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Calculations on Piston, Connecting-Rod and Crankshaft
Assemblies.

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1. Introduction.

This report has been constructed with the objective of outlining how to use basic relationships between stress, strain and moments, the fatigue safety factors of ICE internal parts can be estimated.

The parts considered within the investigation were the Gudgeon pin (wrist pin), the connecting rod and then the crankshaft within Ford Motor Company's 1600cc GTDI EcoBoost internal combustion engine. These parts are the areas of highest concern, given their nature of operation, being consistently under stress.

Having in-cylinder pressure data provided and using the equations governing the inertial force within the engine, the load experienced on these parts is easily found and then used with idea that these internals all operated within the elastic limit, it could be seen that; $E = \frac{\sigma}{\epsilon}$.

This equation is the fundamental for determining the limits of stress that the individual parts are subjected too. And by use of a 'Goodman' diagram, the "actual" limits of stress for the materials can be found, to therefore determine the 'safety factor' of the part.

Hence, the discussion of this theory will construct the bulk report and is to be analysed regarding its validity.

Objectives –

- Manipulate the provided data into a format for which the maximum forces on each part can be obtained.
- Provide an outline of the potential safety factors achieved by Ford Motor Company for their; Gudgeon pin, Connecting-Rod & Bolts and the Crankshaft.
- Demonstrate how to build a relationship in terms of forces and their effective areas to determine the maximum stress seen.
- Put into context, how using stress concentration diagrams, the geometry of the parts can cause issues for controlling fatigue.

2. Principles.

Within an internal combustion engine, the parts are subjected to a magnitude of different stresses and therefore strains. This cyclic stress issue, coupled with potential stress concentrations on the surface of the part, can and will cause a fatigue failure of the part (Arijit, 2014).

With the cyclic bending of parts being the most common reason for failure, it's clear that the gudgeon pin, joining the connecting rod and piston together (shown with the diagram below), is going to be the area of most concern.

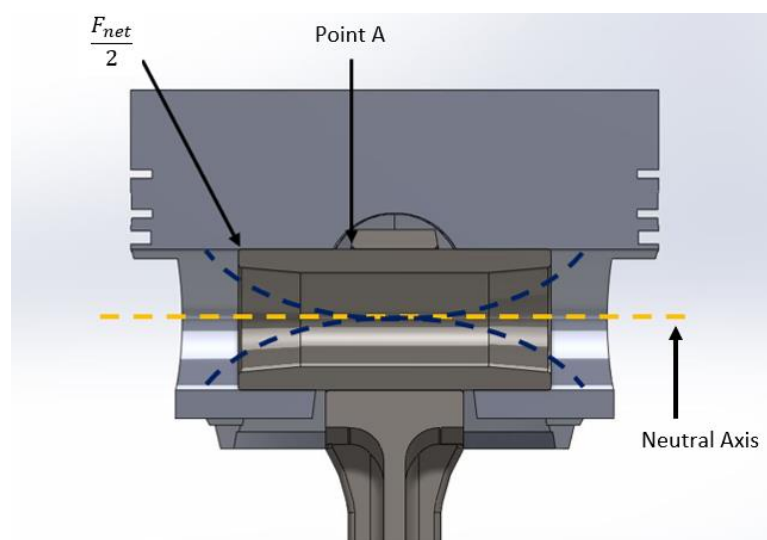


Figure 1 Depiction of the connection between the pin, rod and piston.

The 'point A' is where the bending moment is going to cause large stresses. Due to this, the study was done such that, considering the one side; the net force acting is equally split either side, the force is only acting on the one point of the pin, the difference in width from the top to bottom of the hoop on the rod is neglected and taken as a constant distance of 12mm (6mm from centre). This will be discussed in chapter 3.

The next section to be discussed is the connecting rod, designed to transfer the reciprocating motion through to rotatory at the crankshaft. Within this component, the forces seen are essentially only in the realm of tensile and compressive, during the exhaust and combustion cycles respectively, because of its pin jointed situation.

Meaning that the chance for fatigue failure isn't as high in comparison to the gudgeon pin, but again due to cyclic nature of stresses and the introduction of geometrically induced stress concentrations, fatigue is possible.

Fatigue failure testing is expensive and time consuming, because when looking at the experimentally informed graph;

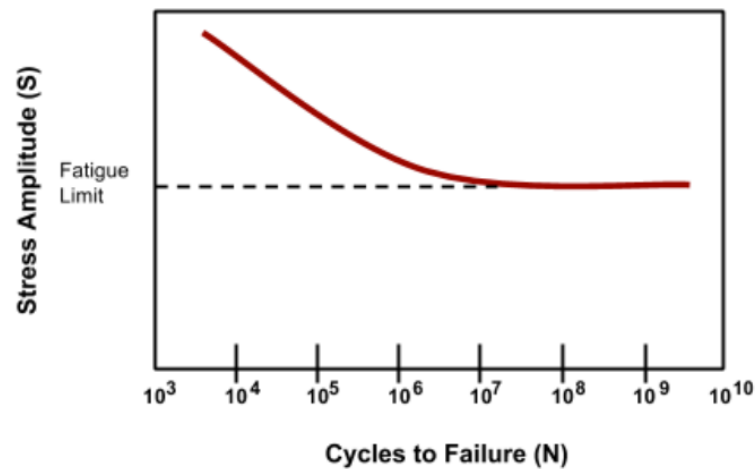


Figure 2 Stress Vs Number of cycles failure curve (Inspectioneering, n.d.).

The point at which the part is expected to ‘never fail’ is at 10^7 cycles, which with respect to a physical testing procedure, isn’t easy to get too. Therefore, by being able to predict the lifetime of the part is key for design, so using simple equations to estimate this can be a cost-time effective procedure, as it to be shown within chapter 4.

Connecting rod bolts are subjected to a considerable amount of stress, however the reality of the range of stress isn’t anywhere near that of the rod or pin, because of the initial compression state of the bolt when installed. The real issue and objective with the bolt is to pre-stress the bolt strategically, in order to position it within the plastic region of the material.

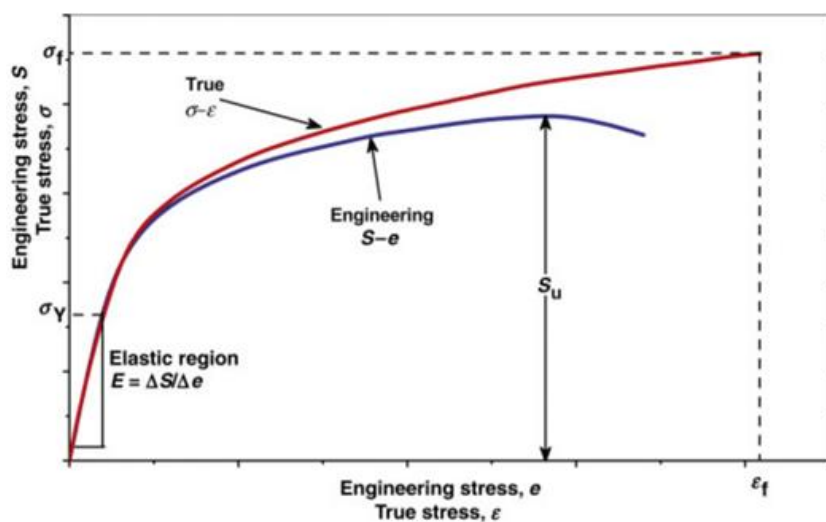


Figure 3 Typical monotonic stress–strain curve (McKeen, 2016)

The target area is shown in the figure above as the points between which the yield point is surpassed and the ultimate tensile point (UTS), denoted as ' S_u ' hasn't been reached. The idea is that within this area, the material no longer follows the linear relationship between stress and strain, because of this, it's theorised that there is a point at which the gradient of the line is zero. With this being the case, the material has reached a point of infinite stiffness and could withstand any stress without a change in length.

Ford Motor Company, through experimentation found that to achieve this, they are to tighten their bolts in order of; 18Nm and then a further 35°. This allows for far more accurate resultant clamping force because the stretch seen by the bolt should remain consistent, regardless of frictional losses. And as a product of this, the pre-stress should be within a tolerable region every time during installation. Calculations carried out to demonstrate this will be in chapter 5.

The final section of this study was based upon the compressive and tensile stresses that the crankshaft experiences. As with all the other parts, there is a clear section to which is going to be the primary area of which the part may fail, in this case it's the section of the crank web between the main journal and 'big end' journal.

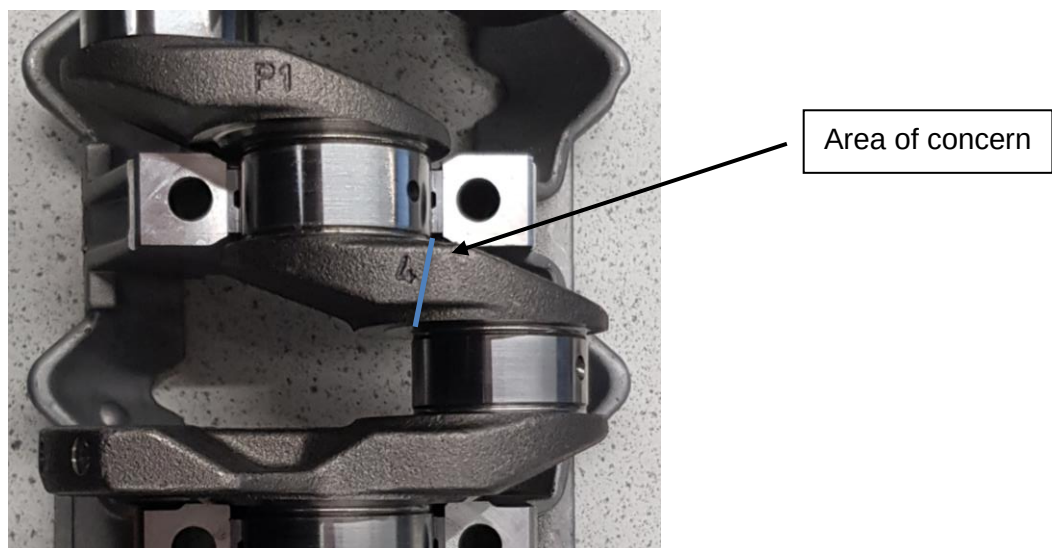


Figure 4 1600cc EcoBoost crankshaft with point of concern.

This area (highlighted in blue) is coupled with nearby stress concentrations that could cause issues for the lifetime of the part and may result in failure due to fatigue. So, again through a simplified situation of bending stresses taken from a single point, an investigation was conducted to obtain the estimated safety factor of the part.

3. Gudgeon pin.

As stated, the main stress concern for the gudgeon pin, is in the form of bending. This bending stress is also the main reason for fatigue failure in parts subjected to cyclic situations and is combined with a respective shear stress because of the joint nature, with the two independent parts at the two positions used in the study.

Starting with the initial characterisation of dimensions and masses, they were found to be;

Gudgeon Pin		
L_1	9.75	mm
L_2	5.27	mm
r_1	10.50	mm
r_2	7	mm
Overall length	46.50	mm
Second moment of area	7660.82	mm ⁴
Mass of pin out	0.0338	Kg
Youngs Modulus	2.05E+05	N/mm ²
Density	7850	Kg/m ³
UTS	670	N/mm ²
YS	435	N/mm ²

Table 1 Gudgeon pin properties.

For the standard engine the material of the pin was taken to be AISI 4130 steel normalised at 870°C (ASM Aerospace Specification Metals Inc., n.d.). Which, from the following calculations, can be shown to withstand the stresses acting upon it. It also has a diamond-like carbon coating (DLC) that will add protection to the surface from frictional wear and helps in initialising compression within the surface of the pin, but the extent of this is hard to predict and was therefore neglected.

The calculations done throughout this report, are done so using net force acting at the point, where; $F_{net} = F_{gas} - F_{inertia}$.

The gas force remains as constant for each area (direct from data) and the inertial forces are calculated based on $F = m \cdot a$. In the case of the gudgeon pin, the inertial forces are a product of the piston assembly mass above. With the piston assembly mass being; 266.17g.

Using the following equation for each case where the reciprocating forces are involved;

$$F_{inertia} = M_{rec} \cdot r\omega^2 \cdot \left[\cos(\theta) + \frac{r}{l} \cdot \cos(2\theta) - \frac{1}{4} \cdot \frac{r^3}{l^3} \cdot \cos(4\theta) \right]$$

(Nigus, 2015)

Under the combustion cycle the tensile and compressive force acting at each engine speed can be seen in the following graph;

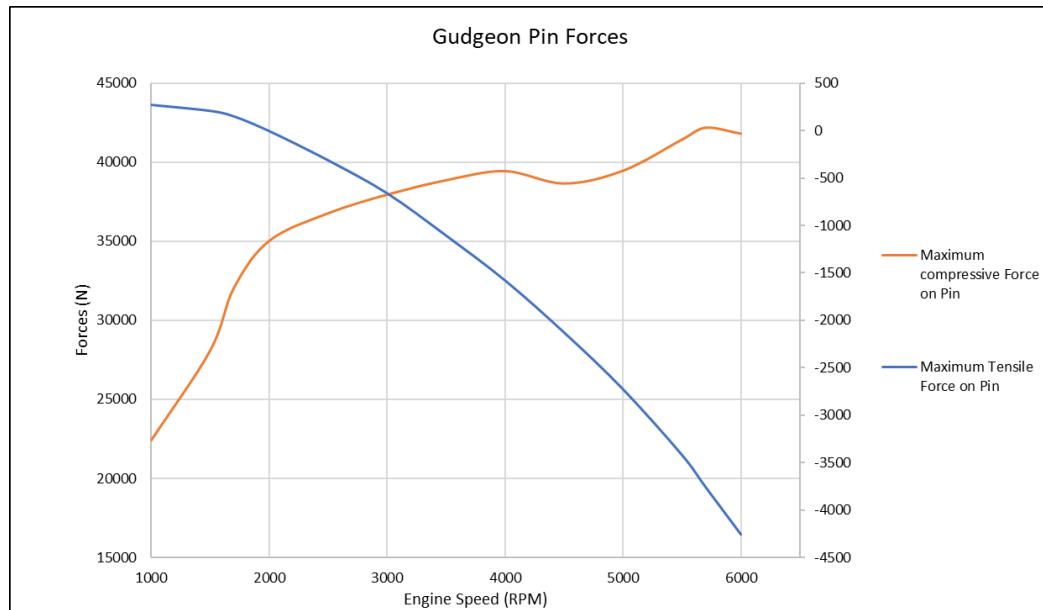


Figure 5 Forces acting on the gudgeon pin during combustion cycle.

When looking at tensile and compressive forces, it's considered that compressive are to be of a negative 'direction' and tensile being positive, because of the more destructive nature of tensile in terms of cracks forming. In the figure above, the values are opposite, this is due to the fact the study was done when looking at point A and this causes the normally compressive forces, to become tensile in this situation.

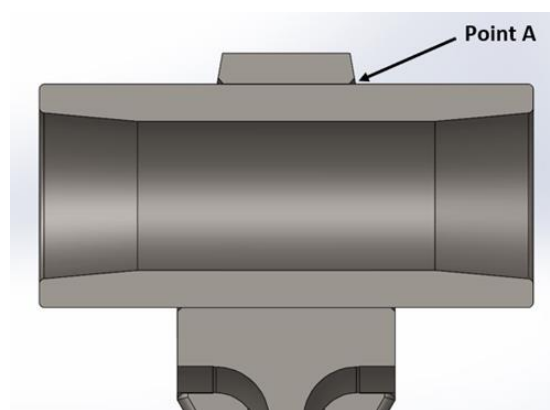


Figure 6 Point A on the gudgeon.

Bending stress can be shown to be equal to;

$$\sigma_B = \frac{M_B \cdot y}{I}$$

With M_B being the bending moment, y is the distance from neutral axis to the point of consideration (10.5mm) and I is the second moment of area.

To obtain the bending moment taken at point A during combustion, the two counteracting forces come from the net force acting clockwise (tensile) and the pin inertial forces acting anticlockwise (compressive), at distances L_1 & L_2 respectively.

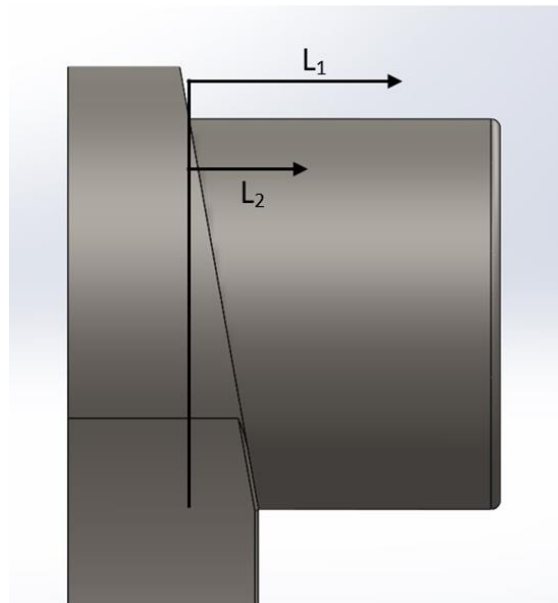


Figure 7 Depiction of the distances used for bending moments.

The overall moment acting on point A is therefore;

$$M_B = \left(\frac{F_{net}}{2} \cdot L_1 \right) + (Pin_{inertia} \cdot L_2)$$

With ' L_1 ' being the centre point of the piston face bearing on the pin, acting at 9.75mm from A. Then ' L_2 ' as the distance to the centre of mass from A, found using SolidWorks and is 5.27mm.

With the F_{net} being positive and $Pin_{inertia}$ being negative, the resultant moments across the engine speeds were;

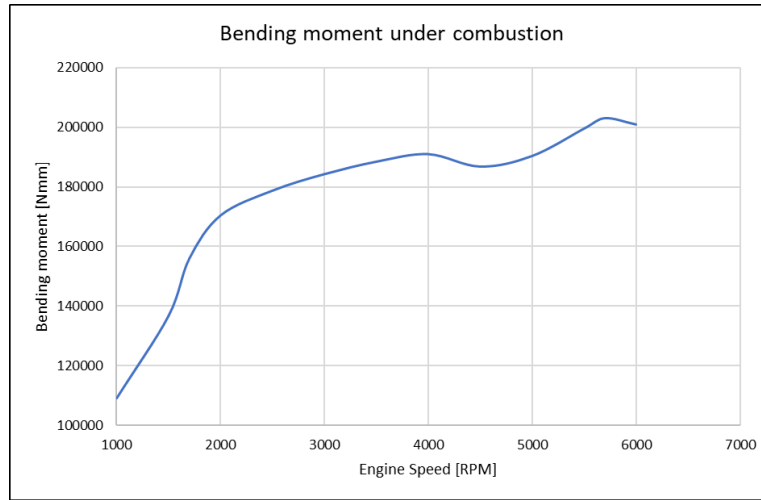


Figure 8 Bending moment during combustion.

Unsurprisingly it follows the same pattern as the gas force seen in figure 5, because of the massively overwhelming force being from the combustion expansion.

As with the exhaust stroke, the inertial forces acting on point A become much larger than that of gas. Using the minimum value of net force across one cycle as the maximum inertial force (occurring at TDC), in conjunction with the inertia of the pin itself, the bending moment becomes;

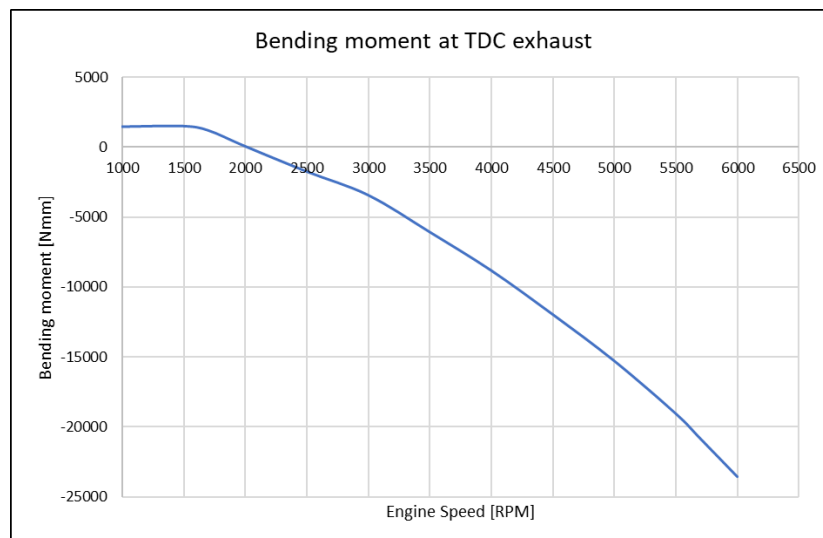


Figure 9 Bending moments at A for TDC exhaust.

Within the graph, the lower engine speeds [1000 – 2000] are showing as positive values, demonstrating the condition of the pin being tension, not compression. This is because the forces from gas pressure within the cylinder are still larger than that of the inertial forces, causing a tensile situation. But after this point, the inertial forces have begun to overcome the gas force and therefore causing the compressive moment around point A.

The last part of the bending stress equation is the second moment of inertia, this is shown as;

$$I = \frac{\pi \cdot (d_1^4 - d_2^4)}{64}$$

Where d_1 & d_2 are the outer and inner diameters of the gudgeon pin (21mm & 14mm).

The resulting bending stress is therefore;

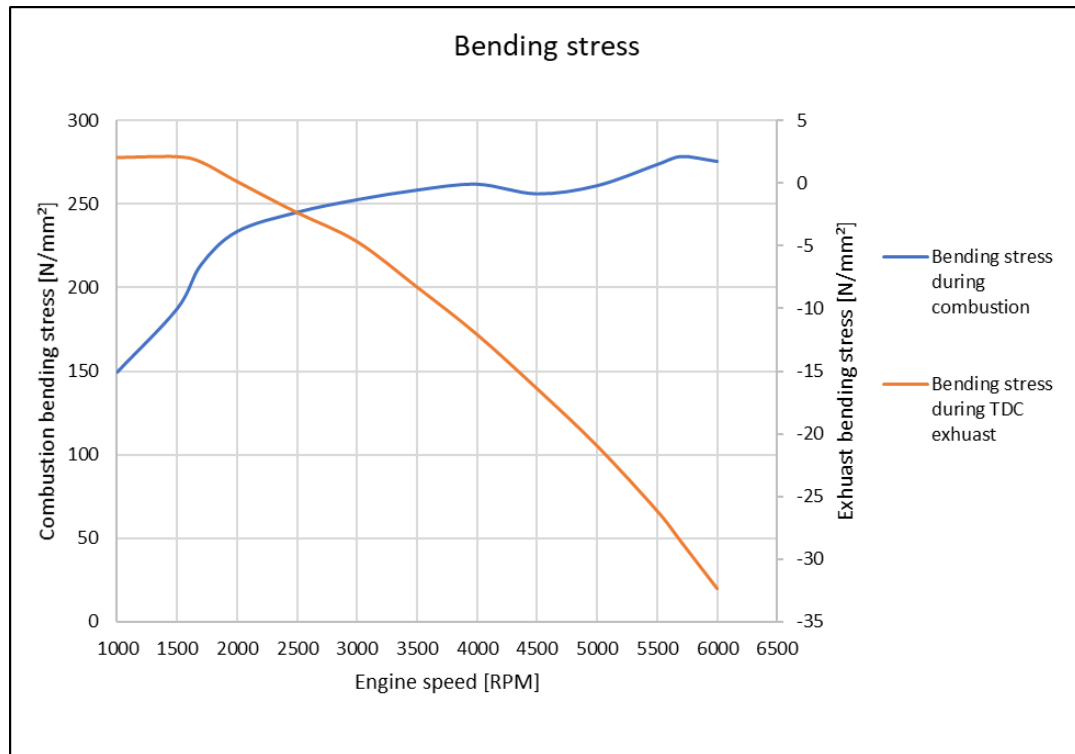


Figure 10 Bending stresses experienced by the gudgeon pin.

The total stress within the gudgeon pin is comprised of both bending and shear stress because of the unbalance in bending moments. In order to calculate the shear stress occurring through the range, the equation used is;

$$\tau = \frac{\left(\frac{F_{net}}{2} + Pin_{inertia}\right)}{\pi \cdot \frac{(d_1^2 - d_2^2)}{4}}$$

The shear stress at each situation can be seen below;

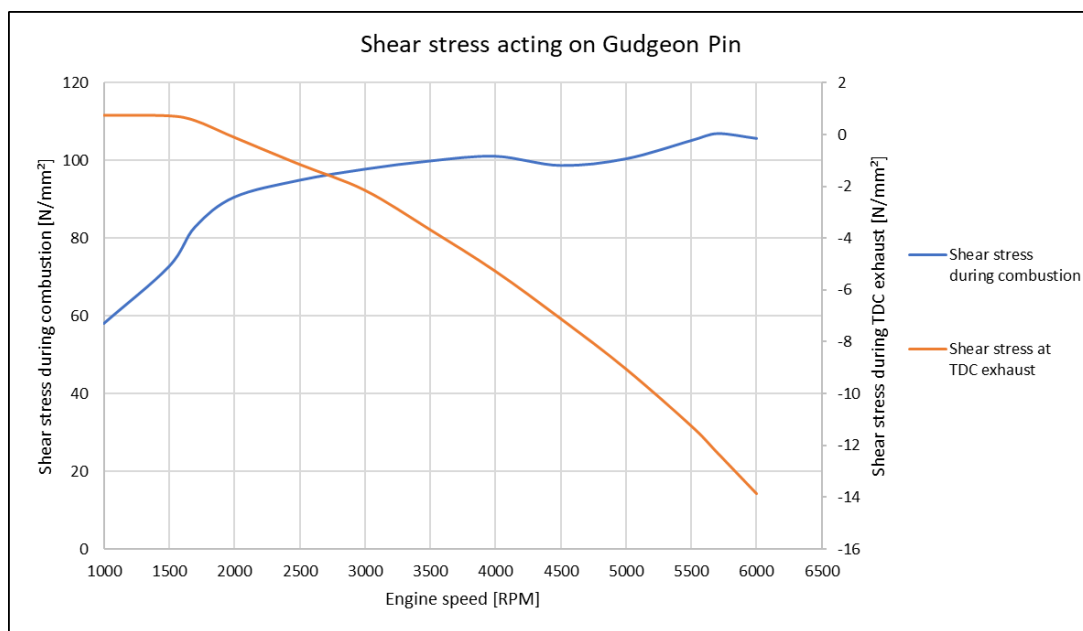


Figure 11 Shear stress on the gudgeon pin.

As is evident, the shear stress experienced is far less than that of bending and as expected, is around half of the bending stress.

This is because with the bending stress being a tensile (or compressive) situation, the final limit of the part is governed by the UTS. With shear stress, dependent upon the theory used, it can be said that the part will have a shear strength of; $\frac{1}{\sqrt{3}} \cdot UTS$.

(Christensen, 2019)

Therefore, given this relationship, its expected to be obtaining values for shear stress within the region of half bending.

The resulting stresses are calculated via;

$$\sigma_{res} = \sqrt{\sigma_B^2 + (2 \cdot \tau)^2}$$

The resultants work out to be;

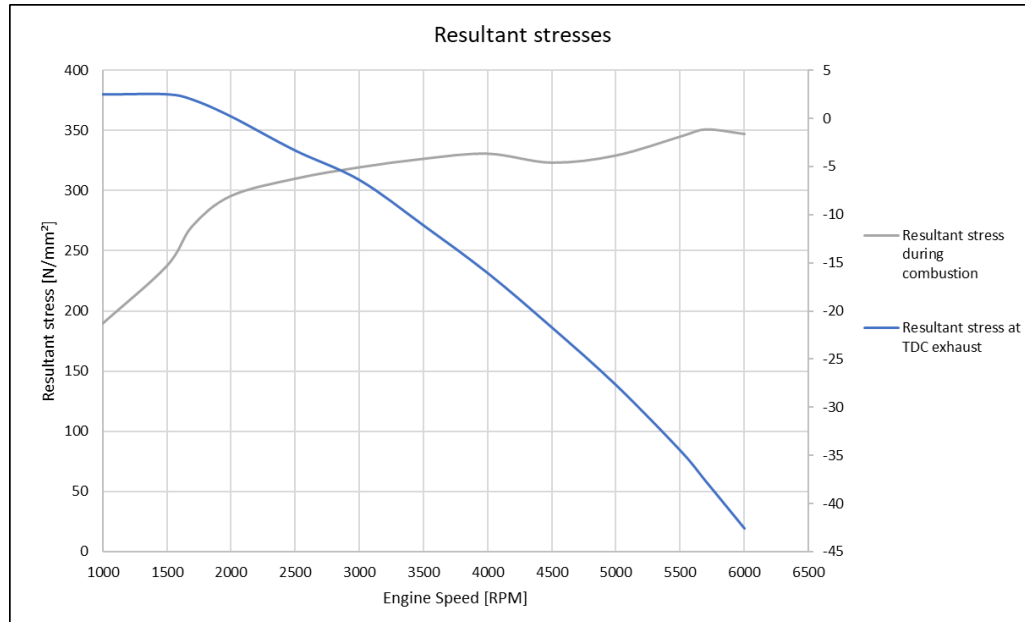


Figure 12 Resultant stresses on the gudgeon pin.

From here, the range of stress and mean stress occurring at each engine speed can be found and then used in conjunction with a Goodman diagram to determine the 'fatigue safety factors' of the part/s.

Range of stress being;

$$Range = \sigma_{max} - \sigma_{min}$$

Mean stress is;

$$Mean = \frac{(\sigma_{max} + \sigma_{min})}{2}$$

[Main notice when calculating the range and mean, is to be sure of the direction in terms of positive/negative being used, depending on the type of stress concerned, and being aware of the correct notation corrections.]

The range and mean stresses across the engine speeds were;

	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000
Stress Range on Pin [N/mm²]	186.93	234.88	268.72	295.43	313.31	325.81	337.65	346.82	345.05	356.99	379.51	388.68	389.76
Mean Stress on Pin [N/mm²]	95.98	119.97	136.34	147.95	153.35	156.54	157.74	157.36	150.83	150.80	155.27	156.64	152.30

Table 2 Range & Mean stress experienced by the pin for each engine speed.

These results show the clear reality of the pin being subjected to far more tensile stress situations than that of compressive and is unfortunately a component of which fatigue is a major concern, hence the importance of correlative safety factors.

Using a 'Goodman' diagram, the idea is being able to predict the "allowable" range of stress occurring at a given mean, all within the restrictions of the materials UTS and the situation dependent endurance limits, in terms of bending or tensile/compression.

Load Type	Steel
Tension/compression	0.30...0.45
Bending	0.40...0.55

Table 3 Endurance limit multipliers negating stress concentrations (BOSCH, 2002).

The endurance limits (fatigue limits) are situated as in both the positive and negative side of the x-axis because of the alternating stresses occurring. Throughout this report, the endurance limit multiplier was taken to be 0.45, for all situations discussed.

The Goodman diagram for the gudgeon pin is as follows;

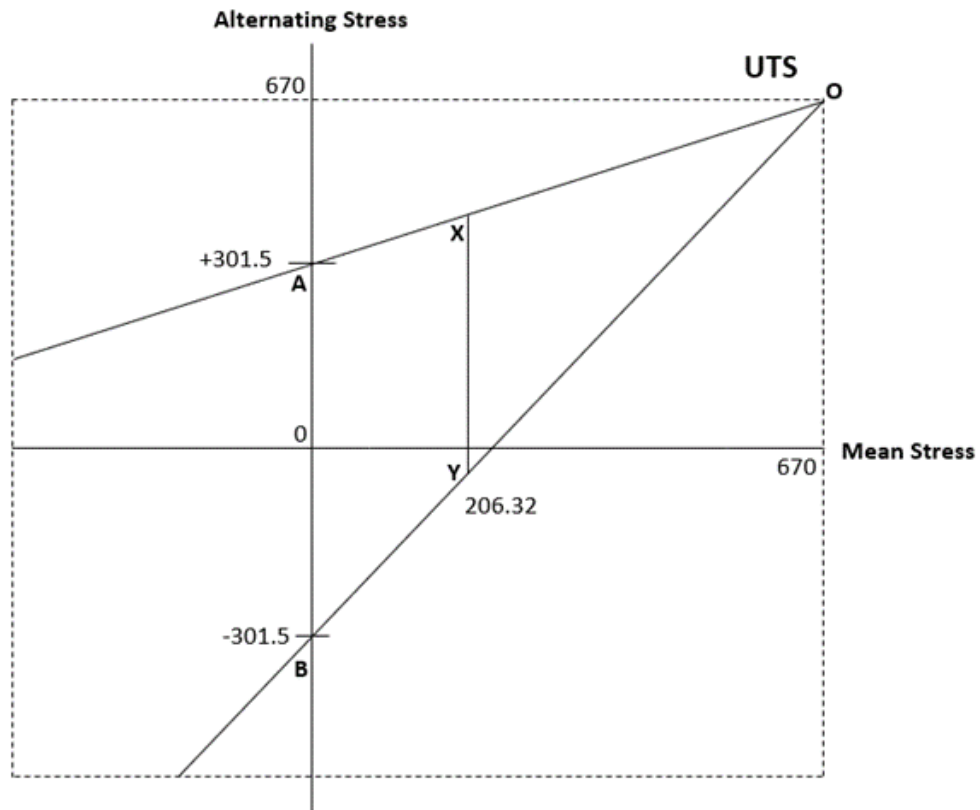


Figure 13 Goodman diagram for the AISI 4130 Gudgeon pin.

The diagram helps to give a visual representation of what the allowable range will be, however there is multiple trigonometric relationships that allows an equation to be developed. This can then be used to obtain the correct value for the allowable range of stress at a specific mean.

From the diagram, using the triangles created in respect to 'OAB' and 'OXY', the relationship is;

$$\frac{XY}{AB} = \frac{OX}{OA}$$

Following on to creating two additional triangles, 'OXC' & 'OAD', such that the graph becomes;

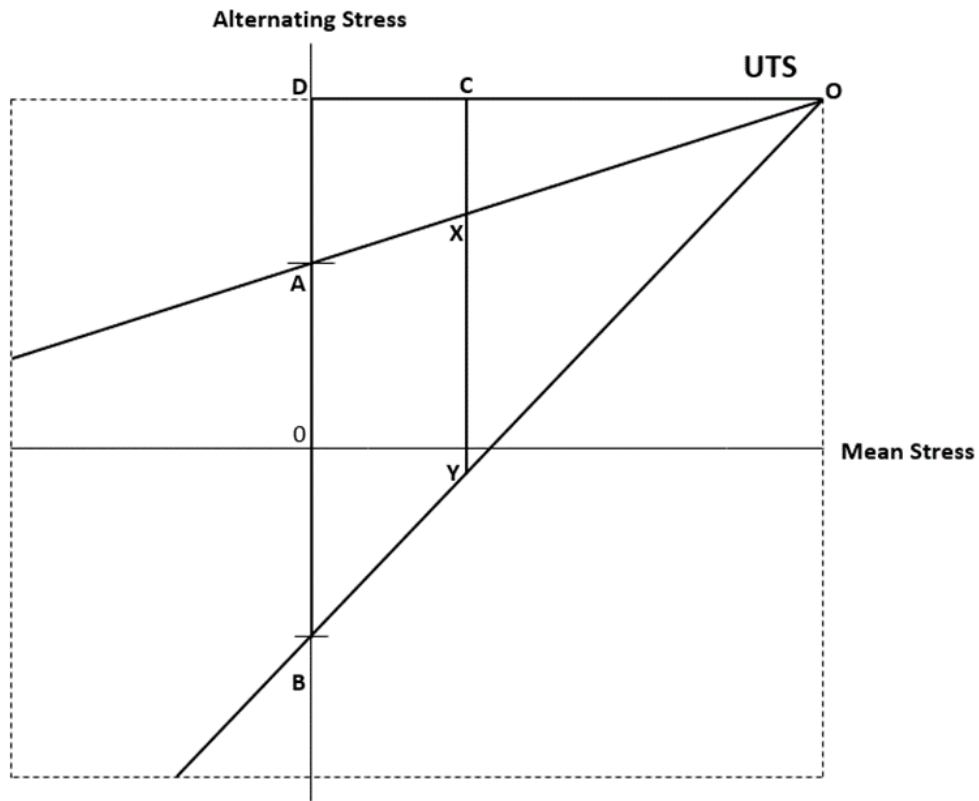


Figure 14 Triangle construction and relationships on the Goodman diagram.

The relationship becomes;

$$\frac{OX}{OA} = \frac{OC}{OC} = \frac{XY}{AB} = \frac{(UTS - mean)}{UTS}$$

Therefore, the final respective relationship is;

$$XY = AB \cdot \left(\frac{1 - mean}{UTS} \right)$$

And subsequently the equation to be used becomes;

$$XY = (2 \cdot EL) \cdot \left(\frac{1 - mean}{UTS} \right)$$

With 'EL' being the endurance limit of the material used.

Upon having implemented the equation for 'allowable range' and having found the maximum range and mean stresses seen by the part (consistently at 6000RPM), the fatigue safety factor is noted as;

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

The values found were;

Range at 0	Allowable	FSF
603	465.93	1.20

Table 4 Range at zero, Allowable range and the FSF of the AISI 4130 gudgeon pin.

And as can be seen, the fatigue safety factor [FSF] was found to be 1.2. Therefore, insinuating that the part shouldn't fail from fatigue. Because it's above 1 showing that the actual range of stress experienced is less than the predicted 'allowable' range. (This clarified by the clear operation of the engine and a static load test completed within SolidWorks [appendix 1.]).

Considering there are multiple applications of the 1600cc EcoBoost™ engine in the racing world, albeit with modifications, it's of interest to find out if the stock parts could be used under the high demand, high speed conditions. For this, a simple study was conducted using a linear extrapolation tool for the gas force and then a 5th order polynomial curve fit for the tensile, all within Microsoft's Excel software. The predicted values of gas force and inertial forces when the engine speed was taken to 9000RPM can be seen below;

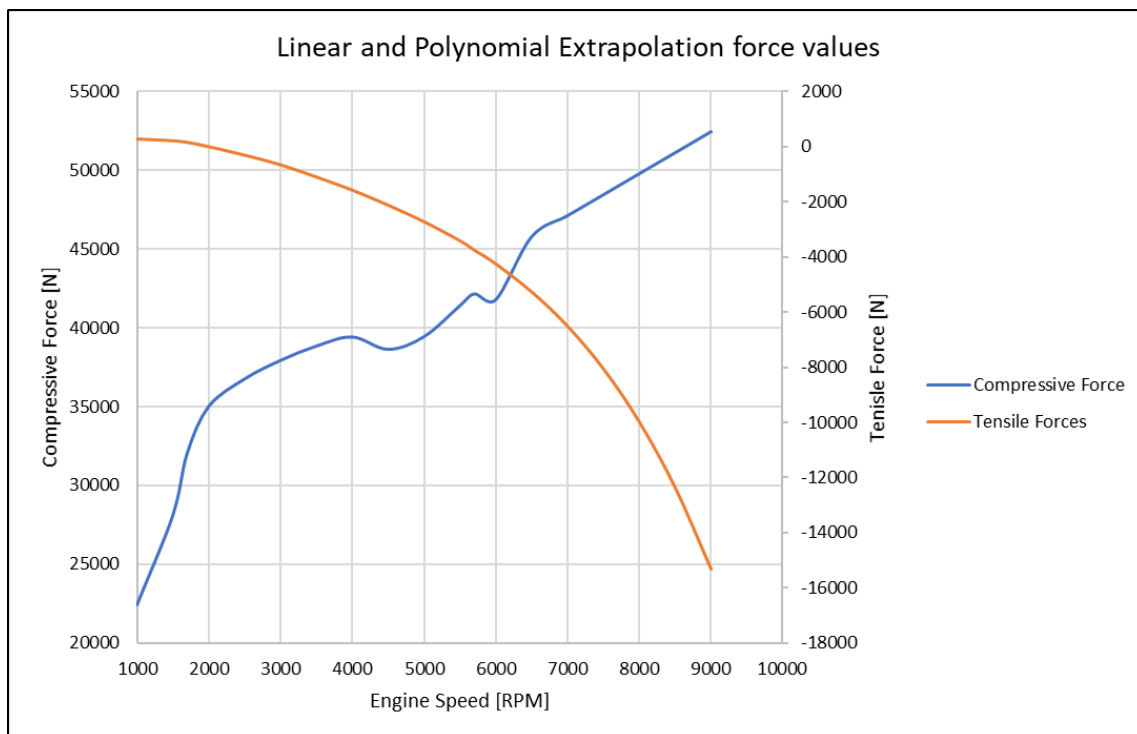


Figure 15 Linear & Polynomial Extrapolation force values on the gudgeon pin up to 9000RPM.

Because of the linearity seen within the compressive gas forces, it's questionable as to how realistic the values would be, given the real-life characteristics of combustion within a GTDI engine in respect to detonation due to excessively high initial cylinder pressures. Along potential choke and fuelling issues, both of which could be potentially overcome with a larger turbocharger and injectors (or potentially change their duty cycles).

However, the values were used regardless and considered as somewhat of a worst-case scenario, to which it was found that the AISI 4130 steel material presumed on the standard gudgeon pin would have eventually failed due to fatigue.

Using the AISI 4130 steel the values were found to be;

Range at 0	Allowable	FSF
603	448.18	0.96

Table 5 Values found for AISI 4130 steel gudgeon pin at 9000RPM.

The FSF was found to be less than 1. Showing that the predicted allowable range of stress the part can see is less than what it would see and therefore cause a fatigue issue of which failure will occur within the near future.

Hence, in this situation, if the material of the gudgeon pin was to be changed to H-13 tooling steel (ZRP, 2017). Material properties known to be;

H-13 Tool Steel		
Youngs Modulus	210	N/mm ²
Density	7800	Kg/m ³
UTS	1200	N/mm ²
YS	1000	N/mm ²

Table 6 H-13 Tool Steel material properties (AZoM, 2013).

Having implemented the new material properties, the FSF was then found to be;

Range at 0	Allowable	FSF
603	925.18	1.98

Table 7 Values found when using H-13 Tool steel for the gudgeon pin.

The fatigue safety factor has now been increased to just over two times that of the AISI 4130 and would therefore show no issues with being subjected to the potential stresses seen at 9000RPM and would be a brilliant fit for racing applications.

4. Connecting Rod.

Moving on from the gudgeon pin, the next link within the system is the connecting rod. Another major component that requires a high level of manufacturing because of the amplitude of stresses in both tension and compression (There is also potential for bending within the rod, however this is not included within this study). The rod itself is constructed of a powder forged steel, of which the material properties were assumed to be;

Powder Forged Steel		
Youngs Modulus	200	N/mm ²
Density	7600	Kg/m ³
UTS	760	N/mm ²
YS	520	N/mm ²

Table 8 Con-Rod powder forged steel material properties (GKN Sinter Metals, 2018).

Three points across the rod were chosen to conduct a stress study; A, B & C, relative positions shown below;

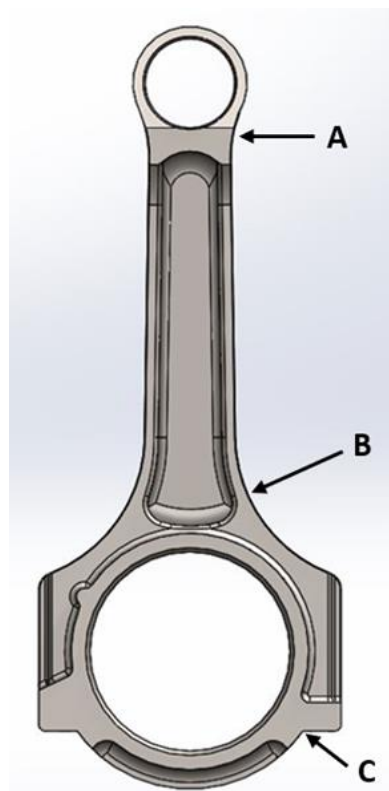


Figure 16 Depiction of the points across the connecting rod.

Given the pinned joint situation that the connecting rod is subjected to, the main forces that could cause an issue are the tensile forces occurring from the inertial loads induced by the reciprocating and rotational masses. Along with the compressive force from the gas pressures, both of which are now to only be look at, as if the situation is exclusively acting normal to the points, i.e. tension/compression not bending.

The inertial forces acting at point are calculated using the same principles as previous stated, but now including the mass of gudgeon pin & the small amount above the point. Making $M_{rec} = 363$ grams. The tensile and compressive forces acting on A were found to be;

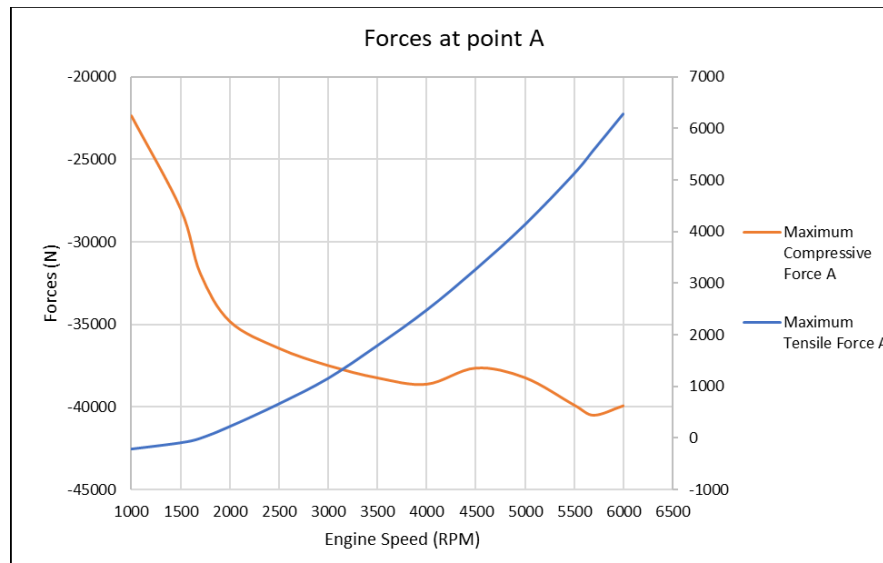


Figure 17 Forces acting on A.

The main note with these forces, is the change in direction (positive and negative). As with the previous stress, the gas forces were the tensile and inertial causing the compressive. But the now the gas forces have become the compressive, hence the negative direction on the values, due to the situation of the study now being only tensile/compressive and not bending. Therefore, there isn't just a single point being considered, it's now an effective cross-sectional area.

So, the stress seen is equated using;

$$\sigma \left[\frac{N}{mm^2} \right] = \frac{Force [N]}{Area [mm^2]}$$

The effective area was found using the CAD model on the connecting-rod built in Solidworks, found to be;

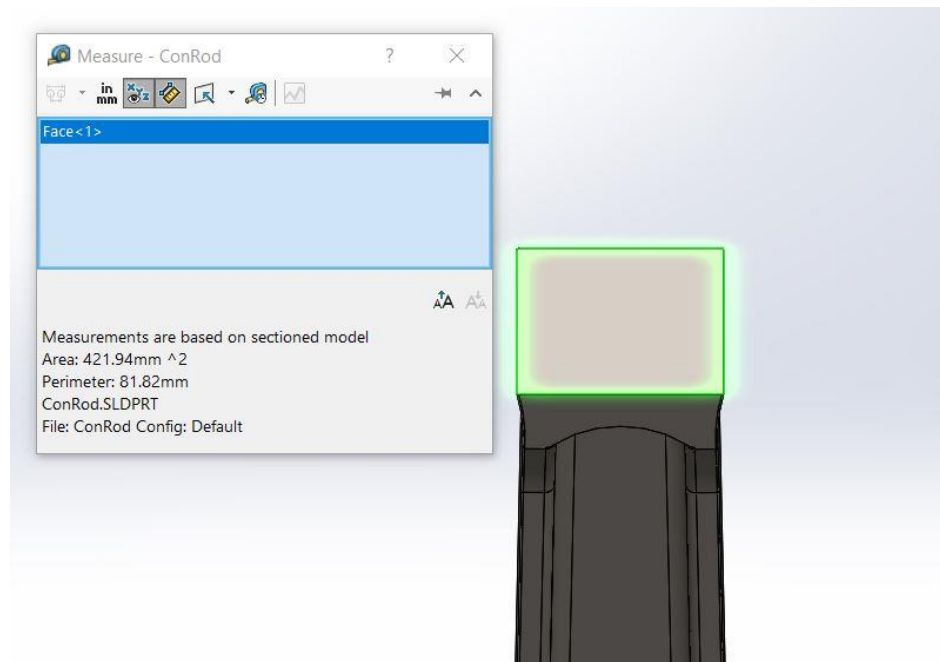


Figure 18 Area of A.

The range & mean calculated are shown in the table below;

	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000
Stress range at A [N/mm²]	52.45	66.11	75.59	83.04	87.95	91.60	94.88	97.43	96.99	100.41	106.68	109.25	109.56
Mean stress at A [N/mm²]	-26.75	-33.29	-37.83	-41.01	-42.42	-43.07	-43.21	-42.85	-40.75	-40.40	-41.20	-41.39	-39.89

Table 9 Range and mean stress for point A.

Then, again implementing the trigonometric relationship equation discussed previously, the fatigue safety factor for this point was;

0 Range	Allowable	FSF
684	719.90	6.571

Table 10 FSF values for point A on the con-rod.

The difference with the points across the connecting rod is the introduction of stress concentration factors, induced by the radii, fillets and points occurring across the rod.

Research conducted found that author Walter D. Pilkey wrote a book containing multiple scenarios effective stress concentration that will occur due to geometry, notches/grooves and fillets. All of which are cases dealt with, within this study.

For point A and subsequently point B (to be discussed), the representative graph was the same because of their geometry being of similar nature.

Below is the graph chosen for these points;

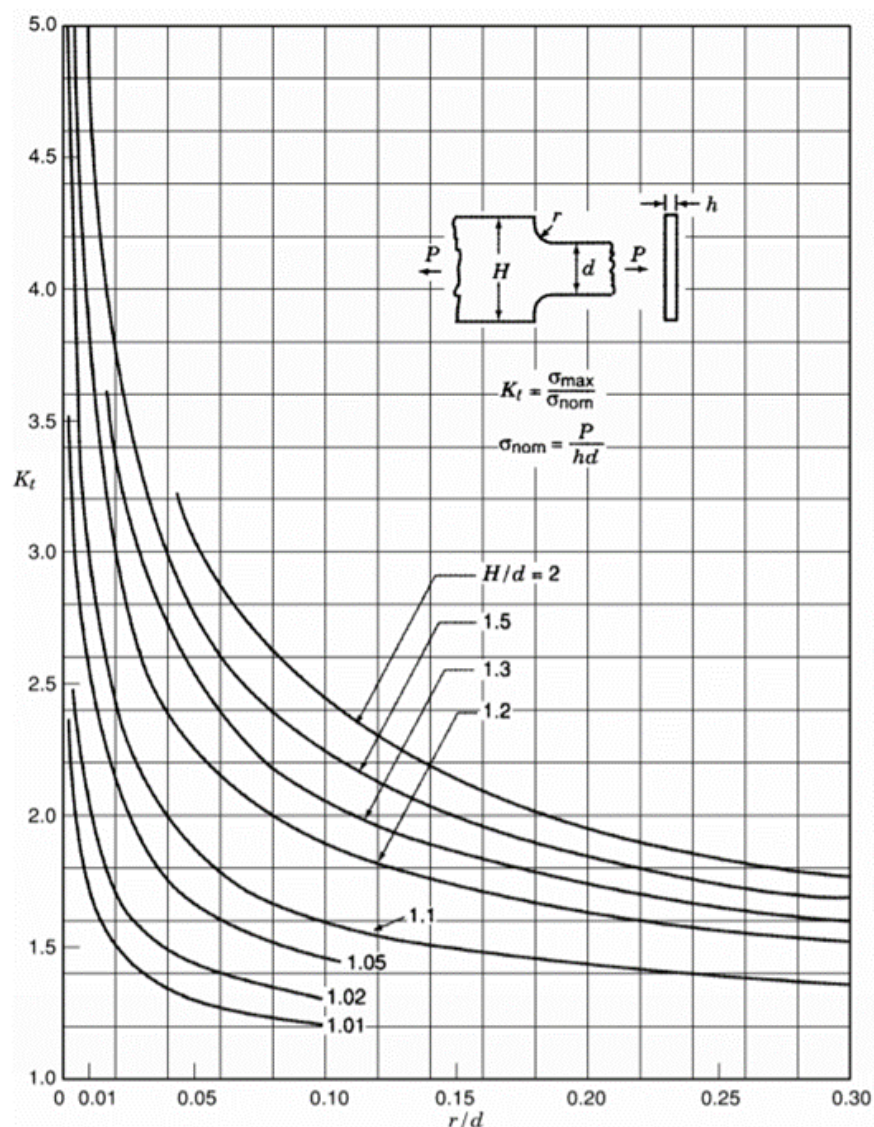


Chart 3.1 Stress concentration factors K_t for a stepped flat tension bar with shoulder fillets (based on data of Frocht 1935; Appl and Koerner 1969; Wilson and White 1973).

Figure 19 Stress concentration graph that was considered for points A & B (Pilkey, 1997).

To use this specific graph, the values of; ' H ', ' d ' and ' r ' were required, again referring to the CAD model of the rod;

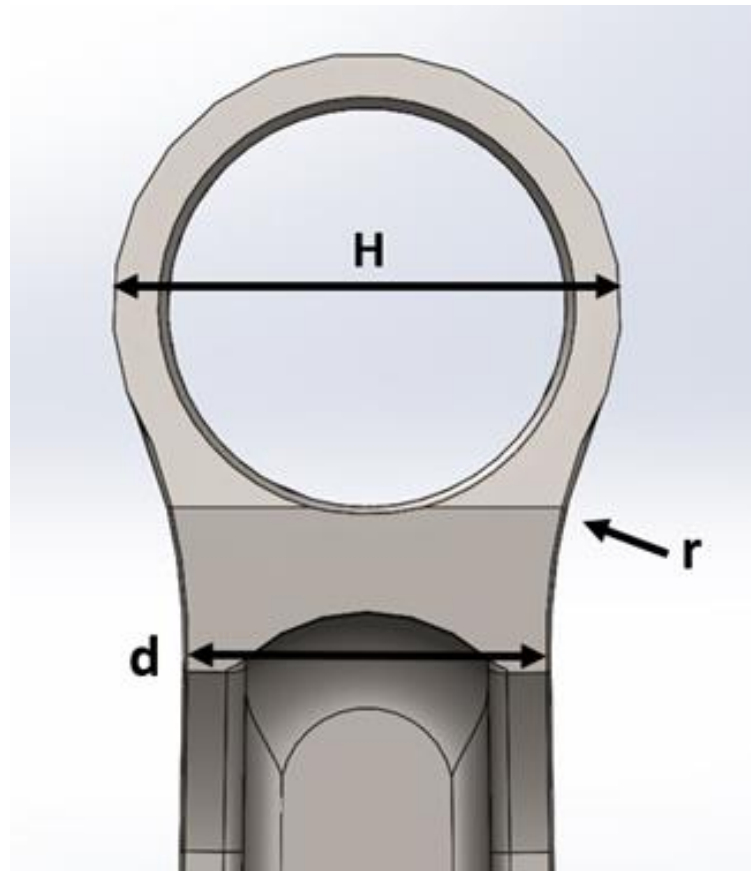


Figure 20 Highlighting corresponding subtext positions at A.

Values used		
H'	26.73	mm
d'	19.37	mm
r'	30	mm

Table 11 Values for stress concentrations at point A.

Using these to obtain values of, ' $\frac{r}{d}$ ' found to be, 1.38 and ' $\frac{H}{d}$ ' as 1.55, it's clear that within this situation because of the radius at the curve being so large, the value is somewhat trivial in terms of effect. Therefore, it was neglected from point A and the endurance limit wasn't affected.

With point B, the same style situation occurred, but now having larger forces acting on this point, because of the added mass within the inertia force equation ($M_{rec} = 473\text{g}$). And now because of the waste within the centre, creating the distinctive 'I' beam shape, the effective area was now substantially smaller than point A;

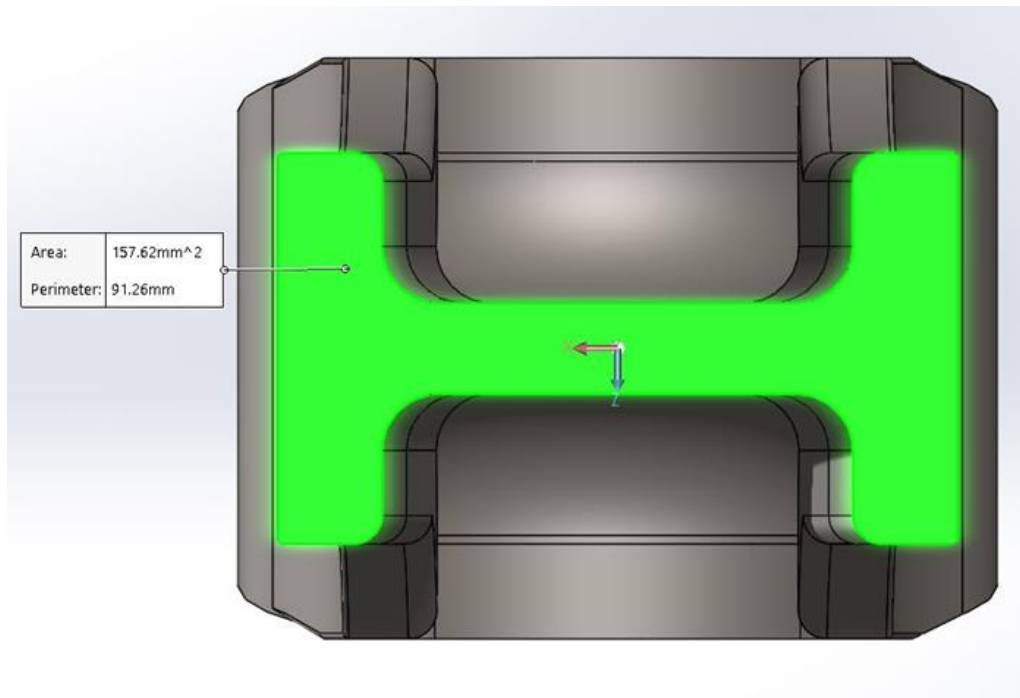


Figure 21 Area at point B.

The stress now occurring here were calculated as;

	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000
Stress range at B [N/mm²]	139.9	176.4	201.7	221.6	234.8	244.6	253.6	260.5	259.7	269.1	285.7	292.5	293.3
Mean stress at B [N/mm²]	-71	-88	-99.8	-108	-111	-112	-111	-108	-101	-98.32	-98.28	-97.87	-92.44

Table 12 Range & Mean stresses at point B.

When looking at the difference between the stress range occurring at A in comparison to B, the values have over double because of the massive reduction in cross-sectional area of which the force is being put through. And although there was more mass added in the effective inertia, there isn't much change or at least enough to be the cause of this great increase. Shown within the following figure;

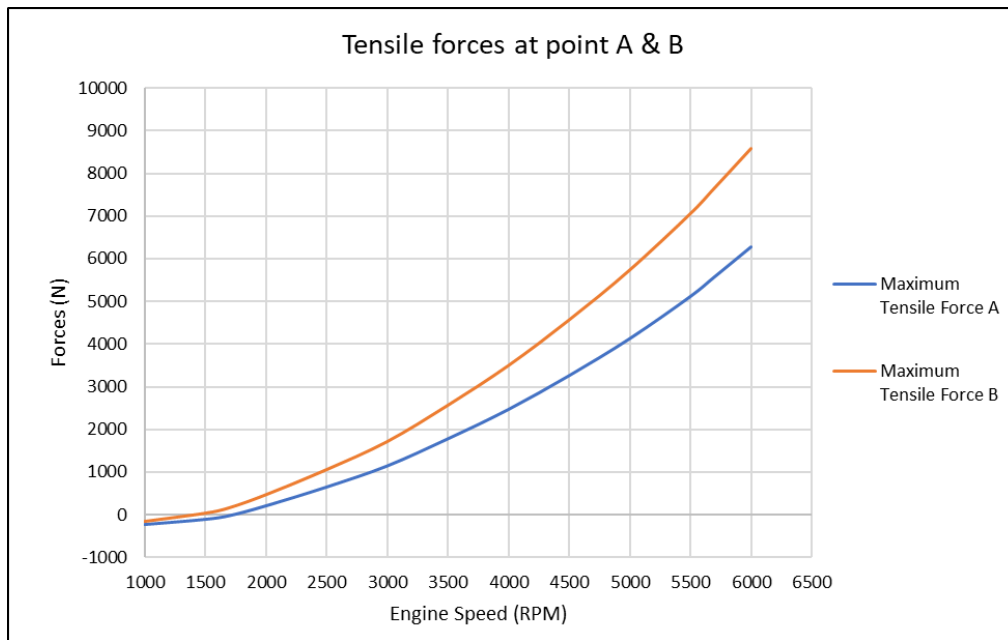


Figure 22 Tensile forces due to inertia at points A & B (TDC exhaust).

It's clear that although the extra mass has most certainly added to the tensile forces on point B in comparison to A, but they most certainly aren't of a magnitude of double the forces across A. Clarifying that the area is the main cause for the issues here, but because of the material properties of the rod and the geometric gains created by the 'I' beam used, there isn't an issue with fatigue.

This is again backed up by the calculated FSF;

Range at 0	Allowable	FSF
684	767.20	2.615

Table 13 FSF values for point B.

It's much smaller than the FSF at point A, but still higher enough to insinuate the idea of no possible fatigue failure issues.

As stated, given the geometry of the rod at point B, there is a possible stress concentration to be induced. However, with the values of; ' $\frac{r}{d}$ ' at 1.64 and ' $\frac{H}{d}$ ' as 3.33 it's clearly another trivial value that can for one, not be found from the graph and would be very close to one, having no affect on the endurance limit.

[Refer to appendix 2]

The third and final area on the rod was point C, situated on the connecting rods bearing cap. This area is relatively small regarding cross-sectional area, along with many intricate notches and grooves across the faces adding to multiple stress concentrations.

For example;

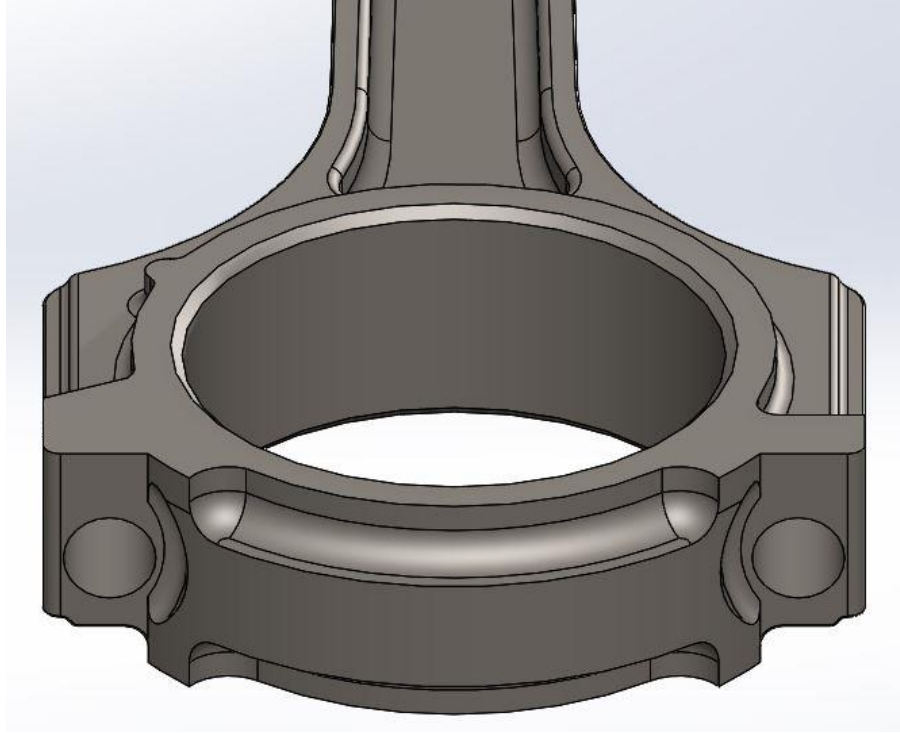


Figure 23 Connecting rod bearing cap.

The major change for point C, is the addition of both mass and rotational inertia. Because of this, the equation for inertia became;

$$F_{inertia} = \left(M_{rec} \cdot r\omega^2 \cdot \left[\cos(\theta) + \frac{r}{l} \cdot \cos(2\theta) - \frac{1}{4} \cdot \frac{r^3}{l^3} \cdot \cos(4\theta) \right] \right) + (M_{rot} \cdot r\omega^2)$$

With $M_{rec} = 473g$, denoted as; $(\frac{1}{3} \text{ Conrod mass} + \text{Piston assembly mass})$.

And $M_{rot} = 161.9g$, denoted as; $(\frac{2}{3} \text{ Conrod mass} - \text{Bearing cap mass})$.

Another factor personal to point C, is that it doesn't experience (at least in this study) the compressive forces created during combustion. Hence, the maximum forces coming from the inertial effects of masses being accelerated & decelerated [change in direction] up to and at TDC exhaust. All calculations were done on the idea of symmetry, with $\frac{F_{net}}{2}$.

The area used for point C was also found using the CAD model in Solidworks;

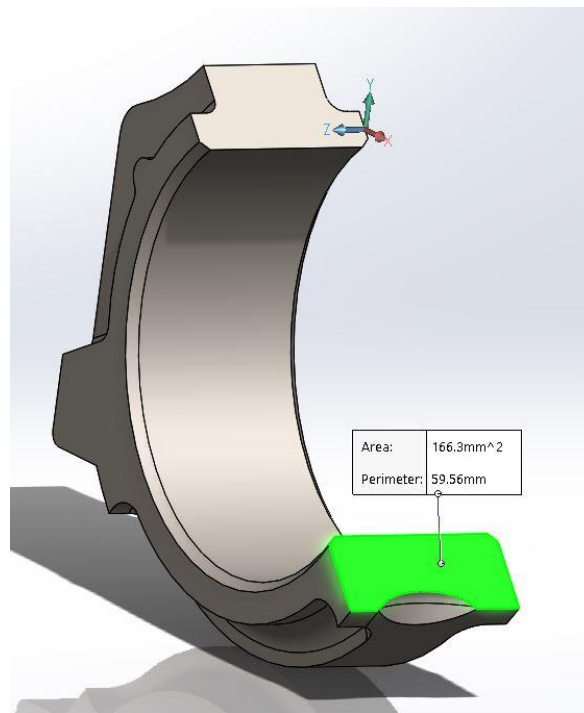


Figure 24 Effective area at point C.

The forces calculated showed the same exponential trend as before;

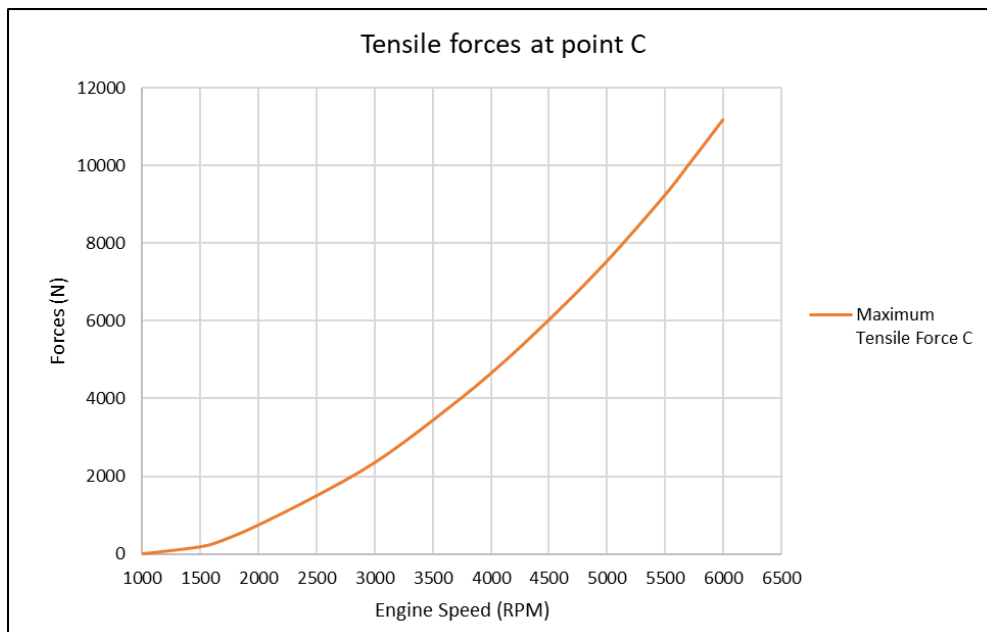


Figure 25 Tensile forces at C.

Showing a consistent increase in maximum force as the mass is increased. However, in this case, because the bearing cap is not affected by the compressive forces induced by the gases within the cylinder, this graph starts from 0 and continues up. When studying the previous graphs of points A & B, it can be seen that they both start within the negative range [1000 – 2000RPM], this is because the inertial forces created have yet to overcome the resistive gas force, therefore they remain in a compressive state. But this isn't possible at point C, hence at 1000RPM, the effective force is zero.

Again, because of the lack of compressive forces, the range on mean stress is far less than the previous points because the only force is brought about by inertia, of which is nothing in comparison to the gas forces. Therefore, the range of stress is simply the stress induced by inertia and the mean is half range;

	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000
Stress range at C [N/mm²]	0	0.62	1.21	2.50	5.02	7.87	11.51	15.55	20.16	25.24	30.93	33.49	37.43
Mean stress at C [N/mm²]	0	0.31	0.60	1.25	2.51	3.93	5.75	7.78	10.08	12.62	15.47	16.75	18.71

Table 14 Range & Mean stresses at C.

Obtaining the fatigue safety factor for point C follows the same procedure as before, but this time there is a stress concentration factor that certainly needs to be accounted for.

After research, again in Walter Pilkey's book, a suitable graph was chosen based on the idea that the shape incurred by the filleting and cut made around point C, was that of a hyperbolic curve. Thus, the graph used for this was;

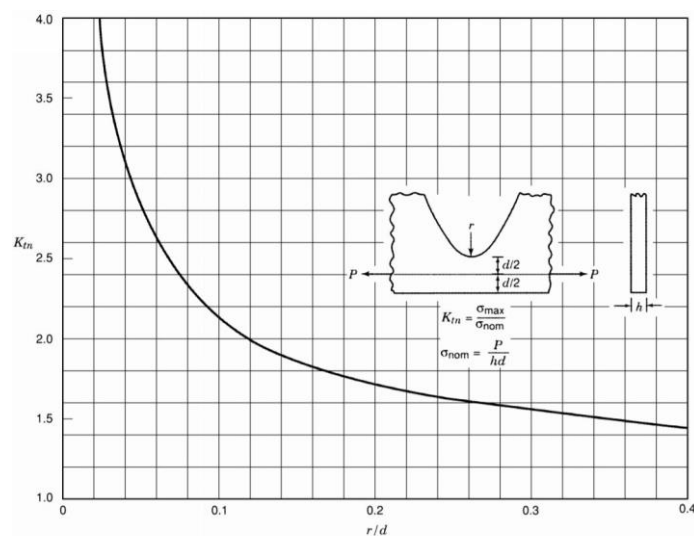


Chart 2.8 Stress concentration factors K_{tn} for tension loading of a semi-infinite thin element with a deep hyperbolic notch, tension loading in line with middle of minimum section (approximate values; Neuber 1958).

Figure 26 Stress concentration graph used for point C (Pilkey, 1997).

The reasoning for this graph is because when studying the model, the shape at which point C demonstrates a hyperbolic curve, of which the stresses passing through this area are travelling along the plane shown in the graph, with the effective distance of 'd'.

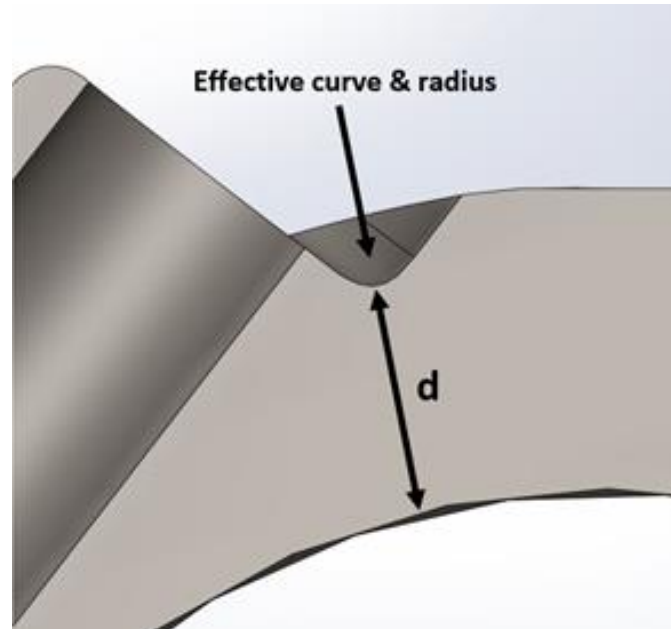


Figure 27 Reasoning for stress concentration graph choice at C.

The radius at the centre of the curve is 2mm and the distance from there to the inside of the rod 'd' is 6.8mm. Causing ' $\frac{r}{d}$ ' to 0.2941, hence the stress concentration factor for point C was taken to be 1.55. Although it was thought that the situation would be worse than it seems, because the FSF was found to be;

Range at 0	Allowable	FSF
441.29	430.42	11.5

Table 15 FSF values for point C.

With a value this high it shows that there shouldn't be an issue within this section of the cap and despite the notching, the result isn't too dramatic. But in reality, there will be multiple forces going through this area than those used in this study, to which could cause a fatigue issue. The other point is that there is a curve across the other direction of the curve used, meaning the stress concentration experienced at this point could be far higher than first thought. Because of the complicated nature of the geometry and mathematics, further investigation using a computational method would be required. It could also be done experimentally, but this will be difficult and costly.

5. Connecting Rod bolts.

The bolts play a major part within the construction and to be frank are the part that will most likely fail because of the small area coupled with the stress concentrations occurring from the threading.

The bolts share their load with the rod itself and can be designed to reach a point of which they won't cause any issue, because they're installed to a point past yield which will change the material properties to state of phenomena in that the material becomes very soft and flexible, whilst easily withstanding the alternating loads [refer to appendix 3].

To understand how the bolt and connecting rod share the load, the ratio of stiffness is required.

Based on the assumption of operation within the elastic region of a stress Vs strain graph meaning that the materials remain governed by Young's modulus, stiffness can be found using;

$$K = \frac{E \cdot A}{L}$$

With E as Young's modulus, A is the effective cross-sectional area and L being the length of the part.

To obtain the ratio of stiffnesses;

$$\frac{E_{cap} \cdot A_{cap}}{E_{bolt} \cdot A_{bolt}} = \frac{K_{cap}}{K_{bolt}}$$

The materials properties and areas were;

Youngs modulus of Rod	200	Mpa
Area of Rod	148	mm ²
Youngs modulus of Bolt	202	Mpa
Area of Bolt	39.1637	mm ²

Table 16 Values for ratio of stiffness between the cap and bolt.

The bolt was thought to be manufactured from chrome molybdenum steel with the average Young's modulus of 202 MPa (Perry, et al., 2014). The bolt is a metric fine 8x1mm threaded bolt, of which the tensile effective area is stated to be 39.2mm² (BOSCH, 2002). And by using the CAD model to find the area of the cap, the ratio is;

$$\frac{E_{cap} \cdot A_{cap}}{E_{bolt} \cdot A_{bolt}} = 3.742$$

The length of the bolt was measured to be 37.5mm, but the theory is that the second engaged thread is subjected to the worst stress, therefore the length to the threaded section within the rod is 23.5mm, with an additional 2mm added to take the length to the second thread. The stiffness of the bolts is therefore;

$$\frac{E_{bolt} \cdot A_{bolt}}{L_{bolt}} = 310.24 \text{ N/mm}$$

To understand the range of stress that the bolt is going to see, the installation state of the bolt needs to be found. These bolt's are a one time use, correctly tightened within the manufactures building process and taken into a point of yield, by tightening 18Nm, then to the yield point (thought of as 40Nm (Perry, et al., 2014)), with a final 35° turn. This controlled tightening procedure is majorly important and is usually found through multiple stages of experimental testing. The more compressed the bolt, the less a range of stress experienced, hence wanting to effectively tighten the bolt to the same point of pre-stress every time.

Common practice for reusable bolts is to tighten to 90% of the yield point, as to ensure that with any deformation of the part/material, it should return back to original state with no issues, unlike the uncertainty of the plastic region where deformation will most likely remain permanent.

The UTS of the bolt is known as 1100MPa, stated on the bolt itself;



Figure 28 UTS printed on the bolt head.

Based on the bolt having this UTS, it was thought of as a hybrid bolt falling between the standard ratings 10,9 & 12,9. Both of which identify the UTS and the yield strength as being either 1000 or 1200 Mpa with yield being 0.9 times UTS. Made on the assumption that this bolt follows the same principle, yield strength is 990Mpa.

When tightening to 90% yield, the axial force required is thought of as 60% of yield. With stress being;

$$\sigma = \frac{Force}{Area}$$

Then 60% axial stress is; $0.6 \cdot 0.9 \cdot 0.9 \cdot UTS = 535MPa$

Multiplying this value by the effective tensile area of the bolt (39.2mm²), the force is 20936.91 N.

When tightening the bolt, frictional losses are incurred by part of the head of the bolt and the treads. The coefficients of friction were presented as 0.18 across the threads and 0.12 for the head of the bolt (BOSCH, 2002).

Using the axial force with these values and the effective radii of the head and bolt, the frictional torques required were;

$$Friction Torque_{thread} = (F_{axial} \cdot \mu) \cdot r_{thread}$$

$$Friction Torque_{head} = (F_{axial} \cdot \mu) \cdot r_{head}$$

$$Total torque = Friction Torque_{thread} + Friction Torque_{head}$$

With ' r_{thread} ' as 3.3675×10^{-3} (m) and ' r_{head} ' as 5.25×10^{-3} (m).

The torque required to overcome the friction and obtain 90% of yield compression within the bolt is 27.04 Nm.

Because of the installation stress on the bolt, the range of stress seen within the bolt isn't very high, none the less the effective stress concentration factor is. With the bolt being in the same situation as point C of the bearing cap, the forces seen are only occurring at TDC exhaust coming from the inertial forces of the accelerated masses. As previously stated, the bolt and connecting rod share the stresses seen because of the jointed nature, hence the need for the stiffness ratio in order to obtain a value for the potential extra force.

$$\text{Extra force} = \frac{\frac{F_{net}}{2}}{(1 + \text{stiffness ratio})}$$

Because the study was done for one side, assuming symmetry, the net force is divided by two. Upon using this equation, the extra force and then subsequent range and mean stress can be seen below;

	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000
Extra Force	0	19.56	38.04	78.88	158.15	247.87	362.67	490.10	635.35	795.23	974.80	1055.49	1179.47
Alternating Stress	0	0.50	0.97	2.01	4.04	6.33	9.26	12.51	16.22	20.31	24.89	26.95	30.12
Max Stress	891	891.50	891.97	893.01	895.04	897.33	900.26	903.51	907.22	911.31	915.89	917.95	921.12
Mean Stress	891	891.25	891.49	892.01	893.02	894.16	895.63	897.26	899.11	901.15	903.45	904.48	906.06

Table 17 Extra force, Alternating stress, Maximum stress and the mean stress on the standard bolt.

Regarding the stress concentration factor of the bolt, using the following graph returned a value of 5. Although it was thought as excessively high, research showed that the values weren't inconceivable and then backed up with the fact that the FSF remained above 1.

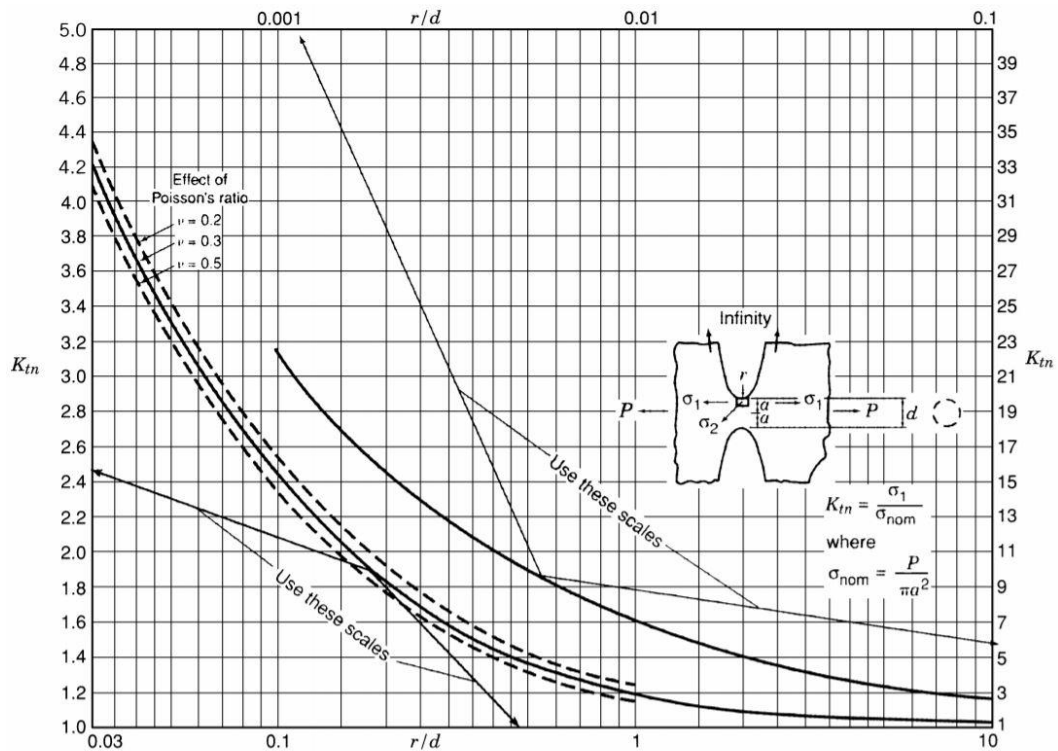


Chart 2.18 Stress concentration factors K_{tn} for a deep hyperbolic groove in an infinitely wide member, three dimensional case, in tension (Neuber 1958 solution).

Figure 29 Stress concentration factor graph used for the bolt thread (Pilkey, 1997).

Using the values presented by BOSCH in their automotive handbook and the diagram showing the representative relationships for the notations of a thread, the radius at the root and the distance between were found.

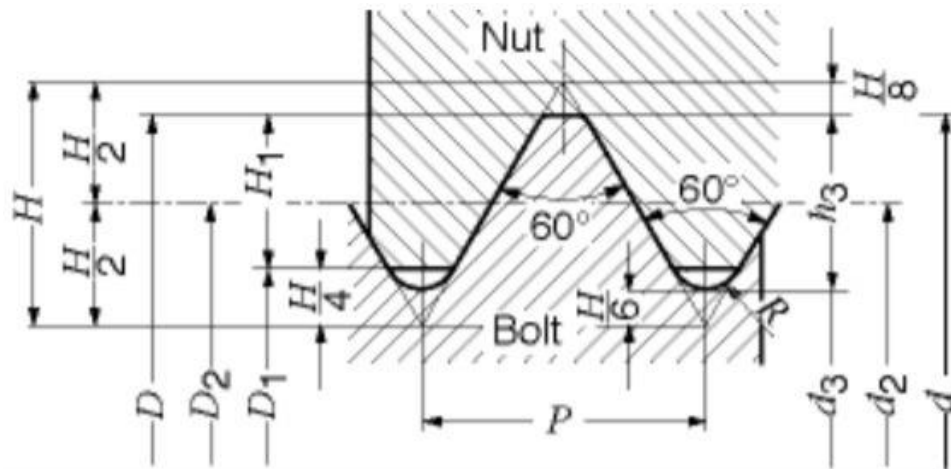


Figure 30 Diagram representing the notations used for thread geometry (BOSCH, 2002).

Using simple trigonometry, the value for 'H' was found easily as 0.866mm for a 1mm pitched bolt thread. It's stated that the root radius is equivalent to the distance denoted as ' $\frac{H}{6}$ ', and is therefore 0.1443mm. The distance between the thread is noted as 'd₃' and is said to be 6.773mm (BOSCH, 2002).

Using these values within the stress concentration graphic, ' $\frac{r}{d}$ ' was found to be 0.021 therefore the value for a stress concentration was found to be 5.

Having implemented this value into the endurance limit, the FSF values were;

Range at 0	Allowable	FSF
99	34.91	1.16

Table 18 FSF values for the standard bolt.

So, as shown, the standard bolt managed to retain an FSF value above one, implying that there shouldn't be an issue with failure due to fatigue of the part.

However, when the study was done based on a racing application of the engine speed limit increased to 9000RPM, it was a different scenario.

The bolt would fail is taken to 9000RPM in terms of fatigue because the allowable was found to be less than the actual range predicted.

Therefore, researching into standard materials used when larger UTS's are required, ARP state about using 8740 Chrome Molybdenum having a UTS of around 1200Mpa. With the forces acting on the bolt now shown as;

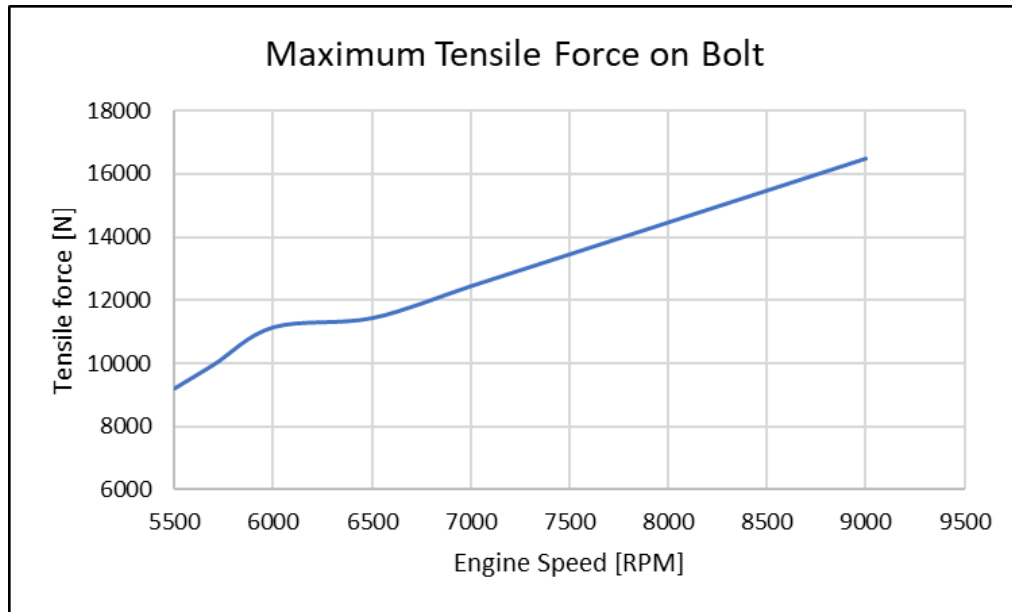


Figure 31 Forces on the uprated bolt.

The range and mean of stress therefore become;

	5500	5700	6000	6500	7000	7500	8000	8500	9000
Range of Stress	916.52	918.63	921.88	922.68	925.47	928.26	931.05	933.84	936.64
Mean Stress	903.76	904.82	906.44	906.84	908.23	909.63	911.03	912.42	913.82

Figure 32 Range and Mean stress on new bolt.

Although within the world of motorsport, the complete redesign of the bolt is often required, usually because they're to be used within a different connecting rod than the standard. Hence, they are designed to work with a specific rod, done so by modifying the bolts, average areas and inducing different stiffnesses across sections of the bolt, in order to control how the bolt behaves and as to where the bolt sees stretch. Because it's known that the area around the second engaged thread is where the worst stress is seen, if the section of the bolt prior to this could be made to have a different stiffness by using the relationship of;

$$K_{bolt} = \frac{E \cdot A}{L}$$

Because the area below the thread doesn't required any interaction with the surface of the rod, this could be 'wasted'. Meaning there is no need to thread this area, removing a multitude of stress concentrations, but the effective area can be reduced, therefore reducing the stiffness of this section, and therefore the stiffness. Because the stiffness has become a lot less than that of the threaded section, this will then become the point that see's the more stretch. By doing this, it takes stress away from the second engaged thread and therefore increases the expected lifetime and fatigue safety factor.



Figure 33 Standard bolt.



Figure 34 Modified bolt.

In terms of the redesign the major change is the reduction in height by the head of the bolt, changing the stiffness in this area. The wasted shank in the new bolt will have then drastically changed the stiffness here and would have now saved weight, machining and remove unnecessary thread induced stress concentrations. Then finally, the threaded section was shortened in order to reduce mass again, but also the number of engage threads with the female thread of the rod, reducing frictional losses when tightening the bolt.

Using the redesigned bolt that has a UTS of 1200MPa, Youngs modulus of 210GPa causing the drop in stiffness ratio of connecting rod to bolt, from 3.742 to 3.543. The final effect of this is the FSF now becoming 1.411 at 9000RPM, highlighting it can easily withstand the stress and won't fail from fatigue.

6. Crankshaft.

Naturally the last area of this study was the crankshaft, of which followed the same principles of the gudgeon pin in terms of bending and shear stress all taken from a certain point, in this case the centre of the main bearing journal.

The area of the crankshaft studied was the small section of the web between the overlap of both the 'big end' journal and the main journal, highlighted below;

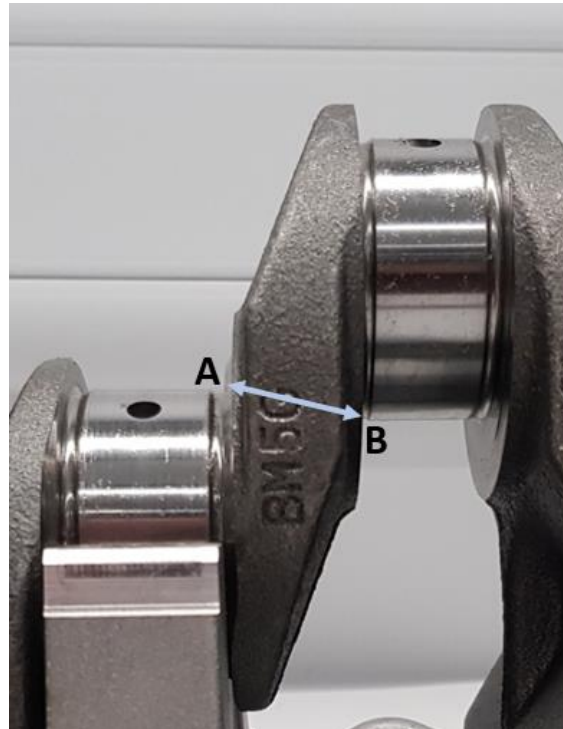


Figure 35 Section of the crank web used to pick points of concern.

Having measured the crankshaft to find the values for the diameters of the journals, their widths and the horizontal distance across the crank web from A to B.

AB	22	mm
Width of Main	24	mm
Diameter of Main	48	mm
Width of Big End	21	mm
Diameter of Big End	44	mm

Table 19 Values of crankshaft geometry.

The distance from the centre of the main journal to A is 12mm and to B is 32.8mm. Calculating the bending moments at each point during both combustion and TDC exhaust, the resulting forces and moments for both A & B are below;

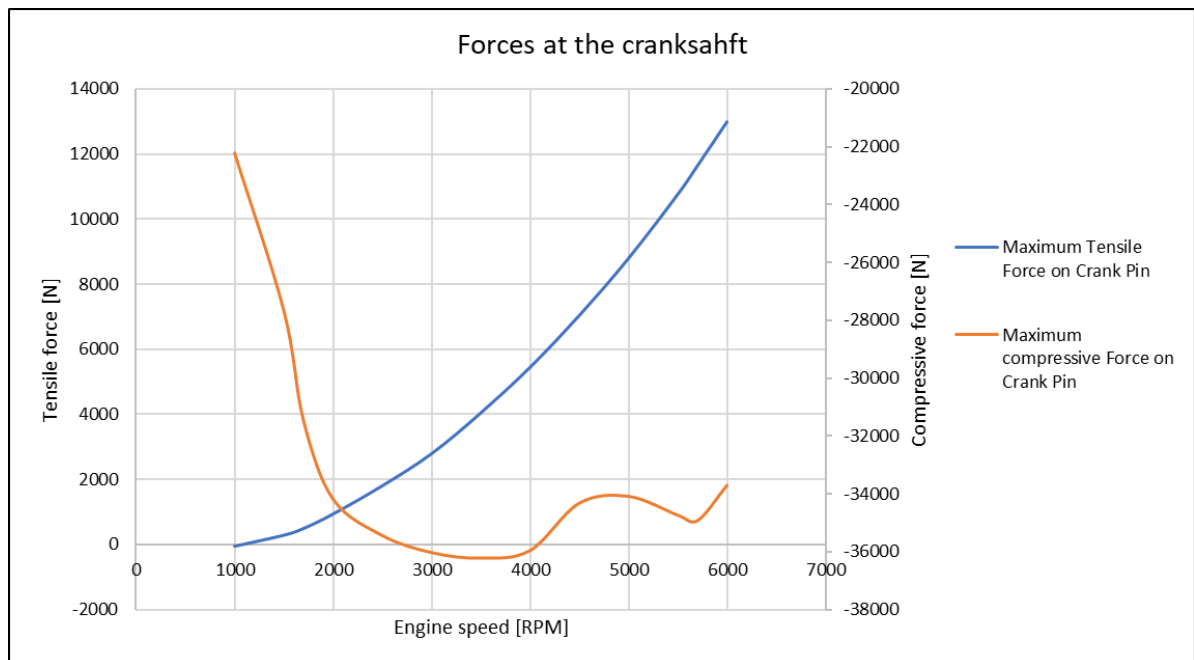


Figure 36 Forces occurring on the crankshaft.

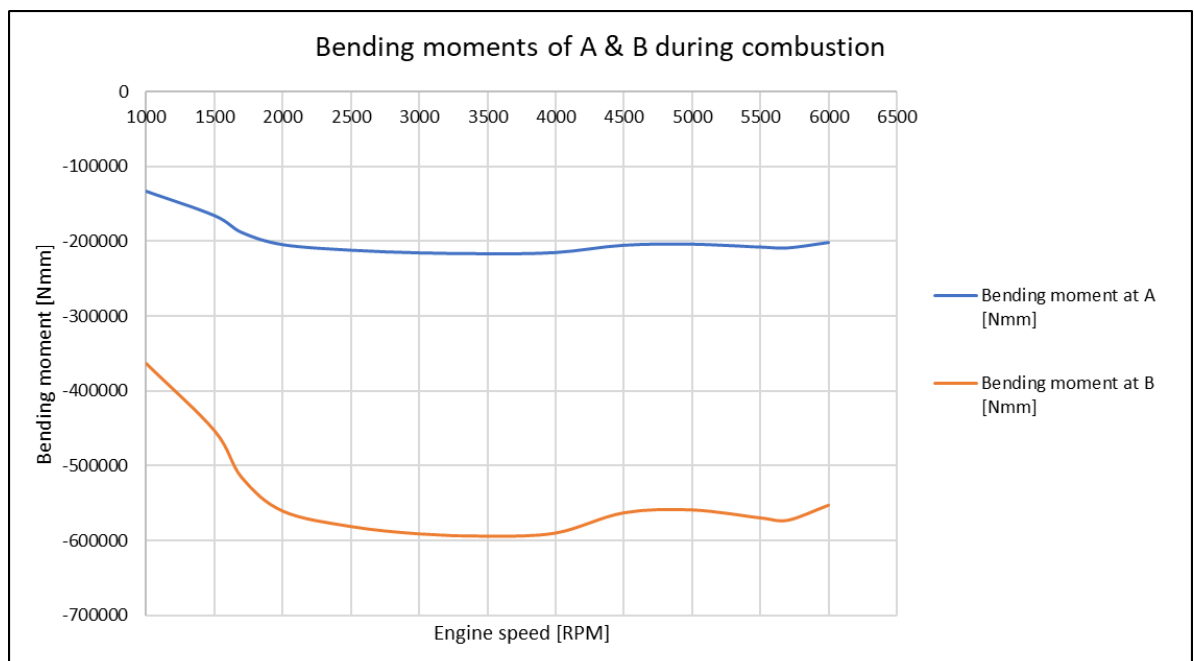


Figure 37 Bending moments of A & B during combustion.

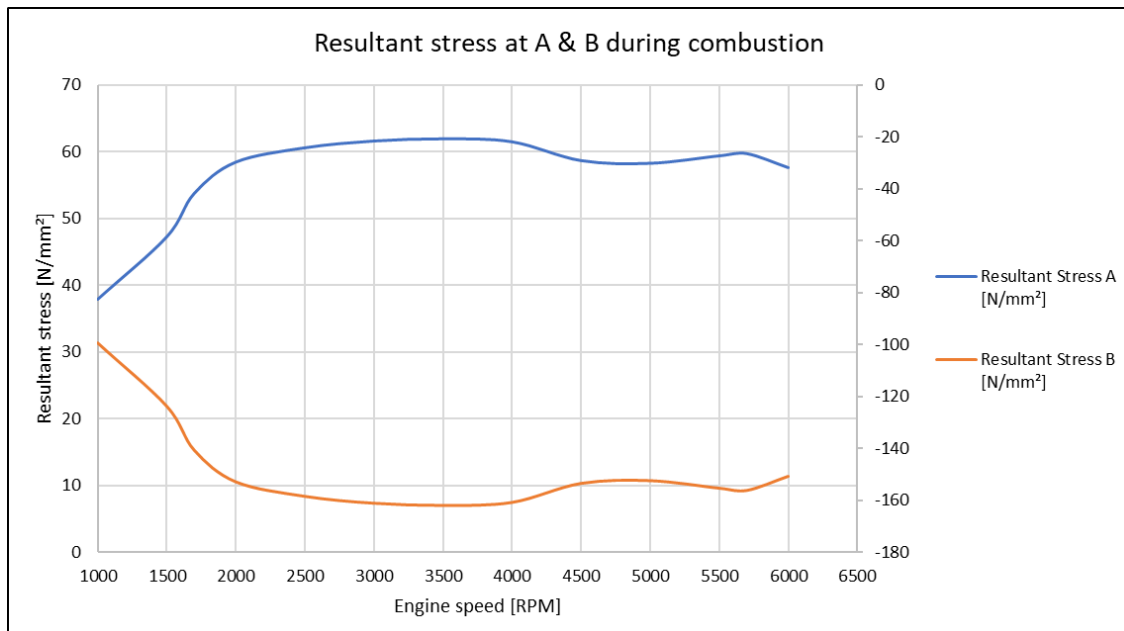


Figure 38 Resultant stress of A & B during combustion.

When looking at the stresses occurring at both A & B, A is put into a tensile situation and therefore is the point of main concern during combustion, hence the positive resultant stress shown in the figure above. Because of this, B is therefore put into compression and isn't of real concern during combustion.

When looking at TDC exhaust, the situations change, and A is then put into a compressive situation and B the tensile. Results of which follow;

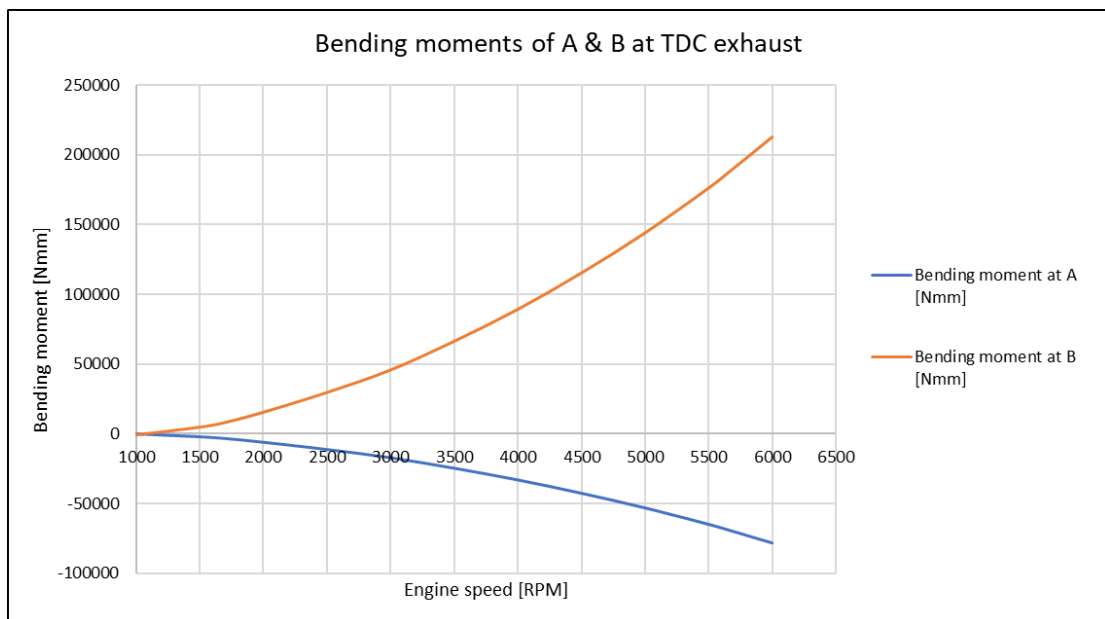


Figure 39 Bending moments of A & B at TDC exhaust.



Figure 40 Resultant stresses of A & B at TDC exhaust.

Because of the much larger distance from the centre of the main journal, B is clearly experiencing much larger stresses because of the bending moment. Therefore, the removal of any and all stress concentrations possible is so important. The range of stress at A & B were found to be;

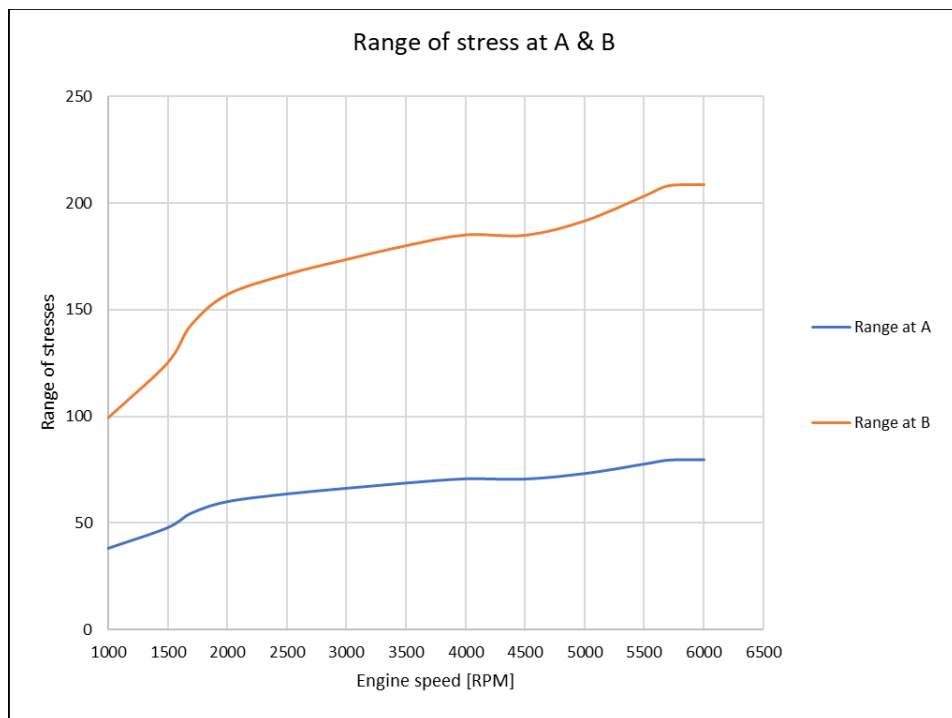


Figure 41 Range of stress for A & B of the crankshaft.

The figure above clearly shows how dramatic the distance away impacts the forces happening at certain points. Because as can be seen, the range at B is over double that at A, which is why when following through to the FSF, it is considerably lower for B than A.

To calculate the FSF, stress concentration factors were required because of the semi-circular notches made into the journals of the crankshaft, presumably to allow travel of oil behind the bearing shells, whilst making machining of the part easier. Based on its shape and the circular nature of the journal, the following graph was chosen to calculate the stress concentration factor;

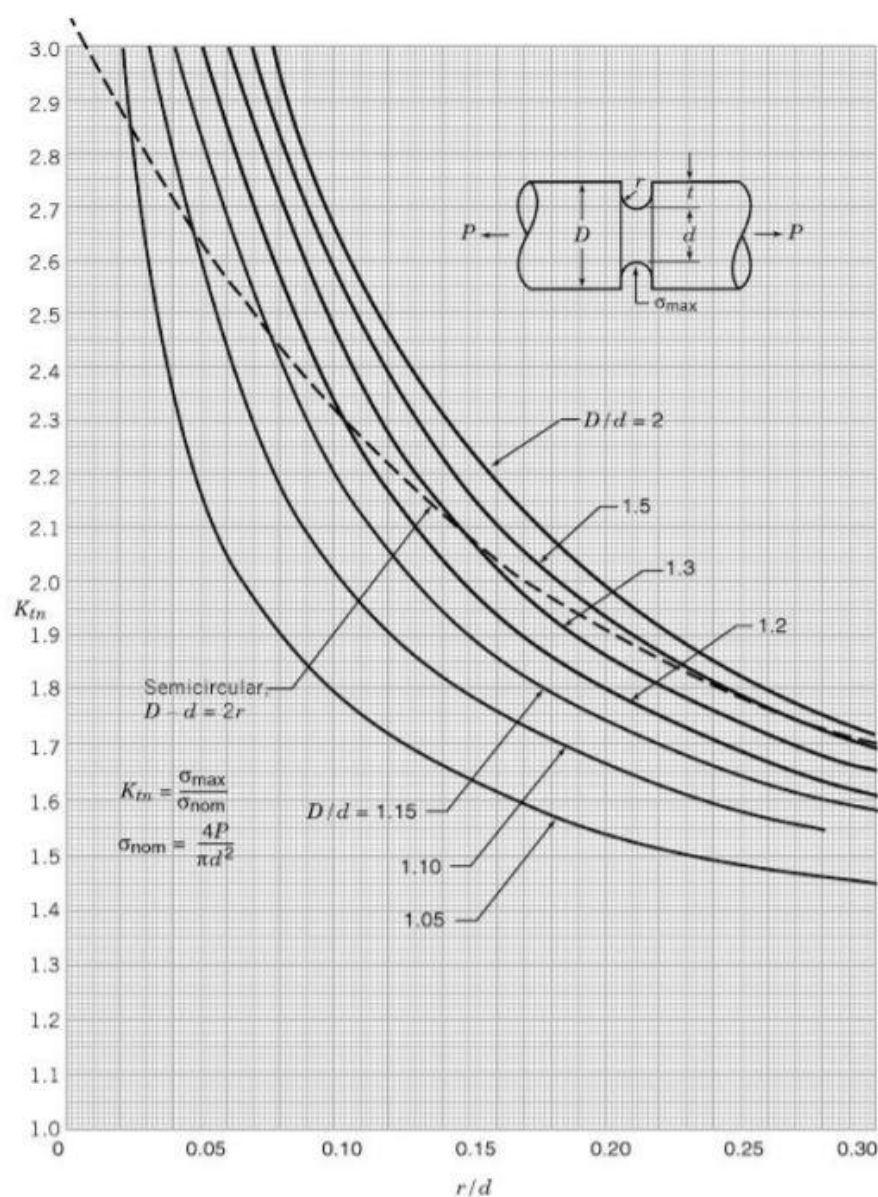


Chart 2.19 Stress concentration factors K_{tn} for a tension bar of circular cross section with a U-shaped groove. Values are approximate.

Figure 42 Stress concentration graph for the crankshaft (Pilkey & Pilkey, 2008).

For the main journal the values of ' $\frac{D}{d}$ ' and ' $\frac{r}{d}$ ' were 1.0748 and 0.0374 respectively, meaning that the stress concentration factor is 2.85. And for the big end journal those values were 1.0821 and 0.0411, again respectively, causing the stress concentration factor to be 2.8.

Implementing these into the endurance limits, the FSF values were found to be;

Range at 0	Allowable	FSF
120.54	235.38	2.95

Table 20 FSF values for point A on the crankshaft.

Range at 0	Allowable	FSF
118.42	251.46	1.21

Table 21 FSF values for point B on the crankshaft.

From these values, it's clear to see that although the allowable range of stress at B is higher than that of A, because of the smaller mean value, the fatigue safety factor is far less. This is because of the greater range of stress experienced at B, due to the distance from the main journal centre.

Under these situations, it is believed that the part will not fail due to a fatigue issue, even with the stress concentrations experienced at these points. Of course, this is backed up with the fact of which, the real crankshaft within the EcoBoost engines has no history of fatigue issues and can clearly reach the target cycle value of 10^7 , implying its "infinite" lifecycle.

7. Conclusion.

In conclusion, although there is multiple assumptions made and certain factors neglected, using simple mathematics and the laws governing material properties the fatigue safety factors of the parts used can be estimated with confidence.

The values presented in this report would require validation in terms of both advanced computational modelling to gain a better understanding of the stresses seen within the part, along with a series of real life testing, be it through destructive and non-destructive testing methods to understand how the materials deform due to cyclic operation of an ICE.

Throughout the report, it was consistently found that the safety factors of the ford parts were sufficiently high to show that fatigue shouldn't become an issue within the 1600cc EcoBoost engine, to which is backed up with the fact that there isn't any reports of failures to a catastrophic level during the years of manufacture and or since they were taken out of manufacture.

Ford motor company have many years of experience within the automotive world, to which is why they can easily achieve these levels of safety within the parts, whilst maintaining a cheap enough build cost, all of which led to this engine being a staple within their history and a favourite of many.

8. References

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9. Appendices

1. Gudgeon Pin static load tests.

Static loading tests completed within SolidWorks on the standard gudgeon pin and then an upgraded H-13 tool steel pin.

The standard gudgeon pin was ran through a static force test, to which assuming the net force was equally divided either side (hence half force used), the force applied was 21082.91N at the end of the pin. This distance is slightly further than where the force would act is using the assumption of the halfway point from inside the bearing face of the piston. Therefore, this test was more of a potential worst case than the pin should see, the resultant stress was found to be;

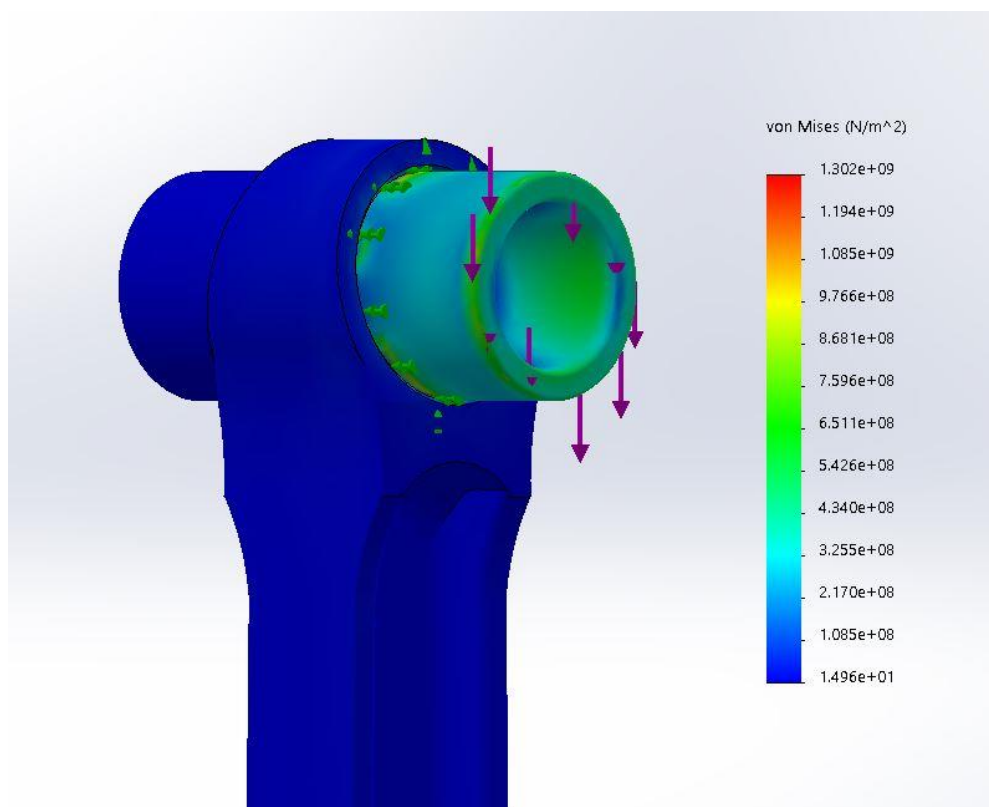


Figure 43 Stress induced from a Static load force test conducted within SolidWorks on the presumed standard gudgeon pin.

And as seen, the yield strength of the pin hasn't been reached in any sense (the point is shown with a red arrow on the colour scale axis, if met).

The strain experienced by the part was also seen to be a trivial value;

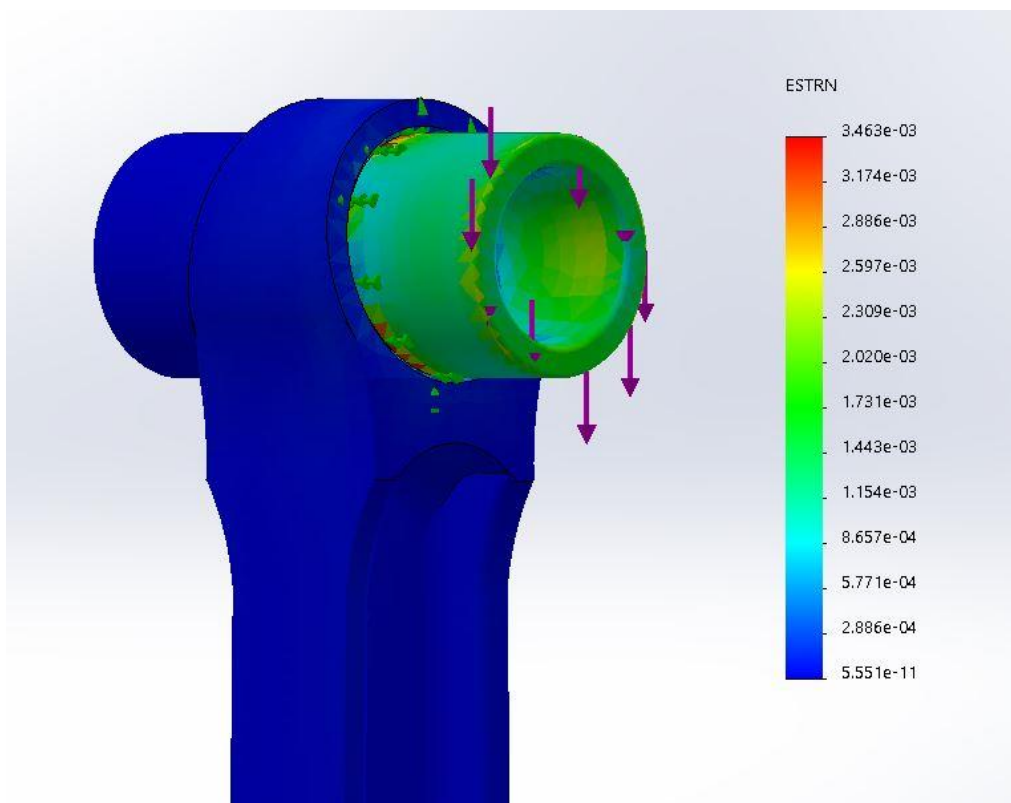


Figure 44 Strain calculated within the static load test on the standard gudgeon pin.

The worst strain experienced was a value of 0.00346 with a maximum displacement value of 0.08902 mm of change.

2. Stress concentrations at points A & B on the connecting rod.

As stated within the chapter, stress concentration factors for points A & B were negated from the study because the radii of the sections are so large that the resultant values required were far too large, to achieve a value from the graph. But because this isn't reality, the geometry will induce some form of stress concentration, as shown during the static load study done on the part;

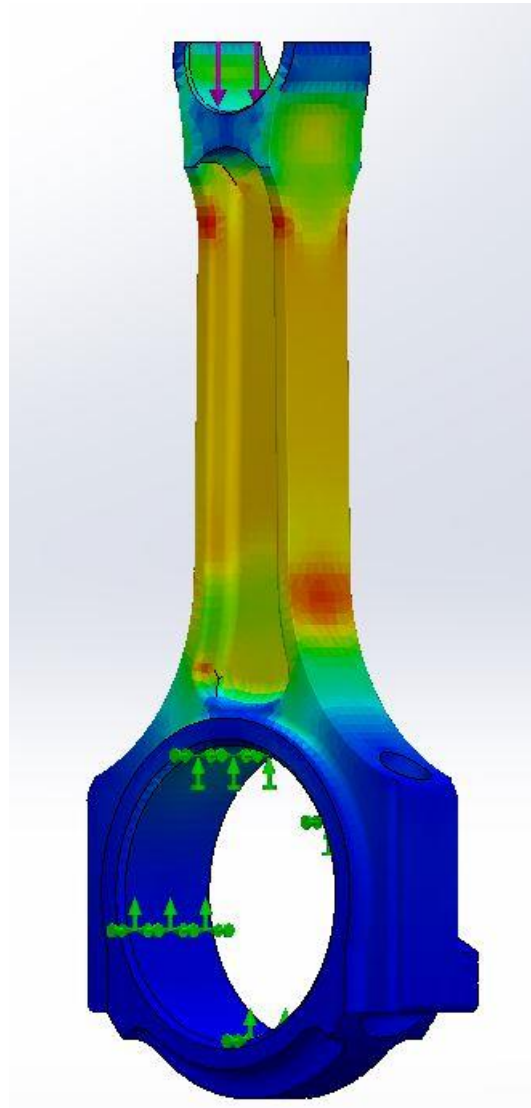


Figure 45 Strain for across the rod for a static load test.

There is a clear point around the area of B that is showing signs of concentration and because of the linearly directed force, A hasn't shown to be a large concentration. However, in a real-world situation the force would most likely be at an angle down the rod, due to the reciprocating – rotary action within the ICE, that

would cause A to see a concentrated stress. Because if B has a larger radius and diameters and is still seeing the concentration, A would indefinitely be worse given the smaller geometries.

Also, when looking at the more expensive and uprated versions of connecting rod designs, there is an obvious effort taken to reduce the radius at these points, or at the very least an attempt to make it a more streamlined arc. For example, comparing PEC's upgraded forged 'H' beam rods to the original rod;



Figure 46 PEC Forged 'H' beam connecting rod (Gumtree, 2015).



Figure 47 Ford standard connecting rod.

There is an obvious difference across the rod, from the removal of small notches and grooves, especially around the cap. But also, the change to point A, with having essentially removed the radius of the side and then the smoothing out of the radius around point B, whilst simultaneously increasing the diameter of this point, helping again to reduce the effect of a concentration.

3. Testing of the standard bolts to the point of failure.

A short study was conducted on the standard bolts to try categorising the properties of the material and understand how it will behave past the elastic limit.

Using the idea that Ford tighten their bolts by either, tightening to 18Nm and then 35° to take it into yield, which through the process of calculating the tightening torque seemed to be the realistic situation. However, the bolt showed a vastly different set of characteristics when being tested, therefore causing dispute as to whether this is true.

Using a DTI gauge to measure how much the bolt is being stretched as it is tightened, the first process was done by tightening to 18Nm, then 35° to which caused the bolt to stretch by 0.22mm. And although it was difficult to identify, the bolt didn't reach a 'soft' point, demonstrating that it had made it to the plastic region.

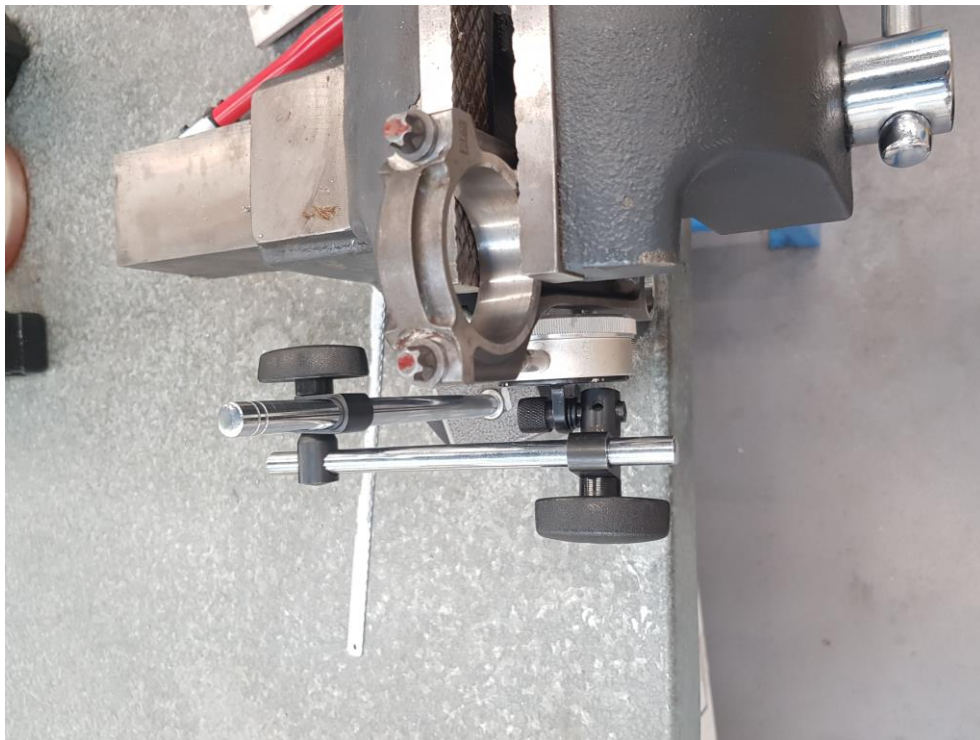


Figure 48 Setup used to measure the stretch of the bolt.



Figure 49 Initial stretch from tightening as presumed was correct.

From this point, the bolt was then turned a further 900° to which there was no notable change in material properties when tightening and the bolt had then by this point stretched by 2.43mm without braking. It was then removed to check the bolt and it was clear where the bolt was going to break;



Figure 50 Deformation and stretch of the bolt.



Figure 51 Tested bolt against original.

A test was then conducted on the secondary bolt where it was tested at different torques up to the believed yield point of 40Nm.

Torque	Stretch
18	0.08
24	0.1
30	0.13
35	0.15
40	0.185

Table 22 Torque and the measured stretch for the bolt.

The second bolt was then retightened under the process of 18Nm, yield of 40Nm and then 35°;

Torque	Stretch
18	0.08
40	0.2
35°	0.3

Table 23 Second test torques and stretches.

This bolt was then tightened to a further 900° alike the other bolt, to which the point of failure was found, but not in the bolt threads but with the head.



Figure 52 Failed bolt head.

This bolt failed around the mark of 830° and whilst tightening felt far more restricted compared to the first test, possibly due to contamination in the threads or due to stretch hardening occurring within the bolt. Although the results are to be disputed on the fact that it's unknown if they were previously tested, the equipment not being exact and through human error within tightening.

Either way, it's now thought that the materials plastic region is very unpredictable and therefore hard to mathematically decipher, but it's now thought that Ford tighten the bolts initially when built to 18Nm, 40Nm (or closest yield point) and then a further 35° to take it to a desired point within the plastic region.