



Prifysgol Cymru
Y Drindod Dewi Sant
University of Wales
Trinity Saint David

University of Wales Trinity Saint David

School of Engineering

Application of CAE Processes

Ross Burnside

2110614

Motorsport Engineering (BEng)

15/12/2024

Structural & Fluid Analysis – ACHA6010

Word Count: (3276 Words)

1.0 – Contents:

2.0 – Introduction.

3.0 – Methodology.

3.1 – Kinematic Analysis Methodology.

3.2 – Static Structural Analysis Methodology.

3.3 – Steady-State-Thermal Analysis Methodology.

4.0 – Results Analysis.

4.1 – Kinematic Analysis.

4.2 – Static Structural Analysis.

4.3 – Steady-State-Thermal Analysis.

4.4 – Steady-State-Thermal & Static Structural Combined Analysis.

4.5 – Mesh Independence Study.

5.0 – Discussion & Conclusions:

6.0 – References.

2.0 – Introduction.

“Automotive engine pistons are most frequently made from aluminium alloys, usually with high silicone content to improve stiffness while reducing overall density”.
(McCune, R.C. – 2001).

A piston converts combustion energy into mechanical motion; operating under high pressures, elevated temperatures, and dynamic forces. This makes its structural and thermal performance critical for engine reliability and efficiency.

This study investigates a 6061-T6 aluminium piston, combining “high strength, good workability, and high resistance to corrosion” as well as being “widely available”.
(ASM.matweb.com – 2024).

Finite Element Analysis (FEA) in ANSYS Workbench is used to evaluate the piston’s behaviour under static structural loads, thermal effects, and their combined interactions. The analysis will assess stresses, deformations, and heat distribution to understand piston performance under realistic operating conditions.

This analysis also highlights the impact of boundary conditions, user inputs, and software limitations on the simulation’s accuracy. The results aim to provide insights into operational behaviour and inform potential design improvements for enhanced performance and durability.

3.0 – Methodology.

3.1 – Kinematic Analysis Methodology.

A kinematic analysis (*figure 1*) of the forces transmitted through the gudgeon-pin to the piston in a reciprocating assembly was performed. The study was conducted using ANSYS Rigid Dynamics simulation to determine the forces in the model's respective X, Y, and Z directions and their combined effects under realistic operational conditions.

Components featured in the assembly included a crankshaft, connecting-rod, gudgeon-pin, and piston.

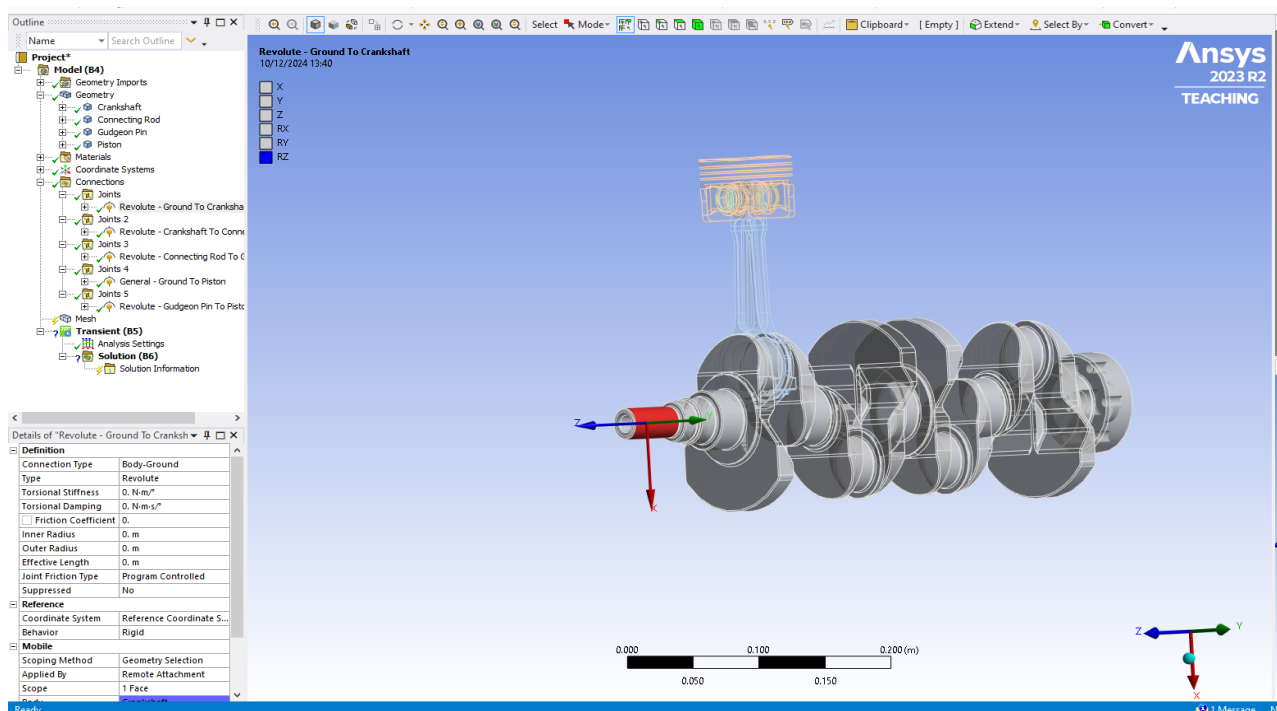


Figure (1); A screenshot from ANSYS Mechanical to show Kinematic Analyses set-up and methodology.

Revolute joints were used to link the crank to the connecting-rod, the rod to the gudgeon-pin, and gudgeon-pin to piston. The piston was defined as free in the X axis (vertical) and constrained in the Y and Z axes, with forces evaluated using a joint probe on the gudgeon-pin/piston interface.

The analysis was conducted under real-world high-speed rotational conditions at 6000RPM, quantifying forces in all three axis directions over a complete cycle. Analysis settings (*figure 2*) were controlled to ensure a clear and manageable amount of data was exported for evaluation.

Details of "Analysis Settings"	
Step Controls	
Number Of Steps	1
Current Step Number	1
Step End Time	3.e-002 s
Auto Time Stepping	Off
Time Step	1.e-005 s
Solver Controls	
Time Integration Type	Program Controlled
Correction Type	Program Controlled
Assembly Type	Program Controlled

Figure (2); Analysis Settings.

3.2 – Static Structural Analysis Methodology.

This analysis assesses the structural response of a 6061-T6 piston under a combustion load applied at 6000 RPM. The study evaluates stress distribution, deformation, and the material's ability to withstand the applied pressure.

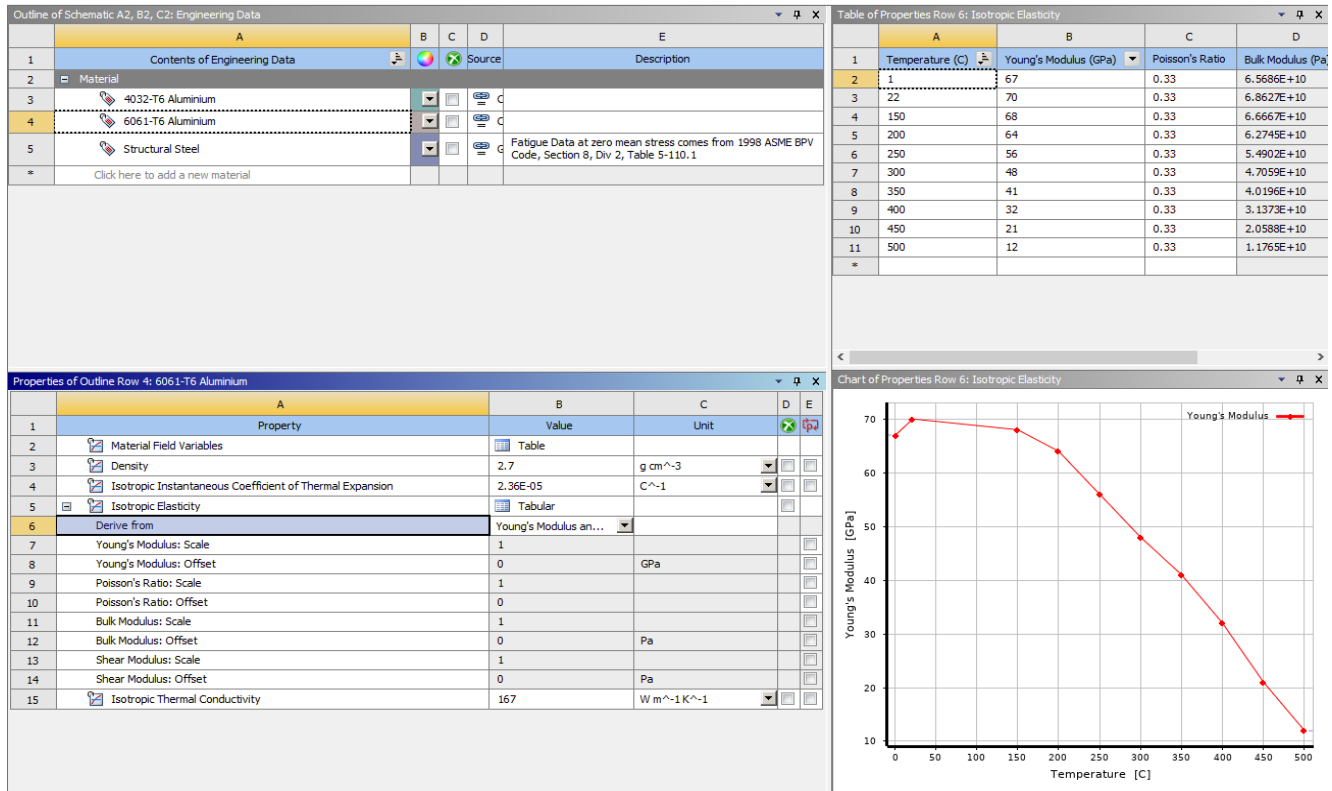
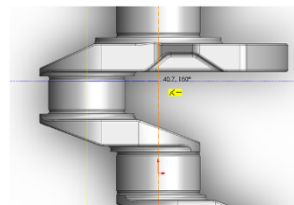


Figure (3); A screenshot to show the input of Young's Modulus over a Temperature Range, alongside the Poisson's Ratio to compute Isotropic Elasticity.

A Static Structural system was added and linked to the standalone Engineering Data system; the piston geometry file was imported applying the required material properties (fig 3) used in this evaluation.

A piston-force-calculator was used to “estimate the force on a piston cylinder system based on the given piston head diameter and cylinder pressure” (Dhari, R. – 2024).

Additional piston parameters (figure 4) were determined through CAD-model access.



$$\text{Stroke} = (2 \times \text{CrankRadius}) = 81.4\text{mm}$$

6061 T6 Piston:
Mass = 256grams
Volume = 94.7cm³

Figure (4); Piston parameters.

Bore diameter ...

70.74 mm

Bore area ...

3,930 mm²

Cylinder pressure ...

4,000,000 Pa

Piston force ...

15,721 N

Share result Reload calculator Clear all changes

Figure (5); Piston Force Calculator. (Dhari, R. – 2024).

Boundary conditions included a static pressure of 2.96MPa, calculated from a realistic combustion force of 15,720N and distributed over the piston crown; in addition to cylindrical supports (*figure 6*) applied to the gudgeon-pinholes, preventing motion in radial and axial directions while maintaining tangential freedom to simulate realistic constraints.

$$F_{combustion} = P_{cylinder} \times A_{crown}$$

$$Piston_D = 70.74mm = 0.0704m$$

$$A_{crown} = \pi \left(\frac{0.0704}{2} \right)^2 = 0.00393m^2$$

e.g. for a typical pressure of 4MPa:

$$F_{combustion} = 4,000,000 \times 0.00393 = 15,720N$$

$$F_{inertia} = m_{piston} \times a_{piston}$$

Max. accel of piston = TDC

$$\omega = \frac{2\pi \times RPM}{60} = \frac{2\pi \times 6000}{60} = 628.32rad/s$$

$$a_{piston} = 0.0407 \times 628.32 = 16,050m/s^2$$

$$F_{inertia} = 0.256 \times 16,050 = 4108.8N$$

$$F_{net} = F_{combustion} - F_{inertia}$$

$$F_{net} = 15,720 - 4108.8 = 11611.2N$$

$$P_{applied} = \frac{F_{net}}{A_{crown}} = \frac{11,611.2}{0.00393} = 2.96MPa$$

The geometry was discretised using a tetrahedral mesh to prepare it for finite element analysis.

Particular attention was give to refining the mesh in critical areas such as the edges of the crown, ring-grooves, and gudegoen-pinholes, where stress concentrations are expected due to the applied load and geometry transitions.

To ensure the accuracy of the results, a mesh independence study was conducted. This involves iteratively refining the mesh until the results for stress and deformation stabilises.

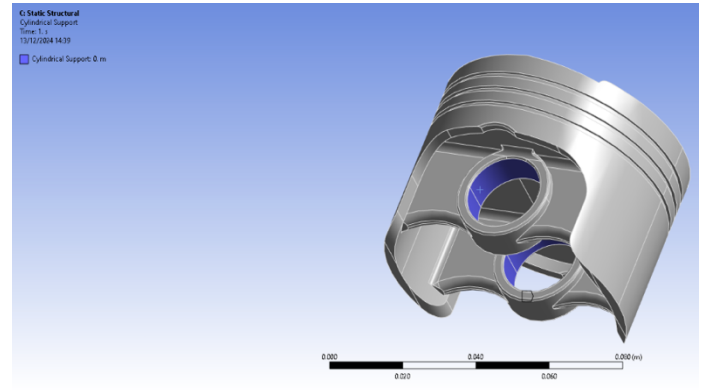


Figure (6); Cylindrical Support applied to pin-holes.

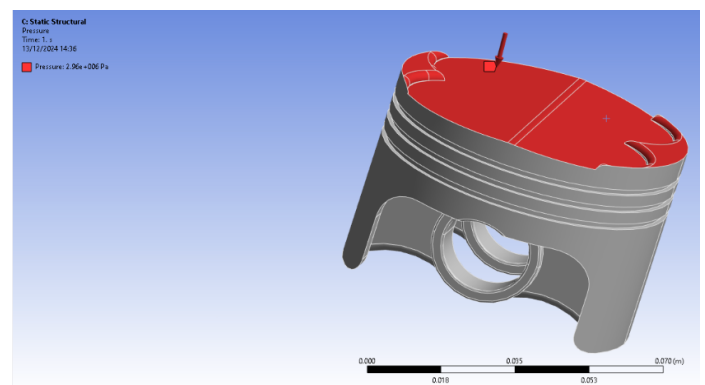


Figure (7); Piston crown showing the applied 2.96MPa pressure.

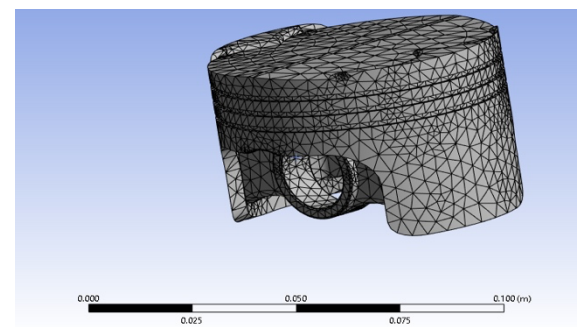


Figure (8); Mesh example.

3.3 – Steady-State-Thermal Analysis Methodology.

The steady-state thermal analysis of the 6061-T6 piston was performed to evaluate temperature distribution across the piston under thermal loads comparable to a combustion engine during operation.

It was noted that for the piston crown, “typical temperature values [...] are [between] 270-350 °C” for a “gasoline engine [utilising] direct injection” (Jaaskelainen, H. – 2017).

Thermal inputs included a fixed crown temperature of 350 °C (figure 9) and convection coefficients (figure 10) applied across the crown and surrounding regions.

Film Coefficient (figure 11) for the Crown surface area was set as 2000W/m² °C, Crown Edge as 1500W/m² °C, 1st Ring-Land Bridge as 1000W/m² °C, with values then descending as 800, 600, 450, and 200W/m² °C.

Although there is no “fixed operating temperature for all engines [...] we can assume that the operating temperature is [...] between 80 and 120 °C” (Bartoszewicz, B – 2024).

This ensured ambient temperature was set as 100 °C for each convection used during the analysis.

Figure 12 shows results for the piston’s temperature distribution, total and directional heat flux, including temperature probes across the piston’s body to ensure clear heat distribution and effect.

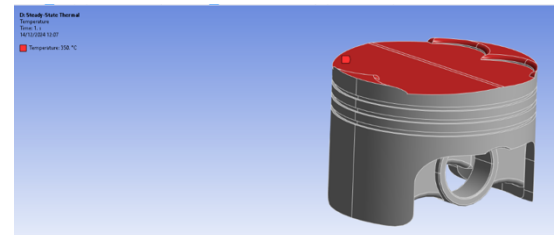


Figure (9); Thermal load application area.

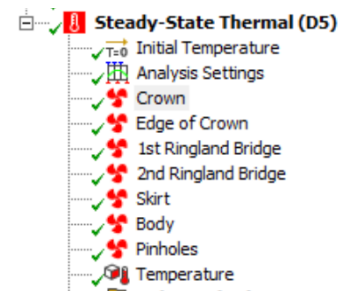


Figure (10); Convection Coefficients.

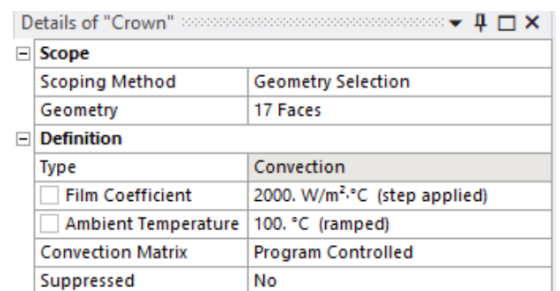


Figure (11); Details of “Crown” Convection.

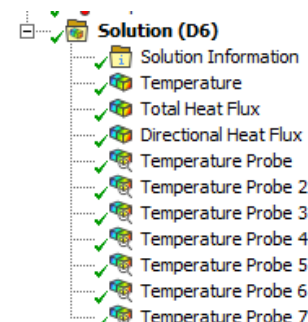


Figure (12); Solution-Design-Tree from ANSYS Steady-State-Thermal System.

3.4 – Steady-State-Thermal & Static Structural Combined Analysis Methodology.

The combined thermal and mechanical analysis of the 6061-T6 piston was conducted by applying the same combustion pressure of 2.96MPa as in the initial structural analysis, incorporating thermal effects derived from the steady-state thermal simulation.

The results demonstrate how thermal expansion and mechanical loading interact to influence the piston's stress and deformation behaviour.

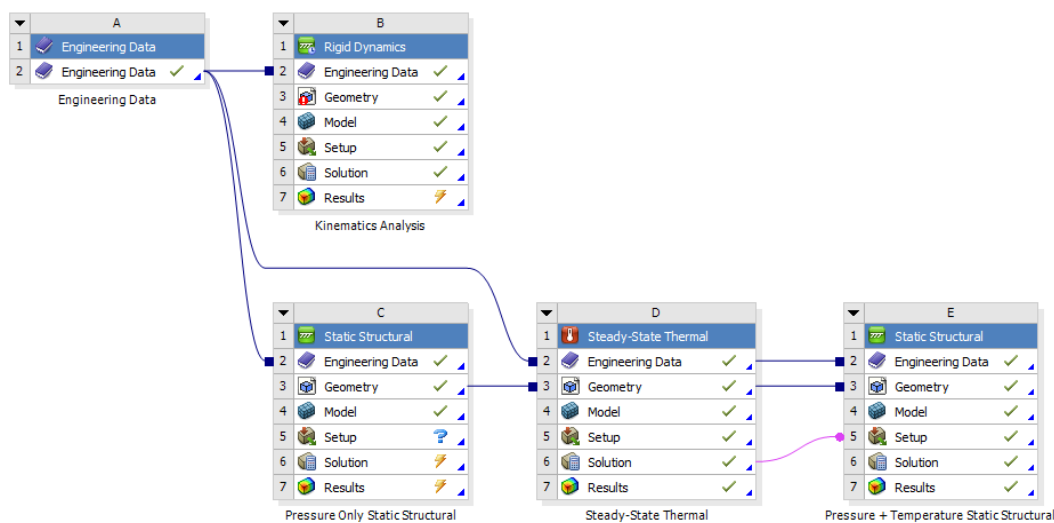


Figure (13); A screenshot from ANSYS Workbench to show simulation models and independent/standalone systems used in this analysis.

The methodology illustrated in figure 13 highlights the systematic workflow for coupling steady-state thermal and static structural simulations.

The results from the steady-state thermal analysis were linked directly to the subsequent static-structural analysis to account for the effects of temperature-induced stresses and deformations alongside the applied combustion pressure.

This integrated approach allows for a comprehensive evaluation of the piston's mechanical performance under combined thermal and mechanical loading, ensuring the simulation closely represents real-world operating conditions.

4.0 – Results Analysis.

4.1 – Kinematic Analysis.

The forces transmitted between the gudgeon-pin and piston were evaluated using ANSYS simulation data. This simulation excludes combustion forces, focusing solely on the inertial effects and dynamic loading due to the reciprocating motion of the piston and the angular oscillation of the connecting rod. The data was analysed along the respective axes, corresponding to the piston stroke direction, lateral (side-to-side) forces, and out-of-plane forces.

The X-axis, reflecting the reciprocating motion of the piston, exhibits the largest forces. These forces range from a maximum of $5329.6N$ to a minimum of $-2941.8N$, oscillating sinusoidally over the cycle. The behaviour is consistent with the primary role of the X-axis in transmitting the piston's motion to the crankshaft via the connecting rod.

In real world conditions, combustion would amplify the downwards forces during the power stroke, significantly increasing the X-axis force magnitudes.

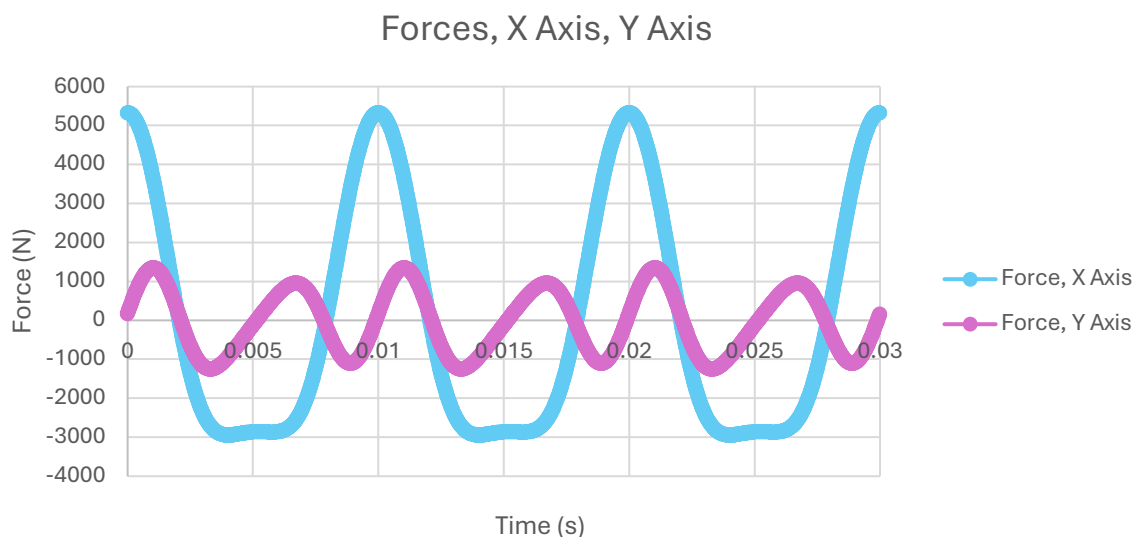
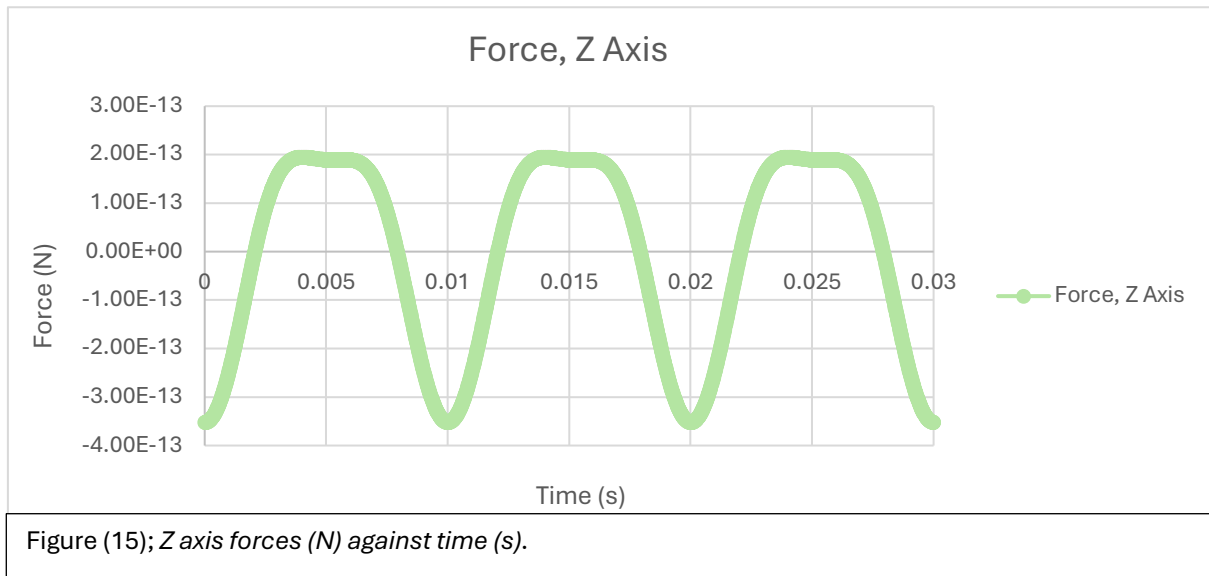


Figure (14); X and Y axis forces (N) against Time (s).

The Y-axis, which is perpendicular to the stroke and captures lateral forces, shows smaller forces ranging from $1347.3N$ to $-1248.1N$. These forces are generated by the angular inclination of the connecting rod as it oscillates through the cycle. These forces are smaller in magnitude than the X-axis forces but still significant, as they cause side-to-side loading on the gudgeon pin.

At mid-stroke position, where the connecting rod reaches its maximum angular offset, Y-axis lateral forces are at their peak. At TDC and BDC, the connecting rod is nearly vertical, and lateral forces approach zero. These Y-axis forces contribute to wear and stress on the gudgeon pin but are not the primary loads.

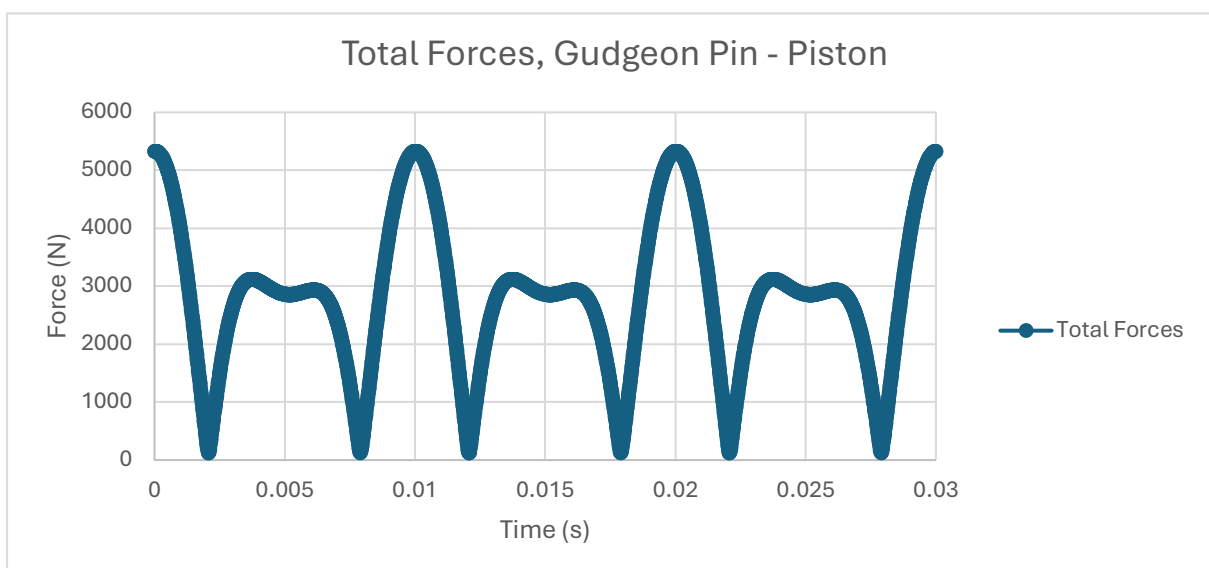
The Z-axis forces, representing out-of-plane dynamics, are the smallest in magnitude. These forces are effectively zero within the limits of the simulation.



This negligible force magnitude confirms that the system is well-constrained in the X-Y plane, and that there are no significant misalignments or vibrations, affecting the gudgeon pin.

The Piston and Gudgeon Pin are constrained to move predominantly in the X-Y plane. Therefore, the magnitude of any 'out-of-plane' forces (Z-axis) are naturally smaller.

The higher forces in the X and Y axes indicate these directions are critical for fatigue analysis. The Gudgeon Pin must withstand cyclic tensile and compressive forces in the X-axis while resisting lateral bending from the Y-axis forces. The smaller Z-axis forces suggest minimal wear or fatigue due to 'out-of-plane' dynamics.



This kinematic analysis shows that forces transmitted between the gudgeon pin and piston are dominated by reciprocating motion along the X-axis. The lateral forces in the Y-axis are secondary but significant, while Z-axis forces are negligible, indicating excellent system alignment.

However, this kinematic analysis is inherently limited as it does not account for the additional forces introduced by combustion; which significantly amplify the axial (X-axis) loading on the piston, gudgeon-pin, and con-rod; particularly during the power stroke when peak cylinder pressures occur.

These forces can induce higher tensile and compressive stresses, which are critical to understanding the true operational loads on the assembly. Furthermore, the thermal stresses from high temperatures during combustion and their combined effects with mechanical loads are not considered. Incorporating these factors would provide a more comprehensive evaluation of the system's performance and durability under real-world operating conditions.

4.2 – Static Structural Analysis.

The static structural analysis of the 6061-T6 piston under a combustion load of 2.96MPa applied to the crown was completed. This analysis evaluates the behaviour of the piston in terms of stress-distribution, total deformation, and the material's response to the applied load.

The Von-Mises stress (*fig. 17*) distribution across the piston reveals several critical regions where stress concentrations are most pronounced. The highest stresses are observed around the edges of the piston crown, the gudgeon pinholes, and ring grooves.

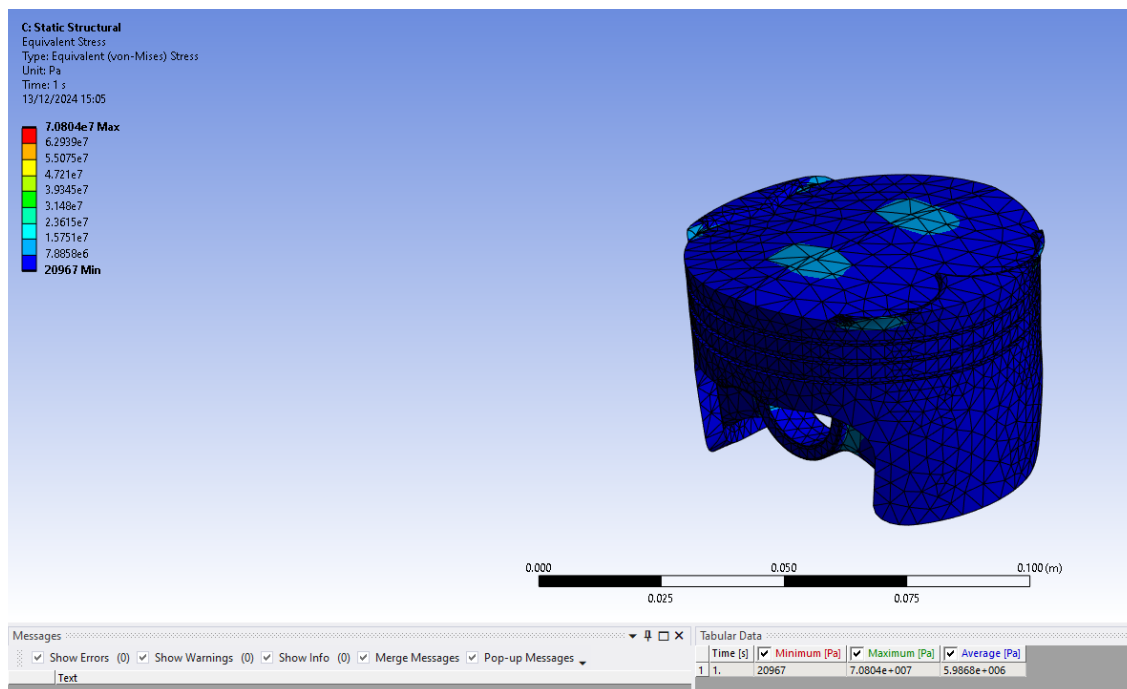


Figure (17); A screenshot to show the Equivalent Stress distributed across the piston.

The maximum stress recorded is 70.804MPa (*figure 17*), which is significantly below the 6061 T6 yield strength of 276MPa, confirming that the piston remains within the elastic deformation range and does not undergo plastic deformation under the applied load.

The stress distribution indicates that the highest stress concentrations occur at the crown edges and gudgeon pinholes (*figure 18*), consistent with the geometry and applied loading conditions.

Most of the piston body experiences relatively low stress levels, as indicated by the predominance of blue regions.

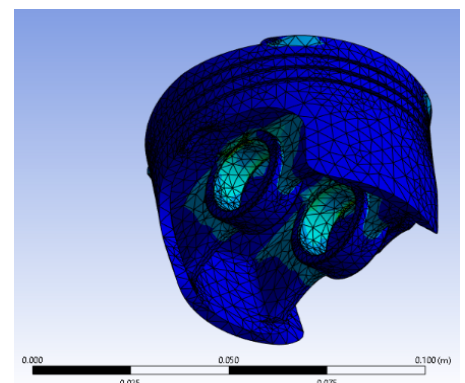


Figure (18); Equivalent Stress.

The observed stress levels suggest that the piston is structurally sound under this loading scenario. However, while the material remains elastic in this static analysis, further investigation into cyclic loading and thermal effects is necessary to fully assess the long-term durability of the piston in real-world operating conditions.

The total deformation results for the piston under the applied static combustion load were evaluated to determine the extent of displacement experienced by the 6061-T6 material under the applied forces.

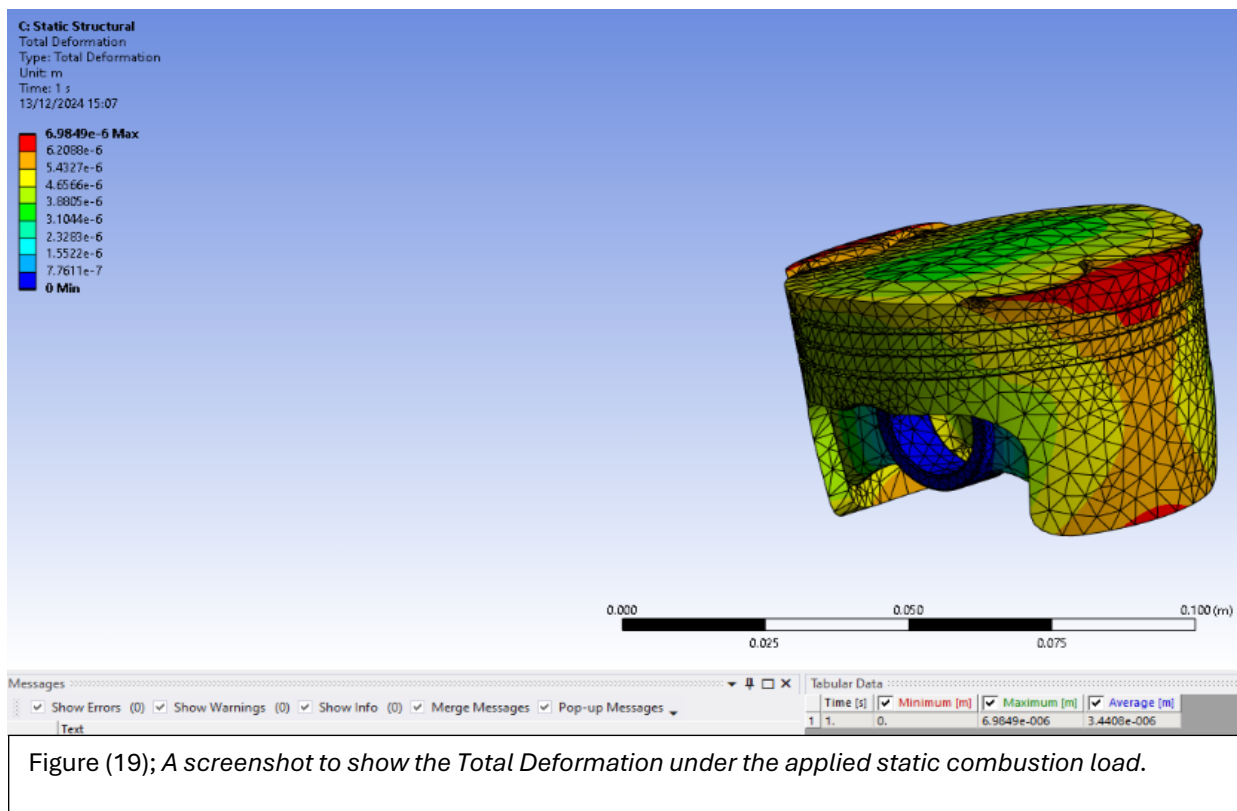


Figure (19); A screenshot to show the Total Deformation under the applied static combustion load.

The analysis shows a maximum deformation of 0.0069849mm, as indicated in the legend of figure 19. This is concentrated near the crown edges, while the remainder of the piston body experiences minimal deformation.

The magnitude of deformation is small, demonstrating that the piston remains highly rigid under the load.

The deformation pattern corresponds closely to the stress distribution observed in the equivalent stress analysis, where regions such as the crown edges and skirt experience slightly greater displacement due to the geometry applied constraints.

The results suggest that the piston remains structurally sound under these static loading conditions. However, as the analysis is static, it does not account for additional effects such as dynamic loading, thermal expansion, or material fatigue; which could amplify deformation in real-world engine conditions.

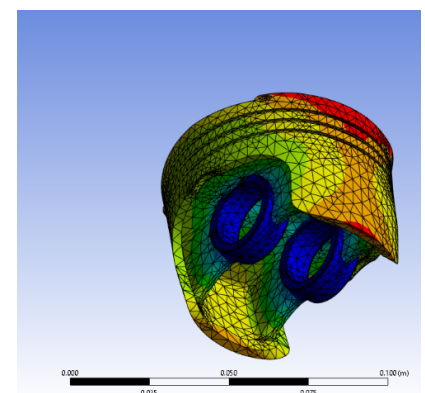


Figure (20); Total Deformation.

4.3 – Steady-State-Thermal Analysis.

The temperature distribution results (*figure 21*) show significant thermal gradients across the piston; with the highest temperatures observed at the crown. This is consistent with the location of direct heat transfer from combustion gasses. The piston crown reached a peak temperature of 350°C , as expected due to the applied fixed temperature.

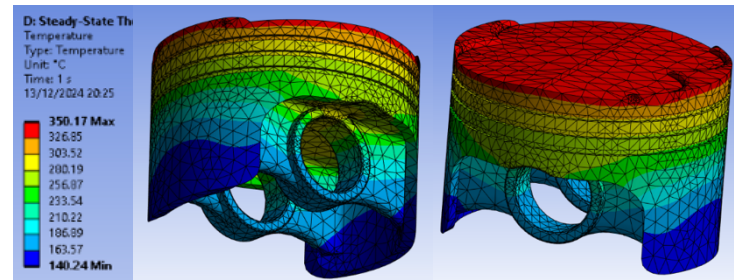


Figure (21); Temperature distribution across the piston.

The thermal gradient reduces as it travels through the ring-lands, into the body, skirt and gudgeon-pin-holes.

Figure 22 shows that the first ring land bridge experiences temperatures around 290°C , while the second recorded approximately 270°C . The gudgeon-pin-hole edges (*figure 23*) and piston skirt demonstrated similar temperatures of 279°C and 268°C .

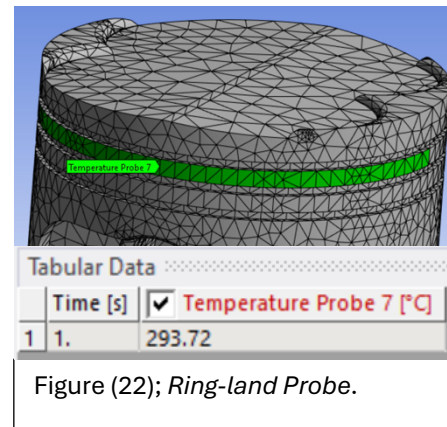


Figure (22); Ring-land Probe.

Reduced temperatures of 178°C in the piston body indicate effective heat dissipation through convection and conduction mechanisms. The thermal distribution results suggest that the piston is subject to significant thermal loading, particularly at the crown, which could potentially impact material properties, such as thermal expansion and stress concentrations in those areas.

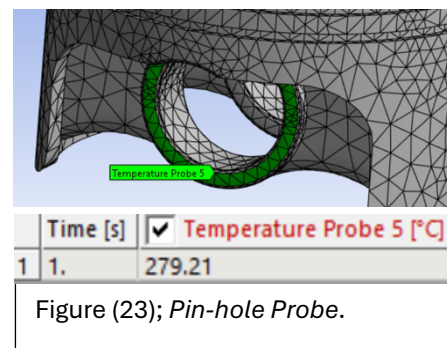


Figure (23); Pin-hole Probe.

The observed thermal gradients emphasise the importance of material selection and adequate cooling mechanisms to maintain the structural integrity of the piston under operational conditions. Further investigation into the combined effects of thermal and mechanical loads is necessary to assess the overall durability and performance of the piston.

The total heat flux results (*figure 24*) indicate the rate of heat transfer throughout the piston due to conduction and convection. The maximum total heat flux value observed is $973,380\text{W/m}^2$, concentrated around the crown and surrounding areas directly exposed to the applied load.

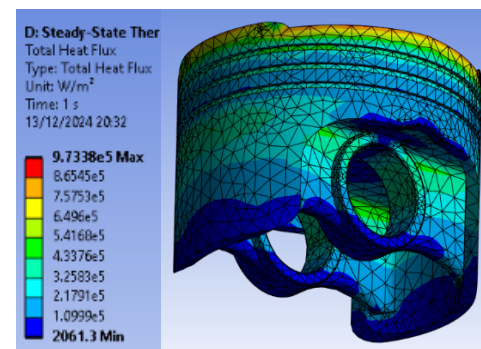


Figure (24); Total Heat Flux.

This high flux corresponds to the temperature gradient between the crown and the cooler areas of the body. The minimum heat flux of 206.13W/m^2 is observed in the skirt and pinhole areas, where the heat transfer is less intense due to their distance from the primary thermal load.

The variation in heat flux distribution highlights the efficiency of heat dissipation through the 6061 material. The high flux near the crown demonstrates thermal energy transfer away from the combustion chamber, whilst lower flux in the skirt indicates reduced conduction due to not interacting with combustion gases directly.

4.4 – Steady-State-Thermal & Static Structural Combined Analysis.

The equivalent stress results for the piston under combined thermal and pressure loading (*figure 25*) indicate a maximum stress of 2.2615GPa (2261.5MPa), as observed in the tabular data and stress contours.

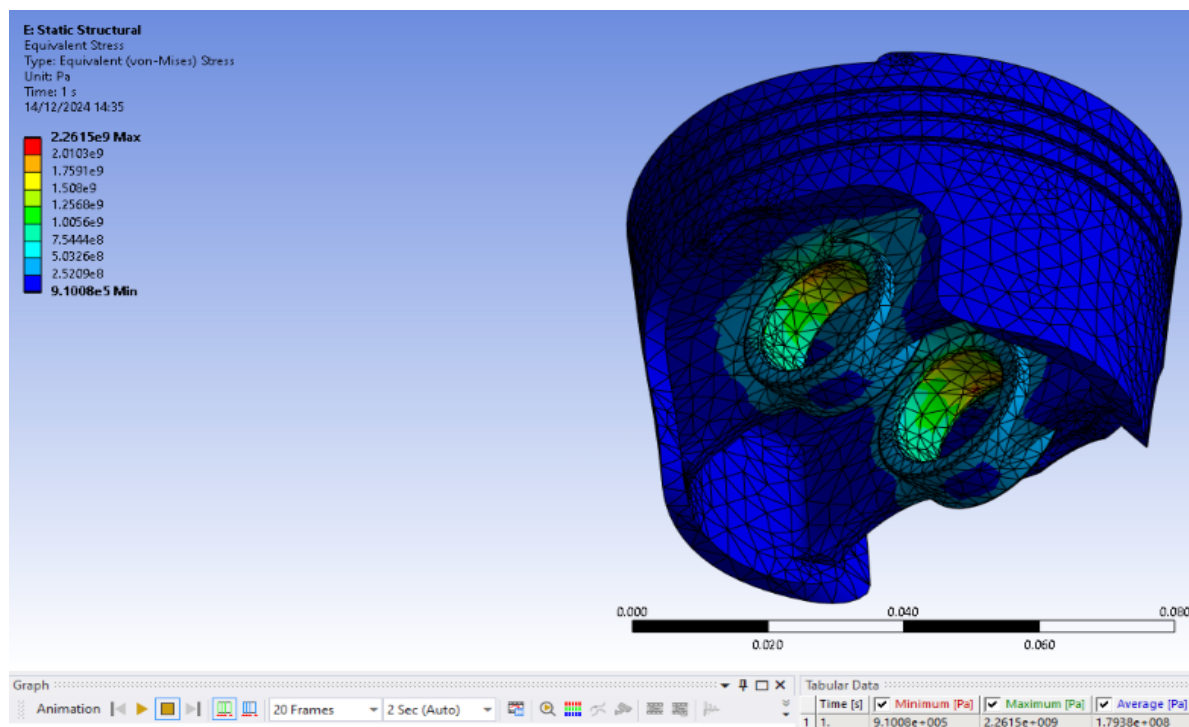


Figure (25); Combined Steady-State-Thermal and Static-Structural Loadings.

Despite the knowledge that “the greatest stress appears on the upper end of the piston pin seat and this stress concentration is one [...] reason for fatigue failure” (Kamble, V.B. – 2018), the positioning and magnitude of this stress raised questions regarding the reliability of the simulation.

The application of cylindrical supports at the pinholes may not adequately replicate realistic support conditions during operation. In a real-world system, forces are translated through movement and into the connecting rod and crankshaft; as opposed to being localised to the gudgeon pinholes. This may result in exaggerated stress values that are specific to the constraints in this model.

In addition to this, the accuracy of results is highly dependent on the input and calculations made during model set-up. Potential inaccuracies in applying thermal load distribution, pressure magnitudes, or material properties could amplify discrepancies in the observed stresses. These factors highlight the limitations of the software, which relies on the accuracy of user-defined inputs to produce meaningful results.

The total deformation of the piston under combined loading reaches a maximum of 0.009811mm, representing a slight increase compared to the 0.006985mm deformation under pressure-only conditions. The deformation is primarily concentrated near the piston crown and ring-lands, where thermal gradients are highest.

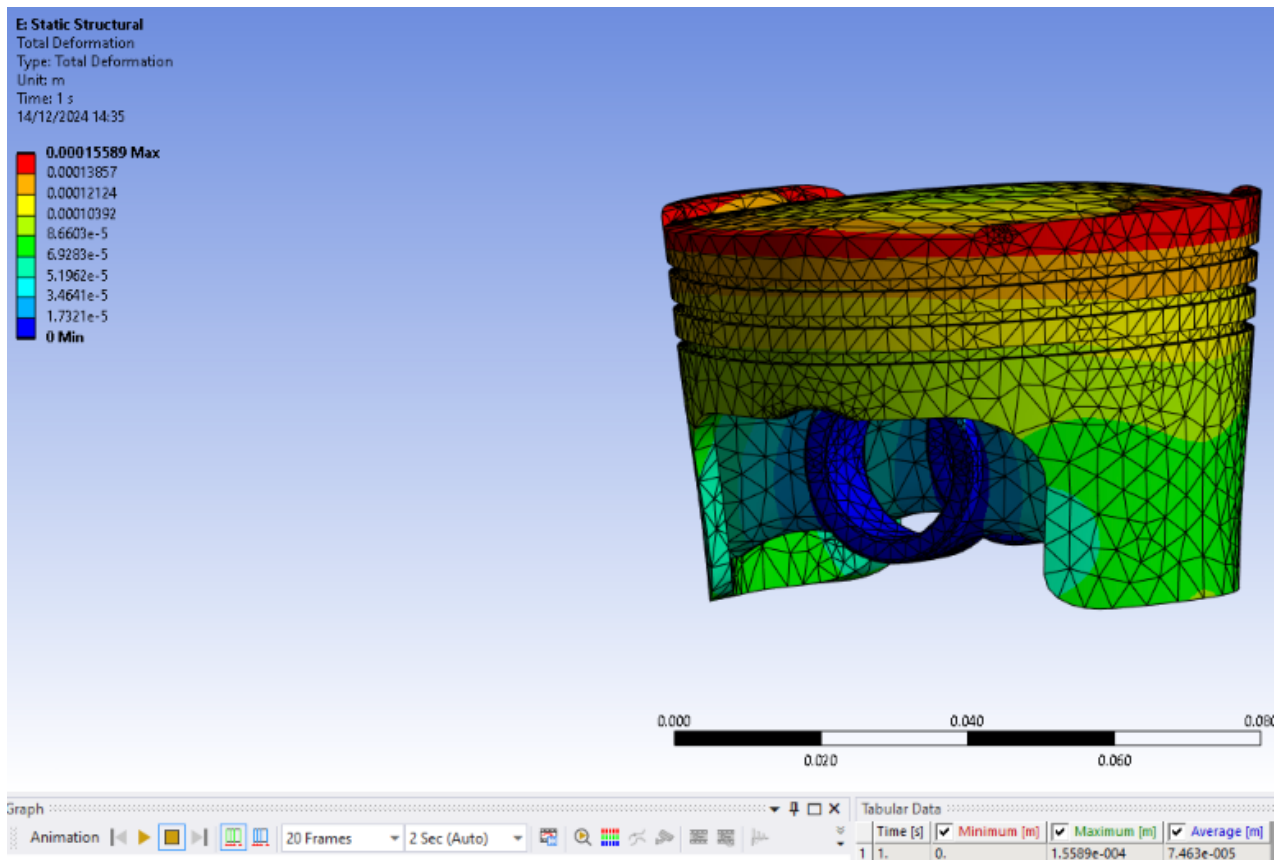


Figure (26); A screenshot to show the Total Deformation of the piston under the applied loads.

While the deformation appears reasonable under combined loading, the exaggerated stress results due to the boundary conditions reduce confidence in the overall conclusions regarding the structural integrity of the piston.

The limitations of the software are compounded by inaccuracies in input parameters, and boundary conditions, which may not fully align with real-world conditions.

While the deformation values remain within expected ranges, the stress concentrations likely overestimate the actual mechanical loads the piston would experience in operation. The results underline the need for careful verification of model inputs and boundary conditions to improve the fidelity of the simulation.

4.5 – Mesh Independence Study.

The original mesh of the model used in this study was generated with the parameters seen in figure 27.

This mesh configuration uses ANSYS's adaptive sizing to focus on areas of higher geometric complexity.

However, this baseline provides the starting point for a mesh independence study to assess whether further refinement improves result accuracy.

This study was performed using the most accurate simulation model available; the original static-structural analysis.

The mesh details were updated (figure 28), significantly increasing the number of elements from 43,824 to 138,027, and nodes from 67,041 to 204,036.

This finer mesh provides more accurate results by better capturing the geometry and stress/strain gradients in critical regions such as the crown edges and pinholes.

The refined mesh results in higher stress and deformation values than the initial, less complex mesh; with the maximum stress increasing to 82.812MPa and the maximum deformation rising to 0.007387mm.

This highlights the importance of performing a mesh independence study to ensure the accuracy and reliability of simulation results.

The differences emphasise that the less complex mesh may underestimate critical metrics in Finite Element Analysis.

The mesh Element Size was then further updated to 0.001m to assess the effect of further refinement on deformation and equivalent stress.

Details of "Mesh"	
Display	Use Geometry Setting
Defaults	
Physics Preference	Mechanical
Element Order	Program Controlled
Element Size	Default
Sizing	
Use Adaptive Sizing	Yes
Resolution	Default (2)
Mesh Defeaturing	Yes
Defeature Size	Default
Transition	Slow
Span Angle Center	Coarse
Initial Size Seed	Assembly
Bounding Box Diagonal	0.11162 m
Average Surface Area	1.4155e-004 m ²
Minimum Edge Length	5.1318e-005 m
Quality	
Inflation	
Advanced	
Statistics	
Nodes	67041
Elements	43824
Show Detailed Statistics	No

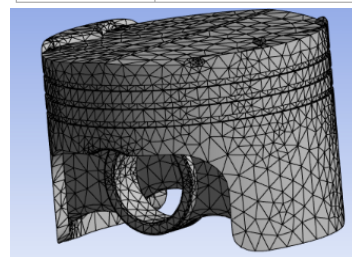


Fig. (27); Original Mesh.

Details of "Mesh"	
Display	Use Geometry Setting
Defaults	
Physics Preference	Mechanical
Element Order	Program Controlled
Element Size	2.e-003 m
Sizing	
Use Adaptive Sizing	Yes
Resolution	Default (2)
Mesh Defeaturing	Yes
Defeature Size	Default
Transition	Slow
Span Angle Center	Coarse
Initial Size Seed	Assembly
Bounding Box Diagonal	0.11162 m
Average Surface Area	1.4155e-004 m ²
Minimum Edge Length	5.1318e-005 m
Quality	
Inflation	
Advanced	
Statistics	
Nodes	204036
Elements	138027
Show Detailed Statistics	No

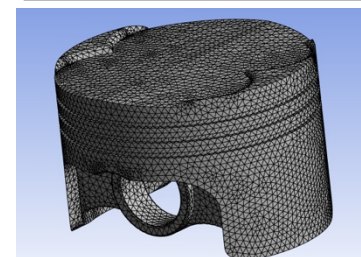


Fig (28); Improved Mesh.

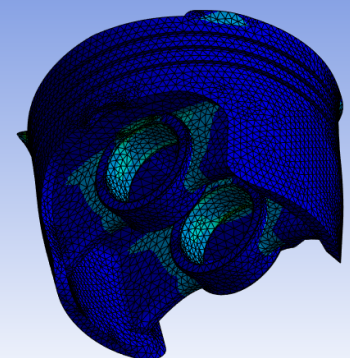
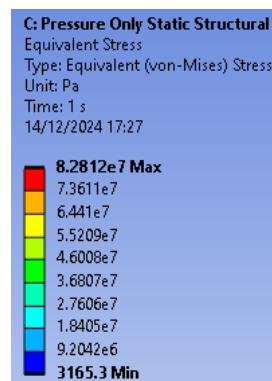


Figure (29); Equivalent Stress results using improved mesh.

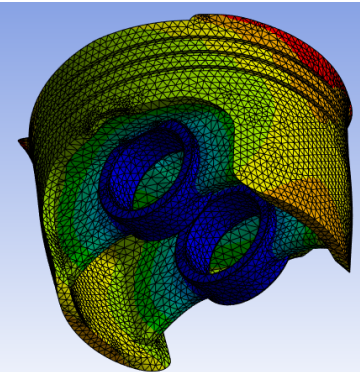
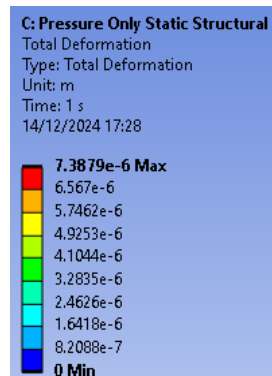


Figure (30); Total Deformation results using improved mesh.

The maximum equivalent stress with the further refined mesh (*figure 31*) increases to 96.962MPa (*figure 32*). This represents an increase of approximately 17% when compared to the previous mesh result of 82.812MPa. This rise in stress value indicates that the finer mesh captures higher stress concentrations more accurately, particularly around critical features in direct contact with excessive loads, such as the gudgeon pinholes, crown edges, and piston body.

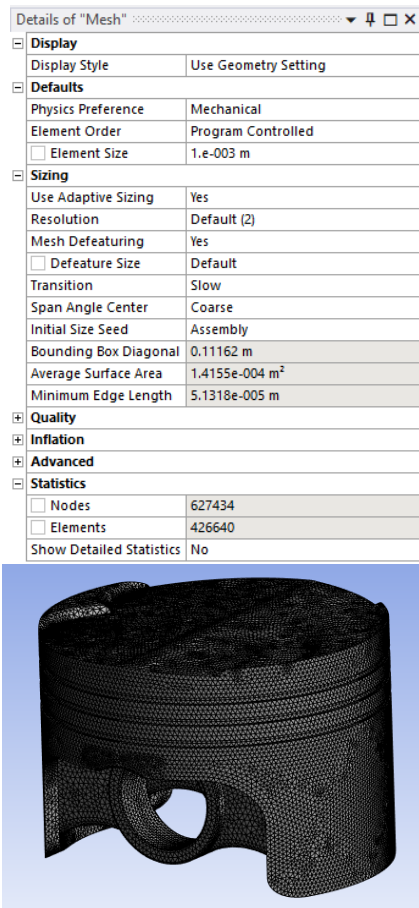


Figure (31); 0.001m Mesh.

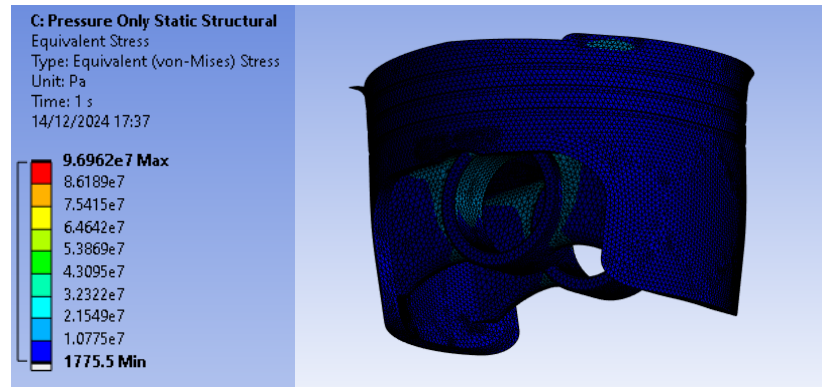


Figure (32); 0.001m mesh Equivalent Stress results.

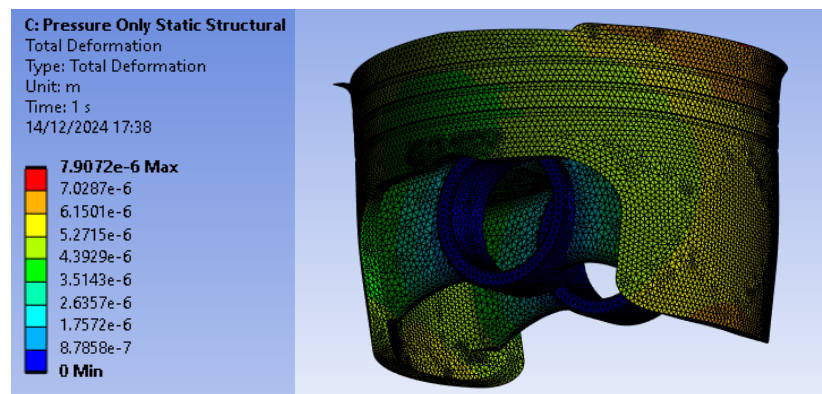


Figure (33); 0.001m mesh Total Deformation results.

The maximum total deformation for the refined mesh is 0.007907mm (*figure 33*), this reflects a slight increase of approximately 7%, showing that the fine mesh better resolves deformation details, likely due to improved element density around the high-deformation areas.

The increase in both stress and deformation values highlights the importance of mesh refinement in improving solution accuracy, particularly for complex geometries like the piston. The changes between the initial and refined meshes suggest that the initial mesh might not be fine enough to fully capture the stress and deformation distribution, reiterating a lack of confidence in results accuracy.

The mesh was further refined to 0.0005m to assess the impact on results. The total deformation increased by approximately 14.23%, indicating the mesh refinement captured more localised deformations, improving the solution's accuracy. The equivalent stress increased by 48.45%, suggesting the refined mesh resolved higher stress concentrations more precisely, especially around critical features such as edges or small fillets.

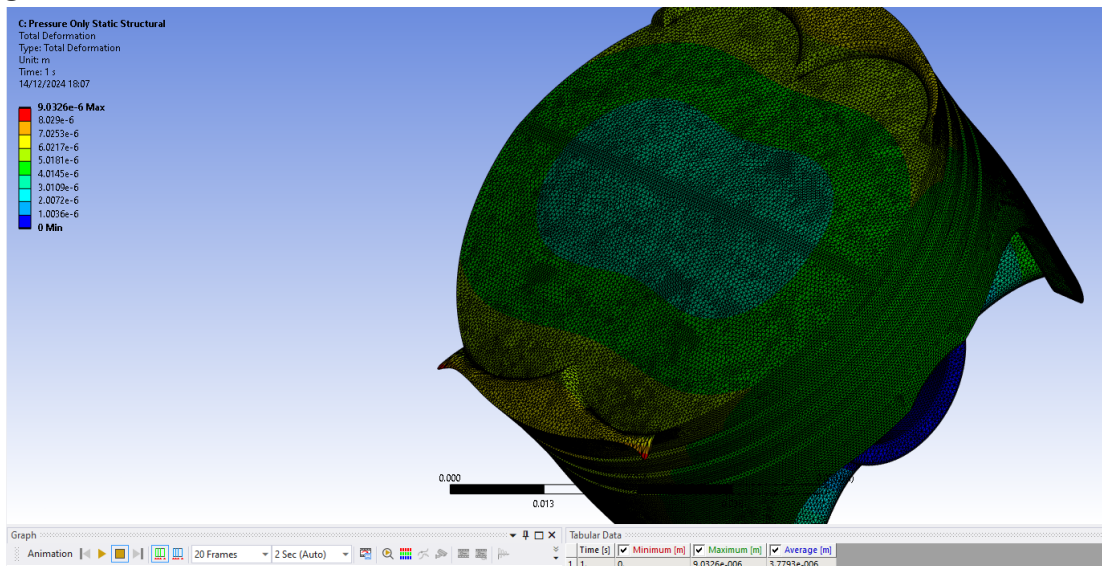


Figure (34); 0.0005m Mesh Total Deformation results.

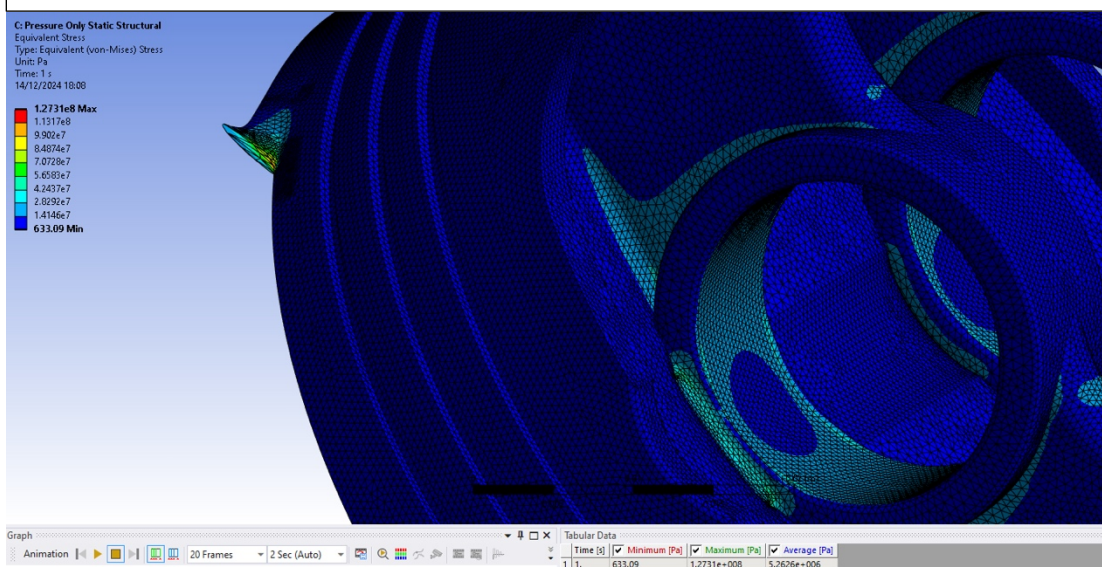


Figure (35); 0.0005m Mesh Equivalent Stress Results.

The use of a convergence tool in this study would provide a systematic approach to ensure the mesh is sufficiently refined to deliver accurate results. By monitoring equivalent stress and total deformation, against mesh refinements, the tool could identify when results stabilise within an acceptable threshold for the application.

This would confirm that further mesh refinement would have negligible impact on the solution, validating the accuracy of the results while optimising computational efficiency. Incorporating this step would enhance the reliability of the analysis and the conclusions drawn.

5.0 – Discussion & Conclusion.

The static structural analysis revealed that the piston could withstand a maximum stress of 70.804MPa under combustion loading alone, remaining well within the material's yield strength of 276MPa. Additionally, the maximum deformation of 0.0069849mm demonstrated the pistons rigidity and suitability for typical operating conditions. These results were consistent with expectations for 6061-T6 aluminium pistons under pressure-only conditions.

The steady-state thermal analysis highlighted the significant thermal gradients across the piston, with the crown reaching a peak temperature of 350 °C. This was accompanied by a maximum heat flux of 973,380W/m², emphasising the effectiveness of the material in dissipating heat from the combustion chamber. The results confirmed that thermal loads, particularly at the crown, could induce stress concentrations and material expansion, necessitating further investigation into their combined effects with mechanical loads.

In the combined analysis of thermal and structural loading, the results revealed significant increases in both stress and deformation. The maximum equivalent stress increased to 2.2615GPa, far exceeding the material's yield strength, while the total deformation rose to 0.009811mm. These findings raised concerns about the reliability of the results due to the boundary conditions applied. The cylindrical supports at the gudgeon-pinholes may have introduced artificial stress concentrations that do not accurately reflect real-world distributions. Furthermore, potential inaccuracies in user-defined-inputs, such as pressure magnitudes, thermal distributions, and material properties highlight the limitations of the simulation software and FEA analysis when user assumptions are not carefully validated.

While the analyses demonstrates the structural integrity of the piston under individual loading conditions, the combined thermal and mechanical loading results underscore the need for more accurate modelling techniques.

The exaggerated stress values observed in the combined analysis suggest the necessity of refining boundary conditions to better represent load transfer through the piston skirt, connecting rod, and crankshaft, highlighting the limitations of FEA.

Additionally, more rigorous validation of input parameters, including thermal gradients, material properties, and applied pressures is essential to improving the reliability of the simulation. This study underscores the importance of validating FEA results against experimental or real-world data to ensure confidence in the conclusions drawn from numerical simulations

6.0 – References.

ASM.matweb.com – 2024

6000 Series Aluminium Alloy; Aluminium Alloy; Metal; Nonferrous Metal.

<https://asm.matweb.com/search/SpecificMaterial.asp?bassnum=ma6061t6>

Accessed 2nd Dec 2024

Bartoszewicz, B. – 2024

Bar-tek.com: The optimal temperature for your engine.

<https://www.bar-tek.com/engine-temperature>

Accessed 14th Dec 2024

Dhari, R. – 2024

Omnicalculator.com: Piston Force Calculator.

<https://www.omnicalculator.com/physics/piston-force>

Accessed 2nd Dec 2024

Jaaskelainen, H. – 2017

Dieselnet.com: Piston Cooling.

[https://dieselnet.com/tech/combustion_piston-cool.php#:~:text=Some%20typical%20temperature%20values%20for,injection\)%20270-350°C](https://dieselnet.com/tech/combustion_piston-cool.php#:~:text=Some%20typical%20temperature%20values%20for,injection)%20270-350°C)

Accessed 14th Dec 2024

Kamble, V.B. – 2018

Researchgate.net: Design and Analysis of Gudgeon Pin to Minimise Stress

Concentration: A Review.

https://www.researchgate.net/publication/324562558_Design_and_Analysis_of_Gudgeon_Pin_to_Minimize_Stress_Concentration_A_Review

Accessed 14th Dec 2024

McCune, R.C. – 2001

Sciencedirect.com: Automotive Engine Materials.

<https://www.sciencedirect.com/topics/engineering/engine-pistons>

Accessed 14th Dec 2024