical analysis is used for analysis of piston. Therefore material properties of aluminium alloy are tabulated in table 3. [8]

Table 3. Material properties of aluminium alloy

Properties	Values
Density in Kg/m^3	2770
Coefficient of thermal expansion in C^{-1}	0.00035
Young modulus in MPa	71000
Bulk modulus in MPa	6960.8
Shear modulus in MPa	2669.2

Tensile yield strength in MPa	280
Compressive yield strength in MPa	280
Tensile ultimate strength in MPa	310
Poissons ratio	0.33

The model of piston is created in ANSYS Design Module as shown in figure 1, as per the calculated dimensions of piston.

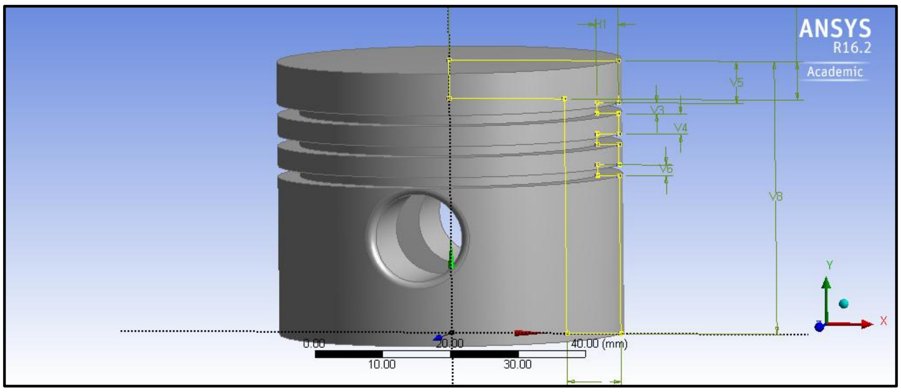


Figure 1. The model of piston is created in ANSYS Design Module

4.1 Meshing of S.I. engine piston

Figure 2, shows meshed model of piston in ANSYS designed module. A tetrahedral element was used for the solid mesh. Table 4, shows meshing properties of existing piston.

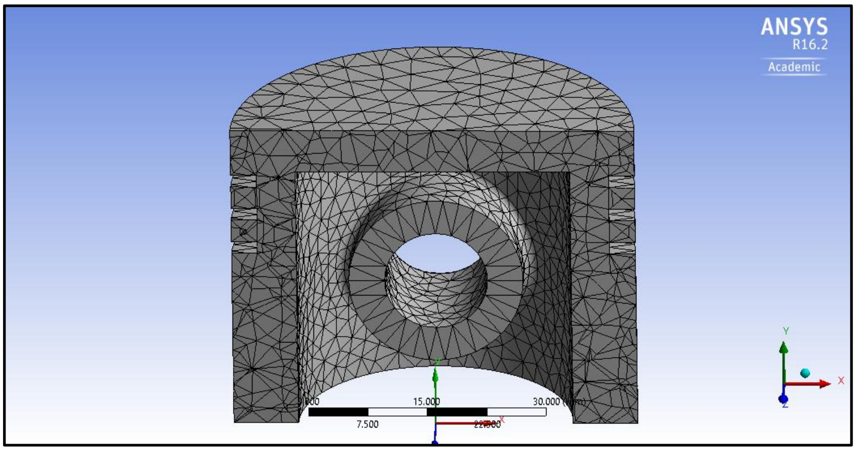


Figure 2. Meshed model of piston in ANSYS designed modular

Table 4. Meshed model of piston in ANSYS designed modular

Number of nodes	Number of elements	Size of elements
22961	13405	4.5 (mm)

4.2 Boundary conditions for analysis of S.I. engine piston using ANSYS

The piston is divided into the areas defined by a series of grooves for sealing rings. The boundary conditions for mechanical simulation were defined as the pressure acting on the entire piston head surface. It is necessary to load certain data on material that refer to both its mechanical and thermal properties to do the coupled thermo-mechanical calculations.

The temperature load is applied on different areas and pressure applied on piston head. The regions like piston head and piston ring regions are applied with large amount of heat (255°C-180°C). The convection values on the piston wall ranges from 350 W/mK to 600 W/mK. The working pressure is 2 Mpa. [4]

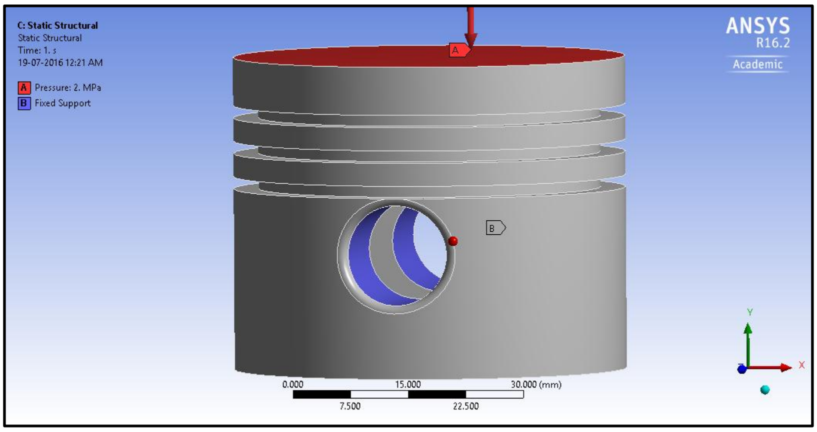


Figure 3. Static structural boundary conditions

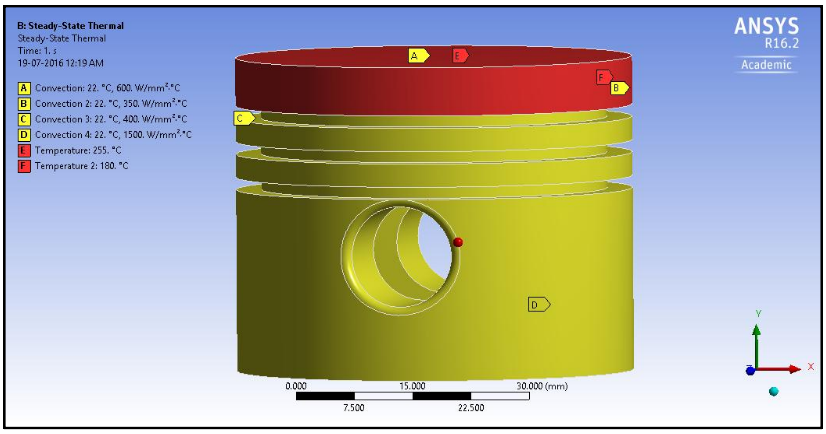


Figure 4. Static thermal boundary conditions

V. THERMO-MECHANICAL ANALYSIS OF S.I. ENGINE PISTON BEFORE OPTIMIZATION

5.1 Von-mises stress distribution due to thermo-mechanical loading

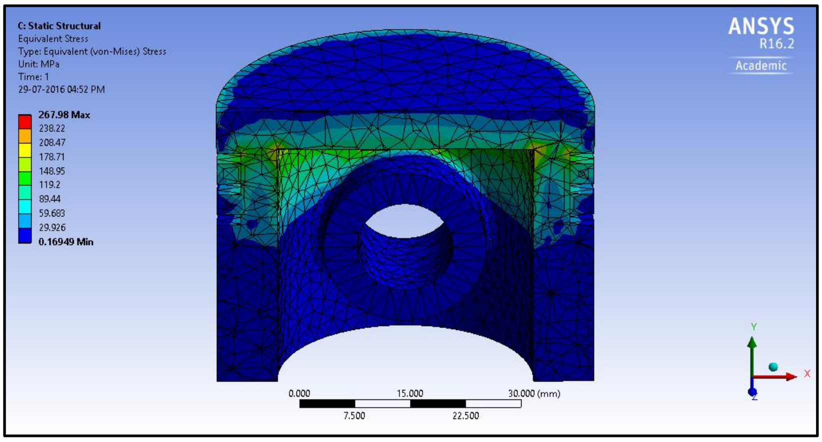


Figure 5. Equivalent von-mises stress distribution in S.I. engine for the given loading conditions

At The figure 5, signifies the von-mises stress distribution in S.I. engine for the given loading conditions. The maximum value is 267.98 MPa and minimum value is 0.16949 Mpa.

5.2 Deformation distribution due to the effect of thermo-mechanical loading

The figure 6, indicates the total deformation distribution in the S.I. engine for the given loading conditions. The maximum value is 0.142597 mm MPa and minimum value is 0 mm.

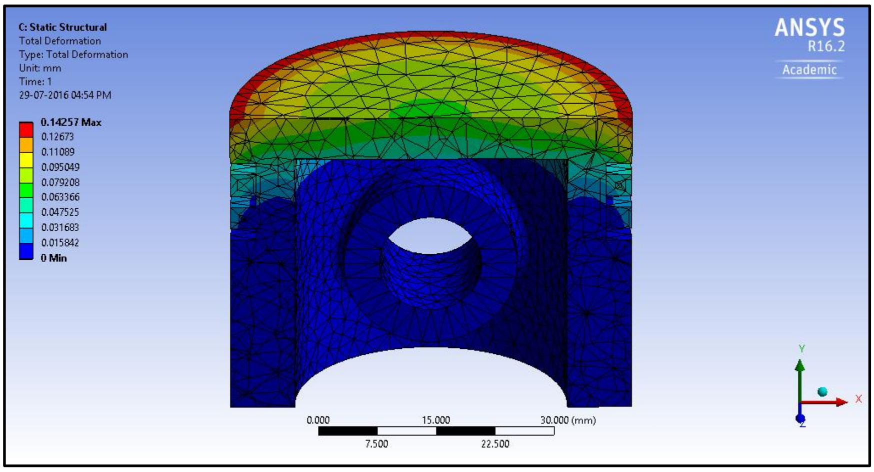


Figure 6. Total deformation distribution in the S.I. engine for the given loading conditions

5.3 Distribution of factor of safety due to effect of thermo-mechanical loading

The figure 7, indicates the Safety factor distribution in the S.I. engine Piston for the given loading conditions. The maximum value is 15 and minimum value is 1.0448.The maximum and minimum values of the Thermo-mechanical analysis results for the given loading conditions of the Aluminium Alloy Piston are mentioned in the following table 5.

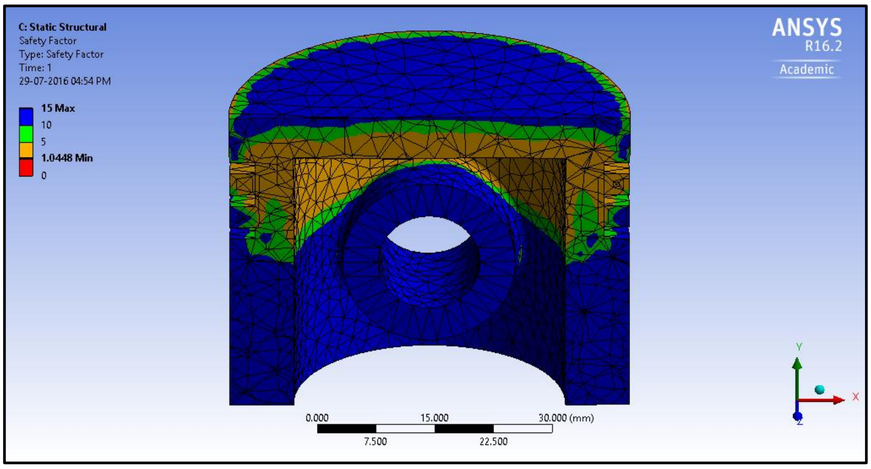


Figure 7. Safety factor distribution in the S.I. engine Piston for the given loading conditions

Table 5. The maximum and minimum values of the Thermo-mechanical analysis

Parameters	Maximum	Minimum
Equivalent Von-Mises stress (MPa)	267.98	0.16949
Total deformation (mm)	0.14257	0
Factor of Safety	15	1.0448

VI. RESPONSE SURFACE OPTIMIZATION

The objective of parametric optimization task was to minimize mass of piston under the effect of compressive gas load and temperature applied in piston parts such that the maximum and equivalent stress amplitude are within the limits of allowable stresses. Optimized model of existing piston is shown in figure. Mathematically stated, the optimization statement would appear as follows:

Objective Function: Minimize mass and increases factor of safety:

(i.)Maximum von-mises stress < Allowable or design stress
(ii.) Factor of safety>1
(iii.)Geometrical constraints with respect to other engine parts
(iv.) Following reasons where scope for material removal and addition

- (a) Axial thickness of the ring, t_2
- (b) Width of the topland, b_1
- (c) Width of the other land, b_2
- (d) Maximum thickness of barrel, t_3
- (e) The piston crown Thickness, t_H
- (f) Radial thickness of the ring, t_1

6.1 Design of experiments

A. Input parameters

(a) P21 - Axial thickness of the ring, $t_2 = 1.6$ mm

Lower Bound	1.003 mm
Upper Bound	1.76 mm

(b) P18 - Width of the topland, $b_1 = 6.234$ mm

Lower Bound	2 mm
Upper Bound	6.8574 mm

(c) P6 - Width of the other land, $b_2 = 3$ mm

Lower Bound	0.8 mm
Upper Bound	3.3 mm

(d) P16 - Maximum thickness of barrel, $t_3 = 8.1534$ mm

Lower Bound	3.8 mm
Upper Bound	8.968751 mm

(e) P17 - The piston crown Thickness, $t_H = 5.6973$ mm

Lower Bound	2 mm
Upper Bound	6.26703 mm

(f) P2 – Cylinder radius R = 25.479 mm

Constant Value	25.479 mm

(g) P22 - Radial thickness of the ring, $t_1 = 1.7247$ mm

Lower Bound	1.55 mm
Upper Bound	3.586 mm

(h) P24 – Pistol length, L = 40.5 mm

Constant Value	40.5 mm

B. Output parameters

(a) Static structural

(i.) P13 - Safety Factor Minimum

Calculated Minimum	0.15079230486212214
Calculated Maximum	1.1127836929138353

(ii.) P14 - Equivalent Stress Maximum

Calculated Minimum	251.62122862064697 MPa
Calculated Maximum	1856.8586789360352 MPa

(iii.)P15 - Total Deformation Maximum

Calculated Minimum	0.12478499032658545 mm
Calculated Maximum	0.16056266554458831 mm

(b) Study state thermal

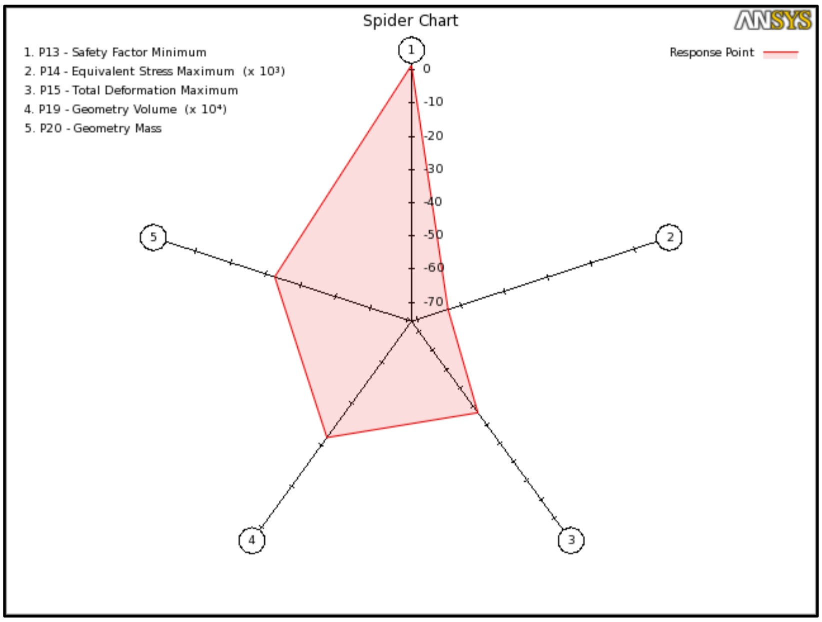
(i.) P19 - Geometry Volume

Calculated Minimum	32358.692246561193 mm^3
Calculated Maximum	50818.1147503642 mm^3

(ii.) P20 - Geometry Mass

Calculated Minimum	0.089633577522974517 Kg
Calculated Maximum	0.14076617785850884 Kg

C. Spider curve



Graph 1. Spider graph for response

6.2 Optimization

A. Objectives and constraints

(a) Maximize P13 (factor of safety); P13 >= 1

Calculated Minimum	-76.093
Calculated Maximum	1.4239

(b) Minimize P20 (Mass of geometry)

Calculated Minimum	0.078289 Kg
Calculated Maximum	0.14814 Kg

Table 6. Comparison of change in dimensions of Piston after optimization

Dimensions	Dimensions Before Optimization (mm)	Dimensions After Optimization (mm)	% Change
Cylinder bore, D	50.957	50.957	0
Piston total length, L	40.5	40.5	0
Thickness of the piston crown, t_H	5.6973	6.2339	9.41↑
Thickness of barrel, t_3	8.1534	3.8906	52.28↓
Width of the topland, b_1	6.234	6.4717	3.81↑
Width of the other land, b_2	3	0.86688	71.1↓
Axial thickness of the ring,	1.6	1.6381	2.38↑
Radial thickness of the ring,	3.26	1.5546	52.31↓

↑ Increase in quantity, ↓ Decrease in quantity

VII.RESULT AND COMPARISON OF S.I. ENGINE PISTON

7.1 Reduction in Geometric mass of piston after optimization

Table 7. Reduction in Geometric mass of piston after optimization

Original	Optimized	Reduction (Percentage)
0.13615 Kg	0.10065 Kg	26.074%

7.2 Von-Mises stress after optimization due to thermo-mechanical loading

The figure 8, indicates the equivalent von-mises stress distribution in the optimized S.I. engine piston for the given loading conditions. The maximum value is 259.99 MPa and minimum value is 0.10009 MPa.

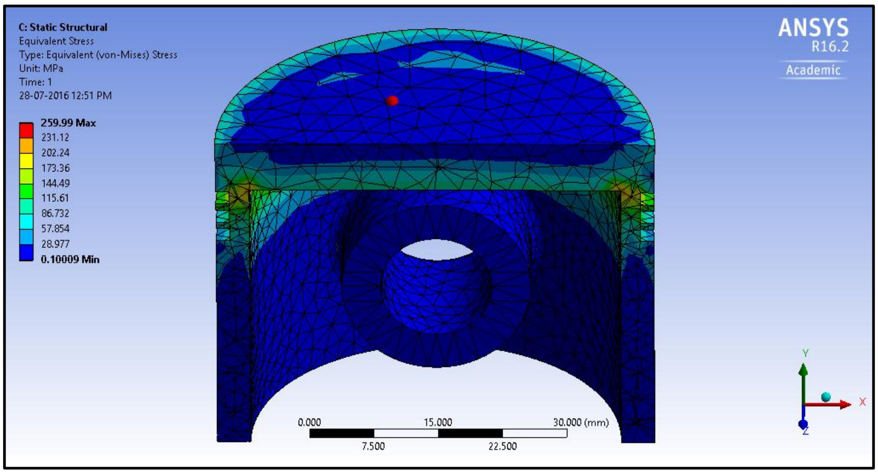


Figure 8. Von-Mises stress after optimization due to thermo-mechanical loading

Table 8. Comparison Von-Mises stress after optimization due to thermo-mechanical loading

Before optimization	After optimization	Reduction (Percentage)
267.98 MPa	259.99 MPa	2.982%

7.3 Deformation after optimization due to thermo-mechanical loading

The figure 9, indicates deformation after optimization due to thermo-mechanical loading. The maximum value is 0.14214 mm and minimum value is 0 mm.

Table 9. Comparison of Deformation after optimization due to thermo-mechanical loading

Before optimization	After optimization	Reduction (Percentage)
0.14257	0.14214	0.3%

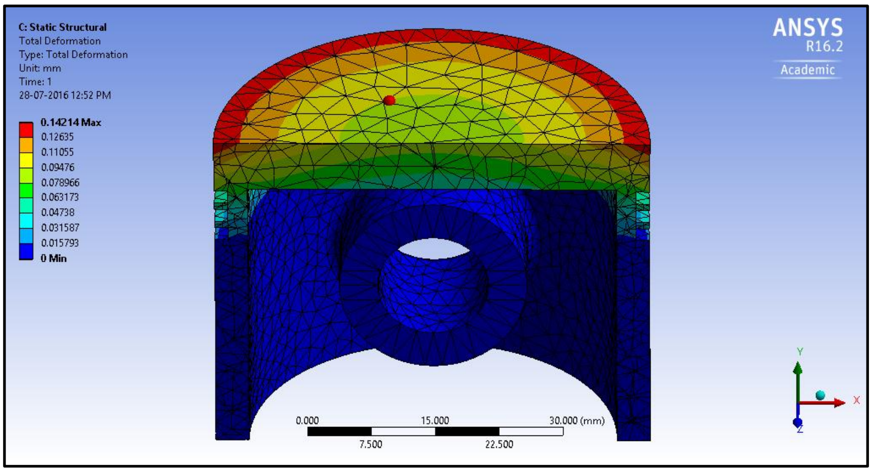


Figure 9. Deformation after optimization due to thermo-mechanical loading

7.4 Factor of safety after optimization due to thermo-mechanical loading

The figure 10, indicates the factor of safety distribution in the optimized S.I. engine piston for the given loading conditions. The maximum value is 15 and minimum value is 0.

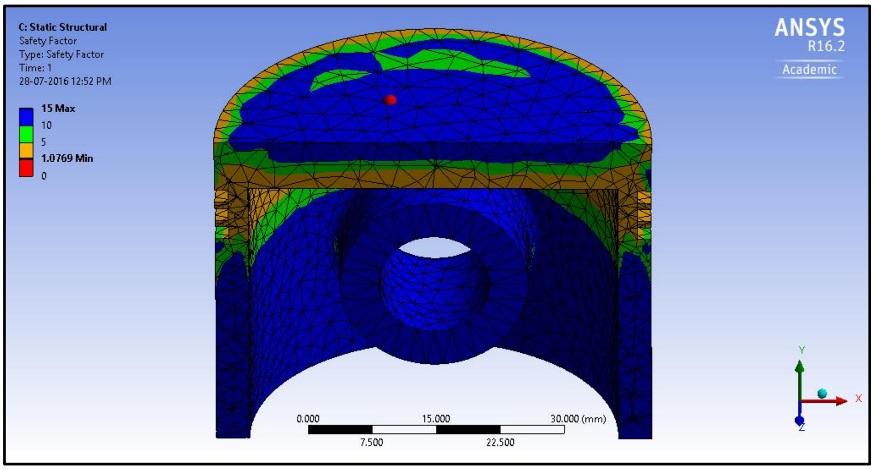


Figure 10. Factor of safety distribution in S.I. piston

Table 10, Comparison of Factor of safety after optimization due to thermo-mechanical loading

Before optimization	After optimization	Reduction (Percentage)
1.0448	1.0769	3.072%

VIII. CONCLUSION

Solid Model of piston has been made using ANSYS 16.2 Geometric module. Thermo-Mechanical (Static Structural Analysis + Steady-State Thermal Analysis) analysis is done for Piston. Here optimization of Piston has been done on Response Surface Optimization module.

On the basis of this study following conclusions have been made:

1. After Response Surface Optimization percentage of weight reduces is 26.074%, Resultant percentage of increase in factor of safety, percentage of decease in equivalent von-mises stress is 3.072%, 2.982% respectively.

2. After optimization Equivalent von-mises stress, factor of safety and weight is 259.99 MPa, 1.0769 and 0.10065 Kg respectively.
3. Here factor of safety is greater than 1 and Maximum Equivalent Von-mises stress is less than yield stress of aluminium alloy so designed model of piston is safe.

Acknowledgements

The authors sincerely thank to Department of Mechanical Engineering, Samrat Ashok Technological Institute, Vidisha for support, co-operation and encouragement that enabled this project.

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