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Analytical Assessment of Engine Internal Components

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Contemporary Powertrain Design (ACHA6006)

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Introduction

The internal components of an engine are subjected to complex cyclic loads, influencing their fatigue life and overall durability. This report conducts an analytical assessment of key engine components, focusing on stress distribution, fatigue safety factors (FSF), and the impact of design modifications. Using first-principle calculations and interpolated data, the forces acting on the Piston, Gudgeon Pin, Connecting Rod, and Bolts are evaluated across a range of engine speeds, identifying critical stress points.

A core aspect of this analysis is the influence of stress concentrations on fatigue performance, particularly in the Connecting Rod Bolts. The study explores geometric optimisations such as varying shank dimensions to enhance FSF while maintaining structural integrity. Additionally, material variations and surface treatments are investigated to assess their potential in improving component fatigue resistance.

Beyond static stress evaluation, this report examines engine torque characteristics for different crankshaft configurations, comparing standard and cross-plane layouts in inline-four and V8 configurations. The impact of these designs on torque fluctuation, power delivery, and drivability is discussed, offering insights into their suitability for various applications.

Sigma Engine Components Stress Analysis

Engine Force Calculations

To accurately determine the Ford Sigma Engine components Fatigue Safety Factors, an analysis of the forces and stresses endured within the system is required. Initial calculations using the first-principle equations determines the estimated Piston Velocity, Piston Acceleration and ultimately Piston Inertia Force.

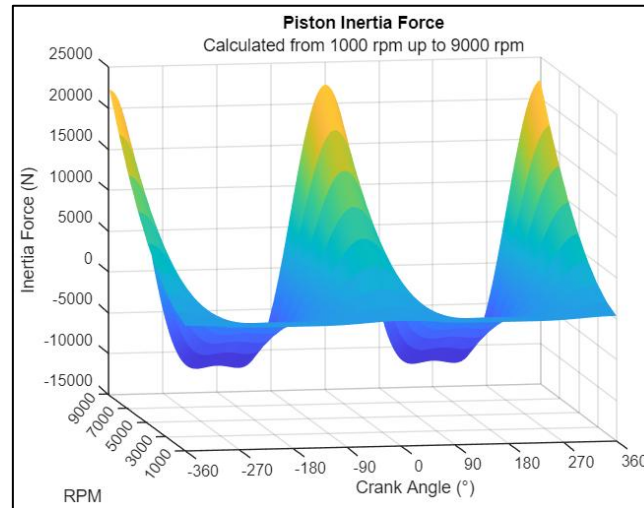


Figure 1: Piston Inertia Force during a single Crank Rotation at various RPMs

Figure 1 shows the calculated inertia force from 1000 to 9000 rpm. The cyclic, non-linear loading between tension and compressive loads increases exponentially placing greater structural fatigue stress demands on the components.

Ford Sigma Provided Pressure Data & Predicted Pressures

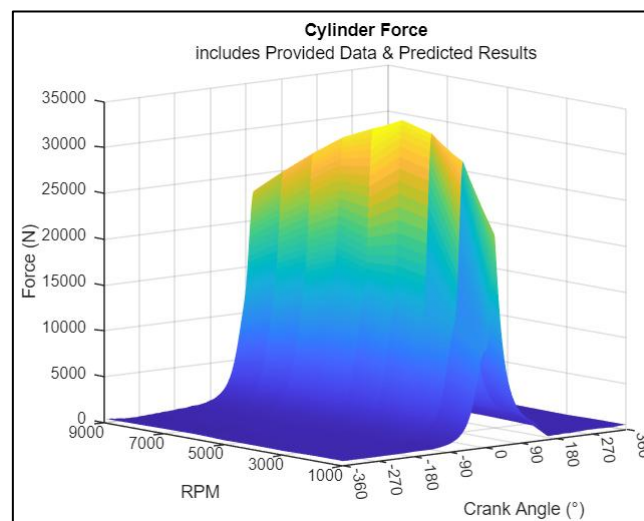


Figure 2: Calculated and Predicted Compression Force within a single Ford Sigma Cylinder

Figure 2 visually represents the cylinder forces applied to the top of the piston. The volumetric efficiency of the combustion within the cylinder has been considered, when predicting 7000 – 9000 RPM, to reflect real-world performance of a naturally aspirated engine, this does allow an element of error within the Fatigue Safety Factor (FSF) calculations.

Resulting Piston Forces

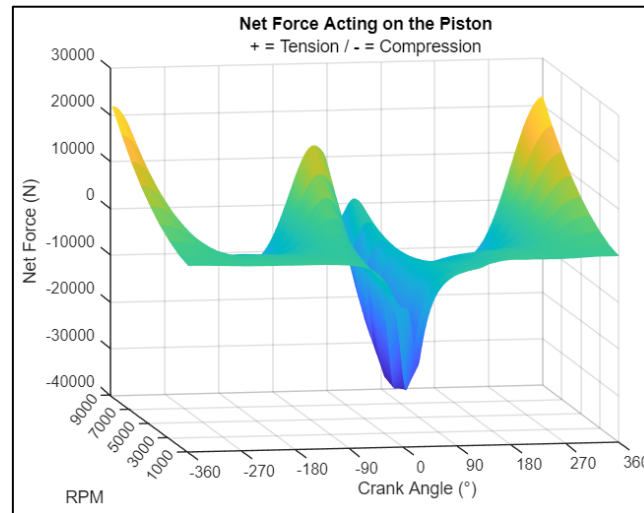


Figure 3: Calculated Combined Force acting on the Piston Top

Figure 3 combines the previous calculated forces to determine the net force experienced by the piston and ultimately at the small end of the connecting rod. The compression of the cylinder force is now considered negative.

Piston Side Frictional Force & Further Analysis Assumptions

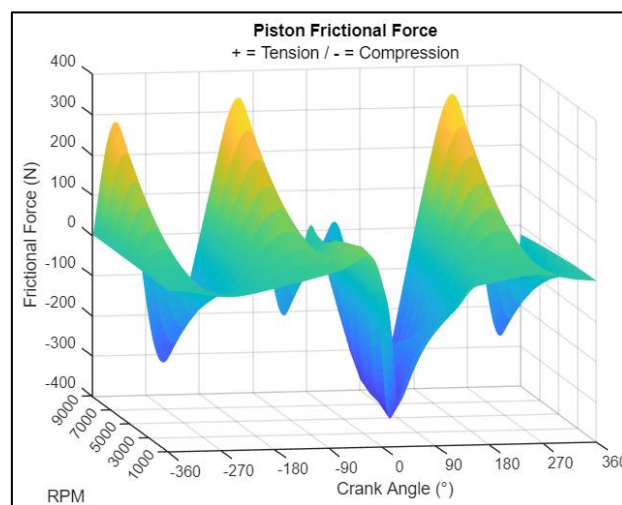


Figure 4: Calculated Piston and Cylinder Side Wall Friction

To further improve the accuracy of the kinematic calculations an assumed frictional loss at μ of 0.1 is displayed in figure 4. However, this has not been used in the final FSF calculations as the worst case (highest force) has been used, this ensures a further level of safety in case of improved friction modifications within the engine.

Kinematic Assumptions:

- Rigid Body: System is considered rigid, neglecting components elasticity
- Steady-State Conditions: Each thousand RPM, ignoring transient effects
- Simplified Mass Distribution: Reciprocating mass is estimated as 1/3 of Connecting Rod
- Interpolated Pressure Data: Data set created from simple engine theory

Connecting Rod Resultant Stress

Table 1: Connecting Rod Goodman Diagram Stress Results based on Ford Sigma Material Specifications

Engine Speed (RPM)	Maximum Compression (MPa)	Maximum Tension (MPa)	Mean Stress (MPa)	Stress Range (MPa)
1000	-164.93	-1.56 (<i>Compression</i>)	-83.25	163.37
2000	-212.11	4.36	-103.87	216.47
3000	-220.18	14.68	-102.75	234.86
4000	-213.70	28.26	-92.72	241.97
5000	-187.25	46.37	-70.44	233.62
6000	-156.14	67.21	-44.47	223.35
7000	-116.06	92.91	-11.57	208.97
8000	-87.72	122.59	17.436	210.32
9000	-104.17	156.23	26.03	260.4

Table 2: Table displaying the Fatigue Safety Factors for each RPM

Sigma Specification Material Fatigue Safety Factor (1000-9000 RPM)								
4.59	3.55	3.27	3.13	3.16	3.20	3.28	3.13	2.50

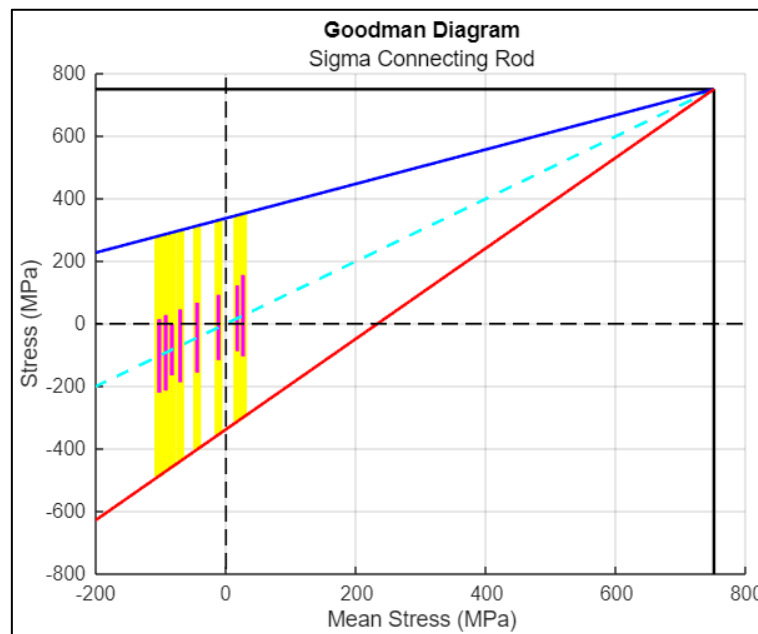


Figure 5: Goodman Diagram for Connecting Rod with Ford Sigma Material Specification (1000 – 9000 RPM)

The increase stress range at higher RPMs indicates the greater cyclic loading, while the mean stress shifts from negative (compression) to positive (tension), highlighting the transition in loading behaviour and the impact of the inertia forces. The FSF results show the over engineering design and added weight of the connecting rod. However, the increased stress range at 9000 rpm indicates that long-term high RPM operation could still lead to fatigue concerns.

Ford Sigma Connecting Rod Varied Material Investigation

The choice of component material helps to optimise strength-to-weight ratio as well as fatigue resistance. The chosen materials consist of a range of potentially possible materials with varying strength and fatigue properties.

Table 3: Displaying the Material Data used for the Material Variation Investigation. The remaining three Goodman Diagrams are included in the Appendix (1,2,3).

Material Name	Material UTS (MPa)	Material Endurance Limit (%)
Sigma Specification Material	750	45
4340 Steel	1080	50
Ti-6Al-4V	950	40
7075-T6 Al	570	35

Table 4: A table displaying the calculated FSF of each Connecting Rod Materials

Engine Speed (RPM)	Sigma Specification Material	4340 Steel	Ti-6Al-4V	7075-T6 Al
1000	4.59	7.12	5.06	2.80
2000	3.55	5.47	3.89	2.18
3000	3.27	5.04	3.59	2.01
4000	3.13	4.85	3.45	1.92
5000	3.16	4.92	3.49	1.92
6000	3.20	5.03	3.56	1.93
7000	3.28	5.22	3.68	1.95
8000	3.13	5.05	3.55	1.84
9000	2.5	4.05	2.84	1.46

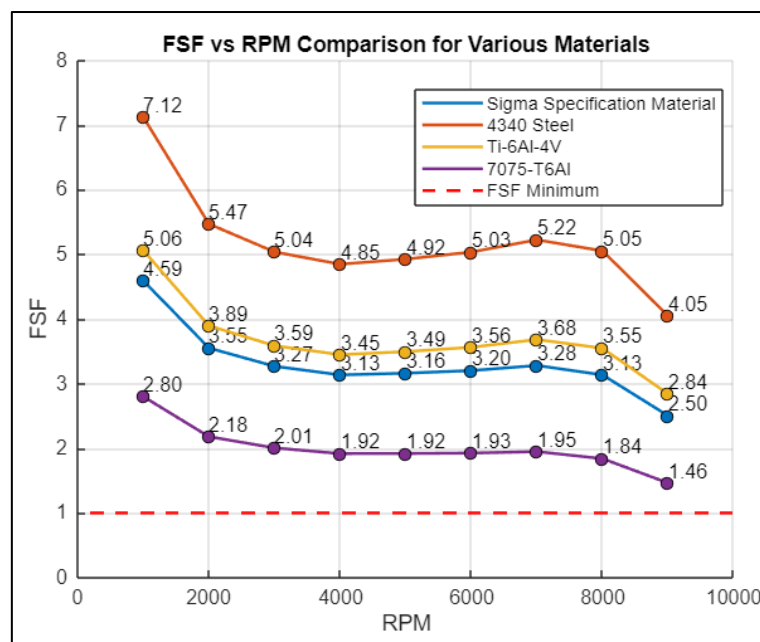


Figure 6: A visual representation of the Material Investigation FSF across each RPM

Titanium offers the best balance, providing a higher FSF while being lighter than steel and the original Sigma Material, making it ideal for weight-sensitive applications. Aluminium's FSF is too low for safe implementation without further analysis (thermal) or durability improvements.

Ford Sigma Connecting Rod Varied Treatment Process Data

Table 5: Hardness of Heat Treatment Process (Jamkhedkat, Ketanmarathe, Khanvikar, Sandeshgaonkar, 2018)

Hardening Process	Hardening & Tempering	Nitriding	Carburising
Surface Improvement due to Process	83 MPa	62 MPa	123 MPa

Surface treatment and finishing processes play a critical role in improving fatigue life, wear resistance and overall durability. Using various techniques reduces the likelihood of crack initiation and propagation under cyclic loading.

Table 6: Stress results for Goodman Diagram for Carburising Treatment Process

Engine Speed (RPM)	Maximum Compression (MPa)	Maximum Tension (MPa)	Mean Stress (MPa)	Stress Range (MPa)	FSF
1000	-287.93	-124.56	-206.24	163.37	5.27
2000	-335.11	-118.64	-226.87	216.47	4.06
3000	-343.18	-108.32	-225.75	234.86	3.74
4000	-336.70	-94.74	-215.72	241.97	3.59
5000	-310.25	-76.63	-193.44	233.62	3.63
6000	-279.14	-55.79	-167.47	223.35	3.69
7000	-239.06	-30.09	-134.57	208.97	3.81
8000	-210.72	-0.41	-105.56	210.32	3.66
9000	-227.17	33.23	-96.97	260.4	2.93

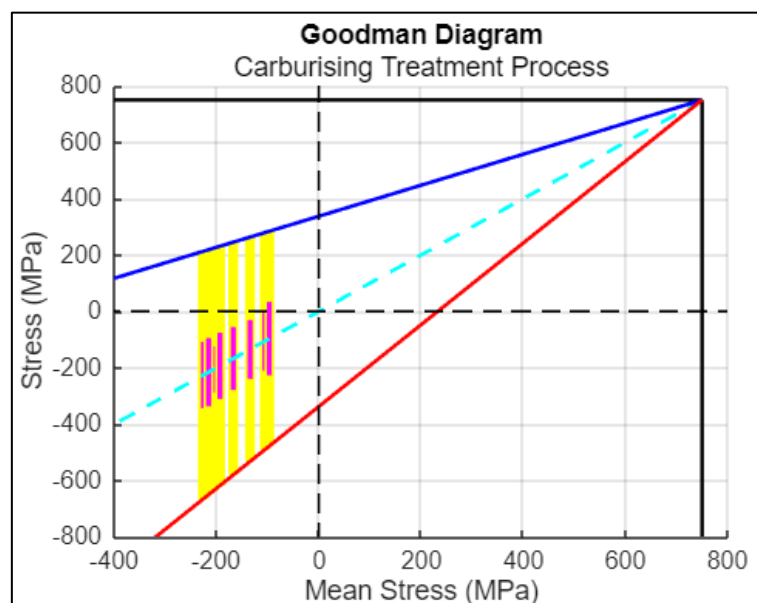


Figure 7: Goodman Diagram for Connecting Rod with Ford Sigma Material Specification and Carburising Treatment (1000 – 9000 RPM)

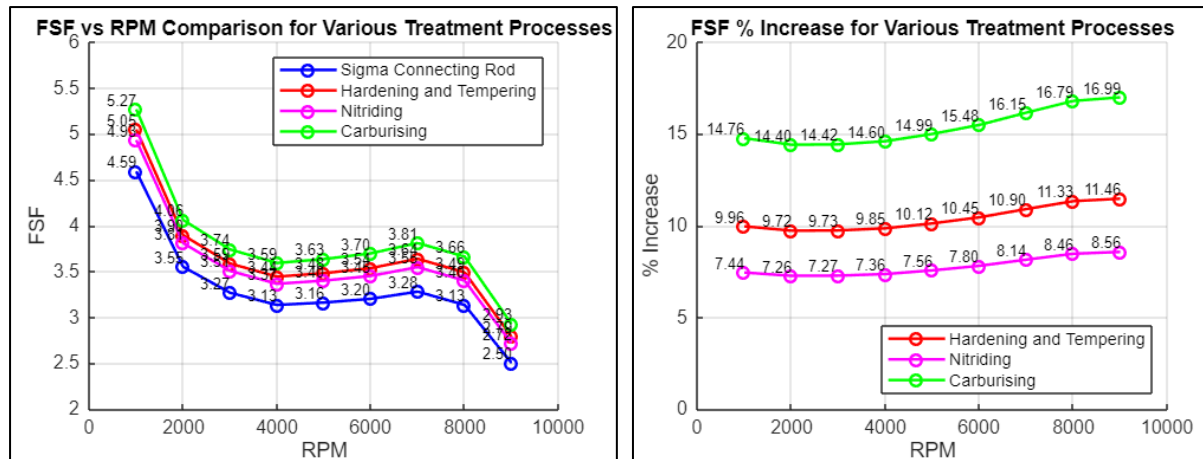


Figure 8: 2 graphs showing the FSF of each Treatment Process and the % Increase from the Original FSF

Carburising improves the fatigue resistance across all RPMs, with the highest increase of FSF with around 15% gain. The Goodman Diagram shows the improved tensile resistance, but the compression remains dominant. While carburising is beneficial, hybrid treatments like shot peening maybe needed for sustained high-performance reliability.

Carburising emits CO₂ and NO_x, while nitriding produces harmful nitrogen by-products. Though energy intensive, Hardening & Tempering is cleaner, benefiting from green energy and no chemical exposure. As seen in figure 8, its slight FSF reduction is justified by lower environmental and social impact. (Cheng, Russell, Shunmugam, Lazoğlu, 1985)

Gudgeon Pin Resultant Stress

The Gudgeon Pin plays a crucial role in transferring forces between the Piston and Connecting Rod, experiencing both bending and shear stresses due to its support conditions and cyclic loading. The forces acting on the pin are a combination of compressive gas pressure forces pushing downward and inertial forces pulling upward, with the net force being shared 50% per side, assuming a symmetric design.

Due to these loading conditions, the pin undergoes both downward bending due to combustion and upward bending due to the inertia forces. Additionally, shear stresses occur at the support points, which define the regions of highest stress concentration.

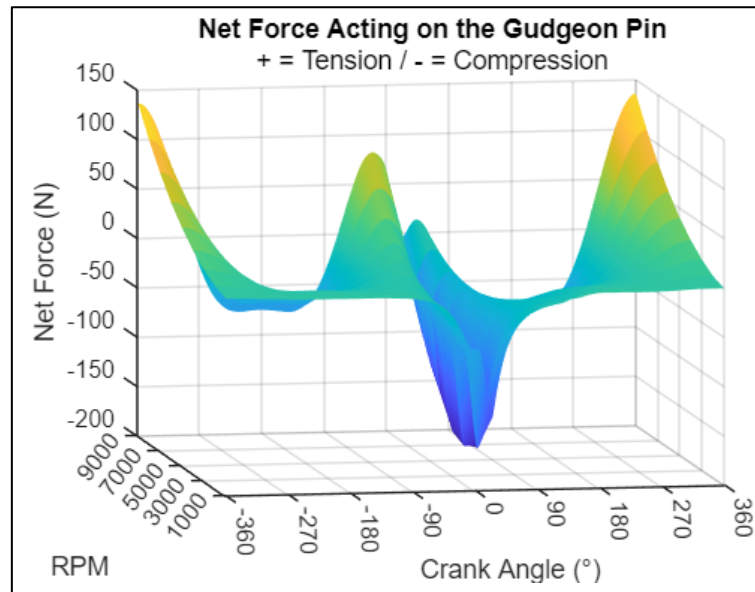


Figure 9: Calculated Combined Force acting on the Gudgeon Pin

The cyclic net force loading is evident in figure 9, from this it is possible to outline the fatigue requirements and ensure appropriate design options to ensure long term durability.

Table 7: Stress results for Goodman Diagram for Gudgeon Pin with Engine Specification Material

Engine Speed (RPM)	Maximum Compression (MPa)	Maximum Tension (MPa)	Mean Stress (MPa)	Stress Range (MPa)	FSF
1000	-128.35	-1.07	-64.71	127.28	5.87
2000	-165.02	4.06	-80.48	169.08	4.49
3000	-170.81	12.94	-78.93	183.75	4.13
4000	-164.71	24.71	-69.99	189.42	3.98
5000	-142.63	40.31	-51.16	182.94	4.05
6000	-116.73	58.42	-29.15	175.15	4.15
7000	-83.55	80.65	-1.45	164.20	4.31
8000	-72.53	106.32	16.89	178.85	4.21
9000	-86.75	135.41	24.33	222.16	3.37

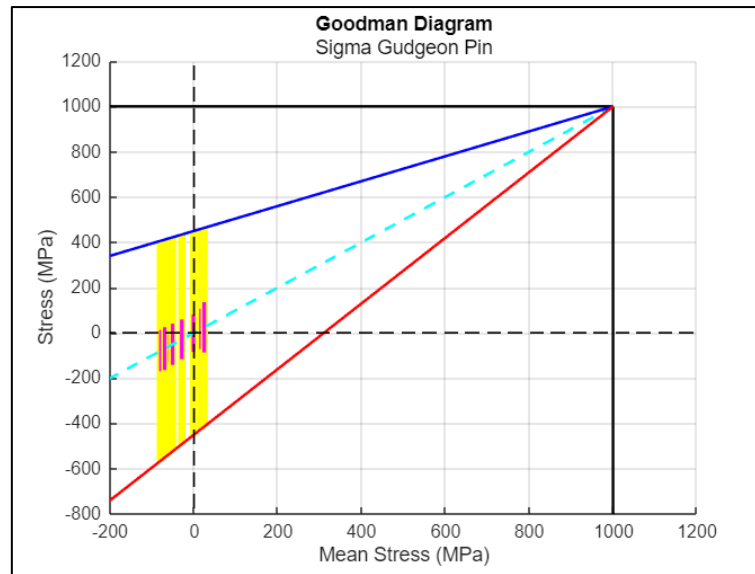


Figure 10: Goodman Diagram for Gudgeon Pin with Ford Sigma Material Specification (1000 – 9000 RPM)

Ford Sigma Gudgeon Pin Varied Material Data

Table 8: Displaying the Material Data used for the Material Variation Investigation. The remaining three Goodman Diagrams are included in the Appendix (4,5,6).

Material Name	Material UTS (MPa)	Material Endurance Limit (%)
Sigma Gudgeon Pin	1000	0.45
52100 Bearing Steel	2000	0.5
17-4 PH Stainless	1200	0.45
7075-T6 Aluminium	570	0.35

Table 9: A table displaying the calculated FSF of each Gudgeon Pin Materials

Engine Speed (RPM)	Sigma Gudgeon Pin	52100 Bearing Steel	17-4 PH Stainless	7075-T6 Aluminium
1000	5.87	12.64	6.97	2.72
2000	4.49	9.61	5.32	2.10
3000	4.13	8.85	4.90	1.93
4000	3.98	8.55	4.72	1.85
5000	4.05	8.78	4.82	1.86
6000	4.15	9.08	4.95	1.88
7000	4.31	9.58	5.17	1.91
8000	4.21	9.43	5.06	1.84
9000	3.37	7.59	4.06	1.47

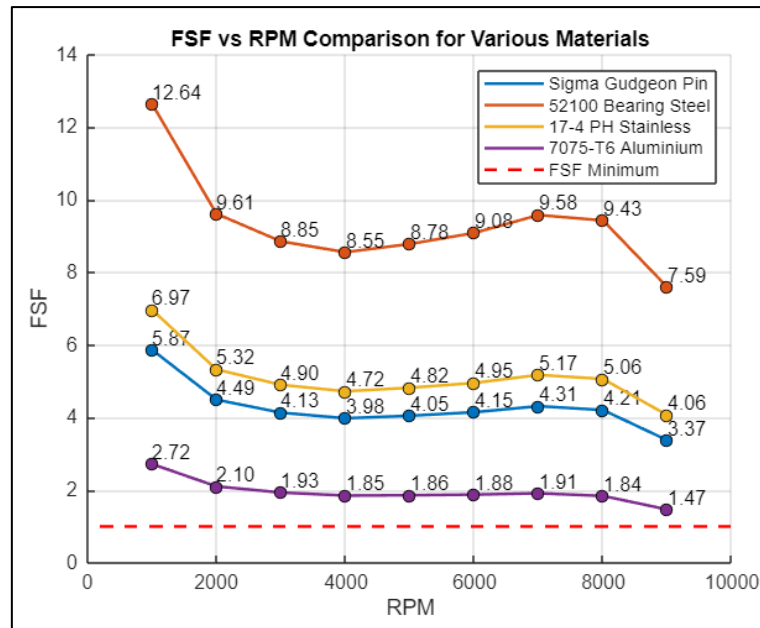


Figure 11: Each Material FSF at Each RPM

The large forces that are experienced by the relatively small component is evident in figure 11. The material selection of the Gudgeon pin significantly impacts the fatigue life and structural integrity. While bearing steel offers superior fatigue resistance, its increased density may impact weight-sensitive designs, where stainless steel or optimised aluminium alloys could be alternatives. However, aluminium's low FSF raises durability concerns, suggesting further investigation into hybrid treatments or coatings to enhance surface strength and wear resistance. Ultimately, 52100 bearing steel is the most viable choice for performance-focused applications, while stainless steel could provide a balance of strength and corrosion resistance.

Connecting Rod Bolt Resultant Stress Analysis

The Connecting Rod Bolts play a crucial role in maintaining the structural integrity of the assembly under these cyclic loads. The bolts are preloaded during assembly, creating a clamping force that holds the Connecting Rod and Cap together. This preload remains constant under static conditions, however during engine operation, additional tensile forces act on the bolts, this leads to bolt elongation and stress variations.

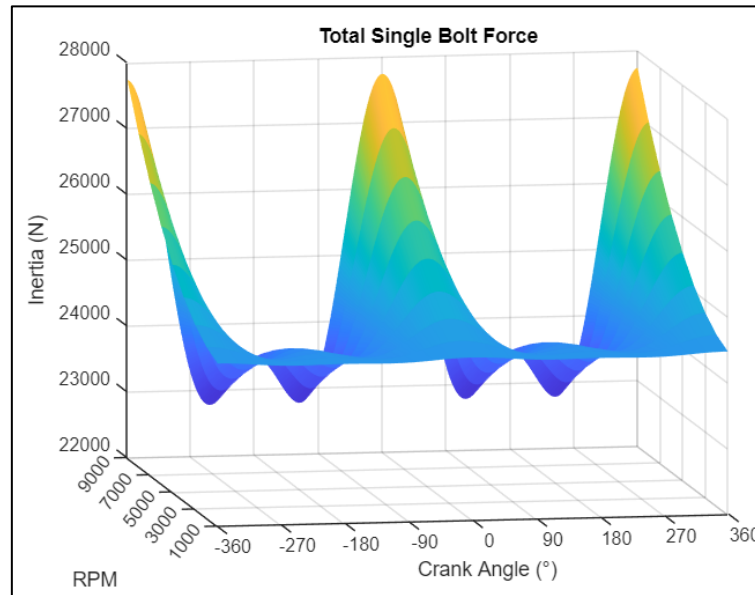


Figure 12: Calculated Combined Force acting on a single Connecting Rod Bolt

As opposed to the previous components figure 12 shows how the Bolt is constantly under tension yet is still under cyclic loads and is increased with RPM.

Table 10: Stress results for Goodman Diagram for Connecting Rod Bolts with Engine Specification Material

Engine Speed (RPM)	Minimum Tension (MPa)	Maximum Tension (MPa)	Mean Stress (MPa)	Stress Range (MPa)	FSF
1000	808.51	809.92	809.21	1.40	186.01
2000	808.51	814.14	811.32	5.62	46.16
3000	808.51	821.17	814.84	12.66	20.26
4000	808.51	831.02	819.77	22.51	11.20
5000	808.51	843.68	826.10	35.17	7.00
6000	808.51	859.16	833.83	50.64	4.72
7000	808.51	877.45	842.9	68.93	3.35
8000	808.51	898.55	853.53	90.04	2.46
9000	808.51	922.47	865.49	113.95	1.85

Note that the Minimum Tension is constant due to the preloaded tension, however, the stress range increases by over 80x, highlighting the increasing fatigue demand on the bolts. Along with this the FSF drastically decreases operating close to its fatigue limits, making them vulnerable to the cyclic loading failures. The Goodman Diagram in figure 10 shows how the mean stress rises, the operating stress points also shift closer to the fatigue failure threshold.

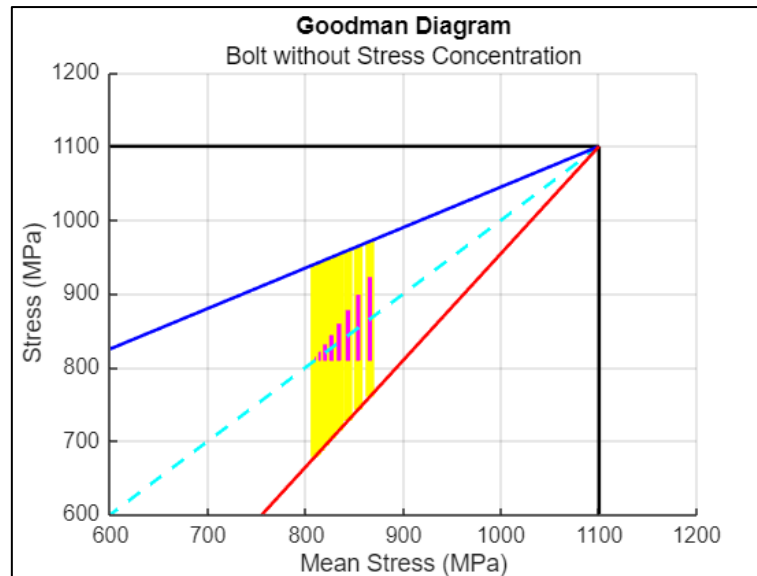


Figure 13: Goodman Diagram for Gudgeon Pin with Ford Sigma Material Specification (1000 – 9000 RPM)

Stress Concentration Impact

Stress concentrations are critical in mechanical design as they can significantly impact the fatigue life and structural integrity of components. They occur at geometric discontinuities such as sharp corners, holes, or sudden changes in cross-section, where applied stress is no longer evenly distributed, leading to localised stress peaks. This is particularly important in high-load cyclic applications like engine components, where stress concentrations can act as initiation sites for cracks and fatigue failure. (Concept, 2024)

Table 11: FSF change due to Bolt Stress Concentration Considerations

RPM	FSF	Stress Concentration FSF
1000	186.01	72.08
2000	46.16	17.89
3000	20.26	7.85
4000	11.20	4.34
5000	7.00	2.71
6000	4.72	1.83
7000	3.35	1.30
8000	2.46	0.95
9000	1.85	0.71

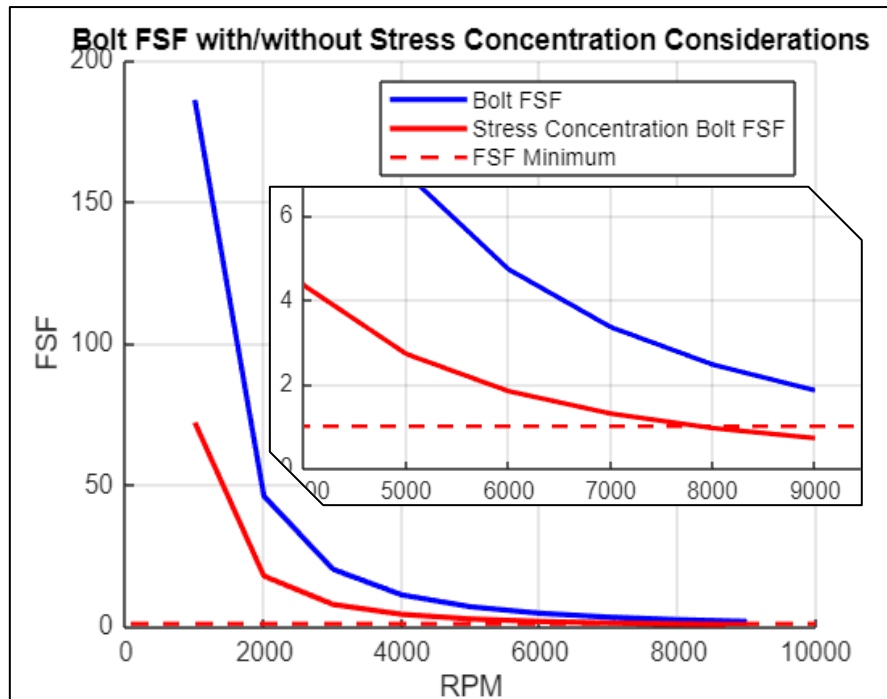


Figure 14: A graph showing the impact of Stress Concentrations on FSF across RPM range

With the stress concentrations taken into consideration the Ford Sigma Specification Bolts would experience fatigue failure as the FSF reaches the critical threshold of 1. As shown in figure 14 the stress concentration-adjusted FSF decreases more aggressively than the standard FSF, reinforcing the importance of accounting for localised stress effects in fatigue analysis. To mitigate this, optimising fillet radii, applying surface treatments or using materials with higher endurance limits further enhance fatigue resistance. (be-cu, 2024)

Bolt Redesign

Bolted joints are typically analysed using stiffness models, treating the bolt as a series of springs in parallel. The effective stiffness is determined by the axial rigidity of different sections, requiring component-wise evaluation. (Lu, Hua, Li, Cui, Li, 2019) By segmenting the bolt into three regions—the head, shank, and threaded portion—engineers can refine stiffness distribution, mitigate stress concentrations, and enhance fatigue safety. (SDC, 2024)

For the Ford Sigma Connecting Rod Bolts some boundary conditions need to be set before redesigning. Firstly, the length of the bolt will remain the same, even though the shank and thread sections will change in length, the overall bolt will remain the same length. Additionally, to this the threaded section will remain the same pitch and diameter. This investigation has specifically looked into the shank length and area to see the impact of saving weight while maintaining/improving the FSF.

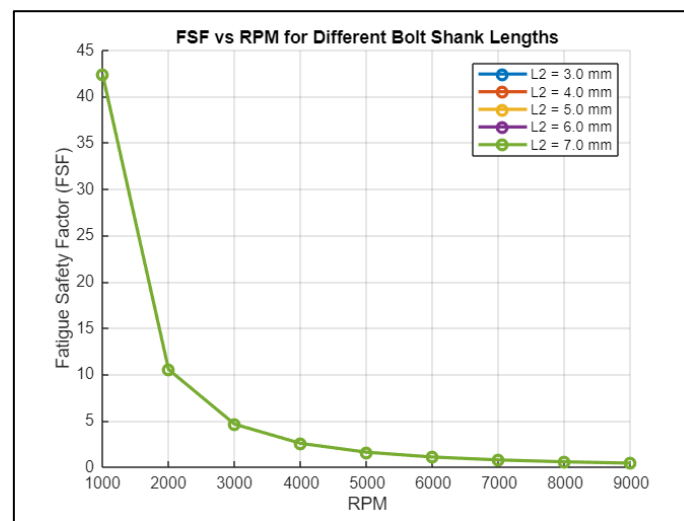


Figure 15: FSF plotted based on Bolt Shank Length change

Table 12: Bolt Shank Length changes FSF

RPM	3mm	4mm	5mm	6mm	7mm
1000	42.43	42.43	42.43	42.43	42.43
2000	10.53	10.53	10.53	10.53	10.53
3000	4.62	4.62	4.62	4.62	4.62
4000	2.55	2.55	2.55	2.55	2.55
5000	1.59	1.59	1.59	1.59	1.59
6000	1.07	1.07	1.07	1.07	1.07
7000	0.76	0.76	0.76	0.76	0.76
8000	0.56	0.56	0.56	0.56	0.56
9000	0.42	0.42	0.42	0.42	0.42

The results show no change in FSF across different shank lengths, indicating that shank length does not significantly impact fatigue performance. This is because the bolt's critical stress concentration points remain at the thread roots, where fatigue failure typically initiates. Since bending stiffness is primarily influenced by cross-sectional area rather than length, altering the length has little effect on stress distribution or fatigue resistance. The preload and clamping force also remain unchanged, further explaining why FSF is unaffected. (Hardlock, 2022)

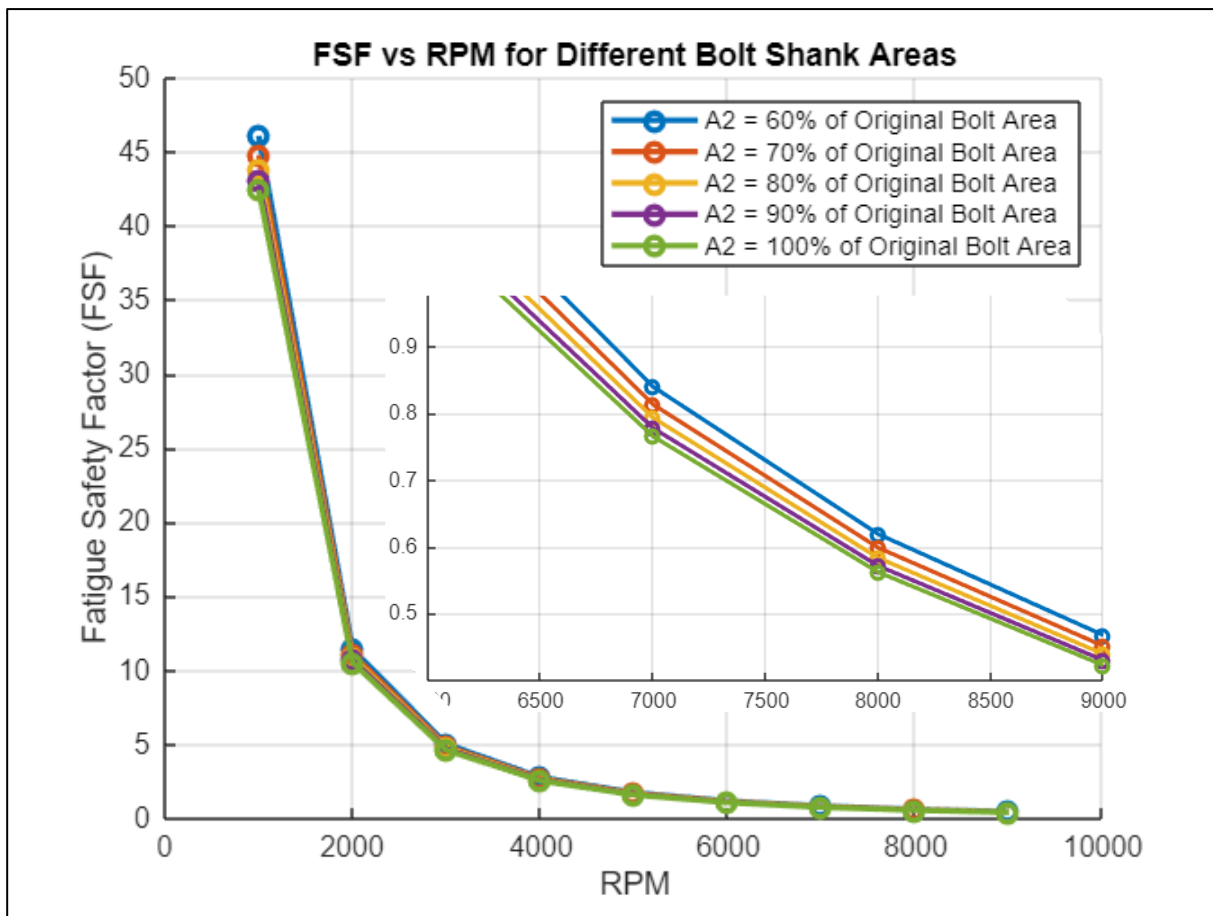


Figure 16: FSF plotted based on Bolt Shank Area change

Table 13: Bolt Shank Area changes FSF

RPM	60%	70%	80%	90%	100%
1000	46.08	44.77	43.79	43.03	42.43
2000	11.44	11.11	10.87	10.68	10.53
3000	5.02	4.88	4.77	4.69	4.62
4000	2.78	2.70	2.64	2.59	2.55
5000	1.74	1.69	1.65	1.62	1.59
6000	1.18	1.14	1.11	1.09	1.07
7000	0.83	0.81	0.79	0.77	0.76
8000	0.61	0.59	0.58	0.57	0.56
9000	0.46	0.45	0.43	0.43	0.42

Reducing the shank cross-sectional area increases the FSF by improving stress distribution and lowering inertia forces. A smaller shank reduces bolt stiffness, allowing more even load distribution and decreasing peak stress concentrations. Additionally, reduced mass lowers inertia forces at high RPMs, mitigating fatigue failure. (Ha, 2020) However, while this enhances FSF, it also affects clamping force, requiring a balance between stiffness and fatigue performance. At high speeds, stress concentrations remain a limiting factor despite the benefits of mass reduction.

Engine Torques & Power

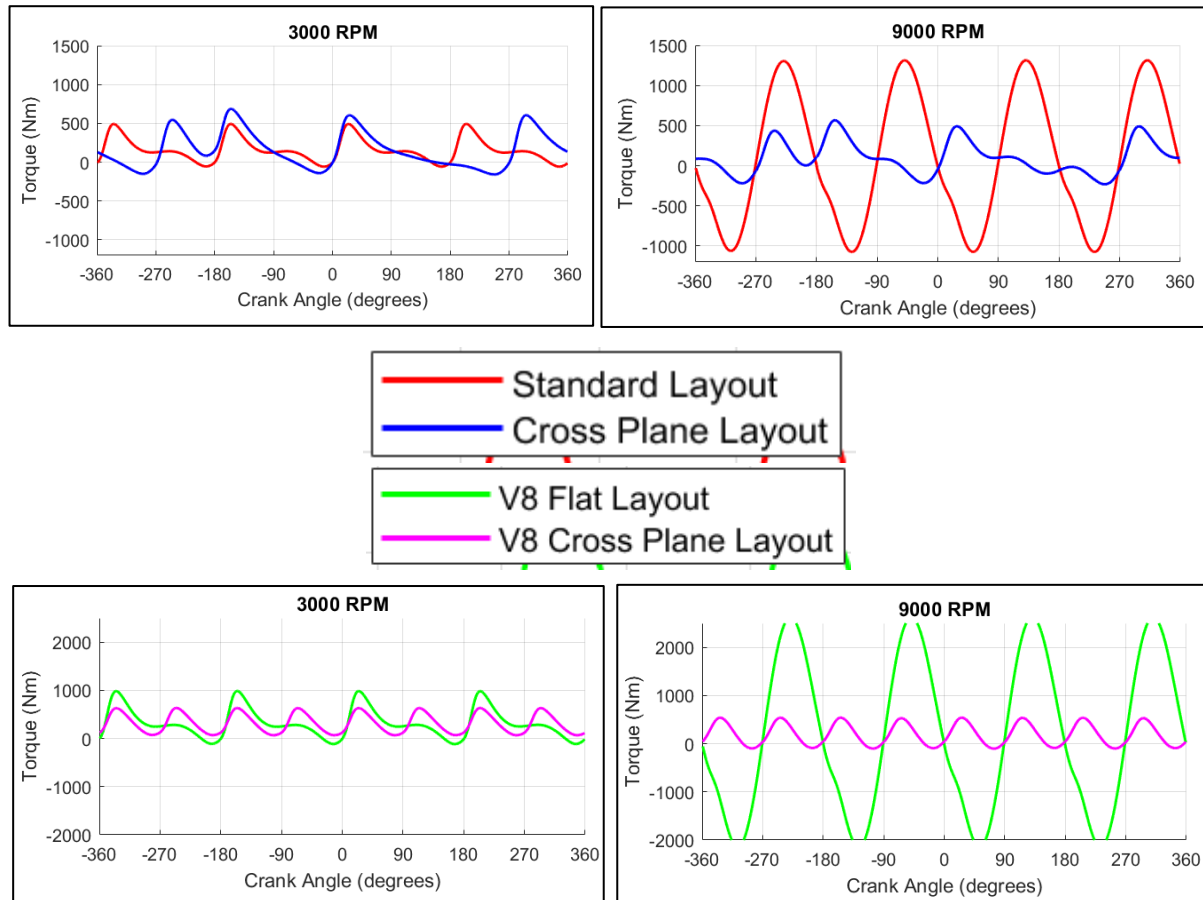


Figure 17: Series of graphs comparing the Combined Torque for each investigated engine layout

The standard Inline-Four shows high torque fluctuations, especially at increased RPM, due to paired reciprocation. The Crossplane Inline-four smooths these peaks, reducing vibrations for better drivability.

For V8s, the Flat-Plane crank maintains high fluctuations but offers fast throttle response, ideal for motorsport. The Crossplane V8 significantly reduces torque variation, enhancing smoothness and traction, making it preferred for high-power applications.

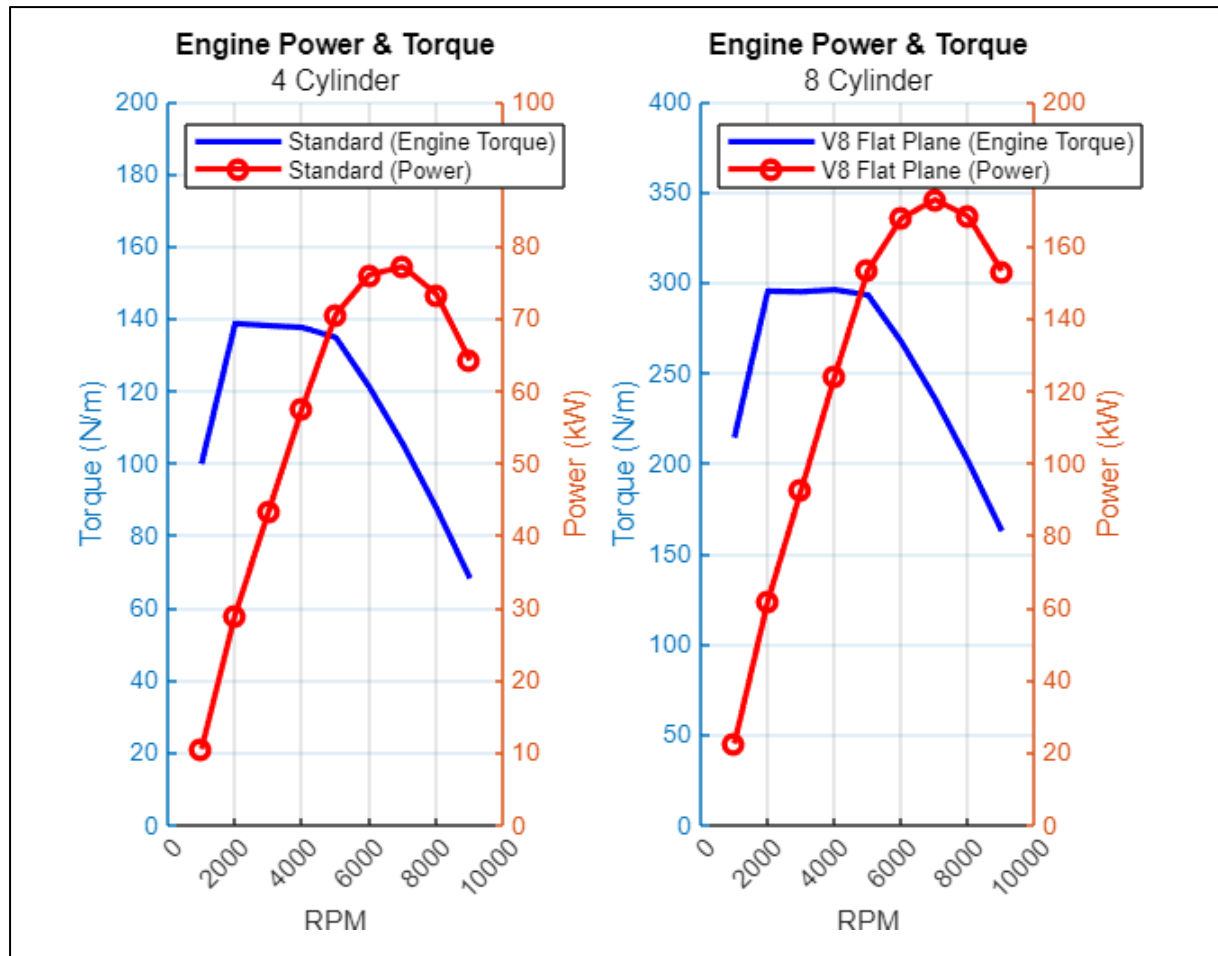


Figure 18: A visual comparison of the power and torque curves for different engine layouts

Both engine layouts produce similar Power and Torque curves as the V8 is essentially two 4-cylinder configurations together. However, figure 18 doesn't show an exact doubling in the results. The V8 sustains a slightly longer torque band before dropping off due to the increased number of power strokes per revolution, reducing the gap in power output.

Conclusion

The findings of this report highlight the critical balance between structural integrity, material selection, and dynamic engine performance in high-speed applications. The fatigue safety factor analysis confirmed that stress concentrations, particularly in the Connecting Rod Bolts, drastically reduce fatigue resistance at elevated RPMs. Without mitigation, these bolts risk failure under prolonged high-speed operation. However, geometric refinements, such as reducing the bolt shank area, showed potential in redistributing stresses and improving FSF while maintaining overall stiffness. This emphasises that weight-saving strategies must be carefully implemented to avoid compromising durability.

Material investigations further reinforced that strength-to-weight ratio plays a pivotal role in component longevity. While titanium alloys provided an optimal balance of weight reduction and fatigue resistance, aluminium alloys, despite their lightweight properties, lacked sufficient endurance limits for these high-stress applications. Surface treatments like carburising demonstrated a significant FSF improvement, indicating that enhancing surface hardness is an effective method to extend component life without the need for expensive material substitution. However, environmental considerations, such as CO₂ and NO_x emissions, suggest that alternative treatments like nitriding or hybrid processes should be explored further, or investment into offsetting maybe required.

From a performance perspective, the torque and power evaluations illustrated the inherent trade-offs between engine configurations. Standard inline-four engines exhibited significant torque fluctuations, whereas cross-plane layouts mitigated these variations, enhancing drivability and reducing stress on the drivetrain. Similarly, the V8 flat-plane configuration showed greater fluctuations but increased power potential, while the cross-plane V8 provided a smoother and more controlled torque delivery, making it preferable for high-power road-going applications. The results reaffirm that while flat-plane cranks are advantageous in high-revving motorsport settings, cross-plane cranks offer superior refinement and long-term durability.

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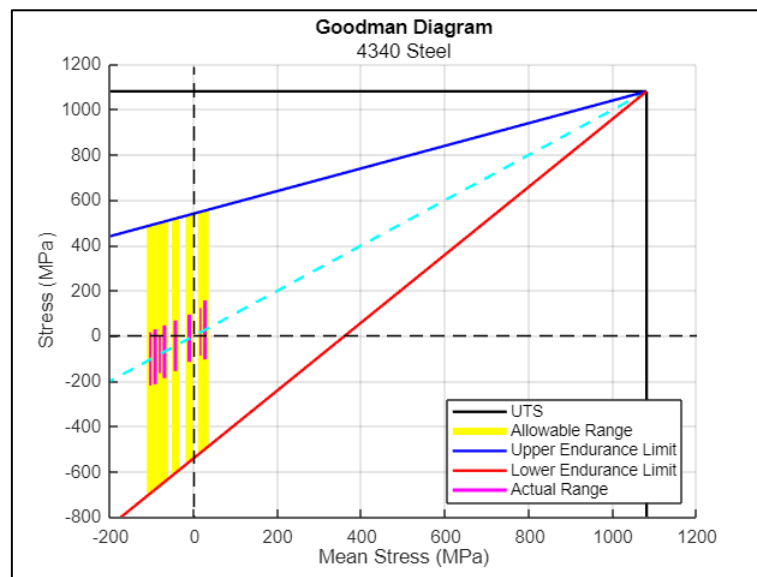
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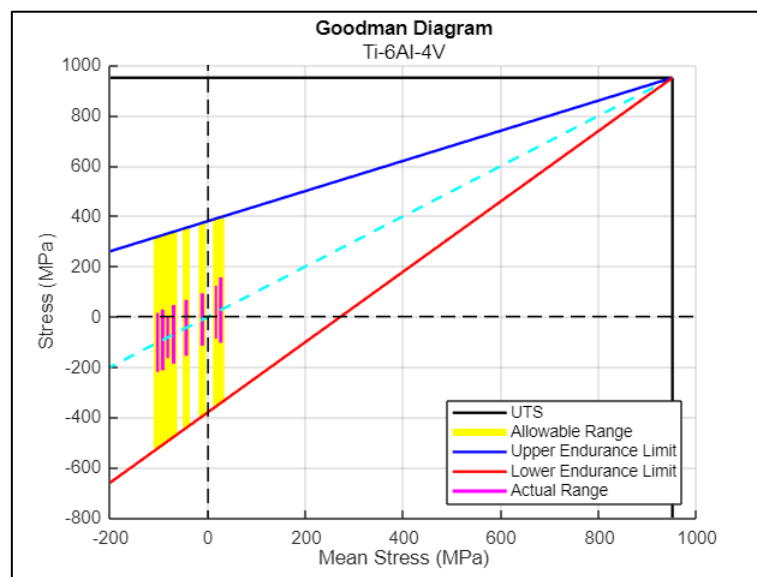
Lu, S., Hua, D., Li, Y., Cui, F. and Li, P. (2019) 'Stiffness Calculation Method and Stiffness Characteristic Analysis of Bolted Connectors', *Mathematical Problems in Engineering*, 2019(1). Available at: <https://doi.org/10.1155/2019/6206092>.

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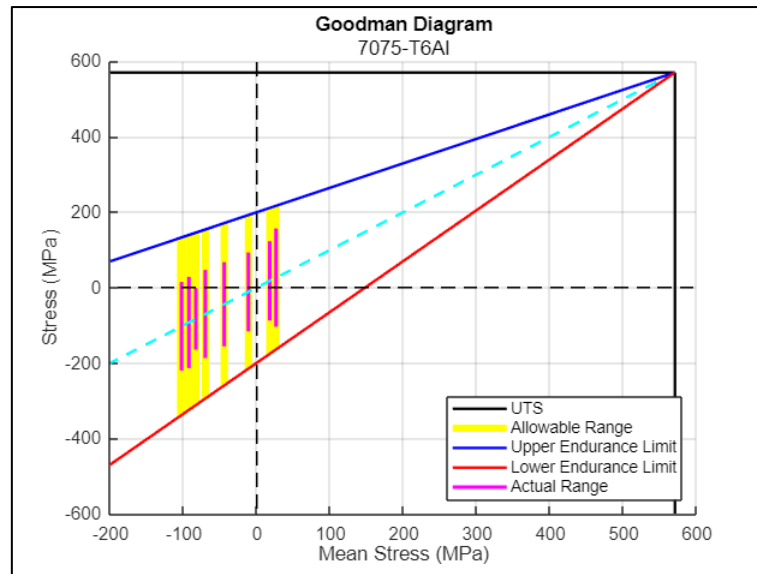
Appendix



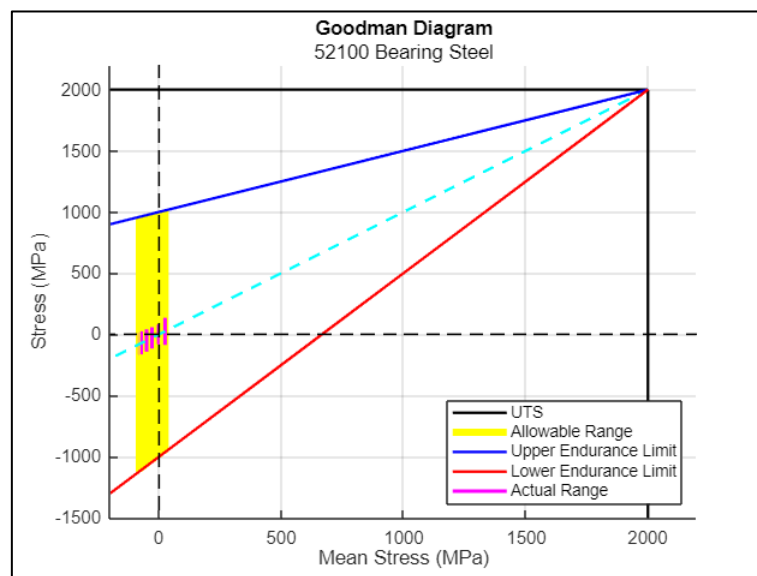
Appendix 1: Goodman Diagram for Connecting Rod with Steel Material



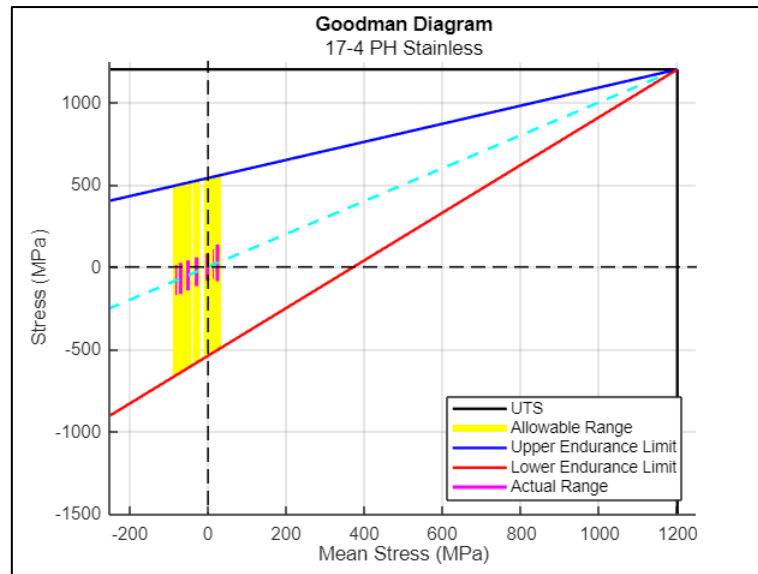
Appendix 2: Goodman Diagram for Connecting Rod with Titanium Material



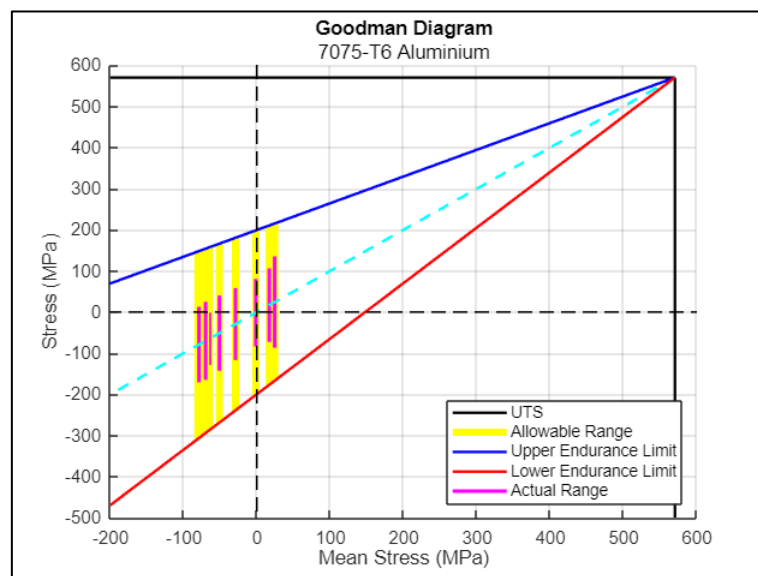
Appendix 3: Goodman Diagram for Connecting Rod with Aluminium Material



Appendix 4: Goodman Diagram for Gudgeon Pin with Bearing Steel Material

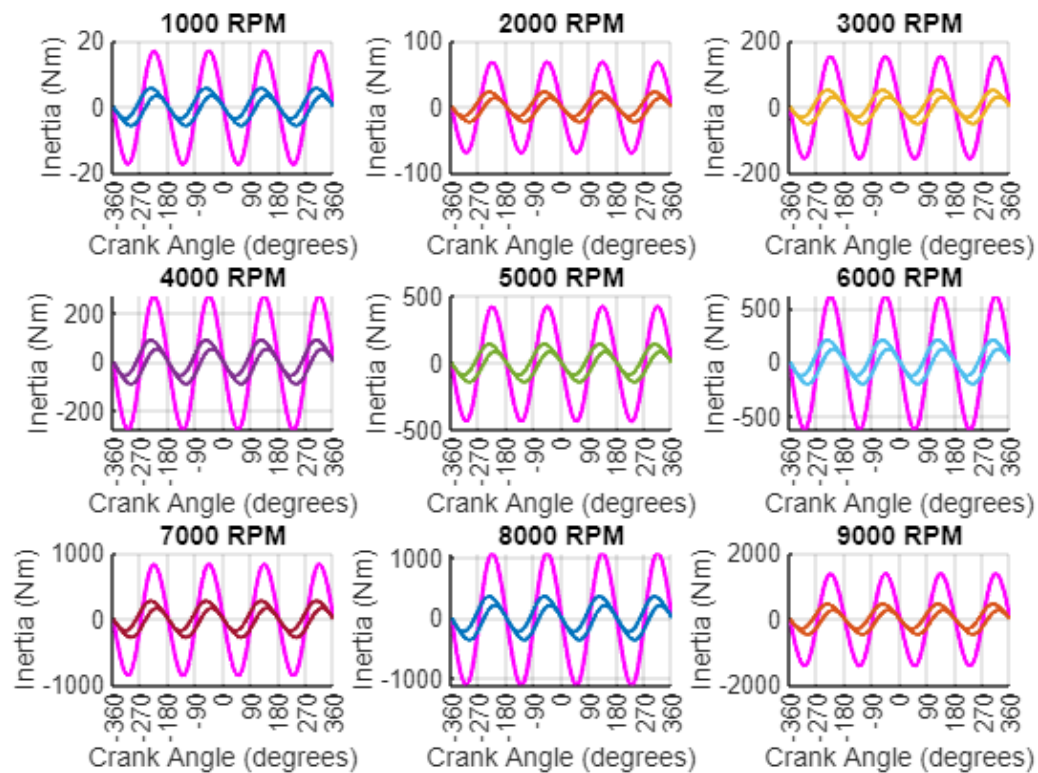


Appendix 5: Goodman Diagram for Gudgeon Pin with Stainless Steel Material



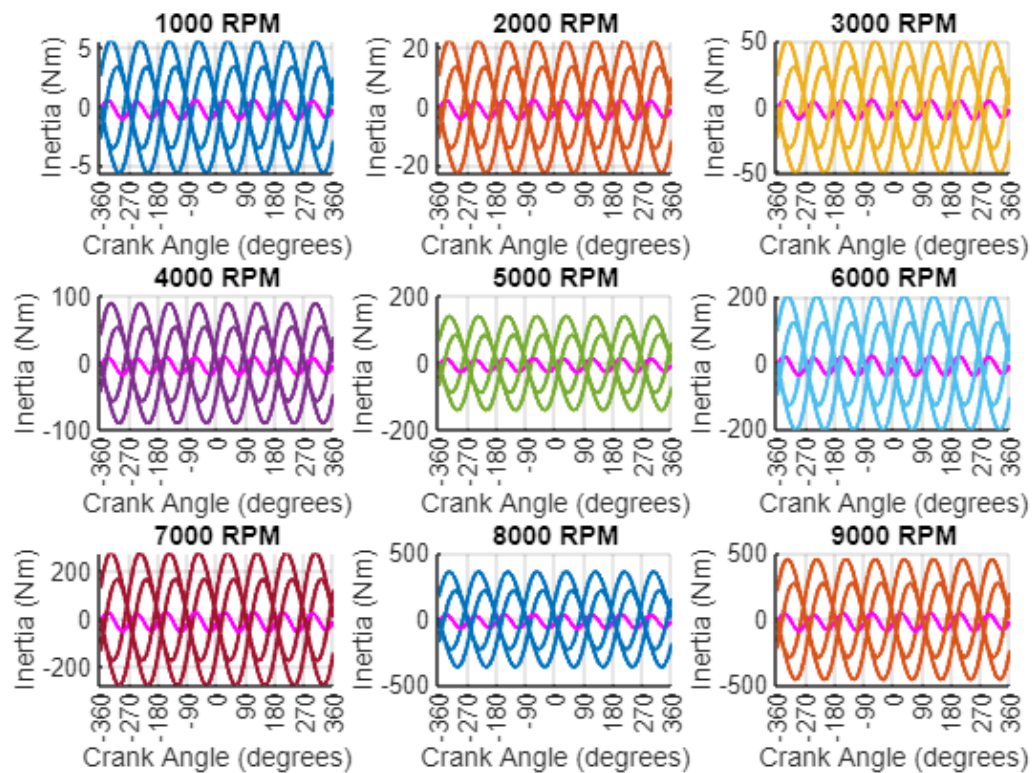
Appendix 6: Goodman Diagram for Gudgeon Pin with Aluminium Material

Standard Layout - Torque Inertia vs Crank Angle



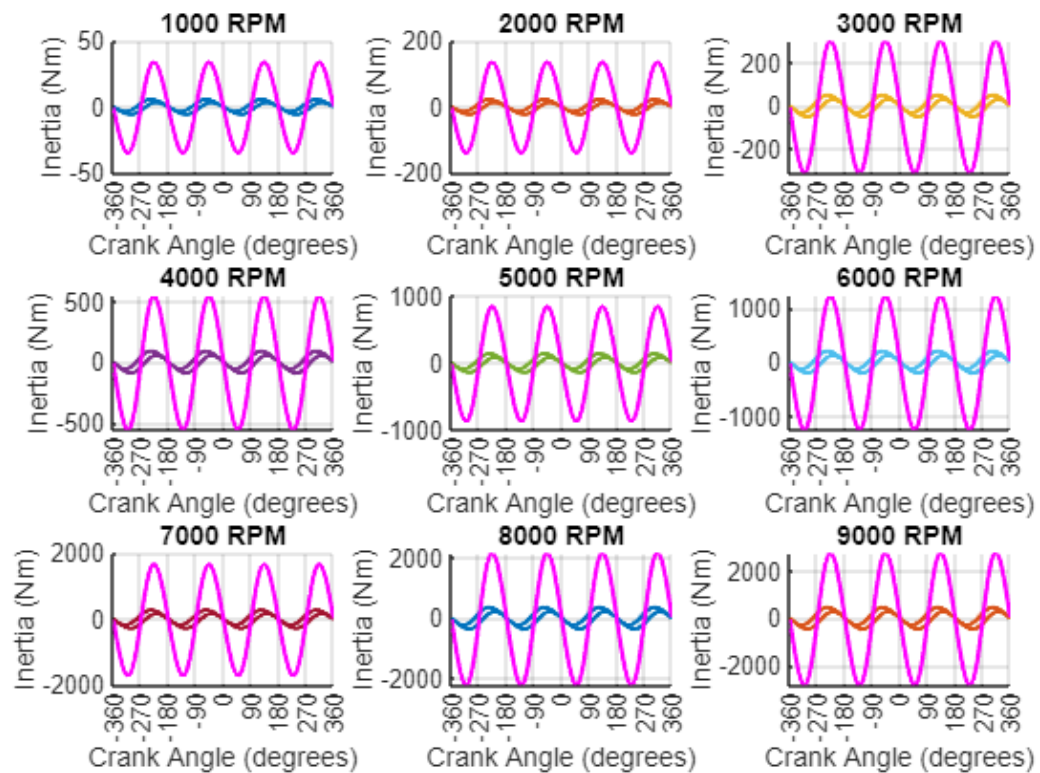
Appendix 7

Cross Plane Layout - Torque Inertia vs Crank Angle



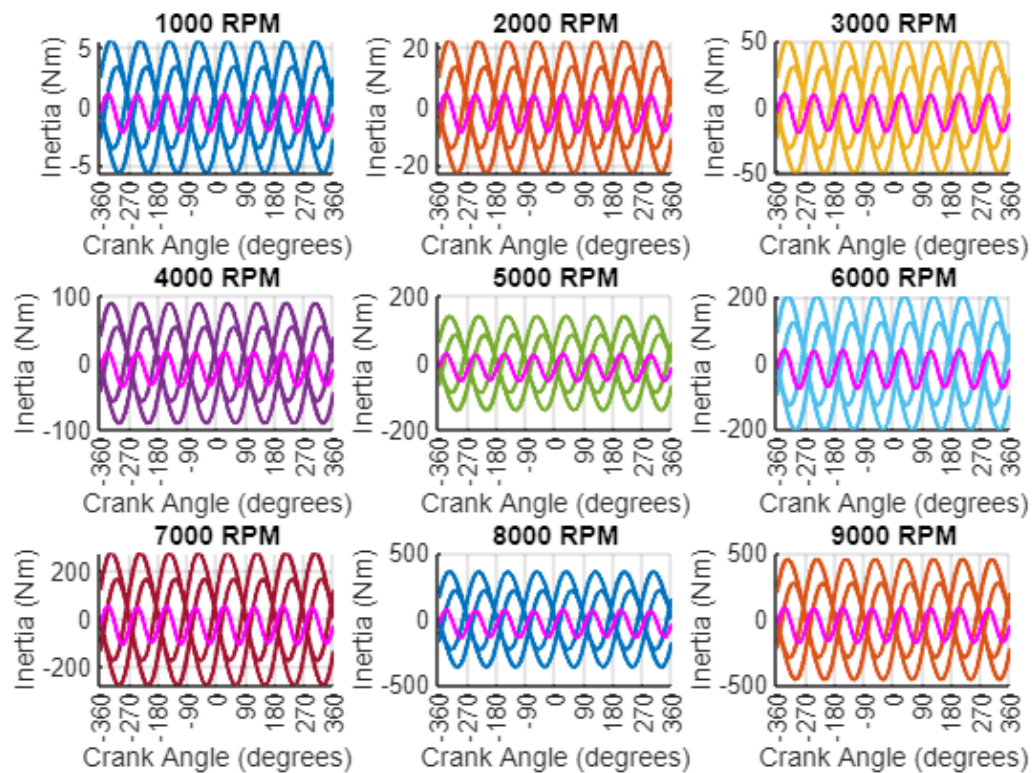
Appendix 8

V8 Flat Layout - Torque Inertia vs Crank Angle

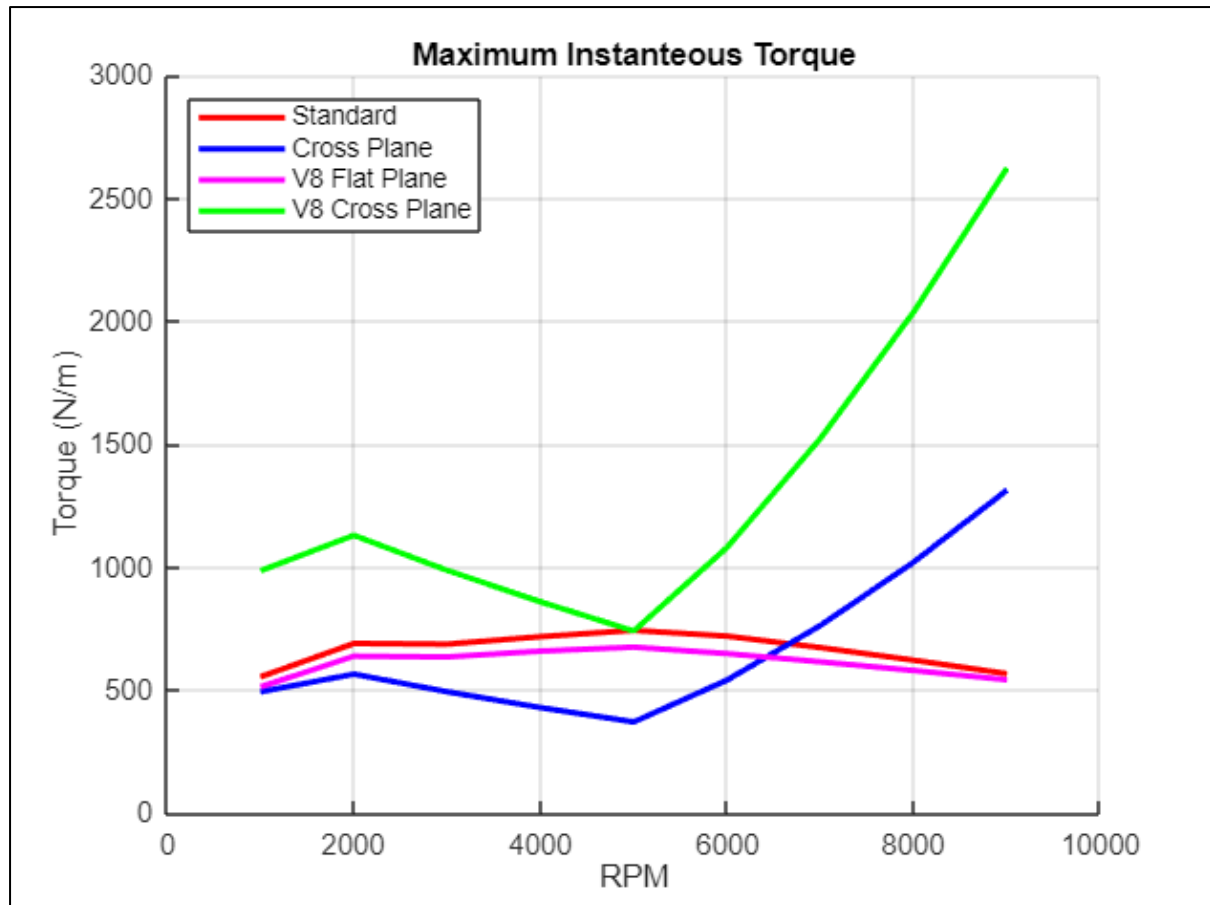


Appendix 9

V8 Cross Plane Layout - Torque Inertia vs Crank Angle



Appendix 10

*Appendix 11*