

University of Wales Trinity Saint David

Institute of Science



**Prifysgol Cymru**  
Y Drindod Dewi Sant  
**University of Wales**  
Trinity Saint David

# Contemporary Powertrain Design

ACHA6006

James Edward Vine

2106621

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## 1 Introduction

The internal components of an engine undergo complex cyclic loading, affecting durability and performance. This report analyses key components of the Ford 1.6 Sigma engine, focusing on the connecting rod (Conrod) bolts, midpoint of the connecting rod, and gudgeon pin. Using MATLAB, cylinder pressure data from 1000 to 6000 RPM is processed, with extrapolated values up to 9000 RPM to evaluate fatigue safety factors (FSF).

To address increased stresses at higher RPMs, the report explores optimisations in material selection, geometry, and thread design for the connecting rod bolts. Inconel 718 and titanium are compared to the original steel bolts for strength-to-weight and fatigue resistance. Additionally, geometry modifications, including a wasted shank design, and the implementation of rolled threads are examined to mitigate stress concentrations and enhance bolt longevity.

The study also investigates crankshaft configurations, comparing a conventional flat-plane inline-four with a crossplane design. By analysing torque contributions from gas and inertia forces, the effects on power delivery, vibration, and overall engine performance are assessed.

This report integrates theoretical analysis with empirical calculations to determine optimal design choices for high-RPM durability while maintaining reliability. The findings provide insight into best practices for internal combustion engine component design and highlight areas for further development.

## 2 Kinematic and Force Analysis

This section evaluates critical components of the Ford 1.6 Sigma engine, including the midpoint of the connecting rod, the gudgeon pin at point A, and the connecting rod bolts. Using pressure data and kinematic equations, forces acting on these components are computed, followed by stress analysis and FSF evaluations to assess durability and potential overengineering.

Engine pressure data from 1000 to 6000 RPM was processed in MATLAB, with results forming the basis for assessing mechanical loads and structural integrity.

### 2.1 Calculating Forces

This section derives piston motion and force equations to calculate inertia and gas forces, forming the basis for net force analysis throughout the engine cycle.

#### 2.1.1 Piston Motion Equations

##### 2.1.1.1 Piston Displacement (S)

The equation for S calculates the piston's position as a function of crank angle, accounting for crank radius (R) and rod length (L). This forms the basis for velocity and acceleration calculations, which are critical for determining inertia forces acting on the connecting rod and bolts.

$$\phi = \sin^{-1} \frac{R \sin \theta}{L}$$

Equation 1 – Connecting Rod Angle (Harrison ,2025)

$$S = R + L - (R \cos \theta + L \cos \phi)$$

Equation 2 – Piston Displacement (Harrison, 2025)

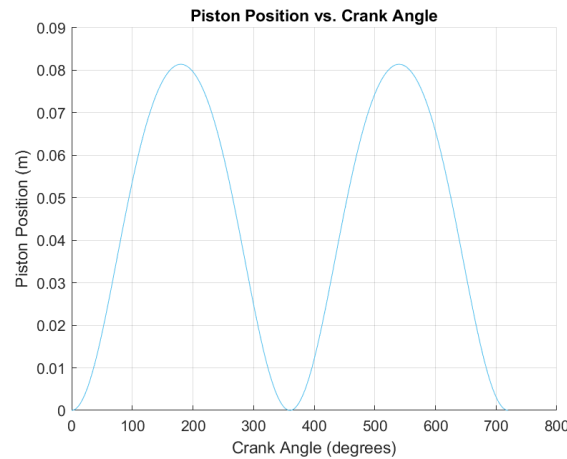


Figure 1 - Piston Position Against Crank Angle (Vine,2025)

##### 2.1.1.2 Velocity

$$\frac{dS}{dt} = R\omega \sin \theta + \left( \frac{R^2 \omega}{2L} \sin 2\theta \right)$$

Equation 3 - Piston Velocity (Harrison, 2025)

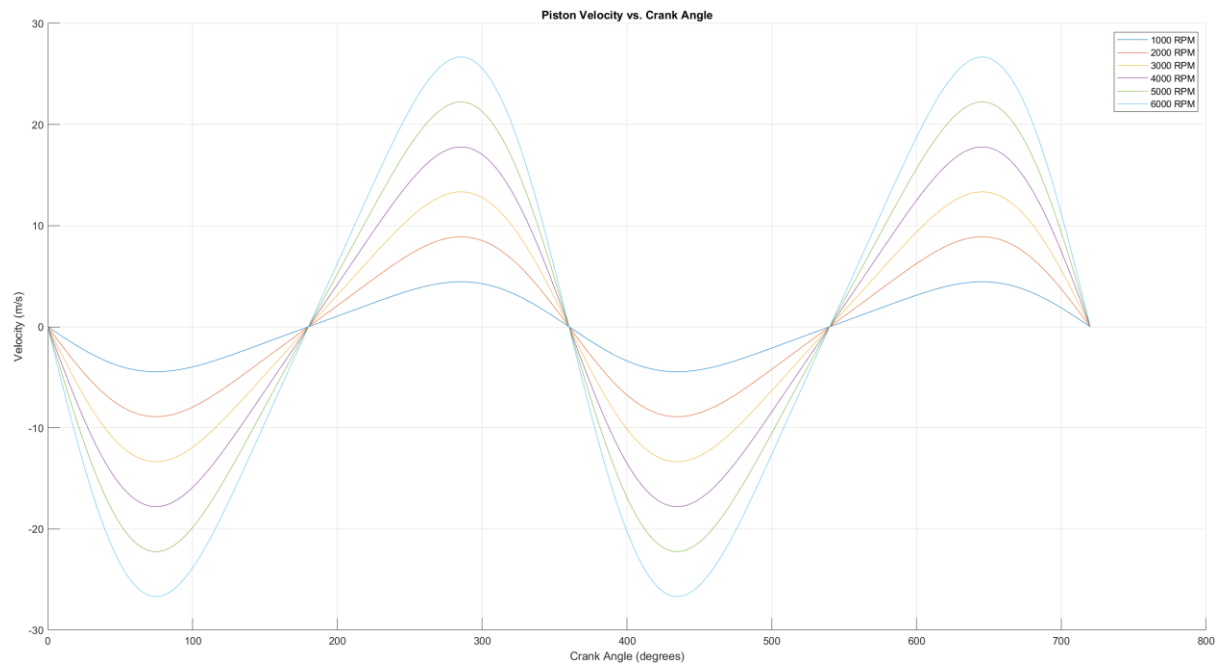


Figure 2 - Piston Velocity Against Crank Angle (Vine, 2025)

The velocity of the piston is found by differentiating the displacement equation with respect to time. This accounts for the change in piston position as the crank rotates.

#### 2.1.1.3 Acceleration

$$\frac{d^2S}{dt^2} = R\omega^2 \cos\theta + \frac{R^2}{L}\omega^2 \cos 2\theta$$

Equation 4 – Piston Acceleration (Harrison, 2025)

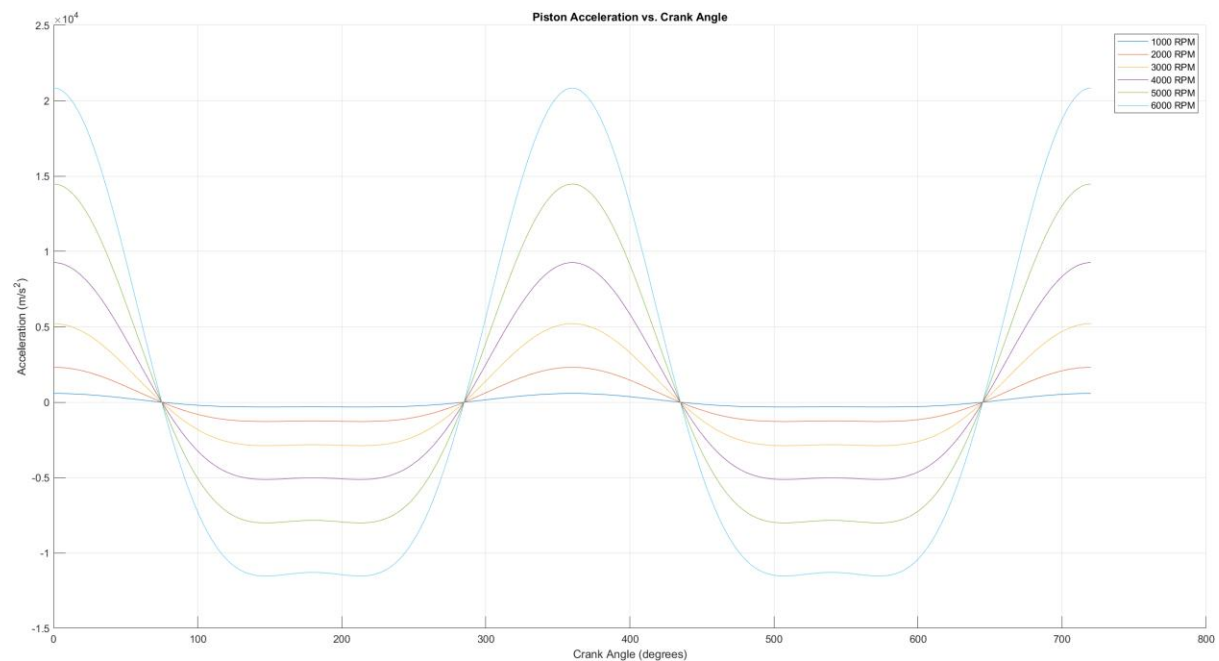


Figure 3 - Piston Acceleration Against Crank Angle (Vine, 2025)

Differentiating velocity gives piston acceleration, which is crucial for calculating inertia forces acting on the piston assembly.

### 2.1.2 Inertia Forces

As engine speed increases, the inertia forces acting on the reciprocating components grow significantly, scaling with the angular velocity. At TDC, the tensile force stretches the connecting rod, increasing stress on the bolts, which must withstand repeated cyclic loading to prevent fatigue failure. At BDC, compressive forces act on the rod, raising the risk of buckling, which can be mitigated through material selection and geometry optimisation (Xu et al., 2021).

$$F_{inertia} = m_{rec} * a$$

Equation 5 – Inertia Force (Harrison, 2025)

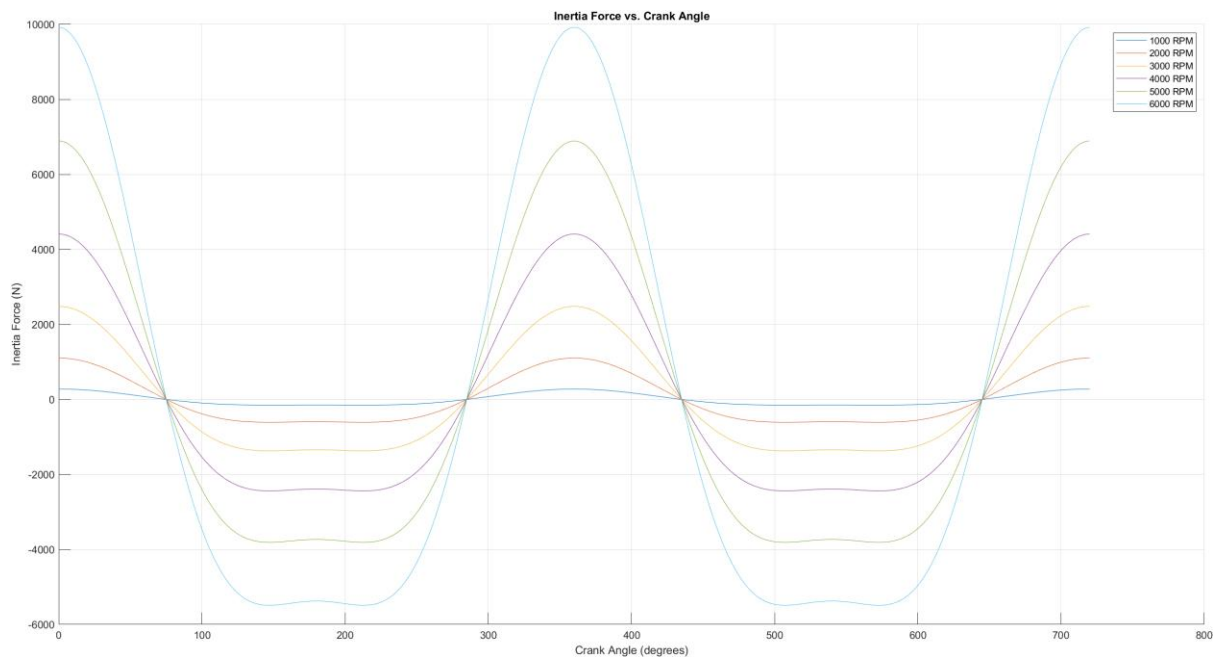


Figure 4 - Inertia Force Against Crank Angle (Vine, 2025)

### 2.1.3 Gas Force

Gas forces fluctuate throughout the engine cycle, peaking after combustion and reducing as the piston moves downward. These forces drive the connecting rod in compression, opposing inertia forces during the expansion stroke. As RPM increases, the time available for combustion shortens, potentially affecting pressure profiles and peak force values (EPI Inc., 2021).

$$F_{gas} = P_{cyl} * A$$

Equation 6 – Gas Force (Harrison, 2025)

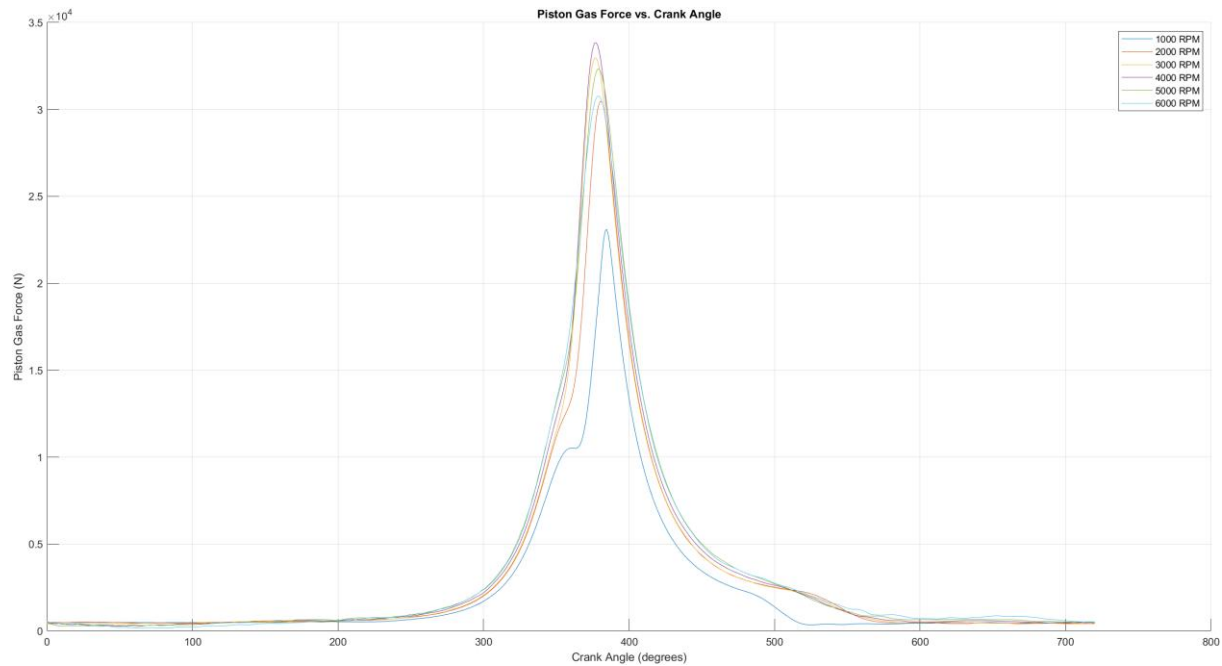


Figure 5 - Piston Gas Force Against Crank Angle (Vine, 2025)

#### 2.1.4 Net Force

The net force can be divided into compressive and tensile components, as shown in the *figure 6* and *7*. Compressive Forces occur during the power stroke, where the high gas pressure dominates, pushing the piston down and generating compressive loads in the connecting rod and gudgeon pin. Tensile Forces occur during the exhaust and intake strokes, particularly near TDC, where inertia forces overcome gas forces, pulling on the connecting rod and bolts. This is critical because tensile loading is what drives fatigue failures in the bolts (Schreier, 2003).

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

Equation 7 – Net Force (Harrison, 2025)

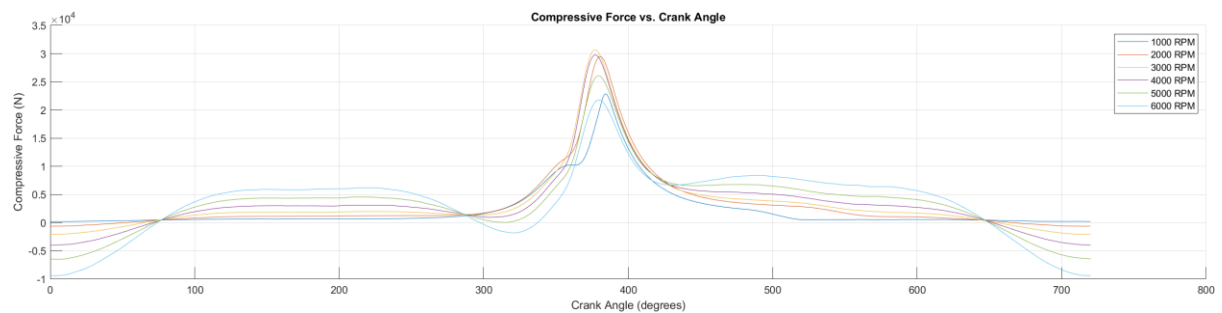


Figure 6 - Compressive Force Against Crank Angle (Vine, 2025)

By switching the arrangement of the  $F_{net}$  equation we can find the tensile force directly.

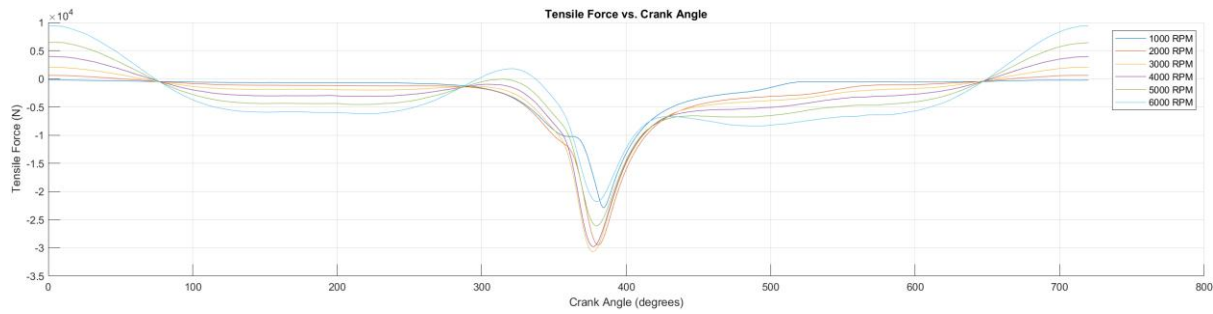


Figure 7 - Tensile Force Against Crank (Vine, 2025)

## 2.2 Stress Calculations for Components

### 2.2.1 Midpoint of the Connecting Rod (Conrod)

The midpoint of the conrod experiences alternating tensile and compressive stresses due to the cyclic loading from combustion forces and inertia effects. The stress at this location is calculated based on the net force acting on the rod and the cross-sectional area.

The Goodman diagram for this point is shown in *figure 8*. The key findings include, and FSF of 2.59, indicating the component operates well within safe limits. A minimum stress of -221.42 MPa, Max stress of 66.09 MPa, showing that the rod experiences predominantly compressive loading. The real stress range (yellow) is well within the allowable range (blue), confirming that the rod is not at risk of fatigue failure under current conditions.

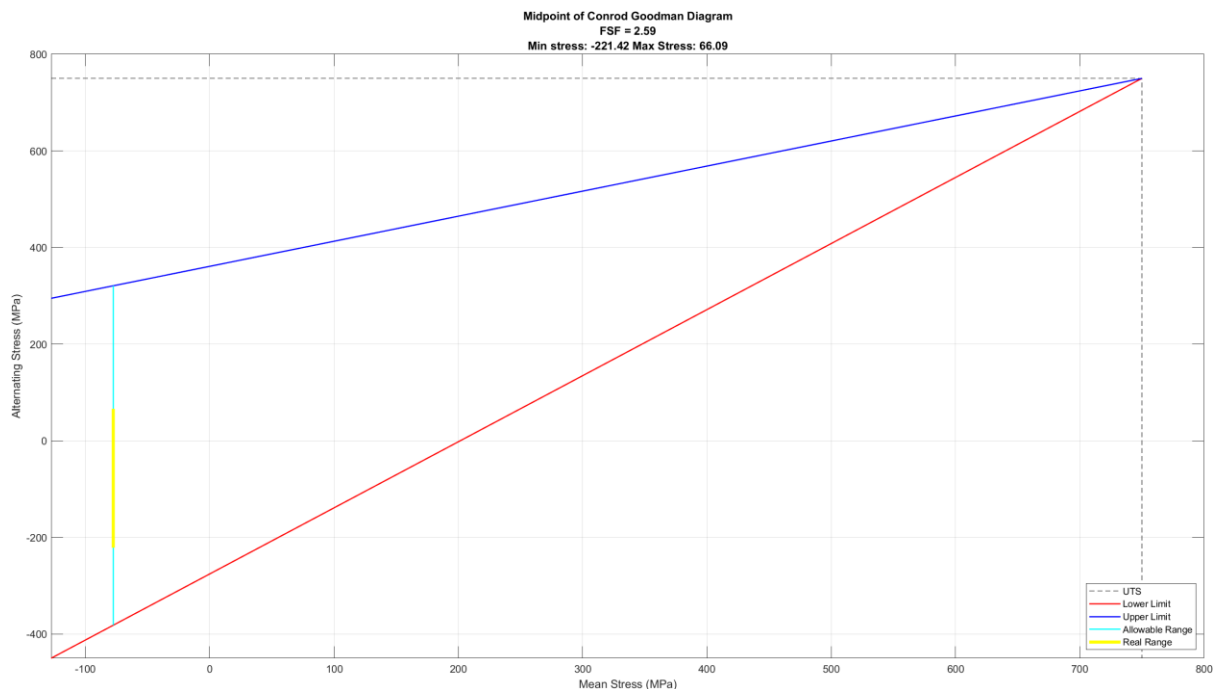


Figure 8 - Midpoint of Conrod Goodman Diagram (Vine, 2025)

### 2.2.2 Surface Pressure

Looking at *figure 9* and *10*, an initial estimate of -100 MPa (Harrison, 2025), was initially used to represent surface pressure corrections. However, research suggests that actual compressive residual stresses in peened steel connecting rods can range from 200 MPa to 400 MPa (Rautio et al., 2022). These compressive stresses reduce the mean stress experienced by the component, effectively increasing the allowable stress range and improving the FSF. To better reflect industry practice, a

revised surface pressure correction of -200 MPa was also considered, showing an increase in FSF by shifting the stress state further into compression, thereby delaying fatigue failure.

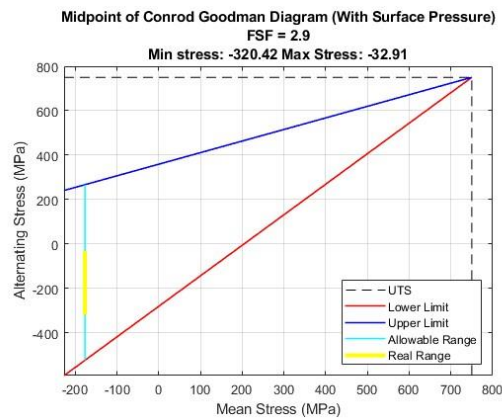


Figure 9 -100MPa, Comparing Surface Pressure Effect on the Conrod (Vine, 2025)

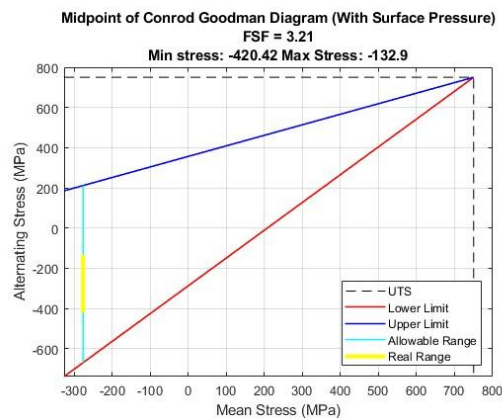


Figure 10 - 200MPa Surface Pressure Effect on the Conrod (Vine, 2025)

### 2.2.3 Gudgeon Pin at Point A

The gudgeon pin plays a critical role in transmitting forces between the piston and conrod, experiencing bending and shear stresses due to combustion and inertia loads, making fatigue analysis essential. Figure 11 an FSF value of 2.68, indicating a reasonable margin against fatigue failure. A minimum stress of -320.93 MPa and a max stress of 59.25 MPa, showing a predominantly compressive stress state. The real stress range, is well within the allowable range, confirming that under current conditions, fatigue failure is unlikely.

Given that the gudgeon pin is subjected to bending, its material selection and surface treatment are critical for fatigue life. Typically, gudgeon pins are made from low-carbon case-hardened steel, which undergoes carburization to increase surface hardness while retaining a tough core. The material can be refined and hardened through oil quenching, enhancing its wear resistance and fatigue strength (Kharade et al., 2019). Additional surface treatments, such as nitriding or DLC coatings, further improve durability and bending fatigue resistance by reducing surface defects and increasing load-bearing capacity (Coker Engineering, 2025). Although this report primarily focuses on the conrod bolts, it is acknowledged that as engine RPM increases, the gudgeon pin will experience greater inertia forces, which may reduce its FSF.

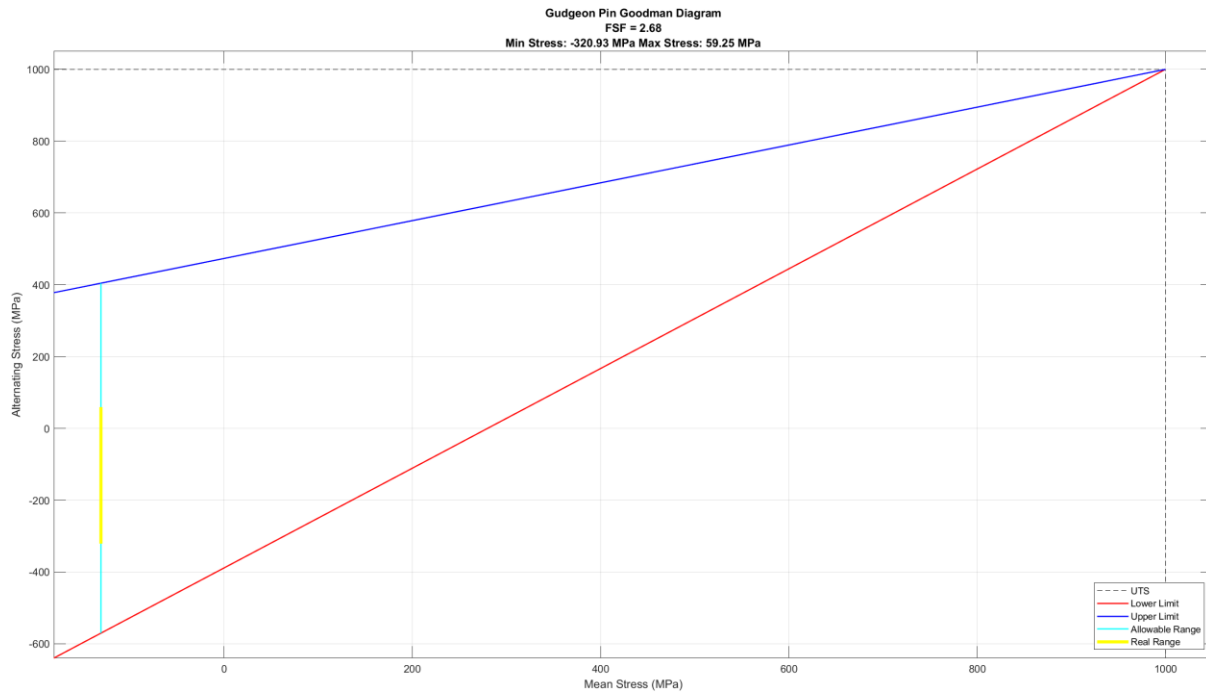


Figure 11 - Gudgeon Pin Goodman Diagram (Vine, 2025)

#### 2.2.4 Conrod Rod Bolts

The conrod bolts experience high cyclic tensile loading, particularly near TDC, where inertia forces stretch the bolts. Unlike the gudgeon pin and connecting rod, which undergo both compressive and tensile loading, the bolts are primarily subjected to tensile stress (Becker, 2017), making the FSF analysis crucial.

##### 2.2.4.1 Bolt Forces

Forces are calculated as

$$Bolt_{Inertia} = (mass_{recipitating} * acceleration_{piston}) + (mass_{rotational} * acceleration_{crank})$$

Equation 8 – Bolt Inertia (Harrison, 2025)

$$Max\_Bolt_{inertia} = \frac{\max(Bolt_{inertia})}{2}$$

Equation 9 – Maximum Bolt Inertia (Harrison, 2025)

This accounts for the total force split across both bolts.

##### 2.2.4.2 Bolt Stiffness

The stiffness of the bolt and cap system determines how loads are shared between the fastener and its surrounding structure.

$E_{bolt}$  = Module of elasticity of the bolt material

$A_{bolt}$  = Cross sectional area of the bolt

$L$  = Effective length of the bolt

The stiffness of the bolt and cap is calculated using:

$$K_{bolt} = \frac{E_{bolt} * A_{bolt}}{L}$$

Equation 10 – Bolt Stiffness (Harrison, 2025)

The same calculation is also carried out for the cap, this refers to the lower part of the connecting rod, which is clamped to the big-end bearing journal of the crankshaft by the conrod bolts. The interaction between these components is accounted for in the load-sharing equation

$$F_{extra} = \frac{F_{inertia\_bolt} * K_{bolt}}{K_{bolt} + K_{cap}}$$

Equation 11 – Combining Forces and Stiffness (Harrison, 2025)

A stiffer cap absorbs more of the load, reducing the stress amplitude on the bolt, whereas a more flexible cap transfers a higher proportion of the force to the bolt. For this report the cap will remain the same throughout.

#### 2.2.4.3 Bolt Loads

To determine how the bolt withstands the applied forces

$$Yield_{Strength} = 0.9 * Material\_UTS$$

Equation 12 – Yield Strength (Harrison, 2025)

$$ProofLoad = 0.9 * YieldStrength$$

Equation 13 - ProofLoad (Harrison, 2025)

$$S_{axial} = 0.6 * ProofLoad$$

Equation 14 – S axial (Preload) (Harrison, 2025)

Carrying over calculated values from section 2.2.4.2

$$F_{axial} = S_{axial} * A_{bolt}$$

Equation 15 – F axial (Harrison, 2025)

Therefore, the total bolt forces are

$$F_{bolt\_new} = F_{axial} + F_{extra}$$

Equation 16 – Total Bolt Force (Harrison, 2025)

#### 2.2.4.4 Stress Concentration Factor (KT)

To calculate the stress concentration factor (Kt), the M8 × 1.0 (AJU Motorsport, 2025) thread specifications were obtained from data table, table 1, The h/r ratio determines the appropriate coefficients used in the calculation.

Table 1 - Stress Concentration Data (Harrison, 2025)

| Nominal Size ISO MF | Thread Form Type | Major Diameter mm d=D | Pitch mm p | Root Radius mm r | Pitch Diameter mm d2=D2 | Minor Diameter Male Thd. d3 | Minor Diameter Female Thd. D1 | Thread Height Male Thd. h3 | Thread Height Female Thd. H1 | Tap Drill Diameter mm |
|---------------------|------------------|-----------------------|------------|------------------|-------------------------|-----------------------------|-------------------------------|----------------------------|------------------------------|-----------------------|
| 1.0x0.2             | M                | 1.00                  | 0.20       | 0.029            | 0.870                   | 0.755                       | 0.783                         | 0.123                      | 0.108                        | 0.80                  |
| 1.1x0.2             | M                | 1.10                  | 0.20       | 0.029            | 0.970                   | 0.855                       | 0.883                         | 0.123                      | 0.108                        | 0.90                  |
| 1.2x0.2             | M                | 1.20                  | 0.20       | 0.029            | 1.070                   | 0.955                       | 0.983                         | 0.123                      | 0.108                        | 1.00                  |
| 1.4x0.2             | M                | 1.40                  | 0.20       | 0.029            | 1.270                   | 1.155                       | 1.183                         | 0.123                      | 0.108                        | 1.20                  |
| 1.6x0.2             | M                | 1.60                  | 0.20       | 0.029            | 1.470                   | 1.355                       | 1.383                         | 0.123                      | 0.108                        | 1.40                  |
| 1.8x0.2             | M                | 1.80                  | 0.20       | 0.029            | 1.670                   | 1.555                       | 1.583                         | 0.123                      | 0.108                        | 1.60                  |
| 2x0.25              | M                | 2.00                  | 0.25       | 0.036            | 1.838                   | 1.693                       | 1.729                         | 0.153                      | 0.135                        | 1.75                  |
| 2.2x0.25            | M                | 2.20                  | 0.25       | 0.036            | 2.038                   | 1.893                       | 1.929                         | 0.153                      | 0.135                        | 1.95                  |
| 2.5x0.35            | M                | 2.50                  | 0.35       | 0.051            | 2.273                   | 2.071                       | 2.121                         | 0.215                      | 0.189                        | 2.10                  |
| 3x0.35              | M                | 3.00                  | 0.35       | 0.051            | 2.773                   | 2.571                       | 2.621                         | 0.215                      | 0.189                        | 2.60                  |
| 3.5x0.35            | M                | 3.50                  | 0.35       | 0.051            | 3.273                   | 3.071                       | 3.121                         | 0.215                      | 0.189                        | 3.10                  |
| 4x0.5               | M                | 4.00                  | 0.50       | 0.072            | 3.675                   | 3.387                       | 3.459                         | 0.307                      | 0.271                        | 3.50                  |
| 4.5x0.5             | M                | 4.50                  | 0.50       | 0.072            | 4.175                   | 3.887                       | 3.959                         | 0.307                      | 0.271                        | 4.00                  |
| 5x0.5               | M                | 5.00                  | 0.50       | 0.072            | 4.675                   | 4.387                       | 4.459                         | 0.307                      | 0.271                        | 4.50                  |
| 5.5x0.5             | M                | 5.50                  | 0.50       | 0.072            | 5.175                   | 4.887                       | 4.959                         | 0.307                      | 0.271                        | 5.00                  |
| 6x0.75              | M                | 6.00                  | 0.75       | 0.108            | 5.513                   | 5.080                       | 5.188                         | 0.460                      | 0.406                        | 5.20                  |
| 7x0.75              | M                | 7.00                  | 0.75       | 0.108            | 6.513                   | 6.080                       | 6.188                         | 0.460                      | 0.406                        | 6.20                  |
| 8x0.75              | M                | 8.00                  | 0.75       | 0.108            | 7.513                   | 7.080                       | 7.188                         | 0.460                      | 0.406                        | 7.20                  |
| 8x1.0               | M                | 8.00                  | 1.00       | 0.144            | 7.350                   | 6.773                       | 6.917                         | 0.613                      | 0.541                        | 7.00                  |

These values were using the stress concentration factor equation

$$K_t = C_1 + C_2 \left( \frac{2h}{d} \right) + C_3 \left( \frac{2h}{d} \right)^2 + C_4 \left( \frac{2h}{d} \right)^3$$

Equation 17 – Stress Concentration for 2 u-notches (Harrison, 2025)

Where  $C_1, C_2, C_3$  and  $C_4$  are coefficients based on the thread geometry.  $K_t$  is implemented in the allowable stress calculation when calculating the FSF.

#### 2.2.4.5 FSF

The original conrod bolt at 6000 RPM serves as the baseline for analysis, with an FSF of 3.41. This represents a safe margin for fatigue resistance, ensuring long-term reliability under moderate RPM conditions. As the experimental data rises to 9000rpm this FSF will serve as a target.

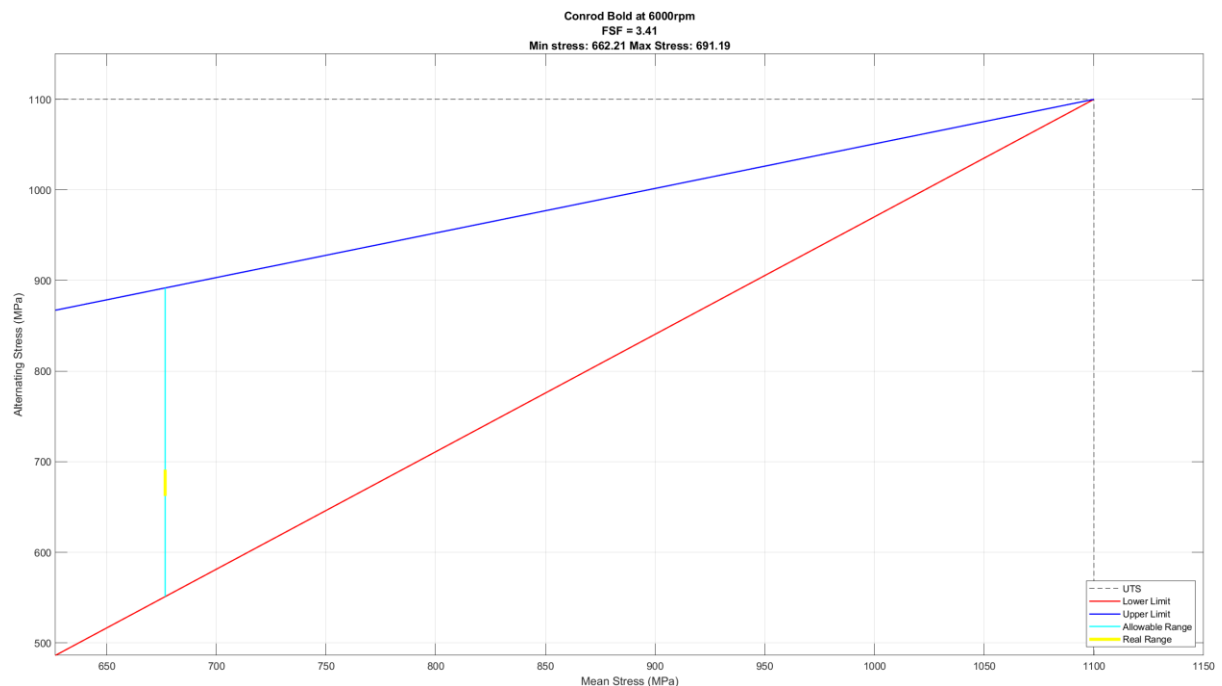


Figure 12 - Conrod Bolt at 6000 RPM (Vine, 2025)

### 3 Extending the RPM Range

This section will cover the process for raising the rpm and its effect on the conrod bolts at higher RPMs.

#### 3.1 Interpolation Process

To evaluate component stresses at higher RPMs, cylinder pressure data was interpolated up to 6000 RPM and extrapolated for 7000–9000 RPM using an exponential growth model.

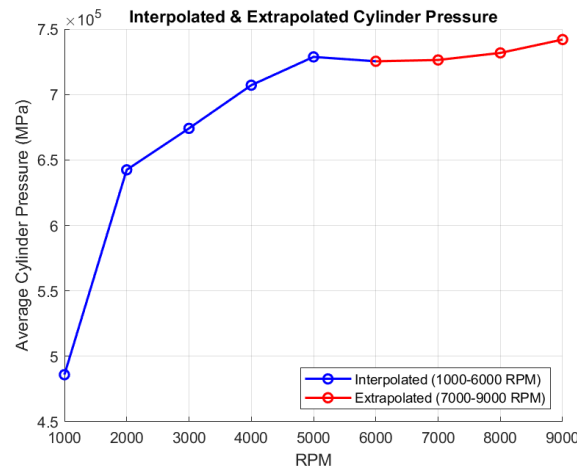


Figure 13 - Interpolated, Extrapolated Plot for Higher RPMs (Vine, 2025)

#### 3.2 Conrod Bolt FSF at 9000rpm

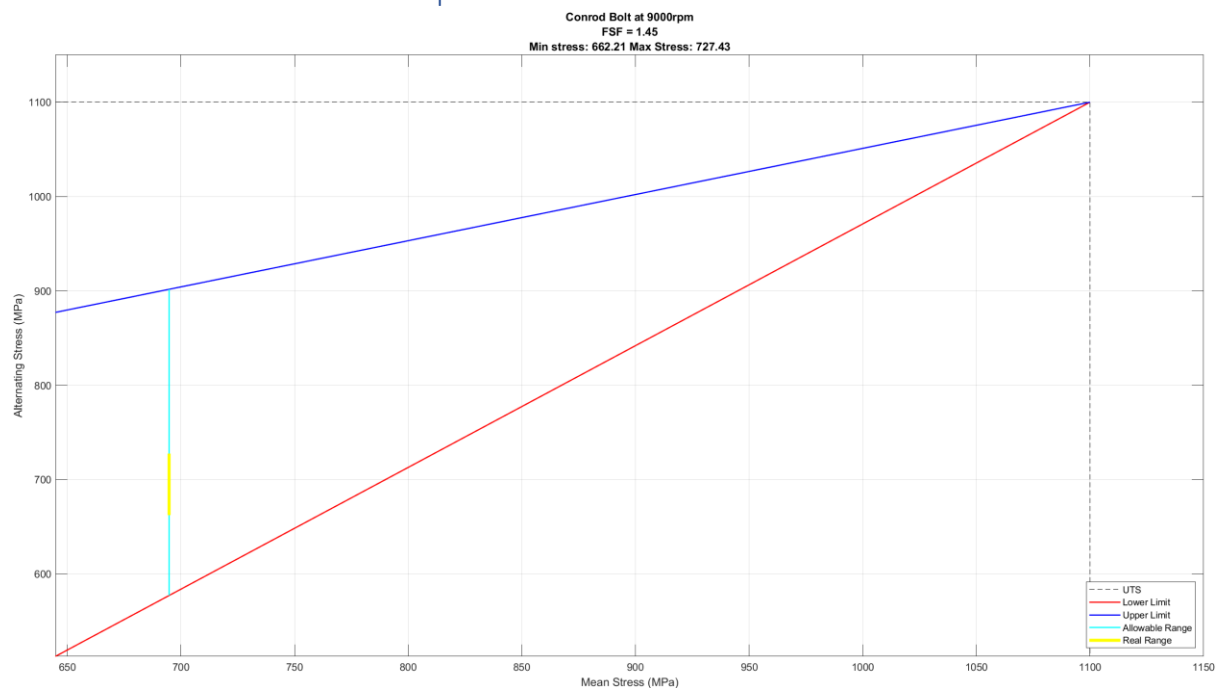


Figure 14 - Conrod Bolt at 9000 RPM (Vine, 2025)

## 4 Conrod Bolt Optimisation

This section evaluates material selection, geometry modifications, and thread design to improve strength, weight reduction, and fatigue resistance. By comparing steel, Inconel, and titanium across wasted shank and rolled thread designs, the goal is to identify the optimal bolt configuration for high-RPM durability.

### 4.1 Table of Results

Table 2 - Stresses and FSF Results (Vine, 2025)

| Material          | RPM  | UTS  | Module of Elasticity | Component Weight (kg) | Max Stress (MPa) | Minimum Stress (MPa) | FSF  |
|-------------------|------|------|----------------------|-----------------------|------------------|----------------------|------|
| Original Geometry |      |      |                      |                       |                  |                      |      |
| Steel (Original)  | 6000 | 1100 | 210000               | 0.035                 | 691.19           | 662.21               | 3.41 |
| Steel (Original)  | 9000 | 1100 | 210000               | 0.035                 | 727.43           | 662.21               | 1.45 |
| Inconel           | 9000 | 1375 | 200000               | 0.0315                | 890.43           | 827.76               | 1.92 |
| Titanium          | 9000 | 1030 | 110000               | 0.02                  | 657.55           | 620.07               | 2.43 |
| Wasted Shank      |      |      |                      |                       |                  |                      |      |
| Steel (Original)  | 9000 | 1100 | 210000               | 0.035                 | 718.15           | 621.21               | 1.71 |
| Inconel           | 9000 | 1375 | 200000               | 0.0315                | 881.44           | 827.76               | 2.26 |
| Titanium          | 9000 | 1030 | 110000               | 0.02                  | 651.78           | 620.07               | 2.9  |
| Rolled Thread     |      |      |                      |                       |                  |                      |      |
| Steel (Original)  | 9000 | 1100 | 210000               | 0.035                 | 718.15           | 621.21               | 1.88 |
| Inconel           | 9000 | 1375 | 200000               | 0.0315                | 881.44           | 827.76               | 2.49 |
| Titanium          | 9000 | 1030 | 110000               | 0.02                  | 651.78           | 620.07               | 3.19 |

### 4.2 Material Changes

Table 2 material data was acquired from Probolt, (2024) and Special Metals, (2007).

The first material is Inconel 718 is used in high-stress fasteners such as connecting rod bolts due to its superior tensile strength, fatigue resistance, and corrosion resistance, even at room temperature, outperforming steel of equivalent strength 1600 MPa (Ward, 2011).

The second material is titanium, which is increasingly utilised in high-performance automotive applications such as connecting rods, engine valves, and fasteners. Its exceptional strength-to-weight ratio and corrosion resistance make it ideal for reducing reciprocating mass and improving engine efficiency (Isaka et al., 2022).

#### 4.2.1 Inconel 718

Showing an FSF of 1.92, indicating the bolt is nearing its fatigue limit but remains within acceptable safety margins. With minimum and maximum stresses of 827.76 MPa and 890.43 MPa, the bolt experiences high cyclic loading, and the real stress range is close to the allowable limit, suggesting limited remaining fatigue life. While Inconel offers superior strength over steel, the FSF dropping below 2 raises concerns for long-term reliability.

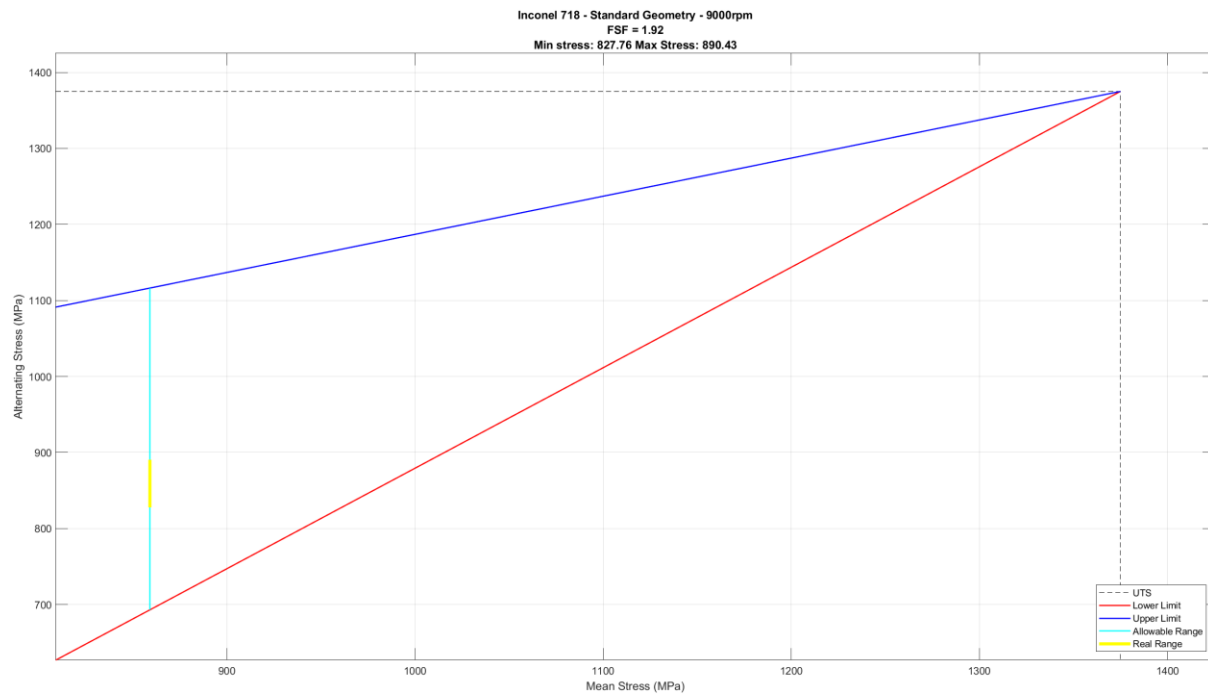


Figure 15 - Inconel 718 Goodman Diagram

#### 4.2.2 Titanium

An FSF of 2.43, indicates better fatigue performance than Inconel and steel. With minimum and maximum stresses of 620.07 MPa and 657.55 MPa, the titanium bolt experiences lower cyclic loading, attributed to its lower density and reduced mass. While titanium offers a 43% weight reduction (Lee, 2023), its lower UTS compared to Inconel results in higher strain under load, which may impact preload retention.

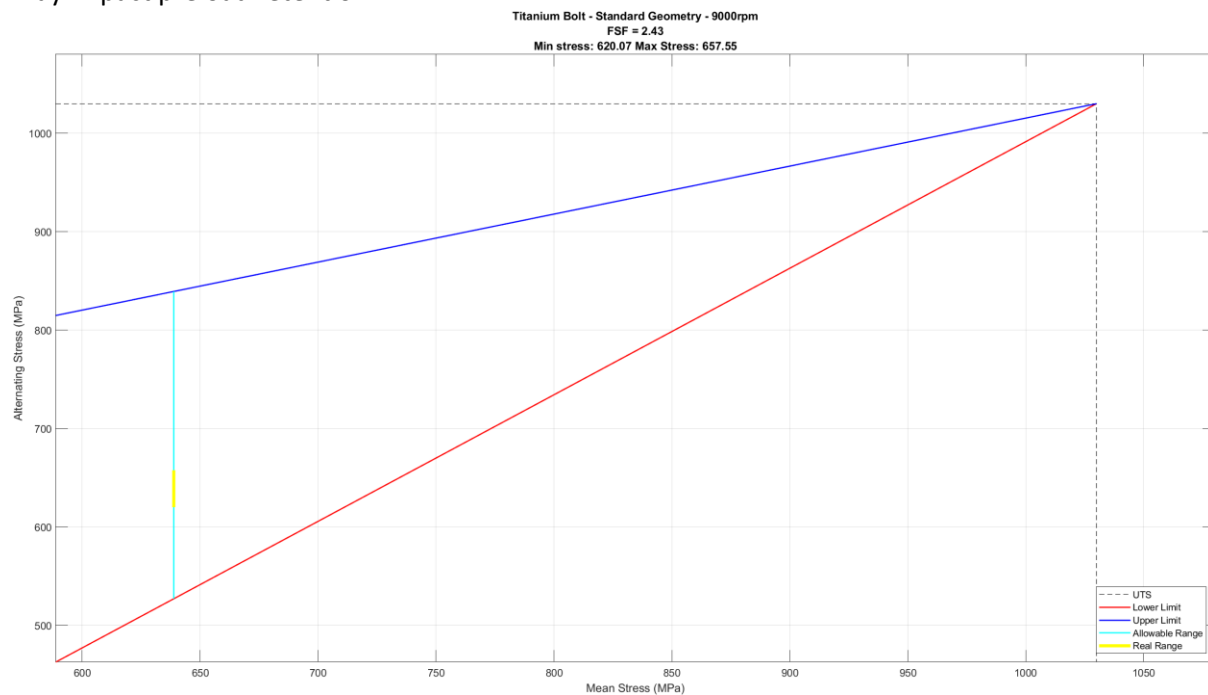


Figure 16 - Titanium Goodman Diagram

### 4.3 Geometry Changes

The wasted shank modification reduces the cross-sectional area. The bolt is divided into three sections: L1 (full-width under head), L2 (wasted shank), and L3 (threaded section). The wasted shank (L2) is 20% smaller than the previous section, lowering inertia forces and reducing cyclic stress without compromising load-bearing capacity. Despite this reduction, the threaded section (L3) retains the original cross-sectional area to ensure secure fastening. This modification helps distribute stress more efficiently, improving fatigue resistance.

```
% Section Lengths (Total must be 27.26mm)
L1 = 8;      % mm (Full-width under head)
L2 = 12;     % mm (Wasted shank)
L3 = 7.26;   % mm (Thread interacting section)
L_total = L1 + L2 + L3; % Check total length
disp(L_total);

% Cross-Sectional Areas
A1 = 0.9 * 42.43; % mm^2 (Shanked section, 90% of original)
disp(A1);
A2 = 0.8 * A1;    % mm^2 (Wasted shank, 80% of A1)
disp(A2);
A3 = 42.43;      % mm^2 (Threaded section, same as original)
disp(A3);
```

Figure 17 - New Geometry and Area Calculations in MATLAB (Vine, 2025)

#### 4.3.1 Wasted Shank - Steel (Original Bolt)

With and FSF of 1.45 in the original design, the new value of 1.71 indicates a slight improvement in fatigue resistance. The maximum stress 718.15 MPa is lower than the original 727.43 MPa, suggesting a more efficient stress distribution, but the minimum stress 662.21 MPa remains unchanged. Despite the FSF improvement, the bolt is still approaching fatigue limits, highlighting the need for further optimisation.

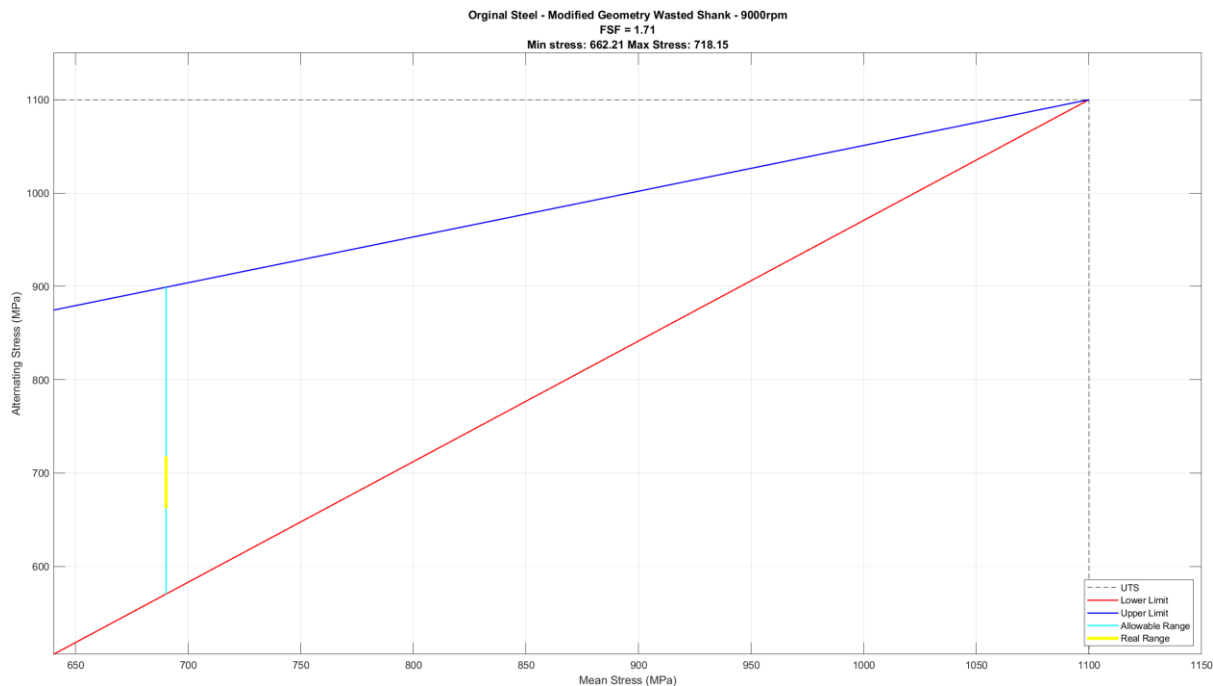


Figure 18 - Original Material and Wasted Shank (Vine, 2025)

#### 4.3.2 Wasted Shank - Inconel 718

The Goodman diagram for Inconel shows an FSF of 2.26, up from 1.92 in the original design, indicating improved fatigue resistance. The maximum stress has slightly reduced to 881.44 MPa from 890.43 MPa, suggesting a modest decrease. This indicates that the wasted shank design helps distribute stress more efficiently without compromising structural integrity. However, stress levels remain relatively high, and further refinements can be made.

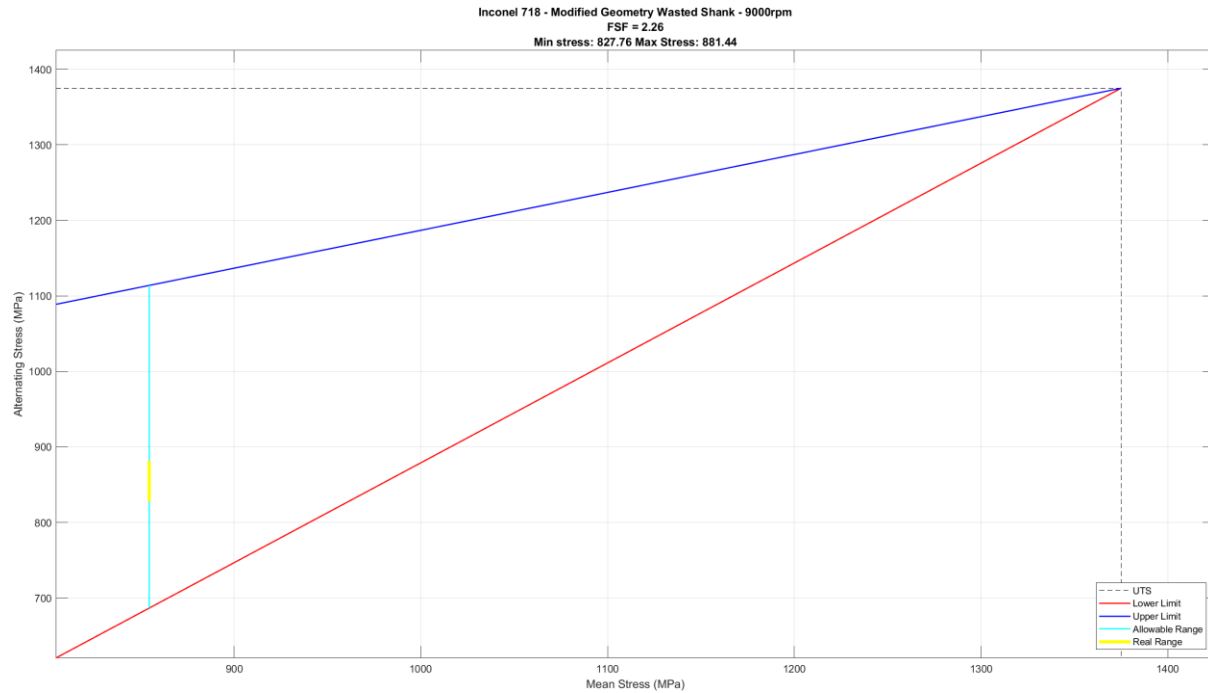


Figure 19 - Inconel 718 and Wasted Shank (Vine, 2025)

### 4.3.3 Wasted Shank – Titanium

The Titanium bolt exhibits the lowest tensile stress values among all configurations, with a maximum stress of 651.78 MPa and minimum stress of 620.07 MPa. This contributes to the highest FSF of 2.9, making it the best-performing design in terms of fatigue resistance.

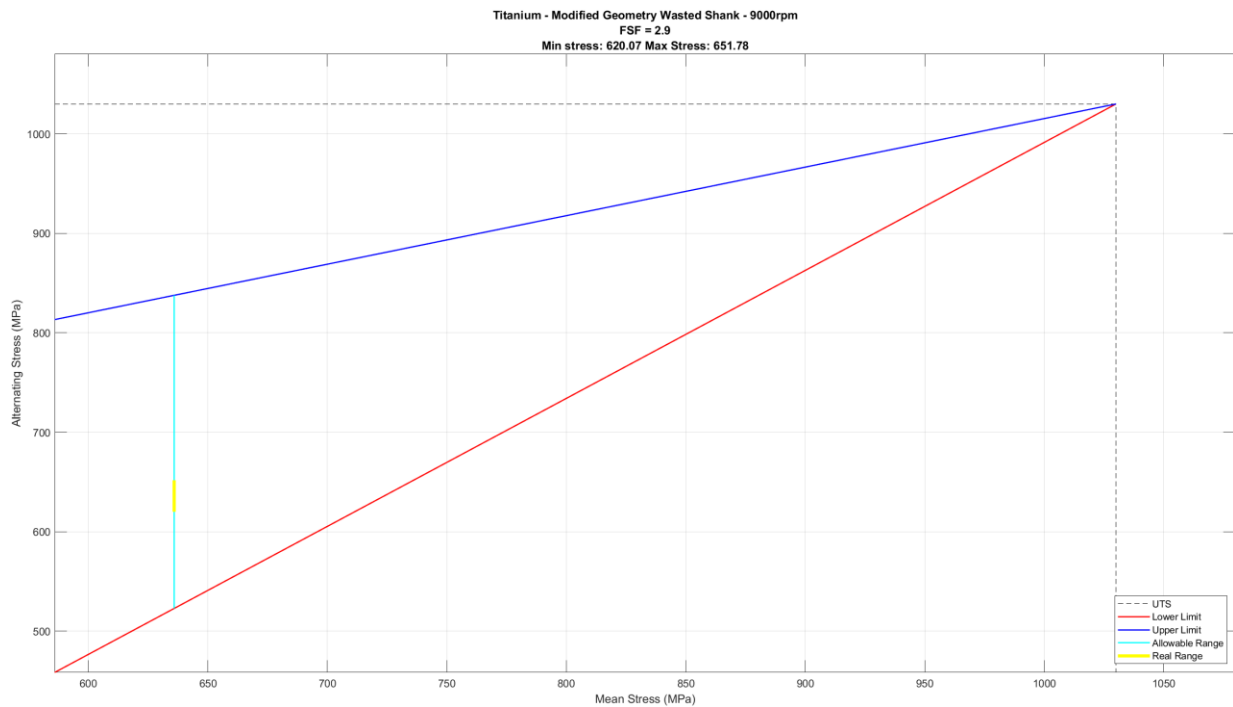


Figure 20 - Titanium Wasted Shank (Vine, 2025)

## 4.4 Thread Changes

Rolled threads offer significant advantages over cut threads, particularly in enhancing bolt fatigue strength. Mohandesi et al. (2007) found that rolled threads can improve fatigue strength by up to 55% compared to machined threads, primarily due to the introduction of compressive residual stresses and increased dislocation density from the rolling process. Additionally, the smoother surface finish of rolled threads lowers stress concentrations, further enhancing fatigue performance.

To improve fatigue resistance while maintaining ISO compatibility, the M8 x 1.0 thread profile was optimised by increasing the root radius ( $r$ ) from 0.144 mm to 0.18–0.20 mm and slightly reducing the thread height ( $h$ ) from 0.613 mm to 0.5 mm. This modification is based on studies demonstrating that increasing the root radius significantly reduces stress concentration factors ( $K_t$ ) and improves fatigue performance by distributing load more evenly across the thread profile (Connolly, 2020).

### 4.4.1 $K_t$ comparison rolled threads

Table 3 - Rolled Threads and the Change to  $K_t$  Value (Vine, 2025)

| Bolt Option   | $K_t$ Value |
|---------------|-------------|
| Standard      | 3.8599      |
| Rolled Thread | 3.5048      |

Since the stress concentration factor ( $K_t$ ) is highly sensitive to the root radius and thread height, reducing  $h/r$  while increasing  $r$  leads to a lower peak stress at the thread root, mitigating the risk of fatigue failure, particularly at high-RPM. The selected dimensions ensure compliance with ISO 965-1 (ISO, 2013) tolerances while optimising the thread profile for high-load and fatigue-critical environments.

#### 4.4.2 Rolled Threads and Wasted Shank – Steel

Rolled threads predominantly influenced fatigue life rather than peak stress levels. Showing another improvement over the wasted shank, raising the FSF from 1.71 to 1.88.

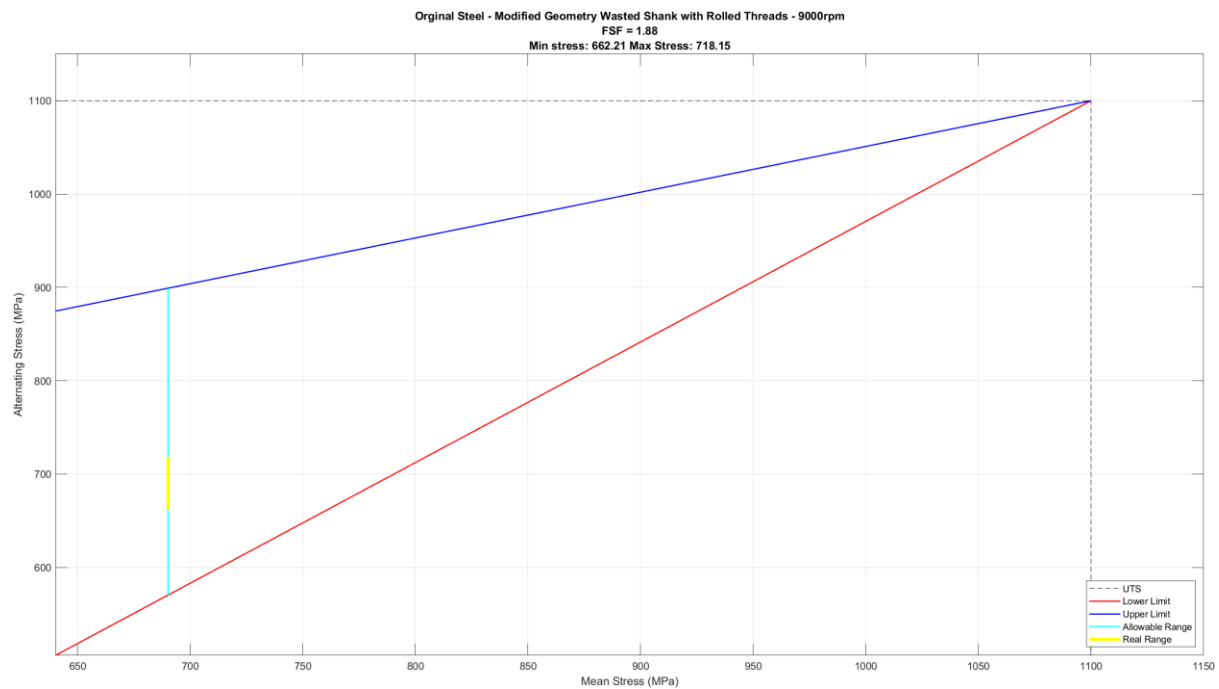


Figure 21 - Rolled Thread Original Material (Vine, 2025)

#### 4.4.3 Rolled Threads and Wasted Shank – Inconel 718

Exhibiting a similar increase to the steel the FSF have improved, increasing to 2.49 from 2.26 in the wasted shank test.

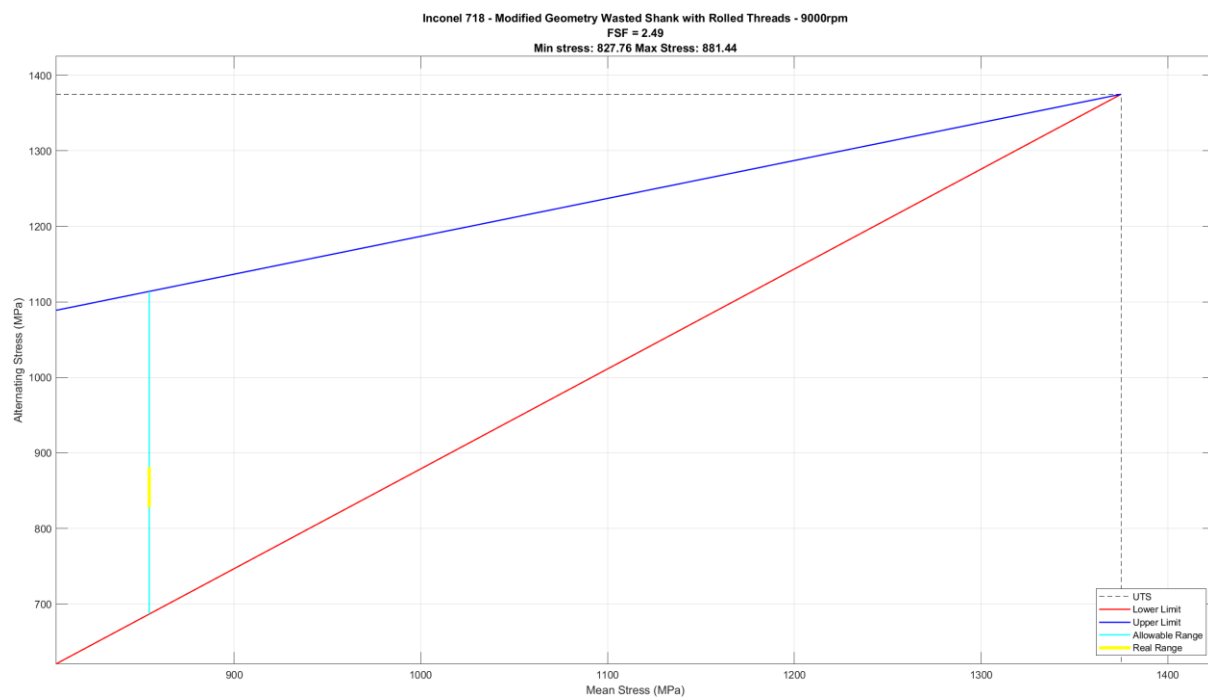


Figure 22 - Rolled Thread Inconel 718 (Vine, 2025)

#### 4.4.4 Rolled Threads and Wasted Shank – Titanium

The Titanium bolt with a wasted shank and rolled threads at 9000 RPM achieved an FSF of 3.19, making it the closest configuration to the original steel bolt at 6000 RPM (FSF 3.41). This indicates that the combination of weight reduction, improved fatigue resistance from rolled threads, and lower overall stresses has successfully mitigated the effects of increased RPM. The maximum stress (651.78 MPa) and minimum stress (620.07 MPa) are also the lowest among the tested configurations, reinforcing the benefits of titanium's high strength-to-weight ratio and reduced inertia forces. This result highlights that a proper balance of material selection and geometry optimization can help maintain fatigue safety at elevated engine speeds.

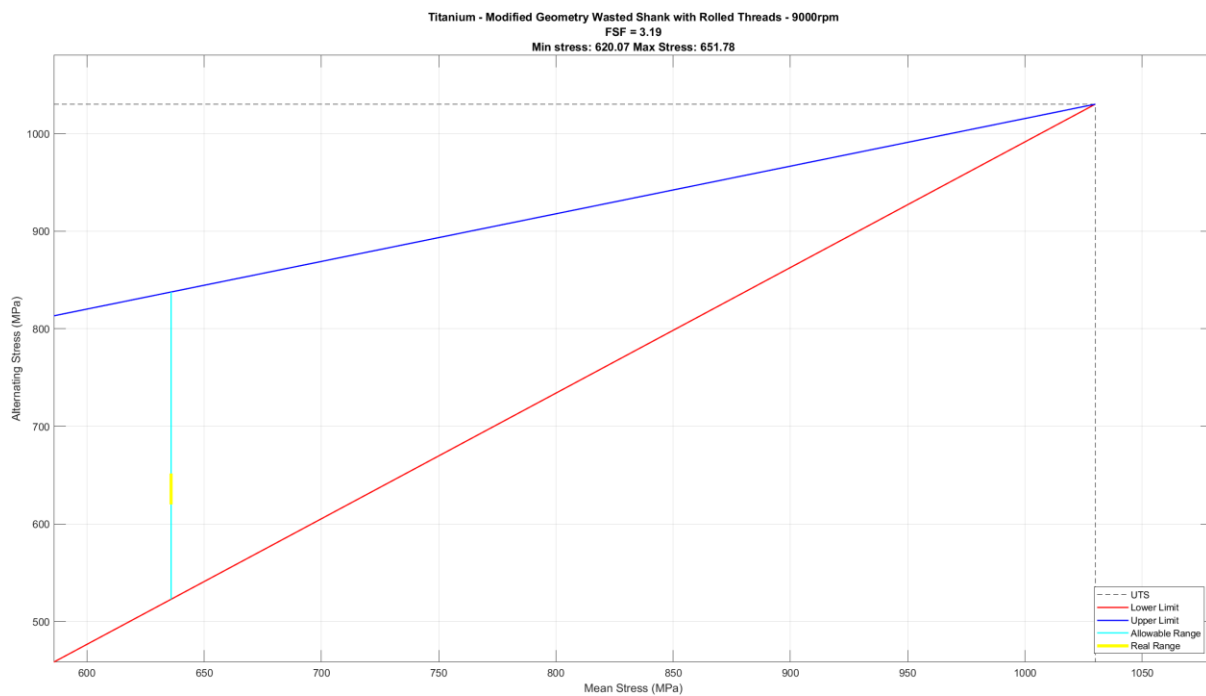


Figure 23 - Titanium Rolled Thread (Vine, 2025)

#### 4.5 Tightening Torque

Tightening torque plays a critical role in ensuring the preload force in conrod bolts is sufficient to resist fatigue while avoiding excessive stress that could lead to failure. The applied torque must overcome both frictional resistance at the threads and the underhead contact surface. This section calculates the required tightening torque for different materials using the axial forces determined earlier and a friction coefficient of 0.12 (Eccles, 2023).

```
%Tightening Torque
r= 0.5*((8+13)/2);
disp(r);
BoltCoe = 0.12; %% (Eccles,2023)
F_friction_bolt = BoltCoe*F_axial;
disp(F_friction_bolt)
Thf = BoltCoe *F_axial *r/1000;
disp(Thf);
Tft=F_friction_bolt * r/1000;
disp(Tft);
Ttotal= Thf+Tft; %Nm
disp(Ttotal);
```

Figure 24 - MATLAB Code for Tightening Torques (Vine, 2025)

The MATLAB script in Figure 24 automates tightening torque calculations. Table 4 shows Inconel 718 requires the highest torque 42.32 Nm due to its higher axial force, while Titanium requires the least 31.70 Nm due to its lower stiffness. The original steel bolt, 33.86 Nm offers a balance between preload and applied torque.

*Table 4 - Tightening Torque after Adding Coefficient (Vine, 2025)*

| Material         | Axial Force (n) | Tightening Torque (Nm) |
|------------------|-----------------|------------------------|
| Steel (Original) | 26872           | 33.86                  |
| Inconel 718      | 33590           | 42.32                  |
| Titanium         | 25162           | 31.70                  |

## 5 Engine Configuration and Torque Characteristics

The layout of an engine's crankshaft and firing order significantly influences its torque characteristics, vibration levels, and overall performance. While the study primarily focuses on inline-four configurations, other layouts, such as a V4, offer alternative approaches to balancing forces and optimizing power delivery.

### 5.1 Inline 4 Cylinder

The flat-plane crankshaft in four-cylinder engine features  $180^\circ$  crankpin intervals, resulting in a uniform  $180^\circ$ - $180^\circ$ - $180^\circ$ - $180^\circ$  firing sequence. This configuration enables high-revving efficiency due to balanced secondary forces but produces larger inertial torque fluctuations (Tromans, 2024), leading to increased vibration. In contrast, the crossplane crankshaft, as used in the Yamaha R1, spaces crankpins at  $90^\circ$  intervals ( $0^\circ$ ,  $90^\circ$ ,  $180^\circ$ ,  $270^\circ$ ) and follows a  $270^\circ$ - $180^\circ$ - $90^\circ$ - $180^\circ$  firing sequence (Trevitt, 2021). This design reduces peak inertial torque fluctuations, improving power delivery and traction while minimizing torque spikes. The analysis compares torque contributions from gas and inertia forces, illustrating the differences in torque smoothness between these configurations.

#### 5.1.1 4 Cylinder Inertia Torque

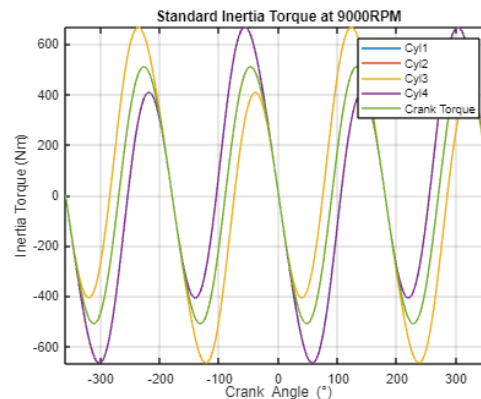


Figure 25 - Flat Plane Inertia (Vine, 2025)

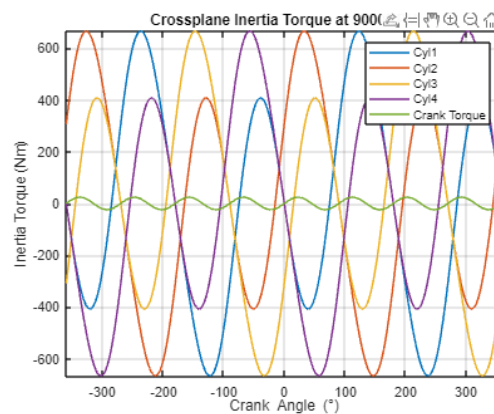


Figure 26 - Crossplane Inertia (Vine, 2025)

The inertia torque graphs in Figures 25 and 26 further illustrate the differences between flat-plane and crossplane crankshafts. The flat-plane design produces higher peak-to-peak variations in inertia torque, contributing to greater oscillations in engine balance. In contrast, the crossplane configuration shows a more evenly distributed inertia torque, reducing oscillations and enhancing overall smoothness.

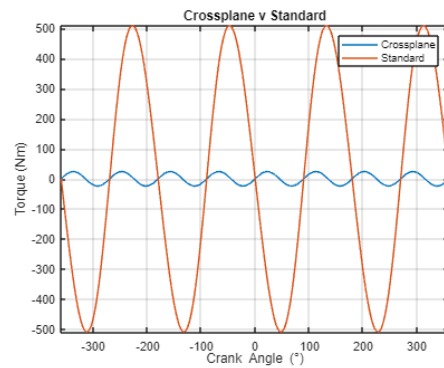


Figure 27 - Crossplane v Standard Comparison (Vine, 2025)

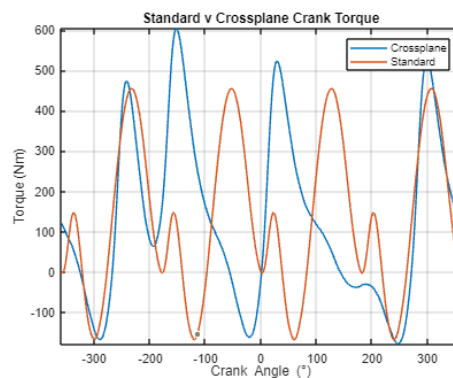


Figure 28 - Crank Torque Comparison (Vine, 2025)

The crank torque graph highlights the difference in torque delivery between flat-plane and crossplane configurations. The flat-plane crank exhibits sharper fluctuations, leading to increased vibrations, while the crossplane design smooths torque output, enhancing traction and control. This results in improved stability at the cost of a slight reduction in peak torque.

## 5.2 Other Configurations

### 5.2.1 V5 Crossplane

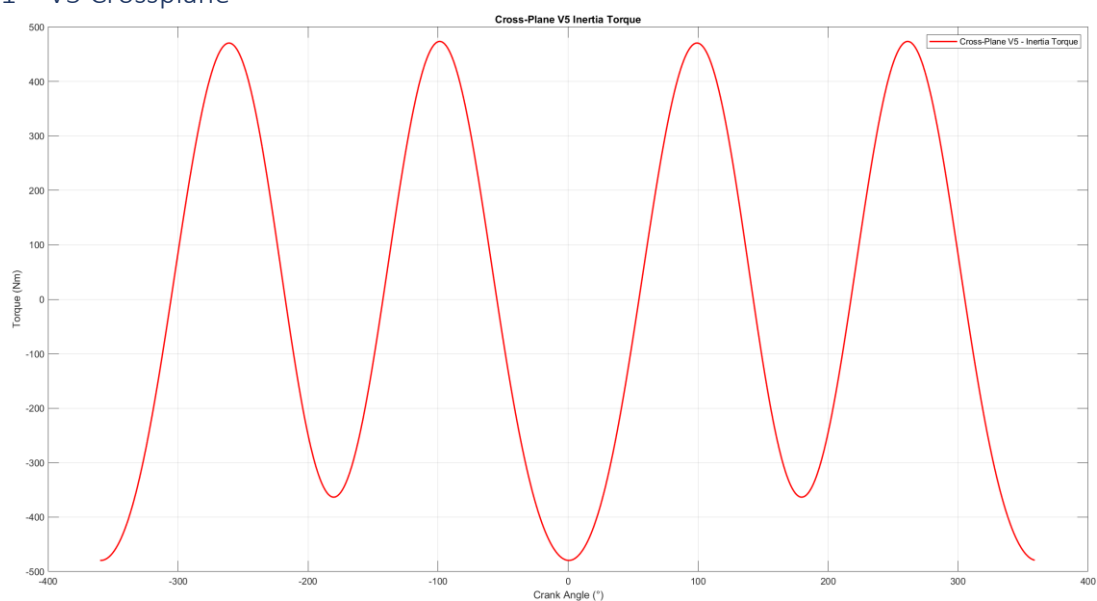


Figure 29 - V5 Inertia Torque (Vine, 2025)

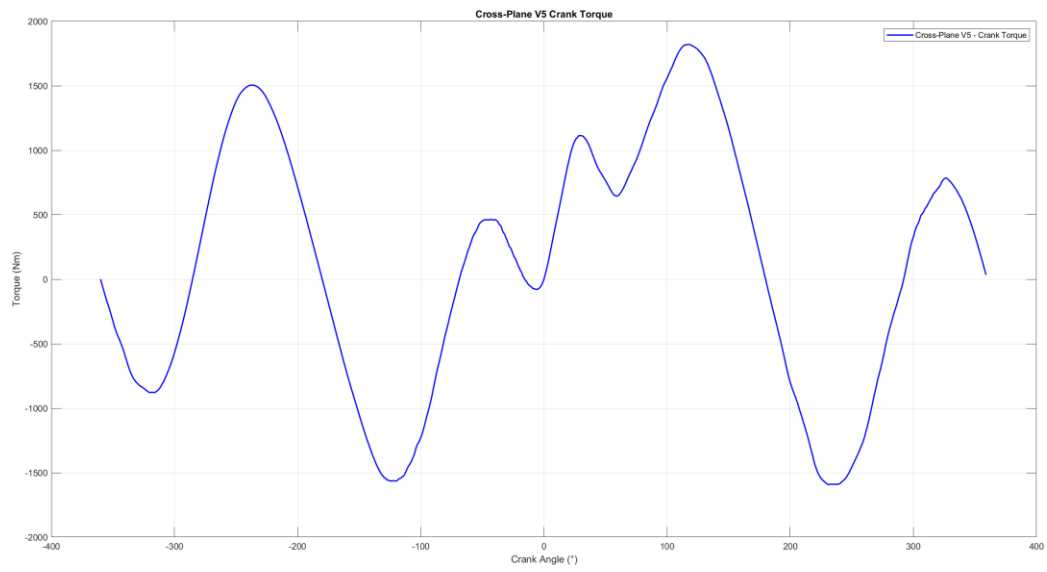


Figure 30 - V5 Crank Torque (Vine, 2025)

## 6 Conclusion

The optimisation of conrod bolts through material and geometric modifications has demonstrated significant improvements in FSF values and stress management at an increased RPM. Material changes, notably with Titanium, coupled with design adaptations like wasted shank and rolled threads, have enhanced the FSF, ensuring greater durability under high-stress conditions. While the baseline FSF for the original steel bolts at 6000 RPM was 3.41, the improvements have brought the titanium configurations with rolled threads and wasted shank close to this figure at 9000 RPM, showing an FSF of 3.19. This progress underscores our success in increasing the operational limits of engine components, which could lead to more efficient and reliable high-performance engines.

**Word Count:** 2682

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## 8 Appendix

```

clc;
clear;
close all;
rpm = readmatrix("1600SigmaPressureData.xlsx","Range","B2:G2"); % rpm data up to 6000 rpm
crank = readmatrix("1600SigmaPressureData.xlsx","Range","A5:A724"); % crank angle
pressure9000 = readmatrix("1600SigmaPressureData.xlsx","Range","B5:G724")*10^5; % pressure data up to 6000 rpm
rpm_ext = 7000:1000:9000;
pressure_ext = zeros(length(crank), length(rpm_ext));
for i = 1:length(crank)
    pressure_ext(i, 1:length(rpm)) = interp1(rpm, pressure9000(i,:),rpm,'pchip');
    growth_rate = log(pressure9000(i,end)/pressure9000(i,end-1))/1000;
    for j = 1:length(rpm_ext)
        delta_rpm = rpm_ext(j)-6000;
        pressure_ext(i,length(rpm)+j) = pressure9000(i, end)*exp(growth_rate*delta_rpm);
    end
end
rpm = 1000:1000:9000;
pressure9000=pressure_ext;

```

Figure 31 - MATLAB Code for Extracting and Expanding RPM (Vine, 2025)

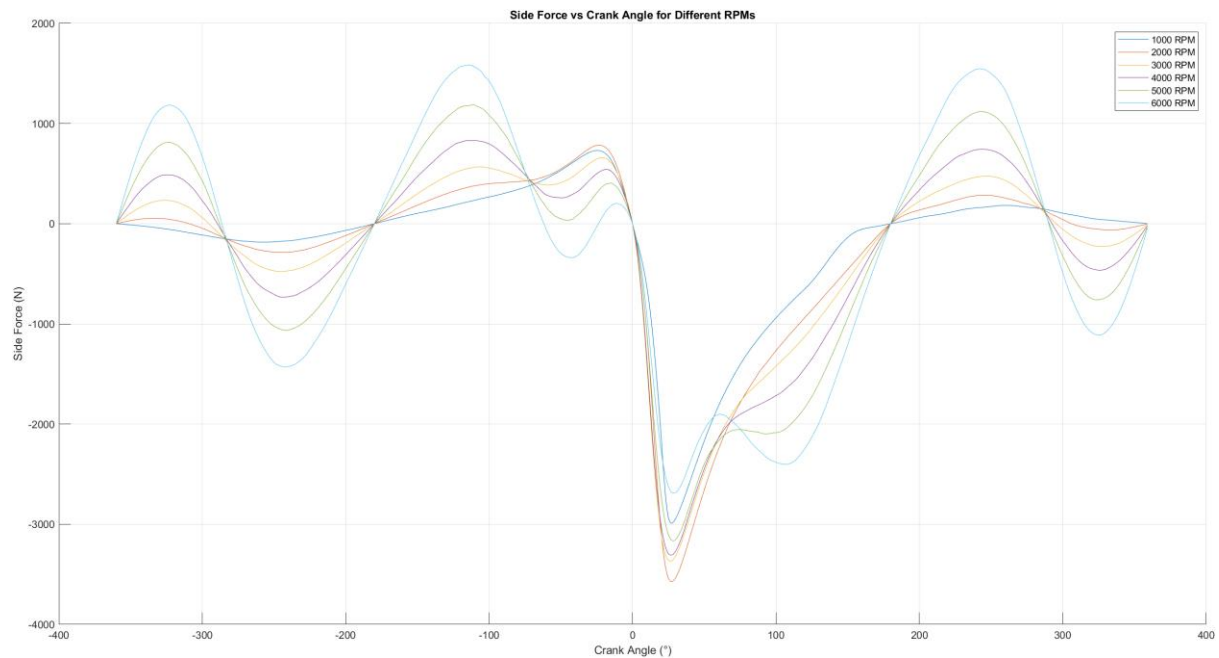


Figure 32 - Side Forces on the Piston (Vine, 2025)

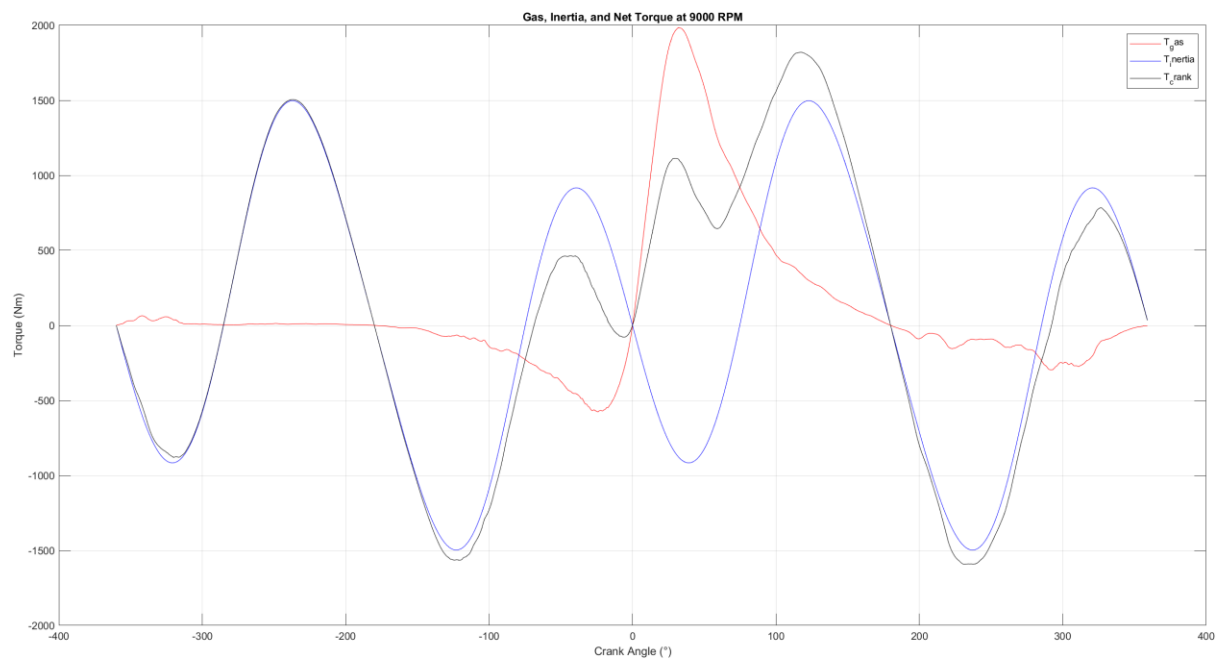


Figure 33 - Gas, Inertia and Net Torque Comparison (Vine, 2025)

```

d = 8; % Bolt diameter (mm)
h = 0.613; % Thread height (mm)
r = 0.144; % Root radius (mm)

h_r_ratio = h / r;
% Select appropriate coefficients based on h/r ratio
if h_r_ratio >= 0.1 && h_r_ratio <= 2.0
    C1 = 0.850 + 2.628 * sqrt(h_r_ratio) - 0.413 * sqrt(h_r_ratio)^2;
    C2 = -1.119 - 4.826 * sqrt(h_r_ratio) + 2.575 * sqrt(h_r_ratio)^2;
    C3 = 3.563 - 0.514 * sqrt(h_r_ratio);
    C4 = -2.294 + 2.713 * sqrt(h_r_ratio) + 0.340 * sqrt(h_r_ratio)^2;
elseif h_r_ratio > 2.0 && h_r_ratio <= 50.0
    C1 = 0.833 + 2.069 * sqrt(h_r_ratio) - 0.090 * sqrt(h_r_ratio)^2;
    C2 = 2.732 - 4.157 * sqrt(h_r_ratio) + 0.176 * sqrt(h_r_ratio)^2;
    C3 = -8.859 + 3.327 * sqrt(h_r_ratio) - 0.320 * sqrt(h_r_ratio)^2;
    C4 = 6.294 - 3.239 * sqrt(h_r_ratio) + 0.154 * sqrt(h_r_ratio)^2;
else
    error('h/r ratio is outside the valid range for the given formula.');
```

```

end

Kt = C1 + C2 * ((2*h)/d) + C3 * ((2*h)/d)^2 + C4 * ((2*h)/d)^3;
disp(Kt);

```

Figure 34 - Stress Concentration Script in MATLAB (Vine, 2025)

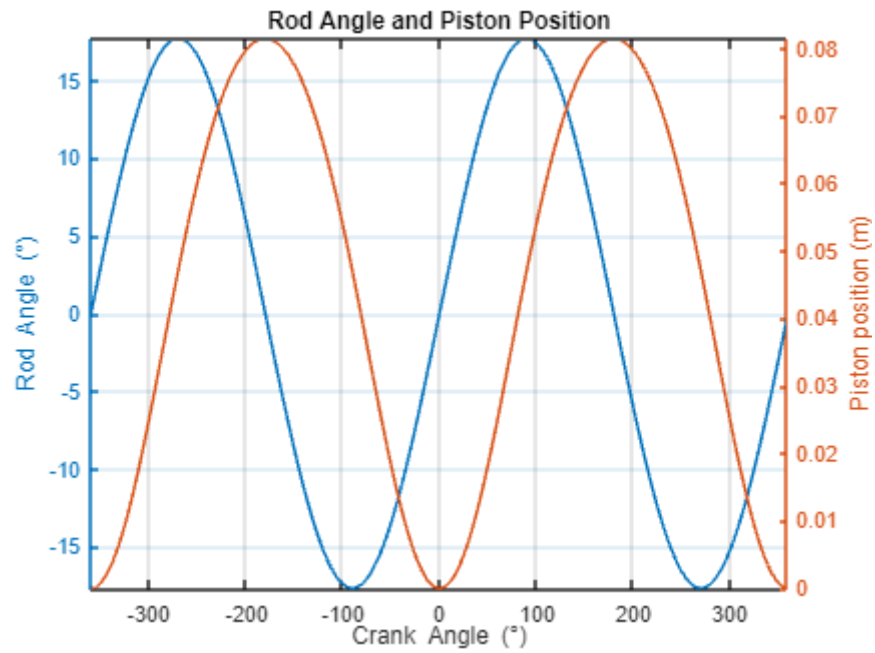


Figure 35 - Rod Angle and Piston Position (Vine, 2025)

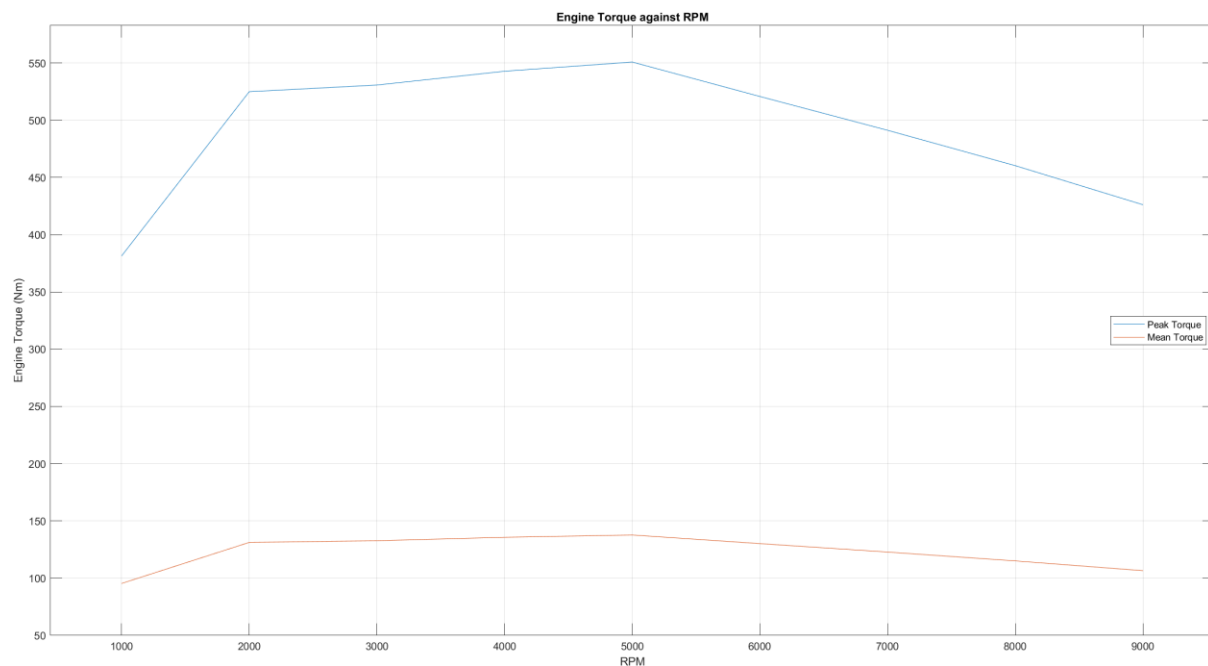


Figure 36 - Engine Torque v Rpm 4 cylinder (Vine, 2025)

**Technical specifications**

|                                         |                                                   |
|-----------------------------------------|---------------------------------------------------|
| <b>Engine code</b>                      | AGZ                                               |
| <b>V-angle</b>                          | 15°                                               |
| <b>Displacement</b>                     | 2324 cm <sup>3</sup>                              |
| <b>Bore</b>                             | 81.0 mm                                           |
| <b>Stroke</b>                           | 90.2 mm                                           |
| <b>Compression ratio</b>                | 10.0 : 1                                          |
| <b>Firing order</b>                     | 1 - 2 - 4 - 5 - 3                                 |
| <b>Mixture preparation and ignition</b> | Bosch Motronic M3.8.3                             |
| <b>Fuel</b>                             | 95 RON unleaded premium                           |
| <b>Exhaust gas aftertreatment</b>       | Three-way catalytic converter with lambda control |

*Figure 37 - Volkswagen V5 Specifications (Volkswagen AG, 1997)*