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Analytical Assessment of Engine Internal Components
Contemporary Powertrain Design

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1 – Introduction

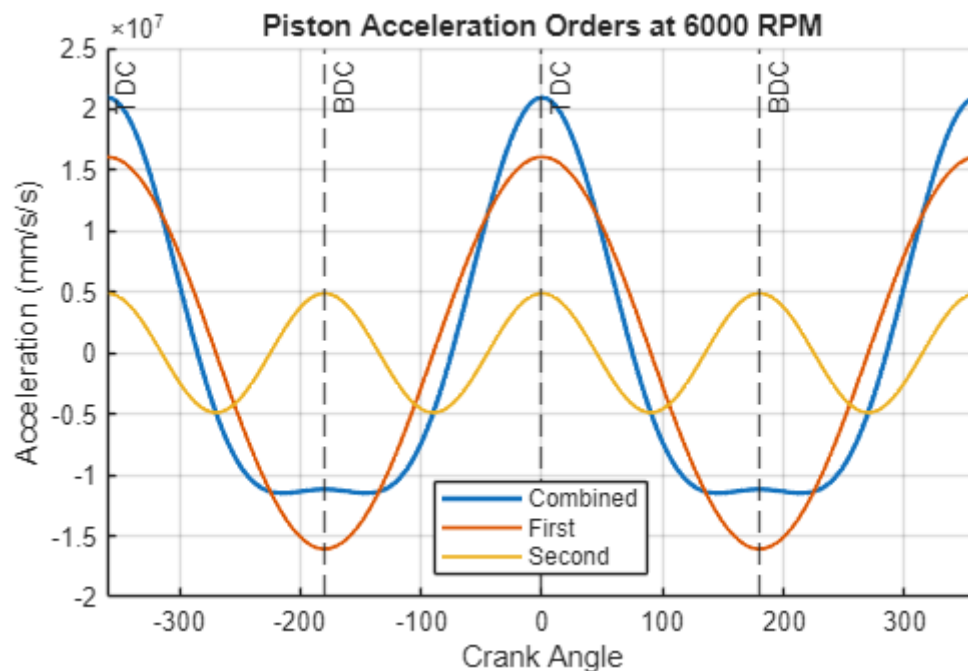
Bottom-end engine components operate under repeated combined gas and inertia loading, making fatigue resistance critical to reliability and safe high-speed operation. This study uses the supplied Ford 1.6 GTDi EcoBoost engine data to calculate component forces, stresses and fatigue safety factors (FSF), and identify the most critical region in the bottom end. The conrod bolt is then reassessed at 9000 Rotations-per-minute (RPM) and redesigned to improve fatigue margin. Instantaneous torque is also compared for standard and cross-plane crankshaft arrangements to examine how crank phasing affects dynamic loading and application suitability.

2 – Methodology

For this study pre-recorded data was taken and a series of calculations was made to create useable information required to investigate the structural properties and performance characteristics of the engine. Equations used can be found in appendix B.

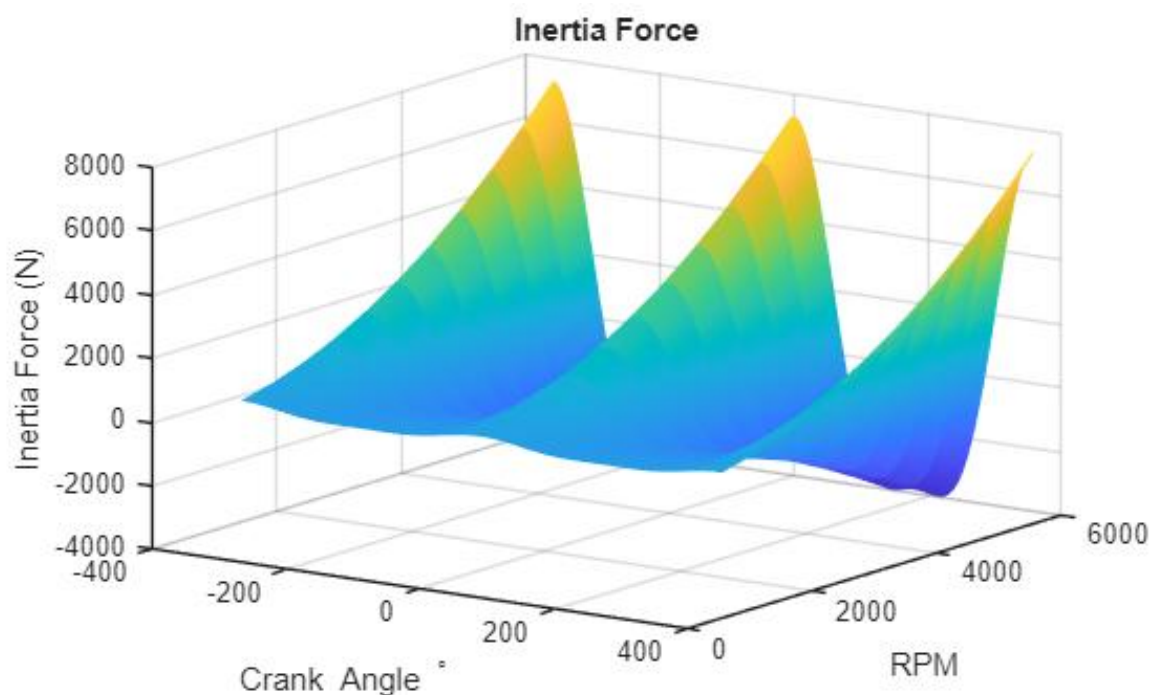
Results were extrapolated using linear extrapolation, extending the existing trend beyond the original data range. This assumed that the relationship between RPM and the calculated loading continued linearly, allowing the stresses and corresponding FSF at 9000 RPM to be estimated.

3 - Pressure Data, Kinematics and Force Derivation



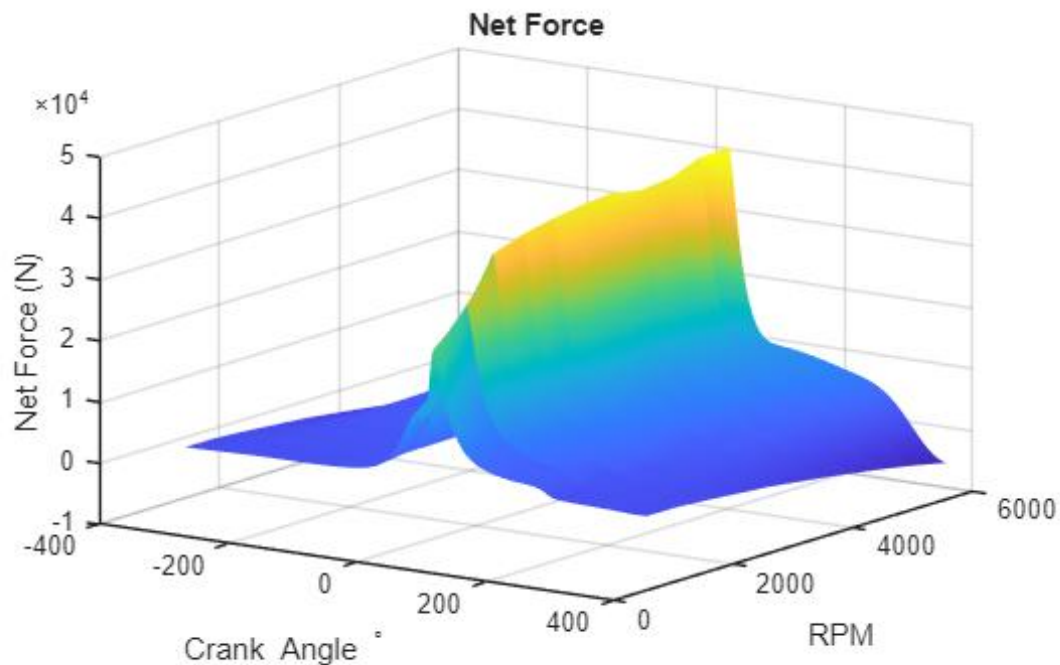
Graph 1: Piston Acceleration Orders at 6000 RPM Against Crank Angle

Combined piston acceleration is made up of 1st and 2nd order terms, with the 2nd order repeating at twice crank speed, which explains the asymmetric shape around TDC and BDC. Acceleration is the basis of the inertia-force calculation, so it links the kinematics directly to the structural loading on the engine.



Graph 2: Inertia Force 3D Graph Against Crank Angle and RPM

The acceleration result is converted into force, allowing identification of where tensile inertia loading becomes significant across crank angle and RPM. This is important for fatigue-sensitive parts in the engine.



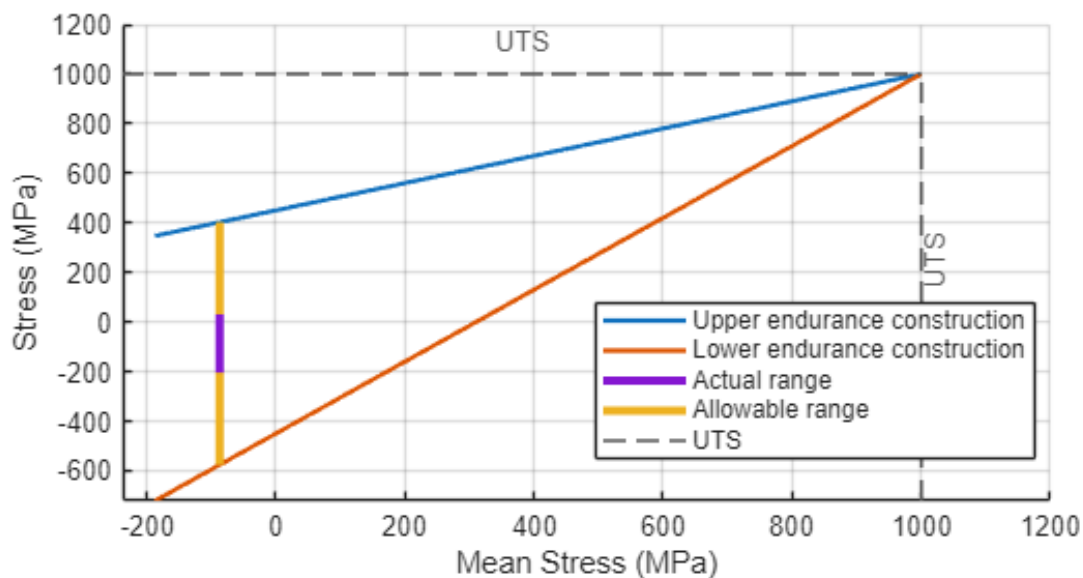
Graph 3: Net Force 3D Graph Against Crank Angle and RPM

Graph 3 combines the pressure driven gas force with the inertia effect to give the actual load transmitted through the bottom-end components. Net force is used as the basis for conrod stress calculations, FSF and Goodman range diagrams.

4 - Fatigue Safety Factor Analysis

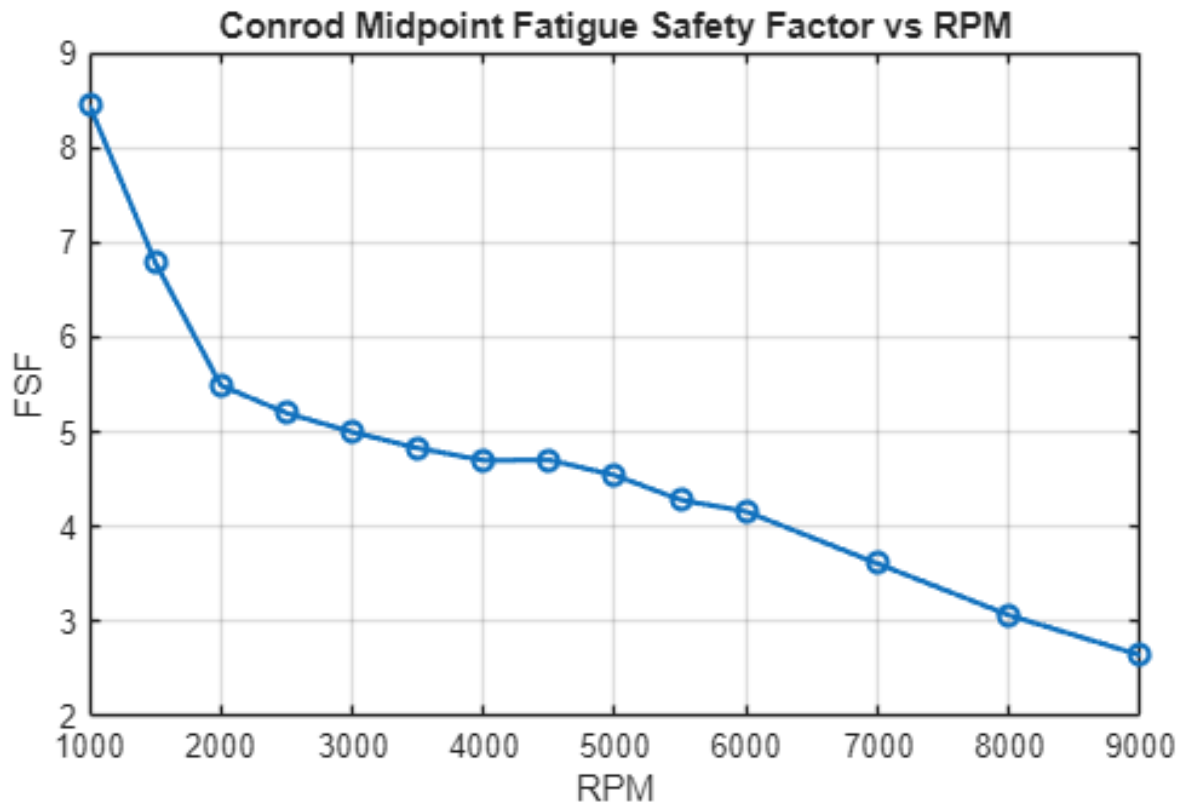
4.1 Conrod Midpoint

Goodman Diagram - Conrod Midpoint at 6000 RPM
Mean = -87.34 MPa | Actual Range = 235.26 MPa | Allowable Range = 978.60 MPa
FSF = 4.1597



Graph 4 - Conrod Midpoint Goodman Range Diagram

The conrod beam is primarily loaded in compression during combustion, making the mean stress negative. Stress is based on the net force and the cross-sectional area (CSA) of 196.39 mm². This section of the conrod is safely within the fatigue limit, with an FSF of 4.16.



Graph 5: Conrod Midpoint FSF against RPM

FSF decreases as RPM increases because inertia loading rises significantly, increasing the cyclic loading carried by the conrod. The midpoint is unlikely to be a fatigue failure location, published conrod studies show critical stresses are typically concentrated near changes in cross-section such as bolt-hole regions (A. Strozzi et al, 2016), while low-stress regions may allow for material reduction. Given the calculated FSF of 4.16 at the midpoint, it may be relatively overengineered and is not a limiting feature in the design.

4.1.1 - Thinnest-Section Comparison

To show the effect of area on FSF, the conrod beam was split at the smallest section, reducing CSA from 196.39mm² to 139.6mm².

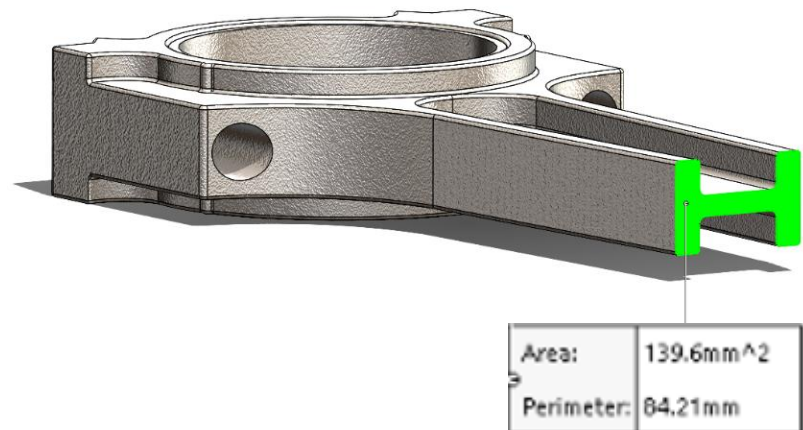
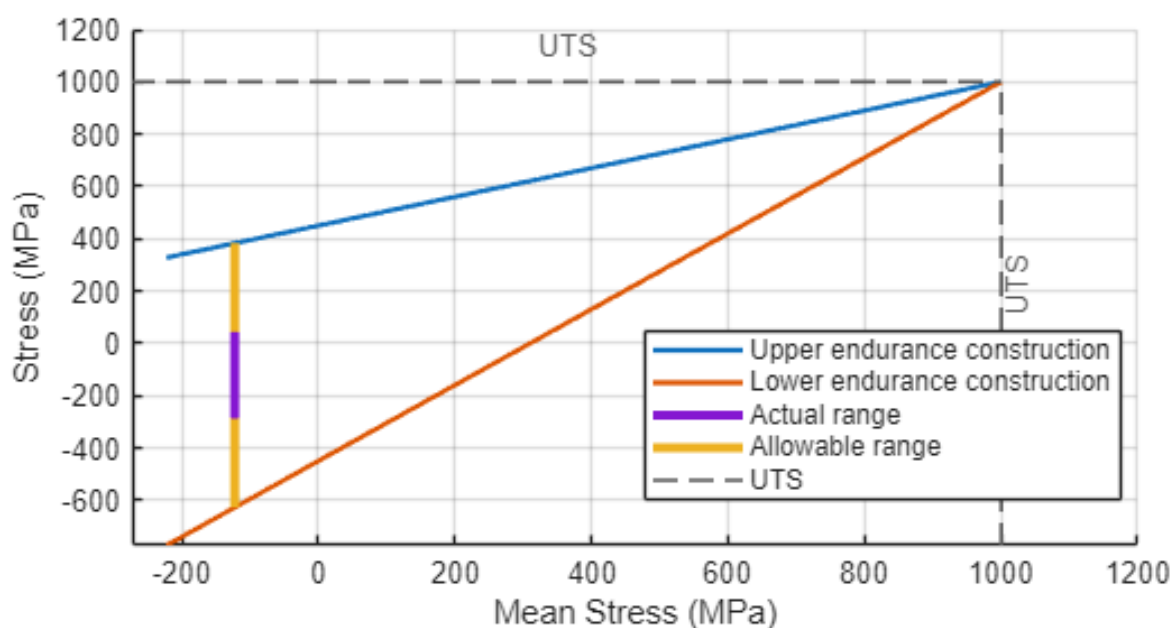


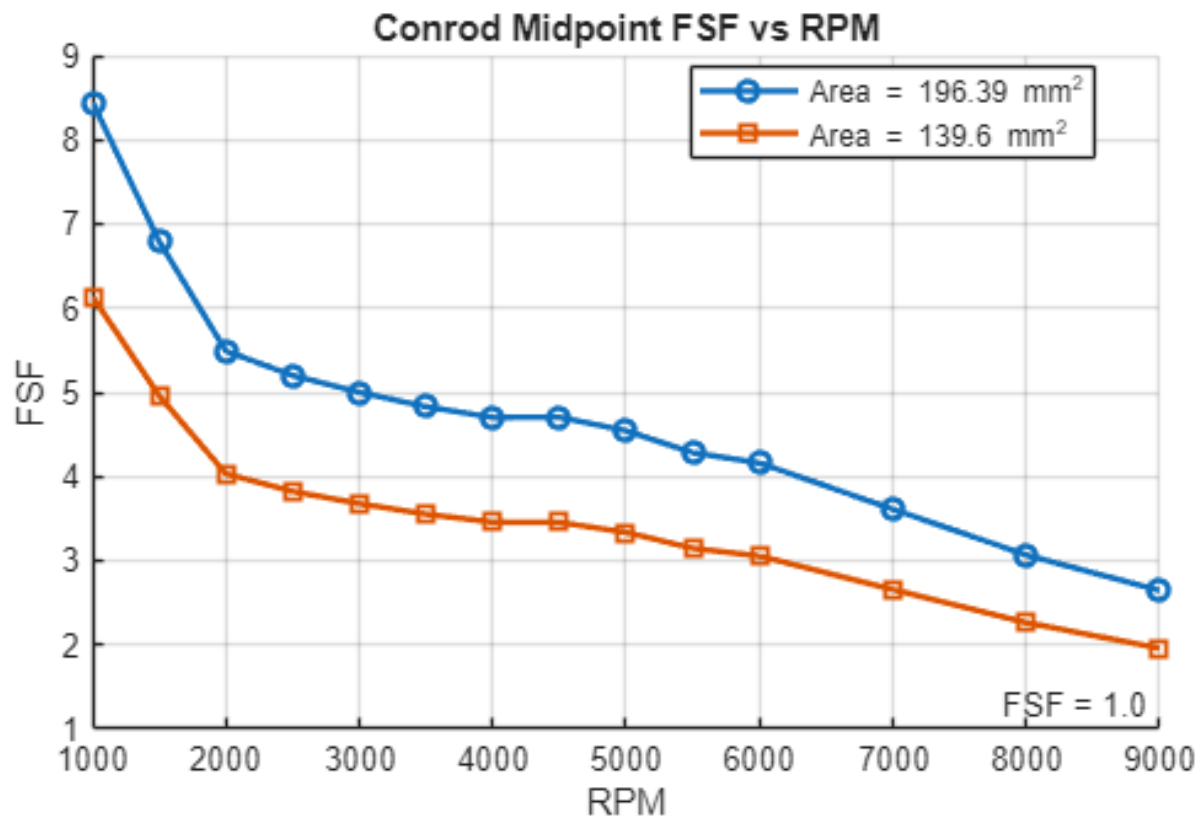
Figure 1: New Conrod Midpoint Intersection

Goodman Diagram - Conrod Midpoint (Area = 139.6 mm²) at 6000 RPM
 Mean = -122.87 MPa | Actual Range = 330.96 MPa | Allowable Range = 1010.58 MPa
 FSF = 3.0535



Graph 6: New Conrod Midpoint Goodman Range Diagram

This reduced the FSF from 4.16 to 3.05, this means the thinner section is more critical than the original estimate. Using the larger midpoint area overestimates the fatigue margin and the minimum section provides a more conservative assessment of beam strength.

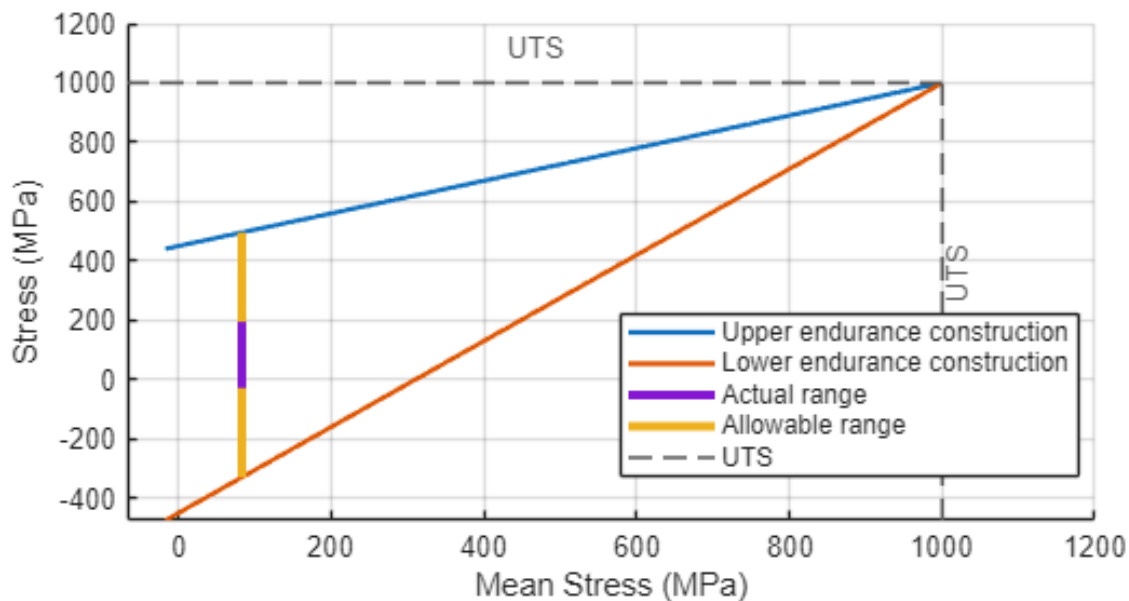


Graph 7: Old Vs New Conrod Midpoint FSF against RPM

This comparison shows that the reduction in FSF is maintained across the full speed range, so the thinner section does not just reduce safety but gives a consistently lower fatigue margin at all RPM.

4.2 - Gudgeon Pin at Point A

Goodman Diagram - Gudgeon Pin A at 6000 RPM
Mean = 82.86 MPa | Actual Range = 223.18 MPa | Allowable Range = 825.43 MPa
FSF = 3.6985



Graph 8: Gudgeon Pin A Goodman Range Diagram

Unlike the midpoint, the gudgeon pin sees bending stress giving a positive mean stress, meaning it operates in a lower fatigue region. Point A has the highest bending stress because it's on the outer material away from the neutral axis. Tensile mean stress reduces the allowable alternating stress, so even though the pin is still safe at 6000 RPM with an FSF of 3.6985, it has a smaller fatigue margin than a component loaded mainly in compression.

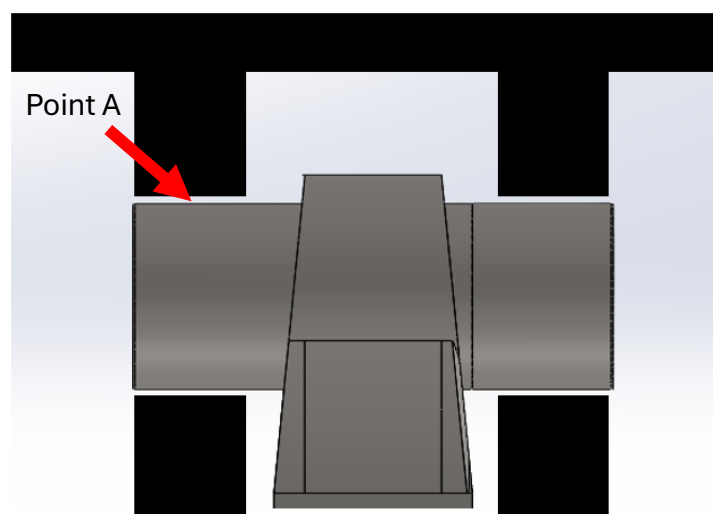
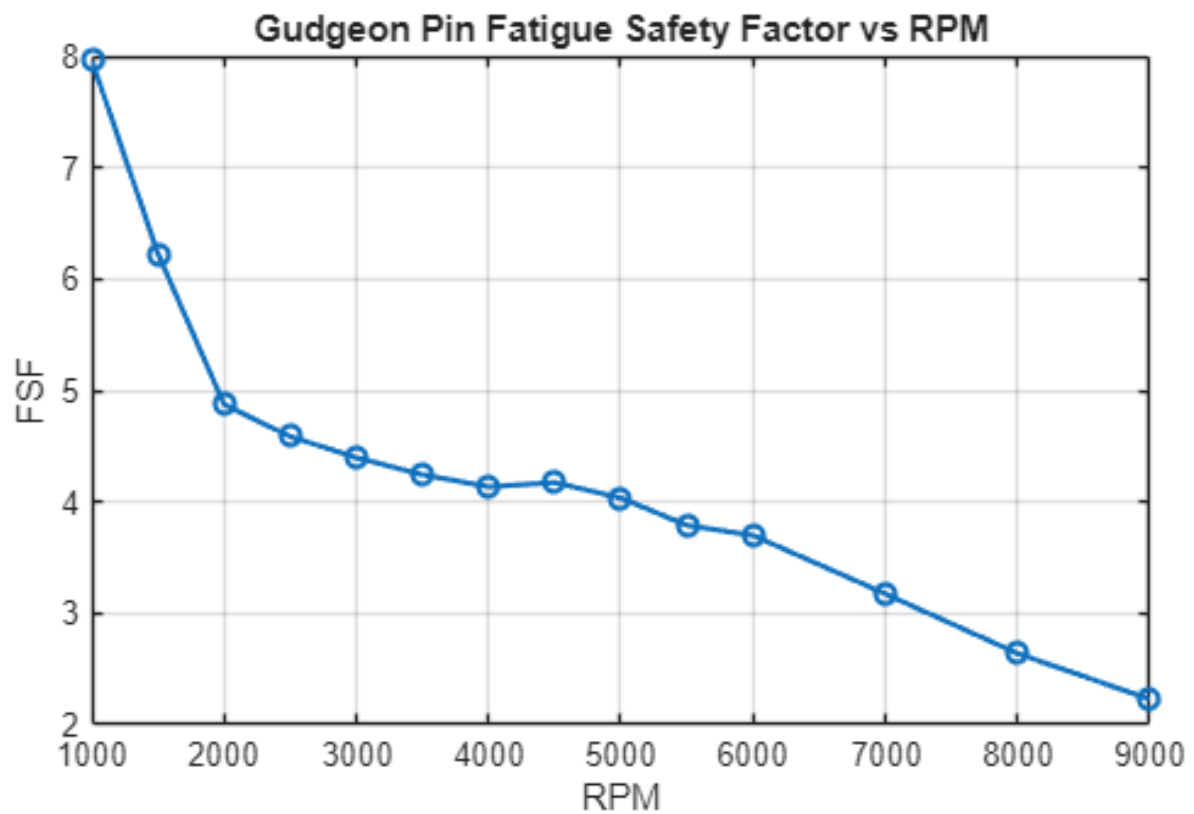


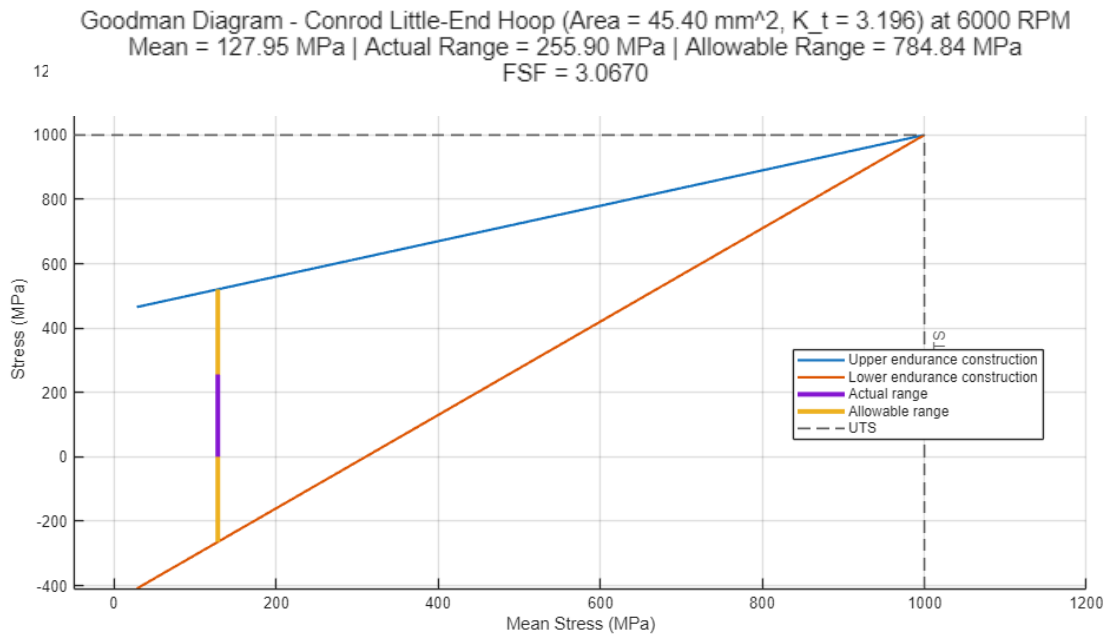
Figure 2: Gudgeon Pin, Point A



Graph 9: Gudgeon Pin FSF against RPM

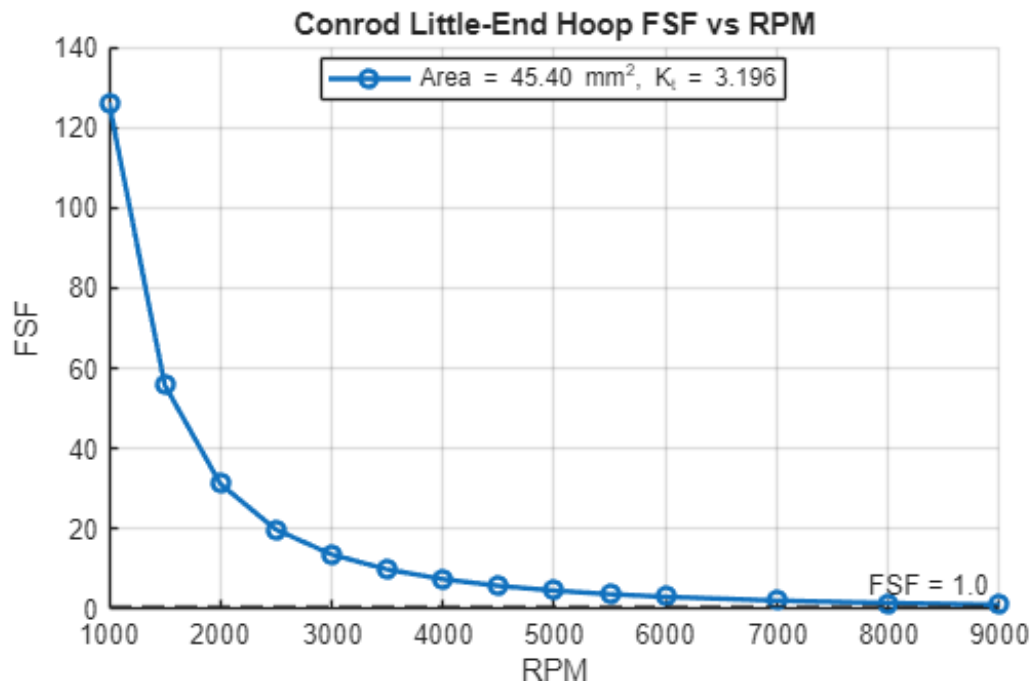
Cyclic bending load on the pin increases with RPM. But the pin is not predicted to fail at 9000 as FSF stays above 1. However, this result depends on simplified assumptions.

4.3 - Conrod Hoop



Graph 10: Conrod Little-End Hoop Goodman Range Diagram

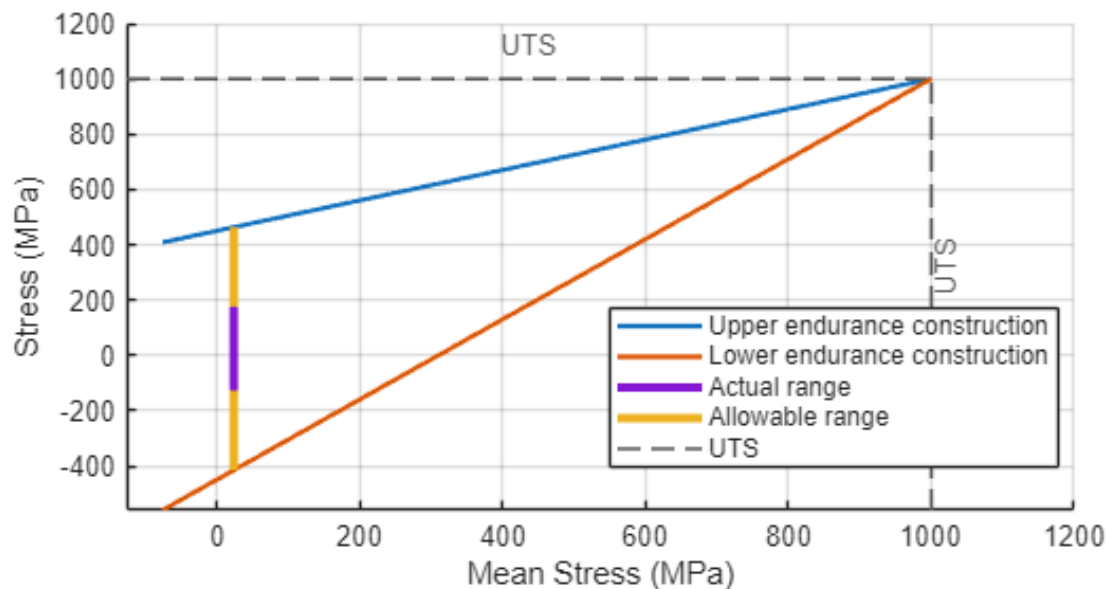
Stress Concentration (K_t) was required as a local filleted edge is involved, so nominal stress alone would underestimate the peak fatigue stress. There was no Roark's formula available which directly related to the hoop fillet, therefore the formula in appendix BC was used. It is not identical to the hoop, but it has a similar fillet and gave a realistic K_t value for the conrod hoop.



Graph 11: Conrod Little-End Hoop FSF against RPM

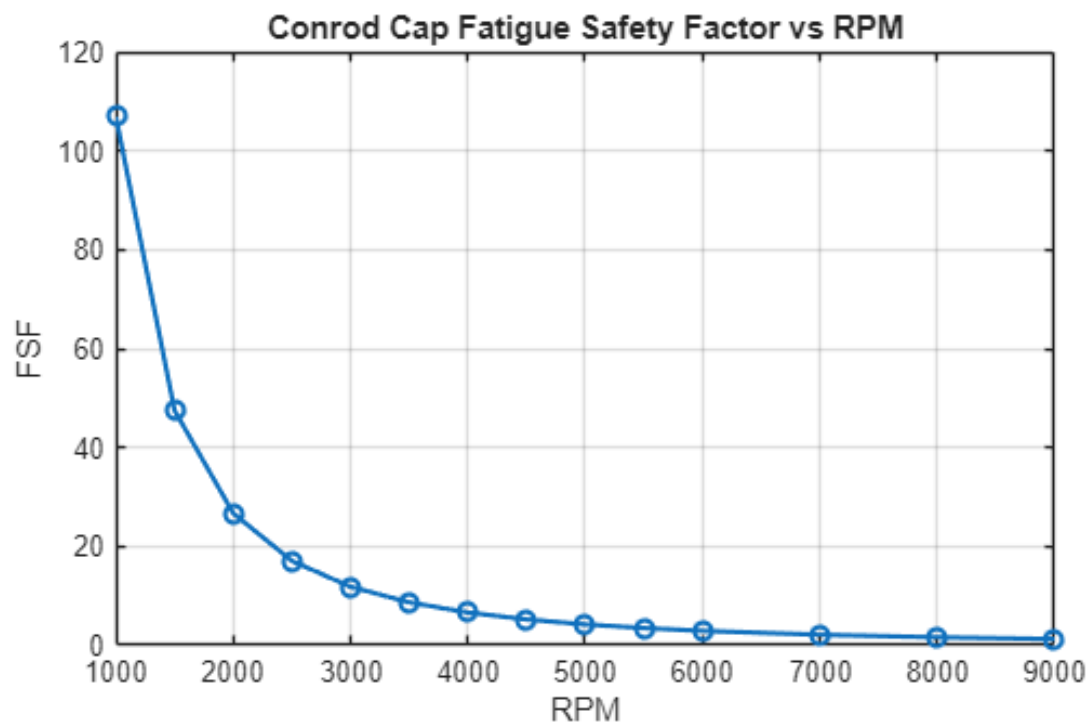
4.4 – Conrod Cap

Goodman Diagram - Conrod Cap at 6000 RPM
 Mean = 24.19 MPa | Actual Range = 302.99 MPa | Allowable Range = 878.23 MPa
 FSF = 2.8985



Graph 12: Conrod Cap Goodman Range Diagram

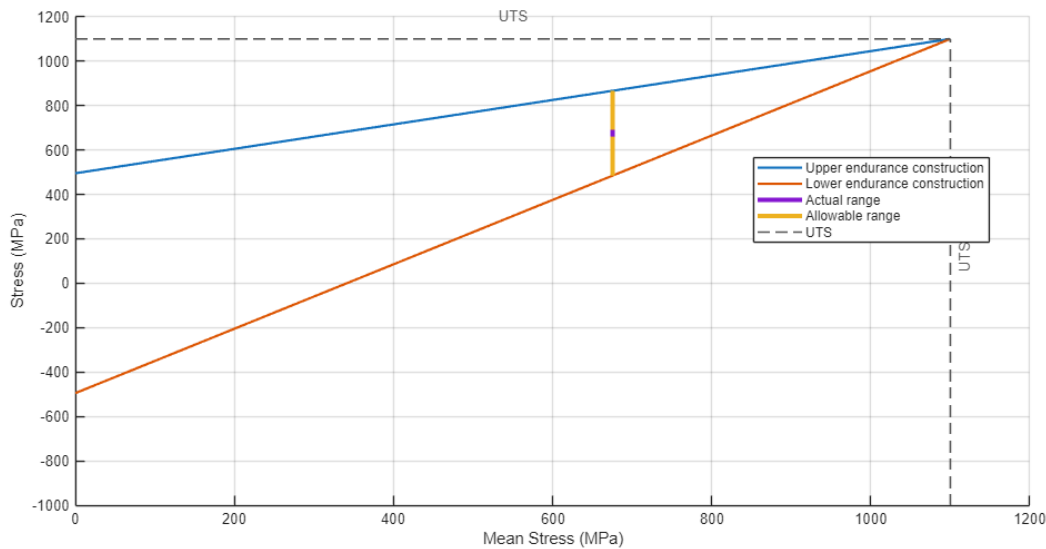
The conrod cap has a lower FSF because its effective area is reduced by the bolt holes and split interface, creating local stress concentrations (A. Strozzi et al, 2016). The cap mainly sees inertia-driven loading, so it does not benefit from compressive stress during combustion.



Graph 13: Conrod Cap FSF against RPM

4.5 - Conrod Bolt Baseline Analysis

Goodman Diagram - Conrod Bolt (M8x1.0, A = 42.43 mm², K_t = 4.302) at 6000 RPM
 Mean = 675.42 MPa | Actual Range = 30.839 MPa | Allowable Range = 382.12 MPa
 FSF = 2.8800



Graph 14: Conrod Bolt Goodman Range Diagram

Publicly available sources state the Ford 1.6 EcoBoost conrod bolt to be M8x1.0 (ARP, 2026a), FSF was calculated to be 2.88 at 6000 RPM. The bolt is more fatigue-sensitive than the conrod having a smaller area and is affected by stress concentration. Heywood (1962) also states that fatigue failure normally occurs in the bolt threads about one turn in from the loaded face, where the thread load transfer is greatest. Although this source is old, the underlying fatigue and thread-loading principles remain applicable to modern bolt design.

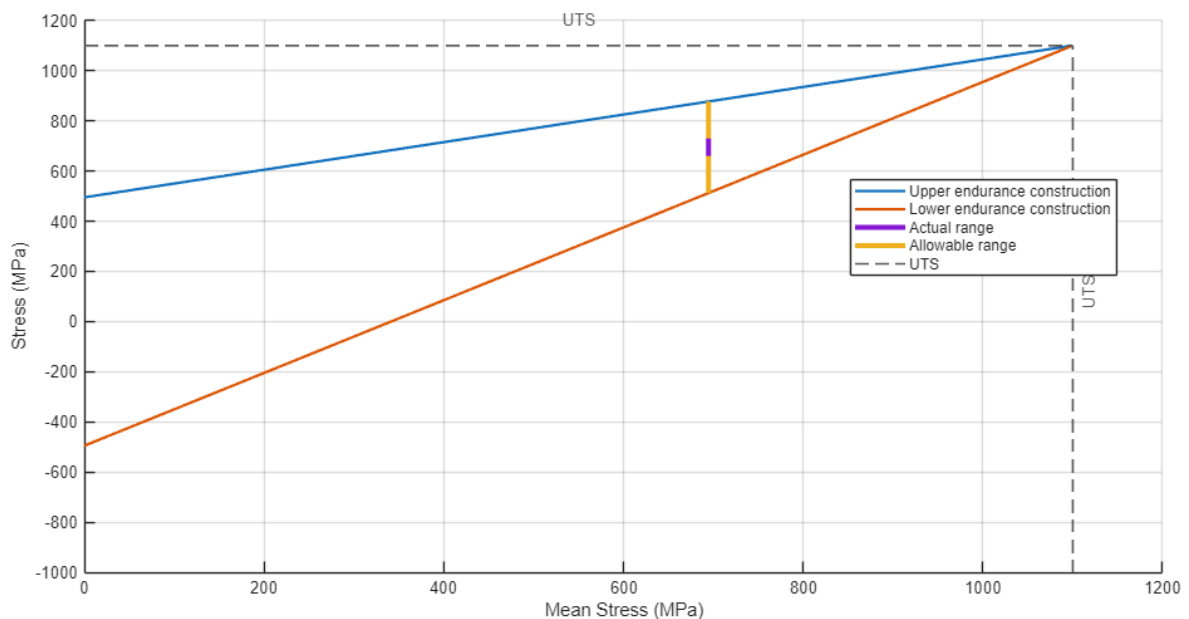
4.6 - Goodman Diagrams Summary

The Goodman diagrams show that within the simplified model, the calculated stresses remain within the material failure limits for the components considered. At 6000 RPM this suggests a healthy fatigue margin, where durability and reliability were prioritised over minimum material use. However, as RPM is extended towards 9000 the margin reduces, so the design cannot be said to be as reliable at higher speeds. Euro 7 requires durability over 200,000 km and 10 years of use (European Commission, 2022). The FSF values at the intended operating condition are consistent with long service life, but they do not alone prove that such durability will be achieved in real life.

5 - Conrod Bolt Extended RPM Analysis

5.1- Interpolation / Extrapolation to 9000 RPM

Goodman Diagram - Conrod Bolt (M8x1.0, A = 42.43 mm², K_t = 4.302) at 9000 RPM
Mean = 694.69 MPa | Actual Range = 69.387 MPa | Allowable Range = 364.78 MPa
FSF = 1.2219

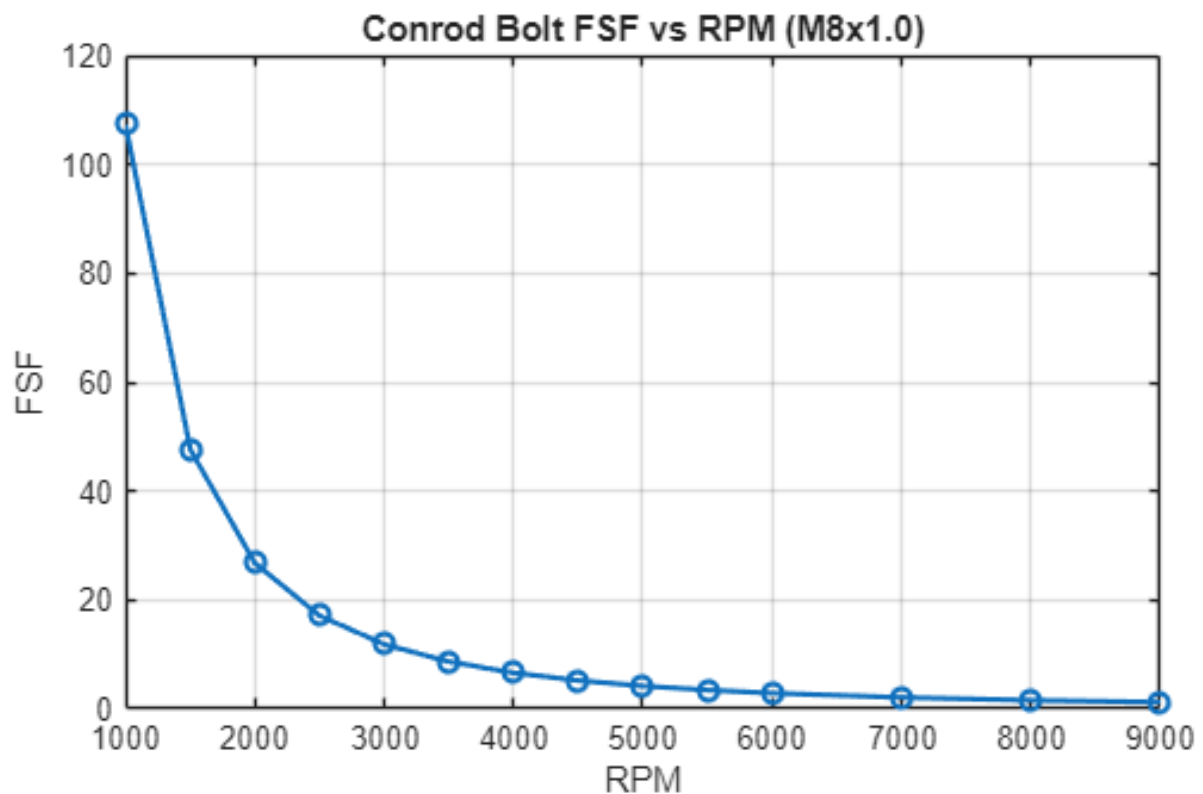


Graph 15: Extrapolated Conrod Bolt Goodman Range Diagram

When extrapolated to 9000 RPM, the bolt is not predicted to immediately fail. However, 1.22 is very close to the limit, if the loading on the bolt increases by only 22% it would fail. So, the bolt may tolerate short over engine speed events, but repeated exposure to this speed could lead to failure.

On top of this, there are many real-life factors which are not considered in the calculations such as low tightening force, design defects especially at the thread root radius, material or fabrication defects (Griza et al, 2009), which can further reduce FSF. So, you can't rely on a FSF of 1.22 being completely accurate.

5.2 - Bolt FSF Versus RPM



Graph 16: Conrod Bolt FSF against RPM

5.3 - Operation and Life Expectancy

The baseline conrod bolt is expected to have a long operating life at 6000 RPM, where the FSF remains high. As speed is extended towards 9000 RPM, that margin falls sharply, so repeated operation in this range would reduce life expectancy and would require redesigning to operate at this RPM regularly.

6 - Conrod Bolt Redesign

6.1 - Thread Pitch

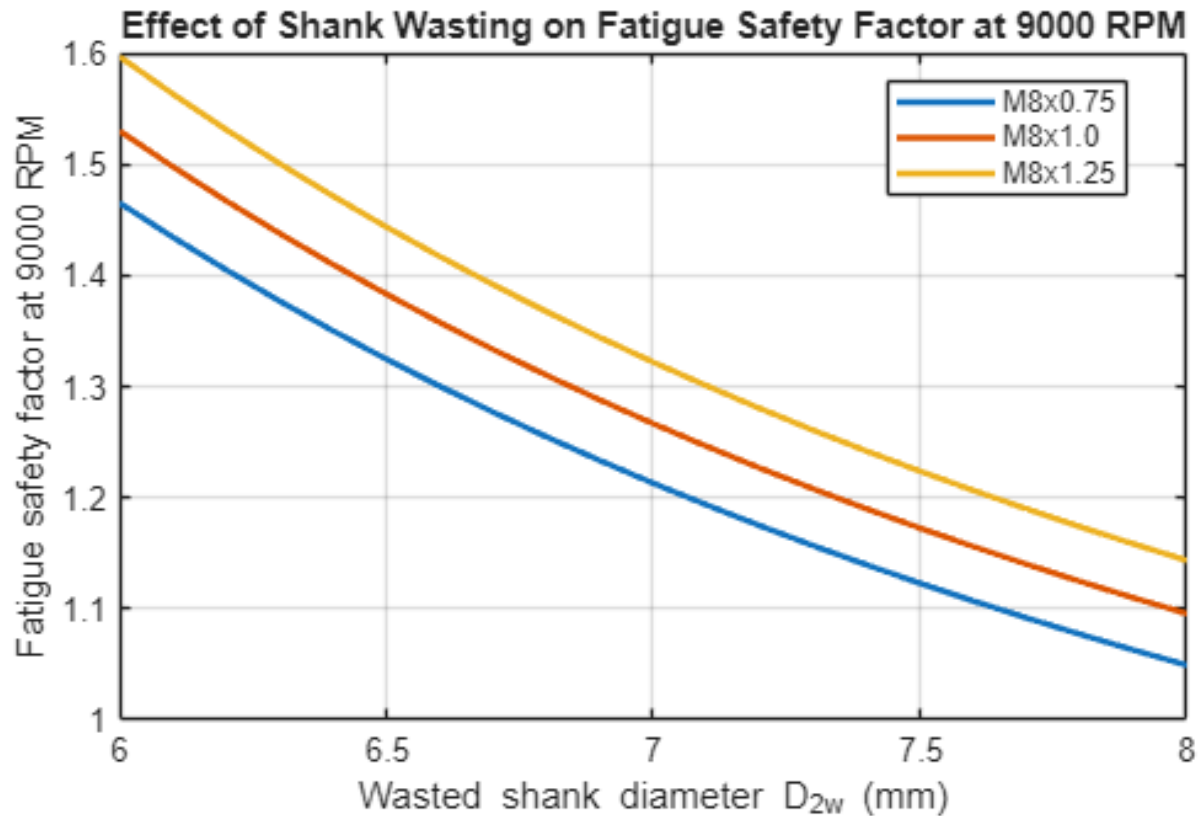
Table 1: Conrod Bolt Thread Comparison

Conrod Bolt Thread Comparison (9000 RPM)			
Nominal Size ISO MF	FSF	Thread Root Stress Concentration Factor	Tightening Torque (Nm)
8.0x0.75	1.171	4.49	36.58
8.0x1.0	1.222	4.30	36.17
8.0x1.25	1.277	4.12	35.76

The highest bolt stresses occur at the thread root, and a coarser thread has a larger pitch and root radius, so the thread root is less sharp, reducing the stress concentration. A larger, smoother thread-root radius spreads the load more gradually, so the peak local stress is lower (Barrett, 1990, Chalupník, 1968).

The tightening torque also changes slightly because thread geometry affects how applied torque is translated into preload through the friction at the thread surfaces. Overall, the main benefit of the coarser thread is better stress distribution.

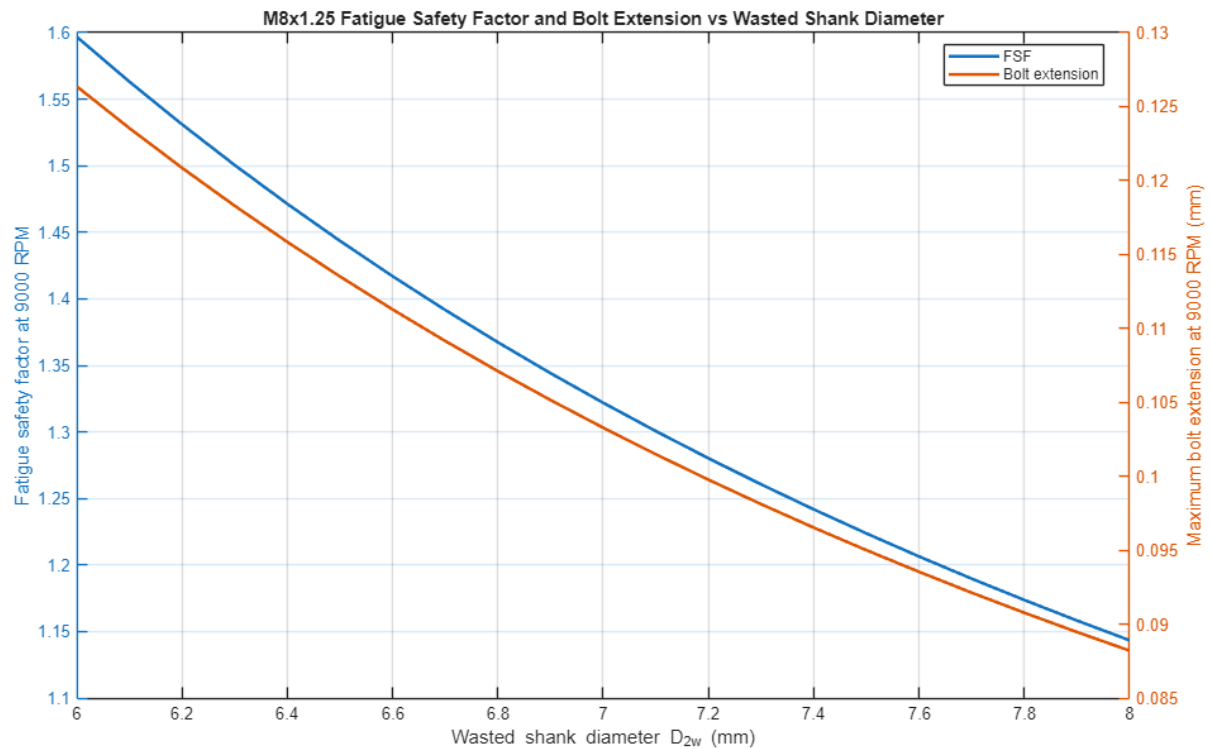
6.2 - Shank Size (Shank Wasting)



Graph 17: Wasted Shank against FSF

The original Ford bolt has no shank wasting. ARP states that a reduced-diameter shank is the standard preferred rod-bolt configuration (ARP, 2026b).

A smaller shank makes the bolt more flexible, so more of the cyclic strain is absorbed along the shank instead of being concentrated at the thread root. Graph 17 shows that controlled shank wasting can improve fatigue performance by giving better stress distribution through the bolt.



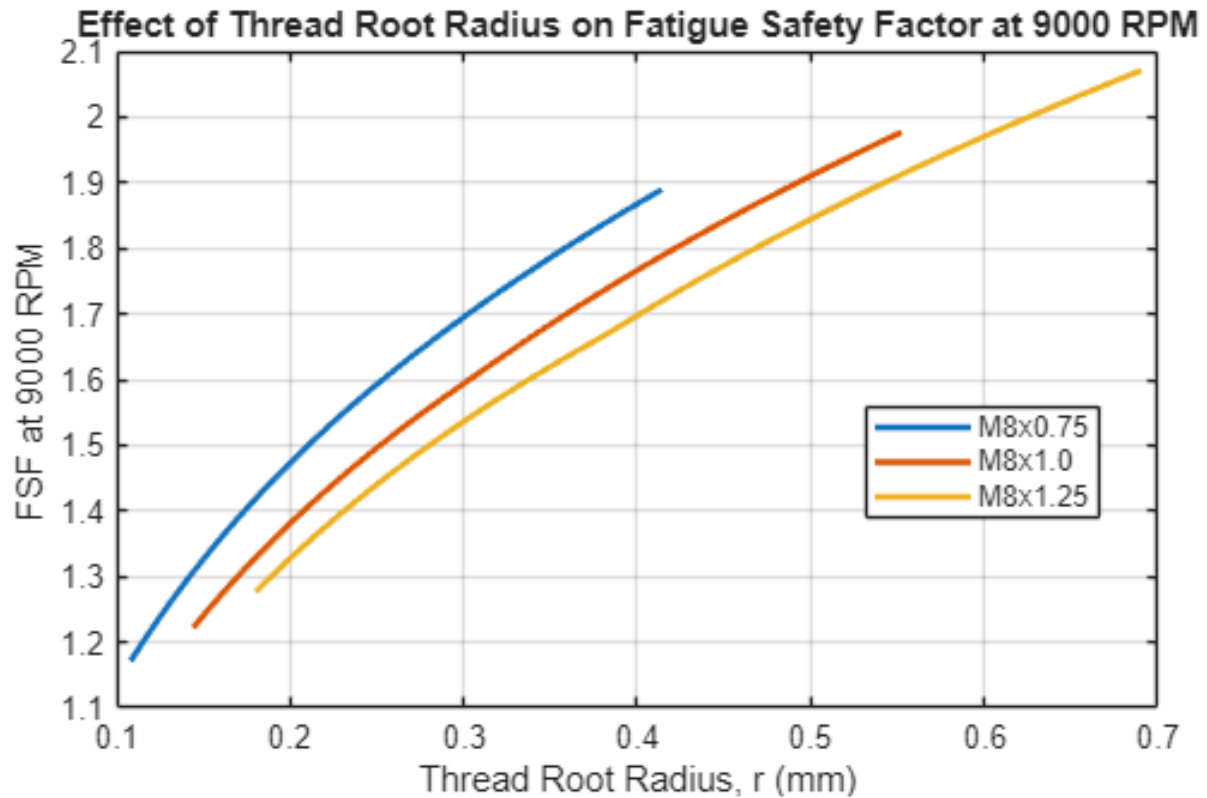
Graph 18: FSF / Bolt Extension against Wasted Shank

Reducing shank diameter increases FSF but also increases bolt extension. A lower bolt stiffness for the same axial force gives a larger extension, shown in the equation below. If the bolt extends too much, the cap can deflect more under load, increasing the risk of joint separation/catastrophic failure. As a result, shank wasting must be limited to a balanced design rather than maximised.

$$K = \frac{F}{x}$$

Equation 1: Bolt Stiffness
in Relation to Axial Force
and Bolt Extension

6.3 - Thread Root Radius



Graph 19: Effect of Root Radius on FSF

Increasing the thread root radius raises FSF because a smoother root reduces stress concentration at the threaded region, which Heywood (1962) identifies as the fatigue-critical part of the bolt. Graph 19 shows that this is beneficial, although it becomes less effective as the radius increases. This is consistent with Peterson's notch-sensitivity relation, where further increases in root radius give smaller reductions in notch severity (Peterson, 1974).

$$q = \frac{1}{1 + \alpha/\rho}$$

Figure 3: Peterson's Notch Sensitivity Equation

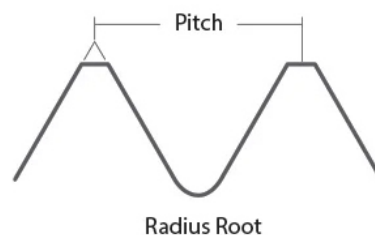
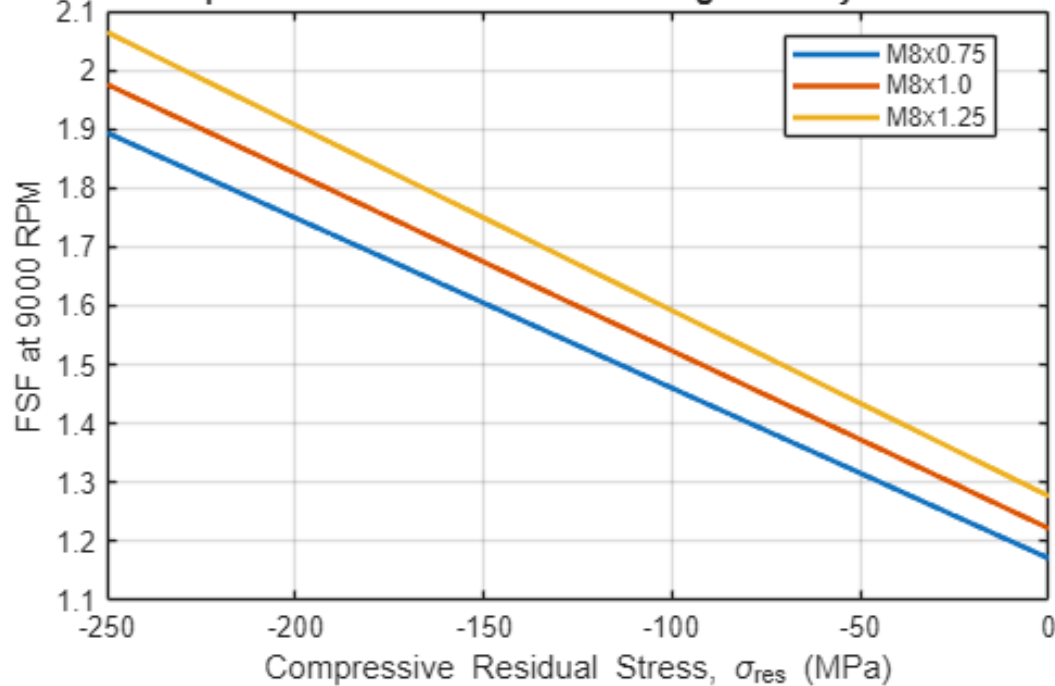


Figure 4: Taken from Earnest (2026)

6.4 - Residual Stress / Rolled Thread Effect

Effect of Compressive Residual Stress on Fatigue Safety Factor at 9000 RPM



Graph 20: Effect of Compressive Residual Stress on FSF

Thread rolling is where the thread is cold formed rather than cut. This process displaces and work-hardens the material at the thread root, leaving it in compression (STANLEY, 2015). Applying a residual stress is treating the bolt as if rolling has already left a compressive stress at the root. When the bolt is loaded in tension, part of the applied tensile stress is first offset by this built in compression, so the effective mean stress at the critical root is reduced and fatigue performance improves (Rolled Threads, 2026).

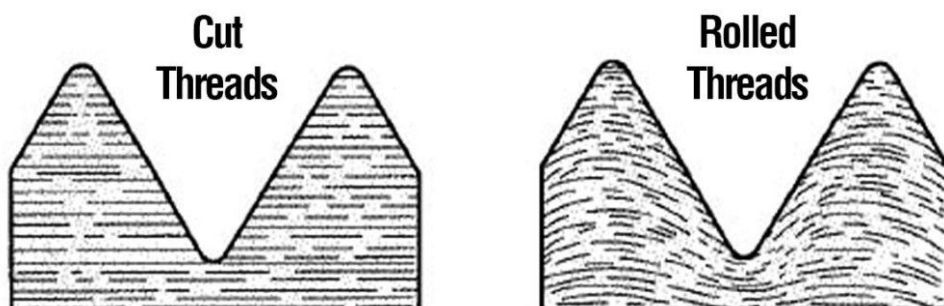


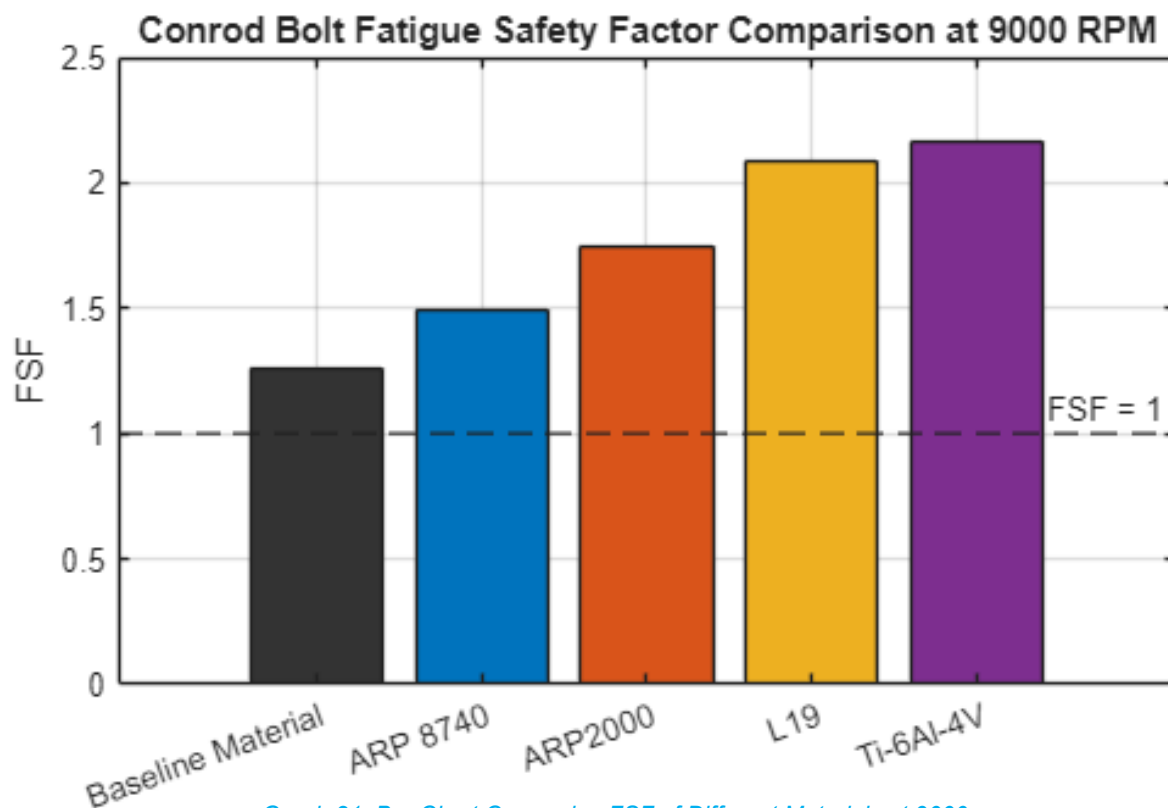
Figure 5: Taken from UC Components (2026)

6.5 - Material Study

Table 2: Material Specification (AZOM, ARP (2), TIMET (2026))

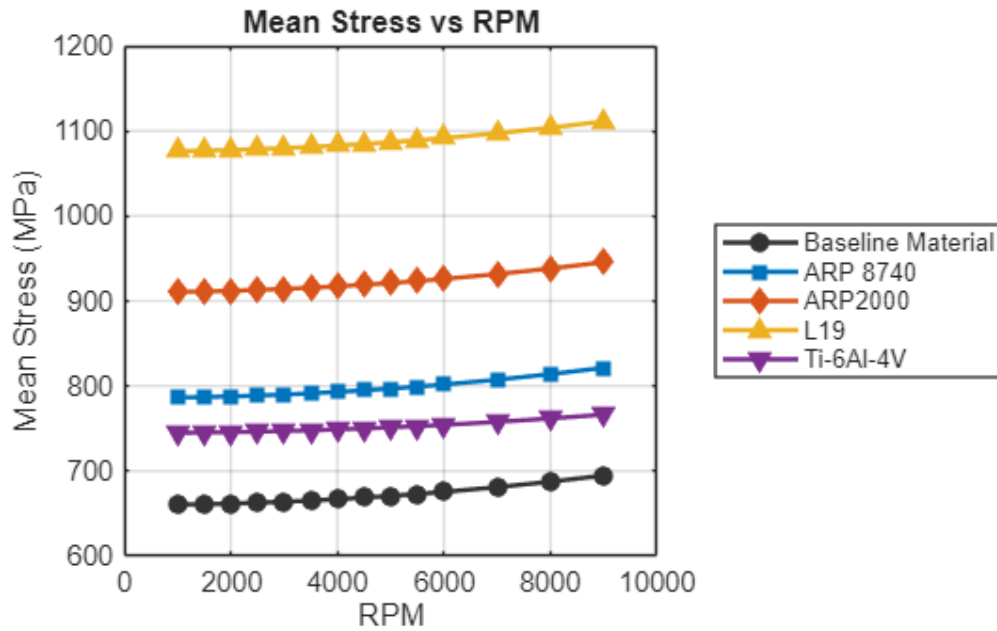
Material	UTS (Mpa)	Yield (Mpa)	Endurance Limit (Mpa)	Modulus (Gpa)	Density (g/cm ²)
Baseline Material	1100	990	495	200	7.85
ARP 8740	1310	1172	590	205	7.85
ARP2000	1517	1379	683	205	7.85
L19	1793	1517	807	205	7.85
Ti-6Al-4V	1241	1103	510	114	4.43

Four materials were compared to the baseline material, three types of steel commonly used for conrod bolts and one titanium material.



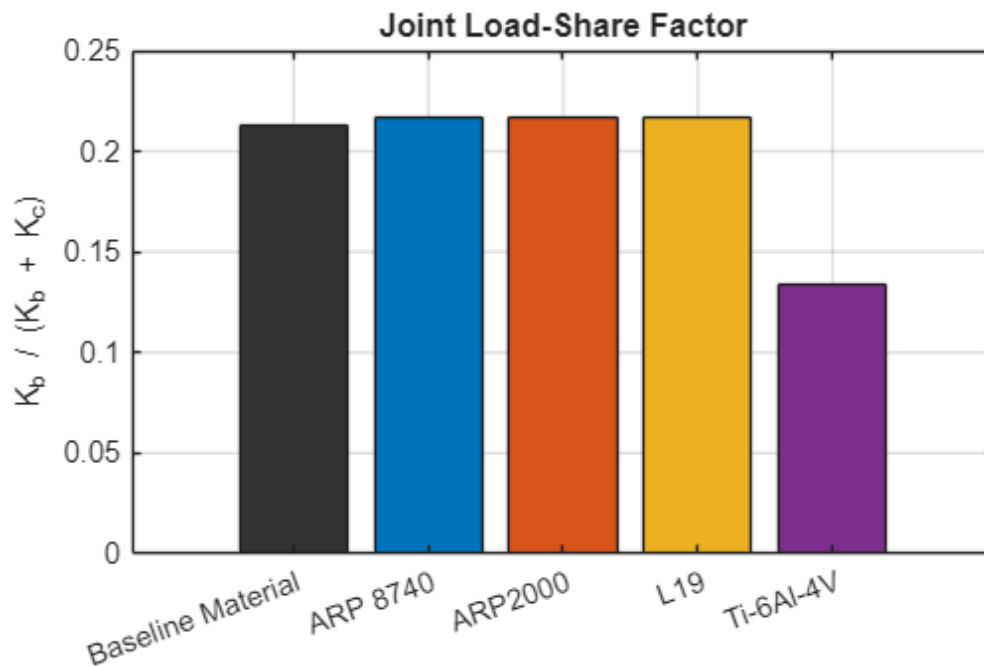
Graph 21: Bar Chart Comparing FSF of Different Materials at 9000

Titanium gives the highest FSF because its lower stiffness reduces the cyclic load carried by the bolt. On FSF alone it appears best, but this does not mean it is the best overall design choice for a conrod bolt.



Graph 22: Mean Stress against RPM

Generally increasing mean stress decreases fatigue life, however it also gives a higher preload which decreases alternating stress (Suresh. S, 1998). Titanium improves the bolt's apparent FSF allowing for large elastic extension. L19 provides a better balance between fatigue performance and joint integrity, maintaining clamp load more effectively.



Graph 23: Bar Chart Comparing Joint Load-Share of Different Materials at 9000

The titanium bolt has the lowest joint load-share factor, less of the load is being carried by the bolt and more is being transferred onto the conrod cap. This is why titanium isn't generally used for conrod bolts, although the bolt is safer in fatigue, the lower stiffness can reduce clamp effectiveness and increase the risk of cap movement/extension.

6.6 - Final Recommended Bolt Design

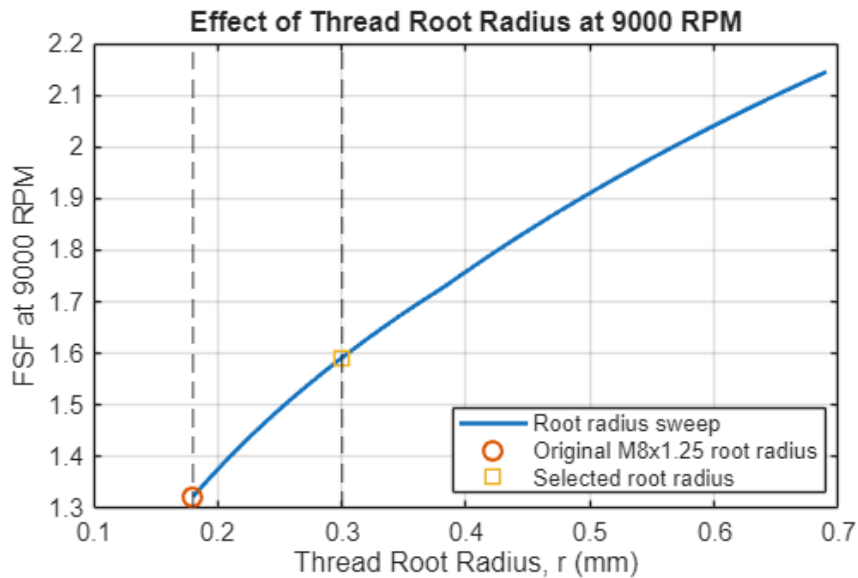
Table 3: Original and New Bolt Specification

Parameter	Original Bolt	Final Recommended Bolt
Thread Designation	M8x1.0	M8x1.25
Material	Baseline Material	L19
Kt	4.3	3.42
Tightening Torque (Nm)	36.17	58.29
Shank Diameter (mm)	7.4	7
Thread Root Radius (mm)	0.144	0.3
Residual Stress (MPa)	0	-50
UTS (MPa)	1100	1793
Endurance Limit (MPa)	495	807
FSF at 9000 RPM	1.22	2.82

The final design combines geometric and material changes to improve the FSF of the conrod bolt at 9000 RPM from 1.22 to 2.82.

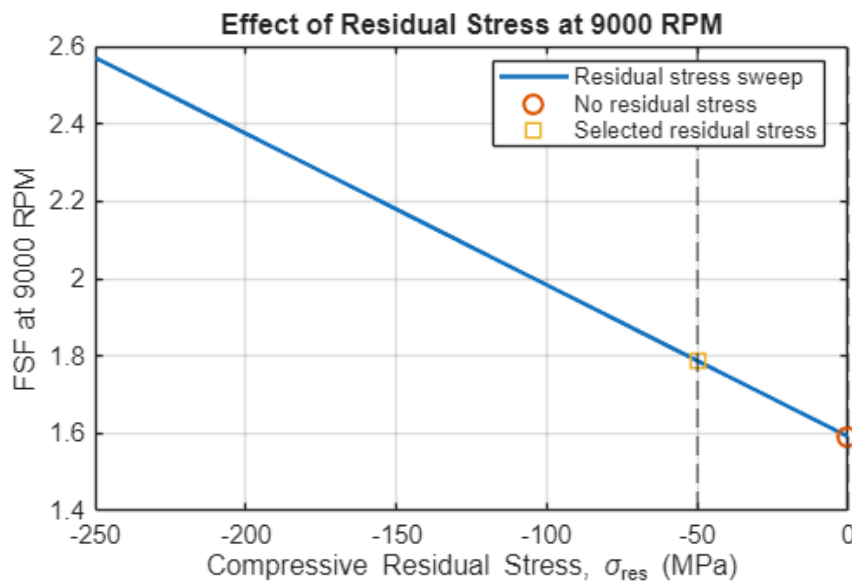
Material wise L19 has been selected. L19 is easily contaminated and subject to stress corrosion, so it must be kept well-oiled and not exposed to moisture. It also requires special care during manufacturing to avoid hydrogen embrittlement (ARP, 2026c). But, as the bolt is constantly submerged in oil, this is not an issue if care is taken during production and instalment. L19 is also a premium steel and therefore costly, however this study is looking at improving FSF without cost factors.

The original bolt shank is 7.35mm, so reducing it by 0.35mm to 7mm provides a safe balance of improved FSF without excessive bolt extension.



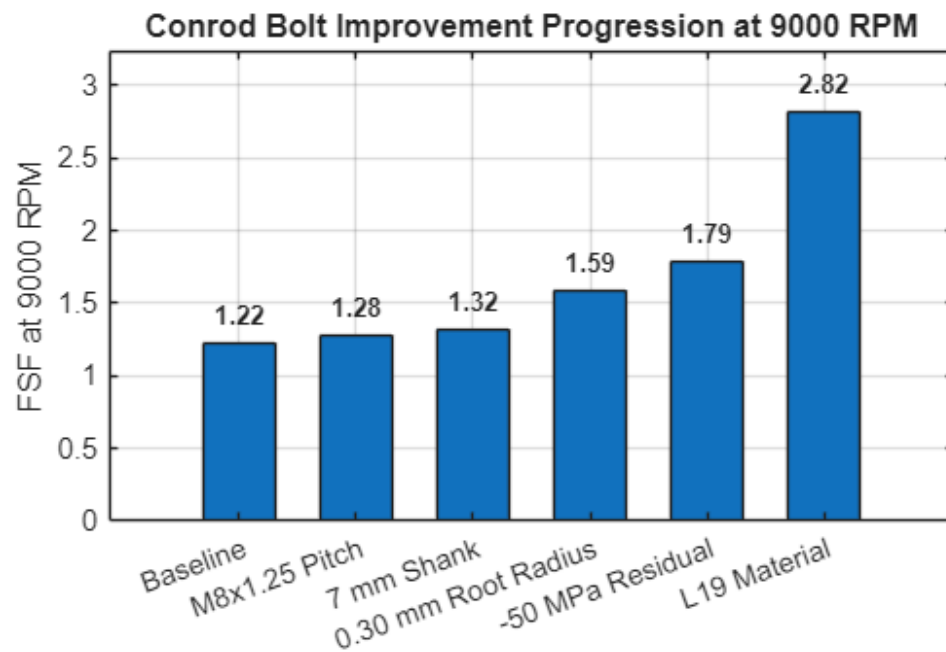
Graph 24: Root Radius Selection

A root radius of 0.30mm increases FSF without excessively changing the M8x1.25 thread. Since standard M-profile geometry gives a minimum external root radius of 0.156mm, and published ISO-derived limits allows rounded thread root up to about 0.30mm, any further increase would begin to reduce effective thread height and move the thread away from the standard profile (Bolt Science, Machining Doctor, 2026). This makes 0.30mm a practical thread-root geometry limit.



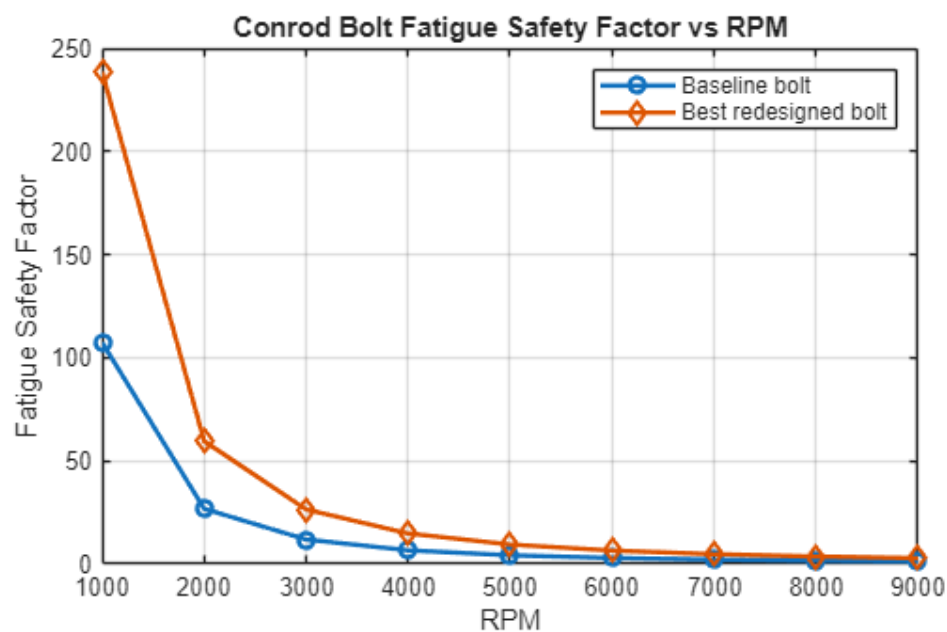
Graph 25: Residual Stress Selection

A value of -50 MPa was selected to represent the fatigue benefit of rolled threads. Published work shows that thread rolling can generate compressive residual stresses at the thread root ranging from a few hundred MPa to over 1000 MPa (Furukawa and Hagiwara, 2015). Although -50 MPa is low relative to these values, it still shifts the Goodman operating point to a safer position and increases FSF, while avoiding reliance on an unrealistic assumption.



Graph 26: Bar Chart Showing Conrod Bolt Improvement Iterations

Graph 26 shows that the redesign works cumulatively, with each modification adding a further increase in FSF. The largest single increase comes from L19, making material the most crucial factor when designing a conrod bolt.

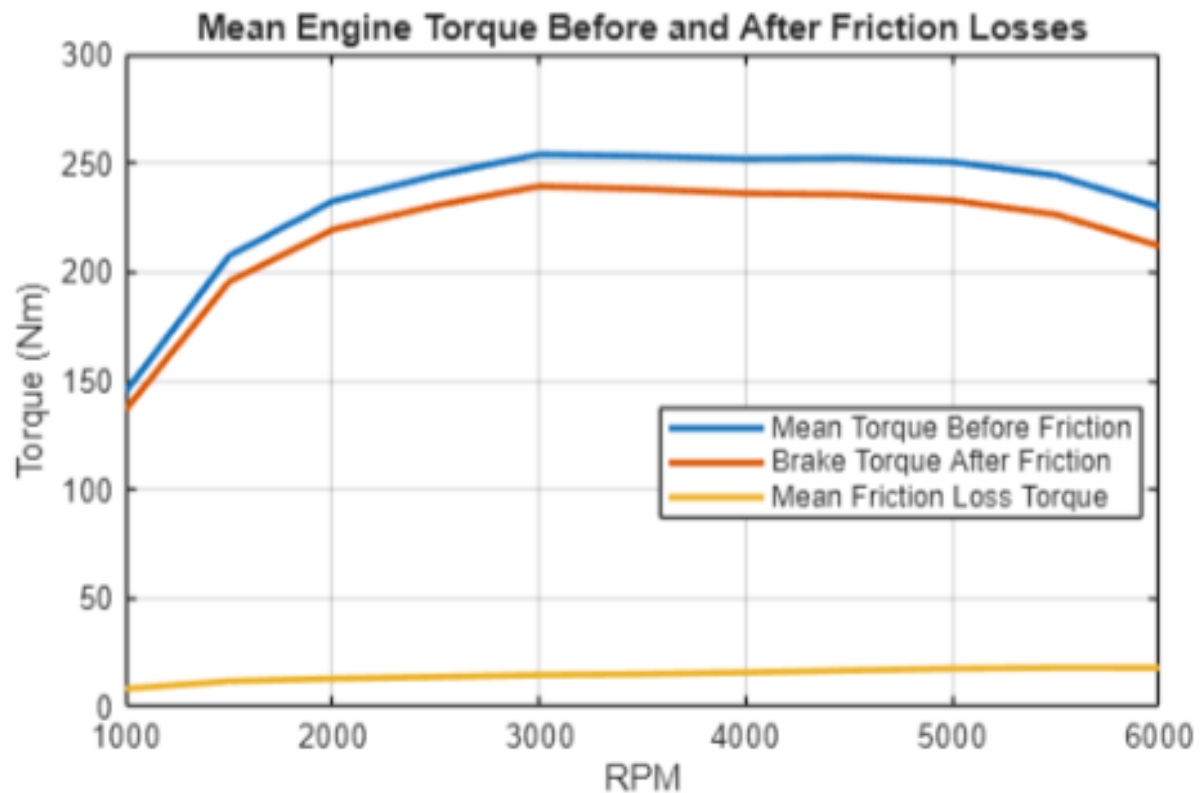


Graph 27: Old Vs New Conrod Bolt FSF against RPM

The redesigned bolt maintains a higher FSF across the full speed range, showing that the benefit is not limited to one operating point. Because of the design changes, the bolt is now improved significantly.

7 - Instantaneous Torque Analysis

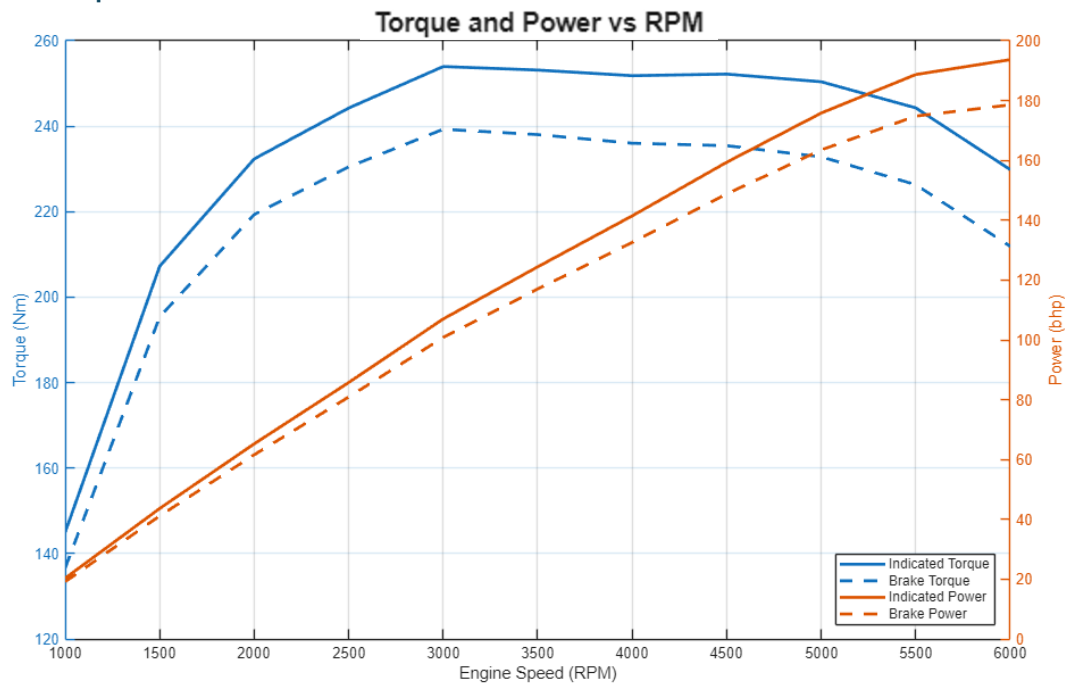
7.1 - Combined Crank Torque



Graph 28: Engine Torque Before and after Friction Losses

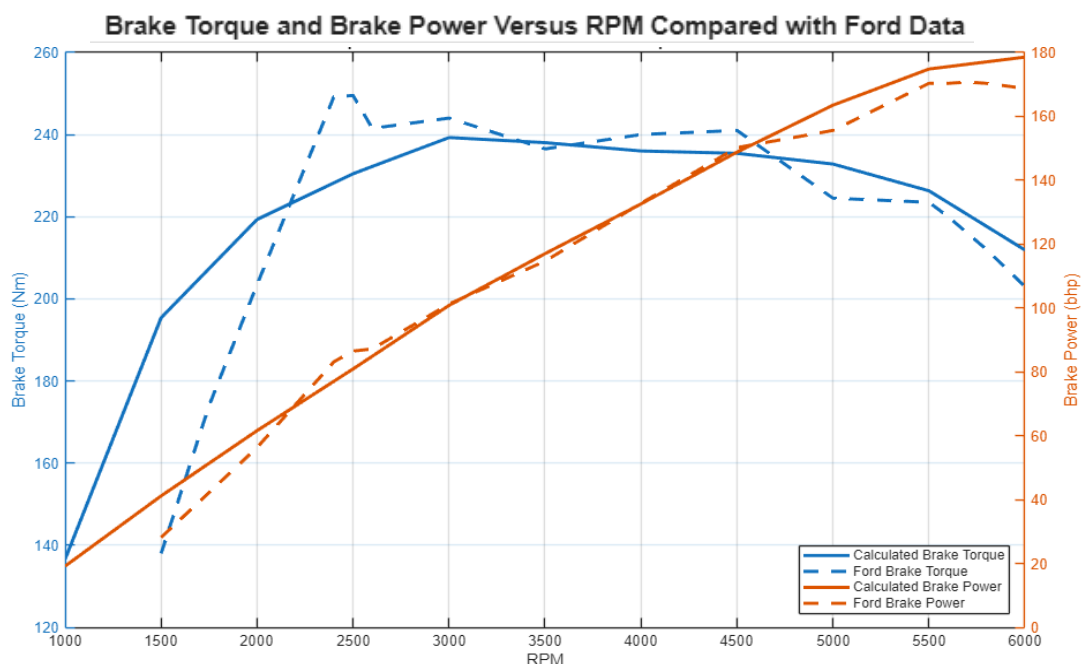
Piston ring friction reduces torque, with losses increasing as RPM increases because the rings slide faster against the chamber walls. If the oil film becomes too thin to maintain full surface separation, metal contact can occur, increasing friction even further (Ferguson. C. R, 2001).

7.2 – Torque and Power



Graph 29: Torque/Power against RPM

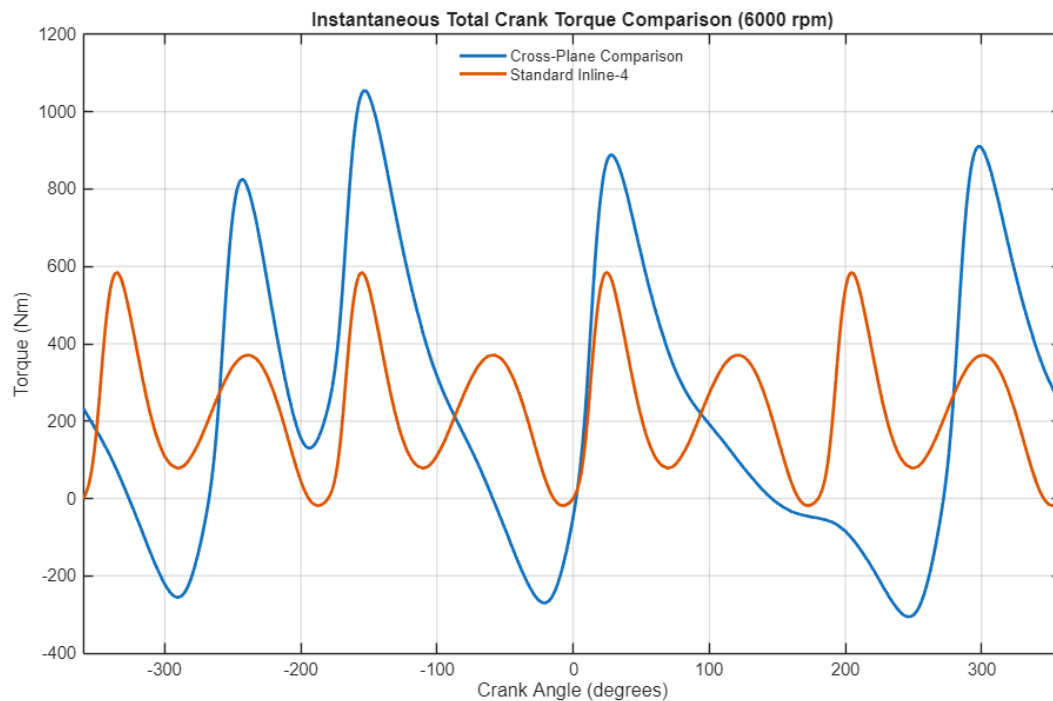
Indicated torque and power represent the work produced inside the cylinders, whereas brake torque and power are the usable output after mechanical losses have been removed.



Graph 30: Comparison Between Ford Posted Figures and Calculated Figures

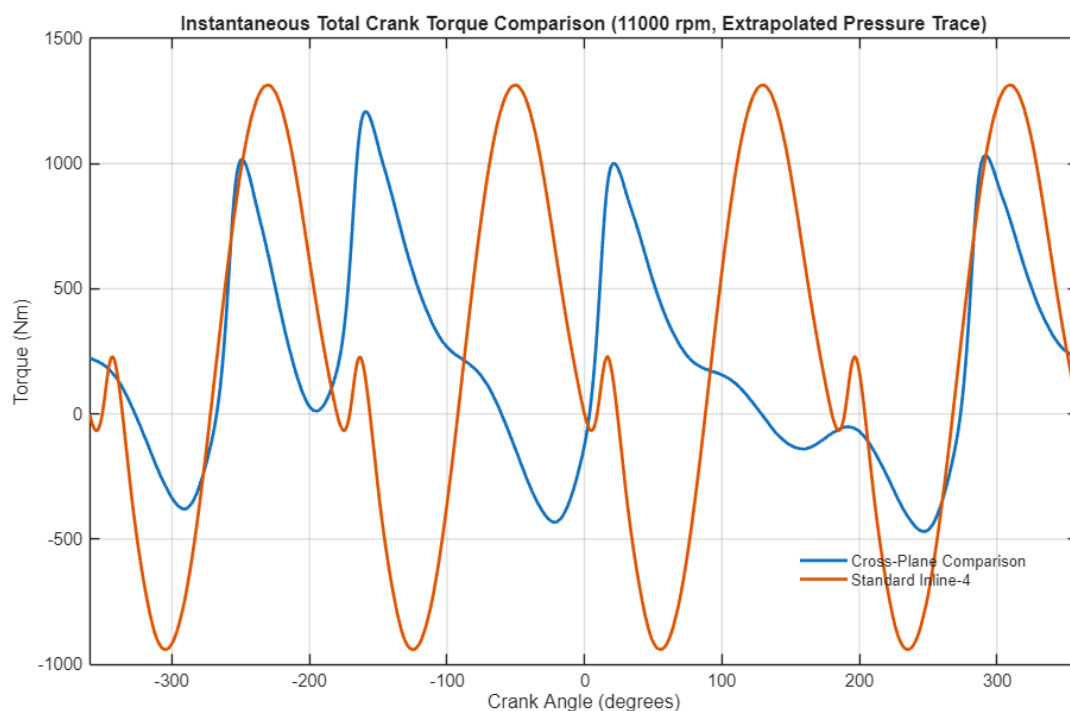
Compared to posted engine performance data by Ford (Ford Component Sales, 2015), the calculated figures come out as slightly higher. This is because the calculations used are simplified only taking piston ring friction and side-force effects into consideration. Real engines also lose output through bearing friction, other mechanical components, and driving essential engine accessories (Heywood. J. B, 2018).

7.3 - Suitability for Different Applications



Graph 31: Cross Plane and Flat Plane Torque against Crank Angle

11000 RPM was used for a comparison case as it clearly shows the difference between crank phasing where inertia speeds are amplified.



Graph 32: Cross Plane and Flat Plane Torque against Crank Angle Extrapolated to 11000 RPM

At 6000 RPM, the flat-plane inline-4 produces the smaller instantaneous total crank-torque range, giving smoother torque delivery across its operating range. Combustion torque still dominates the overall response, so its second-order torque ripple at twice crankshaft speed is not large enough to outweigh its smoother mid-range behaviour yet.

Fluctuations in instantaneous crank torque act as torsional excitations within the crankshaft and driveline. If one of these excitation orders matches a natural torsional frequency of the system, harmonic resonance can amplify vibration response, increasing audible noise/cyclic loading (Ross Performance Parts, 2024). However, at 6000 RPM these effects are not yet dominant. This makes the flat-plane configuration the better choice for a standard car, where simplicity, and acceptable noise, vibration, and harshness (NVH) are more important than performance and high-speed.

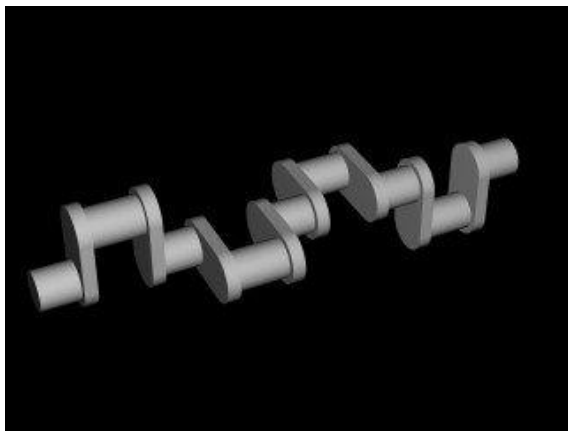


*Figure 6: Cross-Plane Crank Shaft
(David Fuller, 2022)*

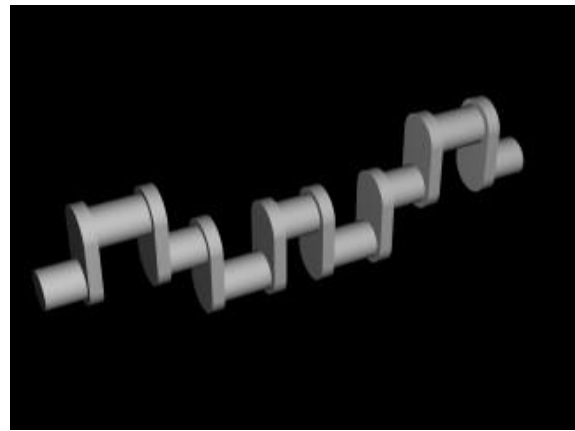


*Figure 7: Flat-Plane Crank Shaft
(David Fuller, 2022)*

At 11000 RPM, inertial effects rise sharply and begin to dominate the instantaneous torque response. As torque ripple increases, the resulting torsional excitation is more likely to align with one of the crankshaft system's natural frequencies, which could amplify vibration rather than just transmit it. This would worsen NVH, increase torsional oscillation and raise alternating stresses, potentially reducing durability.

At this speed, the cross-plane configurations 90° phasing spreads piston accelerations more evenly and reduces inertia-driven torque fluctuation (Altuğ. E, 2019, Berruga García. V, 2018). Therefore, making it more suitable when limiting excitation of torsional modes for improved performance at high-speed is more important than driver comfort and regular driving. However, cross-plane crankshafts require greater counterweight mass making them more complex, while the flat-plane design remains lighter and simpler, making them ideal for mass production (Rupp. S, 2022).

8 - Limitations and Assumptions

A major limitation of this study is the use of linear extrapolation to calculate values above 6000 RPM. Linear extrapolation may oversimplify dynamic systems like engines where the behaviour is not stable over time. This can lead to significant errors when the data exhibits complex behaviours or when external factors influence the system unpredictably (Penn State, 2026). As a result, all extrapolated data must be considered with these limitations in mind.

Another limitation is the lack of variables considered in the calculations. In real life there are many more factors which can negatively affect FSF. This includes component temperatures, mechanical efficiency, and maintenance (Heywood. J. B, 2018). Temperature has a massive effect (Altair, 2025), and components like the gudgeon pin experience large changes in temperature during operation. Errors in the estimation of material properties can also create uncertainty in the analysis of fatigue results (Suresh. S, 1998). Therefore, the actual values could be lower/higher than what was calculated in this study.

9 – Conclusion

Overall, the analysis showed that the component with the highest risk of failure was the conrod bolt. The other sections had very high FSF making them potentially overengineered. Even the thinnest conrod midpoint giving a FSF of 3.05, which remains above a published conrod minimum of 1.5 (Wenxiang Gao et al, 2022). However, that value came from a low-speed diesel study at 167 RPM using detailed fatigue analysis, so it shouldn't be treated as a direct comparison. For the bolt, the baseline result of 1.22 at 9000 RPM was slightly below published conrod bolt studies, where local FSF of about 1.25-1.35 were reported in critical thread regions for an engine at 4000–5500 RPM (Dewen Jia et al, 2025). Because the operating speeds and inputs differ, some difference is expected and the values would likely be closer with more accurate modelling. The redesign improved the bolt, reducing peak stress and increasing FSF from 1.22 to 2.88, giving a safer margin even at increased RPM than the baseline design.

The crankshaft comparison showed that flat-plane/standard configurations are generally more suitable for normal road-car operation, where simplicity, reliability, and acceptable NVH at typical engine speeds are most important. While the cross-plane arrangement is more suited for high RPM. Taken together, the study shows that effective powertrain design is a compromise between fatigue strength, dynamic loading and intended vehicle application.

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11 - Appendices

Appendix A - Input Data Tables

Ford Ecoboost 1.6 GTDI specifications		
Specification	Value	Unit
Bore Diameter	79	mm
Crank Radius	40.7	mm
Connecting Rod Length	134	mm
Reciprocating Mass	347	g
Rotational Mass	313	g
Piston Mass	268	g
Ring Masses	9	g
Clips Mass	5	g
Pin Mass	28	g

Connecting Rod Specifications		
Specification	Value	Unit
Cap Interface Cross Sectional Area	210	mm ²
Modulus of Elasticity	170000	N/mm ²
Length of Connecting Rod Section Interacting with Bolt	31	mm
UTS of Material	1000	Mpa
Cross Sectional Area at Midpoint	196.39	mm ²
Hoop Width	13.11	mm
Hoop Thickness	3.45	mm

Gudgeon Pin Specifications		
Specification	Value	Unit
L1 Inner Length	7	mm
L2 Outer Length	7.75	mm
Exposed Mass	23.28	g
Cross Sectional Area	227.54	mm ²
D1 Inner Diameter	12.3	mm
D2 Outer Diameter	21	mm
Neutral Axis	10.5	mm
Moment of Inertia	8423	Kg/mm ²
UTS of Material	1000	Mpa
Endurance Limit of Material	45	% of UTS

Connecting Rod Bolt Specifications		
Specification	Value	Unit
Cross Sectional Area of Bolt	42.43	mm ²
Modulus of Elasticity	200000	N/mm ²
Length of Interacting Bolt Section	21.26	mm
Length to 2nd Engaged Thread (Effective Working Length)	27.26	mm
UTS of Material	1100	Mpa
Yield Strength of Material	90	% of UTS
Endurance Limit of Material	45	% of UTS

Nominal Size ISO MF	Thread Form Type	Major Diameter mm d=D	Pitch mm p	Root Radius mm r
8.0x0.75	M	8	0.75	0.108
8.0x1.0	M	8	1	0.144
8.0x1.25	M	8	1.25	0.18

Nominal Size ISO MF	Pitch Diameter mm d2=D2	Minor Diameter Male Thd. d3	Minor Diameter Female Thd. D1
8.0x0.75	7.513	7.08	7.188
8.0x1.0	7.35	6.773	6.917
8.0x1.25	7.188	6.466	6.647

Nominal Size ISO MF	Thread Height Male Thd. h3	Thread Height Female Thd. H1	Tap Drill Diameter mm
8.0x0.75	0.46	0.406	7.2
8.0x1.0	0.613	0.541	7
8.0x1.25	0.767	0.677	6.8

Appendix B – Worked Equations and Sample Calculations

Appendix BA – Equations Used for FSF

$$F_{Inertia} = m_{rec}a$$

$$\sigma_{mean} = \frac{\sigma_{min} + \sigma_{max}}{2}$$

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

$$\sigma_{range} = \sigma_{max} - \sigma_{min}$$

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

$$\sigma_{allowable} = 2\sigma_{End-Lim} \times \left(1 - \frac{\sigma_{mean}}{UTS}\right)$$

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

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

$$\sigma_{maxNew} = \sigma_{max} + \sigma_{SurfaceFinishPressure}$$

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

$$\sigma_{minNew} = \sigma_{min} + \sigma_{SurfaceFinishPressure}$$

$$K = \varepsilon \frac{A}{L}$$

$$\sigma_{meanNew} = \frac{\sigma_{minNew} + \sigma_{maxNew}}{2}$$

$$\frac{1}{K_{bolt}} = \frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3}$$

$$\sigma_{allowableNew} = 2\sigma_{End-Lim} \times \left(1 - \frac{\sigma_{meanNew}}{UTS}\right)$$

$$F_{axial} = \sigma_{axial} \times \text{Cross Sectional Area}$$

$$FSF_{NEW} = \frac{\sigma_{allowableNew}}{\sigma_{rangeNew}}$$

$$F_{friction} = \mu \times F_{axial}$$

$$F_{extra\ bolt} = \frac{F_{inertia\ bolt} K_{bolt}}{(K_{bolt} + K_{cap})}$$

$$F_{inertia} = m_{rec} \left(R\omega^2 \cos\theta + \frac{R^2}{L} \omega^2 \cos 2\theta \right) + m_{rot} (R\omega^2 \cos 2\theta)$$

Appendix BB – Equations Used for Torque and Power

$$Work_{Torque} = \sum Crank Torque$$

$$Brake_{Torque} = Work_{Torque} - Work_{Loss total}$$

$$Indicated_{power} = EngineTorque \times \omega$$

$$Friction_{power} = Work_{Loss total} \times \omega$$

$$Brake_{power} = Indicated_{power} - Friction_{power}$$

$$Work_{Gas} = \sum Torque_{Gas}$$

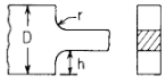
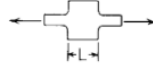
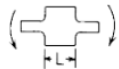
$$IMEP = \frac{Work_{Gas}}{R \times 2 \times A}$$

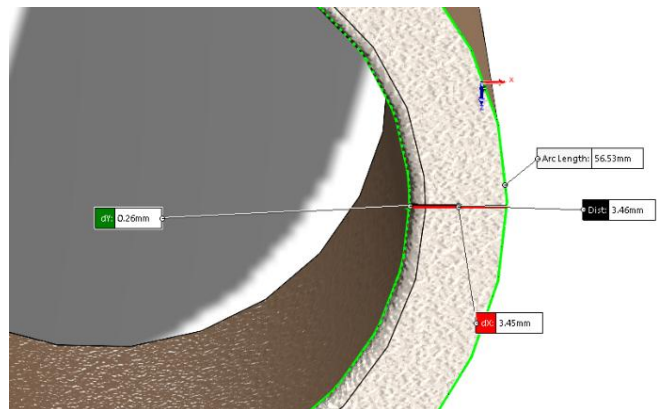
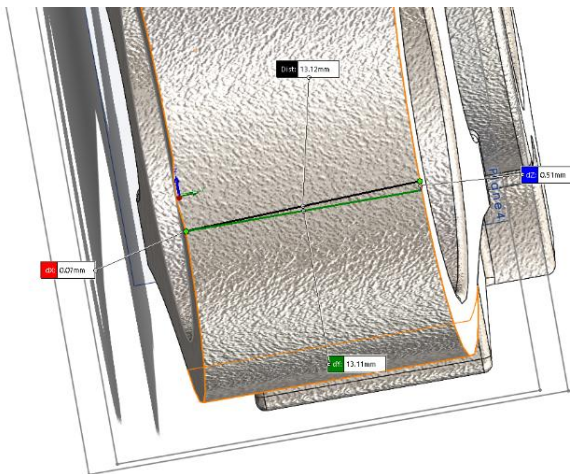
$$FMEP = \frac{Work_{Loss total}}{R \times 2 \times A}$$

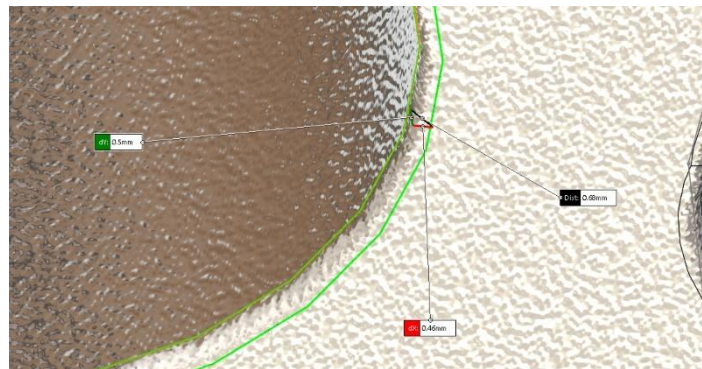
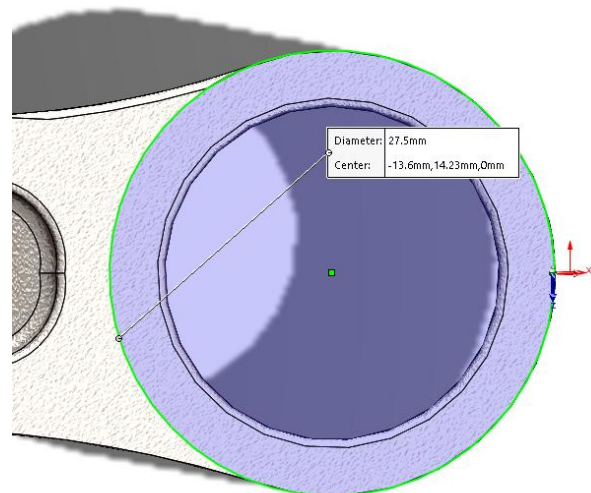
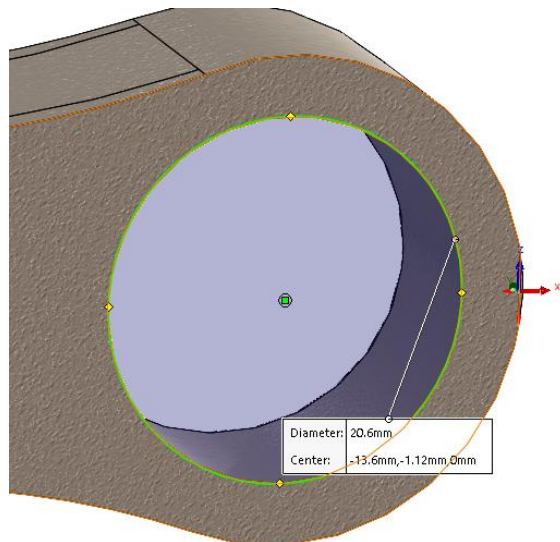
$$BMEP = IMEP - FMEP$$

Appendix BC – Calculating Kt for Conrod Hoop

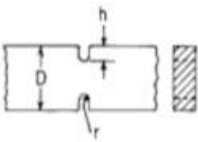
TABLE 17.1 Stress concentration factors for elastic stress (K_t) (Continued)

Type of form irregularity or stress raiser	Stress condition and manner of loading	Stress concentration factor K_t for various dimensions															
5. Square shoulder with fillet in a member of rectangular section 	5a. Elastic stress, axial tension 	$K_t = C_1 + C_2 \left(\frac{2h}{D} \right) + C_3 \left(\frac{2h}{D} \right)^2 + C_4 \left(\frac{2h}{D} \right)^3$ where $\frac{L}{D} > \frac{3}{[r/(D-2h)]^{1/4}}$ and where <table><tr><th></th><th>$0.1 \leq h/r \leq 2.0$</th><th>$2.0 \leq h/r \leq 20.0$</th></tr><tr><td>C_1</td><td>$1.007 + 1.000\sqrt{h/r} - 0.031h/r$</td><td>$1.042 + 0.982\sqrt{h/r} - 0.036h/r$</td></tr><tr><td>$C_2$</td><td>$-0.114 - 0.585\sqrt{h/r} + 0.314h/r$</td><td>$-0.074 - 0.156\sqrt{h/r} - 0.010h/r$</td></tr><tr><td>$C_3$</td><td>$0.241 - 0.992\sqrt{h/r} - 0.271h/r$</td><td>$-3.418 + 1.220\sqrt{h/r} - 0.005h/r$</td></tr><tr><td>$C_4$</td><td>$-0.134 + 0.577\sqrt{h/r} - 0.012h/r$</td><td>$3.450 - 2.046\sqrt{h/r} + 0.051h/r$</td></tr></table> (Refs. 1, 8, 11, and 19) For cases where $\frac{L}{D} < \frac{3}{[r/(D-2h)]^{1/4}}$ see Refs. 1, 21, and 22.		$0.1 \leq h/r \leq 2.0$	$2.0 \leq h/r \leq 20.0$	C_1	$1.007 + 1.000\sqrt{h/r} - 0.031h/r$	$1.042 + 0.982\sqrt{h/r} - 0.036h/r$	C_2	$-0.114 - 0.585\sqrt{h/r} + 0.314h/r$	$-0.074 - 0.156\sqrt{h/r} - 0.010h/r$	C_3	$0.241 - 0.992\sqrt{h/r} - 0.271h/r$	$-3.418 + 1.220\sqrt{h/r} - 0.005h/r$	C_4	$-0.134 + 0.577\sqrt{h/r} - 0.012h/r$	$3.450 - 2.046\sqrt{h/r} + 0.051h/r$
	$0.1 \leq h/r \leq 2.0$	$2.0 \leq h/r \leq 20.0$															
C_1	$1.007 + 1.000\sqrt{h/r} - 0.031h/r$	$1.042 + 0.982\sqrt{h/r} - 0.036h/r$															
C_2	$-0.114 - 0.585\sqrt{h/r} + 0.314h/r$	$-0.074 - 0.156\sqrt{h/r} - 0.010h/r$															
C_3	$0.241 - 0.992\sqrt{h/r} - 0.271h/r$	$-3.418 + 1.220\sqrt{h/r} - 0.005h/r$															
C_4	$-0.134 + 0.577\sqrt{h/r} - 0.012h/r$	$3.450 - 2.046\sqrt{h/r} + 0.051h/r$															
5b. Elastic stress, in-plane bending 	$K_t = C_1 + C_2 \left(\frac{2h}{D} \right) + C_3 \left(\frac{2h}{D} \right)^2 + C_4 \left(\frac{2h}{D} \right)^3$ where $\frac{L}{D} > \frac{0.8}{[r/(D-2h)]^{1/4}}$ and where <table><tr><th></th><th>$0.1 \leq h/r \leq 2.0$</th><th>$2.0 \leq h/r \leq 20.0$</th></tr><tr><td>C_1</td><td>$1.007 + 1.000\sqrt{h/r} - 0.031h/r$</td><td>$1.042 + 0.982\sqrt{h/r} - 0.036h/r$</td></tr><tr><td>$C_2$</td><td>$-0.270 - 2.404\sqrt{h/r} + 0.749h/r$</td><td>$-3.599 + 1.619\sqrt{h/r} - 0.431h/r$</td></tr><tr><td>$C_3$</td><td>$0.677 + 1.133\sqrt{h/r} - 0.904h/r$</td><td>$6.084 - 5.607\sqrt{h/r} + 1.158h/r$</td></tr><tr><td>$C_4$</td><td>$-0.414 + 0.271\sqrt{h/r} + 0.186h/r$</td><td>$-2.527 + 3.006\sqrt{h/r} - 0.691h/r$</td></tr></table> (Refs. 1, 11, and 20) For cases where $\frac{L}{D} < \frac{0.8}{[r/(D-2h)]^{1/4}}$ see Refs. 1 and 20.		$0.1 \leq h/r \leq 2.0$	$2.0 \leq h/r \leq 20.0$	C_1	$1.007 + 1.000\sqrt{h/r} - 0.031h/r$	$1.042 + 0.982\sqrt{h/r} - 0.036h/r$	C_2	$-0.270 - 2.404\sqrt{h/r} + 0.749h/r$	$-3.599 + 1.619\sqrt{h/r} - 0.431h/r$	C_3	$0.677 + 1.133\sqrt{h/r} - 0.904h/r$	$6.084 - 5.607\sqrt{h/r} + 1.158h/r$	C_4	$-0.414 + 0.271\sqrt{h/r} + 0.186h/r$	$-2.527 + 3.006\sqrt{h/r} - 0.691h/r$	
	$0.1 \leq h/r \leq 2.0$	$2.0 \leq h/r \leq 20.0$															
C_1	$1.007 + 1.000\sqrt{h/r} - 0.031h/r$	$1.042 + 0.982\sqrt{h/r} - 0.036h/r$															
C_2	$-0.270 - 2.404\sqrt{h/r} + 0.749h/r$	$-3.599 + 1.619\sqrt{h/r} - 0.431h/r$															
C_3	$0.677 + 1.133\sqrt{h/r} - 0.904h/r$	$6.084 - 5.607\sqrt{h/r} + 1.158h/r$															
C_4	$-0.414 + 0.271\sqrt{h/r} + 0.186h/r$	$-2.527 + 3.006\sqrt{h/r} - 0.691h/r$															

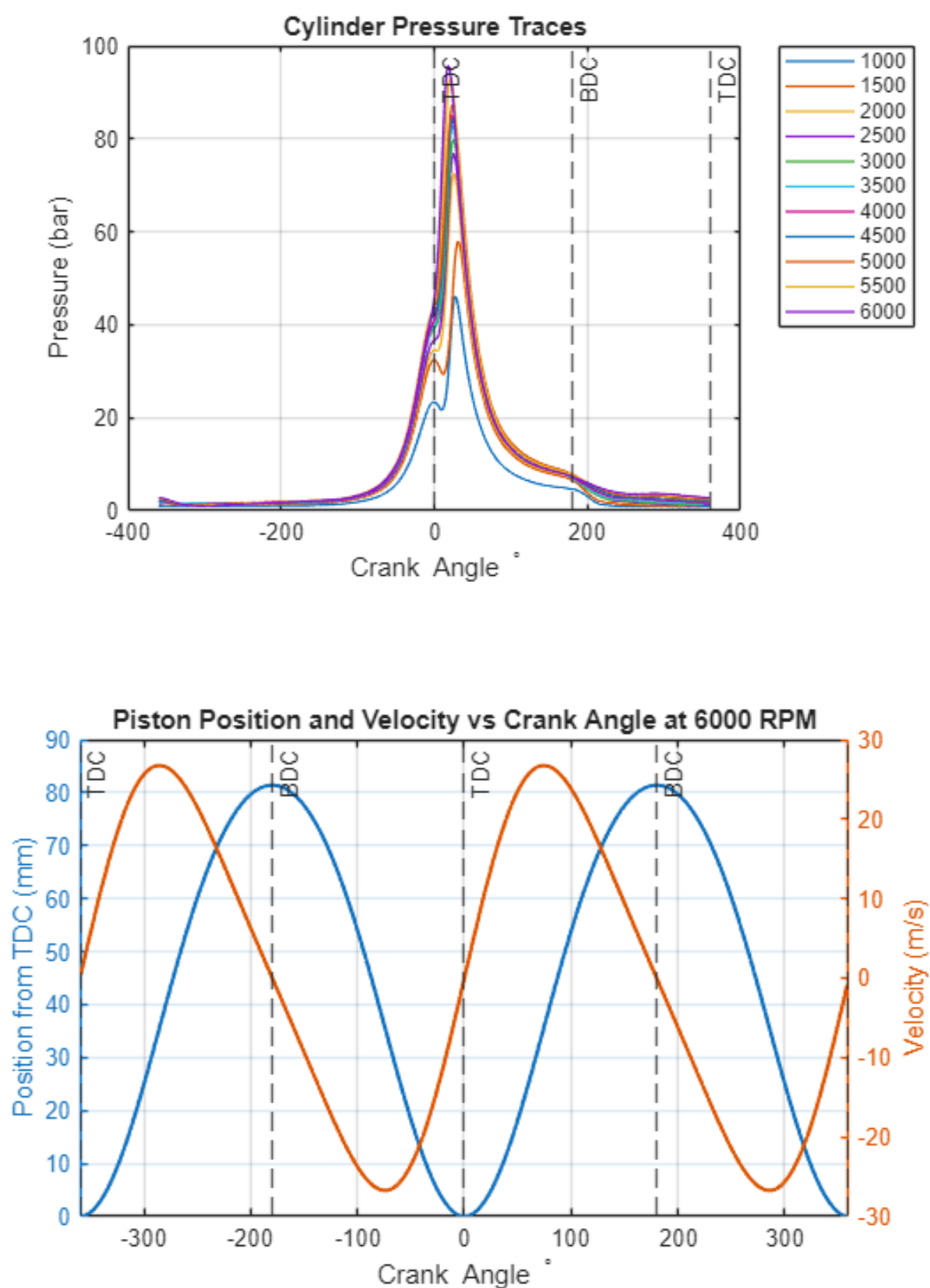


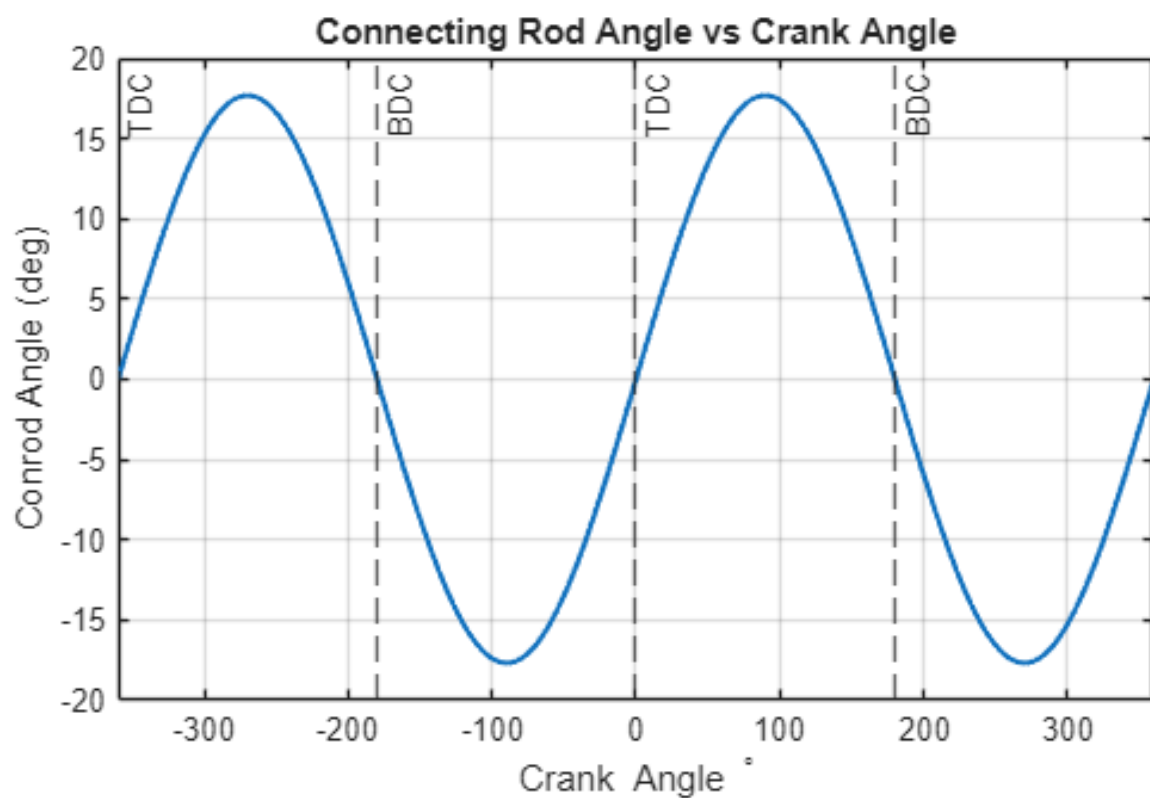
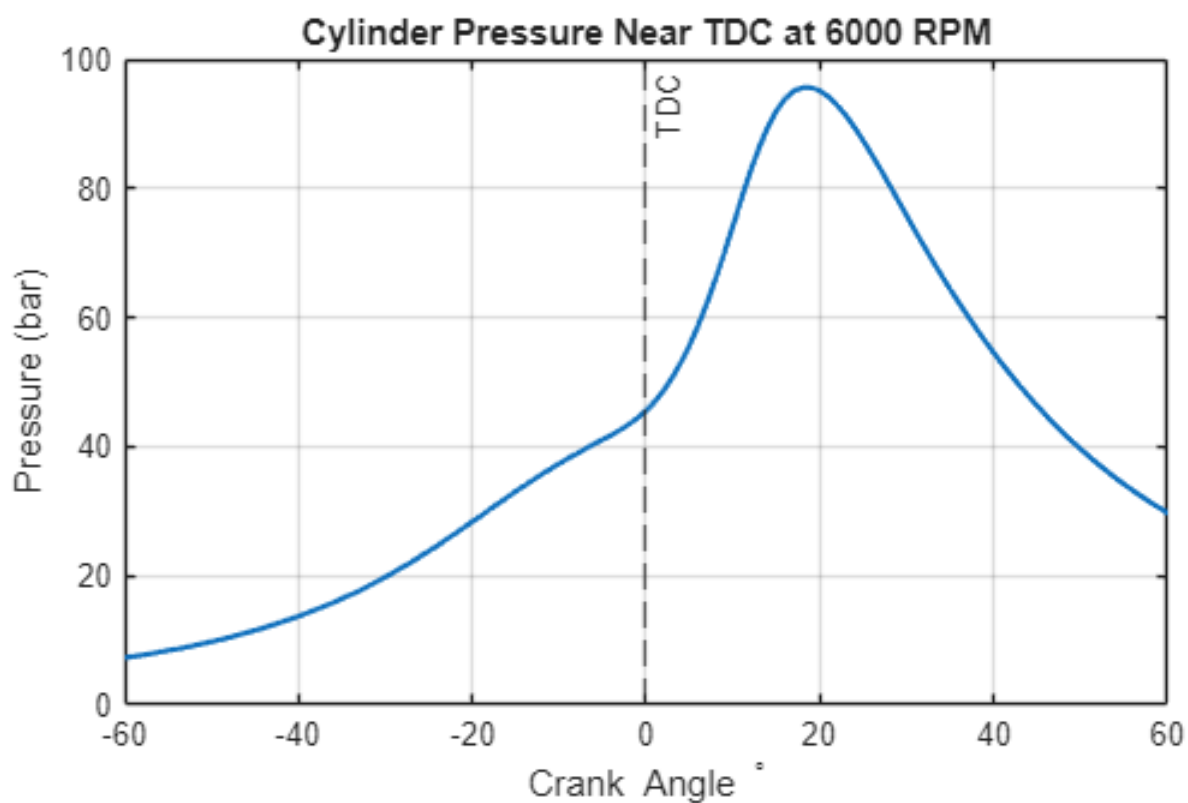


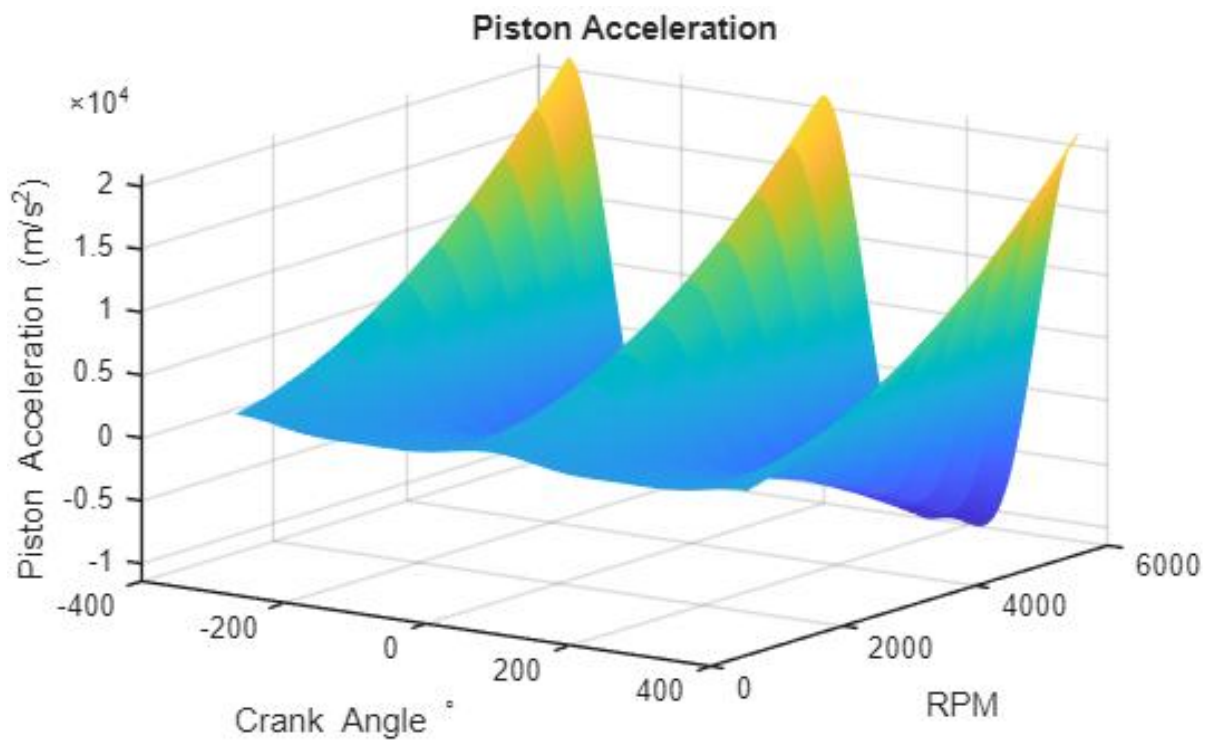
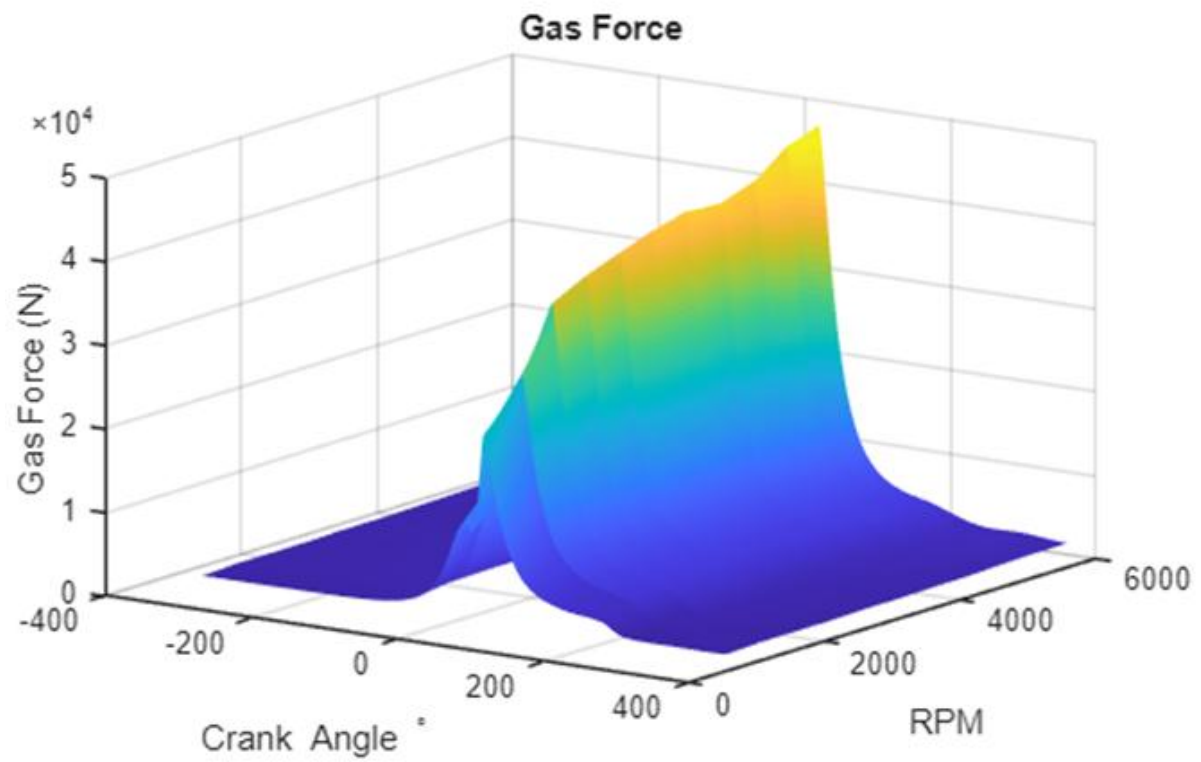
Appendix BD – Calculating Kt for the Conrod Bolt

Type of form irregularity or stress raiser	Stress condition and manner of loading	Stress concentration factor K_t for various dimensions															
1. Two U-notches in a member of rectangular section 	1a. Elastic stress, axial tension	$K_t = C_1 + C_2 \left(\frac{2h}{D} \right) + C_3 \left(\frac{2h}{D} \right)^2 + C_4 \left(\frac{2h}{D} \right)^3$ where <table border="1"> <thead> <tr> <th></th><th>$0.1 \leq h/r \leq 2.0$</th><th>$2.0 \leq h/r \leq 50.0$</th></tr> </thead> <tbody> <tr> <td>C_1</td><td>$0.850 + 2.628\sqrt{h/r} - 0.413h/r$</td><td>$0.833 + 2.069\sqrt{h/r} - 0.009h/r$</td></tr> <tr> <td>$C_2$</td><td>$-1.119 - 4.826\sqrt{h/r} + 2.575h/r$</td><td>$2.732 - 4.157\sqrt{h/r} + 0.176h/r$</td></tr> <tr> <td>$C_3$</td><td>$3.563 - 0.514\sqrt{h/r} - 2.402h/r$</td><td>$-8.859 + 5.327\sqrt{h/r} - 0.320h/r$</td></tr> <tr> <td>$C_4$</td><td>$-2.294 + 2.713\sqrt{h/r} + 0.240h/r$</td><td>$6.294 - 3.239\sqrt{h/r} + 0.154h/r$</td></tr> </tbody> </table> For the semicircular notch ($h/r = 1$) $K_t = 3.065 - 3.370 \left(\frac{2h}{D} \right) + 0.647 \left(\frac{2h}{D} \right)^2 + 0.658 \left(\frac{2h}{D} \right)^3$		$0.1 \leq h/r \leq 2.0$	$2.0 \leq h/r \leq 50.0$	C_1	$0.850 + 2.628\sqrt{h/r} - 0.413h/r$	$0.833 + 2.069\sqrt{h/r} - 0.009h/r$	C_2	$-1.119 - 4.826\sqrt{h/r} + 2.575h/r$	$2.732 - 4.157\sqrt{h/r} + 0.176h/r$	C_3	$3.563 - 0.514\sqrt{h/r} - 2.402h/r$	$-8.859 + 5.327\sqrt{h/r} - 0.320h/r$	C_4	$-2.294 + 2.713\sqrt{h/r} + 0.240h/r$	$6.294 - 3.239\sqrt{h/r} + 0.154h/r$
	$0.1 \leq h/r \leq 2.0$	$2.0 \leq h/r \leq 50.0$															
C_1	$0.850 + 2.628\sqrt{h/r} - 0.413h/r$	$0.833 + 2.069\sqrt{h/r} - 0.009h/r$															
C_2	$-1.119 - 4.826\sqrt{h/r} + 2.575h/r$	$2.732 - 4.157\sqrt{h/r} + 0.176h/r$															
C_3	$3.563 - 0.514\sqrt{h/r} - 2.402h/r$	$-8.859 + 5.327\sqrt{h/r} - 0.320h/r$															
C_4	$-2.294 + 2.713\sqrt{h/r} + 0.240h/r$	$6.294 - 3.239\sqrt{h/r} + 0.154h/r$															

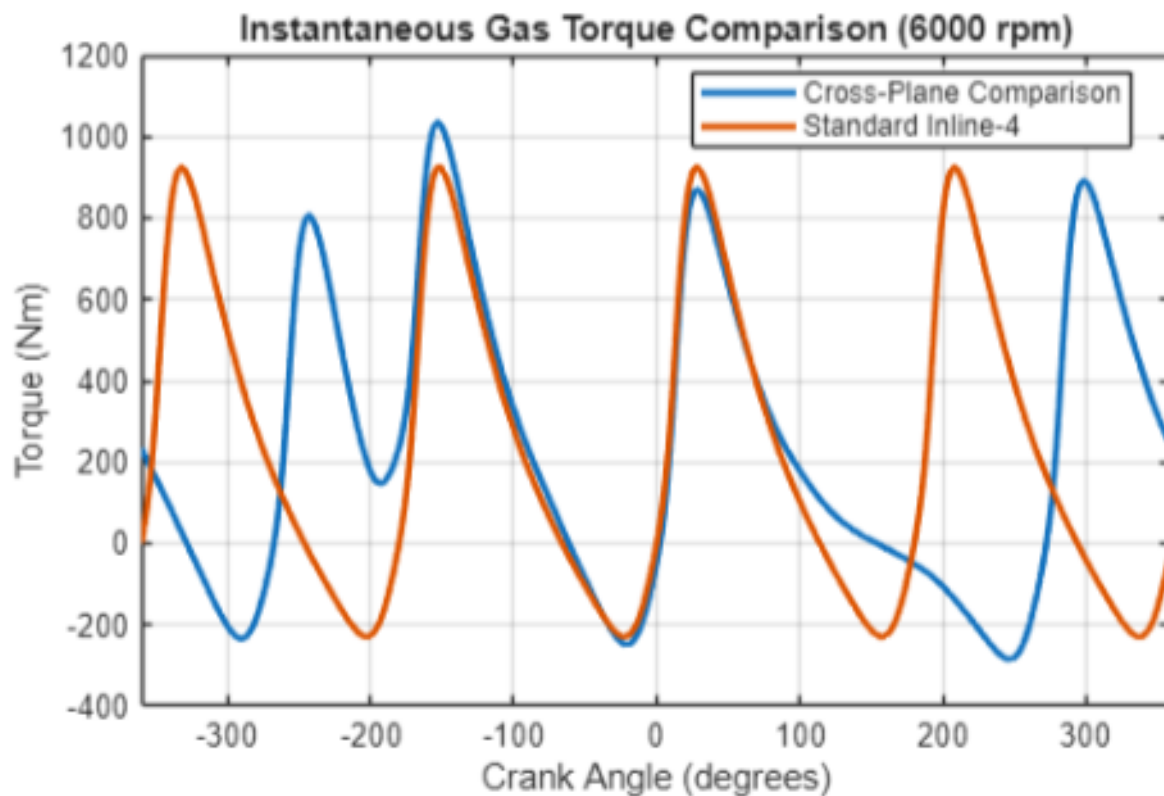
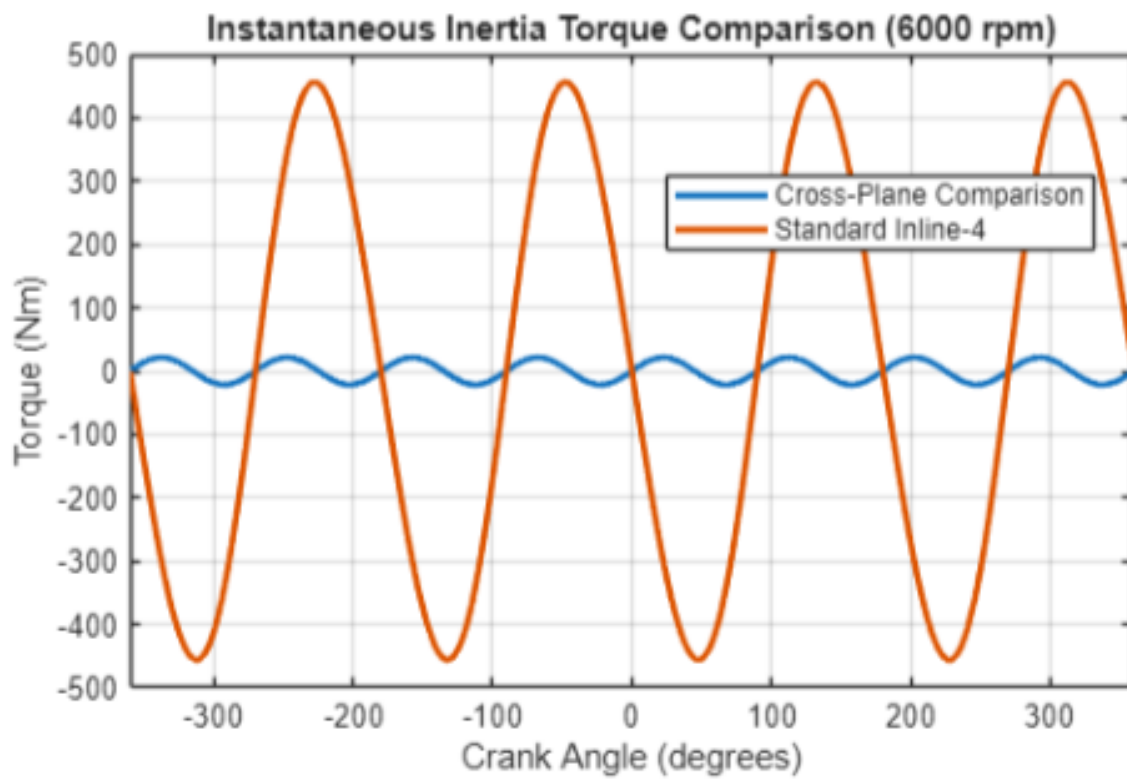
Appendix C – Supplementary Kinematics / Pressure / Force Plots

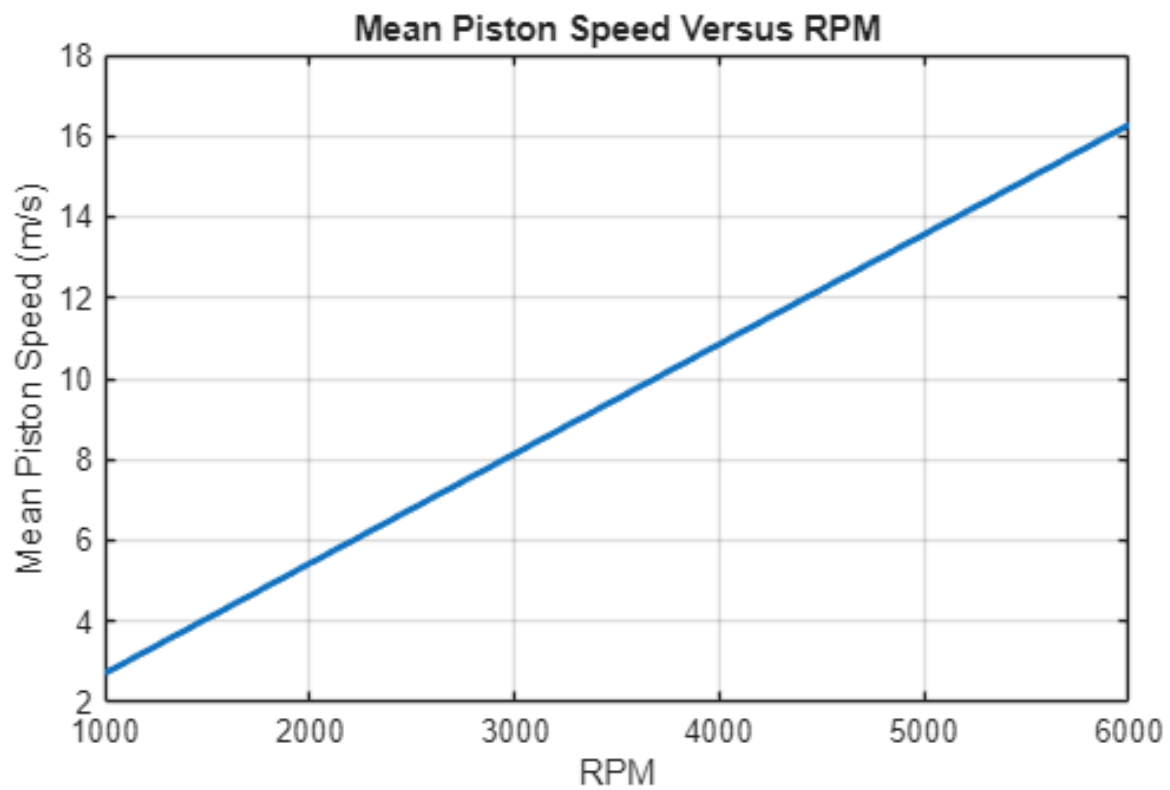
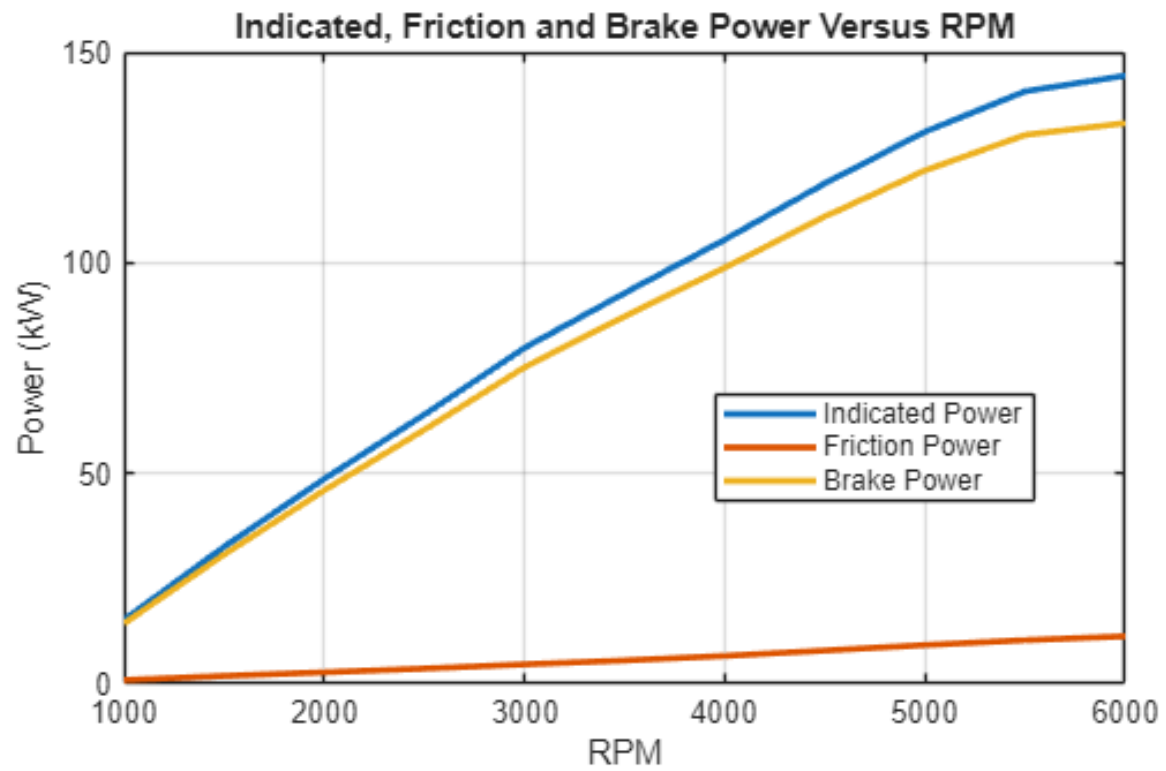


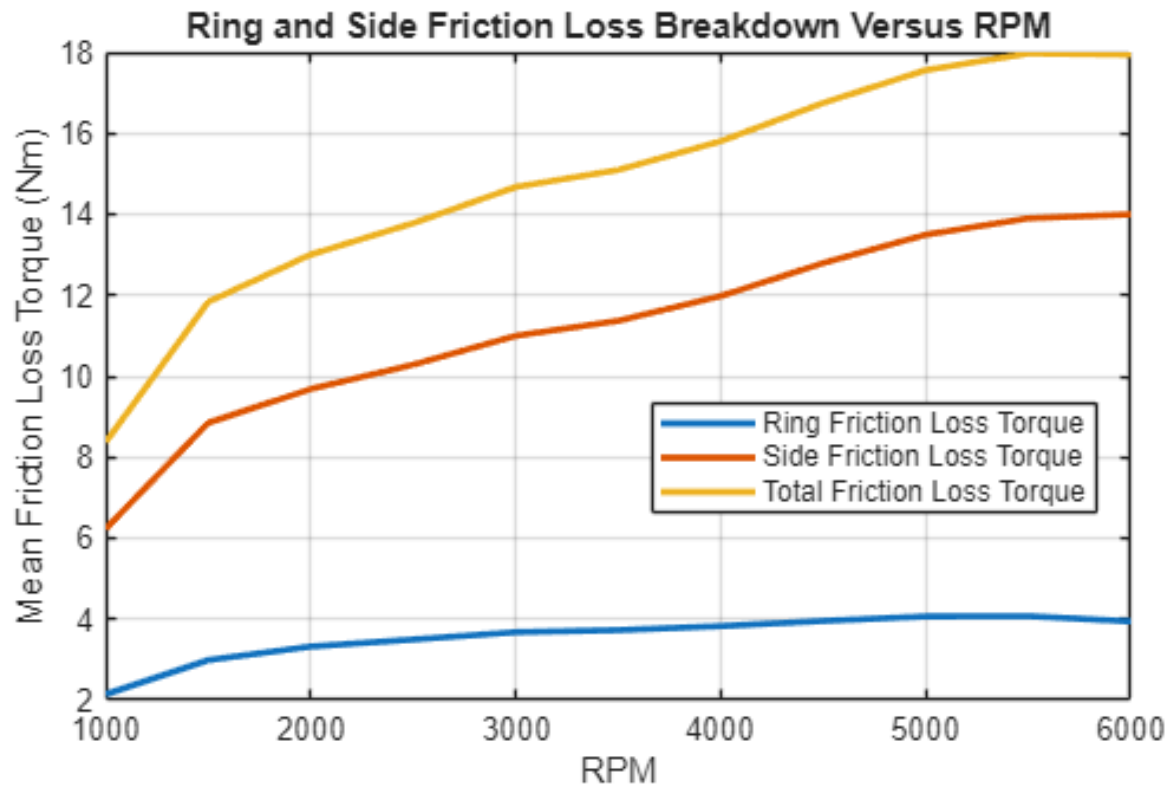




Appendix D – Supplementary Instantaneous Torque and Power Graphs







Appendix E – RPM Extrapolation to 9000 RPM

