

# **University of Wales Trinity Saint David**

**Wales Institute of Science & Art**



## **Application of Computer Aided Engineering Processes.**

**Structural Design & Fluid Analysis / ACHA6010**

Dylan Joseph Arthur

2004556

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## Contents

|       |                                              |    |
|-------|----------------------------------------------|----|
| 1     | Introduction .....                           | 2  |
| 1.1   | Methodology.....                             | 2  |
| 2     | Validation Study .....                       | 3  |
| 2.1   | Analytical validation.....                   | 3  |
| 2.1.1 | Cantilever beam UDL .....                    | 3  |
| 2.1.2 | Bimetallic Strip Thermal Load .....          | 5  |
| 2.2   | Mesh Independence study.....                 | 7  |
| 3     | Baseline Piston testing .....                | 9  |
| 3.1   | Rigid Dynamics Study .....                   | 9  |
| 3.2   | Steady state thermal study .....             | 9  |
| 3.3   | Full linked study / Static Structural .....  | 10 |
| 3.4   | Standard Piston Results .....                | 12 |
| 4     | Piston Modification.....                     | 13 |
| 4.1   | Initial Modification.....                    | 13 |
| 4.2   | Rigid Dynamics Study.....                    | 14 |
| 4.3   | Steady State Thermal Study.....              | 15 |
| 4.4   | Fully Linked System / Static Structural..... | 16 |
| 4.5   | Modified piston results .....                | 18 |
| 4.6   | Further Work.....                            | 18 |
| 5     | CAE and its Limitations.....                 | 19 |
| 6     | Conclusion.....                              | 20 |
| 7     | References .....                             | 21 |

# 1 Introduction

Computer-Aided Engineering (CAE) tools have transformed the engineering landscape by enabling precise design, analysis, and optimization. This report assesses the CAE process, its limitations, and practical implementation through the simulation of a real-world system. The main objective is to model an engine assembly comprised of a crankshaft, connecting rod, gudgeon pin and piston, and then to perform dynamic, static structural, and thermal analyses aiming to optimise the piston while considering these effects. Validation studies and mesh sizing evaluations will be completed to verify the investigation methods and will ensure software reliability. This study exemplifies how CAE tools enhance engineering design and addresses their constraints.

## 1.1 Methodology

The process of this study is shown below.

- Complete analytical validation cases to verify experimental methods.
- Complete mesh sizing study on piston model to aid selection of suitable mesh sizing.
- Complete baseline fully linked structural analysis of supplied piston model.
- Modify and improve piston model, to reduce mass and improve structural integrity.
- Complete a fully linked structural analysis of the new improved pistons design.
- Compare results, noting any strengths or limitations of the testing methods, conclude.

## 2 Validation Study

The validation study was split into two sections, the first being theoretical cases to compare with calculated results from analytical methods, and the second being a mesh independence study to aid testing method and mesh sizing selection.

### 2.1 Analytical validation

Two validation cases were completed, those being a cantilever beam with a uniformly distributed load (UDL) to validate static structural results, and a Bimetallic strip with a thermal load to validate thermal loading results. The specification for both validation examples was supplied by Dlubal Software, a developer of engineering programs for FEA, structural analysis and design for over 30 years (Dlubal, n.d.).

#### 2.1.1 Cantilever beam UDL

##### 2.1.1.1 Analytical

The specification for the components of the cantilever beam with UDL is shown below.

Table 1: Cantilever beam specification table (Dlubal, n.d.)

|          |                 |                       |       |            |     |
|----------|-----------------|-----------------------|-------|------------|-----|
| Material | Elastic-Plastic | Modulus of Elasticity | $E$   | 210000.000 | MPa |
|          |                 | Poisson's Ratio       | $\nu$ | 0.000      |     |
|          |                 | Shear Modulus         | $G$   | 105000.000 | MPa |
| Geometry | Plate           | Length                | $L$   | 1.000      | m   |
|          |                 | Width                 | $w$   | 0.050      | m   |
|          |                 | Thickness             | $t$   | 0.005      | m   |
| Load     | Plate           | Pressure              | $p$   | 2750.000   | Pa  |

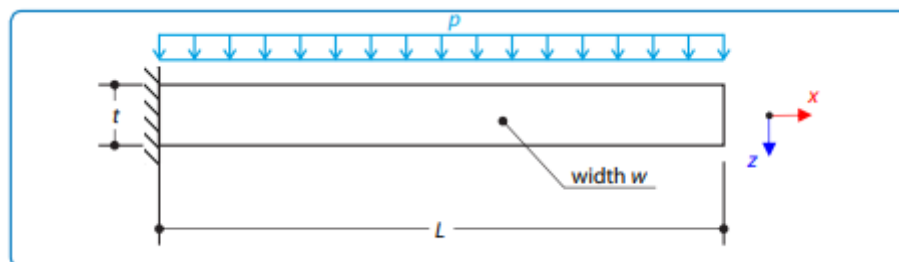


Figure 1: Cantilever beam with UDL Diagram (Dlubal, n.d.)

Table 1 shows the material specification and part geometry for the cantilever beam, and figure 1 shows a sketch of the simulated experiment. The beam was modelled using Solidworks and then imported into Ansys Workbench 2021 to be used in a static structural study, as well as specifying the material properties within the Engineering data section.

This study calculates the max deflection using the Euler-Bernoulli theory, as show below.

$$El_y \frac{d^2 u_2}{dx^2} = M$$

Finding the second integral of this equation, and provided maximum displacement occurs at  $x = L$ , gives the following equation.

$$u_{z,max} = \frac{ql^4}{8EI_y} = \frac{3pL^4}{2Ew^2t} = 157.143mm$$

As shown the max deflection when calculated analytically is found to be 157.143mm

#### 2.1.1.2 Simulated

Due to the dimensions of the model, a mesh size of 2.5mm was selected so that the thinnest face of the model, at 5mm contained 2 elements. Once meshed, and taken into a static structural study, one end of the beam was fixed with a fixed support and the UDL of 2750 Pa was applied on the top face of the beam. Figure 2 shows the results of this test.

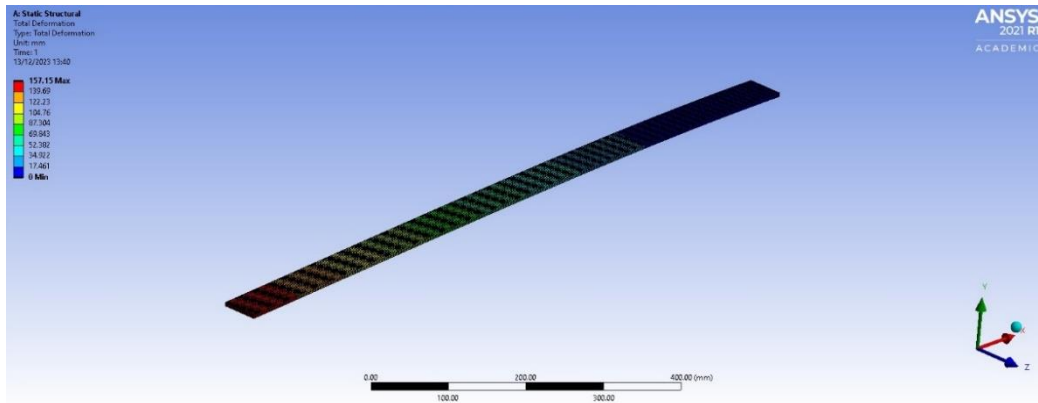


Figure 2: Cantilever beam with UDL static structural solver

The simulated study provided a max deflection of 157.15mm, when compared with the analytical solution of 157.143mm a percentage error of 0.00445% is present. FEA solutions state that FEA results should lie within 5% of hand calculation results (FEA Solutions, 2023), therefore this percentage error is perfectly acceptable and within tolerance.

## 2.1.2 Bimetallic Strip Thermal Load

### 2.1.2.1 Analytical

The specification for the components of the Bimetallic strip is shown below.

Table 2: Bimetallic strip specification table (Dlupal, n.d.)

|          |        |                                  |                 |                        |                         |
|----------|--------|----------------------------------|-----------------|------------------------|-------------------------|
| Material | Invar  | Modulus of Elasticity            | $E_i$           | 137000.000             | MPa                     |
|          |        | Poisson's Ratio                  | $\nu_i$         | 0.280                  | —                       |
|          |        | Coefficient of Thermal Expansion | $\alpha_i$      | $1.200 \times 10^{-6}$ | $^{\circ}\text{C}^{-1}$ |
|          | Copper | Modulus of Elasticity            | $E_c$           | 130000.000             | MPa                     |
|          |        | Poisson's Ratio                  | $\nu_c$         | 0.354                  | —                       |
|          |        | Coefficient of Thermal Expansion | $\alpha_c$      | $2.000 \times 10^{-5}$ | $^{\circ}\text{C}^{-1}$ |
| Geometry |        | Cross-section Width              | $w$             | 5.000                  | mm                      |
|          |        | Layer Thickness                  | $t_i = t_c = t$ | 1.000                  | mm                      |
|          |        | Length                           | $L$             | 100.000                | mm                      |
| Load     |        | Thermal Loading                  | $T_c$           | 100.000                | $^{\circ}\text{C}$      |

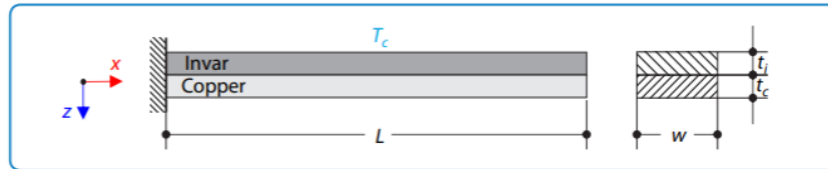


Figure 3: Bimetallic strip Thermal loading diagram (Dlupal, n.d.)

Table 2 shows the material specification and part geometry for the cantilever beam, and figure 3 shows a sketch of the simulated experiment. As before, the strip was modelled using Solidworks and into Ansys Workbench 2021 to be used in a static structural study with a thermal loading, as well as specifying the material properties within the Engineering data section.

This study calculates the maximum deflection using the dimensionless coefficient (K1)

$$K_1 = 4 + 6 \frac{t_c}{t_i} + 4 \left( \frac{t_c}{t_i} \right)^2 + \frac{E_c}{E_i} \left( \frac{t_c}{t_i} \right)^3 + \frac{E_i t_i}{E_c t_c}$$

The maximum deflection of the strip is then defined by the following formula:

$$u_{z,max} = \frac{6(a_i - a_c)(t_i + t_c)T_c L^2}{t_s^2 K_1} \approx -7.049mm$$

As shown the max deflection when calculated analytically is found to be 7.049mm.

#### 2.1.2.2 Simulated

Due to the dimensions of the model, a mesh size of 0.5mm was selected so that the thinnest face of each strip, at 1mm contained 2 elements. Once meshed, and taken into a static structural study, one end of the strip was fixed with a fixed support and the thermal loading of 100°C above ambient (22°C) was applied to the strip. Figure 4 shows the results of this test.

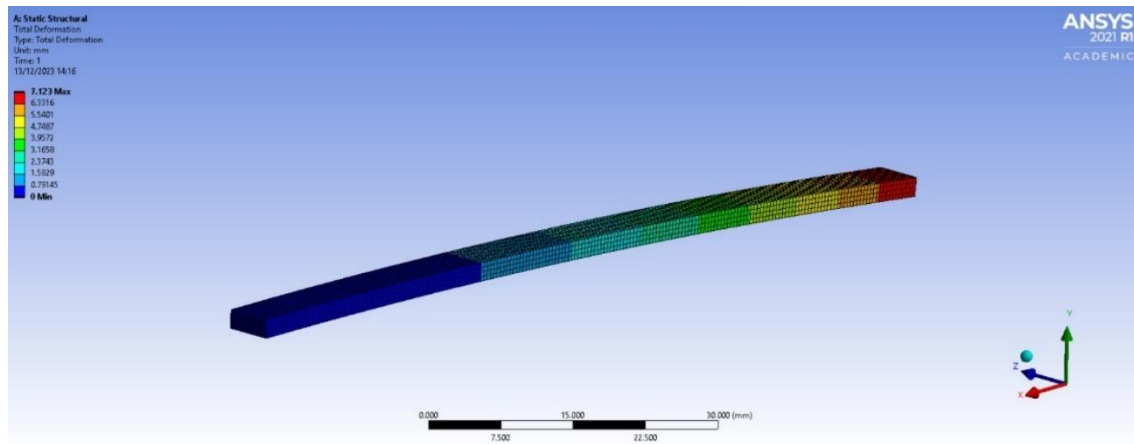


Figure 4: Bimetallic strip with thermal loading diagram

The simulated study provided a max deflection of 7.123mm, when compared with the analytical solution of 7.049mm a percentage error of 1.049% is present. Although this percentage error is still low, it is much higher than the cantilever beam example so to investigate this some different mesh sizes were tested using the same method, which includes 1mm, 0.25mm and 0.1mm element sizes so see if they had any effect on the percentage error. Figure 5 shows these results.

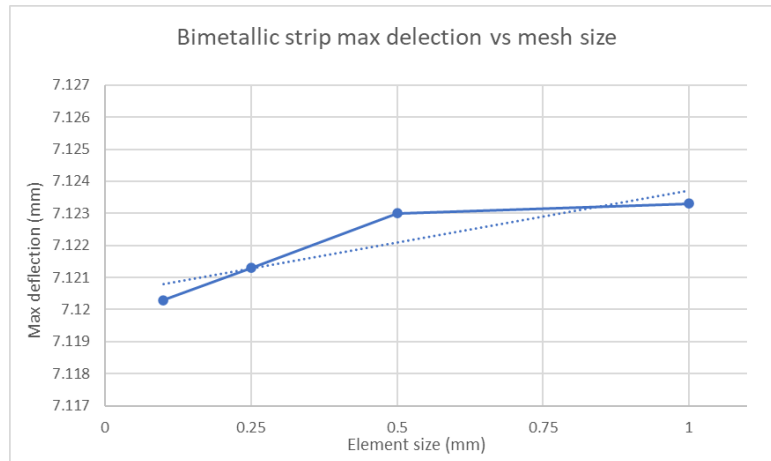


Figure 5: Bimetallic strip Element size vs max deflection

As seen in figure 5 reducing the mesh element size in this case does start to trend towards the analytical solution, however the at 0.1mm element size which would be unusable to use for the piston study due to the high time and computing power demands this would have.

## 2.2 Mesh Independence study

To further validate the chosen methods before drawing any conclusions, a mesh independence study was completed using the standard unmodified piston model, tested through a range of mesh size to view the effect on max deflection and min deflection when run in the same loading conditions.

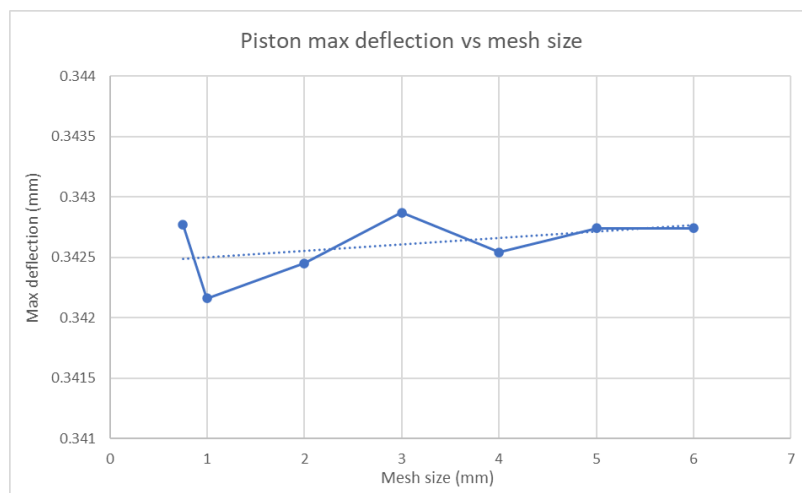


Figure 6: Piston Element size vs max deflection

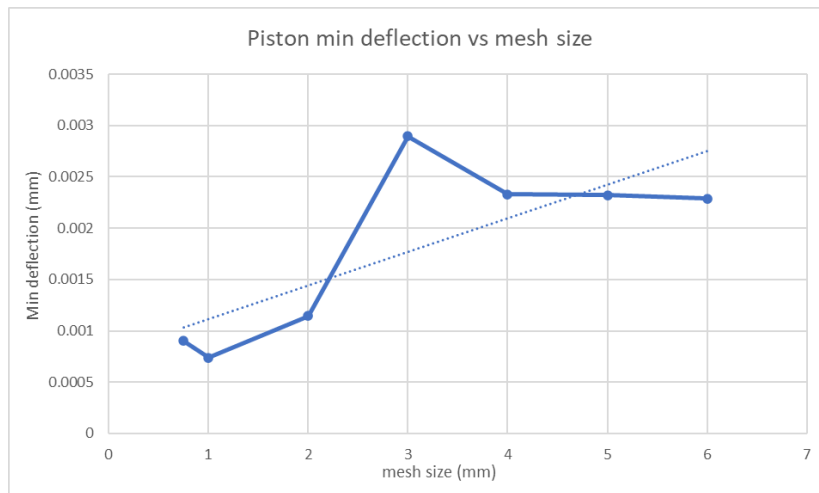


Figure 7: Piston element size vs min deflection

Figures 6 and 7 show the max and min deflection when altering the mesh element size on the standard piston model, as can be seen the max deflection has little variance with the change in mesh size with only 0.207% between the largest and smallest values, whereas the min deflection varied more with a 104.885% difference between largest and smallest values, this is likely due to the smaller element sizes able to locate areas of very low deflection that were missed with larger elements. Looking at these results the mesh element size chosen was 2mm with 1mm for local sizing as this allows for areas of minimum deflection to better be located and also provides a consistent max deflection.

### 3 Baseline Piston testing

To analyse and modify the existing piston geometry, the motion loads, deflection, and stress must be calculated to understand how changes made affect these factors.

#### 3.1 Rigid Dynamics Study

The first study completed involved calculating the forces and moments acting on the piston during an engine cycle at a given RPM, in this case 6000RPM was used as it provides a reasonable representation of engine characteristics at higher engine speed for a small naturally aspirated petrol engine, such as the Ford Sigma. Figure 8 shows the result from this section, with a total force at the gudgeon pin to piston joint of 6899.3N.

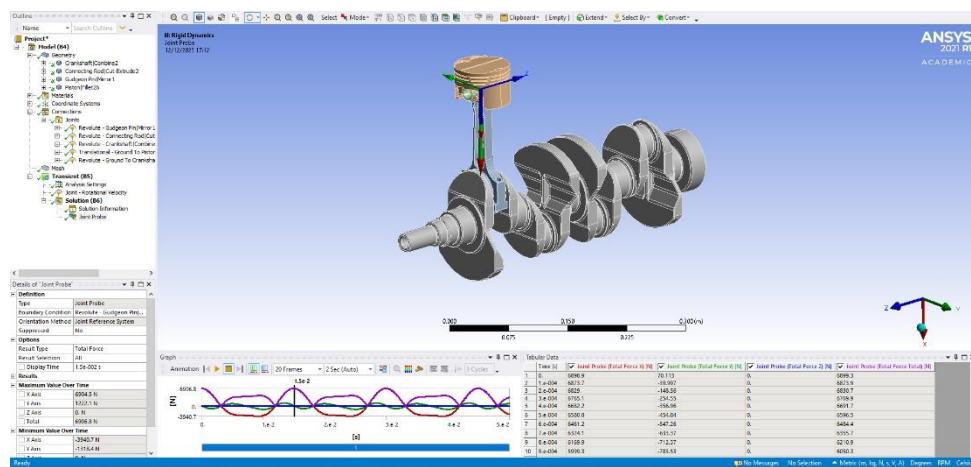


Figure 8: Baseline piston in rigid dynamics solver

#### 3.2 Steady state thermal study

Next the heating conditions that the piston experiences were applied and visualised, each section of the pistons heating coefficients and expected temperatures were applied, and a mesh generated. Using the mesh independence study as reference a 2mm mesh was applied and a 1mm sizing applied to the piston crown where the largest temperature is experienced. Figures 9 and 10 show the mesh and results from this study.

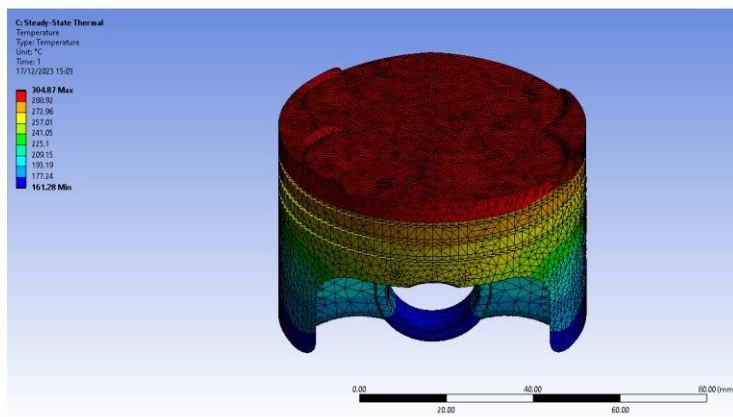


Figure 9: Baseline piston max temperature top view

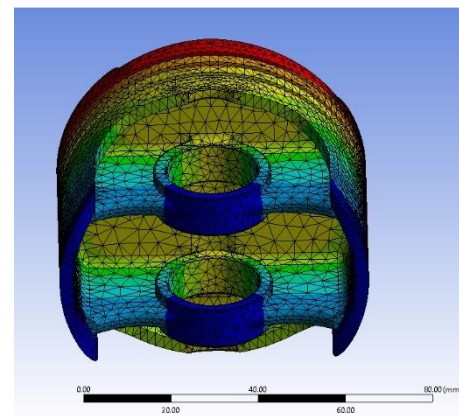


Figure 10: Baseline piston max temperature bottom view

In standard specification the piston is shown to experience a maximum thermal loading of 304.87°C, peaking across the crown surface as expected, with temperatures decreasing as distance away from the combustion chamber increases.

### 3.3 Full linked study / Static Structural

With rigid dynamics and steady state thermal results collected the static structural study can be completed, combining all forces and loadings experienced by the piston to show the overall deformation, stress and strain when under normal operating conditions at 6000RPM. A force was also applied to the top surface of the piston to simulate the cylinder pressure at this RPM, in this case 100.067 bar which is measured Ford Sigma 1.6L cylinder pressure at 6000RPM supplied by Dr Owen Williams. The results from this linked study are shown below in figures 11 and 12.

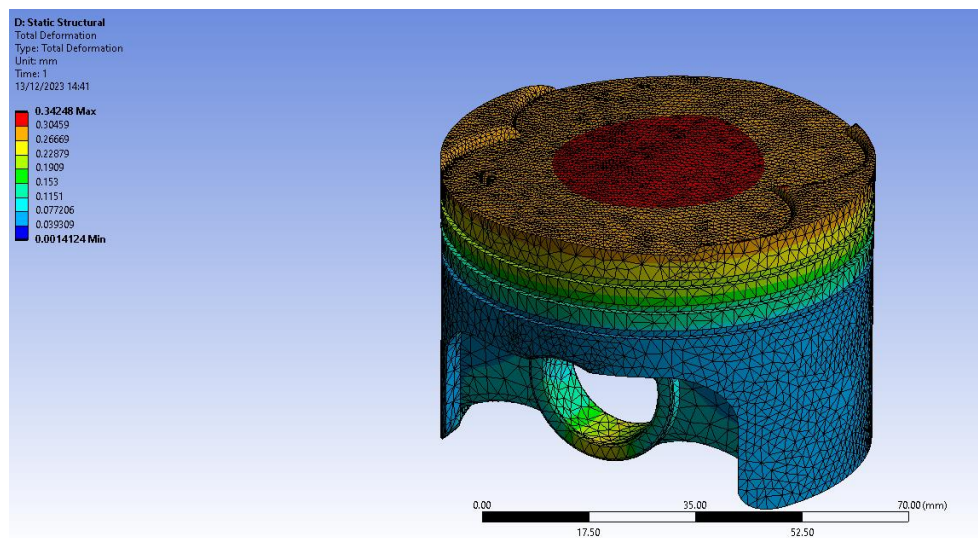


Figure 11: Baseline piston max deflection top view

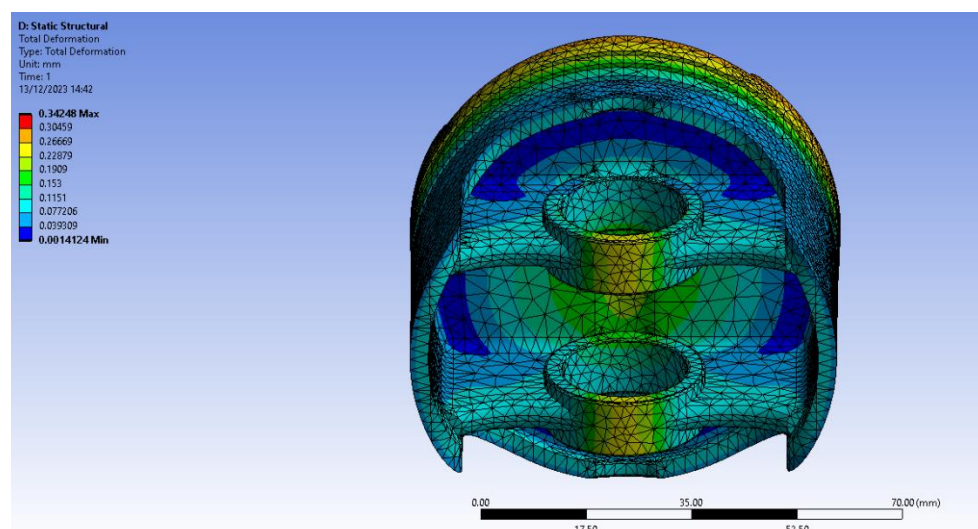


Figure 12: Baseline piston max deformation bottom view

Figures 11 and 12 show the areas of high and low deformation on the standard piston, the maximum deformation is stated to be 0.34248mm, which will aim to be improved on the modified piston. This high deformation area is the centre of the piston crown, with a deformation between 0.27mm and 0.30mm shown on the rest of the crown and also on the bottom of the gudgeon pin boss, this is due to force under tension when the piston reaches TDC and then returns down the cylinder. Another useful conclusion from this data is the areas that experienced very little deflection or stress could be removed or reduced in order to reduce piston mass, such as areas around the skirt and also the underside.

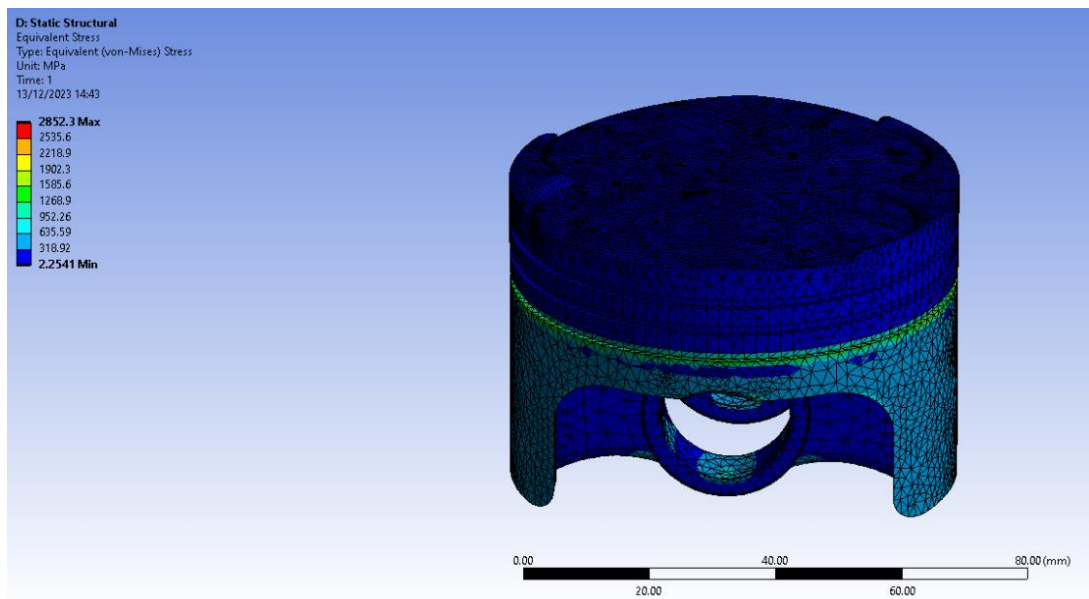


Figure 13: Baseline piston equivalent stress top view

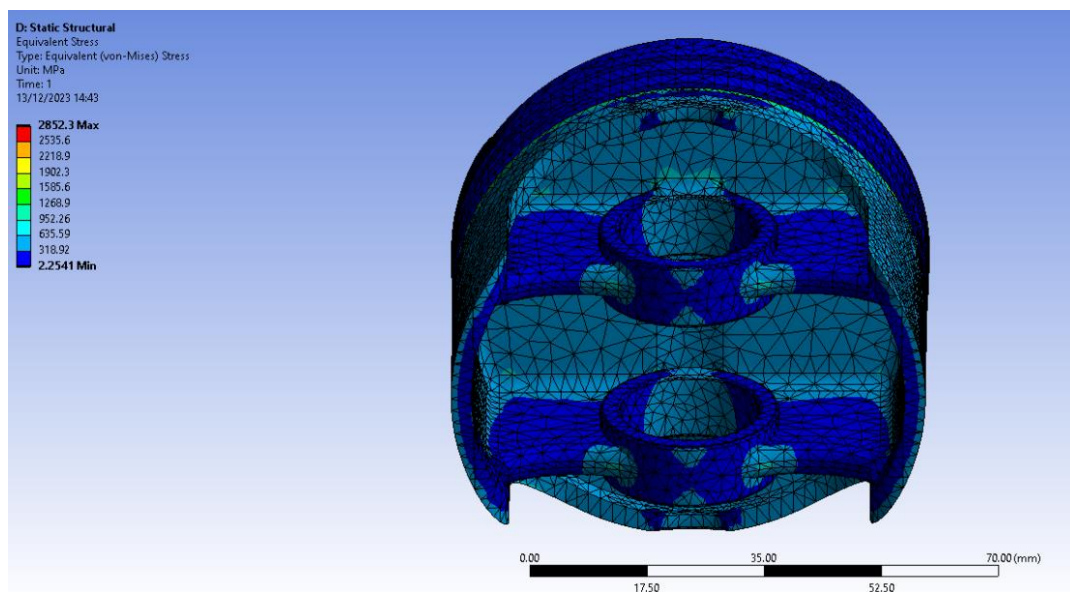


Figure 14: Baseline piston equivalent stress bottom view

Figures 13 and 14 show the stress experienced by the piston under the given conditions, with a maximum value of 2852.3 MPa, experienced around the bottom oil groove.

### 3.4 Standard Piston Results

The main points to consider after fully testing the piston in standard configuration, is that reducing the mass would have large benefits to both performance and reliability. Reducing the piston mass would reduce overall reciprocating mass within the engine allowing it to rev faster therefore improving responsiveness and performance, also reducing weight of reciprocating components has a direct effect upon the magnitude of unbalance in the engine, so lightening the pistons has great influence towards vibration reduction (Bull, A.A 1924). There are also many areas identified on the piston where mass could be removed without negatively affecting the structural integrity of the piston.

## 4 Piston Modification

### 4.1 Initial Modification

With the area of improvement decided as removing mass while not causing any increased deformation of the piston, the existing provided model was modified with this goal in mind. The standard piston when specified with 6061 T6 Aluminium as the material has a mass of 255.77g.



Figure 15: Baseline piston CAD model top



Figure 16: Baseline piston CAD model bottom

Figures 15 and 16 show the piston in stock configuration, with figures 17, 18 and 19 showing the first iteration of piston modification, with a reduced skirt size and large amount of material removed from the underside while maintaining the essential dimensions such as the pin boss and importantly piston crown, and valve cutouts. Alteration to these parts in particular would cause unwanted change in compression ratio and potentially cause catastrophic valve piston interference.

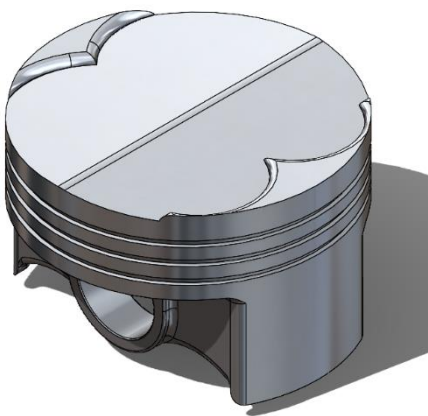


Figure 17: Modified piston top view

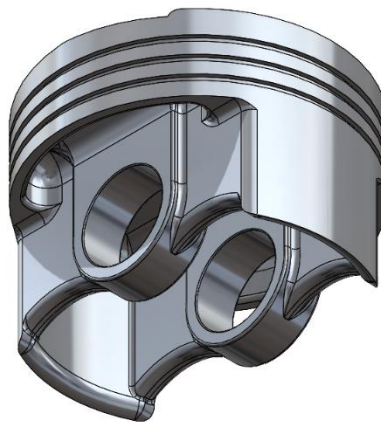


Figure 18: Modified piston bottom iso view

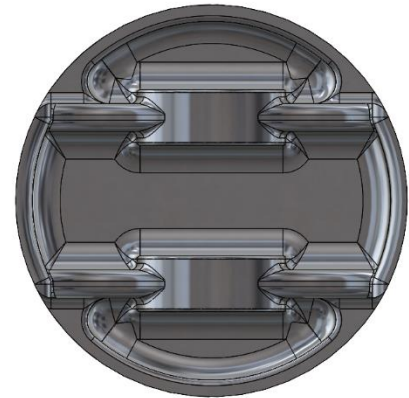


Figure 19: Modified piston bottom view

The piston in this new lightweight configuration weighs 186.7g, a 69.07g or 27.0% decrease in mass compared to the standard 255.77g. This has many benefits including less inertia of the piston, as well as increased horsepower and a faster revolution speed, all of which are benefits of lighter reciprocating parts (Machol, M.R. 1913). This reduction in mass will also cause the piston to experience less force from the motion loads when operating at the same RPM.

## 4.2 Rigid Dynamics Study

To ensure results are comparable and accurate the new piston was tested in an identical manner to the standard piston, starting with a rigid dynamics study to analyse the forces from motion acting on the piston at 6000RPM, the results can be seen in figure 20.

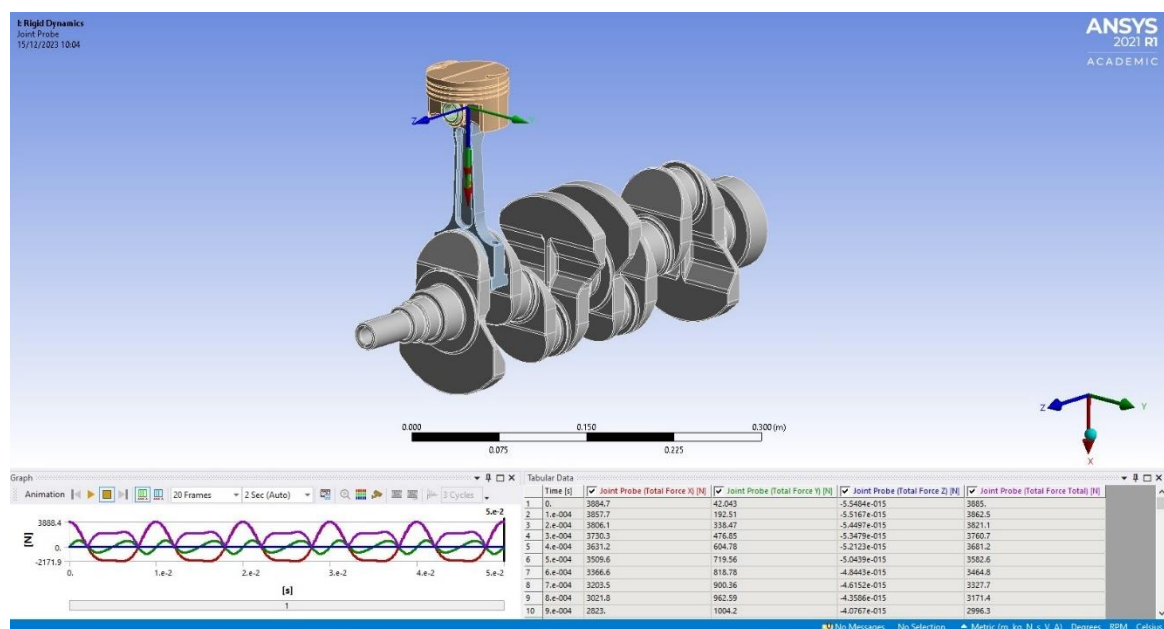


Figure 20: Modified piston in rigid dynamics solver

Figure 20 shows the forces from motion acting on the piston at 6000RPM, peaking at 3885N which when compared to the standard forces of 6899.3N is a 43.7% decrease in force on the piston to gudgeon pin joint. As the gudgeon pin boss has not been modified this will reduce the chance of failure at this point, resulting in a more reliable system.

### 4.3 Steady State Thermal Study

Looking at the thermal results of the modified piston, it becomes clear that although the peak temperature at 294.24°C appears lower than in standard configuration, the area of increased temperature on the underside of the piston is larger. This is due to the removed material causing heat to penetrate to the lower surfaces of the piston more easily.

The material used is 6061 T6 Aluminium, which suffers a loss in strength at temperatures over 150°C with substantial weakening at 200°C (Czerwinski, F. 2020), however this temperature is present for a very short duration during combustion. An aluminium alloy such as this has a melting point of around 600°C, the maximum piston temperature must not exceed 66% of this value (Yao, Z. 2019) and in this case a maximum of 294.24°C is 49.04% of the melting point temperature and therefore suitable for use. Figures 21 and 22 show the results from this study.

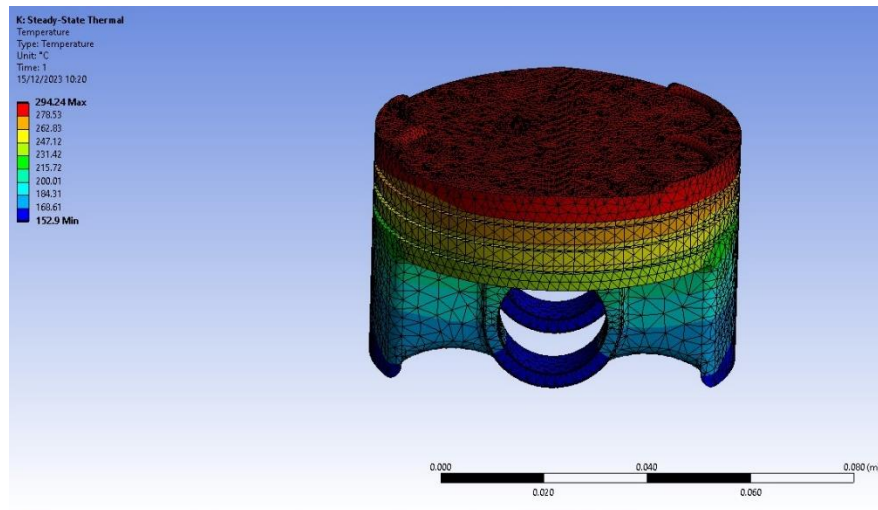


Figure 21: Modified piston in steady state thermal solver top view

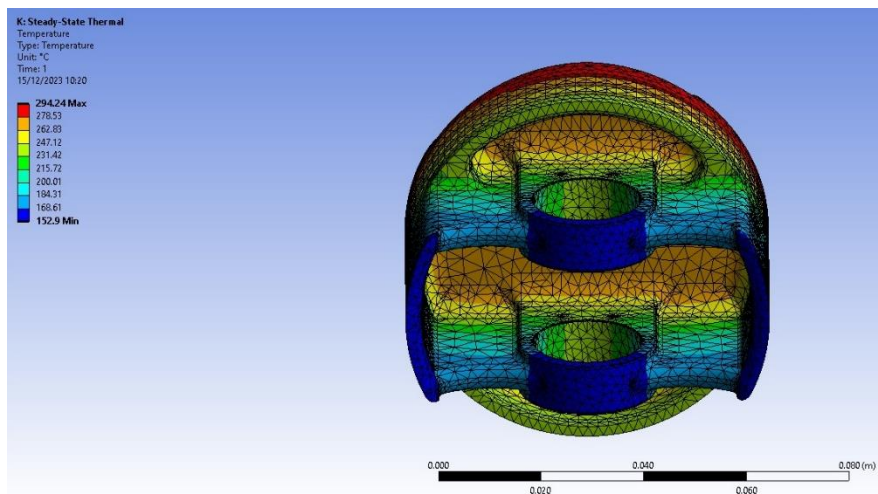


Figure 22: Modified piston in steady state thermal solver bottom view

#### 4.4 Fully Linked System / Static Structural

With all required data collected and implemented into the static structural study, the results for the deformation, stress and strain can be calculated for the modified piston at 6000RPM to be compared against the baseline model. Figures 23 and 24 show the deflection of the piston under these conditions.

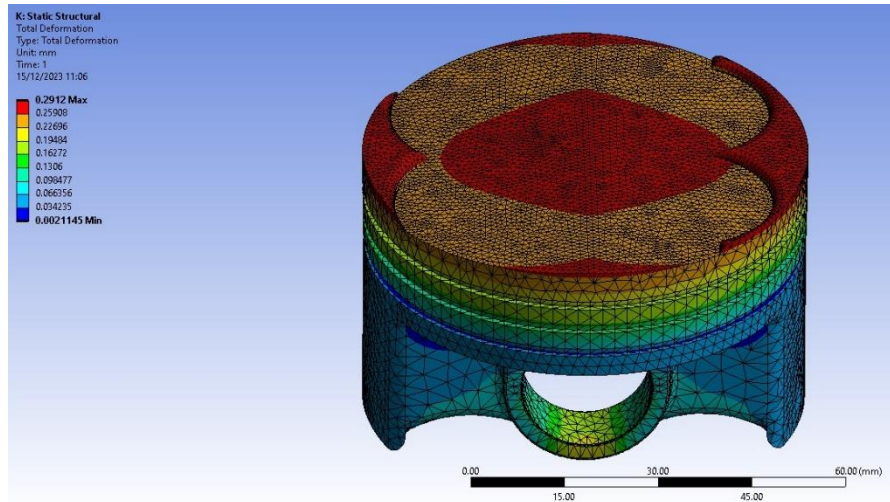


Figure 23: modified piston max deflection top view

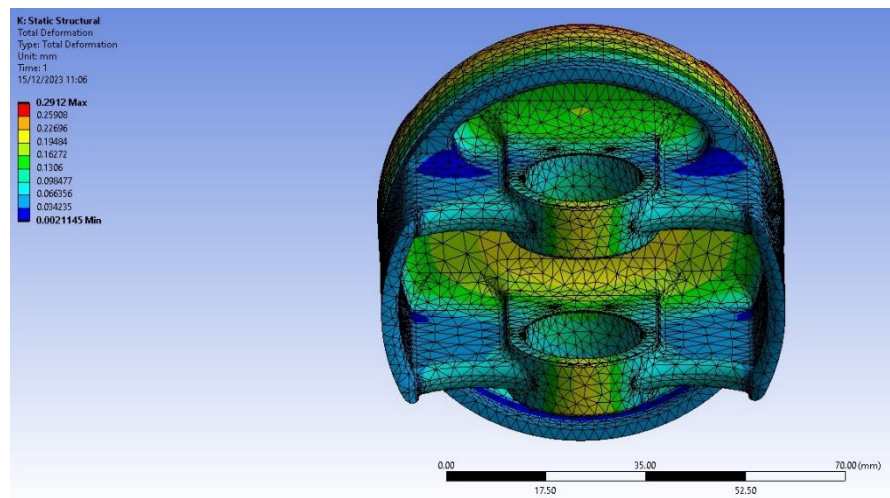


Figure 24: Modified piston max deflection bottom view

Shown in figures 23 and 24 the maximum deflection of the piston is 0.2912mm, 14.97% lower than that of the standard piston. This is likely due to the reduced forces acting on the piston due to the reduced mass, the gudgeon pin boss also shows a reduced deflection when compared to the standard piston. Overall there are no concerning areas of deflection after the modifications, and still some areas shown by the dark blue that experienced very little deflection and therefore could be modified further in future design iterations.

Figures 25 and 26 show the stress on the top and bottom surfaces of the modified piston, with a maximum of 1474.9 MPa which is a 48.64% decrease from standard. This is likely due to the decrease in mass at the top of the piston, and also the reduction in skirt size around the oil groove where stress was peaking in standard configuration.

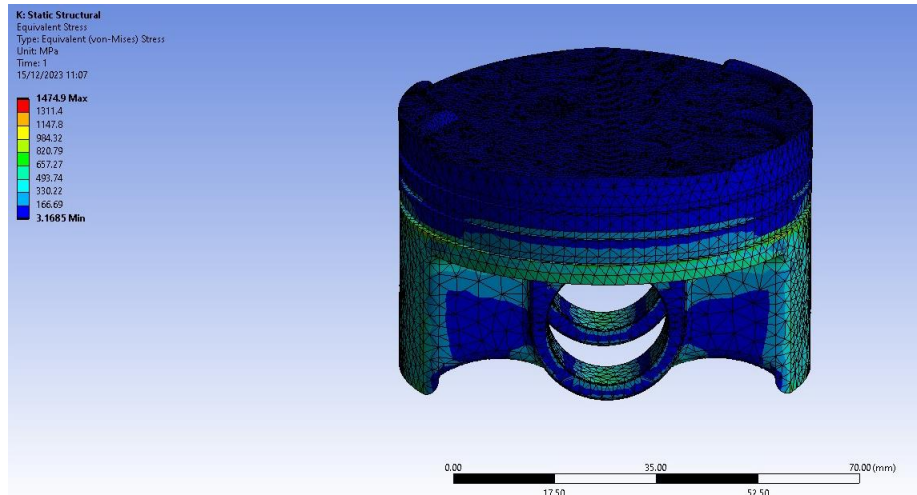


Figure 25: Modified piston equivalent stress top view

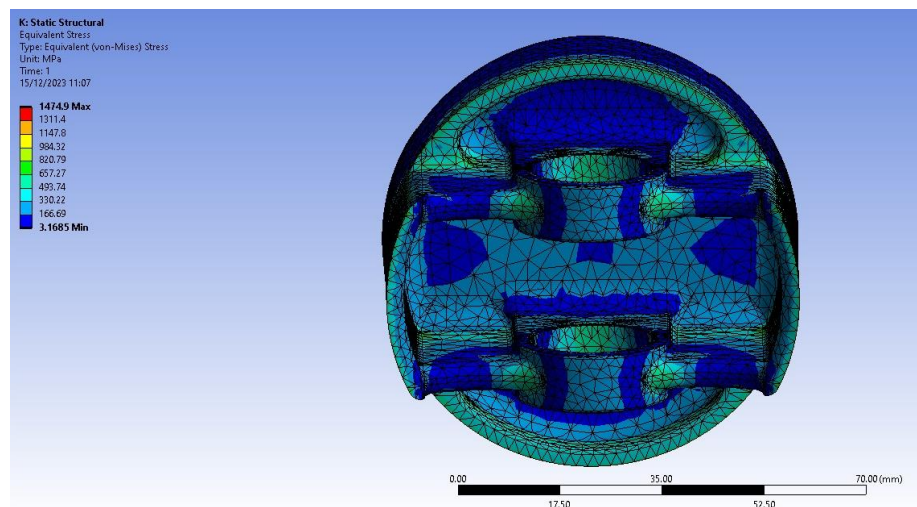


Figure 26: Modified piston equivalent stress bottom view

## 4.5 Modified piston results

The modifications made to the piston gave many positive results, these are shown below.

- 27% reduction in mass.
- 43.7% reduction in forces due to motion.
- 3.5% reduction in max temperature.
- 14.97% reduction in maximum deflection.
- 48.64% reduction in maximum stress.

These factors all improve the characteristics of the piston, reducing reciprocating mass within the 4-cylinder engine by a total of 276.28g whilst maintaining strength required for high RPM scenarios. The reduction in reciprocating mass also reduces the required size for crankshaft counterweights, therefore reducing mass further and reducing problems occurring due to torsional vibration (Cameron, K. 2023).

## 4.6 Further Work

To further this study, a more in-depth analysis would be completed testing the new piston designs at higher engine RPMs to investigate how decreasing mass affects their performance when encountering larger forces due to motion, and also more piston iterations would be created and tested, using topology optimisation to further reduce mass and improve performance of the piston.

## 5 CAE and its Limitations

In the context of piston development, Computer-Aided Engineering (CAE) faces several limitations. The accuracy of CAE simulations largely depends on the fidelity of the models and the quality of input data, with simplifications and assumptions potentially leading to results that might not fully represent real-world conditions. Another significant challenge is accurately simulating the complex behaviour of materials under extreme conditions, such as high temperatures and pressures. These simulations also demand considerable computational resources, which can be costly and time-consuming (TWI global, n.d.), and the use of these systems can require extensive and advanced training to operate (Dassault Systemes n.d.).

Results from CAE also need to be validated and verified against real-world testing, particularly for critical components like engine pistons. CAE may not always accurately predict all possible failure possibilities, especially those involving complex interactions or rare events, which in an engine can be catastrophic. Another factor not taken into consideration in this study is fatigue life of the component, as all the testing completed tested the piston under a set parameter and for one cycle, whereas in an engine the components can fail due to fatigue and also overheating which will be caused by heat build up on the piston rather than a momentary high temperature. The most common causes of piston failures are insufficient cooling, insufficient lubrication, thermal fatigue, and combustion damage, none of which are simulated with the study that has been completed (Ambrozik,T. , Lagowski,P. 2015).

The interaction of a piston with other engine components is another aspect that CAE models might not fully capture, potentially leading to inaccuracies in results. Finally, the complexity of CAE output necessitates skilled interpretation, and misinterpretation of data can lead to incorrect decisions in the design and development process.

Overall, while CAE is a vital tool in engine piston development, these limitations underscore the importance of supplementing CAE with experimental testing and other validation methods to ensure the reliability and accuracy of the design process.

## 6 Conclusion

In conclusion, CAE is a very useful and powerful tool for development within engineering, and this study completed was ultimately successful in optimising a component using a fully linked Dynamic, thermal and structural study to do so, allowing for clear comparisons between a stock and modified component when put in identical load cases. The results showed that reducing the mass of the piston overall had a positive effect on its performance and reliability from the tests completed, however a more in-depth analysis looking particularly at fatigue failures is necessary to truly understand the characteristics of the component under the constant and repetitive loadings it will be subject to in an engine, therefore this study provides a positive start to a full development of a piston however further work is required in order to come to more decisive conclusions.

This study provided a good insight into the benefits and limitations of CAE systems, and how multiple systems can be linked to provide a detailed analysis of a single component acting within a system.

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