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**Analytical Assessment of
Engine Internal Components**

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AI Acknowledgement

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Abstract

This report presents an analytical assessment of the bottom-end components of the Ford EcoBoost 1.6 GDTi inline-four-cylinder engine, evaluating fatigue safety factors at the connecting rod midpoint, gudgeon pin, and connecting rod bolt under both standard and extended operating conditions. A kinematic model was developed using manufacturer-specified geometry and cylinder pressure data across 1000–6000 RPM, with the modified Goodman fatigue criterion applied throughout. At 6000 RPM, fatigue safety factors of 4.160, 3.795, and 3.715 were obtained for the midpoint, gudgeon pin, and bolt, respectively, confirming infinite fatigue life with substantial design margins. Linear extrapolation of pressure data to 9000 RPM revealed a 57.1% reduction in bolt FSF to 1.593, driven by the quadratic relationship between angular velocity and inertia force. A wasted shank bolt redesign incorporating rolled threads and reduced stress concentration restored the FSF to 3.435 at 9000 RPM, a 115.6% improvement, limited by a fundamental material yield constraint. Investigation of alternative bolt materials demonstrated that Ti-6Al-4V achieved the highest FSF despite the lowest UTS, attributable to its lower elastic modulus, which reduced dynamic load sharing. Instantaneous torque analysis confirmed the standard crankshaft configuration as optimal for the Ford EcoBoost operating envelope, with the cross-plane alternative offering lower inertia torque fluctuations suited to high-revving motorsport applications. The mean torque was identical across configurations at 229.8 Nm.

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1. Introduction

The connecting rod assembly is one of the most critically loaded subsystems within an internal combustion engine, experiencing cyclic alternating stresses from both combustion gas pressure and reciprocating inertia forces throughout every engine cycle (Shenoy and Fatemi, 2006). Fatigue failure of bottom-end components, particularly connecting rod bolts, remains a primary cause of catastrophic engine failure, as these small cross-section fasteners must withstand high mean stresses from preload combined with dynamic inertia loading at elevated engine speeds (Xu *et al.*, 2024). Quantifying the fatigue life of such components under both standard and extended operating conditions is therefore essential to ensuring structural integrity and informing design optimisation in both production and motorsport contexts.

This report presents an analytical assessment of the bottom-end components of the Ford EcoBoost 1.6 GDTi inline-four-cylinder engine. Cylinder pressure data over 1000–6000 RPM and manufacturer-specified component properties were used to calculate the forces, stresses, and fatigue safety factors at the connecting rod midpoint, at the gudgeon pin at point A, and at the connecting rod bolt. The modified Goodman fatigue criterion was applied throughout, consistent with established practice for high-cycle fatigue assessment of engine components (Stephens *et al.*, 2001), with thread stress concentration effects determined using Roark's formulations (Young and Budynas, 2002).

The aims of this work were to: quantify component fatigue safety factors under standard operating conditions; critically evaluate the design margins and potential over-engineering present in the OEM configuration; assess bolt integrity under extended operation to 9000 RPM; propose geometry and material modifications to restore adequate fatigue life at elevated speed; and compare instantaneous torque characteristics between standard and cross-plane crankshaft configurations.

2. Baseline Component Analysis

2.1 Methodology

A kinematic model was constructed for the Ford EcoBoost 1.6 GDTi engine using manufacturer-specified geometry ($R = 40.7$ mm, $L = 134$ mm, bore = 79 mm) and mass data (Reciprocating mass = 347 g, Rotational mass = 313 g). Piston acceleration was derived from the standard two-term approximation, incorporating first and second-order harmonic components. Gas forces were calculated from the provided cylinder pressure data across 720 crank angle positions at each of the eleven RPM points (1000–6000 RPM). The net force acting on the connecting rod was determined as the difference between inertia and gas forces, then resolved along the connecting rod axis. Fatigue safety factors were computed using the modified Goodman criterion, in which the allowable stress range is a function of the material's endurance limit and the mean stress relative to UTS (Stephens *et al.*, 2001). All calculations were performed in base SI units using MATLAB. The modified Goodman criterion was selected over alternatives such as the Gerber parabola or Soderberg line as it provides the most conservative infinite-life prediction for high-cycle fatigue, which is appropriate for engine components subjected to millions of loading cycles per operating hour.

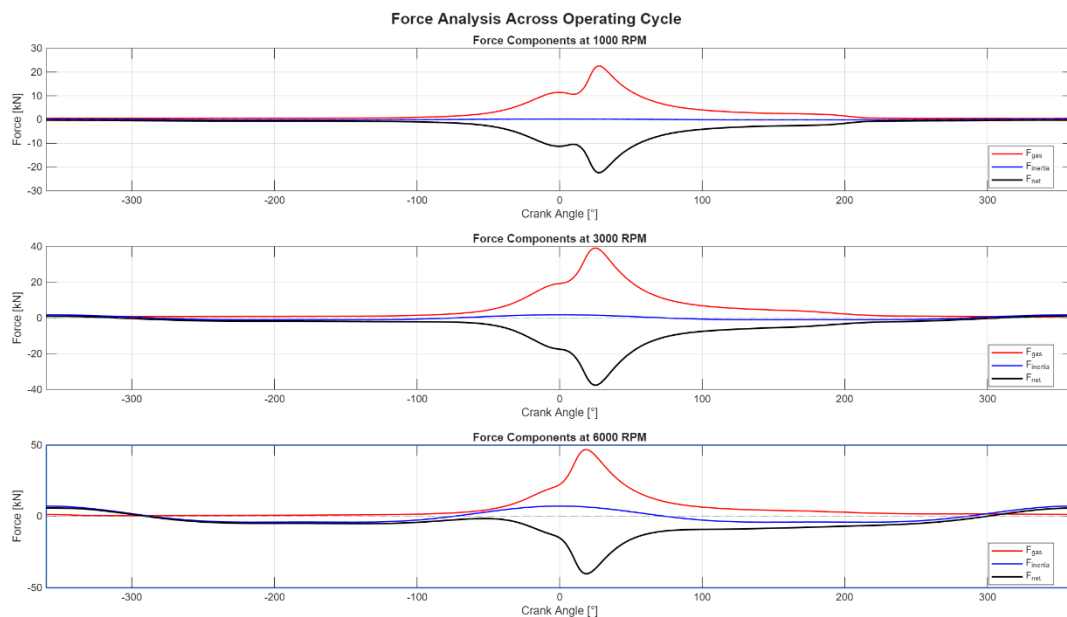


Figure 2.1 - Force analysis across the operating cycle at 6000 RPM, showing gas force, inertia force, and net force components versus crank angle.

2.2 Connecting Rod Midpoint

The midpoint experienced compressive stresses during combustion events and tensile stresses near exhaust TDC, with the stress range increasing with engine speed as inertia forces grew with the square of angular velocity. At 6000 RPM, the FSF was 4.160, indicating substantial over-engineering relative to the infinite-life threshold of 1.0. Even at the lowest analysed speed of 1000 RPM, the FSF remained high, reflecting the dominance of combustion-induced compressive loading at low speeds where inertia contributions are minimal. This is consistent with Shenoy and Fatemi (2005), who adopted a minimum failure index of 0.60 (equivalent to FSF ≈ 1.67) as their design constraint during connecting rod optimisation, confirming that production manufacturers embed substantial margins above this threshold to accommodate manufacturing variability and thermal effects. The corresponding Goodman diagram is presented in Figure 2.2.

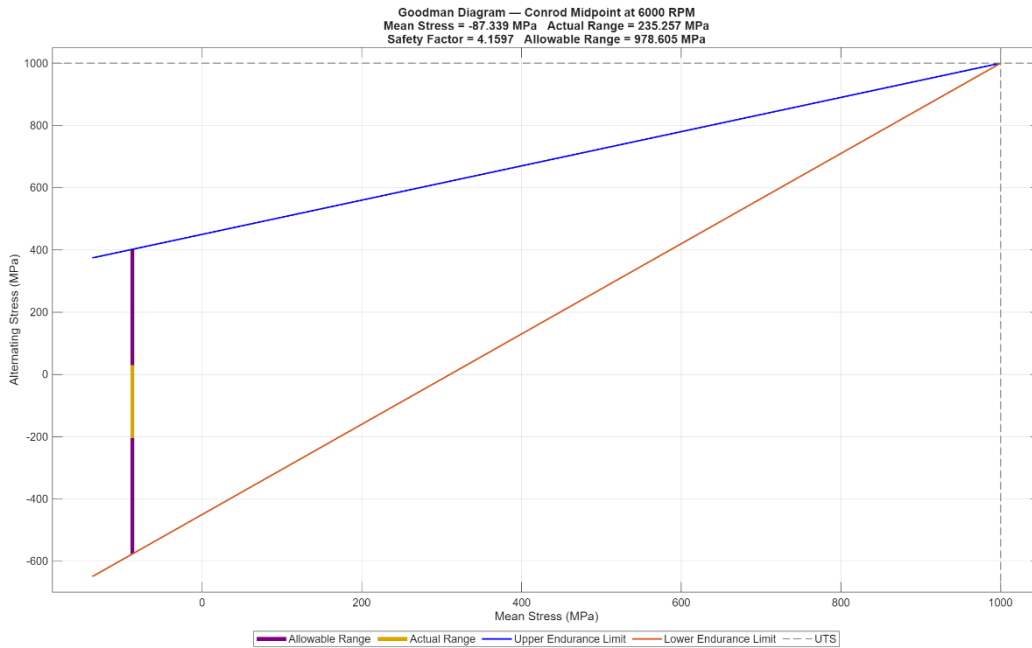


Figure 2.2 - Goodman diagram for the connecting rod midpoint at 6000 RPM, showing the operating point relative to the material failure envelope.

2.3 Gudgeon Pin at Point A

The gudgeon pin analysis employed a simplified three-point bending model with the bending moment calculated at the larger exposed distance $L_2 = 7.75$ mm. Resultant stresses were determined by combining bending and shear components using the relation $\sigma_{resultant} =$

$\sqrt{\sigma_{bending}^2 + 2\sigma_{shear}^2}$. Critically, the shear stress formulation differs between load cases: during

combustion, inertia provides partial relief at the support $\sigma_{shear} = \frac{\frac{F_{net}}{2} - F_{pin inertia}}{A}$, whereas during

peak inertia, all forces act in tension with no counterforce $\sigma_{shear} = \frac{F_{net}}{A}$. At 6000 RPM, the gudgeon pin FSF was 3.795, marginally below the midpoint value but still well above unity. This is consistent with the OEM design philosophy, in which the gudgeon pin, being a precision-ground component with a favourable surface finish, can tolerate tighter safety margins than the beam section. The Goodman diagram is presented in Figure 2.3.

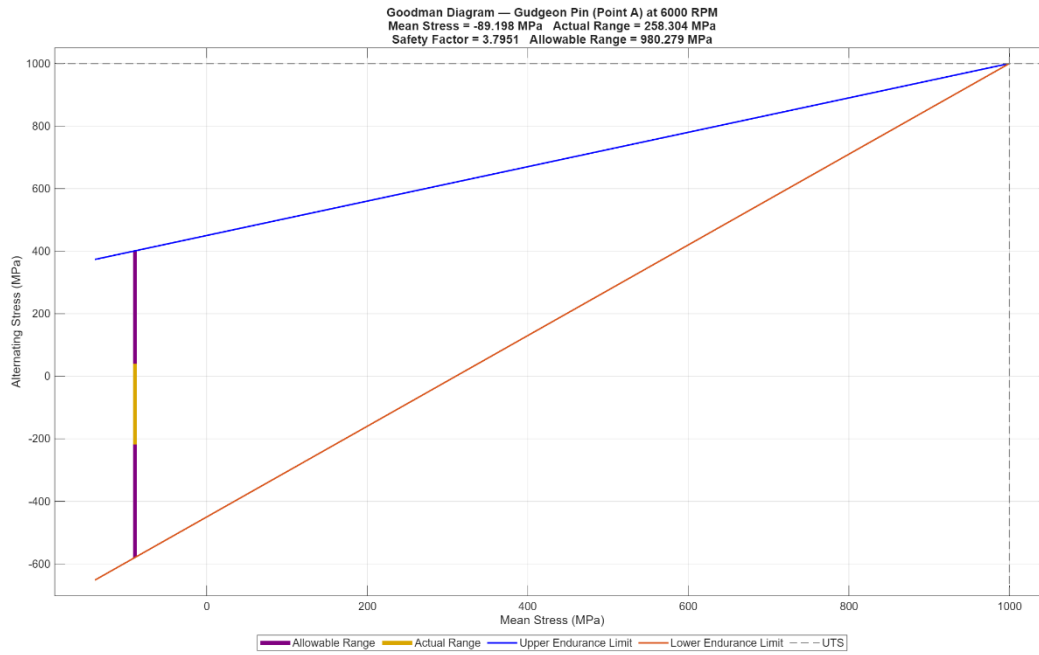


Figure 2.3 - Goodman diagram for the gudgeon pin at point A at 6000 RPM.

2.4 Connecting Rod Bolt

The bolt analysis incorporated the additional rotational inertia component (rod whip) not present in the midpoint or pin calculations, since the bolt must support both the reciprocating assembly and two-thirds of the connecting rod mass acting as a rotational load. A thread stress concentration factor of $K_t = 4.32$ was applied to the endurance limit, derived from Roark's two-sided U-notch model for an M8×1.25 thread (Young and Budynas, 2002). Bolt preload was calculated as $\sigma_{(axial)} = 0.6 \times 0.9 \times \sigma_{(Yield)} = 534.6$ MPa, with the additional dynamic force determined through stiffness-based load sharing. At 6000 RPM, the bolt FSF was 3.715 with the SCF applied, the lowest of the three components. It should be noted that the preload derivation assumes ideal lubricated conditions; frictional variations in service would shift the mean stress and, correspondingly, affect the FSF, confirming the bolt as the critical element for fatigue life, consistent with the observation that connecting rod bolt failure is the predominant mechanism of catastrophic bottom-end failure, with Liu *et al.* (2025) reporting that 65% of bolt fatigue fractures initiate at the first engaged thread, where stress concentration is highest. The Goodman diagram is presented in Figure 2.4.

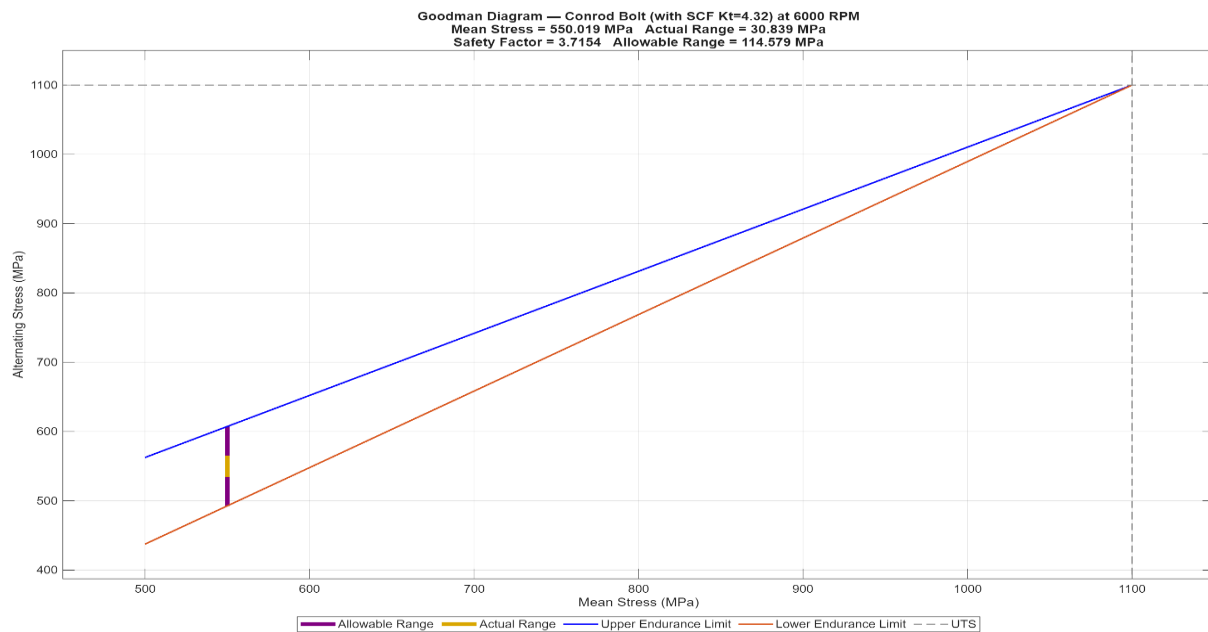


Figure 2.4 - Goodman diagram for the connecting rod bolt at 6000 RPM, incorporating Kt = 4.32.

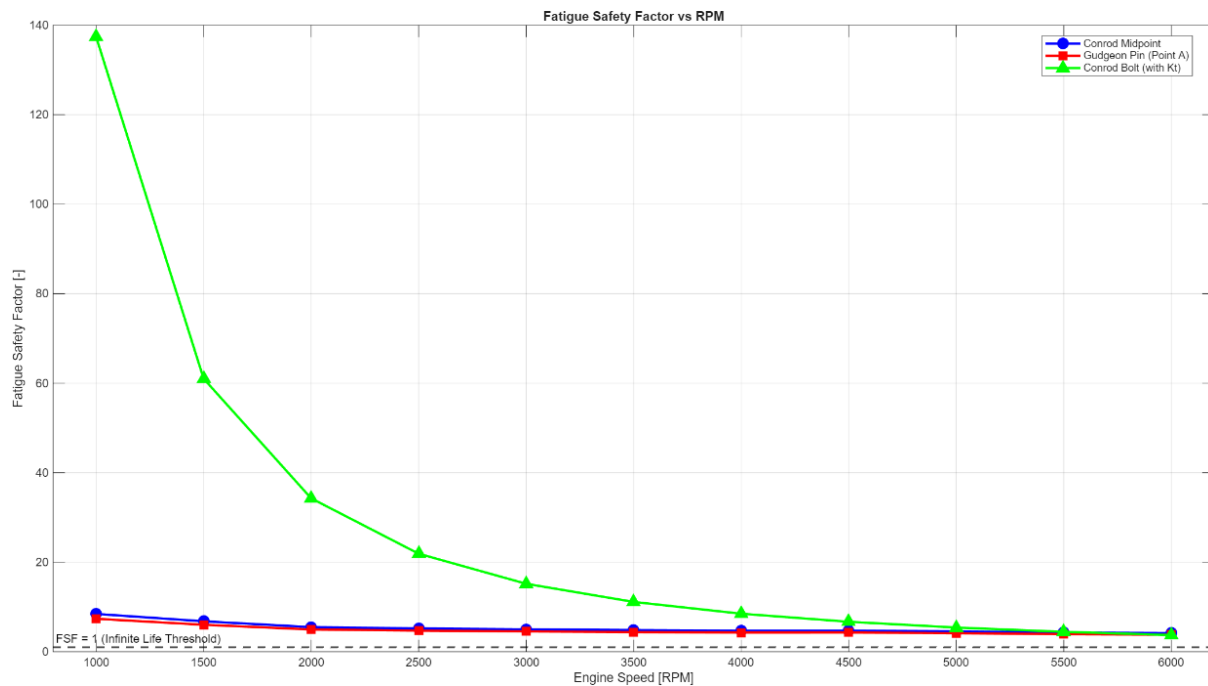


Figure 2.5 - Fatigue safety factor versus RPM for all three components across the 1000–6000 RPM operating range.

Table 2.1 — Summary of fatigue safety factors at 6000 RPM for all three analysed components.

Component	UTS (MPa)	EL _{eff} (MPa)	σ_{mean} (MPa)	σ_{range} (MPa)	FSF
Conrod Midpoint	1000	450.0	−87.3	235.3	4.160
Gudgeon Pin (Pt A)	1000	450.0	−89.2	258.3	3.795
Conrod Bolt (Kt=4.32)	1100	114.6	550.0	30.8	3.715

3. Extended RPM Analysis

3.1 Extrapolation Method

The cylinder pressure data were extended from 6000 to 9000 RPM in 500-RPM increments using per-crank-angle linear regression across all 11 original data points. This method was selected after evaluating cubic spline, PCHIP, and quadratic polynomial alternatives, all of which produced unphysical results during extrapolation. The cubic spline generated negative pressures at 227 of 720 crank angles, while quadratic fitting caused peak pressures to decrease at high RPM, which is physically unreasonable for a turbocharged engine maintaining boost pressure. Linear regression yielded zero unphysical values and physically reasonable peak pressure growth, though it represents an extrapolation of 50% beyond the measured range, assuming unchanged pressure trends. Moreover, linear regression tends to overestimate pressure growth at high RPM, leading to lower FSFs, and the appropriate direction for safety-critical design assessment should be conservatively evaluated. In practice, real engines experience volumetric efficiency losses and potential compressor surge at extreme speeds, so the extrapolated pressures should be treated as indicative rather than precise. Furthermore, the extrapolation does not account for changes in combustion phasing, knock-limited spark advance, or fuel enrichment strategies typically employed by OEMs at elevated engine speeds.

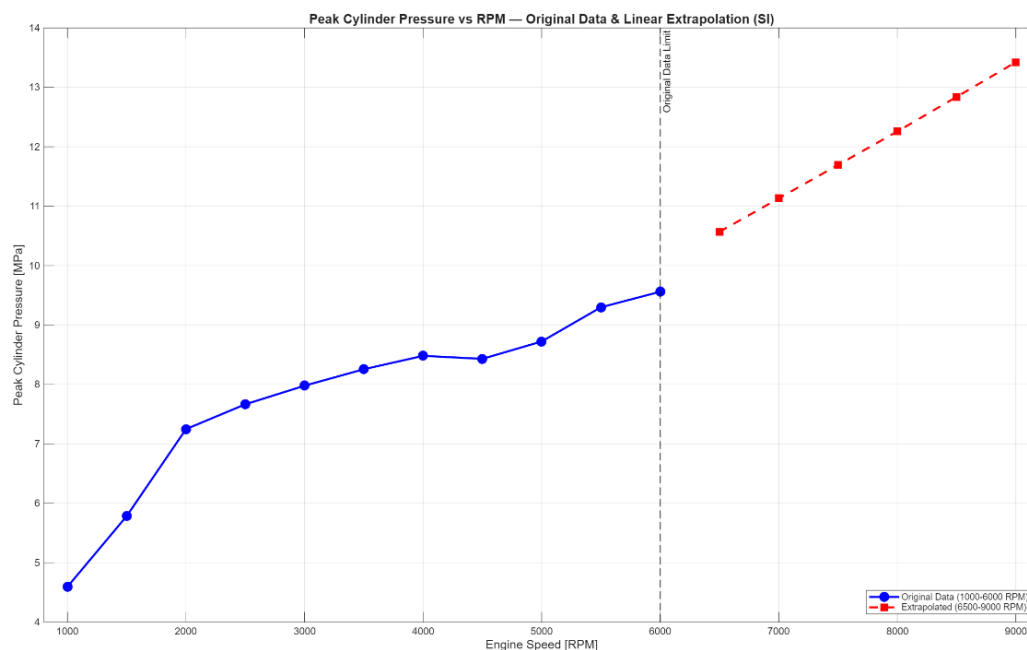


Figure 3.1 - Peak cylinder pressure versus RPM, showing original data points (1000–6000 RPM) and linear extrapolation to 9000 RPM.

3.2 Bolt Fatigue at Extended RPM

The bolt-specific inertia force, incorporating both the reciprocating component and the rotational rod whip term, increased by a factor of 2.25 between 6000 and 9000 RPM, consistent with the theoretical ω^2 relationship. This quadratic growth is the dominant factor in bolt FSF degradation, as the stress range scales directly with the additional dynamic force transmitted through the stiffness-based load-sharing model. At 9000 RPM, the bolt FSF was calculated as 1.593, representing a 57.1% reduction from the 6000 RPM baseline of 3.715. While this remains above unity, indicating the bolt would theoretically retain infinite fatigue life, the substantially reduced safety margin raises concerns regarding operational reliability under sustained high-speed conditions. Manufacturing variability, thermal softening, and transient over-speed events could erode the remaining margin, potentially reducing the FSF below unity in service (Liu *et al.*, 2025). The FSF versus RPM trend exhibited a nonlinear decline, with the rate of degradation accelerating above 7000 RPM as the inertia force component increasingly dominated the gas force component.

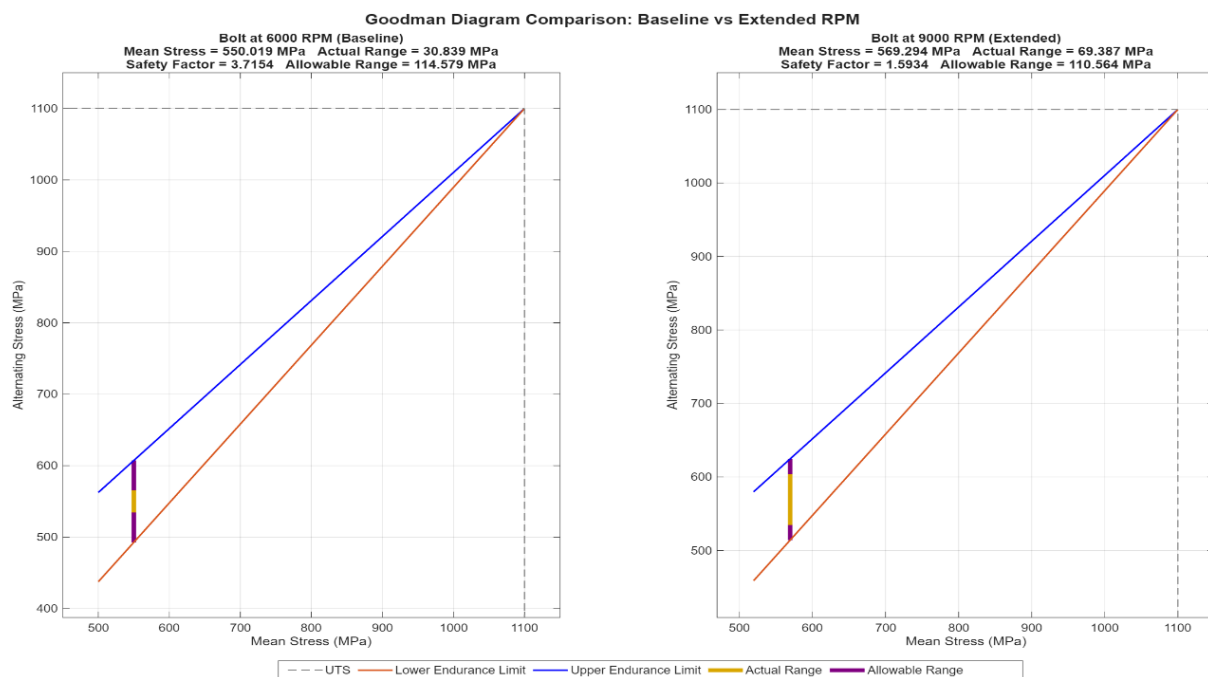


Figure 3.2 - Goodman diagram comparison for the connecting rod bolt at baseline (6000 RPM) versus extended (9000 RPM) conditions.

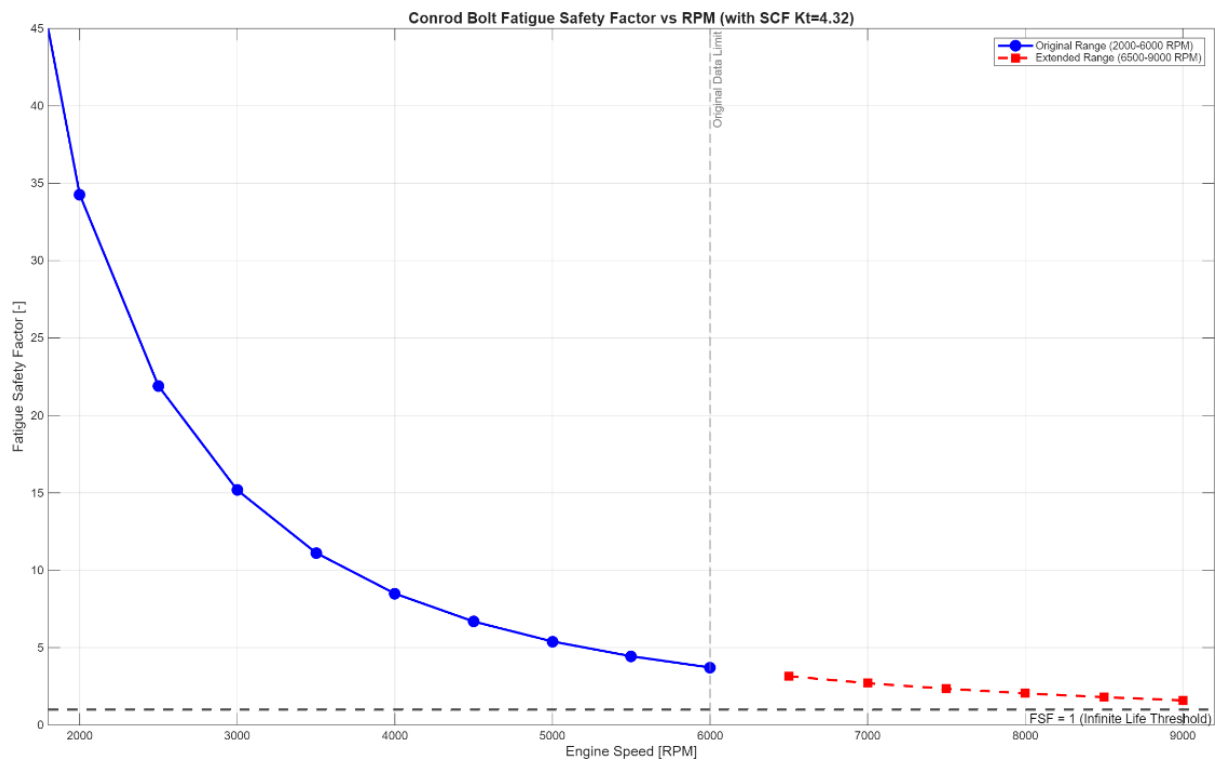


Figure 3.3 - Connecting rod bolt fatigue safety factor versus RPM across the full 1000–9000 RPM range, with the infinite life threshold ($FSF = 1$) indicated.

4. Bolt Redesign

4.1 Design Approach

The reduction in bolt FSF from 3.715 at 6000 RPM to 1.593 at 9000 RPM demonstrated the need for geometric modification to restore an adequate safety margin. The redesign strategy focused on reducing bolt stiffness by using a wasted-shank configuration, thereby lowering the proportion of external inertia load transmitted to the bolt through the stiffness-based load-sharing mechanism.

The bolt was modelled as a two-section series spring system: a reduced-diameter shank section ($L = 21.26$ mm) and the original thread engagement section ($L = 6.00$ mm, $A = 42.43$ mm²), with the combined stiffness calculated as $\frac{1}{K_{bolt}} = \frac{1}{K_{shank}} + \frac{1}{K_{thread}}$. This approach is consistent with established practice for improving fatigue resistance in bolted joints, where reducing fastener stiffness relative to the clamped members decreases the dynamic load amplitude experienced by the bolt (Shenoy and Fatemi, 2005).

4.2 Progressive Redesign

Three progressive design iterations were evaluated through a parametric sweep of wasted shank cross-sectional area, each building upon the previous:

Design A applied geometry optimisation only, reducing the shank area to decrease bolt stiffness while retaining the original thread stress concentration factor ($K_t = 4.32$). Design B also incorporated rolled-thread manufacturing, imparting approximately 100 MPa of residual compressive stress at the thread root, thereby reducing the mean stress and increasing the allowable stress range. Design C further accounted for the reduced stress concentration factor resulting from the smoother root radius produced by thread rolling (K_t reduced from 4.32 to 3.456), consistent with published findings that rolled threads significantly improve fatigue resistance compared to cut threads (Liu *et al.*, 2025). All designs were subject to a yield constraint ensuring that the peak stress in the wasted section did not exceed the material yield strength of 990 MPa.

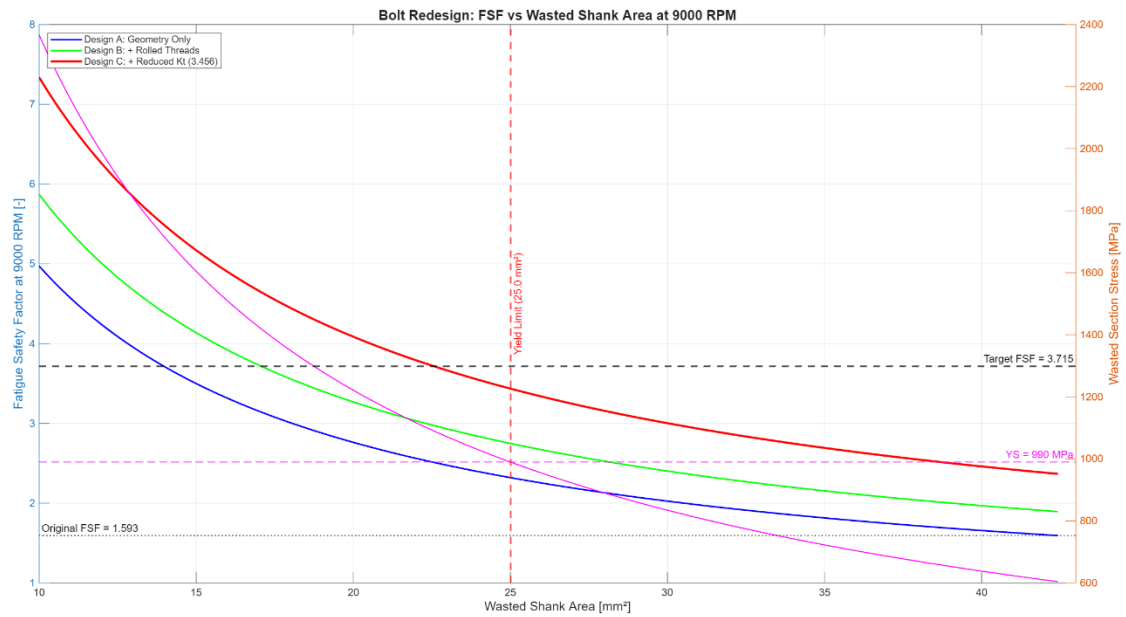


Figure 4.1 - Parametric sweep of fatigue safety factor versus wasted shank cross-sectional area at 9000 RPM, showing Design A, B, and C improvements.

4.3 Results and Yield Constraint

Design C achieved an FSF of 3.435 at 9000 RPM, representing a 115.6% improvement over the original bolt (1.593), but it fell short of the 6000 RPM baseline target of 3.715. This shortfall is attributable to a fundamental yield constraint: reducing the shank area below 25.01 mm² would cause the peak stress under preload plus dynamic loading to exceed the material yield strength. This represents a physical limit of the OEM bolt material at 9000 RPM, not a design deficiency. The wasted shank area cannot be reduced further without risking plastic deformation and loss of preload. The bolt stiffness was reduced by 35.2% from its original value, lowering the load-sharing ratio from 21.3% to 14.9% and, correspondingly, reducing the stress range transmitted to the bolt.

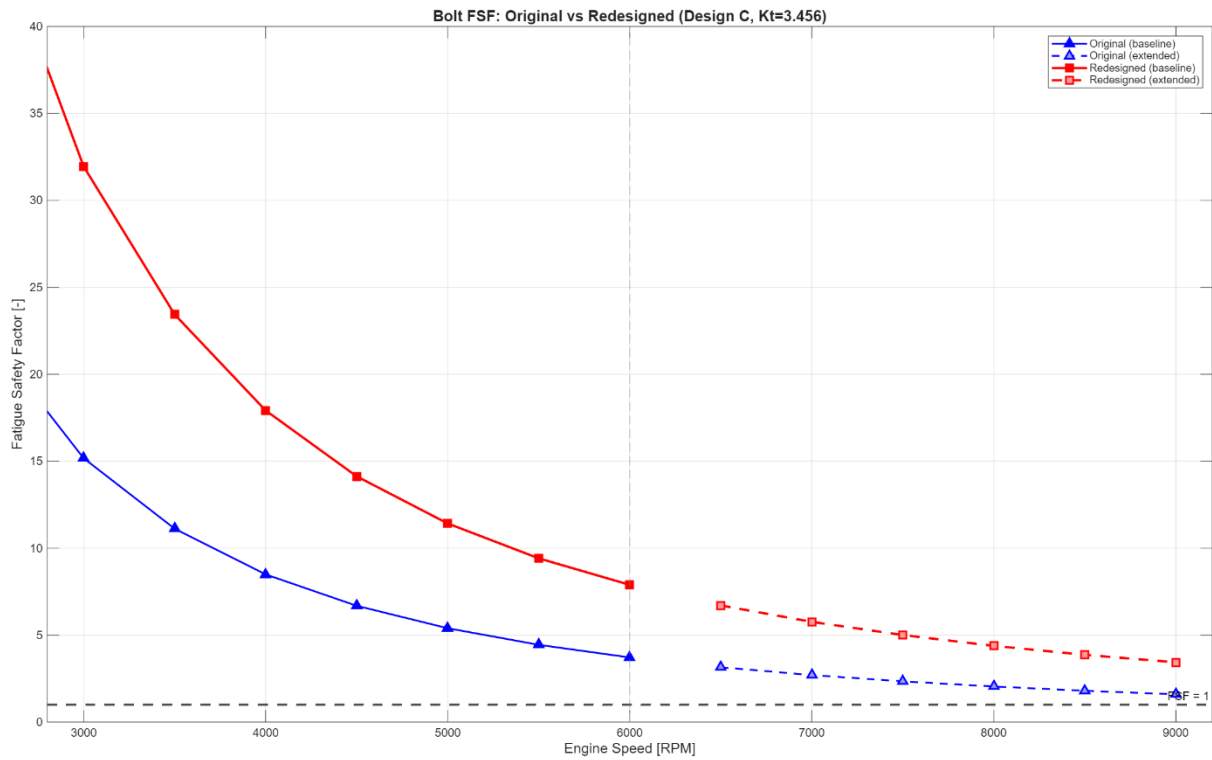


Figure 4.2 - Fatigue safety factor versus RPM comparison between the original and redesigned (Design C) connecting rod bolt.

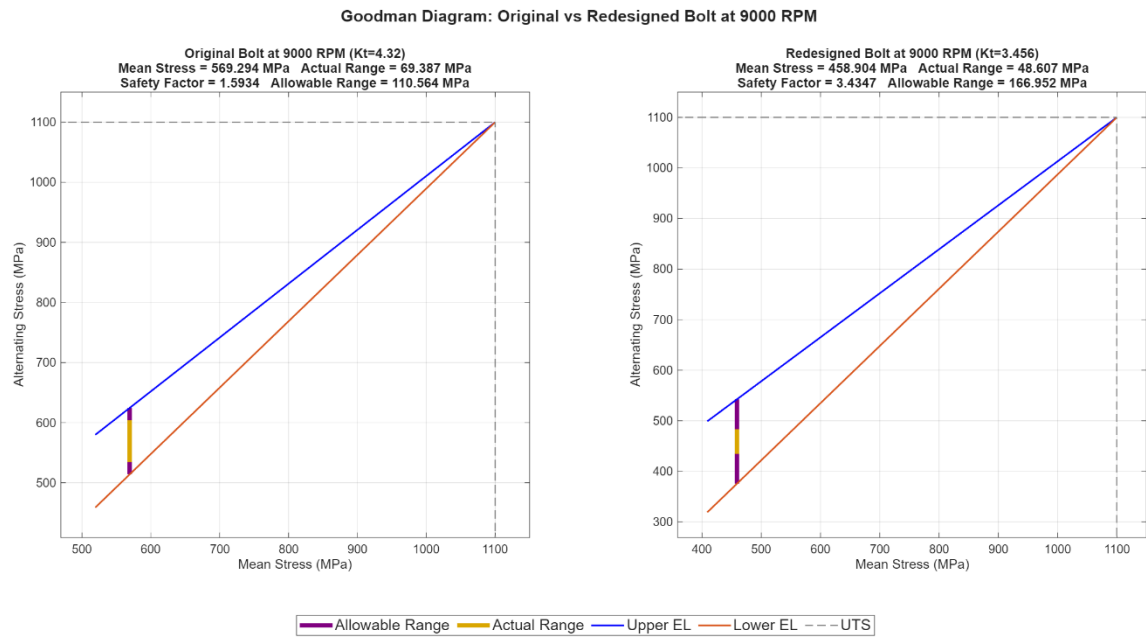


Figure 4.3 - Goodman diagram comparison between the original and redesigned bolt at 9000 RPM.

5. Crankshaft Torque Comparison

5.1 Standard and Cross-Plane Configurations

Instantaneous torque characteristics were calculated for both standard flat-plane and cross-plane crankshaft configurations using the manufacturer-provided pressure data. Gas torque was computed as $T_{gas} = F_{gas} \times \tan \phi \times R \cos \theta + F_{gas} \times R \sin \theta$, while inertia torque used the corresponding expression with reversed sign on the $\sin \theta$ term, consistent with the opposing nature of inertia-induced crank moment. Single-cylinder torques were offset and summed across four cylinders using the standard firing order 1-3-4-2 with evenly spaced 180° gas intervals, and the cross-plane order 1-3-2-4 with uneven intervals of 270° , 180° , and 90° from cylinder one.

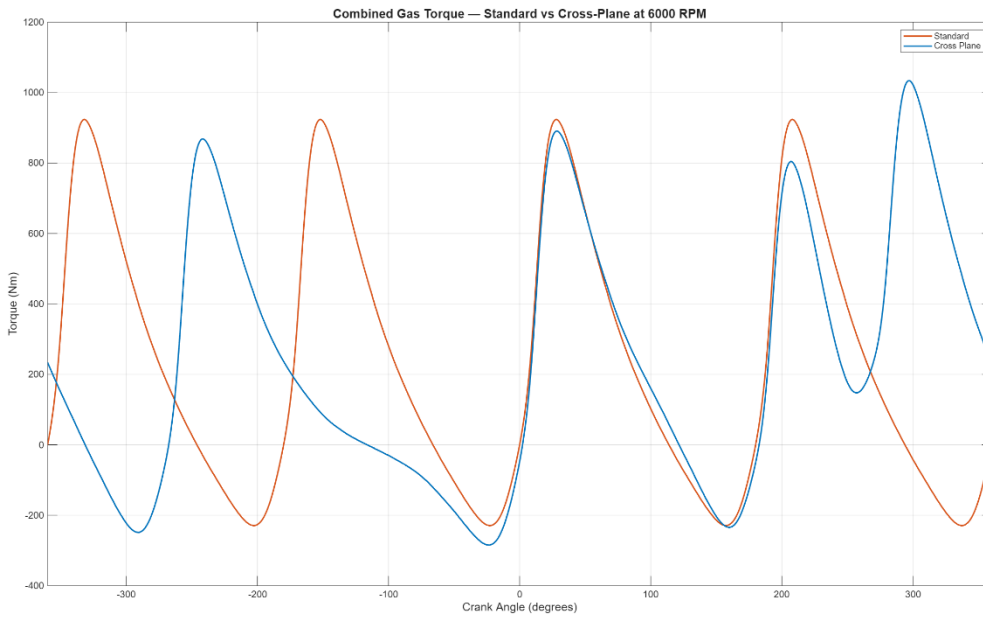


Figure 5.1 - Combined gas torque for standard and cross-plane crankshaft configurations at 6000 RPM.

At 6000 RPM, the standard configuration produced a combined crank torque range of 602.0 Nm compared to 1360.6 Nm for the cross-plane, a 126% increase in torque fluctuation, confirming the standard configuration's suitability for the Ford EcoBoost 1.6 GDTi operating envelope. This advantage arises because at moderate engine speeds, gas torque dominates the combined output, and the evenly spaced firing events of the standard crank distribute combustion pulses uniformly across the cycle. The cross-plane configuration, by contrast, exhibited a higher gas torque range due to its uneven firing intervals, which create periods of closely spaced and widely spaced combustion events.

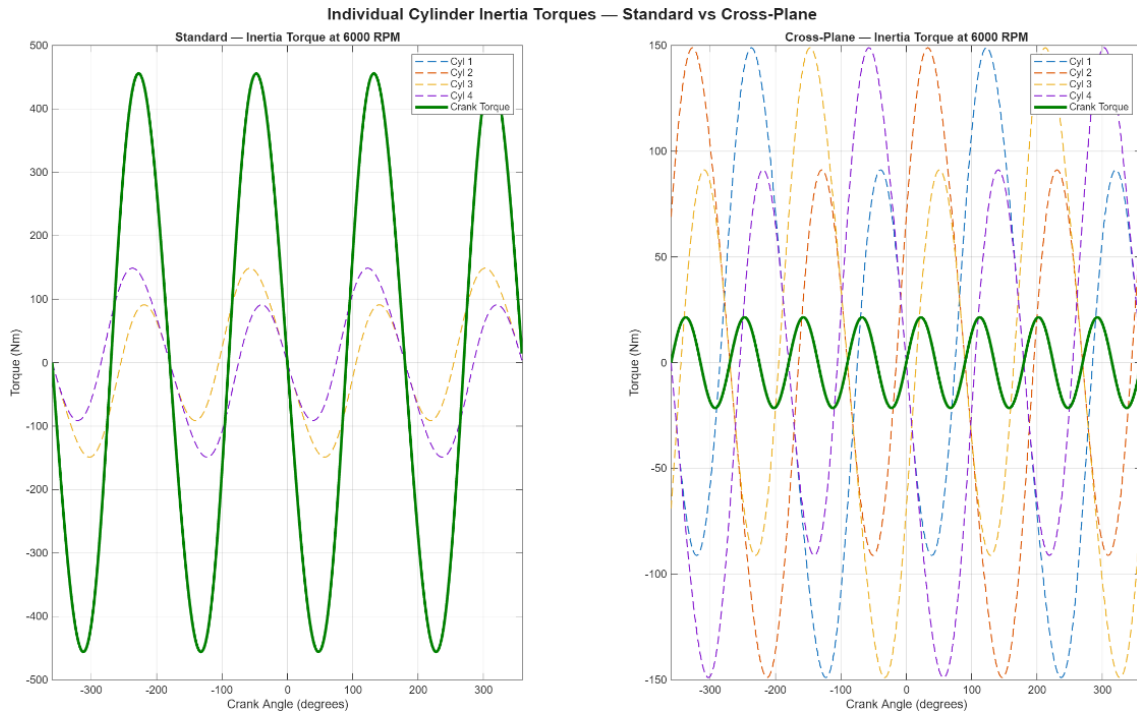


Figure 5.2 - Individual cylinder inertia torque contributions for standard and cross-plane configurations at 6000 RPM.

5.2 Application Suitability

The cross-plane configuration demonstrated a substantially lower inertia torque range of 42.9 Nm compared to 911.7 Nm for the standard crank, a 21.3× reduction. The 90° pin offset spacing causes the four cylinders to partially cancel each other's inertia contributions, whereas the standard crank's paired pistons (cylinders 1&4 together, 2&3 together) reinforce second-order inertia effects. This inertia advantage becomes increasingly significant at higher engine speeds, where inertia forces grow with ω^2 , and it explains why cross-plane inline-four configurations are employed in high-revving motorsport applications such as the Yamaha YZF-R1, which revs to 14,000+ RPM. Mean torque was identical between configurations at every analysed RPM, confirming energy conservation. It should be noted that this analysis assumes rigid crankshaft behaviour and does not capture torsional vibration effects, which become significant at high RPM and would further differentiate the two configurations. For the Ford EcoBoost 1.6 GDTi, operating at 1000–6000 RPM, where gas forces dominate, the standard configuration remains optimal, as it minimises torque fluctuations and NVH concerns for a production vehicle.

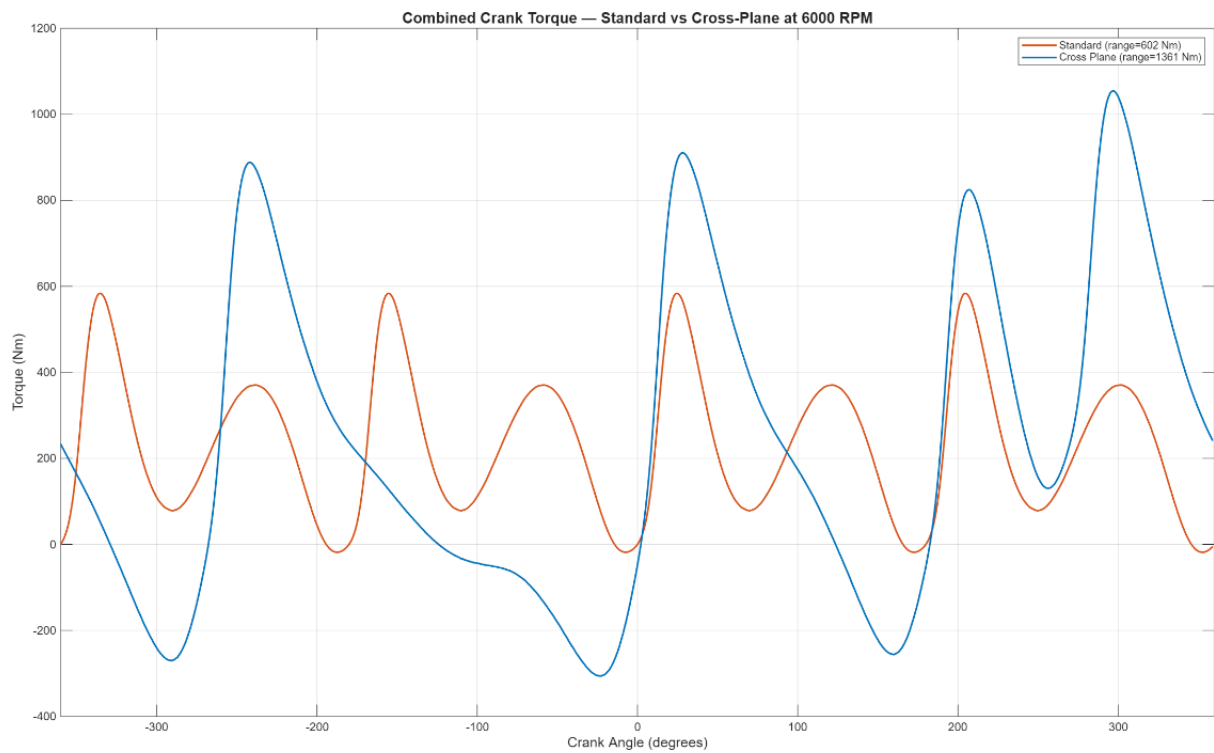


Figure 5.3 - Combined crank torque for standard and cross-plane configurations at 6000 RPM.

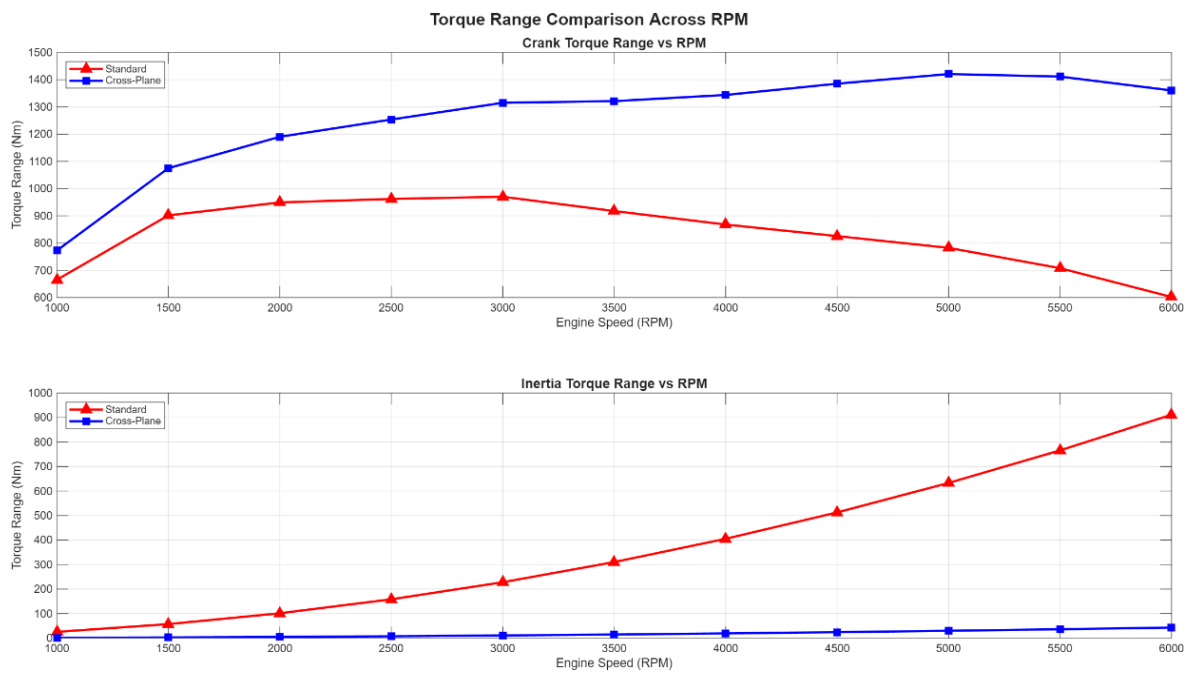


Figure 5.4 - Torque range comparison across the full RPM range for standard and cross-plane configurations.

6. Alternative Bolt Materials

6.1 Material Comparison

Four bolt materials were evaluated under identical loading and redesigned geometry (Design C) conditions at 9000 RPM: the OEM steel (UTS = 1100 MPa, E = 200 GPa), ARP2000 (UTS = 1517 MPa, E = 200 GPa), Ti-6Al-4V (UTS = 1020 MPa, E = 114 GPa), and Inconel 718 (UTS = 1450 MPa, E = 205 GPa). Material properties were sourced from manufacturer datasheets and published characterisation data (Carpenter Technology Corporation, 2000; Special Metals Corporation, 2007; Automotive Racing Products, 2026). Each material's preload was recalculated using the full derivation chain ($\sigma_{\text{preload}} = 0.6 \times 0.9 \times \sigma_{\text{Yield}}$) to ensure physically consistent bolt tension across alloys.

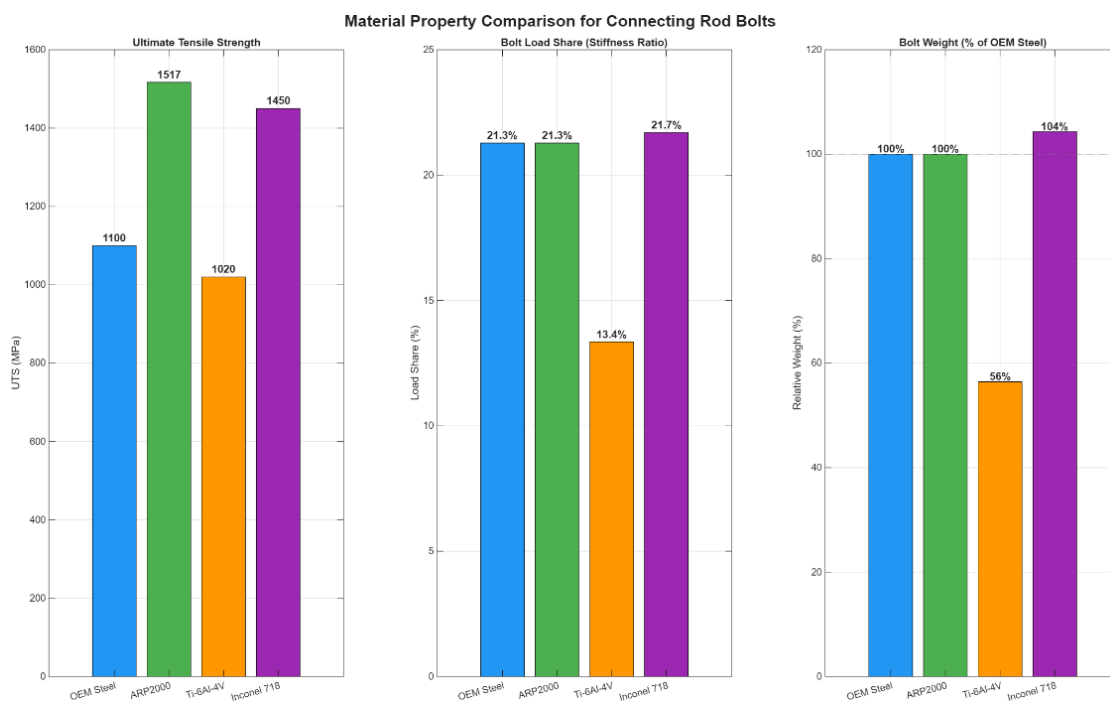


Figure 6.1 - Material property comparison for the four connecting rod bolt materials evaluated.

6.2 Key Findings

Ti-6Al-4V had the highest FSF among all materials despite having the lowest UTS. This counterintuitive result is explained by its significantly lower elastic modulus (114 GPa versus 200–205 GPa for the steel alternatives), which reduces bolt stiffness by 43% and consequently lowers the proportion of dynamic inertia load carried by the bolt through the stiffness-based load-sharing mechanism. The reduced extra force more than compensates for the lower material strength, demonstrating that in stiffness-governed bolted joints, elastic modulus can be a more significant design parameter than ultimate tensile strength. Under the redesigned geometry, Ti-6Al-4V achieved the highest FSF of 5.796, followed

by Inconel 718 at 4.592, ARP2000 at 4.583, and OEM steel at 3.435. This finding is consistent with published observations that titanium alloys achieved the highest safety factors in connecting rod applications despite lower absolute strength (Xu et al., 2024). Inconel 718 and ARP2000 both outperformed the OEM steel due to their substantially higher UTS and endurance limits, though neither matched the Ti-6Al-4V FSF because their stiffness values remained comparable to steel. From a practical standpoint, Ti-6Al-4V is costly and presents manufacturing challenges. Titanium fasteners require specialist tooling, controlled-atmosphere processing, and are susceptible to galling without appropriate surface treatments (Bettaieb, Lenain and Habraken, 2012). ARP2000, therefore, represents the most practical performance upgrade, offering meaningful FSF improvement over the OEM bolt with direct dimensional compatibility and established motorsport pedigree.

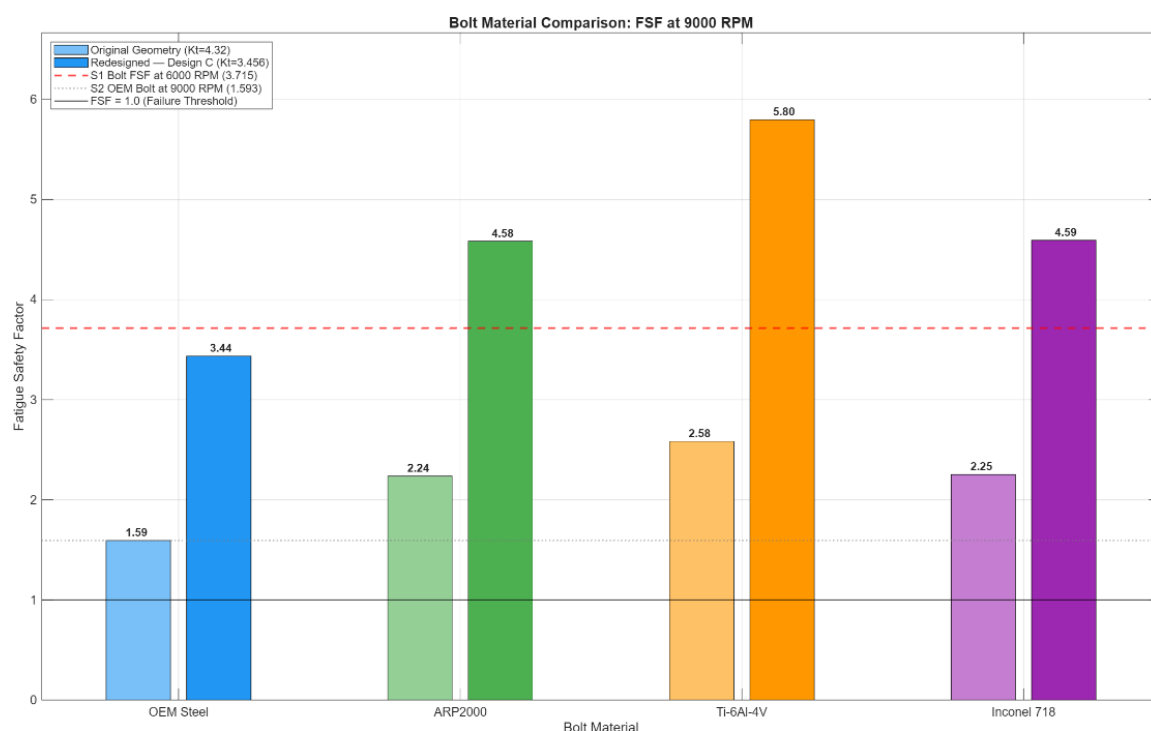


Figure 6.2 - Fatigue safety factor comparison at 9000 RPM for OEM steel, ARP2000, Ti-6Al-4V, and Inconel 718 under redesigned geometry.

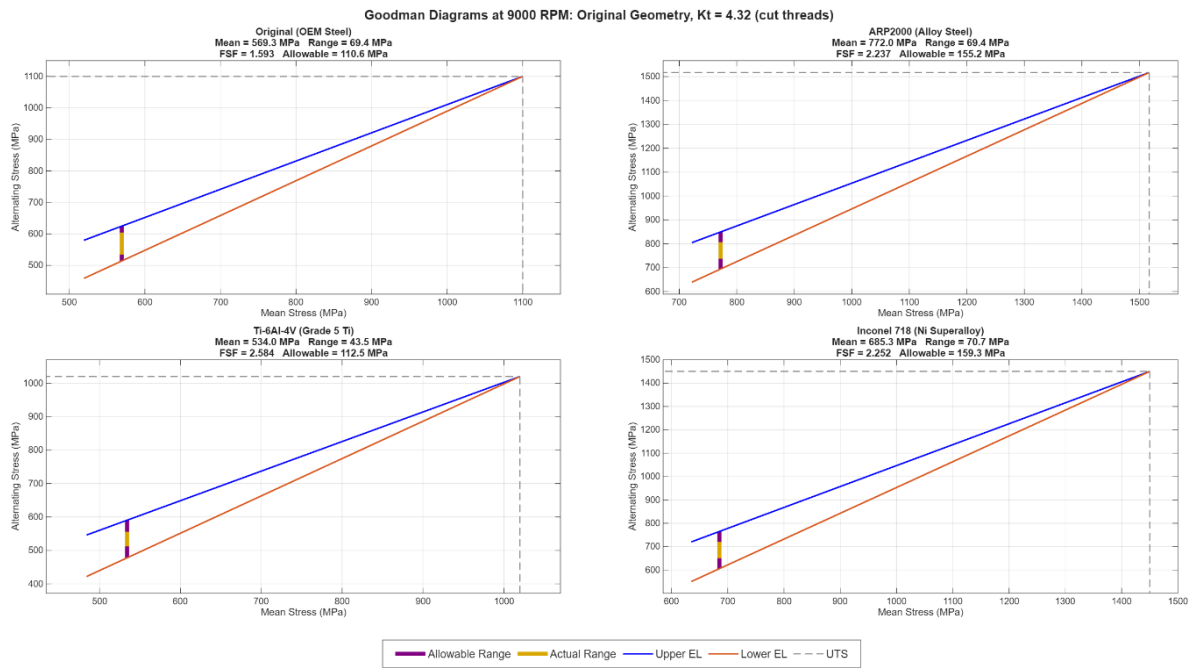


Figure 6.3 - Goodman diagrams at 9000 RPM for all four bolt materials under original bolt geometry.

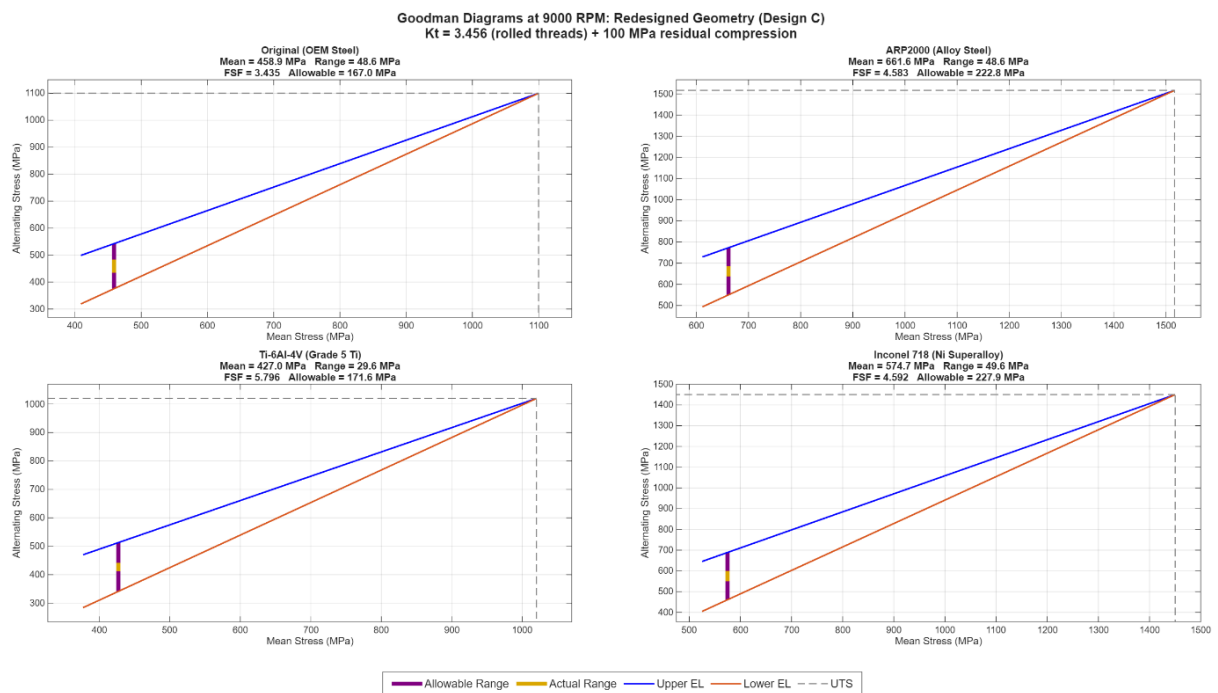


Figure 6.4 - Goodman diagrams at 9000 RPM for all four bolt materials under redesigned (Design C) geometry.

7. Conclusions

The analytical assessment of the Ford EcoBoost 1.6 GDTi bottom-end components yielded the following conclusions against the aims stated in the Introduction:

At the standard operating limit of 6000 RPM, fatigue safety factors were calculated as 4.160 for the connecting rod midpoint, 3.795 for the gudgeon pin, and 3.715 for the connecting rod bolt. All three components demonstrated infinite fatigue life with substantial design margins, indicating a degree of over-engineering consistent with production-engine requirements for manufacturing tolerances, thermal variation, and transient loading. The bolt was confirmed as the critical component with the lowest FSF of the three, attributable to the combined effects of thread stress concentration ($K_t = 4.32$) and the additional rod whip inertia loading.

Extension of the operating range to 9000 RPM, based on linear extrapolation of pressure data, revealed a 57.1% reduction in bolt FSF to 1.593, driven by the quadratic relationship between angular velocity and inertia force. While this value remains above unity, the substantially reduced margin raises significant concerns for sustained high-speed operation.

A wasted shank bolt redesign (Design C), incorporating optimised geometry, rolled threads, and reduced stress concentration, restored the FSF to 3.435 at 9000 RPM, a 115.6% improvement. Full recovery to the 6000 RPM baseline of 3.715 was prevented by a fundamental yield constraint of the OEM material.

Investigation of alternative bolt materials demonstrated that Ti-6Al-4V achieved the highest FSF despite having the lowest UTS, attributable to its lower elastic modulus, which reduced bolt stiffness and dynamic load share. ARP2000 was identified as the most practical performance upgrade, offering improved FSF with direct dimensional compatibility.

The standard crankshaft configuration was confirmed as optimal for the Ford EcoBoost 1.6 GDTi operating envelope, producing a lower combined torque range than the cross-plane alternative at all analysed speeds up to 6000 RPM. The mean torque was identical across configurations at 229.8 Nm, confirming energy conservation. For applications requiring sustained operation above 7000 RPM, ARP2000 with the wasted-shank geometry (Design C) is recommended, as it addresses both the stiffness-driven load-sharing limitation identified in the redesign analysis and the material-yield constraint that prevented full FSF recovery with OEM steel, while maintaining direct dimensional compatibility.

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