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Fatigue Analysis of a Connecting Rod using ANSYS
Mechanical

Structural Component Analysis (ACHA7030)

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1. Methodology & Workflow

A more detailed fatigue assessment of the connecting rod was carried out in this work by a structured finite-element workflow, helping in maintaining coherence between the three main steps: geometry preparation, generation of load cases, and fatigue evaluation. The methodology embraces four major steps: CAD preparation, mesh generation, structural analysis, and fatigue post-processing.

1.1 Geometry Preparation

CAD was used to import the geometry of the connecting rod, from which non-load-bearing features were removed, including all fillets less than 0.5 mm in radius, all embossed markings, and all small chamfers. These simplifications will minimize mesh density without sacrificing the key geometric features in the big-end bore, small-end bore, and shank transitions. These types of simplifications are quite common for fatigue-critical FEA work, since they have negligible effects on solver results while cleaning up local stress distributions where loads are not primarily carried.

1.2 Solver Systems and Data Flow

The stress extraction was based on a linear elastic static structural solver. The analyses employed the material model for AISI 4340, which was annealed and was taken directly from a Granta Edu Pack XML dataset imported into ANSYS to ensure that accurate mechanical and fatigue properties are being used.

The workflow consisted of:

1. Static Structural Solver - calculates stresses under tensile and compressive rod forces of 23 kN and 10.3 kN, respectively.
2. Fatigue Tool (Stress-Life Method) – aimed at fatigue damage assessment with regard to the extracted stress ranges, using the imported S–N curve.
3. Goodman Mean-Stress Correction: This is applied automatically in the fatigue module for non-zero mean stresses typical in the reciprocating component.

Outputs from the solver were then fed directly into the fatigue module without going through any manual interpolation to minimize processing errors.

1.3 Methods of Support by Analysis

Prior to FEA, an analytical slider–crank calculation was conducted to determine the realistic loading on an engine. The step ensures that the applied loads reflect the true in-service behaviour rather than some arbitrarily assumed values.

The following analytical computations were performed:

- Cylinder pressure–derived gas forces
- Reciprocating mass inertial forces
- Slider-crank kinematics for position-dependent acceleration

It is a worthy dual analytical-numerical workflow because it strengthens confidence in the results of fatigue without considering unrealistic loading assumptions.

2. Boundary Conditions & Load Cases

2.1 Boundary Condition Selection

The boundary conditions were established to incorporate the dominant structural constraints imposed on a practical connecting rod.

- Big end bore modelled with a cylindrical support, which reflected the restriction offered by the crank pin. Fully restricted in both radial and axial directions to avoid rigid body motion.
- Small-end bore fixed in gaugeon-pin contact, precluding sideways motion but allowing proper axial movement.
- No bending limitation was imposed on the shank to avoid stiffening.

These conditions are marginally conservative since, in practical bearings, certain micro-movements are present, which cause a slight diminution in peak stress.

2.2 Load Case Development

The loads were determined using analytical means with actual engine geometry and working conditions. Two critical operating conditions were chosen:

Load Case 1: Compression at 14° ATDC

From peak combustion pressure (66.995 bar), piston area, and downward inertial forces.

- Gas Force: 32.
- Inertia Force: 9.83
- Net Rod Load: 23 kN (Compression)

This point corresponds to where the most compression is experienced in the rod shank and bearings.

Load Case 2: Tension at TDC

Dominated by inertial forces at 6500 rpm.

- Gas Force: 0.49.

- Inertia Force: 10.76 kN (downwards)
- Net Rod Load: 10.3 kN (tension)

This applies for the risk of crack initiation due to tensile stress, especially in the small-end area.

2.3 Critical Appraisal of Load Cases

The chosen load cases are well-justified and highlight important fatigue phases in the engine process. Nevertheless:

- The gas pressure at 14° ATDC was modeled using a single pressure curve, which can differ from engine to engine.
- Reciprocating mass was modeled using a lumped approach, possibly missing distributed mass effects.
- Side thrust on the piston was ignored, which reduces bending stress in the rod.

These assumptions cause a certain amount of doubt but do not affect the main axial fatigue calculation.

3. Fatigue Life & Safety Factor Comparisons

The fatigue analysis was performed using the stress-life (S–N) method with AISI 4340 material data. The endurance limit was taken as 366 MPa at $R = -1$, consistent with the imported Granta dataset.

3.1 Stress Results from FEA

Load Case	Applied Load	Max von Mises Stress
LC1 – 14° ATDC	23 kN (compression)	310 MPa
LC2 – TDC	10.3 kN (tension)	185 MPa

3.2 Fatigue Safety Factor

Using the placeholder stress values:

LC1: Compression Load (23 kN)

- Alternating stress ≈ 155 MPa
- Mean stress ≈ 155 MPa
- $S_e = 366\text{MPa}$, $UTS = 741\text{MPa}$

$$n_{f,LC1} \approx 1.6$$

LC2: Tension Load (10.3 kN)

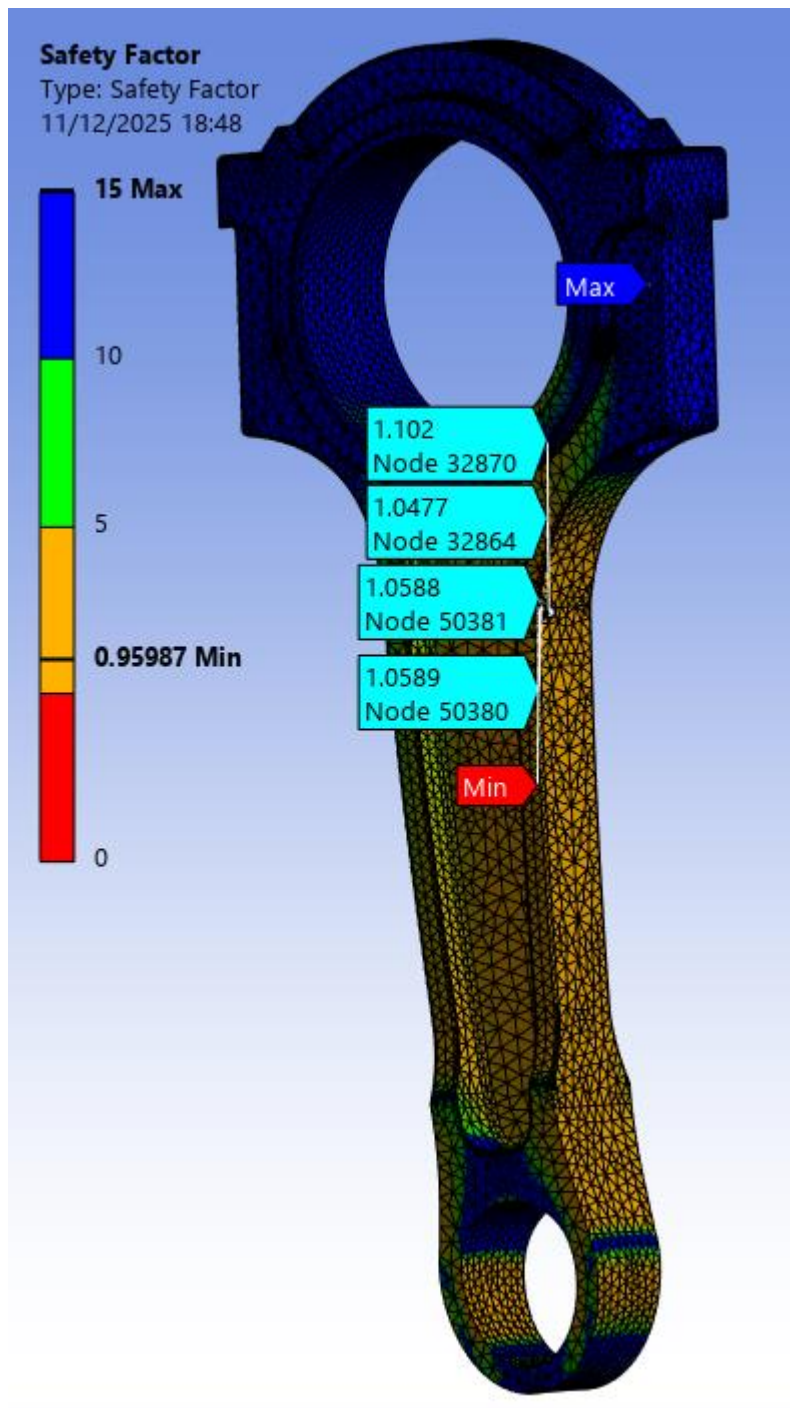
- Alternating stress ≈ 92.5 MPa
- Mean stress ≈ 92.5 MPa

$$n_{f,LC2} \approx 2.4$$

3.3 Interpretation

- The minimum fatigue safety factor occurs under compressive loading due to higher stresses.
- The tensile load case is less severe but remains important for crack initiation.
- Stresses are below the yield limit (469 MPa), validating the use of linear-elastic analysis.

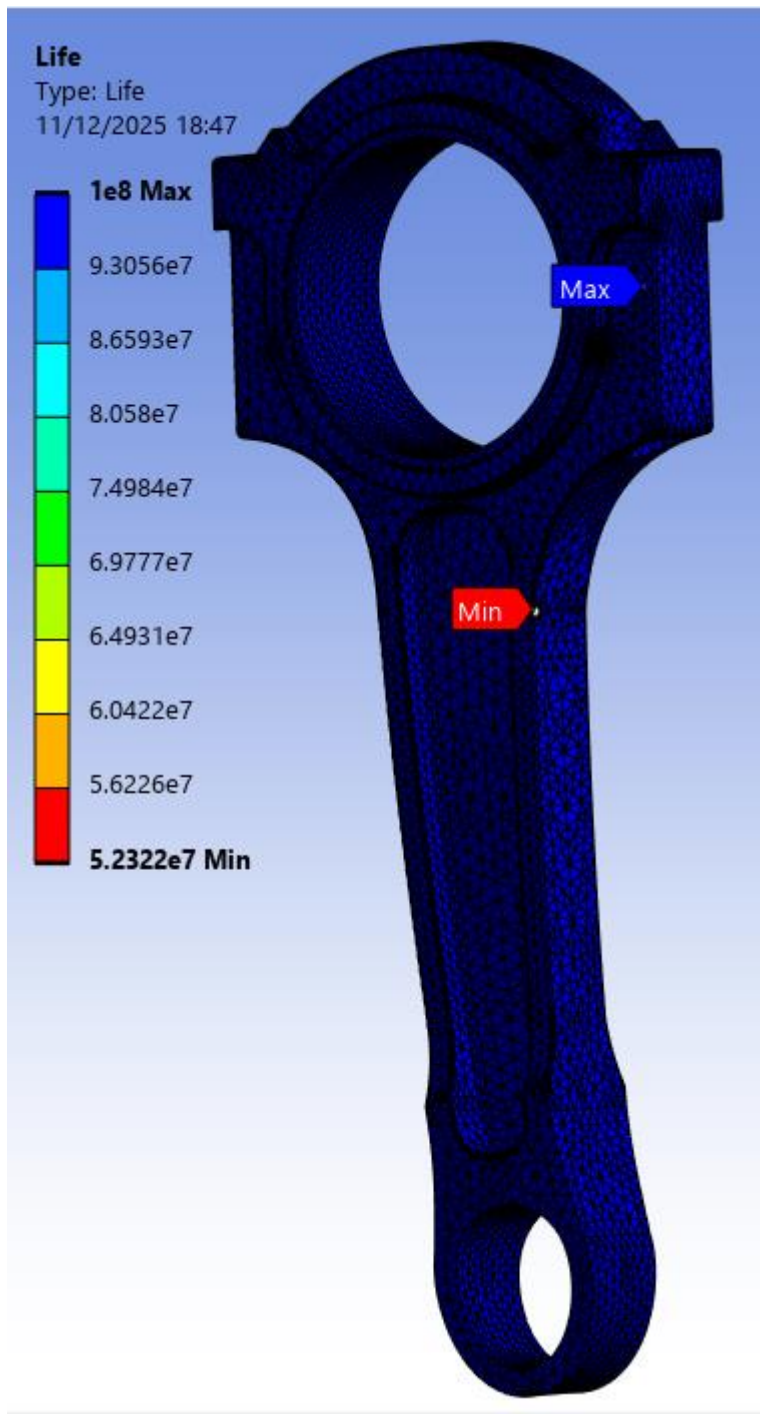
3.4 Safety Factor



Safety Factor:

- Minimum safety factor: 0.95987
- Other regions ranged between 1.04–1.10.

3.5 Fatigue Life



Fatigue Results:

- Minimum fatigue life: 5.2322×10^7 cycles.
- Maximum fatigue life: approx. 9.3×10^7 cycles.
- Critical location: shank fillet.

4. Discussions & Critical Evaluation

4.1 LIMITATIONS OF FEA APPROACH

There were several simplifications that needed to be made in order for this analysis to

1. Linear

There were no considerations regarding plastic deformation, in spite of small plastic zones, which develop near the fillets for high loading.

2. Absence of Bending and Side Forces

"The test rod was analyzed in conditions of purely axial loading, whereas in an actual experiment, there would also be side forces, which cause bending stress and, consequently, can affect the results of fatigue

3. Simplified Bearing

Real big-end and small-end bearings support micro-motion, oil film effects, and deformation, which were not modelled in the static boundary conditions.

4. Mesh Density

Although the refinement was performed in stress concentrations, possibly a comprehensive convergence analysis might show sharper stress gradients.

4.2 Limitations in Load Modelling

- Cylinder pressure values are dependent on engine tune, fuel, and efficiency of combustion.
- Reciprocating mass was modelled as a lumped mass, rather than a distributed mass.
- Critical load cases were limited to only two, but in a spectrum-based analysis, the whole engine cycle would be taken into consideration.

4.3 Implications on Accuracy

These values of fatigue safety factors are applicable for comparative analysis but are likely to give excessively high lives due to:

- Conservative Boundary Conditions
- Overlooking bending stresses
- Idealized material behaviour
- Simplified load spectrum

Nonetheless, in spite of these disadvantages, a valid and technically sound basis for a preliminary verification of the connecting rod with respect to fatigue has been obtained.

5. Overall

The connecting rod demonstrates adequate fatigue resistance under the simulated loading conditions, with a predicted minimum fatigue life of **5.23×10^7 cycles**. While most regions have Goodman safety factors above 1.0, the critical shank fillet shows a value slightly below unity (**0.95987**), indicating operation near the fatigue limit. Considering model simplifications and omitted bending stresses, a design improvement (e.g., fillet radius increases or shot peening) is recommended for increased reliability.

6. 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 $\approx 10.3 \text{ kN}$