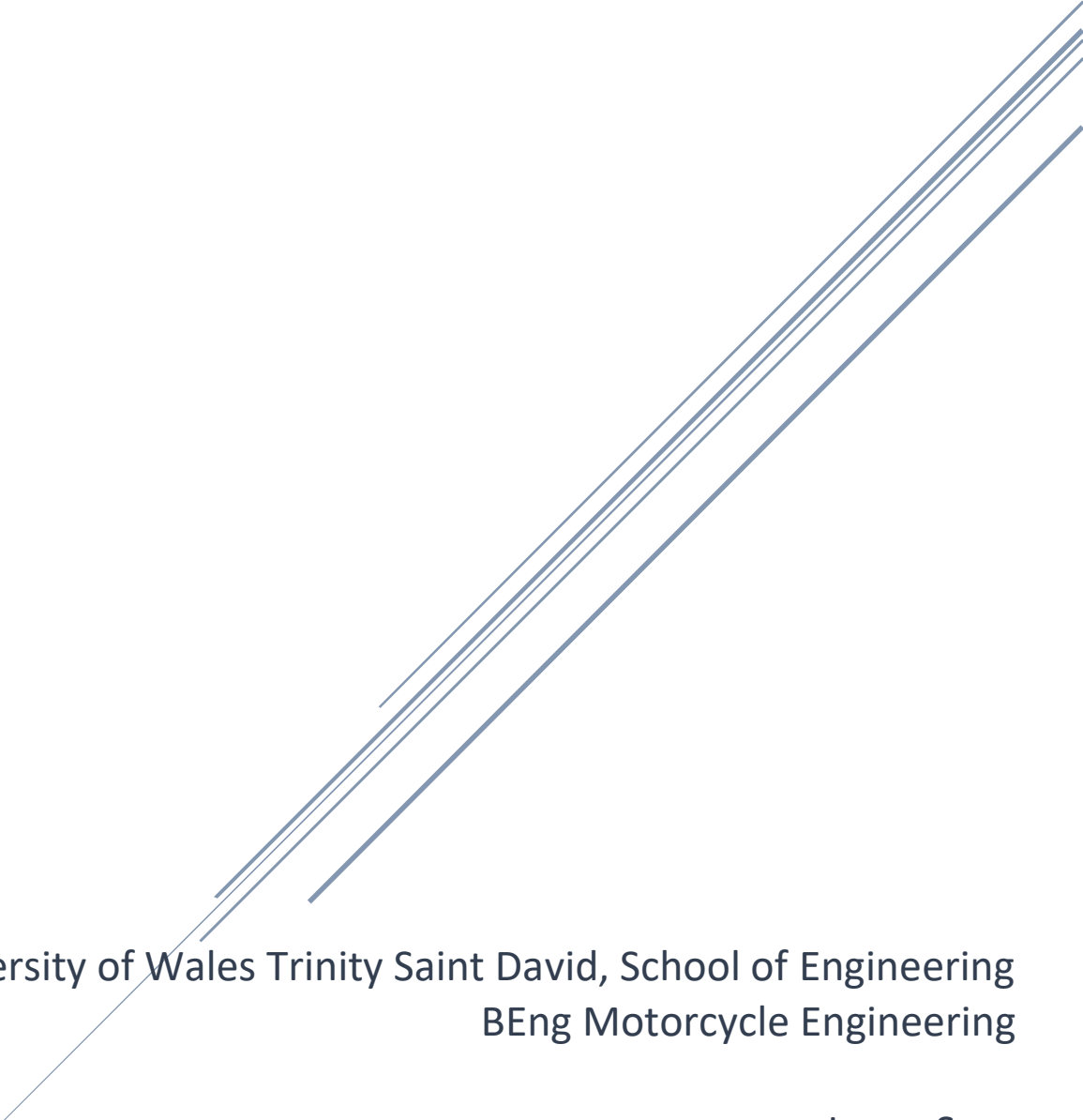


ASSIGNMENT 1(A)

Advanced Engine Design, (M6X00410)



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Introduction

In order for correct operation of an ICE the internal reciprocating components need to be properly designed for the operating conditions and applied loads that all the parts will endure. During combustion and other critical times during the four stroke cycle the connecting rod is put under immense stress, and thus the connecting rod has historically undergone a great deal of research in order to produce a component that is able to withstand these large forces.

This report will cover how to determine these forces by numerically analysing a Ford 1.6L GTDI EcoBoost connecting rod and reverse engineering the generated results. This has been accomplished by taking the real world in-cylinder pressure data and then back calculating the applied forces in order to determine the reasoning behind the material / design choices the Ford engineers decided upon when designing the connecting rod.

Summary

The following data highlights the key results generated from the report:

- The critical loads generated from the pressure data occurred at 20 degrees after TDC at 5700 RPM (compressive) instead of 6000 RPM due to the lower inertia forces, and the other critical load (tensile) was 1 degree after TDC for the exhaust stroke at 6000 RPM.
- The associated stress / strain results generated a safety factor of 3.72 in a compressive mode and 20.81 in a tensile mode at the maximum load cases for the OEM connecting rod.
- This report clearly demonstrated the importance of selecting an appropriate material and associated design shape (ref. Appendix A ANSYS results) in order to create a component that is able to withstand the high compressive / tensile loads and are able to stay within clearance tolerances.
- The tensile mode failure was not an issue due to the low revving RPM ceiling. The only time when the tensile failure is applicable is for high revving engines, which supports why motorcycles commonly have connecting rods failures in tensile modes.

Connecting Rod Numerical Analysis

The following results have been calculated using a quasi-static approach in order to ensure the critical loads and resulting forces were considered for every degree of crank angle. The input variables for the OEM Ford 1.6L GTDI EcoBoost connecting rod can be seen below, Table 1.

Table 1: OEM Piston Specifications

OEM Specifications		
Piston Weight	0.258	kg
Piston Component Weight	0.140	kg
Piston dia.	79.000	mm
Stroke	0.0814	m
Cyl Volume	9.98E-05	m ³
Number of Cylinders	4	-
Radius of Crank	0.0407	m
Conrod Length	0.1340	m
Conrod cross section	195.0	mm ²
Piston Area	4901.67	mm ²
Offset	1.50	mm
Offset / Length	0.01	-
W=	628.32	rad/s
N=	3.29	-
UTS	750.00	N/mm ²
Youngs Modulus E	170000.00	N/mm ²

Production engine, (Reference Appendix B for further clarification)

A) 1. *Peak gas force* = *Piston Area x Combustion Force* = **46862.91 N at 6000 RPM**

Table 2: Peak Gas Forces

RPM	Max Pressure (Bar)	Peak Gas Force (N)	Crank Angle
1000	45.96	22526.60	28.00
1500	57.86	28363.02	31.00
1700	66.10	32397.71	29.00
2000	72.49	35531.47	26.00
2500	76.66	37575.47	25.00
3000	79.79	39111.04	25.00
3500	82.56	40468.92	25.00
4000	84.81	41573.15	24.00
4500	84.31	41324.14	24.00
5000	87.24	42759.72	24.00
5500	92.99	45579.77	21.00
5700	95.26	46690.86	20.00
6000	95.61	46862.91	18.00

- The peak gas force is calculated using the average pressure data across all cylinders, meaning that cylinder variation is not taken into account.

B) 1. Inertia loads on connecting rod exerted from piston, Figure 1.

$$\text{Inertia Force} = \text{Mass of Piston} \times \text{Piston Acceleration}$$

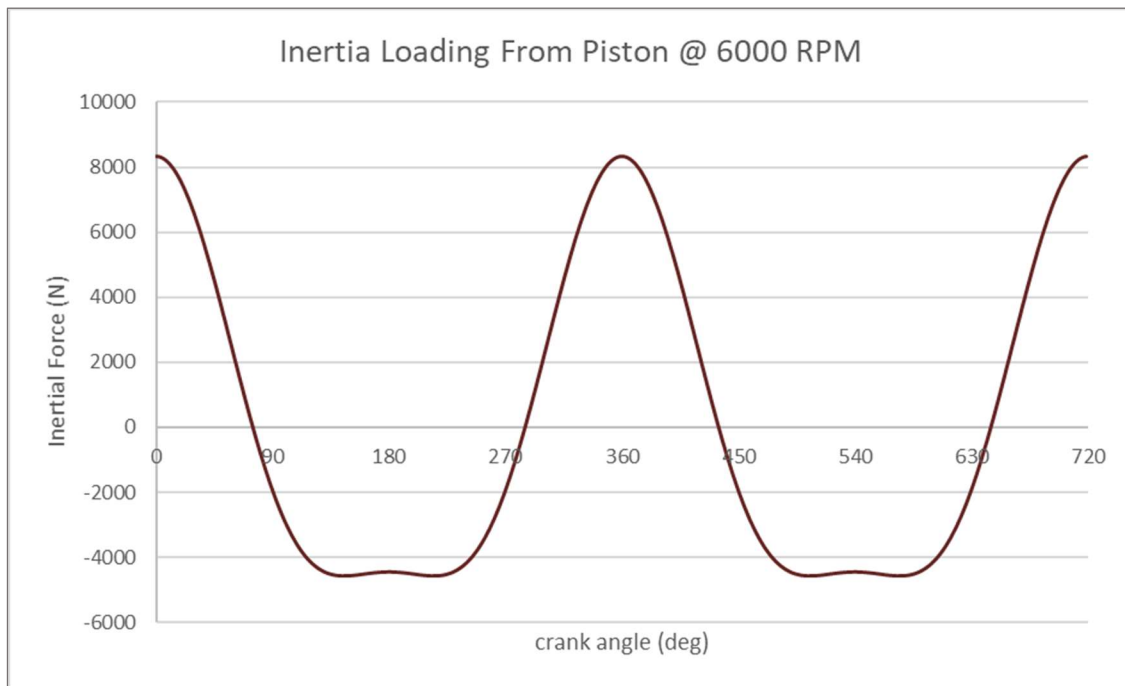


Figure 1: Connecting Rod Inertia Load

- 1/3 of the connecting rod mass was not included due to the limited scope of the assignment.
- The inertia force in the rod was calculated by taking the vertical inertia force and taking the rod force component by dividing by the cosine of the rod angle.

C) 1.
$$\text{Max Compressive Force} = \frac{(\text{Combustion Force} - \text{Inertia Force})}{\cos(\text{Rod Angle})} - \text{Combined Friction Force}$$

Max Compressive Force = 39282.82 N at 5700 RPM 20 degrees after TDC.

- Largest combined net force, usually when gas pressure is the highest due to combustion happening in the cylinder, (Usually around 15° – 20° after TDC, (Iliev, 2012))
- Even though the maximum peak gas force in Part A occurs at 6000 RPM, the maximum compressive force is occurring at 5700 RPM because the inertial loads and friction forces are less due to the lower rotational velocity.

$$D) \quad 1. \text{ Max Tensile Force} = \frac{(\text{Combustion Force} - \text{Inertia Force})}{\cos(\text{Rod Angle})} - \text{Combined Friction Force}$$

Max Tensile Force = 7027.66 at 6000 RPM, 1 degree after TDC of exhaust stroke.

- Smallest combined net force, meaning when piston is generating the highest inertia force when compared to the gas pressure force, (Usually around TDC for exhaust stroke).

$$E) \quad 1a. \text{ Max Compressive Stress} = \frac{\text{Maximum Compressive Stress (Part C)}}{\text{Con Rod Shank Minimum Cross-sectional Area}}$$

Max Compressive Stress = 201.45 $\frac{N}{mm^2}$ at 5700 RPM

$$1b. \text{ Max Tensile Stress} = \frac{\text{Maximum Tensile Stress (Part D)}}{\text{Con Rod Shank Minimum Cross-sectional Area}}$$

Max Tensile Stress = 36.04 $\frac{N}{mm^2}$ at 6000 RPM

$$F) \quad 1a. \text{ Compressive Safety Factor} = \frac{\text{UTS of Material}}{\text{Maximum Compressive Stress (Part E)}}$$

Compressive Safety Factor = 3.72 at 5700 RPM

- Based on historical data the compressive safety factor for OEM connecting rods is usually around 3 – 5 depending on the application (ILIEV, 2015), meaning the generated value is validated. This of course is based on using the Ultimate Tensile Strength.

$$1b. \text{ Tensile Safety Factor} = \frac{\text{UTS of Material}}{\text{Maximum Tensile Stress (Part E)}}$$

Tensile Safety Factor = 20.81 at 6000 RPM

- It is interesting to note that during standard operating conditions the OEM connecting rod is nowhere near failure in the Tensile direction.

*Importantly, the safety factors are based off the UTS of the material and not the yield strength due to the ease of comparing historical data (Centre, 2014) and also because it is easier to test the design to failure. Although, in practical terms designing to the yield strength would generate more reliable data.

G) 1a.
$$\text{Max Compression Distance} = \frac{\text{Maximum Compressive Stress (Part E)} * \text{Cross-Section Area}}{\text{Material Stiffness (k)}}$$

Max Compression Distance = 0.159 mm at 5700 RPM

1b.
$$\text{Max Stretch} = \frac{\text{Maximum Tensile Stress (Part E)} * \text{Cross-section area}}{\text{Material Stiffness (k)}}$$

Max Stretch = 0.028 mm at 6000 RPM

Modified engine, (Reference Appendix C for further clarification)

*All formulas used apart for Part H are identical to the above Production engine calculations.

- In order to properly compare the effects that the inertial loads / gas pressures and material choice has when designing a connecting rod, the real world pressure data was extrapolated for a higher revving engine, Figure 2. This was achieved by combining a linear-polynomial method of extrapolating the pressure data for 8000 RPM. The justification of the higher revving engine is that the modified engine is able to rev higher due to improved valve springs to prevent float and larger cams in order to stop the choking effect seen at higher RPM's on a dyno graph.

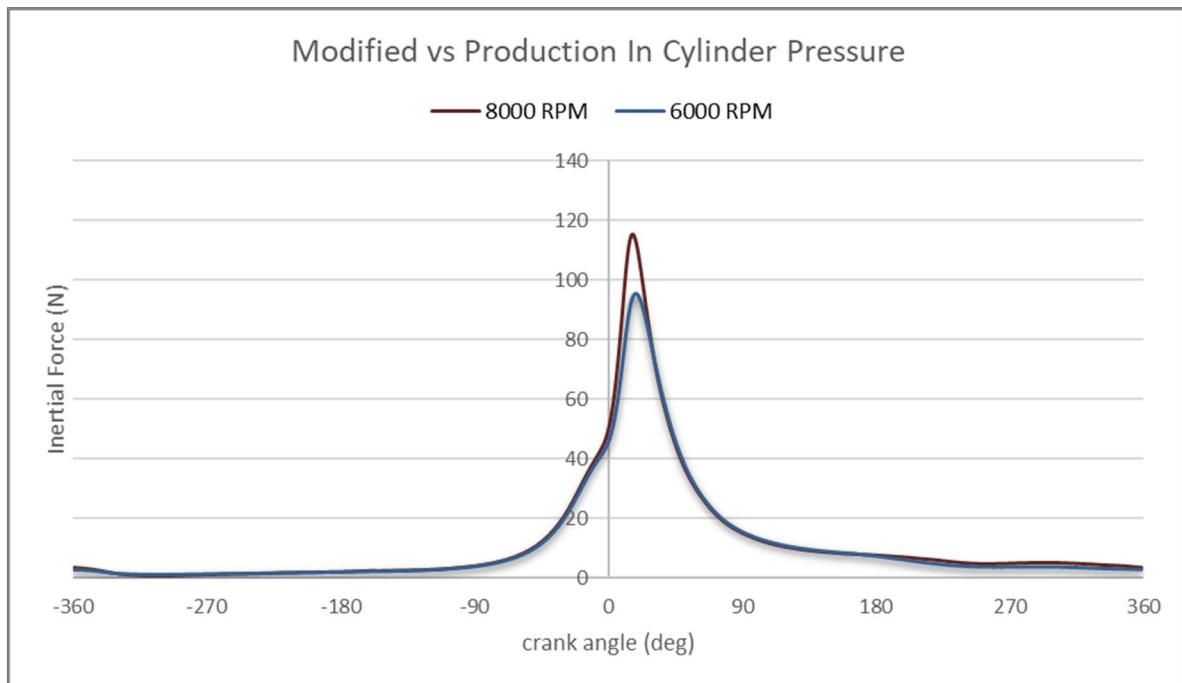


Figure 2: Modified vs Production in Cylinder Pressure

A) 2. **Peak gas force = 56654.64 N at 8000 RPM**

B) 2. Inertia loads on connecting rod exerted from piston, Figure 3.

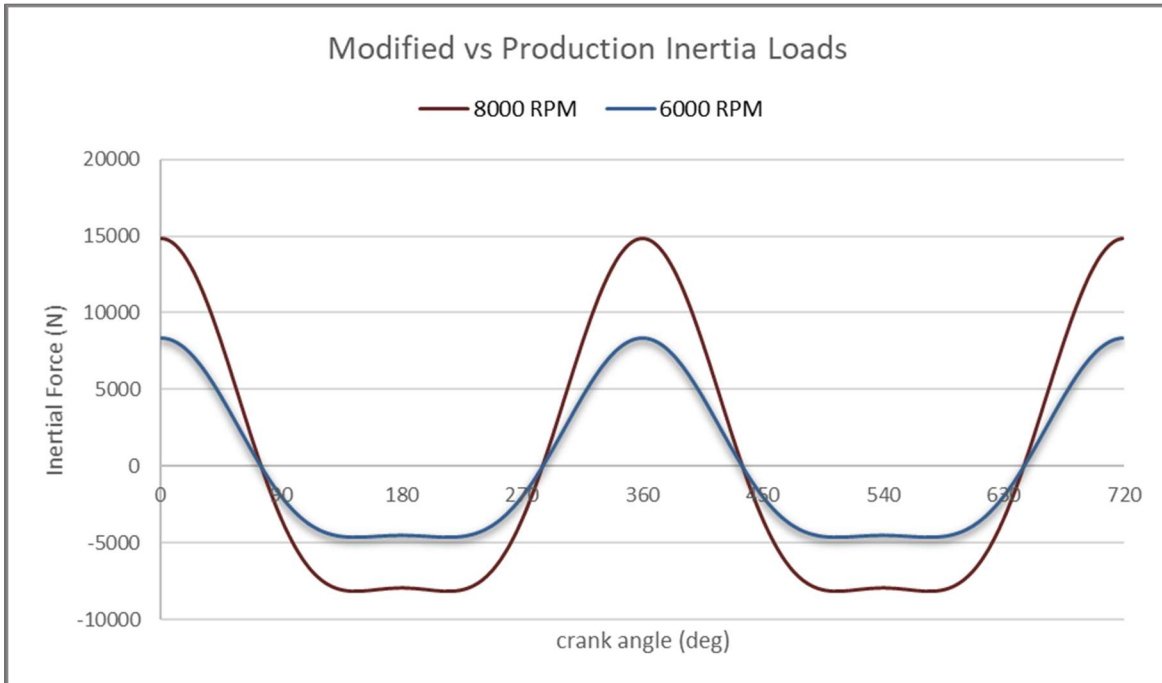


Figure 3: Modified vs Production Inertia Loads

- The graph above clearly demonstrates that the reciprocating mass has a much greater effect at higher rpm due to the increase of the piston acceleration, which is due to the rise in crank angular velocity.

C) 2. **Max Compressive Force = 41959.47 N at 8000 RPM**, 16 degrees after TDC.

- An increase of **6.81%** over the production results at 6000 RPM.

D) 2. **Max Tensile Force = 14840.760 N at 8000 RPM**, 1 degree before TDC of exhaust.

- An increase of **137.18%** over the production results at 6000 RPM.

*The results from part C and D for the modified engine highlight the dramatic difference between the maximum compressive and tensile force that the connecting rod experiences for higher revving engines.

E) 2a. **Max Compressive Stress = $215.18 \frac{N}{mm^2}$ at 8000 RPM**

2b. **Max Tensile Stress = $76.11 \frac{N}{mm^2}$ at 8000 RPM**

F) 2a. **Compressive Safety Factor = 3.49 at 8000 RPM**

- An **8.03%** reduction when compared with the production engine results at 5700 RPM.

2b. **Tensile Safety Factor = 9.86 at 8000 RPM**

- A **46.87%** reduction when compared with the production engine results at 5700 RPM.

G) 2a. **Max Compression Distance = 0.170 mm at 8000 RPM**

2b. **Max Stretch Distance = 0.060 mm at 8000 RPM**

H) The following input variables were used to calculate an identical connecting rod design in a Titanium alloy, Table 3.

Table 3: Titanium Alloy Properties

Titanium Ti4 v6AL		
<i>UTS</i>	950.00	N/mm ²
<i>Youngs Modulus E</i>	110000.00	N/mm ²
<i>Stiffness (K)</i>	160074.63	N/mm

1a. **Max Compression Distance = 0.262 mm at 8000 RPM**

1.b **Max Stretch Distance = 0.093 mm at 8000 RPM**

- Due to a smaller stiffness value for the Titanium material the maximum compression and stretch increased by **54.54%** at 8000 RPM when switching to a Titanium connecting rod, Figure 4.

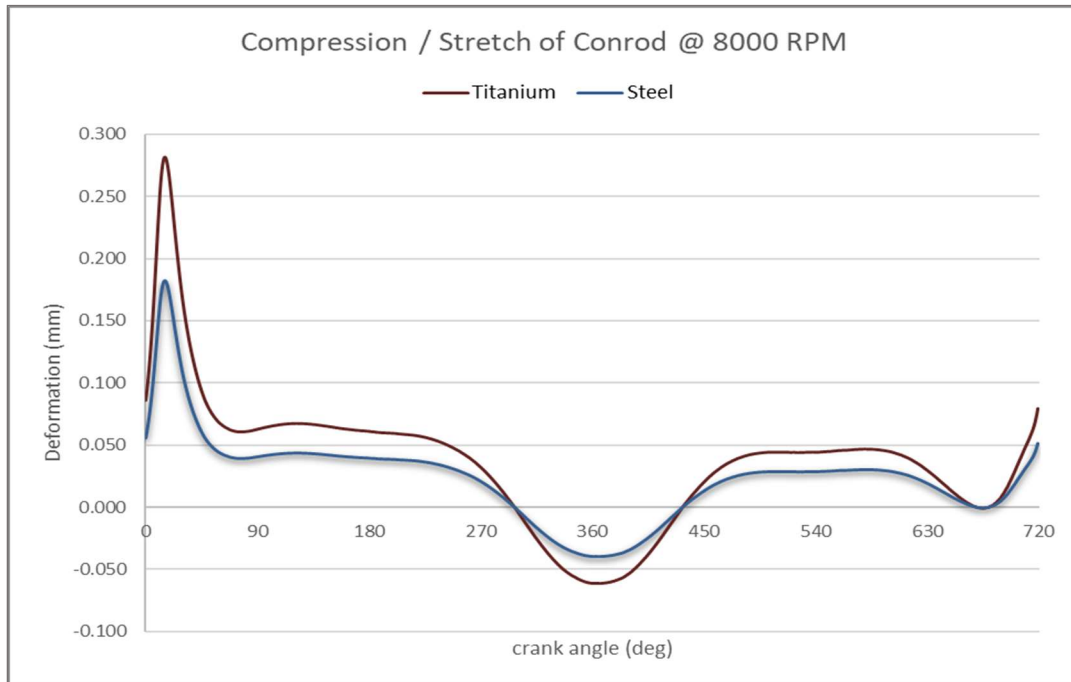


Figure 4: Titanium vs Steel Deformation

More importantly this could cause problems for the operating clearances due to the piston being able to protrude farther into the clearance volume and thus potentially making contacting with a valve. Although, this is not assuming a change in mass for the Titanium connecting rod, meaning that the inertial forces have not decreased which would then lower the tensile stretch values. Which is why this would need to be investigated beyond the scope of the assignment to provide a definitive result.

Interestingly, in order for the titanium rod to generate identical compressive and tensile safety factors the cross sectional area would be **154.0 mm²**, assuming the connecting rod is a simple rectangular beam. This change in cross sectional area would then generate a compressive distance of **0.357 mm** and tensile stretch distance of **0.078 mm**. This simple change in cross sectional area perfectly demonstrates that designers must change the overall shape of the titanium connecting rod and produce a structure that has more mass further away from the centre (commonly an “H” type design) (Bari, 2017). The “H” beam allows for designers to produce a lighter titanium rod without sacrificing the stiffness and resistance to buckling.

Conclusion

The results generated from this report have proven that by extrapolating real world in cylinder pressure data into the related forces the connecting rod design can be either validated or improved upon for future designs. This report however was limited in the overall scope and thus many additions were made to try and create a better overall simulation. For example, the following additions were made: piston pin offset, skirt friction, ring friction, associated torque values, strain values, mechanical efficiency and a variety of mean effective pressure modes. All these additions can be viewed in the following appendices, although much more research needs to be done beyond the scope of this assignment to definitely generate reliable results for designing a connecting rod.

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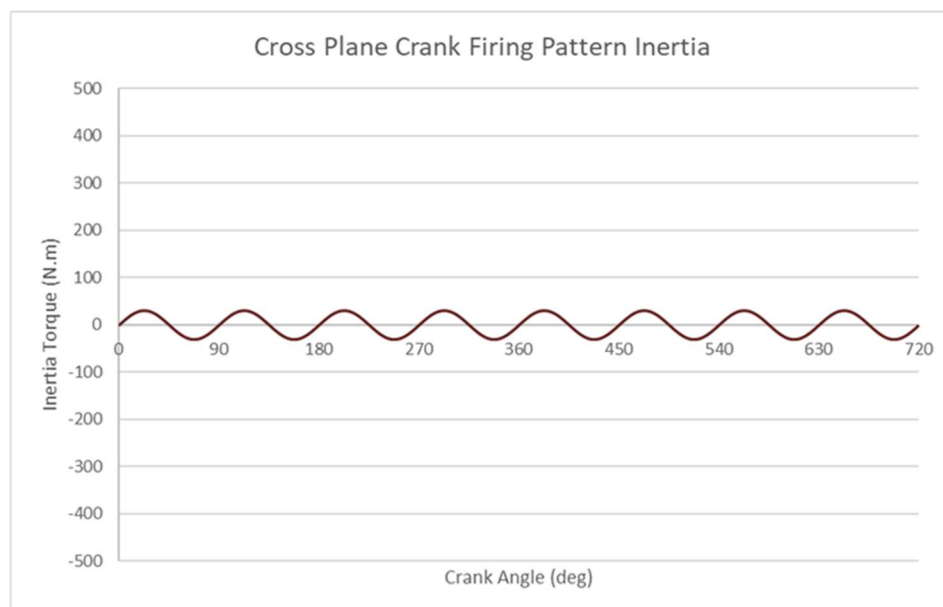
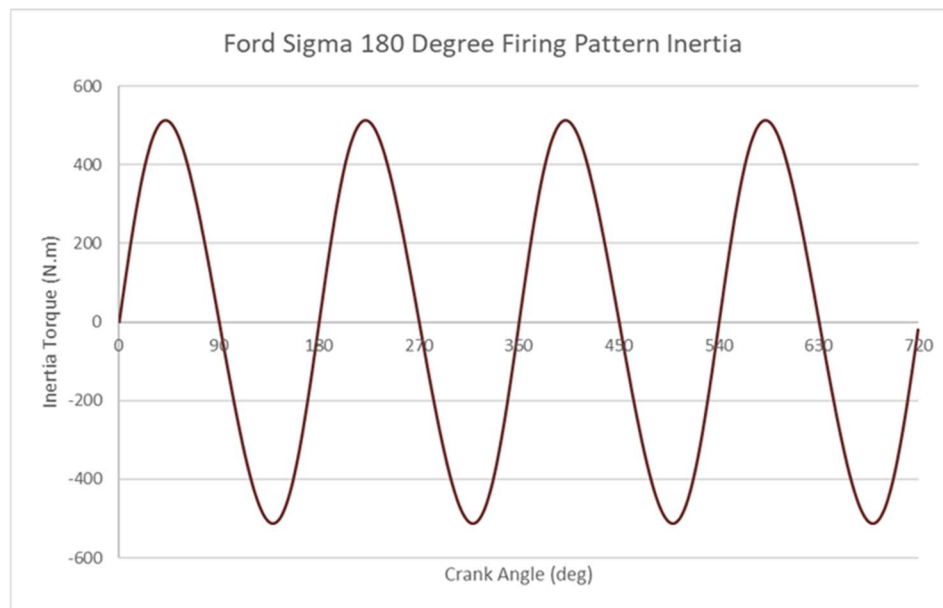
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Appendix A – Results

OEM Specifications Results															Modified
	RPM	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000	
Max Compressive Force (N)		22024.49	27582.74	31419.29	34256.239	35818.361	36796.492	37466.402	37771.106	36730.430	37207.827	38720.797	39282.822	39644.529	8000
Max Tensile Load (N)		198.170	52.203	46.313	303.285	792.311	1347.532	2051.591	2818.634	3696.830	4658.722	5752.598	6257.176	7027.661	9866.401
Max Compressive Stress (N/mm^2)		112.946	141.450	161.125	175.673	183.684	188.700	192.135	193.698	188.361	190.809	198.568	201.450	198.177	292.910
Max Tensile Stress (N/mm^2)		1.016	0.268	0.238	1.555	4.063	6.910	10.521	14.455	18.958	23.891	29.501	32.088	36.039	64.068
Compressive Safety Factor		6.640	5.302	4.655	4.269	4.083	3.975	3.903	3.872	3.982	3.931	3.777	3.723	3.784	2.561
Tensile Safety Factor		738.002	2801.557	3157.875	482.220	184.587	108.532	71.286	51.887	39.561	31.393	25.423	23.373	20.811	11.706
Max Compressive Distance (mm)		0.089	0.111	0.127	0.138	0.145	0.149	0.151	0.153	0.148	0.150	0.157	0.159	0.156	0.231
Max Tensile stretch (mm)		0.001	0.000	0.000	0.001	0.003	0.005	0.008	0.011	0.015	0.019	0.023	0.025	0.028	0.051
Max Compressive Strain		0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.002
Max Tensile Strain		0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
Stiffness (k)		2.47E+05	2.47E+05	2.47E+05	2.47E+05	2.47E+05	2.47E+05	2.47E+05	2.47E+05	2.47E+05	2.47E+05	2.47E+05	2.47E+05	2.47E+05	1.95E+05
Indicited Torque (N.m)		139.302	199.099	215.523	223.423	234.842	244.070	243.267	241.970	242.260	240.462	234.512	229.236	220.505	195.015
Instantaneous Peak Torque (N.m)		558.869	774.602	831.205	851.849	894.554	932.724	934.859	948.001	970.312	992.454	978.353	961.830	933.376	933.488
Imep (Bar)		11.244	16.102	17.421	18.057	19.001	19.778	19.747	19.684	19.765	19.672	19.243	18.839	18.176	16.260
Fmep (Bar)		0.687	0.970	1.033	1.067	1.133	1.210	1.248	1.310	1.392	1.464	1.503	1.505	1.506	1.655
Bmep (Bar)		10.558	15.132	16.388	16.990	17.868	18.568	18.498	18.373	18.373	18.208	17.740	17.335	16.669	14.605
Mechanical Efficiency (%)		0.939	0.940	0.941	0.941	0.940	0.939	0.937	0.933	0.930	0.926	0.922	0.920	0.917	0.898

Titanium Ti4 v6AL Results @ 8000 RPM		
<i>UTS</i>	950.00	N/mm ²
<i>Youngs Modulus E</i>	110000.00	N/mm ²
<i>Stiffness (K)</i>	126417.91	N/mm
<i>Compressive safety factor</i>	3.24	-
<i>Tensile safety factor</i>	14.83	-
<i>Max Compressive distance</i>	0.357	mm
<i>Max Tensile stretch</i>	0.078	mm
<i>Max Compressive Strain</i>	2.66E-03	-
<i>Max Tensile Strain</i>	5.82E-04	-



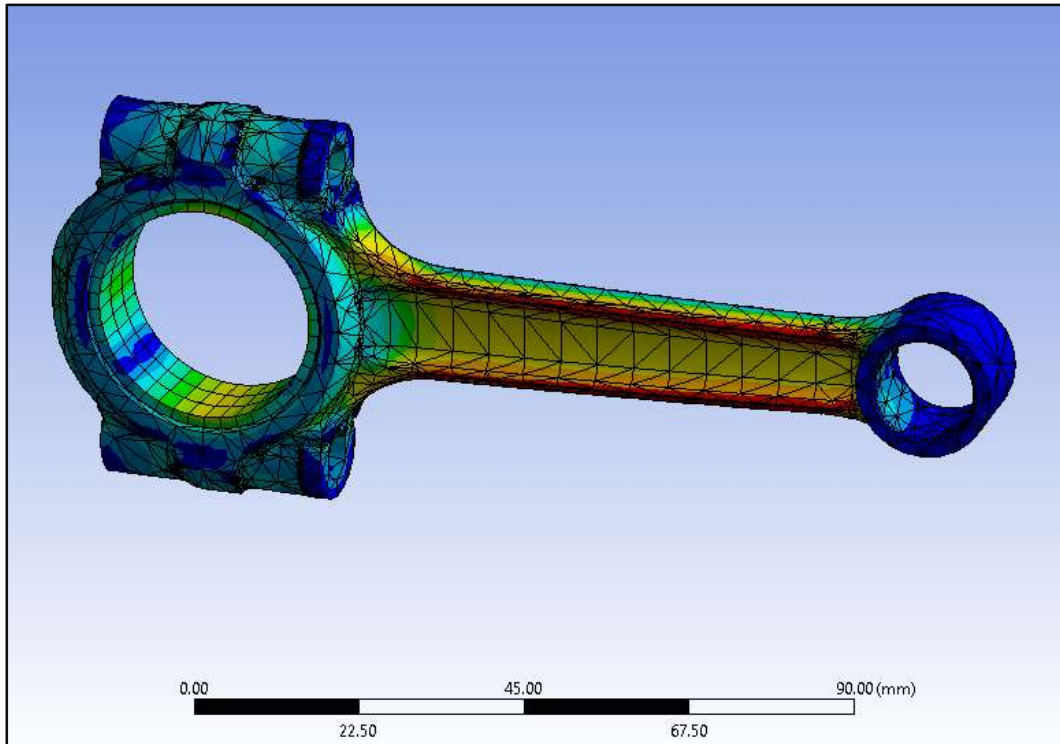


Figure 5: OEM Rod

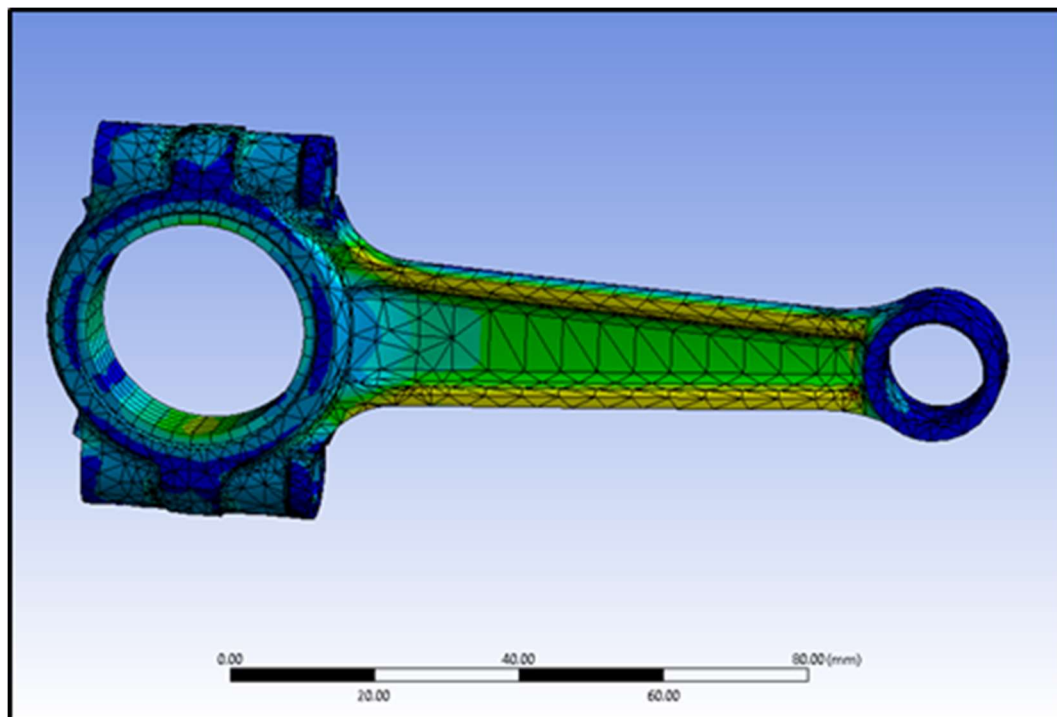
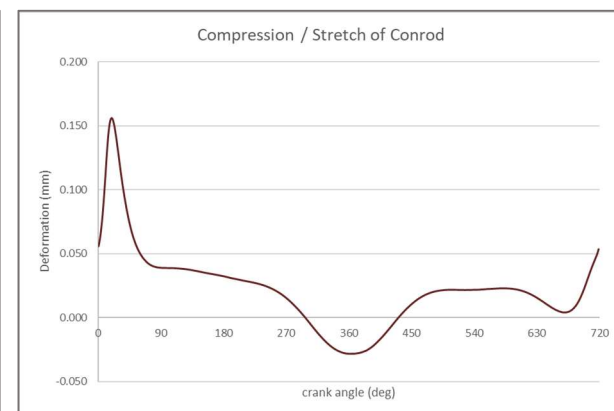
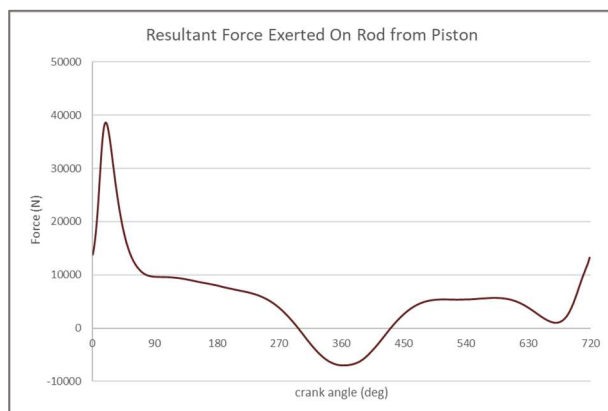
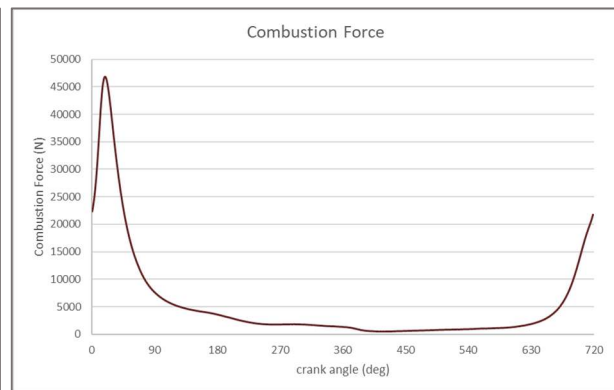
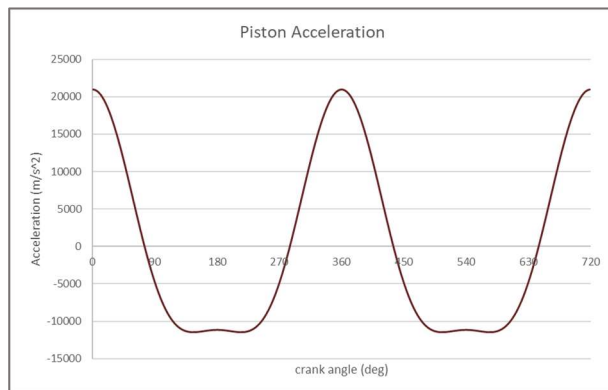
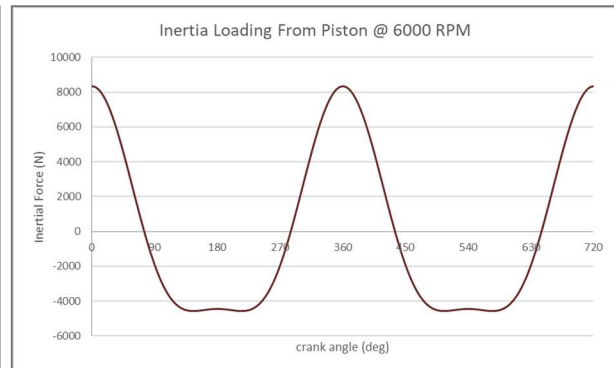
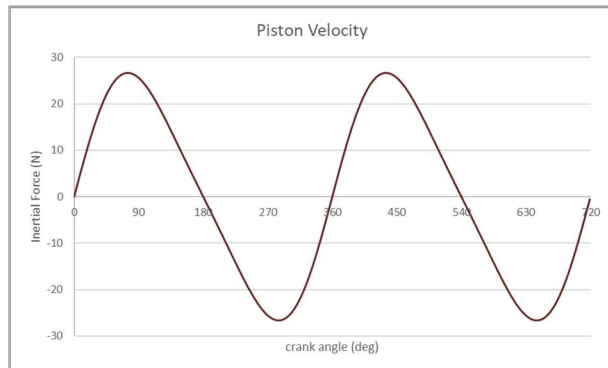


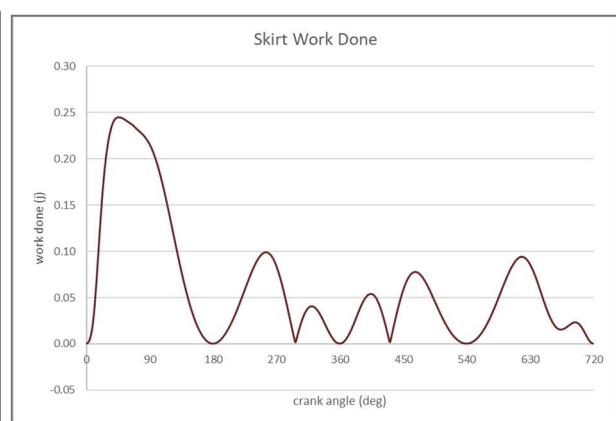
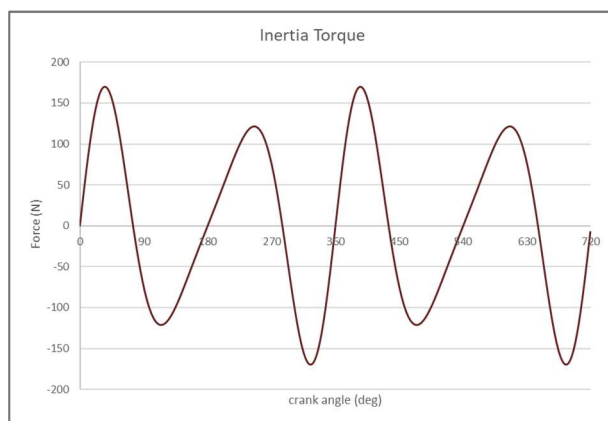
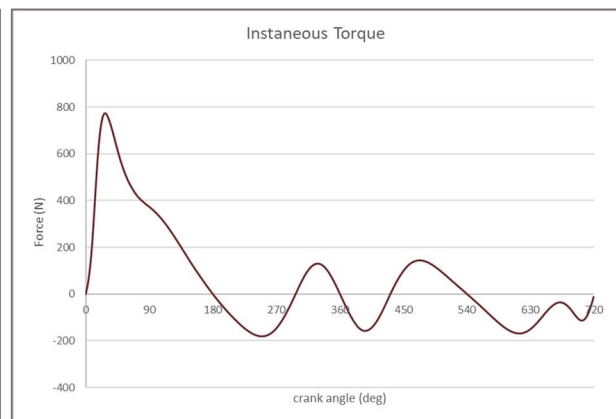
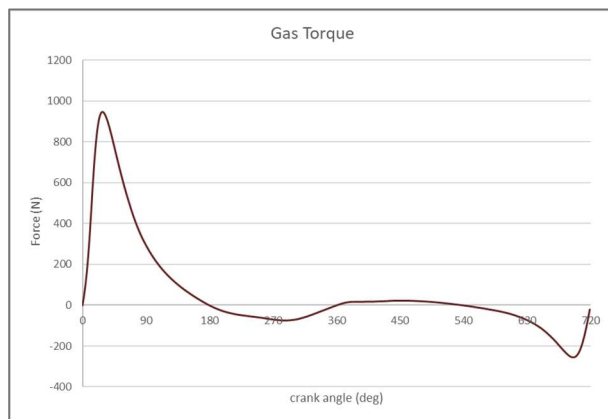
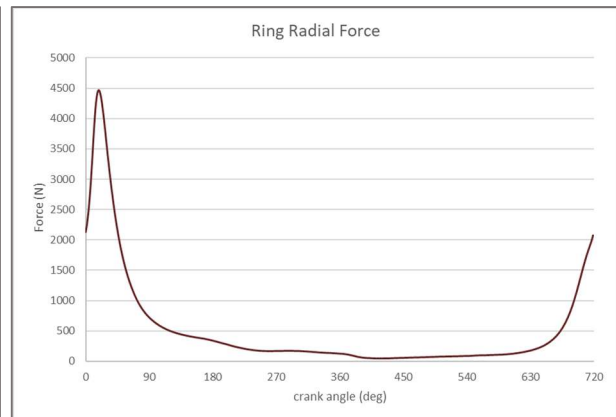
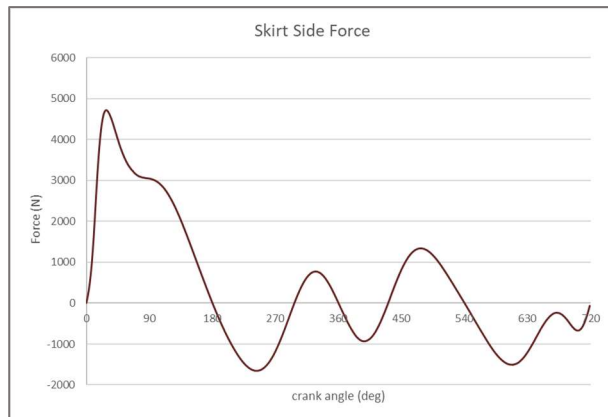
Figure 6: Titanium Rod, Larger Cross Section

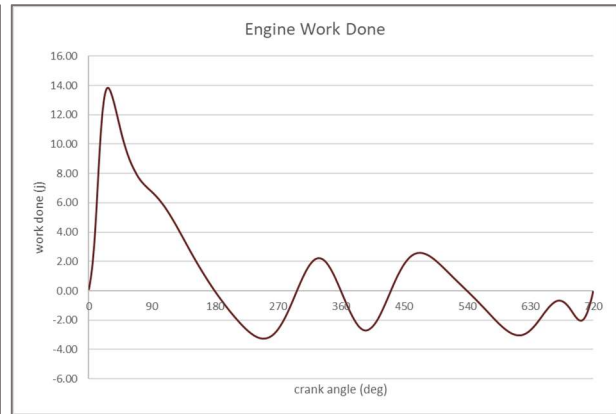
Appendix B – Connecting Rod Analysis, Production Engine

The following graphs and input values were generated for validation / proof of method for the excel workbook calculations, specifically these are samples for 6000 RPM:

Friction Variables		
<i>Coefficient of Friction</i>	0.1	-
<i>Diameter of Rings</i>	0.0790	m
<i>Height of Rings</i>	0.0019	m



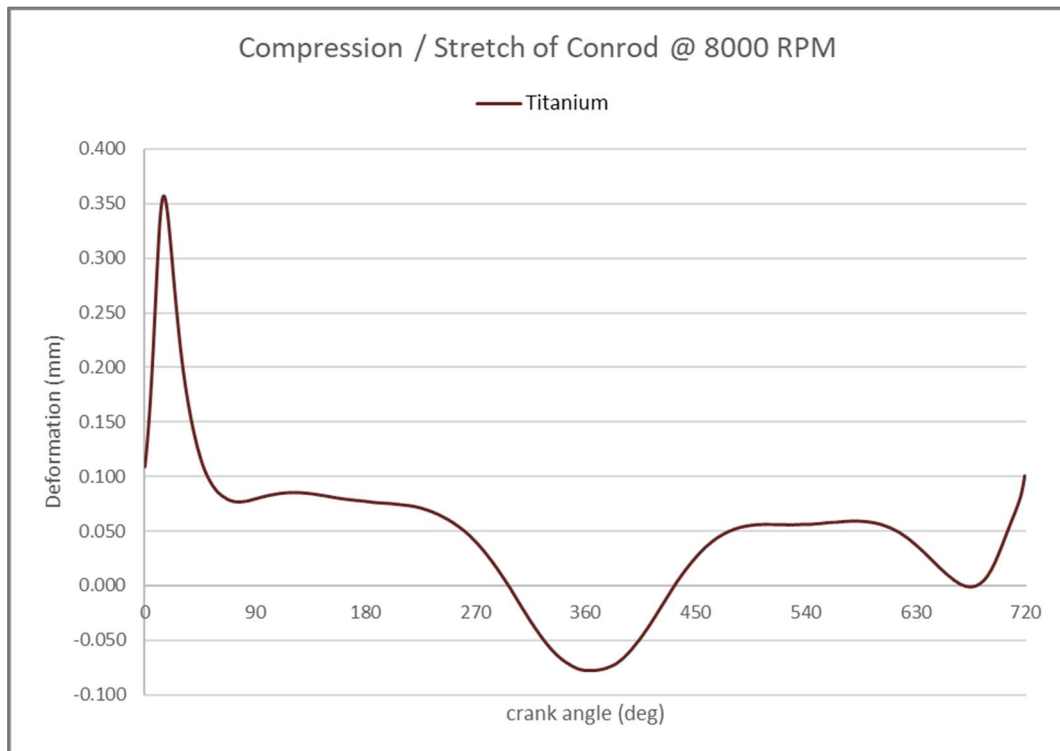




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Appendix C – Connecting Rod Analysis, Modified Engine



Appendix D – Other Supporting Data

