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Analytical Assessment of Engine Internal Components

Contemporary Powertrains (ACHA6006)

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Table of Contents

1	Introduction	4
2	Engine loading and force paths.....	5
2.1	Forces acting through the components	5
2.2	Load transfer through the piston, gudgeon pin and connecting rod	6
2.3	Differences in fatigue behaviour between the assessed components	7
3	Fatigue assessment of the original components	8
3.1	Connecting rod midpoint.....	8
3.2	Gudgeon pin	9
3.3	Connecting rod bolt.....	11
3.4	Comparison of component severity	12
4	Connecting rod bolt behaviour at 9000 rpm	13
4.1	Effect of engine speed on bolt loading.....	13
4.2	Performance of the original bolt at 9000 rpm	14
5	Bolt improvement study and final design selection	16
5.1	Design logic behind the improvement routes.....	16
5.1.1	Shank diameter changes	16
5.1.2	Shank length changes.....	17
5.1.3	Rolled thread improvement.....	20
5.1.4	Material selection	22
5.2	Final bolt design selection	24
6	Torque and Power comparisons	26
6.1	Understanding the differences between the two types of crankshafts.....	26
6.2	Torque range comparison between crank designs	27
6.3	Comparison of calculated metrics to that of the manufacturer	28
6.4	Instantaneous torque comparisons and engineering discussion	29

7	Conclusions	31
8	References	32
Graph 1	- Surface plot showing the Net Force seen by the internal components of the engine	5
Graph 2	-Piston acceleration overlaying the 1 st and 2 nd order components and combined acceleration	6
Graph 3	- Forces on the internal engine components at 6000rpm	6
Graph 4	- FSF per RPM for the Conrod Midpoint.....	8
Graph 5	- Conrod Midpoint Goodman Diagram at 6000 RPM.....	9
Graph 6	- FSF per RPM for the Gudgeon Pin.....	9
Graph 7	- Gudgeon Pin Goodman Diagram at 6000 RPM.....	10
Graph 8	- FSF per RPM for a Conrod Bolt up to 6000 RPM	11
Graph 9	- Conrod Bolt Goodman diagram at 6000 RPM	12
Graph 10	- Surface plot showing the Cylinder pressures extrapolated to 9000 RPM	13
Graph 11	- FSF for the Conrod Bolt up to 9000 RPM.....	14
Graph 12	- Goodman Diagram for the Conrod Bolt at 9000 RPM	15
Graph 13	- Waisted Shank FSF for different diameters compared to the clamp force	16
Graph 14	- Goodman Diagram showing the FSF for the Bolt modified Shank waist at 9000 RPM	17
Graph 15	- FSF comparison for the bolt plotting different shank lengths and diameters.....	17
Graph 16	- Remaining clamp force for the bolt shank length and waist redesign at 9000 RPM.....	18
Graph 17	- Goodman diagram for the selected shank length and diameter.....	19
Graph 18	- Comparison of cut vs rolled threads for different bolt lengths and diameters.....	20
Graph 19	- Bar chart comparing the cut threads vs the rolled threads for the selected bolt geometry	21
Graph 20	- Goodman Diagram for the new Bolt with rolled threads	21
Graph 21	- Bar chart comparing the FSF for different materials and different bolt specifications.....	23
Graph 22	- Goodman diagram for the final selected bolt redesign specifications	24
Graph 23	- Torque comparison for the cross-plane and flat-plane crank	27

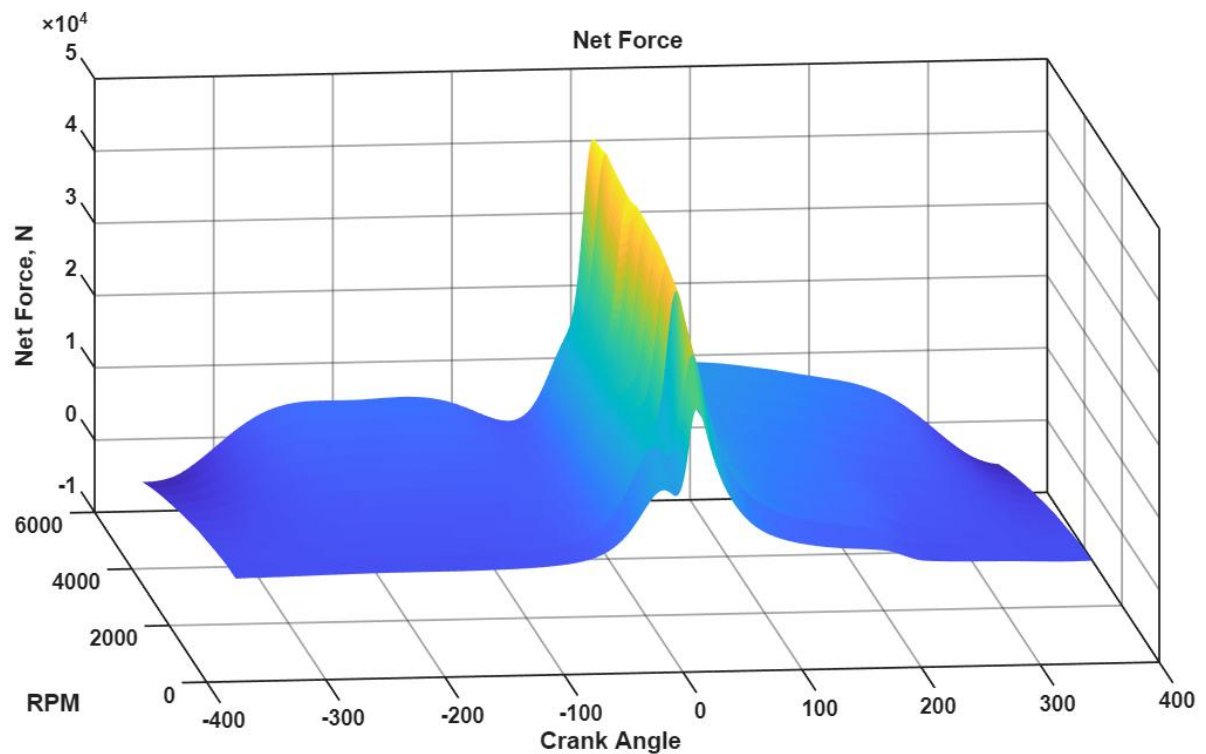
Graph 24 - Calculated Power and Torque compared to the Ford Data	28
Graph 25 - Instantaneous torque at 6000 RPM.....	29
Graph 26 - Instantaneous torque at 9000 RPM.....	29
Table 1 - Stress characteristics applied per component.....	7
Table 2 - FSF and specification for all components at 6000 RPM.....	12
Table 3 - Selected material specification	22
Table 4 - Comparison of the FSF of the original Bolt and the redesigned Bolt.....	25
Figure 1 – Flat-plane crank.....	26
Figure 2 - Cross-plane crank.....	26

1 Introduction

As engine development for motorsport and high-performance automotive applications continues to push operating speeds beyond the original design specification, the structural integrity of bottom-end components becomes increasingly important. The Ford 1.6 Ecoboost GDTI engine provides as basis for a production unit whose internal components must be analytically evaluated to determine whether they remain suitable when subjected to loading conditions beyond their intended operating range. The connecting rod midpoint, gudgeon pin and connecting rod bolt are assessed as the primary load-bearing components of the crank-slider mechanism, while all three share the same force path from the piston crown to the crankshaft, each experiences a fundamentally different loading mode and therefore a different fatigue risk. Connecting rod bolt failure is a well-documented consequence of high-speed operation, where inertia-driven cap-opening loads grow with the square of engine speed and rapidly erode the fatigue margin of a preloaded joint (Robert L. Norton, 1998). This report quantifies the forces acting on these components, calculates fatigue safety factors using the Goodman criterion, identifies the governing fatigue-critical component, extends the bolt assessment to 9000 rpm using linear extrapolation of the pressure data, investigates redesign changes to recover fatigue margin at the higher speed, and compares the instantaneous torque behaviour of flat-plane and cross-plane crankshaft configurations for different engineering applications.

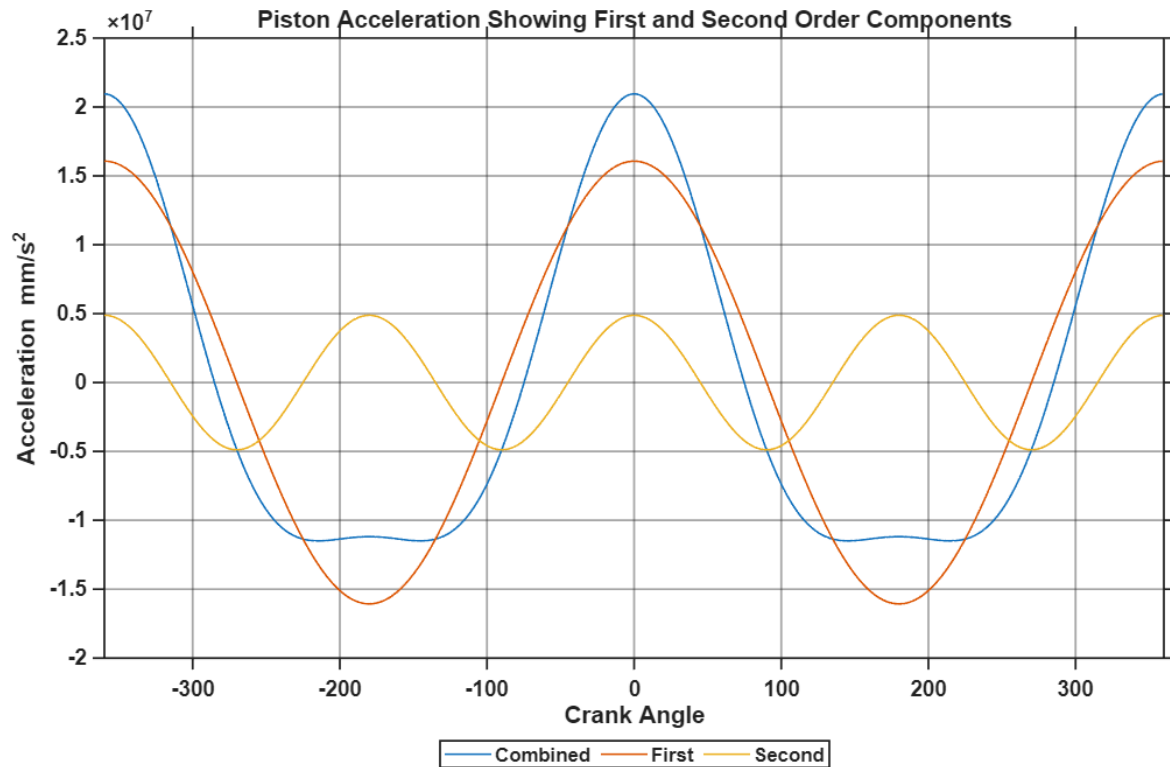
2 Engine loading and force paths

2.1 Forces acting through the components



Graph 1 - Surface plot showing the Net Force seen by the internal components of the engine

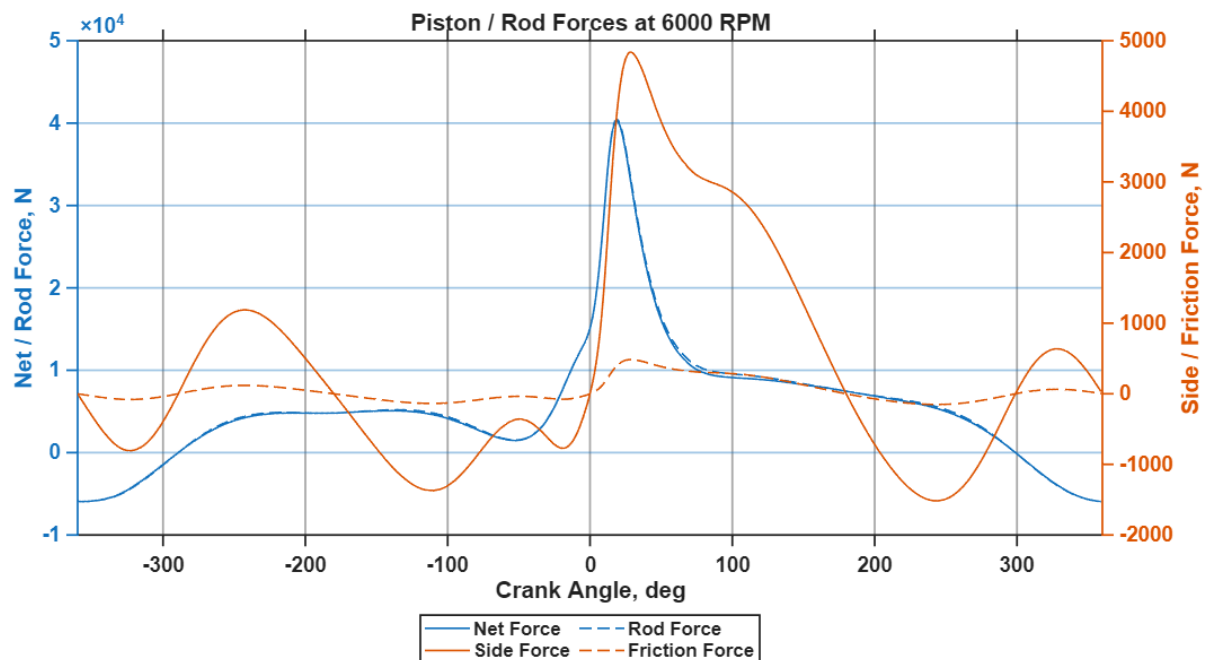
Figure 1 shows that the net piston force is strongly crank-angle dependent and combines a narrow high compressive peak around combustion with lower-magnitude tensile and compressive regions through the rest of the cycle. This confirms that the piston assembly is not loaded by gas pressure alone, the applied force varies continuously with the interaction between combustion loading and reciprocating inertia.



Graph 2 -Piston acceleration overlaying the 1st and 2nd order components and combined acceleration

Figure 2 shows that the inertia term is non-sinusoidal because the piston acceleration contains both first- and second-order components, so the cyclic loading becomes asymmetric through the cycle and increasingly speed-sensitive as RPM rises.

2.2 Load transfer through the piston, gudgeon pin and connecting rod



Graph 3 - Forces on the internal engine components at 6000rpm

The forces graph shows the resulting load transfer through the mechanism at 6000 rpm. The rod force follows the same overall trend as the net piston force, confirming that the main load path is from the piston crown, through the gudgeon pin, into the connecting rod and then to the crankshaft. However, rod angularity also generates a significant side force on the piston skirt, particularly around the main load peak, while friction remains much smaller in magnitude. The assembly therefore carries both axial and lateral reactions rather than behaving as a purely one-dimensional system.

2.3 Differences in fatigue behaviour between the assessed components

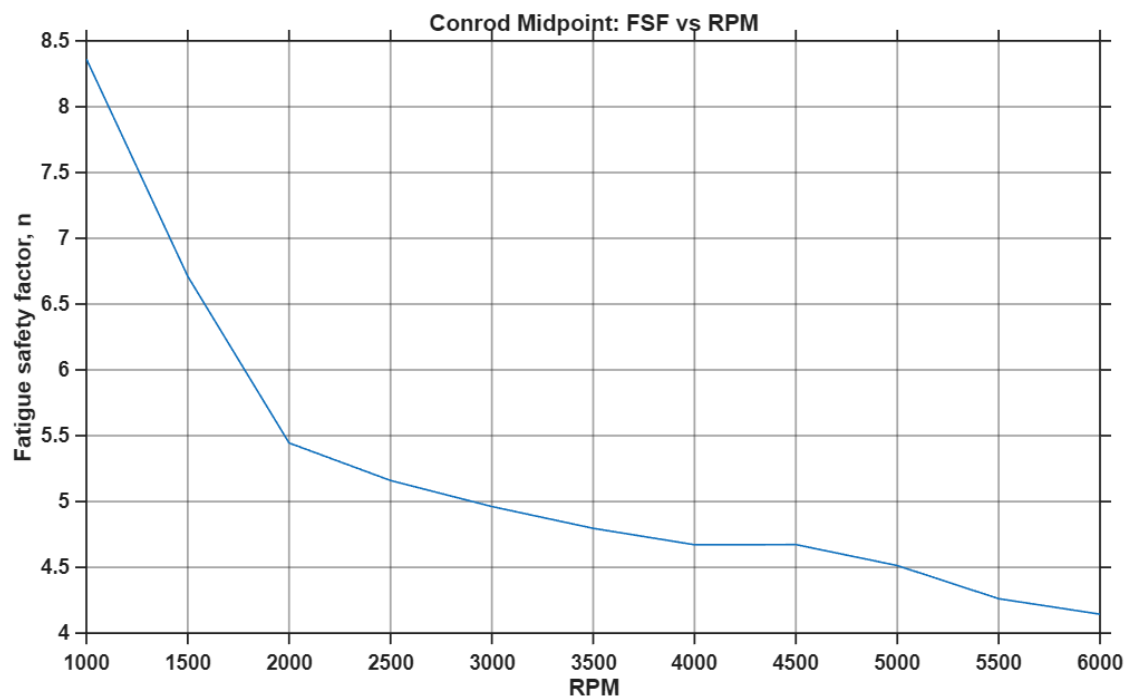
Table 1 - Stress characteristics applied per component

Component	Dominant loading mode	Mean stress characteristic	Governing risk
Rod midpoint	Axial tension/compression	Low/moderate	Alternating axial stress
Gudgeon pin	Bending with local shear/contact reaction	Reversing	Cyclic bending fatigue
Conrod bolt	Preloaded axial tension + cyclic opening load	High tensile mean stress	Fatigue at critical section/thread root

Table 1 summarises the fatigue implications of this load path. The rod midpoint is dominated by axial tension and compression, the gudgeon pin by cyclic bending with local support reactions, and the conrod bolt by high preload with a superimposed opening load. These differences in loading mode and mean stress condition explain the different fatigue sensitivities assessed in Section 3.

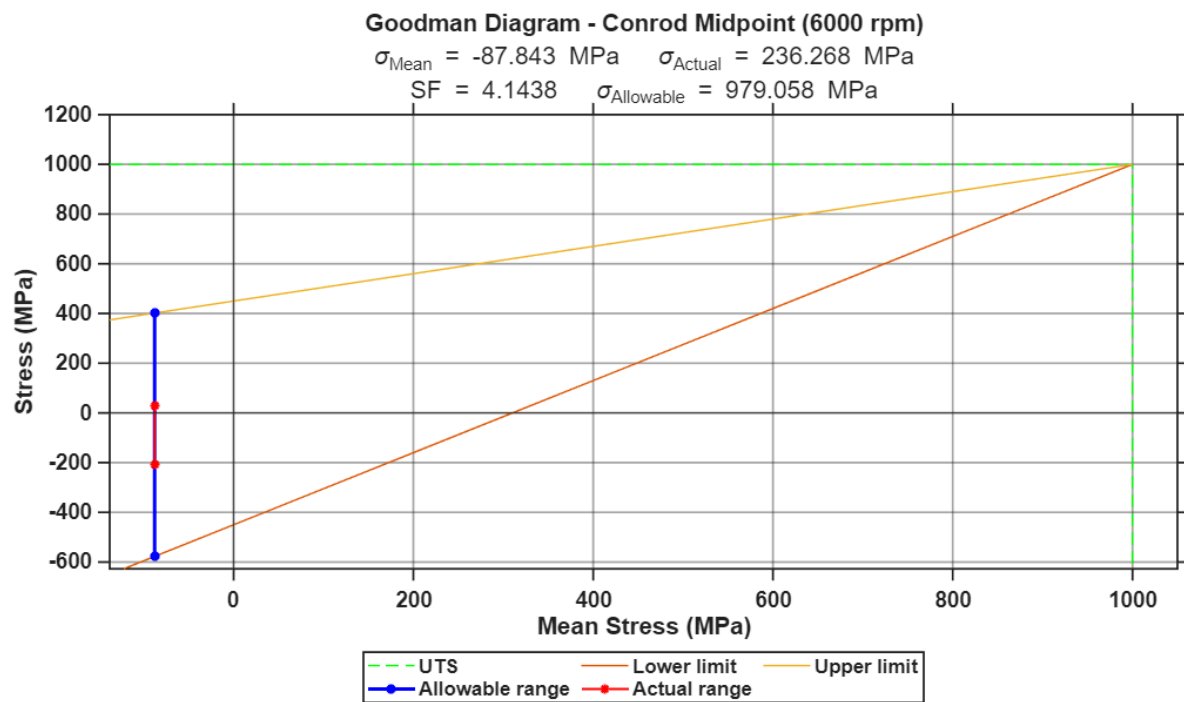
3 Fatigue assessment of the original components

3.1 Connecting rod midpoint



Graph 4 - FSF per RPM for the Conrod Midpoint

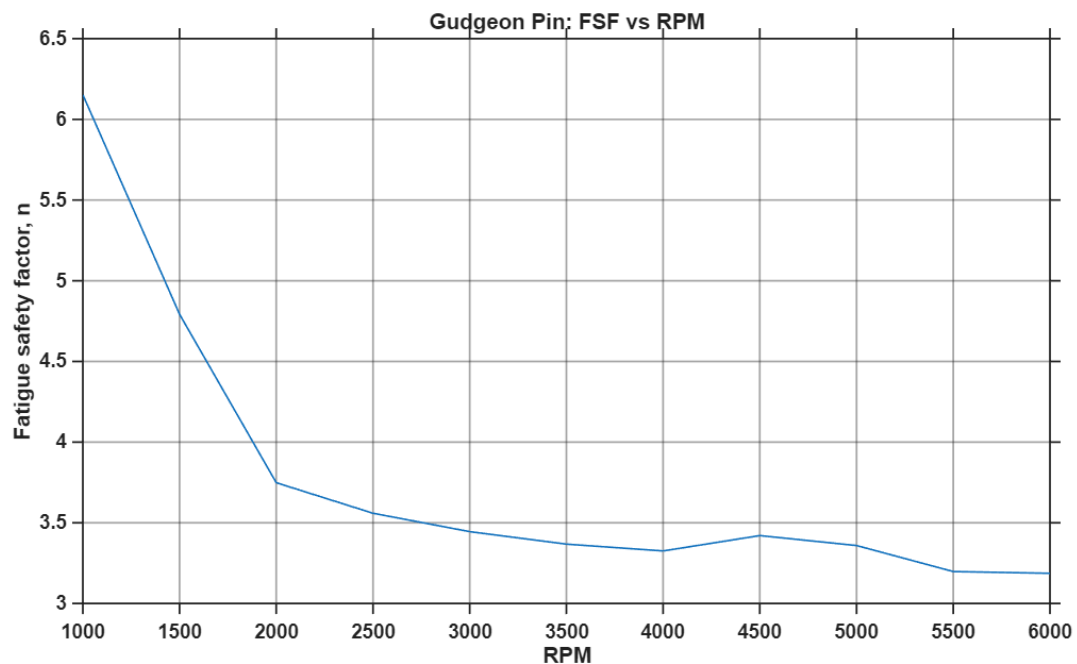
The conrod midpoint is the least critical component because its loading is predominantly axial, cycling between combustion-driven compression and inertia-driven tension. (Bailey, 1968). Its Fatigue Safety Factor (FSF) falls from approximately 8.4 at 1000 rpm to 4.14 at 6000 rpm, but the margin remains high throughout the operating range.



Graph 5 - Conrod Midpoint Goodman Diagram at 6000 RPM

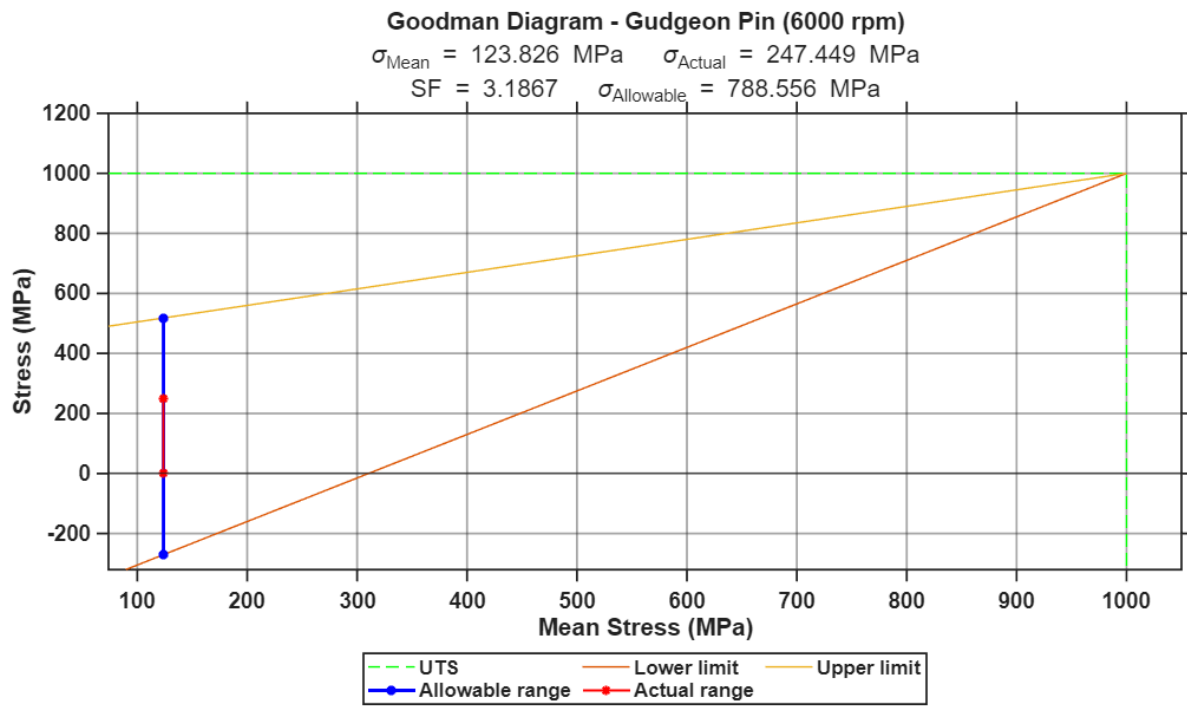
At 6000 rpm the Goodman result gives an alternating stress of 236.3 MPa within an allowable range of 979.1 MPa, and the negative mean stress is beneficial because it increases the permissible fatigue range. The rod midpoint can therefore be regarded as conservative rather than fatigue limited.

3.2 Gudgeon pin



Graph 6 - FSF per RPM for the Gudgeon Pin

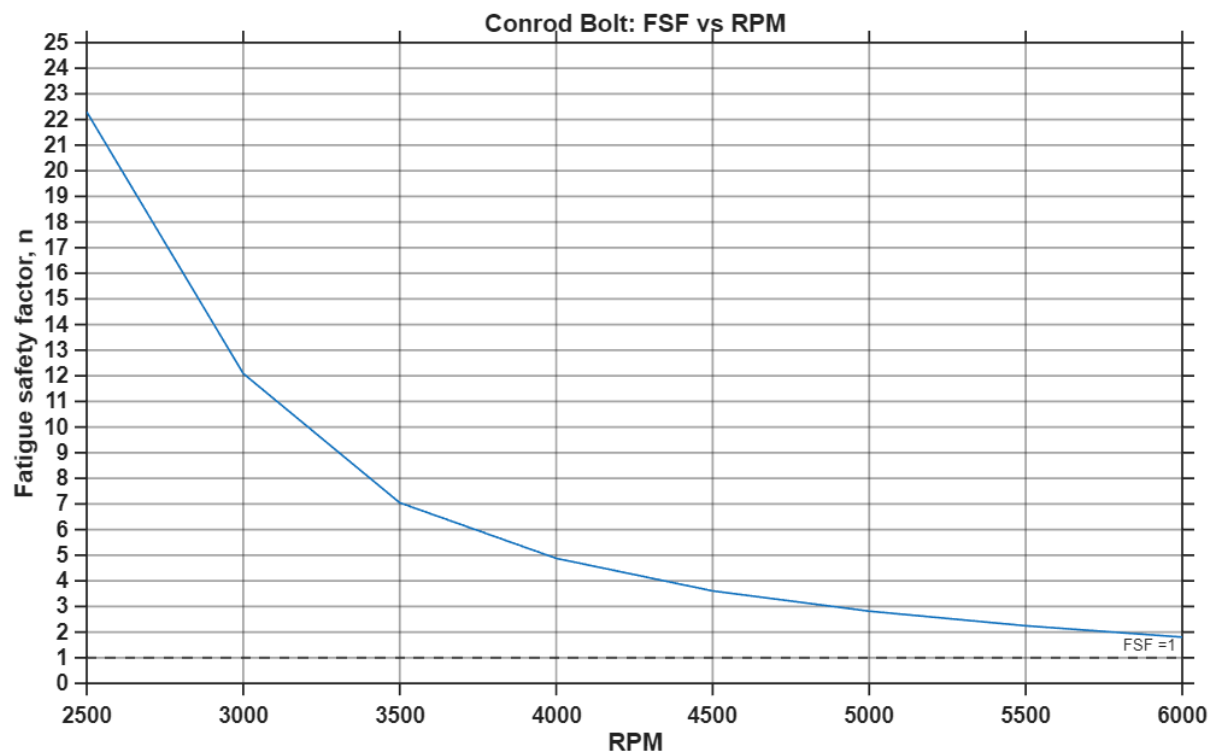
The gudgeon pin is more fatigue-sensitive because it is bending-dominated rather than purely axial. Combustion and inertia loads act in opposite directions, creating a reversing bending condition with local support reactions (Roark et al., 2002a), so the FSF remains below that of the rod across the full speed range. It decreases from about 6.1 at 1000 rpm to 3.19 at 6000 rpm.



Graph 7 - Gudgeon Pin Goodman Diagram at 6000 RPM

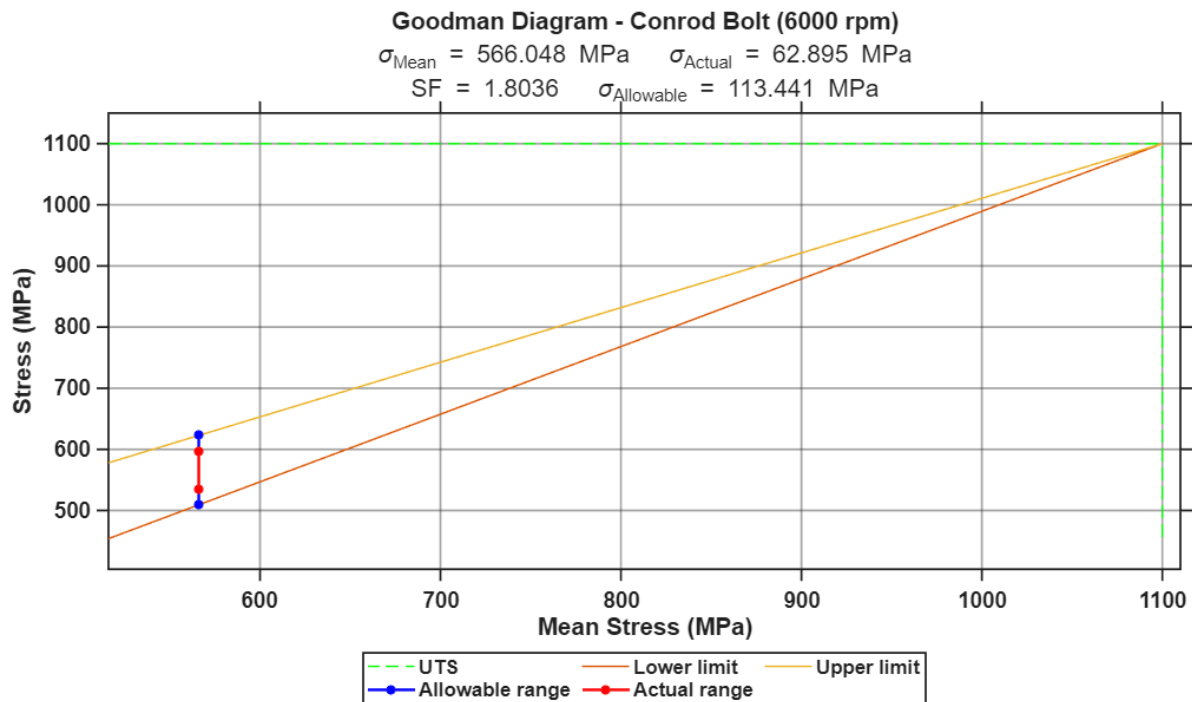
At 6000 rpm the Goodman diagram gives a mean stress of 123.8 MPa and an alternating stress of 247.4 MPa, with an allowable range of 788.6 MPa. The positive mean stress is less favourable than the rod condition, but the remaining margin still indicates an acceptable and moderately conservative design.

3.3 Connecting rod bolt



Graph 8 - FSF per RPM for a Conrod Bolt up to 6000 RPM

The conrod bolt is the governing component. Unlike the rod and pin, it operates as a preloaded tensile member, so the inertia-driven opening load is superimposed onto an already high tensile mean stress. The valid FSF trend is presented only above 2500 rpm because the low-speed region gives artificially high values when gas loading exceeds the opening load. Within the valid range, FSF falls sharply from 22.2 at 2500 rpm to 1.80 at 6000 rpm, showing much greater RPM sensitivity than the other components. The FSF for the bolt is consistent with the values presented by (Jia et al., 2025) and (Griza et al., 2009) validating the method used.



Graph 9 - Conrod Bolt Goodman diagram at 6000 RPM

At 6000 rpm the Goodman result gives a mean stress of 566.0 MPa, an alternating stress of 62.9 MPa and an allowable range of only 113.4 MPa. Although the alternating stress is relatively low, the high tensile mean stress severely restricts fatigue capacity.

3.4 Comparison of component severity

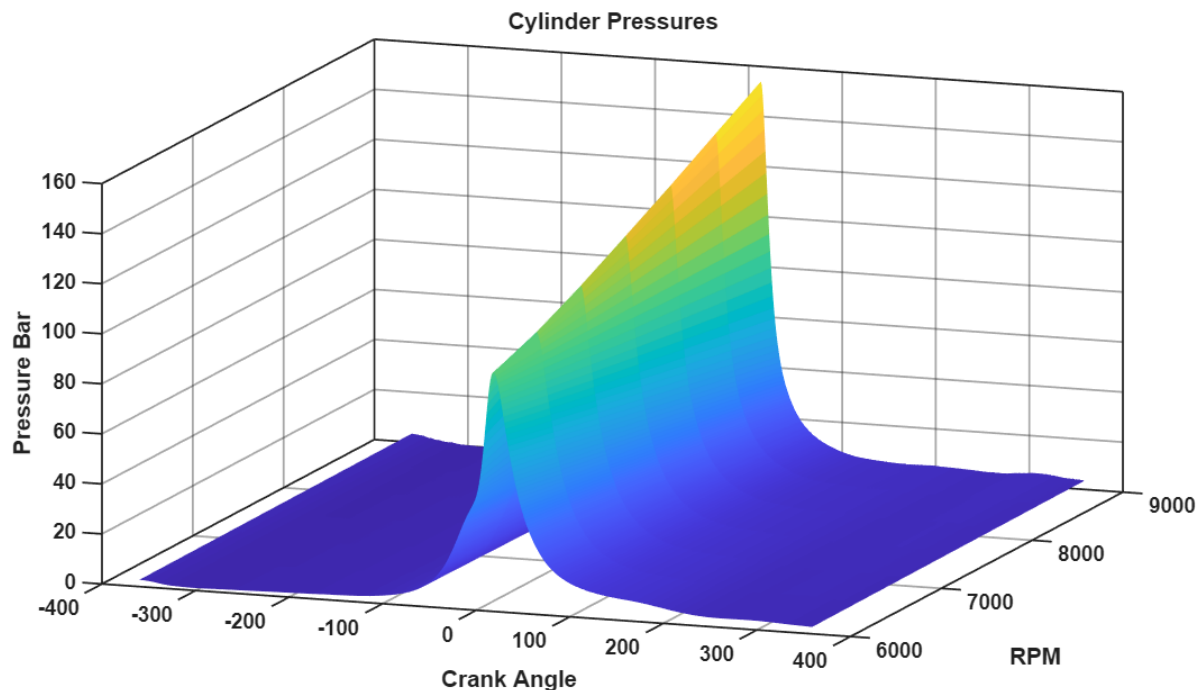
Table 2 - FSF and specification for all components at 6000 RPM

Component	FSF at 6000 rpm	Trend with RPM	Dominant loading mode	Design judgement
Conrod midpoint	4.14	Gradual decrease with increasing RPM	Axial tension and compression	Conservative, over-safe
Gudgeon pin	3.19	Moderate decrease with increasing RPM	Reversing bending	Acceptable, moderately conservative
Conrod bolt	1.80	Steep decrease with increasing RPM	Preloaded cyclic tension	Critical, redesign priority

At 6000 rpm the bolt gives the lowest FSF, compared with 3.19 for the gudgeon pin and 4.14 for the rod midpoint, so it was identified as the most critical and most design-sensitive component for further redesign.

4 Connecting rod bolt behaviour at 9000 rpm

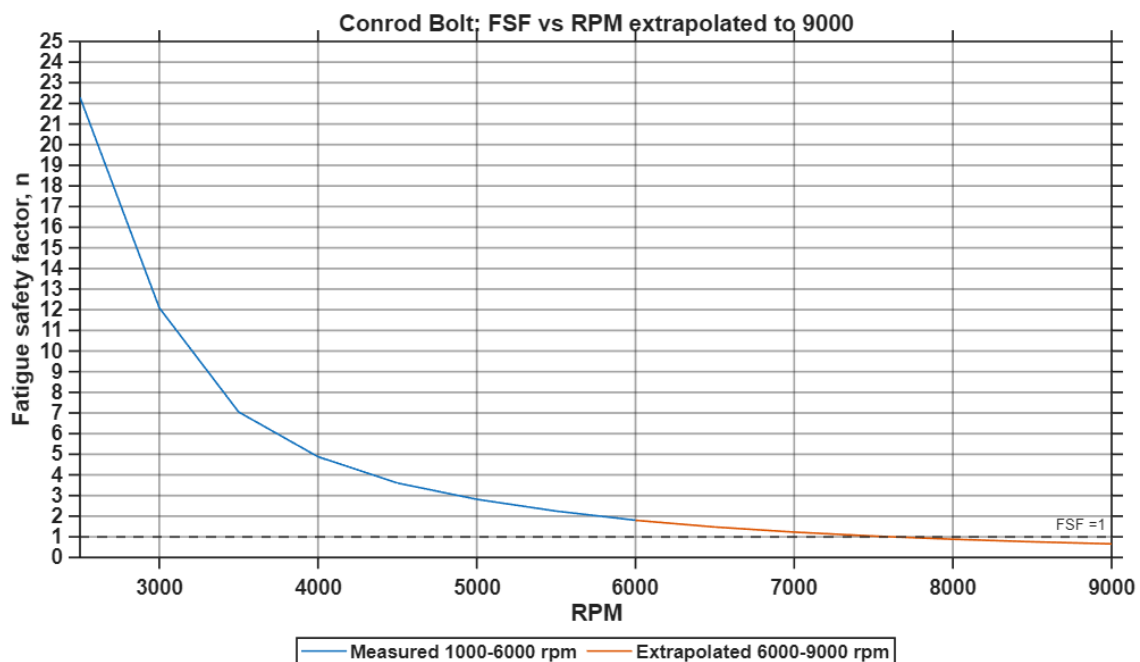
4.1 Effect of engine speed on bolt loading



Graph 10 - Surface plot showing the Cylinder pressures extrapolated to 9000 RPM

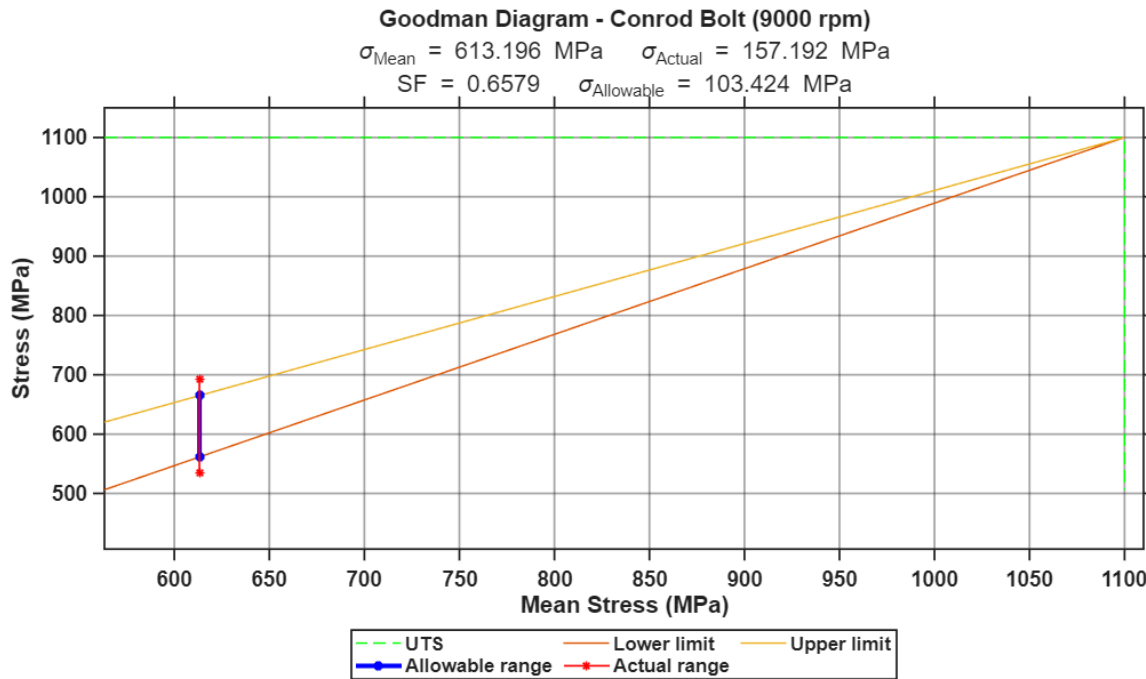
The cylinder pressure data were extended to 9000 rpm using the MATLAB `interp1` function with linear extrapolation. This method was appropriate given the broadly consistent pressure trend across the supplied RPM range, with no discontinuity evident that would require a higher-order approach. While linear extrapolation may introduce minor inaccuracy at the upper speed limit, this has limited effect on the fatigue outcome as the dominant bolt loading comes from the inertia-driven cap-opening force rather than gas pressure.

4.2 Performance of the original bolt at 9000 rpm



Graph 11 - FSF for the Conrod Bolt up to 9000 RPM

When the bolt assessment is extended to 9000 rpm, the fatigue margin falls below the acceptable limit. The extrapolated FSF trend shows a continued reduction from 1.80 at 6000 rpm to 0.66 at 9000 rpm, indicating that the original bolt no longer satisfies the fatigue requirement at the higher speed. This is confirmed by the 9000 rpm Goodman diagram, which gives a mean stress of 613.2 MPa and an alternating stress of 157.2 MPa, compared with an allowable stress range of only 103.4 MPa. Relative to the original assessed range, this represents a substantial loss of margin and shows that the bolt becomes the limiting feature of the assembly once the operating speed is extended.



Graph 12 - Goodman Diagram for the Conrod Bolt at 9000 RPM

This behaviour is expected from the loading mechanism. As engine speed increases, the inertia-driven cap-opening load rises rapidly and is superimposed onto the high tensile mean stress already created by preload. The bolt is therefore forced to operate further up the Goodman diagram, where the allowable alternating range is much smaller. In addition, the critical section remains the most highly stressed local region, so the increased cyclic load is applied at the point most susceptible to fatigue damage. The combined effect is a rapid deterioration in fatigue resistance even though the bolt geometry itself is unchanged.

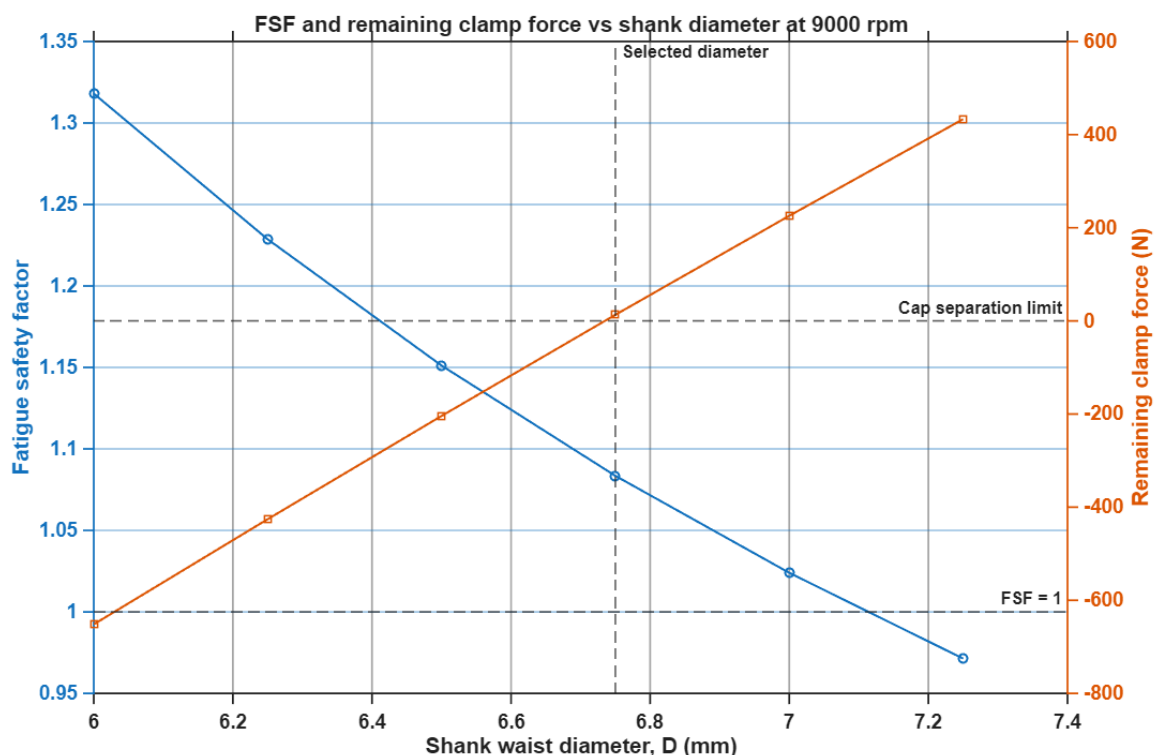
The original production bolt may be acceptable within the intended operating range, but the 9000 rpm assessment shows that its fatigue margin reduces significantly once inertia-driven opening loads become dominant.

5 Bolt improvement study and final design selection

5.1 Design logic behind the improvement routes

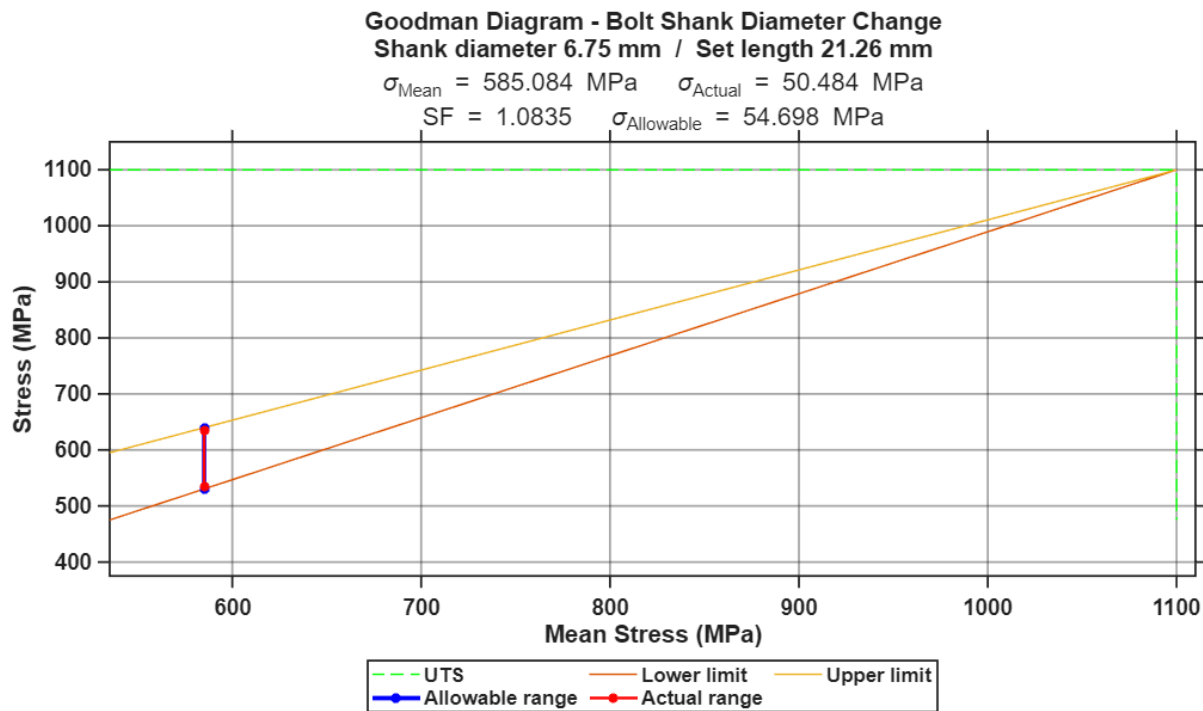
Three redesign routes were assessed: geometry changes to lower cyclic stress severity, rolled threads to reduce notch severity at the most fatigue-critical region without altering the bolt envelope (Da Silva et al., 2016), and material upgrade to raise the allowable stress range (Pedersen, 2013). Each route carries constraints in isolation, so the most effective redesign was expected to come from combining all three modifications.

5.1.1 Shank diameter changes



Graph 13 - Waisted Shank FSF for different diameters compared to the clamp force

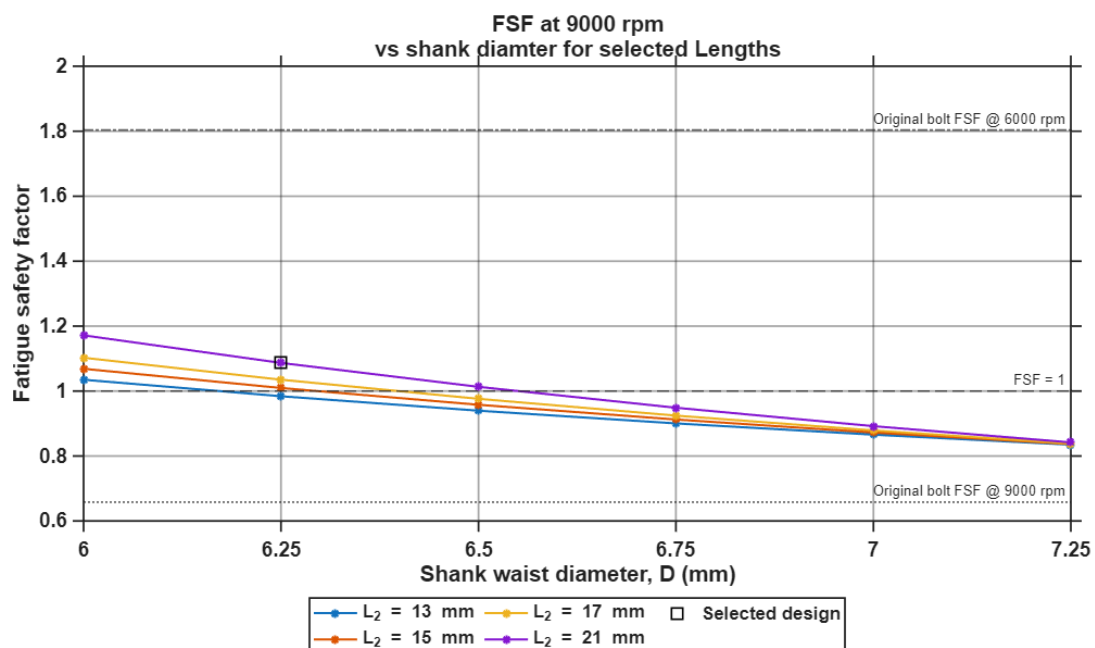
The above graph shows that reducing shank waist diameter improved FSF at 9000 rpm, rising from 0.97 at 7.25 mm to 1.32 at 6.0 mm as the bolt became more compliant. However, the same trend reduced the remaining clamp force, with the smallest diameters crossing into cap separation and therefore becoming unacceptable despite the higher FSF. The critical result is therefore the compromise between fatigue improvement and joint integrity rather than the maximum FSF alone (Kaundabalarman, 2017). On this basis, 6.75 mm was selected as the best diameter-only solution, as it remained close to the separation limit while keeping FSF above 1.



Graph 14 - Goodman Diagram showing the FSF for the Bolt modified Shank waist at 9000 RPM

The Goodman diagram shows that this change only produced a marginal improvement in FSF. Although the smaller diameter lowered the cyclic stress, the bolt still operated close to the fatigue limit because the high tensile mean stress remained largely unchanged. The diameter change was therefore a useful first improvement, but not sufficient on its own to provide a robust redesign.

5.1.2 Shank length changes

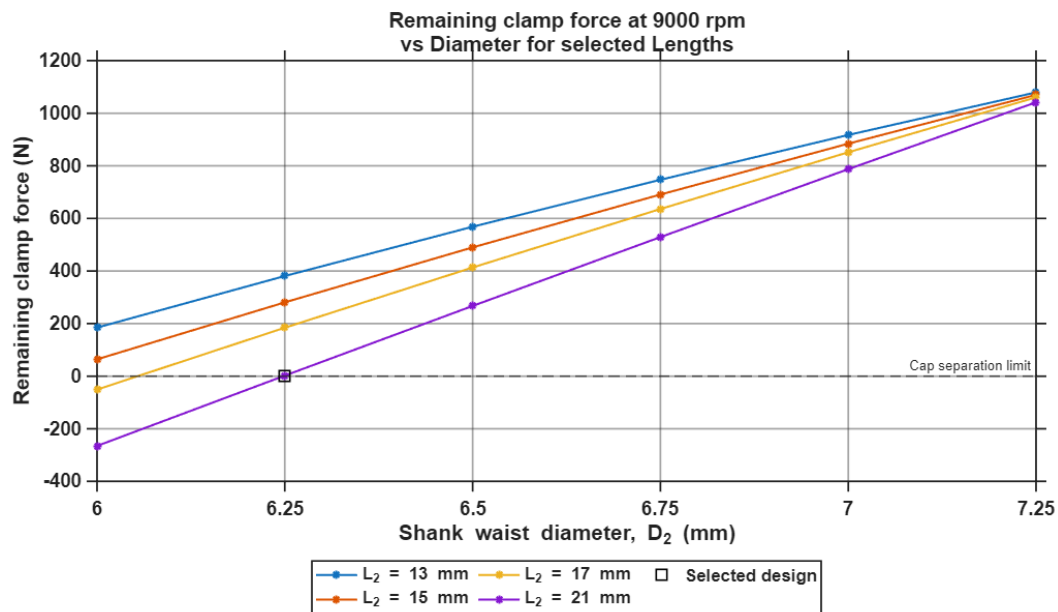


Graph 15 - FSF comparison for the bolt plotting different shank lengths and diameters

Graph 15 proves that increasing shank length improved FSF across the full diameter range, with the 21 mm length consistently giving the highest values and the 13 mm length the lowest. As in the diameter-only study, FSF still decreased as diameter increased, but the upward shift of the curves confirms that added compliant length was more effective at reducing cyclic stress severity than diameter change alone. This behaviour is governed by the bolt load-sharing relationship. A longer compliant shank reduces bolt stiffness relative to the clamped joint members, which lowers the load factor,

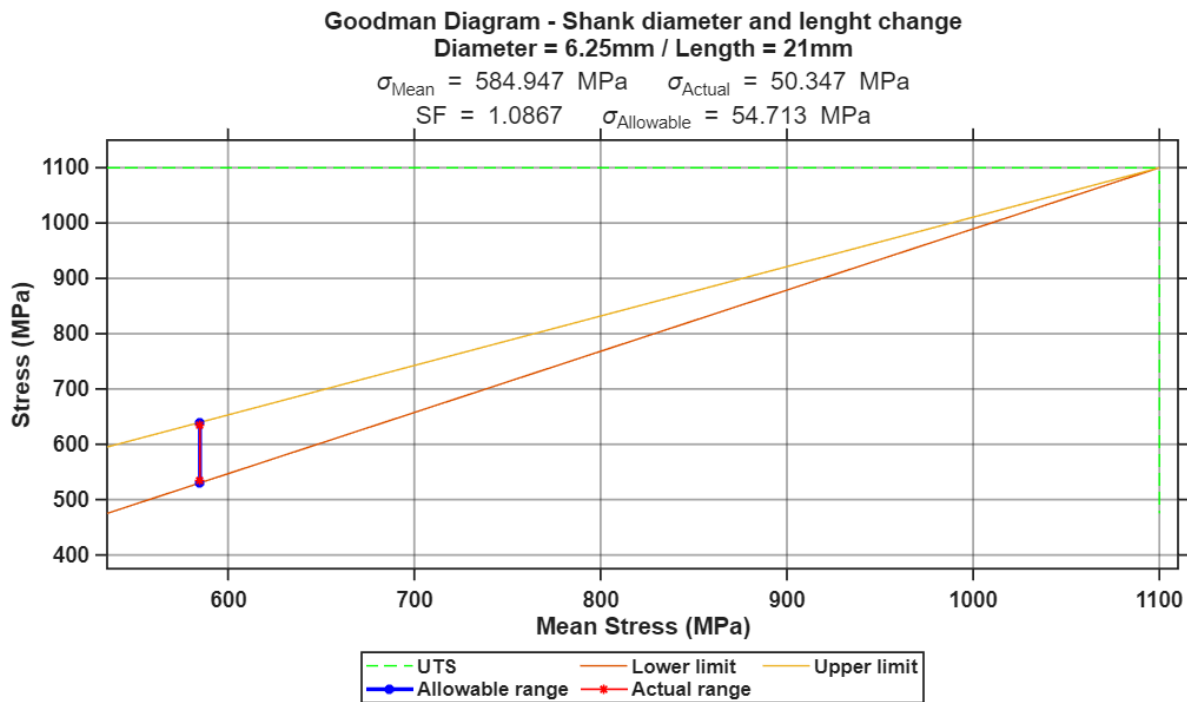
$$\Phi = k_b / (k_b + k_m)$$

where k_b is bolt stiffness and k_m is joint member stiffness. (Persson et al., 2021)



Graph 16 - Remaining clamp force for the bolt shank length and waist redesign at 9000 RPM

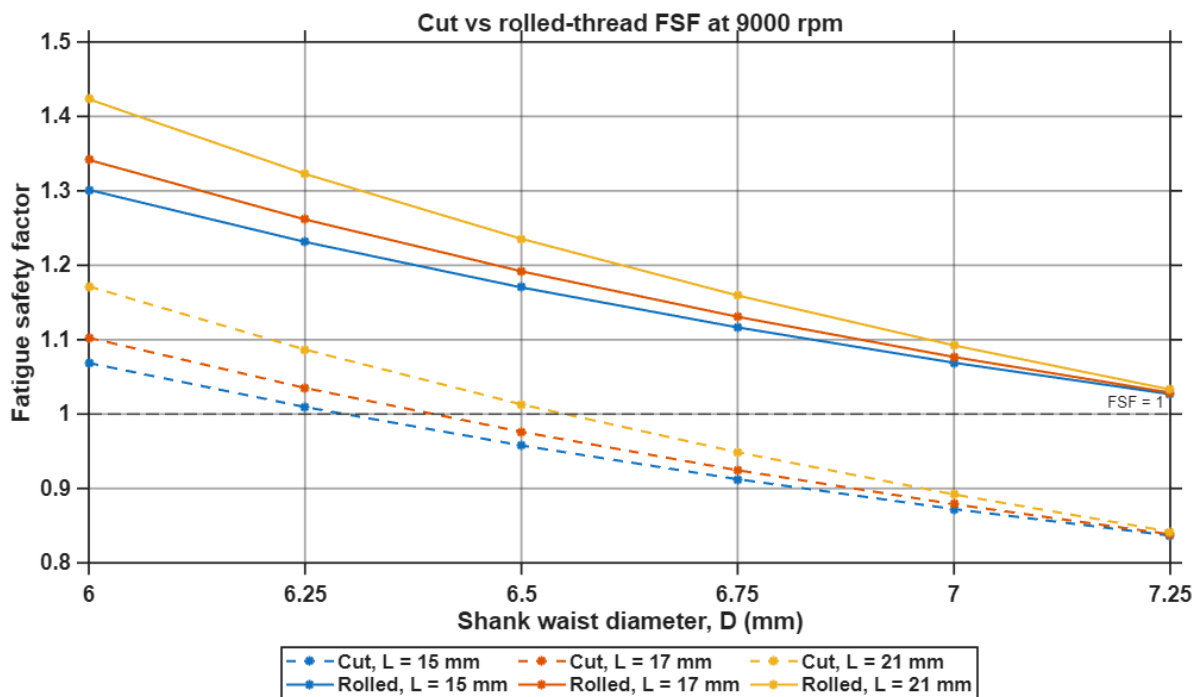
Graph 16 shows the limiting effect of this approach. For all lengths, remaining clamp force increased with diameter, while longer compliant sections reduced clamp retention at a given diameter. The 21 mm length therefore gave the best fatigue response but also placed the joint closest to cap separation. On this basis, 6.25 mm diameter and 21 mm length were selected as the best combined geometry, as this condition sits above the separation limit while still maintaining FSF above 1.



Graph 17 - Goodman diagram for the selected shank length and diameter

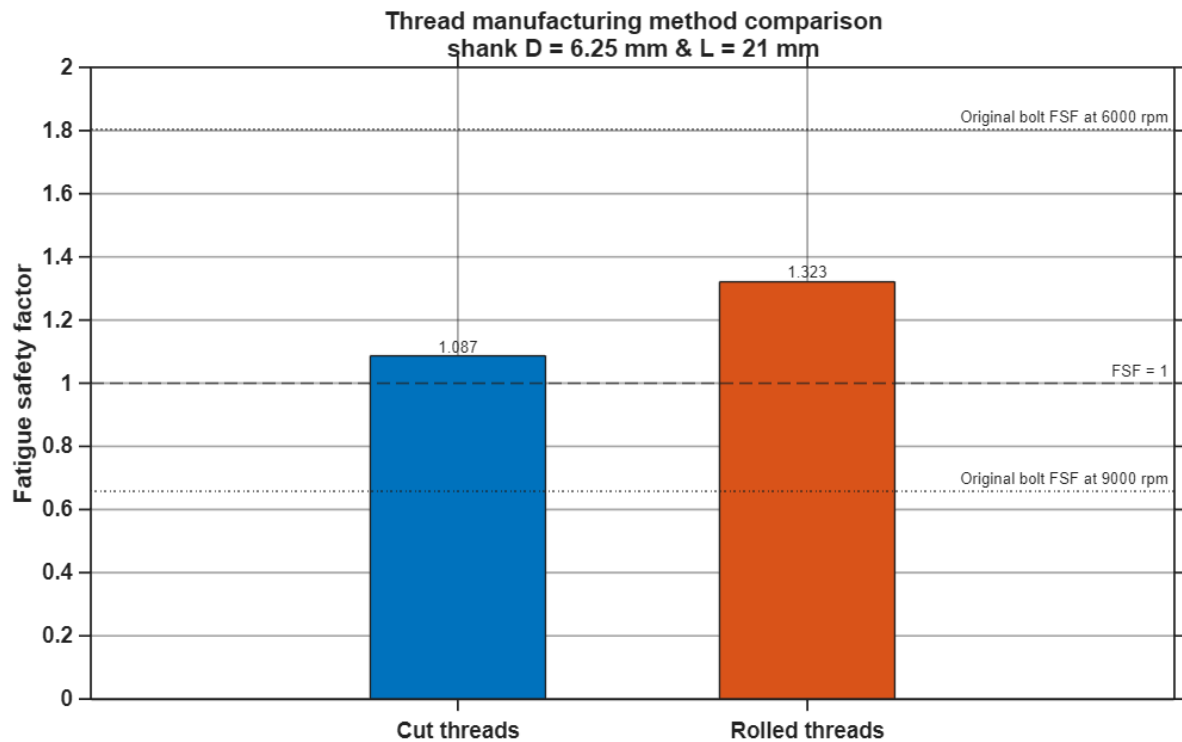
The goodman diagram shows that the selected geometry remained only marginally acceptable. The FSF confirms that the added compliance reduced the alternating stress effectively, but the bolt still operated close to the fatigue limit because the tensile mean stress remained high. Increasing compliant length therefore improved the design more effectively than diameter change alone, but the benefit remained constrained by cap separation and the persistently high mean stress.

5.1.3 Rolled thread improvement



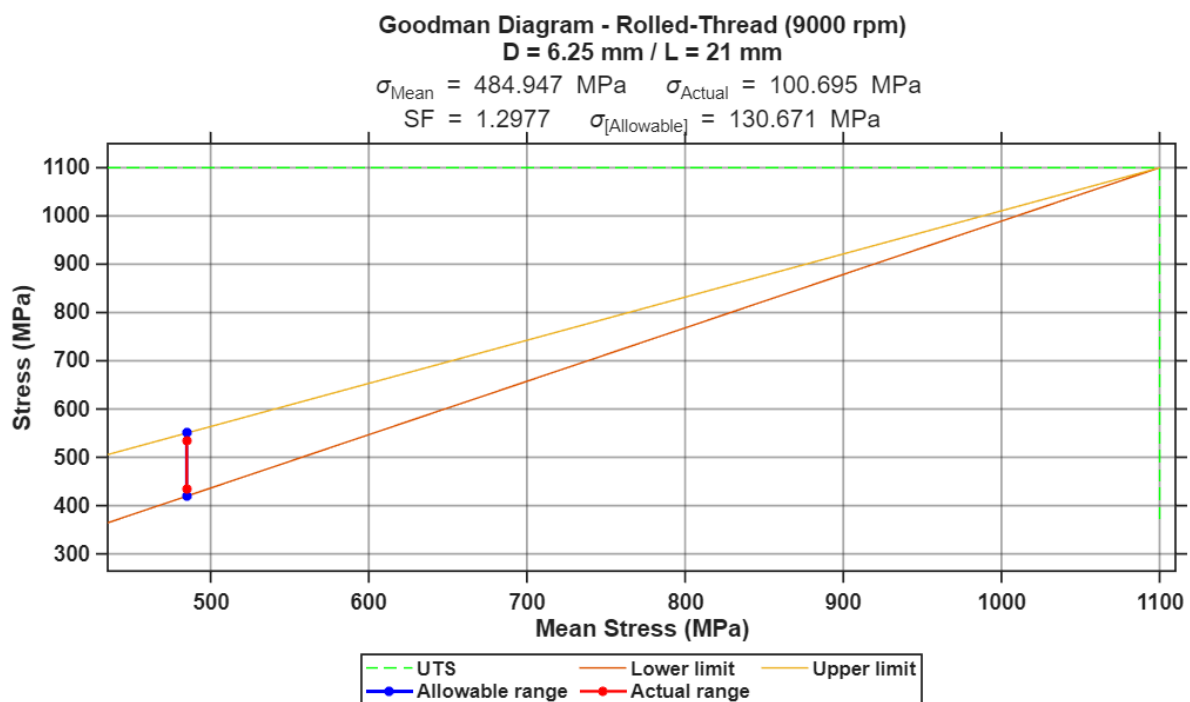
Graph 18 - Comparison of cut vs rolled threads for different bolt lengths and diameters

The graph above proves that rolled threads improve the FSF across the full assessed diameter range for all selected shank lengths, with every rolled-thread curve remaining above the corresponding cut-thread result. This confirms that improving the thread-root condition gives a direct fatigue benefit, which is important because conrod bolt failure is most likely to initiate at the threaded section where local stress concentration is highest.



Graph 19- Bar chart comparing the cut threads vs the rolled threads for the selected bolt geometry

The chart above shows the effect more clearly at the selected 6.25 mm / 21 mm geometry, where FSF increased from 1.087 for cut threads to 1.323 for rolled threads, an improvement of approximately 22%.



Graph 20 - Goodman Diagram for the new Bolt with rolled threads

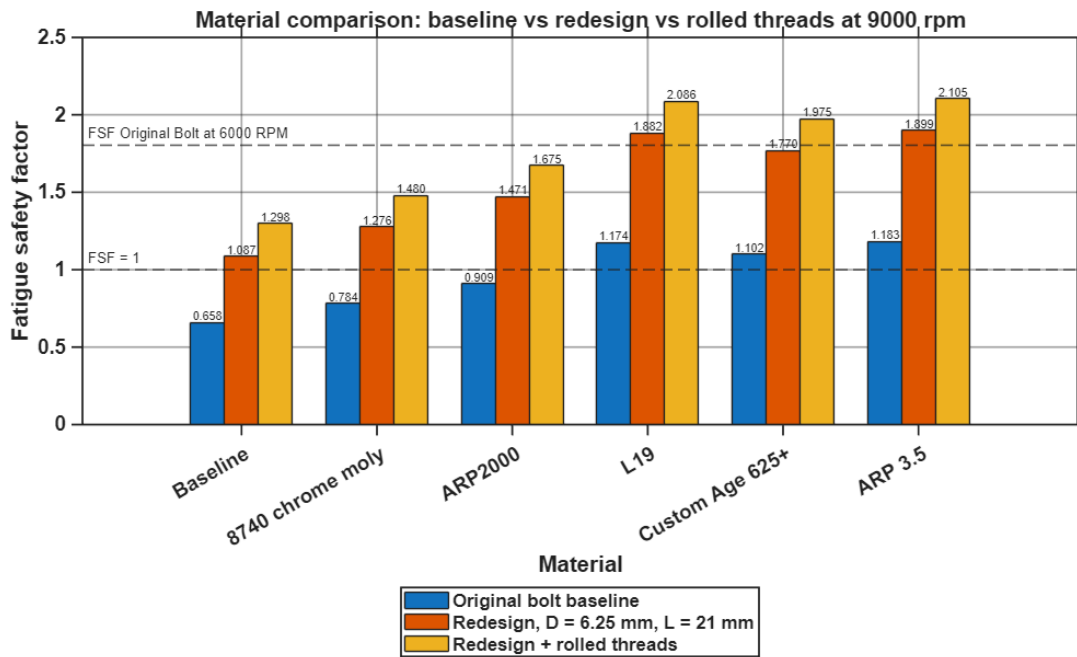
The Goodman diagram above confirms that the rolled-thread condition moved the operating stress range further inside the allowable Goodman boundary, giving a more credible fatigue margin at 9000 rpm. Rolled threads therefore provided one of the most effective single refinements, because they improved the behaviour of the most fatigue-critical region without changing the overall bolt geometry (Roark et al., 2002b).

5.1.4 Material selection

Table 3 - Selected material specification

Material	Yield strength (MPa / N/mm²)	Tensile strength (MPa / N/mm²)	Modulus of Elasticity	(E) in N/mm²
8740 chrome moly	1172	1310	207 GPa	206800 N/mm²
ARP2000	1379	1517	207 GPa	206800 N/mm²
L19	1517	1793	207 GPa	206800 N/mm²
Custom Age 625+	1622–1793	1795–1931	207 GPa	206800 N/mm²
ARP 3.5	1517–1724	1793–1931	205–207 GPa	205000–206800 N/mm²

The material parameter table shows that the assessed materials have similar Modulus of elasticities, so changing material had little effect on joint stiffness or compliance. The main benefit therefore came from increased yield and tensile strength, which raised the allowable Goodman stress range at the critical bolt section. This is important because the preceding geometry changes reduced cyclic stress severity but did not remove the underlying high mean-stress condition (2026 ARP Catalog, n.d.).

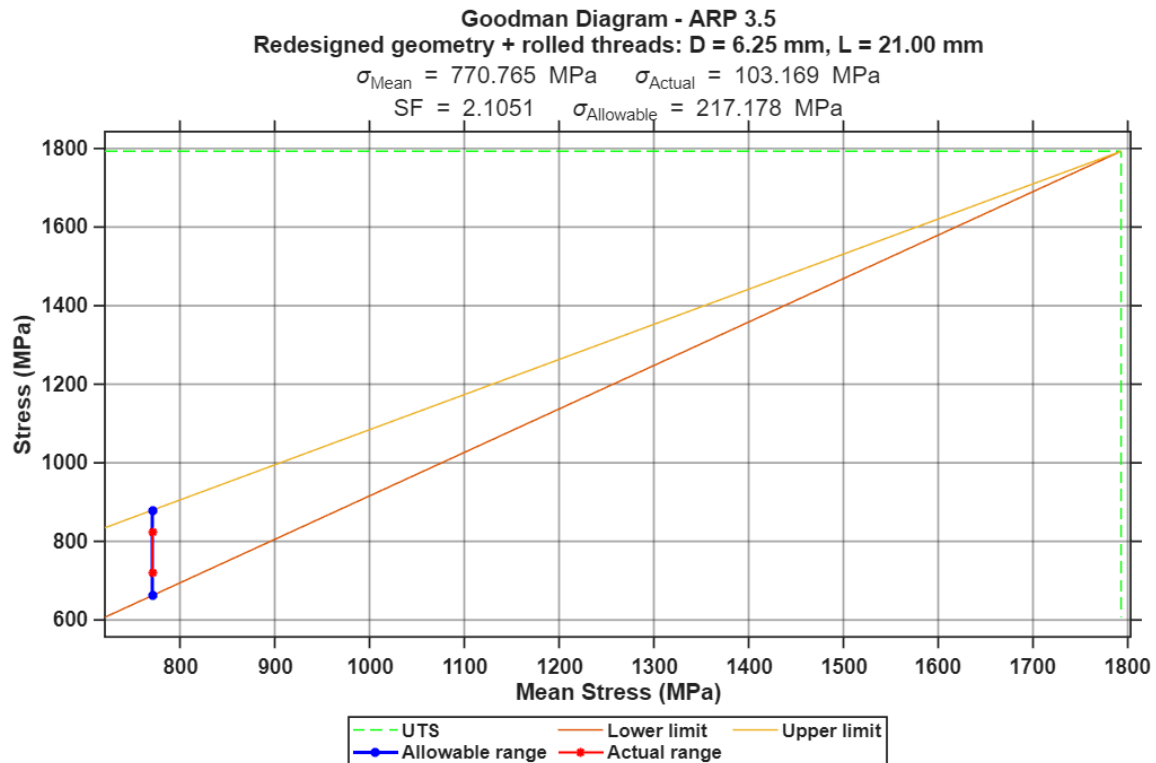


Graph 21 - Bar chart comparing the FSF for different materials and different bolt specifications

Graph 21 shows that for every material, the reduced-shank redesign improved FSF relative to the original baseline, and rolled threads increased it further, confirming that material strength acted as a multiplier on the geometry and thread improvements rather than a substitute. ARP 3.5 gave the highest final FSF at 2.105, followed closely by L19 at 2.086 and Custom Age 625+ at 1.975, while the weaker baseline and 8740 chrome moly remained below the original 6000 rpm condition. Material upgrade alone would not have solved the problem, as the bolt would still have carried the same stiffness and thread-root stress concentration; higher-strength material provided the final margin increase only once compliance and thread condition had been improved.

5.2 Final bolt design selection

The final design uses a 6.25 mm waist, 21 mm length, rolled threads and ARP 3.5 material.



Graph 22 - Goodman diagram for the final selected bolt redesign specifications

The final Goodman diagram shows that this combination gave the strongest Goodman margin of the assessed designs, while the table below confirms that it increased FSF to 2.11 at 9000 rpm, exceeding both the original bolt at 9000 rpm (0.66) and the original production condition at 6000 rpm (1.80). This is the key design achievement, as the redesigned bolt not only remained suitable at the extended speed range but also delivered a higher FSF than the original bolt the intended operating range. In motorsport terms, this demonstrates the value of using increased compliance, improved thread-root fatigue behaviour and a higher-strength material to support increased engine speeds.

Table 4 - Comparison of the FSF of the original Bolt and the redesigned Bolt

Bolt condition	Material	Geometry / thread condition	FSF
Original bolt at 6000 rpm	Baseline	Original geometry, cut thread	1.80
Original bolt at 9000 rpm	Baseline	Original geometry, cut thread	0.66
Final redesigned bolt at 9000 rpm	ARP 3.5	6.25 mm waist, 21 mm compliant length, rolled thread	2.11

6 Torque and Power comparisons

6.1 Understanding the differences between the two types of crankshafts

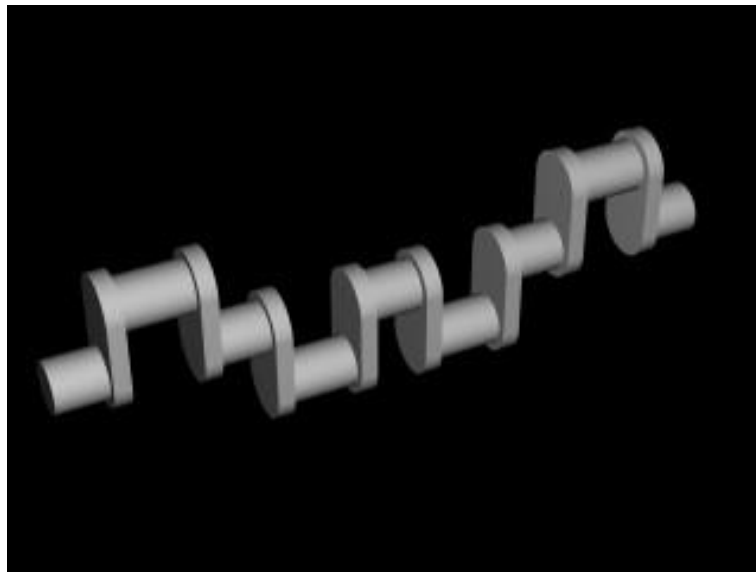


Figure 1 – Flat-plane crank

(Flat Plane Crankshafts vs. Cross-plane Crankshafts, n.d.)

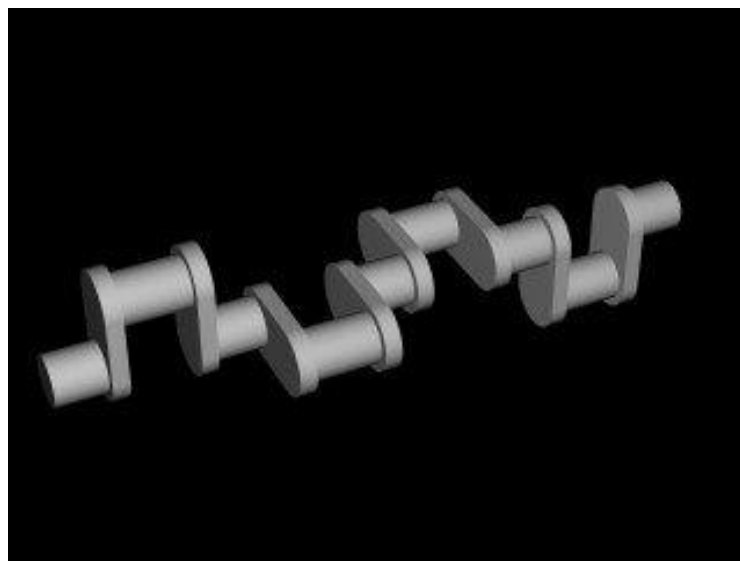
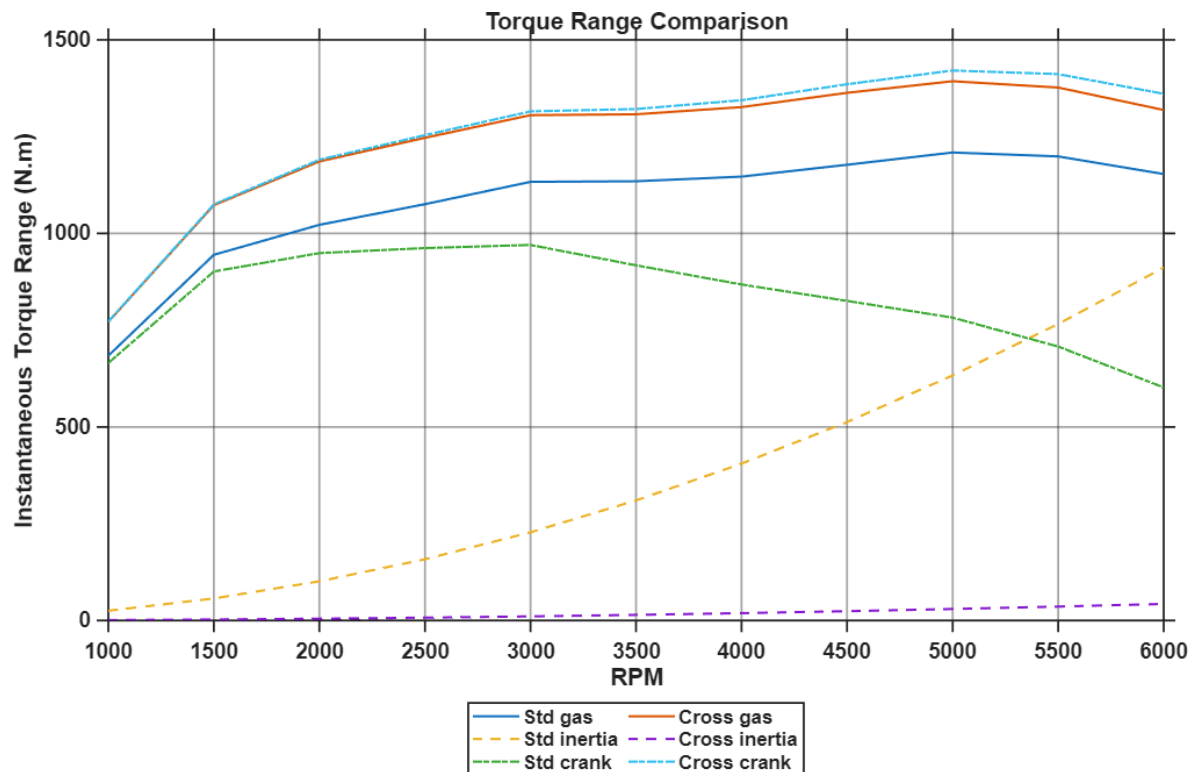


Figure 2 - Cross-plane crank

(Flat Plane Crankshafts vs. Cross-plane Crankshafts, n.d.)

Figures 1 & 2 show the geometric difference between the two crankshaft layouts. The flat-plane crank uses 180° crank pin phasing, giving a simpler, lighter arrangement with lower rotational inertia, while the cross-plane crank uses 90° phasing, improving inherent balance and refinement at the cost of higher mass, greater counterweight demand and more complex exhaust pulse behaviour. The engineering trade-off is therefore between rotational simplicity and dynamic smoothness.

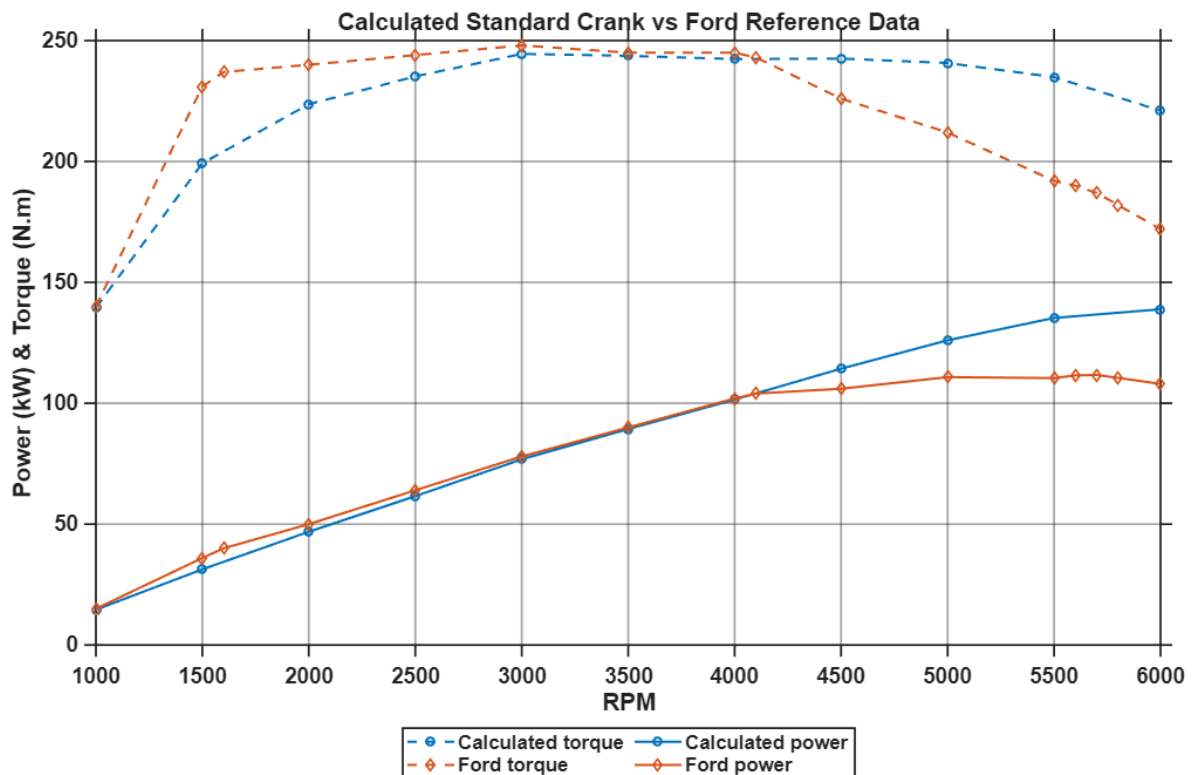
6.2 Torque range comparison between crank designs



Graph 23 - Torque comparison for the cross-plane and flat-plane crank

Graph 23 shows that the cross-plane crank gives a consistently larger instantaneous torque range than the flat-plane arrangement across the assessed speed range. This difference is not driven primarily by gas torque, which remains similar for both layouts, but by the crankshaft torque component generated by the different phasing of the cylinder contributions. Instantaneous torque varies with crank angle even at constant speed because the crank throw moment arm changes continuously and the cylinder gas load is not constant through the cycle. The two layouts therefore produce different summed torque traces under the same operating conditions.

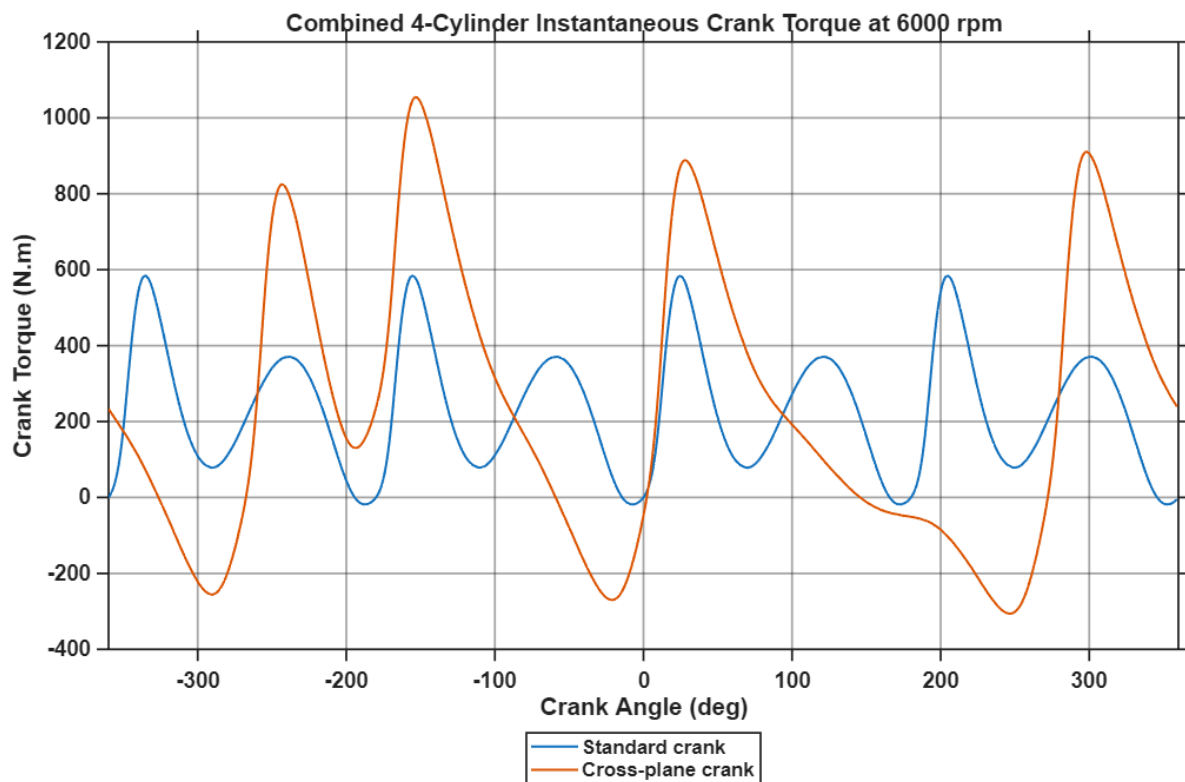
6.3 Comparison of calculated metrics to that of the manufacturer



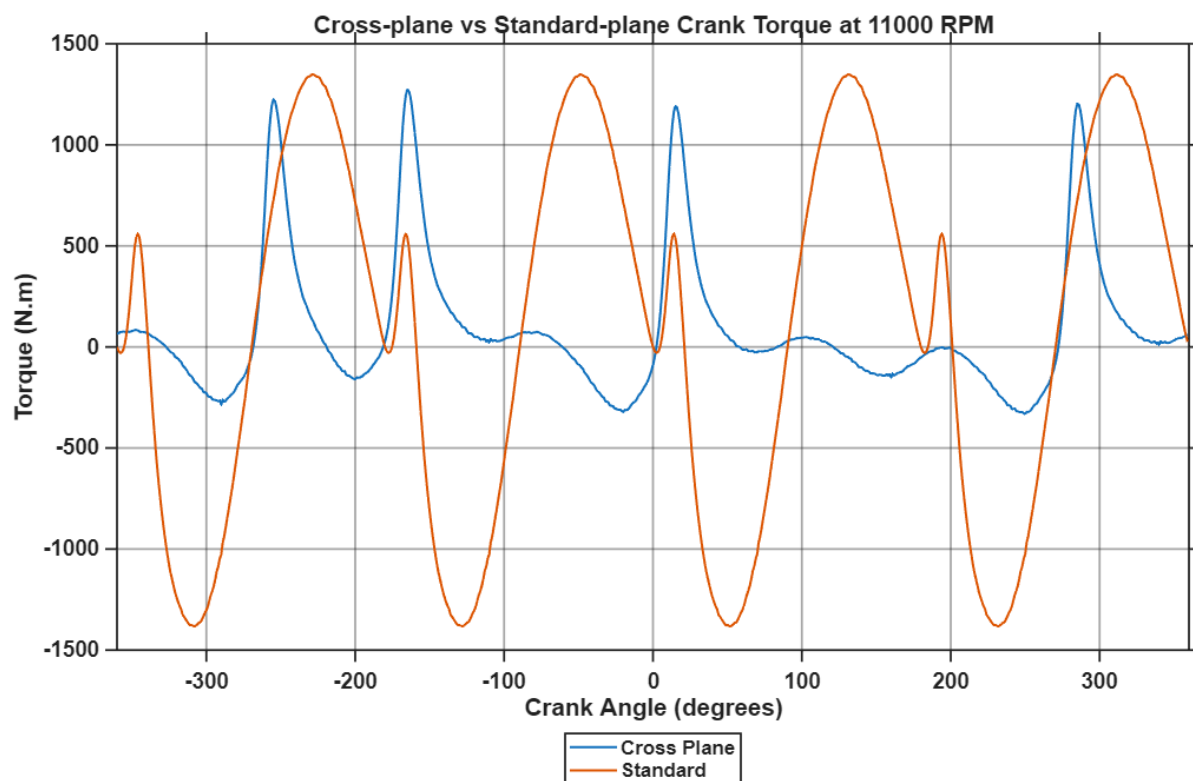
Graph 24 - Calculated Power and Torque compared to the Ford Data

The power and torque comparison plot shows that the calculated torque curve follows the general shape of the Ford reference data well in the mid-range, with the closest agreement between 3000 and 4000 rpm. (16-EcoBoost-GTDI-Torque-Power, n.d.) At low speed the model underpredicts torque slightly, while above 4500 rpm it increasingly overpredicts both torque and power. This is expected because the calculation captures the basic crank-angle torque generation but does not fully represent the speed-dependent losses of the real engine, including breathing limitation, friction and pumping losses.

6.4 Instantaneous torque comparisons and engineering discussion



Graph 25 - Instantaneous torque at 6000 RPM



Graph 26 - Instantaneous torque at 11000 RPM

The two graphs above show the instantaneous torque comparison at 6000 rpm and 11000 rpm. At 6000 rpm, the flat-plane crank gives the narrower torque range, while the cross-plane crank shows larger local fluctuations. At 11000 rpm, the flat-plane trace becomes much more severe, whereas the cross-plane layout gives the tighter overall range. This indicates that as inertia torque becomes dominant at very high speed, cross-plane phasing reduces torque ripple more effectively. The flat-plane crank is therefore better suited to motorsport applications where low inertia, sharper response and favourable exhaust scavenging are prioritised, while the cross-plane crank is better suited to high-performance road cars where smoother delivery, lower NVH and improved driver refinement are more important.

7 Conclusions

The fatigue assessment showed a clear difference in fatigue margin between the three components. The connecting rod midpoint gave the highest FSF at 4.14, confirming that this component was conservative relative to the applied loading. The gudgeon pin gave an FSF of 3.19, indicating a more moderate but still acceptable design, while the connecting rod bolt was identified as the critical component with an FSF of 1.80. This was driven by its Goodman stress state, where a high tensile mean stress of 566.0 MPa restricted the allowable alternating range to 113.4 MPa. When the bolt assessment was extended to 9000 rpm, the FSF fell to 0.66, with alternating stress rising to 157.2 MPa against an allowable value of only 103.4 MPa, confirming that the original bolt was not suitable for operation at this speed.

The redesign study showed that restoring an adequate fatigue margin required a combined geometry, thread and material approach rather than a single modification. Geometry changes alone raised the bolt FSF to 1.087, confirming that increased compliance reduced cyclic stress but did not provide sufficient margin on its own. The introduction of rolled threads improved this by 22% to 1.323 by reducing stress concentration at the thread root. The final redesign, combining the optimised geometry with rolled threads and ARP 3.5 material, achieved an FSF of 2.11 at 9000 rpm. This represents a 220% improvement over the original bolt at 9000 rpm and a 17% increase above the original production condition at 6000 rpm, confirming that the redesign restored a satisfactory and quantifiable fatigue margin across the extended speed range.

8 References

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