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Thermal Stress Analysis and Performance Optimisation of Marine Diesel Engine Piston

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ABSTRACT
This research carries out thermal stress analysis and performance optimisation of Marine Diesel Engine (MDE) piston. Improper piston design, apart from causing poor combustion, could lead to severe thermal stress, wear of moving components, crack of skirt, etc. To mitigate the thermal-stress induced failure of MDEpiston, ANSYS Mechanical Modeller is implemented to produce an optimal design by combining the best thermal and geometric propertiesfor the operating conditions. The effects of thermal stress distribution and concentration within the operating range are characterisedfor different pistonthicknesses.The results reveal that the investigated MDE piston experiences a maximum equivalent stress of 6408.6MPa at the piston crown and a minimum equivalent stress of 270.05MPa at the skirt. The piston's radial thickness is most affectedbecause of the very rapid temperature fluctuations and collision against the liner. Thus, its optimal thickness is 4.0 mm rather than 5.24 mm. Also, the largest thermal stress concentration occurs on the piston crown. The maximum stress reduces from 85 MPa to 55 MPa, and the biggest deformation drops from 0.051762 mm to 0.025884 mm. To prevent adverse pistondeformation, crack, and wear: piston materials should have high stiffness and low expansion. Therefore, proper material selection and geometry are essentials for optimal MDE piston design.

Keywords: Thermal Stress, Analysis, Piston, Optimization, Efficiency, ANSYS

1. INTRODUCTION

The efficiency of a diesel engine is proven to be higher than that of any other power plants of the same size. However, the piston as shown in figure 1 is one of the basic complex components of internal combustion engines that experiences high level of damage in operation. It is found that the ratio of the piston height to the diameter is found to be reducing due to the improvements in modern design. The progressive reductions in piston height and circumferential extent of its skirt imply corresponding reduction in the piston mass, and a consequent decrease in the inertial forces. This also results in lesser friction losses and fuel consumption (Matteo, et al., 2014). The improvement in design and the material properties of piston show that, the pistons are stronger, more durable and lighter than previous engine models (Matteo, et al., 2014).

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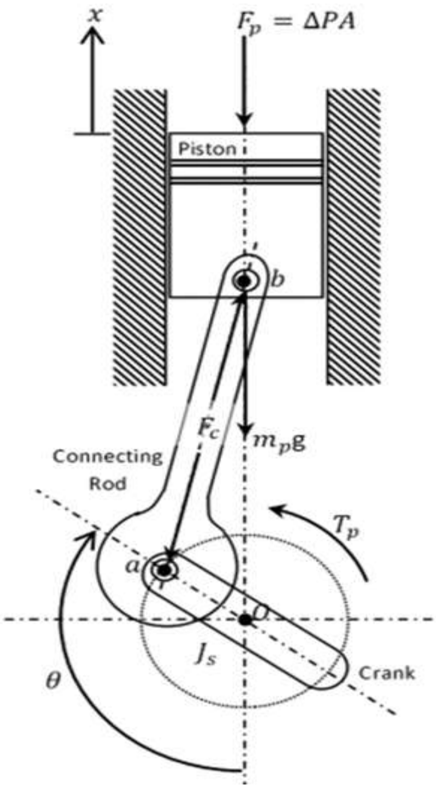

Fig. 1: Simplified internal components of MDE (Murphy, et al., 2015)

Often, in the design of mechanical components, the major focus has always been on the mechanical stress analysis of parts as it is very much crucial in the determination of the geometry of such parts. Lesser attention is given to the aspect of thermal stress analysis (Calbureanu, et al., 2013). This is presumably due to its lesser effects in the design of components of an MDE. However, the drastic reduction in the size of modern MDE has mandated a more thorough investigation into the stress caused by thermal transfer especially to the piston.

This work intends to analyse the energy absorbed by the piston and how it can affect the overall performance of the power plants.It investigates the various aspects of optimisations and adopts a more modern method of design for efficient performance of the components of interest. The effects of thermal stress on the performance of the MDE piston are evaluated with a view to optimising the thermal properties and geometric parameters of the piston.

II. ANALYTICAL MODEL

In this research, FEA of heat transfer and residual stress in the marine diesel piston will be carried out and compared to experimental data generated during the current research. There are many FEM software programs available to proffer various engineering solutions. However, the ANSYS finite element program will be used in the current research. This is because it is widely available within the university. Hence the aforementioned data are specific to the ANSYS program. 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 design, 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.

III. METHODOLOGY

In this research, finite element analysis (FEA) of heat transfer and residual stress in the marine diesel piston will be carried out and compared with experimental data generated during the current research. There are many finite element method software programs available to proffer various engineering solutions. However, the ANSYS finite element program will be used in the current research. This is because it is widely available within the university. Hence the aforementioned data are specific to the ANSYS program. The design of the piston starts with the definition of the piston geometry using 3D computer aided design software.

Analysing the piston using this software will indicates how the piston would behave in a real engine. It also enables the Engineer to see where the stresses and temperatures would be the greatest and how the piston would respond. The analysis of the piston with ANSYS is done to optimise the stresses and minimize the weight. The Mathematical model of optimisation is formulated first, and then FEA results on ANSYS software are correlated for validation. Based on the analysis of optimal result, the stress concentration on the piston would provide a better reference for redesigning of the piston.

A. Piston Design Parameters:

The parameters for a piston design consist of the following:

(i) Thickness of piston head (t_H)
(ii) Heat flows through the piston head (H)
(iii) Radial thickness of the ring (t_1)
(iv) Axial thickness of the ring (t_2)
(v) Width of the top land (b_1)
(vi) Width of other ring lands (b_2)

(i) Thickness of Piston Head (t_H):

Thickness of the piston head is calculated using Grashoff's formula:

$$t_H = D\sqrt{\left(\frac{3}{16}\right)\cdot\left(\frac{P}{\sigma_t}\right)} \quad (1)$$

Where:

P = maximum pressure in N/mm²

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

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

Here the material is a particular grade of AL-Si alloy which permissible stress is 50 MPa- 90Mpa. Before calculating thickness of piston head, the diameter of the piston has to be specified. The given piston length is 152mm and the piston diameter is 140mm.

(ii) Heat Flow through the Piston Head (H):

The heat flow through the piston head is calculated using the formula

$$H = 12.56 \times t_H \times K \times (T_C - T_e) \quad (2)$$

Where:

K = Thermal conductivity of material which is 174.15W/mk

Tc = temperature at centre of piston head in °C.

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

(iii) Radial Thickness of Ring (t_1)

$$t_1 = D\sqrt{\frac{3 \times P_w}{\sigma_t}} \quad (3)$$

Where:

D = cylinder bore in mm

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

(iv) Axial Thickness of Ring (t_2)

The thickness of the rings may be taken as $t_2 = 0.7t_1$ to t_1. Let us assume t_2 = 5mm

Then the minimum axial thickness (t_2)

$$t_2 = \frac{D}{10 \times n_r} \quad (4)$$

Where: n_r = number of rings

(v) Width of the top land (b_1)

The width of the top land varies from, $b_1 = t_H$ to $1.2 \times t_H$ (5)

(vi) Width of other lands (b_2)

Width of other ring lands varies from, $b_2 = 0.75 \times t_2$ to t_2 (6)

(vii) Maximum Thickness of Barrel (t_3)

$$t_3 = 0.03 \times D + b + 4.5\text{mm} \quad (7)$$

Where: b = radial depth of piston ring groove

$$\text{Also, } b = t_1 + 0.4 \quad (8)$$

From the above expressions, tabulated parameters are calculated below: Source: Manufacturer's manual of the MDE onboard MT Otuoke at NPA Port, Rivers State

Table 1: Design Specification before optimisation

S/No	Design1 – Dimensions	Size(mm)
1	Length of the Piston(L)	152
2	Cylinder bore/outside diameter of the piston(D)	140
3	Radial thickness of the ring (t_1)	5.24
4	Axial thickness of the ring (t_2)	5
5	Maximum thickness of barrel (t_3)	14.34

International Journal of Advances in Engineering and Management (IJAEM)
Volume 3, Issue 7 July 2021, pp: 3585-3596 www.ijaem.net ISSN: 2395-5252

6	Width of the top ring land (b_1)	10.84
7	Width of other ring lands (b_2)	4

B. Thermal and Geometric Properties of the Piston Material

It is important to calculate the piston temperature distribution in order to control the thermal stresses and deformations within acceptable levels. 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 (IC) engine pistons are made of an aluminium alloy which has a thermal expansion coefficient of 80% higher than the cylinder bore material made of cast iron**.** The thermal and geometric properties are as shown in Table 3.2: Source: Manufacturer's manual of the MDE onboard MT Otuoke in NPA, Rivers Port

Table 2: Thermal and Geometric Properties of Diesel Engine Piston

Property	**Aluminium Alloy**	**Zirconium**
Young's Modulus	70 x 10^3MPa	220 x 10^3MPa
Poisson Ratio	0.31	0.35
Thermal Conductivity	234W/mK	7W/mK
Co-efficient of Thermal Expansion	23 x 10^{-6}/K	10 x 10^{-6}/K

Thus, the dimensions for the piston are calculated and these are used for modelling the piston in ANSYS software. In the above procedure, the ribs in the piston are not taken into consideration, so as to make the piston model simple in design. In modelling a piston, considering all factors will become a tedious process. Therefore, a symmetric model is developed using the above dimensions. The piston is modelled in ANSYS as shown in Figures 2 and 3.

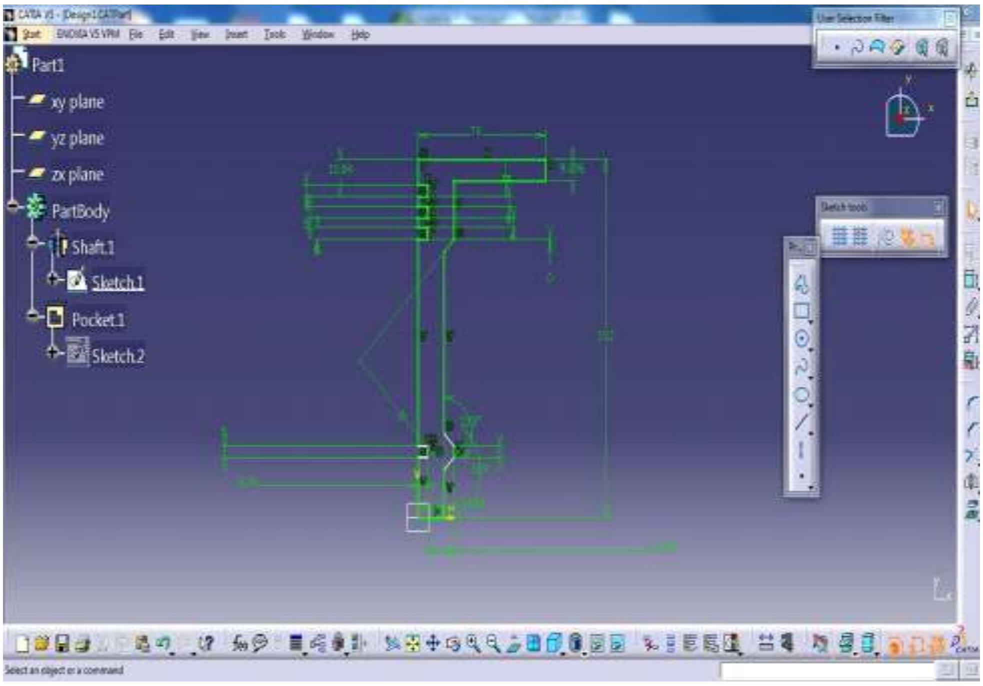

Fig. 2: Sketch of the Piston before optimization

International Journal of Advances in Engineering and Management (IJAEM)
Volume 3, Issue 7 July 2021, pp: 3585-3596 www.ijaem.net ISSN: 2395-5252

Fig. 3: Piston Model

IV. RESULT AND ANALYSIS

The current model is undergone through Thermal Analysis and followed by Static Analysis, together known as Coupled Field Analysis. The meshed component is analysed to find the thermal stresses of the piston. The component is subjected to the influence of heat conduction at the top of the piston and heat convection to side lands. The following images are shown for result deformation and Von-Misses stresses before and after optimisation.

A. Total Deformation

The MDE piston experiences a maximum total deformation of 2.4187mm at the piston crown and a minimum total deformation of 0.0094155mm at the skirt. This is as a consequence of the influence of heat conduction at the top of the piston and heat convection to side lands as revealed in Fig. 4. The piston head is the top surface of the piston which is subjected to pressure fluctuation, thermal stresses and mechanical load during normal engine operation. By the forces of combustion, piston reciprocates inside the cylinder.

International Journal of Advances in Engineering and Management (IJAEM)
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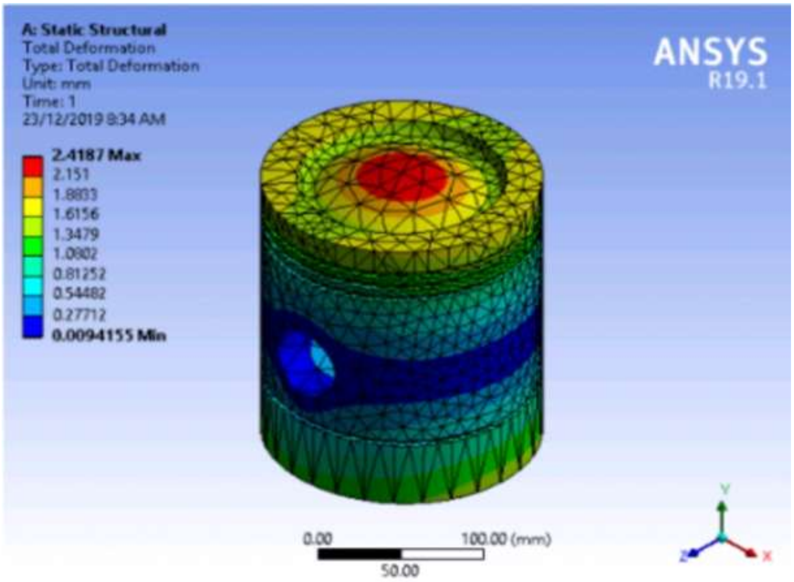

Fig. 4: Total deformation of the MDE piston

B. Directional Deformation

The MDE piston experiences a maximum directional deformation of 0.39843mm and a minimum directional deformation of -0.39943mm. This is due to the influence of heat conduction at the top of the piston and heat convection to side lands as shown in Figure 5.Thermal deformations under the operating bowl rim temperature are constrained by the surrounding material. This causes large compressive stresses on the total bowl rim circumference that often exceeds the yield strength of the material.

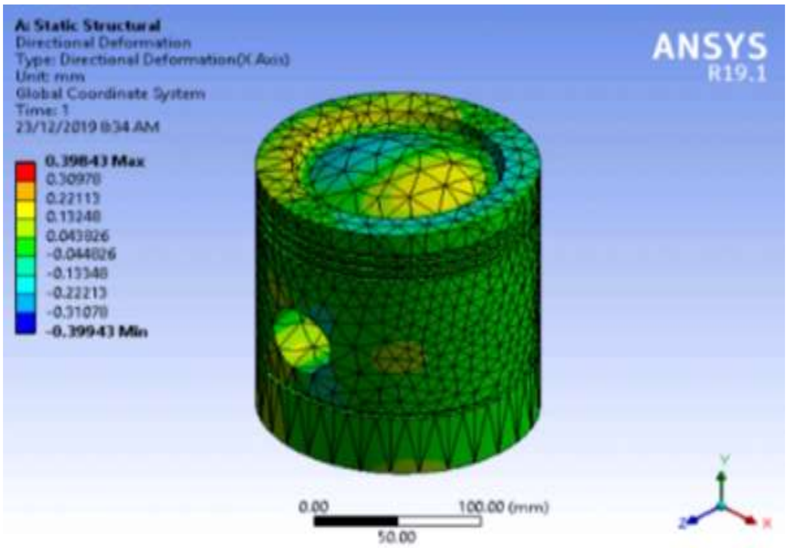

Fig. 5: Directional Deformation of the MDE piston

C. Total Heat Flux

The MDE piston experiences a maximum total heat flux of 23.941W/mm^2 at the piston crown and a minimum total heat flux of 0.015389W/mm^2 at the skirt. This, in turn, is caused by the effect of heat conduction at the top of the piston and heat convection to side lands as indicated in Fig. 6

International Journal of Advances in Engineering and Management (IJAEM)
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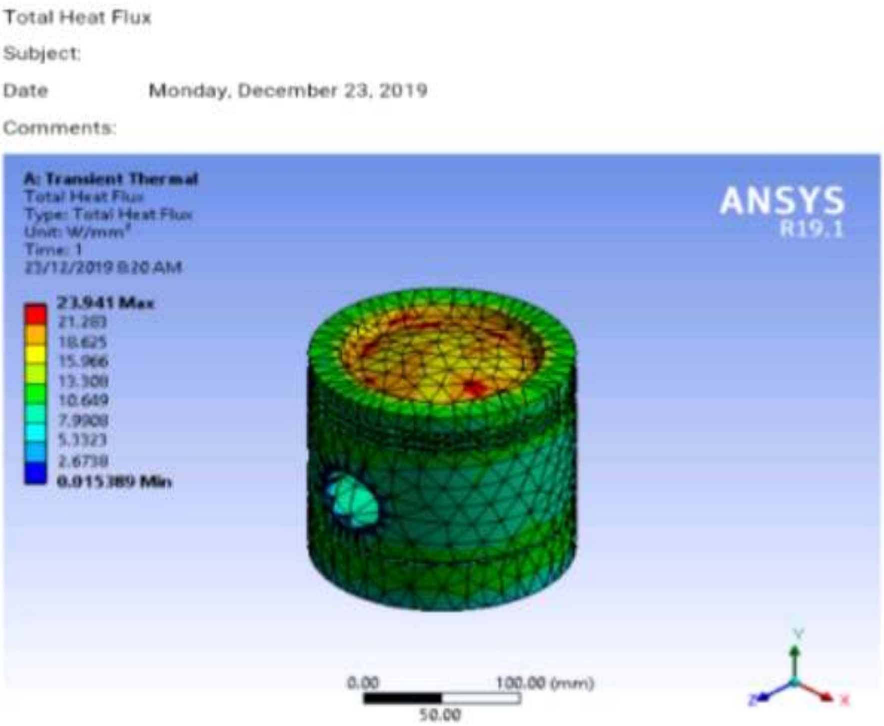

Fig.6: Total heat flux of the MDE piston

D. Directional Heat Flux

The MDE piston experiences a maximum directional heat flux of 14.277W/mm^2 at the piston crown and a minimum directional heat flux of -14.346W/mm^2 at the skirt. This is as a result of heat conduction at the top of the piston and heat convection to side lands as revealed in Fig. 7

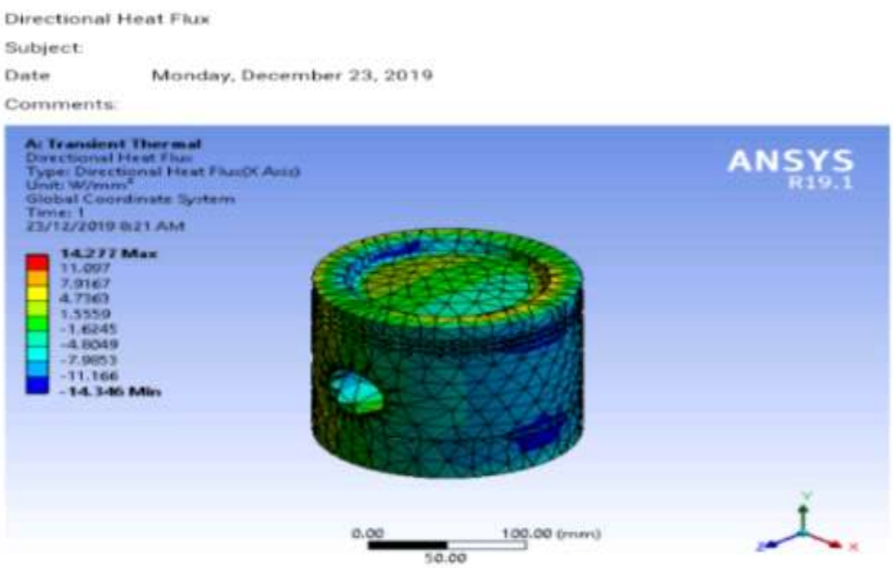

Fig. 7: Directional Heat flux of the MDE piston

E. Temperature

The MDE piston experiences a maximum temperature of 2179.6°C at the piston crown and a minimum temperature of 80.89°C at the skirt. This, in turn, is subjected to the influence of heat conduction at the top of the piston and heat convection to side lands as revealed in Fig. 8.

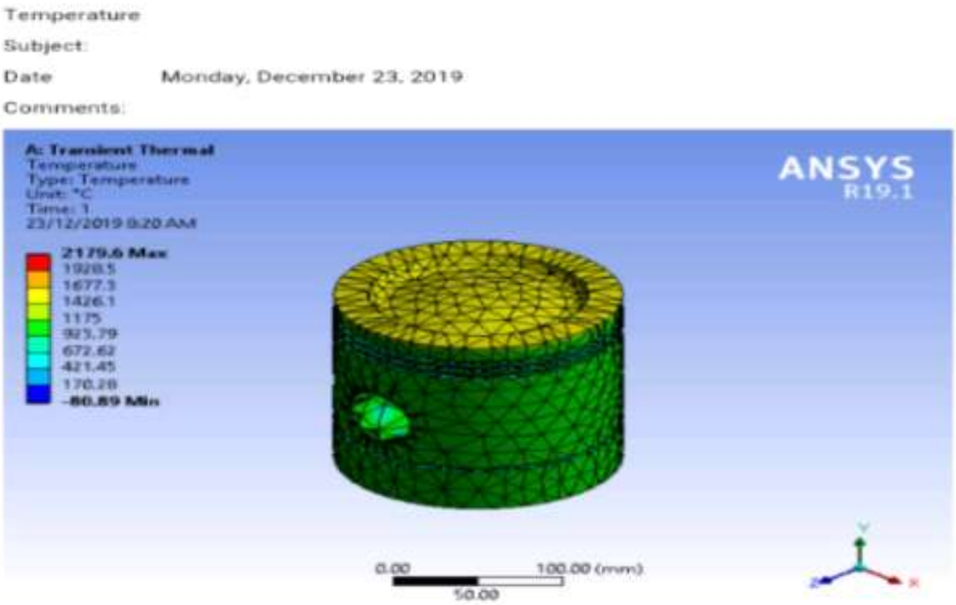

Fig. 8 Temperature of the MDE piston

F. Equivalent Elastic Strain

The MDE piston experiences a maximum equivalent elastic strain of 0.03259mm/mm at the piston crown and a minimum equivalent elastic strain of 0.0016228mm/mm at the skirt. This, in turn, is subjected to the influence of heat conduction at the top of the piston and heat convection to side lands as revealed in Fig.9

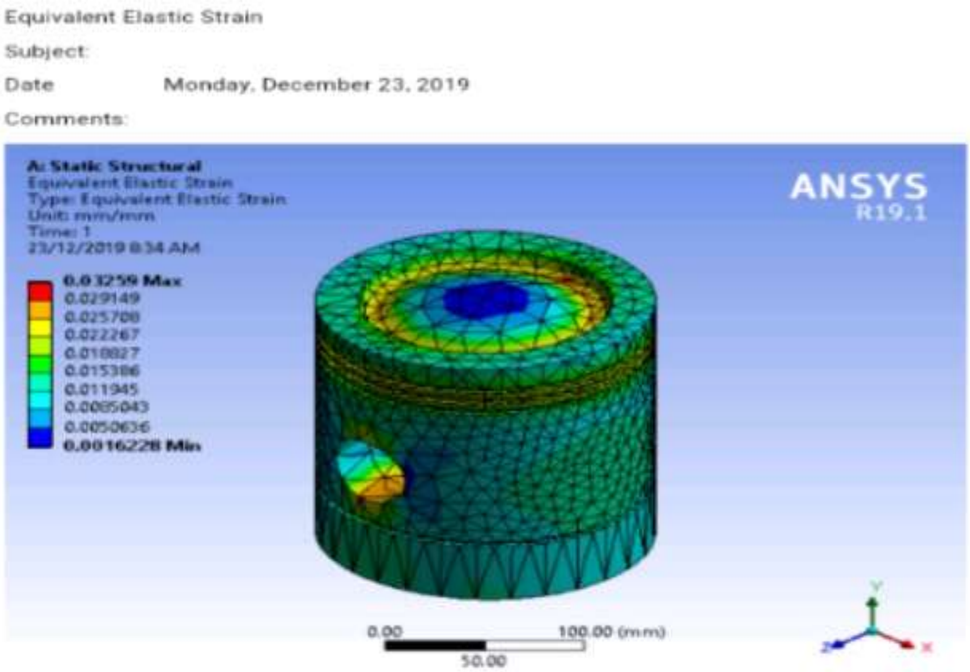

Fig. 9 Equivalent Elastic Strain of the MDE piston

International Journal of Advances in Engineering and Management (IJAEM)
Volume 3, Issue 7 July 2021, pp: 3585-3596 www.ijaem.net ISSN: 2395-5252

G. Principal Elastic Strain

The MDE piston experiences a maximum principal elastic strain of 0.021177mm/mm at the piston crown and a minimum principal elastic strain of 0.0025653mm/mm at the skirt. This, in turn, is subjected to the influence of heat conduction at the top of the piston and heat convection to side lands as revealed in Fig. 10.

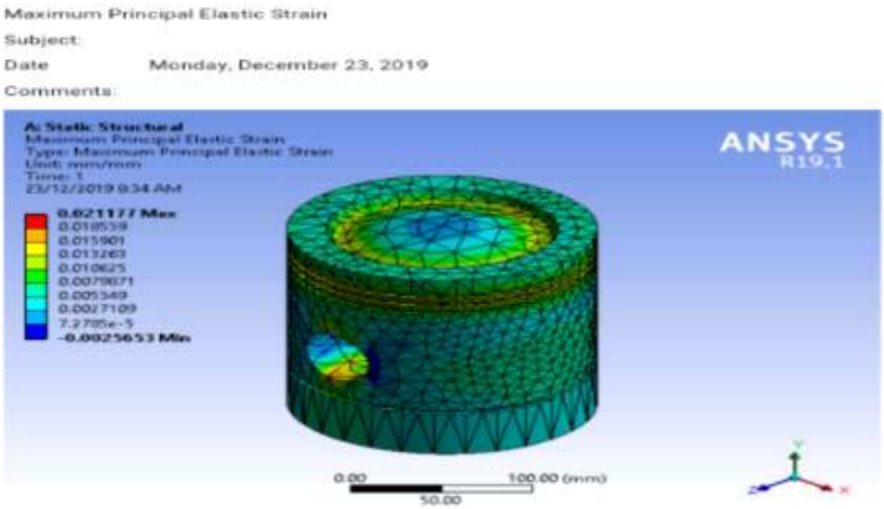

Fig. 10 Principal Elastic strain of the MDE piston

H. Equivalent Stress

The MDE piston experiences a maximum equivalent stress of 6408.6MPa at the piston crown and a minimum equivalent stress of 270.05MPa at the skirt. This, in turn, is subjected to the influence of heat conduction at the top of the piston and heat convection to side lands as revealed in Fig. 11.

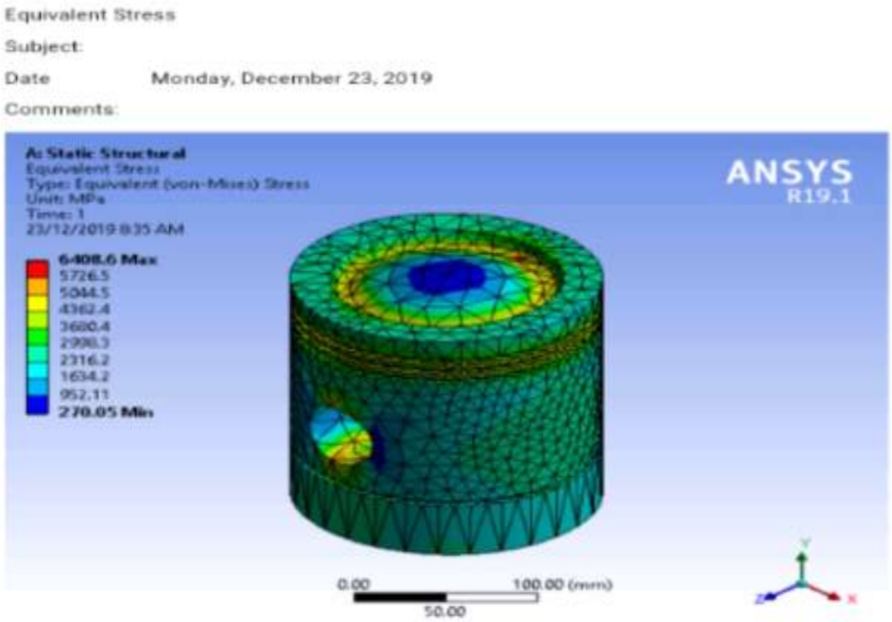

Fig. 11 Equivalent Stress on the MDE piston

International Journal of Advances in Engineering and Management (IJAEM)
Volume 3, Issue 7 July 2021, pp: 3585-3596 www.ijaem.net ISSN: 2395-5252

I. Optimisation Of Piston

After generating a finite element model using ANSYS software, a strategy for the optimisation workflow is defined. Target of the optimisation is to reach a mass reduction of the piston**.** Therefore, the objective function is to minimize mass, subject to the following constraints:

(i) Allowable or design stress should not exceed the maximum Von-Misses stress
(ii) Manufacturing constraints should not be violated
(iii) Detailed stress-loading analysis on the piston should be carried out to ensure that areas where excess materials are removed do not exceed the allowable maximum von-Misses stress with factor of safety kept at 1.5

The following are considerations for material removal from the piston:

- Allowable radial thickness of the ring
- Allowable axial thickness of the ring
- Maximum thickness of the barrel
- Design width of the top land
- Design width of other ring lands

The optimised values of piston design parameters after optimisation using ANSYS are given in Table 3. Source: Manufacturer's manual of the MDE onboard MT Otuoke in NPA, Rivers Port

Table 3: Piston Design Parameters after Optimisation

S/No	Piston Design Parameters (mm)	Before Optimisation	After Optimisation	Design Consideration
1	Radial Thickness of the ring (t_1)	5.24	3.46	4.0
2	Axial thickness of the ring (t_2)	5.0	3.52	4.0
3	Maximum thickness of the barrel (t_3)	14.34	9.08	9.0
4	Width of the top land (b_1)	10.84	9.36	10.0
5	Width of the ring land (b_2)	4.0	3.24	3.0
6	Von-Misses Stress [MPa]	85.6439	55.6685	69.0
7	Deflection	0.051762	0.025884	0.05884

The initial value of the width of the top land (b_1) i.e., before optimisation is 10.84mm. It is changed after applying pressure directly on the head i.e., the piston crown which shape is like a bowl.The value after optimisation is obtained as 9.36mm and it is rounded to10mm. This value is considered for the optimal design.

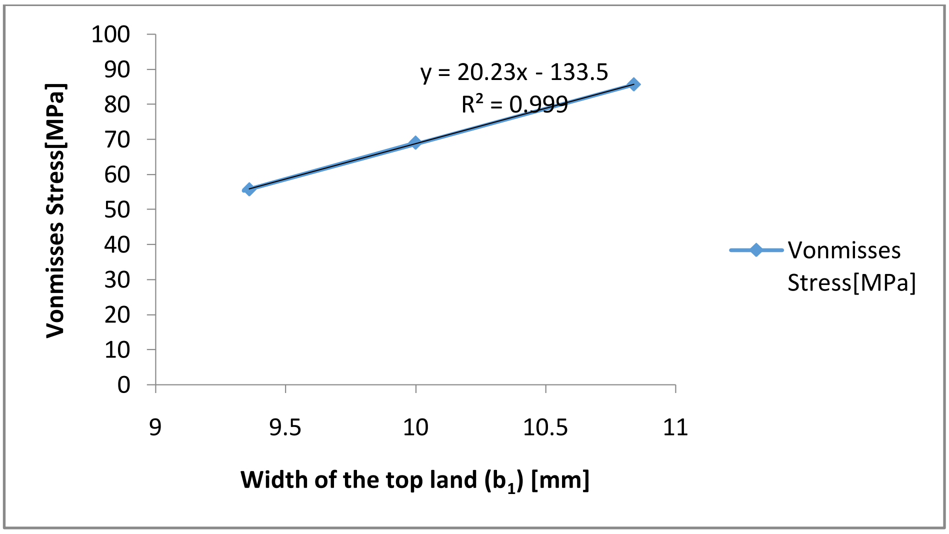

Fig. 12: Von-Misses stress at different width of the piston top land

The Figure12shows that the Von-Misses stress on the MDE piston are 85.6439MPa, 55.6685MPa and 69MPa at the respective piston top-land width of 10.84mm, 9.36mm and

International Journal of Advances in Engineering and Management (IJAEM)
Volume 3, Issue 7 July 2021, pp: 3585-3596 www.ijaem.net ISSN: 2395-5252

10mm.The width of the top land near piston rings is 5mm in size and is changed due to pressure and heat applied on rings through groves.

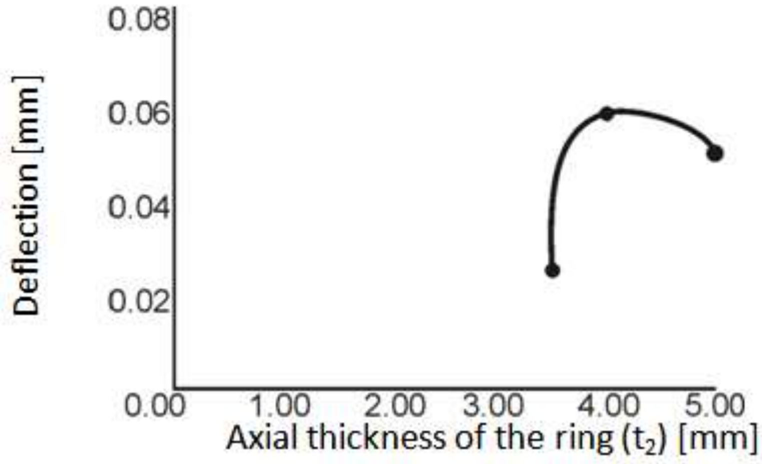

Fig. 13: Deflection at different axial thickness of the piston ring

The value after optimisation is 3.52mm and is rounded to 4mm.The figure 13. showed that the deflections on the MDE Piston are 0.051762mm, 0.025884mm and 0.05884mm at the respective piston axial thickness of 5mm, 3.52mm and 4mm.

V. CONCLUSION

The length 152mm and the diameter 140mm are assumed to be constant. It is not considered in terms of the variations in piston length and diameter of the piston. The radial thickness of the piston is affected more as it is very small in size. However, the temperature and heat flow are very high to the size of thickness. Initially, optimisation value was given as 5.24mm and later obtained after optimisation as 3.46mm. This is rounded to the next higher integer i.e., 4mm and is taken into consideration for design. The axial thickness of the piston ring before optimisation is 5mm. It is however, changed to 3.52mm after optimisation, since more heat and stress are experienced in the groves nearest to the head of the piston. This is rounded to the next higher integer i.e., 4mm, which is taken for design.

The maximum thickness of the barrel before optimisation is 14.34mm. However, it is much affected by thermal expansion after being subjected to high thermal loads. It is changed to 9.08mm and rounded to the next higher integer, i.e., 10mm which is taken for the design. The width of the top land initial value i.e., before optimisation is 10.84mm and is changed after applying pressure directly on the head i.e., top of the piston. This is as a result of the fact that, the shape of the piston on top will become just like a bowl. The value after optimisationis obtained as 9.36mm, and it is rounded to10mm. This value is considered for optimal design.

The stress distribution on the piston mainly causes the deformation of piston. Therefore, in order to reduce the stress concentration, the piston crown should have enough stiffness to reduce the deformation. The optimal mathematical model considers deformation of piston crown and quality of piston and piston skirt. The FEA is carried out for standard piston model used in Diesel Engine. The result of analysis indicates that, the maximum stress changes from 85MPa to 55MPa and biggest deformation has been reduced from 0.051762 mm to 0.025884 mm, respectively.

Based on the results and findings, it is advisable to choose piston materials with good thermo-mechanical properties, wear resistant and suitable lubricating characteristics. Piston deformation is caused by uneven stress distributions. To reduce the stress concentration, the piston crown configuration, thickness and stiffness must be suitable for the combustion flame progression. Finally, the optimisation successfully reduced the maximum stress from 85 MPa to 55 MPa, and the biggest deformation from 0.051762 mm to 0.025884 mm precisely.

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