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Analytical Assessment of Engine Internal Components

Contemporary Powertrains

ACHA6006

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Abstract

This report presents a fatigue analysis of key connecting rod components in a Ford 1.6 EcoBoost engine, calculating fatigue safety factors for the conrod midpoint, gudgeon pin and conrod bolt across 1000 to 6000 RPM. The analysis is extended to 9000 RPM using PCHIP interpolation, followed by a bolt redesign, alternative material investigation, and a comparison of instantaneous torque between flat plane and cross plane crankshaft configurations.

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Introduction

This assignment will demonstrate skills on performance improvement and alterations to the weight characteristics of the next generation engines, including an emphasis on powertrain sustainability. Furthermore, this paper will critically analyse development opportunities of internal combustion engines and how they are implemented into electric and hybrid powertrains, in order to meet future emissions and efficiency requirements within the automotive industry.

Engine based force calculations are made based around information provided from the 1.6L Ford Sigma engine, such as engine data and a specification sheet. These calculations will be used to determine the forces experienced by the engine and its components during its operating cycles. The components to be assessed is Connecting Rod mid-point, Gudgeon Pin and the Connecting Rod bolt. Calculations and information for this paper have been calculated through processes on MATLAB, with graphs and important data displayed.

1. Literature Review

1.1 What is Engine kinematics

The study of the engine kinematics is how the study of geometry of motion within the internal combustion engine, its main focus being how these parts move throughout the engine cycle (Hailemariam Nigus, 2016). The overall theory behind engine kinematics and the calculations required, is that it is used to predict how the engine system and the components behave, which can be crucial for managing component loads (Huping An, et al, 2025).

1.2 What are Fatigue safety factors

Fatigue safety factors (FSF) is a design ratio, which is used to represent a material and/or components margin of safety against any structural failure, damage or cyclic loading. The FSF is usually defined to be the ratio of fatigue strength or endurance limit, to the real alternating stresses. The usage of the FSF values is to predict how a component is to survive its intended purpose, when accounting for its geometry and giving an insight into its survival life and how it can resist cracking (Faction Bay).

FSF is commonly calculated using empirical theories that account for the mean stresses in a component, these theories include the likes of Goodman and Gerber methods.

1.3 Goodman Method

The Goodman method refers to calculations made to create a Goodman diagram and modified Goodman diagram. The Goodman diagram is a presentation of the fatigue life test. It is used to predict the fatigue life at specific given stress ratios that are based on

the properties of materials. The diagram and calculations required take the geometry of the component into consideration. When using these types of diagrams, it can help determine the safest possible operating stress when the component is undergoing fatigue loading, therefore balancing mean and alternating stresses. (Nani Bai et al, 2021)

2. Calculating the Engine Kinematics

The engine kinematics is what defines the engine cycle that will be used throughout the rest of the later studies. The kinematics information has been gathered through a cycle simulation of the engine. Using this engine data spreadsheet, calculations of various kinematic engine forces have been calculated such as the piston acceleration, inertia force and gas forces over the course of one engine cycle.

2.1 Inertia Force

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

Equation 1 – Inertia Force Equation

This equation is calculated through using acceleration, which had been previously calculated. As well as using the reciprocating mass, which considers anything that lies above the big end of the connecting rod, this is because the other various parts and 1/3 of the conrod move in an up and down motion within the cylinder, whereas the big end (2/3 conrod) of the conrod moves in a rotational motion around the crankshaft and therefore counted as rotational mass.

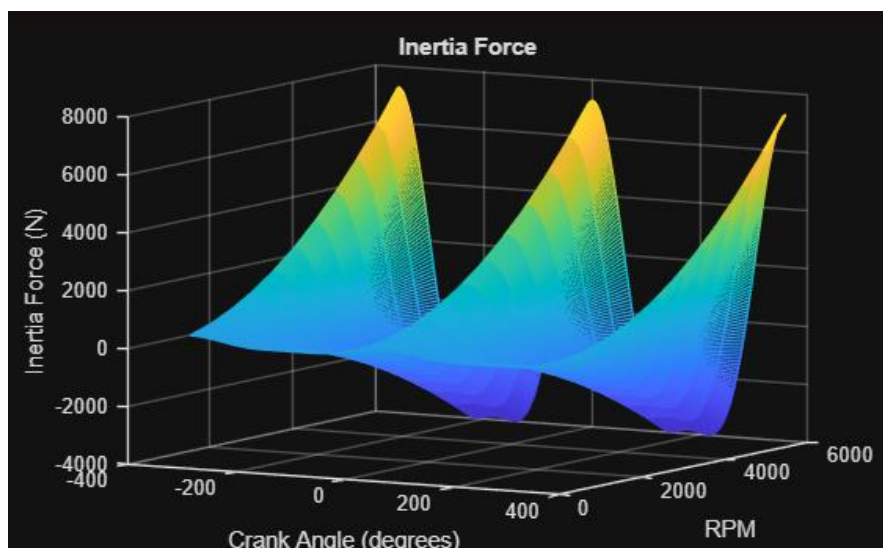


Figure 1 – Mesh Graph Displaying Inertia Force Over All RPMs in the Engine

Displayed in Figure 1 – Mesh Graph Displaying Inertia Force Over All RPMs in the Engine an overview of how the inertia force changes during the engine cycle. As shown in the graph there is a trend that can be seen where the lower the RPM the less the inertia forces peak, then when it gets higher is where the largest change in inertia forces can be seen.

When at 6000 RPM the peak inertia force reads at around 8000N which indicates it can be highly stressing to the bolts and other various components, this may later show signs of failure or close to if pushed further.

3. Fatigue safety Factors of Engine Components

By calculating the fatigue safety factors, it can help gain an understanding of the stress performance for each of these selected engine components. The list of the components is as follows:

- Connecting Rod (specifically looking at the midpoint)
- Gudgeon Pin
- Connecting Rod Bolts

3.1 Engine Kinematic Factors

When considering FSF values it is important that the gas force is considered as well as the inertia force. The main reason for this is the gas force adds compression to the conrod during the combustion phase while the inertia force pulls the conrod upward when at TDC on the exhaust stroke in the cycle, effectively making the conrod endure tension. Therefore, the gas force has to also be calculated in order to set the con rods loading conditions properly, which goes towards the calculations.

3.1.1 Gas Force

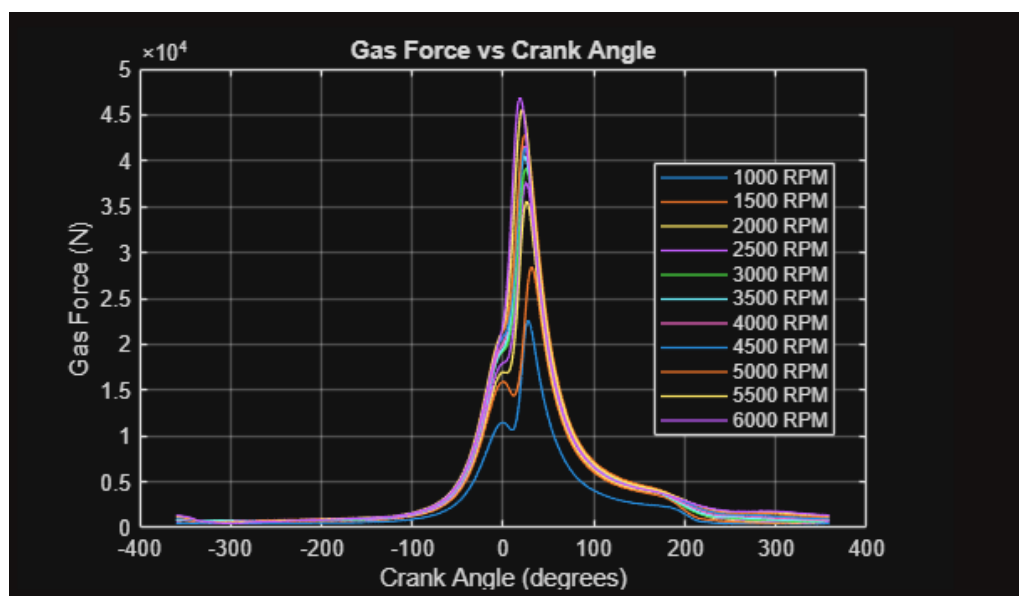


Figure 2 – Gas Force vs Crank Angle Graph

Depicted in Figure 2 – Gas Force vs Crank Angle Graph is the graph that plots the gas force per degree of crank angle during a whole cycle for each available RPM. As is displayed a sharp spike around 25 degrees past TDC, which indicates where peak cylinder pressure

occurs. At this peak is where the gas force is applying the compression onto the conrod, this drives the minimum stress.

3.1.2 Net Force

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

Equation 2 – Net Force Equation

Using the equation shown above, it can be used to calculate the net force, using this calculation it can be plotted via MATLAB script and show the net force across the whole engine cycle.

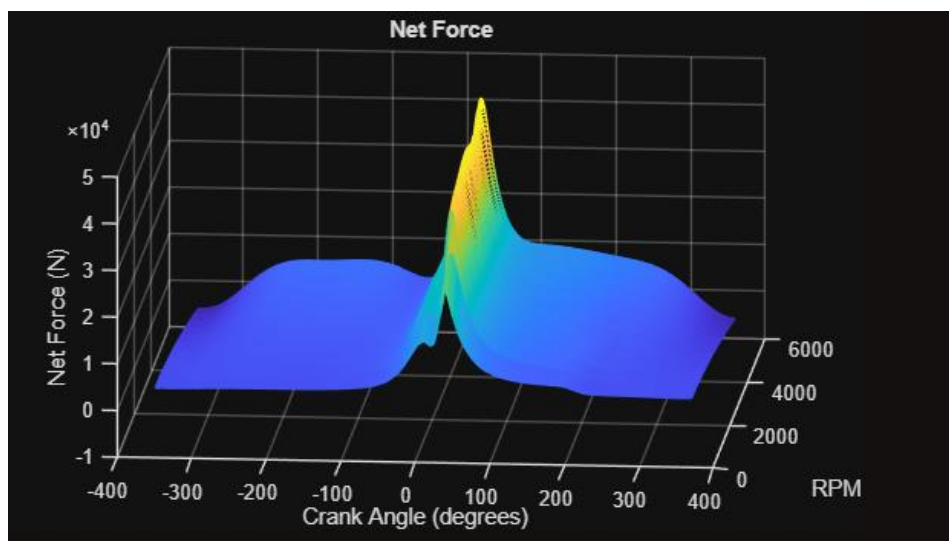


Figure 3 – Net Force vs Crank Angle Graph

From the graph shown in Figure 3 it helps provide for the max stress value. This is shown through the plateauing nature around TDC of the exhaust stroke where the gas force is at or around zero. In this graph the stress range can be assumed, this is seen through the total distance between the bottom of the combustion rise and through the highest point of the inertia plateau. This range between the two points is what can be used to determine if the conrod can survive the fatigue, therefore the larger that gap is the lower the FSF will be.

3.2 Connecting Rod Mid-Point

3.2.1 Connecting Rod Mid Point Equations

Using values for the gas force and net forces along with the forces that the connecting rod experiences that have been calculated earlier, is used for setting up the calculations required in order to form the FSF calculations.

$$F_{rod} = \frac{F_{net}}{\cos\phi} \qquad \sigma = \frac{F}{CSA}$$

Equation 3 – Rod Force Calculation

Equation 4 –Stress Calculation for Rod midpoint

Using these particular equations, they can be used to calculate both for the minimum and max stresses. These two equations work in unison to convert forces into stress at a specific crank angle position. When at the peak combustion phase the net force is dominated by the gas force, when dividing by the Cos Theta it calculates the rod force along the axis, when dividing by the cross sectional area a negative compressive stress is produced, therefore this is deemed as the minimum stress. When at TDC, specifically at the exhaust stroke it is the opposite effect, this is what leads to the max stress in this instance.

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

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

Equation 5 – Stress Range Calculation

Equation 6 – Mean Stress Calculation

Once the min and maximum values are calculated, these equations are used to characterise the stresses through the engine cycle. The stress range equation depicts how much the material and component are being stressed during each engine cycle. Regarding the mean stress it demonstrates the overall medium or average value of the cycle, displaying what type of stress (Tension or compression) the component is under on average.

3.2.2 Connecting Rod Mid Point FSF

The reason for selecting specifically the mid point for the FSF calculation, is due to how critical it is to the component, as it experiences the full axial loads that get transmitted from the surrounding components during the engine cycle.

Table 1 – Connecting Rod Mid Point FSF

RPM	FSF
1000	13.395
1500	11.818
2000	7.049
2500	6.366
3000	6.043
3500	5.582
4000	5.312
4500	5.61
5000	5.383
5500	4.747
6000	4.54

As seen in the table there is a relatively consistent downward trend in FSF, as expected the higher the RPM the lower the FSF due to the nature of the higher stresses and loads going through the engine components. There is an anomaly at 4500 RPM where the FSF goes up slightly, possibly due to a random variation in cylinder pressure. Despite this however the values remain above the critical limit highlighting the conrods structure and resistance to fatigue.

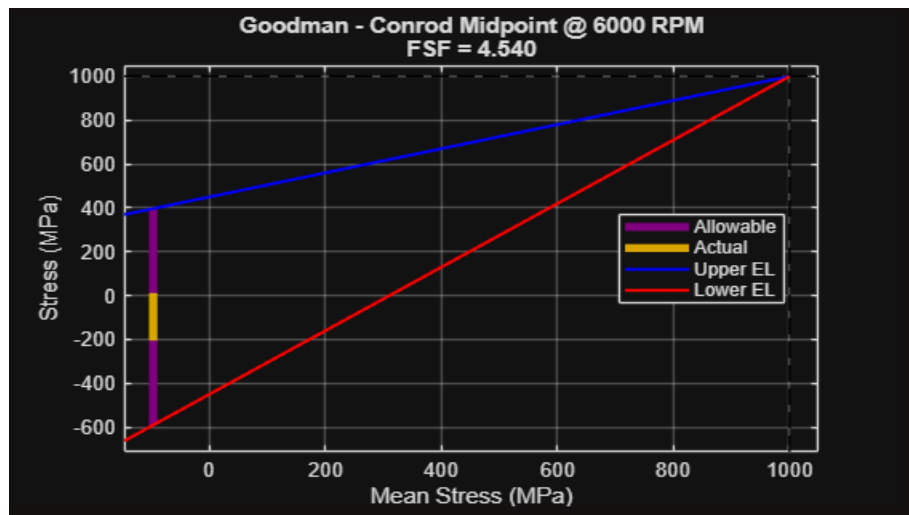


Figure 4 – Goodman Diagram of CRMP at 6000 RPM

In this particular Goodman diagram, it proves the FSF of the conrod mid point. When analysing the graph, it is seen that there is plenty of allowable stress indicating how the conrod is able to withstand the higher stresses at 6000 rpm and is safe from the effects of fatigue. The negative mean stress region indicates how the conrod mid point spends most of the time under compressive forces in the engine cycle. Overall this graph indicates how the conrod can operate safely in this operating condition.

3.3 Gudgeon Pin

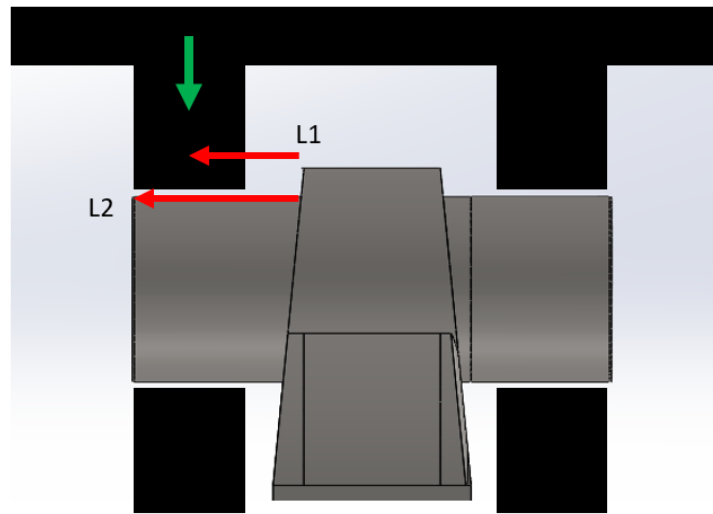


Figure 5 – Example of the gudgeon pin and L1 and L2 for reference to bending moment equations

3.3.1 Gudgeon Pin Equations

$$I = \frac{\pi}{64} (D2^4 - D1^4)$$

Equation 7 – Area MOI calculation

$$\sigma_b = \frac{My}{I}$$

Equation 9 – Bending stress equation

$$M_{max\ bending} = \left(\frac{F_{net\ max}}{2} \times L2 \right)$$

Equation 8 – Max bending moment

$$\sigma_{shear} = \frac{Force}{Cross\ Sectional\ Area}$$

Equation 10 – Shear stress equation

Each of these equations are used to calculate their representative values, these individual values help towards calculating for the Min and Max stresses along with the mean and range stresses. These respective values then allow to calculate the allowable stress which will then be used for the FSF.

3.3.2 Gudgeon Pin FSF



Figure 6 – Goodman Diagram of FSF for Gudgeon Pin

This diagram displays the FSF for the Gudgeon pin, the value itself being lower than that of the conrod mid point, this is due to the fact that the pin is more critically loaded than that of the conrod, identified through the fact it faces bending and shear stresses simultaneously. However, the FSF is still above the ideal minimum value for FSF as shown through the difference between actual and allowable stresses.

Table 2 – Gudgeon Pin FSF all RPM

RPM	FSF
1000	11.341
1500	10.087
2000	6.236
2500	5.582
3000	5.264
3500	4.812
4000	4.546
4500	4.717
5000	4.47
5500	3.92
6000	3.692

As shown in the table the pattern remains relatively the same as previously shown through the conrod data. The anomaly that had already been highlighted can be seen at 4500RPM, however this time it is a larger jump up, indicating this jump in the pressure data is stressing the pin more than the conrod. Again, as stated previously the FSF value remains higher than the critical limit.

3.4 Connecting Rod Bolt

3.4.1 Connecting Rod Bolt FSF

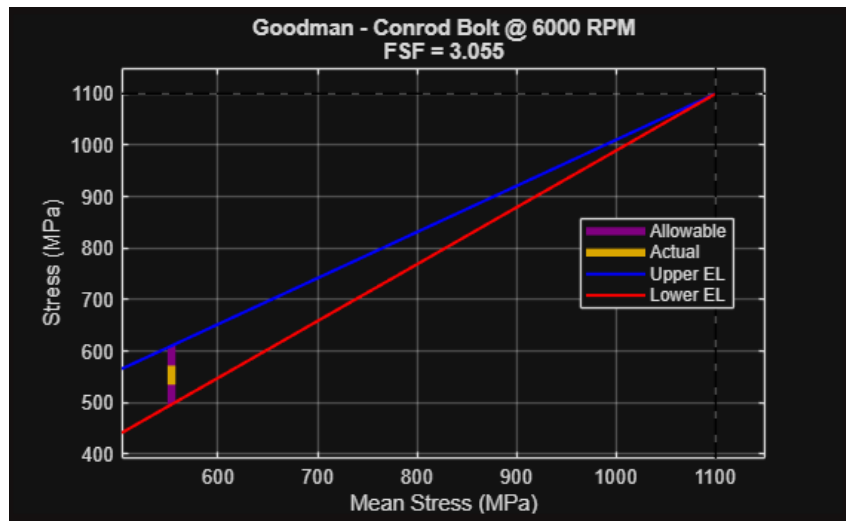


Figure 7 – Goodman Diagram for ConRod Bolt at 6000 RPM

As can be seen, similar to the previous diagrams, the depiction is relatively the same. What this graph details is how the bolt is working within its stress range with an FSF of 3.055, much like the other components. However, as can be seen the endurance limits are more restrictive and therefore small allowable space is available, possible due to a combination of the nature of the component's geometry and the axial forces that go through it.

Table 3 – ConRod Bolt FSF

RPM	FSF
1000	113.577
1500	50.421
2000	28.316
2500	18.085
3000	12.527
3500	9.176
4000	7.001
4500	5.51
5000	4.443
5500	3.654
6000	3.054

Depicted in the table is the FSF for all RPMs, as can be seen the anomaly that has been viewed on the previous tables at 4500 can not be seen here, this is due to the fact that the calculations prior to the FSF calculation for the bolt are independent of any pressure information. This leads to the constant decreasing values as to be expected and can conclude that there is a very brief variation in the pressure data around that RPM.

4. RPM Extension Analysis

4.1 Interpolation Method

The used method is called PCHIP (Piecewise Cubic Hermite Interpolating Polynomial). This method is used to estimate what the data should look like using the already given data, PCHIP uses these known data points and fits a curve to it, it then further predicts what the curve will look like based on the patterns already known. PCHIP had been used due to the fact it preserves the local shape between datapoints and does not produce any anomalies in the data. A lower bound of 1 bar had also been included so the data will not drop below atmospheric pressure. (Burden, R.L and Faires J.D, 2010) (Hadeef et al, 2012)

4.2 Bolt Fatigue at 9000 RPM

Table 4 – Extended RPM FSF

RPM	FSF
6000	3.055
6500	2.588
7000	2.217
7500	1.918
8000	1.673
8500	1.47
9000	1.3

The table displays the range starting from 6000 and ending at 9000 RPM. It becomes increasingly more obvious that the bolt is prone to failure the higher it is pushed especially shown when at 8500-9000 RPM where it is exceedingly close.

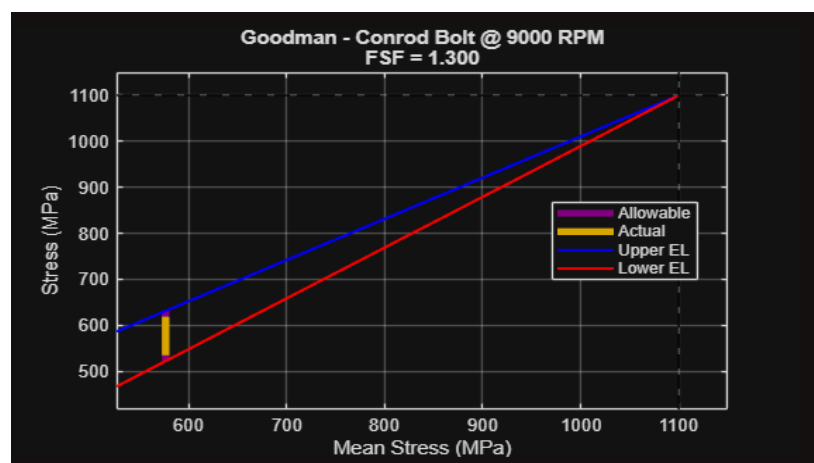


Figure 8 – Goodman diagram For ConRod Bolt at 9000 RPM

This Goodman diagram further backs up what has been previously mentioned, by showcasing the FSF at 9000 RPM it clearly displays how the bolt has very little allowable stress and therefore indicating potential failure.

5. Bolt Redesign

5.1 Redesign Approach

The redesign follows the waisted shank method, splitting the single bolt section into three. The reasoning for this is that reducing cross sectional area of the middle section the overall stiffness is reduced. The reduction lowers the extra forces transferred into the bolt in operation through the stiffness ratio $F_{extra} = F_{inertia} \times K_{bolt} / (K_{bolt} + K_{cap})$, meaning a smaller proportion of the inertia separating force at TDC and narrowing the stress range (*Barrett, 1990*) (*Griza et al, 2009*).

On this occasion everything apart from the middle shank, which features a reduced diameter, section stays the same

5.2 Redesign FSF

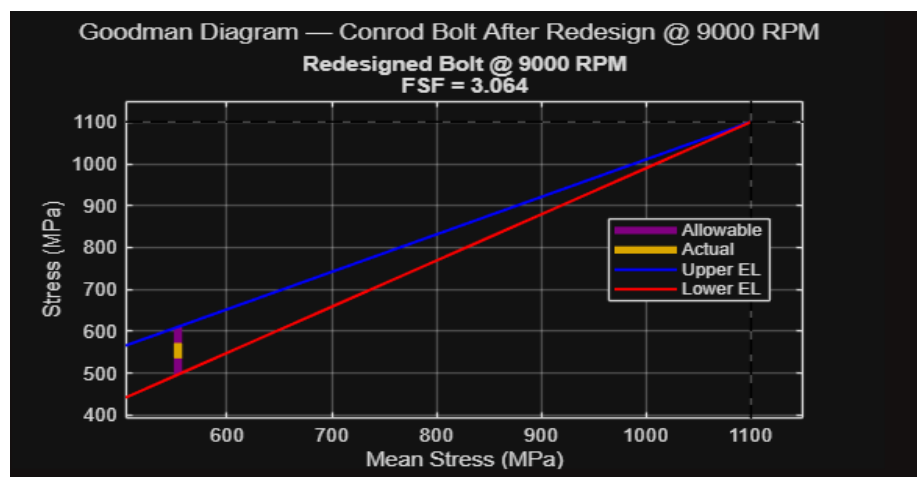


Figure 9 - Goodman diagram of Redesign at 9000 RPM

When looking at this diagram it can be seen how the redesign means the bolt has more allowable stress than before, further backed up by the FSF value displayed of 3.064 compared to the 1.3 of before.

6. Crankshaft Torque comparison

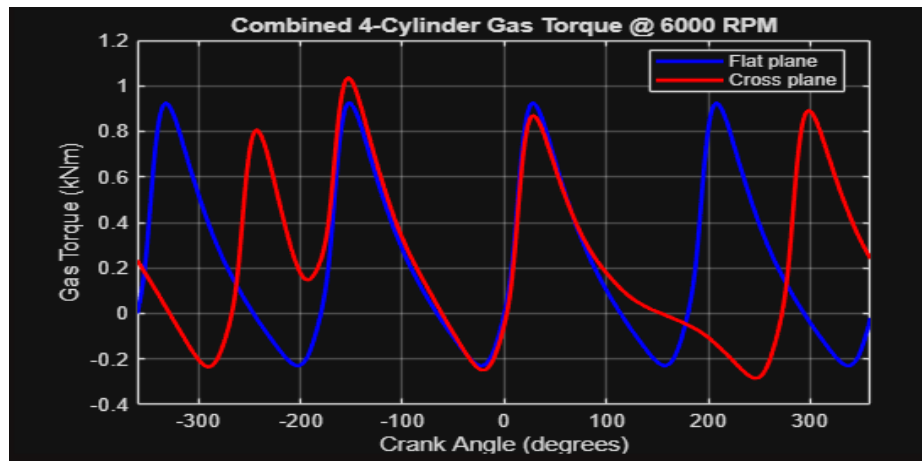


Figure 9 – Combined 4cyl Gas torque at 6000 RPM

From Figure 9 it can be seen how the red line indicating the cross plane shows much larger fluctuations, this is due to the ignition point spacing.

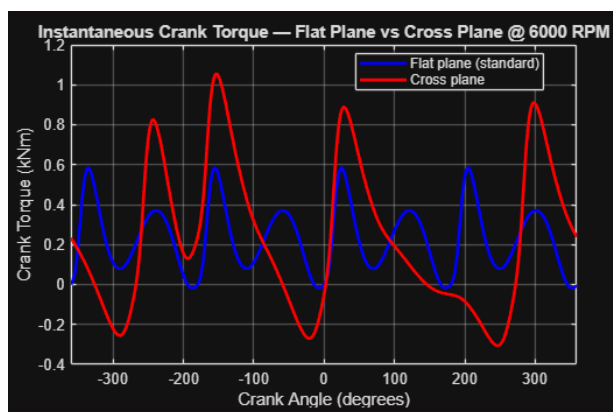


Figure 10 – Flat Plane vs Cross Plane Comparison Graph

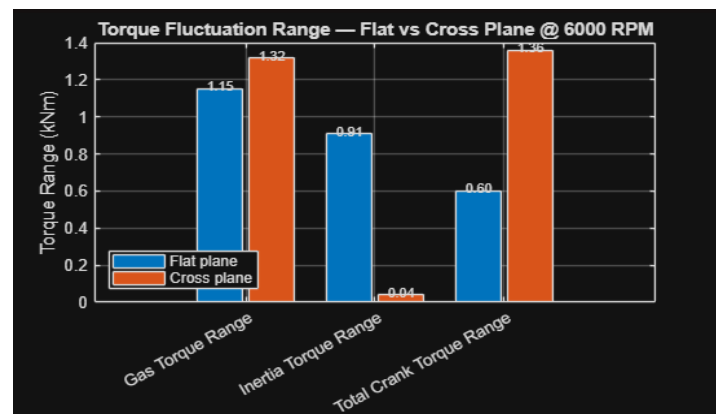


Figure 11 – Fluctuation difference chart

Figure 10 displays the instantaneous crankshaft torque between standard and cross plane configurations. The cross plane shows examples of absorbing energy as evident through the troughs when it goes below zero, whereas the flat plane configuration remains to be smooth and consistent, proving that the standard plane has a lower range of instantaneous torque, further detailed in Figure 11.

7. Alternative Bolt Material Study

For this study 3 alternative types of metals have been considered:

- Grade 12.9 Steel – Upgrade from 10.9 steel, with a higher UTS of 1220MPa, cost effective and increased strength. (*Bickford, JH, 2008*)
- AerMet 100 steel – Common in motorsport and aerospace applications, UTS of 1965MPa, raised endurance limit. (*Lesuer, D.R, 1996*)
- Ti-6Al-4V – Titanium alloy, high strength keeps the allowable stress competitive despite lower UTS than the other alternatives. (*Lutjering, G, 2007*)

7.1 Material Study Analysis

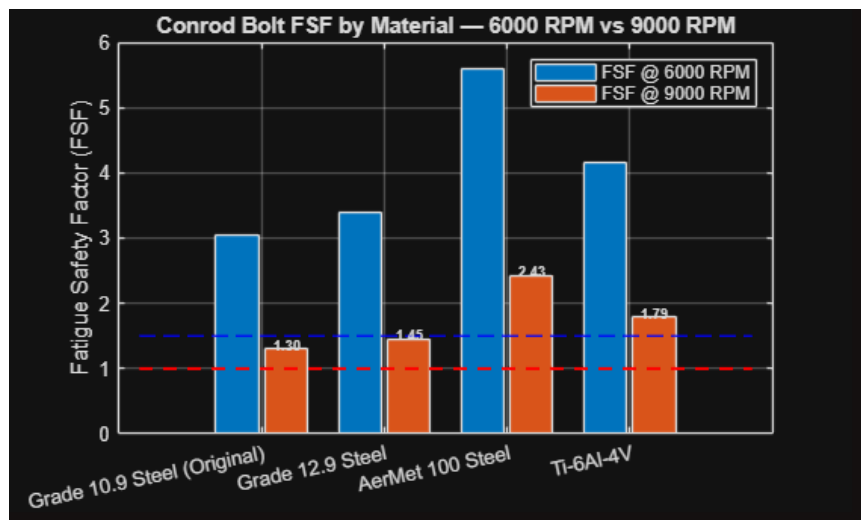


Figure 12 – All Materials Comparison Chart

From figure 12 it can be visualised the advantages of these other materials when comparing them against the original, all of which perform better at both 6000 and 9000 RPM. The AerMet 100 Steel proving to be the best performing, highlighting its higher UTS than the others. However, due to Ti-6Al-4V being lighter and offering good performance it also makes it a highly competitive argument for usage as the new material.

8. Conclusion

This report has successfully analysed the fatigue safety factors of three critical connecting rod components across the full operating range of 1000 to 6000 RPM, demonstrating that all three components remain above the failure threshold throughout with the conrod bolt producing the lowest FSF of 3.715 at 6000 RPM. The analysis was extended to 9000 RPM through PCHIP interpolation of the existing pressure data, revealing a significant deterioration in bolt FSF, which highlighted the inadequacy of the original bolt design at extended operating speeds. A waisted shank redesign was implemented by splitting the bolt into three, successfully recovering the FSF at 9000 RPM back to the original 6000 RPM target through purely geometric means without any material change. Alternative materials were investigated, with AerMet 100 providing the

greatest absolute FSF improvement and Ti-6Al-4V offering the most efficient solution through its benefit of reduced stiffness and high specific strength. Finally, the instantaneous torque analysis demonstrated that while the flat plane configuration delivers smoother and more consistent torque output suited to road car applications, the cross plane configuration significantly reduces inertia torque fluctuations making it better suited to high performance applications.

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Appendix 1

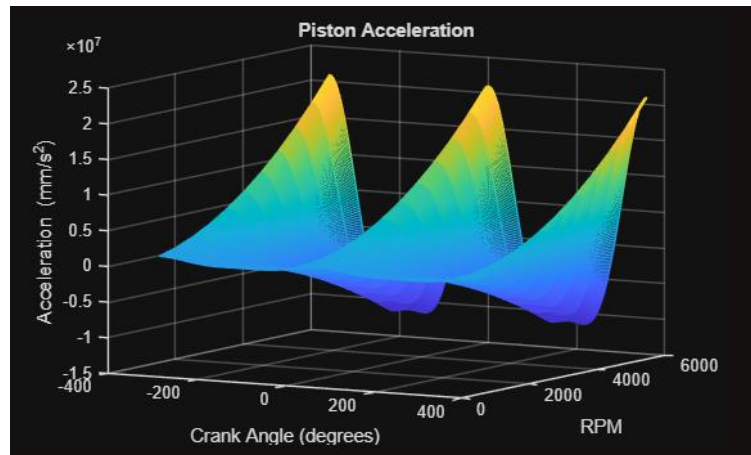


Figure 13 – Piston acceleration vs crank angle vs RPM

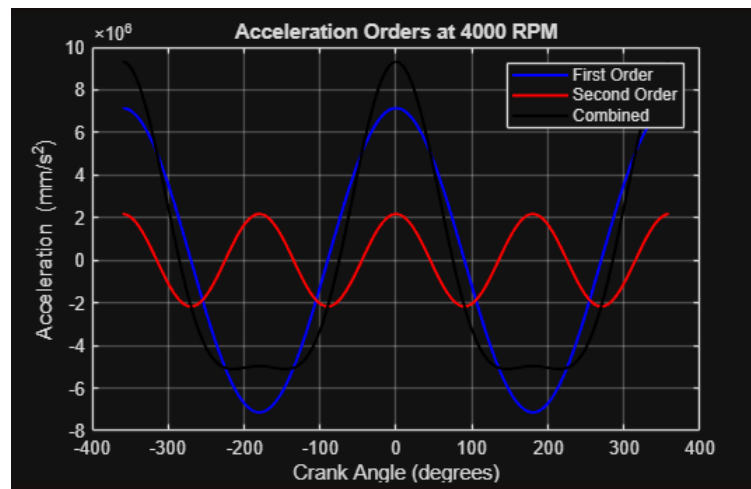


Figure 14 – Acceleration orders at 4000 RPM

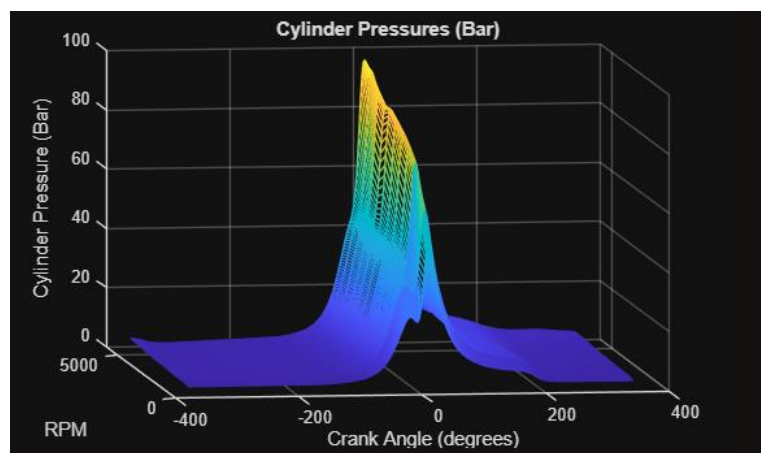


Figure 15 – Cylinder pressures in Bar per crank angle

Appendix 2

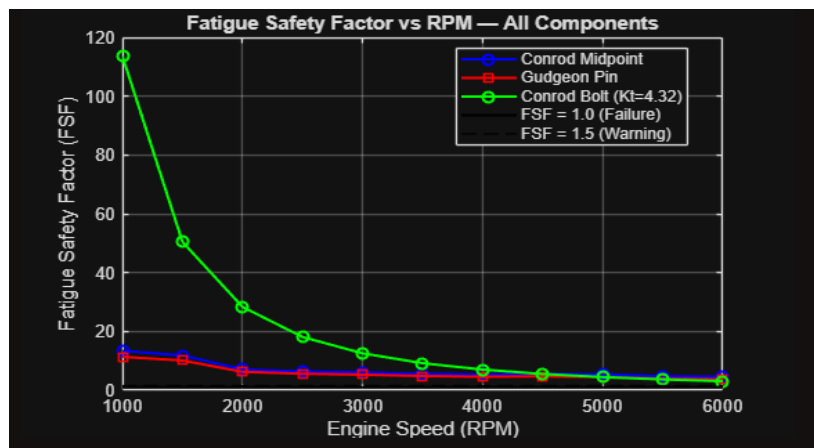


Figure 16 – FSF per RPM all components

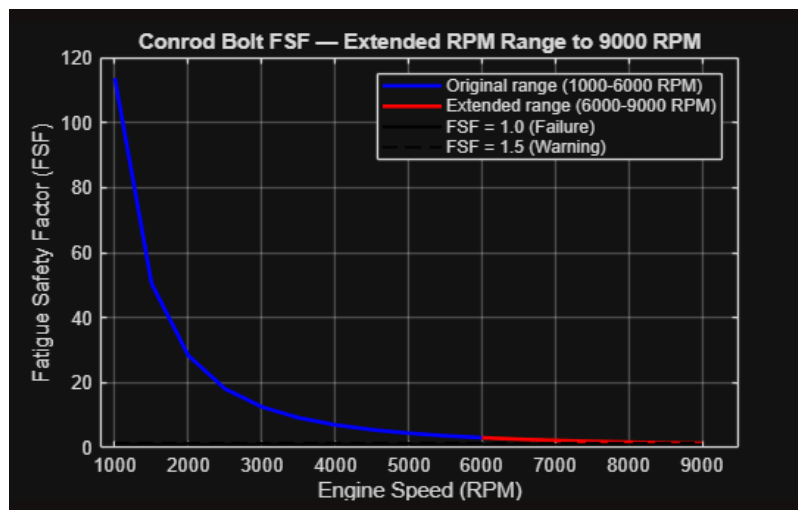


Figure 17 – ConRod Bolt FSF at extended ranges

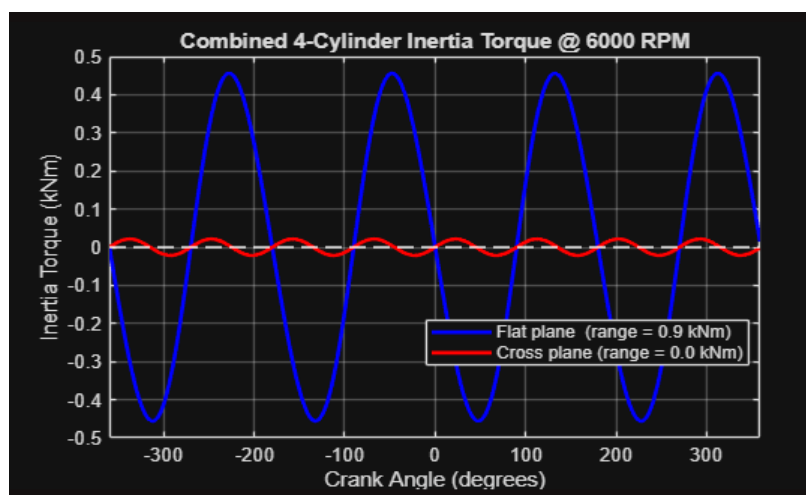


Figure 18 – Inertia Torque Flat Plane and Cross Plane at 6000RPM

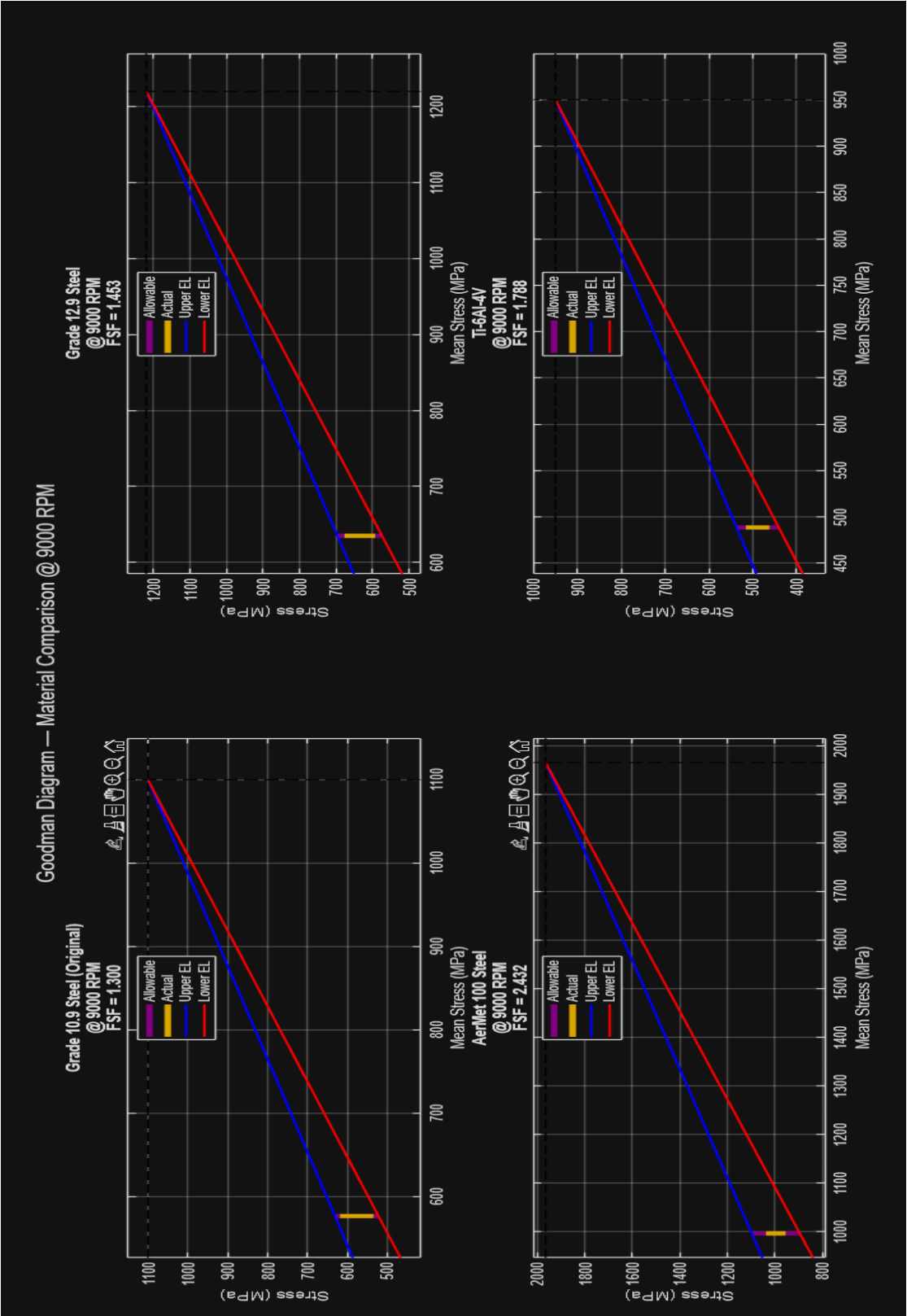


Figure 19 – All Materials Goodman diagrams