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Con-rod accelerations, forces and displacements – M6X00410

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1 Introduction

This report aims to discuss the accelerations, displacements and forces acting on a connecting rod during engine operation. The connecting rod is a fundamental component in an internal combustion engine, it '*...transmits the gas forces from the piston to the crankshaft.*'

(Rao, 2013)

The connecting rod converts the reciprocating motion of the pistons into rotary motion of the crankshaft. Both con-rod and crankshaft are subjected to many forces during operation (see *figure 1*).

Data is from a Ford 1.6l EcoBoost GTDI, a turbocharged direct injection inline four-cylinder engine, with DOHC twin variable camshaft timing. This engine is efficient and runs lean, rated at Euro 5 emissions standard.

The results are estimations since the crankshaft angular velocity is assumed to be constant.

Piston pin offset has been disregarded. First and second order harmonics have been considered during the Fourier series. However since normal values for rod ratio are usually around 0.3, the higher-level terms have been ignored. Forces exerted on the piston are assumed to be symmetric in the x direction however in reality the piston deforms during operation.

'The cylinder has actually become elliptical due to the rotation of the piston about the wrist pin when the piston moves toward the bottom dead centre. In addition, the high temperature during the operation process also affects the deformation of the piston and cylinder. Consequently, the forces exerted on the piston are not symmetric in the x direction.'

(Duy, 2018)

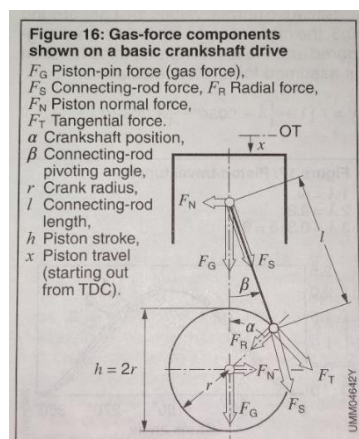


Figure 1 - Force components shown on a basic crankshaft drive (Dipl. - Ing. Karl-Heinz Dietsche, 2018)

2 Equations used

<i>Gas force,</i>	$F_{Gas} = \text{Average Pressure} \times \frac{\pi(\text{Bore})^2}{4}$
<i>Acceleration</i>	$= R\omega^2 \cos \theta + \frac{R}{L} R\omega^2 \cos 2\theta$
<i>Stroke</i>	$= R \left(1 - \cos \theta + \frac{L}{R} \left(1 - \sqrt{1 - \left(\frac{R}{L}\right)^2 \sin^2 \theta} \right) \right)$
<i>First order inertia force</i>	$= m_{piston} \cdot R\omega^2 \cos \theta$
<i>Second order inertia force</i>	$= m_{piston} \cdot \frac{R}{L} R\omega^2 \cos 2\theta$
<i>Inertia force,</i>	$F_{Inertia} = m_{piston} \left(R\omega^2 \cos \theta + \frac{R}{L} R\omega^2 \cos 2\theta \right)$ $= m_{piston} \cdot \text{Acceleration}$
<i>Net force,</i>	$F_{Net} = F_{Gas} - F_{Inertia}$
<i>Rod angle,</i>	$\varphi = \sin^{-1} \left(\frac{R}{L} \sin \theta \right)$
<i>Rod force,</i>	$F_{Rod} = \frac{F_{Net}}{\cos \varphi} = \frac{F_{Net} - F_{Inertia}}{\cos \varphi}$
<i>Side force,</i>	$F_{Side} = F_{Net} \cdot \tan \varphi$
<i>Frictional force,</i>	$F_F = \mu \cdot F_{Side} = 0.1 \times F_{Side}$
<i>Instantaneous torque</i>	$= F_{Rod} \cdot R \cdot \sin(\theta + \varphi)$
<i>Inertia torque</i>	$= F_{Inertia} \cdot R \cdot \sin(\theta + \varphi)$
<i>Gas torque</i>	$= F_{Gas} \cdot R \cdot \sin(\theta + \varphi)$
<i>Compressive force</i>	$= \text{Maximum Rod force}$
<i>Tensile force</i>	$= \text{Minimum Rod force}$
<i>Compressive stress</i>	$= \frac{\text{Compressive force}}{\text{Conrod cross sectional area}}$
<i>Tensile stress</i>	$= \frac{\text{Tensile force}}{\text{Conrod cross sectional area}}$
<i>Static Safety Factor</i>	$= \frac{\text{Ultimate Tensile Strength}}{\text{Nominal Stress}}$
<i>Maximum compression distance</i>	$= \frac{\text{Compressive force} \times \text{Conrod length}}{\text{Young's Modulus} \times \text{Conrod cross sectional area}}$
<i>Maximum Stretch</i>	$= \frac{\text{Tensile force} \times \text{Conrod length}}{\text{Young's Modulus} \times \text{Conrod cross sectional area}}$
<i>Buckling load</i>	$= \text{Maximum gas force} \times \text{Factor of safety}$

3 Component Specifications

	A	B	C	D	E	F	G	H	I	J	K
1	PRODUCTION ENGINE				MODIFIED ENGINE						
2	Conrod	0.2674	kg		Conrod	0.18718	kg		6AL-4V Ti rods		
3	Conrod cap	0.1038	kg		Conrod caps	0.07266	kg		Forged Wossner pistons		
4	Conrod bolts	0.03519	kg		Conrod bolts	0.0355	kg		ARP 2000 series bolts		
5	Conrod assembly	0.4064	kg		Conrod assembly	0.29534	kg				
6	Gudgeon pin	0.08022	kg		Gudgeon pin	0.08022	kg				
7	Piston	0.2577	kg		Piston	0.282	kg				
8	Piston ring	0.00549	kg		Piston ring	0.00549	kg				
9	Circlip	0.00077	kg		Circlip	0.00077	kg				
10	Reciprocating Assembly	0.3504	kg		Reciprocating Assembly	0.38023	kg				
11	Conrod cross sectional area	0.00014	m ²		Conrod cross sectional area	0.00016	m ²				
12											
13											
14											
15	PRODUCTION ENGINE				MODIFIED ENGINE						
16	Conrod length	0.134	m		Conrod length	0.13402	m				
17	Bore	0.079	m		Bore	0.0795	m				
18	Stroke	0.0814	m		Stroke	0.0814	m				
19	Compression ratio	10.1 : 1			Compression ratio	9.5 : 1					
20	Crank throw	0.0407	m		Crank throw	0.0407	m				
21	UTS	7.5E+08	N/m ²		UTS	9.5E+08	N/m ²				
22	Young's modulus	1.7E+11	N/m ²		Young's Modulus	1.10E+11	N/m ²				
23	Piston ring height	0.00012	m		Piston ring height	0.00012	m				
24											

Figure 2 - Component and material specifications for the production and modified engine

The production engine has cast hypereutectic pistons, which contain a high silicon content and maintain their strength better at high temperatures than standard cast pistons. This offers increased reliability over standard cast pistons; however, they are not as strong as forged pistons.

'Piston crowns and top ring lands feature a performance enhancing coating, while a friction-reducing coating is found on the asymmetrical skirts.' (Ferrara, 2016)

The piston pins feature a DLC coating in order to increase wear resistance and durability.

The connecting rods use *'... a hot-forged powder metal process and cracked rod cap process is used in the manufacturing process. While those processes add strength, we are concerned that the overall cross sectional area of the rod beams may not be enough.'* (Ferrara, 2016)

The OEM con-rod bolts are a torque-to-yield design, therefore cannot be reused. The bolts Ultimate Tensile Strength (UTS) is approximately 1100N/mm².

The modified engine features forged Wössner 4032 T6 aluminium pistons with PTFE coated skirts.

The crowns of the pistons are polished to minimise the chance of detonation, lowering the compression ratio so the engine can take more boost before detonation. The modified Wössner pistons also have a 0.5mm wider bore.

The modified engine features custom 6Al-4V titanium connecting rods, with 14.7% larger cross-sectional area since the Young's Modulus of titanium is less than powder forged steel.

Since the con-rod bolts are generally the least durable component in a modified engine it also has ARP 2000 series connecting rod bolts with a UTS of approximately 1300N/mm².

4 Peak Gas Forces

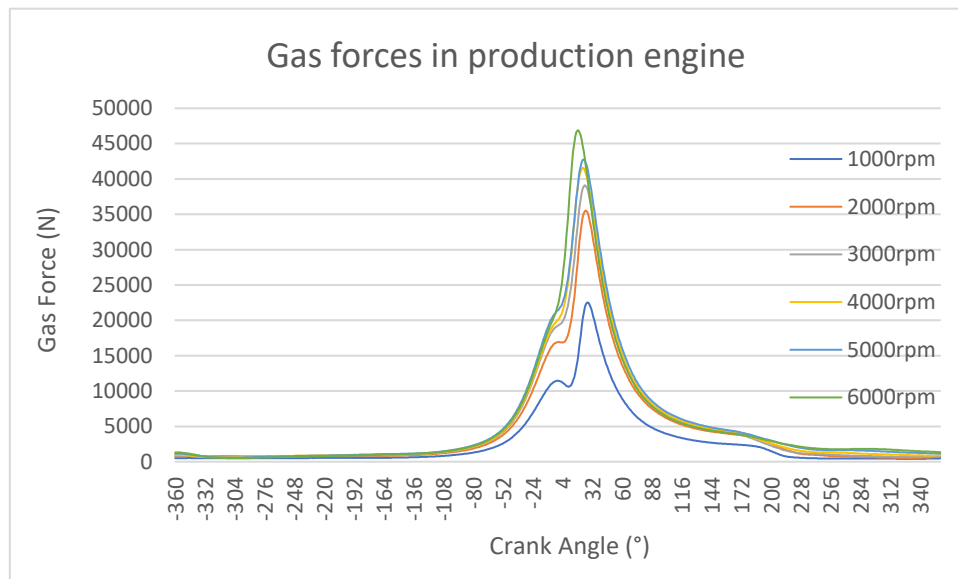


Figure 3 - Gas force in the production engine for two crankshaft revolutions

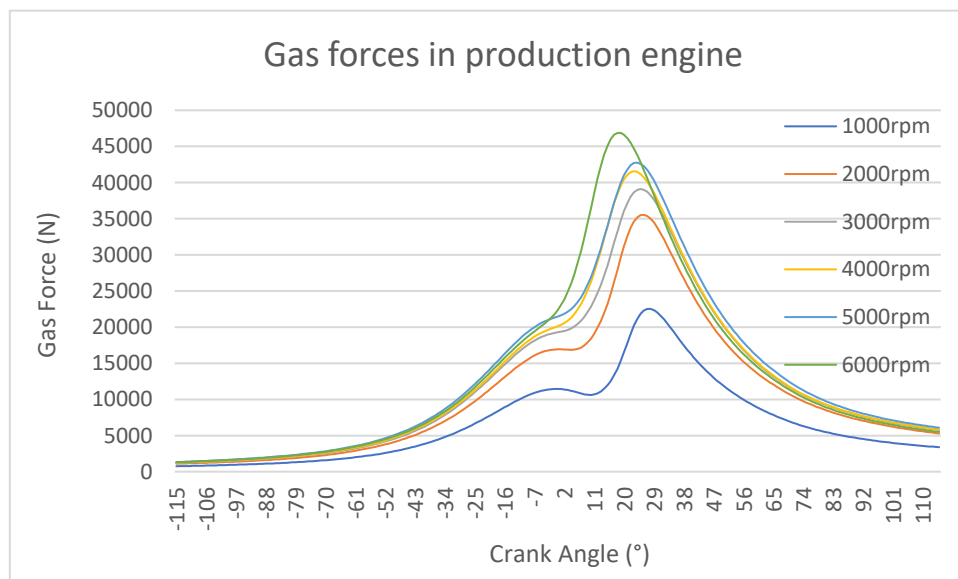


Figure 4 - Gas Force in the production engine from -115° to 115°

Peak gas force occurs shortly after Top Dead Centre (TDC) at each RPM (see *figure 3 and 4*). Gas force is produced by the combustion of the air-fuel mixture. If peak pressure occurred at TDC then there would be a very high compressive force on the piston pin, con-rod and crankshaft journal bearing. Having peak force at TDC will mean the gas force will produce maximum rotational acceleration of the crankshaft. Peak gas force occurs at 28° crank angle at 1000rpm and 18° at 6000rpm. The peak gas force occurs earlier at 6000RPM because the ignition timing has been advanced. The time taken for the air-fuel mixture to burn is constant, therefore at higher RPMs spark needs to be earlier.

The values for gas pressure are estimates since crankcase pressure acts on the underside of the piston opposing the gas force.

'In addition, the crankcase pressure generated by oil pressure on the underside is normally determined in the experiment. This force is very small compared to the total force exerted on the top of the piston.'

(Duy, 2018)

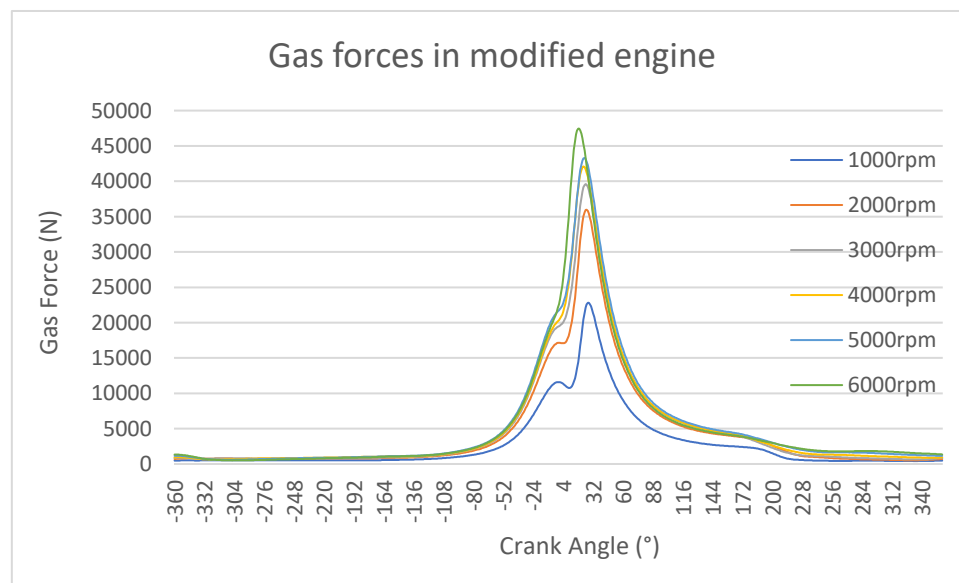


Figure 5 - Gas force in the modified engine for two crankshaft revolutions

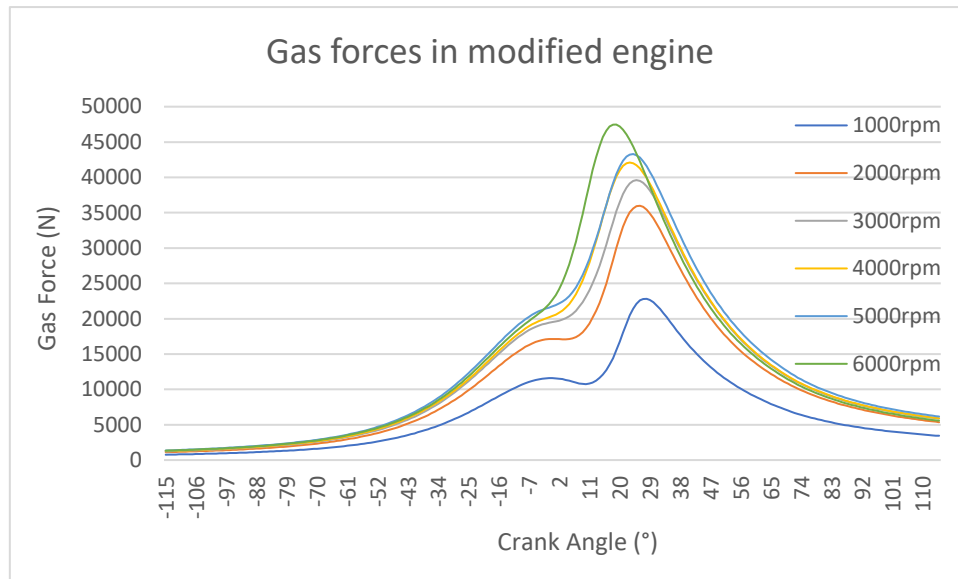


Figure 6 - Gas Force in the modified engine from -115° to 115°

The modified engine's bore was widened by 0.5mm, therefore gas forces are slightly larger than the production engines (see figure 5 and 6) as the piston surface area is larger.

'The advantages of using a large cylinder bore are the reduction in overall cylinder height because of reduced crank throw and length of rod, a stiffer crankshaft because of the increased overlap of pin and journal giving greater torsional rigidity and the reduced mean piston speed desirable for better durability.'

(J.G.Giles, 1968, pp. 24-25)

5 Inertia Loads

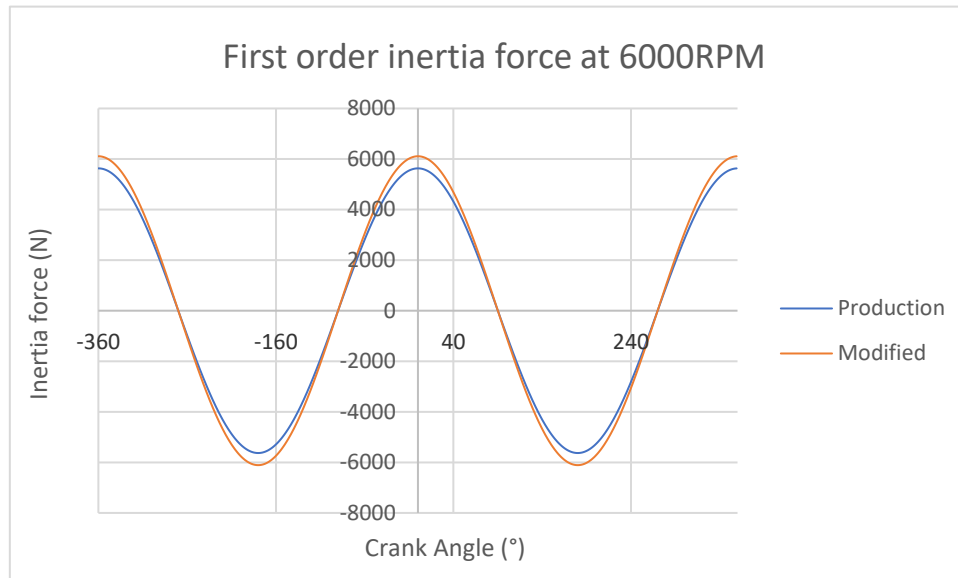


Figure 7 - First order inertia force at 6000RPM for production and modified engine

Figure 7 shows first order inertia loads for both the production and modified engine. The first order inertia load is larger for the modified engine since the piston weight is heavier than in the production engine. Maximum first order inertia load at 6000rpm of production engine is 5630.13N and modified engine is 6109.43N. The first order inertia force occurs from the reciprocating components inside the engine. First order inertia force can be balanced by counterweights on the crankshaft.

'Four-cylinder. All primary forces and couples balanced. Secondary forces out of balance acting along bore axes varying from one maximum positive to zero to maximum negative twice every engine revolution.'

(J.G.Giles, 1968, p. 45)

Since the 1.6 EcoBoost has a flat plane crankshaft it is in primary balance.

'An even-firing inline-four engine is in primary balance because the pistons are moving in pairs, and one pair of pistons is always moving up at the same time as the other pair is moving down. However, piston acceleration and deceleration are greater in the top half of the crankshaft rotation than in the bottom half, because the connecting rods are not infinitely long, resulting in a non sinusoidal motion.'

(Georgiev, 2011)

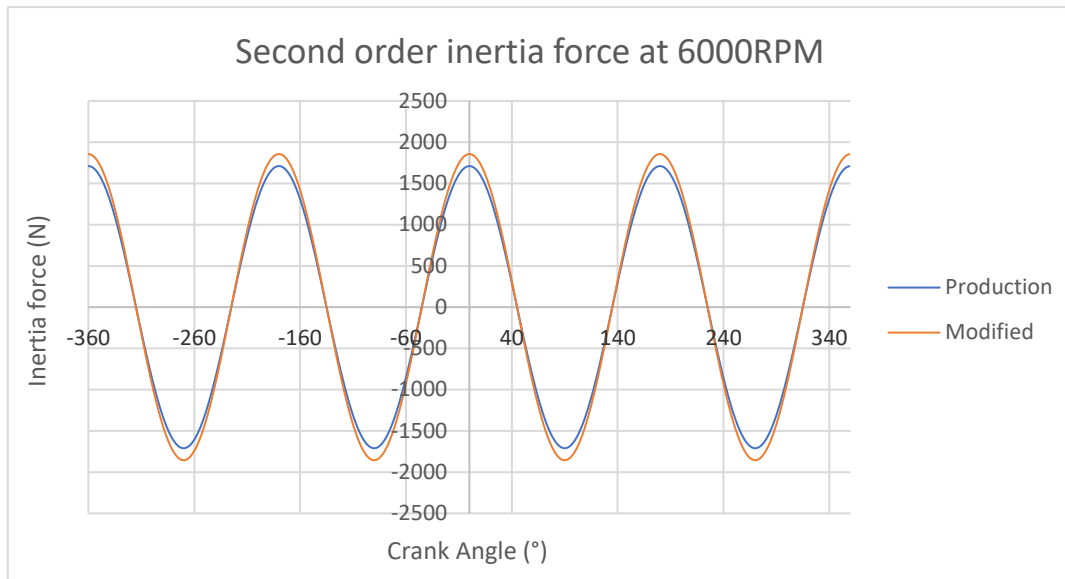


Figure 8 - Second order inertia force at 6000RPM for production and modified engine

As a result of the non-sinusoidal motion of the crankshaft '...two pistons are always accelerating faster in one direction, while the other two are accelerating more slowly in the other direction, which leads to a secondary dynamic imbalance that causes an up-and-down vibration at twice crankshaft speed. This imbalance is tolerable in a small, low-displacement, low-power configuration, but the vibrations get worse with increasing size and power.' (Georgiev, 2011)

Second order inertia load cannot be balanced by the crankshaft as they are at twice engine speed (see figure 8). A balance shaft at twice engine speed can reduce second order inertia forces. Increasing the length of the conrod will also reduce second order inertia forces.

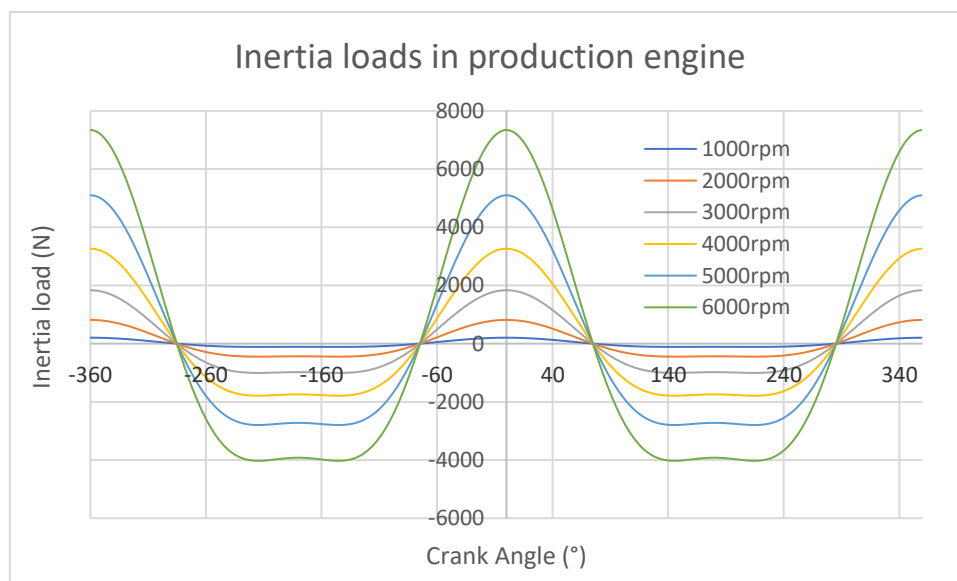


Figure 9 - Total inertia load at each RPM for production engine

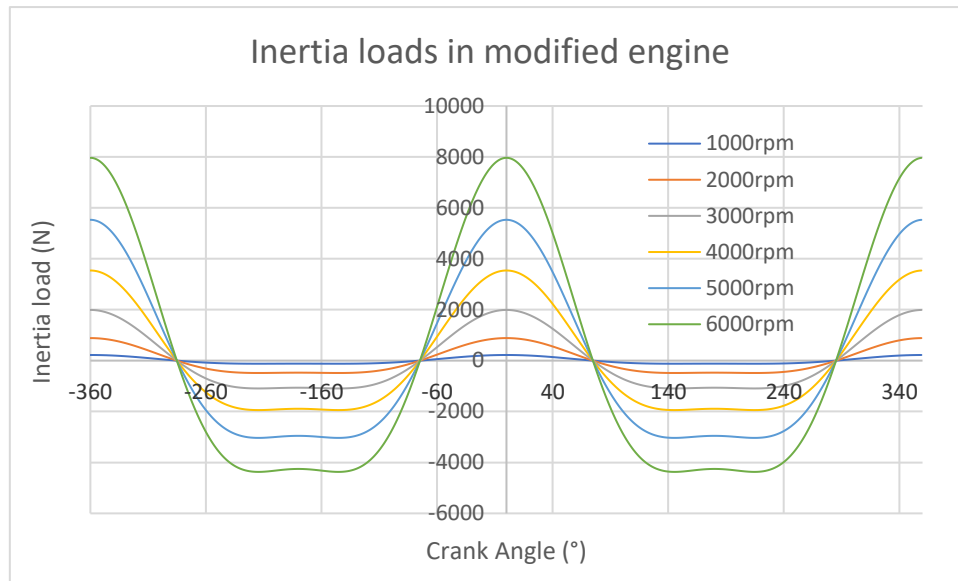


Figure 10 - Total inertia load at each RPM for modified engine

Figure 9 and figure 10 show the inertia loads in the production engine and in the modified engine. The higher order balance has been disregarded. The difference in values at 1000RPM and 6000RPM are very large since inertia force increases with the square of RPM.

6 Highest Net Compressive Force

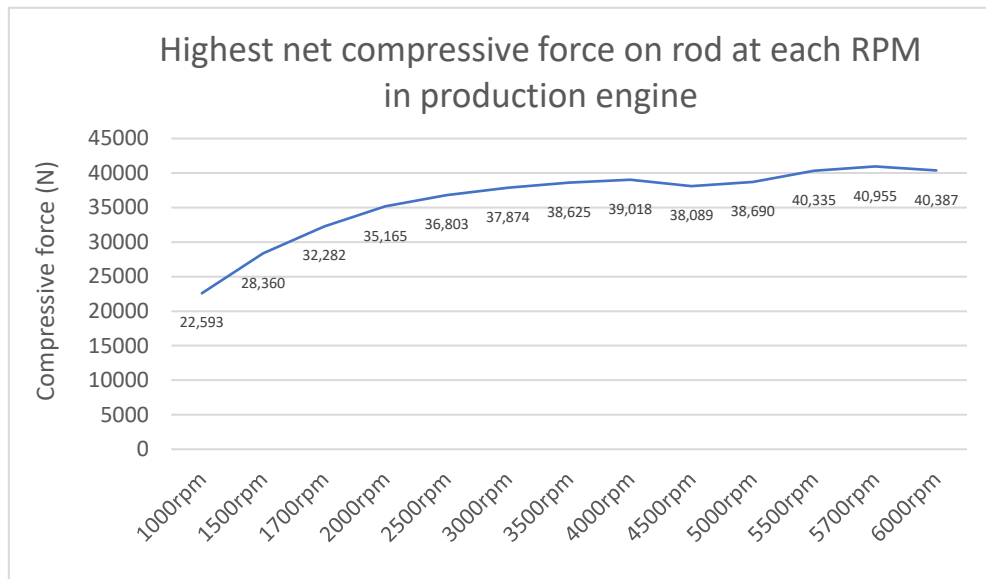


Figure 11 - Highest net compressive force on con-rod at each RPM for production engine

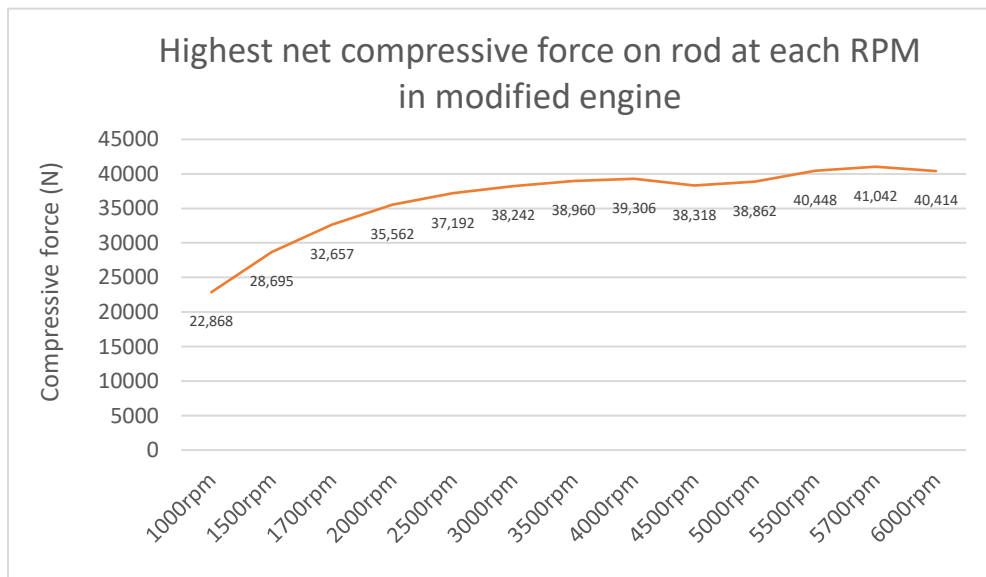


Figure 12 - Highest net compressive force on con-rod at each RPM for modified engine

Figure 11 and figure 12 show that the net compressive force on the con-rod increases with RPM. Maximum net compressive force occurs at 5700RPM for both production and modified engine. Highest net compressive force occurs at 28° at 1000RPM, 25° at 3000RPM and 19° at 6000RPM for both production and modified engine.

7 Highest Net Tensile Load

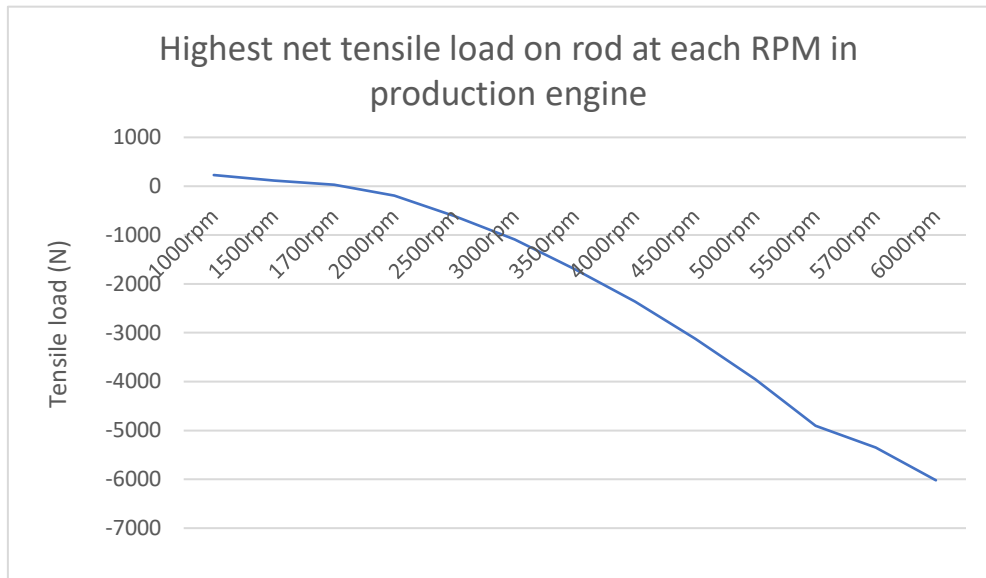


Figure 13 - Highest net tensile load on con-rod at each RPM for production engine

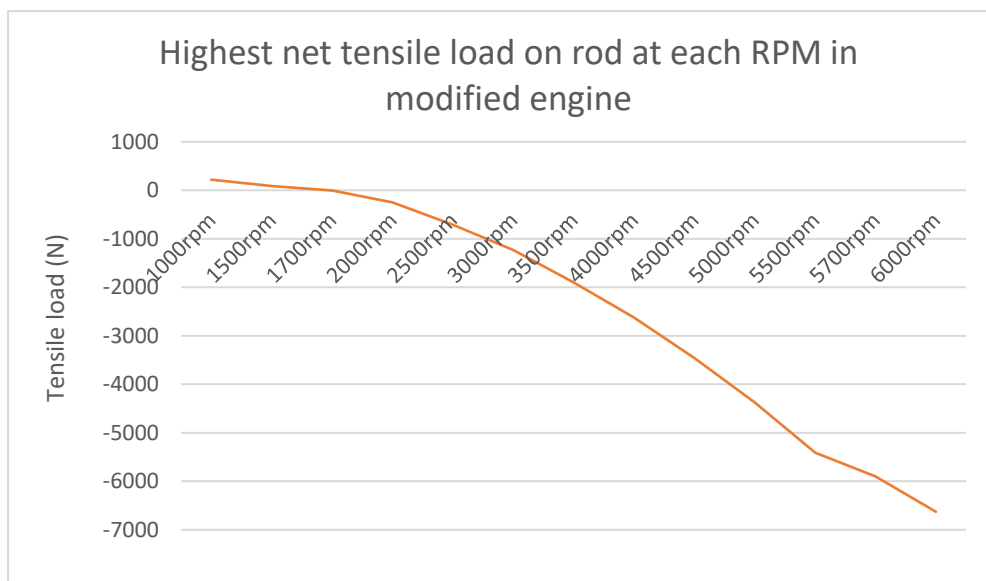


Figure 14 - Highest net tensile load on con-rod at each RPM for modified engine

The values for tensile load are positive up to 1700RPM for the production engine and negative from 2000RPM to 6000RPM (see *figure 13*).

Tensile loads are positive up to 1500RPM for the modified engine and from 1700RPM to 6000RPM (see *figure 14*). The negative values are likely due to tensile load being equal to minimum rod force

and tensile load is acting in an opposing direction to compressive load. Both graphs indicate that tensile load is negligible from 1000RPM to 1700RPM for both engines.

The tensile load for the production engine decreases from -189.6N at 2000RPM to -6020.1N at 6000RPM. The tensile load for the modified engine decreases from -248.3N at 2000RPM to -6627.5N at 6000RPM. These results are as expected since tensile loads are smaller than the compressive loads.

8 Max Compression and Tensile Stresses

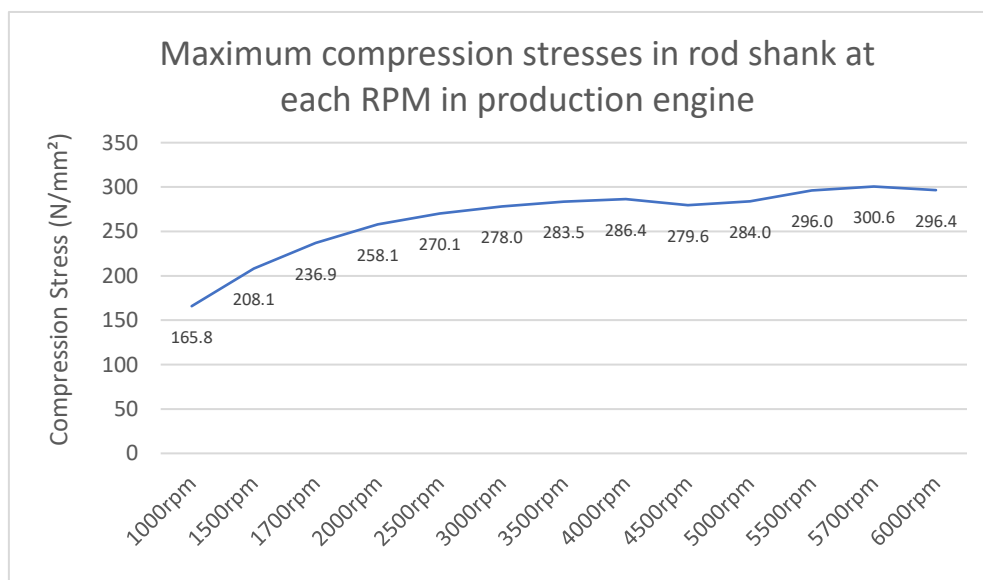


Figure 15 - Maximum compression stresses in rod shank at each RPM in production engine

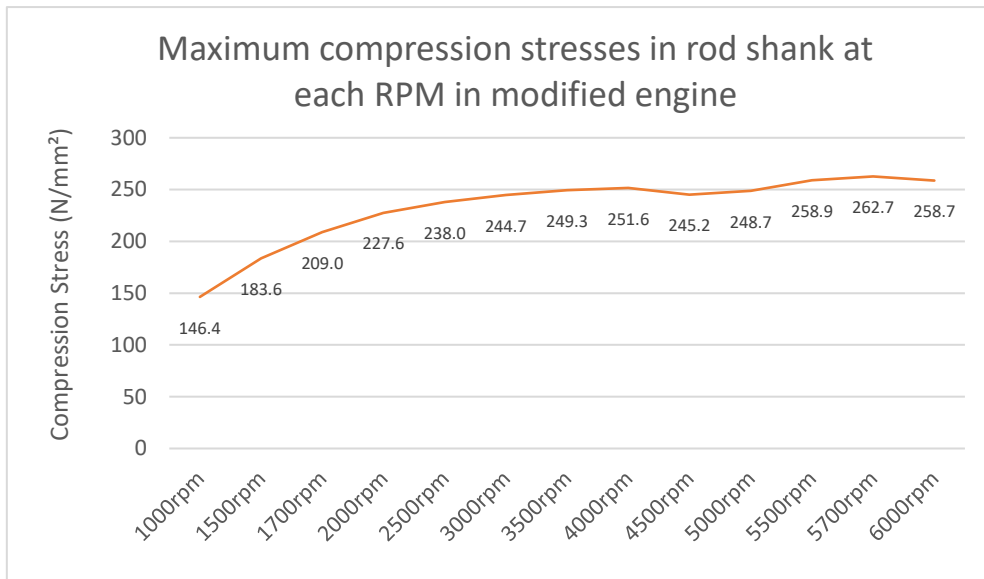


Figure 16 - Maximum compression stresses in rod shank at each RPM in modified engine

Maximum compression increases with RPM for production and modified engine (see *figure 15* and *figure 16*). It follows the same trend as maximum compression force. The values for compression stresses are smaller for the modified engine than the production engine, as the con-rod cross sectional area is larger for the modified engine. Interestingly, at 4500RPM both the production and modified engine compressive stress is less than at 4000RPM.

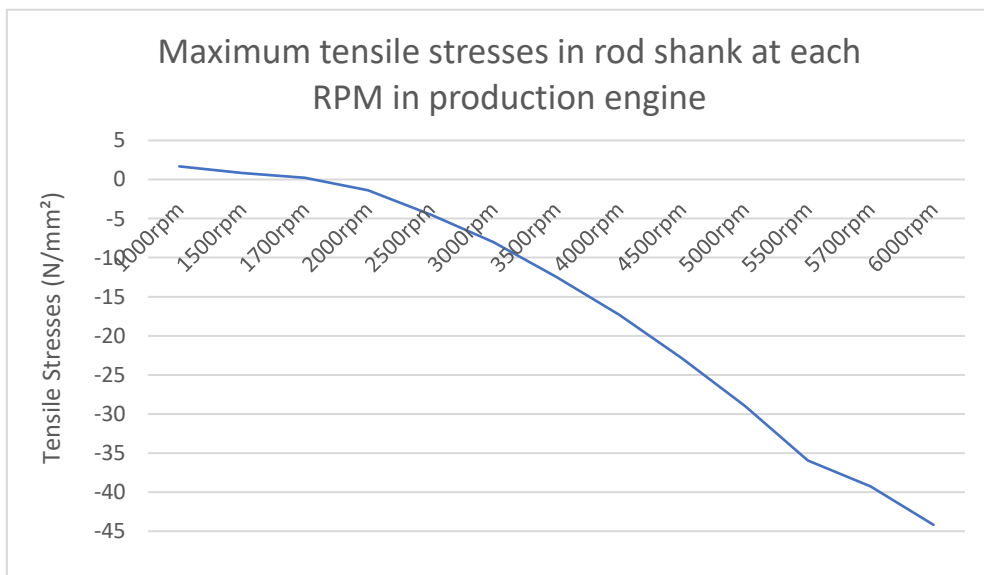


Figure 17 - Maximum tensile stresses in rod shank at each RPM in production engine

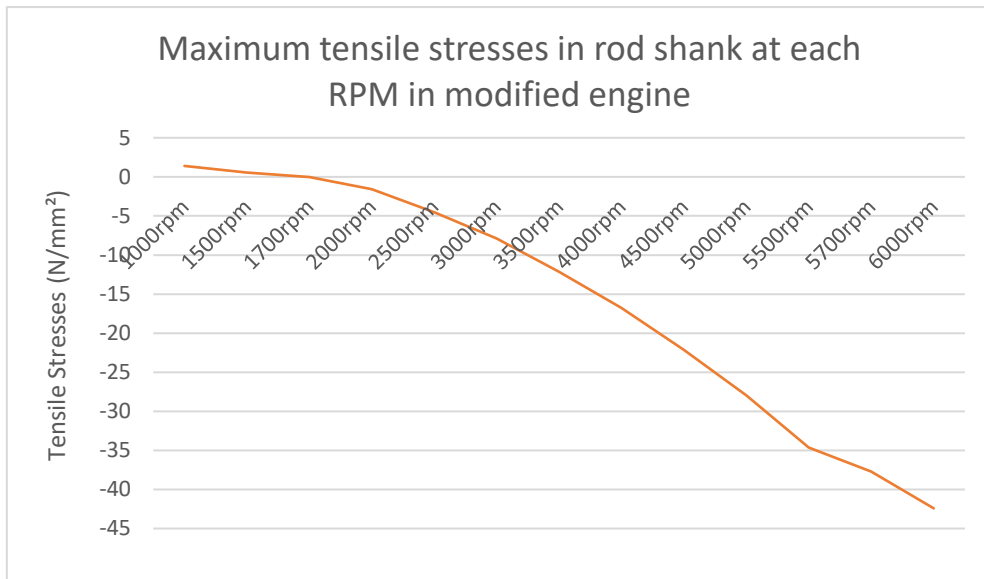


Figure 18 - Maximum tensile stresses in rod shank at each RPM in modified engine

The maximum tensile stresses in the production and modified engine decrease as RPM increases (see *figure 17* and *figure 18*). The maximum tensile stress occurs at 6000RPM for the production and modified engine, but the value is smaller for the modified engine since it has a larger con-rod cross sectional area. The maximum tensile stress is 4.15% larger for the production engine.

9 Nominal Static Safety Factors

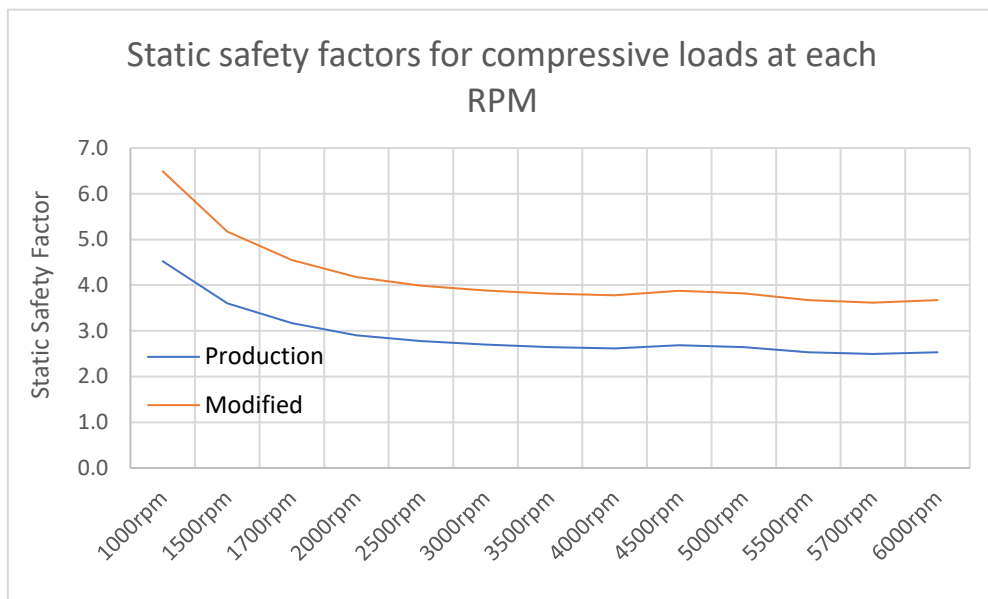


Figure 19 - Static safety factors for compressive loads at each RPM for production and modified engine

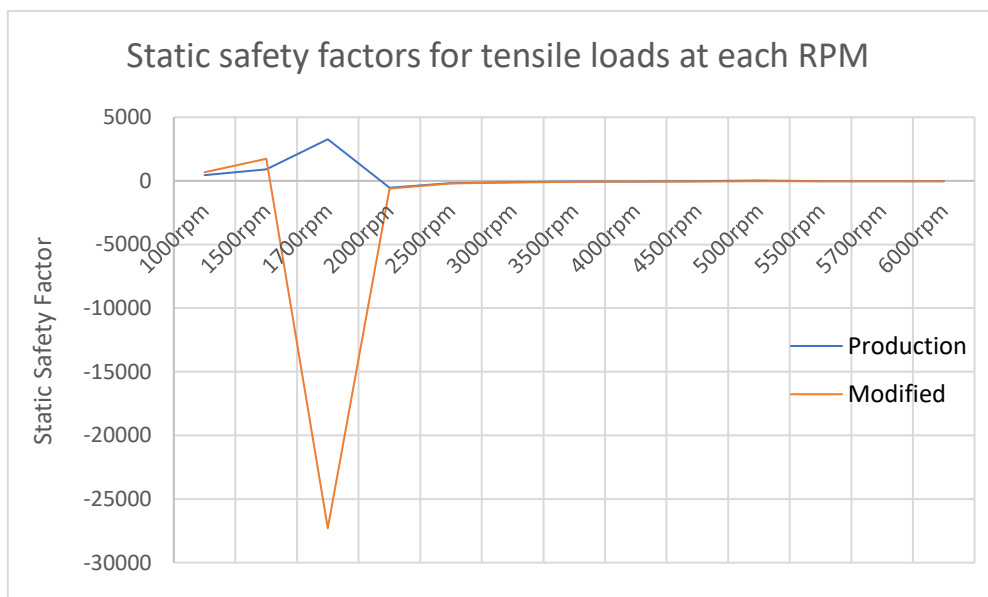


Figure 20 - Static safety factors for tensile loads at each RPM for production and modified engine

The static safety factor for compressive loads decreases for the production and modified engine as RPM increase (see *figure 19*). The modified engine has a larger factor of safety, and therefore its buckling load will be higher and it will be less likely to fatigue. For both engines the safety factor is almost constant from 3500RPM to 6000RPM.

Figure 20 illustrates the static safety factor for tensile load which appears rather constant over the rev range, possibly due to the safety factor value at 1700RPM for both the production and modified engine. These results must be incorrect, as they do not follow any trend.

10 Maximum Compression Distance

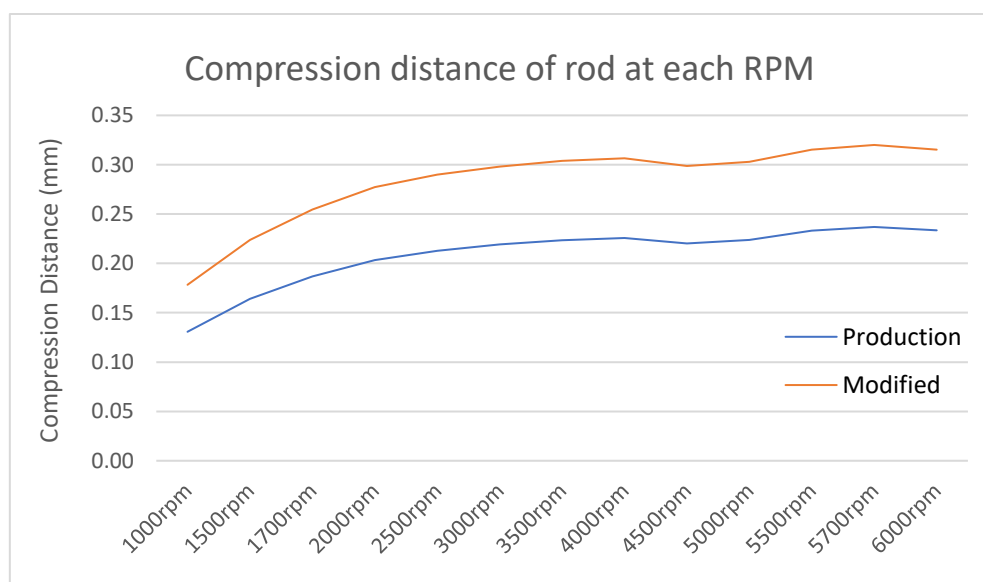


Figure 21 - Maximum compression distance of con-rod at each RPM for production and modified engine

The modified engine has a 6Al 4V titanium conrod as opposed to the powder forged steel rod this results in a higher compression distance than the production engine (see *figure 21*). 6Al 4V Titanium has a smaller Young's Modulus than the powder forged steel, therefore it is more elastic. The maximum compression distance of the titanium rod, 0.32mm occurs at 5700RPM.

11 Maximum Stretch

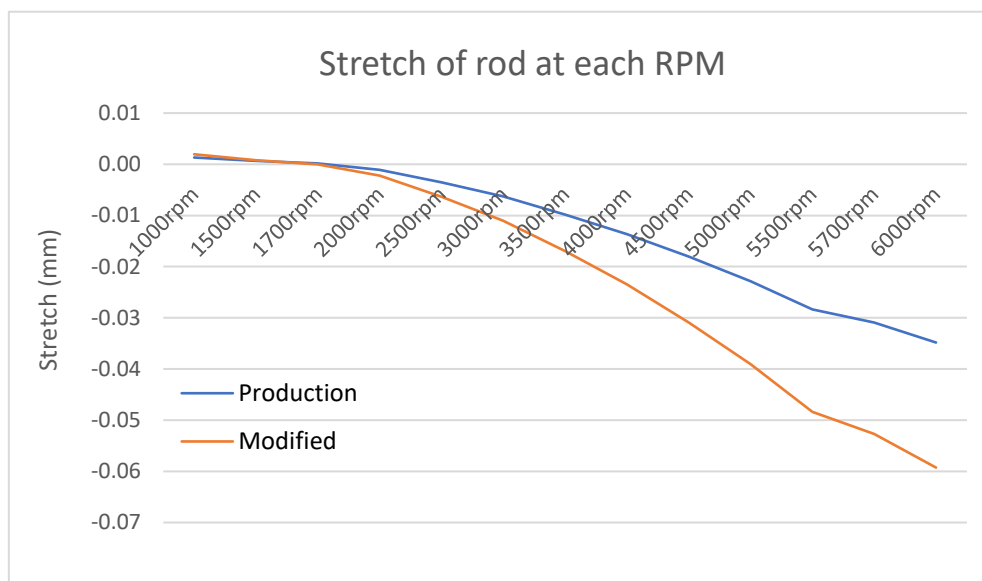


Figure 22 - Maximum stretch of con-rod at each RPM for production and modified engine

According to *figure 22* stretch of the con-rod decreases as RPM increase. The negative values for stretch indicate that their direction is opposite to the compression distance. Therefore, the negative sign can be ignored. The modified engine has a larger stretch with its maximum of 0.059mm occurring at 6000RPM. The production engine has a maximum stretch of 0.034mm at 6000RPM. Since the stretch of the modified engine is 73.4% larger than the production engine stretch a higher compression ratio will be achieved although it will likely be a negligible change. The main concern with piston stretch is the piston may come into contact with the valves, which could lead to engine failure. The deck height of the engine is 203.85mm, therefore a change in 0.059mm shouldn't affect engine operation.

12 Conclusion

To conclude, it is clear from the data collected there are many considerations when designing a con-rod. In order to produce more accurate results, piston pin offset would need be considered, and the displacement of the pistons during operation. In general the results collected are as expected, however static safety factors do not appear to be correct at 1700RPM.

Computational methods can be used to produce better con-rod designs. Technology like Topology optimization can be used in order to minimise the mass of a con-rod design. By minimising the mass of the con-rod quicker piston acceleration can be achieved. By calculating the forces on a con-rod, accurate external loads can be inputted into the topology optimization, therefore making the design as light and stiff as possible. By considering piston pin offset, more accurate results could have been calculated, and therefore a better topology optimization. Von Mises stresses on the piston can be calculated using Finite Element Analysis and can be used to verify if the topology design is feasible.

13 Appendix

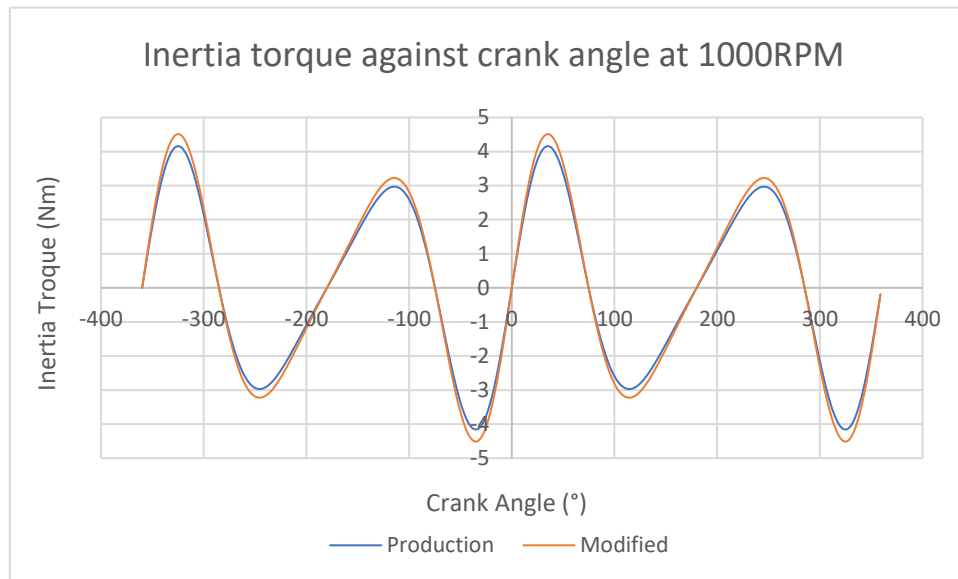


Figure 23 - Inertia torques against crank angle at 1000RPM for production and modified engine

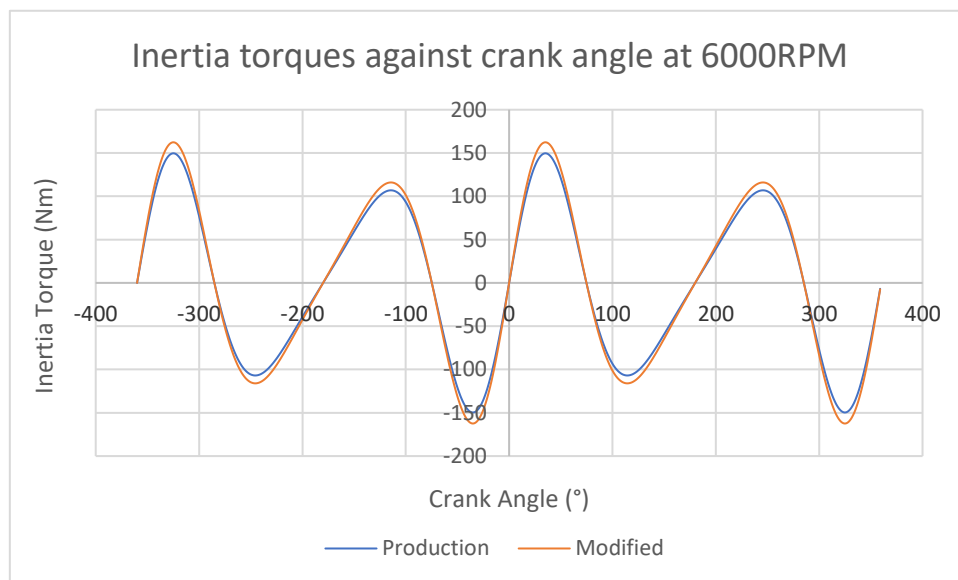


Figure 24 - Inertia torques against crank angle at 6000RPM for production and modified engine

Figure 23 and figure 24 show inertia torque for production and modified engine. Inertia torque is 0 at -360° , -180° , -75° , 0° , 75° , 180° and 360° . To validate the results mean inertia torque per cylinder was calculated at 1000RPM and 6000RPM. The results were 0, as expected.

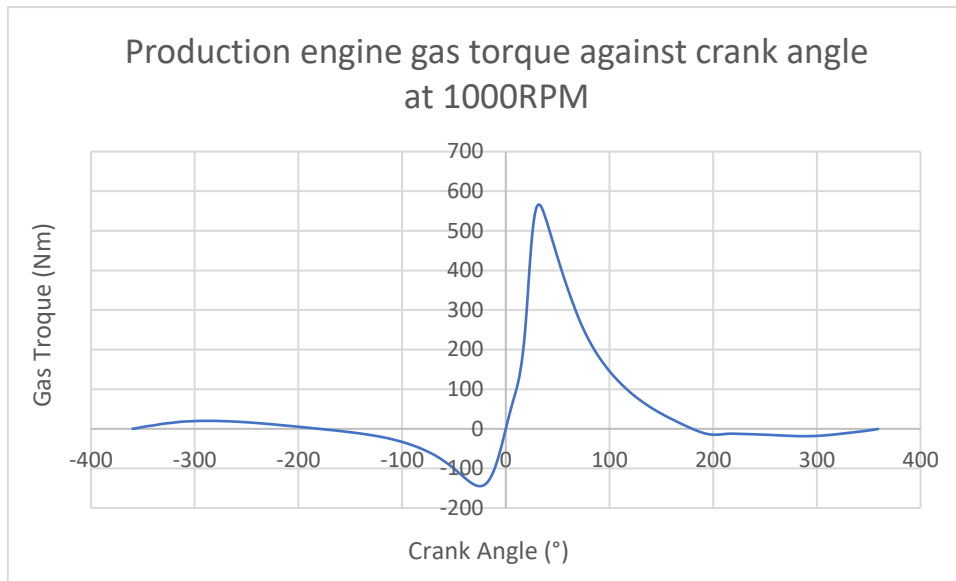


Figure 25 - Gas torque against crank angle at 1000RPM for production engine

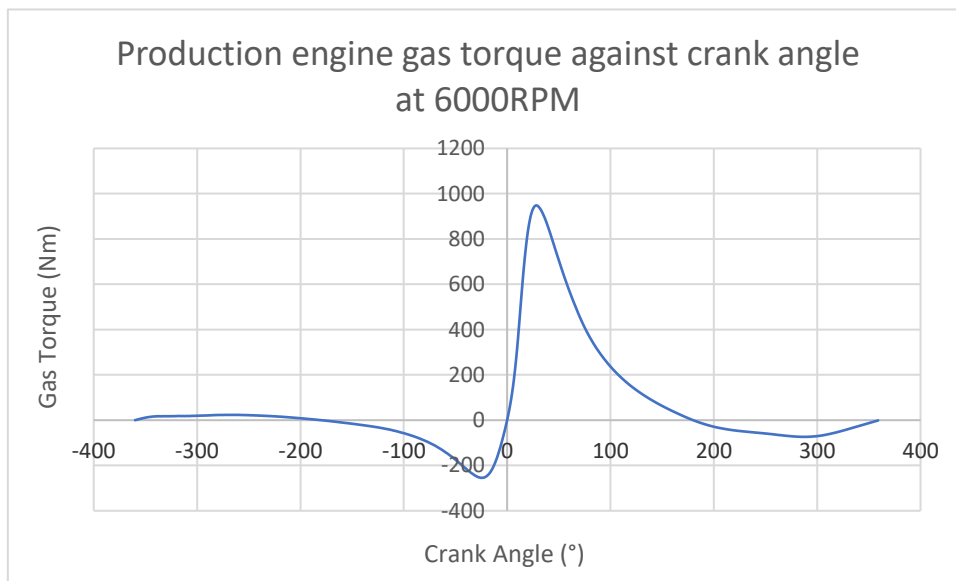


Figure 26 - Gas torque against crank angle at 6000RPM for production engine

Gas torque remains relatively constant for the intake and exhaust stroke at 6000RPM (see figure 26). During the expansion stroke at gas torque decreases to -254.45N at -25°. During the combustion stroke gas torque increases to its maximum value of 948.08N at 28°.

Gas torque at 1000RPM follows the same trend (see figure 25). Maximum gas torque is 566.22N at 32°.

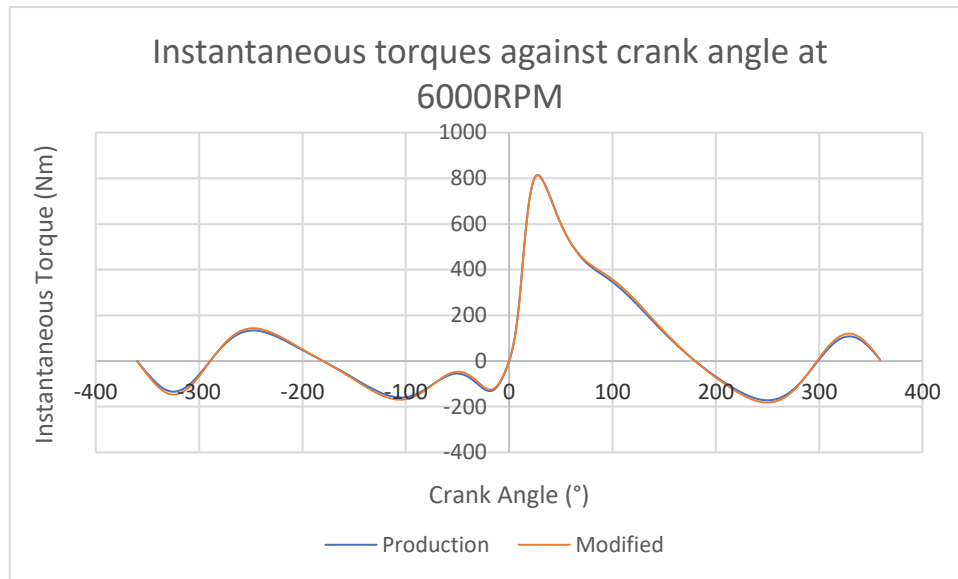


Figure 27 - Instantaneous torques against crank angle at 6000RPM for production and modified engine

Instantaneous torques can be used as a diagnostics tool and can be used for advanced controller designs (see figure 27). It can be used to highlight cyclic variations.

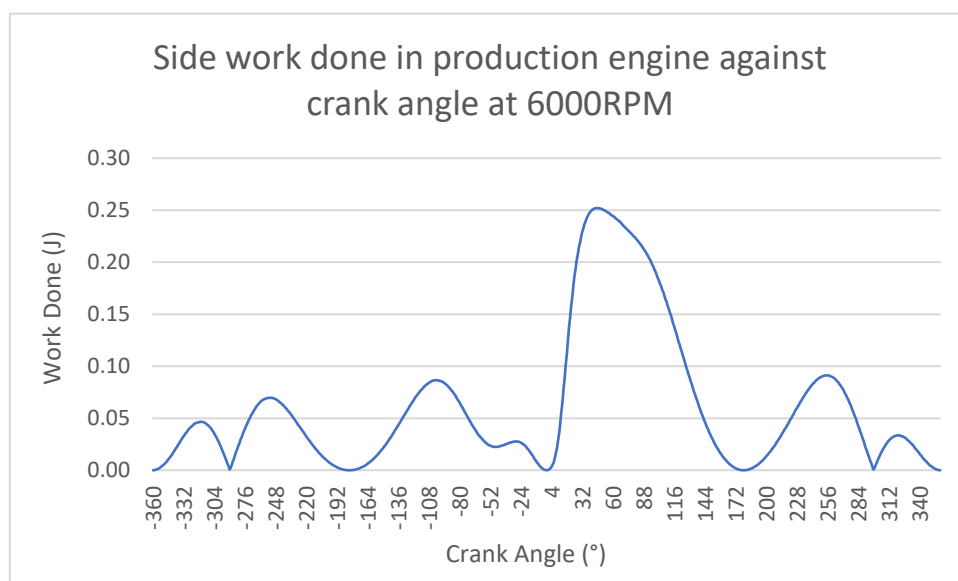


Figure 28 – Side work done against crank angle at 6000RPM for production engine

The maximum value for side work done occurs at crank angle 45° (see figure 28), this is when the rod angle is at its maximum. The piston pin offset allows the pressure to rotate the piston clockwise before TDC to get it onto the left-hand cylinder wall for the power stroke. When the rod angle is at its maximum the torque trying to rotate the piston will be at its maximum.

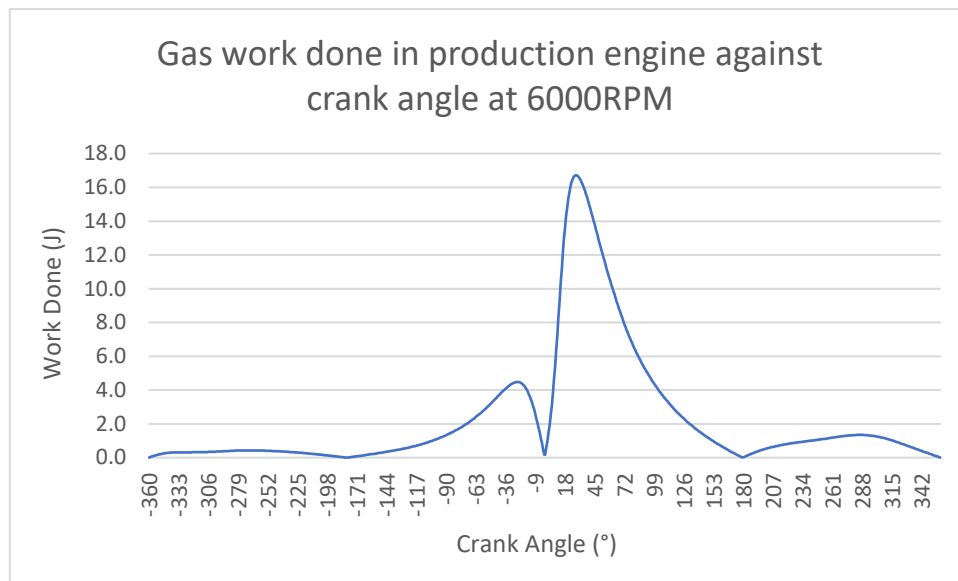


Figure 29 - Gas work done against crank angle at 6000RPM for production engine

Figure 29 shows how the gas work done is mainly during the compression and power stroke. This is because the combustion chamber has a fixed volume for these two strokes. Maximum gas work done is during the power stroke.

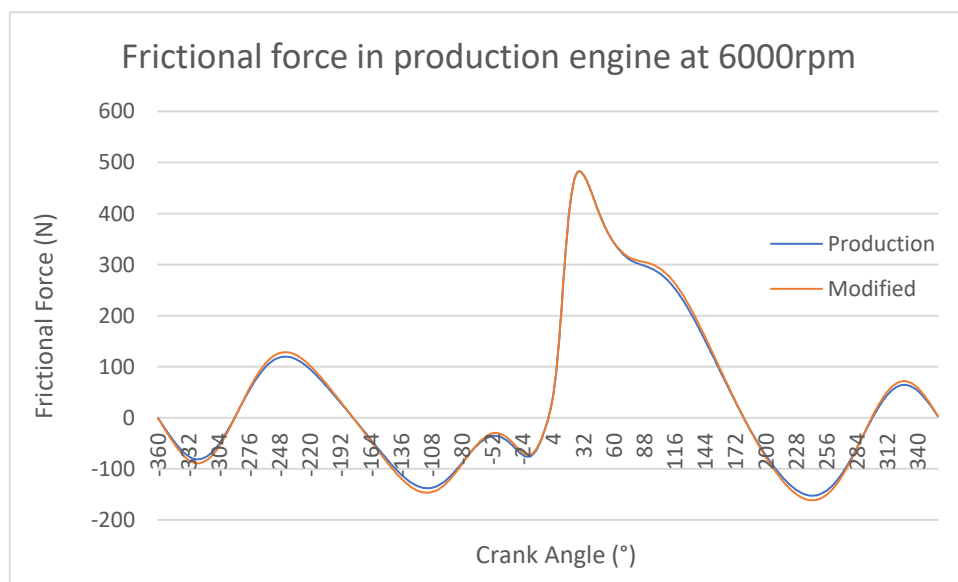


Figure 30 - Frictional force against crank angle at 6000RPM in production engine

Figure 30 indicates that the frictional force is most prominent in the power stroke.

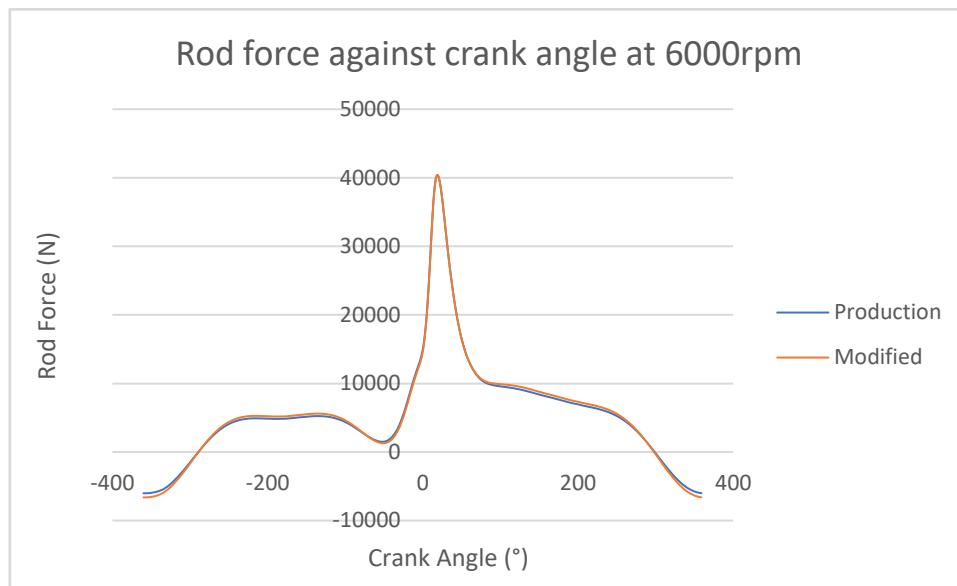


Figure 31 - Rod force against crank angle at 6000RPM for production and modified engine

Figure 31 shows how peak rod force occurs at crank angle 19°, one degree after peak gas force.

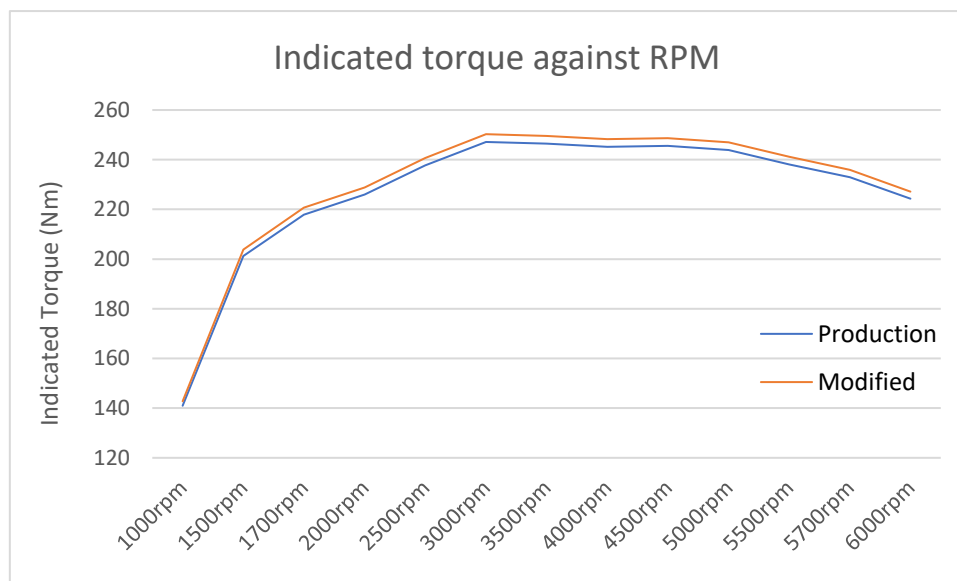


Figure 32 - Indicated torque against RPM for production and modified engine

The stated torque specification for the production engine is 240Nm from 1600-5000RPM. The results show how these values correlate with the data collected in *figure 32*. The modified engine has slightly higher torque across the rev range.

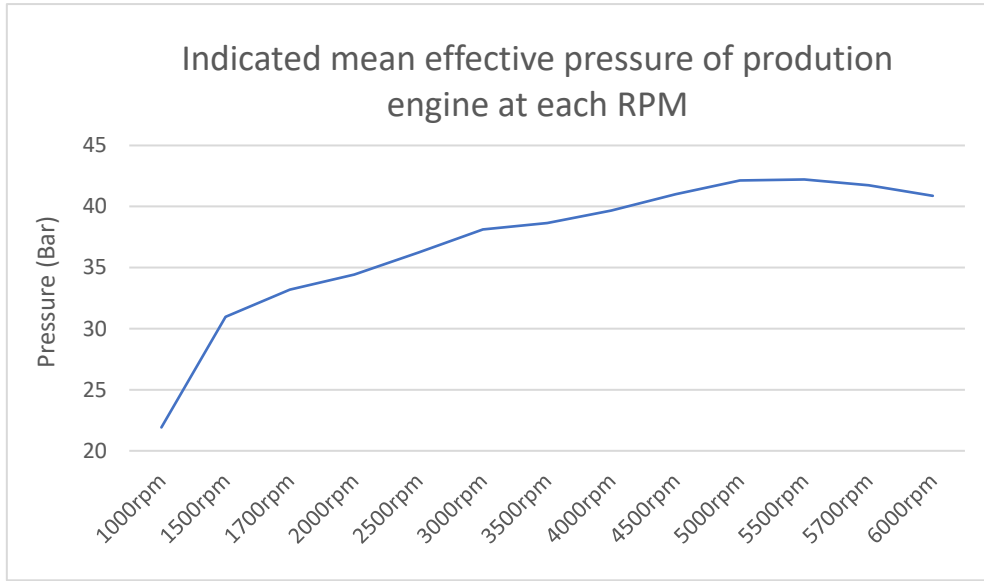


Figure 33 - Indicated mean effective pressure against RPM for production engine

Indicated Mean Effective Pressure (IMEP) was calculated by dividing gas work done with cylinder displacement.

$$\Delta WD_{Gas} = \left| \frac{(F_{G1} + F_{G2})}{2} \right| \times |S_2 - S_1|$$

$$IMEP = \frac{\Delta WD_{Gas}}{cyl. displacement}$$

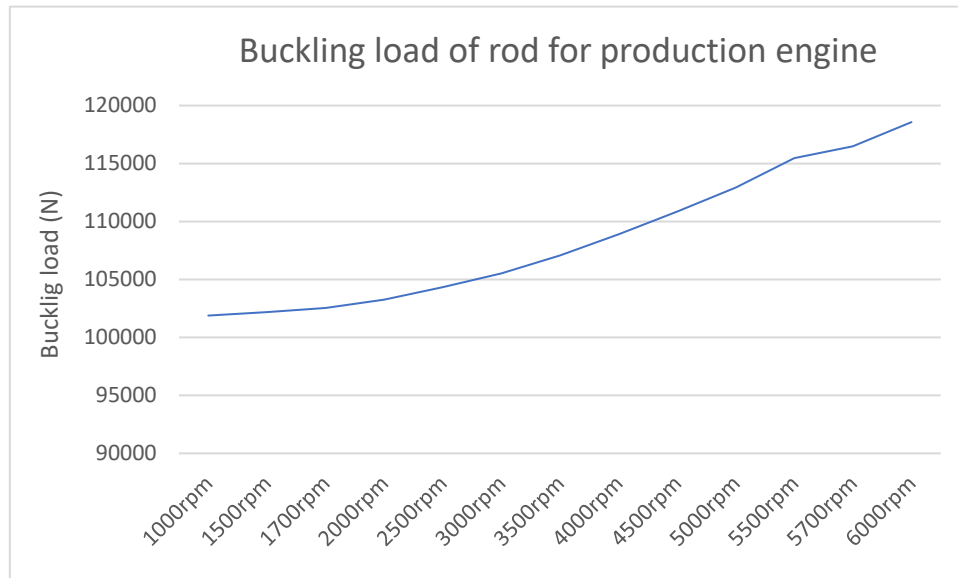


Figure 34 -Buckling load of con-rod against RPM for production engine

Figure 34 shows how the buckling load increases as the RPM increases. The equation used was:

$$\text{Buckling load} = \text{Maximum gas force} \times \text{Factor of safety} \quad (\text{Hussin, et al., 2014})$$

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