

Analytical assessment of engine internal components

Contemporary Powertrain Design (ACHA6006)

Jay Wexler

2305581

Lecturer: Andrew Noble

University of Wales Trinity Saint David

School of Engineering

Motorcycle Engineering BEng

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Abstract

This study evaluates the forces, stresses, and fatigue behaviour of key bottom-end engine components using kinematic modelling and pressure data. Fatigue performance was assessed using the Goodman criterion, identifying the connecting rod bolt as the fatigue-critical component due to high mean stress from preload. Extending operation to 9000 RPM showed a significant reduction in fatigue safety factor, driven by increasing inertia forces. A redesigned bolt incorporating reduced stiffness and improved thread geometry improved fatigue performance while maintaining acceptable behaviour at standard operating speeds. Torque analysis highlighted differences between standard and cross-plane crankshaft configurations, demonstrating their suitability for different performance applications.

1 Introduction

Internal combustion engines operate under cyclic loading conditions where components are subjected to combined gas and inertia forces throughout the engine cycle, leading to potential fatigue failure in critical components (Heywood 1988; Stone and Macmillan 2012). Components such as the connecting rod, gudgeon pin, and connecting rod bolt are particularly susceptible due to their role in transmitting dynamic loads within the reciprocating system.

To understand the behaviour of these components, kinematic analysis, force resolution, and fatigue assessment are combined. The Goodman fatigue criterion is used here to evaluate the combined effects of mean and alternating stresses, while stress concentration effects at critical locations, such as bolt threads, must also be considered (Budynas et al. 2015).

The aim of this study is to evaluate the loading and fatigue behaviour of key engine components and assess their suitability under increased engine speeds. This is achieved by developing a kinematic and dynamic model to determine forces and stresses, calculating fatigue safety factors for each component, and identifying the critical component governing fatigue life. The analysis is extended to 9000 RPM using interpolated pressure data to assess the effect of increased operating speed, followed by a redesign of the connecting rod bolt to improve fatigue performance. Finally, the influence of crankshaft configuration on instantaneous torque characteristics is evaluated.

2 Methodology

2.1 Engine Data & Assumptions

The analysis uses supplied engine geometry and in-cylinder pressure data to determine piston motion, forces, and resulting stresses over a full cycle. Pressure data was converted to SI units for use within MATLAB.

Slider-crank kinematics, defined by crank radius, connecting rod length, and bore, govern piston displacement, velocity, and acceleration, and hence inertia forces. The system is modelled as a rigid-body mechanism with linear elastic behaviour, neglecting thermal effects. Friction is represented by a constant coefficient, reciprocating mass is lumped along the piston axis, and the connecting rod is assumed to carry axial load only. These assumptions capture the dominant dynamics while keeping the model computationally efficient (Heywood 1988; Stone and Macmillan 2012).

2.2 Kinematic Analysis

Figure 1 shows piston displacement and velocity over a full engine cycle. Piston motion was modelled using slider-crank relations to determine displacement, velocity, and acceleration as functions of crank angle. Piston acceleration was defined as:

$$a = R\omega^2 \cos \theta + \frac{R^2}{L} \omega^2 \cos 2\theta$$

This relation shows that acceleration, and therefore inertia force, varies non-linearly with crank angle and increases with angular velocity squared. These kinematic outputs define piston motion and provide the basis for inertia forces, which peak near TDC as piston acceleration reverses direction (Heywood 1988; Stone and Macmillan 2012).

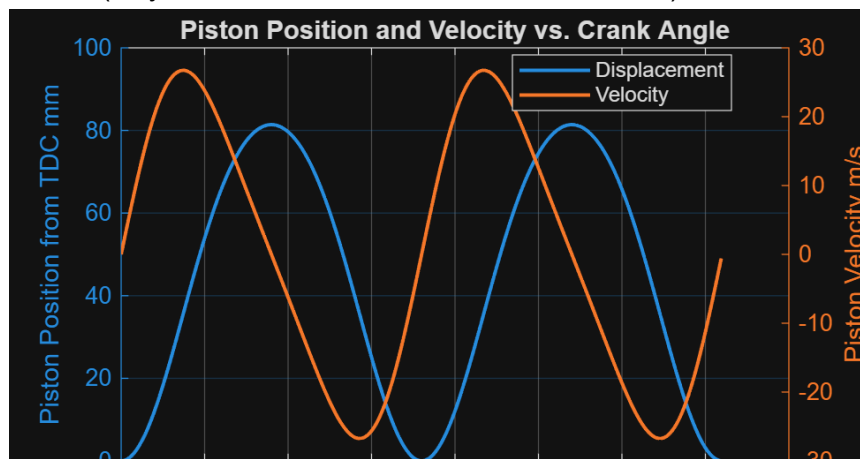


Figure 1: Variation of piston displacement and velocity with crank angle over a full engine cycle

2.3 Force Analysis

Gas force was calculated from cylinder pressure and piston area using $F_{gas} = PA$, varying with crank angle and engine speed (Figure 2), with a peak during combustion shortly after TDC. Inertia force was obtained from reciprocating mass and piston acceleration, $F_{inertia} = m_{rec}a$, increasing with ω^2 . The net piston force was then determined from $F_{net} = F_{gas} - F_{inertia}$, representing the load transmitted through the connecting rod.

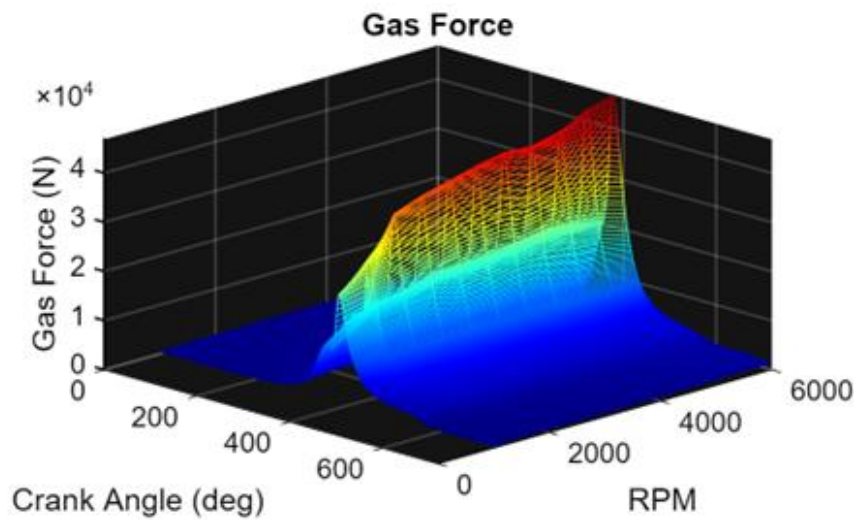


Figure 2: Gas force as a function of crank angle and engine speed.

The net piston force, defined as the difference between gas and inertia forces, represents the load transmitted through the connecting rod (Figure 3).

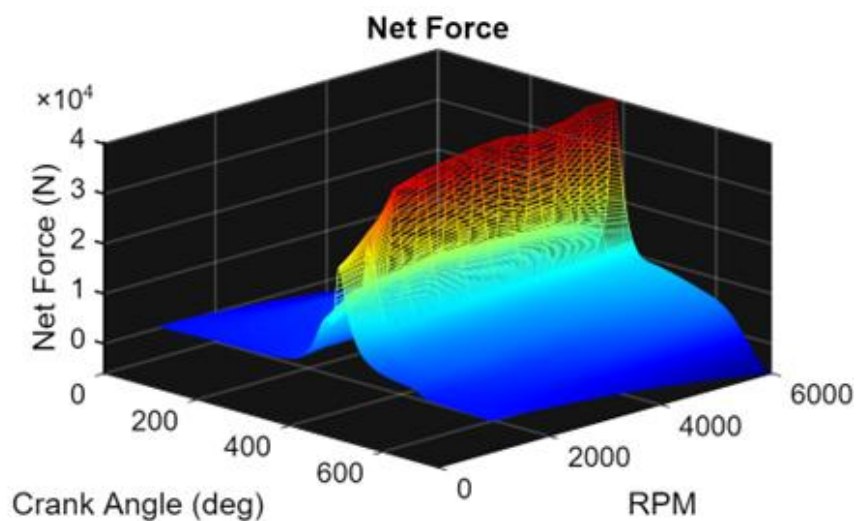


Figure 3: Net force acting on the piston as a function of crank angle and engine speed.

Figure 4 compares gas, inertia, and net forces at 6000 RPM, showing gas force dominance during combustion due to peak cylinder pressure following ignition, while inertia forces dominate outside this region due to piston acceleration. These force histories form the basis for stress and fatigue analysis.

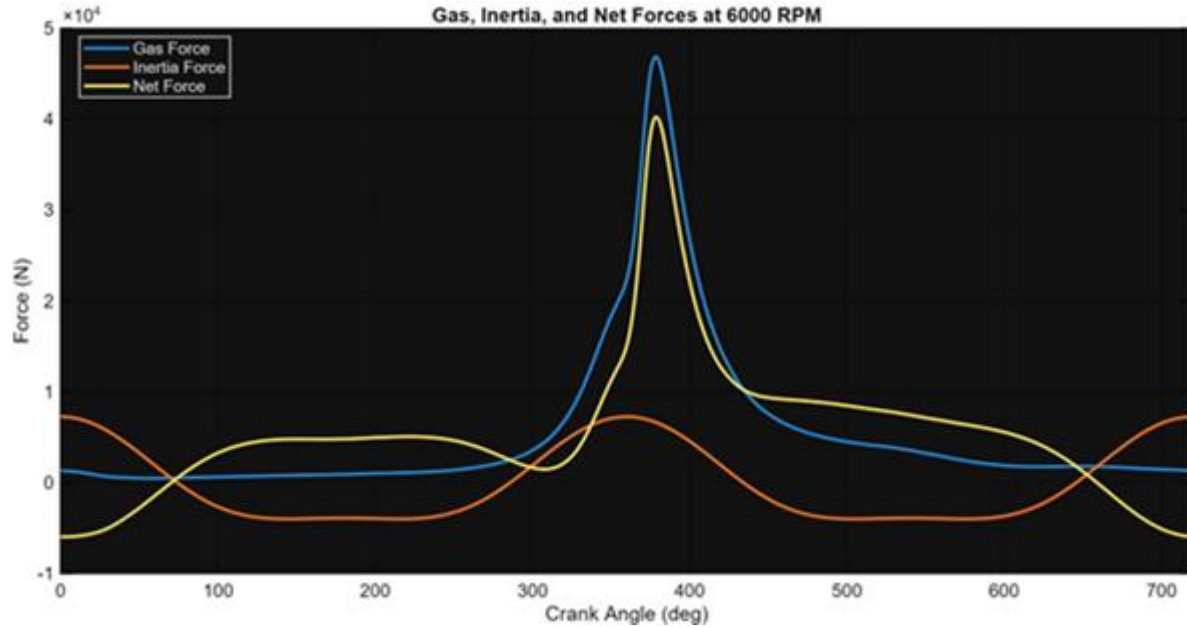


Figure 4: Gas, inertia, and net forces over a full engine cycle at 6000 rpm.

2.4 Stress Analysis

The calculated forces were used to determine stresses in the gudgeon pin and connecting rod bolt. Axial and bending stresses were evaluated from $\sigma = F/A$ and $\sigma_b = My/I$.

The gudgeon pin was modelled as a beam under transverse loading due to piston side forces, producing bending and shear stresses from the distributed load and pin geometry. The connecting rod bolt was modelled as a preloaded joint, where preload generates high mean stress and inertia introduces an alternating component. Load sharing between the bolt and clamped members was determined from relative stiffness, with stiffness defined as $K = EA/L$.

Stress histories were obtained over the cycle and across all engine speeds, from which mean and alternating stresses were calculated for fatigue assessment.

2.5 Stress Concentration Factors (K_t)

Geometric discontinuities such as fillets and threads introduce local stress concentrations, increasing peak stress beyond nominal values (Pilkey and Peterson 1997). This effect can be expressed as $\sigma_{max} = K_t \sigma$, and was applied at critical locations, particularly the gudgeon pin supports and bolt threads (Budynas et al. 2015).

Rather than modifying the applied stress directly, the endurance limit was reduced to reflect increased sensitivity to cyclic loading, providing a more realistic fatigue assessment.

2.6 Fatigue Safety Factor (Goodman & FSF)

Fatigue performance was assessed using the Goodman criterion, accounting for combined mean and alternating stress effects. Stress histories were used to determine maximum and minimum stresses, from which mean stress and stress range were calculated using $\sigma_{mean} = (\sigma_{max} + \sigma_{min})/2$ and $\sigma_{range} = \sigma_{max} - \sigma_{min}$.

The allowable alternating stress was defined using material properties, including ultimate tensile strength and endurance limit (Budynas et al. 2015). The fatigue safety factor (FSF) was obtained from $FSF = \sigma_{allowable}/\sigma_{range}$, with results presented using Goodman diagrams to identify components approaching fatigue limits (Dowling et al. 2020).

3 Results & Analysis — Component Behaviour

Fatigue behaviour was assessed using the Goodman criterion, with results expressed as mean stress, stress range, and fatigue safety factor (FSF), enabling direct comparison of component performance.

3.1 Connecting Rod

The connecting rod midpoint exhibits a mean stress of 58.11 MPa, stress range of 113.87 MPa, and FSF of 4.095 (Figure 5). The operating point lies well within the Goodman boundary, with the allowable range (847.70 MPa) far exceeding the applied range, indicating a large fatigue margin.

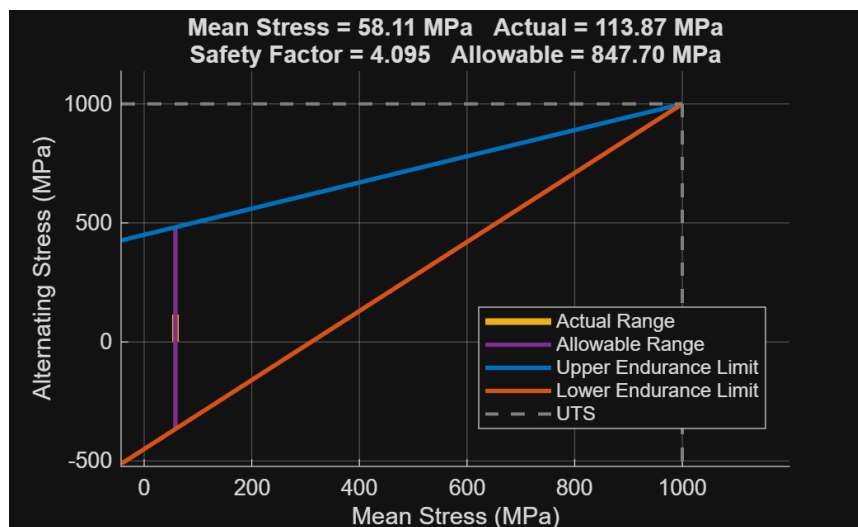


Figure 5: Goodman diagram for connecting rod midpoint.

This is primarily due to the relatively large cross-sectional area and predominantly axial loading, which produces a uniform stress distribution, limiting both mean and alternating stresses and reducing fatigue sensitivity. Fatigue behaviour is governed by a low stress range, with minimal sensitivity to mean stress effects. The high FSF indicates conservative design at this location, suggesting low fatigue risk but potential for weight optimisation without compromising durability.

3.2 Gudgeon Pin

The gudgeon pin shows increased fatigue loading, with mean stress of 85.06 MPa, stress range of 222.94 MPa, and FSF of 3.694 (Figure 6). Although still within the Goodman limit, the reduced margin reflects a more complex loading state.

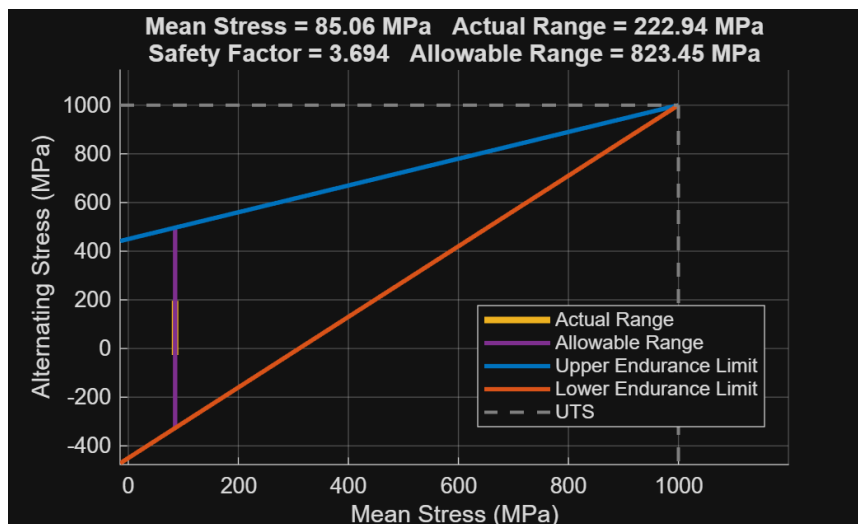


Figure 6: Goodman diagram for gudgeon pin

Unlike the connecting rod, the pin is subjected to combined bending and shear due to load transfer between piston bosses and rod angularity, producing localised peak stresses and increased fatigue sensitivity. This increases both mean stress and stress range. Despite this, the allowable range (823.45 MPa) remains significantly higher than the applied range, confirming a substantial fatigue reserve. However, the higher stress range indicates greater sensitivity to dynamic loading, particularly during peak force transmission. The pin remains conservatively designed but represents a more mechanically demanding component than the rod midpoint.

3.3 Connecting Rod Bolt

3.3.1 Nominal Behaviour

The connecting rod bolt exhibits a mean stress of 761.42 MPa, stress range of 202.84 MPa, and FSF of 1.502 (Figure 7), placing it significantly closer to the Goodman failure boundary and identifying it as the fatigue-critical component.

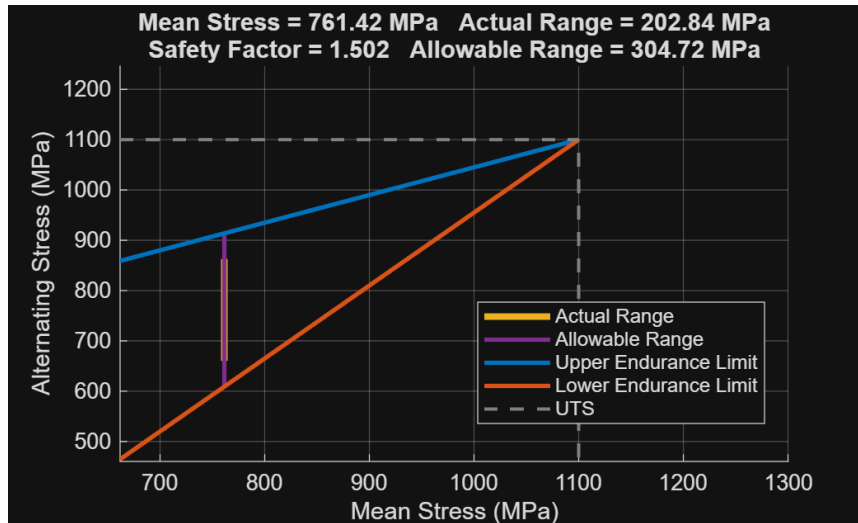


Figure 7: Goodman diagram for connecting rod bolt

Bolt stress variation (Figure 8) is dominated by high tensile preload with superimposed cyclic loading. A peak near TDC corresponds to maximum combustion and inertia-induced separation forces, as piston acceleration reverses direction and reaches a maximum. The high mean stress arises from preload required to maintain joint integrity, while the alternating component is primarily driven by inertia forces, which increase with ω^2 . As a result, fatigue performance is governed by mean stress rather than stress amplitude. The allowable stress range (304.72 MPa) is only moderately higher than the applied range, resulting in a limited fatigue margin compared to other components.

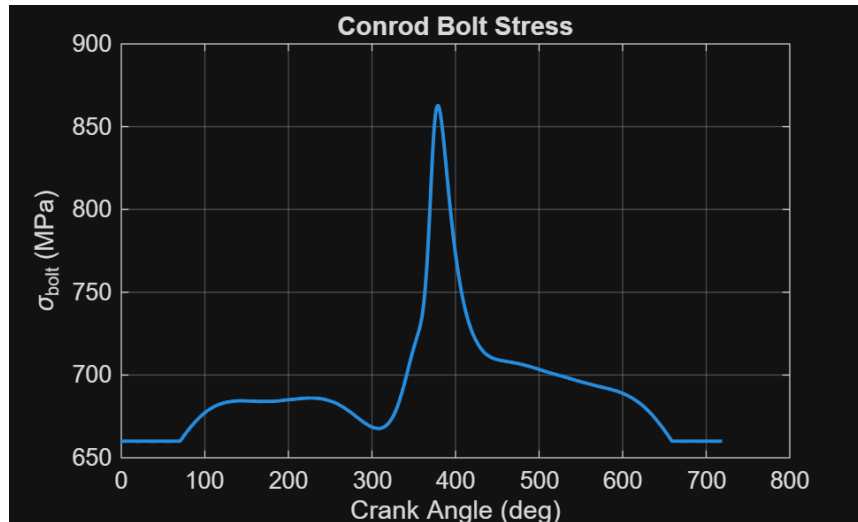


Figure 8: Connecting rod bolt vs. crank angle at 6000 rpm

3.3.2 Thread-Root Stress Concentration Effect

Thread geometry introduces significant stress concentration, making the root the primary site for fatigue crack initiation. This effect was accounted for by reducing the endurance limit, reflecting increased sensitivity to cyclic loading. The resulting reduction in FSF further confirms the bolt as the fatigue-critical component. Given the already high mean stress from preload, local stress amplification significantly increases failure risk. This highlights the importance of design improvements such as optimised thread geometry or reduced shank designs to improve fatigue resistance.

4 Extended RPM Analysis

To evaluate the effect of increased engine speed on fatigue behaviour, the operating range was extended to 9000 RPM. As pressure data was only available up to 6000 RPM, interpolation was used to estimate cylinder pressure at higher speeds.

The interpolated data was then used to recalculate dynamic forces, stresses, and fatigue safety factors for the connecting rod bolt. This enables assessment of how increasing engine speed influences fatigue performance and defines the upper operational limits of the system.

4.1 Interpolation Method

To extend the operating range to 9000 RPM, cylinder pressure data was interpolated across engine speed. For each crank angle, pressure variation with RPM was treated as a continuous

function, allowing values beyond the original 6000 RPM dataset to be estimated while preserving crank-angle resolution.

Interpolation is appropriate as cylinder pressure varies smoothly with engine speed over the available range (Heywood 1988). This enables evaluation of increased engine speed effects without requiring additional experimental data, particularly as inertia-driven loading dominates at higher speeds.

However, extrapolation beyond 6000 RPM introduces uncertainty, as combustion behaviour may change due to reduced combustion duration, altered air–fuel mixing, and increased heat transfer (Stone and Macmillan 2012). The results should therefore be interpreted as an approximation but remain sufficient for identifying trends in fatigue behaviour.

4.2 Results

The recalculated fatigue safety factor for the connecting rod bolt shows a clear reduction with increasing engine speed, as illustrated in Figure 9. This trend is driven by the increase in inertia forces, scaling with the square of angular velocity, increasing cyclic tensile loading and accelerating fatigue damage.

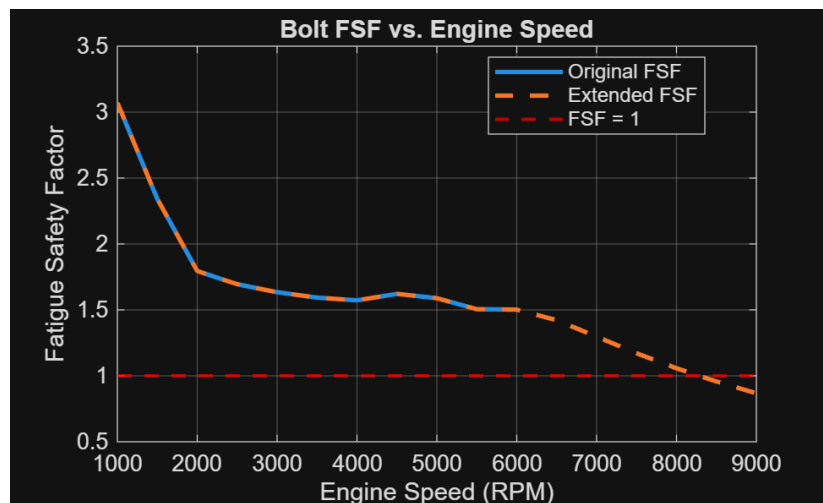


Figure 9: Fatigue safety factor of connecting rod bolt vs engine speed

As RPM increases, the alternating stress component becomes more significant, reducing the allowable stress range under the Goodman criterion. This leads to a decrease in fatigue safety factor, with the value reducing from approximately 1.502 at 6000 RPM to 0.867 at 9000 RPM, representing a reduction of 37.9%, highlighting the sensitivity of bolt fatigue performance to increasing engine speed.

The Goodman diagram at 9000 RPM (Figure 10) shows the operating point approaching the failure boundary, confirming that the reduced safety factor is driven by high mean stress combined with increased alternating stress.

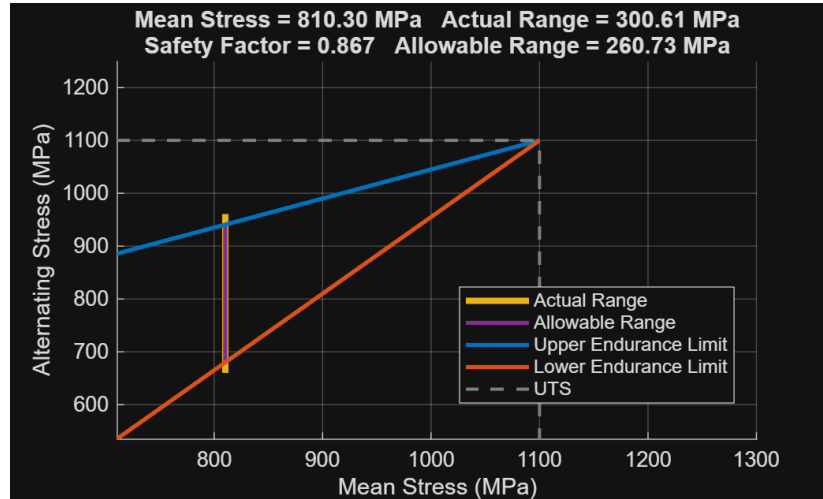


Figure 10: Goodman diagram for connecting rod bolt at 9000 RPM

At the upper end of the extended range, the fatigue safety factor approaches critical levels, suggesting that the bolt may be at risk of fatigue failure under sustained high-speed operation.

4.3 Critical Discussion

The reduction in fatigue safety factor with increasing engine speed highlights the growing influence of inertia forces within the reciprocating system. As these forces scale with the square of angular velocity, they become the dominant contributor to cyclic loading, particularly for the connecting rod bolt.

The results show that while the bolt operates within acceptable limits at lower engine speeds, its fatigue performance deteriorates significantly with increasing RPM. This confirms that the bolt is the limiting component and defines the upper operational boundary of the engine.

From a design perspective, this behaviour emphasises the importance of controlling mean stress and reducing cyclic loading in bolted joints. Without modification, operation at elevated engine speeds would reduce fatigue life and increase the risk of failure.

5 Bolt Redesign

The extended RPM analysis identified the connecting rod bolt as the fatigue-critical component, with a significant reduction in fatigue safety factor at higher engine speeds. The redesign therefore aims to increase fatigue resistance by reducing combined mean and alternating stresses while maintaining sufficient preload for joint integrity.

A reduced shank (wasted shank) design was introduced to decrease bolt stiffness, ensuring a smaller proportion of the external load is carried by the bolt. This reduces the alternating stress experienced during operation, which is particularly important at high engine speeds where inertia forces increase cyclic loading. Increasing the effective working length further reduces stiffness, with $K \propto 1/L$, improving load distribution and reducing stress amplitude. In addition, optimisation of thread geometry reduces stress concentration at the thread root, increasing the effective endurance limit and improving fatigue resistance.

These modifications shift the operating point on the Goodman diagram away from the failure boundary by reducing alternating stress and mitigating mean stress effects, directly addressing the limitations identified in the extended RPM analysis.

5.1 Redesign Outcomes

The fatigue safety factor variation with engine speed (Figure 11) shows a consistent improvement for the redesigned bolt across the full RPM range. While the original design approaches failure at high speeds (FSF $\approx$ 1 at $\sim$ 8300 RPM), the redesigned bolt maintains FSF > 1 surpassing 9000 RPM, confirming improved fatigue performance.

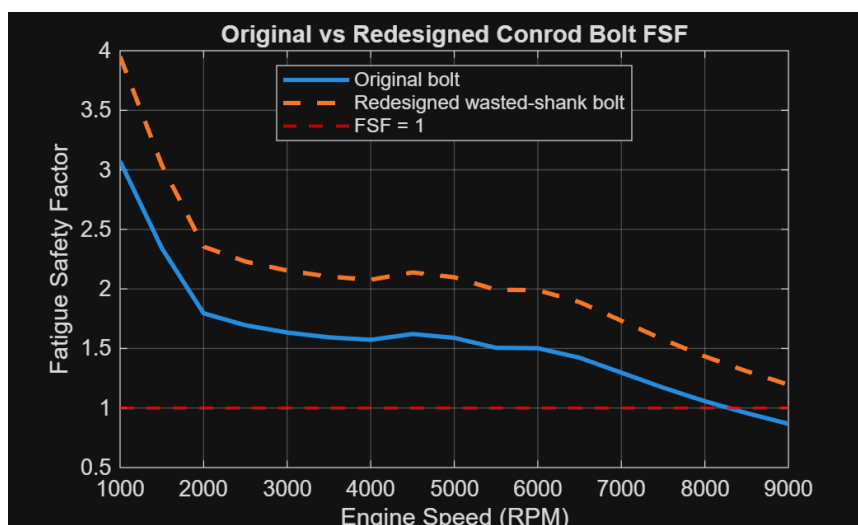


Figure 11: Fatigue Safety Factor comparison across complete RPM range

The redesigned bolt shows a clear improvement in fatigue performance. At 9000 RPM, the fatigue safety factor increases from approximately 0.867 to 1.196, indicating a significant recovery in fatigue margin. At 6000 RPM, the fatigue safety factor increases to 1.989 compared to the original value of 1.502, demonstrating that the redesign improves performance without compromising standard operating conditions.

This improvement is primarily due to reduced alternating stress resulting from decreased bolt stiffness and improved load distribution, combined with reduced stress concentration at the thread root. This is reflected in the Goodman diagrams (Figure 12), where the operating points at both 6000 RPM and 9000 RPM shift away from the failure boundary. The shift is more pronounced at 9000 RPM, indicating that the redesign is most effective under inertia-dominated loading, where increased cyclic stresses would otherwise govern fatigue failure.

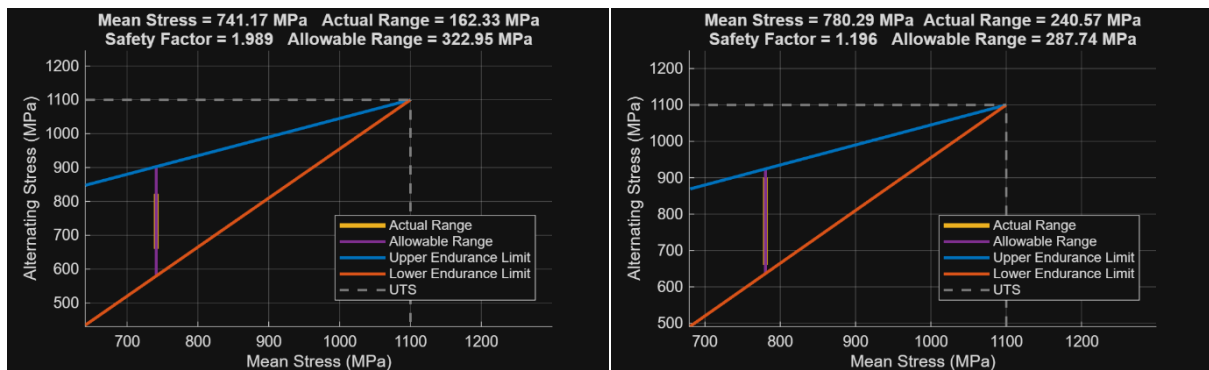


Figure 12: Redesigned bolt Goodman diagrams at 6000 & 9000 RPM

6 Torque Analysis

The instantaneous engine torque was calculated from the gas and inertia forces acting on the crank mechanism by resolving these forces into tangential components. This provided the variation in crankshaft torque over a full engine cycle.

The analysis was then extended to compare standard and cross-plane crankshaft configurations, enabling assessment of their torque characteristics and suitability for different applications.

6.1 Method

The instantaneous crankshaft torque was calculated by resolving gas and inertia forces into tangential components acting on the crank. Gas forces generate positive torque through crank

rotation, while inertia forces produce opposing components depending on piston acceleration. These contributions were combined to obtain net crank torque as a function of crank angle.

The individual contributions of gas and inertia torque, along with the resulting net torque for a single cylinder at 6000 RPM, are shown in Figure 13.

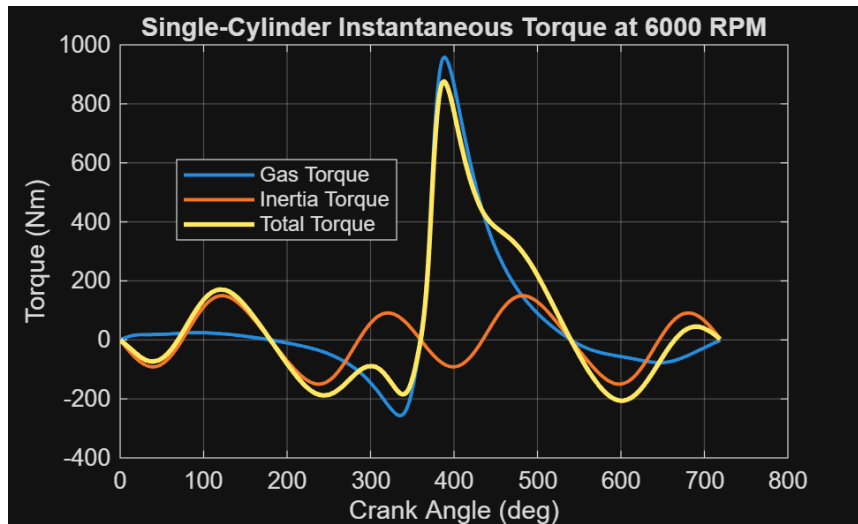


Figure 13: Single-cylinder gas, inertia, and total torque at 6000 RPM

The torque was then evaluated over a full engine cycle and averaged to determine overall engine output. This process was repeated for both crankshaft configurations by realigning the cylinder phasing and firing order.

6.2 Standard vs Cross-plane Results

The instantaneous torque results present clear differences between the standard and cross-plane crankshaft configurations. The standard configuration produces larger torque fluctuations over the engine cycle, with distinct peaks corresponding to combustion events.

In contrast, the cross-plane configuration produces a more evenly distributed torque output, reducing torsional vibration and improving drivability. While peak torque remains similar, the cross-plane arrangement results in a more consistent torque delivery. The average torque over the cycle is comparable for both configurations; however, the distribution of torque differs significantly.

6.3 Discussion

The differences in torque characteristics between the standard and cross-plane crankshaft configurations arise from the phase relationship between cylinder firing events. In the standard

configuration, torque contributions are more evenly spaced, resulting in more frequent but lower amplitude fluctuations.

In contrast, the cross-plane configuration alters the firing order, producing fewer but larger torque pulses. This results in greater peak amplitudes but increased separation between torque events, as shown in Figure 14.

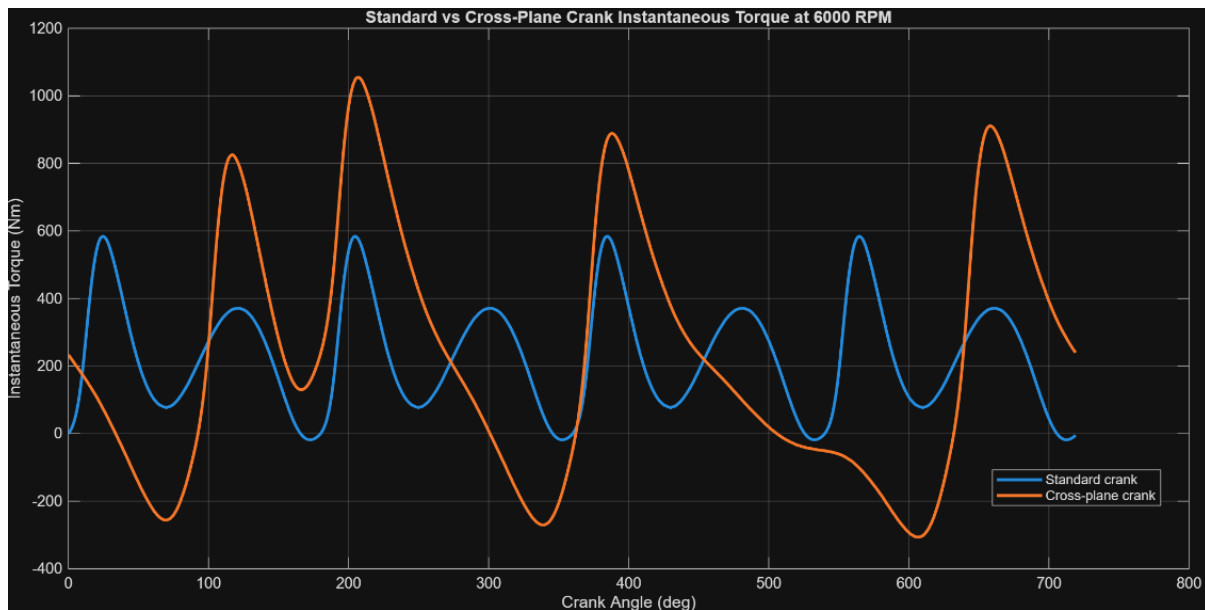


Figure 14: Instantaneous torque comparison between standard and cross-plane crank configurations at 6000 RPM

From an application perspective, the standard configuration provides more consistent torque delivery, making it well suited to high-speed operation where smoothness and reduced vibration are important. The cross-plane configuration, despite larger fluctuations, can improve traction by spacing torque delivery, which is advantageous in certain motorsport applications.

Additionally, the distribution of torque in the cross-plane configuration can reduce the frequency of stress cycles in drivetrain components, although the higher peak loads may increase localised stress. The increased complexity and potential imbalance of cross-plane crankshafts also introduce additional design and manufacturing challenges.

7 Conclusion

The forces, stresses, and fatigue behaviour of key bottom-end components were evaluated using kinematic modelling and pressure data. Fatigue performance is governed by the interaction of mean and alternating stress rather than load magnitude alone. The connecting rod and gudgeon pin exhibited high fatigue safety factors of 4.095 and 3.694, indicating substantial design margin. In contrast, the connecting rod bolt was identified as the fatigue-critical component, with a safety factor of 1.502 due to high mean stress from preload.

Extending operation to 9000 RPM further reduced bolt fatigue safety factor due to inertia forces increasing with ω^2 , indicating that sustained operation beyond this speed would result in unacceptable fatigue life. A redesigned bolt incorporating reduced stiffness and improved thread geometry increased fatigue performance by reducing alternating stress and stress concentration effects, although with reduced joint stiffness and increased manufacturing complexity.

Torque analysis showed that the standard configuration produces more frequent, lower-amplitude fluctuations, while the cross-plane configuration generates larger, separated torque pulses, making each more suitable for different applications.

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