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Analytical assessment of engine internal components

Contemporary Powertrain Design (ACHA6006)

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1 Introduction

The primary aim of this study is to analyze the forces, stresses, and fatigue safety factors of key bottom-end components in the Ford 1.6 Sigma engine, assess the impact of increasing the engine RPM to 9000 RPM, and propose an optimized design for the connecting rod bolts to enhance engine reliability and performance. This will be achieved through computational analysis, interpolation methods, and design optimization techniques using MATLAB/Excel.

Modern engines are constantly pushed to operate at higher RPM ranges for increased performance. However, with higher RPMs the mechanical stress and fatigue loading on components can become a failure point so understanding and mitigating these loads is crucial for reliability, and longevity which is a key challenge in powertrain design.

2 Methodology

This study used MATLAB to calculate the dynamic forces and stresses experienced by the selected bottom-end components of a 1.6 litre Ford Sigma engine across a range of engine speeds. The provided pressure data and engine geometry were used to model the system and evaluate the stress and fatigue values to calculate safety factors by using kinematic equations, dynamic force equilibrium and Newton's Second Law.

2.1 Force Calculations

The equations used in figure 1 below were used to perform the dynamic and structural analysis of the components, the formulae used are fundamental to internal combustion engine (ICE) modelling as they include kinematic relationship and mechanic stress/strain calculations. Rod angle, engine displacement, velocity and acceleration were derived from the crankshaft motion using trigonometric and differential expressions. Force calculations were produced from gas pressure, inertia, net force and the force in the connecting rod.

	Formula
Rod angle	
	$\phi = \sin^{-1} \frac{R \sin \theta}{L}$
Distance from TDC	
	$S = (R + L) - (R \cos \theta + L \cos \phi)$
Velocity	
	$\frac{dS}{dt} = R\omega \sin \theta + \left(\frac{R^2 \omega}{2L} \sin 2\theta \right)$
Acceleration	
	$\frac{d^2 S}{dt^2} = R\omega^2 \cos \theta + \frac{R^2}{L} \omega^2 \cos 2\theta$
Mass reciprocating	
M_{rec} = Piston + Pin + Rings + 1/3 connecting rod	
Piston inertia FORCE	
	$F_{Inertia\ pist} = m_{rec} a$
Area of piston	
	$Area = \frac{\pi \times Bore^2}{4}$
Gas force	
	$F_{gas} = P_{cyl} \times Area$
Net force	
	$F_{net} = F_{gas} - F_{inertia}$
Rod force	
	$F_{rod} = \frac{F_{net}}{\cos \phi}$
Side force	
	$F_{side} = F_{net} \times \tan \phi$
Side friction	
	$F_{friction} = \mu \times F_{side}$
stress	
	$Stress = \sigma = \frac{Force}{Area} = MPa$
strain	
	$Strain = \epsilon = \frac{\delta}{L} = \frac{P \times L}{E \times A} = E \times \epsilon = Ratio$

Figure 1 - Equations List (UWTSD)

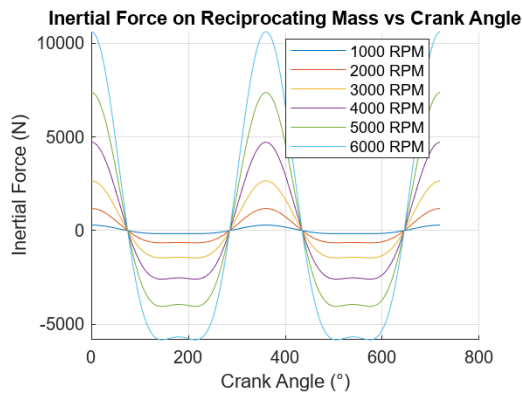


Figure 2 – Reciprocating Mass Inertial Force vs Crank

Figure 2 displays the Inertial force of the reciprocating mass against crank angle across engine speeds from 1000 to 6000 RPM. It shows us that as the RPM increases the force amplitude significantly increases with the peak inertial forces at 6000 RPM, this is due to the relationship between acceleration and angular velocity. At higher RPMs the peaks become more pronounced indicating a greater dynamic load on the reciprocating parts.

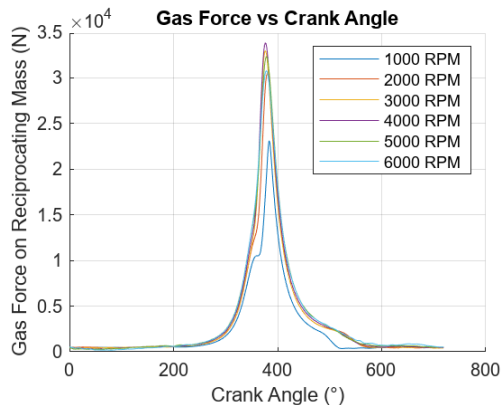


Figure 3 - Gas force vs Crank Angle

The figure above shows the gas force acting on the reciprocating mass for various engine speeds. The gas force spikes sharply at 360 degrees which corresponds with the power stroke with the magnitude constant across the RPM range, this indicates that combustion pressure is relatively stable with RPM. Unlike inertial forces, the gas force does not scale significantly with RPM across the crank as it highlights how combustion dynamics are more dependent on pressure than speed.

2.2 Assumptions

During the analysis of the components the following assumptions were made to enable calculations where direct measurements of specifications were unavailable.

2.2.1 Material Properties

The exact material specifications of the components were not provided to allow the incorporation of the young's modulus and yield strength to allow a more accurate calculation. Simplified results using the UTS and EL were used to focus on fatigue behaviour rather than the elastic strain limits.

2.2.2 Engine Temperatures

The analysis did not account for thermal expansion or changes in material strength with temperature as this data was not provided. This assumes a constant average material behaviour across the engine cycles although real world application would show temperature fluctuations particularly during the power stroke, this can influence the fatigue life of the component('Pistons and engine testing', 2016).

2.2.3 Friction Coefficient

The coefficient of friction between the piston and rings during the engine cycle was given an estimate of 0.1, reflecting typical values for lubricated steel on steel contacts. This may introduce uncertainty in the calculations as it does not consider that the coefficient of friction will change with operating temperatures as the oil film thickness will change as well as expansion of components(Belahcene *et al.*, 2018).

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3 Results

3.1 Midpoint of the Connecting Rod.

The midpoint of the connecting rod is subjected to a combination of stresses such as axial, bending and inertial forces. Axial forces are the most dominant forces due to the gas pressure pushing the piston down during combustion and the inertial forces during acceleration and deceleration of the connecting rod (Pan and Zhang, 2021). These forces act along the length of the rod and alternate between compression and tension depending on the stroke.

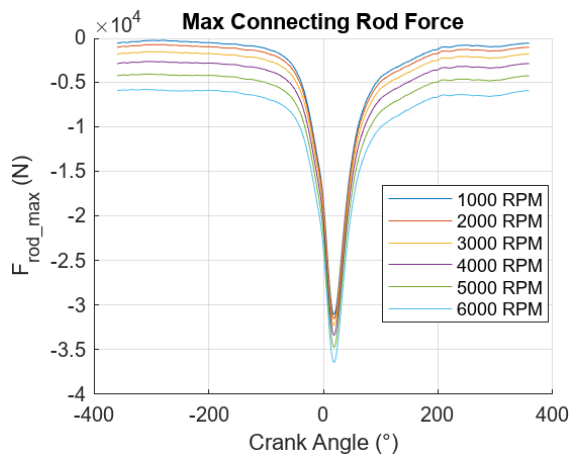


Figure 4

The plot above shows the maximum force experienced by the connecting rod as a function of the crank angle across the rpm range of the engine. The highest tensile force occurs just after TDC where inertial forces from the piston are greatest. As the RPM increases, the magnitude also peaks increasingly especially in the negative direction under tension. This highlights the importance of material strength and fatigue resistance when designing connecting rods, especially for higher RPM requirements.

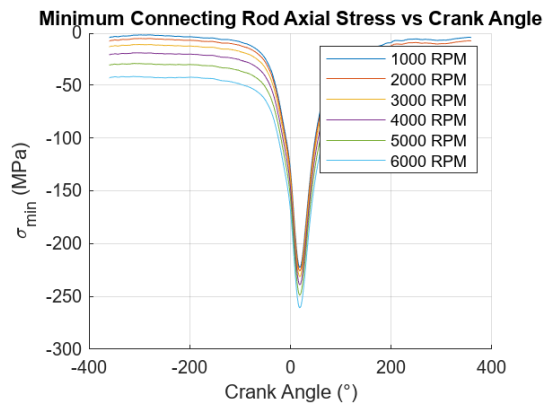


Figure 5 - Connecting rod axial stress

Figure 5 gives us the plot of tensile stress against crank angle for the RPM range of the engine, with a peak of 260MPa occurring during TDC at 6000 RPM, this highlights the importance of material selection and gives us the range of bending stress that the mid point of the connecting rod is subjected to.

3.2 Gudgeon pin at point A.

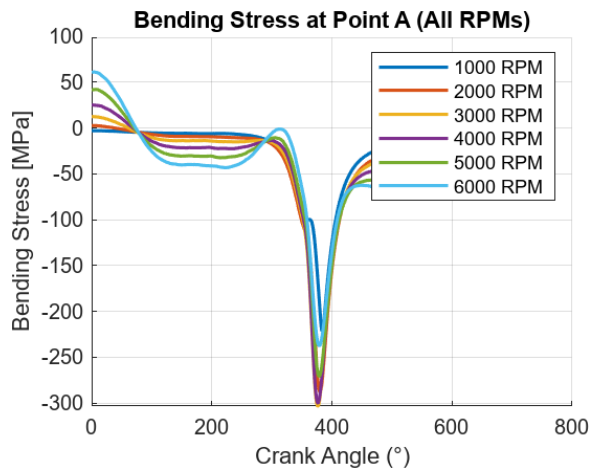


Figure 6 - Gudgeon Pin Bending Stress

Figure 6 above illustrates the bending stress at Point A of the Gudgeon pin across the full engine cycle (720 degrees) for the rpm range of the engine. All RPMs follow a similar stress profile with the compressive stresses dominating. The maximum overall bending stress occurs consistently around the 360-degree crank angle which corresponds to the start of the exhaust stroke where inertial and gas forces are both high and opposing. A peak stress reaches approximately -290MPa which indicates a significant compressive load. The results showed the maximum force occurring at the mid

rpm range of the engine where combustion loads peak due to the available time for the cycles to complete without losing efficiency(Agalarov, Hasanova and Imamalieva, 2024).

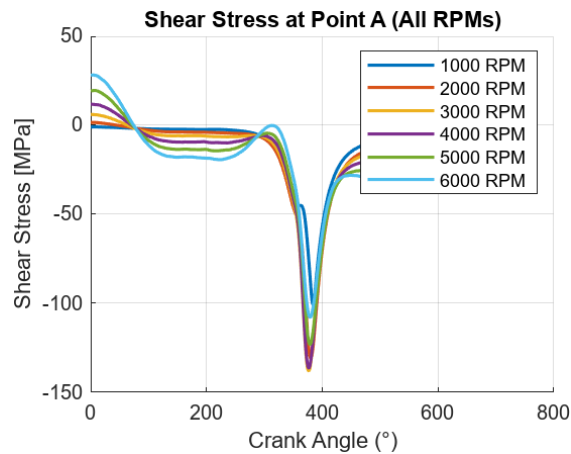


Figure 7 - Shear Stress at Point A

The shear stress shows a peak compression force at the end of the compression stroke, transitioning into the combustion stroke. The force is transferred through the Gudgeon pin into the conrod leading to a bending moment in the pin as well as axial and radial forces.

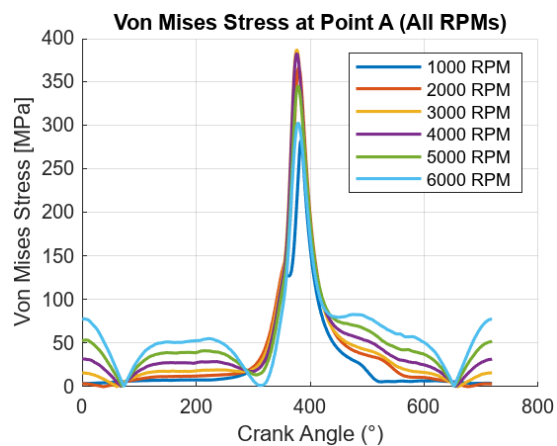


Figure 8 - Von Mises stress at Point A

Figure 8 gives us a von mises stress which combines both the bending (tensile/compressive) force with the shear to give a critical value to compare with the yield strength of the material. The combined maximum force is nearer 400MPa which is well below the UTS of the material which is 1000MPa which means the stresses are well below the failure limit of the material in combined stresses.

Table 1 – Gudgeon Pin Stress Values

	Stress Values					
RPM	σ min (MPa)	σ max (MPa)	σ mean (MPa)	σ range (MPa)	σ allowable (MPa)	FSF
1000	50	180	115	130	796.5	6.126923077
2000	48	200	124	152	788.4	5.186842105
3000	45	220	132.5	175	780.75	4.461428571
4000	40	240	140	200	774	3.87
5000	35	260	147.5	225	767.25	3.41
6000	30	280	155	250	760.5	3.042

Table 1 presents the fatigue analysis of the Gudgeon pin at point A across the rpm range of the engine. The minimum and maximum stress values were extracted from the gas force data on the reciprocating mass and the inertial force. These were then used to compute the mean and range stresses against the Ultimate Tensile Strength (UTS) of 1100 MPa and Endurance Limit (EL) 450MPa of the material. A resulting FSF was then determined using these ranges where the results show that the FSF does decrease with an increase in engine speed which indicates higher cyclic loading and a reduced fatigue margin at increased engine speeds. However, the values remain above 1 which suggests that the Gudgeon pin is operating within a safe fatigue limit.

The results showed that all operating points were within the safe zone the components fatigue limit as both the alternating and mean bending stresses stayed well below the endurance limit even at 6000RPM. This suggests that the Gudgeon pin is not expected to fail likely exceeding the engine's service life. The Gudgeon pin will maintain structural integrity even under full load, high speed conditions. Possible over-engineering observations were that the bending stress levels were less than 30% of the material's UTS with safety factors clearly above 1, shear and axial loads were also not near material limits.

3.3 Con rod bolt

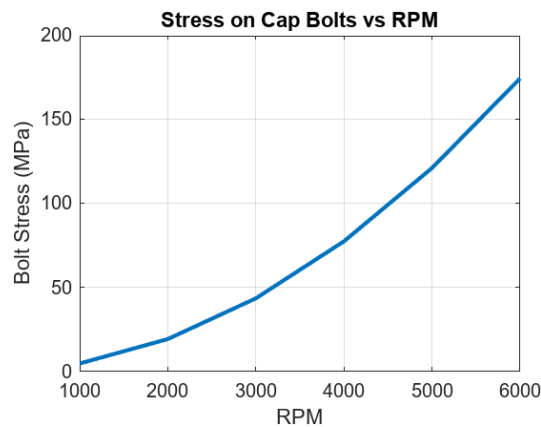


Figure 9

Figure 9 illustrates the relationship between engine speed (RPM) and the tensile force experienced by the connecting rod bolts. As the RPM increases, the tensile force rises due to increased inertial forces acting upon the bolt (Heywood, 2018). The bolts must resist higher loads resulting from increased reciprocating mass acceleration at higher engine speeds that stretch the components. At 6000 RPM the tensile force peaks at approximately 15kN, highlighting the need to select a bolt material that can withstand these forces throughout the components life cycle.

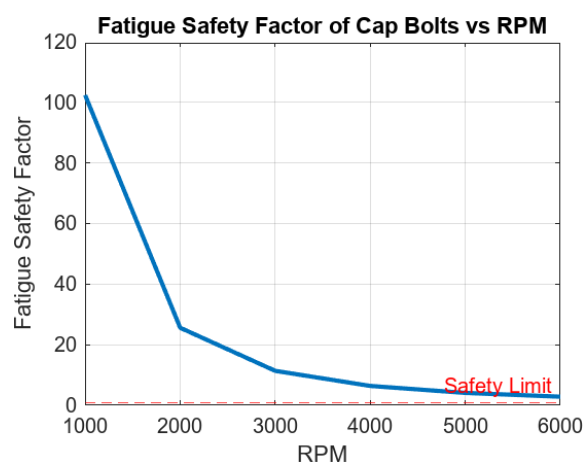
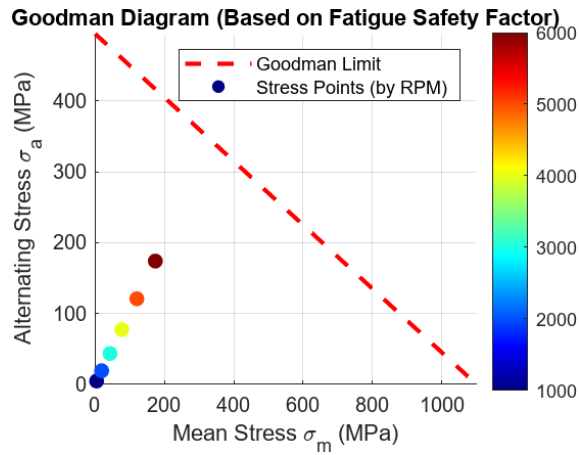


Figure 10

Figure 10 displays how rapidly the FsF declines with an increase in engine speed. At lower RPM, the FsF is extremely high due to minimal loading, however as the RPM increases the FsF drops sharply, approaching the critical fatigue limit of the bolt.



The goodman diagram presents the relationship between the mean stress and alternative stress that acts on the bolts. All stress points are below the failure line, indicating that under these operating conditions the bolt remain within the safe fatigue zone(Savaripour *et al.*, 2025). However, the curve shows a clear upward trend as the RPM increases, so do the stresses. This validates that the FsF decreases at increased engine speeds.

3.4 Analysis of Higher RPM (9000 RPM) Effects

The crank angle at which peak pressure occurs gradually decreases with increasing engine speed, from $\sim 384^\circ$ at 1000 RPM to $\sim 374^\circ$ at 9000 RPM. This reflects the need for earlier combustion as the engine speed increases as well as the inability to get sufficient air into the cylinder for combustion and time to extract the gas during the exhaust stroke. The values for 7000-9000 RPM were interpolated using the pchip function in MATLAB which is a cubic interpolation, the results do not account for physical changes that would occur during the engine cycles when the RPM is increased such as temperature fluctuations, knock control or the air-fuel ratio.

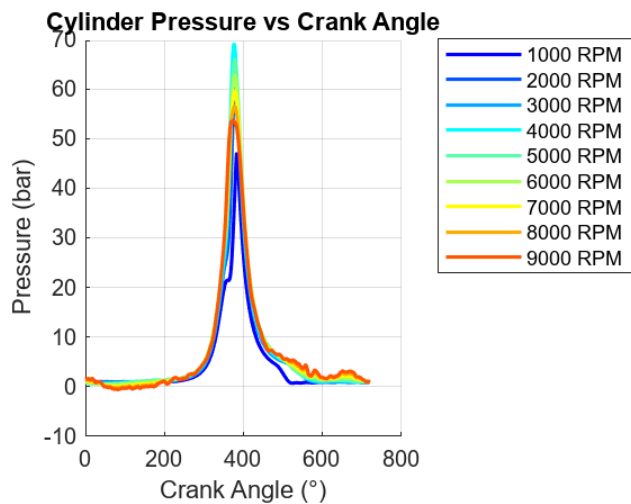


Figure 11 - Cylinder pressure vs Crank Angle

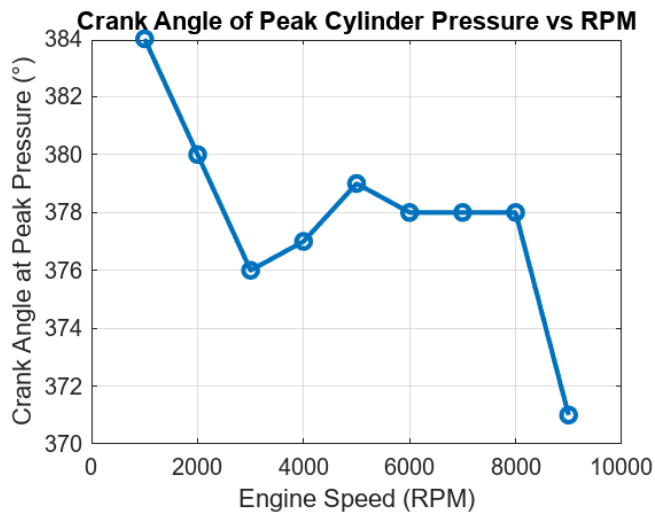


Figure 12 - Crank angle peak pressure vs RPM

Figure 12 shows where the peak pressure shifts crank angles with an engine speed increase. As the RPM increases, combustion must occur earlier to ensure peak pressure aligns with the pistons downward motion during the power stroke. The trend downwards as the RPM increases can be in part due to the engine no longer having time to effectively intake enough air for the combustion phase as well as there not being enough time to remove the gases during the exhaust phase.

Table 2 - Revised FsF

RPM	Stress Values		FSF
	$\sigma_{\max}$ (Mpa)	σ_{mean} (MPa)	
7000	312.56	156.28	3.17
8000	410.19	205.1	2.41
9000	520.82	260.41	1.9

Recalculating the stresses and safety factor of the connecting rod bolts at an increased engine speed reveals a trend of increasing stress on the component. At 7000 RPM the bolts continue to operate within the safe fatigue limits with a comfortable factor indicating a high reliability and service life. However, at 8000 RPM the FsF decreased to 2.41 suggesting a reduced fatigue life and more strain on the components. The maximum engine speed of 9000 RPM gave us the lowest FsF score of 1.9. Whilst still above the minimum threshold for safe operation, it is approaching the fatigue endurance limit and long-term reliability reduces. The components life cycle is significantly reduced as well as an increased risk to fatigue failure especially under high cycle conditions.

Were the engine to operate at this RPM over a sustained period, design modifications would be recommended to ensure long-term durability and safety. These modifications could include

upgrading the material to that of one with a higher UTS to reduce the effects of stress(He *et al.*, 2022).

3.5 Redesign of Connecting Rod Bolts for Improved Performance

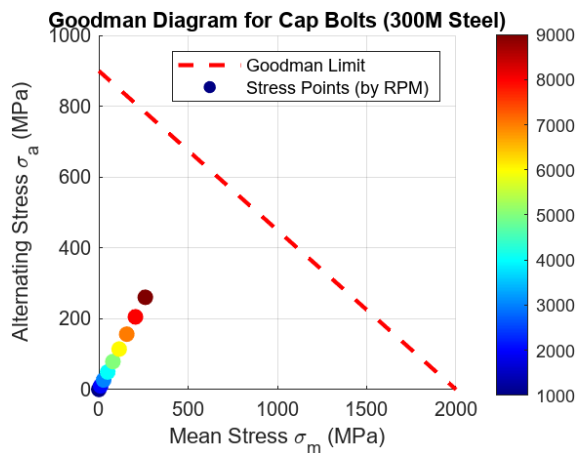
3.5.1 Increased Bolt Head Area

A method to increase the FsF of the bolt could be to increase the bolt head area. Although the strength of the bolt is determined by the cross-sectional area, the bolt head plays a critical role in distributing the preload forces and reducing local stress concentrations on the contact surface. Increasing the area will reduce peak stresses and movement under load. A larger head can also allow for a higher torque value when tightening the bolt which can further improve fatigue life by ensuring the bolt remains in tension and reducing alternating stress amplitude.

3.5.2 Material Selection

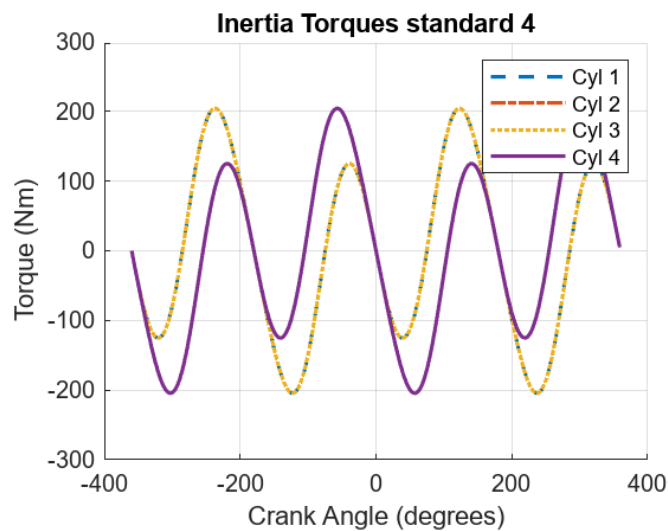
To ensure fatigue reliability of the connecting rod cap bolts, 300M steel was selected as a material to calculate the FsF and stress properties of the bolt compared to the provided properties. 300M is an ultra-high strength steel widely used in the automotive industry for bearing components with a high yield and tensile strength which would make it a suitable choice for connecting rod bolts in high load environments(Zhao *et al.*, 2021). UTS of 300M can be as high as 2000MPa which is almost double that of the default material and with an EL of 900MPa.

3.5.3 Result

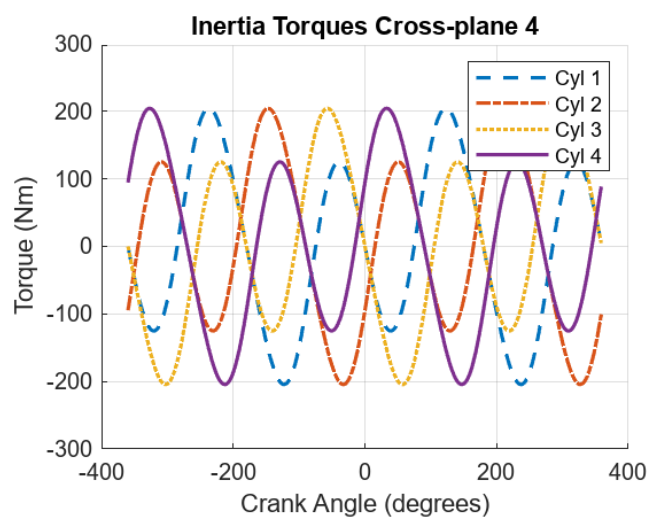


The results using the 300M steel for the connecting rod bolt shows a reduced overall stress and all points remain well within the safe zone. 9000 RPM still shows the highest alternating and mean stress but maintains a substantial margin from the failure limit and a reduction over the original material. The FsF of this material at 9000 RPM has increased to 3.46, an appropriate level for high-cycle fatigue application.

3.6 Torque Analysis for Different Crankshaft Configurations



The standard inline 4 configuration shows overlapping sinusoidal curves with repeating torque pulses. The overlapping pattern results in more consistent combined torque which makes this configuration mechanically balanced for torque output however can produce more vibration.



The cross-plane configuration shows more phase separation between cylinders where each cylinders inertial torque has a noticeable shift and their misalignment causes the total torque to fluctuate noticeably.

3.6.1 Comparison

Standard and Cross-plane crankshafts offer unique characteristics that can influence their suitability for automotive application.

Standard configuration crankshafts are typically lighter than cross-plane designs which allows for quicker revving and higher RPM limits which can improve performance as they would have lower rotational inertia. The disadvantage of a standard configuration is the introduction of significant vibrations due to their firing order and rotation. Low-end torque may be lower than that of the cross-plane crankshaft

Due to their design, cross-plane configurations can reduce vibrations and enhance comfort and driveability due to the primary and secondary balance which can aid in reducing Noise, Vibration and Harshness. They may also offer greater low-end torque which is beneficial for everyday driving. The need for counterweights, cross-plane crankshafts are typically heavier which can reduce the capability for high-revving applications(Albers *et al.*, 2009).

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