



Wales Institute of Science and Art

Analytical assessment of engine internal components

Contemporary Powertrain (ACHA6006)

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Abbreviation Meaning

RPM	Revolutions per minute
TDC	Top dead centre
UTS	Ultimate tensile strength
FSF	Fatigue safety factor
ARP	Automotive Racing Products
MPa	Megapascal

1 Introduction

This report investigates the internal loading of a Ford 1.6 Sigma engine using measured in-cylinder pressure data and engine kinematics to calculate forces, stresses and fatigue safety factors in the conrod mid-section, gudgeon pin and conrod bolts. The analysis is then extended to 9000 rpm, the bolts are redesigned to recover fatigue margin, and standard and cross-plane crank torque characteristics are compared.

2 Engine parameters and Data

The engine parameters used in this study define the basis for the force, stress, and fatigue analysis. These include geometric dimensions, material properties, component masses, and measured in-cylinder pressure data.

2.1 Engine Geometry

Parameter	Symbol	Value	Units
Bore Diameter	D	79	mm
Crank radius	R	40.7	mm
Stroke	S	81.4	mm
Connecting rod Length	L	134	mm
Rod-to-Crank Ratio	R/L	0.304	-
Piston Area	Ap	4903	mm ²

Table 1 - Engine Geometry

(Andrew noble, 2026)

The key geometric parameters include bore diameter, stroke, crank radius, and connecting rod length. The crank radius was defined as half of the stroke, and the piston area was calculated using:

$$A_p = \frac{\pi D^2}{4}$$

This area is used to convert cylinder pressure into gas force acting on the piston. The ratio of crank radius to connecting rod length (R/L) influences piston acceleration and inertia effects. All geometric parameters are summarised in Table 1.

2.2 Material properties

Component	E(N/mm ²)	UTS (MPa)	Yield strength (MPa)	Endurance limit (MPa)
Gudgeon pin	-----	1000	-----	45% of UTS (450)
Connecting rod	170000	1000	-----	45% of UTS (450)
Connecting rod bolt	200000	1100	90% of UTS (990)	45% of UTS (495)

Table 2 - Given Material Properties (E = Young Modulus)

(Andrew noble, 2026)

The material properties provided for each engine component were used as the basis for the stress and fatigue calculations throughout the analysis in table 2.

Material	Description	UTS (MPa)	Yield Strength (MPa)	Endurance Limit (MPa)
Stainless 300	Corrosion-resistant stainless steel for automotive and marine use	1172	1055	528
8740 Chrome Moly	Tough steel with adequate fatigue properties for most racing use	1241–1448	1117–1303	559–652
ARP2000	Higher-strength alloy steel, common upgrade from 8740	1517	1365	683
L19	Very high-strength steel for high inertia load applications	1793	1614	807
Aermet	Martensitic super-alloy with very high clamp load capability	2000–2138	1800–1924	900–962
Inconel 718	Nickel-based super-alloy with good fatigue and corrosion resistance	1448–1586	1303–1428	652–714
ARP3.5	Aerospace-type super-alloy for extreme loads, high RPM and endurance use	1793–1931	1614–1738	807–869
Custom Age 625 Plus	High-strength super-alloy with strong fatigue life and corrosion resistance	1793–1931	1614–1738	807–869
Titanium Ti-6Al-4V	Lightweight corrosion-resistant titanium alloy	1241	1117	559

Table 3 - ARP Alternative Materials (Conrod Bolts)

These ARP material specifications were used for the bolt redesign study. UTS values were used to estimate yield strength and endurance limit, allowing candidate materials to be compared using Goodman fatigue safety factor.

(ARP Bolts, no date)

2.3 Mass

Parameter	Mass (KG)	Type
Piston	0.268	Reciprocating
Ring	0.009	Reciprocating
Clips	0.005	Reciprocating
Gudgeon pin	0.028	Reciprocating
Total reciprocating	0.304	Reciprocating
Rotating mass	0.313	Rotating

Table 4 - Component Mass

(Andrew noble, 2026)

The mass of the system was divided into reciprocating and localised components. The reciprocating mass includes the piston assembly and contributes directly to inertia forces. The gudgeon pin mass was considered separately due to its influence on local.

A lumped-mass approach was used for the inertia calculations. Reciprocating and rotating masses are listed in Table 4.

2.4 In-cylinder pressure data

Measured in-cylinder pressure data was provided as a function of crank angle across multiple engine speeds. This data was used to determine gas force acting on the piston.

The pressure profile shows a peak during combustion, which dominates compressive loading in the conrod, while inertia forces dominate at other parts of the cycle.

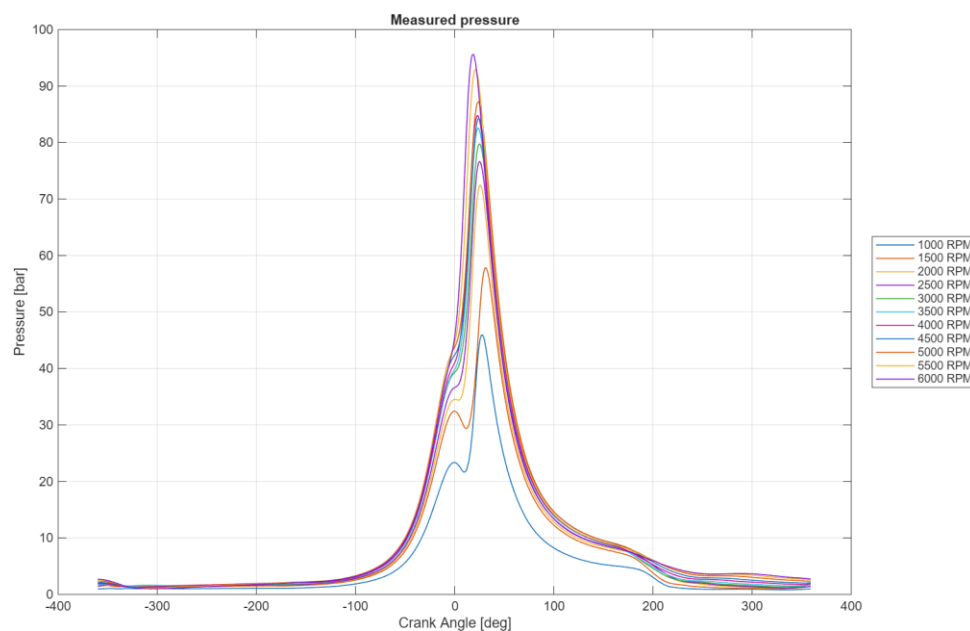


Figure 1 - Ford Sigma Pressure Data

(Andrew noble, 2026)

2.5 RPM extrapolation

Pressure data at 9000 RPM was estimated using extrapolation from the measured datasets. Multiple methods were reviewed within the MATLAB script, including linear, PCHIP, MAKIMA, spline, second-order polynomial fit using the highest three RPM points, and second-order polynomial fit using all measured RPM points.

(MathWorks, no date)

(Huang, Wang and Chou, 2024)

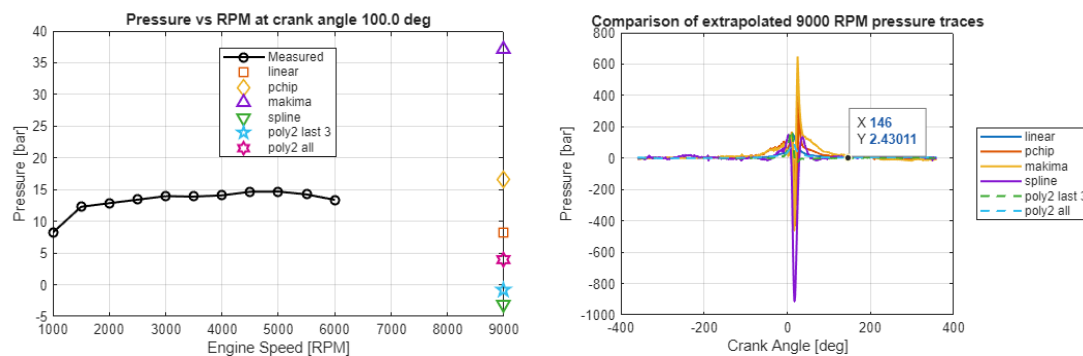


Figure 2 - Interpolation of Pressure Data

Linear extrapolation was selected for the main analysis because it provides the simplest and most conservative extension of the measured trend while avoiding the potential overshoot associated with higher-order smooth fitting methods.

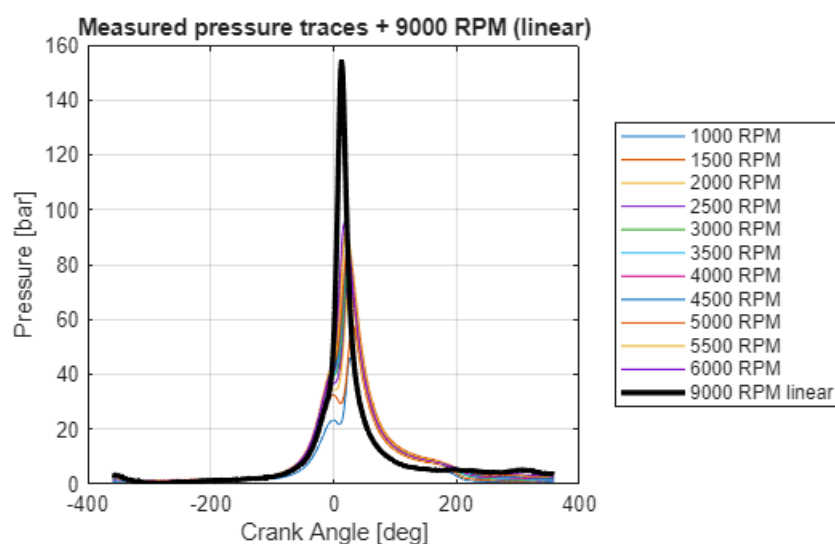


Figure 3 - Extrapolation Results

3 Methodology

This section describes the methods used to determine forces, stresses, and fatigue behaviour of the engine components. A MATLAB-based script was used to process in-cylinder pressure data, calculate dynamic forces, and evaluate performance of the engine components across the full 720-degree crank cycle.

3.1 Engine kinematics

The piston motion was modelled using standard slider-crank kinematics. Crank angle was converted to radians and used to calculate piston position, velocity, and acceleration.

(Alistair, 2024)

Piston position was defined as:

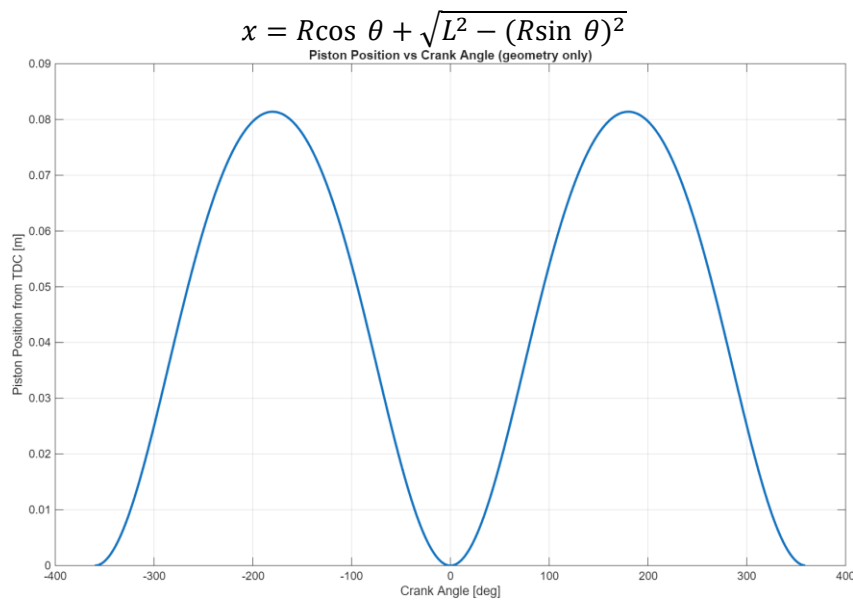


Figure 4 - Piston Position vs Crank Angle

Acceleration was calculated using first and second order components:

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

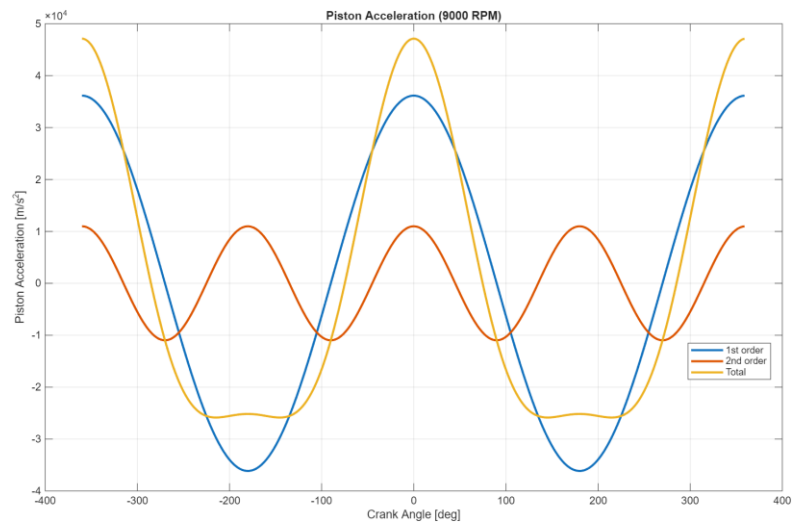


Figure 5 - Acceleration at 9000rpm

This captures both primary and secondary inertia effects, which are significant at higher engine speeds.

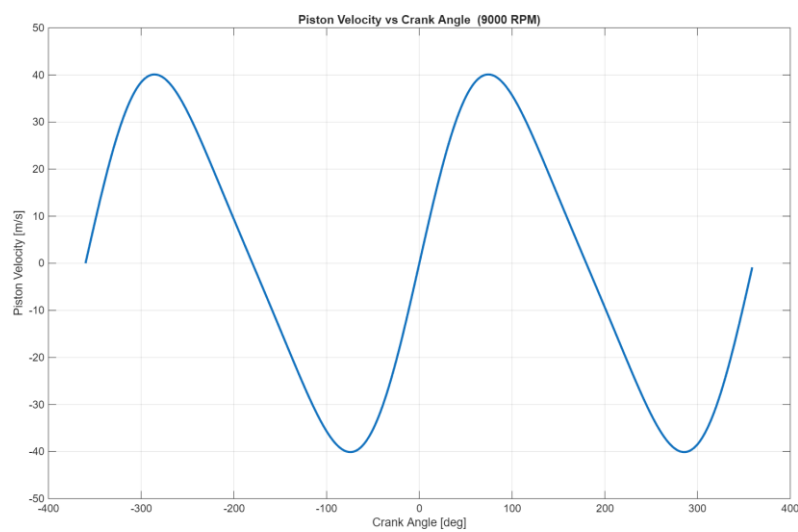


Figure 6 - Piston Velocity vs Crank angle at 9000rpm

3.2 Force analysis

Forces acting on the piston–conrod system were determined from the combined effects of in-cylinder gas pressure and inertial loading due to reciprocating motion.

The gas force acting on the piston was calculated from the measured cylinder pressure:

$$F_{gas} = PA_p$$

where pressure was converted from bar to Pascals prior to application. The inertial force was calculated using the reciprocating mass and piston acceleration obtained from the kinematic model:

$$F_{inertia} = m_{recip}a$$

The net force transmitted through the conrod was initially defined as:

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

where positive values represent tensile loading and negative values represent compressive loading. For clarity of interpretation, results were also presented using a compression-positive convention, allowing combustion-dominated regions to be more easily identified.

The axial force within the conrod was determined by resolving the net force along the rod axis:

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

where the conrod angle ϕ was obtained from slider-crank geometry. The lateral side force acting on the piston skirt was also evaluated as:

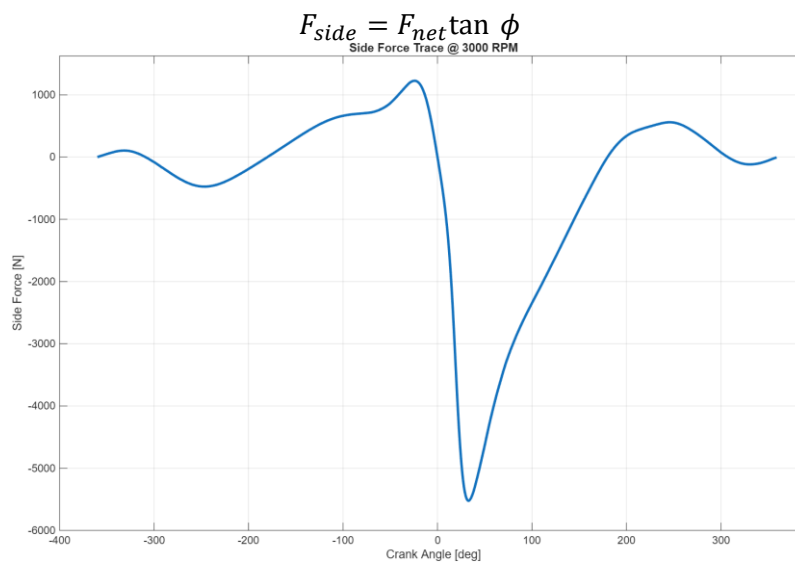


Figure 7 - Side Force's

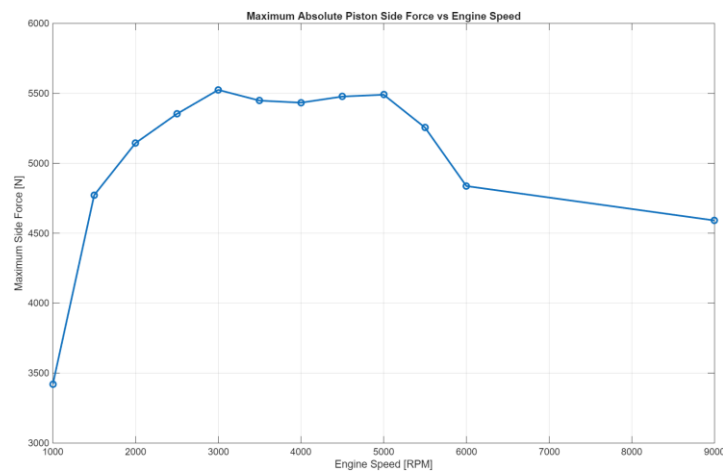


Figure 8 - Side Forces with engine speed

Peak piston side force depends on net piston load $\times$ rod angle, not RPM alone. Since rod angle is fixed by geometry, the drop at high RPM means the gas force at the critical crank angles falls more than inertia increases, so the maximum side forces reduce.

A friction model was applied using a coefficient of friction of 0.1 to represent pistons interaction with the cylinder.

To improve physical interpretation of the loading cycle, the net conrod force was further separated into compressive and tensile components, presented as positive magnitudes. This allows clear identification of regions dominated by combustion pressure and inertia effects.

The results show a pronounced compressive peak near top dead centre, corresponding to maximum cylinder pressure during combustion. Away from top dead centre, inertia effects dominate, producing tensile loading in the conrod as the piston decelerates and reverses direction.

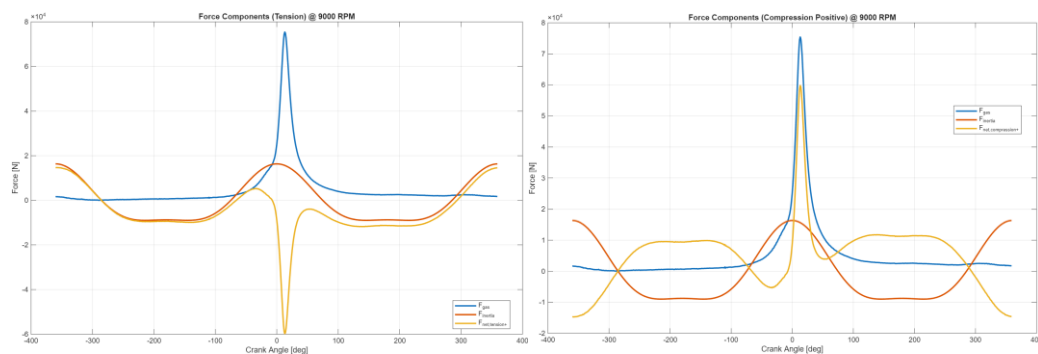


Figure 9 - Tensile and Compressive forces

3.3 Conrod Stress model

The conrod was analysed at the mid-section assuming uniform axial loading.

Stress was calculated using:

$$\sigma = \frac{F_{rod}}{A_{rod}}$$

Maximum and minimum stresses were extracted over the crank cycle for each engine speed. Mean and alternating stresses were then calculated:

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

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

Fatigue performance was assessed using the Goodman relation

3.4 Gudgeon pin stress model

The gudgeon pin was analysed under combined bending and shear loading.

Bending stress was calculated from the applied moment:

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

Shear stress was calculated from combined piston and conrod forces:

$$\tau = \frac{F}{A}$$

An equivalent stress was then determined:

$$\sigma_{eq} = \sqrt{\sigma_b^2 + (2\tau)^2}$$

Fatigue behaviour was assessed using the Goodman approach.

3.5 Conrod bolt model

The conrod bolt was modelled using a preload and stiffness-based load sharing approach.

Initial preload was defined as:

$$F_{preload} = \sigma_{pre} \cdot A_{bolt}$$

The load transferred to the bolt was determined using stiffness ratios:

$$K_{bolt} = \frac{E_{bolt} A_{bolt}}{L_{bolt}}$$

$$K_{cap} = \frac{E_{rod} A_{cap}}{L_{cap}}$$

$$\text{Load fraction} = \frac{K_{bolt}}{K_{bolt} + K_{cap}}$$

Bolt force was then calculated as:

$$F_{bolt} = F_{preload} + F_{extra}$$

Thread stress concentration was included using a calculated K_t , reducing the effective endurance limit.

3.6 Fatigue analysis

Fatigue performance of all components was evaluated using the Goodman failure criterion:

$$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{UTS} = \frac{1}{FSF}$$

The factor of safety against fatigue (FSF) was calculated for each component across all RPM conditions, with particular focus on worst-case loading at 9000 RPM.

3.7 Surface finishing

Surface finishing was considered as an additional fatigue improvement method. For the connecting rod, shot peening was selected because it introduces beneficial compressive residual stress at the surface and is commonly used on fatigue-critical components. A connecting-rod study reported a surface residual compressive stress of –554 MPa after shot peening.

(Kai, 2022)

3.8 Cross plane and standard crank analysis

Torque output can be calculated for both standard inline and cross-plane crank configurations.

Single-cylinder torque contributions were calculated from gas and inertia forces:

$$T = F \cdot r$$

Cylinder phasing was applied using crank angle offsets:

- Standard crank: 0°, 180°, 360°, 540°
- Cross-plane: 0°, 180°, 450°, 630°

Total engine torque was obtained by summing all cylinder contributions over a full 720° cycle.

This allowed comparison of torque smoothness and delivery characteristics between configurations.

4 Results

4.1 Conrod stress fatigue results

RPM	Max Stress	Max Angle	Min Stress	Min Angle	Mean Stress	Alt Stress	Allowable Stress	FSF
1000	-1.2	339	-115	28	-58.1	56.9	952.3	8.363
1500	-0.6	340	-144.4	31	-72.5	71.9	965.3	6.711
2000	0.9	346	-179.1	26	-89.1	90	980.2	5.445
2500	3.1	350	-187.5	25	-92.2	95.3	983	5.16
3000	5.5	349	-192.9	25	-93.7	99.2	984.4	4.962
3500	8.5	353	-196.8	24	-94.1	102.7	984.7	4.796
4000	11.9	355	-198.8	23	-93.5	105.3	984.1	4.672
4500	15.7	358	-194.1	25	-89.2	104.9	980.3	4.673
5000	19.9	-360	-197.2	24	-88.7	108.5	979.8	4.513
5500	24.7	-359	-205.7	21	-90.5	115.2	981.5	4.261
6000	30.3	-358	-206	19	-87.8	118.1	979.1	4.144
9000	75	-358	-305.2	13	-115.1	190.1	1003.6	2.64

Table 5 - Conrod Mid-Point Results

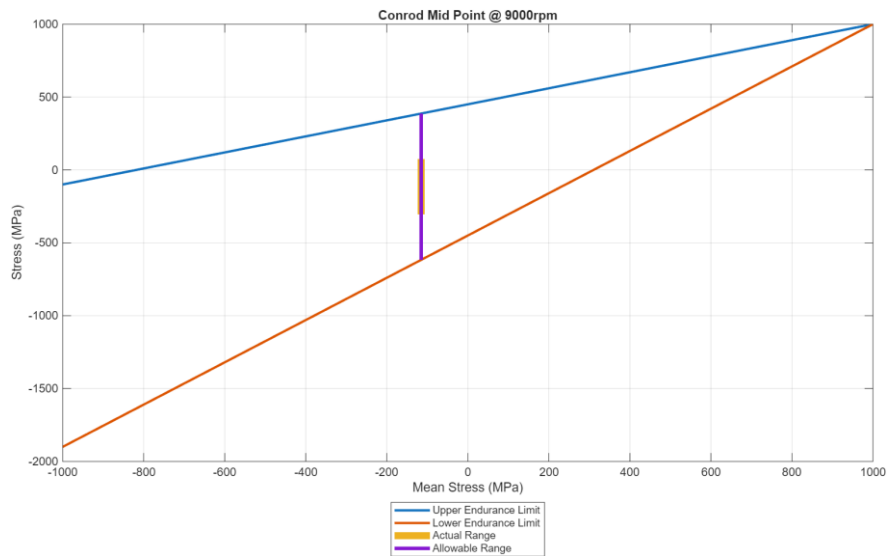


Figure 10 - Conrod Midpoint Goodman Diagram

The connecting rod remains safely within the fatigue limit across the full speed range, although the safety factor decreases as RPM increases. At the stock operating speed of 6000 rpm, the conrod mid-point still has a strong fatigue margin, with a Goodman fatigue safety factor of 4.144. The mean stress stays compressive throughout, which is beneficial for fatigue performance, and even at 9000 rpm the rod remains within the fatigue limit, indicating that it is still structurally suitable at the higher engine speed.

4.1.1 Surface finishing comparison

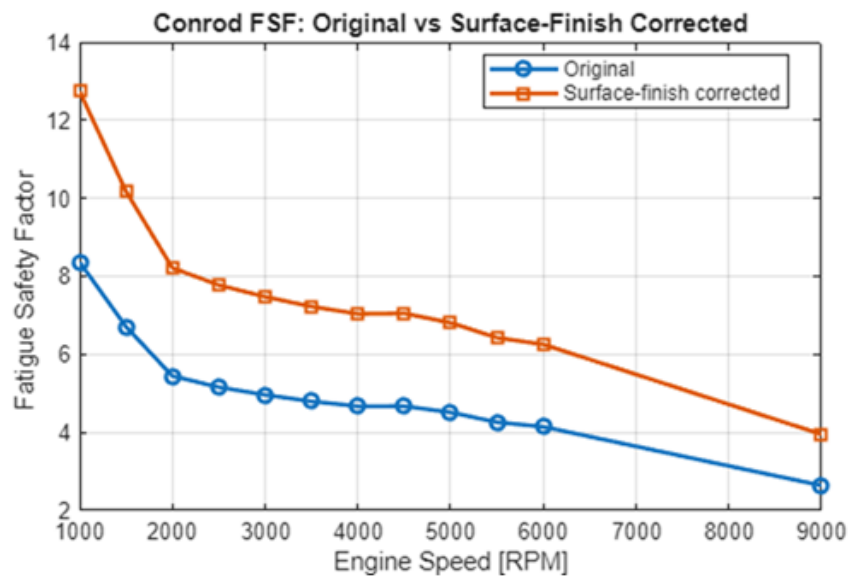


Figure 11 - Surface Finish FSF Comparison

This result shows that applying the surface-finish correction improves the conrod fatigue safety factor across the full speed range. The corrected curve remains above the original curve at all RPM, showing that the added compressive residual stress from shot peening gives a clear fatigue benefit.

At 6000 rpm the safety factor increases noticeably, and even at 9000 rpm the corrected value remains around 4, compared with about 2.6 for the untreated condition. This confirms that shot peening is an effective method for improving fatigue resistance and extending the safe operating margin of the connecting rod at higher engine speeds.

4.2 Gudgeon pin fatigue results

The gudgeon pin remained within the fatigue limit across the full speed range in both shear and combined loading, although the safety factor decreased as engine speed increased. In shear, the fatigue safety factor fell from 18.47 at 1000 rpm to 8.66 at 6000 rpm and 5.41 at 9000 rpm, with a maximum shear stress of 90.7 N/mm² at 9000 rpm. Under combined loading, the resultant fatigue safety factor decreased from 6.24 at 1000 rpm to 3.93 at 6000 rpm and 2.84 at 9000 rpm, with the resultant stress reaching 317.9 N/mm² and a mean stress of 159.7 N/mm² at the highest speed. In both cases, the Goodman plots showed that the operating stress range remained inside the allowable limit, confirming that the gudgeon pin was not a critical component in this analysis.

4.2.1 Pin shear

RPM	Max Stress	Max Angle	Min Stress	Min Angle	Mean Stress	Stress Range	Allowable Stress	FSF
1000	0.2	348	-48.6	28	-24.2	48.7	900	18.47
1500	1.3	346	-60.3	31	-29.5	61.6	900	14.62
2000	3.2	352	-74.2	26	-35.5	77.4	900	11.63
2500	5.8	354	-76.4	25	-35.3	82.2	900	10.95
3000	8.7	353	-77.1	25	-34.2	85.8	900	10.49
3500	12.4	356	-76.7	24	-32.1	89.1	900	10.1
4000	16.6	357	-75.2	23	-29.3	91.8	900	9.8
4500	21.4	359	-70.7	25	-24.7	92.1	900	9.77
5000	26.6	-360	-69	24	-21.2	95.6	900	9.41
5500	32.4	-359	-68.9	22	-18.2	101.4	900	8.88
6000	39	-359	-64.9	19	-12.9	103.9	900	8.66
9000	90.7	-358	-75.5	13	7.6	166.2	900	5.41

Table 6 - Pin Shear Results

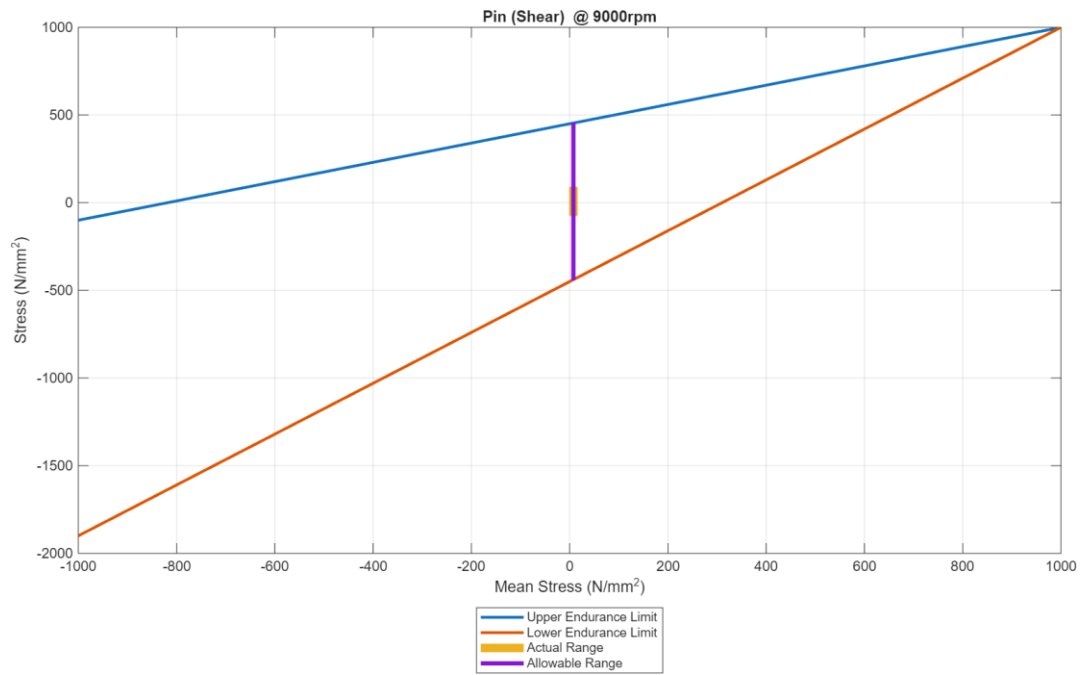


Figure 12- Pin Shear Goodman Diagram

4.2.2 Pin Resultant

RPM	Max Stress	Max Angle	Min Stress	Min Angle	Mean Stress	Stress Range	Allowable Stress	FSF
1000	145.2	28	1	339	73.1	144.2	900	6.24
1500	181.1	31	1.3	315	91.2	179.9	900	5
2000	224.1	26	1.5	304	112.8	222.7	900	4.04
2500	232.9	25	1.7	299	117.3	231.2	900	3.89
3000	237.5	25	1.9	-296	119.7	235.5	900	3.82
3500	239.6	24	1.9	-293	120.8	237.6	900	3.79
4000	238.9	23	1.8	-291	120.4	237.1	900	3.8
4500	229.4	25	1.7	-290	115.6	227.7	900	3.95
5000	229.3	24	1.6	-289	115.5	227.7	900	3.95
5500	235.2	21	1.5	-288	118.4	233.7	900	3.85
6000	230.5	19	1.4	-287	116	229.2	900	3.93
9000	317.9	13	1.5	-285	159.7	316.4	900	2.84

Table 7 - Pin Resultant Results

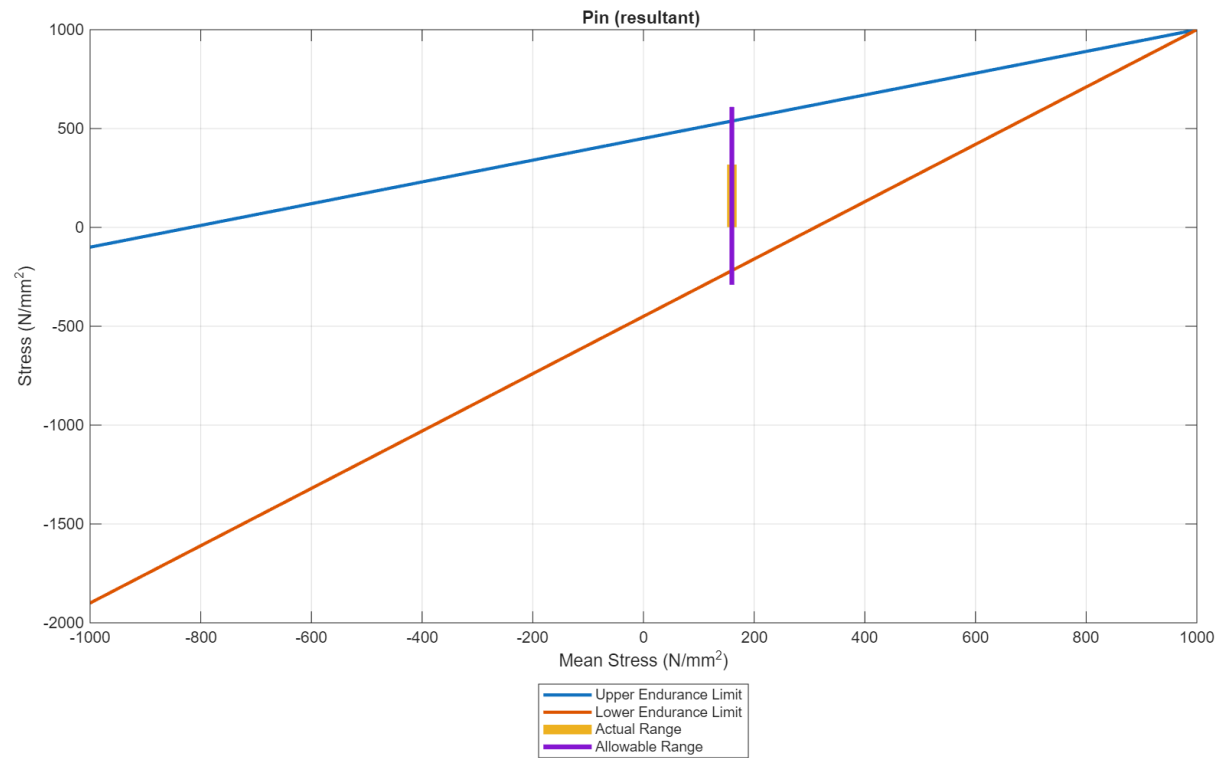


Figure 13 - Pin Resultant Goodman Diagram

4.3 Conrod bolts fatigue results

RPM	Max Stress	Max Angle	Min Stress	Min Angle	Mean Stress	Alternating Stress	Stress Range	Allowable Range	FSF
1000	535.5	-360	534	-180	534.7	0.7	1.5	117.9	2.03
1500	536.5	-360	533.2	-180	534.9	1.7	3.3	117.8	1.997
2000	538	-360	532.1	-180	535.1	3	5.9	117.8	1.952
2500	540	-360	530.7	-180	535.3	4.6	9.2	117.7	1.898
3000	542.3	-360	529	-180	535.7	6.6	13.3	117.7	1.835
3500	545.1	-360	527	-180	536	9	18.1	117.6	1.766
4000	548.3	-360	524.7	-180	536.5	11.8	23.6	117.5	1.693
4500	551.9	-360	522	-180	537	15	29.9	117.4	1.617
5000	556	-360	519.1	-180	537.5	18.5	36.9	117.3	1.539
5500	560.5	-360	515.8	-180	538.2	22.3	44.7	117.1	1.462
6000	565.4	-360	512.3	-180	538.8	26.6	53.2	117	1.386
9000	604	-360	484.3	-180	544.2	59.8	119.7	115.9	0.984

Table 8 - Conrod Bolts Results

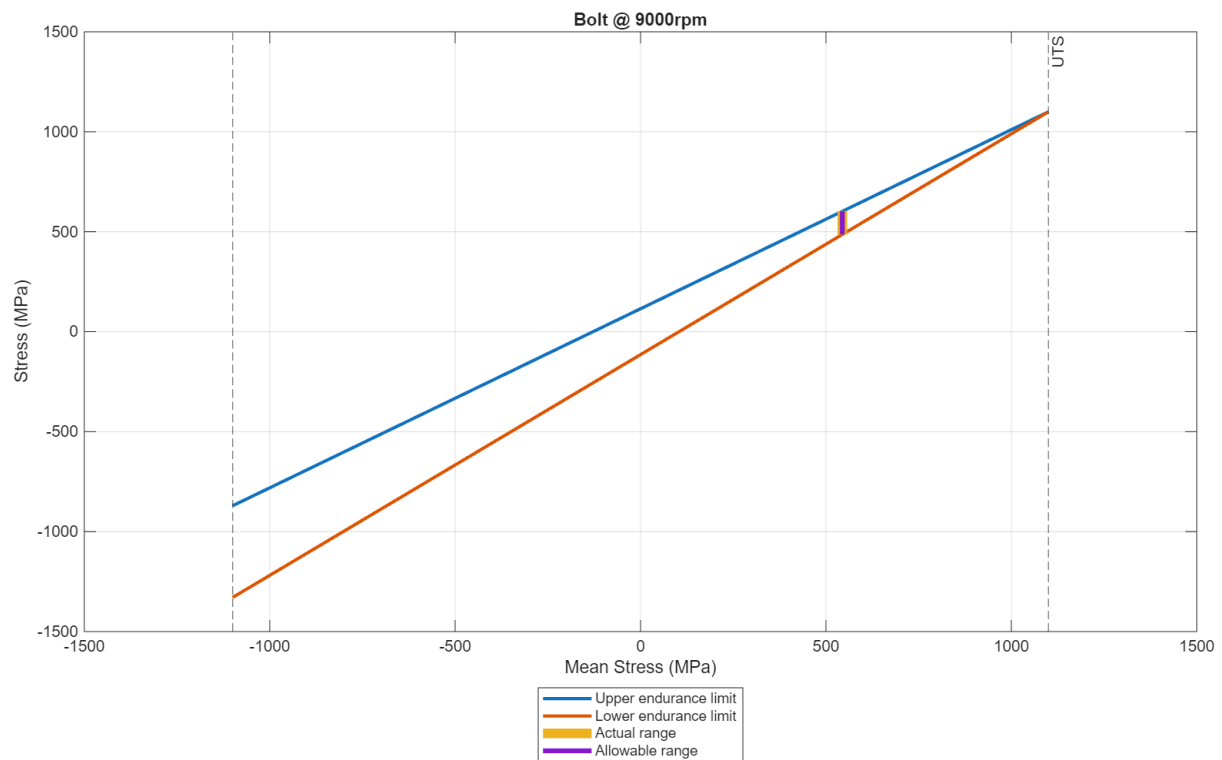


Figure 14 - Conrod Bolts Goodman Diagram

The conrod bolt fatigue margin decreases steadily with engine speed. At 6000 rpm the bolt remains above the Goodman limit with an FSF of 1.386, but at 9000 rpm this falls to 0.984, making the original bolt unsuitable at the higher speed. This is mainly driven by increasing inertia loading rather than gas pressure, as the mean bolt stress changes only slightly while the stress range rises significantly. The Goodman plot shows that at 9000 rpm the actual stress range reaches the allowable limit, so the bolt becomes the critical component and requires redesign.

4.4 Cross plane and standard crank results

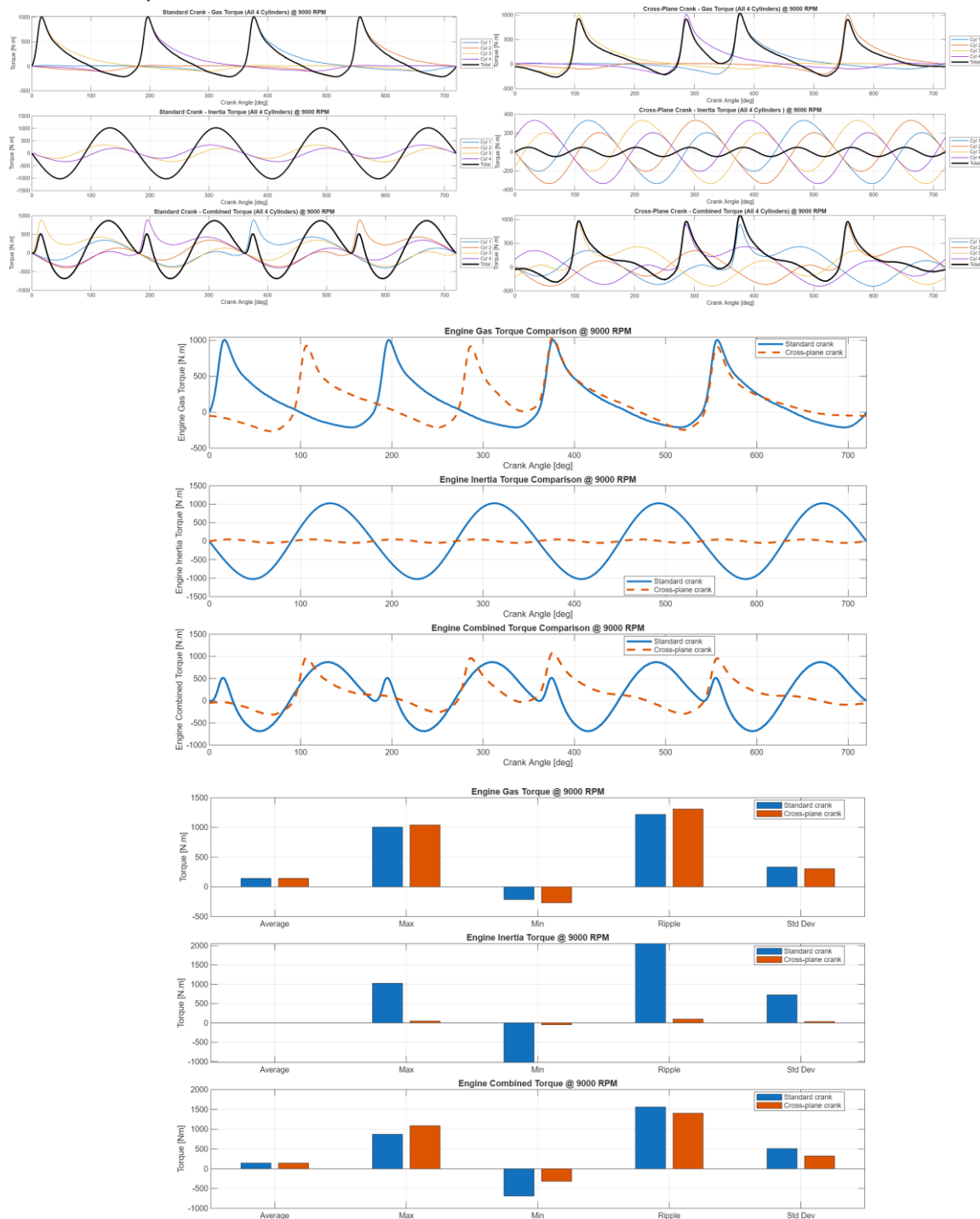


Figure 15 - Cross Plane vs Standard Crank Results

At 9000 rpm, both crank layouts produced a similar average torque, but the instantaneous torque traces were different. The standard crank showed a much larger inertia torque fluctuation, which led to a higher combined torque ripple and larger swings between positive and negative torque. The cross-plane crank gave a smoother combined torque trace with lower ripple and lower standard deviation. The gas torque behaviour was broadly similar between the two layouts, so the main difference came from the inertia torque contribution and the different cylinder phasing.

5 Analysis

At 6000 rpm the original engine components remain within the fatigue limit. The conrod has a fatigue safety factor of 4.144 and the gudgeon pin remains non-critical, while the bolt has the lowest margin with an FSF of 1.386. This shows that the stock engine speed is acceptable for the original design, but also that the conrod bolt is already the most fatigue-sensitive component.

At 9000 rpm the conrod and gudgeon pin remain above the fatigue limit, but the bolt falls to an FSF of 0.984. This identifies the bolt as the limiting component at increased engine speed.

The results therefore justify focusing the redesign on the conrod bolt rather than the other components. The 9000 rpm values are based on extrapolated pressure data and a simplified stiffness model, so the exact values are approximate, but the trend is clear: the original design is acceptable at 6000 rpm, and the conrod bolt becomes the critical part at 9000 rpm.

5.1 Crank Design and Firing order

These results show that the main effect of changing from a standard crank to a cross-plane crank is not a large change in average torque, but a change in how torque is delivered through the cycle. The standard inline-four crank has an even firing order and good primary balance, but in this model, it also produces larger inertia-driven torque oscillations, which makes the total output more uneven. By contrast, the cross-plane crank changes the phase relationship between the cylinders and uses an uneven firing sequence, which reduces the masking effect of inertia torque on combustion torque and gives a smoother combined torque trace. This is consistent with the lower ripple and lower standard deviation found in the present results. In practical terms, this would improve rear-tyre feel, traction and throttle connection, particularly in high-performance riding. However, the cross-plane layout also brings penalties in vibration, balancing requirements, weight, heat and cost, which helps explain why the conventional flat-plane inline-four remains the more common production solution.

(Piggott and Taulbut, no date)

(John Burns, 2016)

6 Bolt Redesign

The bolt redesign was carried out in MATLAB using a full parameter sweep of all candidate ARP materials and bolt geometries. For each material, the script varied bolt diameter, pitch,

root radius and effective length, then recalculated preload, stiffness split, bolt stress and Goodman fatigue safety factor.

The design target was to find a bolt that matched the original 6000 rpm bolt fatigue safety factor at 9000 rpm. This allowed the higher-speed condition to be directly compared against the known acceptable stock-engine condition.

Because no reliable cost or weight data was available for the candidate materials in the analysis, the redesigns were ranked using bolt volume. The final design was therefore selected as the smallest-volume bolt that achieved the target fatigue safety factor at 9000 rpm. This provided a simple and consistent basis for comparing the redesign options while still favouring a compact solution.

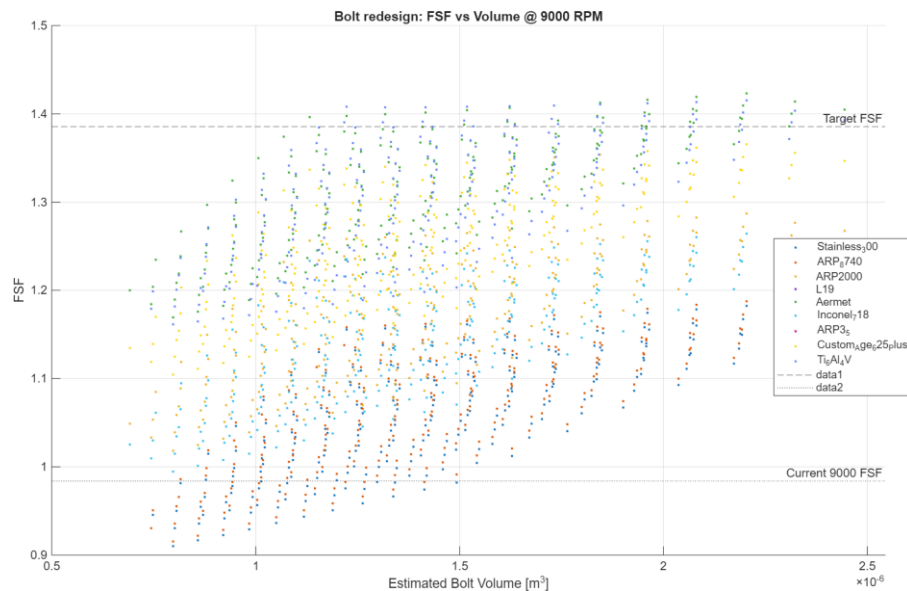


Figure 16 - MATLAB Bolt Redesign Scatter

Material	MajorD_mm	Pitch_mm	RootR_mm	As_mm2	Kt	FSF_9000	Volume_m3
Aermet	7.50	1.25	0.108	31.443	4.0722	1.3964	1.1319E-06
Aermet	7.50	1.25	0.150	31.443	4.0722	1.3964	1.1319E-06
Aermet	7.50	1.25	0.200	31.443	4.0722	1.3964	1.1319E-06
Ti_6Al_4V	7.50	1.00	0.108	33.817	4.2676	1.3904	1.2174E-06
Ti_6Al_4V	7.50	1.00	0.150	33.817	4.2676	1.3904	1.2174E-06
Ti_6Al_4V	7.50	1.00	0.200	33.817	4.2676	1.3904	1.2174E-06
Ti_6Al_4V	7.75	1.25	0.108	33.976	4.1036	1.4078	1.2232E-06
Ti_6Al_4V	7.75	1.25	0.150	33.976	4.1036	1.4078	1.2232E-06
Ti_6Al_4V	7.75	1.25	0.200	33.976	4.1036	1.4078	1.2232E-06
Aermet	7.75	1.25	0.108	33.976	4.1036	1.3978	1.2232E-06

Table 9 - Bolt Redesign Results

The redesign results show that several candidates met the target of matching the original 6000 rpm bolt FSF at 9000 rpm. The best candidate was Aermet, because it achieved the target with the smallest bolt volume.

The selected Aermet design used a 7.5 mm diameter, 1.25 mm pitch, and 31.443 mm² stress area, giving an FSF of 1.3964 at 9000 rpm with a volume of 1.1319×10^{-6} m³. This made it the most efficient option in the sweep, as it provided the required fatigue improvement with the least material.

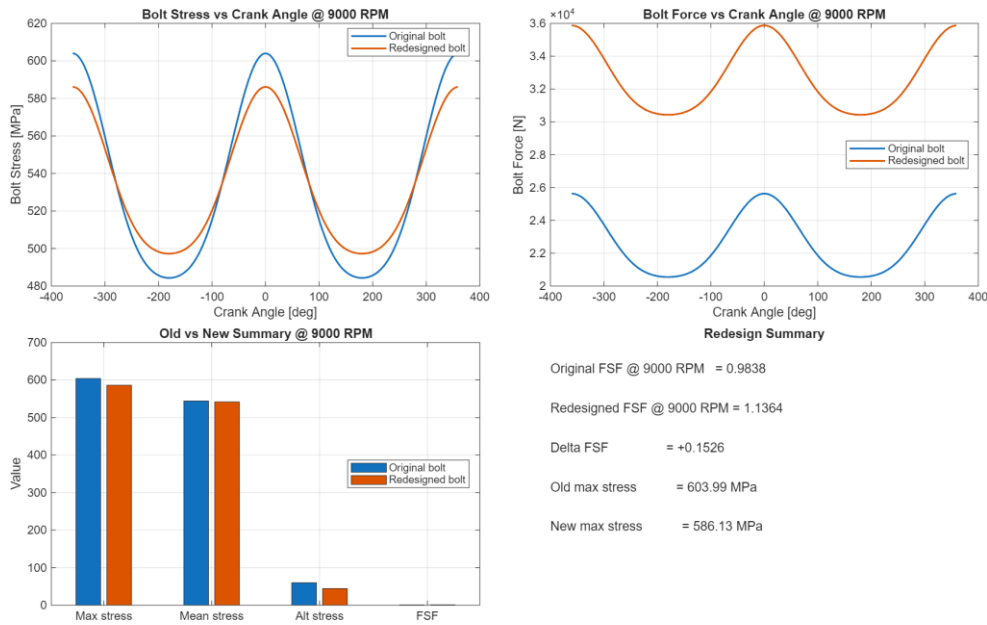


Figure 17 - New Vs Old Bolt Results

7 Conclusion

The results showed that at the stock operating condition of 6000 rpm, the connecting rod, gudgeon pin and conrod bolt all remain within the Goodman fatigue limit, confirming that the original design is suitable for its intended speed range. When the analysis was extended to 9000 rpm, the connecting rod and gudgeon pin remained structurally acceptable, although both showed the expected reduction in fatigue safety factor as inertia loading increased. The conrod bolt, however, became the critical component, with its fatigue safety factor falling below unity. This showed that the limiting factor at higher speed is not the rod or pin, but the bolt preload and its increasing inertia-driven cyclic stress.

The study also showed that shot peening provides a clear fatigue benefit for the connecting rod by increasing the safe operating margin through compressive residual stress. In the crankshaft comparison, the standard crank and cross-plane crank produced similar average torque, but the cross-plane arrangement gave a smoother combined torque trace with lower fluctuation over the cycle. A MATLAB redesign sweep was then used to recover the

original bolt fatigue margin at 9000 rpm. From the candidate ARP materials and geometry combinations, Aermet gave the best solution because it achieved the target fatigue safety factor with the smallest bolt volume.

This analysis does, however, have limitations. The bolt redesign comparison was based mainly on room-temperature material properties, so differences in temperature sensitivity between candidate materials were not included and could affect the real stiffness, strength and fatigue behaviour in service. In addition, changes in bolt material and geometry would also change bolt mass, which would slightly alter the inertia loading and therefore the calculated stresses. Overall, the project shows that the original engine design is suitable at 6000 rpm, that the redesigned conrod bolts are safe at 9000 rpm, and that this limitation can be overcome through more in analysis with less assumptions and more variables in the calculations.

(High-Temperature and Low-Temperature Fatigue, no date)

8 Bibliography

Alistair (2024) *Crank mechanism kinematics - classic analysis*.

Andrew noble (2026) 'Powertrain Assignment'.

ARP Bolts (no date) *ARP Material Specifications*.

High-Temperature and Low-Temperature Fatigue (no date).

Huang, C.C., Wang, Y. and Chou, C.C. (2024) *Comparison of S-Parameter Low Frequency Interpolation Methods*.

John Burns (2016) 'Why Are Yamaha the Only Ones to Use a "Crossplane" Crankshaft?', *motorcycle.com*.

Kai, Z. (2022) 'Study on Shot Peening Technology and Performance of Connecting Rod for Marine Diesel Engine', *Cailiao Baohu*, 55. Available at: <https://doi.org/10.16577/j.issn.1001-1560.2022.0288>.

MathWorks (no date) *Extrapolation for Interpolant Fit Types*.

Piggott, D. and Taulbut, D. (no date) *1 of 24 Grand Prix Motorcycle Engine Development*. Available at: <http://www.grandprixengines.co.uk>.