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Analytical Assessment of Engine Internal Components

Contemporary Powertrain (ACHA6006)

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1 Introduction

Reciprocating engine bottom-end components work under combined loading, where combustion pressure and reciprocating inertia act together through the crank-slider mechanism. This means the highest component stress does not always happen at peak cylinder pressure alone, because the inertia term increases with engine speed squared and can either add to or oppose the gas load depending on crank angle. For this reason, the conrod, gudgeon pin and conrod bolts need to be assessed over the full engine cycle rather than at a single peak-pressure point. (Heywood, 2018)

This report evaluates the Ford Sigma 1.6 engine using the supplied pressure data, slider-crank kinematics and analytical stress methods from the lecture framework. Net piston force is found from the combination of gas and inertia forces, then converted into conrod axial load, gudgeon pin stress and conrod bolt load. Fatigue performance is then assessed using mean stress, stress range and Goodman-based allowable range, allowing all three components to be compared on the same basis.

The study then extends the pressure data to 9000 RPM to check whether the baseline bolt still has enough fatigue margin at the higher speed. Finally, the report compares standard and cross-plane crank phasing using instantaneous torque traces. This is important because crank phasing does not alter total cycle work, but it can change torque ripple, negative torque and likely torsional severity. Overall, the outcome gives both a component-level and system-level view of the engine design.

2 Methodology

2.1 Input Data & Assumptions

The analysis uses the supplied Ford Sigma 1.6 geometry, mass data and cylinder-pressure traces as the main inputs. The pressure map is treated as the combustion input, while bore, crank radius and conrod length define the crank-slider geometry. Reciprocating mass is used for piston inertia, and the rod, pin and bolt dimensions are then used for the later stress calculations. For this reason, the pressure data in Figure 1 becomes the foundation for all force and stress results that follow.

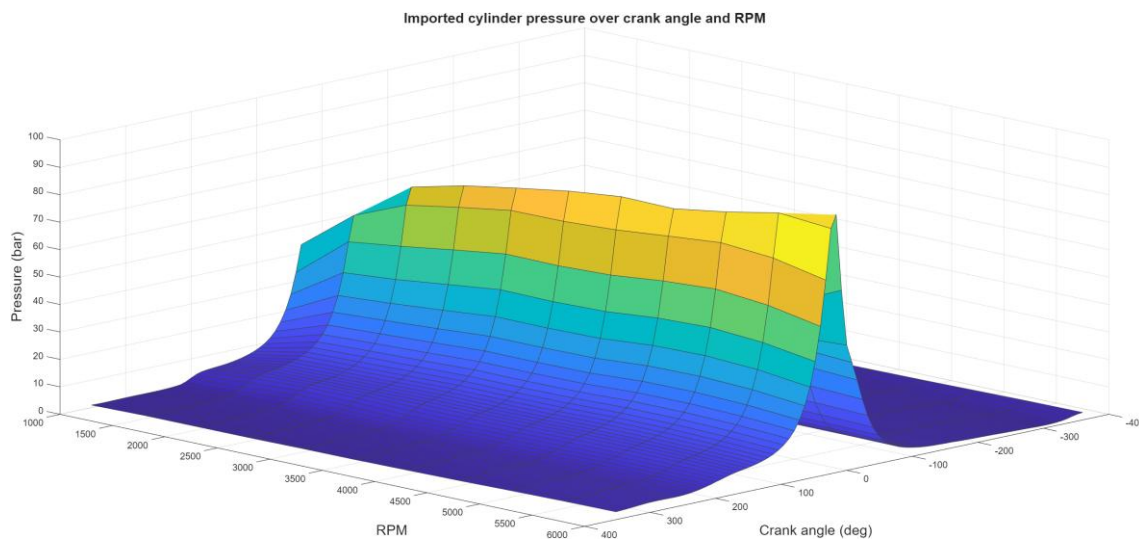


Figure 1: Cylinder pressure data for the Ford Sigma 1.6 engine.

Several simplifying assumptions are required. The stress calculations are analytical and based on idealised sections rather than finite-element stress fields, materials are treated as homogeneous and elastic, the Goodman method is linear, and the 9000 RPM case is created by linearly extrapolating the available pressure data rather than using measured values. These assumptions are acceptable for a comparative design study, but they should still be kept in mind when judging absolute safety margin, particularly for the bolt where the result moves close to the fatigue limit.

2.2 Kinematics and Force

Piston motion is defined using standard slider-crank relationships. Crank angle sets rod angle, piston position and piston acceleration, with the acceleration term containing both first-order and second-order effects. Figure 2 shows that acceleration changes strongly with crank angle and becomes much more severe as speed rises. This is important because inertia force is directly proportional to acceleration, so tensile loading near top dead centre can rise quickly even when cylinder pressure does not.

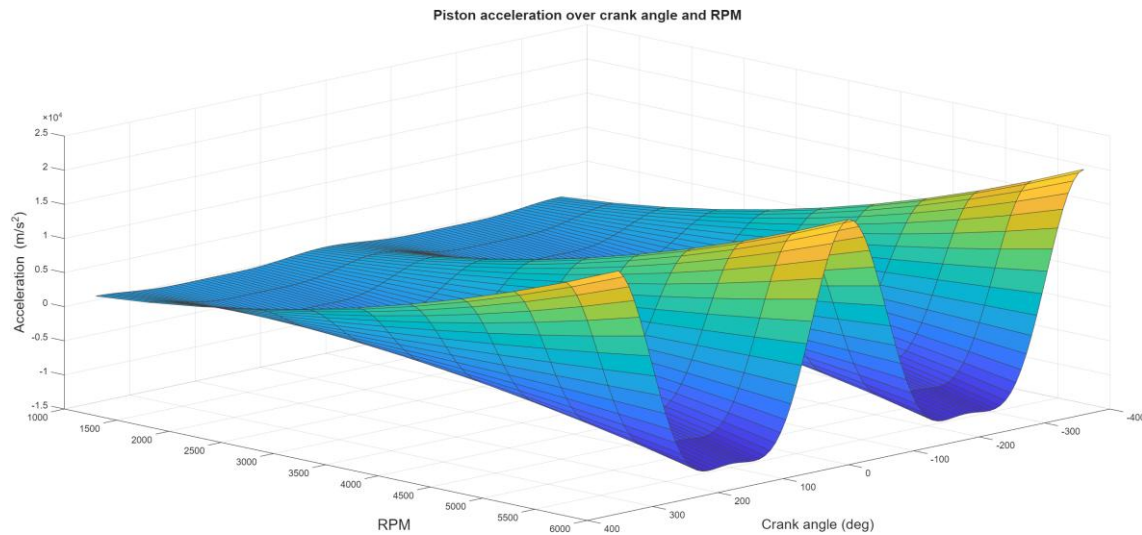


Figure 2: Piston acceleration calculated from engine geometry.

Gas force is calculated from cylinder pressure and piston crown area. Inertia force is then found from reciprocating mass and piston acceleration, and the two are combined to give net piston force. This net force is then resolved through the rod angle to find conrod axial force. The key point is that combustion-driven compression is strongest shortly after top dead centre, while the highest tensile loading appears near top dead centre of the exhaust and intake overlap region where pressure is low and inertia dominates. This change in loading direction is what makes different components critical at different parts of the cycle.

2.3 Fatigue Assessment

For each component, the calculated stress trace is reduced to maximum stress, minimum stress, mean stress and total stress range. These are then compared with a Goodman-based allowable range, modified where required for stress concentration. This is useful because it separates components which see high nominal stress but only small fluctuation from those which see damaging cyclic variation around an already high mean stress. (Pilkey et al., 2020)

The Goodman diagrams are particularly useful here because they show whether the actual stress range remains comfortably inside the allowable region or whether the design is working close to the fatigue boundary. This is more useful than quoting peak stress alone. A component can stay safe at a relatively high mean stress if the alternating part is small, while another can become critical at a lower peak value if the range is large. For this reason, the Goodman plots are used as the main fatigue comparison tool throughout the report.

3 Results & Analysis

3.1 Forces

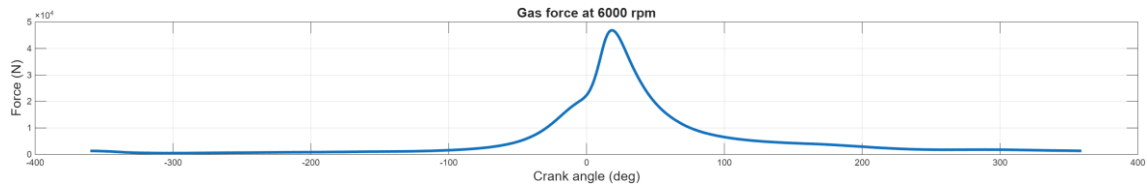


Figure 3: Gas forces at 6000 RPM.

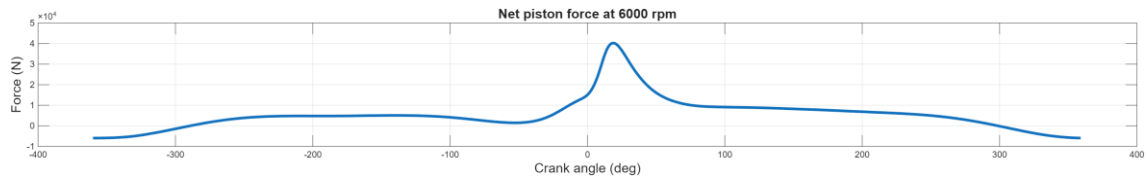


Figure 4: Net piston forces derived from both gas and inertia forces.

Table 1: Piston forces.

Maximum Gas Force (N)	46862.91
Minimum Gas Force (N)	405.74
Maximum Net Piston Force (N)	40253.52
Position at 6000RPM (° ATDC)	19
Minimum Net Piston Force (N)	-5948.64
Position at 6000RPM (° ATDC)	-358

Figures 3 and 4 show that the governing force behaviour is controlled by combustion compression during the early expansion phase and by inertia tension close to top dead centre. The maximum gas force is 46.86 kN, while the maximum net piston force is 40.25 kN at 19° and 6000 RPM. The corresponding maximum conrod axial force is 40.45 kN, showing that the rod-angle correction is small but still worth including for accuracy.

The minimum net piston force is -5.95 kN at -358° and 6000 RPM, almost exactly at top dead centre, where cylinder pressure has dropped and inertia reverses the loading into tension. This is an important result because it shows that one engine cycle creates two very different design cases: a high compression case driven mainly by combustion, and a smaller but still important tensile case driven mainly by inertia. Those two events then feed directly into the later fatigue assessment of the rod, pin and bolt.

3.2 Conrod Midpoint

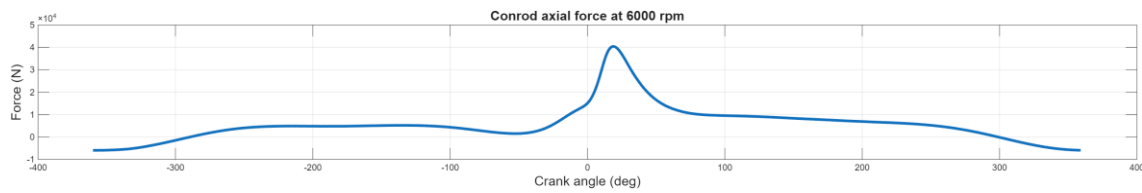


Figure 5: Conrod axial force from piston force.

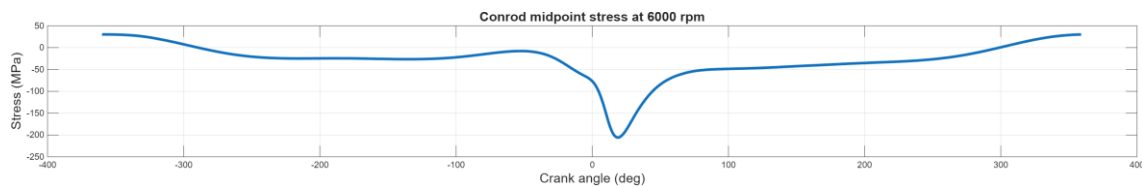


Figure 6: Conrod stress over crank angle.

Table 2: Conrod force and stress results.

Maximum Conrod Axial Force (N)	40451.78
Position at 6000RPM (° ATDC)	19
Minimum Conrod Axial Force (N)	-5948.97
Position at 6000RPM (° ATDC)	-358
Maximum Tensile Stress (MPa)	30.29
Position at 6000RPM (° ATDC)	-358
Maximum Compressive Stress (MPa)	205.98
Position at 6000RPM (° ATDC)	19
Factor of Safety	4.144

The conrod midpoint is assessed from the axial rod load using the mid-shank section properties and the relevant stress concentration treatment. Figure 5 shows that the conrod force follows the same overall pattern as the net piston force, with a strong compressive peak during combustion and a much smaller tensile peak near top dead centre. When converted to stress at the midpoint section, this gives a maximum compressive stress of -205.98 MPa and a maximum tensile stress of 30.29 MPa.

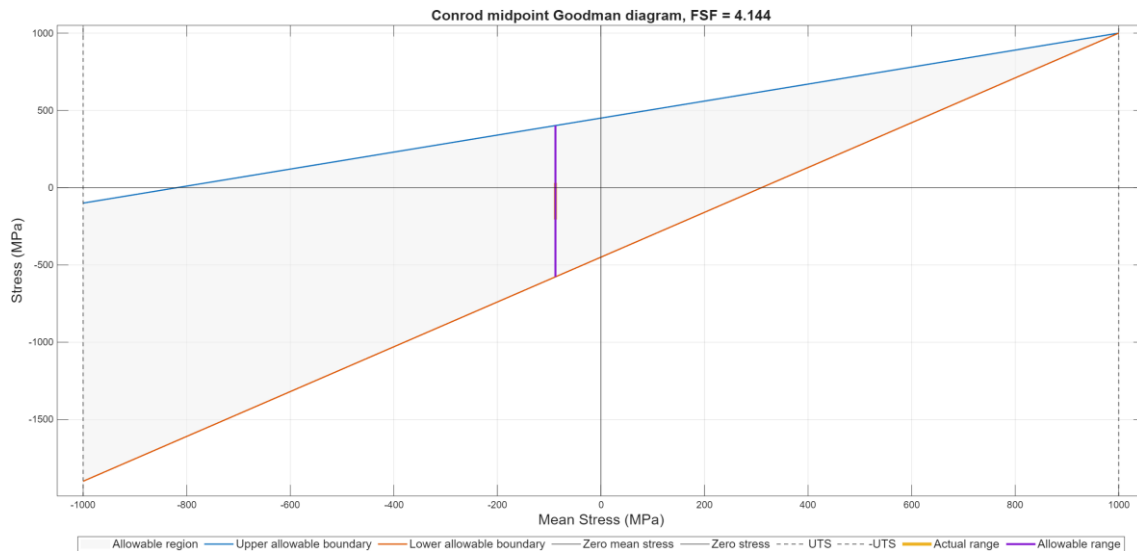


Figure 7: Conrod Goodman diagram.

This imbalance is physically reasonable. The rod acts mainly as a compression member during the power stroke, so the compressive event is set by cylinder pressure acting on the piston area. The tensile event is lower because it depends on reciprocating inertia rather than direct gas loading. The Goodman result in Figure 7 confirms that this section is very safe, with mean stress -87.84 MPa, stress range 236.27 MPa and allowable range 979.06 MPa, giving FSF 4.144. This suggests the midpoint section has a large fatigue margin and is likely conservative for the loading seen here.

3.3 Gudgeon Pin

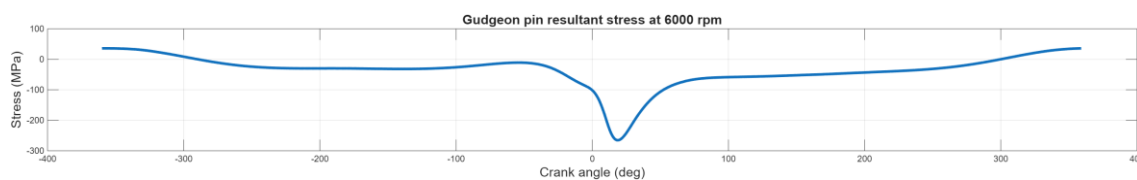


Figure 8: Gudgeon pin stress over crank angle.

Table 3: Gudgeon pin stress results.

Maximum Tensile Stress (MPa)	35.99
Position at 6000RPM (° ATDC)	-358
Maximum Compressive Stress (MPa)	265.48
Position at 6000RPM (° ATDC)	19
Stress Range (MPa)	301.47
Allowable Range (MPa)	1003.27
Factor of Safety	3.328

The gudgeon pin is loaded by the piston-side reaction and is assessed using the pin stress method at point A, where bending and shear combine to give the local resultant stress.

Figure 8 shows that the pin follows the same alternating compression-tension pattern as the rod, but with a larger stress response because the pin is working in local bending rather than mainly axial loading. The peak compressive resultant stress is -265.47 MPa and the peak tensile value is 36.00 MPa.

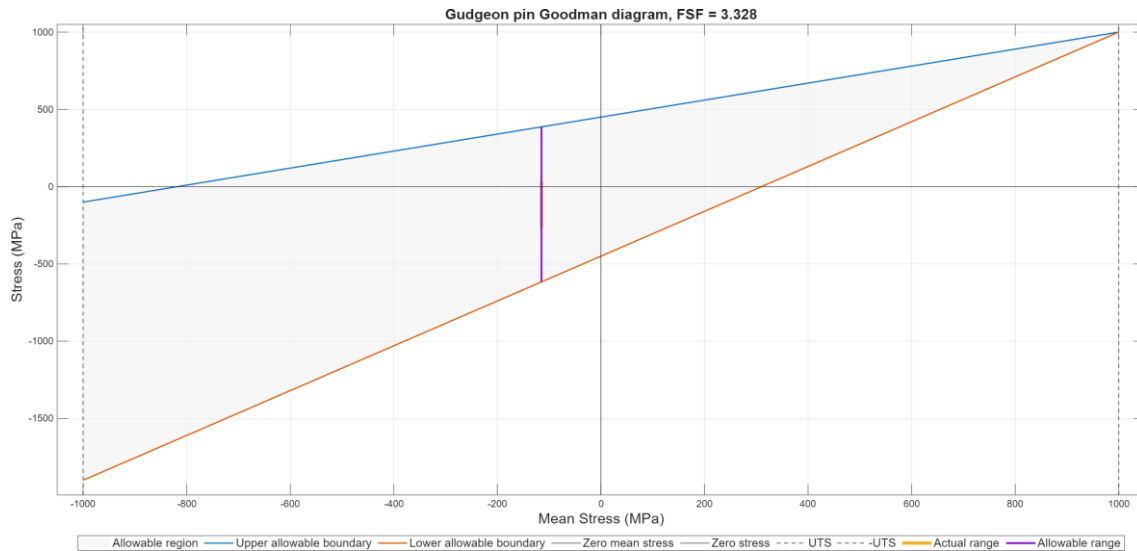


Figure 9: Gudgeon pin Goodman diagram.

The Goodman result in Figure 9 confirms that the pin remains comfortably safe, although it is less conservative than the conrod midpoint. The mean stress is -114.74 MPa and the stress range is 301.47 MPa, while the allowable range is 1003.27 MPa, giving FSF 3.328. This lower margin is expected because the pin sees a more severe local stress state, with bending and shear superimposed in a relatively small section. Even so, the fatigue margin is still strong enough to suggest that the baseline pin is not the limiting component in the analysed range.

3.4 Con-Rod Bolt

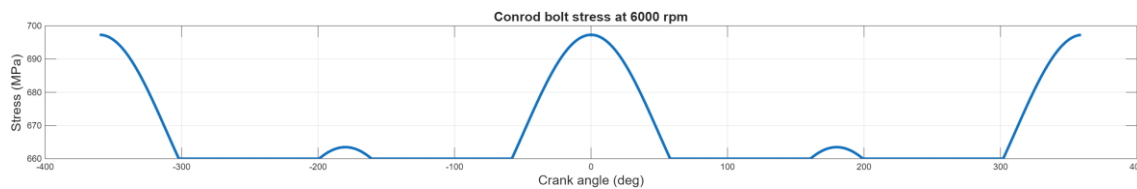


Figure 10: Conrod bolt stress over crank angle.

Table 4: Conrod bolt stress results.

Maximum Stress (MPa)	697.30
Position at 6000RPM (° ATDC)	-360
Maximum Stress (MPa)	660
Stress Range (MPa)	37.30
Allowable Range (MPa)	87.78
Factor of Safety	2.353

The conrod bolt behaves differently from the rod and pin because its baseline condition is set by preload. The bolt is already carrying a high tensile mean stress before the engine is running, and the operating load then appears as an extra fluctuating tensile component generated by inertia in the rod-cap assembly. This makes the fatigue problem much more mean-stress dominated than for the other parts. The result is shown in Figure 10, where the stress trace stays entirely tensile and oscillates around a high baseline value.

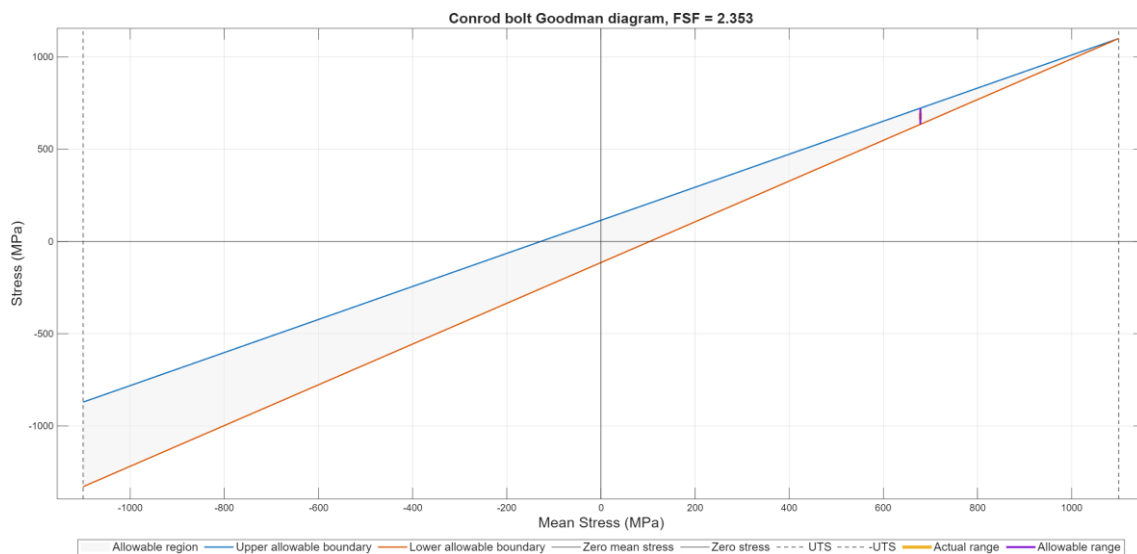


Figure 11: Conrod bolt Goodman diagram.

Although the stress range is only 37.30 MPa, the bolt sits much closer to fatigue failure than the other components because the mean stress is already very high and the allowable range is reduced by thread stress concentration. Figure 11 makes this clear: the allowable range is only 87.78 MPa, giving FSF 2.353. This remains above unity and is therefore acceptable at 6000 RPM, but it is already the lowest margin of the three components. For this reason, the bolt becomes the most critical part to reassess when engine speed is increased.

4 Increased RPM Analysis

To test that speed sensitivity, the supplied pressure data was linearly extrapolated to 9000 RPM, as shown in Figure 12, and the bolt analysis was repeated. Figure 13 shows that the additional bolt load rises strongly with speed because the inertia term scales with ω^2 and the bolt sees this loading directly through the rod cap. Figure 14 then shows the corresponding stress trace, which widens substantially compared with the 6000 RPM case even though the mean stress level changes only slightly.

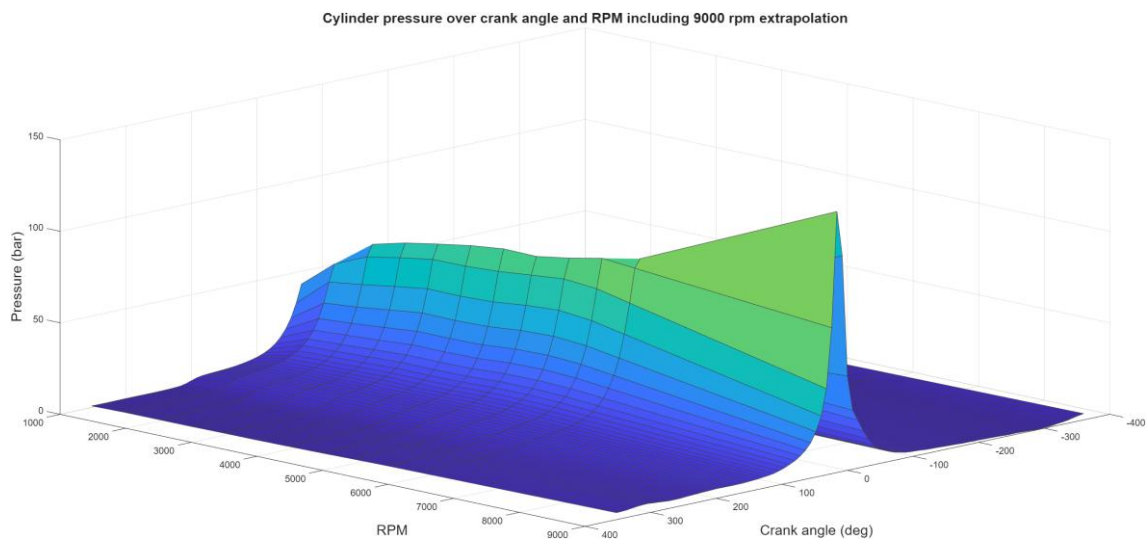


Figure 12: Extrapolated cylinder pressure data.

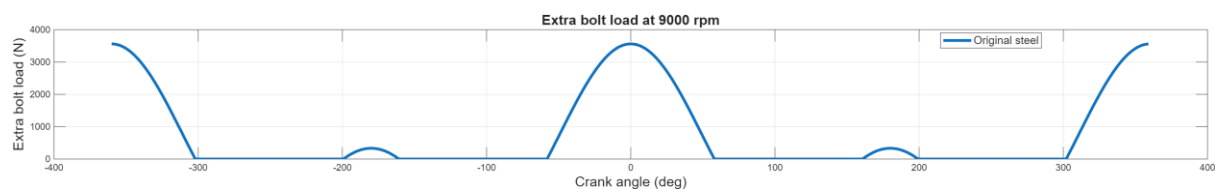


Figure 13: Additional bolt load over crank angle from 6000 RPM to 9000 RPM.

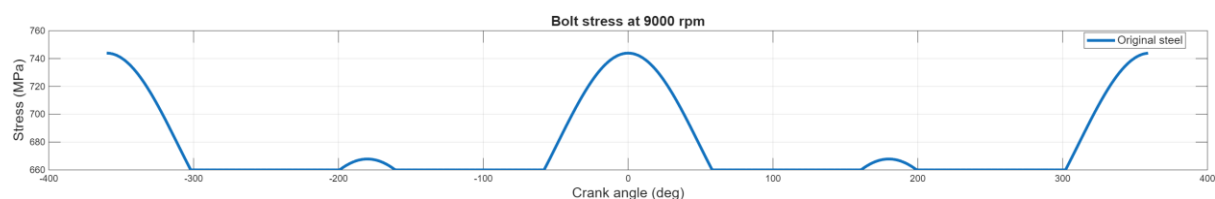


Figure 14: Bolt stress at 9000 RPM.

Table 5: Bolt stress results at 9000 RPM.

Maximum Stress (MPa)	743.93
Position at 6000RPM (° ATDC)	-360
Maximum Stress (MPa)	660
Stress Range (MPa)	83.93
Allowable Range (MPa)	82.92
Factor of Safety	0.988

The consequence is that the bolt stress range rises to 83.93 MPa, while the allowable range falls slightly to 82.92 MPa, giving FSF 0.988. Strictly, this is only marginally below unity, so the result is better described as borderline than as proof of immediate failure. Even so, that still means the original bolt no longer has a reliable fatigue margin at 9000 RPM within the limits of this model. In practical terms, this would reduce confidence in sustained high-speed use and would likely shorten component life, making redesign necessary if this operating speed is to be used regularly.

5 Con-Rod Bolt Redesign

The redesign study focuses on reducing the proportion of external load transferred into the bolt by reducing bolt stiffness. This follows the logic that a more compliant fastener attracts less dynamic tensile load, reducing stress range even if the mean stress remains high. The MATLAB study therefore compares a steel redesign and a titanium redesign, both selected to improve 9000 RPM performance while staying close to the baseline 6000 RPM fatigue factor where possible.

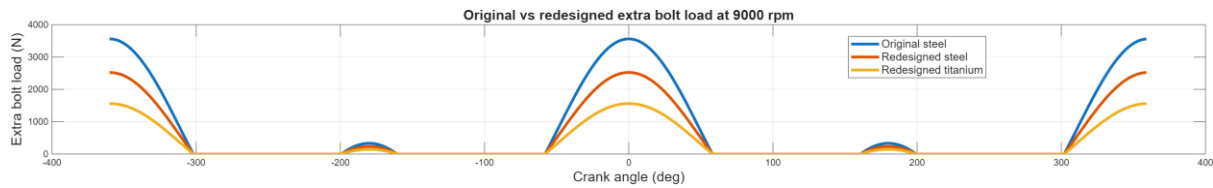


Figure 15: Additional bolt load from 6000 RPM to 9000 RPM, between different designs and materials.

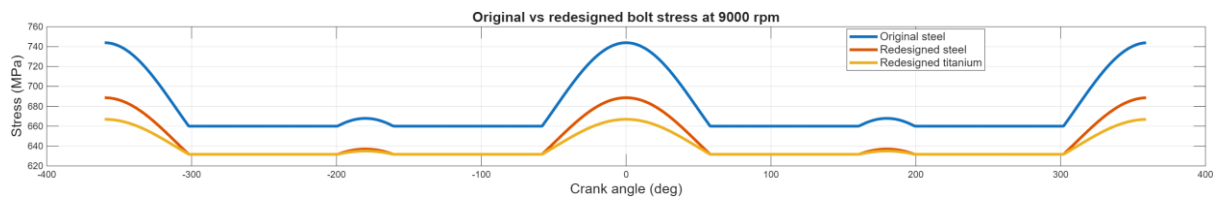


Figure 16: Bolt stress comparison between different designs and materials.

Table 6: Conrod bolt design comparison.

	Original Bolt (Steel)	Optimised Bolt (Steel)	Optimised Bolt (Titanium)
Mass (g)	7.835	12.367	6.979
Thread	M8x1.25	M8x0.75	M8x0.75
Stress Area (mm²)	42.430	44.332	44.332
Shank Diameter (mm)	7.350	7.513	7.513
Shank Length (mm)	21.26	22.000	22.000
Stiffness [K] (N/mm)	399153.000	256789.620	146370.080
Maximum Stress (MPa)	743.93	688.590	666.880
Factor of Safety (9000RPM)	0.988	1.6105	1.7802

For the steel redesign, bolt stiffness fell to 256789.62 N/mm and the joint load fraction fell to 0.1823. This improved the 9000 RPM fatigue factor of safety to 1.610, but only by increasing bolt mass to 12.367 g. That is a major drawback in a reciprocating assembly because any mass increase is undesirable, and in this case it also works against the inertia-driven nature

of the loading problem. While the steel option restores acceptable fatigue margin, it does so in a way that is hard to justify as the best overall engineering solution.

The titanium redesign reduced stiffness further to 146370.08 N/mm and produced the best combined result: FSF increased to 1.780 while bolt mass fell to 6.979 g, which is lower than the original 7.835 g model. This is the most balanced option explored, but it still does not recover the original 6000 RPM bolt margin. The redesign therefore improves the high-speed case to an acceptable level, rather than fully eliminating the penalty of running the engine at 9000 RPM.

6 Instantaneous Torque: Firing Order Comparison

Firing order and crank phasing affect how the single-cylinder torque pulses combine across the engine. Figures 17 to 19 show the single-cylinder gas, inertia and total torque components derived from the pressure and kinematic model. Gas torque dominates the positive part of the trace, while inertia torque produces the negative regions around the cycle. This gives the starting point for comparing how the same four firing events combine in standard and cross-plane crank arrangements.

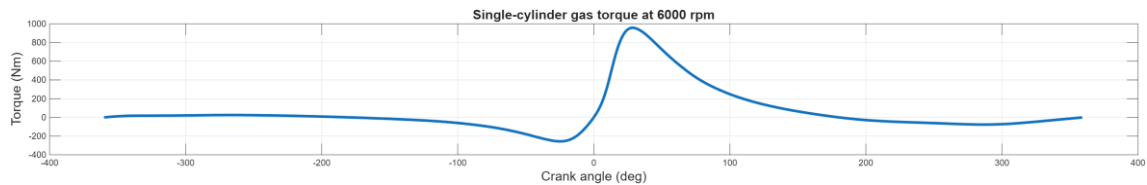


Figure 17: Gas torque generated by a single cylinder

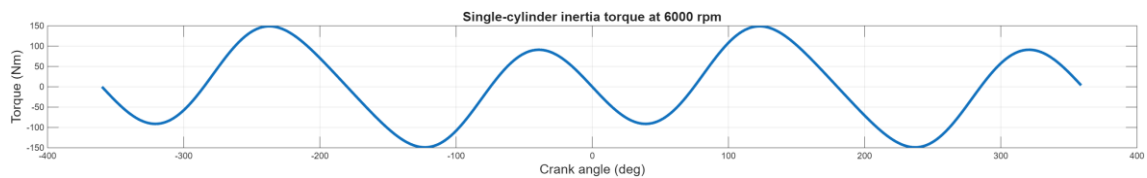


Figure 18: Inertia torque generated by a single cylinder.

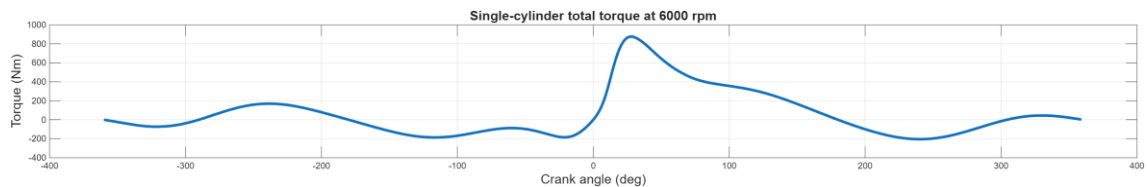


Figure 19: Total torque generated by a single cylinder.

Figures 20 to 22 show that both the standard and cross-plane configurations produce the same mean torque, work per cycle, power and model MEP at a given speed. At 6000 RPM, both layouts produce 229.82 Nm mean torque, 2887.98 J per cycle and 144.40 kW mean power. This is an important analytical check, because crank phasing changes the timing of torque delivery rather than the total energy released by combustion. The difference therefore appears in instantaneous behaviour, not in average output.

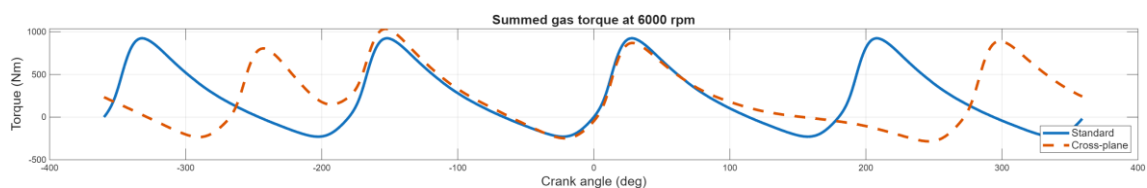


Figure 20: Gas torque generated by four cylinders in both standard and cross-plane configurations.

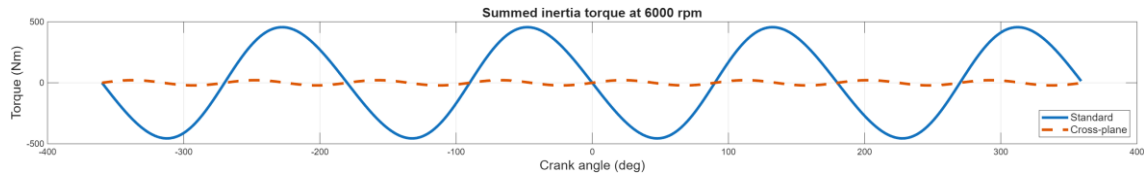


Figure 21: Inertia torque generated by all four cylinders in both standard and cross-plane configurations.

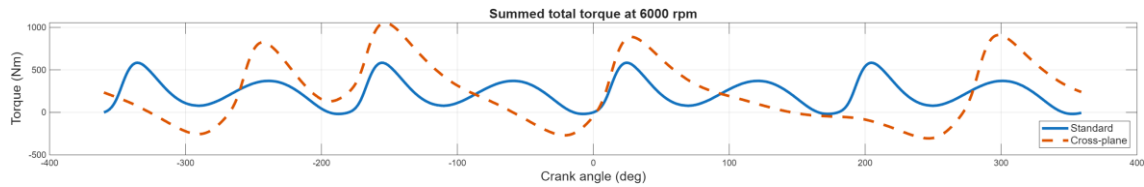


Figure 22: Summed force generated by four cylinders in both standard and cross-plane configurations.

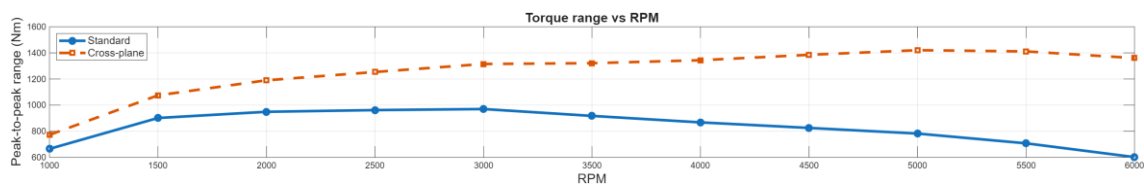


Figure 23: Single-cycle torque range for both standard and cross-plane configurations.

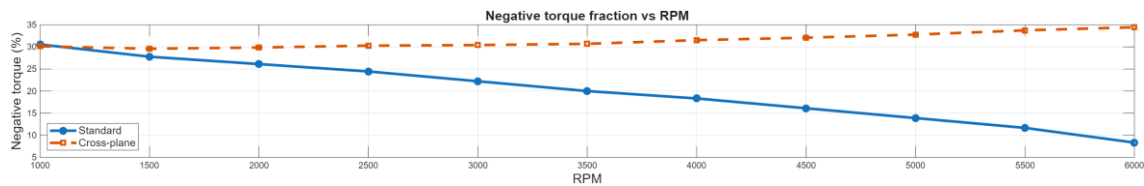


Figure 24: Negative torque percentage for both standard and cross-plane configurations.

Table 7: Firing order comparison.

Peak Torque, Standard (Nm)	583.80
Peak Torque, Cross-Plane (Nm)	1054.56
Minimum Torque, Standard (Nm)	-18.23
Minimum Torque, Cross-Plane (Nm)	-306.04
Torque Range, Standard (Nm)	602.02
Torque Range, Cross-Plane (Nm)	1360.60
Negative Torque, Standard (%)	8.33
Negative Torque, Cross-Plane (%)	34.44

That difference is large in this model. At 6000 RPM the standard crank has a torque range of 602.02 Nm and RMS fluctuation of 162.50 Nm, whereas the cross-plane configuration increases to 1360.60 Nm and 365.94 Nm respectively. Negative torque fraction also rises from 8.33% to 34.44%. Figures 23 and 24 therefore show that, for this engine model, the

standard crank gives much smoother instantaneous torque delivery while the cross-plane layout gives far greater fluctuation. That would likely make the standard arrangement better for refinement and lower torsional loading, while the cross-plane arrangement may still suit applications where a different pulse character is desirable. (Dhamejanl, Menon and Parikh, 1986)

7 Discussion & Conclusions

The overall component ranking is clear. The conrod midpoint is the safest of the three assessed parts with FSF 4.144, followed by the gudgeon pin at 3.328, while the conrod bolt is already the weakest baseline component at 6000 RPM with FSF 2.353. The rod and pin are governed mainly by combustion-driven compression, so they retain a large reserve despite the cyclic loading. By contrast, the bolt is governed by high tensile mean stress from preload plus a speed-sensitive inertia-driven fluctuation. That difference in loading mechanism is the main engineering finding of the report.

The 9000 RPM extension strengthens this conclusion. Extrapolating the pressure data pushes the original bolt to FSF 0.988, showing that the baseline bolt is not suitable for sustained operation at the higher speed within the assumptions of the model. Both redesigns recover safety above unity, but the steel solution does so with an unacceptable mass penalty, whereas the titanium solution gives the best balance of lower stiffness, reduced mass and improved FSF. Even then, the redesign does not fully recover the original 6000 RPM fatigue margin, so the higher-speed condition remains less forgiving than the baseline case.

At system level, the torque study shows that standard and cross-plane crank arrangements are equivalent in mean torque and power but not in instantaneous delivery. The standard crank gives much lower torque range, RMS fluctuation and negative torque throughout the analysed speed range. That points to lower torsional severity in the present model, although the wider real-world effect on traction, refinement and durability would need a fuller engine-vehicle study. The main limitations remain the use of extrapolated 9000 RPM pressure data, linear Goodman fatigue, analytical stress models and assumed titanium properties. Within those limits, the report supports three clear conclusions: the rod midpoint and gudgeon pin are adequately sized in the analysed range, the conrod bolt is the true fatigue-limiting baseline component, and titanium-based bolt compliance is the most promising redesign direction of those explored. (Kadge, 2021)

8 References

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