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***Fatigue Analysis of a Connecting Rod using
ANSYS Mechanical***

Structural Component Analysis (ACHA7030)

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12/12/2025

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1. Methodology & Workflow

To assess fatigue in a connecting rod using FEA, two critical load cases are required: the condition of maximum compression and the condition of maximum tension.

Maximum compression occurs at TDC during peak combustion pressure, while maximum tension occurs when inertial forces dominate as the piston accelerates away from TDC. These two extremes define the stress range used for fatigue evaluation.

The below solver was used to carry the studies.

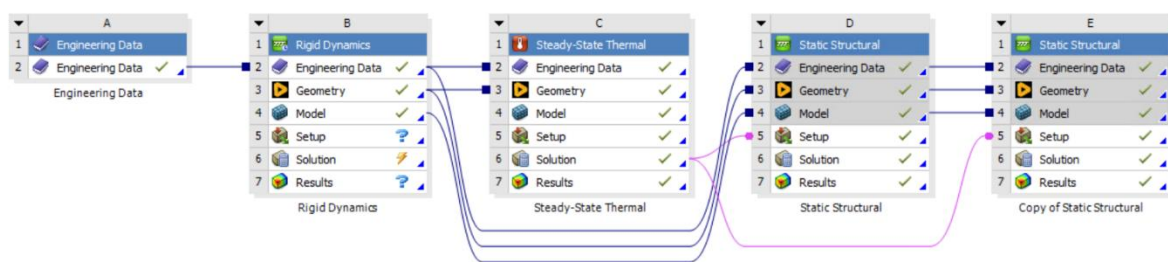


Figure 1.1: Work bench workflow

Engineering data

Material data was exported from Granta Edupack and imported to the ANSYS Workbench.

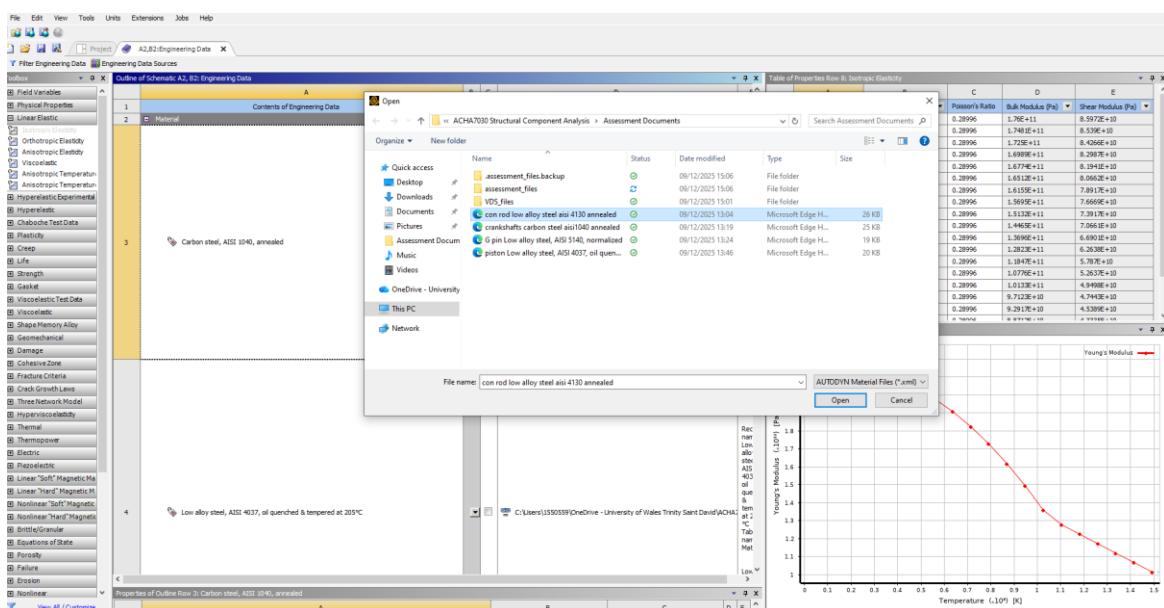


Figure 1.2: Engineering data imported

Rigid dynamics

Rigid dynamic analyses were performed to quantify the initial force experienced by the connecting rod. Following the import of the geometric model, the assembly was rotated 270 degrees to align with the y-axis for improved visualization. Material data was applied according to the components.

Body-to-ground joints were applied to the piston and crankshaft to replicate their fixed interaction with the engine block. The piston was assigned a general joint to permit translational motion along the y-axis, while rotational joints were applied to the remaining components to represent their relative rotational motion.

The time step was set to 0.0000256 s to capture one degree of crankshaft rotation per step at 6500 RPM. A rotational speed of 6500 RPM was applied to the crankshaft. A joint probe was placed between the connecting rod and the gudgeon pin to measure the initial force transmitted through the joint.

Two sets of motion loads were exported for the fatigue analysis. The first corresponds to 360 degrees, where the piston is at top dead centre (TDC) and so experiences maximum inertial loading during the upward stroke. The second corresponds to 14 degrees, where peak gas pressure occurs at 6500 RPM according to the Excel data, producing the highest compressive force on the connecting rod.

To validate the widget's dynamic response, an independent reference analysis was performed in MATLAB. The MATLAB results do not match the widget output exactly, but the discrepancy is minor and the two datasets show close agreement. This indicates that the widget's dynamic computation is consistent with the analytical model.

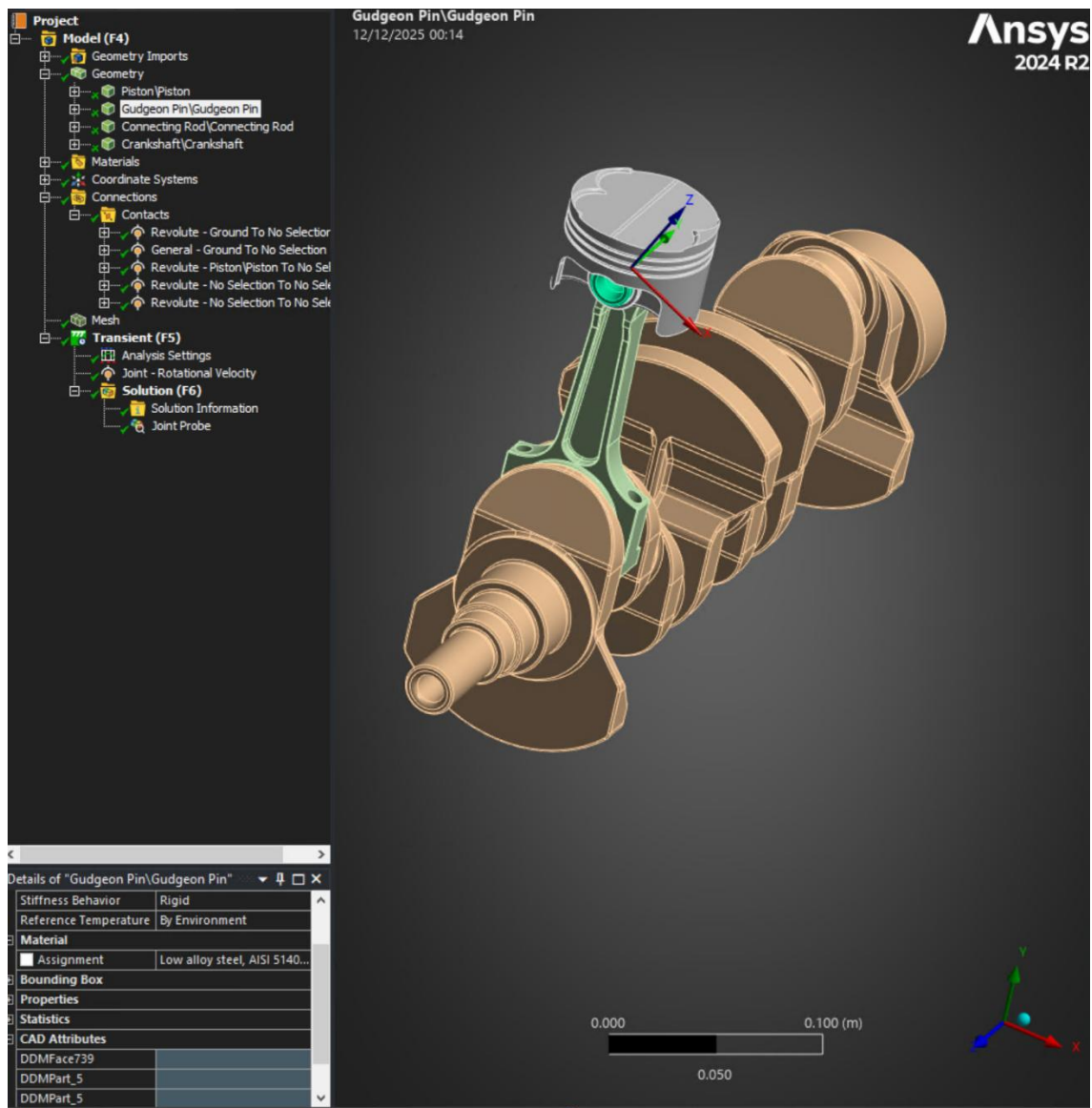


Figure 1.3: Rigid dynamics study

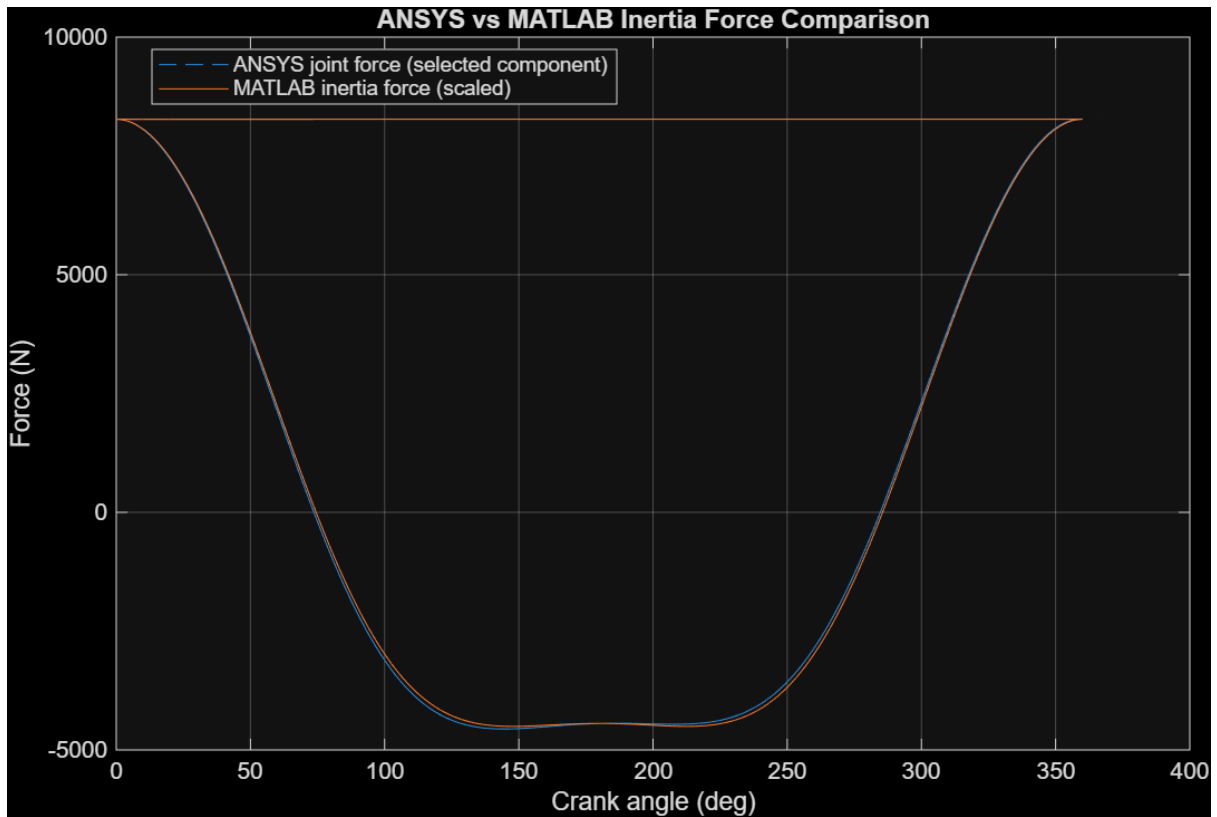


Figure 1.4: MATLAB analytical result vs ANSYS simulation result.

Step time calculation

This equation is used to find the time step per crank-angle increment from the engine speed.

First, the engine speed in RPM is converted to revolutions per second dividing it by 60. This gives the number of crankshaft revolutions completed each second.

Next, the time required for one full revolution is obtained by taking the reciprocal of that value, which gives the number of seconds per revolution.

Finally, the time per degree of crank rotation is found by dividing the time per revolution by 360.

So the time step at 6500 RPM is approximately 2.56×10^{-5} seconds per degree.

$$t_{step} = \frac{1}{360 * \left(\frac{RPM}{60}\right)}$$

Steady-state thermal

The study is linked to the engineering data and to the geometry from the rigid dynamics analysis. Within the studies, only the connecting rod is investigated, all other components are suppressed in the geometry tree. A mesh of 1mm element was created. A temperature of 150 °C is applied to the inner surface of the small end of the connecting rod where it interfaces with the gudgeon pin, and a convection boundary condition with an ambient temperature of 100 °C is applied to the remaining surfaces of the rod to simulate its operating temperature distribution.

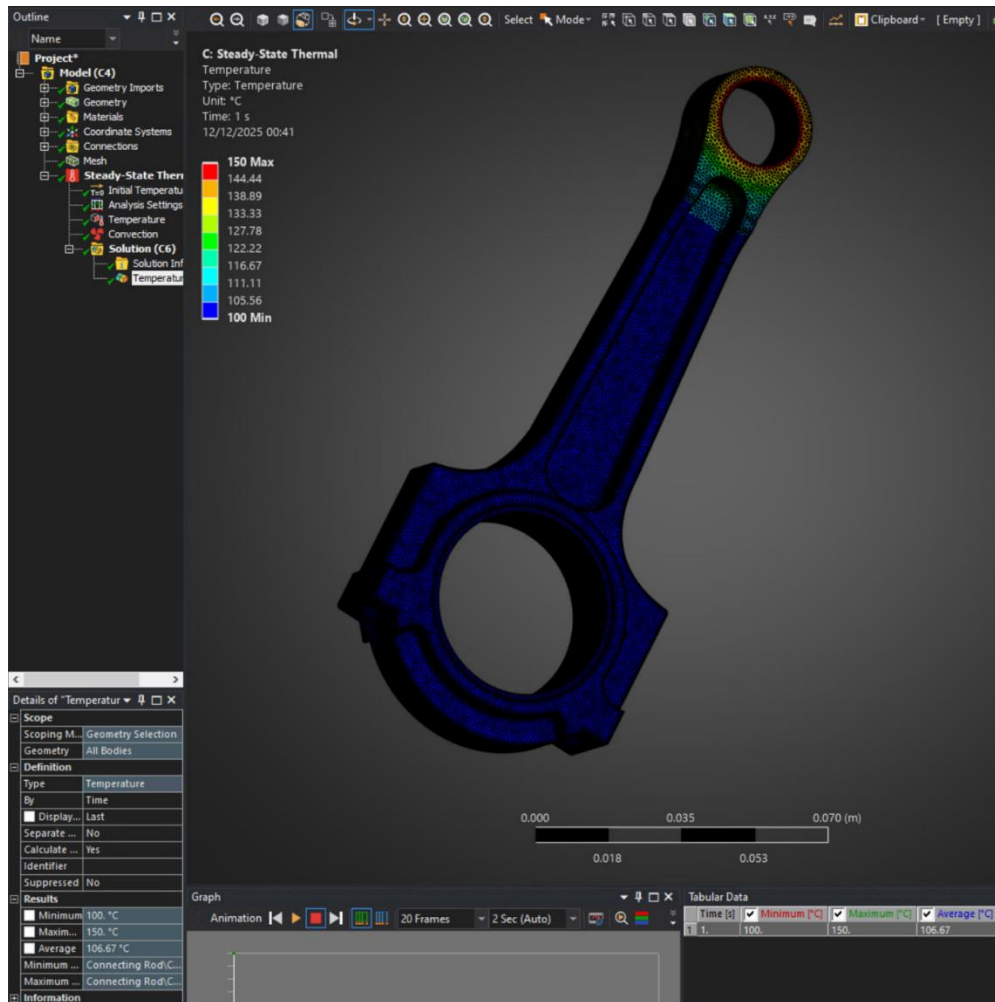


Figure 1.5: Steady state thermal study

Static structural

To obtain the two critical load cases (maximum compression and maximum tension) in the static analysis, the engineering data, geometry, and model are imported from the rigid dynamics study, and the temperature distribution from the thermal study is mapped onto the connecting rod.

Only the connecting rod is investigated in this analysis; all other components are suppressed. The connecting rod stiffness behaviour is changed from rigid to flexible to enable prediction of elastic deformation under the applied loads. The component is discretised using a finite element mesh with a nominal element size of 1 mm.

For the TDC load case, the motion loads corresponding to TDC are imported. Cylindrical supports are applied at both the big end and the small end of the connecting rod, with the radial direction set to free.

For the peak gas-pressure load case at a crank angle of 14 degrees, the corresponding motion loads are imported. A cylindrical support is again applied at the big end, with the radial direction free. Additionally, the calculated load acting at the small end of the connecting rod is applied based on the calculated rod angle.

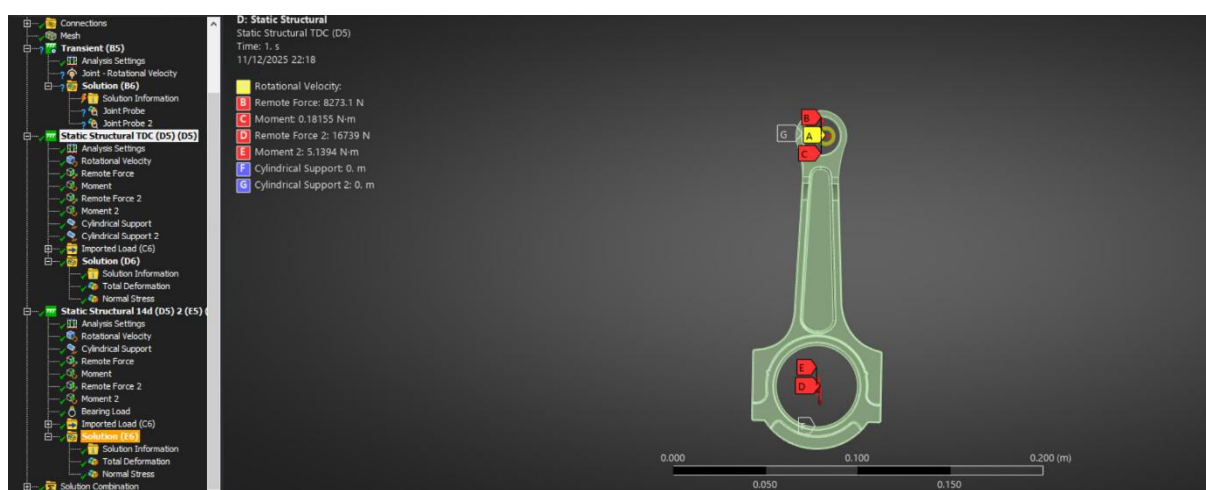


Figure 1.6: Steady state thermal study

Calculate rod angle

Stroke data were obtained from provided specifications for the Ford Sigma engine. The connecting rod length was measured directly from the provided CAD model. The crank angle was defined relative to the crank position at which maximum gas pressure occurred in cylinder one at an engine speed of 6500 RPM.

Parameter	Symbol	Value
Crank angle	θ	14°
Crank radius (stroke/2)	R	40.7 mm
Connecting rod length	L	135.15 mm
Rod angle	ϕ	4.18°

Table 1.1: Input parameters used to calculate connecting rod angle

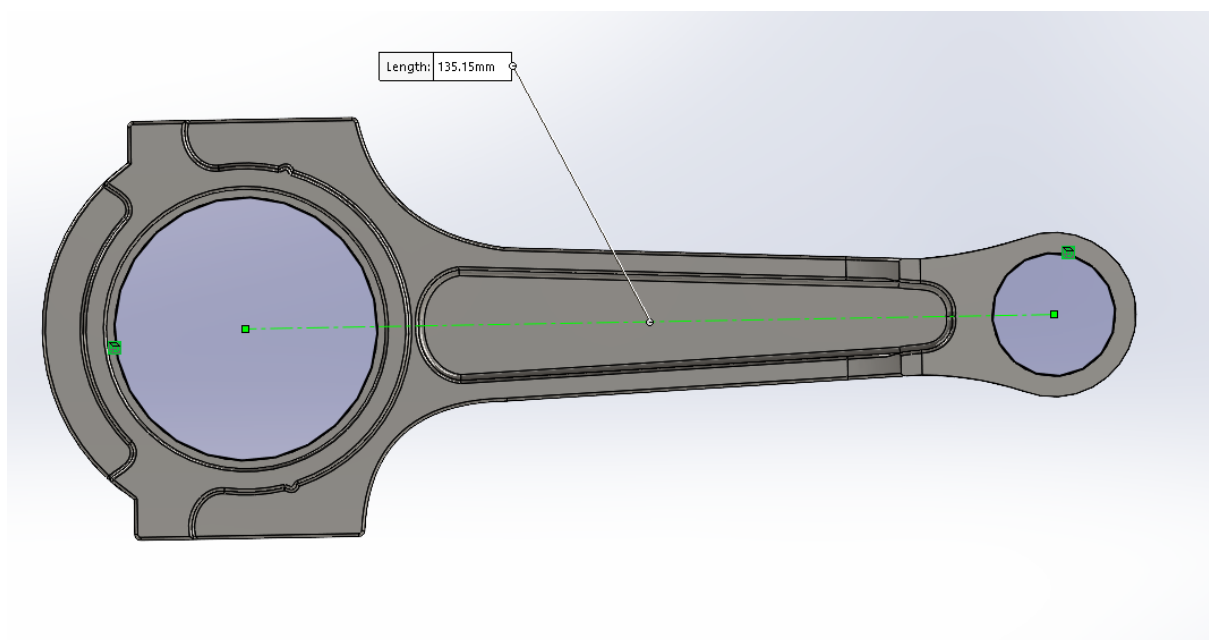


Figure 1.7: Connecting rod length

$$\phi = \sin^{-1}\left(\frac{R \sin \theta}{L}\right)$$

Calculate force on the connecting rod for the gas pressure

Cylinder bore was from provided specifications for the Ford Sigma engine. The max gas pressure occurred in cylinder one at an engine speed of 6500 RPM.

Parameter	Symbol	Value	Units
Cylinder bore	D	79 (0.079)	mm (m)
Gas pressure	P_gas	72.631	bar
Gas pressure	P_gas	7.2631×10^6	Pa
Gas force	F_gas	35.6	kN

Table 1.2: Input parameters and calculated gas force (F_{gas})

$$F_{\text{gas}} = P_{\text{gas}} \cdot A_{\text{piston}}$$

$$A = \frac{\pi d^2}{4}$$

Fatigue tool

The solutions from both loading cases were combined to obtain a fatigue result that accounts for the two extreme operating conditions.

The fatigue analysis was then used to evaluate the component life and factor of safety

2. Boundary Conditions & Load Cases

Materials

Different engine components have different materials data export from Granta Edupack. The selection of these materials is substantiated by the research findings, which demonstrate their suitability for use in a conventional internal combustion engine.

In rigid dynamics study. While the component treated as infinitely stiff with no deformation, the density and mass could affect the realistic dynamics.

List of materials been applied:

Connecting Rod (SteelExpress, n.d.)	Low alloy steel, aisi 4130 annealed
Piston (Garbe, 2024)	Aluminum, 2618 T6
Gudgeon Pin (Myagkov, et al., 2014)	Low alloy steel, AISI 5140 normalized
Crankshafts (Smith & Jones, 2015)	Carbon Steel, AISI 1040 annealed

Table 1.3: Materials Used for Engine Components

Engine speed

According to the Ford Sigma engine specifications, maximum power occurs at 6100 rpm; therefore, the top dead centre (TDC) condition was selected at 6500 rpm to represent a conservative high-speed operating case. Although the maximum torque is reported to occur at 4100 rpm in the specification data, the provided Excel data indicate that the maximum cylinder gas pressure occurs at 6500 rpm in cylinder one.

Consequently, the maximum gas pressure at this operating condition was used to calculate the maximum force acting on the connecting rod.

Temperature

connecting rods for the automotive industry through powder forging was evaluated at engine operating temperatures, ranging between 100 °C and 150 °C (Ilia, et al., 2019)

Elastic deformation (stiffness-controlled): expect only a **small increase** in deformation (typically a few percent) when going from **100 °C to 150 °C**, assuming the same load and boundary conditions. Supported by both Eurocode reduction factors and AISI 4130 modulus-vs-temperature measurements. (Standardization, 2005)

Although the effect of temperature on the connecting rod is expected to be negligible, a small influence on its elastic behaviour remains. Therefore, a steady-state thermal analysis was conducted to account for the potential effects of operating temperature on the elastic response of the component.

Mesh sizing

A global mesh size of 1 mm was applied to the connecting rod, as the geometry contains a moderate level of detail while maintaining an acceptable solution time. A mesh sensitivity study was performed to evaluate the effect of mesh refinement on the results and to ensure that the analysis was not significantly influenced by the chosen mesh size. (Software, 2016)

Gas pressure

The Excel data indicate that the mean gas pressure is highest at 4100 rpm. however, the maximum instantaneous gas pressure occurs at 6500 rpm in cylinder one. Therefore, the gas pressure data at 6500 rpm were used in this analysis, as they represent the most severe loading condition for the connecting rod.

piston side thrust on the cylinder wall

The piston side thrust acting on the cylinder wall was not applied in the static structural analysis of the connecting rod, as this force is a reaction at the piston–cylinder interface and does not constitute an external load acting directly on the connecting rod.

No gas pressure at TDC

To simulate the worst-case scenario, the condition at top dead centre was considered with no cylinder gas pressure acting to counteract the inertial forces. Under this condition, the inertial load reaches its maximum value and represents the most severe loading case for the connecting rod.

3. Fatigue Life & Safety Factor Comparisons

Fatigue life

The fatigue life contour in ANSYS shows that the predicted minimum life of the component is at least 1×10^8 cycles. This does not mean that the component will fail at this number of cycles. Instead, it indicates that the fatigue result has reached the upper life limit (design life cap) defined in ANSYS. Therefore, the connecting rod can be considered to have an **infinite fatigue life** for this case study.

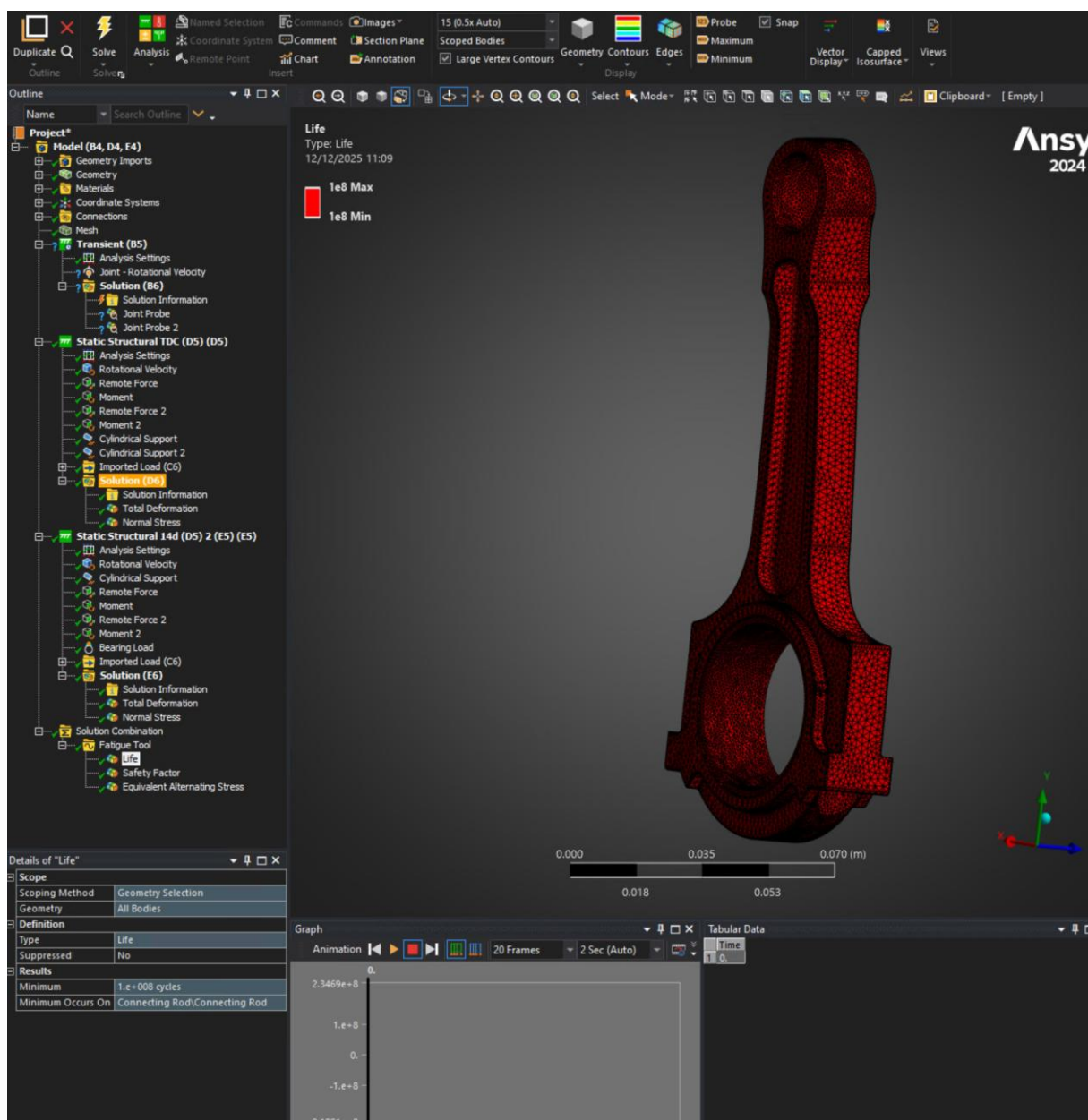


Figure 3.1: Fatigue life contour for the combined extreme cases (overview).

Safety Factor

The minimum safety factor was 1.0889, indicating the design is marginally safe under the combined loading. A value close to unity suggests the fatigue response is governed by **local stress concentrations**, typically at geometric discontinuities such as fillets and bore transitions, where stress gradients are highest.

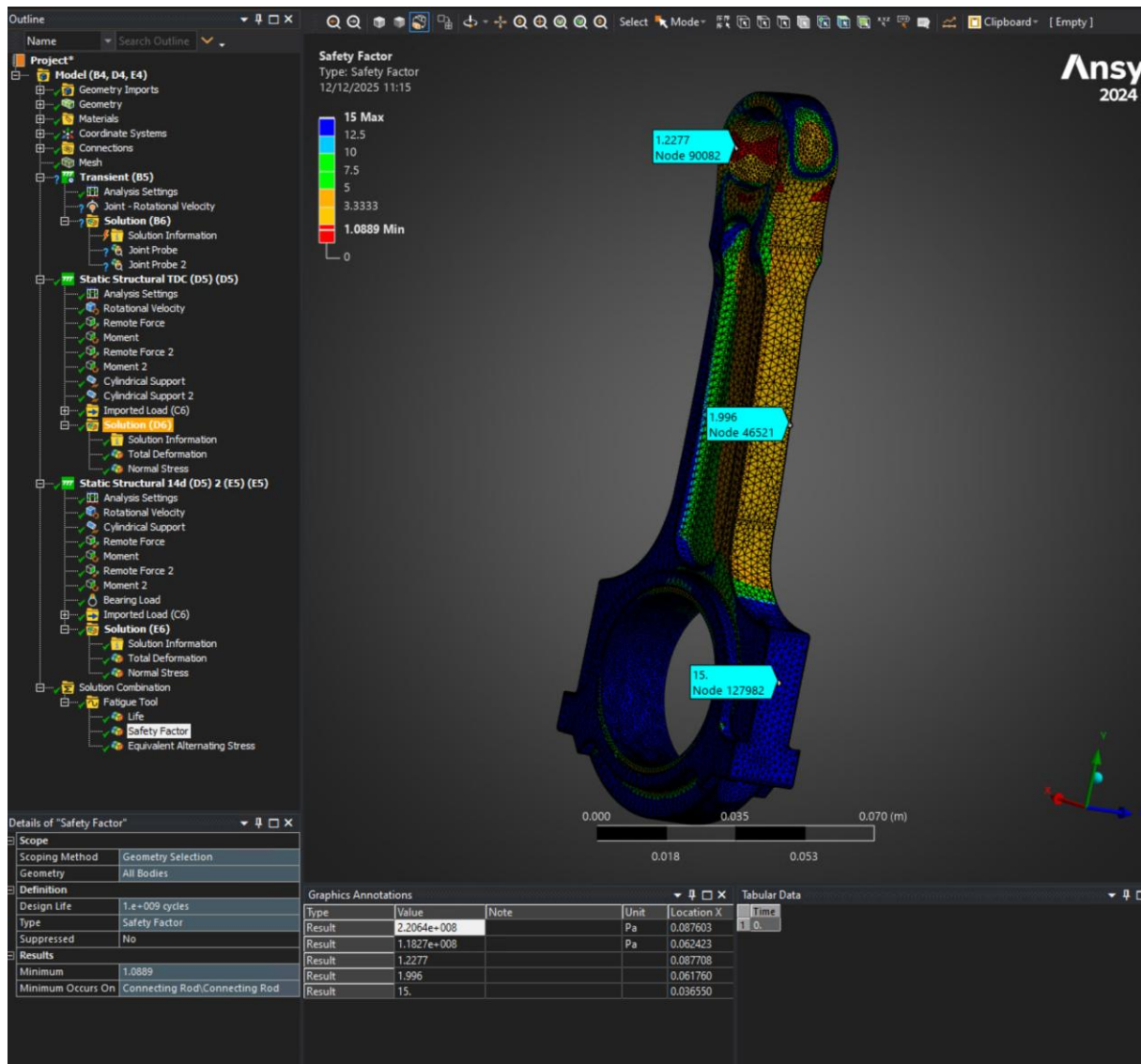


Figure 3.2: Fatigue safety factor contour for the combined extreme cases.

Concussion

the fatigue results indicate that the connecting rod meets a high-cycle fatigue life requirement of $\geq 1 \times 10^8$ cycles, but with a **low safety margin** (minimum safety factor ≈ 1.09). This implies that small changes in loading, stress concentration, or material fatigue properties could reduce the safety factor below unity, and therefore the critical regions identified by ANSYS should be treated as design-driving locations for further refinement

4. Discussions & Critical Evaluation

Geometry and interface simplifications

Suppressing the piston pin, crank pin, bearings, and fasteners removes interface compliance and can alter load transfer. As a result, local stresses at bores and fillet transitions may differ from those in the full assembly, and fatigue hotspots could shift if realistic contact and joint behaviour were included.

In a rigid dynamic analysis, the mass properties of the component are critical because inertial forces are directly proportional to mass. However, the piston geometry used in this study does not match the cylinder bore specified for the Ford Sigma engine. This mismatch may lead to non-representative mass and inertia properties and therefore introduces uncertainty in the predicted inertial loading and the validity of the dynamic results.

Boundary conditions and load application

Remote loads and idealised supports do not fully represent distributed bearing pressures or evolving joint reactions through the crank cycle. Peak stresses can therefore be sensitive to constraint definition, which is particularly important here because the minimum safety factor is close to unity.

Fatigue representation and settings

Peak gas and peak inertia occur at different crank angles; using only two extremes may not capture the true stress history, mean stress, or stress ratio. Fatigue outputs are also sensitive to mesh resolution at stress raisers and fatigue-tool inputs

5. References

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6. Appendix

Axial force in the connecting rod (from gas load)

$$F_{\text{rod}} = \frac{F_{\text{gas}}}{\cos \phi} = \frac{35.6 \text{ kN}}{\cos (4.18^\circ)} \approx 35.7 \text{ kN}$$

$$\boxed{F_{\text{rod}} \approx 35.7 \text{ kN}}$$

Fgas is close to Frod. Fgas was used in the simulation instead of Frod.

side thrust on the cylinder wall (from gas load)

$$F_{\text{side}} = F_{\text{rod}} \sin \phi = F_{\text{gas}} \tan \phi \approx 2.60 \text{ kN}$$

$$\boxed{F_{\text{side}} \approx 2.60 \text{ kN}}$$

Load on con rod

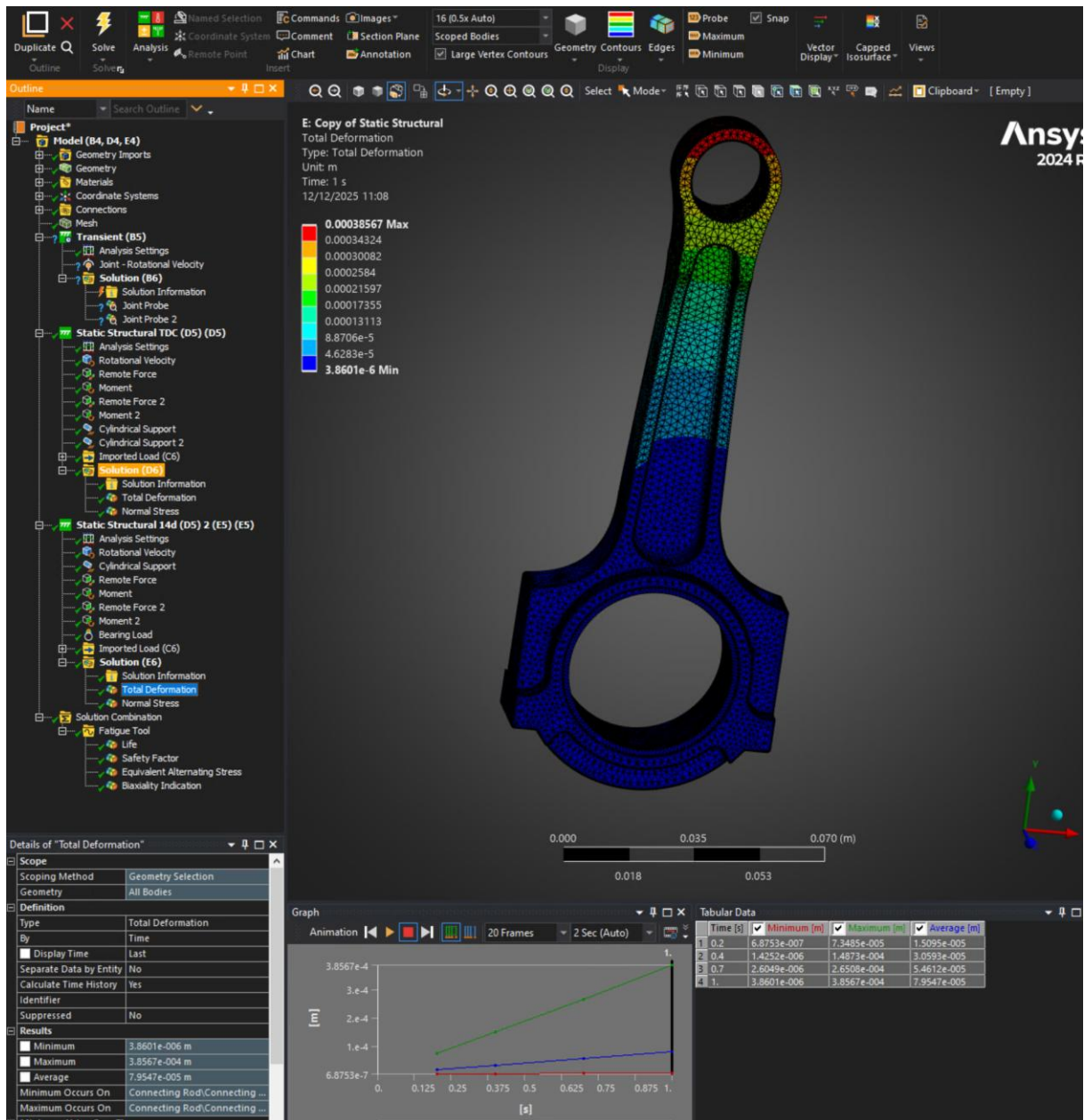
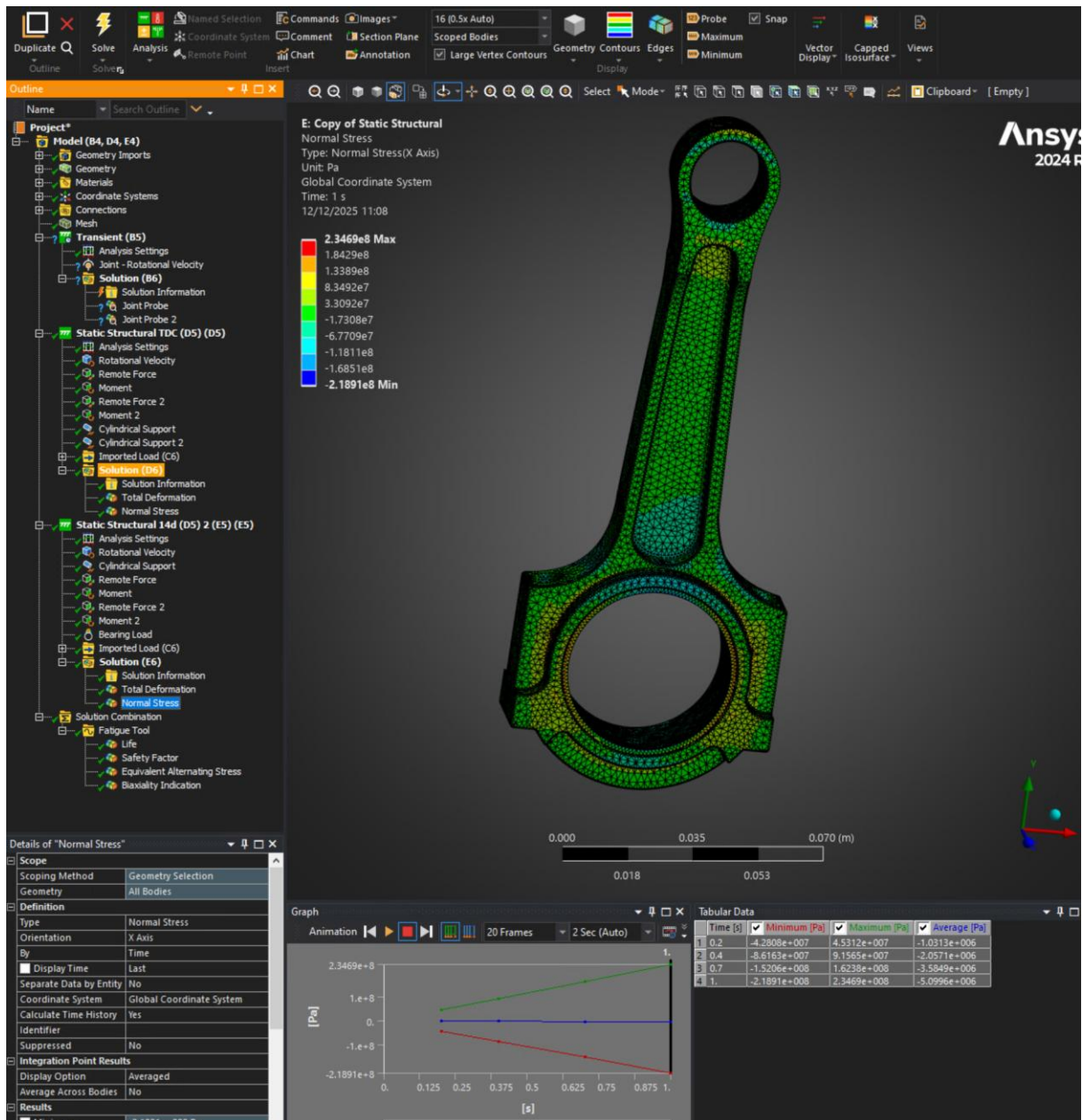
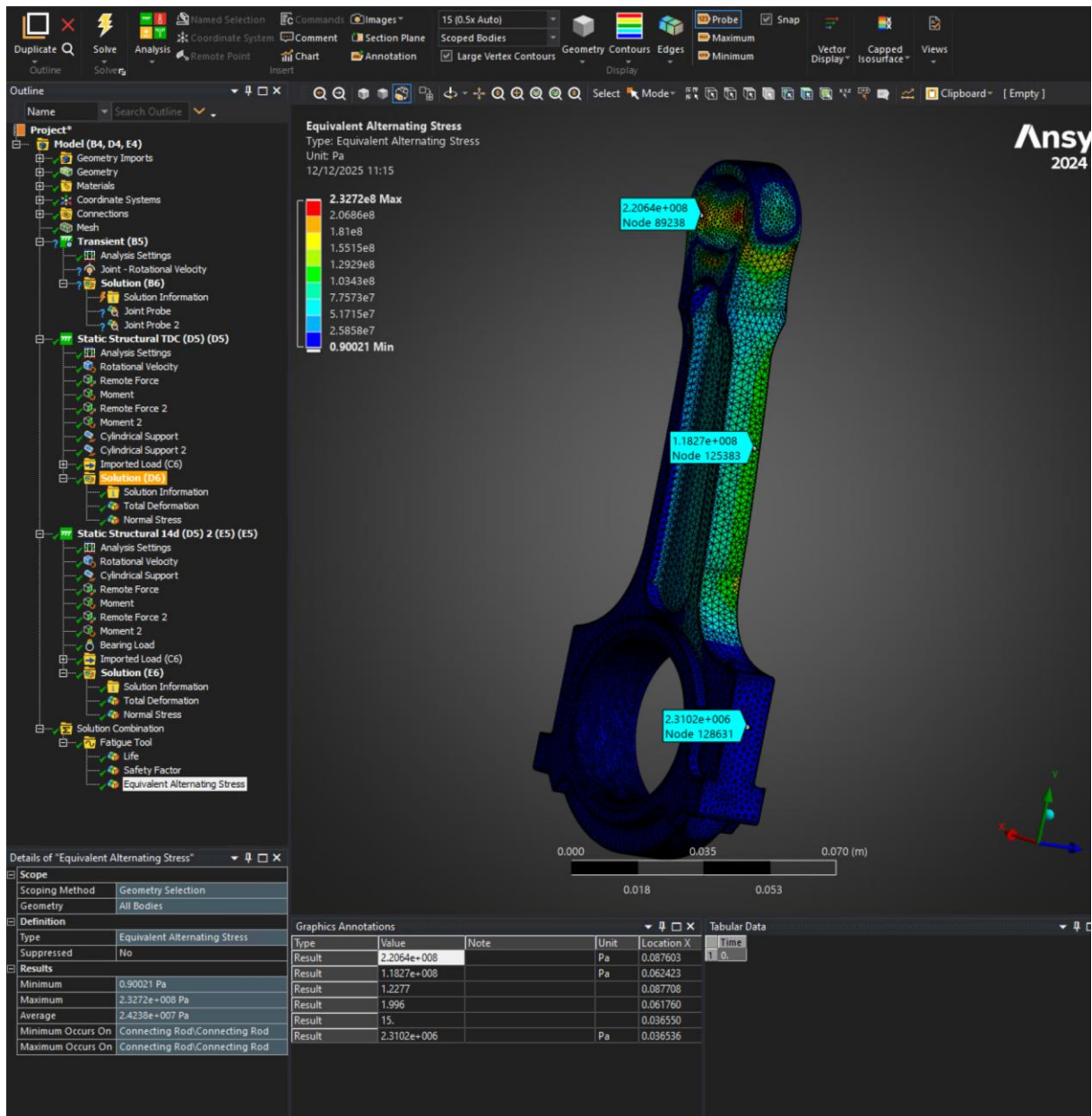


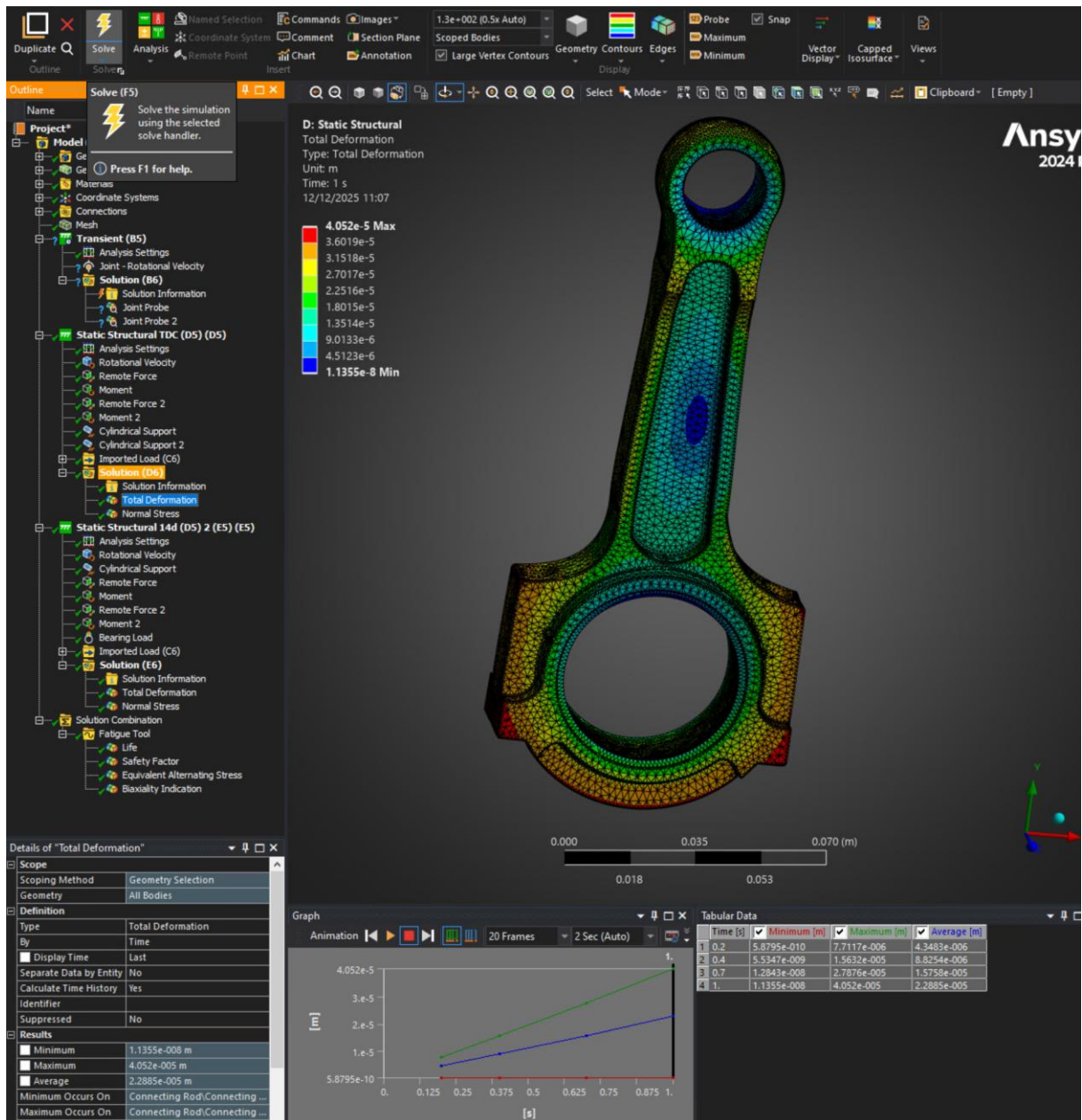
Figure 3.1: Total deformation contour for the peak gas-load case ($\theta \approx 14^\circ$, 6500 rpm).



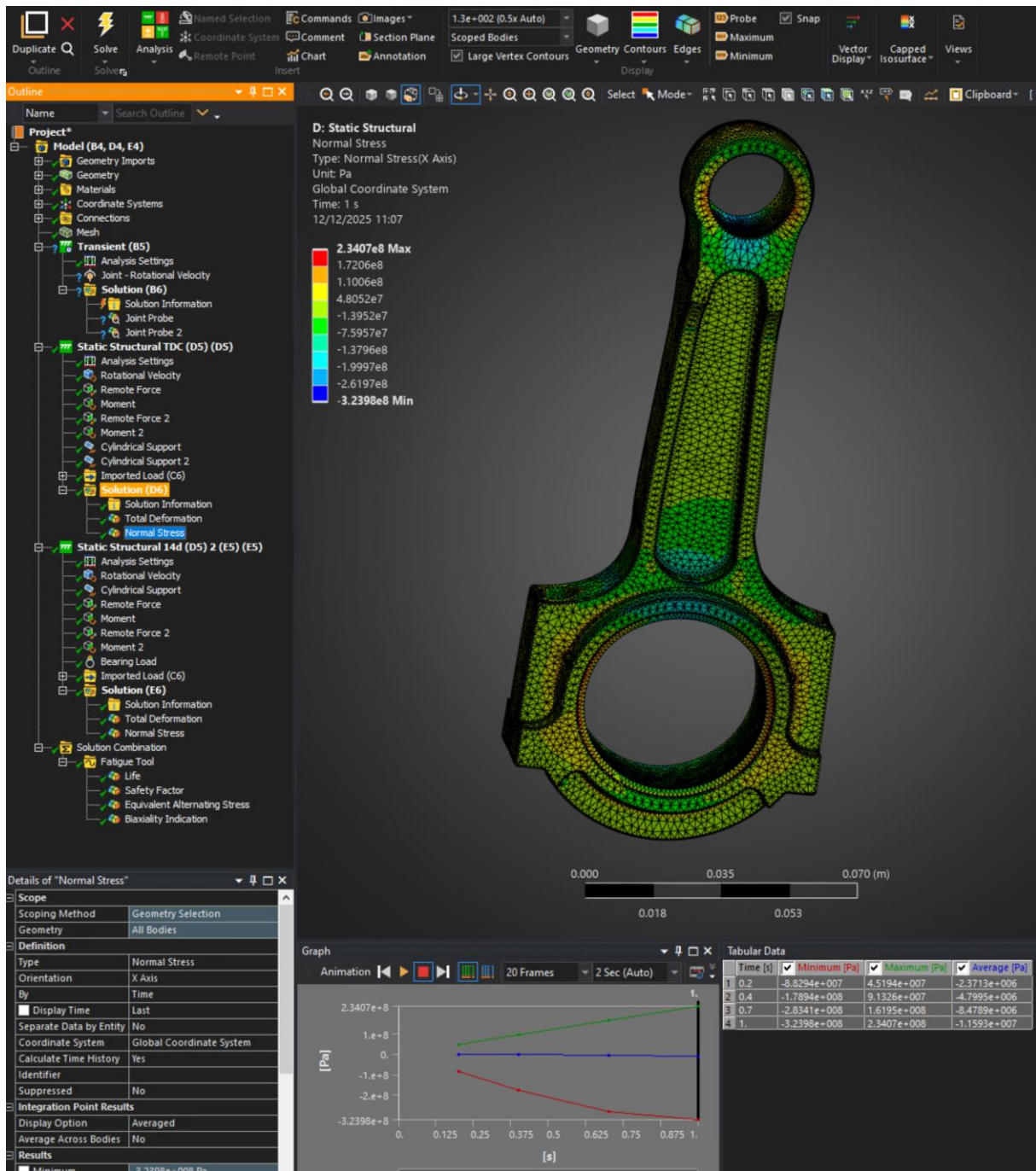
Normal stress (X-axis) contour for the peak gas-load case ($\theta \approx 14^\circ$, 6500 rpm).



Equivalent alternating stress contour for the combined extreme cases.



Total deformation contour for the TDC inertia case (gas pressure neglected).



Normal stress (X-axis) contour for the TDC inertia case (gas pressure neglected).

