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Analytical Assessment of Engine Internal Components

Contemporary Powertrain Design (ACHA6006)

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03/2025



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Introduction

This report investigates the forces and stresses experienced by key bottom-end components of the Ford 1.6 Sigma engine throughout its operating cycle. The primary objective is to determine the structural properties of these components, assess their fatigue life, and evaluate whether they are over-engineered. Calculations are performed using Excel and the results will be critically analysed in terms of stress distribution, fatigue safety factors, and overall component durability.

The first stage of the analysis involves using the provided pressure data and kinematic equations to calculate the forces acting on critical components, specifically the midpoint of the connecting rod, the gudgeon pin, and the connecting rod bolts. From these forces, corresponding stresses and fatigue safety factors will be determined. Goodman diagrams are used to assess fatigue life, and the results will be analysed to determine whether the components are operating within acceptable limits or if excessive design margins exist.

Following this, RPM range is extended to 9000 RPM through interpolation, and the associated pressure data will be extrapolated accordingly. The new forces and stresses acting on the connecting rod bolts will be calculated, and updated fatigue safety factors (FSF) will be assessed. Based on these findings, the geometry/ material of the connecting rod bolts will be changed to restore or improve the original safety factor at the increased RPM.

Finally, instantaneous torque generated by the standard crankshaft configuration and cross-plane configuration will be assessed. The differences in torque characteristics between these configurations will be examined, along with their respective advantages and suitability for different applications.

Part 1

Connecting Rod

To calculate forces, stresses, and fatigue safety factors (FSF) a series of calculations must be completed throughout the 720-degree cycle. As this analysis was completed in excel, angle values must first be converted into radians due to Sin, Cos, and Tan all needing angles in radians.

Piston Position/ Velocity

Initially, Piston position (S) must be calculated using the following equation:

$$S = (R + L) - (R \cos \theta + L \cos \phi)$$

Where:

$$\phi = \sin^{-1} \frac{R \sin \theta}{L}$$

These equations are formed through a series of right-angle trigonometric equations derived from crank rotation, crank radius, and rod length, shown in figure 1.

piston velocity must first be calculated in m/° done simply by deducting value 1 from value 2 of piston position. This can then be converted into m/s by multiplying by the RPM times six.

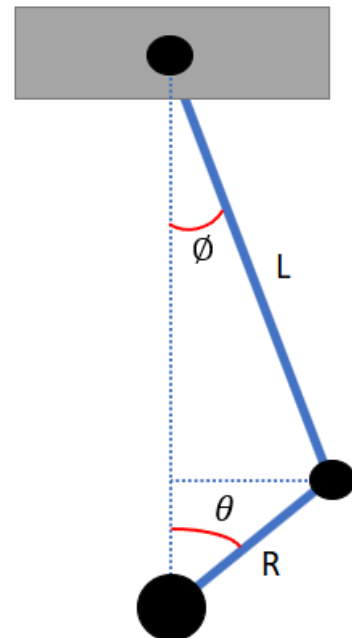


Figure 1- piston position diagram.

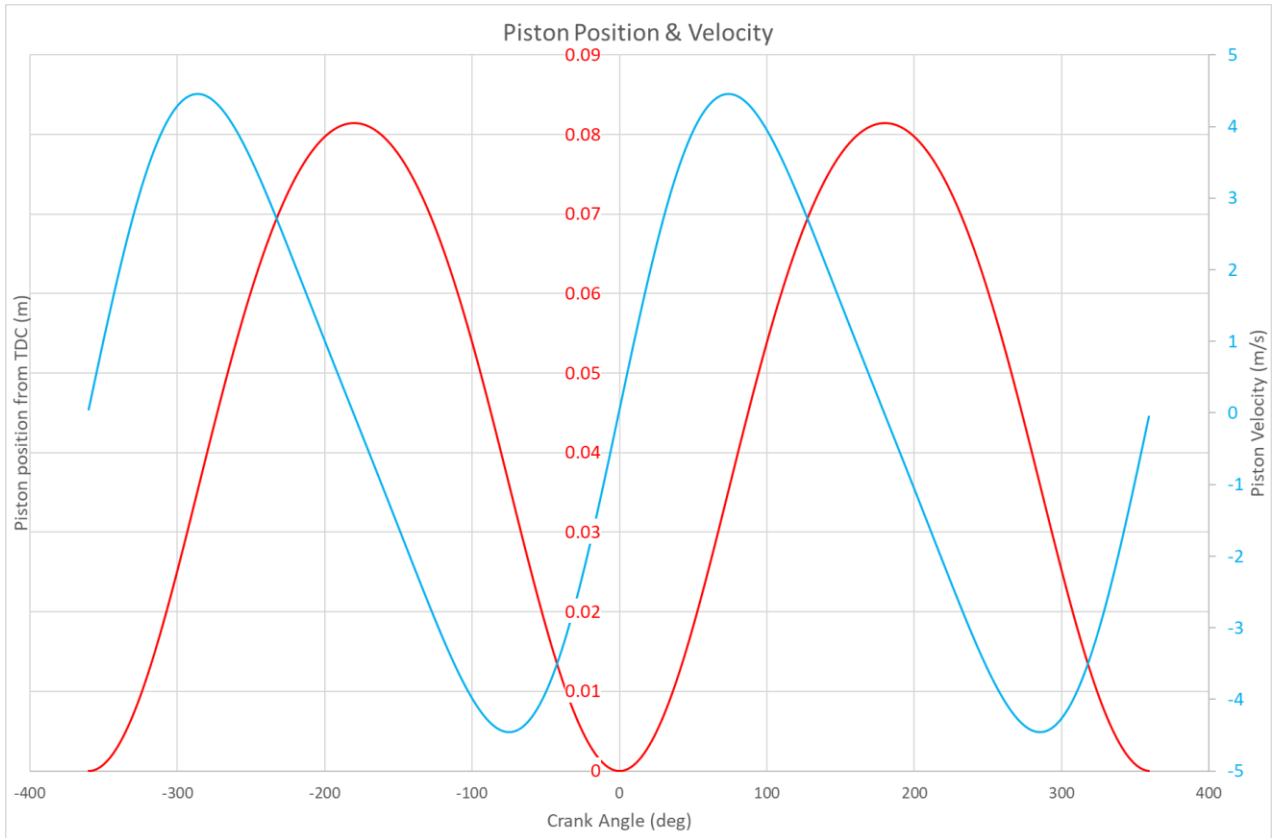


Figure 2- Piston Position and Velocity

Piston Acceleration/ Inertia Force

To calculate piston acceleration a series of equations are combined to produce the following equation:

$$\frac{d^2S}{dt^2} = R\omega^2 \cos\theta + \frac{R^2}{L}\omega^2 \cos 2\theta$$

Formed from the equation for piston position (S), values are first substituted into the S equation where the chain rule is then applied for piston velocity. When differentiated again, the equation above is formed.

From here, Inertia force is calculated by multiplying the acceleration by reciprocating mass, this is the mass of the Piston + Pin + Rings + 2/3 of the Rod. Figure 3 shows the direct relationship between these two values at 1000RPM.

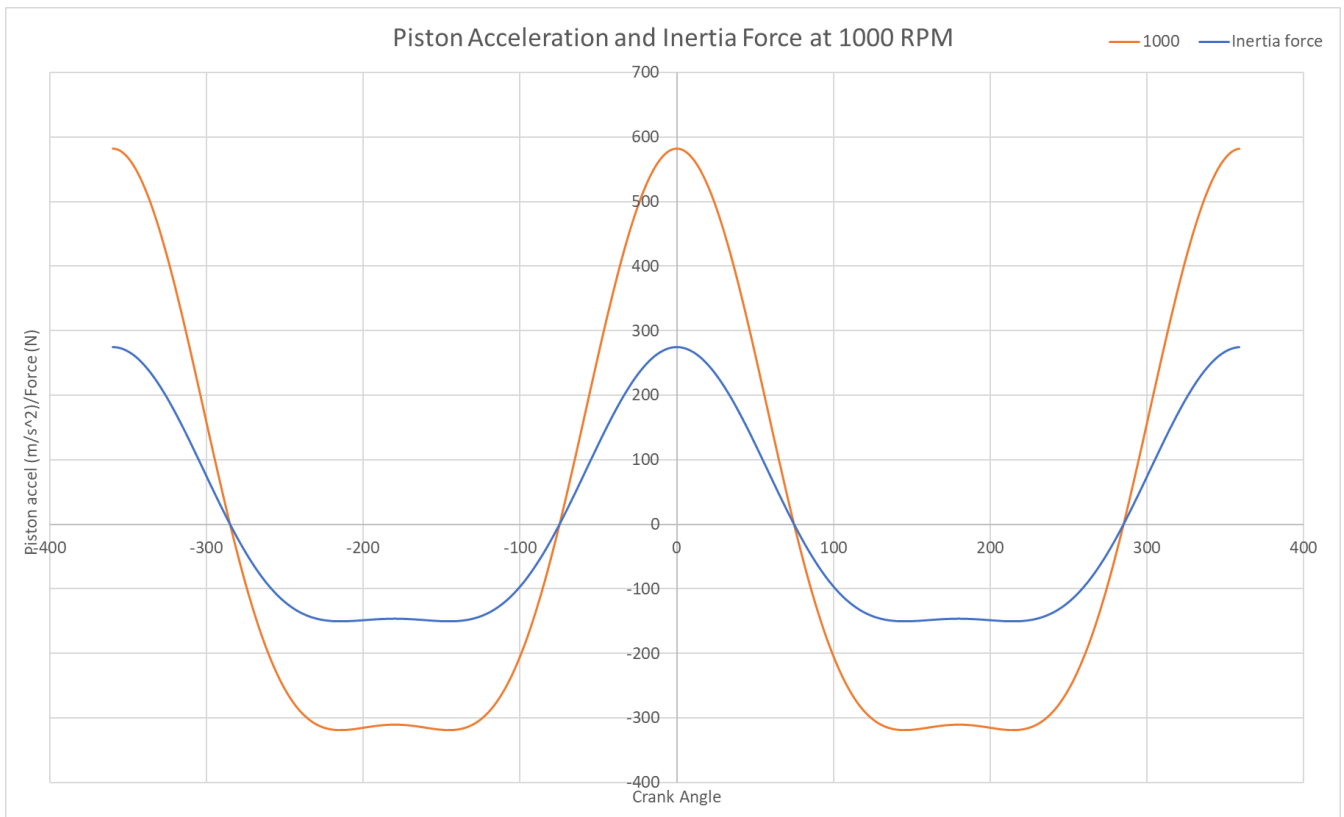


Figure 3- Piston Acceleration and Inertia Force

Piston forces

The next step is to calculate piston Gas force and then Net force. Gas pressure is always active with or against the piston, with gas force being the pressure while accounting for the area of the piston head.

$$F_{gas} = P_{cyl} \times Area$$

Net force is the simply:

$$F_{net} = F_{gas} - F_{inertia}$$

Or

$$F_{net} = F_{inertia} - F_{gas}$$

As compressive forces should be displayed as negative, the second equation is used.

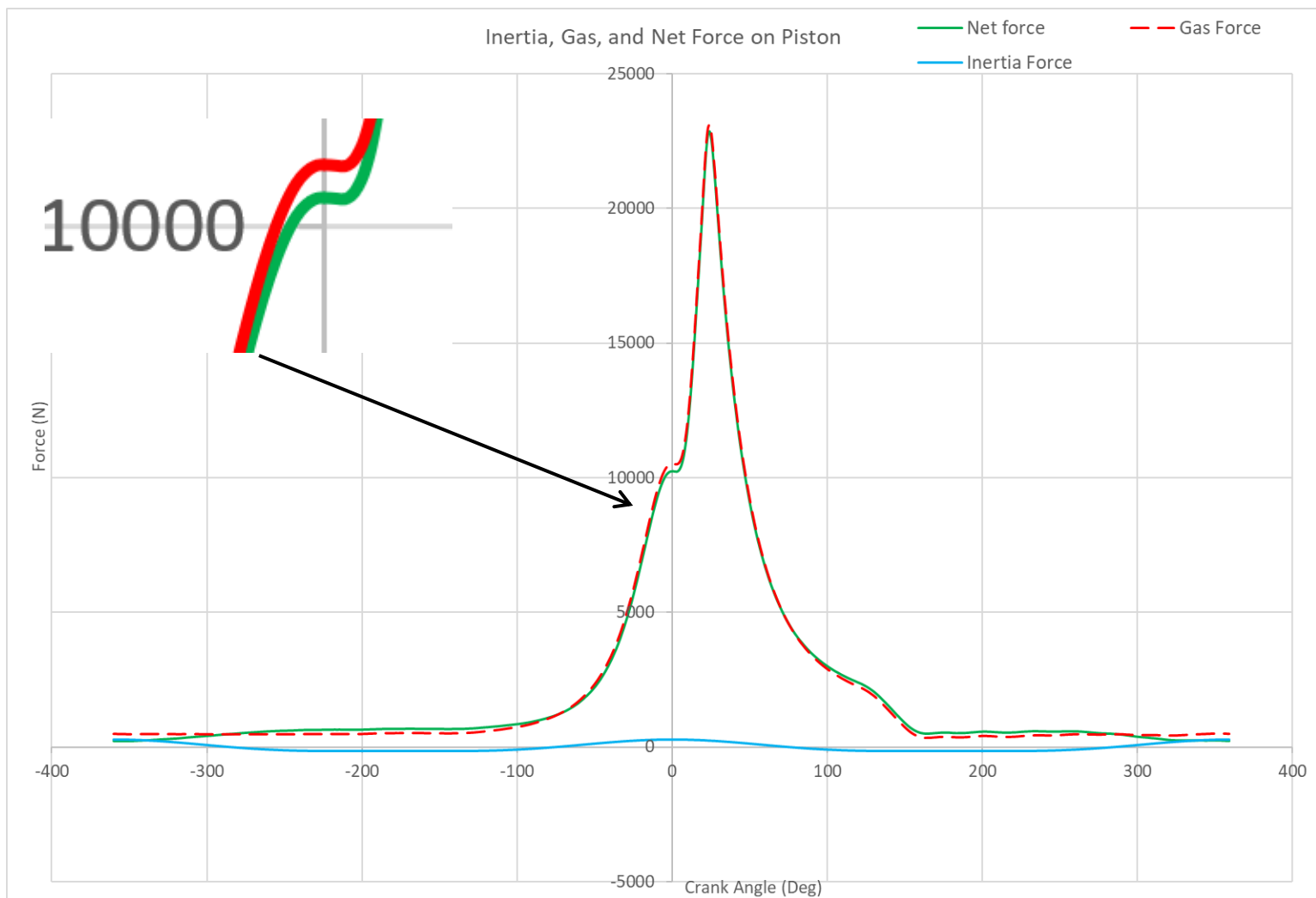
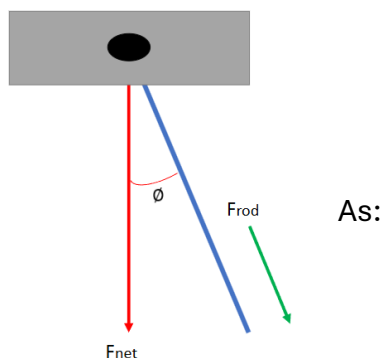


Figure 4- Inertia, Gas, and Net Force on Piston

Figure 4 shows the three forces combined on one graph, highlighting the insignificance of inertia force compared to gas force created through combustion.

This the follows on to the calculation of the rod force itself, being:



As:

$$F_{rod} = \frac{F_{net}}{\cos \phi}$$

$$\cos \phi = \frac{Adj}{Hyp} = \frac{F_{net}}{F_{rod}}$$

Figure 5- Forces Acting from The Piston

Goodman Diagrams

To present goodman diagrams for different RPMs, first the following must be calculated; max/ min stress, mean stress, stress range, allowable stress, and fatigue safety factor. Maximum/ minimum forces on the rod were found which allowed Max/ min stress to be calculated. The remaining values are calculated using the following equations:

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

$$\sigma_{mean} = \frac{\sigma_{min} + \sigma_{max}}{2}$$

$$\sigma_{allowable} = 2\sigma_{End-Lim} \times \left(1 - \frac{\sigma_{mean}}{UTS}\right)$$

$$FSF = \frac{\sigma_{allowable}}{\sigma_{range}}$$

Table 1- Mean Stress, Allowable/ Actual Range, And Fatigue Safety Factor on the Rod

RPM	1000	2000	3000	4000	5000	6000
Mean Stress (MPa)	-83.3	-104.0	-102.9	-92.9	-70.8	-44.9
Allowable Range (MPa)	750.0	768.6	767.6	758.6	738.7	715.4
Actual Range (MPa)	163.5	216.6	235.1	242.3	234.0	224.0
Fatigue Safety Factor	4.59	3.55	3.26	3.13	3.16	3.19

- UTS
- Lower Endurance Limit
- Upper Endurance Limit
- Allowable range
- Actual Range

Figure 6-Legend of Goodman Diagrams

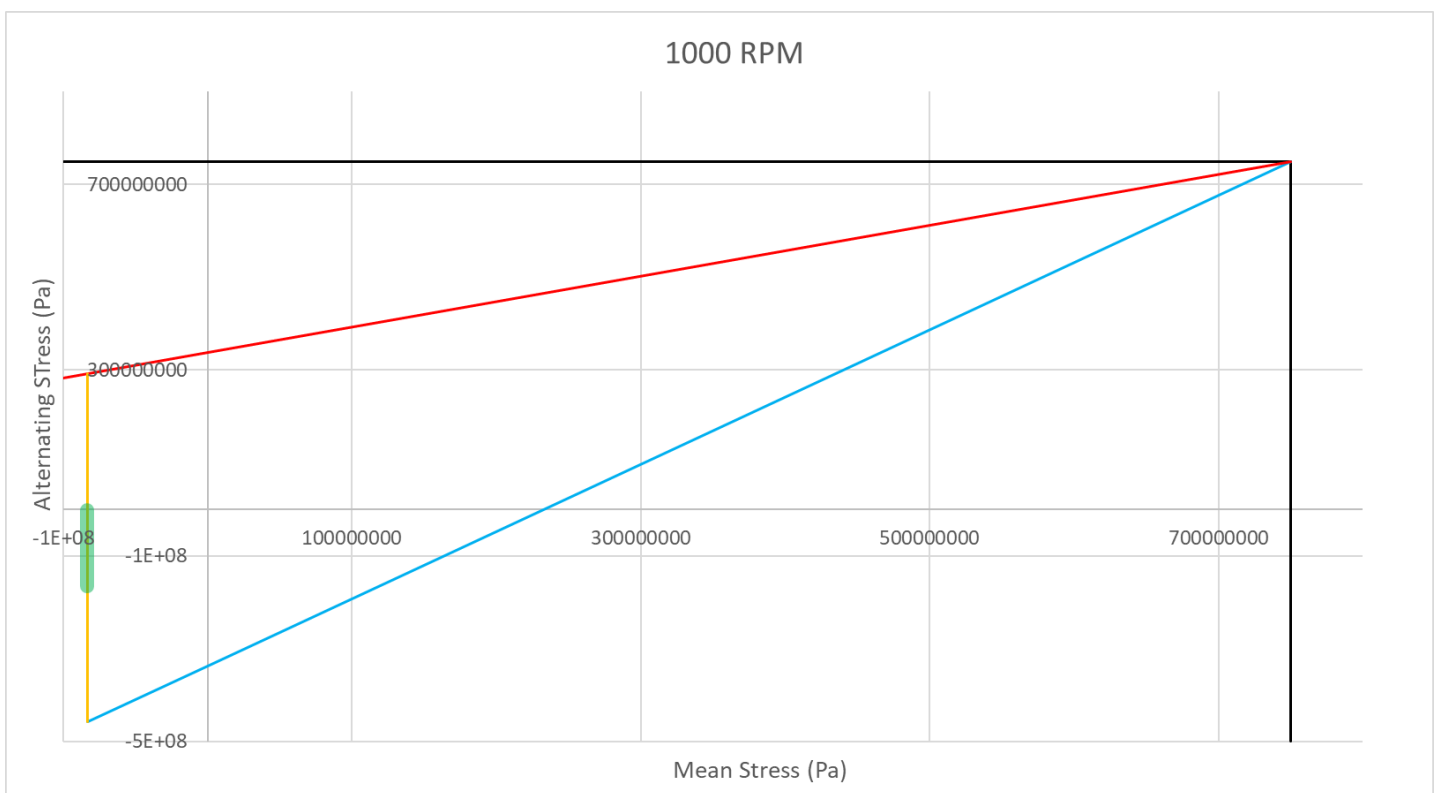


Figure 7- Goodman Diagram for Con Rod at 1000 RPM

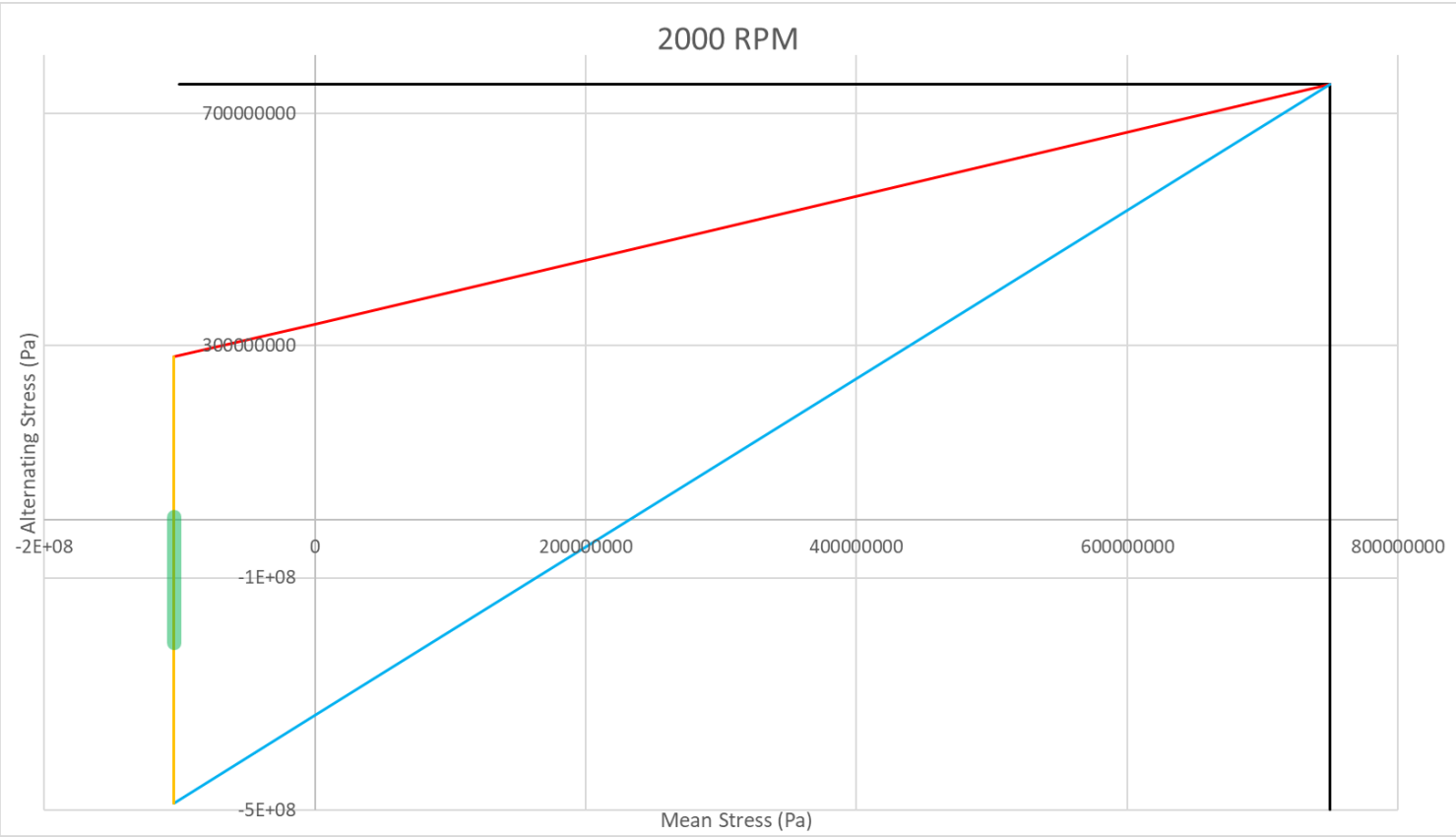


Figure 9-Goodman Diagram for Con Rod at 2000 RPM

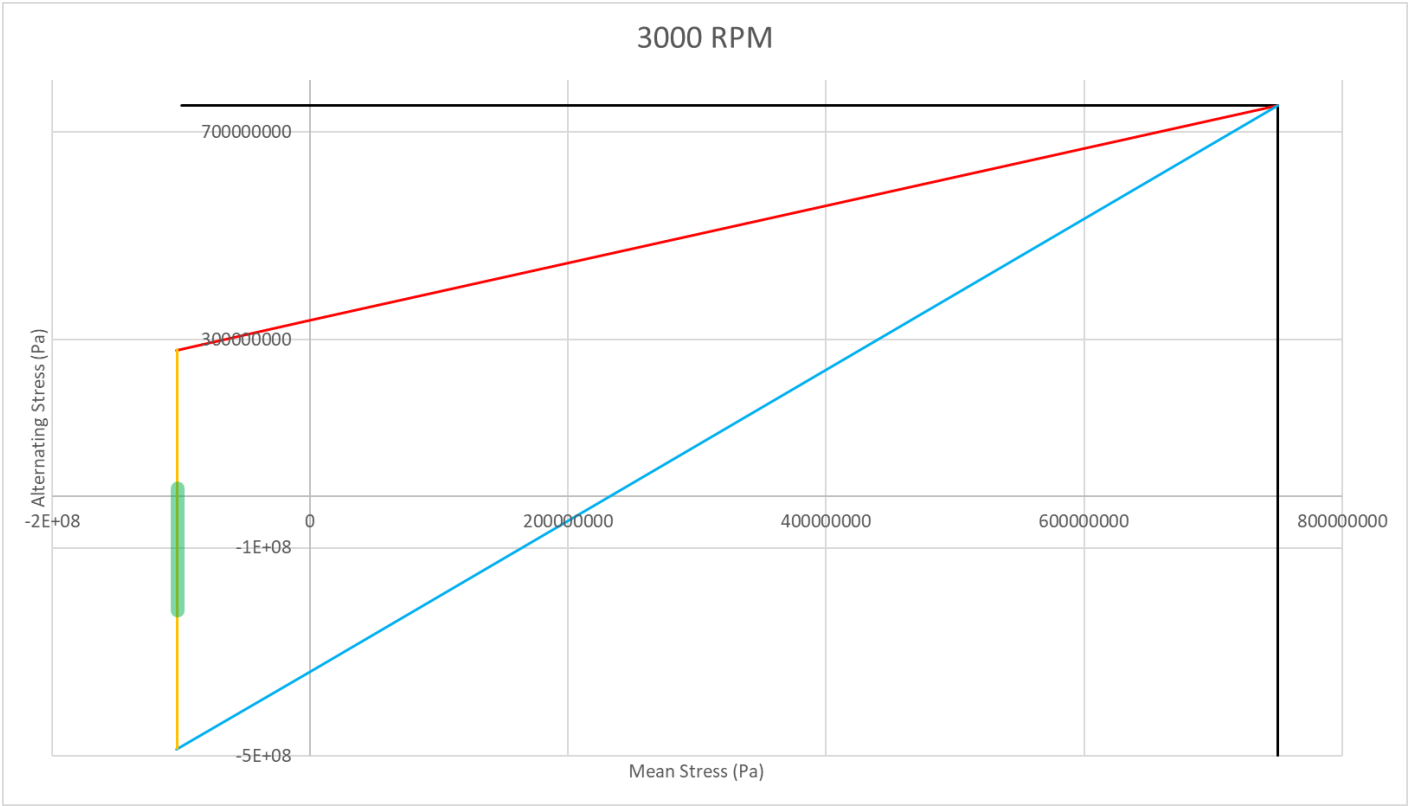


Figure 8-Goodman Diagram for Con Rod at 3000 RPM

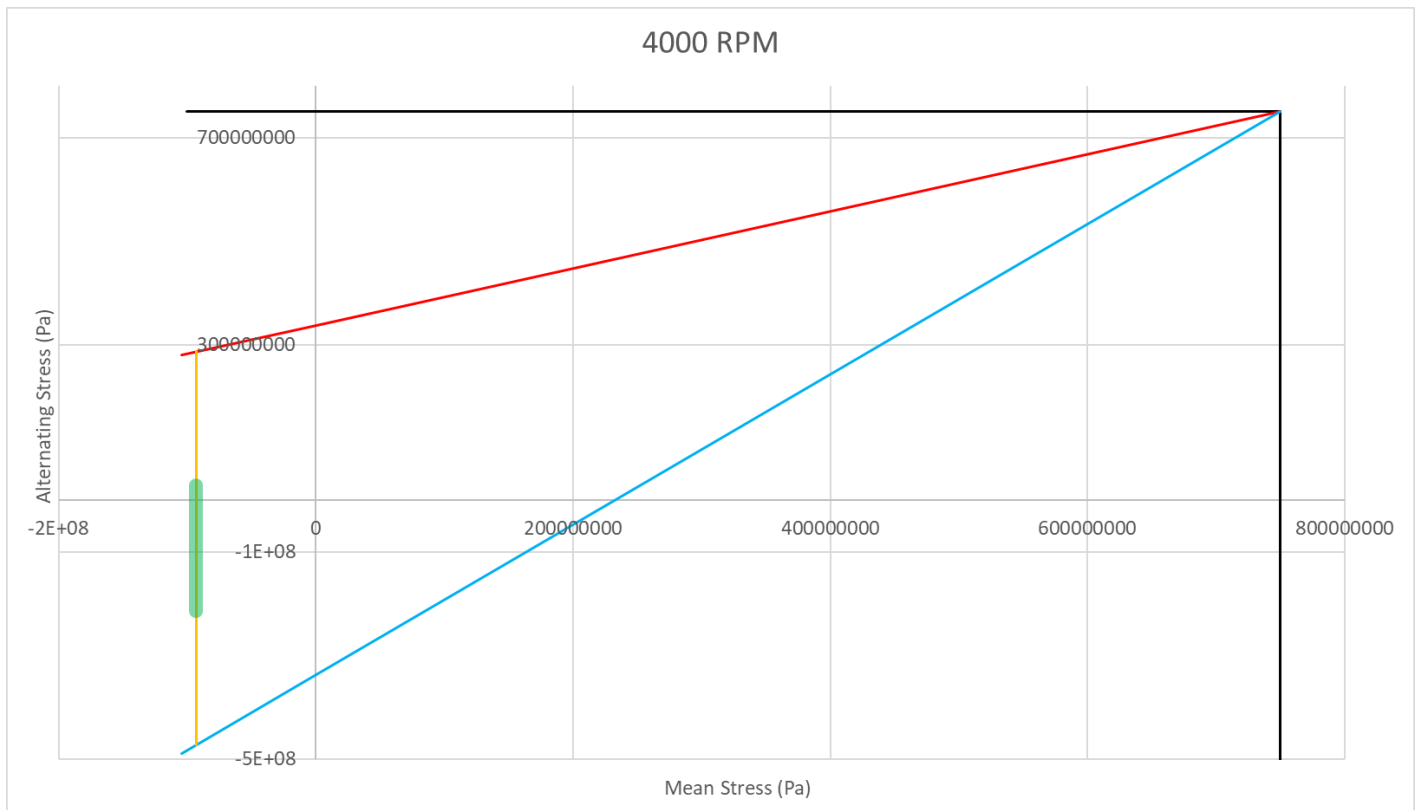


Figure 11-Goodman Diagram for Con Rod at 4000 RPM

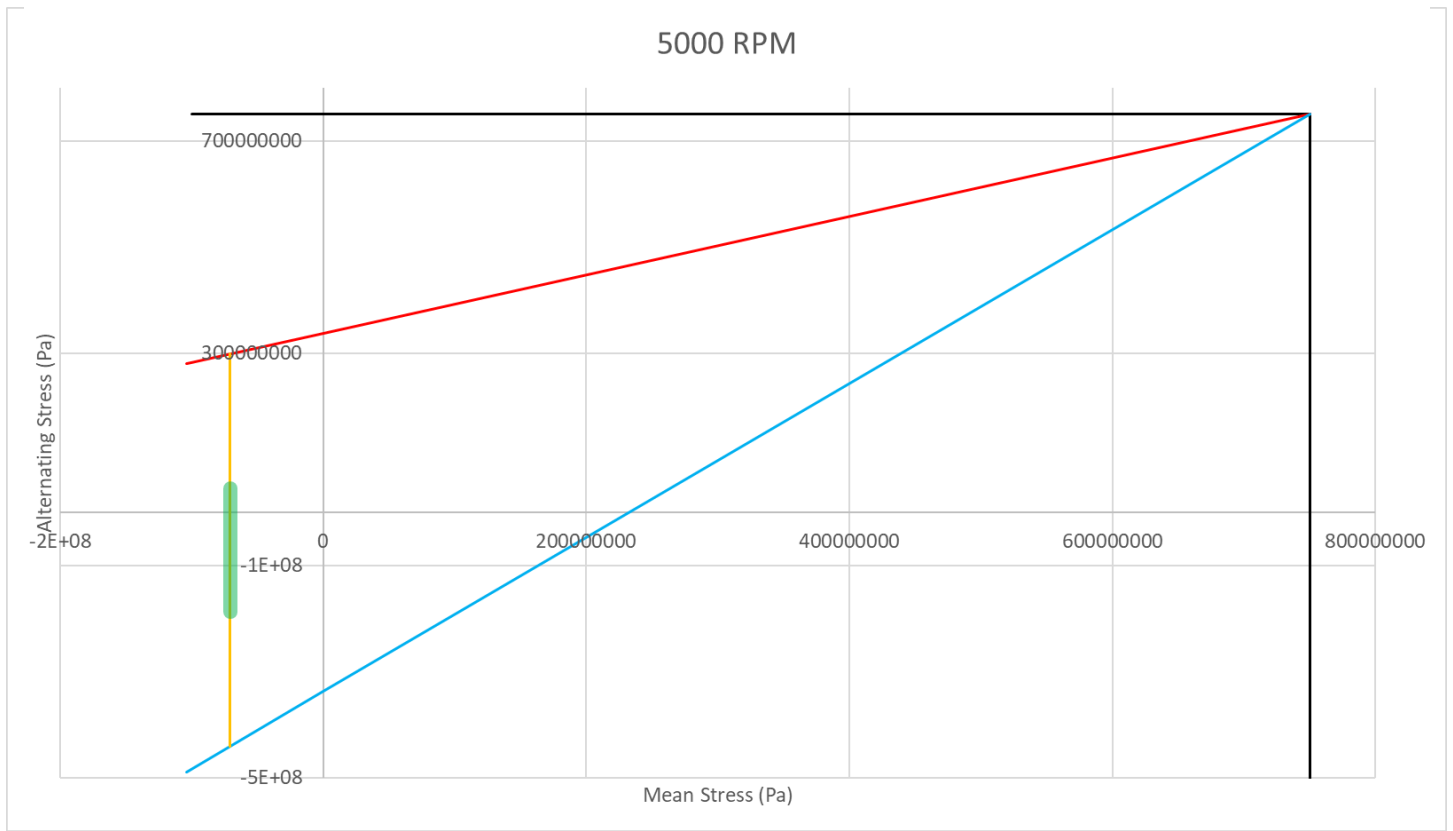


Figure 10-Goodman Diagram for Con Rod at 5000 RPM

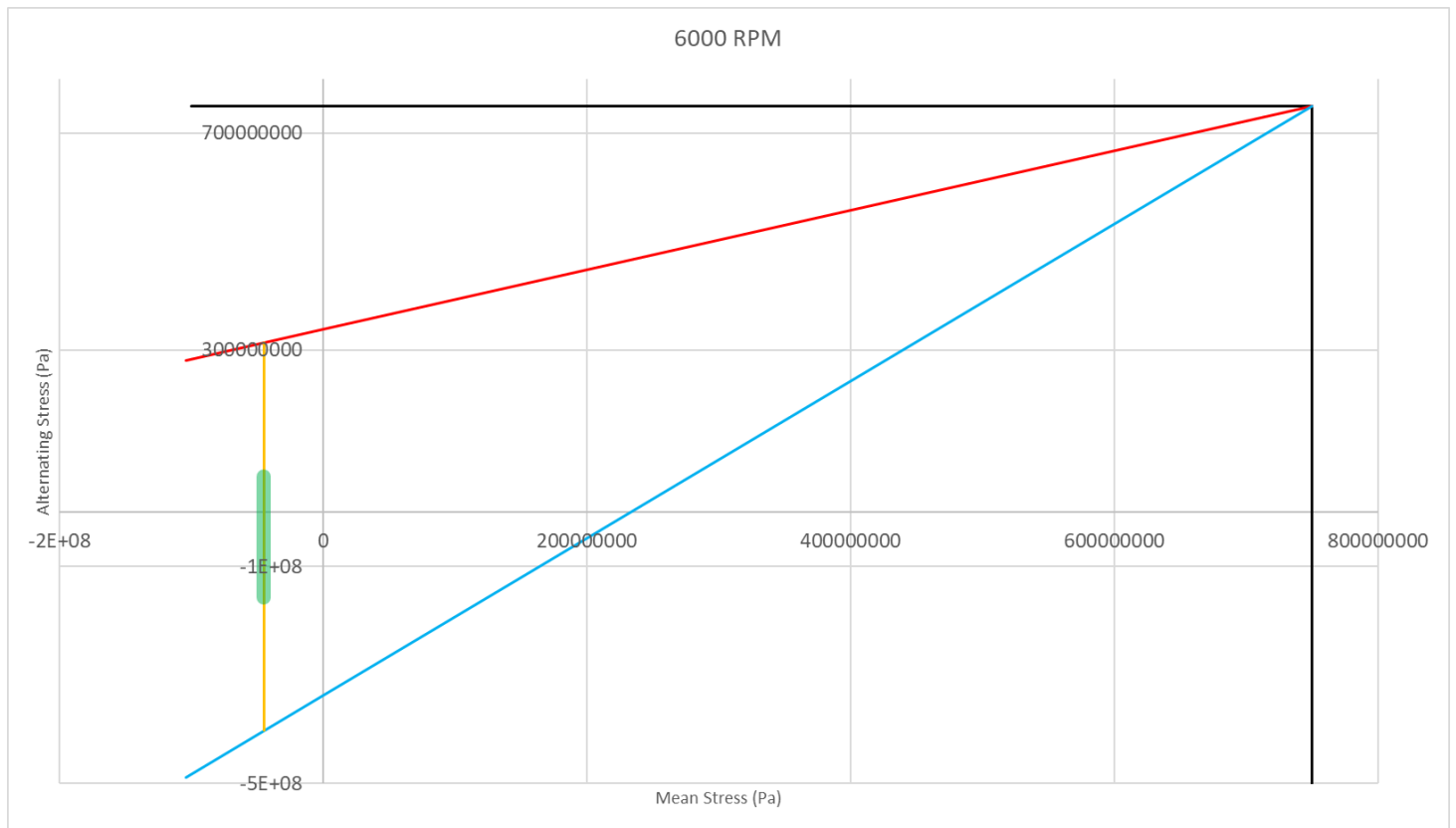


Figure 12- Goodman Diagram for Con Rod at 6000 RPM

The goodman diagrams presented in figures 7 to 12 show minor differences in factors or safety and mean stresses, table 1 shows the minimum factor of safety is at 4000RPM due to a drop off in combustion efficiency after 4000RPM resulting in a decrease in gas pressure. Its also clear to see that even at 6000RPM there is a safety factor of 3.19, but it is important to note stress concentrations have not been considered.

Stress concentrations can be found on the rod in the area shown in figure 13. Roark's Formula to calculate this is shown in figure 14.

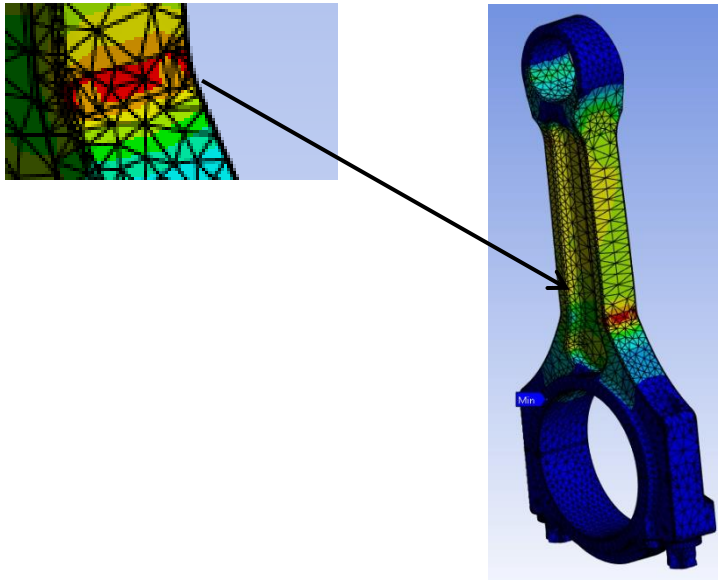
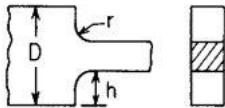
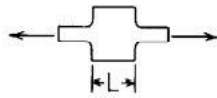


Figure 14-Stress Concentration on Rod

5. Square shoulder with fillet in a member of rectangular section



5a. Elastic stress, axial tension



$$K_t = C_1 + C_2 \left(\frac{2h}{D} \right) + C_3 \left(\frac{2h}{D} \right)^2 + C_4 \left(\frac{2h}{D} \right)^3$$

where $\frac{L}{D} > \frac{3}{[r/(D-2h)]^{1/4}}$ and where

	$0.1 \leq h/r \leq 2.0$	$2.0 \leq h/r \leq 20.0$
C_1	$1.007 + 1.000\sqrt{h/r} - 0.031h/r$	$1.042 + 0.982\sqrt{h/r} - 0.036h/r$
C_2	$-0.114 - 0.585\sqrt{h/r} + 0.314h/r$	$-0.074 - 0.156\sqrt{h/r} - 0.010h/r$
C_3	$0.241 - 0.992\sqrt{h/r} - 0.271h/r$	$-3.418 + 1.220\sqrt{h/r} - 0.005h/r$
C_4	$-0.134 + 0.577\sqrt{h/r} - 0.012h/r$	$3.450 - 2.046\sqrt{h/r} + 0.051h/r$

For cases where $\frac{L}{D} < \frac{3}{[r/(D-2h)]^{1/4}}$ see Refs. 1, 21, and 22.

Figure 13- Roark's Formula for Stress Concentration on The Rod

With these stress concentrations applied, new FSF are as follows.

Table 2- Con Rod FSF with Stress Concentration

	1000	2000	3000	4000	5000	6000
FSF	2.71	2.1	1.93	1.85	1.86	1.89

A static structural fatigue life study was created at 6000RPM to compare FSF. Although values are slightly different, 1.14 on FEA and 1.89 from analytical methods, the stress concentration is present in the same area of that calculated using Roark's. This could be developed further into a full system to improve accuracy, but decision made not to due to time and space constraints.

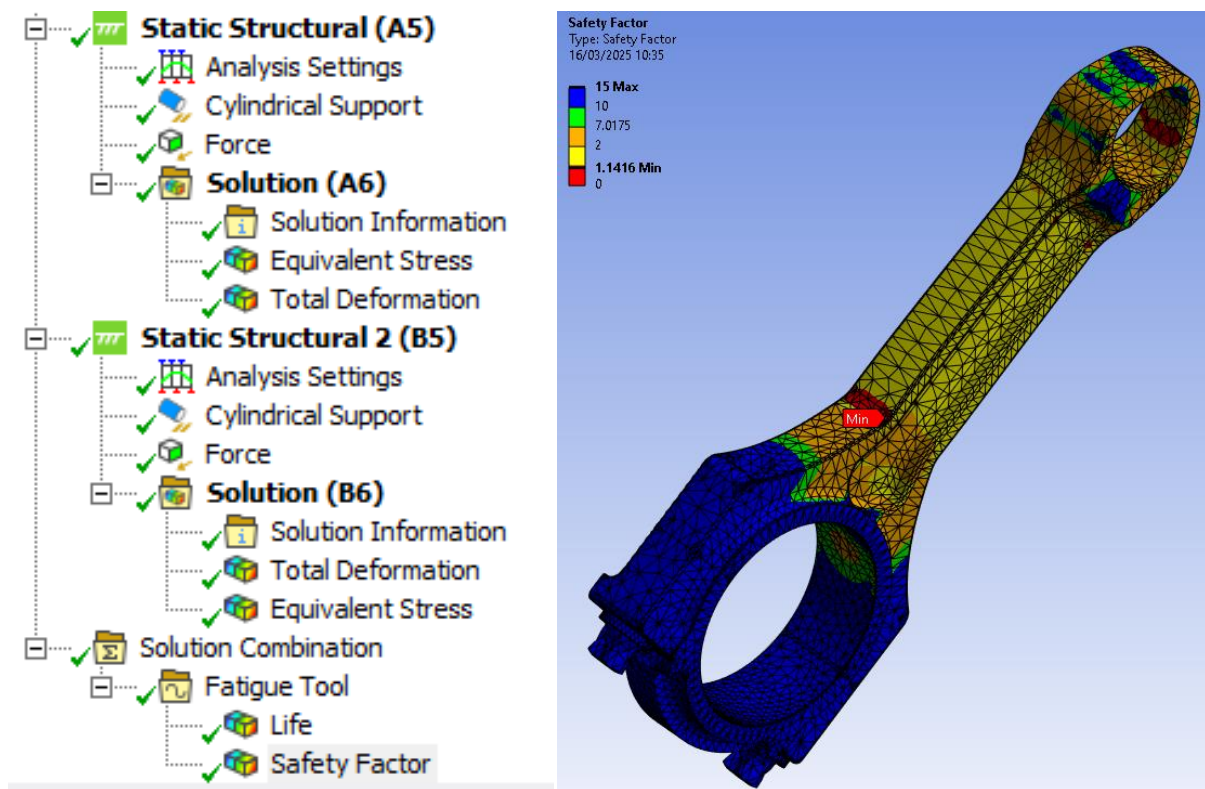


Figure 15- FSF of Con Rod Ansys Static Structural

Gudgeon Pin

The Gudgeon pin will experience stress in two directions:

1. Bending Stress- Due to moment created from force applied from the piston.
2. Shear stress- occurs at support point due to deflection from bending stress.

Therefore, these stresses must be combined to find resultant stress.

Similarly to the rod, the first calculation is to find the Pin Inertia force, calculated by multiplying piston acceleration by the exposed mass on the pin. This can then be multiplied by L1, the distance from centre of gravity on one side to the rods first point of support to give the moment produced from Inertia force.

Moment created by the net force is then calculated by multiplying Net force by L2, the distance between where gas force acts (middle of piston support) to the first point of rod support. These moments can then be combined to calculate total moment acting on the Pin, then the following equation can be used to calculate bending stress:

$$\sigma_b = \frac{My}{I}$$

Where:

y = Distance to neutral axis (m)

I = Second moment of area (m^4)

Shear stress is simply calculated by multiplying total force acting on the pin divided by the Cross-sectional area. This then allows resultant stress to be calculated using the following equation:

$$\sigma_{resultant} = \sqrt{(\sigma_{\max bending})^2 + (2 \cdot \tau)^2}$$

Figures 16 and 17 show bending, shear, and resultant stress at 6000rpm with figure 17 adjusting for compression being negative.

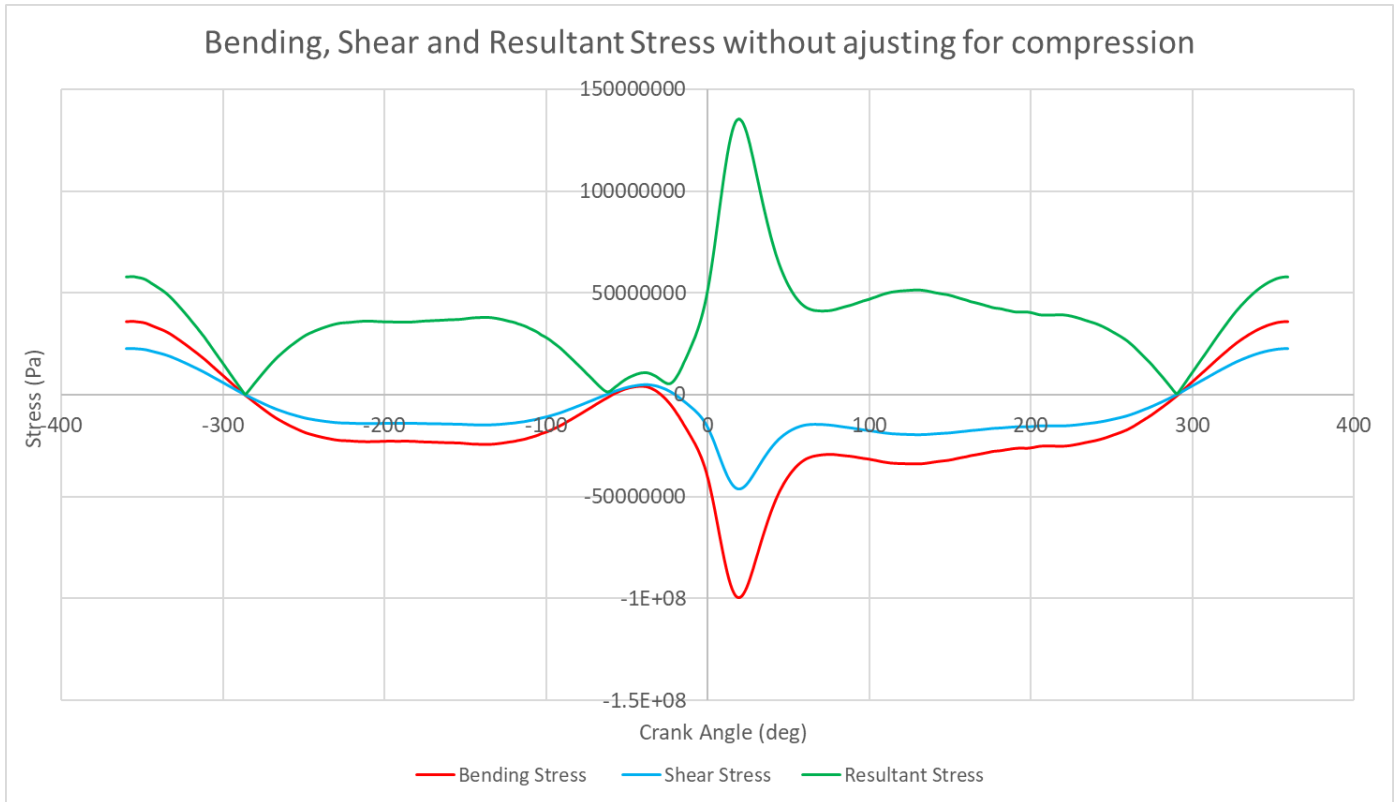


Figure 16- Bending, Shear, and Resultant Stresses on Gudgeon Pin

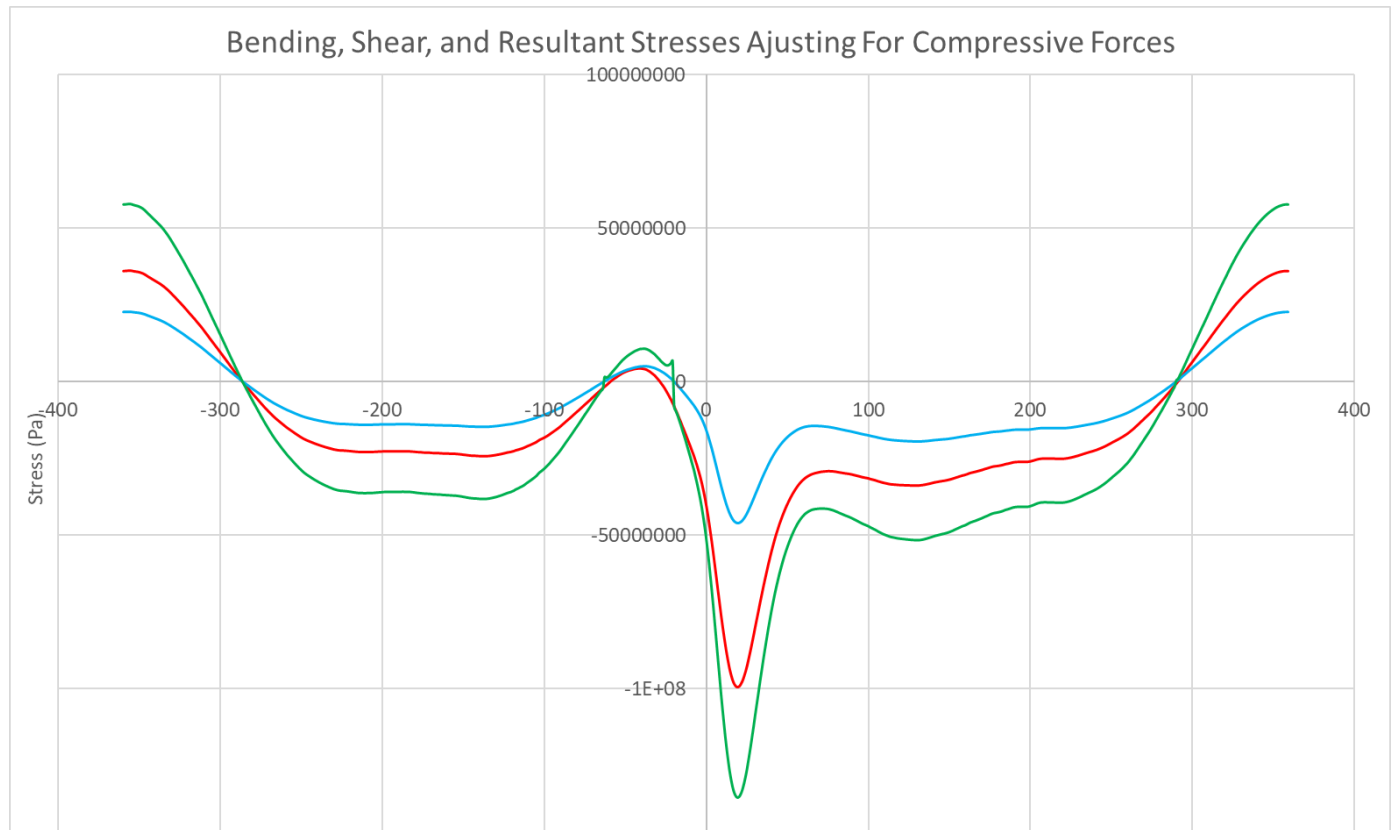


Figure 17-Adjusted Bending, Shear, and Resultant Stresses on Gudgeon Pin

Goodman Diagrams

Table 3- Mean Stress, Allowable/ Actual Range, And Fatigue Safety Factor on Gudgeon Pin

RPM	1000	2000	3000	4000	5000	6000
Mean Stress (MPa)	-50.5	-63.5	-63.6	-58.4	-46.0	-31.6
Allowable Range (Mpa)	945.4	957.1	957.2	952.5	941.4	928.4
Actual Range (Mpa)	98.8	131.3	142.6	147.0	141.8	135.6
Safety Factor	9.57	7.29	6.71	6.48	6.64	6.85

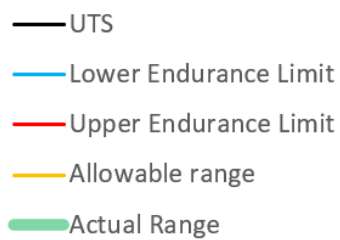


Figure 18- Legend for Goodman Diagrams

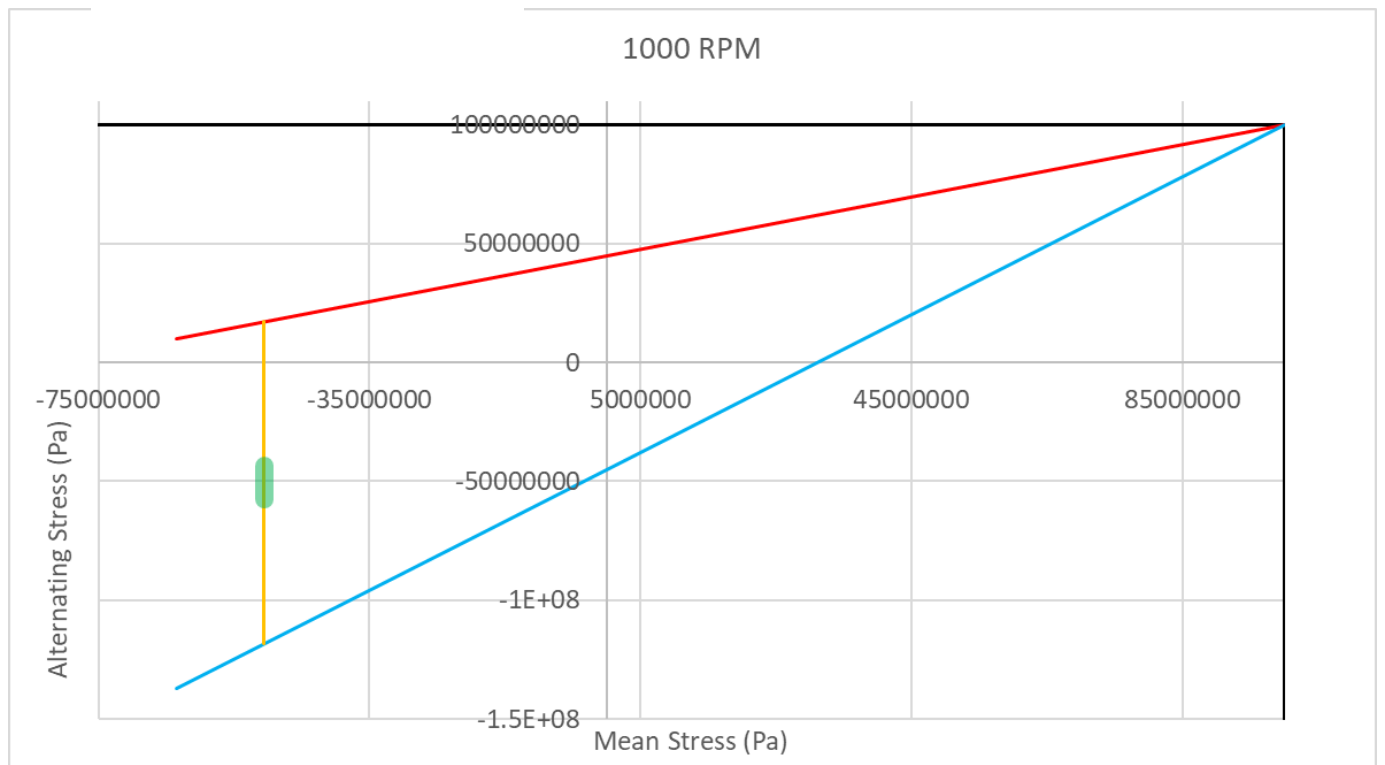


Figure 19-Goodman Diagram for Gudgeon Pin at 1000RPM

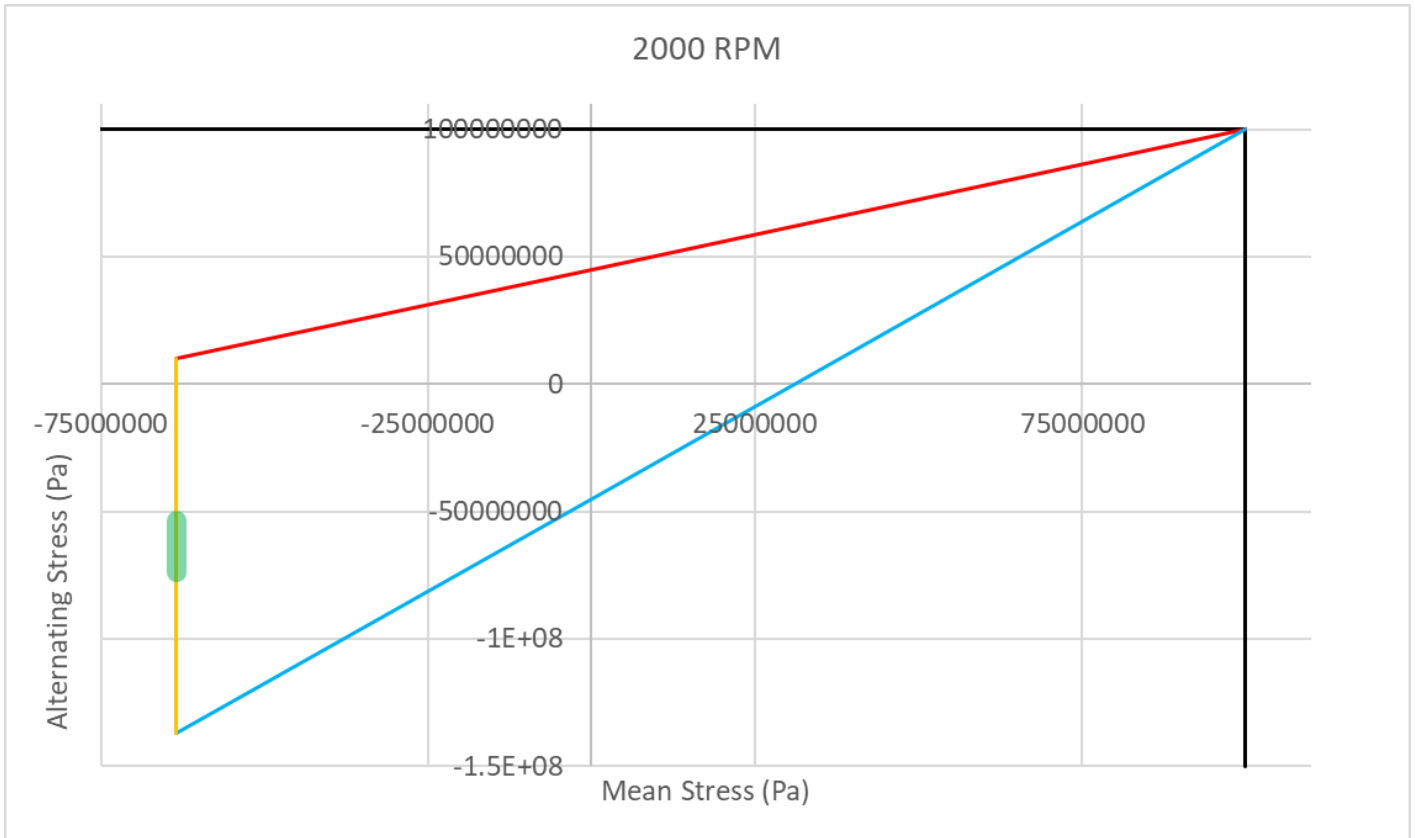


Figure 20-Goodman Diagram for Gudgeon Pin at 2000RPM

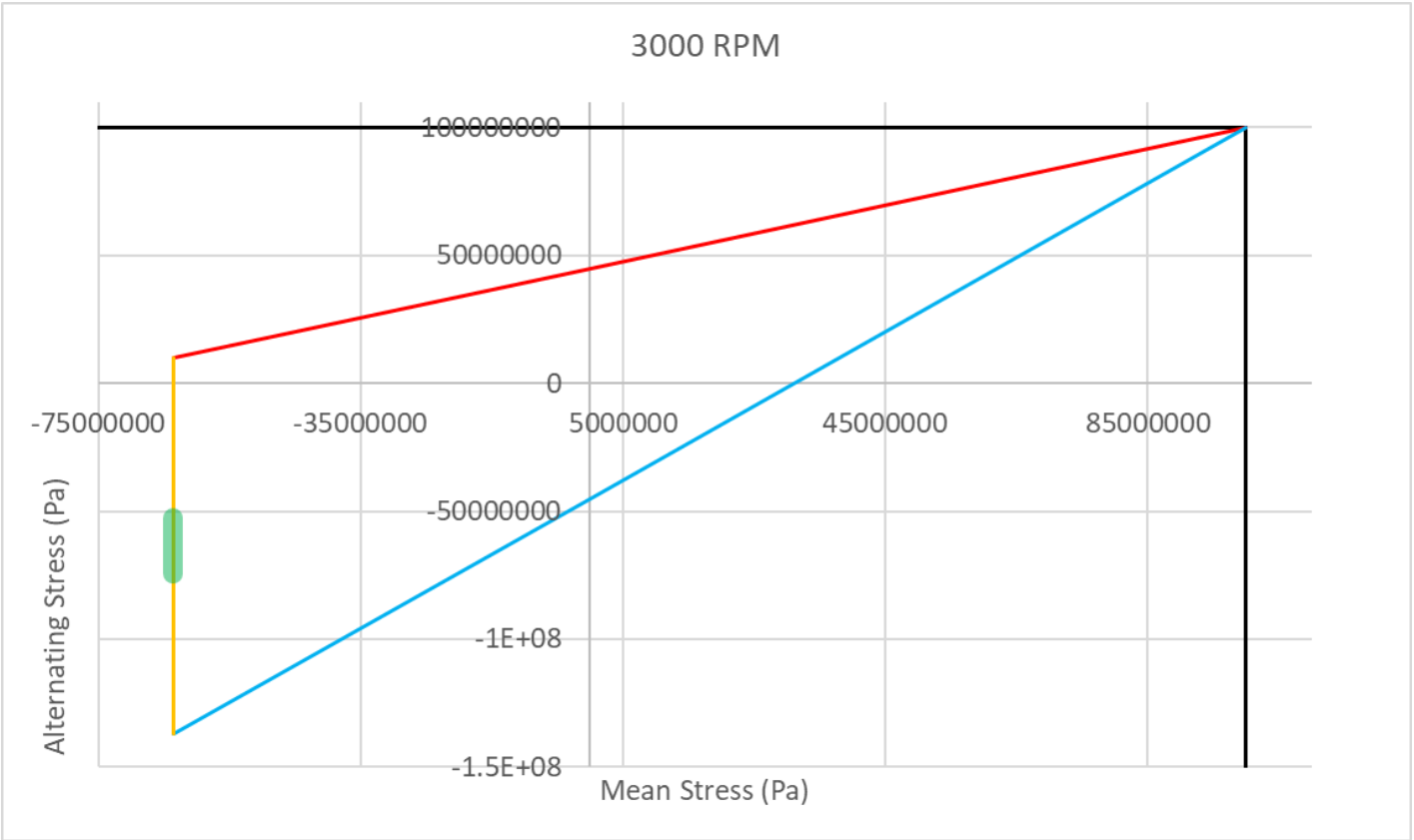


Figure 21-Goodman Diagram for Gudgeon Pin at 3000RPM

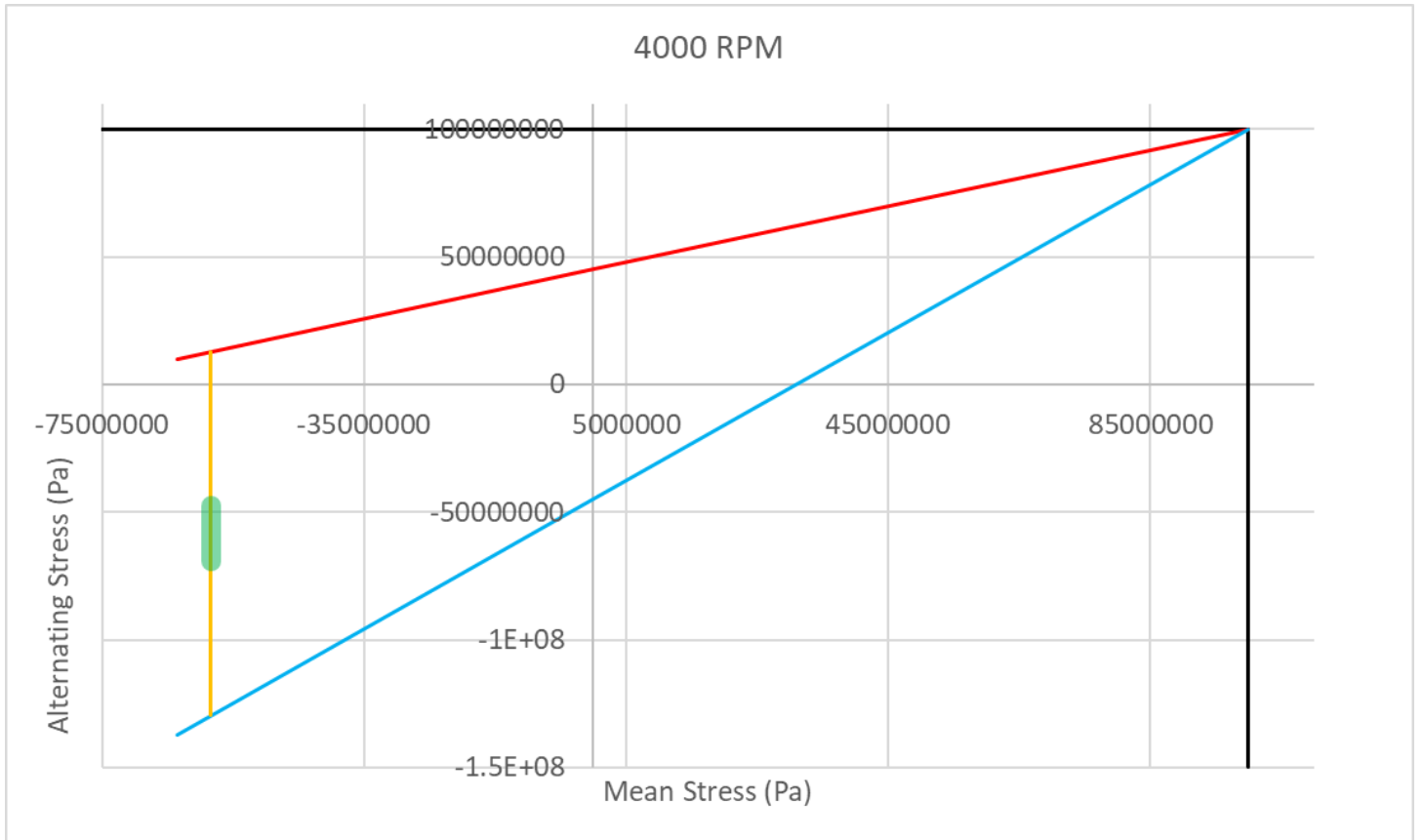


Figure 22-Goodman Diagram for Gudgeon Pin at 4000RPM

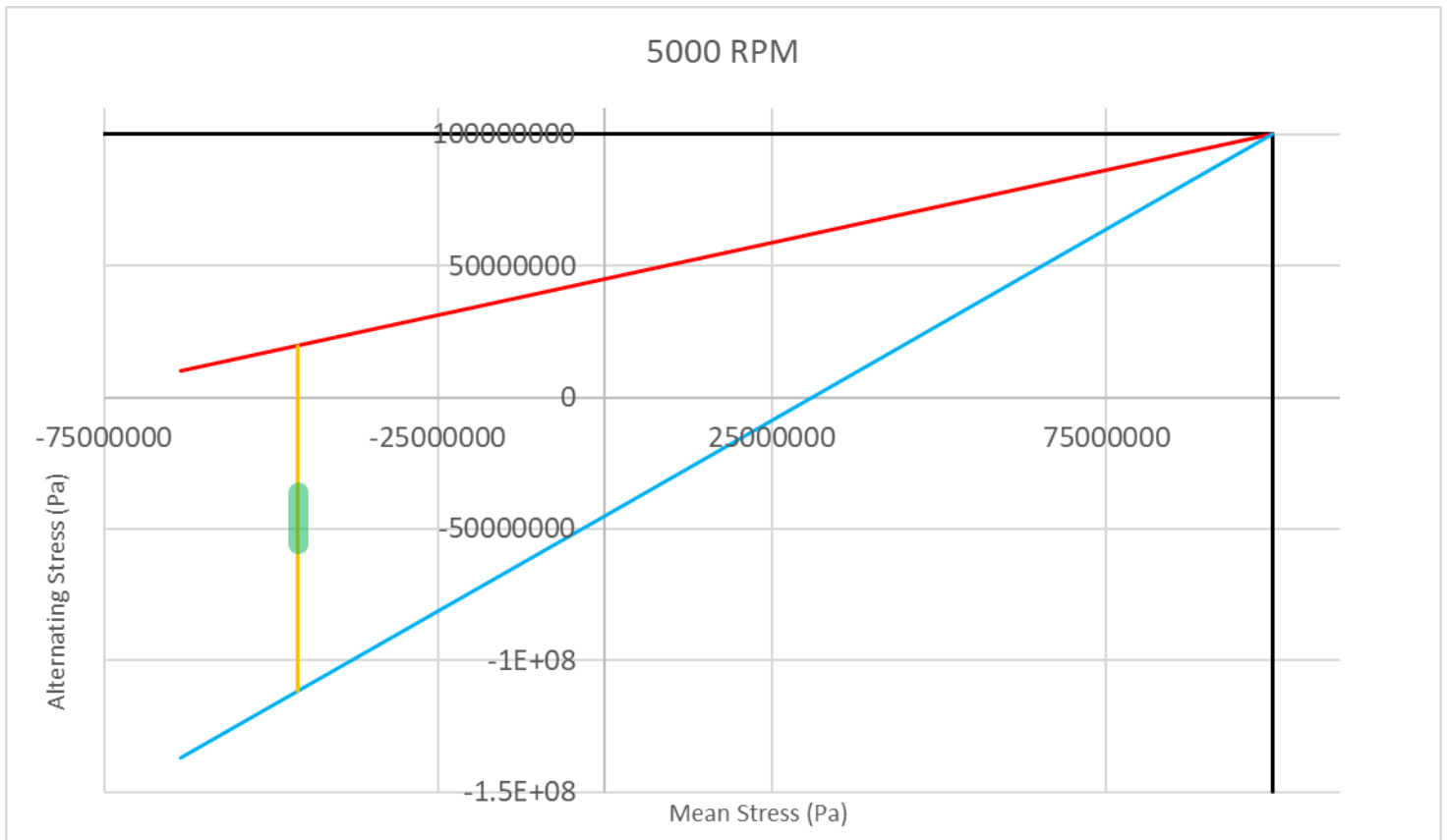


Figure 23-Goodman Diagram for Gudgeon Pin at 5000RPM

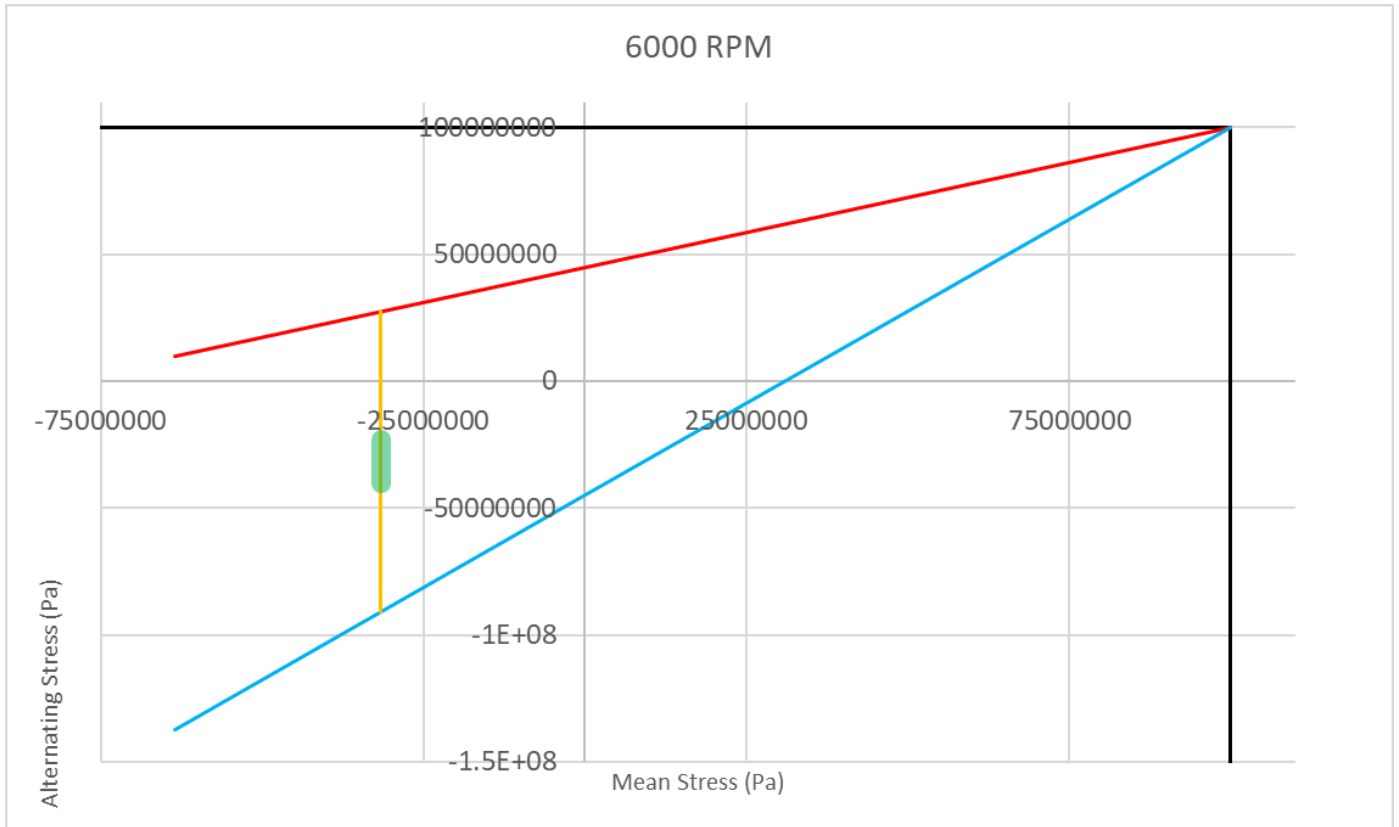


Figure 24-Goodman Diagram for Gudgeon Pin at 6000RPM

On the gudgeon pin, safety factors are even more conservative, again with the minimum at 4000 rpm with a factor of 6.48. This could suggest possible over engineering but again, it is assumed there are no stress concentrations within the component.

Con Rod Bolt

To calculate stresses on the Bolt, first a series of calculations for individual values must be found to complete stress calculations. Key calculations are for axial force, which is found from axial stress, 60% of yield stress which is 90% of UTS. Giving the equation:

$$\sigma_{axial} = 0.6 \times 0.9 \times Yield$$

Bolt/ cap stiffness is also important to calculate the extra bolt force which is calculated with the following equation:

$$F_{extra\ bolt} = \frac{F_{inertia\ bolt} K_{bolt}}{(K_{bolt} + K_{cap})}$$

Other key equations to solve are the cap inertia force, accounting for the forces due to the rotational components that effect the bolt. This is solved using the following equation:

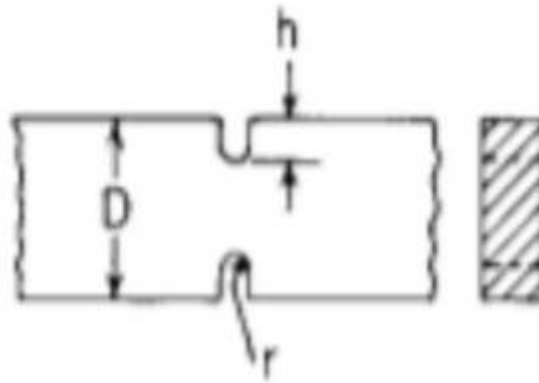
$$F_{inertiaCAP} = m_{rec} \left(R\omega^2 \cos\theta + \frac{R^2}{L} \omega^2 \cos 2\theta \right) + m_{rot} (R\omega^2 \cos 2\theta)$$

With these values, new bolt force can be calculated, and the following is found.

Table 4- Mean stress, Allowable/ Actual Range, FSF

RPM	1000	2000	3000	4000	5000	6000
Mean Stress (Mpa)	724.2	727.8	733.7	742.0	752.6	765.7
Allowable Range (Mpa)	338.2	335.0	329.7	322.2	312.6	300.9
Actual Range (Mpa)	2.37	9.47	21.3	37.87	59.17	85.2
Safety Factor	142.9	35.4	15.5	8.51	5.28	3.53

Bolts have clear stress concentrations which can be calculated using Roark' formulas for stress, in this example the formula in figure 25 is used.



where

	$0.1 \leq h/r \leq 2.0$	$2.0 \leq h/r \leq 50.0$
C_1	$0.850 + 2.628\sqrt{h/r} - 0.413h/r$	$0.833 + 2.069\sqrt{h/r} - 0.009h/r$
C_2	$-1.119 - 4.826\sqrt{h/r} + 2.575h/r$	$2.732 - 4.157\sqrt{h/r} + 0.176h/r$
C_3	$3.563 - 0.514\sqrt{h/r} - 2.402h/r$	$-8.859 + 5.327\sqrt{h/r} - 0.320h/r$
C_4	$-2.294 + 2.713\sqrt{h/r} + 0.240h/r$	$6.294 - 3.239\sqrt{h/r} + 0.154h/r$

Figure 25- Roark's Stress Concentration Formula for Geometries Such As Rod Bolt

$$K_t = C_1 + C_2 \frac{2h}{D} + C_3 \left(\frac{2h}{D} \right)^2 + C_4 \left(\frac{2h}{D} \right)^3$$

With stress concentrations applies, new safety factors are as follows.

Table 5-Mean stress, Allowable/ Actual Range, FSF With Stress Concentration

RPM	1000	2000	3000	4000	5000	6000
Safety Factor	31.0	7.67	3.35	1.84	1.14	0.77

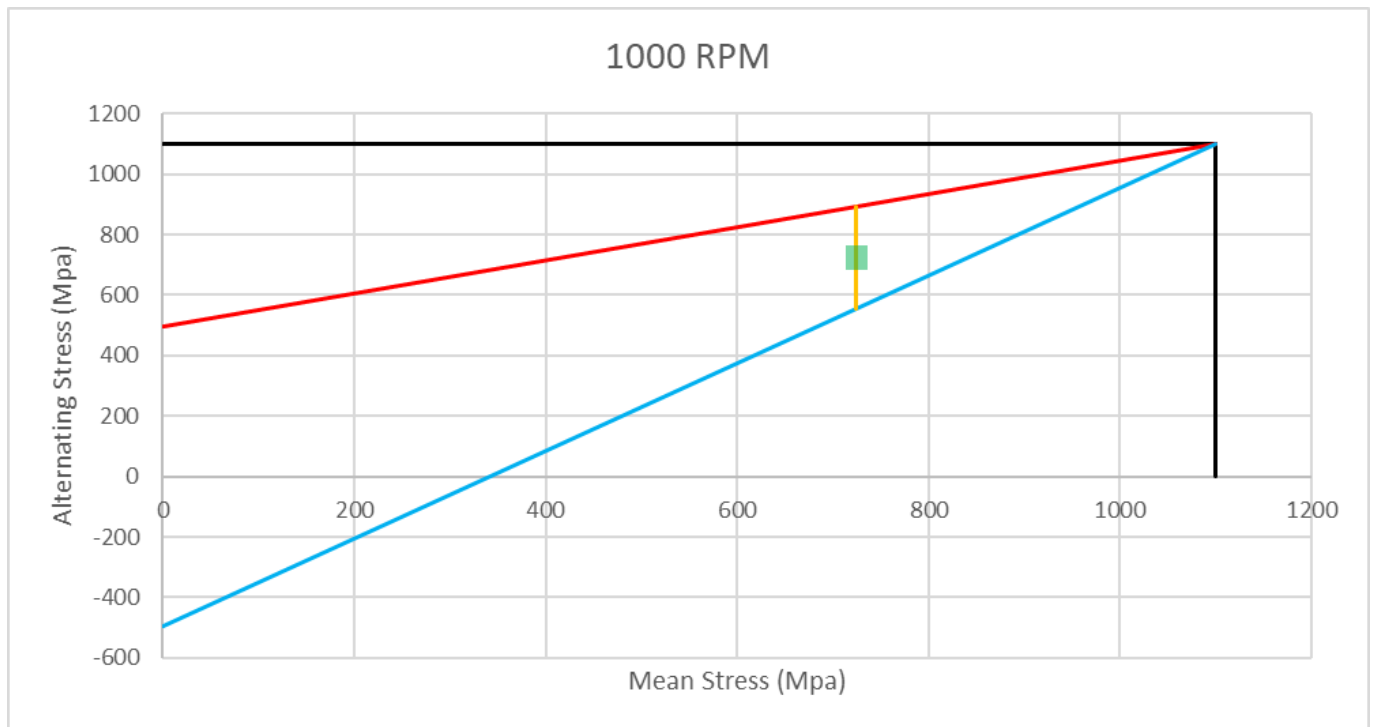


Figure 26-Goodman Diagram for Con Rod Bolt at 1000 RPM

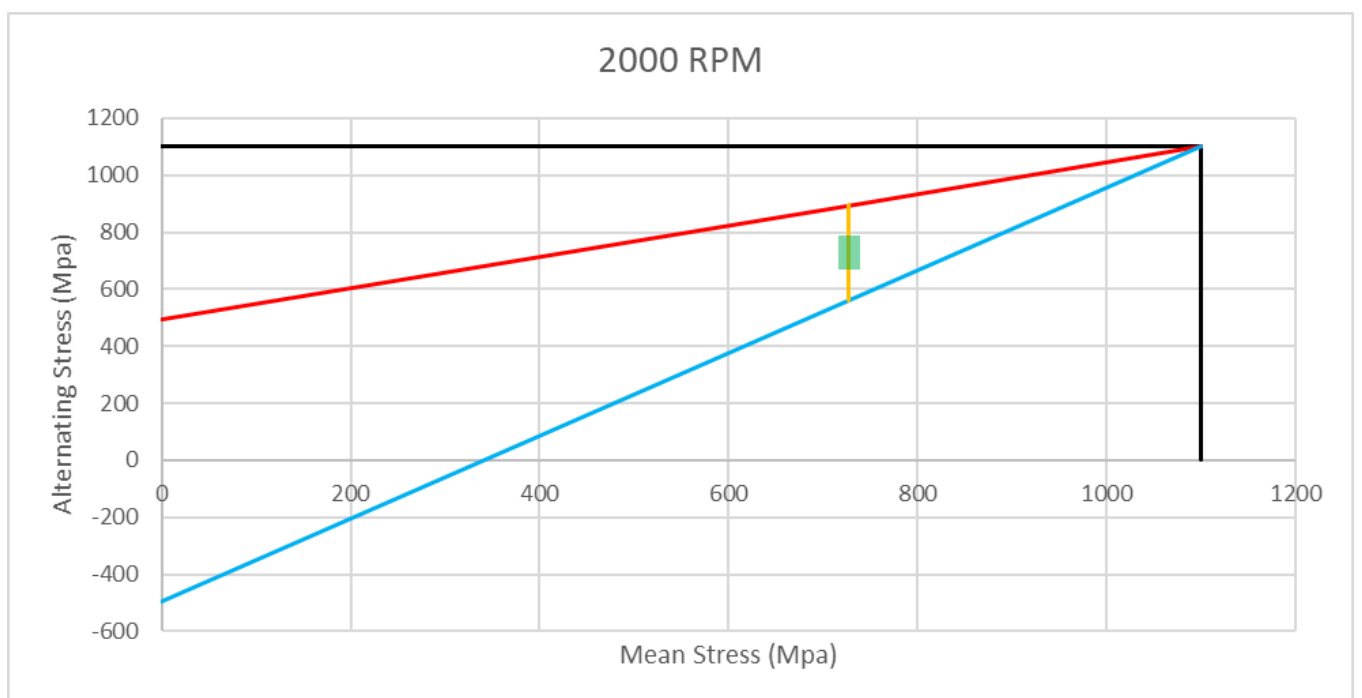


Figure 27- Goodman Diagram for Con Rod Bolt at 2000 RPM

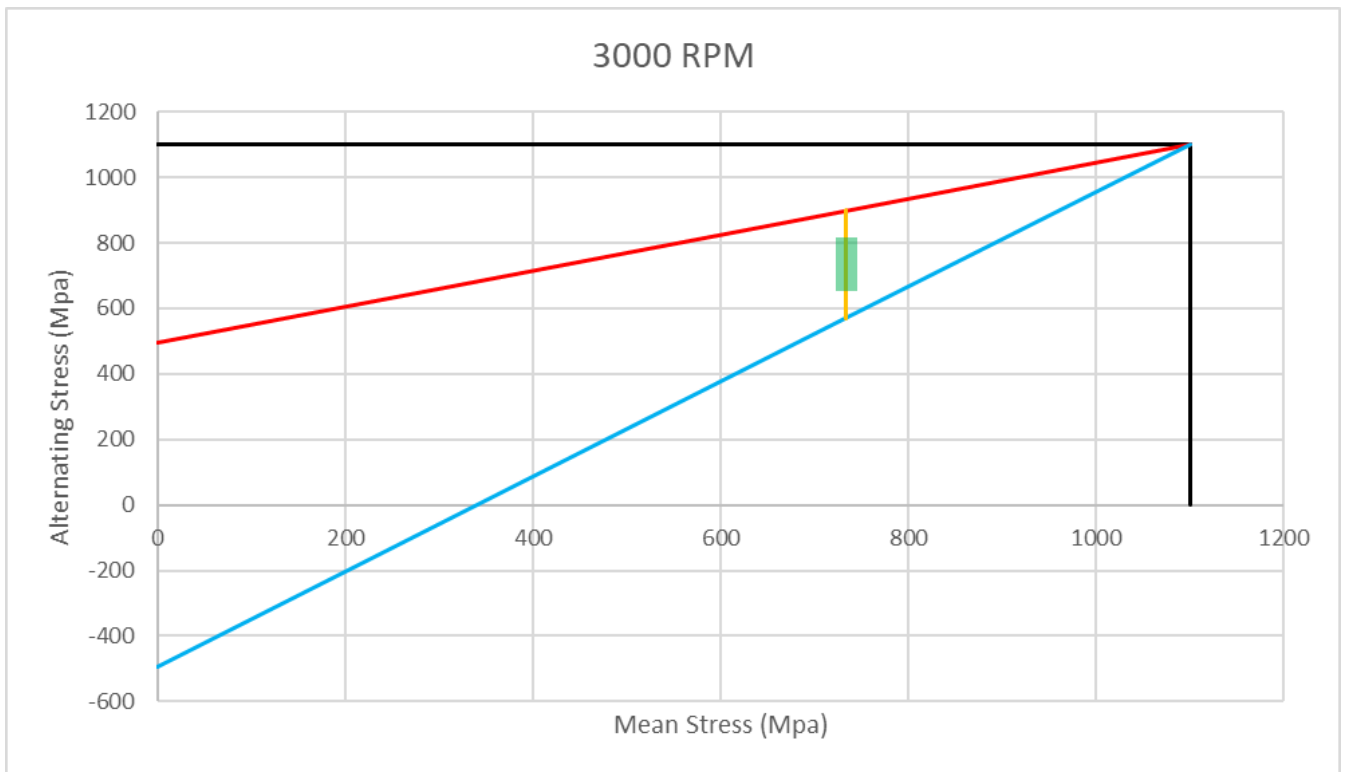


Figure 28-Goodman Diagram for Con Rod Bolt at 3000 RPM

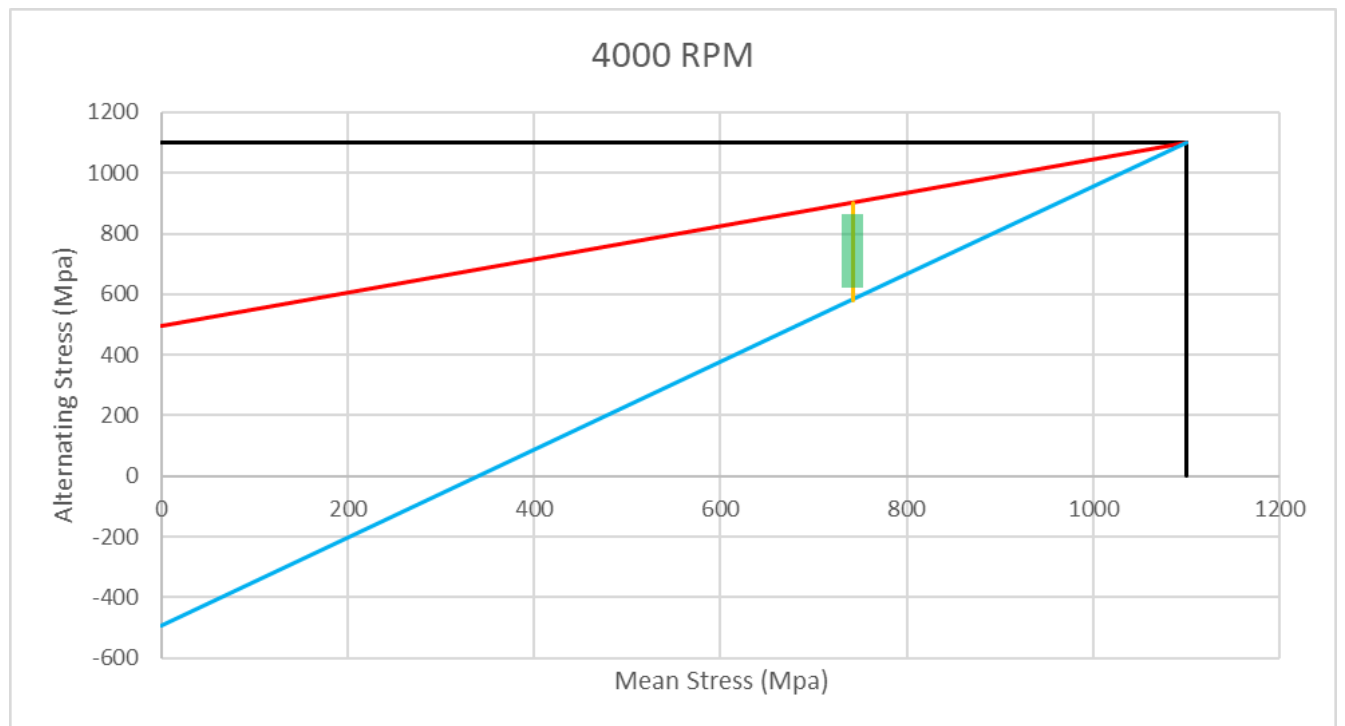


Figure 29-Goodman Diagram for Con Rod Bolt at 4000 RPM

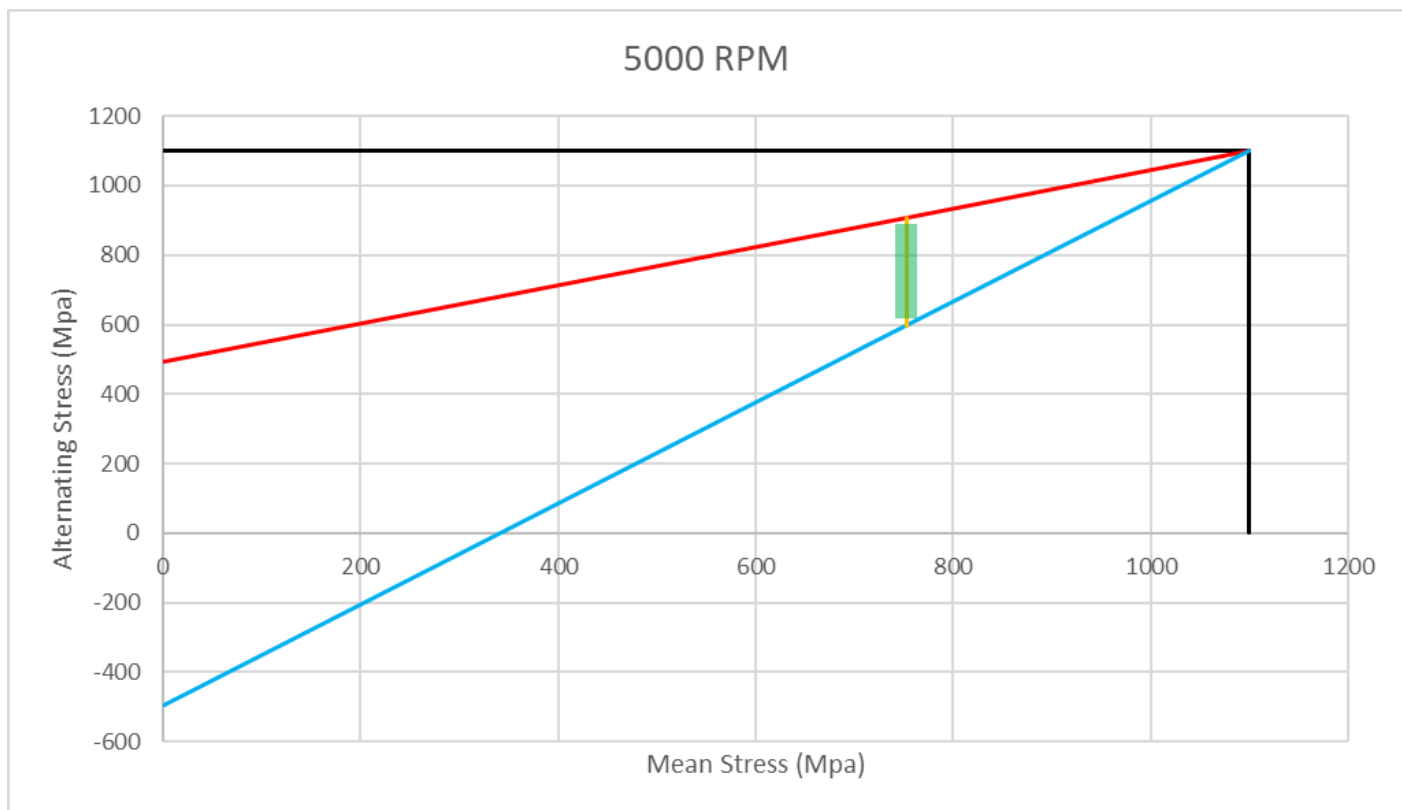


Figure 30-Goodman Diagram for Con Rod Bolt at 5000 RPM

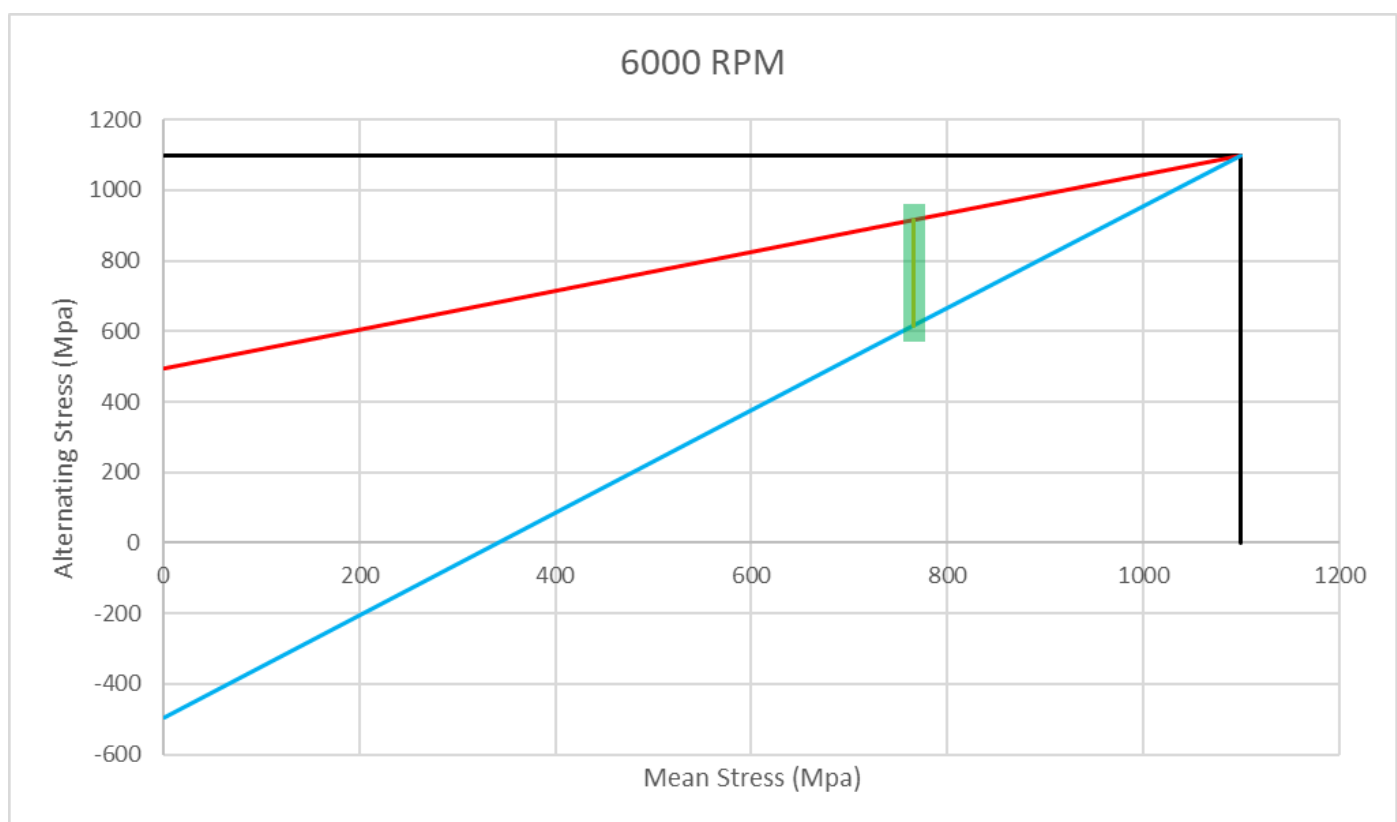


Figure 31-Goodman Diagram for Con Rod Bolt at 6000 RPM

Figures 30 and 31 alongside Table 4 show that at 5 and 6000 rpm, there is little to no factor of safety. At 6000rpm there is a factor of safety of 0.77, reducing the bolts fatigue life when exposed to that rpm, “values less than one indicate failure before the design life is reached” (Sri and Christina, n.d.).

Although under 1, the engine will only spend a small fraction of its life cycle at 6000rpm, in the engines lifecycle it is likely that it will not exceed the cycles required to fatigue the bolt to the point of failure for a long time but if being used in performance applications, changing the bolt would be required.

Part 2

With updated pressure data for 9000 RPM, another goodman diagram can be created.

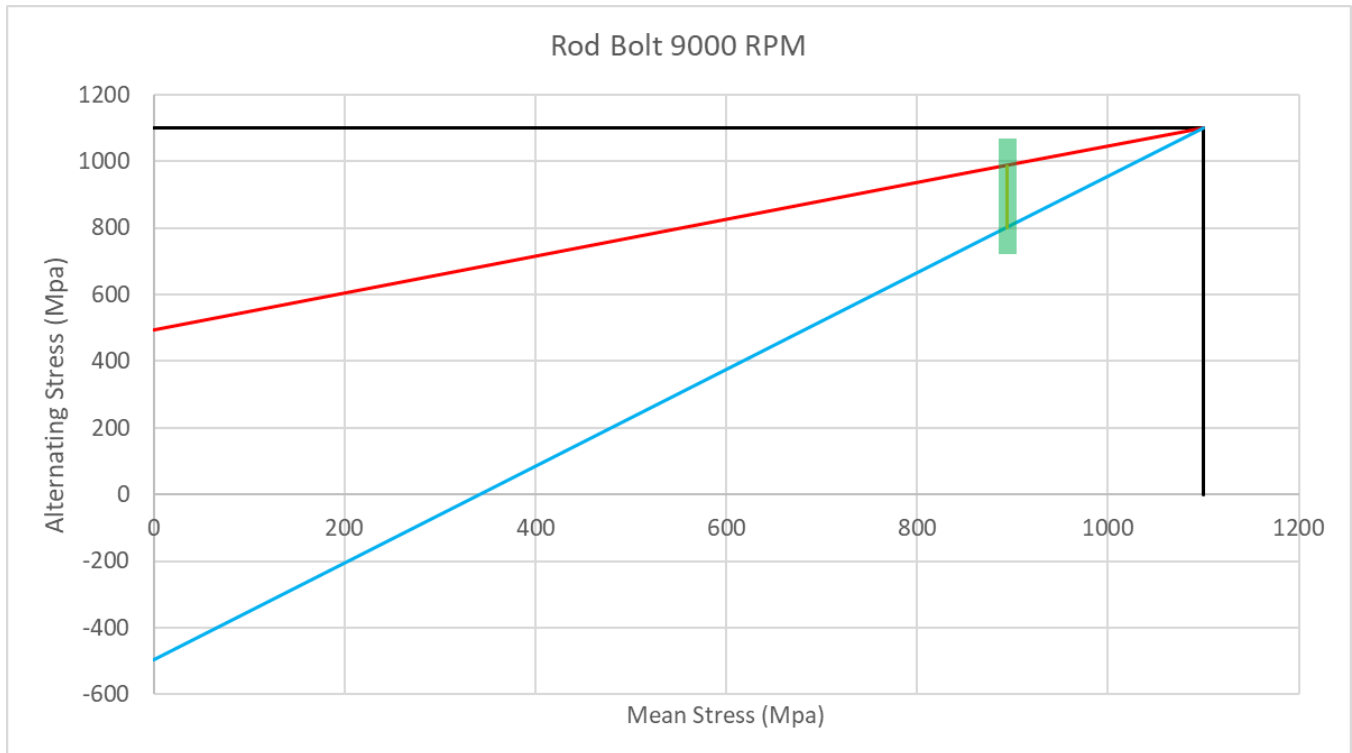


Figure 32- Goodman Diagram of Rod Bolt at 9000 RPM

Here, FSF is 0.53, which excessively exceeds the allowable range, if this were to be performed on the engine with standard bolts, failure would occur very quickly under cyclic loading. This could lead to catastrophic engine failure such as the rod punching through the engine block or damaging other components such as the crank, bearings, and oil system. Even without a catastrophic failure, the bolt will plastically deform leading to a loss in clamping force, increasing wear on bearings and inducing rod knock. There are multiple ways to combat this such as different material, increase bolt diameter, improve heat treatment, optimize preload torque, or redesigning threads and fillets which will be investigated in the next section.

Both the rod and Gudgeon pin maintain a factor of safety of over one, shown by the goodman diagrams in Figures 33 and 34, suggesting reaching that level the bolt would be the only bottom end component that would need upgrading. Its important to note that valve float will occur at high rpms if the valve springs are not stiff enough.

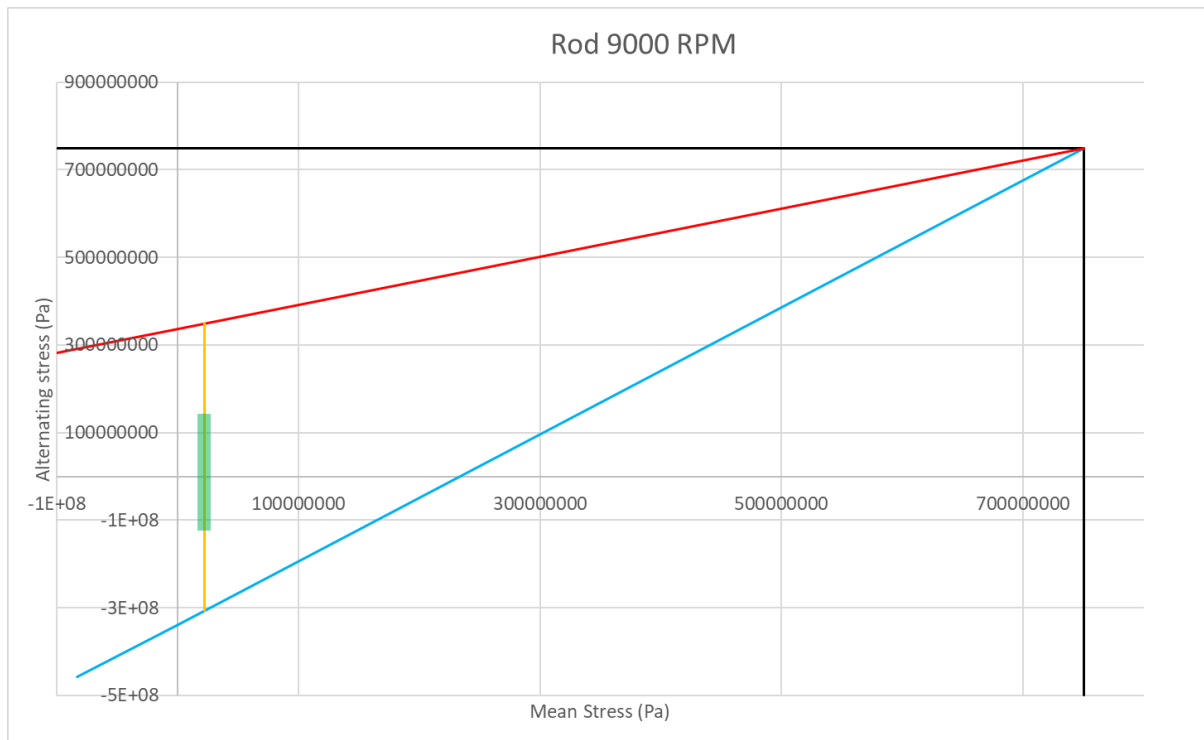


Figure 33-Goodman Diagram of Rod at 9000 RPM

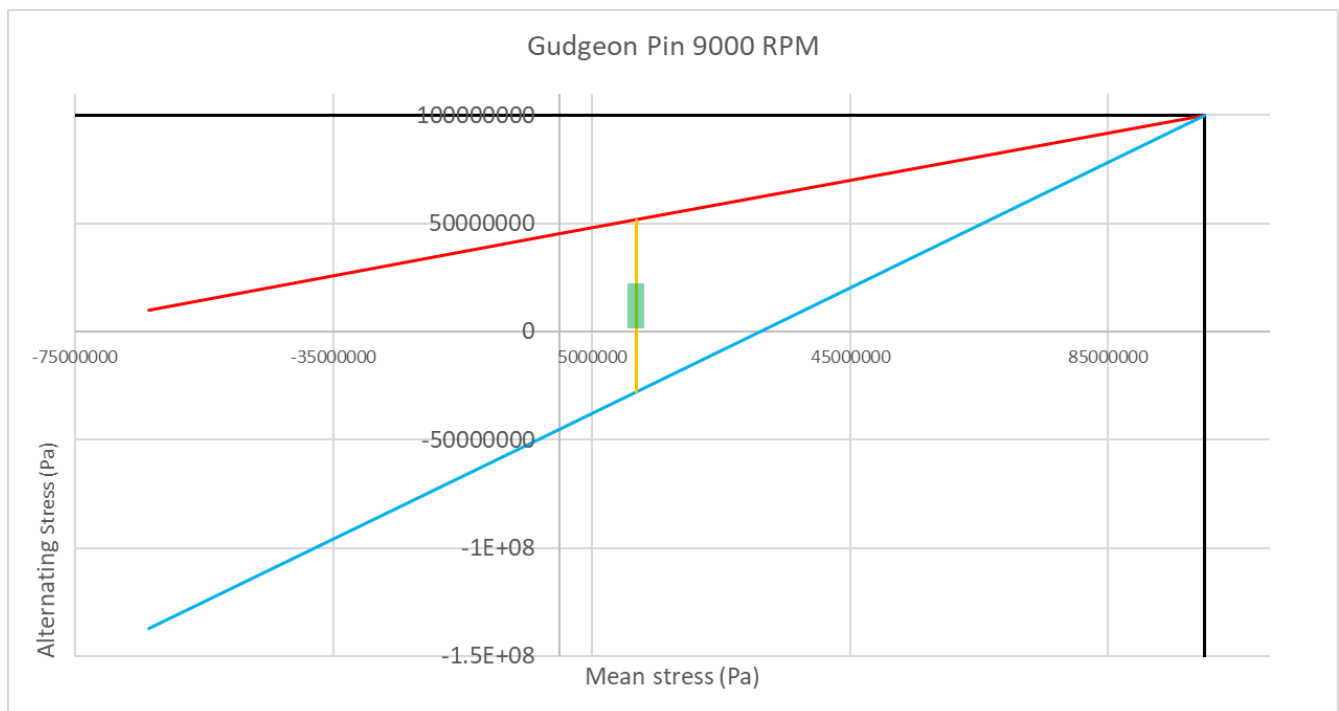


Figure 34-Goodman Diagram of Gudgeon Pin at 9000 RPM

Part 3

With a fatigue safety factor so low, increasing rpm to 9000rpm is not viable. Therefore changes to bolt design must be considered. The most effective way of doing this without having to change the geometry of other components is to change the lengths of various sections of the bolt. The bolt stiffness is a key factor as it contributes to calculating the extra bolt force which in turn effects the bolt force. Reducing bolt stiffness will decrease bolt force and associated stresses, increasing the FSF.

Changing Length

There are three areas of the bolt which can be changed shown in figure 35, with section one being the full width of the bolt under the head, part 2 being the unthreaded shank section, and part 3 being the threaded section which is not engaged.

The first change was to increase all sections by a specific factor shown in Table 6 resulting in the following FSFs. When applying a factor of 1.6 to the lengths, a value close to the original FSF at 6000 RPM was calculated and with a factor of two being applied to lengths FSF exceeds one.

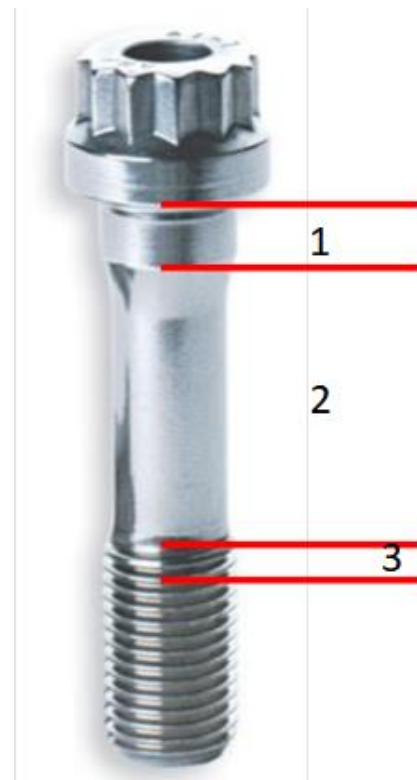


Table 6-Dimensions and FSFs of Bolt Length Changes

Figure 35- Rod Bolt Sections

	multiplier				
	1.2	1.4	1.6	1.8	2
L1	0.24	0.28	0.32	0.36	0.4
L2	5.76	6.72	7.68	8.64	9.6
L3	26.712	31.164	35.616	40.068	44.52
Lt (minus engaged thread)	32.712	38.164	43.616	49.068	54.52
K1	43986250	37702500	32989687.5	29324166.67	26391750
K2	1832760.417	1570937.5	1374570.313	1221840.278	1099656
K3	395204.4025	338746.6307	296403.3019	263469.6017	237122.6
1/K	3.0987E-06	3.61514E-06	4.13159E-06	4.64804E-06	5.16E-06
k	322716.4343	276614.0866	242037.3258	215144.2896	193629.9
FSF	2.9	3.36	3.82	4.28	4.73
FSF with stress concernratio	0.63	0.73	0.83	0.93	1.03

To determine which section had the greatest effect, each of the lengths were changed one at a time with the remaining two set back to standard values, from this the % change of FSF per % of length change can be found and is shown in table 7.

Table 7- Dimensions and FSFs of Bolt Length Changes

	L3 change	L1 change	L2 change
L1	0.2	0.4	0.2
L2	5.76	5.76	9.6
L3	44.52	26.712	26.712
Lt	50.48	32.872	36.512
K1	52783500	26391750	52783500
K2	1832760.417	1832760.417	1099656.25
K3	237122.6415	395204.4025	395204.4025
1/K	4.7818E-06	3.11385E-06	3.45866E-06
K	209126.3867	321145.6559	289129.6012
FSF	4.4	2.92	3.23
FSF with stress concernration	0.95	0.63	0.7
length increase (%)	85.17975055	20.58694057	33.93983859
FSF increase (%)	79.24528302	18.86792453	32.0754717
FSF increase per % length increase	0.930330067	0.916499684	0.945068481

By individually changing each of these values, there is slight difference in the effect they have on FSF in comparison to each other, but changing L1 gives a slightly greater return for length increase. A more efficient way to achieve a FSF of 1 is to make L1 13.5mm, L2 10mm, and L3 30mm. This gives a FSF of 1.01 with an overall length without the engaged thread of 53.5, opposed to 54.5 when evenly increasing all values.

Changing Area

Another option is to change the Bolt areas, the standard bolt has an outer diameter of 7.4mm and a thread pitch of 1mm. The FSF results for M8x1, M9x1, and M10x1.25 are shown in table 8.

Table 8- Dimensions and FSFs of Bolt Area Changes

	Min Stress	Max Stress	Mean Stress	Stress Range	Allowable stress	FSF	FSF with stress concernration
M8	698.3158158	759.8467173	729.0812665	61.53090149	333.8268601	5.425353	1.175502215
M9	534.6480465	581.7576429	558.2028447	47.10959646	487.6174398	10.3507	2.242669609
M10	446.9221221	486.3018991	466.6120106	39.37977696	570.0491905	14.47568	3.136422302
M7.4 (Std)	856.6704798	932.1545312	894.4125055	75.48405136	185.028745	2.45123	0.531103855

Here, FSF can be increased to 1.18 simply by changing to an M8 bolt. This way length does not have to be increased which could cause issues such as the bolt sticking out the top of the rod cap leaving thread that was once engaged to be exposed changing stress values.

Changing Material

The standard rod bolt material has an ultimate tensile strength of 1100 Mpa, this could be a material such as carbon steel AISI 1060 which has a UTS ranging from 900-1100 MPA, if bolt dimensions were to remain the same this would not be suitable to operate at 9000 rpm. ARP 2000 which is a quench and tempered heavily alloyed martensitic steel, is an extremely popular choice for high rpm and high-performance applications. This material has a UTS of 2000 Mpa, and modulus of elasticity of around 210 Gpa, similar to the standard material. When applied to calculate stresses, it is found to have a FSF of 1.04, this combined with a length increase of factor 1.4 gives an FSF with stress concentration applied of 1.41 and 6.48 without. This is within to the recommended value of 6-8 as per Safety culture (SafeCulture, 2024), this will be the final geometry represented in figure 36.

Table 9-Mean stress, Allowable/ Actual Range, FSF with material change

Mean Stress (Mpa)	Allowable Range (Mpa)	Actual Range (Mpa)	FSF	FSF with stress concentration
1586.3	372.3	57.4	6.5	1.4

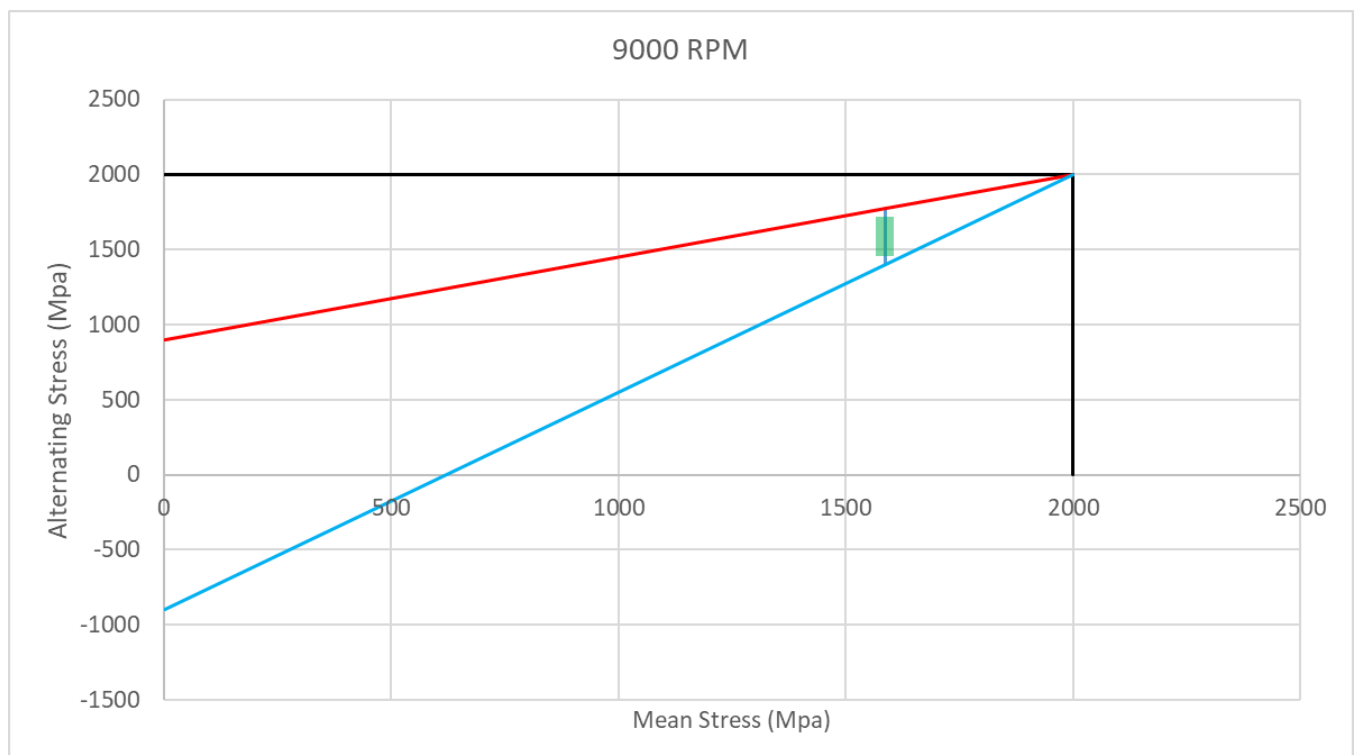


Figure 36- Goodman Diagram of Rod Bolt with ARP 2000 Material

Material Information from Ansys Granta is shown in figure 37, this is not exactly the ARP material as the formula is not public information, but these properties are similar to ARPs material.

Mechanical properties

Young's modulus	①	204	-	215	GPa
Specific stiffness	①	25.9	-	27.4	MN.m/kg
Yield strength (elastic limit)	①	1.75e3	-	1.84e3	MPa
Tensile strength	①	1.95e3	-	2.05e3	MPa

Figure 37- ARP 2000 Material Information

Part 4

When comparing standard vs cross plane crankshafts, the key difference is offset that each cylinder has from each other. A standard crankshaft simply has an offset of 180 degrees between each cylinder. When combined to give resultant torque, the output is shown in figure 38.

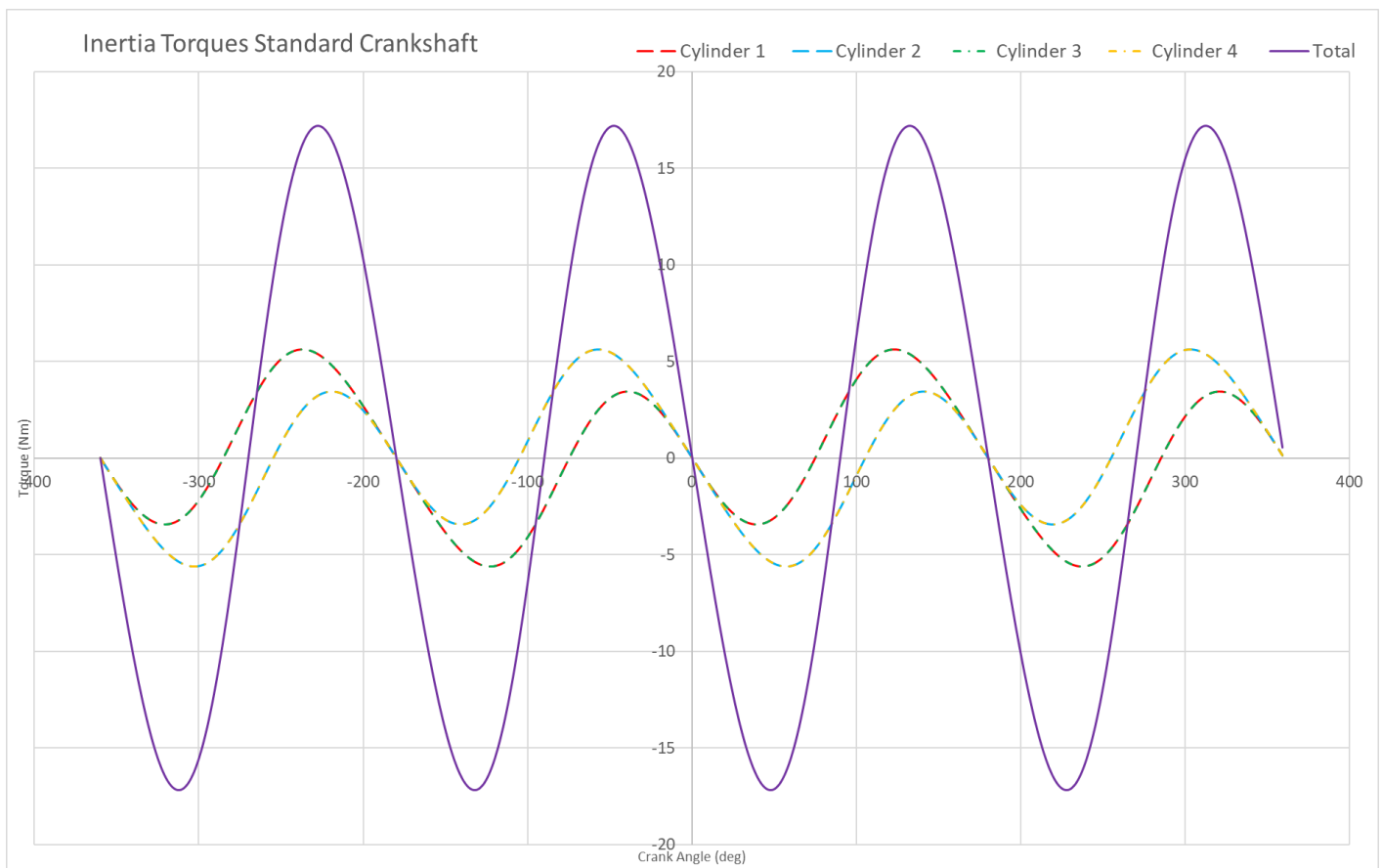


Figure 38- Inertia Torque Produced from Standard Crankshaft

Cross plane crankshafts work differently, they have a firing order of 1, 3, 2, 4 with an offset of 270° on cylinder 3 (-90°), 180° on cylinder 2 (90°), and 90° on cylinder 4 (180°). This results in opposing cylinders inertia force cancelling each other out to decrease the overall inertia torque created shown in figure 39.

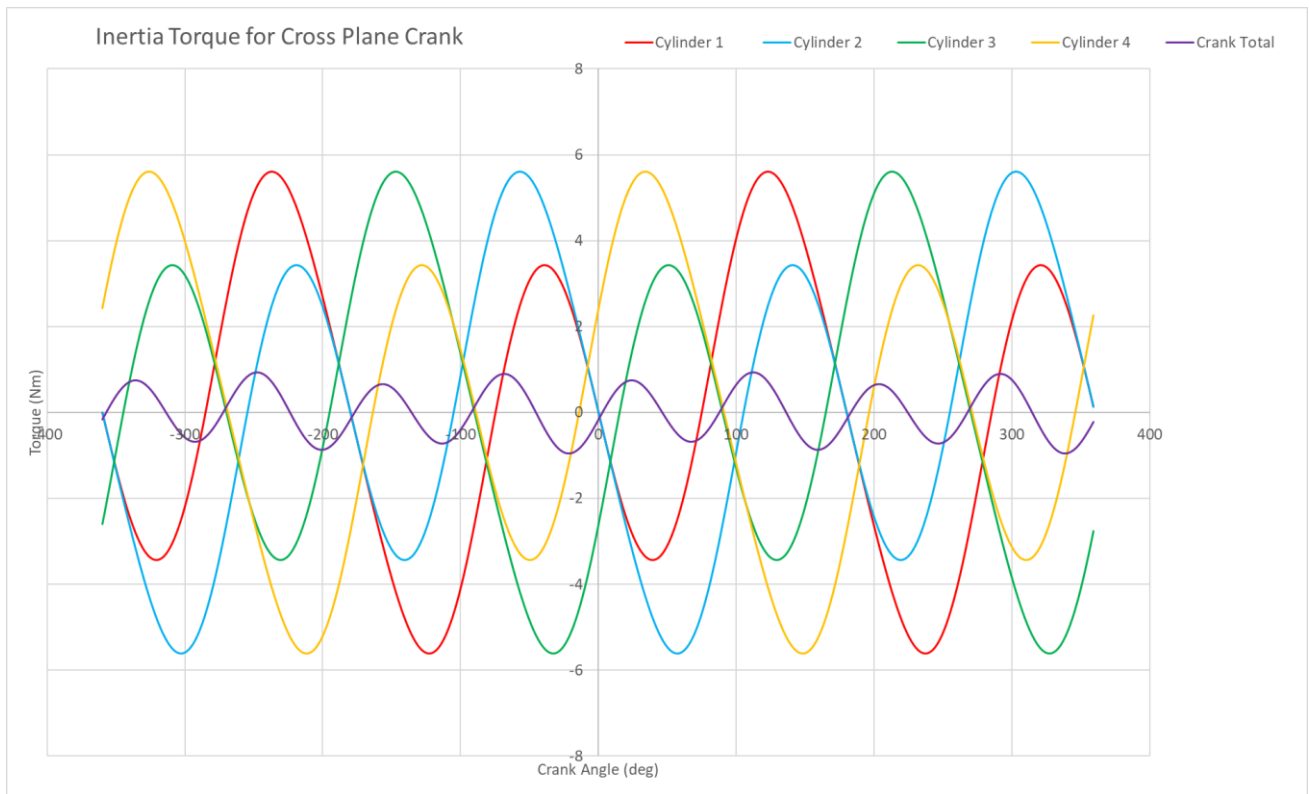


Figure 39- Inertia Torque Produced from Cross Plane Crankshaft

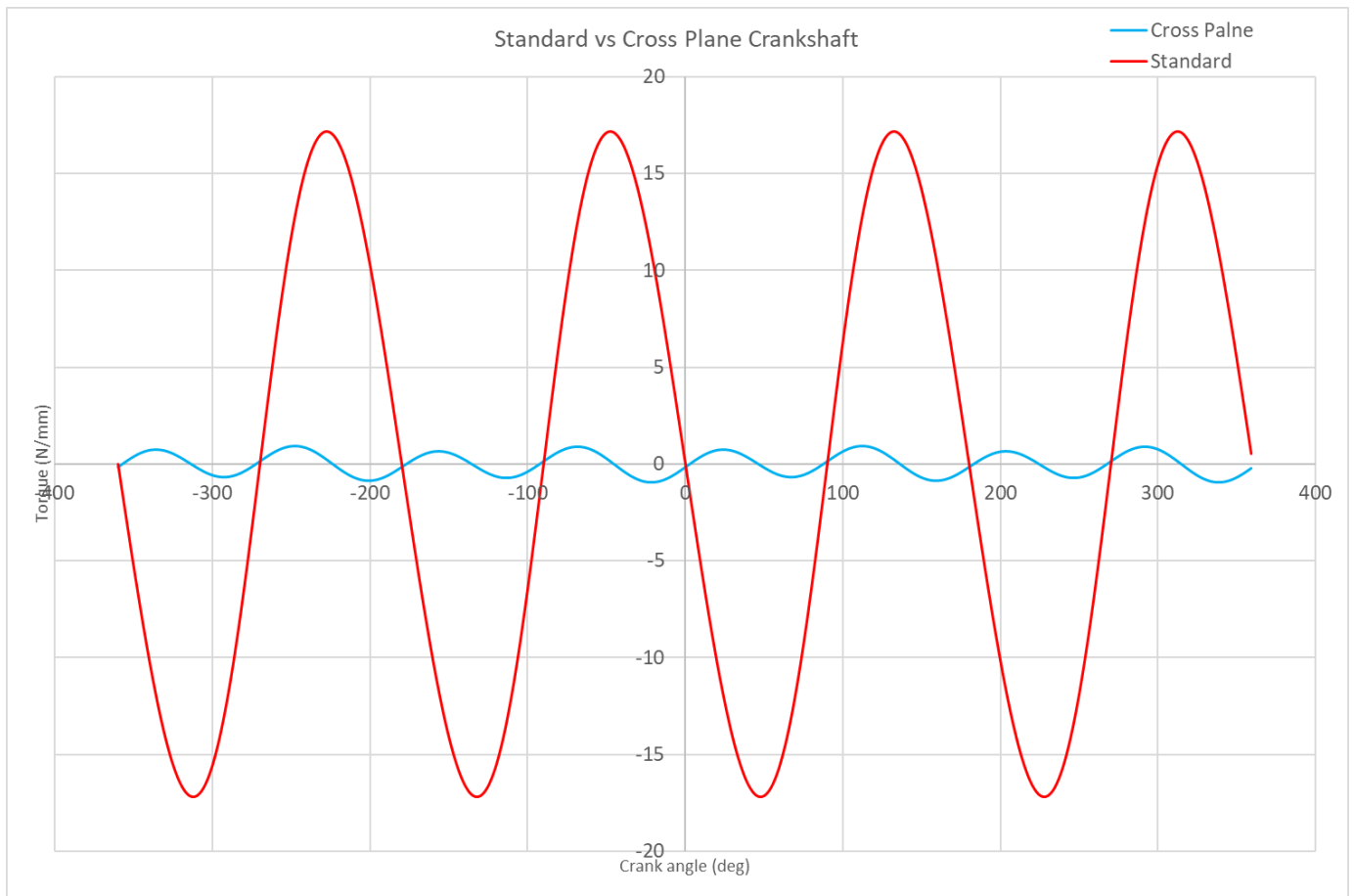


Figure 40- Resultant Torque, Cross Plane vs Standard

Figure 40 shows standard vs cross plane crank and the effect offsetting the cylinders has to resultant inertia torque values. Cross plane crankshafts are a popular option for V8s, especially muscle cars, this is due to their 'smooth running because of consistent firing every 90 degrees and the low-end torque' (Sanchez, 2020) produced from this layout. The primary drawback is the increased weight due to the need to add counterweights, this increases rotational mass resulting in a lower rev limit. On the other hand flat plane cranks offer a more balanced setup reducing rotational mass allowing a higher revving limit alongside quicker engine response and a more compact design. The main drawbacks of the flat plane are vibrations due to 180° offset as well as a lack of torque at lower RPMs. Flat plane is the popular choice in performance and supercar engines, many European supercars manufacture opt for flat plane due to their high revving and lightweight nature.

Conclusion

In conclusion, there are multiple ways to improve the fatigue life of a con rod bolt as discussed throughout this report, both the con rod and gudgeon pin achieve good fatigue lives allowing these components to be left the same if increasing RPM to nine thousand. To ensure the bolt will continue working as expected the geometry mentioned in part 3 can be used which increases the unused bolt length by 40% alongside a material change to ARPs steel alloy. This will allow the bottom end of the engine to manage the increased rpm theoretically infinitely. There are many other ways to change bolt geometry to improve fatigue life but the effect this has on surrounding components should also be considered when developing a component in industry.

2447 words.

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