

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)

Taylor Anais Rohn

2500151

Date 12/12/2025

Contents

1. Methodology & Workflow	2
1.1 Overview	2
1.2 Geometry.....	3
2. Boundary Conditions & Load Cases.....	5
2.1 Constraints	5
2.2 Total Loads.....	11
2.3 Results	12
3. Fatigue Life & Safety Factor Comparisons	14
4. Comparing the ANSYS analysis data with the analytical values shows similar results.....	16
5. Discussions & Critical Evaluation	17
6. References	18

1. Methodology & Workflow

1.1 Overview

This paper focuses on the structural fatigue analysis of a connecting rod from an internal combustion engine simulated on ANSYS Mechanical. The fatigue life of the component under realistic operating conditions is evaluated and then discussed.

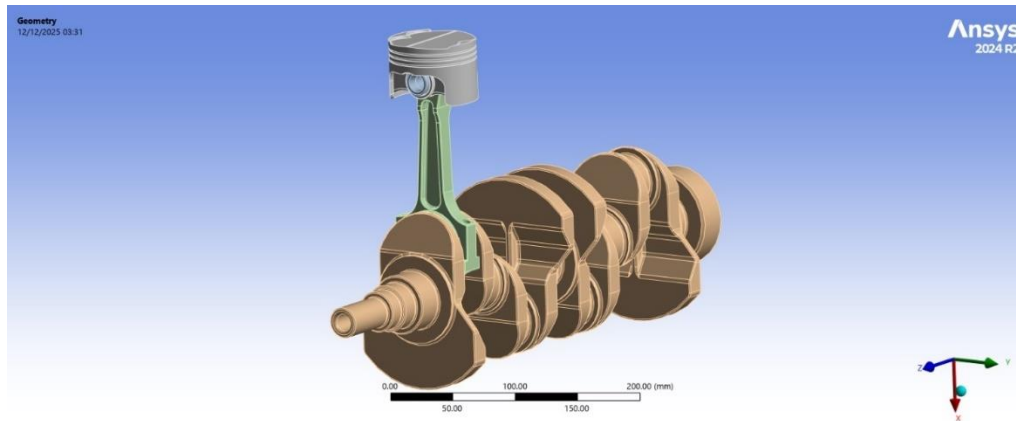
Applying finite element analysis techniques to complex mechanical components such as crankshaft, connecting rod, Gudgeon pin and piston allows to calculate the fatigue life and using computational tools and analytical methods it is possible to critically assess the boundary conditions and loading scenarios in structural simulations.

In a properly designed and assembled engine the connecting rod bolts are intended to be the “weak link” – they should stretch and yield before the main body of the connecting rod itself [1].

Due to simplicity purposes, in this dissertation there are no connecting rods, which does not necessarily mean that the simulation delivers wrong results as the main focus is the connecting rod itself. It is therefore expected that the component will have a long durability if operating in good conditions.

Finally, the collected data will be evaluated against analytical calculations in order to test the efficacy of the described Ansys Mechanical analysis.

1.2 Geometry

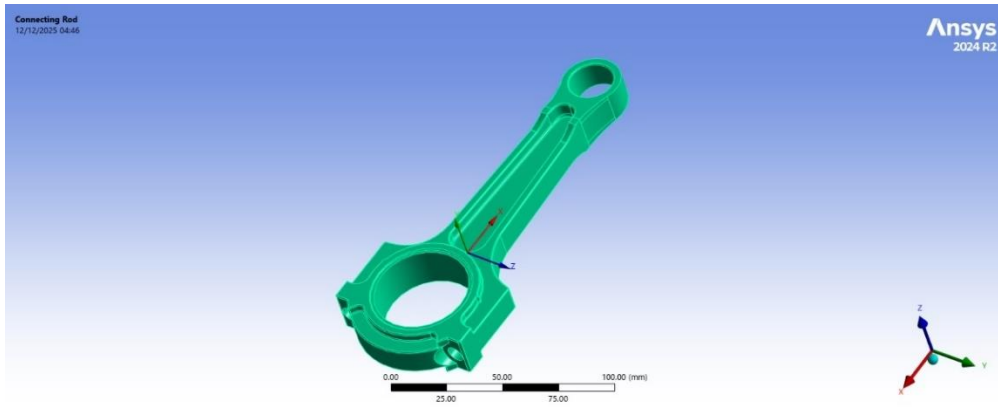


[Figure 1] Crankshaft, connecting rod, Gudgeon pin and piston of an ICE.

The components in study are parts of a Ford Sigma engine with the following specifications:

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	Aluminium
Cylinder block	Aluminium
Camshaft drive	Belt

The materials selected for the Engineering Data were AISI 4340 low alloy steel, normalized, for respectively crankshaft, connecting rod and Gudgeon pin, and Aluminium, 6061, T6 for the piston [Figure 1] . The first offers high strength, toughness, and excellent fatigue/impact resistance [2] whereas the second offers a great balance of lightweight strength, good fatigue resistance for repetitive engine cycles, excellent machinability and decent corrosion resistance [3].

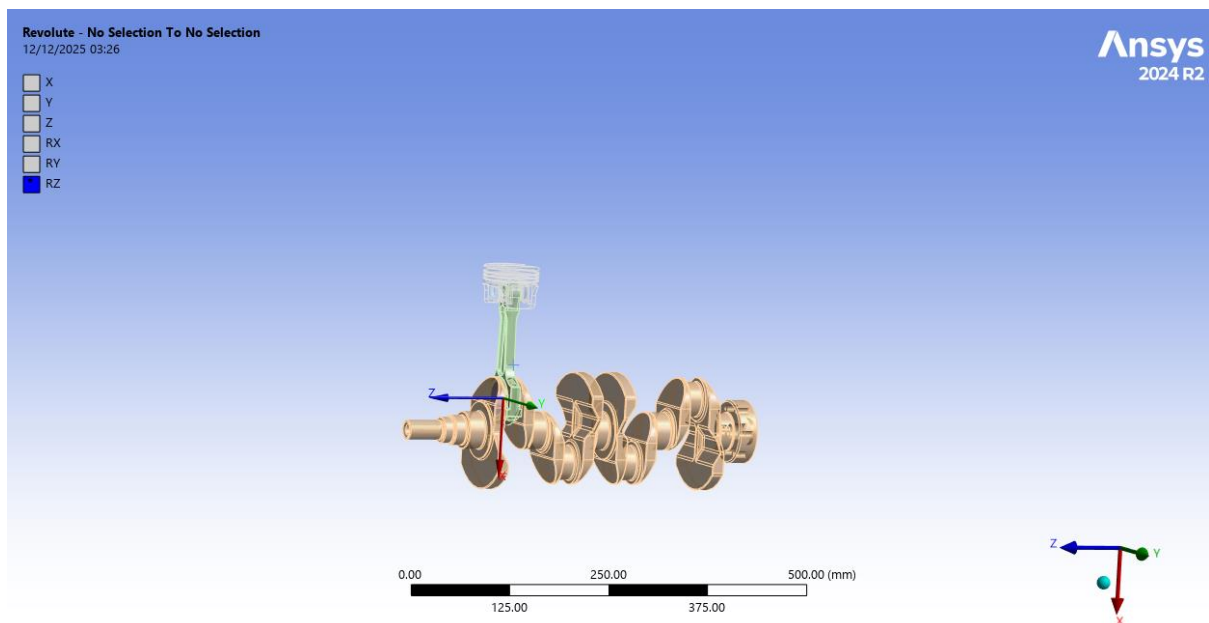


[Figure 2] Connecting rod

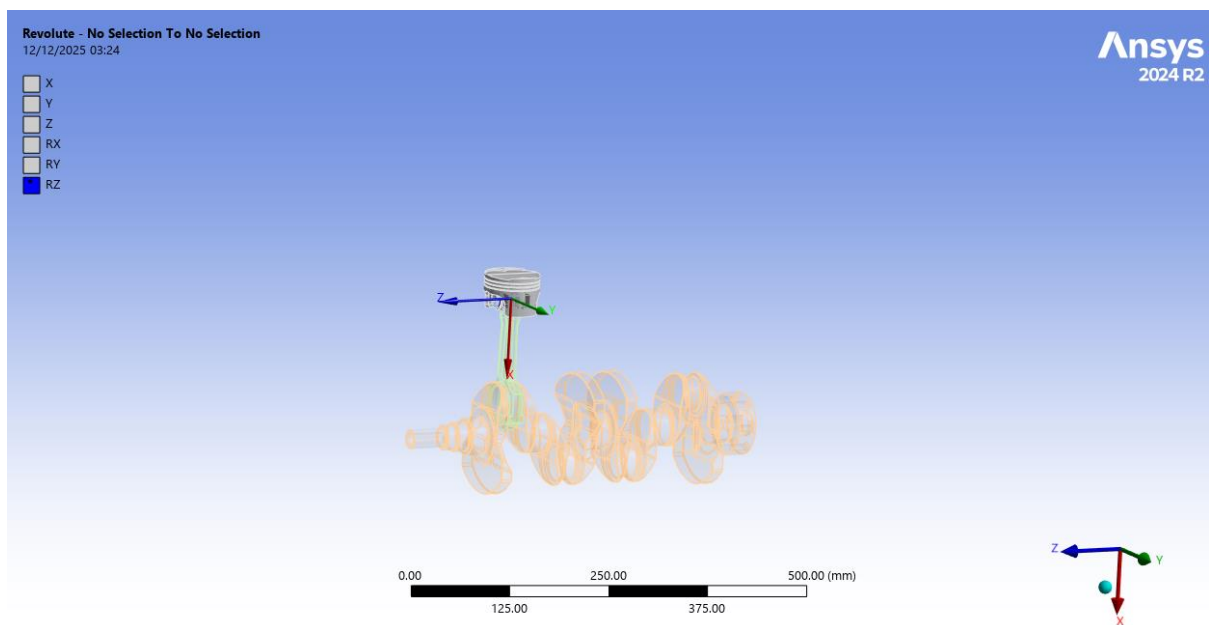
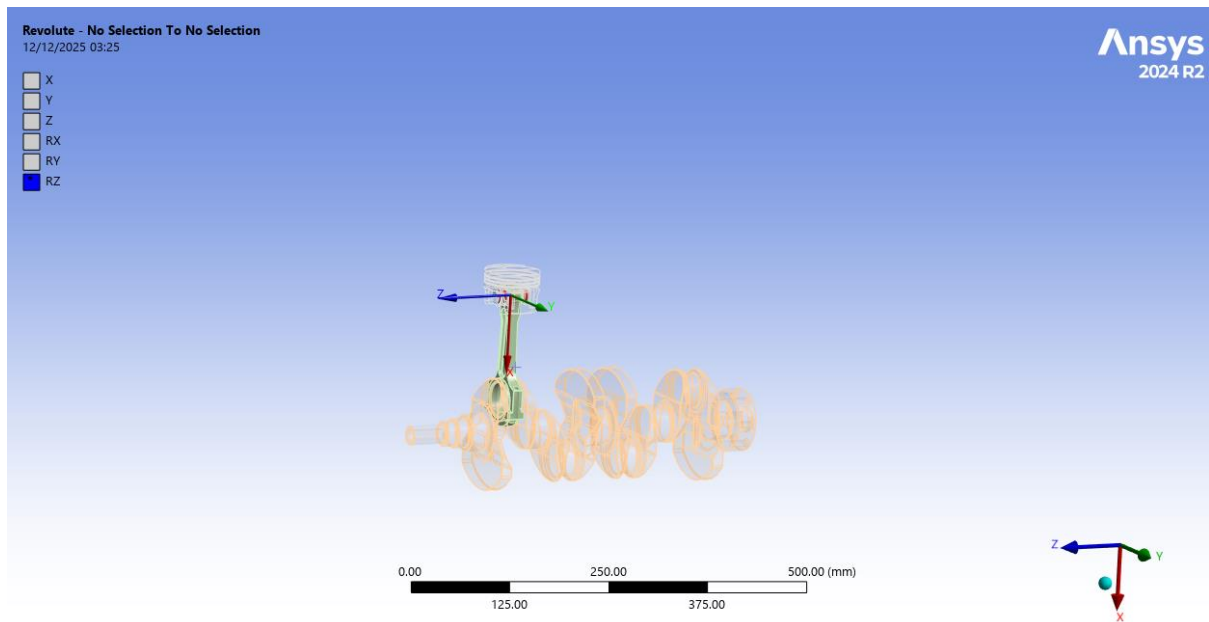
2. Boundary Conditions & Load Cases

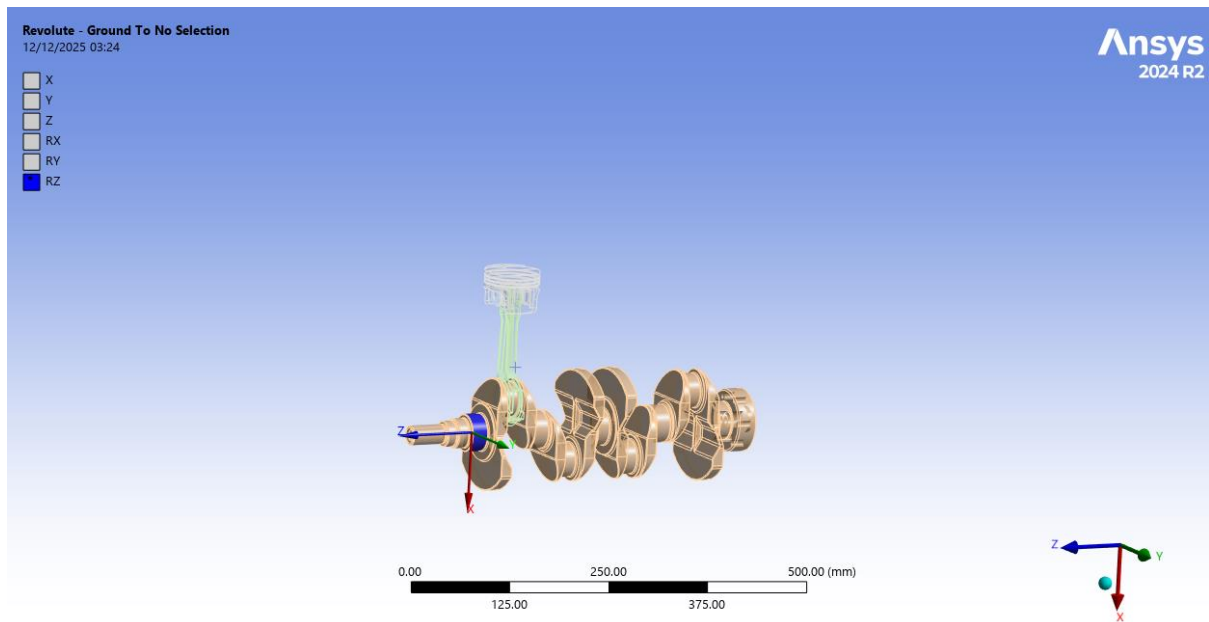
2.1 Constraints

The first step consisted in assigning connections to parts of the ICE, in particular revolute joints constraint respectively between crankshaft and connecting rod [Figure 3], Gudgeon pin and connecting rod [Figure 4], piston and Gudgeon pin [Figure 5], whereas to constraint the crankshaft to a single DOF on z-axis a revolute joint body to ground was implemented [Figure 6] . The piston instead required a single translational DOF following the z-axis obtained applying a general body to ground constraint [Figure 7].

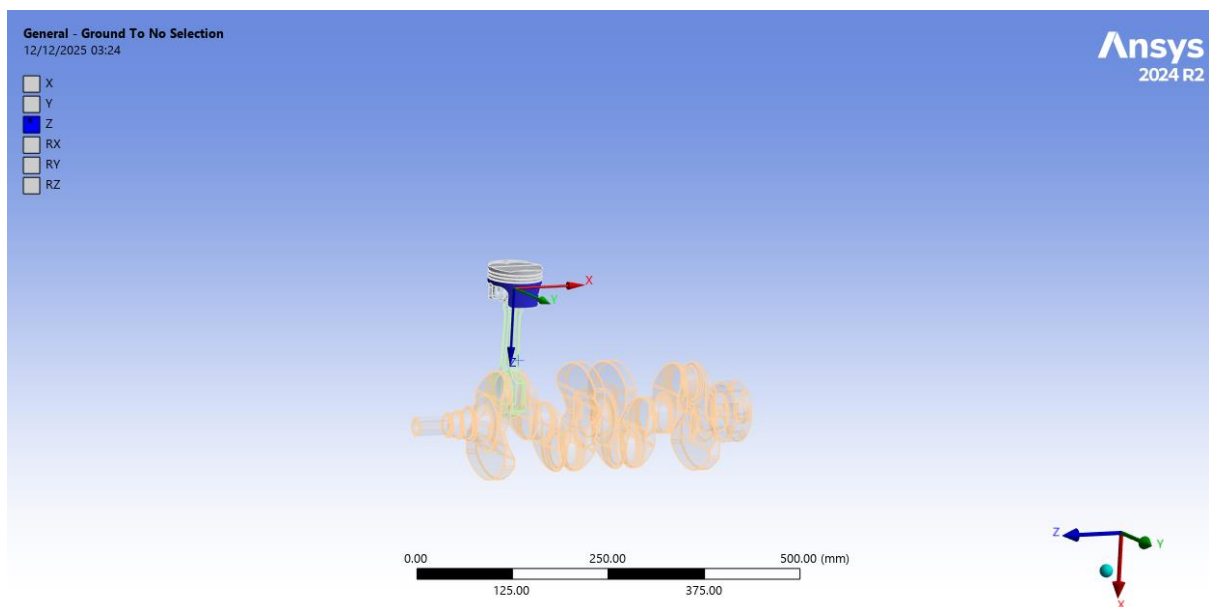


[Figure 3] Crankshaft and connecting rod joint



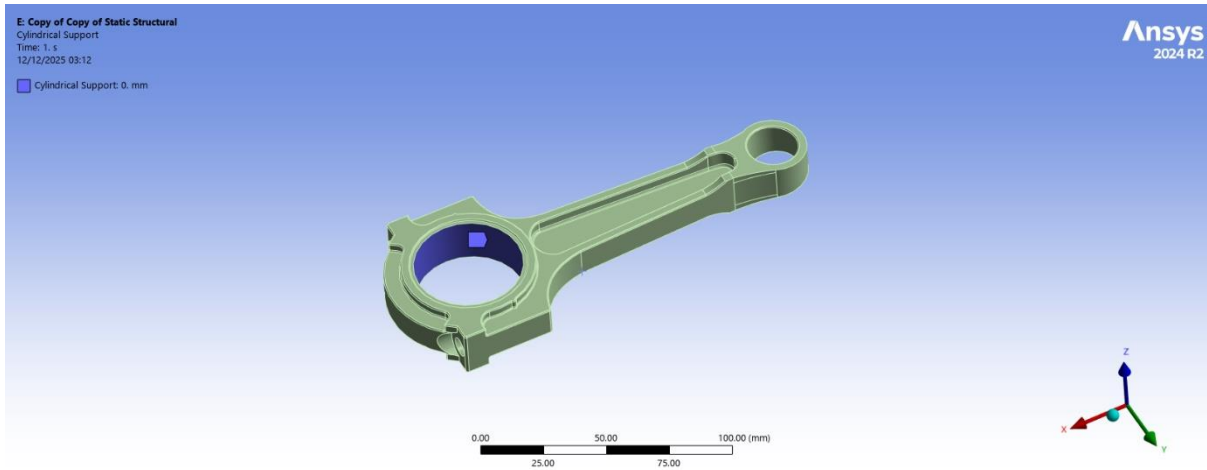


[Figure 6] Rotation of the crankshaft



[Figure 7] Piston in the cylinder

Other boundary condition necessary to carry the rigid dynamic analysis were inserting a cylindrical support on the big end of the connecting rod and a bearing load used to apply the force loadings [Figure 8].

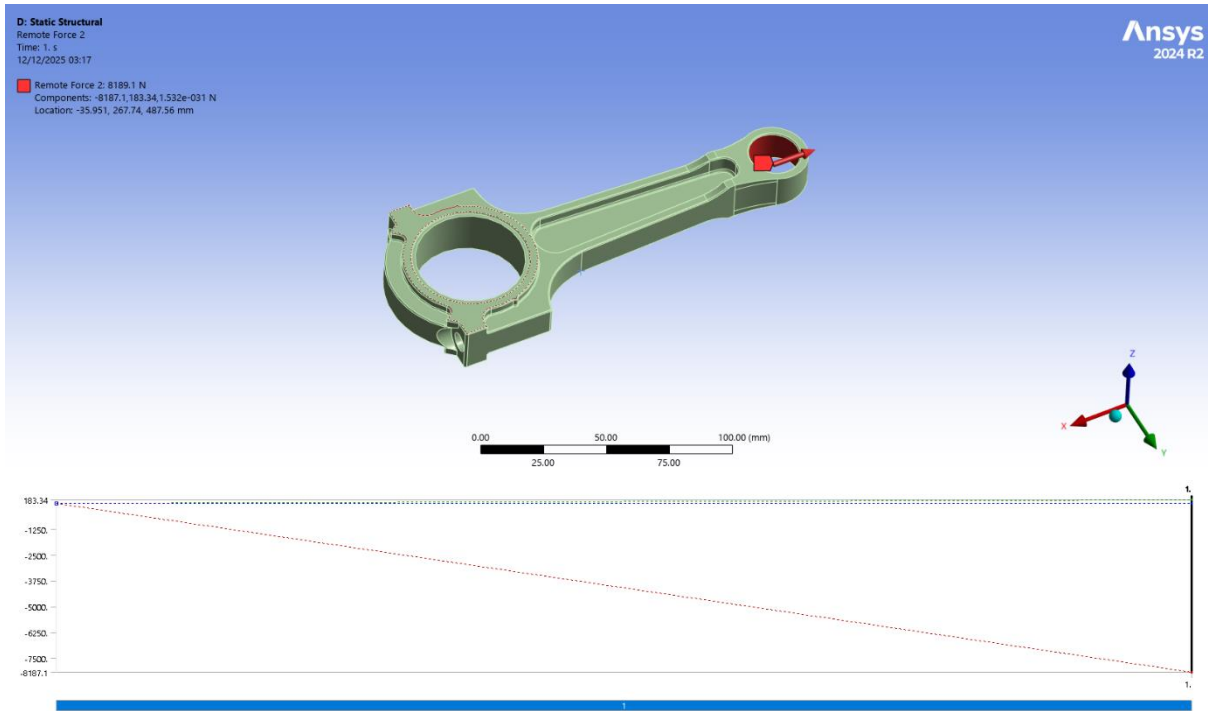


[Figure 8] Cylindrical loading

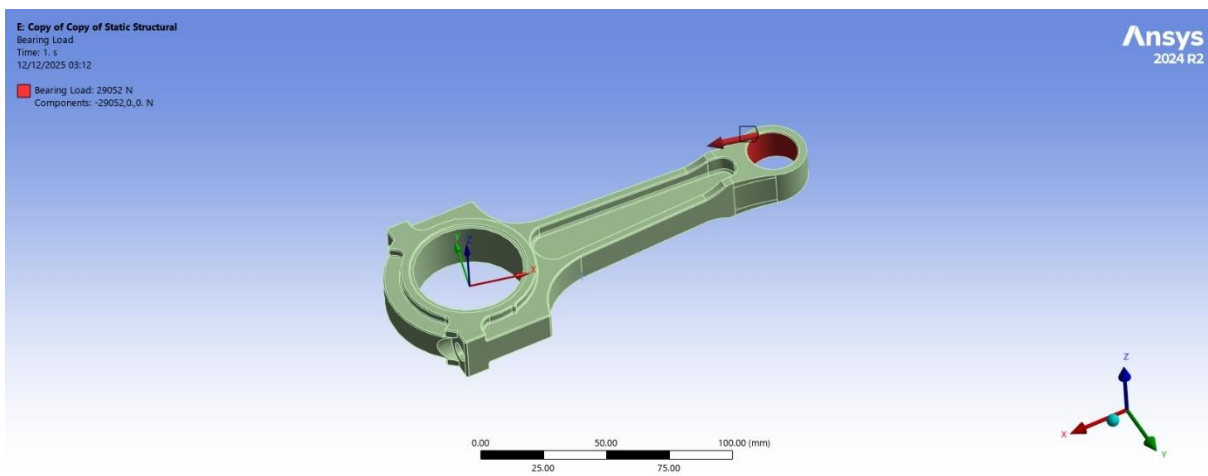
To evaluate the fatigue life, it was necessary to identify the greatest force loadings applied to the connecting rod through the ends of the part, and these were the extreme combustion pressure (compression) during the power stroke and the inertia of the piston/rod assembly (high tensile/pulling forces) at high RPMs.

The highest pressure, 72.631 bar, exercised by the gas on the piston in the cylinder is registered at 6500 rpm, at a 14° degrees crank angle (maximum compression) corresponding to a force of 29,052 N [Figure 9] , while the registered inertia measured was 8189.1 N [Figure 10] .

$$Force = Area [m^2] \times p \left[\frac{N}{m^2} \right] = 4 \times 10^{-3} \times 7,263,100 = 29,052.4 \text{ N}$$



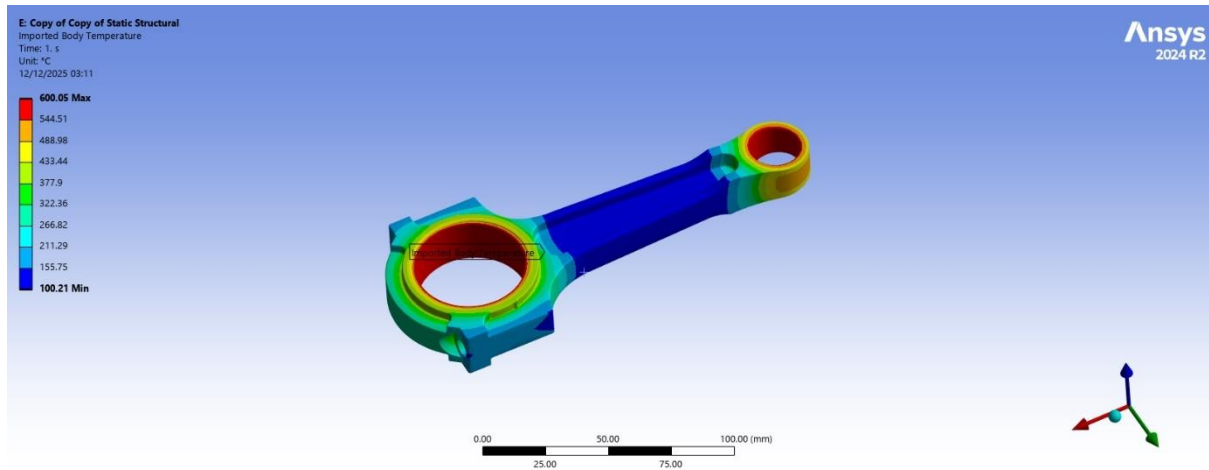
[Figure 9] Maximum tensile strenght



[Figure 10] Maximum compression

A thermal analysis [Figure 11] has been performed to indicate the hottest areas, in particular the heat is caused by the hot aluminium piston in contact with the Gudgeon pin (small end) and the friction against the piston and the crankshaft. An average temperature of oil has been set to 100°. It must be mentioned though that the thermal condition on the connecting rod usually does not affect it [4] since AISI 4340 steel has a

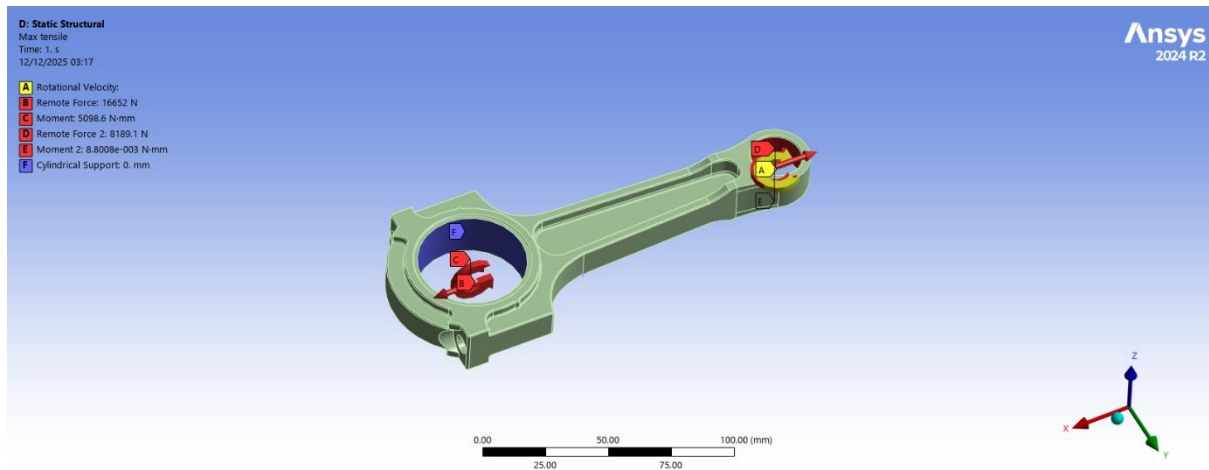
high heat resistance and strength at high temperatures. Excessive heat can cause connecting rods to deform but it's usually symptom of other problems like oil starvation, detonation, or overrevving [5] .



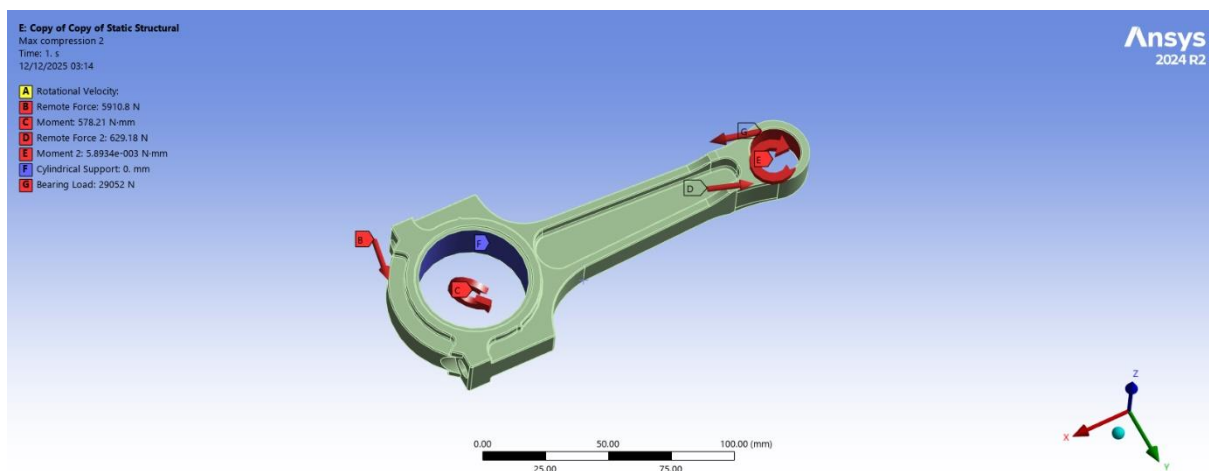
[Figure 11] Thermal analysis

2.2 Total Loads

The total loads measured during the maximum tensile forces applied are shown at [Figure 12], and the max compression forces reached are shown at [Figure 13] .



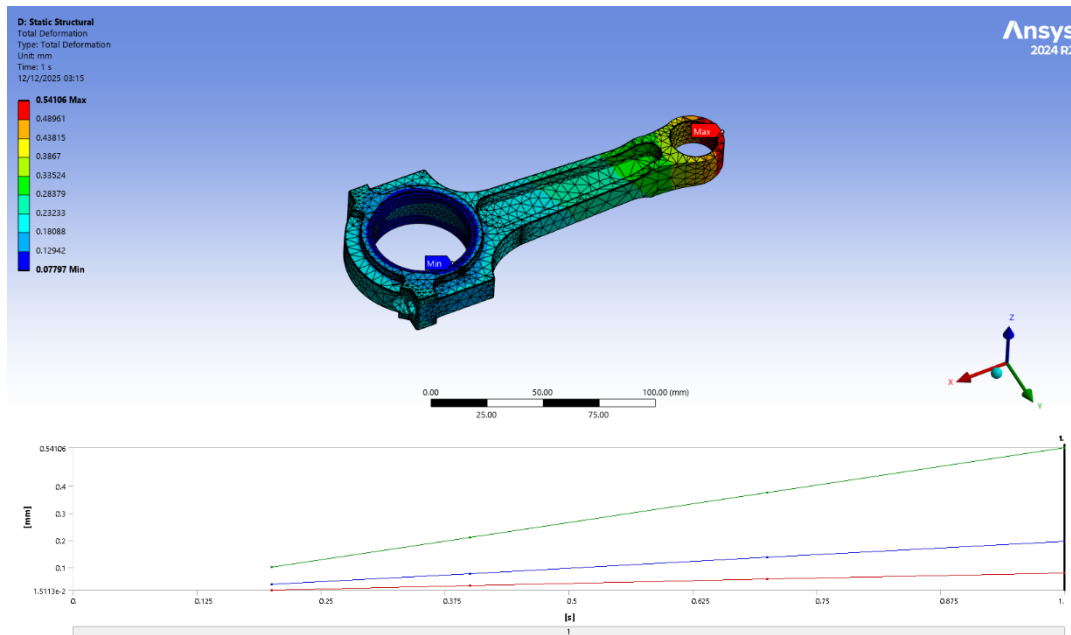
[Figure 12] Maximum tensile loads



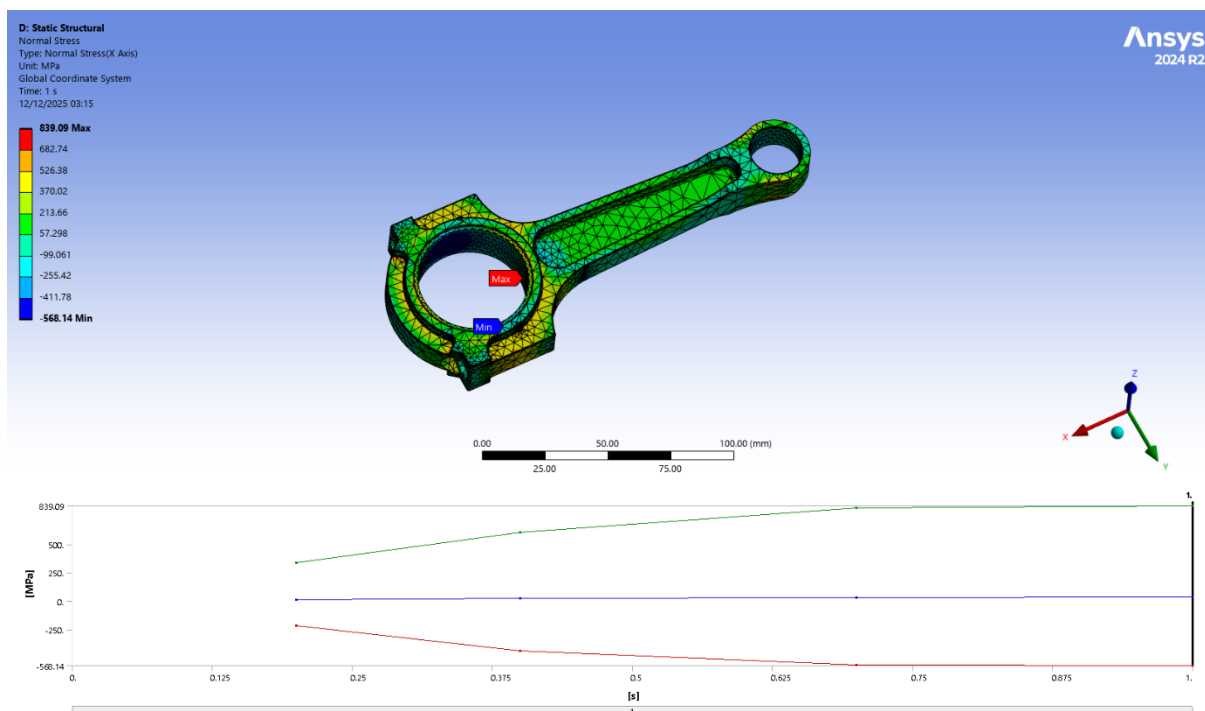
[Figure 13] Maximum compression loads

2.3 Results

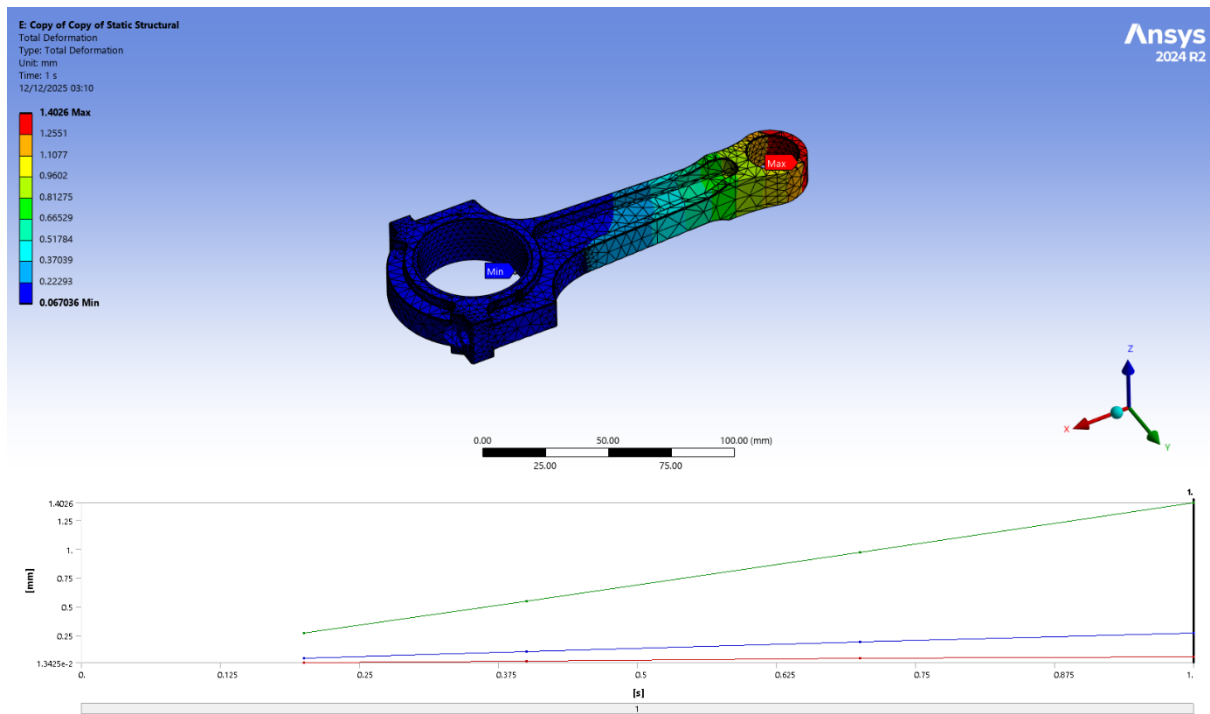
The total deformation [Figure 14] and normal stress [Figure 15] measured at maximum tensile stress are hereby presented with the ones measured during maximum compression [Figure 16], [Figure17].



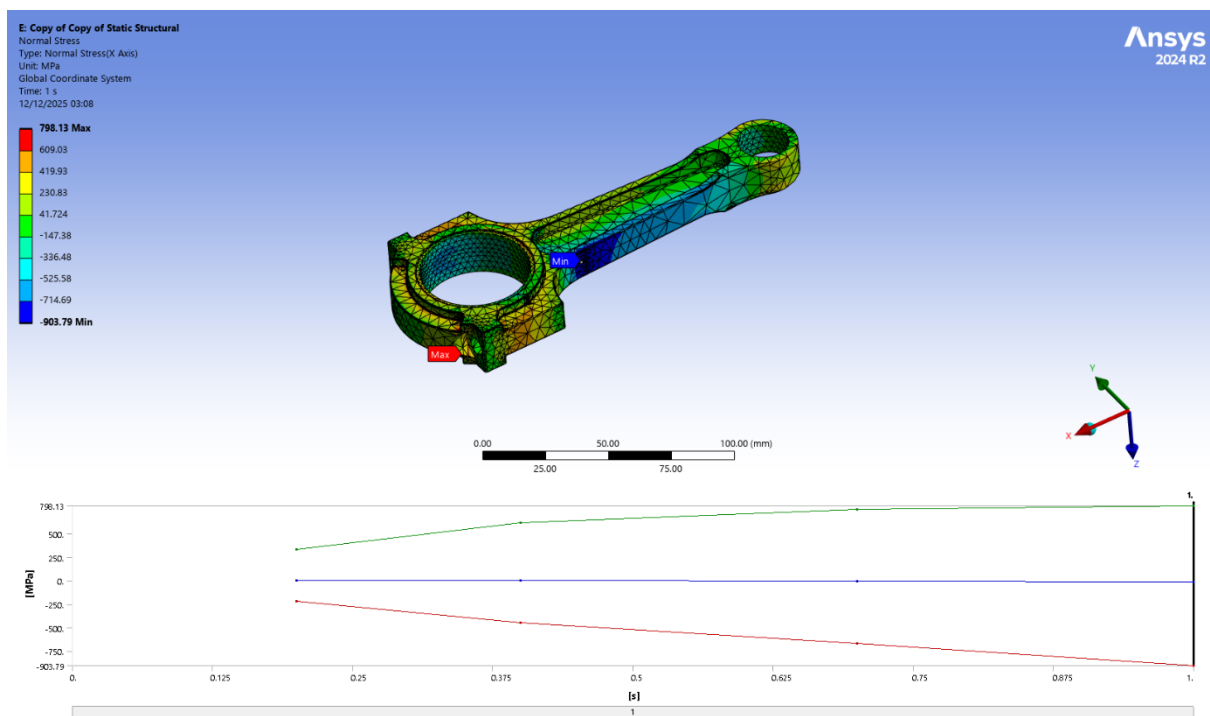
[Figure 14] Total deformation at maximum tensile stress



[Figure 15] Normal stress at max tensile stress



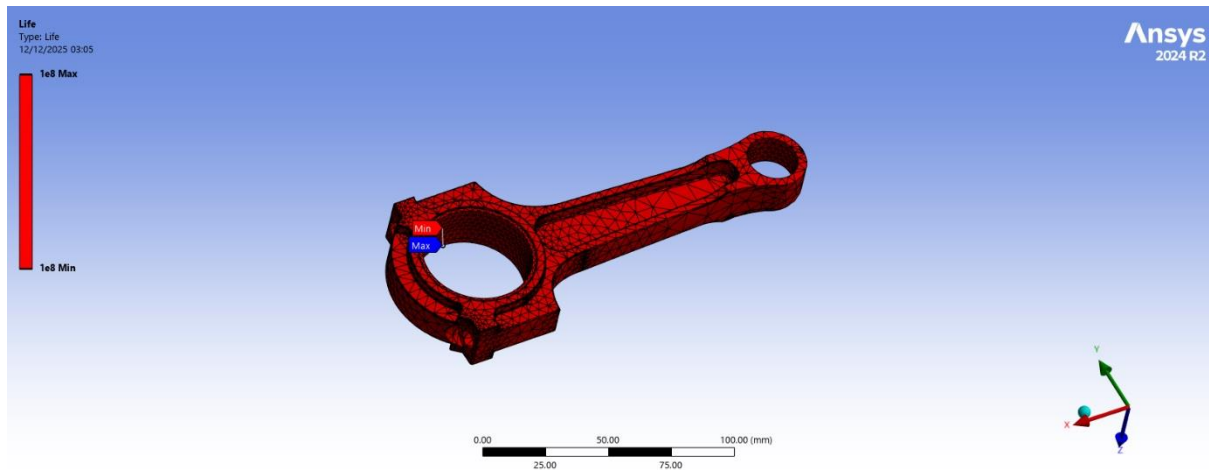
[Figure 16] Total deformation at max compression stress



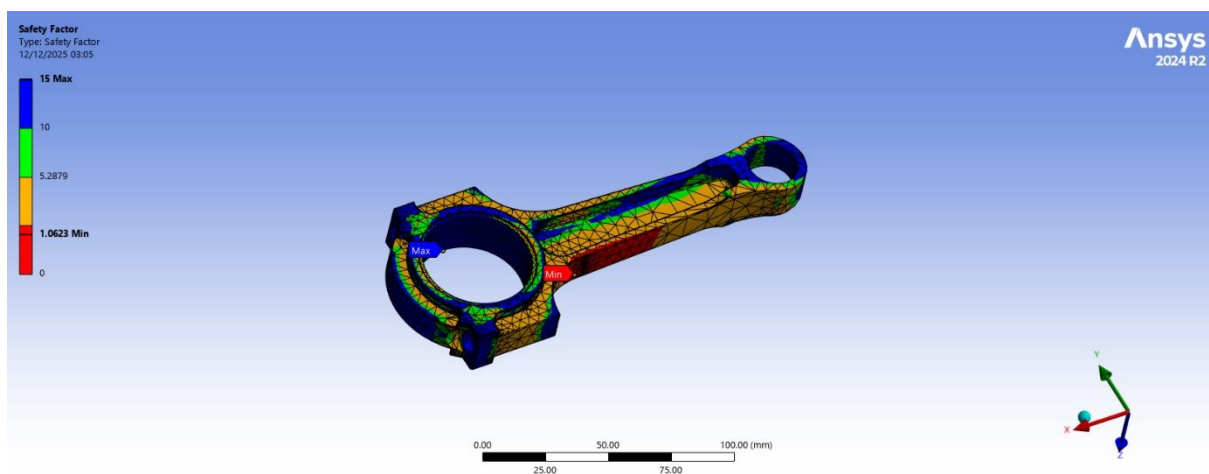
[Figure 17] Normal stress at max compression stress

3. Fatigue Life & Safety Factor Comparisons

The fatigue life and safety factor obtained from combining previous results as shown in [Figure 18], [Figure 19] .



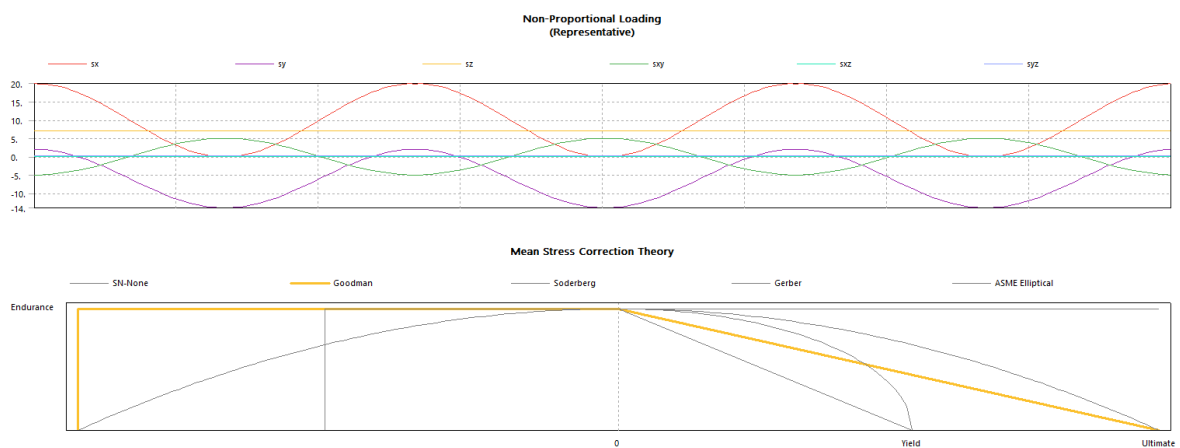
[Figure 18] Fatigue life of the connecting rod



[Figure 19] Safety factor of the connecting rod

Details of "Fatigue Tool"	
Domain	
Domain Type	Time
Materials	
Fatigue Strength Factor (Kf)	1.
Loading	
Type	Non-Proportional
Scale Factor	1.
Definition	
Combination	Last
Options	
Analysis Type	Stress Life
Mean Stress Theory	Goodman
Stress Component	Normal X
Life Units	
Units Name	cycles
1 cycle is equal to	1. cycles

[Figure 20] Parameters implemented in the life and safety factor analysis of the connecting rod



[Figure 21] Fatigue tool trend

4. Comparing the ANSYS analysis data with the analytical values shows similar results.

$$\sigma_{min} = -903.79 \text{ MPa}$$

$$\sigma_{max} = 798.13 \text{ MPa}$$

$$\sigma_{mean} = \frac{\sigma_{min} + \sigma_{max}}{2} = -52.83 \text{ MPa}$$

$$\sigma_{range} = \sigma_{max} - \sigma_{min} = 1701.92 \text{ MPa}$$

$$\text{Material UTS} = 1462.3 \text{ MPa}$$

$$\sigma_{limit} = 0.5 \times \text{UTS} = 731.15 \text{ MPa}$$

$$\sigma_{allow} = 2 \times \sigma_{limit} \left(1 - \frac{\sigma_{mean}}{\text{UTS}}\right) = 1515.13 \text{ MPa}$$

$$\text{Fatigue Safety Factor} = FSF = \frac{\sigma_{allow}}{\sigma_{range}} = 0.8902$$

The fatigue life analysis is quite compatible with the analytical equations and experimental data. The performance of steel in compression versus tension is characterized by a distinction between material properties and structural behaviour. While the raw material often has nearly equal yield strength in both tension and compression, steel structures are more critical in compression due to the risk of buckling [6]. Therefore, the area most affected by the compression phenomenon will also have a shorter fatigue life expectancy and smaller safety factor.

As for the life of the whole connecting rod instead, as previously mentioned, under normal circumstances it will not be the first component of the ICE to fail. Since the Fatigue Safety Factor is below 1 it means that there is a likelihood of fatigue failure, although the number of cycles may still be high. As a matter of fact, an infinite life of the component is unfortunately not realistic, which means that there is space for further implementing better inputs as to get a more genuine and close-to-reality data.

5. Discussions & Critical Evaluation

In this paper a structural fatigue analysis of a connecting rod has been conducted under realistic operating conditions, delivering good but not perfect results.

In particular, the geometry used to conduct the study did not implement connecting rods, notably famous for being the first components of the ICE to yield. In addition to that, even though the ANSYS model shows an infinite fatigue life, it contradicts the Fatigue Safety Factor. It is therefore clear that although ANSYS Mechanical is indeed an extraordinary workbench for simulation, validating the obtained data through empirical testing and analytical equations is still requested.

6. References

[1]	7075 Aluminium vs 6061 Aluminium https://www.thyssenkrupp-materials.co.uk/technical-knowledge-hub/7075-aluminium-vs-6061-aluminium
[2]	Advantages & Uses of 4340 Steel https://www.precisiongroundbars.com/blog/advantages-uses-of-4340-steel/
[3]	Failure analysis of a failed connecting rod cap and connecting bolts of a reciprocating compressor Xiaolei Zhu, Jing Xu, Yang Liu, Bo Cen, Xiaofeng Lu, Zhuo Zeng, School of Mechanical and Power Engineering, Nanjing Tech University, Nanjing 211816, China, 2017.
[4]	Effect of temperature increase on the looseness of connecting rod small end bushing Jiliang Liu, Jianhua Xiang, Wenjie Qin, Zhengxing Zuo, School of Mechanical Engineering, Beijing Institute of Technology, Beijing 100081, China, 2022
[5]	A repertoire of failures in connecting rods for internal combustion engines, and indications on traditional and advanced design methods A. Strozzi, A. Baldini, M. Giacomini, E. Bertocchi, S. Mantovani, Department of Engineering “Enzo Ferrari”, University of Modena and Reggio Emilia, Modena, Italy, 2015
[6]	Strength difference in compression and tension and pressure dependence of yielding in elasto-plasticity Rolf Mahnken, Kleine Pfahlstrasse 22, 30161 Hannover, Germany, 2001