

ANALYTICAL ASSESSMENT OF ENGINE INTERNAL COMPONENTS

**Contemporary Powertrain
Design
(ACHA6006)**

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1. Introduction

Internal combustion engines operate under constant high cyclic loads due to combustion pressures and reciprocating inertia. These loads are transferred through key components such as the con rod, gudgeon pin and con rod bolts. Considerations must be made to ensure repeated stress cycles don't cause fatigue failure. Despite this, care must also be taken not to over-engineer components resulting in an inefficient design with increased inertia. A balance must be struck between these problems for an optimal engine design, and a fatigue safety factor (FSF) is how this balance is measured.

This report aims to investigate the forces and resulting stresses acting upon three key components: the con rod, gudgeon pin and con rod bolts. This will allow for the calculation and discussion of their fatigue safety factors. The Ford engine's RPM limit will then be increased from its current 6000RPM to 9000RPM to observe the effects this has on the forces, stresses and FSF of the con rod bolt. Based on these observations, the con rod bolt will then be redesigned to return its FSF to an acceptable value. Finally, the influence of a change in crankshaft configuration from flat plane to cross plane will be investigated.

2. Engine Component Modelling

2.1. Con Rod Mid-Point

The con rod converts the reciprocating motion of the piston into a rotational motion at the crankshaft; this is how power is transmitted to the drivetrain (Wakshume & Sufe, 2026). This means that the con rod has two forces acting upon it: the gas force pushing down on the piston during combustion and the inertia forces between TDC and BDC.

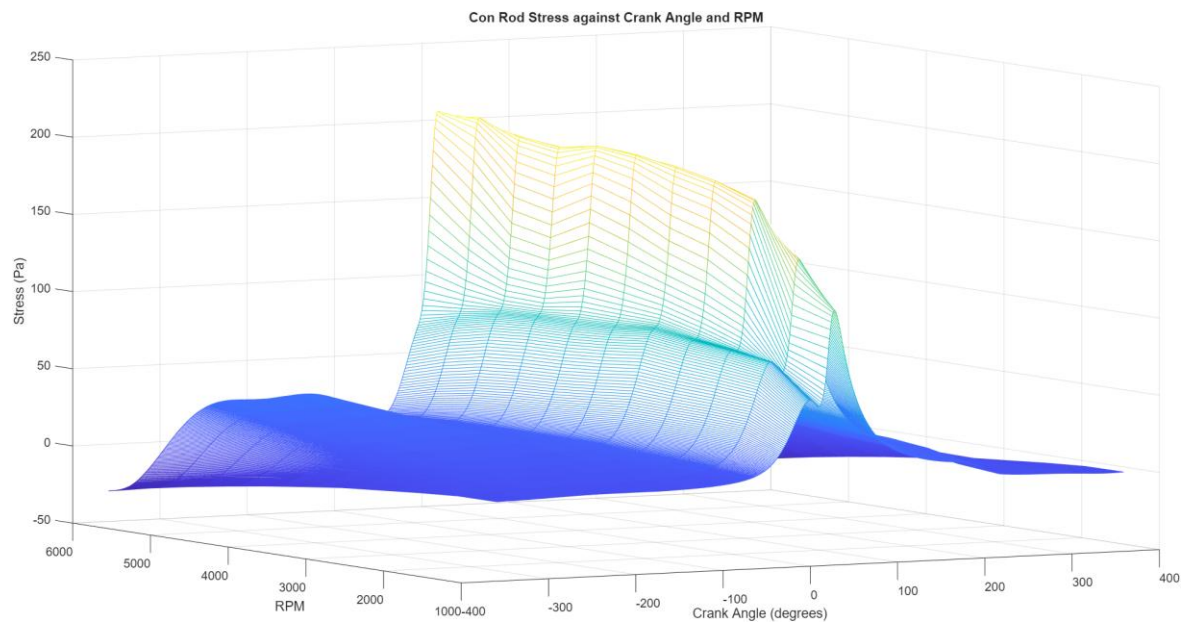


Figure 1 - Con Rod Stress against Crank Angle and RPM

This combination of forces is observed in Figure 1. The con rod stresses at the mid-point of the shank section are prominently compressive; more than 4 times larger than tensile stress at its peak. This is because the forces caused by combustion at TDC are much stronger than those caused by the reciprocating inertia. Tensile stress also appears to be more RPM dependent due to inertia forces' dependency on angular velocity.

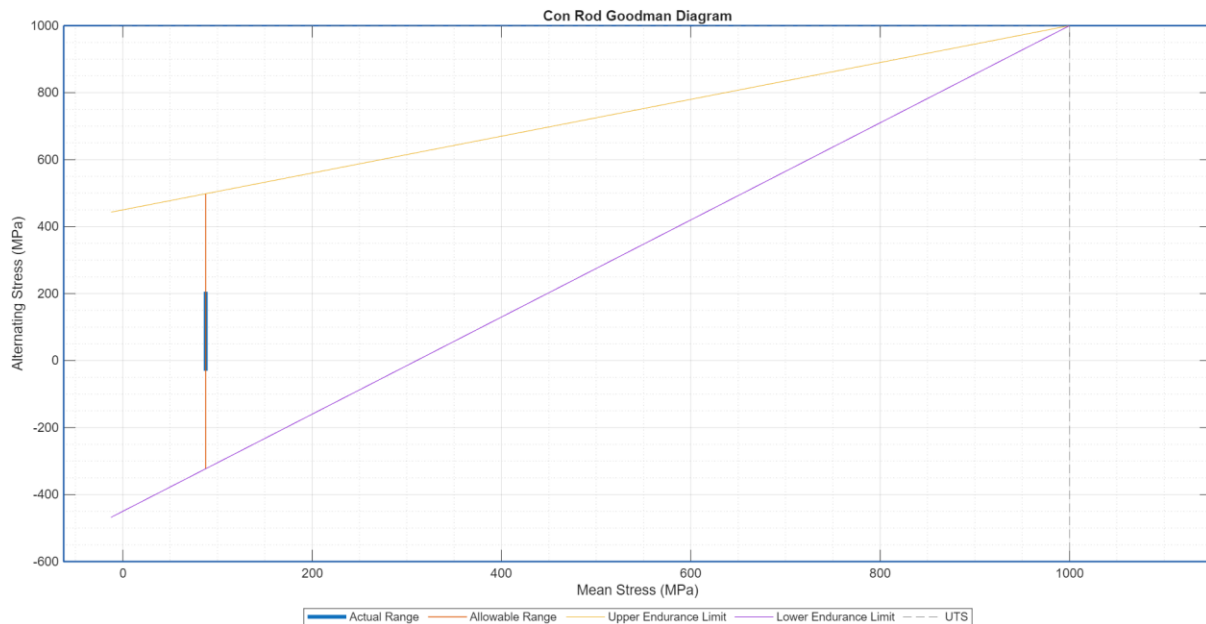


Figure 2 - Con Rod Goodman Diagram

The con rod's FSF of 3.48 suggests that it operates well within safe limits during use. The shank section of the con rod is nowhere near its structural limit. This could be because this engine is not particularly high performance meaning that marginal gains in component efficiency at the cost of durability have not been chased. This is visualised by the Goodman diagram seen in figure 2. The actual stress range of the con rod is comfortably within the allowable stress range at the mid-point of the shank section.

2.2. Gudgeon Pin

The gudgeon pin experiences stress in two directions depending on the loading conditions. During combustion, the pin is bent downwards meaning the upper surface experiences tensile stress and the lower surface is being compressed. The opposite is true for inertia forces which bend the pin upwards. A shear force also acts upon the gudgeon pin due to the way it's supported by the con rod and piston. These stresses combined create the overall stress loading for the gudgeon pin.

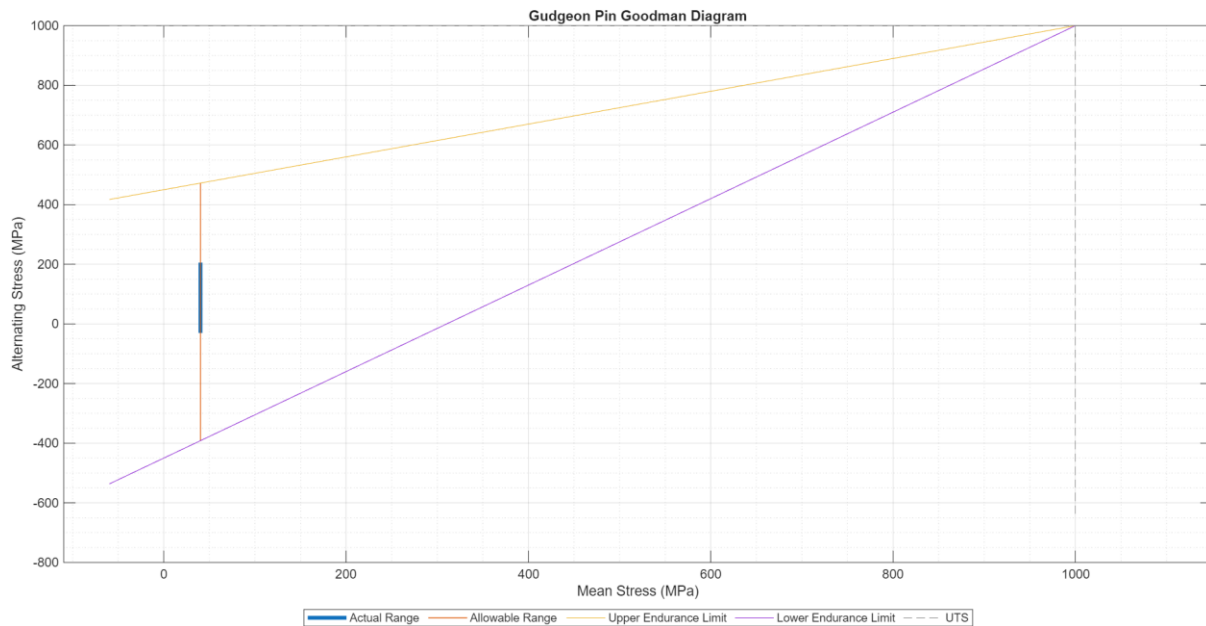


Figure 3 - Gudgeon Pin Goodman Diagram

A fatigue safety factor of 4.15 means that the gudgeon pin is operating well within its allowable fatigue window as expressed in Figure 3. One reason for this is the low mean stress the pin experiences (between -63MPa and 144MPa) which reduces the likelihood of crack propagation. The alternating stress is also very small in comparison to the maximum allowable stress the pin can experience. Based on the calculated data, this pin has been very conservatively designed and can withstand stresses well beyond the current stress range. This could be because the pin is a critical load transfer component that is subject to wear and is difficult to inspect or replace.

2.3. Con Rod Bolt

The con rod bolt's stress loads are unique due to its tightening torque. This creates axial stress under the bolt head and at the thread ends causing the bolt to be in tensile stress even when the engine isn't running. The stresses experienced by the bolt during engine operation are additional to this axial stress meaning the bolt is never in a compressed state.

2.3.1. Fatigue Safety Factor

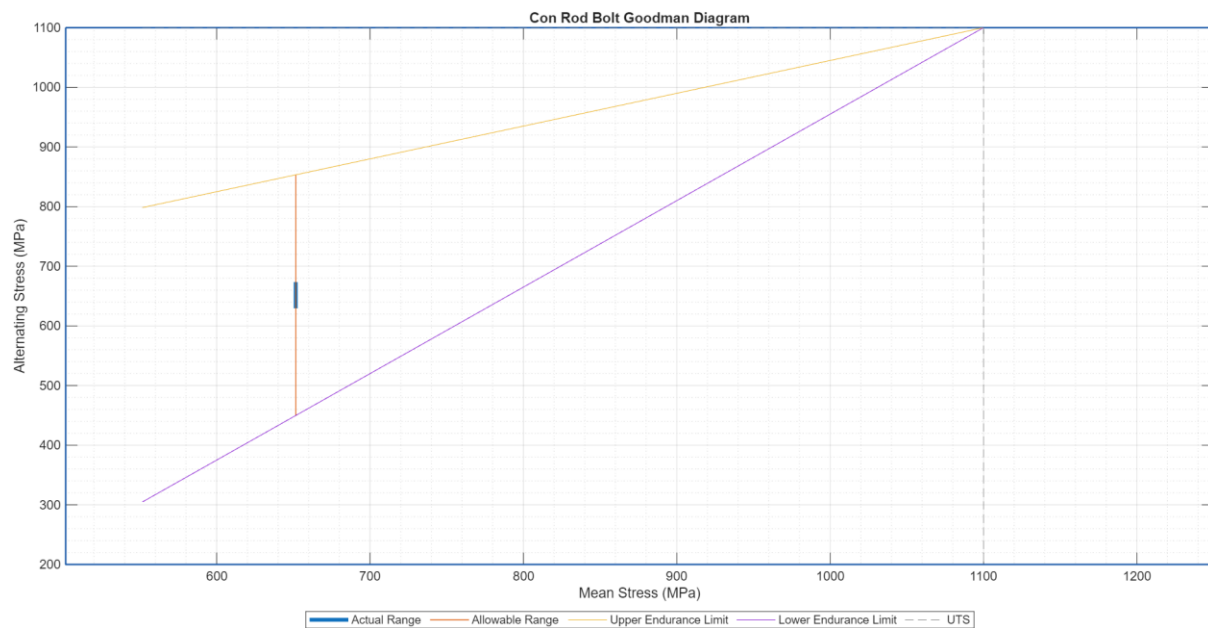


Figure 4 - Con Rod Bolt Goodman Diagram

The prominently tensile load is conveyed clearly in Figure 4 which denotes the bolt's stresses occurring entirely in the positive region. The actual stress range (43.93MPa) in comparison to the allowable stress range (403.61MPa) results in an FSF of 9.19. This makes the bolt seem over-engineered; it's nearly ten times more capable than the requirements of its operational stresses. However, this isn't a fair representation of the bolt as the thread isn't accounted for in these calculations.

2.3.2. Stress Concentrated Fatigue Safety Factor

The con rod bolt's threads are a major source of stress concentration due to the change in geometry. This can be taken into consideration with Roark's equations. The calculated stress concentration factor of 4.69 (based on the thread depth and root radius of the con rod bolt thread) can then be applied to the FSF calculations to provide a more realistic FSF of 1.959. This is noticeably lower than the previous two components due to the constant tensile loading of the bolt causing a higher maximum stress.

3. RPM Extension

The engine's RPM range is a large part of component design considerations because inertial forces increase at a squared rate to RPM. Therefore, any changes to RPM are amplified significantly. A unique consideration to make for the con rod bolt is that its maximum stress occurs at the intake and exhaust strokes rather than during combustion. This is because its stress is inertia dominated and during the intake and exhaust strokes there is no combustion pressure to work against its inertia.

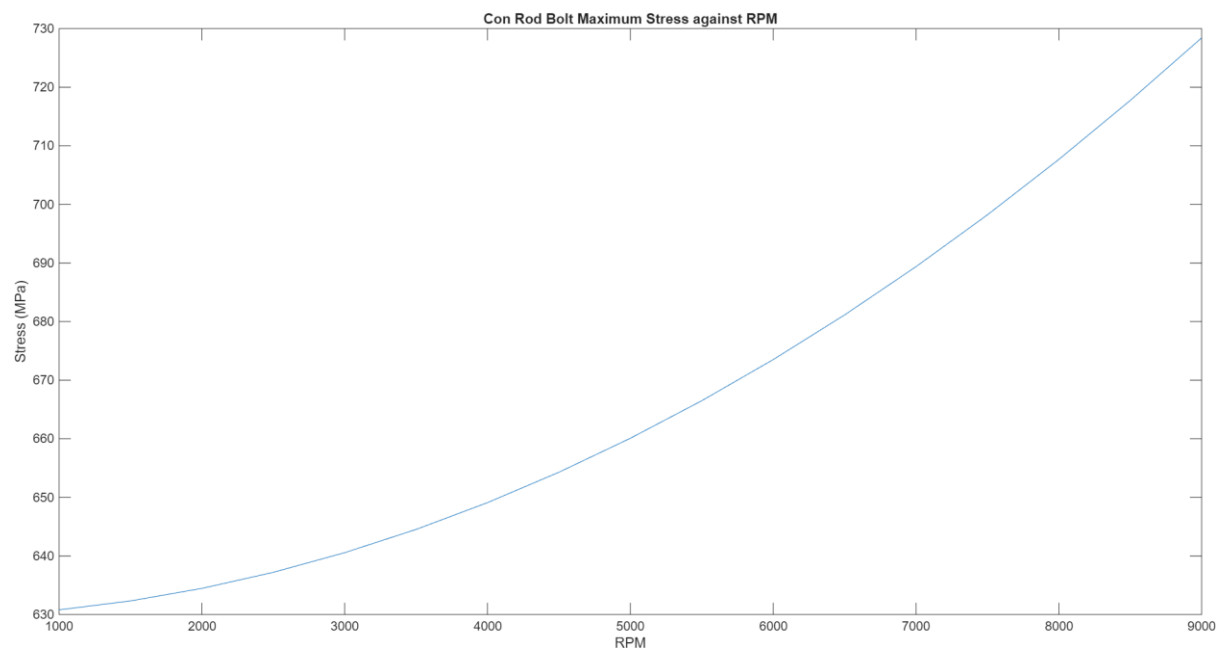


Figure 5 - Con Rod Bolt Maximum Stress against RPM

The non-linear relationship between RPM and con rod bolt stress is clear in Figure 5. Increasing the Ford engine's RPM range from 6000RPM to 9000RPM causes a maximum stress increase of 54.91MPa (from 673.51MPa to 728.42MPa). This occurs due to increased piston acceleration causing an increased deceleration at TDC which increases the tensile stress on the bolt. The increasing gradient at higher RPM demonstrates how inertial forces dominate the bolt's loading.

3.1. Changes to the Fatigue Safety Factors

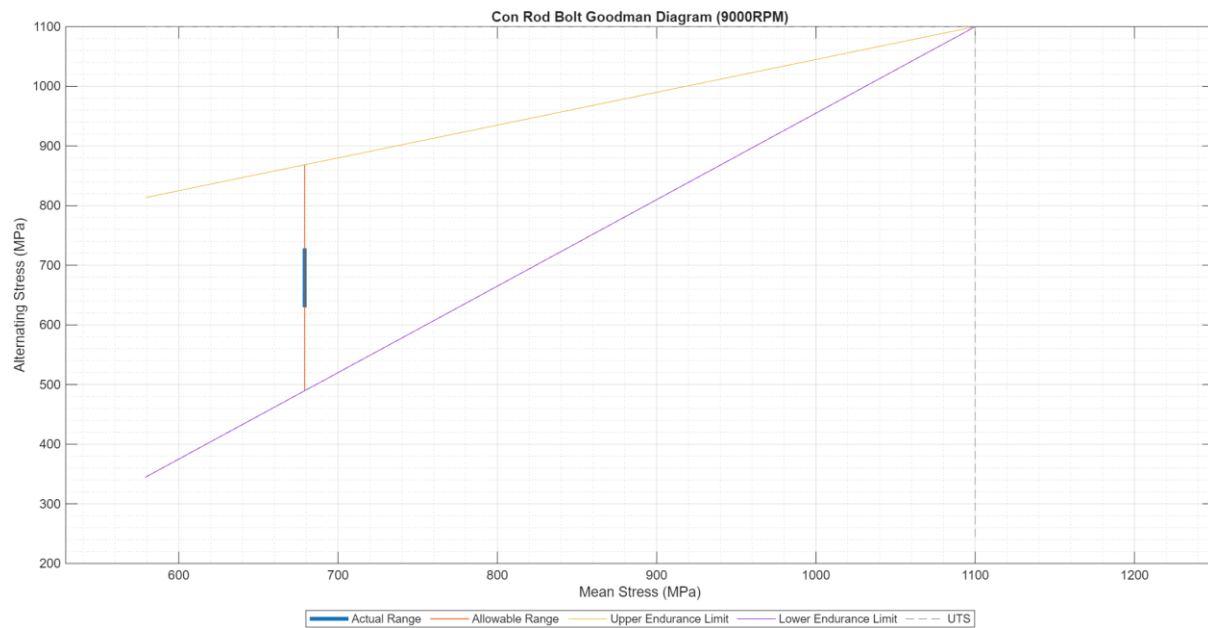


Figure 6 - Extended RPM Con Rod Bolt Goodman Diagram

What this means for the bolt is an FSF reduction from 9.19 to 3.83. The causes of this drop are clear in Figure 6. The bolt's actual range has shifted upwards within the allowable range due to the increased inertial loading and consequent higher tensile forces. The bolt is therefore operating much closer to its fatigue limit. However, this FSF isn't the full picture as it does not consider the stress concentrations caused by the bolt's threads.

Once the stress concentration factors are recalculated, the FSF drops further to 0.82. Under these conditions, the con rod bolt will fail. This is to be expected as the current bolt design isn't suitable for high speed operation; a sudden 3000RPM increase simply hasn't been accounted for. The further drop to the FSF after calculating stress concentration also suggests that cracks and failures will propagate from the thread.

4. Con Rod Bolt Redesign

Design changes need to be made to the bolt following the change in operating conditions. For this to occur, the bolt's length must be separated for each section to have an independent stiffness.

Table 1 - Bolt Length Distribution

Bolt Section	Length (mm)
Grip	1.93
Shank	7.71
Thread	11.62

The lengths in Table 1 have been scaled down from an M8x1.0 bolt from the ARP Catalog (Automotive Racing Products, 2026) to match the length of the original bolt. The thread takes over half the overall length due to the importance of thread length on load distribution. A longer thread will relieve the stress on the first engaged thread which is most prone to fatigue failure (Bailey, 1967). The grip length is relatively small as it's only needed to stop the bolt from swaying.

4.1. Shank Wasting

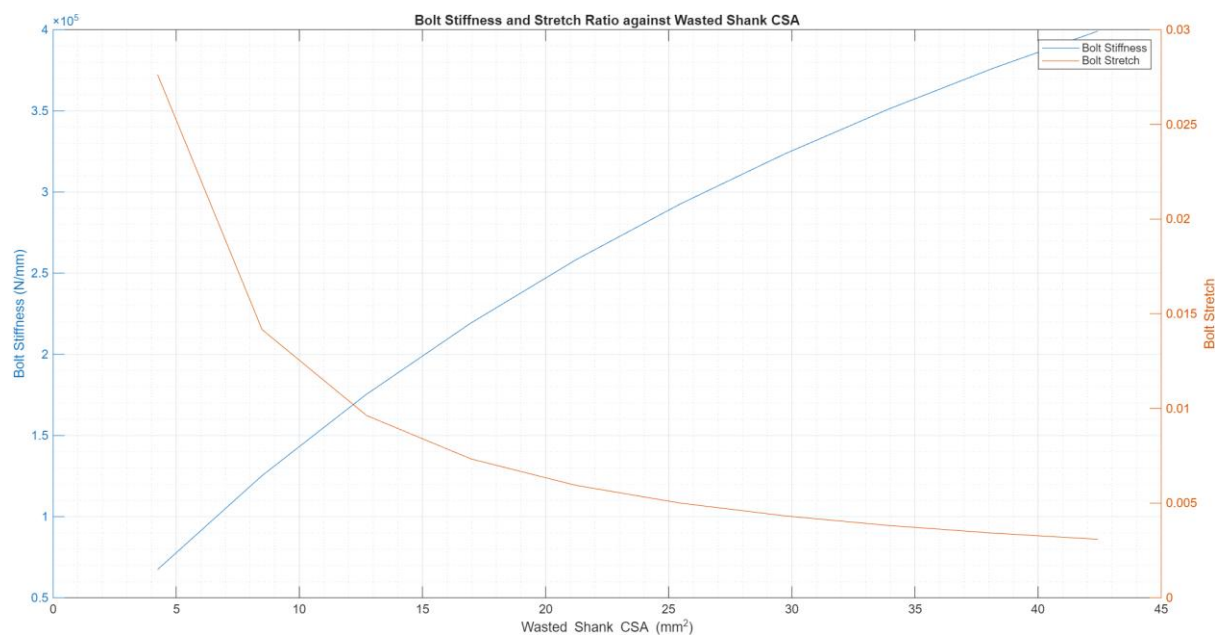


Figure 7 - Wasted Shank CSA's effect on Bolt Stiffness and Stretch Ratio

Shank wasting reduces the cross sectional area (CSA) of the mid-section of the bolt and has the effect of reducing overall bolt stiffness. This means that the bolt will experience less stress variation during cyclic loading. This relationship is displayed clearly in Figure 7. As the wasted shank's CSA decreases, the stiffness does the same leading to an increased stretching of the bolt because deflection is affected by stiffness and not strength (Budynas & Nisbett, 2014). This

will improve the stress distribution of the bolt as the deformation of the shank will relieve the stress concentrations on the threads and increase the bolt's longevity. However, there is a limit to how much the shank can be reduced that isn't considered by these calculations. Excessive reduction of the shank section's CSA can lead to a mean stress beyond the material's yield strength. Therefore, a balance must be struck between stiffness reduction and material strength.

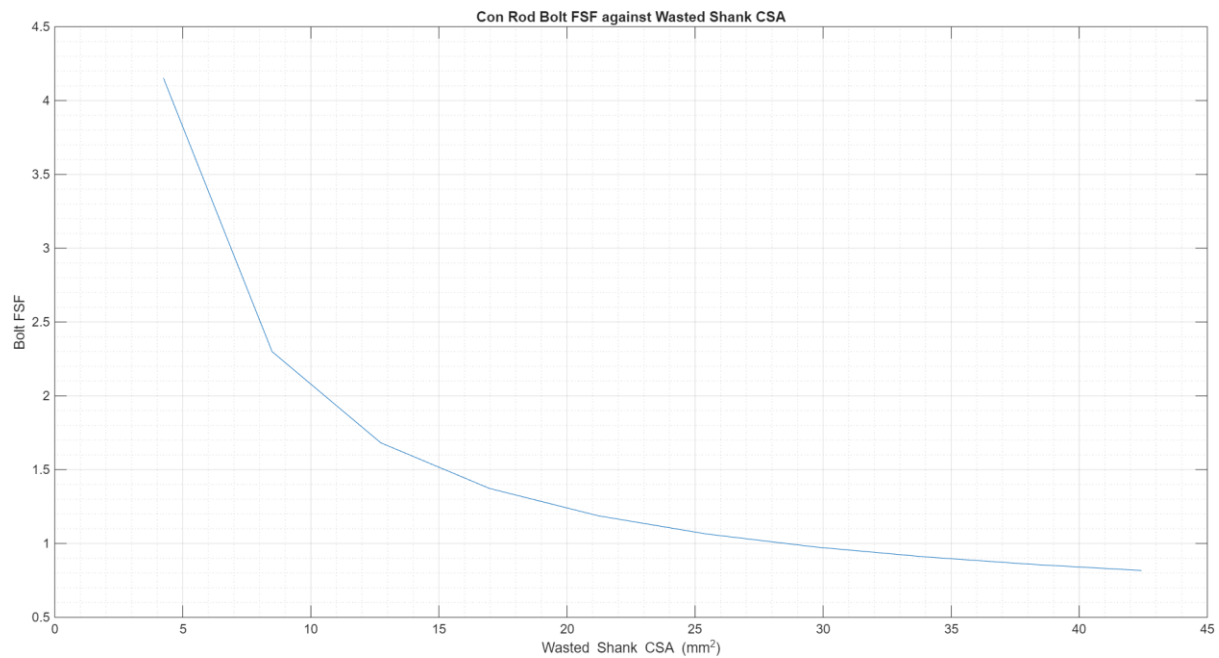


Figure 8 - Wasted Shank CSA's effect on FSF

The increased compliance of the bolt during cyclic loading is made clear in Figure 8. The FSF improves exponentially as the shank CSA decreases. However, the shank will only be reduced by 30% (Takashima, et al., 2003) as this is the most effective balance between stiffness and strength. This reduction of the shank CSA from 42.43mm² to 29.7mm² raises the bolt's FSF to 0.98 meaning fatigue failure would still occur. A further redesign is therefore necessary.

4.2. Material Selection

Another avenue for design alteration is the bolt's material. The current 10.9 grade bolt steel can be improved upon by a higher 12.9 grade bolt steel. As seen in table 1, the only major change is the ultimate tensile strength as steel retains a similar elasticity regardless of carbon content or alloying (Budynas & Nisbett, 2014). 12.9 grade bolt steel's similarity to the 10.9 grade bolt steel makes it a drop in replacement; there are little to no problems caused by this switch.

Table 2 - Original and Proposed Material Properties (Murakami, et al., 1997) (Budynas & Nisbett, 2014)

	10.9 Bolt Steel	12.9 Bolt Steel
Ultimate Tensile Strength	1100 MPa	1220 MPa
Modulus of Elasticity	200000 MPa	200000 MPa

4.3. Final Design Proposal

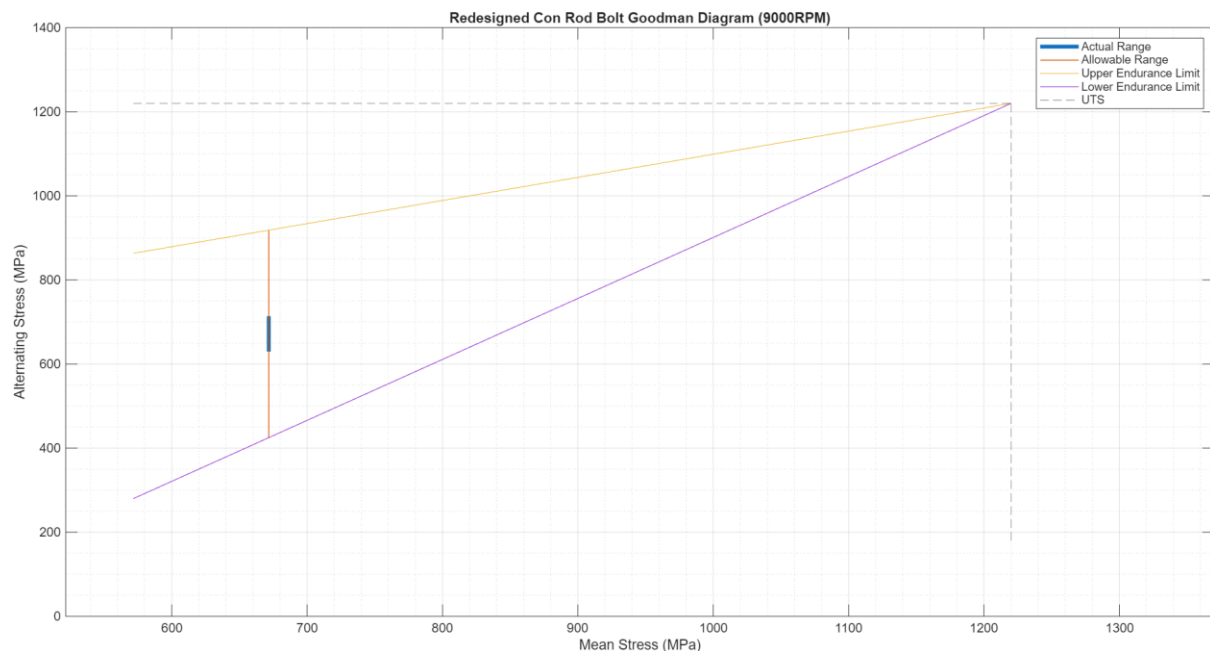


Figure 9 - Final Con Rod Design Goodman Diagram

The combination of the wasted shank and material change give the bolt's final design an FSF of 1.25. This still isn't overly conservative, but the bolt will not fail due to cyclic loading. However, its low tolerance means it may not be suitable for long-life use. Further FSF gains could be made by exploring design changes such as rolled threads which would further reduce the stress concentrations on the bolt thread.

5. Crankshaft Configurations

Another area of design divergence in engines is the crankshaft configuration which will have a major impact on torque delivery and vibrations. Flat plane crankshafts produce large inertia torques due to their pistons reciprocating in pairs. However, cross plane crankshafts have more uniform firing intervals resulting in a smoother torque delivery. This comes with the added benefit of reduced vibrations. The characteristics of these crankshafts will be explored in more detail within this section.

5.1. Inertia Torques

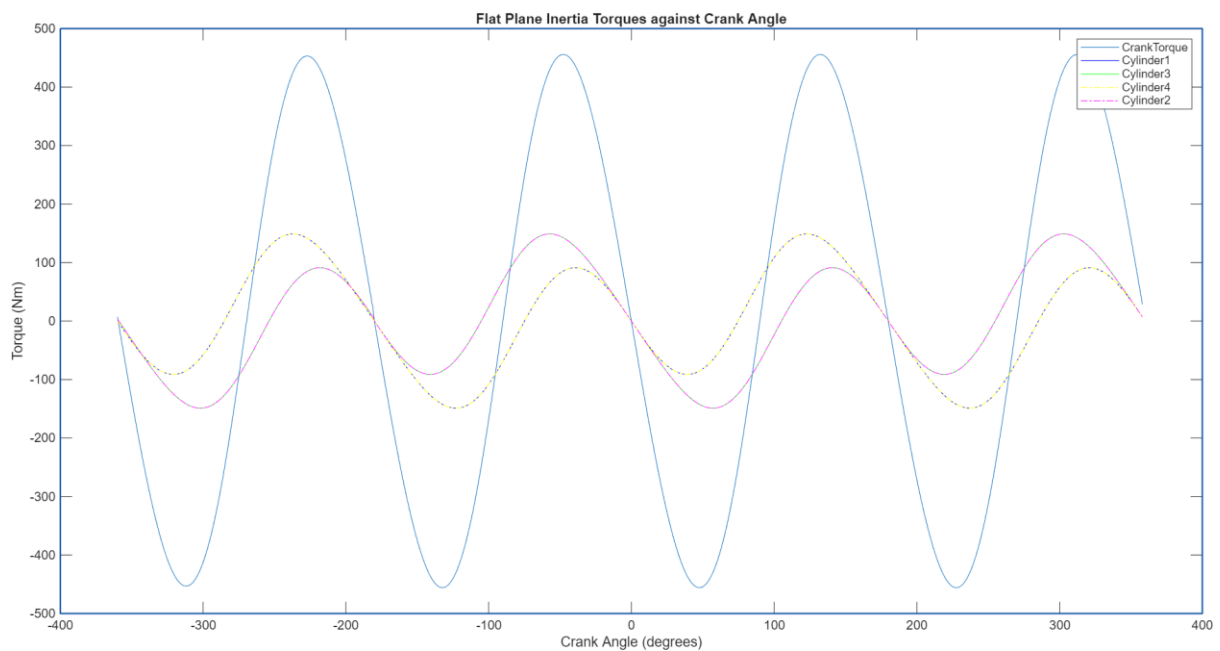


Figure 10 - Flat Plane Crankshaft Inertia Torque

The defining trait of the flat plane inertia torques seen in Figure 10 are the large positive and negative torque fluctuations in crank inertia torque. These are caused by the pistons' limited torque overlap allowing the overall torque to vary wildly. Despite the 180° offset between the cylinders, their paired reciprocating masses mean that little is done to cancel out their generated torques.

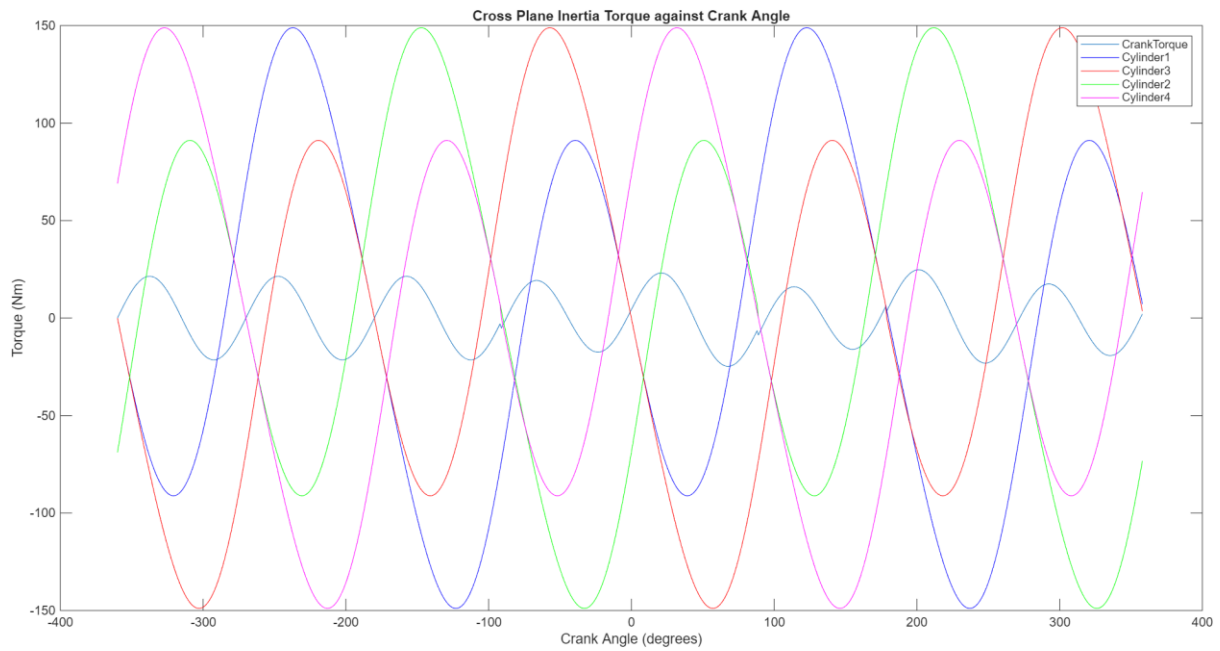


Figure 11 - Cross Plane Crankshaft Inertia Torque

The trends observed in the flat plane crankshaft are reversed completely in Figure 11. The cross plane's individual cylinder inertia torques fluctuate much more than the overall crank inertia torque. This is because in this configuration, the cylinders cancel each other out causing a much reduced overall inertia torque. This is achieved by arranging the pistons to fire at 270° , -180° , -90° , -180° (Ooi, 2024) meaning that combustion occurs every 90° and that two pistons are generating an upwards force as two are creating the opposite.

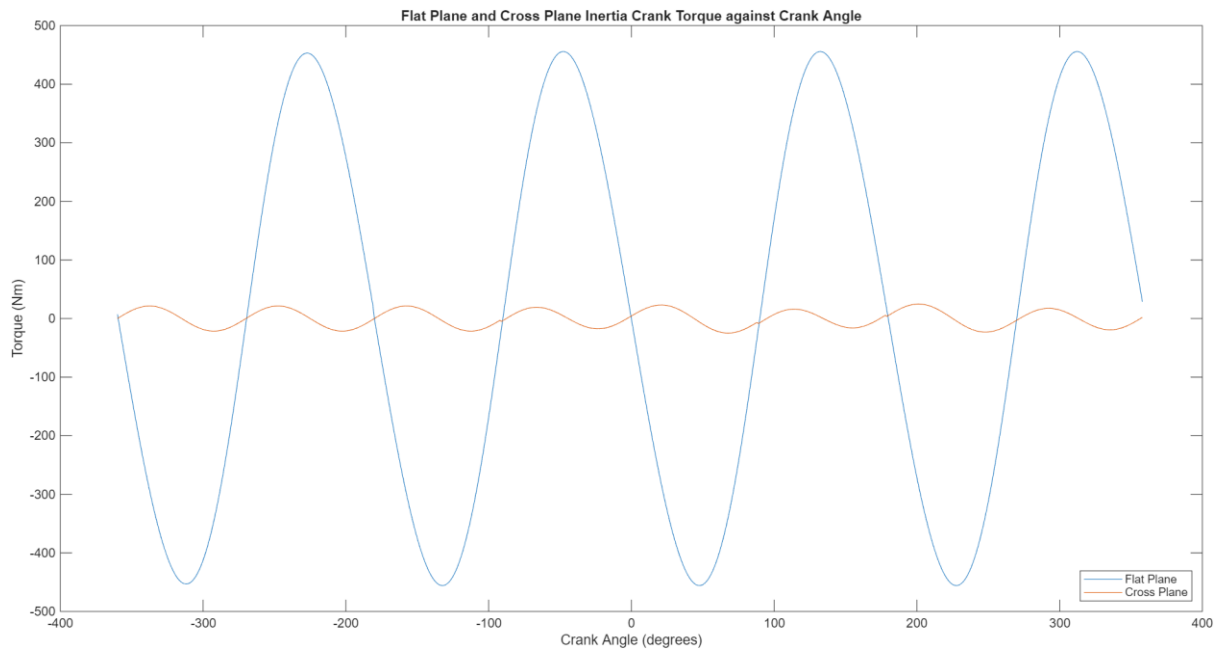


Figure 12 - Inertia Torque Comparison

The difference between both crankshaft configurations is prevalent in Figure 12. The flat plane's inertia torque at the crank fluctuates 21 times more than its cross plane counterpart (450Nm to 21Nm) at the same RPM (6000) in the same four cylinder engine. The changes made to the piston positions and firing order for the cross plane crankshaft make it much more appealing from an inertia perspective alone.

5.2. Gas Torque Comparison

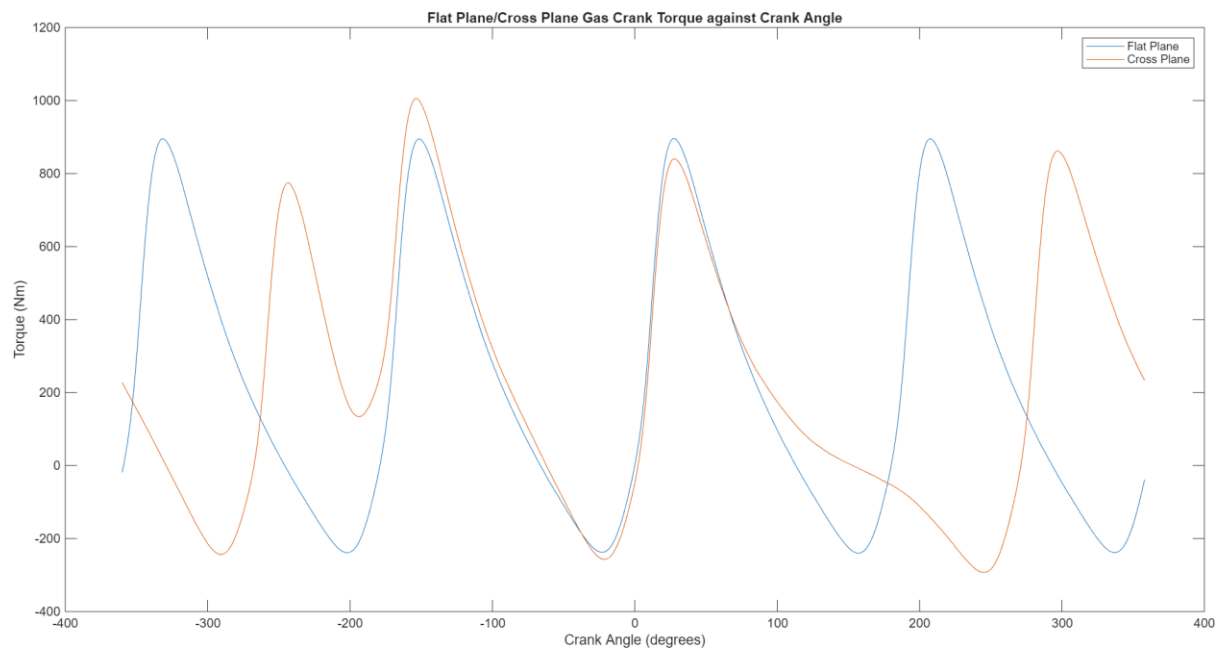


Figure 13 – Flat Plane and Cross Plane Gas Torque Comparison

While figures 10 and 11 conveyed the inertia torques of each crankshaft configuration, figure 13 portrays their gas torques. The flat plane exhibited a large fluctuation in inertia torque. However, its gas torque is evenly spread across the crank range and has the lowest range. On the other hand, the crank plane has differing and irregular torque peaks due to its revised firing order. The spacing between the cross plane gas torque peaks are also uneven due to the firing order. From a gas torque perspective, the flat plane crankshaft is the preferred option thanks to its metronomic and even peaks.

5.3. Total Crank Torque Comparison

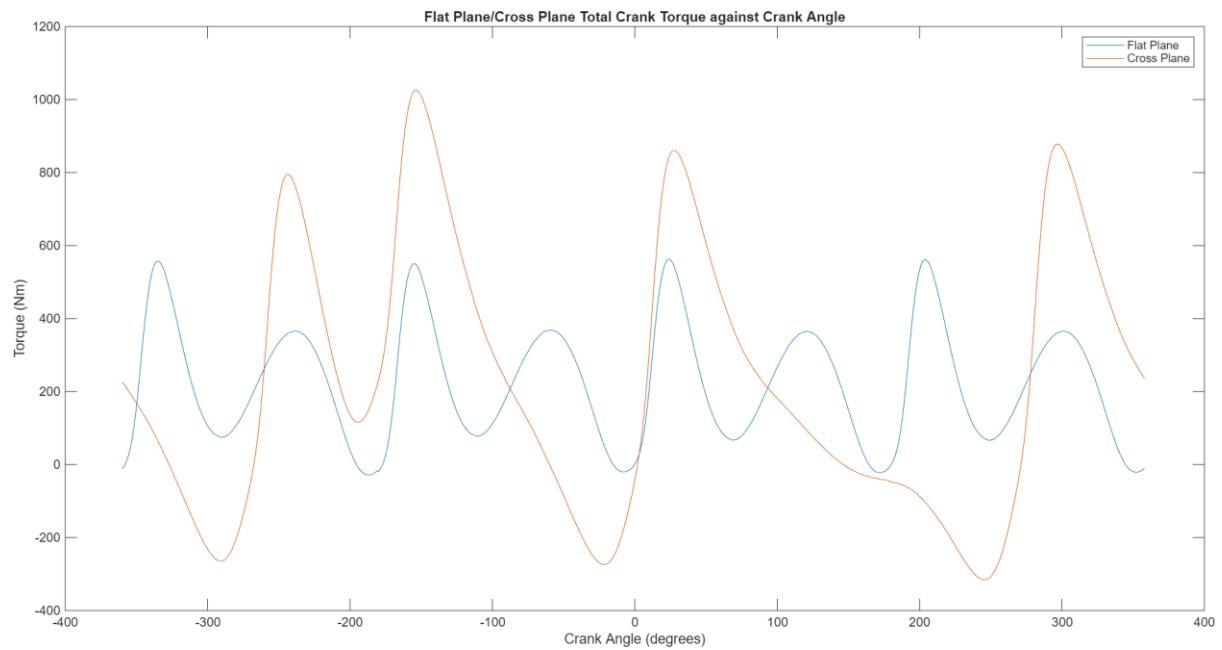


Figure 14 - Total Instantaneous Torque Comparison

The inertia and gas torques combine to create the total instantaneous torques of both crankshaft configurations in Figure 14. Despite the dramatic difference in inertia torques and relatively similar gas torques, the overall instantaneous torque looks favourable for the flat plane crankshaft. This is because the flat plane crankshaft's inertia and gas peaks are aligned to occur at separate points during the crank range. In the cross plane crankshaft's case, the gas and inertia torques overlap which causes the erratic fluctuations in torque.

5.4. Torque Curve Validation

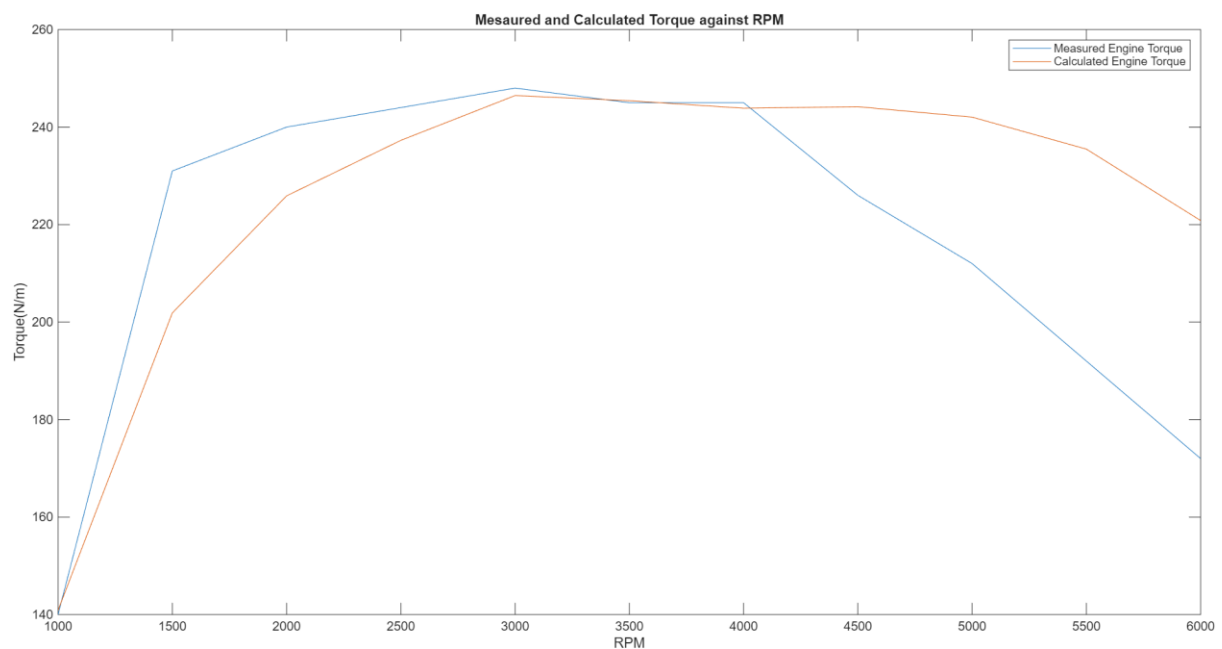


Figure 15 - Measured and Calculated Torque

Figure 15 visualises the calculated torque curve for the flat plane crankshaft configuration and compares it to a measured Ford 1.6GDTi torque curve (Ford Component Sales, n.d.). The flat plane crankshaft has been selected for comparison to match the actual engine.

The correlation between 3000RPM and 4000RPM is encouraging as both curves peak in the same area at a similar RPM. This is most likely because at this RPM range, the measured engine is operating at its highest volumetric efficiency; something that is assumed for the model throughout the RPM range. However, the measured engine torque curve is noticeably steeper at low RPM. This may be a result of the engine's turbocharger having an effect on the combustion pressure at low RPM. At high RPM the model overpredicts. This is because the model isn't considering the mechanical and pumping losses that occur more prevalently at higher RPM thus reducing its accuracy in this region.

6. Conclusion

Under baseline operating conditions, it is clear that both the con rod and gudgeon pin operate comfortably from a fatigue perspective. The con rod bolt is the limiting factor as the component that operates closest to its fatigue limit. This smaller tolerance is punished when the engine speed is raised beyond what the con rod bolt was originally designed for. The resulting FSF of 0.82 has then been tackled with two design changes. Firstly, a wasted shank section to increase compliance and reduce the bolt's alternating stress. Secondly, a change of material to a stronger grade of steel to increase the UTS and endurance limits. Combined, these changes have raised the FSF back up to 1.25. While this value is satisfactory, it remains vulnerable to real world variability. Finally, crankshaft configurations have been explored to observe the torque implications of a change to such a fundamental component. Both have their merits but the flat plane seems the stronger overall package from an automotive perspective.

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