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Stress Optimization of S.I. Engine Piston

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Abstract: ***The performance of any automobile largely depends on its size and working in dynamic conditions. The piston is a "heart" of the engine and its working condition is the worst one of the key parts of the engine in the working environment. The good design of the piston optimization can lead to a mass reduction on the base of stress analysis satisfying the requirements of automobile specifications with cost and size effectiveness. Piston is the part of engine which converts heat and pressure energy liberated by fuel combustion into mechanical works. Engine piston is the most complex among automotive. This paper work describes stress optimization of Piston for I.C. Engine. The stress distribution of piston by using FEM to be investigate and analyze. The stresses due to combustion are considered to avoid the failure of the piston. Intensity of structural stresses should be reduced to have safe allowable limits. This paper introduces an analytical study of the structural effects on the piston head.CAD software Creo is used to model the piston and stress optimization is performed by using Creo-Simulate. The paper describes the piston optimization with using finite element analysis technique to predict the higher stress and critical region on the component. The optimization is carried out to reduce the stress concentration on the head of the piston.***

Keywords: piston head, piston crown, piston skirt, Finite element analysis (FEA), Creo-simulate.

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. 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**.** 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. Failure of piston is due to stress concentration is one of the mainly reason for fatigue failure**.**

General piston is cylinder, but according to the different working conditions and requirements, the construction of the piston can be various. Generally the piston is divided into three parts: head, the skirt and piston pin. The head is the part of the piston top and ring groove. The piston top is completely depending on the requirements of the combustion chamber, the top is designed with flat or nearly flat to be conducive to reduce the area contacting with the high temperature gas, so that the stress can be distributed uniformity. Most gasoline engine uses flat top piston. Piston is designed to be a complex shape, with a depth pit as a part of the combustion chamber, to meet the need of mixture gas in order to improve the combustion efficiency and reduce the deflagration to the minimum extent. The recess of the piston is for mounting the piston ring. The piston ring is sealed to prevent air leakage and prevent the oil entering the combustion chamber.

The skirt is the lower part of the piston. It keeps the piston working in the reciprocating movement of the vertical posture. That is, it‟s the guide portion of the piston. The shape of the piston skirt is very particular, especially like the light passenger cars. The designer considers the skirt from the engine structure and performance to make the engine‟s structure compact and smooth operation. The piston pin is the supporting portion via a piston pin and connecting rod, its located above the piston skirt.

2. Problem Formulation

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.

During power stroke, the pressure on piston will increase and produce high stresses on piston and on piston skirt. Due to high pressure and temperature in the combustion engine friction will increase and the piston skirt will expand. This friction can produce wear and scratch at piston skirt. And it also produces high stresses on piston head. The high stresses and the friction will damage the piston and decrease the power output of the engine.

3. Methodology

- Theoretical Design calculation of piston
- Create a 3D model of piston for two stroke engine using Creo 2.0
- Develop a Finite Element Model for stress optimization of a piston head using Creo-Simulate.
- Analyze piston using static stress analysis method Optimize the model for mass reduction.

3.1 Theoretical Design calculation of piston

The piston is a disc which reciprocates within a cylinder. It is either moved by the fluid or it moves the fluid which enters the cylinder. The main function of the piston of an internal combustion engine is to receive the impulse from the expanding gas and to transmit the energy to the crankshaft through the connecting rod. The piston must also disperse a large amount of heat from the combustion chamber to the cylinder walls.

Main parts of piston

1) **Head or crown.** The piston head or crown may be flat, convex or concave depending upon the design of combustion chamber. It withstands the pressure of gas in the cylinder.
2) **Piston rings.** The piston rings are used to seal the cylinder in order to prevent leakage of the gas past the piston.
3) **Piston Skirt.** The skirt acts as a bearing for the side thrust of the connecting rod on the walls of cylinder.
4) **Piston pin.** It is also called gudgeon pin or wrist pin**.** It is used to connect the piston to the connecting rod.

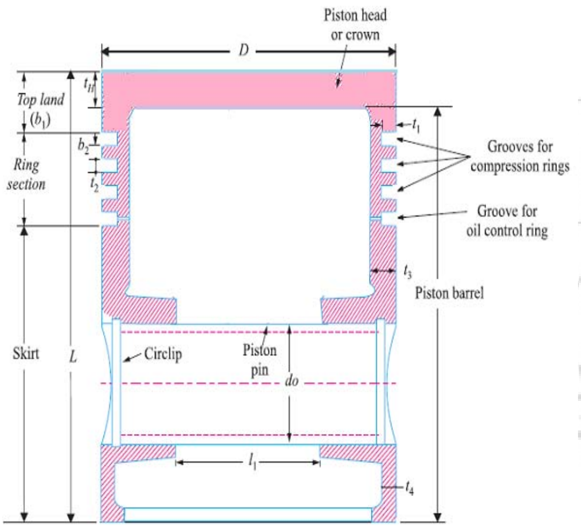

Figure 1: Piston Parts

3.2 Design Considerations for a Piston

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

1) It should have enormous strength to withstand the high gas pressure and inertia forces.
2) It should have minimum mass to minimize the inertia forces.
3) It should form an effective gas and oil sealing of the cylinder.
4) It should provide sufficient bearing area to prevent undue wear.
5) It should have high speed reciprocation without noise.
6) It should be of sufficient rigid construction to withstand thermal and mechanical distortion.
7) It should have sufficient support for the piston pin.

3.3 Material for Pistons

The most commonly used materials for pistons of I.C. engines are cast iron, cast aluminum, forged aluminum, cast steel and forged steel. The cast iron pistons are used for moderately rated engines with piston speeds below 6 m/s and aluminum alloy pistons are used for highly rated engines running at higher piston speeds. The pistons of aluminum alloys are coated with aluminum oxide by an electrical method.

3.4 Procedure for Piston Design parameters:

The procedure for piston designs consists of the following steps:

- Thickness of piston head (tH)
- Thickness of the oil ring (t1)
- Thickness of compression ring (t_2)
- Piston barrel thickness (t_3)
- Piston skirt (P)
- Piston Pin (d0)

3.5 Design of Piston Head or Crown

The piston head or crown is designed keeping in view the following two main considerations, i.e.
1. It should have adequate strength to withstand the straining action due to pressure of explosion inside the engine cylinder.
2. 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 thickness of the piston head (tH), according to Grashoff's formula is given by

$$tH = \frac{\sqrt{3p.D^2}}{16.\sigma t}\ (\text{in mm}) \ .... \mathbf{1}$$

where, p = Maximum gas pressure or explosion pressure in N/mm^2
D = cylinder bore or outside diameter of the piston in mm, and
σt = Permissible bending (Tensile) stress for the material of the piston in MPa or N/mm^2.
It may be taken as 80 MPa to 110 MPa for aluminum alloy on the basis of second consideration of heat transfer, the thickness of the piston head should be such 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

$$tH = \frac{H}{12.56k\ (Tc-Te)}\ (\text{In mm}) .... \mathbf{2}$$

where, H = Heat flowing through the piston head in KJ/s or watts.
k = Heat conductivity factor in W/m/°C. Its value is 46.6 W/m/°C for grey cast iron, 51.25 W/m/°C for steel and 174.75 W/m/°C for aluminum alloys.
Tc = Temperature at the centre of the piston in °C, and
Te = Temperature at the edges of the piston head in °C.
The temperature difference (Tc-Te) may be taken as 220°C for cast iron and 75°C for aluminium.
The heat flowing through the position head (H) may be determined by the following expression, i.e.

H = C x HCV x m x B.P. (in kW)

where, C = constant representing that portion of the heat supplied to the engine which is absorbed by the piston its value is usually taken as 0.005
HCV = Higher calorific value of the fuel in KJ/kg. It may be taken as $45x10^3$ KJ/kg for diesel and $45x10^3$ KJ/kg for petrol.
m = Mass of the fuel used in kg per brake power per second, and

B.P.= Brake Power of the engine per cylinder

The thickness of the piston head (tH) is calculated by using equations (i) and (ii) and larger of the two values obtained should be adopted

When tH is 6 mm or less, then no ribs are required to strengthen the piston head against gas loads. But when tH is greater than 6 mm, then a suitable number of ribs at the centre line of the boss extending around the skirt should be provided to distribute the side thrust from the connecting rod and thus to prevent distortion of the skirt. The thickness of the ribs may be takes as

tH / 3 to tH / 2.

3.6 Piston Ring

The piston rings are used to impart the necessary radial pressure to maintain the seal between the piston and the cylinder bore. These are usually made of grey cast iron or alloy cast iron because of their good wearing properties and also they retain spring characteristics even at high temperatures. The piston rings are of the following two types:

1. Compression rings or pressure rings
2. Oil control rings or oil scraper

The compression rings or pressure rings are inserted in the grooves at the top portion of the piston and may be three to seven in number. These rings also transfer heat from the piston to the cylinder liner and absorb some part of the piston fluctuation due to the side thrust.

The oil control rings or oil scrapers are provided below the compression rings. These rings provide proper lubrication to the liner by allowing sufficient oil to move up during upward stroke and at the same time scrap the lubricating oil from the surface of the liner in order to minimize the flow of the oil to the combustion chamber.

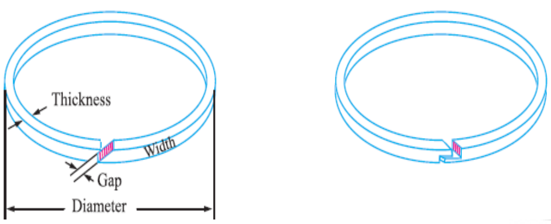

Figure 2: Diagonal cut **fig.3** Step cut

The radial thickness ($t1$) of the ring may be obtained by considering the radial pressure between the cylinder wall and the ring. From bending stress consideration in the ring, the radial thickness is given by

$$D\sqrt{\frac{3Pw}{\sigma t}}$$

Where, D = Cylinder bore in mm

P_w = pressure gas on the cylinder wall in N/mm^2. its value is limited from 0.025 N/mm^2 to 0.042 N/mm^2, and

σt = Allowable bending (Tensile) stress in MPa.

The axial thickness (t2) of the rings may be taken as 0.7t1 to t1

The minimum axial thickness (t2) may also be obtained from the following empirical relation

$$t_2 = \frac{D}{10Nr}$$

Where, Nr = Number of rings.

The width of the top land (i.e. the distance from the top of the piston to the first ring groove) is made larger than other ring lands to protect the top ring from high temperature conditions existing at the top of the piston,

i.e. Width of top land,

b1 = tH to 1.2tH

The width of other ring lands (i.e. the distance between the ring grooves) in the piston may be made equal to or slightly less than the axial thickness of the ring (t2)

i.e. Width of other ring lands,

b2 = $0.75t_2$ to t_2

The depth of the ring grooves should be more than the depth of the ring so that the does not take any piston side thrust. The gap between the free ends of the ring is given by $3.5t_1$ to 4 t_2. The gap when the ring is in the cylinder, should be 0.002D to 0.004D.

3.7 Piston barrel thickness

It is a cylindrical portion of the piston. The maximum thickness (t_3) of the piston barrel may be obtained from the following empirical relation

t_3 = 0.03D + b + 4.5 mm

where, b = Radial depth of piston ring groove which is taken as 0.4 mm larger than the radial thickness of the piston ring (t1)

= t1 + 0.4 mm

Thus, the above relation may be written as

t_3 = 0.03D + t1 + 4.9 mm

The piston wall thickness (t4) towards the open end is decreased and should be taken as 0.25 t3 to 0.35 t3.The outside diameter of the piston pin (*d0*) is determined by equating the load on the piston due to gas pressure (*p*) and the load on the piston pin due to bearing pressure (pb1) at the small end of the connecting rod bushing.

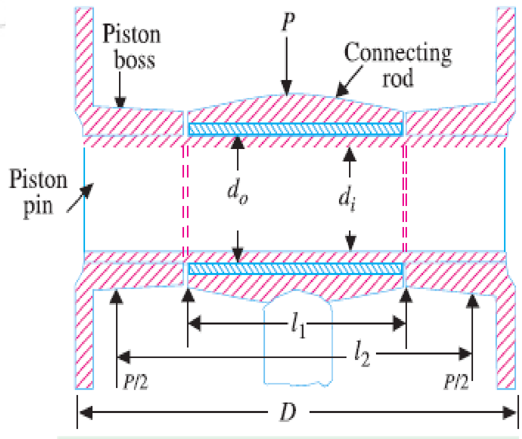

Figure 4: Piston pin details

Let,

d0 = Outside diameter of the piston pin in mm

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l_1 = length of the piston pin in the bush of the small end of the connecting rod in mm. its value is usually taken as 0.45D.

Pb_1 = Bearing pressure at the small end of the connecting rod bushing in N/mm^2. Its value for the bronze bushing may be taken as 25 N/mm^2.

We know that,

load on the piston due to gas pressure or gas load

$$= \frac{\pi D^2}{4} \times p \text{ ......i}$$

and load on the piston pin due to bearing pressure or bearing load

= Bearing Pressure x Bearing area

$= Pb_1 \times d_0 \times l_1$ ii

From equation (i) and (ii), the outside diameter of the piston pin (d0) may be obtained

3.8 Design Specification

Piston size that has been considered here has an L*D specified as 70*60. And from all the above expressions the below tabulated parameters are calculated. The maximum gas pressure acts on piston head is 60 bar.

Table 1: Design Dimension

Sr. No.	Design- Dimensions	Size in mm
1	Length of the Piston (L)	70
2	Cylinder bore / Outside diameter of the piston (D)	60
3	Radial thickness of the ring (t_1)	2.1
4	Axial thickness of the ring (t_2)	2
5	Maximum thickness of barell (t_3)	8.8
6	Thickness of piston head or crown (tH)	5.8
7	Width of the top land (b_1)	5.8
8	Width of other ring lands (b_2)	1.5

The dimensions for the piston are calculated and these are used for modeling the piston in Creo-2.0. In the above procedure the ribs in the piston are not taken into consideration, so as make the piston model simple in its design.

4. Geometry Modeling

Creo-2.0, PTC's parametric, integrated 3D CAD/CAM/CAE solution, is used by discrete manufacturers for mechanical engineering, design and manufacturing. Creo-2.0 was the industry's first successful parametric, 3D CAD modeling system. The parametric modeling approach uses parameters, dimensions, features, and relationships to capture intended product behavior and create a recipe which enables design automation and the optimization of design and product development processes. This powerful and rich design approach is used by companies whose product strategy is family-based or platform-driven, where a prescriptive design strategy is critical to the success of the design process.

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 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.

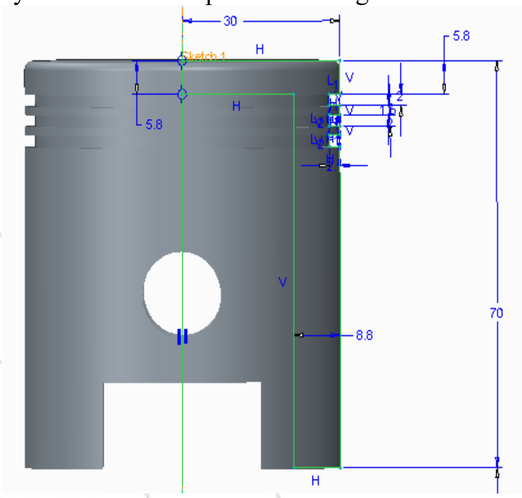

Figure 5: Piston model

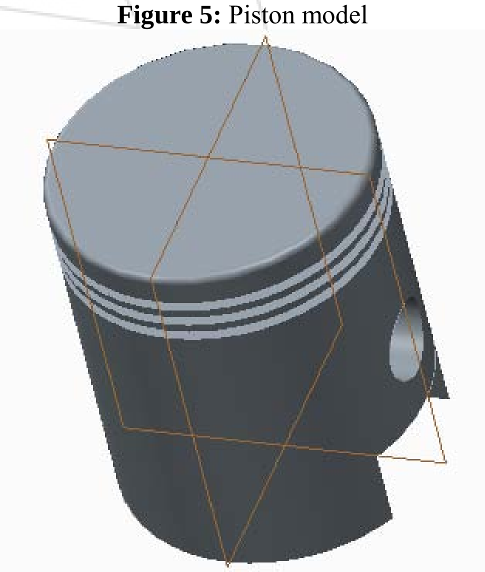

Figure 6: Standard Orientation of the Piston

5. Finite Element Method

The basic idea in the Finite Element Method is to find the solution of complicated problems with relatively easy way. The Finite Element Method has been a powerful tool for the numerical solution of a wide range of engineering problems. Applications range from deformation and stress analysis of automotive, aircraft, building, defense, and missile and bridge structures to the field of analysis of dynamics, stability, fracture mechanics, heat flux, fluid flow, magnetic flux, seepage, and other flow problems. With the advances in computer technology and CAD systems, complex problems can be modeled with relative ease. Several alternate configurations can be tried out on a computer before the first prototype is built. The basics in engineering field are must to idealize the given structure for the required behavior. In the Finite Element Method, the solution region is considered as many small, interconnected sub regions called Finite elements.

➢ **Steps involved in carrying out analysis using Creo-Simulate**

Creo-Simulate is a computer aided engineering tool that allows us to simulate the physical behavior of a part or assembly, to understand and improve mechanical performance of a design. It enables us to analyze and optimize the design for structural, thermal and dynamic requirements. The steps involved in carrying out analysis using Creo-Simulate are given below:-

- **3D part modeling:** Make three dimensional model of piston using Part and Assembly mode of Creo 2.0.
- **Define the FEA model:** At least there are three basic elements to be specified to define a FEA model, i.e., material, loads and constraints.
- **Define material properties, loads and constraints:** Material assign to the piston in the analysis was Steel. The inner uniform pressure on the piston head was taken as 60 bar.
- **Grid Generation:** Mesh generation is called pre-processing for finite element method. Creo-Simulate automatically generates finite element mesh.
- **Run a static analysis:** After analysis was defined completely, it was required to run the analysis.
- **Review the results:** Once the analysis had run successfully, it was important to review the results. The results of the stress analysis of piston are shown in figures.

Material Properties for Aluminum alloy:

Table 2: Material Properties

Density(Kg/m3)	2710
Poisson‟s Ratio	0.3
Young's Modulus (N/mm^2)	6.89E+10
Tensile Ultimate Strength (N/mm^2)	3.10E+08
Tensile Yield Strength (N/mm^2)	2.80E+08
Compressive Yield Strength (N/mm^2)	2.80E+08
Coefficient of thermal expansion (/ ^{0}c)	2.34E-5

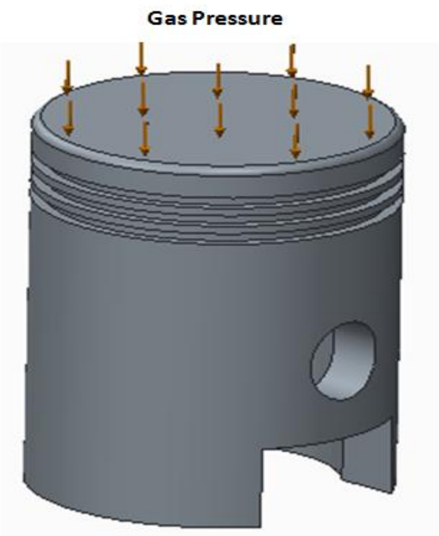

Figure 7: Boundary Condition

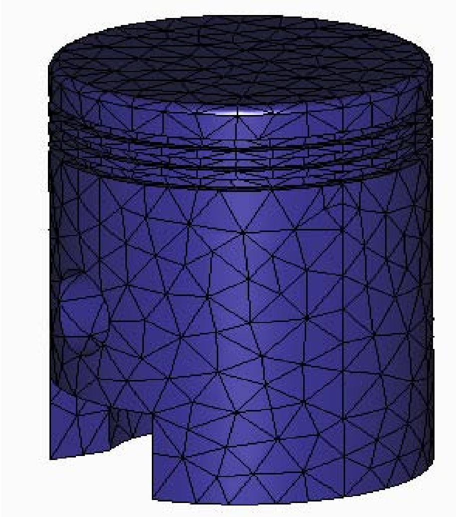

Figure 8: Mesh Model

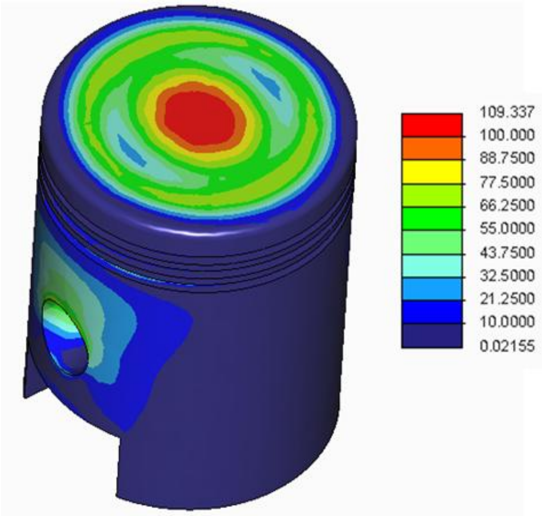

Figure 9: Results of the stress analysis

After reviewing the results, it was found that the stresses developed on piston head was not within the 100 MPa i.e. not in permissible/safe limit.

6. Stress Optimization of the Piston

Optimization is a process of finding an optimal solution satisfying a given number of constraints. An optimization is done in Creo-Simulate with given loading data. The objective of the optimization is to minimize the stress occurred on piston head. The stress distribution on the piston head mainly depends on the deformation of piston head. Therefore, in order to reduce the stress concentration, the piston head should have enough stiffness to reduce the deformation. To study the influence of parameters on piston stress levels, number of iterations are run using optimization tool in Creo-Simulate. 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 minimize the stress obtain on the piston head of non-optimize model 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 . As piston is a critical component we are considering Factor of safety as 2.5. For aluminum alloy, tensile yield strength is 250MPa. And mass of piston is 0.2454Kg.

Working or design stress = 250/2.5 = 100 MPa Based on above analysis the maximum stress induced in the piston is 86.29 MPa, which is less than 100 MPa (allowable stress).

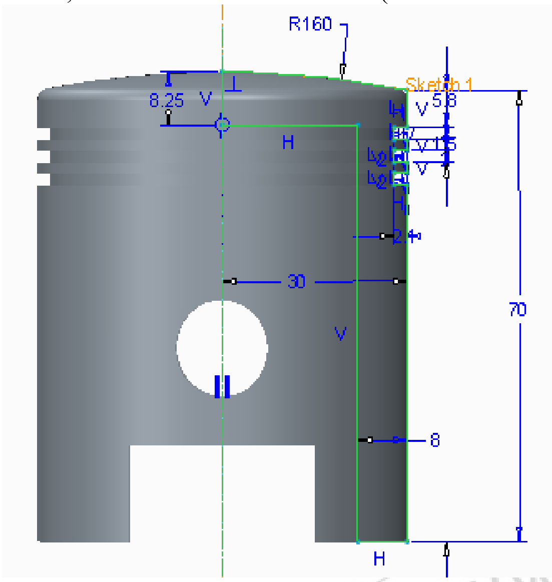

Figure 10: Optimize Piston Model

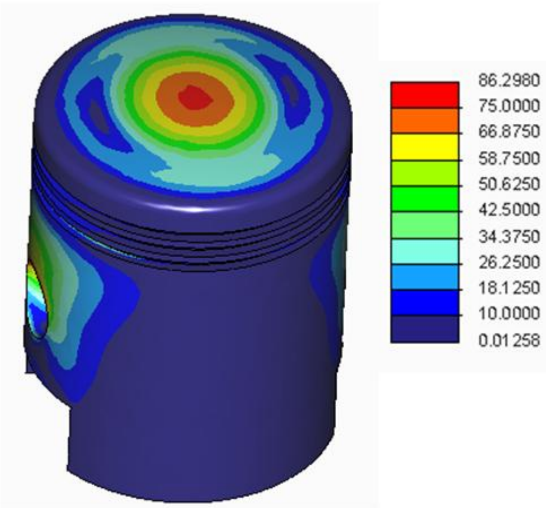

Figure 11: Optimize Result

6.1 Comparison between Original model and Optimize Model

Table 3: Comparison of two model

Parameter	Original model	Optimize model
Maximum Stress occurred	109.33 MPa	86.29 MPa
Mass or weight	0.2574 Kg	0.2454 Kg
Factor of safety	2.28	2.89

Piston Design models are simulated on iteration based and it requires more number of iterations to check whether design is safe or not and to validate the models with the allowable. The optimized model result in Max equivalent stress of 86.29 MPa which is less than allowable stress of 100 MPa & also solid mass is reduced to 0.2454Kg. The result obtained is well below the working stress and mass of piston is also reduced.

7. Conclusion

- The equivalent stress value obtained for the optimized model is well below the permissible value of 100 MPa.
- From optimization results it is clear that there is a scope for reduction in the thickness of piston skirt.
- The optimization of piston is done and it is found that the mass of optimized piston is 0.2454 Kg. Hence percentage reduction in mass compared to non-optimized piston (0.2574 Kg) is 4.66 %.
- 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 piston with reduced material.
- The factor of safety becomes greater than non-optimized model i.e. design become more safe. The factor of safety after optimization is 2.89 obtained.

8. Scope for Future Research

- In this project, only the static FEA of the piston has been performed by the use of the software Creo-Simulate. This work can be extended to study the effect of loads on the piston under dynamic conditions. Experimental stress analysis (ESA) can also be used to calculate the stresses which will provide more reasons to compare the different values obtained.
- Now days a lot of being said about thermal study of the piston model, it has an important role of failure. So the study can be extended to the structural-thermal analysis of the piston.

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