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RESPONSE AND SURFACE OPTIMIZATION ANALYSIS OF S.I. ENGINE THROUGH ANSIS

GAJRAJ SINGH LODHI Master of Technology Student, Department of Mechanical Engineering Oriental Institute of science & Technology, Bhopal, MP-462022, India
PANKAJ KUMAR PANDEY Assistant Professor, Department of Mechanical Engineering Oriental Institute of science & Technology, Bhopal, MP-462022, India
ANKIT KUMAR PANDEY Assistant Professor Department of Mechanical Engineering Oriental Institute of science & Technology, Bhopal, MP-462022, India

Abstract —
In this research article, analyze, and optimize a piston for Scooty Pep engine specifications through ANSYS. The ANSYS Geometric module was used to produce solid models of the piston based on calculated dimensions. ANSYS software was used to perform a thermo-mechanical analysis on the piston to analyze induced stresses, total deformation, strain, and factor of safety distribution in various areas of the piston for analyzing the effect of gas pressure and heat fluctuations. The parts are optimized for strength as well as weight using the Ansys software's Response Surface Optimization module. Optimization result the thickness of the crown head increased by 9.41%, the width of the top land increased by 3.81%, the thickness of the piston barrel decreased by 52.28%, the radial thickness of the ring decreased by 5.31%, and the axial thickness of the ring increased by 2.38%, the resultant mass of the piston decreased by 26.07%, and its factor of safety increased by 3.072%.

Keywords-
Response of SI engine, optimization through ANSYS.

I. INTRODUCTION
The piston's responsibilities are to obtain pressure from the expanding gas and to deliver energy to the crankshaft via the connecting rod. The piston must also carry a significant quantity of heat from the combustion chamber to the cylinder walls as a result of fuel combustion [1].
The piston's primary function remains intact. So, what has altered? - The operational setting. Engines today are quicker, work harder, and run more active than ever before. They are anticipated to run longer and with less maintenance in one go. Examples may be found in the following papers: geometry and combustion flow [2,3]; materials, lubrication, and coating [4-9]; analytical tools- FEA [3-7]; treatment technologies [4], and so on.
Despite this industrial examination, there are still a significant percentage of piston failures. Failures can occur due to mechanical stress, thermal stress, wear and tear processes, and so on. In this study, a piston thermomechanical analysis is perform. Xiqun Lu et al, The reverse heat transfer approach is used to perform thermal numerical analysis on a 4ring articulated S.I. Engine piston of a marine diesel engine and compute the coefficient of heat transfer at each contact in the thermal system. They created metal plugs that were put in the head of an articulated piston and the piston skirt to evaluate temperature distribution. For cylinder temperature measurement, a series of thermal couplings were connected. The numerical simulation boundary condition is validated using experiment results and utilized to estimate the temperature distribution of a new piston design with a minor modification in piston head profile and one fewer ring scheme.[6] Ekrem Buyukkaya et al, Thermal analysis was performed on a standard uncoated diesel piston comprised of aluminum silicon alloy and steel. ANSYS software is then used to conduct a thermal analysis of pistons covered with MgO-ZrO2 material. The maximum temperature of the coated piston was detected around the combustion bowl lip of the piston, according to the data. As a result, oversensitivity must be applied to this location. The maximum surface temperature of the coated piston surface with a low thermal conductivity material rises by 48% for AlSi alloy and 35%

for steel. The highest surface temperature of the coating piston's base metal is 261 °C for AlSi and 326 °C for steel, and the ceramic coating is utilized to test the materials' strength and deformation. [7-9].

Experimental section:

1. MESHING, BOUNDARY CONDITIONS: The pressure operating on the whole piston head surface and providing support on the piston hole were characterized as the boundary conditions. To perform coupled thermo-mechanical calculations, it is important to load data on the material that refers to both its thermal and mechanical properties. Temperature loads are supplied to various places, and pressure is delivered to the piston head. Heat (255°C-180°C) is applied to areas such as the piston ring regions and the piston head. The piston wall convection values range from 350 W/mK to 600 W/mK. 2 MPa is the operating pressure.

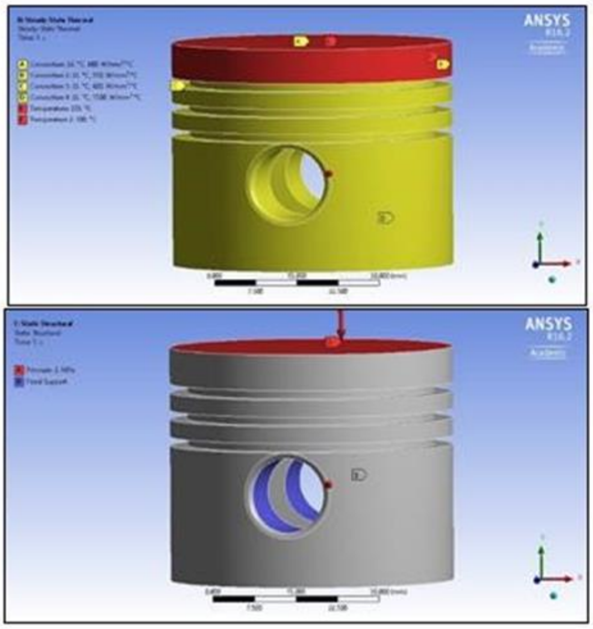

Figure 7, Steady state thermal boundary conditions for piston

$TH = \sqrt{(3 \times P \times D^2)/(16\sigma_t)}$ in mm

TH = 5.6973 mm

- Where, P is the maximum pressure acting on the piston head in N/mm2, P = 6 N/mm2,
- D is the cylinder bore/outside diameter of the piston in mm, D = 50.958 mm,
- and the material is a specific grade of AL-Si alloy with a yield tensile strength of 285 Mpa and an F.O.S. of 2.25.
- Tensile stress permissible for the material in N/mm2, t = 125 in N/mm2.

Based on the second heat transfer consideration, the thickness of the piston head should be such that the heat received by the piston as a result of fuel combustion is swiftly transmitted to the piston and cylinder walls. The piston head's thickness is calculated by treating it as a flat circular crosssectional plate.

$TH = H/(12.526K\,(T_c - T_e))$

TH = 5.195 mm

Where

- Heat flowing through the circular cross-sectional head of the piston in kJ/s or KW,

H = C×HCV×m×B.P. = 2.4295 KW

- K = 175 W/m/C for heat conductivity in W/m/°C
- The temperature difference (TC - TE) for aluminum alloy is 75°C.
- C = 0.05 Coefficient of the fraction of heat delivered to the engine that is absorbed by the piston head

- Greater calorific value of the fuel in kJ/kg, HCV = 47 103 kJ/kg for gasoline

Fuel mass in kg per braking power per second, m= 0.15 KJ/Break/Hr

Break Power, B.P. = 2πNT/60 in KW

= 2.4295 KW

B. Radial thickness of ring

The radial thickness (t1) of the piston ring is determined by considering the bending stress distribution in the ring while assuming the radial pressure between the cylinder wall and the ring.

- Cylinder bore in mm, D = 51 mm gives the radial thickness
- Fuel pressure applied to cylinder wall in N/mm2,
  Pw = 0.025 to 0.042 N/mm2.
- For the current material, t1 = 110 Mpa.

C. Axial thickness of ring

The thickness of the piston rings used as t2 = 0.7t1 to t1, Let t2 = 1.2073 mm

D. Minimum axial thickness

t2 = D/ (10×nr) = 1.6986 mm

Where, Number of rings but the reference number,

nr = 3

E. Width of the top land

The width of the top land varies from,

b1 = tH to 1.2 tH B1 = 6.234 mm

F. Width of other land

The width of other rings lands varies from

b2 = 0.75t2 to t2 b2 = 3 mm

G. Maximum barrel thickness

t3 = 0.03D + b + 4.5 mm = 5.1534 mm

Where, b = Radial depth of four stroke S.I. Engine piston ring groove

Table 3, Designed dimensions of TVS scooty pep+ engine piston for Aluminium Alloy

FINITE ELEMENT ANALYSIS A.: Von-Mises stress distribution in an SI engine piston before optimization due to thermomechanical loading

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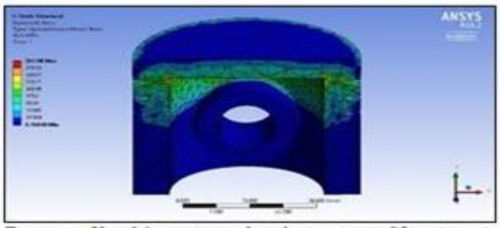

Figure 1 Von-Mises stress distribution in an SI engine piston before optimization due to thermomechanical loading

The figure illustrates the safety factor distribution in the S.I. engine piston under the given loading circumstances, with a maximum value of 15 and a minimum value of 1.0448.

OPTIMIZATION: The objective of the parametric optimization work was to reduce the mass of the piston under the impact of compressive gas load and temperature applied in piston components such that the maximum and equivalent stress values are within the

A. After optimization reduction in Geometric mass of piston

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

B. Comparison of Deformation distribution in S.I. piston after optimization due to thermo- mechanical loading

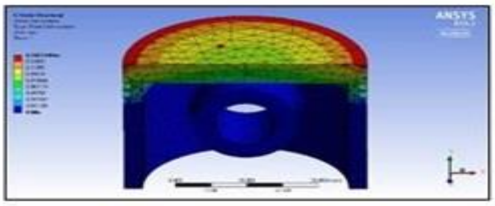

Figure 2. Deformation distribution in S.I. piston after optimization due to thermo-mechanical loading

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

C. Distribution of safety factors in an S.I. piston body after optimization due to thermomechanical stress

D. Distribution of safety factors in an S.I. piston body after optimization due to thermomechanical stress

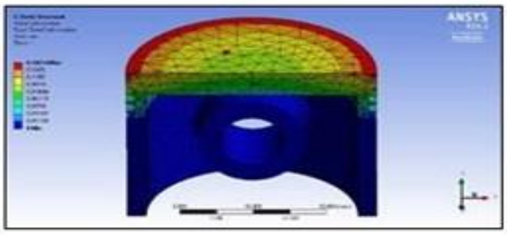

Figure 3 . Distribution of safety factors in S.I. pistons after optimization due to thermomechanical loading

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

CONCLUSION:

The Piston has been optimized using the Response Surface Optimization module. The following results have obtained based on this research: Piston Design

1. The percentage of weight reduced after parametric optimization is 26.074%. The percentage of rise in factor of safety and percentage of decrease in equivalent von-misses stress is changed by 3.072% and 2.982%, respectively.
2. The weight, factor of safety, and equivalent vonmisses stress after optimization are 0.10065 Kg, 1.0769, and 259.99 MPa, respectively. Because the safety factor is larger than one and the maximum equivalent von-misses stress is less than the yield stress of the aluminium alloy, the specified piston model is safe.

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REFERENCES

1) C M Taylo, ―Automobile engine tribology‖ – design considerations for efficiency and durability, Elsevier science publishers, pp 1– 8, 1998
2) H Kajiwara, Y Fujioka, T Suzuki, H Negishi. ―An analytical approach for prediction of piston temperature distribution in diesel Engines‖, JSAE Rev, vol. 23, issue 4, pp 429–434, 2002
3) F Payri, J Benajes, X Margot, A Gil. ―CFD modeling of the in-cylinder flow in directinjection diesel engines‖, Comput Fluids, vol. 33, issue 8, pp 995–1021, 2004
4) C Friedrich, G Berg, E Broszeit, F Rick, J Holland, ―PVD CrxN coatings for tribological application on piston rings‖,
Surf Coat Technol, issue 97 (1–3), pp 661– 708, 1997
5) C E Pinedo, ―The use of selective plasma nitriding on piston rings for performance improvement‖, Mater Des, vol. 24, issue 2, pp 131–5, 2003
6) T Kikuchi, S Ito, Y Nakayama, ―Piston friction analysis using a direct-injection singlecylinder gasoline engine‖ JSAE Rev, vol. 24 issue 1, pp 53–8, 2003
7) T A Stolarski, Q Zhou, ―Temperature– friction characteristics of used lubricant from twostroke cross-head marine diesel engines‖, vol. 252, pp 300–5, 2002
8) M Priest, C M Taylor. ―Automobile engine tribology‖ — approaching the surface, vol. 241 issue 2, pp 193–203, 2000
9) M Takiguchi, H Ando, T Takimoto, A Uratsuka, ―Characteristics of friction and lubrication of two-ring piston‖, JSAE Rev, vol. 17 issue 1, pp 11–6, 1996 10) J R Cho, Y S Joo, H S Jeong, ―The Alpowder forging process‖ - its finite element analysis, J Mater Processing Technol; vol. 111, pp 204–209, 2001