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**Calculation on Piston, Rod and Crankshaft
Assemblies**

Advanced Engine Design (M6X00410)

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1. Introduction

A comparative study will establish the Fatigue Safety Factors (FSF) Ford have engineered into the primary load bearing components of the standard EcoBoost GDI engine. The paper by (Afzal and Fatemi, 2004) explains that a High Cycle Fatigue (HCF) approach can be applied as the intended fatigue life for an engine is assumed to be infinite if the number of cycles exceeds 10^6 . The proposed safety factors will be determined using the Goodman technique, where the relationship between alternating stress and mean stress can be found.

I. Aims

- Carry out fatigue analysis on the Ford GDI gudgeon pin, connecting rod and crankshaft assembly.
- Determine critical points for analysis and explore the geometrically induced stress concentrations in each component.

II. Objectives

- Measure and model the relevant Ford GDI engine components.
- Use MATLAB to process numerical data and form suitable calculations for component stresses.
- Utilise the Goodman method to establish fatigue safety factors for each component
- Examine the interaction between the connecting rod cap and fastener before producing potential design iterations suitable for increased engine speeds.

0. Stress and Fatigue Methodology

I. Fatigue Analysis

As the target is to achieve an infinite fatigue life, it is necessary to choose the extreme net forces from the entire engine speed range. The paper by (Freudenthal, 1963) outlines the differences between using empirical safety factors and statistical reliability methods. In the specific case of an engine, cyclic variation and other factors can contribute to disparities in instantaneous forces which can prove detrimental to the fatigue life if the cumulative worst-case scenario is not considered with an appropriate safety factor. The following calculations will be input to a Goodman diagram for each

mean stress can be found using,

$$\sigma_{mean} = \frac{\sigma_{maxcom} + \sigma_{mincom}}{2}$$

The total range of stress is,

$$\sigma_{range} = \sigma_{maxcom} - \sigma_{mincom}$$

For endurance limit,

$$EL = 0.45 \times UTS$$

The allowable range is determined by,

$$\sigma_{all} = EL \left[1 - \frac{\sigma_{mean}}{UTS} \right]$$

Finally,

$$FSF = \frac{\sigma_{all}}{\sigma_{range}}$$

II. Axial Stress Concentration Factors

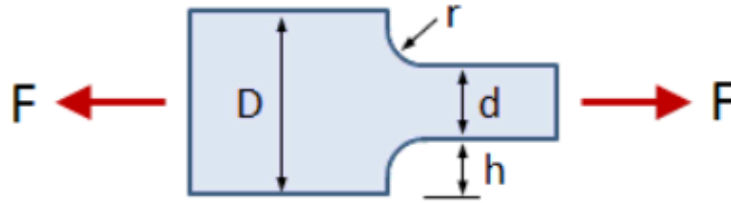


Figure 1 - Axial Stress Concentration Diagram (mechanicalc.com,2020)

Figure 1 will be used to define the geometries around any localised stress concentrations through this investigation. These concentrations are then calculated using Roark's formulas for Stress and Strain (Young and Budynas, 1980).

a) Stress Concentration Criterion

$$\frac{L}{D} > \frac{3}{\left(\frac{r}{D-2h}\right)^{0.25}}$$

b) Coefficients of Stress Concentrations for Additional Criterion

Table 1 - Stress Concentration Coefficients for Axial Shoulder

	$0.1 \leq h/r \leq 2.0$	$2.0 \leq h/r \leq 20.0$
C₁	$1.007 + 1.000 \sqrt{h/r} - 0.031 (h/r)$	$1.042 + 0.982 \sqrt{h/r} - 0.036 (h/r)$
C₂	$-0.114 - 0.585 \sqrt{h/r} + 0.314 (h/r)$	$-0.074 - 0.156 \sqrt{h/r} - 0.010 (h/r)$
C₃	$0.241 - 0.992 \sqrt{h/r} - 0.271 (h/r)$	$-3.418 + 1.220 \sqrt{h/r} - 0.005 (h/r)$
C₄	$-0.134 + 0.577 \sqrt{h/r} - 0.012 (h/r)$	$3.450 - 2.046 \sqrt{h/r} + 0.051 (h/r)$

c) Final Stress Concentration Calculation

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

2. Gudgeon Pin Study

I. Component Overview

Ford have utilized a Diamond-Like Carbon (DLC) coating to the outer surface of the steel gudgeon pin. (Kano, 2006) claims this nano coating can achieve a coefficient of friction lower than 0.01 under boundary lubrication, proven through a series of pin-on-disc and reciprocating sliding tests. The advantage for this material selection is twofold. Firstly, the Ford gudgeon pin is not an interference fit like most other applications, meaning it can freely rotate to distribute the loadings exerted through it. Secondly, This ultra-smooth surface has substantially less stress concentrations, subject to the work completed by (Bayoumi and Abdellatif, 1995) where a correlation between fatigue initiation, final separation life and the fatigue endurance limit of a component can be formed through polynomial equations to relate endurance stress to surface finishes.

In addition, the gudgeon pin has an internal taper to reduce reciprocating mass in areas where there are decreased stresses.

Ford have used this sophisticated technology on the Ecoboost GDI engine as a method for improving mechanical efficiency and to mitigate the additional stresses due to the increased gas forces from forced induction. (Sundararajan, Joshi and Rama Krishna, 2016) discuss the relevance of surface engineering in modern fuel saving methods.

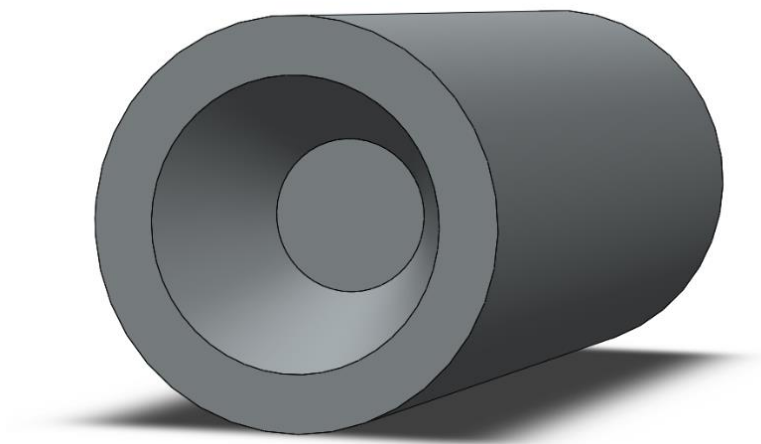


Figure 2 - Solidworks Gudgeon Pin

II. Gudgeon Pin Stress Analysis

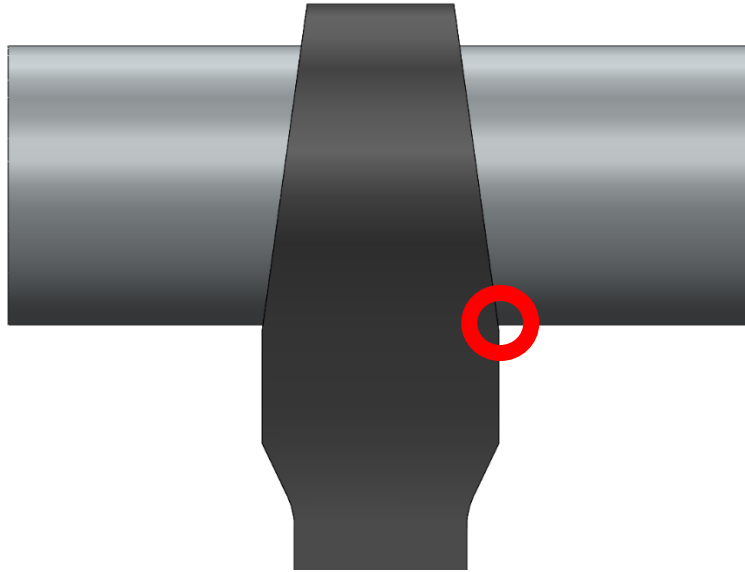


Figure 3 - Gudgeon Pin to Connecting Rod Interface

Figure 3 highlights the point of interest for this study. Solidworks was used to split the exposed gudgeon pin into a separate body to be measured. The applicable values are presented in Table 2.

Note: The tapered small end produces offset lengths for exposed gudgeon pin. This in turn alters the bending moments produced through an engine cycle. Racing applications reduce the length of the exposed gudgeon pin to further reduce the imposed forces.

Table 2 - Gudgeon Pin Geometry

Notation	Value	Description
m	0.02681Kg	Mass of exposed gudgeon pin
y	10.5mm	Distance to neutral axis from surface
D1	21mm	Outer Diameter
D2	12.5mm	Inner Diameter
L1	7.5mm	Distance (edge of Rod to centre of piston pin bore)
L2	6.7mm	Distance (edge of Rod to the centre of gravity position of exposed gudgeon pin)
Fnet	41000.78N @ 20° crank angle	Peak resultant force (gas force & piston inertia) @ 5700rpm

Since,

$$F = ma$$

Where F = Force, m = mass, a = acceleration

Thus,

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

Where F_{pi} = gudgeon pin inertia force(N), R = crank radius (m),

ω = angular velocity(rad/s), L = connecting rod length (m)

The equation above is used to calculate additional inertia forces imposed on the F_{net} value, due to the inclusion of the exposed gudgeon pin.

Since,

$$I = \frac{\pi}{64} \times (D1^4 - D2^4)$$

Where I = 2nd moment of area (mm⁴)

And,

$$M_{bmax} = \left(\frac{F_{net}}{2} \times L1 \right) - (F_{pi} \times L2)$$

Where M_{bmax} = maximum bending moment (Nmm)

Thus,

$$\sigma_{bmax} = \frac{M_{bmax} \times Y}{I}$$

Where σ_{bmax} = maximum bending stress (N/mm²)

The inclusion of shear stress is required to find the combined stress exerted through the gudgeon pin.

These values can be calculated from,

$$\tau_{max} = \frac{\frac{F_{net}}{2} - F_{pi}}{A}$$

Where τ = shear stress (N/mm²), A = cross section area (mm²)

And finally,

$$\sigma_{maxcom} = \sqrt{\sigma_{bmax}^2 + \tau_{max}^2}$$

Where σ_{maxcom} = combined stress (N/mm²)

The measured values from *Table 2* have been introduced into the previous equations. The results of which are compiled in *Table 3*. This forms the maximum stresses due to peak compressive forces occurring at 5700rpm.

Table 3 - Gudgeon Pin Maximum Stress Results

Notation	Value	Description
F_{pi}	455.79 N	Gudgeon pin inertia force
I	8348 mm ⁴	2 nd moment of area
M_{bmax}	153,752.93 Nmm	Maximum bending moment
σ_{bmax}	189.54 N/mm ²	Maximum bending stress
τ_{max}	89.62 N/mm ²	Maximum shear stress
σ_{maxcom}	260.88 N/mm ²	Maximum combined stress

The process is repeated, substituting the peak force from combustion with the peak inertia force: 5963.62 N, -358°crank angle @ 6000rpm. These results can be seen in *Table 4*.

Table 4 - Gudgeon Pin Minimum Stress Results

Notation	Value	Description
F_{pi}	561.03 N	Gudgeon pin inertia force
M_{bmin}	223,635.75 Nmm	Minimum bending moment
σ_{bmin}	23.40 N/mm ²	Minimum bending stress
τ_{min}	10.82 N/mm ²	Minimum shear stress
σ_{mincom}	-31.87 N/mm ²	Minimum combined stress

III. Gudgeon Pin Fatigue Analysis

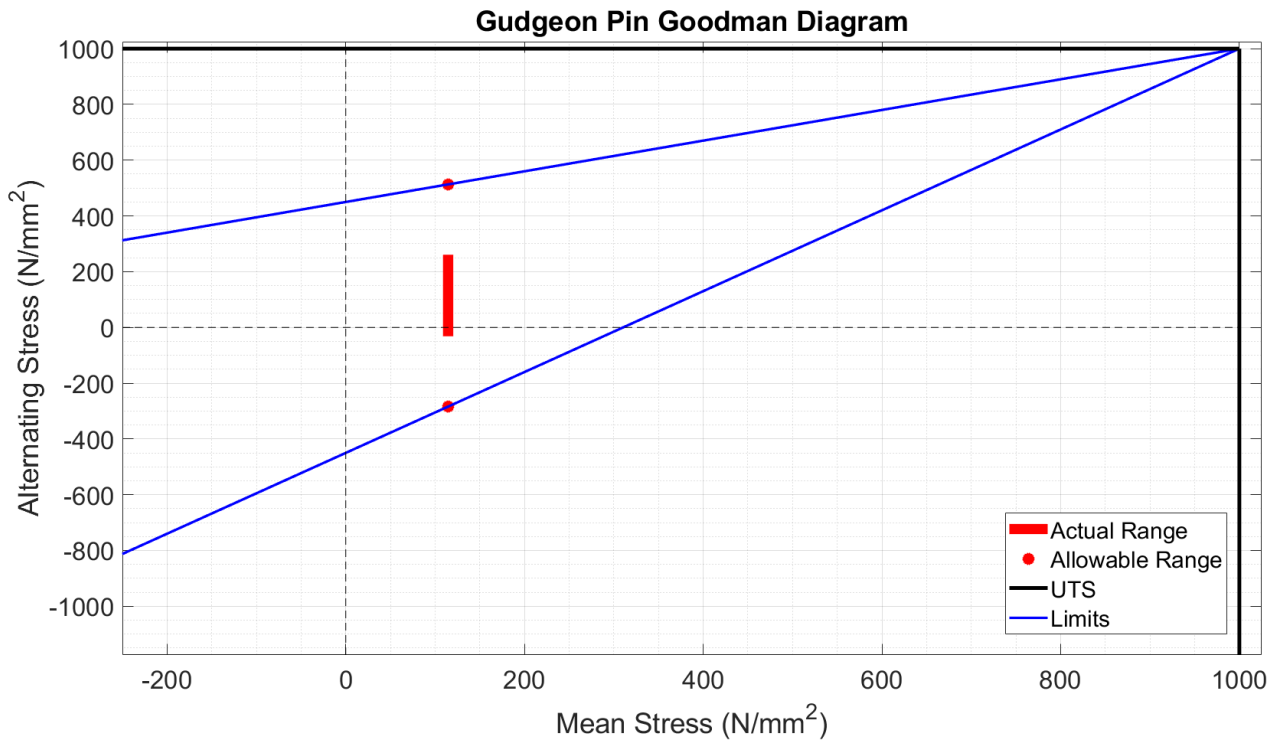


Figure 4 - Gudgeon pin Goodman Diagram (FSF 2.72)

The Goodman diagram is created from processing the stress results through the equations outlined in *Section 2.II*. The calculated Fatigue Safety Factor for the Ford gudgeon pin is 2.72, translating to a 172% increase over the Goodman failure criteria. This implies that Ford have attempted to eliminate the possibility of fatigue failure under typical engine parameters, with no additional stress concentrations due to enhanced surface engineering.

1. Connecting Rod Study

Figure 5 defines the planes through the Ford connecting rod to be explored through this study.

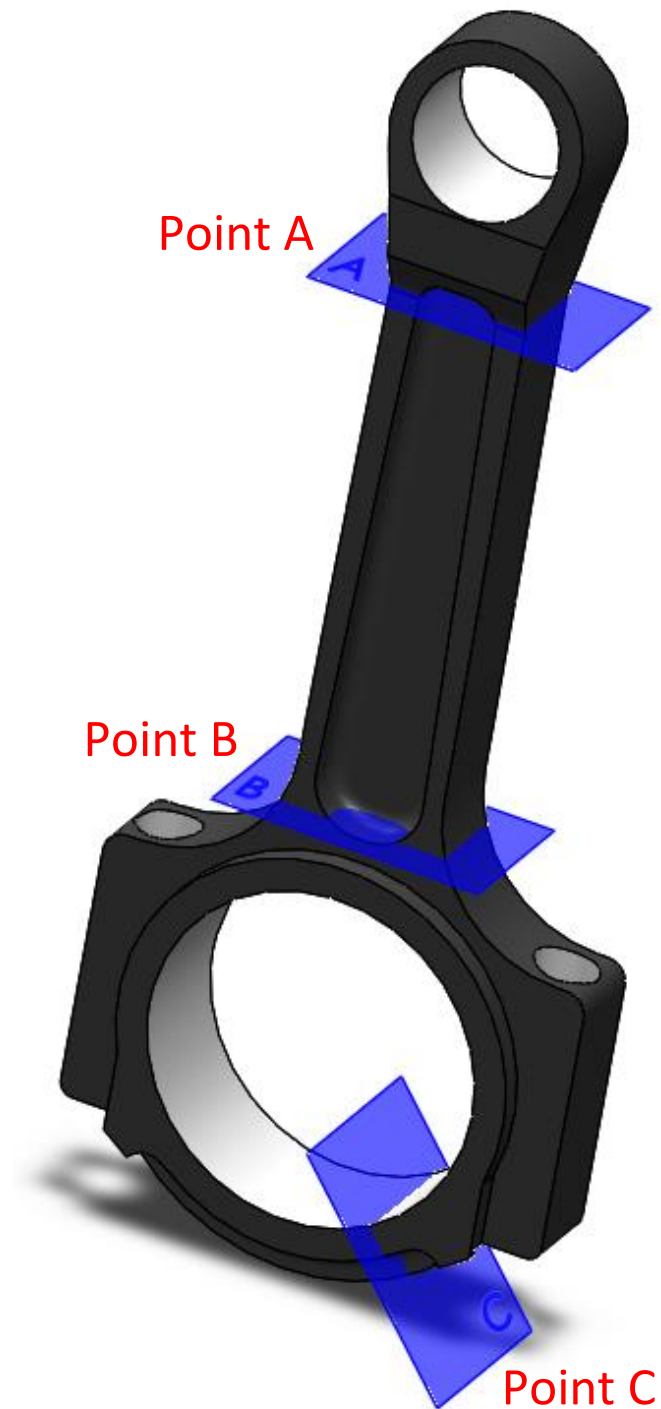


Figure 5 - Solidworks Connecting Rod Critical Points

I. Component Overview



	WHERE	WHEN	WHAT	WITH	HOW
①	Apex of Small End	Max. Inertia forces (no PCP) @ overspeed	Fatigue cracks Rupture Bore deformation	FEA Analysis (Haigh Diagram) Fatigue test Close In Test	Strap width Strap thickness Geometry
②	Bottom of Small End	Max. Compression load (=PCP-Inertia) @ max.torque, torque speed	Pin bore seizure Wear	Rod calc analysis EHL Analysis Dry contact analysis PIMO 3D Analysis	profile lube slots/ pockets pressure oil supply transition to I-Beam
③	I-Beam Minimum Cross Section	Max. Compression load (=PCP-Inertia) @ max.torque, torque speed	Buckling Bending Rupture	Advanced Buckling Calculation $M_{b,max}$ @ $x=1/\sqrt{3}$ Fatigue Test	Optimize X-Section with $I_{xx} : I_{yy} = 4:1$
④, ⑤	Transition Area of I-Beam to Big End	Max. Compression load (=PCP-Inertia) @ max.torque, torque speed	Fatigue cracks Rupture Bore deformation	Rod calc analysis Fatigue analysis (Haigh Diagram) Fatigue test	Smooth wide transition radius Stiffening rib Blind bolt holes
⑥	Fracture Split Joint Face Big End/Cap	Max. Inertia forces (no PCP) @ overspeed max. Compression load @ max. torque, torque speed	Bulging Gap opening Bore distortion	Rod calc Analysis FEA Analysis	Max moment of Inertia Position of bolt hole Clamp Load
⑦	Conrod Bolt	Max. Inertia forces (no PCP) @ overspeed max. Compression load @ max. torque, torque speed	Bolt fatigue Bolt loosening Clamp load reduction	Rod calc Analysis Torque Angle spec. Fatigue test	Thread engagement Bolt grade Rolled Threads Bolt Coating Pitch size
⑧	Conrod Cap	Max. Inertia forces (no PCP) @ overspeed	Fatigue cracks Rupture Bore deformation	FEA Analysis (Haigh Diagram) Fatigue test Close In Test	Strap width Strap thickness Stiffening Rib Geometry around bolt seat

Figure 6 - Connecting Rod Critical Points (Lapp et al., 2010)

The critical points of a typical automotive connecting rod are most poignant where an attempt to reduce mass or increase engine performance is made. The study by (Lapp *et al.*, 2010) suggests the lowest allowable fatigue safety factor in a connecting rod is considered to be 1.2. For this reason, it is expected that Ford have included a greater factor of safety due to the scale of production and range of anticipated operating conditions. *Figure 6* highlights the key areas of interest for engine designers.

a) Point A

This is a typically critical section of any connecting rod due to the transition in profile between the small end and the main shank. The plane through this point has an increased local stress through peak compressive and tensile conditions.

This stress flow is governed by the circumferential stresses due to the gudgeon pin acting through the pin bore and also the radius of the surface transition to the shank as depicted in **FIGURE** (Strozzi *et al.*, 2016).

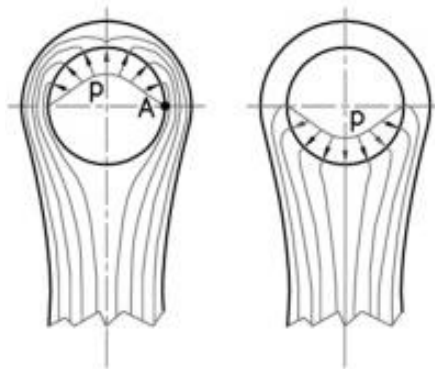


Figure 7 = Small End Stress Distribution (Strozzi *et al.*, 2016)



Figure 8 - Surface Crack Initiated Fatigue Failure (Strozzi *et al.*, 2016)

An example of fatigue-induced crack initiations acting from the surface of the “I Beam” section around this transition region can be observed in *Figure 8*.

Note: There is no requirement for any small end lubrication holes due to the DLC coated gudgeon pin, which also reduces the likelihood of inducing stress concentrations in the small end region of the Ford connecting rod.

b) Point B

The transition from the shank to the big end through the plane at *Point B* poses identical stress distributions to those mentioned in *Section 3.1.a*.

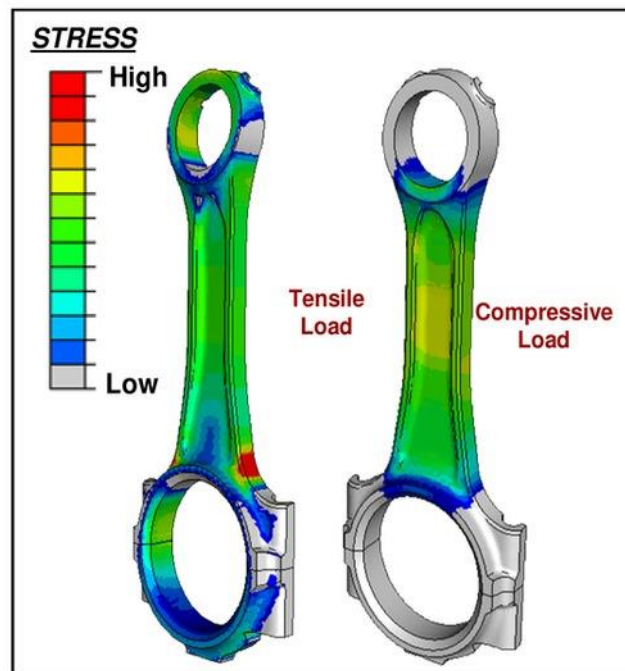


Figure 9 - Tensile Loading Big End Shoulder (Londhe, Yadav and Sen, 2009)

An investigation by (Londhe, Yadav and Sen, 2009) found that fatigue testing a connecting rod through high tensile loadings resulted in failure at the shoulder between the big end and main shank. A correlation was also drawn between surface roughness and fatigue life. The tested connecting rod in *Figure 9* represents a typical automotive application and is similar in design to the Ford GDI connecting rod.

c) Point C

The final plane to investigate is located across to one side of the big end bearing cap. This is significant to the study as there are only tensile loads being transferred in this region through each cycle, due to the crank pin absorbing all the compressive force through the piston assembly. Coupled with a smaller cross section, this plane will provide insight to how the connecting rod cap distributes the force with consideration to the potential stress raising fillet in the region.

The connecting rod cap is put into an amount of compression from the clamping force of the fastener. This is likely to affect the overall stress flow in the region around *point C* on the connecting rod as explained by (De Lima E Silva, 2010). An example of this clamping force is demonstrated in **FIGURE**.



Figure 10 - Connecting Rod Bolt to Cap Clamping Force (De Lima E Silva, 2010)

Note: For this study, any compressive forces through this section due to clamping will be neglected.

Also, an additional element of compression is suspected to act at TDC between the connecting rod cap and crank pin interface (through the shell bearing), this will also be neglected.

II. Connecting Rod Analysis

This section will use the same process as followed in the gudgeon pin study, with the addition of stress concentration factors detailed in *Section 2*. Solidworks was used to find the accurate cross section and correct reciprocating mass for each point as shown in *Figure 11*.

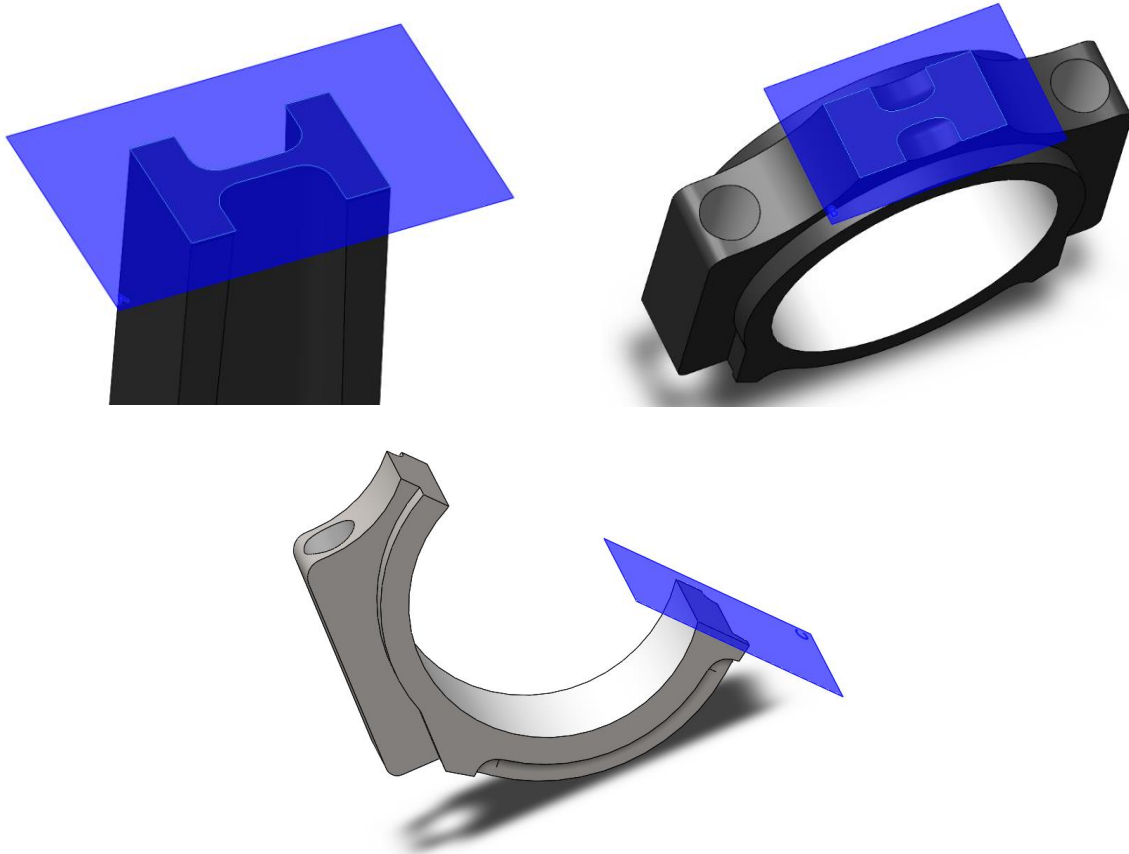


Figure 11 - Cross Sections at Each Plane

a) Stress at Point A

Table 5 – Connecting Rod Stress Results (Point A)

Notation	Value	Description
A	141.43 mm ²	Cross section area
m_{rec}	0.405 kg	Piston Assembly including rod mass above point A
$F_{inertiarec}$	8475.2 N	Reciprocating inertia force
F_{net}	38215.10 N	Resolved force from peak gas and inertia
σ_{min}	-270.21 N/mm ²	Minimum Stress at point A
σ_{max}	59.92 N/mm ²	Maximum Stress at point A

b) Fatigue at Point A

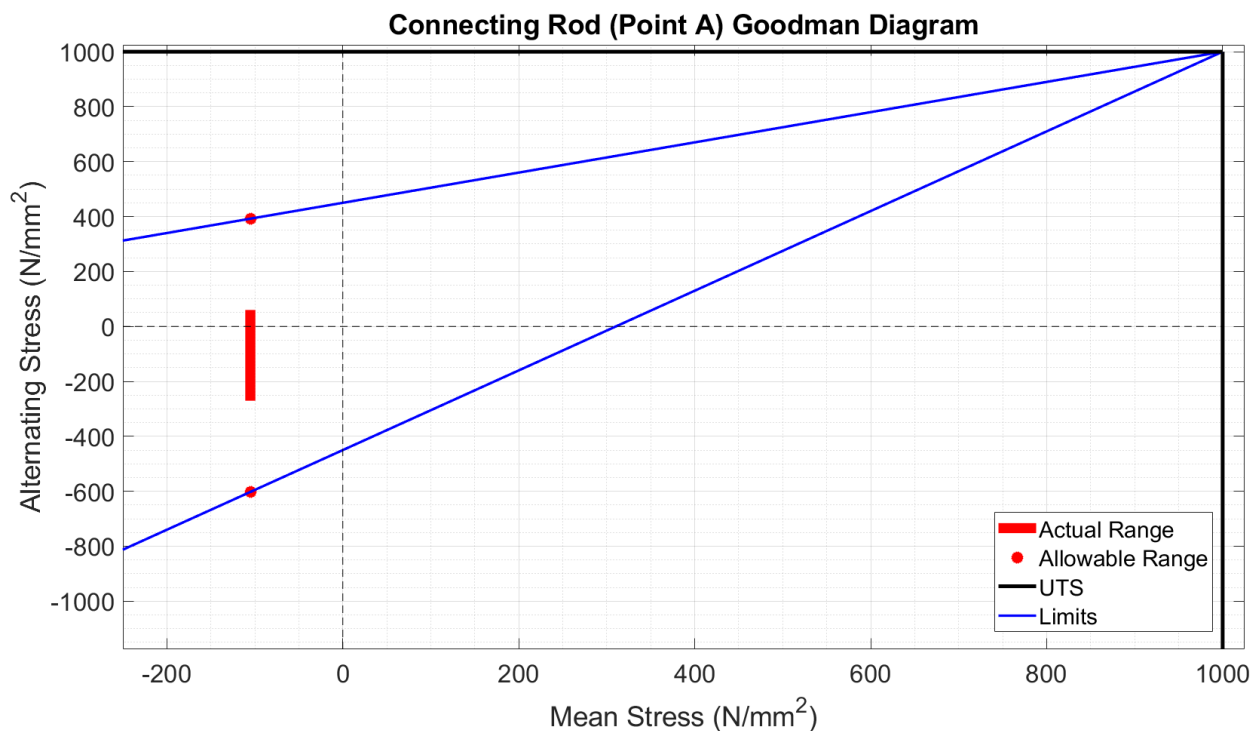


Figure 12 - Connecting Rod (Point A) Goodman Diagram

The FSF for this point of the connecting rod is calculated to be 3.01. This is in-line with the design philosophy of achieving over 10^6 cycles without failure. The calculations for the rod at *Point A* have considered a similar fatigue criterion to the gudgeon pin, where the two largest opposing forces through the engine speed range have been used.

Following the method detailed in *Section 2*, the data from *Table 6* concludes there is no stress concentration induced by the transition from the small end to the shank. The elimination of stress concentrations in this critical region is achieved during the design phase.

Table 6 - Stress Concentration Criteria (Point A)

Notation	Value	Description
<i>r</i>	55.5 mm	Radius of fillet (transition)
<i>D</i>	27.5 mm	Diameter (small end)
<i>d</i>	19.6 mm	Diameter (shank)
<i>h</i>	3.95 mm	(small end diameter – shank diameter) / 2
<i>h/r</i>	0.071	Ratio to define coefficients
<i>L</i>	24 mm	Length of small end

c) Stress at Point B

Table 7 - Connecting Rod Stress Results (Point B)

Notation	Value	Description
A	228.48 mm ²	Cross section area
m_{rec}	0.476 kg	Piston Assembly including rod mass above point B
$F_{inertiarec}$	9960.9 N	Reciprocating inertia force
F_{net}	36729.96 N	Resolved force from peak gas and inertia
σ_{min}	-160.76 N/mm ²	Minimum Stress at point B
σ_{max}	43.6 N/mm ²	Maximum Stress at point B

d) Fatigue at Point B

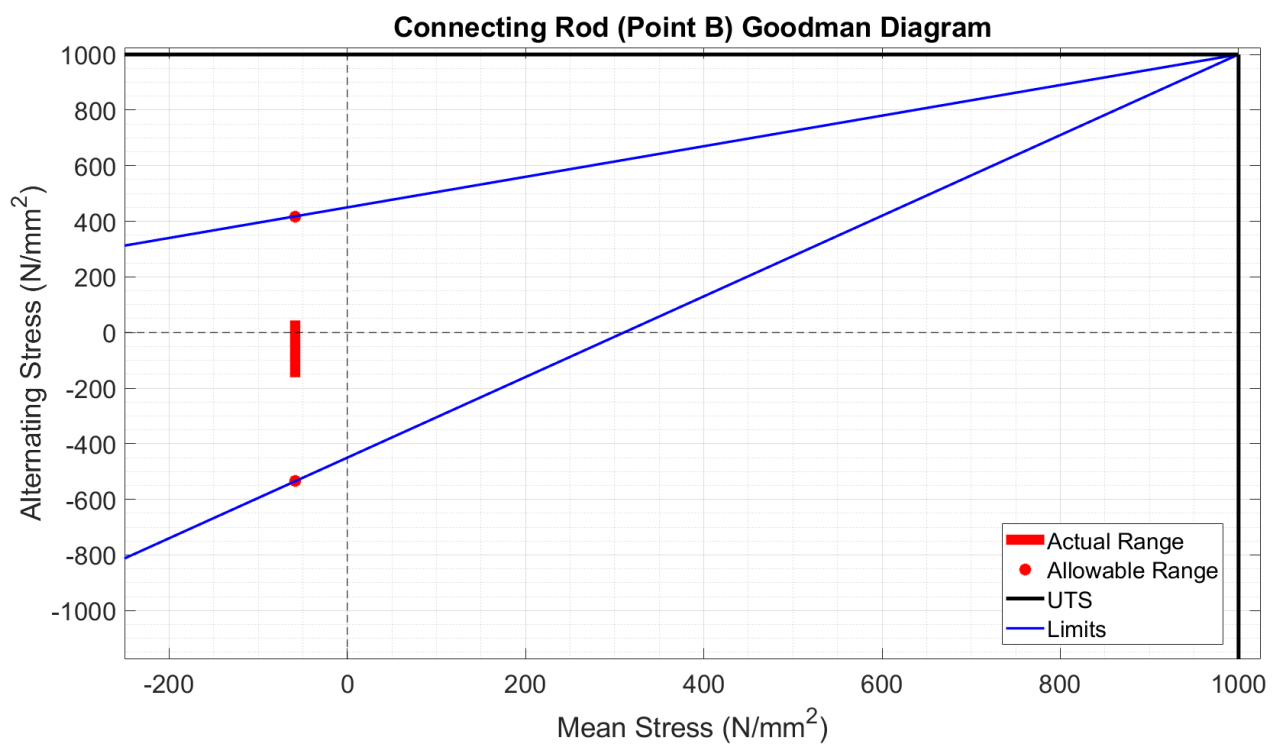


Figure 13 - Connecting Rod (Point B) Goodman Diagram

The results indicate a significantly higher Fatigue Safety Factor (4.6) than *Point A*.

However, contrary to *Point A*, the stress concentration criterion has now been met for the transition fillet from the big end to the main shank. Applying the values in *Table 8* to Roark's equations confirms this.

Table 8 - Stress Concentration Criteria (Point B)

Notation	Value	Description
<i>r</i>	45 mm	Radius of fillet (transition)
<i>D</i>	70 mm	Diameter (big end)
<i>d</i>	21 mm	Diameter (shank)
<i>h</i>	24.5 mm	(Big end diameter – shank diameter) / 2
<i>h/r</i>	0.54	Ratio to define coefficients
<i>L</i>	39 mm	Length of big end

This radius gives a stress concentration (k_t) of 1.14.

The updated Goodman diagram is shown in *Figure 14*, where the corrected endurance limits have been applied. This concludes an FSF of 4.1 for this region.

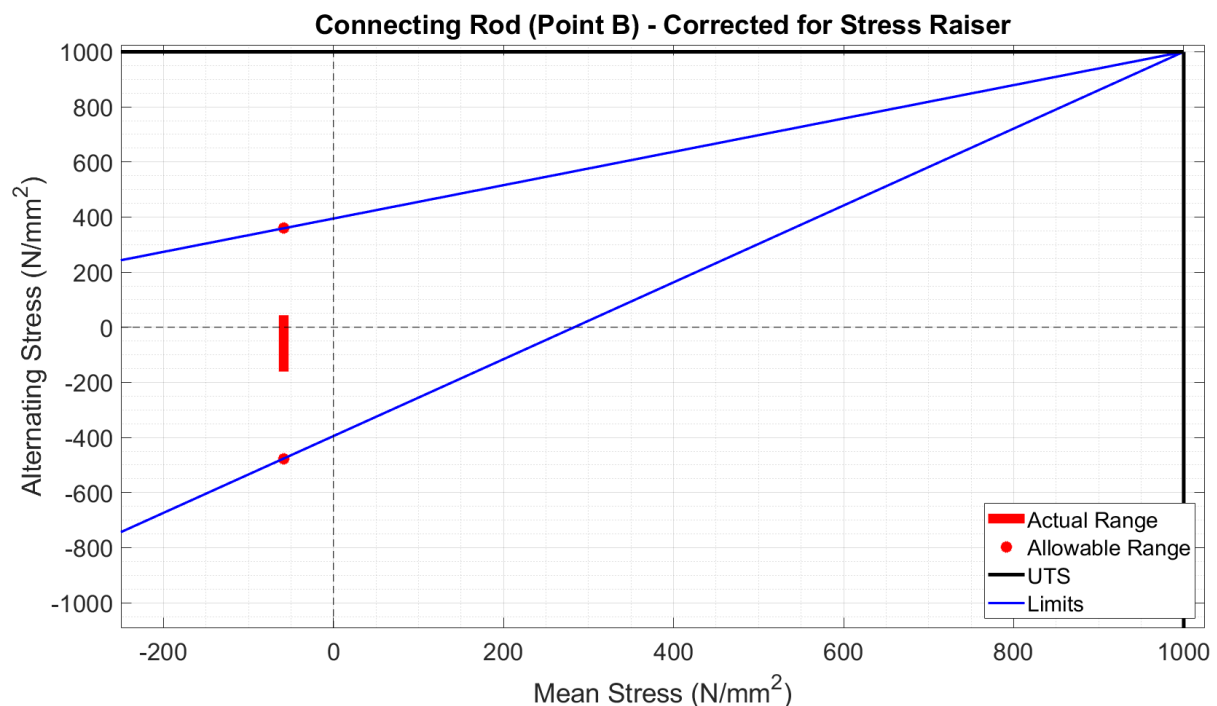


Figure 14 - Connecting Rod (Point B) Corrected for Stress Concentration

e) Stress at Point C

The addition of the inertia force due to rotation (Reif and Dietsche, 2014) is included for *Point C*. This element can be defined by,

$$F_{inertia\ rot} = M_{rot} \times R \times \omega^2$$

Table 9 - Connecting Rod Stress Results (Point C)

Notation	Value	Description
<i>A</i>	136.18 mm ²	Cross section area
<i>m_{rec}</i>	0.563 kg	Piston Assembly including full connecting rod
<i>m_{rot}</i>	0.145 kg	Two thirds of connecting rod
<i>F_{inertiarec}</i>	11782 N	Reciprocating inertia force
<i>F_{inertiarot}</i>	2329.8 N	Rotating inertia force
<i>σ_{min}</i>	0 N/mm ²	Minimum Stress at point C
<i>σ_{max}</i> <i>2</i>	51.81 N/mm ²	Maximum Stress at point C

Fatigue at Point C

The plane cutting through *Point C* is confirmed to have no significant stress raising features as *Table 10* shows.

Table 10 - Stress Concentration Criteria (Point C)

Notation	Value	Description
<i>r</i>	3 mm	Radius of fillet (transition)
<i>D</i>	9.35 mm	Diameter (Full cap)
<i>d</i>	1.68 mm	Diameter (Recess in cap)
<i>h</i>	24.5 mm	(Full cap diameter – recess diameter) / 2
<i>h/r</i>	0.54	Ratio to define coefficients
<i>L</i>	4.7 mm	Length of big end

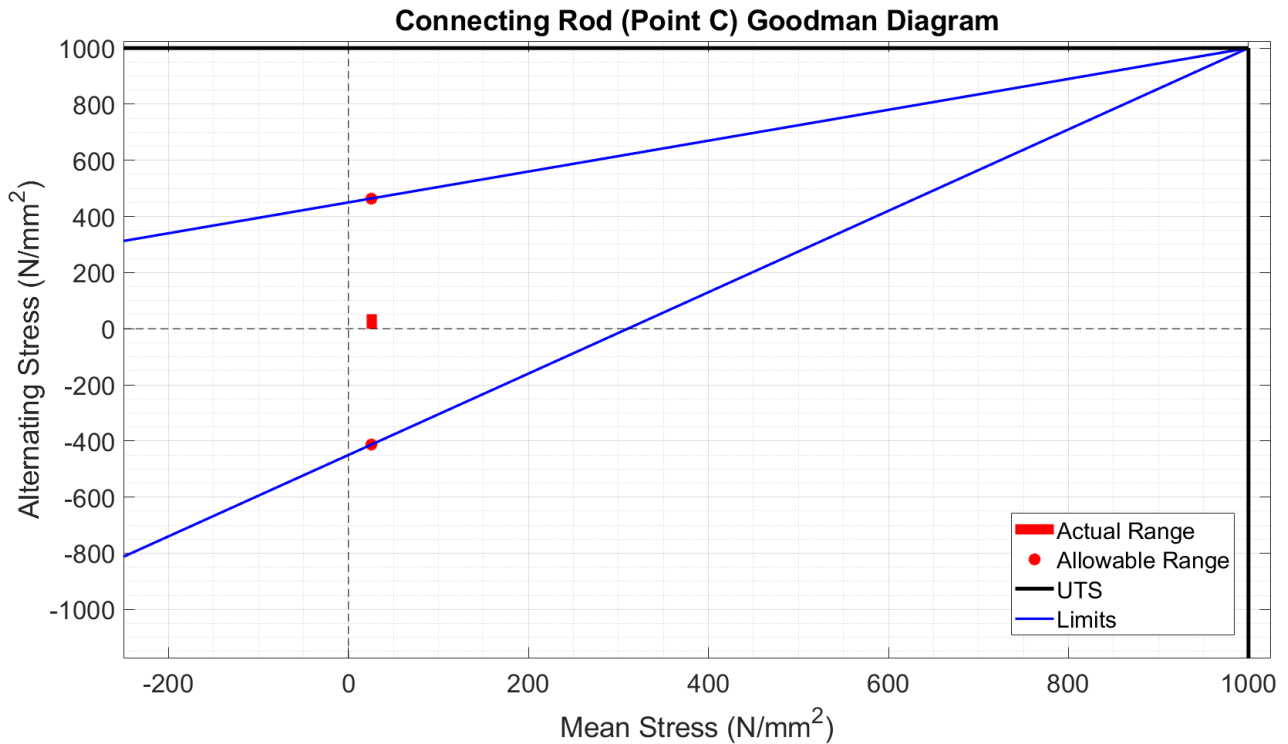


Figure 15 - Connecting Rod (Point C) Goodman Diagram

Figure 15 shows a large fatigue safety factor at *Point C*. The vacancy of any compressive forces and stress concentrations result in a fatigue safety factor of 16.9. This suggests the more critical portions of the connecting rod are at each end of the main shank.

Further critical points under real loading conditions would be found at the centre of the connecting rod cap and the top of the small end (Lapp *et al.*, 2010). These areas would experience peak tensile forces at TDC exhaust, with an element of compression at the cap, where the bearing face is forced against the crank pin.

III. Considerations to Finite Element Method

This connecting rod study would benefit from validation against an accurate FE model, where a broad analysis across all regions of the rod can be achieved. From this, the true fatigue safety factor, and the location of the weakest portion of the component can be found. The proposed FE model should also be correlated with real sample data as discussed by (Londhe, Yadav and Sen, 2009).

2. Connecting Rod Fastening Study

This section will investigate the interactions between the connecting rod, big end cap and assembly fastening bolts in the standard Ford GDI package. The tensile loads from the piston assembly act exclusively through the connecting rod bolts and this is exaggerated at increased engine speeds, presenting a substantial fatigue problem. The connecting rod bolts will displace if the clamping force at rod-to-cap interface is not great enough.

The displacement of the bolt can eventually cause an alternating stress outside of the allowable range for the material, as well as causing the bearing shells around the crank pin to wear, both are circumstances that often result in engine failure. To avoid these conditions, a suitable clamping force *must* be achieved and maintained through 10^6 cycles.

Papers by (Tsuzuku and Tsuchida, 1995), (Kaundabalaraman, 2017) and (Acri *et al.*, 2019) elaborate on the complexities around the design and manufacturing processes required for this vital part of the engine.

I. Comparison of Material Stiffness through Connecting Rod-to-Cap Interface

The individual bolt and cap stiffnesses must be found to accurately reverse engineer this application. from the customary stiffness ratio (Hooke's law),

$$K = \frac{F}{x}$$

Where k = material stiffness, F = force, x = displacement

And Youngs modulus,

$$E = \frac{\sigma}{\varepsilon}$$

Where E = Young's modulus, ε = strain

it is possible to combine the above formula's and rearrange the terms to provide an alternative method for calculating the stiffness of a material where displacement is unknown.

$$K = \frac{EA}{L}$$

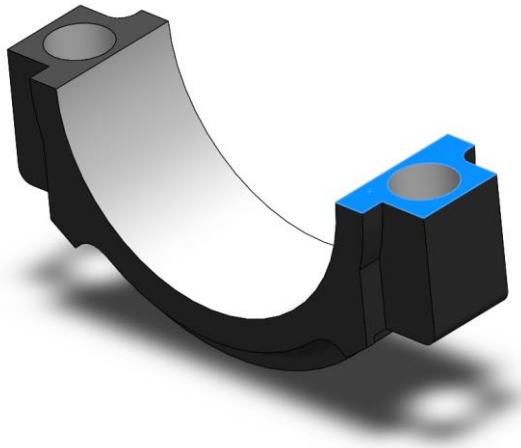


Figure 16 - Solidworks Connecting Rod Cap

The Solidworks model was used to calculate the cross-section areas required for the calculation.

Table 11 outlines the stiffness of the connecting rod bolts and cap, for the Ford GDI.

Table 11 - Bolt & Cap Stiffnesses

Variable	Rod-to-Cap Interface	Connecting Rod Bolt	Unit
Area (A)	128.74	42.43	mm^2
Youngs Modulus (E)	170×10^3	200×10^3	N/mm^2
Length (L)	20.58	20.58	mm
Resulting Stiffness (K)	1.063×10^6	412.34×10^3	N/mm

Note: The final combined stiffness equation eliminates the length variable, for this reason the length of the bolt has been made equal to that of the cap. Bolt shank area calculation from (Reif and Dietsche, 2014).

The resulting stiffness ratio between the cap interface and the bolt is found to be 2.58:1. The designer would have engineered this proportion based on the dynamic loadings at maximum engine speed, as detailed by (Pedersen and Pedersen, 2008).

a) Bolt Stiffness

The stiffness of the OEM bolt is determined by applying the previous formula with the inclusion of the actual bolt length, up to the point of the 2nd engaged thread (Finkelston and Wallace, 1980).

The new length (27mm) returns a value of $314.29 \times 10^3 \text{ N/mm}$.

II. Alternating Bolt Load

The cyclic loadings for the connecting rod bolt are found using the following equation, where the inertia forces from *Section 3* and previous stiffness values are used. An alternating force of

1248.4 N @ 6000rpm is found using,

$$F_{alternating} = \frac{F_{inertia} \times K_{bolt}}{(K_{bolt} + K_{cap})}$$

The displacement of the bolt is calculated by,

$$\Delta x = \frac{F_{alternating}}{(K_{bolt} + K_{cap})}$$

Table 12 - Connecting Rod Bolt Properties

Description	Value	Unit
Material	Steel	N/A
UTS	1100	N/mm²
Yield	990	N/mm²
Youngs Modulus (<i>E</i>)	200	N/mm²
Length	27	mm
Shank area	42.43	mm²
Stiffness	308	GPa
Axial Stress	594	N/mm²
Axial Force	25203.42	N/mm²
Additional bolt force	1248.4	N
Displacement	0.000910576	mm

III. Bolt Tightening Torque

It is common practice for the connecting rod bolts to be termed as single use components. This is due to the measures taken to avoid such failure mechanisms as fatigue and self-loosening, as described by (Hall, Lapp and Krause, 2010) and (Eccles, 1993).

Plastic Region Tightening is used to tighten the bolt beyond the tensile yield point, to increase deformation control (Okubo, Tanaka and Goromaru, 1984).

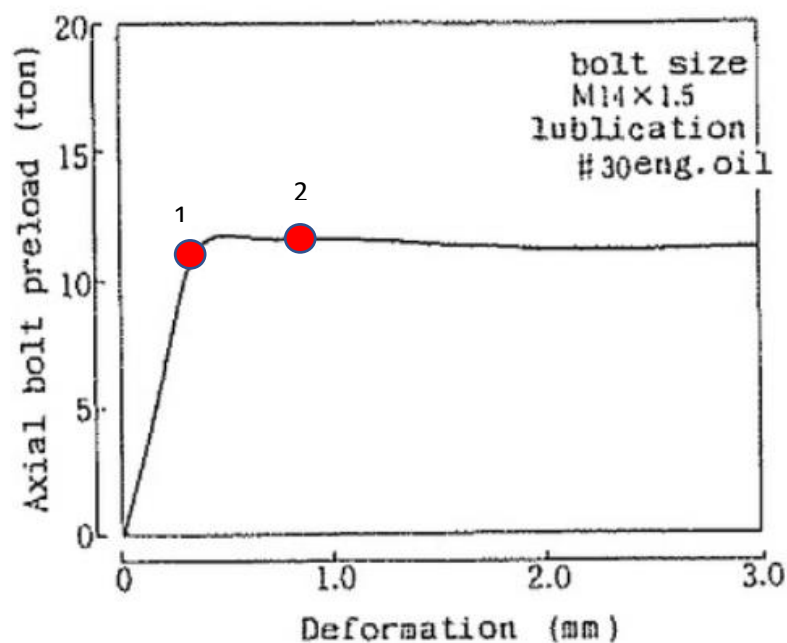


Figure 17 - Plastic Region Tightening (Okubo, Tanaka and Goromaru, 1984)

Figure 17 illustrates a typical tightening curve, where the force is seen to remain stable at a portion after the yield point.

This level of tightening is typically achieved in separate stages. It is first necessary to approach the yield point of the bolt, before turning through an empirically calculated angle (where the bolt is stretched), arriving at a suitable point in the plastic region.

The following method will reach the preload tightening torque (90% yield point).

For tightening torque,

$$T = \mu \times F_{axial} \times r$$

Where μ = Friction, r = bolt pitch radius

Table 13 displays the variables for the bolt head and threads, used to calculate the friction model (separate contacting surfaces).

Table 13 - Related Bolt Properties

Related Bolt Properties		
Description	Value	Unit
Head diameter	12.8	m
Thread size	M8 x 1	ISO Metric (fine)
Thread pitch diameter	7.35	m
Shank radius	3.675	m
Shank area	42.43	m
Bolt head radius	5.038	m

Table 14 - Bolt Tightening Friction Model

Bolt Tightening Friction Model (90% Yield)		
Description	Value	Unit
Axial stress (60% of 90% Yield)	534.6	N/mm ²
Bolt Head		
Axial Force (N)	22682.6	N
Friction Torque (Nm)	13.7	Nm
Bolt Threads		
Axial Force (N)	22682.6	N
Friction Torque (Nm)	15	Nm

A total tightening torque of 28.7Nm is obtained from the friction model in Table 14.

IV. Bolt Fatigue Analysis

This section will determine the engineered FSF for the connecting rod bolt. The inclusion of suitable stress concentration factors will improve the accuracy of the calculations.

a) Stress Concentration Factors

The endurance limit for a component will be influenced by local stress rising features as previously reviewed. In the case of the connecting rod bolts, the only significant stress concentration is the continuous groove in the thread. This particular type of stress raiser is defined by (O'Hara, 1989) as being a ratio of maximum fillet stress normalised to shear transfer rate. Also, the sensitivity to friction is discussed.

This study will adopt a simplified approach, where the boundary conditions are set to only see uniform axial tensile forces through two opposing notches on a rectangular section.

The M8 x 1 thread dimensions (FastenerData, 2020), are included to the most suitable stress concentration factor in Roark's methods in *Figure 18* (Young and Budynas, 1980).

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

where

	$0.25 \leq h/r \leq 2.0$	$2.0 \leq h/r \leq 50.0$
C_1	$0.723 + 2.845\sqrt{h/r} - 0.504h/r$	$0.833 + 2.069\sqrt{h/r} - 0.009h/r$
C_2	$-1.836 - 5.746\sqrt{h/r} + 1.314h/r$	$0.024 - 5.383\sqrt{h/r} + 0.126h/r$
C_3	$7.254 - 1.885\sqrt{h/r} + 1.646h/r$	$-0.856 + 6.460\sqrt{h/r} - 0.199h/r$
C_4	$-5.140 + 4.785\sqrt{h/r} - 2.456h/r$	$0.999 - 3.146\sqrt{h/r} + 0.082h/r$

For the semicircular notch ($h/r = 1$)

$$K_t = 3.065 - 6.269 \left(\frac{2h}{D} \right) + 7.015 \left(\frac{2h}{D} \right)^2 - 2.812 \left(\frac{2h}{D} \right)^3$$

Figure 18 - Stress Concentration Criterion for U Notches (Young and Budynas, 1980)

The stress concentration being introduced by the threads using this technique is 4.2.

Note: A further attempt was made to refine the stress concentration method by substituting the U notch factor into the V notch conditions. However, the $\frac{2h}{D}$ criterion was not met.

b) OEM Bolt Fatigue Safety Factor

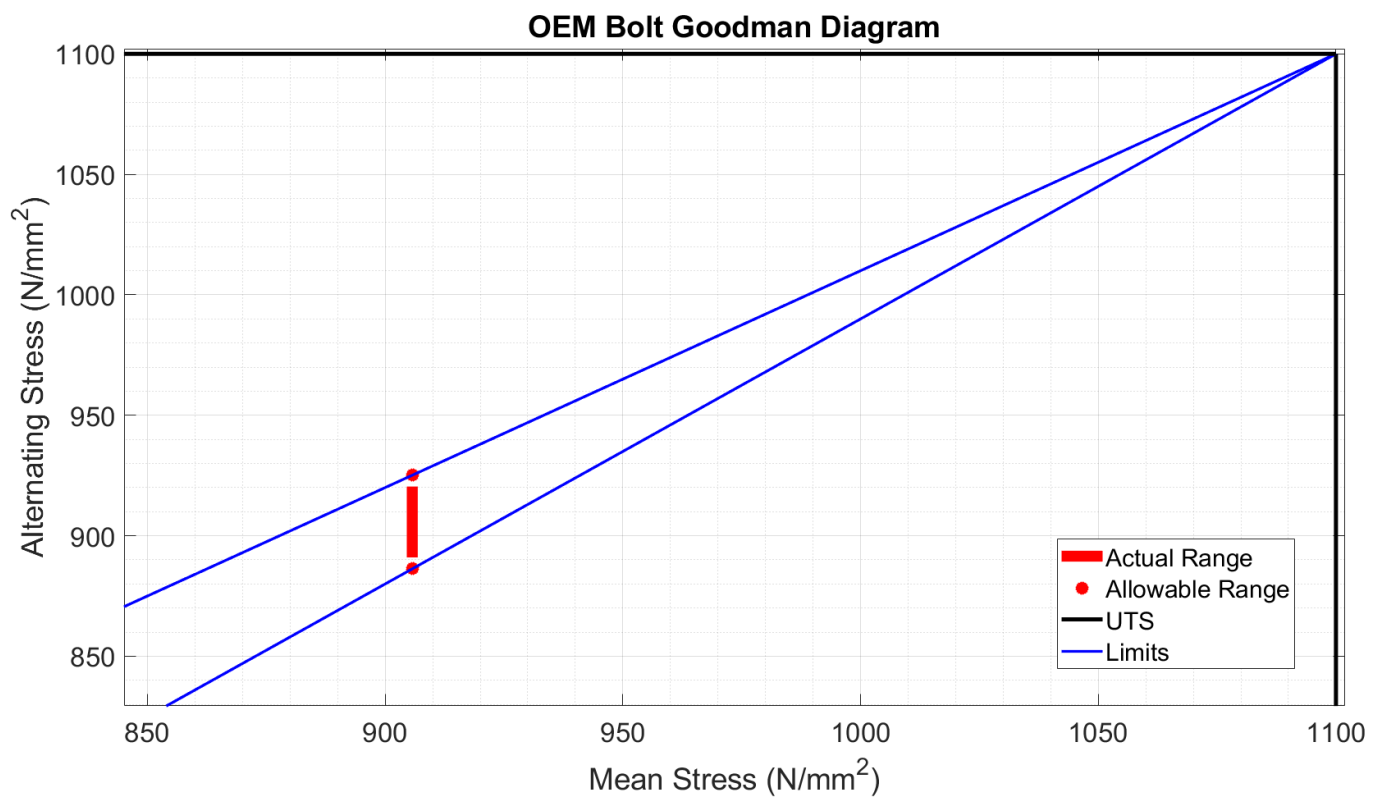


Figure 19 - OEM Bolt Goodman Diagram

The calculated FSF using the Goodman method is 1.32. The designer would have considered the maximum inertia forces at peak engine speed to protect from a worst-case scenario. As the Ford GDI is engineered for a standard road vehicle, the normal operating conditions for its intended drive cycle will cause significantly less stress. The range of stress the bolt experience is also far smaller than previously tested components.

In addition, the trade-off between resistance to fatigue at high speed and mass reduction for improved mechanical efficiency would have been factored in to the final product (Lapp *et al.*, 2010).

Note: the actual fatigue safety factor is likely to be larger than calculated with the addition of engine friction, improved stress concentration calculations and possibly latent cylinder pressures. However, this study provides a good comparison between components.

V. Considerations for Bolt Fatigue in a Modified Engine

An additional study has been carried out to suggest possible design considerations required to increase the engine speed to 9000rpm. Earlier work completed by (Jones, 2019) will be considered for this case.

a) OEM Bolt in Modified Engine

The first phase in this process is to repeat the steps in *Section 4.IV* to find the corrected FSF for the OEM bolt in the modified engine.

Table 15 - Comparison of Inertia Forces Acting on Bolt

Variable	Original Engine	Modified Engine	Percentage Difference
Engine Speed	6000rpm	9000rpm	50 %
Peak Net Tensile Force	5963.62 N	14525.61	144 %
Reciprocating Inertia Force	11782 N	26528.55 N	125 %
Rotational Inertia Force	2329.8 N	5242 N	
Combined Inertia Force	14111.8 N	31770.55 N	
Inertia Force per side	7055.9 N	15885.28 N	
Alternating Bolt Force	1248.4 N	3624.93 N	190%

The OEM bolt sees a 190% increase in additional inertia force as shown in *Table 15*. A Goodman analysis reflects this, with a fatigue safety factor of *0.39*. This would evidently cause failure through fatigue.

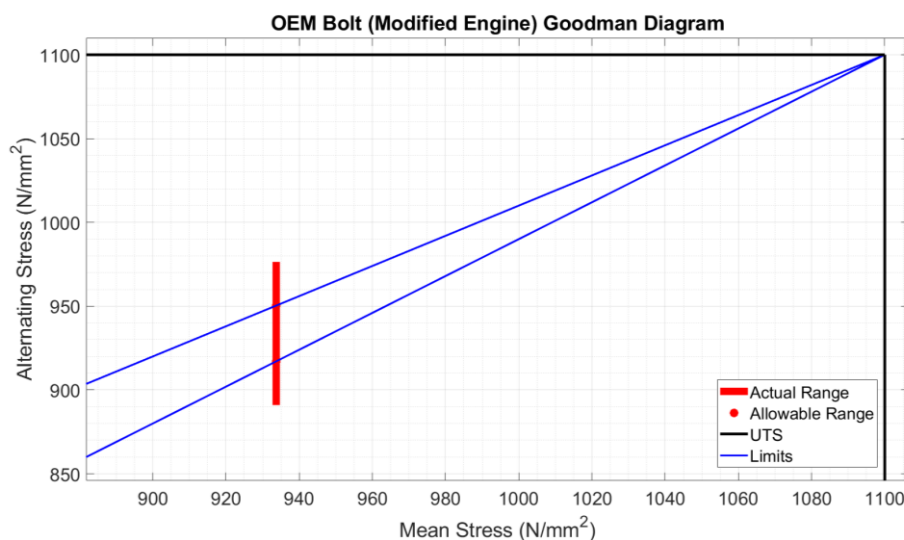


Figure 20 - OEM Bolt (Modified Engine) Goodman Diagram

b) Redesigned Bolt for Modified Engine – Waisted Shank

According to previous calculations, the logical method for reducing the additional forces is to reduce the bolt stiffness. This would subsequently increase the bolt/cap stiffness ratio.

One method in achieving this, whilst maintaining the same thread size (to retain the OEM connecting rod) is to reduce the diameter of the shank. This type of bolt is referred to as having a “waisted shank”.

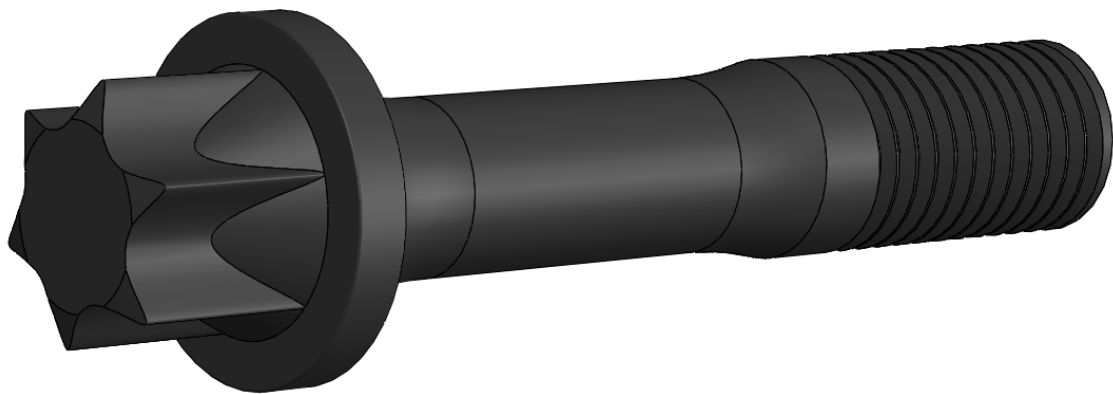


Figure 21 - Modified Bolt - Waisted Shank

This design follows the same stiffness principle as a variable spring, where each section of the bolt can be considered as having a unique stiffness value. The combination of these stiffnesses can be numerically combined by,

$$K_{total} = \frac{1}{1/K_1 + 1/K_2 + 1/K_3}$$

Where each subscript of K signifies the localised stiffness

Note: This technique produces a total stiffness that is lower than the most relaxed component.

The new stiffness for the modified bolt is concluded to be $236.83 \times 10^3 \text{ N/mm}$.

The previous calculations are updated with the new stiffness value, yielding a 25% (730.67 N) reduction in additional bolt force over the OEM bolt.

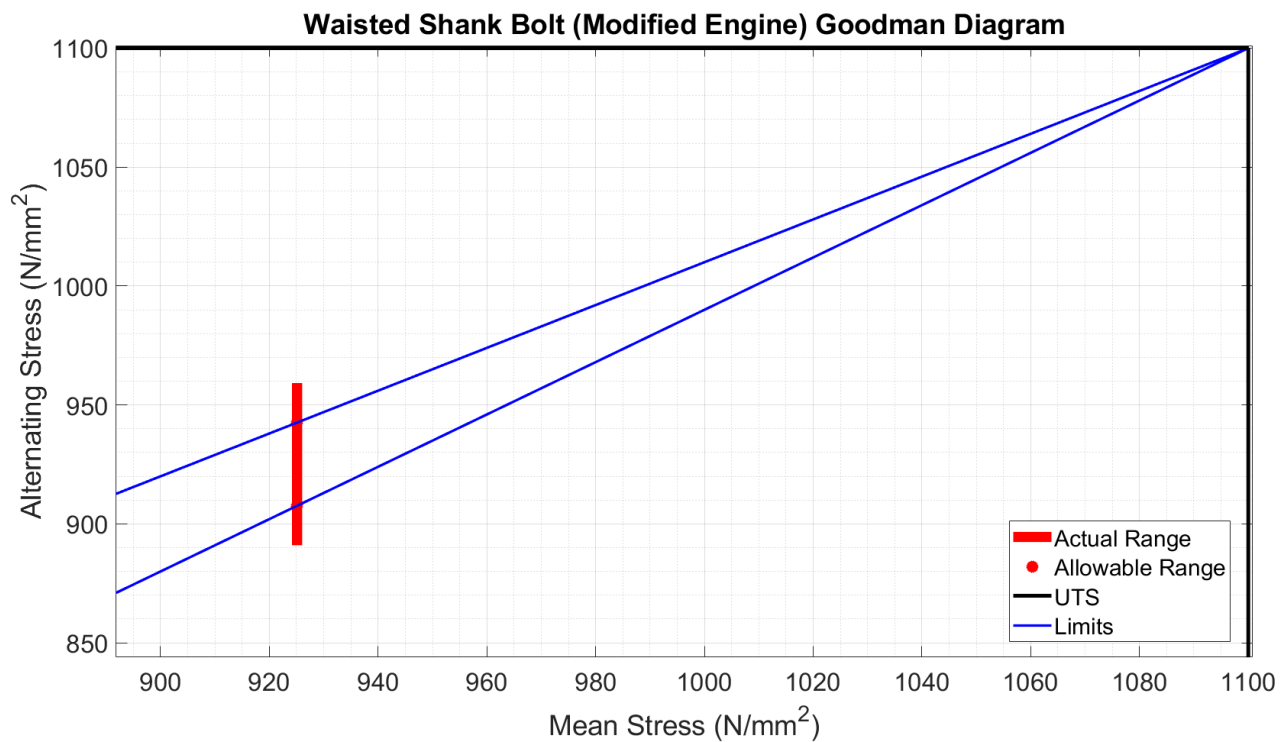


Figure 22 - Waisted Shank Bolt (Modified Engine) Goodman Diagram

Note: The stress concentration factor has been carried over from the original bolt as the modified bolt shares the same thread type. The shank itself is expected to influence a rise in stress, but this is neglected due to the magnitude of the factor introduced by the thread.

The Goodman model reveals an FSF of 0.51, a 30% improvement to that of the OEM bolt in the modified engine. This proves how effective the stiffness ratio is.

A shank optimisation study could be carried out at this point, to reduce the stiffness as far as possible with close consideration to the induced stress concentrations. Following this iterative process could restore the original fatigue safety factor but alternative methods will be considered for the remaining study.

c) Redesigned Bolt for Modified Engine – Required Mass Reduction for Waisted Shank Design

As this is a production engine, Ford have compromised the mechanical efficiency with the costs incurred from development and manufacturing. The numerical model of the piston assembly can be easily manipulated to find the required mass reduction to achieve the original FSF with the waisted shank bolt.

The original reciprocating mass is 563 g. The target mass for the reciprocating components was calculated to be 200 g in order to achieve the original safety factor of 1.3. This is clearly an unrealistic solution but does demonstrate the influence that reciprocating component mass has on the bolt (due to inertia).

d) Redesigned Bolt for Modified Engine – Change of Material for Waisted Shank Design

This section will compare a range of suitable uprated materials, commonly used in high temperature and stress applications (ARP, 2020).

The Goodman analysis will be carried out on the waisted shank bolt for each of the materials presented in *Table 16*.

Table 16 - Comparison of Material Properties

Material	UTS	% UTS to Yield	Description
AMS5844	1931	90%	High cobalt content, superior strength, used in Formula 1 and NASCAR
L19 Steel	1793	85%	Premium steel, very high strength, easily contaminated and not corrosion resistant
Inconel 718	1586	90%	Nickel based, high temperature, super-alloy
8740 Chrome Moly	1379	90%	Steel alloy, moderate strength, requires thread rolling and heat treatment for high tensile use

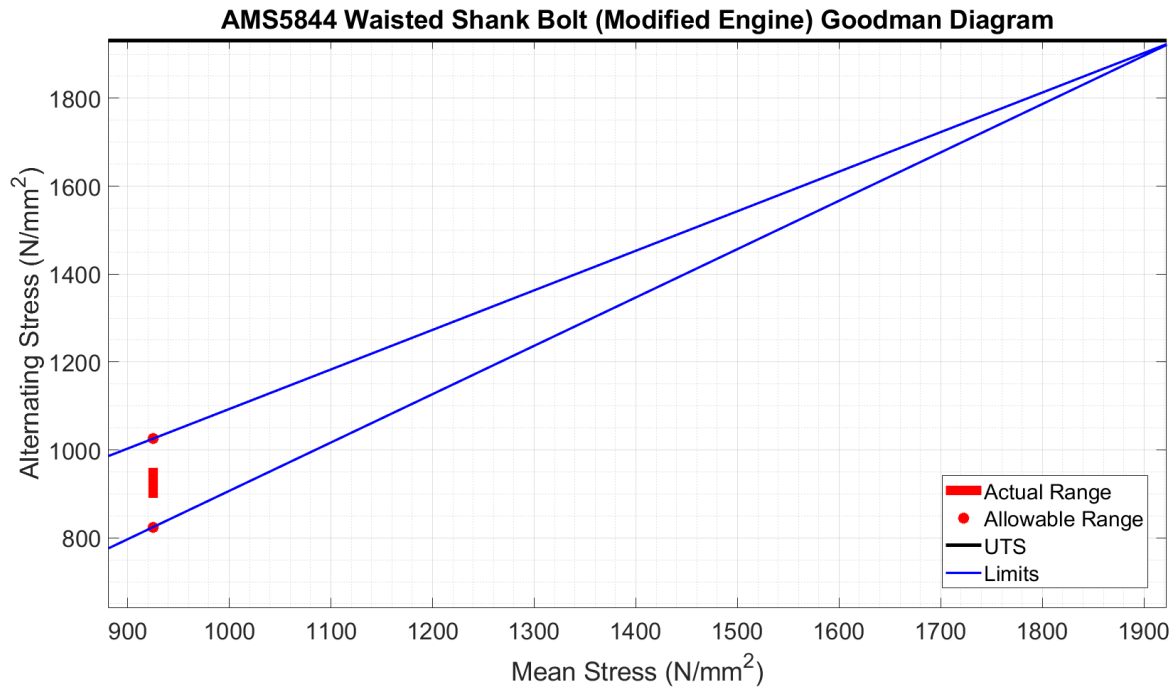


Figure 23 - AMS5844 Waisted Shank Bolt (Modified Engine) Goodman Diagram

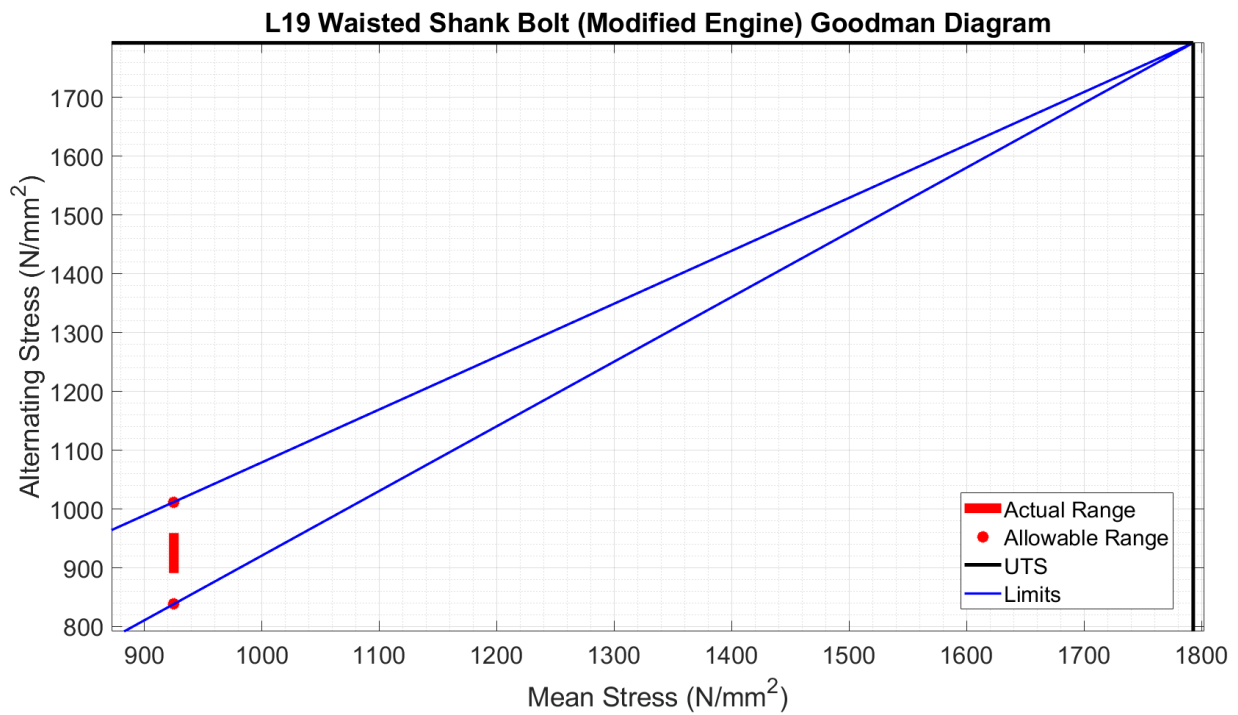


Figure 24- L19 Waisted Shank Bolt (Modified Engine) Goodman Diagram

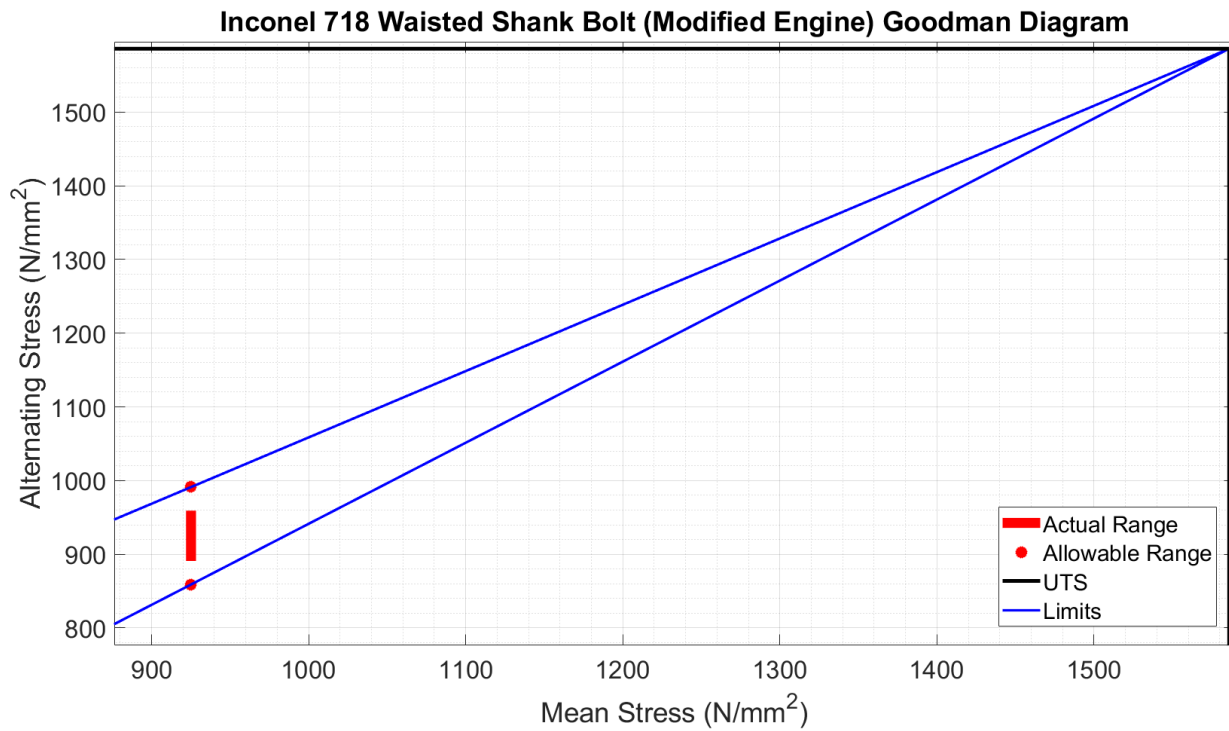


Figure 26 - Inconel 718 Waisted Shank Bolt (Modified Engine) Goodman Diagram

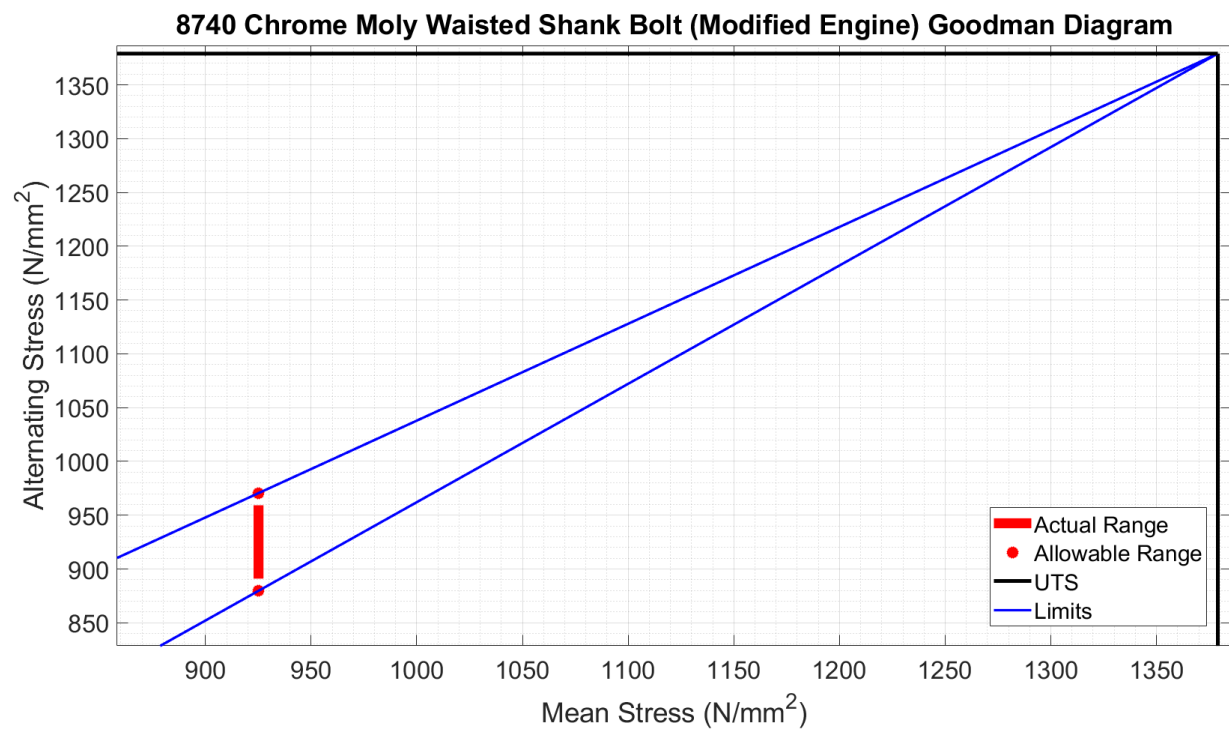


Figure 25 - 8740 Chrome Moly Waisted Shank Bolt (Modified Engine) Goodman Diagram

Table 17 reveals the fatigue safety factors drawn from the material selection. The most practical solution for the Ford Ecoboost engine is the 8740 Chrome Moly, owing to the identical FSF when compared with the OEM bolt. The more exotic materials from the selection would result in significantly overengineering the component for the operating conditions which is unjustified given the standard FSF is proven to reach infinite cycles.

Table 17 - Comparison of FSF in Different Materials

Material	Fatigue Safety Factor
AMS5844	2.95
L19	2.54
Inconel 718	1.94
8740 Chrome Moly	1.33

VI. Bolt Design Summary

This bolt analysis has demonstrated how changing the geometric and material properties can alter the fatigue characteristics of a threaded fastener.

The most effective method for enhancing the resistance to fatigue is proven to be through material selection. It is essential to compare all aspects of the material properties to avoid introducing new problems e.g. thermal conductivity & expansion rates, stiffness and more. One method for comparing likened materials to a set specification is with the aid of material selection programs such as (GRANTA DESIGN, 2020).

A fine pitch thread is preferred for high tensile bolts due to the shallower gradient offering greater resistance to stress. In addition, the required tightening torque is comparatively less than a coarse alternative. This is further supported by (Blake, 1986)

A new tightening torque should be calculated with a different fastener material. In addition, lubricants can be used on the bolt heads to reduce the friction between the bolt and cap during fastening.

3. Crankshaft Study

I. Component Overview

Previous methods will be executed to establish the stresses and fatigue safety factors being transferred through the Ford GDI crankshaft under normal operating conditions.

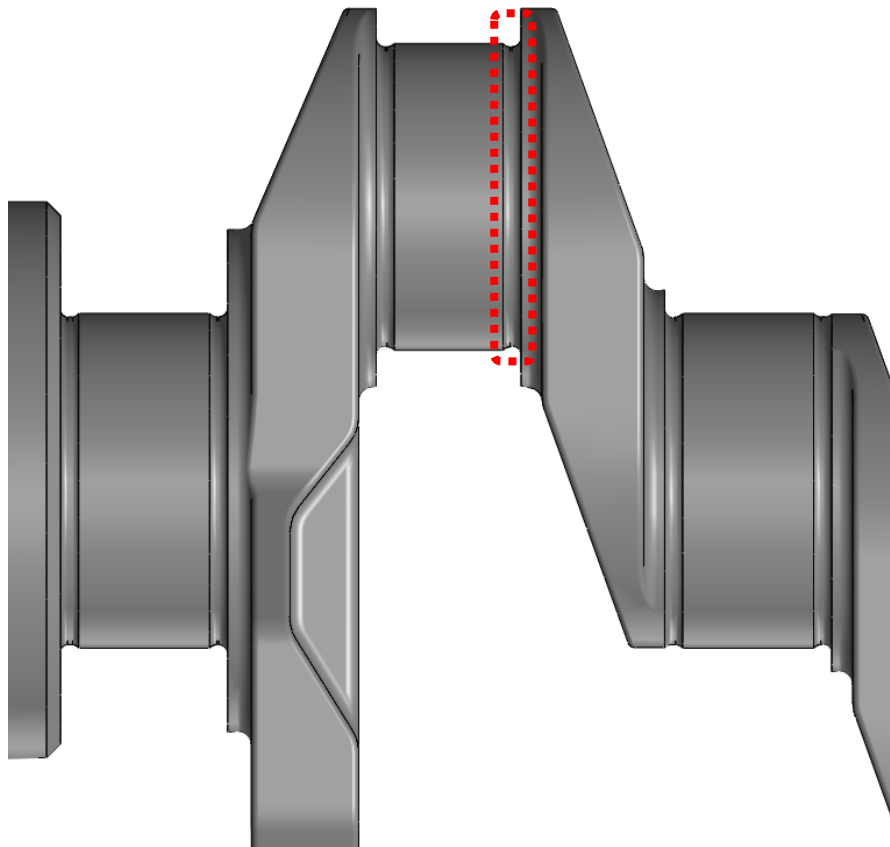


Figure 27 - Critical Fillet Section of Crankshaft

Figure 27 highlights the critical fillet sections that will be investigated around the big end bearing face and main crank bearing face. The most extreme gas and inertia forces will be considered to find the fatigue safety factors, like the gudgeon pin study. Similarly, the stresses due to bending will be examined to form the fatigue safety factor that Ford have engineered into the crankshaft.

Note: The peak force instances occur near TDC. Therefore, the torsional moments acting on the crankshaft are neglected. Reaction forces from the crank journals apply

There are various design techniques to further improve the fatigue strength of the component. The paper by (Çevik and Gürbüz, 2013) highlights the prominence of the fillet region under cyclic loadings. The study compared un-rolled and rolled crankshafts where it was observed that fillet rolling with 15kN rolling load can increase the fatigue strength of the tested crankshaft (ductile iron) by a factor of 4. Another conclusion was the typical crack initiation region is located at the top of the fillet in-line with *Figure 28*.

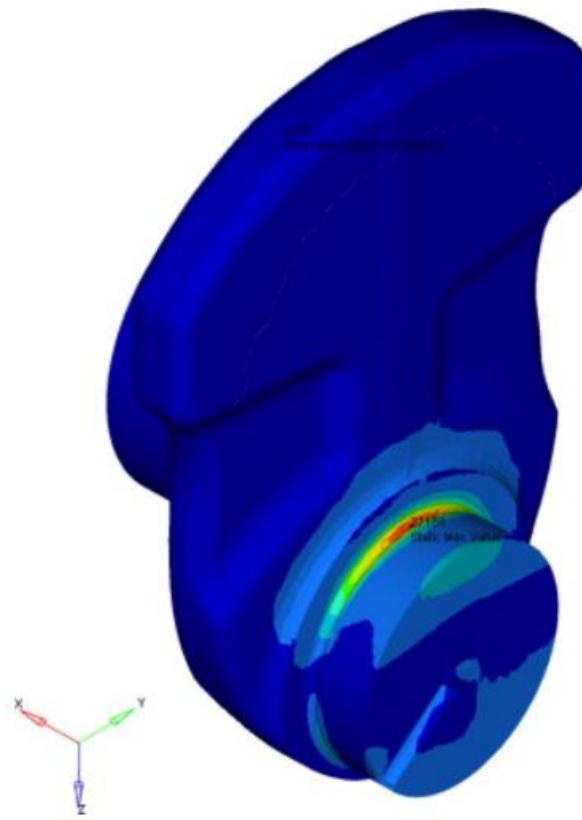


Figure 28 – Typical Stress Distribution across Fillet (Çevik and Gürbüz, 2013)

II. Crankshaft Stress Analysis

It is first necessary to obtain the relevant geometric values.

For second moment of area,

$$b = \frac{D_{crank\ pin} + D_{journal}}{2}$$

Where b = Breadth

$$D = AB$$

Where D = Depth, AB = Distance between fillets

$$I = \frac{bD^3}{12}$$

The equations used to calculate the bending and shear stresses in *Section 2* are recycled into this study. The results of which can be seen below in *Table 18*.

Table 18 - Crankshaft Stress Results

Variable	Value	Unit	Description
I	32306.72	mm ⁴	2 nd moment of area
A	1590.43	mm ²	Area
σ_{bmax}	221.14	N/mm ²	Maximum bending stress
σ_{bmin}	-69.3	N/mm ²	Minimum bending stress
τ_{max}	12.89	N/mm ²	Maximum shear stress
τ_{min}	-4.04	N/mm ²	Minimum shear stress
σ_{maxcom}	222.64	N/mm ²	Maximum combined stress
σ_{mincom}	-69.77	N/mm ²	Minimum combined stress

The stress concentration process detailed in *Sections 3 and 4* revealed a factor of 1.6 for the crankshaft fillet.

III. Fatigue Analysis

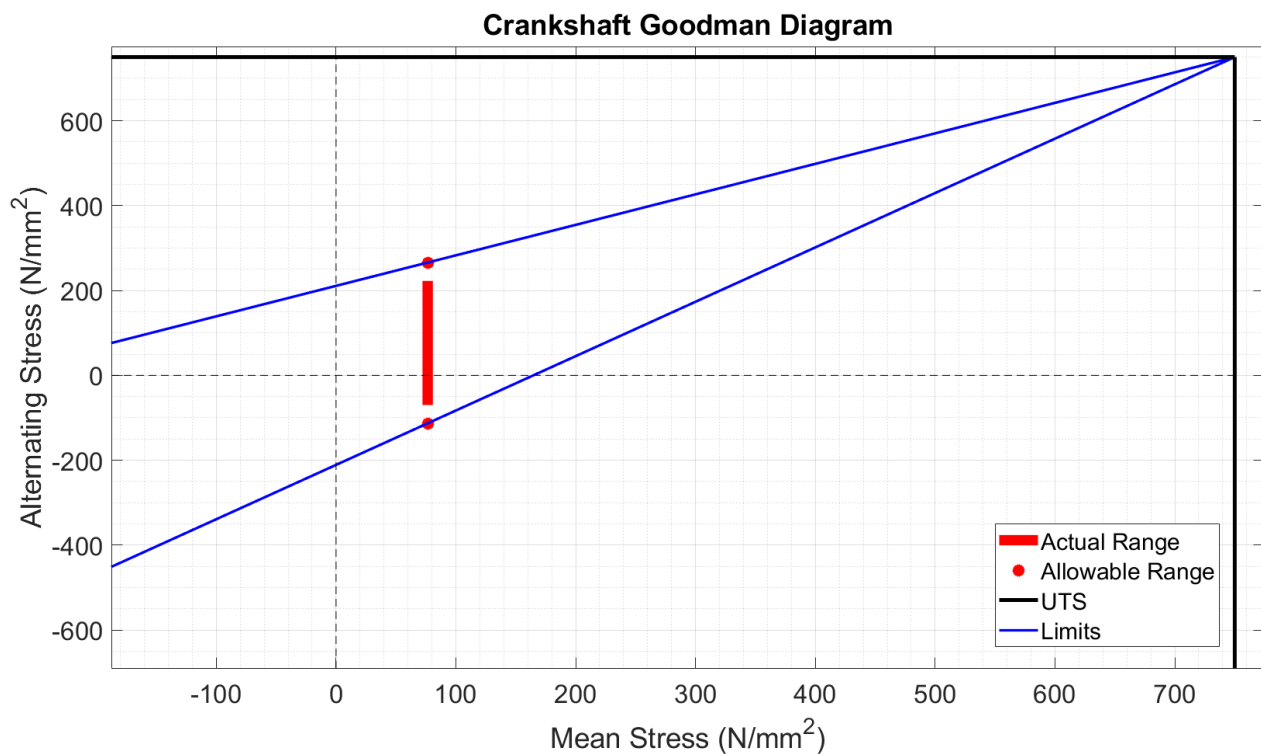


Figure 29 - Crankshaft Goodman Diagram

A Fatigue Safety Factor of 1.3 is found from the Goodman analysis, which yields a 30% margin of safety (assuming no material/surface defects).

This safety factor should be reviewed if the engine is modified for an increase in either engine speed or peak cylinder pressure, as previously demonstrated. If the modification is deemed to put the crankshaft in danger of fatigue, a thermochemical case hardening process known as Plasma Nitriding can be carried out.

The process occurs at low pressure, where a high ionizing glow discharge is produced around the component. As a result of this, the surface area is filled with nitrogen to a controlled amount. This tends to be a significantly thicker case hardening than that found on the gudgeon pin (R. Selvabharathi*, 2018).

The effects of introducing a higher surface compression to the crankshaft can be seen in Figure 30.

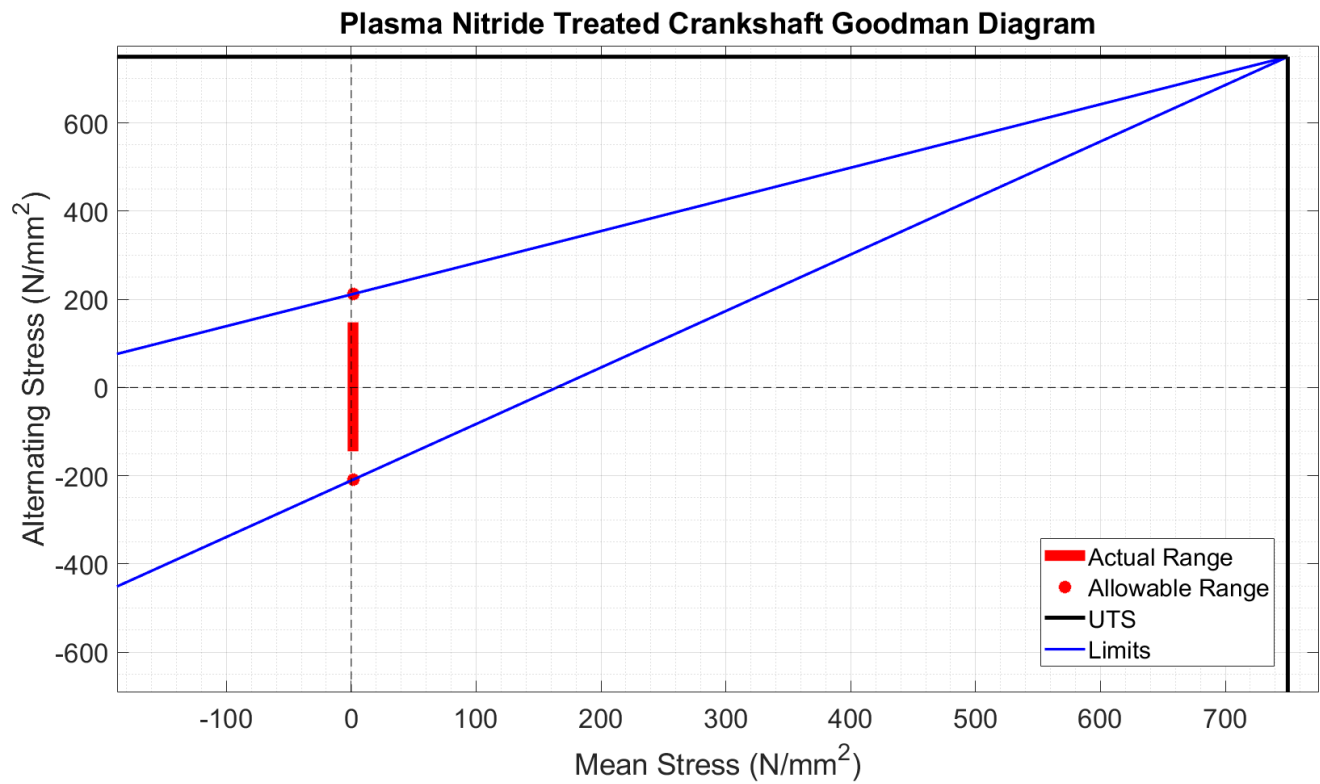


Figure 30 - Plasma Nitride Treated Crankshaft Goodman Diagram

A hypothetical 75 N/mm^2 is applied to the range in alternating stress for the crankshaft. This yields a new safety factor of 1.6, a 23% increase over the OEM version.

4. Conclusion

The methods used through this investigation have provided sound fatigue safety factors for the chosen Ford GDI engine components. Comparisons between this theoretical model and real test procedures can be made for critical points on each component. The lowest fatigue safety factor was found in the crank fillet, closely followed by the OEM connecting rod bolt.

The Goodman technique offers a platform to compare fatigue characteristics across design iterations and different components. This comparison is carried out using basic geometric influences and simplified loadings. Thus, the Goodman technique operates from the following core inputs:

- Material UTS
- Stress Concentration Factor
- Minimum Stress
- Maximum Stress

Although these core variables may include additional factors, it should be remembered there are further constituents to fatigue problems and some examples are detailed below.

- The materials resistance to corrosion and in parallel, moisture content of the system. This can accelerate the decline in component life through corrosion based crack propagation as discussed by (C.Michael, Sue.Hudson, 1976).
- Resistance to contamination and additional reactions to other substances/materials in contact.
- The temperatures through this study are assumed as ambient, yet material properties are susceptible to change with deviations in temperature.
- Resonance in the system. This can severely influence component life if not accounted for.

Ford have evidently accounted for the worst-case scenario and all calculated fatigue safety factors reflect this outcome. This proves the method used is best suited for reverse engineering purposes, where the original safety factor of the component is required for further development. Through this recommended application, the additional fatigue inducing properties will have already been accounted for in the design/development phase.

5. Appendix A – Stress Concentration Calculator in MATLAB

```
%% INPUT VARIABLES FOR SHOULDER WITH FILLET [STRESS CONCENTRATION]

d = 1.58; % INSERT Smallest Diameter
D = 4.68; % INSERT Largest Diameter
L = 4.7; % INSERT length of D section
r = 3; % INSET radius

%% BACKGROUND MATHS, DO NOT ADJUST

h = (D-d) / 2; % DO NOT ADJUST
hr = h/r; % DO NOT ADJUST
Cr = 3 / ( ( r/ (D-(2*h)) ) ^0.25); % DO NOT ADJUST
lond = L/D; % DO NOT ADJUST *****THIS IS L/D*****

%% REMOVE THIS SECTION FROM COMMENT and run IF
%% "lond" is > "cr"      AND 0.1 < hr  < 2.0

c1 = 1.007 + (1 * (sqrt(hr))) - 0.031 * hr;
c2 = -0.114 - (0.585 * (sqrt(hr))) + 0.0314 * hr;
c3 = 0.241 - (0.992 * (sqrt(hr))) - 0.271 * hr;
c4 = -0.134 + (0.577 * (sqrt(hr))) - 0.012 * hr;

kt = c1 + (c2*((2*h)/D)) + (c3*((2*h)/D)^2) + (c4*((2*h)/D)^3);

%% REMOVE THIS SECTION FROM COMMENT and run IF
%% [ lond is > "cr" ] AND [ 2.0 < hr  < 20 ]

% c1 = 1.042 + (0.982 * (sqrt(hr))) - 0.036 * hr;
% c2 = -0.074 - (0.156 * (sqrt(hr))) - 0.0010 * hr;
% c3 = -3.418 + (1.220 * (sqrt(hr))) - 0.0005 * hr;
% c4 = 3.450 - (2.046 * (sqrt(hr))) + 0.051 * hr;
%
% kt = c1 + (c2*((2*h)/D)) + (c3*((2*h)/D)^2) + (c4*((2*h)/D)^3);
```

6. Appendix B – Example of Stress Calculator in MATLAB (Gudgeon Pin Study)

```
%% QUESTION 1 [Fatigue Analysis for Gudgeon Pin]

%% Gudgeon Pin Geometry *** ADJUST AGAINST REAL PIN ***

% Length 1 [Top Length Gudgeon Pin (mm)]
GPL1 = 7.5;
% Length 2 [Bottom Length Gudgeon Pin (mm)]
GPL2 = 6.7;
% Outer Pin Diameter (mm)
OPD = 21;
% Inner Pin Diameter (mm)
IPD = 12.5;
% Y [Distance From Neutral Axis To Surface (mm)]
Y = 10.5;
% I [Using Outer and Inner Pin Dia (mm)]
I = ( (pi/64) * (OPD^4-IPD^4) );
% Mass of Exposed Gudgeon Pin [kg]
M = 0.02681;

%% ENGINE GEOMETRY
% Crank Radius [m]
R = 0.0407;
% Connecting Rod Length [m]
L = 0.134;

%% GUDGEON PIN MATERIAL PROPERTIES

% Ultimate Tensile Strength
GPUTS = 1000;
% 1/2 UTS
GPUSS = 500;
% Endurance Limit
GPEL = 450;

%% MAX COMPRESSIVE

% Engine Speed for Maximun Compressive Force [rpm]

RPMComp = 5700;

% Maximum Compressive Force Acting [N]
MaxComp = 41000.78;

% Maximum Compressive Force Crank Angle

MaxCompAngle = 20;
%% *** STUDY AT 5700RPM [PEAK NET COMPRESSIVE] ***

% Maximum Gudgeon Pin Inertia Force [N]
```

```

FPinInerComp = M*R*((2*pi*RPMComp)/60)^2 * ((cosd(MaxCompAngle)) +
(R/L) * (cosd(MaxCompAngle*2)));

% Bending Moment Due to Maximum Gudgeon Pin Inertia Force

FPinInerBendMomComp = FPinInerComp*GPL2;

% Bending Moment Due to Peak Net Force
FNetBendMomAComp = (MaxComp/2)*GPL1;

% Net Bending Moment

NetBendMomAComp = FNetBendMomAComp - FPinInerBendMomComp;

%% Bending Stress @ A [PER RPM] [N/mm^2]

BendStressAComp = ( (NetBendMomAComp * Y) / I );

%% Net Force @ A [N]

NetForceAComp = (MaxComp/2) - FPinInerComp;

%% Shear Stress (Tau) @ A [N/mm^2]

ShearAComp = ( NetForceAComp / ( (pi/4)* ( (OPD^2)-(IPD^2)) ) ) ;

%% Resultant Stress

ResStressComp = ((BendStressAComp^2) + ((2*ShearAComp)^2))^0.5;

%%
*****
***
%
*****
***

%% MAX TENSILE

% Engine Speed for Maximun Tensile Force [rpm]

RPMTen = 6000;

% Maximum Inertia Force Acting [N]
MaxTen = 5963.62;

MaxTenAngle = -358;

%% *** STUDY AT 6000RPM [PEAK NET TENSILE] ***

% Maximum Gudgeon Pin Inertia Force [N]
FPinInerTen = M*R*((2*pi*RPMTen)/60)^2 * ((cosd(MaxTenAngle)) +
(R/L) * (cosd(MaxTenAngle*2)));

% Bending Moment Due to Maximum Gudgeon Pin Inertia Force

```

```

FPinInnerBendMomTen = FPinInnerTen*GPL2;

% Bending Moment Due to Peak Net Force
FNetBendMomATen = (MaxTen/2)*GPL1);

% Net Bending Moment

NetBendMomATen = FNetBendMomATen - FPinInnerBendMomTen;

%% Bending Stress @ A [PER RPM] [N/mm^2]

BendStressATen = ( (NetBendMomATen * Y) / I );

% Net Force @ A [N]

NetForceATen = (MaxTen/2) - FPinInnerTen;

%% Shear Stress (Tau) @ A [N/mm^2]

ShearATen = ( NetForceATen / ( (pi/4)* ( (OPD^2)-(IPD^2)) ) ) ;

% Resultant Stress

ResStressTen = ((BendStressATen^2) + ((2*ShearATen)^2))^0.5;

%% ***** GUDGEON PIN MEAN, RANGE & SAFETY FACTORS *****

% Mean Stress

MeanStressGP = (ResStressComp + ResStressTen) / 2;

% Range of Stress

RangeStressGP = ResStressComp - ResStressTen;

% Allowable Range

AllRangeGP = ( (GPEL * 2) * (1-(MeanStressGP/GPUTS)) );

% Fatigue Safety Factor

FSFGP = AllRangeGP/RangeStressGP;

%% GOODMAN DIAGRAM
GPUTS1 = [1000 1000]; % UTS & UTS
GPUTS2 = [0 1000]; % Zero & UTS
GPZero = [0,0]; % Zero Y axis
GPEL1 = [450 1000]; % Elastic Limit & UTS
GPEL2 = [-450 1000]; % -Elastic Limit & UTS
GPMS1 = [147.8929 147.8929]; % Mean Stress
GPMS2 = [531.34 -235.55]; % Allowable Range [ZOOM TO FIND LIMITS]

```

7. References

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