

University of Wales Trinity Saint David

Wales Institute of Science & Art



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

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

Structural Component Analysis (ACHA7030)

Koa Brook

2203480

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1. Methodology & Workflow

1.1 Meshing Strategy

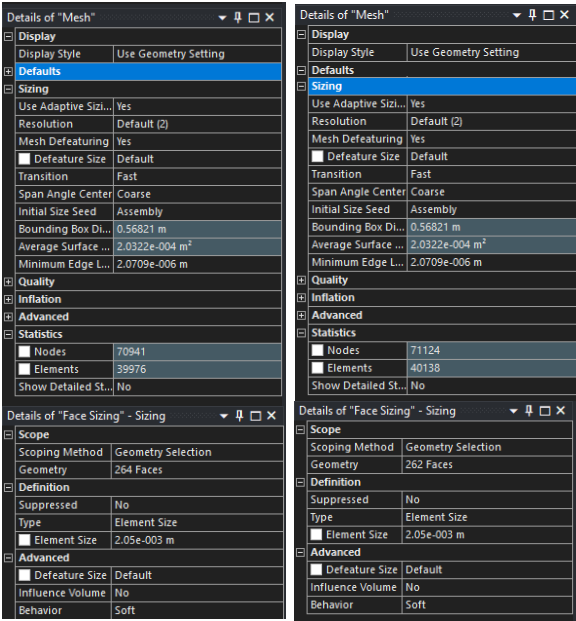


Figure 1: thermal
mesh details

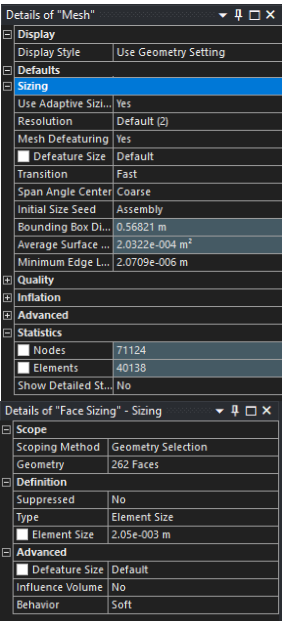


Figure 2:
structural mesh



Figure 3: thermal mesh

A meshing approach based on Gopinatha & Sushma (2015) was used, applying a 2.05 mm element size across the connecting rod. This produced a mesh comparable to the referenced study but with higher node density, ensuring adequate resolution in critical regions (figures 1–3).

1.2 Solver Workflow

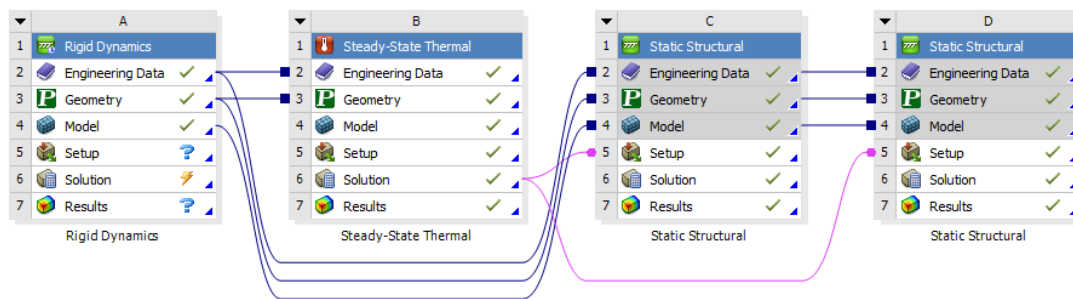


Figure 4: ANSYS Workbench FEA workflow

The workflow (figure 4) consisted of:

- a rigid dynamics simulation to obtain inertia loads,
- a thermal analysis to determine operating temperatures,
- two static structural studies - one for tensile inertia loading and one for compressive pressure loading,
- a fatigue analysis combining both load cases using the Goodman method.

1.3 Material Properties

5	 Low alloy steel, AISI 4140, normalized	3	 Aluminum, EN AC-48000, chill cast, T6
6	 Low alloy steel, AISI 5115, annealed	4	 Carbon steel, AISI 1040, normalized

Figure 5: ANSYS Workbench engineering data materials (imported from Granta)

Materials were selected for each component based of research (figure 5), for the components in the assembly that were not used in the whole study (crank, guidon, piston) sensible materials were selected however the main reason behind this was so that an accurate mass was used by the system for calculating the inertia loadings. The connecting rod (component studied) used AISI 4140 steel, this was determined based of the statement made in Mitchell Repair Information Company, LLC (n.d.), and using MAHLE International GmbH, (2016) as guidance an appropriate material was selected from ANSYS Granta. In ANSYS Granta it contains material properties and common uses,

as one of the common uses for AISI 4140 steel is connecting rods this aids in justifying the material choice.

1.4 Analytical Calculations

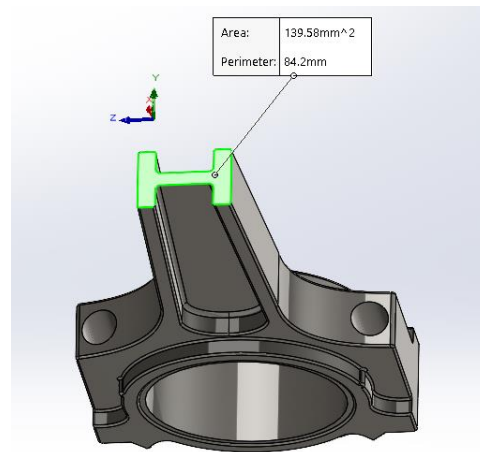


Figure 6: Cross-sectional area of connecting rod (Solid works)

Analytical fatigue assessment used the Goodman approach, consistent with the FEA fatigue tool. The minimum cross-sectional area (figure 6) was taken from SolidWorks, and inertia forces were **calculated** from the component masses and engine speed to obtain the maximum tensile load. The maximum compressive load was **calculated** from the cylinder-pressure trace. These tensile and compressive extremes were then used to determine the mean and alternating stresses, with a UTS range of 915–1130 MPa from Granta, the lower value was used in the Goodman evaluation.

2. Boundary Conditions & Load Cases

2.1 Mechanical Boundary Conditions

During the rigid dynamics study the crank had a revolute connection to ground allowing it to rotate without moving position, the piston had a translational joint to ground allowing it to move back and forth in one direction. The guidon pin had a cylindrical connection to the piston guidon pin holes, this allowed it to rotate but not translate, all other components were connected to each other with revolute joints so they could revolve about each other like real world operation. None of the joints used considered any frictional loadings, as that would increase computational load with negligible impact on the component being studied in terms of stress and loading. For the static structural studies, the rod had a cylindrical constraint at the crank end that also allowed for true deformation of that end of the component.

2.2 Pressure Loading

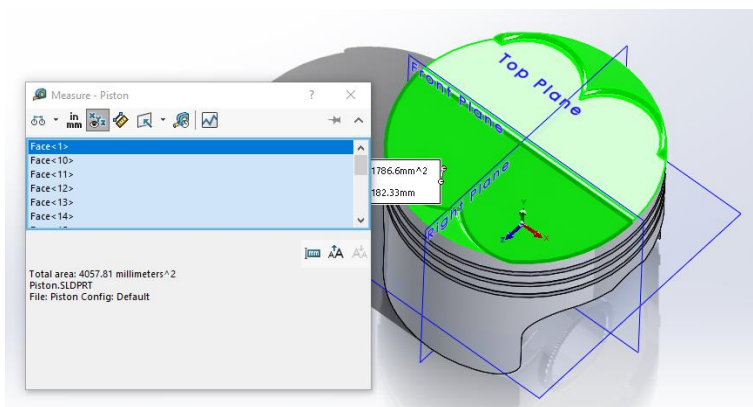


Figure 7: piston crown area (Solid works)

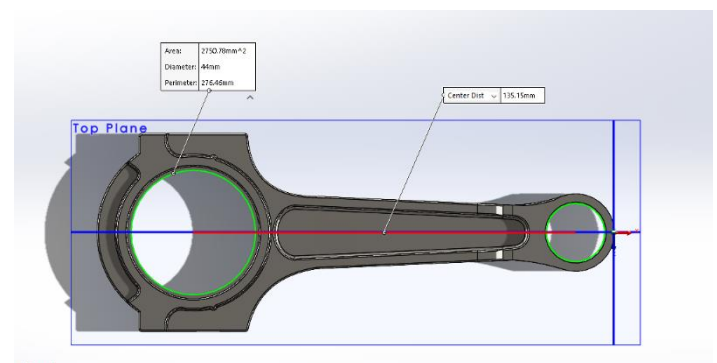


Figure 8: connecting rod length (Solid works)

To determine the loading on the connecting rod due to the pressure the force required calculating, to do this the piston crown are was found in SolidWorks (figure 7).as pressure is force acting upon an area, the compression force acting upon the connecting rod from the cylinder pressure was then calculated by multiplying the max

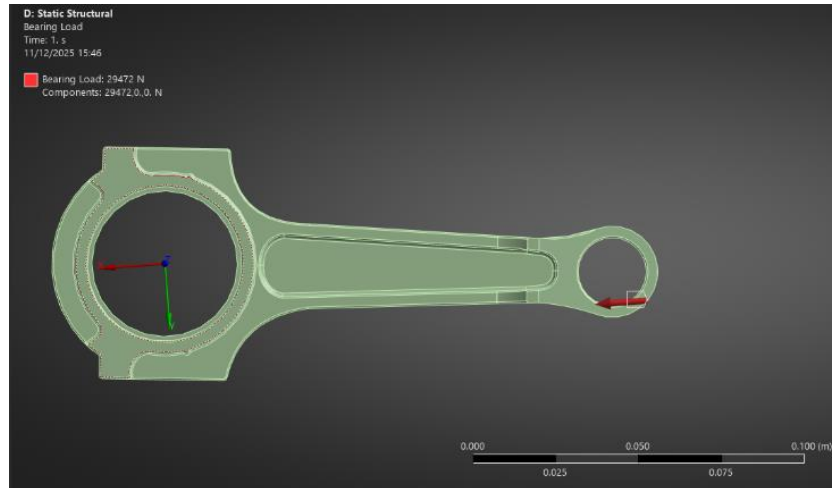


Figure 9: direction of force pressure

cylinder pressure (72.63 bar @ 6500RPM) by the area of the piston crown. This force was then applied as a bearing load on the guidon pin surface, the force was applied at an angle of 4.18° (shown in figure 9) as the pressure acts upon the piston when the crank angle is at 14° ATDC. The angle at which the pressure acts upon the connecting rod was calculated using $\phi = \sin^{-1} \frac{R \sin \theta}{L}$, where θ is crank angle, L is connecting rod length (figure 8), R is crank radius, and ϕ is connecting rod angle. The use of a bearing load rather than a force applied to the inner top end surface means that the load is accurately applied and has distribution across the surface similar to real world applications. Note that this method doesn't account for any force that the guidon pin absorbs due to its own deformation and stress, so this method marginally overestimates the stress acting upon the connecting rod.

2.3 Thermal Load Conditions

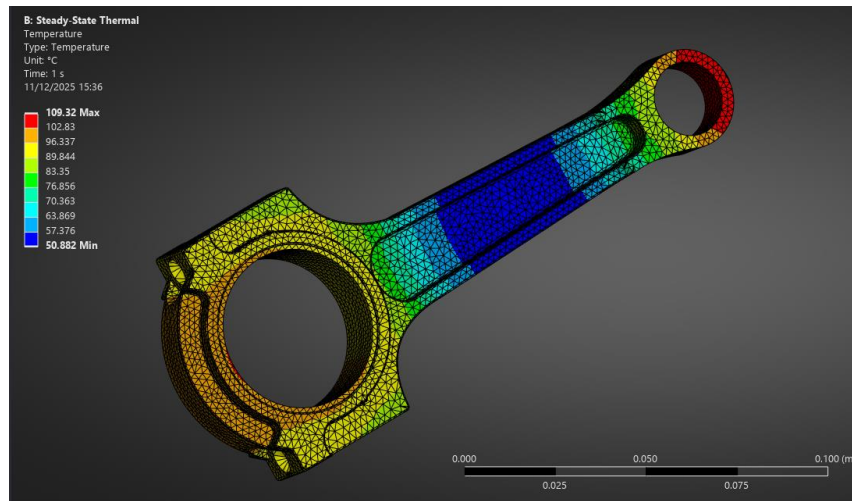


Figure 10: connecting rod temperature distribution

Thermal loading has limited influence on steel connecting rods, as much higher temperatures are needed to significantly affect Young's modulus. Nevertheless, relevant temperature and convection data were applied for completeness. A bulk oil temperature of 150 °C was taken from Ford Power Products Ltd (n.d.). The oil-to-steel convection coefficient was taken as 700 W/m²·°C from Modi et al. (2021), based on crankcase oil-gallery analysis. For the remaining surfaces, a convection coefficient of 110 W/m²·°C and an air temperature of 22 °C were taken from Kolev et al. (2024). These values were applied at the bearing surfaces (oil) and across the rest of the rod (air). As no data were available for crankcase air temperature, ambient conditions were used, which likely underestimates temperature but has minimal effect given the rod's material properties. The model also does not include oil spray effects.

2.4 Kinematic Loads (Inertia)

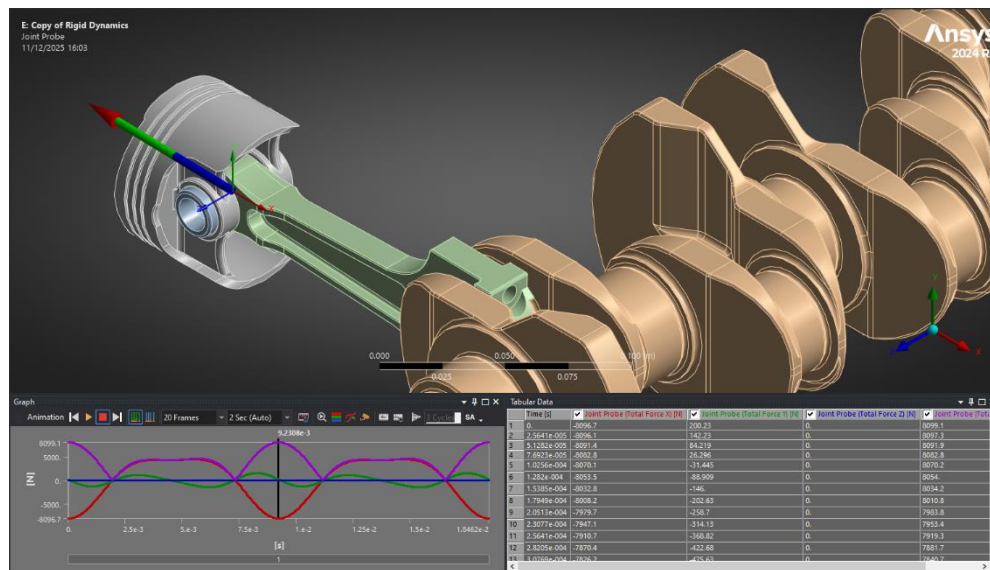


Figure 11: Assembly and inertia forces at TDC

For the kinematic loads the highest RPM of the data given was used (6,500 RPM), this will give the maximum inertia loading which is experienced at TDC (figure 11) due to the significant change of direction of the piston and guidon pin. This gives the maximum tensile loading that the connecting rod experiences for the fatigue study.

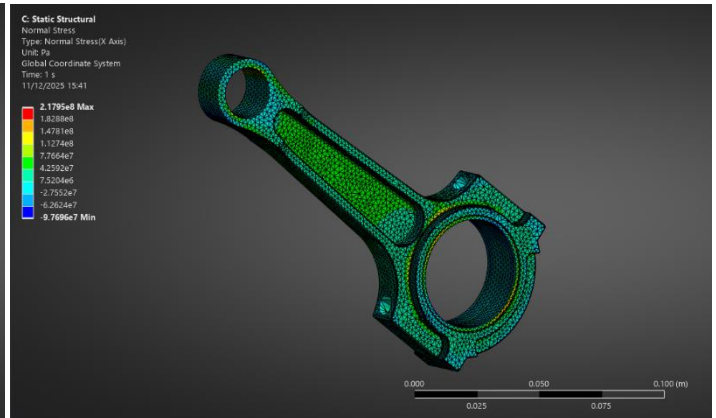
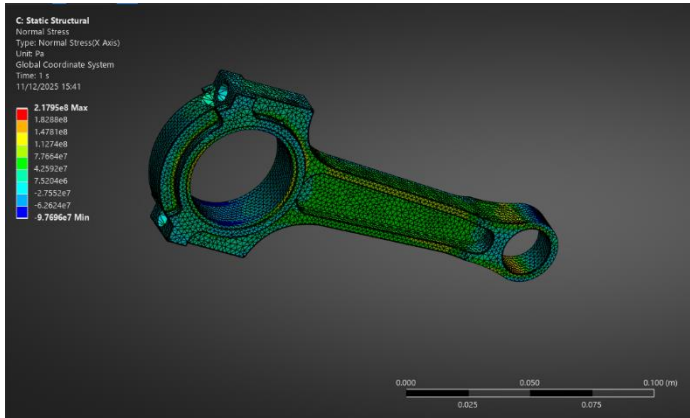
2.5 Critical Evaluation of Load Validity

The accuracy of the inertia loadings is dependent upon component materials for the guidon pin and piston, the material selected will change the mass and therefore the inertia loading. As the materials chosen are supported by research the inertia loadings can be deemed representable of real-world scenarios. The pressure and RPM data were measured from a real engine and therefore can be understood as reliable and be trusted. The thermal loadings are from a larger engine and therefore may be an overrepresentation of the thermal conditions of the component which would increase stress and deformation.

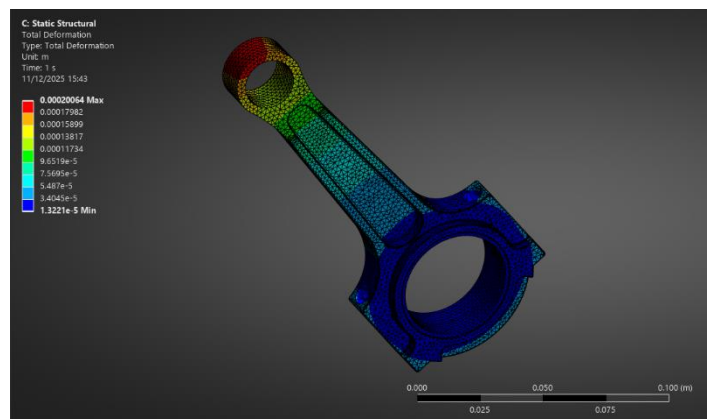
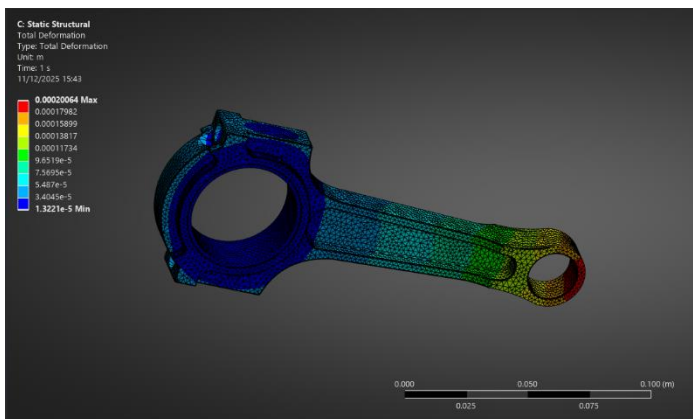
3. Fatigue Life & Safety Factor Comparisons

3.1 Stress Results

3.1.1 Inertia (Tensile)



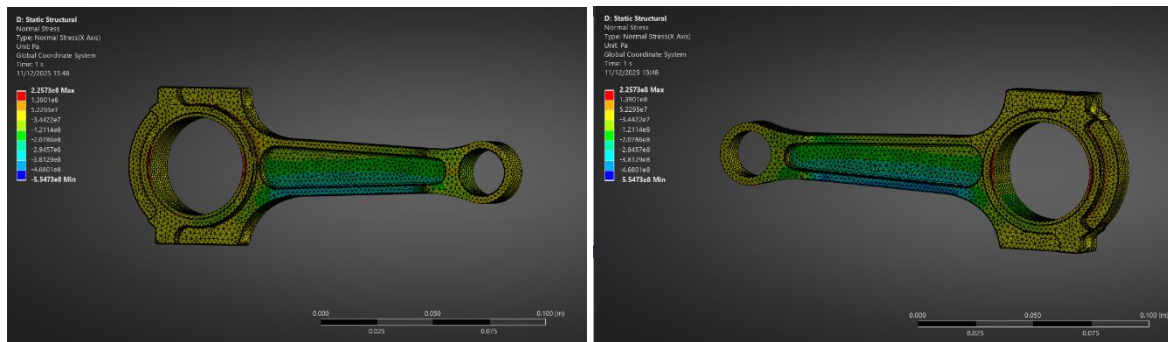
Figures 12 & 13: stress due to inertia loadings



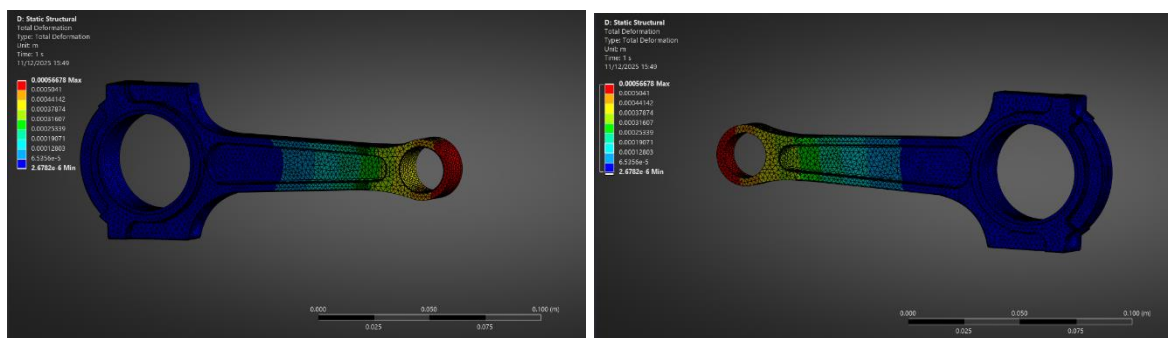
Figures 14 & 15: deformation due to inertia loadings

Figures 14 & 15 show the deformation and stresses acting upon the connecting rod due to inertia loading. We can see that these values are relatively small, this is as the main masses acting on the rod are the guidon pin and the piston. These masses are acting at the smaller end of the connecting rod with maximum deformation centring at the top loop of the rod and the stresses concentrating at the internal edges of the rod (figures 12 & 13). This is as this is the thinnest area of the rod and would expect this part to deform first under tensile loading conditions. We see maximum deformation at the upper edge as the stress on the side edges stretch in the direction of tension moving the upper edge the most.

3.1.2 Pressure (Compression)



Figures 16 & 17: stress due to pressure loadings



Figures 18 & 19: stress due to pressure loadings

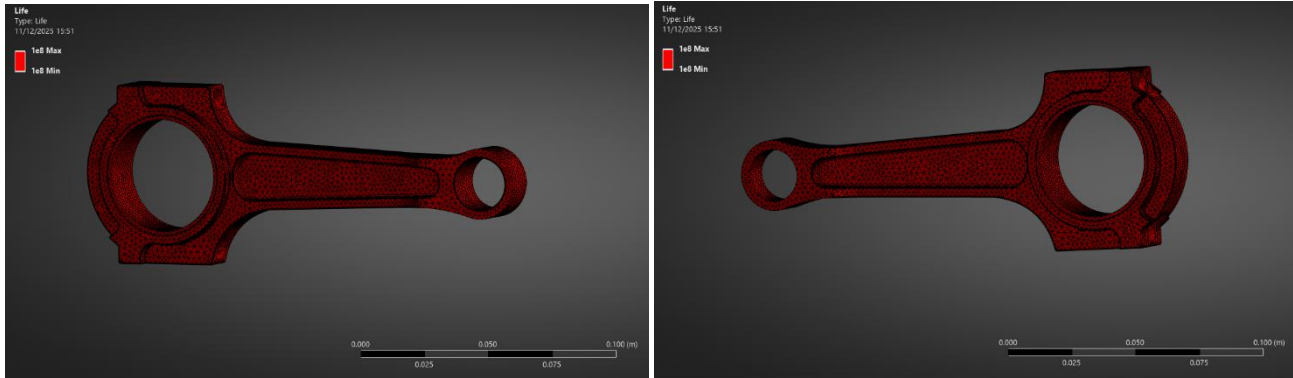
Figures 16 & 17 show the stresses acting upon the connecting rod due to the forces from cylinder pressure, the stresses can be seen as both positive (tensile) and negative (compressive). The asymmetric distribution of stress is due to the pressure acting upon the conrod at an angle rather than normal to it, so one side of the connecting rod is experiencing a compressional stress and the other is experiencing a tensile stress. This force is attempting to bend the connecting rod towards the underside, the high deformation on the edge of the top hoop (figures 18 & 19) may be due to the force focusing on the side edge (perpendicular) deforming this part due to stress tension as it has a lower CSA.

3.2 Fatigue Tool Setup

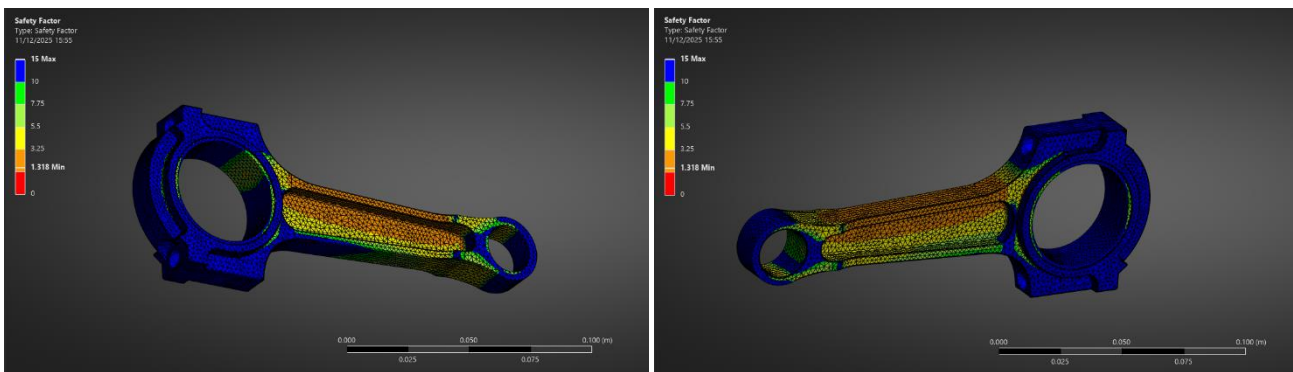
The fatigue tool was setup to use the Goodman method as it is known for giving results for compressional and tensile loadings that represent real world scenarios. The two loading cases used was the inertia force from the reciprocating masses (guidon pin and piston) and then the force resultant of the maximum cylinder pressure. This gives the

tool its maximum loading cases to determine the fatigue safety factor (FSF), and number of cycles the component can withstand.

3.3 Fatigue Life Results (ANSYS)



Figures 20 & 21: Fatigue life results (1×10^8)



Figures 22 & 23: stress due to pressure loadings

The minimum FSF computed by ANSYS is 1.318 (figures 22 & 23), a fatigue safety factor above 1 indicates an infinite component life, this is supported by the cycle life result that shows a component life of 100,000,000 cycles (figures 20 & 21) which in most cases is considered an infinite life. This is shown in (Wits et al., 2023) where it states “... 10^6 cycles after which infinite life is assumed”, as the connecting rod can withstand more than this its life can be deemed infinite. The lowest area of fatigue safety factor is along the edge of the component which exhibits a compressional stress due to cylinder pressure indicating that the most likely failure would be due to bending, however the component should not fail as displayed by the safety factor staying above one.

3.4 Analytical FSF Calculation

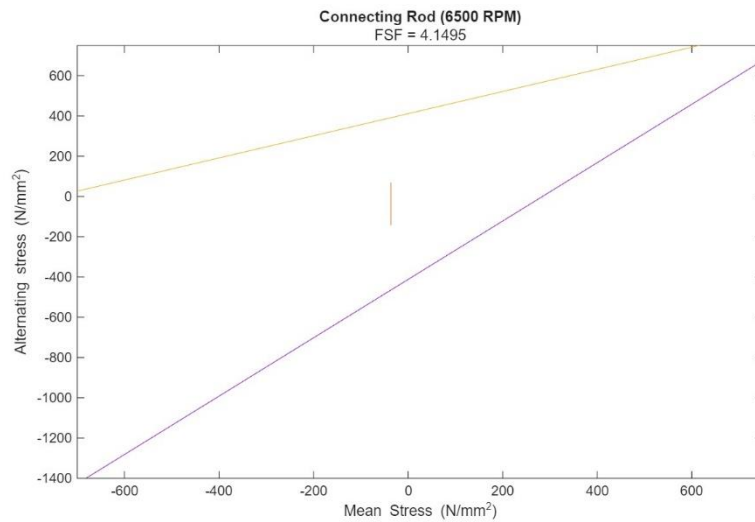


Figure24: Goodman diagram of connecting rod fatigue (analytical)

Figure 24 shows the resultant FSF of 4.15 (2dp), this indicates an infinite component life. This study also accounts for the frictional loadings between components and the changing inertia loadings due to the combustion of other cylinders.

3.5 Comparison: Analytical vs FEA

The analytical study shows a significantly increased value to the lowest in the FEA study, this is as the study finds a value at each node rather than a singular value like the analytical method. Also, the analytical method doesn't account for the angular forces and the bending stress which is why the value is lower in the FEA study. At the same point to what the analytical focuses we can see that if we take an average of values from the FEA the result has a similar value to the analytical study, indicating that had the stresses been in line with the components structure that the analytical would be reflective of the component behaviour. The analytical study doesn't account for component geometry only its CSA, and it also doesn't account for component temperature and the change in Young's modulus due to temperature.

4. Discussions & Critical Evaluation

4.1 Evaluation of Modelling Choices

The meshing strategy gave accurate results to what would be expected from the component in real world behaviour. The values research (convection coefficient, temperature) may not allow for the study to completely reflect real world behaviour, however they're representative and the overestimations allow for a confirmation of fatigue life. As the material specifications were not provided the materials used are representative and are as accurate as possible with research, however with real world materials densities alter changing the inertia loadings as well as other material properties such as Young's modulus. These variations have to be considered with the results and understood that the real world scenario may display differing but similar results.

4.2 Boundary Condition Limitations

The boundary conditions applied introduce simplifications that affect accuracy. Ideal revolute and translational joints ignore friction, clearances and misalignment, all of which influence real load transfer. The cylindrical constraint at the crank end restricts degrees of freedom that may exist during operation due to clearances which allow for some translational freedom, potentially altering local stresses. The bearing load on the gudgeon pin assumes uniform contact pressure, whereas real interactions are non-uniform and dependent on pin deformation as well as roundness. Finally, loads were applied at single crank-angle positions rather than over a full cycle, limiting the representation of transient stresses from multiple cylinders.

4.3 Fatigue Model Limitations

Using the Goodman mean-stress correction gives a simplified fatigue representation suited to uniaxial high-cycle loading. The connecting rod experiences multiaxial, non-proportional stresses, which the model cannot capture. Material S-N data sourced from ANSYS Granta may not reflect the exact alloy or manufacturing process, excluding effects such as forging anisotropy, surface finish and residual stresses. These

assumptions make the fatigue results broadly useful but not fully representative of real conditions.

4.4 Overall Validity of Results

The locations of high stress and the general agreement between analytical and FEA safety factors suggest the modelling approach is reasonable. However, the absolute values are influenced by idealised boundary conditions, assumed material data and simplified thermal loading. The study also considers only peak loading conditions rather than a full operating cycle. While the results indicate infinite-life performance under the defined assumptions, they should be interpreted as indicative rather than definitive.

4.5 Recommendations for improvement

- Apply the full pressure trace over the crank angle to capture realistic loading.
- Include contact and friction at pin and bearing interfaces for more accurate stress distribution.
- Refine mesh locally around fillets and transitions.
- Use validated material and fatigue data specific to the connecting rod.

5. References

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