DESIGN AND STRESS ANALYSIS OF A PISTON FOR OPTIMUM PERFORMANCE

SIDDHNAT PAL

DEPARTMENT OF AUTOMOBILE ENGINEERING,

SRM-IST, CHENNAI.

Abstract

This paper is a study about the deign procedure and considerations to be taken during the designing of the piston. Piston plays a main role in energy conversation. The working condition of the piston is very harsh in comparison of other parts of the internal combustion engine. The main objective of this work is to investigate and analyse the stress distribution of piston. This paper will cover the details about the need of changes to be made in the piston to maximize the performance of the piston to gain a better out put in running in real time conditions through CAE (computer aided engineering). Design of the piston is carried out using Solidworks software, static analysis is performed using Finite Element Analysis (FEA). An analysis of stress and damages due to application of pressure is presented and analysed in this work.

KEYWORDS :

Piston, Dead air space, FEA(Finite Element Analysis), IC engines, Otto cycle, Diesel cycle, ALUMINIUM 7075 T6 PLATE

INTRODUCTION

Engine pistons are one of the most complex components among all automotive or other industry field components. The engine can be called the heart of a car and the piston may be considered the most important part of an engine. There are lots of research works proposing, for engine pistons, new geometries, materials and manufacturing techniques, and this evolution has undergone with a continuous improvement over the last decades and required thorough examination of the smallest details. The fatigue related piston damages play a dominant role mainly due to thermal and mechanical fatigue, either at room or at high temperature. This paper describes the displacement and stress distribution on piston of internal combustion engine by using FEA. The FEA is performed by CAD and CAE software. The main objectives are to investigate and analyse the stress distribution of piston at the real engine condition during combustion process. The paper describes the FEA technique to predict the higher stress and critical region on the component. The optimization is carried out to reduce the stress concentration on the piston.

INTERNAL COMBUSTION ENGINES

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Internal combustion engines can be classified as Continuous IC engines and Intermittent IC engines. In continuous IC engines products of combustion of the fuel enters the engine as the working fluid.

There are two major cycles used in internal combustion engines: Otto and Diesel. The Otto cycle is named after Nikolaus Otto (1832 – 1891) who developed a four-stroke engine in 1876. It is also called a spark ignition (SI) engine since a spark is needed to ignite the fuel-air mixture. The Diesel cycle engine is also called a compression ignition (CI) engine since the fuel will auto-ignite when injected into the combustion chamber. The Otto and Diesel cycles operate on either a four- or two-stroke cycle.

1. In line
2. horizontally opposed
3. radial
4. V

Advantages of internal combustion engines:

- Simple mechanical design.
- Power output is higher per unit weight
- Initial cost is Low cost.
- Higher brake thermal efficiency as only a small fraction of heat energy of the fuel is dissipated to cooling system
- Compact and require less space.

Siddhant pal is currently *pursuing bachelor's degree in automobile engineering ,SRM-IST,Chennai, sidpal99@gmail.com*

THEORY

Factors Considered for Proper Functioning of Piston

- The piston should have huge strength and warmth resistance properties to take the force per unit area and inertia forces.
- They must have minimum weight to attenuate the inertia forces.
- The fabric of the piston should have fast dissipation of heat from the crown to the rings and bearing space to the cylinder walls.
- The material of the piston should possess rigid qualities in order that the piston is ready to take care of adequate surface-hardness unto the operative temperatures.
- The piston should have rigid construction to resist the thermal mechanical distortion and adequate space to stop undue wear.

Forces acting on a piston:

- Every piston has a major and minor thrust side. The major side, due to the direction of rotation and the relative angle of the cylinder sleeve, experiences more side loading than the minor side.
- Inertia force caused by the high frequency reciprocating motion of piston Friction between the cylinder walls and the piston rings Forces due to expansion of gases.

Major development in the design of this piston is to calculate the "DEAD AIR SPACE" the piston and to minimize that area to have increment in the fuel efficiency, engine power and reduction of emissions.

Dead air space is the space between the top ring groove and the top land.

MODELLING

Piston Design The piston is designed according to the procedure and specification which are given in machine design books. The dimensions are calculated in 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:

- 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.
- It should have sufficient support for the piston pin.

The modelling of the piston is done on solidworks software with the calculated variables needed for the correct values of the given parameters of the engine:-

Table-1

Parameters	values
Engine Type	250cc, Liquid Cooled
Bore	76mm
Stroke	55mm
No. Of cylinders	1
Compression ratio	10.7:1
Power	26.5BHP
Torque	22.9Nm
Idle speed	1200 RPM
Dry weight	38KG
Fuel	Gasoline

The material used to model the piston is **ALUMINIUM 7075 T6 PLATE(SS)**

PROPERTIES OF THE MATERIAL: -

Table – 2

Material Properties	Aluminum 6061
Elastic Modulus	68.9GPa
Poisson's Ratio	0.33
Elongation	12%
Mass Density	2700Kg/m^3
Tensile Strength	310MPa
Yield Strength	276MPa

A piston subject to high temperatures and high pressures during its reciprocation in a cylinder of an internal combustion engine is required to have a high strength and a high resistance against wear and to be lightweight. As a material for such a piston, an Al (aluminium) alloy containing Si (silicon) is widely used. The main reasons for adding Si here is to

- improve casting property by lowering the melting point and by facilitating the flow of molten metal,
- restrict deformation at high temperatures by lowering the coefficient of thermal expansion, and
- improve resistance against wear and fatigue due to high speed sliding movement.

CALCULATIONS

Thickness of Piston Head (tH)

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

$$tH = D\sqrt{\frac{3}{16}\frac{P}{\sigma t}}$$

P= maximum pressure in N/mm^2

D= cylinder bore/outside diameter of the piston

σt = permissible tensile stress for the material of the piston

Radial Thickness of Ring (t1)

$$t1 = D\sqrt{\frac{3 \times Pw}{\sigma t}}$$

D = cylinder bore in mm

Pw= pressure of fuel on cylinder wall in N/mm².

Its value is limited from 0.025N/mm² to 0.042N/mm².

σt is 90Mpa

- calculated result = 2.59mm

Axial Thickness of Ring (t2)

The thickness of the rings may be taken a

$$t_2 = 0.7t1 \text{ to } t_1$$

- calculated result = 2.2mm

Width of the top land (b1)

The width of the top land varies from

$$b1 = t_H \text{ to } 1.2 \times t_H$$

- calculated results = 13mm (dead air space)

Width of other lands (b2)

Width of other ring lands varies from

- calculated result = 11.8mm

$$t1 = D\sqrt{\frac{3 \times Pw}{\sigma t}}$$

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

- calculated results = 1.65 to 2.2mm
- 2mm

Maximum Thickness of Barrel (t3)

$$t_3 = 0.03 \times D + b + 4.5mm$$

- Calculated result =10.18mm

Skirt thickness(t_4)

$$t_4 = 0.25\times t_3 * 0.35\times t_3$$

Theoretical Stress Calculation:

The piston crown is designed for bending by maximum gas forces Pz_{max} as uniformly loaded round plate freely supported by a cylinder. The stress acting in MPa on piston crown:

$$\sigma b=\frac{Mb}{Wb}=Pzmax(ri/\delta)2$$

Where , M_b = (1/3) Pzmax ri 3 is the bending moment, MN m;

W_b = (1/3) riδ 2 is the moment of resistance to bending of a flat crown,

Pzmax = Pz, is the maximum combustion pressure,

MPa;=5Mpa

This value varies from 2Mpa-5Mpa in case of aluminium alloy.

$$ri = [D / 2 - (s + t1 + dt)]$$

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is the crown inner radius, m.;

Where, Thickness of the sealing part s = 0.05D= 0.05X76=3.8mm.

Radial clearance between piston ring and channel :dt= 0.0008m

Radial thickness of ring (t1) =3mm.

Therefore

$$, ri=[0.07/2-(0.038+0.003+0.0008)]=0.0274\ m$$

Thickness of piston crown

$$\delta=(0.08 \text{ to } 0.1)\times D= 0.085X76=6.64=7mm.$$

$$\sigma b= 5\times[(0.0274/0.007)^2]\ Mpa= 91Mpa$$

Hence required theoretical stress obtained from calculation is 91Mpa

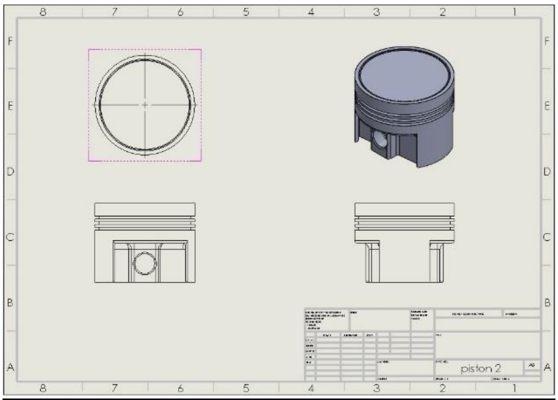

Fig1

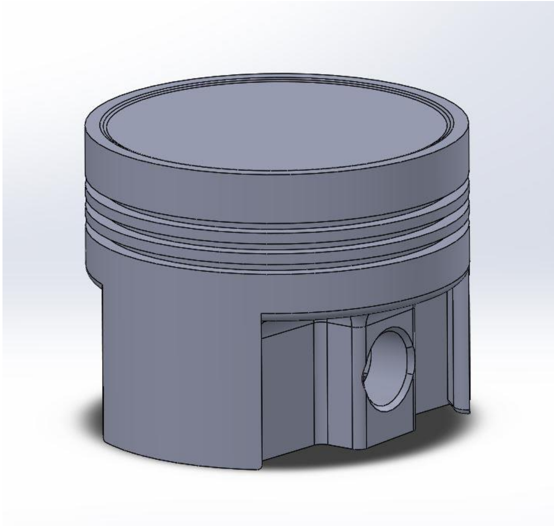

Fig 2

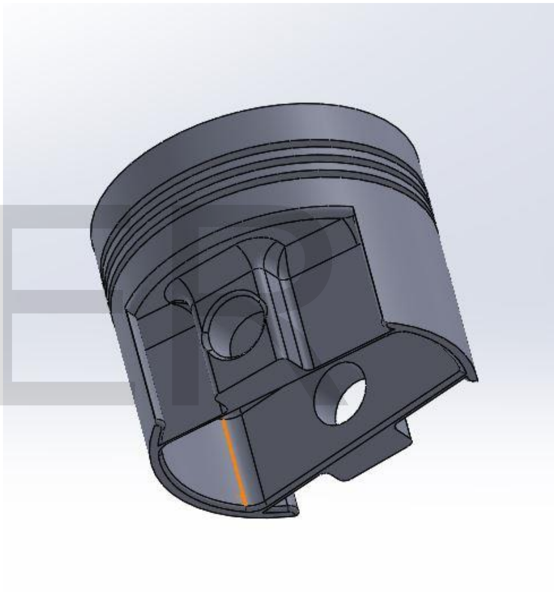

Fig3

ANALYSIS OF THE MODEL:

Here Stress analysis of the piston model has been performed to obtain the value and parameters at which the piston would be damaged. Damages may have different origins: mechanical stresses, wear mechanisms

For this analysis parameters like Pressure, Temperature, Thermal Stress. have been used and to discuss the effects of these parameters on the model are as follows:

PRESSURE:

When air-fuel mixture is ignited, pressure from the combustion gases is applied to the piston head, forcing the piston towards the crankshaft. Due to the pressure at the piston head, there are mainly two critical areas: piston pin holes and localized areas at the piston head. Subsequently will be presented different engine pistons where the cracks initiated on those areas. The pressurized gases travel through the gap between the cylinder wall and the piston. The upward motion of the piston is against the pressure of the gases. The causes a tremendous effect on the piston head leading to its damage and deformation of the piston head.

MESHING THE MODEL:

Mathematically, the structure to be analysed is subdivided into a mesh of finite sized elements of simple shape. Within each element, the variation of displacement is assumed to be determined by simple polynomial shape functions and nodal displacements. Equations for the strains and stresses are developed in terms of the unknown nodal displacements. From this, the equations of equilibrium are assembled in a matrix form which can be easily programmed.

Study name	piston (-Default-)
Mesh type	Solid Mesh
Mesher Used	Standard mesh
Automatic Transition	Off
Include Mesh Auto Loops	Off
Jacobian points	4 points
Element size	3.56855 mm
Tolerance	0.178427 mm
Mesh quality	High
Total nodes	45820
Total elements	29399
Maximum Aspect Ratio	15.958
Percentage of elements with Aspect Ratio < 3	96.2
Percentage of elements with Aspect Ratio > 10	0.0918
% of distorted elements (Jacobian)	0
Time to complete mesh(hh:mm:ss)	00:00:04
Computer name	

Fig4

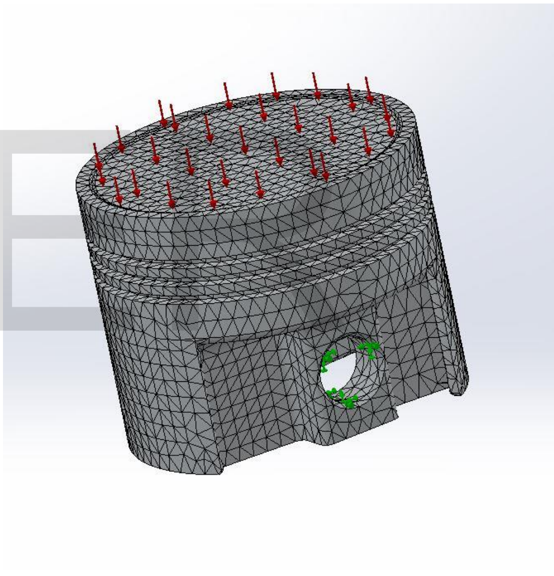

Fig5

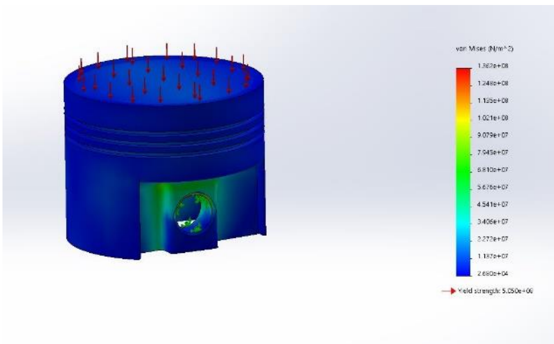

Fig 6 stress plot

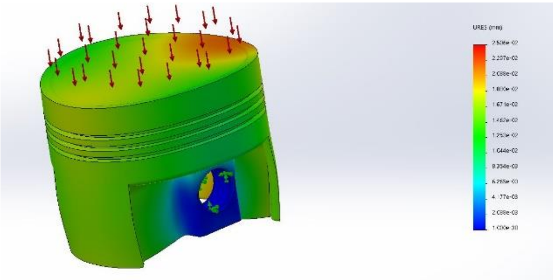

Fig 7 displacement plot

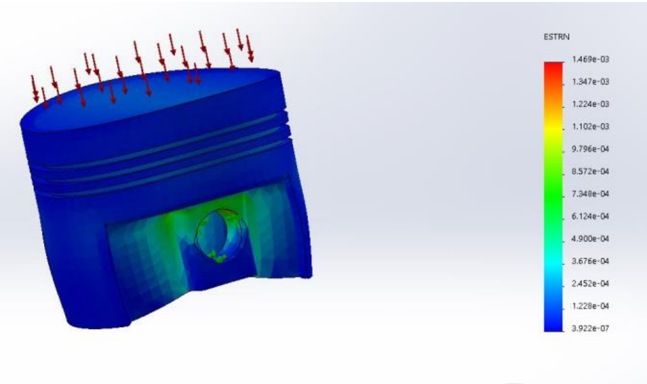

Fig 8 strain plot

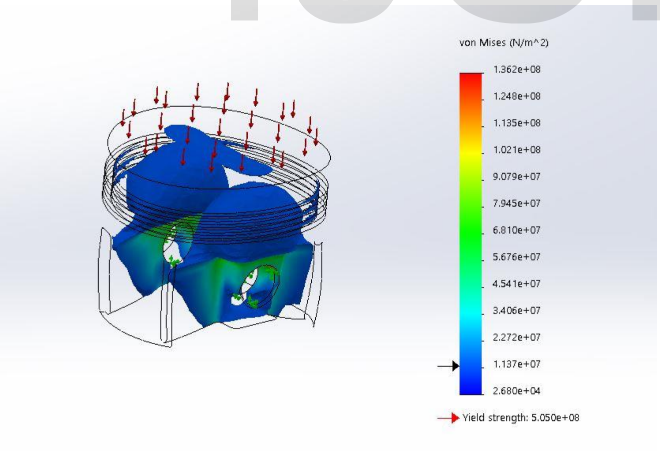

Fig 9 stress iso clipping

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CONCLUSION

- To conclude its safe to say the designed piston is well within the desired parameters and has a reduced dead air space that will help the betterment of the overall performance of the engine output.
- Modelling and analysis of piston is done.
- First Static structural analysis is carried out on piston at 9MPa pressure.
- Maximum stress, deformation, maximum strain and maximum shear stress are noted and tabulated.