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Wales Institute of Science & Art



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

Structural Component Analysis (ACHA7030)

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Date (12/2025)

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1.0 Introduction

This report contains the discussion of the methodologies for analysing the connecting rod of a Ford Sigma Engine 1.6L. The connecting rod and assembly seen in Figure 1 were provided by Andrew Nobel on the 9th December 2026 for analysis using the software ANSYS supplied by the University of Wales Trinity Saint David.

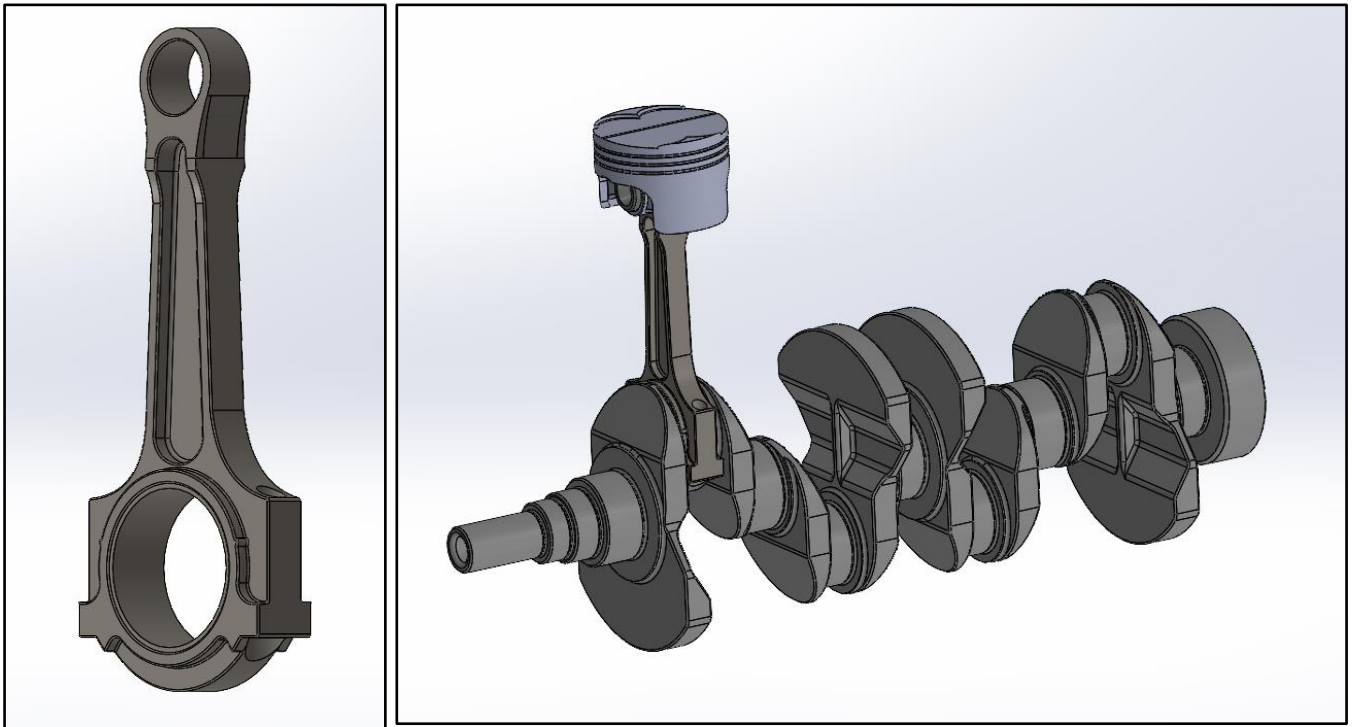


Figure 1: Connecting Rod and Assembly Provided for this Assignment

2.0 Methodology & Workflow

2.1 Research

2.1.1 Engine Specifications

Some of the engine specifications were given in the files for this assignment. Other information was collected from a range of sources.

Specifications	
	Metric
Bore	79 mm
Stroke	81.4 mm
Compression ratio	11.1
Max power @ RPM	123 hp @ 6,000 rpm
Max torque @ RPM	118 lb-ft @ 4,100 rpm
Cylinder head	Aluminum
Cylinder block	Aluminum
Camshaft drive	Belt

Figure 2: Given Engine Specifications

The max engine oil temperature for ford 4 cylinder in-line engines was found to be 150°C (Ford Power Products LTD, 2001, p. 8).

2.1.2 Material Decisions

All materials were imported from Granta which is Ansys' database for material properties (Ansys, 2025). It is also important to note that the materials were unlikely to be the exact materials used as companies generally do not release the specific materials. Finally, the book cylinder 'Components: Properties, Applications, Materials' (Vieweg, 2016) was used to inform the material decisions for this assembly.

The material for the connecting rod was low alloy steel, AISI 4140, normalized. This was selected as general research heavily pointed to connecting rod being made out of a forged alloy steel. Mountune website lists the materials for the '1.6L EcoBoost Engine' in its engine specs sheet as 'Forged 4340' however, Mountune is a website tailored towards race tuned engines. Connecting rods in race tuned engines are more likely to be made from AISI4340 over AISI 4140 due to its greater fatigue resistance(Figure3) (Mountune, 2025).Furthermore, connecting rod is listed as one of the typical uses for low alloy steel, AISI 4140, normalized in Granta Figure4

Pistons	Lightweight 79 mm Bore 10:1 Compression Ratio Forged Pistons w/ Nitrided Rings
Connecting Rods	High-Tensile Strength Forged 4340 Alloy Steel H-Beam Rods w/ ARP 2000 Bolts
Crankshaft	Dynamically Balanced Ford OEM Fiesta ST Forged 4340 Steel 81.4 mm Stroke Crankshaft

Figure 3: Mountune Engine Spec Sheet

Low alloy steel, AISI 4140, normalized

Datasheet view: All attributes Show/Hide Find Similar

Metals and alloys > Ferrous > Alloy steels > Low alloy steel > Wrought > AISI 4100 series (Cr-Mo alloyed) > AISI 4140, 41L40 (0.40% C) > AISI 4140 >

General information

Designation ⓘ

Low alloy steel, AISI 4140, normalized

Condition ⓘ	Normalized
UNS number ⓘ	G41400
US name ⓘ	SAE H41400, SAE 4140, SAE 4142, SAE 4140, ASTM J 70, ASTM J 65, ASTM J 55, ASTM 4140, ASTM G41420, ASTM E7140, ASTM B7M, ASTM B16, ASTM 4142H, ASTM 4142, ASTM 4140H, ASTM 4140, ASTM 4137H, ASTM 41, ~SAE 4137H, ~ASTM B4D, ~ASTM 6
EN name ⓘ	~42CrMoS4, ~42CrMo4
EN number ⓘ	~1.7227, ~1.7223
ISO name ⓘ	~ 42 CrMoS 4, ~ 42 CrMo 4, ~42CrMo4
GB (Chinese) name ⓘ	~42CrMo
JIS (Japanese) name ⓘ	SNB22-5, SNB22-4, SNB22-3, SNB22-2

Tradenames ⓘ

Aristology; Ascometal; Incut; Krupp; Lasalle; Ovako; Pompey; Rycrome; Saerstahl; Usaloy

Typical uses ⓘ

Heavy duty shafts, connecting rods, high tensile bolts, propeller shaft joints, track pins

Figure 4: Material for Connecting Rod Data Page 1

The material used for the piston head was aluminium, A332.0, cast T5. A report discussing pistons and piston materials was read, chapter 1.3 of this report states 'Pistons are produced from cast or forged, high-temperature resistant aluminium

silicon alloys' and later went on to state that a silicon content of 12% is a standard piston alloy (European Aluminium Association, 2011). This aligned with the Ford Engine Specification DHE Range (Ford Power Products LTD, 2001) which names an aluminium alloy as the piston material. The alloy selected from Granta was an Aluminium alloy with a silicon content ranging from 11% to 13%, Figure5. Furthermore, pistons is the only listed use.

Aluminum, A332.0, cast, T5		
Datasheet view:	All attributes	Show/Hide Find Similar
Metals and alloys > Non-ferrous > Aluminum > Cast > 3xx.x series (Si, Cu, Mg-alloyed) > A332.0 >		
General information		
Designation ⓘ		
Aluminum, A332.0, cast, T5 ((a): LM13-TE); This designation has been renamed Aluminum, 336.0		
Condition	ⓘ	T5 (Cooled from an elevated-temperature shaping process and artificially aged)
UNS number	ⓘ	A03320, A03360
Tradenames ⓘ		
Alemite; Alminal; Alpax; Alpro; Aluminumbraz; Amdry; Chromet; Permite; Reptec; Semalloy; Sifalumin; Silumin		
Typical uses ⓘ		
Pistons.		
Included in Materials Data for Simulation	ⓘ	✓
Materials Data for Simulation name	ⓘ	Aluminum alloy, cast, A332.0

Figure 5: Material for Piston Head Data Page 1

The material used for the crankshaft was low alloy steel, AISI 4340, annealed. Figure3 listed this as crankshaft material in the engine spec sheets. While crankshafts are not listed as one of the materials typical uses in Granta Figure6 , high tensile applications is named and crankshafts are subject to high tensile load the reciprocating masses of the assemble are one of the factors that contribute to this load.

Low alloy steel, AISI 4340, annealed

Datasheet view: [Show/Hide](#) [Find Similar](#)

Metals and alloys > Ferrous > [Alloy steels](#) > Low alloy steel > Wrought > AISI 4300 series (Ni-Cr-Mo alloyed) > AISI 4340 (high Cr, 0.40% C) >

General information

Designation ⓘ

Low alloy steel, AISI 4340, annealed

Condition	ⓘ	Annealed
UNS number	ⓘ	G43400
US name	ⓘ	SAE E4340, SAE 4340, SAE E4340, SAE 4340H, SAE 4340, SAE 4330H, ASTM E4340, ASTM 4340, ASTM G43406, ASTM G43400, ASTM E4340H, ASTM E4340, ASTM 1B 1, ASTM B24V, ASTM B23, ASTM 4340H, ASTM 4340, ASTM 4 G, ASME E4340, ~SAE E4340H, ~ASTM B24
GB (Chinese) name	ⓘ	~ 40CrNiMoA
JIS (Japanese) name	ⓘ	SNCM447RCH, SNCM 439, SNB24-5, SNB24-4, SNB24-3, SNB24-2, SNB23-1, ~ SNCM 439, ~SNCM439RCH, ~SNB24-1, ~SNB23-5, ~SNB23-4, ~SNB23-3, ~SNB23-2, G 4103 SNCM 439, SUS 434

Tradenames ⓘ

Lescalloy; Nickelvac; Pompey; Vasco

Typical uses ⓘ

High tensile applications, general engineering parts, through-hardened gears, connecting rods and bolts, gun barrels

Figure 6: Material for Crankshaft Data Page 1

Lastly the gudgeon pin material was low alloy steel, AISI 4140, annealed. The typical uses in Granta implied that this material is used in components that serve a similar purpose to a gudgeon pin.

Low alloy steel, AISI 4140, annealed

Datasheet view: [Show/Hide](#) | [Find Similar](#)

Metals and alloys > Ferrous > [Alloy steels](#) > Low alloy steel > Wrought > AISI 4100 series (Cr-Mo alloyed) > AISI 4140, 41L40 (0.40% C) > AISI 4140 >

General information

Designation ⓘ

Low alloy steel, AISI 4140, annealed

Condition	ⓘ	Annealed
UNS number	ⓘ	G41400
US name	ⓘ	SAE H41400, SAE 4140, SAE 4142, SAE 4140, ASTM J 70, ASTM J 65, ASTM J 55, ASTM 4140, ASTM G41420, ASTM E7140, ASTM B7M, ASTM B16, ASTM 4142H, ASTM 4142, ASTM 4140H, ASTM 4140, ASTM 4137H, ASTM 41, ~SAE 4137H, ~ASTM B4D, ~ASTM 6
EN name	ⓘ	~42CrMoS4, ~42CrMo4
EN number	ⓘ	~1.7227, ~1.7223
ISO name	ⓘ	~ 42 CrMoS 4, ~ 42 CrMo 4, ~42CrMo4
GB (Chinese) name	ⓘ	~42CrMo
JIS (Japanese) name	ⓘ	SNB22-5, SNB22-4, SNB22-3, SNB22-2

Tradenames ⓘ

Aristology; Ascometal; Incut; Krupp; Lasalle; Ovako; Pompey; Rycrome; Saerstahl; Usaloy

Typical uses ⓘ

Heavy duty shafts, connecting rods, high tensile bolts, propeller shaft joints, track pins

Figure 7: Material for Gudgeon Pin Data Page 1

2.1.3 Methodology Decisions

The decision to exclude a thermal study was taken due to its minimal effects on the results. The conclusion was reached as the range of temperatures the part would be subject to (22°C-150°C) would not significantly change the material's (low alloy steel, AISI 4140, normalized) Young's Modulus. Moreover 'Components: Properties, Applications, Materials (Vieweg, 2016) discussed the thermal effects on many components in the assembly but does not consider them for the connecting rod.

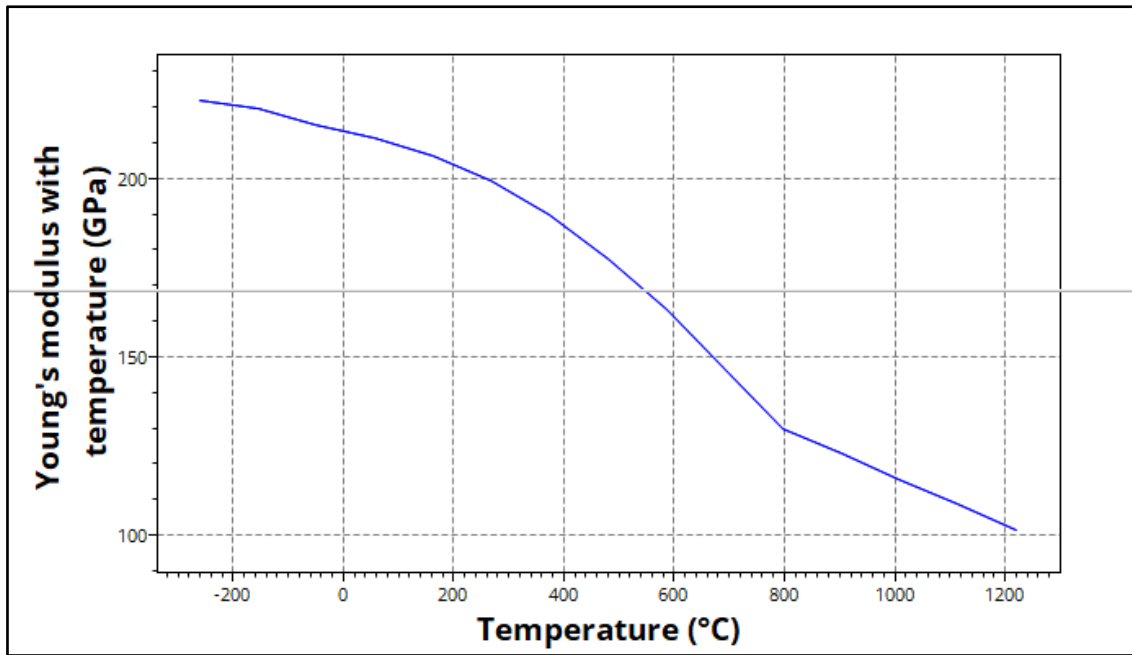


Figure 8: Graph of Low Alloy Steel, AISI 4140, Normalized Temperature against Young's Modulus

The mesh for the connecting rod was set to 1mm element size with soft behaviour. This was done as the mesh was refined to 0.2mm along the two edges highlighted in Figure9, as the sudden change in geometry can cause errors in the study. Refining the mesh reduced the undesirable effects on the solution.

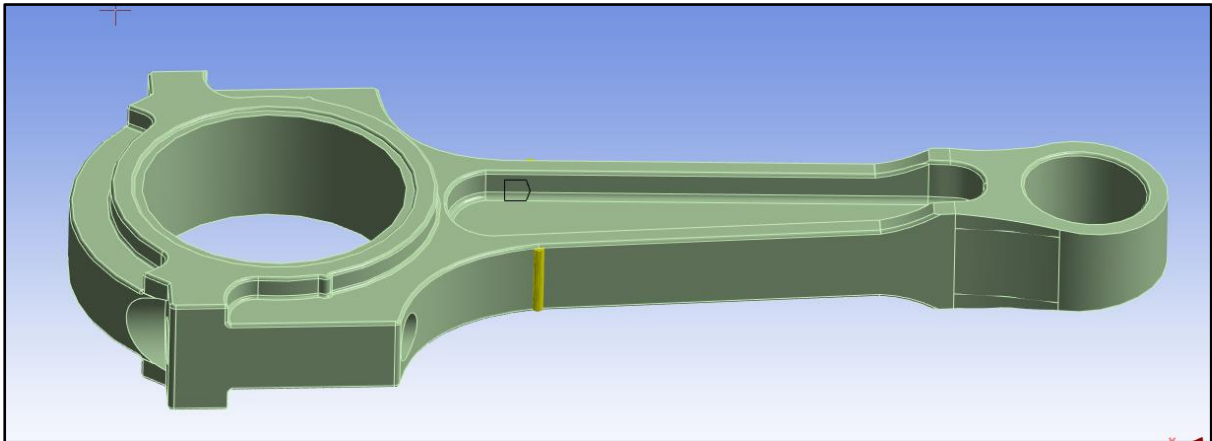


Figure 9: Highlighted Edges for Refined Meshing

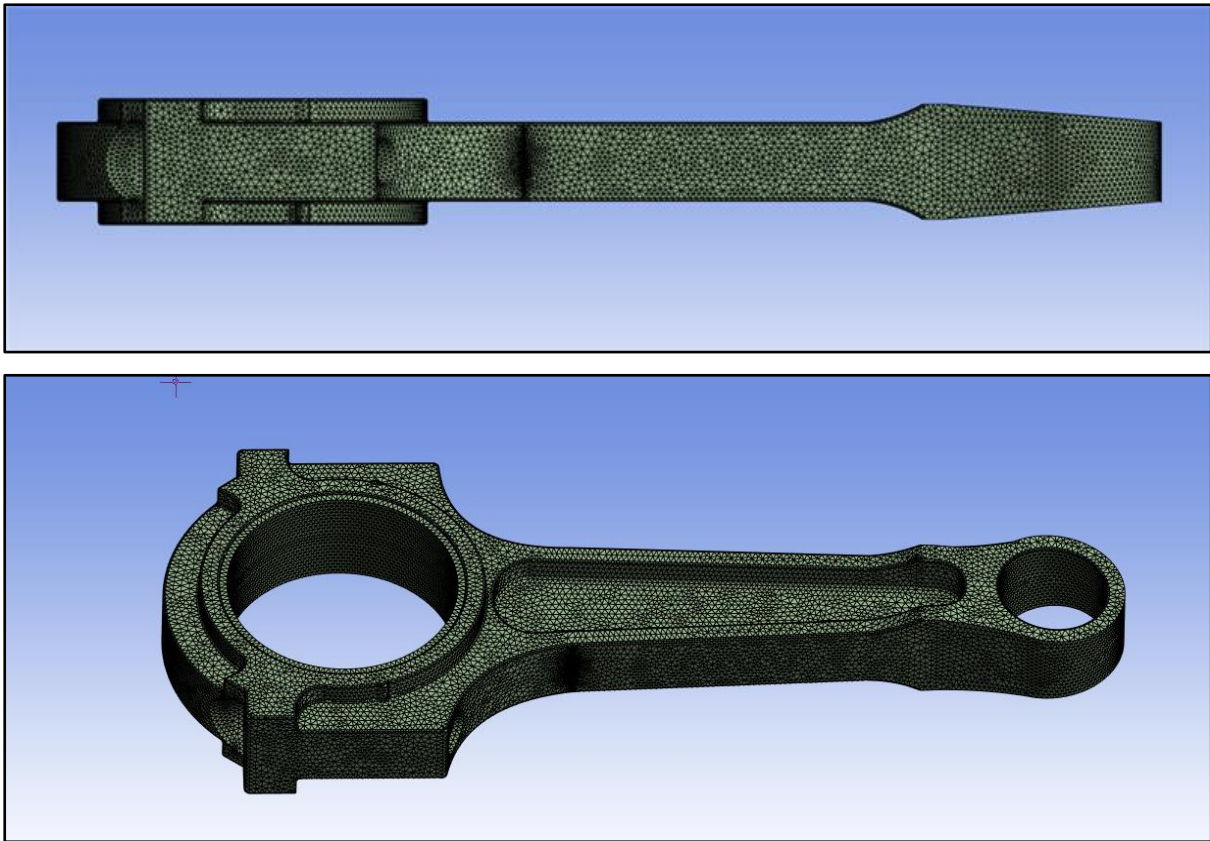


Figure 10: Images of Meshed Connecting Rod

2.2 Workflow

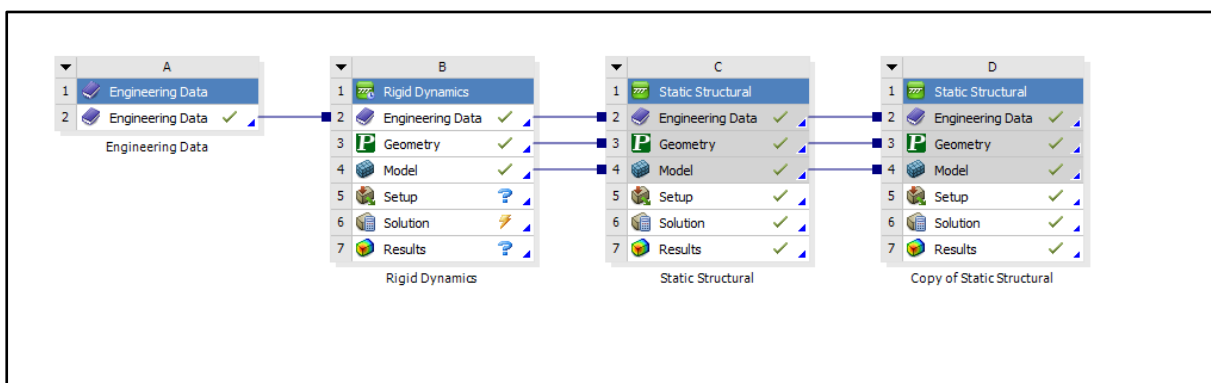


Figure 11: Image of Workflow in Ansys Work Bench

Figure 11 shows the workflow for the analysis process. Engineering Data was opened first to import the materials selected. A Rigid Dynamics study was conducted to generate the motion loads which were then imported into the Static Structural. The first Static Structural contains the data for the connection rod when the compressive

forces are at a maximum for the system and the second Static Structural was the solution for when the system is under its maximum tensile load. The final step was to complete a fatigue study. This could only take place once there were two load cases.

2.3 Simplifications

Failures of connecting rods oftentimes happen at the bolt holes that join the top and bottom of the component, (Alam, Thakur, Kumar PS, & Ghadei, 2018), (Xiaolei, et al., 2017). The geometry has been simplified in the CAD provided to make the connecting rod one part. Thus, this was not an accurate simulation of the component.

3.0 Boundary Conditions & Load Cases

3.1 Rigid Dynamics

The graphs below contain the data that was exported for the Static Structural. This made up the motion loads at peak compressive and peak tensile loads.

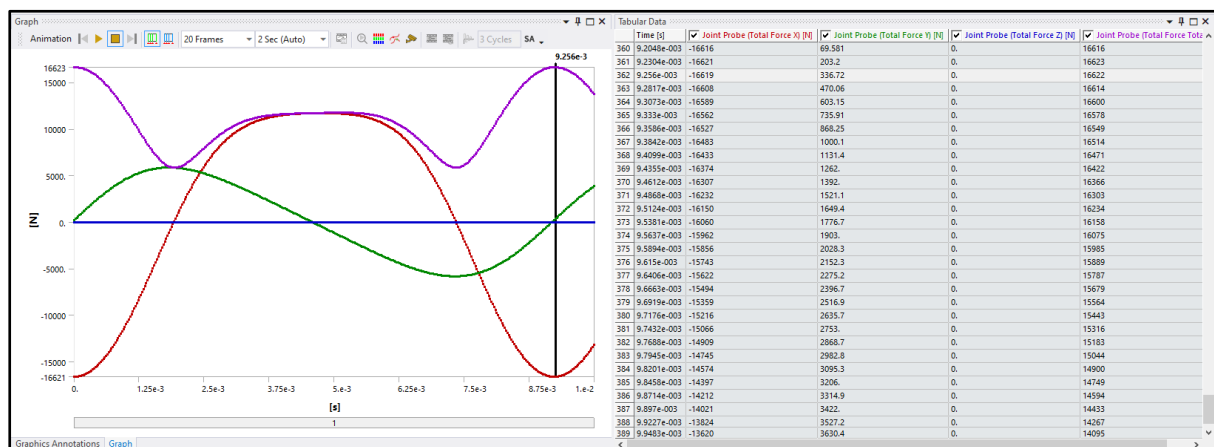


Figure 12: Graph of Force against Time at Rod to Crankshaft Joint top dead centre max in

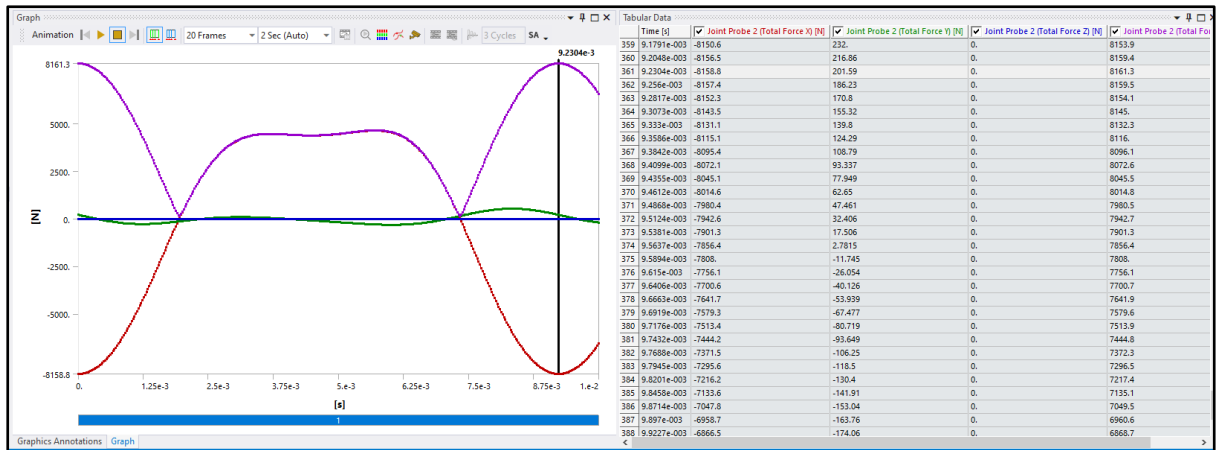


Figure 13: Graph of Force against Time at Rod to Pin Joint top dead centre max in

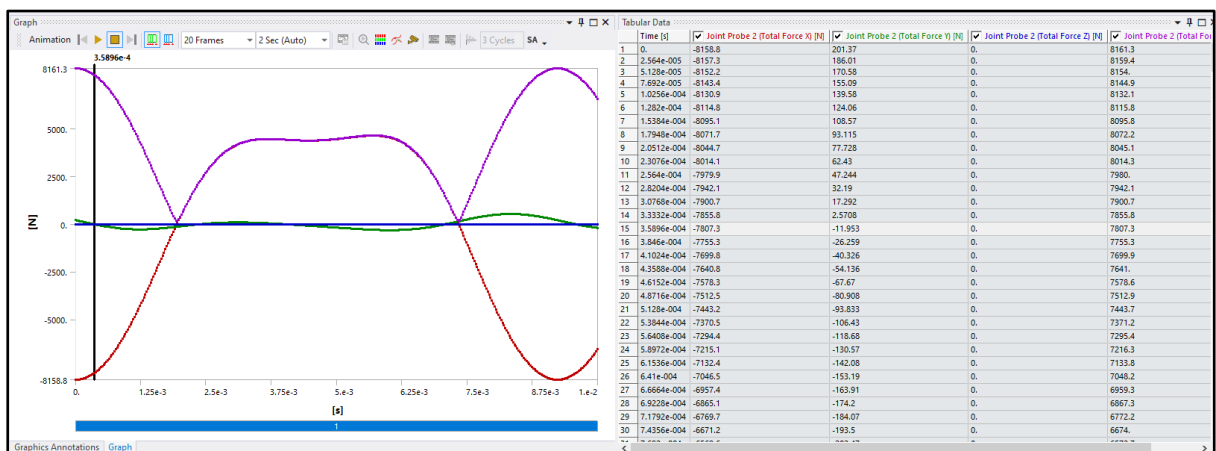


Figure 14: Graph of Force against Time at Rod to Pin Joint max compression 14deg

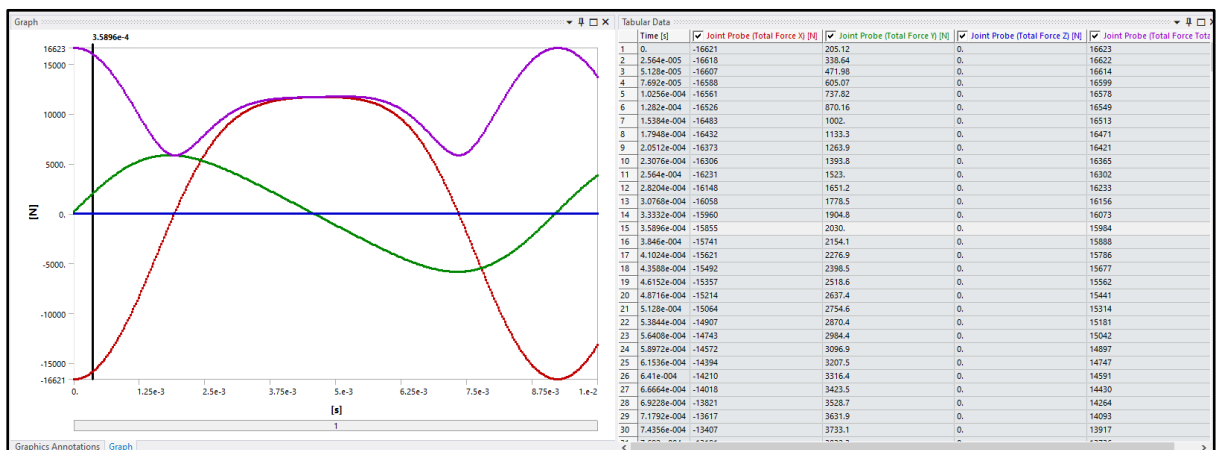


Figure 15: Graph of Force against Time at Rod to Crankshaft Joint max compression 14deg

3.2 Static Structural

For the Static Structural the remote forces and moments were carried over from the Rigid Dynamic. The connecting rod was changed from rigid to flexible and a bearing load of 35601N was added to the small end of the rod. The main flaw with these studies was how they were constrained. Soft springs were not suitable to fix the rod in place. Therefore, a cylindrical support (labelled 'A' in Figure16 and17) was added to the internal face of the large end of the rod. This aimed to replicate the suppressed crankshaft however, the support was fully fixed (axial, radial and tangential). If this study was to be repeated, then the support should be free radially. This would allow for the internal face to deform due to the loads and result in a more accurate study.

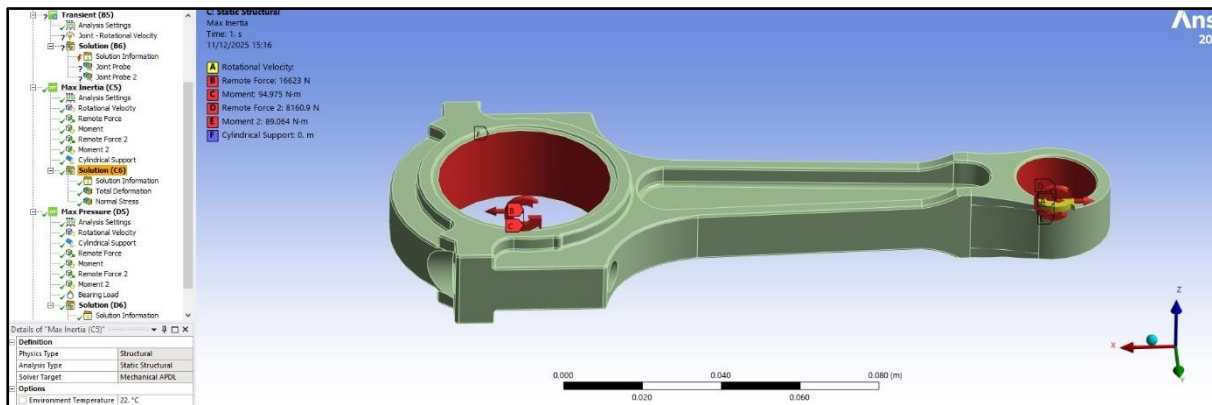


Figure 16: Load Cases for Maximum Inertia

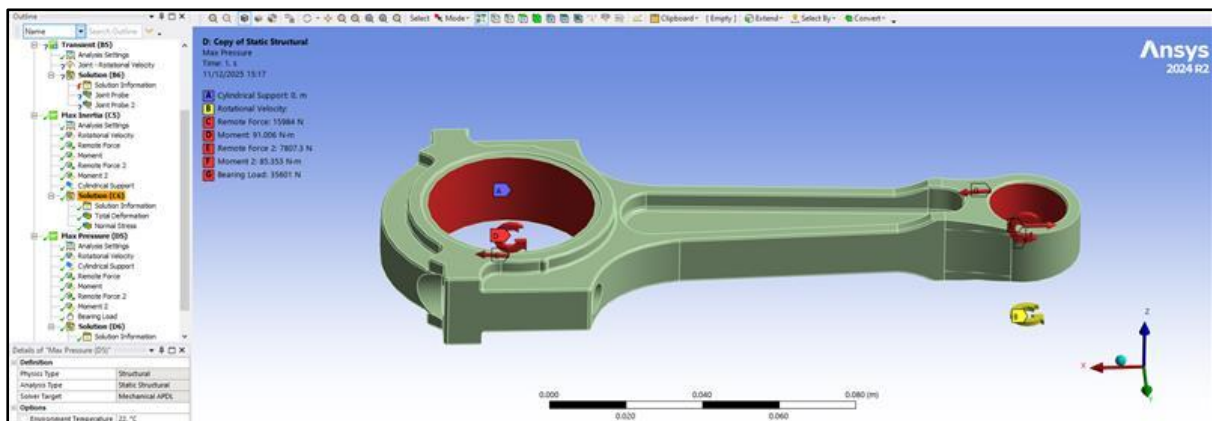
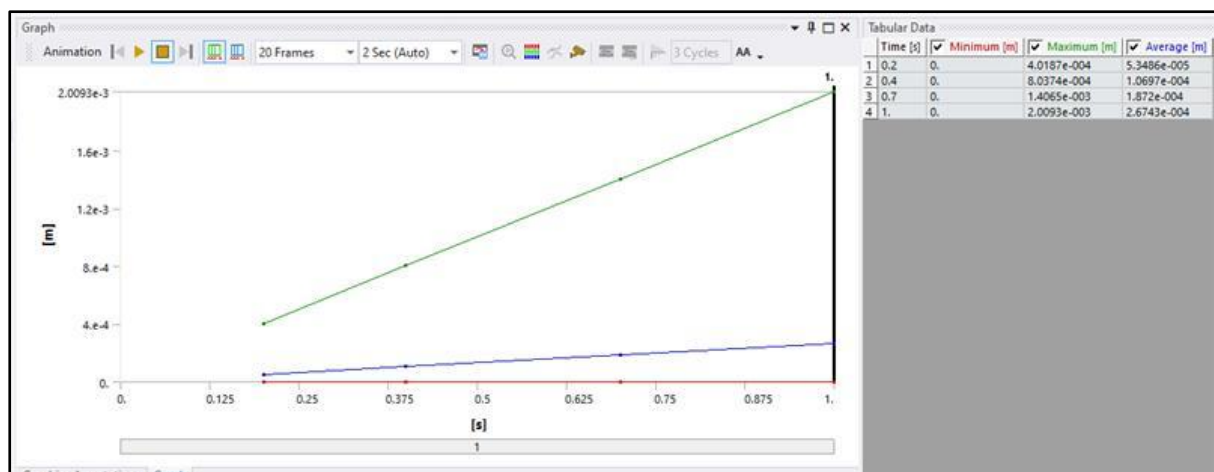
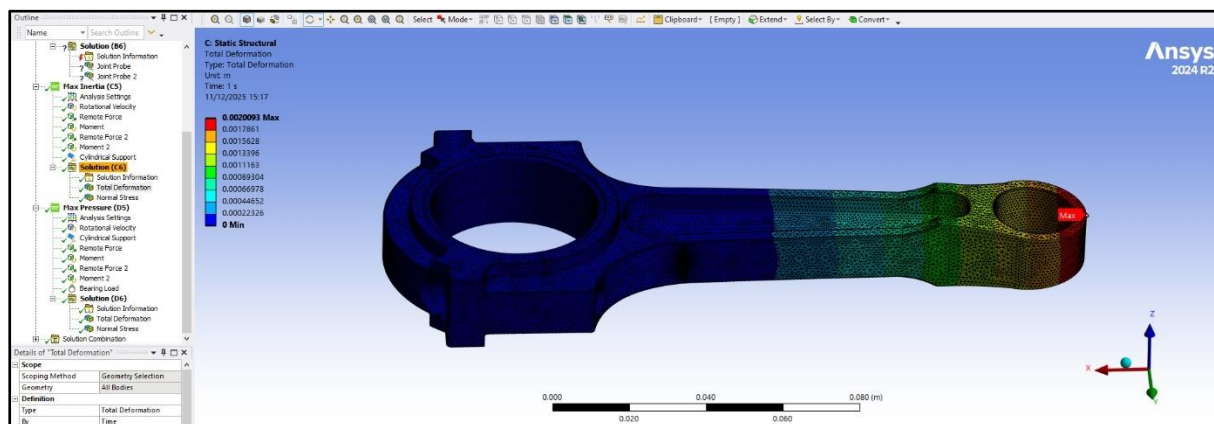


Figure 17: Load Cases for Maximum Compression

4.0 Results

4.1 Total Deformation at Maximum Intertia

Below are the plots for total deformation when the connecting rod is at maximum inertia. The maximum deformation of the part was $2.0093 \times 10^{-3} \text{m}$.



4.2 Normal Stress at Maximum Intertia

The normal stress plot maximum was at $6.3229 \times 10^8 \text{Pa}$. It was vital that the stress solved was for normal stress and not equivalent, as the two static structural were dealing with forces that are acting in alternate directions. Using equivalent data would result in total force not being an accurate representation of the system.

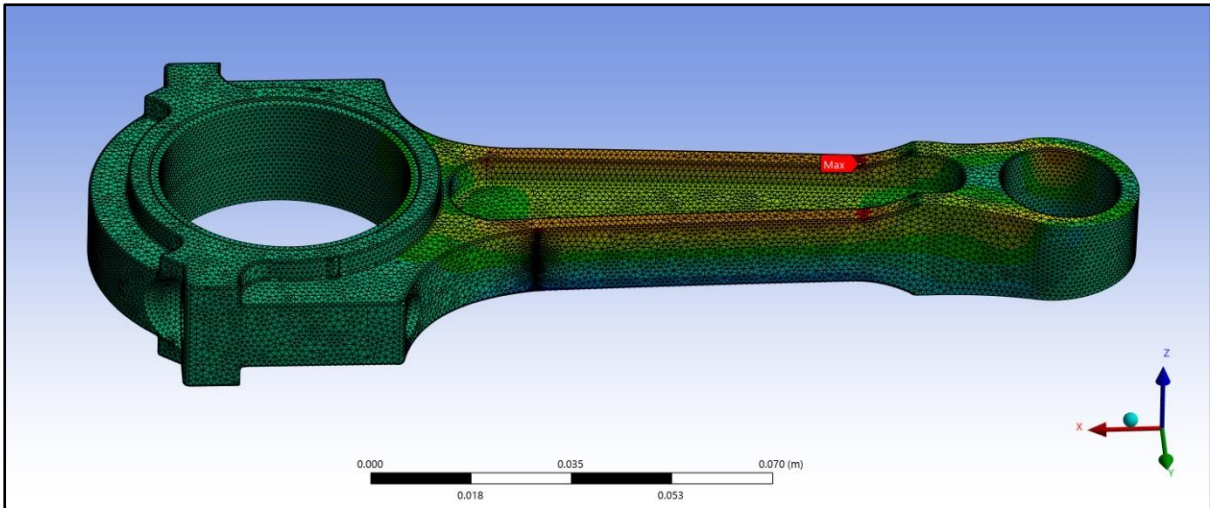


Figure 20: Normal Stress at Maximum Inertia of the Connecting Rod

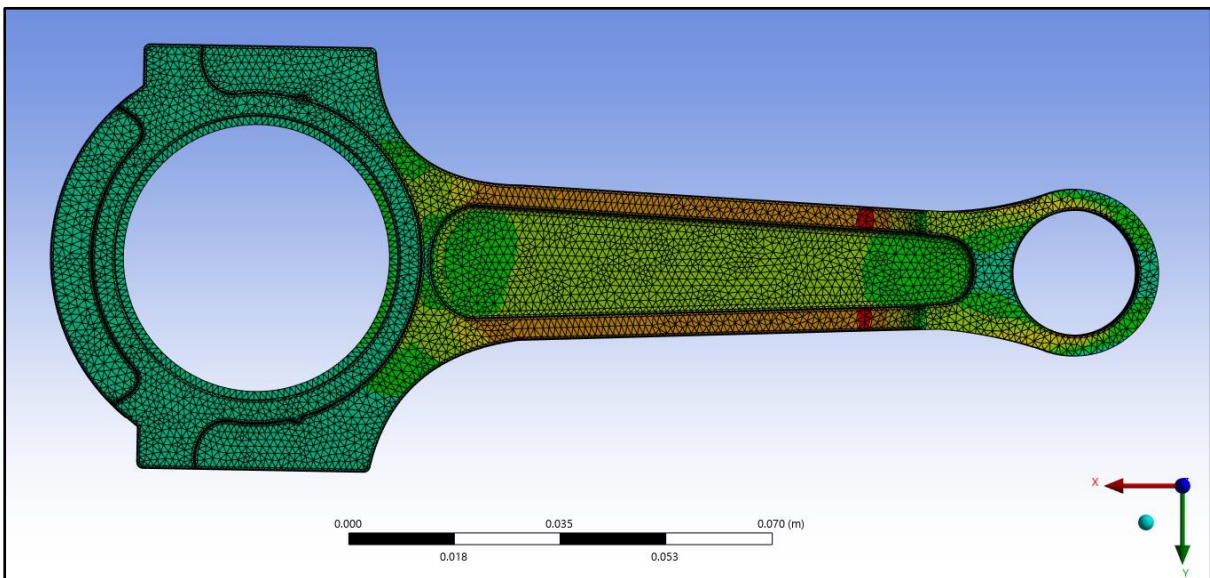


Figure 21: Normal Stress at Maximum Inertia of the Connecting Rod Top Face

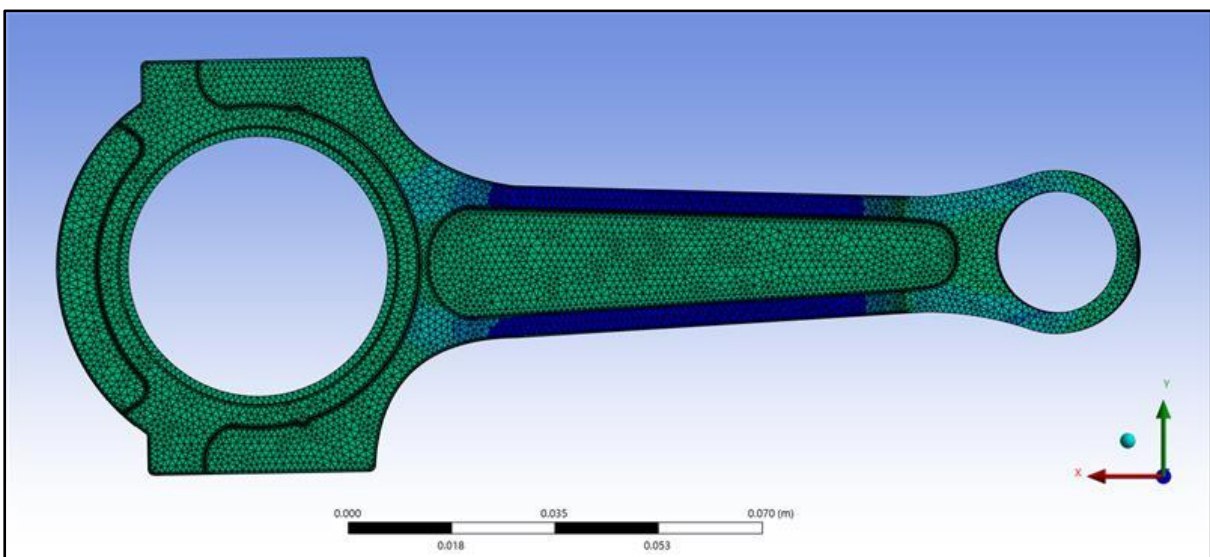


Figure 22: Normal Stress at Maximum Inertia of the Connecting Rod Bottom Face

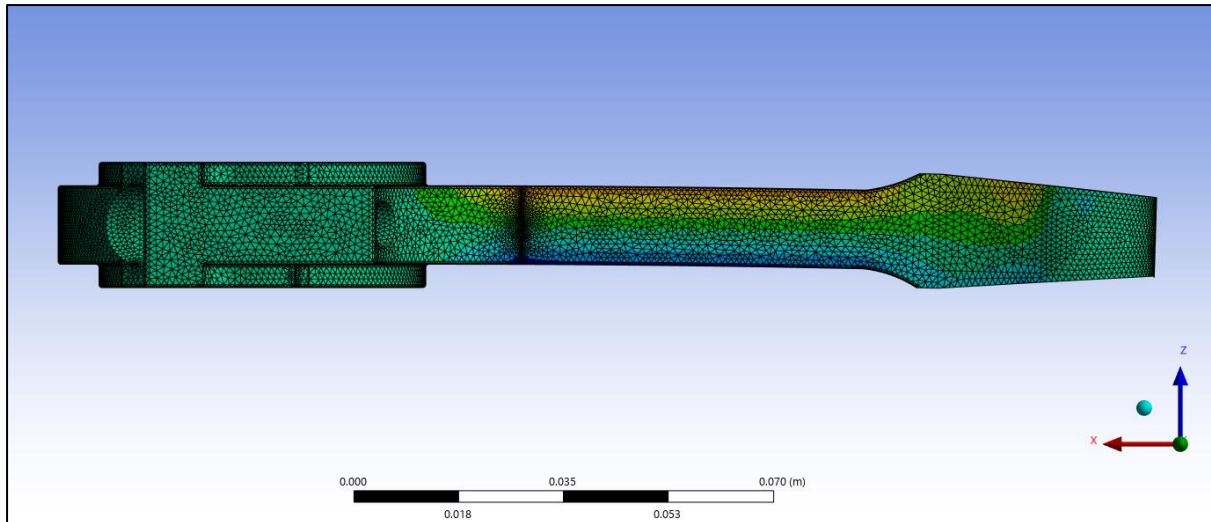


Figure 23: Normal Stress at Maximum Intertia of the Connecting Rod Side View

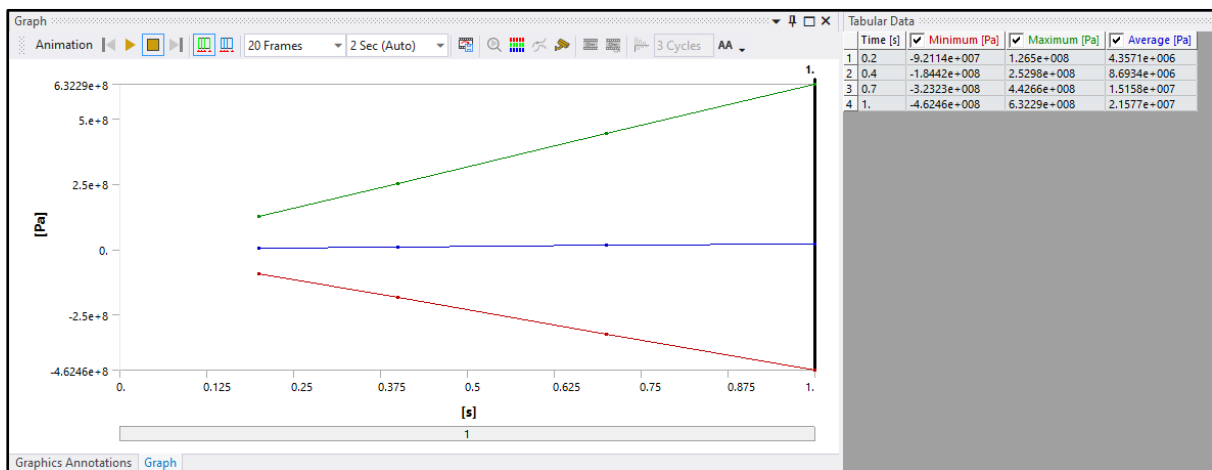


Figure 24: Graph of Normal Stress at Maximum Intertia of the Connecting Rod

4.3 Total Deformation at Maximum Compression

Similar to the previous static structural the maximum deformation occurred at the small end of the rod. The maximum deformation due to compression was $8.7379 \times 10^{-4} \text{m}$.

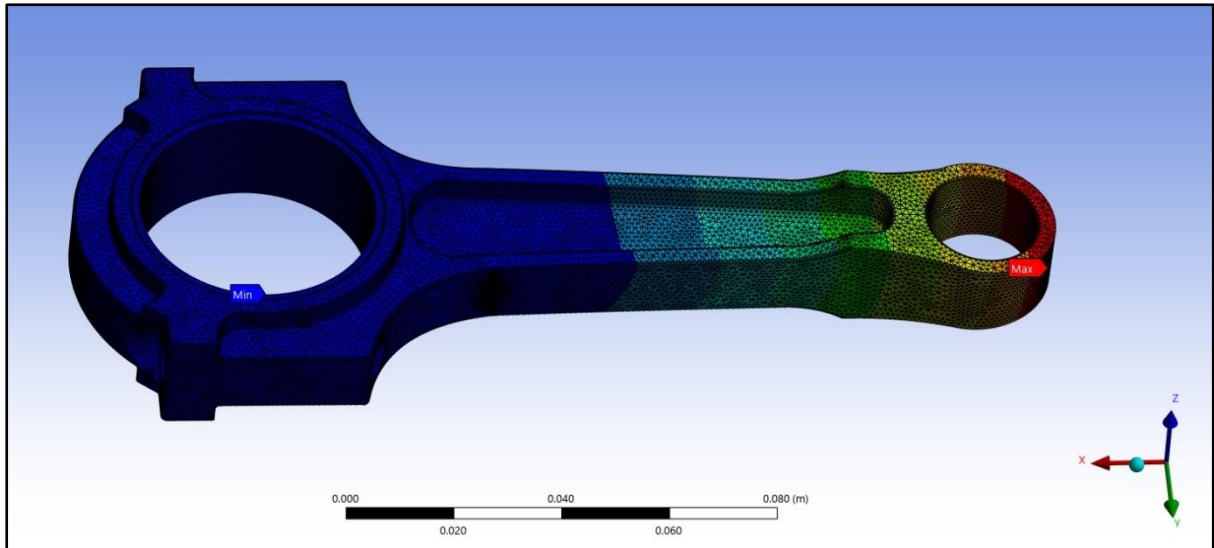


Figure 25: Total Deformation at Maximum Compression of the Connecting Rod

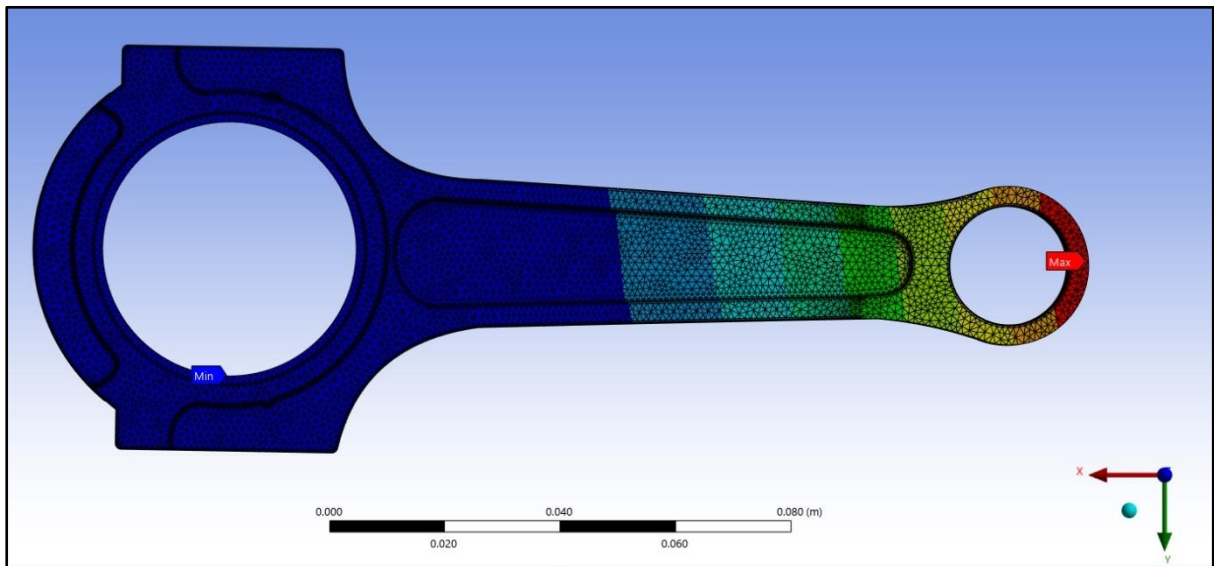


Figure 26: Total Deformation at Maximum Compression of the Connecting Rod Top View

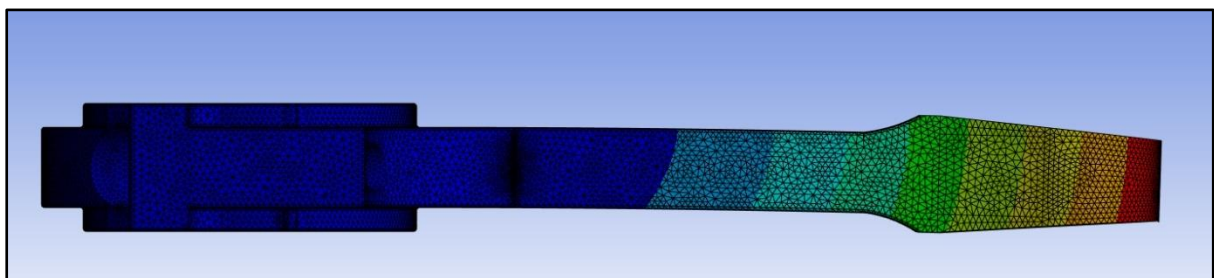


Figure 27: Total Deformation at Maximum Compression of the Connecting Rod Side View

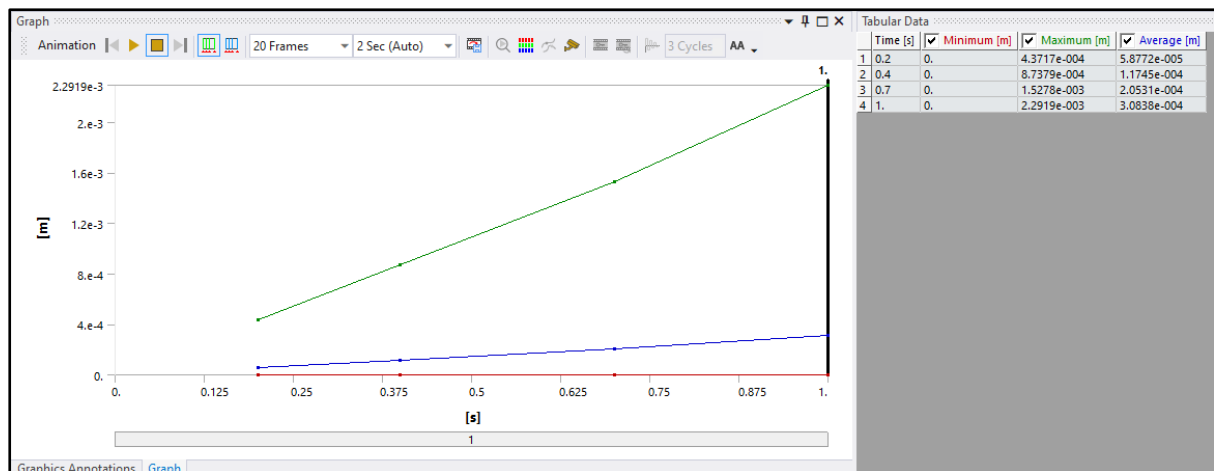


Figure 28: Graph of Total Deformation Stress at Maximum Intertia of the Connecting Rod

4.4 Normal Stress at Maximum Compression

The normal stress plot maximum was at $5.7701 \times 10^8 \text{ Pa}$. In these plots it was very clear that one side of the connecting rod was in tension and the other side in compression. This was lightly due to an imbalance in the piston head which caused the forces to be distributed unevenly over the connecting rod. I was also worth noting that the maximum and the minimum points both lay on edges where there was a change in geometry.

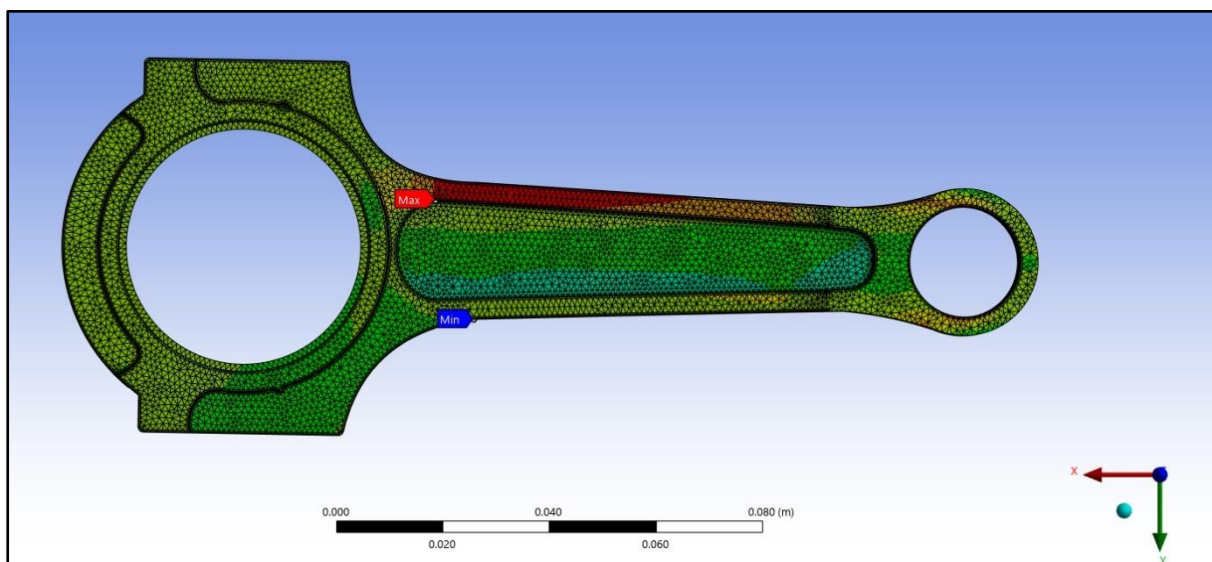


Figure 29: Normal Stress at Maximum Compression of the Connecting Rod

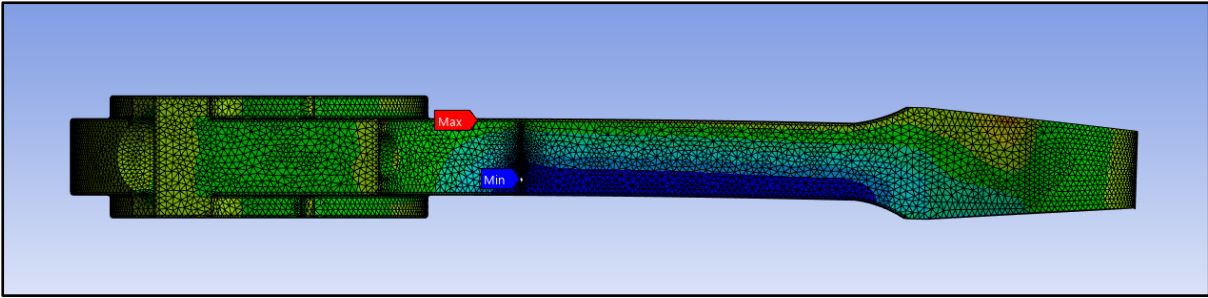


Figure 30: Normal Stress at Maximum Compression of the Connecting Rod Side View

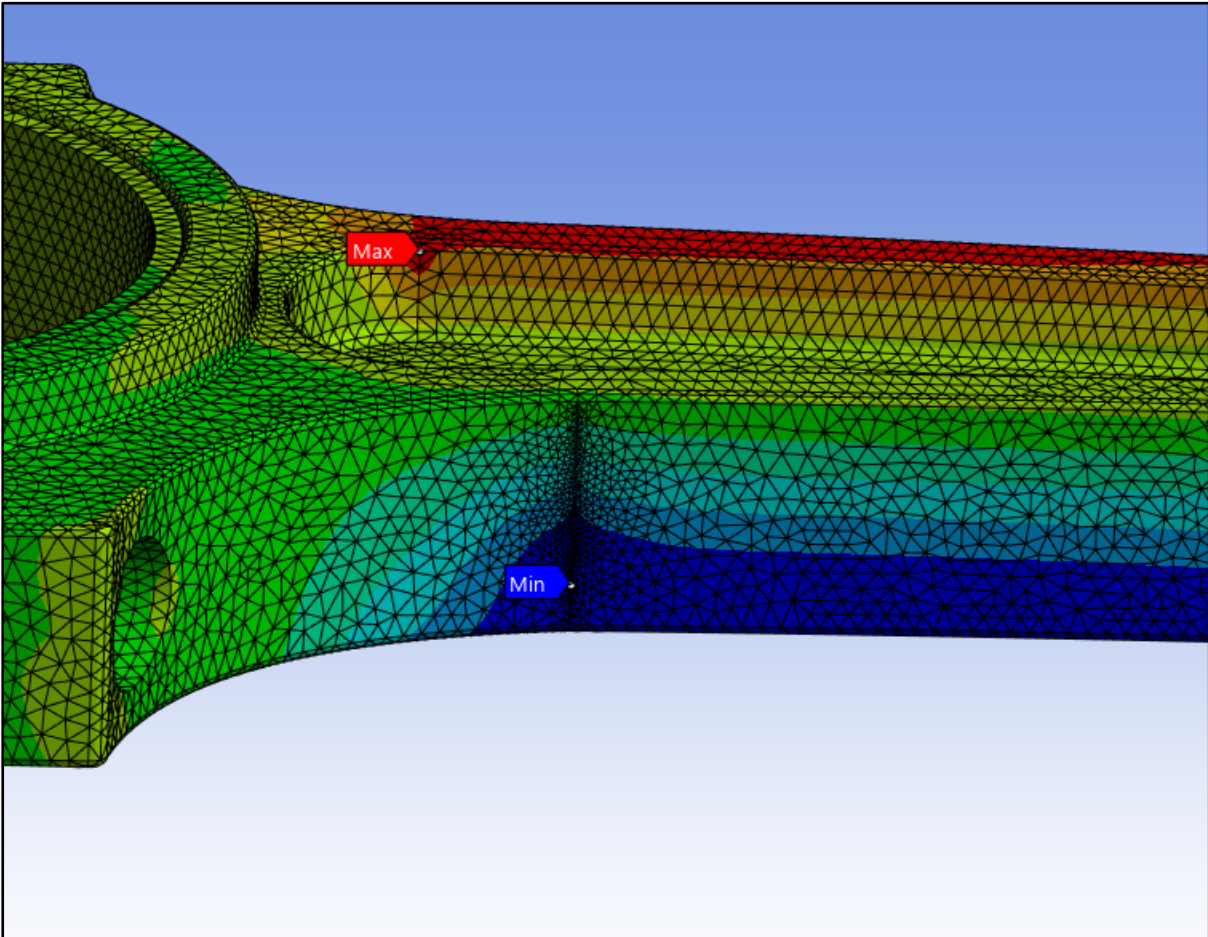


Figure 31: Normal Stress at Maximum Compression of the Connecting Rod Close up view

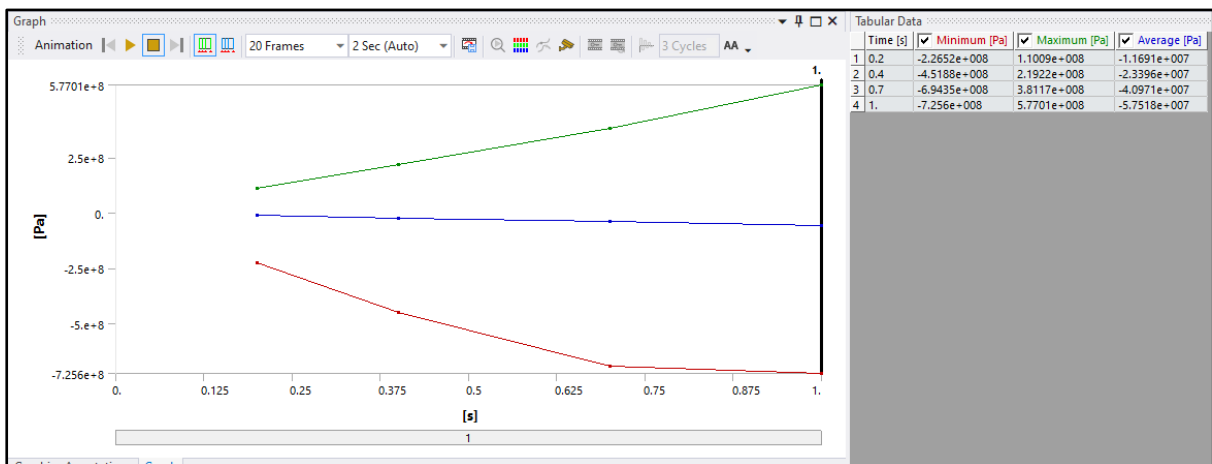


Figure 32: Graph of Normal Stress at Maximum Compression of the Connecting Rod

4.5 Fatigue Life

In Britain it is generally accepted that if a part can survive one million cycles (10^6) then it can be considered to have infinite life (BS 7608 Guide). In the case of this connecting rod its minimum cycles was 1.0333×10^7 . Therefore, it can be considered to have infinite life.

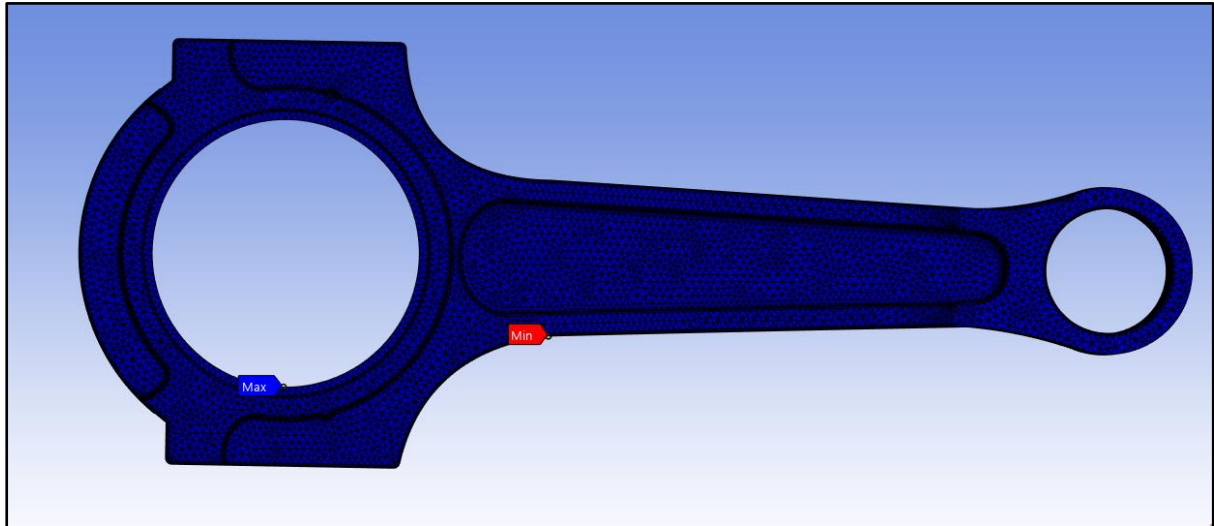


Figure 33: Fatigue life of the Connecting Rod

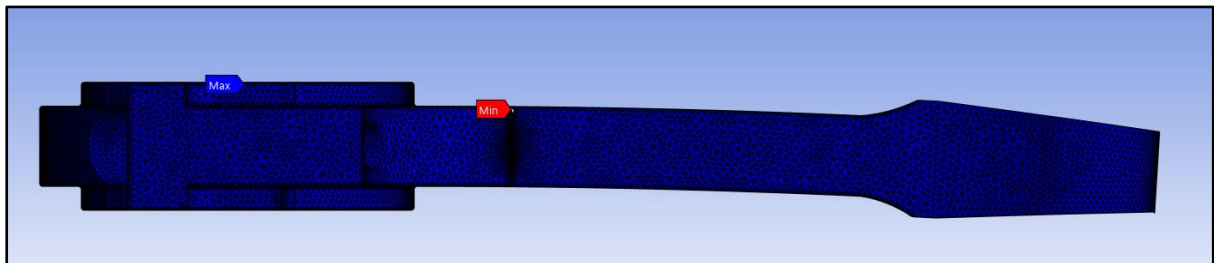


Figure 34: Fatigue life of the Connecting Rod Side View

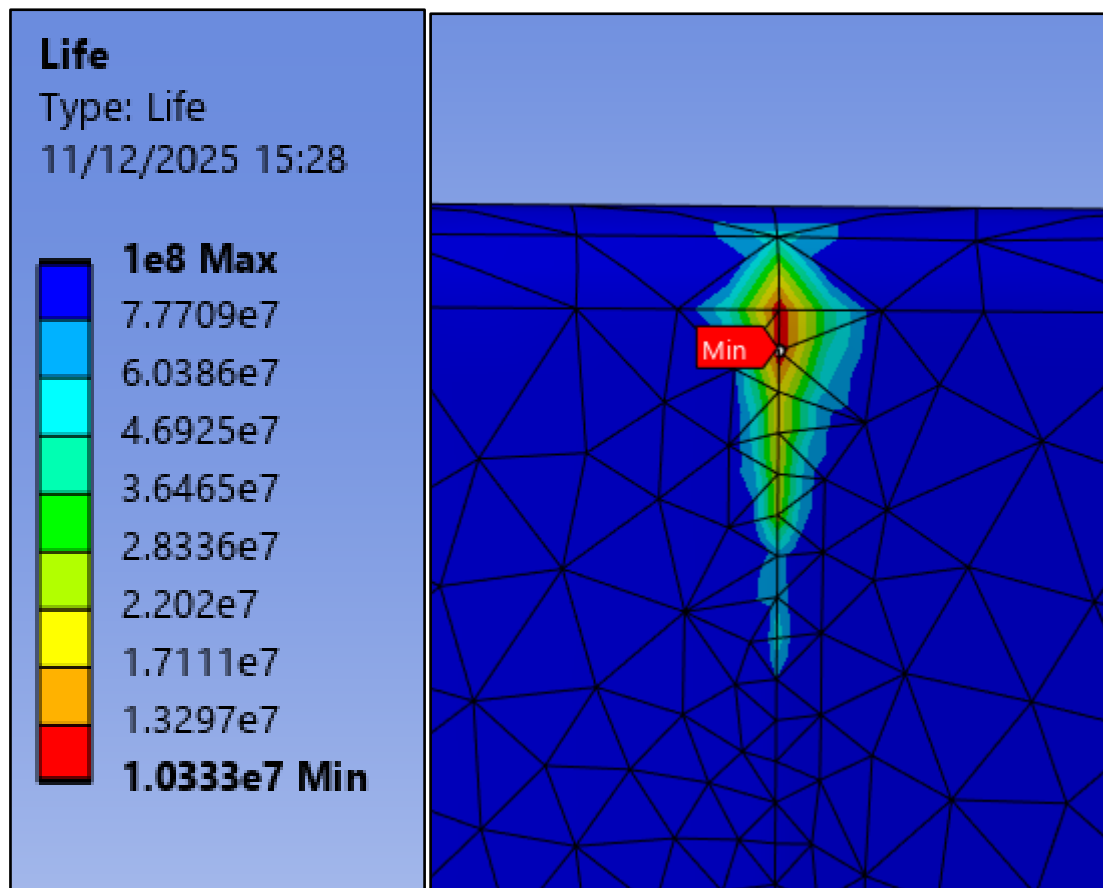


Figure 35: Fatigue life of the Connecting Rod Close up of Minium Value Point

4.6 Fatigue FOS

The connecting rod's factor of safety has a point that was less than one. This was in an area where the geometry changes and an area with a dense mesh.

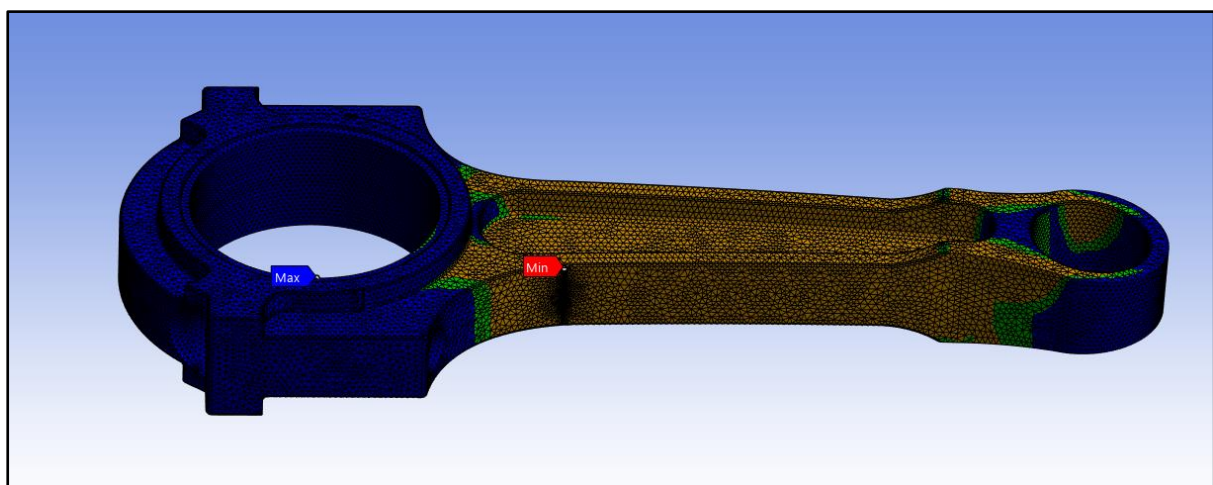


Figure 36: Fatigue FOS of the Connecting Rod

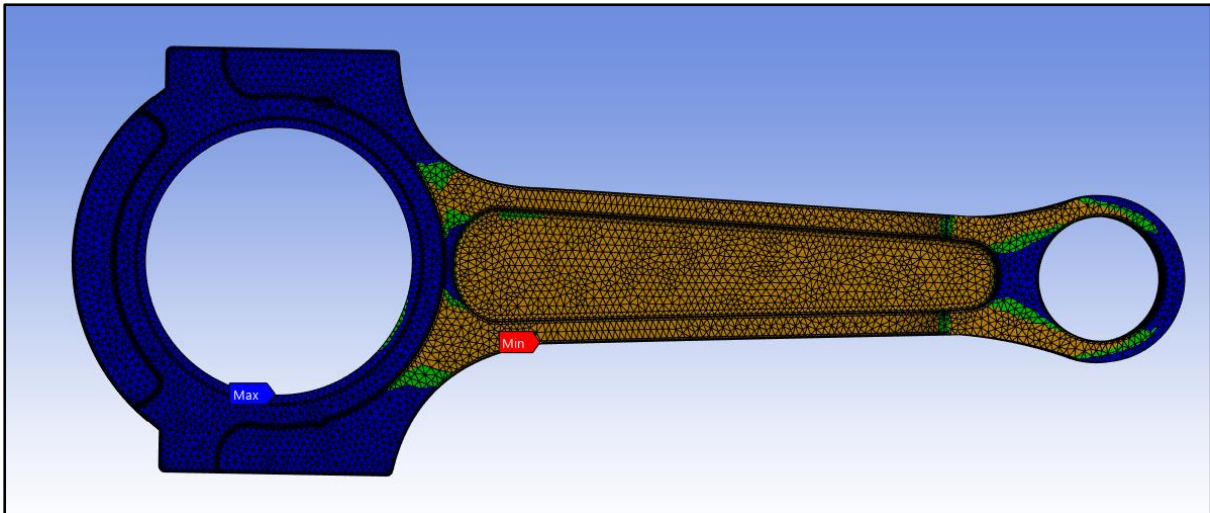


Figure 37: Fatigue FOS of the Connecting Rod Top View

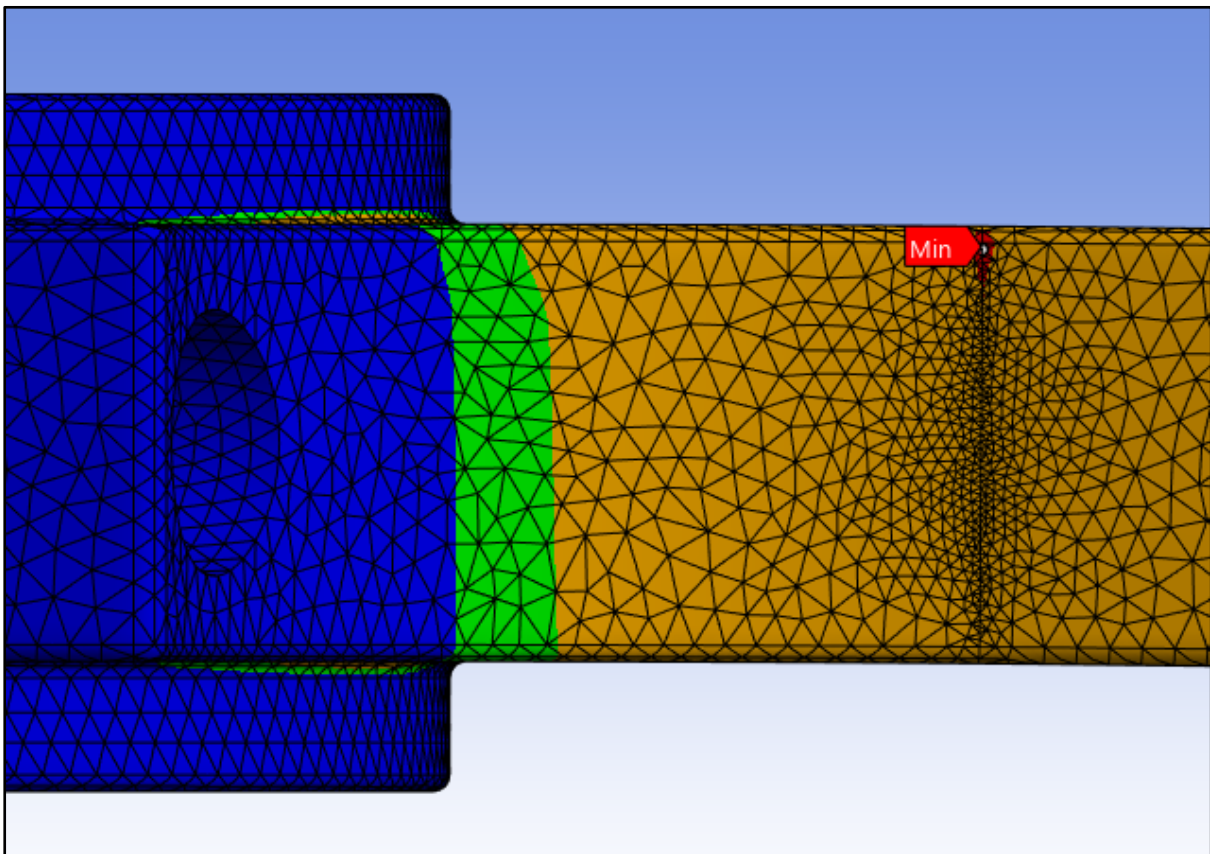


Figure 38: Fatigue FOS of the Connecting Rod Side View of Minium Point

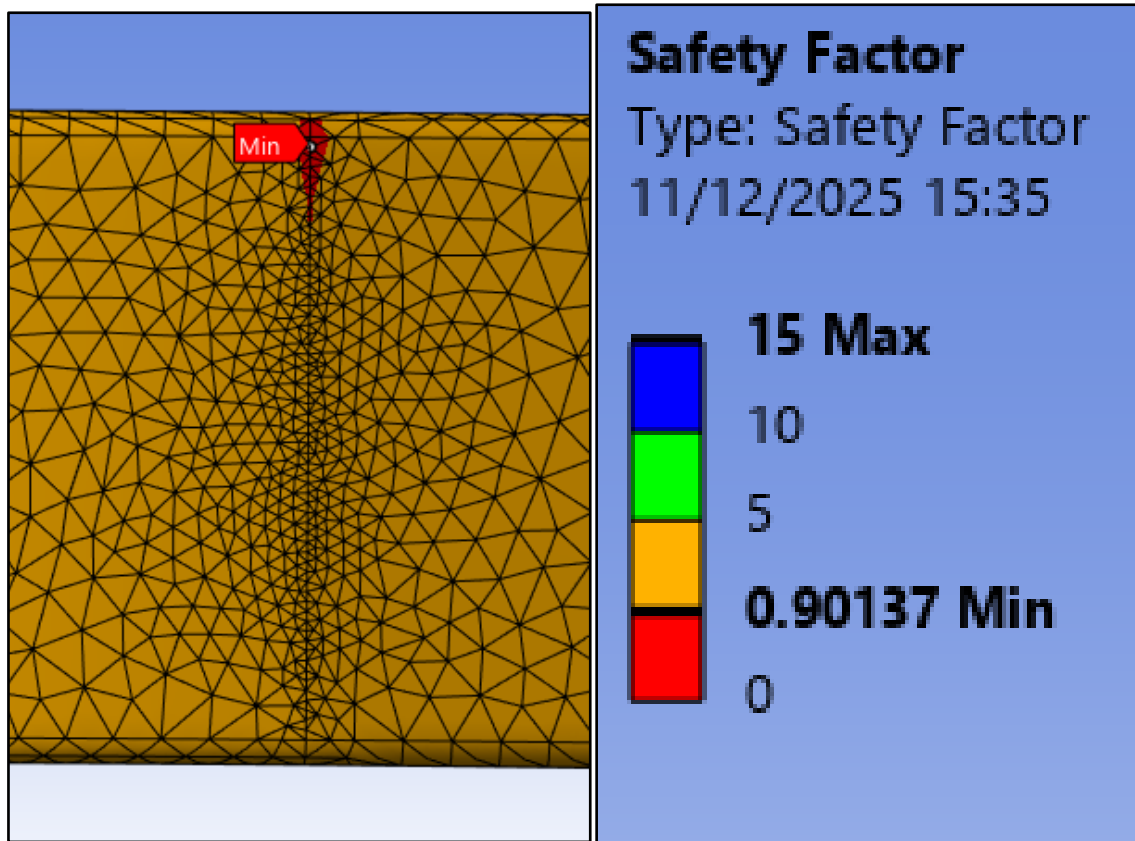


Figure 39: Fatigue FOS of the Connecting Rod Close up View of Minium Point

4.7 Calculations

When hand calculated the fatigue factor of safety is 4.1027. see Figure 40.

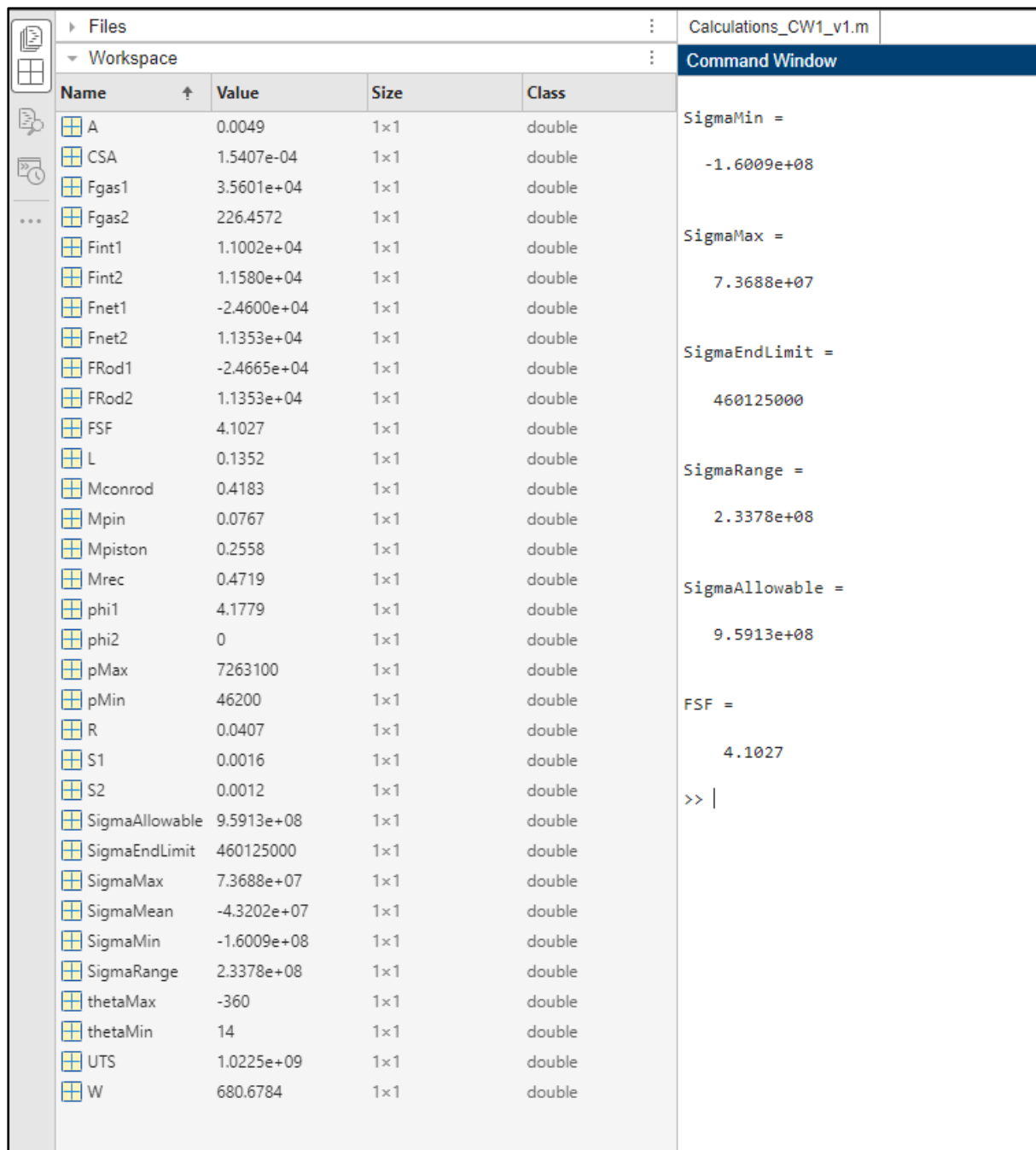


Figure 40: MATLAB Calculations for Fatigue FOS

See appendix for full MATLAB code.

5.0 Discussions & Critical Evaluation

Overall, this simulation was not an accurate representation of the loads that a connecting rod would see. By modelling the rod as one part and not as two (the rod and cap) with bolts the simulation does not account for a common failure type within the connecting rod design. Additionally, the rod was not accurately constrained for the static structural simulations therefore the internal face of the big end of the rod was not able to deform. Lastly, the FOS plot had a minimum value that was four times smaller than that of the hand calculated FOS. Further investigation would be required to fully understand the nature of this error.

6.0 Bibliography

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7.0 Appendix

MATLAB code for calculations

```
clc,clear, close all
```

```
%% Calculations
```

```
%Fixed Inputs
```

```
%R=Crankshaft Radius (half stroke length) (m)
```

```
R=(81.4*10-3)/2;
```

```
%L=Connecting Rod Length (m)
```

```
L=135.15*10-3;
```

```
% Mpiston=Mass of piston (kg) (mass=volume*avg density)
```

```
Mpiston=94730.42*10-9*((2.73*103+2.67*103)/2);
```

```
% Mpin=Mass of gudgeon pin (kg) (mass=volume*avg density)
```

```
Mpin=9774.9*10-9*((7.9*103+7.8*103)/2);
```

```
% Mconrod=Mass of conrod (kg) (mass=volume*avg density)
```

```
Mconrod=53288.27*10-9*((7.9*103+7.8*103)/2);
```

%Mrec=Reciprocating Mass (kg)

Mrec=Mpiston+Mpin+(Mconrod/3);

%Cylinder Head Area (πr^2) (m^2)

A= $\pi*((79/2)*10^{-3})^2$;

%theta=crankshaft Angle at Maxium Gas pressure (deg)

thetaMin=14;

%theta=crankshaft Angle at Maxium Intertia(deg)

thetaMax=-360;

%W=angular velocity of the engine (deg/s) W=RPM*2*pi/60

W=6500*2*pi/60;

%pMax=Max Gas Pressure (N/m^2)

pMax=72.631*10⁵;

%pMin=Min Gas Pressure (N/m^2) for max interia (max tensile)

pMin=0.462*10⁵;

%Cross Section Area of the I-beam

CSA=154.07*10⁻⁶;

%Outputs

%phi=Connecting Rod Angle (deg)

phi1=asind(R*sind(thetaMin)/L); %for max gas (max comp)

phi2=asind(R*sind(thetaMax)/L); %for max interia (max tensile)

%S=piston postion (relative to TDC) (m)

S1=(R+L)-(R*cosd(thetaMin)+L*cosd(phi1));%for max gas (max comp)

S2=(R+L)-(R*cosd(thetaMin)+L*cosd(phi2));%for max interia (max tensile)

%Fgas=Force due to gas pressure (N)

Fgas1=A*pMax;

Fgas2=A*pMin;

%Fint=Force due to Inertia (N)

Fint1=Mrec*(R*W^2*cosd(thetaMin)+((R^2)/L)*W^2*cosd(2*thetaMin)); %for max gas (max comp)

Fint2=Mrec*(R*W^2*cosd(thetaMax)+((R^2)/L)*W^2*cosd(2*thetaMax)); %for max interia (max tensile)

%Fnet=Total Forces (N)

Fnet1=Fint1-Fgas1; %for max gas (max comp)

Fnet2=Fint2-Fgas2; %for max interia (max tensile)

%Force Con Rod

FRod1=Fnet1/cosd(phi1); %for max gas (max comp)

FRod2=Fnet2/cosd(phi2); %for max interia (max tensile)

%% Fatigue Safety Factors

%Inputs

%SigmaMin= (N/m^2)

SigmaMin=FRod1/CSA %for max gas (max comp)

%SigmaMax= (N/m^2)

SigmaMax=FRod2/CSA %for max interia (max tensile)

%UTS= Ulitmate Tensile Strength (N/m^2)

UTS=1.0225*10^9;

%SigmaEndLimit=Limit of stress for the material (N/m^2)

$$\text{SigmaEndLimit}=0.45*\text{UTS}$$

%Outputs

%Mean Stress (N/m²)

$$\text{SigmaMean}=(\text{SigmaMax}+\text{SigmaMin})/2;$$

%Range of Stress Vaules (N/m²)

$$\text{SigmaRange}=\text{SigmaMax}-\text{SigmaMin}$$

%Allowable Stress (N/m²)

$$\text{SigmaAllowable}=2*\text{SigmaEndLimit}*(1-(\text{SigmaMean}/\text{UTS}))$$

%FSF= Fatigue Safety Factor

$$\text{FSF}=\text{SigmaAllowable}/\text{SigmaRange}$$