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Analytical Assessment of Engine
Internal Components

Contemporary Powertrain

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Contents

Abstract.....	1
1. Introduction.....	1
2. Methodology and Governing Equations	2
2.1 Slider-Crank Kinematics	2
2.2 Force Calculations.....	3
2.2.1 Inertia Force	3
2.2.2 Gas Force.....	3
2.2.3 Connecting Rod Force	3
2.3 Stress Analysis.....	4
2.4 Fatigue Analysis	4
2.5 Component Modelling Assumptions	5
2.5.1 Connecting Rod.....	5
2.5.2 Gudgeon Pin.....	5
2.5.3 Connecting Rod Bolt.....	6
3. Results and Analysis 6000 RPM	7
3.1 Connecting Rod	7
3.2 Gudgeon Pin	8
3.3 Connecting Rod Bolt	9
3.4 Comparative Analysis	9
4. Effect of Increasing Engine Speed	10
4.1 Overview	10
4.2 Results and Analysis	10
5. Conrod Bolt Redesign	11
5.1 Design Rationale.....	11
5.2 Material Upgrade – 300M steel.....	12
5.3 Design Evaluation and Failure Consideration	13
6. Crankshaft Torque Analysis	14
6.1 Background	14
6.2 Methodology.....	14
6.3 Results and Analysis	15
6.4 Discussion.....	17
6.5 Balance Shaft Considerations	18
7. Conclusion.....	19
8. References	20

Abstract

This report investigates the stresses, fatigue safety factors and loading conditions experienced by bottom-end components of a 1.6 L gasoline direct injection engine. Using in-cylinder pressure data combined with slider-crank kinematics, the force histories acting on the connecting rod, gudgeon pin and conrod bolt were determined. These loading histories were used to evaluate fatigue behaviour through Goodman analysis.

The study was extended to higher engine speeds by interpolating pressure data up to 9000 RPM, allowing assessment of the effect of increasing inertial forces on component fatigue performance. Results indicate that the conrod bolt is the most critical component, with fatigue safety factor decreasing significantly as engine speed increases. To address this, a redesigned bolt geometry incorporating a wasted shank was analysed, followed by a material upgrade to 300M steel.

Finally, instantaneous crankshaft torque was evaluated for both standard and cross-plane crank configurations to assess the influence of crank phasing on torque fluctuation. The redesigned bolt and improved material selection demonstrated clear improvements in fatigue performance, highlighting the importance of both geometric compliance and material properties in high-speed engine applications.

1. Introduction

Internal combustion engine components are subjected to cyclic loading from combustion pressure and reciprocating inertia. These loads increase with engine speed, making fatigue failure a key design concern. As highlighted by *Rizzoni*, dynamic loading governs durability in engine components.

This study analyses the bottom-end loading of a Ford Ecoboost 1.6 GDI engine using in-cylinder pressure data and slider-crank kinematics. Stresses and fatigue safety factors were determined for the connecting rod, gudgeon pin and conrod bolt using Goodman analysis. The effect of increasing engine speed to 9000 RPM was then assessed, followed by a conrod bolt redesign and material upgrade. Finally, crankshaft torque behaviour was evaluated for standard and cross-plane configurations.

2. Methodology and Governing Equations

This section determines the forces acting on the key bottom-end components and evaluates their fatigue safety factors under engine operating conditions. In-cylinder pressure data was used together with component mass properties and slider-crank kinematics to calculate the dynamic forces within the piston-rod-crank system. These force histories were then used to determine stresses and fatigue behaviour for the connecting rod, gudgeon pin, and conrod bolt. The cylinder pressure trace used for the 6000 RPM analysis is shown in *Figure 1*

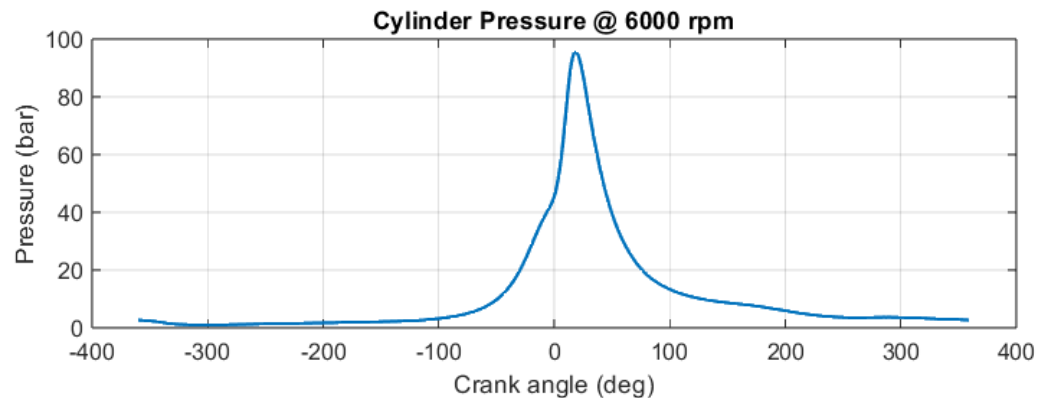


Figure 1 – Cylinder Pressure Trace

2.1 Slider-Crank Kinematics

Piston motion was modelled by using slider-crank kinematics. With displacement, velocity and acceleration expressed as functions of crank angle. The piston acceleration is given by:

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

Where R is crank radius, L is connecting rod length, ω is angular velocity, and θ is crank angle. These relationships allow the inertia forces of the reciprocating mass to be determined. Typical piston speeds were consistent with values reported by *Heywood* (14-20m/s), supporting the validity of the kinematic model.

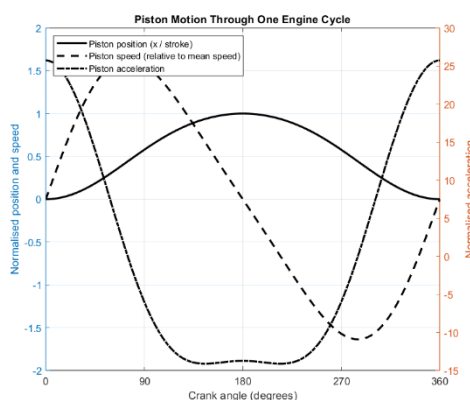


Figure 2 – Calculated Piston Motion

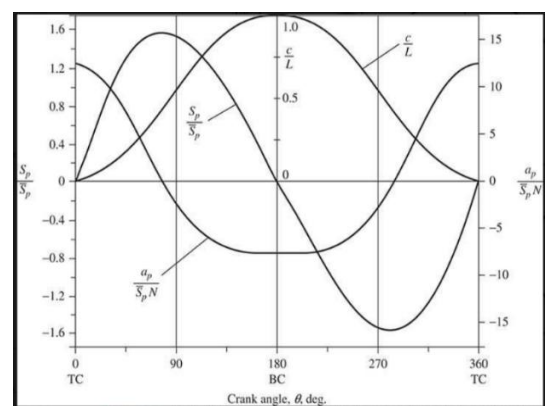


Figure 3 – John Heywood, 2018

2.2 Force Calculations

2.2.1 Inertia Force

The inertia force due to the reciprocating mass is:

$$F_{inertia} = m \cdot a$$

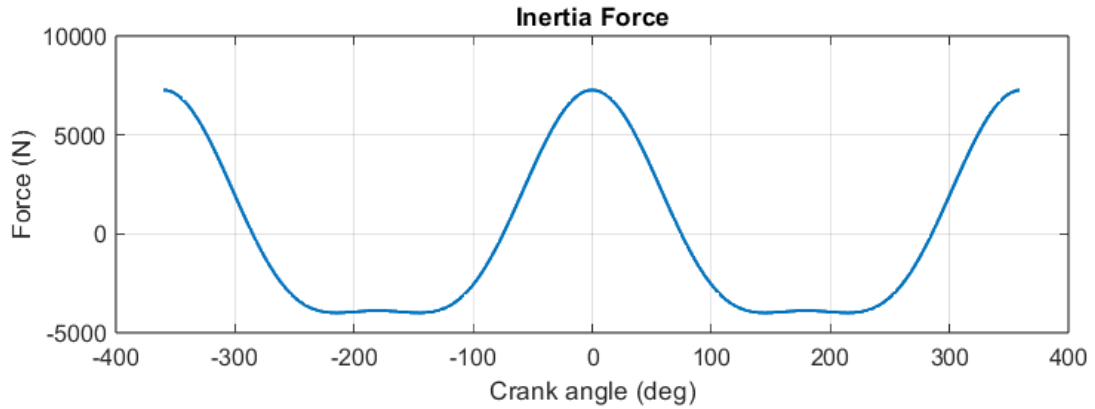


Figure 3 – Inertia Force

2.2.2 Gas Force

The gas force acting on the piston is obtained from the in-cylinder pressure trace:

$$F_{gas} = P(\theta) \cdot A_p$$

Where $P(\theta)$ is cylinder pressure and A_p is piston area.

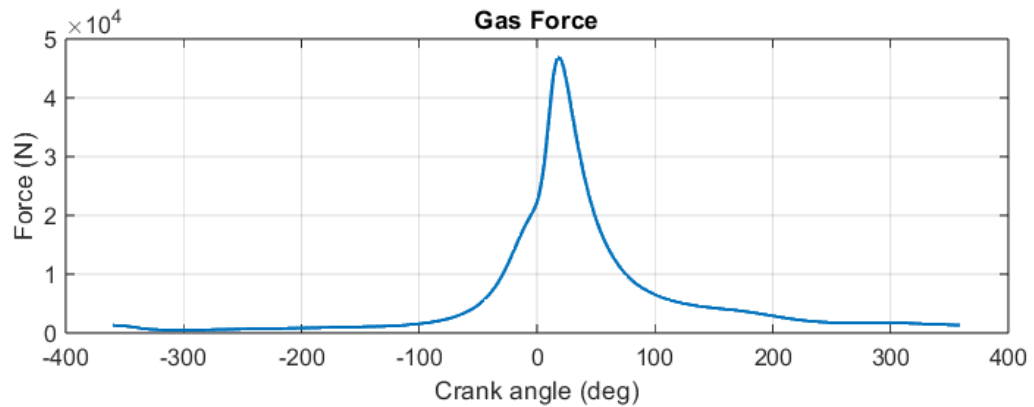


Figure 4 – Cylinder Gas Force

2.2.3 Connecting Rod Force

The axial force in the connecting rod is then resolved as:

$$F_{rod} = \frac{F_{gas} - F_{inertia}}{\cos \phi}$$

Where ϕ is connecting rod angle.

These force histories form the basis for the subsequent stress and fatigue calculations.

2.3 Stress Analysis

Normal stress in each component was calculated using:

$$\sigma = \frac{F}{A}$$

Where A is the relevant cross-sectional area. Different areas were used depending on the component geometry, including the rod midpoint, gudgeon pin section, and bolt shank.

2.4 Fatigue Analysis

Fatigue behaviour was assessed using the Goodman relation:

$$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{UTS}} = \frac{1}{FSF}$$

Where σ_a is alternating stress, σ_m is mean stress, S_e is endurance limit, and S_{UTS} is ultimate tensile strength. This approach accounts for the combined effect of mean and fluctuating stresses, which is essential for components operating under repeated cyclic loading.

2.5 Component Modelling Assumptions

2.5.1 Connecting Rod

The maximum stress was evaluated at the midpoint of the connecting rod (Point A), where the combined effects of gas and inertia forces are greatest. This location corresponds to regions of peak stress identified in *Cylinder Components Analysis* reported by MAHLE GMBH (Figure 2). The connecting rod was assumed to be steel, with an ultimate tensile strength of 1000 MPa and a Young's Modulus 170 GPa

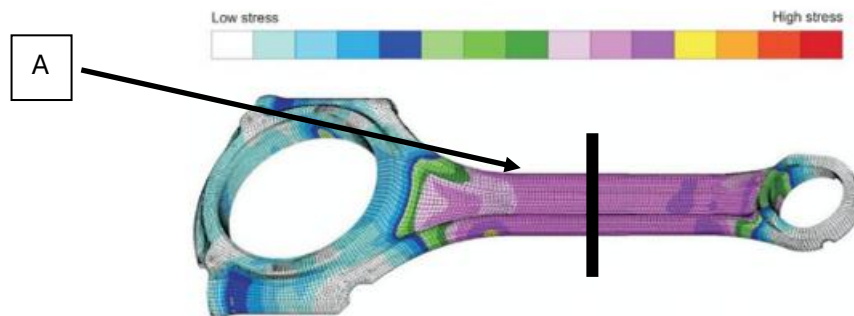


Figure 5 - Mahle Cylinder Components, 2016

2.5.2 Gudgeon Pin

The gudgeon pin was analysed at critical locations where bending stresses are highest due to piston loading and load reversal during engine operation (Point A). The pin experiences bending as the piston applies force through the small end of the rod, and this loading reverses during the engine cycle. The gudgeon pin was assumed to have an ultimate tensile strength of 1000 MPa.

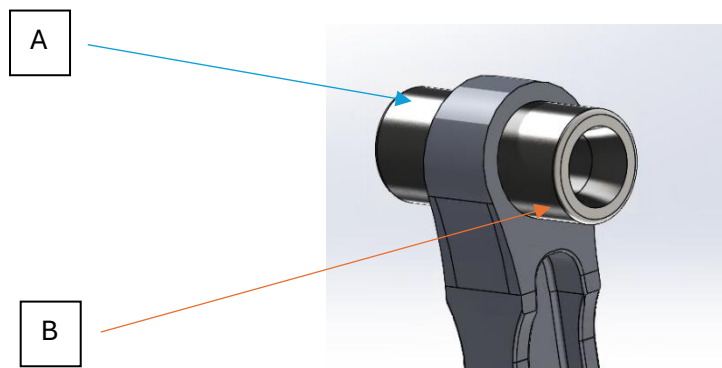


Figure 6 – Ford GDI Connecting Rod Top Assembly

2.5.3 Connecting Rod Bolt

The OEM connecting rod bolt was modelled without a wasted shank, with threads extending over most of its working length. The effective working length was taken as 27.26mm, with an ultimate tensile strength of 1100 MPa. The bolt was analysed under combined preload and external loading, making it particularly sensitive to fatigue due to its high mean stress.



Figure 7 – Ford OEM Connecting Rod Bolt, no date

3. Results and Analysis 6000 RPM

3.1 Connecting Rod

The maximum stress at the midpoint of the connecting rod was 205.98 MPa at 6000 RPM, with a fatigue safety factor of 2.854.

The maximum compressive stress occurred at 19° crank angle, which closely aligns cylinder pressure at 18°. This indicates that connecting rod loading is primarily driven by combustion pressure, with only a secondary contribution from inertia and kinematic effects. The maximum tensile stress occurs at -358°, corresponding to the reversal of piston motion.

The relatively high fatigue safety factor suggests that the connecting rod midpoint is not a critical location under these conditions and could be considered as conservatively designed.

Table 1 – Connecting Rod Performance at 6000 RPM

Parameter	Value
σ Max	205.98 MPa
σ Min	-30.29 MPa
σ Mean	87.84 MPa
σ Alternating	118.13 MPa
Fatigue Safety Factor	2.854

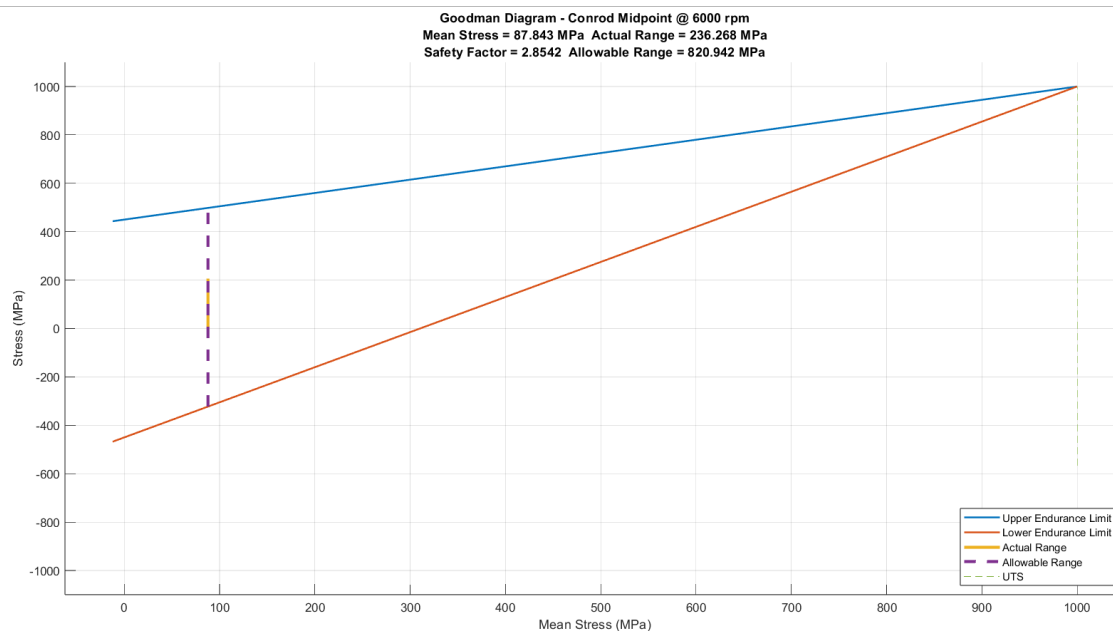


Figure 8 – Connecting Rod Goodman Diagram

3.2 Gudgeon Pin

The Gudgeon pin experienced a maximum stress of 250.04 MPa, with a fatigue safety factor of 2.483.

The loading is dominated by bending, as the piston applies force to the pin while changing direction during the engine cycle. Unlike the connecting rod, the pin is subjected to significant alternating stress due to the continuous reversal of loading direction, resulting in a near symmetric stress cycle.

Despite this, the fatigue safety factor remains above unity, indicating acceptable fatigue performance at 6000 RPM. However, the relatively high alternating stress suggests that the gudgeon pin is more sensitive to cyclic loading than the connecting rod.

Table 2 – Gudgeon Pin Performance at 6000 RPM

Parameter	Value
σ Max	250.04 MPa
σ Min	0.24 MPa
σ Mean	125.14 MPa
σ Alternating	124.90 MPa
Fatigue Safety Factor	2.483

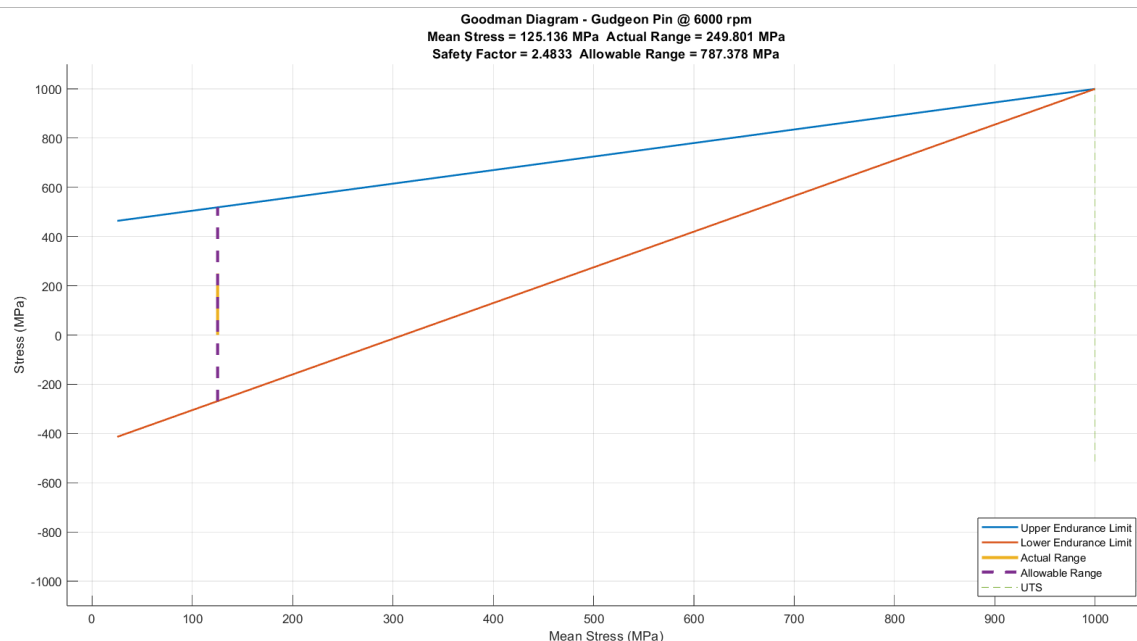


Figure 9 – Gudgeon Pin Goodman Diagram

3.3 Connecting Rod Bolt

The connecting rod bolt experienced the highest mean stress due to preload, with a maximum stress of 715.04 MPa and a fatigue safety factor of 1.561.

Unlike the other components, the bolt operates under high mean stress with relatively small alternating stress. This loading condition is characteristic of preload fasteners, where most of the stress is sustained rather than fluctuating. However, the high mean stress significantly reduces fatigue resistance when assessed using the Goodman relation.

The lower fatigue safety factor indicates that the bolt is the most critical component in the bottom end assembly.

Table 3 – Connecting Rod Bolt Performance at 6000 RPM

Parameter	Value
σ Max	715.04 MPa
σ Min	660 MPa
σ Mean	687.52 MPa
σ Alternating	27.52 MPa
Fatigue Safety Factor	1.561

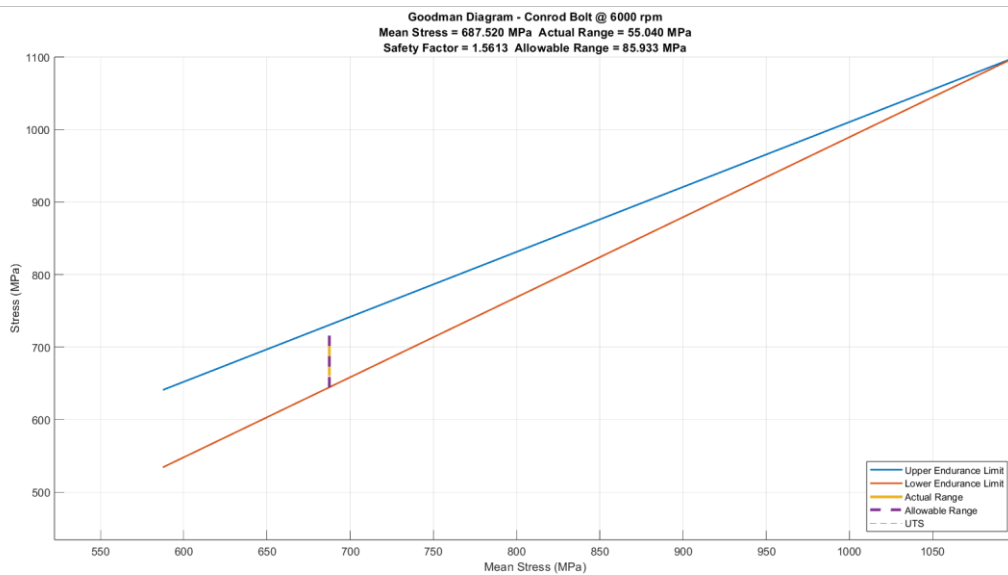


Figure 10 – Connecting Rod Bolt Goodman Diagram

3.4 Comparative Analysis

Comparing all three components, the connecting rod and gudgeon pin both operate with relatively high fatigue safety factors, indicating adequate performance under the loading conditions considered. In contrast, the conrod bolt exhibits a significantly lower fatigue safety factor due to its high mean stress resulting from preload.

This highlights the importance of considering both mean and alternating stress when assessing fatigue behaviour. While the gudgeon pin is dominated by alternating stress and bending, and the connecting rod by compressive loading from combustion pressure, the bolt is governed by sustained tensile loading. This makes the bolt the most fatigue-sensitive component in the assembly and explains why it becomes the focus of later redesign work.

4. Effect of Increasing Engine Speed

4.1 Overview

To assess the influence of engine speed on component loading, the in-cylinder pressure data was interpolated to 9000 RPM. This allowed the effect of increasing inertial forces on fatigue behaviour to be evaluated.

Piston acceleration is given by:

$$a \propto \omega^2$$

And therefore, inertia force:

$$F_{inertia} = ma \propto \omega^2$$

This quadratic relationship means that relatively small increases in engine speed produce significantly higher inertia forces. As noted in *Internal Combustion Engine Fundamentals*, piston acceleration becomes a dominant contributor to cyclic loading at high engine speeds, directly increasing fatigue demand on reciprocating components.

4.2 Results and Analysis

Figure 11 shows the variation of fatigue safety factors with engine speed for all components.

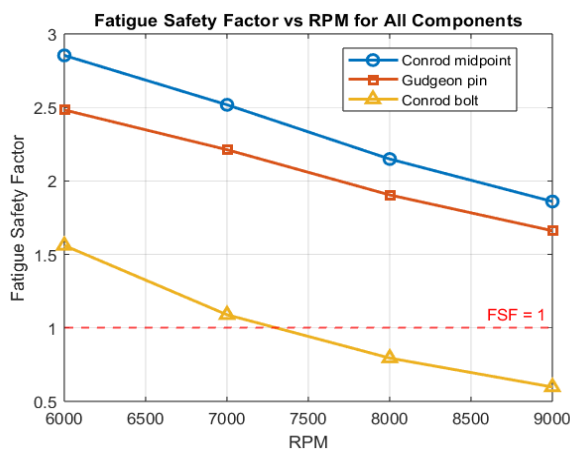


Figure 11 – FSF for all Components

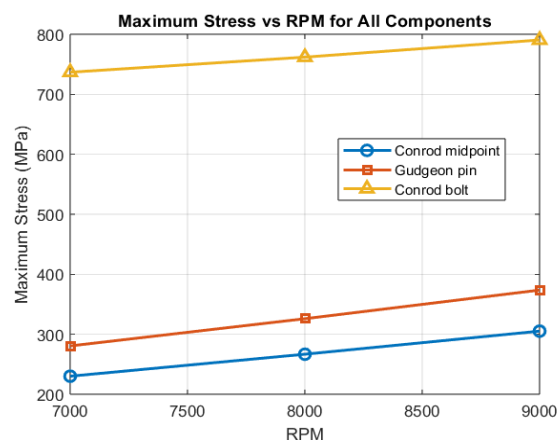


Figure 12 – Maximum Stress for all Components

The connecting rod and gudgeon pin show a gradual reduction in fatigue safety factor with increasing RPM. This behaviour is expected, as both components are primarily influenced by combustion loading, with inertia contributing increasingly at higher speeds.

In contrast, the conrod bolt exhibits a much steeper reduction in fatigue safety factor. This is due to its high mean stress from preload combined with increasing alternating stress from inertia-driven separating forces. At 9000 RPM, the fatigue safety factor drops to 0.6, indicating a high risk of failure. *ARP* emphasise the dangers of “necking”, where the rod bolt is preloaded beyond the materials yield point, leading to failure.

5. Conrod Bolt Redesign

5.1 Design Rationale

The results from section 4 identified the conrod bolt as the limiting component at high engine speeds. To improve fatigue performance, the bolt was redesigned using a reduced shank geometry.

According to *NORD-LOCK group*, bolts used in tension should be more elastic than the clamped members. Reducing the shank diameter increases bolt compliance, improving load distribution and reducing stress concentration in the threaded region.

Bolt stiffness is defined as:

$$k = \frac{E A}{L}$$

And for a stepped geometry:

$$\frac{1}{k_{bolt}} = \sum \frac{1}{k_i}$$

Reducing the shank area therefore lowers overall stiffness, increasing elastic deformation within the bolt.

Additionally, The *VDI2230 guidelines* states that the change in bolt load under external loading is governed by the stiffness ratio between the bolt and joint:

$$C = \frac{k_{bolt}}{k_{bolt} + k_{joint}}$$

A reduction in bolt stiffness decreases C , reducing the proportion of external load transferred to the bolt. This is particularly beneficial for fatigue, as it reduces the alternating stress experienced at the thread roots, which are the most critical locations for crack initiation.

Since preload is sensitive to friction, improving bolt compliance provides a more reliable method of controlling fatigue behaviour than relying solely on torque. Based on this, a wasted shank geometry was developed inspired by motorsport designs. (Table 4)

Table 4 – Geometry Comparison

Section	New Bolt	OEM bolt
D1	8.9	7.35
L1	3	60
D2	4	X
L2	18	X
D3	7.18	X
L3	6.26	x

5.2 Material Upgrade – 300M steel

A material review was conducted using *ANSYS Granta* to identify a suitable high-strength alloy. 300M steel was selected due to its significantly higher ultimate tensile strength compared to the OEM material.

The fatigue performance can be evaluated using the Goodman relation:

$$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{UTS}} = \frac{1}{FSF}$$

An increase in S_{UTS} and S_e therefore directly increases fatigue performance for a given loading condition.

The OEM bolt operates with a fatigue safety factor of approximately 1.5 at 6000 RPM, decreasing rapidly with rising engine speed. The wasted shank redesign improves fatigue performance by reducing alternating stress through increased compliance, resulting in a higher safety factor across the full RPM range.

The addition of 300M steel provides further improvement by increasing material strength limits. This results in the highest fatigue safety factor, maintaining a safe margin even at elevated engine speeds.

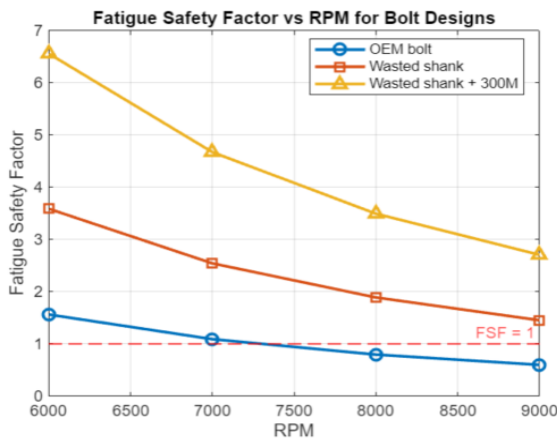


Figure 13 – Bolt FSF at 9000 RPM

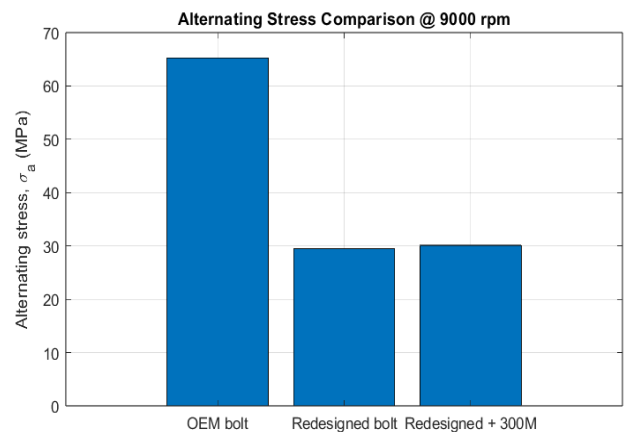


Figure 14 – Alternating Stress Comparison

Table 5 – Material Properties Comparison

Parameter	OEM Bolt	300M Bolt
Modulus of elasticity	200 GPa	210 GPa
UTS	1100 MPa	2030 MPa

5.3 Design Evaluation and Failure Consideration

The results show that fatigue behaviour in the conrod bolt is governed by a combination of high mean stress from preload and increasing alternating stress due to inertia forces. As engine speed increases, inertia-driven separating forces increase with ω^2 , amplifying the alternating stress component and rapidly reducing fatigue life.

The OEM bolt operates close to its fatigue limit at high RPM, indicating a high likelihood of failure. As noted by *ARP*, tensile overload failures in bolts are characterised by yielding and necking prior to fracture, suggesting that insufficient strength or excessive preload can lead to catastrophic failure.

The wasted shank redesign improves performance by:

- Increasing bolt compliance
- Reducing stress concentration at threads
- Lowering alternating stress

The material upgrade further improves fatigue resistance by increasing allowable stress limits.

Overall, the results show that geometric compliance is effective in reducing stress amplitude, while material strength increases the allowable stress envelope. The combination of both provides the most robust solution for high-speed operation.



Figure 15 – ARP Rod Bolt



Figure 16 – Ford OEM Connecting Rod Bolt, no date

6. Crankshaft Torque Analysis

6.1 Background

Instantaneous crankshaft torque varies continuously throughout the engine cycle due to the combined effects of combustion pressure and reciprocating inertia. *Rizzoni (1990)* states that engine torque is governed by the interaction of these forces, producing cyclic variations in crankshaft loading. Similarly, *Crankshaft Design and Optimisation: A review* highlights that both combustion and inertia effects contribute to time-varying torque and vibration, and that crankshaft configuration plays a key role in determining torque smoothness.

6.2 Methodology

The instantaneous crank torque was calculated by resolving piston forces to the crankshaft.

$$F_{gas}(\theta) = P(\theta)A_p$$

$$F_{inertia} = m_{rec}a(\theta)$$

These were converted into torque using slider-crank geometry:

$$T = F \cdot R(\sin\theta + \tan\phi \cos\theta)$$

Where R is crank radius and ϕ is the rod angle.

Total engine torque was obtained by summing four cylinders using crank phasing:

- Standard crank: 180° spacing
- Cross-plane crank: 90° spacing

6.3 Results and Analysis

Figure 17 shows the single cylinder torque components at 6000 RPM. The total torque is dominated by gas pressure near combustion, while inertia torque contributes significant oscillations throughout the cycle.

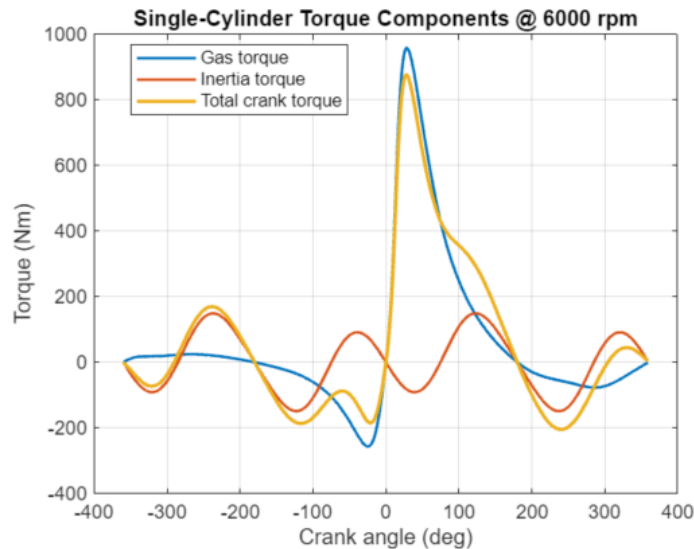


Figure 17 – Single Cylinder Torque Components

Figure 18 compares 4-cylinder gas torque. The standard crank produces evenly spaced torque pulses due to uniform firing intervals, whereas the cross-plane crank exhibits a more uneven distribution.

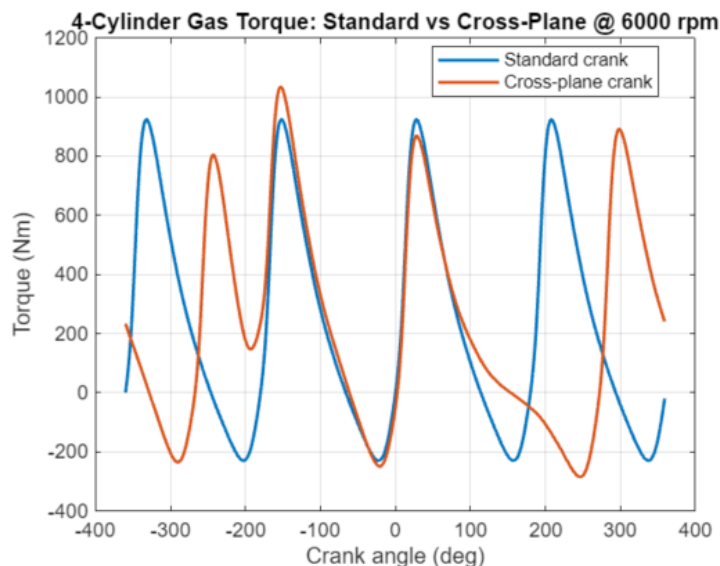


Figure 18 – Gas Torque Comparison

Figure 19 shows the inertia torque comparison. The standard crank exhibits large oscillations due to the 2-up, 2-down piston motion. In contrast, the cross-plane crank significantly reduces inertia torque fluctuations by distributing piston motion across 90° phase differences, improving cancellation over the 720° cycle.

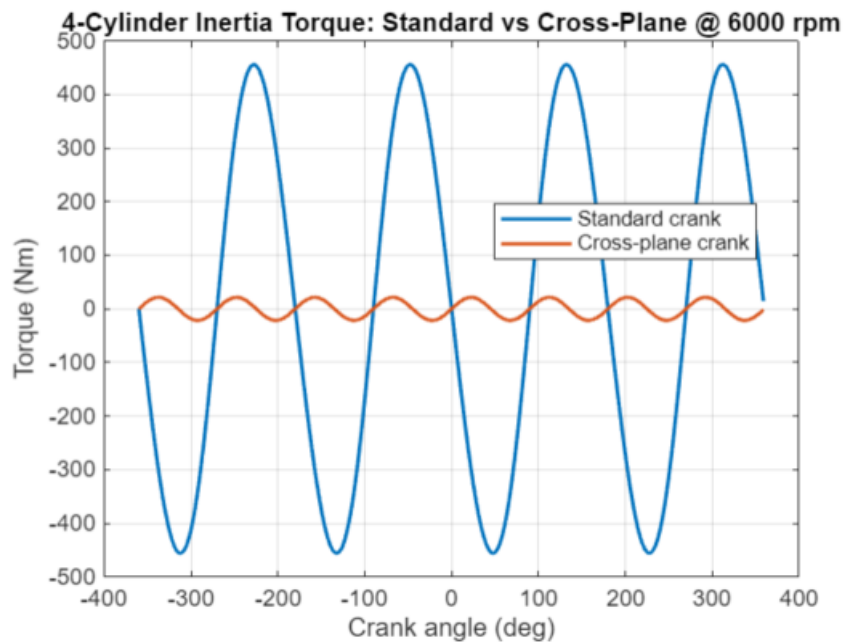


Figure 19 – Inertia Torque Comparison

Figure 20 presents total instantaneous crank torque. Despite reduced inertia effects, the cross-plane crank produces larger overall torque variation due to its uneven gas torque contribution.

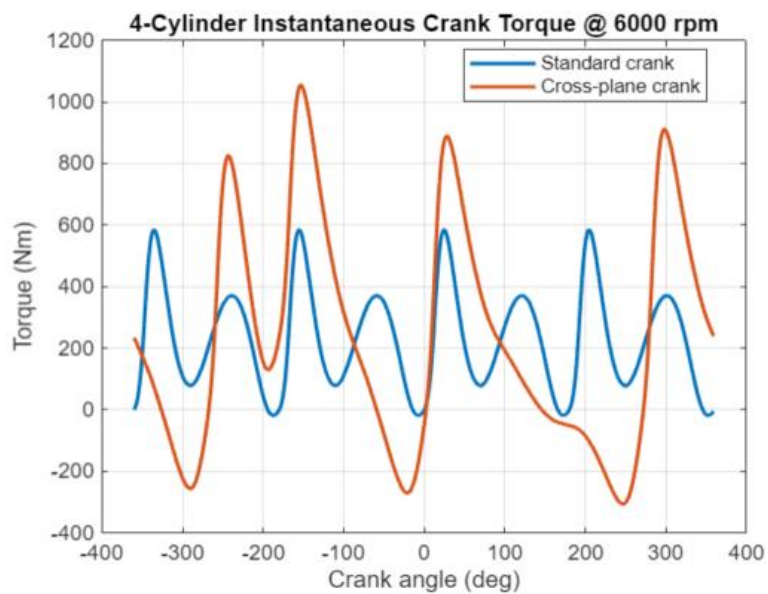


Figure 20 – Instantaneous Torque Comparison

This is further quantified in *Figure 21*, where the cross-plane configuration shows:

- Higher gas torque range
- Significantly lower inertia torque range
- Higher total torque range

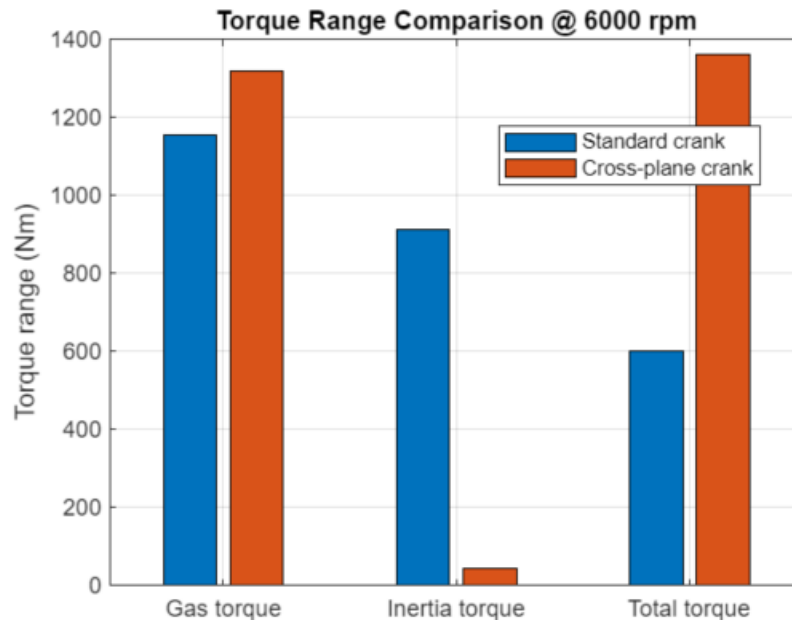


Figure 21 – Torque Range Comparison

6.4 Discussion

The reduction in inertia torque for the cross-plane crankshaft is consistent with improved cancellation of primary and secondary inertia forces:

$$F_{primary} \propto \cos\theta, F_{secondary} \propto \cos(2\theta)$$

Distributing crank throws 90° reduces the alignment of these forces, resulting in smoother dynamic behaviour.

However, as noted in *Crankshaft Design And Optimisation: A Review*, improvements in dynamic balance can come at the expense of torque uniformity. This is evident in the results, where the cross-plane crank produces less uniform gas torque and a higher total torque range.

This demonstrates a key engineering trade-off:

Cross-plane crankshafts improve inertia balance and reduce vibration, but at the cost of increased torque fluctuation due to uneven firing intervals.

6.5 Balance Shaft Considerations

Flat-plane crankshafts are known to generate higher vibration levels due to unbalanced inertia forces. One method of mitigating this is the use of balance shafts.

The study by *Kwon-Hee Suh* shows that engine vibration originates from combustion and inertia forces, which generate unbalanced moments. Balance shafts counteract these forces, improving NVH performance.

In contrast, cross-plane crankshafts inherently reduce these imbalances through crank phasing, reducing the need for additional balancing mechanisms.

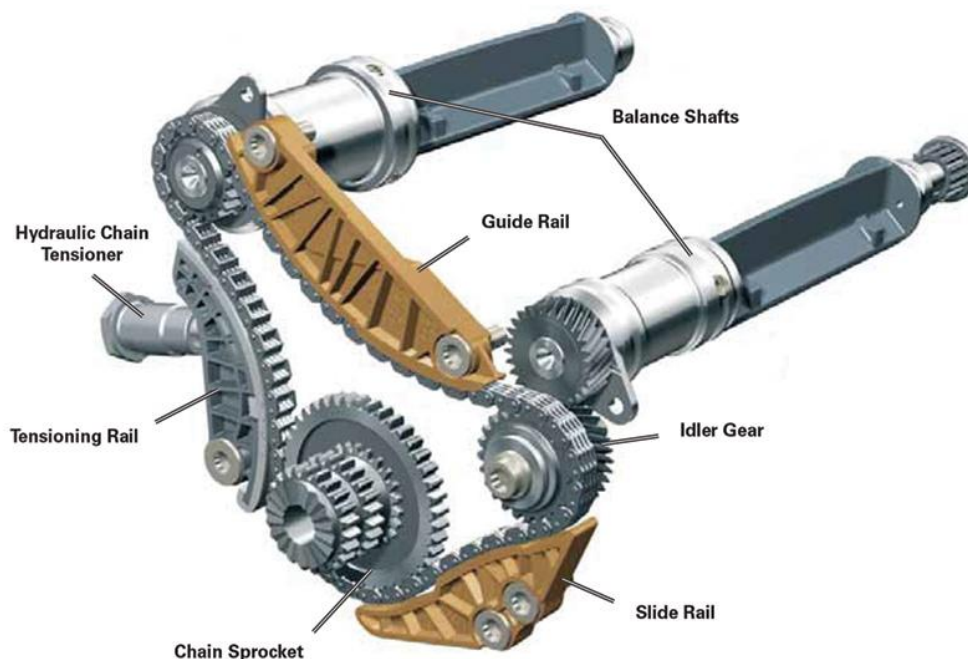


Figure 22 – VWAG Balance Shaft Assembly, 2020

7. Conclusion

This study evaluated the dynamic loading and fatigue behaviour of key engine bottom-end components under realistic operating conditions. Results showed that loading is governed by both combustion pressure and reciprocating inertia, with inertia forces increasing with the square of engine speed and becoming dominant at higher RPM.

At 6000 RPM, the connecting rod and gudgeon pin operated with fatigue safety factors of 2.854 and 2.483 respectively, indicating acceptable performance. In contrast, the conrod bolt was identified as the critical component, with a fatigue safety factor of 1.561, reducing to 0.6 at 9000 RPM, indicating high likelihood of fatigue failure

A redesigned bolt incorporating a wasted shank improved fatigue performance by increasing compliance and reducing alternating stress. The addition of 300M steel further increased the fatigue safety factor to above 2.5 at 9000 RPM, restoring a safe operating margin.

Crankshaft torque analysis at 6000 RPM showed that the cross-plane configuration reduced inertia torque fluctuations (43 Nm vs 912 Nm) but produced a higher total instantaneous torque range (1360 Nm vs 600 Nm) due to uneven firing intervals.

Overall, the results demonstrate that fatigue performance is governed by the interaction of inertia-driven loading and stress concentration effects. Effective design improvements can be achieved through combined geometric compliance and material selection, while crankshaft configuration presents a trade-off between dynamic balance and torque smoothness.

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