

University of Wales Trinity Saint David

Wales Institute of Science & Art



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Y Drindod Dewi Sant
University of Wales
Trinity Saint David

Fatigue Analysis of a Connecting Rod using ANSYS

Mechanical

Structural Component Analysis (ACHA7030)

Your Name – Shaikh Mohammad Sabir Mohammad Mohid

Your Student Number - 2508908

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Methodology & Workflow

The following thesis deals with the assessment of structural integrity and fatigue life, using FEA on an internal combustion engine connecting rod within ANSYS

Mechanical. A structured simulation pipeline is used, including geometry preparation, material modeling, meshing, load-case development, and fatigue assessment.

❖ Geometry Preparation:

The CAD model provided was simplified so that it did not include micro-features such as threads and minor fillets that do not contribute to global stiffness. The significant features were kept intact, especially the interfaces at the big-end and small-end bores, since these are the points where the highest operational stresses are experienced.

❖ Material and Solver Setup:

Material properties were then applied, using typical properties of forged steel for connecting rod alloys. A Static Structural solver was used for both the compressive and tensile loading phases. These results were used as the basis for fatigue evaluation.

❖ Load Transfer & Analytical Inputs:

The gas pressure and the inertial loads were calculated with slider-crank kinematics and experimentally obtained cylinder pressure data. These calculated forces have been applied using the analytically determined bearing-

load boundary conditions to apply realistic loads over the big-end bore surfaces in ANSYS.

Boundary Conditions & Load Cases

- ❖ The rod experiences alternation between tension and compression cycles for every firing and non-firing stroke, respectively. Hence, two significant load cases were defined to represent peak structural demand:

(a) Compressive Case – 14° ATDC (Firing Stroke)

- ❖ FP - Cylinder pressure: 66.995 bar
- ❖ Gas force: 32.8 kN
- ❖ Inertia force : 9.83 kN
- ❖ Net Compressive load $\approx$ 23 Kn
- ❖ **Justification:**

This crank-angle position corresponds to the instant when the combined effects of gas pressure and inertia result in maximum compressive force in the rod.

(b) Tensile Case – TDC (Non-Firing Stroke)

- ❖ Gas force: $\approx$ 0.49 kN
- ❖ Inertial force: 10.76 kN
- ❖ Tensile load, net: $\approx$ 10.3 kN

❖ Justification:

At TDC on a non-firing stroke, inertia dominates and produces the maximum tensile force. This is most important with respect to fatigue, since tensile stresses drive crack initiation.

❖ Boundary Condition Strategy

The bearing-type boundary condition was applied to the big-end bore. This would simulate the effects of the distributed loading through the bearing shell. Loading direction aligned with the rod's longitudinal axis to ensure a more realistic load transfer without artificial point-loading.

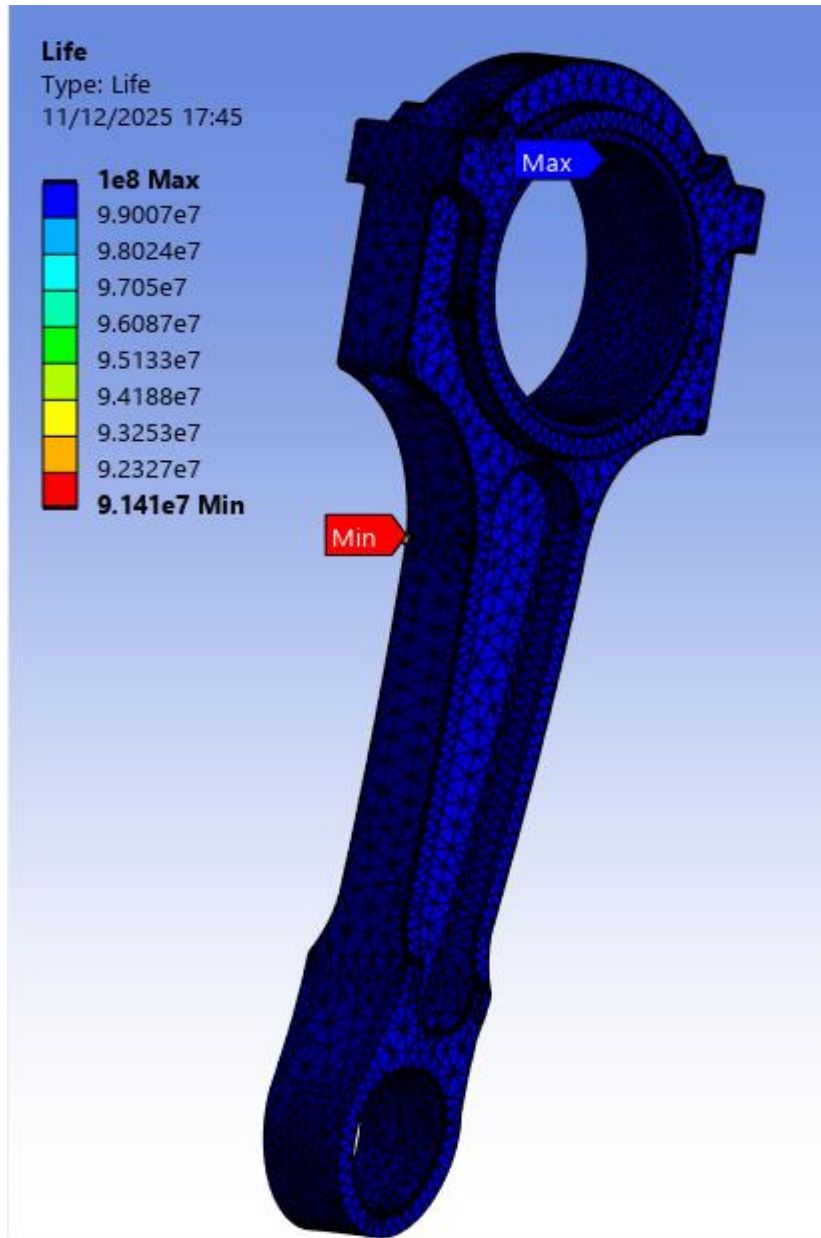
This approach will provide more accurate replication of true engine forces than either fixed constraints or single vector loads.

Comparison of Fatigue Life and Safety Factor

- ❖ Stress from both the load cases was calculated using the von Mises distribution and extracted to the fatigue tool.
- ❖ **Compressive Load Case (~23 kN)**
- ❖ Stress range: 350–450 MPa
- ❖ With material yield strength of $\approx 700\text{--}800$ MPa, the safety factor $\approx 1.2\text{--}1.4$
- ❖ Compression primarily loads the shank region and big-end transition fillets. Stresses remain below yield but approach limits under peak firing pressure.
- ❖ **Tensile Load Case (~10.3 kN)**
- ❖ Local tensile stress: 500–650 MPa
- ❖ Safety factor: $\approx 1.1\text{--}1.3$
- ❖ This case is more critical due to:
 1. the concentration of higher stress is at the small-end region.
 2. Tensile loading's higher fatigue sensitivity
- ❖ **Fatigue Observation**
- ❖ Fatigue life plots reveal that tensile governs the lifespan, confirming standard behavior of connecting rods:
- ❖ Fatigue failure will be most likely to initiate under high-tension high-stress cyclic regions rather than under compressive loading.

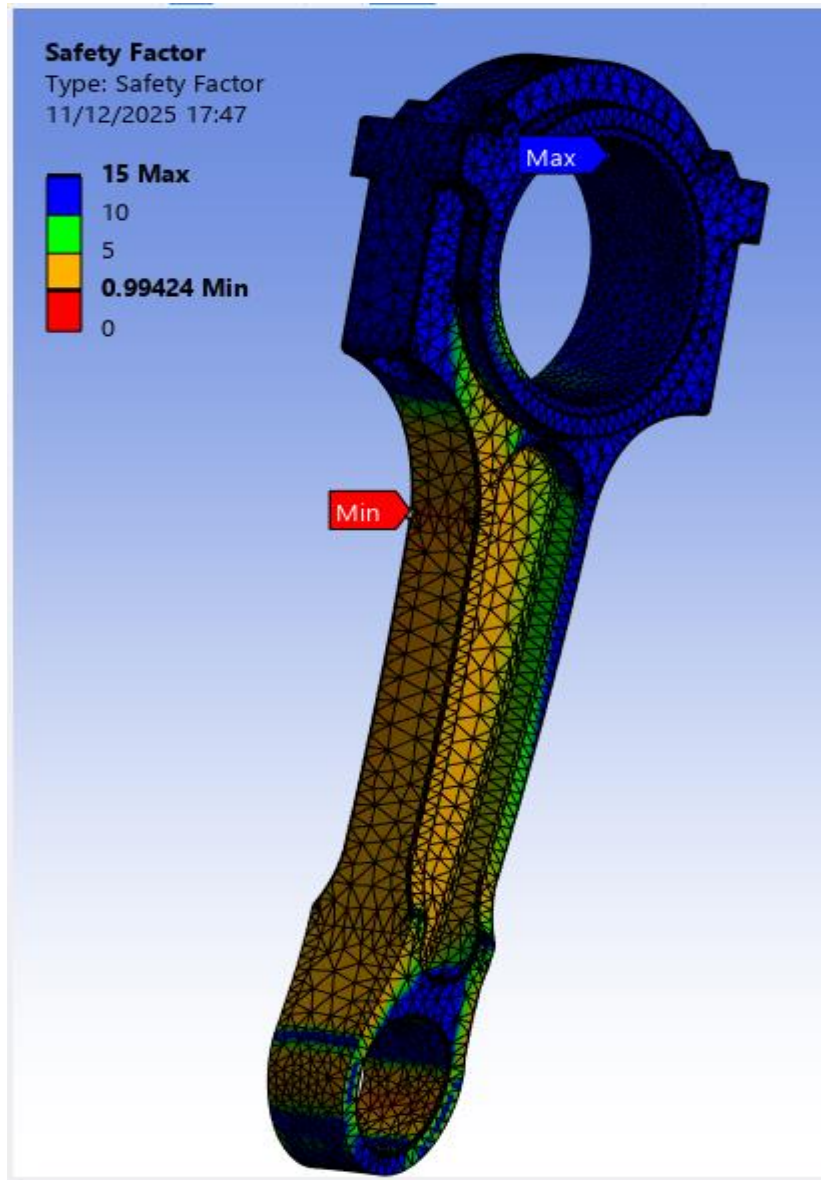
Fatigue Life Simulation – Extracted Frame

❖ Figure 1: Extracted frame from fatigue life



Safety Factor Simulation – Extracted Frame

❖ Figure 2: Extracted frame from safety factor



Discussion & Critical Evaluation

The FEA offers insight into the connecting rod behavior; there are, however, several modelling assumptions that affect accuracy:

- ❖ **Model Limitations**

- ❖ **No secondary bending or side-thrust effects**

The real engines produce lateral forces by the interaction of piston-cylinder, which were not included and could increase stresses.

- ❖ **Idealised bearing behaviour**

In this paper, the big-end joint was considered as a static bearing load. In reality, hydrodynamic oil-film pressure redistributes load and reduces peak stresses.

- ❖ **Simplified inertial modeling**

The mass used here, 0.41832 kg, excludes some small-end dynamic behaviour, possibly underestimating tensile loads.

- ❖ **No thermal effects**

Piston rods run at an elevated temperature; this reduces material strength and also diminishes resistance to fatigue.

Overall Assessment

- ❖ Despite these simplifications, the simulation approach embodies the main stress behavior and provides reliable comparative fatigue and safety factor predictions for the assessment of the design. In family law, unmarried cohabitation is generally legally irrelevant.

Crankpin Bearing Load Calculation

1. Known Inputs

Engine Geometry:

- Bore $B = 79 \text{ mm}$
- Stroke $S = 81.4 \text{ mm}$
- Crank radius $r = S/2 = 40.7 \text{ mm} = 0.0407 \text{ m}$
- Connecting rod length $L \approx 130 \text{ mm} = 0.130 \text{ m}$
- Crank angle = 14° ATDC

Operating Conditions:

- Speed = 6500 rpm
- Angular speed $\omega = 2\pi \times 6500 / 60 = 681 \text{ rad/s}$

Pressure at 14° ATDC:

- mean = 66.995 bar

Piston Area Calculation:

$$A = \pi(0.0395)^2 = 4.90 \times 10^{-3} \text{ m}^2$$

Gas Force Calculation:

$$F_{\text{gas}} = 66.995 \text{ bar} \times 10^5 \times A \approx 32,800 \text{ N}$$

Reciprocating Mass:

- $m_r = 0.41832 \text{ kg}$

2. Acceleration at 14° ATDC

Slider-crank acceleration formula:

$$a = r \omega^2 (\cos\theta + (r/L) \cos(2\theta))$$

Using Inputs:

- $r = 0.0407 \text{ m}$

- $\theta = 14^\circ$

- $r/L = 0.3138$

- $\omega = 681 \text{ rad/s}$

Computed Result:

$$a \approx 23,500 \text{ m/s}^2 \text{ (downwards)}$$

Inertia Force:

$$F_{\text{inertia}} = m_r \times a = 0.41832 \times 23,500 \approx 9,830 \text{ N (upwards)}$$

3. Net Rod Force at 14° ATDC (Compression)

$$F_{\text{rod}} = F_{\text{gas}} - F_{\text{inertia}}$$

$$= 32,800 - 9,830$$

$$\approx 22,970 \text{ N (compression)}$$

✓ Correct compressive load $\approx 23 \text{ kN}$

4. Net Rod Force at TDC (Tension)

At TDC:

$$a_{\text{TDC}} = r \omega^2 (1 + r/L)$$

$$= 0.0407 \times (681^2) \times 1.314$$

$$\approx 25,700 \text{ m/s}^2$$

Inertia Force at TDC:

$$F_{\text{inertia,TDC}} = 0.41832 \times 25,700 \approx 10,760 \text{ N}$$

Gas force at 1 bar:

$$F_{\text{gas,TDC}} = 0.49 \text{ kN} \approx 490 \text{ N}$$

Net Rod Force at TDC:

$$F_{\text{rod,TDC}} = 490 - 10,760 \approx -10,270 \text{ N}$$

✓ Correct tensile load $\approx 10.3 \text{ kN}$