



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

Application of CAE Processes
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Joseph Howard
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Introduction

Computer-Aided Engineering (CAE) is an integral tool across all engineering disciplines. It enables engineers to design components, assess structural integrity under various loads, simulate aerodynamic behaviour such as downforce and drag, and develop complicated assemblies with interacting parts.

This assignment focuses on the reciprocating components of an internal combustion engine, with focus given to the piston. The piston was selected due to its exposure to the highest thermal and mechanical loads in an engine rotating assembly, especially when compared to the crankshaft, wrist pin, connecting rod and flywheel.

In this study, the piston will be subjected to simulated pressure, temperature and inertia loads to reflect real-world engine operation. The results from the finite element simulations will be analysed in detail to understand how thermal and mechanical loads affect stress, strain and deformation. Based on these insights, both material selection and geometry will be optimised to enhance performance, durability, and thermal resilience. Comparative analysis will guide the development of a more reliable piston design under high-temperature and load conditions.

Literature Review

Understanding the mechanical and thermal behaviour of engine pistons is vital for improving performance and reliability. Pistons endure high combustion pressures and significant thermal gradients, both of which influence material behaviour, deformation, and design tolerance requirements.

Finite Element Analysis (FEA) is widely used to study piston performance under combined thermal and mechanical loading. As noted by *Shigley (2020)*, accuracy depends heavily on mesh density and boundary conditions, especially in high-stress areas.

Material selection plays a major role in piston design. Aluminium alloys are preferred for their low weight and thermal conductivity, but they expand significantly with heat. *Hillier (2012)* emphasise the need to allow for this expansion to prevent piston seizure at peak service temperatures.

Thermal behaviour is also critical. *MAHLE (2016)* studies how piston crown temperatures can exceed 400°C, while the skirt remains cooler due to oil flow. This creates thermal gradients that lead to non-uniform expansion. Figures 1 and 2 illustrate how piston geometry affects heat distribution and highlight the need for precise clearances.

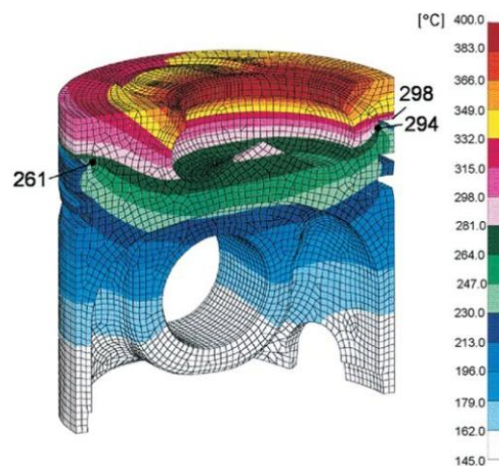


Figure 1 – MAHLE (2016)

Clearance design ensures pistons avoid excessive wall contact during thermal expansion. *MAHLE* data (figure 2) suggests installation clearances for aluminium pistons in four-stroke engines should range from 0.2-0.6% of diameter, with skirt clearances between 1.2-1.8%. *Hillier* also notes that pistons are machined with an oval profile to allow elastic deformation in the skirt rather than the crown- reducing seizure risk.

Table 2.3: Typical installation clearances for light-alloy pistons [% of nominal diameter]

	Gasoline engines			Diesel engines
	Two-stroke	Four-stroke (passenger cars)		Four-stroke (passenger cars)
Engine block material	Aluminum-base alloy	Aluminum-base alloy	Gray cast iron	Gray cast iron
Installation clearance	0.6–1.3	0.2–0.6	0.4–0.8	0.6–0.9
Clearance at upper skirt end	1.4–4.0*	1.2–1.8	1.7–2.4	1.9–2.4

* Only for single-ring designs and high-performance engines (skirt end close to top land)

Figure 2 – MAHLE (2016)

Finally, to define loading conditions, pressure data from *Mark Fall (2024)* using a spark plug sensor shows that four-stroke engines at 6000 RPM reach peak combustion pressures of 95 bar (Figure 3), which is used to inform simulation inputs.

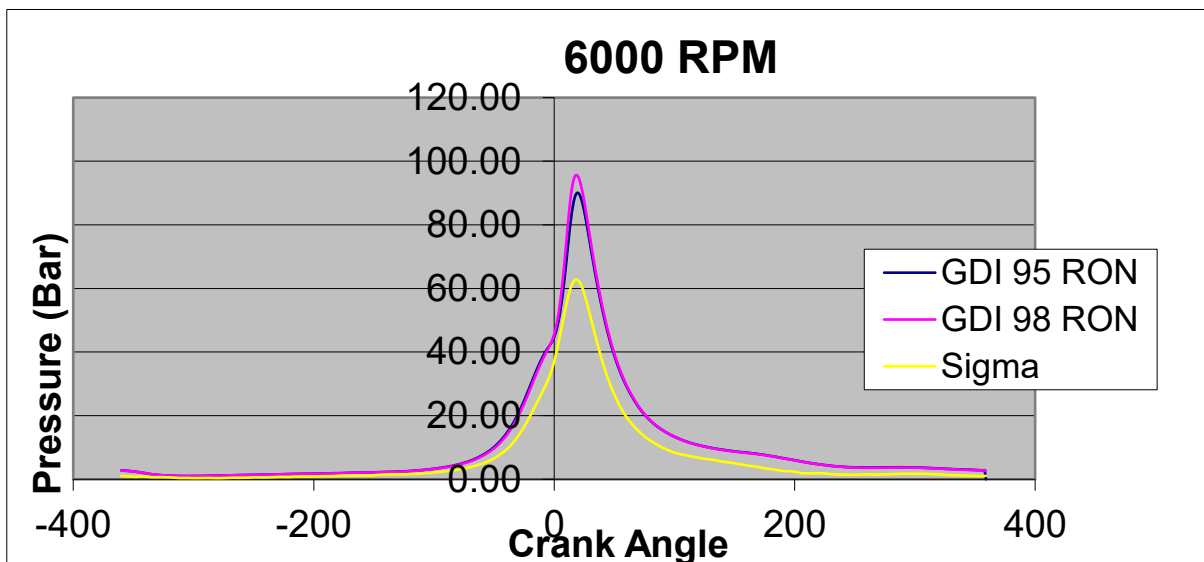


Figure 3 – Mark Fall (2024)

Modelling & Assembly

Materials

This investigation model includes a piston, connecting rod, crankshaft, wrist-pin, and flywheel-components selected for their exposure to varying forces, pressures, and temperatures during engine operation.

To ensure realistic simulation results, accurate material properties were researched for each component. A BMW B48 engine was used as the reference model due to its modern design and available technical data, enabling more accurate inertia calculations for load transfer through the piston.

The baseline piston used 6061-T6 aluminium- a heat treated, high-strength alloy commonly used in performance applications for its strength-to-weight ratio and thermal conductivity. Using realistic material data increases simulation accuracy and supports component comparison throughout testing.

Table 1 shows the material properties used, sourced from *Granta EduPack* and material types gathered from research.

Table 1 – Material Data for Assembly

Component	Material	Density kg/m ³	Youngs Modulus GPa	Poissons Ratio	Tensile Strength MPa	Thermal Conductivity W/m.c	Maximum Service Temperature Celsius	Melting Point Celsius
Wrist Pin ¹	H13 Tool Steel	7.84e3	221	0.295	2.04e3	29	580	1.5e3
Crankshaft ²	Carbon steel 1137	7.9e3	215	0.295	740	53	352	1.51e3
Connecting rod ³	4340 Steel	7.9e3	213	0.295	1.41e3	45	653	1.51e3
Flywheel ⁴	4140 Steel	7.9e3	216	0.295	1.13e3	48	650	1.5e3
Piston ⁵	6061 T6 Aluminium	2.73e3	70	0.33	338	169	150	650

1 - (SuperTech BMW B48 H13 Tool Steel Wrist Pin, no date)

2 – (BMW B48 Technical Manual)

3 – (K1 Technologies)

4 - (MFactory Chromoly-Nitride Racing Flywheel BMW, no date)

5 – (Provided in assignment brief)

Thermal Conditions

To simulate thermal loading accurately, material data for 6061-T6 aluminium was added to the engineering database in ANSYS Mechanical. This included specific heat capacity, coefficient of thermal expansion, and temperature-dependent values for Young's Modulus and Shear Modulus.

Using a range of temperature-dependent values (rather than only ambient data) allows the software to interpolate more realistic material behaviour-particularly for deformation under high thermal loads. This approach improves accuracy in thermal simulations and ensures material performance is realistically represented across the operating temperature range.

Table 2, Additional Aluminium Properties

Specific Heat Constant Pressure	914 J/kg
Coefficient of Thermal Expansion	2.36e-05 Celsius

Table of Properties Row 6: Isotropic Elasticity					
	A	B	C	D	E
1	Temperature (C) 	Young's Modulus (GPa) 	Poisson's Ratio	Bulk Modulus (Pa) 	Shear Modulus (Pa) 
2	22	67	0.33	6.5686E+10	2.5188E+10
3	100	70	0.33	6.8627E+10	2.6316E+10
4	150	68	0.33	6.6667E+10	2.5564E+10
5	200	64	0.33	6.2745E+10	2.406E+10
6	250	56	0.33	5.4902E+10	2.1053E+10
7	300	48	0.33	4.7059E+10	1.8045E+10
8	350	41	0.33	4.0196E+10	1.5414E+10
9	400	32	0.33	3.1373E+10	1.203E+10
10	450	21	0.33	2.0588E+10	7.8947E+09
11	500	12	0.33	1.1765E+10	4.5113E+09

Figure 4, 6061 T6 Temperature dependent properties

Simulation Assumptions and Limitations

All components were modelled in SolidWorks and imported into ANSYS via STEP format. A flywheel was included to better replicate the system's rotational inertia, improving accuracy of dynamic responses.

Manufacturing methods were not modelled. ANSYS assumes ideal materials- homogenous, isotropic, and free from defects- so potential issues like residual stresses, micro-cracks or imperfections from forging or machining are not captured. This can lead to overestimation of durability.

Contact interactions were simplified for stability and efficiency. Tolerances, oil films, and clearances were not considered, meaning the model may underestimate stress concentrations due to misalignment or imperfect contact.

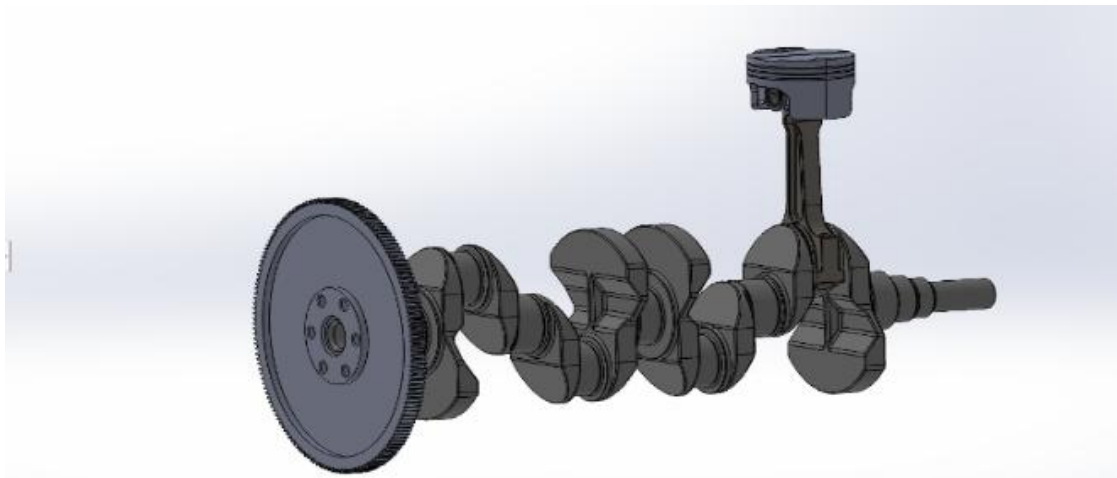


Figure 5 – CAD model of Piston Assembly

Joint Definitions and Assembly Setup

Mechanical joints were applied between components to define movement constraints and simulate realistic kinematics. These joints govern the degrees of freedom (DOF) for each part, enabling motion during dynamic simulation. Table 3 outlines the joint types and permitted motions used in the assembly.

Table 3 – Joint definitions and Functions

Component	Joint	Freedom
Piston to wrist pin	Body – Body Revolute	Rotation in Z axis
Wrist pin to connecting rod	Body – Body Revolute	Rotation in Z axis
Connecting rod to crankshaft	Body – Body Revolute	Rotation in Z Axis
Crankshaft to flywheel	Body – Body Revolute	Rotation in Z Axis
Crankshaft	Body – Ground Revolute	Rotation in Z Axis
Piston	Body – Ground General	Linear Z axis movement

Rigid Dynamics

A rotational velocity of 5000 RPM was applied to the crankshaft in the rigid dynamics system to simulate high-speed engine conditions. This produced a peak reaction force (inertia) of 3761 N, representing dynamic loading. The motion load was then exported for use in the structural analysis.

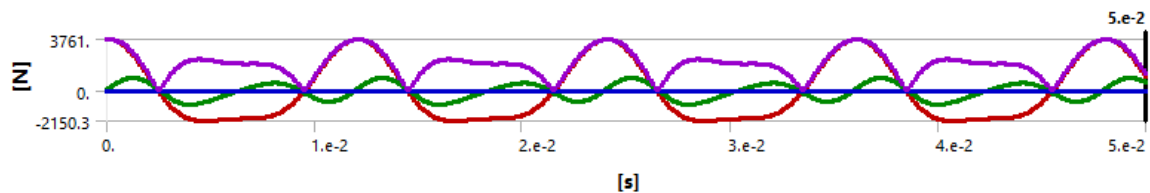


Figure 6 – Rotating assembly inertia

A time step of 0.001s was used, sufficient for capturing steady-state motion, as no acceleration ramping was required. Extending the time scale would increase computational cost without added benefit.

The output load was linked to a static structural system, which shared the same geometry. This load will be used to evaluate stress, strain and deformation, pending finalisation of the thermal study.

Steady State Thermal

After the mechanical simulation, a steady-state thermal analysis was performed to replicate piston temperatures during engine operation. The resulting temperature field was later imported into a thermal-structural simulation to assess temperature-induced stress and deformation.

The piston was meshed using a 0.004 M element size, with refinement in complex regions such as fillets, valve reliefs, and curves to capture steep thermal gradients. Figure 7 shows the mesh used.

Adaptive sizing and slow transitions ensured global element variation, avoiding sudden mesh changes that could reduce accuracy. Quadratic elements were used for better curvature modelling- essential in thermal analysis where temperature gradients are highly localised.

These mesh strategies improved resolution and ensured that critical thermal features like hot spots were accurately captured.

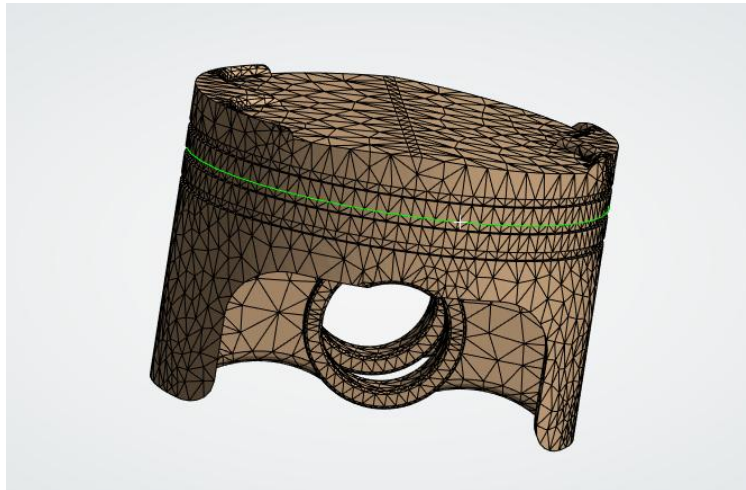


Figure 7 – Piston Mesh (thermal)

Thermal Boundary Conditions and Heat Transfer Inputs

To simulate realistic engine conditions, region-specific thermal boundary conditions were applied in the steady-state thermal analysis. These included heat transfer coefficients (HTC) and corresponding surface temperatures, representing interactions with combustion gases, oil, and other engine systems (Table 4)

Table 4, Thermal Study Boundary Conditions

Component face	Heat Transfer Coefficient Average $W / m^2 k$	Temperature (Celsius) Average
Crown	497.5	928.5
Crown side	447.5	425
Compression Ring 1 Groove	437.5	190
Intermediate Ring 2 Groove	437.5	190
Oil control Ring 3 Groove	338.75	163.75
Underside	675	91.25
Skirt	462.5	107.5

HTC measures how effectively heat transfers between the surface and contacting fluids. Higher HTC values, such as the piston crown, indicate faster heat transfer during combustion. Lower HTC, seen on the underside, reflects slower transfer due to oil cooling.

Surface temperature reflects the steady-state result of this heat exchange. Although the underside has a higher HTC than the ring grooves, it remains colder due to oil contact and limited exposure to combustion heat.

Thermal Simulation

Figures 8-10 show steady-state temperature distribution on the piston. Peak temperatures occur at the crown due to direct combustion exposure, with temperatures decreasing down the piston body.

The gradient is caused by:

- Oil cooling around the lower ring grooves
- Smaller cross-sections (e.g. skirt, oil ring lands) aiding heat dissipation

Lower regions, especially the skirt underside, stay cooler due to contact with the crankcase and continuous oil supply. Thinner sections heat up and cool down faster due to their lower thermal mass, which is evident in the sharp temperature drop along the skirt and wrist-pin boss.

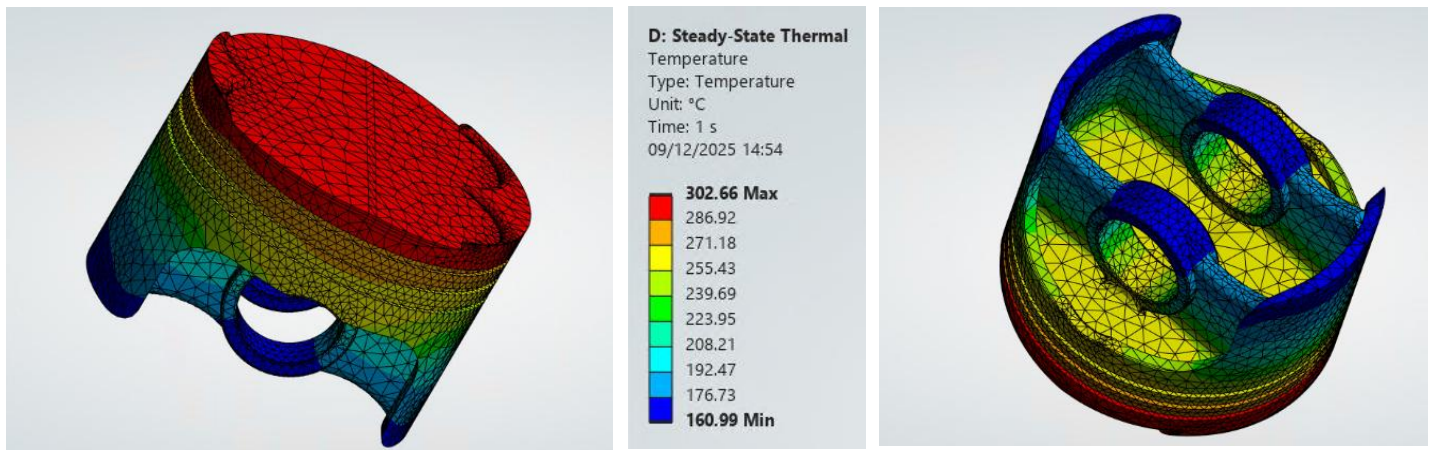


Figure 8, Piston Temperature (top)

Figure 9, Temperature Key

Figure 10, Piston Temperature (bottom)

Static Structural with Thermal Loading

The temperature field from the steady-state thermal analysis was imported into the static structural system to simulate combined mechanical and thermal loading, capturing temperature-dependent expansion and thermal stress for a more realistic in-cylinder representation. A separate mechanical-only case was also analysed to isolate pressure effects and is used for comparison, with full results provided in the appendix.

The mesh strategy remained consistent with previous simulations, using a 2mm global element size with 1mm refinement on the crown, quadratic elements, and smooth transitions.

A 100-bar pressure was applied to the piston crown to represent peak combustion loading. Weak springs were again used to stabilise the model, with minimal influence due to refined meshing constraints.

Total Deformation

Figures 11-13 show a maximum deformation of 0.175 mm, notably higher than the mechanical-only result of 0.0135 mm. This peak occurs at the piston crown centre, where thermal expansion and combustion pressure combine. The valve reliefs also show high deformation due to their thin, raised geometry.

Lower deformation is observed around the skirt and lower pin boss, though moderate displacement occurs at the wrist-pin boss base, attributed to localised heating and thinner material.

While a 0.17 mm crown deformation is within typical limits, it could present issues in high-RPM or forced-induction engines with tight piston-to-head clearances. These results highlight the need to consider thermal distortion in crown and ring land design to prevent interference or wear.

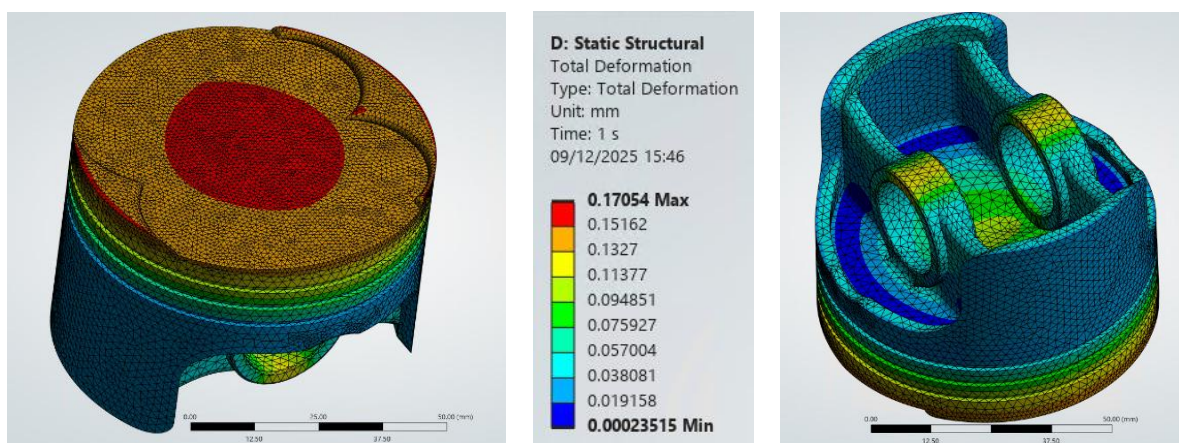


Figure 11, Piston Deformation (top)

Figure 12, Deformation Key

Figure 13, Piston Deformation (bottom)

Equivalent Elastic Strain

Figures 14-16 show a peak strain of 0.0173 mm/mm, notably higher than the 0.0030 mm/mm from mechanical-only loading. This increase is due to the combined effect of thermal expansion and mechanical pressure, especially in thin, high temperature regions.

Strain is concentrated in the pin boss, skirt and filleted ring land areas- zones exposed to both heat and stress. While the crown is the hottest region, its thicker geometry limits deformation, unlike thinner features which show elevated strain.

These results emphasise the need to consider thermal expansion in previously low-strain regions. Sustained high strain could lead to fatigue or plastic deformation in long-term operation.

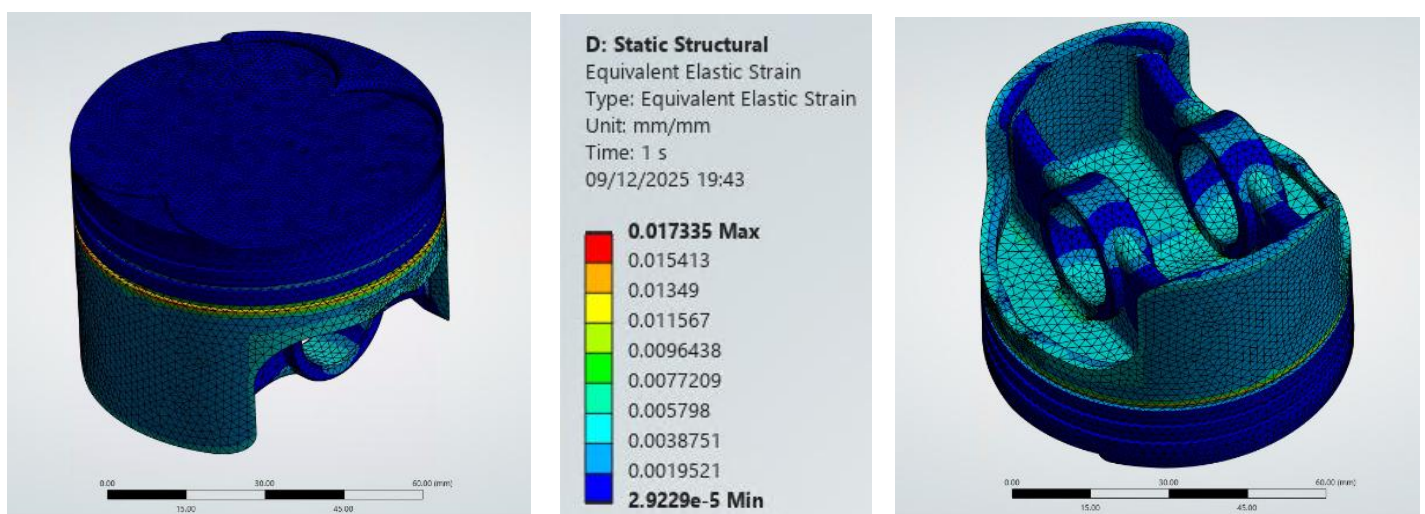


Figure 14, Piston Strain (top)

Figure 15, Strain Key

Figure 16, Piston Strain (bottom)

Von Mises Stress

Figures 17-19 show a peak von-Mises stress of 3462.2 MPa under combined thermal and mechanical loading- far exceeding the 192.5 MPa from the mechanical-only simulation. This surge is driven by thermal expansion constraints and temperature-dependent stress concentrations.

Critical stress appears at the wrist-pin fillets, skirt-ring transitions, and skirt edges, correlating with:

- High thermal gradients
- Thin cross-sectional features
- Fixed boundaries resisting thermal growth

Interestingly, the piston crown, once the highest-stress area, now shows lower stress due to its ability to expand more freely under heat. In contrast, internal regions with constrained geometry experience differential expansion, resulting in stress buildup.

The stress far exceeds the yield strength of 6061-T6 aluminium, (276-310 MPa), indicating plastic deformation or failure risk under prolonged or high-performance use.

This highlights the critical role of thermal stress integration in piston design- omitting it would severely underestimate the real-world stress and impact durability.

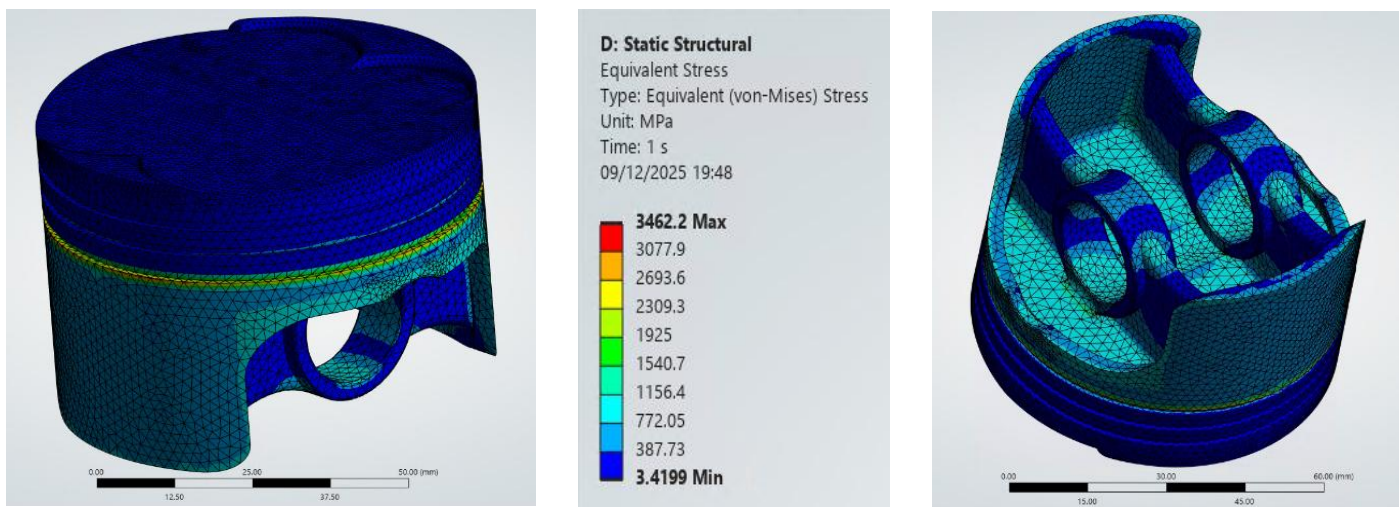


Figure 17, Piston Stress (top)

Figure 18, Stress Key

Figure 19, Piston Stress (bottom)

Piston Performance Improvements

Material Selection

The objective of this section was to improve the structural performance of the piston by upgrading to a stronger material (7075-T6 aluminium) and modifying the geometry to reduce deformation under load. Table 5 below shows material comparison between the stock 6061-T6 aluminium, and the proposed material change.

Table 5 – Material comparison for 6061-T6 vs 7075-T6 Aluminium

Property	6061-T6 Aluminium	7075-T6 Aluminium	Units
Density	2.73x10 ³	2.83x10 ³	Kg/m ³
Young's Modulus (22 degrees)	67	76.6	GPa
Poisson's ratio	0.33	0.33	-
Coefficient of thermal expansion	2.39x10 ⁻⁵	2.41x10 ⁻⁵	°C ⁻¹
Specific Heat Capacity	914	979	J/kg k
Tensile Yield Strength	338	580	MPa

7075-T6 Aluminium Simulation

To assess the impact of using a stronger material, the original piston geometry was re-simulated with 7075-T6 aluminium, replacing the previous 6061 alloy. All boundary conditions and loads remained unchanged, allowing a direct comparison of material performance. Key structural and dynamic results are summarised in table 6.

Table 6 – 7075-T6 Test Results

Result	Value
Max Deformation	0.3541 mm
Max Equivalent Stress	2272.7 MPa
Max Strain	0.0371 mm/mm
Inertia	3898.2 N

Compared to the 6061-T6 piston, using 7075 led to reduced deformation and greater structural rigidity, as evidenced by the lower displacement values. However, the maximum stress rose to 2272.7 MPa, far exceeding the yield strength of 7075-T6 (580 MPa). This suggests that while the part became stiffer, it is still highly susceptible to plastic deformation under thermal-mechanical loading. (Figures 20-22)

Additionally, the rotational inertia increased due to the higher density of 7075-T6, which may affect engine dynamics at high RPM.

Overall, the results indicate that although 7075-T6 improves stiffness, a material upgrade alone is not sufficient to prevent failure- design modifications would also be necessary to see the benefits of stronger materials. Result keys are summarised in table 6 and can also be found in the appendix.

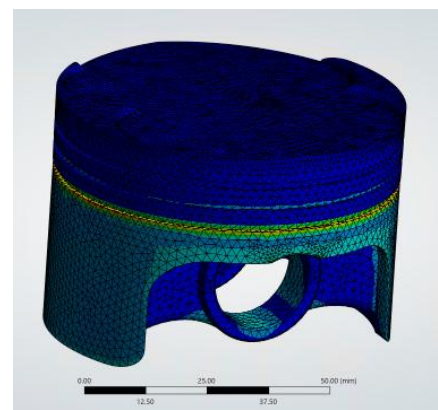
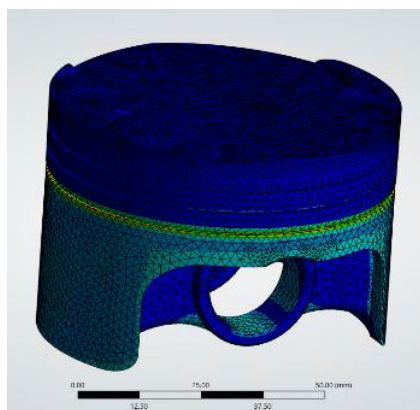
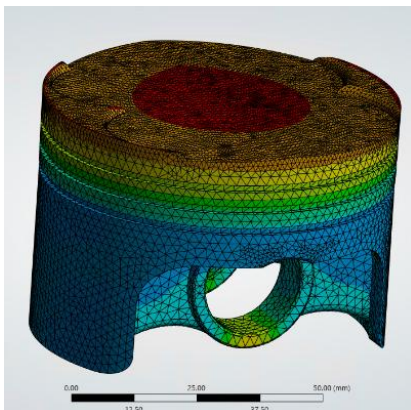


Figure 20 Left, Piston Deformation (top)

Figure 21 Middle, Piston Stress (top)

Figure 22 Right, Piston Strain (top)

Piston CAD Improvements

To further improve the piston's mechanical performance, the CAD geometry was modified to increase structural rigidity. A domed crown was added to enhance strength in the combustion region, and additional support ribs were added near the skirt to reduce deformation under high loads. The figures below show the modified piston design.

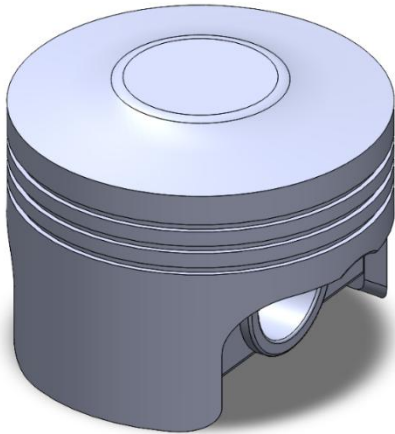


Figure 23, Improved Piston (top)

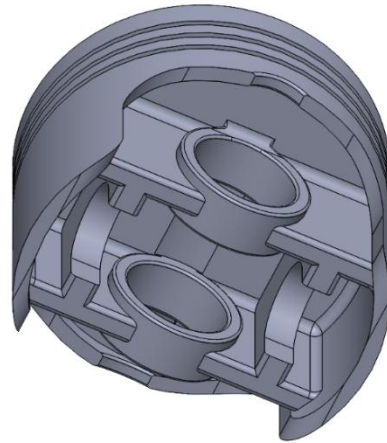


Figure 24, Improved Piston (bottom)

Table 7 – Improved Piston Parameters

Geometry	Volume	Density (7075)	Mass	Inertia at 5000 rpm
Stock Piston	94730.42	2830	0.2679 kg	3898.2 N
Upgraded Piston	113756.71	2830	0.322 kg	4746.7 N

The upgraded piston shows a noticeable increase in volume and mass due to the added geometry. However, this change led to a significant improvement in structural resistance, as confirmed in the simulation results. The inertia increased from 3898.2 N to 4746.7 Newtons, which reflects the additional material. This is a trade-off for improved performance and reduced deformation under thermal and mechanical conditions. The thermal study conducted used the same mesh and temperature constraints as previous simulations and can be found in the appendix.

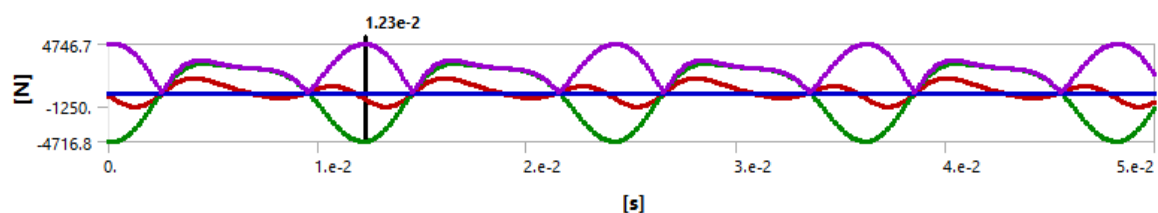


Figure 25, Improved Piston Rotational Inertia (N)

Modified Piston Deformation (7075-T6 Aluminium)

The modified piston design shows a reduction in maximum deformation, now measured at 0.3113 mm, compared to 0.3541 mm in the original piston, with the same material. This enhancement demonstrates the effectiveness of the added structural pieces and dome geometry in resisting bending under load. Although the simulation indicates increased deformation around the lower wrist-pin boss, the overall maximum stress value across the geometry is lower than what was seen in the original piston with 7075-T6 aluminium. This suggests that there is a more efficient load distribution throughout the component.

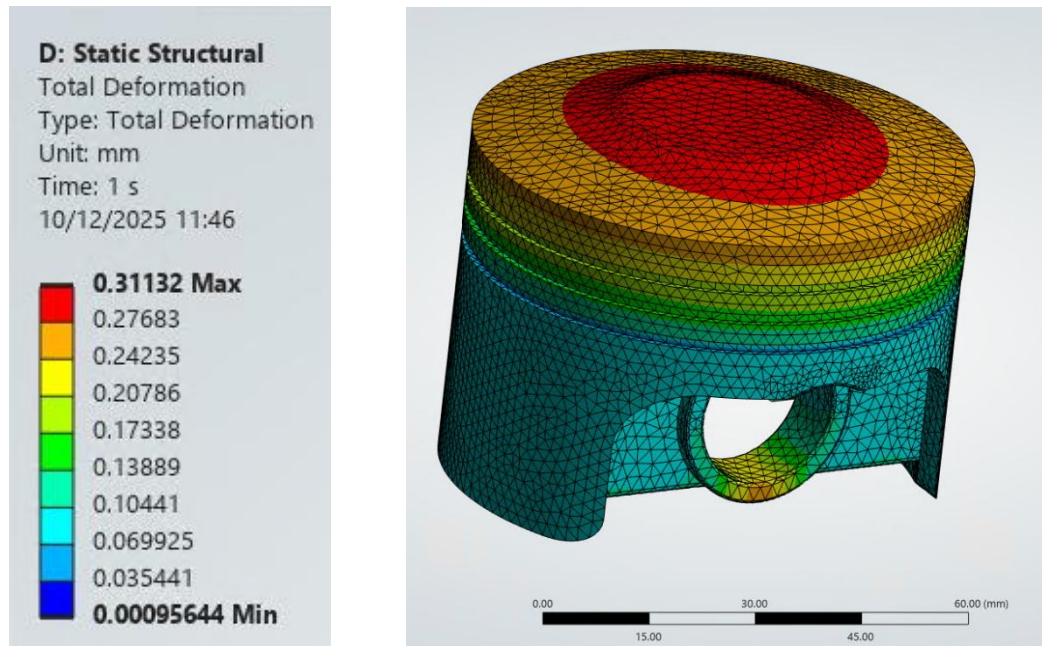


Figure 26, Deformation Key

Figure 27, Improved Piston Deformation (top)

Modified Piston Strain (7075-T6 Aluminium)

The peak equivalent elastic strain in the modified piston is 0.0368 mm/mm, which is slightly lower than the 0.0371 mm/mm observed in the original 7075 piston. This reduction in strain, despite the added mass and geometry complexity, indicates that the redesigned features have contributed to improve stiffness and a slightly more favourable stress distribution. The strain remains concentrated around the upper piston ring grooves, suggesting that while the structural improvements are effective, the piston ring grooves are an area that requires further refinement.

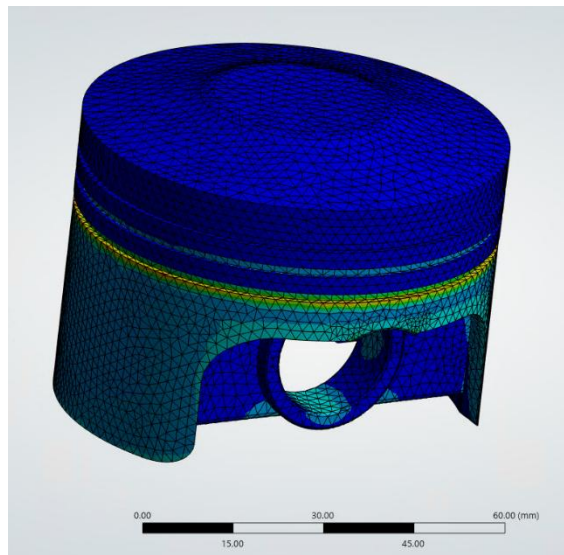
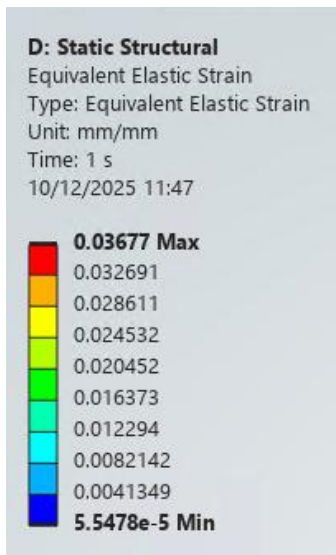


Figure 28, Strain Key

Figure 29, Improved Piston Strain (top)

Modified Piston Stress (7075-T6 Aluminium)

The maximum von Mises stress in the modified piston is 2247.3 MPa, which is slightly lower than the 2272.7 MPa in the original 7075 piston. Although the reduction is minor, it demonstrates that the reinforcements and dome geometry have contributed to a more efficient stress distribution throughout the component. Stress concentrations remain focused near the piston ring grooves, showing that this section still experiences the highest mechanical load under operating conditions. Importantly, the recorded stress is still well above the 7075-T6 aluminium yield strength (580 MPa), meaning that the results are likely to be unrealistic due to the input pressure (110 Bar) or boundary conditions, but can still be used comparatively across iterations.

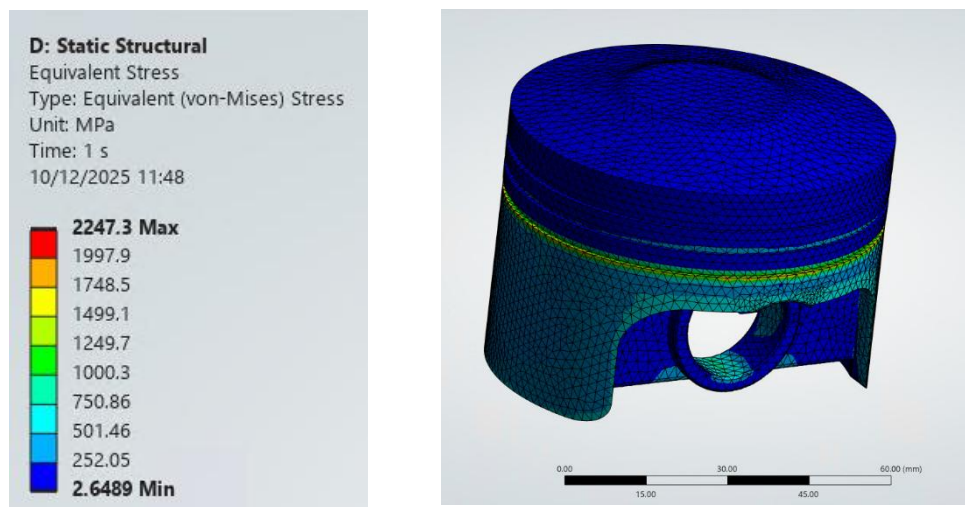


Figure 30, Stress Key

Figure 31, Improved Piston Stress (top)

Results Comparison

Three piston designs were evaluated under identical thermal-mechanical conditions (5000 RPM)

- 6061-T6 Stock
- 7075-T6 stock
- 7075-T6 Modified geometry

Table 8- Performance Comparison

Design	Material	Max Temp (°c)	Max Def (mm)	Max Strain	Max Stress (MPa)	Inertia at 5000 RPM (N)	Mass (kg)
Stock Piston	6061-T6	302.66	0.1705	0.0173	3462.2	3761	0.2586
Stock Piston	7075-T6	313.46	0.3541	0.0371	2272.7	3898.2	0.2679
Modified Piston	7075-T6	308.68	0.3113	0.0367	2247.3	4746.7	0.322

Summary:

- 7075-T6 stock reduced peak stress vs 6061-T6 but increased deformation and strain
- 7075-T6 modified geometry further improved stress distribution, lowering peak stresses while maintaining deformation within limits
- Increased mass and inertia are trade-offs, contributing to increased structural integrity and fatigue resistance
- The modified design shows a more even load distribution, particularly at high-stress areas like the skirt and pin boss.

In conclusion, switching to 7075-T6 improves strength, and geometric refinement enhances durability. Though slightly heavier, the modified piston offers better thermal resistance and fatigue performance under high loads.

Deformation Assessment Against Industry Tolerances

To verify feasibility, simulated deformation results were compared against MAHLE industry guidelines for aluminium pistons. For a piston diameter of 70.74 mm, acceptable deformation ranges from 0.1415 mm to 0.4244 mm (0.2-0.6% of diameter)

Table 2.3: Typical installation clearances for light-alloy pistons [% of nominal diameter]

	Gasoline engines			Diesel engines
	Two-stroke	Four-stroke (passenger cars)		Four-stroke (passenger cars)
Engine block material	Aluminum-base alloy	Aluminum-base alloy	Gray cast iron	Gray cast iron
installation clearance	0.6–1.3	0.2–0.6	0.4–0.8	0.6–0.9
Clearance at upper skirt end	1.4–4.0*	1.2–1.8	1.7–2.4	1.9–2.4

* Only for single-ring designs and high-performance engines (skirt end close to top land)

Figure 32 – MAHLE installation clearances (2016)

Table 9- Tolerance check

Design	Max Def (mm)	Within Tolerance?
Stock Piston (6061)	0.1705	Yes
Stock Piston (7075)	0.3541	Yes
Modified Piston (7075)	0.3113	Yes

All designs fall within the allowable range, confirming structural soundness. Despite increased mass and inertia in the 7075 variants, no excessive deformation was observed that could risk cylinder wall contact.

Validation

Various validation tests were provided from *Dlubal Software (2016)*, which contained instructions for simulation tests, and hand calculations. The specified geometries were produced in SolidWorks and imported to test in ANSYS. The resultant deformation from the specified pressure/temperature was then checked against hand calculations, to verify ANSYS is running as expected.

Simple Beam Validation

Table 10 – Cantilever Validation Parameters

Category	Symbol	Value	Unit
Modulus of Elasticity	E	210,000	MPa
Poisson's Ratio	ν	0.00	-
Shear Modulus	G	105,000	MPa
Length	L	1.00	M
Width	W	0.050	M
Thickness	T	0.005	M
Pressure	p	2750	Pa

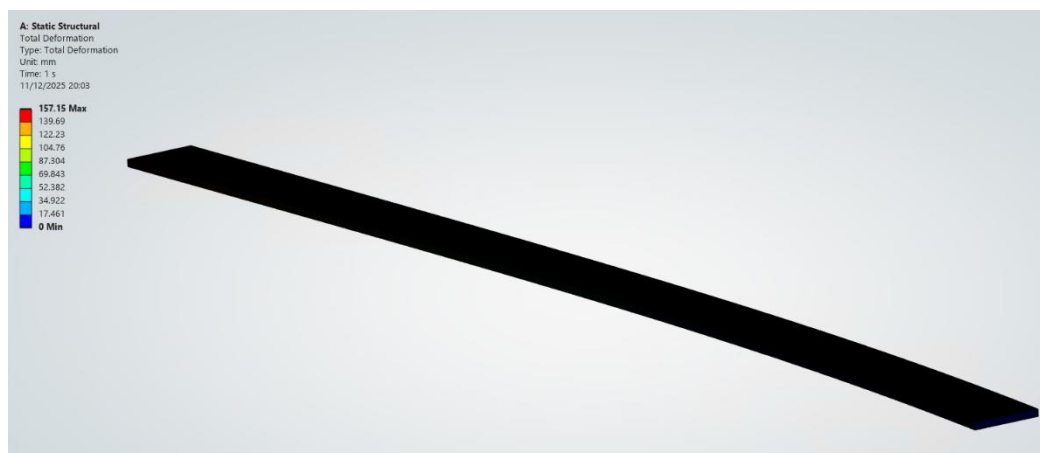


Figure 33 – Cantilever Beam Validation

The provided data was implemented into the ANSYS simulation, and total deformation was recorded. After an applied surface pressure of 2750 Pa, the simulation recorded maximum deformation as 157.15mm.

The following hand calculation was produced, to compare to the simulation.

$$u_{z,max} = \frac{3pL^4}{2Ew^2t} = \frac{3(2750)(1^4)}{2(210,000)(0.050^2)(0.005)} = 157.143 \text{ mm}$$

With a simulation deformation of 157.15 mm, and a hand calculation of 157.143, a difference of 0.004% was observed. This suggests that the ANSYS software is fairly accurate, and that all work in this report is valid.

Double Beam Invar Validation

Table 11 – Bimetallic Strip Thermal Validation Parameters

Category	Property	Symbol	Value	Unit
Material- Invar	Modulus of Elasticity	E_i	137,000	MPa
	Poisson's Ratio	ν_i	0.280	-
	Coefficient of Thermal Expansion	α_i	1.200×10^{-6}	$^{\circ}\text{C}^{-1}$
Material- Copper	Modulus of Elasticity	E_c	130,000	MPa
	Poisson's Ratio	ν_c	0.354	-
	Coefficient of Thermal Expansion	α_c	2.00×10^{-5}	$^{\circ}\text{C}^{-1}$
Geometry	Strip Width	W	5.00	mm
	Layer Thickness	$t=t_c = t_i$	1.00	mm
	Length	L	100	mm
Loading	Temperature Increase	T_c	100	$^{\circ}\text{C}$

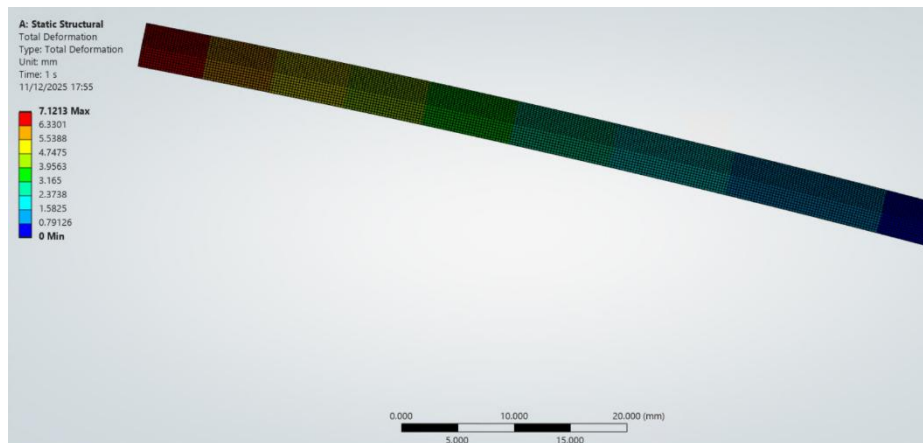


Figure 34 – Bimetallic Strip Simulation

The data provided was implemented into ANSYS, and total deformation was recorded. With an applied temperature increase of 100 degrees (22-122 degrees), a total deformation of 7.1213 mm was determined in the simulation.

The following hand calculations were produced, to compare to the simulation.

$$K_1 = 4 + 6 \left(\frac{t_c}{t_i} \right) + 4 \left(\frac{t_c}{t_i} \right)^2 + \frac{E_c}{E_i} \left(\frac{t_c}{t_i} \right)^3 + \frac{E_i t_i}{E_c t_c} = 4 + 6 \left(\frac{1}{1} \right) + 4 \left(\frac{1}{1} \right)^2 + \frac{130,000}{137,000} \left(\frac{1}{1} \right)^3 + \frac{137,000(1)}{130,000(1)}$$

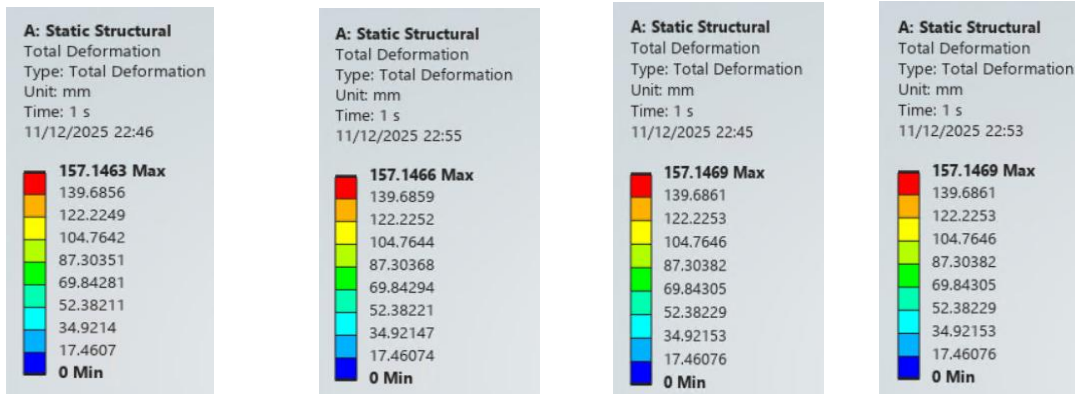
$$u_{z,max} = \frac{6(\alpha_i - \alpha_c)(t_i + t_c)T_c}{t^2 k_1} \frac{L^2}{2} = 7.049 \text{ mm}$$

With a simulation deformation of 7.1213 mm and hand calculation of 7.049 mm, a difference of 1.02% was observed. This again suggests that the ANSYS simulations, and this report- are valid.

Mesh Validation

To verify the accuracy of the ANSYS simulation results, a mesh convergence study was conducted using the cantilever beam model. The beam was analysed with four different mesh densities, with element sizes of 10mm, 5mm, 1mm, and 0.75mm. The purpose of this validation was to determine the influence of mesh resolution on the resulting displacement and assess whether the simulation was mesh independent.

In general, a finer mesh is expected to produce more accurate results, up to the point where further refinement produces negligible changes- this is referred to as mesh convergence.



Figures 35 (a,b,c,d) – Cantilever Beam Deformation (10,5,1,0.75mm mesh)

Table 12 – Mesh Validation results

Mesh Element size (mm)	Max Displacement (mm)	% Difference to Hand Calc
10	157.1466	+0.00229
5	157.1463	+0.00210
1	157.1469	+0.00248
0.75	157.1469	+0.00248

The results demonstrate that mesh convergence was effectively achieved at a mesh size of 1mm. Beyond this point, additional refinement (0.75mm) produced identical displacement values, indicating that the solution had stabilised and was no longer significantly influenced by mesh density.

Interestingly, all simulated values showed a slight overestimation compared to the analytical hand calculation of 157.143 mm. This minor deviation (0.002-0.0025%) is likely due to idealisations in the analytical model, such as the assumptions of ideal material behaviour and boundary conditions. In contrast, the FEA model incorporates a more realistic numerical approach.

Therefore, the convergence study confirms that the ANSYS solver is producing consistent and trustworthy results, and that a mesh size of 1mm offers an optimal balance between computational solve time and accuracy.

Conclusion

This report demonstrated the effective use of FEA simulations to model realistic thermal and mechanical loading on a piston, and how these conditions influence stress, strain and deformation.

The stock piston with 6061-T6 aluminium showed adequate performance but lacked the yield strength to withstand the applied loads. Thermal effects notably increased deformation and stress compared to the mechanical-only case.

Replacing the material with 7075-T6 aluminium improved strength, reducing peak stress from 3462.2 MPa to 2272.7 MPa. However, due to its higher density and thermal conductivity, this change also increased deformation, strain, and inertia. Despite this, the better stress distribution suggests an improvement in fatigue resistance.

The modified piston design further improved performance, reducing peak stress to 2247.3 MPa and lowering peak temperature despite the added mass and inertia. These results confirm that the design changes successfully improved structural integrity and thermal management.

Although deformation increased across the upgrades, all designs remained within acceptable expansion limits as referenced by MAHLE data. While peak stresses exceeded yield strength, these occurred only briefly during combustion and are unlikely to cause failure under normal operation.

Overall, the modified piston using 7075-T6 aluminium offers the best balance of strength, durability, and fatigue resistance, making it the most reliable option under simulated conditions.

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Appendix

7075-T6 Aluminium Thermal Study

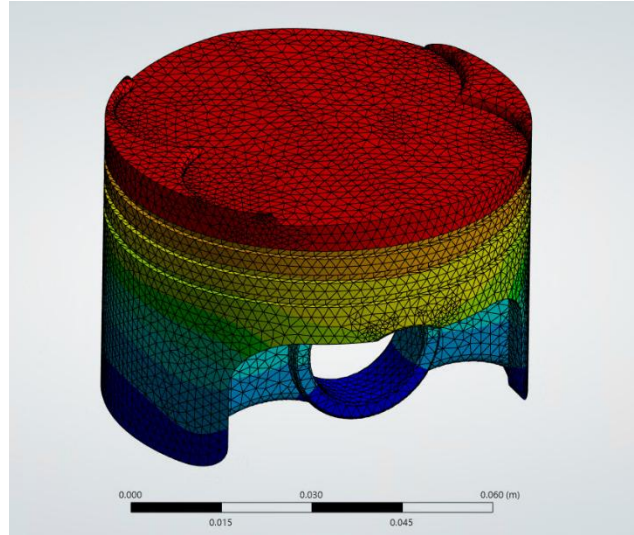
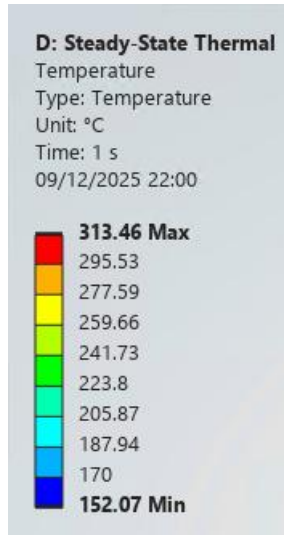


Figure 36, Temperature Key

Figure 37, Improved Piston Temperature (top)

Steady-state thermal simulation results showing peak temperature distribution across piston geometry.

7075-T6 Aluminium Total deformation

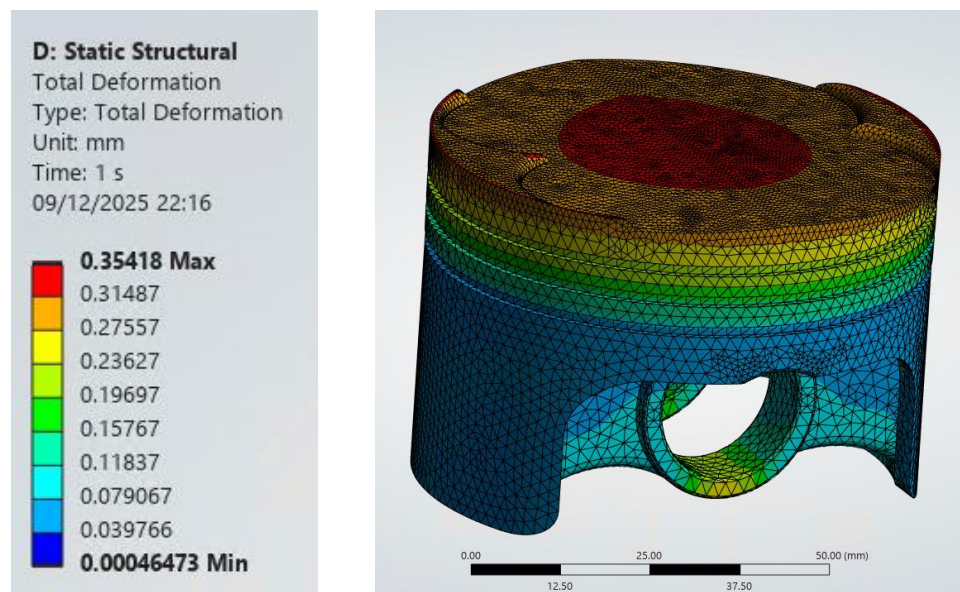


Figure 38, Deformation Key

Figure 39, Improved Piston Deformation (top)

Steady-state structural simulation results showing peak displacement across piston geometry.

7075-T6 Aluminium Equivalent Stress

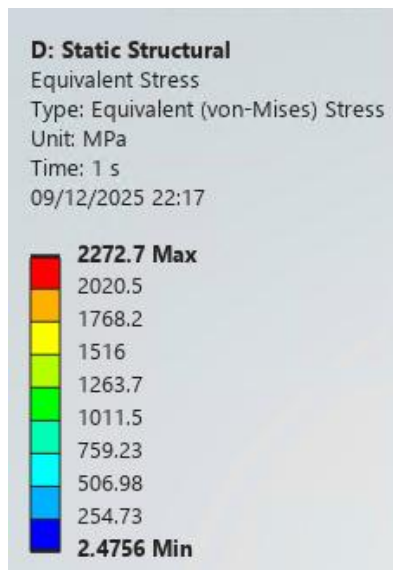


Figure 40, Stress Key

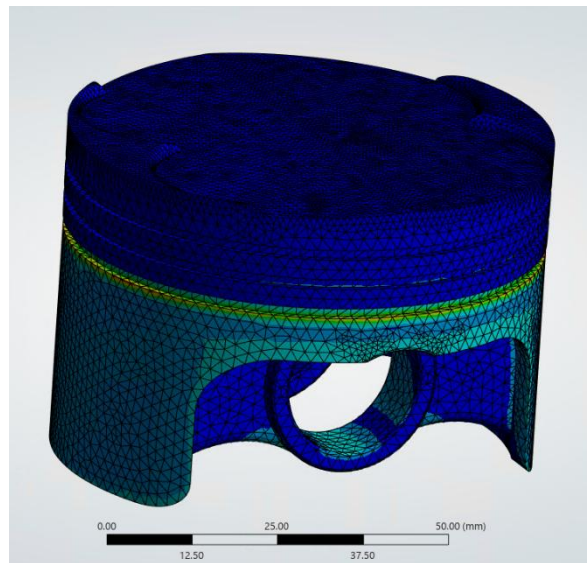


Figure 41, Improved Piston Stress (top)

Steady-state structural simulation results showing peak stress distribution across piston geometry.

7075-T6 Aluminium Equivalent Elastic Strain

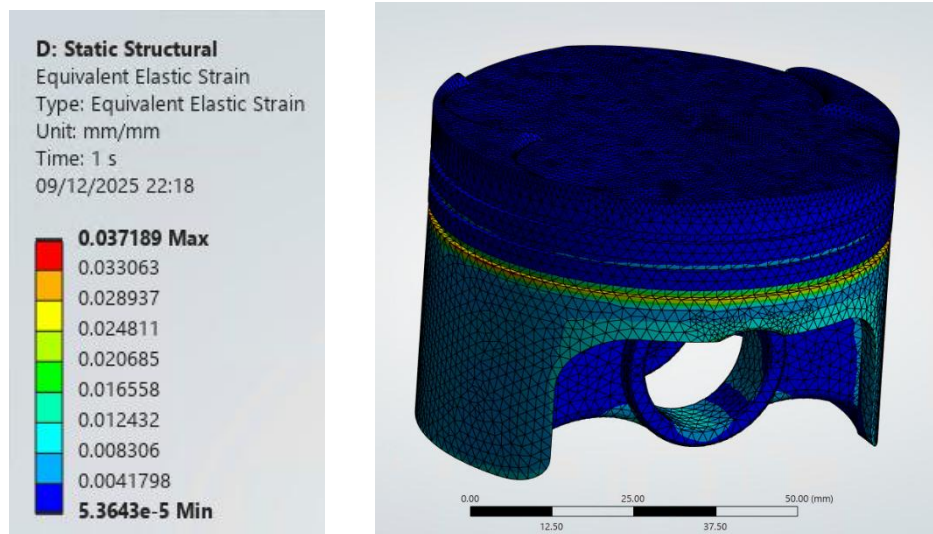


Figure 42, Strain Key

Figure 43, Improved Piston Strain (top)

Steady-state structural simulation results showing peak strain distribution across piston geometry.

Static Structural – No Temperature Influence

To assess the piston's structural behaviour under purely mechanical loading, a static structural simulation was run without thermal effects. This simplified scenario assumed ambient temperature operation, with no thermally induced degradation.

A uniform pressure of 110 bar was applied to the crown in all scenarios, to simulate peak combustion pressure. Weak springs were used to stabilise the model in virtual space, preventing rigid body motion while allowing realistic deformation.

Mesh settings matched those in all simulations. A global 2 mm element size was used, with a refined 1 mm mesh on the crown. Quadratic elements captured curved geometry more accurately. (Figure 44)



Figure 44 – Piston mesh

Total Deformation – Mechanical Loading Only

Under mechanical loading alone, the piston experienced a maximum deformation of 0.0135 mm, concentrated around the valve reliefs on the crown- areas of sharp curvature and geometric discontinuity (Figures 45 and 46)

Despite the 110 bar pressure, deformation across the rest of the piston remained low (0.001- 0.006mm), confirming good structural stability under static loads at ambient temperature.

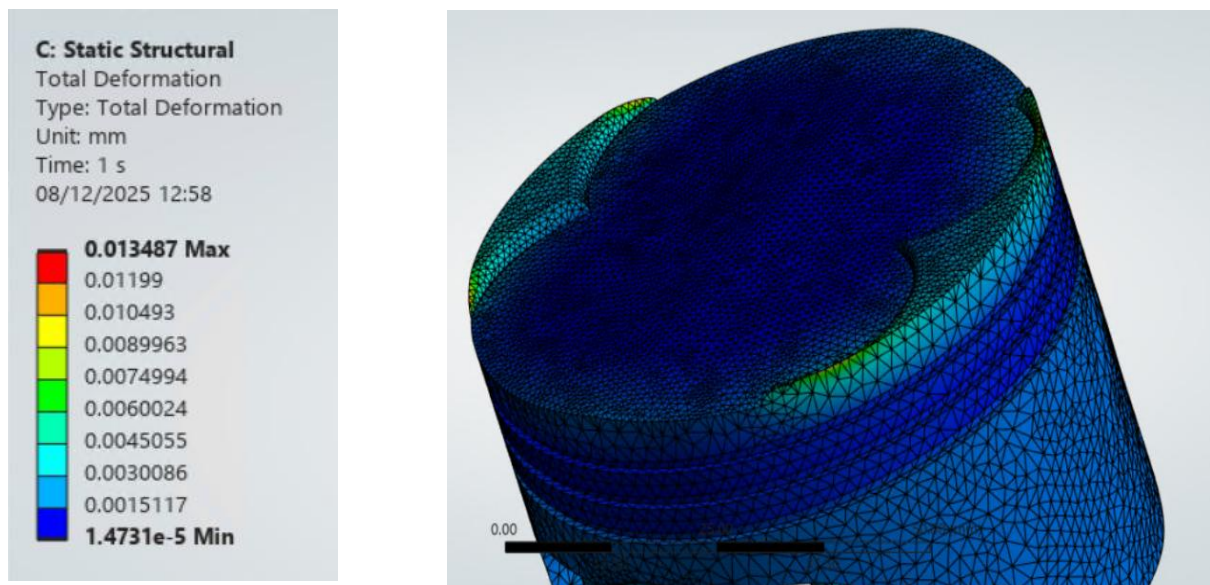


Figure 45, Deformation Key

Figure 46, Piston Deformation (top)

Von-Mises Stress – Mechanical Loading Only

Figures 47 and 48 show the von-Mises stress distribution under mechanical loading. A peak stress of 192.47 MPa was observed at the crown's valve reliefs-locations of sharp curvature and reduced thickness acting as stress concentrators.

Stress in the crown centre remained lower, indicating effective pressure distribution. The skirt and lower body experienced minimal stress (<50 MPa), confirming that static loads are mainly absorbed at the crown face.

As this peak stress is below the 6061-T6 aluminium yield range (276-310 MPa), the piston remains within safe limits under ambient conditions.

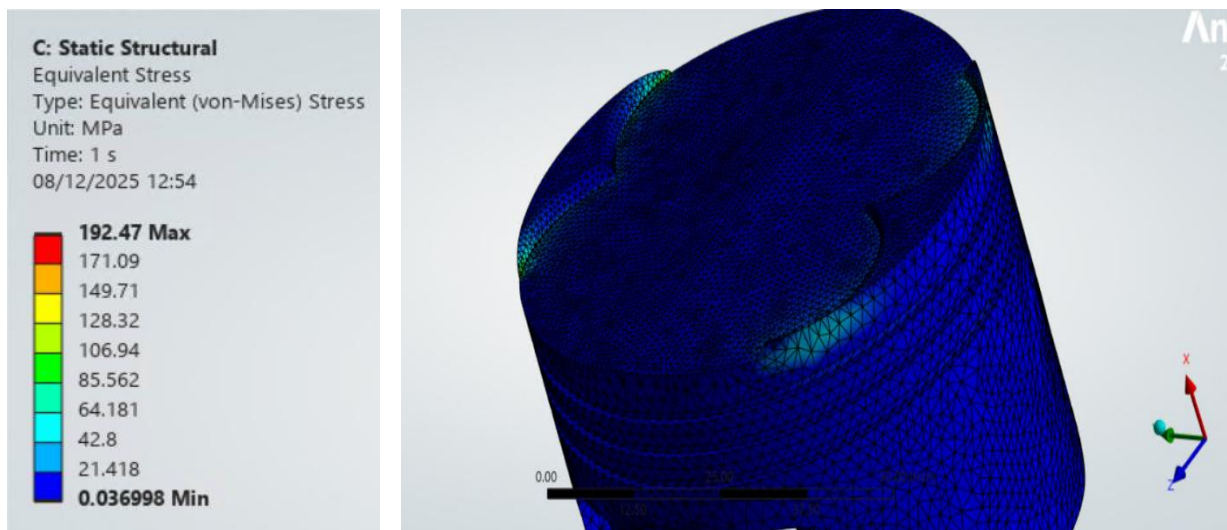


Figure 47, Stress Key

Figure 48, Piston Stress (top)

Equivalent Elastic Strain – Mechanical Loading Only

Figures 49 and 50 show the elastic strain distribution under mechanical loading. A peak strain of 0.00304 mm/mm occurred near the crown's valve reliefs-matching the peak stress zones caused by sharp geometry and reduced stiffness.

Elsewhere, strain remained below 0.001 mm/mm, indicating minimal deformation across the skirt and lower body.

All observed values are within the elastic range for 6061-T6 aluminium, confirming elastic material behaviour. However, localised strain at the crown edges could become critical under thermal or cyclic conditions.

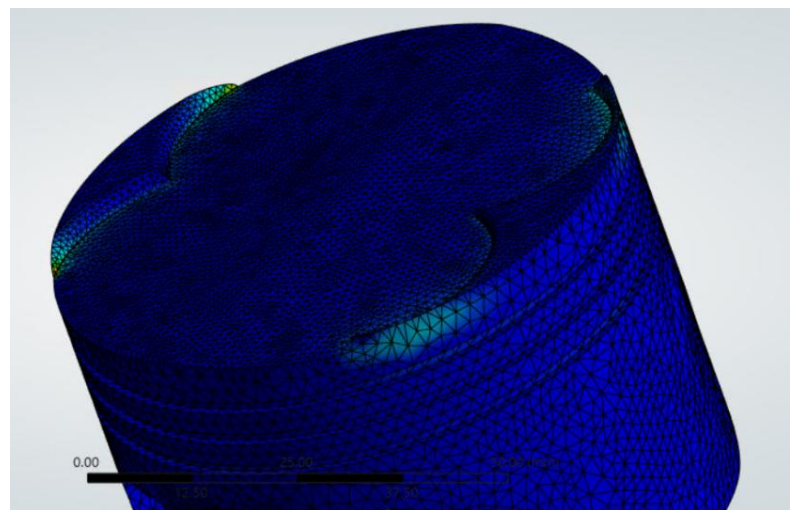
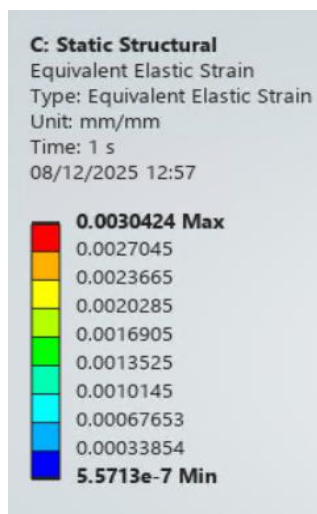


Figure 49, Strain Key

Figure 50, Piston Strain (top)

Comparison of Results

Table 13- Results

Result type	Mechanical only	Mech with Thermal	% Increase
Max Deformation (mm)	0.013	0.17054	1211.85
Max von-Mises Stress (MPa)	192.47	3462.2	1698.2
Max Elastic Strain (mm/mm)	0.003	0.01733	477.66

Introducing a thermal loading resulted in a substantial increase across all measured outputs. Total deformation rose by over 1200%, underscoring the significant impact of thermal expansion on piston behaviour.

The von-Mises stress more than doubled the materials yield strength, indicating a high likelihood of plastic deformation or failure under realistic engine conditions.

Similarly, elastic strain increased by over 470%, particularly in thin regions. This reflects the combined influence of thermal expansion and structural loading, especially in areas unable to accommodate free expansion.

These results clearly demonstrate that ignoring thermal effects in high-temperature components like pistons can lead to underestimated stress levels and inaccurate durability predictions. Integrating thermal loads is therefore essential for realistic simulation results.

7075-T6 Aluminium

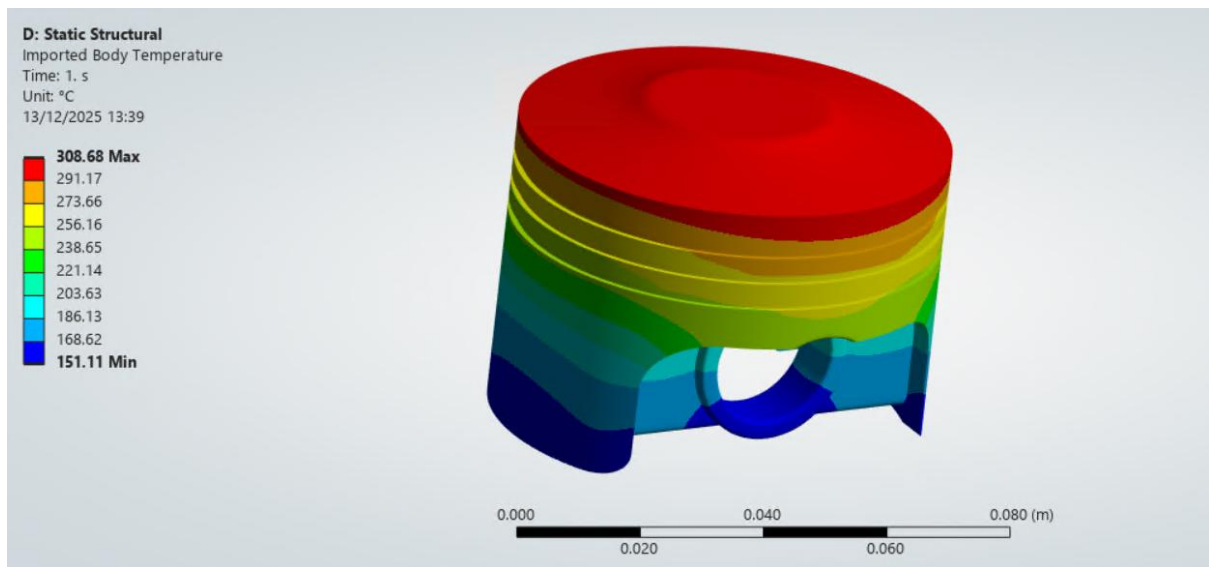


Figure 51, Modified Piston Temperature (top)

Steady-state thermal simulation results showing peak temperature distribution across piston geometry.