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CALCULATIONS ON PISTON, ROD & CRANKSHAFT ASSEMBLIES

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1 Introduction

This investigation looks at computing the fatigue safety factors of Ford's 1.6L GTDI Ecoboost engine components. Author utilises the 0D approach to model component loads, as they occur between the combustion chamber and the crankshaft. The same 'general' method is applied to every investigated component, which it is then adjusted to fit the specific scenario which is being considered by the author. All of the physical dimensions used in this report have been measured by the author and every other model input requirement has been acquired via research. The results are critically analysed to determine model validity.

2 Global Methodology

2.1 Component Masses

Author has taken all relevant component inertias into consideration to assure the accuracy of the loads. Naturally, author has measured all component masses as showed in Figure 2-1, the results can be observed in Table 2-1.

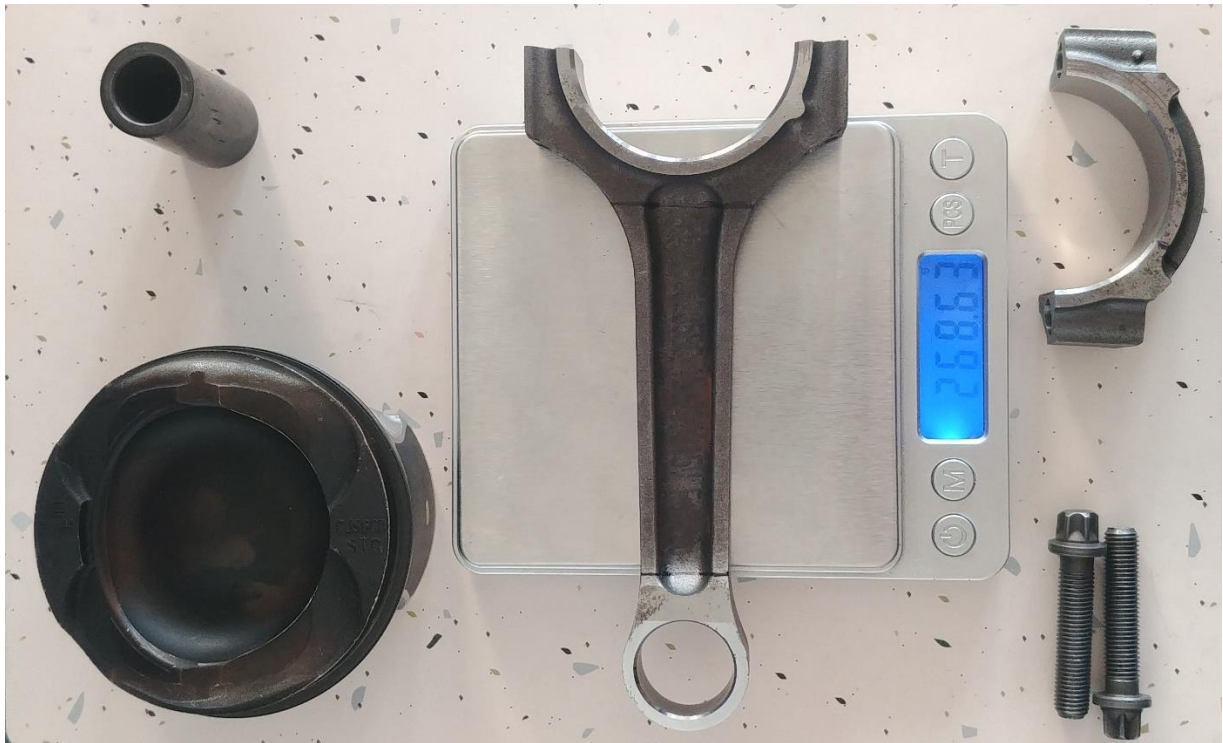


Figure 2-1: Component Mass Measurement

Table 2-1: Component Masses

Masses (g)	OEM
Piston	254.55
Piston Ring	5.89
Circlip	0.82
Piston Assembly	267.97
Total Gaudgeon pin	79.97
unsupported Gaudgeon pin	28.30
Connecting Rod (no cap)	268.64
Connecting Rod Cap	105.10
Connecting Rod Fastener (x1)	17.58
Total Connecting Rod	408.90

2.2 Forces & Stresses

This section outlines the general method of calculating loads at every section between the piston and the crankshaft. Author has applied the same method to every scenario, which is to only consider the loads which occur between the investigated section and the combustion chamber. However, this method is slightly modified in every scenario, the alternations are talked about in the corresponding sections of this investigation.

All forces in this investigation have been calculated every degree of the 4-stroke cycle. Author has decided to do this instead of assuming peak forces at TDC and BDC, since author has tried to keep high accuracy at every step of this investigation.

The applied equations were:

$$F_{gas} = P_{cyl.} * A_{bore}$$

Equation 1: Gas Force

$$F_{inertia} = m_{rec.} * R\omega^2 \cos\vartheta + \frac{R}{L} R\omega^2 \cos 2\vartheta + m_{rot.} * R\omega^2 \cos\vartheta$$

Equation 2: Inertial Force

$$\phi = \sin^{-1}\left(\frac{R}{L} \sin\vartheta - \frac{x}{L}\right)$$

Equation 3: Connecting Rod Angle with Gudgeon Pin Offset

$$F_{friction} = \mu * ((F_{gas} - F_{inertia}) * \tan\phi + P_{cyl.} * \pi B * t_{ring})$$

Equation 4: Frictional Force

$$F_{net} = \frac{F_{gas} - F_{friction}}{\cos\phi} - F_{inertia}$$

Equation 5: Net Force

$$\sigma_{tensional} = \frac{F_{net}}{A_{section}}$$

Equation 6: Tensional Stress

$$\sigma_{bending} = \frac{M_{net}y}{I}$$

Equation 7: Bending Stress

$$M_{net} = F_{net}x - (Fx)_{opposing}$$

Equation 8: Net Moment

$$\sigma_{shear} = F_{net} * A_{perp.}$$

Equation 9: Shear Stress

$$\sigma_{res.} = \sqrt{\sigma_{bending}^2 + \left(\frac{UTS}{YS} \sigma_{shear}\right)^2}$$

Equation 10: Resultant Stress

2.3 Final Endurance Limit

Since this investigation is about the analysis of the component's fatigue over 'infinite' cycles, it was crucial to determine correct Final Endurance Limit (FEL), which is the maximum compressional and tensional stress that component can experience without a failure due to fatigue over 'infinite' life cycle, when the mean stress is 0.

$$F.E.L. = \frac{E.L.*UTS}{K_t}$$

Equation 11: Final Endurance Limit

Author has used table from Appendix 1 to apply endurance limit factors (E.L.) which seemed most appropriate for the application/ material. All stress concentration factors (K_t) were calculated based on the formulas gathered in the second edition of the "Peterson's Stress Concentration Factors". The

following 5 stress concentration types were chosen by author due to his believe of them being the most appropriate for the investigated sections:

- Pin joint with closely fitting pin in tension
- Opposite single notches in finite width element in tension
- Shoulder fillets in thin element in tension and single radius
- Bolt & Nut in tension
- Crankshaft in bending

$$K_{tnb} = \frac{\sigma_{max} dh}{P}$$

Equation 12: Pin Joint Stress Concentration Factor (Pilkey, 1997)

$$K_{tn} = 1 + \left(\frac{\frac{t}{r}}{1.55 \frac{H}{d} - 1.3} \right)^{\frac{\frac{H}{d} - 1 + 0.5 \sqrt{\frac{t}{r}}}{\frac{H}{d} - 1 + \sqrt{\frac{t}{r}}}}$$

Equation 13: Opposite Single Notches Stress Concentration Factor (Pilkey, 1997)

$$K_T = 4.84 * \left(0.42 + 0.16 \sqrt{\frac{1}{\frac{r}{d}} - 6.864} \right) * \left(1 + 81 \left(0.769 - \left(0.407 - \frac{s}{d} \right)^2 * \frac{\delta}{r} * \frac{r^3}{d} \right) \right. \\ \left. * \left(0.285 * \left(2.2 - \frac{b}{d} \right)^2 + 0.785 \right) * \left(\frac{0.444}{\frac{t}{d}^{1.4}} \right) * \left(1 - \frac{\left(\frac{s}{d} + 0.1 \right)^2}{4 \frac{t}{d} - 0.7} \right) \right)$$

Equation 14: Crankshaft Stress Concentration factor (Pilkey, 1997)

To calculate K_t of the shoulder fillets author had to calculate the r/d & H/d ratios, which he then used to locate the K_t on the graph shown in Figure 2-2.

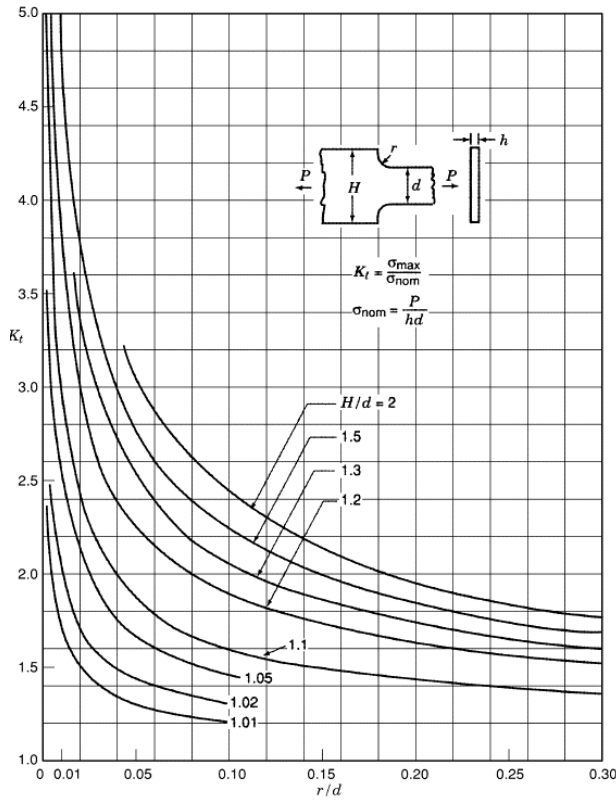


Chart 3.1 Stress concentration factors K_t for a stepped flat tension bar with shoulder fillets (based on data of Frocht 1935; Appl and Koerner 1969; Wilson and White 1973).

Figure 2-2: Shoulder Fillets K_t Graph (Pilkey, 1997)

Sadly, for the stress concentration factor of the fasteners, author was only able to find relationship showed in Equation 15. The range showed is between tapered and standard nut.

$$2.1 \leq K_t \leq 2.7$$

Equation 15: Bolt & Nut Stress Concentration Factor Relationship (Pilkey, 1997)

2.4 Goodman's Diagram

Since the mean stress at any section isn't 0, author had to use Goodman's Diagram to predict the allowable range so that fatigue safety factor (FSF) could be calculated. Figure 2-3 shows an example of a Goodman's Diagram with UTS of 1000 Nmm^{-2} and F.E.L. of 400 Nmm^{-2} .

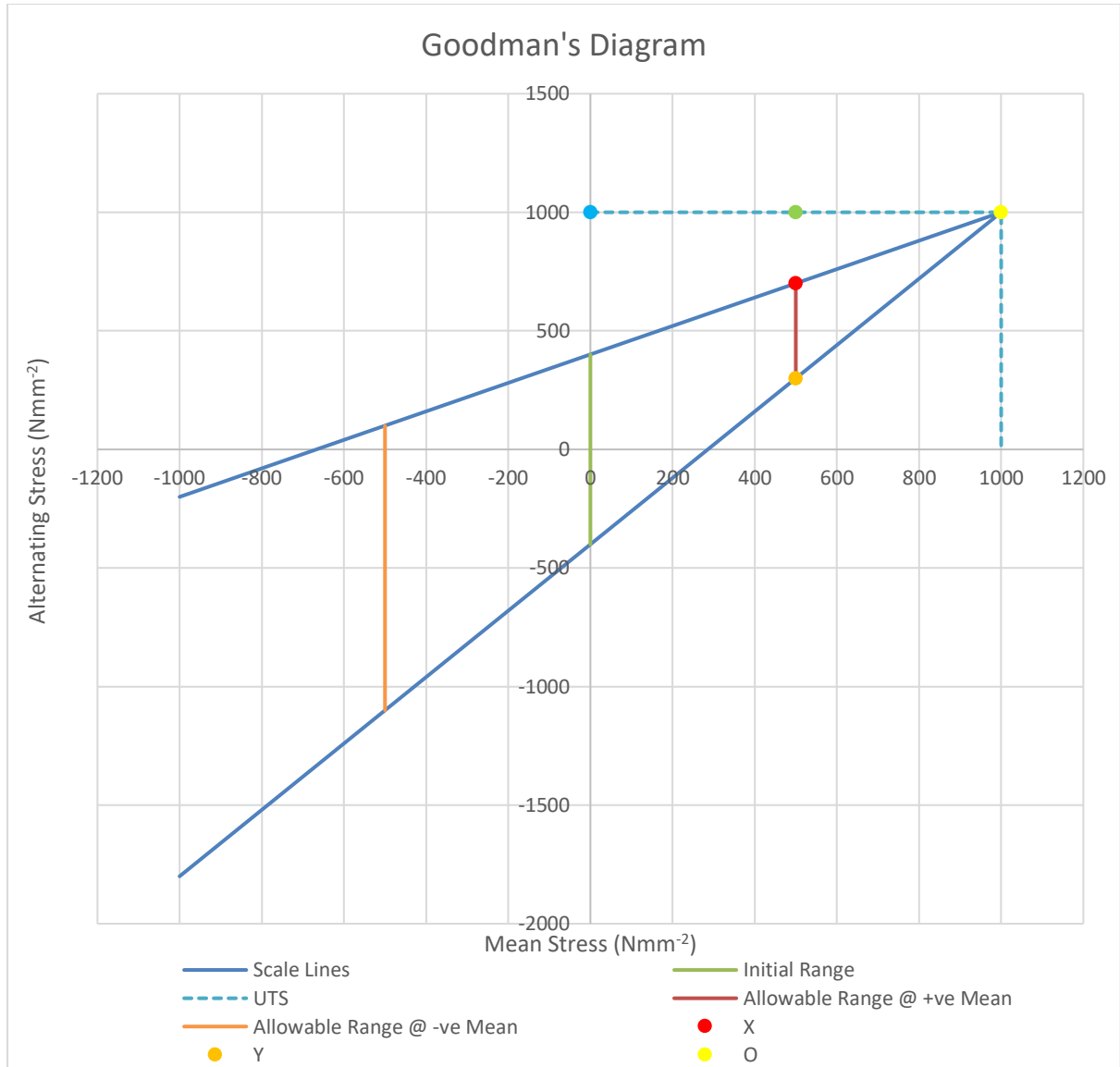


Figure 2-3: Goodman's Diagram Example

To find the allowable range at the new mean, author used the length ratio relationships between points X, Y, O, C and D, as shown in Equation 16 through Equation 19.

$$\frac{XY}{AB} = \frac{OC}{OD}$$

Equation 16: Length Ratio Relationship

$$XY = AB \frac{OC}{OD}$$

Equation 17: Rearranged Length Ratio Relationship

$$\sigma_{allowable\ range} = 2 * F.E.L.* \left(\frac{UTS - \sigma_{mean}}{UTS} \right)$$

Equation 18: Initial Allowable Range

$$\sigma_{allowable\ range} = 2 * F.E.L.* \left(1 - \frac{\sigma_{mean}}{UTS} \right)$$

Equation 19: Final Allowable Range Equation

Finally, the FSF was calculated by Equation 20.

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

Equation 20: Fatigue Safety Factor

3 Fatigue Safety Factor of Gaudgeon Pin

3.1 Measurements

Author has measured the gaudgeon pin dimensions observed in Table 3-1, as shown by Figure 3-1. Author believes that the used digital vernier delivered more than required accuracy.

Table 3-1: Gaudgeon Pin Dimensions

Gaudgeon Pin Specification	
Dimensions:	
Outer Diameter (mm)	21
Inner Diameter (mm)	11.9
A to Centre of Pin Bore (mm)	10.85
A to unsupported CoG (mm)	7.97



Figure 3-1: Gaudgeon Pin Measurement

Author has calculated the unsupported gaudgeon pin mass by dividing the total gaudgeon pin mass by the volume of the CAD model that author has created. This density was then multiplied by the CAD model volume of the one half of the unsupported gaudgeon pin, as showed in Figure 3-2.

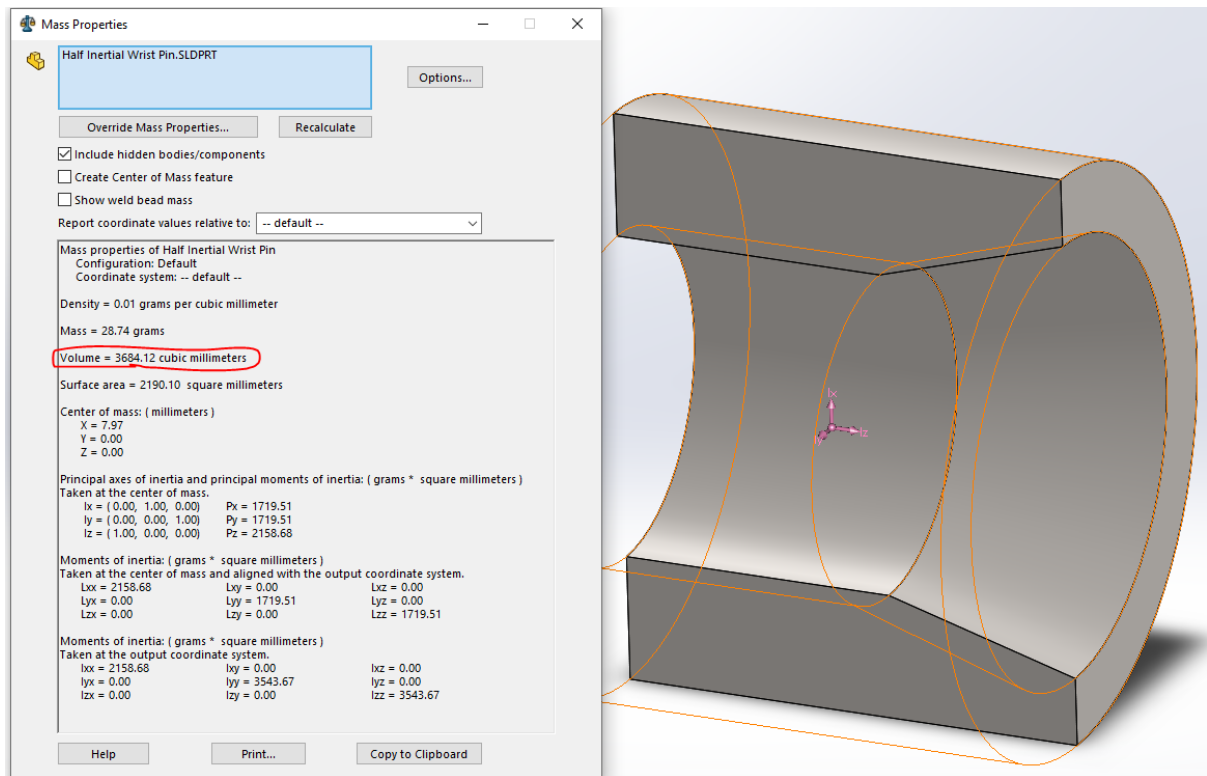


Figure 3-2: CAD model of the one half of unsupported gaudgeon pin section

This also allowed author to accurately measure the centre of gravity of the unsupported section of the gaudgeon pin.

The CAD model has shown that the reason behind chamfering gaudgeon pin was to decrease the distance between the unsupported CoG and point A, as this decreased the moment caused by the gaudgeon pin inertia.

3.2 Methodology Alternations

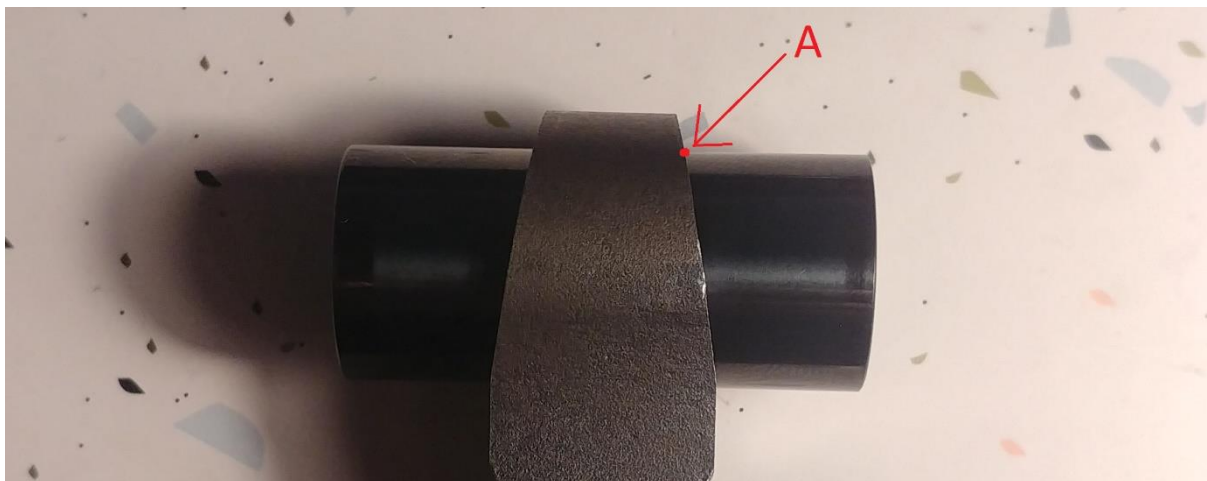


Figure 3-3: Location of Point A

Figure 3-3 shows where author calculated the gaudgeon pin stresses. The shown point was chosen due to the likely hood of it experiencing the biggest forces, since this point is furthest away from where loads are acting.

Another difference compared to the method showed in section 2 is that the inertial force of the component being studied was accounted for as a moment, which counteracted the net force moment.

Author has assumed the gaudgeon pin material to be typical alloy steel, with Ultimate Tensile Strength (UTS) of 720 Nmm^{-2} and Yield Strength (YS) of 620 Nmm^{-2} (SolidWorks, 2018).

Lastly, as shown in Figure 3-3, the OEM gaudgeon pin is coated in diamond like coting (DLC). Therefore, author has assumed additional 200 Nmm^{-2} of intrinsic stress, since author believes that Ford wouldn't be able to afford elaborate coating methods. (Joan & Enric, 2015).

3.3 Results

The combined stress as function of crank angle is shown in Appendix 2.

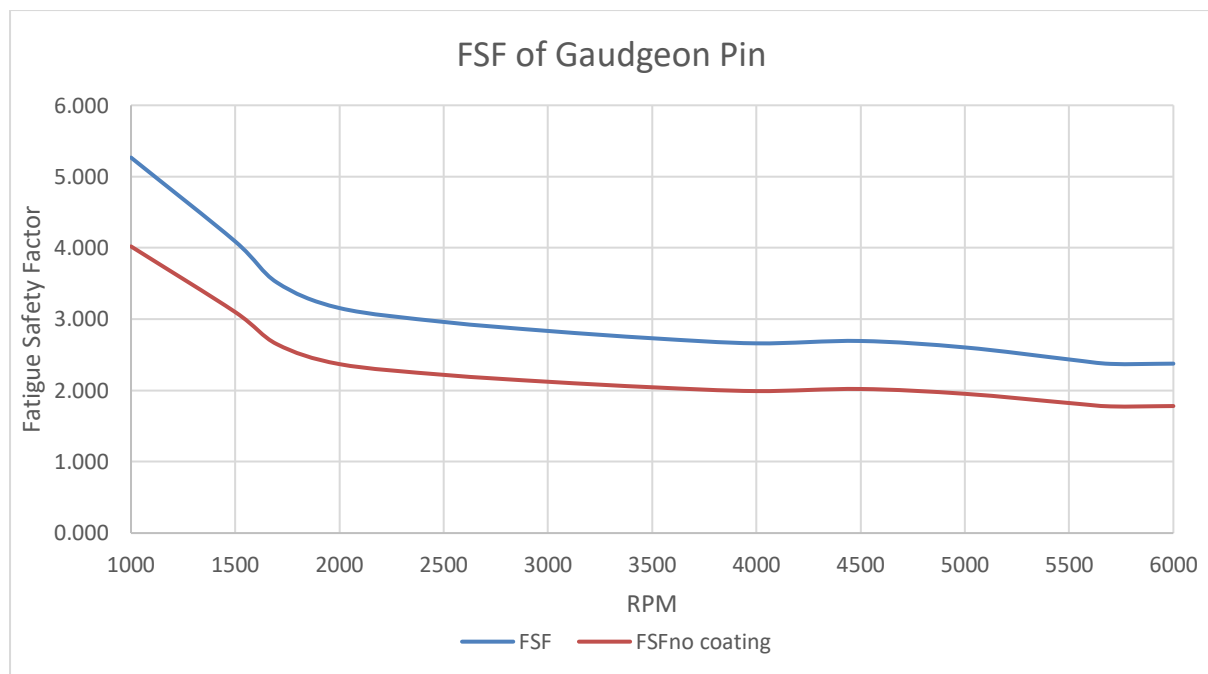


Figure 3-4: Graph of the fatigue Safety Factor of the Gaudgeon Pin

Figure 3-4 shows the calculated gaudgeon pin FSF. As author expected, the FSF decreases with rpm. This is caused by the raise of the mean stresses as well as the actual range of stress, as shown in Figure 3-5. Both of which are due to the raise of peak cylinder pressures and rpm. Since they: increase the compressive stress due to more gas force being produced, overcoming the frictional and inertial forces and the increase of inertia at TDC exhaust, respectively. The initial big decrease between 1,000rpm and 2,000rpm is caused by a rapid increase of peak cylinder pressures.

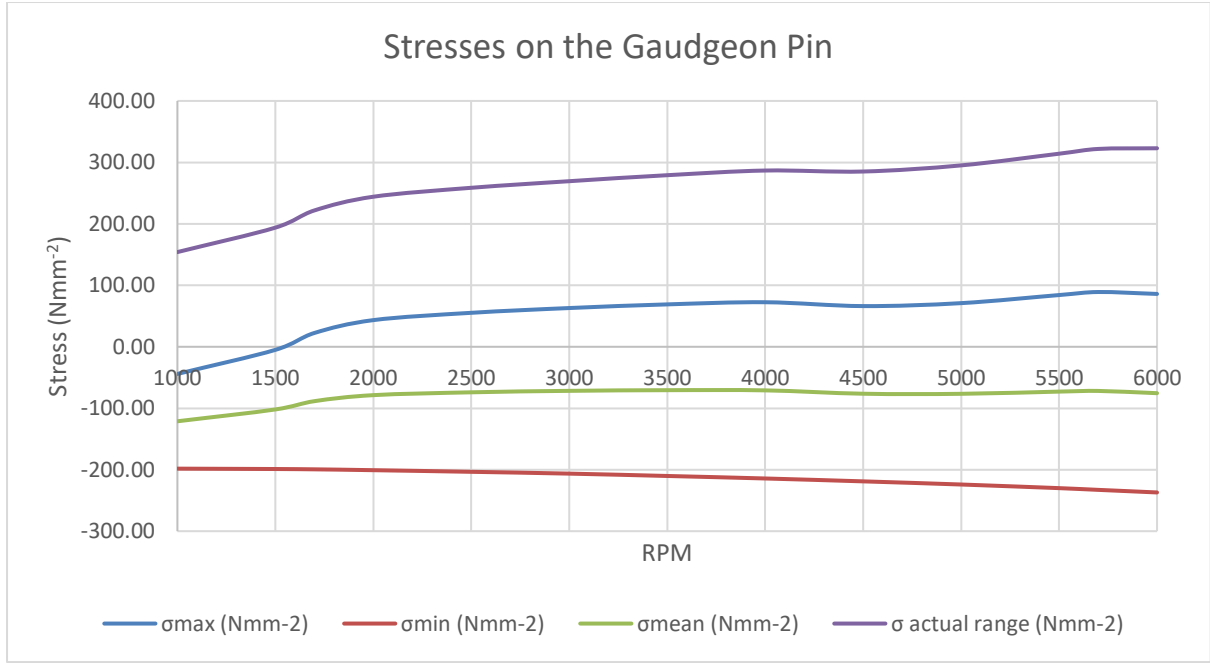


Figure 3-5: Stresses on the Gaudgeon Pin

Figure 3-4 also shows a comparison between coated and uncoated FSFs. At the minimal FSF which occurs at 6,000rpm, the 200 Nmm-2 of intrinsic stress causes the FSF to increase by 33.3%, which is considerable. This occurs since the extra compressive stress decreases the mean stress, causing the allowable range to increase. Author believes this shows that application of plasma coatings such as DLC to be extremely advantageous if properly applied, as it allows rotating loaded components such as gaudgeon or crank pins to be decreased in diameter, which decreases frictional losses as showed by Equation 21. Additionally, more sophisticated plasma coatings can yield intrinsic stresses as high as 1,800 Nmm⁻² (Joan & Enric, 2015).

$$W_{frictional} = \mu \frac{\omega}{d} (2\pi)^2 LR^3$$

Equation 21: Frictional Word Done due to Fluid Drag

Lastly, author believes the calculated FSFs to be overestimated, as mass manufacturer such as Ford couldn't afford to waste material, tooling, transport and engine performance on such high FSF. Therefore, author believes the model to be incomplete, introduction of more factors such as temperature and frequency would definitely decrease the FSF further. Despite the overestimation, author believes the results to still be useful, since the same method can be applied in reverse to design a component and the reliability of it should be about as good as long as the minimum FSF is kept. Author also acknowledges that another explanation of high FSF could be to allow another component to implement a better design.

4 Fatigue Safety Factor of the Connecting-Rod Sections

4.1 Measurements

As shown in Figure 4-1, author has measured all connecting rod dimensions, including radii, of which results are shown in Table 4-1. Additionally, Figure 4-2 shows the location and names of the considered sections.

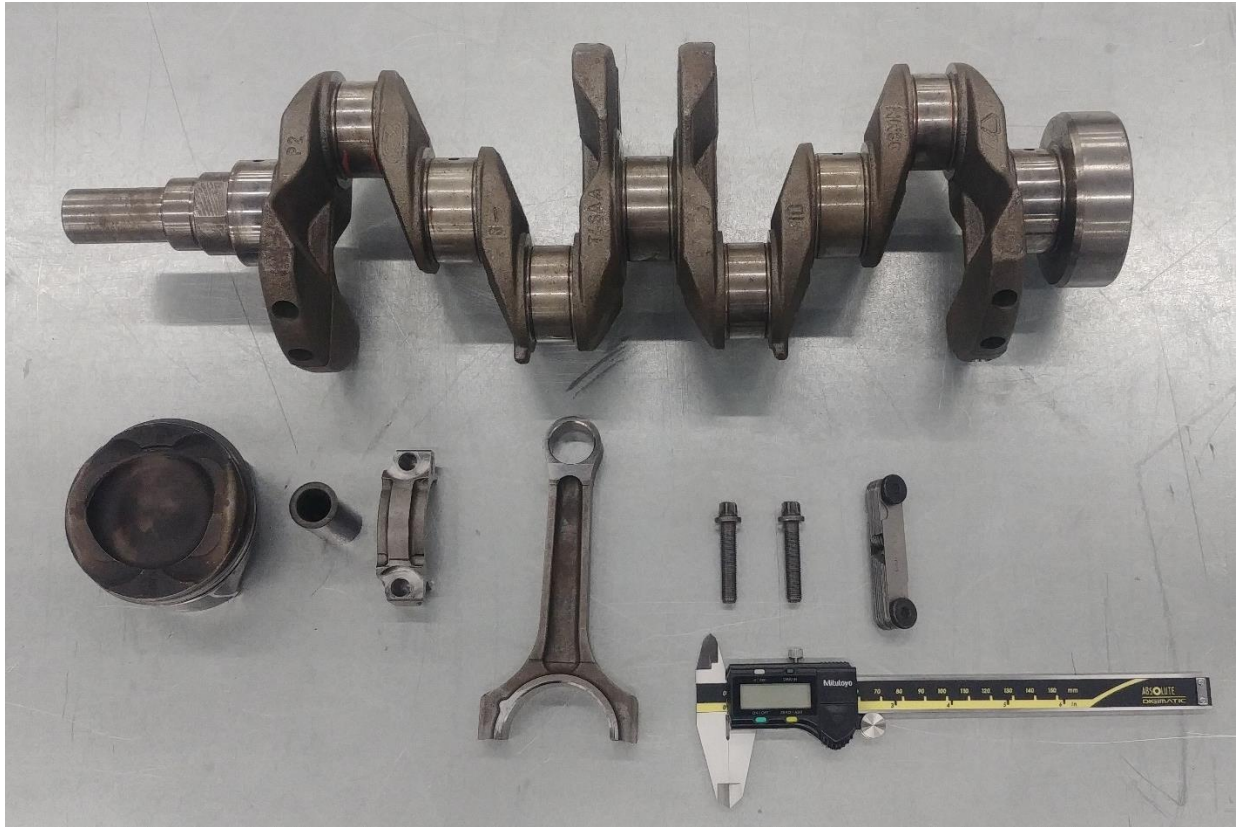


Figure 4-1: Engine Components Measurements



Figure 4-2: Connecting Rod Studied Sections Placement

Table 4-1: Connecting Rod Dimensions

Connecting Rod Specifications	
Dimensions:	
A1	
A_1 (mm ²)	49.4
A_{1d} (mm)	20.99
A_{1H} (mm)	3.3
$A_{1 \text{ position}}$ (mm)	0
A2	
A_2 (mm ²)	72.2
A_{2t} (mm)	5.5
A_{2r} (mm)	3.5
A_{2H} (mm)	16.2
A_{2d} (mm)	5.2
$A_{2 \text{ position}}$ (mm)	21
B	
B (mm ²)	142.6
B_t (mm)	5
B_r (mm)	3.5
B_H (mm)	13.15
B_d (mm)	3.15
B_{position} (mm)	101
C	
C (mm ²)	130.1
C_r (mm)	3
C_H (mm)	20.85
C_d (mm)	9.5
$C_{r/d}$ (mm)	0.316
$C_{H/d}$ (mm)	2.195

4.2 Methodology Alternations

Firstly, author has multiplied the total inertial force of the connecting rod by the ratio of the section location from the gaudgeon pin bore and the connecting rod length, since author has assumed the inertial forces ‘below’ the studied section to be not loading the studied section.

Secondly, the forces and cross-sectional areas have been halved, since author has assumed equal load distribution on each of the side of the centreline between the bores.

Thirdly, the connecting rod material was assumed to be P/F-10C50, with UTS of 830 Nmm^{-2} and YS of 550 Nmm^{-2} (Edmond Ilia, 2002).

To calculate stress concentration factors for sections A1, A2 and B, author used Equation 12, Equation 13 and Equation 13 respectfully. The Figure 2-2 was also used to locate K_t at point C.

Table 4-2: Material Specification & K_t of the Connecting Rod Sections

Material Specification:	
Ultimate Tensile Strength (Nmm^{-2})	830
Yield Strength (Nmm^{-2})	550
Endurance Limit Factor	0.49
K_{TA1}	1.190
K_{T2A}	1.518
K_{TB}	1.329

4.3 Results

The Net Force acting on connecting rod sections A1, A2, B and C are all shown in the Appendix 3 Appendix 4 Appendix 5 and Appendix 6 respectfully.

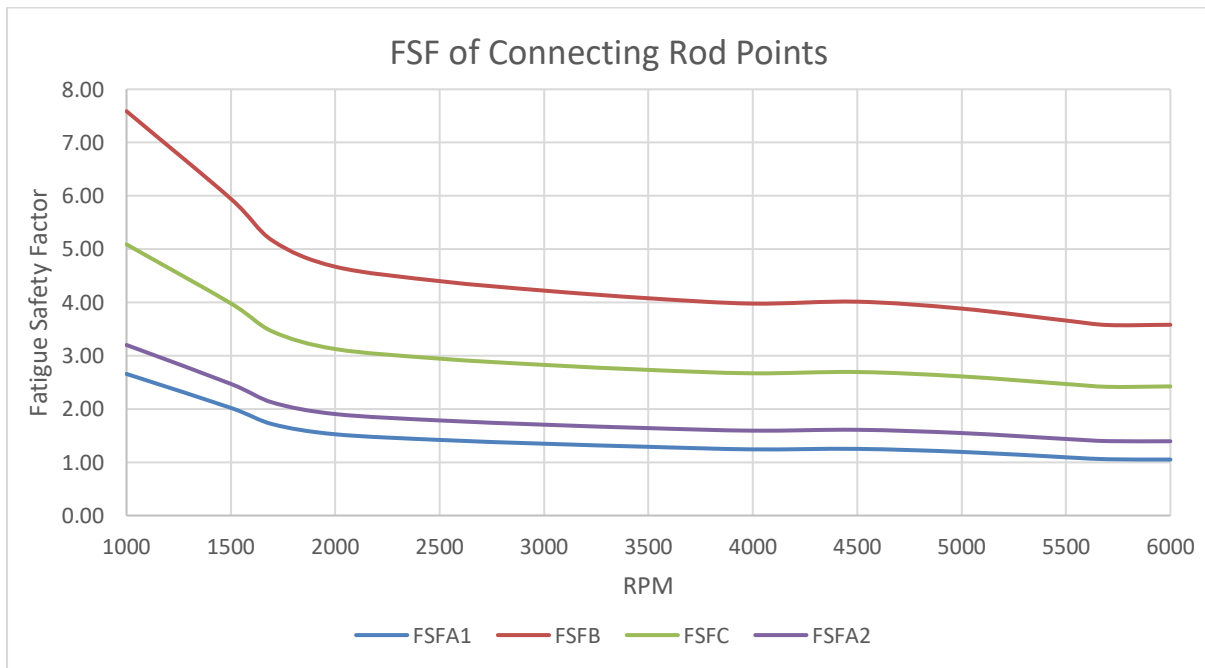


Figure 4-3: Fatigue Safety Factor of Connecting Rod Points

Figure 4-3 shows the FSF for all 4 studied connecting rod sections.

As author has expected, the overall profile of all FSFs is very similar to that of the gaudgeon pin from the section 3.3 of this investigation, since the same relationships seen in Figure 3-5 apply to the conrod too. Naturally, the magnitudes differ very much due to very different cross-sectional areas as well as the implementation of the stress concentration factors.

The FSF of the section B is the greatest. Author has expected this since this section has the largest cross-sectional area, causing that section to experience the least amount of stress. Large cross-sectional area is also the reason why section C has the 2nd largest FSF, as it is only 8.77% smaller than the area at section B. The reason why the difference between FSFs is larger than the mentioned difference, is due to the stress concentration factor at C being 35.44% higher than at section B.

Interestingly, despite the least amount of conrod inertia, the lowest FSFs occurred at sections A1 and A2. This too was due to the cross-sectional area, of which section A1 was the smallest, which is showed by the total minimum FSF of just 1.05.

This clearly shows that on a well-designed connecting rod, the peak stresses occur at the very top of the connecting rod.

Author was very surprised to see such small FSF, since it contradicts the conclusion made in the section 3.3 of this investigation. However, it also supports the statement made after the conclusion. Author believes that to find the truth, FEA study would have to be carried out on the gaudgeon pin and the connection rod. This is however outside the scope of this investigation.

Lastly, author believes the calculated stress concentration factors to be inaccurate, as despite carefully choosing the most applicable methods, the fact remains that author wasn't able to find stress concentration methods which would perfectly fit in the studied scenarios. This was due to the complexity of the conrod profile as well as the direction of loads. This too is another reason why author beliefs an FEA study to be crucial in this case.

5 Fatigue Safety Factor of The Connecting-Rod Cap Fasteners

5.1 OEM Fastener

5.1.1 Measurements

As previously shown in Figure 4-1, author has also measured the OEM fasteners. It was assumed that the stretch region of the fastener to end at the 2nd engaged thread.

5.1.2 Methodology Alternations

The net force was naturally halved, since author assumed the loading to be equal on both fasteners.

Since the connecting rod inertia doesn't act on the fasteners at TDC, which is where peak forces occur, author has decided to exclude its mass from the calculations.

Author choose stress concentration ratio of 2.7 based on Equation 15, since female component is standard and not tapered.

Author has assumed Young's Modulus of the fastener to be that of tool steel and the conrod that of P/F-10C50, of which values are 210 kNmm^{-2} and 170 kNmm^{-2} respectfully (Edmond Ilia, 2002).



Figure 5-1: Fastener UTS

As Figure 5-1 shows, the UTS of the fasteners was 1100 Nmm^{-2} . Since the UTS was relatively high, author has assumed the UTS to YS ratio to be 0.9. Author also assumed the fasteners to be tightened up to 90% of YS, consequently author assumed that the axial ratio to be 0.6. All other important values are shown in Table 5-1.

Table 5-1: Fastener Material Specification

Material Specification:	OEM
E_{Fastener} (Nmm ⁻²)	210000
E_{Cap} (Nmm ⁻²)	170000
UTS _{Fastener} (Nmm ⁻²)	1100
UTS to Yield Ratio	0.9
Tightened to % of Yield	90
Axial % of Yield	60
K_T	2.7
Endurance Limit Factor	0.45
μ_{Thread}	0.18
μ_{Head}	0.12

5.1.3 Results

The inertial and net loads acting on the OEM fastener as a function of crank angle can be found in Appendix 7 and Appendix 8.

Table 5-2 shows important calculations done on the OEM fasteners and the connecting rod cap.

Table 5-2: OEM Fastener Calculations

Calculations:	OEM
K_{Fastener} (KNmm ⁻¹)	252.3
K_{Cap} (KNmm ⁻¹)	651.5
Cap to Fastener Ratio	2.6
Clamping Force (kN)	22.2
FEL (Nmm ⁻²)	183.3
Bolt Tightening Torque (Nm)	27.9

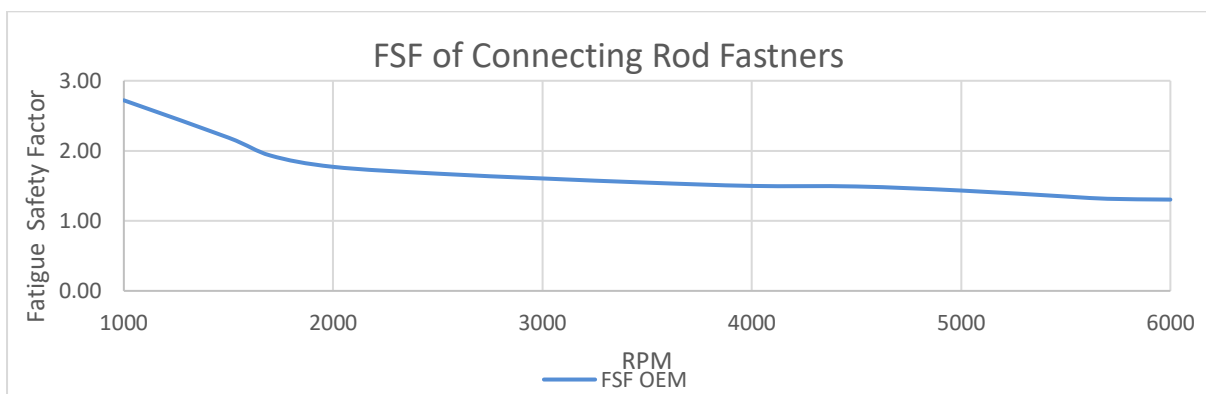


Figure 5-2: Fatigue Safety Factor of Connecting Rod Fasteners

Figure 5-2 shows the calculated FSFs of the OEM fasteners. As expected, the profile was very similar to the FSF profiles of the previously studied components. The minimum FSF was 1.30 at 6,000rpm. Since the clamping force of 22.2 kN opposes gas force, the fasteners are always in tension as shown in Appendix 8 (which is what keeps the conrod cap attached to the conrod). This results in the mean stress to be negative. As Figure 2-3 showed, if the mean stress is -ve, the allowable range increases. However, author wasn't convinced this was the case in real life, despite this, author has assumed this to be correct for the rest of this investigation since study of such topic is outside the scope of this particular investigation.

5.2 Modified Engine Fastener

5.2.1 Modified Engine Specification

Since author has decided to compute stresses through the entire 4 stroke cycle, author had to create the modified engine cylinder pressures. This was done by multiplying the OEM cylinder pressures by the ratio of the peak cylinder pressure from the modified and OEM engines. OEM cylinder pressures at 3,000rpm were chosen due to them delivering the peak net indicated mean effective pressure of 20.245 (Bar) and the modified engine peak cylinder pressures were taken from author's previous work: "Con-Rod Accelerations, Forces & Displacements" (Kaczmarczyk, 2019). The calculated modified engine cylinder pressures are shown in Appendix 9.

5.2.2 Methodology Alternations

The method of calculating the fastener FSF is the same as in section 5.1. However, author has created a sub-method which fed all of the required fastener information, as to aid the design process.

Firstly, author has added the ability to utilise wasted shank in the design of the new fasteners. This was done by implementing Equation 22.

$$K_{Fastener} = \frac{1}{\frac{1}{\frac{E * \pi r_1^2}{L_1}} + \frac{1}{\frac{E * \pi r_2^2}{L_2}} + \frac{1}{\frac{E * \pi r_3^2}{L_3}}}$$

Equation 22: Combined Fastener Stiffness of 3 Sections

Secondly, author has utilized the excel solver tool to rapidly create fastener designs of which the minimal FSF is equal to that of the minimal FSF of the OEM fastener. Author has achieved this by creating a simple fastener design constraints table, as shown in Table 5-3.

Table 5-3: Fastener Design Constraints

Fastener Design		
FSF _{min}	1.304	1.354
SSF _{min}	1.927	2.063
Inputs	Min	Max
Thread	6.0	12.0
Cap Thread Area Ratio	3.0	5.0
D ₁ (mm)	8.0	12.0
L ₁ (mm)	1.0	5.0
D ₂ (mm)	1.0	9.2
UTS * 100 (Nmm ⁻²)	11.0	16.0
Tightened to % of Yield	70.0	100.0
Piston Lightening %	0.0	10.0
Conrod Lightening %	0.0	20.0

Then, author has created many design constraints in the solver tool, as shown in Figure 5-3, based on the values in the Table 5-3.

Figure 5-3: Solver Parameters Input Menu

This allowed author to use the computational power of the machine to generate the designs with the required minimal FSF value. To make this sub-method more realistic, author has also assumed that the modified conrod's mass would equal to the original mass multiplied by the cap area ratio between the

OEM and modified conrod caps. This has ensured that the added inertial forces due to heavier connecting rod will counteract the increase of the cap to the fastener area ratio. Also, author has made sure that the stresses are calculated at the minimal cross-sectional area of the fastener.

5.2.3 Results

The modified engine fastener loads are shown in Appendix 10.

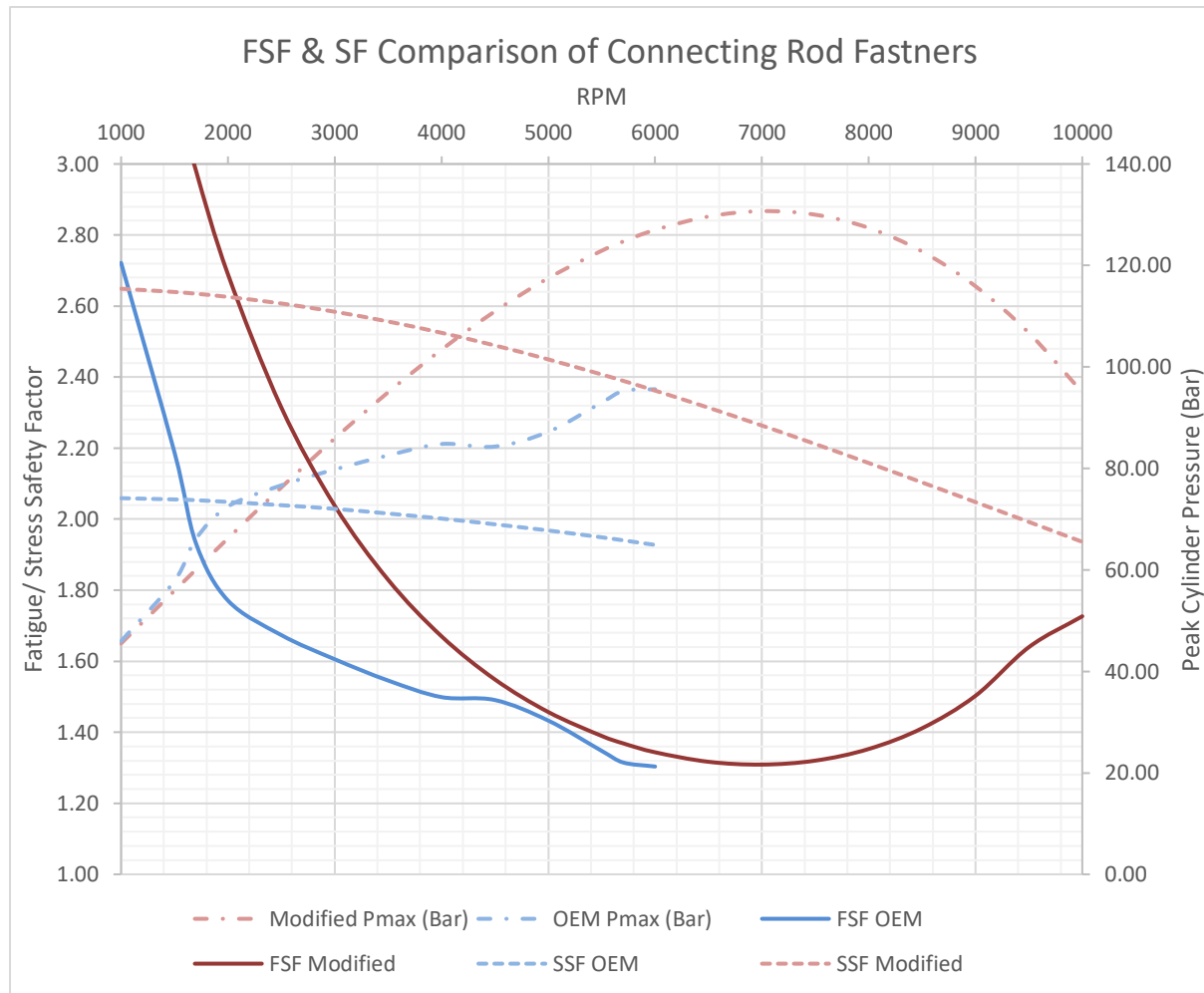


Figure 5-4: Fatigue & Stress Safety factor Comparison Between OEM & Modified Connecting Rod Fasteners

Figure 5-4 shows the FSF & SSF comparison between the OEM and the modified fasteners. In order to achieve the same minimal FSF in the modified fastener, the minimal fastener area had to be increased by 29.0%, due to much higher fastener stresses due to increased cylinder pressures and rpm. This has caused the FSF of the modified fastener below 2,000rpm to be over 3.0.

As expected, both FSF profiles are very similar but with different magnitudes. The why FSF of the modified fastener increases after 7,000r rpm is because FSF is inversely proportional to the cylinder pressure. However, the FSF at 10,000rpm is 5.7% lower than at 3,500rpm, despite the same cylinder pressures. This is due to the much higher inertial loads, which is well showcased in Appendix 11.

Since the fasteners are always in tension, the SSF of the modified fastener decreases at higher rate, due to much higher gas forces.

Table 5-4 shows the final specifications and Figure 5-5 shows the profile comparison of the new fastener designed for the modified engine.

Table 5-4: Modified Fastener Design Specification

Modified Fastener Design:	
Thread Diameter (mm)	9.0
Head Diameter (mm)	12.5
Cap Thread Area Ratio	3.2
Piston Lightening (%)	10.0
Connecting Rod Lightening (%)	20.0
Total Piston Mass Increase (%)	-10.0
Total Connecting Rod Mass Increase (%)	-44.3
EFastener (Nmm-2)	210000.0
ECap (Nmm-2)	170000.0
UTSFastener (Nmm-2)	1200.0
Tightened to % of Yield	70.0
KFastener (KNmm-1)	213.2
KCap (KNmm-1)	444.4
Cap to Fastener Ratio	2.1
Clamping Force (kN)	12.8
FEL (Nmm-2)	200.0
Bolt Tightening Torque (Nm)	17.1

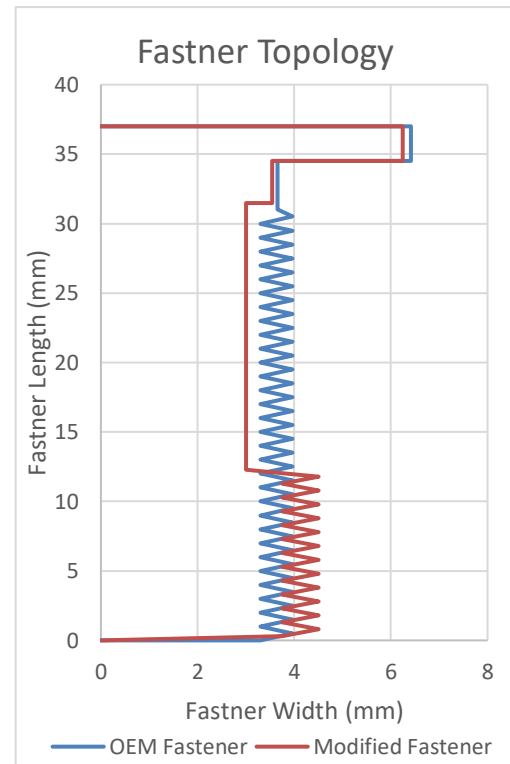


Figure 5-5: Fastener Profile Comparison

The thread side was increased to M9 to increase the cross-sectional area, since no combination of the wasted shank, UTS etc. resulted in acceptable FSF. The Cap area was decreased by 31.8% due to the wasted shank, this has caused an overall connecting rod mass decrease of 44.3%, with assumed 20% lightening due to improved design for motorsport use. This did not only improve the FSF of the fastener but also the rest of the components which experience inertial loads due to the connecting rod, this means that other engine components are more likely to survive the modified engine's much increased performance. This has resulted in stiffness ratio of 3.2 which is practically the same as in the OEM design. Interestingly, these results were achieved with UTS increase of just 100 Nmm^{-2} (9.1% increase), which author beliefs to be viable in high volume production of motorsport engines, such as the 'rally inspired' modified engine used in this investigation. The tightening to % of yield was decreased by 20% to decrease the clamping force, which helped to increase the minimal SSF back up to acceptable value of 1.937 (0.52% increase), since the clamping force cases tension.

5.3 Fastener Design Comparison

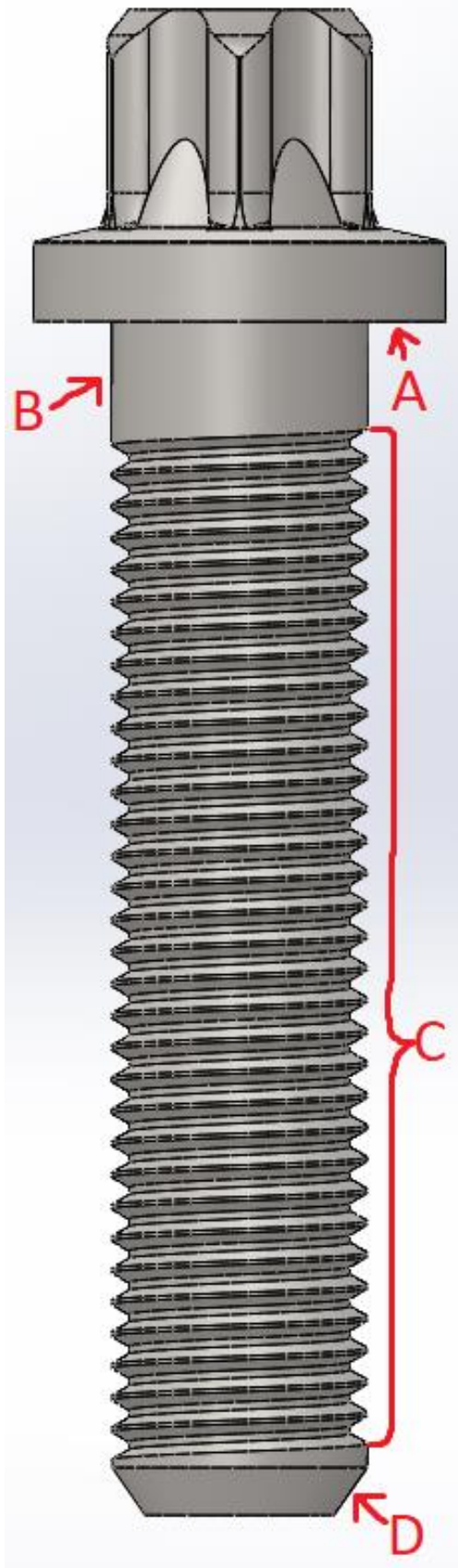


Figure 5-6: OEM Fastener Sketch

Figure 5-6 shows the CAD model of the OEM fastener that author has created. All design features have been annotated in red:

At section A, the surface of the bolt head is very wide. This ensures that there is a lot of material for the fastener to grip on to the conrod cap, as well as to deform once loads are applied.

At section B, part of the shank is smooth. If this was not the case, the thread at the shank to head transition (between A & B) would cause a very big stress concentration which could potentially cause fatigue failure.

Section C shows that the fastener has been threaded most of the shank's length, despite the connecting rod's interface being about half the length of the section. This was done as an inexpensive stress distribution method, since the stress concentration factor is constant throughout the section's length, this prevents high stress constrain occurring at specific threads. As well as making the unengaged threaded section to have lower stiffness, which takes the load off of the threaded section.

Lastly, at section D, the taper not only ensures easy installation, but also prevents the very first threads from experiencing stresses, since those are weakest in most cases due to increased stress concentration, as there is no material beneath those threads to distribute the load.

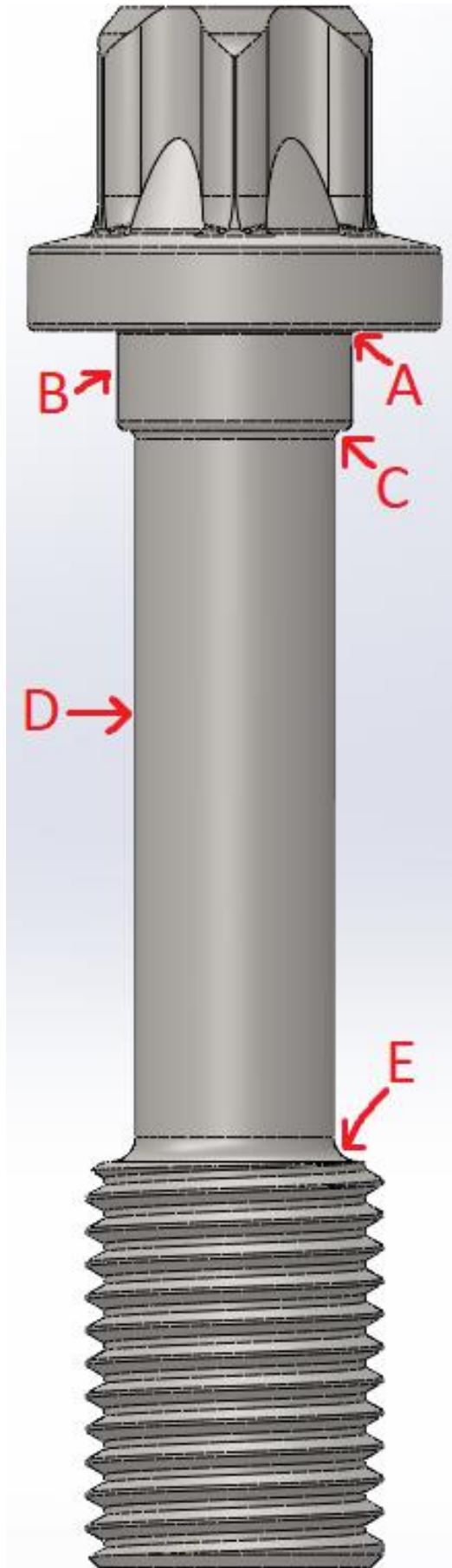


Figure 5-7: Modified Fastener Sketch

Figure 5-7 shows the CAD model of the modified fastener which author has created. All-important design features have been annotated in red:

This design incorporated A and D design features from Figure 5-6.

Sections A, C and E all have a fillet as large as possible, as all of them are considered to be crucial sections, since a profile transition occurs and therefore stress concentration. Furthermore, fillet at section A is there also to prevent strain, ensuring adequate clamping load which prevents the fastener from backing out, as that would definitely cause a catastrophic failure.

At section B, the diameter has been reduced from the thread diameter, this was done to allow a fillet in section A. However, the diameter has been kept as large as possible, just like section B in Figure 5-6, due to the same reasons.

Section D shows wasted shank. This decreases the stiffness of the section, causing much more load to be taken away from the threaded section, compared to the OEM design. This also decreases the overall stiffness of the fastener, changing the conrod cap to fastener stiffness ratio in its favour.

6 Fatigue Safety Factor of The Crankshaft Fillet

6.1 Measurements

As previously shown in Figure 4-1, author has also measured the crankshaft's relevant dimensions, of which values are shown in Table 6-1. Since author wasn't able to fit a radius gauge into the fillet, as it was previously done on other components, author had to predict the radius value based on comparing the radius gauge set on an offset. Therefore, author is not confident in the radius measurement.

Table 6-1: Crankshaft Dimensions

Crankshaft Specifications	
Dimensions:	
Beam Width (mm)	46
Beam Height (mm)	22
Bending Dist. (mm)	30
Bearing Diameter (mm)	48
Bearing Length (mm)	19
Pin Diameter (mm)	44
Depth of undercut (mm)	0.5
Radius (mm)	1.5

6.2 Methodology

To calculate the peak bearing pressure, author has used Equation 23, assuming even distribution of the load.

$$P_{Bearing} = \frac{F_{Net}}{L * D}$$

Equation 23: Bearing Pressure

To calculate the stress concentration factor, author has used Equation 14, since author found it to be the most suitable solution. Despite that, author believes the formula to grossly overestimate the stress concentration factor based on the results shown in section 6.3 of this investigation.

Author also assumed intrinsic stress of 300 Nmm^{-2} , which is relatively low based on author's research (Mariko MATSUDA * 1, 2011).

6.3 Results

The calculated crankshaft loads are shown in Appendix 12.

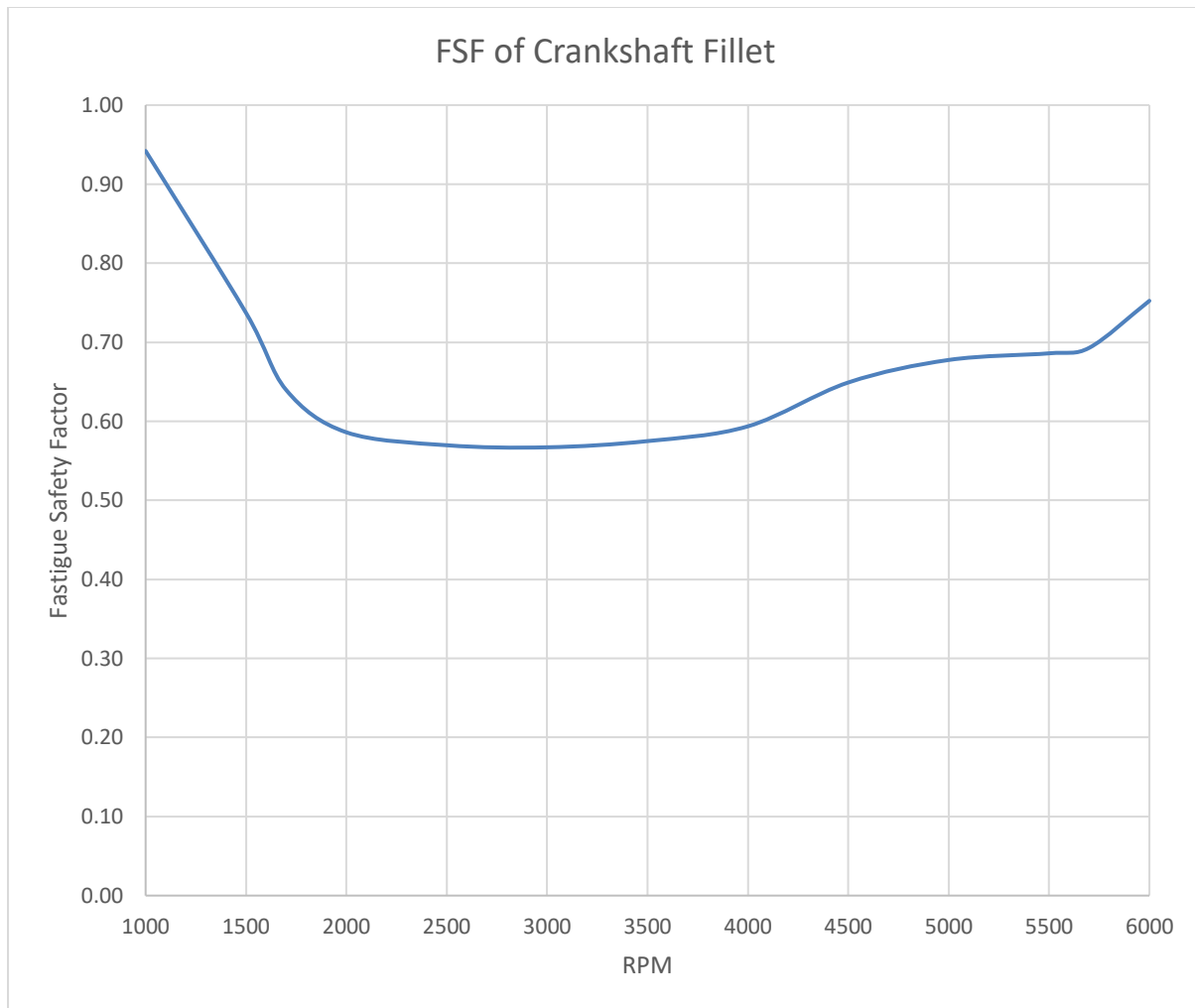


Figure 6-1: Fatigue Safety Factor of the Crankshaft Fillet

Figure 6-1 shows the calculated fatigue safety factor of the crankshaft fillet. It is immediately recognisable that an error exists in the method, since an OEM crankshaft from well renowned company such as Ford would not be released with <1 FSF to mass production. After further investigation author has concluded the method of calculating the stress concentration factor to be very much overestimated. This was found out because such drastically inadequate FSF hasn't occurred at any component before the crankshaft. Due to the similarity of the methods, author was sure the calculated crankshaft loads to be accurate enough for this issue not to occur. Furthermore, the only other drastically different aspect of the calculated values are the relatively small calculated allowable ranges. These were caused by very low F.E.L of just 66.1 Nmm^{-2} , which is 63.9% lower than the 2nd lowest F.E.L calculated in this entire investigation and out of all variables which affect F.E.L the only value which doesn't fit in the ballpark established by previous work, is the stress concentration factor. By decreasing the stress concentration factor by 45%, the new F.E.L became 120.2 Nmm^{-2} , which is only 34.4% lower than the 2nd lowest F.E.L placing it much closer to the ballpark. This change causes the FSFs shown in Figure 6-2.

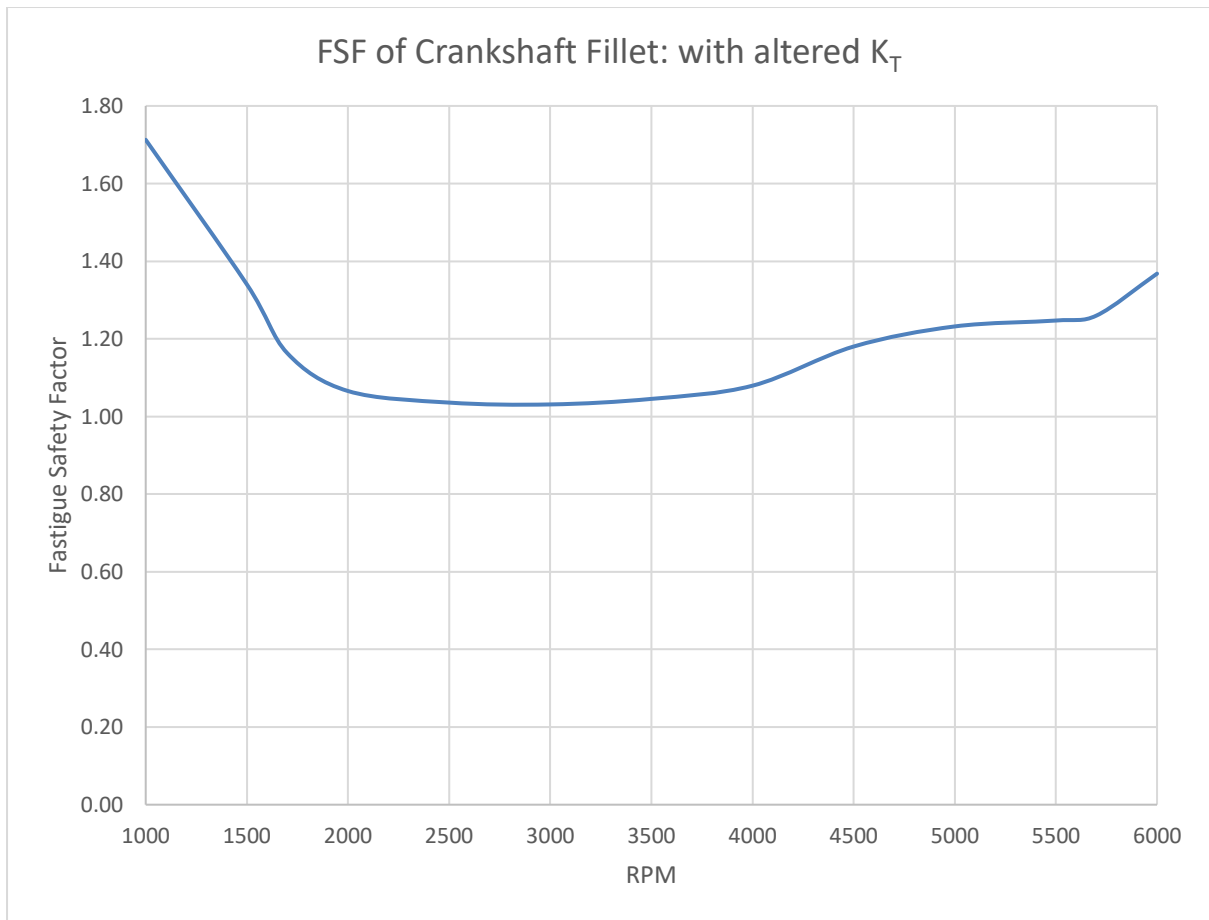


Figure 6-2: Fatigue Safety Factor of the Crankshaft Fillet – with altered K_T

Figure 6-2 shows the FSFs to be much more realistic with the adjusted K_T . However, author still believed the results to be too low since author has neglected to consider other aspects such as temperature of the material, frequency, torsional stresses, friction, bearing contact, compound loads (residual loads created by other cylinders), etc. Author has deemed such work to be outside the scope of this investigation.

7 Conclusion

Author believes the showcased method to be inadequate to produce reliable component design. This is due to many reasons which author has already mentioned in this investigation, in general they are: non-dynamic analysis, 0D FEA model, no thermal stress, frequencies not considered, elevated temperatures not considered, cylinder pressures used were average not the 90th percentile +, too many assumptions were made, certain measurements were inadequate and probably most importantly, author believes the applied method to the Goodman's diagram when the mean is -ve to be incorrect.

Despite these issues, author must acknowledge that reasonable results were calculated for every investigated component, showing the merit and the potential of this model if it was further developed.

Furthermore, author believes the model to be adequate to predict the component's required design changes, which can be modelled in more sophisticated 3D FEA software, for when a design alternation has been carried out in the engine itself, as the sections 5.2 and 5.3 have shown.

Lastly, this investigation has shown that the method created for the purpose of this investigation can be applied to any other engine component. This makes this method very versatile and powerful, considering the time to develop another model utilising this method would be much less time consuming, compared to other more industry mainstream methods such as the utilisation of CAD and FEA programs, not to mention the costs associated with such work.

8 References

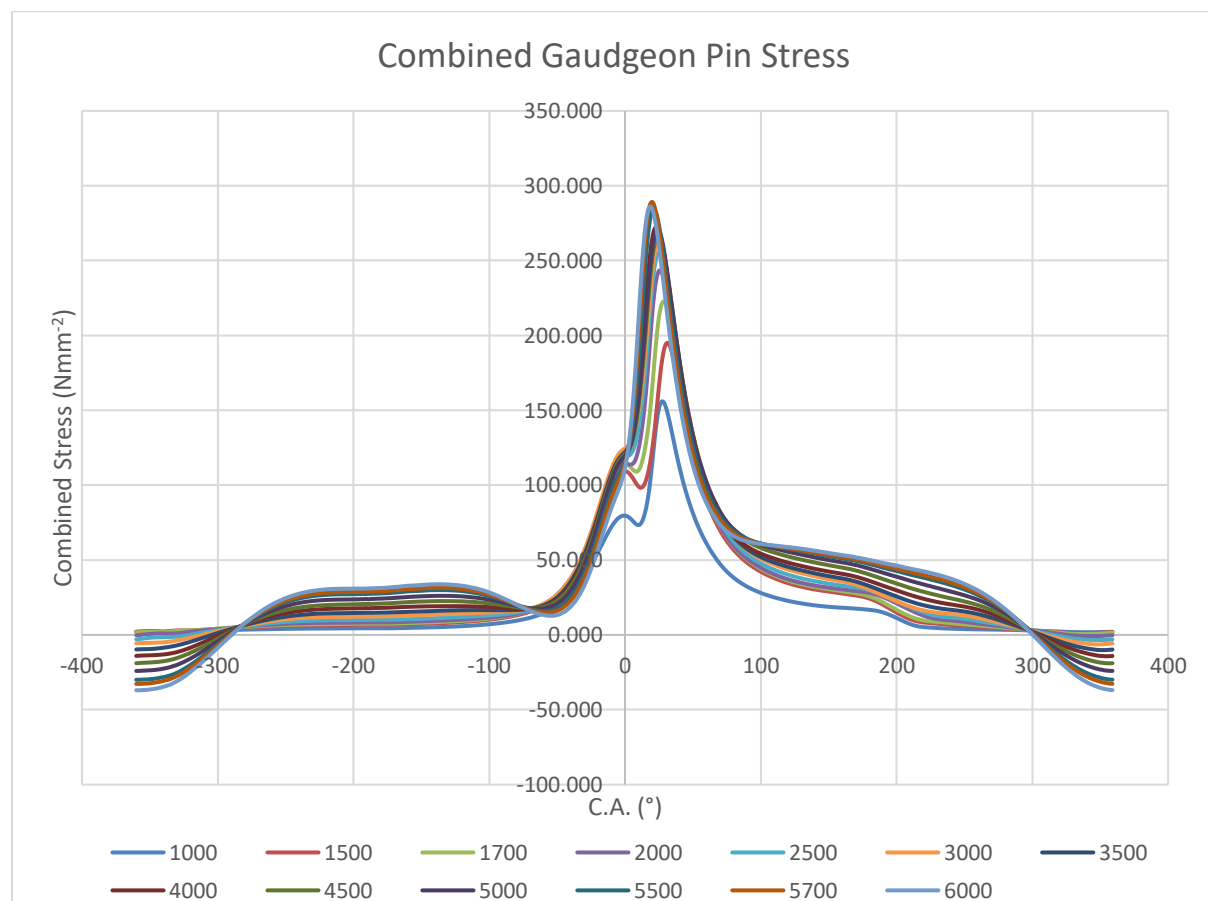
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9 Appendix

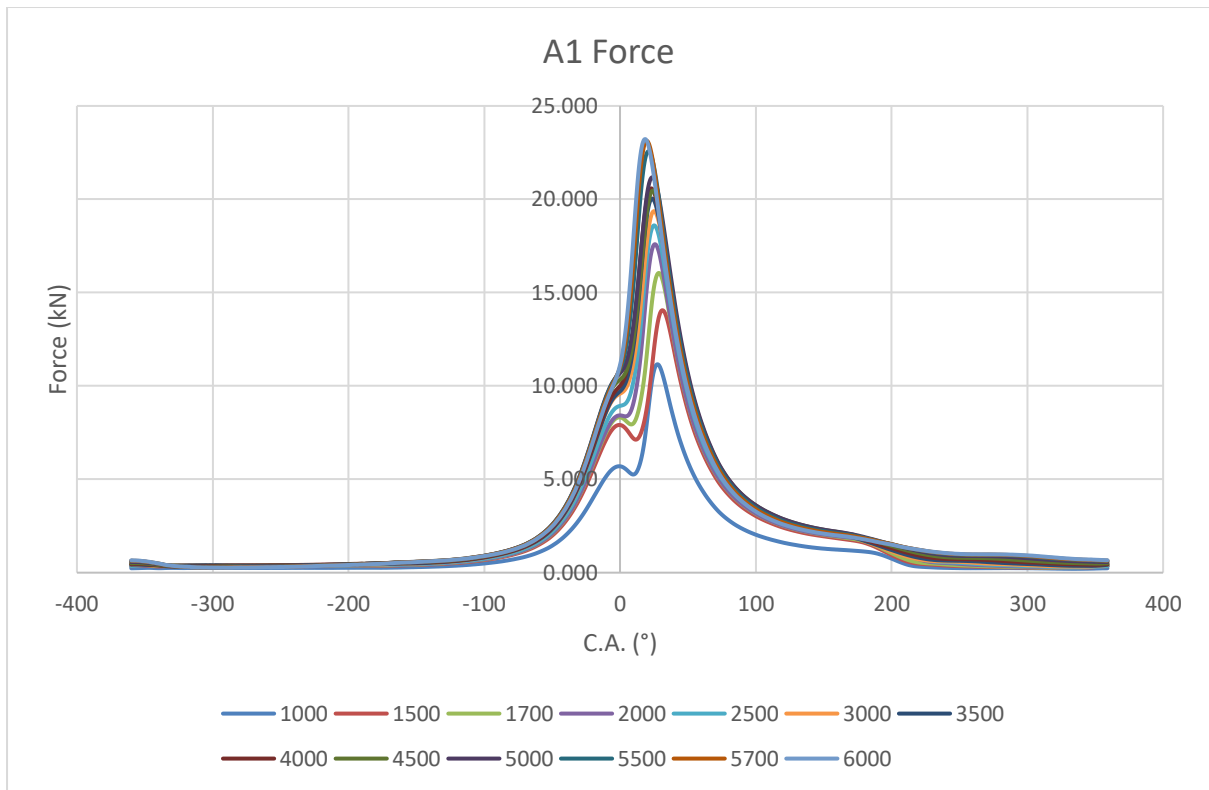
Endurance limit for some steels

Steel	Condition	Tensile strength, σ_u (N mm ⁻²)	Endurance limit, σ_e (Nmm ⁻²)	σ_e/σ_u
0.4% carbon	Normalized	540	270	0.50
(080M40)	Hardened and tempered	700	340	0.49
Carbon, manganese	Normalized	540	250	0.46
(150M19)	Hardened and tempered	700	325	0.53
3% Chrome molybdenum	Hardened and tempered	1000	480	0.48
(709M40)				
Spring steel (735A50)	Hardened and tempered	1500	650	0.43
18,8 Stainless	Cold rolled	1200	490	0.41

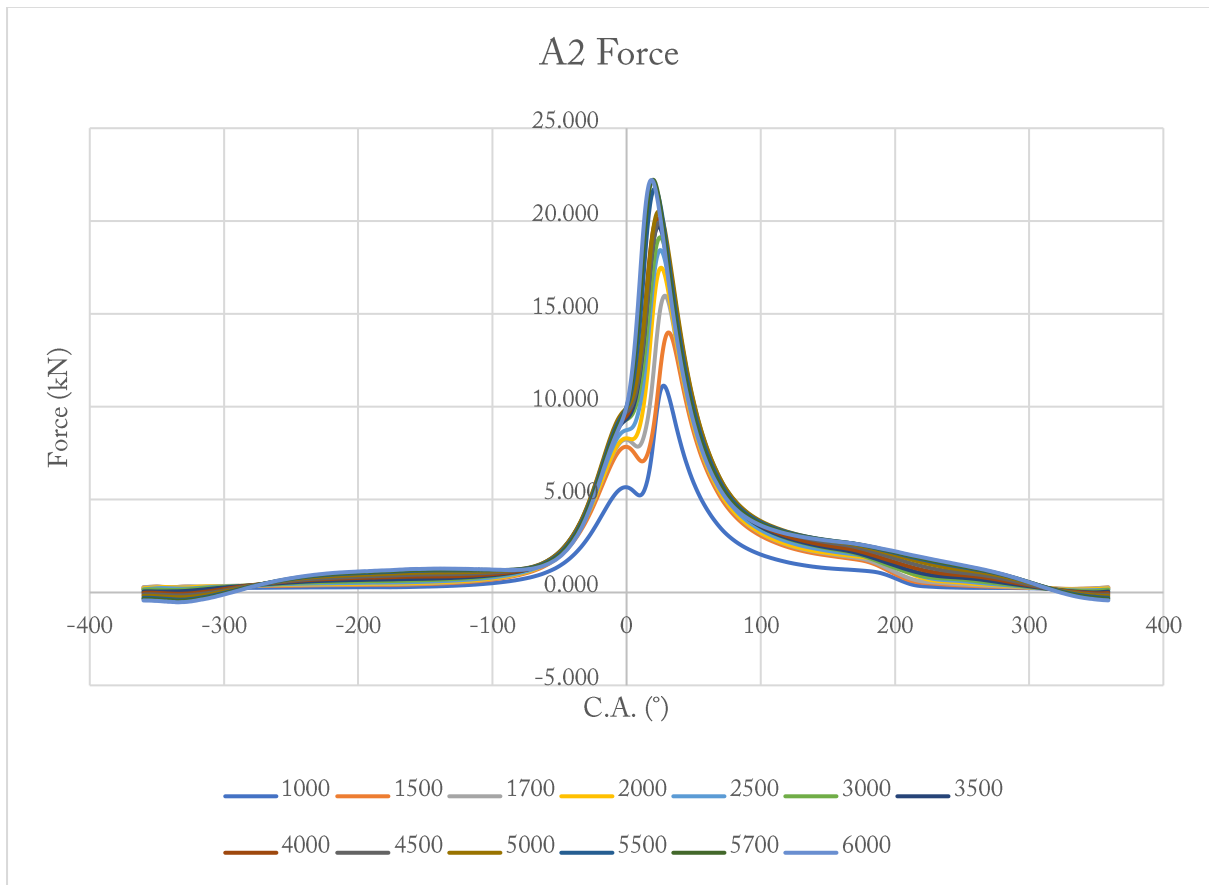
Appendix 1: Endurance Limit Table of Steels (PhD, 1997)



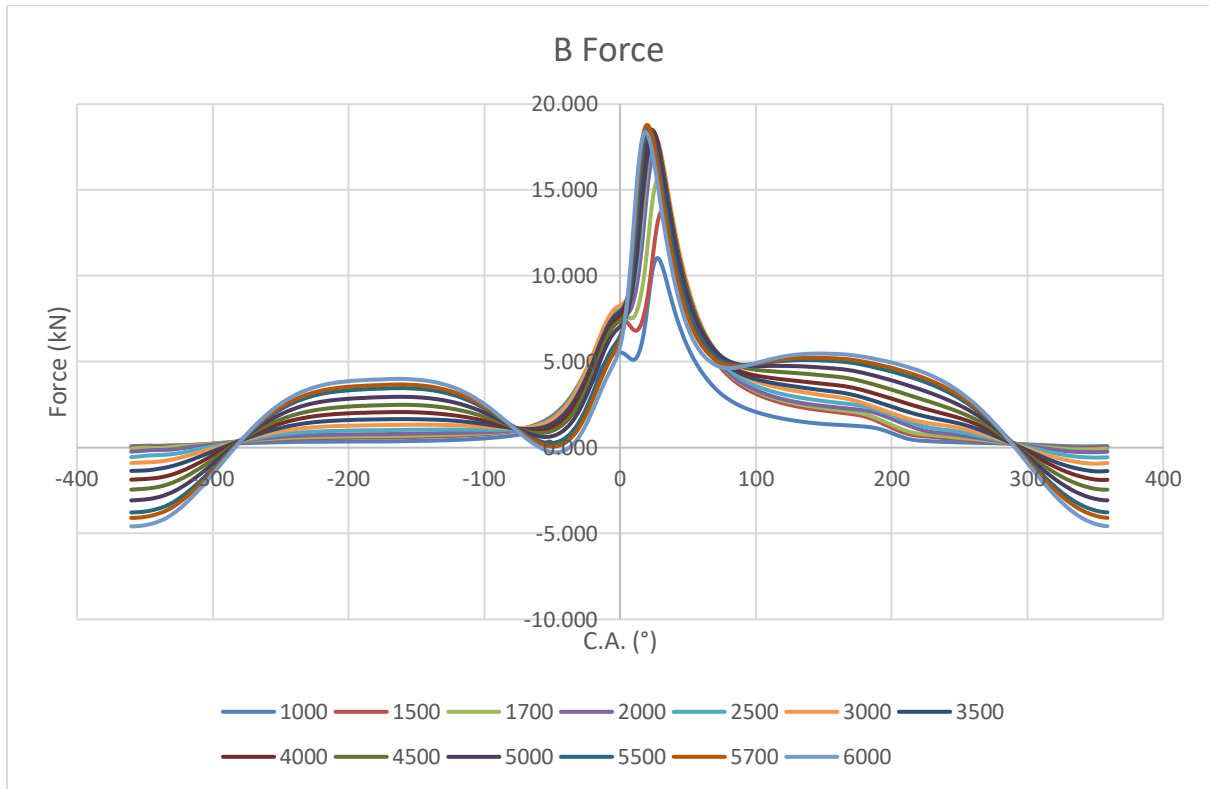
Appendix 2: Combined Gaudgeon Pin Stress



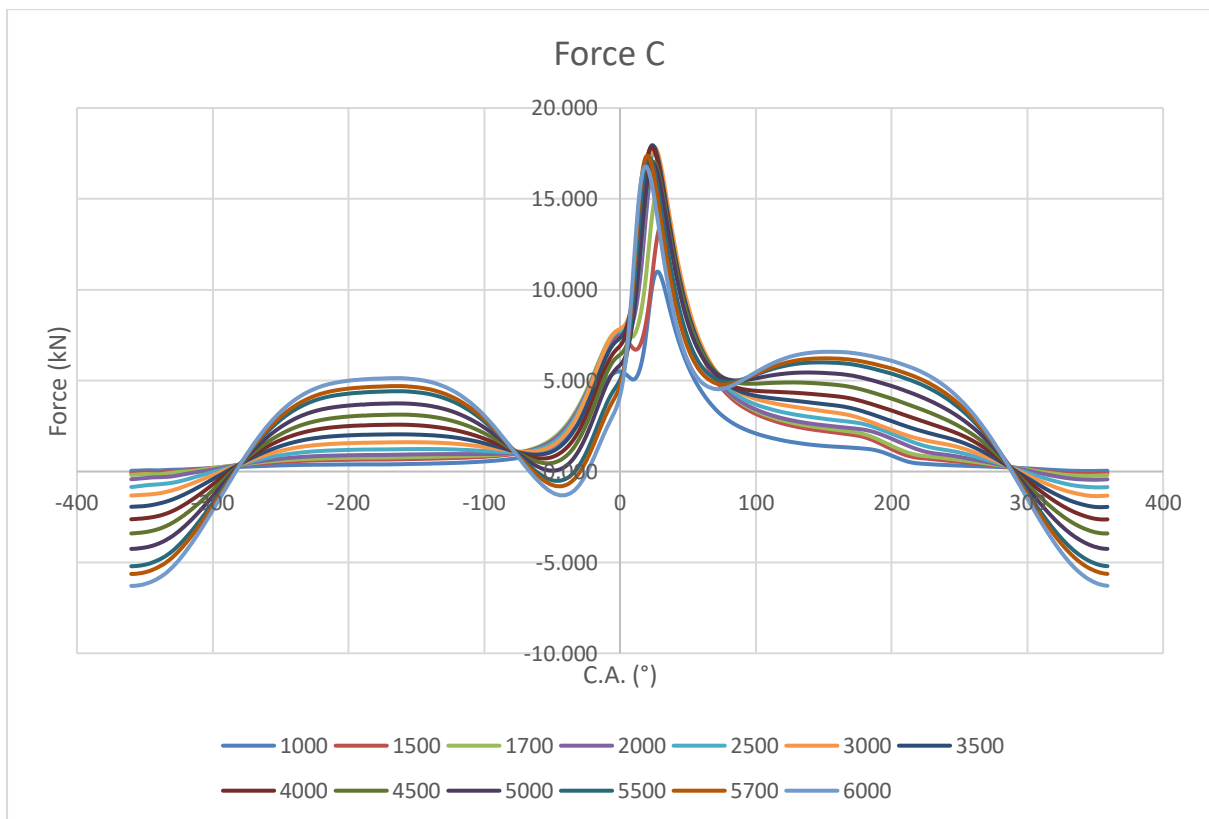
Appendix 3: Connecting Rod A1 Force



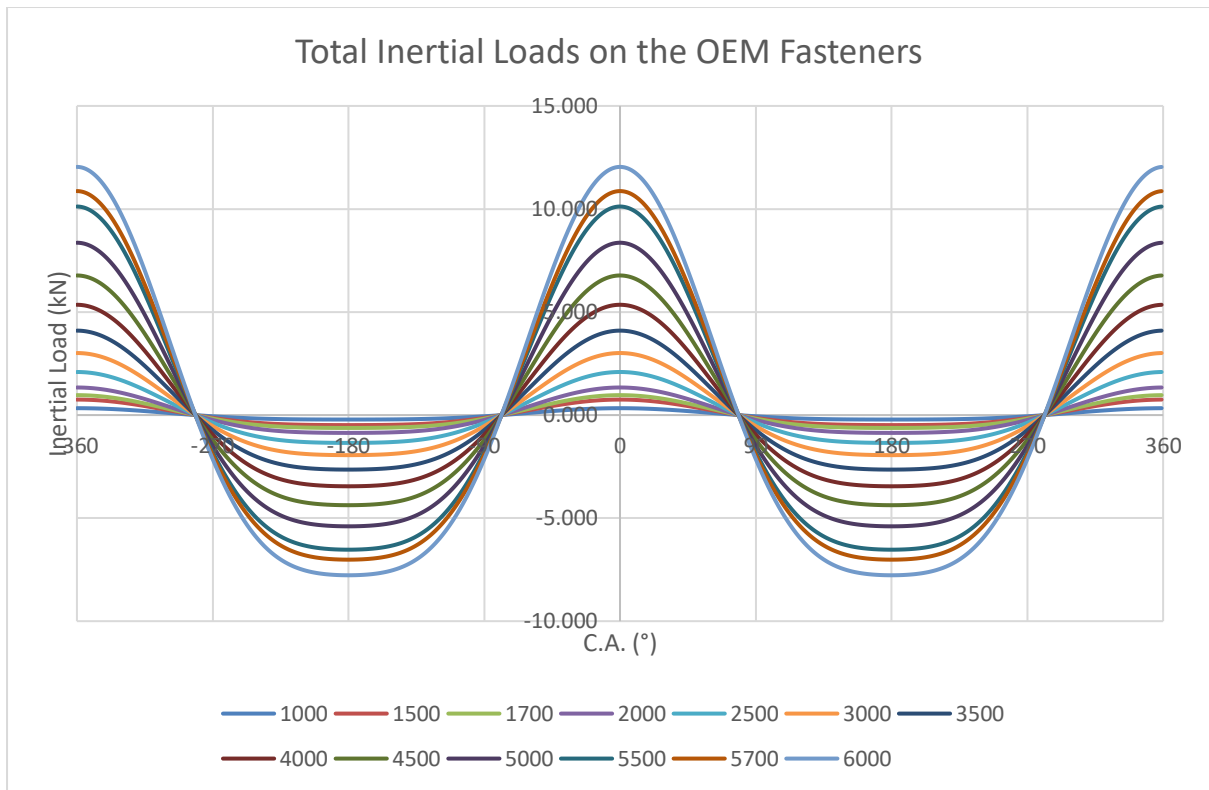
Appendix 4: Connecting Rod A2 Force



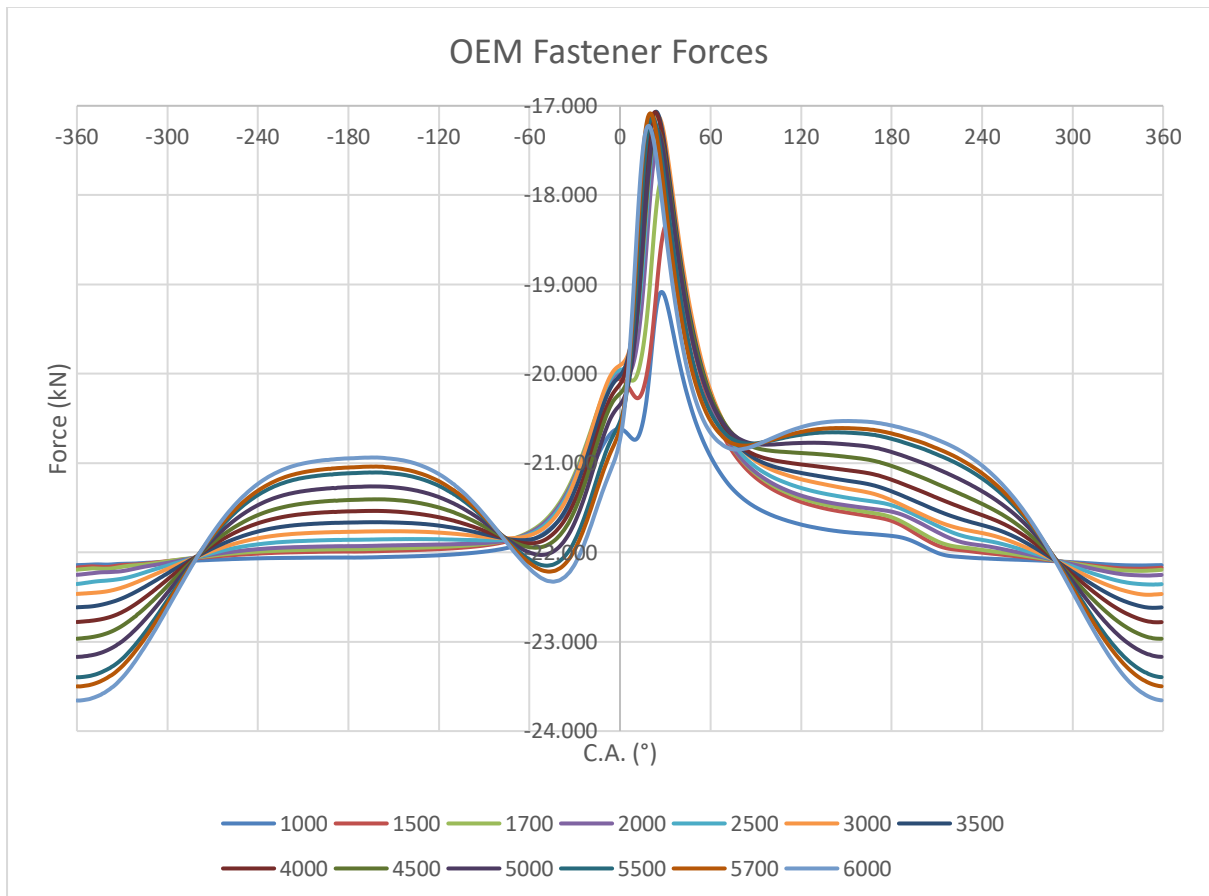
Appendix 5: Connecting Rod B Force



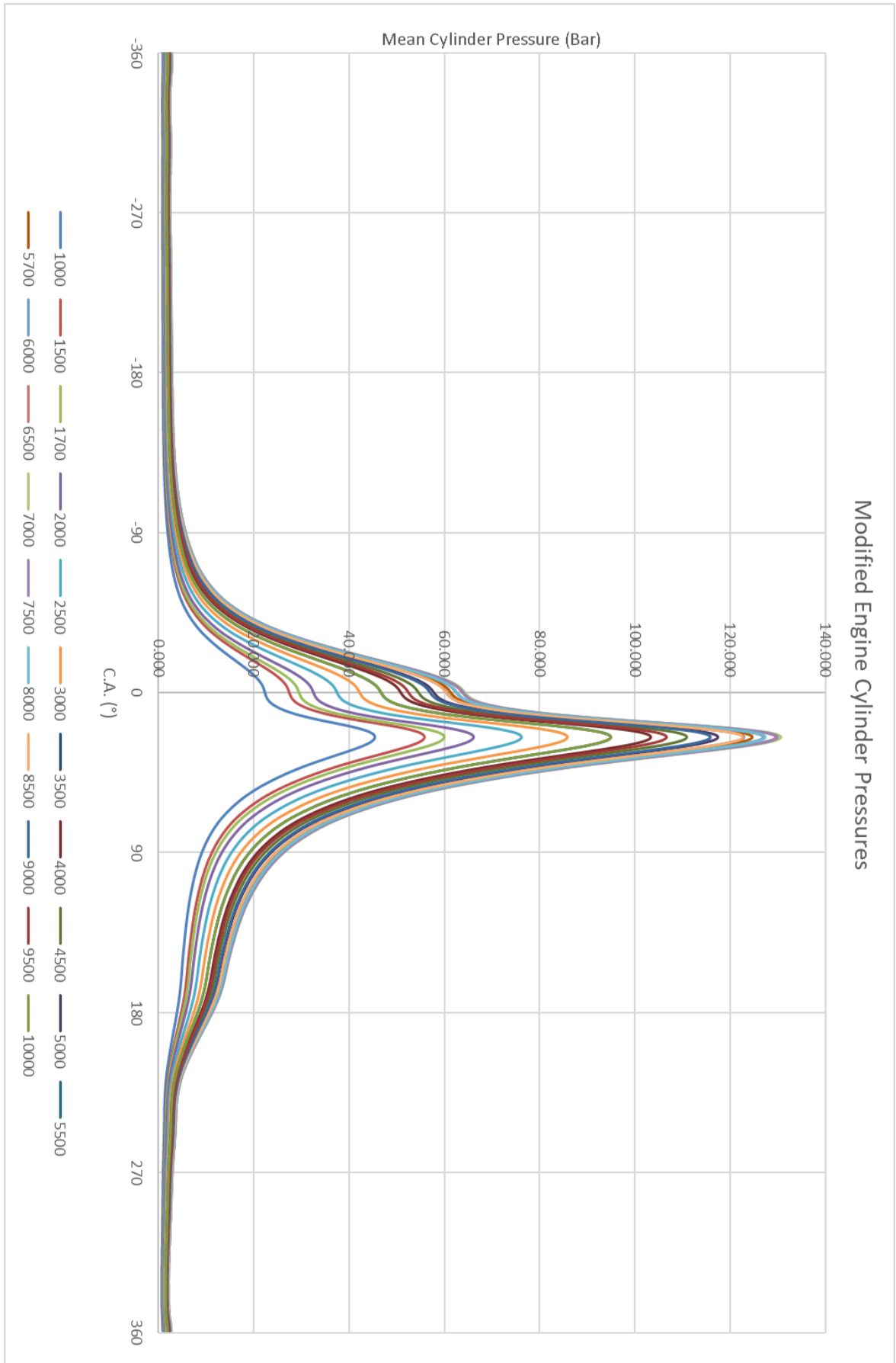
Appendix 6: Connecting Rod C Force



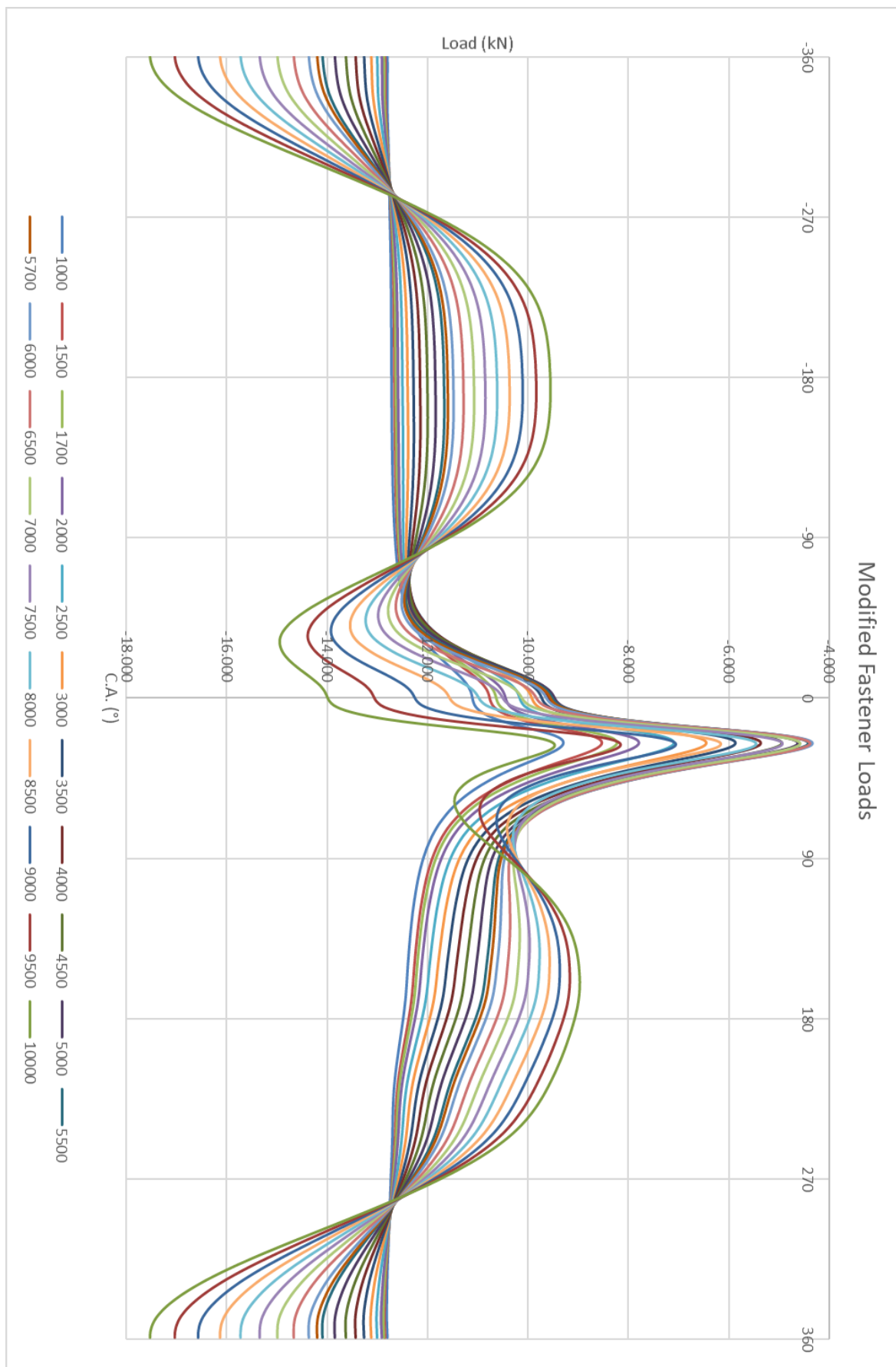
Appendix 7: Total Inertial Loads on the OEM Fasteners



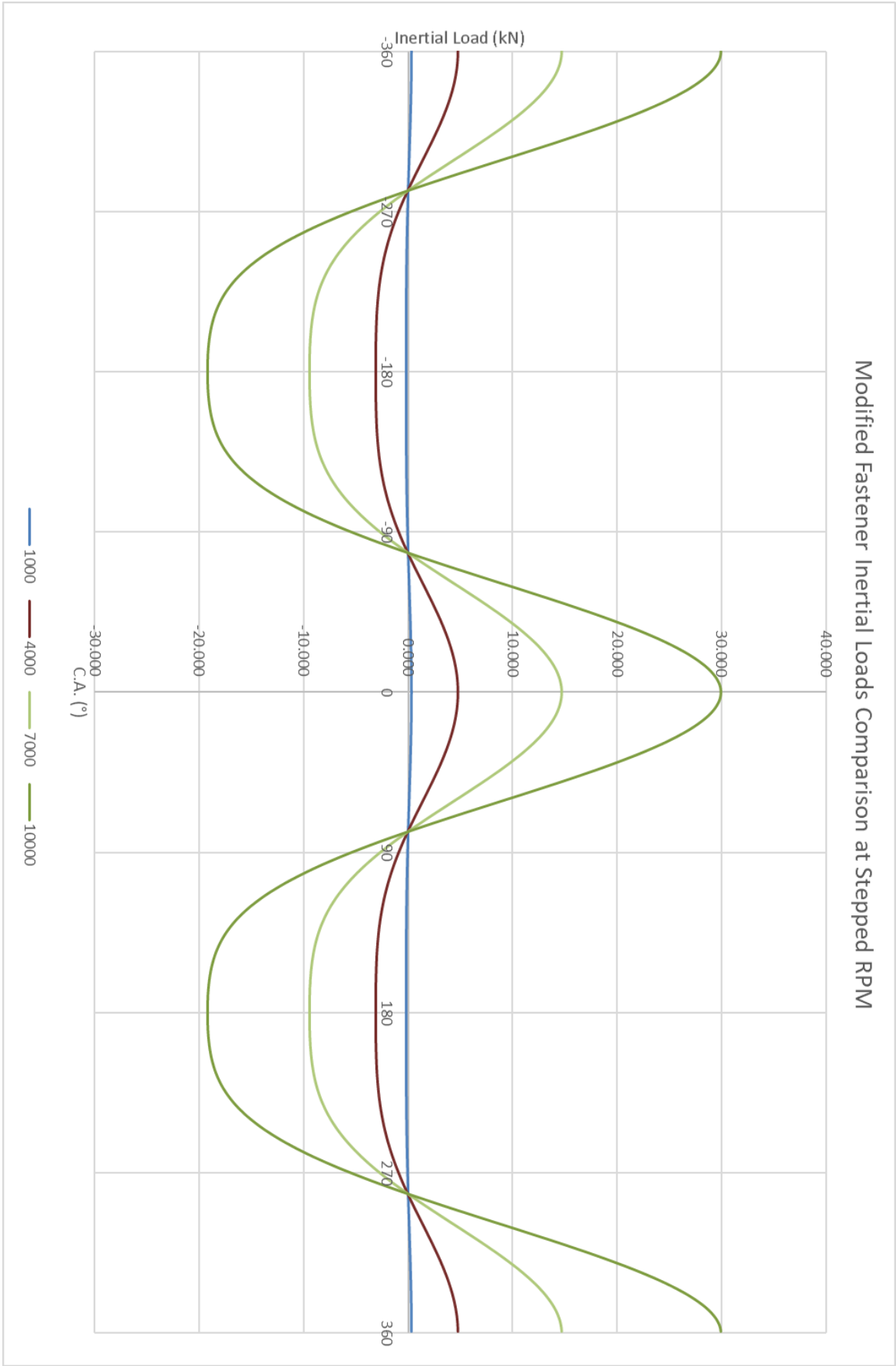
Appendix 8: OEM Fastener Forces



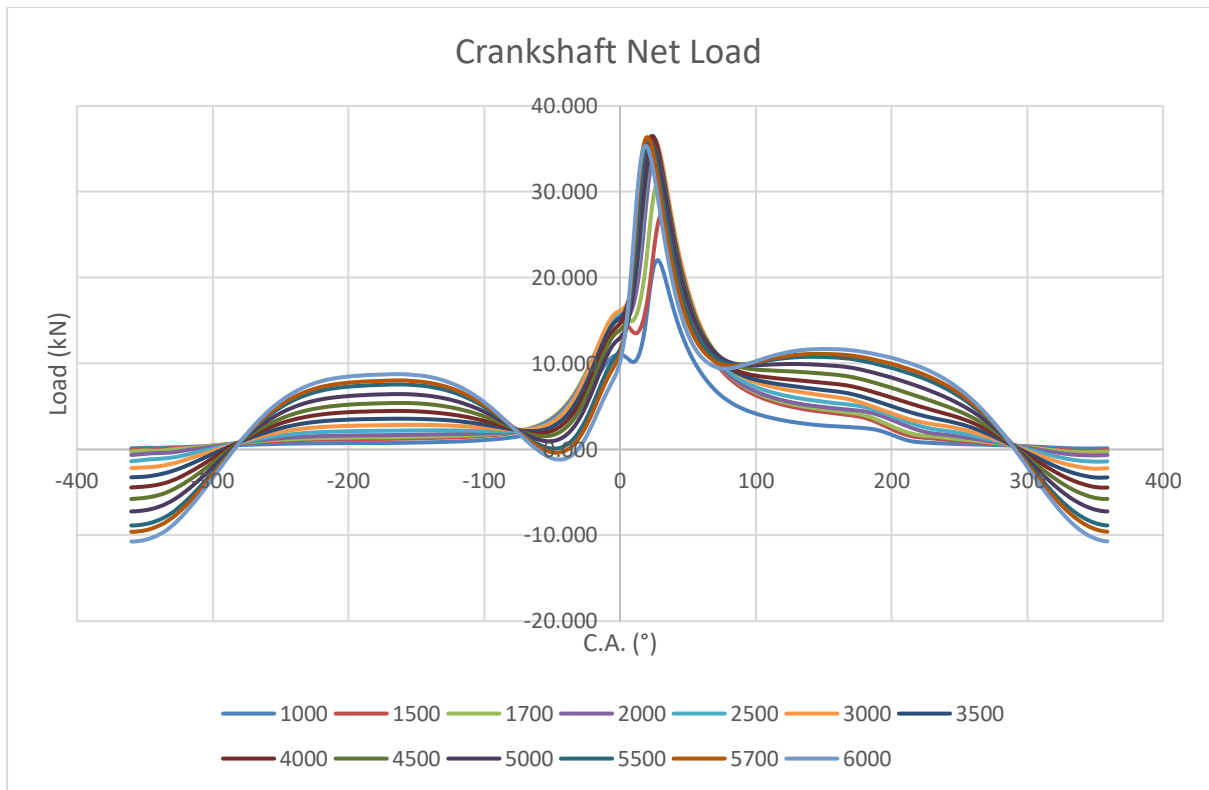
Appendix 9: Modified Engine Cylinder Pressures



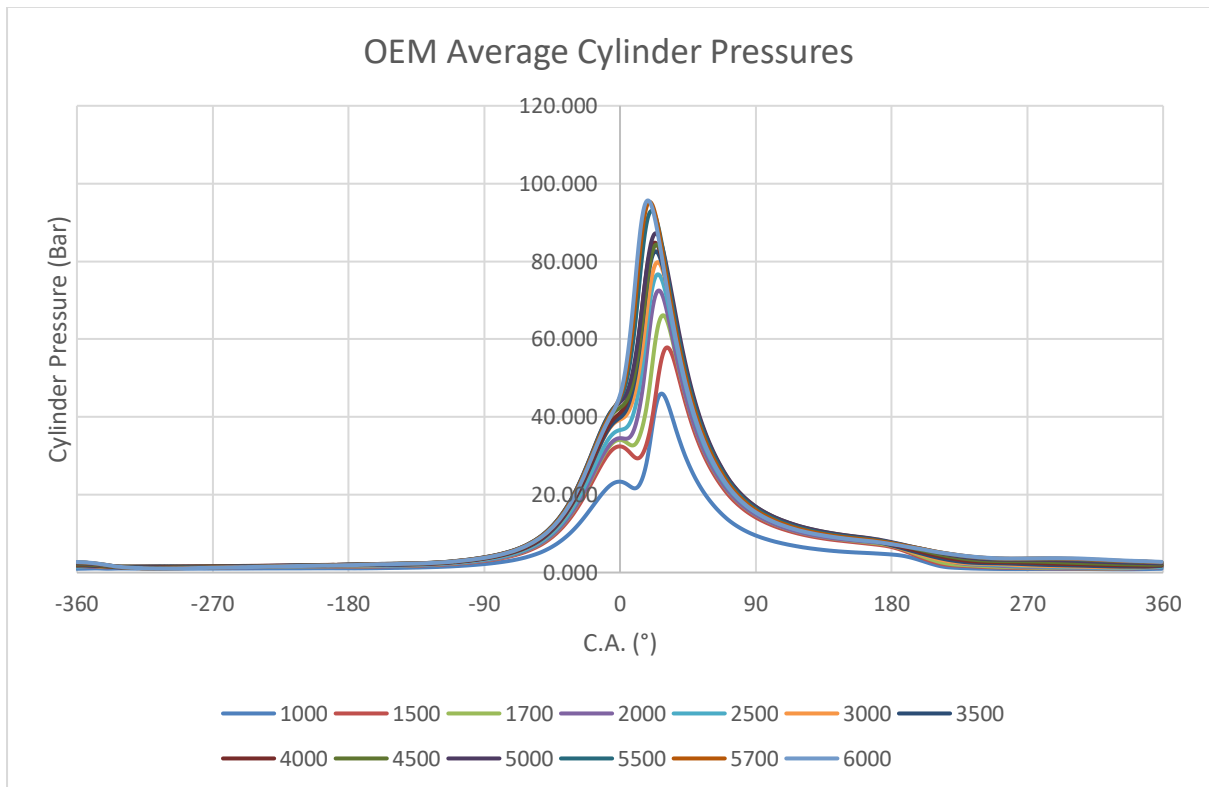
Appendix 10: Modified Fastener Loads



Appendix 11: Modified Fastener Inertial Loads Comparison @ Stepped RPM



Appendix 12: Crankshaft Net Loads



Appendix 13: OEM Average Cylinder Pressures