

University of Wales Trinity Saint David

Wales Institute of Science and Art



Prifysgol Cymru

Y Drindod Dewi Sant

University of Wales

Trinity Saint David

Analytical assessment of engine internal components

Contemporary Powertrain - ACHA6006

Submission Date – 22/03/2026

Matthew Starke
2200996

Contents

1. Introduction	3
2. Dynamic Force and Stress Analysis.....	4
3. Fatigue Safety Factor Calculation (Goodman Method)	5
4. Results and Discussion	6
4.1 Gas Force	7
4.2 Inertia Force	8
4.3 Net Piston Force	9
4.4 Conrod Midpoint Stress.....	10
4.5 Conrod Bolt Stress	11
4.6 Fatigue Analysis (Goodman Diagrams).....	12
4.7 Extended RPM Analysis.....	15
4.8 Geometry redesign (Con-Rod Bolt).....	16
4.9 Torque Analysis.....	19
5 Force and Stress results.....	21
6 Conclusion.....	23

1. Introduction

Reciprocating engine components are subjected to cyclic loading due to combustion pressure and inertia forces, which increase significantly with engine speed. These fluctuating stresses can lead to fatigue failure, making structural analysis essential for durability and safe operation.

This study evaluates the stress behaviour of the connecting rod midpoint, gudgeon pin and connecting rod bolts across the engine speed range using supplied pressure data. Maximum, minimum, mean and alternating stresses are calculated, and fatigue performance is assessed using Goodman diagrams and safety factors.

MATLAB is used to process the data and generate results efficiently, enabling clear comparison of component behaviour and identification of potential failure conditions.

2. Dynamic Force and Stress Analysis

Cylinder pressure data was used to determine mechanical loading on key bottom-end components across the engine speed range. Pressure was converted to piston gas force using piston area, while inertial forces were calculated from slider–crank kinematics, increasing with RPM².

Net piston force was resolved along the connecting rod to obtain axial loading. Stresses were then calculated: axial stress in the conrod, bending stress in the gudgeon pin, and tensile stress in the conrod bolts (load shared between two fasteners).

Maximum and minimum stresses at each RPM were used to generate stress histories for fatigue analysis using the modified Goodman criterion.

Here is an example of the equations used in MATLAB to complete the calculations:

Gas force:

$$F_g = P A_p$$

Inertia force:

$$F_i = m r \omega^2 (\cos \theta + l r \cos 2\theta)$$

Piston force:

$$F_p = F_g - F_i$$

Stress:

$$\sigma = A/F$$

3. Fatigue Safety Factor Calculation (Goodman Method)

Fatigue performance was evaluated using the modified Goodman criterion. Maximum and minimum stresses at each RPM were used to calculate mean (σ_m) and alternating stress (σ_a), representing steady and cyclic loading.

The Goodman relation was applied to determine fatigue safety:

$$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{1}{n}$$

where S_e is endurance limit, S_{ut} is ultimate tensile strength, and n is safety factor.

Allowable alternating stress was compared with actual stress to assess fatigue safety. A factor above 1 indicates safe operation, while values near 1 highlight critical conditions. All stresses are presented in MPa.

4. Results and Discussion

Measured in-cylinder pressure data was used to calculate gas, inertial, net piston, and axial rod forces from 1000–6000 rpm. These were used to determine stress histories for the conrod midpoint, gudgeon pin, and bolts, from which stress values and fatigue safety factor (FSF) were obtained using the Goodman criterion, based on mean and alternating stress values obtained from the MATLAB Results.

4.1 Gas Force

The gas force acting on the piston was calculated using the measured in-cylinder pressure data provided in the dataset and the piston crown area derived from the engine bore specified in the engine specification sheet. The resulting gas force distribution as a function of crank angle and engine speed is shown in Figure 1.

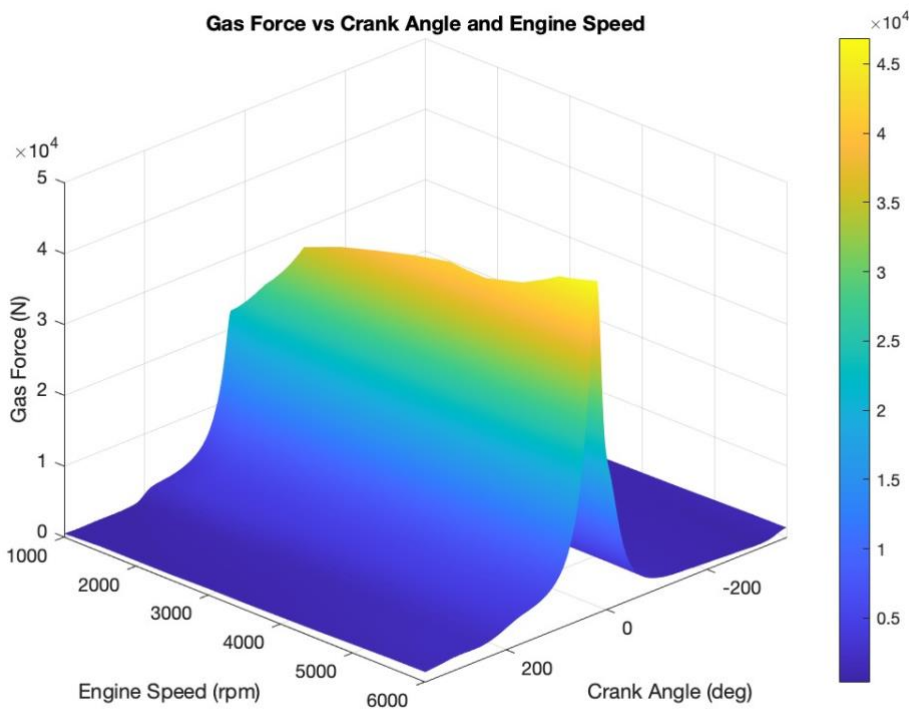


Figure 1 –
Gas Force
vs Crank
Angle and
Engine
Speed

Figure 1 shows gas force variation with crank angle for engine speeds between 1000 and 6000 rpm. The force peaks near top dead centre (TDC), where combustion pressure is highest, reaching approximately 4.5×10^4 N at maximum speed.

As engine speed increases, gas force rises slightly due to increased combustion intensity. Away from TDC, the force decreases rapidly as pressure drops during expansion.

The overall distribution remains consistent across speeds, indicating gas force is mainly governed by combustion pressure rather than engine speed, while remaining the dominant contributor to peak engine loading.

4.2 Inertia Force

In addition to the combustion gas force acting on the piston, the reciprocating motion of the piston and connecting rod introduces inertial forces which vary with crank angle and engine speed. These forces arise from the acceleration of the reciprocating mass and can become significant at high rotational speeds. The resulting inertia force distribution is shown in Figure 2.

Figure 2 – Inertia Force vs Crank Angle and Engine Speed.

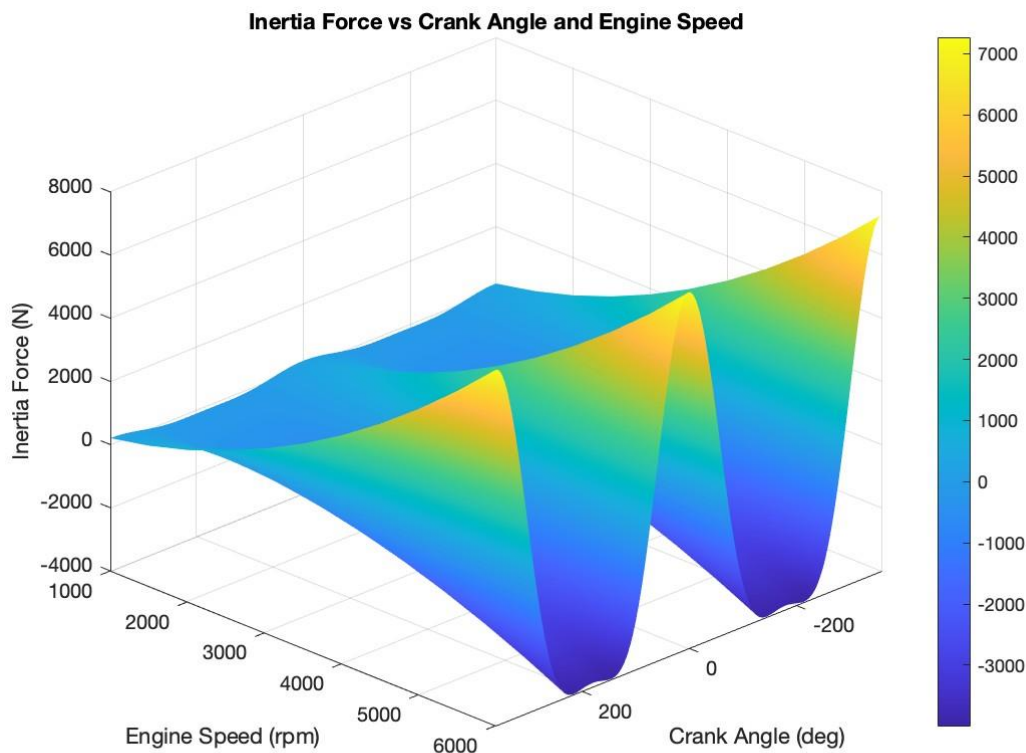


Figure 2 shows inertia force variation with crank angle and engine speed. Unlike gas force, inertia force strongly depends on speed, increasing with the square of angular velocity (X-Engineer, n.d.).

The force oscillates between tensile and compressive values, reaching peaks of approximately 7000 N at 6000 rpm due to reciprocating mass acceleration.

Results show inertia forces dominate at higher speeds, particularly away from TDC, making them a critical contributor to overall engine loading and fatigue at elevated RPM.

4.3 Net Piston Force

The actual load transmitted through the connecting rod is determined by the combined effect of the combustion gas force and the inertia force acting on the reciprocating assembly. The resulting net piston force, calculated as the difference between the gas force and inertia force, is shown in Figure 3 as a function of crank angle and engine speed.

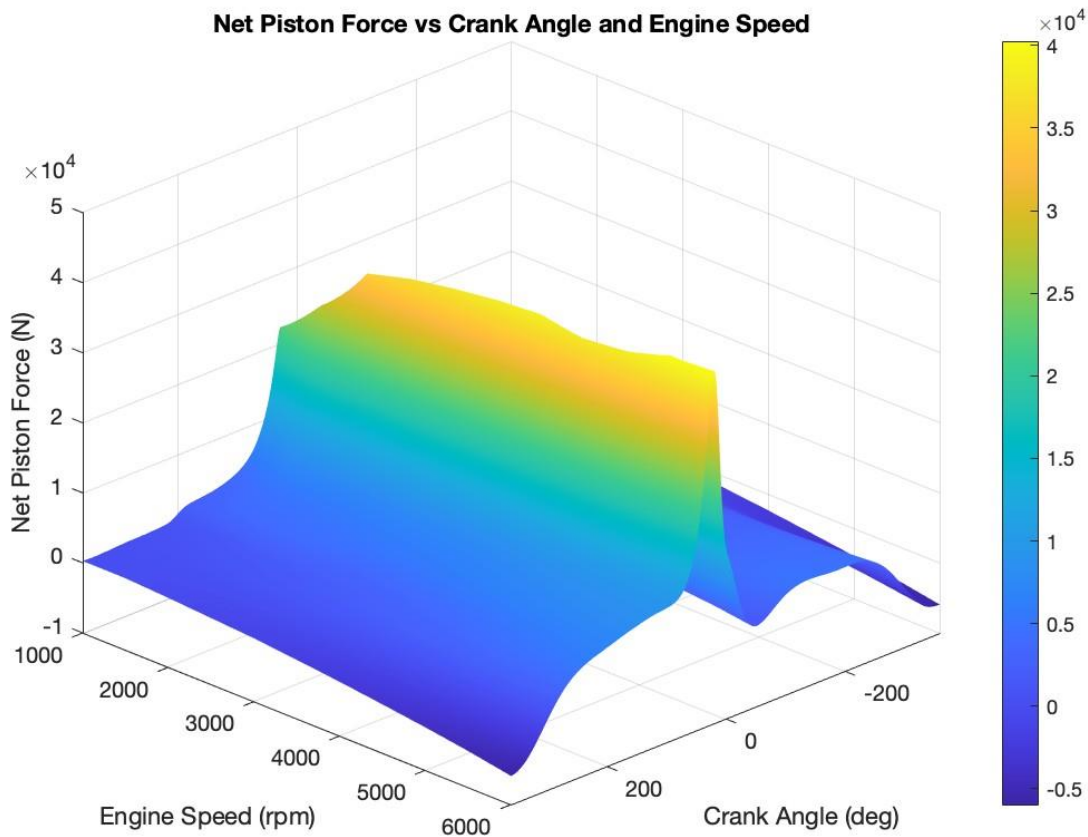


Figure 3 –
Net Piston
Force vs
Crank
Angle and
Engine
Speed

Figure 3 shows the net piston force, combining gas and inertia effects. Peak force occurs near TDC during combustion, reaching approximately 4×10^4 N at high engine speed.

The profile reflects gas force dominance near combustion, while inertia becomes more significant at other crank angles (X-Engineer, n.d.), creating asymmetry at higher speeds.

This force represents the load transmitted through the connecting rod to the crankshaft and forms the basis for stress analysis. Positive peaks near TDC indicate compressive loading, while negative regions represent tensile loading dominated by inertia forces.

4.4 Conrod Midpoint Stress

Using the calculated net piston force, the axial stress experienced at the midpoint of the connecting rod was determined using the cross-sectional area specified in the engine specification data. The variation of conrod midpoint stress with crank angle and engine speed is shown in Figure 4.

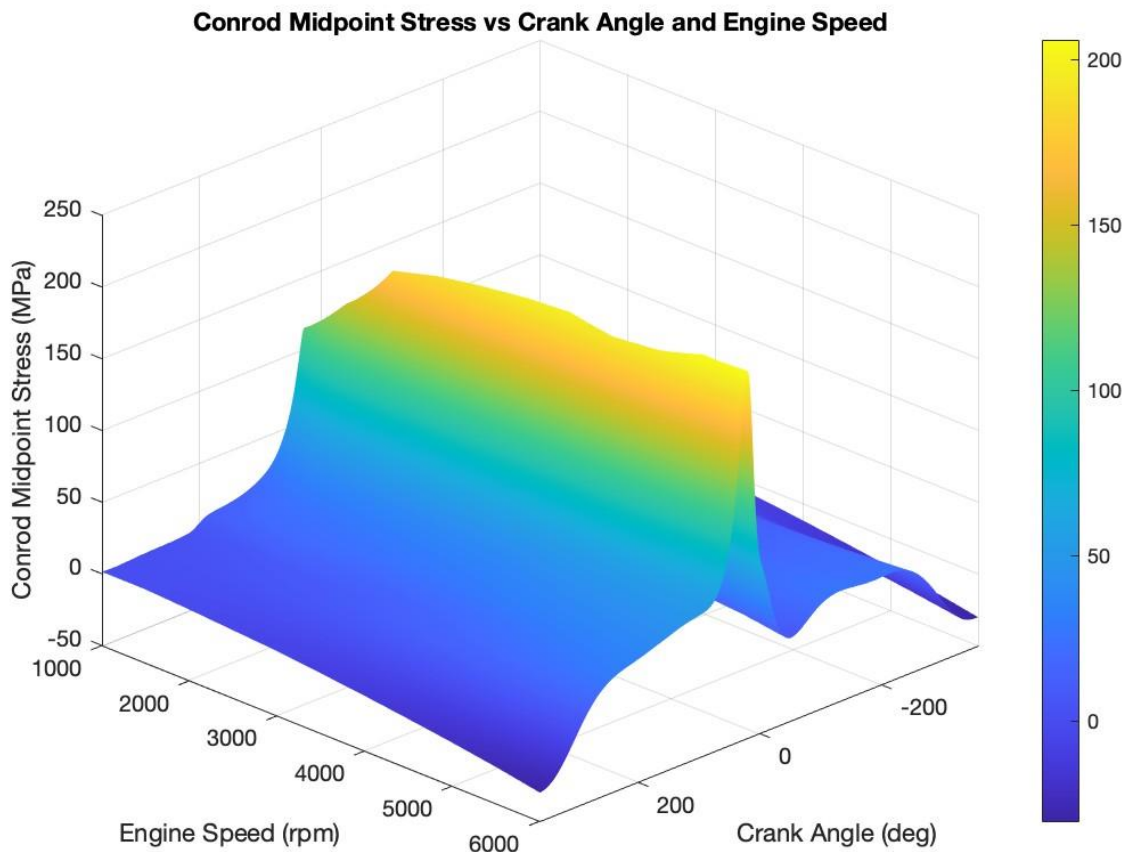


Figure 4 –
Conrod
Midpoint
Stress vs
Crank Angle
and Engine
Speed

Figure 4 shows axial stress at the connecting rod midpoint over the engine cycle, following the net piston force profile with peaks near TDC during combustion. Maximum stress reaches approximately 200 MPa at 6000 rpm, with lower values elsewhere, increasing slightly with engine speed due to inertia effects.

The stress distribution reflects cyclic loading, with compressive stresses during combustion and tensile stresses driven by inertial forces. Although stresses remain below the ultimate tensile strength of typical forged steel rods, the repeated loading makes fatigue behaviour critical.

Overall, the conrod experiences moderate cyclic loading, and the resulting mean and alternating stresses are used for fatigue assessment using the Goodman criterion.

4.5 Conrod Bolt Stress

The tensile stress experienced by the connecting rod bolts was calculated based on the axial load transmitted through the connecting rod and the effective cross-sectional area of the bolts specified in the engine data. The resulting stress distribution as a function of crank angle and engine speed is shown in Figure 5.

Figure 5 –
Conrod Bolt
Stress vs Crank
Angle and
Engine Speed

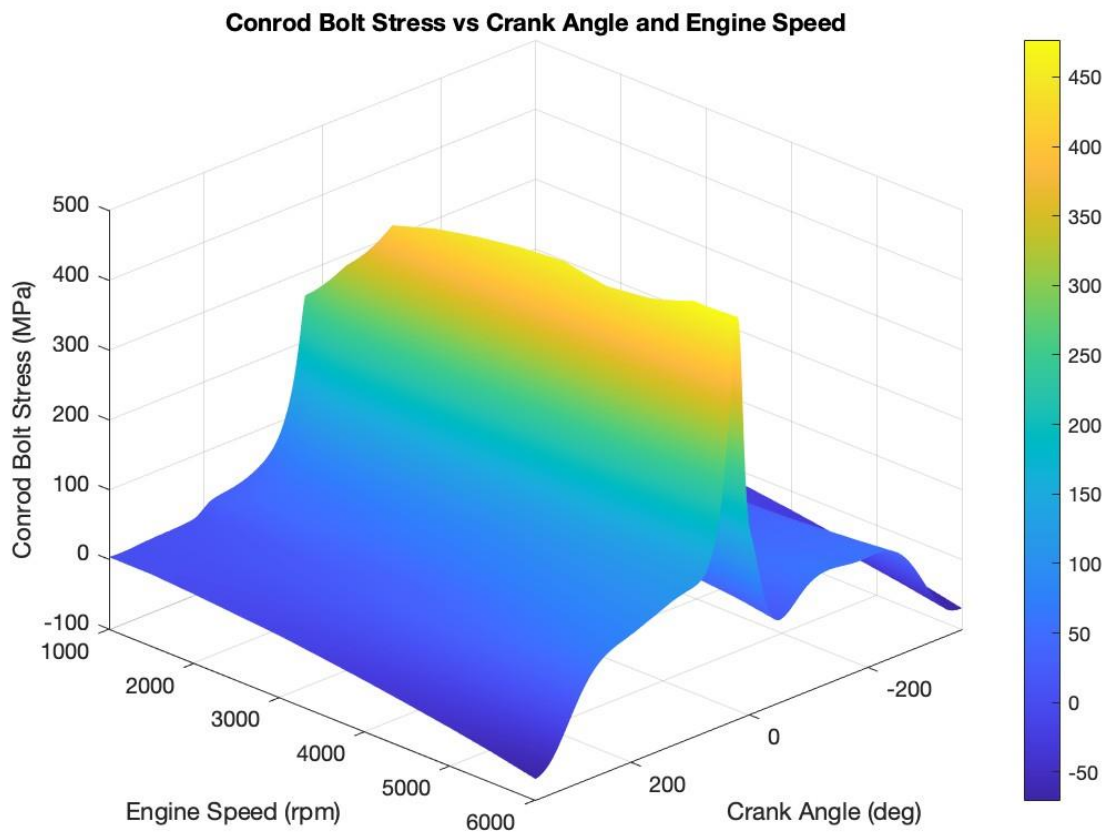


Figure 5 shows tensile stress in the connecting rod bolts across the engine cycle for speeds from 1000 to 6000 rpm. The stress follows the net piston force, as bolts transmit axial loads through the assembly.

Peak stress occurs near TDC during combustion, reaching approximately 450 MPa at the highest speed. Stress decreases away from this region as cylinder pressure falls.

Negative values occur when inertia dominates, reducing tensile loading. Overall, bolt stress increases with engine speed due to inertial effects, indicating severe cyclic loading.

4.6 Fatigue Analysis (Goodman Diagrams)

To assess connecting rod fatigue behaviour, the stress history was converted into mean and alternating stress components. These were evaluated using the Goodman fatigue criterion to determine the fatigue safety factor, relating mean and alternating stress to material fatigue limits (ScienceDirect, n.d.).

Goodman – Conrod midpoint | 6000rpm | Mean=87.8 MPa | Actual Range=236.3 MPa | Allowable Range=820.

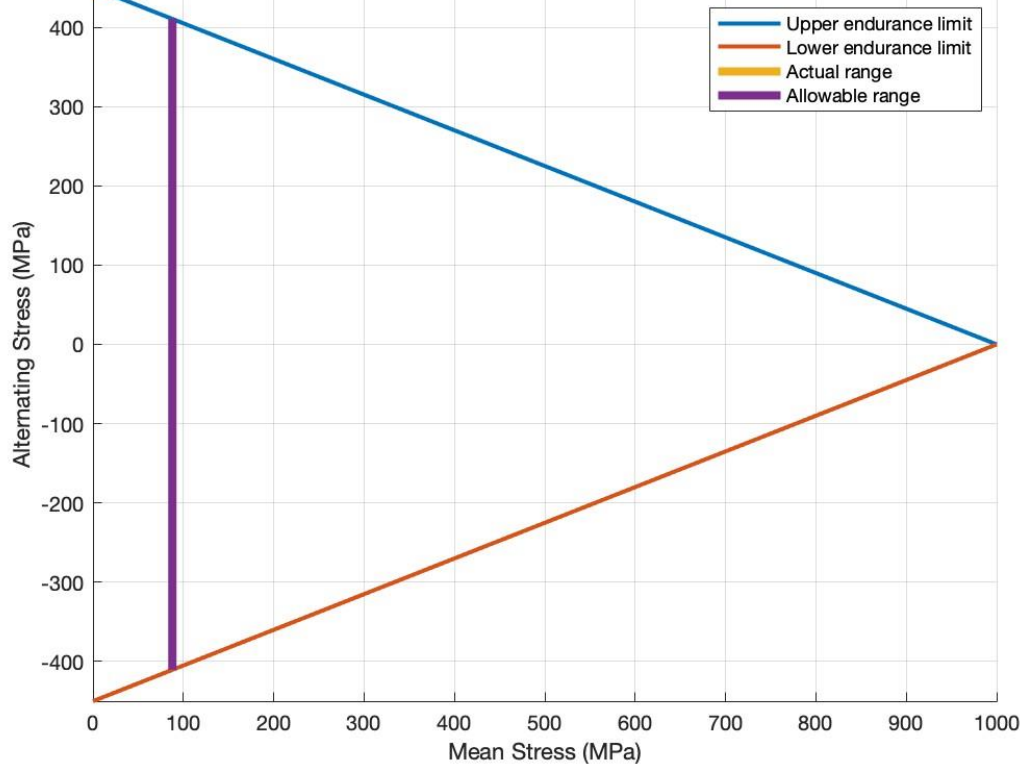


Figure 6 shows the Goodman fatigue diagram for the connecting rod midpoint at 6000 rpm. The mean stress is approximately 87.8 MPa, with a stress range of around 236 MPa.

The actual stress range is represented by the vertical line, while the allowable range from the Goodman criterion is significantly higher. The operating point lies well within the safe region, indicating low cyclic stresses and safe operation within material fatigue limits under these conditions.

Figure 6 –
Goodman
Diagram:
Conrod
Midpoint

The gudgeon pin fatigue behaviour was evaluated using the Goodman criterion, based on bending stress at point A. Mean and alternating stresses from the cycle were compared with material endurance limits to assess fatigue safety.

Goodman – Gudgeon pin at A | 6000rpm | Mean=78.6 MPa | Actual Range=211.8 MPa | Allowable Range=829

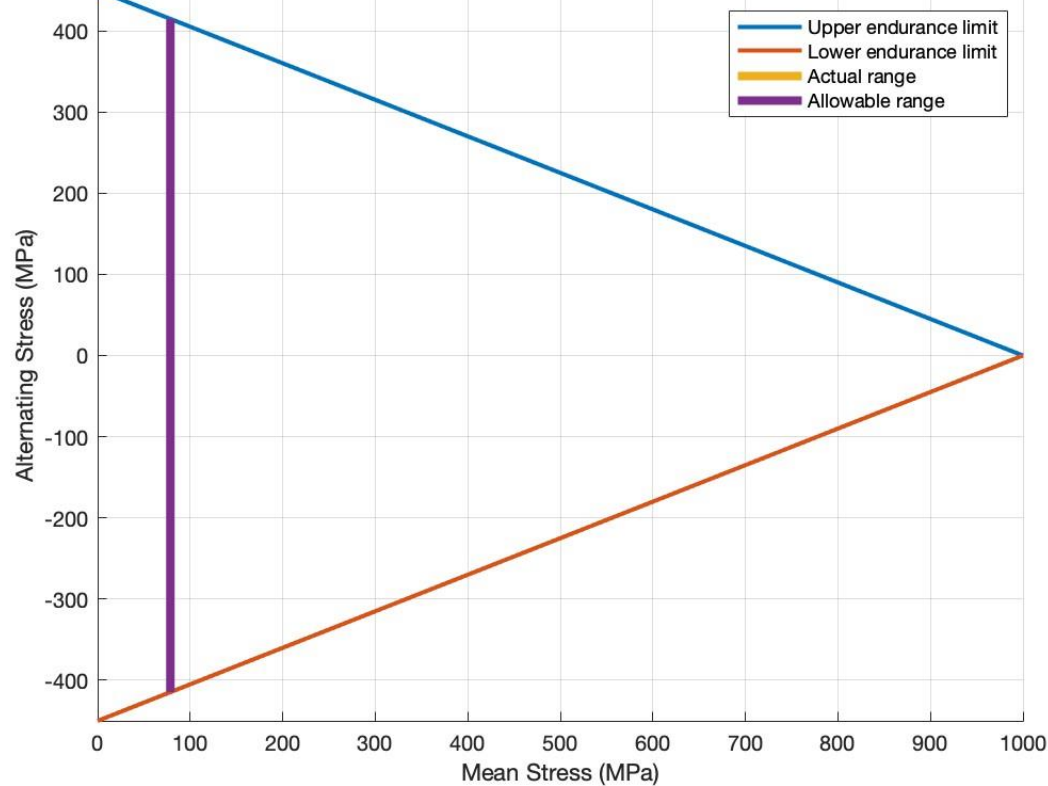


Figure 7 shows the Goodman fatigue diagram for the gudgeon pin at 6000 rpm. The mean stress is approximately 78.6 MPa, with an alternating stress of around 211.8 MPa.

The vertical line represents the actual stress range, while the allowable range from the Goodman criterion is significantly higher at approximately 829 MPa. This indicates the gudgeon pin operates well within the safe fatigue region, with a substantial margin to failure.

Figure 7 –
Goodman
Diagram:
Gudgeon
Pin

The fatigue performance of the connecting rod bolts was evaluated using the Goodman criterion, based on tensile stress from axial rod loading. Mean and alternating stresses at 6000 rpm were used to determine the fatigue safety factor.

Goodman – Conrod bolt | 6000rpm | Mean=238.3 MPa | Actual Range=476.7 MPa | Allowable Range=775.5

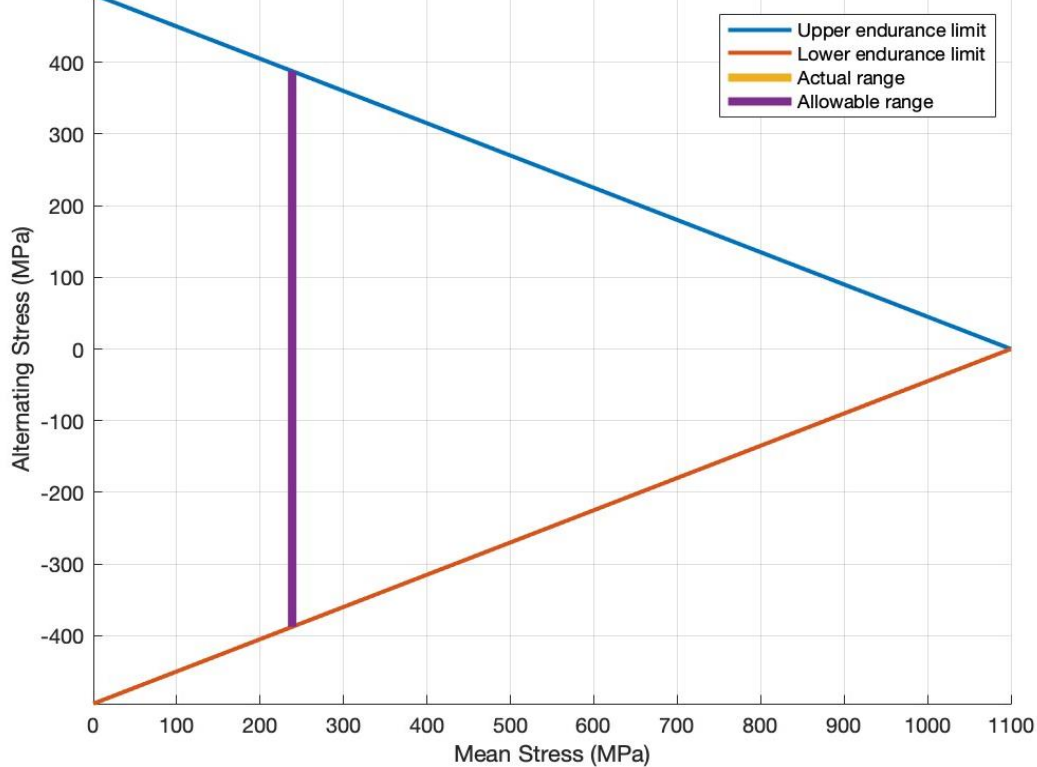


Figure 8 –
Goodman
Diagram:
Conrod Bolt

Figure 8 shows the Goodman fatigue diagram for the connecting rod bolts at 6000 rpm. The mean stress is approximately 238.3 MPa, with an alternating stress of around 476.7 MPa, higher than other components.

The actual stress range remains below the allowable limit of approximately 775.5 MPa, but with a reduced safety margin.

This indicates the bolts experience the most severe fatigue loading, driven by tensile and inertial effects, making them the most critical component for fatigue performance.

Fatigue loading of connecting rod bolts limits high-speed engine operation. Bolts experience high tensile stresses from gas and inertia forces, with failure often initiating at thread roots due to stress concentration (Taylor, 2019; Budynas and Nisbett, 2020).

Bolt fatigue safety factor typically limits maximum RPM, consistent with this analysis.

4.7 Extended RPM Analysis

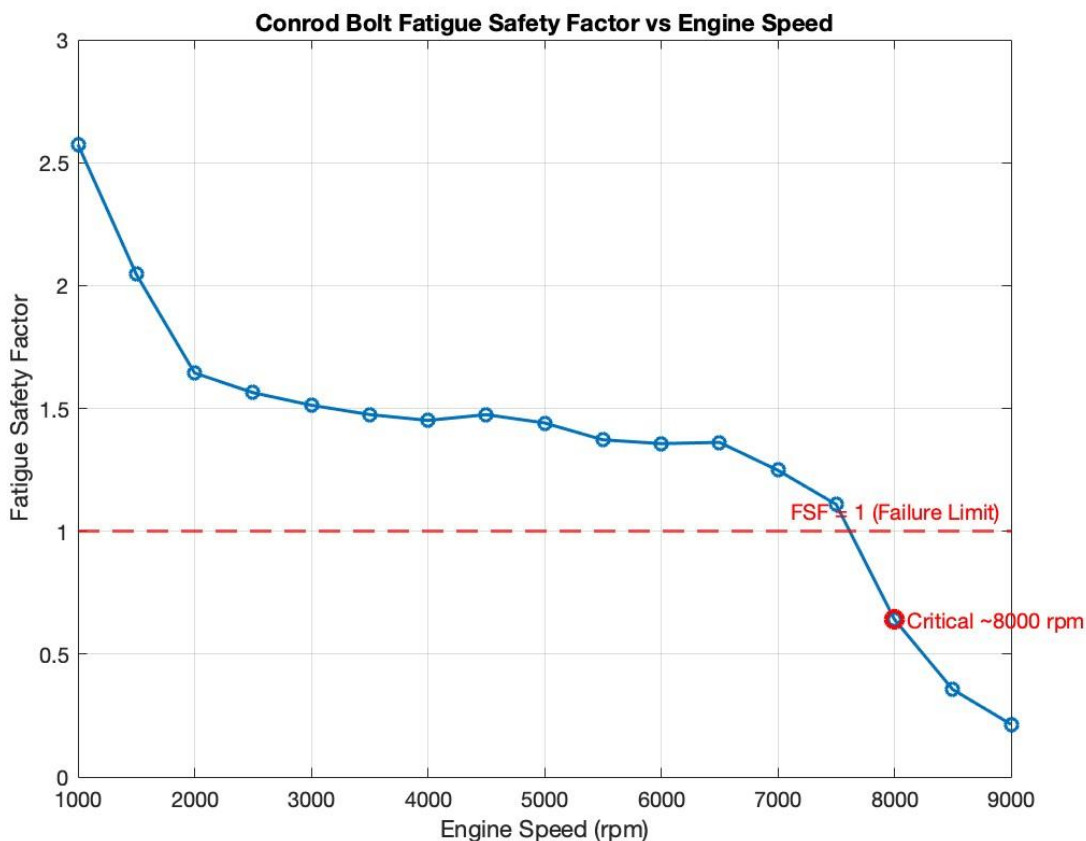


Figure 9: Conrod bolt fatigue safety factor as a function of engine speed (1000–9000 rpm), including the failure threshold (FSF = 1) and identified critical engine speed.

Figure 9 shows fatigue safety factor (FSF) variation for the connecting rod bolts from 1000 to 9000 rpm. FSF decreases from approximately 2.6 at 1000 rpm to around 0.21 at around 8000 rpm, indicating reduced fatigue resistance and failure with increasing speed.

This trend is driven by inertia forces, which scale with engine speed squared and significantly increase alternating stress (Heywood, 2018). While safe at lower speeds, FSF approaches unity as speed rises.

At approximately 8000 rpm, FSF falls below 1, indicating likely fatigue failure. Literature highlights bolt preload reduced reciprocating mass, and high-strength materials as key mitigation strategies.

These findings confirm that connecting rod bolts limit maximum engine speed, explaining typical rev limits of ~6000–6500 rpm in production engines such as the Ford Sigma.

The pressure data was extended from 6000 rpm to 9000 rpm using interpolation at each crank-angle position within MATLAB.

4.8 Geometry redesign (Con-Rod Bolt)

--- FINAL REDESIGN ---
Original Area = 42.43 mm²
New Area = 268.49 mm²
Original Diameter = 7.35 mm
New Diameter = 18.49 mm
Original FSF @6000 = 1.36
Original FSF @9000 = 0.21
New FSF @9000 = 1.36

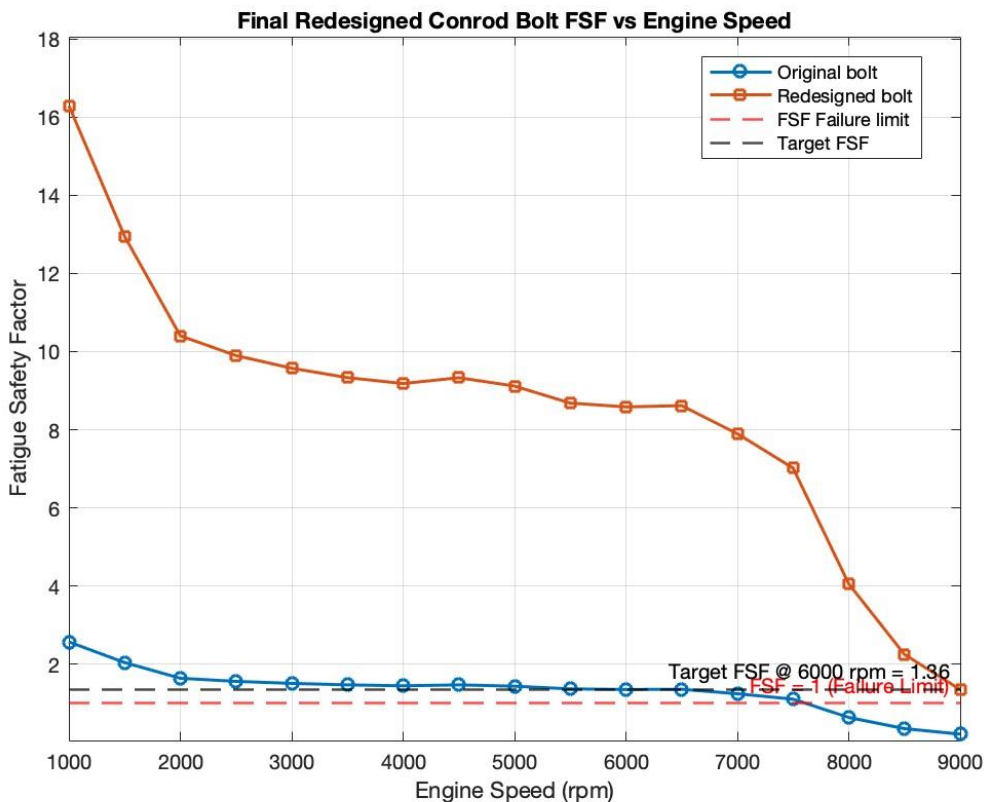


Figure 10: Comparison of fatigue safety factor (FSF) for the original and redesigned connecting rod bolt across the engine speed range.

The connecting rod bolt was redesigned to achieve the same fatigue safety factor at 9000 rpm as at 6000 rpm. The original bolt exhibited a fatigue safety factor of 1.36 at 6000 rpm, which reduced significantly to 0.21 at 9000 rpm, indicating likely failure under high-speed operation.

By increasing the bolt cross-sectional area from 42.43 mm² to approximately 268.49 mm², the fatigue safety factor at 9000 rpm was restored to 1.36, satisfying the requirement specified in the brief. This confirms that increasing load-bearing area is effective in reducing stress and improving fatigue resistance.

However, this required increasing the bolt diameter from 7.35 mm to 18.49 mm, representing a substantial and likely impractical design change due to increased mass, packaging constraints, and potential impacts on engine dynamics and balance.

This highlights that geometry alone is not an efficient solution, and that combining moderate geometric changes with higher-strength materials would provide a more realistic design approach.

Material selection strongly affects bolt fatigue. High-strength alloys (e.g. ARP2000, Ti-6Al-4V) increase allowable stress (ARP, 2021), enabling higher fatigue safety at high RPM without large geometric changes.

4.9 Torque Analysis

Crankshaft torque was calculated by resolving net piston force into its tangential component at the crank radius. Analysis was performed at 6000 rpm, the highest measured speed, ensuring results are based on experimental data rather than extrapolation.

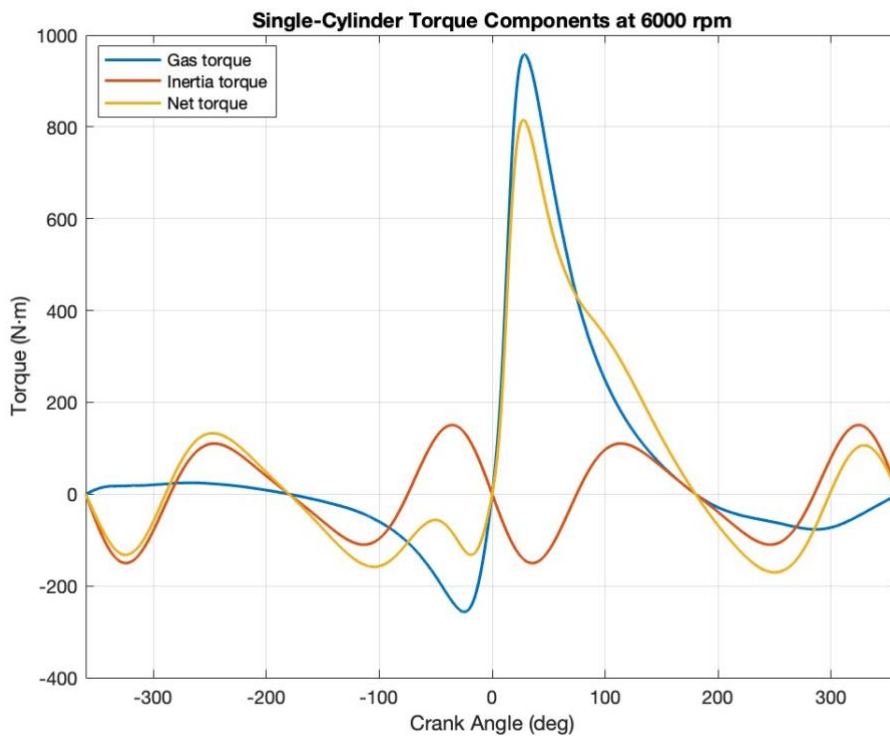


Figure 11: Gas, inertia and net crankshaft torque for a single cylinder at 6000 rpm.

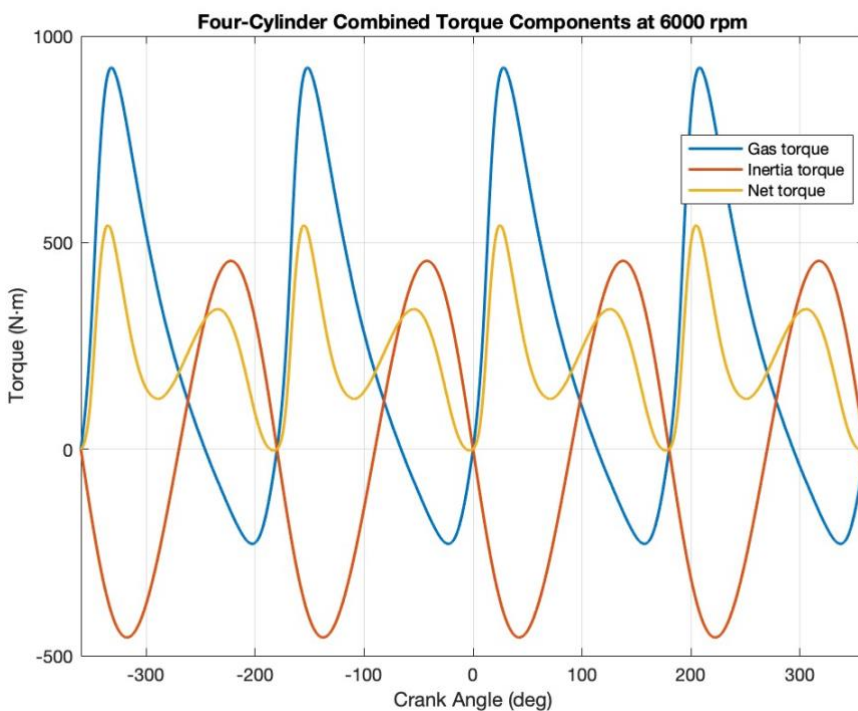


Figure 12: Gas, inertia and net crankshaft torque for 4 cylinders at 6000 rpm.

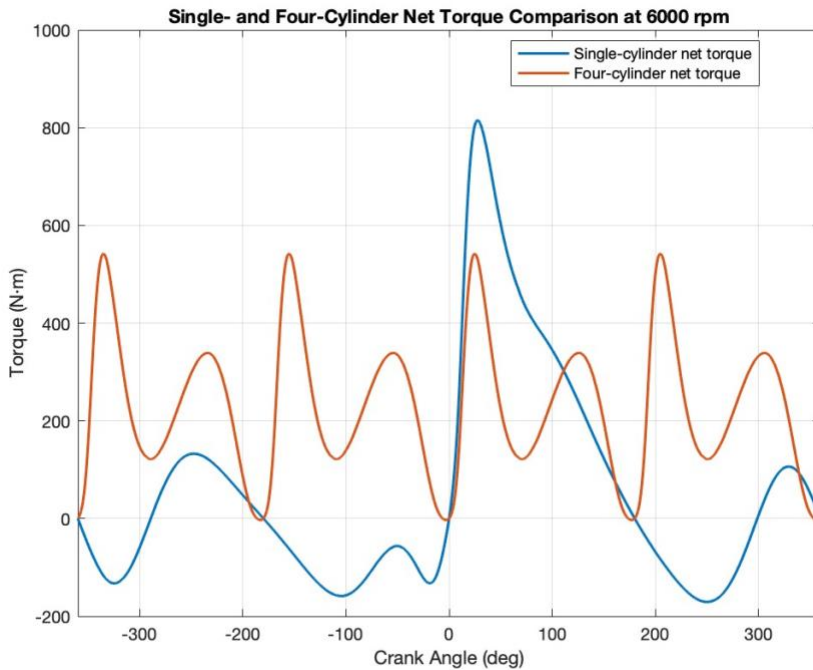


Figure 13: Combined gas, inertia and net crankshaft torque for a four-cylinder engine at 6000 rpm.

The torque results at 6000 rpm show that a single cylinder produces highly fluctuating output, with a dominant peak during the combustion phase and significant negative regions caused by inertia effects. Crankshaft torque was obtained by resolving the net piston force into its tangential component at the crank radius.

The inertia torque varies cyclically and becomes increasingly influential at higher speeds, contributing to the overall variation in net torque. When extended to a four-cylinder configuration with 180° phase spacing, the combined torque output becomes significantly smoother, as individual cylinder contributions overlap.

This reduces the amplitude of torque fluctuations and maintains predominantly positive torque throughout the cycle, demonstrating the effectiveness of multi-cylinder engine configurations in improving torque delivery and reducing vibration. The pressure data was realigned over the full 720° cycle and phase-shifted by 180° intervals.

The torque results are based on an inline four-cylinder crankshaft with 180° firing intervals, producing relatively smooth torque output. In contrast, a cross-plane crankshaft introduces different cylinder phase relationships, improving balance but creating more complex torque fluctuations. Therefore, the inline configuration provides a simplified and representative model for analysing engine torque.

5 Force and Stress results

Calculated forces increased with engine speed due to inertial loading, proportional to the square of angular velocity, resulting in higher stresses across all components.

At 6000 rpm, the conrod midpoint experienced a maximum stress of 265.96 MPa and minimum of -8.49 MPa (mean 128.73 MPa, alternating 137.23 MPa). The gudgeon pin reached 184.56 MPa and -26.82 MPa (mean 78.87 MPa, alternating 105.69 MPa).

The conrod bolts experienced the highest stresses, with a maximum of 476.69 MPa and minimum of -70.10 MPa (mean 203.29 MPa, alternating 273.40 MPa).

Overall, stress amplitude increased with engine speed, with the most severe loading at maximum speed.

6 Conclusion

This study evaluated the forces, stresses, and fatigue behaviour of key engine components using analytical methods and measured pressure data. The results showed that gas forces dominate near top dead centre, while inertia forces increase significantly with engine speed, leading to higher cyclic loading.

Stress analysis indicated that the connecting rod midpoint and gudgeon pin operate within safe limits, exhibiting relatively low mean and alternating stresses. In contrast, the connecting rod bolts experienced the highest stress levels and were identified as the most critical component.

Fatigue assessment using the Goodman criterion confirmed that the bolt safety factor decreases with increasing engine speed, falling below unity at approximately 8000 rpm. This indicates a high risk of fatigue failure at elevated speeds. This is consistent with typical engine design, where connecting rod bolts are often the limiting component at high engine speeds (Budynas and Nisbett, 2020).

Overall, the findings highlight that conrod bolt design is the primary limiting factor for engine durability and maximum operating speed.

7 References

- University of Wales Trinity St David, 2026. *ACHA6006 Engine Data and Specification Sheets*. Provided coursework material.
- University of Wales Trinity St David, 2026. *ACHA6006 Engine Data.xlsx*. Provided coursework material.
- X-Engineer, n.d. *Kinematic analysis of the ICE piston*. Available at: <https://x-engineer.org/kinematic-analysis-piston/> [Accessed 6th March 2026].
- ScienceDirect, n.d. *Goodman diagram - an overview*. Available at: <https://www.sciencedirect.com/topics/engineering/goodman-diagram> [Accessed 17 March 2026].
- Budynas, R.G. and Nisbett, J.K., 2020. *Shigley's Mechanical Engineering Design*. 11th ed. New York: McGraw-Hill.
- **Taylor, C., 2019.** *Fatigue behaviour of high-strength fasteners in reciprocating engines*. SAE Technical Paper 2019-01-0123.
- **Higgins, J. and Pidaparti, R., 2020.** *Structural optimisation and fatigue assessment of forged steel connecting rods*. International Journal of Fatigue, 135, p.105558.
- **Rahnejat, H., 2018.** *Tribology and dynamics of engine components*. Woodhead Publishing.
- **ARP, 2021.** *Fastener material properties and performance guidelines*. ARP Technical Bulletin. Automotive Racing Products, California.
- **Heywood, J.B., 2018.** *Internal Combustion Engine Fundamentals*. 2nd ed. New York: McGraw-Hill.