



University of Wales Trinity Saint David

School of Engineering

Calculations on Piston, Connecting Rod & Crankshaft
Assemblies

Advanced Engine Design

M6X00410

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P148699

January 2020

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1 Nomenclature

M_{Rec} – Reciprocating Mass

M_{Rot} – Rotational Mass

K_B – Bolt Stiffness

K_{ROD} – Connecting Rod Cap Stiffness

S_x – Bolt Stretch

UTS – Ultimate Tensile Strength

ω – Angular Velocity

y – Distance from the Neutral Axis

COG – Centre of Gravity

E – Youngs Modulus

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6 Introduction

This report consists of a study continuing from a previous experiment that was carried out by the author regarding the Con-Rod Accelerations, Force & Displacements within a 1600cc Ford EcoBoost GDI four-cylinder in-line engine, with high-pressure direct fuel injection.

This paper is an investigation into the forces and stresses exerted upon certain components within the engine, calculating the safety factors to compare to what Ford are required to work to in real life. The components investigated within this study are as follows; the Gudgeon Pin, Connecting Rod, Connecting Rod Bolts and the Crankshaft, all calculations were carried out with the use of Microsoft Excel. The report will explain the methodology used behind the calculations along with an analysis of each component and an overall conclusion.

7 Gudgeon Pin

7.1 Methodology

All the following calculations were carried out at an engine speed of 6000RPM as this is when the engine experiences its peak compressive and tensile forces. Furthermore, the gas force acting upon the piston is putting the gudgeon pin into tension. Therefore, the gas force compressing the piston is considered tensile on the gudgeon and the inertia force is considered as a compressive force acting upon the pin.

$$Piston\ Inertia\ Force = Mass_{Rec} \times Piston\ Acceleration$$

Equation 1 Piston Inertia Force

The reciprocating mass used within equation one is the piston assembly minus the pin, the masses can be found in Appendix 13.1, and the piston acceleration values were calculated against each degree of crank angle in the previous assignment (Assignment 1A, Con-Rod Accelerations, Forces & Displacements).

$$Acceleration = R\omega^2 \cos\theta + \frac{R}{L} R\omega^2 \cos 2\theta$$

Equation 2 Acceleration

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

Equation 3 Engine Speed Rad/s

Where R is equal to crank throw and ω is referring to angular velocity in Rad/s, Please Refer to Appendix 13.2.

$$NET\ Force = Gas\ Force - Piston\ Inertia\ Force$$

Equation 4 Net Force

To calculate the compressive and tensile forces acting upon the piston the following two equations were used.

$$Maximum\ Compressive\ Force = Maximum\left(\sum Net\ Force\right)$$

Equation 5 Maximum Compressive Force

$$Maximum\ Tensile\ Force = Minimum\left(\sum Net\ Force\right)$$

Equation 6 Maximum Tensile Force

Gudgeon Pin Specifications				
UTS	670.00	N/mm^2		
Endurance Limit	301.50	N/mm^2		
Outer Diameter	21.00	mm	0.0210	m
Outer Radius	10.50	mm	0.0105	m
Inner Diameter	14.00	mm	0.0140	m
Inner Radius	7.00	mm	0.0070	m
y (Distance from neutral axis)	10.50	mm	0.0105	m
Distance from top edge of rod to centre of pin bore (A)	7.75	mm	0.0078	m
Distance from bottom edge of rod to centre of pin bore (B)	7.90	mm	0.0079	m
Length	46.00	mm	0.0460	m
Length of pin sticking out	17.25	mm	0.0173	m
Distance from Rod Edge to COG Gudgeon Pin	8.01	mm	0.0080	m

Table 1 Gudgeon Pin Specifications

Unfortunately, the exact material of the gudgeon pin was unattainable. Research was carried out on commonly used materials, and the assumption was made to use AISI 4130 grade Steel, with a UTS of 670 N/mm², a modulus of Elasticity of 205 GPa and a density of 7850 kg. Next, the endurance limit could be calculated.

$$\text{Endurance Limit} = \text{UTS} \times 0.45$$

Equation 7 Endurance Limit

Next, in order to be able to calculate the bending moment, the distance from the centre of gravity to the bending point of the pin was calculated from the SolidWorks Cad model as shown in figure 1 below.

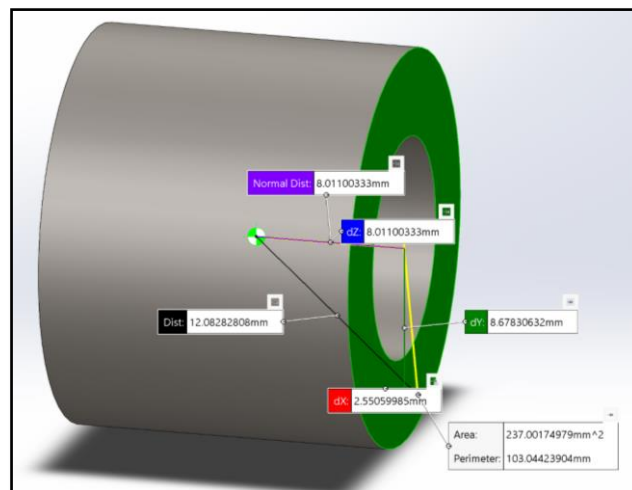


Figure 1 Distance from Rod Edge to COG of Gudgeon Pin

In addition to this, the Inertia force on the pin sticking out also needed to be calculated first in order to calculate the bending moment.

$$\text{Mass of Pin Sticking Out} = \text{Length} \times \pi(R_2^2 - R_1^2) \times \text{Density of Steel}$$

Equation 8 Mass of Pin Sticking Out

The mass of the pin sticking out can be found in Appendix 13.2.

$$\text{Pin Inertia Force} = \text{Mass of Pin Sticking out} \times \text{Acceleration}$$

Equation 9 Pin Inertia Force

$$\text{Bending Moment} = \left(\left(\frac{F_{net}}{2} \right) \times L_2 \right) - (\text{Pin Inertia Force} \times L_1)$$

Equation 10 Bending Moment

$$I = \frac{\pi}{64} \times (D_1^4 - D_2^4)$$

Equation 11 Second Moment of Area

The result for the second moment of Area can also be found in Appendix 13.2. Now, the following equations could be calculated in order to produce a Goodman's diagram based on the stresses exerted onto the gudgeon pin during the engines cycle.

$$\sigma = \frac{\text{Bending Moment} \times y}{I}$$

Equation 12 Bending Stress (σ)

$$\tau = \frac{\frac{F_{net}}{2} - \text{Inertia Pin Force}}{\frac{\pi}{4} \times (D_2^2 - D_1^2)}$$

Equation 13 Shear Stress

$$\sigma_{Resultant} = \sqrt{\sigma_{bending}^2 + (2 \times \tau)^2}$$

Equation 14 Resultant Stress

Equation 14 is used to calculate the resultant stresses, also referred to as the von mises stress equation. Von Mises is used to determine when a ductile material reaches the point of yield. Therefore, meaning the deformation is no longer be considered linear and is instead considered to be plastic. The Von Mises concept originates from the distortion of energy theory “Yielding occurs when the distortion energy reaches a critical value” (Sim Scale, n.d.). Regarding the Gudgeon pin, the Compressive resultant stress is used for σ_{Min} and the tensile resultant stress is σ_{Max} .

$$\text{Range of Stresses} = \sigma_{Max} - \sigma_{Min}$$

Equation 15 Range of Stress

$$\text{Mean Stress} = \frac{\sigma_{Max} + \sigma_{Min}}{2}$$

Equation 16 Mean Stress

$$\text{Allowable Range} = 2 \times \text{Endurance Limit} \times \left(1 - \frac{\text{Mean Stress}}{UTS}\right)$$

Equation 17 Allowable Range

$$\text{Fatigue Safety Factor} = \frac{\text{Allowable Range}}{\text{Range of Stress}}$$

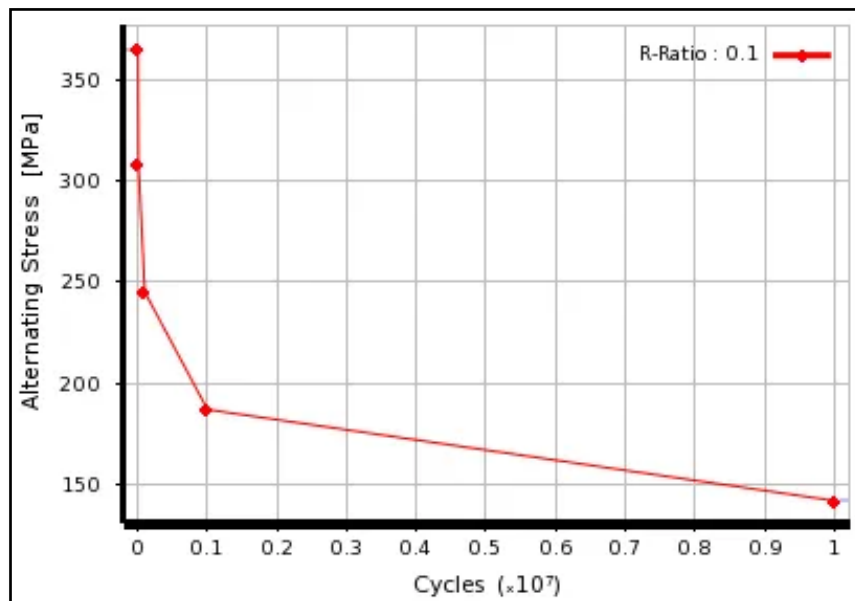
Equation 18 Fatigue Safety Factor

7.2 Results & Analysis

Range of Stresses	336.8086	N/mm ²
Mean Stress	138.0758	N/mm ²
Allowable Range	478.7318	N/mm ²
Fatigue Safety Factor	1.4214	

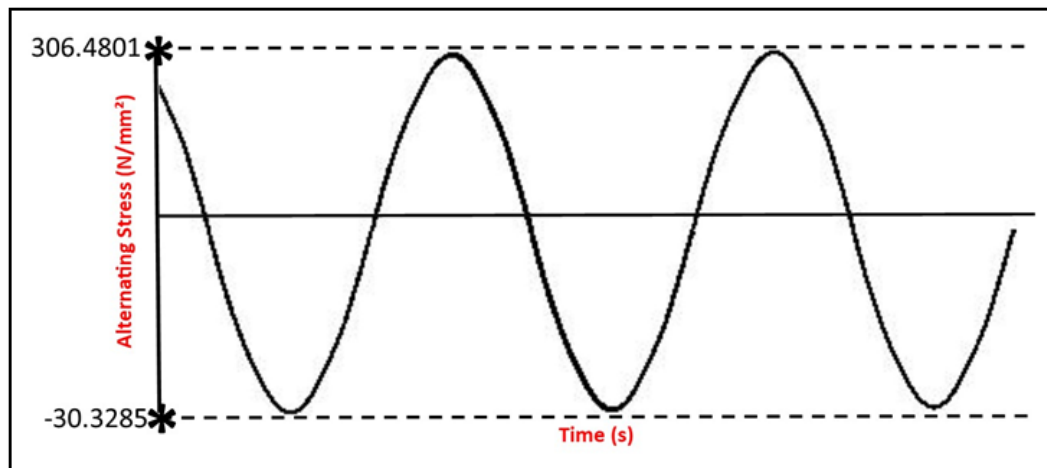
Table 2 Gudgeon Pin results

Table 2 displays all the variables required to produce a Goodman's diagram. It also shows the fatigue safety factor of the Gudgeon pin. The safety factor of a component is described as “the load carrying capability of a product beyond the actual load” (CAE bay, 2018). In other words, it determines how much stronger a component is than what is required of it under the intended loading conditions. a component with an SF of 1 will only support its design load, anymore and the product will fail. Therefore, for the gudgeon pin, 1.4214 times the design load can be exerted upon it before it will fail.



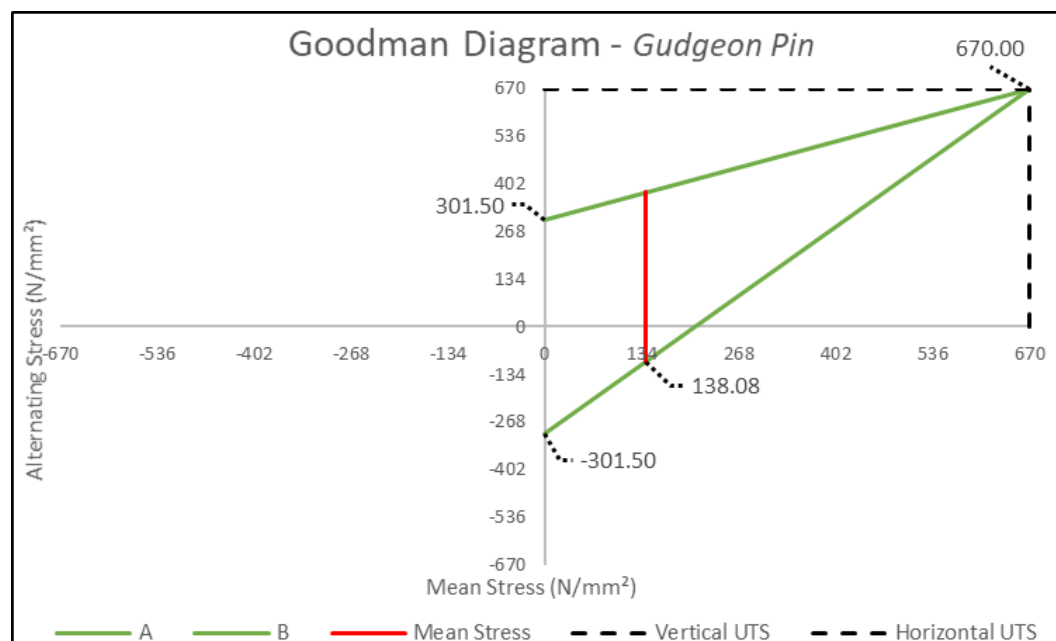
Graph 1 Fatigue Life AISI 4340 Steel (CAE bay, 2018)

Graph 1 shows the fatigue life for AISI 4340 steel, which as previously stated within this study, is the material used for the Gudgeon pin. The fatigue life of a component is defined as the number of stress cycles of a specified character that a component sustains before failure occurs, “*The number of cycles is related to engine speed*” (Xin, 2004). The safety factor of a component directly correlates with its fatigue life, the larger the FSF the more cycles it will be able to withstand before failure.



Graph 2 Gudgeon Pin Alternating Stresses

Graph 2 shows the alternating stress between the minimum and maximum σ values on the gudgeon pin. The Alternating stress can be defined as how much the stress deviates from the mean or “*the Stress on a material created by forces that are changing their direction or magnitude with respect to time*” (Material Grades, 2016). Graph 2 displays this regarding the stresses acting upon the Gudgeon pin, anything between the minimum and maximum force is regarded as the stress range.



Graph 3 Gudgeon Pin Goodman Diagram

From reviewing the Goodman diagram the gudgeon pin has an endurance limit of 301.50N/mm^2 , the endurance limit is equal to the amount of alternating stress the component can withstand. Furthermore, the red line on the diagram displays the mean stress, which is the average stress over a duration, the diagram shows that the mean stress of the gudgeon pin is within tolerance, as it is between the upper and lower endurance limit values.

8 Connecting Rod

8.1 Methodology

The Connecting Rod was split into three sections, Point A, B & C calculating the stresses acting upon each area.

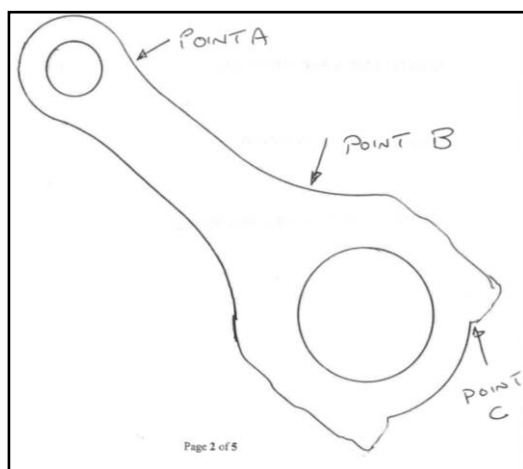


Figure 2 Sketch of Con-Rod



Figure 3 Con-Rod

Figures 4 & 5 show where the cross-sectional area was taken from for part A.



Figure 4 Con-Rod Point A (1)

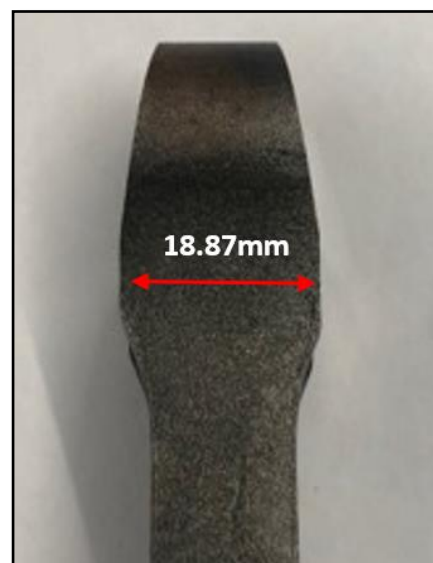


Figure 5 Con-Rod Point A (2)

	Point A	
Thickness	18.8700	mm
width	20.4200	mm
Mass	0.0112	Kg

Table 3 Point A

The mass of the hoop at point A was acquired from the SolidWorks CAD model of the Con-Rod.

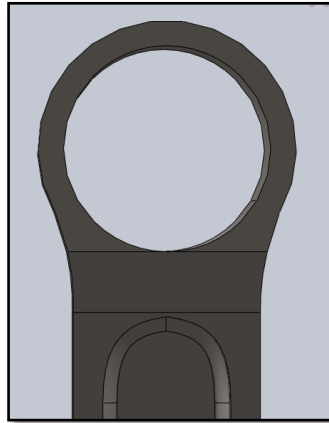


Figure 6 SolidWorks Con-Rod Point A

Then to calculate the Inertia Force acting within the hoop at Point A the mass was multiplied by the total piston acceleration from assignment A at each crank angle as previously stated in section 7.1 of the study. Next, the Net force acting upon Point A was calculated using the following Equation.

Point A Net Force

$$= \text{Gas Force} - (\text{Pin } F \text{ Inertia} + \text{Piston } F \text{ Inertia} + \text{Point A Hoop } F \text{ Inertia})$$

Equation 19 Point A Net Force

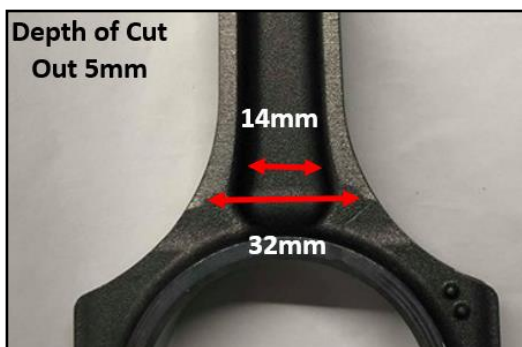


Figure 7 Con-Rod Point B (1)

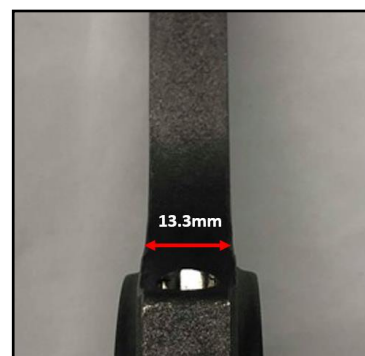


Figure 8 Con-Rod Point B (2)

Figures 7 & 8 show the dimensions used to calculate the CSA for point B. Next, the Reciprocating Mass at Point B was calculated using the following equation;

$$\text{Reciprocating Mass} = \text{Piston Assembly} + \text{Gudgeon Pin} + \left(\frac{1}{3} \times \text{Con Rod}\right)$$

Equation 20 Reciprocating Mass at Point B

	Point B	
Thickness (1)	13.3000	mm
Width (1)	32.0000	mm
Cut Out Depth	5.0000	mm
Width (2)	14.0000	mm
Mass	0.4902	kg

Table 4 Point B

Furthermore, once again following Newtons second Law the Force exerted upon Point B was calculated.

$$F = Ma$$

Equation 21 $F=Ma$, Newtons Second Law

Finally, the required measurements of point C were taken in order to calculate the CSA.

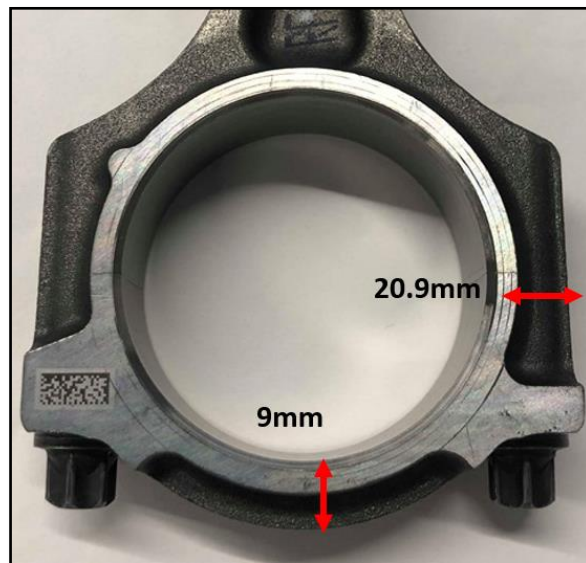


Figure 9 Con-Rod Point C

First Rotational Mass was calculated;

$$\text{Mass}_{\text{Rot}} = \left(\frac{2}{3} \times \text{Con Rod}\right) - \text{Con Rod Cap}$$

Equation 22 Rotational Mass

	Point C	
Width	9.0000	mm
Thickness	20.9000	mm
Mass	0.6573	kg

Table 5 Point C

$$\text{Point C Inertia Force} = (\text{Mass}_{\text{Rec}} + \text{Mass}_{\text{Rot}}) \times \text{Acceleration}$$

The Reciprocating and Rotational mass are both considered when calculating the Inertia force at point C.

$$\text{Point C Force}_{\text{Net}} = \frac{\text{Force}_{\text{Gas}} - \text{Force}_{\text{Inertia}}}{2}$$

Equation 23 Point C Net Force

Equation 23 shows that the Net force is halved this is done as the area is only taken from one side of the cap (Figure 9). Therefore, only half the Net force is taking place within that area. Next with the use of equations 3 & 4 from section 7.1 the tensile and compressive forces at each point were calculated.

	NET Force		
	Point A	Point B	Point C
Tensile	6877.2740	8944.7985	5554.5918
Compressive	-39408.6842	-37527.7223	

Table 6 Con-Rod Forces

Table 6 shows that no compressive forces are considered when looking at point c, this is because the gas force pushes the piston on to the con rod which in turn compresses it onto the crank pin and as the cap is situated below this point it experiences no compressive force.

Con-Rod Specifications		
UTS	750.0000	N/mm ²
Endurance Limit	337.5000	N/mm ²
Second Moment of Area	6526.1490	mm ⁴
Final Endurance Limit C	217.7419	N/mm ²

Table 7 Con-Rod Specifications

Table 7 shows the connecting rod specifications, it is made of grade PMET 4206 F Powder forged steel (GKN Sinter Metals, n.d.). The second moment of area and endurance limit using equations 5 & 11

from section 7.1. As displayed in table 7 the final endurance limit for point C is different to the standard endurance limit which was used for point A & B, this is due to stress concentrations.

A Stress concentration is a location in an object where stress is concentrated and is defined as *“Localised stress considerably higher than average (even in uniformly loaded cross sections of uniform thickness) due to abrupt changes in geometry or localised loading”* (Engineers Edge , 2020). There are various methods that can be implemented with the aim of reducing stress concentrations, one being averting from sharp corners by adding fillets where the bigger the radius the lower the concentration level as the transition between cross sections will be more gradual and less abrupt as shown in figure 10 below.

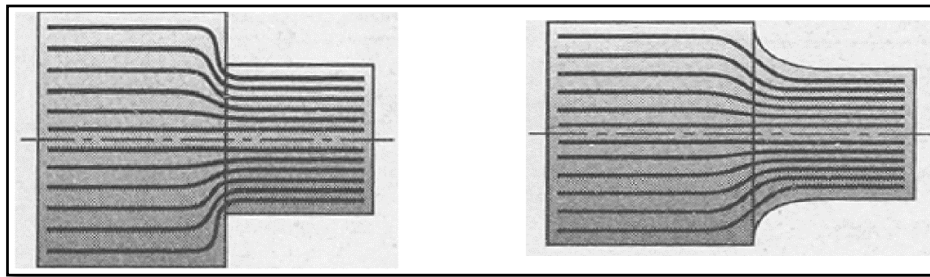


Figure 10 Stress Flow (Krystal Nanan, 2019)

Points A & B of the rod both have shoulder fillets. Therefore, in order to calculate the stress concentration values for both the following graph was used taken from Peterson's stress concentration Factors.

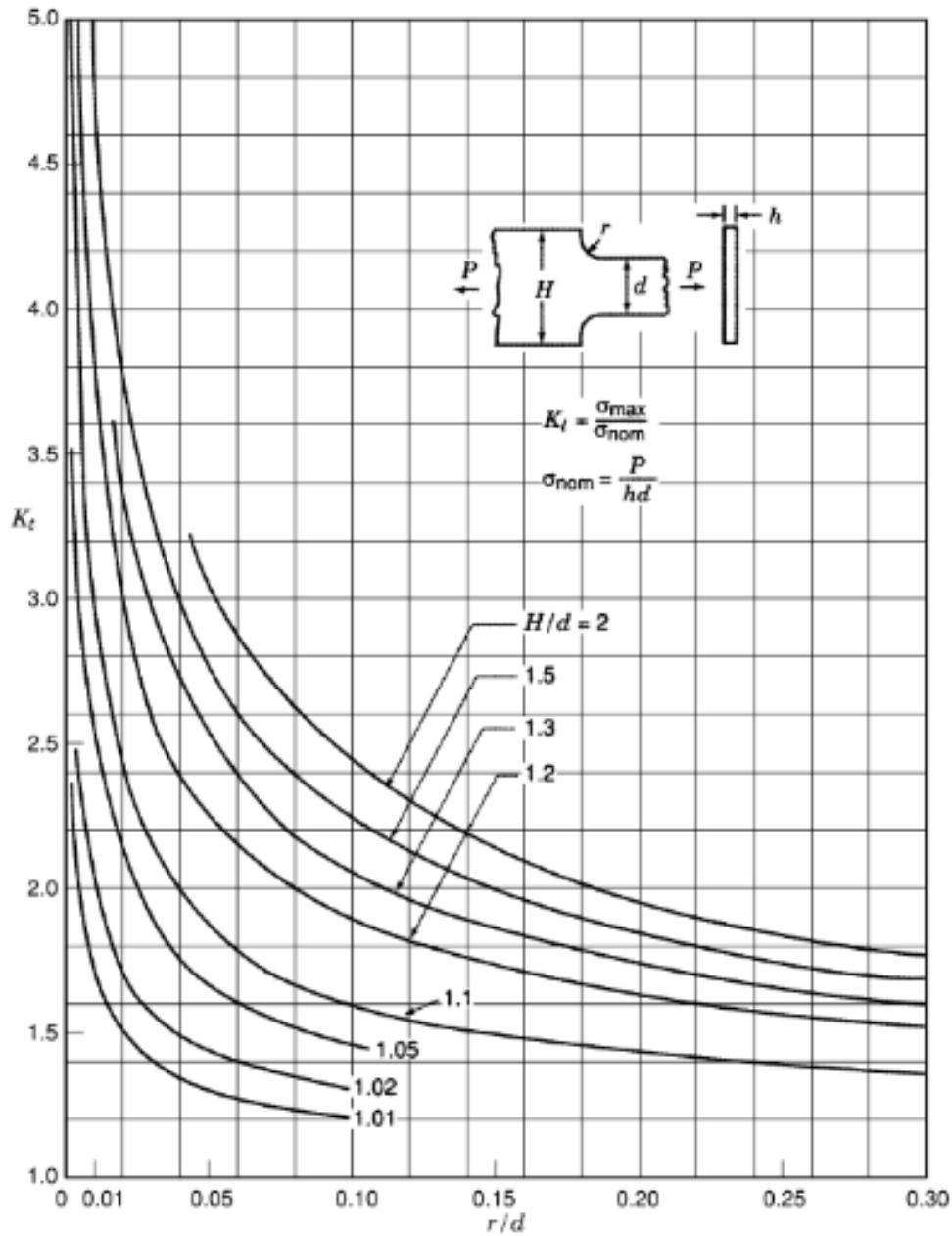


Chart 3.1 Stress concentration factors K_t for a stepped flat tension bar with shoulder fillets (based on data of Frocht 1935; Appl and Koerner 1969; Wilson and White 1973).

Graph 4 Shoulder Fillets (Pilkey)

As shown in the following figures and tables the radiuses at both point A & B are so large that the r/d values cannot be seen on the X-axis on the stress concentration graph (Graph 11) therefore, making a stress concentration value on the points negligible. Hence, why the endurance limit used at point A and B is just 45% of the ultimate tensile strength as previously stated within the report.



Figure 11 Con-Rod Point A Stress Concentration

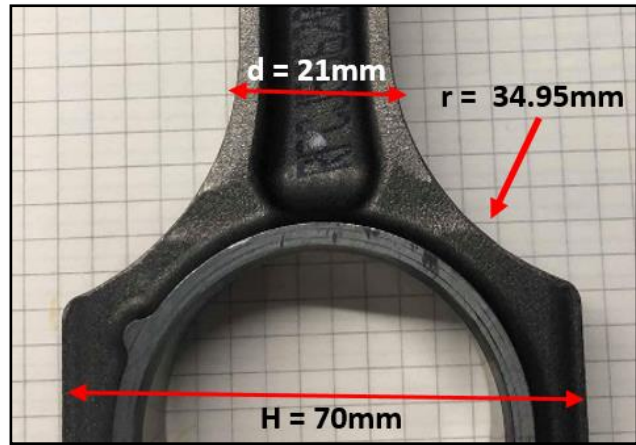


Figure 12 Con-Rod Point B Stress Concentration

r	100.0000	mm
d	19.0000	mm
H	27.4000	mm
r/d	5.2632	
Stress Concentration	N/A	

Table 8 Con-Rod Point A Stress Concentration

r	34.9500	mm
d	21.0000	mm
H	70.0000	mm
r/d	1.6643	
Stress Concentration	N/A	

Table 9 Con-Rod Point B Stress Concentration

Subsequently, at Point C there is a deep hyperbolic notch as opposed to the shoulder fillets found in Point A & B as displayed in figure 15 below.

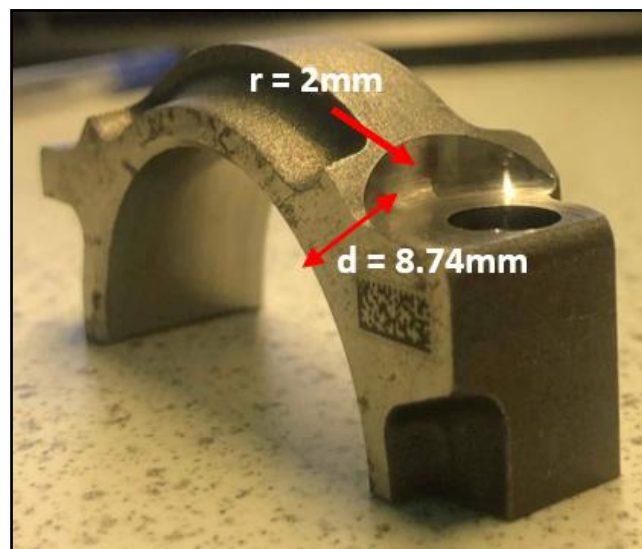
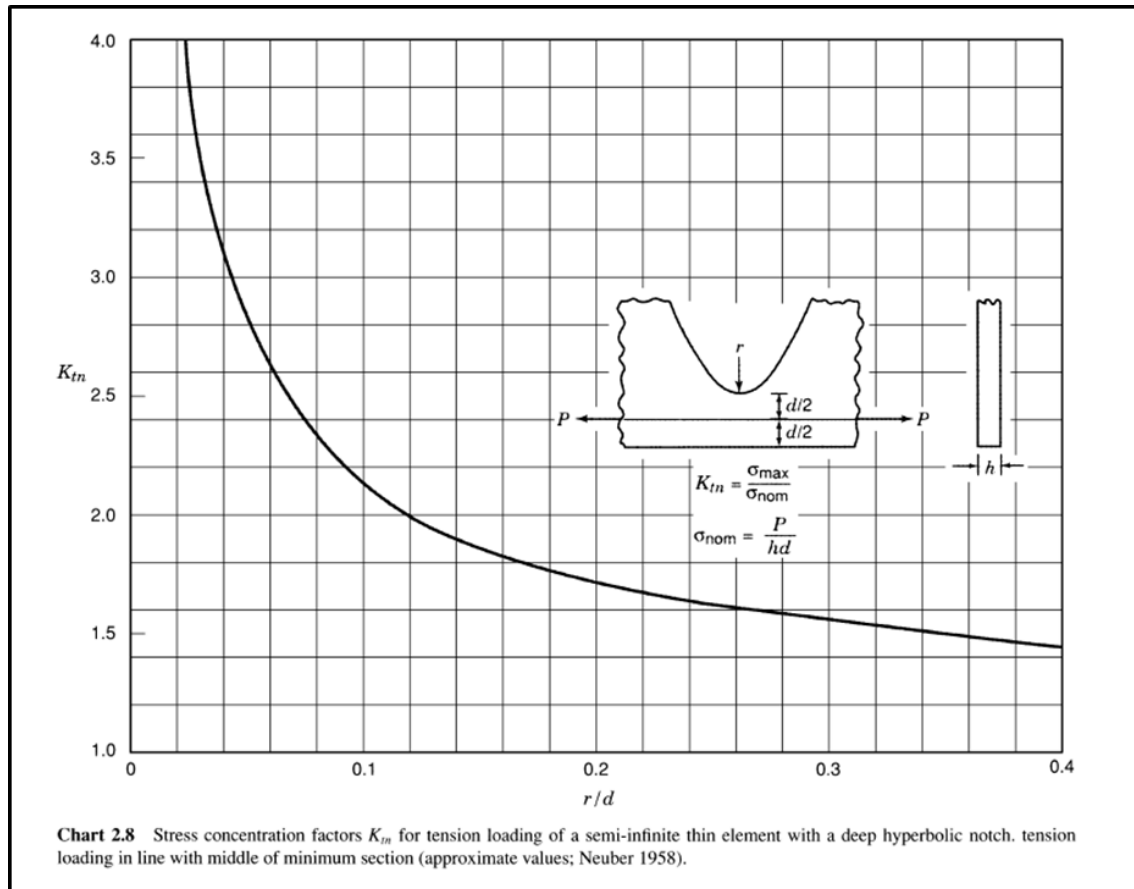


Figure 13 Con-Rod Point C Stress Concentration

Therefore, the following graph also from Petersons stress concentrations book was used.



Graph 5 Deep hyperbolic Notch

r	2.0000	mm
d	5.9400	mm
r/d	0.3367	
Stress Concentration	1.5500	

Table 10 Con-Rod Point C Stress Concentration

A Stress Concentration of 1.55 was determined from graph 5. Therefore, as previously stated in table 7 Point C has a different endurance limit compared to points A & B.

$$\text{Final Endurance Limit} = \frac{\text{Endurance Limit}}{\text{Stress Concentration}}$$

Equation 24 Final Endurance Limit

Next the minimum and maximum nominal stresses were able to be calculated using the following equations;

$$\sigma_{Max} = \frac{Tensile\ Force}{Area}$$

Equation 25 σ_{Max}

$$\sigma_{Min} = \frac{Compressive\ Force}{Area}$$

Equation 26 σ_{Min}

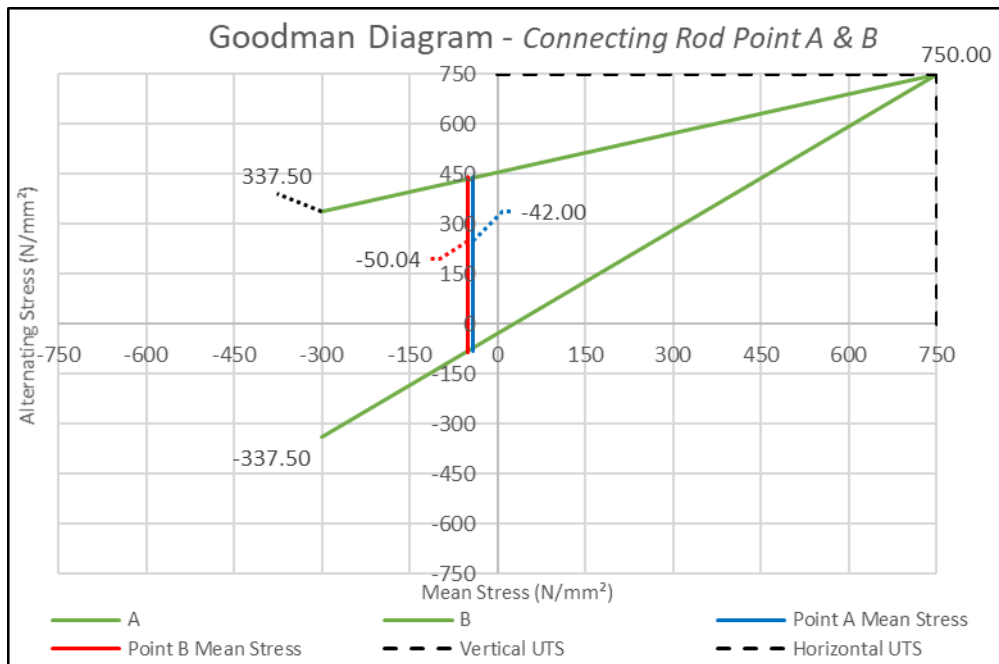
Next with the use of equations 15, 16, 17 & 18 as previously used in section 7.1 the mean nominal and range of nominal stress could be calculated alongside the allowable range and the FSF.

8.2 Results & Analysis

	Point A	Point B	Point C	
CSA	385.3254	285.6000	188.1000	mm ²
Max Nominal Stress	17.8480	31.3193	29.5300	N/mm ²
Min Nominal Stress	-102.2738	-131.3996	N/A	N/mm ²
Mean Nominal Stress	-42.2129	-50.0401	29.5300	N/mm ²
Range Nominal Stress	120.1217	162.7189	29.5300	N/mm ²
Allowable Range	712.9916	720.0361	418.3374	N/mm ²
FSF	5.9356	4.4250	14.1665	

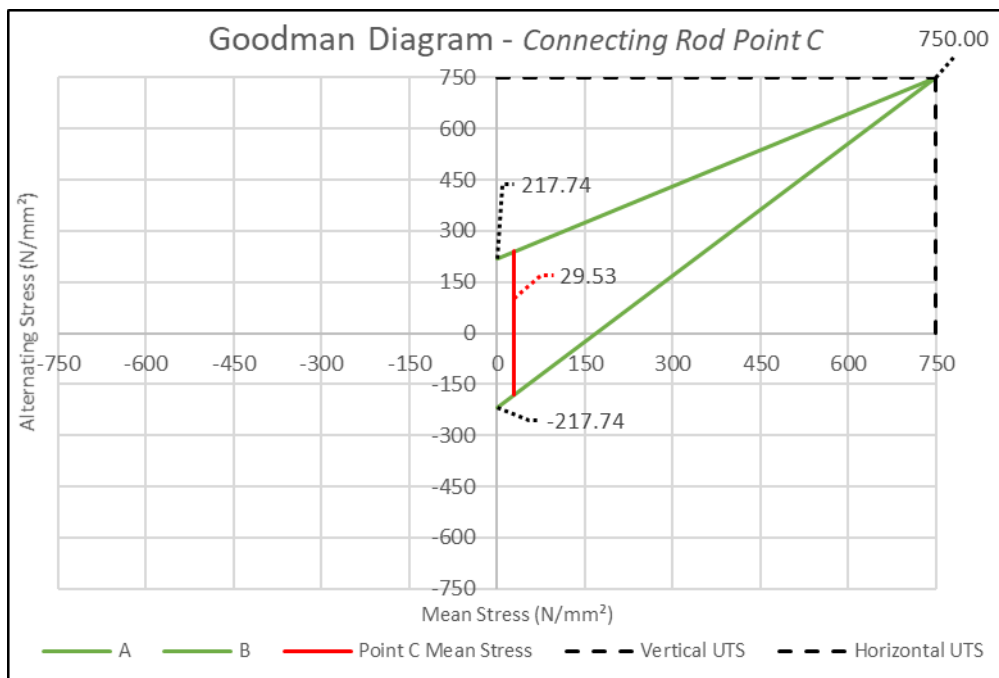
Table 11 Con-Rod Results

Table 9 shows the results for all three points considered when studying the stress being experienced at different sections of the Con-Rod. The safety factor at point C is clearly more substantial than that at point A & B. This may be since any stress seen at the Con-Rod Cap bolt in the region of C has not been taken into consideration, whereas if it was, extra force may be exerted onto the area which in turn may have reduced the FSF. Furthermore when studying Point C, as previously stated in section 5.1 no compressive forces are considered, this in turn will reduce the range of stress, which is a factor within the FSF equation, which may also explain the larger number in comparison to Points A & B. Subsequently, the FSF values at each point are considerably higher than one meaning, theoretically they should last forever providing no more than the intending loads are exerted upon them.



Graph 6 Con-Rod Point A & B - Goodman Diagram

Graph 5 shows the Goodman diagram for points A & B, these were both plotted on the same diagram as they share an endurance limit, as the stress concentration is negligible as previously discussed. It shows that the mean stress is both within tolerance and are relatively close values with a 17% difference. In addition to this, both the mean stress at A and B are negative values this due to the compressive stress values being larger than the tensile stresses, which can be seen in table 7 back in section 5.1, “*In Physics and engineering, tensile stress is considered positive and compressive is considered negative*” (Kean University, 2012).



Graph 7 Goodman Diagram - Connecting Rod Point C

Finally, graph 6 shows the Goodman diagram for Point C. The mean stress once again is within limit even with the smaller final endurance limit due to the stress concentration factor.

9 Connecting Rod Bolts

9.1 Methodology

9.1.1 Standard Bolts



Figure 14 Standard Con-Rod Bolt



Figure 15 Standard Con-Rod Bolt (UTS)

The UTS of the Con-Rod bolts is 1100 N/mm^2 as stated on top of the bolts, please refer to figure 18. Furthermore, research was conducted and assumption was made on the grade of steel of the bolts to be an Alloy-steel Grade 34CrMo4, which is quenched and tempered and the UTS value of the steel lied within an acceptable range as to that of the Con-Rod bolt (Otai Steel, n.d.). Furthermore, quenching and tempering is a strengthening & hardening process, which involves heating the material and then quickly cooling it to set components into place as quickly as possible. (ShapeCUT Steel, 2015). In addition to this a Youngs Modulus value of 210 GPa was used, retrieved from an online source (Ovako, 2016).

9.1.1.1 Bolt & Rod-Cap Stiffness

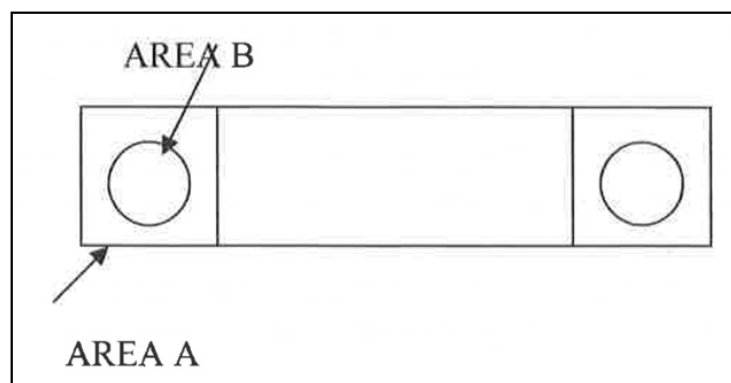


Figure 16 Area A & B

Firstly, the area of the Con-Rod Bolt and the area of the Con-Rod Cap needed to be calculated, also referred to as Area A or Area B as see figure 19. Initially, Measurements of the cap were taken with a Vernier calliper the dimensions of which are shown below.

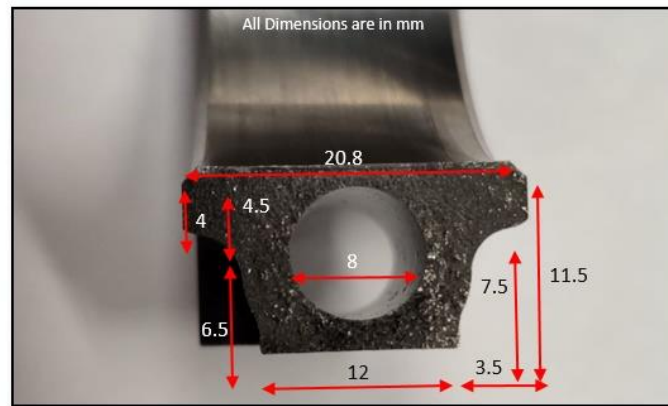


Figure 17 Rod-Cap Dimensions

Using these measurements the area of the Rod-Cap was calculated, the results of which can be found in table 11 below.

A		
Outer Width 1 (mm)	20.8000	mm
Outer Length 1 (mm)	11.5000	mm
Area 1	239.2000	mm ²
Deducted area Width	3.5000	mm
Deducted Area Length	7.5000	mm
Decucted Area	26.2500	mm ²
Area	186.7000	mm ²
Area of Hole	50.2655	mm ²
Total Cap Area	136.4345	mm ²
Deducting Bolt Hole	136.4345	mm ²

Table 12 Area A

The tensile stress area of the bolt (Area B) was sourced from the Bosh Automotive Handbook along with the pitch diameter and nominal thread diameter, as shown in table 14.

Metric fine thread							
Example of designation: M8 x 1 (Nominal thread diameter: 8 mm; pitch: 1 mm)							
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_3	H_1	A_s in mm ²
8	1	7.350	6.773	6.917	0.613	0.541	39.2

Table 13 M8 X 1 (Bosch, 2002)

Next the stiffness ratio could be calculated using equation 28.

$$\text{Stiffness Ratio} = \frac{E_{\text{Cap}} \times A_{\text{Cap}}}{E_{\text{Bolt}} \times A_{\text{Bolt}}}$$

Equation 27 Stiffness Ratio 1.

Approximate Stiffness Ratio
3.3147

Table 14 Approximate Stiffness Ratio



Top to Bottom of Cap	20.3000	mm
Cap to rod Surface	3.5000	mm
Pitch	1.0000	mm
2nd Engaged Thread	2.0000	mm
Bolt head to second engaged thread	25.8000	mm
Length of bolt	37.5000	mm
Length of bolt with head	47.5000	mm
Diameter	8.0000	mm

Table 15 Bolt Measurements

Equation 28 Distance to 2nd Engaged Thread

Measurements were taken of the Con-Rod Bolt to determine L, which is the compressed length of the Cap and Rod and was taken as the length up to the second engaged thread. The Bolt and cap stiffness were then calculated using the following equations.

$$K_B = \frac{E_B \times A_B}{L}$$

Equation 29 Bolt Stiffness

$$K_{\text{Cap}} = \frac{E_{\text{Cap}} \times A_{\text{Cap}}}{L}$$

Equation 30 Cap Stiffness

$$K_{\text{Total}} = K_B + K_{\text{Cap}}$$

Equation 31 Total Stiffness

Bolt Stiffness (K _b)	
319069.7674	N/mm ²
319.0698	kN/mm ²

Table 16 Bolt Stiffness

Cap Stiffness (K _{cap})	
1057631.9189	N/mm ²
1057.6319	kN/mm ²

Table 17 Cap Stiffness

9.1.1.2 Maximum value of the Alternating Load on the Bolt

Firstly, in order to calculate the maximum alternating loads acting upon the bolt, the maximum inertia load on the cap or bolts at TDC was calculated with the use of equation 32, which takes speed into account rather than crank angle.

$$F_{Inertia} = M_{Rec} \times R\omega^2 \times \left(1 + \frac{R}{L}\right) + \left(\frac{2}{3} \times Rod - Mass_{cap}\right) \times R\omega^2$$

Equation 32 Highest Inertia Force on Cap/Bolt

Furthermore, to calculate the Net force, the inertia force was taken away from the gas force at TDC. The extra force in the bolt can then be calculated using the following equation;

$$Extra\ Force = \frac{\left(\frac{Net\ Force}{2}\right)}{1 + \left(\frac{E_{cap} \times A_{cap}}{E_{Bolt} \times A_{Bolt}}\right)}$$

Equation 33 Extra Force in Bolt (1)

Also, Stretch is calculated as;

$$S_x = \frac{\left(\frac{Net\ Force}{2}\right)}{(K_B + K_{cap})}$$

Equation 34 Bolt Stretch

Therefore, Extra Force can be more simply expressed as;

$$Extra\ Force = K_B \times S_x$$

Equation 35 Extra Force (2)

Finally, the Maximum alternating load is then calculated using equation 36.

$$Max\ Alternating\ Load = \frac{Extra\ Force\ in\ Bolt}{A_B}$$

Equation 36 Maximum Alternating Load

9.1.1.3 Required Bolt Tightening Torque

Firstly the Extra Nominal Stress at the root of the thread was calculated using equation 37, where d is the pitch diameter 7.350mm as previously stated in table 14.

$$Stress = \frac{Extra\ Force}{\pi \times \frac{d^2}{4}}$$

Equation 37 Extra Nominal Stress

The Yield point for most high tensile bolts can be expressed as;

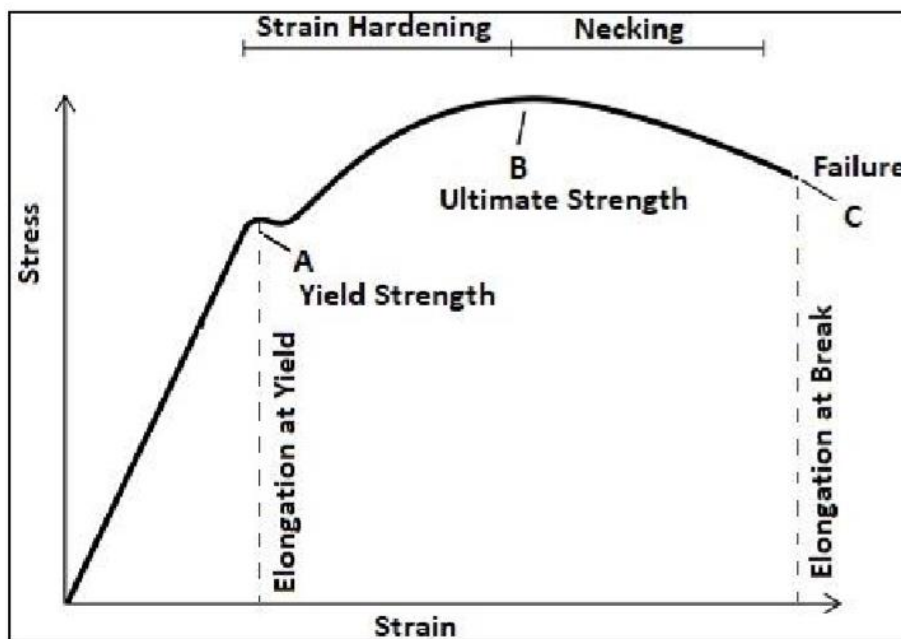
$$Yield\ Point = 0.9 \times UTS$$

Equation 38 Yield Point

Furthermore, as it is required to calculate the bolt tightening torque to 90% yield where axial stress is 60%, therefore;

$$90\% \text{ Yield} = 0.9 \times Yield$$

Equation 39 90% Yield



Graph 8 Typical Stress-Strain Curve (Omnexus , n.d.)

Graph 8 displays a typical stress strain curve and shows where the point of yield is situated. Yield being the amount of stress a component can have exerted upon it before the deformation becomes plastic as opposed to elastic, “the point where the material begins to have permanent deformation”. (Leonghuat, 2016).

Next, the clamping force was calculated.

$$\text{Clamping Force} = \text{Area}_B \times (0.6 \times 90\% \text{ Yield})$$

$$\therefore = \text{Area}_B \times \text{Axial Stress}$$

Equation 40 Clamping Force

In order to calculate the Tightening Torque there are 2 areas of friction to consider as shown in table 19. These values were acquired from Advanced Engine Design lectures with Dr Malcom Williams and can also be found within the Bosh automotive Handbook. (Bosch, 2002).

<i>Coefficient of Friction (μ)</i>	
Effective Thread Friction	0.18
Friction Beneath Head	0.12

Table 18 Coefficient of Friction

the circumferential frictional forces and resisting torque at both the threads and seating face were then calculated

$$\text{Circumferential Frictional Forces} = \text{Clamping Force} \times \mu$$

Equation 41 Circumferential Frictional Force

$$\text{Resisting Torque} = r \times \text{Circumferential Frictional Force}$$

Equation 42 Resisting Torque

Where r is the radius. For the resisting torque at the threads this was half of the average of the nominal thread and the pitch diameter and for the seating face is half of the average of the bolt head diameters. Finally, the total bolt tightening torque required when tightened to 90% yield was a sum of the two added together;

$$\text{Tightening Torque} = \text{Thread Resisting Torque} + \text{Seating Face Resisting Torque}$$

Equation 43 Tightening Torque

Extra Nominal Stress at Root of Thread	31.7462	N/mm ²
Clamping Force on Bolt	20956.3200	N
Circumferential Frictional force at Threads	3772.1376	N
Resisting Torque at Threads	14.4756	N/m
Circumferential Frictional force at Seating Face	2514.7584	N
Resisting Torque at Seating Face	13.2025	N
Total Torque When Tightened	27.6781	N/m

Table 19 Tightening Torque Standard Bolt

9.1.1.4 Stress Concentration Factor

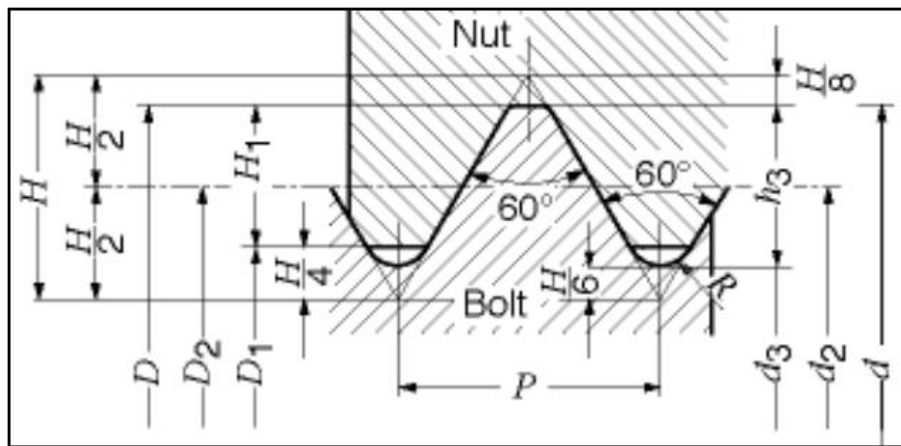


Figure 18 Bolt Thread Stress Concentration (Bosch, 2002)

Figure 21 is from the Bosh Automotive handbook, where the h_3 and d_3 values are also found, please refer to table 14. Subsequently, research was carried to find the H value, this was found to be 0.8660mm (Engineers Edge, n.d.) (Metric Fine Thread ISO DIN 13 (MF), n.d.).

And can be expressed as;

$$H = \frac{\sqrt{3}}{2} \times \text{Pitch}$$

$$\therefore = 0.8660 \times \text{Pitch}$$

$$\therefore = 0.8660 \times 1$$

$$\therefore = 0.8660$$

$$\text{Equation 44 } H$$

Furthermore as displayed in figure 21 r is calculated by; $\frac{H}{6}$. Then, using the graph below the stress concentration for the bolt was calculated.

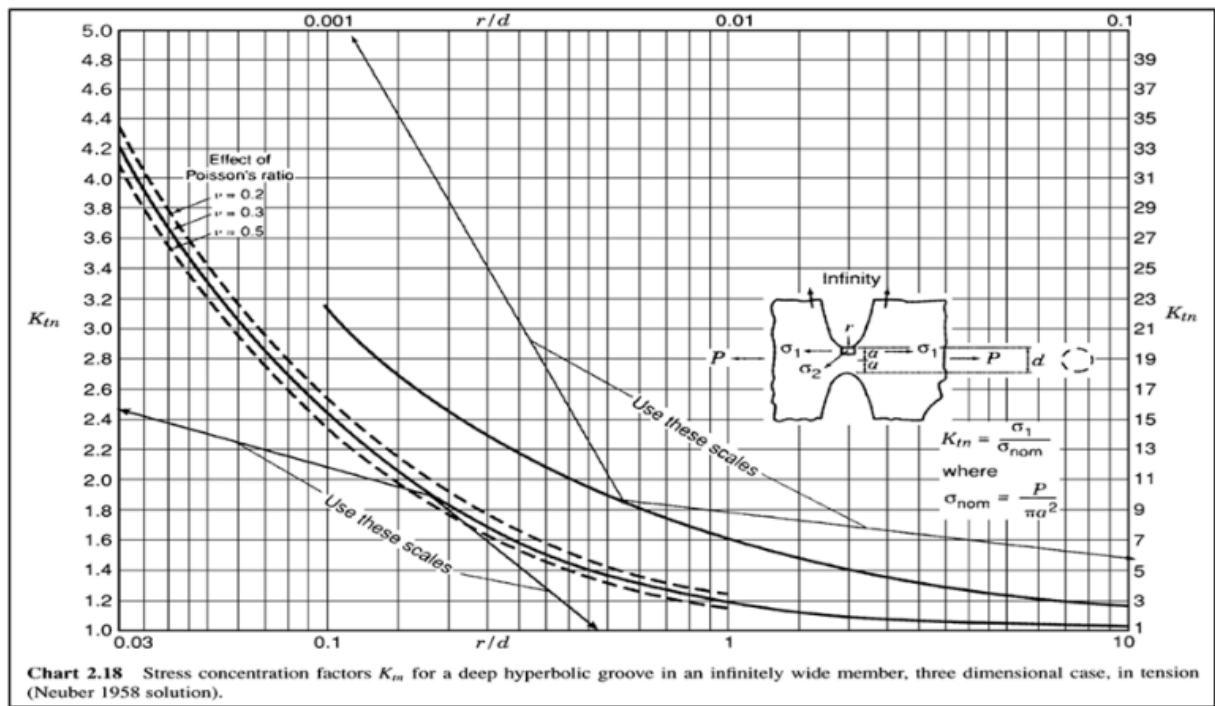


Figure 19 Bolt Stress Concentration

h3	0.6130	mm
d3	6.7730	mm
H	0.8660	mm
r	0.1443	mm
r/d	0.0213	mm
Stress Concentration	4.0000	

Table 20 Bolt Stress Concentration Factor

Due to the stress Concentration factor a final endurance limit of 123.7500N/mm² was calculated. Next the minimum and maximum stress values were calculated using the following two equations.

$$\sigma_{Min} = 90\% \text{ Yield}$$

Equation 45 Minimum Stress

$$\sigma_{Max} = \sigma_{Min} + \text{Alternating Load}$$

Equation 46 Maximum Stress

The allowable range and the FSF calculated using the same equations as used previously within the study.

9.1.2 Upgraded Bolts

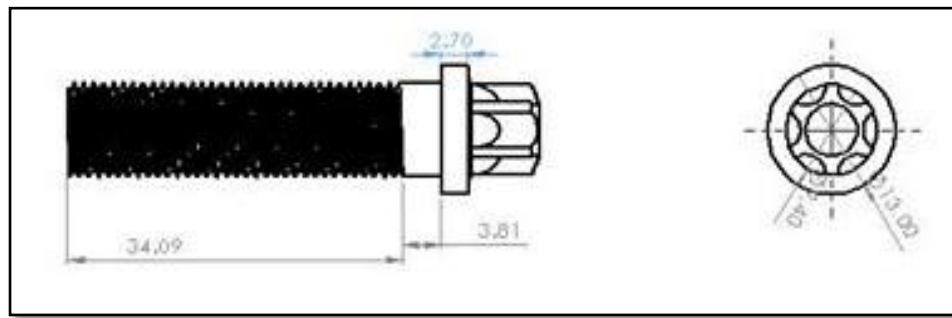


Figure 20 Standard Con-Rod Bolt Technical Drawing

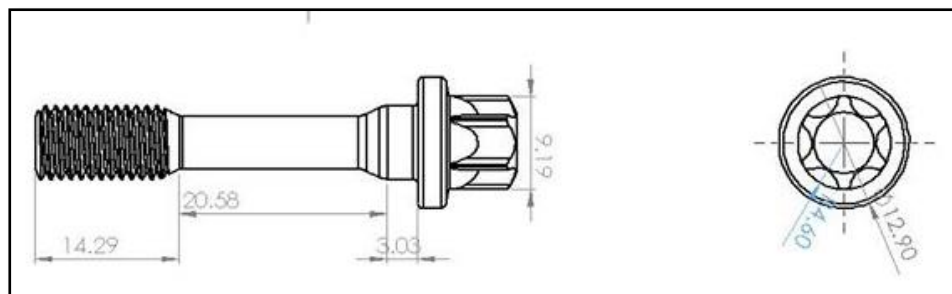


Figure 21 Upgraded Bolt Technical Drawing

The initial paper drawing of both the standard and modified Con-Rod Bolts can be found in Appendix 9.1.

Figure 24 shows the technical drawing generated from the SolidWorks model of the upgraded bolt, where the grade of steel was changed to heat treated 8740 Chrome Moly Steel, which ARP use for their upgraded and high-performance Con-Rod Bolts. This is because they have a Tensile strength of 1,378.9520 N/mm² which is an increase of 22.5056% compared to the steel used for the standard bolts. Furthermore, it is seen as a “good tough steel, with adequate fatigue properties” and is also classified as quench and temper steel. (ARP, 2019). Furthermore, a modulus of elasticity of 207 GPa was used (Puran Singh, 2015) .

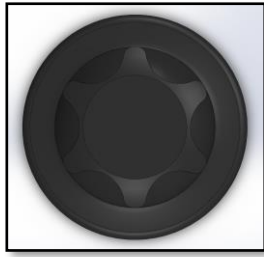


Figure 22 Standard Con-Rod Bolt Head



Figure 24 Standard Con-Rod Bolt Solid Works Model

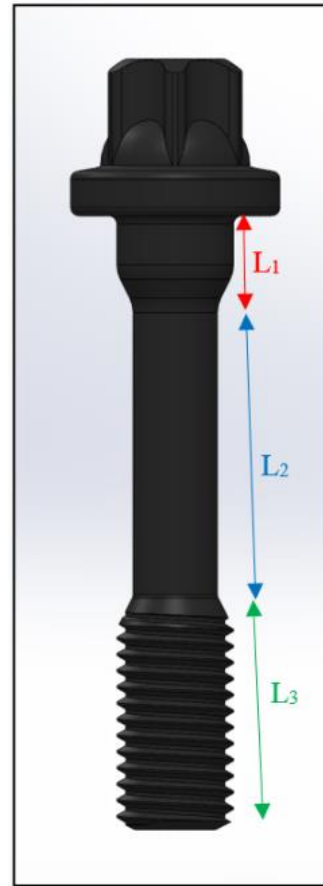


Figure 23 Upgraded Con-Rod Bolt SolidWorks Model

The bolt head, nominal thread and pitch diameter and the overall length of the bolt was kept the same as the standard bolt allowing it to be used with the same Con-Rod without modification of the Rod having to be made. Furthermore, the aim was to reduce bolt stiffness by reducing the diameter of the shank to reduce bolt stiffness in turn reducing extra force in threads, whilst also removing unnecessary material and reducing the mass. The same stress concentration was used at the thread as of that of the standard bolt as the pitch diameter and nominal thread diameter were not altered.

	Section 1	Section 2	Section 3	
Length	3.5000	19.4000	13.0000	mm
Diameter	7.2000	6.0000	7.6750	mm
Area	40.7150	28.2743	39.2000	mm ²
Stiffness	2.41E+06	3.02E+05	6.24E+05	N/mm ²
Stress Concentration	1.5000	1.5000	4.0000	
Final Endurance Limit	413.6856	413.6856	155.132	N/mm ²

Table 21 Modified Bolts Specifications

Moreover, as L_1 , L_2 & L_3 will all have different areas, in order to calculate the overall stiffness of the bolt the following equations were used.

$$K_{1,2,3} = \frac{E \times A_{1,2,3}}{L_{1,2,3}}$$

Equation 47 $K_{1,2,3}$

$$\frac{1}{K_t} = \frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3}$$

Equation 48 $1/K_t$

Total Bolt Stiffness		
1/Kt	5.3320E-06	N/mm ²
Kt	1.8755E+05	N/mm ²
	187.5458	kN/mm ²

Table 22 Total Bolt Stiffness Upgraded Bolt

9.2 Results & Analysis

	Standard	Upgraded	
90% Yield	891.0000	1116.9511	N/mm ²
σ Min	891.0000	1116.9511	N/mm ²
σ Max	924.9830	1139.6793	N/mm ²
Mean Stresses	907.9915	1128.3152	N/mm ²
Range	33.9830	22.7282	N/mm ²
Allowable Range	43.2019	56.3933	N/mm ²
FSF	1.2713	2.4812	

Table 23 Standard & Up-Graded Bolts

Table 21 shows the results for both the standard and upgraded bolts whilst the engine is running at 6,000RPM, the upgraded bolt has a safety factor of more than double that of the standard bolt. The aim was to reduce bolt stiffness by a method of reducing the diameter of the shank to in turn reducing the extra force the threads experience as previously stated in section 6.1.2. Furthermore, the new bolt had a reduction in stiffness of 51.9226% and a decreased extra force in bolt of 47.6333% proving the method successful.

Standard Bolt at 8,000RPM		
σ Min	891.0000	N/mm ²
σ Max	955.1450	N/mm ²
Mean Stresses	923.0725	N/mm ²
Range	64.1450	N/mm ²
Allowable Range	39.8087	N/mm ²
FSF	0.6206	

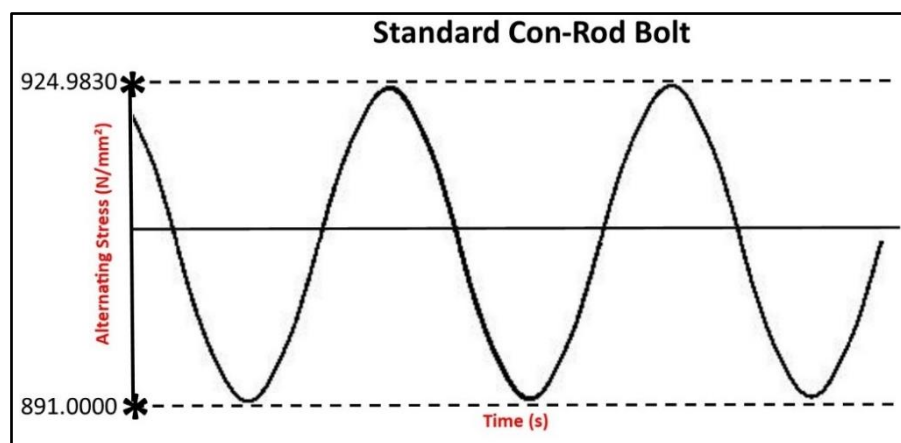
Table 24 Standard Bolt at 8,000RPM

Modifying the engine speed to 8,000RPM shows the FSF of the standard bolts to drop below one, ultimately meaning that they will fail under normal operating conditions of the engine meaning that the

Con-rod bolts needed to be upgraded. Continuing with this, as seen in table 23 below, the FSF of the upgraded Con-rod bolts at 8,000RPM which is complementary of that of the standard con-rod bolts at 6,000RPM with a 0.3388% difference which can be considered trivial.

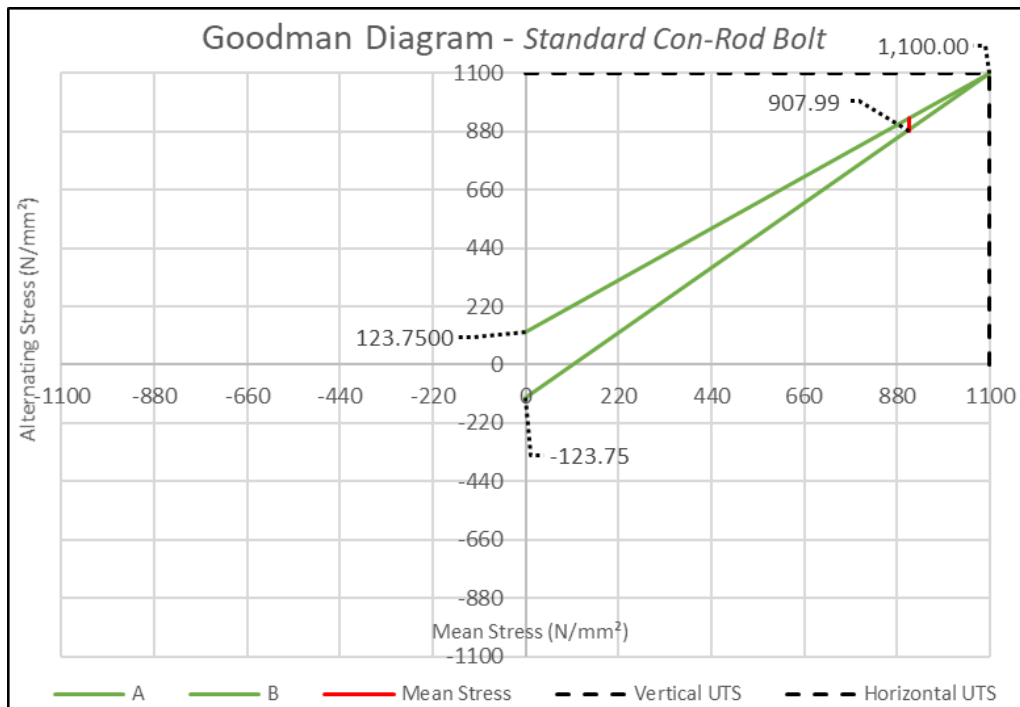
	Standard Bolt at 6,000RPM	Upgraded Bolt at 8,000RPM	
90% Yield	891.0000	1116.9511	N/mm ²
σ Min	891.0000	1116.9511	N/mm ²
σ Max	924.9830	1159.6843	N/mm ²
Mean Stresses	907.9915	1138.3177	N/mm ²
Range	33.9830	42.7332	N/mm ²
Allowable Range	43.2019	54.1427	N/mm ²
FSF	1.2713	1.2670	

Table 25 Standard Bolt at 6,000 RPM and Upgraded Bolt at 8,000 RPM



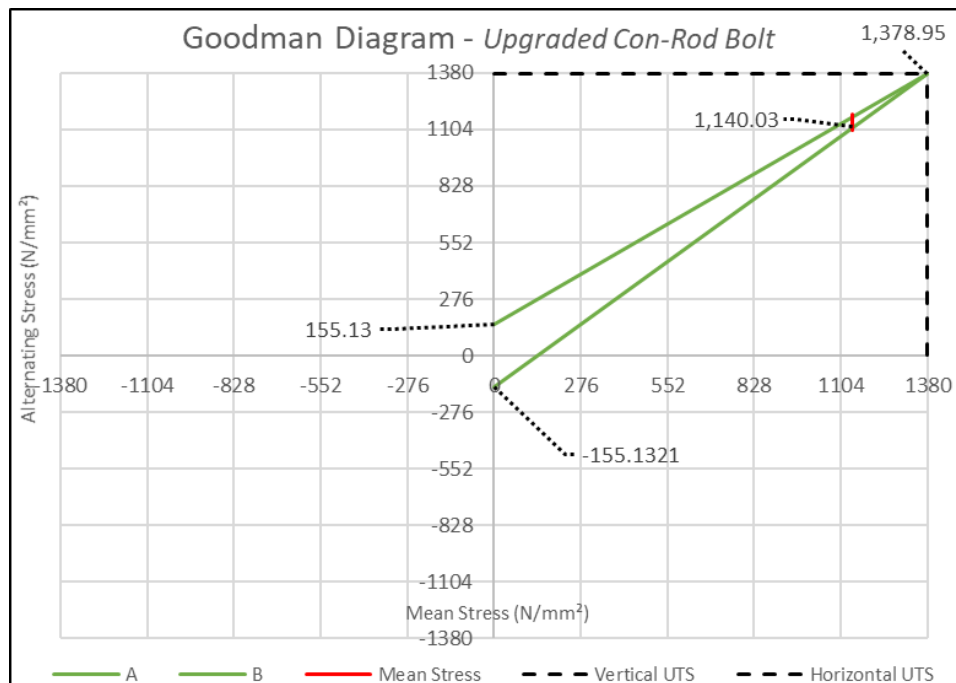
Graph 9 Standard Con-Rod Bolt Alternating Stresses

Graph 7 shows the alternating stress of the standard con-rod bolt with a stress range of 33.9830N/mm² to see the same diagram regarding the alternating stress the upgraded Con-rod bolt experiences please refer to Appendix 13.2.



Graph 10 Goodman Diagram Standard Bolt

Graph 8 shows the Goodman diagram regarding the standard bolt at 6,000RPM whereas Graph 9 shows the Goodman diagram of the upgraded bolt. The mean stress of both is within tolerance of the negative and positive endurance limits.



Graph 11 Goodman Diagram Upgraded Bolt

10 Crankshaft

10.1 Methodology

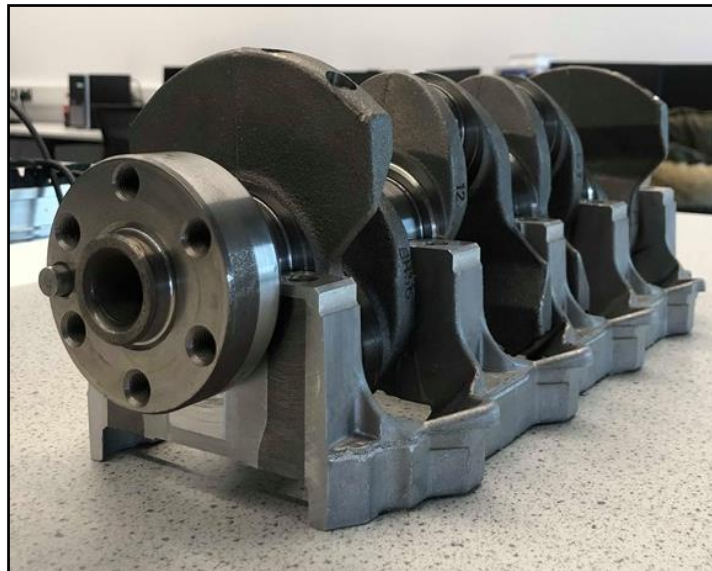


Figure 25 Crank Shaft.

Figure 28 shows the Crankshaft used within the study, which is made from Spherical Graphite Iron, the mechanical properties can be found in table 26 below.

UTS	750.0000	N/mm ²
Endurance Limit	337.5000	N/mm ²
Final endurance Limit @ A	232.7586	N/mm ²
Final endurance Limit @ B	228.0405	N/mm ²

Table 26 Crankshaft Mechanical Properties

Firstly, the required measurements were taken and can be found in table 27 below.

Main Bearing Length	15.0000	mm
Main Bearing Thickness	3.0000	mm
Crankshaft Main Bearing Diameter	48.0000	mm
Crank Shaft Main Bearing Journal Length	18.9000	mm
Horizontal Projected Bearing Area	720.0000	mm
Crankshaft Con-Rod Journal Diameter	44.0000	mm
Effective Width of Critical Beam (k)	46.0000	mm
D	22.3400	mm
Second Moment of Area (I)	42,739.17	mm ⁴
Thickness of the counterweight	20.8000	mm
Width of the Journal	21.0000	mm
A to Main Bearing Centre	10.5000	mm
B to Main Bearing Centre	31.4000	mm

Table 27 Crank Shaft Measurements

Where K, is an average diameter of the Main Bearing and Connecting Rod Journals.

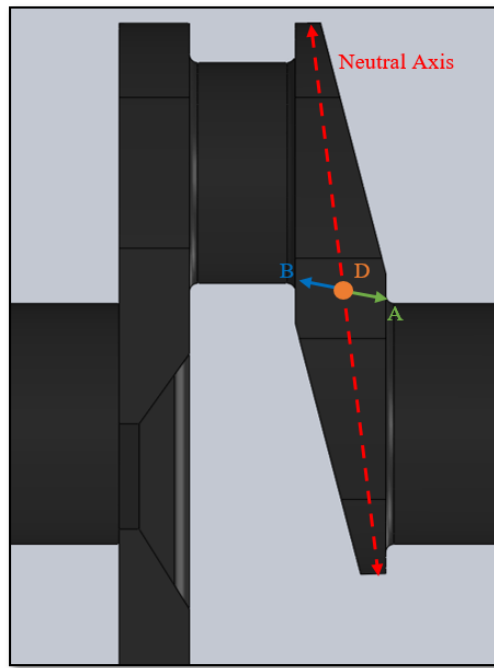
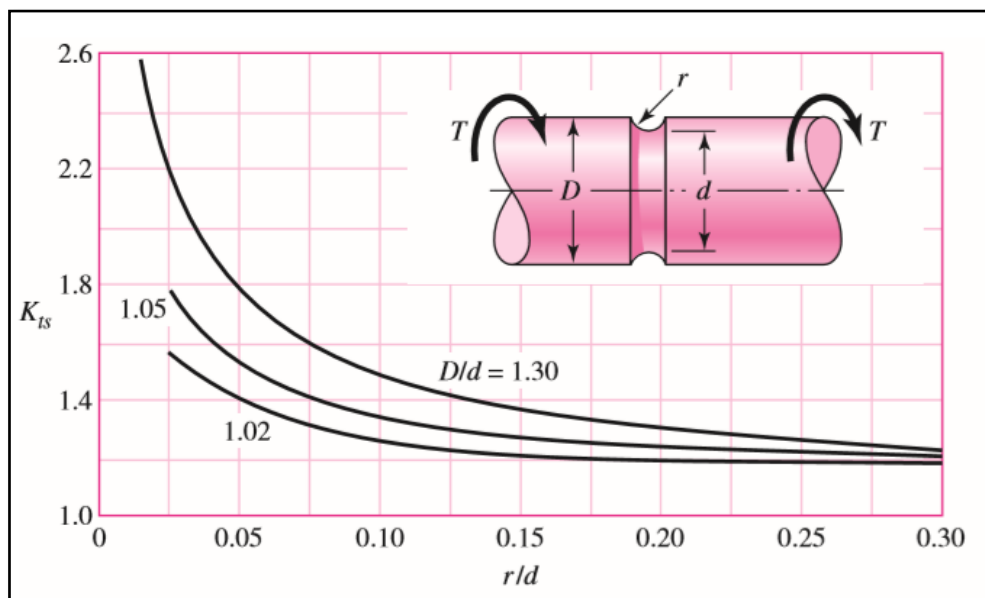


Figure 26 Crankshaft SolidWorks Model

Figure 29 shows the CAD model of the crank shaft and shows D which is the distance from Point A to Point B and the neutral axis which is a 90° line between point A & B. From the SolidWorks Model a fillet radius of 1.7mm was calculated, meaning that the stress concentration at each point will have different values as the diameter of each journal is different. Therefore, meaning the final endurance value of each journal will be different, the following graph was used to determine the stress concentration factors.



Graph 12 Crankshaft Stress Concentration Factor (Pilkey)

Stress Concentrations					
	r	D	d	r/d	Stress Concentration
A	1.7000	48.0000	44.6000	0.0381	1.4500
B	1.7000	44.0000	43.1500	0.0394	1.4800

Table 28 Crank Shaft Stress Concentrations

Next using the same process used in section 7.1 the bending moments and bending stress at Point A & B were both calculated.

	A	B
Bending Moment	210622.3564	629861.1419
Bending Stress	55.0467	164.6159

Table 29 Bending Moment & Stress at Point A & B

From this point onwards, all calculations regarding the crankshaft followed the same processes and methods used throughout the study to be able produce a Goodman's diagram.

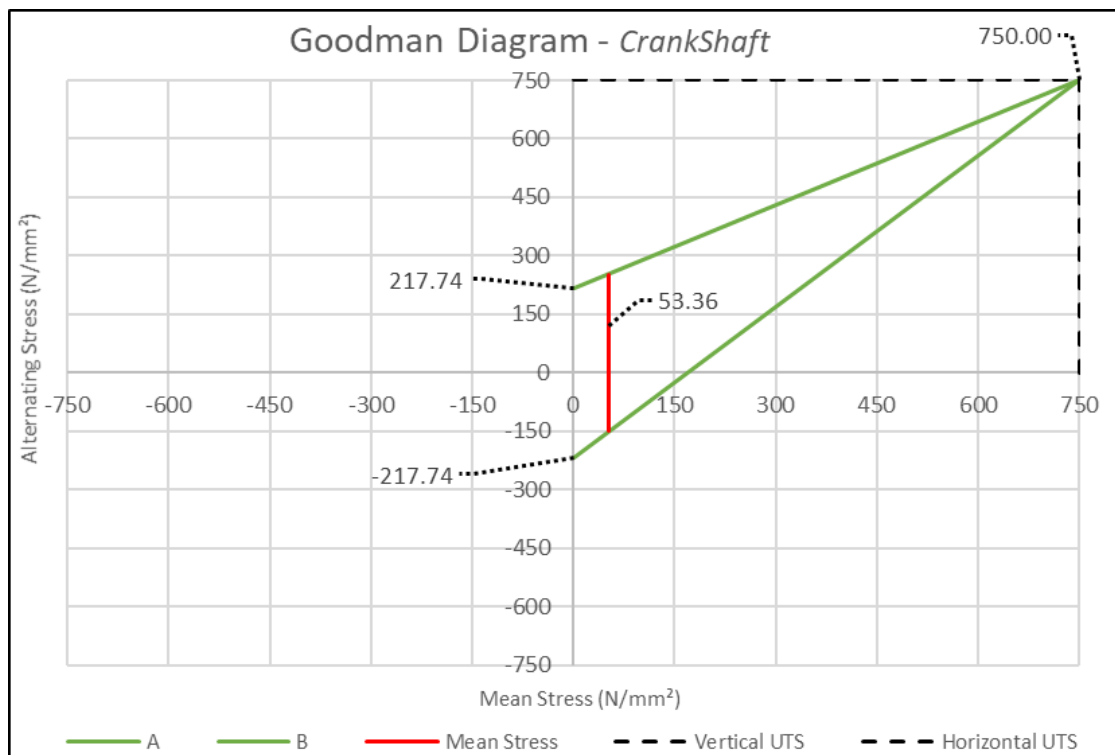
10.2 Results & Analysis

The Results acquired to generate a Goodman's Diagram have only been calculated regarding Point B. This was done because, when the engine is firing point B of the crankshaft goes into tension and is more likely to crack than Point A, which is in compression. Therefore, it endures the largest bending moment and has more stress exerted onto it, in turn resulting in a lower FSF, making it a more applicable point of study than point A.

Shear Stress at B	13.1923	N/mm ²
Max Resultant Stress at B	166.7170	N/mm ²
Min Stress at B	-59.9955	N/mm ²
Range of Stress at B	226.7125	N/mm ²
Mean Stress at B	53.3608	N/mm ²
Allowable Range at B	404.5002	N/mm ²
FSF	1.7842	

Table 30 Results at Point B

Furthermore, Table 25 shows that at point B the crankshaft is showing an FSF of 1.7842 so as previously stated, it should theoretically be able to withstand all the Forces and stresses placed upon it by the standard engines operating conditions.



Graph 13 Goodman Diagram Crankshaft

11 Conclusion

To conclude, this paper has clearly displayed the importance of calculating the stress, safety factor and fatigue life of a critical ICE component during the design process before production to ensure it will be able to withstand the extreme operating conditions the engine will place it under. On the other hand, there were clear limitations within the study, such as assumptions having to be made on the materials used for certain components which questions the validity of the results.

In addition to this, if the experiment was to be carried out again, further work may include a Finite Element Analysis of the components using CAE software, and taking thermal and frictional stresses into account, as they were disregarded in the study. This may have a knock-on effect on the accuracy of the results. Also, other crucial engine components could have been investigated such as the Valve train etc.

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13 Appendix

13.1 Engine Component Masses

Masses		
Conrod	267.4000	0.2674
Conrod big End Cap	103.8000	0.1038
Conrod Bolts	35.1900	0.0352
Conrod Assembly	406.4000	0.4064
Gudgeon Pin	79.0000	0.0790
Piston	257.7000	0.2577
Piston Ring	5.4900	0.0055
Circlip	0.7700	0.0008
	g	kg

Table 31 Engine Component Masses

13.2 Gudgeon Pin

Mass Pin Sticking Out	
0.0261	Kg
26.0564	g

Table 32 Mass of Pin Sticking out

Con-Rod Length	134.0000	mm	0.1340	m
Stroke	81.4000	mm	0.0814	m
Crank Throw (R)	40.7000	mm	0.0407	m
Angular Velocity	628.3185	Rad/s		

Table 33 Crank Throw and Angular Velocity

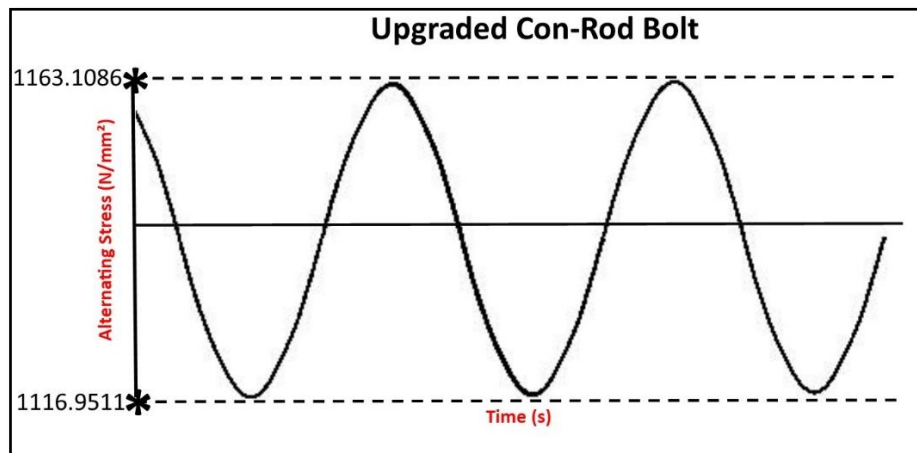
Second Moment of Area	
7660.8228	mm ⁴

Table 34 Second Moment of Area

Point A	Compressive on Piston (Tensile on Pin)	Tensile on Piston (Compressive on Pin)	
Maximum Force	41,622.5483	-4,443.8285	
Bending Moment	158,410.8534	-14,343.3143	N/mm
Bending Stress (σ)	217.1195	-19.6591	N/mm ²
Shear Stress (τ)	108.1540	-11.5471	N/mm ²
Combined Stress	306.4801	-30.3285	N/mm ²
	σ Max	σ Min	
Maximum Inertia Pin Force	545.8295	545.8295	N

Table 35 Gudgeon Pin Stress Results`

13.3 Upgraded Con-Rod Bolt Alternating Stress



Graph 14 Goodman Diagram - Standard Con-Rod Bolt

13.4 Initial Con-Rod Bolt Designs

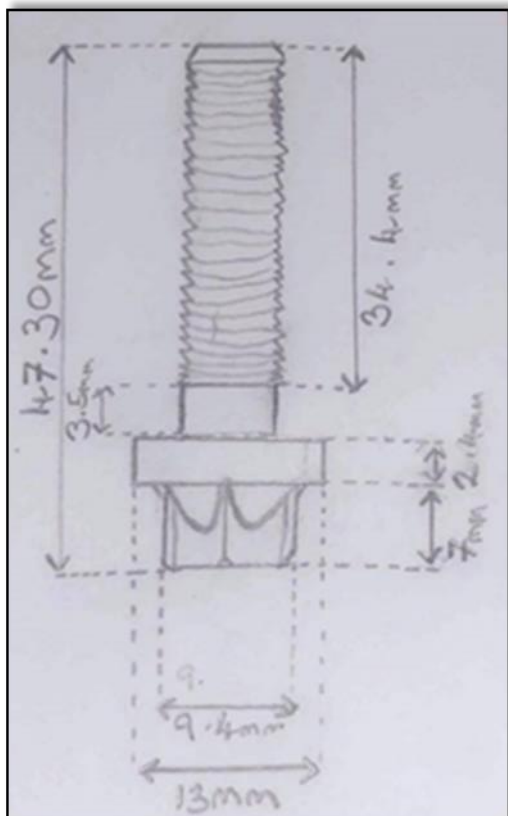


Figure 27 Original Bolt Sketch

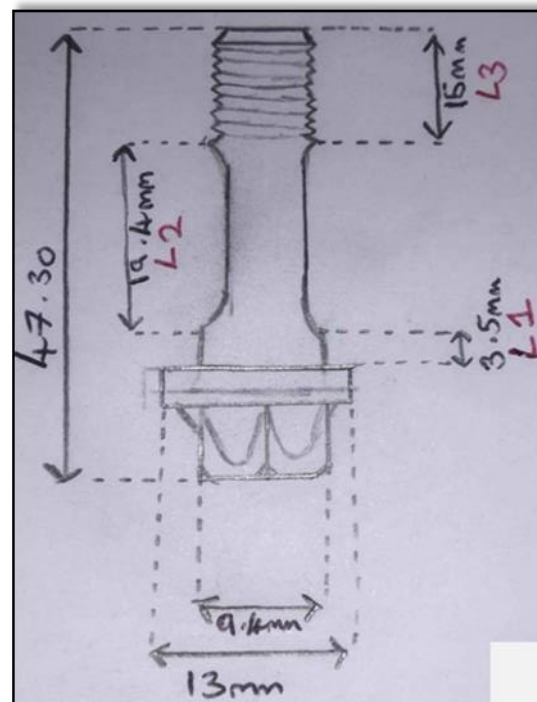


Figure 28 Upgraded Bolt Sketch