



University of Wales Trinity Saint David

School of Engineering

Con-Rod Accelerations, Forces &
Displacements

Kacper Oskar Kaczmarczyk

1701199

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Advanced Engine Design M6X00410

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1 Introduction

This paper is about investigating the connecting rod performance from Ford 1.6l GTDI engine by utilising supplied cylinder pressure data and measuring the engine components. The same method is then used to investigate con-rod performance under higher cyl. pressure, inertial loads and material choices, as well as the connecting rod alterations required to meet the same safety standards as in the Ford production engine.

2 Modified Engine Specifications

For the purpose of this investigation, author has made-up the following base specifications for the M.E.:

- 130 Bar peak cylinder pressure @ 7,000rpm
- Redline @ 10,000rpm, with 95 Bar cylinder pressure

These were primarily chosen to enable the author to study the difference of magnitude of the stresses that cylinder pressure and inertia cause.

To calculate the peak pressure curve for the M.E., author has created 3rd order polynomial, Equation 1:

$$y = -0.0000000002x^3 + 0.0000007x^2 + 0.02x + 25$$

Equation 1: Peak Pressure Curve for Modified Engine

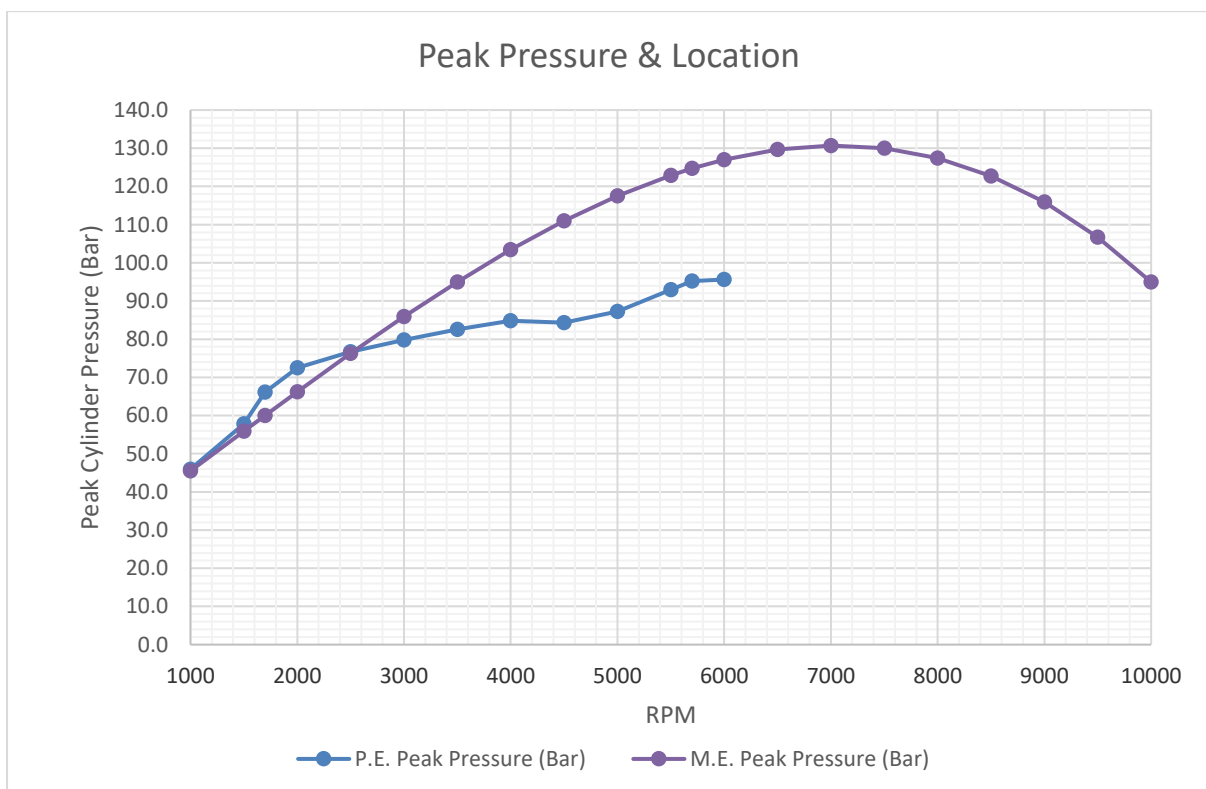


Figure 1: Peak Pressure & Location Graph

Figure 1 shows the peak pressure comparison between the Ford 1.6l GTDI and the M.E. The peak pressure after the 7,000rpm mark has been deliberately decreased to increase the emphasis on the difference between the effective magnitudes of gas and inertial forces.

To make sure that the calculated peak pressure curve seen in Figure 1 is plausible, author has decided to use the relation between peak pressure and gross indicated mean effective pressure (GIMEP) from the Ford 1.6l GTDI, which has been calculated using Equation 2, to come up with an empirical equation: Equation 3 , as seen in Figure 2.

$$GIMEP = \sum_{-180^{\circ}}^{180^{\circ}} PdV$$

Equation 2: Gross Indicated Mean Effective Pressure in terms of Work Done

The calculated work done can be observed in Appendix 3.

$$GIMEP_{pp \text{ Ford 1.6 GTDI}} = 0.2494x + 0.9147$$

Equation 3: GIMEP & peak pressure relation for Ford 1.6l GTDI

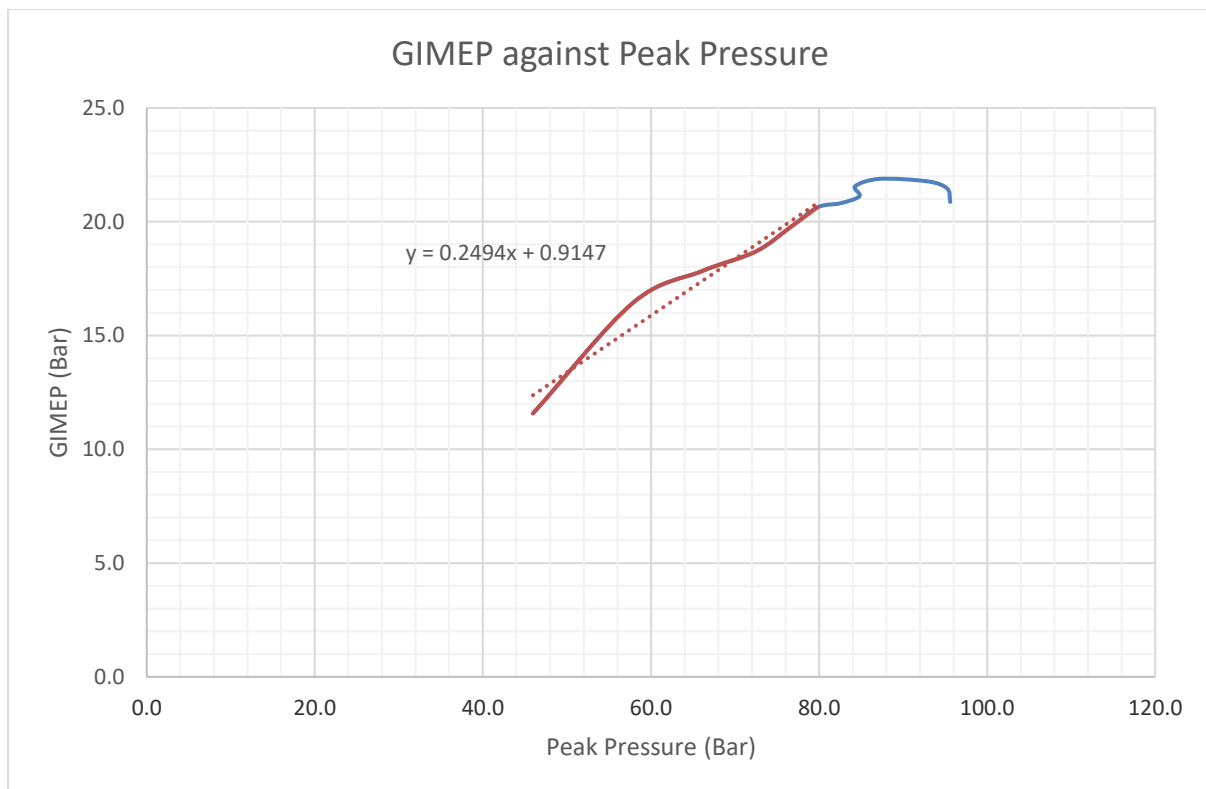


Figure 2: GIMEP against Peak Pressure Graph

Author has decided to not take into consideration peak pressure above 80Bar due to poor convergence.

The GIMEP has then been used to calculate the indicated power and torque of the M.E., by using Equation 4 and Equation 5, as seen in Figure 3.

$$\dot{W}_i = GIMEP * n * V_{sv} * \frac{rpm}{120}$$

Equation 4: Indicated Power in terms of GIMEP (Blair, 1999)

$$\dot{W}_i = \frac{\pi Z_b rpm}{30}$$

Equation 5: Indicated Power in terms of Indicated Torque (Blair, 1999)

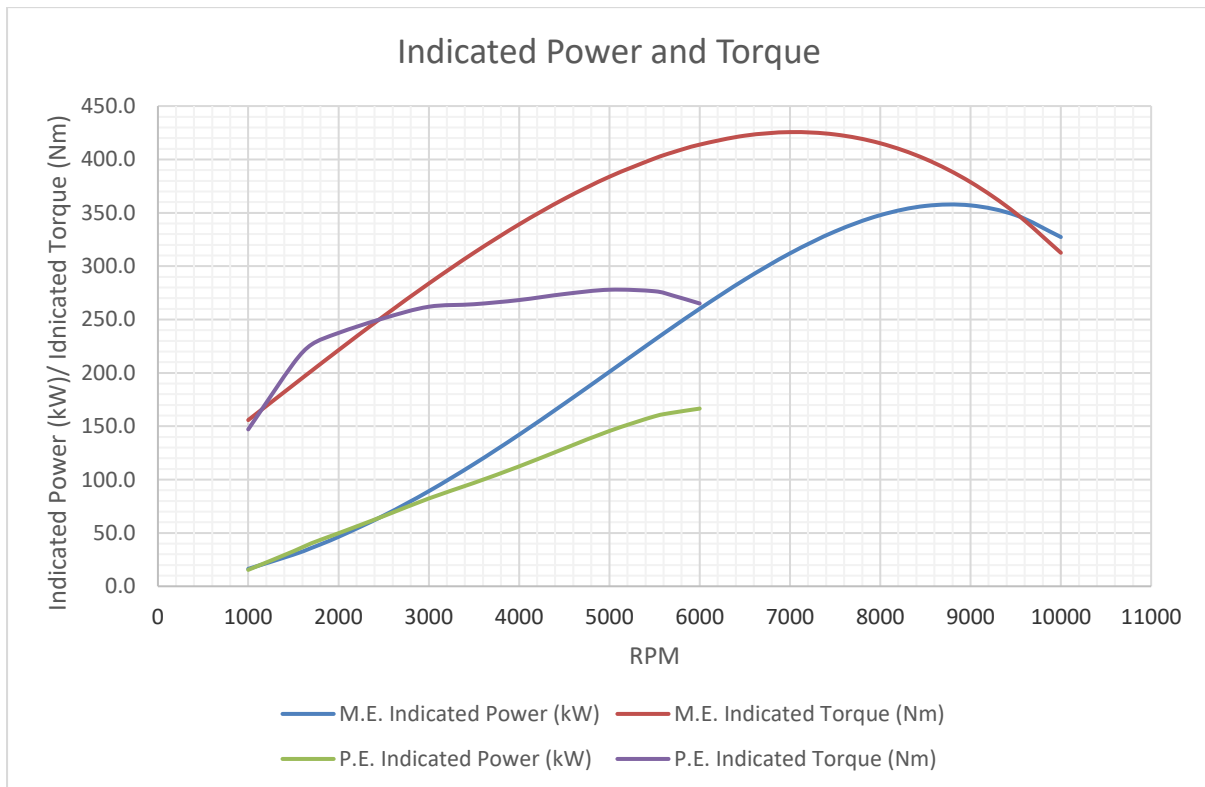


Figure 3: Indicated Power & Torque Graph

Figure 3 shows that the power curve of the M.E. is relatively typical for the use in motorsport. However, since frictional (FMEP) and pumping (PMEP) losses haven't been accounted for, the power and torque at the crank (brake) would decrease as rpm increase, author believes that accounting for losses isn't required as the indicated power and torque show the engine performance characteristics well enough for the purposes of this investigation.

3 Gas Forces

3.1 Production Engine

The supplied cylinder pressures can be observed in Figure 4. It shows that the highest peak pressures as well as pressure sum (area under the curve) occur at 5,700rpm and 6,000rpm. Because of this, as well as the fact that inertial forces will be the greatest due to the highest engine speed, author expects the highest stresses to occur at 5,700rpm and 6,000rpm. Additionally, author believes that the thermal loading on the connecting rod would be the greatest also at the same rpm, due to the directly proportional relation between Pressure and Temperature seen in Equation 6 as well as heat soak.

$$PV = mRT$$

Equation 6: Ideal Gas Law

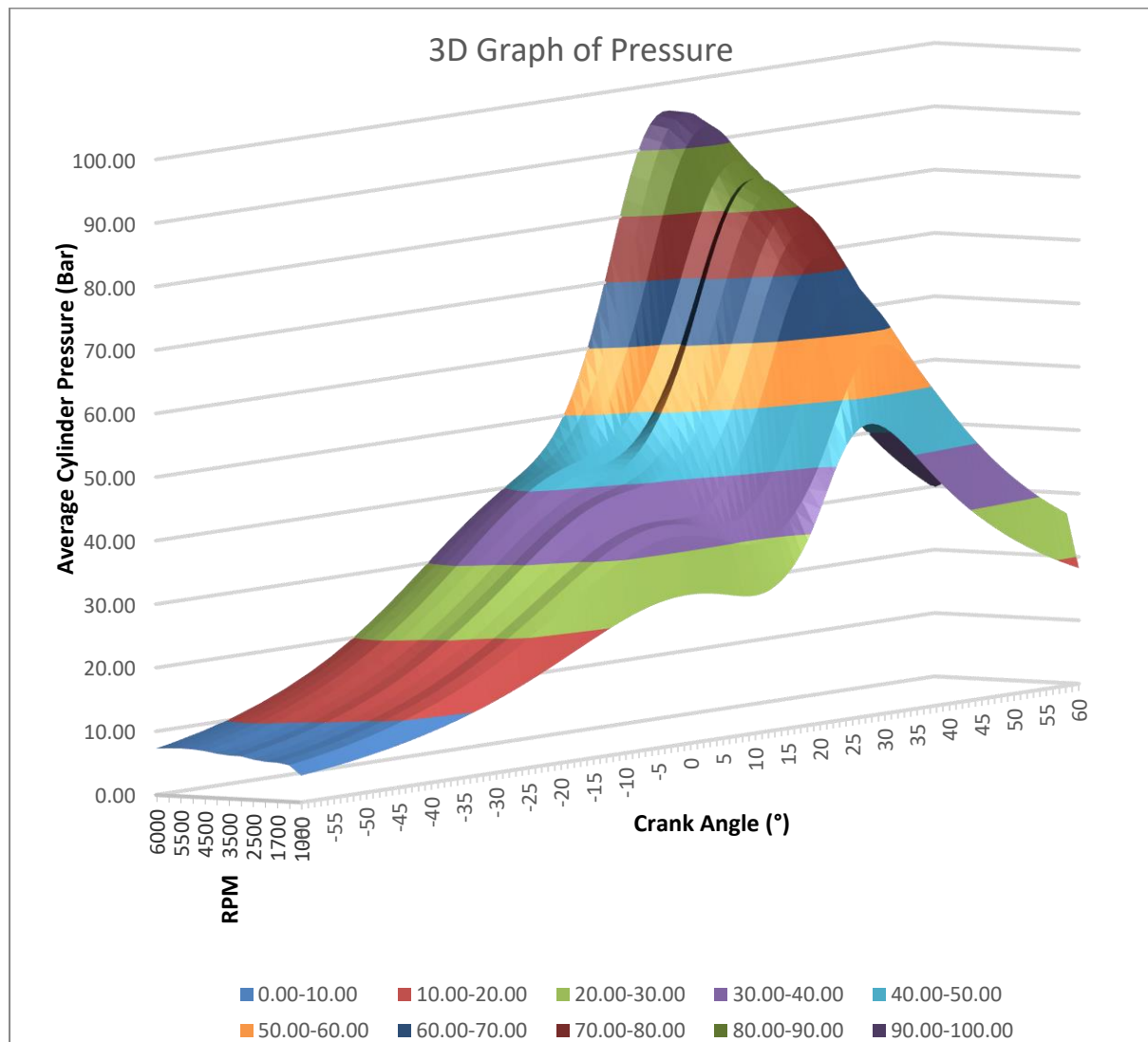


Figure 4: 3D Graph of Pressure

To calculate the gas force, which is applied to the piston crown, the Equation 7 has been used.

$$F_{gas} = P_{cyl.} * \frac{\pi d_{bo}^2}{4}$$

Equation 7: Gas Force (McDonald, 2019-2020)

The results of which can be observed in Appendix 4.

The cylinder pressure is the only variable in Equation 7, as a result the graphs of F_{gas} and cylinder pressure look the same. Due to this, as author expected, the highest gas forces of 46.691kN (4.760t) and 46.863kN (4.787t) occur at 5,700rpm and 6,000rpm respectively. The rest of peak gas forces can be observed in Appendix 1.

3.2 Modified Engine

Due to the established relation between the cylinder pressure and gas force in section 3.1, the gas forces for the M.E. have been directly calculated from the peak cylinder pressures from section 0.

As expected, the highest gas force occurred at 7,000rpm: 64.065kN (6.531t).

The rest of peak gas forces can be observed in Appendix 2.

4 Inertial Loads

To calculate the net inertial force Equation 8 has been used.

$$F_{NI} = m_{rec.} * a_{rec.}$$

Equation 8: Net Inertial Force (McDonald, 2019-2020)

Author has measured the reciprocating masses as in Figure 5; results can be observed in Appendix 1.



Figure 5: Picture of measuring piston mass

To calculate the acceleration of the reciprocating assembly in respect of time, Equation 9 has been used.

$$a_{rec.} = \frac{d^2S}{d\vartheta^2} = R\cos\vartheta + \frac{R}{L}R\cos 2\vartheta$$

Equation 9: Acceleration of Reciprocating Assembly (McDonald, 2019-2020)

Equation 9 can be further simplified into 1st and 2nd order accelerations, the equations for which are separated by “+” in Equation 9. Naturally, to calculate the 1st and 2nd order inertial forces, the corresponding acceleration must be multiplied by $m_{reciprocating}$. The results for which can be seen in

Figure 6. This graph shows that the 1st order inertial force occurs 2 times per engine cycle, whereas the 2nd order inertial forces occurs 4 times per cycle (once per stroke), however, with over 3 times lower magnitude. The 2nd order inertial force is the cause of the ‘flat’ portion of the F_{NI} at and around BDC. It also causes the rate of change of F_{NI} as well as the magnitude at TDC to both increase.

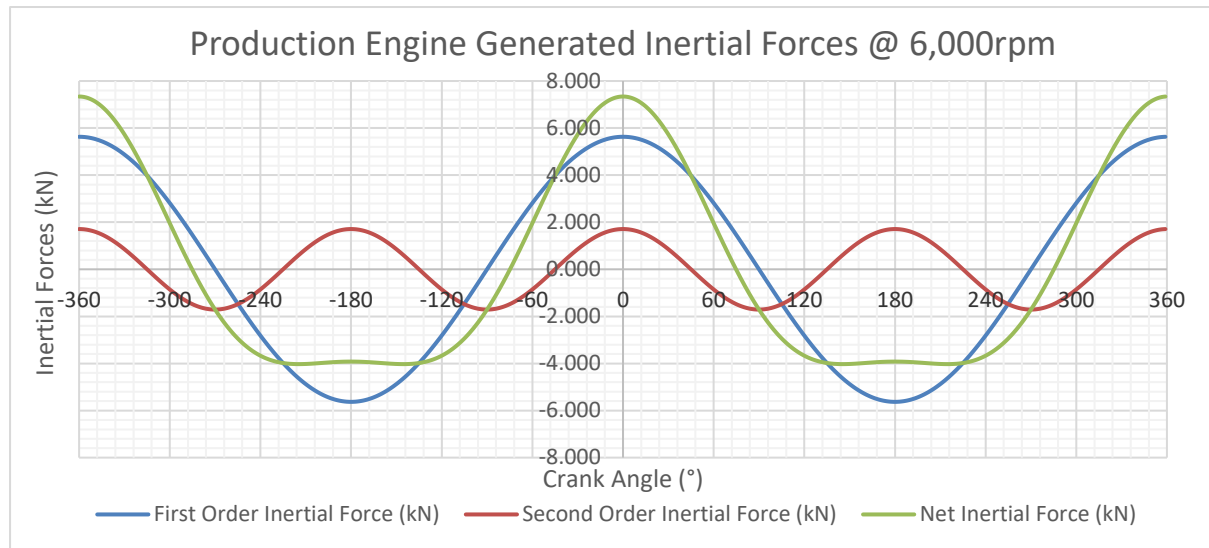


Figure 6: Production Engine Generated Inertial Forces @ 6,000rpm

Author has applied the same method to the M.E. Naturally, the F_{NI} of both engines has stayed the same up until 6,000rpm, due to equal m_{rec} and engine geometry, as Figure 7 shows.

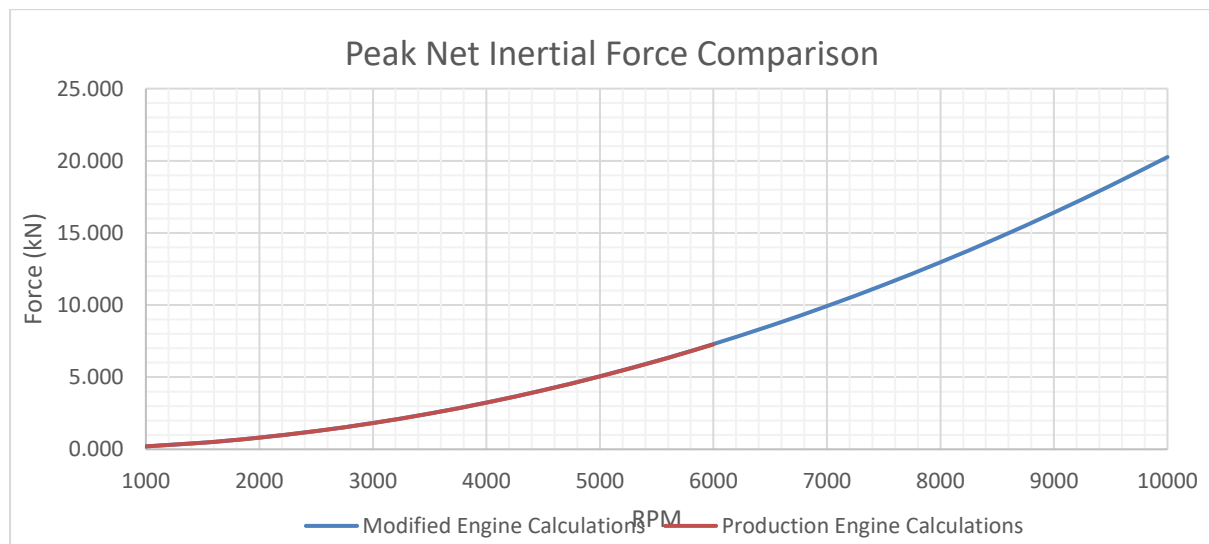


Figure 7: Peak Net Inertial Force Comparison

Due to the squared nature of inertia, as displayed in Figure 7, the F_{NI} @ 5,000rpm and 10,000rpm are 5.065kN and 20.260kN, this means that the force quadruples when rpm doubles. Author expects this to work in favour for the higher performance M.E. as the inertial forces counteract the increased cylinder pressure, as the Appendix 5 shows.

5 Sidewall & Ring Friction

5.1 Method

Author has considered two sources of friction due to the reciprocating assembly:

- Piston Skirt
- Piston Ring

5.1.1 Piston Skirt Force

To calculate piston skirt force Equation 10 has been applied.

$$F_{skirt} = (F_{gas} - F_{NI})\tan\phi$$

Equation 10: Piston Skirt Force (McDonald, 2019-2020)

For M.E. the cyl. pressure and as a result the F_{gas} , author used the peak pressure for compressive calculations. However, for tensile calculations author had to average the cylinder pressure of the Ford 1.6l GTDI in the tensile region. To estimate the pressure above 6,000rpm author has created Figure 8 and used the corresponding equation.

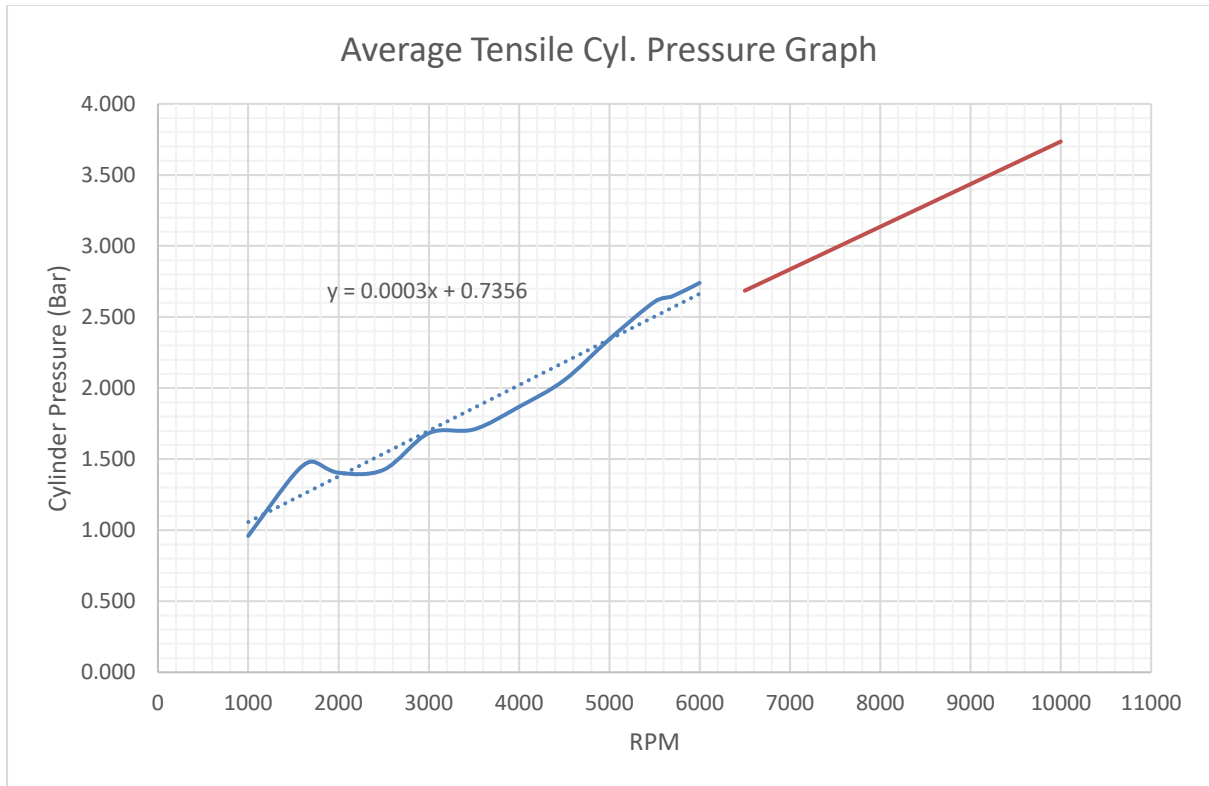


Figure 8: Average Tensile Cylinder Pressure Graph

Author has decided to consider the gauging pin offset to increase the accuracy of Equation 10, this has been achieved with Equation 11.

$$\phi = \sin^{-1} \left(\frac{R}{L} \sin \theta - \frac{x}{L} \right)$$

Equation 11: Conrod Angle Considering Gauging Pin Offset (McDonald, 2019-2020)

The effect of gauging pin offset on the connecting rod angle can be observed in Appendix 6.

Author has measured, as in Figure 9, the gauging pin offset to be +0.5mm towards the exhaust side.



Figure 9: Picture of measuring gauging pin off-set

5.1.2 Piston Ring Force

To calculate piston ring force Equation 12 has been applied.

$$F_{radial} = P\pi Dh$$

Equation 12: Piston Ring Force (McDonald, 2019-2020)

However, D is the measurement when the piston ring makes full contact with the cylinder bore. Therefore, author has measured the piston ring width and took 2 widths away from the cylinder bore as author has assumed full contact. Author is aware that these dimensions will alter due to thermal expansion and the minimisation of the ring gap, however, author believes that such matter is beyond this investigation and accounting for such mechanism would yield a minuscule change.

5.1.3 Frictional Force

To calculate the friction force

$$F_f = \mu(F_{skirt} + F_{radial})$$

Equation 13: Friction Force (McDonald, 2019-2020)

Author has assumed μ to be 0.1, due to (Mohamed Kamal Ahmed Ali, 2017) and (Zhang, 2016).

However, Equation 13 does not have a sign convention relative to the direction of motion, which is why author had to program in the sign convention so that friction always opposes the direction of motion, this can be observed in Figure 10, note that the direction of F_{gas} is considered to be positive.

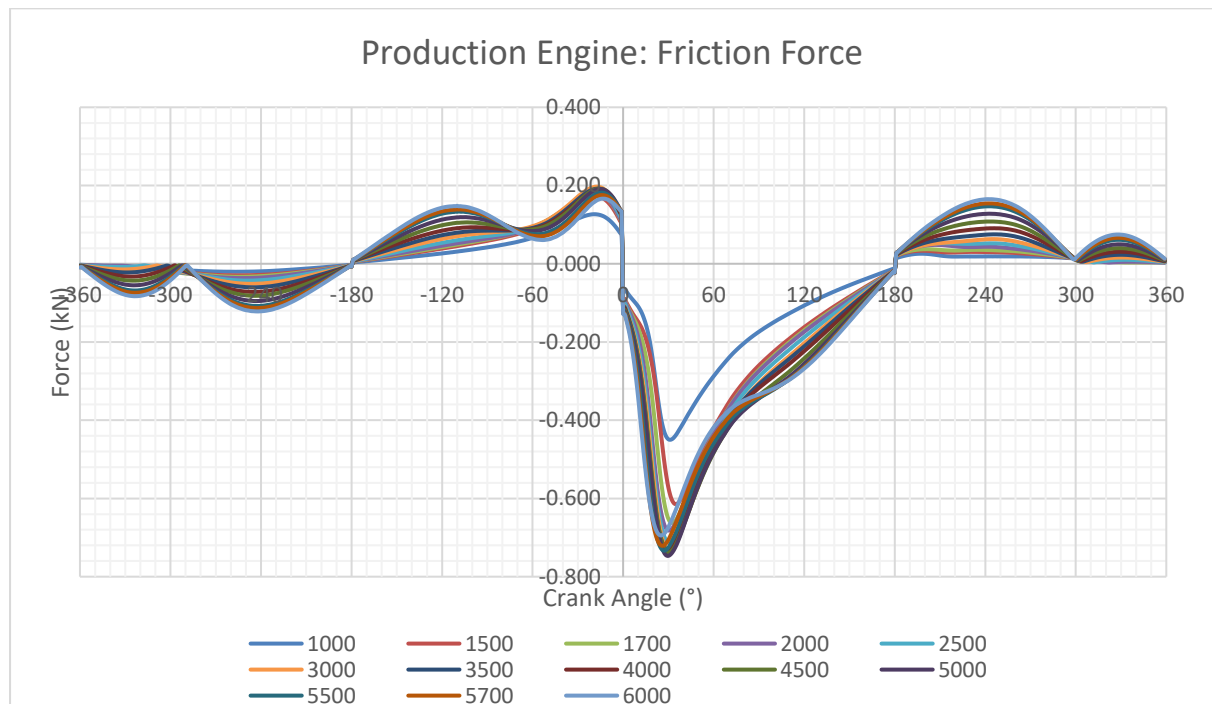


Figure 10: Production Engine Friction Force

The ‘humps’ on the intake and exhaust strokes in Figure 10 are caused by Equation 10; as F_{NI} counteracts F_{gas} , this flips the direction the F_{skirt} is applied, however, the direction of piston motion stays the same, which is why the F_f is negative and positive on intake and exhaust strokes respectively.

5.2 Results

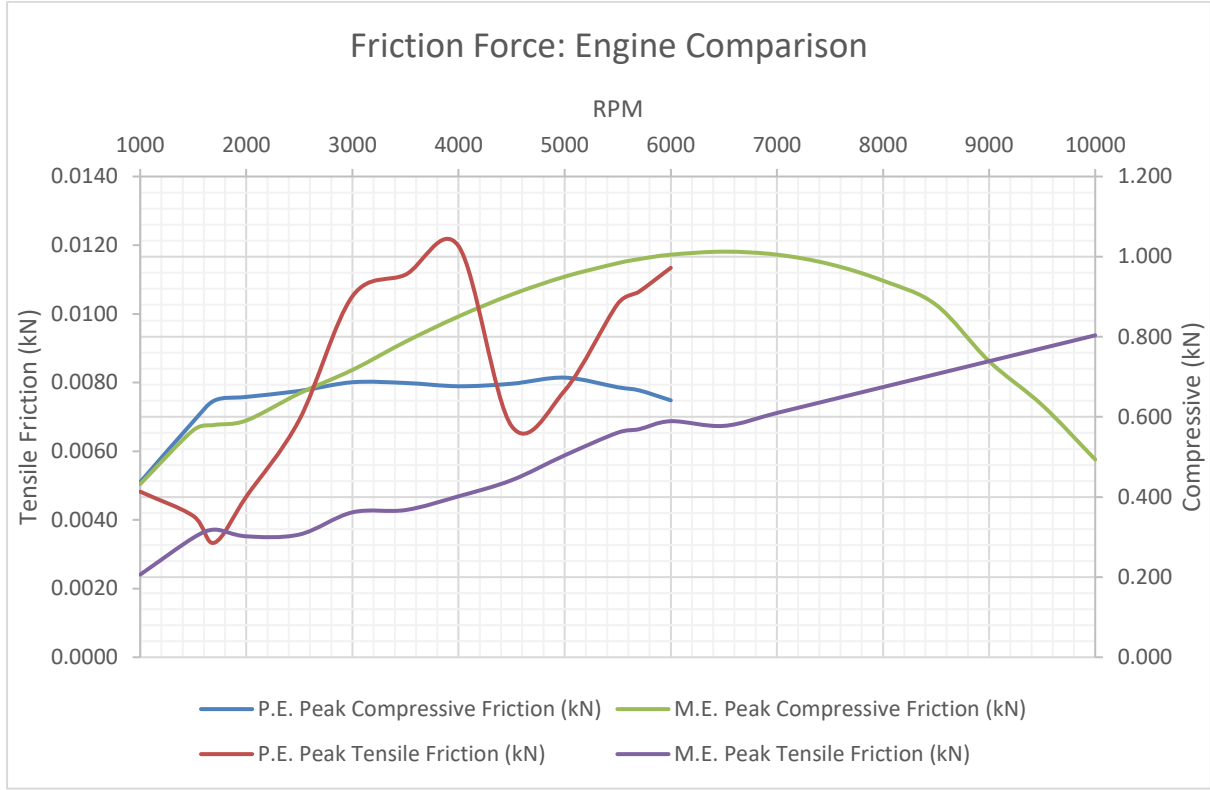


Figure 11: Friction Force Engine Comparison

Figure 11 shows the compressive and tensile friction forces at the crank angle which corresponds to peak compressive and tensile forces from section 0.

As author expected, the compressive frictional forces follow closely the peak pressure profile since pressure is the major variable driving the Equation 13. The squared nature of F_{NI} cause the P.E. compressive F_f to be relatively flat, as oppose to a steady incline as seen in Figure 1, F_{NI} also causes the M.E. compressive F_f to start decreasing at lower rpm and at higher rate than the peak pressure profile seen in Figure 1.

M.E. tensile F_f has relatively steady increase rate compared to other curves, this is due to the estimated pressure (from section 5.1.1) and minuscule F_{NI} , both can be observed in Appendix 2.

Lastly, the reason for the relatively big dispersion of the P.E. tensile F_f is the equal dispersion of the crank angle at which the peak tensile forces occur. Both of which can be observed in Appendix 1.

5.3 Dyno Data Comparison

To find out the % of FMEP comprised of friction due to F_{skirt} and F_{radial} , author has decided to calculate the FMEP of the Ford 1.6l GTDI. To calculate the FMEP due to F_{skirt} and F_{radial} , author has used Equation 14.

$$FMEP_{skirt+radial} = \left| \sum_{-360^{\circ}}^{360^{\circ}} F_F dS \right|$$

Equation 14: Friction Mean Effective Pressure due to skirt and ring force

The component $F_F dS$ has been plotted in Figure 12.

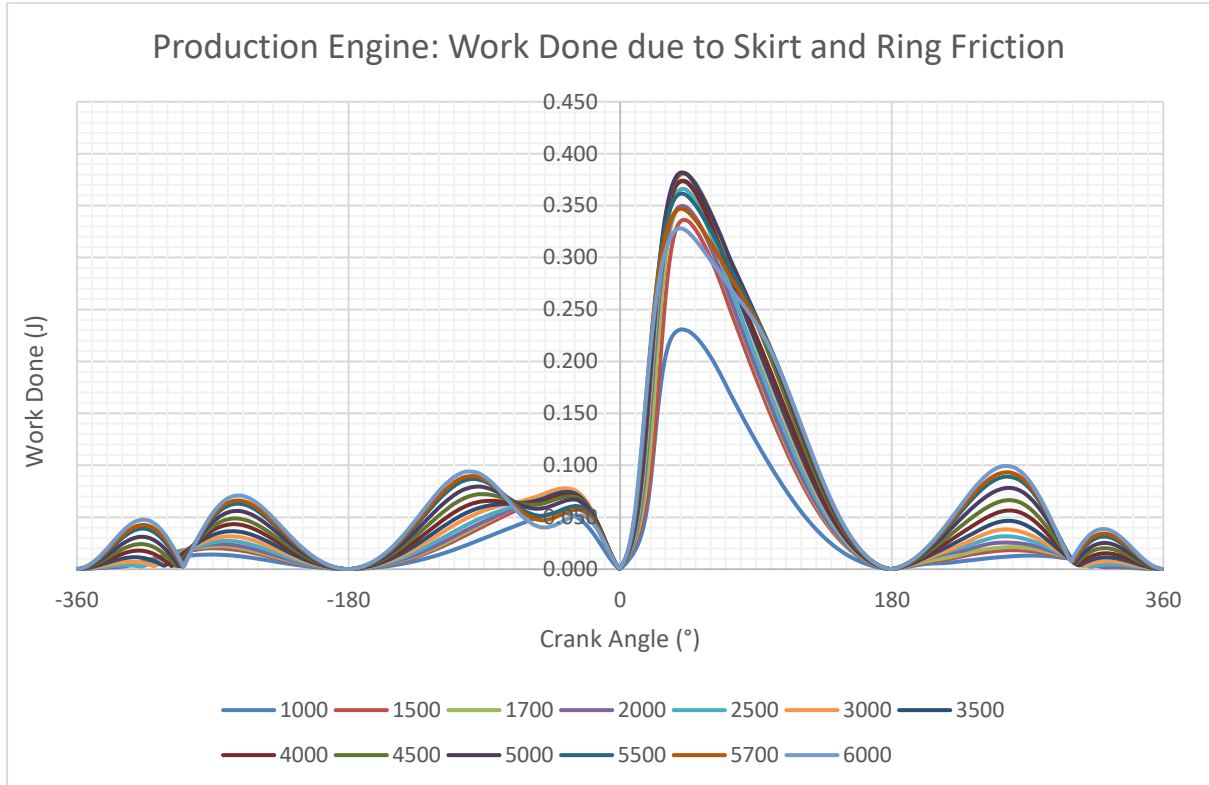


Figure 12: Production Engine Work Done due to Skirt and Ring Friction

Author has made all values in Figure 12 of the same sign since the work done due to friction always counteracts any energy driving the engine (e.g. friction can only take energy out of the system).

Author has then utilised Equation 2 with range altered to intake and exhaust strokes, to calculate PMEP. This has then allowed the author to calculate NIMEP, from which the $FMEP_{skirt+radial}$ has been taken away to calculate the BMEP.

After that author has used the real engine dyno power output (Ford Motor Company, n.d.) to calculate the $BMEP_{real}$ by rearranging Equation 4. Figure 13 shows the comparison between 3 different power and torque types.

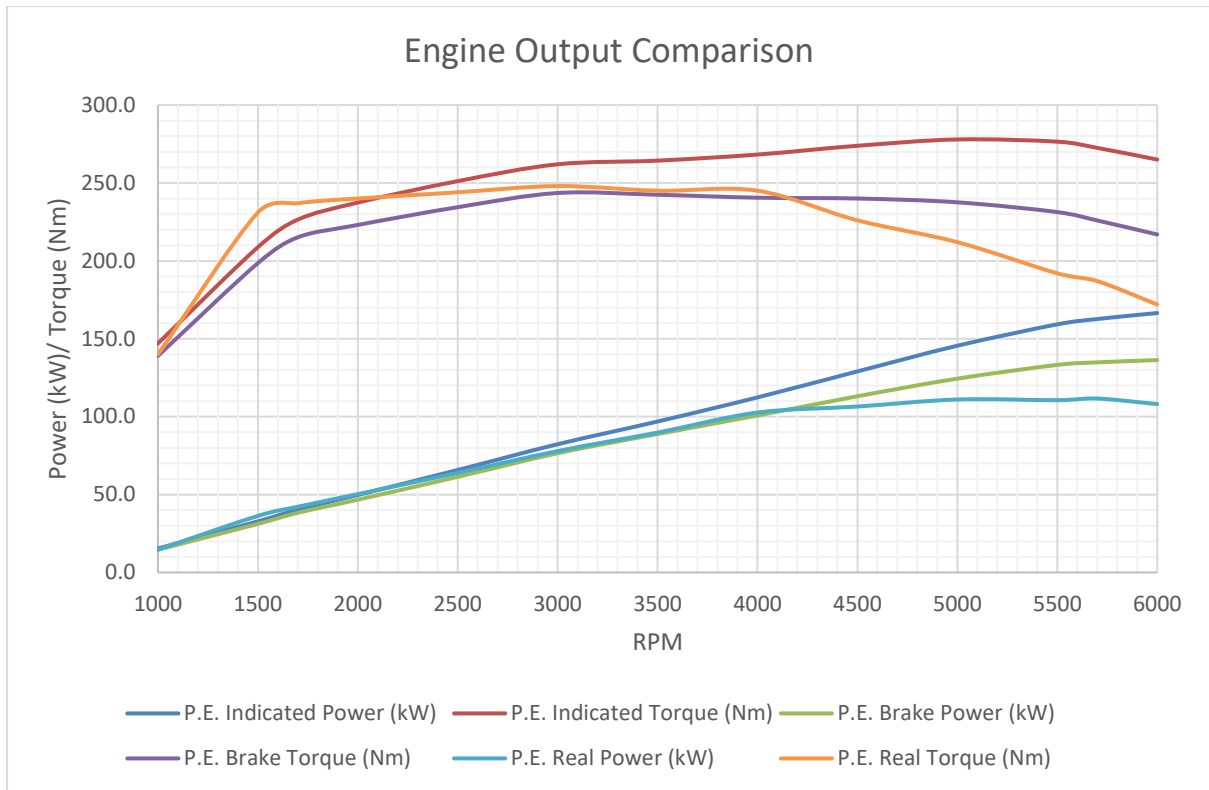


Figure 13: Engine Output Comparison

The Figure 13 suggests that $FMEP_{skirt+radial}$ is in the correct ballpark since the brake torque and real torque between 3,000rpm and 4,000rpm are relatively similar. Above 4,000rpm the difference increases at relatively steady state, this isn't surprising since there are many more sources of friction like valvetrain that haven't been considered in brake torque. However, the lack of steady difference or rate of difference between the brake and real torque suggests that either: calculation of $FMEP_{skirt+radial}$ is incorrect or the given pressure data has been recorded at development state when certain set up parameters like boost were different. This is further emphasised by Figure 14.

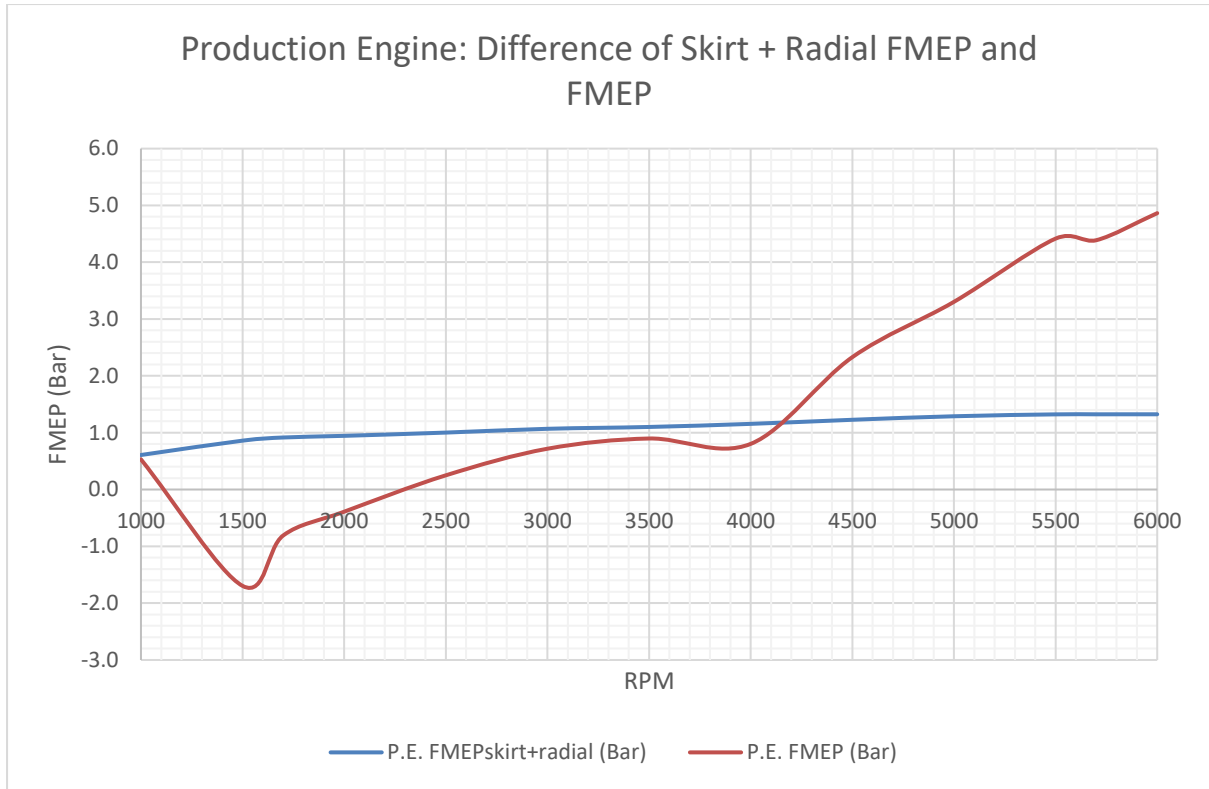


Figure 14: Production Engine Difference of Skirt + Radial FMEP and FMEP

The real FMEP is negative between estimated 1,100rpm and 2,300rpm. This further suggests that the given cyl. pressure data does not match the production engine. Furthermore, $FMEP_{skirt+radial}$ occurs at more or less the middle of the FMEP, further proving the calculation validity.

Author believes that incorporation of other sources of friction like valvetrain or bearings, would cause the rate of calculated FMEP to increase, possibly causing it to match the rate of real FMEP.

However, author believes this to be outside of the scope of this investigation as the method used to calculate the F_f has been proven to be good enough.

All of these results can be found in Appendix 1.

6 Net Forces & Stresses

To calculate the net force acting on the connecting rod author used Equation 15, the results of which can be seen in Figure 15.

$$F_{rod} = \frac{F_{gas} - F_{NI}}{\cos \phi} + F_f$$

Equation 15: Rod Force

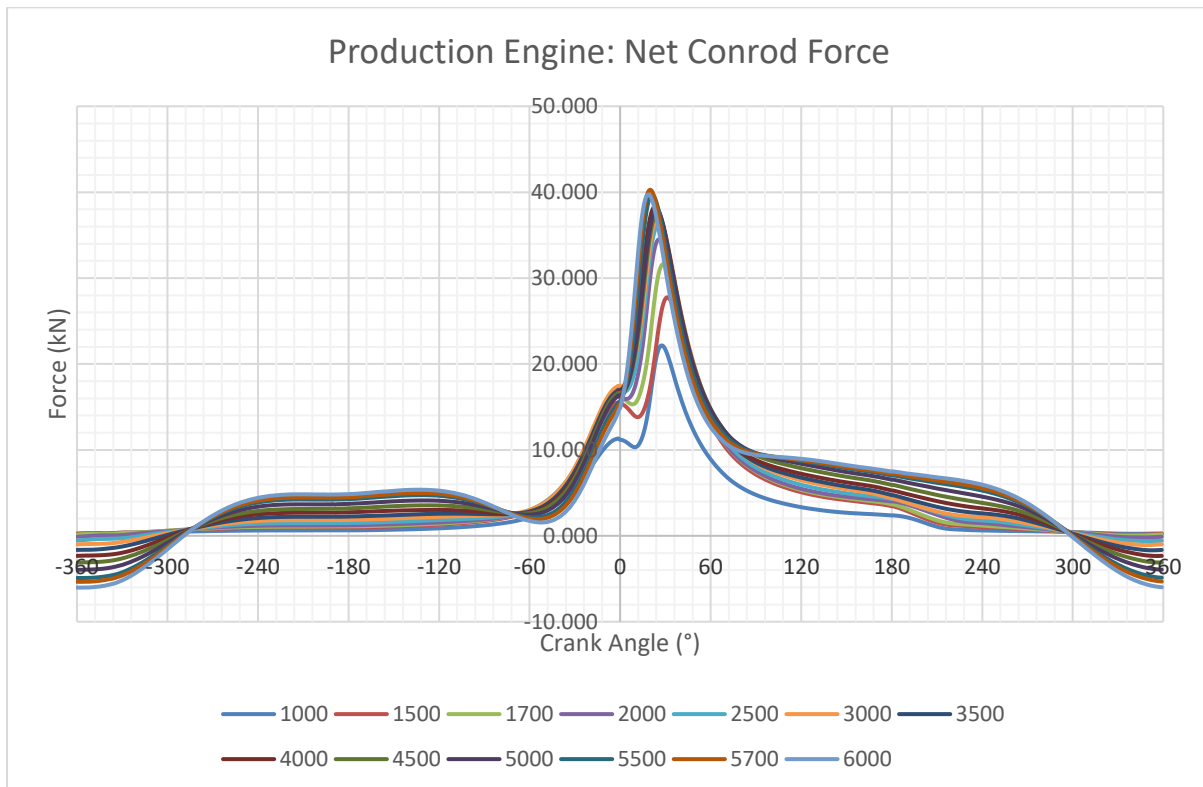


Figure 15: Production Engine Net Conrod Force

The sign in Figure 15 shows whether the connecting rod is under compressive or tensile force. As such, the peak compressive F_{rod} tends to occur at and around 25° ATDC, in fact the peaks occurred at the same crank angle as the peak pressure, this allowed author to assume the same for the modified engine. Whereas the peak tensile F_{rod} always occurred at or very closely to TDC at the end of exhaust stroke.

To calculate the peak compressive and tensile stresses author has divided the F_{rod} by minimal cross-sectional area of the con-rod shank, which author measured. Author concluded that the measurement accuracy was poor as it is impossible to find out where exactly the minimal area actually resides on the shank in the first place. Furthermore, multiple measurements had to be taken, resulting in every

dimension being not measured at the same point. Also, author didn't have a reliable measurement technique to utilise when measuring the shank fillets. However, author believes that the used method was good enough given that results of further calculations won't be affected in great extent by aerial error of what author estimates is 2mm^2 .

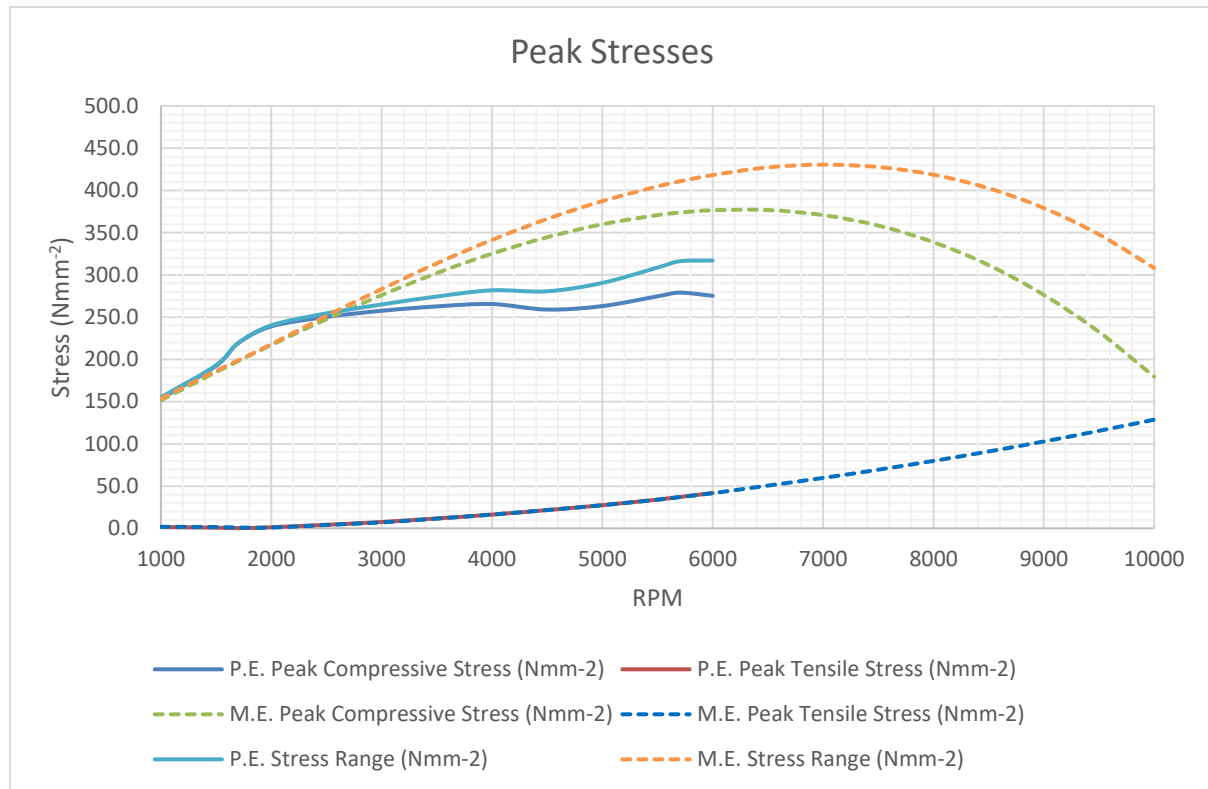


Figure 16: Peak Stresses

Figure 16 shows the results of the stress calculations. Author believes that the tensile stress profile is caused by combination of comparably low pressures and high inertial forces, causing the familiar squared like curve, previously seen in Figure 7, this is further supported by M.E. profile matching that of P.E.

As expected, the compressive stress is higher than tensile. However, author didn't expect such high magnitude difference, this causes the tensile stress to be almost minuscule (at typical production car engine speeds) compared to the compressive stress. However, author is aware of the fact that materials behave differently under compression and stretch. Therefore, what seems like minuscule stress could be significant enough to causes excess fatigue, especially since tensile stress increases the total deformation range.

Naturally, the significance of tensile stress rises with engine speed, at 10,000rpm tensile and compressive stresses of the M.E. almost match. Therefore, even though the peak stresses are relatively low, the deformation range is equal to that happening at 3,500rpm.

Interestingly and unexpectedly, the stress range profiles match their peak pressure counterparts almost perfectly. This has been portrayed in Figure 17.

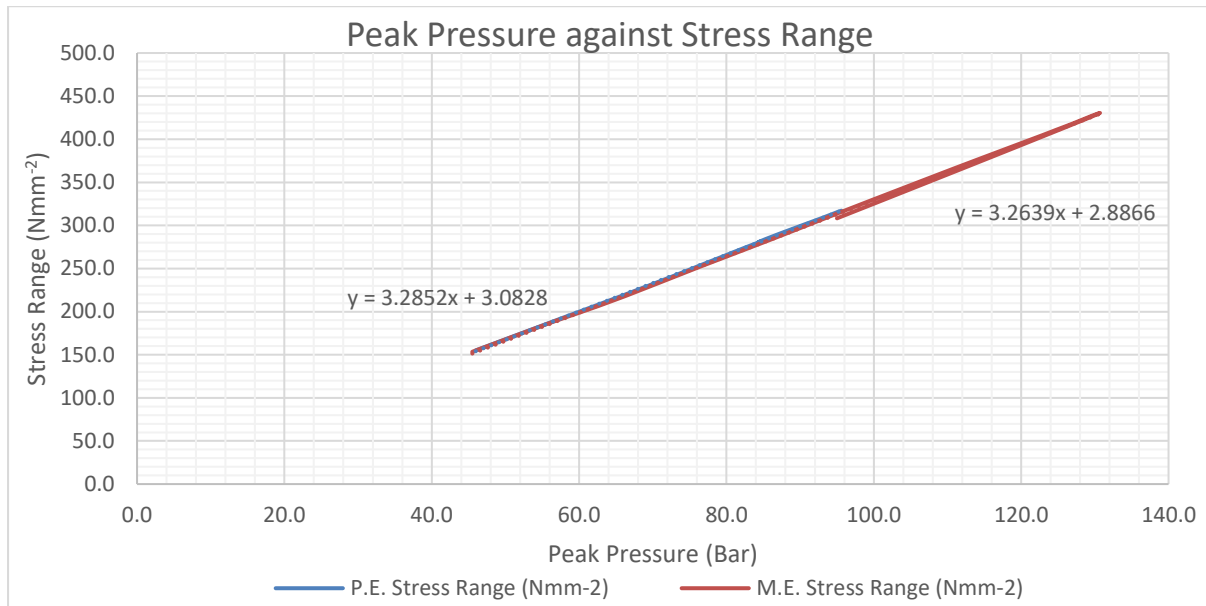


Figure 17: Peak Pressure against Stress Range

Naturally, the stress is affected by the minimal cross-sectional area, however, this can be mitigated by converting the stress range in Figure 17 into force range, due to constant area in both cases, the correlation will stay the same. This is what author has done in Figure 18.

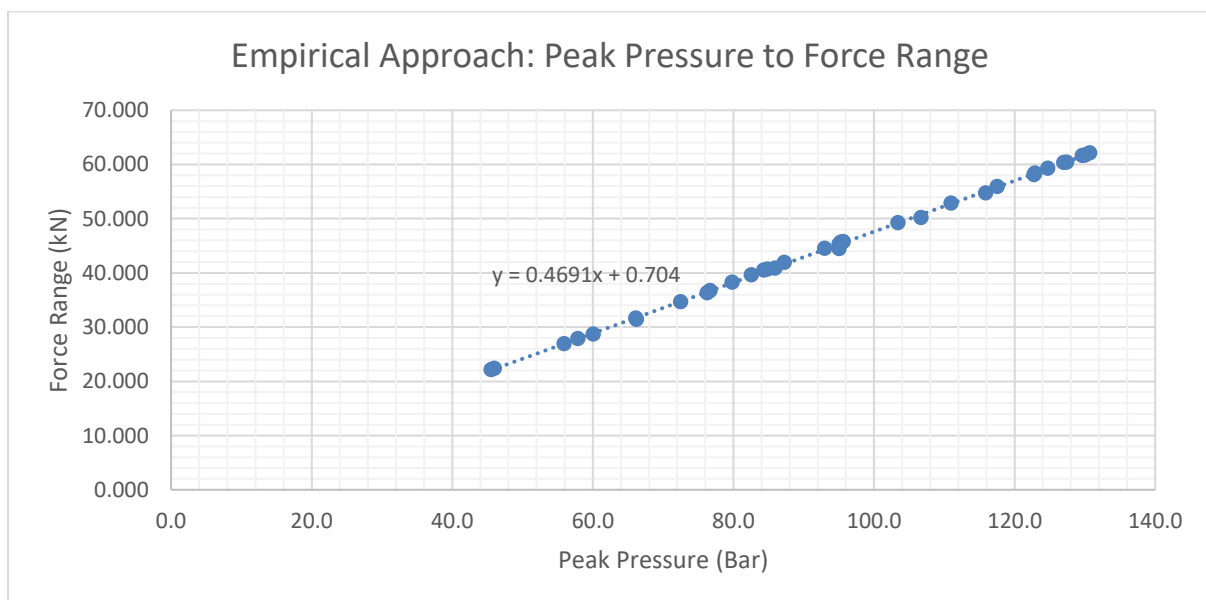


Figure 18: Empirical Approach Peak Pressure to Force Range

This method allows estimation of conrod total deformation through the cycle by just knowing the peak pressure. Naturally another driving factor is m_{rec} , however, author believes that creation of accurate empirical equations is beyond the scope of this investigation.

7 Nominal Safety Factor

Safety factor is a ratio between material strength (typically UTS or YS) and the applied stress. Author has assumed that the material used in the connecting-rod is the typical powder forged steel: P/F-10C50. The utilised material specifications can be observed in Appendix 7, for which, all information was gathered from (Edmond Ilia, 2002) and (Vikrant Garud, 2014).

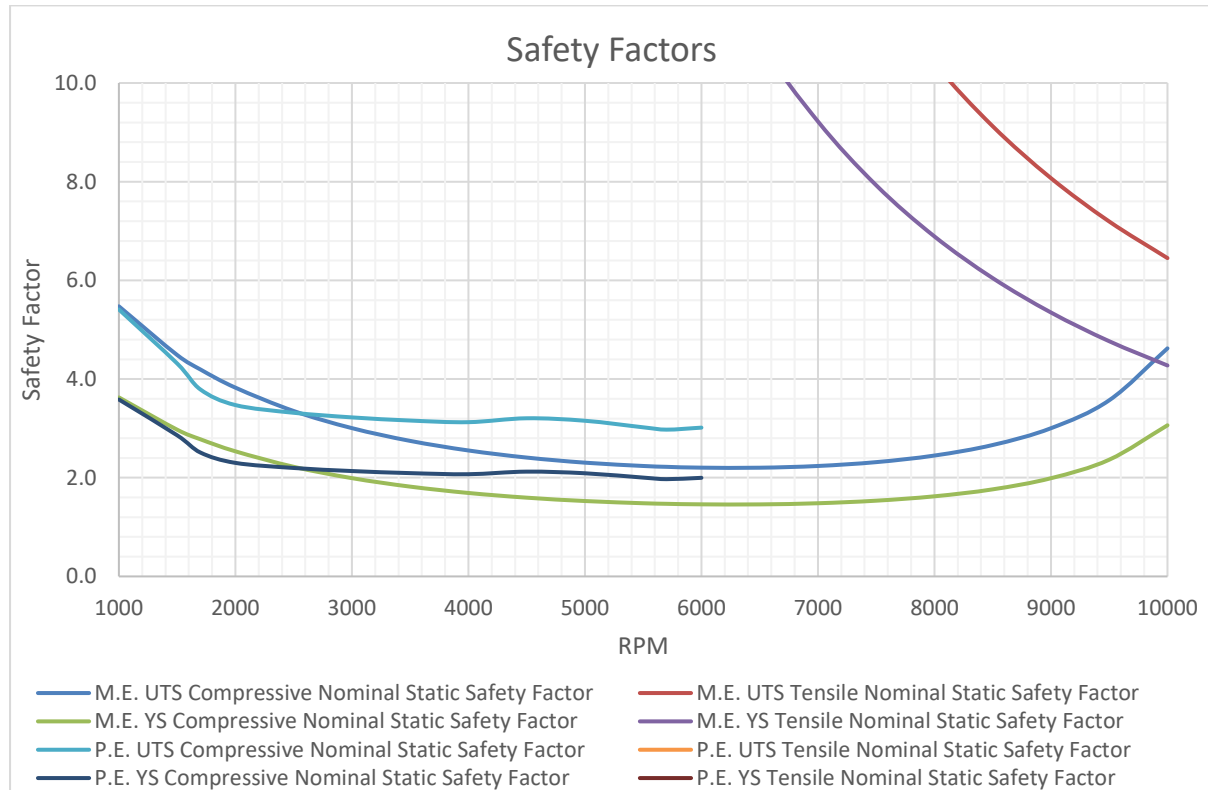


Figure 19: Nominal Safety Factors for P/F-10C50

Figure 19 shows the calculated results. It was previously established in section 6 that tensile stress is much lower than compressive stress. Therefore, as author expected, the tensile safety factors tend to be abnormally high, resulting in them not even fitting on the safety factor scale of 10, in Figure 19, until 7,000rpm is reached. This further supports the hypothesis from section 6.

Figure 19 also shows that the lowest UTS and YS safety factors are 3.0 and 2.0 respectively. This further improved the author's confidence in the accuracy of calculations, since author found those safety factors to be common in researched literature.

Naturally, due to higher peak cylinder pressure of the M.E. both safety factors fall short of their P.E. counterparts. The minimal UTS and YS nominal safety factors under compressive stress are 2.2 and 1.5 respectively, that is equal to safety factor reduction of 26.7% and 25.0%. However, despite the reduction of safety, author believes that due to the hypothetical use of the M.E. in motorsport, the

same connecting-rods could most likely be utilised, as long as frequent engine rebuilds would be carried out to mitigate the risk of con-rod failure.

To increase the minimal safety factors back to that of the production engine, as shown in Figure 20, the minimal cross-sectional area had to be increased by 36.1% to 196.45mm².

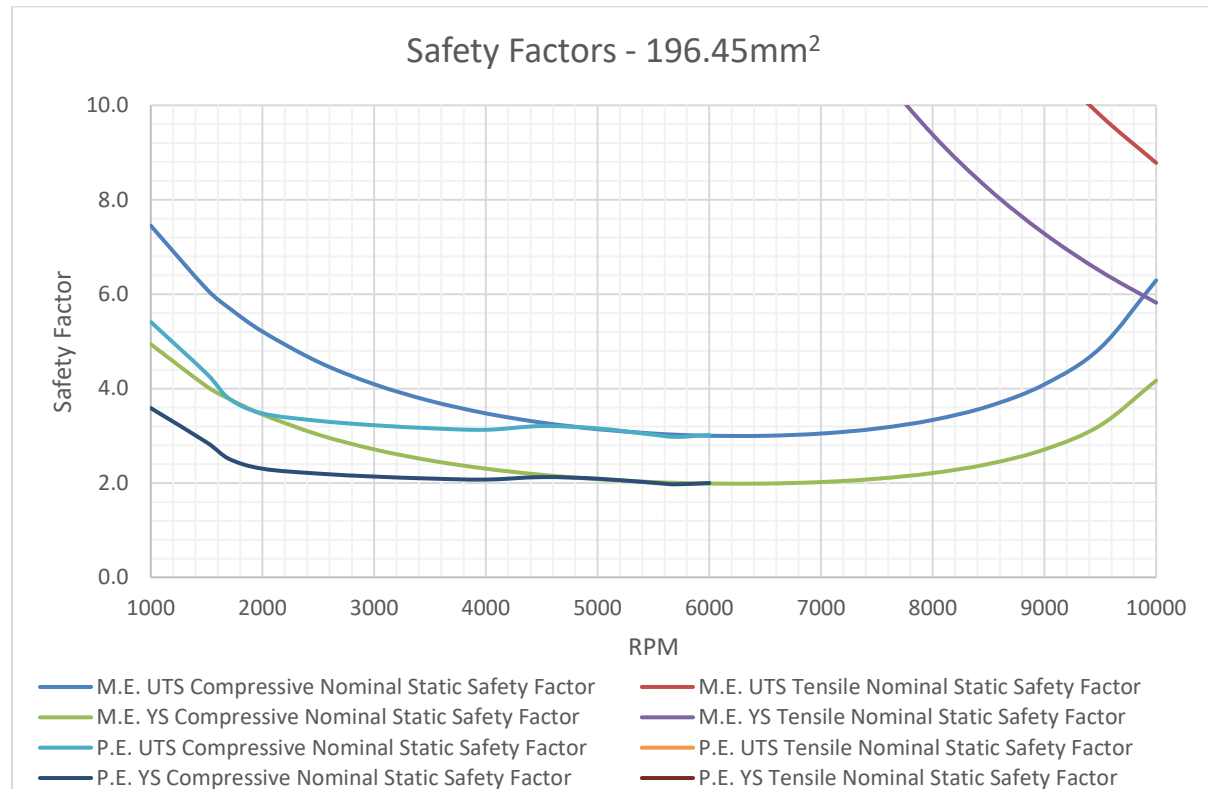


Figure 20: Safety Factors - 196.45mm²

Interestingly, the minimal compressive safety factor of the M.E. all occurs 1,000rpm before the peak cylinder pressure of 130Bar @ 7,000rpm. This is due to very similar gas and frictional forces, but higher inertial forces at 7,000rpm.

8 Con-rod Deformation

If the connecting-rod shad is considered to be a spring, then the Hook's law for linear springs can be incorporated and made the subject of the Young's Modulus, as seen in Equation 16: Shank Stiffness .

$$K_{shank} = \frac{EA}{L_{shank}}$$

Equation 16: Shank Stiffness (McDonald, 2019-2020)

Author is only considering the con-rod shank, as to calculate the total linear deformation of a connecting-rod, one would have to also consider the deformation of the big and small ends, author believes this to be beyond the scope of this investigation.

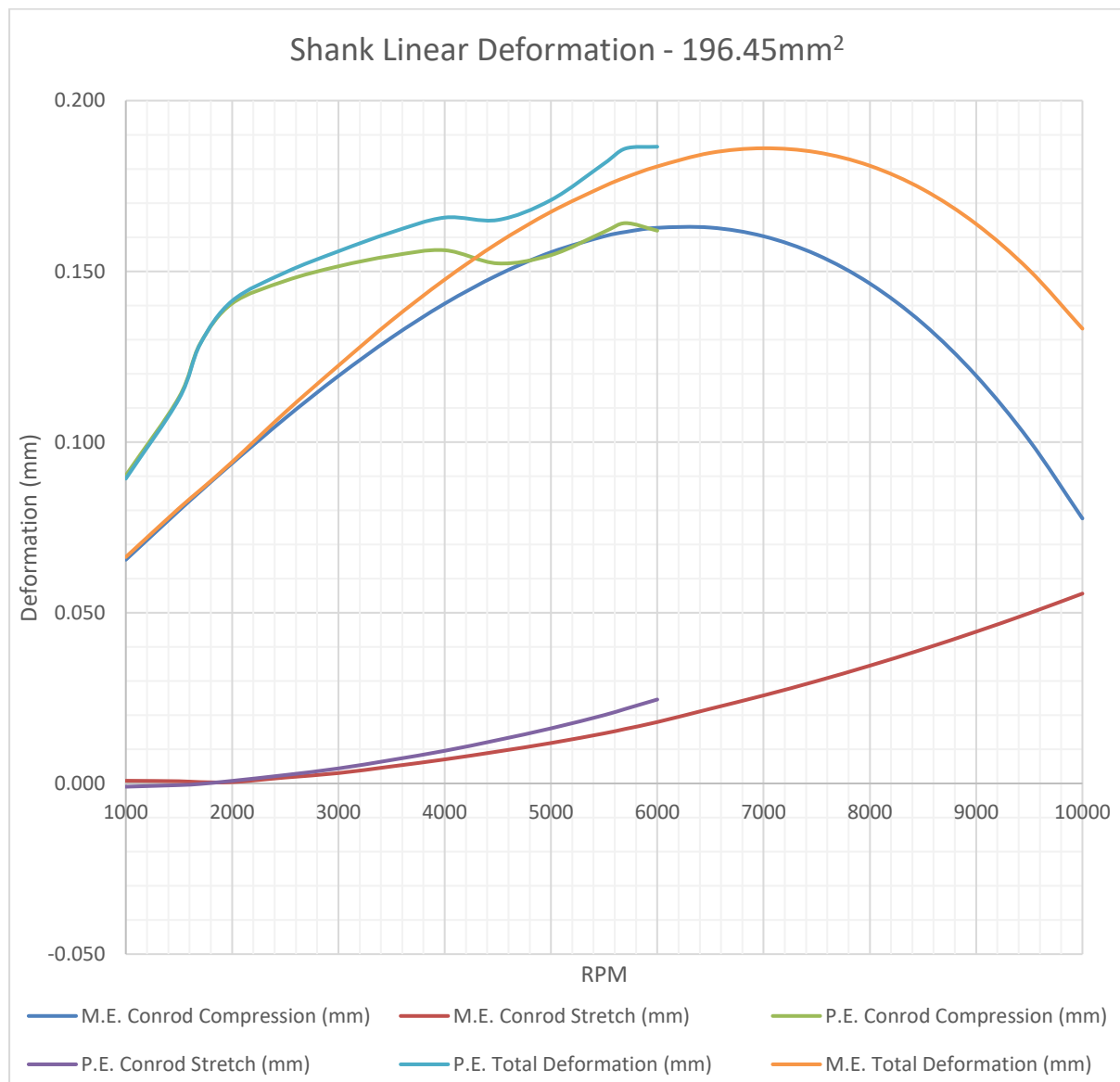


Figure 21: Shank Linear Deformation - 196.45mm²

Figure 21 shows the results of the Equation 16 utilisation. Due to the matching compressive safety factors, the shank compression never exceeded 0.163mm in both cases. However, due to bigger minimal cross-sectional area and relatively similar tensile forces, the shank stretch was decreased by 0.007mm or 28.0%. However, due to much higher redline, the peak shank stretch was 0.056mm.

Most importantly, Figure 21 suggests that due to total deformation, most shank fatigue would occur at the same rpm as the maximum peak cylinder pressure, which is 6,000rpm and 7,000rpm for P.E. and M.E. respectively, just as Figure 1 showed. Author believes this to be important as this knowledge allows con-rod design to focused at just one rpm and peak cylinder pressure, as oppose to the entire rpm range.

9 Con-rod Deformation Utilizing Titanium

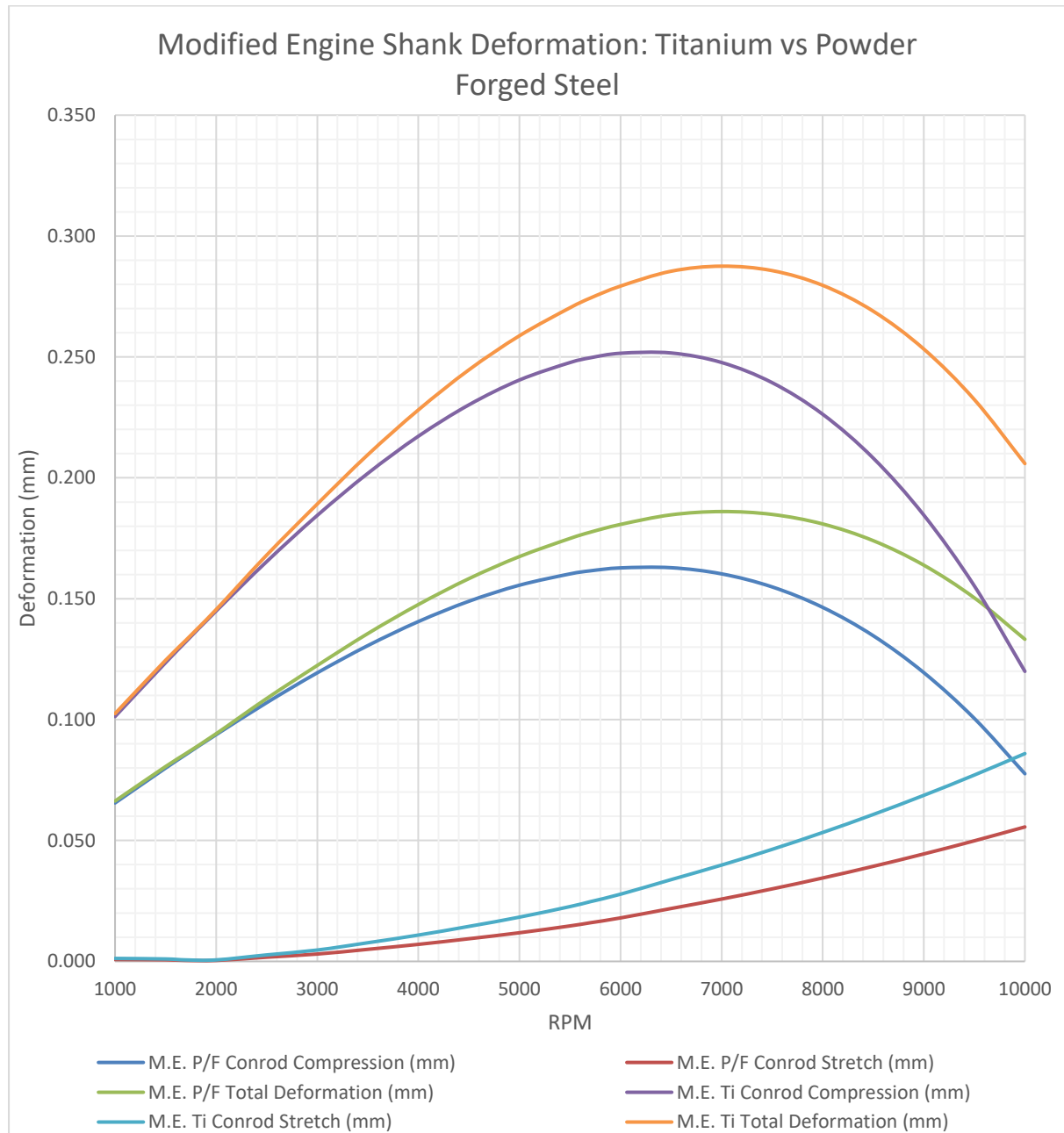


Figure 22: Modified Engine Shank Deformation Titanium vs Forged Steel

Figure 22 shows the shank deformation comparison between titanium and powder forged steel. Due to 35.3% reduction of Young's modulus, the titanium shank deforms 54.8% more at peak total deformation, than its steel counterpart. This increases the fatigue by a big factor, which in turn, could cause a catastrophic con-rod failure.

This could cause major problems, as at 10,000rpm, titanium shank stretches 0.03mm more, this could cause contact between the piston crown and valves. However, 0.03mm is a relatively small increase, especially when typical piston crown to valve clearance @ 12° BTDC are considered.

Nonetheless, to mitigate this issue, the already enlarged minimal cross-sectional area would have to be increased yet again by another 54.5% to 303.69mm², that is equal to a total aerial increase of 110.4%. As a result, the minimal compressive UTS and YS safety factors increased to 5.3 and 4.6 respectively, as shown in Figure 23.



Figure 23: Safety Factors - 303.69mm²

10 Conclusion

Author believes that the investigation was a success since the utilised methods returned results which were both ‘reasonable’ and supported by the researched literature.

However, author believes that several issues occurred throughout the investigation which require further work:

- Throughout the investigation the $m_{rec.}$ has been kept constant, this naturally would not be the case if certain changes were carried out to the engine, especially those seen in sections 2, 6, 7, 8 and 9.
 - The mass of the connecting-rod hasn’t been considered, this was done mainly due to the additional complication since a con-rod is rotating and reciprocating at the same time. However, con-rod’s own inertia would also act on itself, causing the F_{NI} to be higher.
- The accurate measurement of minimal cross-sectional area has proven to be a challenge, to improve this author suggests utilising a 3D scanning technique such as CT Scan.
- Author also didn’t consider how stresses would spread through the con-rod, especially around the big and small ends, as that would have to be considered to compute the total linear deformation of a con-rod.
- Further research would have to be carried out on material performance under compression and tensile forces.

Author believes that most of these issues can be mitigated by the utilisation of computational FEA package such as SolidWorks Simulation or Ansys.

11 References

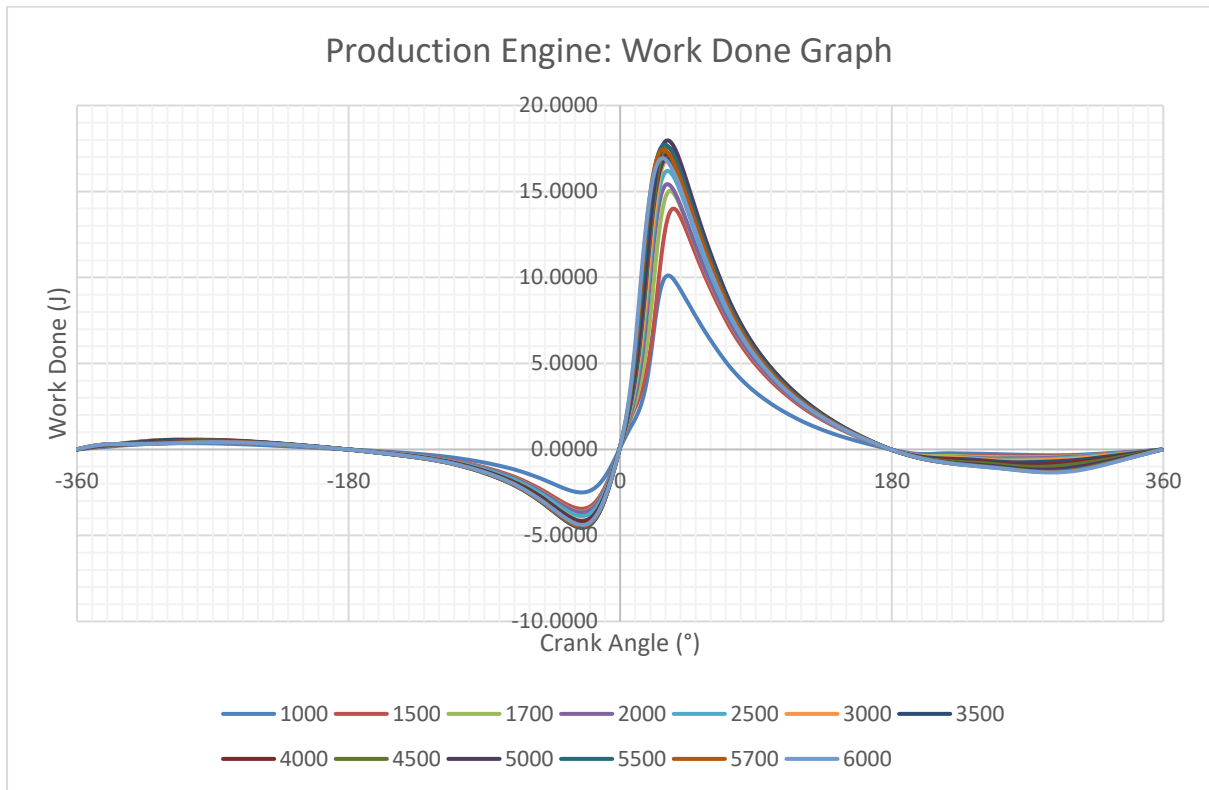
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12 Appendix

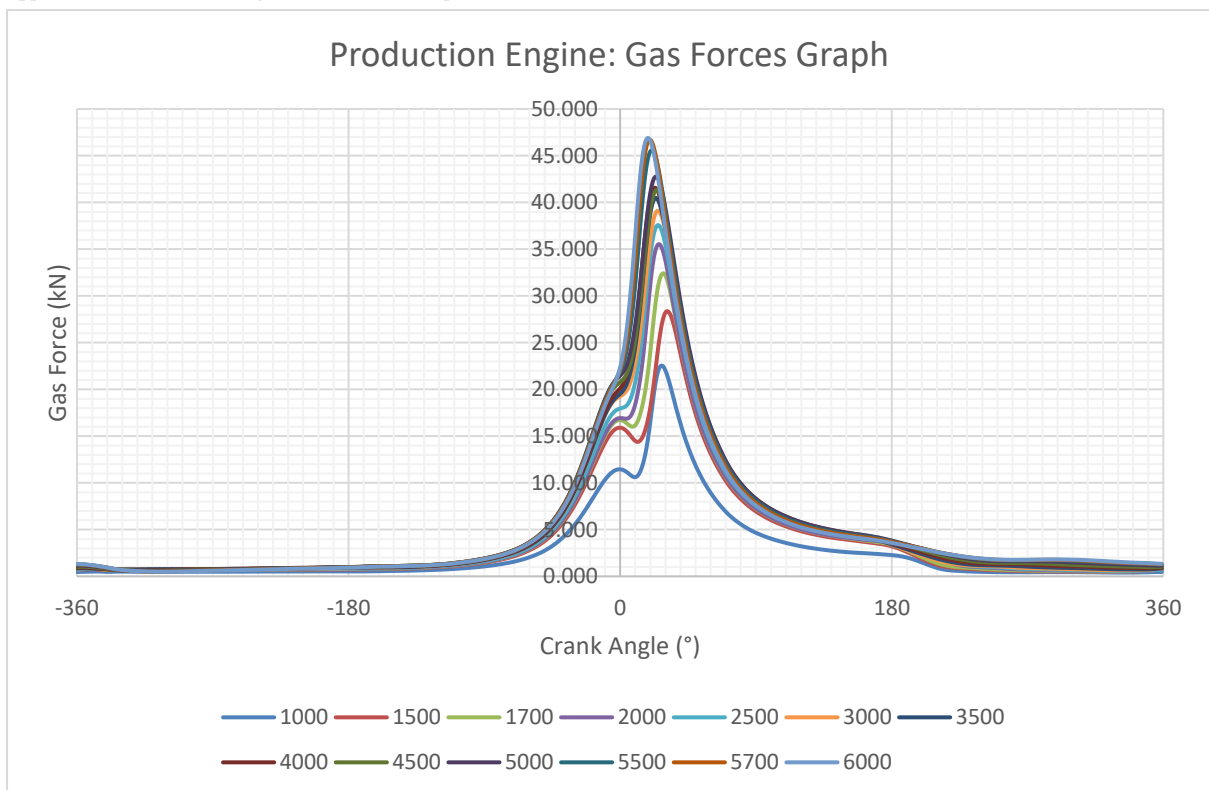
Production Engine Calculations																											
RPM		6000.0		RPM		1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000									
Bore (mm)		79.0		P.E. GIMEP (Bar)		11.6	16.4	17.8	18.7	19.8	20.6	20.8	21.1	21.6	21.9	21.8	21.5	20.9									
Stroke (mm)		81.4		P.E. P.MEP (Bar)		0.0	0.1	0.0	-0.2	-0.3	-0.4	-0.6	-1.0	-1.4	-1.9	-2.2	-2.3	-2.5									
Pin Offset (mm)		0.5		P.E. NIMEP (Bar)		11.6	16.5	17.9	18.5	19.5	20.2	20.2	20.1	20.1	20.0	19.5	19.1	18.4									
CR		10.1		P.E. FMEP _{indicated} (Bar)		0.6	0.9	0.9	0.9	1.0	1.1	1.1	1.2	1.2	1.3	1.3	1.3	1.3									
No. of cylinders		4		P.E. BMEP (Bar)		10.9	15.6	16.9	17.6	18.5	19.2	19.1	18.9	18.9	18.7	18.2	17.8	17.1									
Coefficient Of Friction		0.1		P.E. Indicated Power (kW)		15.4	32.8	40.3	49.7	65.8	82.3	96.9	112.4	129.1	145.5	159.2	162.7	166.5									
Ring Inner Diameter (mm)		73.2		P.E. Indicated Torque (Nm)		147.0	208.8	226.5	237.4	251.2	262.0	264.3	268.2	273.9	277.9	276.5	272.5	265.0									
Ring Height (mm)		1.17		P.E. Brake Power (kW)		14.6	31.2	38.3	46.7	61.4	76.5	88.9	100.8	113.1	124.4	133.2	134.8	136.3									
Connecting Rod Specifications:				P.E. Brake Torque (Nm)		139.0	198.5	215.1	223.0	234.4	243.5	242.4	240.5	240.0	237.6	231.3	225.9	216.9									
Material:		P/F-10C50		P.E. Real Power (kW)		14.7	36.3	42.2	50.3	63.9	77.9	89.8	102.6	106.5	111.0	110.6	111.6	108.1									
Type:		Powder Forged Steel		P.E. Real Torque (Nm)		140.0	231.0	237.0	240.0	244.0	248.0	245.0	245.0	226.0	212.0	192.0	187.0	172.0									
Conrod Length (mm)		134		P.E. FMEP (Bar)		0.5	-1.7	-0.8	-0.4	0.2	0.7	0.9	0.8	2.3	3.3	4.4	4.4	4.9									
Shank Length (mm)		100		P.E. Peak Pressure (Bar)		46.0	57.9	66.1	72.5	76.7	79.8	82.6	84.8	84.3	87.2	93.0	95.3	95.6									
Rod Shank minimum cross-sectional area (mm²)		144.34		P.E. Peak Pressure Crank Angle (°)		28.0	31.0	29.0	26.0	25.0	25.0	24.0	23.0	24.0	24.0	21.0	20.0	18.0									
UTS (Nmm-2)		830		P.E. Peak Gas Force (kN)		22.527	28.363	32.398	35.531	37.575	39.111	40.469	41.573	41.324	42.760	45.580	46.691	46.863									
E (Nmm ⁻² * 10 ³)		170		P.E. Peak Inertial Force (kN)		0.192	0.433	0.556	0.769	1.201	1.730	2.355	3.076	3.893	4.806	5.815	6.245	6.920									
Displacement (cc)		1596.0		P.E. Peak Compressive Friction (kN)		0.437	0.588	0.641	0.650	0.665	0.686	0.685	0.676	0.683	0.698	0.674	0.666	0.641									
K _{peak} (Nmm ⁻¹)		245378.0		P.E. Peak Compressive Force (kN)		22.143	27.755	31.623	34.498	36.121	37.169	37.923	38.324	37.378	37.973	39.644	40.273	39.730									
Masses (g)				P.E. Peak Compressive Force Crank Angle (°)		28.0	31.0	29.0	26.0	25.0	25.0	24.0	23.0	24.0	24.0	21.0	20.0	19.0									
				The same Crank Angle as Peak Gas?		Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	No									
Conrod		267.38		P.E. Peak Tensile Friction (kN)		0.0048	0.0041	0.0033	0.0047	0.0069	0.0105	0.0111	0.0120	0.0067	0.0078	0.0103	0.0106	0.0113									
Conrod Cap		103.8		P.E. Peak Tensile Force (kN)		0.234	0.118	0.035	-0.185	-1.079	-1.688	-2.348	-3.121	-3.960	-4.911	-5.359	-6.032										
Conrod Bolts		35.19		P.E. Peak Tensile Force Crank Angle (°)		339.0	340.0	336.0	346.0	350.0	349.0	353.0	355.0	360.0	360.0	358.0	358.0	358.0									
Conrod Assembly		406.37		P.E. Peak Compressive Stress (Nmm ⁻²)		153.4	192.3	219.1	239.0	250.2	257.5	262.7	265.5	259.0	263.1	274.7	279.0	275.3									
Gudgeon pin		80.22		P.E. Peak Tensile Stress (Nmm ⁻²)		1.6	0.8	0.2	1.3	4.2	7.5	11.7	16.3	21.6	27.4	34.0	37.1	41.8									
Piston		257.7		P.E. Stress Range (Nmm ⁻²)		155.0	193.1	219.3	240.3	254.4	265.0	274.4	281.8	280.6	290.5	308.7	316.1	317.0									
Piston Ring		5.49		P.E. Compressive Nominal Static Safety Factor		5.41	4.32	3.79	3.47	3.32	3.22	3.16	3.13	3.21	3.15	3.02	2.97	3.02									
Circlip		0.77		P.E. Tensile Nominal Static Safety Factor		512.25	1014.04	3469.51	647.23	197.84	111.00	70.96	51.03	38.38	30.26	24.40	22.35	19.86									
Reciprocating Assembly		350.44		P.E. Conrod Compression (mm)		0.090	0.113	0.129	0.141	0.147	0.151	0.155	0.156	0.152	0.155	0.162	0.164	0.162									
				P.E. Conrod Stretch (mm)		-0.001	0.000	0.000	0.001	0.002	0.004	0.007	0.010	0.013	0.016	0.020	0.022	0.025									
				P.E. Total Deformation (mm)		0.089	0.113	0.129	0.141	0.150	0.156	0.161	0.166	0.165	0.171	0.182	0.186	0.186									

Appendix 1: Production Engine Calculations Table

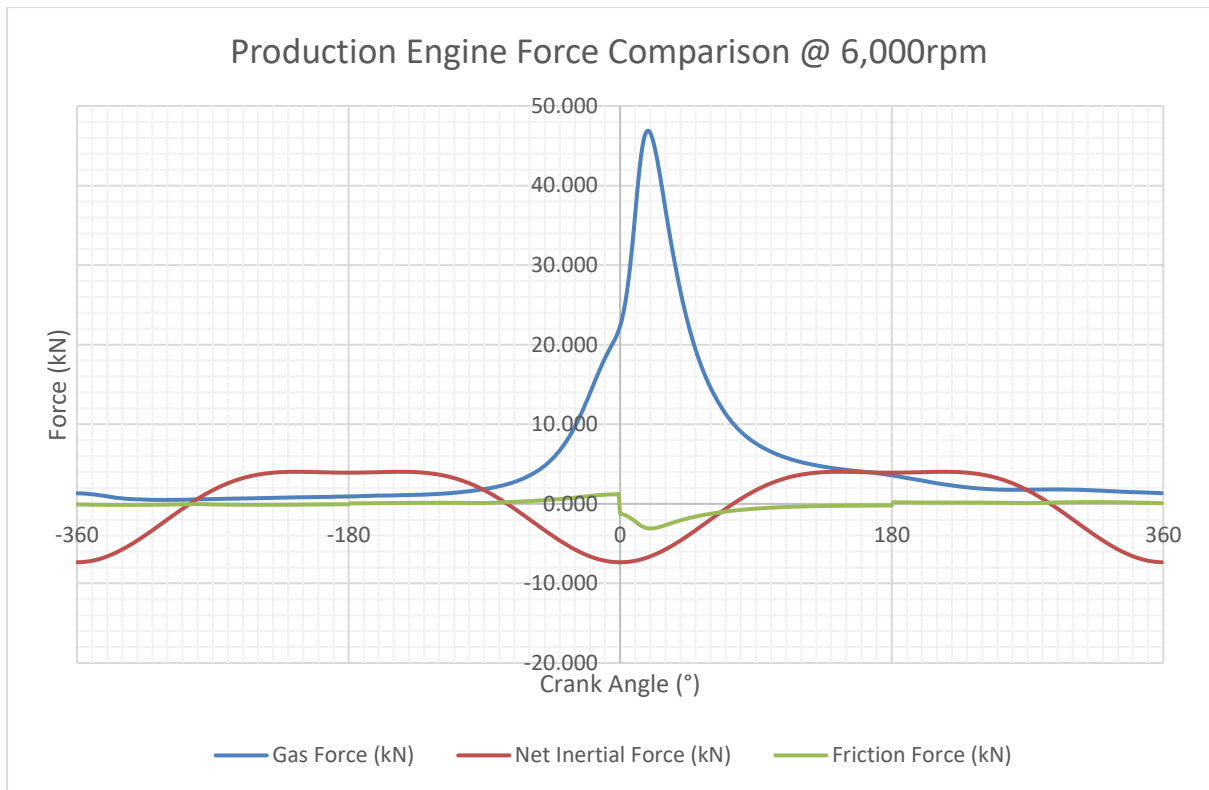
Modified Engine Calculations																													
RPM	6000.0	RPM	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000	6500	7000	7500	8000	8500	9000	9500	10000						
Bore (mm)	79.0	M.E. GIMEP (Bar)	12.3	14.9	15.9	17.4	19.9	22.3	24.6	26.7	28.6	30.2	31.6	32.0	32.6	33.2	33.5	33.3	32.7	31.5	29.8	27.5	24.6						
Stroke (mm)	81.4	M.E. Indicated Power (kW)	16.3	29.6	35.9	46.3	66.3	89.1	114.5	142.1	171.1	201.0	230.9	242.7	260.1	287.4	312.0	332.5	347.8	356.4	356.9	347.8	327.3						
Pin Offset (mm)	0.5	M.E. Indicated Torque (Nm)	155.7	188.7	201.8	221.3	253.1	283.7	312.8	339.1	363.0	383.8	400.9	406.6	413.9	422.3	426.6	423.4	415.2	400.4	378.7	349.6	312.5						
No. of cylinders	4	M.E. Peak Pressure (Bar)	45.50	55.90	60.04	66.20	76.25	85.90	95.00	103.40	110.95	117.50	122.90	124.70	127.00	129.65	130.70	130.00	127.40	122.75	115.90	106.70	95.00						
Coefficient Of Friction	0.1	M.E. ϕ Crank Angle (°)	29	31	29	26	25	24	24	24	24	24	24	24	24	24	24	24	24	24	21	20	19						
Ring Inner Diameter (mm)	73.2	M.E. Peak Gas Force (kN)	22.303	27.400	29.430	32.440	37.370	42.105	46.566	50.683	54.380	57.596	60.242	61.136	62.251	63.590	64.095	63.722	62.447	60.189	56.810	52.301	46.966						
Ring Height (mm)	1.17	M.E. Peak Inertial Force (kN)	0.203	0.456	0.586	0.810	1.266	1.823	2.482	3.242	4.103	5.065	6.129	6.582	7.294	8.560	9.927	11.396	12.966	14.638	16.411	18.285	20.260						
Reciprocating Assembly Mass (g)	350.44	M.E. Peak Compressive Force (kN)	0.432	0.567	0.580	0.593	0.659	0.717	0.788	0.851	0.905	0.950	0.984	0.994	1.005	1.012	1.005	0.981	0.940	0.880	0.799	0.629	0.494						
Connecting Rod Specifications:		M.E. Average Tensile Cyl. Pressure (Bar)	0.960	1.387	1.482	1.404	1.425	1.683	1.709	1.869	2.058	2.344	2.607	2.647	2.740	2.696	2.836	2.989	3.136	3.286	3.436	3.586	3.736						
Material:	Ti6AL4V	M.E. Peak Compressive Force (kN)	21.894	26.697	29.569	31.319	35.734	39.857	43.616	46.935	49.741	51.960	53.503	53.945	54.351	54.377	53.525	51.724	48.900	44.981	39.895	33.559	25.305						
Type:	Annealed Titanium	M.E. Peak Tensile Friction (kN)	0.0024	0.0035	0.0037	0.0035	0.0036	0.0042	0.0043	0.0047	0.0052	0.0059	0.0065	0.0066	0.0069														
Shaft Length (mm)	134	M.E. Peak Tensile Force (kN)	0.264	0.240	0.240	0.335	0.515	0.726	0.932	1.140	1.348	1.556	1.764	1.972	2.180	2.388	2.596	2.804	3.012	3.220	3.428	3.636	3.844						
Thrust Length (mm)	100	M.E. Peak Compressive Stress (N/mm²)	70.21	87.91	88.1	103.1	137.17	132.12	143.6	155.4	166.8	178.1	176.2	177.6	179.9	176.7	170.3	167.0	168.2	161.3	158.1	153.1	110.5	85.3					
Rod Shaft minimum cross-sectional area (mm²)	303.69	M.E. Peak Tensile Stress (N/mm²)	0.97	0.7	0.4	0.4	1.9	3.3	5.5	7.7	10.3	13.0	15.6	17.6	19.8	21.4	28.3	33.0	37.9	43.2	48.9	54.8	61.1						
UTS (N/mm²)	110	M.E. Stress Range (N/mm²)	72.9	88.6	84.5	103.5	110.6	134.6	149.1	162.3	174.1	184.1	192.4	195.2	198.7	203.1	204.6	203.3	199.0	191.4	180.2	165.3	146.8						
E (N/mm² × 10)	110	M.E. Compressive Nominal Static Safety Factor	13.2	10.8	10.1	9.2	8.1	7.2	6.6	6.1	5.8	5.6	5.4	5.3	5.3	5.3	5.4	5.6	5.9	6.4	7.2	8.6	11.1						
Displacement (mm)	1596.0	M.E. Tensile Nominal Static Safety Factor	109.1	132.5	128.0	126.0	127.7	129.1	129.4	129.3	129.3	129.3	129.3	129.3	129.3	129.3	129.3	129.3	129.3	129.3	129.3	129.3	129.3						
K_{axial} (N/mm²)	334059.0	M.E. Corrod Corrosion Factor	0.081	0.081	0.090	0.100	0.116	0.131	0.146	0.160	0.174	0.188	0.202	0.216	0.230	0.244	0.258	0.272	0.286	0.299	0.313	0.327	0.341						
		M.E. Corrod Stress Factor	0.001	0.001	0.000	0.000	0.002	0.003	0.005	0.007	0.009	0.012	0.015	0.016	0.018	0.022	0.026	0.030	0.034	0.039	0.044	0.050	0.056						
		M.E. Total Deformation (mm)	0.066	0.081	0.086	0.094	0.100	0.127	0.136	0.148	0.158	0.167	0.175	0.177	0.181	0.185	0.186	0.185	0.181	0.174	0.164	0.150	0.136						



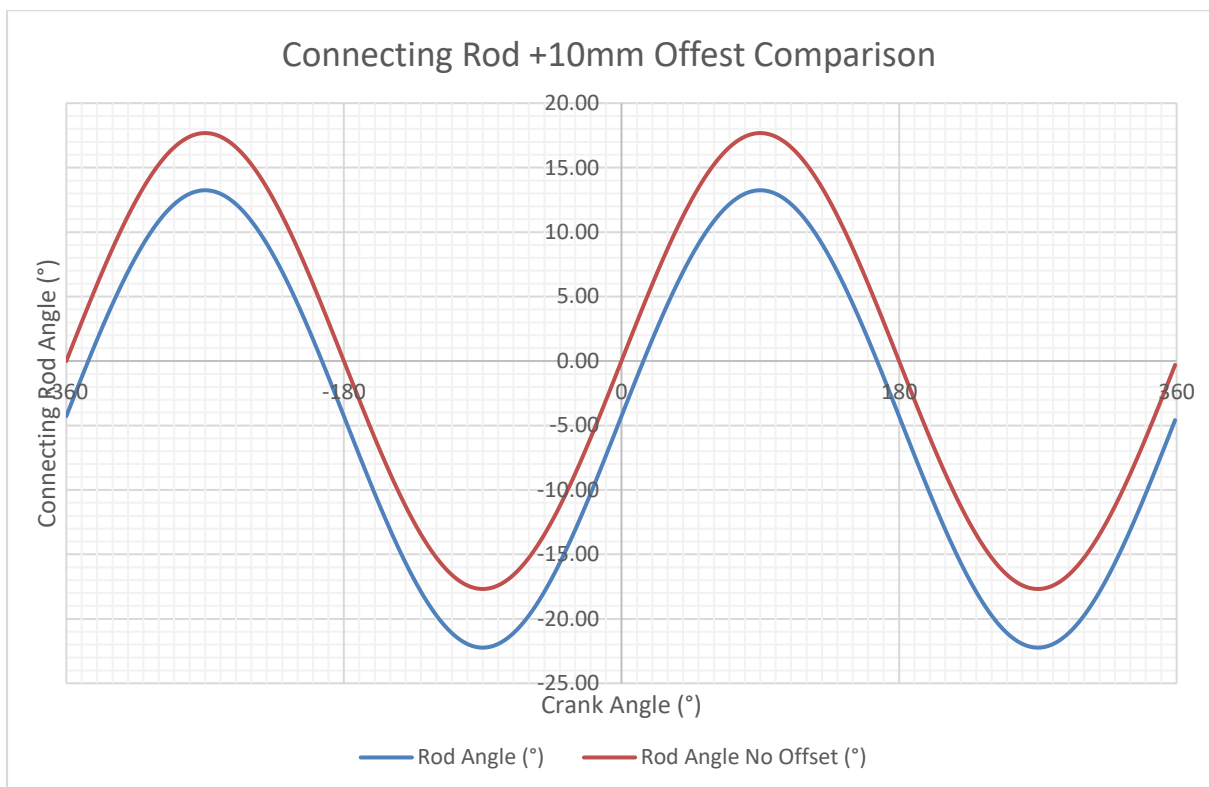
Appendix 3: Production Engine Work Done Graph



Appendix 4: Production Engine Gas Forces Graph



Appendix 5: Production Engine Force Comparison @ 6,000rpm



Appendix 6: Connecting Rod +10mm Offset Comparison

Materials

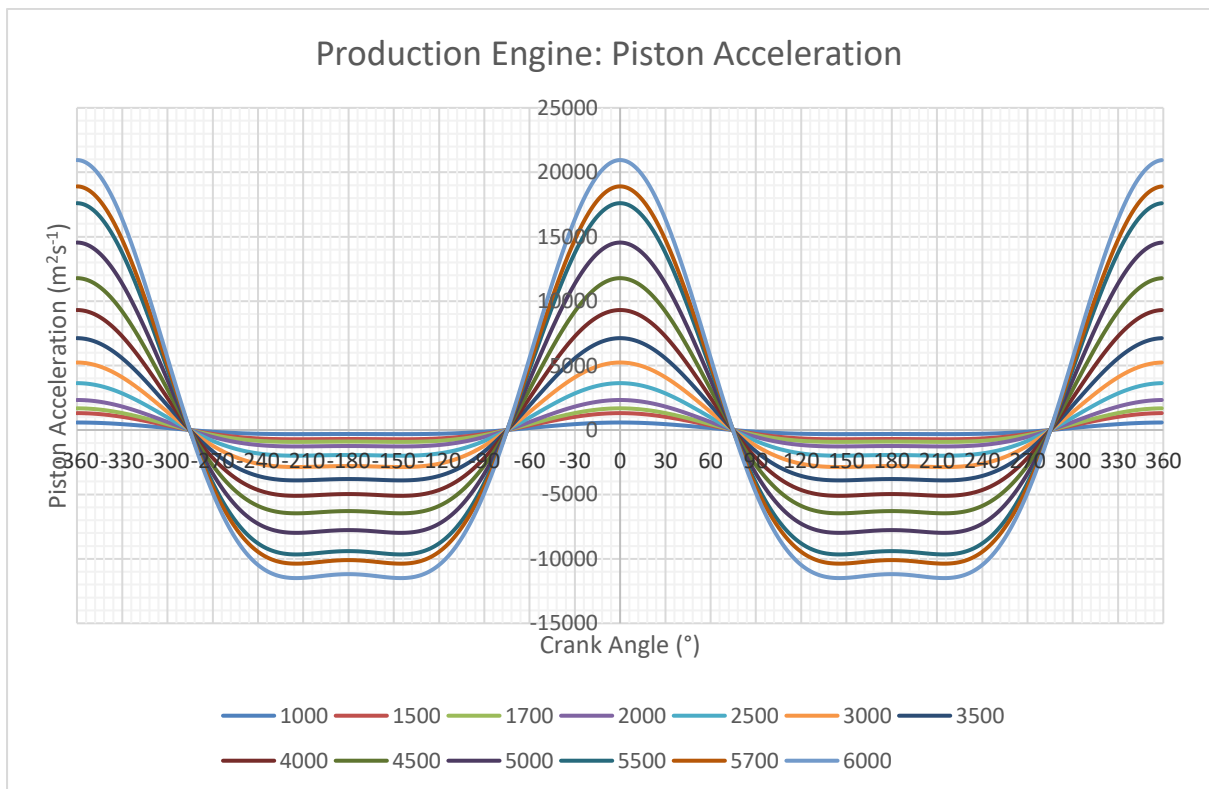
Name:	P/F-10C50	Ti6AL4V
Type:	Powder Forged Steel	Annealed Titanium
UTS (Nmm ⁻²)	830	950
YS (Nmm ⁻²)	550	827
E (Nmm ⁻² * 10 ³)	170	110

Appendix 7: Material Specification

Empirical Peak Pressure to Force Equation

RPM	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000	6500	7000	7500	8000	8500	9000	9500	10000
Peak Pressure (Bar)	46.0	57.9	66.1	72.5	76.7	79.8	82.6	84.8	84.3	87.2	93.0	95.3	95.6	45.50	55.90	60.04	66.20	76.25	85.90	95.00	103.40	110.95	117.50	122.90	124.70	127.00	129.65	130.70	130.00	127.40	122.75	115.90	106.70	95.00
Force Range (kN)	22.377	27.873	31.657	34.683	36.726	38.248	39.611	40.671	40.500	41.933	44.555	45.632	45.762	22.148	26.915	28.700	31.447	36.313	40.872	45.281	49.287	52.868	55.916	58.419	59.280	60.356	61.683	62.134	61.738	60.422	58.112	54.728	50.214	44.490

Appendix 8: Empirical Peak Pressure to Force Equation Table



Appendix 9: Production Engine Piston Acceleration