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Analytical Assessment of Engine Internal Components
Contemporary Powertrain ACHA6006

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Abstract

This study evaluates the fatigue behaviour of crank components in a Ford 1.6 GDTi engine using cylinder pressure data and kinematic force modelling. Goodman diagram analysis shows that the connecting-rod and gudgeon pin maintain high fatigue margins at 6000RPM, while the con-rod bolt becomes critical when thread stress-concentration is applied. Extending the engine speed to 9000RPM highlights likely bolt failure, and redesign analysis indicates that upgrading the material offers an effective fatigue improvement.

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1 Introduction

The aim of this report is to calculate the fatigue safety factors of key components within a Ford 1.6 GDTi engine and extend the pressure data to 9000RPM in order to reassess component durability and propose improvements to the connecting-rod bolt design. Fatigue safety factors are essential in powertrain engineering, as they indicate whether a component can endure the combined gas and inertial loads experienced through it. These loads fluctuate significantly through the engine cycle, making accurate stress evaluation critical for predicting fatigue behaviour.

While several components contribute to the overall load path through the crank assembly, this report focuses primarily on the connecting-rod bolts, examining how their fatigue safety factor changes with increasing rotational speed and identifying design or material modifications that could enhance performance. The connecting-rod and gudgeon pin are also discussed to contextualize how forces are transferred through the assembly.

Modern engine development places increasing emphasis on durability and sustainability. Extending component lifespan reduces material consumption and manufacturing energy, meaning components designed with sufficiently high fatigue resistance therefore support both performance and environmental objectives in modern engineering.

2 Methodology

2.1 Data sources

Cylinder pressure data was taken directly from the assignment information and imported into MATLAB for processing. The kinematic parameters required for the force calculations, including crank radius, connecting-rod length and reciprocating mass, were obtained from the provided 'Engine Specification Sheet' (Appendix). These parameters formed the basis for calculating gas force, inertial force, axial force and subsequent stress values throughout the engine cycle.

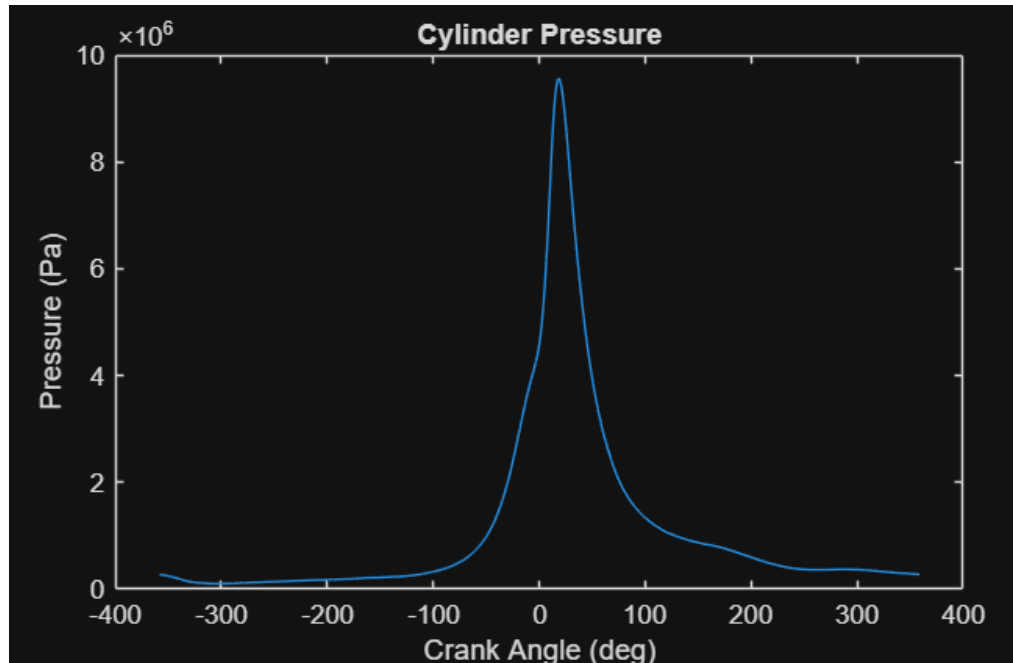


Figure 1 Cylinder Pressure

The cylinder pressure exhibits typical characteristics of a spark ignition engine cycle. Pressure increases rapidly following ignition near top dead centre (TDC=0°) during combustion phase, and reaching a peak of 134Bar shortly after TDC (at 18°ATDC). This peak pressure generates maximum compressive loading within the con-rod.

Another notable feature of the pressure trace is the significantly lower values during intake and exhaust stroke, due to cylinder pressure falling close to atmospheric. Under these conditions the combustion force acting on the piston is minimal and the net force within the con-rod is dominated by inertia forces from the reciprocating mass. This is particularly significant near the exhaust-intake transition around TDC, where piston acceleration is maximum.

2.2 Force Calculations and Trends

Piston displacement, velocity and acceleration were calculated as functions of crank angle using the engine parameters from the appendix. These variables were required to determine the forces generated throughout an engine cycle.

The inertia force was calculated by multiplying the reciprocating mass with acceleration. The gas force acting on the piston was determined by multiplying cylinder pressure by the cross-sectional-area of the piston crown. The total force on the connecting-rod was obtained by combining gas force and inertia force, accounting for the connecting-rod angle ϕ , which varies with crank angle.

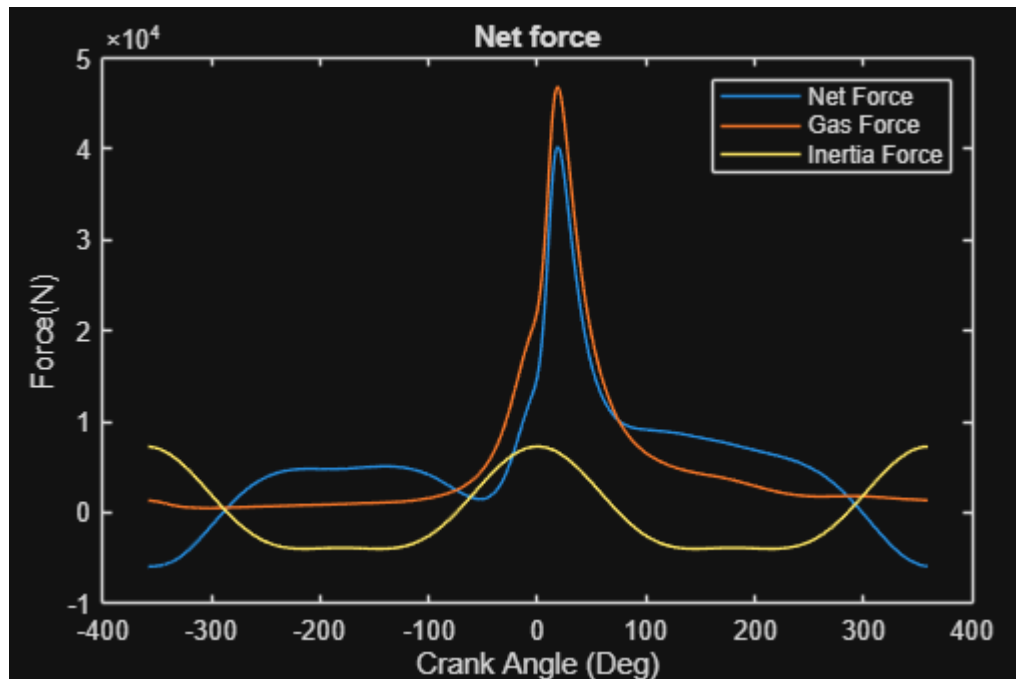


Figure 2 Net Force

The net force along the rod was calculated using inertia subtracted from gas force. The net force varies significantly throughout the engine cycle, following a similar trend to pressure. During the early expansion stroke and combustion, gas pressure produces large compressive forces on the con-rod. As the piston moves towards bottom dead centre the cylinder pressure decreases resulting in a reduction in compressive force.

When cylinder pressures approach atmospheric levels during intake and exhaust strokes, inertia forces become the dominant load. Peak tensile forces are generated when piston acceleration reaches a maximum value near the exhaust-intake transition.

Using the cross-sectional area of the connecting-rod midpoint (figure 3), the resulting axial stresses were calculated from the force acting along the rod axis.

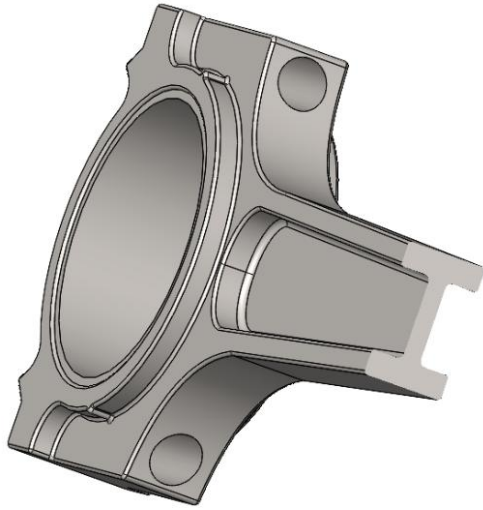


Figure 4 Connecting Rod midpoint

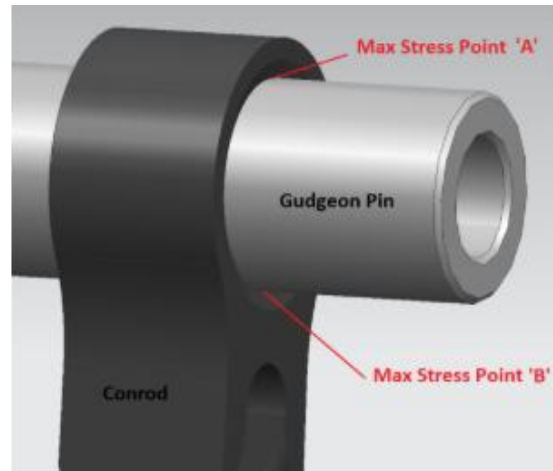


Figure 3 Gudgeon Pin max stress points (Fall, 2013)



Figure 5 Connecting Rod Bolt M8 (FORD, 2016)

Table 1 Material Properties

Property	Rod and Pin (MPa)	Bolt (MPa)
Ultimate Tensile Strength	1000	1100
Endurance Limit	450	495
Modulus of Elasticity	170,000	200,000
Yield Strength	900	990

Fatigue safety factors were determined using the equation below. Minimum and maximum stresses experienced by each component were used to calculate the mean and alternating stress values. Using these stress values and the material endurance limit and UTS, goodman diagrams were constructed.

$$\sigma_{allowable} = 2\sigma_{End-Lim} \times \left(1 - \frac{\sigma_{mean}}{UTS}\right) \quad \text{Equation 1}$$

$$FSF = \frac{\sigma_{allowable}}{\sigma_{range}} \quad \text{Equation 2}$$

There are limitations to the stress calculations due to simplifications, these include:

- Frictionless gudgeon pin
- Uniform material
- Constant temperature
- Rigid Connecting-rod
- Constant rotational speed
- Bolt preload is constant and evenly distributed

3 Stress and Fatigue Analysis

3.1 Connecting Rod Midpoint

The connecting-rod midpoint is a critical location for fatigue assessment due to it being under high alternating axial load generated by the gas pressure and inertia forces. While this analysis focuses on axial force, Shenoy and Fatemi (2006) show that bending stresses are also significant, meaning the current results may under predict realistic numbers.

The stress pattern follows similar behaviour to net force due to it being calculated using the net force, at the rod angle, as a factor of the midpoint cross-sectional area. It shows a large tensile peak at TDC caused by maximum piston deceleration, followed by a compressive region during expansion, where cylinder pressure is highest. These alternating forces produce the mean and alternating stresses used for fatigue assessment.

Table 2 Connecting Rod Stresses

Stress	Value (MPa)
Minimum	-30.2916 (at 1° crank)
Maximum	205.9768 (at 18° ATDC)
Mean	87.8426
Range	236.2684
Allowable	820.94
FSF	3.4746

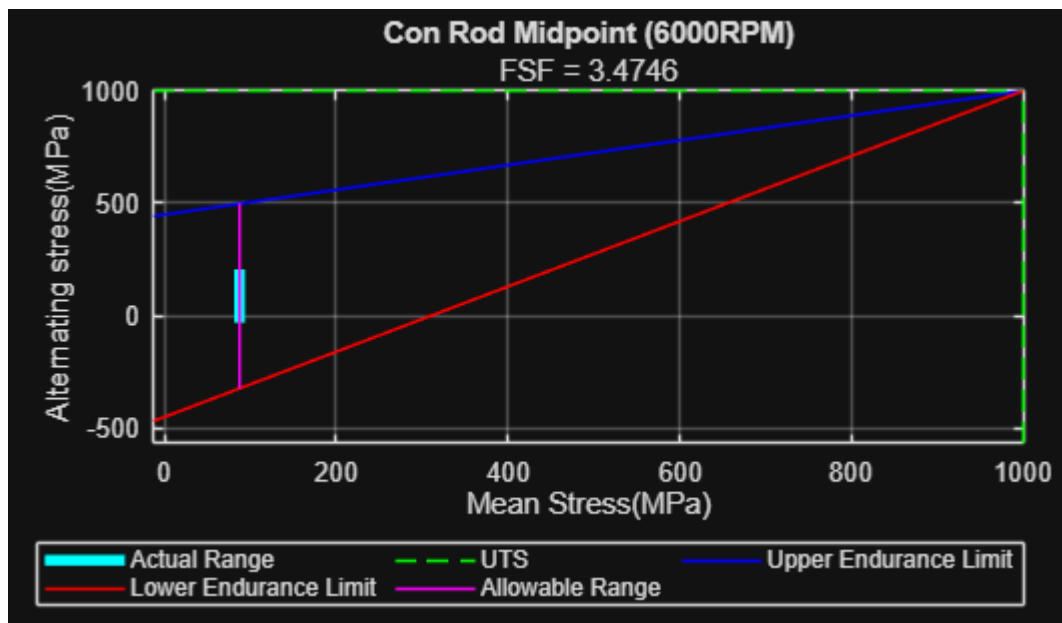


Figure 6 Connecting Rod midpoint Goodman Diagram

The goodman diagram for the connecting-rod midpoint illustrates how the rod material responds to cyclic loading. Its sloping endurance limit lines show that steels can only withstand high alternating stresses when the mean stress is low or compressive, as compressive loading suppresses crack growth. In contrast, tensile mean stress reduces fatigue resistance by promoting crack opening. The UTS line defines the maximum static stress the material can tolerate before single-cycle failure, providing an upper boundary for safe operation. The position of the operating stress point on the diagram is governed by the stress cycle generated through the interaction of gas pressure and inertia forces during the engine cycle. Together these produce the mean and alternating stresses that determine fatigue performance. The resulting fatigue safety factor of 3.47 indicates that the connecting-rod is capable of enduring more than 3 times the stresses encountered in normal operation,

demonstrating a robust design. MAHLE Industries, Inc. has defined the lowest allowable fatigue safety factor to be 1.2 (Lapp et al., 2010). And as noted by Marines et al. (2003), high cycle components require elevated safety factors to ensure durability under continuous, repeated loading.

3.2 Gudgeon Pin

The Gudgeon pin receives a compressive load through the piston which is transferred through the small end of the connecting-rod. In addition to axial compression, the pin also experiences bending and shear stresses caused by side thrust during the intake and expansion strokes. These combined loads make fatigue analysis essential to confirm the pin's ability to withstand repeated cyclic loading. Because the piston reverses direction twice per cycle, the load acting on the pin alternates between directions, causing fluctuations in both bending and inertia forces. This alternating behaviour is essential to understanding the fatigue performance of the component.

The axial component of the stress arises from gas pressure acting on the piston crown, while the bending moment is generated by side thrust and evaluated at the crank angle where net force on the pin is highest. As the pin also acts as a bearing surface, it is exposed to elevated contact pressures, and for this reason is commonly nitrided (MAHLE, 2016) to reduce friction and increase fatigue strength. In this analysis, the pin is assumed frictionless to simplify calculations.

Table 3 Gudgeon Pin Stresses

Stress	Value (MPa)
Minimum	-265.26
Maximum	34.99
Mean	-115.14
Range	300.24
Allowable	1003.6
FSF	3.3427

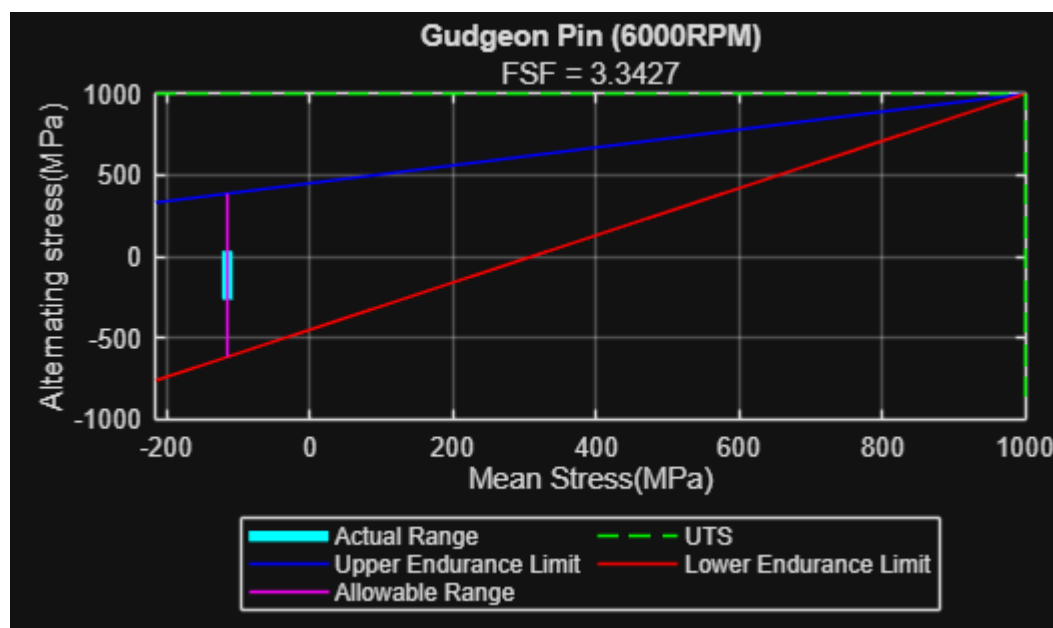


Figure 7 Gudgeon Pin Goodman diagram

Peak bending in the gudgeon pin occurs at 18° ATDC, where side thrust is maximised by high combustion pressure. The resulting safety factor indicates that the component is heavily over-engineered, with its operating point positioned centrally between the upper and lower endurance-limit. This implies infinite fatigue resistance by suppression of crack growth, reinforcing that the pin has a large, conservative safety margin.

3.3 Connecting Rod Bolt

The connecting-rod bolt is subjected to some of the highest stresses within the engine, mostly due to the combined effects of gas pressure and inertial forces. The maximum stress is when the piston hits maximum deceleration during exhaust/intake stroke, putting the bolt under huge tensile stress. With little gas force to counteract this, the bolt carries almost the full inertial load, elevating local tensile stress in the threaded region.

The initial simplified analysis modelled the bolt under a uniform axial load, producing a fatigue factor of 6.9. While this appears safe, it ignored the significant stress location of the thread roots. With the inclusion of this stress concentration factor (calculation found in Appendix, value 4.489), the FSF drops to 1.54, highlighting that bolt durability is governed highly by local stress amplification. This inclusion is necessary due to the first engaged thread carrying the majority of the load transfer between the rod and cap (Hui-li and Si-feng, 2014), making it the primary site for fatigue crack initiation.

$$EL_{new} = \frac{EL_{old}}{K_t} \quad \text{Equation 3}$$

$$\sigma_{allowable} = (2 \times EL_{new}) \times \left(1 - \frac{\sigma_{mean}}{UTS_{bolt}}\right) \quad \text{Equation 4}$$

Table 4 Gudgeon Pin Stresses

Stresses	Before Kt (MPa)	After Kt (MPa)	% Change
Min	594	594	0
Max	655.68	655.68	0
Mean	624.84	624.84	0
Range	61.68	61.68	0
Allowable	427.65	95.26	350.16
FSF	6.9336	1.5446	348.89

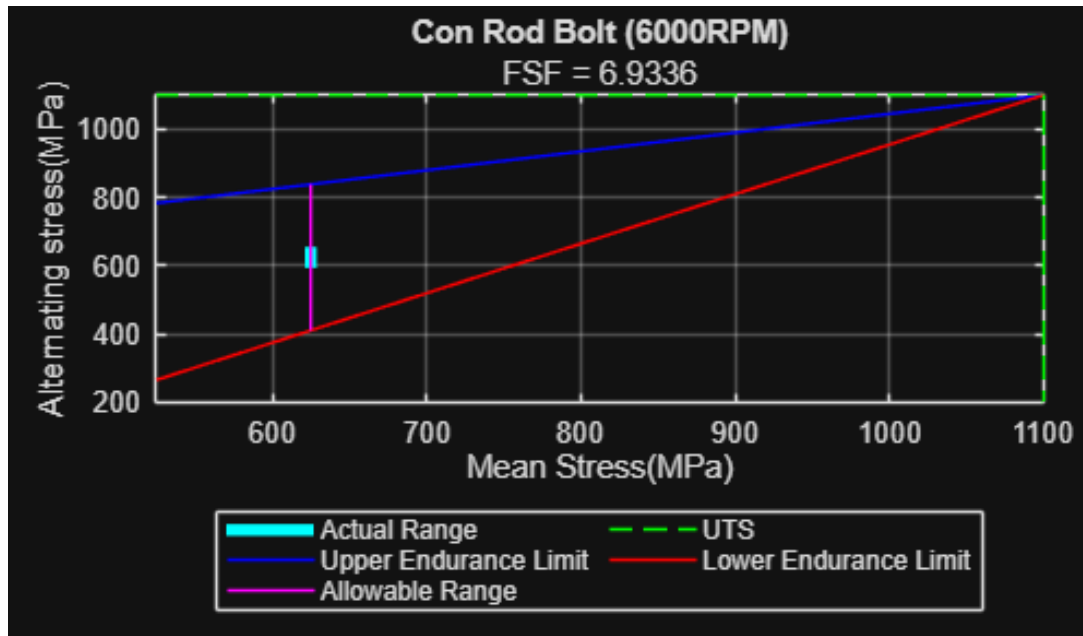


Figure 8 Connecting Rod Bolt Goodman diagram before Kt

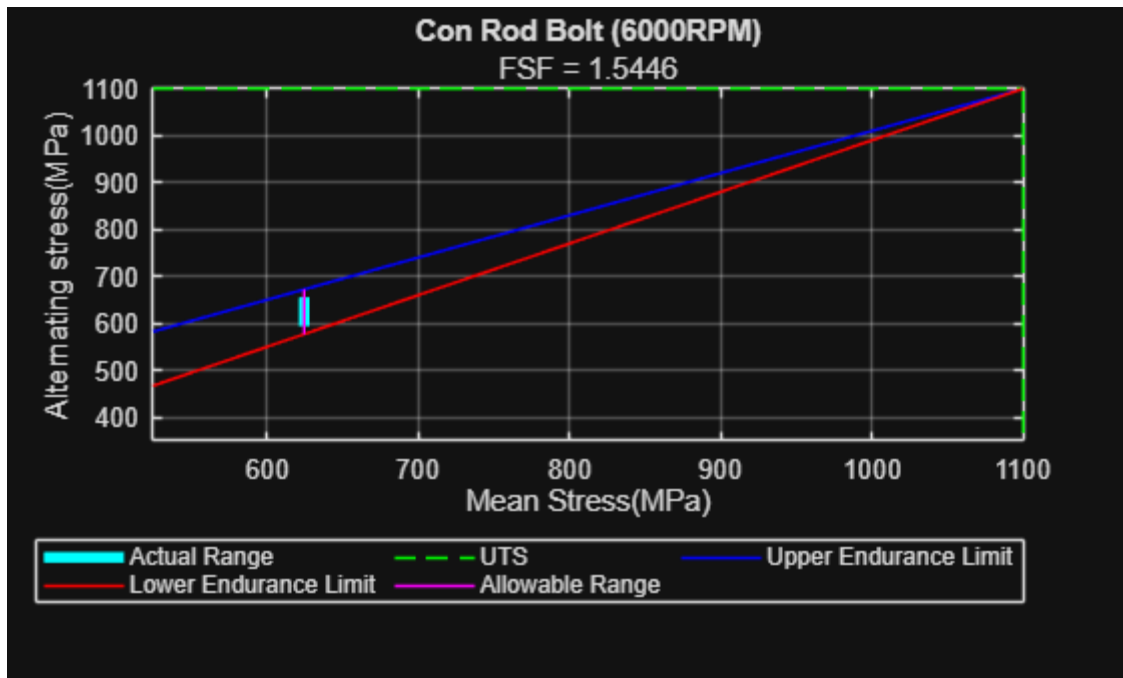


Figure 9 Connecting Rod Bolt Goodman diagram after K_t

Table 4 reflects the stress values before and after K_t was included in the calculations. The stress concentration factor changes the Endurance Limit of the bolt, reducing it by a factor of 4.489, and therefore having a drastic effect on the allowable stress. The goodman diagram reflects this behaviour clearly: the operating point shifts closer to the failure factor of less than 1, once K_t is applied, showing that the material experiences higher effective mean and alternating stresses. The diagram therefore highlights the sensitivity of the bolt material to tensile mean stress and demonstrates that fatigue performance is governed by the thread root rather than the nominal bolt section. The goodman diagram behaviour reinforces that the material becomes increasingly sensitive to tensile mean stress at the thread root, making this the true limiting factor in bolt durability.

4 Extension to 9000RPM

The pressure data was extrapolated in excel before being imported into MATLAB. The extrapolation cannot be an exact replica of 9000RPM due to assumptions of the trends similar to 6000RPM. This pressure data is used to recalculate the forces and stresses acting on the con-rod bolt at higher loads. Peak inertia force increases as a scale factor of 2.2 from 7269kgm² to 16355 kgm² with RPM increase, due to its proportional relationship to engine speed squared. Whereas, maximum gas force only increases by ~40% because combustion pressure is less sensitive to RPM changes than piston acceleration.

As a result, the total tensile load on the con-rod bolt becomes increasingly dominated by inertia rather than gas pressure, driving a sharp rise in alternating stress and a corresponding reduction in fatigue safety factor. Because inertia grows non-linearly with engine speed, the bolt experiences a rapid escalation in peak tensile load at high rpm, pushing the stress range beyond the materials allowable limit.

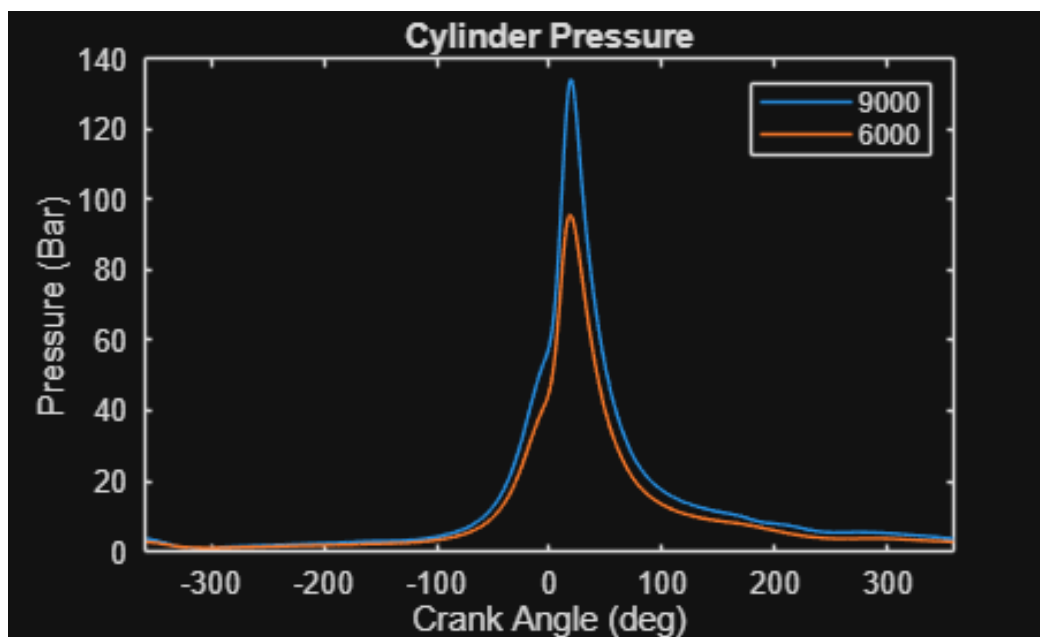


Figure 10 Cylinder pressure 6000 and 9000RPM

Table 5 Connecting Rod Stresses at 9000RPM

Stress	Value (MPa)	% change from 6000RPM
Minimum	594	0
Maximum	732.77	11.76
Mean	663.38	6.17
Range	138.77	125
Allowable	87.54	-7.86
FSF	0.6308	-59.16

Table 5 shows the stress values on the con-rod bolt at 9000RPM and the percentage change from 6000RPM. The fatigue safety factor decreases by 59.16% at 9000RPM due to the large increase in inertial forces and alternating stresses. This reduction indicates that the bolt would fail at 9000RPM, as the existing preload already places the bolt under high mean tensile stress and therefore the additional inertial load pushes the alternating component beyond the endurance limit, causing fatigue failure.

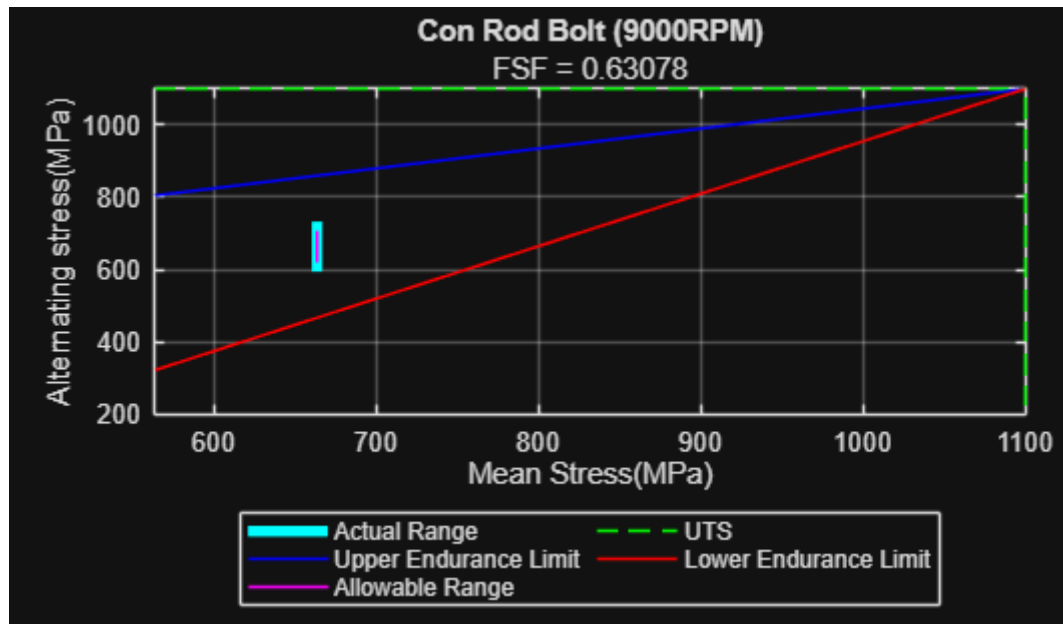


Figure 11 Connecting Rod Bolt Goodman diagram at 9000RPM

This safety factor is still likely to be an over-estimate due to the linear assumptions applied to the gas pressure extrapolation. The Goodman plot confirms that the actual alternating stress range exceeds the materials allowable limit, indicating a high-cycle fatigue risk. To restore an acceptable fatigue margin, the bolt requires redesign.

5 Redesign of Con Rod Bolts

The stress calculations highlight that the existing con-rod bolt design is unable to withstand the increased inertial loading at 9000RPM, leading to an insufficient fatigue safety factor. In motorsport applications, such limitations must be addressed before production, prompting redesign strategies that improve durability without adding unnecessary mass. Potential approaches include using higher-grade materials, reducing reciprocating mass or modifying bolt geometry. This section evaluates redesign options with the aim of increasing fatigue life at high engine speeds while minimising any negative impact on overall assembly mass.

5.1 Redesign 1: Shank Wasting

The initial bolt redesign involved reducing the shank diameter, a process known as shank wasting. Because the highest stresses occur at the thread root, decreasing the shank diameter shifts a greater portion of the load into the bolt head and cap, thereby lowering the stress concentration at the threads. This is effective because reducing the shank diameter helps minimise stress intensification at the thread roots (Majzoobi et al., 2014), improving fatigue performance. Reducing the shank diameter also increases elasticity of the bolt, reducing the alternating stress amplitude caused by cyclic loading.

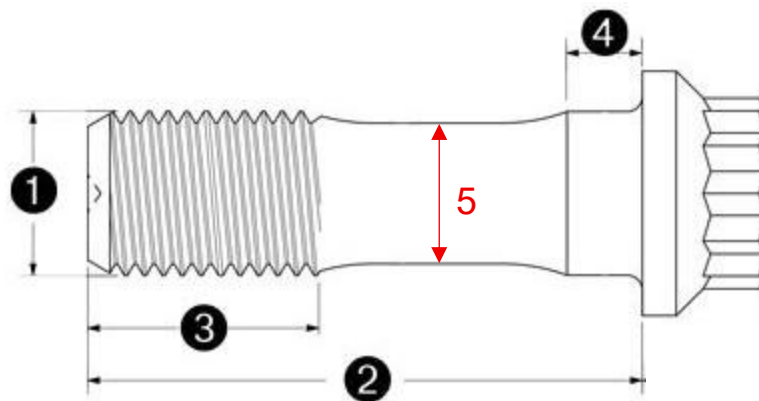


Figure 12 (1) 7.2mm, (2) 27.26mm, (3) 9mm, (4) 4mm, (5) reduced from 8mm to 6mm (JP Racing, 2026)

Table 6 Wasted Shank Con-Rod Bolt Stresses

Stress	Value (MPa)	% change from stock 6000RPM	% change from stock 9000RPM
Minimum	594	0	0
Maximum	680.47	3.78	-7.137685016
Mean	637.24	1.98	-3.942118251
Range	86.47	40.20	-37.68940868
Allowable	92.78	-2.34	5.989677343
FSF	1.0729	-30.54	70.09099845

Table 6 shows that the redesigned bolt limits the rise in maximum stress by only 3.78% of 6000RPM, a substantial improvement over the original 11% increase. The fatigue safety factor also rises above 1, indicating that the bolt can withstand high-cycle loading for longer, though it may still be insufficient for the extreme conditions of typical motorsport engines.

The main drawback of this redesign is the potential reduction in bolt stiffness, which could influence joint behaviour or introduce slight bending with the rod-cap interface. However, the reduced shank diameter also lowers reciprocating mass, marginally decreasing inertia forces and further reducing overall bolt loading.

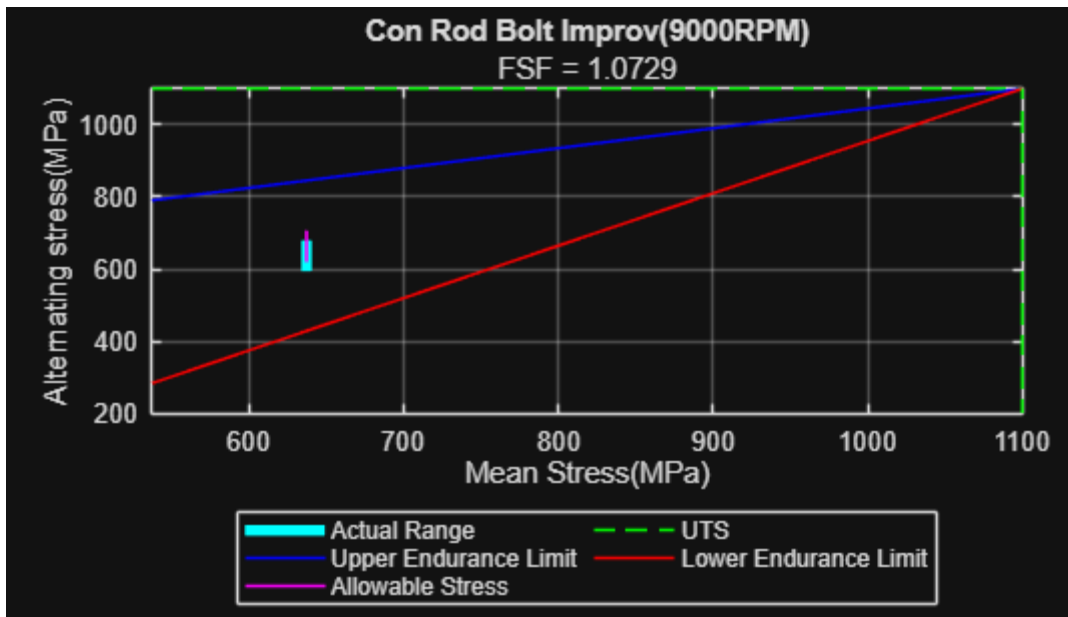


Figure 13 Wasted Shank Con-Rod Bolt Goodman diagram at 9000RPM

5.2 2: Change of material

The material selected for the bolt redesign was ARP2000, sourced from ARP's race-spec con-rod bolt documentation (ARP, 2026). ARP2000 is closely related to 8740 Chrome Moly but offers significantly higher strength and fatigue resistance. Its key mechanical properties are shown in Table 7.

Table 7 ARP2000 material properties (Alloy International Inc., 2020)

Property	Value
Ultimate Tensile Strength	1378.95 MPa
Yield Strength	1241.06 MPa
Modulus of Elasticity	205 GPa
Endurance Limit	138.23 MPa (including Kt)

Using ARP2000 increases the fatigue safety factor by 24% at 6000RPM and by 204% at 9000RPM. This improvement comes from its higher endurance limit, allowing the bolt to withstand greater alternating stresses without initiating fatigue damage. The increased UTS also raises the allowable mean stress (as the Goodman diagram shows), meaning the operating stress range remains within the materials fatigue range. Additionally, the higher yield strength permits greater preload without risk of yielding, which reduces the alternating stress amplitude and further enhances fatigue life.

Table 8 ARP2000 Con-Rod Bolt Stresses

Stress	Value (MPa)	% change from stock 6000RPM	% change from stock 9000RPM
Minimum	744.63	25.35909	25.35909091
Maximum	807.52	23.15778	10.20010262
Mean	776.08	24.20412	16.98682669
Range	62.88	1.957443	-54.68553187
Allowable	120.8691	27.23063	38.07993548
FSF	1.9221	24.43998	204.7179682

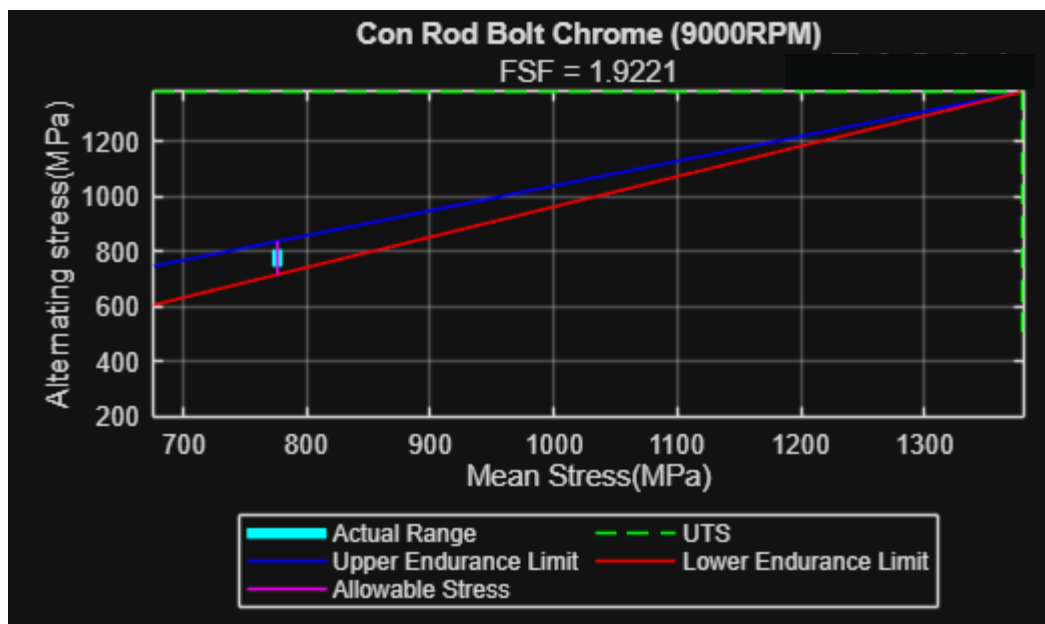


Figure 14 ARP2000 Con-Rod Bolt Goodman diagram at 9000RPM

Although ARP2000 provides superior performance, the threads must be rolled after heat treatment to achieve maximum tensile strength. Overall the material offers a much larger fatigue margin under high rpm operation due to its ability to tolerate higher mean and alternating stresses than the original bolt material.

Overall, the material upgrade provides a much more reliable increase in fatigue strength than geometric modification, making it the best option for an effective bolt redesign.

6 Instantaneous Torque

Instantaneous torque at 9000RPM is compared between a standard inline-four crank with a cross-plane configuration to evaluate the differences in torque delivery. Torque fluctuations arise from the interaction of piston-inertia forces and combustion-generated gas forces. This comparison highlights when each cylinder contributes torque and how they combine to form the overall crankshaft torque profile.

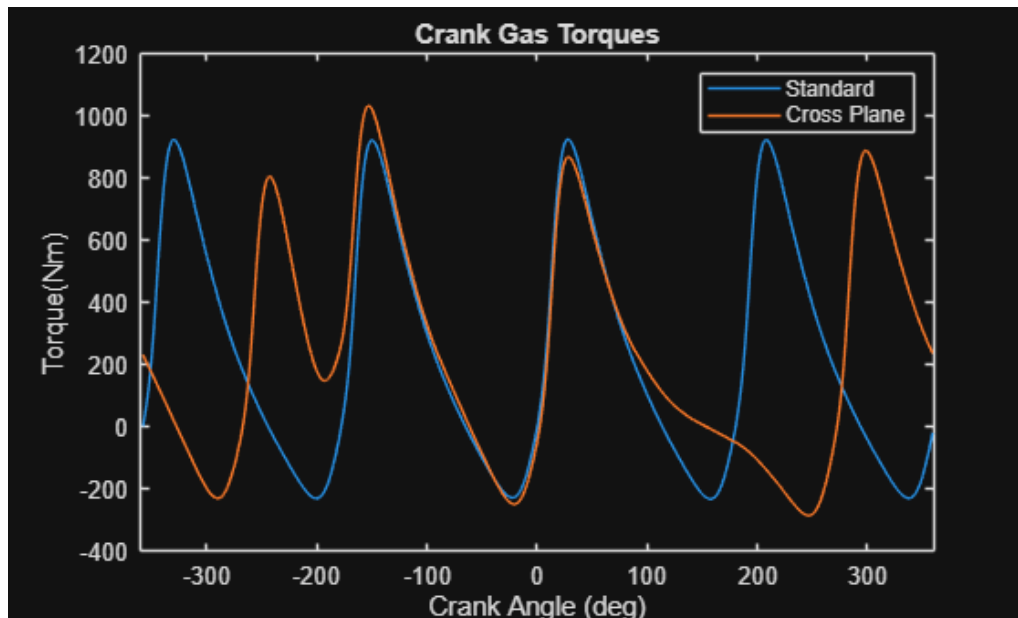


Figure 15 Gas Torque of Standard inline-four vs cross-plane crankshaft

The standard inline-four shows distinct gas torque peaks during combustion, while inertia torque becomes strongly negative during rapid piston deceleration. Cylinders 1-4 and 2-3 move in pairs, producing large combined torque pulses every 180° resulting in a high but irregular torque output. In contrast, a cross-plane crank fires every 90° and its pistons move with 90° phase separation. This reduces net torque magnitude but produces a much smoother torque graph, with extended periods of low or negative gas torque due to the wider spacing between combustion events.

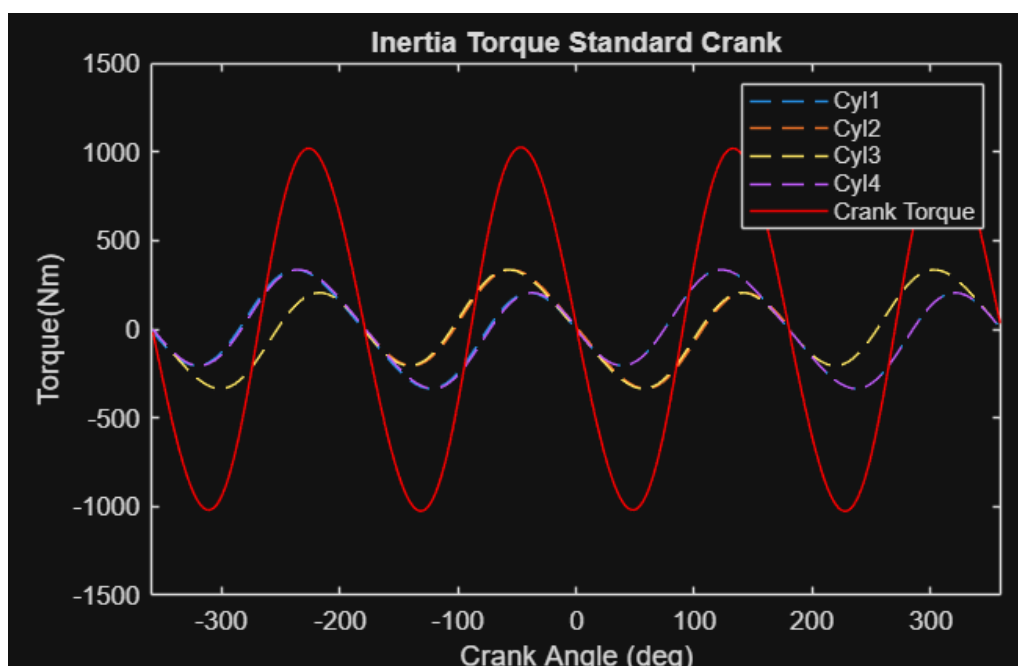


Figure 16 Inertia Torque of an inline-four crankshaft

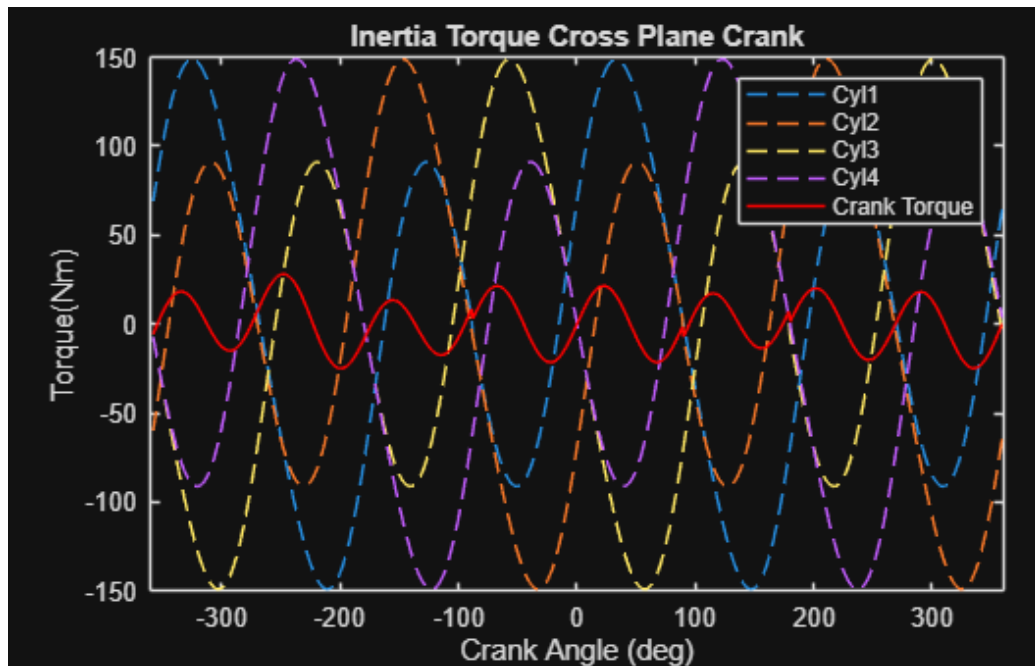


Figure 17 Inertia Torque of a cross-plane crankshaft

Smoother torque delivery improves drivability and reduces vibration loading on engine components, which is directly relevant to fatigue performance analysed in (Bo et al., 2022). Lower torsional oscillation reduces bending stresses in the crankshaft and minimises cyclic loading transmitted to the con-rod and its fasteners. However, cross-plane designs require additional counterweights to control secondary vibrations, increasing rotating mass and therefore inertia.

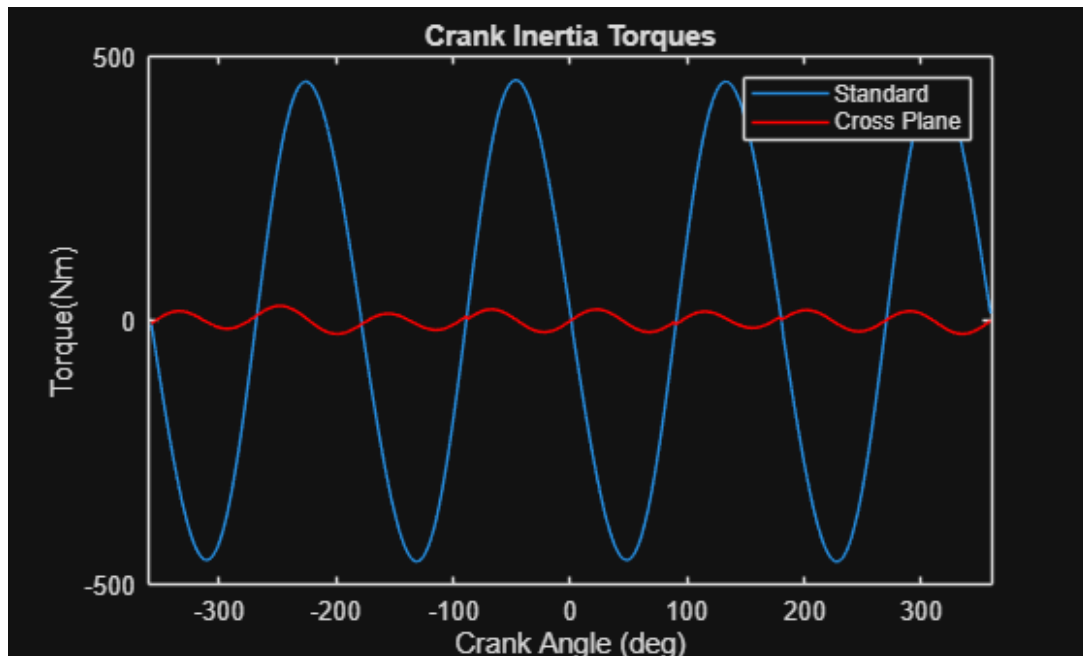


Figure 18 Inertia Torque of an inline-four and cross-plane crankshaft

In summary, lower inertia and smoother torque delivery reduce the cyclic stresses that drive high-cycle fatigue. Flat-plane cranks suit smaller, higher-revving engines where low mass is critical, while cross-plane cranks are preferred in larger engines for their smoother torque characteristics, provided appropriate balancing is used.

7 Conclusion

The fatigue analysis shows that the connecting-rod (**FSF 3.47**) and gudgeon pin (**FSF 3.34**) operate within large margins of material limits and are safe for use at 6000 and 9000RPM. In contrast, the con-rod bolt becomes close to fatigue limit once thread stress-concentration is included, with the safety factor dropping from **6.9** to **1.54**. Extending the operating engine speed to 9000RPM increases the maximum stress by 12% and reduces the safety factor to **0.63**, indicating failure would occur. Out of the proposed redesigns, shank-wasting provided an improvement of 70% (**FSF 1.07**) from 9000RPM, whereas upgrading to ARP2000 material increases the safety factor to **1.92** (204% from 9000RPM). Therefore providing a strong improvement in fatigue performance. Instantaneous torque analysis further supports the need for improved bolt durability due to the high-cyclic loads transmitted through the crank assembly.

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Appendix

Engine Parameters used for calculations

Parameter	Symbol	Value	Units
Bore	B	79	mm
Crank Radius	R	40.7	mm
Connecting-Rod Length	L	134	mm
Reciprocating Mass	m _{rec}	0.347	kg
Rotational Mass	m _{rot}	0.313	kg
CSA Rod	A _{rod}	196.39	mm ²
CSA Pin	A _{pin}	227.54	mm ²
CSA Bolt	A _{bolt}	42.43	mm ²
CSA Cap	A _{cap}	210	mm ²

Stress Concentration Factor equation:

$$K_t = C_1 + C_2 \left(\frac{2h}{D} \right) + C_3 \left(\frac{2h}{D} \right)^2 + C_4 \left(\frac{2h}{D} \right)^3$$

Where h = 0.46mm and D = 8mm