International Journal of Marine Engineering Innovation and Research, Vol. 3(4), Mar. 2019. 148-155
(pISSN: 2541-5972, eISSN: 2548-1479)

Design and Optimisation of Marine Two-Stroke Diesel Engine Piston

Ifunanya Stella Ezeoye[1], Mohammad Orangian[2]

(Received: 06 February 2019 / Revised: 21 March 2019 / Accepted: 25 March 2019)

***Abstract*— in the cylinder of conventional compression ignition engine, temperature distribution and stress variation set up in the entire piston due to the inherent nature of combustion. This project will investigate the causes and effects of high gas pressure and temperature on marine two-stroke diesel engine piston. The investigation has shown that stress-related piston damages play a dominant role mainly due to mechanical and thermal stress or a combination of both. The simulation parameters that will be considered in this report are piston material, combustion temperature, and combustion pressure. In the present study, conventional marine two-stroke diesel engine piston will be modeled using Autodesk Inventor software. The working condition of the piston will be simulated based on the selected parameters using a finite element analysis method to analyze the magnitude of mechanical stresses acting on the piston. The piston will be re-designed using a material superior to the conventional piston material and will be optimized by reducing the stresses acting on the piston. Two materials will be used in the analysis after which the results will be compared and the best will be proposed. The structure and stress analysis (mechanical, thermal and thermal-mechanical stress) of the design will be modeled using the software mentioned earlier.**

Keywords— autodesk inventor, factor of safety, finite element method, stress, von-mises.

I. Introduction

Ships can be able to maneuver themselves in water with the use of propulsion forces. Successful movement of the ship in water by propulsion system is no longer the major concern nowadays. The major concern is how the best mode of propulsion system can be used to ensure a better safety standard for marine ecosystem along with cost efficiency. The major propulsion systems used in ships are diesel propulsion system, Gas turbine propulsion system and steam turbine propulsion system. The latter being the traditional propulsion system in shipping used especially in Liquefied natural gas (LNG) carriers [1]. However, conventional steam propulsion system has long start-up time, high fuel consumption and lower efficiency when compared to other types of propulsion systems. Due to these, currently, the steam propulsion plant is in competition with the new technologies such as diesel-electric system, dual-fuel diesel engine, and boiled-off gas re-liquefied system. Today, two-stroke or four-stroke diesel engines dominate the marine steam engines. Regular development of merchant ships to be more efficient is continuous. Base on an increasing market demand like lowest fuel consumption, the lowest possible propeller speed, higher power output, lower lubrication oil consumption, emission control and easy adjustment of the engine parameters. As a result of the above demands, engine layout, engine design parameters, and engine control need to be re-evaluated.

Modifications, optimization, and design of new engines are the current goal of engine designers, engineers, and manufacturers. For example; as it was read from MAN B & W “engine project guides”, small bore 7S35MC engine launched in 1998 with a mean effective pressure of 19.1bar at 173 rpm, thus providing 5180KW/Cylinder was modified to its ME-B series, 6S35ME-B9 with mean effective pressure of 21bar at 167 rpm producing 5220KW/Cylinder. The comparison between the new 6-cylinder engine and conventional 7-cylinder engine resulted in 40KW/Cylinder more power (MAN B&W, 2011). Also, MAN B&W and Sulzer have developed high-powered engines with maximum cylinder pressure of about 140-155 bar [2].

II. Method

A. Technical Proposal

Design of internal combustion engine piston is rather complex. The piston consists of three main parts; piston head or crown, piston ring, and piston pin. In this report, the design will be particularly based on the piston head as it is the crucial part of the piston where gas pressure is acted upon:

1) Piston

The piston is one of the most important components of the diesel engine. It is a disc that reciprocates within the cylinder. Functions of the piston in a two-stroke diesel engine:

- It converts the force produced by the combustion gases to mechanical power through its reciprocating motion to the crankshaft through connecting rod without loss of gas.
- It disperses a large amount of heat from the combustion chamber to the cylinder walls.
- The air in the cylinder is compressed by the piston crown while the piston skirt acts as a guide during the movement of the piston from top dead center (TDC) to bottom dead center (BDC).
- The piston crown and skirt should have enough stiffness to endure the pressure and friction between contacting surfaces and resist corrosive

Ifunanya Stella Ezeoye, South Shields Marine School United Kingdom, Email : ifuustella@gmail.com
Mohammad Orangian, South Shields Marine School United Kingdom, Email Mohammad.Orangian@stc.ac.uk

combustion products like heavy fuel oil or marine diesel oil.

2) Design requirements of piston material:

- Material with higher strength and heat resistance properties.
- It must be rigid enough to overcome mechanical and thermal loading without distortion**.**
- It must be able to dissipate heat to other components.
- The coefficient of expansion of the material should be at the minimum [3].

B. Approach

The initial design dimensions of the piston will be obtained from the piston available in South Shields Marine School workshop.

For the design approach, the piston will be modeled using Autodesk inventor software while mechanical stress analyses will be performed on the piston using the current material which is forged steel, this will be repeated using Aluminium alloy and heat resistant steel alloyed with chromium and molybdenum, the results will be compared while the best will be proposed for the manufacture of marine piston. Although "Aluminum alloy" is normally used for industry and aircraft pistons but the investigation has shown that the addition of the new alloys makes it have the advantage to retain strength at high temperature, due to this, tests will be performed on it to know if it can serve the same purpose for the marine piston. This is an overview of the intended materials to be used, detailed explanations and selection of the most suitable material will be shown in the next phase of the report.

1) Piston Force

The piston crown transfers combustion gas forces to the crankshaft via the connecting rod. The forces on the piston can be illustrated using the diagram in figure 1.

The gas pressure against the piston crown and the oscillating inertial forces referred to in the following as inertia force of the piston and the connecting rod constitute the piston force F_K . Due to the reduction of the piston force in the direction of the connecting rod (rod force is F_{ST}) an additional component arises following the force parallelogram namely the lateral force F_S also known as the normal force. This force presses the piston skirt against the cylinder bore. During a combustion cycle, the lateral force changes direction several times, which presses the piston from one side of the cylinder bore to the other [4].

2) Gas Pressure

The piston is subjected to the equilibrium of gas, inertia and supporting forces. The supporting forces are the resultant of the conneting rod and lateral forces. The maximum gas pressure in the combustion cycle has critical significance for the mechanical loads. At a maximum gas pressure in the combustion cycle of 150 bar (Assumed value obtained from higher power engines), for the piston with a diameter of 890mm (Dimension took from the marine workshop); the load which the piston is subjected to is given as:

$$P = \frac{F}{A} \tag{1}$$

Where:

P : pressure (N/m^2)

F : forces (N)

A : area (m^2)

3) Piston Clearance

In piston design, some clearance must be allowed. The clearance allowed depends on the actual engine type and rating. A common figure for diametral clearance of piston and cylinder in way of piston ring location in two-stroke cycle engines is given as:

Piston clearance in cylinder bore=0.2% of the cylinder bore.[5]. For this design, knowing that the cylinder bore 890 mm. Due to the maximum gas pressure, this imposes some stress on the piston crown.

4) The Nature of the Stresses acting on the Piston

The The stresses to which a piston is subjected are compressive and tensile caused by bending action due to gas pressures, inertia effects, and thermal stresses. When the crown of a piston is subjected to gas pressures the top surface of the piston is under compressive loading and the lower surface is under tensile loading. The piston crown is then behaving somewhat like a uniformly loaded beam. When the piston is moving upward towards the end of its stroke, retardation occurs and the inertia effects tend to cause the piston to bow upwards so that the top surface of the piston, together with the sides, is under tensile loading and the lower surface of the crown is under compressive loading. The pressure on the top of the piston nullifies the inertia effects when the piston approaches the top center position in the upward direction.

When the piston is retarded on its approach downwards to bottom-center, the piston crown tends to bow downwards, and its upper surface and the piston walls are in compression. The lower surface of the piston crown is then in tension. As the stresses from inertia effects are in the same direction as those caused by gas pressure on the piston crown the two stresses become additive; thus when the piston approaches bottom-dead-center the inertia stresses increase the stresses caused by gas pressure.

The thermal stresses set up in a piston are caused by the difference in temperatures across a section. The free expansion of the hot side is resisted by the cold side which does not want to expand so much, as it is cooler. This section then sets up thermal stresses in the material of the piston, these stresses being greatest where the difference in temperature of the material across any section is greatest [5].

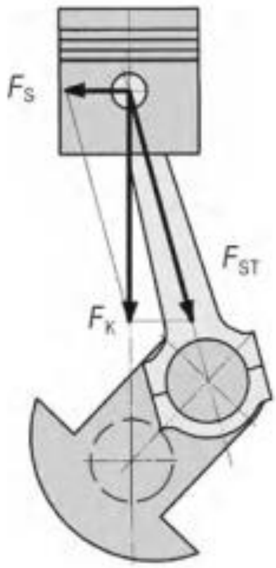

Figure. 1. Forces on the piston

C. Piston Design

1) Geometry Modeling

The basic structure of the piston is a hollow cylinder closed on one side with the segments piston crown with ring grooves.

The piston model was developed in Autodesk Inventor using the dimensions obtained from the piston available in South Shields marine workshop E122. Piston measured data:

Piston diameter	:890 mm
Length of piston head	:450 mm
The thickness of piston head	:100 mm
Maximum cylinder pressure (assumed)	:150 bar

The diagram on figure 2 is the piston model drawn in the inventor using the measured dimensions. The sketch was done by activating a cylinder tool on a 3D modeling tab in the inventor after which the required diameter was inserted. Extrude tool was used by filling the required distance after which the command was executed. Features like revolving, sweep loft, fillet holes, mirror, etc. were used for the sketch. Five compression ring grooves and one oil scraper ring groove were created using a rectangular pattern.

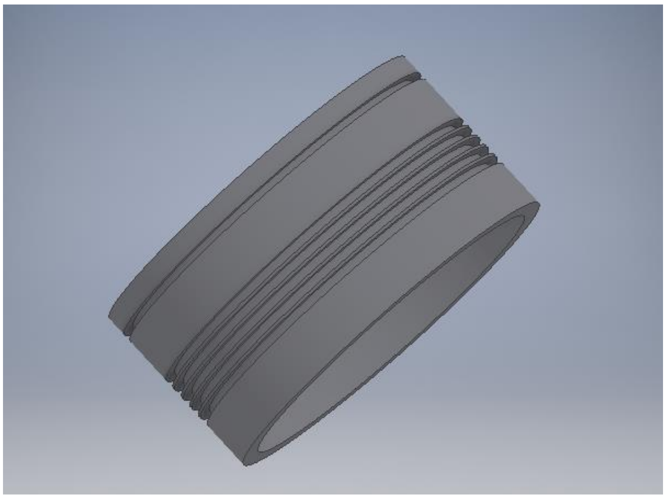

Figure. 2. Original design of piston model

D. Testing

The materials selected for the design of the piston (refer table 1) will be used to conduct a strength analyses on the piston model to identify the material with lower stress concentration which will be proposed to the marine industry. As it was discussed earlier that due to the complexity of the piston model, the design cannot be done analytically, therefore, the design test and analysis will be done by simulating the piston model using Autodesk Inventor software using Finite Element Analysis method.

The material used for the design of piston will be assessed based on their mechanical, thermal and chemical properties. The table below shows the mechanical properties of piston materials.

TABEL 1.
MECHANICAL PROPERTIES OF THE PISTON MATERIALS

Mechanical Properties	Forged steel (AISI 4340)	Aluminium Alloy (4032)	Heat resistance steel (42CrMo4)
Young's Modulus (GPa)	209	79	212
Poisson's Ratio	0.28	0.33	0.3
Shear Modulus (GPa)	85	29.26	81.54
Density (kg/m^3)	7800	2700	7800
Yield strength (MPa)	500	315	750
Tensile strength (MPa)	750	380	900

1) Finite Element Method

This is the mathematical idealization of the real system. It is a computer-based method that breaks geometry into elements joined by nodes and a series of the equation to which elements are formed and then solved simultaneously to evaluate the behavior of the entire system.

2) Design Assumptions

It is quite impossible to avoid assumptions in design works due to the fact that actual design condition varies with respect to time. The assumptions are always made depending upon the details and accuracy required in the modeling. The piston design is based on the following assumptions:

- The materials of the piston are considered as homogeneous and isotropic.
- The piston is assumed to be in static condition.
- The effects of inertia and body force are negligible during the analysis.
- Before the application of the analysis, the piston is stress-free.
- The material's thermal conductivity is uniform throughout.
- The material's specific heat is constant throughout and does not change with temperature.

3) Stress Analysis

For the analysis, linear static stress analysis simulation was created after which material was assigned to the piston model. The materials needed for the simulation were not present in the material library of the inventor; therefore, the new material library was created by inserting the material's properties needed for the design in the inventor material library. Then the next step is setting up of boundary conditions. The boundary condition for the mechanical simulation was defined as a fixed constraint at the outer diameter of the piston skirt. The next boundary condition was defined as the pressure on the entire piston head surface (maximum firing pressure in the cylinder which is assumed to be 150 bar). This acts as a uniform distributed load on the piston head surface. Then, the mesh of 1.5mm was applied to the model. The essence of meshing the geometry is to obtain more accurate results. These steps lead to the calculation of stresses, displacements, and factor of safety of the piston model.

Step 1: Firstly, the piston model was tested using the conventional material which is forged steel. The maximum stress which the piston is subjected to due to the pressure of 150 bar acting on the piston head surface is shown in figure 3.

Step 2: The stress analysis was repeated on two alternative materials selected for the design which are Aluminium alloy and Alloy steel by applying the same pressure on the piston head surface. The figure 3 displays the magnitude of the calculated stresses for the three different materials. Figure 3 shows the maximum stress the piston is subjected to when designed with the conventional material and two alternative materials.

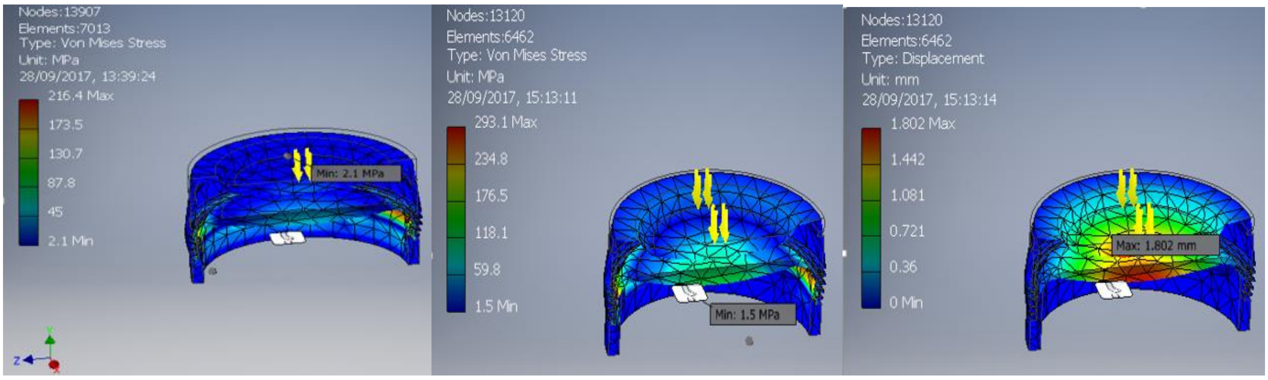

Figure. 3. Stress analysis results in three different materials

Figure 4 shows the maximum displacement the piston is subjected to due to the maximum stress on the piston head surface when designed with the conventional material and two alternative materials.

The diagram below displays the magnitude of the calculated factor of safety for the three different materials.

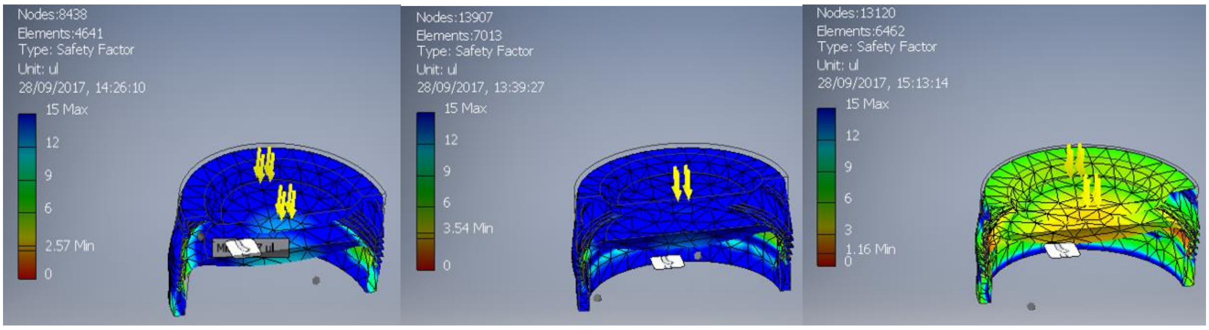

Figure. 4. Safety factor results in three different materials

Table 2 shows the results obtained from static stress analysis when the pressure of 150 bar is applied on the surface of the piston head.

Table 3 shows the results obtained from static stress analysis when the pressure of 150 bar is applied on the surface of the piston head.

TABEL 2.
RESULTS OBTAINED FROM LINEAR STATIC STRESS ANALYSIS UNDER MECHANICAL LOADING

Materials	Von-misses stress (MPa)	Displacement (mm)	Factor of safety
Forged steel (medium carbon steel)	221.8 max	0.3416 max	2.57min
Al4032	293.1 max	1.802 max	1.16 min
42CrMo4	216.4 max	0.3083 max	3.54 min

TABEL 3.
RESULTS OBTAINED FROM STATIC STRESS ANALYSIS UNDER MECHANICAL LOADING

Materials	Von-misses stress (MPa)	Displacement (mm)	Factor of safety
Original piston with Forged steel (AISI4340)	221.8 max	0.7204 max	2.57 min
Piston with new material 42CrMo4	216.4 max	0.3083 max	4.07 min
Optimized piston (with a cavity) with new material	217.3 max	0.3416 max	3.54 min

III. RESULTS AND DISCUSSION

A. Results after Optimization

The results obtained from the finite element analysis were inputted in an excel sheet to obtain a bar chart. The results obtained were represented with a bar chart from figure 6 to 11. Figure 6 shows the stress which the piston is subjected to in terms of Von Mises stress criterion. The maximum stress on forged steel piston is 221.8Mpa while that of Aluminum is 293.1Mpa and that of Heat resistance steel is 216.4Mpa.

From figure 7, piston designed with Aluminum alloy has the highest displacement of 1.802mm, the piston designed with forged steel is displaced by 0.7204mm while the piston designed with heat resistance steel has the lowest displacement of 0.3083mm. Referring to figure 8, the piston designed with heat resistance steel has the highest factor of safety of 4.07 while that of Aluminum is 1.16 and forged steel is 2.57. From the values listed above, the piston designed with Aluminum will fail quickly while in service as it has the highest Von Mises stress with the lowest factor of safety when compared to the other two materials. Also, the piston designed with the conventional material which is forged steel is more stressed than the piston designed with the new material which is heat resistant steel. Therefore, the piston designed with heat resistant steel is an excellent choice for piston design as it has the lowest stress with the highest safety factor with less displacement due to stress.

Furthermore, the piston designed with the new material was optimized by creating a cooling cavity. The results obtained after the finite element analysis of the optimized piston were compared to the piston designed with forged steel and heat resistant steel without a cavity. Figure 9 shows the Von Mises stress obtained from the optimized piston with cavity compared to that designed with the same material but without a cavity and forged steel. The piston with a cavity has maximum stress of 217.3Mpa while the piston with the same material but without cavity has a maximum stress of 216.4Mpa while that of forged steel is 221.8Mpa. From figure 10, the piston with a cavity is displaced by 0.3416mm while that of the same material without the cavity is displaced by 0.3083mm and that of forged steel is 0.7204mm. Referring to figure 11, the safety factor of the piston with the cavity is 3.54 minimum while that of forged steel is 2.57minimum and piston without a cavity is 4.07. These results show that stress increased by 0.9Mpa after the piston with new material was optimized. This increased the displacement from 0.3083mm to 0.3416mm while the factor of safety was reduced by 0.53.

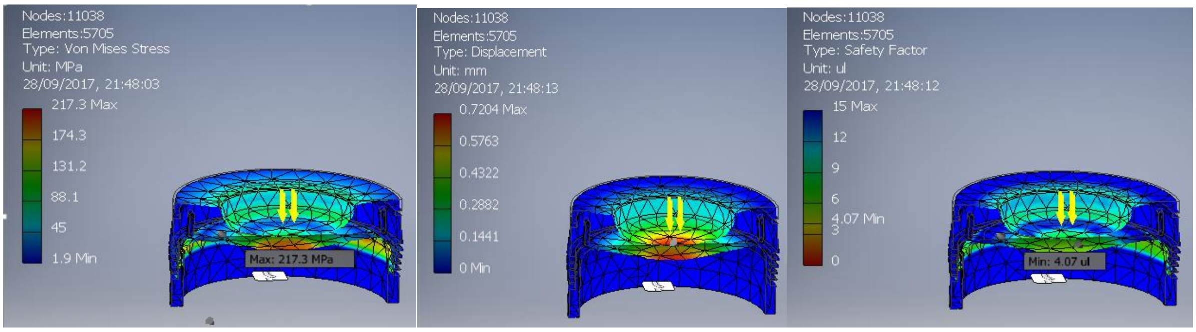

Figure. 5. Stress analysis results in the optimized piston

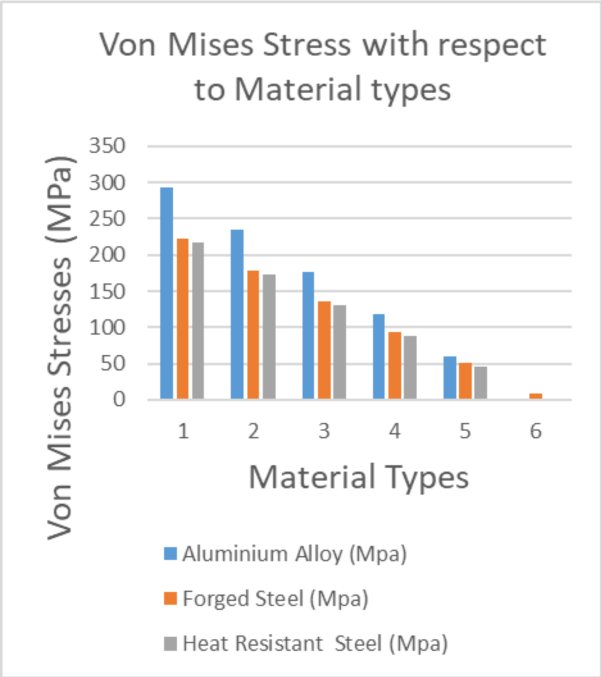

Figure. 6. Von Mises stress graph of three different materials

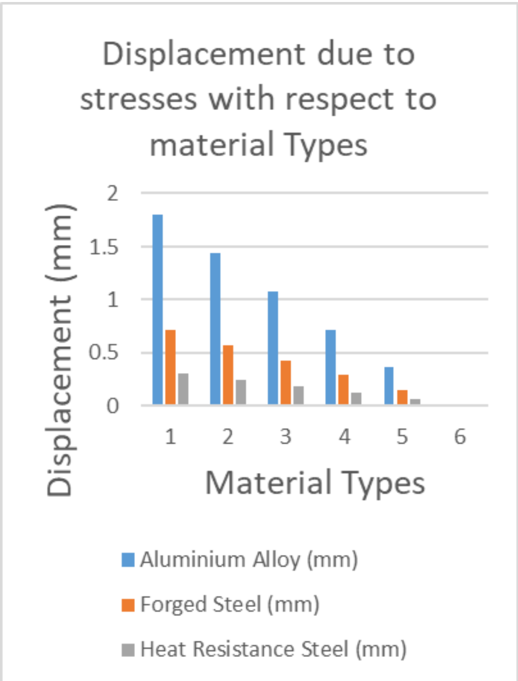

Figure. 7. Displacement graph of three different materials

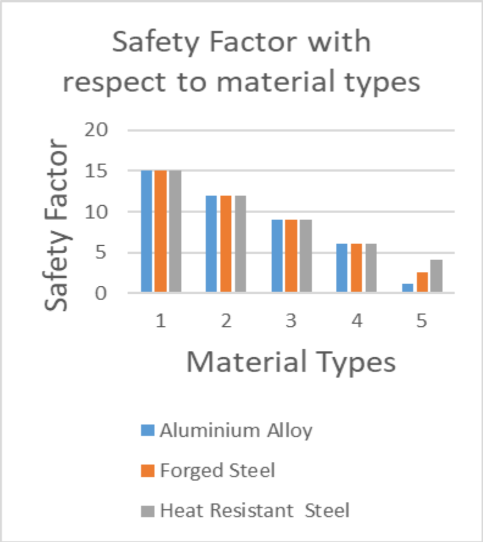

Figure. 8. Safety factor graph of three different materials

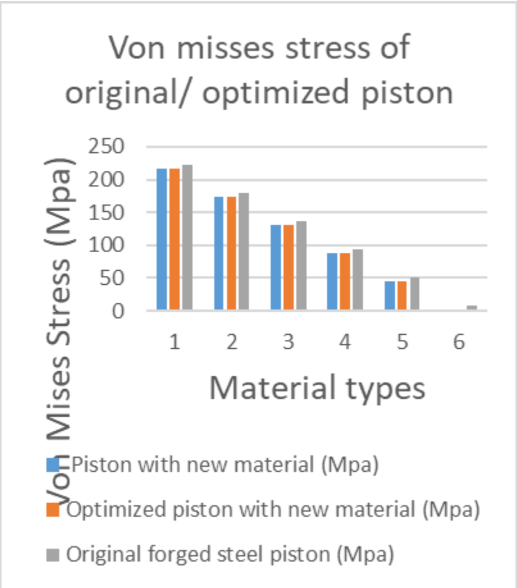

Figure. 9. Von Mises stress graph of optimized piston compared to the conventional piston

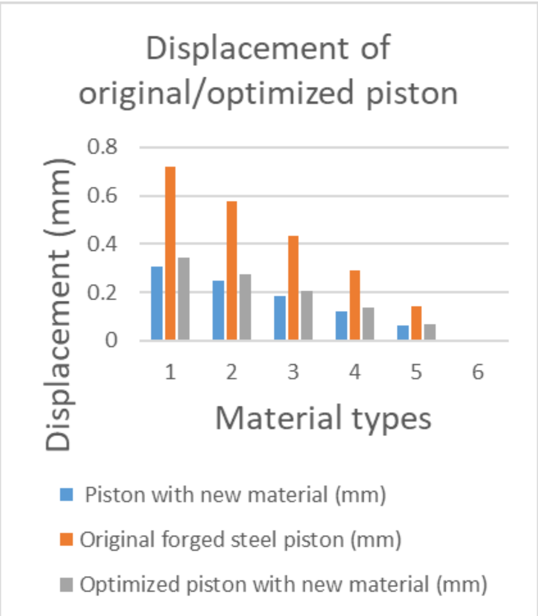

Figure. 10. Displacement graph of optimized piston compared to conventional piston.

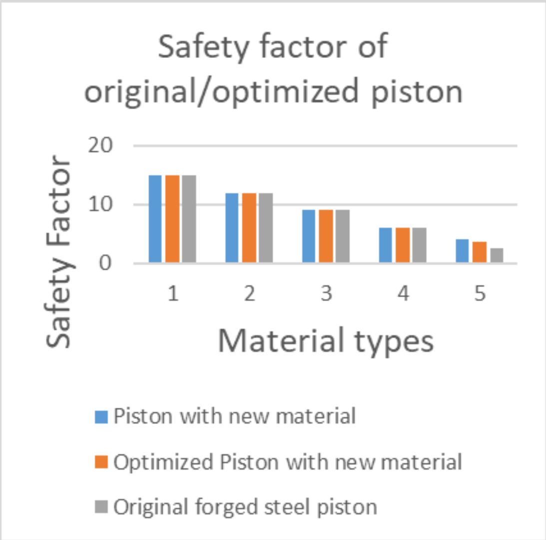

Figure. 11. Safety factor graph of optimized piston compared to conventional piston

IV. Conclusion

Investigations were done on the design of marine two-stroke engine piston. The dimensions measured from the marine workshop were used to develop the piston model successfully as stated in the objective. Stress analysis was performed on the piston using finite element method on the original material which is forged steel and two alternative materials which are Aluminium alloy and Heat Resistant Steel. Results obtained showed that piston designed with Heat Resistant Steel is less stressed compared to others. This material was used in the proposed design after which it was optimized by creating a cooling cavity on the piston. This increased the Von Mises stress to 217.3Mpa with the factor of safety of 3.54 and displacement of 0.3416mm. Comparing this values to the values obtained from piston designed with conventional material which has maximum stress of 221.8Mpa, safety factor of 2.57 and maximum displacement of 0.7204mm; the piston designed with Heat Resistant Steel has higher strength compared to that of the piston designed with conventional material thus the objective which is to design a piston with higher strength has been achieved.

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