

University of Wales Trinity Saint David School of Engineering

*Calculation on Piston, Rod, Crankshaft
Assemblies*

Advanced Engine Design M6X00410

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1 Introduction

In this report the author will investigate the fatigue safety factors for infinite fatigue life for the duty cycle of a Ford 1.6L GDI engine. In current times components of an engine are becoming lighter and lighter so its important to still make sure that they can still take the loads of the internal combustion engine. This will be done by calculating forces and stresses acting on different parts of the engine and then will use this is calculate the safety factors designed by Ford. It will then be used to adapt a bolt so that it is lighter weight than the original with a similar safety factor using Goodman diagrams (P.S.Veers, 2011).

2 Background

2.1 Stress

Stress can be split into a number of different forms. The main in this paper are compression and tension. The actual values are not calculated but from these forces the shear and bending stresses are calculated. There is also torsion which is caused from an object twisting (Parsons, n.d.).

2.1.1 Bending Stress

Bending stress is quite self-explanatory. It is the stress which is put through an object when it is bending. This is when half of a bar or gudgeon pin is in compression and the other half is in tension. But between these there is a line which does not change size in pure bending called the neutral axis (E.Gillam, 1969) which is deemed to have zero stress acting on it. This is a line which will go through the centre of the material which does not physically exist but it is a point which calculation are based off. The bending stress is calculated from the bending moment, distance from neutral axis and the 2nd moment of area.

$$\text{Bending Moment } (M) = F \times L \quad \text{Equation 1-Bending Moment}$$

$$\text{2nd Moment of Area } (I) = \frac{\pi \times D^4}{64} \quad \text{Equation 2-2nd moment of area}$$

$$\text{Bending Stress } (\sigma_B) = \frac{MY}{I} \quad \text{Equation 3-Bending stress}$$

F=Force

L=Length to point being measured

D=Outer Diameter of bar

Y=Distance from neutral axis to outer diameter

2.1.2 Shear Stress

Shear stress is the stress which has a dislocating force acting through the atomic make up of a material. The resistive force is from the friction properties within the material (Jones & Ashby, 2019). This can also be in fluids which is related to the viscosity of the fluid.

$$\text{Shear Stress } (\tau) = \frac{\text{Force}}{\text{Area}} \quad \text{Equation 4-Shear stress}$$

2.2 Measurements

For the dimensions and masses the author used a mixture of his own measurements and measurements used in previous paper (Thomas, 2019).

Component	Value (Kg)
Piston With 3 rings and 2 circlips	0.26677
Just the Con rod Mass	0.2669
Con Rod Cap	0.105
Con Rod Bolt	0.0176
Gudgeon Pin	0.08

Table 1- Masses



Figure 1-measuring the masses

3 Stresses on Gudgeon Pin

The gudgeon pin is acted on from the force of the piston. Because of this it creates a net bending moment. This takes into account the inertia of the pin which is different to the rest of the piston assembly which in turn creates bending and shear stress. The reciprocating mass was taken as the mass of the piston, gudgeon pin and 1/3 of the con rod.

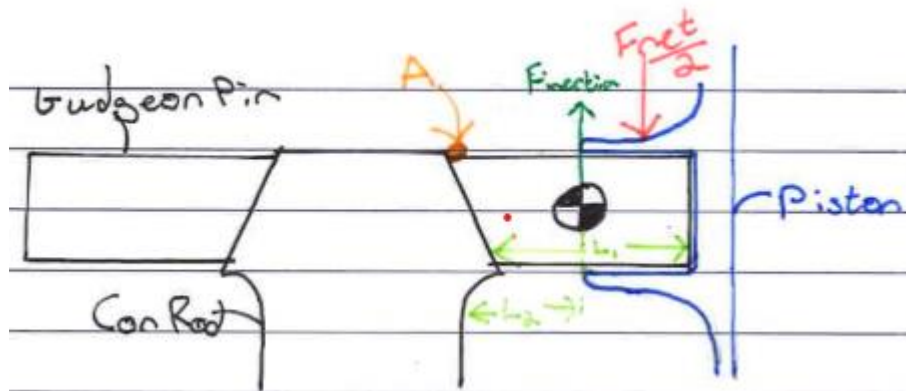


Figure 2-Diagram of piston, pin and con rod

As can be seen the author looked at one side of the pin so the F_{net} is halved. Once the bending and shear stress is calculated the combined stress can then be calculated. This was to make an overall stress so that the values could be compared. The author did this at maximum F_{net} which would be at firing and minimum which is at exhaust. From these values the author then calculated the mean, minimum and range.

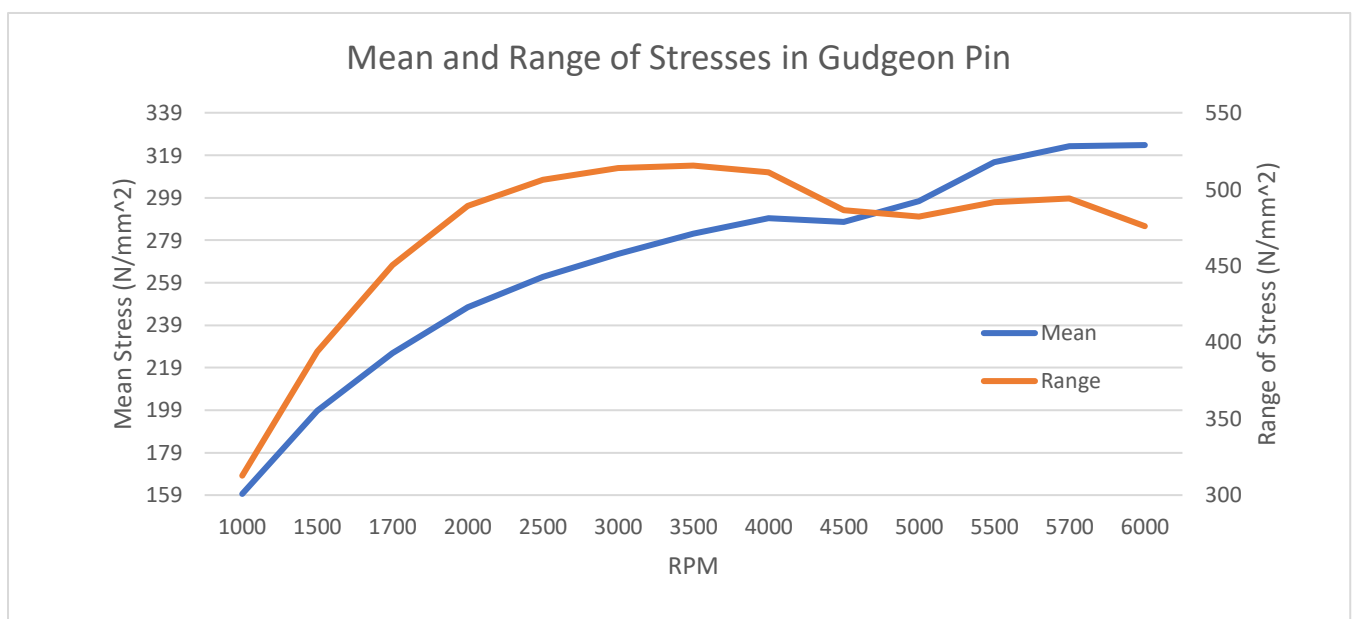


Figure 3-mean and range of stresses on the gudgeon pin

As can be seen the mean stress increases through the rev ranges as the minimum stress only changes with the inertia affecting the RPM but when firing the RPM increases and so

does the average gas pressure so that is why it increases overall. Interestingly though is the shape of the range. As it increases then gets to point (4000rpm) and decreases. This could be perhaps as the rpm increases the gas pressure proportionally increases but at 4000rpm the inertia is increasing more than the gas force so perhaps this causes the range to decrease as inertia subtracts from the net force.

From the mean, a Goodman diagram was made so that the allowable range could be found and therefore the Fatigue Safety Factor (FSF). This is the capability of a product to withstand a load which is higher than the normal or in our case alternating load (CAEbay, n.d.).

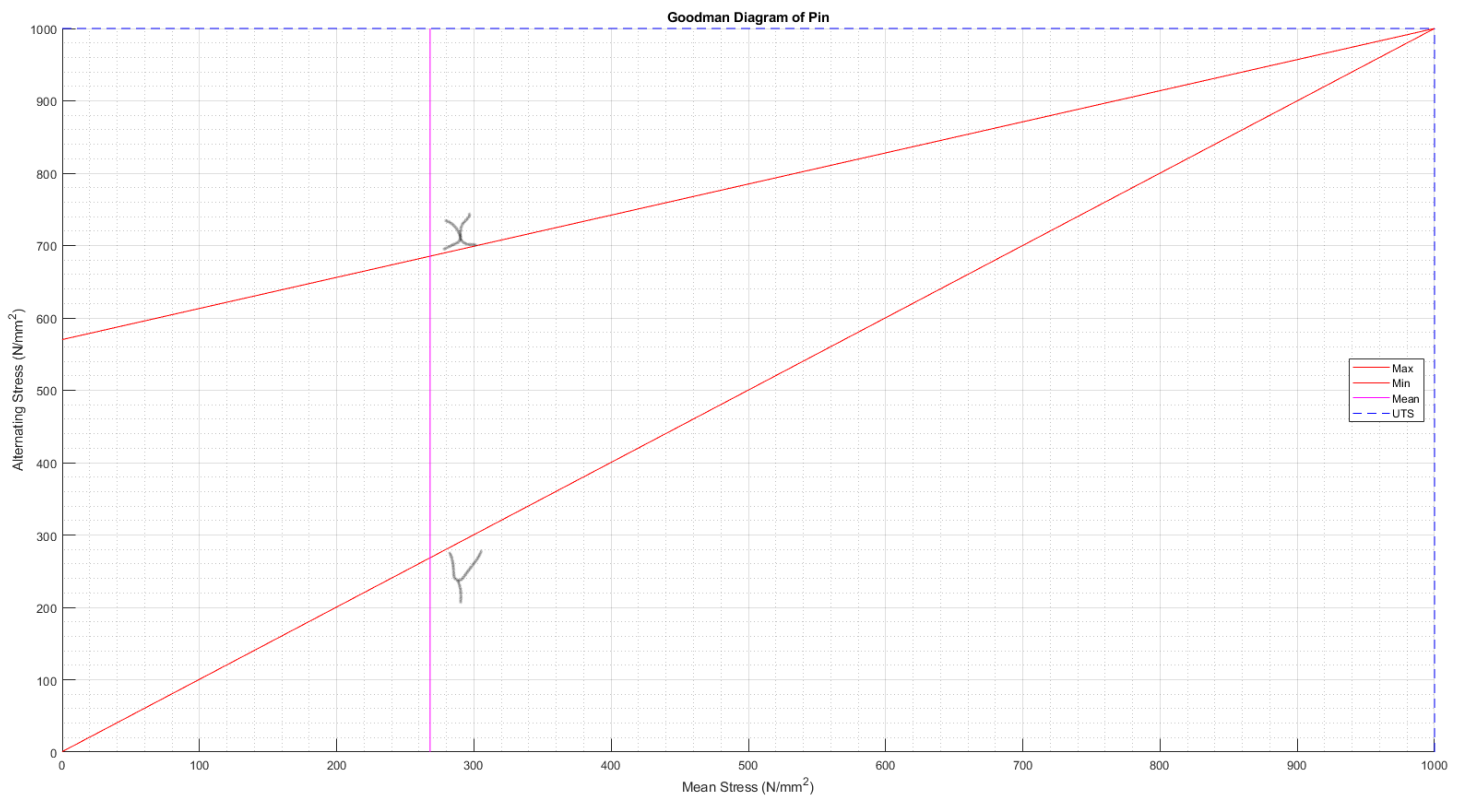
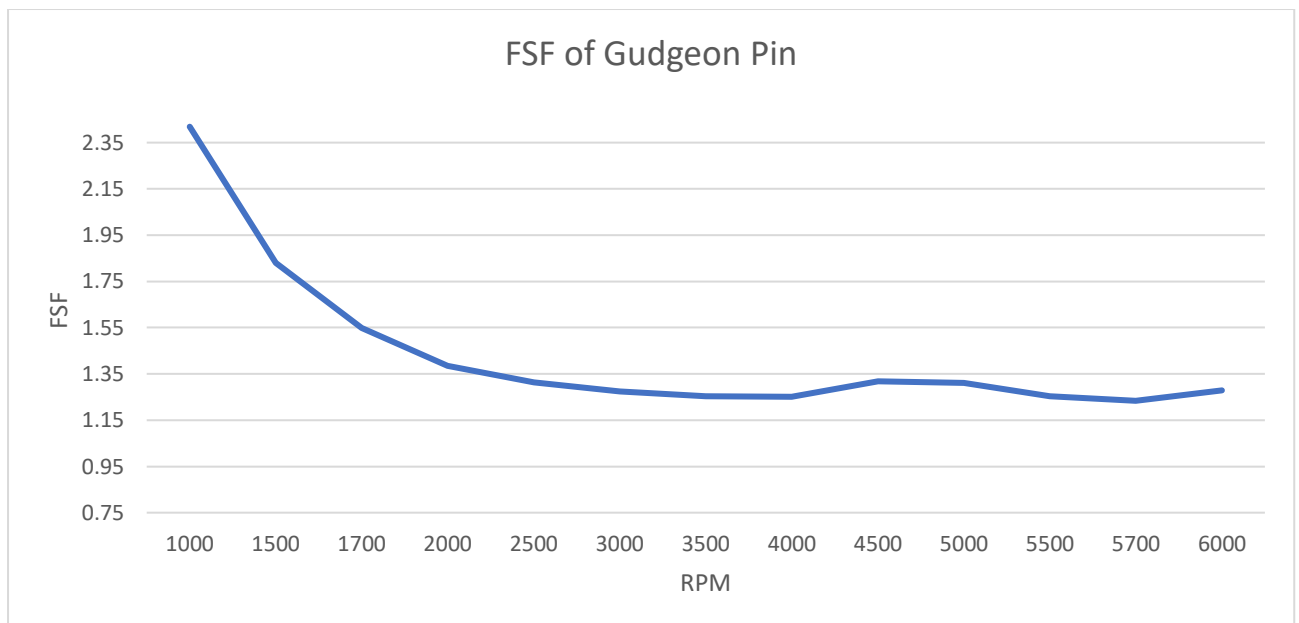


Figure 4-Goodman diagram of the gudgeon pin

From the allowable range the author found that the mean FSF was 1.436074. This is good as if it were below 1 then it will only be able to take the average loads on the pin but Ford have designed the pin so that the FSF has some leeway. However, this was the mean, so the author then calculated the FSF through the RPM's to make see the effects at the extreme loading.



As can be expected the FSF reduces as the RPM increases. This is because there is a much higher load going through the pin. What is interesting is that it is almost like the mean but inverse. This makes sense because it is changing proportionally to the alternating load going through it. The author believes that the calculations could perhaps be correct as the safety factor never really drops below 1 at any point having a 0.2 tolerance. If the car were to over rev the author believes that it would go below 1 as the pin has not been designed to take the extra loads associated with the higher RPM so it would have to be redesigned.

4 Stresses on Con Rod

The stresses on the con rod are different than on the gudgeon pin. The main are compressive and tensile. Compressive is the stress which occurs when the object is pressed, and the atoms are pushed or compressed together. Tensile occurs when the object is being stretched. There is however very little shear acting on a con rod as both ends are free so there is very little lateral load acting on the con rod. To add to that as the author is looking at maximum and minimum loads on the con rod the shear stress generally does not have an effect at the crank angle at which this takes place. The author calculated the stresses acting on the con rod at three different points.

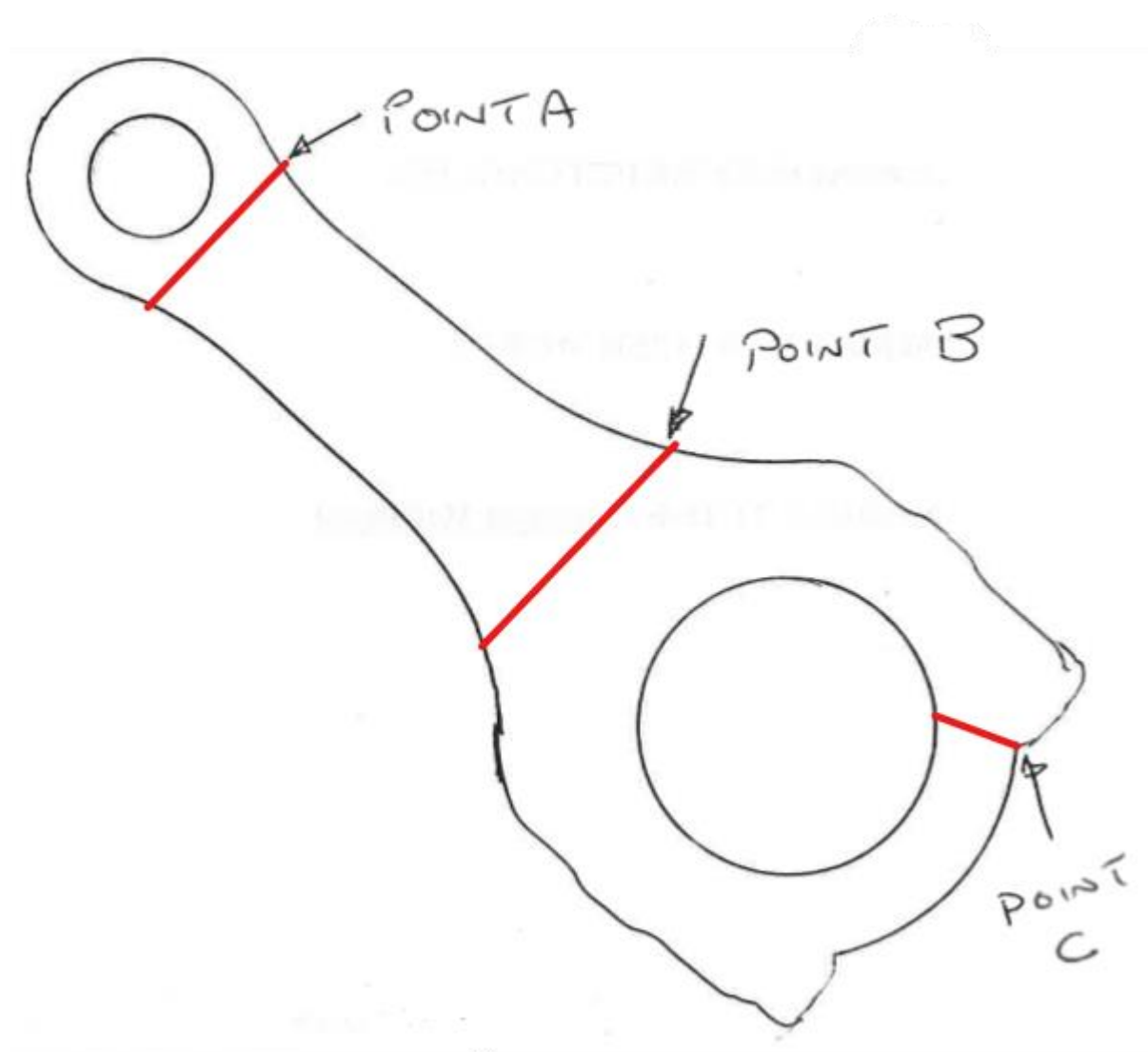


Figure 5-Con rod diagram

4.1 Point A

The author used a CAD file of the con rod and then worked out the cross-sectional area at that point (422 mm^2). The author used the same reciprocating mass as for the gudgeon

pin, but this time added the mass of the gudgeon pin. The author then went through calculating the inertia at that point then using this and the gas force calculated the net force. From the maximum and minimum net force, the nominal stress was calculated which was force over area. The maximum was deemed as compressive where the minimum was deemed as being tension. The writer then went through and calculated the mean of the maximum and minimum and the range of them. They also calculated the polished endurance limit. This is the endurance limit which is considered 45-50% of the UTS when the surface of the material is smooth and has no notches in it (EPI Inc, 2014). The author then used this to calculate the allowable range to calculate the fatigue safety factor.

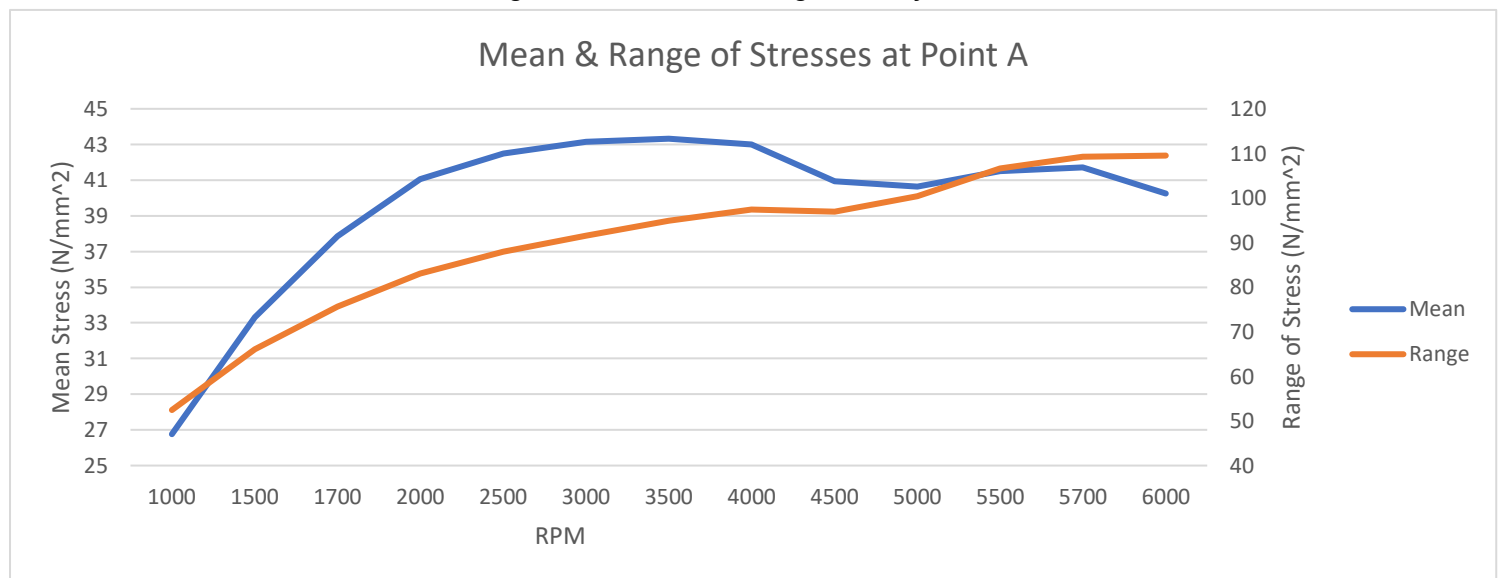
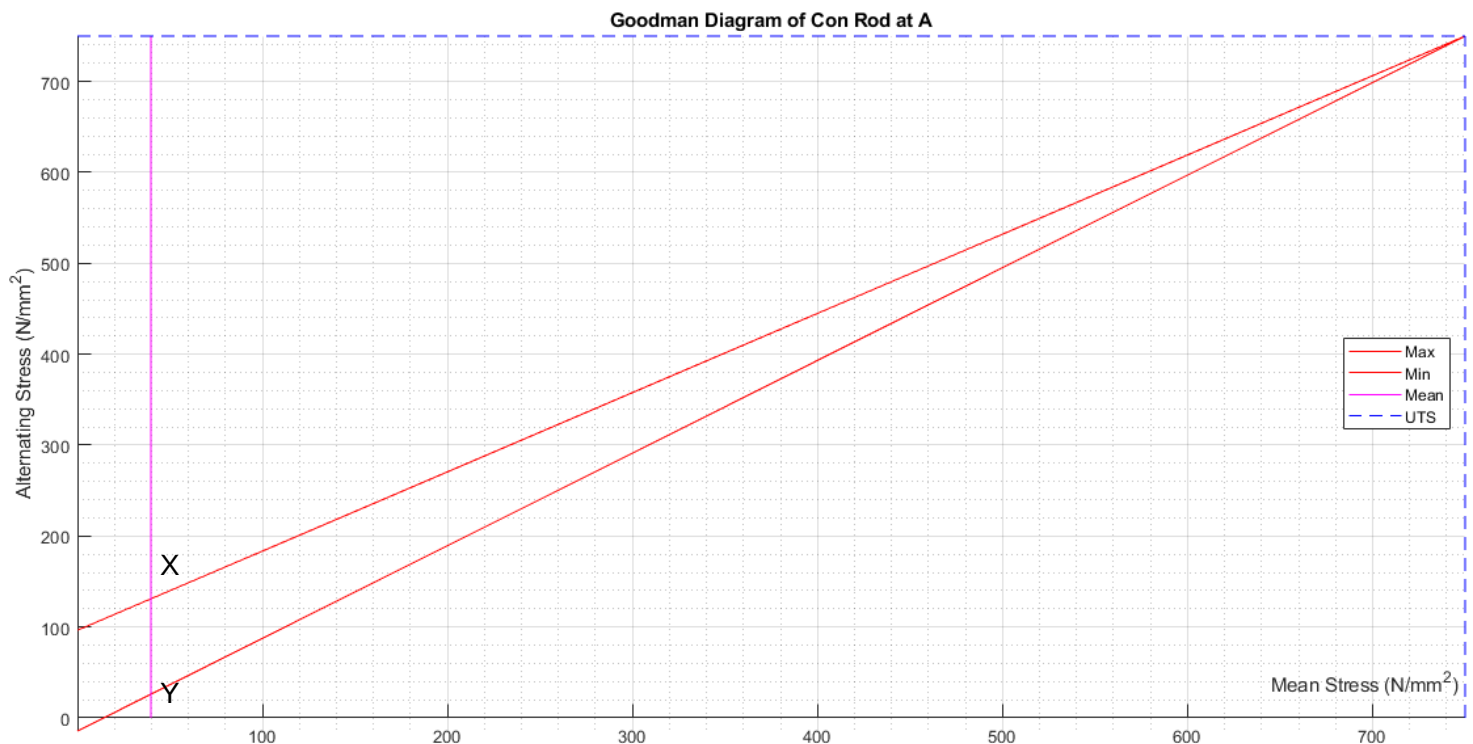


Figure 6-Graph of mean and range of stress acting at point A on gudgeon pin

Interestingly, the range and mean have switched trends from the gudgeon pin. This may be down to the inertia changing as it is now taken as one mass. So, the inertia becomes greater than the gas pressure, therefore the mean stress begins to decrease as the inertia is acting to the opposite direction to the gas force. To add to that the value of the mean stress has also starts to decrease. This could be perhaps due to that there was a bending moment acting on the pin which will create a greater stress as there is more a lever for it to act. To add to that as only compressive and tensile stress is acting on the rod these have less of an effect than shear and bending. The range increasing this way makes sense as the compressive stress increases much more than the minimum hence why the range increases.



The author found the mean FSF was 7.411127. This is interesting because if the writers calculations are correct it suggests that this section is massively over engineered for the expected load. The author also investigated stress concentration factors on this section. However, as the radius was so large the author found that it did not have much of an effect on the stress.

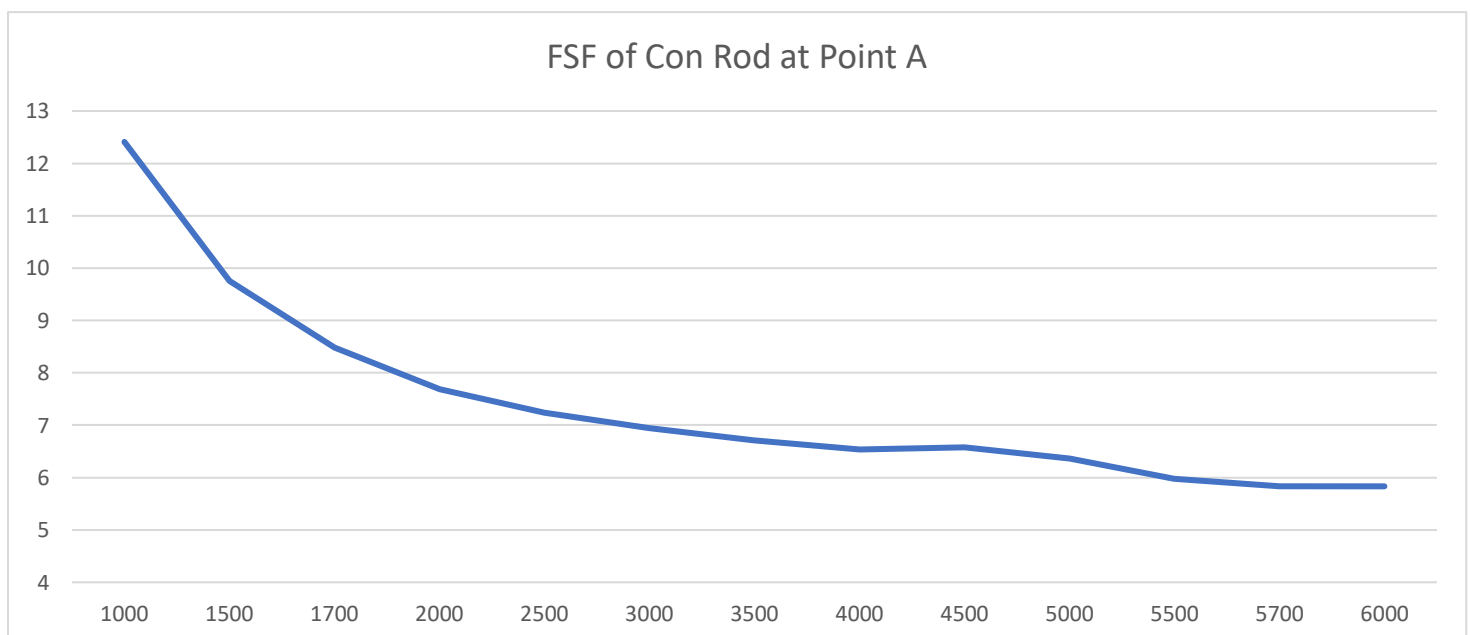


Figure 7- Graph of FSF at Point A on Con rod

This follows the trend that was on the gudgeon pin that the higher the RPM the lower the FSF. In this case it's because the range increases and as the con rod is constantly going in and out of compression, this fatigues the con rod greatly. This graph also suggests that the

con rod at these sections is more susceptible to the range of stress rather than the magnitude. This graph also shows that the FSF never goes below zero but is massively over 1. This suggests that it is either the rod is over engineered at this point or it has been done on purpose to support another section of the rod.

4.2 Point B

Point B is very similar to A, just much further down the rod. To take this into account the author has done the same calculations but using the mass of the piston, gudgeon pin and 2/3 of the con rod to calculate the inertia. To add to that the surface area at that point is also smaller because of the cut outs within the rod.

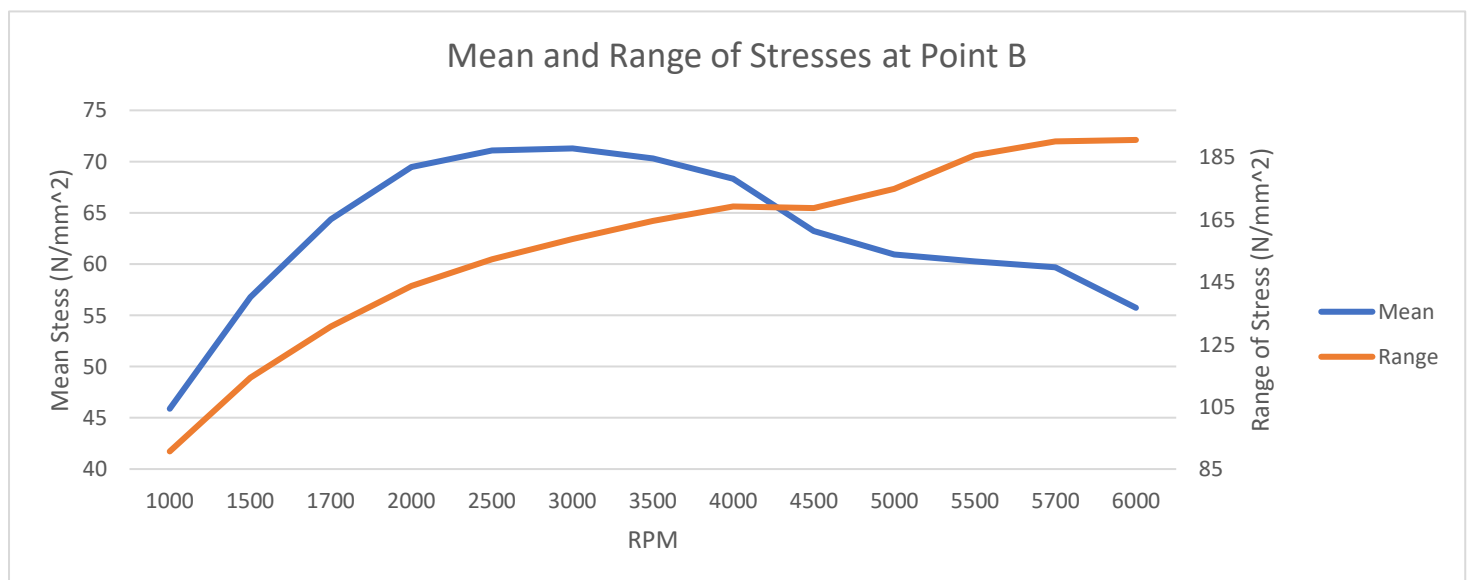


Figure 8-Mean and Range of Stress at Point B on the Con Rod

This is different to point A as the decrease after the peak stress is much larger. This will be because there is even more inertia occurring from the extra mass and no extra gas force. This causes there to be extra tensile stress acting on the rod which will bring the mean down. The range follows the same trend at A with it steadily increasing. There is a small dip at 4500rpm which from previous work the author believes to be from an error in the data (Thomas, 2019). It does appear that the range does start to flatten out the higher the rpm gets. This maybe related to the fact that the mean is beginning to decrease whilst the inertia is still increasing so it may cause the range to flatten out.



Figure 10-Goodman Diagram of con rod at point B

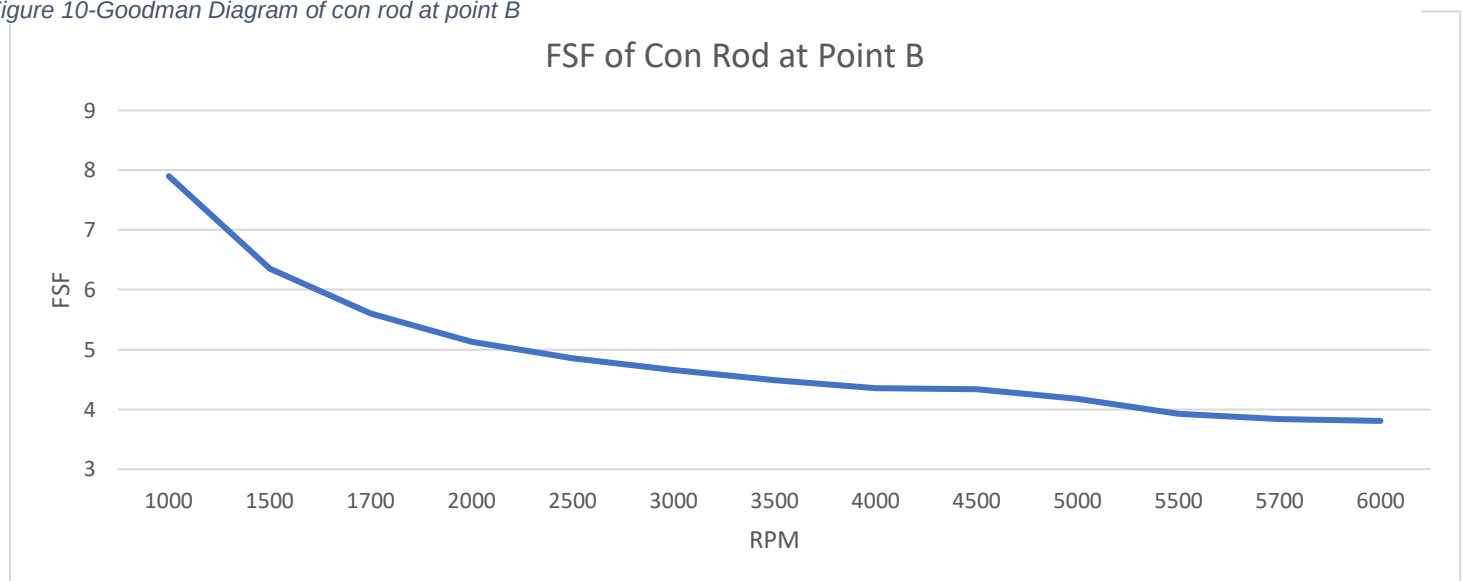


Figure 9-FSF at point B

The mean FSF is much lower at point B than at A being 4.878853. This is down to there being more force going through it with quite a smaller surface area. But again, the author found that this section is over engineered to have an FSF massively over 1. This maybe from Ford making sure that the part will definitely last a long time. But this would be extra mass which would affect the performance of the engine. This leaves scope for a redesign to optimise the conrod. It could however be that the author has left a force out of the calculation which means that it needs to be this strong. On the other hand, it could be due to this section getting influenced by the hoop stresses which may add extra stress at that cross section, so had to make it stronger.

4.3 Point C

Point C is different to the other two points as it has the mass of the whole con rod to contend with but the force acting on it is halved as it is split around the big end. The mass here is considered as the piston, the whole of the conrod and the bolts.

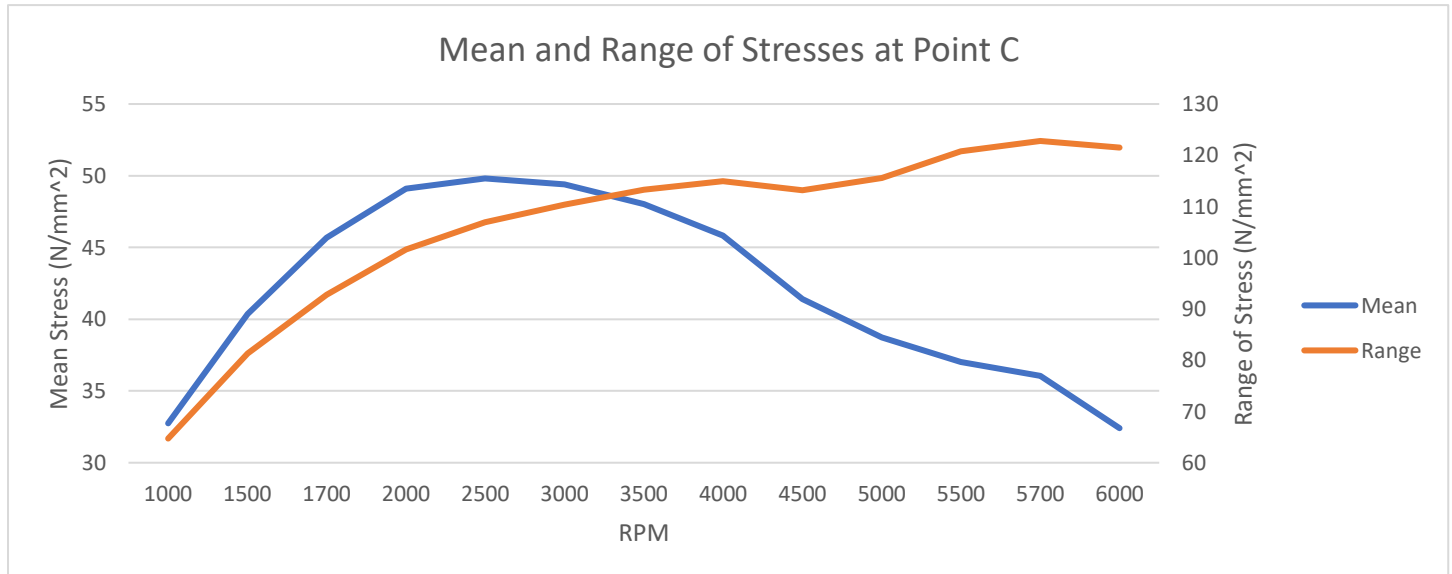


Figure 11-Mean and Range of Stress at Point C

This follows the same trend as point B, but the stress is smaller. This is interesting as you would expect the stress to go up as the surface area is smaller, but the force is halved so it balances the effect of the increase in mass.

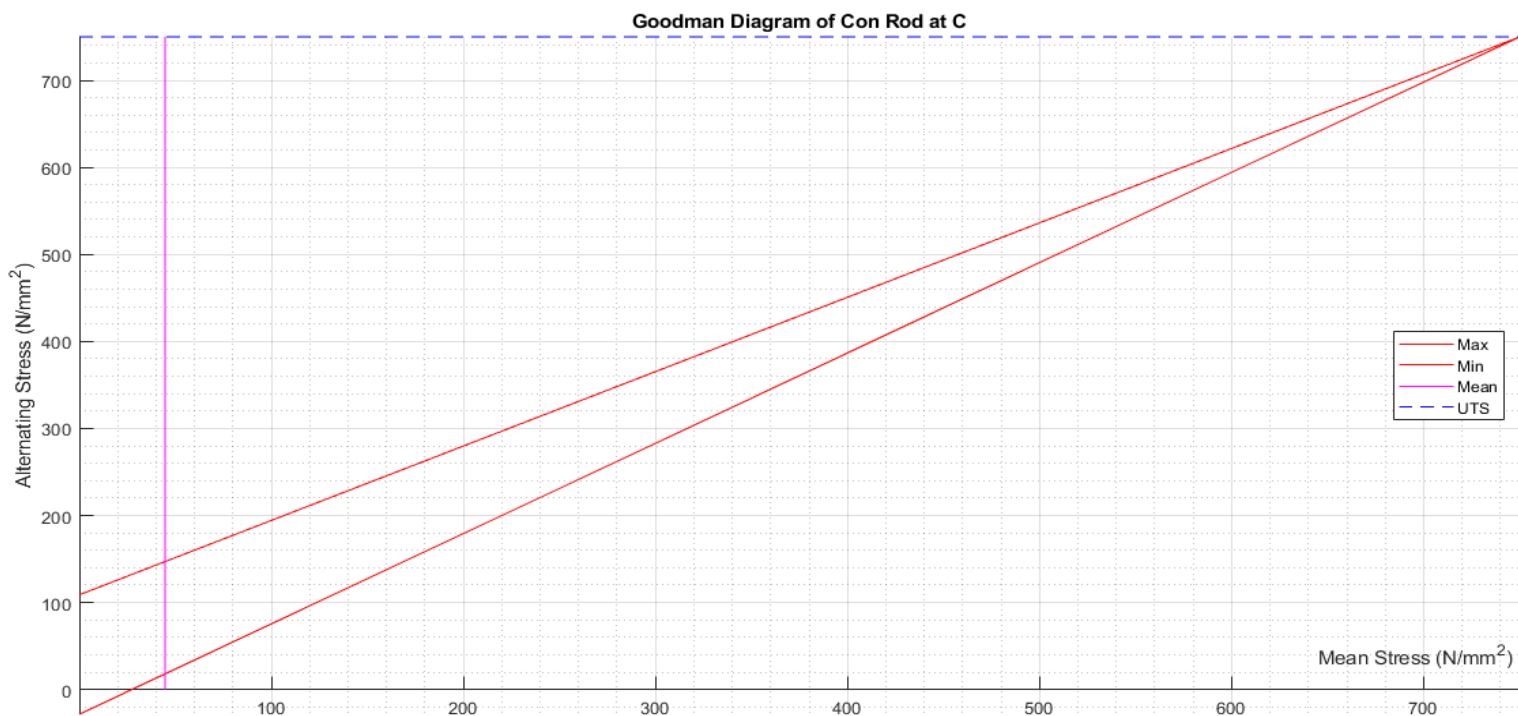


Figure 12-Goodman Diagram at Point C

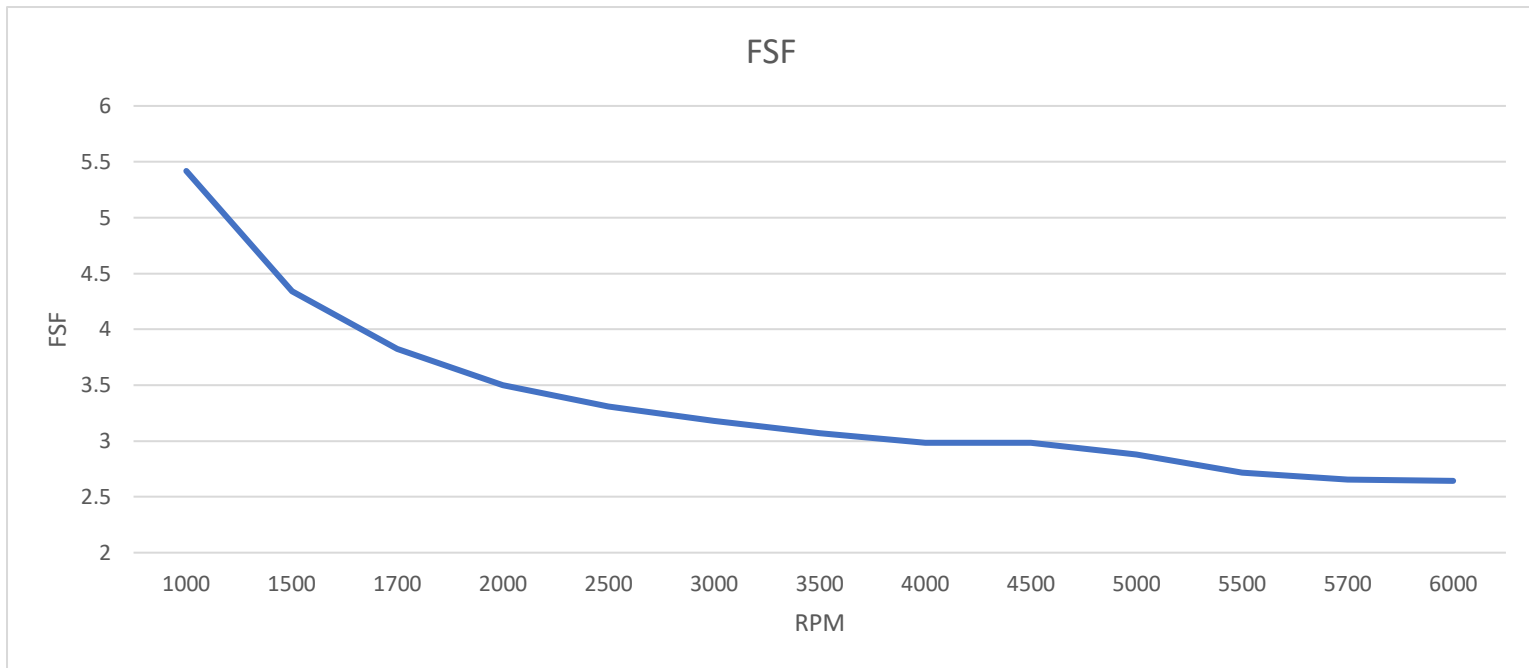


Figure 13-FSF at Point C

This FSF was calculated slightly different as the radius was great enough to use a stress concentration factor. The author measured the radius of the corner using a gauge and compared that with the diameter at that point and found the stress concentration factor using (Pilkey, 1997). The author found the stress concentration factor to be 2. This means that once the polished endurance limit is found this is divided by the stress concentration factor to find the final endurance limit. Because of the stress concentration the FSF has been reduced with an average of 3.346226. To increase this the radius could be increased so to spread the stress over a bigger area rather than in one place.



Figure 14-Measuring the radius to calculate the stress concentration

5 Stress on Bolt

The writer then went onto calculating the stresses acting on the con rod bolt. The author first calculated the ratio of areas. This can give a rough idea of the stiffness ratio as the UTS of the bolt and cap are similar. The author got a value of 2.907143. to calculate the stress acting on the bolt the bolt stiffness has to be calculated. The author got a value of 322823.5294 N/mm using an average width of 7.35mm (Bosch, 2002). Once this was done the author also calculated the cap stiffness which he then used to calculate the stress at the base of the thread. This was done by calculating the peak inertia using the equation below. The reciprocating mass was taken as the mass of the piston, pin and a 1/3 of the con rod and the rotational mass was taken as 2/3 of the con rod.

$$F_{inertia} = M_{rec} \cdot R\omega^2 \left(1 + \frac{R}{L}\right) + M_{rot} \cdot R\omega^2 \quad \text{Equation 5-Maximum Inertia Load for Bolt}$$

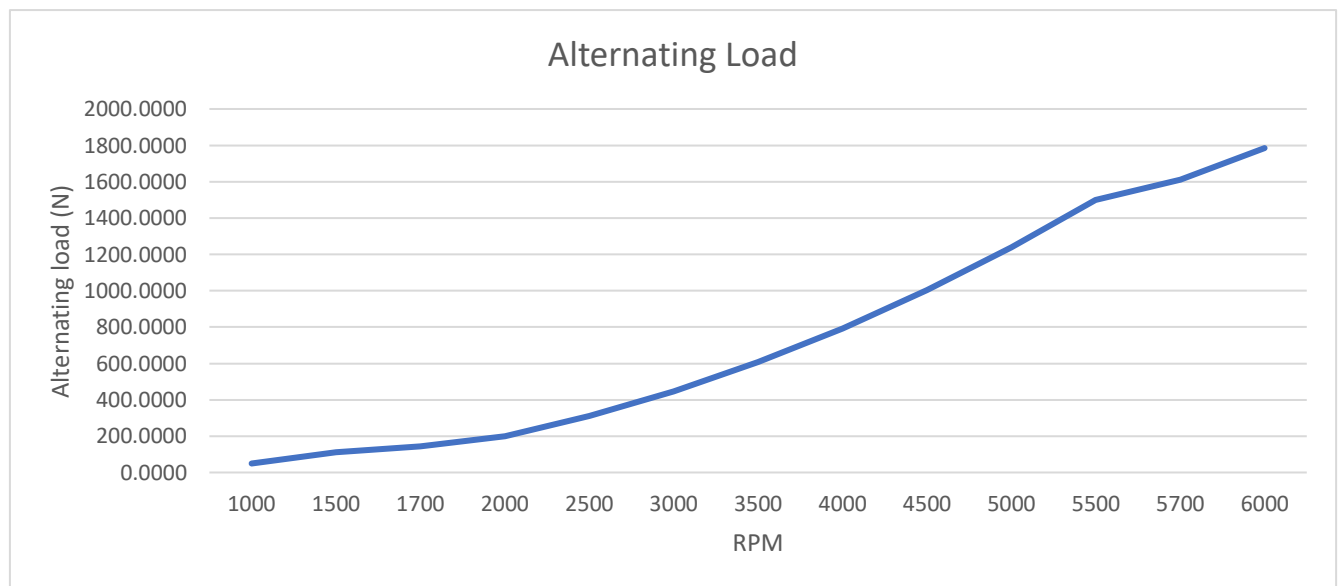


Figure 15-Alternating Load acting on the bolt

As can be seen the alternating load increase proportionally with RPM. This is expected as the main force acting on the bolt is inertia which increase linearly with RPM. Using this the author then calculated the stress acting on the bolt which went up at the same rate as alternating load as it is essentially force over area.

The author then calculated the total torque that was needed to tighten the bolt to 90% yield. First by finding the yield (90% of UTS) then going through each equation stated below.

$$\text{Axial Stress} = 60\% \text{Yield} \quad \text{Equation 6-Axial Stress}$$

$$\text{Clamping Force}(F_c) = \text{Axial stress} \times \text{Bolt Thred Surface area} \quad \text{Equation 7-Clamping Force}$$

$$\text{Friction} = F_c \times \mu \quad \text{Equation 8-Friction}$$

$$\text{Resisting Torque} = \text{Radius} \times \text{Friction} \quad \text{Equation 9-Resisting Torque}$$

The resisting torque was calculated for the thread and then was added to get a total torque needed of 29.58334Nm. The author also investigated tightening the bolt to 18Nm then an extra 35 degrees turn, but this will be in the appendices.

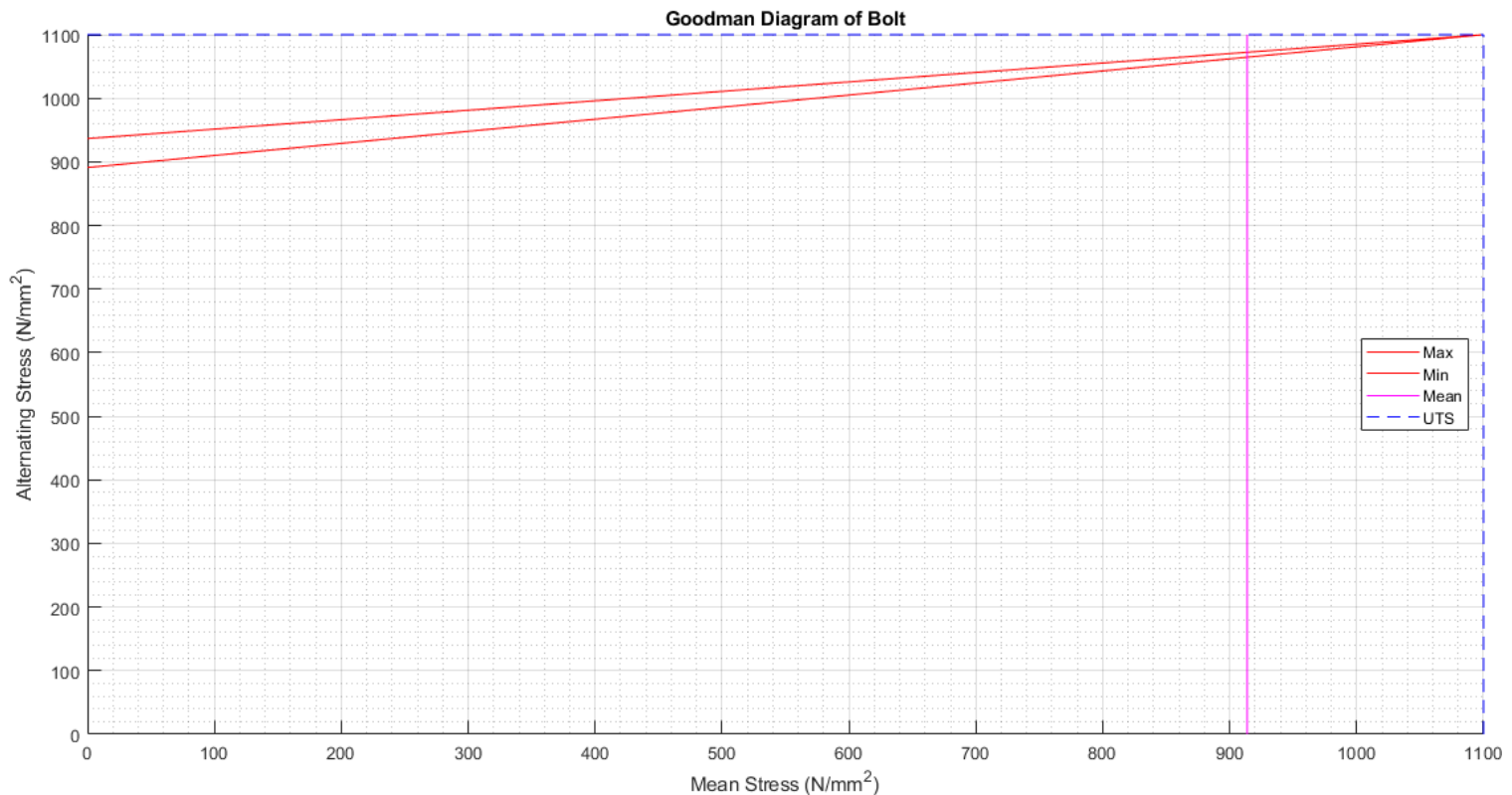


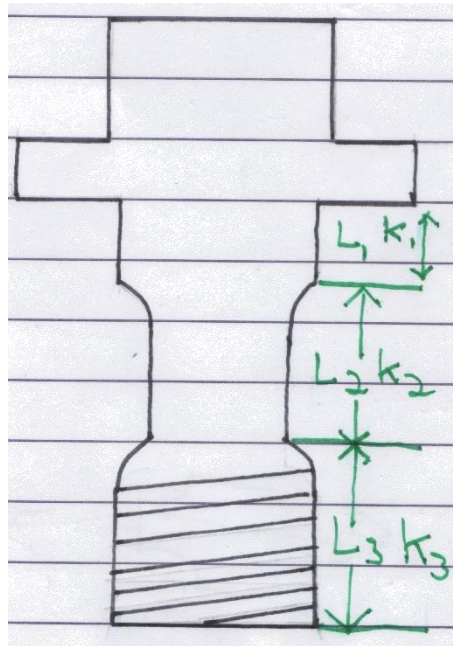
Figure 16-Goodman diagram for bolt

The author then calculated the minimum stress which was 90% of the yield. The author then assumed that the range of the stress was the maximum stress acting on the bolt calculated earlier. Using these he then worked out the maximum and therefore the range. The author calculated the endurance limit using a stress concentration factor of 2 making the final endurance limit 247.5N/mm². Using these the author then calculated the FSF of 1.84. This seems to be a reasonable number as it is above 1 but not massively so that it is not over engineered.

5.1 Bolt Redesign

The author then investigated optimising the bolt by using a different UTS then reducing the mass. The author decided to use a material of a UTS of 1861 called Custom Age 625 Plus® as ARP had used it for applications where high rpm is needed (ARP, 2020). First the author investigated what would happen to the standard bolt if it was taken to 9000rpm. The author found that it would have a FSF of 0.69 which suggests that the bolt would not last very long. The author then changed the material to the higher UTS and this came to a FSF of 3.97 which is quite over engineered. So, to get the bolt back to the original FSF the author would

insert a shank. He did this by first calculating the minimum shank diameter that the bolt could go to by comparing the UTS and dimension and came to an answer of 5.65mm



This is the basic outline the author went for as this is what other manufactures have used so it should work. The author first investigated the following dimension whilst keeping the threaded and top section diameters the same.

L1	3mm
L2	21mm
L3	6mm
D2	5.65mm

Table 2-Table of new bolt dimensions

With these the individual stiffnesses were calculated so the writer could calculate the overall stiffness to be 203.2214N/mm. This is a significant reduction in stiffness but the FSF for this design was 1.17 which is very close to the original. To add to that because of a reduction of volume of around 364.51mm³ there was a reduction of mass of around 2.84g per bolt. This is however assuming that the young's modulus was the same between the 2 materials. It also means that the tightening torque is reduced due to the reduced stiffness of the bolt.

6 Stress on Crank

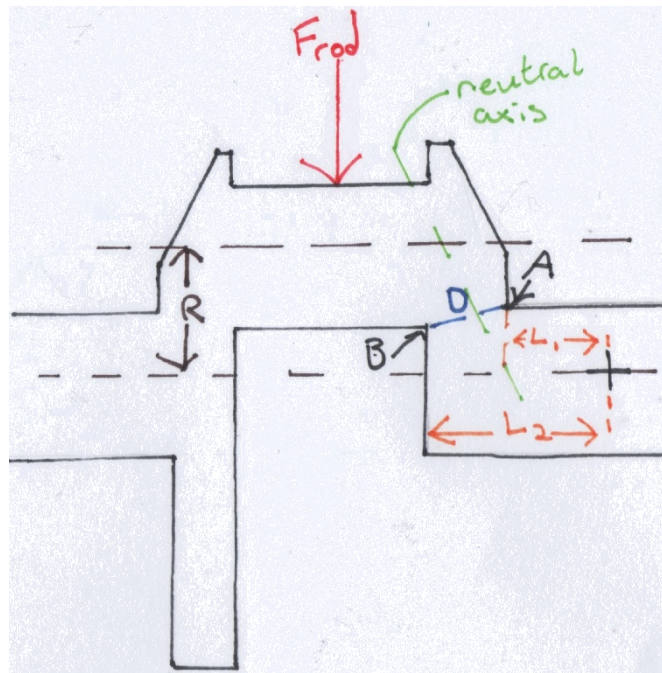
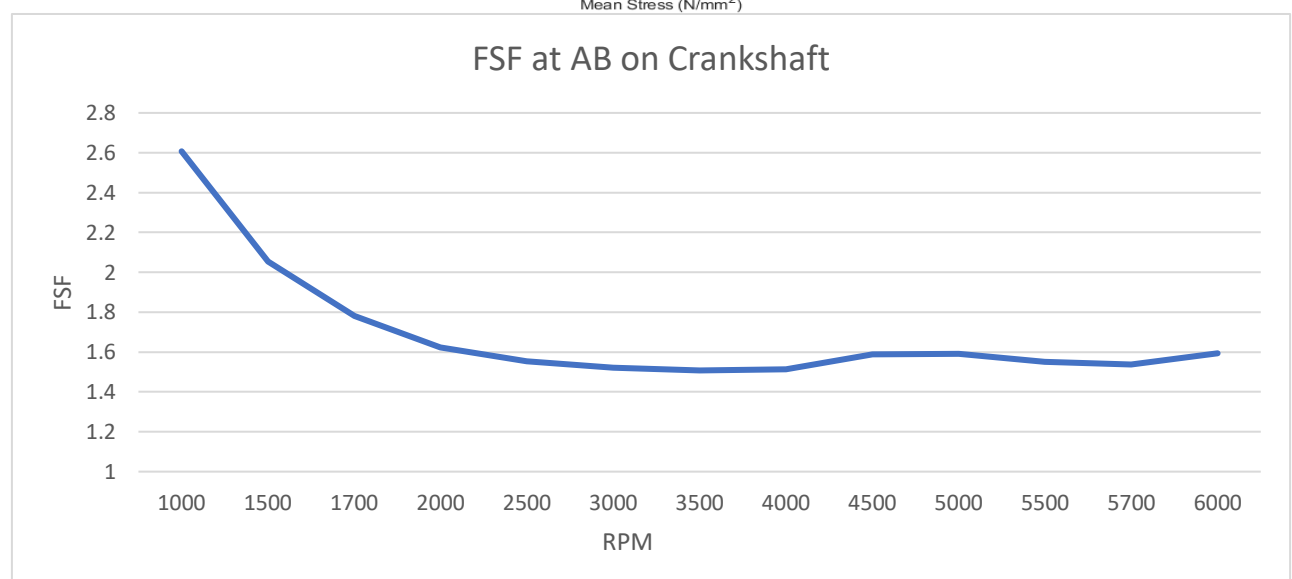
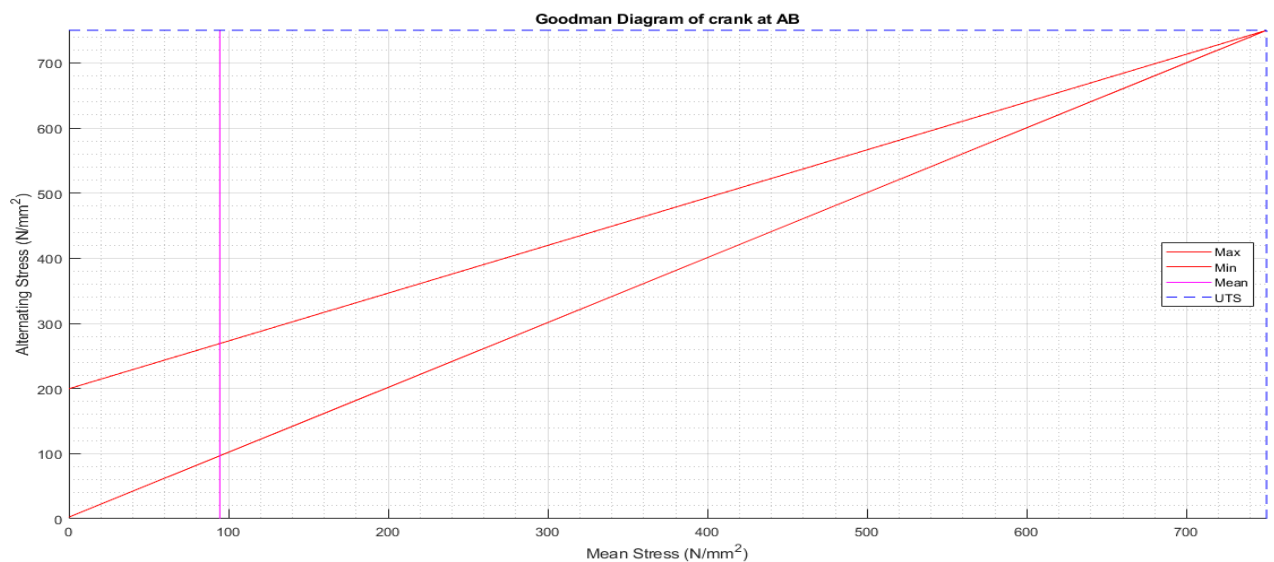


Figure 17- Diagram of Crankshaft



Figure 18-Image of broken crankshaft

The author measured the dimensions and then calculated the stress at point B. He did this by calculating F_{rod} which would be halved on the bearings. He then calculated the bending moment to then calculate the bending stress. Similar to the gudgeon pin there is also a shear stress, so the author then calculated this to then get the combined stress. From these the maximum, minimum, mean and range was found at each RPM. A Goodman diagram was then used to work out the FSF at that section.



The author found that the FSF never goes below 1 which is a good as that suggests that the calculation was correct. Another support to this is that the FSF is similar to other values calculated on other parts.

7 Conclusion

In conclusion the author found that ford as a minimum has a fatigue safety factor of 1.5. This does increase on other parts, but this may be an error in the authors calculations. If the author was to do further research into this matter, then perhaps he would run an FEA solver to validate the calculations and the stress concentrations that he used.

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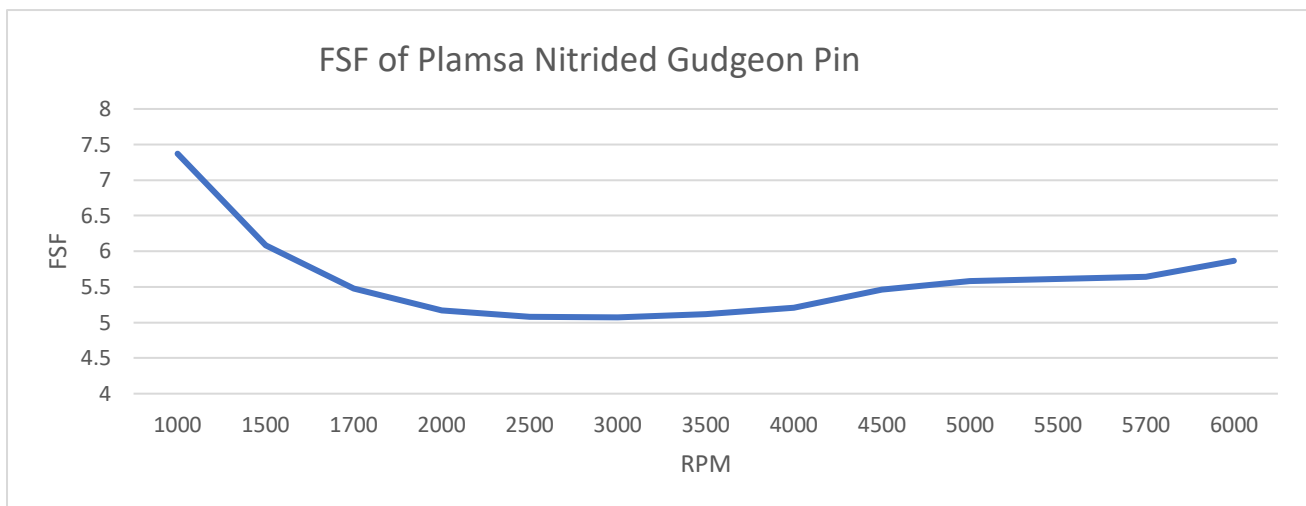
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9 Appendices

9.1 Pulse Plasma Nitriding

The author also made a small investigation pulse plasma nitriding. This is where the surface of a metal is hardened by charging the surface ions and forming a nitrogen rich layer (Bodycote, 2019). This means that a higher fatigue life can be reached or very little effort. The effect varies for case by case, but the author found a value of around -800N/mm^2 (Soleimani, et al., 2012). This meant that the range was the same, but the mean stress reduced significantly. This then meant that the FSF increased greatly.

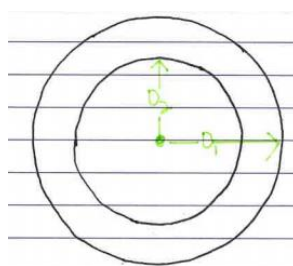


9.2 Different Tightening

The writer investigated tightening the bolt to 18Nm then an extra 35° . This was to see how tightening the bolt in a different manner can effect the static stresses on the bolt. The author first found what the initial stress was when tightened to 18Nm and then calculated the stress that occurred when that extra 35° was added. The author found that the total static stress would be 1142.62N/mm^2 . This is not only well into yield but was also slightly higher than the UTS. This suggests that if that method was used then the bolt would snap rather than tighten any more.

9.3 Equations

$$\text{Mass of Pin Sticking out of con rod} = L \times \pi \times (R_2^2 - R_1^2)$$



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

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

$$FSF = \frac{\text{Allowable Range}}{\text{Actual Range}}$$

$$\text{Polished EL} = UTS \times 0.45$$

$$\text{Final EL} = \frac{\text{Polished EL}}{K_T}$$

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

$$I = \frac{\pi D^4}{64}$$