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Contemporary Powertrain

ACHA6006

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2205525

12/03/2026

Contents

1. Introduction.....	3
2. Methodology	4
2.1. Engine Geometry and Input Parameters	4
2.2. Piston-Crank Kinematics	5
2.3 Force Model	6
3. Structural Analysis of Engine Components	7
3.1. Connecting Rod Midpoint Stress.....	7
3.2. Gudgeon Pin Stress Analysis	8
3.3. Connecting Rod Bolt Analysis.....	9
4. High RPM Analysis	11
4.1. Pressure Data Interpolation	11
4.2. Bolt Fatigue Safety Factor Across RPM Range	11
5. Connecting Rod Bolt Redesign	13
5.1. Design Requirements	13
5.2. Optimised Bolt Design.....	14
5.3. Discussion of Structural and Bolt Findings.....	15
6. Instantaneous Crankshaft Torque Analysis	16
6.1. Torque Calculation Method	16
6.2. Standard Crankshaft Configuration.....	17
6.3. Cross-Plane Crankshaft Configuration	18
6.4. Crankshaft Comparison.....	19
6.5. Discussion of crankshaft comparison.....	20
7. Conclusions.....	21
8. References	22

List of Figures

Figure 1 Piston velocity Vs Crank angle	5
Figure 2 Piston acceleration Vs Crank angle.....	5
Figure 3 Piston side force Vs Crank angle.....	6
Figure 4 Gas force Vs Crank Angle	6
Figure 5 Inertia force Vs Crank angle.....	6
Figure 6 Connecting rod Goodman diagram	7
Figure 7 Gudgeon pin Goodman diagram	8
Figure 8 Connecting rod bolt stress Vs Crank angle	9
Figure 9 Connecting rod bolt Goodman diagram	10
Figure 10 9000rpm Connecting rod bolt Goodman diagram	12
Figure 11 9000rpm interpolation Connecting rod bolt Vs Crank angle.....	12
Figure 12 Optimised bolt Goodman diagram	14
Figure 13 Single Cylinder Instantaneous Torque.....	16
Figure 14 Standard Crankshaft Torque overlay.....	17
Figure 15 Cross-plane Crankshaft torque overlay	18
Figure 16 Standard Vs Cross-plane Instantaneous Torque Comparison	19
Figure 17 Standard Vs Cross-plane Instantaneous Torque Difference	20

List of Tables

Table 1 Engine geometry & masses.....	4
Table 2 Engine material properties	4
Table 3 Connecting rod stress results	7
Table 4 Gudgeon Pin Stress Results	8
Table 5 Connecting rod bolt parameters and Stress results	9
Table 6 9000rpm bolt stress results	11
Table 7 Redesigned connecting bolt parameters and results	13
Table 8 Single Cylinder Torque Results	16
Table 9 Standard Crankshaft Torque	17
Table 10 Cross-Plane Torque Results	18
Table 11 Crankshaft Comparison	19

1. Introduction

This report examines the bottom-end behaviour of the Ford 1.6 Sigma engine using a MATLAB-based analytical model and 720-point crank-angle pressure data from 1000 to 6000 rpm, with the dataset extended to 9000 rpm by interpolation. The focus of the study is on the connecting rod, gudgeon pin and connecting rod bolts, as these components are subjected to repeated gas and inertia loading and are therefore vulnerable to fatigue, particularly as engine speed increases (Lawton and Crutcher, 1966; Strozzi et al., 2016). Piston-crank kinematics were used to determine the piston and rod loading through the cycle. These loads were converted into stresses and assessed based on mean stress, stress range and Goodman theory, with stress concentration factors included for the bolt. The analysis then considers whether the original bolt design remains suitable at higher speed and evaluates potential redesign options. As the model is intended as a first-stage mechanical assessment, the results should be interpreted as analytical design indicators rather than full durability predictions, since factors such as lubrication, thermal effects, material variation, wear and local plasticity are not included.

2. Methodology

2.1. Engine Geometry and Input Parameters

The analysis used a 720-point crank-angle pressure dataset covering 1000 to 6000 rpm, with all geometry, mass and material inputs defined in SI units. The principal inputs are given in Tables 1 and 2, with additional dimensions included later for the gudgeon pin and bolt calculations. Together, these values provided the basis for the kinematic, force and stress analysis (Lawton and Crutcher, 1966; Cardoso et al., 2020).

Table 1 Engine geometry & masses

Parameter	Value
Bore (mm)	79
Crank radius (mm)	40.7
Connecting rod length (mm)	134
Piston area (mm ²)	4901.7
Reciprocating mass (g)	347
Rotational mass (g)	313
Piston (g)	268
Rings (g)	9
Circlips (g)	5
Gudgeon pin (g)	28

Table 2 Engine material properties

Component	UTS	Endurance fraction
Connecting rod	1000 MPa	0.45
Gudgeon pin	1000 MPa	0.45
Bolt	1100 MPa	0.45

2.2. Piston-Crank Kinematics

Piston-crank relations were used to calculate rod angle, piston displacement, velocity and acceleration throughout the cycle. These quantities define the reciprocating motion of the assembly and thus the inertia loading that increases with engine speed.

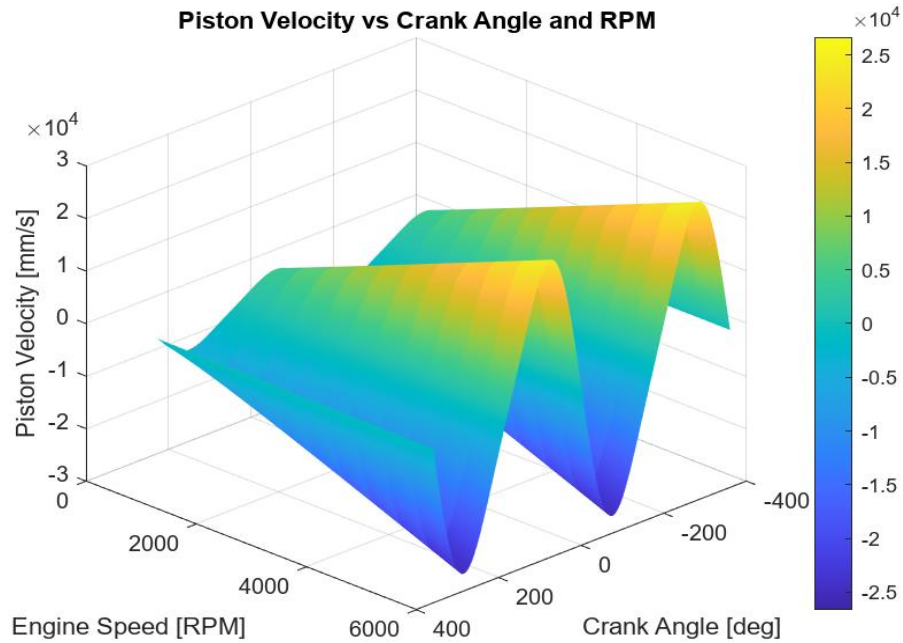


Figure 1 Piston velocity Vs Crank angle

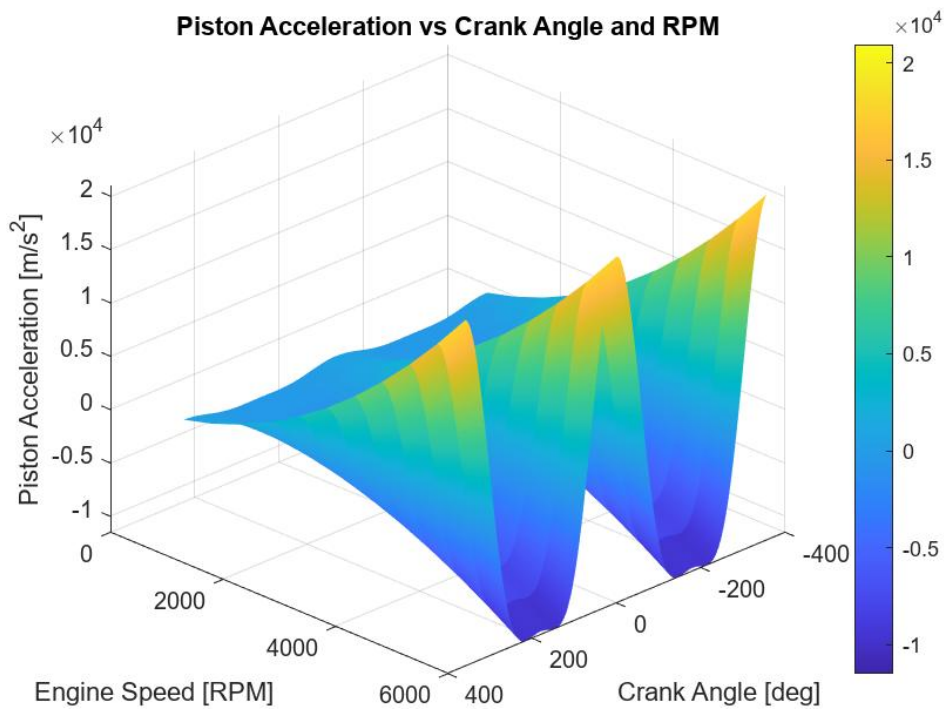


Figure 2 Piston acceleration Vs Crank angle

2.3 Force Model

Gas force was calculated from cylinder pressure and piston area, while inertia force was obtained from the reciprocating mass and piston acceleration. These were then combined to determine net piston force, rod force and piston side force. The results showed that combustion pressure dominated the compressive loading, while inertia became more significant in the tensile parts of the cycle near TDC. This distinction is important because it explains why different components become critical under different operating conditions, which is consistent with work by Lawton and Crutcher (1966). Gas, inertia and net forces can be seen in figures 3, 4 and 5.

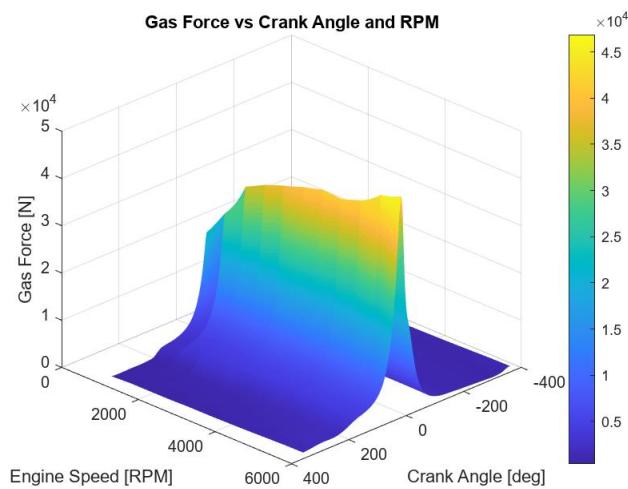


Figure 3 Gas force Vs Crank Angle

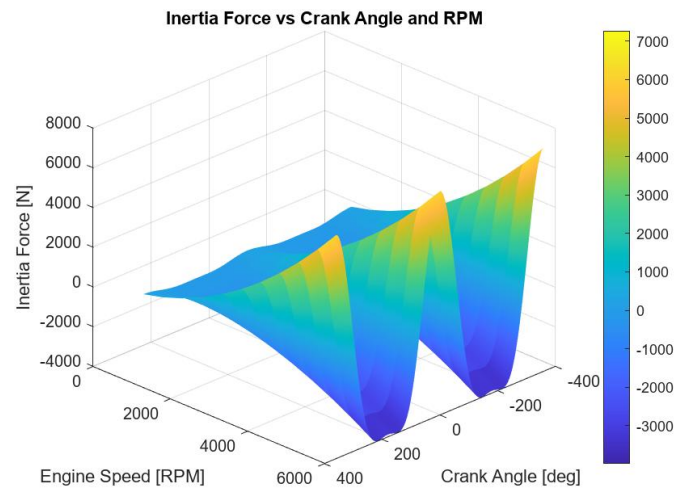


Figure 4 Inertia force Vs Crank angle

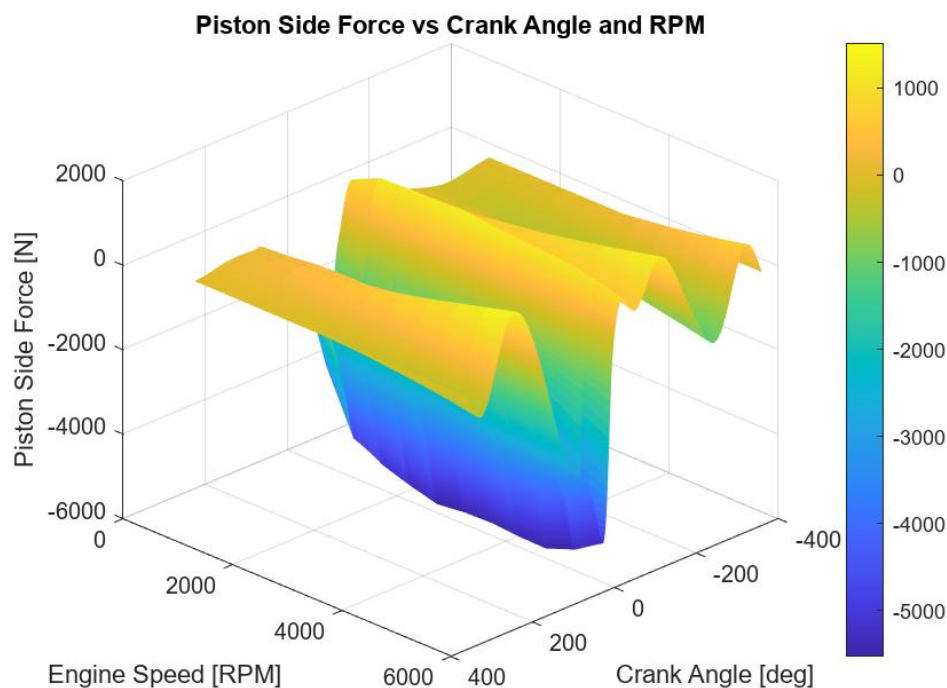


Figure 5 Piston side force Vs Crank angle

3. Structural Analysis of Engine Components

3.1. Connecting Rod Midpoint stress

The connecting rod midpoint stress was calculated from the rod force and the specified cross-sectional area. The results showed this section remained predominantly compression loaded, with a relatively large fatigue margin at 6000 rpm of 4.14. This indicates that the mid-shank is not the critical feature in the present model, as the loading is well within the capacity of the section. This is consistent with the literature, where connecting rod failures are more commonly associated with local stress concentrations at the small end, big end or cap interface rather than the central shank region (Rabb, 1996; Strozzi et al., 2016).

Table 3 Connecting rod stress results

Parameter	Value
Maximum stress (MPa)	30.292
Minimum stress (MPa)	-205.977
Mean stress (MPa)	-87.843
Stress range (MPa)	236.268
Allowable range (MPa)	979.058
Fatigue safety factor	4.1438

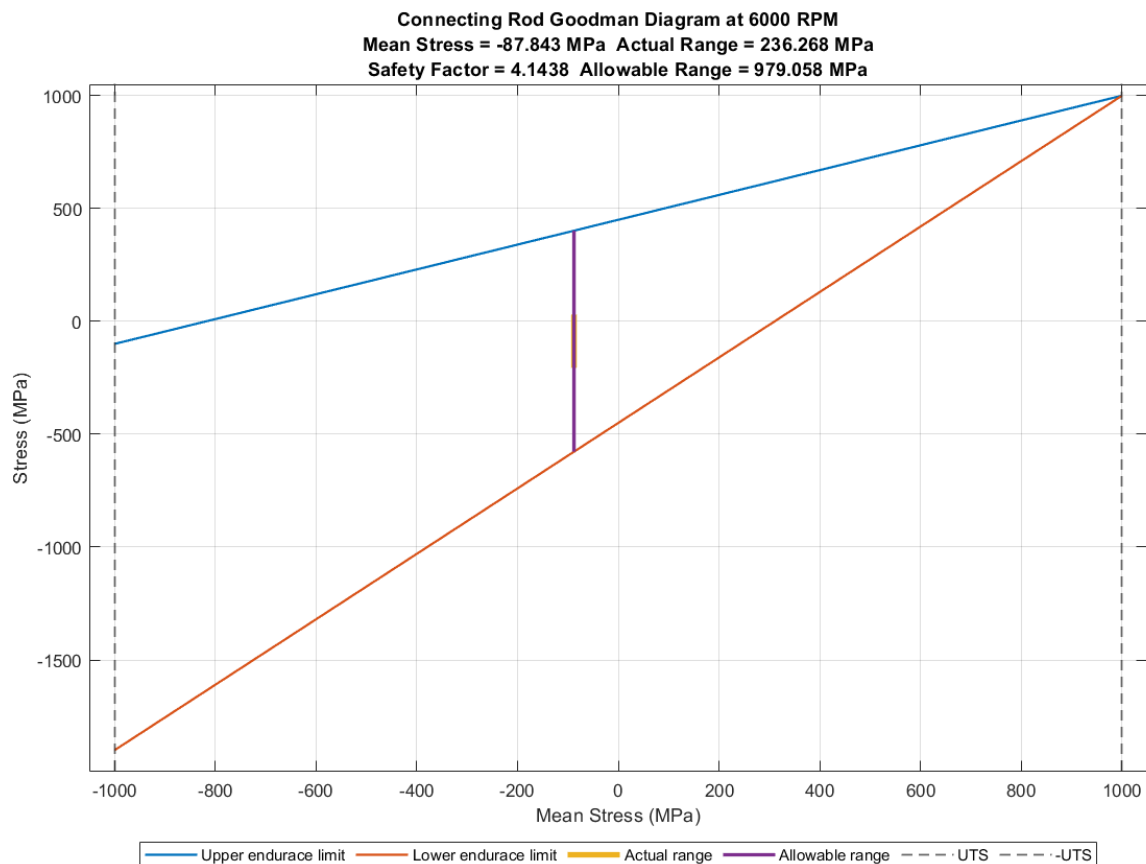


Figure 3 Connecting rod Goodman diagram

3.2. Gudgeon Pin Stress Analysis

The gudgeon pin transfers load between the piston and the connecting rod small end throughout the cycle. In this study, its stress was assessed by combining bending and shear, with the loading model accounting for exposed pin mass, piston-side inertia and net piston force. The results showed that the pin operated with a lower fatigue margin than the connecting rod midpoint but remained structurally adequate within the analysed speed range. This suggests that the pin is more sensitive to cyclic loading than the rod shank, although it does not become the limiting component in the present model.

Table 4 Gudgeon Pin Stress Results

Parameter	Value
Maximum stress (MPa)	257.397
Minimum stress (MPa)	0.152
Mean stress (MPa)	128.774
Stress range (MPa)	257.245
Allowable range (MPa)	784.103
Fatigue safety factor	3.0481
Max bending stress (MPa)	32.986
Max shear stress (MPa)	15.213

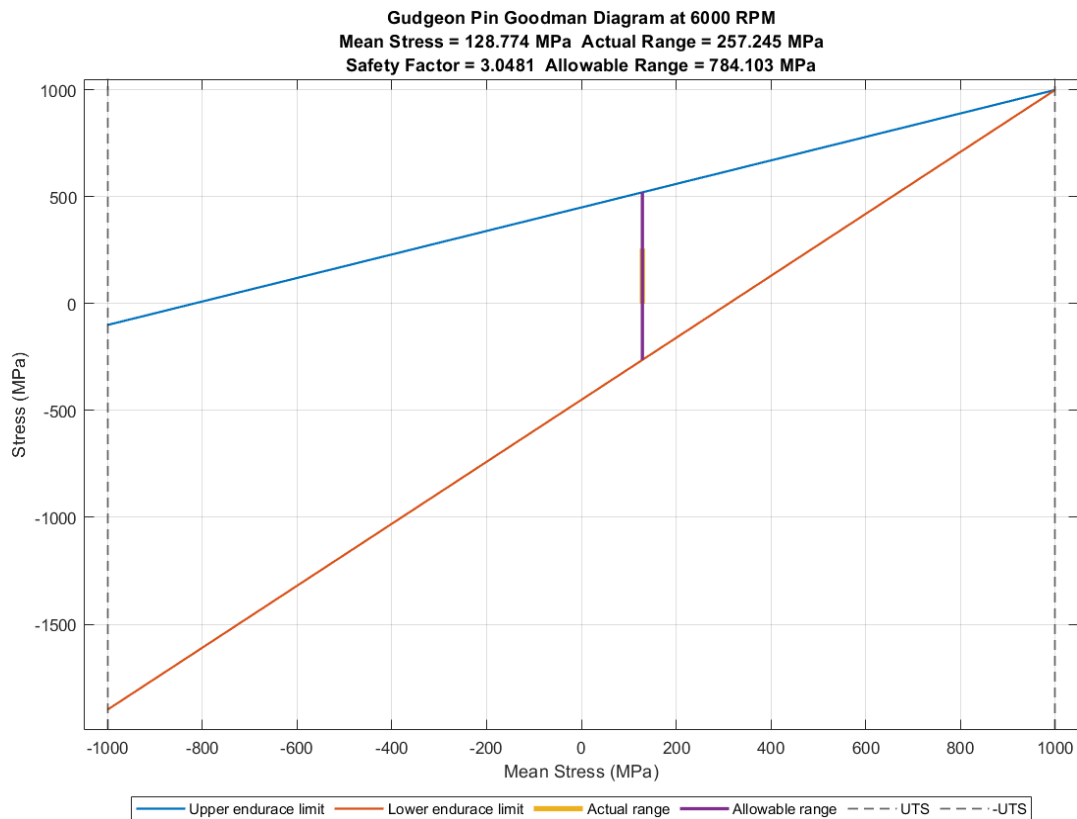


Figure 4 Gudgeon pin Goodman diagram

3.3. Connecting Rod Bolt Analysis

The connecting rod bolts were assessed separately because their behaviour is governed by joint mechanics as well as the applied separating load. As discussed by Benhaddou et al. (2014), bolt response depends not only on the separating force, but also on preload, fastener stiffness, cap stiffness and the way load is shared through the joint. In this case, preload provides the clamping force that resists cap separation, while the stiffness ratio controls how much of the operating load is transferred into the bolt itself. This is why the bolt operates at a high mean stress even when the fluctuating component is relatively low. The analysis showed that the bolt remained acceptable at 6000 rpm, with a fatigue margin similar to the gudgeon pin and connecting rod once stress concentrations were included. This is not fully consistent with literature, which often identifies bolts as fatigue-critical because their behaviour is highly sensitive to preload condition, joint stiffness and thread-root severity rather than peak load alone (Chalupnik, 1968; Pedersen, 2013).

Table 5 Connecting rod bolt parameters and stress results

Parameter	Value
Bolt UTS (MPa)	1100
Yield strength (MPa)	990
Preload stress (MPa)	534.6
Preload force (N)	22683.08
Bolt stiffness (N/m)	3.113E-08
Cap stiffness (N/m)	1.152E-09
Load-sharing factor	0.21279
Maximum stress (MPa)	565.439
Minimum stress (MPa)	534.6
Mean stress (MPa)	550.019
Stress range (MPa)	30.839
Alternating stress (MPa)	15.419
Allowable range (MPa)	110.264
Fatigue safety factor	3.5755

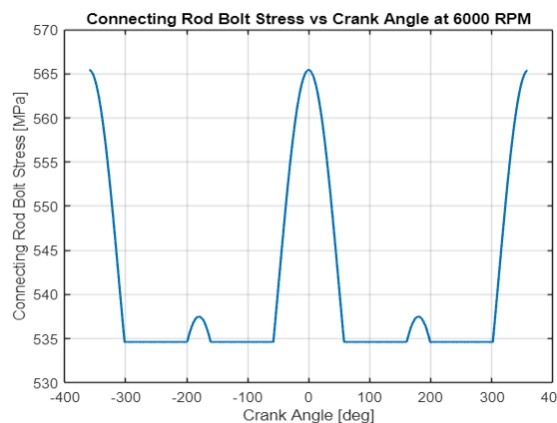


Figure 5 Connecting rod bolt stress Vs Crank angle

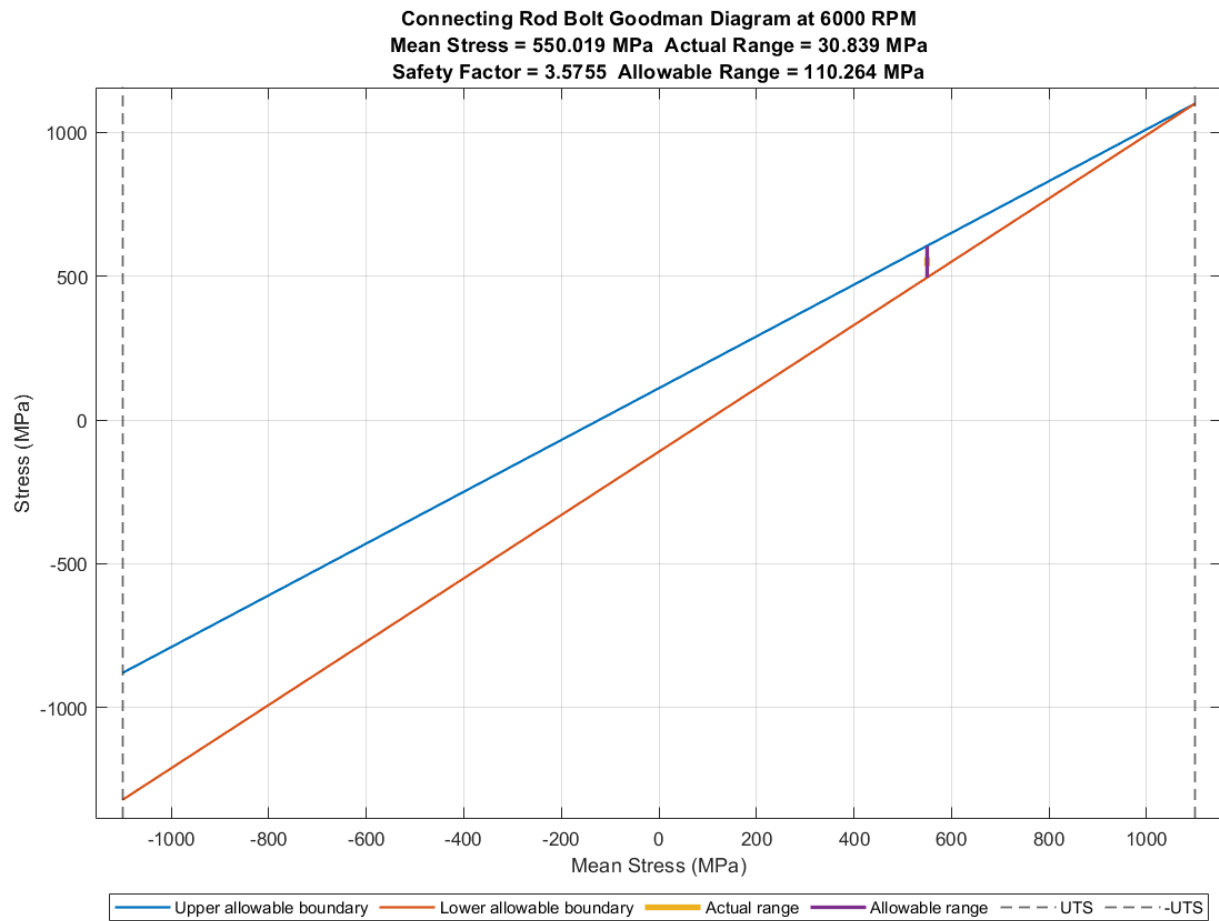


Figure 6 Connecting rod bolt Goodman diagram

4. High RPM Analysis

4.1. Pressure Data Interpolation

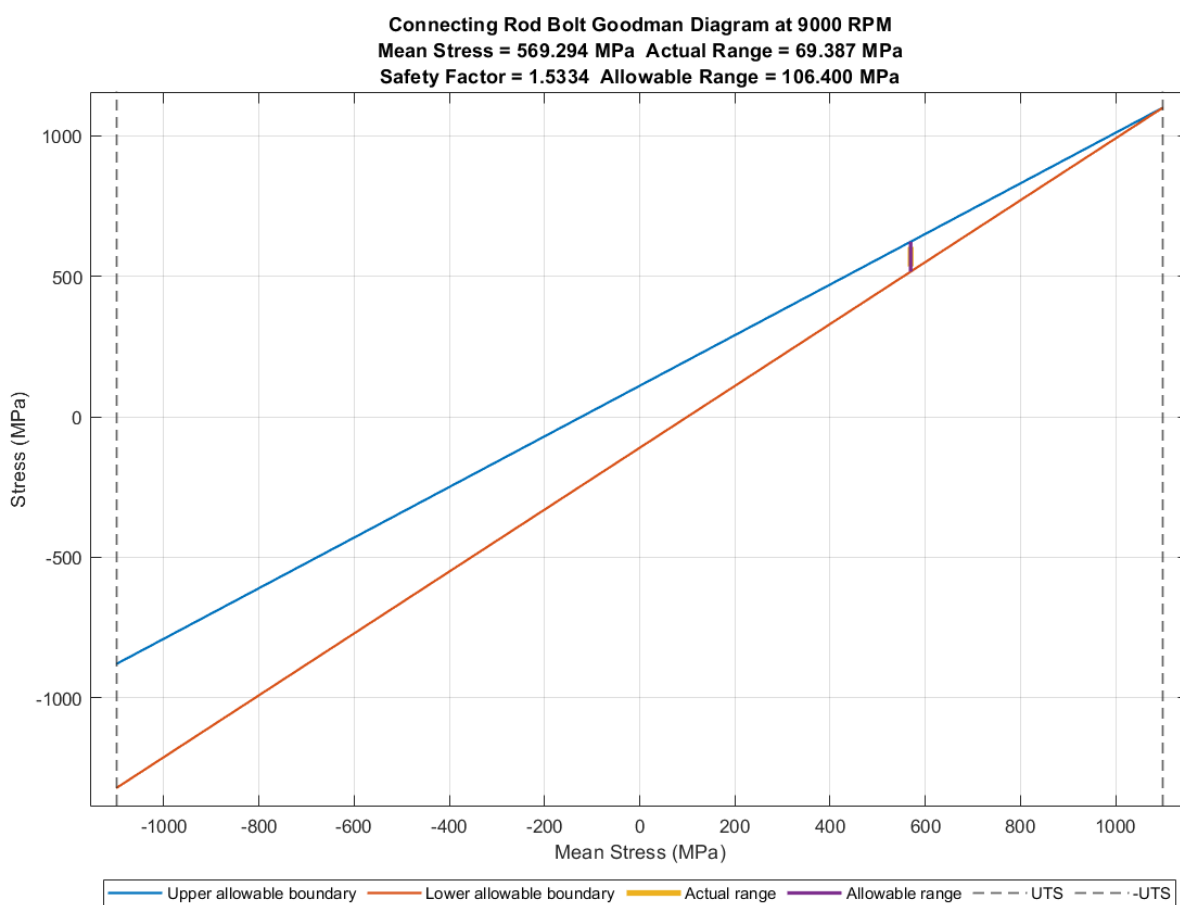
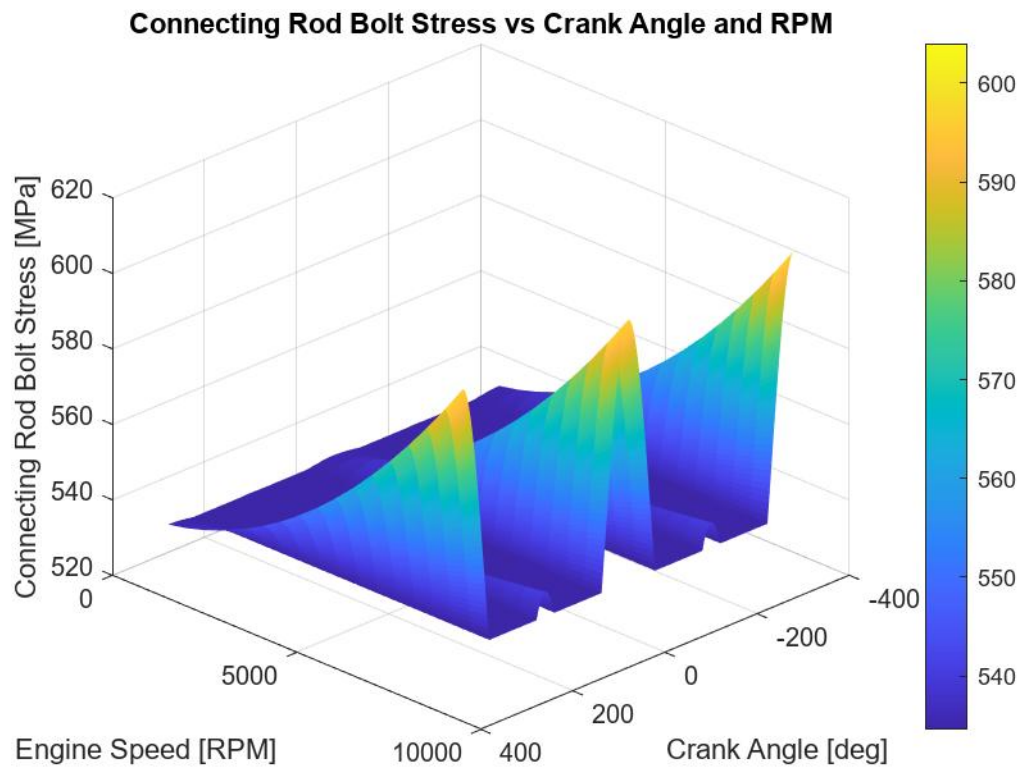
The pressure data was extended to 9000 rpm so that the effect of higher engine speed on component loading could be examined. This is important because increasing rpm raises inertia loading sharply, making the tensile parts of the cycle more severe even though the engine dimensions remain unchanged. The interpolation therefore allowed the model to assess how fatigue margin changed at higher speed without altering the underlying mechanical layout. However, the method assumes that pressure variation between the measured speeds is smooth, so the extended results should be treated as comparative design indicators rather than a direct substitute for measured high-rpm data. This is an important limitation, as literature consistently shows that high-speed engine behaviour is sensitive to loading conditions (Benhaddou et al., 2014).

4.2. Bolt Fatigue Safety Factor Across RPM Range

When the force model was extended to 9000 rpm, both gas and inertia loading increased, but the rise in inertia was particularly important because it increased the cap-separating load carried by the bolts. The calculations therefore showed a clear reduction in bolt fatigue margin, while the connecting rod and gudgeon pin remained within acceptable limits. However, the factor of safety did not fall below 1.0 as expected from the assignment context, and this discrepancy should be acknowledged rather than overlooked. It suggests that one or more parts of the bolt model may be under-predicting the true severity of the high-speed condition. Possible causes include holding preload constant, simplifying the cap-separating load, a thread-stress treatment that is not sufficiently conservative or a possible error in the MATLAB script. The interpolated pressure trace may also contribute to this outcome, since any smoothing of the 9000 rpm combustion profile is likely to reduce the peak loads seen by the bolt. As seen in research, fatigue predictions in highly loaded fasteners are very sensitive to modelling assumptions and local stress treatment (Li et al., 2022). The overall trend is still credible, with the bolt becoming more critical as rpm rises, but the absolute 9000 rpm safety factor should be treated with caution and may be optimistic.

Table 6 9000rpm bolt stress results

Parameter	Value
Maximum stress (MPa)	603.987
Minimum stress (MPa)	534.6
Mean stress (MPa)	569.294
Stress range (MPa)	69.387
Allowable range (MPa)	106.4
Fatigue safety factor	1.5334



5. Connecting Rod Bolt Redesign

5.1. Design Requirements

The redesign objective was to recover as much of the original 6000 rpm fatigue margin as possible while staying within realistic geometric and packaging constraints. In practical terms, this required an improvement in fatigue performance without introducing excessive mass or an impractical bolt form. Because the bolts operate at high mean stress, the most important changes were those affecting stiffness and stress concentration factors. The redesign was therefore approached as a trade-off between fatigue improvement, mass and packaging. Three upgrade concepts were examined:

1. reduced shank diameter
2. increased shank length to thread length ratio
3. material upgrade.

The first two concepts were intended to alter bolt stiffness and therefore reduce the proportion of separating load carried by the fastener. This reflects the wider literature, where bolt compliance and joint load sharing are recognised as major influences on fatigue behaviour (Benhaddou et al., 2014; Pedersen, 2013). By contrast, the material upgrade increased strength and endurance capability, but did not change the applied stress itself, so the mean stress penalty remained largely unchanged. The results showed that stiffness-cantered geometry changes were more effective than material substitution alone, which is consistent with bolted-joint fatigue literature (Chalupnik, 1968; Pedersen, 2013; Benhaddou et al., 2014).

Table 7 Redesigned connecting bolt parameters and results

Design	Mass (g)	Thread	Shank Diameter (mm)	Shank Length (mm)	Max Stress (MPa)	FSF @9000 RPM	FSF Change
Original Bolt	7.835	M8×0.75	7.35	21.26	603.99	1.5334	-
Upgrade 1 - Reduced Shank	8.213	M8×0.75	6.5	14.26	481.08	2.0706	0.3503
Upgrade 2 - Longer Shank	11.347	M8×0.75	7.8	18.26	601.1	1.6045	0.0463
Upgrade 3 - Material Upgrade	9.847	M8×0.75	7.8	14.26	731.26	1.7506	0.1417

5.2. Optimised Bolt Design

Upgrade 1, the reduced shank diameter design, gave the strongest overall result, improving fatigue margin without a substantial mass penalty. This suggests that bolt performance in the present model is influenced more by stiffness and joint load sharing than by material strength alone. Upgrade 2 produced only a modest improvement despite the additional mass, indicating that increasing shank length was less effective than directly reducing shank stiffness. Upgrade 3 was also less effective, showing that a stronger material cannot fully compensate for an unfavourable loading condition if the joint still transfers too much of the separating load into the bolt. This trend is consistent with the literature, where improved bolt compliance is associated with lower load concentration in the threaded region, and better fatigue performance in highly stressed joints (Pedersen, 2013; Benhaddou et al., 2014). The reduced-shank result is therefore mechanically credible, as a more compliant bolt attracts a smaller share of the separating load, and reduces the stress range at the thread root. Even so, the result should be treated cautiously. If the baseline 9000 rpm model is already under-predicting bolt severity, the absolute improvement may also be optimistic. There is also a practical trade-off, since reducing shank diameter improves compliance but may increase sensitivity to local stress if taken too far. Upgrade 1 is therefore the preferred redesign within the scope of this analysis, but it should be regarded as the most effective option rather than a fully validated final design.

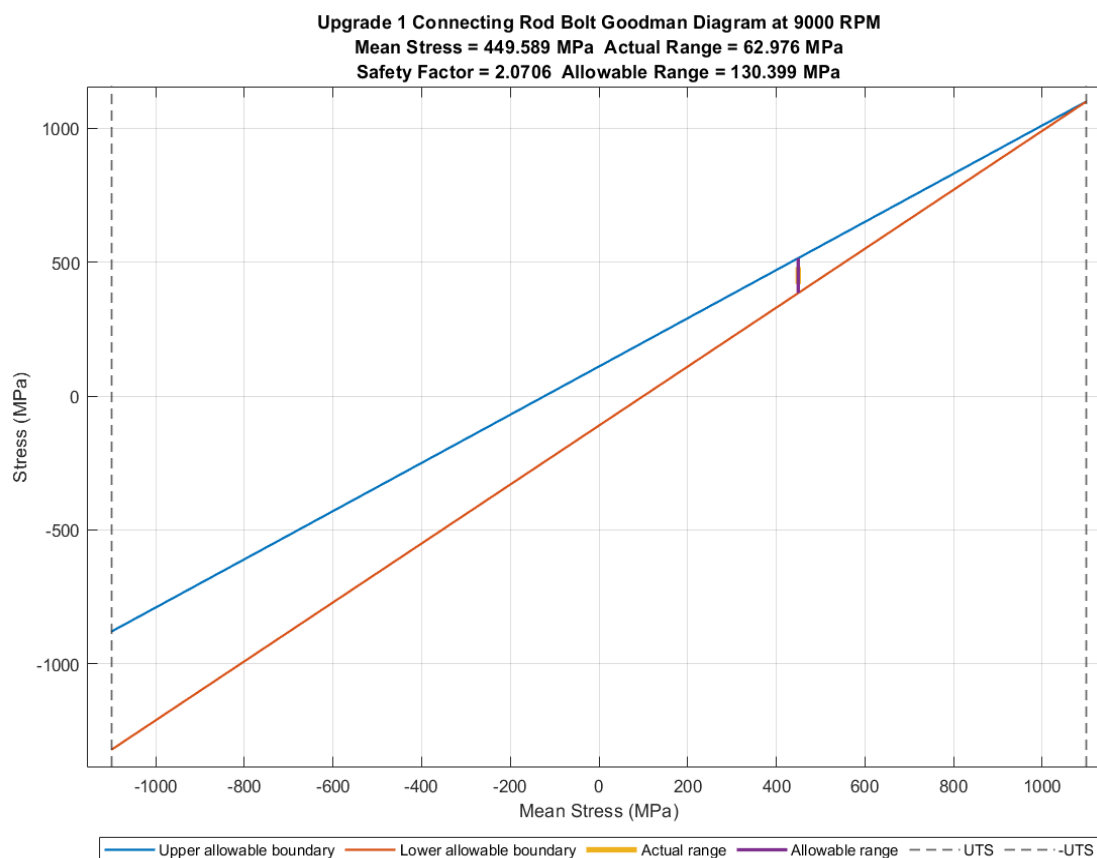


Figure 7 Optimised bolt Goodman diagram

5.3. Discussion of Structural and Bolt Findings

The analytical model showed a clear difference in how the component safety margins changed as engine speed increased. At 6000 rpm, the connecting rod, gudgeon pin and connecting rod bolt all remained within acceptable fatigue limits, with the bolt showing a safety margin that was still comparable with the other assessed parts. At 9000 rpm, however, the bolt safety factor reduced markedly, yet remained above unity when the expected behaviour suggested failure was more likely. This weakens confidence in the absolute value of the predicted margin and indicates that the model is likely under-predicting in the most demanding operating condition. In practical terms, this means the bolt may be closer to its true fatigue limit than the analytical result suggests, so the reported reserve of strength should be treated cautiously. The result is still useful because it identifies the correct trend and confirms that bolt behaviour becomes increasingly important at high speed, but it also shows that the design judgement should not rely on the numerical safety factor alone. Even so, the redesign study demonstrated that stiffness-based optimisation was more effective than material upgrade alone, with the reduced-shank diameter concept giving the most convincing improvement in fatigue performance.

6. Instantaneous Crankshaft Torque Analysis

6.1. Torque Calculation Method

Instantaneous crank torque was determined from the single-cylinder gas-force and inertia-force components at 6000 rpm. Gas torque represents the useful turning effect generated by combustion pressure acting through the crank, while inertia torque represents the opposing effect of accelerating the reciprocating mass. These components were combined to produce the net single-cylinder torque trace over the full 720° cycle. This approach is widely used in the literature because it captures how torque is delivered through the cycle rather than reducing it to a single mean value, which is important when assessing smoothness and torsional behaviour (Cardoso et al., 2020; Pfabe and Woernle, 2016). The resulting torque traces can be seen in figure 13.

Table 8 Single Cylinder Torque Results

Parameter	Value
Max gas torque (Nm)	957.864
Min gas torque (Nm)	-256.572
Max inertia torque (Nm)	148.917
Min inertia torque (Nm)	-148.917
Max crank torque (Nm)	876.095
Min crank torque (Nm)	-204.862
Mean torque (Nm)	57.455

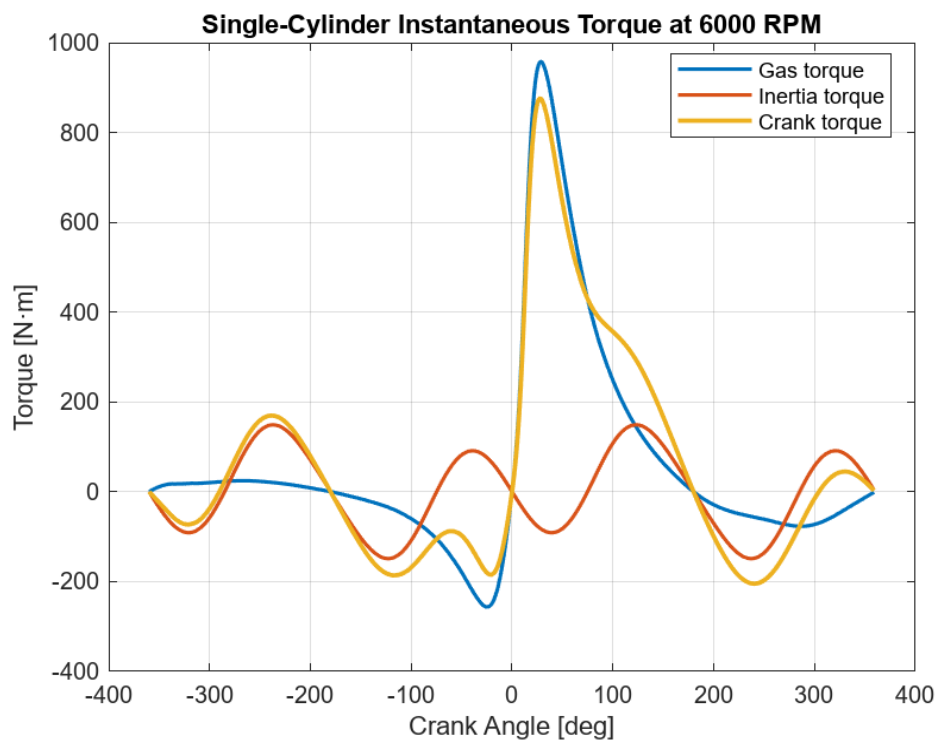


Figure 8 Single Cylinder Instantaneous Torque

6.2. Standard Crankshaft Configuration

The standard crankshaft case used cylinder phase offsets of 0, 180, 360 and 540 degrees before summing the individual cylinder torque traces. This produced a total engine torque trace with a clear repeating pattern across the cycle, where the individual cylinder peaks combine to give four strong positive torque pulses. Between these peaks, the total torque falls but remains relatively controlled, with only a small negative region near the end of the cycle. Although each cylinder produces a highly unsteady torque contribution, the phase relationship between them smooths the combined response and distributes the variation more evenly over the 720 degree cycle. This indicates that the standard crank arrangement provides a stable overall torque delivery, with the summed response governed by the overlap of individual firing events rather than isolated cylinder peaks.

Table 9 Standard Crankshaft Torque

Parameter	Value
Max torque (Nm)	583.798
Min torque (Nm)	-18.226
Mean torque (Nm)	229.819
Torque range (Nm)	602.024
Torque ripple ratio	2.6196

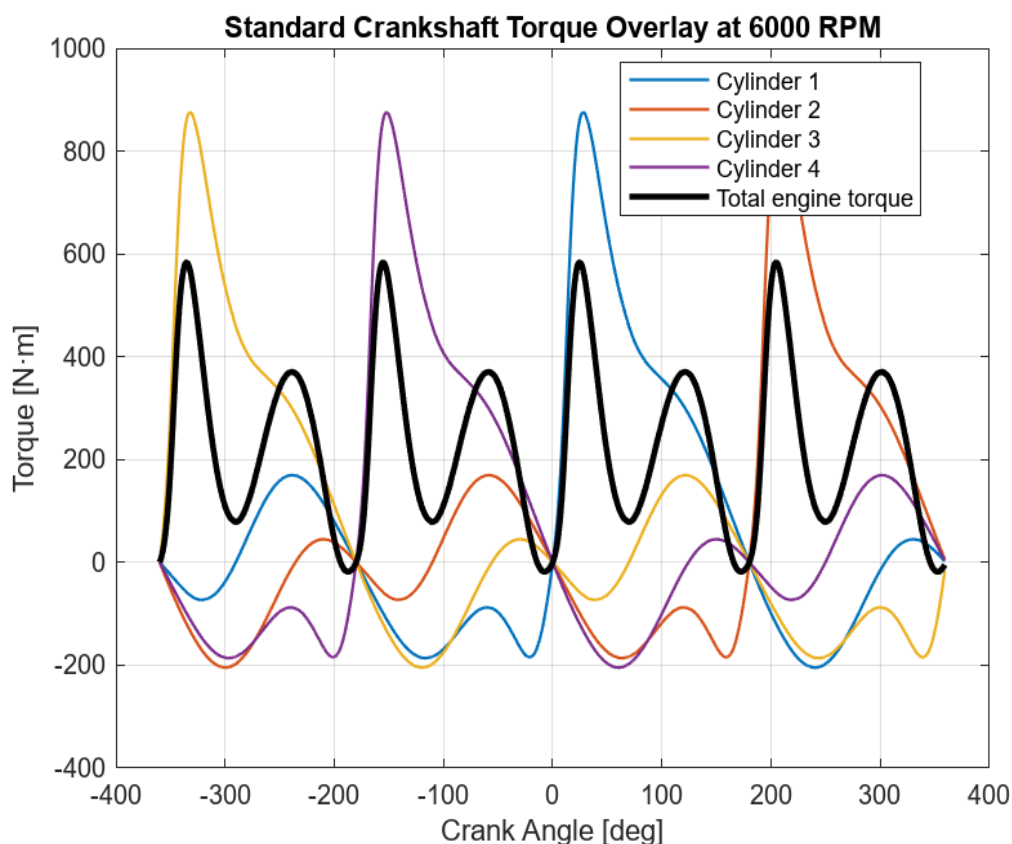


Figure 9 Standard Crankshaft Torque overlay

6.3. Cross-Plane Crankshaft Configuration

The cross-plane case used cylinder phase offsets of 0, 270, 450 and 630 degrees. This produced a total torque trace with a less regular pattern across the 720 degree cycle, where the individual cylinder contributions combine to create several sharp positive peaks and more pronounced negative troughs. The graph shows that torque delivery is concentrated into a smaller number of stronger pulses, rather than being spread more evenly through the cycle. Although the total output over the cycle is unchanged, the instantaneous response is much more uneven because the firing events are distributed differently. As highlighted in the literature, fluctuations of this kind are more likely to increase torsional disturbance within the drivetrain (Pfabe and Woernle, 2016).

Table 10 Cross-Plane Torque Results

Parameter	Value
Max torque (Nm)	1332.443
Min torque (Nm)	-270.117
Mean torque (Nm)	229.819
Torque range (Nm)	1602.56
Torque ripple ratio	6.9732

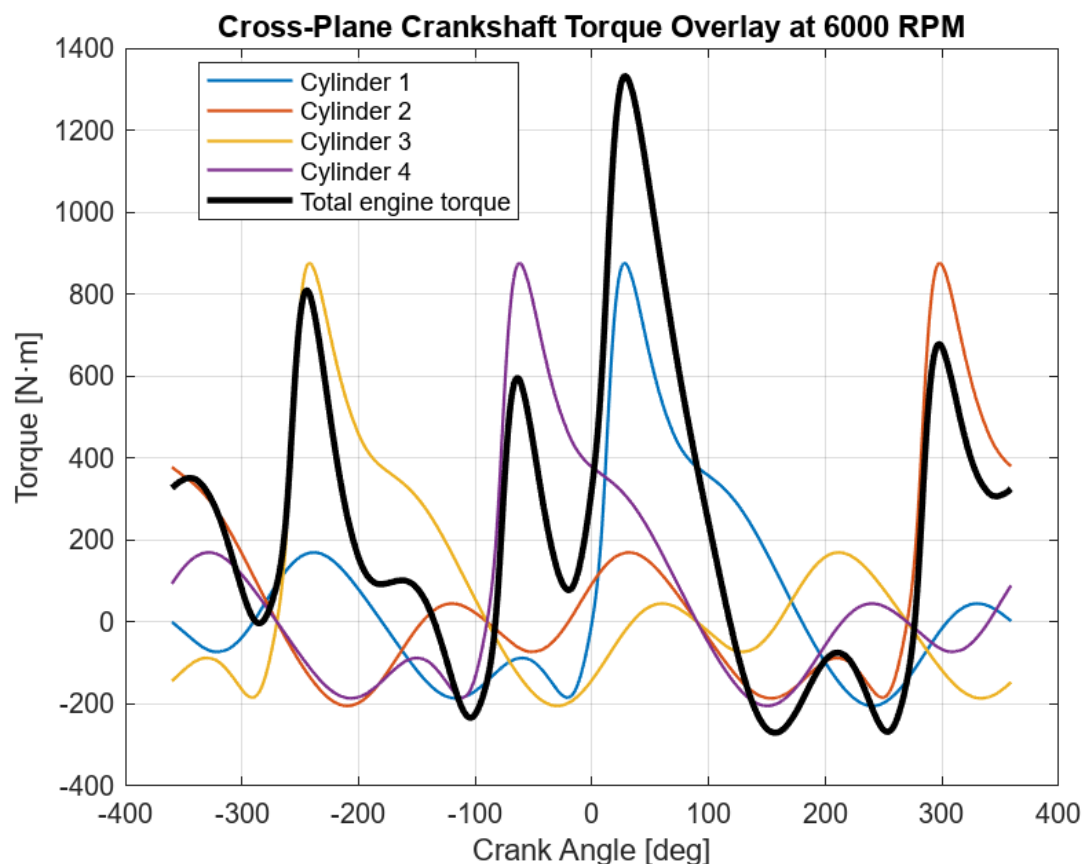


Figure 10 Cross-plane Crankshaft torque overlay

6.4. Crankshaft Comparison

Direct comparison showed that the cross-plane arrangement produced a much wider torque range than the standard configuration, while mean torque remained effectively unchanged. Both crankshafts therefore delivered the same average torque over the cycle, but the standard arrangement did so in a more even and controlled manner. Its torque trace remained more regular, with smaller fluctuations between peaks and troughs, showing that the individual cylinder contributions were distributed more consistently through the cycle. In contrast, the cross-plane phasing concentrated torque into fewer, larger pulses, which increased the overall torque ripple and made the instantaneous delivery far less uniform. Within the limits of this model, the standard arrangement is therefore the more suitable configuration where smoother torque transmission is required.

Table 11 Crankshaft Comparison

Metric	Standard	Cross-plane
Max torque (Nm)	583.8	1332.4
Min torque (Nm)	-18.23	-270.12
Mean torque (Nm)	229.82	229.82
Torque range (Nm)	602.02	1602.56
Torque ripple ratio	2.62	6.97

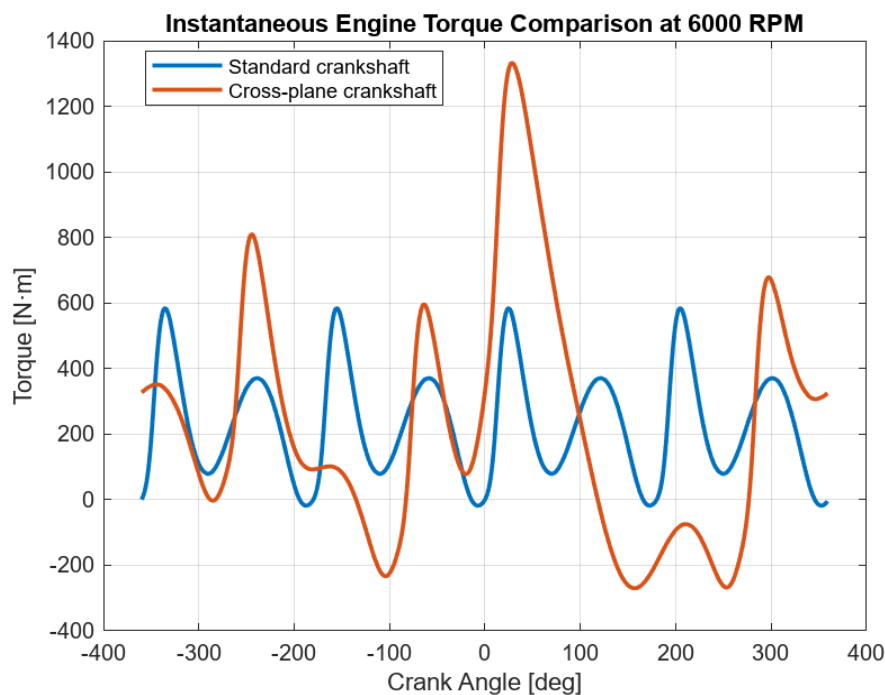


Figure 11 Standard Vs Cross-plane Instantaneous Torque Comparison

6.5. Discussion of crankshaft comparison

The torque analysis gave a clear result, but it should still be interpreted with caution. Although both crankshaft arrangements produced the same mean torque, their instantaneous delivery differed substantially. In this model, the standard configuration gave the smoother response, while the cross-plane arrangement produced greater torque ripple, higher peak torque and a larger negative excursion. This does not mean that every cross-plane crankshaft is inherently less smooth, but it does show that the phasing adopted here generated a less uniform torque trace. This is consistent with the literature, where fluctuating engine torque is linked to torsional crankshaft vibration and increased dynamic disturbance in the drivetrain (Pfabe and Woernle, 2016; Wu et al., 2022). From a design perspective, the standard arrangement is therefore more favourable where smoothness, refinement and reduced torsional loading are priorities. However, this comparison was only carried out at 6000 rpm, so it represents one operating condition rather than the full engine range. Even so, it shows that average torque alone is not enough to describe dynamic behaviour; the timing and fluctuation of torque delivery are equally important when judging crankshaft performance.

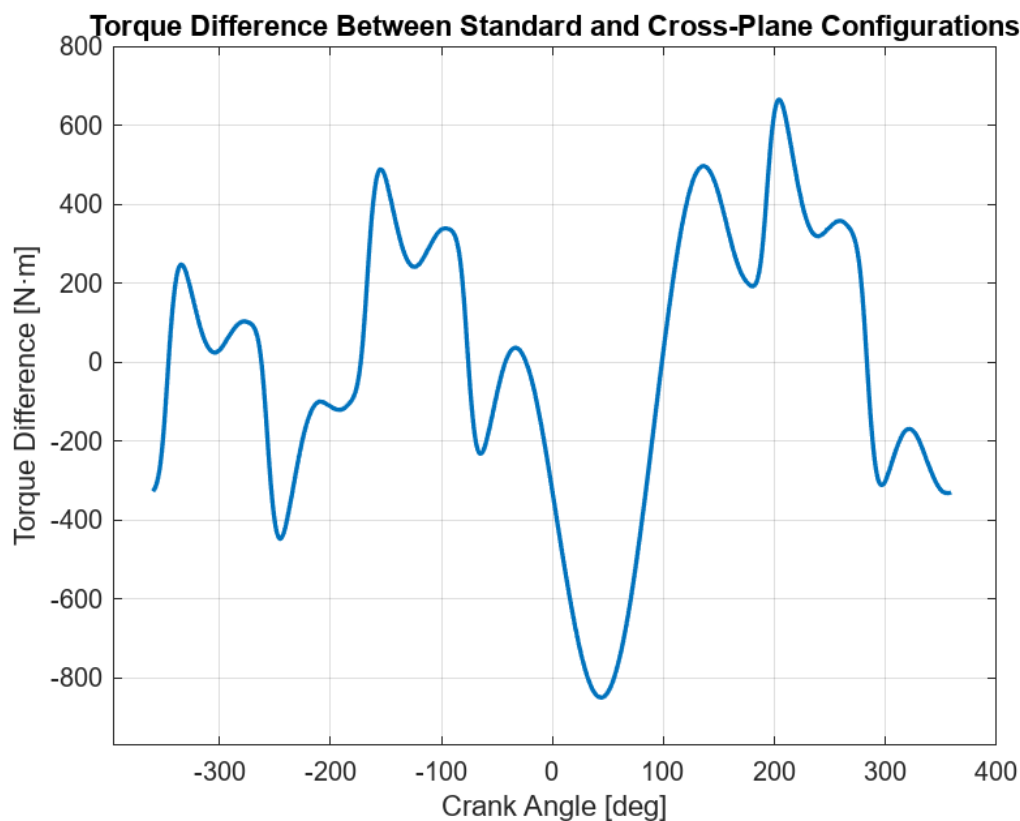


Figure 12 Standard Vs Cross-plane Instantaneous Torque Difference

7. Conclusions

This report has shown that the supplied Ford 1.6 Sigma pressure dataset can be used within a MATLAB analytical framework to assess bottom-end loading, fatigue margin and instantaneous crankshaft torque behaviour. The results indicated that the connecting rod bolt emerged as the critical component as engine speed increased. Bolt fatigue margin reduced significantly at 9000 rpm, confirming that high-speed operation placed the greatest demand on this part of the assembly. The redesign study showed that the reduced-shank concept gave the most effective improvement, offering a better balance between fatigue performance and mass than the alternative upgrade options. In the torque analysis, both crankshaft configurations produced the same mean torque, but the cross-plane arrangement generated much greater torque ripple, so the standard configuration provided the smoother response. This showed that crankshaft phasing had a strong influence on instantaneous torque delivery even when average output was unchanged. It also highlighted that mean torque alone is not sufficient for judging dynamic performance.

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