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Analytical Assessment of Engine Internal Components

**ACHA6006**

*Contemporary Powertrain Design*

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## 1. Introduction

“Fatigue life is the number of cycles a material or component can endure before failure occurs due to repeated loading and unloading.” Higher stress levels generally lead to a shorter life, making fatigue a major consideration in component longevity and reliability (SDC Verifier – 2025).

This report presents an analytical assessment of the forces, stresses, and fatigue safety factors acting on bottom-end components of the 1.6-litre Ford Sigma engine using MATLAB.

Pressure data obtained from experimental sources was used to calculate gas and inertial forces, which were then applied to determine the stress distribution across critical components. Goodman diagrams were generated for fatigue analysis to evaluate the cyclic stress endurance of components, predicting failure points and highlighting areas of potential over-engineering.

Furthermore, this study explores the implications of increasing engine speed to 900RPM, extending the initial dataset through interpolation to assess the components under higher mechanical loads. The effects of different crankshaft considerations were also investigated to determine their impact on instantaneous torque characteristics.

Additionally, the influence of surface treatments such as nitriding on fatigue resistance was evaluated to assess potential durability enhancements.

This analysis was conducted using course-specific materials from the ACHA6006 module, along with external academic and industry sources to validate methodologies and findings. The PowerPoint presentations and module documentation provided theoretical framework and formulae for force and fatigue analysis in engine components. These derivations included:

- A Kinematic Analysis detailing gas force equations from cylinder pressure and inertial force calculations.
- An application of FSF tables and Goodman diagrams to evaluate fatigue resistance in specified component areas.
- A Crankshaft Torque Analysis, investigating fluctuations and their impact on engine performance.

## 2. Methodology & Simulation Set-Up.

### 2.1 - Engine Data and Initial Script Input.

To analyse the piston kinematics and forces acting in the Ford engine, MATLAB was used to process cylinder pressure data and compute piston displacement, velocity, and acceleration across a full engine cycle.

The Sigma Pressure Data spreadsheet provided pressure values in BAR at varying crank angles for RPMs ranging from 1000-6000RPM. This data was imported into MATLAB alongside specific engine geometry, cylinder parameters and dynamics; allowing for an initial kinematic modelling of the 1.6-litre engine.

Piston position was determined using the equation:

$$s = (r + l) - (r \cos \theta + l \cos \phi)$$

Where:

$s$  = Piston Position

$r$  = Crank Radius

$l$  = Connecting Rod Length

$\theta$  = Crank Angle

$\phi$  = Connecting Rod Angle  $= \sin^{-1} \frac{r \sin \theta}{l}$

This formula allowed for accurate tracking of the piston's motion during crankshaft rotation over a full 720° combustion cycle.

The resulting piston position (figure 1) against crank angle data was visualised to illustrate how piston movement changes with varying crank angle, providing insight into correct determination of ( $s$ ) and accurate excel data layout.

Since velocity is the rate of change of position, the piston velocity was obtained by differentiating ' $s$ ' with respect to crank angle.

A numerical differentiation approach was used with the 'central difference method' applied:

$$\frac{ds}{dt} = r\omega \sin \theta + \left( \frac{r^2 \omega}{2l} \sin 2\theta \right)$$

Where  $\omega = RPM \times \frac{2\pi}{60}$

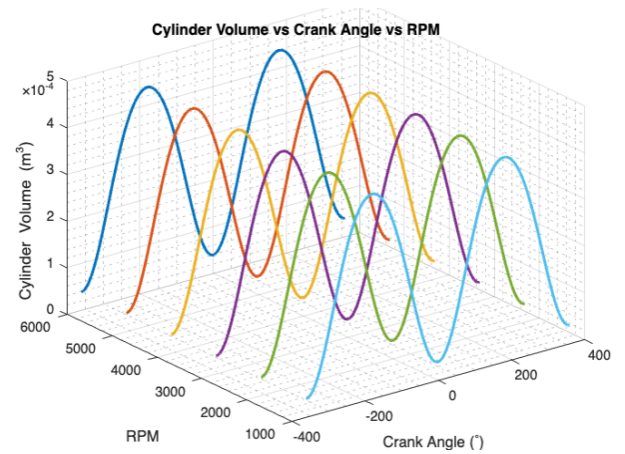


Figure-1: Cylinder Vol. vs Crank Angle

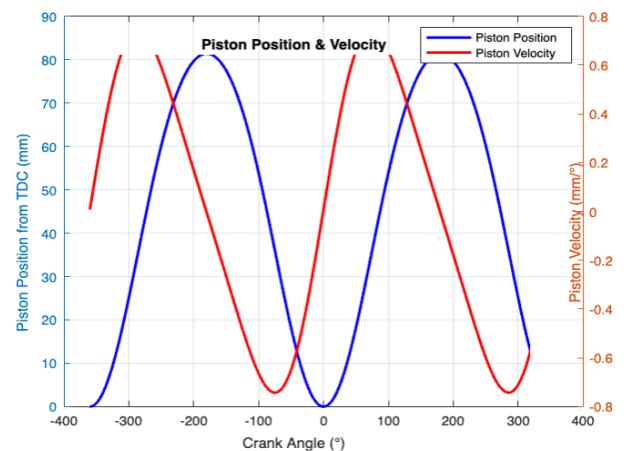


Figure-2: Piston Position & Velocity.

“Acceleration is [...] the first derivative of the velocity curve and the second derivative of the position [or displacement ‘s’] curve”. It can be measured as “how rapidly velocity is changing, usually expressed with reference to time” (EPI Inc. – 2021). This allows Piston acceleration to be derived as the rate of change of velocity, using the second derivative of displacement:

$$\frac{d^2s}{dt^2} = r\omega^2 \cos\theta + \frac{r^2}{l}\omega^2 \cos 2\theta$$

This piston acceleration “is the sum of [...] two separate acceleration orders: primary and secondary” which combine to produce the total acceleration curve. (EPI Inc. – 2021).

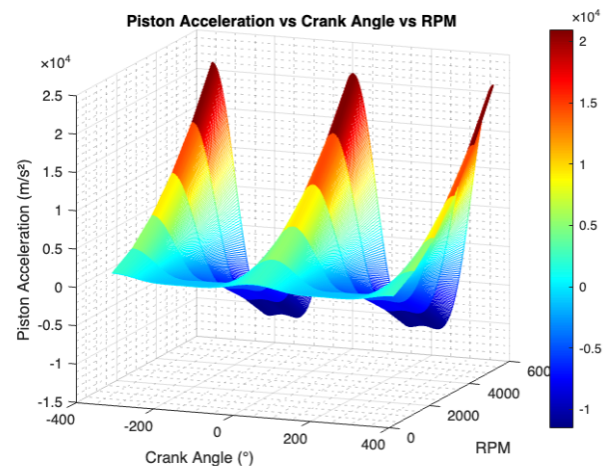


Figure-3: Piston Acceleration vs Crank Angle.

The 1<sup>st</sup> order acceleration ( $r\omega^2 \cos\theta$ ) displays the “piston motion produced by the component of crankpin movement projected onto the vertical plane”.

The 2<sup>nd</sup> order acceleration ( $\frac{r^2}{l}\omega^2 \cos 2\theta$ ) is due to “additional piston movement between TDC and the max velocity point” as well as subtracting from piston motion between the maximum velocity point and BDC (EPI Inc. – 2021).

The maximum positive value of [piston] acceleration occurs [around] TDC [until] the piston stops speeding up and begins to slow down”, the acceleration then changes direction (and turns from a positive number to a negative number) and “momentarily passes through zero” (EPI Inc. – 2021).

As both the primary and secondary acceleration components act in the same direction, they amplify the total acceleration magnitude and force. As the piston moves toward bottom dead centre (BDC), the second order acceleration acts in the opposite direction to the first order; offsetting the effect and reducing the total acceleration magnitude at BDC.

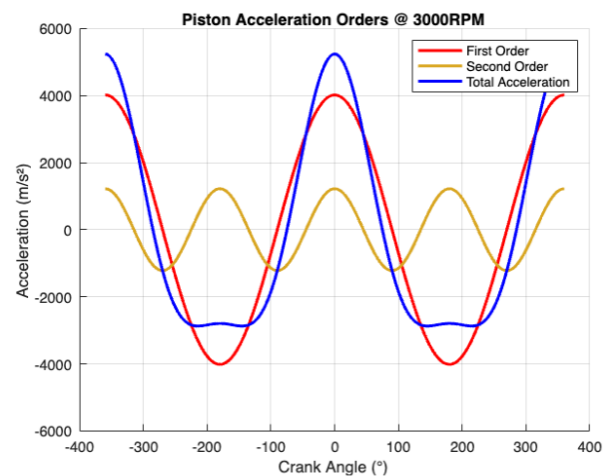


Figure-4: Piston Acceleration Orders.

This coincides with research showing that “the acceleration magnitude at TDC is substantially larger than that at BDC [with] peak piston speed [occurring] closer to TDC than to BDC” (Mosedale, A. – n.d.)

“Connecting rods are subject to forces generated by mass and fuel combustion. These two forces [result] in axial and bending stresses” (Kumar, M. – 2017).

Inertial forces are present due to the acceleration of the reciprocating assembly, consisting primarily of the piston, gudgeon pin, piston rings, and connecting rod. These forces act on the connecting rod and crankshaft; influencing engine balancing, stress distribution, and the fatigue life of the components.

To determine the inertial forces exerted on the connecting rod by the reciprocating components, Newton’s second law applies:

$$F = ma$$

Where:

$$F_{inertia} = m_{rec} \times a$$

$$m_{rec} = \text{Reciprocating Mass}$$

$$a = \text{Piston Acceleration}$$

Since piston acceleration was already defined, the inertial force was equated as:

$$F_{inertia} = m_{rec} \times \left( r\omega^2 \cos\theta + \frac{r^2}{l} \omega^2 \cos 2\theta \right)$$

The resultant 3D plot of Inertia confirmed that peak inertial forces occur at TDC and BDC, where acceleration is greatest. The force distribution also follows the same pattern as the acceleration curve but scaled by mass with results displayed in Newtons.

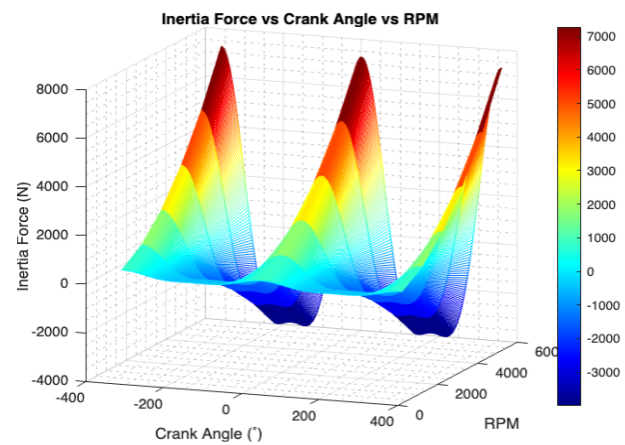


Figure-5: Inertia Force vs Crank Angle

Gas forces are present due to the combustion process, where air/fuel mixture is compressed in the cylinder by the piston acting as a part of the reciprocating assembly. The gas force acting on the piston is computed using the equation:  $F_{gas} = P_{cyl} \times A$

Cylinder pressures of the Sigma 1.6 were provided in BAR and converted to Pascals for the equation.

The pressure peaks around TDC, correlating with the expected timing of the maximum combustion force exerted on the piston during the power stroke.

At low-to-mid RPM ranges, combustion pressure is more stable and reaches higher peak values.

Higher RPM ranges show peak pressure occurring later, possibly limited by ignition timing capabilities of the engine, where “as RPM increases, more ignition timing is needed [...] due to slower flame speed” (Szabo, B. – 2017).

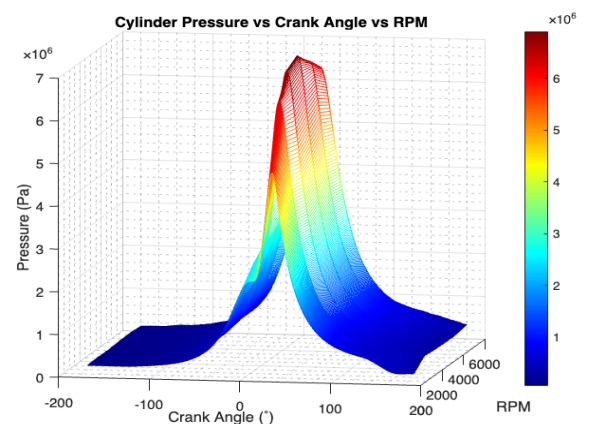


Figure-6: Cylinder Pressure vs Crank Angle

The Net force acting on the piston is determined by considering both the inertial forces and gas forces acting on the reciprocating components.

This is calculated as:  $F_{net} = F_{inertia} - F_{gas}$

This order of formula components ensure compressive forces are represented as negative values, following the convention of tensile loads being positive and compressive loads negative.

Figure 7 shows a plot of net force against crank angle for different RPMs. The most significant compressive force can be seen at TDC, aligning with cylinder pressure and gas force expectations during the power stroke. The largest tensile forces also coincide with correct crank angle ranges, showing peak inertial forces on intake and exhaust strokes.

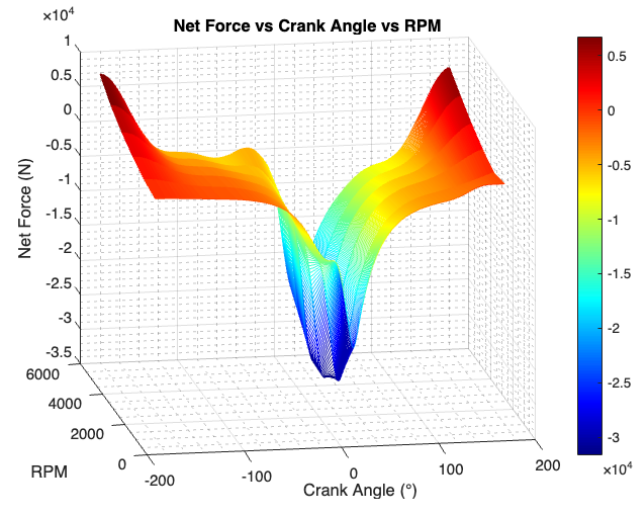


Figure-7: Net Force. vs Crank Angle

It was understood that “connecting rods are subject to forces generated by mass and fuel combustion. These two forces [result] in axial and bending stresses”. This makes endurance limit “a primary design criterion” with factors such as fatigue “often used as evaluation indexes for the performance of engine connecting rods” (Kumar, M. – 2017).

Con. rod force  $F_{rod}$  follows the same trend as the  $F_{net}$  due to being directly derived from net force as:

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

Since  $F_{net}$  is calculated from both gas and inertial force over crank angle and RPM, the connecting rod force reflects their behaviour throughout the rotation of the crank shaft.

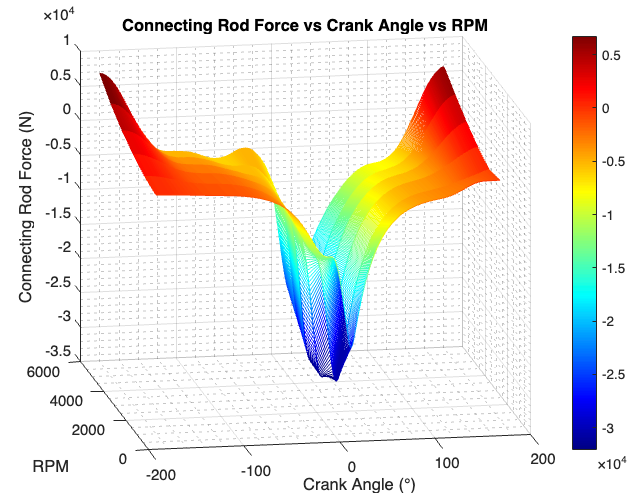


Figure-8: Con.-Rod Force. vs Crank Angle

The force applied to the connecting rod is influenced by the angle of the rod relative to the crankshaft. The amplitude of the connecting rod force has marginally increased due to the denominator effect of the crankshaft rotation ( $\cos\phi$ ).

Because of this, component design “and material must provide a high fatigue resistance and a low weight, leading to low dynamic loading of the crank shaft and piston rod” (Humelnicu, C. – 2018).

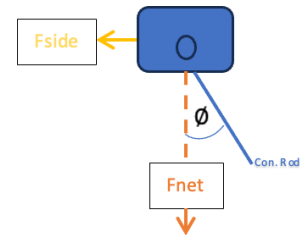
Piston lateral force ( $F_{side}$ ) is a reactional force that occurs due to the angular inclination of the connecting rod ( $\phi$ ) during the reciprocation of the assembly. Since the connecting rod is not perfectly vertical throughout the cycle, a lateral force is exerted on the piston, pushing it against the cylinder wall causing “contact stresses between the piston skirt and cylinder” (Humelnicu, C. – 2018).

The force is derived from the net force acting along the axis of the piston, using Pythagorean theory:

$$F_{side} = F_{net} \times \tan\phi$$

Where:

$$\tan\phi = \frac{Op}{Adj} = \frac{F_{side}}{F_{Net}}$$



The force alternates direction depending on crankshaft position. At TDC and BDC the connecting rod angle is small, meaning the lateral force approaches zero. The maximum lateral force occurs at mid-stroke where  $\phi$  is largest.

Higher RPM ranges increase the magnitude of lateral force, as the larger inertial forces amplify net force values.

Lateral force significantly impacts piston frictional force, causing figures 9 and 10 to mirror results, with frictional force scaled by the coefficient of friction:

$$F_{friction} = \mu \times F_{side}$$

As the crank angle changes, the lateral force reverses direction, causing friction to alternate between the thrust and the non-thrust sides of the cylinder, eventually leading to cylinder ovaling.

Friction is typically higher during mid-stroke, where  $F_{side}$  reaches peak values due to maximum connecting rod inclination.

The coefficient of friction is dictated by oil viscosity, film thickness, and engine temperature. Typical values of piston/cylinder wall frictional coefficient can range between 0.02 and 0.12 (Anggraini, L. – 2020).

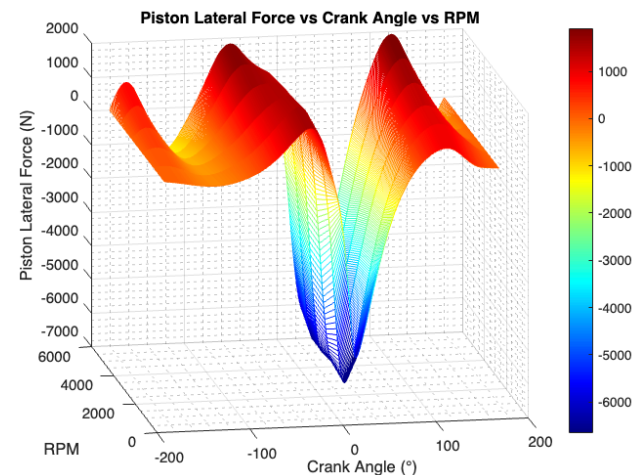


Figure-9: Piston Lateral Force. vs Crank Angle

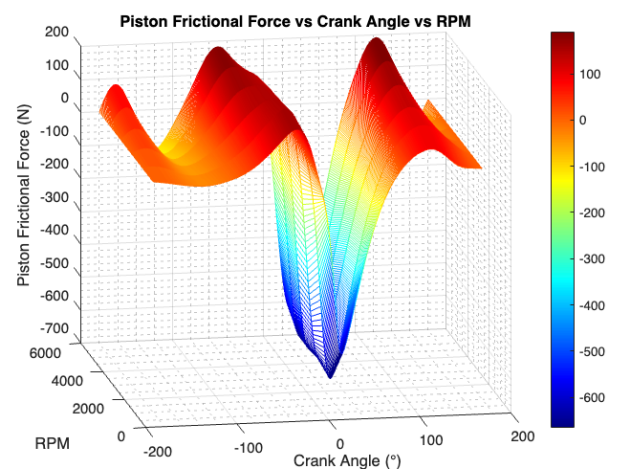


Figure-10: Piston Frictional Force. vs Crank Angle

### 3. Results & Analysis.

#### 3.1 - Engine Data and Initial Script Input.

Fatigue Safety Factors were determined by evaluating the stress conditions at the connecting rod's midpoint across the full range of engine speeds. Material properties for the connecting rod were provided in the Ford EcoBoost 1.6 GDTi Engine Geometry Specifications and used for the following equations:

$$\sigma_{max} = \frac{F_{max}}{CSA} \quad \sigma_{min} = \frac{F_{min}}{CSA}$$

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

$$\sigma_{range} = \sigma_{max} - \sigma_{min}$$

$$FSF = \sigma_{allowable} / \sigma_{range}$$

To assess fatigue performance, the allowable stress range was determined using the Goodman relation, incorporating the ultimate tensile strength (UTS) and the material's endurance limit (45% of the UTS):  $\sigma_{allowable} = 2 \times \sigma_{endurance\ lim.} \times \left(1 - \frac{\sigma_{mean}}{UTS}\right)$

(1000-6000 RPM) FSF - Con. Rod Midpoint:

| RPM  | Max_Tensile_MPa | Min_Compressive_MPa | Mean_Stress_MPa | Stress_Range_MPa | FSF    |
|------|-----------------|---------------------|-----------------|------------------|--------|
| 6000 | 48.208          | -190.13             | -70.961         | 238.34           | 3.1001 |
| 5000 | 31.433          | -211.23             | -89.897         | 242.66           | 3.1151 |
| 4000 | 18.718          | -228.55             | -104.91         | 247.27           | 3.1117 |
| 3000 | 9.2767          | -229.51             | -110.12         | 238.79           | 3.2418 |
| 2000 | 1.7411          | -218.62             | -108.44         | 220.36           | 3.5061 |
| 1000 | -1.2958         | -168.75             | -85.022         | 167.45           | 4.4879 |

Figure-11: Con.-Rod at Midpoint A, FSF Table.

The Goodman diagram provides a visual representation of the fatigue behaviour of the connecting rod midpoint across the range of engine speeds.

While the actual stress ranges appear reasonable and remain within expected operating limits, the allowable stress range exceeding the upper and lower endurance limits suggests potential inaccuracies in the calculation. This discrepancy could stem from incorrect material properties, an overestimated endurance limit, errors in the applied goodman relation, or human input error. However, since the actual stress ranges remain within expected boundaries, the connecting rod is still operating in a safe fatigue region.

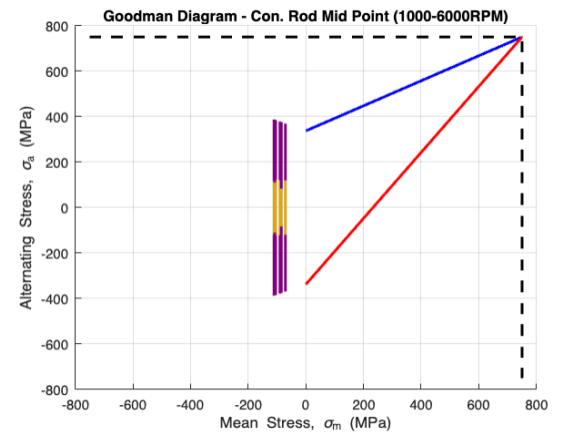


Figure-12: Con. Rod Goodman Diagram.

### 3.2 – Interpolation to 9000RPM.

In order to extend the analysis beyond the provided 1000-6000 RPM range interpolation techniques were applied to the cylinder pressure, forces, and stresses. Since experimental data at higher engine speeds was unavailable, the 'interp1' function was used in MATLAB to extrapolate results assuming a linear trend.

This enabled visualisation of fatigue and force trends at 9000rpm to help determine if failure risks increased significantly beyond the standard tested range.

Interpolation to 9000RPM

```
% Extended RPM range to 9000
rpm_extended = 9000:-1000:1000;

% Interpolated Pressure Data
[Crank_Angles_Grid_Ext, RPM_Grid_Ext] = meshgrid(linspace(-180, 180, 720), rpm_extended);
Pressure_Grid_Ext = interp1(rpm, Pressure_Grid, rpm_extended, 'linear', 'extrap');

% Pressure Grid Extension
Pressure_Pa_Grid_Ext = Pressure_Grid_Ext * 1e5;

% Gas Force Extension
F_gas_Ext = Pressure_Pa_Grid_Ext * Piston_Area;

% Inertia Force Extension
F_inertia_Ext = interp1(rpm, F_inertia, rpm_extended, 'linear', 'extrap');

% Net Force Extension
F_net_Ext = F_inertia_Ext - F_gas_Ext;

% Con. Rod Force Extension
F_rod_Ext = F_net_Ext ./ cosd(Phi);

% Piston Lateral Force Extension
F_side_Ext = F_net_Ext .* tand(Phi);

% Piston Frictional Force Extension
F_friction_Ext = mu * F_side_Ext;
```

Figure-13: 9000RPM Pressure Interpolation

The extended Goodman diagram shows that the actual stress range (gold) remains within expected range and the endurance limits (red and blue), indicating that the connecting rod operates within safe fatigue limits.

However, the allowable stress range (purple) extends beyond these limits, suggesting a potential issue with the input values or calculation assumptions.

Despite this discrepancy, the actual stress behaviour remains reasonable, supporting the validity of the analysis while highlighting the need for verification of the allowable range methodology.

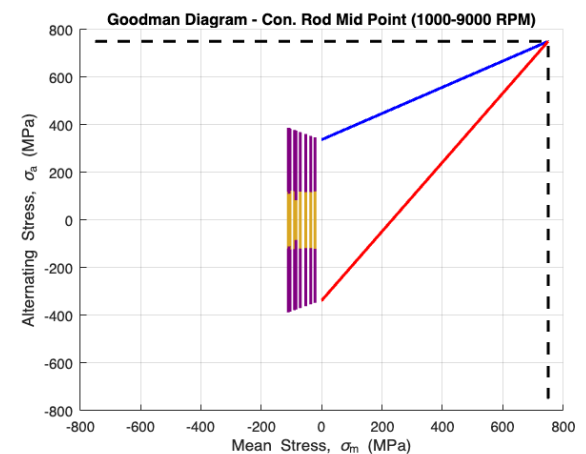


Figure-14: Extended Con. Rod Midpoint Goodman Diagram

(1000–9000 RPM) FSF – Gudgeon Pin at Point A:

| RPM  | Max_Tensile_MPa | Min_Compressive_MPa | Mean_Stress_MPa | Stress_Range_MPa | FSF     |
|------|-----------------|---------------------|-----------------|------------------|---------|
| 9000 | 295.29          | -413.27             | -58.99          | 354.28           | 0.67255 |
| 8000 | 245.02          | -447.77             | -101.38         | 346.4            | 0.71539 |
| 7000 | 194.75          | -499.09             | -152.17         | 346.92           | 0.74727 |
| 6000 | 144.47          | -558.29             | -206.91         | 351.38           | 0.77282 |
| 5000 | 94.201          | -621.35             | -263.57         | 357.78           | 0.79465 |
| 4000 | 56.095          | -674.99             | -309.45         | 365.54           | 0.806   |
| 3000 | 27.801          | -678.34             | -325.27         | 353.07           | 0.84455 |
| 2000 | 5.2176          | -642.56             | -318.67         | 323.89           | 0.91606 |
| 1000 | -3.8833         | -493.09             | -248.49         | 244.6            | 1.1484  |

Figure-15: Con.-Rod at Midpoint, Extended RPM FSF Table.

The Goodman diagram for the Gudgeon Pin at point A shows the actual stress range (gold) and allowable stress range (purple) remain well within the endurance limits (red and blue), indicating safe operation under the cyclic loading; with no immediate risk of fatigue failure.

Unlike previous cases, the allowable range does not exceed the endurance limits, suggesting that input values and calculations are correctly aligned in this instance.

### 3.3 - Surface-Finish Investigation.

Nitriding was investigated as a method of surface-finishing engine components for improved “hardness, temperature-resistance, and *[reduced]* coefficient of friction”; to prevent excessive fatigue accumulation (Pankew, D. – 2020).

The nitrided gudgeon pin Goodman diagram presents a significant improvement in fatigue performance compared to the non-nitrided case.

“Nitriding is a type of case-hardening process that hardens the outer layer of a part by adding nitrogen to its surface. The added nitrogen combines with iron and other alloying elements in the metal’s composition to form hard metallic nitrides” (Sild, S. – 2023).

This effect increases the allowable stress range and enhances fatigue resistance by reducing tensile stress concentrations and improving the overall durability under cyclic loading. A significant improvement was found in the fatigue safety factors, with an average FSF increase of 1.48 shown.

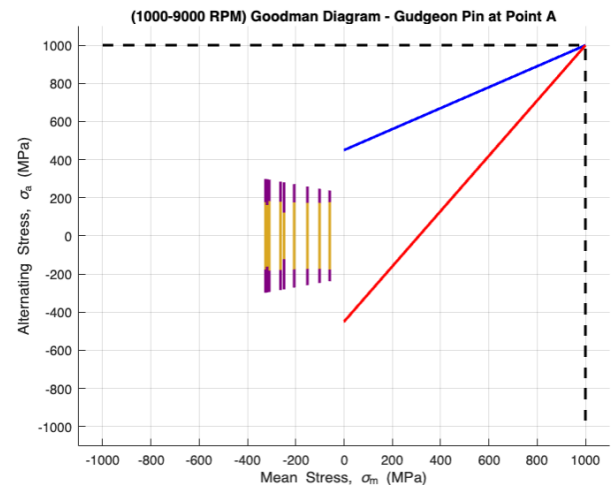


Figure-16: Gudgeon Pin (Point A) Goodman Diagram

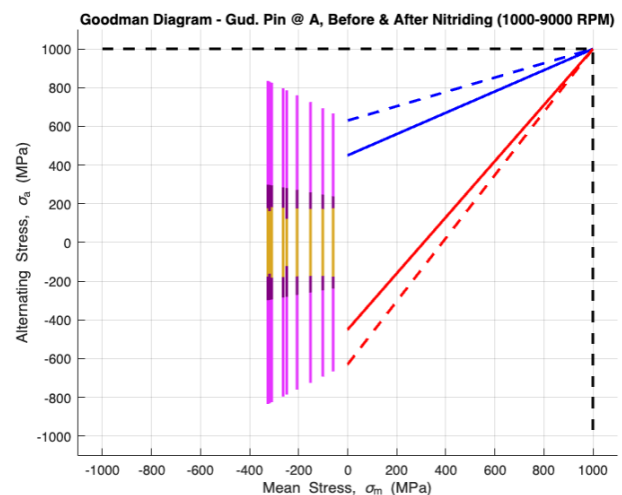


Figure-17: Nitrided Gudgeon Pin (Point A) Goodman Diagram

FSF Before and After Nitriding:

| RPM  | Max_Tensile_MPa | Min_Compressive_MPa | Mean_Stress_MPa | Stress_Range_MPa | FSF_Original | FSF_Nitrided |
|------|-----------------|---------------------|-----------------|------------------|--------------|--------------|
| 9000 | 295.29          | -413.27             | -58.99          | 354.28           | 0.67255      | 1.8831       |
| 8000 | 245.02          | -447.77             | -101.38         | 346.4            | 0.71539      | 2.0031       |
| 7000 | 194.75          | -499.09             | -152.17         | 346.92           | 0.74727      | 2.0923       |
| 6000 | 144.47          | -558.29             | -206.91         | 351.38           | 0.77282      | 2.1639       |
| 5000 | 94.201          | -621.35             | -263.57         | 357.78           | 0.79465      | 2.225        |
| 4000 | 56.095          | -674.99             | -309.45         | 365.54           | 0.806        | 2.2568       |
| 3000 | 27.801          | -678.34             | -325.27         | 353.07           | 0.84455      | 2.3647       |
| 2000 | 5.2176          | -642.56             | -318.67         | 323.89           | 0.91606      | 2.565        |
| 1000 | -3.8833         | -493.09             | -248.49         | 244.6            | 1.1484       | 3.2156       |

Figure-18: Gudgeon Pin at Point A, FSF Table.

### 3.4 – Torque Investigation.

Figure 19 shows a torque graph at 3000RPM illustrating the summed inertia torque contributions from individual cylinders, demonstrating the oscillatory nature of inertia forces in a four-cylinder engine.

As expected, peak inertia torque occurs near TDC and BDC, where the piston acceleration is highest. The total inertial torque exhibits periodic sinusoidal waveform, reinforcing the effects of the rotating mass.

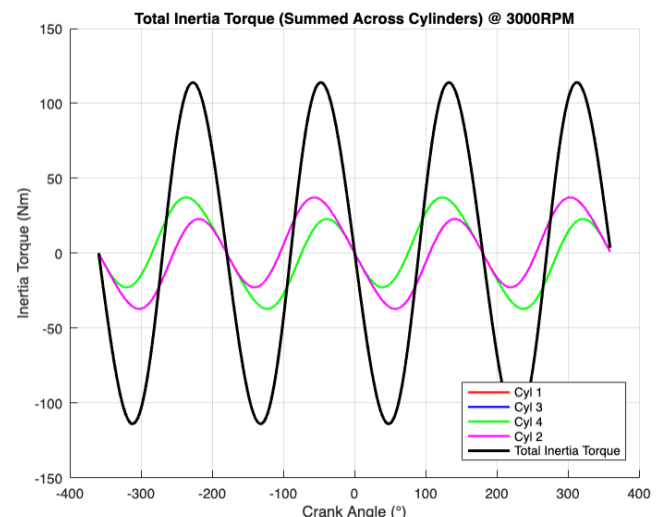


Figure-19: Total Inertia Torque.

Figure 20 illustrates the inertia torque contributions from each cylinder in a cross-plane crankshaft configuration at 6000RPM. The individual cylinder torques exhibit a sinusoidal nature, with each cylinder's contribution offset due to the 90° crankshaft phasing.

The total crank torque appears significantly smoother compared to the individual inertia torque peaks, demonstrating the effect of torque cancellation due to the 90-degree offset.

Cross-plane crankshafts benefit from "high torque at low RPMs *[and]* smooth running due to a constant crank rotation as a consequence of having *[a]* cylinder firing every 90°" (Sanchez, J. – 2020).

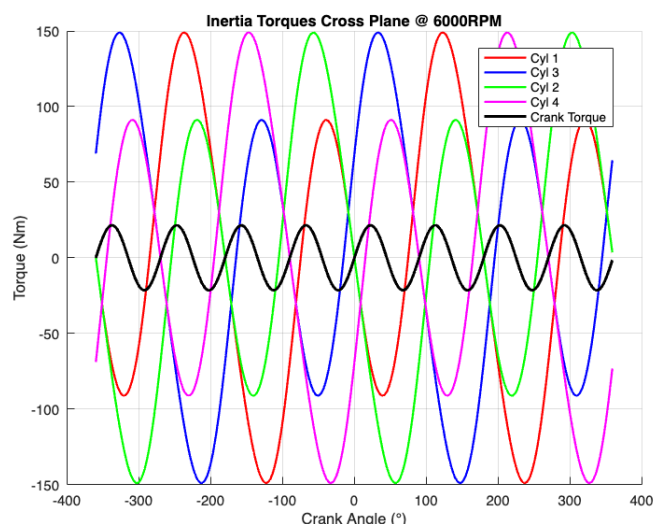


Figure-20: Cross-Plane Inertia Torque

This results in a more evenly distributed inertia torque profile, reducing peak-to-peak fluctuations and improved crankshaft rotational uniformity.

Figure 21 compares the inertia torque characteristics of the standard Sigma crankshaft and a cross-plane example. The standard crankshaft exhibits a larger amplitude of oscillations due to the simultaneous reciprocation of paired cylinders; leading to more pronounced fluctuations in inertia torque.

In contrast, the cross-plane crankshaft significantly reduces these fluctuations by offsetting the cylinder firing at  $90^\circ$  intervals; resulting in a smoother inertia torque curve with increased frequency of oscillations.

However, cross-plane crankshaft examples also require “heavier counterweights (additional rotational mass), [a] lower revving limit, [and] larger crankcase” (Sanchez, J. – 2020).

Figure 22 compares the gas torque characteristics of the standard Sigma crankshaft and the cross-plane example. The standard crank evenly spreads gas torque due to its uniform firing intervals, resulting in the lowest range of instantaneous torque variation.

In contrast, the cross-plane crankshaft displays higher and lower instantaneous torque fluctuations due to the spacing of ignition points. This leads to a greater variation in torque delivery, impacting power characteristics across the crankshaft rotation.

The Peak and Mean torque comparison highlights the variation in torque output across the engine speed range.

Peak Torque increases with RPM before dropping off around 5000RPM; likely due to limitations in volumetric efficiency, combustion characteristics, or increased frictional losses at higher engine speeds.

Mean Torque gradually increases with RPM before stabilising, again declining beyond 5000RPM. This aligns with expectations where engine torque is calculated as the average crank torque over a full cycle; highlighting the contrast between instantaneous torque peaks and the sustained, usable output.

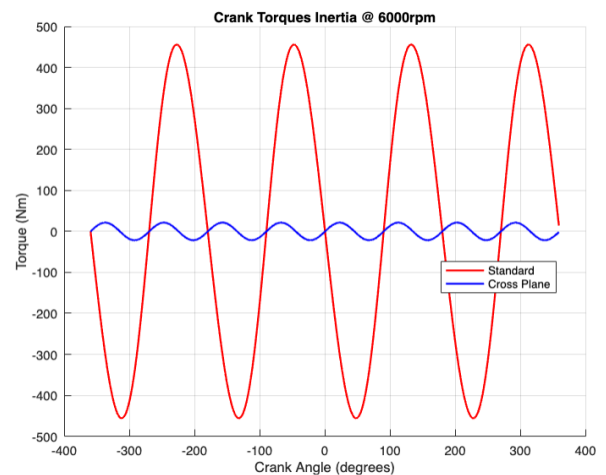


Figure-21: Inertia Torque Comparison

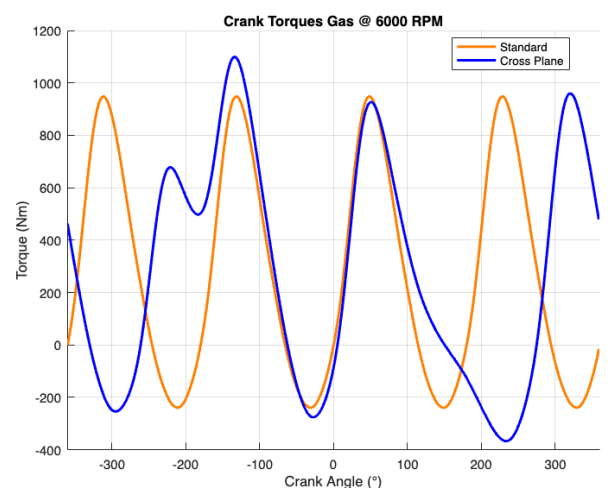


Figure-22: Gas Torque Comparison

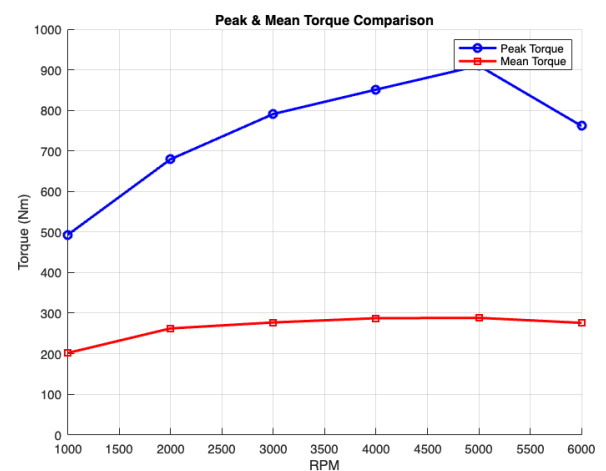


Figure-23: Peak & Mean Torque Comparison

### 3.5 – Big End Investigation.

Figure 24 illustrates how increasing engine speed amplifies the forces acting on the connecting rod due to inertia.

At lower RPMs, the forces remain relatively low and stable, but as RPM increased, the magnitude of rod-whip-force also increases; reaching extreme values at 6000RPM.

The cyclic nature of these forces align with the crank angle, peaking near TDC and BDC, where piston acceleration is at its maximum.

The Total Cap Force shows a similar trend, highlighting how the forces on the con-rod cap escalate with RPM. These forces are primarily influenced by reciprocating inertia, which contributes to the overall stress experienced by the bearing surfaces and fasteners.

At 6000RPM, peak forces exceeded 25,000N, emphasizing the necessity for high-strength fasteners and adequate cap design to ensure durability under cyclic loading.

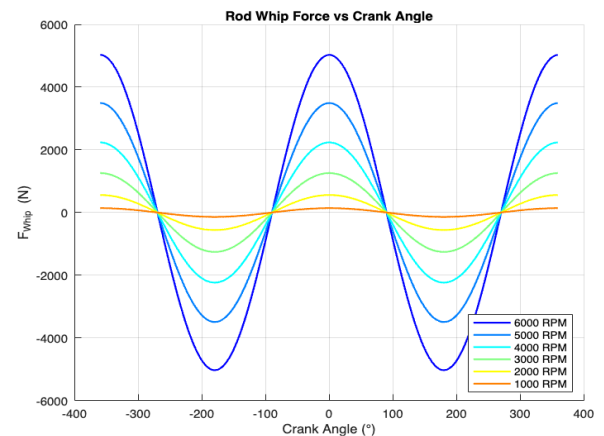


Figure-24: Rod Whip Force.

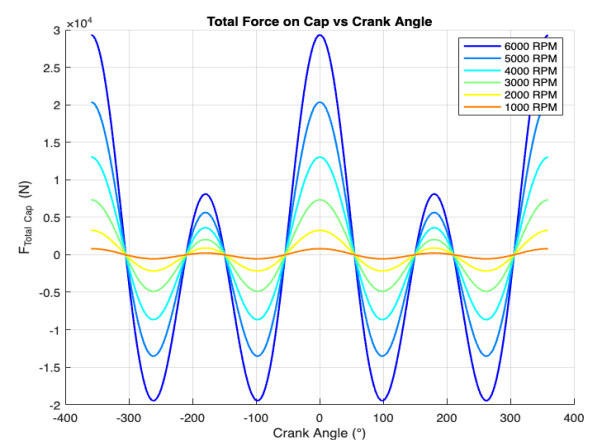


Figure-25: Total Force on Cap

An investigation into the effects of nitriding connecting-rod bolts resulted in a noticeable improvement in FSF across all tested RPMs. The standard and nitrided bolts (figure 26) FSF values were consistently higher, with values increasing 0.373 on average.

This increase is due to the nitriding process introducing a hardened outer layer, which enhances “hardness, temperature-resistance, and *[gives a reduced]* coefficient of friction”; to prevent excessive fatigue accumulation (Pankew, D. – 2020).

Fatigue Safety Factor (FSF) for Standard Bolt at Each RPM:

| RPM  | Max_Tensile_MPa | Min_Compressive_MPa | Mean_Stress_MPa | Stress_Range_MPa | FSF_Standard |
|------|-----------------|---------------------|-----------------|------------------|--------------|
| 6000 | 0.00034542      | 9.595e-06           | 0.00017751      | 0.00033582       | 2.4723       |
| 5000 | 0.00034527      | 9.5907e-06          | 0.00017743      | 0.00033567       | 2.4736       |
| 4000 | 0.00034481      | 9.5779e-06          | 0.00017719      | 0.00033523       | 2.4775       |
| 3000 | 0.00034404      | 9.5567e-06          | 0.0001768       | 0.00033448       | 2.4841       |
| 2000 | 0.00034297      | 9.5269e-06          | 0.00017625      | 0.00033344       | 2.4933       |
| 1000 | 0.00034159      | 9.4887e-06          | 0.00017554      | 0.00033211       | 2.5053       |

Fatigue Safety Factor (FSF) for Nitrided Bolt at Each RPM:

| RPM  | Max_Tensile_MPa | Min_Compressive_MPa | Mean_Stress_MPa | Stress_Range_MPa | FSF_Nitrided |
|------|-----------------|---------------------|-----------------|------------------|--------------|
| 6000 | 0.00034542      | 9.595e-06           | 0.00017751      | 0.00033582       | 2.8431       |
| 5000 | 0.00034527      | 9.5907e-06          | 0.00017743      | 0.00033567       | 2.8446       |
| 4000 | 0.00034481      | 9.5779e-06          | 0.00017719      | 0.00033523       | 2.8491       |
| 3000 | 0.00034404      | 9.5567e-06          | 0.0001768       | 0.00033448       | 2.8567       |
| 2000 | 0.00034297      | 9.5269e-06          | 0.00017625      | 0.00033344       | 2.8673       |
| 1000 | 0.00034159      | 9.4887e-06          | 0.00017554      | 0.00033211       | 2.8811       |

Figure-26: FSF Table Comparison, Standard vs Nitrided Con.-Rod Bolts.

#### 4. Conclusion

Nitriding significantly improved fatigue resistance, increasing FSF across both tests. The hardened surface reduced stress concentrations and enhanced durability without affecting overall stress levels. This confirms nitriding as an effective method for improving component longevity in high-load applications.

Crankshaft configuration plays a crucial role in torque delivery. Cross=plane cranks reduced peak-to-peak fluctuations through improving rotational smoothness but requiring additional counterweights. The standard crankshaft exhibited more pronounced oscillations but maintained a more uniform torque distribution.

Rod whip and cap forces increased significantly with RPM due to inertia effects. The cap experienced 25,000N, emphasising the need for high-strength materials and precision fastening. Ensuring proper fastener integrity and cap design is critical to sustaining durability under extreme cyclic loadings.

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