



University of Wales Trinity Saint David
School of Engineering

Calculations on Piston, Rod & Crankshaft Assemblies

Michael Anderson
P151178

12/11/2019

Advanced Engine Design
M6X00410

Malcolm McDonald

Contents

1) Introduction.....	7
2) Methodology	8
2.1. Gudgeon Pin Forces.....	8
2.2. Connecting Rod Forces	14
2.2.1. Point A.....	14
2.2.2. Point B.....	18
2.2.3. Point C.....	21
2.3. Connecting Rod Bolt Forces	24
2.4. Crankshaft Forces.....	33
3) Results.....	40
3.1. Gudgeon Pin Forces.....	40
3.1.1. Max Net Forces	40
3.1.2. Mean Stress	41
3.1.3. Range of Stress.....	42
3.2. Con Rod Forces.....	43
3.2.1. Point A.....	43
3.2.2. Point B.....	46
3.2.3. Point C.....	49
3.3. Con Rod Cap & Bolts Forces	52
3.3.1. Standard Bolt.....	52
3.3.2. Modified Bolt.....	55
3.4. Crankshaft Forces.....	57
3.4.1. Max Net Forces	57
3.4.2. Mean Stress	58
3.4.3. Range of Stress.....	59
4) Conclusion.....	60
5) References	61

6) Appendix.....	63
6.1. Appendix A – Stress Concentrations Factors for Con Rod Points ‘A’ & ‘B’	63
6.2. Appendix B – Stress Concentrations Factors for Con Rod Point ‘C’	64
6.3. Appendix C – Stress Concentrations Factors for The Con Rod Bolts.....	65
6.4. Appendix D – Stress Concentration Factors for Crankshaft Fillets	66
6.5. Appendix E – Stress Concentration Factors for Crankshaft Fillets	67
6.6. Appendix F – Solidworks Static Study Performed Using 42kN Gas Force.....	68
6.7. Appendix G – Photographs for Crankshaft Measurements.....	69

Table of Tables

Table 1 – Measurements Taken for Ecoboost Gudgeon Pin	10
Table 2 – Specifications for The Con Rod Cap & Bolts	25
Table 3 – Stress Results from The Standard Ford Bolt	30
Table 4 – Results from Study Conducted to See When The Standard Bolts Would Break ..	30
Table 5 – Modified Bolt Calculations and Results.....	32
Table 6 – Measurements and Specifications for The Standard Ford Crankshaft	33
Table 7 –Max Compressive & Tensile Forces Acting on The Gudgeon Pin vs Engine Speed	40
Table 8 – Mean Stress Acting on The Gudgeon Pin vs Engine Speed	41
Table 9 – Range of Stress Acting on The Gudgeon Pin vs Engine Speed.....	42
Table 10 – Max Compressive & Tensile Forces Acting on The Con Rod Point ‘A’ vs Engine Speed	43
Table 11 – Mean Stress Acting on The Con Rod Point ‘A’ vs Engine Speed	44
Table 12 – Range of Stress Acting on The Con Rod Point ‘A’ vs Engine Speed	45
Table 13 – Max Compressive & Tensile Forces Acting on The Con Rod Point ‘B’ vs Engine Speed	46
Table 14 – Mean Stress Acting on The Con Rod Point ‘B’ vs Engine Speed	47
Table 15 – Range of Stress Acting on The Con Rod Point ‘B’ vs Engine Speed	48
Table 16 – Max Tensile Forces Acting on The Con Rod Point ‘C’ vs Engine Speed.....	49
Table 17 – Mean Stress Acting on The Con Rod Point ‘C’ vs Engine Speed.....	50
Table 18 – Range of Stress Acting on The Con Rod Point ‘C’ vs Engine Speed	51
Table 19 – Max Inertia & Net Forces Acting on The Con Rod Bolt vs Engine Speed	52
Table 20 – Extra Force Acting on The Con Rod Bolt vs Engine Speed	53

Table 21 – Alternating Load Acting on The Con Rod Bolt vs Engine Speed	54
Table 22 – Extra Force Acting On The Modified Con Rod Bolts vs Engine Speed	55
Table 23 – Alternating Load Acting on The Modified Con Rod Bolt vs Engine Speed	56
Table 24 – Max Compressive & Tensile Forces Acting on The Crankshaft vs Engine Speed	57
Table 25 – Mean Stress Acting on The Crankshaft vs Engine Speed.....	58
Table 26 – Range of Stress Acting on The Crankshaft Point vs Engine Speed	59

Table Of Figures

Figure 1 - Con Rod & Gudgeon Pin Assembly With Lengths & Point 'A' Annotated (Richardson, 2020)	8
Figure 2 – Goodman diagram for Fatigue Safety Factor for The Gudgeon Pin	12
Figure 3 – CAD Model of The Con Rod With Point 'A' Highlighted (Richardson, 2020)	14
Figure 4 – CAD Model of Con Rod Mass Above Point 'A' (Richardson, 2020).....	14
Figure 5 – Required Dimensions for Calculating Stress Concentration Factors for Point A (Richardson, 2020)	15
Figure 6 – Goodman Diagram for Con Rod Point 'A'	16
Figure 7 - CAD Model of The Con Rod with Point 'B' Highlighted (Richardson, 2020)	18
Figure 8 – Required Dimensions for Calculating Stress Concentration Factors for Point 'B' (Richardson, 2020)	18
Figure 9 - Goodman Diagram for Con Rod Point 'B'	19
Figure 10 - Section View of The Con Rod & Bolt Assembly Highlighting Point 'C' (Richardson, 2020)	21
Figure 11 – Required Dimensions for Calculating Stress Concentration Factors for Point C (Richardson, 2020)	22
Figure 12 – Visual Interpretation of The Stress Concentration Factor Graph for 'Point C' (Pilkey, 1997)	22
Figure 13 - Goodman Diagram for Con Rod Point 'C'	23
Figure 14 – CAD Model of The Con Rod Cap (Left) & Cross-Sectional Area Measurement of The Cap-Rod Interface (Right) (Richardson, 2020)	24
Figure 15 – CAD Model of The Standard Con Rod Bolt with Length To 2 nd Engaged Thread Highlighted (Anderson, 2020).....	24
Figure 16 – Dimensions of An M8 1mm Pitch Threaded Bolt (Bosch, 2002)	26

Figure 17 – Visual Interpretation of The Stress Concentration Factor Graph for The Con Rod Bolts (Pilkey, 1997)	28
Figure 18 - Goodman Diagram for The Con Rod Bolts.....	29
Figure 19 – Modified Bolt Dimensions.....	31
Figure 20 - CAD Models of Both The Standard And Modified Bolts (Anderson, 2020).....	32
Figure 21 - CAD Model of The Standard Ford Crankshaft with Points 'A' & 'B' and Distance 'D' (Richardson, 2020)	33
Figure 22 - Petersons Stress Concentration Diagram (Left) (Pilkey, 1997) & CAD Model of Ford Crankshaft With Dimensions in 'mm' (Right) (Richardson, 2020)	35
Figure 23 - Stress Concentration Factors Calculations For Points 'A' (Left) & 'B' (Right) (Pilkey, 1997).....	36
Figure 24 – Stress Concentration Factor for The Standard Ford Crankshaft (Pilkey & Pilkey, 2008)	37
Figure 25 - Goodman Diagram For Crank Shaft Point 'A'	38
Figure 26 - Goodman Diagram for Crankshaft Point 'B'.....	39
Figure 27 – Gudgeon Pin Max Compressive & Tensile Net Forces vs Engine Speed	40
Figure 28 – Gudgeon Pin Mean Stress vs Engine Speed.....	41
Figure 29 – Gudgeon Pin Range of Stress vs Engine Speed	42
Figure 30 – Max Compressive & Tensile Net Forces Acting on Point 'A' vs Engine Speed .	43
Figure 31 – Max Compressive & Tensile Net Forces Acting on Con Rod Point 'A' vs Engine Speed	44
Figure 32 – Range of Stress Acting on Con Rod Point 'A' vs Engine Speed	45
Figure 33 – Max Compressive & Tensile Net Forces Acting on Point 'B' vs Engine Speed .	46
Figure 34 – Mean Stress Acting on Con Rod Point 'B' vs Engine Speed.....	47
Figure 35 – Range of Stress Acting on Con Rod Point 'B' vs Engine Speed	48
Figure 36 – Max Tensile Net Forces Acting on Point 'C' vs Engine Speed	49
Figure 37 – Mean Stress Acting on Con Rod Point 'C' vs Engine Speed.....	50
Figure 38 – Range of Stress Acting on Con Rod Point 'C' vs Engine Speed	51
Figure 39 – Max Inertia & Tensile Net Forces Acting on Con Rod Bolt vs Engine Speed	52
Figure 40 – Extra Force Acting on Con Rod Bolt vs Engine Speed	53
Figure 41 – Alternating Load Acting on Con Rod Bolt vs Engine Speed.....	54
Figure 42 – Extra Force Acting on The Modified Con Rod Bolt vs Engine Speed.....	55
Figure 43 – Alternating Load Acting on Modified Con Rod Bolt vs Engine Speed	56
Figure 44 – Max Compressive & Tensile Net Forces Acting On The Crankshaft vs Engine Speed	57
Figure 45 – Mean Stress Acting on Crankshaft Point 'A' vs Engine Speed.....	58
Figure 46 – Range of Stress Acting on Crankshaft Point 'A' vs Engine Speed	59

Figure 47 – Stress Concentration Factors for A Stepped Flat Tension Bar with A Shoulder Fillet (Pilkey, 1997)	63
Figure 48 – Stress Concentration Factors for A Semi-Infinately Thin Element with A Deep Hyperbolic Notch (Pilkey, 1997)	64
Figure 49 – Stress Concentration Factors for A deep Hyperbolic Groove In An Infinitely Wide Member (Pilkey, 1997)	65
Figure 50 – Stress Concentrations Factors for A Crankshaft In Bending (Pilkey, 1997)	66
Figure 51 – Stress Concentration Factors for A U-Shaped Grooved Shaft In Bending (Pilkey & Pilkey, 2008).....	67
Figure 52 – SolidWorks Static Study Showing Stress & Strain Results	68
Figure 53 – Measuring The Horizontal Distance From Point 'A' To 'B' (Anderson, 2020).....	69
Figure 54 – Measuring The Angled Distance From Point 'A' to 'B' (Anderson, 2020).....	69
Figure 55 – Measuring The Diameter Of The Crank Journal (Anderson, 2020)	70
Figure 56 – Measuring The Length Of The Crank Journal (Anderson, 2020)	70
Figure 57 – Measuring The Diameter Of The Big End / Crank Pin (Anderson, 2020)	70
Figure 58 – Measuring The Length Of The Big End / Crank Pin (Anderson, 2020).....	70
Figure 59 – Measuring The Diameter Of The Crank Journal Bearing Shell (Anderson, 2020)	70
Figure 60 – Measuring The Length Of The Crankshaft Bearing Shell (Anderson, 2020)	70

1) Introduction

An experiment was conducted into multiple components that are vital to an engine's operation.

These include:

- Gudgeon Pin
- Connecting Rod
- Connecting Rod Bolts
- Crankshaft

Each component was examined separately within a theoretical environment using data provided by the Ford Motor Company from their 1.6L Ecoboost engine.

The inertia force from each component was calculated then subtracted from the gas force to create a net force. This aided in determining the stress that was exerted throughout each engine cycle.

Stress concentration factors were acknowledged and accounted for throughout this experiment, each section of this report features a snippet of each graph used for the calculations, the full graphs have been included within the Appendix.

The results from this entire experiment have been analysed to find common trends and occurrences throughout each component.

2) Methodology

2.1. Gudgeon Pin Forces

To begin this experiment, measurements were taken of the Ford 1.6 Ecoboost engine Piston, Gudgeon pin, Connecting Rod (Con Rod) & Bolts as well as the Crank Shaft, by Michael Anderson and Danny Richardson. These measurements allowed the creation of CAD models that accurately represent the engines components, provided for this report courtesy of Danny Richardson. The CAD files were used throughout this study to ascertain measurements, areas and masses which correlated to those taken in real-life.

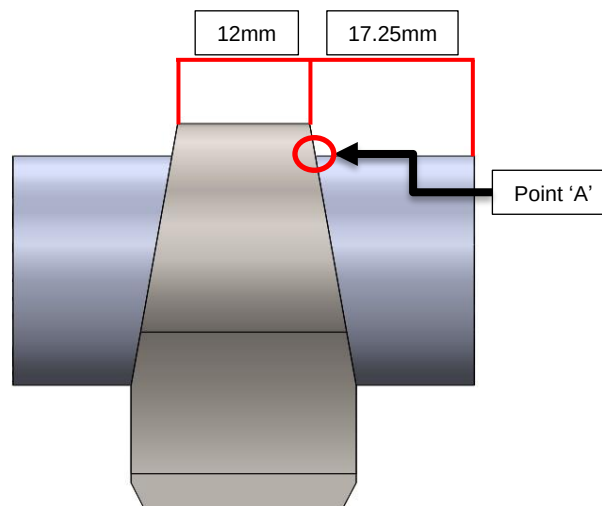


Figure 1 - Con Rod & Gudgeon Pin Assembly With Lengths & Point 'A' Annotated (Richardson, 2020)

To begin with, the inertia of the gudgeon pin needed to be calculated. To simplify the calculations, only one side of the pin was considered, this meant that the mass of the overall pin could be halved for the equations used below.

The overall length of the gudgeon pin was 46.5mm long, when halved this was 23.25mm, the top point of the con rod (Figure 1) was 12mm wide, taken as 6mm when halved. When subtracted from the half length of the gudgeon pin, this left 17.25mm of the pin sticking out from the con rod at point 'A'. By using this length, together with the inner and outer diameters of the gudgeon pin, converted into radii, the following equations can be used to attain vital information.

$$\text{Mass of Pin Sticking Out (kg)} = \text{Length} * (\pi * (R_2^2 - R_1^2)) * 7800$$

2.1-1

Where:

Length = Length of the gudgeon pin sticking out (m)

R_2 = Outer radius of the pin (m)

R_1 = Inner radius of the pin (m)

7800 = Density of steel (kg/m^3)

Now the inertia created by the mass of the gudgeon pin sticking out from its reciprocating motion could be attained, this was created into a column within Excel for each engine speed (1000 – 6000RPM).

$$\text{Pin Inertia (N)} = \text{Mass of Pin Sticking Out} * \text{Acceleration}$$

2.1-2

As:

$$\text{Accerelation (m/s}^2\text{)} = (R\omega^2\text{Cos}\theta) + \left(\frac{R}{L}R\omega^2\text{Cos}2\theta\right)$$

2.1-3

Where:

R = Crank radius/throw (half engine stroke) (m)

L = Length of the connecting rod (m)

ω = Engine speed (rad/s)

θ = Crank angle per degree ($^\circ$)

The force created by the gas within the engine cylinder was previously calculated by means of the following formula.

$$\text{Gas Force (N)} = P * \frac{\pi * \text{Bore}^2}{4}$$

2.1-4

Where:

P = Average Cylinder Pressure per degree of crank angle (Pa)

Bore = Engine Cylinder Bore (m)

To determine the correct Net Force applied to the gudgeon pin

$$\text{Net Force (N)} = \text{Gas Force} - \text{Piston Inertia}$$

2.1-5

Piston Inertia is calculated the same way as the pin inertia, but instead of the mass of the gudgeon pin, the mass of the piston minus the gudgeon pin is multiplied by the acceleration.

Using the CAD models previously created, a measurement was taken to the centre of the piston bore from where the end of the gudgeon pin sits, to the inside edge of the piston. This enabled the collection of the length from point 'A' to the centre of the piston bore, where the downforce applied by the gas force is applied. By modifying the CAD model of the gudgeon pin, half of the pin was removed, then by means of the 'Mass Properties' manager, the centre of mass (COM) could be obtained and such, the length from the centre of the pin to the COM.

This was then used to formulate the Net Bending moment applied to point 'A'. A column of the bending moment around point 'A' was created within Excel to attain the results per degree of crankshaft rotation. Where the maximum value equated to the most tensile force applied to the pin, and the minimum value represented the most compressive force.

$$\text{Bending Moment (Nmm)} = \left(\frac{\text{Net Force}}{2} * L_1\right) - (\text{Pin Inertia} * L_2)$$

2.1-6

Where:

L_1 = Length from point 'A' to the centre of the piston bore (mm)

L_2 = Length from point 'A' to the centre of mass (COM) (mm)

The neutral axis (notated by 'y') represents the distance from the outer edge of the object, to the exact centre line of the object that is neither stretched nor compressed and where the longitudinal stress is zero. (Merriam Webster, 2020)

$$y \text{ (mm)} = \frac{\text{Outer Diameter}}{2} = \frac{21}{2} = 10.5\text{mm}$$

2.1-7

The second moment of area (notated by 'I') is a measure of the effectiveness of the cross-sectional area of a shape to resist the bending caused by the loading of a force upon the shape, the bigger the number, the more resistance to bending the shape has. (Learn Easy, 2020)

$$I \text{ (mm}^4\text{)} = \frac{\pi}{64} * (D_2^4 - D_1^4) = \frac{\pi}{64} * (21^4 - 14^4) = 7660.82\text{mm}^4$$

2.1-8

Where:

D_1 = The inner diameter of the gudgeon pin (mm)

D_2 = The outer diameter of the gudgeon pin (mm)

Gudgeon Pin Measurements		
Pin Outer Diameter	21.0	mm
Pin Inner Diameter	14.0	mm
Pin Distance To Neutral Axis	10.50	mm
Pin Second Moment Of Area	7660.82	mm ⁴
Pin Length	46.5	mm
Half Pin Length	23.25	mm
Pin Weight	79	g
Piston on Pin (1 Side)	15	mm
Half Piston On Pin	7.25	mm
Con Rod Small End Top Thick	12.0	mm
Half Small Top End	6	mm
Con Rod Small End Bottom Thick	19	mm
Half Small Bottom End	9.5	mm
Pin Sticking Out Con Rod	17.25	mm
Pin Length To Piston Centre	9.75	mm
Length To COM	5.27	mm
Weight Sticking Out	0.02589	Kg

Table 1 – Measurements Taken for The Ecoboost Gudgeon Pin

The bending stress (notated by 'σ') around 'Point A' can be attained by using the respective compressive or tensile moment to account for the stress seen throughout the gudgeon pin.

$$\sigma \text{ (N/mm}^2\text{)} = \frac{\text{Bending Moment} * y}{\text{Second Moment of Area}}$$

2.1-9

Shear stress (notated by 'τ') can also be equated by using half of the net force applied to the gudgeon pin, then dividing it by the area of the pin between the two diameters.

$$\tau = \frac{\frac{\text{Net Force}}{2}}{\frac{\pi}{4} * (D_2^2 - D_1^2)}$$

2.1-10

Both bending (σ) and shear (τ) stresses can be combined into a single value.

$$\text{Combined Stress} = \sqrt{\sigma^2 + (2 * \tau)^2}$$

2.1-11

The two answers from equation 2.1-11, represent the maximum (tensile / σ_{Max}) or minimum (compressive / σ_{Min}) stresses exerted onto the gudgeon pin throughout the engine cycle, shown below:

$$\sigma_{Max} = 365.09 \text{ N/mm}^2$$

$$\sigma_{Min} = -42.54 \text{ N/mm}^2$$

With both of these values, both the range and mean stress can now be equated to help gain an understand of the alternating stress that the gudgeon pin is under during the engine's operation.

$$\text{Range of Stress (N/mm}^2\text{)} = \sigma_{Max} - \sigma_{Min} = 365.09 - (-42.54) = 407.63 \text{ N/mm}^2$$

2.1-12

$$\text{Mean Stress } (\sigma_{Mean}) \text{ (N/mm}^2\text{)} = \frac{\sigma_{Max} - \sigma_{Min}}{2} = \left(\frac{365.09 + 42.54}{2} \right) = 161.27 \text{ N/mm}^2$$

2.1-13

Since the Ultimate Tensile Strength (UTS) of the gudgeon pin was known to be 670 MPa for 4130 Normalised Steel (Aerospace Specification Metals Inc., 2020), the Endurance Limit (EL) of the gudgeon pin can be equated. The standard utilisation of the Fatigue / Endurance Limit for Cast Iron / Steel is to multiply the UTS value by 0.45 (Carvill, 1993), this value is the “maximum amplitude of stress you can go up to, from mean of zero, and down to and last forever” (McDonald, 2019).

$$\text{Endurance Limit (N/mm}^2\text{)} = 0.45 * \text{UTS} = 0.45 * 670 = 301.5 \text{ N/mm}^2$$

2.1-14



From Triangles OXY & OAB:

2.1-15

2.1-16

2.1-17

$$XY = \text{Allowable Range} \quad 2.1-18$$

$$OX = UTS - \sigma_{Mean} \quad 2.1-19$$

$$OA = UTS \quad 2.1-20$$

Giving:

$$\text{Allowable Range (N/mm}^2\text{)} = 2 * EL * \left(1 - \frac{\sigma_{Mean}}{UTS}\right) \quad 2.1-21$$

So:

$$\text{Allowable Range (N/mm}^2\text{)} = (2 * 301.5) * \left(1 - \left(\frac{161.27}{670}\right)\right) = 457.85 \text{ N/mm}^2 \quad 2.1-22$$

To determine whether a component can withstand the forces and alternating stresses exerted upon it, by comparing the allowable range against the actual range calculated within equation 2.1-12. The Fatigue Safety Factor (FSF), shown in equation 2.1-23 compares these values as a ratio, if this value is below '1' the part cannot tolerate the amount of stress seen within the situation, whereas, the higher above the value '1', this part would theoretically last forever, or at the very least, outlast other components. (Jelaska, 2011)

$$FSF = \frac{\text{Allowable Range}}{\text{Actual Range}} = \frac{457.85}{407.63} = 1.123 \quad 2.1-23$$

Since the gudgeon pin has an FSF above 1, it should theoretically withstand all the forces, pressures and stresses seen within the standard engine.

2.2. Connecting Rod Forces

This portion of the experiment was conducted upon the connecting rod. By picking 3 unique points on the con rod, the forces, stresses, their respective Stress Concentration Factors (SCF) as well as their FSF ratios, such as those done previously. Most of the methodology for this section is extremely similar, as a result, all that is described within section 2.2.1 can be assumed to be carried on for points 'B' & 'C'.

2.2.1. Point A

As shown in **Figure 3**, point 'A' was selected on the thickest part of the con rod shank, just under the gudgeon pin hoop, where the full amount of the net force is being applied, with both compression and tensile loads.

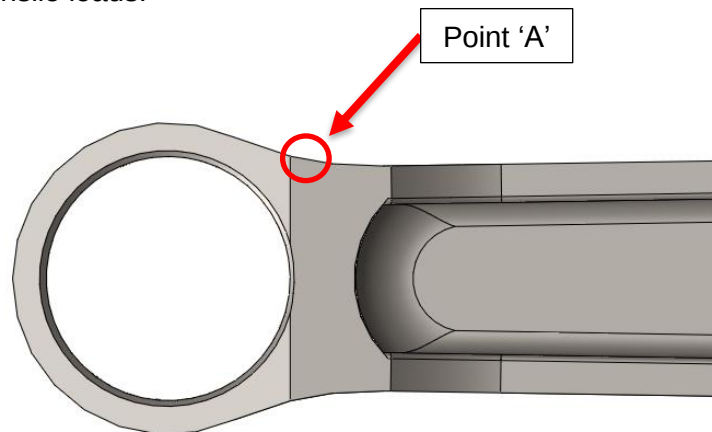


Figure 3 – CAD Model of The Con Rod With Point 'A' Highlighted (Richardson, 2020)

The first action taken was to determine the cross-sectional area. Using the CAD models created previously, the function 'Section View' became handy by effectively 'cutting' the model, then using the 'Measure' tool, the area of the face is calculated with ease for use within the upcoming equations.

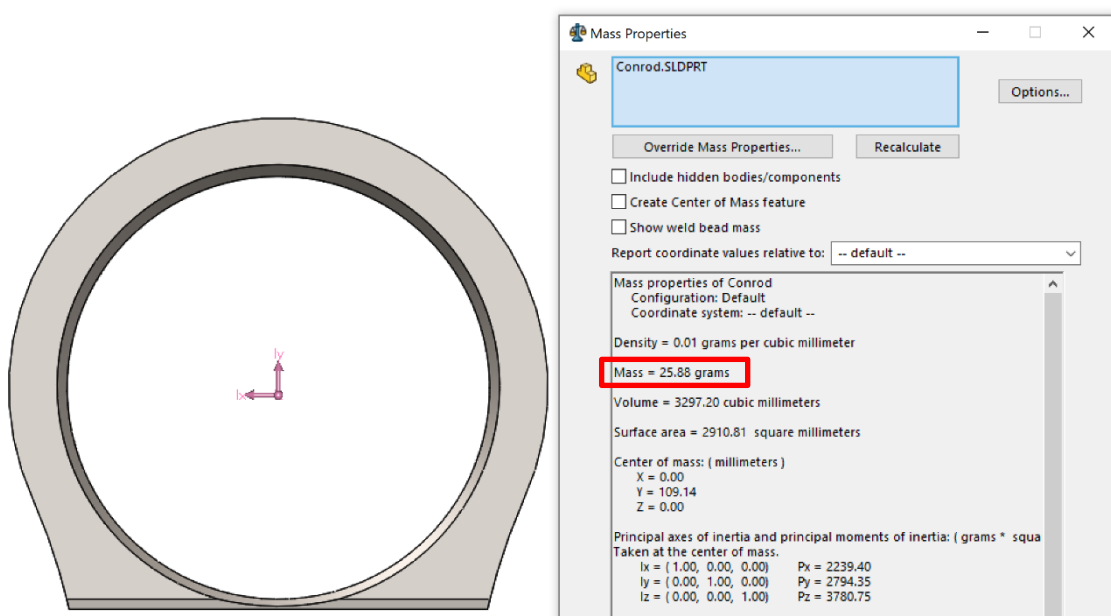


Figure 4 – CAD Model of Con Rod Mass Above Point 'A' (Richardson, 2020)

Now by creating a plane at point 'A', a sketch was created around the part, then 'Extrude Cut' the mass of the con rod hoop above 'Point A' could be measured by means of the 'Mass Properties' tool (Figure 4).

Armed with the extra mass, a new inertia force could be calculated using the acceleration from equation 2.1-3, including the mass of both the piston and the gudgeon pin. As before, this was created within a column in Excel over each degree of crankshaft rotation, This was subtracted from the gas force (equation 2.1-4) to create a new net force, as within equation 2.1-5.

Now the same process as within section 2.1 was applied. Compressive and tensile stresses were calculated using the respective maximum and minimum net force applied to the rod shank and dividing by the area. Where compression stress is negative and tensile is positive.

$$\text{Nominal Stress (N/mm}^2\text{)} = \frac{\text{Net Force}}{\text{Cross - Sectional Area}}$$

2.2-1

The mean and range of stresses are equated using the same method as previously shown in equations 2.1-12 & 2.1-13 combining both the compressive and tensile forces.

In the previous section, the endurance limit method shown in equation 2.1-14, is known as the polished endurance limit since it neglects any radius or notches within the component. This time however, the con rod has multiple radii so each point, ('A' 'B' & 'C') so to accurately represent stress concentrations within the component the Endurance Limit (EL) is divided by the Stress Concentration Factor (SCF), if applicable, to create the Final Endurance Limit (FEL).

$$FEL = \left(\frac{0.45 * UTS}{SCF} \right)$$

2.2-2

The SCF was calculated with the aid of "Peterson's Stress Concentration Factors" (Pilkey, 1997) page 150, where a graph showing the ability to calculate the stress concentration for a stepped flat tension bar featuring a shoulder fillet (Appendix 6.1, Figure 47).

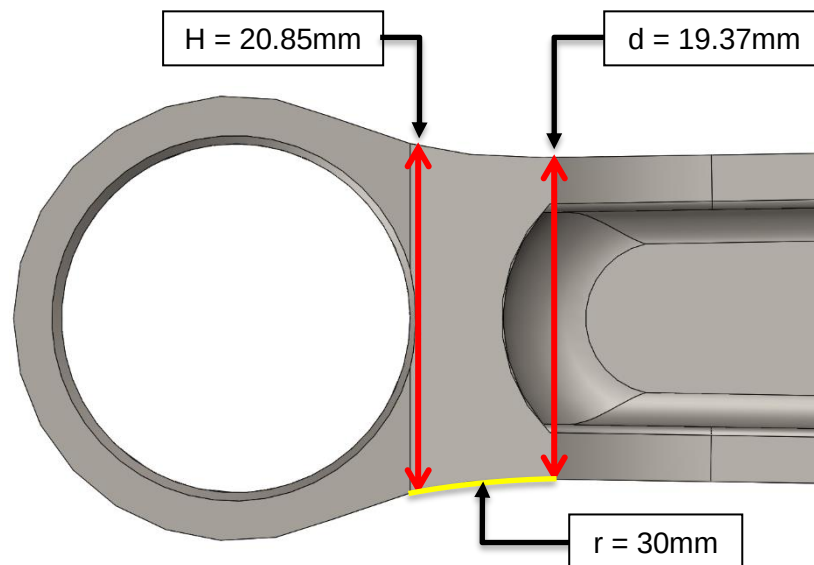


Figure 5 – Required Dimensions for Calculating Stress Concentration Factors for Point A (Richardson, 2020)

Using the CAD model of the con rod, measurements were taken from point 'A' relative to those seen within **Figure 47**, notated as 'H', 'd' and 'r', the measurements are shown within (**Figure 5**). Now the appropriate line can be selected from within the graph.

$$\text{Appropriate Stress Line} = \frac{H}{d} = \frac{20.85}{19.37} = 1.076$$

2.2-3

This enabled the selection of the correct line from within the graph to choose in relation to the SCF. Next, the precise point along the x-axis of the graph should be calculated.

$$X - \text{Axis Value} = \frac{r}{d} = \frac{30}{19.37} = 1.548$$

2.2-4

The highest value along the x-axis of the graph is 0.3, considering the value for point 'A' equated to 1.548, meaning the stress concentration was small enough to be neglected, due to the greatness of the radius throughout this point.

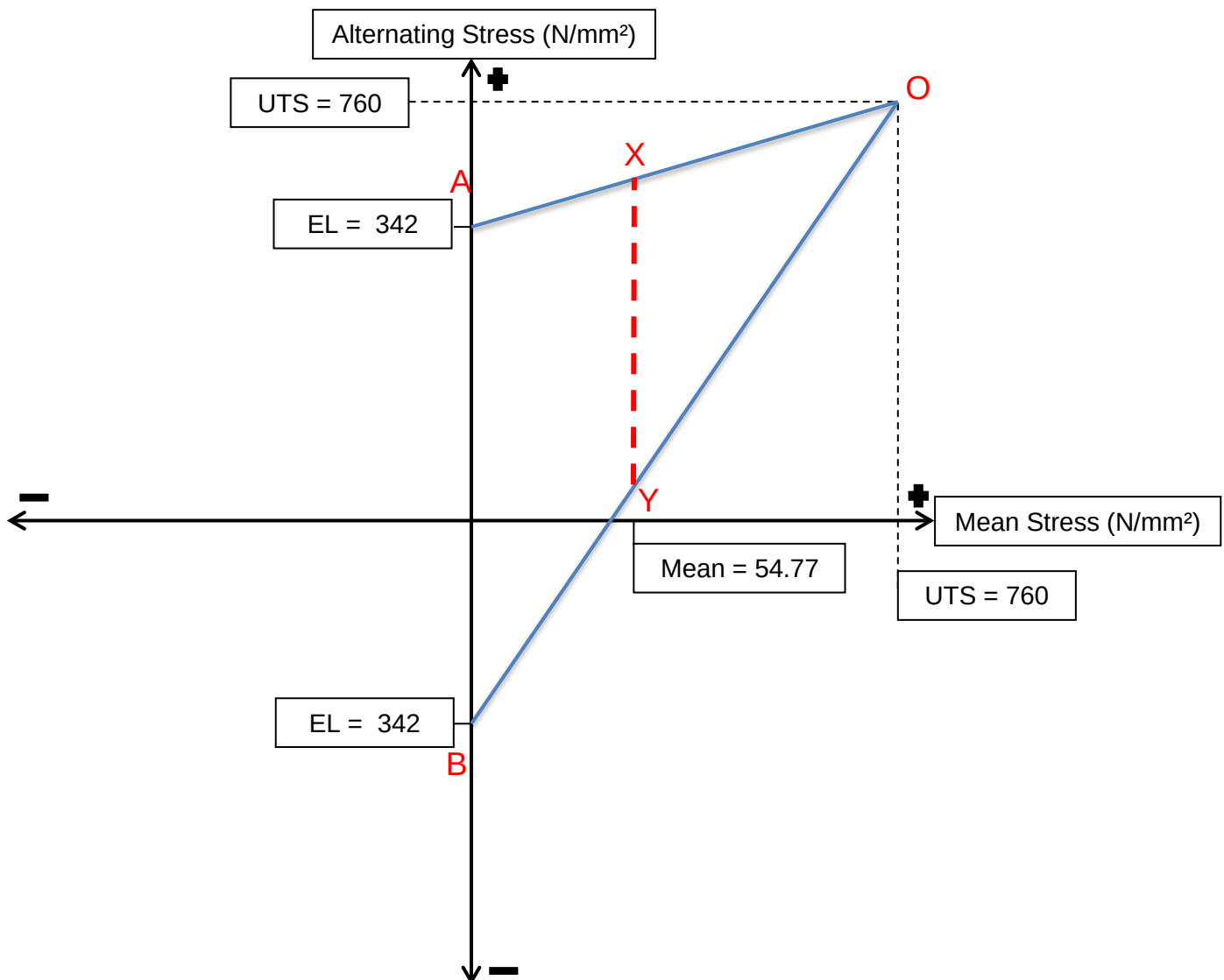


Figure 6 – Goodman Diagram for Con Rod Point 'A'

A Goodman diagram could now be created to determine the fatigue safety factor using the same method by translation of triangles and the ratio of allowable over actual range, as described previously in Gudgeon Pin Forces 2.1. This led to the result seen within equation 2.2-5.

$$FSF = \frac{\text{Allowable Range}}{\text{Actual Range}} = \frac{366.65}{85.63} = 4.28$$

2.2-5

This FSF shows that point 'A' of the con rod is significantly strong enough to resist both the compressive and tensile forces exhibited throughout the engine's operation.

2.2.2. Point B

Shown within **Figure 7** is where point 'B' was selected. This was at the bottom of the 'I' beam shank which is where the most amount of compressive force is displayed, as seen within the Solidworks study (**Appendix 6.6**).

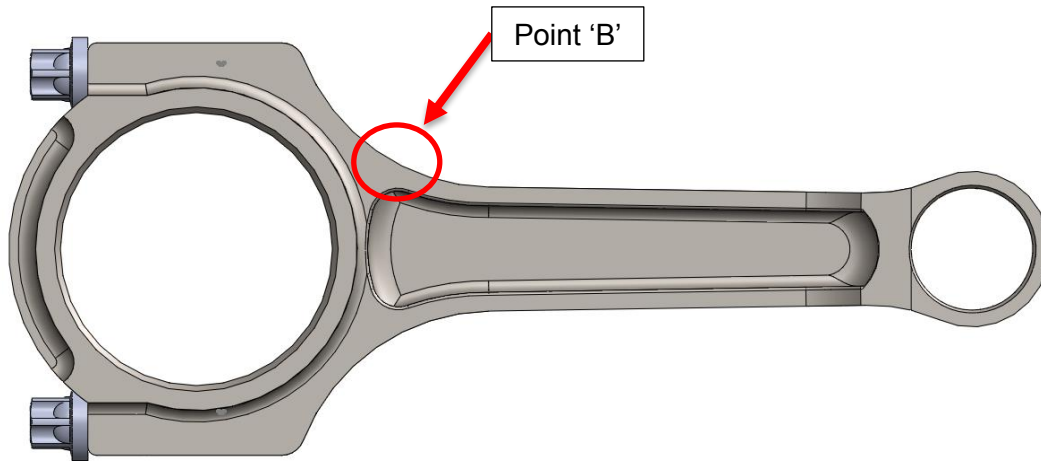


Figure 7 - CAD Model of The Con Rod with Point 'B' Highlighted (Richardson, 2020) (Anderson, 2020)

The same process of attaining the cross-sectional area and mass from within Solidworks was taken as above. These measurements enabled the creation of a new inertia force, and hence, a new net force which would show different compressive and tensile stresses through point 'B', leading to new mean and range stresses using the same process as for point 'A'.

Now the stress concentration needed to be calculated for point 'B', as this would affect the endurance limit, and thus, the allowable range. The measurements are shown below within **Figure 8**.

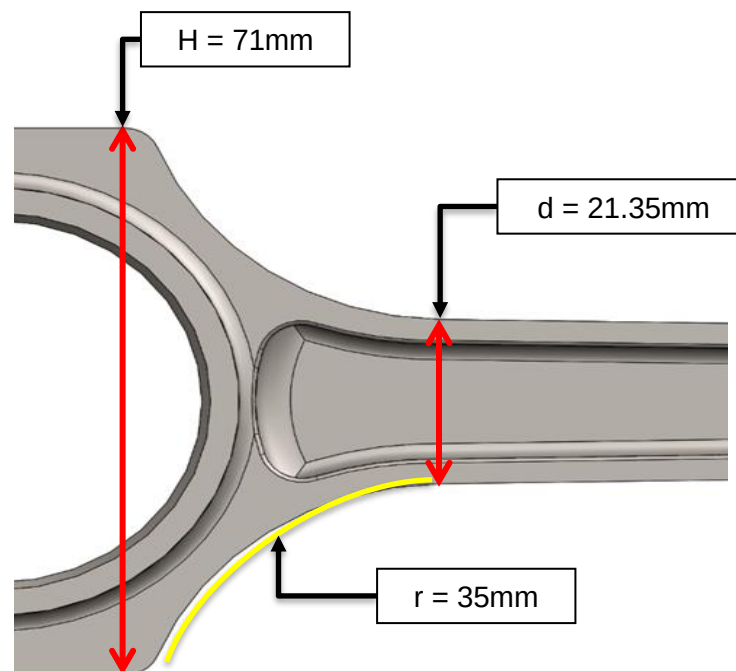


Figure 8 – Required Dimensions for Calculating Stress Concentration Factors for Point 'B' (Richardson, 2020)

This enabled the selection of the correct stress line to follow within the graph.

$$\text{Appropriate Stress Line} = \frac{H}{d} = \frac{71}{21.35} = 3.326$$

2.2-6

The value for the lines within the graph only go up to '2', so this was selected although it may be low in comparison. Next, the precise point along the x-axis of the graph should be calculated.

$$X - \text{Axis Value} = \frac{r}{d} = \frac{35}{21.35} = 1.639$$

2.2-7

The highest value along the x-axis of the graph is 0.3, considering the value for point 'B' equated to 1.639, meant that the stress concentration was small enough to be neglected, due to the greatness of the radius and the ratio of ' H/d ' throughout this point.

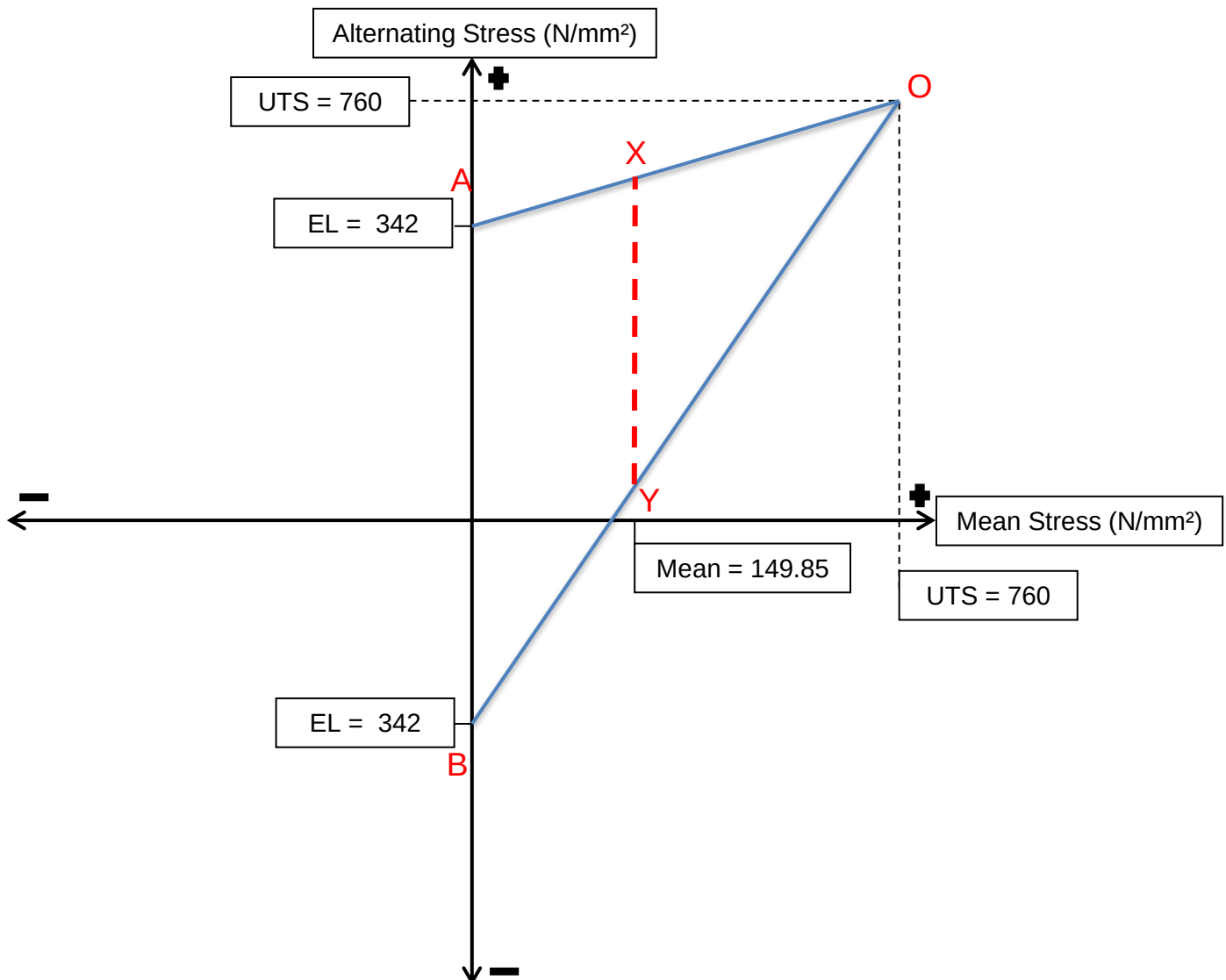


Figure 9 - Goodman Diagram for Con Rod Point 'B'

$$FSF = \frac{\text{Allowable Range}}{\text{Actual Range}} = \frac{409.43}{186.87} = 2.19$$

2.2-8

This FSF shows that point 'B' of the con rod is strong enough to resist the forces applied to the con rod throughout the standard engine cycle. Out of the three points chosen from throughout the con rod, point 'B' sees the worse safety factor due to the reduction of area experiencing the same compressive and tensile forces.

2.2.3. Point C

Throughout the con rod, the worst concentration of stress should theoretically be exhibited upon the small radius that is seen near the face of the con rod bolt, shown within (Figure 10).

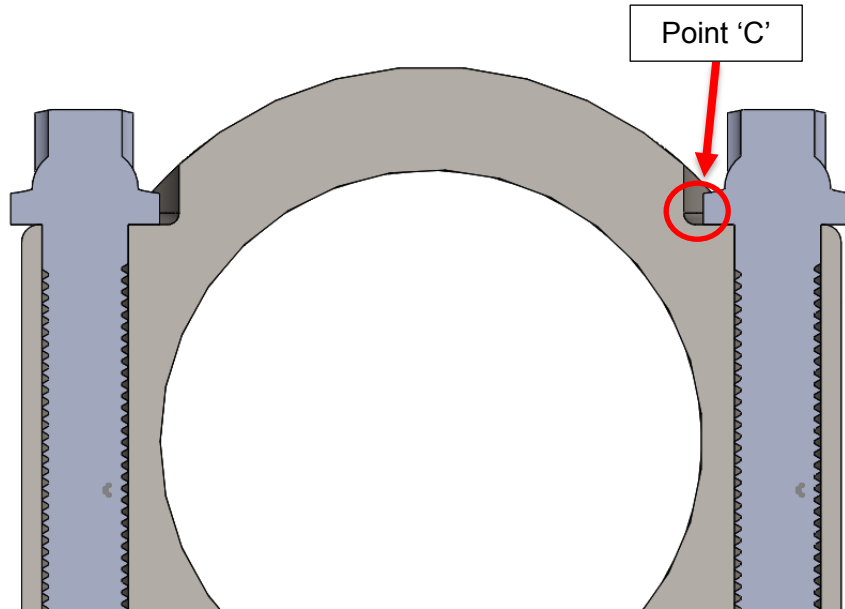


Figure 10 - Section View of The Con Rod & Bolt Assembly Highlighting Point 'C' (Richardson, 2020) (Anderson, 2020)

Now, because the radius is situated on the cap of the con rod, there is no compressive force applied to the cap, only tensile, representing σ_{Max} , therefore the compressive force (σ_{Min}) equals zero. Only one side of the cap is being considered, the net force used within the equations for calculating the stress is halved, as the same force would be applied to the opposite side equally.

For this portion, the acceleration now includes the rotating mass as well as the reciprocating.

$$Acceleration = M_{Rec} * R\omega^2 \left(1 + \frac{R}{L}\right) + M_{Rot} * R\omega^2$$

2.2-9

Where:

$M_{Rec} = \text{Piston Assembly} + 1/3 \text{ Con Rod}$

$M_{Rot} = 2/3 \text{ Con Rod} - \text{Cap Mass}$

The same method was used to calculate the tensile stress as before, this time however, since there is no compressive force, the mean and range of stress, still have two terms, but the compressive force is represented by zero, shown below.

$$Range \text{ of Stress } (N/mm^2) = \sigma_{Max} - \sigma_{Min} = 38.75 - 0 = 38.75 N/mm^2$$

2.2-10

$$Mean \text{ Stress } (\sigma_{Mean}) (N/mm^2) = \frac{\sigma_{Max} - \sigma_{Min}}{2} = \frac{38.75 + 0}{2} = 19.38 N/mm^2$$

2.2-11

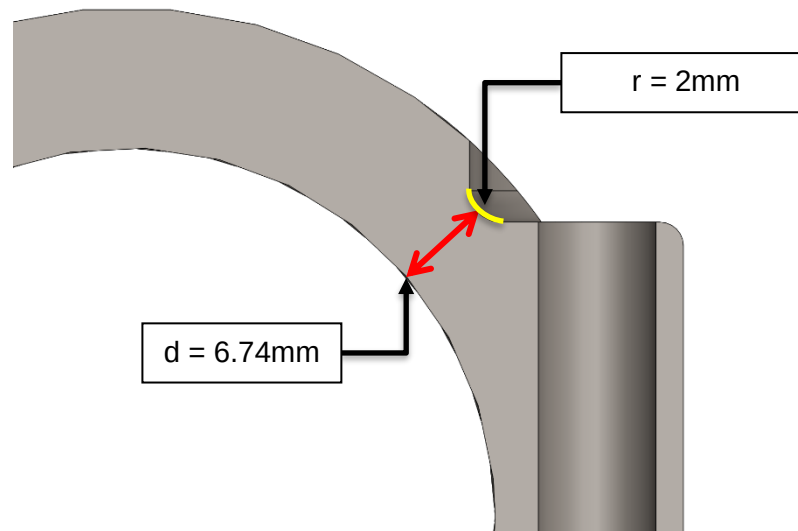


Figure 11 – Required Dimensions for Calculating Stress Concentration Factors for Point C (Richardson, 2020)

The stress concentration will now be evaluated for point 'C'. This time, we are dealing with a hyperbolic radius within the con rod so a suitable graph within "Peterson's Stress Concentration Factors" (Pilkey, 1997) page 88 (Appendix 6.2, Figure 48). Measurements were taken as shown within (Figure 11)

Since there was only one line on the graph, the only equation required was to attain the x-axis value.

$$X - Axis Value = \frac{r}{d} = \frac{2}{6.74} = 0.2967$$

2.2-12

The value for the stress concentration of point 'C' can now be equated by viewing the graph within Appendix 6.2, an annotated version can be seen in Figure 12.

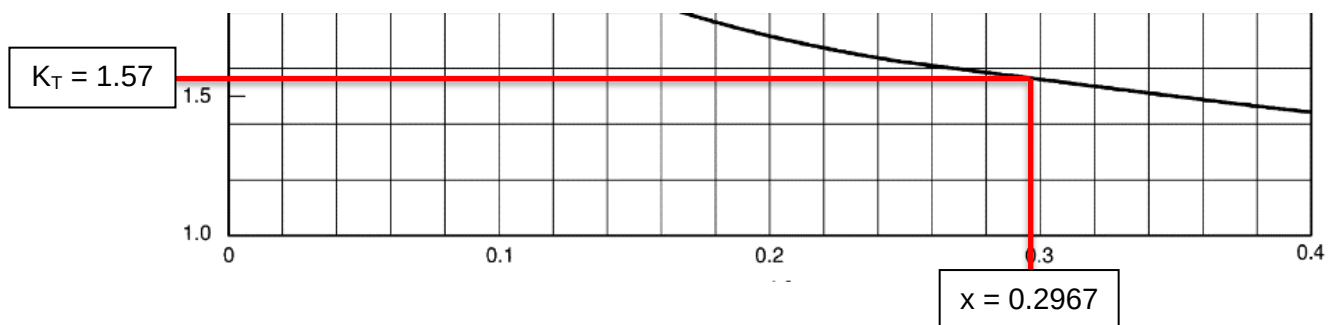


Figure 12 – Visual Interpretation of The Stress Concentration Factor Graph for 'Point C' (Pilkey, 1997)

The polished endurance limit (EL) is divided by the stress concentration factor (K_T) to achieve the final endurance limit (FEL).

$$FEL (N/mm^2) = \frac{0.45 * UTS}{K_T} = \frac{0.45 * 760}{1.57} = 217.83 N/mm^2$$

2.2-13

Now a Goodman diagram can be constructed, and the correct fatigue safety factor can be ascertained.

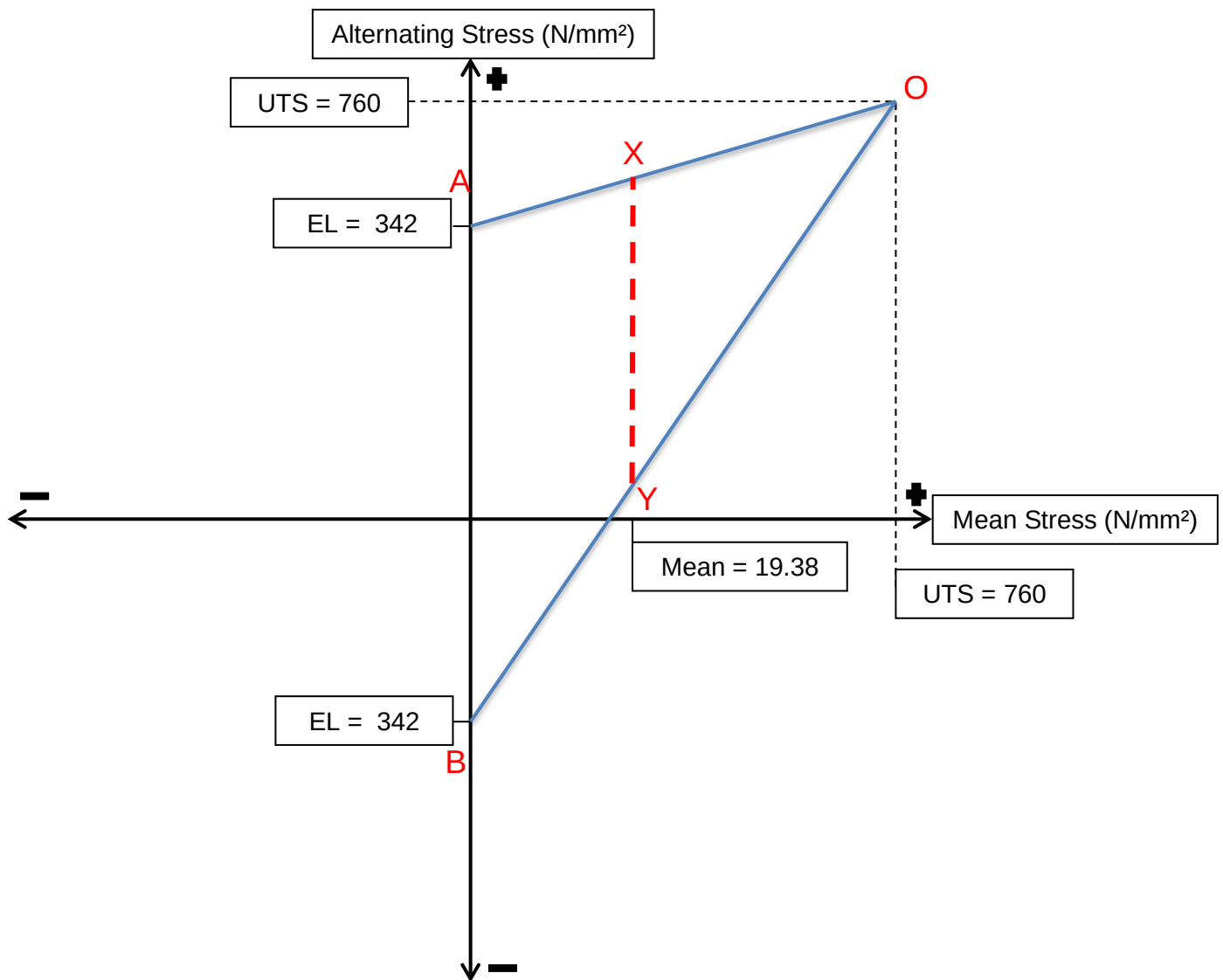


Figure 13 - Goodman Diagram for Con Rod Point 'C'

$$FSF = \frac{\text{Allowable Range}}{\text{Actual Range}} = \frac{446.77}{38.75} = 11.53$$

2.2-14

This FSF shows that point 'C' of the con rod is strong enough to resist the forces applied to the con rod throughout the standard engine cycle.

2.3. Connecting Rod Bolt Forces

This portion of the experiment now turned its focus to the bolts that join the con rod cap to the con rod itself around the big end bearing.

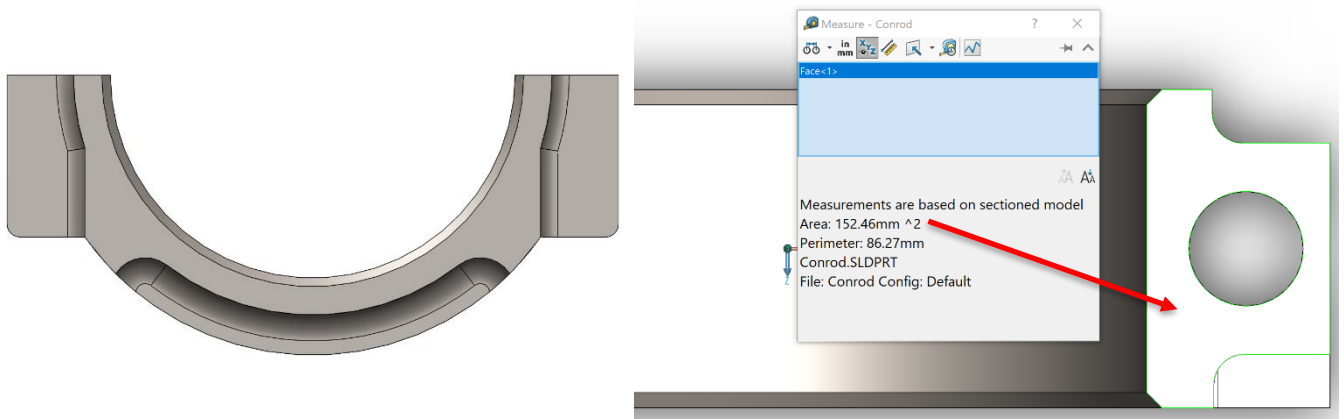


Figure 14 – CAD Model of The Con Rod Cap (Left) & Cross-Sectional Area Measurement of The Cap-Rod Interface (Right) (Richardson, 2020)

The first task was to determine the stiffness of both the con rod cap and the rod bolt. The materials of both components were known; the cap was made from PMET 4206F steel (GKN Sinter Metals, 2019), whereas, the bolt is made from strong 34CrMo4 Chromium-Molybdenum steel (Steel Number, 2020).

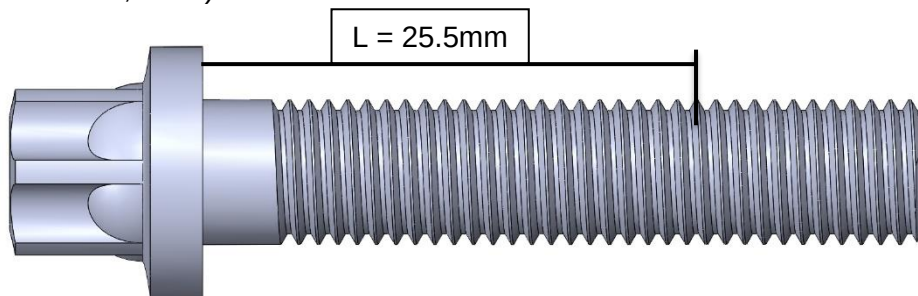


Figure 15 – CAD Model of The Standard Con Rod Bolt with Length To 2nd Engaged Thread Highlighted (Anderson, 2020)

Both sources included the Young's Modulus (E) ("describes the elastic properties of a solid undergoing tension or compression in only one direction" (Encyclopedia Britannica, 2020)) for these materials.

The cross-sectional area (A) needed to be determined, from which Solidworks became of use for the cap as 152.46mm² (Figure 14 – Right), whereas, the Bosch Automotive Handbook (Bosch, 2002) states that the tensile stress area of an M8 bolt is 39.2mm².

The length until the 2nd engaged thread was measured, as this would be used to equate the individual stiffness (K_T) of the cap and the bolts (Figure 15).

Each individual stiffness can be calculated using:

$$K_T (N/mm) = \frac{E * A}{L}$$

2.3-1

The ratio of stiffnesses can be written as:

$$K_T \text{ Ratio} = \frac{K_{Cap}}{K_{Bolt}} \quad 2.3-2$$

By substituting each stiffness (2.3-1) into equation 2.3-2:

$$K_T \text{ Ratio} = \frac{\frac{E_{Cap} * A_{Cap}}{L}}{\frac{E_{Bolt} * A_{Bolt}}{L}} \quad 2.3-3$$

But since each stiffness uses the same length, it can be cancelled, leaving the final equation:

$$K_T \text{ Ratio} = \frac{E_{Cap} * A_{Cap}}{E_{Bolt} * A_{Bolt}} = \frac{200 * 10^3 * 152.46}{202 * 10^3 * 39.2} = 3.851 \quad 2.3-4$$

The stiffness of bolt and cap also be equated, as below:

$$K_{Cap} (N/mm) = \frac{E_{Cap} * A_{Cap}}{L} = \frac{200 * 10^3 * 152.46}{25.5} = 1195.76 \text{ kN/mm} \quad 2.3-5$$

$$K_{Bolt} (N/mm) = \frac{E_{Bolt} * A_{Bolt}}{L} = \frac{202 * 10^3 * 39.2}{25.5} = 310.53 \text{ kN/mm} \quad 2.3-6$$

All the required dimensions and measurements are contained with **Table 2** below:

Bolt + Cap Specs		
Bolt - UTS	1100	Mpa
Bolt Area CSA	39.2	mm ²
Bolt Young's Modulus	202	GPa
Cap - UTS	760	Mpa
Rod Cap Interface CSA	152.46	mm ²
Cap Young's Modulus	200	GPa
Cap-Bolt Stiffness Ratio	3.851	
Cap From Bottom To Split	19.25	mm
Length Till Thread In Rod	4.25	mm
Length To 2nd Thread	2	mm
Total Engaged Thread Length	25.50	mm
Bolt Stiffness	310.53	kN/mm
Cap Stiffness	1195.76	kN/mm

Table 2 – Specifications for The Con Rod Cap & Bolts

As explained before, both the con rod cap & bolts do not experience compressive forces throughout the engine's operation, so the maximum tensile net force was taken for each engine speed. To ascertain the correct inertia force that is experienced on the bolt and the cap, equation 2.2-9 is used but this time, including the mass of the con rod cap, this is subtracted from the gas force to create the net force applied onto the bolts.

To determine the extra force distributed through the bolts, half of the tensile force is multiplied by a form of the stiffness ratio.

$$Extra\ Bolt\ Force\ (N) = \frac{1}{1 + \frac{K_{Cap}}{K_{Bolt}}} * Tensile\ Force = \frac{1}{1 + \frac{1195.76}{310.53}} * 11430.44 = 1178.21\ N$$

2.3-7

Nominal thread diameter	Pitch	Pitch diameter	Minor diameter	Thread depth	Tensile stress area
$d = D$	P	$d_2 = D_2$	d_3	D_1	H_1
8	1	7.350	6.773	6.917	0.613
					0.541
					A_s in mm ²
					39.2

Figure 16 – Dimensions of An M8 1mm Pitch Threaded Bolt (Bosch, 2002)

The extra nominal stress exhibited upon the root of the bolt thread. By dividing the extra bolt force by the surface area of the bolt by using the average diameter of the bolt, given by the Bosch Automotive Handbook (Bosch, 2002), seen within Figure 16 as d_2 (7.35mm). This stress is the alternating load on the bolt throughout the engine's operation.

$$Extra\ Nominal\ Stress\ (N/mm^2) = \frac{Extra\ Bolt\ Force}{Bolt\ Surface\ Area} = \frac{1178.21}{\frac{\pi * 7.35^2}{4}} = 27.77\ N/mm^2$$

2.3-8

As the bolts were tightened to 90% of yield, the required bolt tightening torque to overcome friction can be determined. If yield is 90% of the UTS, then 90% of yield can be equated to:

$$90\% \text{ of Yield} = 0.9 * 0.9 * UTS = 0.9 * 0.9 * 1100 = 891\ N/mm^2$$

2.3-9

60% of this value is made up of Axial stress, the rest is comprised of Torsional.

$$Axial\ Stress\ (N/mm^2) = 0.6 * 90\% \text{ of Yield} = 0.6 * 891 = 534.6\ N/mm^2$$

2.3-10

The clamping force once assembled can also be calculated by multiplying the Axial stress by the cross-sectional area of the bolt using the average thread diameter.

$$Clamping\ Force\ (N) = Bolt\ Surface\ Area * Axial\ Stress = \frac{\pi * 7.35^2}{4} * 534.6 = 22682.64\ N$$

2.3-11

The frictional values for the thread ($\mu = 0.18$) and the head ($\mu = 0.12$) of the bolt were supplied by Malcolm McDonald during an 'Advanced Engine Design' lecture (McDonald, 2019). These were used to calculate the required tightening torque. The circumferential force of the bolt thread was analysed first.

$$Circumferential\ Friction\ Force\ (N) = \mu * Clamping\ Force = 0.18 * 22682.64 = 4082.87\ N$$

2.3-12

By multiplying the circumferential force by the average radius the torque can be equated. This is divided by 1000 to turn it from N/mm to N/m.

$$Threads\ Resisting\ Torque\ (N/m) = \frac{\frac{Avg\ Diameter}{2} * Circumferential\ Thread\ Force}{1000}$$

2.3-13

$$\text{Thread Resisting Torque (N/m)} = \frac{\frac{7.35}{2} * 4082.87}{1000} = 15.005 \text{ N/m} \quad 2.3-14$$

To calculate the friction caused by the head of the bolt, the clamping force is multiplied by the head frictional coefficient.

$$\text{Circumferential Friction Force (N)} = \mu * \text{Clamping Force} = 0.12 * 22682.64 = 2721.92 \text{ N} \quad 2.3-15$$

To progress further, the average diameter of the head is required. By measuring the outside diameter of the seating face, and using the outer diameter of the thread, the average seating face diameter can be attained.

$$\text{Average Seating Face Diameter (mm)} = \frac{\text{Outer} + \text{Inner}}{2} = \frac{13 + 8}{2} = 10.5 \text{ mm} \quad 2.3-16$$

Now the radius of this seating face is used to gain the torque required to overcome the head friction.

$$\text{Threads Resisting Torque (N/m)} = \frac{\frac{\text{Avg Diameter}}{2} * \text{Circumferential Head Force}}{1000} \quad 2.3-17$$

$$\text{Head Resisting Torque (N/m)} = \frac{\frac{10.5}{2} * 2721.92}{1000} = 14.29 \text{ N/m} \quad 2.3-18$$

The torque can be totalled to determine the torque required to tighten the bolt.

$$\text{Total Torque (N/m)} = \text{Head Torque} + \text{Thread Torque} = 14.29 + 15.005 = 29.29 \text{ N/m} \quad 2.3-19$$

The extra nominal stress is the extra load applied to the bolts from static (equation 2.3-8), if these bolts were tightened to 90% of yield, then that is the minimum amount of stress the bolts will endure. Where the maximum stress is 90% of yield, plus the range of stress.

$$\sigma_{Max} \text{ (N/mm}^2\text{)} = 891 + 27.77 = 918.77 \text{ N/mm}^2 \quad 2.3-20$$

$$\sigma_{Min} \text{ (N/mm}^2\text{)} = 891 \text{ N/mm}^2 \quad 2.3-21$$

To verify that the range (alternating load) on the bolt is correct, it can be back calculated.

$$\text{Range of Stress (N/mm}^2\text{)} = \sigma_{Max} - \sigma_{Min} = 918.77 - 891 = 27.77 \text{ N/mm}^2 \quad 2.3-22$$

The mean stress can also be ascertained by the same method previously shown.

$$\text{Mean Stress } (\sigma_{Mean}) \text{ (N/mm}^2\text{)} = \frac{\sigma_{Max} - \sigma_{Min}}{2} = \frac{918.77 + 891}{2} = 904.88 \text{ N/mm}^2 \quad 2.3-23$$

As seen before, stress concentrations are an important factor to consider for the design of any part as they severely impact the endurance limit (EL) that the component can endure. By using the stress concentration graph in **Appendix 6.3 Figure 49**, it was possible to attain the correct value the correct EL value that would impact the Fatigue Safety Factor (FSF).

By knowing that high carbon steel has a value of 0.295 (The Engineering Toolbox, 2020) the suitable stress line of 0.3 was chosen. The radius of the thread was measured at 0.1443mm and Bosch stated the distance between the radii as 6.773mm.

$$X - Axis Value = \frac{r}{d} = \frac{0.1442}{6.773} = 0.0213$$

2.3-24

With the x-axis value, the graph can be used to attain the correct stress concentration factor.

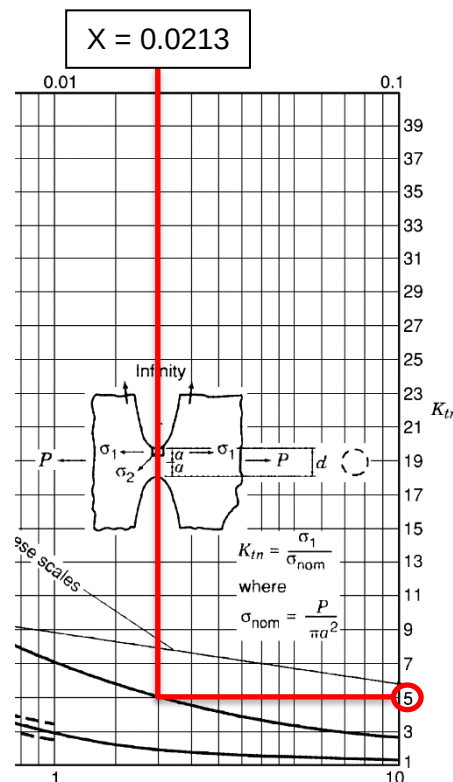


Figure 17 – Visual Interpretation of The Stress Concentration Factor Graph for The Con Rod Bolts (Pilkey, 1997)

Using the stress concentration factor of 5, the Final Endurance Limit (FEL) can be found to plot the Goodman diagram correctly as done previously.

$$FEL (N/mm^2) = \frac{0.45 * UTS}{K_T} = \frac{0.45 * 1100}{5} = 99 N/mm^2$$

2.3-25

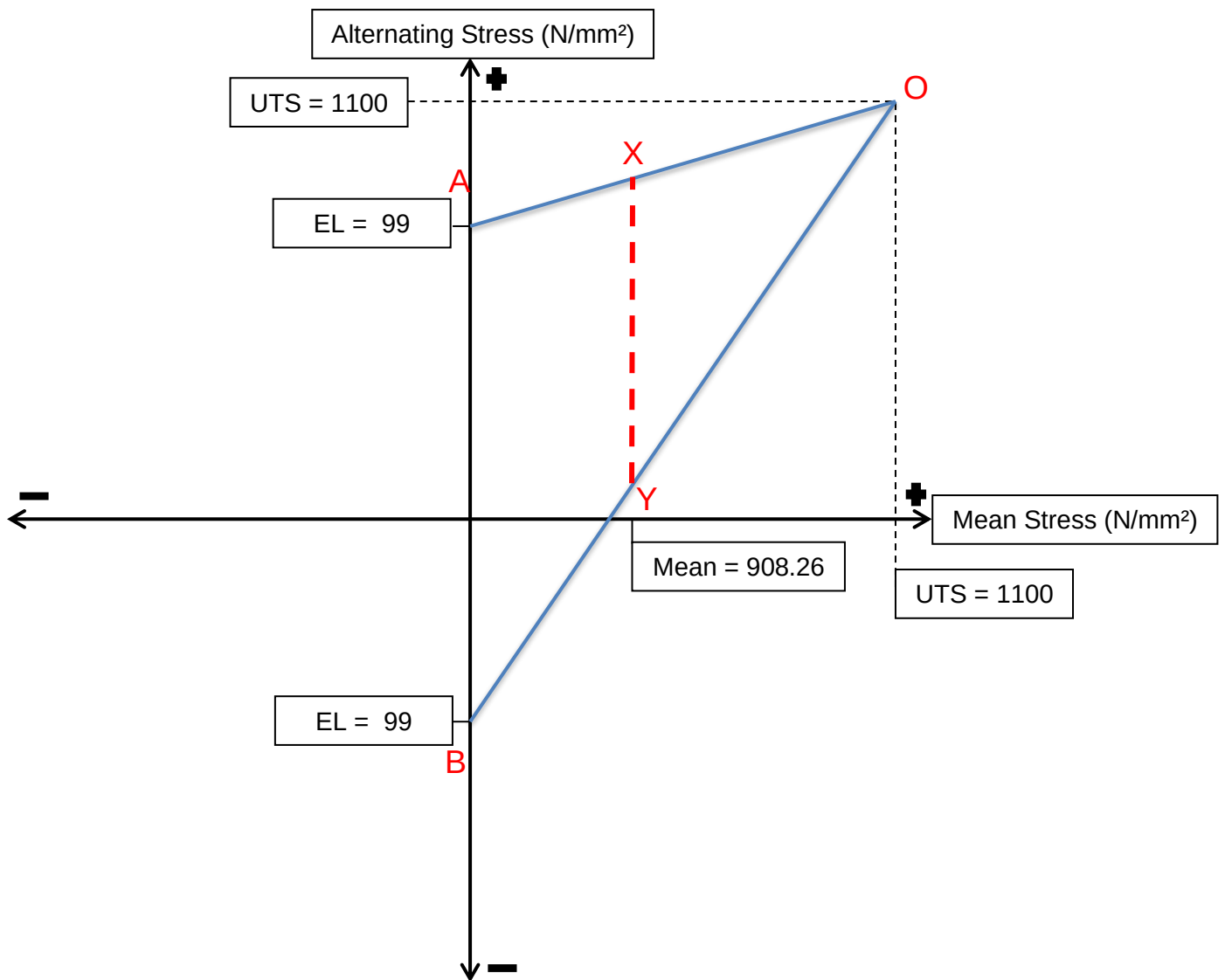


Figure 18 - Goodman Diagram for The Con Rod Bolts

$$FSF = \frac{\text{Allowable Range}}{\text{Actual Range}} = \frac{35.12}{27.77} = 1.265$$

2.3-26

This FSF shows that the con rod bolts are strong enough to resist the forces throughout the standard engine cycle at 6000RPM. Some of the results from this experiment conducted here can be seen within

Bolt		
Concentration Factor	5.00	
Actual Range Stress	27.769	N/mm ²
Actual Stress	918.769	N/mm ²
Mean Stress	904.884	N/mm ²
Polished Endurance Limit	495.000	N/mm ²
Final Endurance Limit	99.000	N/mm ²
Allowable Stress Range	35.121	N/mm ²
Fatigue Safety Factor	1.265	

Table 3 – Stress Results from The Standard Ford Bolt

A quick investigation was conducted to see when the standard bolts reach a safety factor below one, showing that they would fail. Below (Table 4) are the results from a linear interpolation of the gas forces created from a higher RPM, from which, the other necessary calculations were formulated.

This experiment shows the exact RPM that the bolts Fatigue Safety Factor (FSF) reaches a value of '1' to ten decimal places, after this exact point, the FSF drop to '0.99', meaning the bolts would fatigue and eventually fail.

RPM	7105.826171	
Bolt		
Concentration Factor	5.00	
Actual Range Stress	34.514	N/mm ²
Actual Stress	925.514	N/mm ²
Mean Stress	908.257	N/mm ²
Polished Endurance Limit	495.000	N/mm ²
Final Endurance Limit	99.000	N/mm ²
Allowable Stress Range	34.514	N/mm ²
Fatigue Safety Factor	1.0000000000	

Table 4 – Results from Study Conducted to See When The Standard Bolts Would Break

Another experiment was conducted to see whether using modified bolts, with a waisted shank and a higher UTS would theoretically be able to endure higher stress and RPM.

Using Solidworks, a CAD model for the modified bolt was created as a representation, and to gain cross-sectional areas to be used within the calculations.

The modified bolt would use the same tensile stress area as the previous bolt, the same length and the same head design for an accurate comparison. The same threaded diameter would also be kept so no modification to the con rod would need to take place.

All calculations that have been made on the previous standard bolt would be made to test the new modified bolt.

The modified bolt will be made from H13 tool steel, with a UTS of 1990 MPa (Elgin Fastener Group, 2020). The bolt features a shorter thread length and a narrow mid shank.

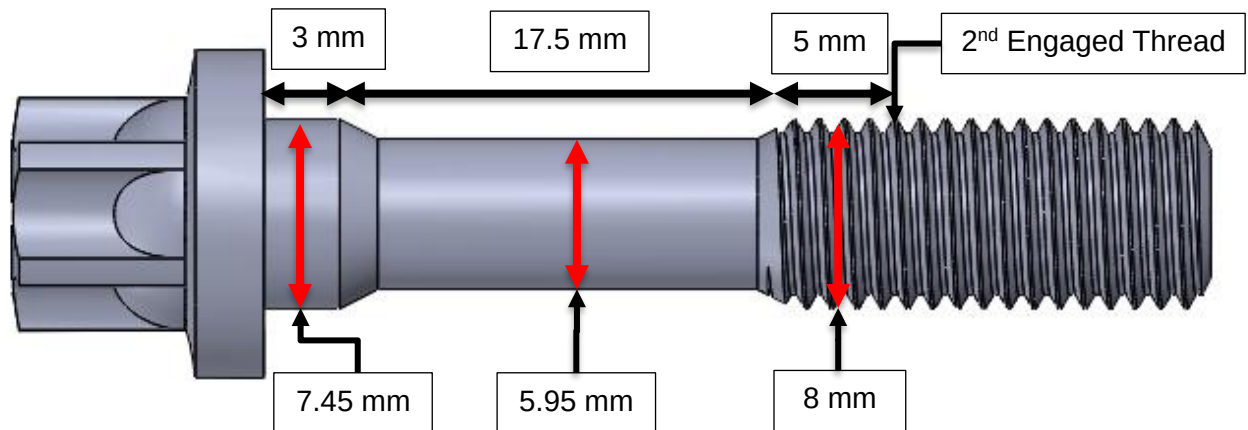


Figure 19 – Modified Bolt Dimensions

The cross-sectional areas were calculated for each section of the bolt shank. Since there was different areas and lengths, each section would also have a new stiffness that is required. Using Solidworks measure the lengths and areas of each section, Excel was used to store this information before using the following equation to determine the overall new bolt stiffness.

$$\text{Bolt Stiffness (kN/mm)} = \left(\frac{1}{\frac{E_1 * A_1}{L_1}} \right) + \left(\frac{1}{\frac{E_2 * A_2}{L_2}} \right) + \left(\frac{1}{\frac{E_3 * A_3}{L_3}} \right) \quad 2.3-27$$

$$\text{Bolt Stiffness (kN/mm)} = \left(\frac{1}{\frac{210 * 10^3 * 43.59}{3}} \right) + \left(\frac{1}{\frac{210 * 10^3 * 27.28}{17.5}} \right) + \left(\frac{1}{\frac{210 * 10^3 * 50.27}{5}} \right) \quad 2.3-28$$

This gave an overall bolt stiffness of 549.33 kN/mm, with a new cap-to-bolt stiffness ratio of 3.704.

The previous method was now conducted onto the bolt using the same equations and forces exerted as on the standard bolt to determine whether this new bolt has an acceptable fatigue safety factor.

The results form this study are entailed within Table 5.

Modded Bolt Specs - H13 Tool Steel		
Bolt - UTS	1990.00	Mpa
Bolt Area CSA	39.2	mm ²
Bolt Young's Modulus	210.00	GPa
Bolt Stiffness	549.33	kN/mm ²
Cap-Bolt Stiffness Ratio	3.704	
Extra Force In Bolt Ratio	0.2126	
Concentration Factor	5.00	
Actual Range Stress	30.994	N/mm ²
Actual Stress	1642.894	N/mm ²
Mean Stress	1627.397	N/mm ²
Polished Endurance Limit	895.500	N/mm ²
Final Endurance Limit	179.100	N/mm ²
Allowable Stress Range	65.269	N/mm ²
Fatigue Safety Factor	2.106	

Bolt Torques		
90% Yield	1612	N/mm ²
Axial Stress	967.1	N/mm ²
Torsional	644.8	N/mm ²
Axial Force	37911.89	N
Thread Circumferential Force	6824.14	N
Thread Tightening Torque	24.339	N/m
Head Circumferential Force	4549.43	N
Head Tightening Torque	23.88	N/m
Total Tightening Torque	48.224	N/m

Table 5 – Modified Bolt Calculations and Results

As seen above, the resultant safety factor was determined to be 2.106 N/mm². This was 0.841 better and should withstand a greater rise in RPM and stress than that of the standard bolt.

In terms of tightening torque, this resulted in a total of 48.22 N/m in order to tighten to 90% yield. With an increase in 890 MPa in UTS, it would result in a higher torque to get past the elastic portion of the steels modulus and reach the plastic region.

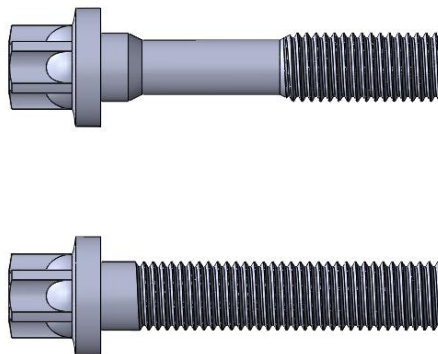


Figure 20 - CAD Models of Both The Standard And Modified Bolts (Anderson, 2020)

There are a few critical features within the con rod bolts. The main feature to consider is that of the 2nd engaged thread, this is where the most force occurs when tightened past 90% yield and between this point and the head of the bolt is where the most stretch takes place when the bolt enters the plastic region of its modulus.

The shank diameter is also a massive consideration upon design, the main reason for a waisted bolt is to reduce the stiffness of the bolt to reduce the amount of extra force that is exhibited upon the engaged thread at TDC, but reducing this too far can create a weak point that, when tightened, will enter yield too early and snap, either during fitment or operation.

2.4. Crankshaft Forces

Within this study, the forces applied to the crankshaft under standard operation of the Ford 1.6L Ecoboost engine. For this, measurements were taken of three critical areas of the shaft; the big end journal, the crank journal and the crank journal bearing shell, pictures of these are featured within [Appendix 6.7](#).

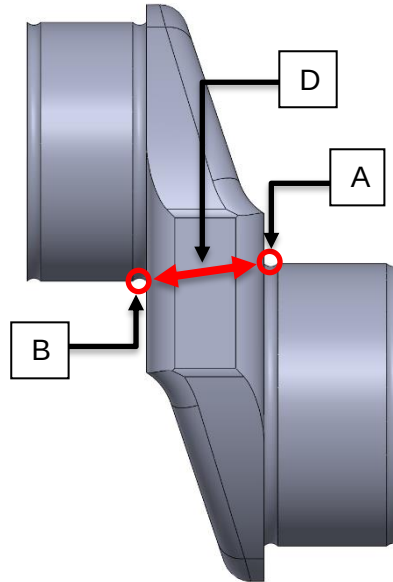


Figure 21 - CAD Model of The Standard Ford Crankshaft with Points 'A' & 'B' and Distance 'D' (Richardson, 2020)

The worst points within the crankshaft are the critical fillets either side of the crank web, shown in [Figure 21](#) labelled as 'A' & 'B'.

Crankshaft Specs		
Crank Journal Diameter	48	mm
Crank Journal Length	24.07	mm
Crank Journal Neutral Axis	24	mm
Crank Journal Projected CSA	1155.36	mm ²
Crank Pin Diameter	44.03	mm
Crank Pin Length	21.03	mm
Crank Pin Neutral Axis	22.015	mm
Crank Pin Projected CSA	925.95	mm ²
Bearing Shell Diameter	54.03	mm
Bearing Shell Length	15.12	mm
Bearing Shell Neutral Axis	27.015	mm
Bearing Projected CSA	816.93	mm ²
A to B Diagonal	21.98	mm
A to B Length	20.98	mm
A to B Neutral Axis	11.0	mm
L1	12.04	mm
L2	33.02	mm
Bredth	46.02	mm
Second Moment Of Area	40719.39	mm ⁴

Table 6 – Measurements and Specifications for The Standard Ford Crankshaft

For this study, only one crank journal is considered, so, as before the net force is halved because the same force is being applied through both bearings. The inertia force was calculated by taking the full amounts of mass for reciprocating and rotating mass components. The projected cross-sectional area is determined by multiplying the diameter by the length, as seen within Table 6.

$$\text{Projected Area} = \text{Diameter} * \text{Length} = 48 * 24.07 = 1155.36 \text{ mm}^2 \quad 2.4-1$$

Armed with the projected area and the net force, the nominal bearing pressure can be calculated.

$$\text{Nominal Bearing Pressure} = \frac{\frac{\text{Net Force}}{2}}{\text{Projected Area}} \quad 2.4-2$$

The distance from point 'A' to 'B' was measured to be 21.98mm, (Figure 21, labelled 'D') the section between 'A' & 'B' is the weakest point for bending within the crankshaft, this distance can be reduced by creating a shorter stroke, this reduces the crank throw of the engine and increase the highest RPM that can be achieved.

The lengths from each point was measured to the centre of the crank journal, these lengths would be used to calculate the bending moment exhibited through each point under both compressive and tensile forces.

$$\text{Bending Moment} = \frac{\text{Net Force}}{2} * \text{Distance To Centre Of Crank Journal} \quad 2.4-3$$

The distances from both 'A' to 'B' is halved to find the neutral axis between these points, where the bending moment centres around.. To calculate the bending stress, the second moment of area is required. For this, the effective width of the critical beam would be taken as the average diameter between the crank pin and the crank journal.

$$\text{Effective Width of the Critical Beam (mm)} = \frac{\text{Crank Pin Diameter} + \text{Crank Journal Diameter}}{2} \quad 2.4-4$$

$$\text{Effective Width of the Critical Beam (mm)} = \frac{44.03 + 48}{2} = 46.02 \text{ mm} \quad 2.4-5$$

Using this value as the breadth, the depth can be taken as the distance between points 'A' & 'B' to calculate the second moment of area.

$$\text{Second Moment of Area (mm}^4\text{)} = \frac{\text{Breadth} * \text{Depth}^3}{12} = \frac{46.02 * 21.98^3}{12} = 40719.39 \text{ mm}^4 \quad 2.4-6$$

As before (equation 2.1-9) the bending stress can be formulated, now the shear stress can be equated. For point 'A' the diameter of the crank journal is used, whereas, with point 'B' the diameter used is that of the crank pin.

$$\text{Shear Stress (N/mm}^2\text{)} = \frac{\frac{F_{\text{net}}}{2}}{\frac{\pi * \text{Diameter}^2}{4}} \quad 2.4-7$$

After calculating the resultant stress for both points, along with the mean and range of stress exhibited, the experiment can continue. Stress concentration factors are important within the critical fillets of the crankshaft. This means that, as seen within section 2.2.3, the endurance limit is severely altered, decreasing the amount of stress that specific component could withstand. The graph for Peterson's Stress Concentrations below can be found with **Appendix 6.4, Figure 50**.

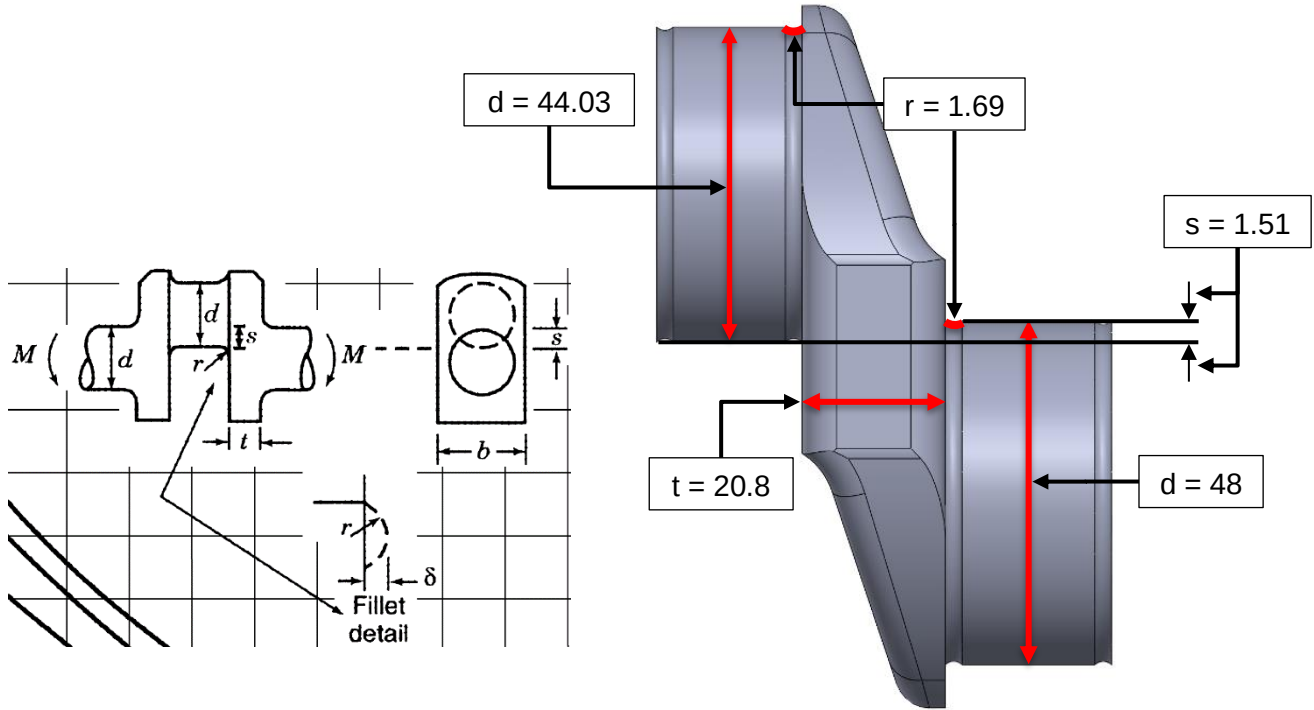


Figure 22 - Peterson's Stress Concentration Diagram (Left) (Pilkey, 1997) & CAD Model of Ford Crankshaft With Dimensions in 'mm' (Right) (Richardson, 2020)

These dimensions, along with the Peterson's graph, enabled the selection of the correct stress line to follow within the graph.

$$\text{Point 'A' Appropriate Stress Line} = \frac{s}{d} = \frac{1.51}{48} = 0.0315 \quad 2.4-8$$

$$\text{Point 'B' Appropriate Stress Line} = \frac{s}{d} = \frac{1.51}{44.03} = 0.0343 \quad 2.4-9$$

The lines within the graph use the value for $\frac{s}{d}$, with this, the x-axis value can now be formulated.

$$\text{Point 'A' X - Axis Value} = \frac{t}{d} = \frac{20.8}{48} = 0.43 \quad 2.4-10$$

$$\text{Point 'B' X - Axis Value} = \frac{t}{d} = \frac{20.8}{44.03} = 0.472 \quad 2.4-11$$

With the x-axis value, the graph can be used to attain the correct stress concentration factor. Unfortunately, as it will become apparent, this must not be the correct graph to use for the experiment.

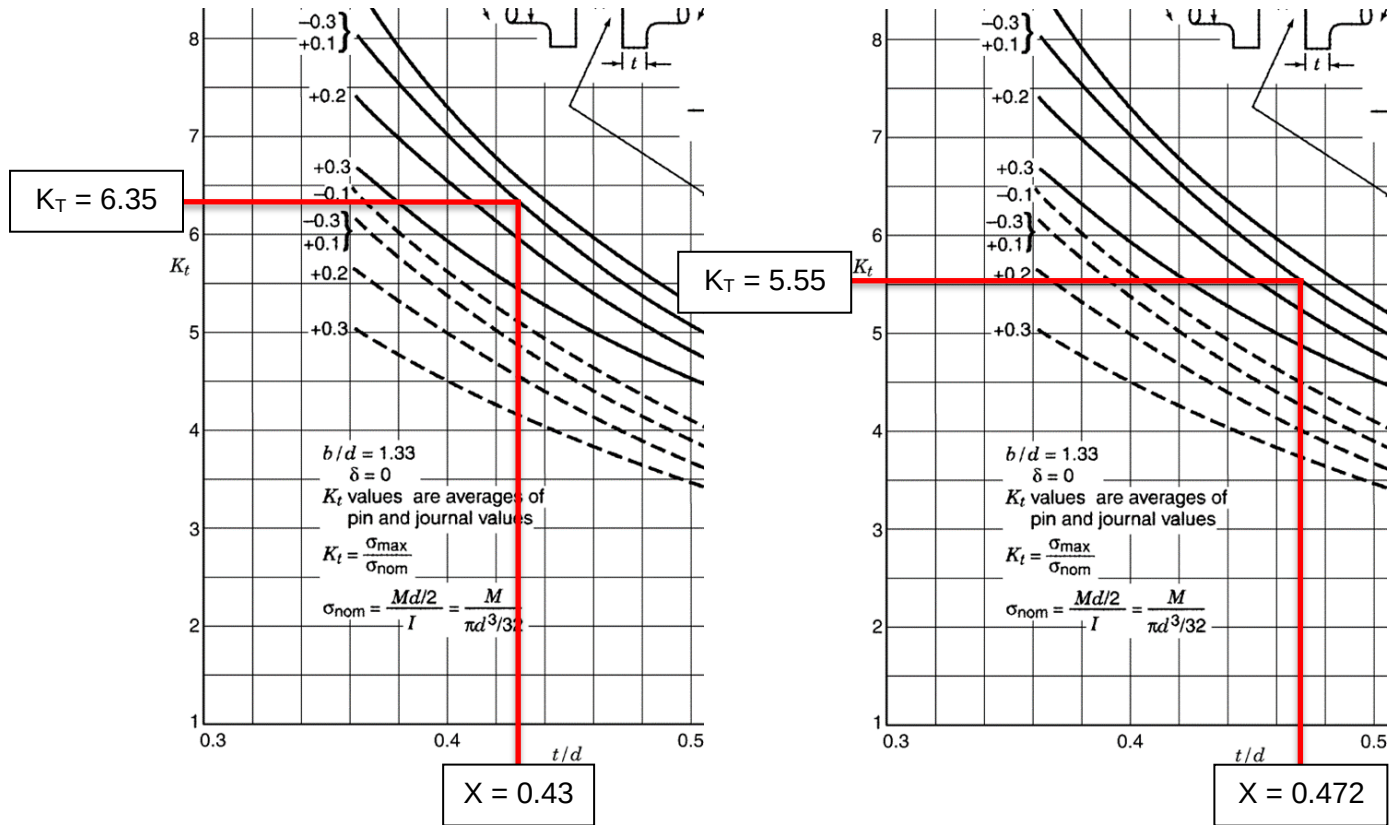


Figure 23 - Stress Concentration Factors Calculations For Points 'A' (Left) & 'B' (Right) (Pilkey, 1997)

By using the stress concentration factor for points 'A' & 'B' respectively (Figure 23), each fatigue safety factor can be equated.

$$\text{Point 'A' FEL (N/mm}^2\text{)} = \frac{0.45 * UTS}{K_T} = \frac{0.45 * 750}{6.35} = 53.15 \text{ N/mm}^2 \quad 2.4-12$$

$$\text{Point 'B' FEL (N/mm}^2\text{)} = \frac{0.45 * UTS}{K_T} = \frac{0.45 * 750}{5.55} = 60.81 \text{ N/mm}^2 \quad 2.4-13$$

Both points 'A' & 'B' are now analysed to determine whether their individual FSF is high enough to withstand the stress exhibited through the fillets. The allowable and actual range at 6000 RPM can be compared to attain the value for the fatigue safety factor.

$$\text{Point 'A' FSF} = \frac{\text{Allowable Range}}{\text{Actual Range}} = \frac{103.88}{79.46} = 1.307 \quad 2.4-14$$

$$\text{Point 'B' FSF} = \frac{\text{Allowable Range}}{\text{Actual Range}} = \frac{128.44}{208.16} = 0.617 \quad 2.4-15$$

This FSF shows that the fillet at point 'B' within the crankshaft is not strong enough to resist the forces throughout the standard engine cycle due to such a high stress concentration factor. It is evident to see that point 'B' sees more range of stress than point 'A' due to the bigger bending moment caused by the greater distance from 'B' to the centre of the crank journal which causes it to fail.

A search was conducted to find a more suitable stress concentration graph that was more applicable for the standard Ford crankshaft. As seen within [Appendix 6.5 Figure 51](#), a suitable graph was attained from the 3rd edition of the Peterson's Stress Concentration instead of 2nd edition that was previously used. (Pilkey & Pilkey, 2008)

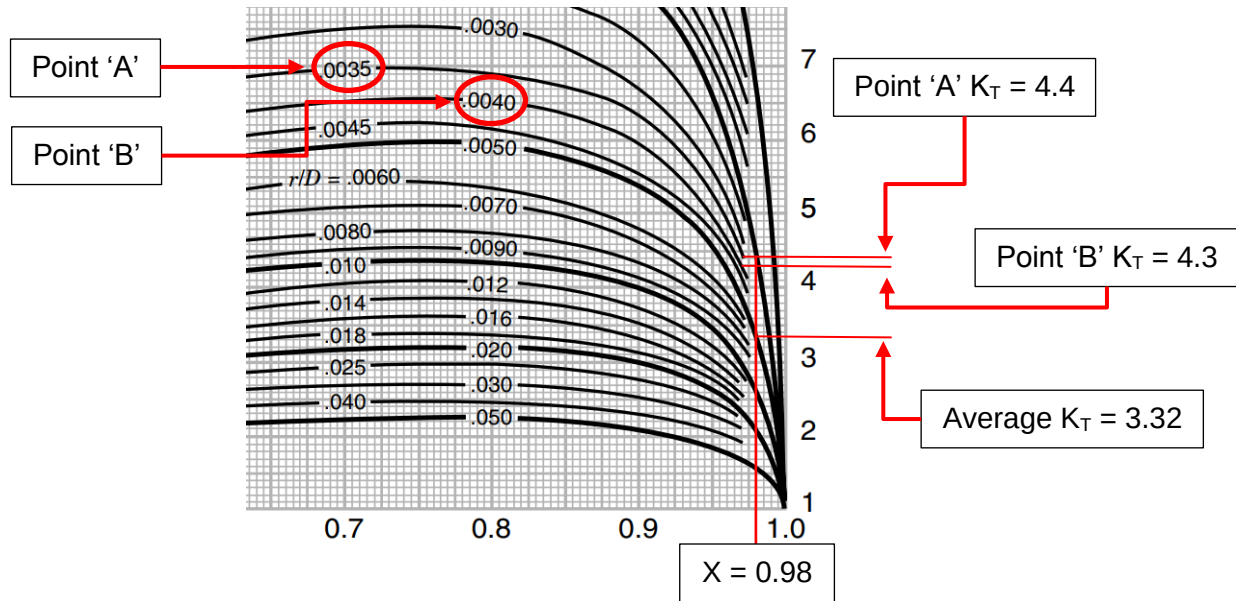


Figure 24 – Stress Concentration Factor for The Standard Ford Crankshaft (Pilkey & Pilkey, 2008)

By taking the outer diameter of the crank journal and crank pin (D), and subtracting the depth of the radius, the inner diameter can be gained (d) the following x-axis values can be ascertained.

$$\text{Point 'A' Appropriate Stress Line} = \frac{r}{D} = \frac{1.69}{48} = 0.03521 \quad 2.4-16$$

$$\text{Point 'B' Appropriate Stress Line} = \frac{r}{D} = \frac{1.69}{44.03} = 0.03838 \quad 2.4-17$$

The lines within the graph use the value for $\frac{r}{D}$, because at this point of the graph the separate lines dissipate, the value for K_T was taken from the line for '0.05' instead. With this, the x-axis value can now be formulated.

$$\text{Point 'A' X - Axis Value} = \frac{d}{D} = \frac{47}{48} = 0.9792 \quad 2.4-18$$

$$\text{Point 'B' X - Axis Value} = \frac{d}{D} = \frac{43.03}{44.03} = 0.9773 \quad 2.4-19$$

This now meant that both points 'A' & 'B' used the same stress concentration factor, which would mean they shared the same endurance limit.

$$FEL (N/mm^2) = \frac{0.45 * UTS}{K_T} = \frac{0.45 * 750}{3.32} = 101.66 N/mm^2 \quad 2.4-20$$

Both fillets were analysed to determine whether their FSF is high enough to withstand the stress exhibited through the fillets.

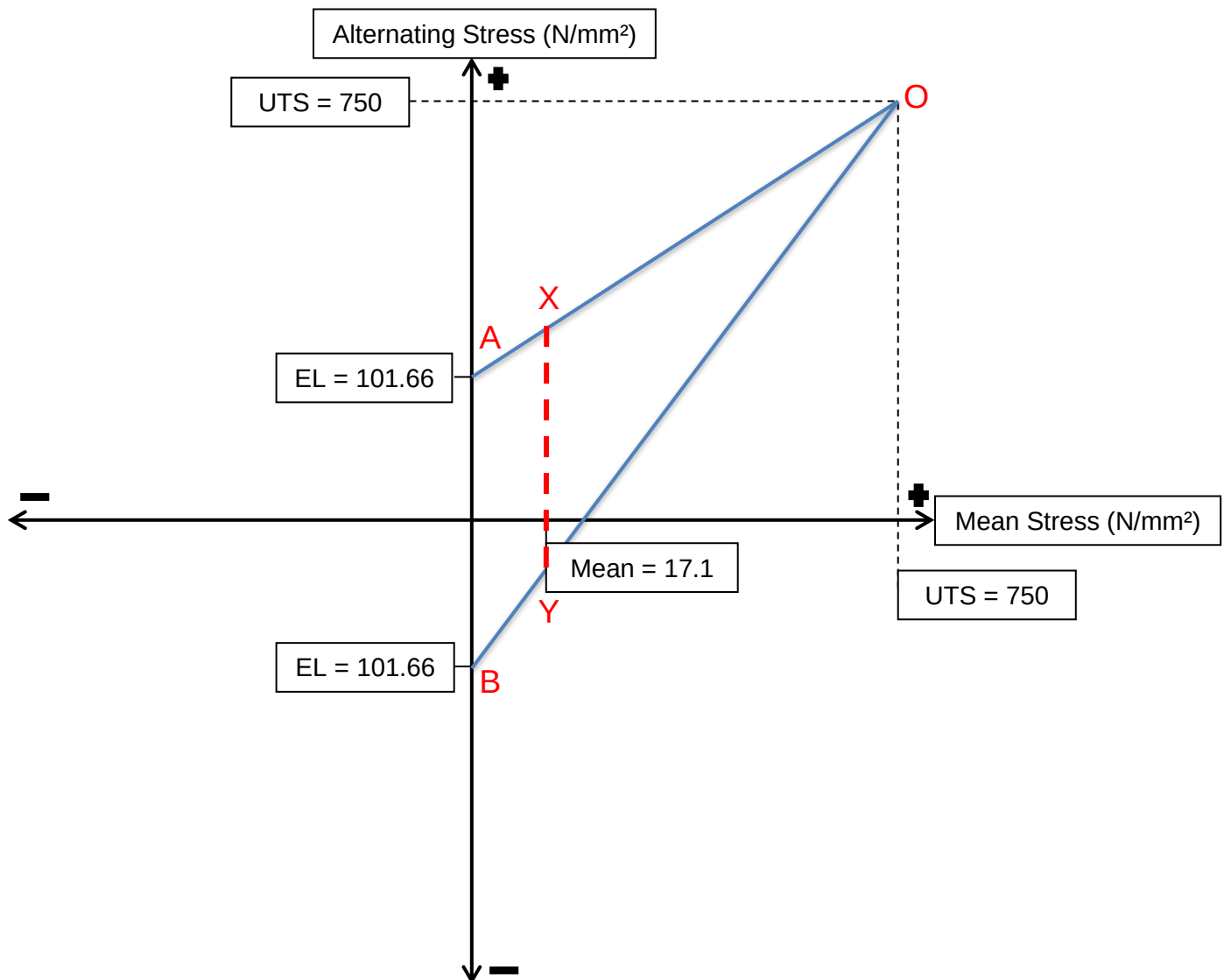


Figure 25 - Goodman Diagram For Crank Shaft Point 'A'

The allowable and actual range at 6000 RPM can be compared to attain the value for the fatigue safety factor.

$$\text{Point 'A' FSF} = \frac{\text{Allowable Range}}{\text{Actual Range}} = \frac{198.68}{79.46} = 2.5$$

2.4-21

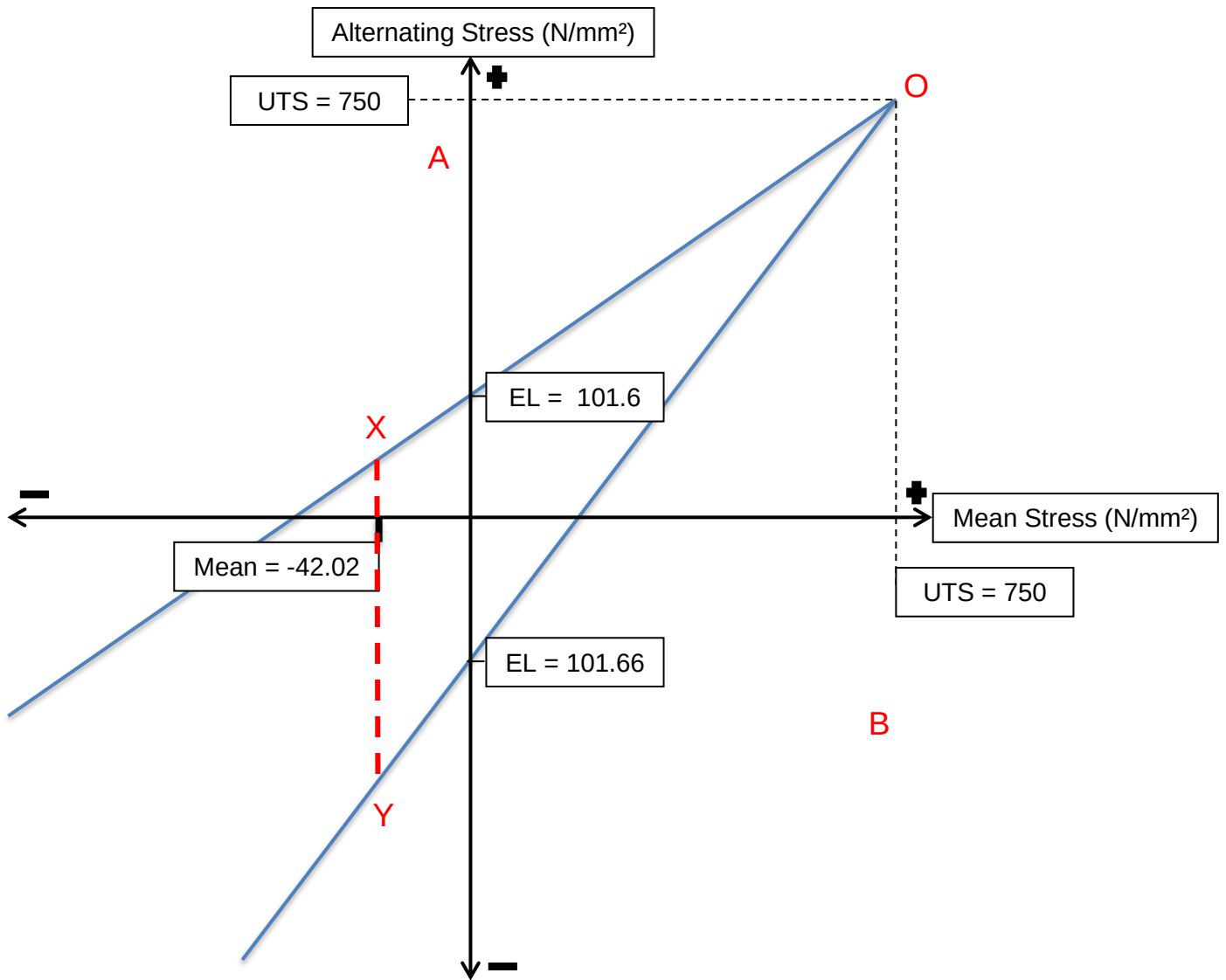


Figure 26 - Goodman Diagram for Crankshaft Point 'B'

$$\text{Point 'B' FSF} = \frac{\text{Allowable Range}}{\text{Actual Range}} = \frac{214.7}{208.16} = 1.031$$

2.4-22

This FSF shows that the fillets within the crankshaft can endure the forces throughout the engine cycle, but point 'B' is only just strong enough to resist the forces and should the RPM increase, this point would fail, leading to the conclusion that the engine is limited to 6000 RPM because after which, the crankshaft will explode.

3) Results

3.1. Gudgeon Pin Forces

3.1.1. Max Net Forces

Gudgeon Pin Max Forces @ A (N)													
Engine Type	RPM	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	6000
Production Engine													
Compression		22401.53	28095.79	32042.17	35015.51	36757.58	37933.28	38843.69	39422.43	38637.54	39442.92	41413.66	42165.81
Tensile		-271.20	-207.98	-149.56	5.72	314.08	661.40	1106.69	1580.12	2127.77	2726.02	3418.95	3756.86

Table 7 – Max Compressive & Tensile Forces Acting on The Gudgeon Pin vs Engine Speed

As expected, the compressive forces are significantly greater than the tensile forces acting on the gudgeon pin during operation from point 'A'. During low engine speeds (< 2000 RPM) the gudgeon pin is still in compression, whereas, after 2000 RPM the gas forces overcome the pin's inertia force causing it to be in tension.

The forces experienced by the gudgeon pin consist of negative (compressive) & positive (tensile), the results shown within both Table 7 and all those further throughout the experiment, have had their signs flipped so that each force is represented as a positive value within its own right.

By looking at Figure 27, it is obvious to see that the tensile forces increase at an exponential rate, whereas, the compressive forces starts off with an exponential increase, the rate of which then drops off, although still increases before dropping off and then increases again.

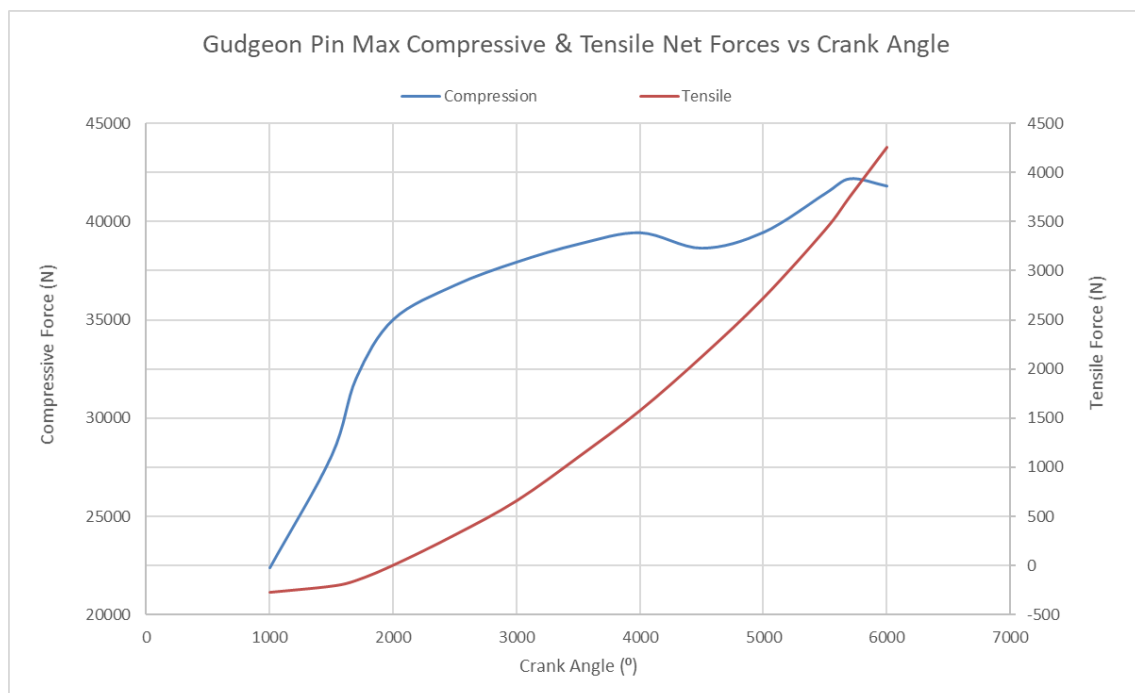


Figure 27 – Gudgeon Pin Max Compressive & Tensile Net Forces vs Engine Speed

3.1.2. Mean Stress

Gudgeon Pin Mean Stress @ A (N/mm ²)													
	RPM												
	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000
	94.01	117.93	135.01	148.64	154.81	158.65	160.63	161.19	155.64	156.77	162.68	164.67	161.27

Table 8 – Mean Stress Acting on The Gudgeon Pin vs Engine Speed

Table 8 features the results from calculating the mean stress that the gudgeon pin experiences.

It is hard not to notice that the pattern that occurs within Figure 28, is extremely similar to the compressive force seen within Figure 27 due to the 'hunchback' effect.

Because the tensile force occurs at a constant exponential rate, whereas, the compressive force features a lot of noise, this makes the compressive force the dominant and deciding factor in the value of mean stress exerted throughout the engines operation.

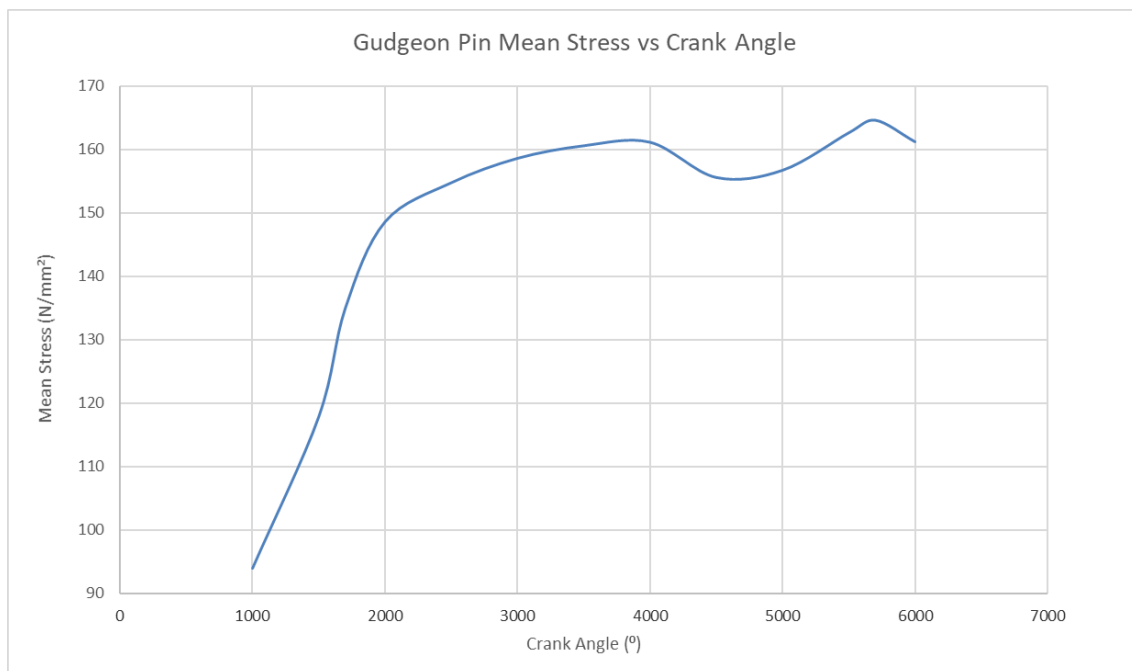


Figure 28 – Gudgeon Pin Mean Stress vs Engine Speed

3.1.3. Range of Stress

Gudgeon Pin Range Of Shear Stress @ A (N/mm ²)													
	RPM												
	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000
	193.05	240.94	273.98	297.74	316.21	330.00	343.42	354.43	354.64	368.86	394.25	404.65	407.63

Table 9 – Range of Stress Acting on The Gudgeon Pin vs Engine Speed

Above, are the results from calculating the range of stress experienced by the gudgeon pin during the engines operation against the engine's speed.

Although not to the same extent that the mean stress exhibits, the range of stress follows the pattern of the compressive force, shown previously.

It is expected that as the engine increases in speed, so too would the stress incurred upon the components within.

It also makes apparent that, for the same reason as with the mean stress, the compressive force being the greater force caused by the peak gas force during combustion overpowering the inertia of the gudgeon pin.

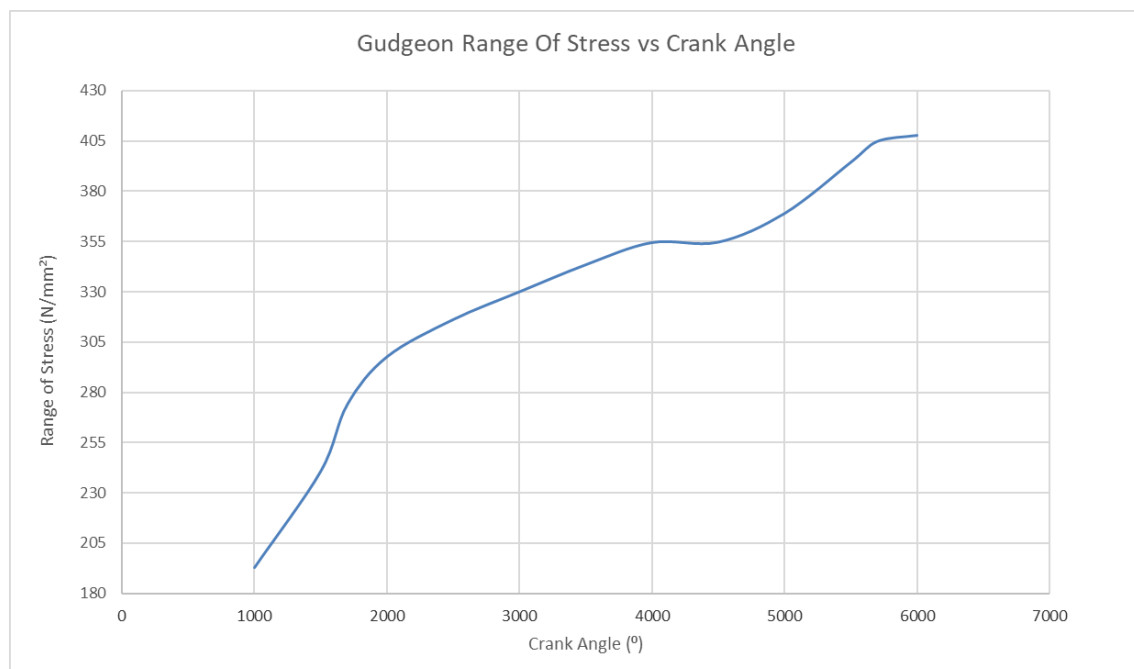


Figure 29 – Gudgeon Pin Range of Stress vs Engine Speed

3.2. Con Rod Forces

3.2.1. Point A

3.2.1.1. Max Net Forces

Rod Max Forces @ A (N)													
Engine Type	RPM	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	6000
Production Engine													
Compression		22384.09	28058.52	31992.59	34943.55	36643.52	37769.03	38617.03	39122.49	38262.86	38980.36	40832.65	41085.67
Tensile		-251.99	-166.92	-98.35	84.96	444.89	848.63	1366.24	1922.55	2564.71	3266.02	4072.18	5033.96

Table 10 – Max Compressive & Tensile Forces Acting on The Con Rod Point 'A' vs Engine Speed

As previously discussed, the net forces seen throughout each engine speed follow the same pattern. This is what you'd expect considering they're calculated from the same gas force, the only difference is the value of the inertia force subtracted from the gas force, where the increase in mass above each point, increases the inertia force.

As seen below, the graph is identical to the previous graph in terms of trends, the only difference is that the compressive force decreases in value as the tensile increases. This is caused by the increase in inertia force each cycle.

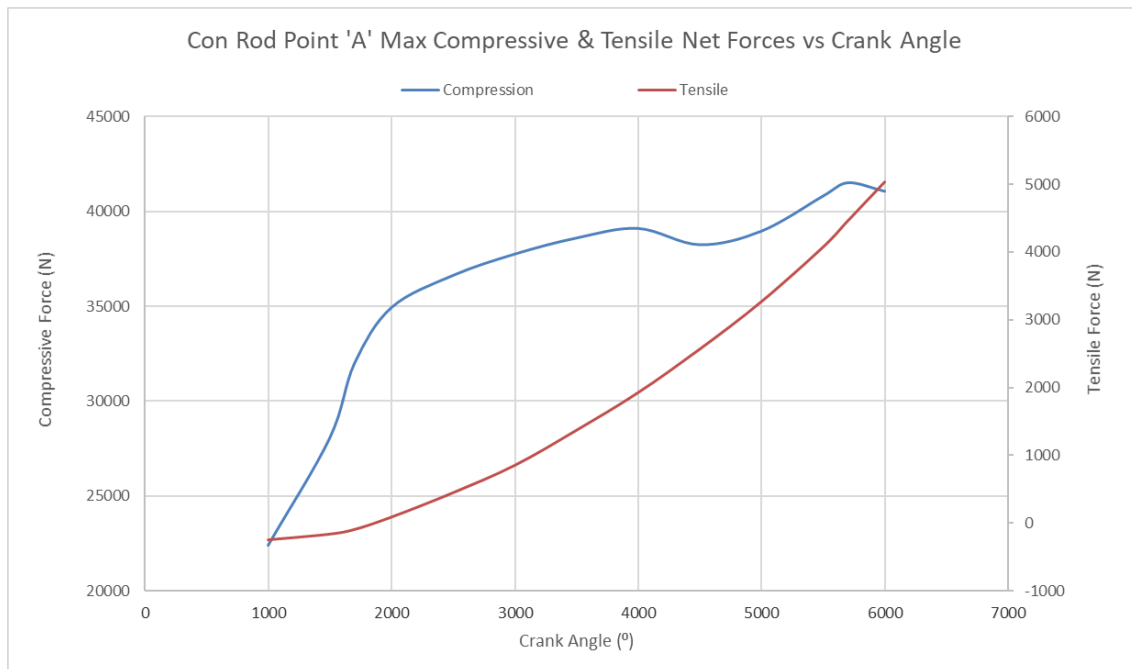


Figure 30 – Max Compressive & Tensile Net Forces Acting on Point 'A' vs Engine Speed

3.2.1.2. Mean Stress

Rod Mean Stress @ A (N/mm ²)													
RPM	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000
	26.29	33.13	37.88	41.60	44.05	45.86	47.49	48.75	48.49	50.17	53.33	54.62	54.77

Table 11 – Mean Stress Acting on The Con Rod Point 'A' vs Engine Speed

As expected, the mean stress through point 'A' followed the pattern of the compressive force exerted through the con rod, although **Figure 31**, strangely replicates the trend seen within **Figure 29**Figure 32.

Point 'A' was chosen as it is the point within the con rod with the most cross-sectional area. This point should show the least amount of stress through the rod due to the same force being displaced over a greater area.

This is evident as out of the sections of the con rod that endure both compressive and tensile forces, it shows the lowest value for mean stress.

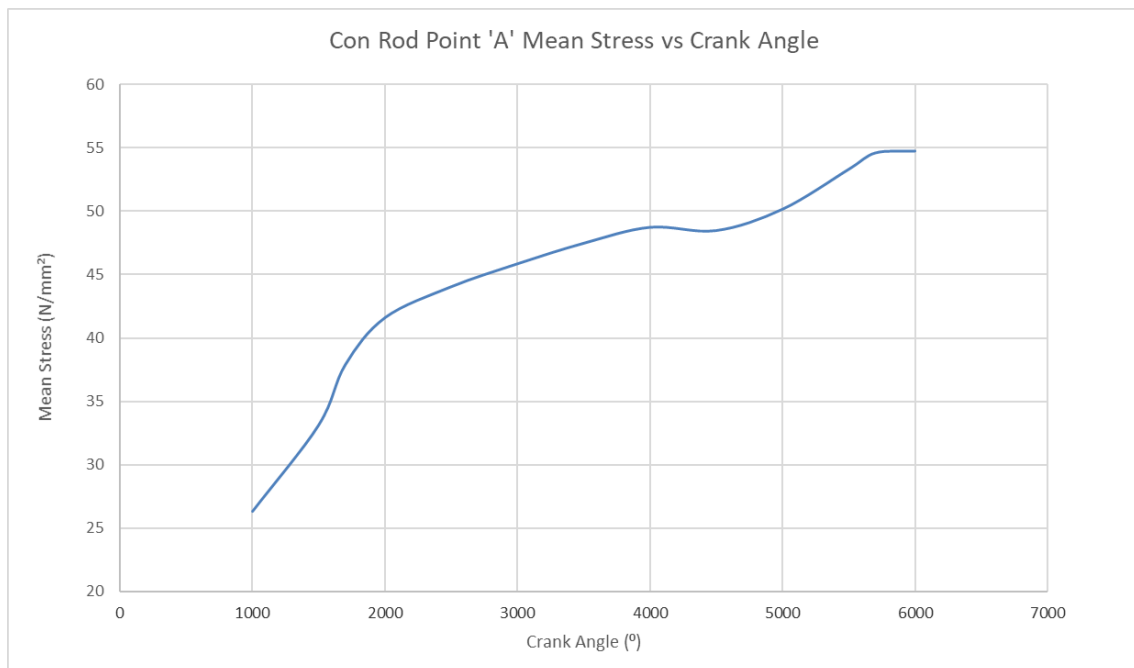


Figure 31 – Max Compressive & Tensile Net Forces Acting on Con Rod Point 'A' vs Engine Speed

3.2.1.3. Range of Stress

Rod Range Of Shear Stress @ A (N/mm²)														
	RPM													
	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000	
	53.77	67.04	76.23	82.80	85.98	87.70	88.48	88.36	84.79	84.83	87.32	88.07	85.63	

Table 12 – Range of Stress Acting on The Con Rod Point 'A' vs Engine Speed

Here are the results for the range of stress that point 'A' is subjected to.

The graph showing the range of stress below, is representative of that seen within **Figure 28**.

Unlike previous graphs, the range on point 'A' actually peaks at 3500 RPM and proceeds to increase and equally decrease as the engine speed increases. This may be correct, but the cause is unknown. The nominal stress, as well as the individual stresses and forces do not peak at this value, they are still increasing.

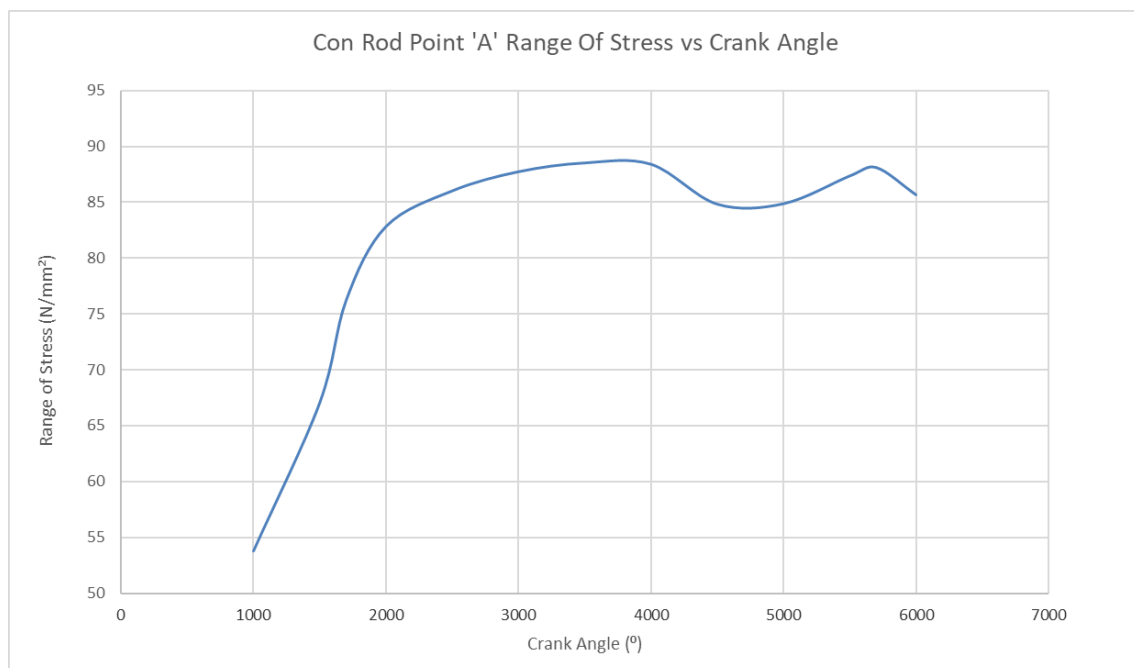


Figure 32 – Range of Stress Acting on Con Rod Point 'A' vs Engine Speed

3.2.2. Point B

3.2.2.1. Max Net Forces

Rod Max Forces @ B (N)													
Engine Type	RPM	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	6000
Production Engine													
Compression		22300.76	27880.47	31755.70	34599.78	36098.58	36984.32	37534.18	37689.52	36477.64	36770.45	38056.87	38519.81
Tensile		-158.49	39.91	171.13	481.05	1078.37	1748.38	2614.39	3564.07	4652.23	5845.86	7192.98	8745.09

Table 13 – Max Compressive & Tensile Forces Acting on The Con Rod Point 'B' vs Engine Speed

As study moves further down the con rod, it is clear to see that due to the more mass acting within the inertia force, the compressive force decreases in magnitude, whereas, the tensile increases in proportion.

When looking at point 'A' it is only until 2000 RPM where the compressive force is overcome by the inertia and becomes tensile changing its direction, point 'B', however, only takes until 1500 RPM until the force is overcome.

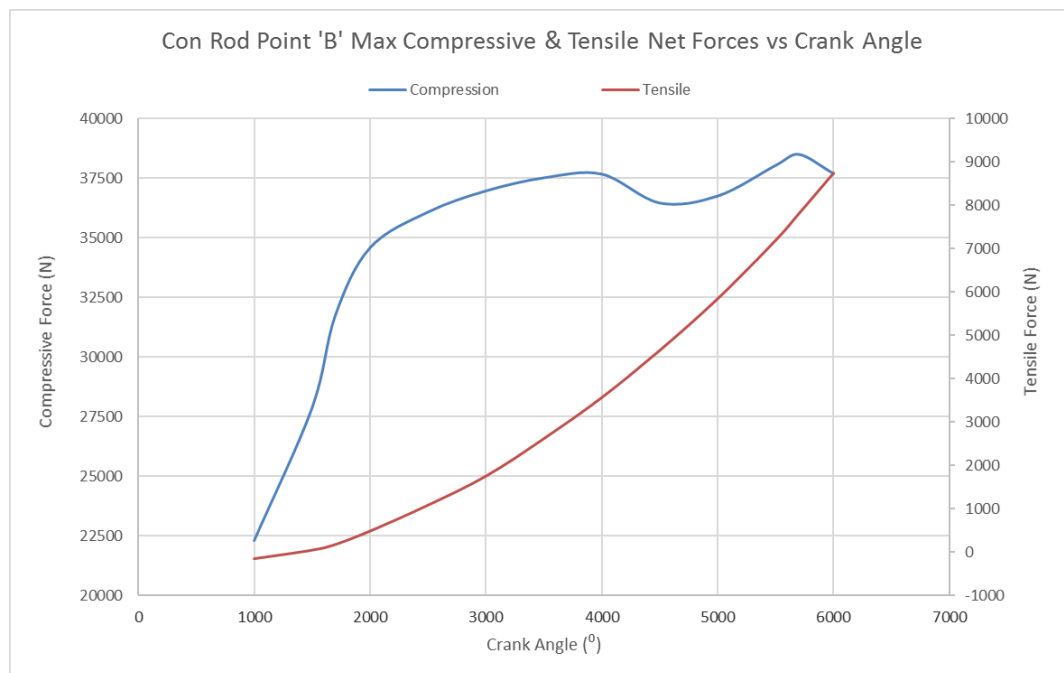


Figure 33 – Max Compressive & Tensile Net Forces Acting on Point 'B' vs Engine Speed

3.2.2.2. Mean Stress

Rod Mean Stress @ B (N/mm ²)												
	RPM											
	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	6000
	71.43	90.07	102.99	113.16	119.93	124.94	129.51	133.08	132.68	137.47	145.97	149.44
												149.85

Table 14 – Mean Stress Acting on The Con Rod Point 'B' vs Engine Speed

As before, the mean stress follows the pattern of the compressive force as in **Figure 33**, but instead of dropping off at 6000 RPM, it plateaus.

The graph below looks identical to that of the mean stress from point 'A', as you would assume from the same component experiencing the same force, although through a smaller area, the amplitude should be the same.

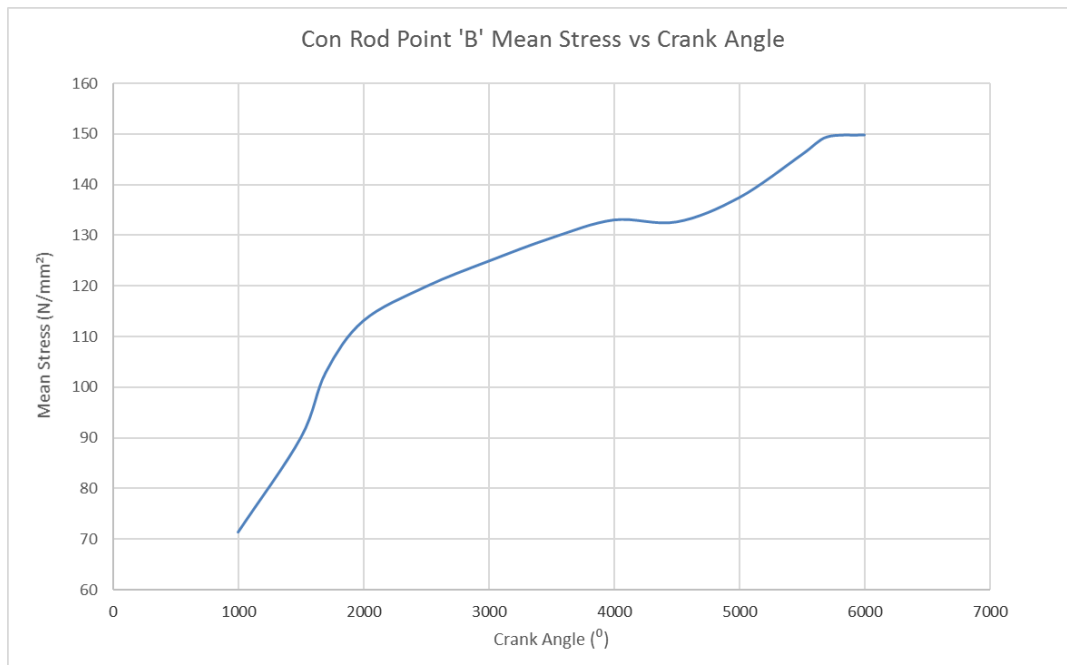


Figure 34 – Mean Stress Acting on Con Rod Point 'B' vs Engine Speed

3.2.2.3. Range of Stress

Rod Range Of Shear Stress @ B (N/mm²)														
	RPM													
	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000	
	144.90	179.62	203.77	220.12	225.94	227.33	225.29	220.16	205.33	199.51	199.12	198.15	186.87	

Table 15 – Range of Stress Acting on The Con Rod Point 'B' vs Engine Speed

The actual range of stress exerted through point 'B' actually peaks at 3000 RPM, whereas before (Figure 32), the peak was at 3500 RPM and then again at 5700 RPM.

The stress throughout point 'B' significantly increases to begin with, plateaus between 2000-4000 RPM before plummeting at 6000 RPM, seen below in Figure 35.

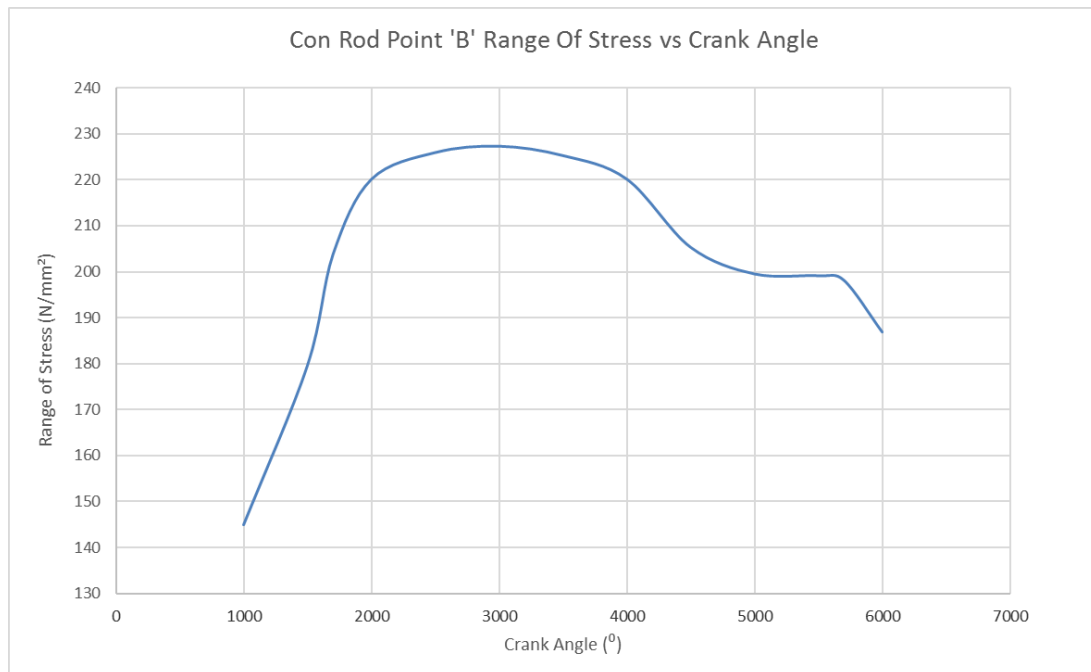


Figure 35 – Range of Stress Acting on Con Rod Point 'B' vs Engine Speed

3.2.3. Point C

3.2.3.1. Max Net Forces

Rod Max Tensile Forces @ C (N)													
Engine Type	RPM	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	6000
		0.00	204.39	385.87	787.86	1566.16	2442.00	3569.55	4822.33	6247.75	7816.48	9576.81	11580.42

Table 16 – Max Tensile Forces Acting on The Con Rod Point 'C' vs Engine Speed

There are compressive forces within the engine, however, these do not impact point 'C' as the con rod cap never experiences compressive forces throughout the engine's operation, since the compressive forces caused by the cap and the con rod being bolted together are neglected.

This follows the same trend exhibited by every other tensile force that is acting upon the con rod that have previously been examined.

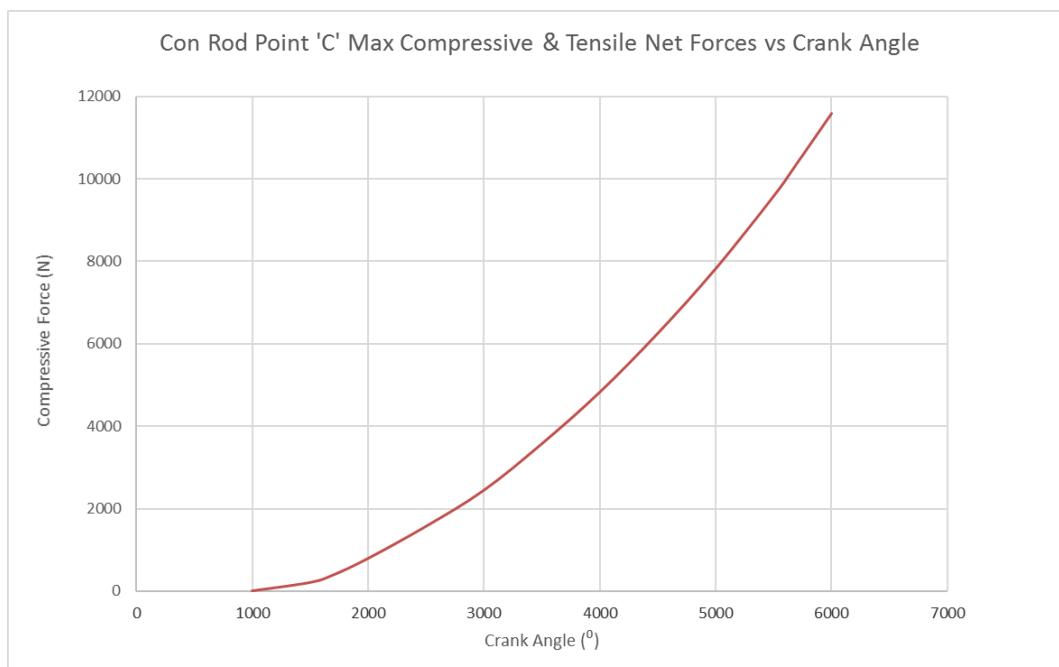


Figure 36 – Max Tensile Net Forces Acting on Point 'C' vs Engine Speed

3.2.3.2. Mean Stress

Rod Mean Stress @ C (N/mm ²)													
	RPM												
	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000
	0.00	0.34	0.65	1.32	2.62	4.09	5.97	8.07	10.45	13.08	16.02	17.34	19.38

Table 17 – Mean Stress Acting on The Con Rod Point 'C' vs Engine Speed

Throughout the entire con rod, this is the lowest mean stress value that is calculated, again this is due to the cap not seeing any compressive forces throughout the operation of the engine.

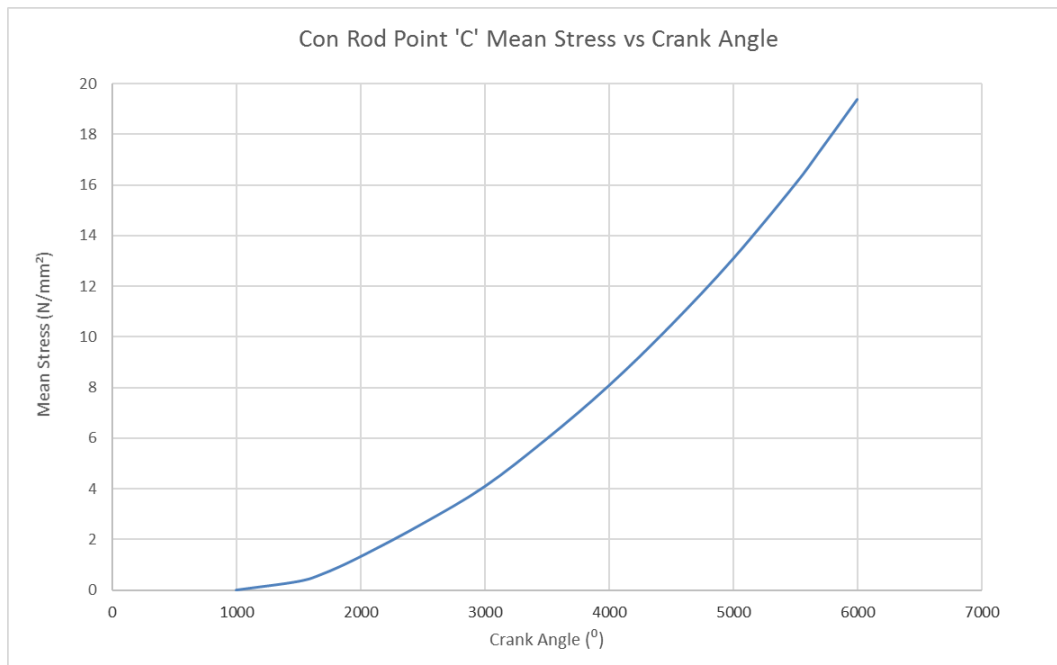


Figure 37 – Mean Stress Acting on Con Rod Point 'C' vs Engine Speed

3.2.3.3. Range of Stress

Rod Range Of Shear Stress @ C (N/mm²)														
	RPM													
	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000	
	0.00	0.68	1.29	2.64	5.24	8.17	11.94	16.14	20.91	26.16	32.05	34.69	38.75	

Table 18 – Range of Stress Acting on The Con Rod Point 'C' vs Engine Speed

Considering there is no compressive force acting upon the con rod cap, the minimum stress is zero, therefore, the range of stress is exactly that of the maximum stress value through the area of point 'C'.

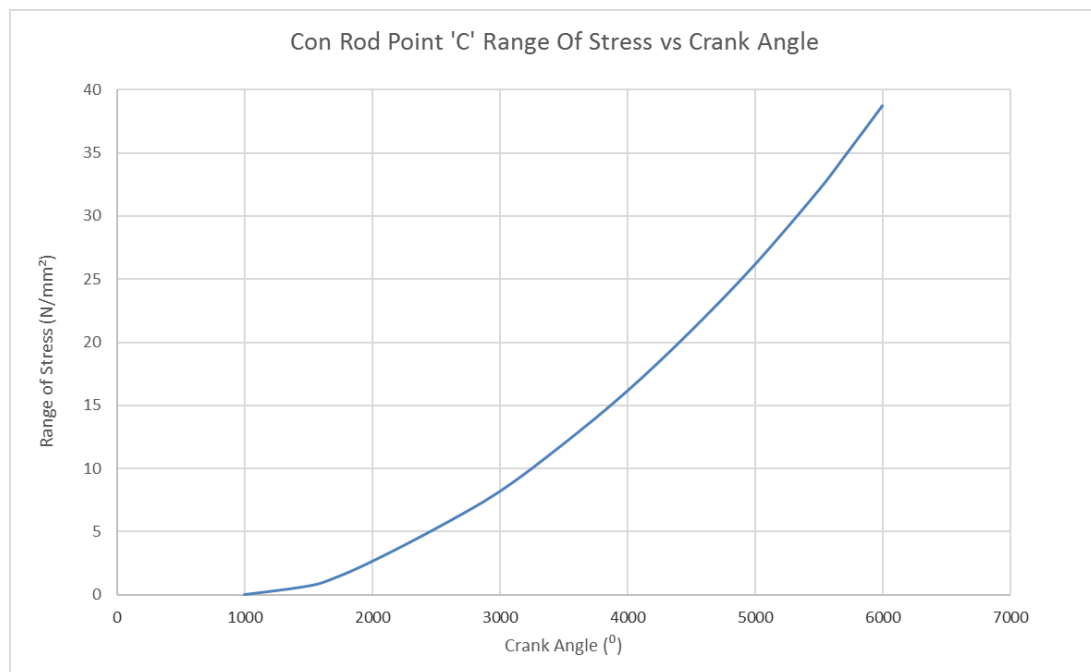


Figure 38 – Range of Stress Acting on Con Rod Point 'C' vs Engine Speed

3.3. Con Rod Cap & Bolts Forces

3.3.1. Standard Bolt

3.3.1.1. Max Inertia & Net Forces

Inertia & Net Forces (N)																
	RPM	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000	6500	7000
Reciprocating		279.67	629.27	808.26	1118.70	1747.97	2517.07	3426.02	4474.80	5663.42	6991.87	8460.17	9086.64	10068.30	11816.27	14121.56
Rotating		74.59	167.83	215.57	298.37	466.21	671.34	913.76	1193.49	1510.51	1864.82	2256.43	2423.52	2685.34	3151.55	3655.05
Total		354.27	797.10	1023.83	1417.07	2214.17	3188.41	4339.78	5668.29	7173.92	8856.70	10716.60	11510.16	12753.64	14967.82	17776.61
Tensile		0.00	207.75	386.71	779.42	1544.57	2419.72	3528.15	4757.56	6162.73	7710.69	9449.42	10230.74	11430.44	11712.77	13771.19

Table 19 – Max Inertia & Net Forces Acting on The Con Rod Bolt vs Engine Speed

Shown within Table 19 are the results for calculating the max inertia forces of both the reciprocating and rotating masses, along with the maximum tensile force that the bolts are enduring throughout the increase in engine speed.

Along with carrying out a linear interpolation of these forces up to 7000 RPM, the experiment carried out within section 2.3 is shown, whereby taking the engine up to a speed where the bolts would theoretically break is included for reference of these forces.

Below shows the comparison of inertia forces and net forces throughout the RPMs. The trends are identical up to 6000 RPM, as there is no compressive force to sway them apart from one another. But during the interpolation section (6000 – 7000 RPM) the tensile force plateaus until the 7105 RPM where it shoots up again exponentially.

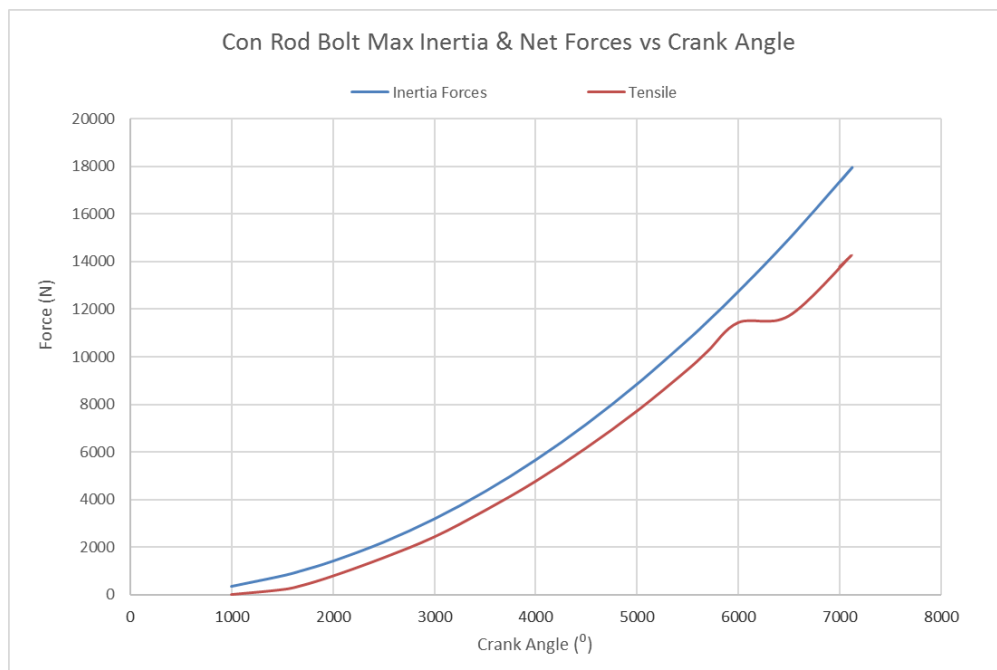


Figure 39 – Max Inertia & Tensile Net Forces Acting on Con Rod Bolt vs Engine Speed

3.3.1.2. Extra Force

Bolt Extra Force (N)															
	RPM														
	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000	6500	7105.826
	0.00	21.41	39.86	80.34	159.21	249.42	363.67	490.39	635.23	794.79	974.01	1054.55	1178.21	1207.31	1419.48

Table 20 – Extra Force Acting on The Con Rod Bolt vs Engine Speed

Table 20 shows the extra force that is applied to the con rod bolts during operation. On the climb to 6000 RPM it a steady but exponential increase as seen within the tensile forces previously.

The interpolation of 7105 RPM shows an increase from 7000 RPM, as seen below. It can be assumed that this force is what is responsible for the bolts fatigue safety factor to drop below '1' at this point.

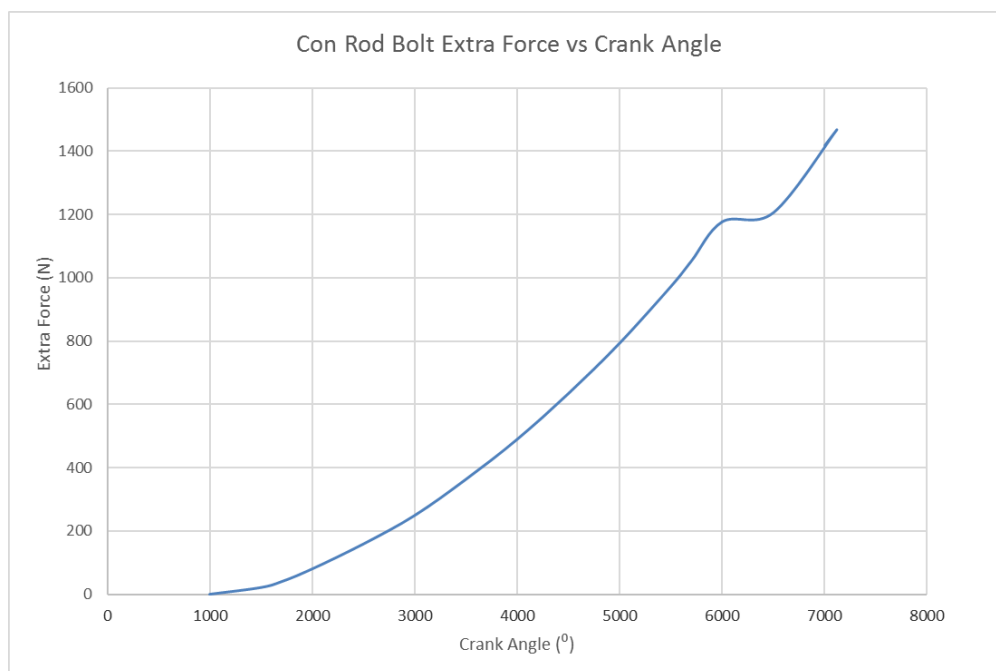


Figure 40 – Extra Force Acting on Con Rod Bolt vs Engine Speed

3.3.1.3. Alternating Load

Bolt Alternating Load (N)															
	RPM														
	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000	6500	7105.826
	0.00	0.50	0.94	1.89	3.75	5.88	8.57	11.56	14.97	18.73	22.96	24.85	27.77	28.45	33.46

Table 21 – Alternating Load Acting on The Con Rod Bolt vs Engine Speed

The alternating load can be seen above as the force that is being applied throughout the operation of the engine, as it is alternating between these values as the maximum stress and zero being the minimum, however this is not the case as there is already stress/load applied to the bolt from the original tightening.

If the bolt was originally tightened to 90% of yield, that is the value for the minimum stress occurred, where the maximum, is the minimum plus the alternating load value.

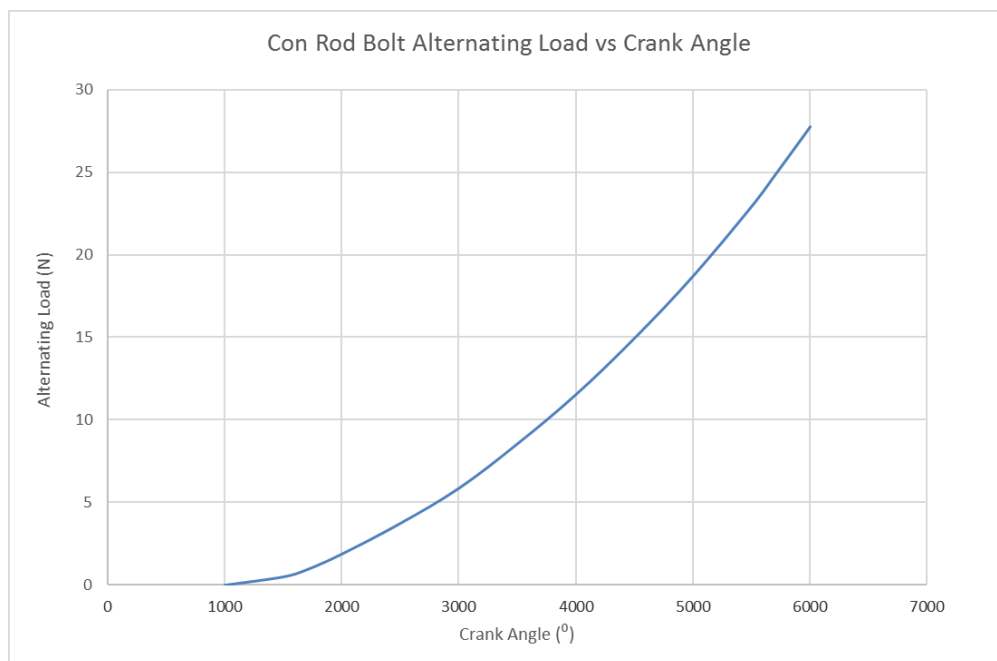


Figure 41 – Alternating Load Acting on Con Rod Bolt vs Engine Speed

3.3.2. Modified Bolt

3.3.2.1. Extra Force

MODDED Bolt Extra Force (N)													
	RPM												
	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000
	0.00	22.08	41.10	82.85	164.17	257.19	375.01	505.68	655.04	819.57	1004.38	1087.43	1214.95

Table 22 – Extra Force Acting On The Modified Con Rod Bolts vs Engine Speed

The extra force exerted onto the modified bolt has increased slightly from that of the standard bolt. The standard bolt exhibited a total of 1207 N, whereas, the modified bolt increased to 1214 N.

This increase come directly from the cap-bolt stiffness ratio, since this decreased by 0.2, making the result for $\frac{1}{1 + \frac{K_{Cap}}{K_{Bolt}}}$ from equation 2.3-4 slightly greater in value.

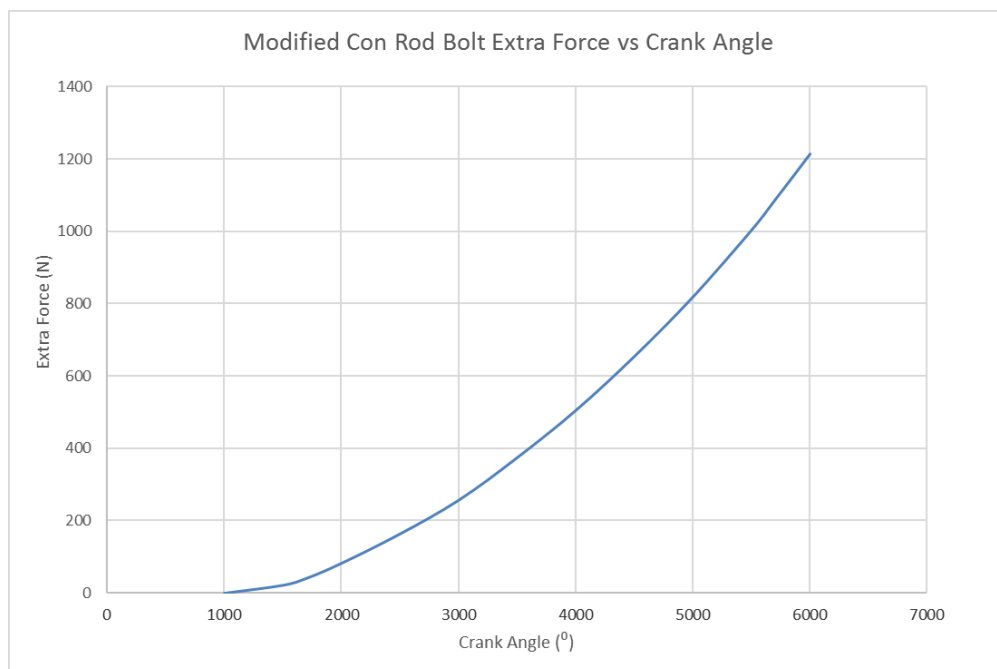


Figure 42 – Extra Force Acting on The Modified Con Rod Bolt vs Engine Speed

3.3.2.2. Alternating Load

MODDED Bolt Alternating Load (N)													
	RPM												
	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000
	0.00	0.56	1.05	2.11	4.19	6.56	9.57	12.90	16.71	20.91	25.62	27.74	30.99

Table 23 – Alternating Load Acting on The Modified Con Rod Bolt vs Engine Speed

If compared to the standard bolt, the modified bolt displays less alternating load during each engine cycle.

With a decrease of 2.47 N each cycle, the modified bolt's extra stiffness helped the situation but only slightly.

The graph below displays the same trend you would expect drawing its trend and magnitude directly from the tensile forces.

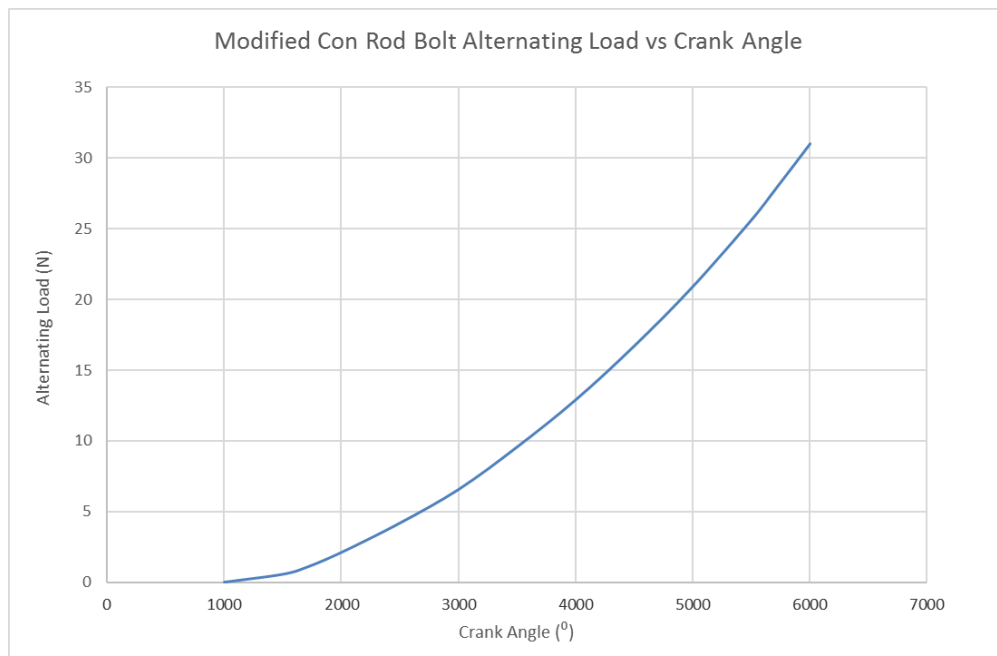


Figure 43 – Alternating Load Acting on Modified Con Rod Bolt vs Engine Speed

3.4. Crankshaft Forces

3.4.1. Max Net Forces

Crank Net Force (N)													
	RPM	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	6000
Compression		22126.01	27461.68	31239.99	33929.08	35071.74	35505.67	35561.62	35163.60	30763.40	32744.81	33461.73	33675.48
Tensile		-5.14	411.30	650.16	1079.62	1948.98	2951.00	4171.93	5803.21	9981.36	9444.36	11581.06	12515.65

Table 24 – Max Compressive & Tensile Forces Acting on The Crankshaft vs Engine Speed

Seen in Table 24 above, and reinforced from Figure 44, there doesn't seem to be a correlation to the forces acting upon the crankshaft, they mimic the compressive and tensile forces discussed previously but at 4500 RPM the both experience an odd peak.

The tensile forces up until now have exhibited a steady exponential increase, with the crankshaft however, at 4500 RPM it seems to experience a sudden surge before settling down to its regular pattern. The compressive also experience this peak but in the other direction where it dips suddenly.

This may be down to a mistake or an error within the data, but with no other data to validate against, it can only be acknowledged rather than rectified.

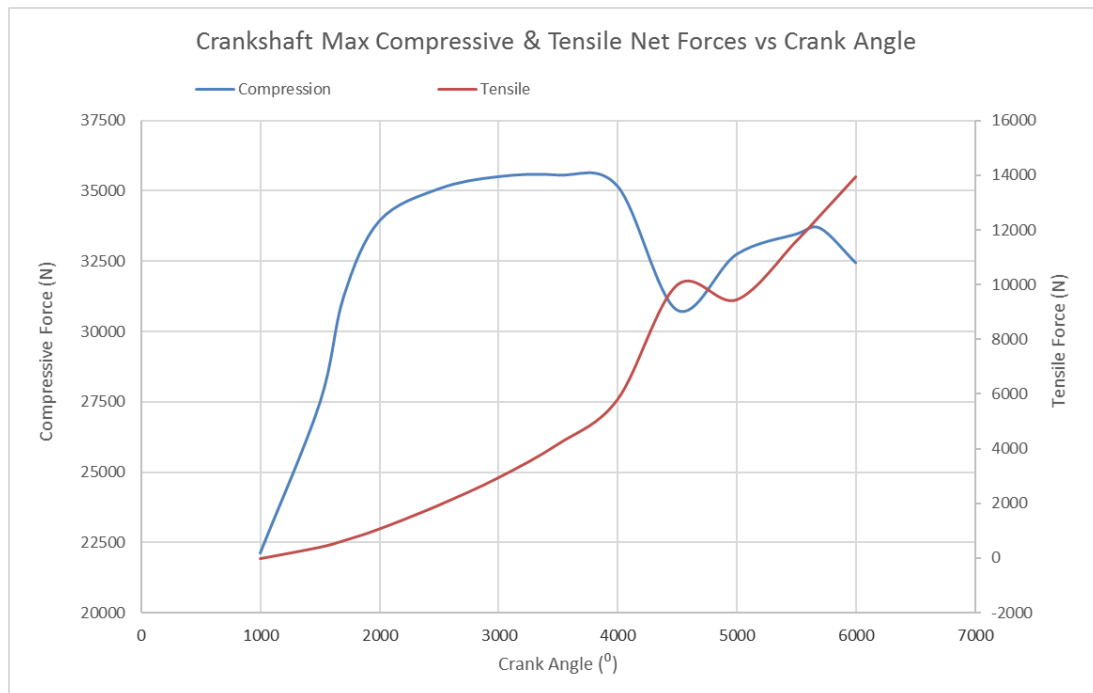


Figure 44 – Max Compressive & Tensile Net Forces Acting On The Crankshaft vs Engine Speed

3.4.2. Mean Stress

Crank Mean Stress (N/mm ²)													
Point	RPM	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	6000
A		19.39	23.72	26.84	28.84	29.14	28.70	27.76	26.09	18.84	21.01	19.91	17.10
B		-49.83	-60.92	-68.90	-73.99	-74.63	-73.38	-70.78	-66.25	-47.04	-52.69	-49.55	-42.02

Table 25 – Mean Stress Acting on The Crankshaft vs Engine Speed

The mean stress for point 'A' follows the same trend that the compressive force displayed earlier on. It peaks at 2500 RPM then decreases until a small rise at 5000 RPM.

Point 'B' follows the same trend as point 'A' flipped upside down. This was expected as when one point is experiencing compression, the other experiences tension.

The numbers within Table 25 are representative of direction, where negative values show compression and positive values indicate tension.

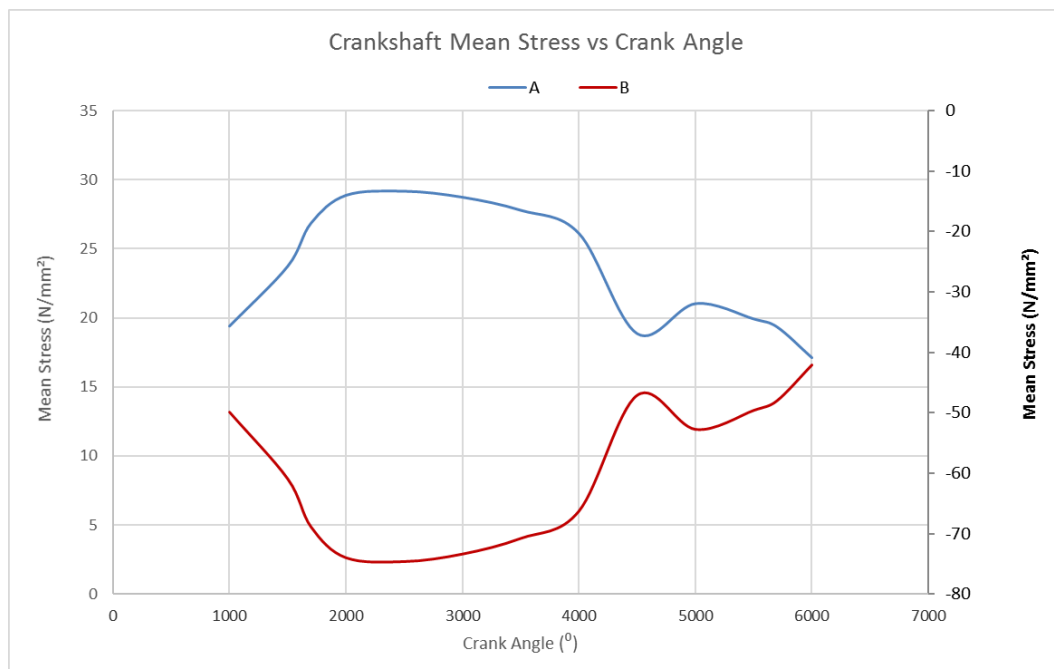


Figure 45 – Mean Stress Acting on Crankshaft Point 'A' vs Engine Speed

3.4.3. Range of Stress

Crank Range Of Stress (N/mm ²)													
Point	RPM	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	6000
A		38.75	48.78	55.78	61.19	64.61	66.99	69.07	71.03	70.10	72.70	77.43	79.46
B		99.62	125.50	143.58	157.61	166.63	173.05	178.74	184.21	183.01	189.54	202.29	208.16

Table 26 – Range of Stress Acting on The Crankshaft Point vs Engine Speed

The range of stress exerted throughout the crankshaft show the effect and magnitude of point 'B' being 20.8mm further away from the bearing centre than point 'A'.

Because point 'B' is further away, it causes it to have a higher bending moment, which in turn, increases the stress exerted upon it.

Within Figure 46, it is obvious to see that both points have a near identical trend within their range of stresses. This graph mimics that seen within Figure 34 with the mean stress of con rod point 'B'.

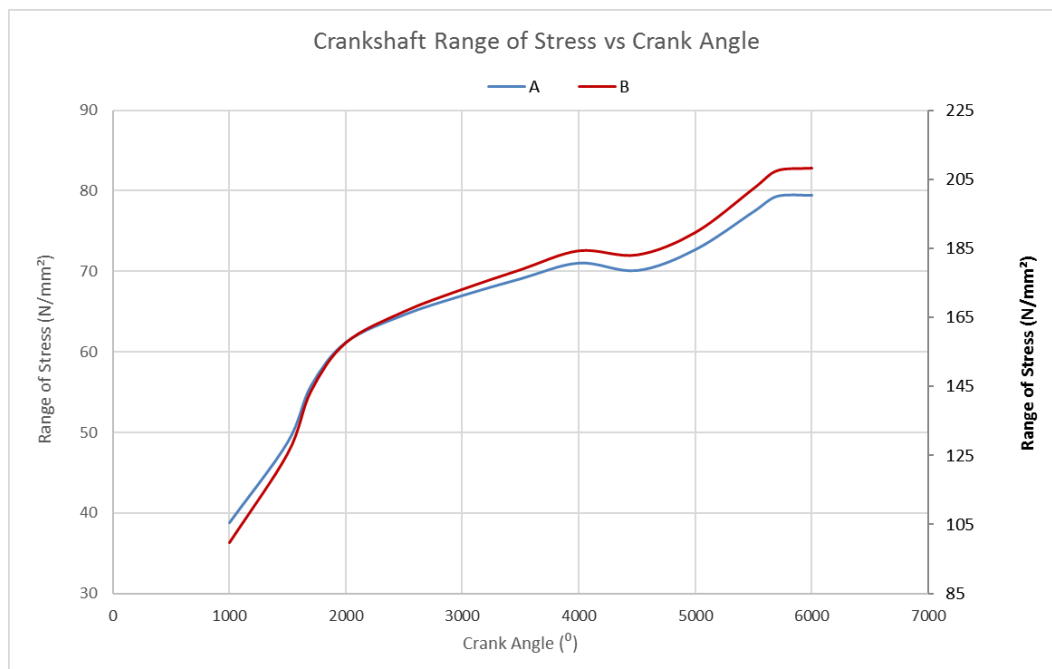


Figure 46 – Range of Stress Acting on Crankshaft Point 'A' vs Engine Speed

4) Conclusion

To conclude this experiment, it has been found how vital the fatigue safety factor is with determining how long a component will last given the stress exerted upon it during the engine's operation.

By decreasing the diameters or cross-sectional area of a component and putting the same force through it, the stress induced can dramatically increase and lead to failure of the component.

If this is the case, such as if during the process of lightening a component, material is taken away, then the best course of action is to change to a material with a higher tensile strength so that it can withstand the force through a smaller area, but the cost would need to be considered if this was a mass produced part.

When considering the design of a component, the impact of having a radius within said component can also drastically affect the endurance limit, by way of the stress concentration factor. Should a component reach a value outside of its recommended endurance limit, that part will not last for 10^7 cycles which is the standardised test for components, should they reach this number of cycles with a fatigue safety factor above '1' and not fail, they would theoretically last forever. (TWI, 2020)

5) References

Aerospace Specification Metals Inc., 2020. *AISI 4130 Steel, normalized at 870°C (1600°F)*. [Online]

Available at: <http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=m4130r>

Anderson, M., 2020. *CAD Models of Standard And Modified Con Rod Bolts*, Swansea: University of Wales Trinity St David.

Anderson, M., 2020. *Photos Taken While Measuring The Crankshaft*. [Art] (University Of Wales Trinity St. David).

Bosch, 2002. *Automotive Handbook*. 1st ed. s.l.:Bentley Publishers.

Carvill, J., 1993. *Mechanical Engineer's Data Handbook*. s.l.:Butterworth-Heinemann.

Elgin Fastener Group, 2020. *Special Steel H13*. [Online]

Available at: <https://elginfasteners.com/resources/materials/material-specifications/special-steel-h13/>

[Accessed 05 02 2020].

Encyclopedia Britannica, 2020. *Young's modulus*. [Online]

Available at: <https://www.britannica.com/science/Youngs-modulus>

[Accessed 06 02 2020].

GKN Sinter Metals, 2019. *Materials and Processes*. [Online]

Available at: <https://www.gknpm.com/globalassets/downloads/powder-metallurgy/2018/materials-and-processes.pdf/>

[Accessed 15 10 2019].

Jelaska, D., 2011. *Fatigue safety factor general formula proposition for the prestressed components subjected to arbitrary CA stress cycling process*, Split, Croatia: University of Split-FESB.

Learn Easy, 2020. *2nd Moment of Area*. [Online]

Available at:

http://www.learneasy.info/MDME/MEMmods/MEM30006A/Area_Moment/Area_Moment.html

McDonald, M., 2019. *Endurance Limit*. Swansea: s.n.

McDonald, M., 2020. *Adanced Engine Design*. Swansea, s.n.

Merriam Webster, 2020. *Definition of neutral axis*. [Online]
Available at: <https://www.merriam-webster.com/dictionary/neutral%20axis>

Pilkey, W. D., 1997. *Peterson's Stress Concentration Factors*. 2nd ed. New York: John Wiley & Sons, Inc..

Pilkey, W. D. & Pilkey, D. F., 2008. *Peterson's Stress Concentration*. 3rd ed. New Jersey: John Wiley & Sons, Inc..

Richardson, D., 2020. *CAD Models Of 1.6 Ecoboost Engine Components*, Swansea: University of Wales Trinity St. David .

Steel Number, 2020. *34CrMo4 (1.7220)*. [Online]
Available at: http://www.steelnumber.com/en/steel_composition_eu.php?name_id=333
[Accessed 29 01 2020].

The Engineering Toolbox, 2020. *Poisson's ratio*. [Online]
Available at: https://www.engineeringtoolbox.com/poissons-ratio-d_1224.html
[Accessed 07 02 2020].

TWI, 2020. *FATIGUE TESTING*. [Online]
Available at: <https://www.twi-global.com/technical-knowledge/job-knowledge/fatigue-testing-078>
[Accessed 06 02 2020].

6) Appendix

6.1. Appendix A – Stress Concentrations Factors for Con Rod Points ‘A’ & ‘B’

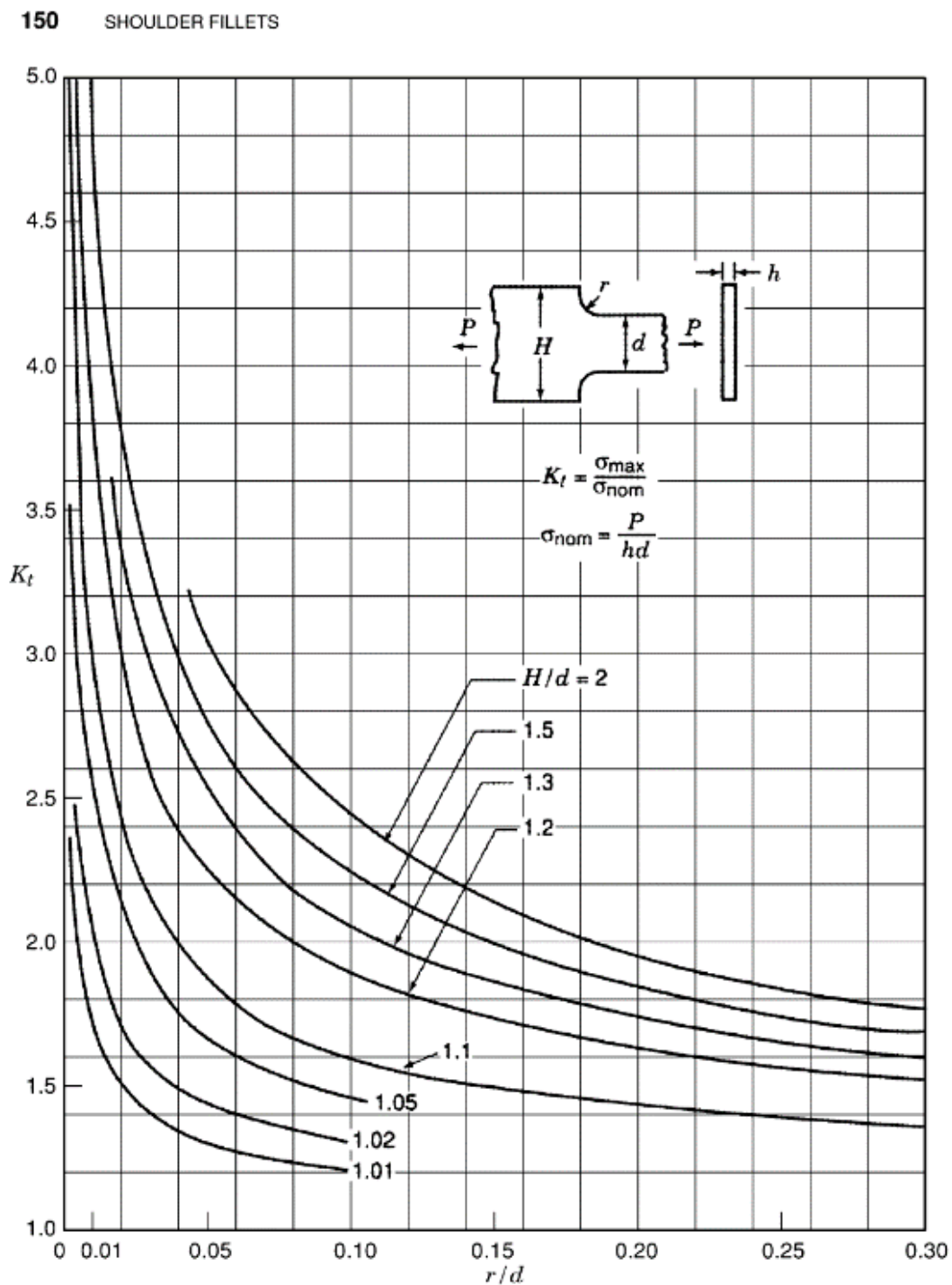


Chart 3.1 Stress concentration factors K_t for a stepped flat tension bar with shoulder fillets (based on data of Frocht 1935; Appl and Koerner 1969; Wilson and White 1973).

Figure 47 – Stress Concentration Factors for A Stepped Flat Tension Bar with A Shoulder Fillet (Pilkey, 1997)

6.2. Appendix B – Stress Concentrations Factors for Con Rod Point ‘C’

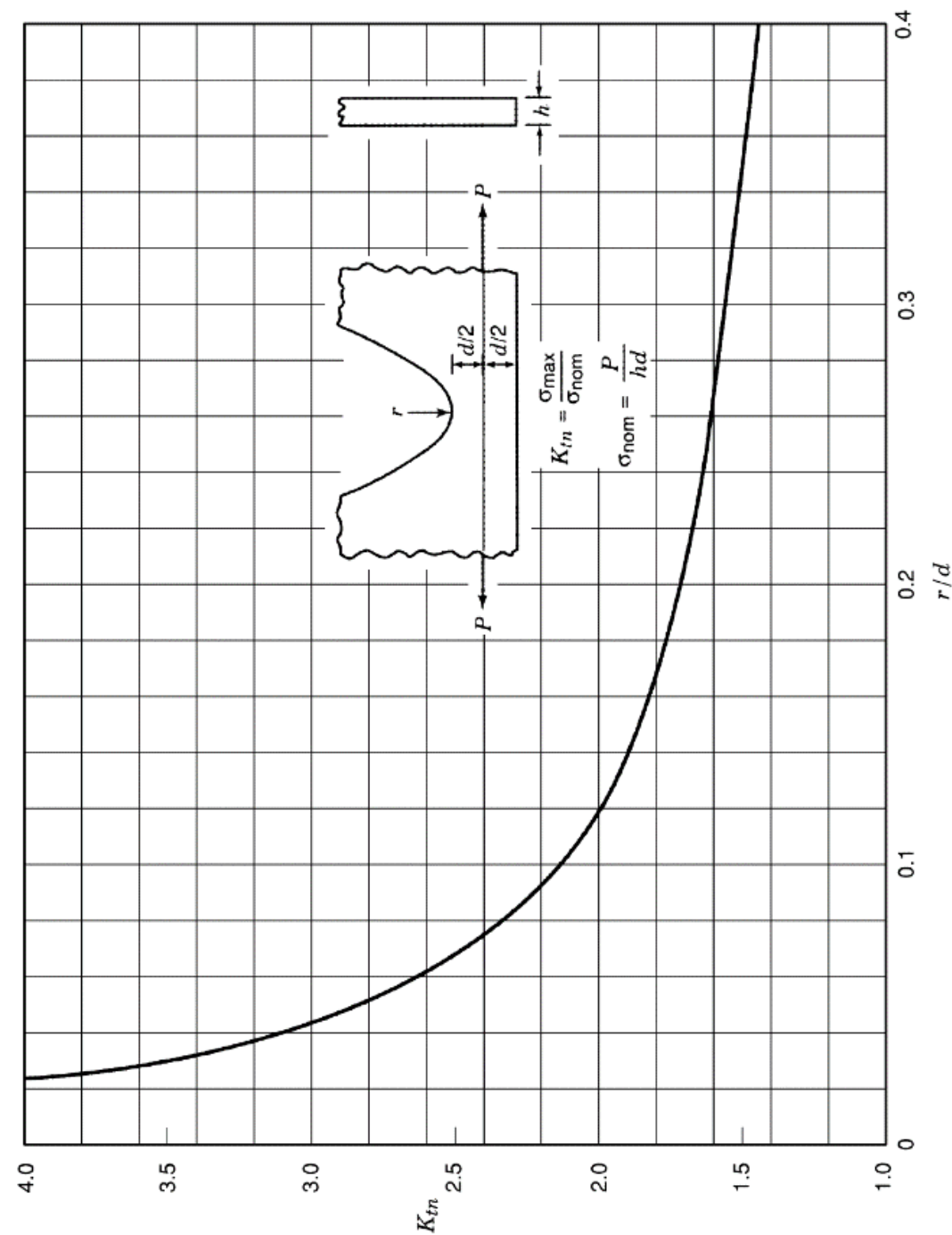


Chart 2.8 Stress concentration factors K_{tn} for tension loading of a semi-infinite thin element with a deep hyperbolic notch. tension loading in line with middle of minimum section (approximate values; Neuber 1958).

Figure 48 – Stress Concentration Factors for A Semi-Infinitely Thin Element with A Deep Hyperbolic Notch (Pilkey, 1997)

6.3. Appendix C – Stress Concentrations Factors for The Con Rod Bolts

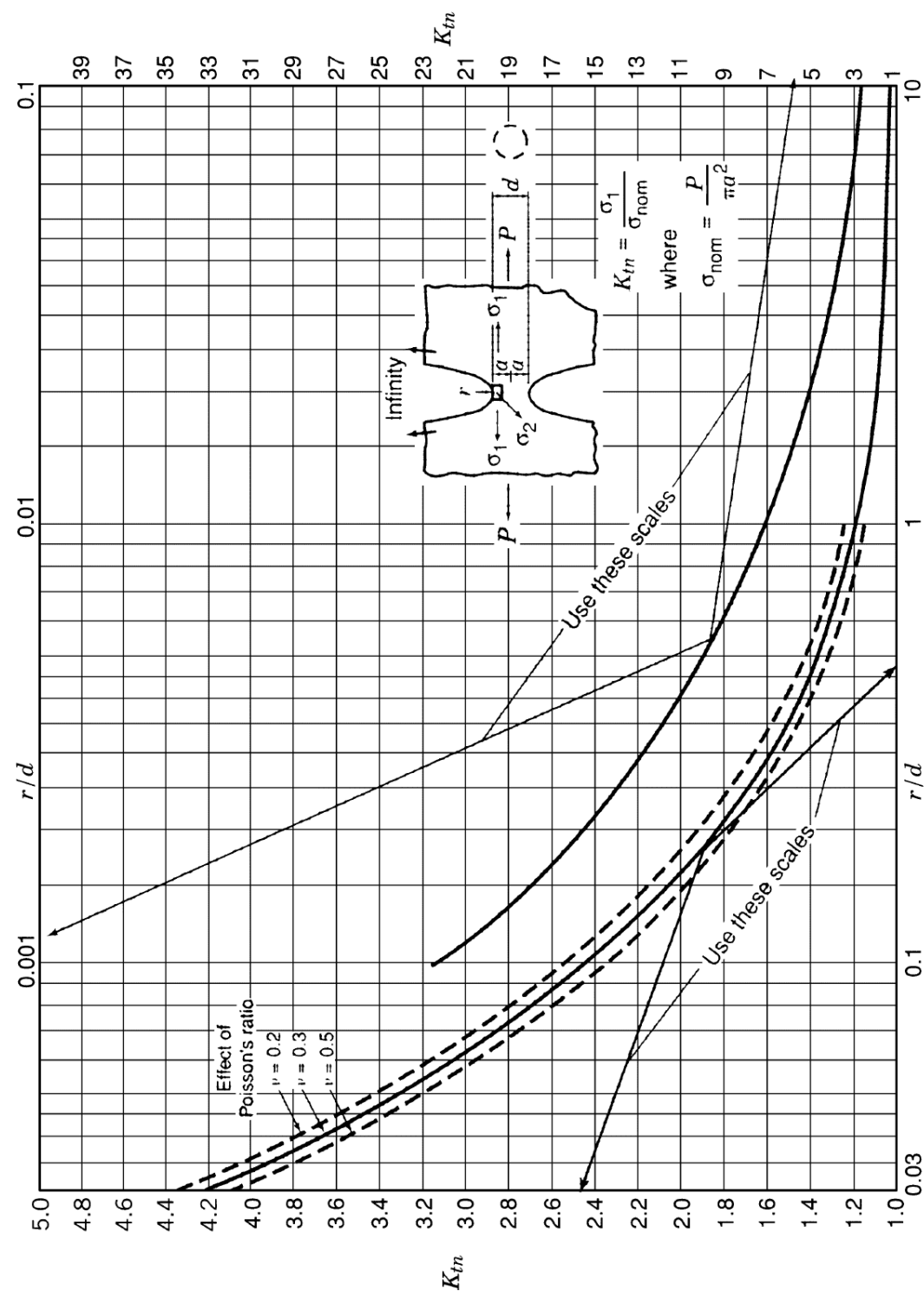


Chart 2.18 Stress concentration factors K_m for a deep hyperbolic groove in an infinitely wide member, three dimensional case, in tension (Neuber 1958 solution).

Figure 49 – Stress Concentration Factors for A deep Hyperbolic Groove In An Infinitely Wide Member (Pilkey, 1997)

6.4. Appendix D – Stress Concentration Factors for Crankshaft Fillets

CHARTS 429

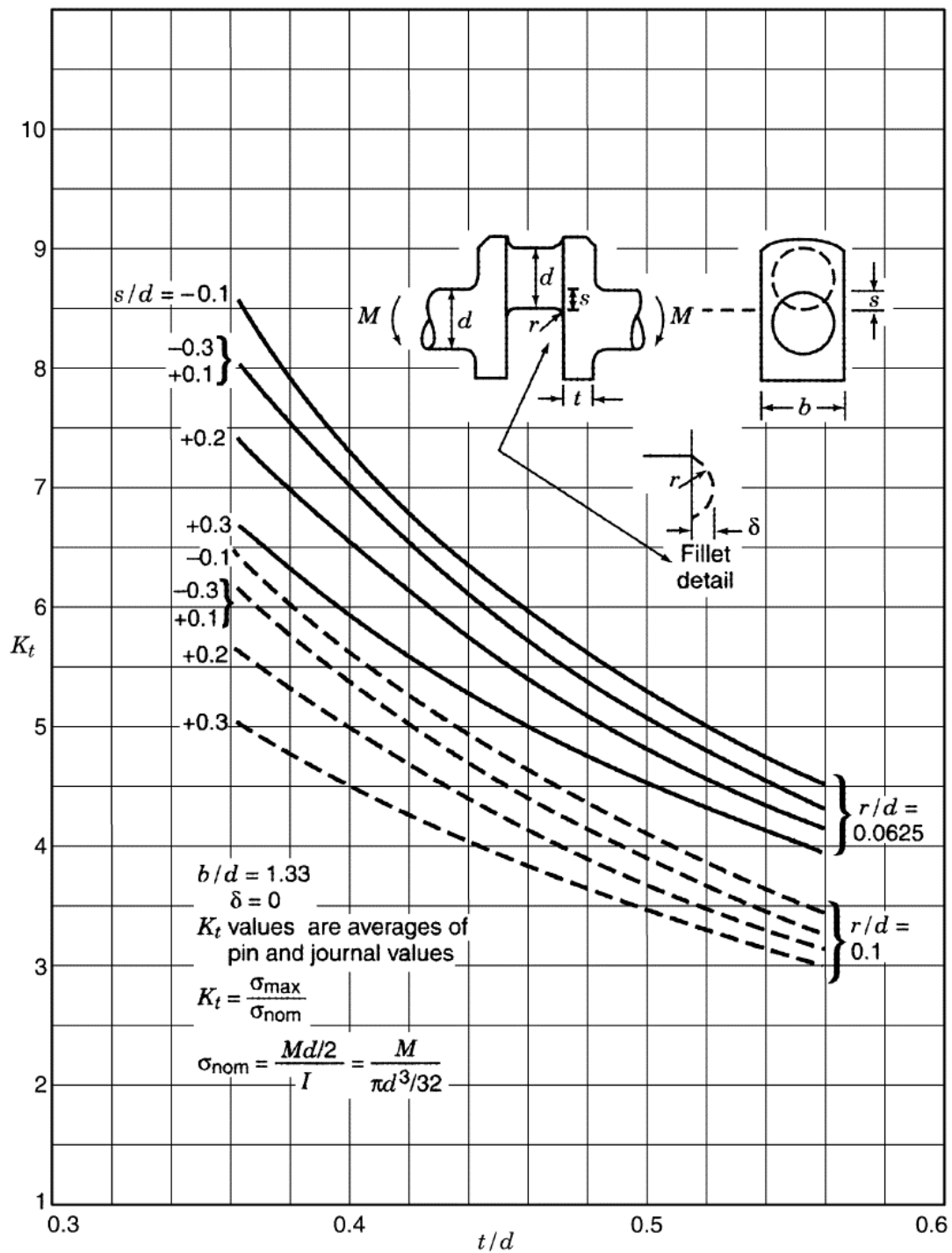


Chart 5.18 Stress concentration factors K_t for a crankshaft in bending (from strain gage values of Arai 1965).

Figure 50 – Stress Concentrations Factors for A Crankshaft In Bending (Pilkey, 1997)

6.5. Appendix E – Stress Concentration Factors for Crankshaft Fillets

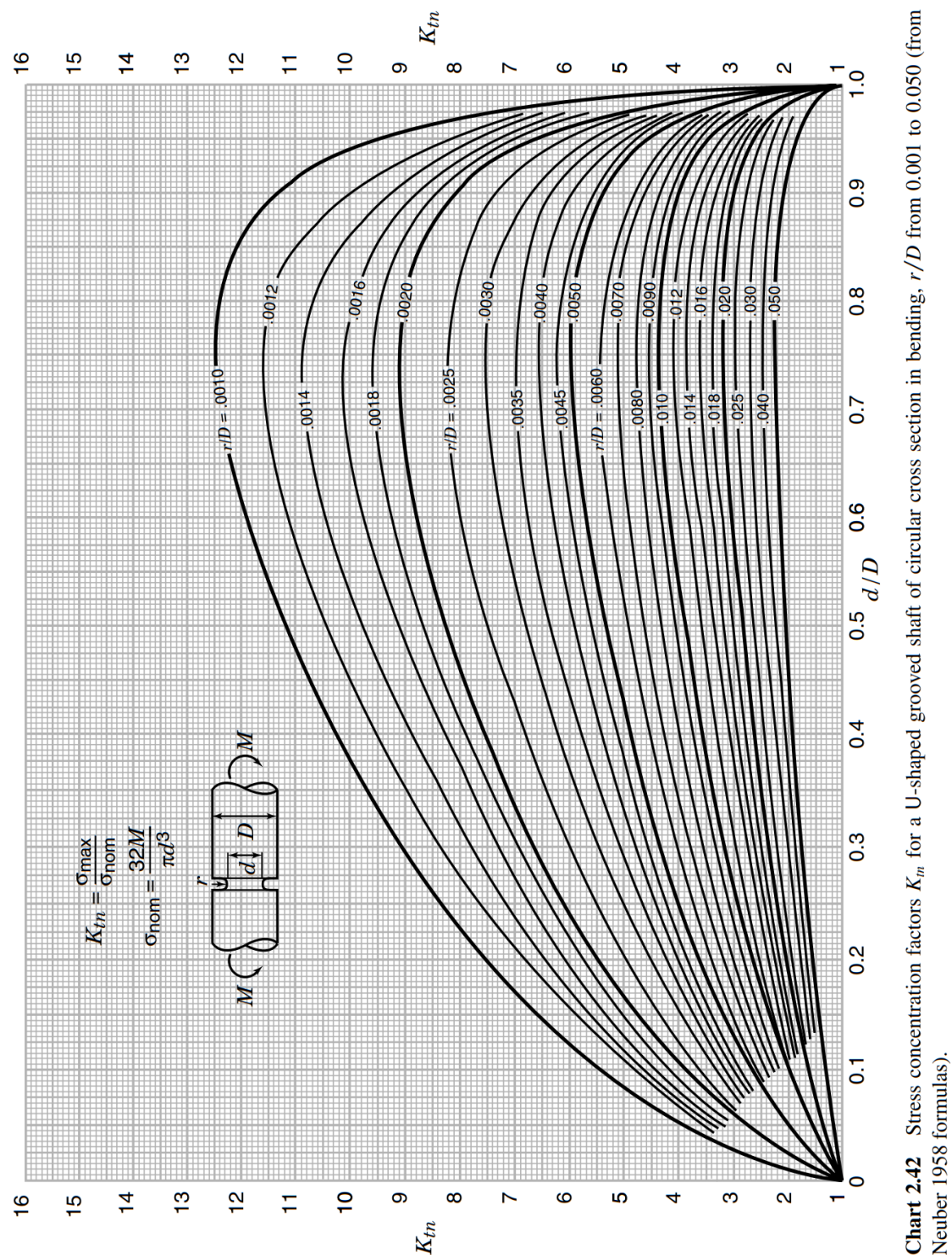


Chart 2.42 Stress concentration factors K_{tn} for a U-shaped grooved shaft of circular cross section in bending, r/D from 0.001 to 0.050 (from Neuber 1958 formulas).

Figure 51 – Stress Concentration Factors for A U-Shaped Grooved Shaft In Bending (Pilkey & Pilkey, 2008)

6.6. Appendix F – Solidworks Static Study Performed Using 42kN Gas Force

This study was conducted on the con rod from point 'A' and below, this neglected the hoop at the top of the component to easily apply the force within Solidworks.

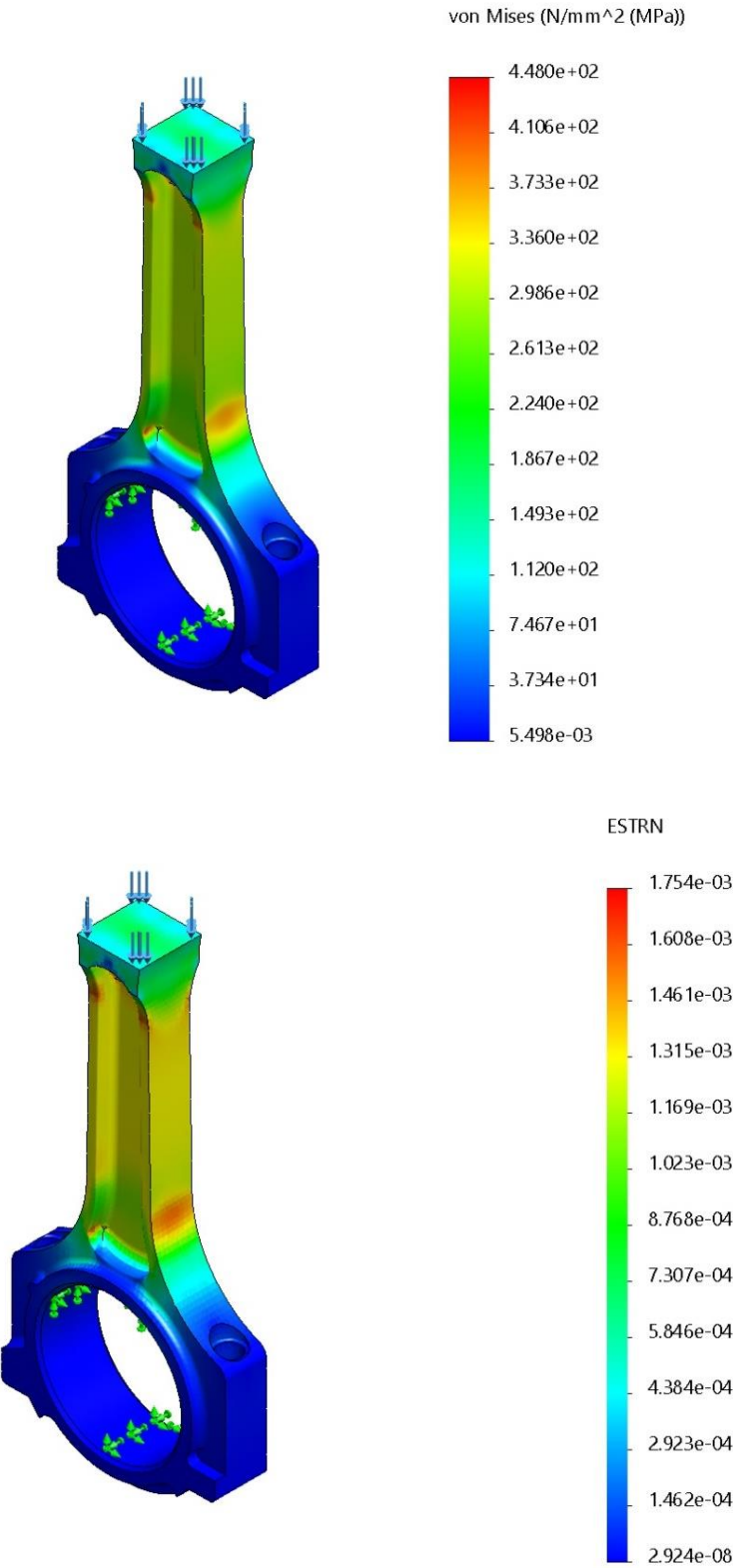


Figure 52 – SolidWorks Static Study Showing Stress & Strain Results

6.7. Appendix G – Photographs for Crankshaft Measurements



Figure 54 – Measuring The Angled Distance From Point 'A' to 'B' (Anderson, 2020)



Figure 53 – Measuring The Horizontal Distance From Point 'A' To 'B' (Anderson, 2020)



Figure 55 – Measuring The Diameter Of The Crank Journal (Anderson, 2020)

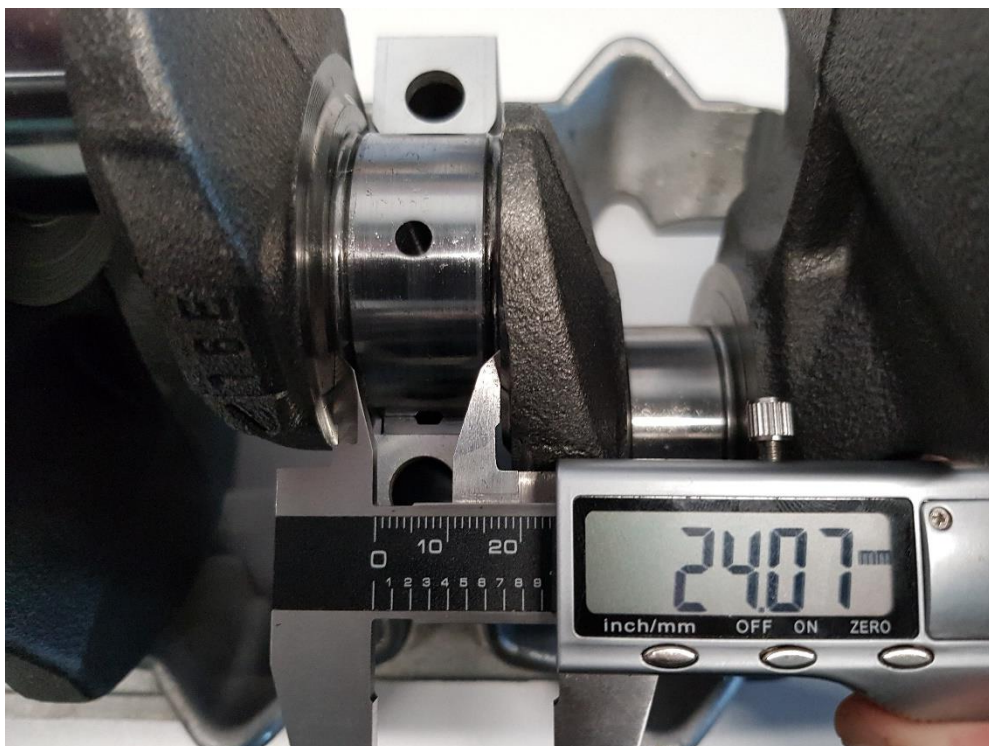


Figure 56 – Measuring The Length Of The Crank Journal (Anderson, 2020)



Figure 57 – Measuring The Diameter Of The Big End / Crank Pin (Anderson, 2020)

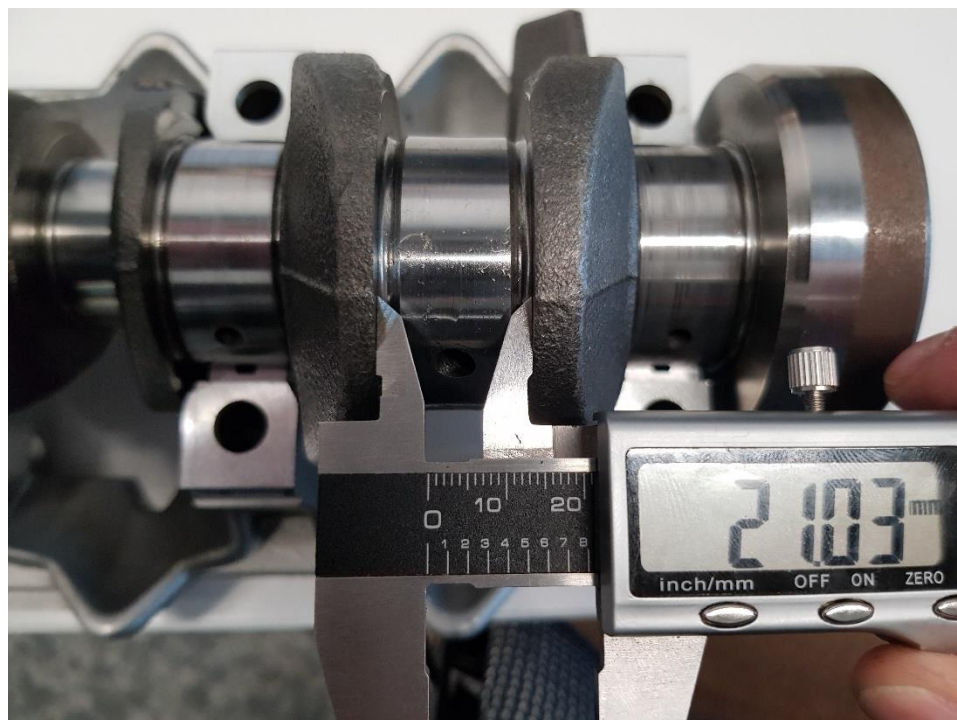


Figure 58 – Measuring The Length Of The Big End / Crank Pin (Anderson, 2020)



Figure 59 – Measuring The Diameter Of The Crank Journal Bearing Shell (Anderson, 2020)

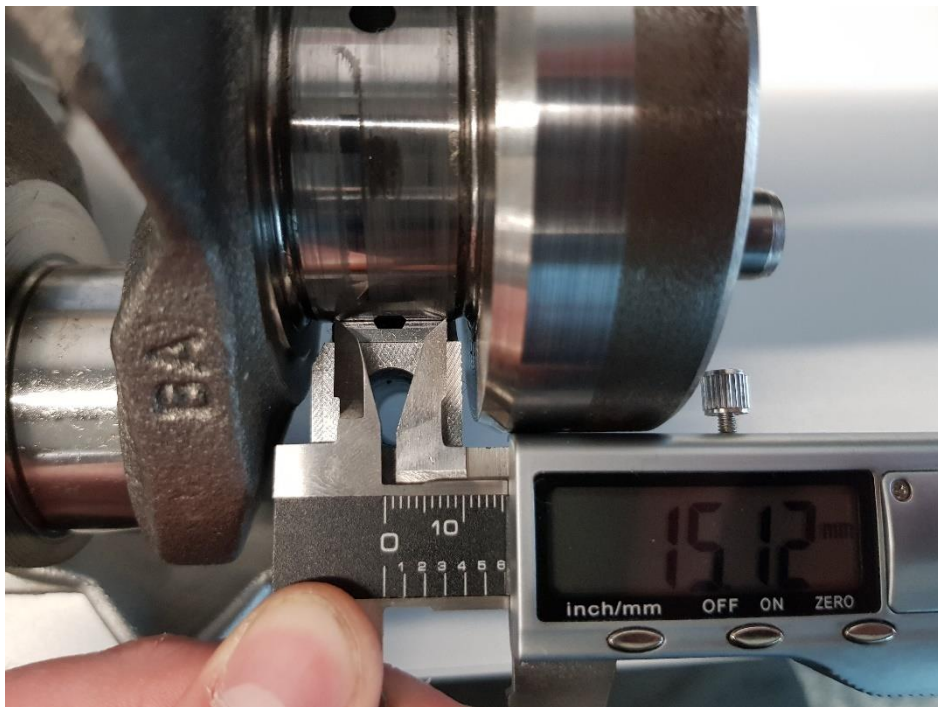


Figure 60 – Measuring The Length Of The Crankshaft Bearing Shell (Anderson, 2020)