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Analytical Assessment of Engine Internal Components

Contemporary Powertrain Design ACHA6006

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22nd March 2026

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1. Introduction

The bottom end components of a reciprocating internal combustion engine operate under severe cyclic loading produced by the combined effects of combustion gas pressure and inertia forces. As a result, components such as the connecting rod, gudgeon pin and connecting rod bolts must be designed to withstand repeated tensile, compressive and bending stresses without fatigue failure, while also avoiding unnecessary mass (MAHLE GmbH, 2016). It is important to evaluate these competing requirements because insufficient fatigue resistances can reduce durability and life expectancy, whereas over-engineering increases reciprocating mass and will limit engine performance. This report therefore focuses on using a Ford 1.6 Sigma pressure data and kinematic equations to calculate component forces, stresses and fatigue safety factors (FSF).

2. Methodology

The analysis was carried out in MATLAB using the supplied Ford 1.6 Sigma in-cylinder pressure data together with the engine geometry, component dimensions and mass properties provided in the assignment. The overall method combined piston motion equations, inertia loading, gas loading and fatigue analysis to assess the structural behaviour of the main reciprocating components. The method was structured in four stages:

1. component forces, stresses and FSF were determined across the original operating range up to 6000 rpm.
2. the pressure data was extended to 9000 rpm by interpolation and the connecting rod bolt FSF was reassessed.
3. the connecting rod bolt was redesigned to improve fatigue performance at the higher speed condition.
4. the cylinder data was phase-shifted to compare instantaneous torque behaviour for flat-plane and cross-plane crankshaft arrangements over a full 720° cycle.

2.1 Kinematics and inertia Loading

Piston motion was calculated using the standard piston and crank equations to determine connecting rod angle, piston displacement, velocity and acceleration as functions of crank angle (Noble, 2026). The reciprocating inertia force was then found from Newton's second law using the calculated acceleration and reciprocating mass. Gas force was determined from the in-cylinder pressure data and piston area, and the net piston force was obtained from the difference between gas and inertia forces. This net force was then resolved through the connecting rod to obtain the axial rod load used in the later stress calculations. The method assumes ideal piston-crank motion and neglects secondary effects such as connecting rod flexibility, piston slap and bearing compliance. (see Appendix 1 – 3 for detailed calculations).

2.2 Stress & Fatigue Analysis

Stress values through the engine cycle were calculated for the connecting rod midpoint, gudgeon pin and connecting rod bolts using the force results together with the relevant section properties. The connecting rod midpoint was assessed using direct axial stress, the gudgeon pin was assessed using combined bending and shear, and the bolt stress was calculated from the sum of the preload and the additional tensile load caused by joint separation forces.

For each component, the maximum and minimum stresses over the cycle were identified and used to calculate mean stress, stress range and allowable stress range. Fatigue behaviour was then assessed using the Goodman method, and the (FSF) was calculated from the ratio of allowable stress range to applied stress range (Appendix 4).

2.3 Extension to 9000 rpm and bolt redesign

An additional pressure trace at 9000 rpm was generated in MATLAB by interpolation and combined with the same kinematic and force calculations to reassess the connecting rod bolt FSF at elevated engine speed.

Following this, the bolt was redesigned to improve fatigue performance at 9000 rpm. The redesign study considered three approaches: geometry modification, substitution of alternative materials, and inclusion of rolled-thread effects through an imposed compressive residual stress of -100 MPa applied to σ_{max} and σ_{min} at the thread root. Material properties were taken from Ansys Granta, and the endurance limit was taken as 45% of ultimate tensile strength to maintain consistency

2.4 Instantaneous Torque Comparison

For the crankshaft comparison, the previously calculated one-cylinder force data were converted into instantaneous torque using crank kinematics. These were then shifted across the full 720° engine cycle in MATLAB using *circshift* to represent the four-cylinder arrangement. Different crank phasing was applied for the flat-plane and

cross-plane layouts, and the individual cylinder torque traces were summed to obtain total instantaneous torque. This allowed the effect of crank phasing on inertia torque, gas torque and combined torque behaviour to be compared directly.

3 Results & Discussion

3.1 Kinematic behaviour and inertia scaling

Figures 1 and 2 show that piston acceleration and reciprocating inertia force increase strongly with engine speed because the inertia term depends on angular speed squared. This means the load increase is disproportionately large at higher rpm, rather than scaling linearly. From an engineering perspective, this is important because it shifts the loading balance of the bottom end as speed rises (Heywood, 2018). At lower speed, combustion gas pressure remains the dominant source of compressive loading over much of the cycle, but at higher speed inertia becomes increasingly influential and can approach or exceed the gas contribution in critical regions.

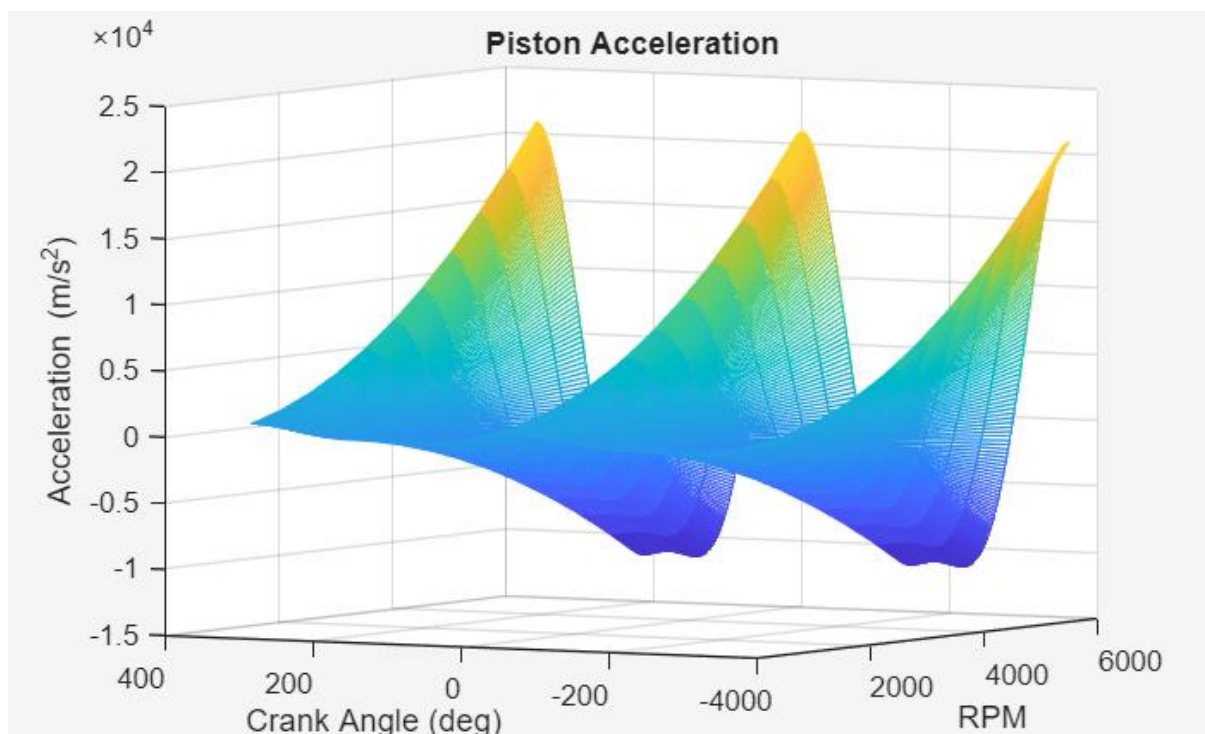


Figure 1: Piston acceleration vs crank angle and engine speed.

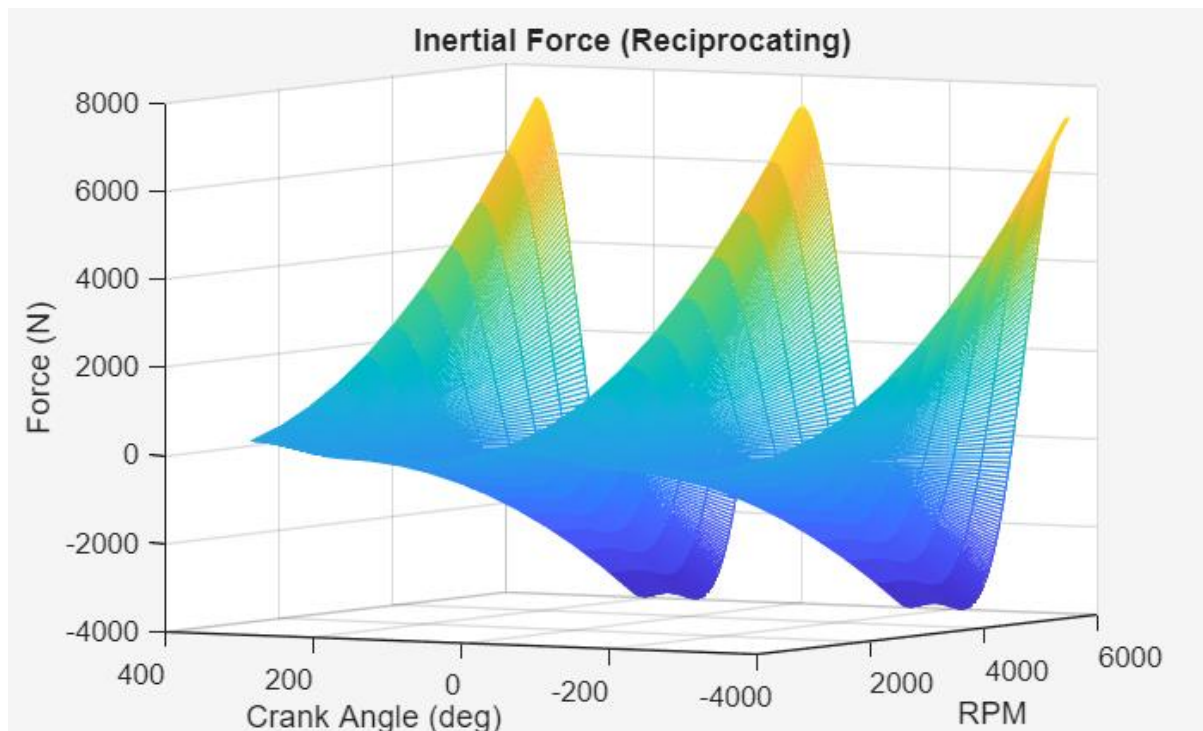


Figure 2: Reciprocating inertia force vs crank angle and engine speed.

3.2 Gas force, net force and rod/lateral force

Fig. 3-5 show that gas force, net piston force and connecting rod force vary substantially through the cycle, with gas pressure mainly driving compression and inertia contributing the main tensile effect. The side force and associated friction follow the same general pattern, but these are secondary to the main durability result (Fig. 6-7). This highlights that the connecting rod bolts are governed by the tensile separating load created when inertia exceeds gas force at the big end (Heywood, 2018). This means the bolt becomes progressively more critical as engine speed increases, not because peak combustion pressure rises dramatically, but because the inertia-driven tendency to pull the joint apart becomes much more severe.

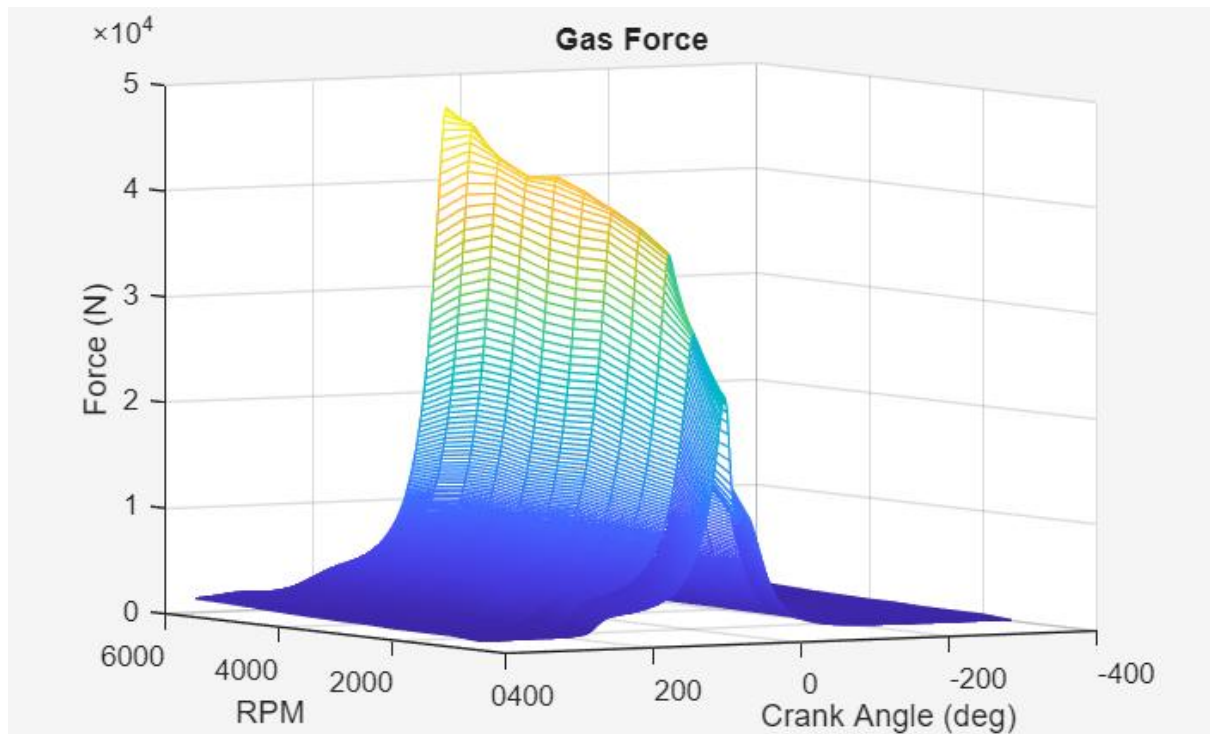


Figure 3: Gas Force vs crank angle and engine speed.

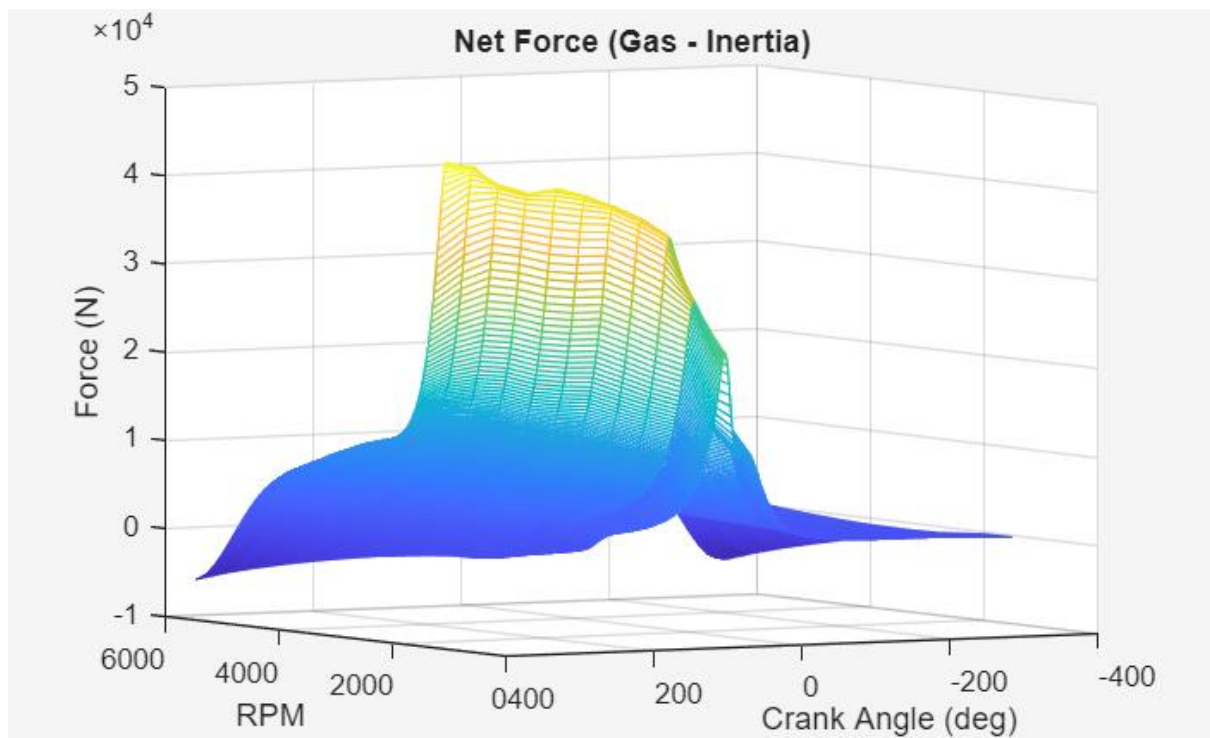


Figure 4: Net piston force (gas – inertia) vs crank angle and engine speed.

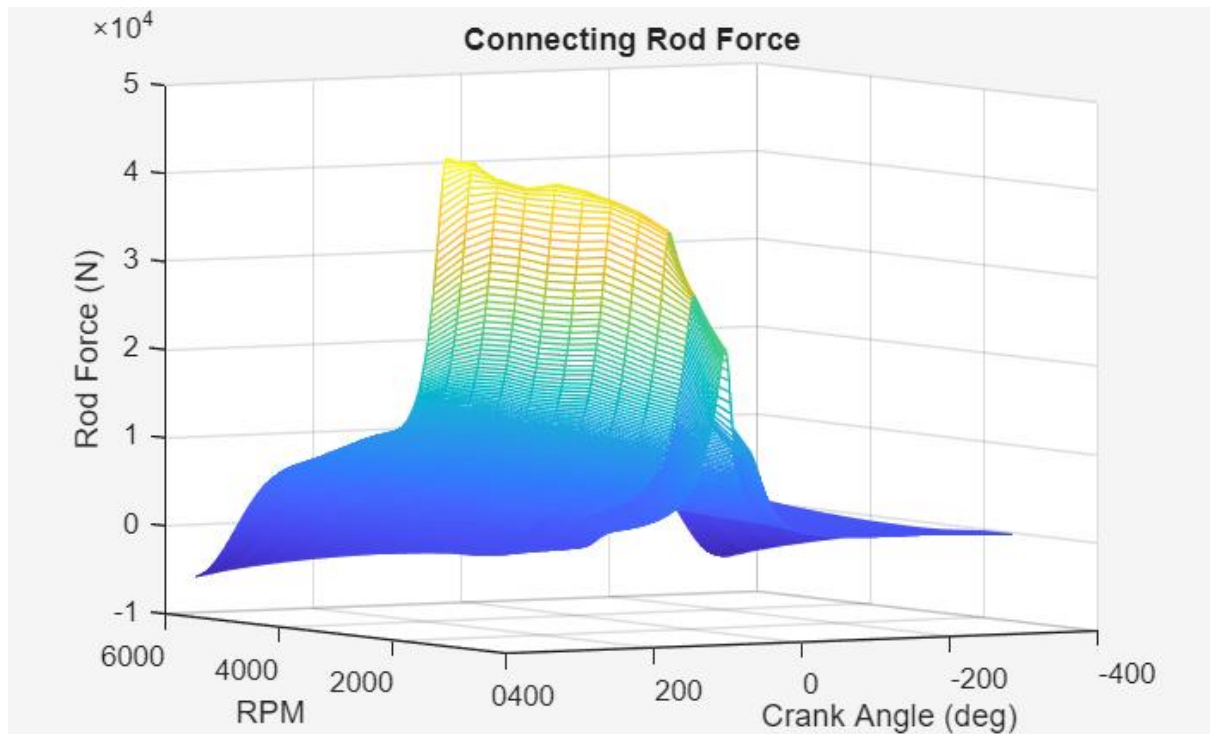


Figure 5: Connecting rod axial force vs crank angle and engine speed.

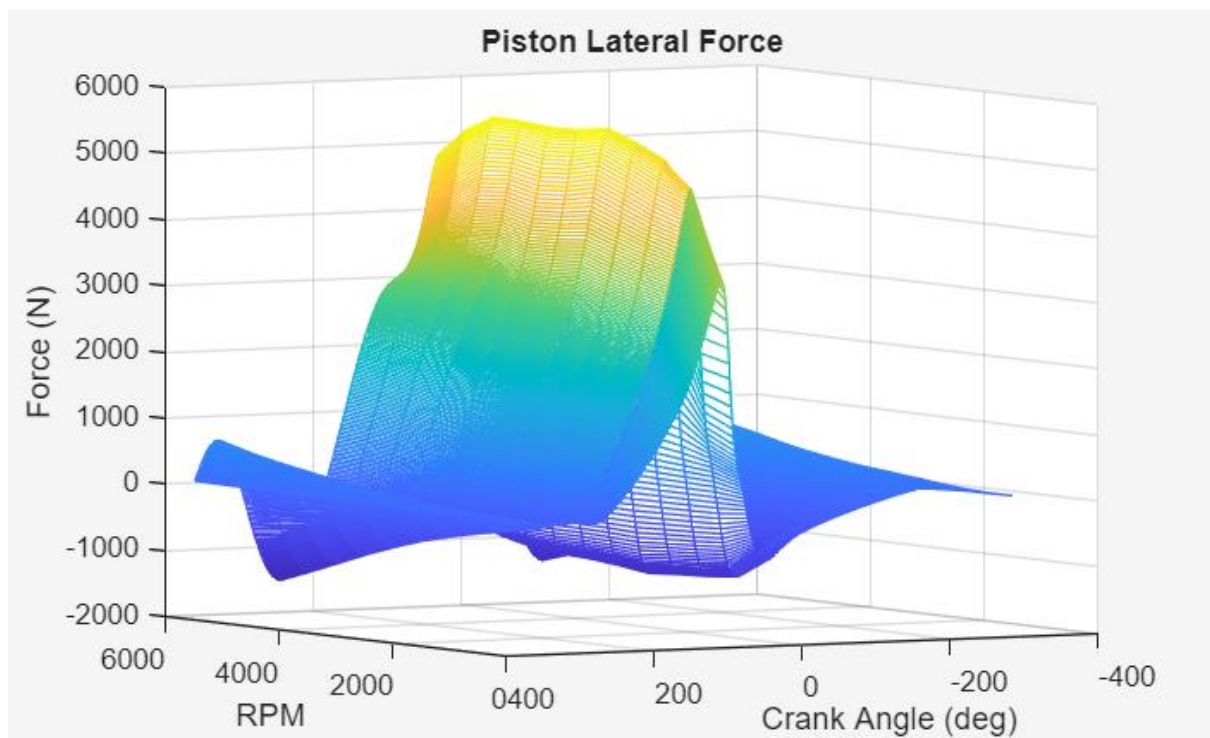


Figure 6: Piston lateral force vs crank angle and engine speed.

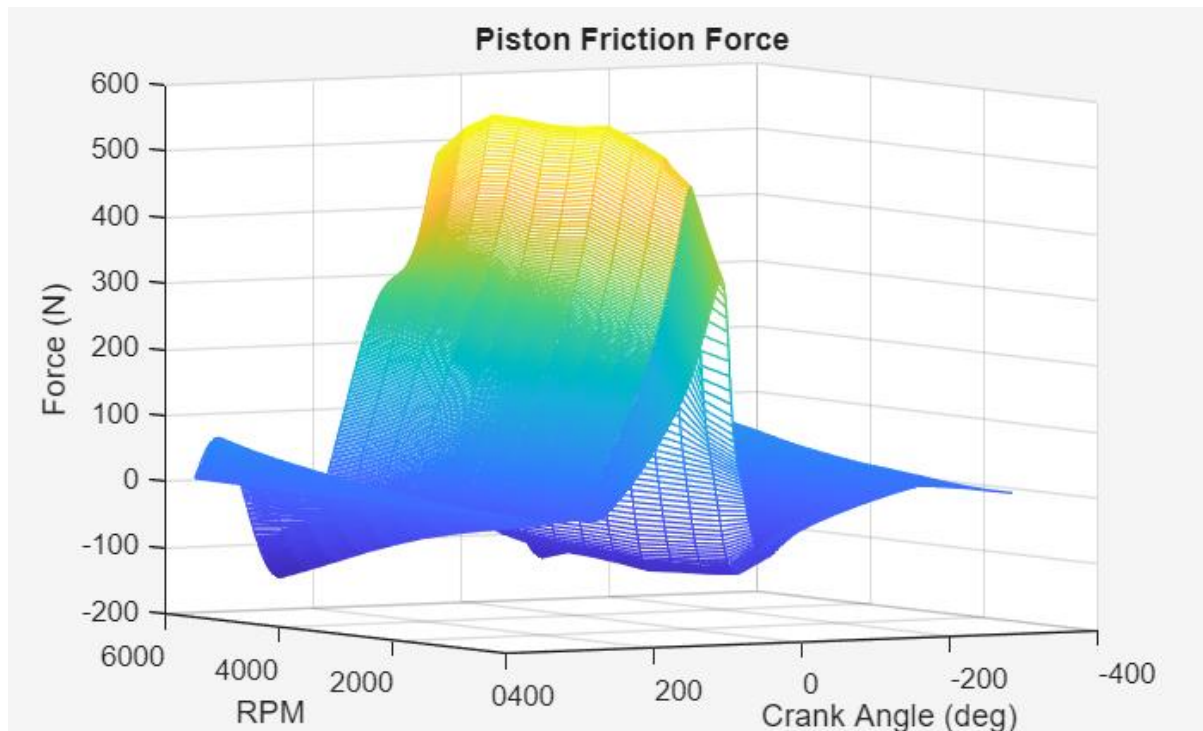


Figure 7: Piston Friction Force vs crank angle and engine speed.

3.3 Baseline Durability at 6000 rpm

The baseline 6000 rpm results show that all three assessed components have FSF above 3.0 (Table 1 and Figures 8-10). The bolt stress range was relatively small at 33.28 MPa, despite a high mean stress of 551.24 MPa, showing that preload dominated the stress state.

Table 1: Baseline 6000 rpm results

Component	Sigma min (MPa)	Sigma max (MPa)	Sigma Mean (MPa)	Sigma Range (MPa)	FSF
Conrod midpoint	-205.98	30.29	-87.84	236.27	4.144
Gudgeon pin	-265.48	35.99	-114.74	301.47	3.328
Conrod bolt	534.60	567.88	551.24	33.28	3.306

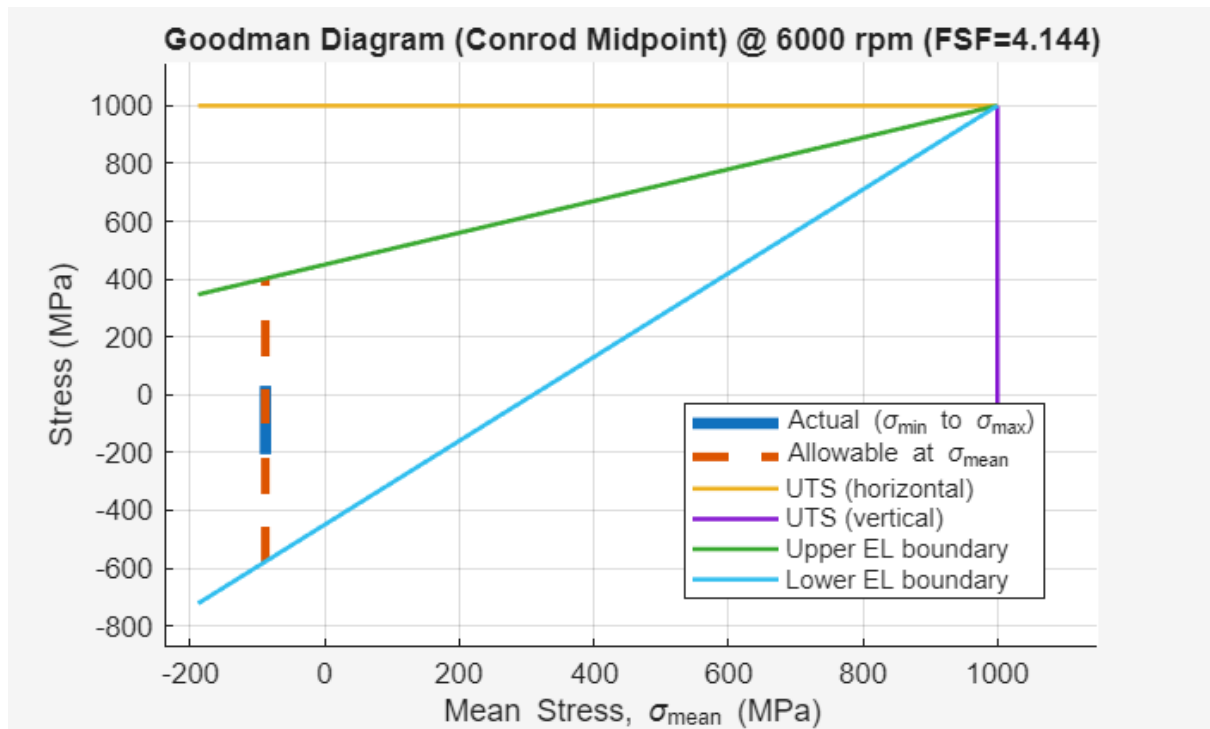


Figure 8: Goodman Diagram for Conrod Midpoint at 6000 rpm.

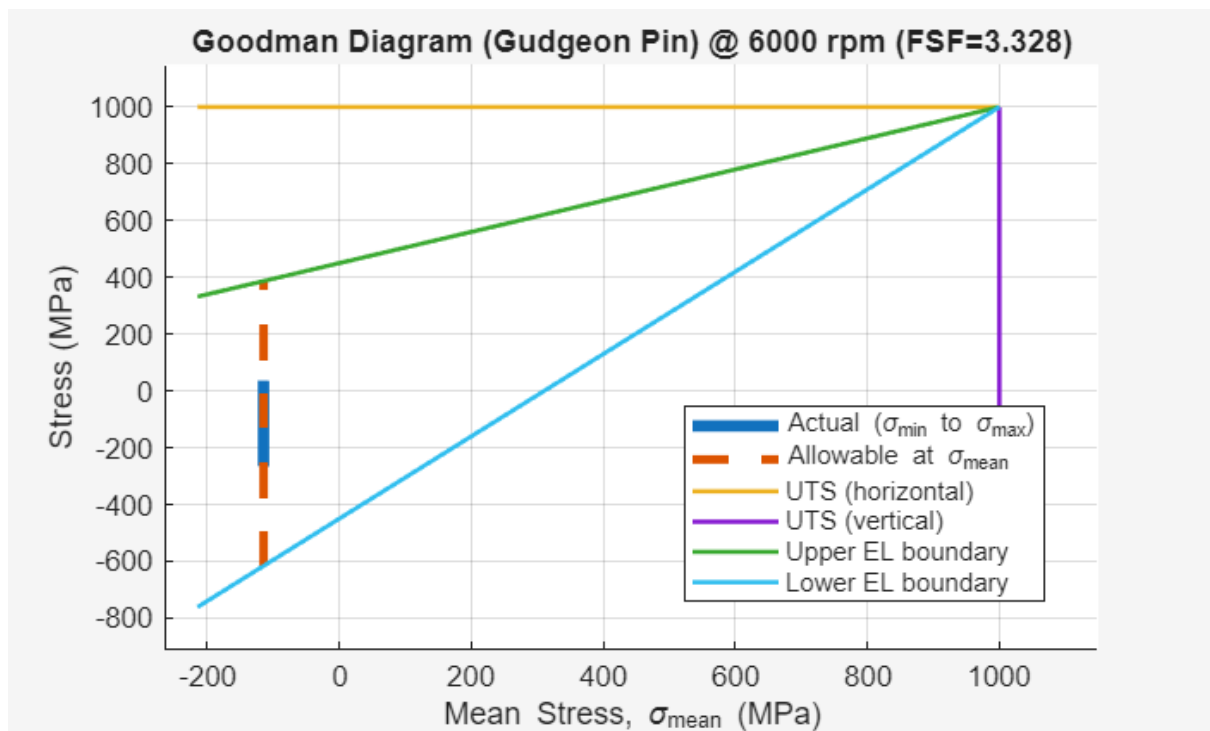


Figure 9: Goodman Diagram for Gudgeon Pin at 6000 rpm.

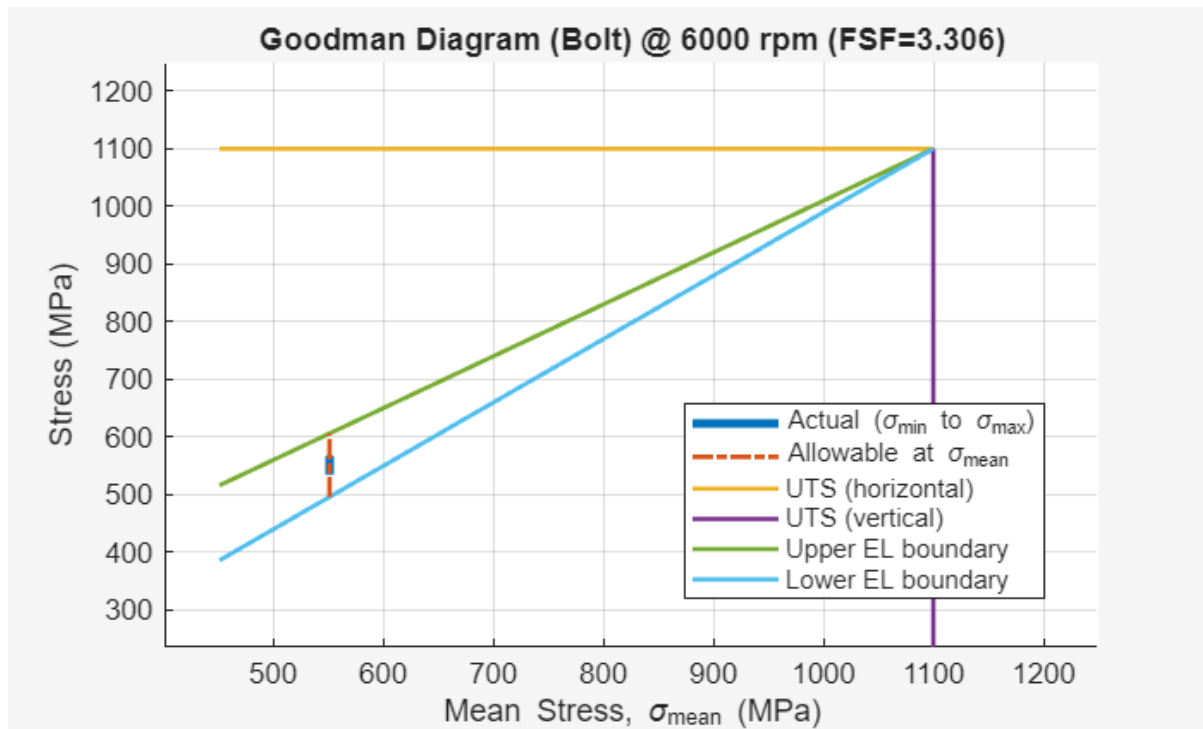


Figure 10: Goodman Diagram for Conrod Bolt at 6000 rpm.

These results suggest that, within the original operating range, the assessed components are not an immediate fatigue concern. The connecting rod midpoint and gudgeon pin both operate with a negative mean stress because the compressive gas load is larger than the tensile inertia load over much of the cycle. In Goodman terms this is favourable, since compressive mean stress increases the allowable alternating stress and therefore improves fatigue margin. This helps explain why both components remain comfortably above an FSF of 1.0 despite relatively large stress ranges.

The gudgeon pin is more critically loaded than the midpoint of the connecting rod. This is because the pin experiences a more severe combined stress state, with bending and shear acting together rather than predominantly axial loading (MAHLE GmbH, 2016). The result is useful because it shows that the more geometrically constrained a part is, the more demanding its fatigue case is.

The most interesting baseline result is the bolt. Although the FSF is also above 3, its behaviour is fundamentally different from the conrod and gudgeon pin. The stress

range is very small, only 33.28 MPa, but the mean stress is very high generated by preload (Table 1). This is typical of connecting rod bolts, where the bolt is preloaded to minimise the alternating stress it experiences (MAHLE GmbH, 2016). The bolt thread remains a fatigue-critical feature because the mean stress is high and local stress concentration is severe (Jia et al., 2025). Literature identifies bolt threads and joint opening behaviour as dominant durability issues in conrod assemblies (Kurata et al., 1989; Griza et al., 2009; de Moraes Teixeira and Rotta, 2011). Therefore, the baseline bolt remains durable at 6000 rpm not simply because it is over-engineered, but because the preload and stiffness distribution keep the dynamic stress under control.

Overall, the 6000 rpm results indicate that the standard design has a comfortable fatigue margin under the original operating range. However, the fact that all three FSFs are above 3.0 also suggests that at least some degree of conservatism exists in the baseline design. This is reasonable for a production engine, where durability and reliability usually take priority over absolute mass minimisation.

3.4 Extended 9000 rpm bolt condition

When the operating range was extended to 9000 rpm, the connecting rod bolt fatigue margin deteriorated sharply. The worst-case separating load increased from 10.97 kN to 26.01 kN, the additional bolt load increased from 1.41 kN to 3.35 kN, and the calculated bolt FSF fell from 3.306 to 1.337 (Fig.11). The corresponding bolt stress range increased from 33.28 MPa to 78.88 MPa.

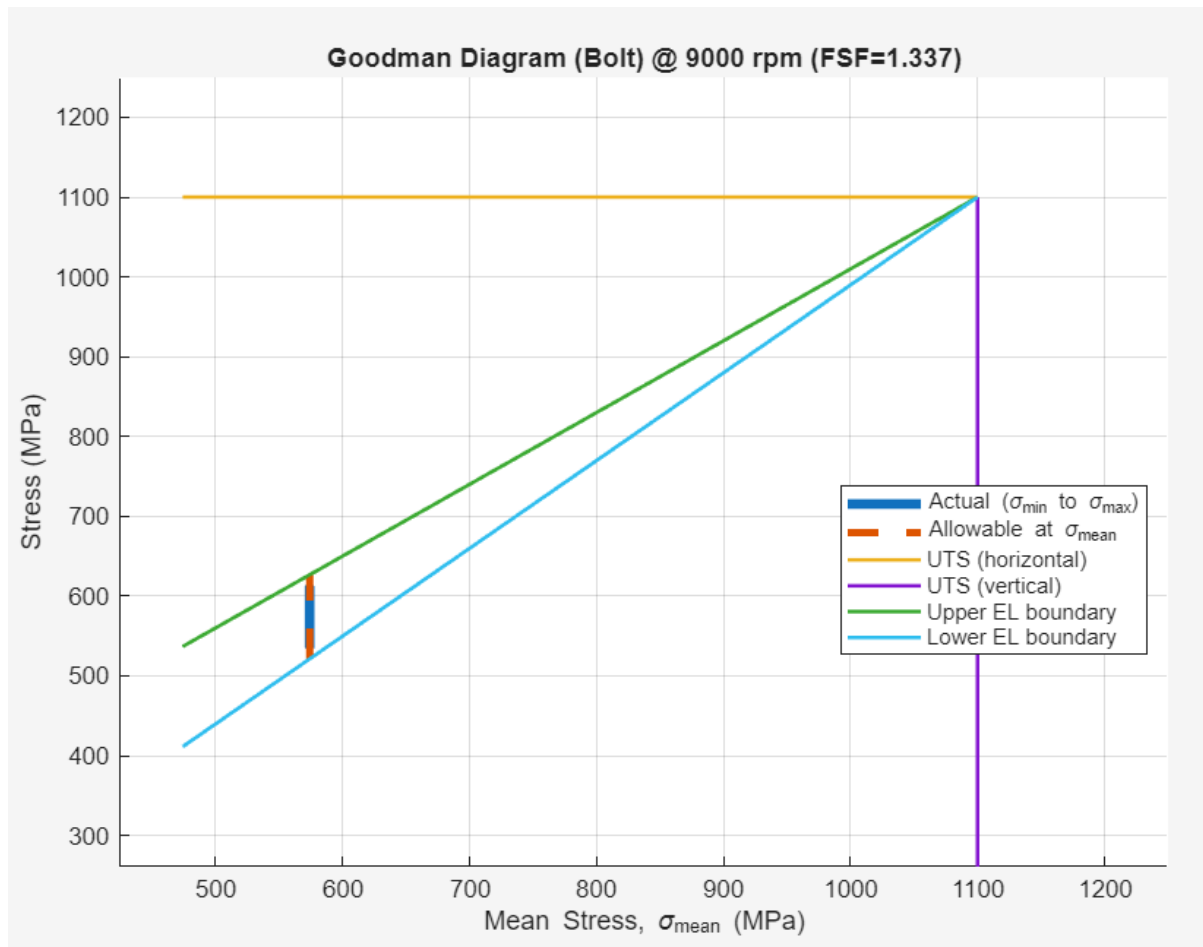


Figure 11: Goodman Diagram for Conrod Bolt at 9000 rpm.

This reduction is in the expected direction and confirms that the bolt becomes the most critical bottom-end component as speed increases. The deterioration is driven much more by inertia than by gas pressure. At the worst bolt condition, gas force only increased from about 1.32 kN to 1.62 kN, whereas bolt inertia force increased from about 12.29 kN to 27.63 kN. The separating load therefore became much more inertia-driven at high speed. In engineering terms, the bolt is no longer governed by combustion pressure, but by the reciprocating system pulling the big-end joint apart near TDC.

The FSF at 9000 rpm is still above 1.0, which does not fully match the assignment expectation that the original bolt should become unsafe. This suggests that the simplified analytical model is not capturing the full severity of the real joint behaviour. Possible reasons include idealised stiffness sharing, simplified treatment of thread-root loading and the assumption that the load is shared symmetrically between the

two bolts. Even so, the reduction from 3.306 to 1.337 is large enough to justify redesign, and the trend itself is physically credible. The 9000 rpm result should therefore be treated as directionally reliable, even if the exact numerical margin is optimistic.

3.4 Bolt Redesign

3.4.1 Geometric Redesign

As the 9000 rpm analysis identified the connecting rod bolt as the limiting component, the first redesign stage focused on geometry. Three studies were carried out to assess the influence of wasted-shank diameter and effective loaded length.

The geometric redesign study showed that reducing wasted-shank diameter had the greatest effect on fatigue performance. In Study 1, decreasing D_2 from 6.5 mm to 5.0 mm increased FSF from 1.834 to 2.465, showing the bolt is highly sensitive to shank diameter (Table 2). This is because a smaller middle section reduces bolt stiffness and therefore reduces the proportion of the separating load transferred into the bolt during the critical tensile part of the cycle. The result indicates that diameter reduction is the most effective geometric route to improving bolt fatigue resistance.

Table 2: Geometric bolt redesign Study 1.

Study 1	Option A	Option B	Option C	Option D
L_1 (mm)	4	4	4	4
L_2 (mm)	14.26	14.26	14.26	14.26
L_3 (mm)	9	9	9	9
Total Length (mm)	27.26	27.26	27.26	27.26
D_1 (mm)	7.8	7.8	7.8	7.8
D_2 (mm)	6.5	6	5.5	5
D_3 (mm)	7.2	7.2	7.2	7.2
FSF	1.834	1.993	2.197	2.465

By comparison, increasing wasted-shank length produced a smaller improvement. In Study 2, increasing L_2 from 12 mm to 15 mm at $D_2 = 6.0$ mm only increased FSF from 1.923 to 2.016, while in Study 3 the same length change at $D_2 = 5.0$ mm increased FSF from 2.321 to 2.513 (Tables 3 & 4). This shows that increasing effective loaded length is beneficial, but less so than reducing shank diameter. Taken together, the studies indicate that stiffness reduction through cross-sectional area is the dominant factor, while length change provides only a secondary gain. The best overall geometry was therefore Study 3 Option D with $L_1 = 3.26$ mm, $L_2 = 15.0$ mm, $L_3 = 9.0$ mm, $D_2 = 5.0$ mm, giving an FSF of 2.513.

Table 3: Geometric bolt redesign Study 2.

Study 2	Option A	Option B	Option C	Option D
L_1 (mm)	6.26	5.26	4	3.26
L_2 (mm)	12	13	14.26	15.00
L_3 (mm)	9	9	9	9
Total Length (mm)	27.26	27.26	27.26	27.26
D_1 (mm)	7.8	7.8	7.8	7.8
D_2 (mm)	6	6	6	6
D_3 (mm)	7.2	7.2	7.2	7.2
FSF	1.923	1.954	1.993	2.016

Table 4: Geometric bolt redesign Study 3.

Study 3	Option A	Option B	Option C	Option D
L_1 (mm)	6.26	5.26	4	3.26
L_2 (mm)	12	13	14.26	15.00
L_3 (mm)	9	9	9	9
Total Length (mm)	27.26	27.26	27.26	27.26
D_1 (mm)	7.8	7.8	7.8	7.8
D_2 (mm)	5	5	5	5
D_3 (mm)	7.2	7.2	7.2	7.2
FSF	2.321	2.385	2.465	2.513

3.4.2 Material redesign

After the best geometry had been selected, a material comparison was carried out using that same geometry (Table 5).

Table 5: Material Properties of candidate materials taken from Ansys Grant EduPack.

Material	Young's Modulus, E (GPa)	Yield Strength (MPa)	Tensile Strength/ UTS (MPa)	*Assumed Endurance Limit 45% of UTS (MPa)	**Density (kg/m ³)	Cost (GBP/kg)
Baseline (high strength steel equivalent)	200	990	1100	495	7.79e3 - 7.87e3 (est. From comparable high strength steel)	1.29 - 1.91 (est. From comparable high strength steel)
A286/ 660	201	651	956	430.2	7.9e3 – 7.98e3	5.88 – 8.79
SAE 4335M	196	1150-1270	1180-1300	531 (using lower UTS limit of 1180 MPa)	7.81e3 – 7.84e3	1.45 – 2.13
Maraging Steel 300	190	1840	1870-2070	841 (using lower UTS limit of 1870 MPa)	7.88e3 – 7.96e3	30.1 – 38.4

Notes for Table 5:

* For consistency with calculations, the endurance limit was assumed as 45% of UTS, using the conservative lower end UTS values.

** Density for baseline material established from comparable high strength steel

The redesigned bolt using A286 gave an FSF of 2.245, which was an improvement over the baseline redesigned geometry but remained lower than the stronger alternatives. SAE 4335M increased FSF to 2.859, while maraging steel 300 produced the highest value at 4.720 (Table 6).

Table 6: Material redesign results.

Material	rpm	Sigma Range (MPa)	FSF
Baseline	9000	78.88	1.337
A286/660		43.57	2.245
SEA 4335M		42.64	2.859
Maraging Steel 300		41.51	4.720

Numerically, maraging steel is the best material. However, it is not necessarily the best engineering choice overall. Its fatigue performance is significantly better, but that benefit would come with a substantial increase in cost and likely with greater manufacturing and procurement implications. In contrast, SAE 4335M gives a clear improvement over the baseline design, while remaining much closer to a practical high-strength steel solution. For that reason, SAE 4335M represents the best overall compromise between fatigue improvement and realistic implementation.

3.4.3 Thread rolling

Thread rolling is a cold-forming process in which the thread profile is formed rather than cut, improving the thread-root surface and introducing beneficial residual compressive stress (Horn, *et al.*, 2006). Because fatigue cracks usually initiate at this highly stressed surface region, the compressive residual stress reduces effective mean stress and improves the fatigue safety factor (Bradley *et al.*, 2007).

For the redesigned bolt with the selected material, SAE 4335M, thread rolling increased FSF from 2.859 to 3.348 (Table 7). The stress range remained 42.64 MPa before and after rolling, showing that the improvement came from a shift to a more compressive mean stress rather than from any reduction in external load. This could be beneficial from a manufacturing perspective because fatigue performance is improved without changing the basic geometry or applied loading.

Table 7: Rolled thread study results.

Material: SEA 4335 M	rpm	Sigma Range (MPa)	FSF
Standard Threads	9000	42.64	2.859
Rolled Threads		42.64	3.348

3.5 Final redesign comparison

The final redesign summary shown in Table 8 and Figure 12 highlights that no single redesign measure is sufficient on its own to recover all of the original fatigue margin. Geometry alone improves the result substantially, while material substitution and thread rolling adds additional gains. The overall conclusion is therefore that the best solution is a combination of all three.

Table 8: Stages of redesign and final redesign summary.

Case	RPM	Sigma Range (MPa)	FSF
Original Geometry	6000	33.28	3.306
Original Geometry	9000	78.88	1.337
Best Redesigned Geometry	9000	54.16	1.993
Best Redesigned Geometry and 'best' material (SAE 4335M)	9000	42.64	2.859
Best Geometry, Material and rolled threads	9000	42.64	3.348

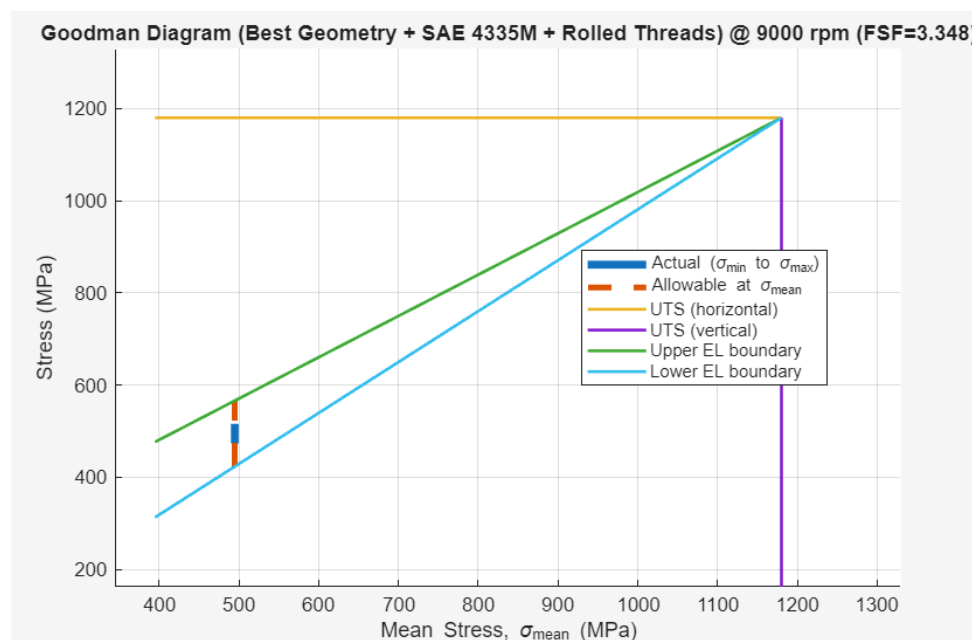


Figure 12: Goodman's diagram of final bolt design.

4. Instantaneous Torque Comparison

The torque comparison was built up from the one-cylinder inertia, gas and net torque traces (Fig.13) before these were phase-shifted across the full 720° cycle to represent the four-cylinder engine (Konstantinos-Marios et al., 2021). The cross-plane arrangement produced a much lower inertia torque ripple than the standard flat-plane arrangement, with peak-to-peak ripple reducing from 911.424 Nm to 41.843 Nm and RMS fluctuation reducing from 321.258 Nm to 14.803 Nm (Fig.14). This confirms that the 90° crank phasing is highly effective at smoothing the inertia component of torque (Heywood, 2018).

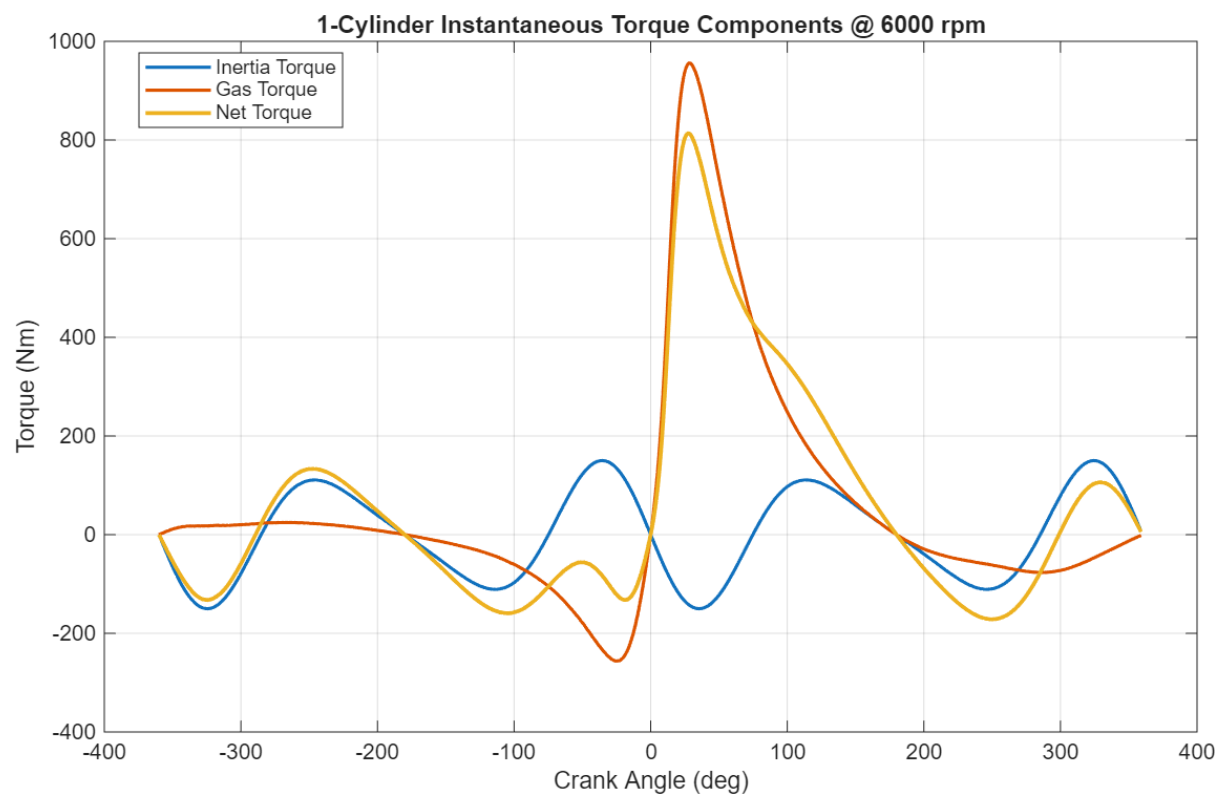


Figure 13: One-cylinder inertia, gas and net torque traces with the flat plan crank.

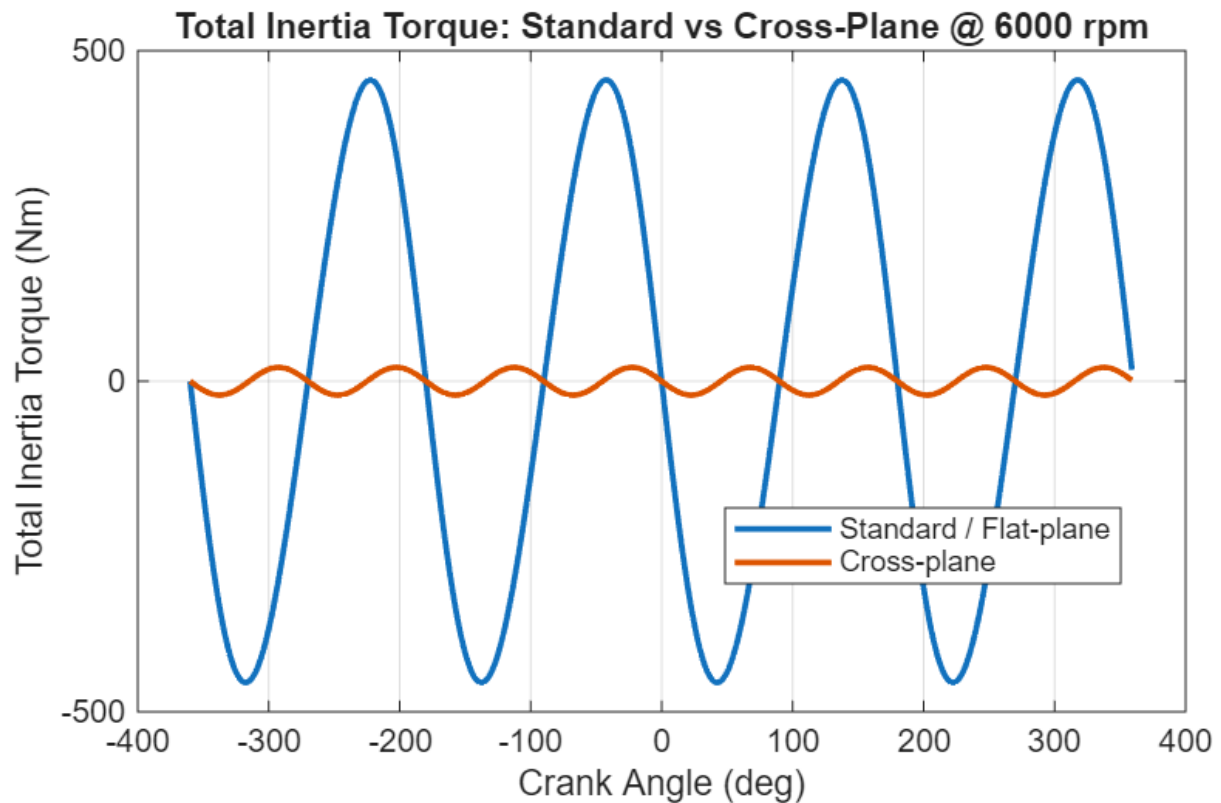


Figure 14: Total Inertia Torque comparison of Flat Plane & Cross Plane arrangements.

In this model, gas torque ripple was identical for both crank layouts because the same one-cylinder pressure trace was shifted using the same 180° firing spacing over the 720° cycle. The difference between the two layouts therefore came from inertia torque phasing rather than from any change in the gas torque summation.

When gas and inertia torque were combined, the flat-plane arrangement gave the lower overall torque ripple (Fig.15). Peak-to-peak combined torque ripple was 543.602 Nm for the flat-plane arrangement compared with 1110.158 Nm for the cross-plane arrangement, while RMS fluctuation was 140.264 Nm and 365.095 Nm respectively (Table 9). This shows that lower inertia ripple does not automatically produce lower total torque ripple once the gas torque contribution is included. Under the assumptions used here, the cross-plane arrangement is more favourable for reducing inertia-driven fluctuation, whereas the flat-plane arrangement gives the smoother total combined torque trace (Franco, Franchek and Grigoriadis, 2007). This suggests that the more suitable layout depends on whether the priority is inertial smoothness or total torque delivery.

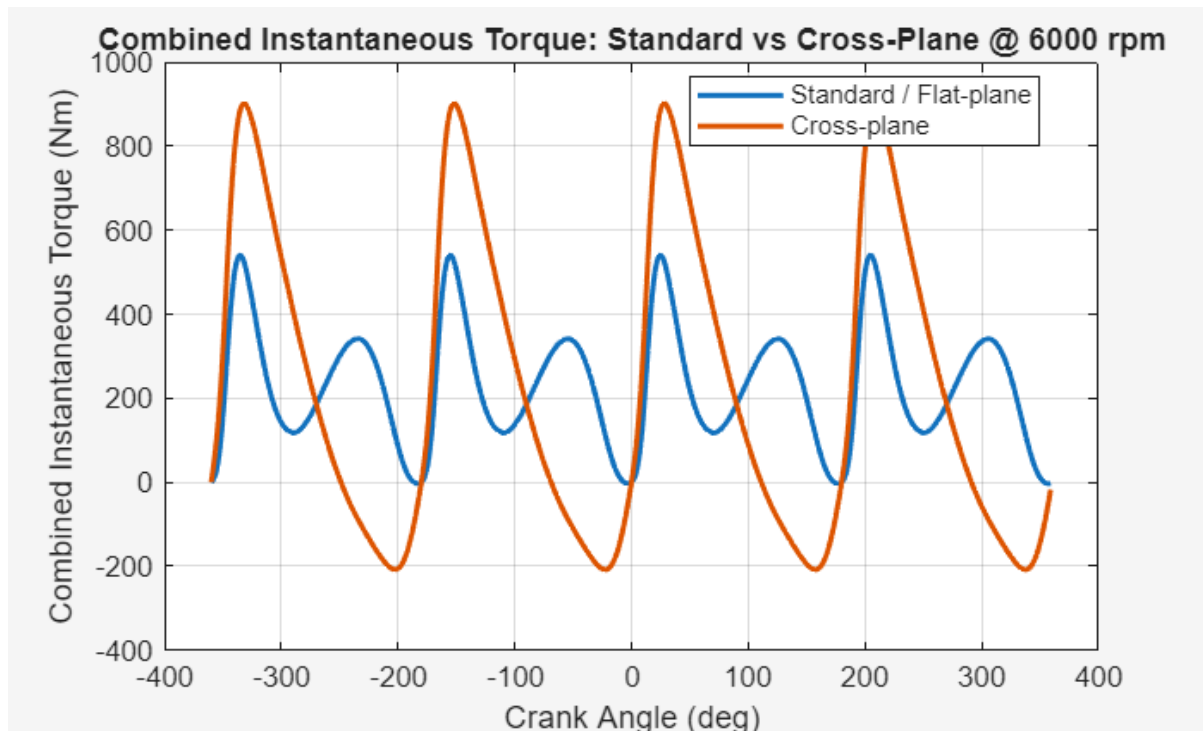


Figure 15: Combined Gas & Inertia Torque comparing flat plane and cross plane arrangements.

Table 9: Summary of instantaneous torque comparison for flat-plane and cross-plane crankshaft arrangements at 6000 rpm.

	Flat Plane	Cross Plane
Inertia Torque (peak-peak) (Nm)	911.424	41.843
Inertia Torque RMS(Nm)	321.258	14.803
Combined Torque (peak-peak) (Nm)	543.602	1110.158
Combined Torque RMS (Nm)	140.264	365.095
Negative Torque Fraction (%)	3.33	37.78

5. Conclusions

This report used the Ford 1.6 Sigma pressure data together with engine kinematics to assess the loading, stress and fatigue safety factors of key bottom-end components. At the original operating condition of 6000 rpm, the connecting rod midpoint, gudgeon pin and connecting rod bolt all showed fatigue safety factors above 3.0, indicating acceptable durability within the baseline speed range. However, extension of the analysis to 9000 rpm showed that the connecting rod bolt became the limiting component, with FSF falling to 1.337 as the inertia-driven separating load increased sharply.

The redesign study showed that bolt geometry, material selection and manufacturing method each contributed to improved fatigue performance. Reducing the wasted-shank diameter gave the greatest geometric benefit, while increased wasted length provided a smaller secondary improvement. Of the alternative materials assessed, maraging steel gave the highest numerical FSF, but SAE 4335M provided the most practical overall solution when strength improvement was balanced against cost. The addition of rolled threads increased the final SAE 4335M redesign from a FSF of 2.859 to 3.348 by shifting the local stress state in a more compressive direction without changing the stress range.

The instantaneous torque comparison showed that the cross-plane arrangement greatly reduced inertia torque ripple, but in this analytical model the flat-plane arrangement produced the lower total combined torque ripple once the gas torque contribution was included. Overall, the study shows that the original design is adequate within the baseline operating range, but that higher-speed operation requires targeted bolt redesign.

6. Word Count

2,573 – Excluding tables, figures, titles, appendices and in-text references.

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8. Appendices

Appendix 1 – Kinematics Calculations

The piston motion was calculated using the standard piston and crank kinematic equations provided. First the connecting rod angle was calculated from:

$$\varphi = \sin^{-1} \frac{(R \sin \theta)}{L}$$

where R is the crank radius, L is the connecting rod length and θ is the crank angle.

The piston displacement S from top dead centre (TDC) was then calculated as:

$$S = (R + L) - (R \cos \theta + L \cos \varphi)$$

Piston velocity was obtained from the first derivative of displacement with respect to time:

$$\frac{dS}{dt} = R\omega \sin \theta + \frac{R^2\omega}{2L} \sin 2\theta$$

and piston acceleration from the second derivative:

$$\frac{d^2S}{dt^2} = R\omega^2 \cos \theta + \frac{R^2}{L} \omega^2 \cos 2\theta$$

The reciprocating inertia force was the calculated from Newton's second law:

$$F_{inertia} = m_{rec} \times a$$

where m_{rec} is the reciprocating mass and a is acceleration.

This stage was necessary because inertia loading increases with the square of angular speed, meaning that at elevated engine speeds it can become comparable to, or greater than, the gas loading. This is particularly important for fatigue-critical components such as the connecting rod bolts.

Appendix 2: Gas Force, Net Force, Connecting Rod Force Calculations

The gas force acting on the piston was calculated from the in-cylinder pressure and piston area. Piston area was found from the bore using:

$$Area = \frac{\pi \times bore^2}{4}$$

The gas force was then calculated as:

$$F_{gas} = P_{cyl} \times Area$$

where P_{cyl} is the instantaneous cylinder pressure.

The net axial force acting on the piston was obtained from the combination of gas force and inertia force:

$$F_{net} = F_{gas} - F_{inertia}$$

Because the connecting rod is inclined to the cylinder axis for most of the crank cycle, the force acting in the rod was resolved as:

$$F_{rod} = \frac{F_{net}}{\cos \varphi}$$

Appendix 3: Pin and Bolt Stress Calculations

The stress at the conrod midpoint was calculated using the axial rod force and cross-sectional area at the midpoint section. Assuming a uniform axial load, direct stress was found from,

$$\sigma_{mid} = \frac{F_{rod}}{A_{mid}}$$

where A_{mid} is the cross-sectional area of the conrod midpoint. This produces a stress range over the engine cycle, from which the maximum and minimum stresses could be identified.

Gudgeon pin stress

The gudgeon pin stresses were calculated from the pin load generated by the piston force calculations. The peak pin loading condition was identified from the force trace, and the corresponding stress was then calculated at the selected pin section. The pin was assessed under the applied loading using bending and shear equations ($\sigma = \frac{My}{I}$; $\tau = \frac{F}{A}$) to determine the stress variation through the cycle. These stress values were then used to calculate the fatigue safety factor discussed in section 2.3.4.

Connecting rod bolt stress

The connecting rod bolt stresses were calculated from the total tensile load acting on the bolt, made up of the initial preload and the additional load caused by joint separation forces during operation. The separating load was determined from the difference between the inertia and gas forces acting at the big end, with only the tensile portion contributing to bolt loading. The share of this separating load carried by the bolt was then found using the bolt and joint stiffness relationship. The final bolt load was converted into stress using the tensile stress area of the bolt, allowing the maximum and minimum bolt stress over the cycle to be determined.

Appendix 4: Fatigue safety factor method

For each component, the maximum and minimum stresses were taken from the calculated stress history. These were then used to calculate the mean stress and stress range and allowable stress range:

$$\begin{aligned}\sigma_{mean} &= \frac{\sigma_{max} + \sigma_{min}}{2} \\ \sigma_{range} &= \sigma_{max} - \sigma_{min} \\ \sigma_{allow} &= 2\sigma_{EL}\left(1 - \frac{\sigma_{mean}}{UTS}\right)\end{aligned}$$

These values were used to assess fatigue behaviour using the Goodman method. The fatigue safety factor was determined by comparing the allowable stress with the stress range:

$$FSF = \frac{\sigma_{allow}}{\sigma_{range}}$$