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**Investigation into the Effect of Lubricants on Fastener
Coefficient of Friction and an In-depth Fatigue
Analysis of a Connecting Rod Bolted Joint.**

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Module: Major Project

Module code: M6X00417

24/04/201

This thesis is the result of the author's original research. It has been composed by the author and has not been previously submitted for examination which has led to the award of a degree.

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Abstract

This paper looks very at a very in depth analysis of a Ford Sigma 1.6L connecting rod bolted joint and the behaviour of lubricants on friction. An experiment was conducted comparing the effects on the coefficient of friction and hence torsional stress present, that engine oil, general grease, specialised fastener grease and no lubricant had. It was found with a sample of 10 bolts for each lubricant that engine oil had a slightly negative effect raising the coefficient of friction by 8%, general grease had a very small positive effect lowering the coefficient of friction by 4% and the specialised lubricant had a large effect lowering the coefficient of friction by 23%. An unused method of calculating the individual coefficient of friction of the head and thread was tried on the results and was found to only give a rough estimate rather than a reliable figure.

A force washer was fabricated using a strain gauge to give more accurate readings of the preload force from the bolt and to see how preload changed as the bolt yielded. This data was used to confirm the ford tightening procedure and to carry out calculations to determine the fatigue safety factor of the standard bolt. An FEA analysis of the con-rod in tightened conditions was carried out to determine the loading planes of the con-rod. The fatigue safety factor of the ford bolt at 6000rpm was then calculated to be 3.26.

A conceptual idea of using a hydraulic tensioning method and a stronger bolt material was then explored and an estimate made of the benefits to the fatigue life and mass savings from this method. It was estimated a weight saving of 2.5g for each bolt could be made and an increase in the fatigue safety factor to 3.94.

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1.0 Introduction

The bolt is an often overlooked aspect of many engineering problems despite being used widely through an almost unlimited range of applications. When explored more in-depth, it becomes clear the complexity of a bolted joint and the methods of ensuring it is safe.

This paper is focussed on an analysis of the connecting rod bolted joint of a Ford Sigma 1.6L and the effect that lubricants have on the coefficient of friction.

An experiment was carried out to determine the effect of 3 different lubricants on the friction of the fastener and the effect this has on the amount of torsional stress present. The length of bolts was measured as they were tightened in torque increments. A method of calculating the individual coefficient of friction of the head and thread was then tested. Using more accurate data from a self-made force washer, and some FEA analysis of the bolted joint, the fatigue life of the standard bolt was then calculated and aspects of the fatigue researched.

Finally, the benefits of a conceptual method of tightening the con-rod using a hydraulic tensioner was explored and estimates made of the positive effect this could have.

2.0 Effect of lubricants on fastener coefficient of friction and torsional stress

2.1 Information

2.1.1 Axial stress

When a bolt is tightened it elongates due to the axial forces created at the thread and creates tensile stress in the engaged length of the bolt. At the same time, this axial force creates friction at the threads which is overcome by the applied torque through the shank of the bolt, creating torsional stresses in the engaged length.

Calculating the nominal axial stress in the shank of the bolt is simply the preload force over the tensile area of the bolt, however, this may not represent the true stress at the surface of the bolt. On bolts with a grip length to diameter ratio of less than 4:1, the stress will vary through the shank of the bolt with a higher stress at the surface than at the centre of the bolt. (Bickford, 2008)

The Ford con-rod bolt being tested has a grip length to diameter ratio of just over 3:1 so will be close to uniform but with slightly greater stress at the surface. Unfortunately, this is a complex problem with the shank of the Ford bolt being threaded and so, for the purposes of these calculations, it is assumed the stress is evenly distributed through the shank.

2.1.2 Torsional stress

The torsional stress in the shank of the bolt peaks on the surface of the shank where it is the furthest distance from the centre and the stress distribution looks like figure 1 below

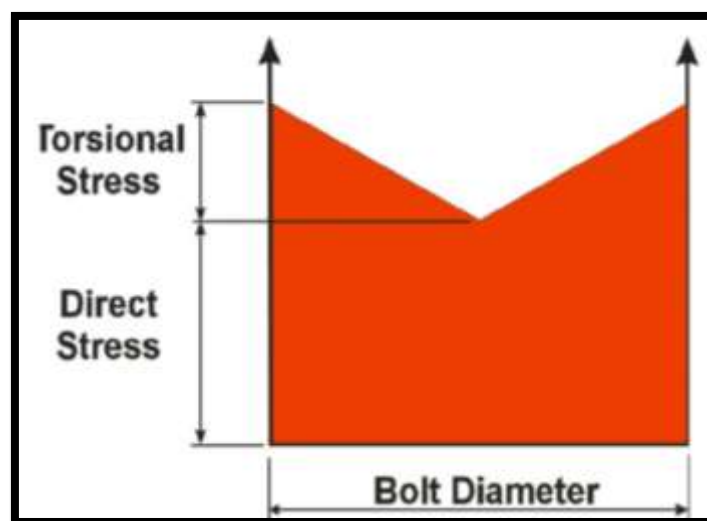


Figure 1

The coefficient of friction at the threads determines the resisting torque to the bolt being tightened and so directly controls the amount of torsional stress that will be seen in the shank. Experimentally determining the coefficient of friction individually for the bearing faces and thread is usually done with a set-up beyond the scope of this experiment involving

a torque transducer at the threaded end to measure torsion. However, a less accurate method will be used to approximate the values.

The tensile and torsional stress combine to give an overall stress in the bolt (this will be discussed in greater depth later in the paper). It is desirable to maximise the amount of tensile stress, as this creates the clamp force, and minimise the amount of torsional stress which provides no clamp force. Because the torsional stress is directly correlated to the coefficient of friction at the threads, using a lubricant can make it possible to utilise a larger proportion of a fasteners stress capability as tensile stress, therefore increasing the clamp force possible from the fastener.

2.1.3 Nut factor

The nut factor is different from the the coefficient of friction in that it summarises all of the factors that affect the required torque into one coefficient, rather than just the friction, and was first used by Shigley in his work on fastener friction (Shigley, 1977) . The largest factor effecting the torque is indeed friction but the nut factor also accounts for effects such as non-parallel surfaces, bending of the joint, bending of the bolt, torsion, localised plastic deformation of threads, material irregularities, and many more factors that are not able to be calculated through theory. (Bickford, 2008)

The nut factor is the most widely used method of determining the required torque in industry and is essentially showing that the gradient of the linear section created in a torque-preload plot from experimental results will give you a constant that can then be used to calculate the required torque should you require a different preload. It is calculated using the equation below:

$$K = \frac{T}{FD}$$

Where:

K = Nut factor

T = Torque (Nmm)

F = Preload force (N)

D = Nominal bolt diameter (mm)

2.2 Experiment

2.2.1 Aims

An experiment was carried out on 40 bolts using 4 lubrication options with the aim of determining the effect the lubricants have on the coefficient of friction of the bearing faces and thread, the nut factor, and hence the amount of torsional stress present in the bolt. The lubricants tested were Silkolene 10W-40 engine oil, Silkolene RG2 high performance general grease, specialised ARP molybdenum disulphide fastener assembly lubricant and no lubricant as a control test.

The stiffness of the bolts and cap were first calculated through both theoretical means and FEA analysis. The lengths of the bolts were measured as they were tightened incrementally in the con-rod through to failure. The bolt length was plotted against the torque applied and from the results, the combined coefficient of friction and the tensile stress in the bolt at

yield was calculated. The torsional stress in the shank and the individual coefficients of friction of the bearing faces and the thread can then be estimated.

2.2.2 Method

- Bolts first had their bottom end ground flat and a chamfer applied for increased accuracy when measuring as shown in figure 2 below:



Figure 2

- The bolts and con-rod were then cleaned in a de-greasing parts washer to remove any residual grease and particles from the threads.
- The length of the bolt was then measured using a micrometer.
- The selected lubricant was then applied to the threads and bearing faces and the bolt was inserted into an inverted con-rod held in a vice and tightened to 20Nm using a clicker type torque wrench.
- The length of the bolt was then measured and recorded against the torque setting.
- This was then repeated at 2Nm increments through to bolt failure.

10 bolts were tested under each condition, changing to a fresh con-rod hole every 5 bolts to avoid excessive wear of the bearing faces and thread.

2.2.3 Experiment notes

10 bolts were tested under each condition, changing to a fresh con-rod hole every 5 bolts to avoid excessive wear of the bearing faces and thread.

20Nm was used as the starting torque due to this being the lowest setting on the available torque wrench.

The torque wrench used was approximately 2 years old and had seen little use since it was new and calibrated. Some error in the specified torque figures will be present however, this is estimated to be small and, as the same torque wrench was used for all bolts, the results will be comparable.

The accuracy of the micrometer is specified as being accurate to within 0.01mm, however bolts were repeatedly and reliably measured with an accuracy of 0.005mm. This was enough to accurately measure the gradient of the torque- stretch plot during the elastic region but some appreciable error was induced when estimating the yield torque.

2.2.4 Calculating combined coefficient of friction

The combined coefficient of friction of the head and thread was calculated from the long form torque equation shown below which is a simplification of the full energy equation from the work carried out by A.R Srinivas. (Srinivas, 1992)

$$T = F \left(\frac{p}{2\pi} + \frac{\mu_t r_t}{\cos \beta} + \mu_n r_n \right)$$

This shows that the input torque is resisted by three reaction torques which are, from left to right, the torque caused from the inclined plane of the thread helix which stretches the bolt, the torque to overcome friction at the threads, and the torque required to overcome friction at the load bearing faces. (Bickford, 2008)

This was rearranged for the combined coefficient of friction:

$$\mu_{total} = \frac{\left(\frac{T}{F} - \frac{p}{2\pi} \right)}{\left(\frac{r_t}{\cos(\beta)} + r_n \right)}$$

Where:

T = Torque (Nmm)

F = Bolt preload force (N)

p = Thread pitch (mm)

μ_t = Thread coefficient of friction

r_t = Effective radius of threads (mm)

β = Half angle of the threads (degrees)

μ_n = Bearing faces coefficient of friction

r_n = Effective radius of bearing faces (mm)

The equation involves an assumption that the thread and bearing face friction are equal in order to make the rearranging possible which induces a small error, however, the error is small and the equation is widely used in industry and research. (Bickford, 2008)

2.2.5 Calculating axial stress

The preload of the bolt can be calculated whilst the bolt is in its elastic region through the relationship of force and a linear spring:

$$F = K \cdot \Delta L$$

Where:

F = Force (N)

K = Stiffness (N/m)

ΔL = Change in length (m)

The nominal axial, or tensile, stress in the shank can then be calculated using the stress equation below:

$$\sigma = \frac{F}{A}$$

Where:

σ = Stress (N/m²)

F = Force (N)

A = Area (m²)

2.2.6 Estimating yield point

Tensile tests were carried out on 8 bolts to determine the Ultimate tensile strength and their yield point. The results are shown below in table 1:

	UTS	Yield point
Bolt 1	1145.12	1033.552
Bolt 2	1098.965	1008.644
Bolt 3	1117.666	1020.979
Bolt 4	1147.905	1034.507
Bolt 5	1133.183	1038.088
Bolt 6	1142.335	1022.571
Bolt 7	1129.602	1038.884
Bolt 8	1147.109	1021.377
Average	1132.736	1027.325

Table 1

By using the above stress equation to calculate the axial stress in a tightened bolt at point of yield and knowing the yield stress of the bolt, the proportion of stress that must be torsional can then be calculated.

When a bolt of high UTS yields, there is not a definitive point where the bolt stiffness changes, rather a smooth curve into yield making it difficult to define the yield point. Because of this several methods have been proposed to attain a closer estimate such as the offset method and gradient method.

The gradient method measures the gradient at individual points along the curve and compares them to the gradient in the elastic region. Once the point where the gradient becomes a set amount higher than the elastic region is called the yield point. (Shigley, Mischke, & Budynas, 2004)

The offset yield method, or non-proportional yield, is most commonly used and is shown in figure 3. This is where a line of the same gradient as the elastic region is drawn from a point of usually 0.2% of the original length and the intersection point of the two plots is called the offset yield point. (Shigley, Mischke, & Budynas, 2004)

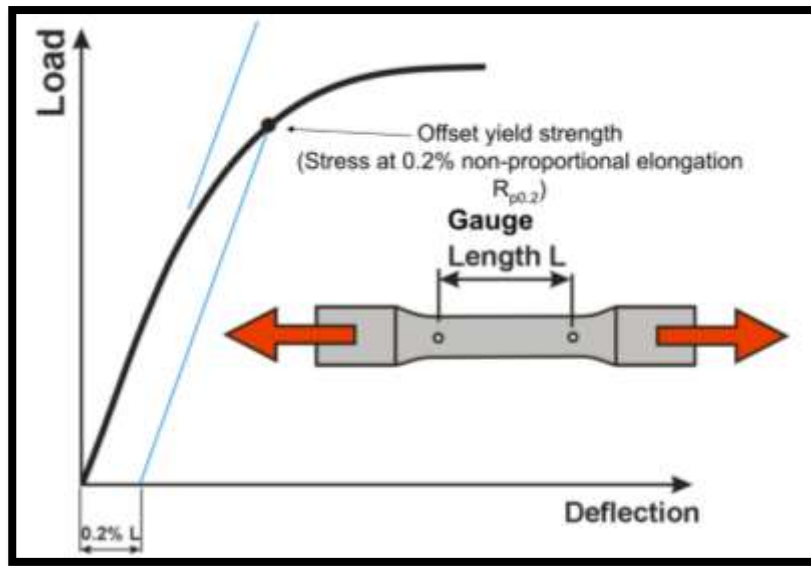


Figure 3

Unfortunately, the data gathered has a lot of 'noise' in it where slip-stick phenomenon of the bolt is occurring and so both of these methods do not work consistently with the data. Instead the yield point of each bolt was determined individually by analysing the figures and data available and accounting for errors and noise on a subjective basis.

2.2.7 Calculating torsional stress

It is important to note that torsional stress is a shear stress and shear stresses cannot be directly added onto a tensile stress with regards to the UTS of a material because they both have different severity to the yielding of the material. Therefore, all that is known at the point of looking at the amount of axial stress at yield is that the remaining stress is a shear stress, represented as a tensile stress with regards to the tensile yield limit of the material.

There are several similar methods of combining tensile and shear stresses together to get an equivalent stress, normally with regards to the UTS. The most applicable and most used for ductile materials are Tresca theory and Von-mises theory. (Shigley, Mischke, & Budynas, 2004)

Von-mises is the industry standard for bolting applications and the equation is shown below:

$$\sigma_r = \sqrt{\sigma^2 + 3\tau^2}$$

Rearranged for shear stress this gives the equation:

$$\tau = \sqrt{\frac{\sigma_r^2 - \sigma^2}{3}}$$

Where:

σ_r = Resultant stress (N/mm²)

σ = Tensile stress (N/mm²)

τ = Shear stress (N/mm²)

With the shear stress now known, the equation for the reaction torque from the thread, that is creating the torsion stress, can be substituted into the 'shear stress in a bar under torsion' equation and can be rearranged to get the thread friction.

The 'shear stress in a bar under torsion equation is shown below:

$$\tau = \frac{Tr}{J}$$

Where:

τ = Shear stress (N/mm²)

T = Torque (Nmm)

r = Radius of bar

J = Polar moment of inertia (mm⁴)

[2.2.8 Calculating individual thread coefficient of friction](#)

The torque, in the case of the bolt, is the reaction torque that causes torsional stress in the shank. The friction under the head must first be overcome before torsional force can be created in the shank so it is the force from the incline plane of the thread and the friction of the threads that create the reaction torque. From the torque equation earlier it can be shown that this torque will be as below in equation:

$$T_t = F \left(\frac{p}{2\pi} + \frac{\mu_t r_t}{\cos\beta} \right)$$

Where:

T_t = Torsional torque (Nmm)

F = Bolt preload force (N)

p = Thread pitch (mm)

μ_t = Thread coefficient of friction

r_t = Effective radius of threads (mm)

β = Half angle of the threads (degrees)

Substituting the 'torsional torque' equation into the 'shear stress in a bar under torsion' equation and rearranging for thread friction gives the following equation:

The rearranging can be found in the appendices.

$$\mu_t = \frac{\cos\beta \left(\frac{\tau J}{Fr} - \frac{p}{2\pi} \right)}{r_t}$$

This can then be subtracted from the combined coefficient of friction to give the bearing faces coefficient of friction.

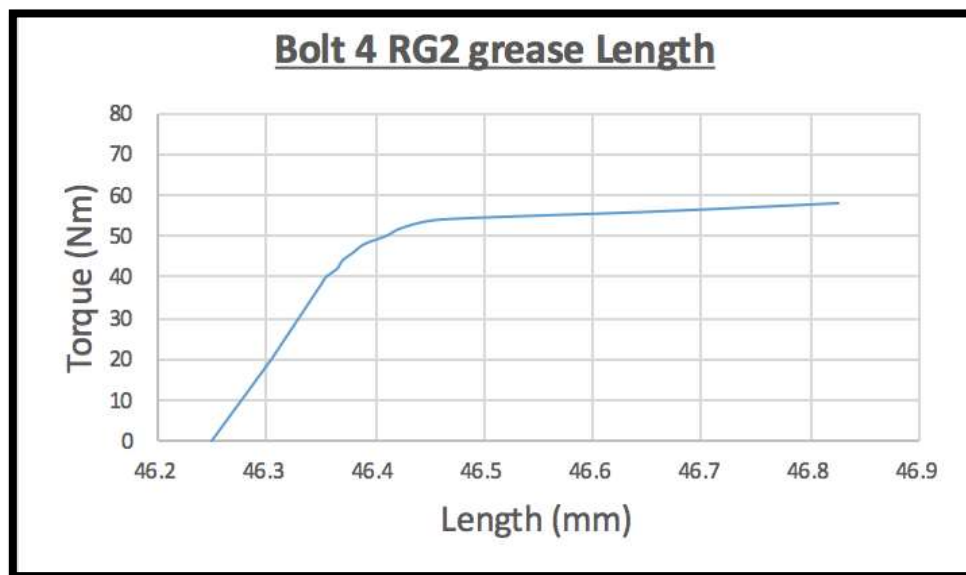
This method of calculating the thread and bearing faces coefficient of friction is not used in industry as very accurate methods are available with the correct equipment. Due to very limited equipment being available for the purposes of this research, the above method was constructed and will be tested. The accuracy of the above method will be in question, largely due to the dependency on the accuracy of the change in length measurements to

determine the force, and having to estimate the yield point from the data. As more factors that rely on the force are brought into the equation, such as the total coefficient of friction equation, a small amount of inaccuracy at the point of measuring will essentially be amplified for the final results.

2.3 Results and analysis

2.3.1 Torque length measurements and strain

The results of the torque to bolt length were recorded and plotted on a graph for analysis. The majority of the bolts tested came out with very smooth curves clearly showing the relationship between the applied torque and the change in length is linear whilst the bolt is in its elastic region. The yield point is able to be determined with reasonable accuracy. It is important to note that the end of the plotted line is not the length of the bolt at point of fracture, the bolt likely continued on past this length but was not measured as the next torque value was not reached. An example is shown below in graph 1:



Graph 1

Because of minute differences in the length of the bolts, the strain was calculated for each bolt so the results could be plotted on a single graph. The strain was calculated using the equation:

$$Strain = \frac{\Delta L}{L}$$

Where:

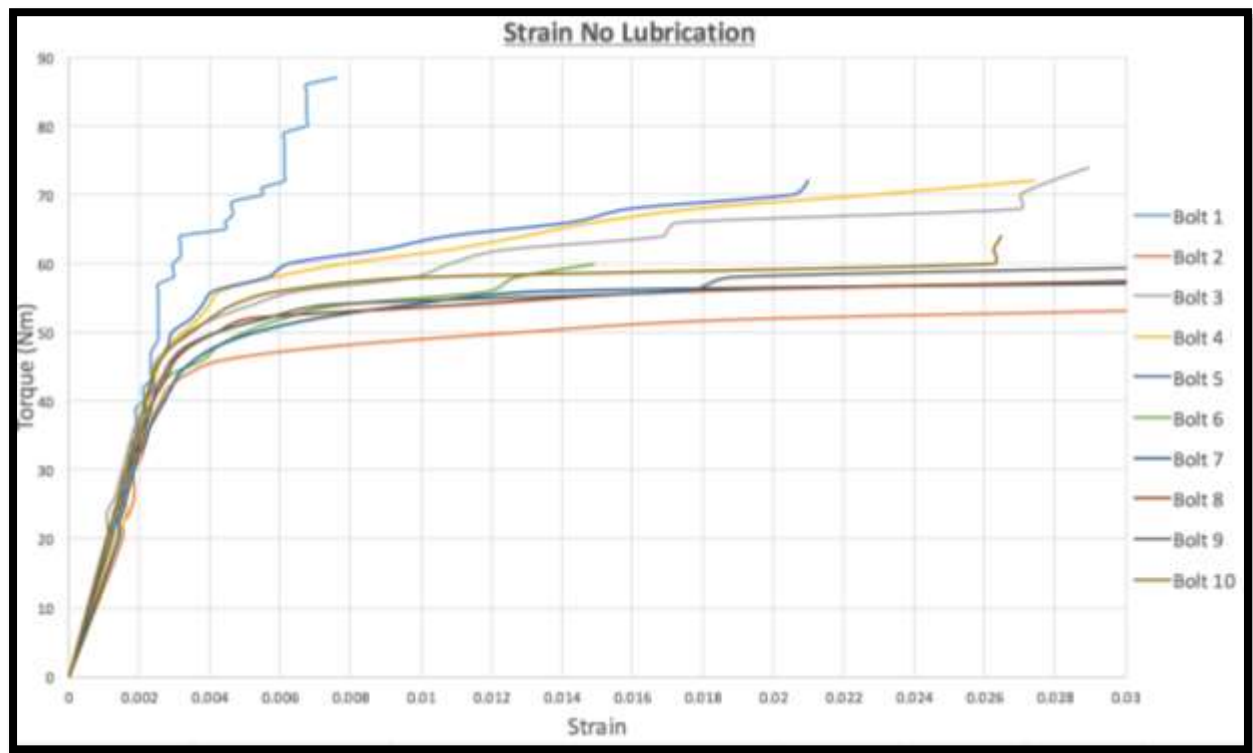
ΔL = Change in length

L = Original length

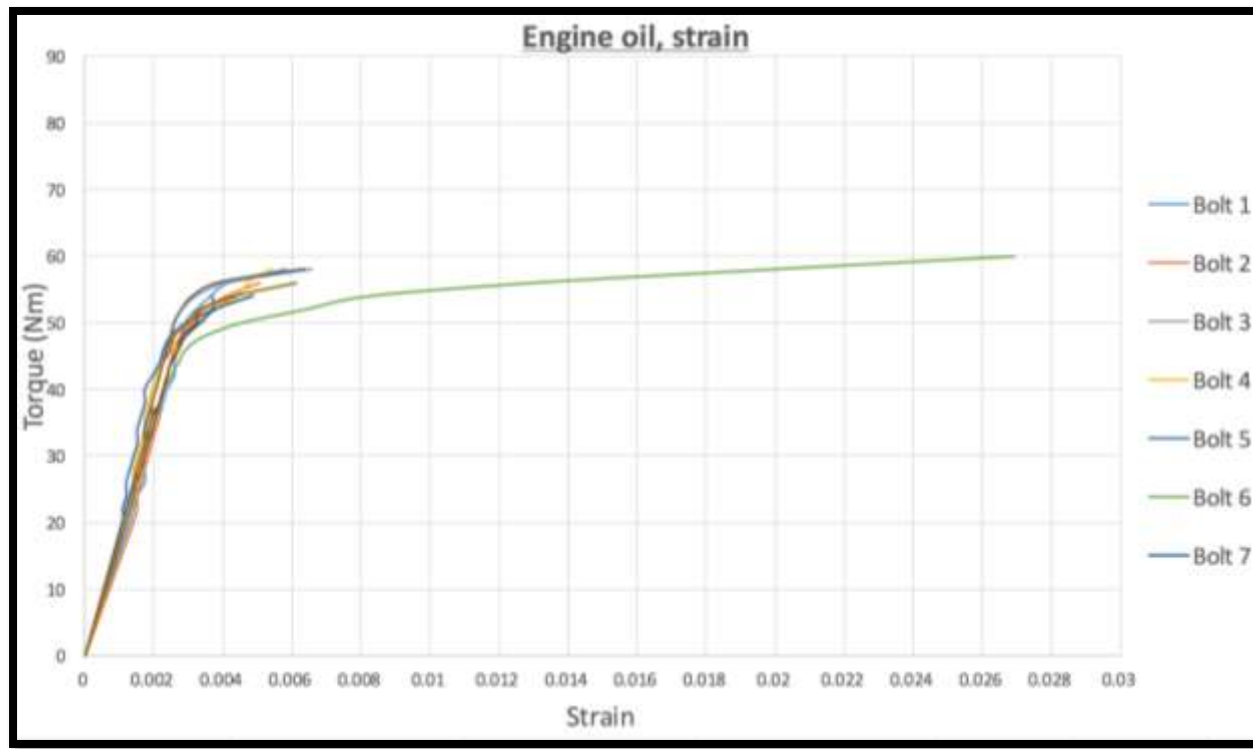
2.3.2 Torque strain results

With the results of 10 bolts on one graph it makes it possible to see the deviation present between bolts and highlights some important aspects of each lubricant.

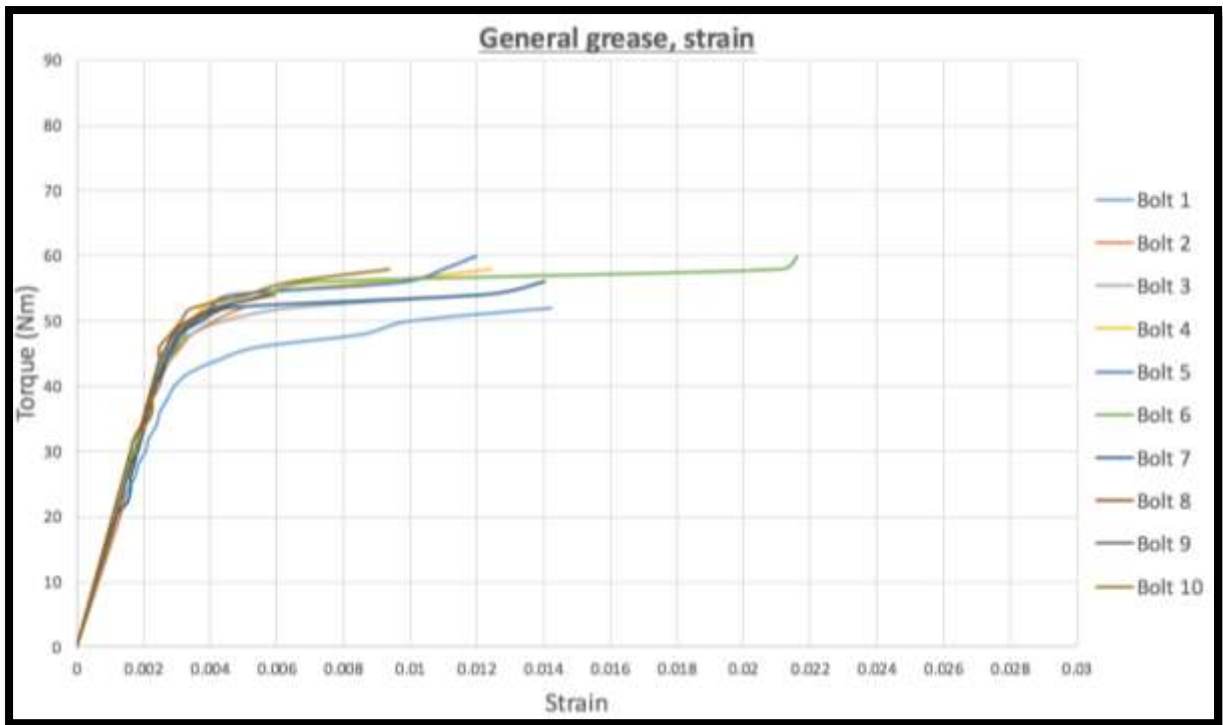
Below in graphs 2,3,4,5 and 6 is the data of the 10 bolts of each lubrication method depicted together on graphs.



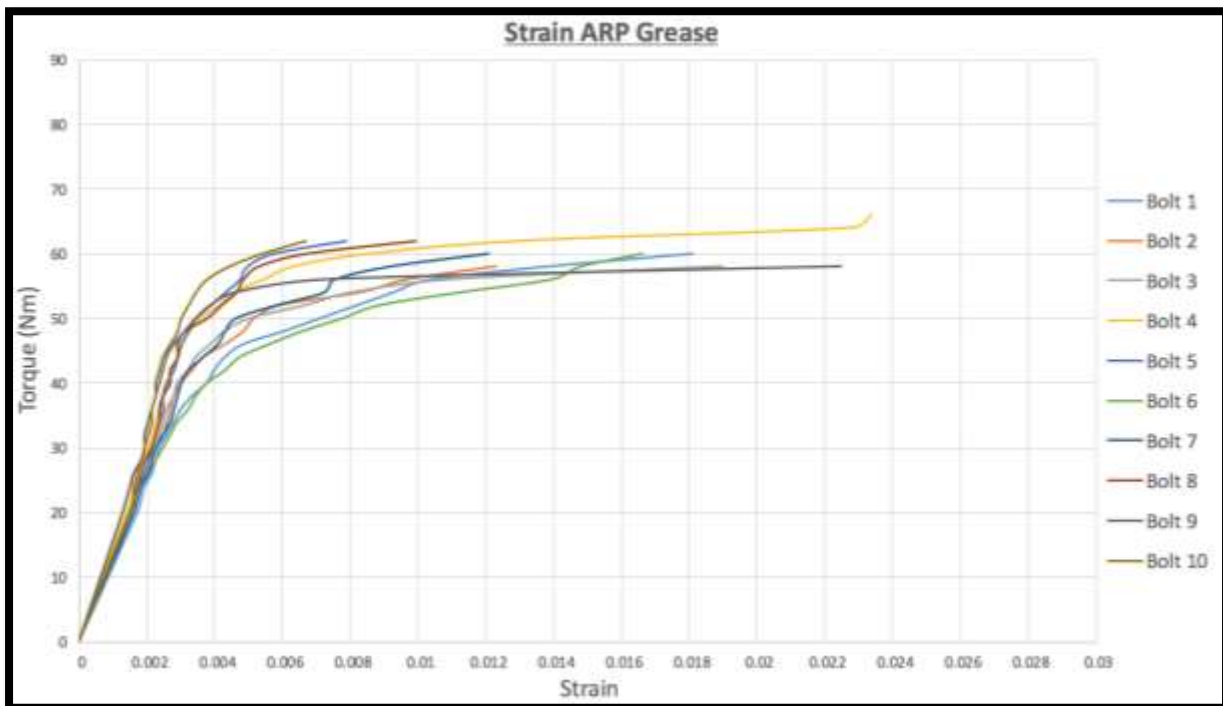
Graph 2



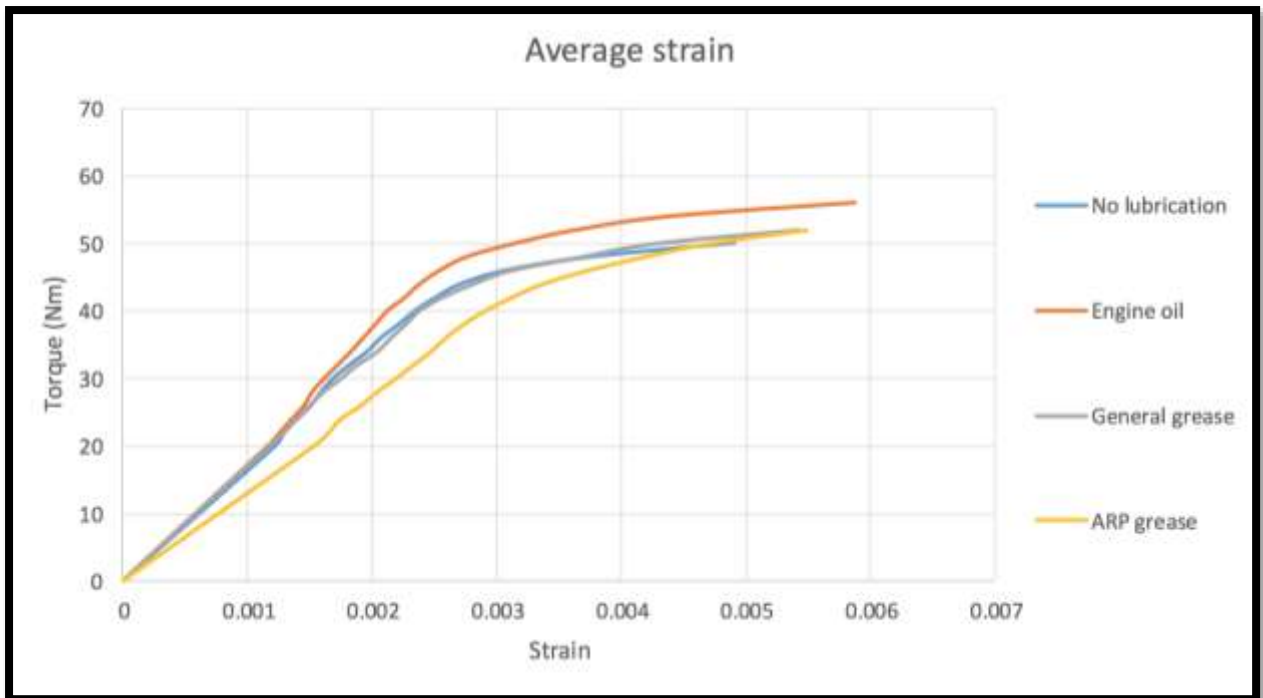
Graph 3



Graph 4



Graph 5



Graph 6

2.3.3 Initial analysis

It can be seen from graph 6 above that the General grease had almost no effect at all on the gradient (which is essentially the nut factor) and engine oil appears to have been slightly worse than no lubricant. This could be down to the small amount of samples used and would need to be repeated with many more samples to definitively show that engine oil increased friction. However, it is clear that the effect both engine oil and grease had was small and the specialised ARP grease did have a large effect.

It can be seen in graph 5 that the gradient of the torque-strain relationship in its linear section is less constant across the bolts with the ARP grease than the other three lubricants. When the order that the bolts were tested in is taken into account, it can be seen that bolts 1 and 6 had the lowest gradient followed by bolts 2 and 7 and the bolts with the highest gradient are bolts 5 and 10. Bolts 1 and 6 were tested whilst the con-rod was fresh, followed by 2 and 7, and finishing with 5 and 10.

It is clear that the gradient with the ARP grease increased with repeated use of the con-rod joint, being lowest on first use. Looking at figure 4 below showing the surface after uses, this is likely due to the lubricant performing very well on a smooth undamaged surface and losing performance as the quality of the mating surfaces dropped with repeated use to the point that bolts 9 and 10 had gradients very similar to using no lube.



Figure 4

This highlights the factors that can affect friction and that a quality lubricant may not be of any benefit at all if the surface quality is low. This would be an interesting area of further research. The other three tests were not seen to be dependent on previous usage of the con-rod hole, adding reason to believe the oil and general grease had very little effect on the friction to begin with.

2.3.4 Effect on combined friction

	No lubrication					
	Total coeff friction	Nut factor	Axial stress at yield (N/mm ²)	Percentage of torsion %	Shear stress (N/mm ²)	Thread friction
Bolt 1						
Bolt 2	0.249714883	0.165719789	785.072449	23.55672357	382.2696473	0.190998476
Bolt 3	0.320660199	0.207149737	752.3610969	26.74186008	403.6011975	0.214238874
Bolt 4	0.281246135	0.184133099	785.072449	23.55672357	382.2696473	0.190998476
Bolt 5	0.281246135	0.184133099	785.072449	23.55672357	382.2696473	0.190998476
Bolt 6	0.281246135	0.184133099	752.3610969	26.74186008	403.6011975	0.214238874
Bolt 7	0.249714883	0.165719789	752.3610969	26.74186008	403.6011975	0.214238874
Bolt 8	0.264650739	0.174441883	785.072449	23.55672357	382.2696473	0.190998476
Bolt 9	0.281246135	0.184133099	719.6497449	29.9269966	423.0182205	0.238343578
Bolt 10	0.29979393	0.194964458	719.6497449	29.9269966	423.0182205	0.238343578
Average	0.278835464	0.182725339	759.6302863	26.03405197	398.4354025	0.209266409
Max	0.320660199	0.207149737	785.072449	29.9269966	423.0182205	0.238343578
Min	0.249714883	0.165719789	719.6497449	23.55672357	382.2696473	0.190998476
Deviation + (%)	14.99979055	13.36672704	3.349282297	14.9532798	6.169837776	13.89480952
Deviation - (%)	10.44364307	9.306618367	5.263157895	9.515723507	4.057308936	8.729510499

Table 2

	Engine oil					
	Total coeff friction	Nut factor	Axial stress at yield (N/mm ²)	Percentage of torsion %	Shear stress (N/mm ²)	Thread friction
Bolt 1						
Bolt 2	0.264650739	0.174441883	785.072449	23.55672357	382.2696473	0.190998476
Bolt 3	0.320660199	0.207149737	719.6497449	29.9269966	423.0182205	0.238343578
Bolt 4	0.320660199	0.207149737	686.9383929	33.11213312	440.7738061	0.263608958
Bolt 5	0.371335425	0.236742556	654.2270408	36.29726964	457.0616221	0.290348035
Bolt 6	0.281246135	0.184133099	752.3610969	26.74186008	403.6011975	0.214238874
Bolt 7	0.29979393	0.194964458	752.3610969	26.74186008	403.6011975	0.214238874
Bolt 8	0.281246135	0.184133099	752.3610969	26.74186008	403.6011975	0.214238874
Bolt 9	0.320660199	0.207149737	752.3610969	26.74186008	403.6011975	0.214238874
Bolt 10	0.29979393	0.194964458	686.9383929	33.11213312	440.7738061	0.263608958
Average	0.302469763	0.196527065	732.7342857	28.65294199	414.0571539	0.229486198
Max	0.371335425	0.236742556	785.072449	36.29726964	457.0616221	0.290348035
Min	0.264650739	0.174441883	654.2270408	23.55672357	382.2696473	0.190998476
Deviation + (%)	22.7677839	20.46308047	7.142857143	26.67903228	10.38611885	26.52091421
Deviation - (%)	12.50340645	11.23773018	10.71428571	17.78602152	7.677081844	16.7712578

Table 3

	General grease					
	Total coeff friction	Nut factor	Axial stress at yield (N/mm ²)	Percentage of torsion %	Shear stress (N/mm ²)	Thread friction
Bolt 1						
Bolt 2	0.223916586	0.150654354	817.783801	20.37158705	358.6821503	0.168322745
Bolt 3	0.264650739	0.174441883	785.072449	23.55672357	382.2696473	0.190998476
Bolt 4	0.281246135	0.184133099	785.072449	23.55672357	382.2696473	0.190998476
Bolt 5	0.281246135	0.184133099	752.3610969	26.74186008	403.6011975	0.214238874
Bolt 6	0.264650739	0.174441883	785.072449	23.55672357	382.2696473	0.190998476
Bolt 7	0.281246135	0.184133099	752.3610969	26.74186008	403.6011975	0.214238874
Bolt 8	0.264650739	0.174441883	785.072449	23.55672357	382.2696473	0.190998476
Bolt 9	0.281246135	0.184133099	752.3610969	26.74186008	403.6011975	0.214238874
Bolt 10	0.264650739	0.174441883	785.072449	23.55672357	382.2696473	0.190998476
Average	0.267215482	0.175939617	775.2590434	24.51226452	388.4435177	0.198027062
Max	0.281246135	0.184133099	817.783801	26.74186008	403.6011975	0.214238874
Min	0.223916586	0.150654354	752.3610969	20.37158705	358.6821503	0.168322745
Deviation + (%)	5.250688514	4.656985478	5.485232068	9.095836742	3.902157995	8.186664883
Deviation - (%)	16.20373767	14.37155734	2.953586498	16.89226824	7.661697533	15.00013022

Table 4

	ARP grease					
	Total coeff friction	Nut factor	Axial stress at yield (N/mm ²)	Percentage of torsion %	Shear stress (N/mm ²)	Thread coeff friction
Bolt 1	0.184226899	0.127476761	915.9178571	10.81617749	268.2184924	0.099919593
Bolt 2	0.202418006	0.138099824	850.4951531	17.18645053	332.3587392	0.145881693
Bolt 3	0.180851038	0.144104165	883.2065051	14.00131401	302.5862243	0.123270301
Bolt 4	0.23620149	0.157828371	817.783801	20.37158705	358.6821503	0.168322745
Bolt 5	0.223916586	0.150654354	850.4951531	17.18645053	332.3587392	0.145881693
Bolt 6	0.176141963	0.122755399	883.2065051	14.00131401	302.5862243	0.123270301
Bolt 7	0.19295863	0.132575831	883.2065051	14.00131401	302.5862243	0.123270301
Bolt 8	0.223916586	0.150654354	817.783801	20.37158705	358.6821503	0.168322745
Bolt 9	0.264650739	0.174441883	785.072449	23.55672357	382.2696473	0.190998476
Bolt 10	0.249714883	0.165719789	817.783801	20.37158705	358.6821503	0.168322745
Average	0.213499682	0.146431073	850.4951531	17.18645053	329.9010742	0.145746059
Max	0.264650739	0.174441883	915.9178571	23.55672357	382.2696473	0.190998476
Min	0.176141963	0.122755399	785.072449	10.81617749	268.2184924	0.099919593
Deviation + (%)	23.95837624	19.129007	7.692307692	37.06566999	15.87402323	31.04880973
Deviation - (%)	17.49778687	16.16847656	7.692307692	37.06566999	18.69729644	31.44267936

Table 5

The tables 2,3,4 and 5 above showing the calculated values confirm what was seen on the torque-strain graphs. The total coefficient of friction and the nut factor are both essentially the gradient of the line from the previous graphs with the nut factor showing the true gradient.

Considering the average of each lubricant, the effect that each one has on the coefficient of friction compared to no lubricant can be calculated:

- Engine oil_+8%
- General grease_-4%
- ARP grease_-23%

When the surfaces were fresh the ARP grease decreased the coefficient of friction by 35% although there were only two samples under these conditions to show this.

2.3.5 Ratio of axial to tensile stress

As expected, when the coefficient of friction decreased there was a rise in the axial stress at the yield point showing a corresponding drop in shear stress.

The average axial stress for no lubrication was 759N/mm² which equates to a clamping force of 29715N.

The engine oil slightly decreased the axial stress to 732N/mm² giving a clamp force of 28658N, a reduction of 1057N.

The General grease raised the axial stress to 775N/mm², a clamp force of 30341N and an increase of 626N.

Given that the yield point was an estimate from analysing the data, the variance of 1057N and 626N will contain error however, they do correspond with the calculated coefficients of friction.

The ARP grease saw the axial stress raised to 850N/mm² corresponding to a clamp force of 33278N. This is an increase of 3563N, 12% higher than using no grease. This highlights the importance of using the specified lubricant.

As this is the axial stress at the yield point, the yield point has been estimated from looking at the data and graph plot for each bolt. Due some bolts experiencing a slip-stick phenomenon, causing fluctuations in the data, and some bolts yielding very smoothly, making it hard to define the point of yield, the axial stress is approximate. However, with the results averaging over 10 bolts and correlating with the results of the coefficient of friction, it is deemed that they are reasonable.

2.3.6 Calculated individual thread and head coefficient of friction

Using the experimental method proposed, the results for the average thread coefficients of friction are:

- No lubrication_0.209
- Engine oil_0.229
- General grease_0.198
- ARP grease_0.146

Looking at many measured values for coefficient of friction of the thread and bearing faces from the Bosh automotive handbook, the coefficient of friction of the bearing faces is on average 0.566 times the thread friction. (handbook, 1996)

Based on the calculated thread coefficients and the above assumption, the bearing face coefficients should be in the region of:

- No lubrication_0.118
- Engine oil_0.129
- General grease_0.112
- ARP grease_0.083

By using these values in the full torque equation with the measured average preload at 34Nm, the results show:

- No lubrication_38Nm
- Engine oil_39Nm
- General grease_38Nm
- ARP grease_35Nm

These are all within 15% of the measured torque and the ARP grease is within 3% of the measured torque.

A lot more trust can be placed into the total coefficient of friction calculated at the start because it involves less variables and more accurate measurements. Assuming the total coefficient of friction to be accurate, it would seem that the estimated values for the thread coefficient are slightly above what would be expected in reality.

Given the findings it would not be recommended to use these coefficients of friction in calculations, however, being within 15% shows the method of calculation may have some merit when more samples are used and a more accurate method of measuring elongation and yield point is used.

3.0 Measuring bolt preload to angle with load cell

3.1 Information

3.1.1 Ford tightening procedure

An official Ford manual giving the tightening procedure of the con-rod bolts could not be found and it appears that it is the intentions of Ford that the main, big end, and small end bearings are non-serviceable items and the 'bottom end' of the engine is to be replaced in the case of failure.

Because of this, two sources had the most reliable information for the tightening procedure. The first is from a meeting with head lecturer, Malcolm McDonald, in April 2018 who states from correspondence with Ford that the tightening procedure is 'tightening the bolt till snug, followed by a 45-degree angle of turn'.

The second is from the Official Ford, 2013 technical regulations for the Formula Ford competition using 1600cc Duratec engines which are the updated engines of the Ford sigma that the Con-rod and bolts are from. The technical regulations simply state for the con-rod bolts: 'Step 1: snug torque. Step 2: tighten to yield'.

Although this is an updated engine, it is the same capacity and often minimal parts are changed and the tightening procedure of the con-rod bolts likely remains the same.

With no solid information on the tightening procedure and the interest of the bolts being tightened into yield, it was determined that more practical research would be carried out into the tightening of the bolts.

With it clear from multiple sources that the tightening procedure is a snug torque followed by an angle of turn of the bolt, possibly into yield, the practical research will involve tightening the bolts in increments of angle and measuring the preload force through into bolt yield. The aims are to be able to compare the preload force at various angles of turn and to see how the preload force changes as the bolt goes through yield.

3.1.2 Force washer

The method of calculating the preload force in the previous experiment from the extension and stiffness only works whilst the bolt is in its elastic region because the stiffness changes once yielded. One of the interests of the new research is to know how the preload force of the bolt changes through yield and so a new method will be required. After some research it was decided to manufacture a force washer using strain gauges.

A force washer is a larger than usual strain gauged washer that is placed within the joint, as a washer would be, and can continuously measure the preload as the bolt is tightened.

A strain gauge is a small flat sensor whose resistance is extremely sensitive to changes in length. It is glued to the side of a material and then changes in length with the material as it is placed under strain and can measure both compressive and tensile strain. (Omega.co.uk) The change in resistance is linear and must be calibrated to a known force before use.

Given access to strain gauges, it was decided to manufacture a force washer that would be inserted into the con-rod and would measure the preload. To keep the force across the washer uniform, a section of the middle of the con-rod was milled out and a mild steel

washer with the strain gauge on placed in. Care was taken to machine the gap in the rod and the washer ends flat to avoid non-parallel faces.

Most force washers use at least four strain gauges and average the result from all of them in case the loading isn't evenly distributed across the top of the washer. The equipment available only made it possible to use one strain gauge. The con-rod and force washer are shown below in figures 5, 6 and 7

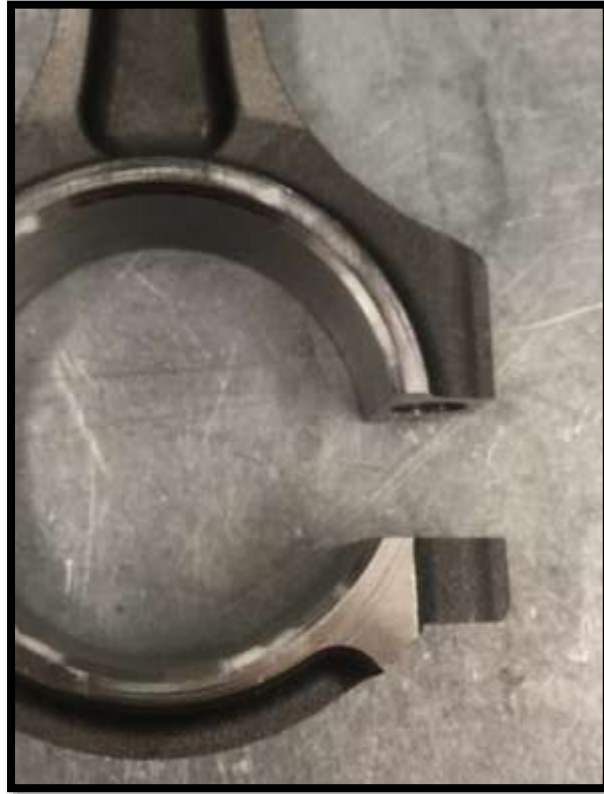


Figure 5

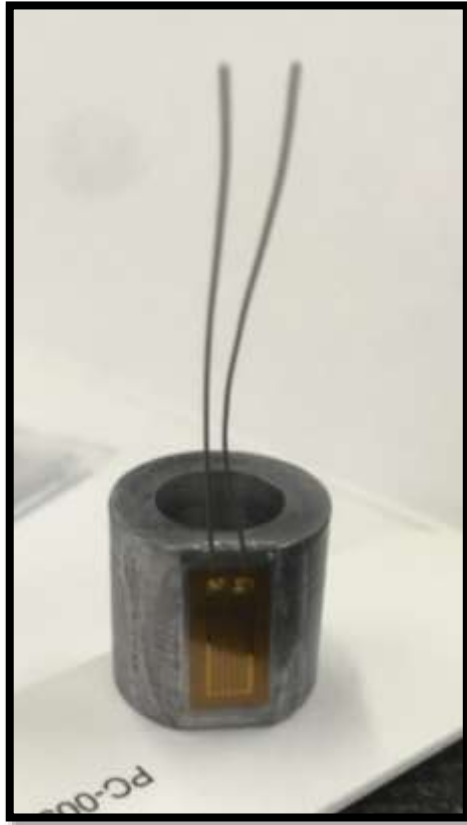
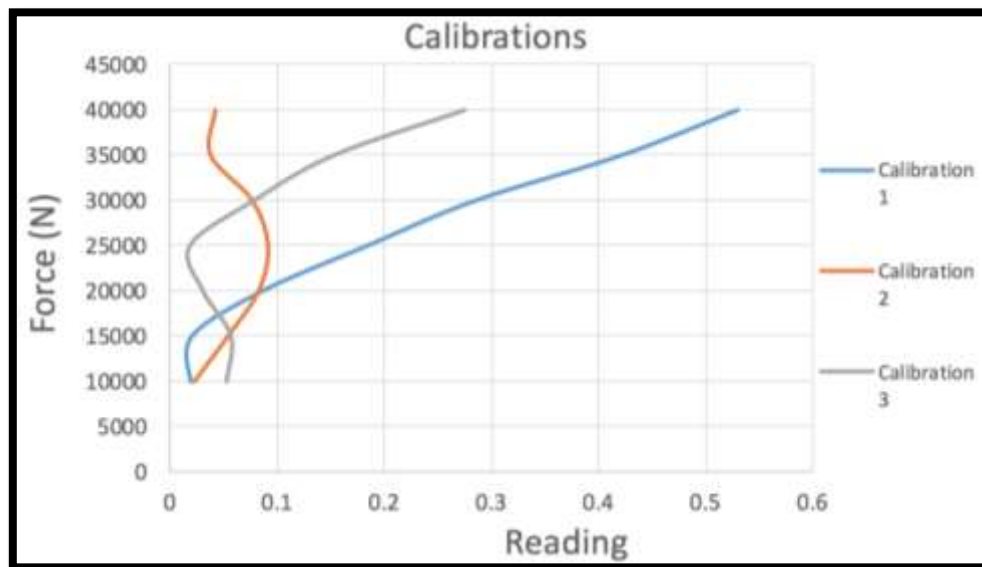


Figure 6



Figure 7

An attempt was made to calibrate the force washer in a hydraulic press with only one strain gauge to see if consistent results could be gained if care was taken to ensure an evenly distributed load. The washer was turned 45 degrees between calibrations and after just 3 calibrations it was clear the load was not even as shown below in graph 7:

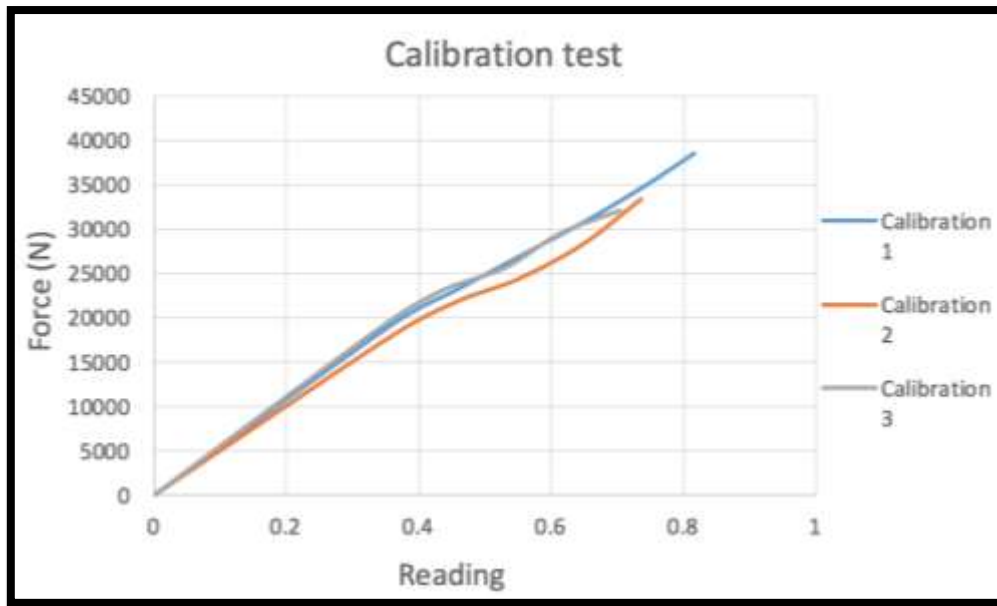


Graph 7

As the equipment for a full force washer was not available, it was decided that the single gauge washer would be calibrated in-place in the con-rod and would always be used in the same orientation for consistency.

For calibration the change in length of the bolt was measured and multiplied by the stiffness whilst in the elastic region to calculate the force and this was plotted against the strain gauge reading. This would give the gradient of the strain gauge reading to force which is a linear relationship and so can be used to calculate the force once the bolt yields.

To check the load was distributed the same each time, the washer was removed and reinserted in the same orientation between the calibrations. The results of the first three calibrations are shown below in graph 8:



Graph 8

This shows that the force washer will give quite a good accuracy once a linear trend line is fitted to each line and an average taken and so the experiment could go ahead.

3.1.3 Research

The turn of nut control was first used in structural steel applications in the 1950's where the nut would be tightened with a torque expected to stretch the bolt to a minimum of 75% of its ultimate tensile strength. The nut would then be turned 180 degrees to guarantee the bolt is well into yield, producing as much tension as the bolt could create. (Bethlehem steel corp, 1977)

This worked well in structural steel applications where the bolts were of a lower strength grade giving them very good ductility and there were no fluctuating loads on the bolt.

This method was adapted by the automotive industry where a much lower snugging torque would be used that created just enough preload to pull the joint fully together, overcoming any bending due to misaligned surfaces etc, followed by a turn of the bolt intended to take the fastener to a defined percentage of yield or a set amount over yield.

Through this method more accuracy of the desired preload can be obtained as turn of the nut is much less sensitive to the friction than torque tightening and can be $\pm 5\%$ as opposed to $\pm 25\%$ when torque tightening alone. It is not however perfect because differences in friction will result in more torsional stress in some bolts than others and hence lower tensile stress is available. (Bickford, 2008)

Snug torques vary depending on each individual application but from research into the ford tightening procedure, almost all anecdotal evidence found from forums and individuals stated between 10 and 20Nm. From current knowledge, this seems very reasonable and so 15Nm was used in the tests.

3.2 Experiment

3.2.1 Aims

The aims of the experiment are:

- Determine the average clamping force produced from angle of turn of the bolt.
- Determine how the preload force changes as the bolt passes through yield.

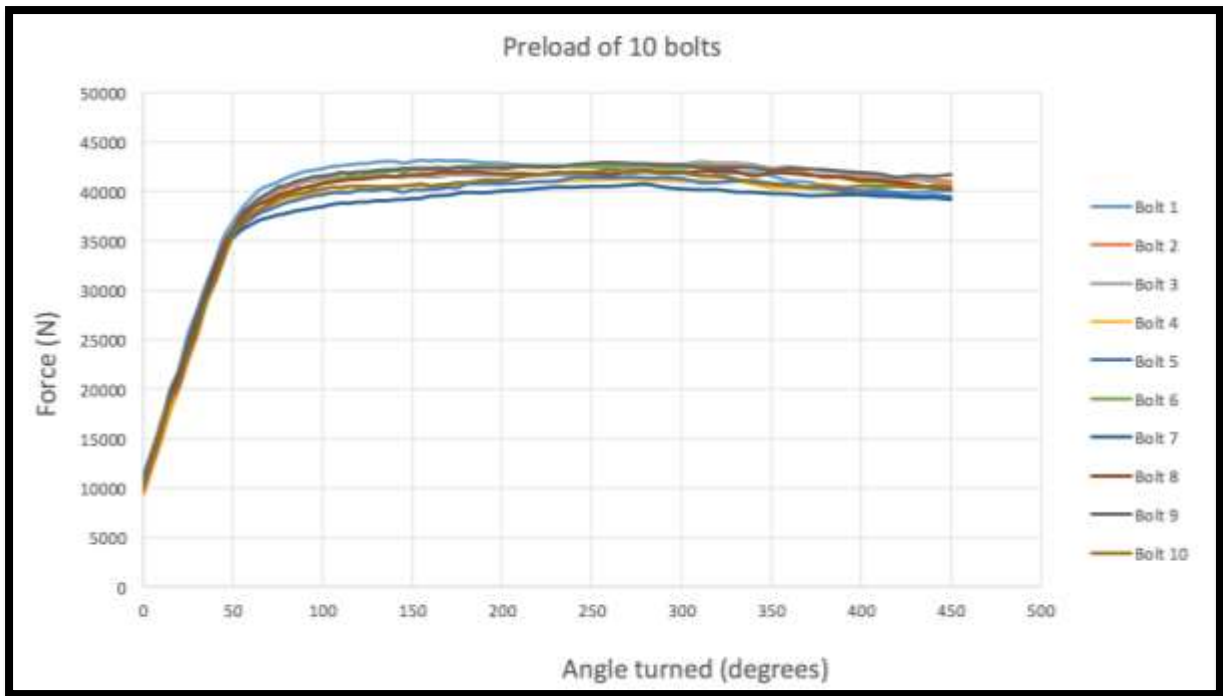
3.2.2 Method

- All bolts and the con-rod were cleaned in a degreasing parts washer.
- The modified con-rod was held in a vice and the force washer inserted into the slot.
- The strain gauge was connected to a wheatstone bridge and then to the reader.
- 6 bolts had their ends ground flat to increase accuracy of length measurements for calibration use.
- 3 bolts were tightened to 36Nm in 4Nm increments, the length measured and the reading recorded.
- The bolt force was then calculated from the change in length and used to calibrate the force washer.
- Using the specialised ARP fastener assembly grease and a large angle gauge, 10 bolts were torqued to 15Nm and tightened in 5 degree increments to 180 degrees and then 10 degree increments to 450 degrees with the force washer reading recorded at each.
- 3 more of the ground calibration bolts were tightened to 36Nm in 4Nm increments, the length measured and the reading recorded.
- The preload was then calculated using the combined average of the 6 calibrations and plotted on graphs.

3.3 Results and analysis

3.3.1 Ford tightening procedure

Graph 9 below shows the force attained from angle of turn from the 10 bolts tested and it can be seen that the gradient whilst in the elastic region is very consistent across the bolts. The bolts quite consistently yield between 45 to 50 degrees of rotation and then vary more significantly in the force they produce through yield.



Graph 9

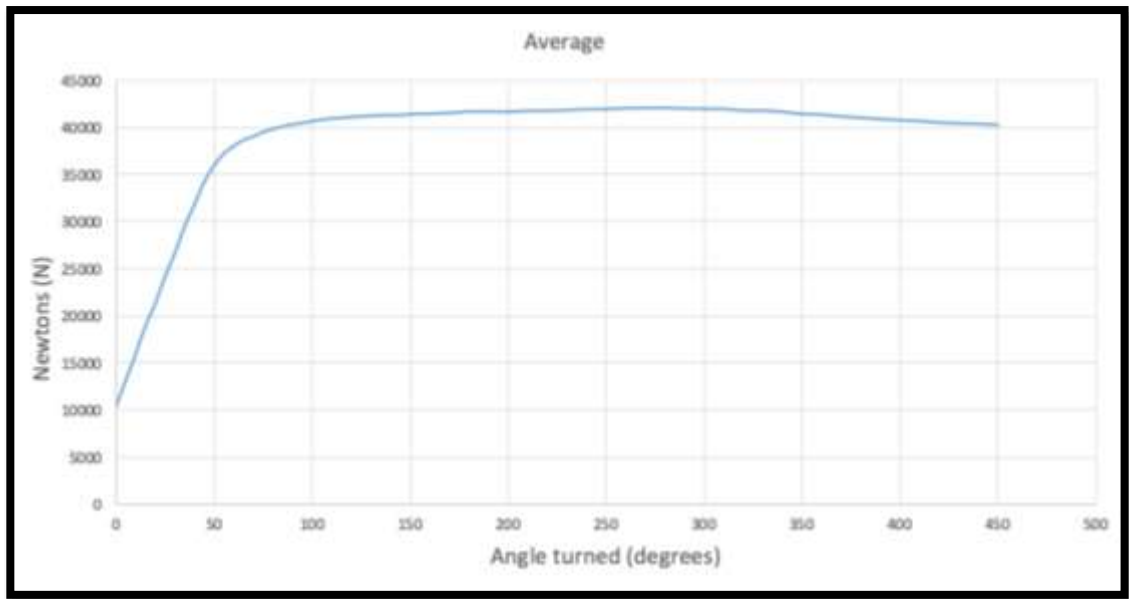
This data shows good consistency with both of the suggested tightening procedures for the Ford bolts, the first being a snug torque followed by a 45 degree turn and the second being essentially tighten to yield.

The 15Nm snug torque followed by 45 degree rotation puts the bolts right up to their yield point and possibly past it if small errors or different snug torques were to be used.

3.3.2 Turn of nut method

Upon closer inspection of the figures of the ten bolts, the maximum and minimum force produced at 40 degrees of rotation, before yield, is 32891N and 30813N, a difference of $\pm 3.5\%$ of the average force which is a very precise scatter in the fastener world.

Due to the small variations from each increment, the yield point cannot be reliably calculated using the 0.2% offset yield method or the average gradient method and so was again determined for each bolt individually and subjectively. However, on this occasion the 'noise' in the data was much smaller and the yield point clearly fell between 45 and 50 degrees of rotation for every bolt with the average yield point coming out a 47 degrees.



Graph 10

The data of the preload as the bolt yields will be used in the design calculations of the upgraded bolt.

4.0 Analysis of Ford connecting rod bolted joint

4.1 Inertia Forces

4.1.1 Information

When an engine is in operation the upwards inertia force from the piston assembly and the main body of the con-rod is reacted by the con-rod cap against the crank pin. This force peaks when the piston is at TDC where the acceleration is maximum and is calculated using Newtons second law of motion, $F=ma$. On the power stroke the gas force acting in the opposite direction greatly overwhelms the inertia force, but on the exhaust stroke there is only a small residual gas force from the cylinder and the cap has to react almost the full inertia force.

4.1.2 Calculating inertia force

The total inertia force will consist of a force due to the reciprocating mass accelerations and a force due to the rotational inertia of the rotating component of the con-rod. The rotating component of the con rod is usually assumed to be two thirds of the mass and the other third is reciprocating. (handbook, 1996)

It can be seen in the Inertia equation below that the inertia force is proportional to the square of the angular velocity and so exponentially increases with engine rpm.

The equation used to calculate the inertia force is shown below:

$$F_{inertia} = M_{recip} \cdot R \cdot \omega^2 \left(1 + \frac{R}{L}\right) + M_{rot} \cdot R \cdot \omega^2$$

Where:

- $F_{inertia}$ = Inertia force (N)
- M_{recip} = Reciprocating mass (Kg)
- M_{rot} = Rotating mass (Kg)
- R = Crank throw (m)
- ω = Angular velocity (rad/s)
- L = Con-rod length (m)

4.1.3 Calculations

Tables 6 and 7 below show the measured component masses and the necessary engine geometries:

Component	Mass (Kg)
Piston and rings	0.272
Gudgeon pin	0.08
Circlips x2	0.0017
Con-rod (whole)	0.372
Con-rod cap	0.105
Con-rod bolts x2	0.035

Table 6

Engine specification	
Bore (m)	0.079
Stroke (m)	0.0814
Crank pin offset (m)	0.0407
Con-rod length (m)	0.137

Table 7

From engine pressure data at 6000rpm, the residual gas force in the cylinder at TDC on the exhaust stroke was found be 2.71 Bar and, using the bore diameter, the force this exerts on the piston is calculated below:

$$2.71 \times 10^5 \times \frac{\pi \times 0.079^2}{4} = \mathbf{1328.4 \text{ Newtons}}$$

The inertia force of interest is the force that the bolts have to react and because the con-rod cap and the bolts themselves are reacted through the crankpin, the mass of these must be removed from the rotational mass component. The Inertia force at 6000rpm was then calculated:

$$0.478 \times 0.0407 \times \left(\frac{6000 \times 2\pi}{60}\right)^2 \times \left(1 + \frac{0.0407}{0.137}\right) + 0.143 \times 0.0407 \times \left(\frac{6000 \times 2\pi}{60}\right)^2$$

$$= \mathbf{12259.7 \text{ Newtons}}$$

Taking into account the residual gas force, this gives an overall inertia force of:

$$12259.7 - 1328.4 = \mathbf{10931 \text{ Newtons}}$$

4.2 Bolt and cap stiffness

4.2.1 Information

The stiffness of a uniformly shaped object placed under a uniformly distributed load can be calculated, given the Young's modulus and the geometry of the object are known. The equation for stiffness is shown below:

$$K = \frac{EA}{L}$$

Where:

K = Stiffness (N/m)

E = Young's modulus (N/m²)

A = Cross sectional area (m²)

L = Length (m)

4.2.2 Bolt stiffness

Tests originally carried out by E.M Slaughter in 1933, and proven in many studies since, showed that a threaded rod will have the same tensile strength as a smooth unthreaded rod that's diameter is equal to the mean of the pitch diameter and minor diameter. (Shigley, Mischke, & Budynas, 2004) This dimension is often called the stress diameter and the cross sectional area found using this dimension is called the stress area of a threaded rod. For calculating the stiffness of the bolt, using the stress area in the calculation will give the most accurate results. The diameters are shown below in figure 8:

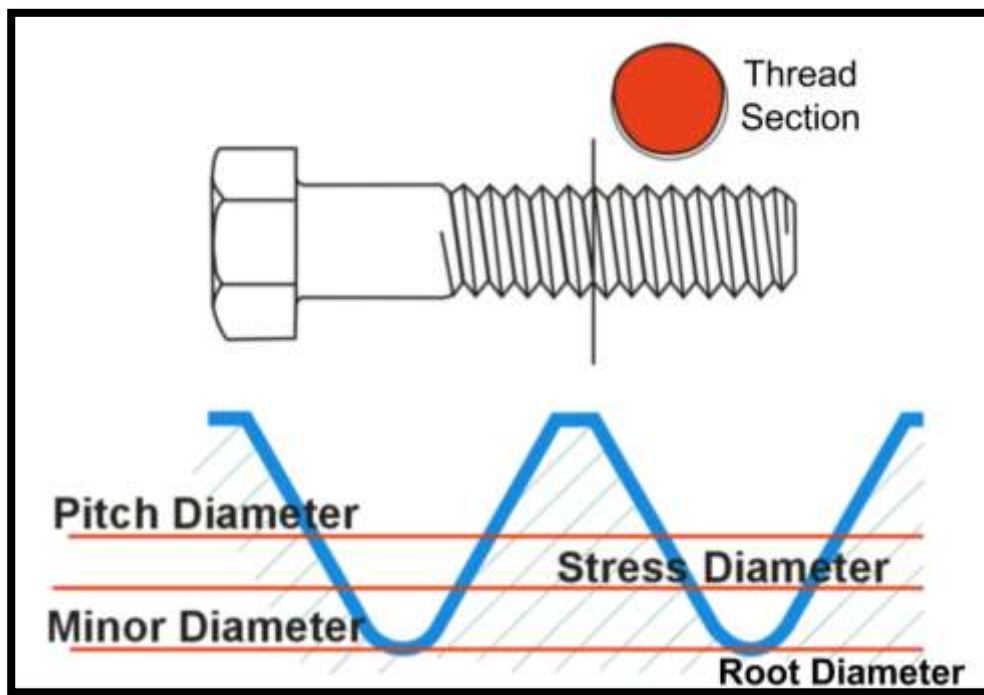


Figure 8

The majority of the tensile load is taken up by the first three threads with the 6th thread seeing almost no load at all, therefore, the force is often assumed to be applied at the second engaged thread. (Bickford, 2008)

Despite being loaded from the threads and the bearing face of the bolt, it cannot be assumed that the engaged length is simply a cylinder of equal length to the grip length. The tensile stress extends into the head of the bolt and through the threads of the bolt. This is a very complex distribution but a simplification, assuming the tensile stress reaches zero at the free ends of the bolt and rises uniformly to reach the body stress at the grip points, gives more accuracy than without. (Bickford, 2008) This is shown below in figure 9:

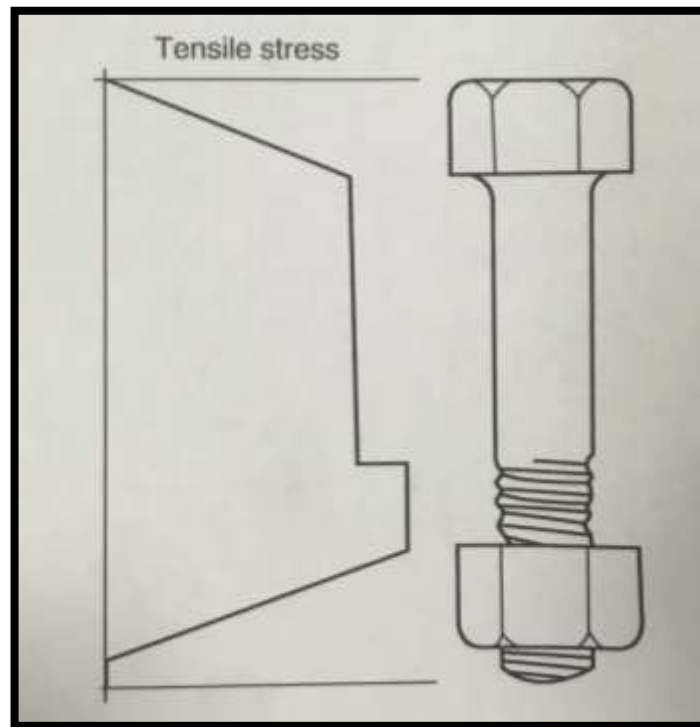


Figure 9

There are many different correction factors suggested from authors but the most common is to add 0.4 of the bolt diameter to the grip length for both the head and the thread meaning 0.8 of the 8mm bolt is to be added to the measured grip length. (Bickford, 2008)

4.2.3 Bolt calculations

The engaged length of the cap and bolt was measured from the bearing face of the rod to the second engaged thread and was found to be 25.2mm.

The major diameter and pitch of the bolt was measured and it was found to be a Metric 8mm fine thread with a pitch of 1mm. The minor and pitch diameters were then found from a 'Zeus precision charts' booklet.

The Young's modulus, or modulus of elasticity, of steel remains fairly constant at around 207GPa regardless of heat treatment, carbon content or alloying. (Shigley, Mischke, & Budynas, 2004)

The required values are shown in table 8 below:

Bolt values	
Pitch diameter (mm)	7.35
Minor diameter (mm)	6.773
Engaged length (mm)	25.2
Youngs modulus (N/mm ²)	207000

Table 8

The stress diameter of the bolt was calculated as the mean of the minor and pitch diameter and used below to calculate the stress area:

$$\frac{\pi \times 7.06^2}{4} = 39.15 \text{ mm}^2$$

The stiffness was then calculated:

$$\frac{207000 \times 39.15}{31.6} = 256457 \text{ N/mm}$$

The calculated bolt stiffness is limited in accuracy due to various factors such as dimensional differences from manufacturing, variation in Young's modulus, and due to localised plastic yielding at stress concentrations. (Bickford, 2008)

4.2.4 Cap stiffness

The load is applied to the cap at both ends of the bolt from the area under the bearing face and from the area of the second engaged thread. Because the cap cross sectional area is larger than where the force is applied, the area of cap put into tension spreads out in a cone like fashion from either end of the bolt encompassing more area the closer it gets to the centre of the engaged length.

On a uniform geometry this is an area of much research with many methods proposed such as Rotchers pressure cone (Shigley, Mischke, & Budynas, 2004) or an equivalent cylinder approach proposed by (Motosh, 1976), however, due to the con-rods complex changing geometry an FEA analysis was carried out to gain a reasonable estimate of the cap stiffness.

4.2.5 Cap Finite element analysis

The FEA analysis of the con-rod was run using the FEA package of SolidWorks. A cross section of the con-rod between the bearing face and second engaged thread was fixed for the simulation and a force of 10000N applied at a diameter of the bearing face of the bolt and second engaged thread to simulate a bolt with a clamp force of 10000N. It was done this way to ensure the pressure cone effect could be taken into account as simply fixing one end with a force at the other would engage all of the fixed surface.

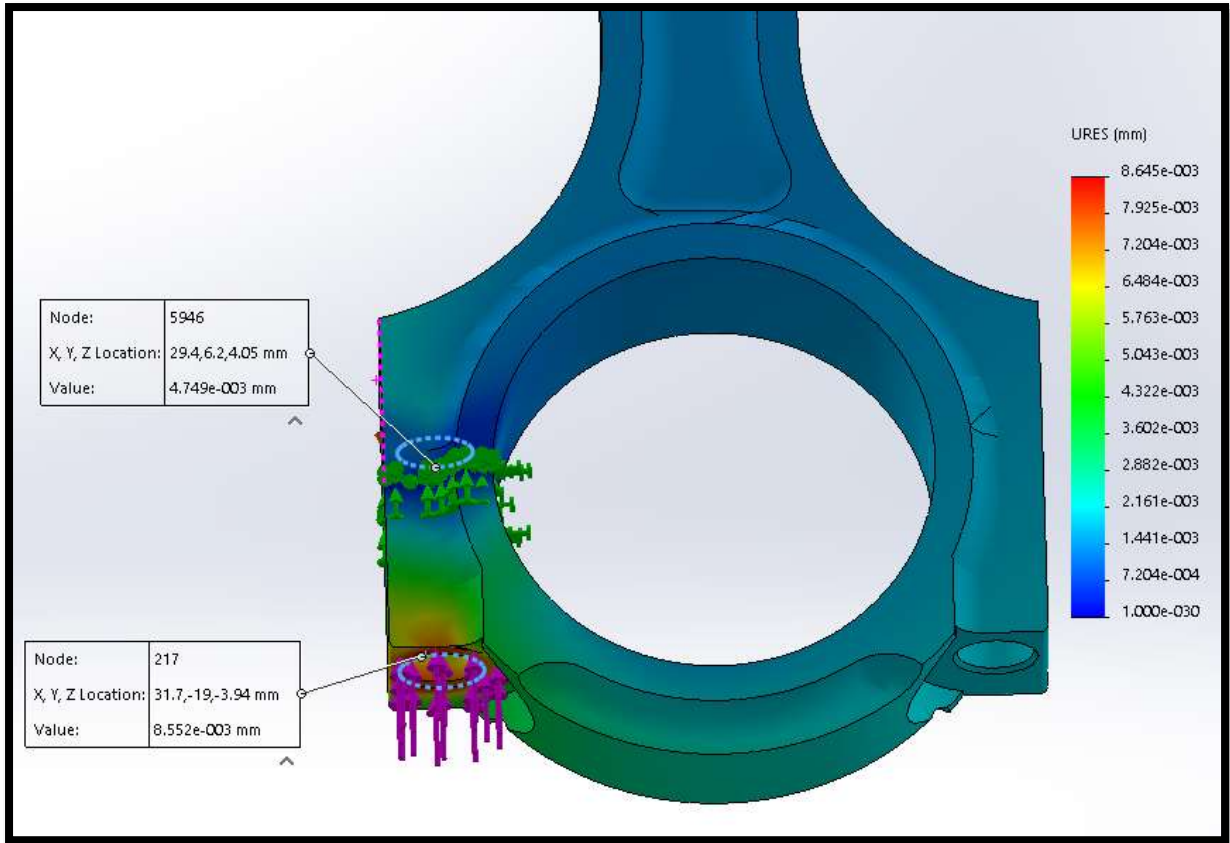


Figure 10

It can be seen from figure 10 above that the maximum deflection at the bearing face was 0.008552mm and 0.004749mm at the thread side. The stiffness was then calculated.

$$\frac{1}{0.008552 + 0.004749} \times 10000 = \mathbf{751823N/mm}$$

4.3 Embedding

4.3.1 Information

The surfaces of the load bearing faces are never perfectly flat and on the microscopic level are made up of many mountains and valleys. Upon tightening, the load bearing faces only contact each other on the high spots, essentially reducing the surface area and thus increasing the stress well beyond the yield stress of the bolt. There is localised plastic deformation on the microscopic level until the contact area has become large enough to support the loads.

This is referred to as embedment of the the contact surfaces and occurs at the bearing faces and the threads. Attempts have been made to formulate equations to define the amount of embedding to be expected from (Meyer & Stelow, 1972), (Prodan, 1974).

(Junker, 1974) proposed the following information in table 9 which is now utilised by the VDI 2230, the worldwide accepted method for the design and calculation of bolted joints.

Average Roughness Height Rz (per DIN 4768)	Typical Manufacturing Process	Main Type of Applied Loading Acting	Guide Values for the Amount of Embedding (in μm)		
			In the thread	Head/nut bearing surface	Interfaces within the joint
<10 μm	Die casting, cold rolling, drawing, polishing, lapping, honing, grinding, smooth milled or turned surface	Tension/Compression	3	2.5	1.5
		Shear	3	3	2
10 μm to <40 μm	Investment casting, forging, milling, planing, shaping, sawing, turning.	Tension/Compression	3	3	2
		Shear	3	4.5	2.5
40 μm to <160 μm	Hot rolling, sand casting, flame cutting.	Tension/Compression	3	4	3
		Shear	3	6.5	3.5

Table 9

The amount of embedding is multiplied by the stiffness to give the force that is lost. This assumes that the joint is tightened to 90% or more of the bolts capacity and it larger than an M5 thread.

These are however just guidelines as many factors affect the amount of embedding and if extreme accuracy is required this must be determined experimentally.

A study carried out by (Bethlehem steel corp, 1977) showed that 5% of the initial preload of structural steel bolts is lost to embedding in the first hour after tightening ad then up to 2% more over the life off the structure.

Further to this, studies carried out by (Bickford, 2008) on studs show that 50% of the torsional stress disappears immediately when the torque is removed and almost all of the remaining torsional stress disappears due to relaxation over the following hour. This is an important and overlooked factor in bolting. If a bolt is tightened until yield and 20% is torsional stress, an hour later 20% of the bolts stress capabilities is now available again to be utilised as tensile stress, potentially from external tensile loading. (Bickford, 2008)

4.3.2 Embedding calculations

From the experiment results on the Ford con-rod, when tightened to the assumed specification of 15Nm and a 45-degree snug torque, the average preload on the joint was 34215N. The maximum and minimum were 35478N and 33231N.

$$F_{pmax} = 35478N$$

$$F_{pmin} = 33231N$$

The embedding losses from these can then be calculated. Table 9 estimates that 3 microns of embedding will occur at the threads, 2.5 microns at the bearing faces, and 1.5 microns at the joint interface giving a total of 0.007mm of loss. This equates to a force loss of:

$$0.007 \times 256457 = \mathbf{1795N}$$

Giving the preload after embedding as:

$$F_{Pmax} = 33683N$$

$$F_{Pmin} = 31436N$$

4.4 Joint behavior under tensile load

4.4.1 Information

When a bolt is tightened, it elongates and provides an axial clamping force onto the joint. The joint experiences this clamping force and becomes compressed until it is providing an equal force opposing the clamping.

Assume a force is applied at the head and thread of the bolt that acts to stretch the bolt very slightly. This force will create a very small change in length of the bolt as it elongates and therefore will also cause an equal change in length of the joint as it 'springs back' from the compression.

Given that the force created from both the bolt and joint is equal to the stiffness multiplied by the change in length, and they have both experienced the same change in length, a joint with a much higher stiffness than the bolt will have a much greater reduction in force than the bolt. Therefore, when an external load attempts to increase the tensile force acting on the bolt, an amount of tensile force will also be removed from the bolt because the joint is relaxing. It can then be seen that the amount of force removed from the bolt due to the joint relaxing is dependent on the ratio of stiffness between the bolt and joint. (Bickford, 2008)

An intuitive way of understanding this is to imagine your hands with a stiff spring in between them and two other springs on the outside that are pushing your hands together, compressing the spring in the middle of your hands. If you are to push each hand outwards by 1cm (the external tensile force) the two outside springs will compress a combined total of 2cm and therefore increase their force they are applying onto your hands. But at the same time the centre spring will relax by 2cm, reducing the force it is applying to your hands. The difference in stiffness of these springs will dictate how much force is lost from the middle spring when the outside springs compress. Once your hands have opened far enough, the middle spring will reach its free length and fall from your hands, this is the point of joint separation.

Another analogy that is similar to the bolting problem is shown below in figure 11 below:

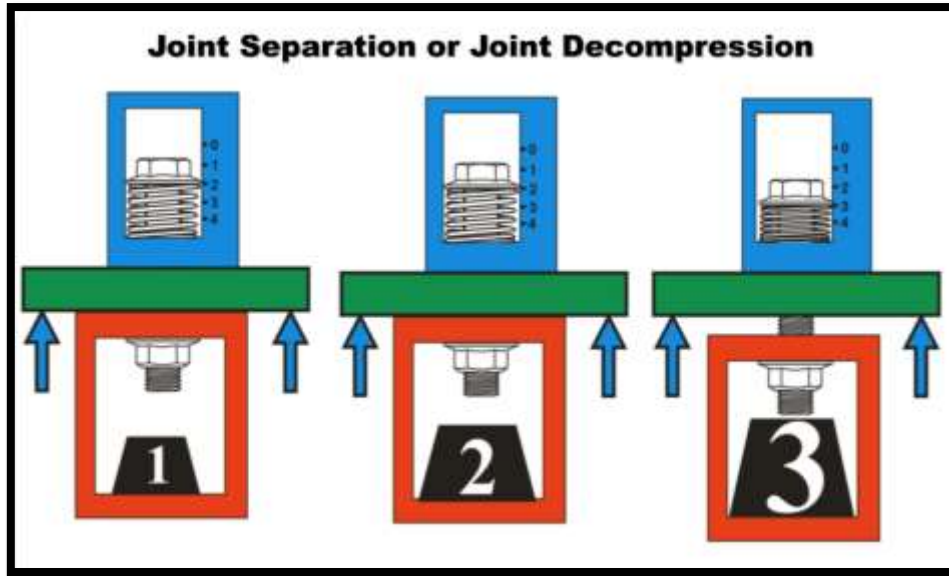


Figure 11

4.4.2 Change in bolt force and load factor

The formula used for determining the change in bolt force is shown below:

$$\Delta F_B = \left(\frac{K_B}{K_B + K_J} \right) \times F$$

Where:

ΔF_B = Change in bolt force (N)

F = External force (N)

K_J = Joint stiffness (N/mm)

K_B = Bolt stiffness (N/mm)

(Bickford, 2008)

The $\left(\frac{K_B}{K_B + K_J} \right)$ section is used often and has its own name, load factor, and its own symbol, (Φ_K). It defines the portion of an external load that will be experienced in the bolt. (Bickford, 2008)

Therefore, the change in bolt force is shown as:

$$\Delta F_B = \Phi_K \times F$$

Where:

ΔF_B = Change in bolt force (N)

F = External force (N)

Φ_K = Load factor

4.4.3 Change in joint force

The change of clamping force in the joint can also be calculated and the equation is shown below:

$$\Delta F_j = (1 - \Phi_K) \times F$$

Where:

ΔF_j = Change in joint force (N)

F = External force (N)

Φ_K = Load factor

When the change in clamp force of the joint is equal to the initial bolt preload, there will be 0 clamp force and the joint will start to separate. At this point the above equations will no longer work and the bolt will now see the full tensile load of any extra applied forces. Substituting ΔF_j for the preload, F_p , and rearranging for F gives you the external force required to cause joint separation which is named F_{crit} and is shown below:

$$F_{crit} = \frac{F_p}{1 - \Phi_K}$$

Where:

F_{crit} = Force required to separate joint (N)

F_p = original bolt preload (N)

Φ_K = Load factor

4.4.4 Loading planes

However, this is a simplified solution as it assumes that the external tensile force is applied at the head and engaged thread of the bolt, whereas in reality this is not the case.

Assume a joint with two equal thickness plates being pulled together by the preload of a bolt and nut. Then assume the separating force is applied to each plate at the joint interface (from the centre). In this scenario the force must first move the plates apart in order to cause any change in length and hence change in force in any of the components. However, the plates have a clamping force equal to the bolt preload pressing them together and this must first be overcome before any separation will occur. (Bickford, 2008)

Going back to the analogy of springs in your hands, this scenario represents having no spring at all in between your hands with the two outside springs pressing your hands together. You must first overcome the force of the springs before they will start to compress and your hands separate.

The reality is that some region of the joint will remain compressed, some region will be able to fully relax and there will be an in between region which will slightly relax. To represent this analytically, a theoretical loading plane is introduced above and below the joint interface and this is considered the plane where the transition occurs from compressed to relaxed. The ratio of the relaxing part of the joint to the full length of the joint is called the loading factor (Bickford, 2008). This is shown below in figure 12:

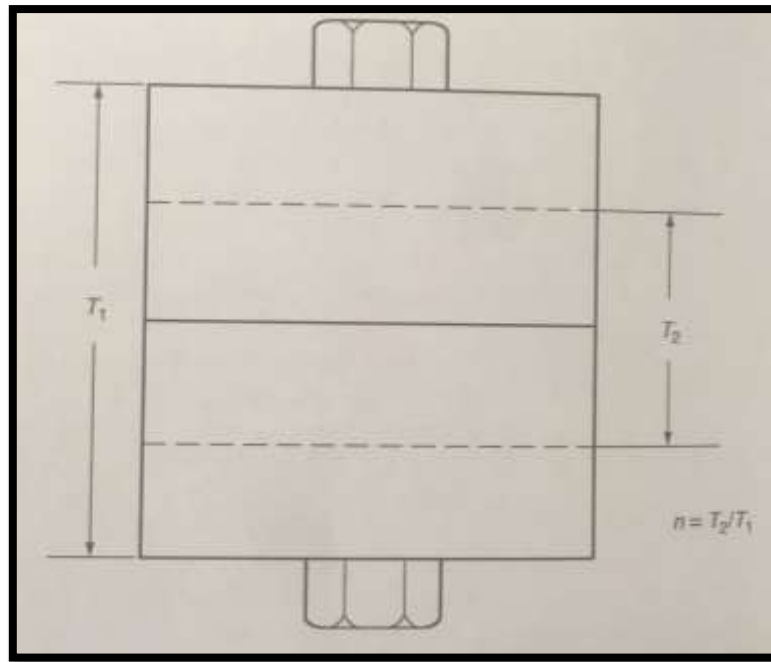


Figure 12

This is a factor that almost entirely depends on the specific application and is most heavily influenced by how the force is applied and the geometry of the part. The most accurate method of determining the loading plane factor is experimentally but requires extremely accurate measuring equipment and a deep knowledge of different values for the specific joint. An FEA analysis can give a good approximation and if this is not available, (Bickford, 2008) recommends using a value of 0.5 as this will get you closer to the truth than ignoring the loading factors all together.

Figure 13 below shows the Ford con-rod and the trend of how the downwards force is applied which helps to highlight how complex the problem can be.

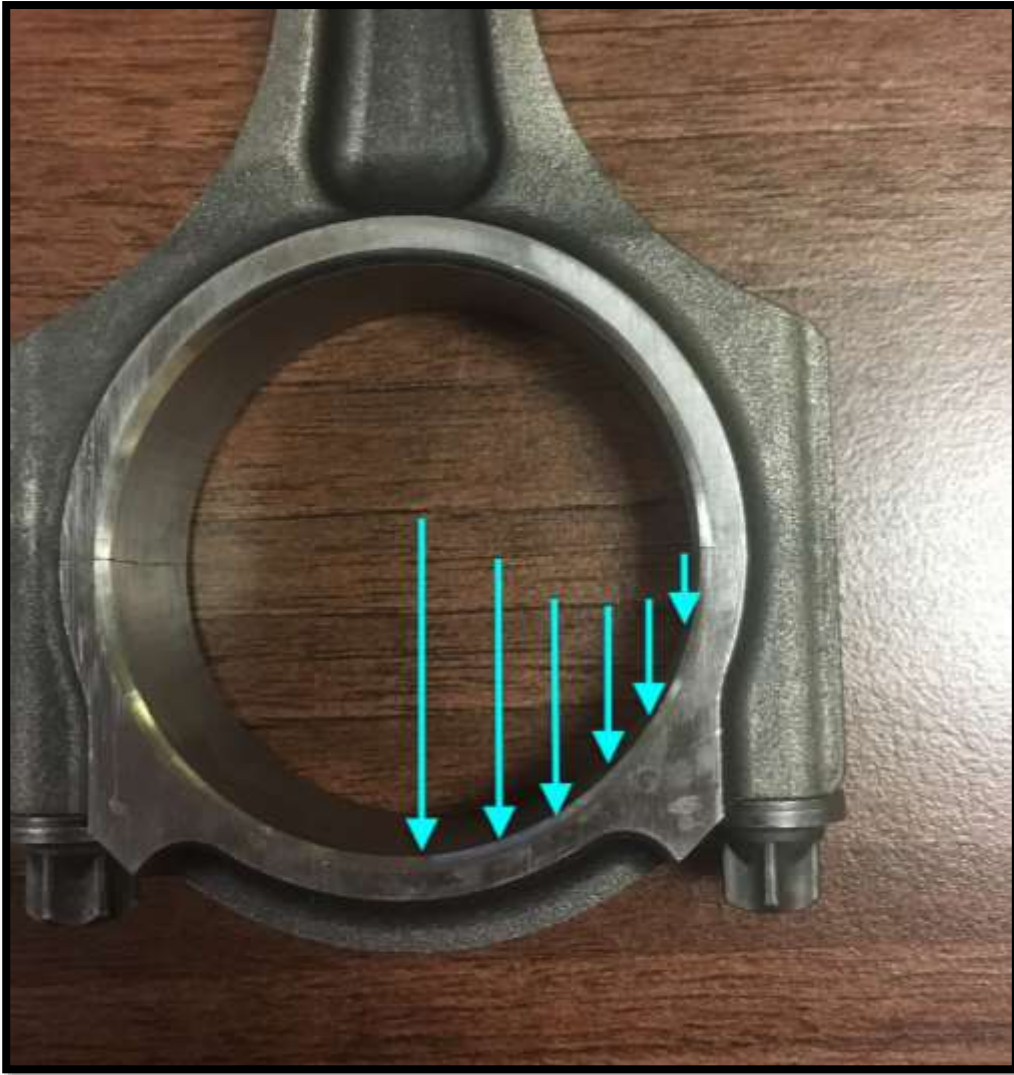


Figure 13

4.4.5 Loading plane corrections

This is denoted by n and this changes the load factor to:

$$\Phi_{KN} = n \times \left(\frac{K_b}{K_b + K_j} \right)$$

The previous equations then need to be corrected and are shown as:

$$\Delta F_B = \Phi_{KN} \times F$$

$$\Delta F_J = (1 - \Phi_{KN}) \times F$$

$$F_{crit} = \frac{F_p}{1 - \Phi_{KN}}$$

Where:

- Φ_{KN} = True load factor
- ΔF_B = Change in bolt force (N)
- ΔF_J = Change in joint force (N)
- F_{crit} = Force required to separate joint (N)
- F_p = Initial preload force (N)
- F = External tensile force (N)

4.4.6 load factor calculations

The load factor then needs to be applied to determine the maximum bolt force under load for each preload scenario. First values for the true load factor must be determined.

From previous calculations the stiffness of the bolt was shown to be 256457N/mm and the stiffness of the cap was 751823N. This gives an initial load factor of:

$$\Phi_K = \frac{256457}{256457 + 751823} = \mathbf{0.25435}$$

However, the loading plane factor needs to be considered. An FEA analysis of the con-rod big end was carried out in SolidWorks with the bolts simulated using the bolt connections option which allows a preload to be applied in the form of a bolt and taken into consideration in the analysis. The previously calculated inertia force of 10931N was then applied to the big end cap through a simulated crank pin and the deformation of the big end analysed. As can be seen in figures 14, 15 and 16 below, there is a section of the con-rod near the bolt that experiences very little deformation, essentially showing that it remains compressed, and how it differs through the cross section of the rod.

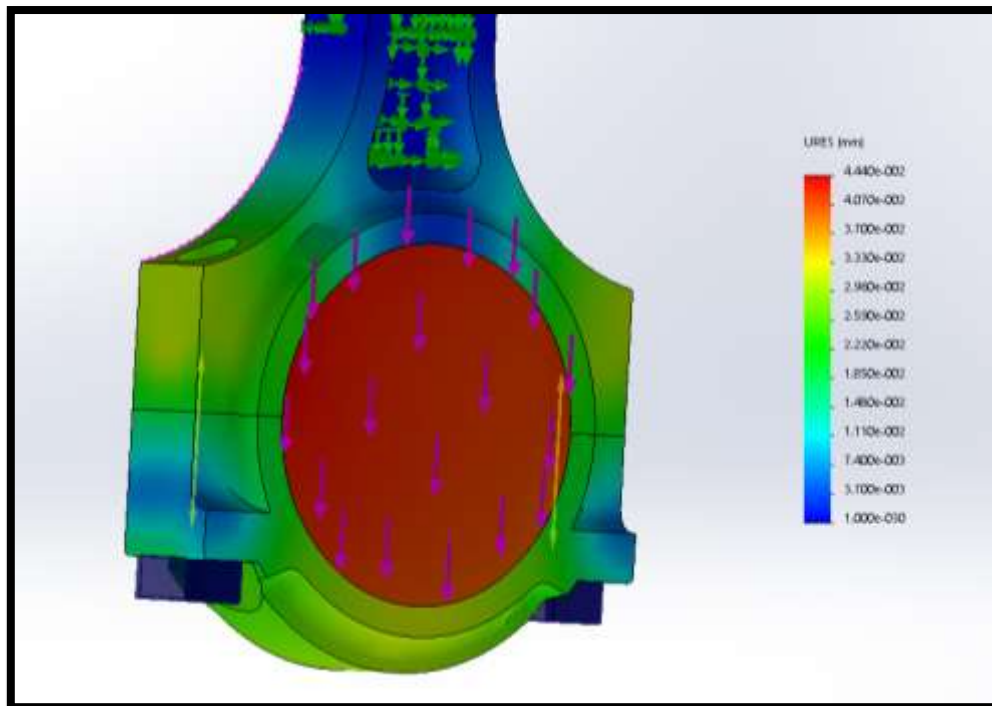


Figure 14

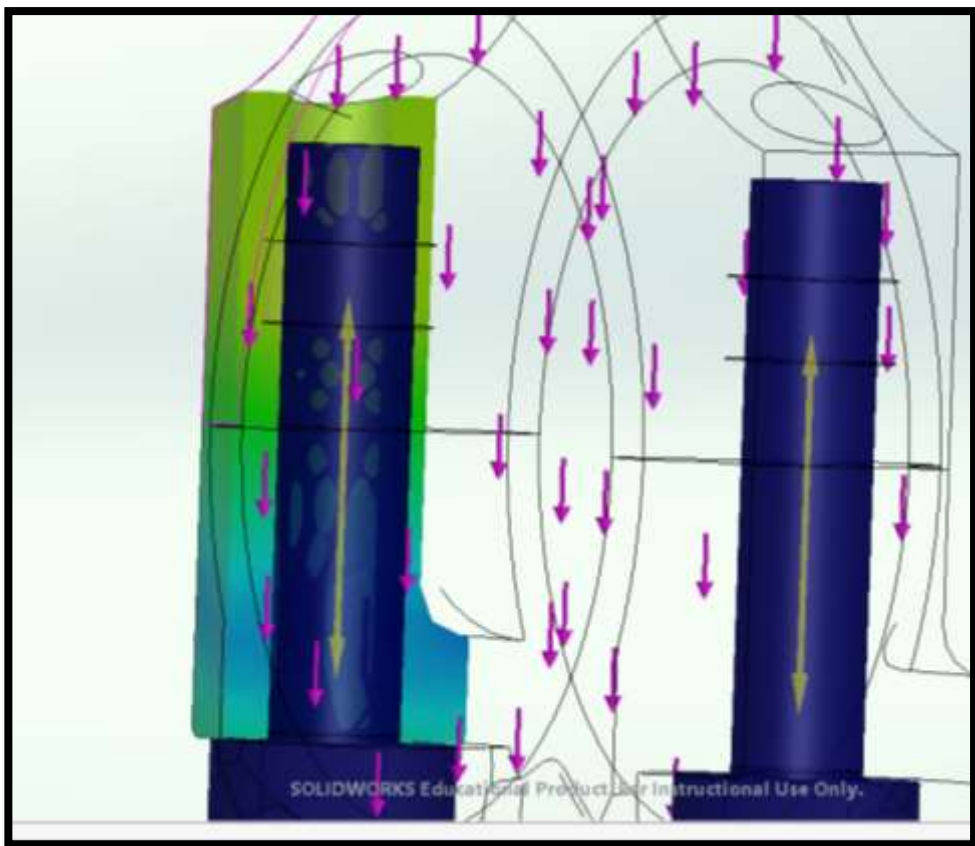


Figure 15

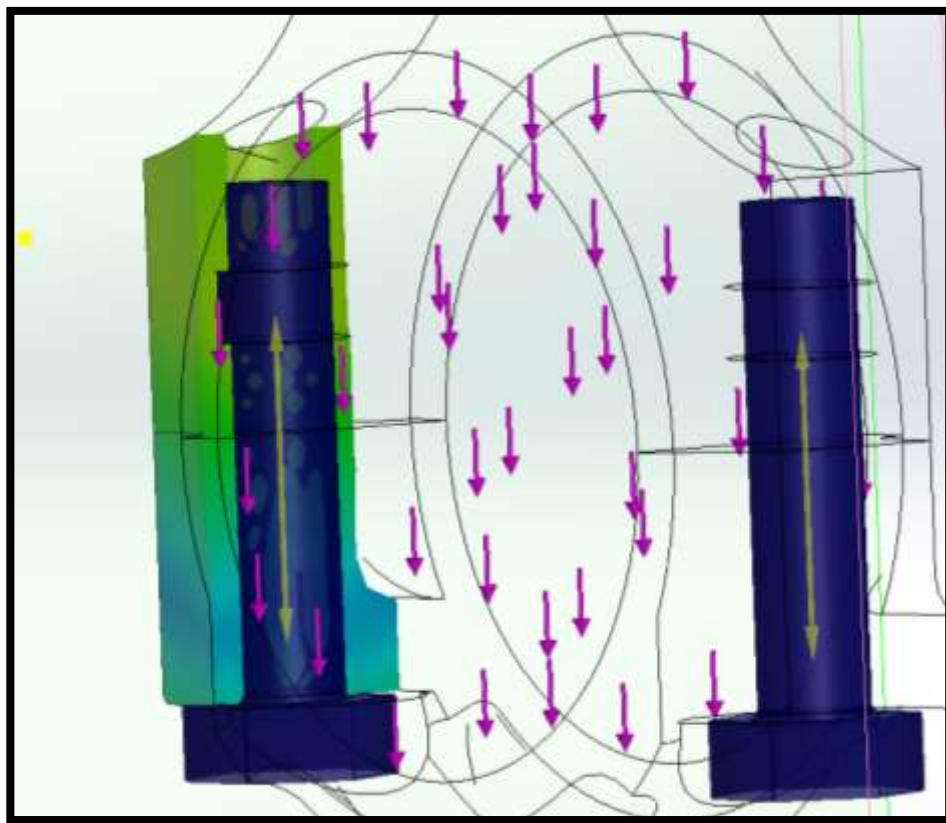


Figure 16

From these results an estimated 50% of the joint remains compressed giving a loading plane factor of 0.5.

Therefore, the true loading plane factor is:

$$\Phi_{K_n} = 0.5 \times 0.25435 = \mathbf{0.1272}$$

4.4.7 Calculating bolt and joint force under service

The maximum bolt force seen is then calculated for both preload scenarios using an inertia force of 5466N as the force is spread across both bolts:

$$\begin{aligned} \text{Max}F_{B_{max}} &= 33683 + (0.1272 \times 5466) = \mathbf{34378N} \\ \text{Max}F_{B_{min}} &= 31436 + (0.1272 \times 5466) = \mathbf{32131N} \end{aligned}$$

The maximum force seen by the bolt is well within its tensile capabilities and will not pose a problem of tensile fracture.

The minimum joint force will be:

$$\begin{aligned} \text{Min}F_{J_{max}} &= 33683 - ((1 - 0.1272) \times 5466) = \mathbf{28912N} \\ \text{Min}F_{J_{min}} &= 31436 - ((1 - 0.1272) \times 5466) = \mathbf{26665N} \end{aligned}$$

In the case with the least amount of bolt preload there is still 26665N clamping the joint together which is a change of 4770N, still more than enough to keep the joint firmly together.

4.5 Fatigue

4.5.1 Information

Fatigue failure of a part is a mechanism of failure that can occur when tensile loads are far below the yield strength of the material but are cyclic in nature. It begins with crack initiation, often starting at an imperfection in the material or a stress concentration, where microscopic tear occurs between the grains.

The crack is then formed at the microscopic tears and begins to spread as the cycling load repeatedly tears more and more material at the leading edge of the crack.

The crack growth increases exponentially with size until there is no longer enough joined material to withstand a cycle of load and the bolt fully ruptures. This can take millions of cycles to occur (Bickford, 2008) (Shigley, Mischke, & Budynas, 2004)

Figure 17 below shows a crack propagating from the root of a thread.

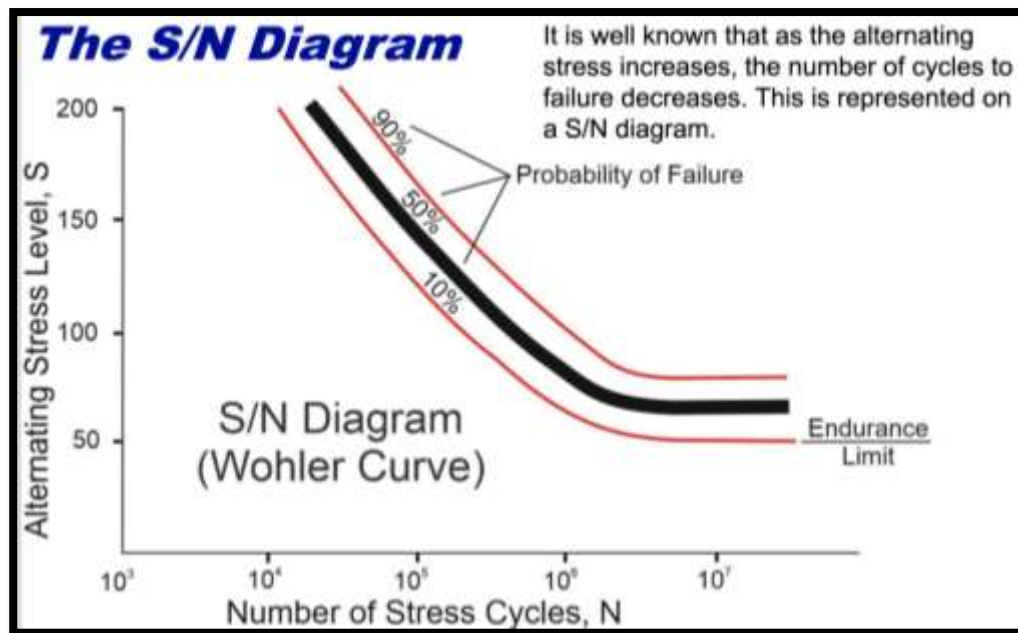


Figure 17

At the point of crack initiation, the crack is too small to be seen with the naked eye and is referred to as a stage 1 crack. At around the point the crack becomes visible to the human eye (if you look closely enough) it is now macroscopic and referred to as a stage 2 crack. Stage 3 is the point of rupture. A telling sign of fatigue is the beach or clamshell marks that propagate out from the point of crack initiation, caused from the surfaces rubbing together under the cycling load. (Shigley, Mischke, & Budynas, 2004)

4.5.2 S/N diagrams

To determine the effects of cyclic loads on materials, fatigue tests are run on smooth polished shafts of different materials where the bar is bent slightly and then rotated about its centre axis placing the side into a repeating compression and tension of equal measure. The bar is run till fatigue failure at different levels of stress and the amount of cycles till failure is plotted on what is referred to as an S-N diagram. An example S-N diagram is shown in graph 11 below:



Graph 11

The stress level at the point of the curve, referred to as the knee, where it becomes almost parallel with the horizontal axis is referred to as the endurance limit or fatigue limit of the material and is where the part will never fail, regardless of the number of cycles and most often falls at around 2×10^6 cycles (Burguete & Patterson, 1995).

It is important to note that this is for a smooth polished bar of material and changes significantly with other geometry and factors.

4.5.3 Mean stress effects

One of the largest of these factors is the mean stress present in the material. The bar test producing the S-N diagram has a zero mean stress. Generally, as the mean stress in a component rises, the endurance limit decreases right up until the point that the mean stress is the UTS of the material and zero alternating force is possible. (Shigley, Mischke, & Budynas, 2004)

Several different methods have been proposed for calculating the endurance limit of smooth polished samples that are free from stress concentrations including Goodman, Gerber, Soderberg and variations of these. (Shigley, Mischke, & Budynas, 2004)

4.5.4 Stress concentration effects

Another factor that heavily influences the fatigue limit is stress concentrations. A stress concentration is caused when there is a sudden change in geometry causing the 'flow' of stress through a part to be concentrated at one point as in figure 18 below. The highest stress concentrations on a bolt are the shoulder where the head and shank meet, the thread runout where the thread meets the shank and the roots of the thread. On a bolt in service, the highest is most often at the first or second engaged thread where the extra deformation from tightening exaggerates the already present stress concentration.

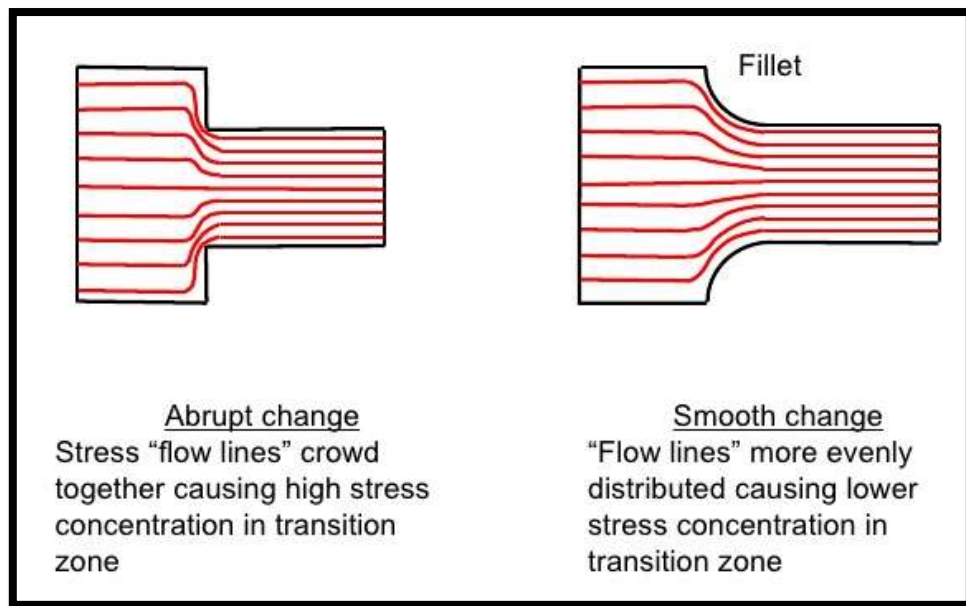
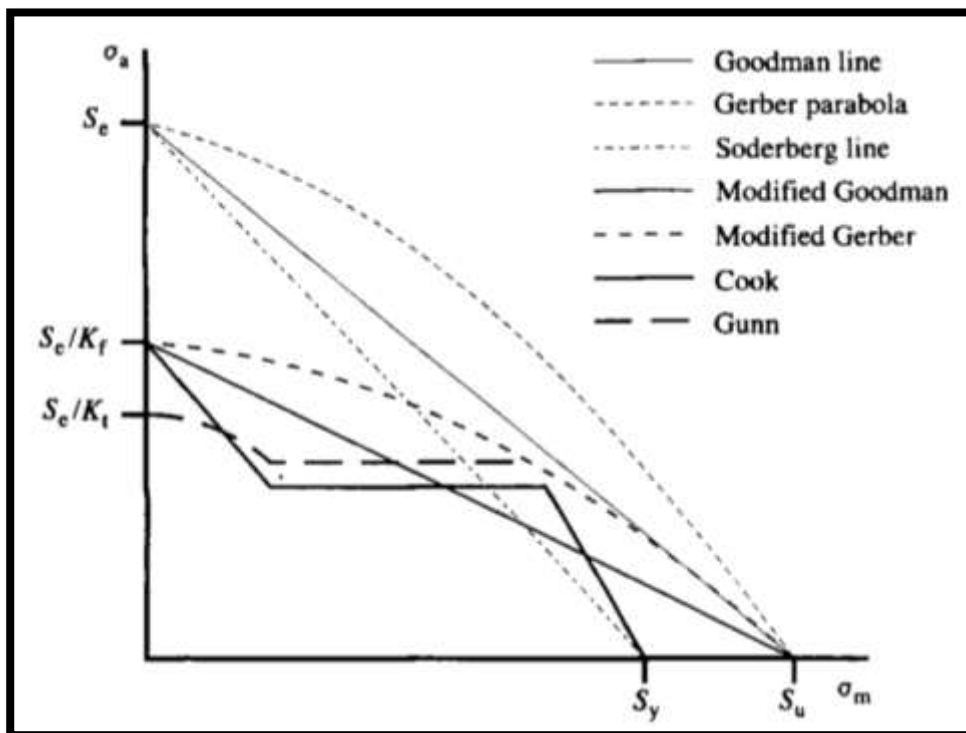


Figure 18

4.5.5 Bolts fatigue limit

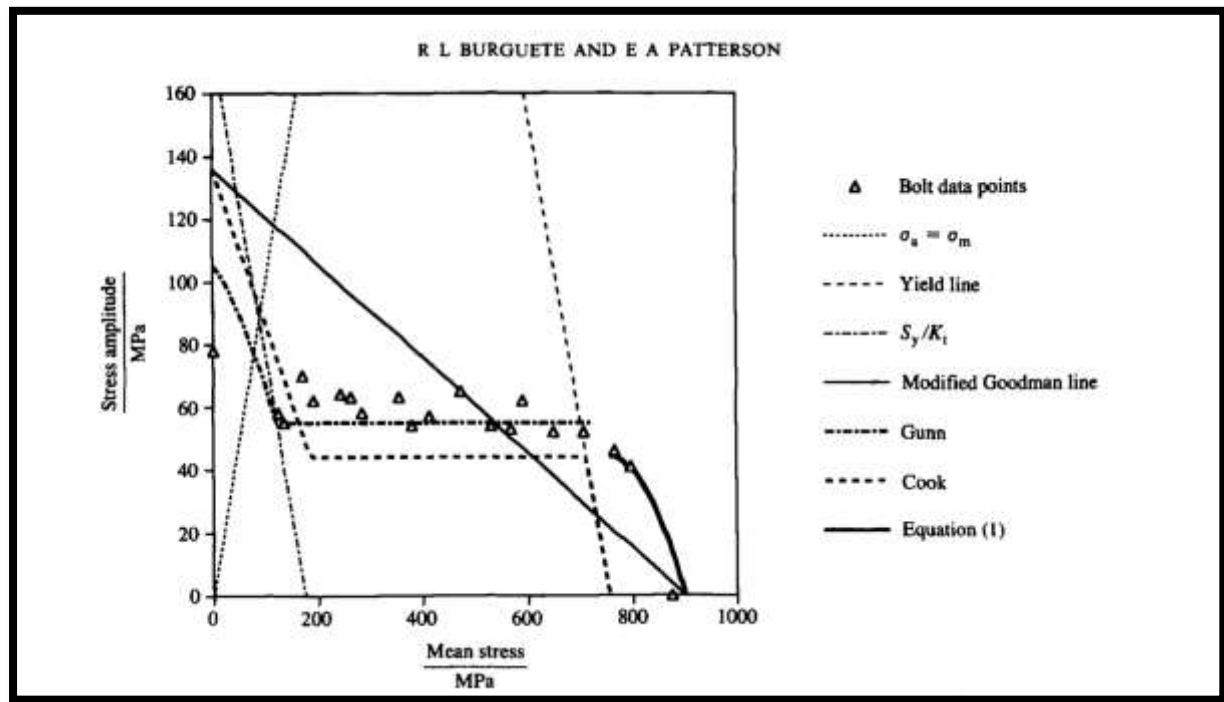
Interestingly, a notched specimen, such as a bolt, acts very differently when considering fatigue life and so the Cook and Gunn method is used to calculate fatigue life of notched specimens. Graph 12 below by (Burguete & Patterson, 1995) shows the different methods on a Haigh diagram where the vertical axis is the stress amplitude and the horizontal axis is mean stress.



Graph 12

Several studies including (Burguete & Patterson, 1995) and (Munn & Li, 2010) show that the fatigue limit remains constant for threaded fasteners once a certain mean stress is reached. For the application of bolting, this means that there is no reason to limit the amount of preload placed on the bolt with the worry of reducing the fatigue life of the fastener. The exact reason this happens is not definitive but most agree that it is because of the localised plastic deformation at the thread root altering the mean stress around that area.

Graph 13 below is directly from the work of (Burguete & Patterson, 1995) and is the results of testing on M12 grade 8.8 bolts.



Graph 13

It can be seen that the Gunn and Cook methods both correlate with the data quite well, with the Cook method being more conservative and therefore it is most often used for rigorous bolt design calculations. It must be noted that both Gunn and Cook methods fail to give reasonable results when the bolt has gone far into yield.

The Cook method was developed by Cook himself and is detailed in 'Mechanics and materials for design (Cook, 1985).

4.5.6 Cook method of estimating fatigue limit

The Cook method is shown below:

$$\sigma_a = \frac{S_e}{K_f} \left(1 - \frac{K_f \sigma_m}{S_u} \right) \quad \text{When } 0 \leq \sigma_m \leq S_{m1}$$

$$\sigma_a = \frac{S_e}{K_f} \left(\frac{1 - \left(\frac{S_y}{S_u} \right)}{1 - \left(\frac{S_e}{S_u} \right)} \right) \quad \text{When } S_{m1} \leq \sigma_m \leq S_{m2}$$

$$\sigma_a = S_y - S_m \quad \text{When } S_{m2} \leq \sigma_m \leq S_u$$

Where:

$$S_{m1} = \frac{S_y}{K_f} - S_a$$

$$S_{m2} = S_y - S_a$$

$$S_a = \frac{S_e}{K_f} \left(\frac{1 - \left(\frac{S_y}{S_u} \right)}{1 - \left(\frac{S_e}{S_u} \right)} \right)$$

$$K_f = 1 + q(1 - K_t)$$

σ_a = Stress amplitude (N/mm²)

σ_m = Mean stress (N/mm²)

S_e = Fatigue limit stress (N/mm²)

S_u = Ultimate tensile stress (N/mm²)

S_y = Yield stress (N/mm²)

S_{m1} = Constant

S_{m2} = Constant

S_a = Constant

K_f = Fatigue reduction factor

K_t = Stress concentration factor

q = Notch sensitivity

By considering what is now known about the shape of the Cook method on the Haigh diagram and the way that notched bars react to fatigue, the Cook equations can be related what happens to the bar.

When $0 \leq \sigma_m \leq S_{m1}$, there is no plastic deformation anywhere on the part and it behaves almost as a normal bar, where an increase in mean stress mean a reduction to the fatigue limit. When $S_{m1} \leq \sigma_m \leq S_{m2}$, there is now localised plastic deformation around the stress concentrations altering the mean stress and the fatigue limit is constant until close to yield. This is the section where the majority of bolts in service will fall in-between. When $S_{m2} \leq \sigma_m \leq S_u$, the full bolt is yielding and the accuracy of the results gets bought into question.

[4.5.7 Mean stress calculations](#)

The calculations can then be carried out on the Ford con-rod bolts to estimate their fatigue life. As will be seen, the highest stress concentration occurs at the thread and so the fatigue life will be solely dependent on the thread concentration. First the mean stress, in both the highest and lowest initial preload scenarios, the fatigue limit and the constants must be calculated to determine which part of the Cook equation must be used.

The mean stress in the bolt in service is then calculated as:

$$Max \sigma_m = \frac{33683 + 34378}{2 \times 39.15} = \mathbf{869.23 \text{ N/mm}^2}$$

$$Min \sigma_m = \frac{31436 + 32131}{2 \times 39.15} = \mathbf{811.84 \text{ N/mm}^2}$$

To calculate the fatigue reduction factors, the stress concentrations of the bolt and its notch sensitivity must be known.

4.5.8 Thread stress concentration factor

The thread stress concentration factor will vary between applications as it is dependent on the tolerance of the thread and the method of production and is also partly a function of the female thread tolerance and geometry. Data for thread concentrations is very hard to come by and can vary largely in the results.

(Blake, 1986, p. 17) suggests that it is between 6 and 3, (Eccles, 2012) suggests between 4 and 6, (Patterson, 1990) used photoelasticity measurements on M12 bolts and determined a stress concentration of 4.2, and perhaps most reliably (Lehnhoff & Bunyard) used FEA analysis on M8 (course thread) bolts of min and max tolerance of male and female threads and had stress concentration factors between 1.18 and 4.8.

The stress concentrations at the thread runout and shoulder will be lower than this.

Considering all of these findings a conservative stress concentration of 5 will be used.

4.5.9 Notch sensitivity

The notch sensitivity of 0.7 was obtained graphically from (Shigley, Mischke, & Budynas, 2004, p. 336)

4.5.10 Fatigue limit of smooth polished bar

There is very little data on the fatigue limit for the high UTS steel alloys used in bolts. There is a general consensus that the fatigue limit is mostly between 40%-60% of the UTS in strong steels and (Burguete & Patterson, 1995) stated that data found from the 'world metal index' city library, Sheffield, a polished boron steel bar (commonly used for bolts) with a UTS of 1050N/mm² has a fatigue limit of 50% the UTS.

Therefore 50% of the UTS will be used for these calculations.

4.5.11 Fatigue safety factor calculation

The fatigue reduction factor and constants were then calculated:

$$K_f = 1 + 0.7 \times (5 - 1) = \mathbf{3.8}$$

$$S_a = \frac{550}{3.8} \times \left(\frac{1 - \left(\frac{990}{1100} \right)}{1 - \left(\frac{550}{1100} \right)} \right) = \mathbf{28.95}$$

$$S_{m1} = \frac{990}{3.8} - 28.95 = \mathbf{231.6}$$

$$S_{m2} = 990 - 28.95 = \mathbf{961.1}$$

It can then be seen that the mean stress of both the initial preload scenarios are in between the constants S_{m1} and S_{m2} showing that the middle of the three equations must be used. Looking back at the Cook method shows that, under these circumstances, σ_a is equal to S_a which was just calculated as:

$$\sigma_a = \frac{550}{3.8} \times \left(\frac{1 - \left(\frac{990}{1100} \right)}{1 - \left(\frac{550}{1100} \right)} \right) = \mathbf{28.95 \text{ N/mm}^2}$$

This is the allowable stress amplitude which means the stress that is allowed above and below the mean stress. So for infinite fatigue life, the total change in stress calculated in the bolt must not exceed twice this value:

$$\text{Allowable } \Delta\sigma = 2 \times 28.95 = \mathbf{57.9 \text{ N/mm}^2}$$

The bolts both had the same change in stress of:

$$\text{Max } \Delta\sigma = \frac{34378 - 33683}{39.15} = \mathbf{17.75 \text{ N/mm}^2}$$

The fatigue safety factor of the bolt can now be calculated:

$$\text{Fatigue Safety Factor} = \frac{57.9}{17.75} = \mathbf{3.26}$$

This is likely to be a conservative estimate of the safety factor and the true value would probably be even higher. The stress concentration used was likely higher than reality and the Cook method gives a value roughly 15% lower for the allowable stress amplitude than what is statistically expected.

[4.5.12 Notes](#)

Conversely, the exact stiffness of the bolt and rod was not known and was instead determined through FEA software and calculations which will carry a certain amount of error. Ideally the stiffness of the bolts would be measured with an unmodified bolt placed into tension, and tensile tests would be carried out on test pieces machined from the con-rod to exactly determine the Young's modulus which would then be used in the FEA analysis.

The loading plane factor likely also carries some error due to the difficulties of determining this accurately but tests to gain more accuracy are far beyond the scope of this paper and the FEA analysis gave quite clear results.

The embedding of the bolts will also change on a bolt to con-rod basis but the deviation is small. Also the fracture split Ford con-rod will likely have almost zero embedding due to the grain structure already being matched but it has been included in the calculations to be conservative.

Finally, there are many factors that will drastically change the results such as damage to the threads, non-parallel surfaces, incorrectly tightened bolts, etc however these are more of a quality control problem that should be accounted for at the bolt manufacturing and transport stage.

4.5.13 Prying

There may also be additional loading to the bolt through the action of prying. Prying is when the force is applied to the joint away from the axis of the bolt which has a levering effect, increasing the force experienced by the bolt and also introducing a bending force into the bolt. Because the cap is bolted at either side and the force applied in the middle, a lot of potential prying force would be absorbed as tension in the cap first. This is also a likely reason that the clamp force is roughly 6 times greater than the applied inertia force because increasing preload has been shown to reduce the effects of prying even if joint separation was not occurring at a lower preload (Bickford, 2008, p. 251)

Prying in this scenario of the con-rod is a complex problem that would be a research paper of its own and so has been ignored for the purposes of these calculations and any effect it does have will be 'absorbed' by the conservative approach taken.

5.0 Conceptual Hydraulic tensioning method

5.1 Research

5.1.1 Hydraulic tensioning

Hydraulic tensioning is a method of tightening the joint with the use of almost no torque, eliminating any torsional stress in the bolt and utilising a stud and nut instead of a bolt. A threaded section of the tensioner is threaded over the end of the stud after the nut has been applied. Hydraulic pressure is then fed into the tensioner which extends the stud and the nut is then tightened against the face. When the hydraulic tension is removed, the tension in the stud is taken up by the nut and maintains the compression on the joint. The process is shown below in figure 19:

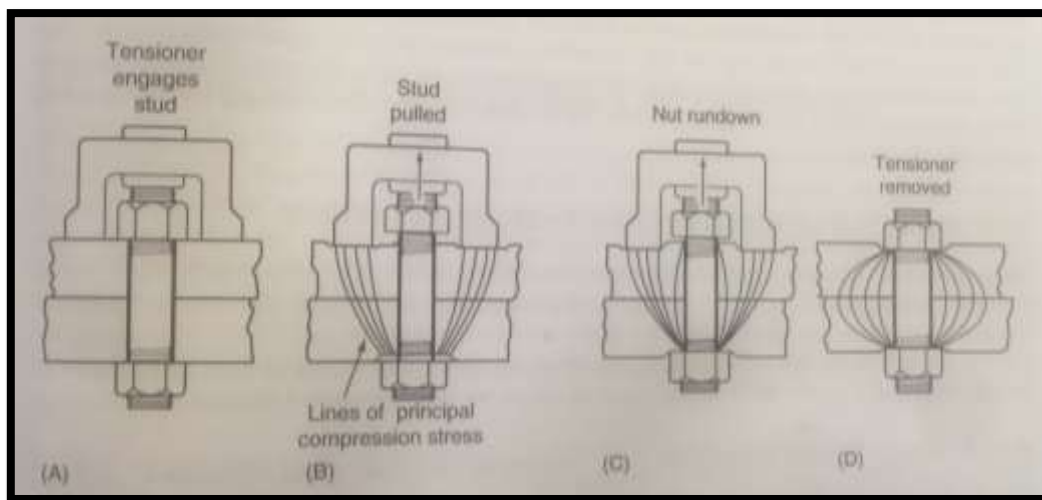


Figure 19

5.1.2 Hydrogen embrittlement

When using bolts above grade 8.8, hydrogen embrittlement begins to be a factor that must be accounted for. Hydrogen embrittlement occurs when hydrogen is introduced into the bolt at the manufacturing stage or occasionally whilst in service and later causes embrittlement followed with cracking, normally within a few hours of tightening. It is currently an area of much research and the exact mechanism of which it forms cracks is not known, however it is clear that bolts above grade 8.8 can be potentially susceptible and fasteners above grade 10.9 become susceptible. (Bickford, 2008, p. 360)

Fortunately, hydrogen is normally exposed to hydrogen when plating is taking place and engine fasteners are not plated due to constantly being covered in oil. There is still risk, however this can mostly be avoided by buying from respectable manufacturers that have taken care and 'baked' the bolts for over 24 hours. The majority of bolts will fracture from hydrogen embrittlement within the first few hours of tightening so inspecting fasteners for failure after 24 hours can almost entirely negate the risk. (Eccles, 2012)

5.1.3 Fatigue resistant thread forms

To minimise the stress concentrations at the thread root for fatigue critical situations, the ISO MJ thread was developed in the 1970's which has a minimum thread root radius of 0.15 of the thread pitch, a standard M thread has a minimum of 0.125 of the pitch. In reality the pitch radius is normally larger than 0.15 of the pitch and special attention must be paid to the tolerances of the female thread. An MJ thread can reduce the stress concentration factor at the thread to below 3. (Blake, 1986, p. 17).

5.1.4 Fatigue resistant manufacturing methods

Almost all manufactured threads are now rolled instead of cut due to the cost savings of rolling. Rolling a thread moulds the material into the shape of the thread between two rollers and is usually done before heat treatment. When the thread is rolled, it introduces compressive stress into the surface of the bolt which is beneficial to fatigue life. However, when the bolt is then heat treated, the residual compressive stress from rolling is dissipated. When the threads are rolled after heat treatment, the beneficial residual compressive stress remains but the cost of manufacture rises due to the wear to the rolling dies. (Eccles, 2012)

5.2 Application to connecting rod joints

5.2.1 Proposal

The hydraulic tensioning method is most commonly used for bolts above M20, however, with the benefits of creating no torsional stress during tightening and good accuracy of bolt preload being attainable, the benefits are going to be explored should the technology be used for tightening con-rod bolts. The complexity, costs and time of this method would rule it out for production means and would most likely only be utilised at the highest level should it be found to be beneficial.

The stud proposed is a grade 12.9 bolt with the final dimensions shown in figure 20 below. A grade 12.9 bolt will be slightly more susceptible to hydrogen embrittlement but the risk is almost zero if bought from a reputable manufacturer and they are checked 24 hours after tightening. The geometry allows for 6.4mm of engaged threads, the amount of an 8mm nut, as after a length of thread 0.8 of the bolt diameter, the successive threads are supporting no load. (Bickford, 2008)

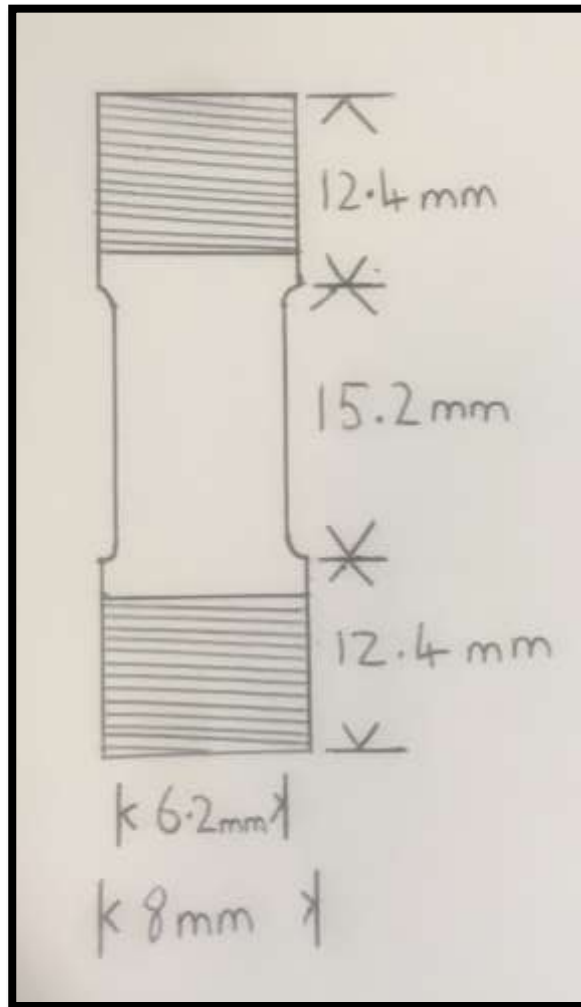


Figure 20

To avoid placing stress concentrations, close together and having a cumulative effect, space for two threads has been left from the bottom of nut to separate the engaged threads and thread runout. A further length of two threads has been left between the thread runout and the radius at the change of diameter. (Eccles, 2012)

The radius was chosen based on the calculations shown shortly.

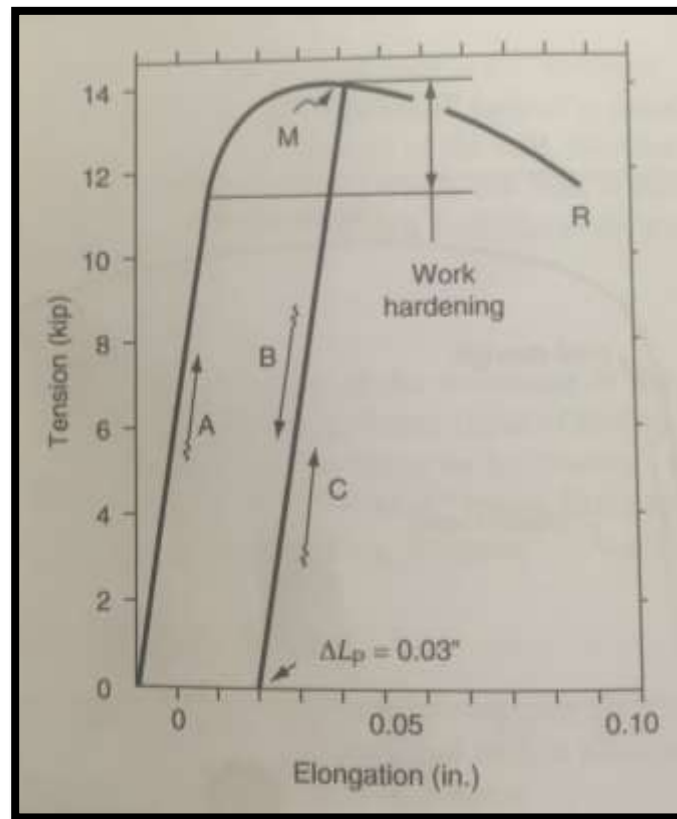
Note that initially the stud would need to have a longer length to allow for the hydraulic tensioner and would be cut to length once the joint is tensioned.

5.2.2 Yield behaviour of the shank

The threads must remain the same size to fit the con-rod thread, however with a higher strength grade of bolt, the shank can be waisted down to save on weight and reduce the stiffness. To get the shank as thin as possible it is preferable to use as much of the tensile capabilities as possible and given that the fatigue problems occur at the threads, the shank can be tightened slightly into yield.

When a bolt reaches yield it doesn't instantly break because of steel's ability to work harden. This is the reason that the load it can support still increases after yielding. If the bolt is loosened after yielding it will not return back to its original length but will then act

elastically again up to the maximum tension it was under before the load was applied. (Bickford, 2008, p. 42) This is shown below in graph 14:



Graph 14

This phenomenon can be used to negate losing some preload force to embedding. With 7 microns of embedding in the elastic region, the joint would lose preload at the rate of the gradient of the elastic region. When tightened into yield the gradient is much less and so the force lost is then much less. It is assumed that the stud will be lengthened in the tensioner so that after embedding the stress will be at the yield of the material of 1080 N/mm². In reality higher values could be reached safely.

5.2.3 Dimension calculation

It is preferable to maintain the same clamp force as the standard bolt due to distortion of the big end at different clamp forces. Therefore, the average clamp force after embedding from the standard bolt of 32560N is the design aim. By rearranging the stress formula to make the radius the subject, the desired radius to achieve this can then be calculated:

$$r = \frac{\sqrt{F}}{\pi\sigma}$$

Where:

r = Radius (mm)

F = Force (N)

σ = Stress (N/mm²)

$$r = \frac{\sqrt{32560}}{\pi \times 1080} = \mathbf{3.1mm}$$

This shows that the diameter of the smooth section of shank needs to be 6.2mm
The stiffness of the bolt can then be calculated. At the base and top of the stud there will each be 8.2mm of engaged length of thread, accounting for the stress gradient reaching 0 at the end of the nut, and 15.2mm of smooth shank.

[5.2.4 Stiffness calculation](#)

The stiffness of the bolt can then be calculated using the equation for stiffness in a varying diameter:

$$\frac{1}{K_{total}} = \frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3}$$

Using the above values and the diameter of the shank and thread, the stiffness was calculated:

$$K_t = \frac{1}{4.45594 \times 10^{-6}} = \mathbf{224420 \text{ N/mm}^2}$$

This is a reduction of 32037N/mm from the previous bolt due to the waisted shank.

[5.2.5 Load factor and bolt force calculation](#)

This will then effect the load factor of the joint and so the maximum bolt force must be recalculated:

$$F_{Bmax} = 32560 + \left(0.5 \times 5466 \times \frac{224420}{224420 + 751823} \right) = 33188N$$

[5.2.6 Fatigue safety factor calculation](#)

The mean stress at the thread for fatigue can then be calculated:

$$\sigma_m = \frac{32560 + 33188}{2 \times 39.15} = \mathbf{839.69 \text{ N/mm}^2}$$

The S_{m2} constant can then be calculated to determine which part of the Cook equations must be used:

$$S_{m2} = 1080 - \left(\frac{600}{3.8} \times \left(\frac{1 - \frac{1080}{1200}}{1 - \frac{600}{1200}} \right) \right) = \mathbf{1048N/mm^2}$$

This shows the stress at the thread is more than S_{m1} and less than S_{m2} meaning the allowable stress amplitude is:

$$\sigma_a = \frac{600}{3.8} \times \left(\frac{1 - \frac{1080}{1200}}{1 - \frac{600}{1200}} \right) = 31.58 \text{ N/mm}^2$$

The allowable change in stress is then twice the amplitude at 63.16 N/mm². It can be seen from the change in force of the bolt that the change in stress at the thread has reduced to 16.04 N/mm². The safety factor of the bolt can then be calculated:

$$\text{Fatigue safety factor} = \frac{63.16}{16.04} = 3.94$$

The safety factor has been increased by 20% from the standard bolt due to the lower stiffness of the bolt reducing the extra bolt force by 67N down to 628N and the higher strength material increased the allowable range of stress by 2.63N/mm² up to 31.58N/mm².

5.3 Analysis

This was without any potential benefits from using an MJ8 thread form or from rolling the threads after heat treatment. These were ignored because it was not determined if the standard bolt had an MJ8 thread form or was rolled after heat treatment so the worst case was assumed.

Because of the hydraulic tensioning, all of the stress capabilities of the fastener could be utilised as tensile stress whilst the bolt is in operation. To estimate the weight saving from the reduced shank and slightly shorter length, a rough volume for the standard bolt was calculated using the head diameter and pitch diameter of 2222mm³ and with one bolt weighing 17.7g, this gives a density of 7.97g/cm³ which is a reasonable density for steel. Calculating the volume of the stud and nut using the pitch diameter and a 12mm diameter for the nut gives a volume of 1913mm³ and a mass of 15.2g using the same density. This is an estimated weight saving of 2.5g per bolt, a weight reduction of 14%. Although this is a small weight saving, a total of 20g would be saved from the rotating mass an in-line 4 engine using 8 bolts, not an insignificant amount in the world of very high performance engines.

The savings of using the hydraulic tensioning method are very small and would most likely not be justifiable in the real world maybe other than where small gains are very hard to come by.

6.0 Conclusion

As expected, the ARP specialised fastener lubricant had the largest effect on the coefficient of friction, reducing it by 23%, showing why molybdenum disulphide greases are often specified for con-rod bolts and the importance of using the specified lubricant. Interestingly,

the engine oil raised the coefficient of friction by 8% which was not expected. In the case of the con-rod bolts this is not a problem as they have normally have a specified lubricant, however other less critical engine fasteners are often assembled either intentionally or unintentionally with engine oil and this could have a negative effect to the clamping force on a joint. This would be an area for more interesting research. Using more samples for each sample and an externally calibrated force washer would increase the accuracy and give a better idea of the scatter.

When using the angle of turn method to tighten the con-rod bolts a preload of $\pm 3.5\%$ was obtained. The preload scatter normally stated for turn of nut method is $\pm 7.5\%$ and this may be more realistic if more samples had been tested. The load cell gave consistent results but was still calibrated using micrometer measurement of the bolts and the bolt stiffness calculation. With more equipment a full force washer could be made that would be accurately calibrated externally. This would have the potential to explore the coefficient of friction with much more accuracy and may give credit to the method of calculating individual thread and head friction coefficients proposed in the first experiment.

The fatigue analysis of the ford con-rod bolt was comprehensive with factors considered and the largest area that was missed out was any effect of prying on the mean stress in the bolt. Despite this, the estimate was conservative and so any error induced from ignoring prying will likely be 'absorbed' by this. The accuracy of the fatigue analysis could be improved by experimentally measuring the stiffness of both the rod and cap, the amount of embedding losses and the stress concentrations, however given the safety factor of over 3, this would not be necessary as the bolt is clearly not being limited by fatigue.

The conceptual method of hydraulically tensioning a higher strength stud shows potential for increased resistance to fatigue and mass savings for the rotational mass but the costs and time involved of developing the equipment and for assembly might outweigh any benefits. Only when extensive effort has gone into developing an engine and the builder is looking for very small gains where money is freely available might this be a worthy investment.