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Contemporary Power Train Design
ACHA6006

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Introduction

Bottom-end internal combustion engine components are subjected to cyclic loading from cylinder pressure and reciprocating inertia forces, both of which increase with engine speed. As a result, fatigue assessment is required to determine whether key components can provide adequate durability without unnecessary mass or over-conservative design. This is particularly important for the connecting rod, gudgeon pin and con-rod bolt, as failure of these parts would significantly affect engine reliability and safe operation.

This report analyses the supplied Ford 1.6 GDTi engine data using pressure traces, engine geometry and slider-crank kinematics to evaluate the connecting rod midpoint, gudgeon pin and con-rod bolt. Goodman diagrams and fatigue safety factors are used to identify the most critical component over the operating range and assess whether the design appears conservative or over-engineered.

Part 1 Methodology

The Part 1 analysis used the supplied crank-angle pressure data and engine specification data, including bore, crank radius, connecting rod length, reciprocating mass, component areas and material properties. Pressure at each crank-angle position was converted into piston gas force using piston area, while slider-crank relations were used to calculate piston acceleration and the associated inertia force. These were combined to obtain the net axial load through the mechanism.

This load was then used to calculate stress in the three components. Connecting rod midpoint stress was found from axial force and cross-sectional area, gudgeon pin stress was determined using a bending-stress approach, and con-rod bolt stress was obtained from the tensile opening load at the big end, shared equally between the two bolts. For each component and engine speed, maximum stress, minimum stress, mean stress and alternating stress were then calculated and used in the Goodman fatigue assessment.

Part 1 - Results

The following results show the calculated mean stress and alternating stress for the connecting rod, gudgeon pin and con-rod bolt, over the engine speed range of 1000rpm-6000rpm, together with the data used for the Goodman fatigue assessment.

Connecting Rod

RPM	RodMax	RodMin	RodMean	RodAlt	RodFS
1000	115.05	1.1749	58.112	56.937	5.416
1500	144.43	0.60113	72.514	71.912	4.3044
2000	179.09	-0.92724	89.083	90.01	3.459
2500	187.45	-3.0558	92.198	95.254	3.2908
3000	192.93	-5.4589	93.734	99.193	3.1831
3500	196.78	-8.529	94.127	102.66	3.1032
4000	198.81	-11.852	93.481	105.33	3.0529
4500	194.12	-15.68	89.221	104.9	3.1024
5000	197.22	-19.868	88.677	108.54	3.0313
5500	205.65	-24.652	90.501	115.15	2.8869
6000	205.98	-30.292	87.843	118.13	2.8542

(Figure 1. Calculated stress values for the connecting rod from 1000-6000rpm)

Table 1 shows the connecting rod midpoint stress results from 1000 to 6000 rpm. The maximum stress increased from 115.05 MPa at 1000 rpm to 205.98 MPa at 6000 rpm, while the minimum stress decreased from 1.17 to -30.29 MPa. The alternating stress rose steadily from 56.94 MPa to 118.13 MPa, and the fatigue safety factor fell from 5.42 to 2.85. the most critical condition appeared at 6000 rpm.

Gudgeon Pin

PinMax	PinMin	PinMean	PinAlt	PinFS
98.581	1.0068	49.794	48.787	6.3207
123.75	0.51508	62.134	61.619	5.0235
153.46	0.0029512	76.73	76.727	4.0447
160.62	0.030878	80.326	80.295	3.8646
165.31	0.015945	82.664	82.648	3.7548
168.62	0.092687	84.354	84.261	3.6819
170.36	0.034637	85.196	85.161	3.6438
166.33	0.06778	83.201	83.133	3.7321
168.99	0.081998	84.537	84.455	3.6736
176.22	0.1692	88.193	88.024	3.5236
176.49	0.074908	88.284	88.209	3.5174

(Figure 2. Calculated stress values for the gudgeon pin from 1000 to 6000 rpm)

For the gudgeon pin, maximum stress increased from 98.58 MPa to 176.49 MPa across the speed range, while minimum stress remained close to zero. Alternating stress increased from 48.79 MPa to 88.21 MPa, and the fatigue safety factor reduced from 6.32 to 3.52. The most critical condition again occurred at 6000 rpm.

Con-Rod Bolt

BoltMax	BoltMin	BoltMean	BoltAlt	BoltFS
716.66	660	688.33	28.329	1.4642
731.12	660	695.56	35.562	1.4201
748.2	660	704.1	44.098	1.3714
752.31	660	706.16	46.157	1.3602
755.01	660	707.51	47.505	1.3529
756.91	660	708.45	48.454	1.3478
757.91	660	708.95	48.955	1.3452
755.6	660	707.8	47.799	1.3513
757.12	660	708.56	48.562	1.3472
761.28	660	710.64	50.639	1.3363
761.44	660	710.72	50.718	1.3359

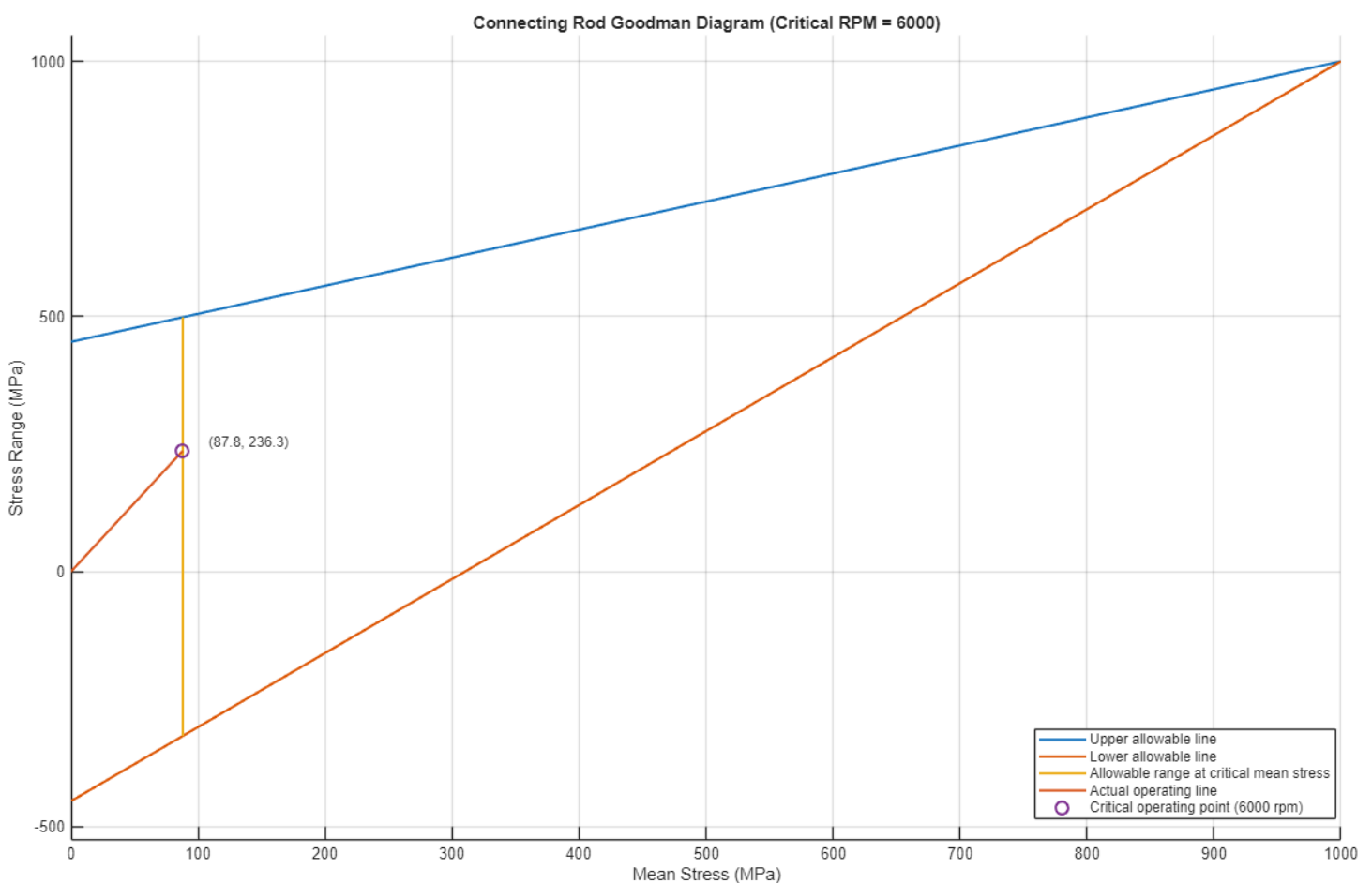
(Figure 3. Calculated stress values for con-rod bolt from 1000 to 6000 rpm)

For the con-rod bolt, maximum stress increased from 716.66 MPa at 1000 rpm to 761.44 MPa at 6000 rpm, while minimum stress remained constant at 660 MPa due to the preload condition. Alternating stress increased from 28.33 MPa to 50.72 MPa, and the fatigue safety factor decreased from 1.46 to 1.34. The most critical condition occurred at 6000 rpm.

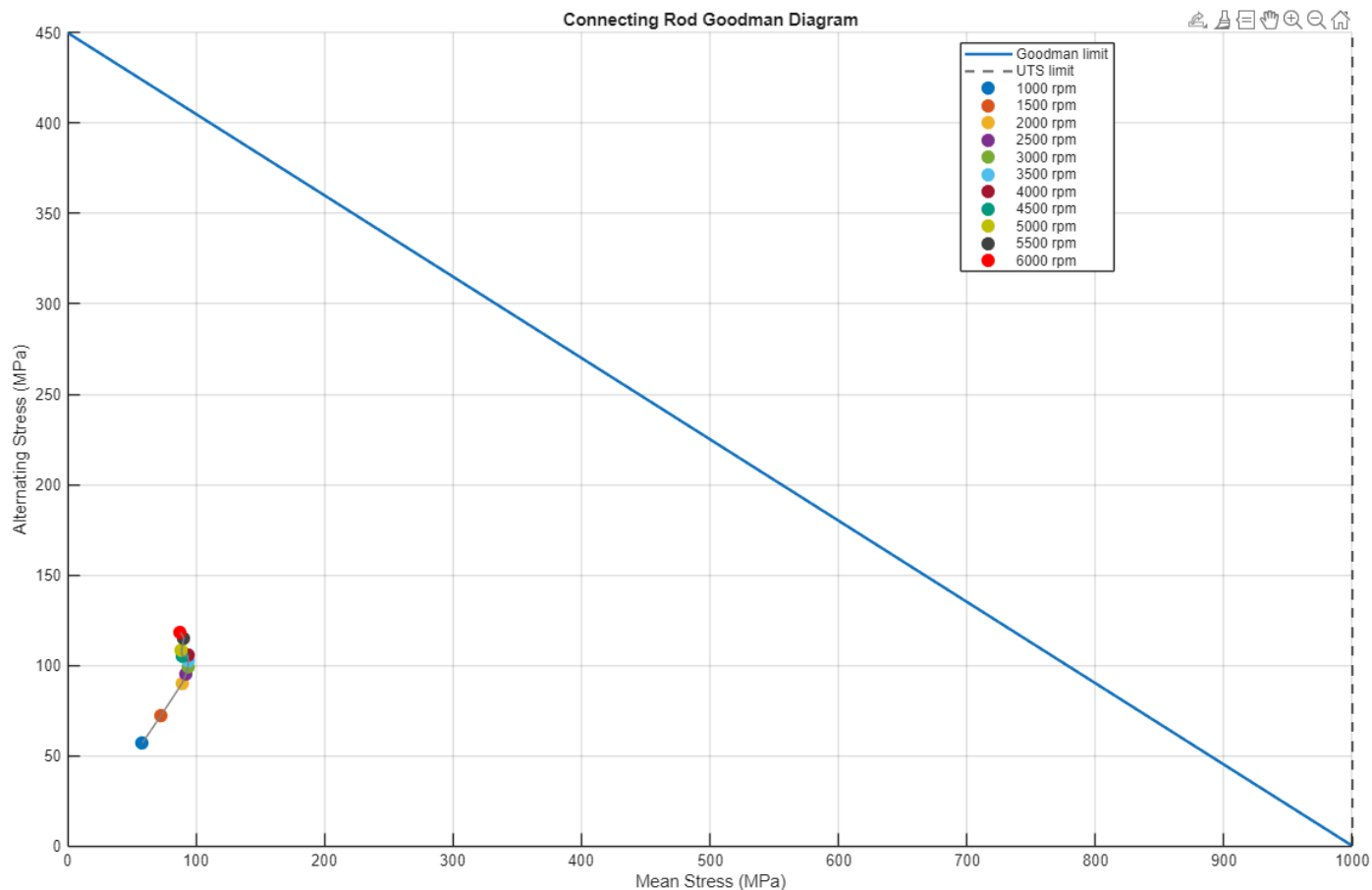
Part 1 - Analysis

This section analyses the calculated stresses, fatigue safety factors and Goodman diagrams for the connecting rod midpoint, gudgeon pin and con-rod bolt. The results are used to identify the critical operating conditions and to assess likely component durability and possible over-engineering.

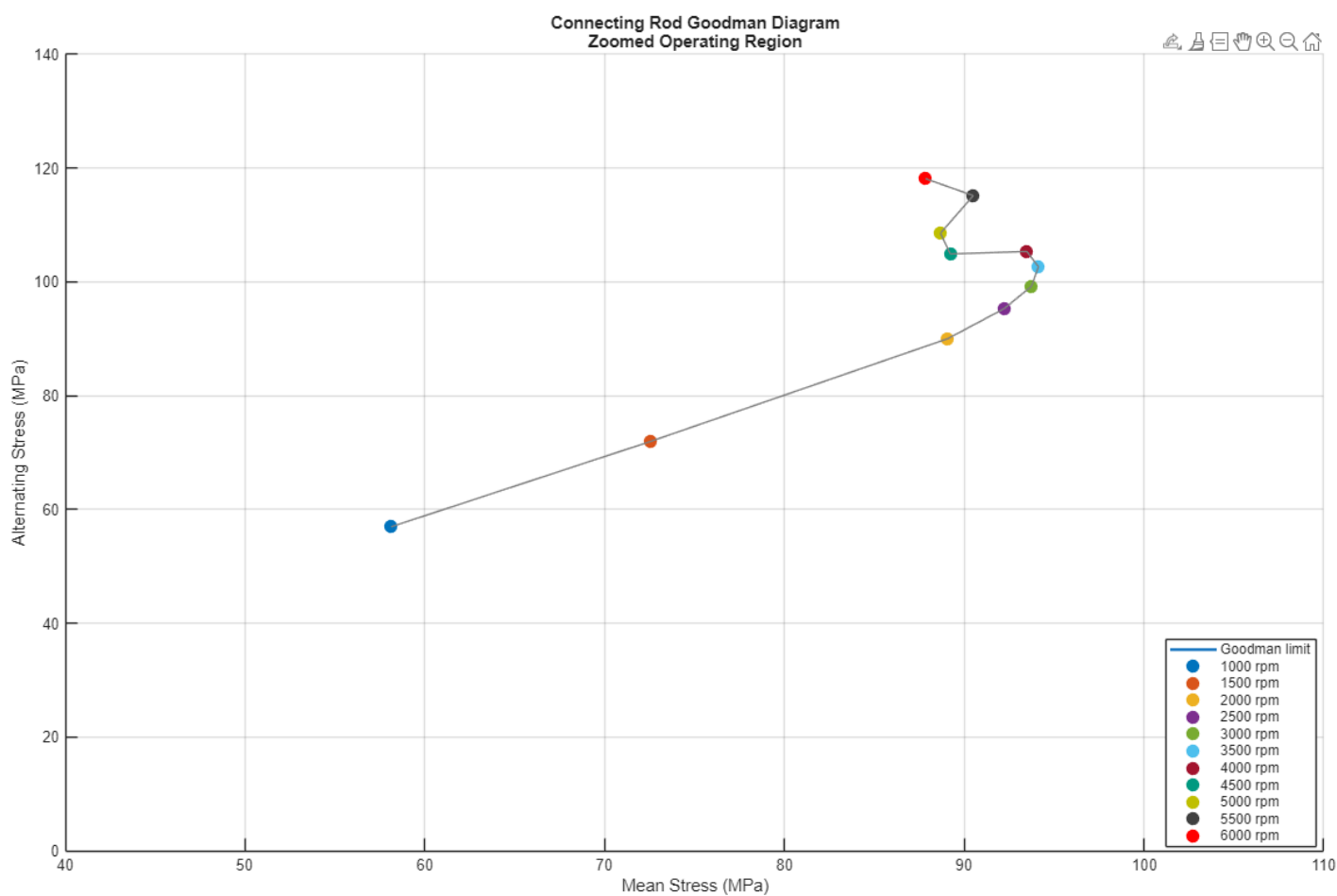
Connecting Rod



(Figure 4. Connecting rod critical rpm Goodman diagram)



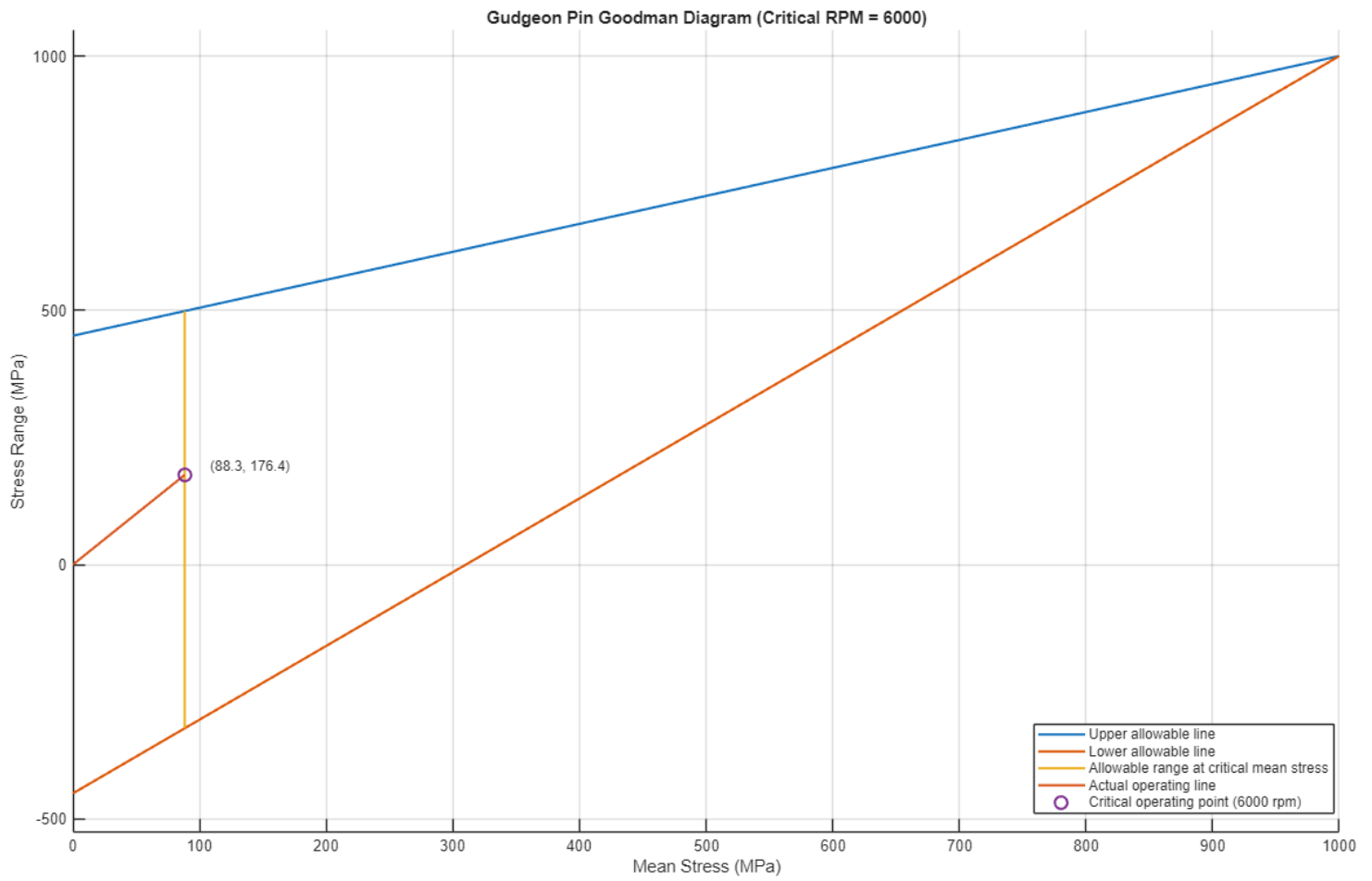
(Figure 5. Connecting rod Goodman Diagram)



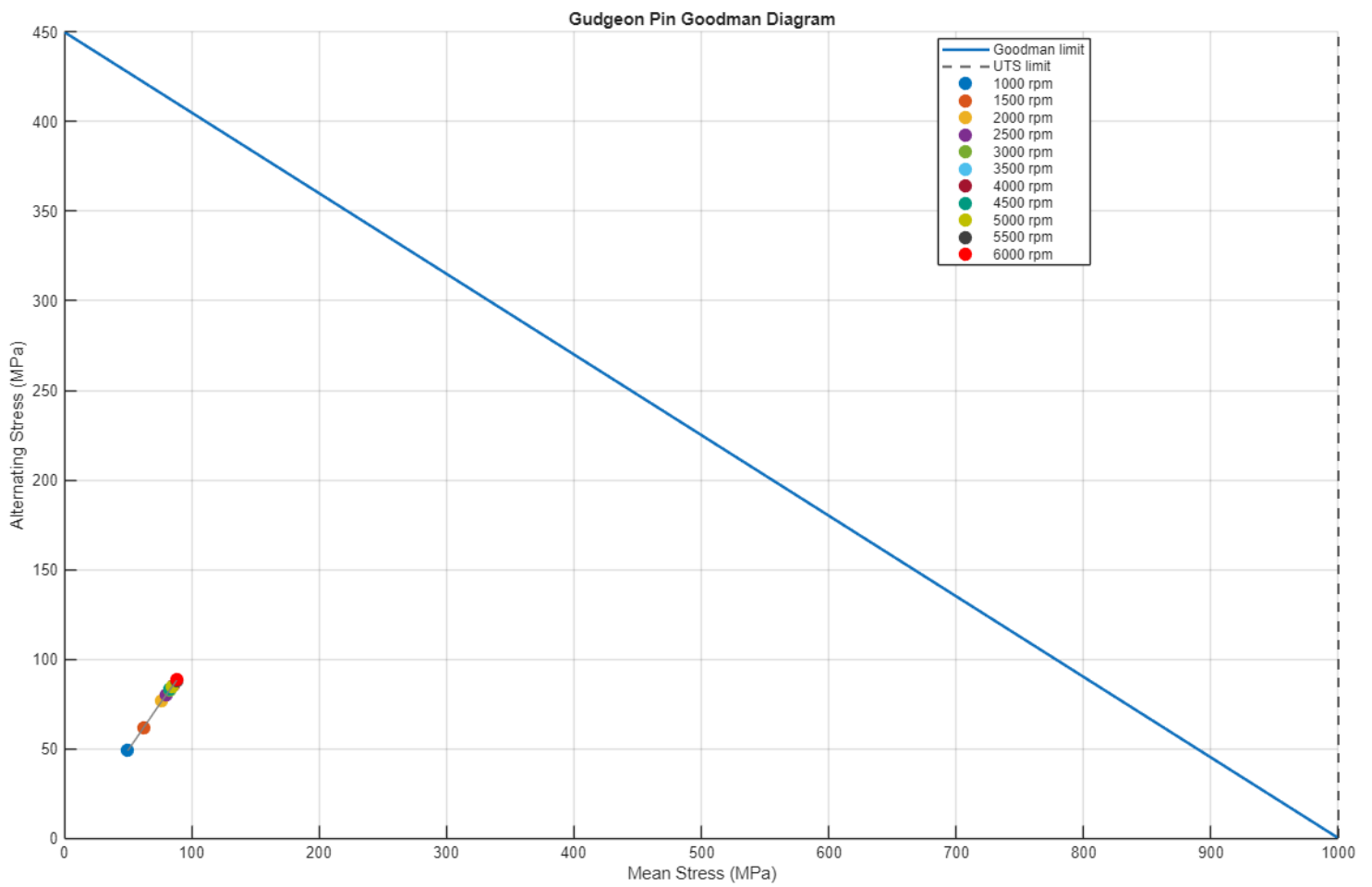
(Figure 6. Connecting rod Goodman diagram zoomed operating region)

The Goodman diagrams show that the connecting rod remains well below the fatigue failure limit across the full 1000–6000 rpm range, with all operating points clustered in a low-stress region. The minimum fatigue safety factor of 2.85 at 6000 rpm still represents a strong margin against failure, while values above 5 at low rpm indicate considerably more fatigue capacity than is required. The rise in alternating stress with rpm is mechanically expected, since increasing engine speed raises piston acceleration and therefore inertia loading, which increases the cyclic stress variation in the rod. Although 6000 rpm is the most severe condition, the component remains comfortably within the safe region, indicating that the connecting rod is unlikely to be the limiting component over the original operating range. This points to a conservative design with substantial durability margin but also suggests a degree of over-engineering under the analysed loading conditions.

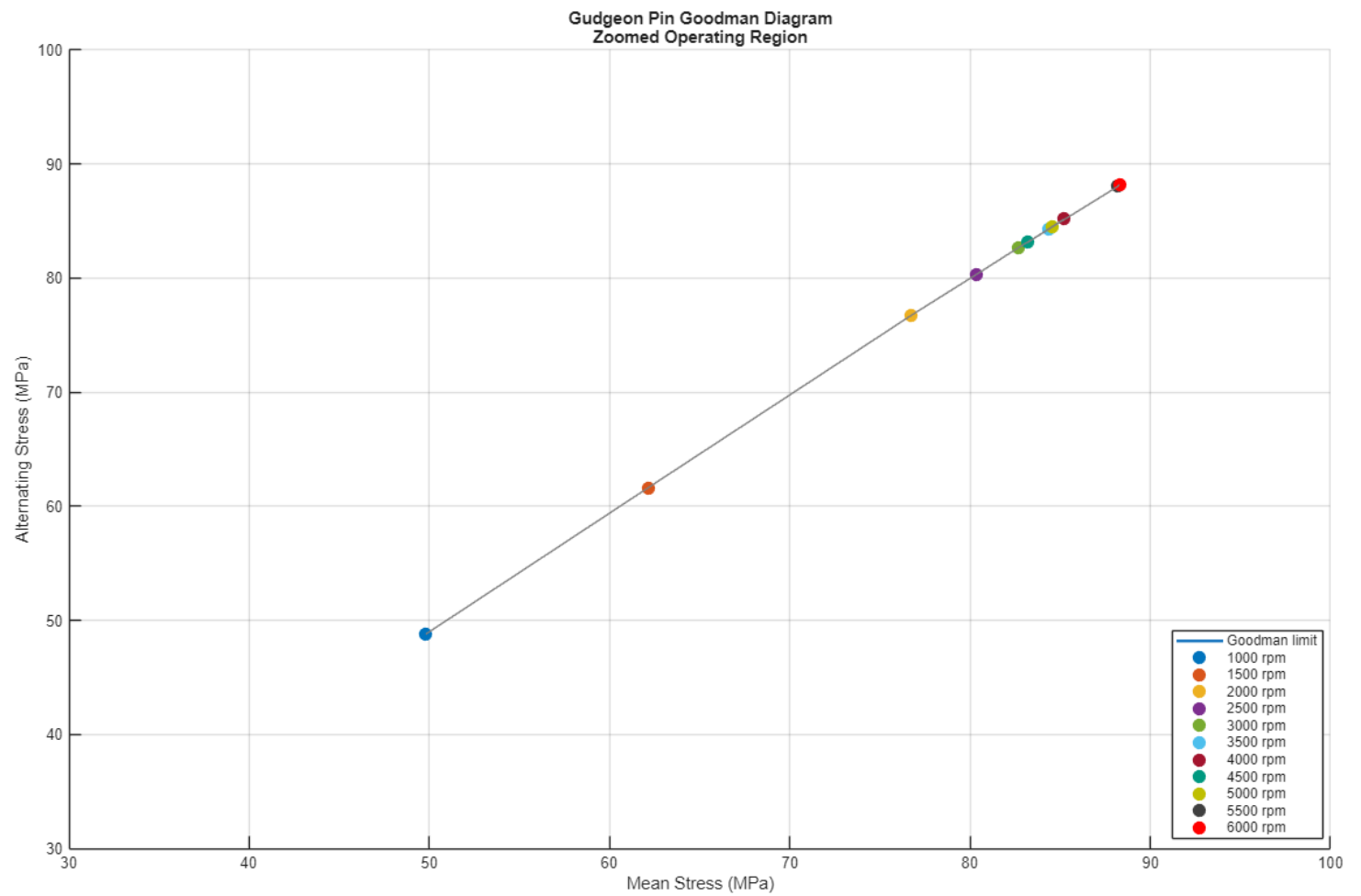
Gudgeon Pin



(Figure 7. Gudgeon pin critical rpm Goodman diagram)



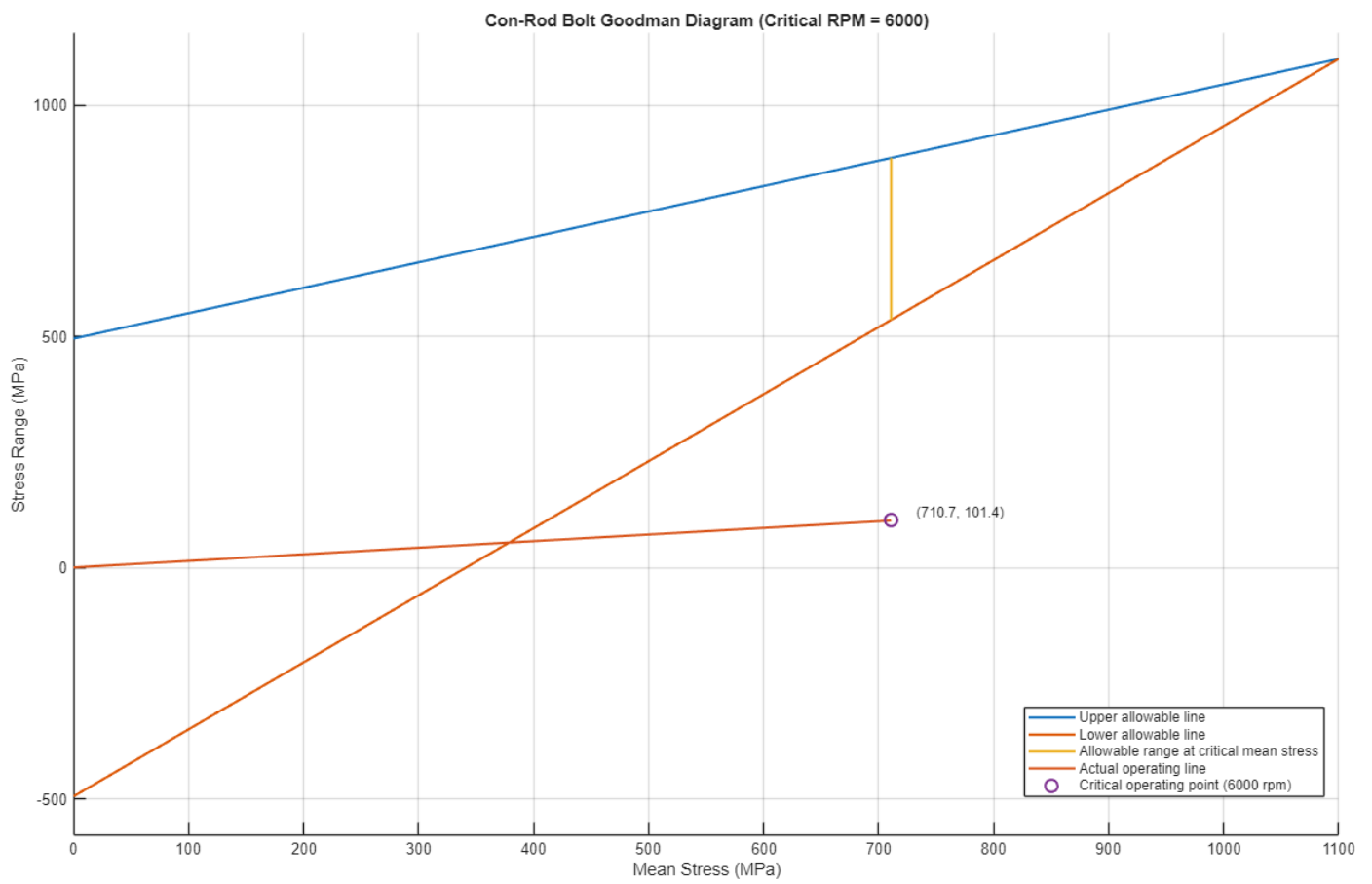
(Figure 8. Gudgeon pin Goodman diagram)



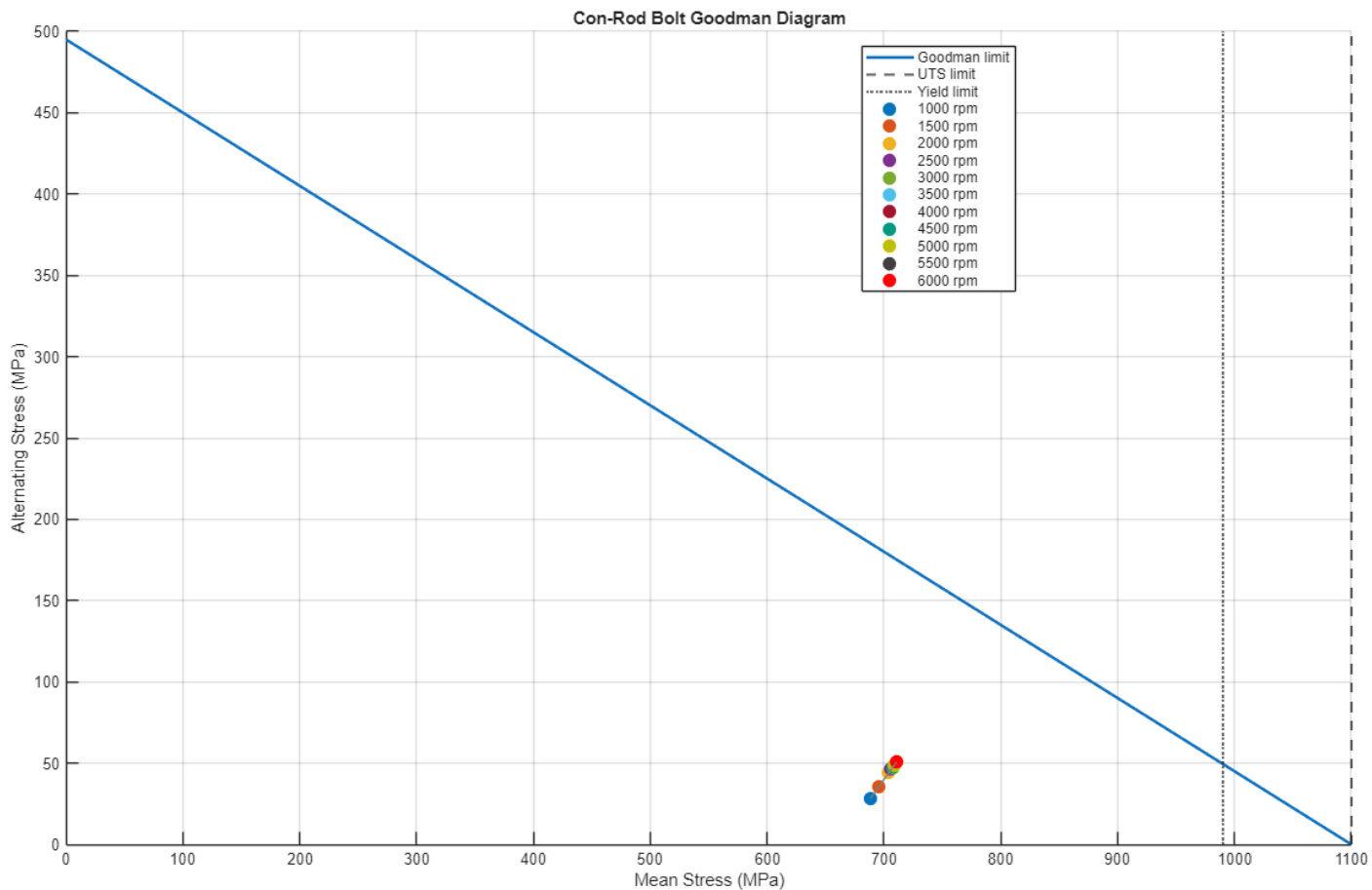
(Figure 9. Gudgeon pin Goodman diagram zoomed operating region)

The gudgeon pin Goodman diagrams show that the component remains safely below the fatigue failure boundary throughout the speed range, but with a smaller reserve than the connecting rod. Both mean and alternating stress increase progressively with rpm, with 6000 rpm giving the most severe condition, and fatigue safety factor falling from 6.32 at 1000 rpm to 3.52 at 6000 rpm. These values still indicate a durable design, but they also show that the gudgeon pin is more highly utilised relative to its allowable fatigue capacity than the connecting rod. However, the operating points remain comfortably within the Goodman limit, so the pin is not fatigue-critical over the original operating range. The results therefore suggest good life expectancy, but with less evidence of over-engineering than for the connecting rod.

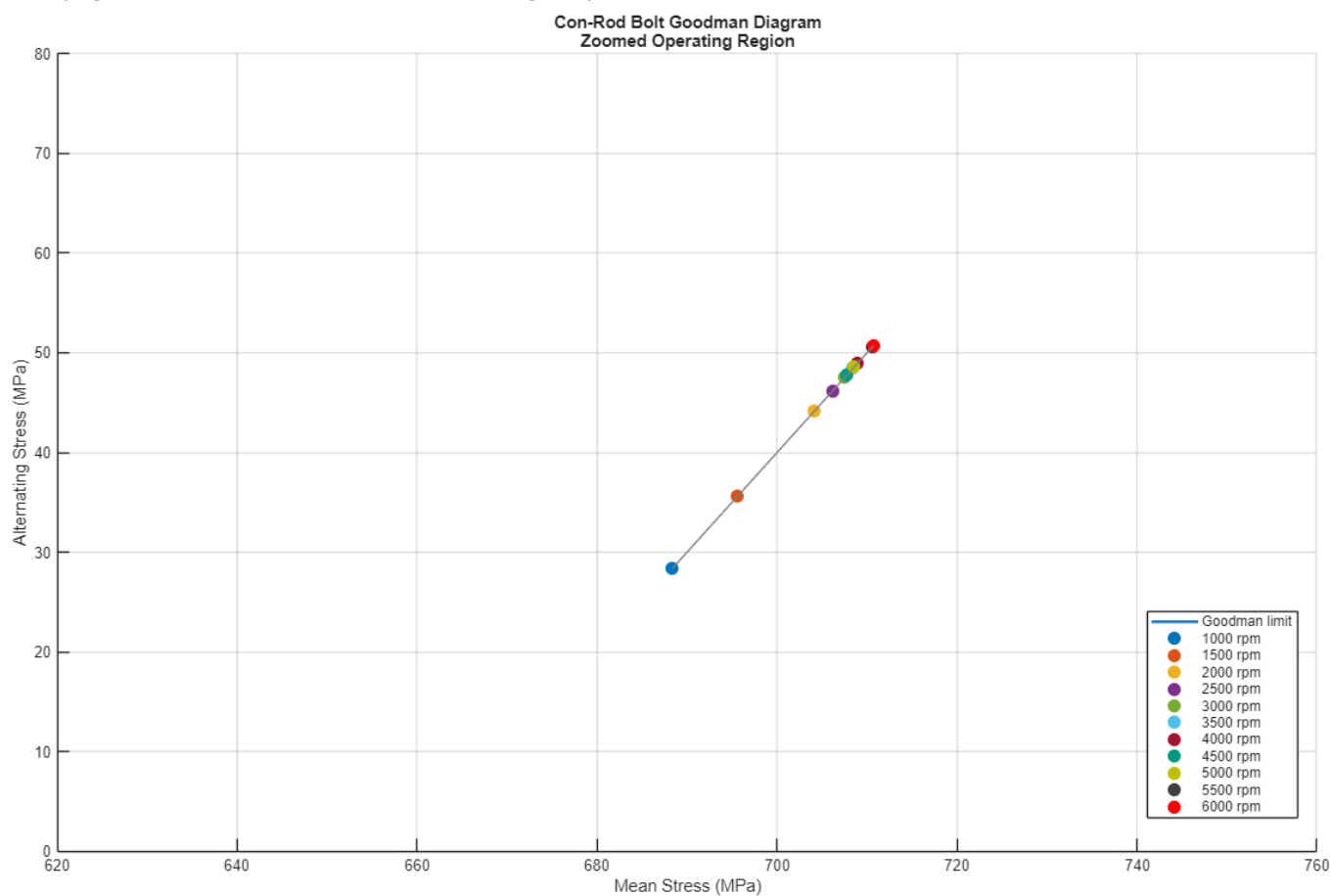
Con-rod bolt



(Figure 10. Con-rod bolt critical rpm Goodman diagram)



(Figure 11. Con-rod bolt Goodman diagram)



(Figure 12. Con-rod bolt Goodman diagram zoomed operating region)

The con-rod bolt remains within the safe region across 1000–6000 rpm, but it operates much closer to the fatigue limit than the connecting rod or gudgeon pin. Its fatigue safety factor falls from 1.46 to 1.34, making it the most highly utilised component in the assembly. Unlike the other components, the bolt carries a very high mean stress due to preload, while alternating stress rises only gradually with rpm. This shows that bolt fatigue is governed mainly by tensile mean stress rather than large stress fluctuation. Although the 6000-rpm condition is still acceptable, the small fatigue margin indicates a design that is adequate rather than conservative, so the bolt is the component most likely to become critical if engine speed increases.

Part 2 – Methodology

In Part 2, the pressure data were extended to 9000 rpm by interpolation and extrapolation across the original RPM range. The con-rod bolt stresses and Goodman fatigue safety factors were then recalculated using the same kinematic and preload model as in Part 1, allowing the effect of increased engine speed to be assessed.

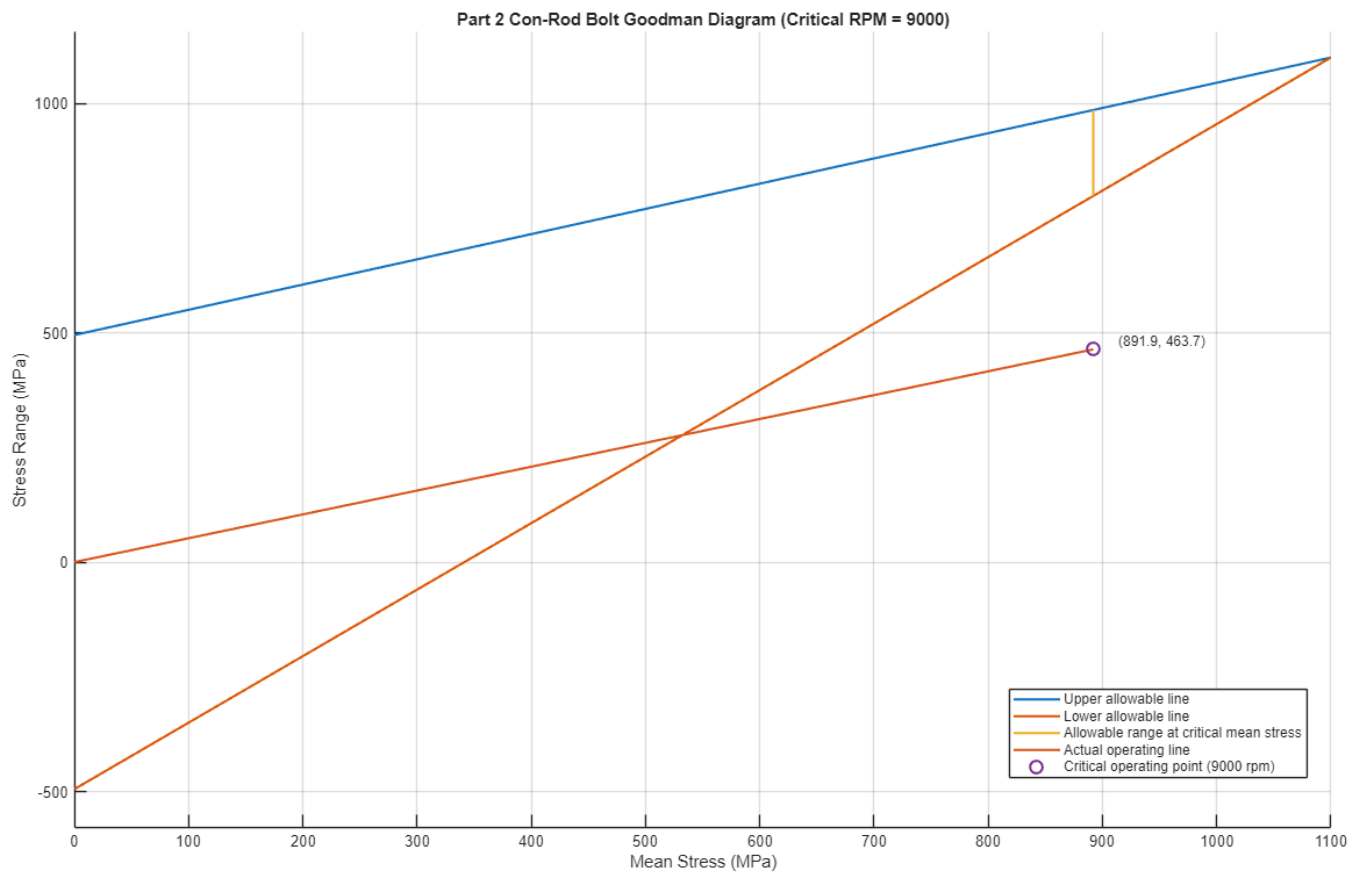
Part 2 – Results

RPM	BoltMax	BoltMin	BoltMean	BoltAlt	BoltFS
1000	716.66	660	688.33	28.329	1.4642
1500	731.12	660	695.56	35.562	1.4201
2000	748.2	660	704.1	44.098	1.3714
2500	752.31	660	706.16	46.157	1.3602
3000	755.01	660	707.51	47.505	1.3529
3500	756.91	660	708.45	48.454	1.3478
4000	757.91	660	708.95	48.955	1.3452
4500	755.6	660	707.8	47.799	1.3513
5000	757.12	660	708.56	48.562	1.3472
5500	761.28	660	710.64	50.639	1.3363
6000	761.44	660	710.72	50.718	1.3359
6500	759.8	660	709.9	49.9	1.3402
7000	768	660	714	54.002	1.3189
7500	780.31	660	720.15	60.153	1.2883
8000	835.62	660	747.81	87.808	1.1666
8500	949.17	660	804.58	144.58	0.97701
9000	1123.7	660	891.86	231.86	0.78175

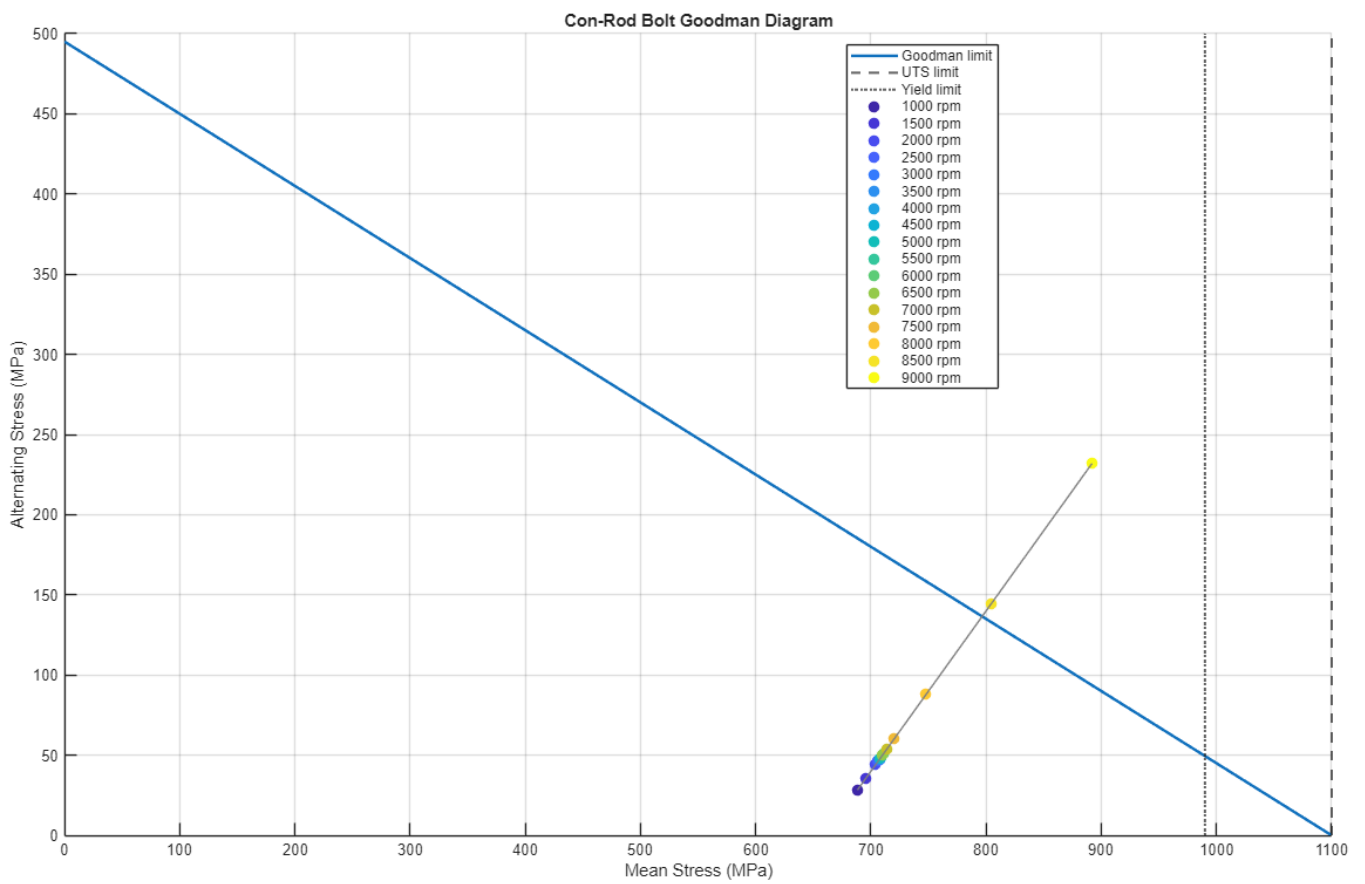
(Figure 13. Results for extended rpm range)

Extending the operating range to 9000 rpm caused a clear increase in con-rod bolt stress and a reduction in fatigue safety factor. While minimum stress remained constant at 660 MPa due to preload, maximum stress increased to 1123.7 MPa and alternating stress rose to 231.86 MPa at 9000 rpm. As a result, the fatigue safety factor fell from 1.34 at 6000 rpm to 0.78 at 9000 rpm, indicating that the bolt becomes unacceptable at the extended speed range.

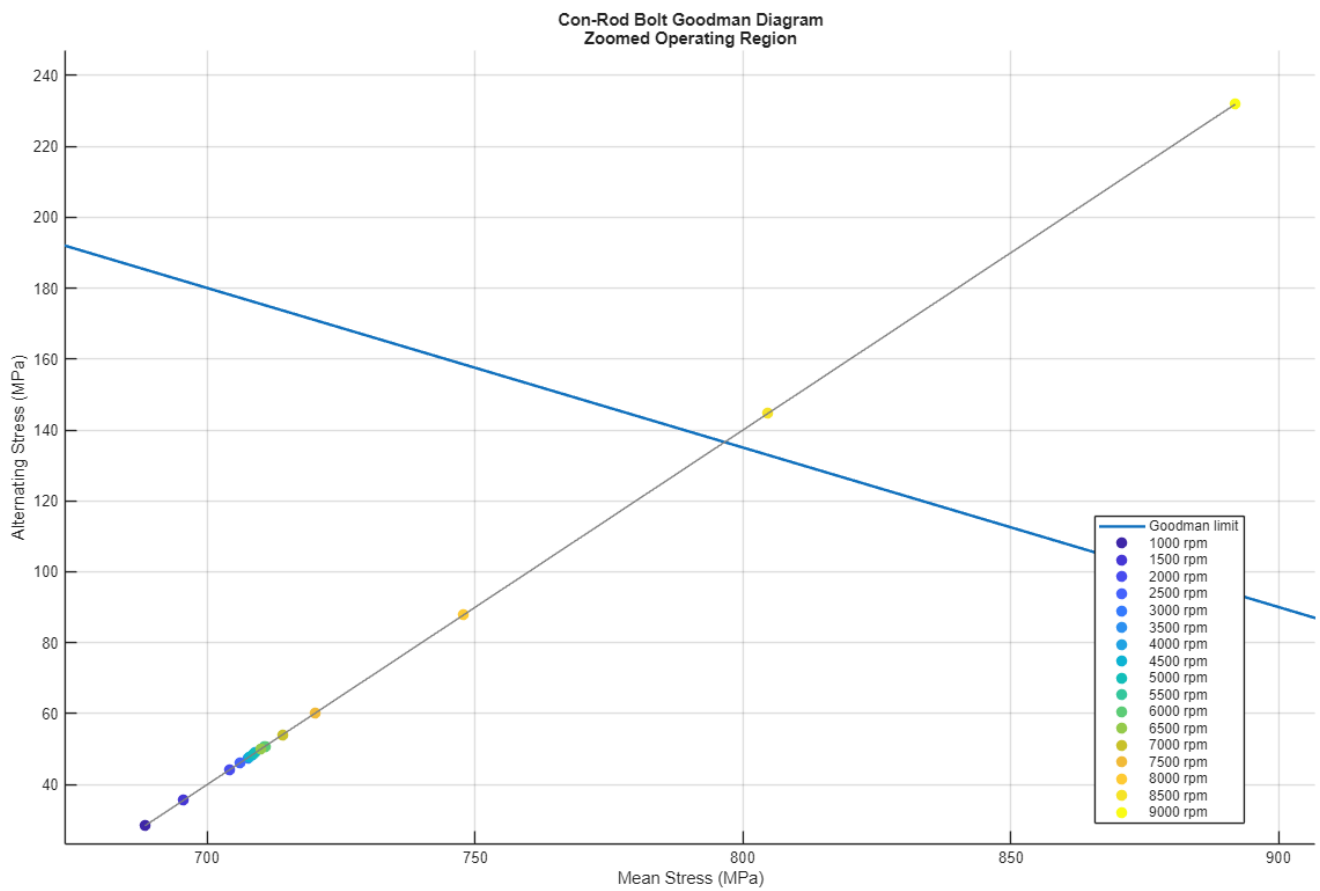
Part 2 – Analysis



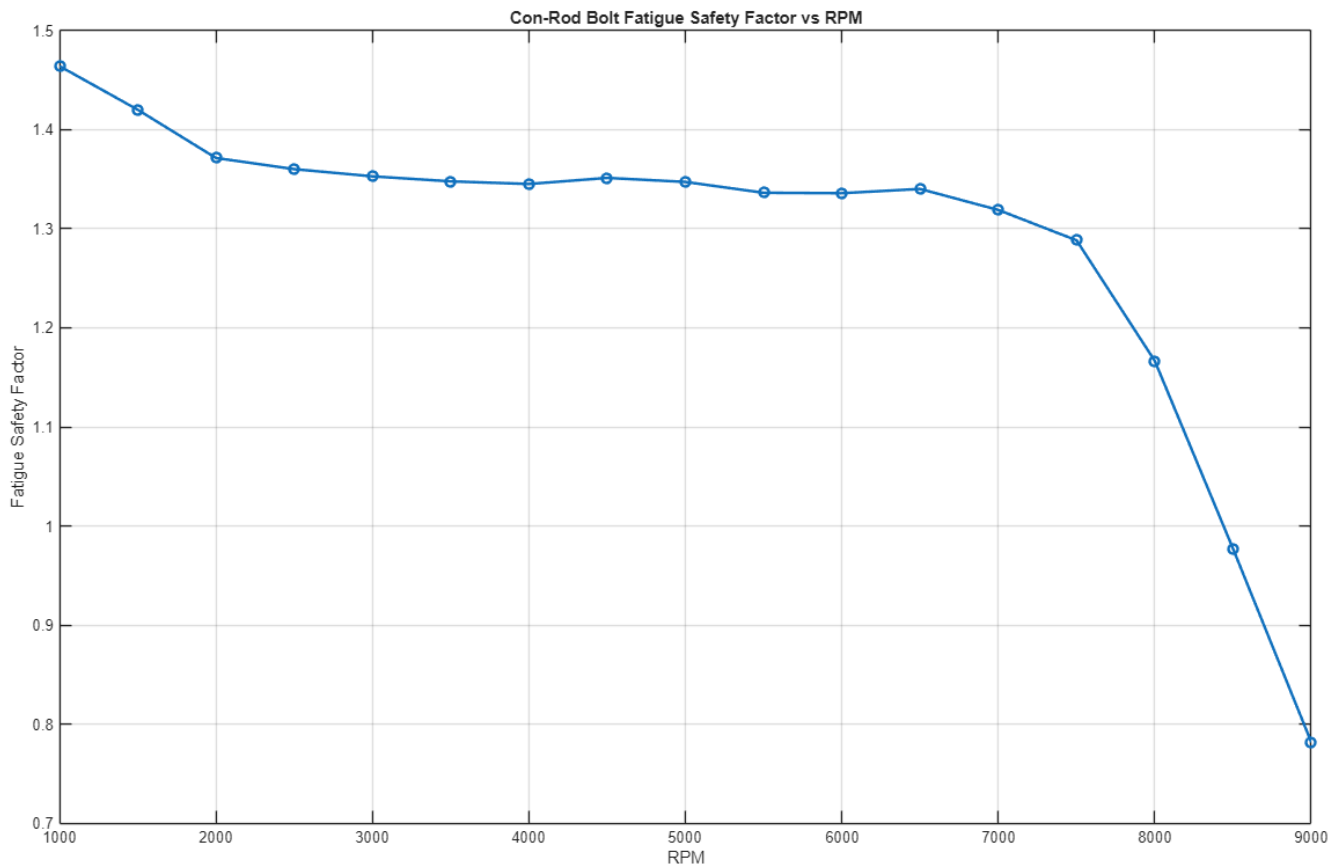
(Figure 14. Extended rpm con rod bolt critical rpm Goodman diagram)



(Figure 15. Extended rpm con-rod bolt Goodman diagram)



(Figure 16. Extended rpm range con-rod bolt Goodman diagram zoomed operating region)



(Figure 17. Extended rpm FSF against rpm range)

The Part 2 results show that extending the speed range beyond 6000 rpm has a severe effect on con-rod bolt fatigue performance. The FSF versus rpm plot remains stable up to around 7000–7500 rpm, indicating that the bolt still operates with an acceptable margin over this range, but beyond this point the safety factor drops sharply as engine speed increases. This is reflected in the Goodman diagrams, where the operating points move upward and to the right with rising mean and alternating stress. At 8500 and 9000 rpm the points exceed the Goodman limit and fatigue safety factor falls below 1, indicating that fatigue failure would be expected under sustained operation. This shows that the original bolt design is adequate over the baseline range but has very limited reserve capacity beyond it, making the bolt the limiting component at higher speeds.

Part 3 – Methodology

In Part 3, the con-rod bolt geometry was redesigned by carrying out a parametric sweep of bolt stress area at the 9000-rpm condition. For each trial area, the bolt stress history and fatigue safety factor were recalculated using the same loading model as Part 2, while keeping the preload force constant. The redesigned area was taken as the first value that restored the 9000-rpm fatigue safety factor to the target level taken from the original 6000 rpm condition.

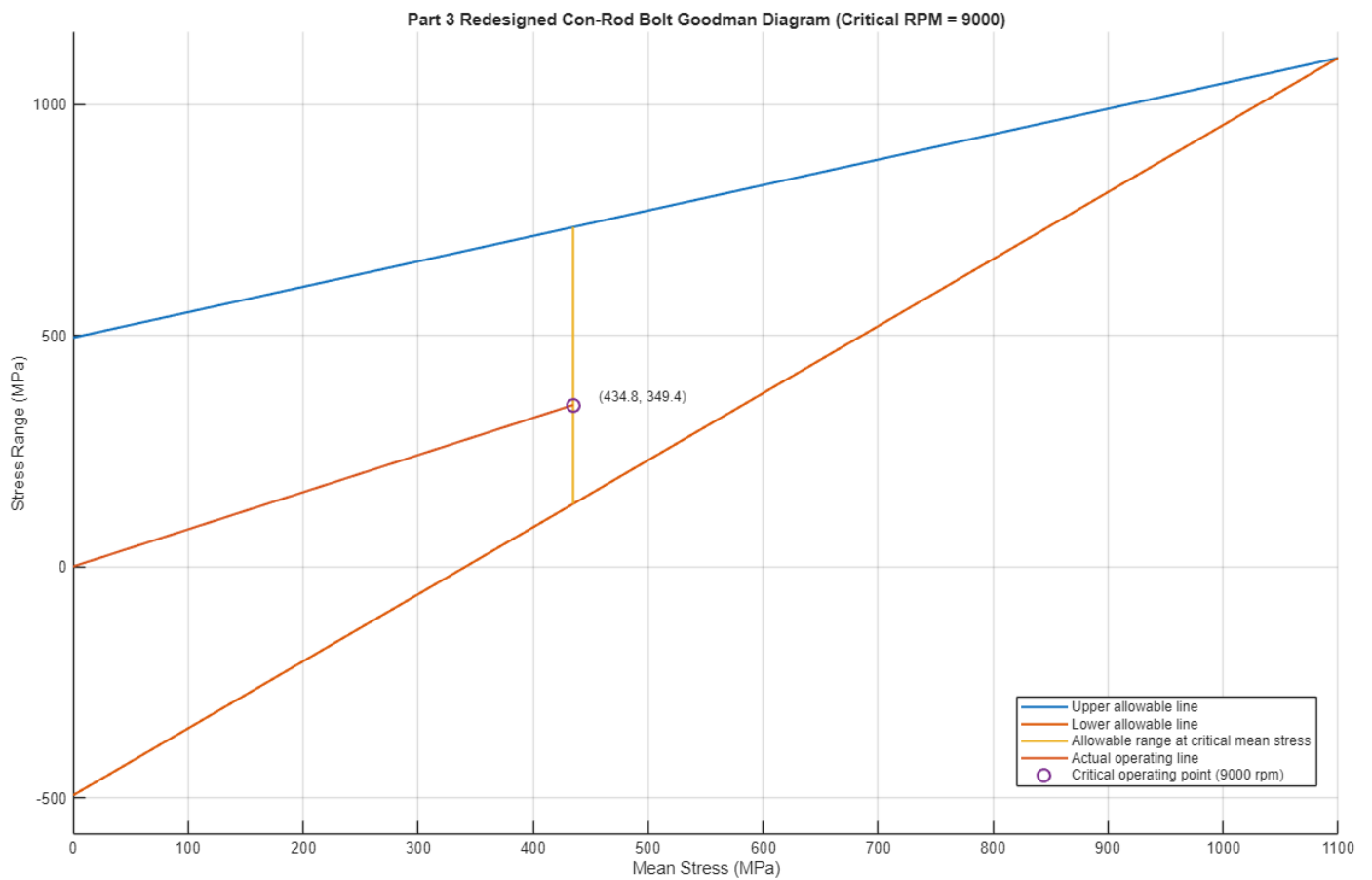
Part 3 – Results

OriginalArea_mm2	OriginalEqDiameter_mm	TargetFSF	RequiredArea_mm2	RequiredEqDiameter_mm	AchievedFSF
42.43	7.3501	1.3359	107.68	11.709	1.3366

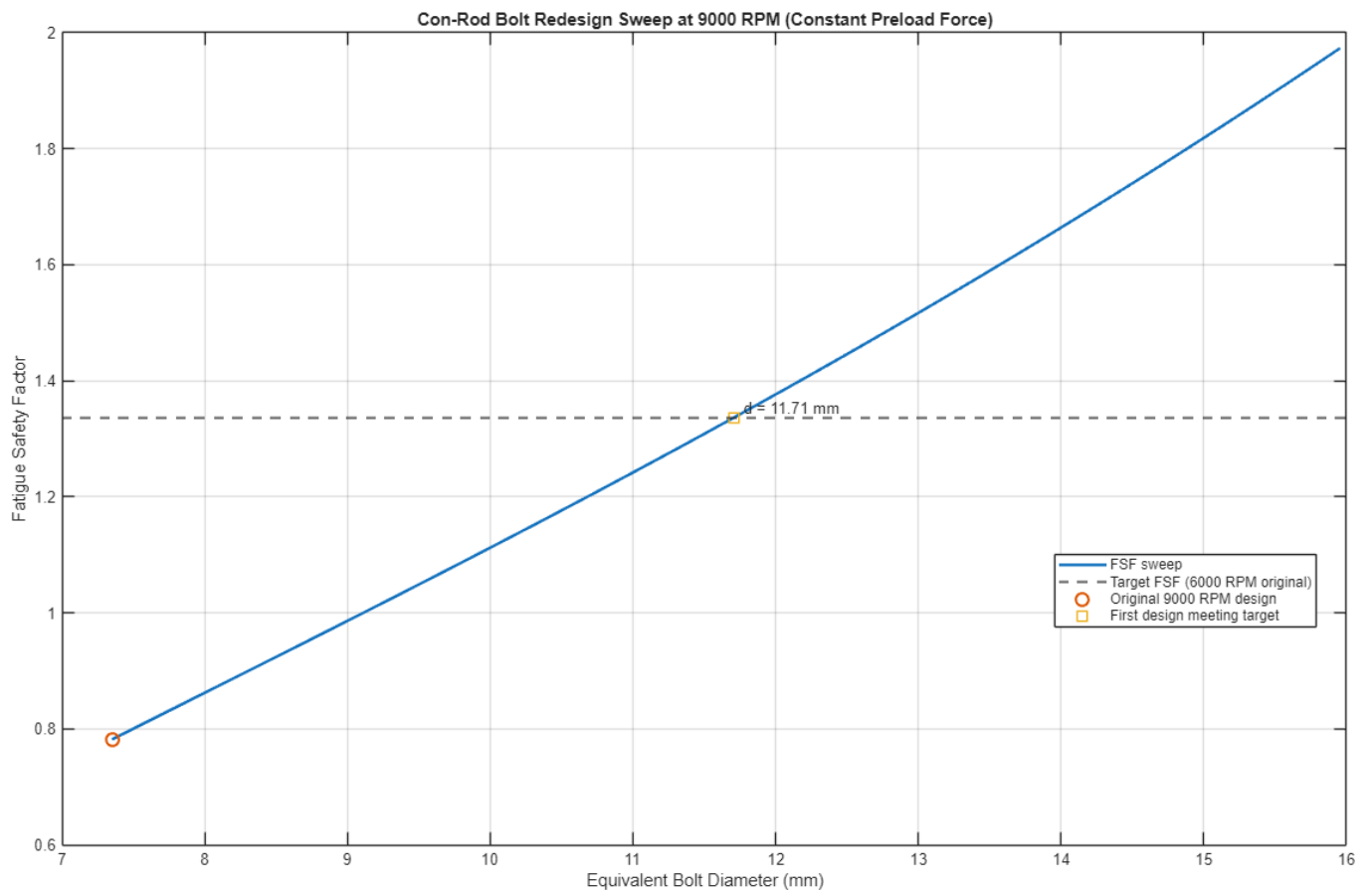
(Figure 18. Results for redesigned geometry)

The redesign results show that increasing the bolt stress area from 42.43 mm² to 107.68 mm² increased the equivalent diameter from 7.35 mm to 11.71 mm. This achieved a fatigue safety factor of 1.3366 at 9000 rpm, effectively matching the target value of 1.3359 taken from the original 6000 rpm condition.

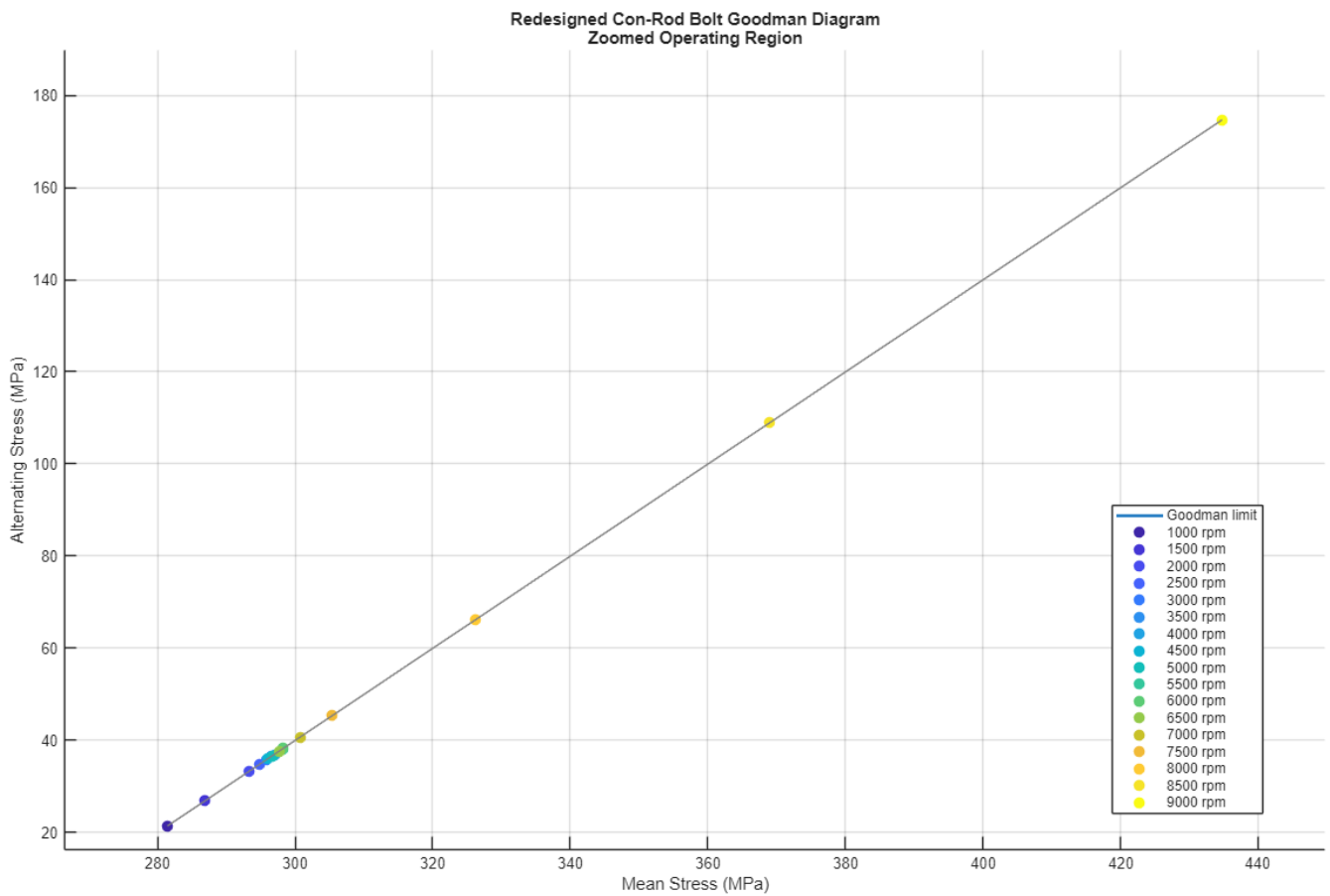
Part 3 – Analysis



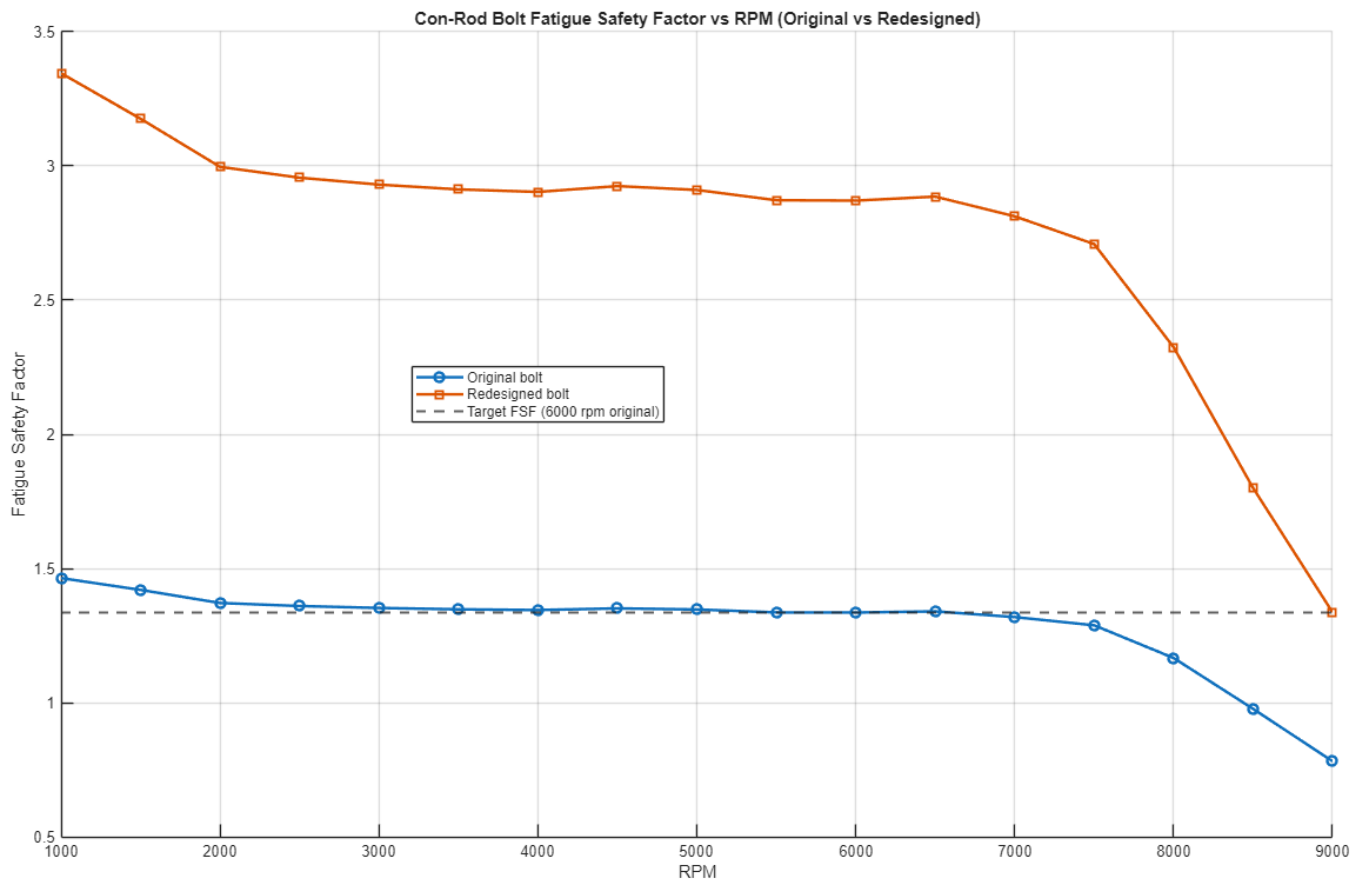
(Figure 19. Redesigned geometry con-rod bolt critical rpm Goodman diagram)



(Figure 20. Con-rod bolt redesign sweep until desired FSF value)



(Figure 21. Redesigned con-rod bolt Goodman diagram zoomed operating conditions)



(Figure 22. Con-rod bolt FSF value against rpm original compared to redesigned)

The Part 3 results show that the original bolt geometry was insufficient to maintain the required fatigue safety factor at 9000 rpm, and that a substantial increase in bolt stress area was required to restore acceptable performance. The redesign sweep demonstrates that the target fatigue safety factor of 1.3359 was reached at a stress area of 107.68 mm², corresponding to an equivalent diameter of 11.71 mm. This confirms that the original geometry did not provide adequate reserve capacity for the extended speed range, and that redesign of the bolt was necessary rather than optional. The FSF versus rpm comparison further shows that the redesigned bolt remains above the target safety factor across the full extended operating range, only approaching the target again at 9000 rpm. This indicates that the redesign was successful in restoring high-speed fatigue performance, but it also introduces a much larger safety margin at low and mid-range rpm than is strictly required. In this sense, the redesign is effective but somewhat conservative outside the critical high-speed condition. The redesigned Goodman diagrams support this conclusion, showing that the new operating points are shifted back into the safe region, while the critical 9000 rpm stress-range plot confirms that the redesigned operating point remains below the allowable stress range at the corresponding mean stress. Critically, this demonstrates that increasing bolt geometry alone can recover the required fatigue margin, but only at the expense of a large increase in cross-sectional area, which would carry penalties in mass, packaging and manufacturing practicality. Therefore, the redesign is mechanically successful, but the scale of the required increase also suggests that the original bolt was highly optimised for the baseline range and not intended for sustained operation at the extrapolated higher engine speeds.

Part – 3b Methodology

In Part 3b, the bolt redesign procedure was repeated for three different bolt materials: A4-80, the current model material and property class 12.9 steel. For each case, the material strength values were updated, the bolt area required to restore the target fatigue safety factor at 9000 rpm was recalculated, and the resulting fatigue performance over the full 1000–9000 rpm range was compared.

Material	UTS_MPa	YieldStrength_MPa	EstimatedEnduranceLimit_MPa
{ 'A4-80' }	800	600	360
{ 'Current Model' }	1100	990	495
{ '12.9' }	1220	1100	549

(Figure 23. Material properties for extended study)

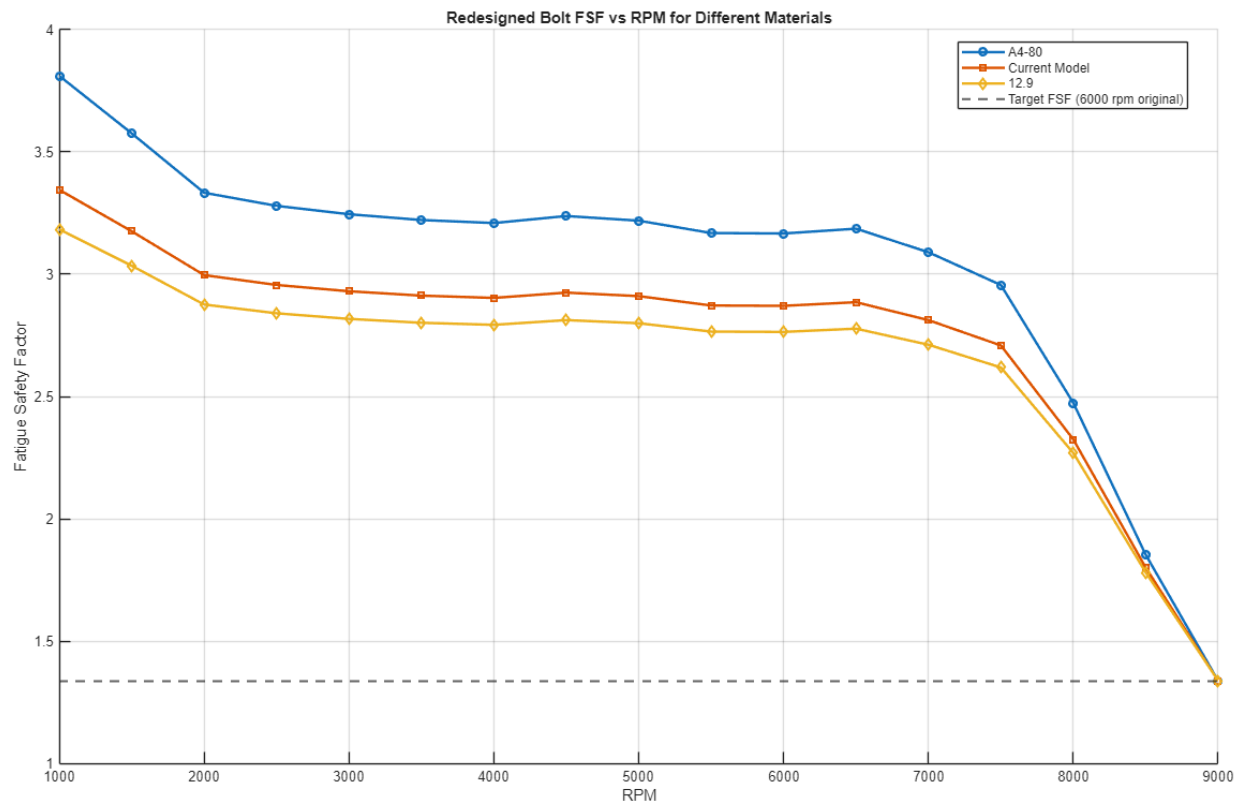
Part 3b – Results

Material	UTS_MPa	EstimatedSe_MPa	RequiredArea_mm2	RequiredEqDiameter_mm	AchievedFSF
{'A4-80' }	800	360	179.68	15.125	1.337
{'Current Model' }	1100	495	107.68	11.709	1.3366
{'12.9' }	1220	549	90.43	10.73	1.338

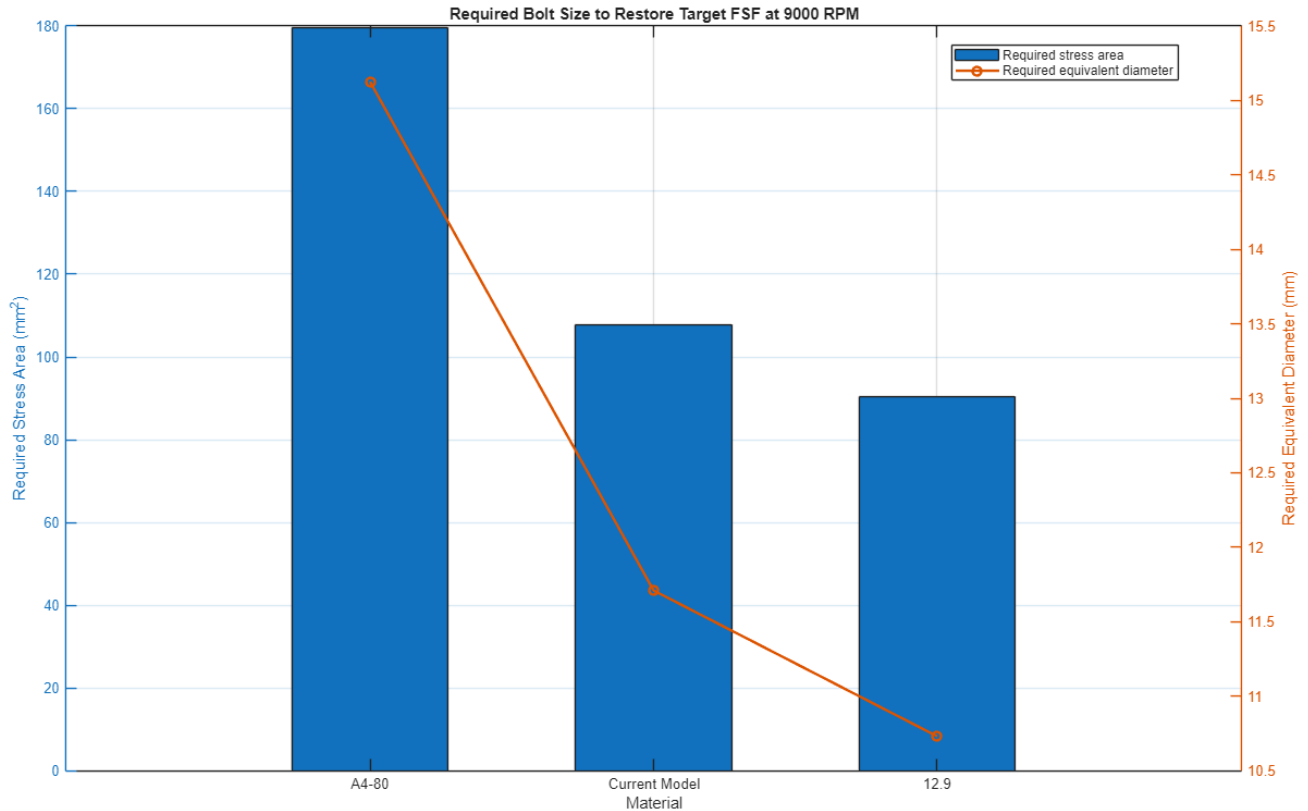
(Figure 24. Results table for Extended material study for con-rod bolt design)

The material comparison shows that increasing bolt material strength reduces the geometry required to restore the target fatigue safety factor at 9000 rpm. The weaker A4-80 material required the largest redesign, with a stress area of 179.68 mm² and equivalent diameter of 15.13 mm, while the current model material required 107.68 mm² and 11.71 mm. The stronger 12.9 material required the smallest redesign, reducing the required area to 90.43 mm² and equivalent diameter to 10.73 mm, while still achieving the target fatigue safety factor.

Part 3b – Analysis



(Figure 25. Extended material study for con-rod bolt FSF against rpm)



(Figure 26. Extended material required con-rod bolt size)

The material comparison shows that bolt material strength has a clear effect on the redesign required to restore fatigue performance at 9000 rpm. The weaker A4-80 material required the largest increase in geometry, while the stronger 12.9 material required the smallest, confirming that increasing material strength reduces the bolt size needed to achieve the target fatigue safety factor. The FSF versus rpm plot also shows that all three redesigned cases remain above the target value throughout the extended operating range and only approach it again at 9000 rpm, meaning each redesign is effective once the geometry is adjusted accordingly. Critically, however, the differences in fatigue safety factor across the rpm range are relatively small compared with the differences in required bolt size, which suggests that material selection mainly affects packaging and geometric efficiency rather than changing the overall fatigue trend. This indicates that adopting a stronger material such as 12.9 steel would be the more practical redesign solution, as it restores the required fatigue margin with a smaller increase in area and diameter than either the current material or the weaker A4-80 option.

Part 4 – Methodology

In Part 4, gas and inertia torque were calculated for a single cylinder using the pressure data and slider-crank model. These traces were then phase-shifted and summed for both the standard and cross-plane crankshaft configurations to obtain the instantaneous total crank torque, allowing the effect of crank arrangement on torque fluctuation to be compared.

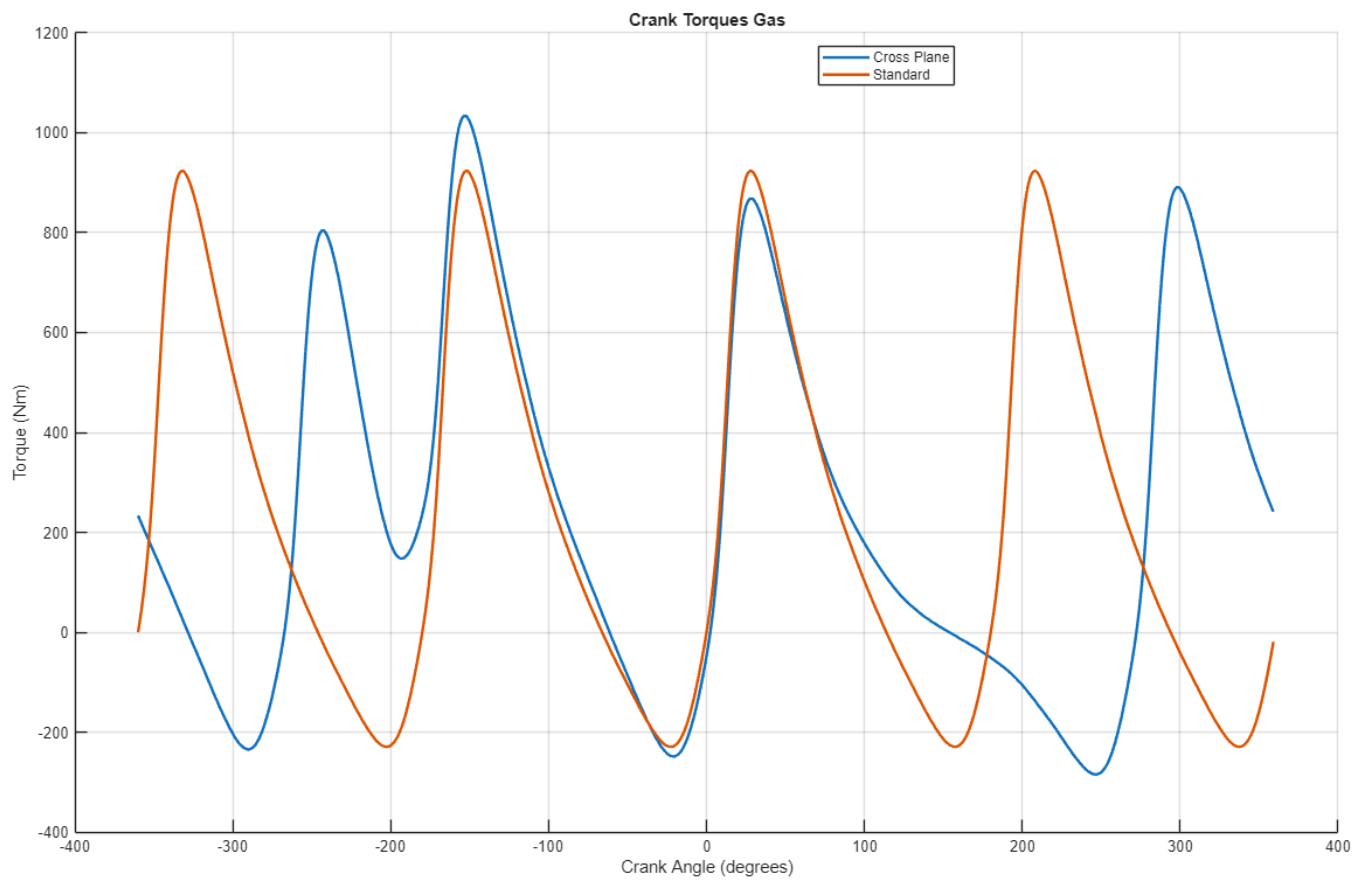
Part 4 – Results

Configuration	GasRange	InertiaRange	TotalRange	MeanTorque	PeakTorque	MinTorque
Standard / Flat-plane	1153.412	911.609	545.303	NaN	542.221	-3.082
Cross-plane	1318.776	42.862	1276.954	NaN	1013.779	-263.175

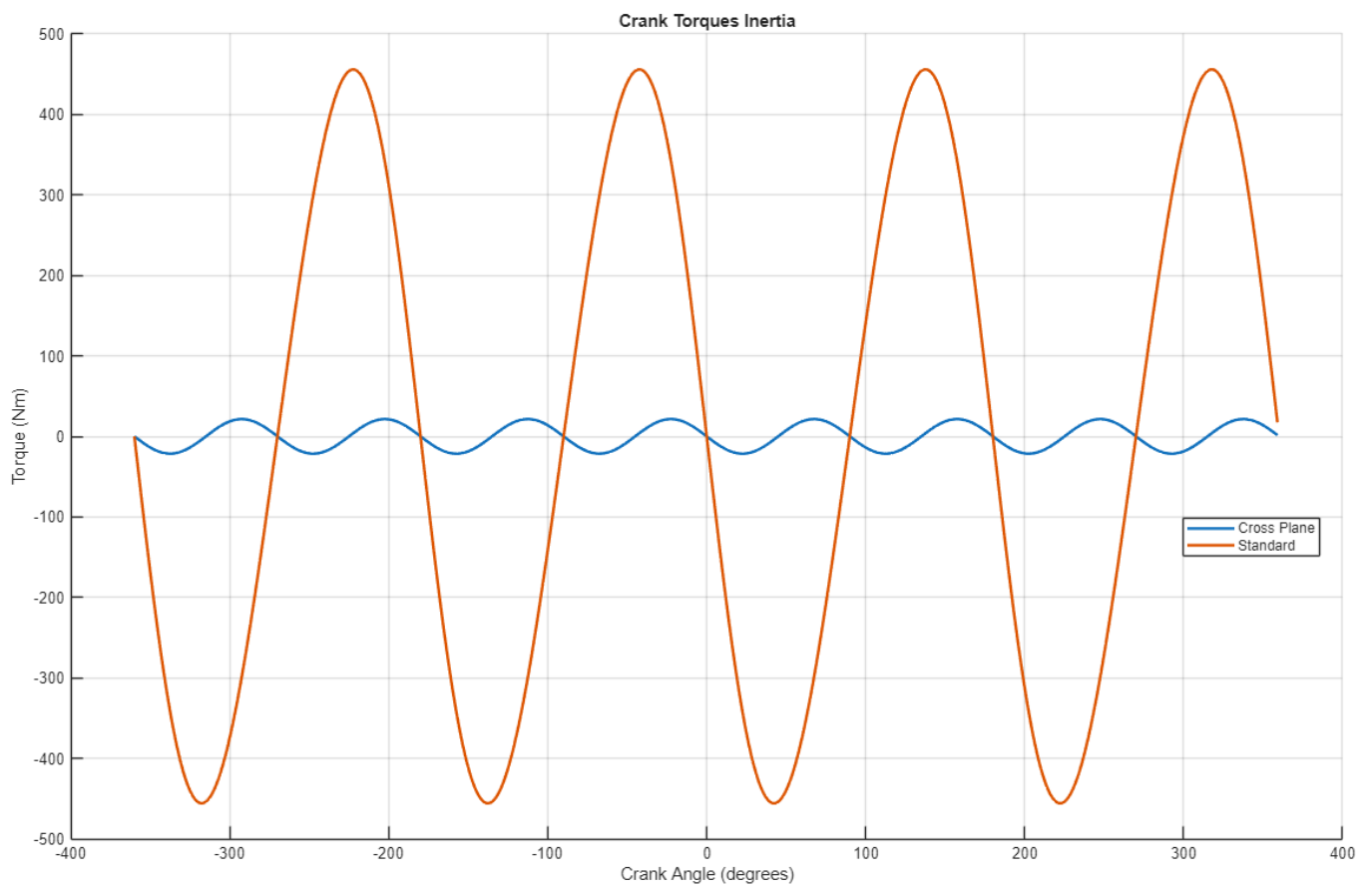
(Figure 27. results for torque values comparison between flat-plane and cross-plane)

The Part 4 torque results show that the cross-plane configuration produces a larger gas torque range and a much larger total instantaneous torque range than the standard flat-plane configuration. At 6000 rpm, gas torque range increased from 1153.41 Nm to 1318.78 Nm, while total torque range increased from 545.30 Nm to 1276.95 Nm. In contrast, inertia torque range reduced substantially from 911.61 Nm for the standard configuration to 42.86 Nm for the cross-plane configuration.

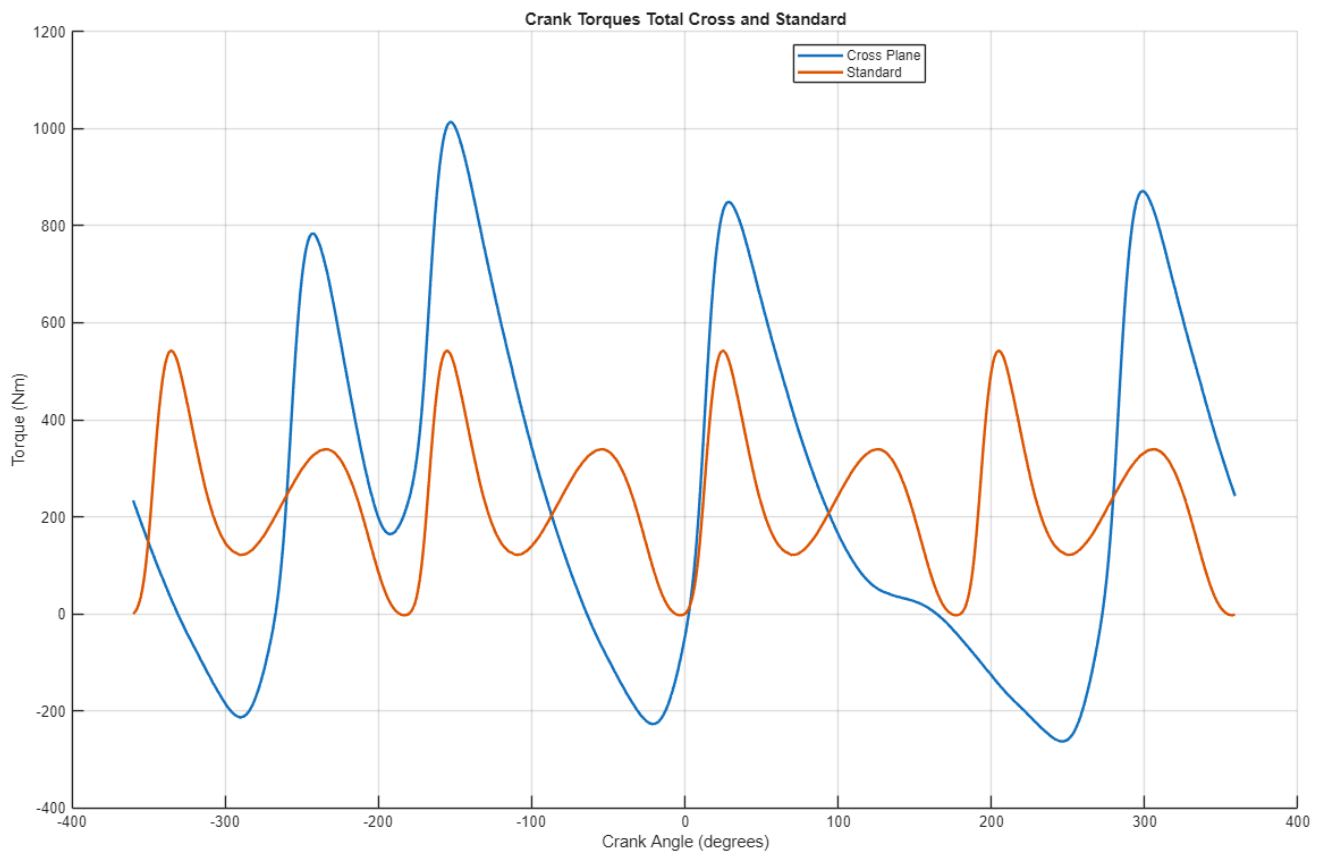
Part 4 – Analysis



(Figure 28. Crank gas torque comparison between flat-plane and cross-plane)



(Figure 29. Crank inertia torque comparison between flat-plane and cross-plane)



(Figure 30. Crank total torque comparison between flat-plane and cross-plane)

The torque plots show that the two crank configurations produce different instantaneous torque behaviour. The cross-plane arrangement reduces inertia torque fluctuation substantially, but the total torque plot shows a much larger overall torque range than the standard configuration. This means that, in the present model, the cross-plane setup improves inertia balance but does not give the smoothest total torque delivery, as gas torque phasing dominates the combined response. The standard configuration therefore appears more suitable where smoother torque output is required.

Conclusion

This report evaluated the fatigue behaviour of the connecting rod midpoint, gudgeon pin and con-rod bolt using pressure-based loading, slider-crank kinematics and Goodman analysis. Over 1000–6000 rpm, all components remained within the safe region, but the con-rod bolt was the most highly utilised, with the lowest fatigue safety factor of 1.34 at 6000 rpm, compared with 2.85 for the connecting rod and 3.52 for the gudgeon pin. Extending the range to 9000 rpm reduced bolt fatigue safety factor to 0.78, showing that the original bolt design was not suitable for sustained higher-speed operation. Increasing bolt stress area from 42.43 mm^2 to 107.68 mm^2 restored the 9000-rpm fatigue safety factor to 1.3366. The material study showed that stronger materials reduce the bolt size required to achieve this target, with 12.9 steel giving the smallest redesign. The torque study showed that the cross-plane arrangement reduced inertia torque fluctuation but produced a larger total torque range than the standard configuration. Overall, the con-rod bolt was the critical component, and any increase in operating speed would require redesign to maintain acceptable fatigue performance.

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