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CALCULATION ON PISTON, ROD, CRANKSHAFT ASSEMBLIES

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1. Introduction

The following report investigates the stresses acting on a gudgeon pin, a conrod, a conrod bolt and a crankshaft. Hand calculations are used to calculate these stresses and the fatigue safety factors of each component are also found.

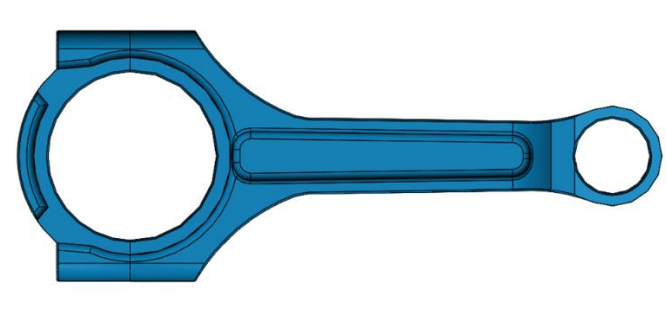
2. CAD models and Material Data

The following components were measured and modelled in Solidworks:

2.1 Conrod



Picture 1 – Conrod



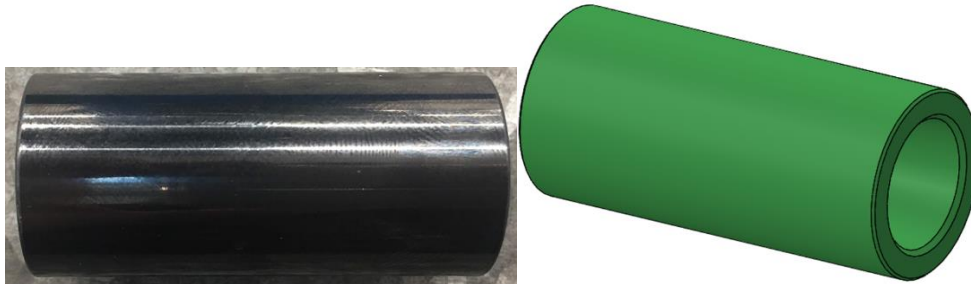
Model 1 – Conrod CAD

The material used for the Conrod is 34CrNiMo6, with the following properties:

Material 1 – Conrod Material 34CrNiMo6 (Ovako, 2020)

Measurement	Value
Youngs Modulus (E)	210 GPa
Poisson Ratio	0.3
UTS	1000 MPa
Density	7800 Kg/m ³
Yield Stress	750MPa
Shear Modulus	80 GPa

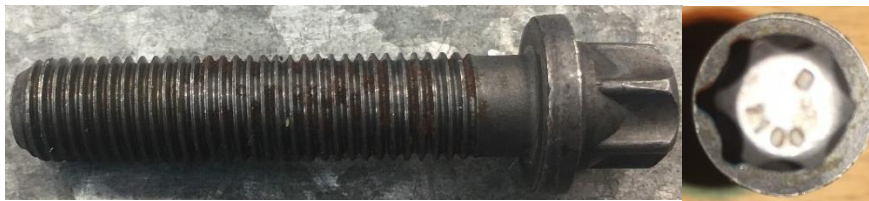
2.2 Gudgeon Pin



Picture 2 – Gudgeon Pin

The Gudgeon pin material is assumed to be the same as the Conrod.

2.3 Conrod Bolt



Picture 3 – Conrod Bolt, on the right the grade mark can be seen.



Model 2 – Conrod Bolt CAD

A paper done by a previous student at UWTSD, Samuel Mousley, conducted research into the the Effect of Lubricants on Fastener Coefficient of Friction and an In-depth Fatigue Analysis of a Connecting Rod Bolted Joint, from this paper the material properties of the bolt can be found. (Mousley, 2018)

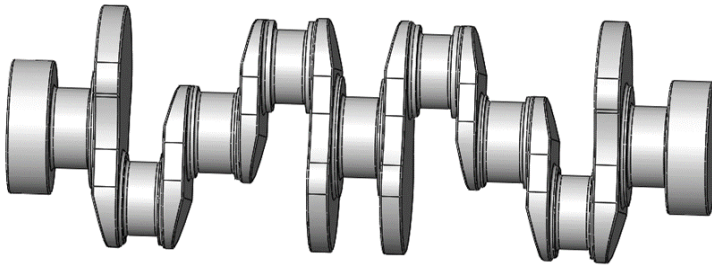
Material 2 – Conrod Bolt Material (Mousley, 2018)

Measurement	Value
Youngs Modulus (E)	207 GPa
UTS	1100 MPa
Density	7800 Kg/m ³
Yield Stress	990 MPa

2.4 Crankshaft



Picture 4 - Crankshaft



Model 3 – Crankshaft CAD

Ductile iron has been used to produce Ford Crankshafts in the past according to a paper by Song Liu, (Liu, 2012). The material chosen is a grade 150/100/7 ADI (Keough, 1998)

Material 3 - grade 150/100/7 ADI (Keough, 1998)

Measurement	Value
Youngs Modulus (E)	155 GPa
UTS	1100 MPa
Density	7800 Kg/m ³
Yield Stress	830MPa

3. Stresses on Gudgeon Pin

Two loads act on the pin, gas pressure load from the piston head and a inertia load from the mass of the piston. These loads alternate on the pin giving a range of stresses. A Goodman diagram is used to understand how the fatigue produced by these stresses affect the pin. (Taylor C. F., 1985)

3.1 Calculations

The following equations where used to calculate the stresses acting on the pin and from the data acquired from these calculations a Goodman diagram can be done:

Table 1 – Tensile Stress on Pin, Equations

Tensile Stress on Pin	
Pin Acceleration	$Pin\ Acc = R\omega^2 \cos \theta + \frac{R}{L} R\omega^2 \cos \theta$
Pin Inertia Force	$Pin\ Inertia\ Force = Pin\ Acc * Pin\ Mass$
Pin Inertia Load Moment	$M_p = Pin\ Inertia\ Force * L2$
Gas Load Moment	$M_F = Net\ Gas\ Force * L1$
Pin Second Moment of Area	$I = \frac{\pi * (D_1^4 - D_2^4)}{64}$
Tensile Bending Stress	$\sigma_B = \frac{M}{I} * y$
Tensile Shearing Stress	$\tau = \frac{(Net\ Gas\ Force - Pin\ Inertia\ Force)}{Pin\ Area}$
Tensile Von Mises Stress	$\sigma_R = \sqrt{\sigma_B^2 + (2 * \tau)^2}$

Table 2 – Compressive Stress on Pin, Equations

Compressive Stress on Pin	
Inertia Load Moment	$M_I = Inertia\ Force * L1$
Pin Inertia Load Moment	$M_p = Pin\ Inertia\ Force * L2$
Total Load Moment	$M = M_I + M_p$
Compressive Bending Stress	$\sigma_B = \frac{M}{I} * y$
Compressive Shearing Stress	$\tau = \frac{(Net\ Gas\ Force + Pin\ Inertia\ Force)}{Pin\ Area}$
Compressive Von Mises Stress	$\sigma_R = \sqrt{\sigma_B^2 + (2 * \tau)^2}$

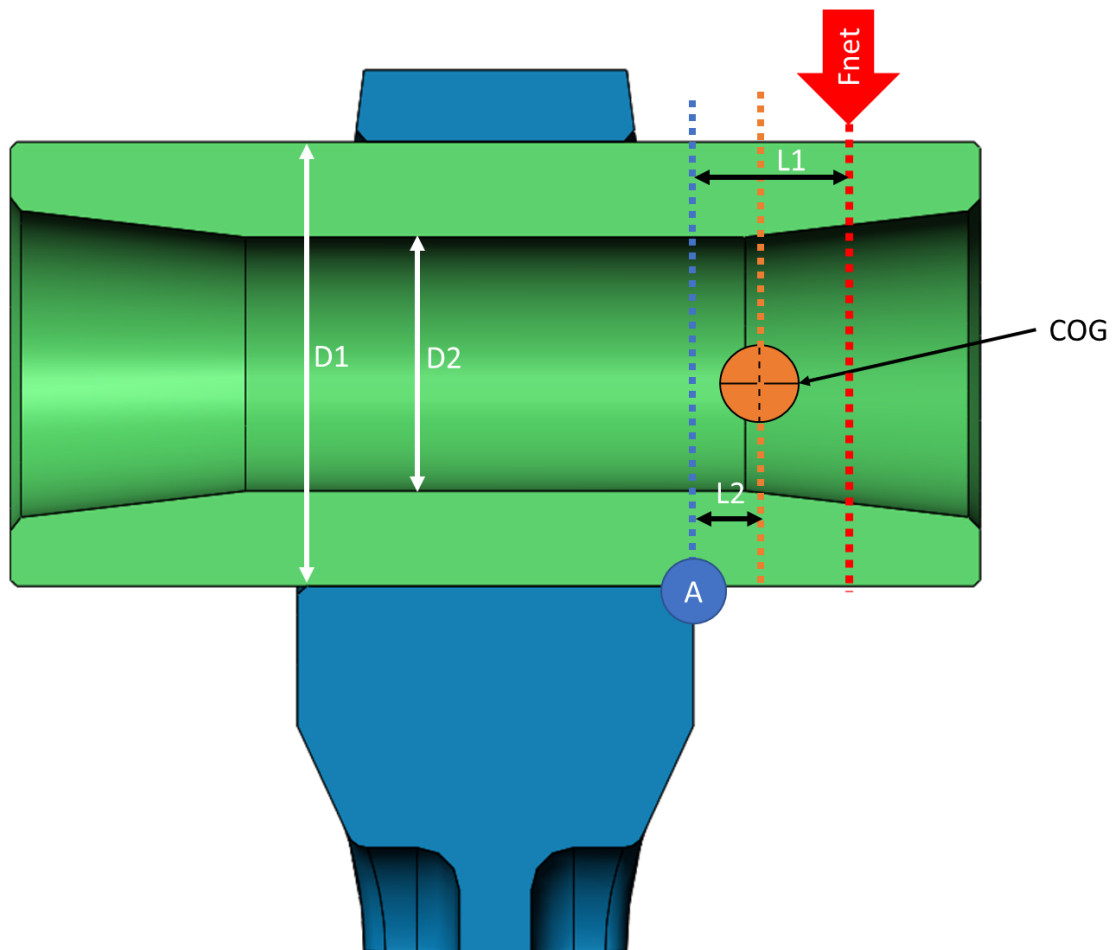


Figure 1 – Cross Section of Gudgeon pin and conrod



Figure 2 – Pin Section Sticking out from conrod

Point A, as seen in Figure 1, is the point where the pin will experience its maximum stress concentrations, due to the location and the nature of the loads acting on the point. The equations from Tables 1 and 2, are used to calculate the stresses happening at that point A.

- The Net Gas Force is taken at 5700 RPM from a previous report done by the author, this is because it is at its highest at that RPM level.
- The Inertia Force is taken at 6000 RPM from a previous report done by the author, this is because it is at its highest at that RPM level.
- The pin inertia moment is calculated by multiplying the “Pin Inertia Force”, by the distance “L2” from the centre of gravity, as seen in Figure 1, which is the Centre of Gravity of the mass of the pin sticking out from the conrod, Figure 2, to point A.
- The gas load moment is calculated by multiplying the “net gas force” by the distance “L1” from the Net Gas force, red arrow in Figure 1, to point A.
- The forces under tensile loading are acting in opposite directions, therefore they are subtracted from each other.
- The forces under compression are acting in the same direction, so they are added together.
- The second moment of Area is calculated with the inner and outer diameters of the pin, as seen in Figure 1
- “y” is the distance from the centre axis of the pin.
- The Von Mises Stress, combines both stresses together and gives the user a resultant stress. Furthermore, the compressive and tensile resultant stresses are used in a Goodman Diagram.

Table 3 - Measurements

Name	Value
D1	21mm
D2	12mm
L1	8.298mm
L2	6.43mm

Table 3 shows the values for the measurements in Figure 1.

Table 4 -Calculated Tensile Stresses on Pin

Calculated Tensile Stresses on Pin	
Tensile Bending Stress	204.86 MPa
Tensile Shear Stress	85.543 MPa
Tensile Von Mises Stress	266.9 Mpa

Table 5 – Calculated Compressive Stresses on Pin

Calculated Compressive Stresses on Pin	
Compressive Bending Stress	34.611 Mpa
Compressive Shear Stress	14.995 Mpa
Compressive Von Mises Stress	45.797 Mpa

Table 6 – Fatigue Safety Factor Equations

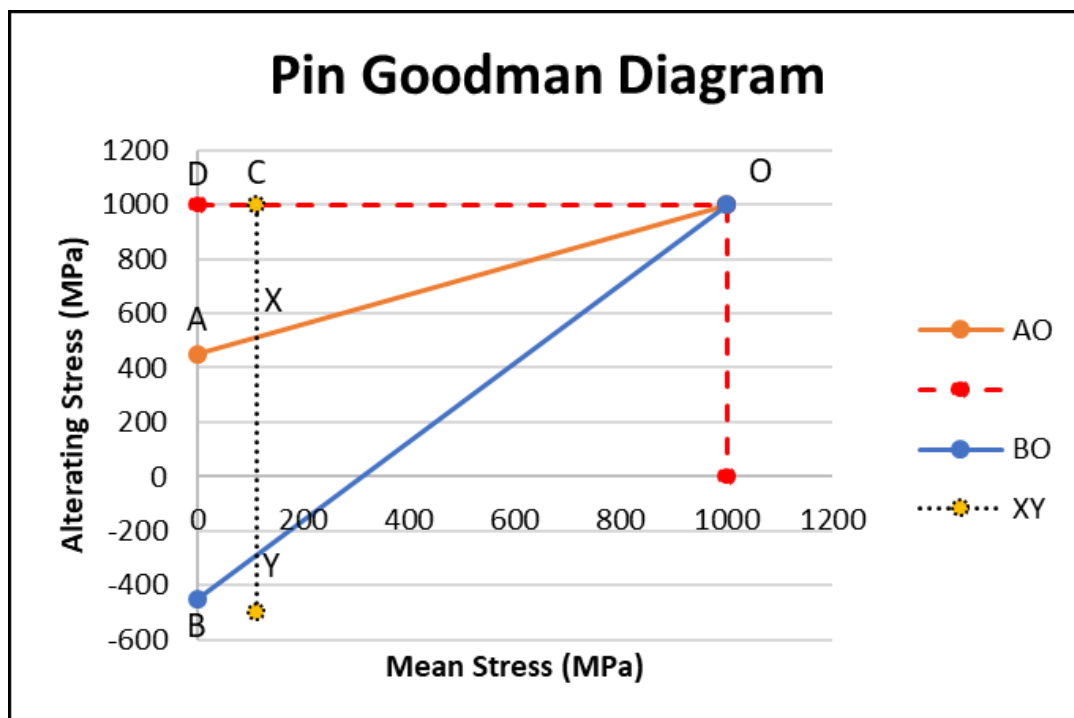
Fatigue Safety Factor of Pin	
Mean Stress	$Mean\ Stress = \frac{(Max\ Stress + Min\ Stress)}{2}$
Max Stress	$Max\ Stress = \sigma_{max}$
Min Stress	$Min\ Stress = \sigma_{min}$
Range of Stress	$Range\ of\ Stress = \sigma_{max} - \sigma_{min}$
Allowable Stress Range	$XY = AB * \frac{OC}{OD}$
Allowable Stress	$XY = AB * \frac{(UTS - Mean)}{UTS}$
Fatigue Safety Factor	$FSF = \frac{Allowable\ Stress}{Range\ of\ Stress}$

Analysing Table 4:

- The Max Stress is the Tensile von Mises Stress from Table 1.
- The Min Stress is the Compressive von Mises Stress from Table 2.
- The mean stress is used within the Goodman diagram to find out the range of allowable stress.

3.2 Calculated data for Gudgeon Pin

Using the equations and data from Section 3.1 it is possible to produce the Goodman Diagram.



Graph 1 – Pin Goodman Diagram

Table 7 – Calculated Fatigue Safety Factor of Pin

Calculated Fatigue Safety Factor of Pin	
Name	Value
Mean Stress	110.544 MPa
Range of Stress	312.702 Mpa
Fatigue Safety Factor	2.559

The Safety Factor of 2.559, shown in table 7, is calculated using the Goodman Diagram, Graph 1, theoretically this tells the user that the pin can be made lighter. One possibility to why the pin wasn't made lighter is, the dimensions of the pin are set in that manner so when in use the displacement of the pin is correct, if the geometry is changed for saving weight this might have a knock on effect with other components of the engine.

4. Stresses acting on conrod

There are three stress points to be investigated on the conrod, Point A, Point B and Point C, as seen in Figure 3.

Points A and B, experience tensile and compressive stresses. On the other hand, Point C only experiences tensile stresses due to it being located under the crank. At these three points, due to their location being on sharp edges stress concentrations will happen. Furthermore, this means additional calculations will have to be done to calculate these stress concentrations.

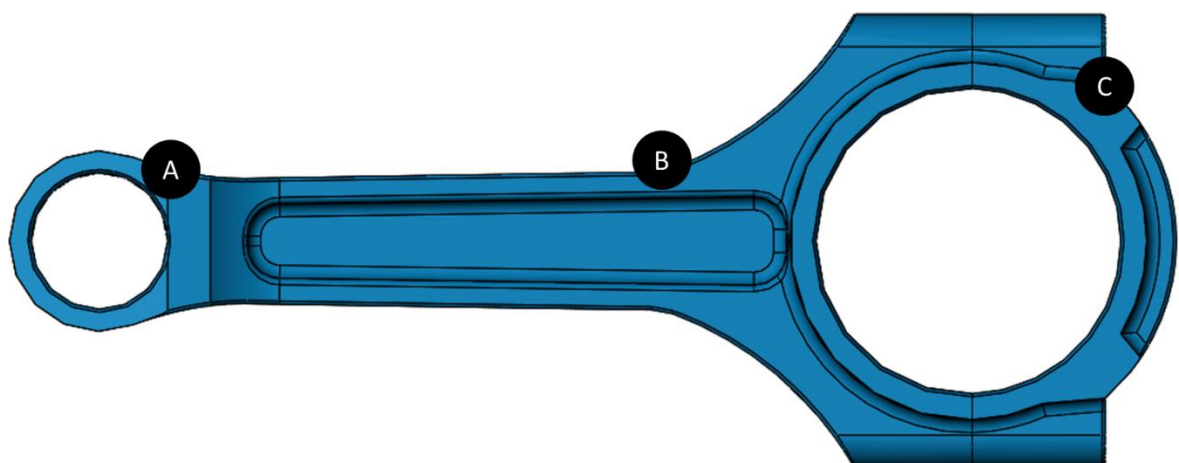


Figure 3 – Stress points acting on Conrod

4.1 Stress concentration Factor used

The Stress concentration is due to a shoulder fillet, this is the most frequently found while machining.
 (Pilkey & Pilkey, 2008)

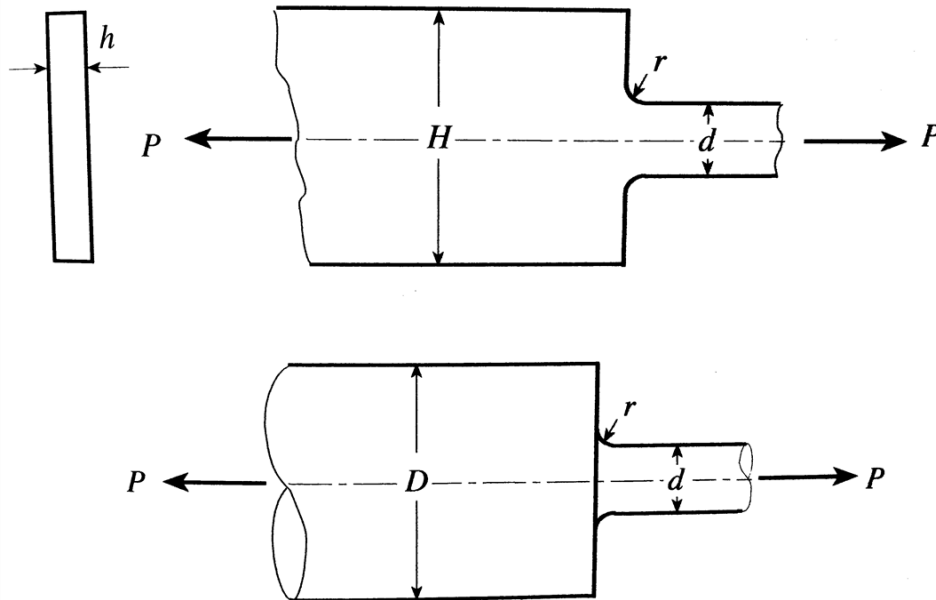


Figure 4 – Fillets in a thin element (top) and a circular bar (bottom)

Table 8 – Stress concentration factor equations

Stress Concentration Factor Equations	
Name	Equation
Stress Concentration Factor	$K_t = \frac{\sigma_{max}}{\sigma_{nom}}$
Nominal Stress	$\sigma_{nom} = \frac{4 * P}{\pi * d^2}$
Stress Concentration Factor (2)	$K_t = C_1 + C_2 * \left(\frac{2t}{D}\right) + C_3 * \left(\frac{2t}{D}\right)^2 + C_4 * \left(\frac{2t}{D}\right)^3$

The equations used are based on Data from (Pilkey & Pilkey, 2008), as seen in Figure 5.

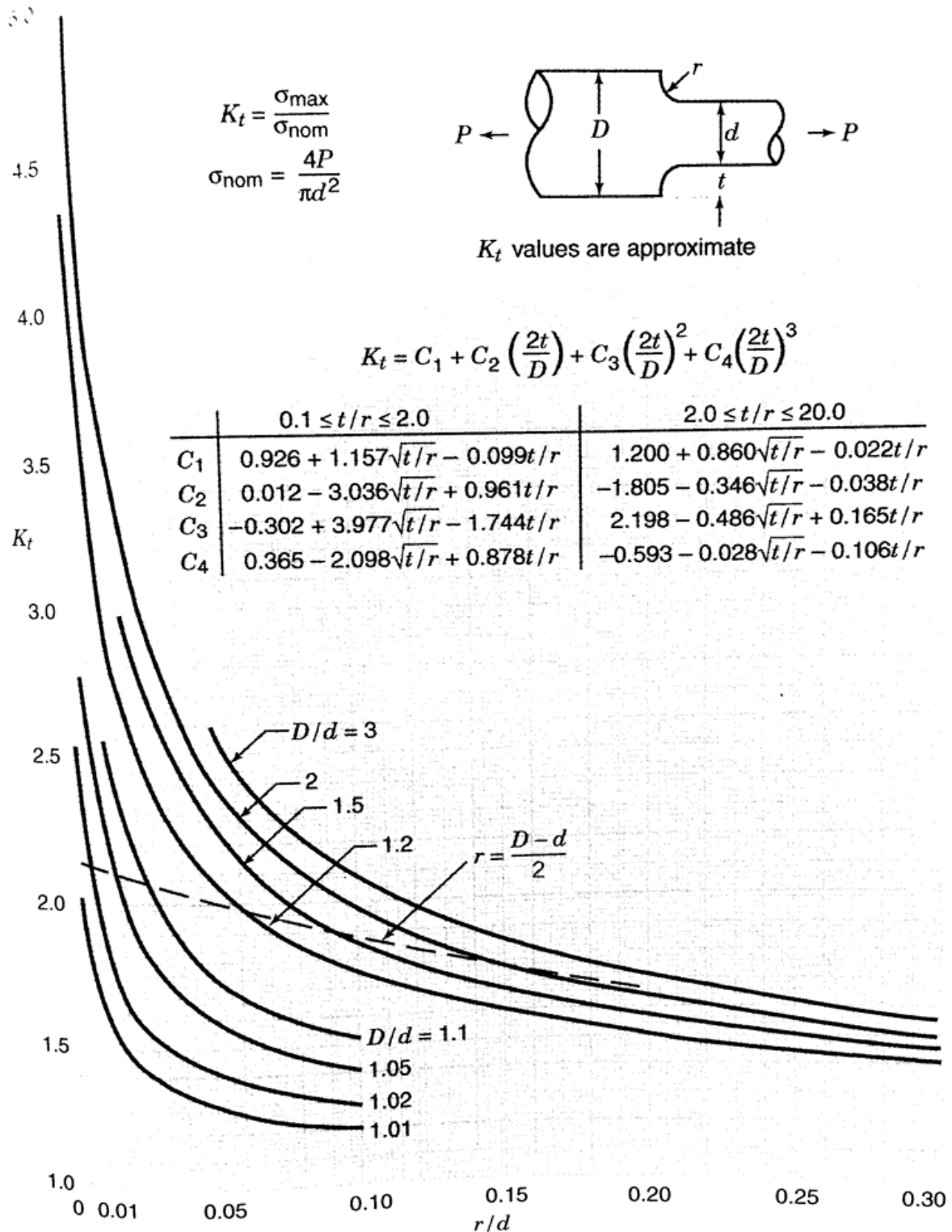


Figure 5 – Stress Concentration Factor for a stepped tension bar of circular cross section with shoulder fillet (Pilkey & Pilkey, 2008)

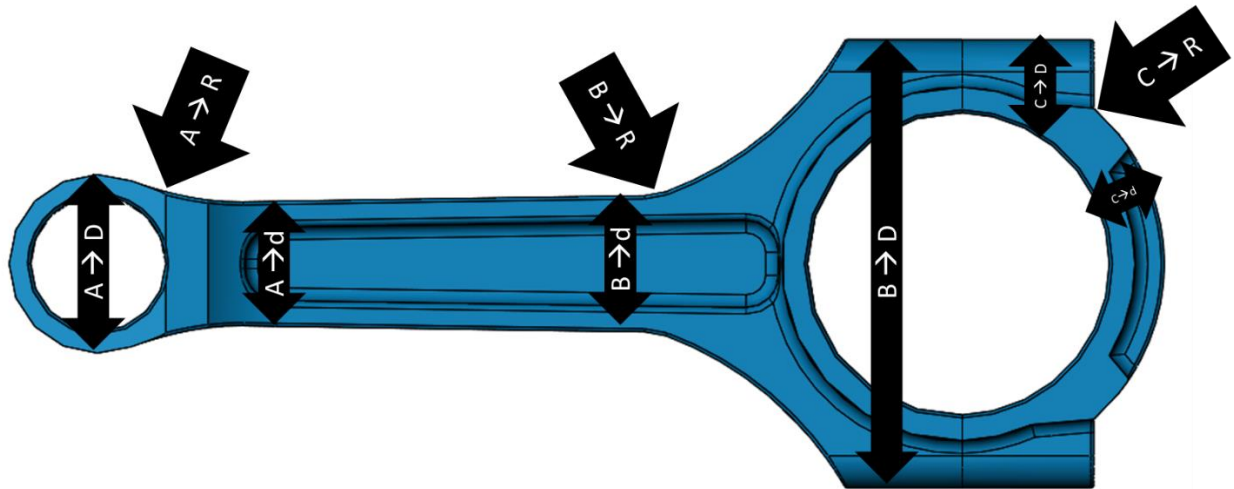


Figure 6 – Stress concentration factor measurements for equation

Figure 6 showcases the measurements needed for using the equation in Table 8 for calculating the Stress Concentration Factor, additionally data from Figure 5 is also used.

Table 9 – Measurements for Stress Concentration Factor Conrod

Measurements for Stress Concentration Factor Conrod	
Measurement	Value
A → R	40 mm
A → D	27.5 mm
A → d	19.2 mm
B → R	54 mm
B → D	69.6 mm
B → d	21.05 mm
C → R	0.5 mm
C → D	11.58 mm
C → d	9.34 mm

Table 10 – Calculated Stress Concentration Factors for Conrod

Calculated Stress Concentration Factors	
Name	Value
Point A, K_t	1.0537
Point B, K_t	0.5728
Point C, K_t	2.1526

Combining the equations and data from Table 8 and 9, as well as the data from Figure 5, the stress concentration factors can be calculated, as seen in Table 10. The reasoning for Point C to have the

largest stress concentration factor is due to its smaller Radius of 0.5mm, while the lowest concentration factor at Point B, is due to it having the largest radius of 54mm.

4.3 Stresses acting on Point A and Point B

The equations in the following table were used for calculating the stresses at Point A and B:

Table 11 – Equations for calculating Stresses at Point A and Point B

Equations for Stresses on Point A and Point B	
Name	Equation
Tensile Stress	$\sigma_T = \frac{\text{Net Inertia Force} * \text{Inertia Force (A or B)}}{\text{Area (A or B)}}$
Compressive Stress	$\sigma_c = \frac{\text{Net Gas Force}}{\text{Area (A or B)}}$
Fatigue Safety Factor	$FSF = \frac{\text{Allowable Stress}}{\text{Range of Stress}}$

- The Net inertia force used is the same one as used in the gudgeon pin study at 6000RPM
- The Net Gas Force is the same one as used in the gudgeon pin study at 5700RPM.
- Areas A of 141 mm^2 and B of 146.72 mm^2 are the cross-sections of the conrod, as seen in Figure 7.
- The Mass of each section is calculated using the volumes in Figure 8, multiplied by the density of the material.

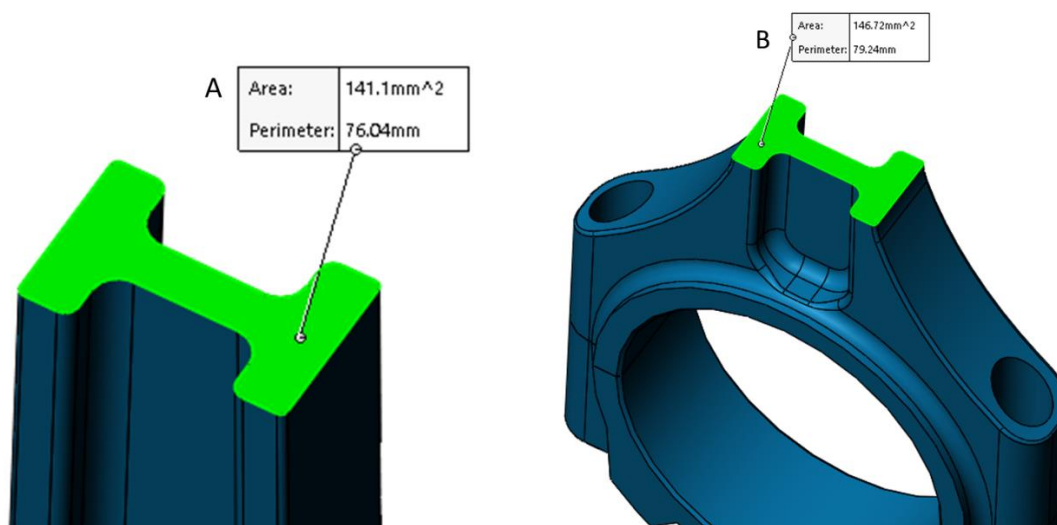


Figure 7 – Area Point A (Left) and Area Point B (Right)

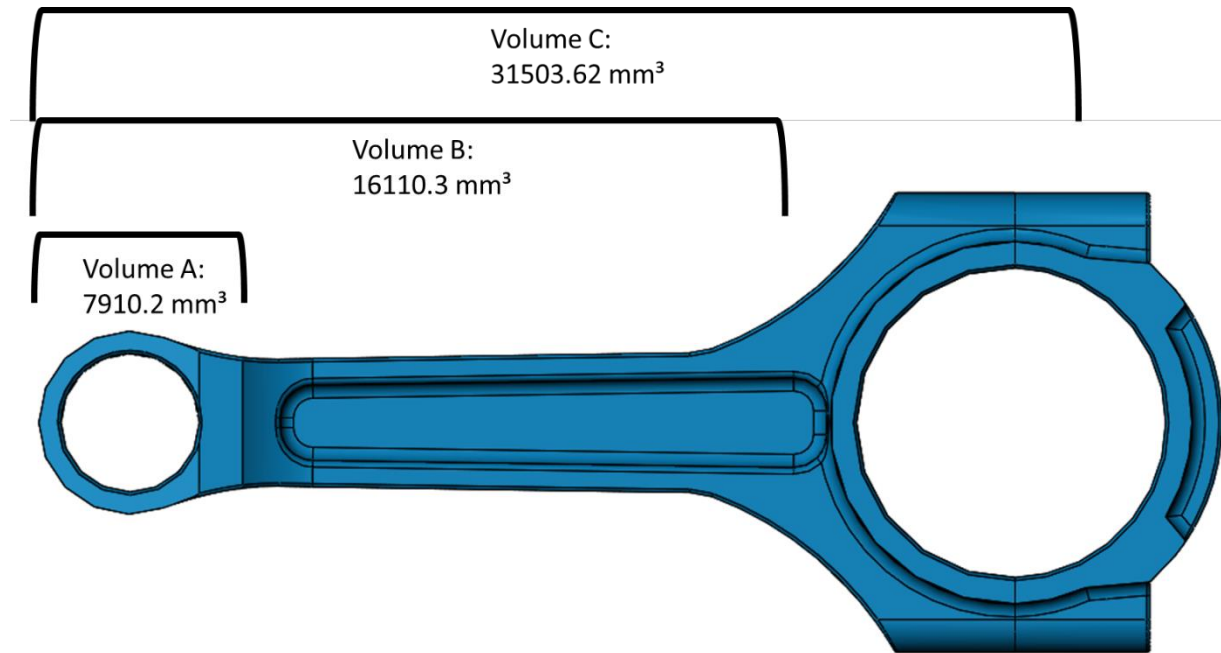


Figure 8 – Volumes for Calculating Stresses on Point A, B and C

4.4 Stresses acting on Point C

The following equations are used for Point C:

Table 12 – Equations for Stresses at Point C

Equations for Stresses at Point C	
Reciprocating Mass	$M_{rec} = \text{Piston Mass} + \text{Pin Mass} + \frac{1}{3} * \text{Rod Mass}$
Rotating Mass	$M_{rot} = \frac{2}{3} * \text{Rod Mass} - \text{Cap Mass}$
Inertia Force (C)	$\text{Inertia Force} = \frac{1}{2} \left(M_{rec} R w^2 \left(\cos \theta + \frac{R}{L} \cos 2\theta \right) \right) + (M_{rot} R w^2 \cos \theta)$
Tensile Stress	$\sigma_T = \frac{\text{Inertia Force}}{\text{Area (C)}}$
Fatigue Safety Factor (C)	$FSF = \frac{\text{Allowable Stress}}{\text{Range of Stress}}$

- At this point only inertia force is present, tensile force. Therefore, compressive load can be assumed to be 0.
- There are two types of inertia acting at this point due to its location near the crank, the reciprocating mass inertia and the rotating mass.
- The area of 138.03 mm², was taken from Figure 9.

- The mass was calculated using the volume of 31503.62 mm^3 , as seen in Figure 8, multiplied by the density of the material.

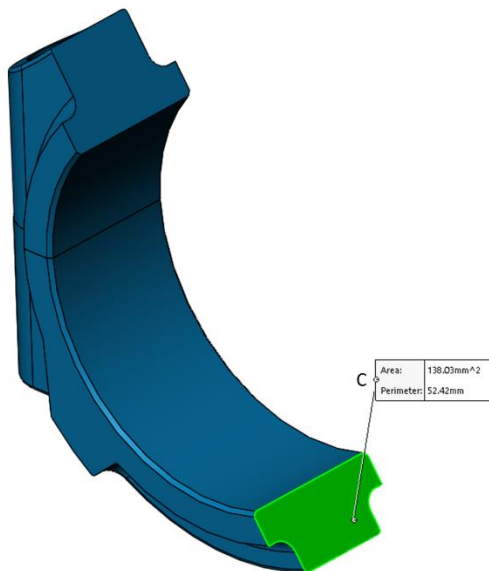
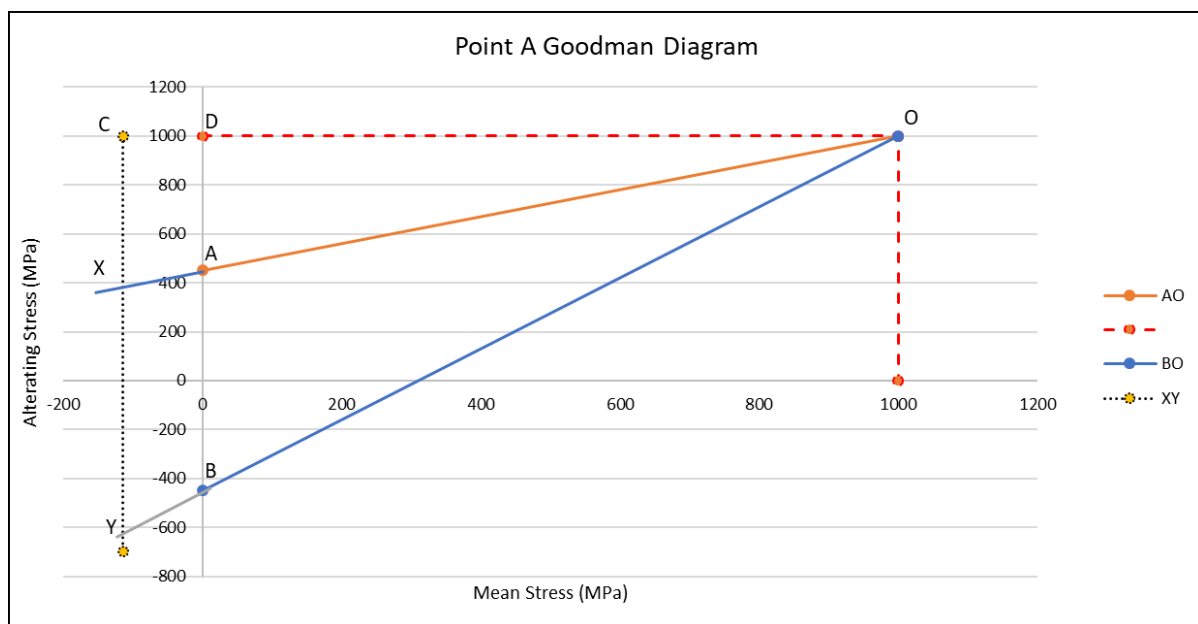


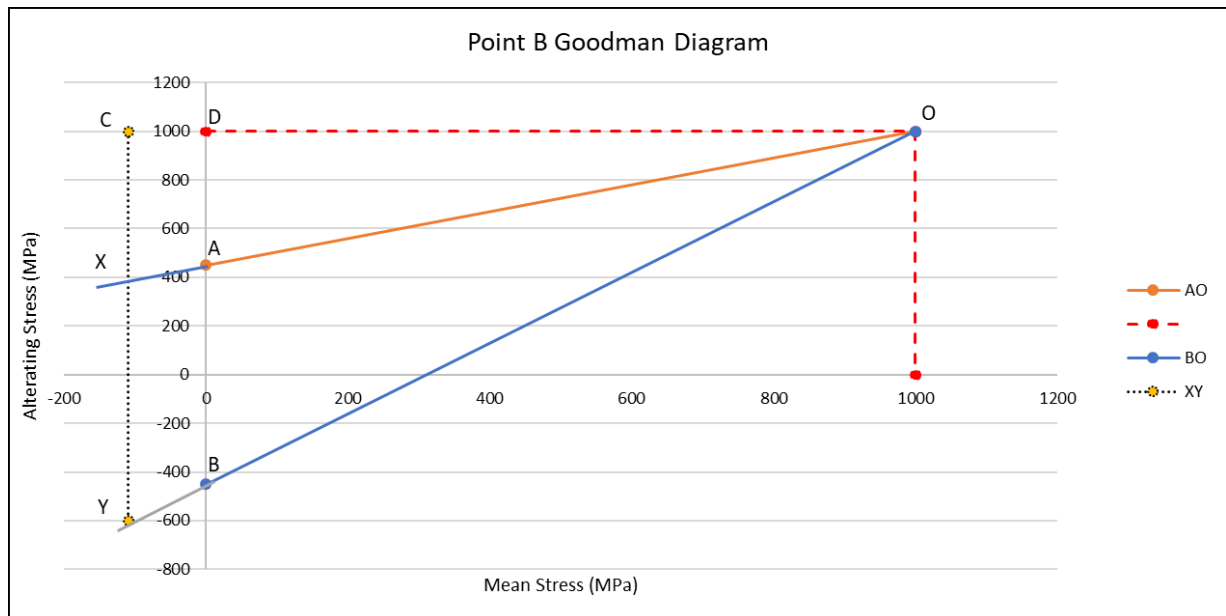
Figure 9 – Area for Point C

4.5 Calculated Data for Points A, B and C

The stress concentration factors for Points A and B are so low that they weren't added into the material endurance limit.



Graph 2 – Point A Goodman Diagram



Graph 3 – Goodman Diagram at Point B

The reason the allowable range in the Goodman Diagrams for Point A and Point B are negative is because the compressive stress is larger than the tensile stress, thus the negative stress is larger.

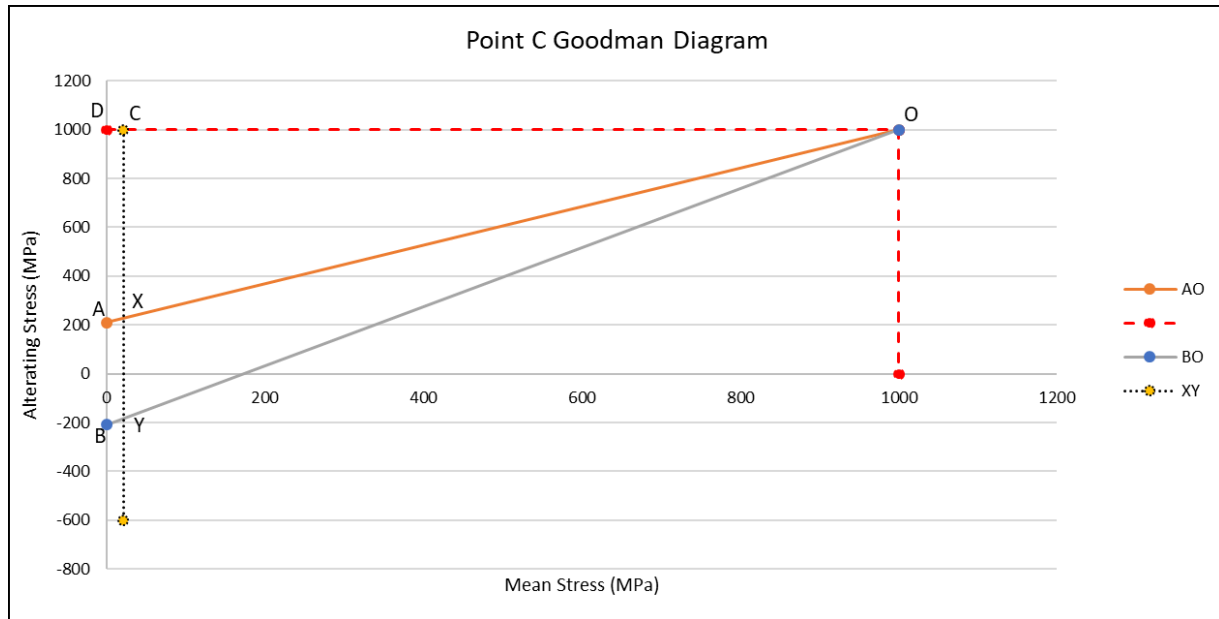
The Safety Factor at Point A is 2.2762, see table 13, and at Point B is 2.3786, see table 14, as expected it is larger at point B thanks to its larger radius the stresses are lower.

Table 13 – Calculated Data for Point A

Calculated Data for Point A	
Tensile Stress	61.3 MPa
Compressive Stress	-289.06 MPa
Mean Stress	-113.9 MPa
Allowable Fatigue Range	797.51 MPa
Range of Stress	350.36 MPa
Fatigue Safety Factor (A)	2.2762

Table 14 – Calculated Data for Point B

Calculated Data for Point B	
Tensile Stress	58.952 MPa
Compressive Stress	-277.99 MPa
Mean Stress	-109.5 MPa
Allowable Fatigue Range	801.43 MPa
Range of Stress	336.94 MPa
Fatigue Safety Factor (B)	2.3786



Graph 4 – Point C Goodman Diagram

The Safety factor at Point C is of 9.578, see table 15, this large safety factor is probably due to not considering the stress of the bolt on that point.

Table 15 – Calculated Data for Point C

Calculated Data for Point C	
Tensile Stress	42.7149 MPa
Compressive Stress	0 MPa
Mean Stress	21.357 MPa
Allowable Fatigue Range	409.16 MPa
Range of Stress	42.715 MPa
Fatigue Safety Factor (C)	9.5789

5. Stresses on Conrod Cap bolt

The conrod bolts are used to hold together tightly the rod cap and the conrod. The two main stresses acting on the bolt are tensile due to inertia load from the conrod and torsional due to the tightening load of the bolt. (Taylor C. F., 1985)

5.1 Equations used to calculate stresses on bolt

Table 16 shows the equations used for calculating Extra Force on bolt, while Table 17 shows the equations used to calculate Tightening Torque on bolt for 90% yield. Furthermore Table 18 shows the equations used to calculate the Fatigue Safety Factor of the bolt.

Table 16 – Equations to calculate extra force on bolt

Equations for calculating extra force on bolt	
Cap Stiffness	$K_{cap} = \frac{E_{cap} * Area\ of\ cap}{Working\ Length}$
Bolt Stiffness	$K_{bolt} = \frac{E_{bolt} * Area\ of\ cap}{Working\ Length}$
Mecc Mass	$Mecc\ Mass = Piston + Pin + Mass\ (B)$
Mrot Mass	$Mecc\ Mass = Mass\ (B) - Cap\ mass$
Alternating Load	$Alt\ Load = \frac{1}{2} \left(M_{ecc} R w^2 \left(\cos \theta + \frac{R}{L} \cos 2\theta \right) \right) + (M_{rot} R w^2 \cos \theta)$
Extra Force on Bolt	$Extra\ F = \frac{Alt\ Load}{\left(1 + \frac{K_{cap}}{K_{bolt}} \right)}$

Table 17 – Equations to calculate bolt torque at 90% yield

Equations for calculating tightening torque on bolt	
Axial Force	$Axial\ F = (90\% \ of\ yield * 0.6) * Shank\ Area$
Average Bolt Radius	$Avg\ Bolt\ r = \frac{1}{2} * \frac{Shank\ Dim + Outer\ Head\ Shoulder\ dim}{2}$
Head Resting Friction Torque	$HRFT = Head\ Coefficient\ of\ Friction * Axial\ F * Avg\ Bolt\ r$
Thread Resting Friction Torque	$TRFT = Thread\ Coefficient\ of\ Friction * Axial\ F * Thread\ Shank\ Radius$
Bolt Torque for 90% Yield	$T = Head\ Friction\ Torque + Thread\ Friction\ Torque$

- The tightening load is directly correlated with the torque value the bolt is at. To calculate this torque value the Head and Thread of the bolt must be considered as two different entities and then added together as in Table 17.

- There is an axial and a torsional force acting on the bolt. For this report it is assumed that 60% of the force is axial, this is inputted into the axial force equation seen in Table 17.
- The alternating load arises when the engine is in operation and comes from the inertia force. It is divided by two as there are two bolts holding the cap onto the conrod. However, this bolt doesn't take the whole stress to itself the cap does as well, this is where K_{cap} and K_{bolt} come into play. The working section is from the start of the bolt shank to the second bolt thread of engagement. (Mcdonald, 2020)
- The Extra Force is used to identify what extra stress from the alternating load.

Table 18 – Fatigue Safety Factor Equations

Equations for calculating Fatigue of bolt	
Bolt Torque for 90% Yield	$T = \text{Head Friction Torque} + \text{Thread Friction Torque}$
Extra Force on Bolt	$\text{Extra } F = \frac{\text{Alt Load}}{\left(1 + \frac{K_{cap}}{K_{bolt}}\right)}$
Extra Stress in Bolt	$\text{Extra Stress} = \frac{\text{Force}}{\text{Bolt Area}}$
Min Stress	$\text{Min Stress} = \text{Bolt UTS} * 0.9 * 0.9$
Max Stress	$\text{Max Stress} = \text{Min Stress} + \text{Extra Stress}$
Endurance Limit	$\text{Endurance Limit} = \frac{\text{Material Limit}}{\text{Stress concentration factor}}$
Fatigue Safety Factor	$FSF = \frac{\text{Fatigue Range}}{\text{Stress Range}}$



5.2 Stress Concentration Factors on bolt

The closest Petersons Concentration Factors found is the Concentration Factor for a flat tension bar with opposite V shaped notches (Pilkey & Pilkey, 2008). The calculation of this concentration factor is done in two steps, firstly the stress concentration factor on a U-shaped notch is found and then this is combined with the following equations to find the stress concentration in a V notch.

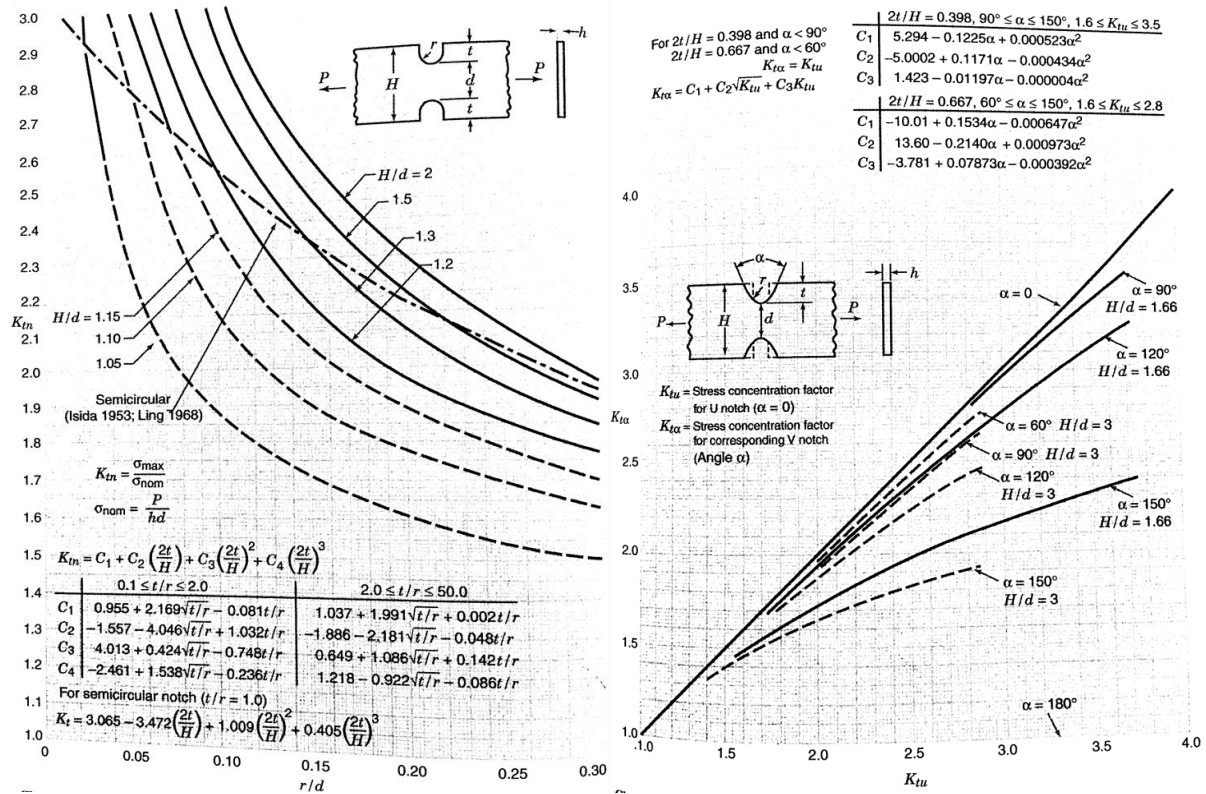


Figure 10 – U shaped stress concentration Factor (Left), V shaped stress concentration factor (Right) (Pilkey & Pilkey, 2008)

Using the equations and data provided in Figure 10 for stress concentrations in axial tension, as well as the measurements in the following table 18 the stress concentration factor of the bolt can be taken.

Table 19 – Bolt measurements for Stress Concentration Factor

Bolt Measurements for Stress Concentration Factor	
H	8 mm
t	0.541 mm
r	0.144 mm
α	60°

Table 20 – Calculated Stress Concentration Factors

Calculated Stress Concentration Factors	
U notch	4.11
V notch	3.58

Table 19 shows the stress concentration factors calculated. The one used in the study is the V notch Factor, the reason it is lower than the U notch is due to the sharper angle a V notch is at than a U notch.

5.3 Measured Data for calculations

The following measurements, Table 21, were taken from the Bosch Engineering Handbook and notes from lectures. (BOSCH, 2000) (McDonald, 2020). The Bolt in question is an M8 x 1.0

Table 21 – Data collected for bolt equations

Data collected for use in equations	
Shank Diameter	7.8mm
Outer Head Shoulder Diameter	12.8mm
Bolt Thread Coeff of Friction	0.18
Bolt Head Coeff of Friction	0.12
Thread Shank Radius	3.675mm
Thread Shank Working Area	39.2mm ²
Working Length	24.58mm
Cap Area for Cap Stiffness	138.03mm

5.4 Calculated Data for Standard Bolt

The Data calculated has been taken when the engine is at 6000RPM

Table 22 – Bolt Torque Data

Bolt Torque	
Axial Force	20956.32 N
Average Head Radius	5.15 mm
Head Resting Friction Force	12.95 Nm
Thread Resting Friction Force	13.86 Nm
Bolt Torque at 90% Yield	26.81 Nm

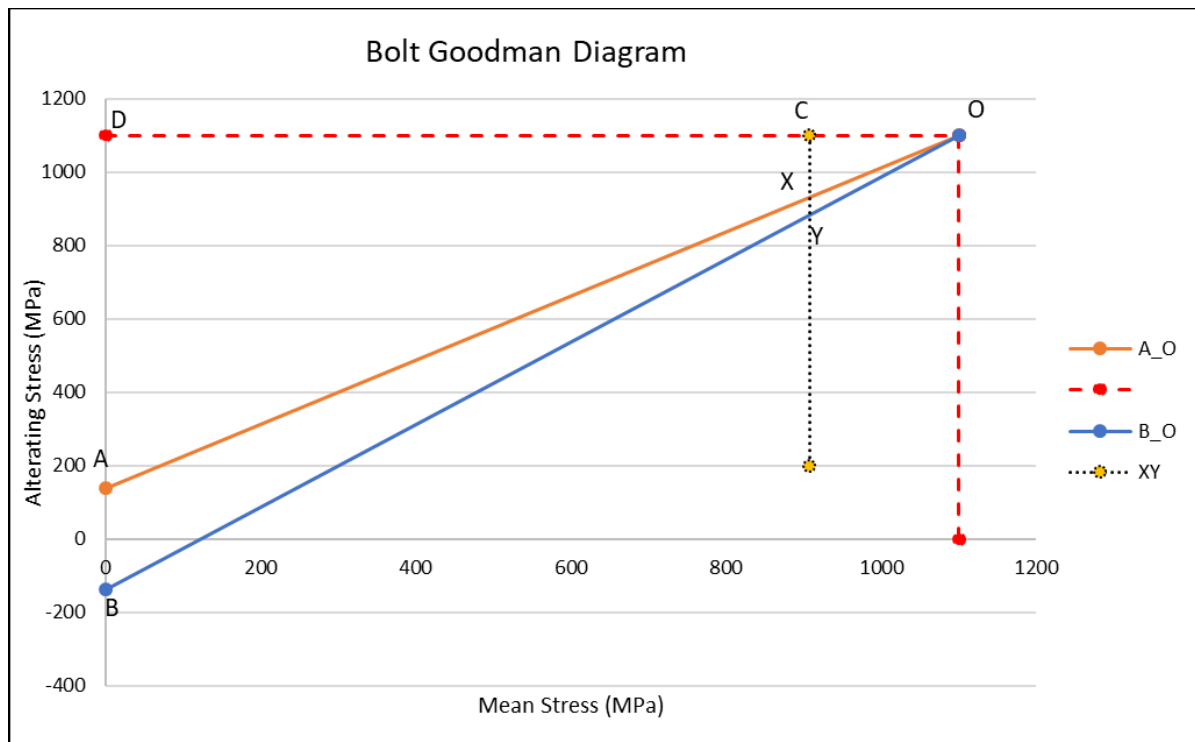
The tightening of the bolt is of 26.81Nm with the data inputted.

Table 23 – Extra Force on Bolt and Stiffness Ratio

Extra Force on Bolt and Stiffness Ratio	
Cap Stiffness	1179263.629 N/mm
Bolt Stiffness	330122.05 N/mm
Stiffness Ratio	3.572
Alternating Load	5895.940 N
Extra Force	1289.518 N
Extra Stress in bolt	32.90MPa

- The stiffness ratio can be calculated by dividing the Cap Stiffness by the Bolt Stiffness, giving a ratio of 3.572
- From the alternating load the Extra Force can be found.
- The extra stress in the bolt is calculated by dividing the Extra Force by the Thread Shank Area, giving a result of 32.90MPa
- The Stress concentration factor is 3.58, as seen in Table 20

With this data a Goodman Diagram can be done.



Graph 5 – Bolt Goodman Diagram at 6000RPM

Graph 5 shows that there is an allowable stress range of 48.38 MPa, with that in mind the Fatigue Safety Factor can be calculated as shown in Table 24. A Safety Factor of 1.47 is found.

Table 24 – Fatigue Safety Factor for bolt

Fatigue Safety Factor for bolt	
Max Stress	923.90 MPa
Min Stress	891 MPa
Mean Stress	907.447 MPa
Allowable Fatigue Range	48.381 MPa
Range of Stress	32.90 MPa
Fatigue Safety Factor (B)	1.47

6. Super Bolt For an uprated engine

The same engine is uprated to 9000RPM, the aim is to redesign the bolt to achieve the same Fatigue Safety factor as the standard bolt at 6000RPM. The conrod is to remain standard, therefore the head and thread on the redesigned bolt is to stay the same.

The material used is ARP2000 used by ARP in their M8x1.0 Performance Conrod Bolts, the UTS of the material, as seen in Table 2, is of 1516.84 MPa and a yield point of 1378.95 MPa.

Table 25 – Materials used by ARP in their Fasteners (Automotive Racing Product, 2019)

QUICK REFERENCE GUIDE TO MATERIALS USED IN FASTENERS				
MATERIAL	USE?	YIELD STRENGTH	TENSILE STRENGTH	USED FOR
Grade 5	No	90,000 psi	120,000 psi	Accessory bolts & studs
Grade 8	No	120,000 psi	150,000 psi	Accessory bolts & studs
"Stainless 300"	Yes	140,000 psi	170-190,000 psi	Accessory bolts & studs
Custom 450®	Yes	150,000 psi	170-190,000 psi	Accessory bolts & studs, head bolts
8740 chrome moly	Yes	180,000 psi	190,000 psi	Rod bolts, head & main studs & bolts
A286	Yes	170,000 psi	200,000 psi	Head bolts, accessory bolts
ARP2000®	Yes	200,000 psi	220,000 psi	Rod bolts, head & main studs
L19®	Yes	200-230,000 psi	260,000 psi	Connecting rod bolts
Inconel 718	Yes	190-210,000 psi	210-230,000 psi	head bolts, case studs & bolts
Custom Age 625+	Yes	235-255,000 psi	260-280,000 psi	Head studs, connecting rod bolts
ARP 3.5®	Yes	220-250,000 psi	260-280,000 psi	Connecting rod bolts
AerMet®	Yes	260,000 psi	290-310,000 psi	Special products
Titanium	Yes	160,000 psi	180,000 psi	Head studs, accessory bolts

The design of the Super bolt was based on the design done by ARP for their Ford Performance Bolts, as seen in Figure 11



Figure 11 – ARP Ford M8x1.0 Performance Bolt (ARP, 2020)

Figure 11 the shank has been thinned, as well as the number of threads reduced drastically.

6.1 Calculated data of forces acting on Super bolt

The same equations are used as for the Standard bolt.

Table 26 – Super Bolt Torque value

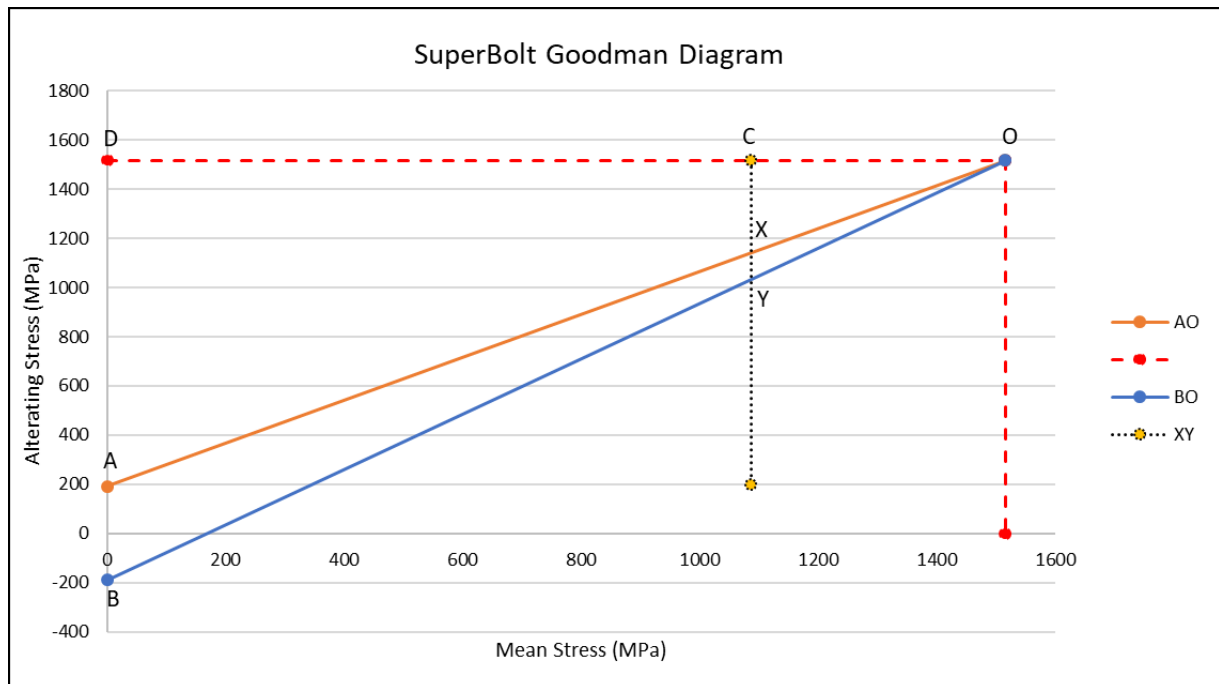
Super Bolt Torque	
Axial Force	20956.32 N
Average Head Radius	5.15 mm
Head Resting Friction Force	12.95 Nm
Thread Resting Friction Force	13.86 Nm
Bolt Torque at 76.37% Yield	26.81 Nm

Due to its higher UTS the yield percentage for the Super Bolt to match the standard bolt would have to lower to 76.37%

Table 27- Extra Force on Super Bolt and Stiffness Ratio

Extra Force on Super Bolt and Stiffness Ratio	
Cap Stiffness	1179263.6 N/mm
Bolt Stiffness	107967.37 N/mm
Stiffness Ratio	10.92
Alternating Load	13078.653 N
Extra Force	1096.981 N
Extra Stress in bolt	68.97 MPa

The change in bolt stiffness, changes the stiffness ratio to 5.44



Graph 6 – Super Bolt Goodman Diagram

Fatigue Safety Factor for Super bolt	
Max Stress	1122.08 MPa
Min Stress	1053.104 MPa
Mean Stress	1087.591 MPa
Allowable Fatigue Range	107.854 MPa
Range of Stress	68.97 MPa
Fatigue Safety Factor (B)	1.56

The Fatigue Safety Factor of the Super Bolt is of 1.56, this increase is due to its higher UTS.

6.2 Drawings of Standard Bolt and Super Bolt

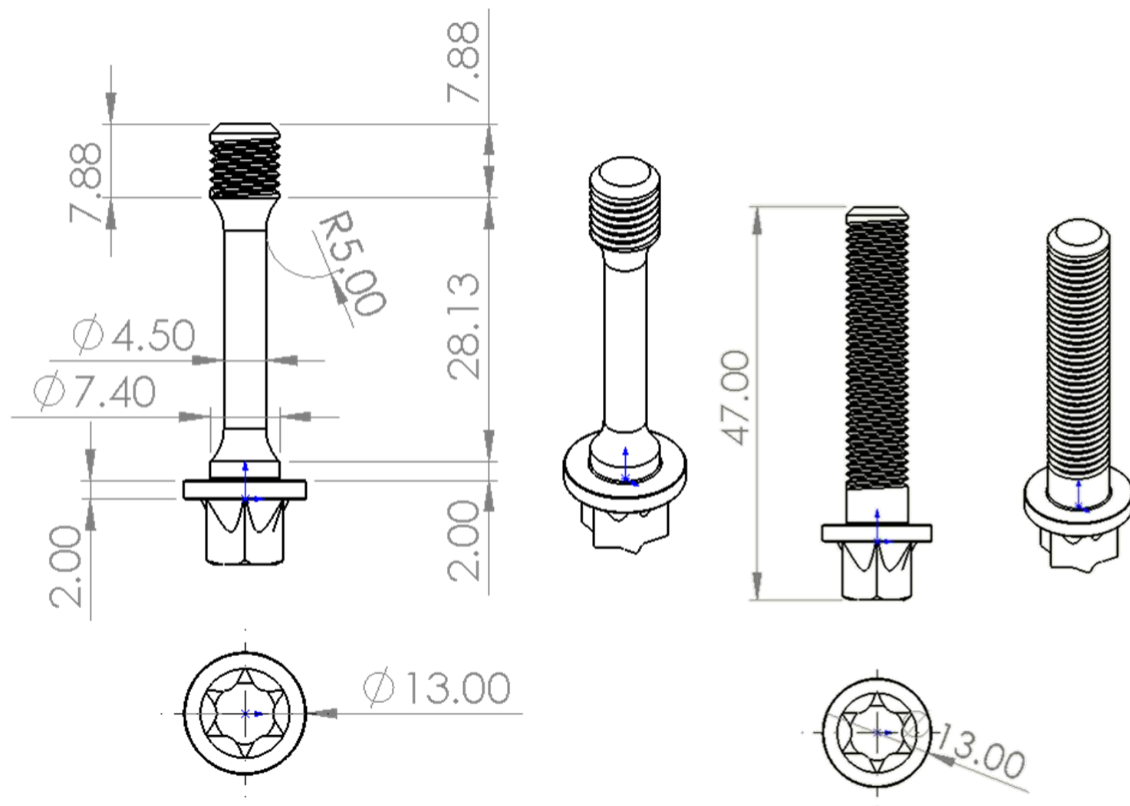


Figure 12 – Super Bolt Technical Drawing (Left), Standard Bolt Drawing (Right)

- The 4.5 mm diameter shank, could be reduced even more if wanted to, as the current safety factor of the bolt is 1.56, the shank diameter can be smaller to achieve the standard bolts 1.36
- The Head of the bolt is the same on both Standard and Super, as well as the thread size.
- The step from the shank to the rest of the bolt has a 5mm fillet to dissipate as much as possible stress concentrations in that area.

7. Stresses acting on Crankshaft

There are three loads acting on the crank bending, shear and torsional.

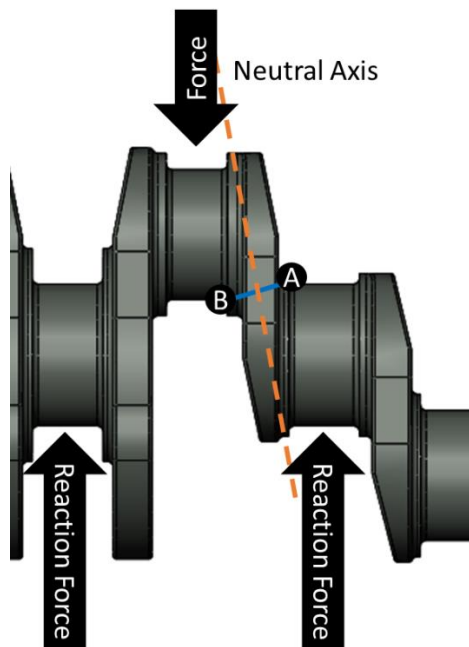


Figure 13 – Crankshaft Section being investigated

Looking at Figure 13, a simple beam scenario is being developed, with the two reaction forces on the outside and the Bending Force. Point B is the point where the max bending stress is happening due to the U notch and where the conrod is located. Point B is in tension when the compressive forces through the conrod are applied and are transferred through the crankshaft, it is in compression through the inertia forces.

7.1 Equations to calculate stresses on crankshaft

The following tables showcase the equations used to calculate the tensile and compressive stresses happening at Point B, as well as the equation used for the Fatigue Safety Factor.

Table 28 – Equations for calculating Crank Tensile stress

Equations for calculating Tensile stress	
Bending Moment	$M = Gas\ Force * L$
Tensile Bending Stress	$\sigma_B = \frac{M}{I} y$
Shear Stress	$\tau = \frac{Gas\ Force}{Area}$
Von Mises Stress	$\sigma_R = \sqrt{\sigma_B^2 + (2\tau)^2}$

Table 29 – Equations for calculating Compressive Stress

Equations for calculating Compressive stress	
Reciprocating Mass	$M_{rec} = \text{Piston Mass} + \text{Pin Mass} + \frac{1}{3} * \text{Rod Mass}$
Rotating Mass	$M_{rot} = \frac{2}{3} * \text{Rod Mas} + \text{Cap Mass}$
Inertia Force Bending Stress	$\text{Inertia Force} = \frac{1}{2} \left(M_{rec} R W^2 \left(\cos \theta + \frac{R}{L} \cos 2\theta \right) \right) + (M_{rot} R W^2 \cos \theta)$
Shear Stress	$\sigma_T = \frac{\text{Inertia Force}}{\text{Area}}$
Bending Moment	$M = \text{Inertia Force} * L$
Compressive Bending Stress	$\sigma_B = \frac{M}{I} y$
Von Mises Stress	$\sigma_R = \sqrt{\sigma_B^2 + (2\tau)^2}$
Fatigue Safety Factor	$FSF = \frac{\text{Allowable Stress}}{\text{Range of Stress}}$

At point A torsional stresses are acting on the crankshaft the following equations in table 30 are used for its calculations.

Table 30- Equations for torsional stresses at Point A

Equations for torsional stresses at Point A	
Torque Force	$T = F_{gas} r \cos \theta + F_{side} r \sin \theta$
Torsional Shear Stress	$\tau = \frac{Tr}{J}$
Second Moment of Area	$J = \frac{\pi * d^4}{32}$

- The Torque values where taken at 5000rpm as it clocked the highest value of 811.70Nm
- For the Gas load and inertia load in tables 28 and 29, the values are taken at 5700rpm and 6000rpm respectively.
- Distance y is 12.23mm and distance L is 35.22 mm
- The Area is of 43035.15 mm⁴

7.2 Stress Concentrations on Crankshaft

There are two types of stress concentrations happening on the crank, as mentioned before in this report there are U fillets in the areas of Point B and A. Furthermore, when under torsion there are torsional stress concentrations happening.

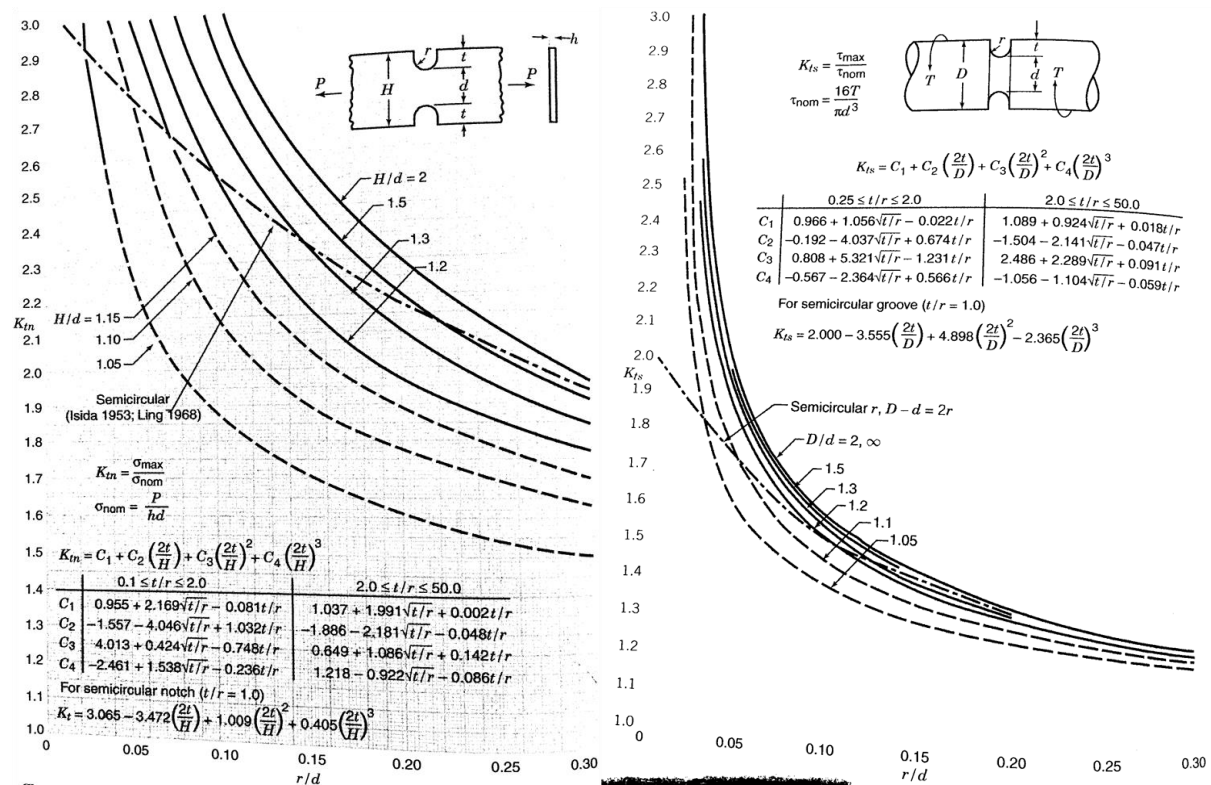


Figure 14 – U shaped stress concentration Factor (Left) and stress concentration factor of torsion of a bar (right) (Pilkey & Pilkey, 2008)

The following measurement in table 31 are used to calculate the stress concentration factors in Figure 14.

Table 31 -Measurements for stress concentration Factor

Measurements for Stress Concentration Factor U groove	
H	7.8 mm
d	44 mm
r	1.5 mm
t	0.5 mm
h	50 mm
Measurements for Stress Concentration Factor torsion	
D	48 mm

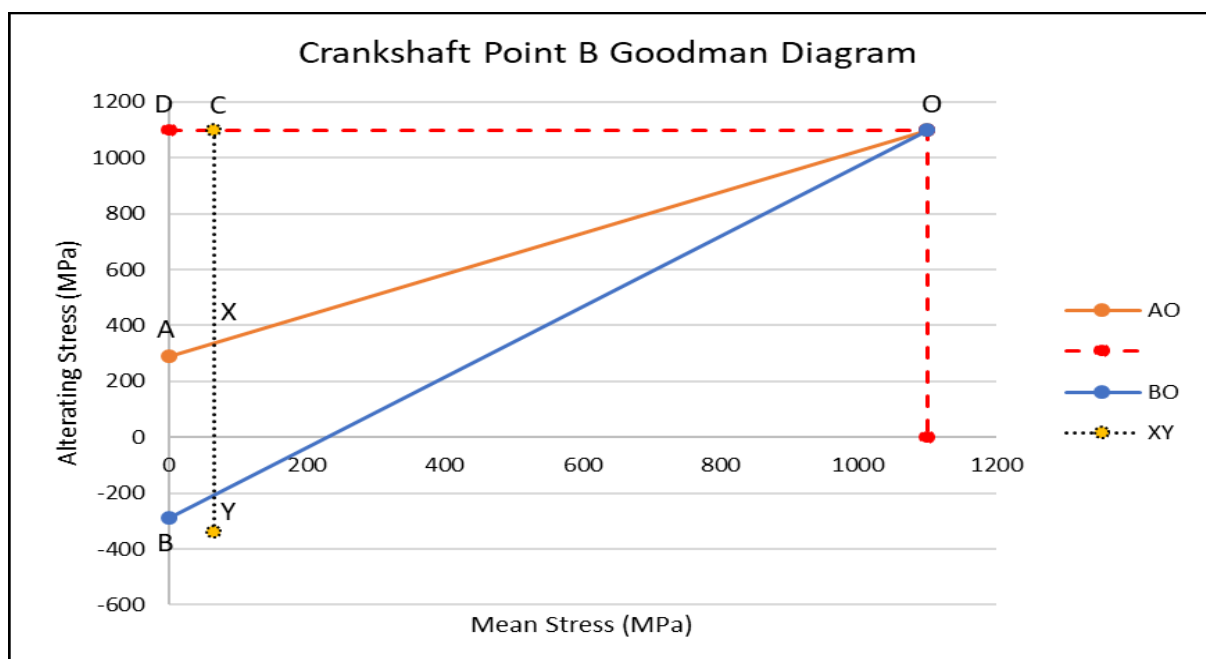
Using the measurements in Table 31 the following Stress Concentration Factors can be found

Table 32 – Calculated Stress Concentration Factor

Calculated Stress Concentration Factors	
U notch	1.71
Torsion of crank	1.521

The stress Concentration Factors are then inputted into its respective Goodman Diagrams in the results section.

7.3 Calculated Stresses on Crankshaft



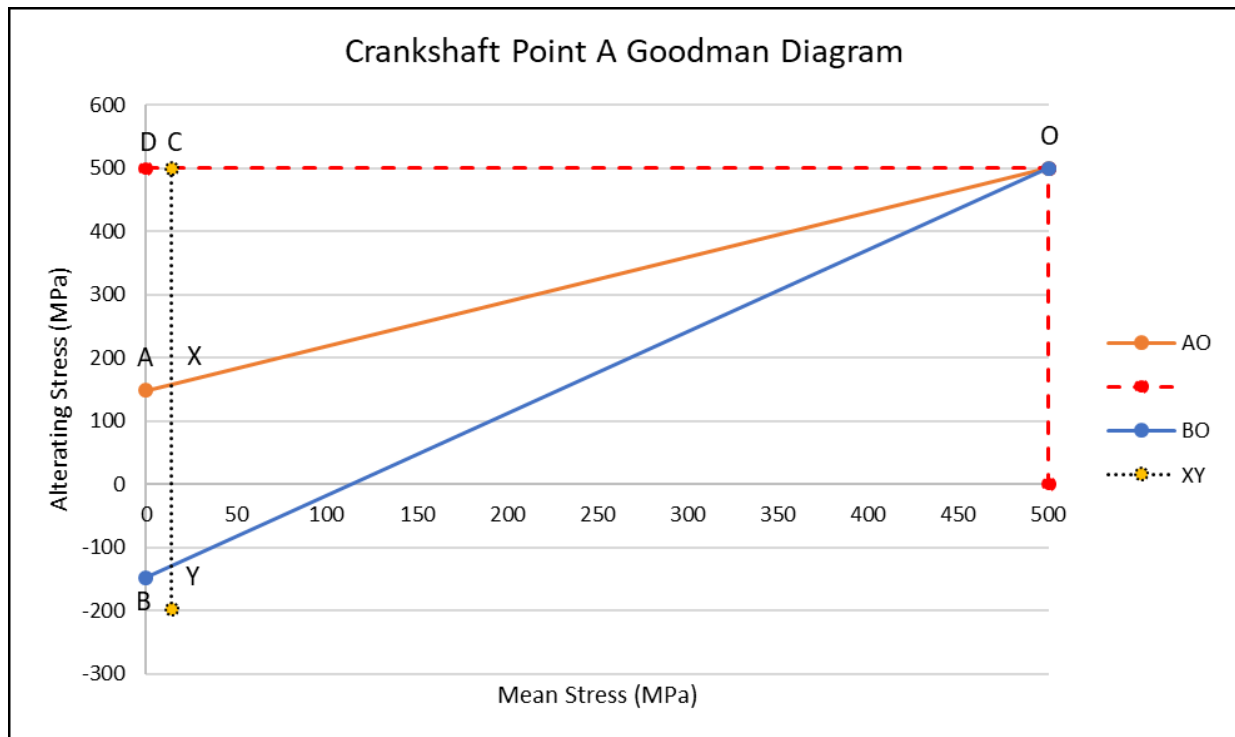
Graph 7 – Goodman Diagram for Point B

From the Goodman Diagram at point B the Fatigue Safety Factor can be found.

Table 33 – Fatigue Safety Factor For Point B

Fatigue Safety Factor for Point B	
Max Stress	206.04 MPa
Min Stress	-75.796 MPa
Mean Stress	65.122 MPa
Allowable Fatigue Range	542.74 MPa
Range of Stress	281.84 MPa
Fatigue Safety Factor (B)	1.925

The Max stress and Min stress are the Von Mises Stress in tensile and compressive bending, and the fatigue safety factor is of a value of 1.925, see table 33.



Graph 8 – Crankshaft Point A Goodman Diagram

Table 34 – Fatigue Safety Factor for Point A

Fatigue Safety Factor for Point A	
Max Stress	37.38 MPa
Min Stress	-9.12 MPa
Mean Stress	16.37 MPa
Allowable Fatigue Range	285.98 MPa
Range of Stress	50.98 MPa
Fatigue Safety Factor (B)	5.608

The max stress and min stress in table 34 are taken from the torsional shear stress in the crank. The fatigue safety factor of the crank is of 5.608

8. Conclusion

This report teaches the user what basic calculations can be done to find what stresses and what fatigue safety factor each element of the assembly has.

Further work can be done in seeing what additional forces are acting on the components, such as thermal forces and friction forces.

Finite Element Analysis can be done for each element, to correlate the hand calculations with FEA, also real world testing can be undertaken, for example by applying tensile and compressive loads to the conrod, using the initial hand calculations the rod can be tensile tested within its elastic region.

9. References

- ARP. (2020). *Ford CVH M8 x 1.0 rod bolt kit*. Retrieved from ARP Store: <https://www.arpstore.com/arp-151-6004-ford-cvh-m8-x-1-0-rod-bolt-kit.html>
- Automotive Racing Product. (2019). 2019 Catalog and Tech Guide. Ventura, USA.
- BOSCH. (2000). *Automotive Handbook*. Germany: Robert Bosch.
- Keough, J. R. (1998, August). *Ductile Iron Data for Design Engineers*. Retrieved from Ductile Iron Society: <https://www.ductile.org/didata/Section4/4intro.htm#Mechanical%20Properties>
- Liu, S. (2012). *Foundry and Development of Ductile Iron Crankshaft*. Chongqing: The Open Mechanical Engineering Journal.
- Mcdonald, M. (2020). *Advanced Engine Design*. Swansea: UWTSD.
- Mousley, S. (2018). *Investigation into the Effect of Lubricants on Fastener Coefficient of Friction and an In-depth Fatigue Analysis of a Connecting Rod Bolted Joint*. Swansea: UWTSD.
- Ovako. (2020). *Material data sheet Steel Grade*. Retrieved from Ovako.
- Pilkey, W. D., & Pilkey, D. F. (2008). *Peterson's Stress Concentration Factors*. New Jersey: John Wiley & Sons.
- Taylor, C. F. (1985). *The Internal Combustion Engine in Theory and Practice*. Massachusetts: The MIT Press.
- Taylor, C. F. (1985). *The Internal Combustion Engine in Theory and Practice Volume 1* (2nd Edition ed.). Massachusetts: MIT Press.

10. APPENDIX

10.1 APPENDIX PIN

PIN CALCULATIONS

Pin Measurements				
L1	8.298	mm	0.008298	m
L2	6.43	mm	0.00643	m
L3	11.085	mm	0.011085	m
L4	7.79	mm	0.00779	m
D1	21	mm	0.021	m
D2	12	mm	0.012	m
L5	14.2	mm	0.0142	m
L6	16.97	mm	0.01697	m

Properties		Goodman			
UTS	1000	MPa	x	y	
USS	500	MPa	Point 0	1000	1000
E.L Top	450	MPa	Point A	0	450
E.L Bottom	-450	MPa	Box	0	1000
				1000	1000
				1000	0
			Point 0	1000	1000
			Point B	0	-450
			XY	0	1000
				0	-500

Tensile Bending Stress		
RPM	5700	N
Net Gas Force	40786.55	
Half of Net Force	20393.28	
Crank angle	20	Deg

L	0.134	m
R	0.0407	m
Density	7800	Kg/m³
Mass Pin	0.025836	Kg
Pin Inertia Force	439.2312	N
I	8528.688	mm⁴
y	10.5	mm

Compressive Moment	169223.4	Nmm
Pin Inertia Moment	2824.257	Nmm
Total Moment	166399.1	Nmm

Bending Stress	204.8605	MPa
----------------	----------	-----

Tensile Shear Stress		
Force	19954.04	N
Area	233.2633	mm²
Shear Stress	85.54302	MPa

Tensile Von Stress	266.9049	MPa
--------------------	----------	-----

Mean Stress	110.5538	Mpa
Allowable Fatigue Range	800.5016	Mpa
Range Of Stress	312.7023	Mpa

Fatigue Safety Factor	2.559948	
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Compressive Bending Stress		
Net Inertia Force	6019.789	N
Half of Net Force	3009.895	N
RPM	6000	
Crank angle	-358	Deg

L	0.134	m
R	0.0407	m
Density	7800	Kg/m³
Mass Pin	0.025836	Kg
Pin Inertia Force	487.943	N
I	8528.688	mm⁴
y	10.5	mm

Inertia Monment	24976.1	
Pin Inertia Monment	3137.474	
Total Monment	28113.58	Nmm

Bending Stress	34.61172	Mpa
----------------	----------	-----

Compressive Shear Stress		
Force	3497.837529	N
Area	233.2632545	mm²
Shear Stress	14.9952359	MPa
Compressive Von Stress	45.79737676	MPa

Pin Goodman Diagram

Legend:

- AO: Solid orange line
- BO: Solid blue line
- XY: Dotted blue line

10.2 APPENDIX CONROD





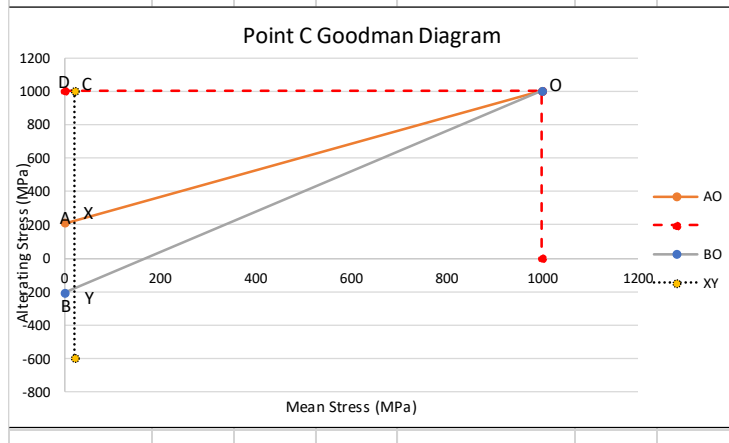
Conrod Measurements			
C → D	11.58 mm	0.01158 m	
C → d	9.34 mm	0.00934 m	
C Area	138.03 mm	0.13803 m	
C → R	0.5 mm	0.0005 m	
Volume (B)	16110.3 mm ³	1.61E-05 m ³	
Volume (C)	31503.62 mm ³	3.15E-05 m ³	
Rod Mass (B)		0.12566 Kg	
Rod Mass (C)		0.245728 Kg	
Piston Mass	249.9 g	0.2499 Kg	
Pin Mass	80.1 g	0.0801 Kg	
Cap Mass	105.2 g	0.1052 Kg	

Properties		Goodman		
UTS	1000 MPa	x	y	
USS	500 MPa	Point 0	1000	1000
E.L Top	209.046 MPa	Point A	0	209.046
E.L Bottom	-209.046 MPa			
		Box	0	1000
			1000	1000
			1000	0
		Point 0	1000	1000
		Point B	0	-209.046
		XY	21.35746083	1000
			21.35746083	-600

Point C Tensile Stress	
Net Inertia Force	6019.79 N
RPM	6000
Area	141.1 mm ²
Crank angle	-358 Deg
L	0.134 m
R	0.0407 m
Density	7800 Kg/m ³
Inertia Force (C)	5895.941 N
Stress	42.71 MPa

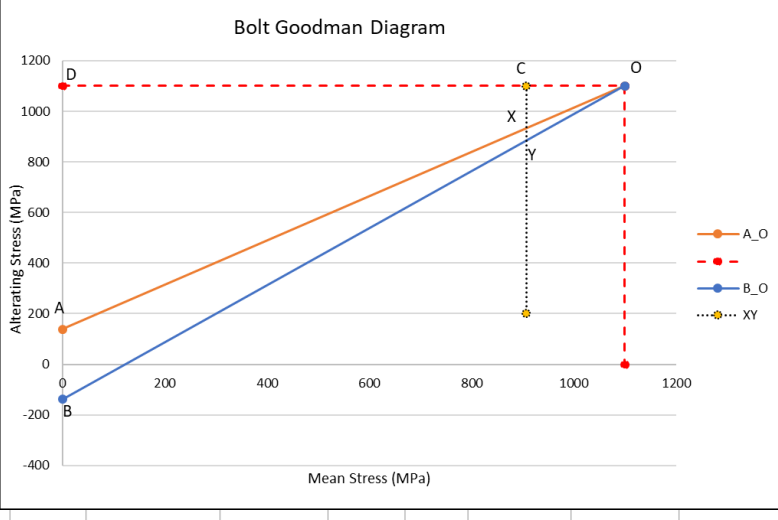
Point C Stress Concentration Factor	
t	2.24 mm
r	0.5 mm
D	11.58 mm
d	9.34 mm
t/r	4.48
2t/D	0.386874
C1	2.931387
C2	-2.10872
C3	0.302602
C4	-0.14219
Concentration Factor	2.1526

Mean Stress	21.35746
Fatigue Rating	409.1627
Range Of Stress	42.71492
SF	9.578917



10.3 Appendix bolt and super bolt

Bolt Measurements		Properties		Goodman	
Shank Dim	7.8 mm	Bolt E	207000 MPa	x	y
Outer Head Shoulder	12.8 mm	UTS	1100 MPa	Point O	1100 1100
Thread Shank Radius	3.675 mm	Yield Stress	990 MPa	Point A	0 138.1946018
Thread Shank Area	39.2 mm ²	90% of Yield	891 MPa	Box	0 1100
Bolt Head Coefficient of Friction	0.12	Endurance Limit	495 MPa		1100 1100
Bolt Thread Coefficient of Friction	0.18	E.L Top	138.1946018 MPa		1100 0
Cap E	210000 MPa	E.L Bottom	-138.1946018 MPa	Point O	1100 1100
Cap Area	138.03 mm ²			Point B	0 -138.1946018
Working Length	24.58 mm			XY	907.4479338 1100
					907.4479338 200
Calculations		Bolt Stress Concentration Factor U notch		Bolt Stress Concentration Factor V notch	
Axial Force	20956.32 N	t	0.541 mm	t	0.541 mm
Average Head Radius	5.15 mm	r	0.144 mm	r	0.144 mm
Head Resting Friction Torque	12.95101 Nm	H	8 mm	H	8 mm
Thread Resting Friction Torque	13.86261 Nm	α	60 Deg	α	60 Deg
Bolt Torque For 90% of Yield	26.81361 Nm	t/r	3.756944444	t/r	3.756944444
Cap Stiffness (Kcap)	1179264 N/mm	2t/H	0.13525	2t/H	0.13525
Bolt Stiffness (Kbolt)	330122.1 N/mm	C1	4.903637117	C1	-3.1352
Alternating Load At 6000RPM	5895.941 N	C2	-6.293730501	C2	4.2628
Extra Force In Bolt	1289.518 N	C3	3.287462417	C3	-0.4684
Extra Stress In Bolt	32.89587 MPa	C4	-0.89219497	CF (V notch)	3.581905468
Min Stress	891 MPa	CF (U notch)	4.110338827		
Max Stress	923.8959 MPa				
Mean Stress	907.4479				
Fatigue Rating	48.38119				
Range Of Stress	32.89587				
SF	1.470738				



Bolt Goodman Diagram

The diagram plots Alternating Stress (MPa) on the y-axis (ranging from -400 to 1200) against Mean Stress (MPa) on the x-axis (ranging from -400 to 1200). A blue line (A_O) represents the Goodman line, starting at point A (0, 138.19) and ending at point O (1100, 1100). A red dashed line (B_O) represents the yield limit, starting at point B (0, -138.19) and ending at point O (1100, 1100). A yellow dotted line (XY) represents the stress path, starting at point X (907.45, 1100) and ending at point Y (907.45, 200). Points C (907.45, 1100) and D (0, 1100) are also marked on the graph.



Bolt Measurements	
Shank Dim	7.8 mm
Outer Head Shoulder	12.8 mm
Thread Shank Radius	3.675 mm
Thread Shank Area	39.2 mm^2
Bolt Head Coefficient of Friction	0.12
Bolt Thread Coefficient of Friction	0.18
Cap E	210000 MPa
Cap Area	138.03 mm^2
Working Length	24.58 mm

Calculations	
Axial Force	20956.32 N
Average Head Radius	5.15 mm
Head Resting Friction Torque	12.95101 Nm
Thread Resting Friction Torque	13.86261 Nm
Bolt Torque For 76.37% of Yield	26.81361 Nm
Cap Stiffness (Kcap)	1179264 N/mm
Bolt Stiffness (Kbolt)	107967.4 N/mm
Alternating Load At 9000RPM	13078.65 N
Extra Force In Bolt	1096.981 N
Extra Stress In Bolt	68.97379 MPa
Min Stress	1053.104 MPa
Max Stress	1122.078 MPa

Mean Stress	1087.591
Fatigue Rating	107.8544
Range Of Stress	68.97379
SF	1.5637

Stiffness Shank	
D1	7.4
L1	2
A1	43.0084
D2	4.5
L2	28.13
A2	15.90431
L3	4
A3	39.2
K1	4451370
K2	117034.9
K3	2028600

Bolt Stress Concentration Factor U notch	
t	0.541 mm
r	0.144 mm
H	8 mm
α	60 Deg
t/r	3.756944
2t/H	0.13525
C1	4.903637
C2	-6.29373
C3	3.287462
C4	-0.89219
CF (U notch)	4.110339

Bolt Stress Concentration Factor V notch	
t	0.541 mm
r	0.144 mm
H	8 mm
α	60 Deg
t/r	3.756944444
2t/H	0.13525
C1	-3.1352
C2	4.2628
C3	-0.4684
CF (V notch)	3.581905468

SuperBolt Goodman Diagram

10.4 Appendix Crank

Crank Measurements			Properties		Goodman	
L	35.22 mm	0.03522 m	UTS	1100 MPa	x	y
2nd moment	43035.15 mm ⁴		USS	500 MPa	Point 0	1100 1100
Y	12.23 mm		E.L Top	288.4489 MPa	Point A	0 288.4489
			E.L Bottom	-288.449 MPa	Box	0 1100
						1100 1100
						1100 0
					Point 0	1100 1100
					Point B	0 -288.449
					XY	65.12238 1100
						65.12238 -338.449

Tensile Bending Stress		Tensile Shear Stress		Compressive Bending Stress		Compressive Shear Stress	
Net Compressive Force	40786.55 N	Force	20393.28 N	RPM	6000	Force	7502.029 N
Half of Net Force	20393.28	Area	1452.2 mm ²	Area	mm ²	Area	1452.2 mm ²
RPM	5700			Mecc	0.452937 Kg		
Compressive Monment	718251.1 Nmm	Shear Stress	14.04302 MPa	Mrot	0.344114 Kg	Shear Stress	5.165975 MPa
Bending Stress	204.1171 MPa	Tensile Von Stress	206.0404 MPa	Crank angle	-358 Deg	Compressive Von Stress	75.79561 MPa
				L	0.134 m		
				R	0.0407 m		
				Density	7800 Kg/m ³		
				Inertia Force	7502.029 N		
				Total Moment	264221.4 Nmm		
				Bending Stress	75.08812 Mpa		

U notch Concentration Factor		Mean Stress	
t	0.5 mm	Mean Stress	65.12238
r	1.5 mm	Fatigue Rating	542.7441
H	7.8 mm	Range Of Stress	281.836
d	44 mm		
t/r	0.333333		
2t/H	0.128205		
h	50		
C1	2.214817		
C2	-4.35412		
C3	6.347201		
C4	-21.282		
CF (U notch)	1.716075	SP	1.925745

Crankshaft Point B Goodman Diagram

