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ANALYTICAL ASSESMENT OF ENGINE INTERNAL COMPONENTS
(ACHA6006)

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1 INTRODUCTION

This report calculates the relevant structural forces on the connecting rod assembly of the Ford Sigma engine and justifies sound judgement of its limitations or overestimations with respect to stresses present, fatigue life and safety factor. With correct calculation, justified design modification of the connecting rod has been made to meet or exceed the requirement of the engine's performance in its common use case in addition to increased engine speeds outside of the normal use case.

The torque produced by the engine has been calculated with consideration to the standard configuration and for a cross-plane crank shaft arrangement. Characteristic changes derived from the differences has been discussed.

3 ASSUMPTIONS

The report assumes uniform pressure distribution on the account that all contact surfaces are perfectly fitting. The study assumes ideal assembly conditions with no movement that would be expected due to tolerance or backlash. It assumes the piston is moving uniformly in its axial plane with no rocking in the cylinder and no input from skirt contact, if any. The study neglects any secondary dynamics, especially at higher RPM, such as bending considerations within the component.

The calculations assume an adiabatic relationship between contact surfaces, resulting in zero change in material properties with a change of temperature(Morad & Alrajhi, 2014).

The real gas pressure data will have accounted for this with respects to the data it exposes, but further calculations made within this study are assumptions on these fixed real-world variables.

4.1 PISTON FORCES

The piston position has been plotted (Figure 1) demonstrating the sinusoidal wave pattern due to the engine geometry. First and second derivatives of the angular velocity result in the pistons linear velocity and linear acceleration, Figure 2. Given Newtons Second Law of motion, $F=ma$, our maximum forces exerted on the piston should be observed at the highest rates of acceleration (and deceleration).

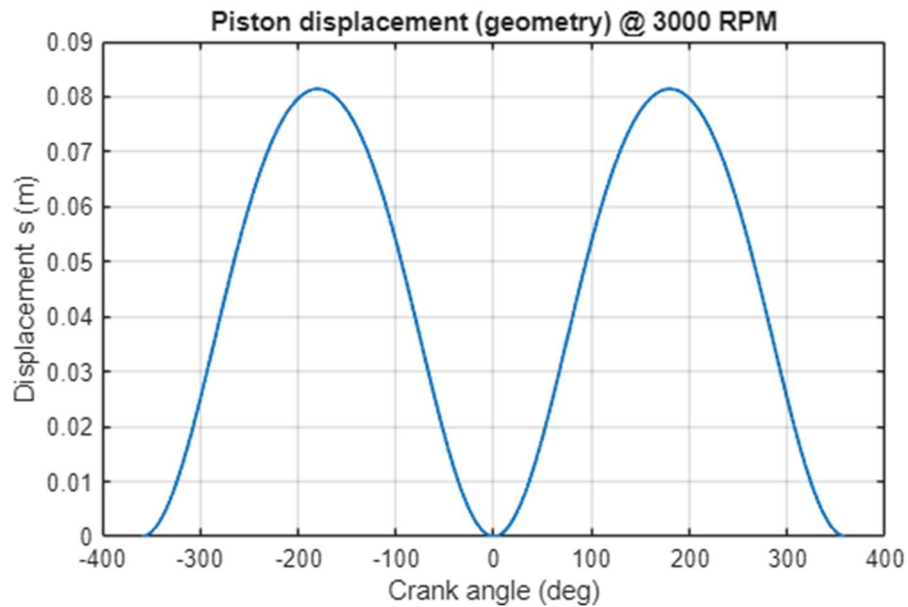


Figure 1 - Pison position per degree

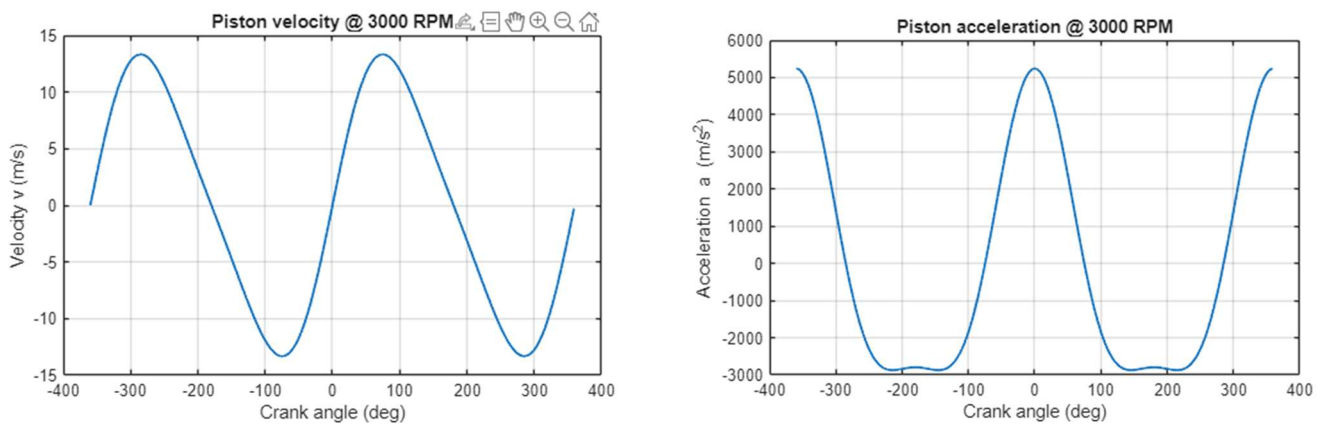


Figure 2 - Piston linear rate of velocity and acceleration

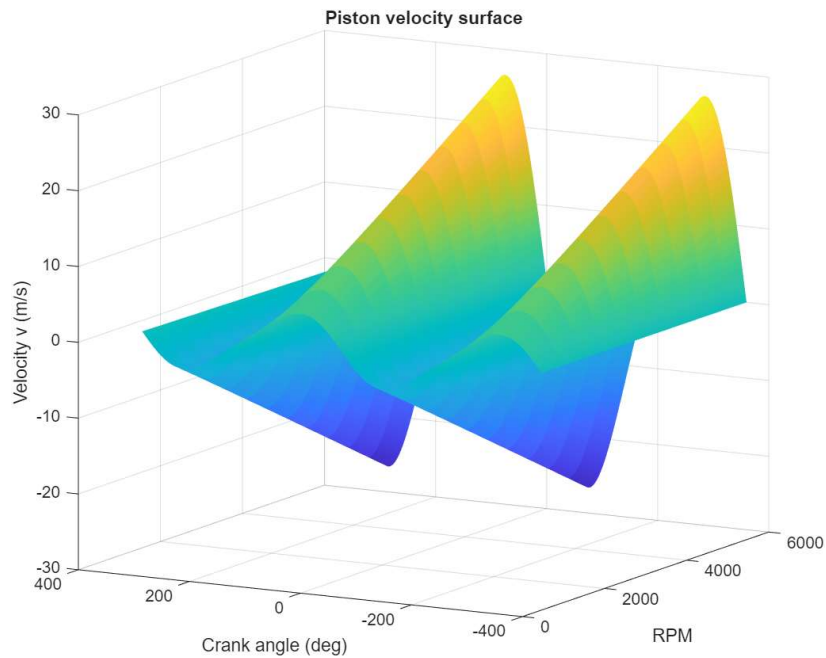


Figure 3 - Piston velocity across RPM range

Figure 3 demonstrates the fixed geometry across the RPM range with expected results whereby the piston velocity is proportional to the crank speed.

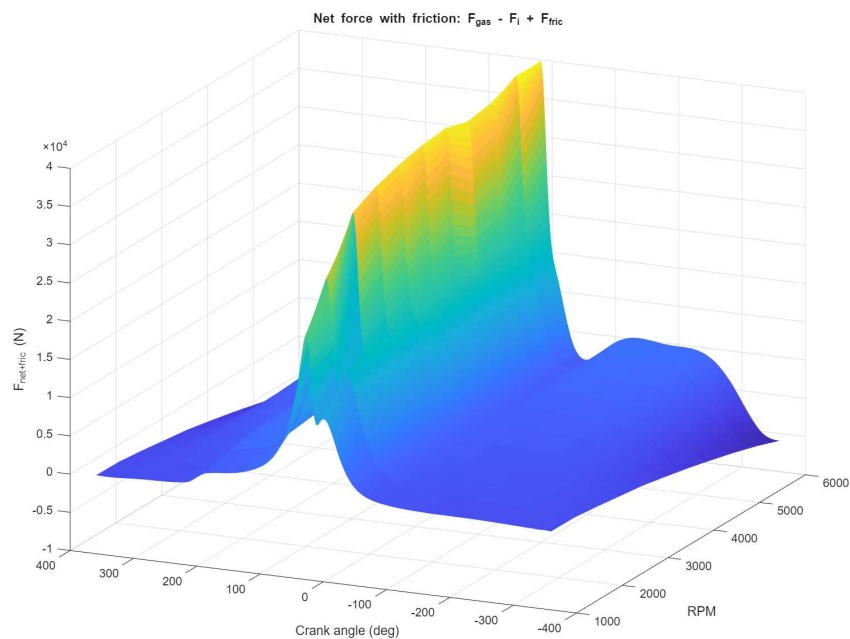


Figure 4 - Net Force

The gas pressure data has been used to accurately plot the net forces acting on the piston. The addition of calculated forces added by angular friction from the piston rings has also been included to generate a comprehensive net force plot, see figure 4.

The angular friction presented by the piston rings is not linear, as the conrod angle, proportionate to the crank angle, causes angular velocity on the piston rings against the cylinder wall. The peak frictional forces are consistent, between 29° - 36° , after TDC, and have correlation to the peak pressure angle. It can be concluded that frictional

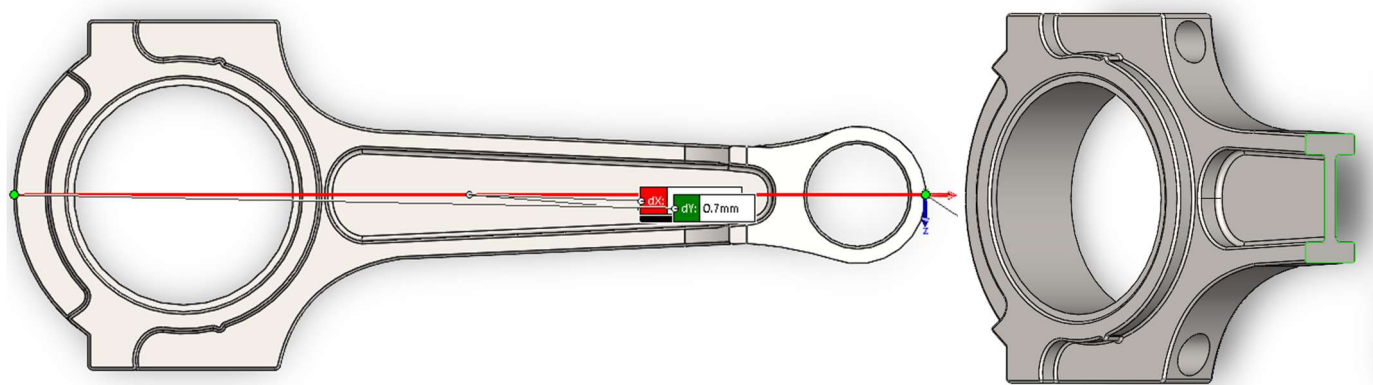
forces are directly harmonised to gas pressures and not where crank angle is maximum relative to the normal of the piston ring or velocity is at maximum.

RPM	Ffric_MaxAbs_N	Ffric_Angle_deg	Fnet_Angle_deg
1000	341.94	33	28
1500	477.22	36	31
2000	514.44	32	26
2500	535.28	33	25
3000	552.44	33	25
3500	544.79	32	24
4000	543.29	32	23
4500	547.7	33	24
5000	549	33	24
5500	525.48	30	21
6000	483.7	29	19

Table 1- Frictional Force detailed table

4.2 MIDPOINT OF CONNECTING ROD

The net pressure has been used to calculate the stresses at the midpoint of the conrod. The conrod had a midpoint length from big and small end centres defined as 66mm, resulting in a cross-sectional area of 196.39mm².



4.2.1 FORCE PROFILE

Peak net force was found at engine speed of 6000RPM. Most of the forces acting on the conrod are consistently positive values, with only the inertia working as an opposing force in the phases around BDC.

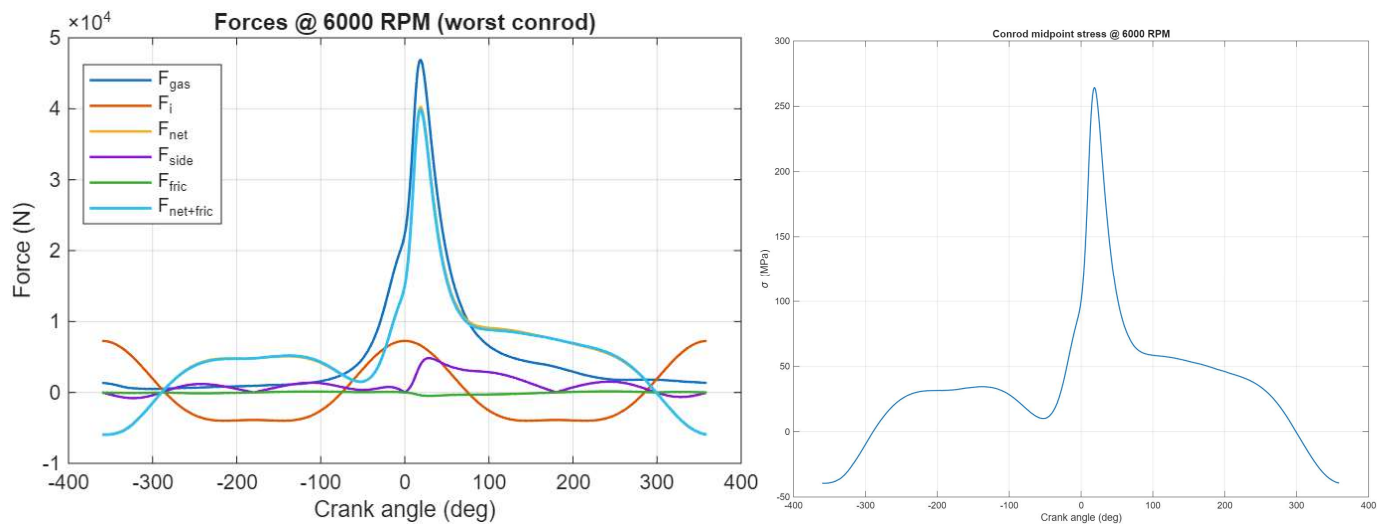


Figure 5 - Conrod forces at 6000RPM

The peak conrod force occurs at a crank angle of 19°. The maximum force calculated at the midpoint of the conrod was found to be 39854N and accounts for total net forces, table 2.

RPM	ConrodMidpointForce_N
1000	22041
1500	27571
2000	34391
2500	36037
3000	37089
3500	37873
4000	38306
4500	37351
5000	37957
5500	39709
6000	39854

Table 2 – Conrod midpoint forces across RPM range

RPM	ConrodPeakAbsStress_MPa	ConrodStressAtPeak_MPa	ConrodPeakStressAngle_deg	PinPeakAbsStress_MPa	PinStressAtPeak_MPa	PinPeakStressAngle_deg	BoltPeakAbsStress_MPa	BoltStressAtPeak_MPa
1000	112.23	112.23	28	136.64	136.64	28	891.06	891.06
1500	140.39	140.39	31	171.05	171.05	31	892.93	892.93
2000	175.11	175.11	26	213.56	213.56	26	894.43	894.43
2500	183.5	183.5	25	224.1	224.1	25	896.35	896.35
3000	188.85	188.85	25	231.03	231.03	25	898.71	898.71
3500	192.84	192.84	24	236.39	236.39	24	901.49	901.49
4000	195.05	195.05	23	239.68	239.68	23	904.71	904.71
4500	190.19	190.19	24	234.37	234.37	24	908.35	908.35
5000	193.28	193.28	24	238.85	238.85	24	912.42	912.42
5500	202.19	202.19	21	250.66	250.66	21	916.91	916.91
6000	202.93	202.93	19	252.59	252.59	19	921.84	921.84

Table 3 - Conrod forces and stresses

The peak compressive stress was calculated to be 202.93MPa, and tensile stress 30MPa. These stresses are happening at the highest RPM range as would be expected, where the inertia forces are greatest. There is direct correlation to the stress and the peak force, as the maximum stress occurs at the same crank angle, 19°m as the peak force is achieved.

4.2.2 CONROD GOODMAN DIAGRAM

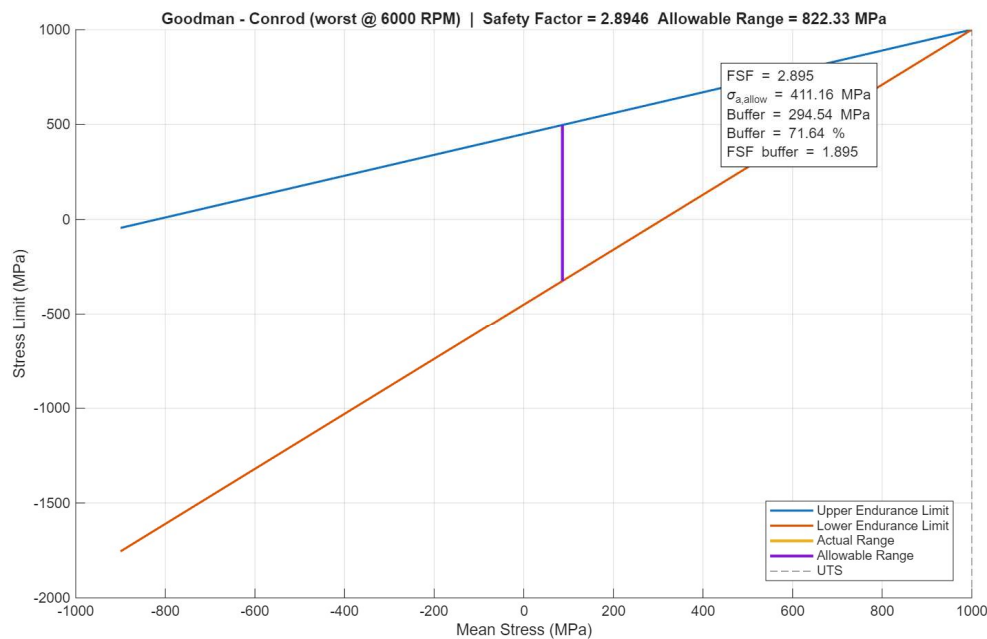


Figure 6 – Conrod Goodman diagram

The goodman diagram visually demonstrates the limits of the material under this load case. The lowest factor of safety is located below the goodman line indicating infinite life with a large buffer towards alternating stresses of 71.64%. Overall factor of safety calculated as 2.865.

4.2.3 CONROD FATIGUE SAFETY FACTOR

RPM	FSF_ConrodMid
1000	5.553
1500	4.428
2000	3.537
2500	3.361
3000	3.251
3500	3.166
4000	3.111
4500	3.165
5000	3.091
5500	2.934
6000	2.895

Table 4 - Conrod fatigue safety factor

The calculation of the fatigue safety factor for the conrod found that the conrod operated well within the fatigue safety factor limits for its allocated engine speed. Fatigue safety factor was lowest at peak RPM proving the fatigue safety factor is completely reliant on conrod net forces.

4.3 GUDGEON PIN AT POINT A

The calculated forces from the conrod have been used to calculate the stresses transferred into the gudgeon pin. The forces have been calculated per degree, a full cyclic load graph can be produced.

RPM	PinPointAForce_N	PinPointAAngle_deg
1000	11032	28
1500	13809	31
2000	17240	26
2500	18090	25
3000	18648	25
3500	19078	24
4000	19341	23
4500	18910	24
5000	19269	24
5500	20219	21
6000	20370	19

Table 5 - Gudgeon pin forces

The forces seen at point A see an increase in 85% across the RPM range. As the RPM increase, the peak force advances closer to TDC.

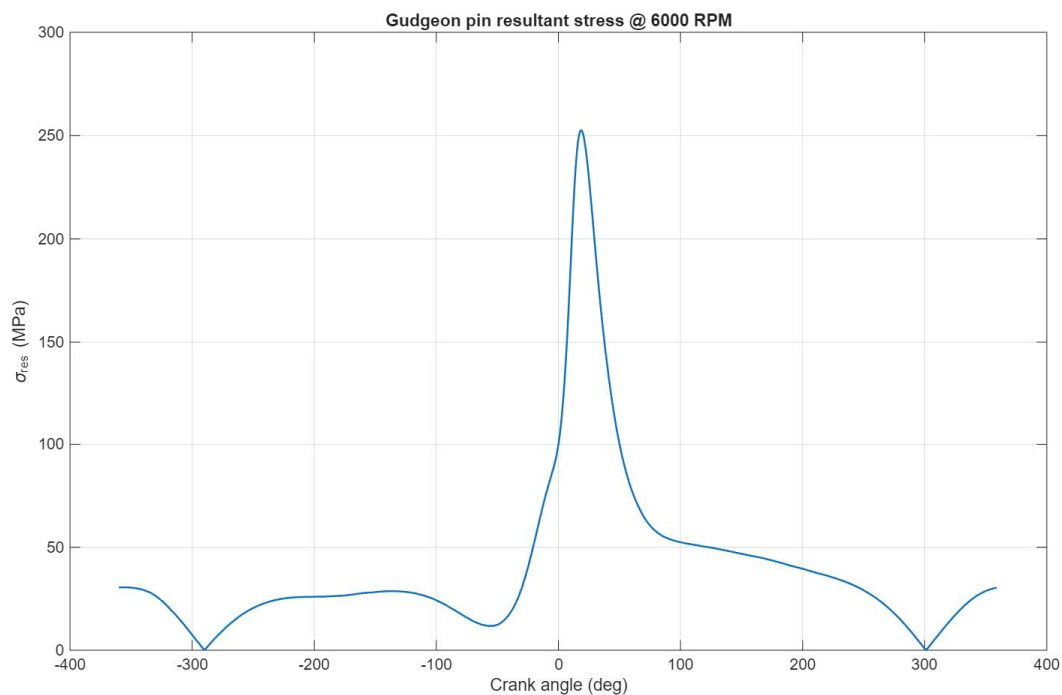


Figure 7 - Gudgeon pin forces across crank cycle

Figure 7 demonstrates how the force is proportional to the conrod forces, as the gudgeon pin forces are a derivative of the conrod and piston.

4.3.1 GUDGEON PIN GOODMAN DIAGRAM

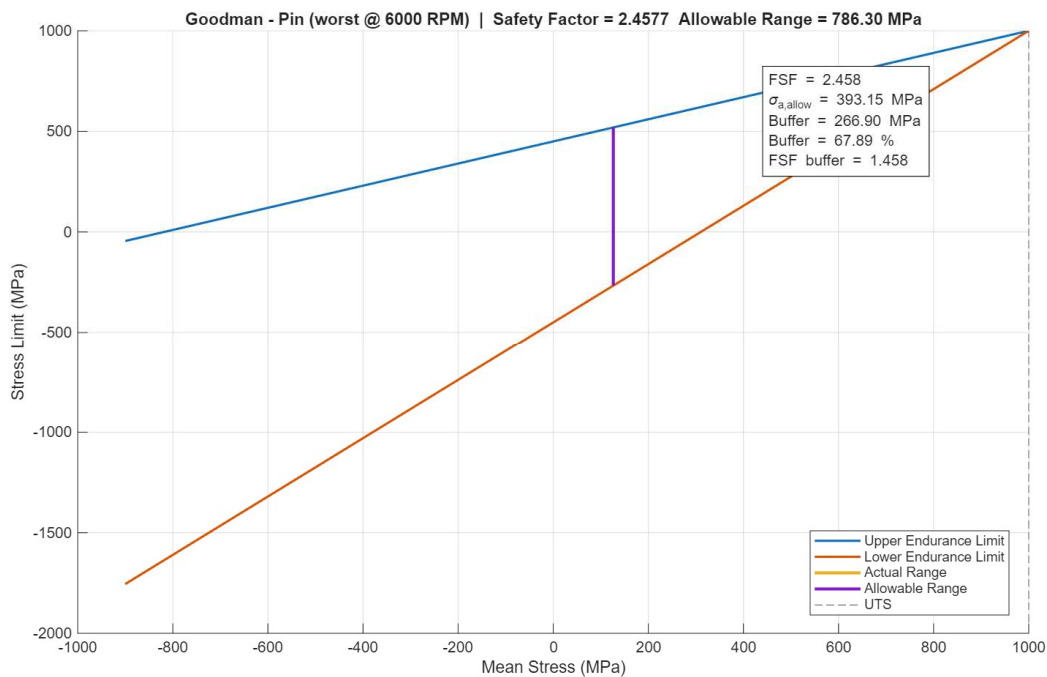


Figure 8 - Gudgeon pin goodman diagram

The goodman diagram for the gudgeon pin demonstrates that the gudgeon pin is working within an infinite life range with a 67.89% buffer before reaching a safety factor of 1. The gudgeon pin retrieved a factor of safety value of 2.458.

4.3.2 GUDGEON PIN FATIGUE SAFETY FACTOR

RPM	FSF_GudgeonPin
1000	4.563
1500	3.637
2000	2.907
2500	2.77
3000	2.687
3500	2.626
4000	2.59
4500	2.649
5000	2.599
5500	2.477
6000	2.458

Table 6 - Gudgeon pin fatigue safety factor across RPM

Table shows the safety factor of the pin inversely proportional to engine speed. It can be concluded that the safety factor of the gudgeon pin is related to the inertial force.

4.4 CONROD BOLTS

Conrod bolts assume they are tightened to 90% UTS.

RPM	ConrodBoltForce_N	ConrodBoltAngle_deg
1000	33849	28
1500	34438	31
2000	35163	26
2500	35338	25
3000	35450	25
3500	35534	24
4000	35580	23
4500	35478	24
5000	35543	24
5500	35729	21
6000	35745	19

Table 7 - Conrod bolt peak force across RPM

The con rod bolts experience their highest force at the maximum engine speed. As force increases with engine speed, the peak force is experienced earlier after TDC, see table 7.

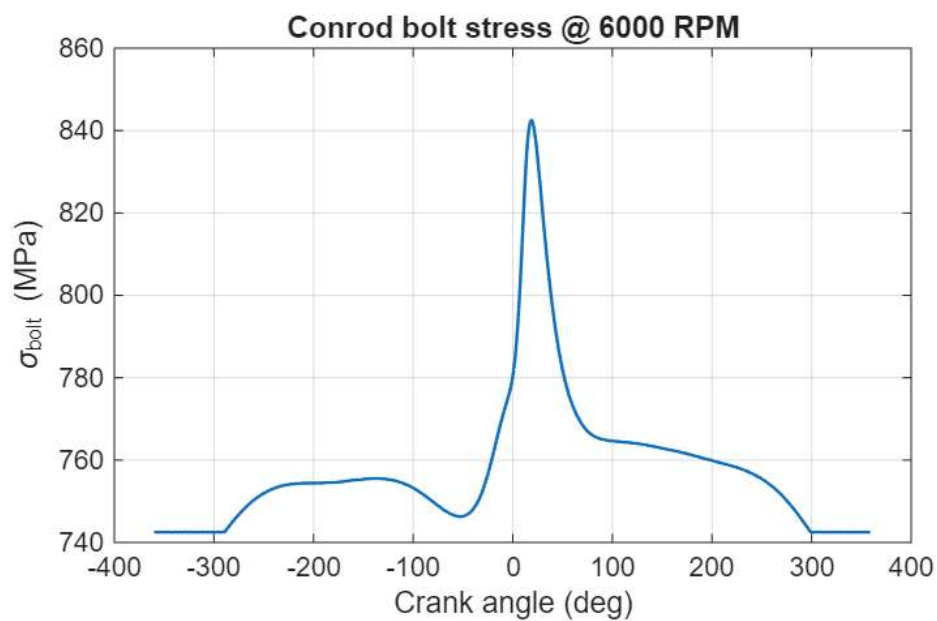


Figure 9 - Conrod bolt stress across crank cycle

Figure 9 above shows correlation to the forces experienced by the conrod and the pin, aligning with the correlation between the components in the system.

4.4.1 GUDGEON PIN GOODMAN DIAGRAM

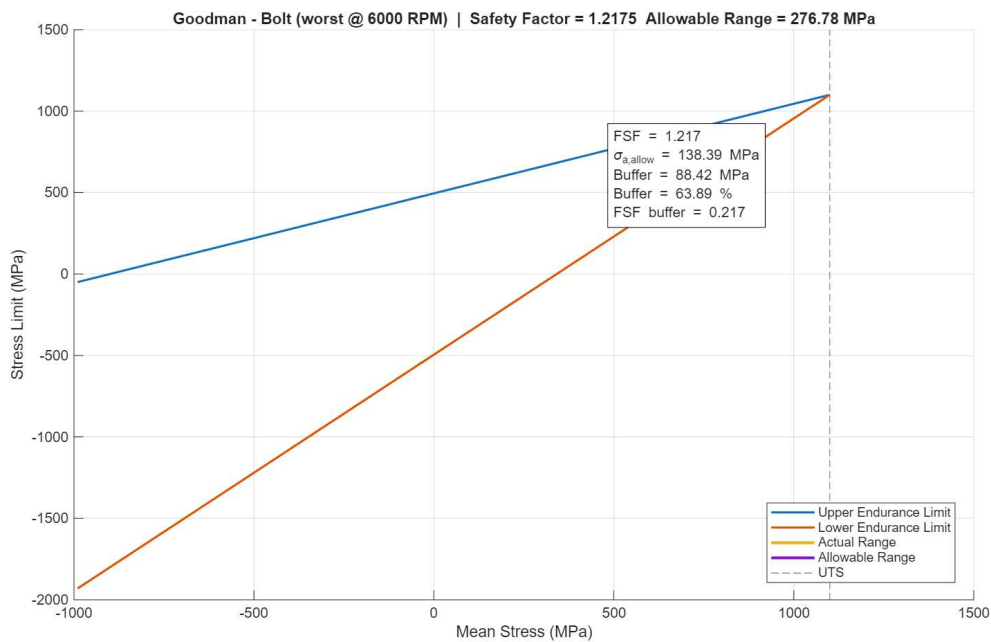


Figure 10 - gudgeon pin goodman diagram

The bolts were calculated to being the weakest component of the system, retrieving a safety factor of only 1.217. the buffer was found to be 63.89%. This is close to the limit of safety for this component considering these calculations are for a perfect system. With engine wear and external forces from potholes for example, there is a minute possibility that the bolts would fail under normal operating conditions (Tien Han et al., 2021).

The bolts have a more consistent factor of safety throughout the engine speed range due to the preload on the bolts, in this simulated case, 90% of the UTS.

4.4.2 CONROD BOLTS FATIGUE SAFETY FACTOR

RPM	FSF_ConrodBolt
1000	1.323
1500	1.288
2000	1.248
2500	1.239
3000	1.233
3500	1.228
4000	1.226
4500	1.231
5000	1.228
5500	1.218
6000	1.217

Figure 11 - Conrod bolt fatigue safety factor

The safety factor of the conrod bolts is close to one throughout the entire engine speed range. The bolts are exposed to rotational and reciprocating force. Although, given the low safety factor throughout the entire range, the dominant force loading can be attributed to the bolt preload.

4.5 DISCUSSION OF OVER-ENGINEERING

RPM	FSF_ConrodMid	FSF_GudgeonPin	FSF_ConrodBolt
1000	5.553	4.563	1.323
1500	4.428	3.637	1.288
2000	3.537	2.907	1.248
2500	3.361	2.77	1.239
3000	3.251	2.687	1.233
3500	3.166	2.626	1.228
4000	3.111	2.59	1.226
4500	3.165	2.649	1.231
5000	3.091	2.599	1.228
5500	2.934	2.477	1.218
6000	2.895	2.458	1.217

Figure 12 - Component fatigue safety factor comparison

All the components in the assembly achieved infinite life, exceeding a fatigue safety factor of one. The conrod bolts have the lowest level of safety, with only a difference of 0.217 at the RPM peak. The fatigue safety factor for the bolts is close to the value of one all the way throughout the RPM range. Considering this, it can be assumed that the engine forces are having less of an effect on the conrod bolts than the other components in the system. The preloading force appears to be a dominant stressor within the bolts, alluding to a poor initial design or material choice.

5 EXTENDED RPM ANALYSIS (9000 RPM)

By means of extrapolation the engine speed range has been extended to 9000RPM. A line of best fit was made from the existing gas data and the gradient traced across the RPM range to 9000RPM.

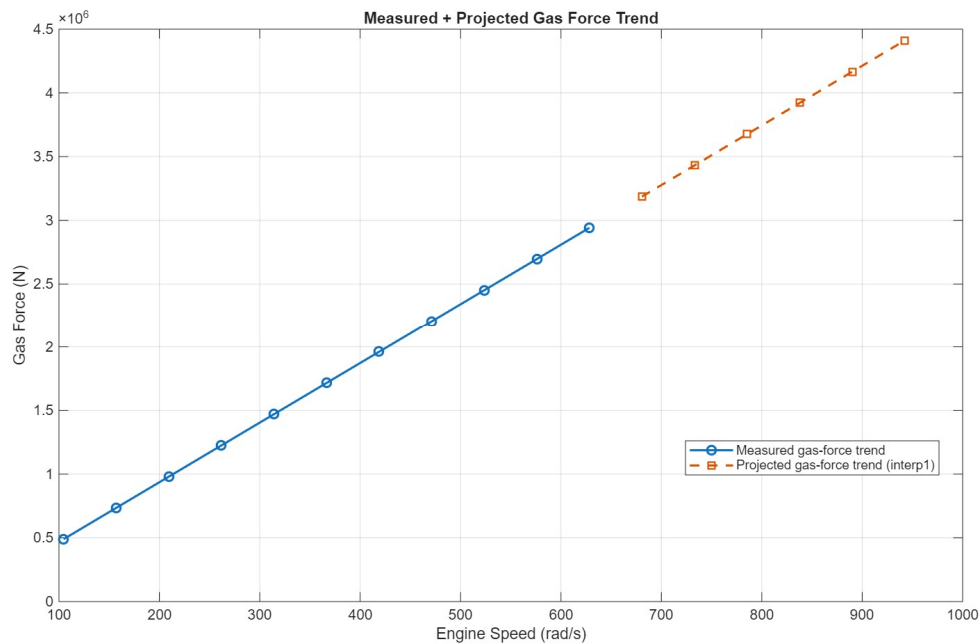


Figure 13 - Extrapolation of gas force curve

The resultant forces for velocity and acceleration enable the safety factor to be analysed for the same components at the new engine speed.

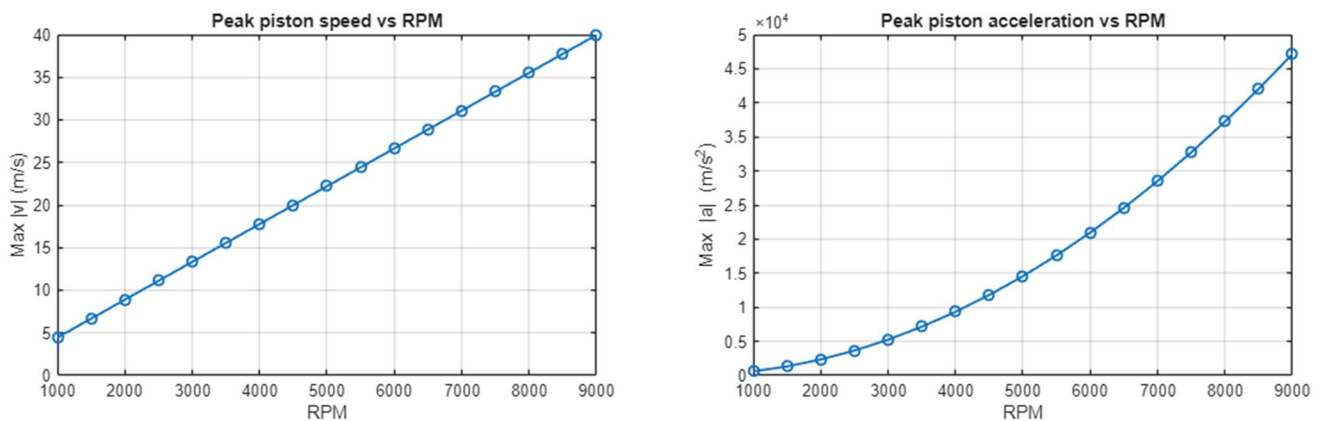


Figure 14 - Piston velocities

The fixed geometry extends the velocity and acceleration of the piston proportionally. A new net force was simulated using the same method as previous.

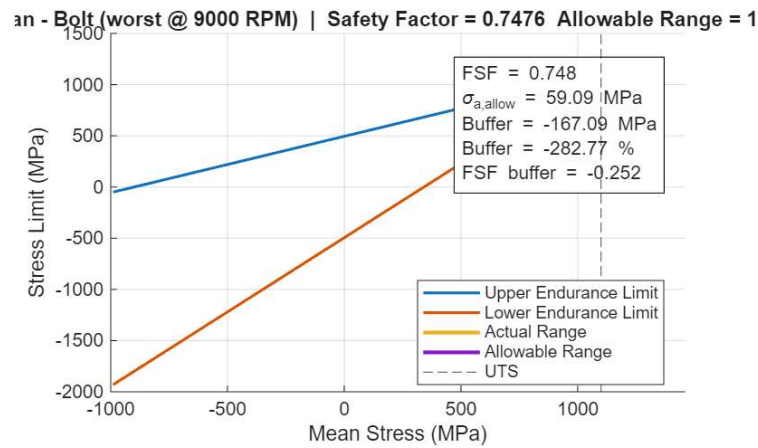


Figure 15 - bolt safety factor at 9000RPM

With the new forces, the bolt safety factor was calculated to be 0.748. Given the calculated safety factor being below zero, the conrod bolts can be deemed to have zero life and will fail at the new maximum engine speed of 9000RPM.

RPM	FSF_ConrodMid	FSF_GudgeonPin	FSF_ConrodBolt
1000	5.553	4.563	1.323
1500	4.428	3.637	1.288
2000	3.537	2.907	1.248
2500	3.361	2.77	1.239
3000	3.251	2.687	1.233
3500	3.166	2.626	1.228
4000	3.111	2.59	1.226
4500	3.165	2.649	1.231
5000	3.091	2.599	1.228
5500	2.934	2.477	1.218
6000	2.895	2.458	1.217
6500	2.892	2.472	1.22
7000	2.646	2.276	1.202
7500	2.356	2.041	1.177
8000	1.665	1.436	1.08
8500	1.033	0.878	0.919
9000	0.653	0.55	0.748

Table 8 - Increased RPM range fatigue safety factors

The zero-life failure at 9000RPM, it is justified that the maximum safe engine speed for these bolts to be 8000RPM. It can be observed that the gudgeon pin shares this engine speed limit with a safety factor receding below zero between 8000-8500RPM. The conrod is only overengineered from the system marginally, with the conrod limit only being ~500RPM greater than both, where the safety factor falls below one after 8500RPM. It could be argued that the assembly has been designed well in unison, aimed at working within its normal operating range. In hindsight, a lack of over engineering for these components makes it unsuitable for engine speed increase without modification. All components should be modified to some extent when exceeding 8000RPM.

6.1 REDESIGN CRITERIA

The target fatigue safety factor is to meet or exceed the standard bolt fatigue safety factor, 1.217. A material choice has not been defined; thus, an alternative material may be explored and the effects of different material properties considered.

There are some packaging limitations that should be considered when redesigning engine components. To ensure the fit is the same, and replacement of the bolts is a simple remove and replace scenario, the overall dimensions should remain the same; M8 bolt, overall length not exceeding 35mm.

6.2 GEOMETRIC MODIFICATION

The conrod assembly, consisting of conrod, piston, gudgeon pin and bolt all play an important role in the safe transfer of chemical energy to kinetic energy. When modifying the engine, the cost should be considered. Not only are the bolts the smallest component of the assembly, but they are also the easiest to access, where the other components would need more comprehensive engine teardown for replacement. Considering female thread depth on the conrod, limitations on how much thread can be accepted from the bolt exist. There will likely be a predefined limit to the bolt length that can be used and a maximum diameter to ensure enough “land” is left between the edge of the hole and edge of conrod material.

The current script enables quick modification to the conrod bolts dimensions to reverse engineer the bolt to bring the conrod bolt within a safe working life value. Considering the limitations with bolt size, a waisted neck design has been utilised to reduce the stress concentration factor.

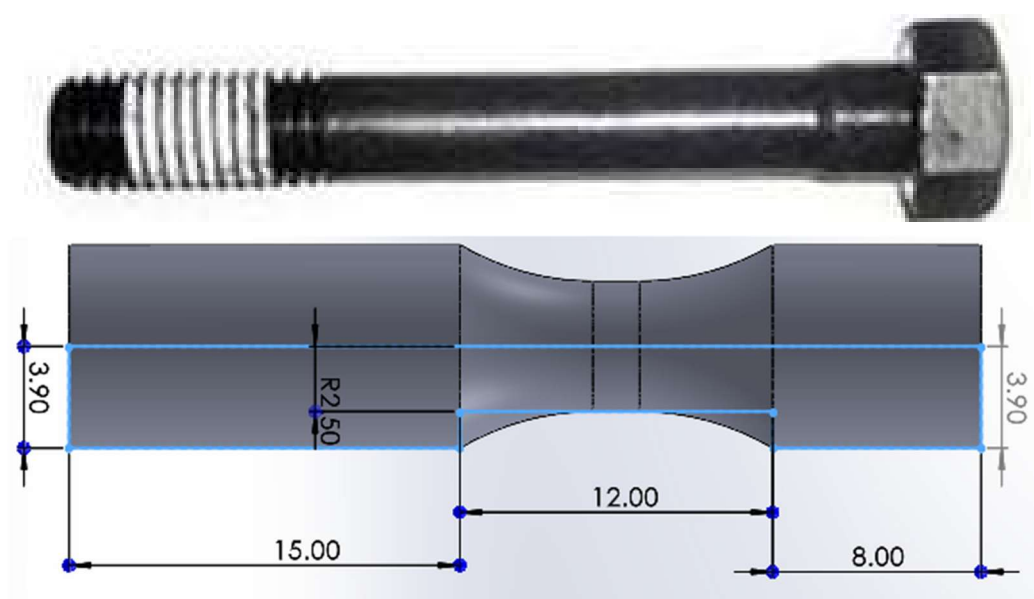


Figure 16 - Redesign geometry

The above geometry was not enough to increase the fatigue safety factor without compromising the compatibility with the conrod unless modified. Therefore, in addition to the above geometry, a change to the material has been justified. Invar 48 has a modulus of elasticity of 160Gpa. This addition of this material to the formula resulted in the fatigue safety factor of 1.034. this is comparable to the fatigue safety factor calculated for the conrod bolts at 6000RPM.

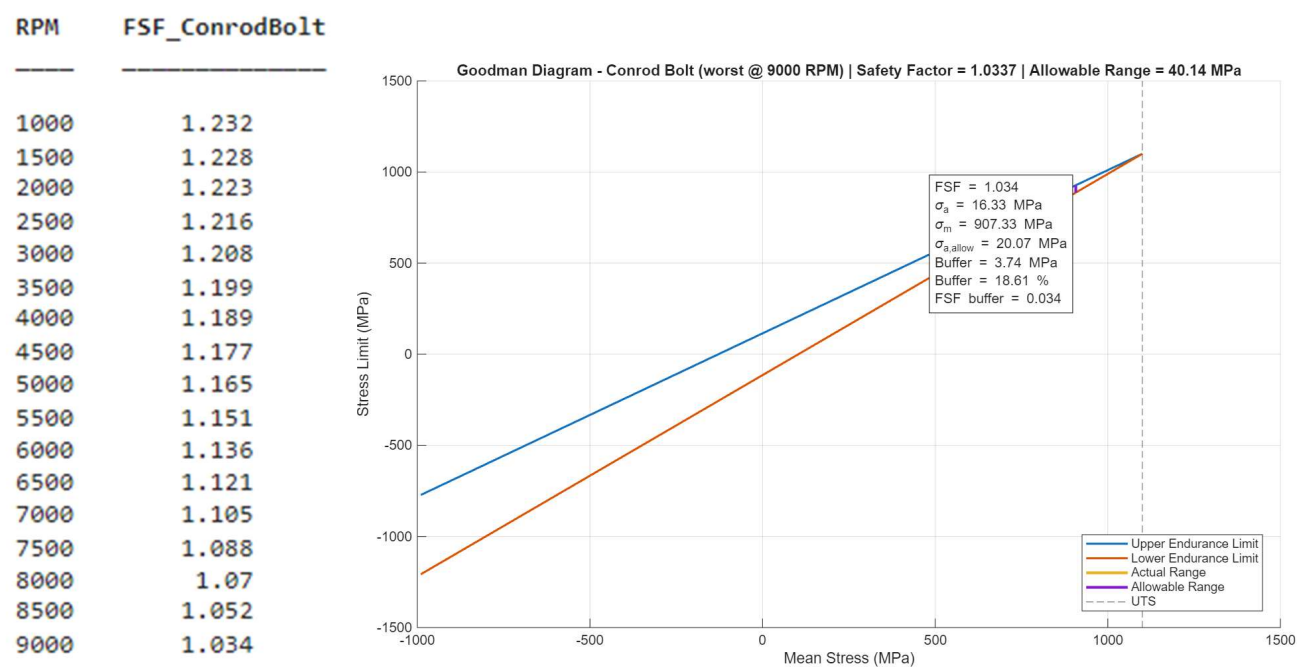


Figure 17 - Extended RPM fatigue safety factor and goodman diagram

Given the UTS and modulus of elasticity specified, the bolt is likely to be made from a high carbon grade of steel. An assumed density has been allocated to the new bolt, 7830kg/m², 42CrMo4, 1.7225, 42CrMoS4, AISI 4140, 42CD4, 708M40 - Alloy Steel (n.d.).

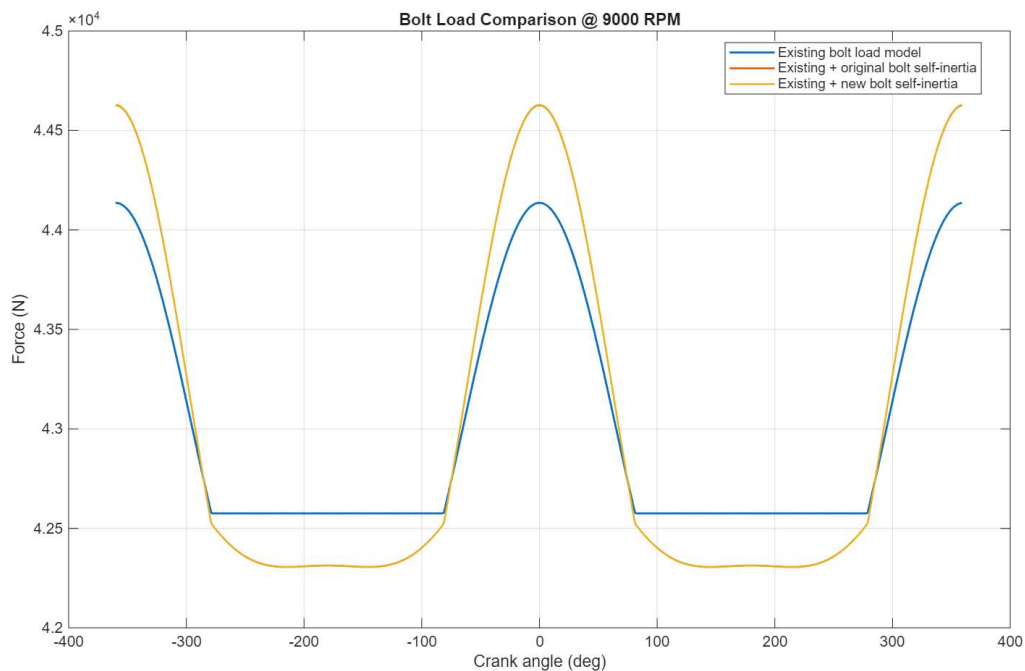


Figure 18 - Conrod bolt forces at 9000PRM

Peak forces exists where cap separation force is highest and constant load on the bolt from preload. The new bolt exhibits self-inertia forces which the original bolt was not exposed to due to increase inertia forces at elevated RPM.

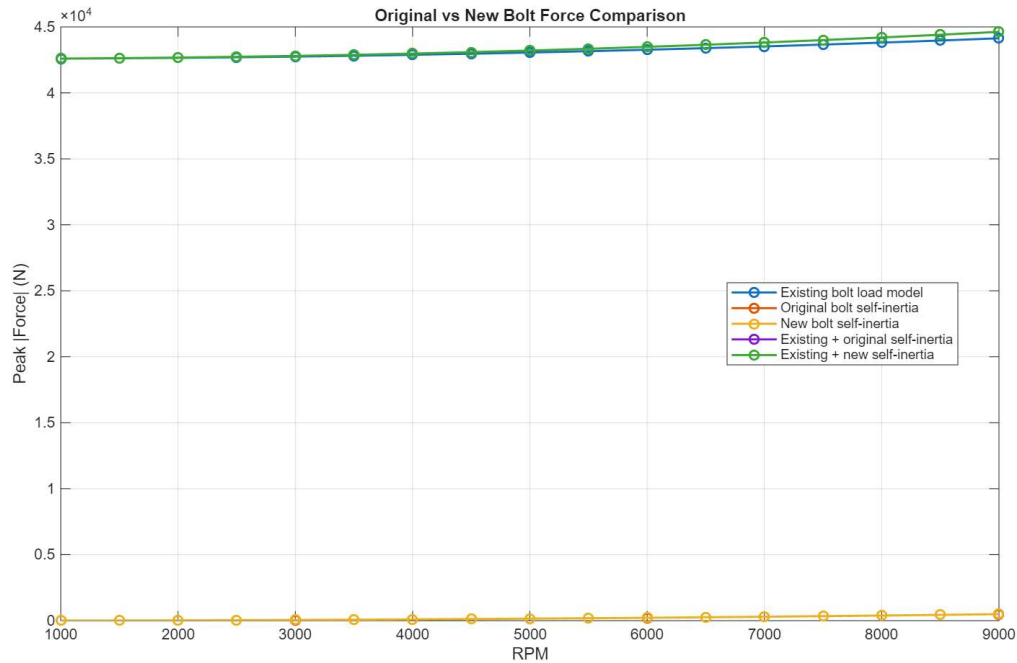


Figure 19 - Force components on conrod bolts

Increased inertia force seen in figure 19 may have an adverse effect on the performance of the engine, as the mass is proportional to inertia in rotational parts will cause the engine speed to decline more slowly, as the system carries more momentum. From an engineering perspective, the priority of the specific component should be safety, longevity and structural integrity, within normal operating boundaries, and not overall performance.

7.1 TORQUE CALCULATION

Using the force data, the overall forces have been applied to calculate the engine piston and engine torque outputs for a standard and a cross plane engine configuration.

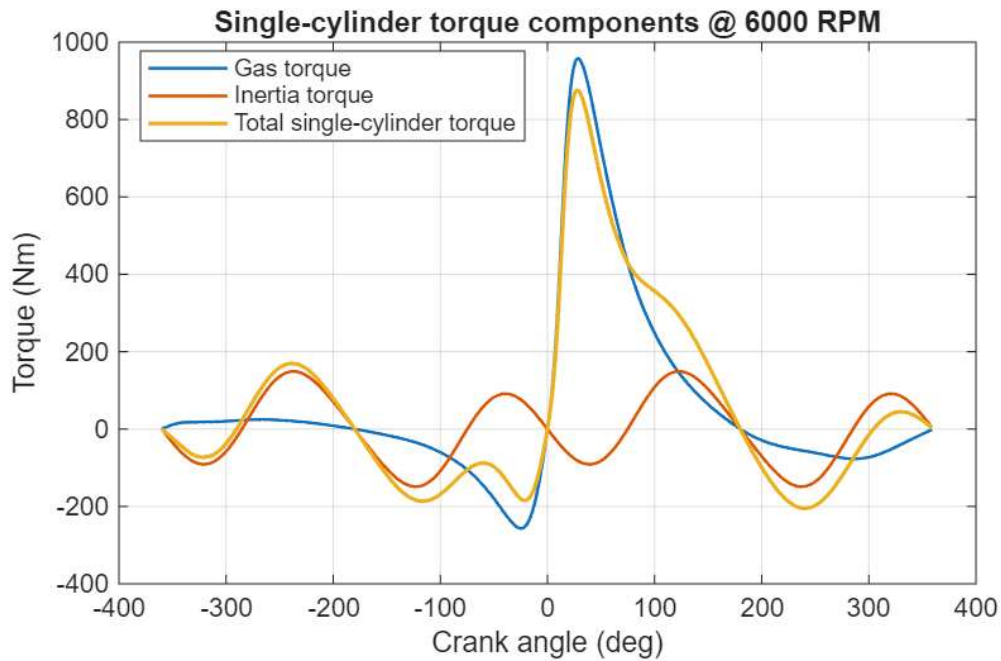


Figure 20 - Torque components

The torque components acting on a single cylinder in the standard and the cross plane configuration can be seen in figure 21.

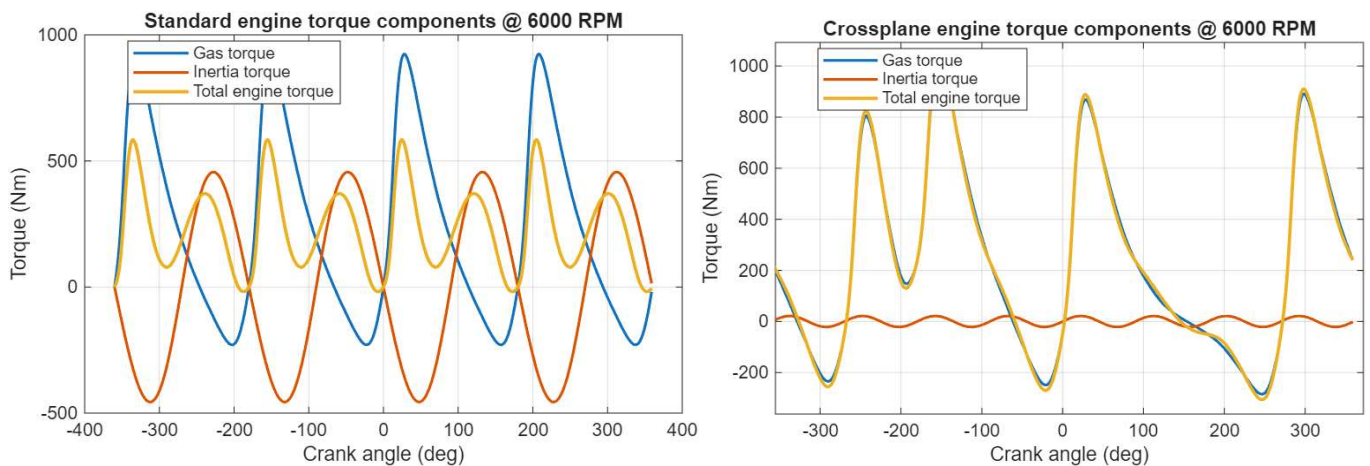


Figure 21 Comparison of average crank torque through crank cycles

With the addition of the crank cylinder arrangement and phasing, a complete torque output per cycle for the engine can be produced. This covers the entire 4 stroke cycle and shows where the torque outputs occur.

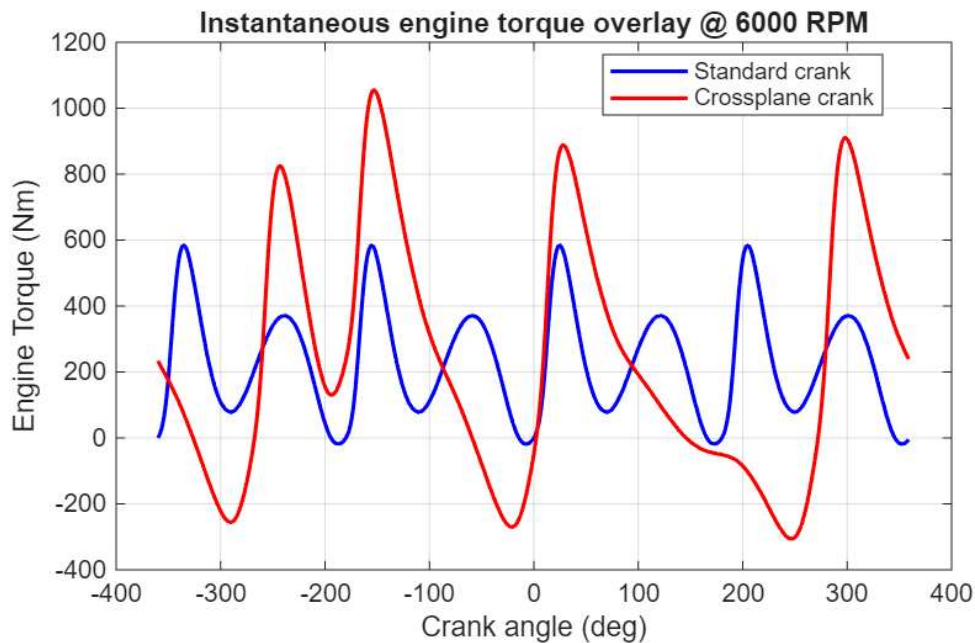


Figure 22 - Standard vs cross plane engine torque output

Comparing the resultant forces from the piston torques shows that at 6000RPM, the cross plane is producing ~500NM more torque when comparing peak values. It is noticeable how the peaks of each torque burst from each configuration does not share the same frequency. The cross plane arrangement had 3 peaks in quick succession, and another later in the crank phase, whereas the standard crankshaft has smaller, and more even interval spread between peaks of torque output.

7.2 TORQUE CURVE COMPARISON

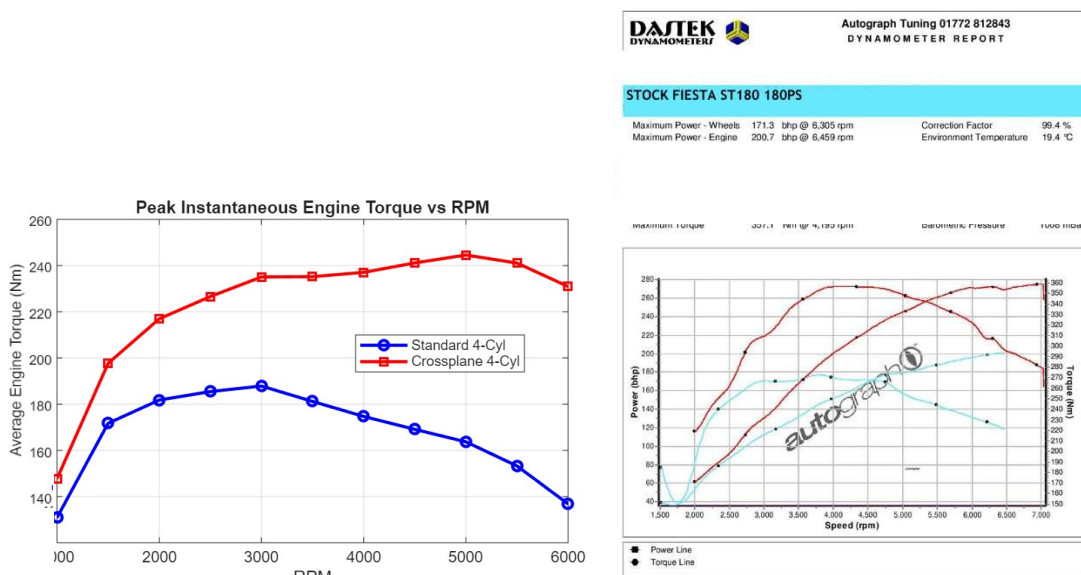


Figure 23 - Engine torque comparison to real output

Comparing the simulated data to real dyno data from a vehicle fitted with the same engine proves that the simulated data method to have similarities to the real world data. the steepest rate of change of torque is consistent across both, with highest delta values being found in the 1000RPM-2000RPM range. The overall bell shape of the

torque curve is like real world results, having a simulated peak torque at 3000RPM and a real world peak value at 3800RPM. Torque fall off from peak is similar, and ends at 6000RPM in a similar region, 138NM in simulation, 130NM real data.

Peak torque values showed discrepancies. The sample real world dyno curve displays torque curve when using an overboost configuration, whereas it is unknown if the gas pressure used from the data acquired for simulation was using the base setting or an overboost. The peak simulated torque showed 188NM of torque, where as the dyno curve demonstrated 272NM. Ford spec sheet for this engine details a peak torque of 240 in standard boost operation, figure 24.

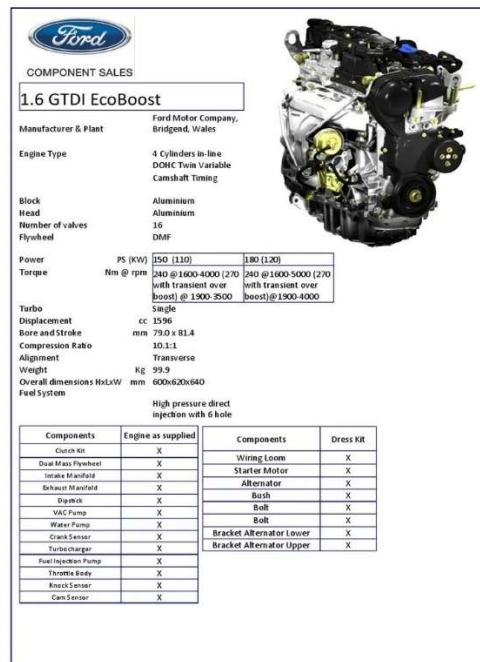


Figure 24 - Ford 1.6 gdti spec sheet

The difference between the real and the simulated torque outputs may be attributed to the calculation method. The simulation method uses IMEP, being the pressure measured in the cylinder. Therefore, all subsequent calculations are derived from this data. The dyno charts are a final torque figure (BMEP) with a correction factor for losses. Although crude, it is accurate for the proposes of understanding engine performance for before and after comparison when tuning.

The discrepancy of ~30% suggests inaccuracies in unmodelled losses. This could be because that temperature can have on components within the engine or pumping loses that don't have a linear or consistent loss input across the RPM range.

The adiabatic assumption when extrapolating the gas force is a likely area of error. Charles's Law would imply that the gas force should increase exponentially as the RPM increases due to the increase in rate of change of temperature.

Finally, it is completely reasonable to assume that these is an error in the calculation.

8 APENDIX

8.1 ENGINE GEOMETRIC PARAMETERS

Bore diameter = 79mm

Crank Radius = 40.7mm

Conrod overall length = 182.97mm

Conrod overall midpoint length = 91.485mm

Overall CSA of midpoint = 196.39 mm^2

Centre to centre con rod length = 134mm

Centre to centre mid CSA = 196.39 mm^2

Pin L1 = 7.75mm

Pin L2 = 7mm

Cross sectional area 227.54 mm^2

D1 = 21mm

D2 = 12.3mm

Neutral axis 10.5mm

Moment of inertia = 8423 Kg.m^2

Cap interface cross sectional area = 210 mm^2

Length of connecting rod section interacting with bolt = 31mm

CSA at Upper Shank = 196.39 mm^2

Reciprocating Mass = 347g

Rotational Mass = 313g

Piston Mass = 268g

Ring Masses = 9g

Clips Mass = 5g

Pin Mass = 28g

Cross sectional area of bolt = 42.43 mm^2

Length of interacting bolt section = 21.26mm

Length to 2nd engaged thread (effective working length) =
27.26mm

8.2 MASS PROPERTIES

Con rod Reciprocating Mass = 347g

Conrod Rotational Mass = 313g

Piston Mass = 268g

Ring Masses = 9g

Clips Mass = 5g

Pin Mass = 28g

Pin exposed mass = 23.28g

8.3 MATERIAL PROPERTIES

Pin UTS = 1000MPa

Pin endurance limit = 45% of UTS, 450MPa

Conrod UTS = 1000MPa

Conrod modulus of elasticity = Modulus of Elasticity = 170,000 N/mm²

Bolt UTS = 1100MPa

Bolt yield strength = 90% of UTS

Bolt endurance limit of material = 45% of UTS

Bolt yield strength of material = 90% of UTS

Blot modulus of elasticity = 200,000 N/mm²

9 REFERENCES

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