# Design and Stress Analysis of Marine Diesel Engine Piston Using Solid Works and Finite Element Method (FEM)

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**Abstract:** Piston plays a very critical role in energy conversation as far as internal combustion engines are concern. Piston is a major component of the internal combustion engine and directly bears the high temperature and pressure developed due to combustion of fossil fuel in the combustion chamber. As a result, the piston is subjected to high thermal and structural stresses due to high temperature, reciprocating speed and loads fluctuations. If these stresses exceed the designed values, piston failure will occur. In this work, Aluminum alloy have been selected for structural and thermal analysis of piston and the stresses due to high combustion temperature are analyzed using finite element method in order to predict areas of the piston susceptible to failure. The piston in this work is designed using SOLID WORKS and imported to ANSYS R19.1 software for thermal and structural analysis. The result obtained shows that finite element method offers a reliable means to predict failure characteristics of an aluminum alloy piston. The finite element result analyses indicate that the maximum stress is $3.2566 \times 10^6 Pa$ and the maximum expected deformation is 1.7852mm.

## 1. INTRODUCTION

Piston is one of the most important components of an internal combustion engine which reciprocates within the cylinder. The main function of the piston is to transfer force from combustion gas in engine, piston endures the cyclic gas pressure and the inertial forces at work, and this working condition may cause the fatigue damage of piston, such as piston side wear, piston head/crown cracks and so on (1); (8). The investigations indicate that the greatest stress appears on the upper end of the piston. Stress concentration in piston is one of the main reasons for fatigue failure (2). On the other hand piston overheating-seizure can only occur when something burns or scrapes away the oil film that exists between the piston and the cylinder wall (3). There are lots of research works proposing for engine pistons, new geometries, materials and manufacturing techniques, and this evolution has go on with a continuous improvement over the last decades and required thorough examination of the smallest details.

Notwithstanding, there are a huge number of dam aged pistons having different causative origins which are mainly: wear, temperature, and fatigue related. But more than wear and fatigue, damage of the piston is mainly due to stress development, namely- Thermal stress and Mechanical stress (4).

In engine, transfer of heat takes place due to difference in temperature and from higher temperature to lower temperature (5). Thus, there is heat transfer to the gases during intakes stroke and the first part of the compression stroke, but the during combustion and expansion processes the heat transfer take place from the gases to the walls. So the piston crown, piston ring and the piston skirt should have enough stiffness which can endure the pressure and the friction between contacting surfaces (6); (7). The thermal and mechanical deformation causes piston cracks (7). Therefore, it is very essential to analyses the stress distribution, temperature distribution, heat transfer, mechanical load in order to minimize the stress at different load on the piston (2); (8).

This works describes the stress distribution on the piston of marine diesel engine by using ANSYS software base on Finite Element Method (FEM). The main objective of this research is to investigate and analyze the thermal and mechanical stresses distribution of pistons at real engine condition during combustion process.

The piston is designed according to the standard procedures and specifications. The pressure applied on piston head, temperatures of various areas of the piston, heat flow, stresses, strains, length, diameter of piston and hole, thicknesses are all taken into consideration.

In designing a piston for an engine, the following points should be taken into consideration (9): 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 have sufficient support for the piston pin.

**2.1 Engine Specifications:**

Engine: Detroit V8: 71 Power Marine Diesel Engine
Type: Four Stroke, Water Cooled Diesel
Bore Diameter: 94mm
No. of cylinder: 8 Cylinder Engine
Stroke Length: 280mm
Temperature: 800℃
Gas Pressure: 55bar
Compression Ratio: 18.7
Combustion Principle: Compression Ignition



**TABLE 1.1: Piston Dimension**

| S/N | DESCRIPTION | NOTATION | UNIT | VALUE USED |
|---|---|---|---|---|
| 1 | Length of Piston | $l$ | Mm | 115.2 |
| 2 | Outside diameter of Piston | $d$ | Mm | 94 |
| 3 | Length of skirt | $l_s$ | Mm | 96.2 |
| 4 | Thickness of Piston Head | $t_H$ | Mm | 15 |
| 5 | Radial thickness of the ring | $t_1$ | Mm | 2 |
| 6 | Axial thickness of the ring | $t_2$ | Mm | 2 |
| 7 | Maximum Thickness of barrel | $t_3$ | Mm | 6 |
| 8 | Width of the top land | $b_1$ | Mm | 2 |

| | | | | |
|---|---|---|---|---|
| 9 | Width of other ring lands | $b_2$ | Mm | 5 |
| 10 | Piston pin diameter | $d_o$ | Mm | 25 |
| 11 | Number of rings | $n_r$ | Mm | 6 |

The diagram on Fig. 2.1a to 2.1c is the piston model drawn in solidworks software using the measured dimensions. The sketch was done by activating a cylinder tool on a 3D modeling tab in the software 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, fillet holes, mirror, etc. were used for the sketch. fourcompression ring grooves and two oil scraper ring groove were created using a rectangular pattern.The table below shows the mechanical properties of piston materials.

### 2.2 Material Properties

Aluminum alloys are widely used in engineering str uctures and components wherelight weight or corro sion resistance is required (4); (9).



Table 2.1 Piston Properties

| Mechanical Properties | Aluminium Alloy |
|---|---|
| Young's Modulus | 7.1e+010 Pa |
| Poisson's Ratio | 0.33 |
| Shear Modulus | 2.6692e+010 Pa |
| Density | 2770 $kg/m^3$ |
| Tensile Yield Strength | 2.8e+008 Pa |
| ensile Ultimate Strength | 3.1e+008 Pa |

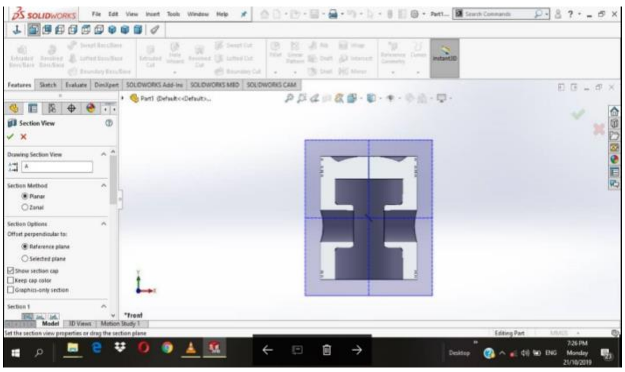

Fig 2.1a Piston Design Step

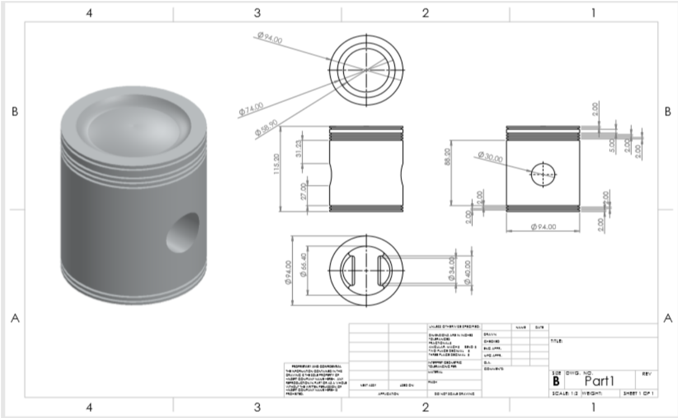

**Fig. 2.1b** Piston Drawing Sheet

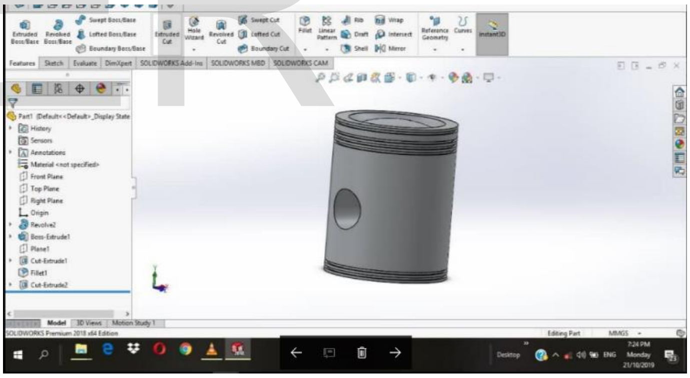

Fig. 2.1c 3D Piston Model

## 3. RESULT AND DISCUSSION

From the methodology presented in this work, the following analyses were performed and results obtained are presented as follows:

### 3.1 Static Structural Result

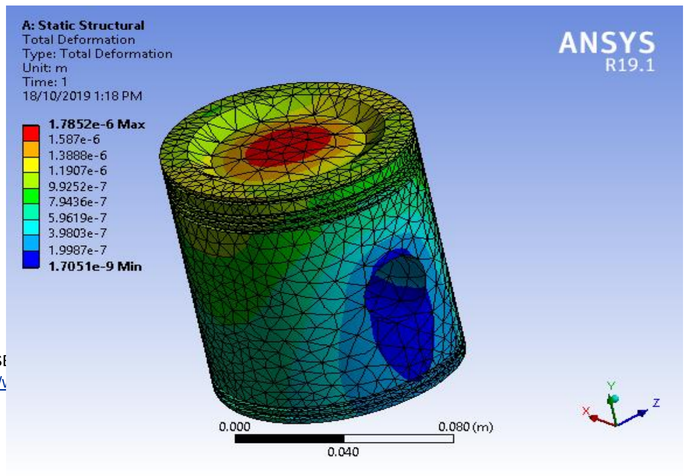

Fig 3.1Thetotal deformation of the piston during operation

The maximum deformation of the piston is expected at the mid region of the piston at seen in Fig. 3.1. This expected deformation is of the range of 1.705× $\mathbf{10^{-6}}$mm to 1.7852mm. The minimum deformation will arise during startup of the engine when the piston is running at ambient temperature and relatively close to such. The maximum deformation will occur at full load of the engine and after the piston has ran for some time; this is due to both the high action of the thermal and structural stresses.

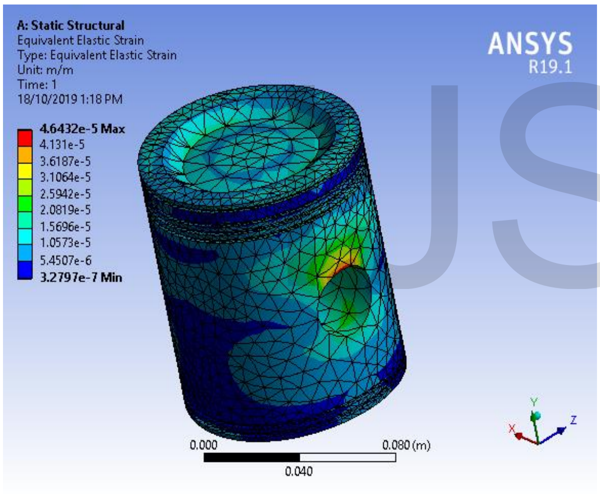


Fig 3.2 The Equivalent Elastic Strain Of The Piston During Operation

The expected elastic strain on the piston is 3.2797× $\mathbf{10^{-7}}$ for its minimum stress (thermal and structural), and 4.6432× $\mathbf{10^{-5}}$for its maximum expected stress. As seen in Fig. 3.2 and Fig. 3.3, the maximum strain is expected to arise due to the action of the gudgeon pin on the piston due to the push up effect of engine connecting rod.

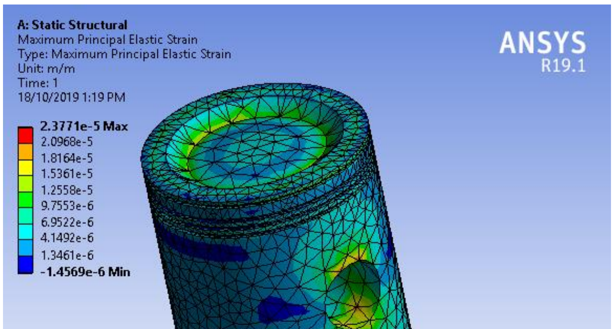


Fig 3.3 the maximum principal elastic strain

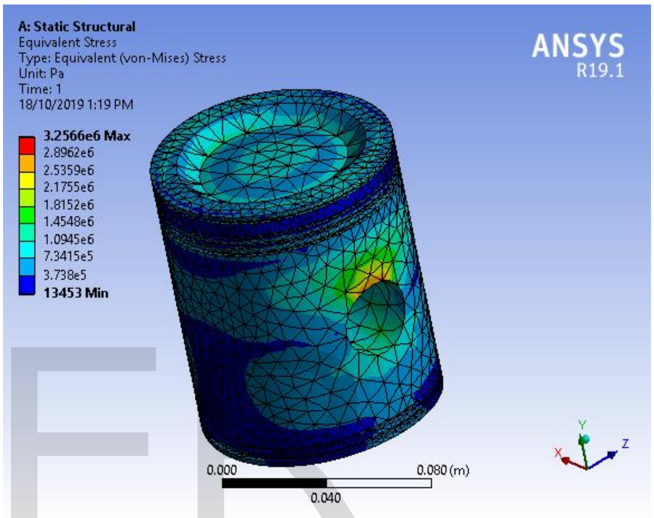


Fig 3.4 The Equivalent Stress Of The Piston During Operation

During operation the piston received pressure action from the gudgeon pin. This is reflected as the equivalent stress. As seen in Fig. 3.4 the equivalent stress is maximum at the gudgeon pin opening. The expected equivalent stress on the piston is 13453Pa (minimum stress), and $3.2566 \times 10^{6} Pa$ (maximum stress).

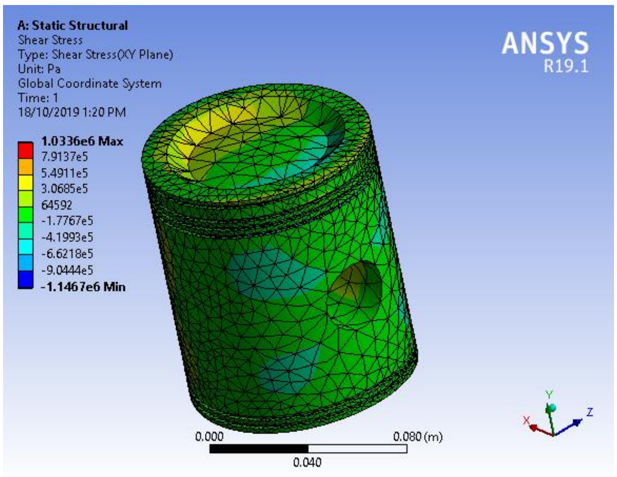

Fig 3.5 The Shear Stress Of The Piston During Operation

The axial shear on the piston is due to the explosion during piston operation. This action is mostly felt at the crevice of the piston head. The maximum permissible axial shear stress on the piston is 1.0336×$\mathbf{10^6}$Pa as shown in Fig. 3.5.

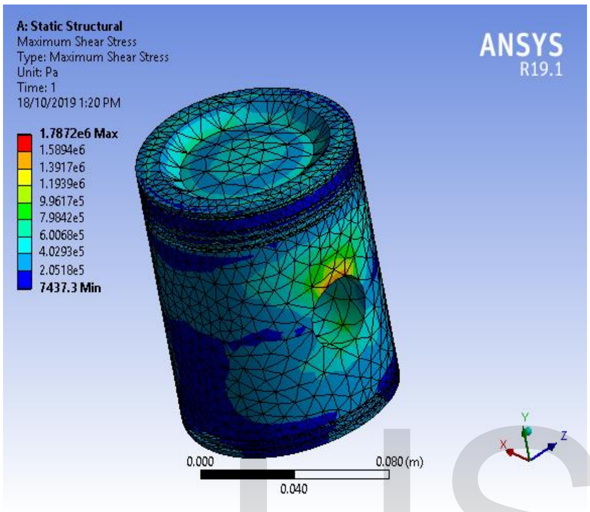


Fig 3.6 The maximum Shear Stress Of The Piston During Operation

TABLE 3.1: Static Structural Results

The maximum shear stress of the piston is expected at the gudgeon pin opening of the piston at seen in Fig. 3.6. This is due to the action of the gudgeon pin on the piston.

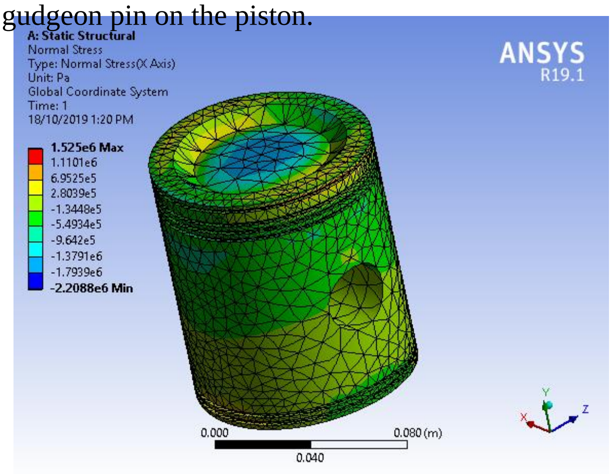


Fig 3.7 The Normal Stress Of The Piston During Operation

The normal stress is maximum at the point where the downward pressure of the fuel combustion and the action of the gudgeon pin on the piston is most felt. The normal stress is of the range of -2.2088x$\mathbf{10^6}$Pa (minimum stress) and 1.525×$\mathbf{10^6}$Pa as can be seen in Fig. 3.7.

| Object Name | *Total Deformation* | *Equivalent Elastic Strain* | *Maximum Principal Elastic Strain* | *Equivalent Stress* | *Shear Stress* | *Maximum Shear Stress* | *Normal Stress* |
|---|---|---|---|---|---|---|---|
| State | Solved | | | | | | |
| **Scope** | | | | | | | |
| Scoping Method | Geometry Selection | | | | | | |
| Geometry | All Bodies | | | | | | |
| **Definition** | | | | | | | |
| Type | Total Deformation | Equivalent Elastic Strain | Maximum Principal Elastic Strain | Equivalent (von-Mises) Stress | Shear Stress | Maximum Shear Stress | Normal Stress |
| By | Time | | | | | | |
| Display Time | Last | | | | | | |
| Calculate Time | Yes | | | | | | |

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| History | | | | | | | |
| Identifier | | | | | | | |
| Suppressed | No | | | | | | |
| Orientation | | | | | XY Plane | | X Axis |
| Coordinate System | | | | | Global Coordinate System | | Global Coordinate System |
| **Results** | | | | | | | |
| Minimum | 1.7051e-009 m | 3.2797e-007 m/m | -1.4569e-006 m/m | 13453 Pa | -1.1467e+006 Pa | 7437.3 Pa | -2.2088e+006 Pa |
| Maximum | 1.7852e-006 m | 4.6432e-005 m/m | 2.3771e-005 m/m | 3.2566e+006 Pa | 1.0336e+006 Pa | 1.7872e+006 Pa | 1.525e+006 Pa |
| Average | 7.3484e-007 m | 1.0004e-005 m/m | 4.8699e-006 m/m | 6.6885e+005 Pa | -1108.5 Pa | 3.689e+005 Pa | -1.7136e+005 Pa |
| Minimum Occurs On | piston-Free Parts | | | | | | |
| Maximum Occurs On | piston-Free Parts | | | | | | |
| **Information** | | | | | | | |
| Time | 1second | | | | | | |
| Load Step | 1 | | | | | | |
| Substep | 1 | | | | | | |
| Iteration Number | 1 | | | | | | |
| **Integration Point Results** | | | | | | | |
| Display Option | | Averaged | | | | | |
| Average Across Bodies | | No | | | | | |



### 3.2 Transient Thermal Result

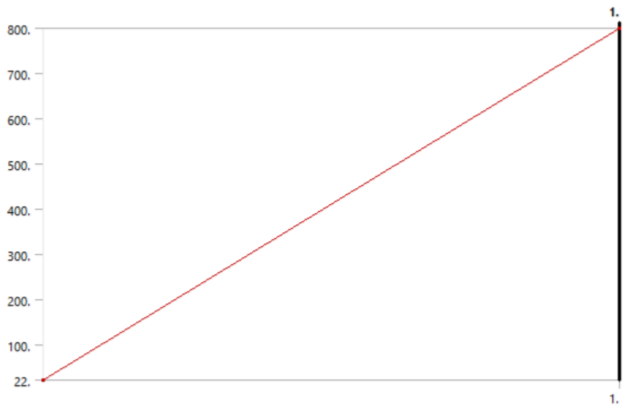


Fig 3.8 shows the thermal increase of the system with time. As shown, the temperature of the piston is minimum at startup of the engine and increases linearly until it reaches its peak (maximum temperature of 800℃.)

Fig 3.8 Graph showing the temperature of the

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Piston during operation

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**TABLE 3.2 Transient Thermal Results**

| Object Name | *Temperature* | *Total Heat Flux* | *Directional Heat Flux* |
|---|---|---|---|
| Minimum | 149.04 °C | 38520 W/m² | -2.0917e+007 W/m² |
| Maximum | 800. °C | 2.5671e+007 W/m² | 2.0883e+007 W/m² |
| Average | 531.15 °C | 7.0151e+006 W/m² | 18754 W/m² |
| Minimum Occurs On | piston-Free Parts | | |
| Maximum Occurs On | piston-Free Parts | | |
| **Minimum Value Over Time** | | | |
| Minimum | 5.9943 °C | 175.39 W/m² | -2.0917e+007 W/m² |
| Maximum | 149.04 °C | 38520 W/m² | -1.381e+006 W/m² |
| **Maximum Value Over Time** | | | |
| Minimum | 38.45 °C | 2.9418e+006 W/m² | 1.3753e+006 W/m² |
| Maximum | 800. °C | 2.5671e+007 W/m² | 2.0883e+007 W/m² |



As seen in table 3.2, the maximum temperature in of the piston is 149.04℃ at first run, and 800℃ at full load. Consequently, the minimum temperature of the piston is 5.99℃ conditionally, and 38.45℃ at full load.

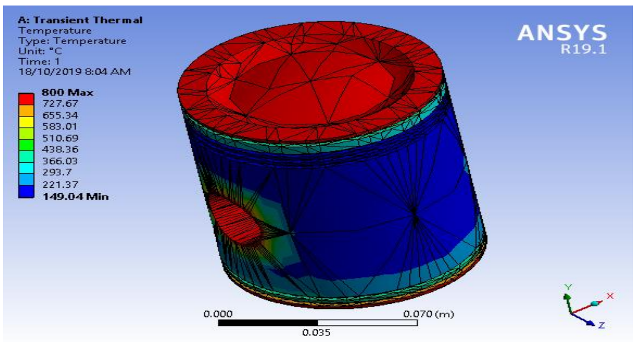


Fig 3.9 the temperature of the piston during operation

Fig 3.9 shows the temperature distribution of the piston. The top of the piston which receives direct heat from the combustion of gases would see the highest temperature concentration. High temperature is also expected on the crevice of the gudgeon pin opening due to the frictional action of the gudgeon pin on the piston surface.

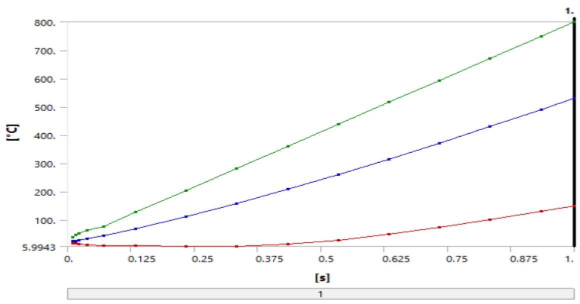


Fig 3.10 Graph showing the temperature rise of the piston within 1s.

TABLE 3.3 Transient Thermal Temperature

| Time [s] | Minimum [°C] | Maximum [°C] | Average [°C] |
|---|---|---|---|
| 1.e-002 | 18.147 | 38.45 | 24.361 |
| 1.5727e-002 | 16.278 | 45.85 | 25.77 |
| 2.1453e-002 | 14.718 | 51.839 | 27.26 |
| 3.8004e-002 | 12.118 | 63.085 | 32.101 |
| 7.0223e-002 | 9.2665 | 76.633 | 42.839 |
| 0.13406 | 9.4475 | 126.3 | 67.275 |
| 0.23406 | 5.9943 | 204.1 | 110.55 |
| 0.33406 | 6.7145 | 281.9 | 157.56 |
| 0.43406 | 13.985 | 359.7 | 207.56 |
| 0.53406 | 28.017 | 437.5 | 260.13 |
| 0.63406 | 48.196 | 515.3 | 314.96 |
| 0.73406 | 73.971 | 593.1 | 371.83 |
| 0.83406 | 101.41 | 670.9 | 430.49 |
| 0.93406 | 129.1 | 748.7 | 490.77 |
| 1. | 149.04 | 800. | 531.15 |

Table 3.3 shows the temperature rise of the piston within one second, analyzed evenly within one minute. As seen, the minimum temperature rises evenly from 18.147℃ to 149.04℃, as the maximum temperature rises from 38.45℃ to 800℃.

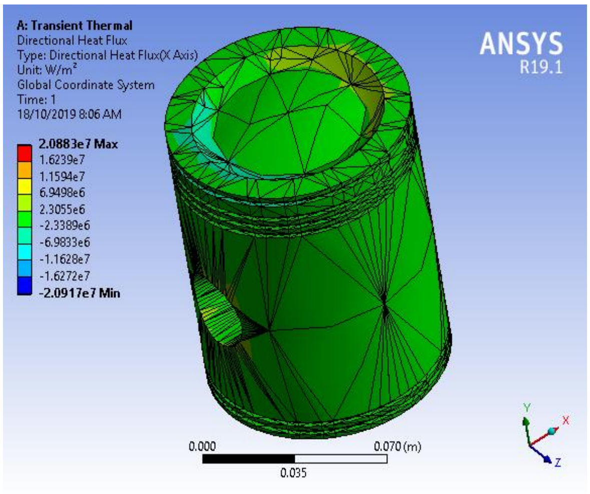


Fig 3.11 the total heat flux

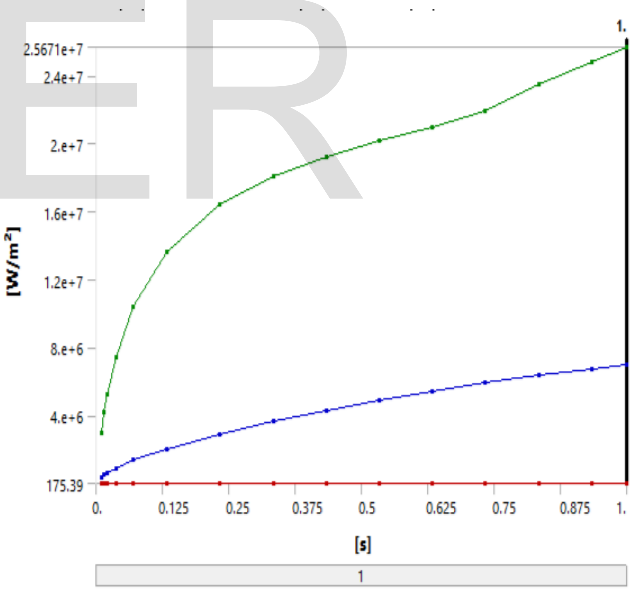


Fig 3.12 graph showing the total heat flux

TABLE 3.4 Transient Thermal Total Heat Flux

| Time [s] | Minimum [W/m²] | Maximum [W/m²] | Average [W/m²] |
|---|---|---|---|
| 1.e-002 | 376.79 | 2.9418e+006 | 3.3952e+005 |
| 1.5727e-002 | 270.44 | 4.1649e+006 | 4.8668e+005 |
| 2.1453e-002 | 175.39 | 5.1997e+006 | 6.151e+005 |
| 3.8004e-002 | 206.72 | 7.4378e+006 | 9.1176e+005 |
| 7.0223e-002 | 297.19 | 1.0348e+007 | 1.3548e+006 |
| 0.13406 | 750.77 | 1.3632e+007 | 2.0264e+006 |
| 0.23406 | 2287.3 | 1.6385e+007 | 2.8862e+006 |
| 0.33406 | 4203. | 1.8031e+007 | 3.6357e+006 |
| 0.43406 | 2344.6 | 1.9179e+007 | 4.2966e+006 |
| 0.53406 | 8448.5 | 2.0129e+007 | 4.8902e+006 |
| 0.63406 | 12687 | 2.0946e+007 | 5.4283e+006 |
| 0.73406 | 6302.8 | 2.1928e+007 | 5.9172e+006 |
| 0.83406 | 12878 | 2.3446e+007 | 6.3604e+006 |
| 0.93406 | 34327 | 2.482e+007 | 6.7643e+006 |
| 1. | 38520 | 2.5671e+007 | 7.0151e+006 |

Table 3.4, Fig 3.11 and 3.12 shows the heat concentration (flux) of the piston during operation. The minimum heat flux rises steadily from 376.79W/m$^2$ to 38.52kW/m$^2$ as the piston operates between its maximum and minimum temperatures. Consequently, the maximum heat flux rises from 2.9418MW to 25.671MW.

## 4. CONCLUSION

From the analyses, it has been shown that although thermal stress is not responsible for biggest slice of damaged pistons, it remains a problem on engine pistons, and its solution remains a goal for manufacturing robust piston for marine diesel engine application. Issues posed by thermal stress on pistons will remain a problem for a while. This is because efforts in reducing fuel consumption and quest for power increase will push to limit weight reduction, unavoidably leading to thinner walls and higher thermal stresses. In satisfying all the critical requirements for robust piston design suitable for marine application, knowledge on piston behavior under working conditions is paramount which this work intents to bring to lime light by putting forward a solid work model of the piston coupled with an ansys analysis used to obtain the thermal stress field only caused by the uneven temperature distribution when piston works. Analytically, the stress distribution on the piston basically depends on the piston deformation. It is obvious from this research point of view, to reduce the stress concentration; the piston crown should have enough stiffness to withstand this stress and deformation. Similarly, due to this deformation, the greatest stress concentration is caused on the piston's upper end. Also the stress distribution on the piston mainly depends on the deformation of piston. The FEM prove to be a very veritable tool in proffering the solutions to mitigate the effect of these stresses and deformations during piston design and manufacturing.

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