Design Analysis and Optimization of Piston using CATIA and ANSYS

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ABSTRACT

Aim: Design, Analysis and optimization of piston which is stronger, lighter with minimum cost and with less time. Since the design and weight of the piston influence the engine performance.
Study Design: Analysis of the stress distribution in the various parts of the piston to know the stresses due to the gas pressure and thermal variations using with Ansys.
Methodology: 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.
Brief Results: The volume of the piston is reduced by 24%, the thickness of barrel is reduced by 31%, width of other ring lands of the piston is reduced by 25%, Vonmisses stress is increased by 16% and Deflection is increased after optimization. But all the parameters are well with in design consideration.

Key Words: CATIA V5R16, ANSYS, Vonmisses Stress, Computer CAD, CAE, Optimization.

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. R&D and testing 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 products [1]. A piston is a moving component that is contained by a cylinder and is made gas-tight by piston rings. In an engine its purpose is to transfer from expanding gas in the cylinder to the crank shaft via piston rod and or connecting rod. As an important part in an engine piston endures the cyclic gas pressure and inertia forces at work and this working condition may cause the fatigue damage of the piston. The investigations indicate that greatest stress appears on the upper end of the piston and stress concentration is one of the mainly reason for fatigue failure[2]

MODELING

Piston Design

The piston is designed according to the procedure and specification which are given in machine design and data hand books. The dimensions are calculated in terms of SI Units. The pressure applied on piston head, temperatures of various areas of the piston, heat flow, stresses, strains, length, diameter of piston and hole, thicknesses, etc., parameters are taken into consideration

Design Considerations for a Piston

In designing a piston for an engine, the following points should be taken into consideration:

- It should have enormous strength to withstand the high pressure.
- It should have minimum weight to withstand the inertia forces.
- It should form effective oil sealing in the cylinder.
- It should provide sufficient bearing area to prevent undue wear.
- It should have high speed reciprocation without noise.
- It should be of sufficient rigid construction to withstand thermal and mechanical distortions.

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- It should have sufficient support for the piston pin.
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Procedure for Piston Design

The procedure for piston designs consists of the following steps:

- Thickness of piston head (t_H)
- Heat flows through the piston head (H)
- Radial thickness of the ring (t_1)
- Axial thickness of the ring (t_2)
- Width of the top land (b_1)
- Width of other ring lands (b_2)

The above steps are explained as below:

Thickness of Piston Head (t_h)

The piston thickness of piston head calculated using the following Grashoff's formula,$t_H = \sqrt{(3pD^2)/(16\sigma_t)}$ in mm

Where

P= maximum pressure in N/mm²

D= cylinder bore/outside diameter of the piston in mm.

σ_t=permissible tensile stress for the material of the piston.

Here the material is a particular grade of AL-Si alloy whose permissible stress is 50 Mpa-90Mpa. Before calculating thickness of piston head, the diameter of the piston has to be specified.The piston size that has been considered here has a L*D specified as 152*140.

Heat Flow through the Piston Head (H)

The heat flow through the piston head is calculated using the formulaH = $12.56 * t_H * K * (T_c - T_e)$ Kj/sec

Where

K=thermal conductivity of material which is 174.15W/mkT_c = temperature at center of piston head in °C.

T_e = temperature at edges of piston head in °C.

Radial Thickness of Ring (t_1)

$t_1 = D\sqrt{3p_w/\sigma_t}$

Where D = cylinder bore in mm

P_w= pressure of fuel on cylinder wall in N/mm². Its value is limited from 0.025N/mm²to 0.042N/mm². For present material, σ_t is 90Mpa

Axial Thickness of Ring (t_2)

The thickness of the rings may be taken as

$t_2 = 0.7t_1$ to t_1Let assume t_2 =5mm

Minimum axial thickness (t_2)

= $D/(10*n_r)$

Where n_r = number of rings

Width of the top land (b_1)

The width of the top land varies from

$b_1 = t_H$ to $1.2\, t_H$

Width of other lands (b_2)

Width of other ring lands varies from

$b_2 = 0.75t_2$ to t_2

Maximum Thickness of Barrel (t_3)

$t_3 = 0.03*D + b + 4.5$ mm

Where

b = Radial depth of piston ring groove

Thus, the dimensions for the piston are calculated and these are used for modeling the piston in CATIA V5R16. In the above procedure the ribs in the piston are not taken into consideration, so as make the piston model simple in its design. In modeling a piston considering all factors will become tedious process. Thus, a symmetric model is developed using the above dimensions [3].

The Piston Model before optimization

The following are the sequence of steps in which the piston is modeled.

- Drawing a half portion of piston
- Exiting the sketcher
- Developing the model
- Creating a hole
- Applying fillets

Piston was modeled using CATIA V5 software which is shown in Fig.1

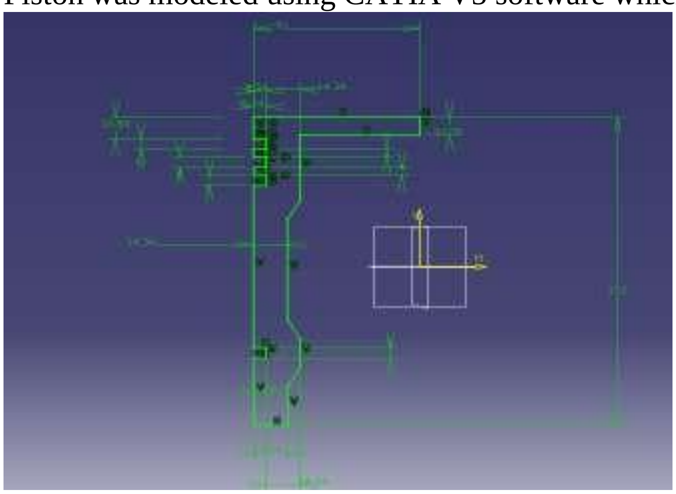

Fig.1.Sketch of the Piston before optimization

It was then imported to ANSYS 11.0. for analysis and optimization. Specifications of piston before optimization are shown in Table 1.

Table1.Design Specification before optimization

S.No.	Dimensions	Size in mm
1	Length of the Piston(L)	152
2	Cylinder bore/outside diameter of the piston(D)	140
3	Thickness of piston head (t_H)	9.036

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4	Radial thickness of the ring (t_1)	5.24
5	Axial thickness of the ring (t_2)	5
6	Width of the top land (b_1)	10
7	Width of other ring lands (b_2)	4

The CAD and FEA of Piston

The design of the piston starts with the definition of the piston geometry using 3D CAD software. This 3D CAD geometric model is then imported to FEA software and analysed under the predicted service conditions before anything is made. That speeds up the design and testing process, reduces the lead time to create new pistons designs, and produces a better product. The idea behind finite analysis is to divide a model piston into a fixed finite number of elements. Computer software generates and predicts the overall stiffness of the entire piston. Analyzing the data it is possible predict how the piston will behave in a real engine and allows the engineer to see where the stresses and temperatures will be the greatest and how the piston will behave [4].
Analysis of the piston is done to optimize the stresses and minimize the weight using ANSYS. The mathematical model of optimization is established firstly, and the FEA is carried out by using the ANSYS software. Based on the analysis of optimal result, the stress concentrates on the piston has become evaluate, which provides a better reference for redesign of piston.

2.5 Meshing of piston before optimization

Element used is 20 node Tetrahedron named soilid90 [5]. The element size is taken as 5, then total number elements were 57630 and nodes were 91176 found in meshed model. The Figure 2. Shows meshed model of the piston.

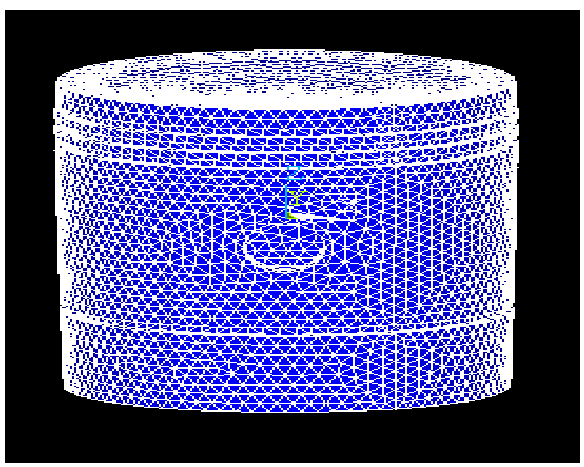

Fig. 2. Meshed Model of the Piston

Thermal and Geometric Properties

It is important to calculate the piston temperature distribution in order to control the thermal stresses and deformations within acceptable levels. The temperature distribution enables us to optimize the thermal aspects of the piston design at lower cost, before the first prototype is constructed. As much as 60% of the total engine mechanical power lost is generated by piston ring assembly. The piston skirt surface slides on the cylinder bore. A lubricant film fills the clearance between the surfaces. The small values of the clearance increase the frictional losses and the high values increase the secondary motion of the piston. Most of the Internal Combustion

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(IC) engine pistons are made of an aluminum alloy which has a thermal expansion coefficient, 80% higher than the cylinder bore material made of cast iron. This leads to some differences between running and the design clearances. Therefore, analysis of the piston thermal behaviour is extremely crucial in designing more efficient engine[6]. The thermal and geometric properties are as shown in below table 2

Table 2. Thermal and Geometric Properties

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, Dens	2.77e-6 kg/mm³

Applying Temperatures, Convections and Loads

The piston is divided into the areas defined by a series of grooves for sealing rings. E. The boundary conditions for mechanical simulation were defined as the pressure acting on the entire piston head surface (maximum pressure in the engine cylinder). 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 [7].

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 (160°C-200°C). The convection values on the piston wall ranges from 232W/mK to 1570W/mK. The working pressure is 2Mpa.

3. OPTIMIZATION

After generating an accurate finite element model a strategy for the optimization workflowwas defined. Target of the optimization was to reach a mass reduction of the piston[8].

Objective Function: Minimize massSubject to constraints:

(i) Maximum Vonmises stress<Allowable or design stress
(ii) Factor of safety>1.2
(iii) Manufacturing constraints
(iv) After carrying out static structural analysis the stresses in each loading conditions were studied and then area where excess material can be removed were decided so that maximum vonmises stress does not exceed allowable and factor of safety is kept above 1.5 .
(v) Following reasons where scope for material removal
a. Radial Thickness of the ring
b. Axial Thickness of the ring
c. Maximum Thickness of the Barrel
d. Width of the Top Land
e. Width of other ring lands

Meshing of piston after optimization

The meshing of the piston after optimization is done with the same element structure and size i.e., taken before optimization. The total number elements were 78221 and nodes were 47286 found in meshed model.

The Piston Model after optimization

The optimized model of the piston is as shown in below Fig. 3.

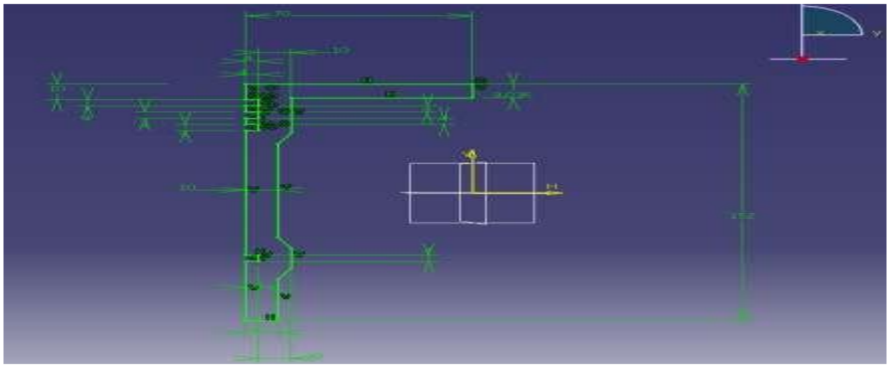


Fig. 3. Sketch of the Piston after optimizationThe model of the piston as shown in below Figure 4.

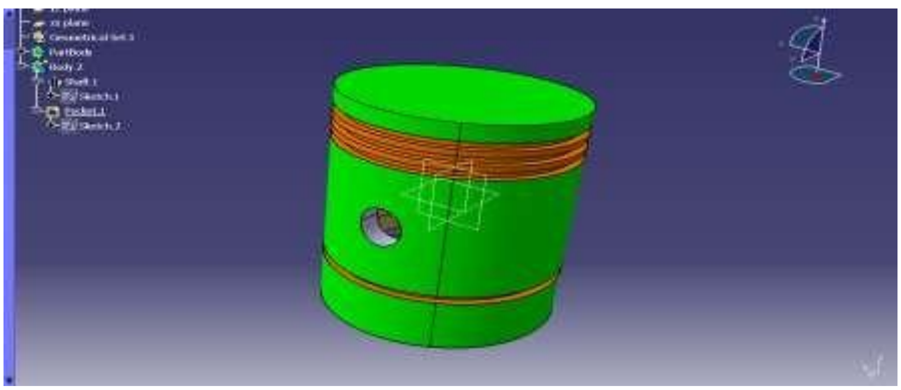
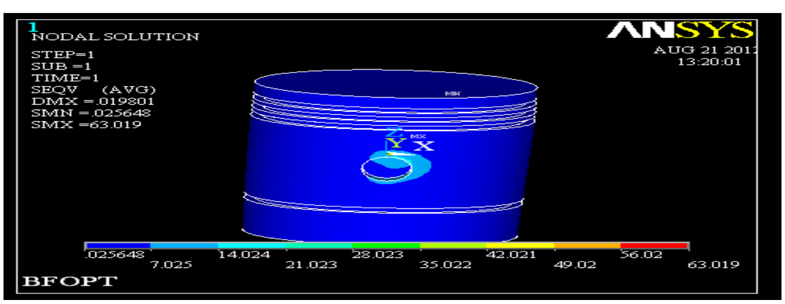


Fig. 4. Model of the piston

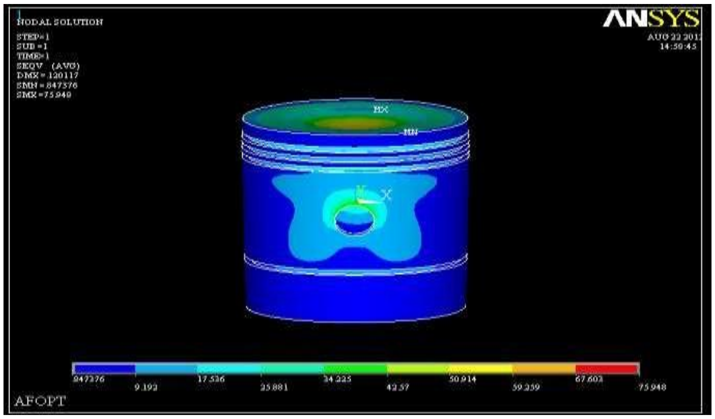
4. RESULTS AND DISCUSSION

The following images are shown for resulted von-mises stresses before and after optimization

i). Vonmisses Stress before Optimization



(ii). Vonmisses Stress after Optimization



The optimized values after optimization using ANSYS are given in the following Table 3

Table 3. Design values after optimization

S.No.	Parameter	BEFORE OPTIMIZATION	AFTER OPTIMIZATION	DESIGN CONSIDERATION
1	Volume	997021 mm^3	752994mm^3	752994mm^3
2	Radial thickness of the ring (t_1)	5.24 mm	3.46 mm	4 mm
3	Axial thickness of the ring (t_2)	5 mm	3.52 mm	4 mm
4	Maximum Thickness of Barrel (t_3)	14.34 mm	9.08 mm	10 mm
5	Width of the top land (b_1)	10.84 mm	9.36 mm	10 mm
6	Width of other ring lands (b_2)	4 mm	3.24 mm	3 mm
7	Von mises stress	63.019Mpa	75.95Mpa	63-76Mpa
8	Deflection	0.0198 mm	0.120mm	0.120mm

The length 152mm of the piston obtained is same as before and after optimization process. The value after optimization is taken into consideration. It is not considerable that the variations in piston length. Applying the temperature and pressure loads on different areas of the piston and heat flow has not affected the length.

The diameter also same even after optimization process i.e., 140mm and is taken into consideration for design purposes. The heat flow in the piston material and pressure on the head has not affected in length and diameter as these are larger than other parameters. So the piston can withstand easily on sizes of these parameters.

The volume has varied after applying temperature and pressure loads over the piston as volume is depends on not only on length and diameter and also on thicknesses. The volumetric size after optimization is taken into consideration. The volume of piston before optimization is 997021 mm^3 and for the same after optimization is 752994mm^3

The radial thickness of the piston has affected more as it is very small in size and the temperature and heat flow are very high to this size of thickness. Before optimization value is given as 5.24mm and obtained after optimization is 3.46mm. This is rounded to next highest value i.e., 4mm and is taken into consideration for design. The axial thickness of the piston ring before optimization is 5mm, it is changed to 3.52mm after optimization, since the more and more heat and stress applied through groves as it is very near to the head of the piston. This is rounded to next highest value i.e., 4mm is taken into consideration for design.

The maximum thickness of the barrel before optimization is 14.34mm has much affected in variation of size after applying pressure and temperature loads and is changed to 9.08mm and rounded to next highest value i.e., 10mm taken into consideration.

The width of the top land has not much affected while comparing with the maximum thickness of the barrel. The initial value i.e., before optimization is 10.84mm and is changed after applying pressure which is directly applied on the head i.e., top of the piston as a result the shape of the piston on top will become just like a bowl. The value after optimization is obtained as 9.36mm and it is rounded to 10mm. This value is considerable for design.

The width of the other lands i.e., near piston rings are 4mm in size and is changed due to pressure and heat applied on rings through groves. The value after optimization is 3.24mm and is rounded to 3mm.

The von misses stress initially was 63.019Mpa, after optimization it is obtained as 75.95Mpa and it is permissible up to 90Mpa. So the piston with these considerations can withstand easily.

The deflection due to pressure applied is more than that of temperature applied. In this analysis the pressure as well as temperature loads are taken into consideration for applying on the piston. The deflection before optimization is given as 0.0198mm and after optimization it is obtained that 0.120mm, this value is taken into consideration for design purpose.

The values obtained after applying the loads are depend on the area chosen by the user. The factor of safety is as follows:

A. The mass of the piston before optimization is 2.72. and factor of safety of is 1.42

B. The mass of the piston after optimization is 2.08. and factor of safety of is 1.2

5. CONCLUSION

The deflection due to pressure applied after optimization is more than before optimization and this value is taken into consideration for design purpose. The stress distribution on the piston mainly depends on the deformation of piston. Therefore, in order to reduce the stress concentration, the piston crown should have enough stiffness to reduce the deformation.

All the phases in this project given can be extended to the piston design with reduction of material at bottom. The material is removed to reduce the weight of the piston so as to improve the efficiency. It is essential to obtain the optimized results for new piston with reduced material.

In brief:

1 The optimal mathematical model which includes formation of piston crown and quality of piston ad piston skirt.

2. The FEA is carried out for standard piston model used in diesel engine ceramic coating on crown and the result of analysis indicate that the maximum stress has changed from 63.019Mpa to 74.95Mpa.

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