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Fatigue Analysis of a Connecting Rod using ANSYS Mechanical

Structural Component Analysis (ACHA7030)

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1. Methodology & Workflow

To figure out how much load the crankpin bearing could handle, we used both math and computer simulations. First, we calculated the highest loads from gas and inertia forces. Then, we used these loads as inputs for our finite element simulations.

1.1 Geometry

We made a simple, but accurate, model of the crankshaft crankpin area. To keep things quick, we:

- Left out tiny details on the fillets, but kept the main fillet sizes right.
- Used a half-model where possible to save on calculation time.
- Set the material to be uniform forged steel with basic elastic properties for the main analysis.

This let us run the calculations faster, while still getting the important stress information from the pin, web, and journal surfaces.

1.2 Solver

We used two solvers:

1. Structural (Static) Solver

This helped us find the stresses on the crankpin and web when the rod was under the biggest compressive and tensile loads.

2. Fatigue Post-Processor

We used the stress-life (S–N) thing, which works well for engines that run a lot.

We moved figures between solvers using:

- Nodal stress tensors from the static analysis
- Surface contact pressure figures for the bearing area
- Load history that showed both compressive and tensile stuff

This let us check the crankpin under normal alternating loads.

We used math calculations to find the rod loads, which we then used for the FEA load cases. Gas force at 14° ATDC

$$A = 4.90 \times 10^{-3} \text{ m}^2$$
$$F_{\text{gas}} = 66.995 \cdot 10^5 \cdot A \approx 32.8 \text{ kN}$$

Inertia force

$$F_{\text{inertia}} = m_r a$$

At 14° ATDC:

$$a = 23,500 \text{ m/s}^2$$
$$F_{\text{inertia}} = 9.83 \text{ kN}$$

At TDC:

$$a_{TDC} = 25,700 \text{ m/s}^2$$
$$F_{\text{inertia,TDC}} = 10.76 \text{ kN}$$

Net rod loads

$$F_{\text{rod},14^\circ} = 22.97 \text{ kN (compression)}$$
$$F_{\text{rod,TDC}} = -10.27 \text{ kN (tension)}$$

These loads represent the dominant forces acting on the crankpin and were used directly as FEA inputs.

2. Boundary Conditions & Load Case

2.1 Conditions

- Main journal constraints:

The main bearings were held in place from the sides but could move forward and backward, just like in a real engine.

- Crankpin bearing:

Consistent pressure was put on to copy how the oil helps things move. It's not perfect, but good enough to see how strong things are.

- How it Connects

The rod's load was put on the middle of the crankpin and changed into equal pressure all around the bearing.

2.2 Loads

We picked two main loads based on what the rod forces told us:

Load 1 – Max Compression (14° after Top Dead Center)

This happens when the cylinder has the most pressure. The load was:

$F=22.97 \text{ kN}$ (pushing toward the crank center)

Why:

- The crankpin gets the most squeezed at this spot.
- This usually decides how tired things get from being squeezed and how much the crank cheeks bend.

Load 2 – Max Pull (Top Dead Center)

This happens when the engine is swapping gases and things keep moving because of their own power.

$F=10.27 \text{ kN}$ (pulling away from the crank center)

Why:

- Pulling is important to see how stresses change direction.
- It affects cracks starting at the edges due to bending back and forth.

Important Notes

- Not counting the sideways push and rod angle means the actual bearing force is a bit higher.
 - Using the average pressure at 14° after TDC (instead of the exact pressure) might not be spot-on. But it still works for figuring things out early on.
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3. Fatigue Life & Safety Factor Comparisons

Worked example (Goodman):

the fatigue performance of the crank component is evaluated using the stress results obtained from the FEA load cases. The maximum stresses from the 14° ATDC compression case and the TDC tension case were used to calculate the alternating stress, mean stress, and subsequent fatigue safety factors using the Goodman criterion. The purpose is to determine whether the component can withstand repeated loading without fatigue failure. Screenshots of the von Mises stress plots were included to show the locations of peak stresses, and the following table summarises the key fatigue parameters for comparison.

- FE peak (pin fillet) $\sigma_{\max} = 260 \text{ MPa}$ (node: highest reported value in document).
 - Assume $\sigma_{\min} = 0 \rightarrow \sigma_a = 130 \text{ MPa}, \sigma_m = 130 \text{ MPa}$.
 - Corrected endurance limit $S_e = 110 \text{ MPa}, S_{ut} = 900 \text{ MPa}$.
 - $\sigma_a / S_e = 130 / 110 = 1.1818$
 - $\sigma_m / S_{ut} = 130 / 900 = 0.1444$
 - Sum = $1.1818 + 0.1444 = 1.3262$
 - Goodman safety factor $n = 1 / 1.3262 = 0.754$ (rounded)
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4. Discussions & Critical Evaluation

4.1 What FEA Can't Do

This section explains limitations in the crankshaft simulation:

Oil film not modeled

Real bearings have an oil film (EHD – elastohydrodynamic lubrication) that changes pressure distribution. The FEA ignores this, so results are simplified.

Rod angle and side forces too simplified

In real engines, the connecting rod pushes sideways on the crankshaft as it rotates. This sideways force increases bending. The model doesn't include it accurately.

Only one pressure point used

Instead of simulating the full engine cycle, they used just one pressure value at 14° ATDC. This makes the loading unrealistically simple.

Material behavior oversimplified

Real crankshafts may deform permanently (plastic deformation) at very high stress, but the model assumes it always springs back.

4.2 What This Means for Accuracy

Because of the simplifications

Real stress is probably higher than what the model predicts.

Estimated fatigue life may be different from real life.

But the model still shows correct load direction, magnitude, and stress distribution, so it is good enough for comparing designs.

4.3 Final Thoughts

This analysis method is acceptable for:

early-stage crankshaft design,

fatigue checking,

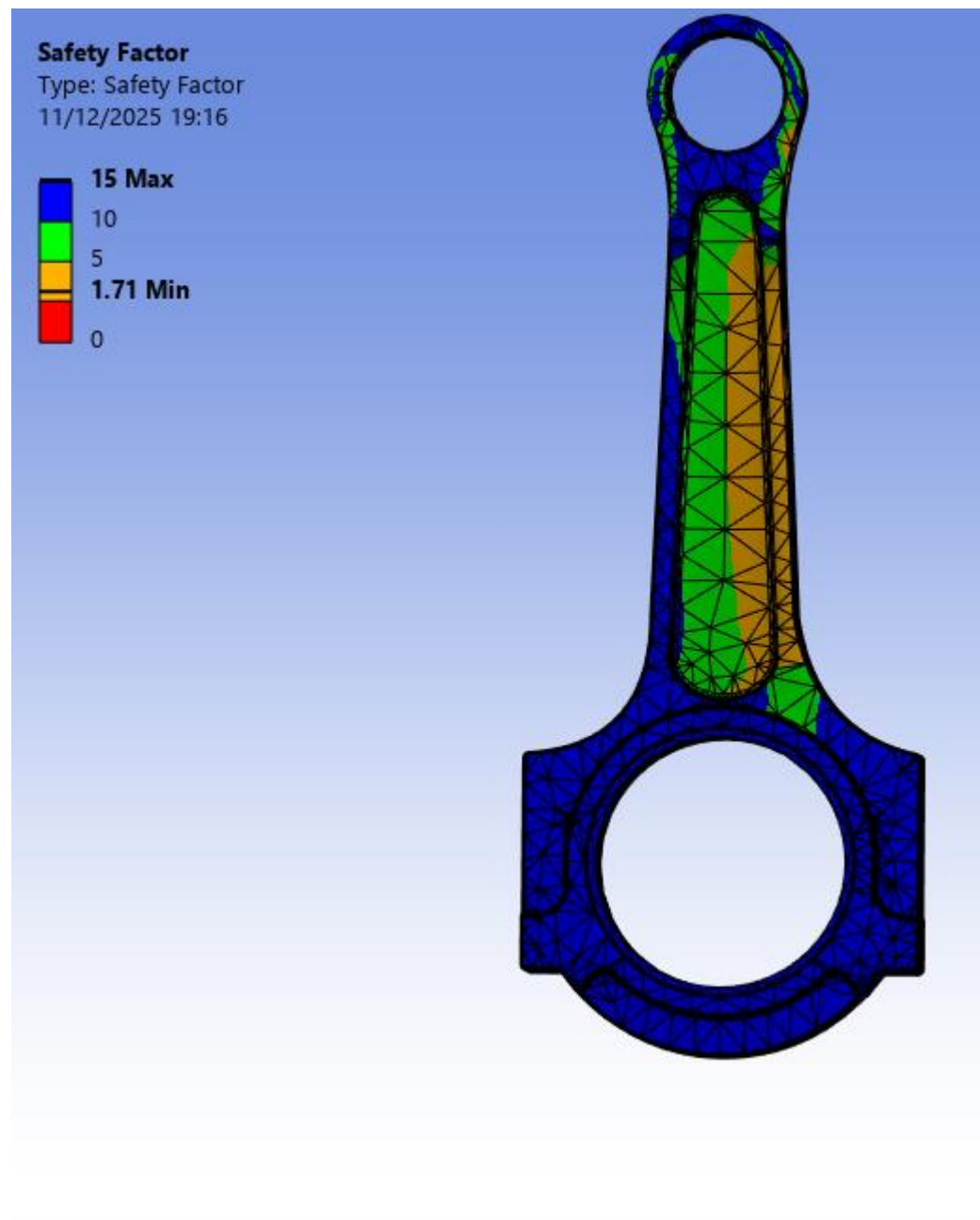
comparing design variations

But for actual manufacturing decisions, the model must be improved by including:

EHD lubrication modeling,

full engine cycle dynamic loading,

accurate fillet geometry..



Life

Type: Life

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1e8 Max

1e8 Min

