



University of Wales Trinity Saint David

Wales Institute of Science & Art

Analytical Assessment of Engine Internal Components

Contemporary Powertrain

ACHA6006

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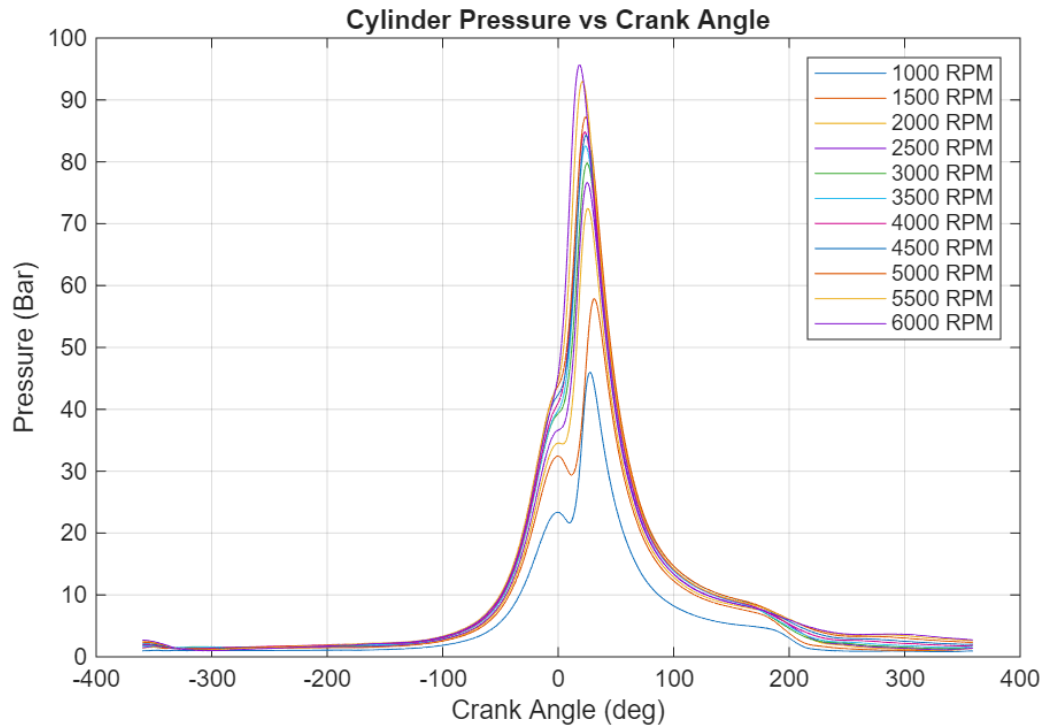
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1 Introduction

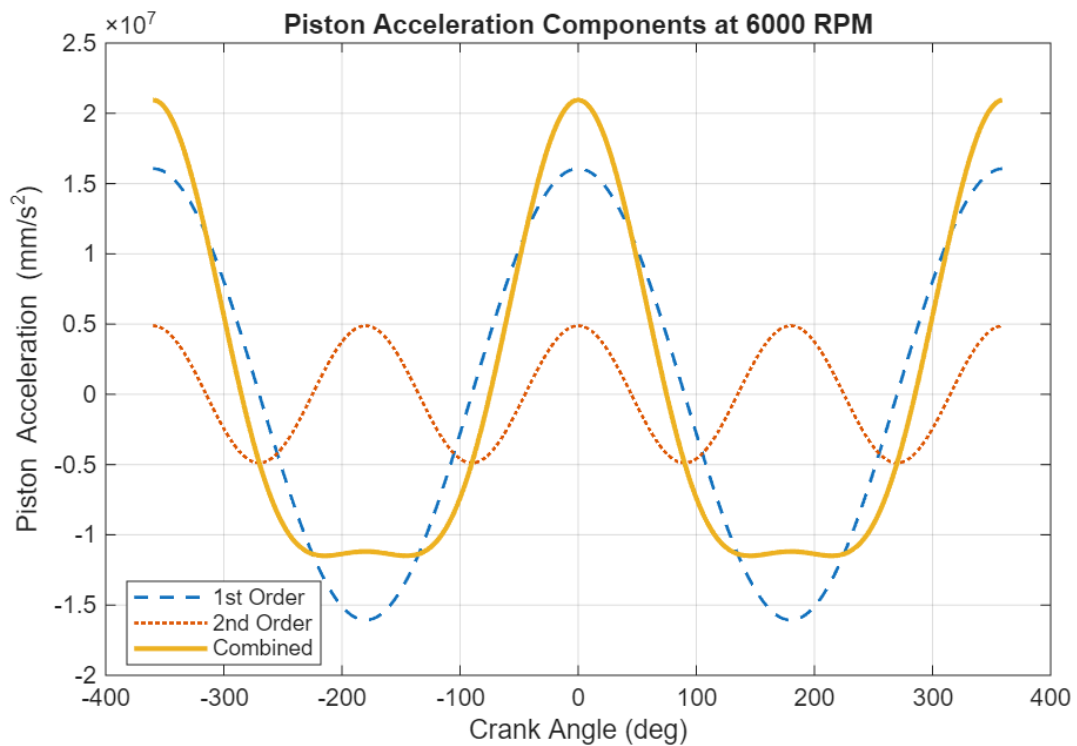
Internal combustion engine components are subjected to high cyclic loading due to the combined effects of combustion pressure and inertia forces. These conditions make fatigue failure a critical design consideration, particularly at elevated engine speeds where inertia effects become increasingly dominant. This study evaluates the fatigue performance of key bottom-end components within the Ford GTDi EcoBoost engine using analytical methods. Stresses and fatigue safety factors are determined across standard and extended operating conditions to assess component reliability. A torque and power analysis is also conducted, considering frictional losses and the influence of crankshaft configuration. Different crankshaft designs are analysed to determine their suitability for varying applications, providing further context to the loading conditions experienced.

2 Engine Kinematics and Force Calculations

Graph 1 shows the raw pressure data taken from a Ford GTDi which will be used to calculate kinematics and forces. Piston position and velocity were then calculated and plotted. (Appendix A, B)



(Graph 1: Ford GTDi Cylinder Pressure vs Crank Angle)



(Graph 2: 6000RPM Piston Acceleration Components)

The piston deviates from simple harmonic motion due to second order acceleration which is an inertia caused by conrod angularity, occurring at twice the crankshaft speed. (Heywood, J. B., 2018)

Acceleration components were calculated with the equations shown below:

First Order:

$$R\omega^2 \cos(\theta)$$

Second Order:

$$\frac{R^2}{L} \omega^2 \cos(2\theta)$$

Combined:

$$R\omega^2 \cos(\theta) + \frac{R^2}{L} \omega^2 \cos(2\theta)$$

The pressure and combined acceleration could then be converted into forces using the following equations:

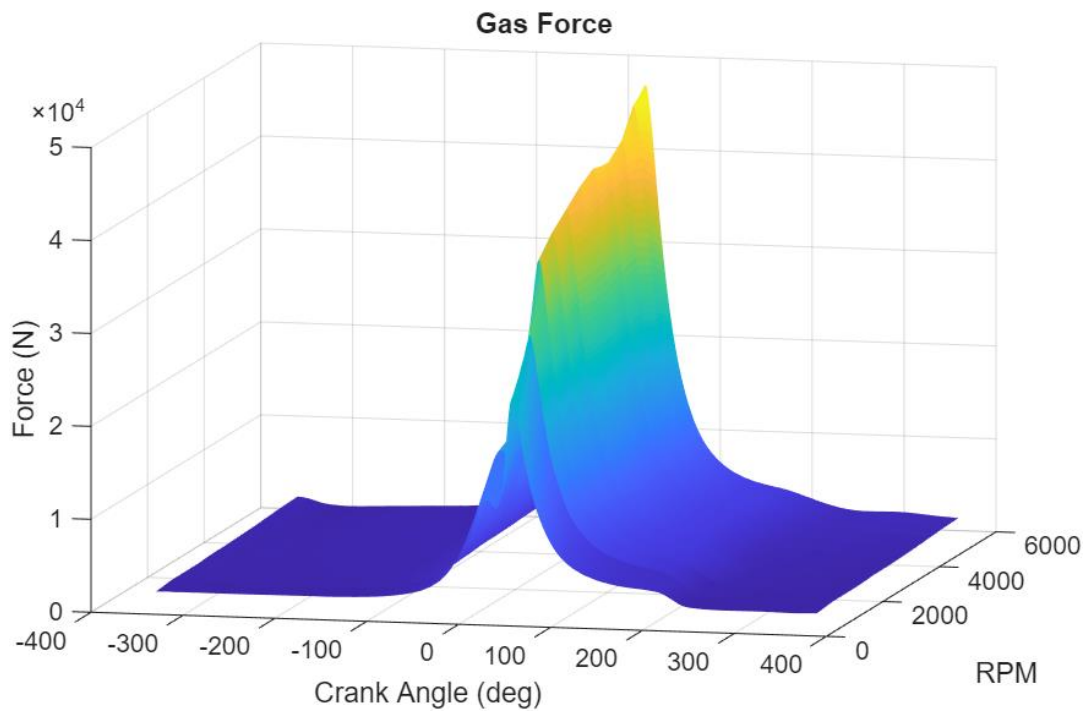
Gas Force:

$$Pressure (Pa) \times \frac{\pi \times Bore^2}{4}$$

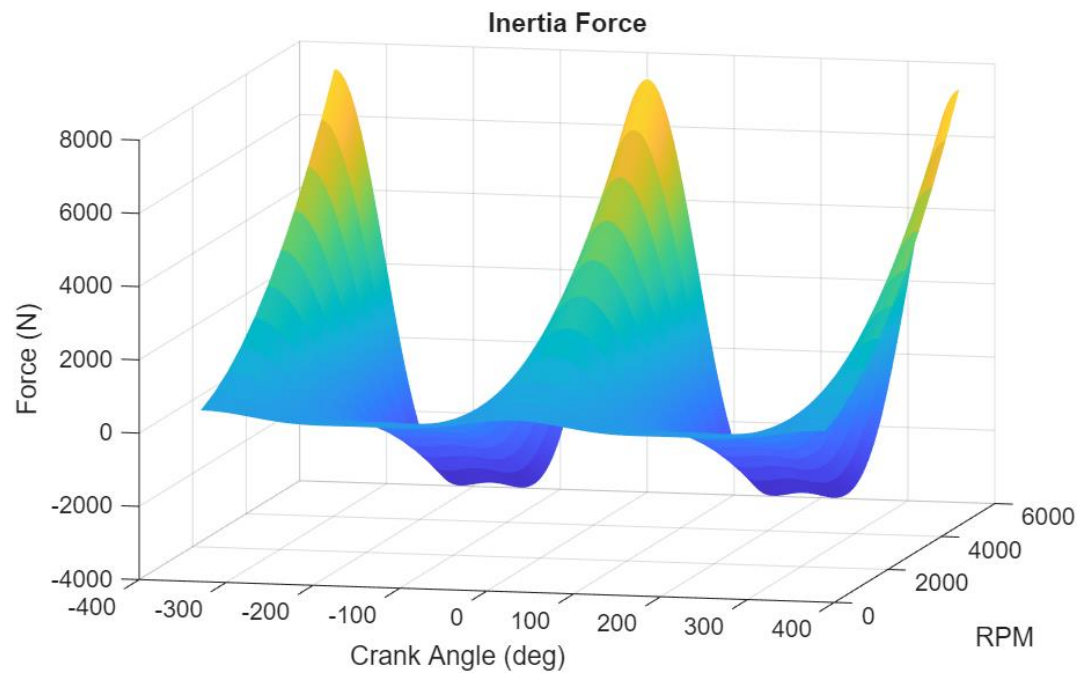
Inertia Force:

$$Acceleration (mm/s^2) \times Reciprocating Mass (Kg)$$

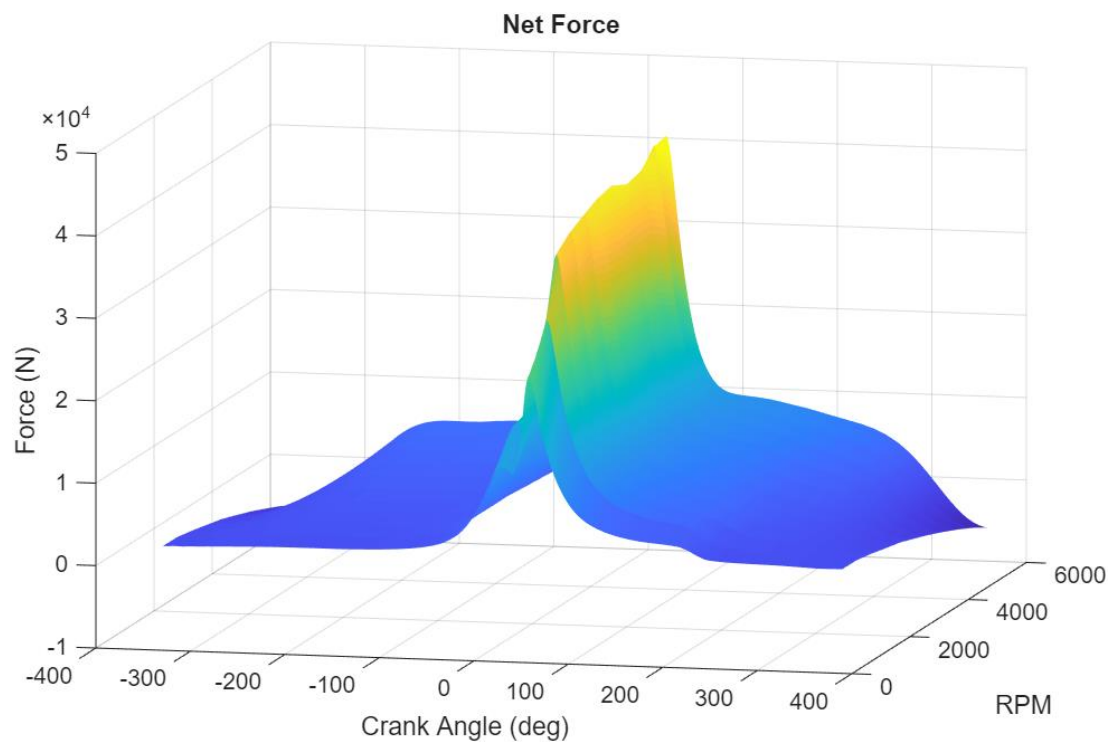
The graphs below display these forces separately and as net force.



(Graph 3: Original Limit Gas Force)



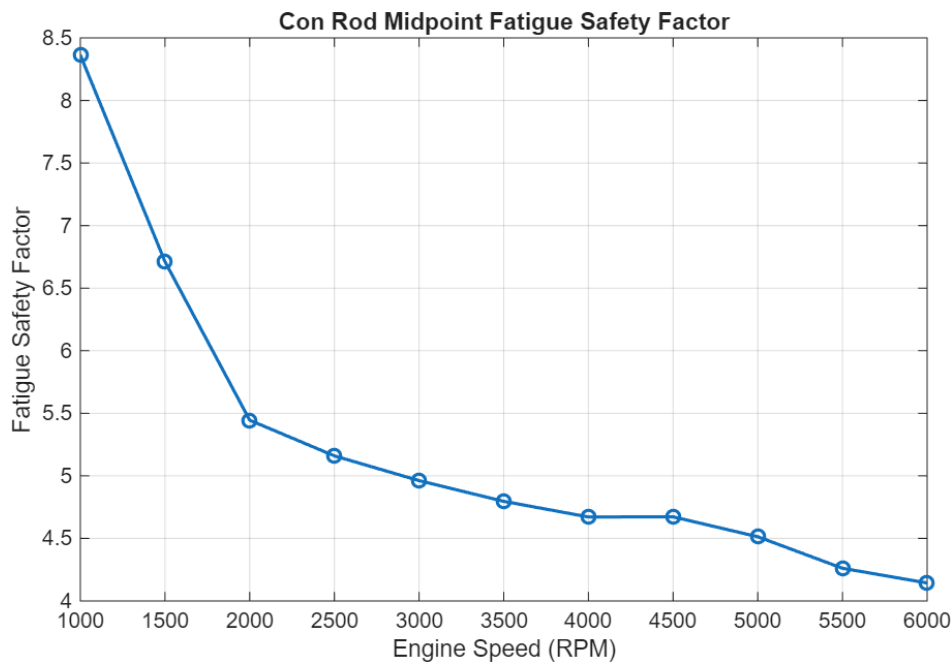
(Graph 4: Original Limit Inertia Force)



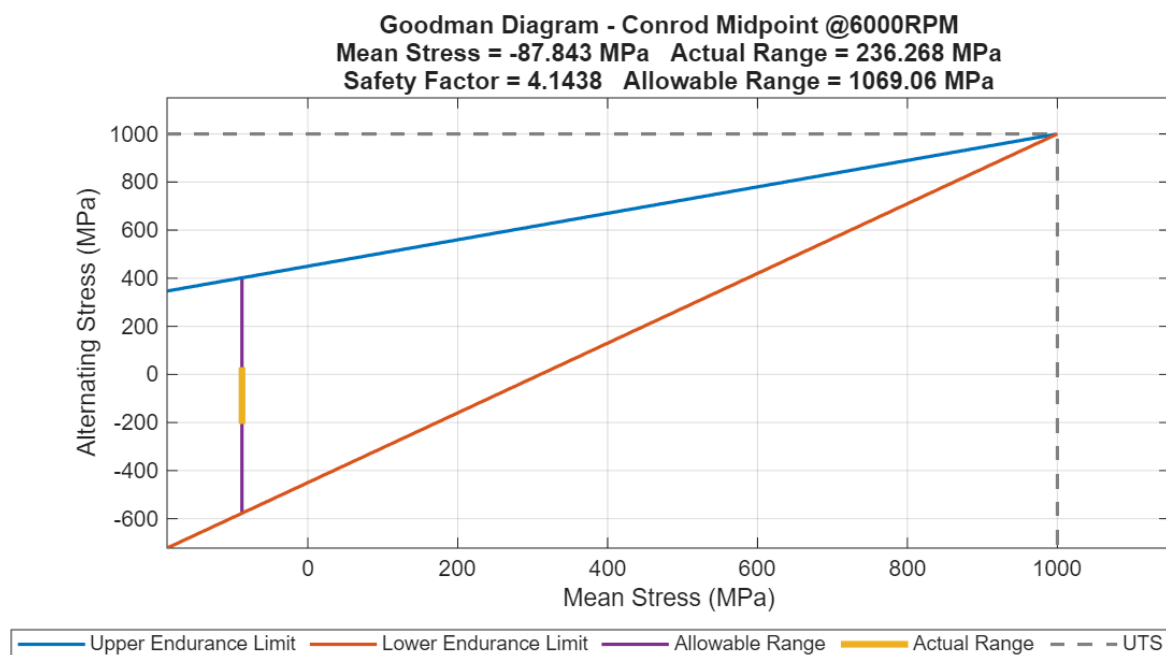
(Graph 5: Original Limit Net Force)

3 Component Fatigue Analysis at 6000RPM

3.1 Ford GTDi 6000RPM – Connecting Rod Midpoint



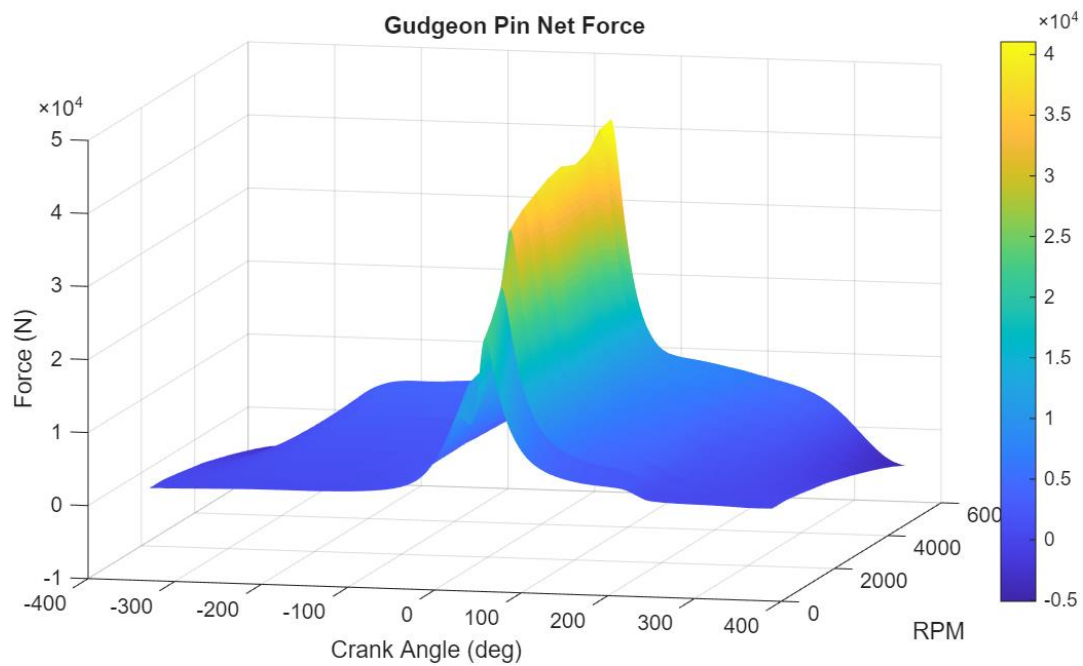
(Graph 6: Original Limit Midpoint Fatigue Safety Factor vs Engine Speed)



(Graph 7: 6000RPM Midpoint Goodman Diagram)

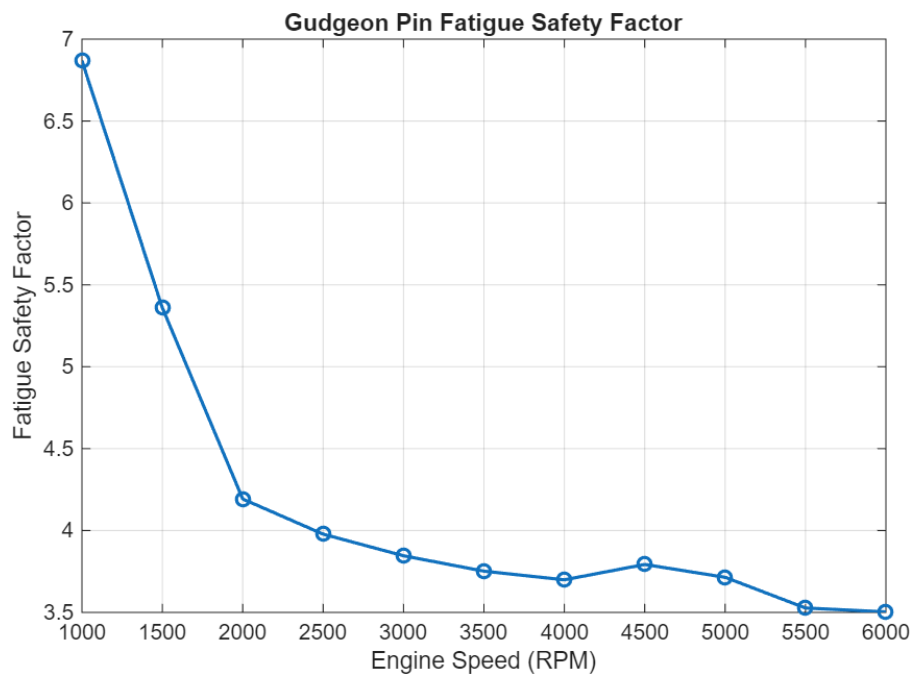
The Goodman diagram details the stress and fatigue safety factor (FSF) at 6000 revolutions per minute (RPM) which is the limit (highest forces) of this engine. With a FSF of 4.14, the midpoint is considered as operating well below its endurance limit. Whilst this could be built-in fatigue tolerance, the loads are predictable therefore, there could be scope for this component having structural weight reduction and currently is over-engineered. (Bin Z., Yongqi L., Lixia J., 2010).

3.2 Ford GTDi 6000RPM – Gudgeon Pin



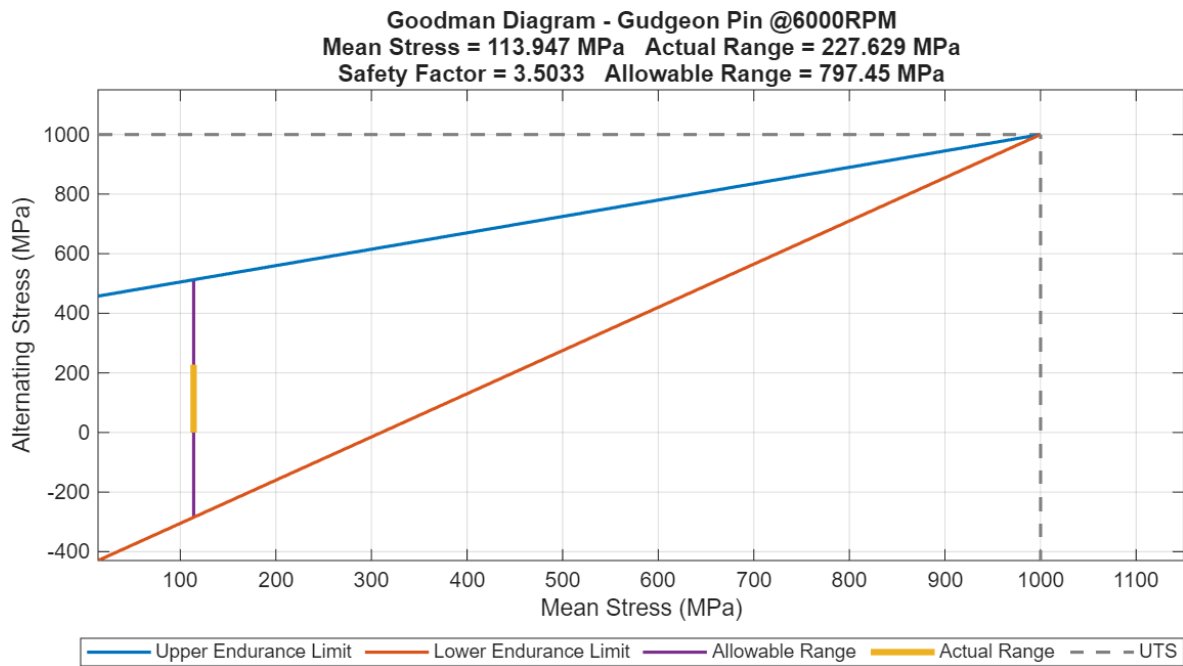
(Graph 8: Original Limit Gudgeon Pin Resultant Stress)

The gudgeon pin has bending and shear stress to take into consideration, these were calculated and plotted on graph 8 above as resultant stress.



(Graph 9: Original Limit Gudgeon Pin Fatigue Safety Factor vs Engine Speed)

Due to the extra stresses seen on the gudgeon pin, the FSF is slightly lower at 3.5 at 6000RPM. The FSF range is lower as the forces scale less with RPM in comparison to the midpoint.



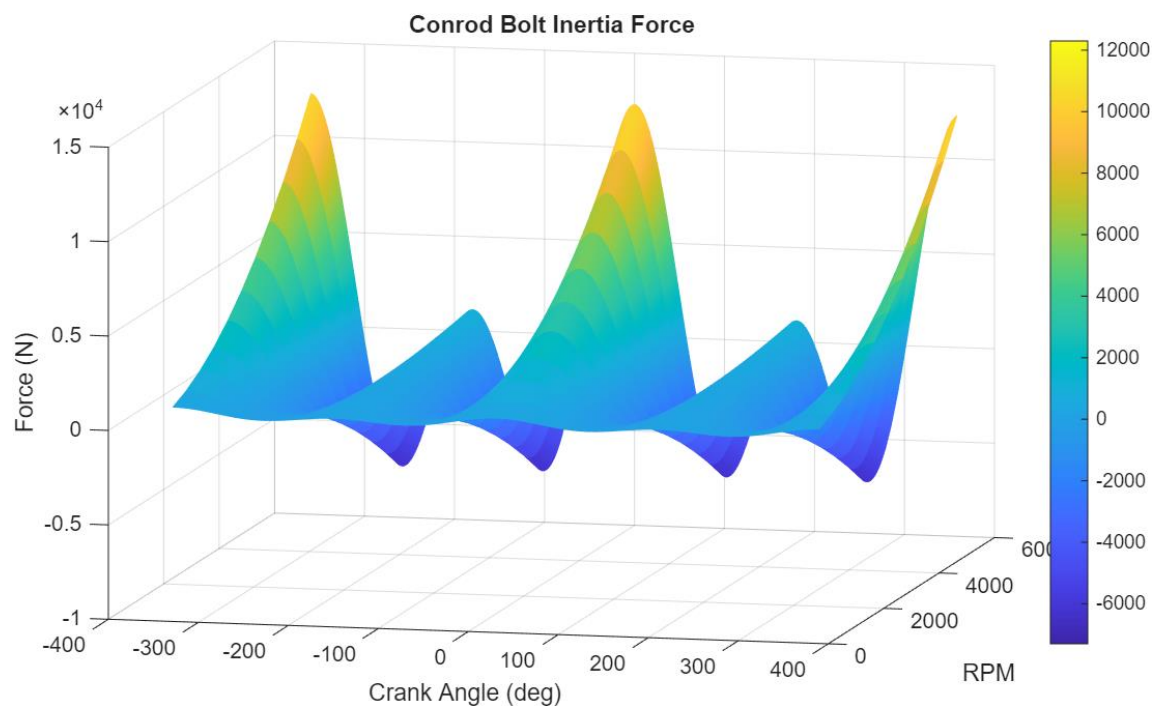
(Graph 10: 6000RPM Gudgeon Pin Goodman Range Diagram)

The Goodman diagram above details the conservative design, as the actual range sits well within the allowable range with a low mean stress through the component. This indicates that the component is slightly over-engineered, as the operating stress range remains significantly below the endurance limit. Geometry changes could be made without compromising fatigue performance. Research suggests a FSF in the range of 2.4 however, details the importance of temperature loads around the joint and piston, which weren't considered in this analytical model. (MAHLE GmbH, 2010)

3.3 Ford GTDi 6000RPM – Connecting Rod Bolt

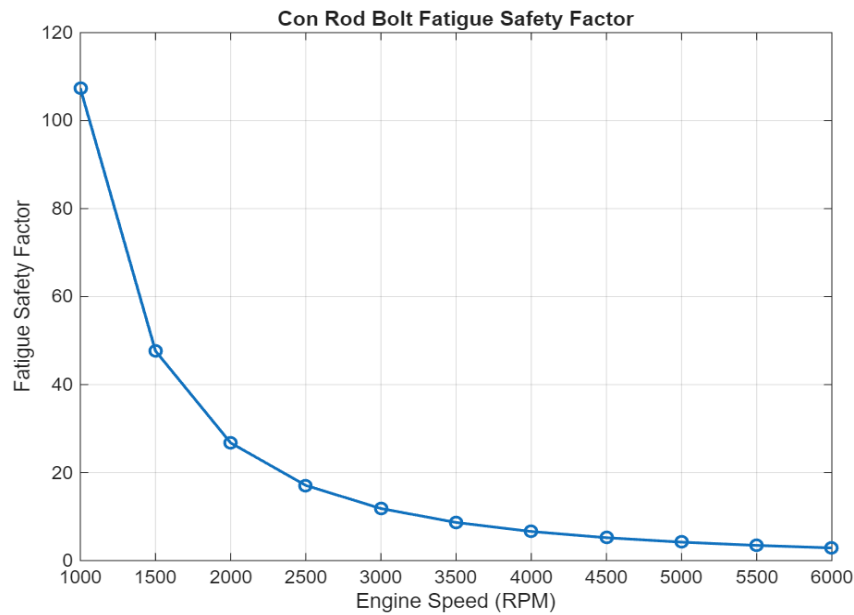
Conrod bolts have preload (axial stress) considerations which need to be accounted for during calculations. These have been included by using the assumption of when permanent fasteners are tightened to 90% of the yield strength (Childs, P., 2014), axial stress equals 60% of the ultimate tensile strength (UTS).

Once preload was converted to axial force, the tightening torque could be calculated using approximate frictional coefficient and radius values, outputting a tightening torque of 37.8Nm. This sits within an acceptable range for an M8 bolt, shown in Appendix C.



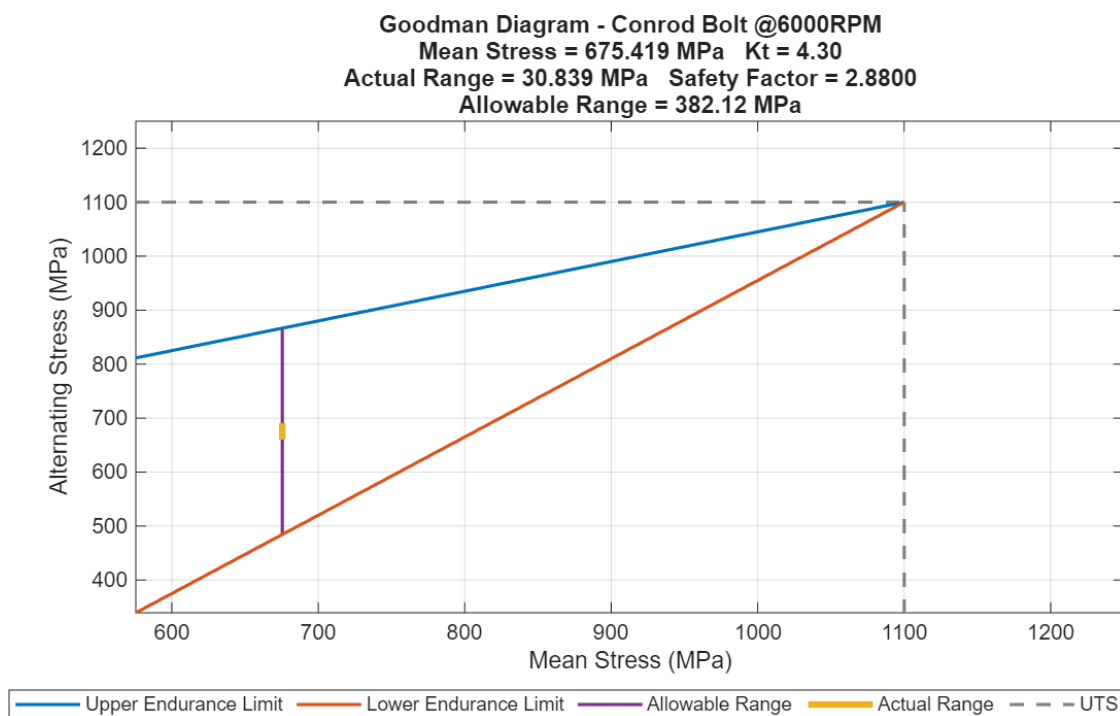
(Graph 11: Original Limit Bolt Inertia Force)

Graph 11 shows that the bolt is under slightly different inertia loading conditions determined by the motion of the crankshaft and the weight of rotational components.



(Graph 12: Original Limit Bolt Fatigue Safety Factor vs Engine Speed)

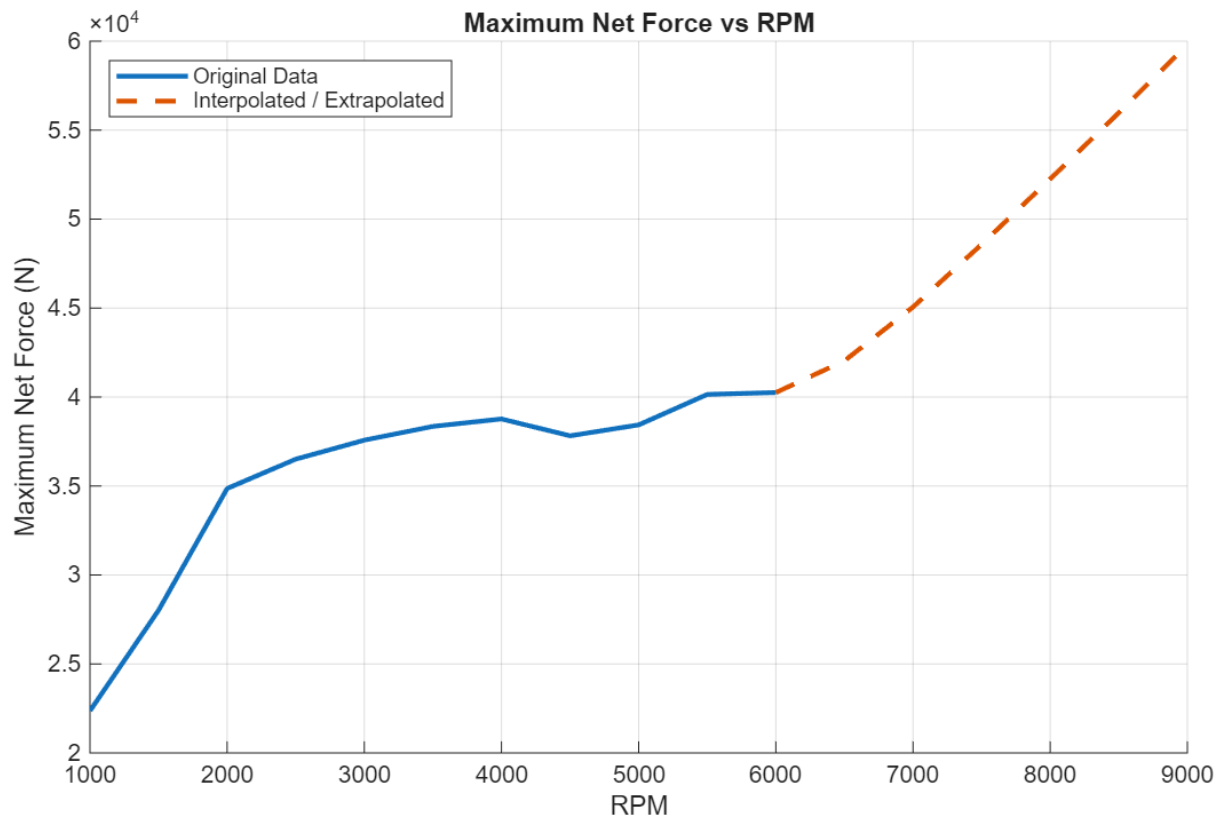
Due to thread design, stress concentration (K_t) builds up and should be considered within the operating conditions. This value was calculated as 4.3 based on the bolt threads using Roark's Formula (Appendix D).



(Graph 13: 6000RPM Bolt Goodman Range Diagram)

The Goodman diagram for the bolt highlights the relatively low variation in stress compared to the allowable range. This behaviour is expected, as most of the bolt stress arises from the initial preload, while dynamic engine loads introduce only small fluctuations about this mean stress. The resulting fatigue safety factor of 2.88 indicates that the bolt is operating within safe limits. This behaviour is consistent with bolted joint design principles, where preload is used to minimise stress variation and improve fatigue life. (Shigley J. E., Mischke C. R., Budynas R. G., 2004)

4 RPM and Force Extension

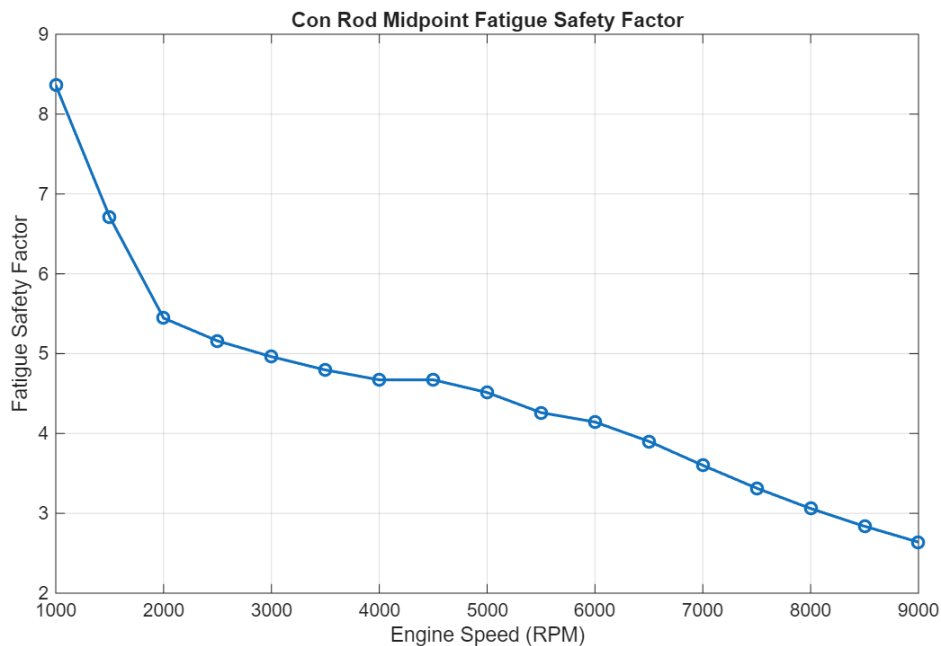


(Graph 14: Net Force Extension)

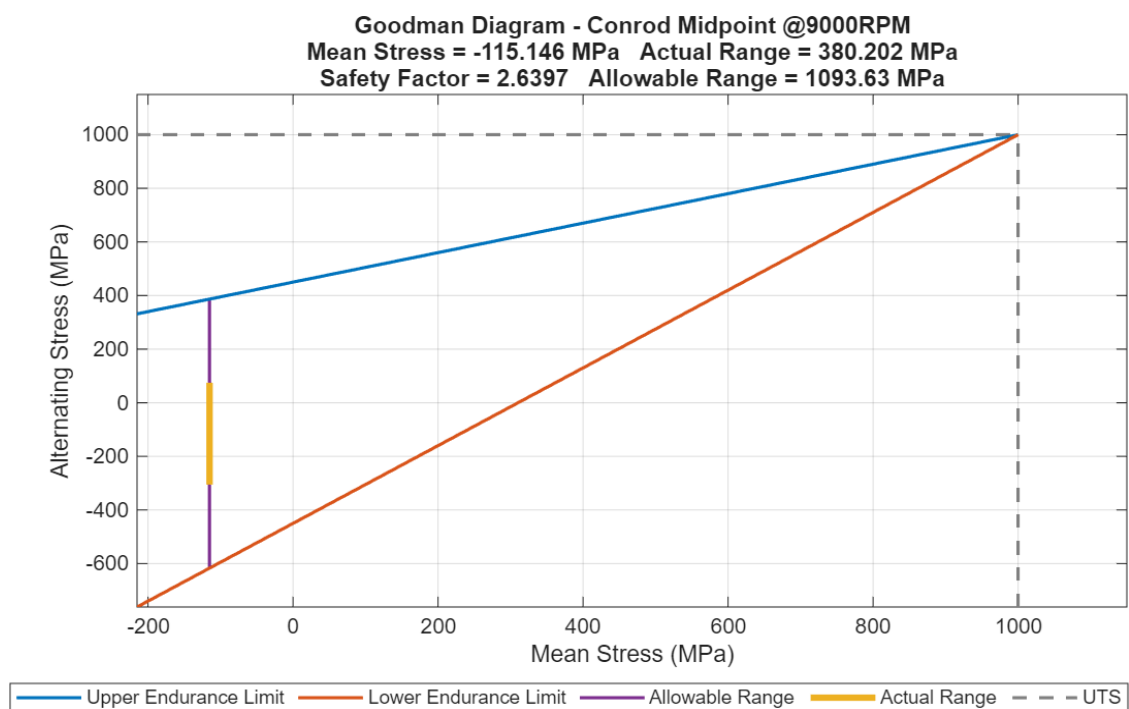
The gas and inertia forces were interpolated and extrapolated beyond the rpm limit of the Ford, then plotted as net force. As this is a prediction there will be a slight loss of accuracy compared to previous calculations but will approximate a new worst-case condition for the components at 9000RPM.

5 Component Fatigue Analysis at 9000RPM

5.1 Ford GTDi 9000RPM – Connecting Rod Midpoint



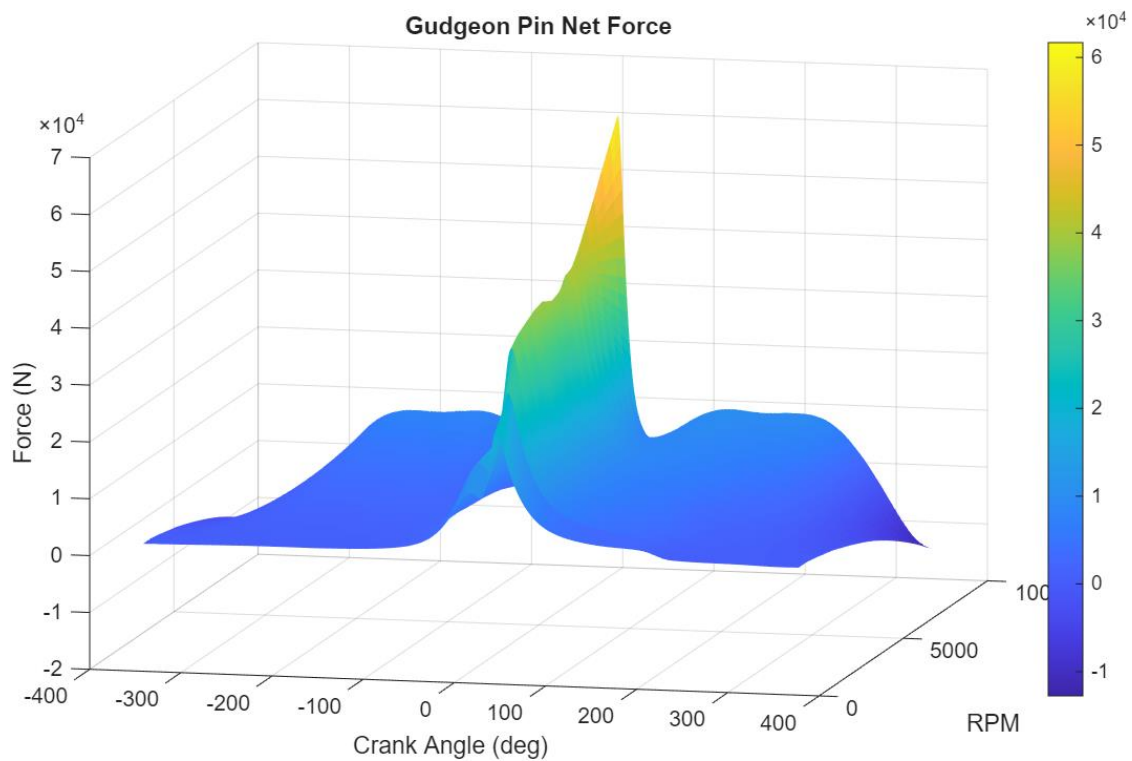
(Graph 16: Raised Limit Midpoint Fatigue Safety Factor vs Engine Speed)



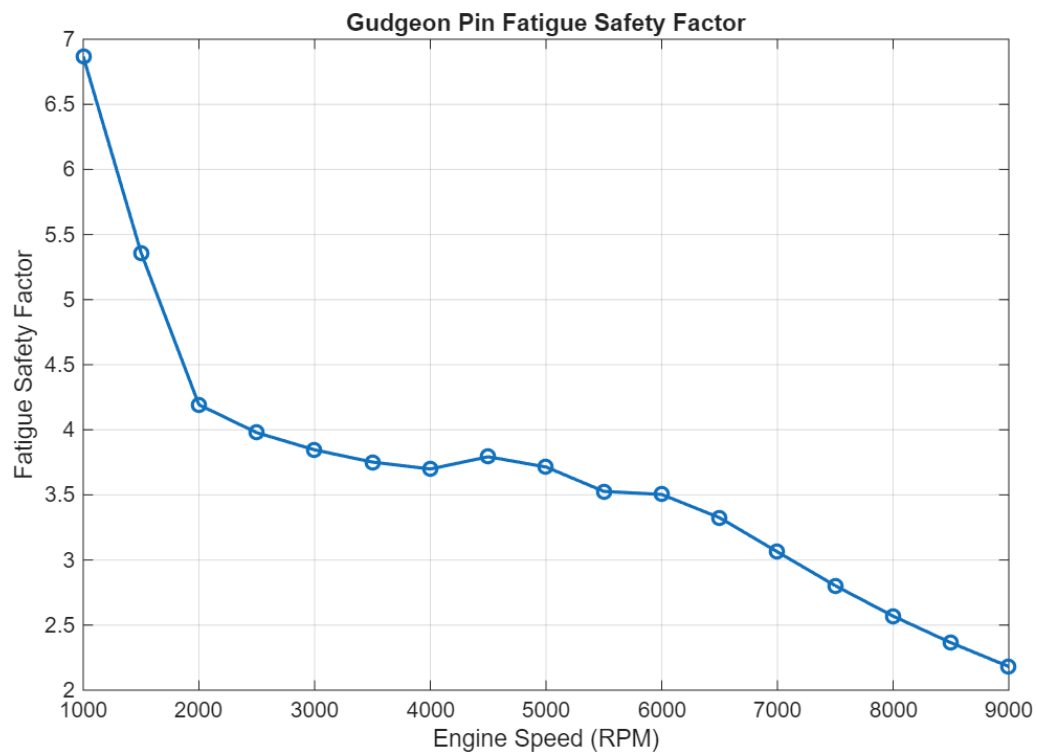
(Graph 15: 9000RPM Midpoint Goodman Range Diagram)

The Goodman diagram shows that at the raised limit, the midpoint is within a comfortable range, suggesting strong fatigue life. Therefore, the conrod could have slightly less conservative geometry and weight but still suit the load conditions at 6000RPM, with an acceptable safety margin.

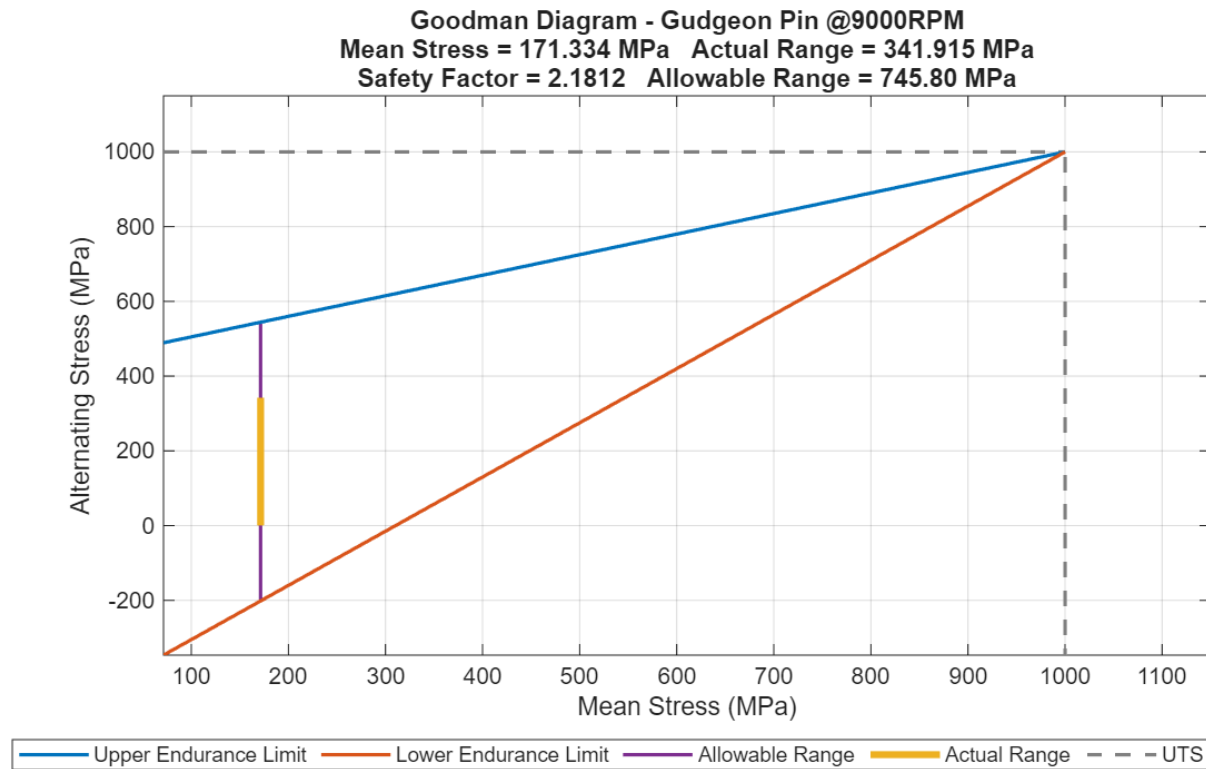
5.2 Ford GTDi 9000RPM – Gudgeon Pin



(Graph 17: Raised Limit Gudgeon Pin Net Force)



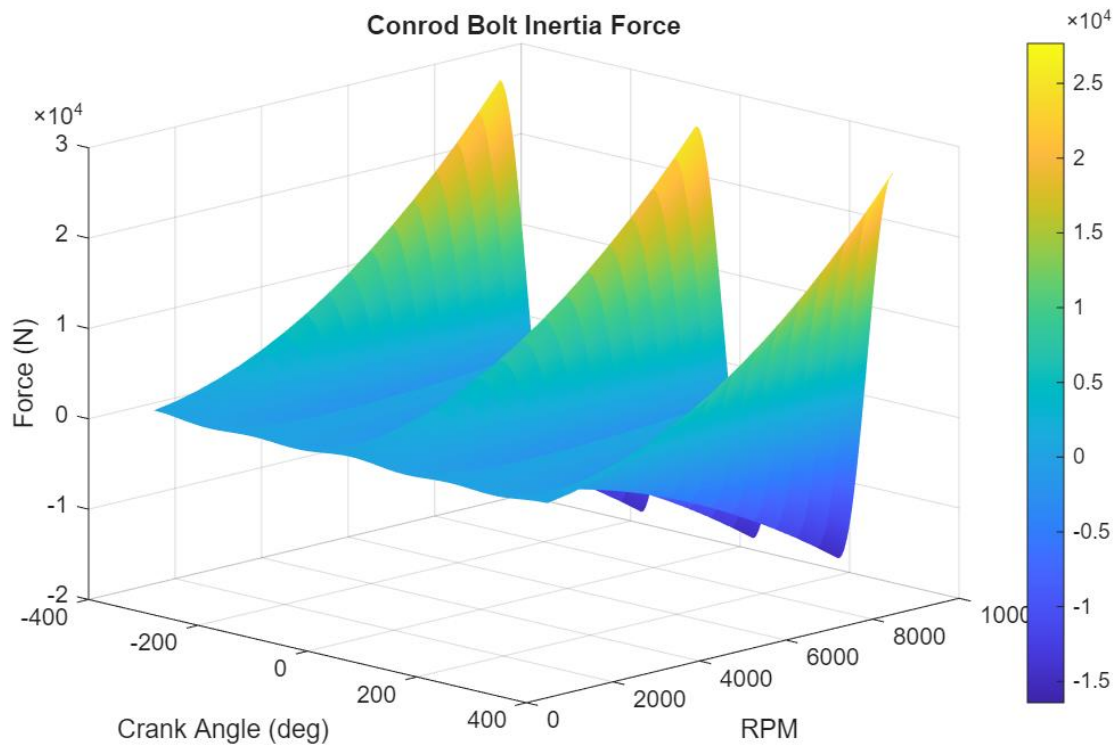
(Graph 18: Raised Limit Gudgeon Pin Fatigue Safety Factor vs Engine Speed)



(Graph 19: 9000RPM Gudgeon Pin Goodman Range Diagram)

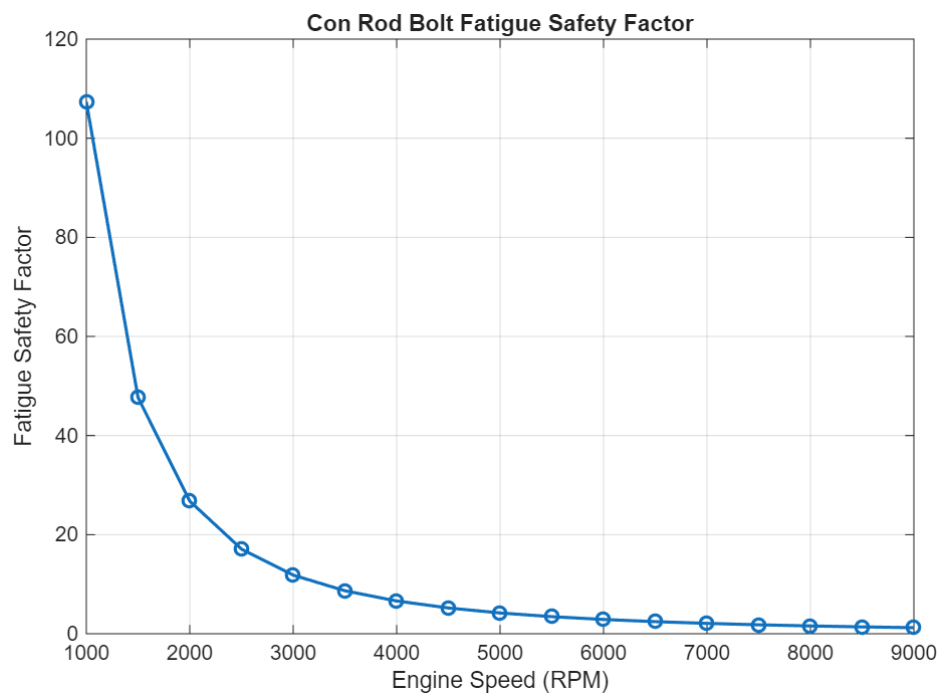
The actual range is a larger proportion of the allowable range for the gudgeon pin at 9000RPM, shown above. The safety factor drops considerably but is still within the range that there is low probability of failure. This follows the previous conclusion that the design of the pin is conservative and there is room for small design changes within a practical application margin.

5.3 Ford GTDi 9000RPM – Connecting Rod Bolt

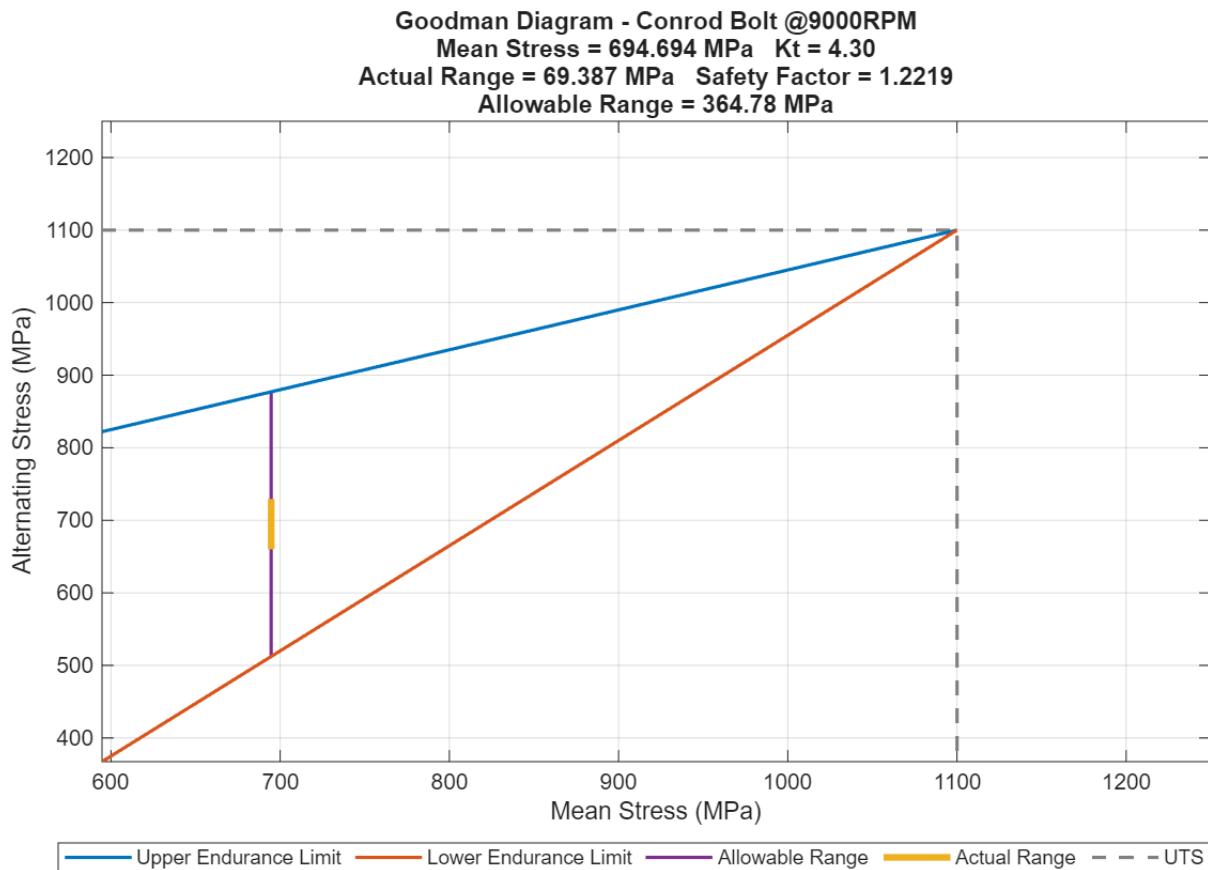


(Graph 20: Raised Limit Bolt Inertia Force)

Graph 20 shows the forces at 9000RPM on the conrod bolt increase by over double compared to 6000RPM. This is because inertia forces relate to piston acceleration, which increases with the square of piston velocity. (Heywood, J. B. 2018)



(Graph 21: Raised Limit Bolt Fatigue Safety Factor vs Engine Speed)



(Graph 22: 9000RPM Bolt Goodman Range Diagram)

The range diagram above shows the FSF value drops to 1.22 at 9000RPM. Whilst this doesn't necessarily indicate immediate failure, it is close to the limit and a small variation in stress would cause failure. There are also limitations within the calculations which, if accounted for, could cause the bolt to fail in a practical application.

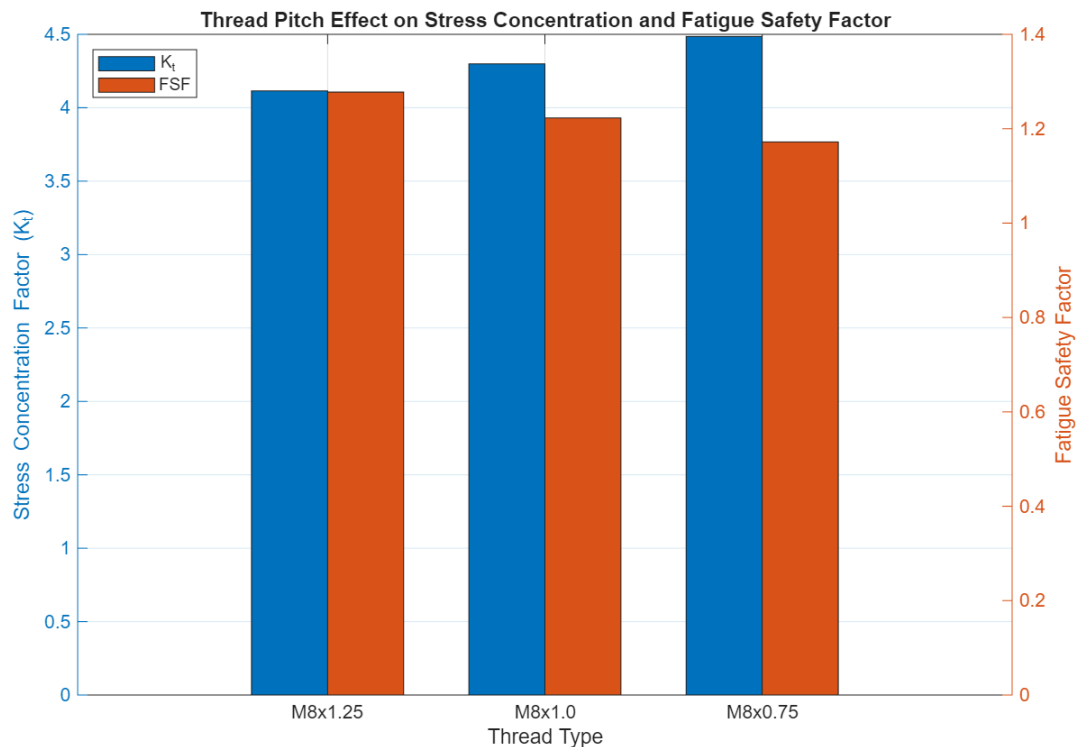
Limitations include the lack of a temperature condition that the bolt may be operating under, which likely results in over-estimated FSF, as elevated temperatures may reduce material strength and accelerate fatigue damage. Another limitation is that the stress concentration was calculated strictly for the threads, which is where most of the stress is, but neglects manufacturing imperfection stress concentrators or geometry changes such as the shank to the head. (Hearn, E. J. 1997)

6 Connecting Rod Bolt Redesign

The bolt will need redesigning to meet the 9000RPM conditions, with a similar FSF as the original bolt at the original limit. The following graphs and comparisons will be strictly at 9000RPM conditions.

6.1 Bolt Geometry Redesign

6.1.1 Thread Pitch

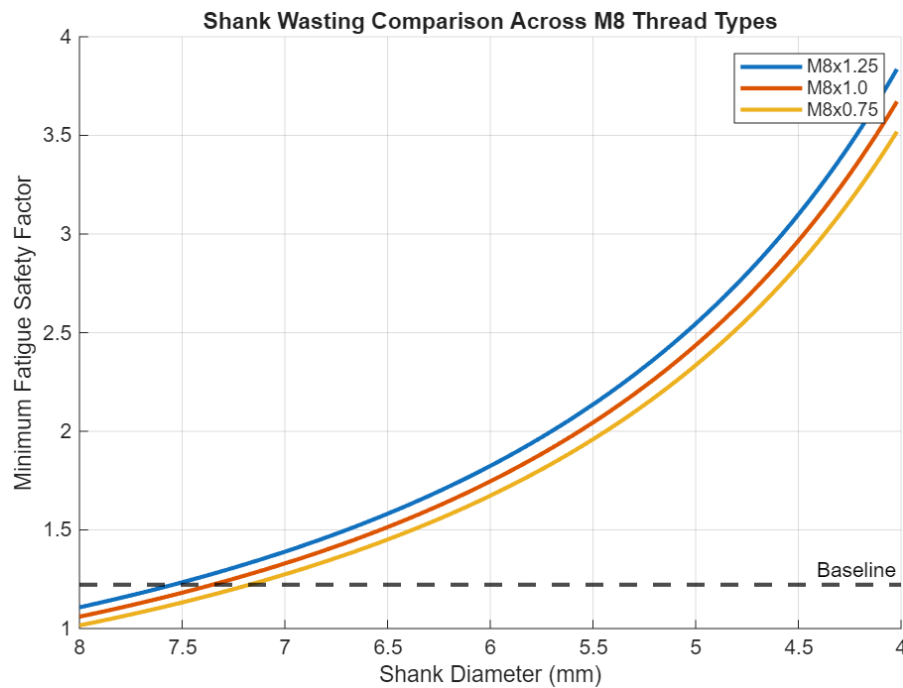


(Graph 23: Thread Pitch Stress Concentration and Fatigue Safety Factor)

To begin the redesign process, the thread pitch was adjusted to reduce stress concentration and improve FSF. As given in the specification, the bolt used for this engine is an M8x1.0 thread type and as shown by the graph above, performed better than a finer thread type (M8x0.75). However, the M8x1.25 bolt provided the lowest K_t therefore marginally improving the FSF to 1.27.

Whilst this proves a courser thread would be best for this application, all thread types will be evaluated in each of the upcoming design changes.

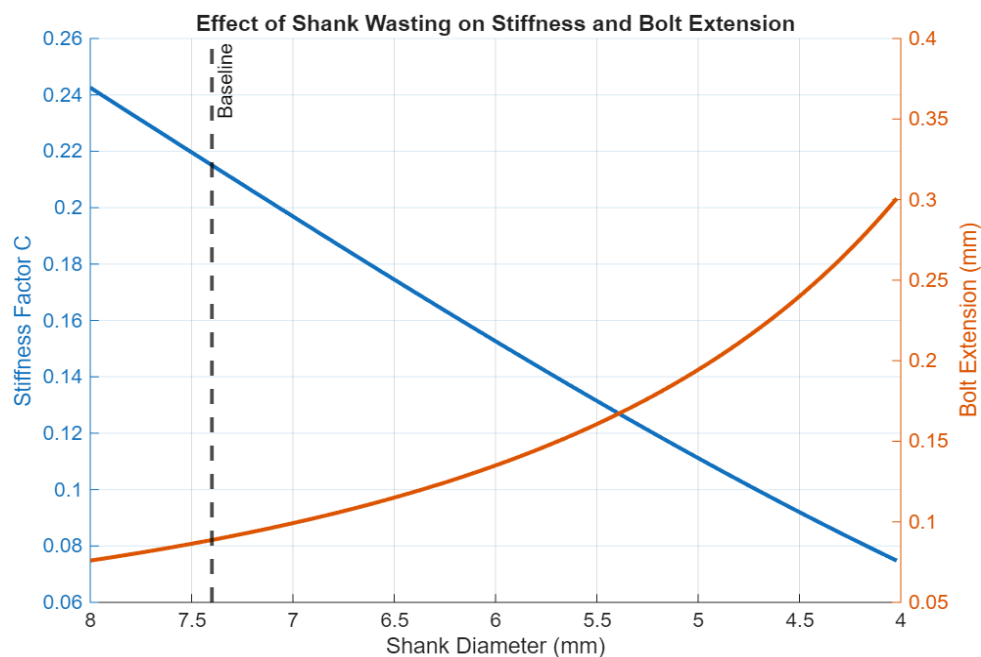
6.1.2 Shank Wasting



(Graph 24: Shank Wasting Fatigue Safety Factor)

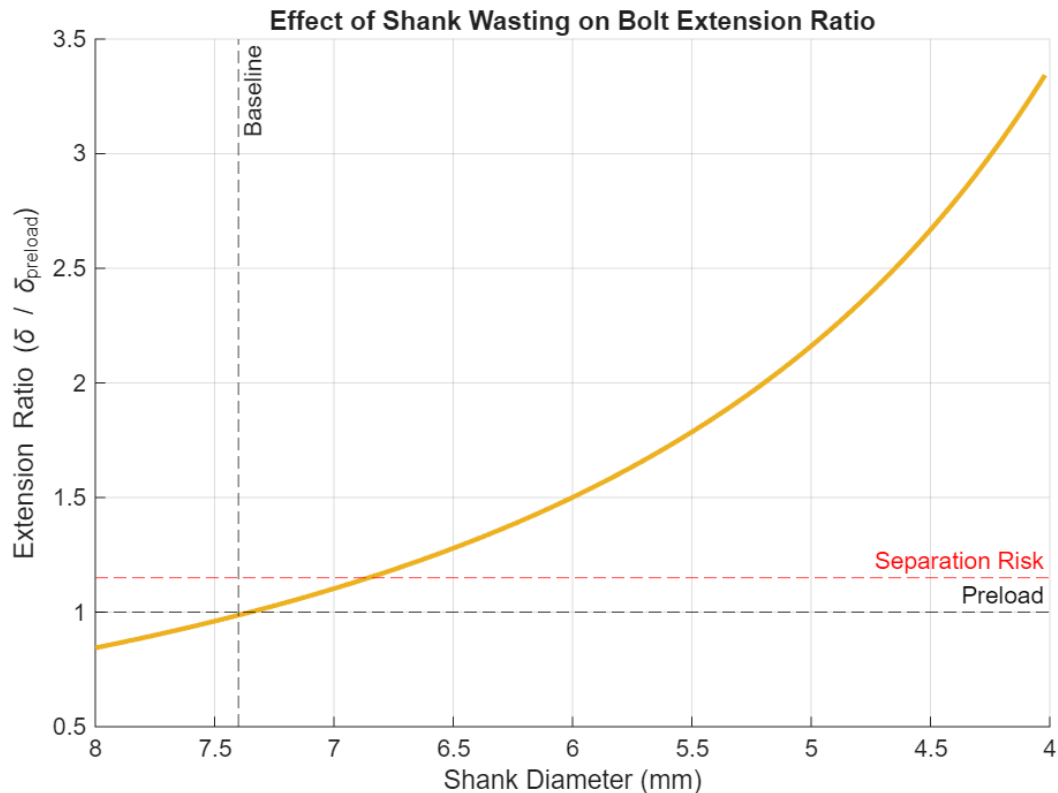
Shank wasting is considered as a useful method for improving FSF through a reduction in bolt stiffness between the threads and the head (equation Appendix E). This allows the bolt to extend more to reduce the stress across the main section of the bolt. The results indicate that reducing shank diameter improves fatigue performance, with an uptrend that would suggest wasting even smaller than 4mm would provide better performance.

However, graph 25 below shows the reduction in stiffness heavily affecting the bolt extension, to the point where the cap could be subject to clamp separation from the crank. (Equations Appendix F)



(Graph 25: Shank Wasting Stiffness and Bolt Extension)

Bolts have a preload extension that shouldn't be largely exceeded, otherwise joint separation will occur, labelled the extension ratio. (Barrett, R. T., 1990)

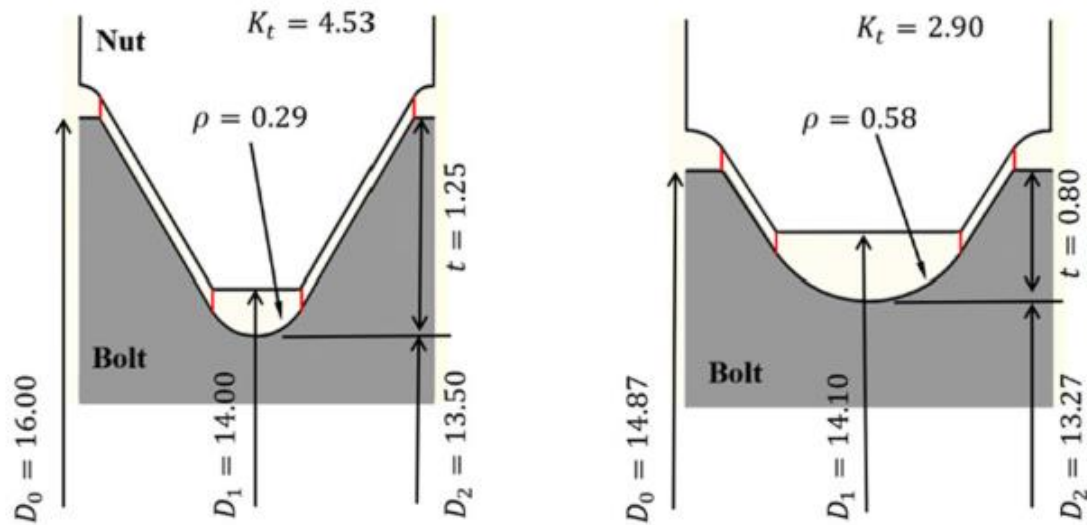


(Graph 26: Shank Wasting Extension Ratio)

A ratio of 1.15 was considered as at risk of separation because once this is passed the reduction in compression becomes significant enough that joint separation may occur under loading. This correlates to around 6.9mm shank diameter, showing that wasting beyond this may improve bolt fatigue but could reduce clamp force leading to joint failure.

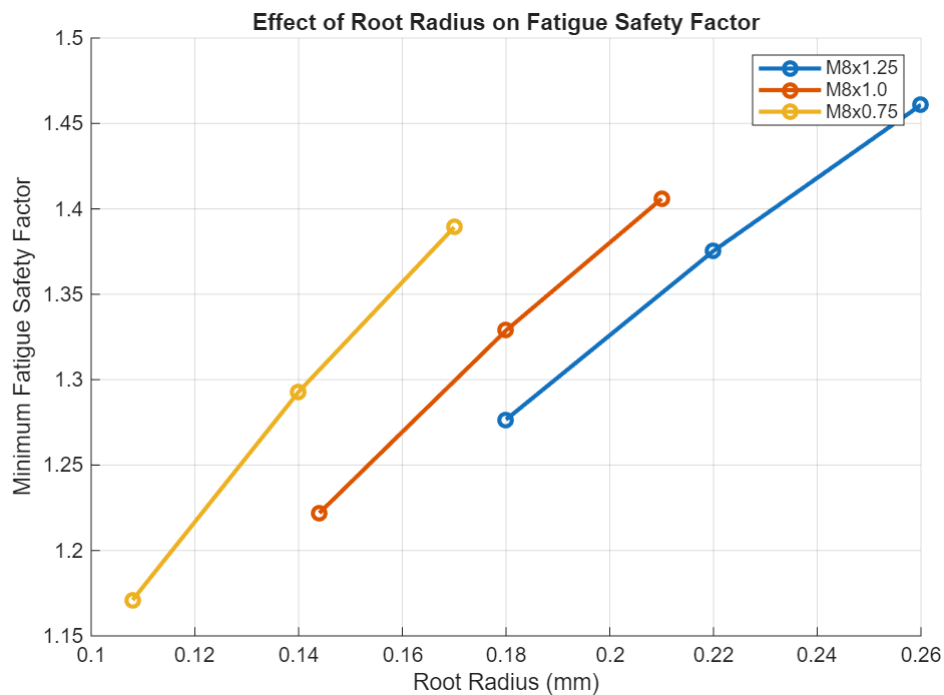
Based off this graph, the optimal chosen value of shank diameter is 7mm because it provides an increase in fatigue but avoids separation risk by a safe margin at an extension ratio of 1.1.

6.1.3 Root Radius Enlargement



(Figure 1: Root Radius Enlargement Affecting K_t (Noda, N.-A. et al. 2021))

Increasing the root radius has the effect of reducing the stress concentration between the conrod tapped threads and the female bolt threads due to reduced sharpness of the contact.

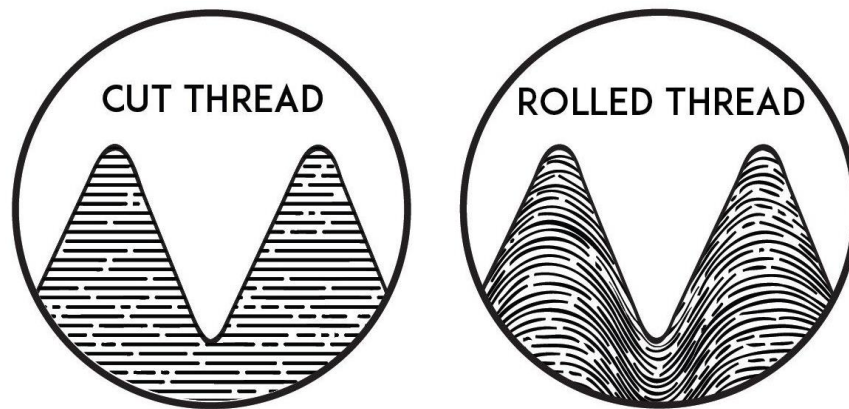


(Graph 27: Root Radius Fatigue Safety Factor)

Each thread type has a slightly different range of radius change due to thread geometry. (Machining Doctor, 2016)

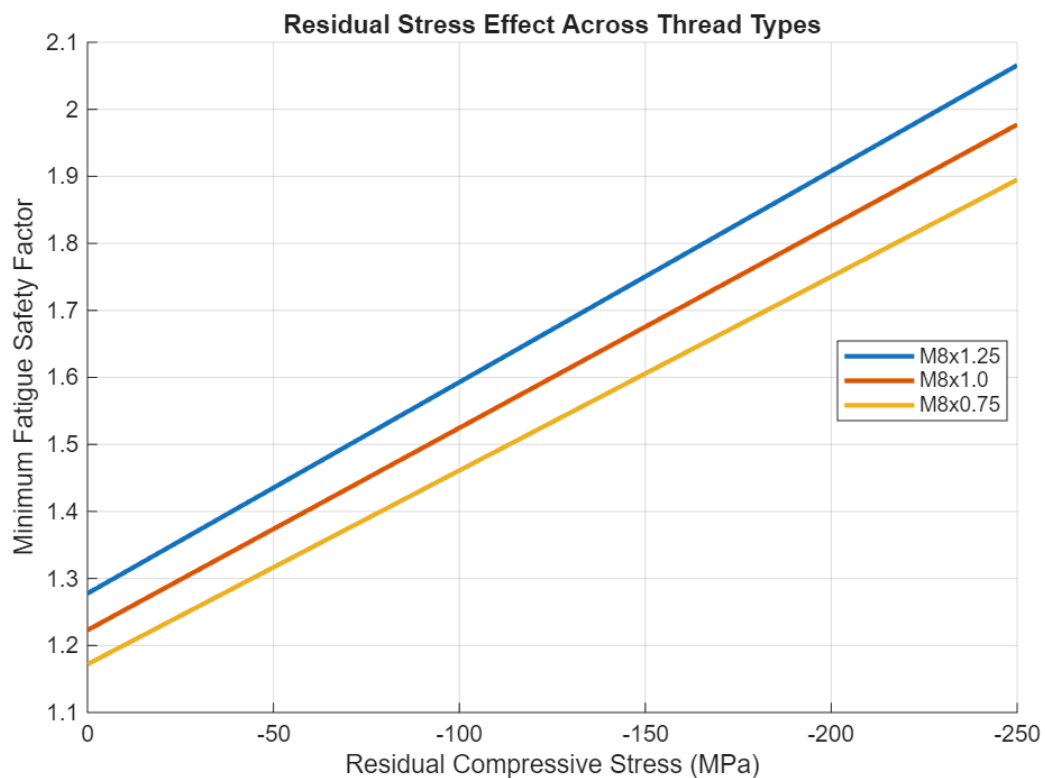
M8x0.75 improved the FSF the most across its range but M8x1.25 extended to the highest FSF of this design alteration at 1.46.

6.1.4 Thread Rolling



(Figure 2: Cut vs Rolled Thread Grain Structure (Rolled Threads Unlimited, 2024))

Rolled threads introduce a compressive stress during manufacturing which creates a tighter grain structure around the detailed area of threads, shown in the figure above.



(Graph 28: Residual Compressive Stress Fatigue Safety Factor)

Compressive stress was added to simulate the effect of rolled threads. A maximum of -250MPa of stress was used to sit in a comfortable range of realistic compressive stress created by rolling threads. (Unglaub, J., Wimpory, R.C. and Hensel, J. 2022)

The percentage increase in FSF values remained identical across all thread types at around 61% improvement which is within expected fatigue strength increase (Rolled Threads Unlimited, 2026).

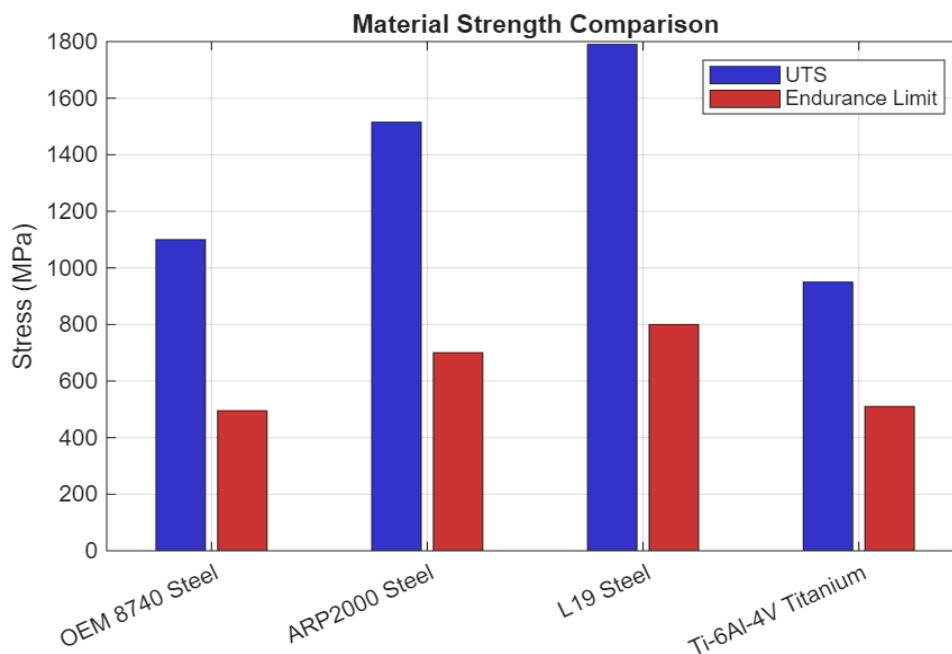
6.2 Bolt Material Redesign

(Table 1: Redesign Materials)

Material	Elastic Modulus (GPa)	Ultimate Tensile Strength (MPa)	Yield Strength (MPa)	Endurance Limit (MPa)	Density (Kg/m ³)
OEM (8740 Chrome Moly)	200	1100	990	495	7850
ARP 2000 Steel	200	1515	1380	700	7850
L19 Steel	205	1790	1700	800	7850
Ti-6Al-4V Titanium	114	950	880	510	4430

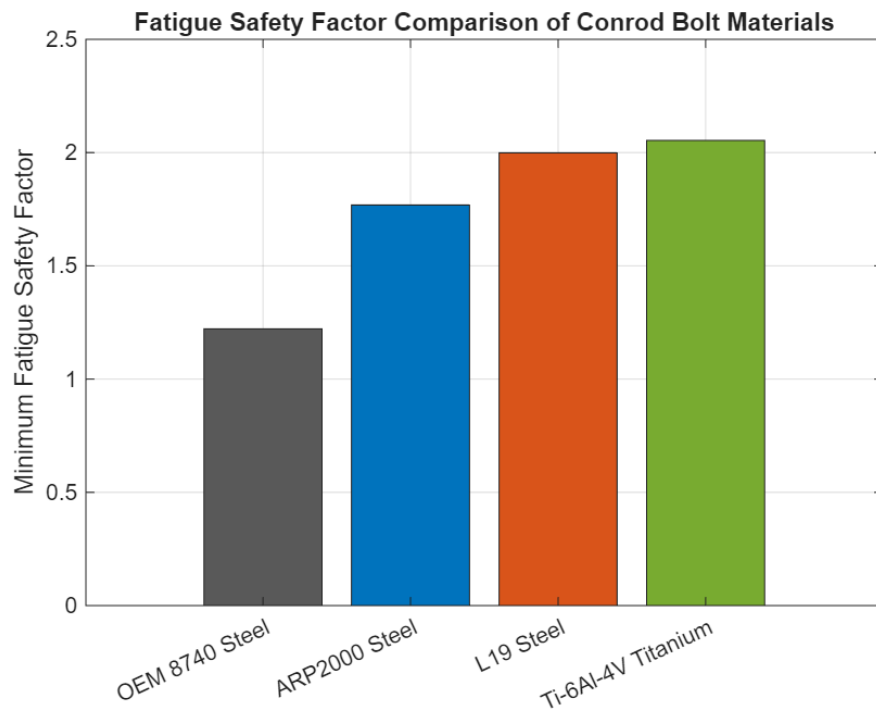
The material values in the engine specification were within a similar range as original equipment manufacturer (OEM) 8740 Chrome Moly Steel, a commonly used conrod bolt material, so the baseline will be classed as this material moving forward. Properties for the other conrod bolt materials were obtained from ARP fastener technical specifications. (ARP, 2026)

6.2.1 Material Fatigue & Strength



(Graph 29: Material Strength Properties Comparison)

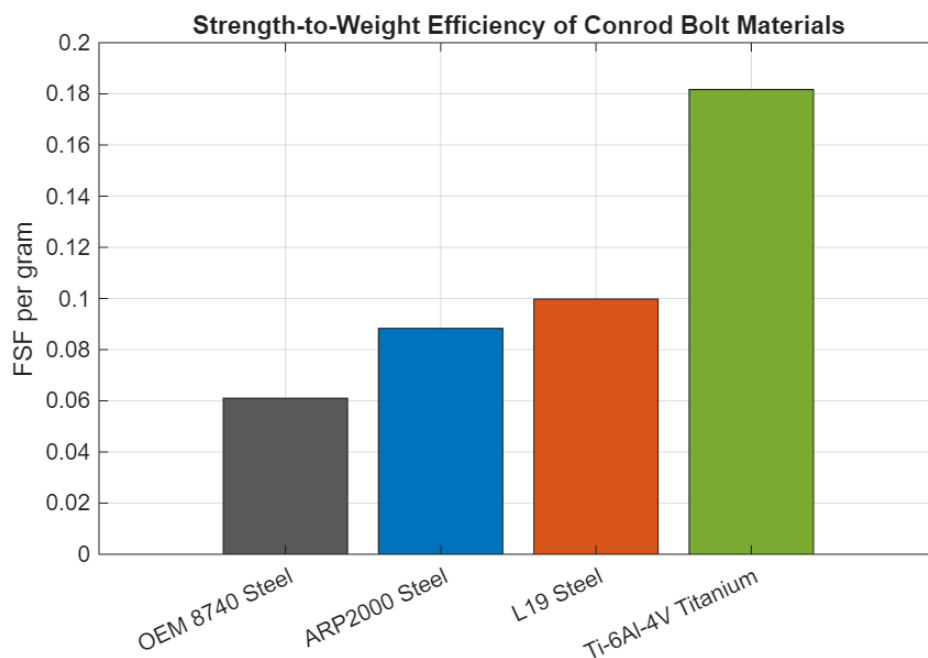
As shown by graph 29, the L19 Steel has the strongest tensile strength properties compared to the OEM and ARP Steel. Titanium shows as the lowest tensile strength and endurance limit however due to the low elastic modulus it is comparable to the steel in this application due to stiffness being an important factor in bolt design.



(Graph 30: Recalculated Fatigue Safety Factor)

FSF was recalculated, showing the benefits of the high strength L19 and the more elastic titanium deflecting the load.

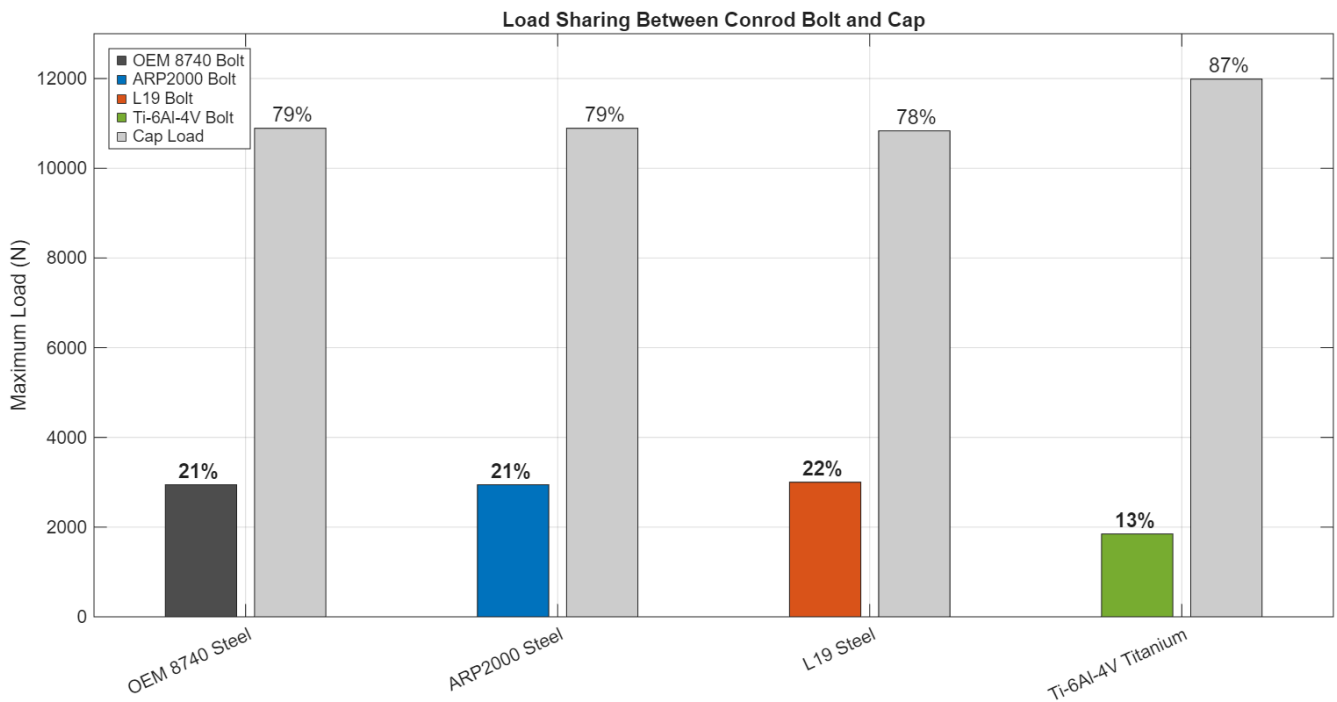
6.2.2 Material Weight



(Graph 31: Strength-to-Weight Efficiency)

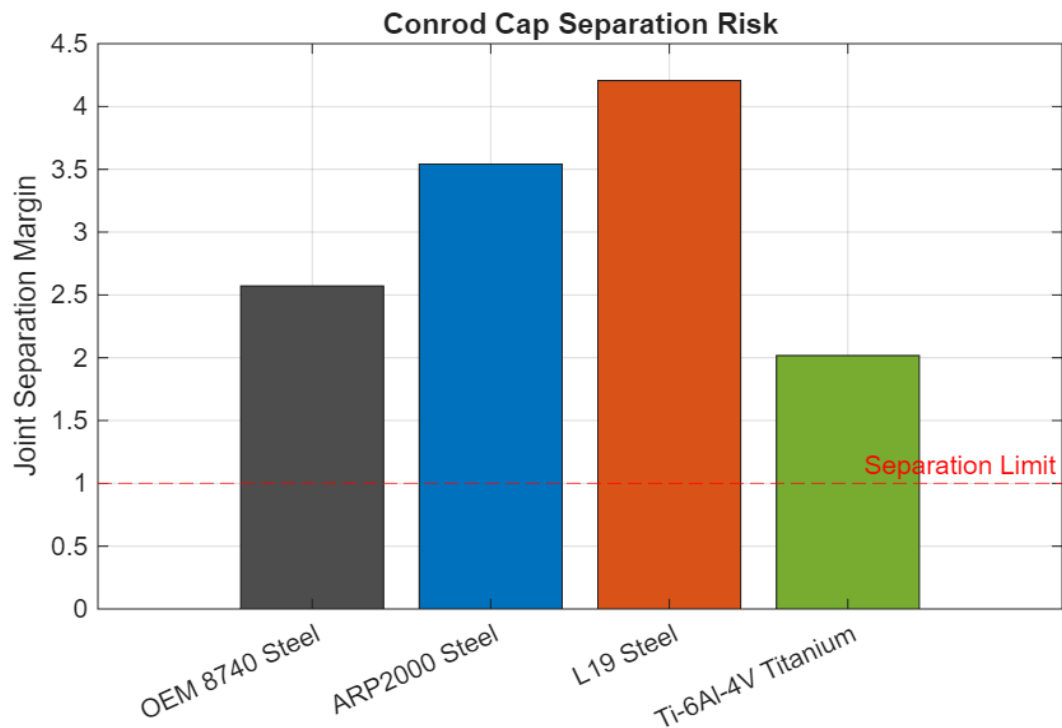
As expected, the low-density titanium provided the best strength-to-weight by a considerable margin. However, L19 was also an increase of 0.4 FSF per gram over the OEM material as the steel materials have identical density, despite increases in strength.

6.2.3 Material Stiffness & Load Variation



(Graph 32: Conrod Bolt and Cap Load Sharing)

Due to the reduced stiffness, the titanium bolt load falls by nearly 50% compared to the original. This would provide much better load resistance for the bolt however, the crankshaft to cap joint begins to separate and become a concern at low bolt stiffness. The reduced bolt load is transferred to the cap which peaks at 87% for the titanium.



(Graph 33: Conrod Cap Separation Risk)

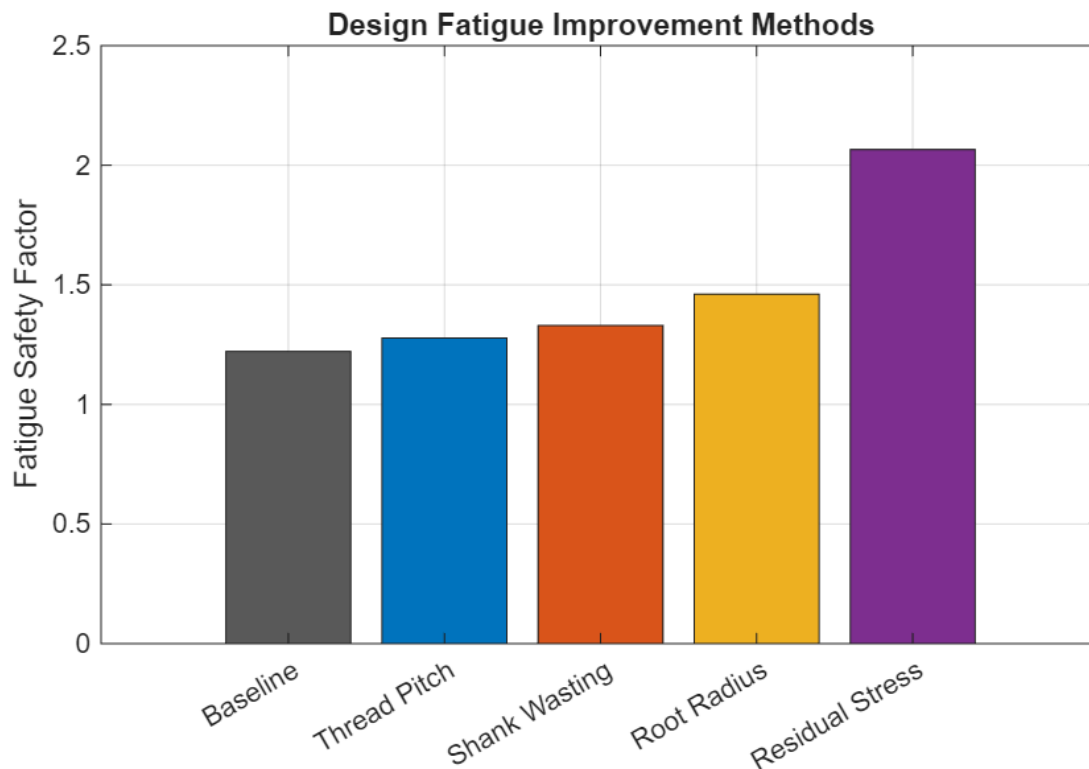
The separation margin plotted for each of the materials shows titanium is the closest to joint separation overall however, still above the limit. The lower cap load through the steels tested reduces risk largely.

Separation risk was evaluated using the separation margin, calculated as the ratio of separation load to external load (equations Appendix G). The separation limit of 1 is the point where the separation force exceeds the external load, with a range typically between 2.0 to 2.5 (Manning, J. 2013).

6.3 Optimised Bolt Comparison

6.3.1 Optimum Design

To find the optimum design changes, the best FSF for each was compared to the baseline value. For every method the highest value was taken, apart from shank wasting where there was a compromise to avoid excessive bolt extension.



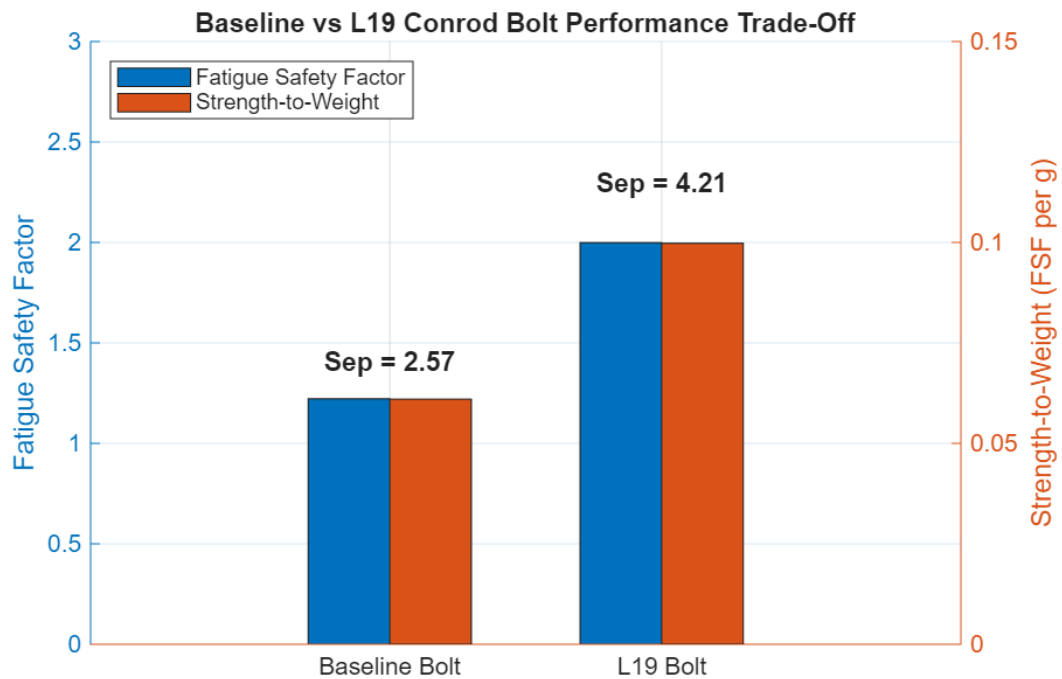
(Graph 34: Design Focused Fatigue Improvements)

To incorporate the design changes, calculation values need to be extracted to recalculate the FSF of the optimum bolt. The corresponding values include improved K_t , improved stiffness and residual stress introduction.

(Table 2: Optimal Calculation Values)

Variable	Optimal Value
Thread Pitch (mm)	1.25
Thread K_t	4.1147
Shank Diameter (mm)	7
Bolt Stiffness	0.1969
Root Radius (mm)	0.26
Root K_t	3.5986
Residual Compressive Stress (MPa)	-250

6.3.2 Optimum Material



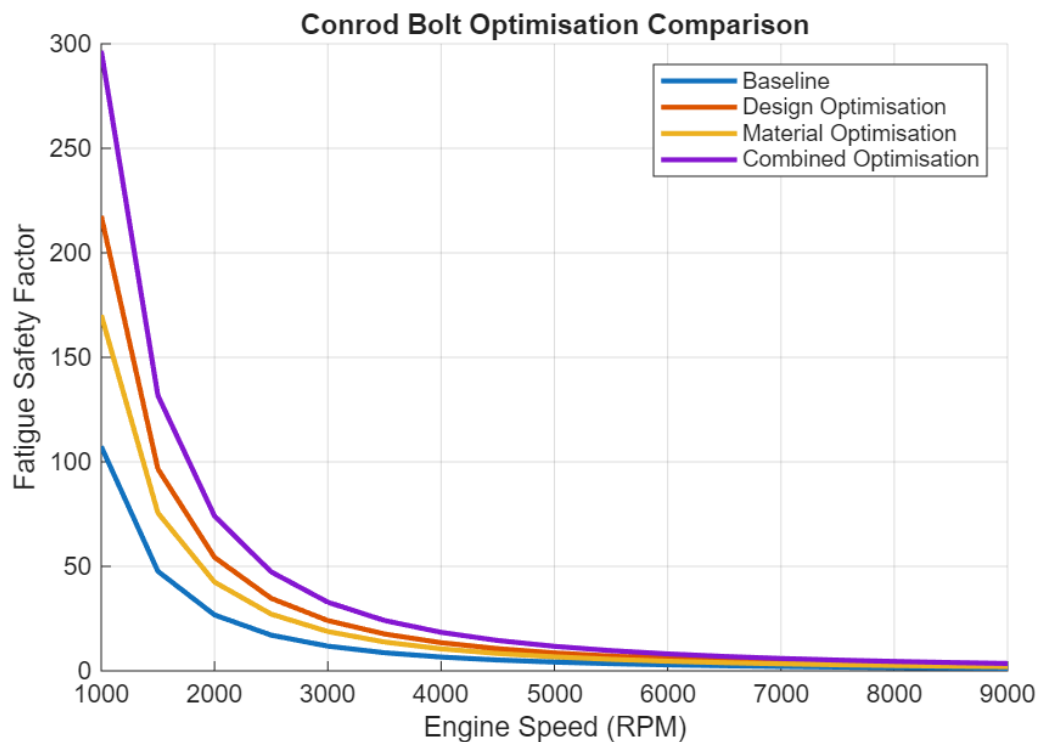
(Graph 35: Material Focused Improvements)

Graph 35 shows the chosen optimal material compared with the baseline for FSF, strength-to-weight and joint separation factor (Sep). The reason for choosing L19 steel was due to a balance of increased fatigue life, good strength-to-weight and low risk of cap separation.

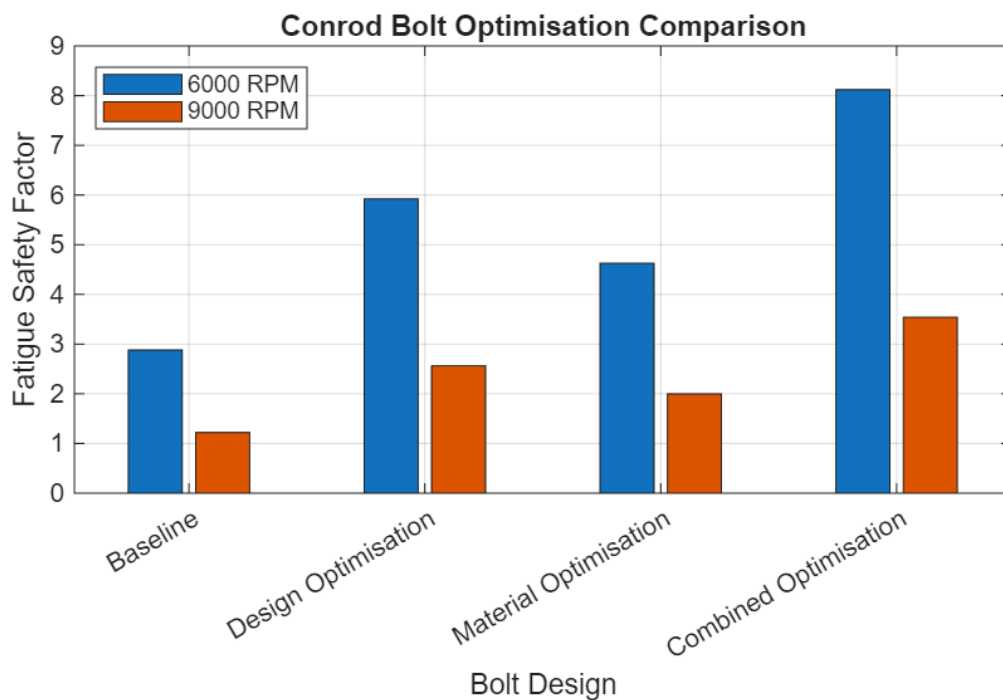
There are some drawbacks to using high strength steel like L19 as they are susceptible to hydrogen embrittlement which causes stress corrosion reducing ductility therefore, reducing fatigue resistance. (Gong P. et al. 2022)

To avoid contamination or stress corrosion it must be kept well-oiled and not exposed to moisture (ARP, 2026).

6.3.3 Material & Design Combined

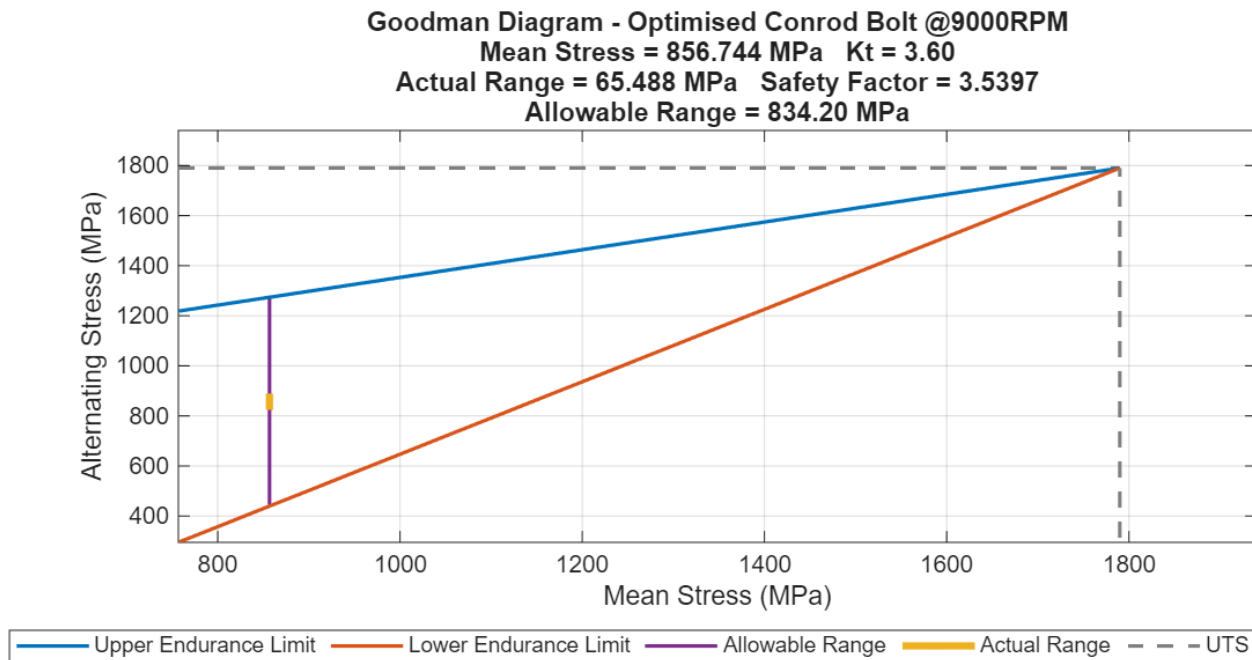


(Graph 36: Bolt Optimisation Comparison vs RPM)



(Graph 37: Bolt Optimisation Comparison at Normal and Raised Limit)

The comparison graphs above show how the bolt has become more suitable to be able to meet the raised limit conditions. With only design optimisation the FSF at the raised limit is just slightly below the 6000RPM value. If changing bolt material incurs large extra cost and manufacturing effort, then just design optimisation would be suitable for this application.

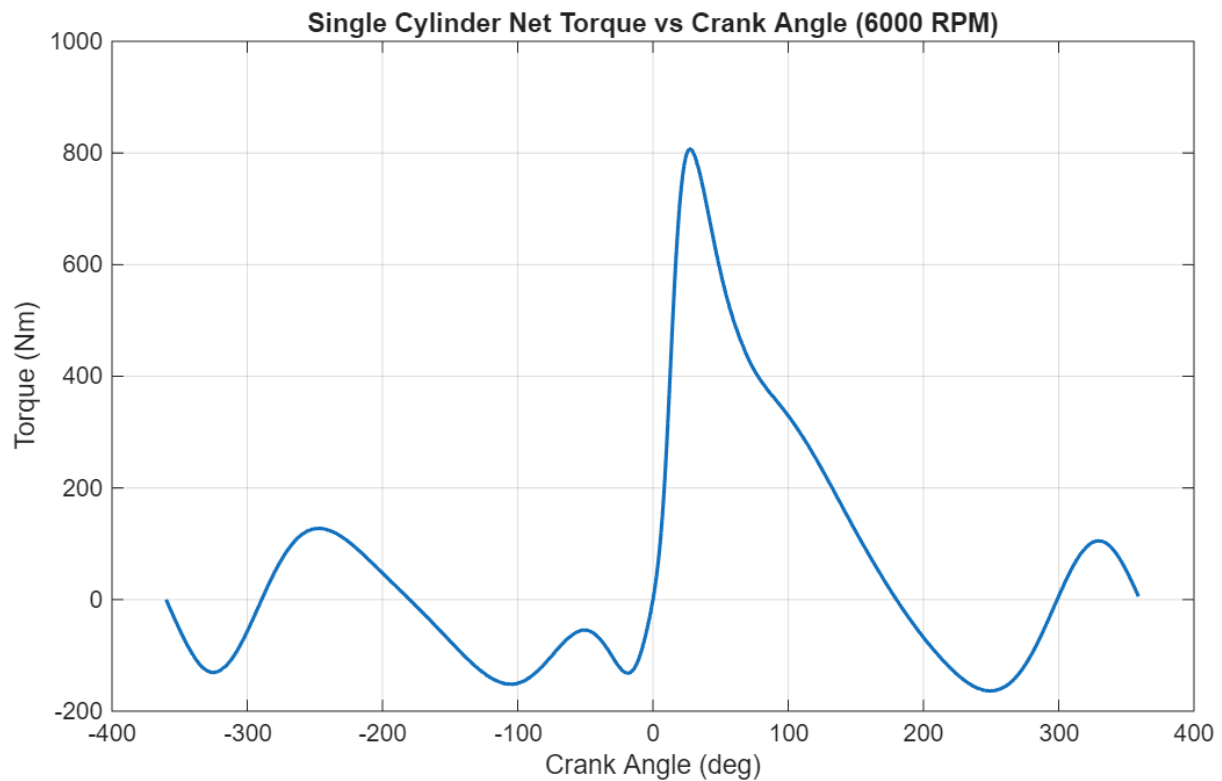


(Graph 38: Combined Optimised Goodman Range Diagram)

The Goodman diagram shows that the mean stress has increased by around 19%. The allowable range has also increased, with a slight decrease in actual range. Whilst the mean stress may be higher the proportionally small alternating stress and high UTS means that the bolt has increased fatigue life at the raised limit.

7 Instantaneous Torque Analysis

Net torque (Graph 39) was calculated using inertia and gas torque (Appendix H, I), which will be utilised for instantaneous torque analysis.



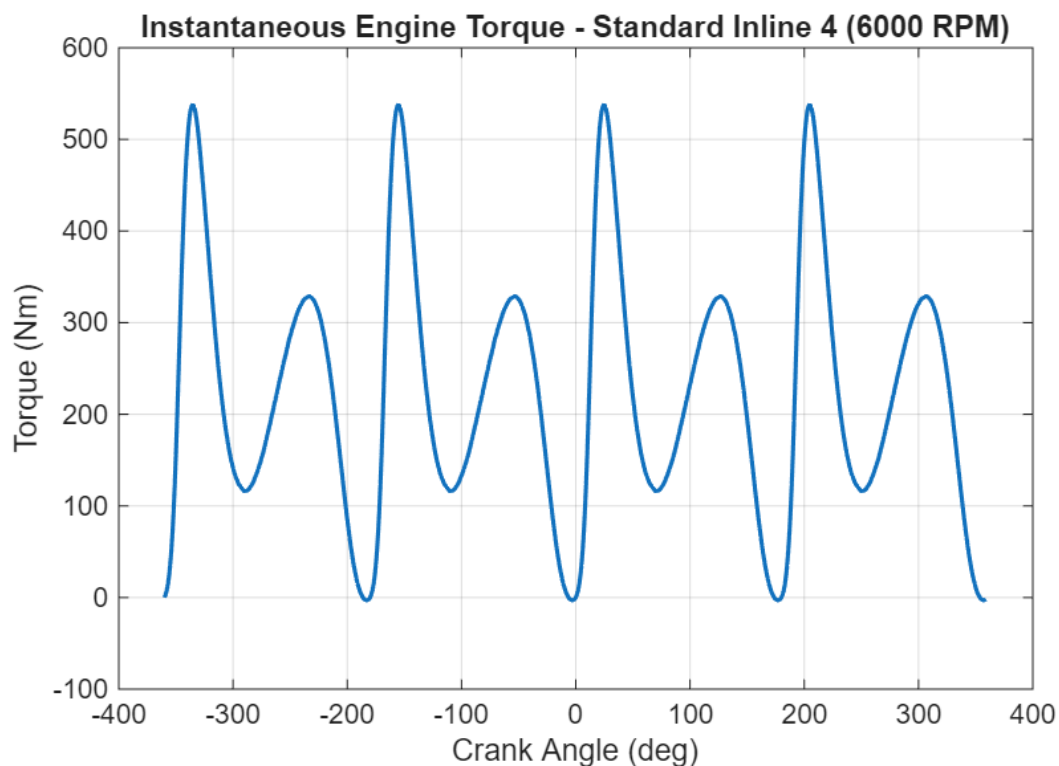
(Graph 39: Ford GTDi Net Torque)

7.1 Flat-Plane Crank



(Figure 3: Flat-Plane Crank Assembly (Cycle World, 2015))

Pictured above is a typical 4-cylinder flat-plane crank assembly which is used in engines like the Ford GTDi, utilising primary balance qualities and low weight without the need for large counterweights. Due to the low weight the engine revs quicker with better throttle response. (Tromans P. 2024)



(Graph 40: Flat-Plane Instantaneous Torque)

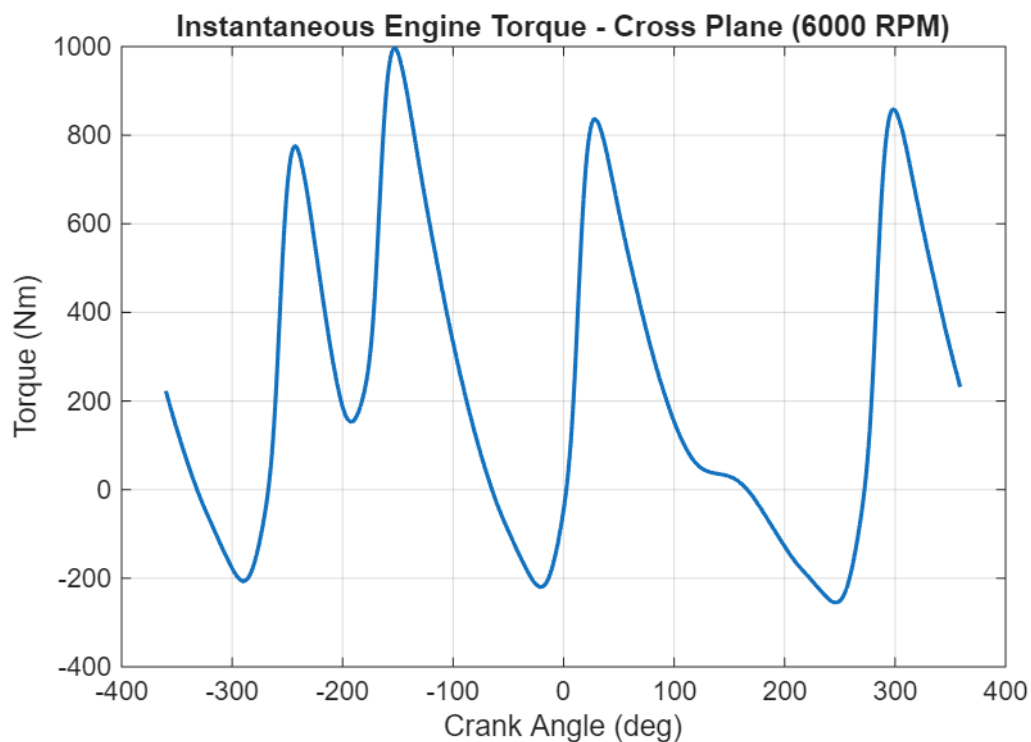
Graph 40 shows repetitive torque delivery with very low compressive (negative) torque. The reason Ford may spec this in their engines would be easier manufacturing, with improved responsiveness, and smaller low-end torque aiding predictability for the average driver.

7.2 Cross-Plane Crank



(Figure 4: Cross-Plane Crank Assembly (Yamaha, 2015))

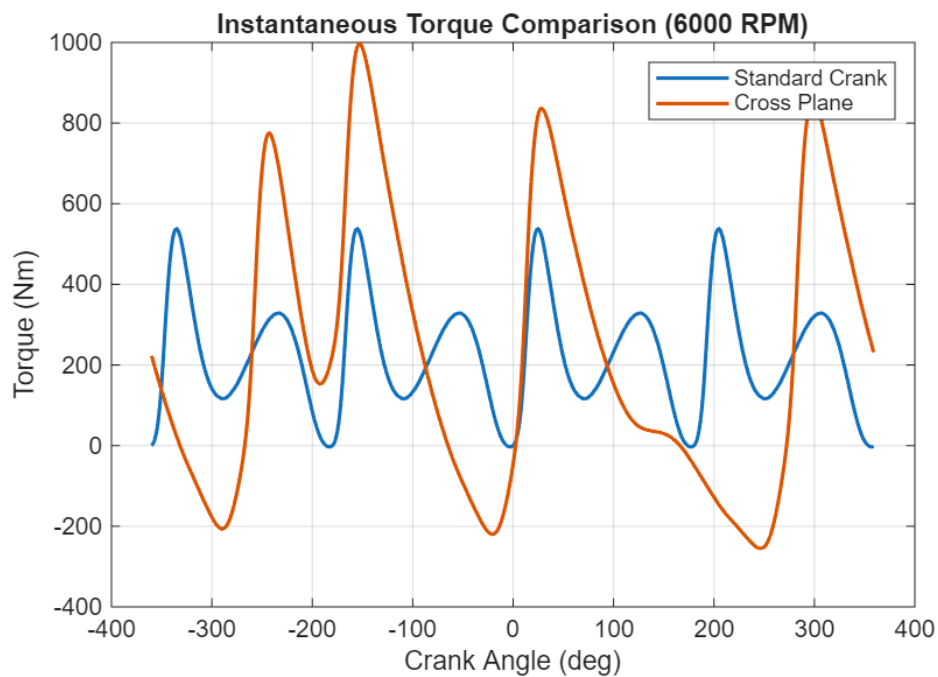
Cross-plane crankshafts are rarer for 4-cylinder applications, but is used in the Yamaha R1 engine, seen above. The benefits of cross-plane include low secondary vibrations therefore, improved smoothness and low-end torque delivery. (Rupp, S. 2022)



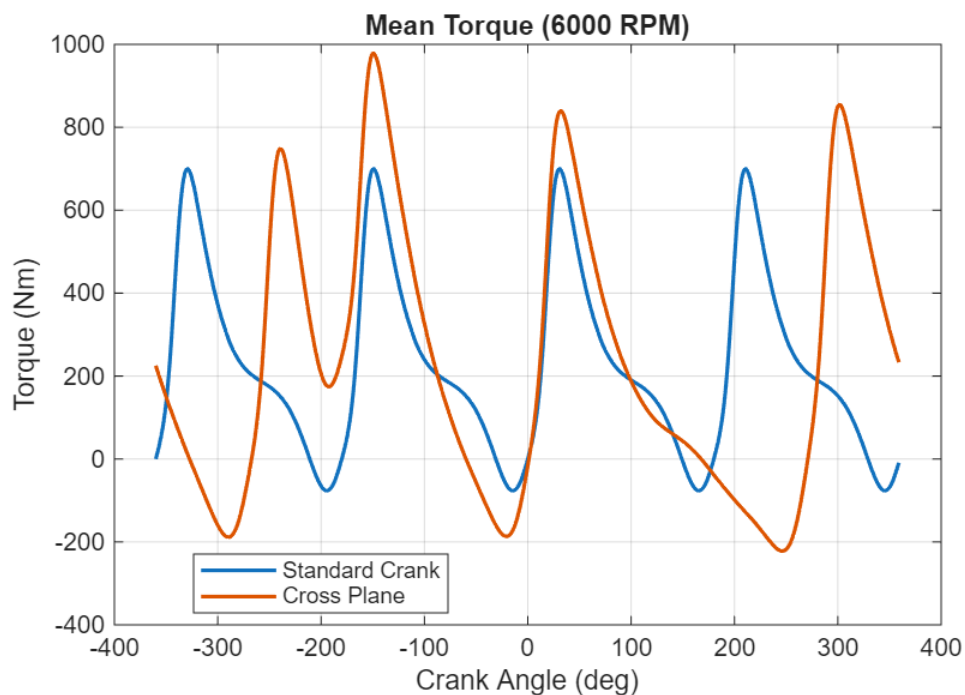
(Graph 41: Cross-Plane Instantaneous Torque)

Graph 41 shows the what the GTDi instantaneous torque would look like if specified with a cross-plane crankshaft. Ford may have avoided this configuration due to reduced responsiveness alongside necessary manufacturing changes with added design and production complexity.

7.3 Crank Design Comparison

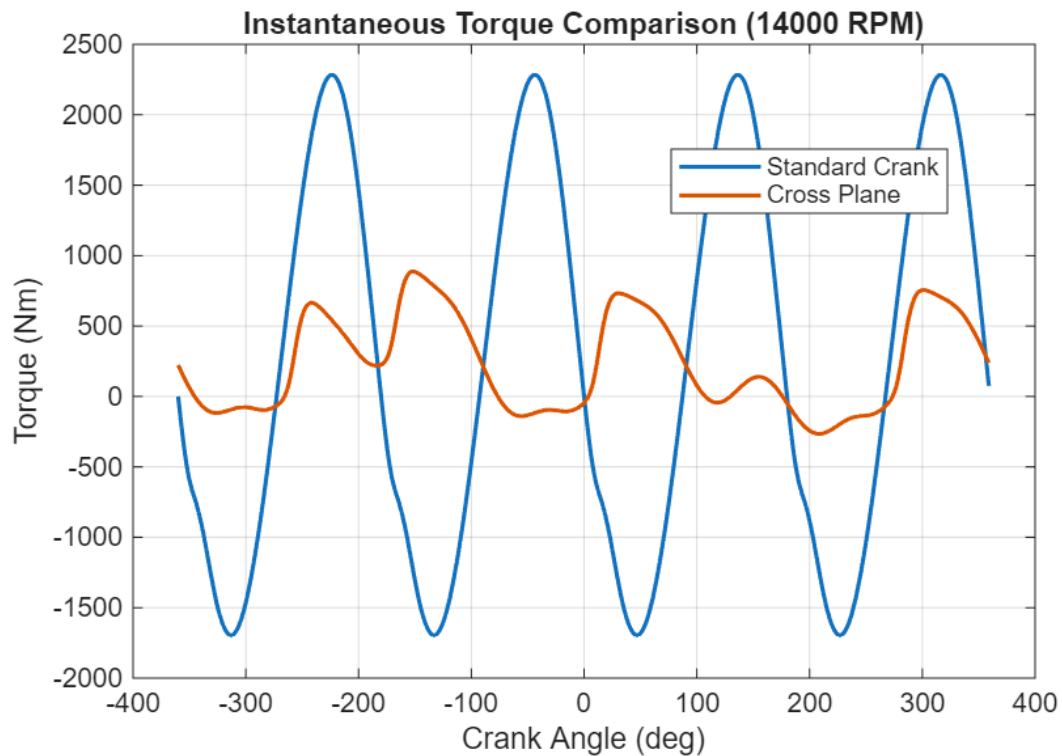


(Graph 42: Instantaneous Torque Comparison)



(Graph 43: Mean Torque Comparison)

Whilst the instantaneous torque is a clear comparison, it makes sense to compare the average (mean) torque as across a cycle to see flat-plane frequency benefits. Flat-plane shows its disadvantage of lower peak torque, but the lack of compressive torque means there is frequent delivery during the cycle at 6000RPM.



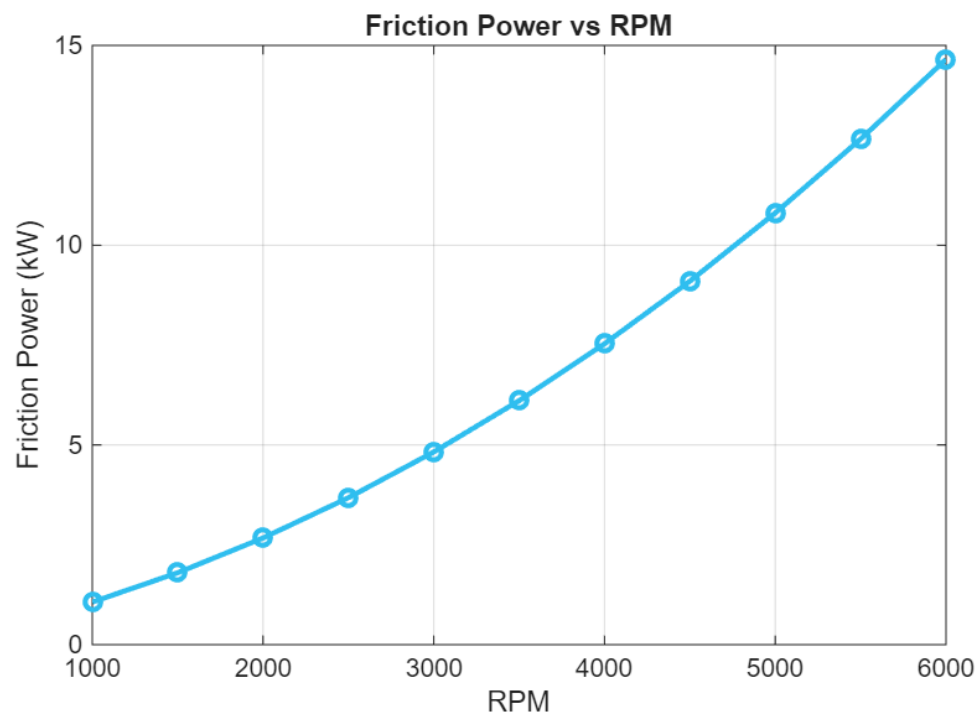
(Graph 44: Instantaneous Torque Comparison Raised Limit)

To evaluate the true benefit of the cross-plane secondary balance, the limit has been raised to 14000RPM, which wouldn't be possible for this engine, however, shows a cross-plane crank would benefit high loading conditions. The standard crank would cause engine failure due to uncontrolled torque spiking from very low compressive to very high peak torque.

Therefore, cross-plane design would suit an application that requires low-end torque and smoothness, along with secondary balance and resistance to failure at high RPM, like the Yamaha YZF-R1 (Yamaha, 2025).

8 Torque & Power

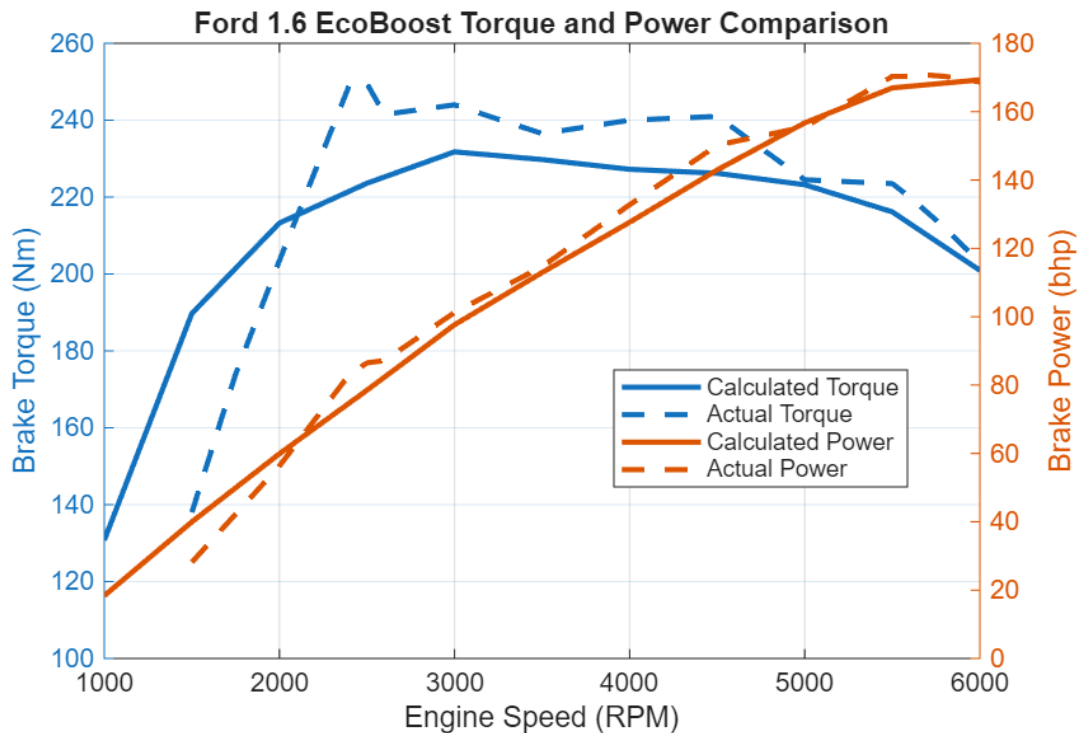
8.1 Friction Power



(Graph 45: Friction Power vs Engine Speed)

There are numerous losses within the operation of an engine including friction from ring and side forces, oil/viscosity and pumping losses. The graph isn't linear because at increased RPM there is higher mean piston speed, increasing oil and pumping losses. This was calculated using friction mean effective pressure (FMEP), equation Appendix J. Engines built for high pressure, for example compression-ignition, see higher losses and use increased constants (Blair, G. P. 1999)

8.2 Engine Curve Comparison



(**Graph 46:** Calculated vs Actual Torque and Power (Ford UK, 2010))

Brake power is the outcome of subtracting frictional power from calculated power. Ford published a torque and power curve for this engine as shown on the graph above. There is good correlation between power however, the sharper peaks in the measured torque curve are likely due to combustion effects and unmodelled dynamic losses, which are not captured in the analytical model.

The losses vary between engines due to differences in temperature, which affects oil viscosity, engine condition and accessory loads. These heavily influence the FMEP used to quantify total frictional losses. Without knowing these losses, the constants used in calculations won't match, causing discrepancies between calculated and real-world data.

9 Critical Conclusions of Results

This study evaluated the fatigue performance of key bottom-end components within the Ford GTDi EcoBoost engine under cyclic loading at 6000RPM and an extended condition of 9000RPM. At 6000RPM, all components operated within safe limits, with the conrod midpoint exhibiting a FSF of 4.14, indicating a conservative design with potential for weight reduction.

At 9000RPM, fatigue performance decreased significantly due to the quadratic increase in inertia forces with engine speed. The conrod bolt was identified as the critical component, with FSF reducing to 1.22, indicating minimal tolerance for additional real-world effects and justifying redesign. The exclusion of thermal and extra stress concentration effects likely resulted in over-estimated safety factors.

The torque and power analysis was conducted to compare flat-plane and cross-plane crankshaft configurations and their suitability for this application. The flat-plane crank showed more frequent torque delivery and improved responsiveness, whereas the cross-plane configuration offered smoother operation and greater high-speed balancing. Justification as to why the flat-plane crank was specified by Ford was highlighted through these results. Unknown practical losses modelled through FMEP contributed to deviations between calculated and real data.

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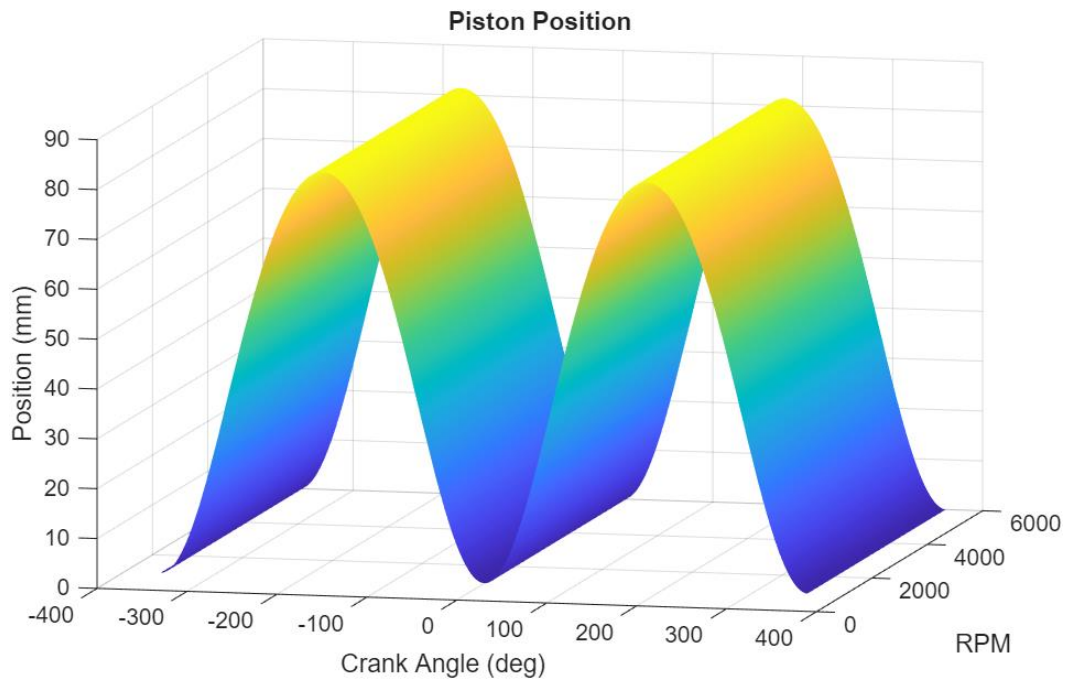
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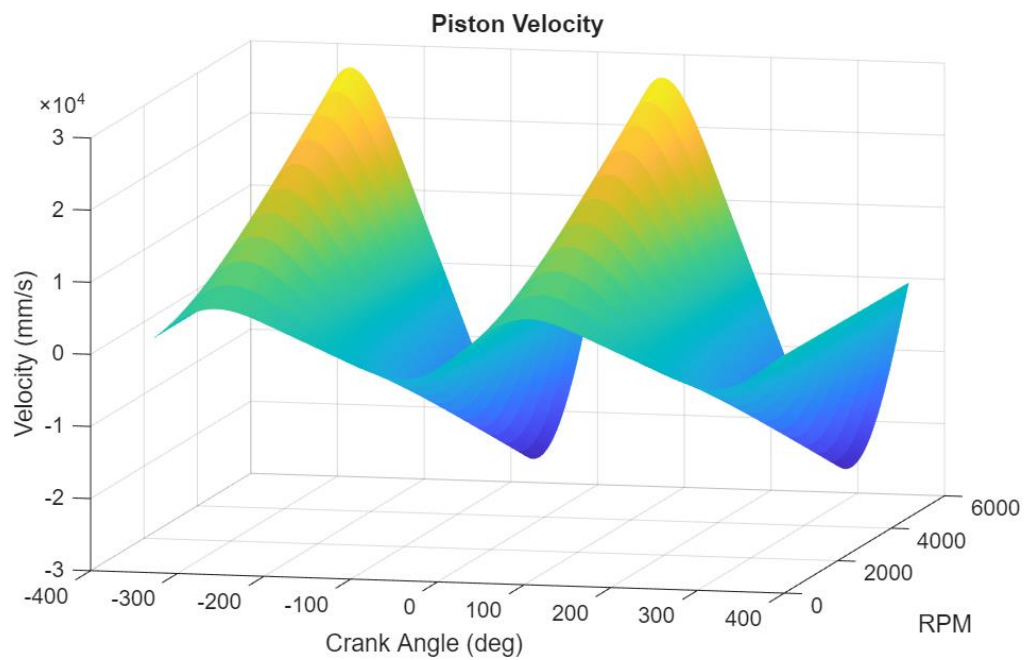
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Appendices

Appendix A – Piston Position Graph



Appendix B – Piston Velocity Graph

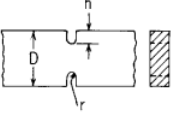


Appendix C – Conrod Bolt Tightening Torque Table

Recommended Tightening Torque (Nm)					
Size (mm)	Grade 8.8	Grade 9.8	Calculated	Grade 10.9	Grade 12.9
M8	28.8	32.3	37.8	41.3	48.3

(Fractory, 2026)

Appendix D – Roark's Stress and Strain Formula

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> <tr> <th></th><th>$0.1 \leq h/r \leq 2.0$</th><th>$2.0 \leq h/r \leq 50.0$</th></tr> <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> </table> (Refs. 1–10) 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	$-2.294 + 2.713\sqrt{h/r} + 0.240h/r$	$6.294 - 3.239\sqrt{h/r} + 0.154h/r$															

(Young, W.C. and Budynas, R.G., 2002)

Appendix E – Bolt Stiffness Equations

$$\text{Bolt Stiffness } (B_k) = \text{Elastic Modulus } (\varepsilon) \times \frac{\text{Bolt Area}}{\text{Bolt Length}}$$

Appendix F – Joint Stiffness Factor Equation

$$\text{Cap Stiffness } (C_k) = \text{Elastic Modulus } (\varepsilon) \times \frac{\text{Cap Area}}{\text{Interacting Length}}$$

$$\text{Joint Stiffness Factor } (C) = \frac{\text{Bolt Stiffness}}{\text{Bolt Stiffness} + \text{Cap Stiffness}}$$

(Shigley J. E., Mischke C. R., Budynas R. G., 2004)

Appendix G – Joint Separation Equations

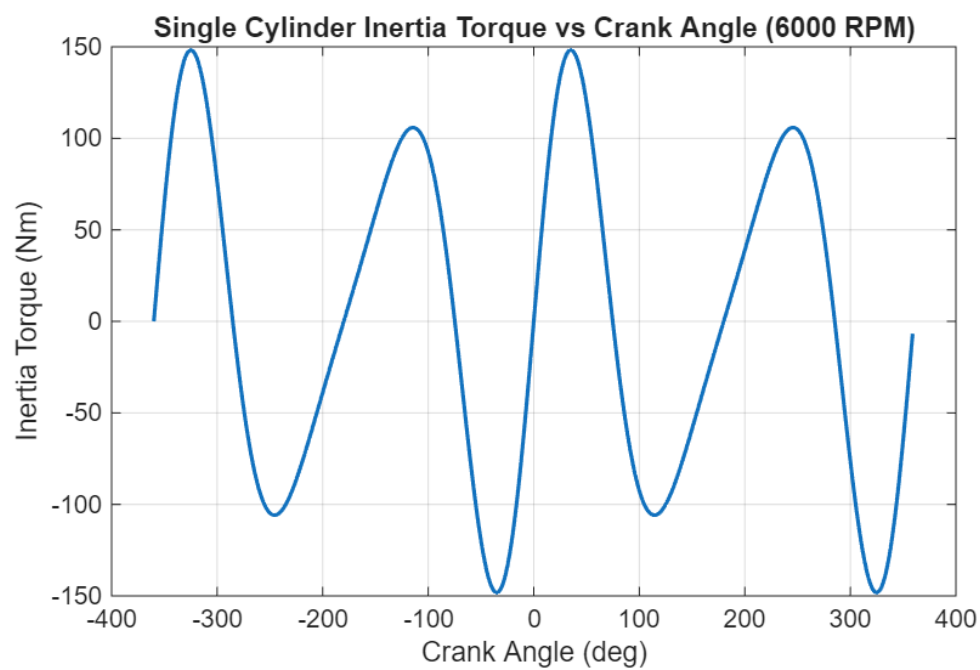
$$F_{external} (1 - C) = F_{preload}$$

$$F_{seperation} = \frac{F_{preload}}{1 - C}$$

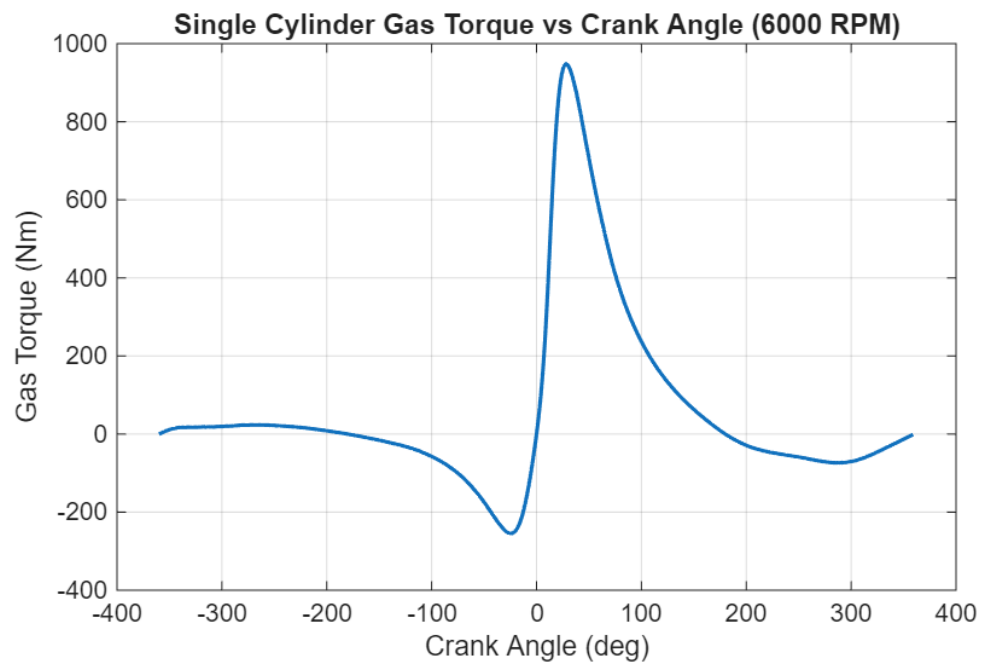
$$Seperation\ Margin = \frac{F_{seperation}}{Max\ F_{external}}$$

(Shigley J. E., Mischke C. R., Budynas R. G., 2004)

Appendix H – Inertia Torque Graph



Appendix I – Gas Torque Graph



Appendix J – FMEP Equation

$$FMEP = \text{Constant } (a) + (\text{Constant } (b) \times \text{Stroke } (L) \times \text{Engine Speed } (N))$$

(Blair, G. P. 1999)