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Contemporary Powertrain Design

Analytical Assessment of Engine Internal Components (ACHA6006)

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1 Introduction and Methodology

The bottom-end components of an engine are subjected to constantly fluctuating loads from both combustion gas pressure and piston inertia throughout operating cycles. In an engine like the Ford 1.6L Sigma, these forces vary significantly with engine speed and crank position, making it imperative to understand how they translate into component stresses. This report analyses those forces using supplied cylinder pressure and kinematic data in order to calculate stresses and fatigue safety factors for the connecting rod [henceforth con-rod] midpoint, gudgeon pin, and con-rod bolts across 1000 to 6000 RPM using Goodman diagrams.

The analysis is then extended to 9000 RPM through MATLAB polynomial interpolation, allowing bolt safety factors to be assessed under extreme conditions and for the proposition of a revised bolt geometry. The report concludes with a comparison of standard flat plane [henceforth standard plane] and cross plane crankshaft configurations, discussing how each affects instantaneous torque output and suitability for different applications.

2 Critical Analysis of Data

Goodman diagrams were used to evaluate the fatigue performance of the con-rod, gudgeon pin, and bolt assemblies. These were conducted at engine speeds between 1000 and 6000 RPM. This analysis presents clear differences in how each component responds to cyclic loading, highlighting both strengths in the design and areas where stress behaviour becomes critical at higher rotational speeds.

For each RPM condition, the peak and minimum values of axial stress at the con-rod midpoint, bending stress at the gudgeon pin, and bolt stress were extracted across the complete 720° engine cycle. Mean stress (σ_m) and alternating stress (σ_a) were computed as:

$$\sigma_m = (\sigma_{max} + \sigma_{min}) / 2$$

$$\sigma_a = (\sigma_{max} - \sigma_{min}) / 2$$

The modified Goodman criterion for fatigue safety factor is:

$$N_f = 1 / (\sigma_a / S_e + \sigma_m / UTS)$$

Where S_e is the material endurance limit and UTS is the ultimate tensile strength. The #DIV/0! Values near -355° at 1000 RPM occur where the net force crosses zero, producing infinite apparent safety factors; these are not physical failures.

Below is figure 1 taken from the excel, it shows #DIV/0! in the final column in the 1000 RPM sheet, along with other stress/force examples:

Piston acceleration (m/s ²)	Inertia Force (N)	Net Piston Axial Force (N)	Con-rod axial force (N)	Con-rod stress midpoint (Mpa)	Pin force taken for stress (N)	Pin bending moment at A (Nmm)	Pin stress (Mpa)	Bolt Load Per Bolt (N)	Bolt Stress (Mpa)	Safety Factor Con_Rod	Safety Factor Pin	Safety Factor Bolt
581.89	201.92	271.59	271.59	1.38	271.59	950.55	1.18	0.00	0.00	115.05	98.58	0.00
581.74	201.86	274.70	274.71	1.40	274.71	961.47	1.20	0.00	0.00	1.17	1.01	0.00
581.29	201.71	276.33	276.34	1.41	276.34	967.21	1.21	0.00	0.00	58.11	49.79	0.00
580.53	201.45	281.37	281.40	1.43	281.40	984.92	1.23	0.00	0.00	56.94	48.79	0.00
579.48	201.08	284.92	284.98	1.45	284.98	997.45	1.24	0.00	0.00	423.85	427.59	495.00
578.13	200.61	288.33	288.43	1.47	288.43	1009.51	1.26	0.00	0.00	7.44	8.76	#DIV/0!
576.48	200.04	293.68	293.83	1.50	293.83	1028.40	1.28	0.00	0.00			
574.53	199.36	299.14	299.34	1.52	299.34	1047.69	1.31	0.00	0.00			
572.29	198.59	304.69	304.97	1.55	304.97	1067.38	1.33	0.00	0.00			
569.76	197.71	310.35	310.70	1.58	310.70	1087.46	1.36	0.00	0.00			
566.93	196.73	314.52	314.96	1.60	314.96	1102.35	1.37	0.00	0.00			
563.82	195.64	320.38	320.92	1.63	320.92	1123.21	1.40	0.00	0.00			
560.42	194.46	324.62	325.27	1.66	325.27	1138.45	1.42	0.00	0.00			
556.73	193.19	325.90	326.67	1.66	326.67	1143.33	1.43	0.00	0.00			
552.76	191.81	325.81	326.69	1.66	326.69	1143.42	1.43	0.00	0.00			
548.52	190.34	322.50	323.50	1.65	323.50	1132.26	1.41	0.00	0.00			
544.00	188.77	317.70	318.82	1.62	318.82	1115.86	1.39	0.00	0.00			
539.21	187.11	311.39	312.63	1.59	312.63	1094.20	1.36	0.00	0.00			
534.15	185.35	306.90	308.26	1.57	308.26	1078.91	1.34	0.00	0.00			
528.83	183.51	303.97	305.46	1.56	305.46	1069.12	1.33	0.00	0.00			
523.26	181.57	301.24	302.88	1.54	302.88	1060.09	1.32	0.00	0.00			
517.42	179.55	303.27	305.08	1.55	305.08	1067.78	1.33	0.00	0.00			
511.34	177.44	308.57	310.58	1.58	310.58	1087.04	1.36	0.00	0.00			
505.01	175.24	312.35	314.58	1.60	314.58	1101.02	1.37	0.00	0.00			
498.45	172.96	320.88	323.36	1.65	323.36	1131.76	1.41	0.00	0.00			
491.65	170.60	327.90	330.63	1.68	330.63	1157.22	1.44	0.00	0.00			
484.62	168.16	335.12	338.13	1.72	338.13	1183.45	1.48	0.00	0.00			
477.36	165.64	342.29	345.59	1.76	345.59	1209.57	1.51	0.00	0.00			
469.89	163.05	346.48	350.05	1.78	350.05	1225.19	1.53	0.00	0.00			
462.20	160.38	350.74	354.60	1.81	354.60	1241.11	1.55	0.00	0.00			
454.31	157.65	352.01	356.14	1.81	356.14	1246.48	1.55	0.00	0.00			

Figure 1: Example of Excel sheet used (1000 RPM)

Below is the table of inputs used in order to calculate forces and stresses in excel:

Bore (mm)	79.00
Crank radius r (mm)	40.70
Con-rod length (mm)	134.00
Reciprocating mass (kg)	0.35
Rotational mass (kg)	0.31
Piston area (mm ²)	4901.67
Piston area (M ²)	0.00
Con-rod midpoint area (mm ²)	196.39
Gudgeon Pin area (mm ²)	227.54
Gudgeon pin D1 (mm)	12.30
Gudgeon pin D2 (mm)	21.00
Gudgeon pin neutral axis y (mm)	10.50
Gudgeon pin (mm ⁴)	8423.00
Gudgeon pin L1 (mm)	7.00
Gudgeon pin L2 (mm)	7.75
Bolt Area (mm ²)	42.43
Bolt UTS (MPa)	1100.00
Rod UTS (MPa)	1000.00
Rod endurance limit (MPa)	450.00
Pin UTS (MPa)	1000.00
Pin endurance limit (MPa)	450.00
Number of bolts	2.00
Bolt Endurance Limit (MPa)	495.00

2.1 Con-Rod

The con-rod midpoint is primarily loaded in axial compression during the combustion stroke, with small tensile sections during the inertia-dominated portion of the cycle near TDC of the gas exchange stroke. The peak compressive stress is approximately 88.87 MPa, and occurs at 3000 RPM rather than at the highest speed. This reflects the relationship between engine speed and cylinder pressure. At higher speeds, inertia forces acting with the gas force moderate the net axial load through the rod.

Above 4000 RPM, stresses stabilise in the range of 76 to 89 MPa as the competing effects slightly cancel out one another. Applying the Goodman criterion with an endurance limit of 450 MPa and UTS of 1000 MPa, safety factors are comfortable throughout, ranging from effectively infinite at 1000 RPM to a minimum of approximately 4.9 at 3000 RPM. Above 4000 RPM the factor slightly improves to values of 7 to 9; inertia pre-loading shifts the stress cycle in a beneficial direction. Infinite fatigue life is confidently predicted across the full speed range.

The con-rod is significantly over-engineered relative to its fatigue loading. The component presents no durability issues across the entire operating range and requires no design modification for sustained high-speed use, provided the condition is maintained to avoid introducing bending loads into what is modelled as a purely axial member.

2.2 Con-Rod Analysis Figures

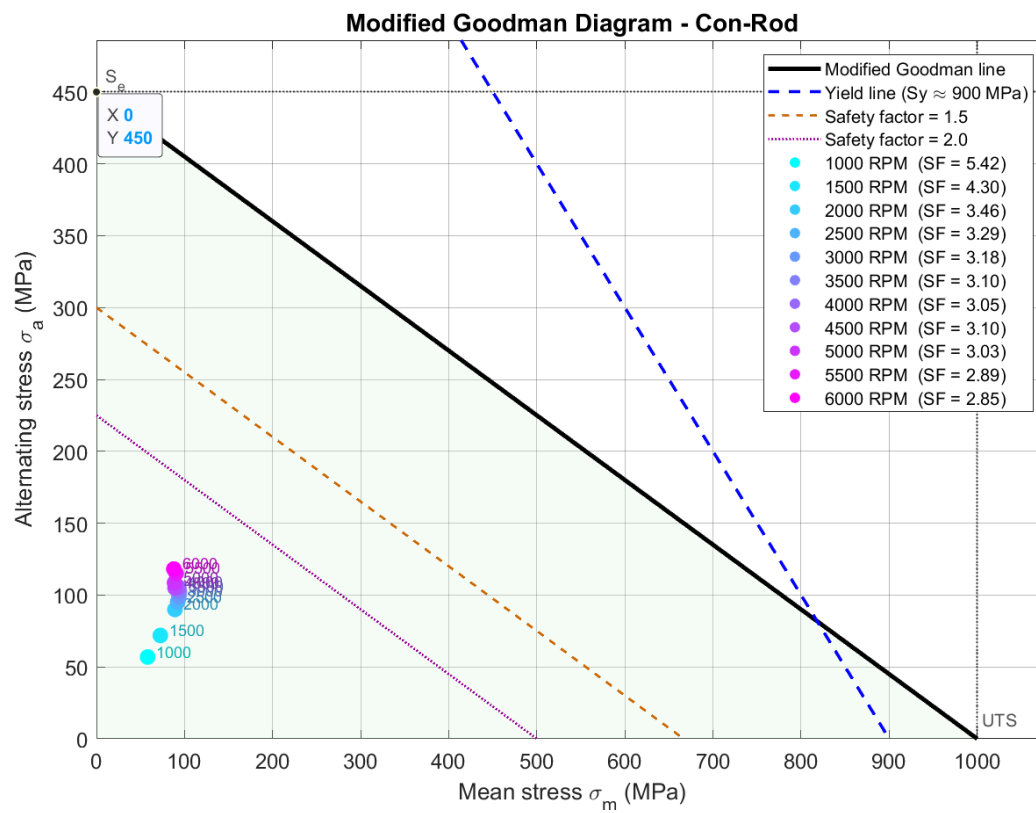


Figure 2: Modified Goodman Diagram: Con-Rod

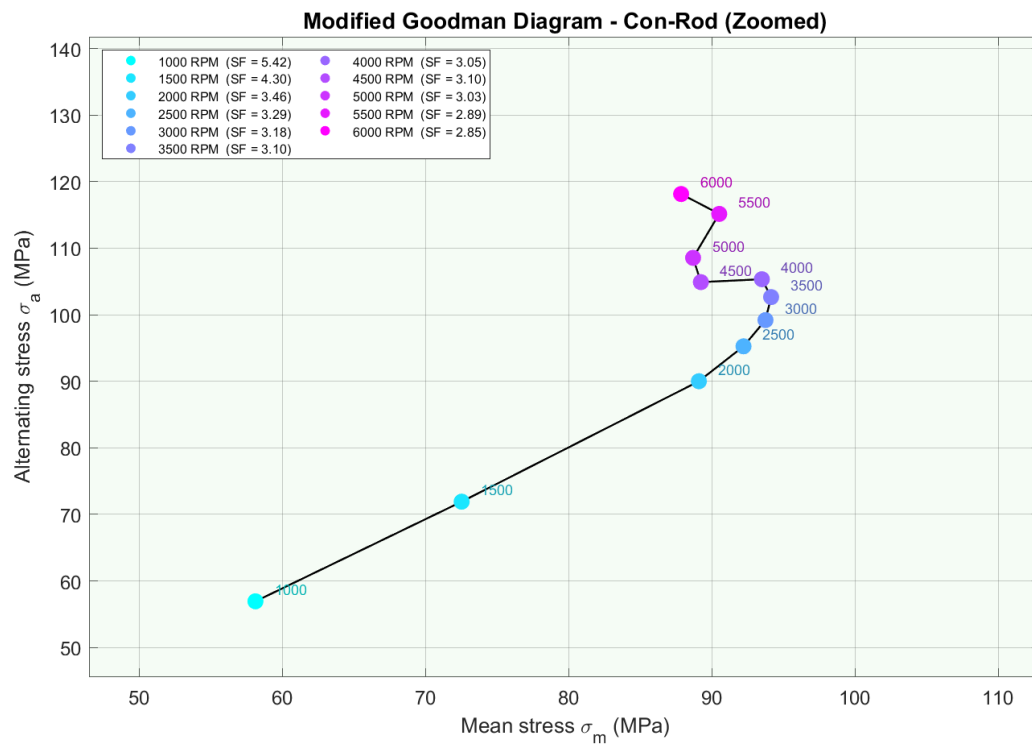


Figure 3: Con-Rod Safety Factor (Zoomed)

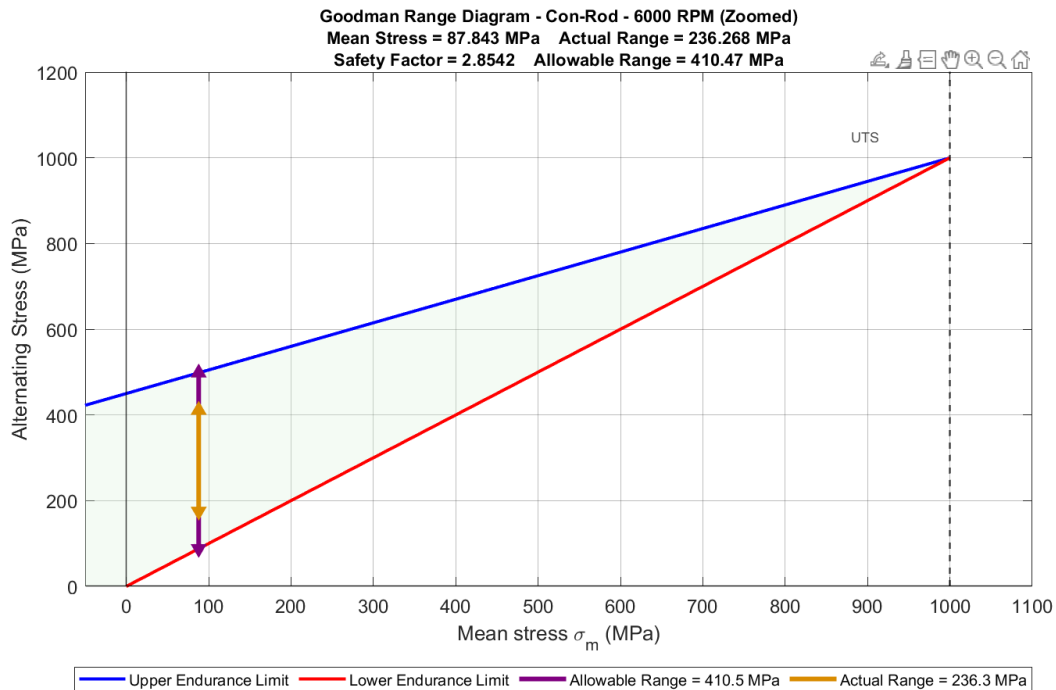


Figure 4: Con-Rod Goodman Range Diagram

2.3 Gudgeon Pin

The gudgeon pin is loaded in bending at point A, both the mean and alternating stress are positive. The growth in combustion pressure and inertia loading result in the peak bending stress rising from 73.64 MPa at 1000 RPM to 206.04 MPa at 6000 RPM. The Goodman safety factor falls from approximately 5.9 at 1000 RPM to a plateau of 2.2 to 2.3 between 4000 and 6000 RPM, where mean and alternating stresses grow near proportionally. While this prevents further deterioration, it offers no improvement. This plateau is the tightest margin of the three components analysed, and the pin is the life-limiting component in this assembly under ideal conditions.

The critical concern is that the safety factor of 2.2 to 2.3 makes no allowance for fretting fatigue at the boss interfaces. In practice, this can reduce the effective endurance limit by 20% to 40%. If the endurance limit had derated accordingly to 270 to 360 MPa, the safety factor at high speed would fall to approximately 1.4 to 1.7. This would cause an approach to the lower boundary of acceptable design and raise concern about finite fatigue life under sustained high-speed operation. Ionic surface

chemical heat treatment can be used to increase wear resistance for operation at high-rpm, alongside verification that small-end lubrication is adequate.¹

1 ¹ Wu Xiaochun, Chen Rui, Zhang Jun, *Surface Treatment Process for Automobile Engine Piston Pins* (Shanghai: Shanghai University, 2005)

2.4 Gudgeon Pin Analysis Figures

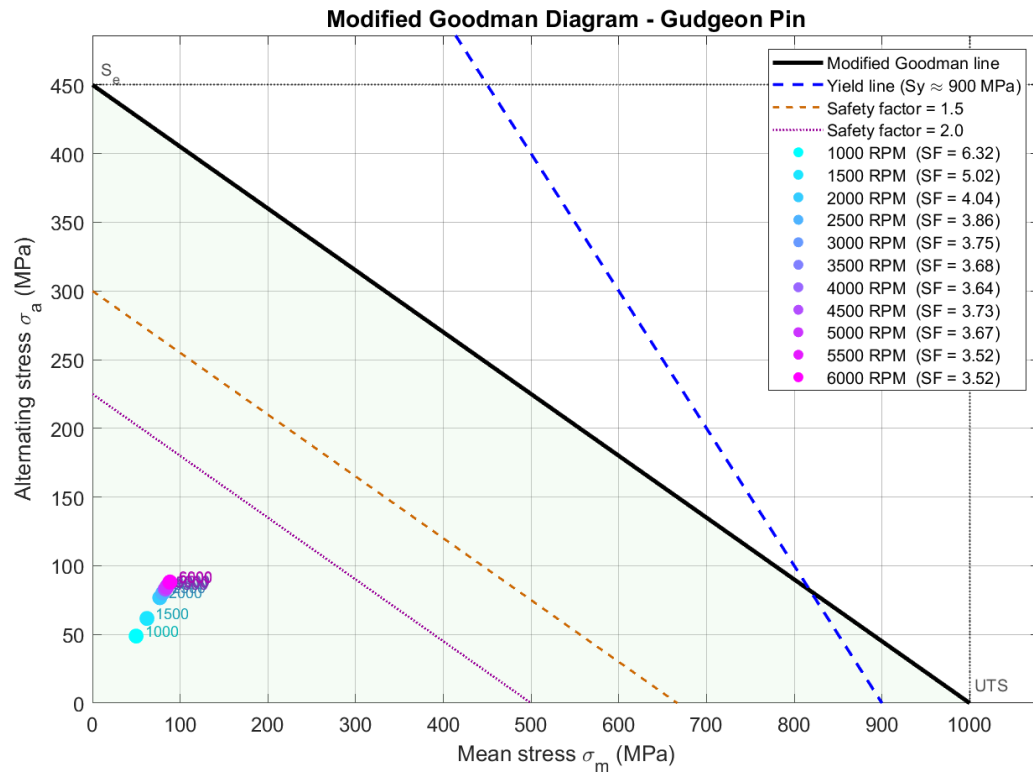


Figure 5: Modified Goodman Diagram Gudgeon Pin

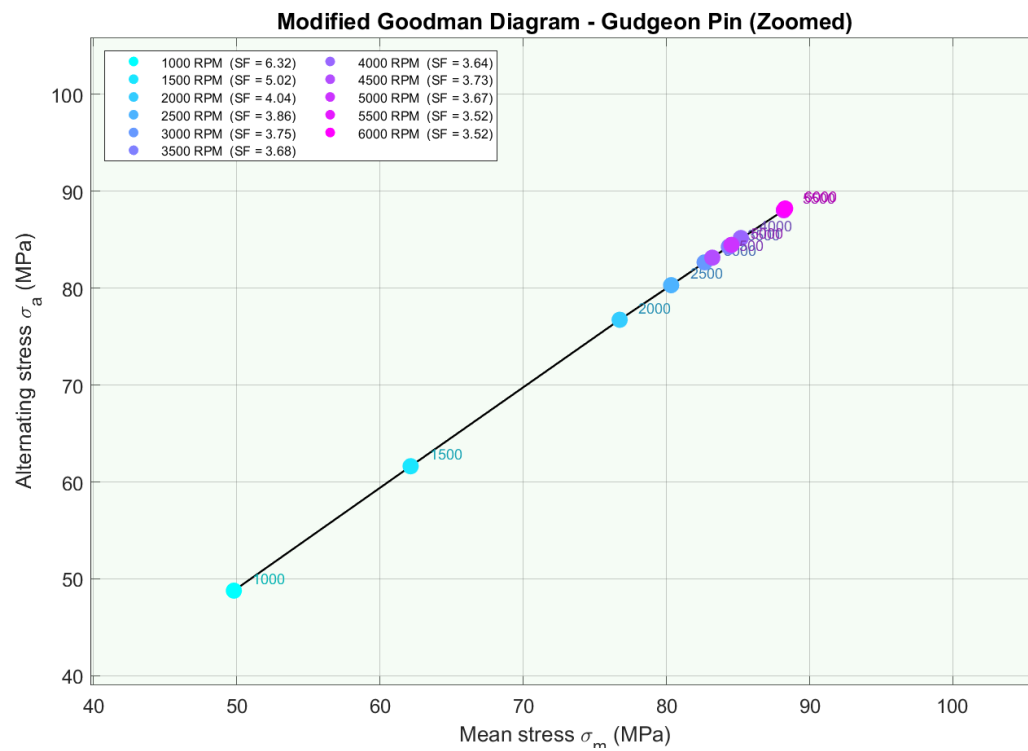


Figure 6: Gudgeon Pin Safety Factor (Zoomed)

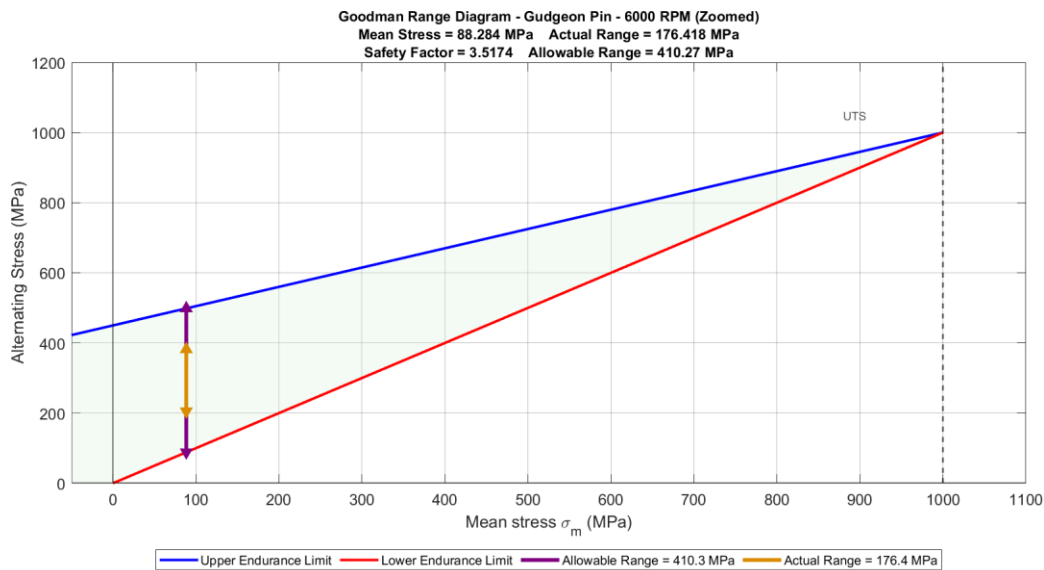


Figure 7: Gudgeon Pin Goodman Range Diagram

2.5 Bolt

The bolt loading does not undergo stress through most of the engine cycle. Tensile spikes present in a fluctuating manner and emerge above 2000 RPM. Simultaneously, inertia forces at TDC of the gas exchange stroke act to separate the big-end joint. Peak bolt stress rises from approximately 179 MPa at 2000 RPM to 206 MPa at 6000 RPM. Here, a plateau above 4000 RPM mirrors what is presented in the pin data. Safety factors of 2.4 to 2.7 appear adequate, but these figures are substantially misleading because bolt preload is not included in the dataset.

In practice, a representative preload of 60% to 80% of proof strength would raise the mean bolt stress to approximately 750 MPa at 6000 RPM, reducing the true safety factor to approximately 1.27. This result is marginal and reveals that fatigue has a more significant impact on the bolt than figures suggest.

Thus, strict preload control is the most important mitigation for bolt fatigue. Both under-tightening and over-tightening, however, are harmful; the former risking dynamic joint separation, and the latter increasing mean stress and directly reduces Goodman margin. Regular inspection of torque is highly recommended for continuous high-speed use. Of the three components, the bolts are the most assembly-sensitive and their condition directly controls big-end joint reliability at high speed.

2.6 Bolt Analysis Figures

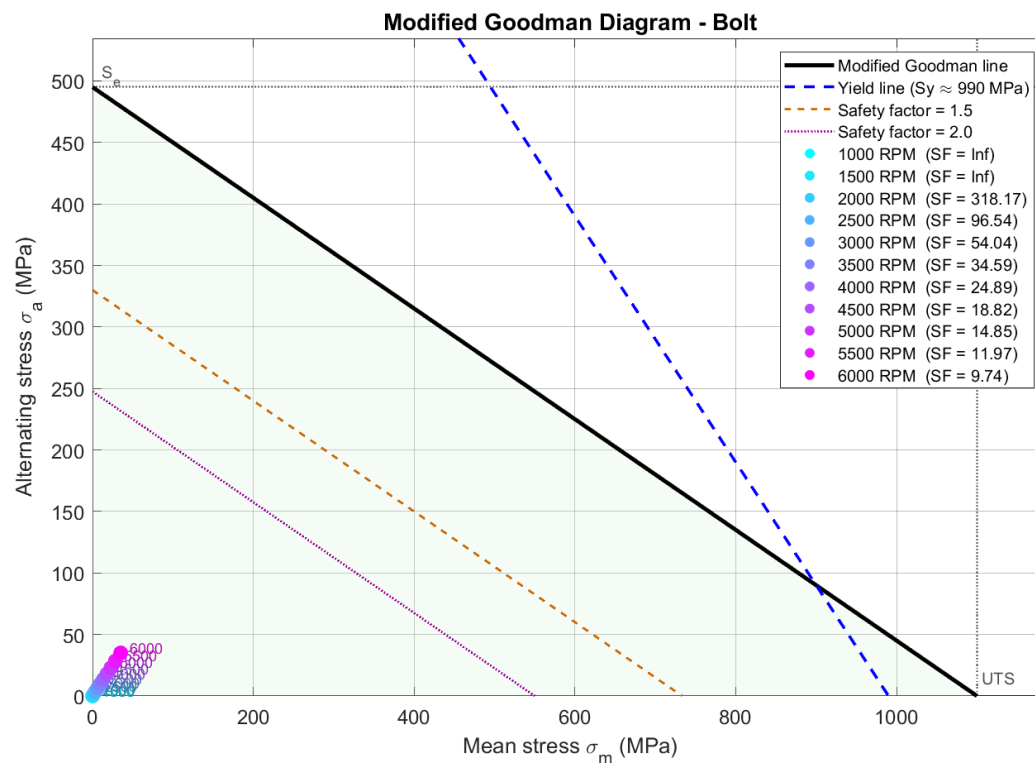


Figure 8: Modified Goodman Diagram Bolt

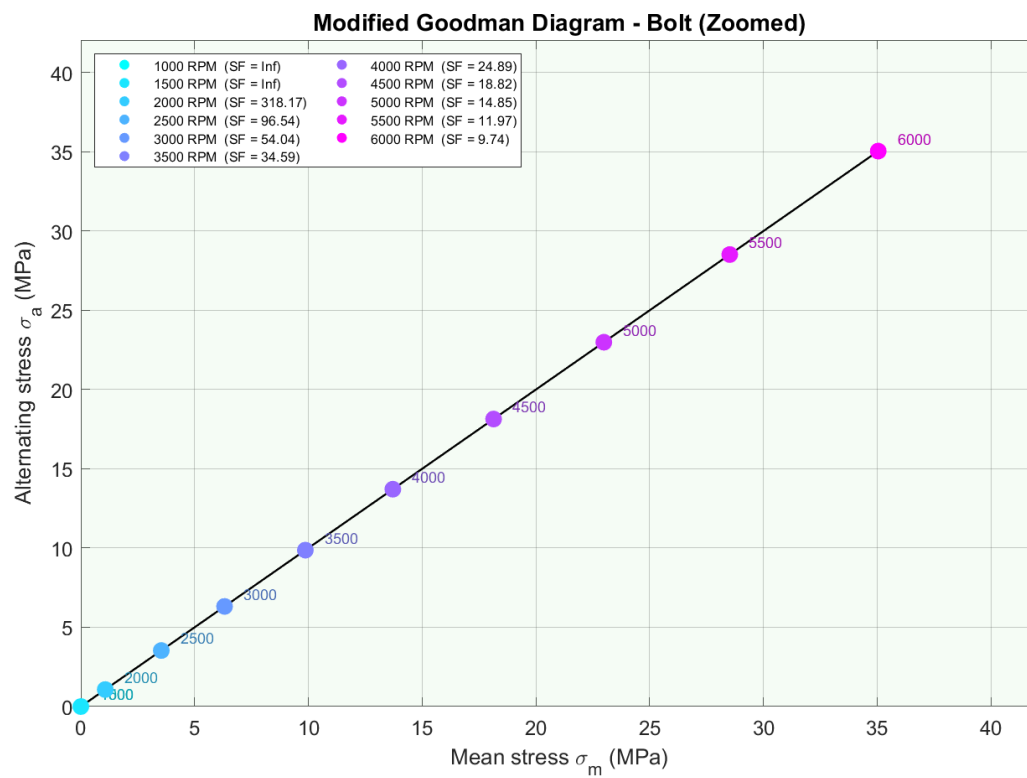


Figure 9: Bolt Safety Factor (Zoomed)

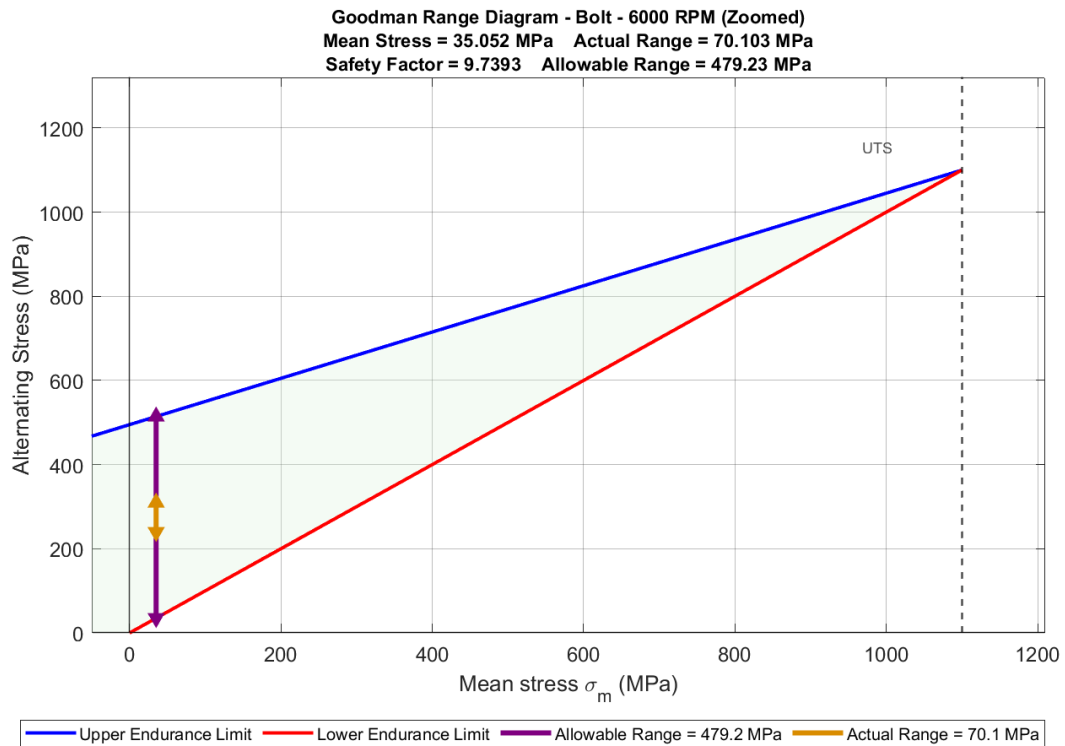


Figure 10: Bolt Goodman Range Diagram

3 Extended RPM Analysis

The MATLAB polynomial interpolation extends the bolt stress data to 9000 RPM. Both the mean and alternating stress are equal at every speed due to the fluctuating tension cycle during which minimum stress remains at 0. At 6500 RPM the extrapolated mean and alternating stress are both 41.14 MPa, rising to 76.84 MPa at 9000 RPM. The stress range increases from 69.57 MPa at 6000 RPM to 153.69 MPa at 9000 RPM, near doubling across a relatively small RPM range. The polynomial fit curves show a continuously accelerating upward trend above 6000 RPM with no sign of the plateau behaviour shown between 4000 and 6000 RPM. The inertia force, growing with ω^2 , becomes the dominant and rapidly increasing driver.

The safety factors decline consistently from 9.74 at 6000 RPM to 4.44 at 9000 RPM. The full Goodman diagram confirms all operating points sit well inside the safe zone, and the zoomed diagram shows the extrapolated points from 6500 RPM onward continuing the near-linear trajectory. At 9000 RPM the actual stress range of 153.7 MPa remains substantially below the allowable range of 460.4 MPa. This gives visible margin to the endurance boundaries. On this basis alone the bolt appears comfortably safe across the extended range.

However, the same issues discussed in the earlier analysis of the bolt regarding preload torque also apply here. Adding a realistic assembly preload of 700 MPa to the mean stress reduces the Goodman factor at 9000 RPM from 4.44 to approximately 1.17. This figure is very close to failure (Goodman Line). It should also be noted that polynomial extrapolation carries inherent uncertainty; the accuracy of the predicted values decrease with distance from the last measured data point at 6000 RPM. Any change of actual combustion pressure from the extrapolated trend would alter these results.

For sustained operation above 6000 RPM the bolt represents a durability risk upon properly accounting for preload. Use of a higher-grade bolt material with improved UTS and endurance limit, combined with preload control would be the minimum recommended mitigations for any application requiring consistent operation above 7000 RPM. The bolt, as opposed to the con-rod body or gudgeon pin, is the component most likely to affect the fatigue limit of this assembly.

3.1 Extended RPM Analysis Figures

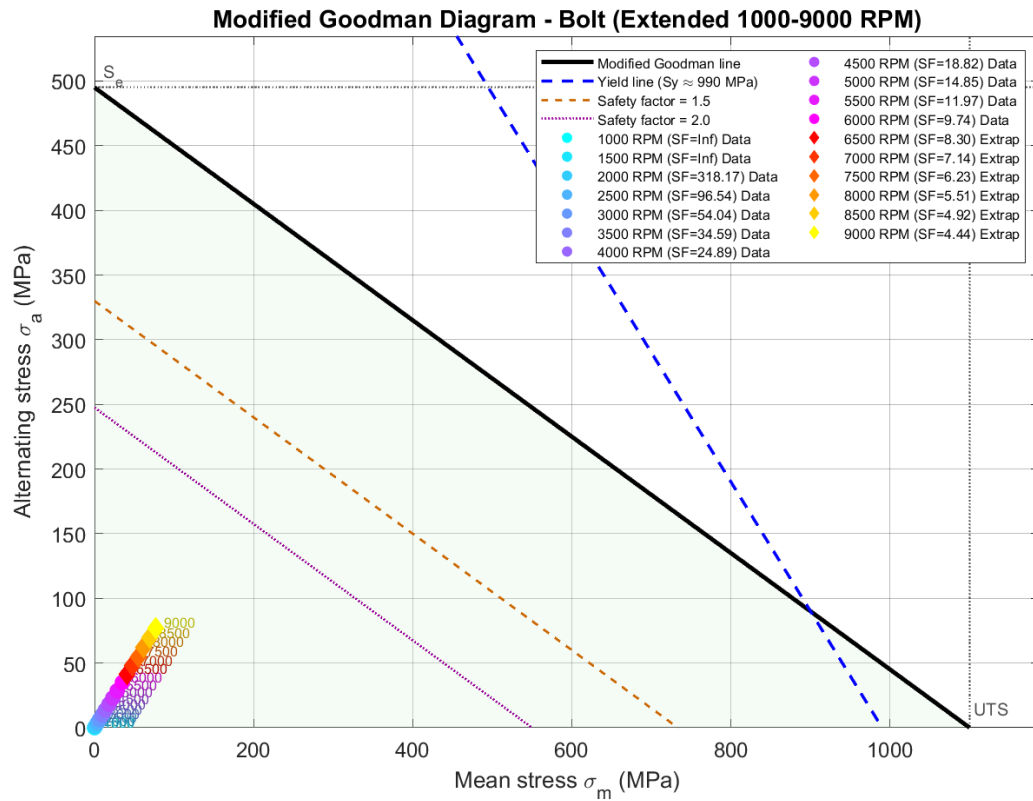


Figure 11: Extended RPM Modified Goodman Diagram

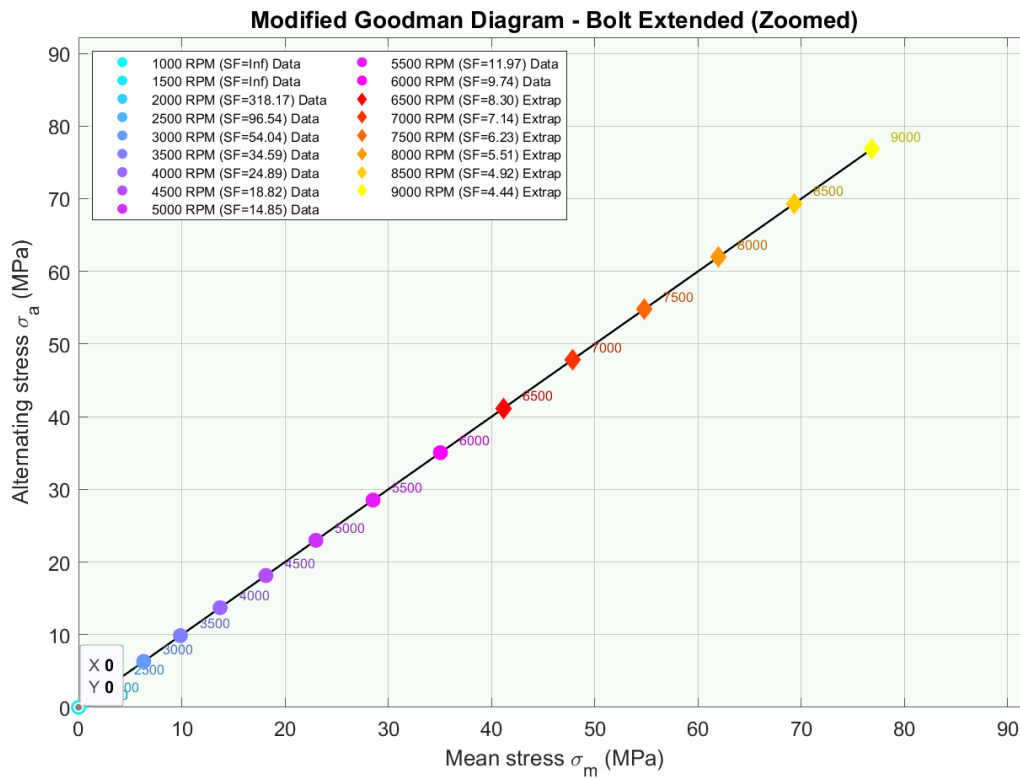


Figure 12: Extended RPM Safety Factor (Zoomed)

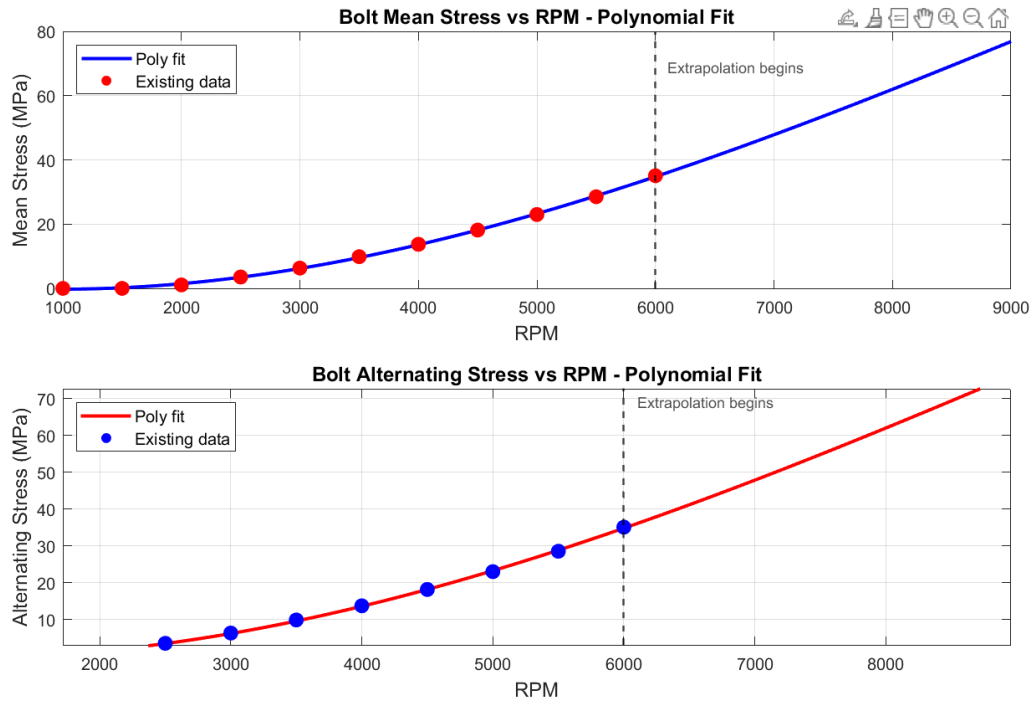


Figure 13: Bolt Mean/Alternating Stresses VS RPM

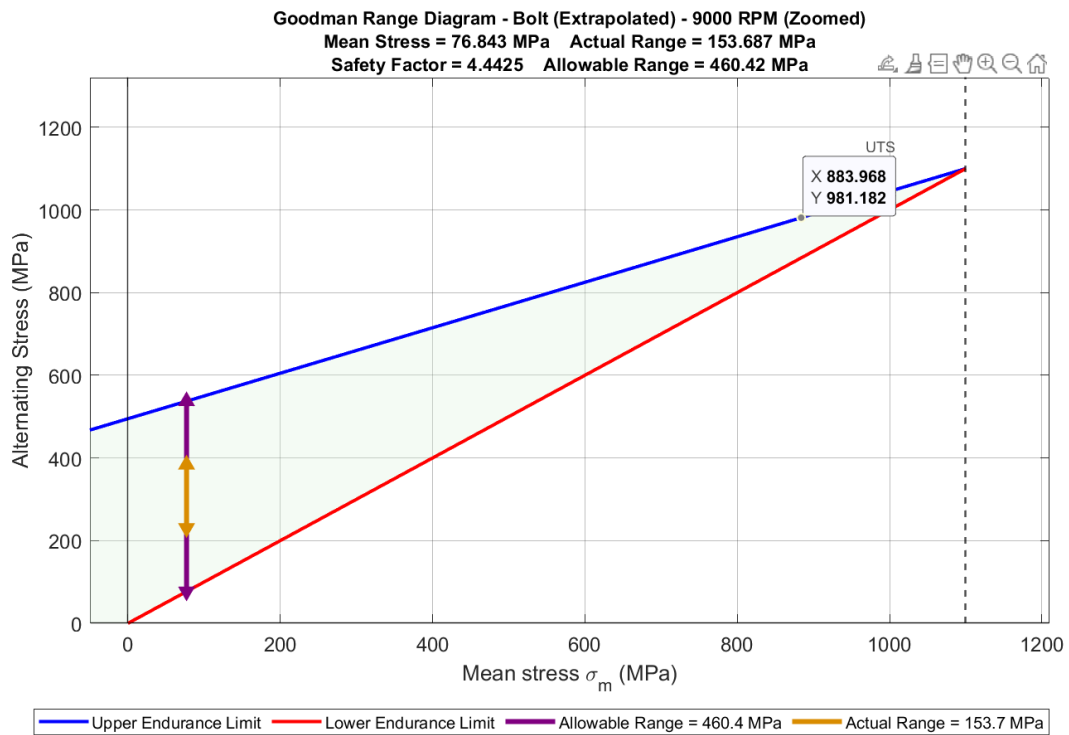


Figure 14: Extended RPM Goodman Range Diagram

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BOLT STRESS SUMMARY - Extended RPM Range					
=====					
RPM	Mean (MPa)	Alt (MPa)	Range (MPa)	SF	Source

1000	-0.248	-0.248	-0.496	Inf	Data
1500	0.199	0.199	0.398	1716.488	Data
2000	1.454	1.454	2.907	234.856	Data
2500	3.468	3.468	6.935	98.444	Data
3000	6.193	6.193	12.386	55.122	Data
3500	9.582	9.582	19.164	35.628	Data
4000	13.585	13.585	27.171	25.129	Data
4500	18.156	18.156	36.311	18.803	Data
5000	23.244	23.244	46.489	14.686	Data
5500	28.804	28.804	57.608	11.852	Data
6000	34.785	34.785	69.571	9.814	Data
6500	41.141	41.141	82.282	8.298	Extrap
7000	47.822	47.822	95.645	7.138	Extrap
7500	54.781	54.781	109.563	6.232	Extrap
8000	61.970	61.970	123.940	5.509	Extrap
8500	69.340	69.340	138.680	4.923	Extrap
9000	76.843	76.843	153.687	4.443	Extrap

Figure 15: Bolt Stress Summary Incl. Extended RPM

4 Bolt Diameter Redesign

In order to restore the bolt safety factor at 9000 RPM to a level comparable to the original 6000 RPM, the bolt cross-sectional area was increased until the target safety factor of 9.74 was reached. The MATLAB redesign output shows the required bolt diameter increases from 7.36mm to 10.88mm, representing a diameter increase of 48.06% and an area increase from 42.43 mm² to 93.02 mm². This is a 119.23% increase. The larger cross-section directly reduces both mean and alternating stress through the distribution of the same inertia-driven force over a greater area. This brings the mean stress at 9000 RPM down from 76.84 MPa to 35.05 MPa and halves the actual stress range from 153.69 MPa to 70.10 MPa. The target safety factor of 9.74 is achieved, as confirmed by the MATLAB output. The Goodman diagram comparing the original and redesigned bolt at 9000 RPM clearly illustrates the improvement.

The original bolt operating point sits further from the origin than the redesigned point, and while both remain comfortably within the safe zone, the redesigned bolt sits much closer to the cluster of original 1000-6000 RPM data points. The range diagram for the redesigned bolt at 9000 RPM shows an actual stress range of 70.1 MPa against an allowable range of 479.2 MPa. This is a close mirroring of the margin available at 6000 RPM with the original geometry.

The area increase of 119.23% is significant and has notable practical implications. Naturally, the larger bolts will need correspondingly larger through-holes in both the big-end cap and con-rod shank, which would serve to reduce the net cross-sectional area of those components and in turn could introduce localised stress concentration at the bolt boss. The additional bolt mass also adds to the rotating assembly weight at the big end, which increases bearing loads and may require rebalancing. Having said that, the geometry change is straightforward to implement and does not require a material change. Due to this, the bolt specification is to be retained at a larger diameter.

To reiterate, the preload inaccuracy applies to the redesigned bolt. The safety factor of 9.74 at 9000 RPM assumes no preload, and adding a realistic preload stress would substantially reduce this figure. Despite this, the redesign does provide considerably more headroom before the true factor approaches the Goodman line compared to the original geometry. Strict torque control remains essential for any practical implementation at extended RPM. The redesigned bolt should be treated as a minimum change rather than a complete solution. For example, the process of heat treatment should be considered in order to gain increased wear resistance.

4.1 Bolt Diameter Redesign Figures

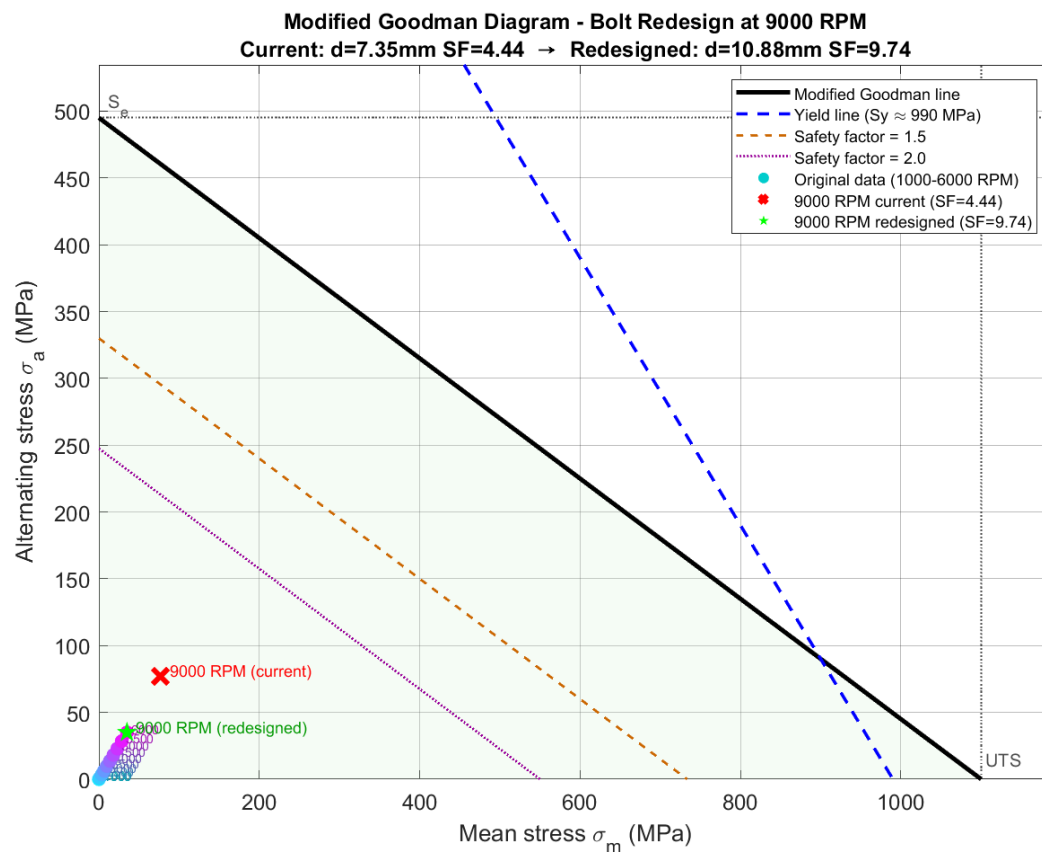


Figure 16: Redesigned Bolt Diameter Modified Goodman Diagram

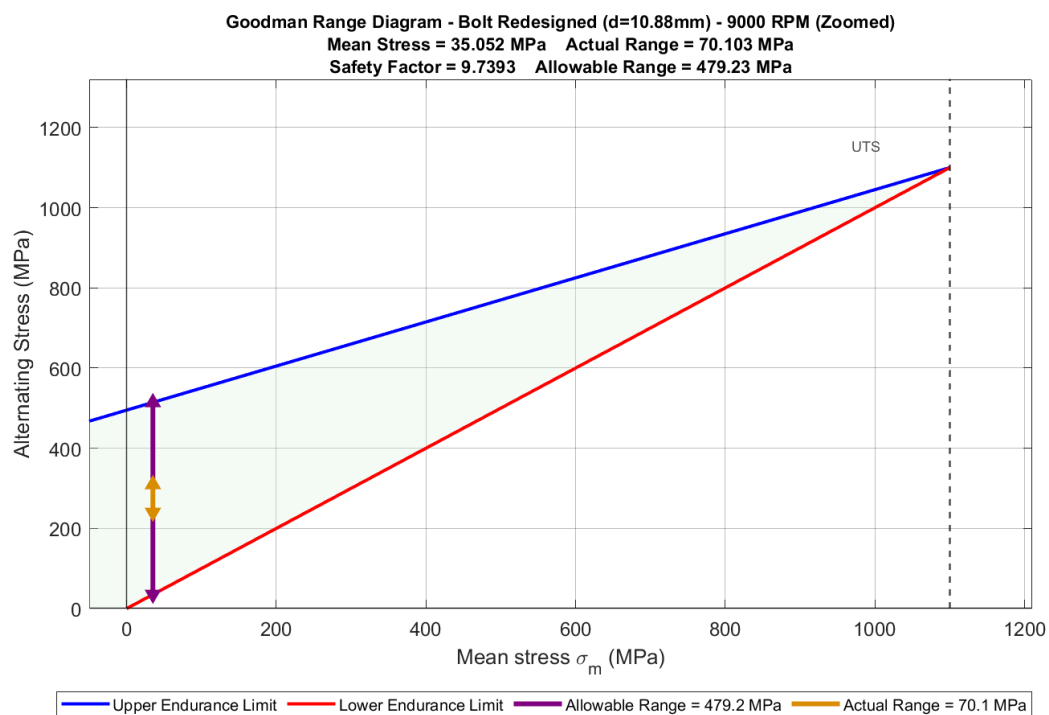


Figure 17: Redesigned Bolt Diameter Goodman Range Diagram

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BOLT REDESIGN RESULTS
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Current diameter  = 7.3501 mm
Required diameter = 10.8828 mm
Diameter increase = 48.06%
Current area      = 42.4300 mm^2
Required area     = 93.0187 mm^2
Area increase     = 119.23%
Target SF         = 9.7393
Achieved SF       = 9.7393
=====

```

Figure 18: Bolt Redesign Table Output

5 Standard and Cross Plane Analysis

The instantaneous torque outputs for both crankshaft configurations have been calculated across the full RPM range through the combination of the gas torque and inertia torque from all four cylinders; the firing order has been realigned to suit each configuration. The torque summary table shows that the mean torque values are identical for both configurations at every RPM. As expected at 6000 RPM both produce a mean of 379.82 Nm. The mean torque is determined by the thermodynamic work done per cycle independently of crankshaft geometry. The distinction between the two configurations lies entirely in how the instantaneous torque fluctuates around a set mean; the character of each layout becomes apparent here.

Looking at the inertia torque plots at 6000 RPM, the standard plane configuration produces oscillations with a peak-to-peak amplitude of around 900 Nm, while the cross plane configuration produces a smaller and more irregular oscillation of roughly 300 Nm peak-to-peak. This difference becomes apparent because in a standard plane crank all four con-rods share the same two crank planes, causing the inertia forces to add and subtract in a regular pattern. The cross plane crank is spaced at 90° intervals, which causes inertia contributions from adjacent cylinders to partially cancel, reducing the net inertia torque variation. Conversely, the gas torque plots in the cross plane layout produces higher peak gas torques approaching 1,130 Nm compared to around 930 Nm for the standard plane. The 90° spacing means combustion events from different cylinders overlap less and each pulse is more isolated. This produces sharper and taller torque peaks, contrasting the broader peaks of the standard plane. The total torque plots combine both effects, at 6000 RPM the cross plane produces peaks approaching 1250 Nm against 590 Nm for the standard plane, but also deeper negative troughs reaching around -450 Nm compared to the few negative values for the standard plane.

The all-RPM total torque plots show how these characteristics scale against RPM. The standard plane configuration maintains a relatively consistent torque delivery across the RPM range, with limited negative drops and steep combustion peaks. The cross plane plot shows more complex waveforms with sharper peaks and more pronounced dips. The negative torque in the 200° to 360° crank angle region increases with RPM as inertia effects increase. This is reinforced by the piston acceleration surface plot, which demonstrates that peak accelerations, and thus inertia forces by extension, increase with RPM. This affects the cross plane layout where partial cancellation of inertia torques becomes less effective as the individual cylinder contributions increase.

These characteristics make the two configurations suited to different applications. Due to its smooth, consistent torque delivery and lower peak values, the standard plane crankshaft is well suited to high-revving applications where balanced power is valued over raw torque intensity. This is why standard plane cranks are traditionally found in Formula 1 and high-performance V8s like the Ferrari 458. Conversely, the cross plane configuration produces the uneven firing intervals and higher torque peaks that result in the strong mid-range pull of V8 muscle car engines. The greater torque variation is managed through a heavier flywheel. For this four-cylinder engine, the cross plane configuration results in higher instantaneous torque peaks which could improve throttle response and power delivery at high RPM. The deeper negative torque dips and more complex vibration signature would demand a more robust drivetrain and more careful NVH management over the more refined output of the standard configuration.

5.1 Standard and Cross Plane Analysis Figures.

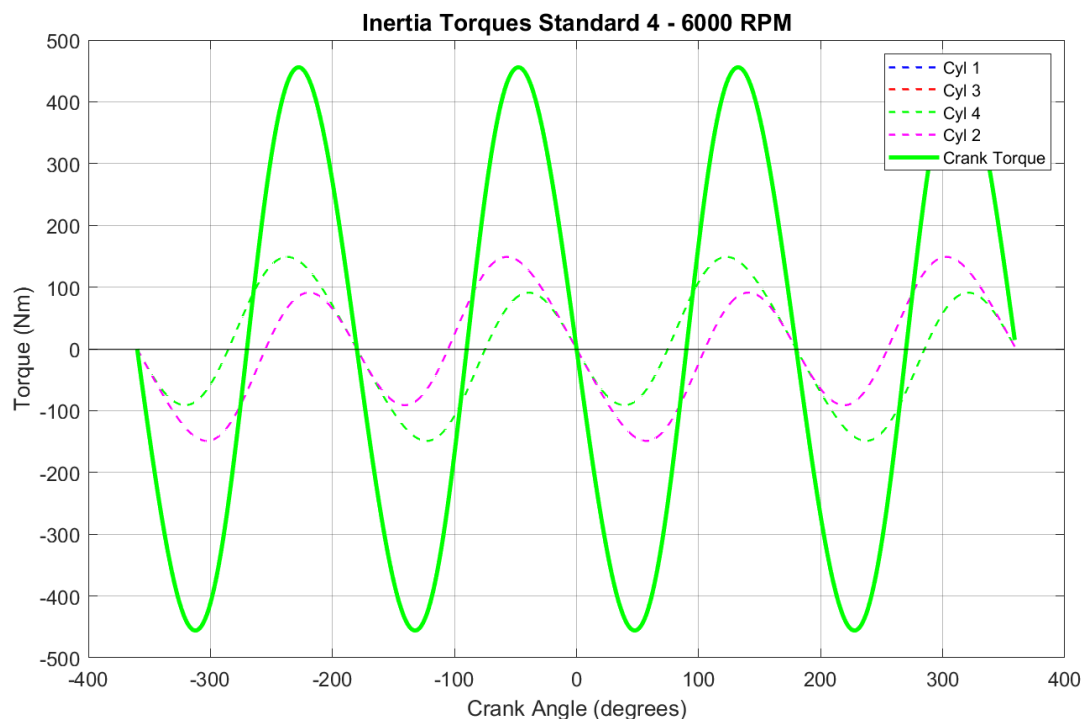


Figure 19: Inertia Crank VS Torque Standard Plane (Offset by 180 Deg per cylinder)

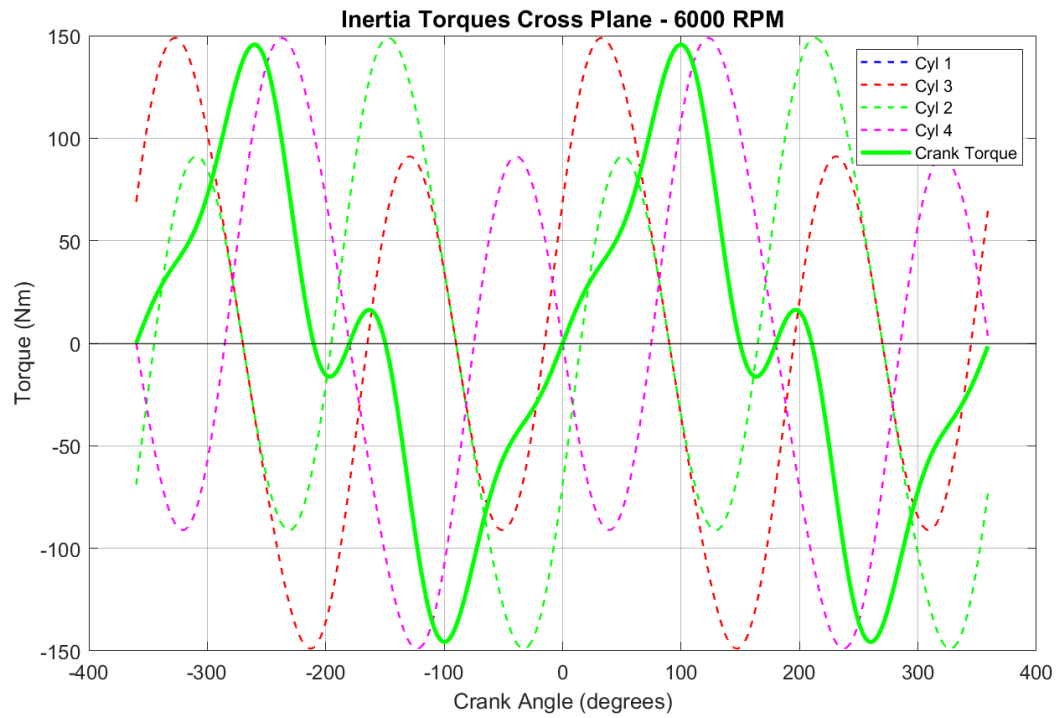


Figure 20: Inertia Crank VS Torque Cross Plane (Offset by 180 deg per cylinder)

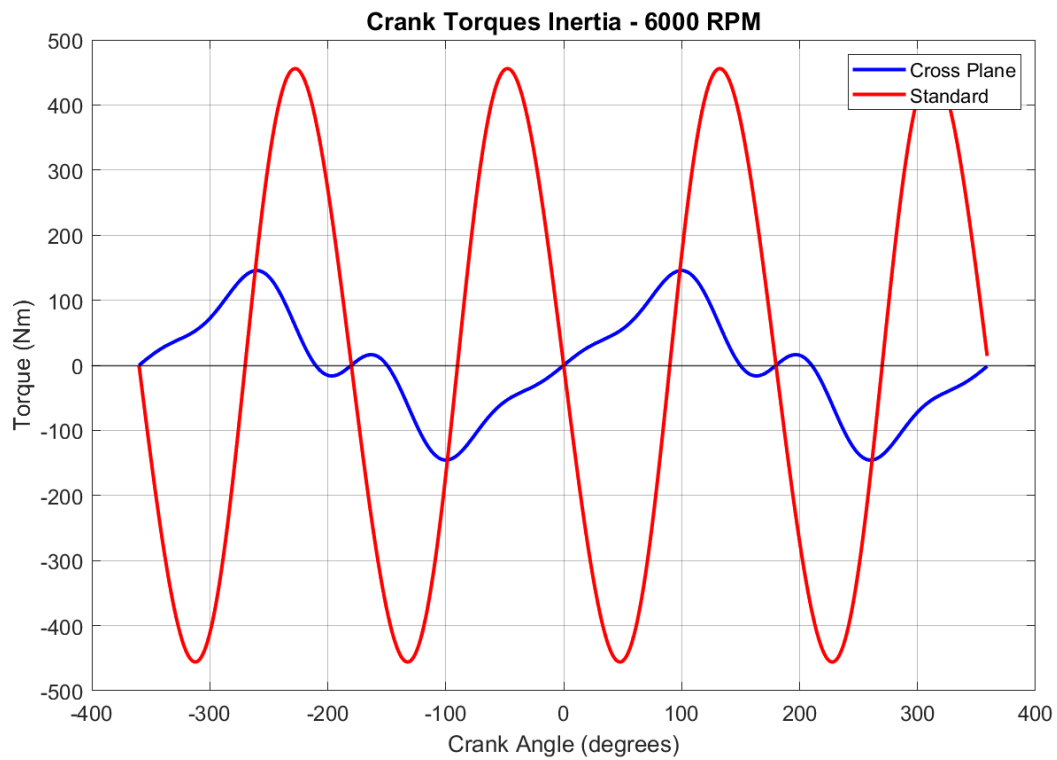


Figure 21: Crank Inertia @ 6000RPM

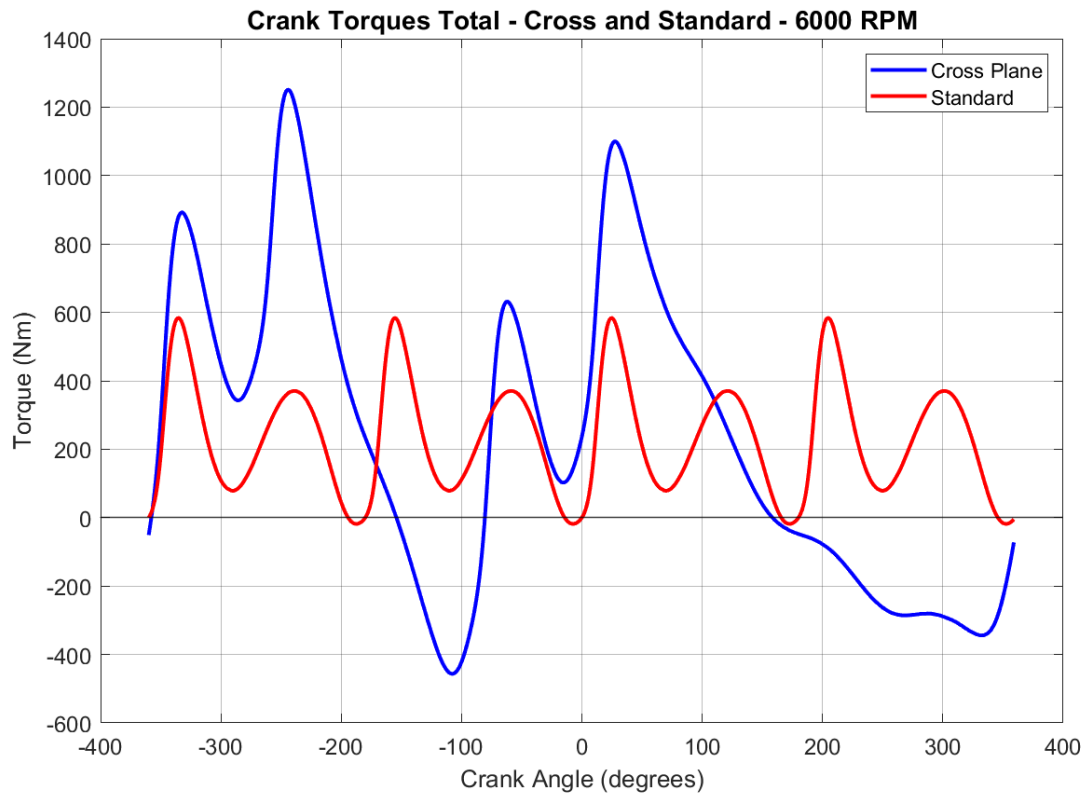


Figure 22: Crank Torque Total - Standard and Cross Plane @ 6000RPM

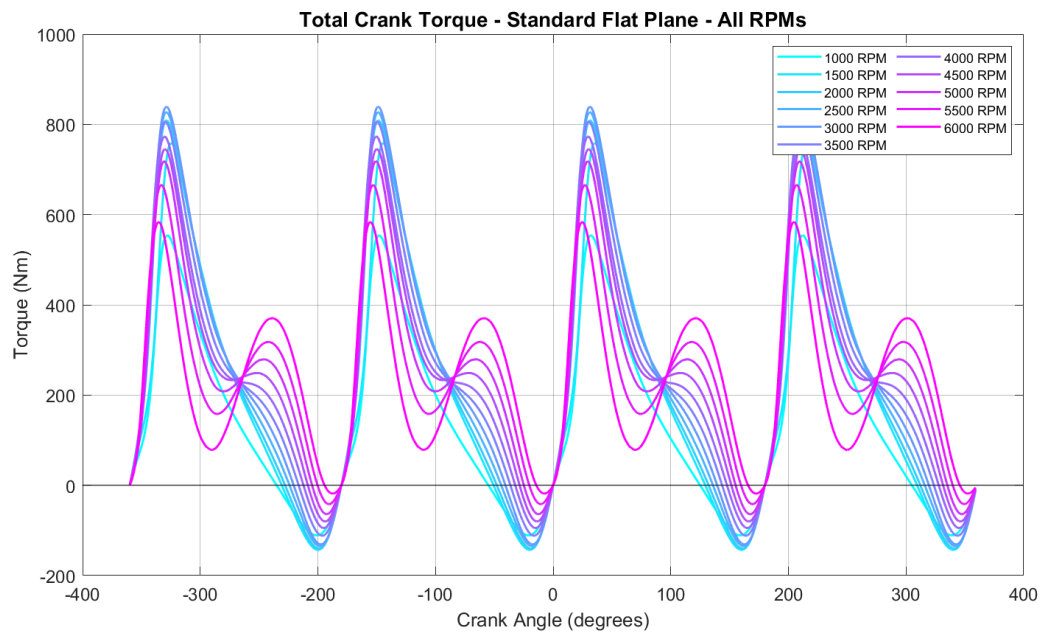


Figure 23: Total Crank Torque - Standard Plane @ All RPMs

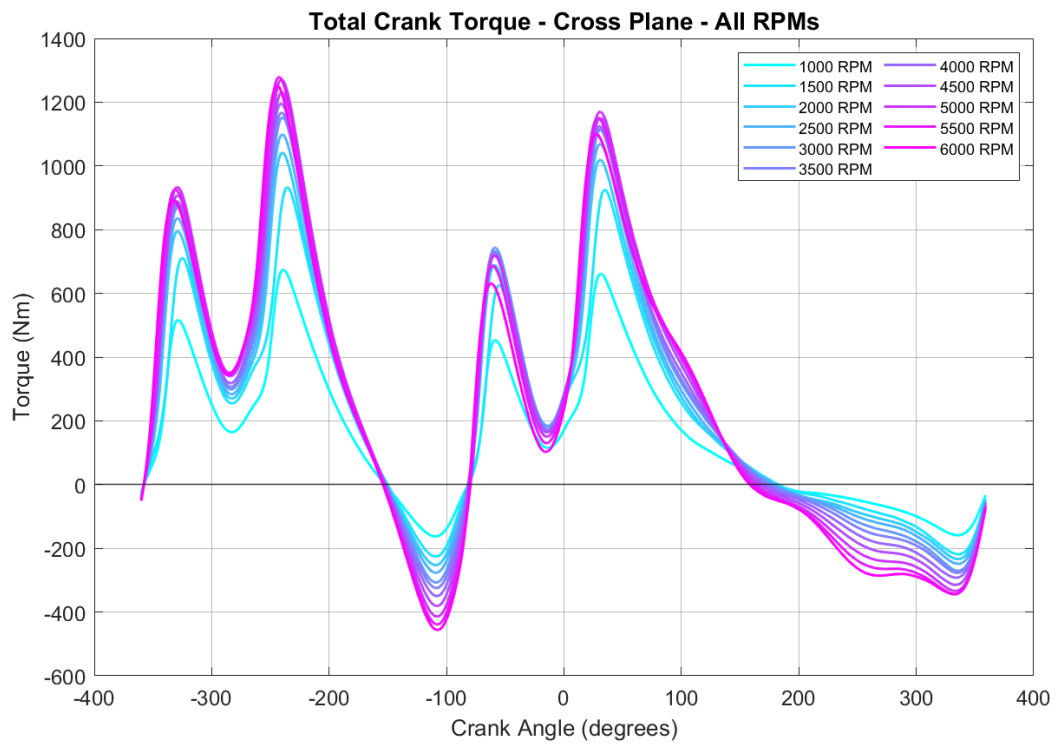


Figure 24: Total Crank Torque - Cross Plane @All RPMs

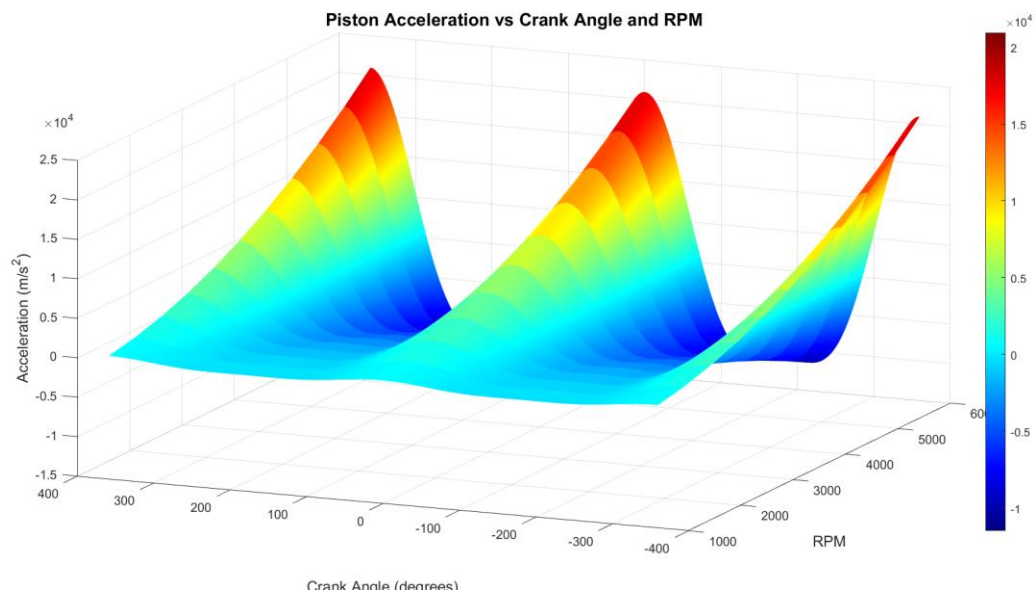


Figure 25: Piston Acceleration vs Crank Angle and RPM

6 Conclusion

This report has analysed the fatigue behaviour of three key bottom-end components of the Ford 1.6L Sigma engine, from 1000 to 6000 RPM, and extended to 9000 RPM through polynomial interpolation.

The con-rod presented over-engineering with infinite fatigue life predicted throughout. The gudgeon pin was the most critical component, with safety factors plateauing at 2.2 to 2.3 above 4000 RPM. This margin reduces further when fretting fatigue is considered. The con-rod bolt analysis highlighted the significant impact of preload on true fatigue margin. The safety factors substantially overestimated real-world performance. With strict preload control remaining essential, the proposed diameter increase from 7.36 mm to 10.88 mm successfully restores the 9000 RPM safety factor to the original 6000 RPM level.

Finally, the crankshaft comparison confirmed that while both configurations produce identical mean torque, the standard plane favours high-revving refinement whereas the cross plane delivers stronger torque peaks. This suggests that the two configurations are better suited to different performance applications.

7 References

- Xiaochun, Wu; Rui, Chen; Jun, Zhang, *Surface Treatment Process for Automobile Engine Piston Pins* (Shanghai: Shanghai University, 2005) [Accessed: 22/03/2026]