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Investigation and Analysis of Conventional Piston and Wankel Rotary Internal Combustion Engine Efficiency

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Major Project: M6X00417

Declaration of Authenticity:

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

Signed:.....Date:.....

Abstract:

The following Major Project paper will focus on efficiency comparisons between Conventional Piston Engines and the Wankel Rotary Engine.

Comparisons will be established via the use of two Ford piston engines, the Ford 1.6 Sigma and the Ford 1.6 Turbocharged Fox engine. These engines will be placed under comparison with the 13b-MSP Renesis Wankel Rotary engine extracted from a Mazda Rx8 192 model.

Several Combustion Calculations will be produced within this paper. Creation of Solid Works models of vital engine components will be produced and placed under Finite Element Analysis via the use of numerical values established from the combustion calculations. Consequently, enabling further analysis of engine components.

Furthermore; thermal calculations are produced to promote acknowledgment of thermal activity within an engine, dependant on material.

Subsequently assisting further understanding of complex calculations. As well as providing a physical understanding of engine processes and vital components within an engine via disassembly.

Consequently, through application of all methods mentioned, a detailed understanding of efficiency comparisons will be established.

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Chapter 2.0 - Introduction:

The unique design of the Wankel Rotary Internal Combustion Engine (ICE) and the benefits the engine offers consequently produced an investigation to fully understand the reasons the conventional piston engine is perceived as superior.

The alternative design has been undertaken by several manufacturers and used in several automotive applications, including motorcycles, automobiles, aircraft and most recently range extenders in hybrid vehicles.

Mazda Motor Corporation are the only manufacturer who have successfully created several mass-produced production models with the Wankel Rotary engine platform. Production began with the Mazda Cosmo L10A in 1967 and production ceased with the Mazda Rx8 R3 in 2012 due to being unable to meet tightened Euro 6 emission regulations, to be introduced in 2014. (The AA, 2017)

Mazda aims to overcome these limitations with the next-generation Mazda Rx-Vision Concept using Skyactiv-R technology, which was revealed at the 44th Tokyo Motor Show in 2015. (EVO, 2017)

Consequently, an investigation into these limitations of the Wankel Rotary engine will be implemented and an analysis using data acquired against the conventional piston engine will be completed.

Therefore, the Ford Sigma 1.6 Naturally Aspirated (Otherwise known as the Ford Duratec) and Ford Fox 1.6 Turbocharged (Otherwise known as the Ford EcoBoost) Conventional Piston engines were selected to perform the comparison. Finite Element Analysis (FEA) will be used in tandem with Solid Works 2016 to produce engine components and visually display component deformation dependant on material. Microsoft Excel will be used to investigate specific variations in results which will occur over the revolution range of the engines.

Throughout this Major Project, results will be obtained and comparisons will be established. Consequently, enabling advantages and disadvantages of each individual engine to be attained and documented.

2.0.1 – Chapter References:

1. The AA 2017, Limits to Improve Air Quality and Health, Viewed 26th March 2017, <https://www.theaa.com/driving-advice/fuels-environment/euro-emissions-standards>
2. EVO 2015, Mazda Rx-Vision, Viewed 26th March 2017, <http://www.evo.co.uk/mazda/16734/mazda-rx-vision-new-pics-of-stunning-rotary-engined-concept>

2.0.2 – Chapter Literature Review:

1. The AA – Obtained Euro 6 Emissions Regulations.
2. EVO – Information gained upon where Skyactiv-R technology was revealed.

Chapter 3.0 – Aims and Objectives:

3.0.1 – Aims:

The Major Project aims to investigate efficiency of various Internal Combustion Engines. Consequently, the Wankel Rotary ICE was selected and compared to a Naturally Aspirated and Turbocharged Conventional Piston ICE.

Inefficiencies and limitations of the Wankel Rotary Engine compared to the Conventional Piston Engine will be produced via Thermal and Mechanical Efficiency calculations. Subsequently Advantages and Disadvantages of the Wankel Rotary Engine are created and an investigation into methods to overcome these limitations will be produced.

3.0.2 – Objectives:

The Aims listed above will be achieved via the use of Solid Works 2016, to produce detailed models of vital engine components which are required to be placed under FEA. Beforehand vital engine components are required to be measured accurately using Digital Vernier Callipers.

Microsoft Excel is essential to produce spreadsheets of technical data acquired on each engine. Consequently, enabling multiple graphs to be produced on each engine platform and comparisons to be formed.

Chapter 4.0 – Research:

4.0.1 – Internal Combustion Engine Specification:

Three engines were selected to enable the comparison of engine efficiency to be obtained. The Wankel Rotary Engine was selected due to essentially being a topic of high interest to the author and thus enabling a high level of motivation to be maintained throughout the year of study, to further the authors understanding of this engine platform.

The Ford Sigma and Fox engines were selected as they were both easily accessible from the University's School of Engineering. Consequently, enabling a vast yield of data to be obtained and used in comparison to the Wankel Rotary Engine.

4.0.1.1 – Mazda Rx8 13b-MSP (Multi-Side Port) Renesis Naturally Aspirated, Wankel Rotary:



Power = 192PS
Torque = 220Nm
Displacement = 1308cc
Cylinders = Twin Rotors
CO₂ = 267 g/Km
(Parkers, 2017)

Figure 1.0 – Mazda Rx8 13b Renesis Engine (Jay, 2016)

4.0.1.2 – Ford Sigma 1.6 Naturally Aspirated:



Power = 115BHP
Torque = 155Nm @ 4150 RPM
Displacement = 1596cc
Cylinders = 4
(Work Shop Manuals, 2017)
CO₂ = 164g/Km (Energy EU, 2013)

Figure 1.0.1 – Ford Sigma 1.6 (In Auto News, 2012)

4.0.1.3 – Ford EcoBoost 1.6 Turbocharged:



Power = 178BHP

Torque = 253.1Nm @ 4002 RPM

Displacement = 1600cc

Cylinders = 4

(Malcolm McDonald, 2016)

CO₂ = 137g/Km (Next Green Car, 2017)

Figure 1.0.2 – Ford 1.6 EcoBoost (Western Slope Auto, 2013)

4.0.2 – Contact Information:

Unlike for the data required for the conventional pistons engines; a lack of available information within the university for the Wankel Rotary Engine required the author to search elsewhere for the information. Consequently; several companies and individuals were approached to obtain the information required.

Companies Contacted:

- Ben Dunn of Rotary Revs – Rotary Engine Rebuild Specialists
- John Hickman of De Verde Hydrogen – Hydrogen Hybrid Conversions.
- Lewis Corps of LC Rotary – Rotary Engine Rebuild Specialists.
- Alistair Hackwood of Mazda Car Dealership– Clevedon England.
- Mazda Motor Corporation – Head Office (Japan).
- Mazda Motor Corporation – Head Office (UK).
- Chris Terry – Vapour Blasting Specialists.

Individuals Contacted:

- Barry Bordes – Rotary Car Club USA.
- Dave Cooper – Technician UWTSD.
- Lewis Grey – eBay.
- Malcolm McDonald – UWTSD Lecturer.
- Peter Hamilton – Rx8 Owners Club UK.
- Tim Tudor – UWTSD Lecturer.

As listed above, several companies and individuals were contacted to obtain information. Firstly, De Verde Hydrogen were contacted prior to the decision to investigate efficiency of Conventional Piston Engines compared to the Wankel Rotary Engine. Prior to establishing the route of this Major Project, Hydrogen fuel cells were investigated.

Once the route of the Major Project was established Malcolm McDonald was approached and Ford Sigma and Ford Fox engine data was obtained. However, Wankel Rotary engine data was limited at the university. Consequently, the author was required to independently source the essential engine data.

Therefore, several Rotary Engine Rebuild Specialists including LC Rotary and Rotary Revs were contacted. However, both were unable to provide the data required. Consequently, Alistair Hackwood of Clevedon Mazda Dealership was approached, who transferred communications to Mazda Motor Corporation UK. Limited information of the Wankel Rotary Engine at the UK headquarters inevitably required Mazda Motor Corporation Japan to be contacted, due to being the home of research and development (R&D) of the Wankel Rotary Engine.

However further issues transpired, due to Mazda's Technical Manager advising, "Due to the Rotary Engine being solely for Mazda, the information requested may be classed as sensitive which is classified only for our factory. However, I can confirm we have approached Japan and asked for this to be released on this occasion to help with the dissertation" (Mazda Motor Corporation, 2017)

Further correspondents with Mazda abruptly ended with a recommendation to contact Rotary Engine Rebuild Specialists. Which had previously been pursued and been unsuccessful. Therefore, further alternative methods of obtaining Wankel Rotary data began. Consequently; Barry Bordes of the Rotary Car Club USA and Peter Hamilton of the Rx8 Owners Club UK were sourced and contacted. Thankfully these individuals could supply the Wankel Rotary Engine data and recommendations to useful books to further knowledge of the Wankel Rotary ICE's. Consequently, a Mazda Rx8 192PS 13b Renesis Wankel Rotary Engine was sourced from Lewis Grey via the Vehicle Parts and Accessories category on eBay, to enable engine disassembly and physical measurements to be obtained to produce Solid Work Models and eventually Finite Element Analysis.

4.0.2.1 – Data Obtained:

The data obtained in Figure 2.0 was acquired via Peter Hamilton of the Rx8 Owners Club UK. The data included Volumes for 270° of eccentric shaft rotation.

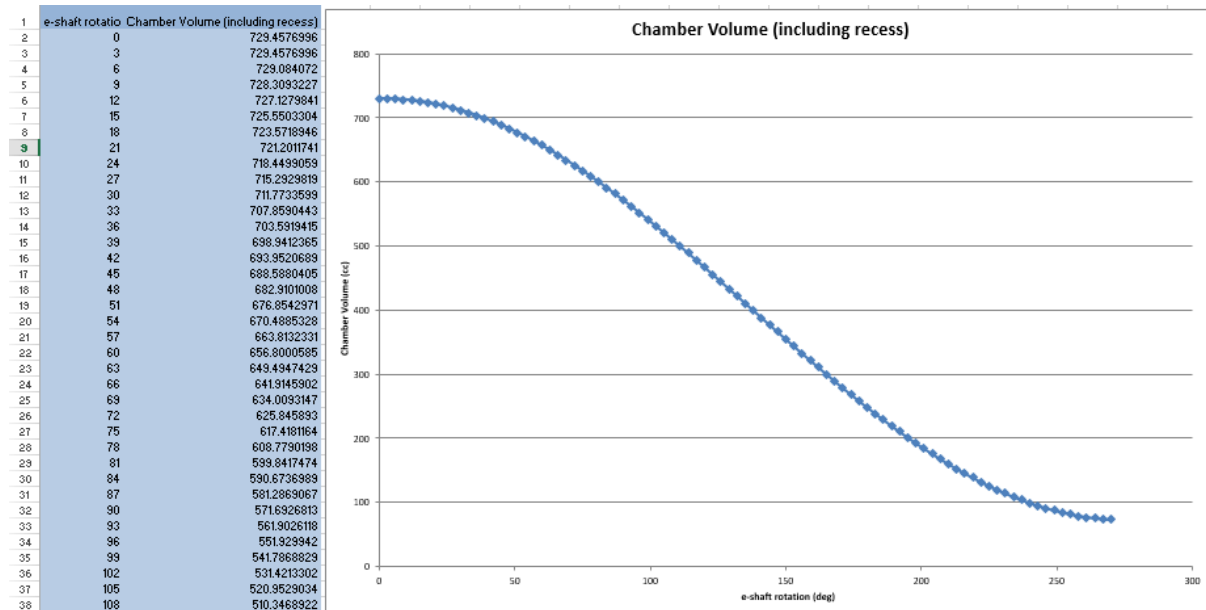


Figure 2.0 – Volumes Obtained.

The following data was obtained via the Kenichi Yamamoto Rotary Engine book and displayed the Cylinder Pressure in Figure 2.0.1 and the Burn Rates in Figure 2.0.2.

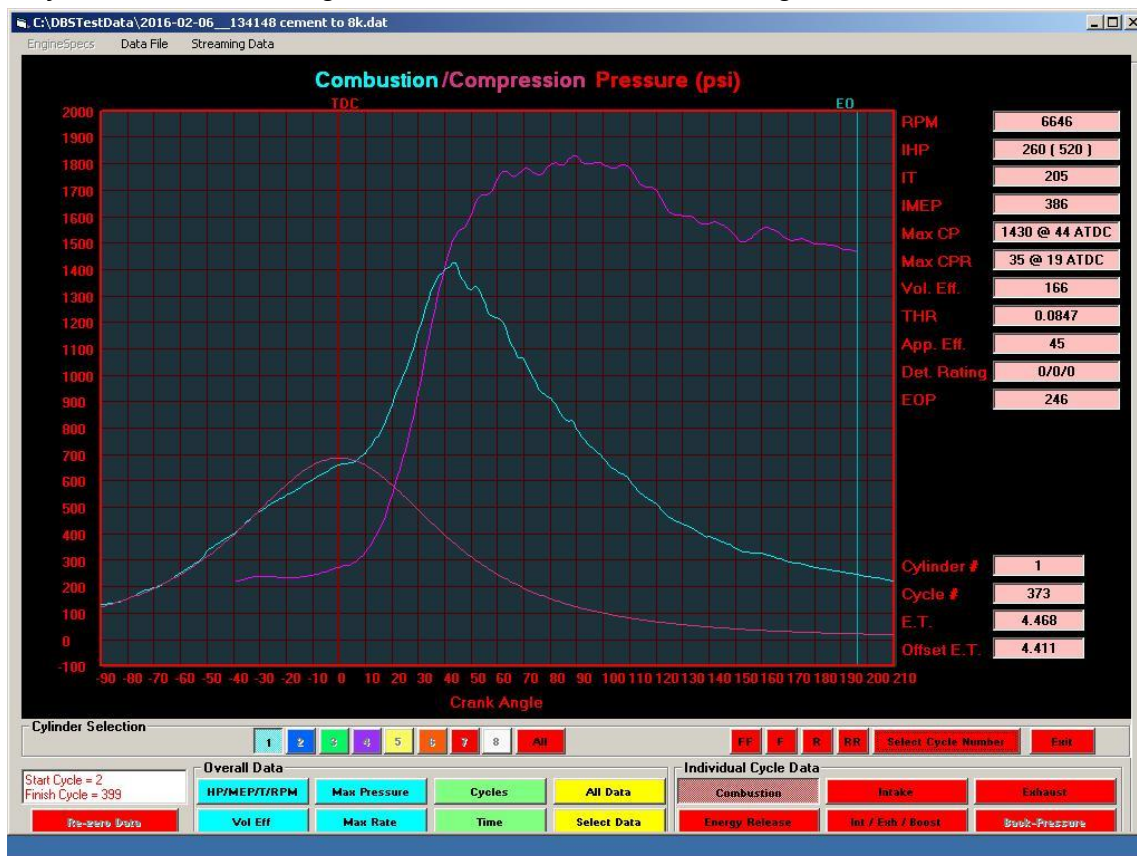


Figure 2.0.1 – Kenichi Yamamoto Cylinder Pressure. (Yamamoto, 1981)

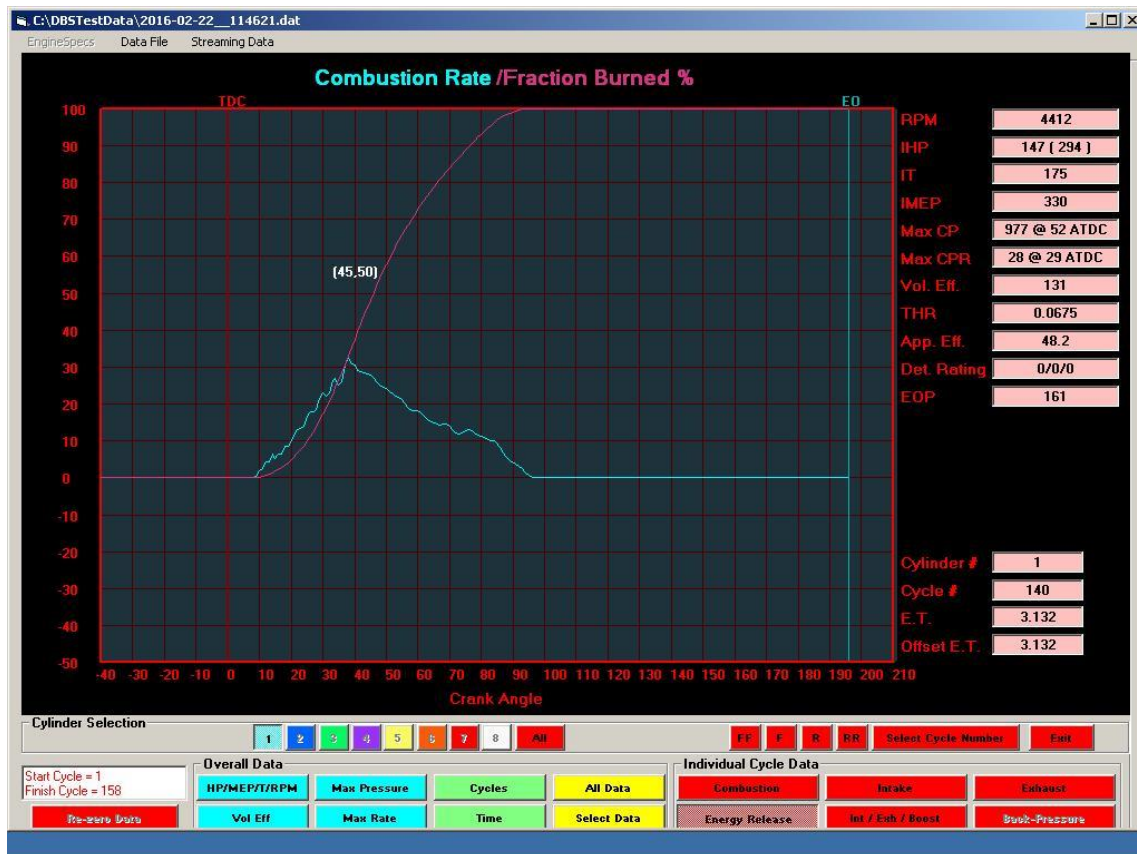


Figure 2.0.2 – Kenichi Yamamoto Burn Rates. (Yamamoto,1981)

4.0.3 – Obtaining Engine Components and Vapour Blasting:

As previously mentioned in Chapter 4.0.2, the Wankel Rotary engine was sourced and disassembled. Full description of engine disassembly is provided in Chapter 7.0.1. Ford Sigma and Fox engine components required were readily available from the School of Engineering and obtained via lecturer Malcolm McDonald.

However once Wankel Rotary engine disassembly was complete; the engine was sent off to Chris Terry Vapour Blasting Specialists. Where the engine transformed from the condition in Figure 3.0 to the presentable condition of Figure 3.0.1 for the first interim VIVA.

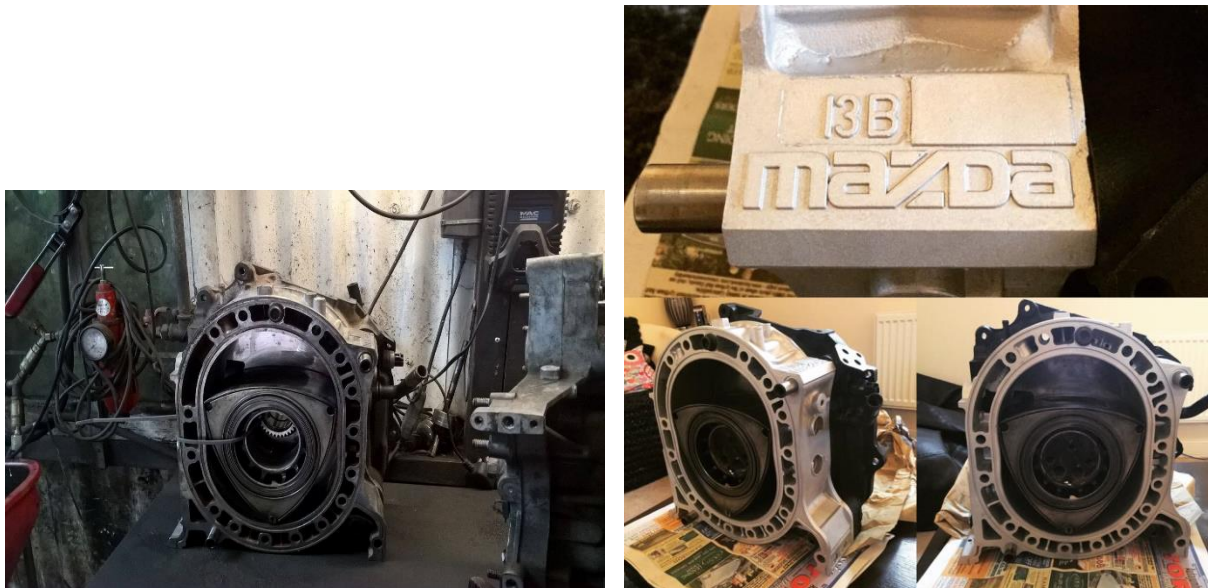


Figure 3.0 – Prior Vapour Blasting. (Jay, 2016) Figure 3.0.1 – Post Vapour Blasting (Jay, 2016)

Vapour Blasting, also known as Aqua Blasting is used as an alternative to Sand Blasting and other similar methods, due to providing a less aggressive process via the flow of the water borne abrasive and consequently obtaining a finer finish to the engine. (Vixen, 2016)

4.0.4 – Ignition:

The Major Project will investigate ignition systems via Mass Fraction Burn Rates. Consequently, the dual spark ignition of the Mazda Rx8 13b Renesis engine was investigated.

4.0.4.1 – Dual Spark Ignition (DSI):

DSI systems have been used in production vehicles, as well as in racing application. Multiple vehicle manufacturers have used this method of spark plug formation, including Alfa Romeo with their Twin-Spark (T-Spark) ignition system.

The Twin or Dual Spark Ignition System increases the likelihood of ignition and consequently reduces the total burn time. This is due to an increase flame front, which is the area in which the combustion reaction occurs.

DSI's are predominantly required with engines which have a large bore, to increase the probability of ignition. An increase in power can be achieved with the use of DSI, due to an increase in burn rate of the gas. Therefore, ignition can be retarded and thus reduce force required to move the piston. Consequently, improving the pumping efficiency of the engine.

(JCM Machine & Coatings, 2017)

The Wankel Rotary engine required a DSI system, due to the length of the combustion chamber. A single spark plug would be unable to ignite all the air and fuel mixture and consequently reduce efficiency.

4.0.5 – Rotary Engine History:

Prior to the Wankel Rotary Engine, several Rotary Engine platforms have been developed. These developments transpired from the Rotary piston type water pump. Several diverse Rotary engine methods are displayed below.

Production of the Ramelli's Pump occurred in 1588, displayed in *Figure 4.0*. The principle of this method was for use as a water pump.

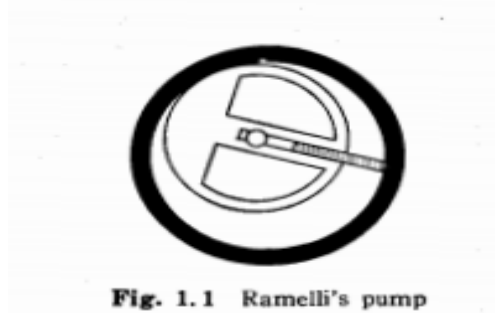


Figure 4.0 – Ramelli's Pump. (Yamamoto, 1981)

Murdoch's Rotary Steam engine was a reciprocating External Combustion Engine (ECE) produced in 1799, presented in *Figure 4.0.1*.

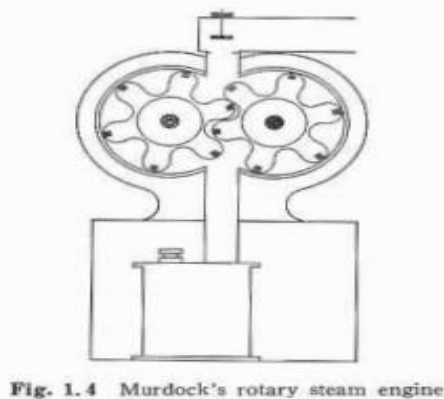


Figure 4.0.1 – Murdoch's Rotary Steam Engine. (Yamamoto, 1981)

Cooley's Rotary engine produced the first Rotary engine with visual similarities to the Wankel Rotary engine of the present. This was also a steam engine and production began in 1901. Displayed below in *Figure 4.0.2*.

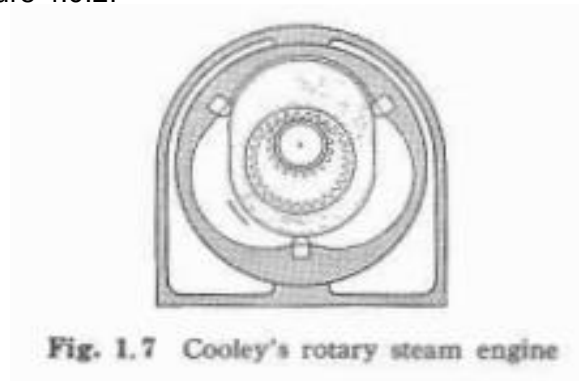


Figure 4.0.2 – Cooley's Rotary Steam Engine. (Yamamoto, 1981)

The final Rotary method produced prior to the Wankel Rotary engine was the Maillard's Rotary Compressor, produced in 1943. Where vast similarities between the Wankel Rotary engine occurred. This compressor was derived from the application of the rotary theory and is presented in *Figure 4.0.3*.

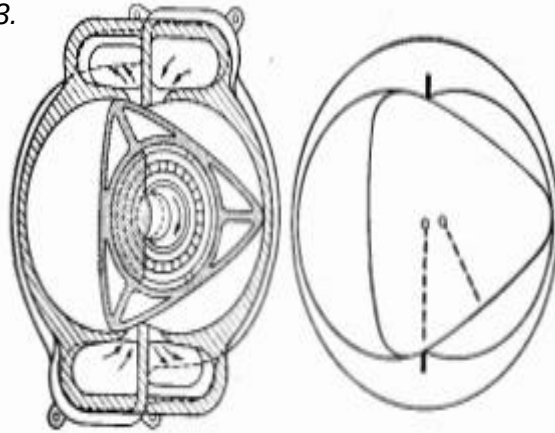


Figure 4.0.3 – Maillard's Rotary Compressor. (Yamamoto, 1981)

4.0.5.1 – Wankel Rotary Engine History:

Wankel approached NSU (Neckarsulm Strickmaschinen Union), who were currently producing motorcycles. After communications began; the men who were responsible for acquiring Wankel to NSU, Dr. Victor Frankenberger and Dr. Walter Froede visited Wankel and were shown multiple designs and reports Wankel had produced. Consequently, Froede was highly impressed with Wankel's work and shortly after in 1951 a Wankel Rotary engine was installed on an NSU motorcycle as a supercharger, otherwise known as a compressor.

NSU applied the Wankel Rotary compressor to the 500cc two cylinder and could produce 120 horsepower. The NSU Quickly 50cc motorcycle set a speed record of 120mph at the Bonneville Salt Flats via Wankel Rotary compressor application.

(Hege, 2001)

In 1959 a 250cc single rotor NSU-Wankel type rotary engine succeeded durability test. Consequently, Mazda heavily invested R&D into the Wankel Rotary engine and in 1963 a concept of the Mazda Cosmo Sports transpired. The Cosmo was a 400CC Twin-Rotor design and was displayed at Tokyo Motor Show for the first time.

NSU rivalled Mazda with the NSU Spider, which became the first ever rotary engine production vehicle on the market in 1964. Subsequently Mazda responded with production of their Cosmo Sports concept in 1967. This occurred the same year NSU produced the Ro80, until production ceased in 1977.

The NSU Ro80 caused the collapse of the company due to highly unreliable Wankel Rotary engines failing after only 15,000 miles, due to Apex seal failure. Consequently, NSU replaced hundreds of engines under warranty and were eventually bought by Volkswagen (VW) and eventually formed Audi Motor Corporation.

(Motoring Research, 2015)

Therefore, Mazda Motor Corporation were left unrivalled and could produce a range of production vehicles. Consequently, the Mazda Rx7 FB was fashioned in 1978, the second-generation Mazda Rx7 FC in 1985 and finally the third-generation Mazda Rx7 FD was produced in 1991. The same year Mazda entered the rotary engine into and won the 24 Hours of Le Mans. Thus, becoming the first Japanese vehicle manufacturer to win this event.

(Yamamoto, 1981)

Finally, the Mazda Rx8 was manufactured in 2003, until production ceased in 2011 due to failing current Euro market emissions standards.

4.0.6 – Wankel Rotary Engine Advantages:

The major benefit to the Wankel Rotary engine is the simplistic design. Only three moving parts; two rotors and an eccentric shaft are required. Consequently, Noise Vibration and Harshness (NVH) is dramatically reduced compared to a conventional piston engine.

Many car enthusiasts are intrigued by the high revving nature of the engine. A 9,000 RPM redline is achieved from the factory in the Mazda Rx8 231 model. However, with modification a maximum of 10,600RPM is achievable before internal components require upgrading.

The compact and lightweight design of the engine also enables 50:50 weight distribution to be obtained with ease. Further benefit to the Wankel Rotary engine is it is immune to catastrophic failure, unlike conventional piston engines.

(National Speed, 2009)

4.0.6.1 – Wankel Rotary Engine Disadvantages:

The Wankel Rotary engine also has several disadvantages. As previously mentioned in Chapter 4.0.6, Wankel Rotary Engines are immune to catastrophic failure. However, compression loss is a common failure, where the vehicle will produce a limited amount of power until the engine eventually dies.

Further disadvantages are produced with the requirement of high maintenance and high use of oil, especially in later Series 2 model Rx8 produced from 2008, which acquired a third oil injector into the combustion chamber to reduce high friction occurrence at the centre of the Apex seals. Experienced rotary engine enthusiasts found oil changes were required every 3000 miles to encourage a healthy engine, which was not practical for daily drivers. (Rx8 Club, 2008)

Low Miles per Gallon (MPG) figures of approximately 16 to 22 MPG for the 1.3 Litre engine were viewed as appalling. Consequently, the low torque figure and flooding which occurred if the engine was turned off before reaching adequate temperature caused further problems.

(National Speed, 2009)

4.0.7 – Future of the Wankel Rotary Engine:

Conventional piston engines aren't the only ICE of the future, multiple engine designs are still produced to this day, such as the MAHLE Powertrain partnership of Duke Engine technology. (Duke Engines, 2017)

Consequently, Mazda did not admit defeat with their final production rotary engine vehicle and unveiled the Mazda Rx-Vision concept with SkyActiv-R Technology in the 2015 Tokyo Motor Show, pictured in *Figure 5.0*. (Inside Mazda, 2017)



Figure 5.0 – Mazda Rx-Vision Concept. (Inside Mazda, 2017)

Mazda have recently filed an application for a patent for the new rotary engine design, displayed in *Figure 5.0.1*. Consequently, producing hope for the return of a more powerful, reliable and emissions friendly rotary engine. However, rumours of the engine being used as a generator for an electric motor have surfaced.

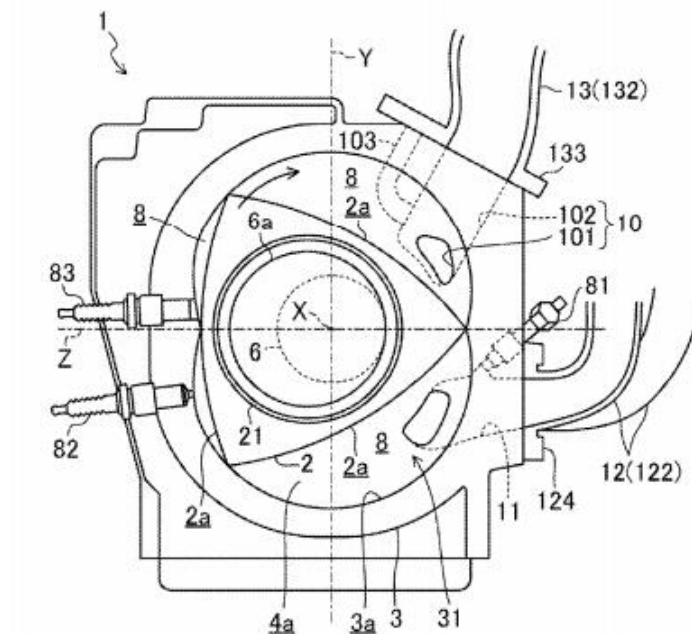


FIG. 4

Figure 5.0.1 – Mazda Patent for Rotary Piston Engine Mounted on Vehicle. (United States Patent and Trademark Office, 2016)

4.0.8 – Engine Materials:

Several material properties were investigated for use within calculations and Solid Works models at a later date.

Ultimate Tensile Strength (UTS):

- Plain Carbon Steel → 440MPa (AXOM Materials, 2012)
- Aluminium 6061 → 310MPa (ASM, 2017)
- Cast Carbon Steel → 414MPa (Total Materia, 2003)

4.0.9 – Chapter References:

1. ASM 2017, Aluminium 6061-T6, Viewed 5th April 2017, <http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=ma6061t6>
2. AZOM Materials 2012, AISI 1018 Mild/Low Carbon Steel, Viewed 5th April 2017, <http://www.azom.com/article.aspx?ArticleID=6115>
3. Duke Engines 2017, Technology Overview, Viewed 21st April 2017, <http://www.dukeengines.com/technology/overview/>
4. Energy EU 2013, CO₂ Emissions, Viewed 7th April 2017, <https://www.energy.eu/car-co2-emissions/ford.php>
5. Hege, 2002, Engineering, The Wankel Rotary Engine – A History, McFarland & Company Inc Publishers, North California.
6. In Auto News 2012, Ford Begins to export its 1.6 Sigma to Europe, viewed 29th March, <http://www.inautonews.com/ford-begins-to-export-its-16-sigma-engine-to-europe>
7. Inside Mazda 2017, Mazda Rx-Vision Rotary Sports Car Concept, Viewed 14th March 2017, <https://insidemazda.mazdausa.com/mazda-news/mazda-new-sports-car-concept/>
8. JCM Machine & Coatings 2017, Engine Combustion Dynamics – Normal Combustion, Viewed 9th April 2017, <https://www.jcmmachineandcoatings.com/understanding-combustion/>
9. Motoring Research 2015, NSU Ro80, Viewed 21st April 2017, <https://www.motoringresearch.com/car-news/features/nsu-ro80-the-car-of-the-year-that-killed-the-company>
10. National Speed 2009, Advantages and Disadvantages of a Rotary Engine, Viewed 11th December 2016, <http://nationalspeedinc.com/advantages-and-disadvantages-of-a-rotary-engine/>
11. Next Green Car 2017, Ford Focus 1.6T Zetec S EcoBoost 180PS, Viewed 7th April 2017, <http://www.nextgreencar.com/view-car/36040/ford-focus-1.6t-zetec-s-ecoboost-182ps-petrol-manual-6-speed/>

12. Parkers 2017, Mazda RX-8 (2003 – 2010) Specs & Dimensions, Viewed 7th April 2017, [http://www.parkers.co.uk/mazda/rx-8/coupe-2003/\(192ps\)-4d/specs/](http://www.parkers.co.uk/mazda/rx-8/coupe-2003/(192ps)-4d/specs/)
13. Rx8 Club 2008, Mechanical Changes on Rx8 Series 1 to 2, Viewed 21st April 2017, <http://www.rx8club.com/series-ii-technical-trouble-shooting-160/mechanical-changes-rx-8-series-i-ii-161665/>
14. Total Materia 2003, Cast Carbon Steels, Viewed 5th April 2017, <http://www.totalmateria.com/page.aspx?ID=CheckArticle&site=kts&NM=98>
15. United States Patent and Trademark Office 2016, Rotary Piston Engine Mounted on Vehicle, Viewed 21st April 2017, <http://pdfaiw.uspto.gov/.aiw?PageNum=0&docid=20160084158&IDKey=32A0F77DE744&HomeUrl=http%3A%2F%2Fappft.uspto.gov%2Fnetacgi%2Fnph-Parser%3Fsect1%3DPTO1%2526sect2%3DHITOFF%2526d%3DPG01%2526p%3D1%2526u%3D%25252Fmetahtml%25252FPTO%25252Fsrchnum.html%2526r%3D1%2526f%3DG%2526l%3D50%2526s1%3D%25252220160084158%252522.PGNR.%2526OS%3DDN%2F20160084158%2526RS%3DDN%2F20160084158>
16. Vixen 2016, A Guide to the Vapour Blasting Process, Viewed 16th November 2016, <http://www.vaporblastingequipment.com/wet-blasting/what-wetblasting-guide-vapor-blasting-process>
17. Western Slope Auto 2013, 2014 Ford Fusion Engine Choices, Viewed 7th April, <http://www.westernslopeauto.com/blog/2014-ford-fusion-engine-choices/>
18. Work Shop Manuals 2017, Ford Workshop Manuals, Viewed 7th April 2017, [http://workshop-manuals.com/ford/focus_2004.75_07.2004/mechanical_repairs/3_powertrain/303_engine/303-01b_engine_1.6l_duratec-16v_ti-vct_\(sigma\)/specifications/](http://workshop-manuals.com/ford/focus_2004.75_07.2004/mechanical_repairs/3_powertrain/303_engine/303-01b_engine_1.6l_duratec-16v_ti-vct_(sigma)/specifications/)
19. Yamamoto, 1981, Automotive, Rotary Engine, Toyo Kogyo CO LTD, Japan

4.0.9.1 – Chapter Literature Review:

1. ASM – Obtained material properties of Aluminium.
2. AZOM – Obtained material properties of Plain Carbon Steel.
3. Duke Engines – Used as an example for alternative engines of the future.
4. Energy EU – Obtained Ford CO₂ Emissions.
5. Hege – Obtained Historical information about the Wankel Rotary ICE.
6. In Auto News – Obtained Ford Sigma picture.
7. Inside Mazda – Researched the Mazda Rx-Vision Concept.
8. JCM Machine & Coatings – Investigated Dual Spark Ignition Systems.
9. Motoring Research – Investigated the history of NSU.
10. National Speed – Investigated Advantages and Disadvantages of the Rotary Engine.
11. Next Green Car - Obtained further Ford EcoBoost CO₂ Emissions data.
12. Parkers – Obtained Torque figures of Mazda Rx8 192PS.
13. Rx8 Club – Obtained information on mechanical differences between series one and two model Rx8's.
14. Total Materia – Obtained material properties of Cast Carbon Steel.
15. United States Patent and Trademark Office – Investigated Mazda's new Rotary engine patent application.
16. Vixen – Investigated Vapour Blasting Process.
17. Western Slope Auto – Obtained picture of Ford 1.6 EcoBoost engine.
18. Work Shop Manuals – Obtained Ford Sigma displacement.
19. Yamamoto – Obtained information on the history of the rotary engine.

Chapter 5.0 – Progress Seminar:

The progress seminar enabled the collection of feedback via lecturer Tim Tudor. Consequently, production of the poster in *Figure 6.0* transpired. The poster obtained early Major Project results and investigations.



An Investigation into Efficiencies of the Wankel Rotary Engine

Rupert Jay

Supervisor: Malcolm McDonald

School of Engineering, Manufacturing & Logistics
BEng (Hons) Advanced Engine Design

Introduction

Due to having ownership of a rotary engine vehicle and the motivation to further understand the processes which occur within the rotor housing and why these engines suffer from lack of efficiency, this subject was decided upon.



(Rupert Jay, 2016)

Research:

- Mazda Head Office contacted → Currently awaiting information from **Mazda Japan**.
- Emailed a forum in **America** → Received good information and sources on the Rotary Engine.
- Obtained 2 vital books by **Kenichi Yamamoto** on detailed information on the internal processes of the Rotary Engine.
- Purchased **13b-Renesis** engine to strip-down.
- Obtained a damaged **13b-REW** rotor housing from an Mazda Rx7 to compare efficiency loss to the working order 13b-Renesis from Mazda Rx8.
- Obtained Ford Sigma 1.6 data and produced **Cylinder Pressure graphs** at various RPM to then compare to Cylinder Pressures on Mazda Rx8.

Aims:

1. Investigate reasons for lack of efficiency of the Wankel Rotary Engine
2. Investigate mechanical efficiencies and compare to piston engine
3. Investigate Thermal efficiencies and compare to piston engine

Practical Work:

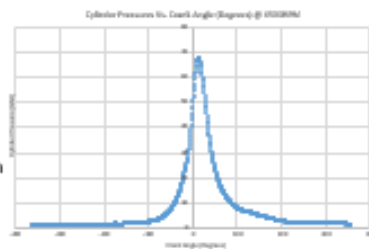


Figure 1.0 – Cylinder Pressure (BAR) Vs. Crank Angle per degree graph @ 6500RPM

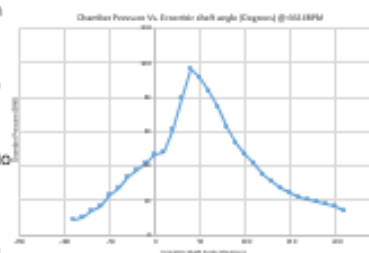


Figure 1.1 – Chamber Pressure (BAR) Vs. Eccentric Shaft Angle per 10 degrees @ 6646RPM.

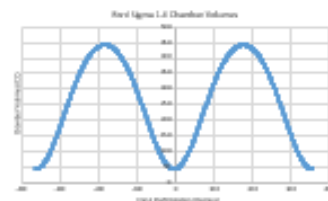


Figure 1.2 – Cylinder Volumes of Ford Sigma 1.6 per degree of crankshaft rotation.

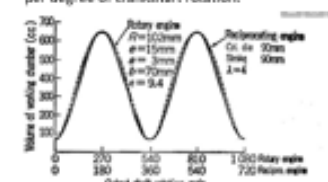


Figure 1.3 – Chamber Volumes of Wankel Engine per degree of eccentric shaft rotation (Kenichi Yamamoto, 1981).

Strip-down of 13b-Renesis Engine:



(Rupert Jay, 2016)

- 13b-Renesis engine was stripped down to show internal components.
- These components were then measured.
- Measurements will then be used to produce a scale model of a Rotary Engine on CAD.

References

- Yamamoto, 1981, Engineering, Rotary Engine, Tokyo Japan.
- Ansdale, 1988, Engine Design and Performance, The Wankel RC Engine, Frome England.

School of Engineering, Manufacturing & Logistics

Figure 6.0 – Progress Seminar Poster.

6.0.1.1 – Ford Sigma Excel for 1000RPM:

Firstly; a Mean Cylinder Pressure vs Crank Angle graph was produced using the data provided, displayed in *Figure 7.0.1*.

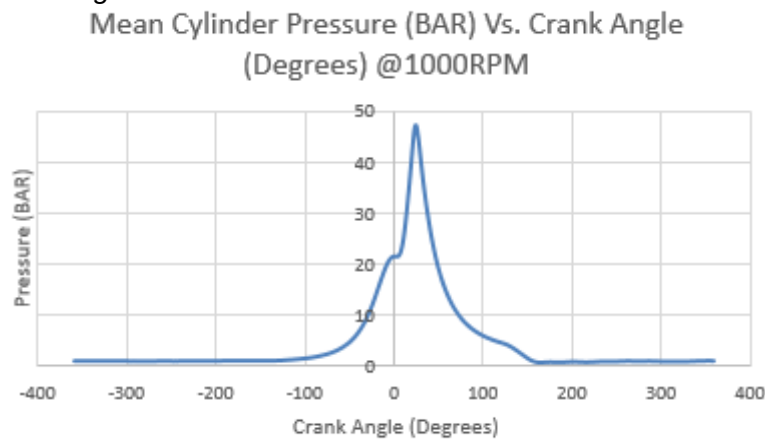


Figure 7.0.1 – Mean Cylinder Pressure (BAR) vs Crank Angle (Degrees).

Secondly; the data provided enabled the creation of a pressure vs volume graph, displayed in *Figure 7.0.2*.

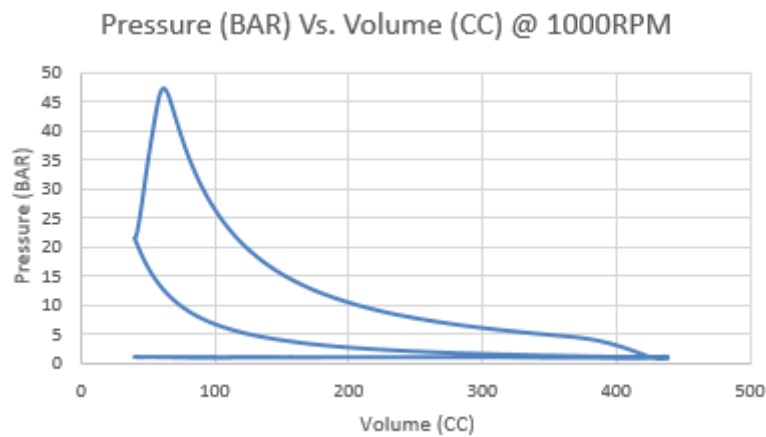


Figure 7.0.2 – Pressure (BAR) vs Volume (CC).

The logarithm of Pressure and Volume was then calculated via the “log equation tool” in Microsoft Excel. Consequently, a logarithm of Pressure and Volume was obtained for each degree of crank angle rotation, from -360 to +359 degrees. The first 10 degrees of calculations at 1000RPM are displayed in *Figure 7.0.3*.

C.A.(DEG)	LogV	LogP
-360	1.594061	0.0086
-359	1.594503	0.007321
-358	1.595827	0.006466
-357	1.598024	0.004321
-356	1.600973	0.002166
-355	1.604874	0.002166
-354	1.609488	0
-353	1.614897	0
-352	1.621072	0
-351	1.627878	-0.0013
-350	1.635484	-0.00218

Figure 7.0.3 – Pressure and Volume Logarithms per Degree of Crank Angle Rotation.

These calculations enabled the creation of a LogP vs LogV graph, presented in *Figure 7.0.4*.

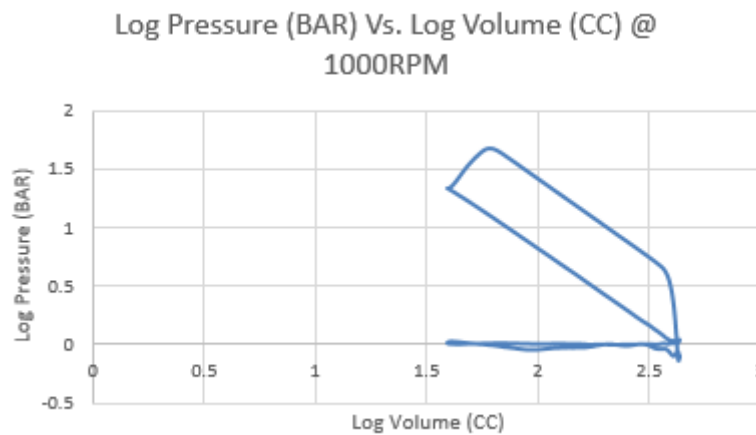


Figure 7.0.4 – LogP (BAR) vs LogV (CC).

The Extra Pressure (BAR) from Combustion (ΔP) was required for calculation. However, the expansion or compression dependant on the engine being Before Top Dead Centre (BTDC) or After Top Dead Centre (ATDC), otherwise known as “n”, was required to be calculated first.

Therefore; the following equation is completed, for each degree of crank angle rotation, using data previously obtained for pressure and volume.

$$n = \frac{(\text{Log}P2 - \text{Log}P1)}{(\text{Log}V1 - \text{Log}V2)}$$

The first 10 degrees of calculations for “n” are displayed in *Figure 7.0.5*.

n
2.893227
0.645833
0.976049
0.730906
0
0.469462
0
0
0.191738
0.114656

Figure 7.0.5 – Calculations for “n” per Degree of Crank Angle Rotation.

Due to “n” differing dependant on the engine being BTDC or ATDC, Top Dead Centre (TDC) was required to be located. This data clearly identified TDC occurring at 0 Degrees Crank Angle, due to being the centre number within the crank angle Microsoft Excel spreadsheet layout.

For the calculation of ΔP , an average of “n Compression” and “n Expansion” was required. Consequently, ten results of approximately 1.3 before and after TDC were selected to produce an average, as presented in *Figure 7.0.6 and 7.0.6.1*. The number 1.3 was selected, due to approximately being the average value of “n” BTDC and ATDC. The ten degrees of crank angle selection for the average value of “n” will differ for each RPM.

1.302861	n (Compression) 1.301979962	
1.297185		
1.305948		
1.291861		
1.312211		
1.326224		
1.285095		
1.310507		
1.281042		
1.306866		

Figure 7.0.6 – “n Compression”.

1.306954	n (Expansion) 1.339763914	
1.346239		
1.329516		
1.361811		
1.332765		
1.347593		
1.332243		
1.354667		
1.349261		
1.336591		

Figure 7.0.6.1 – “n Expansion”.

ΔP can now be calculated using the following equation:

$$\Delta P = \text{Mean } P1 - \text{Mean } P2 \left(\frac{V1}{V2} \right)^n$$

Where:

Mean P = Mean Cylinder Pressure

n = Expansion/Compression

Pressure and volume data obtained and “n” values previously calculated are entered into the equation for ΔP and a value is produced per degree of crank angle rotation and displayed in graph form in *Figure 7.0.7*.

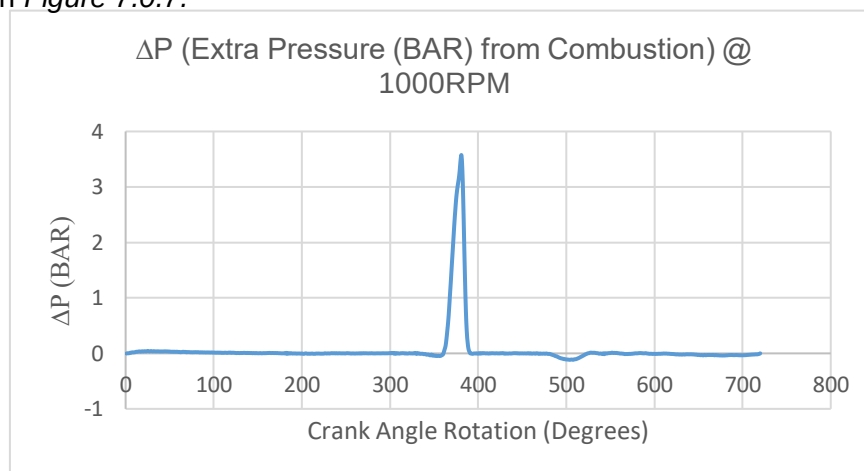


Figure 7.0.7 - ΔP (BAR).

The next step required Start of Combustion (SOC) and End of Combustion (EOC) to be located dependant on ΔP . Consequently, a trend of numbers starting at zero, increasing and then finishing at zero was required to be located.

From the Sigma data, SOC was located at 0 Degrees crank angle rotation for 1000RPM, as displayed in *Figure 7.0.8*. However; as RPM increases ignition timing is advanced and consequently SOC occurs earlier.

C.A.(DEG)	LogV	LogP	LogV	n	Delta P (P2N)
-5	1.604874	1.321288	1.052286	-0.0475	-0.0475
-4	1.600973	1.325393	0.985321	-0.04584	-0.04584
-3	1.598024	1.328298	0.925934	-0.04075	-0.04075
-2	1.595827	1.330333	0.857574	-0.02908	-0.02908
-1	1.594503	1.331468	0.366238	-0.02045	-0.02045
0	1.594061	1.33163	0.274666	0.022383	0.022383
1	1.594503	1.331508	0.16826	0.076431	0.076431
2	1.595827	1.331286	0.082974	0.135859	0.135859
3	1.598024	1.331103	-0.43225	0.257098	0.257098
4	1.600973	1.332378	-0.805	0.413142	0.413142
5	1.604874	1.335518	-1.18362	0.580017	0.580017

Figure 7.0.8 – SOC occurs at 16 Degrees BTDC.

Figure 7.0.8 displays the point of SOC and combustion duration highlighted in green, which occurs at TDC, where “n Compression” becomes a positive number and TDC is highlighted in orange. EOC occurs 31 Degrees ATDC, as seen in *Figure 7.0.8.1*, where “n Expansion” becomes a negative number.

21	1.750817	1.645167	-1.29726	3.321101
22	1.763203	1.661235	-0.77544	2.780149
23	1.775756	1.670969	-0.19211	2.064425
24	1.788451	1.673408	0.330125	1.370356
25	1.801266	1.669177	0.749691	0.790157
26	1.814114	1.659545	1.029132	0.407705
27	1.827046	1.646237	1.181145	0.202432
28	1.840043	1.630885	1.2743	0.080461
29	1.853029	1.614338	1.311796	0.033017
30	1.865992	1.597333	1.339398	0.000414
31	1.878924	1.580012	1.340926	-0.00127

Figure 7.0.8.1 – EOC occurs at 52 Degrees ATDC.

Simple Digital Systems declares combustion is complete between 20 and 25 Degrees ATDC, this matches with the Sigma results, as “n expansion” gradually decreases and equals less than one at 25 Degrees ATDC, as displayed in *Figure 7.0.8.1*. (Simple Digital Systems, 1998)

However, despite the combustion process completion at 25 degrees, “n expansion” doesn’t become less than zero until 31 degrees ATDC, consequently this is where EOC occurs for 1000RPM.

Next; the sum of ΔP is calculated over the combustion duration period, via the addition of ΔP values over the combustion period, presented in *Figure 7.0.9*.

0.022383	0.022383
0.076431	=W373+V374
0.135859	0.234674
0.257098	0.491772
0.413142	0.904915
0.580017	1.484931
0.800881	2.285812
1.057018	3.342829
1.301826	4.644655
1.570501	6.215156

Figure 7.0.9 – The Sum of ΔP (BAR).

Finally; Mass Fraction Burn (MFB) percentage can be calculated. However, Maximum Brake Torque (MBT) is required first. MBT is obtained through the addition of ΔP over the combustion duration period, via calculating the sum of the values. Once MBT is obtained, MFB is calculated via division of the Sum of ΔP by MBT for the combustion duration period. Consequently, producing a set of results over the combustion period, as displayed in *Figure 7.0.10*.

C.A (Deg)	ΔP	Sum of ΔP	MFB
0	0.022383	0.022383	0.000463021
1	0.076431	0.098815	0.002044073
2	0.135859	0.234674	0.004854456
3	0.257098	0.491772	0.010172784
4	0.413142	0.904915	0.018719025
5	0.580017	1.484931	0.030717223
6	0.800881	2.285812	0.047284204
7	1.057018	3.342829	0.069149628
8	1.301826	4.644655	0.096079145
9	1.570501	6.215156	0.128566454
10	1.825011	8.040167	0.16631856
11	2.084279	10.12445	0.209433874
12	2.343344	12.46779	0.257908189
13	2.574902	15.04269	0.311172506
14	2.782885	17.82558	0.368739144
15	2.932634	20.75821	0.429403477
16	3.047917	23.80613	0.492452548
17	3.149186	26.95531	0.557596477
18	3.289823	30.24514	0.62564961
19	3.477949	33.72309	0.697594322
20	3.568671	37.29176	0.771415703
21	3.321101	40.61286	0.840115854
22	2.780149	43.39301	0.897625889
23	2.064425	45.45743	0.940330496
24	1.370356	46.82779	0.968677614
25	0.790157	47.61794	0.985022766
26	0.407705	48.02565	0.993456542
27	0.202432	48.22808	0.997644039
28	0.080461	48.30854	0.999308445
29	0.033017	48.34156	0.999991438
30	0.000414	48.34197	1

Figure 7.0.10 – Results over SOC period.

MFB is produced over the combustion duration period and is measured over a scale between 0 and 1, where zero denotes the SOC and the maximum of one identifies where the combustion process is complete and consequently EOC occurs, shown in *Figure 7.0.11* for 1000RPM.

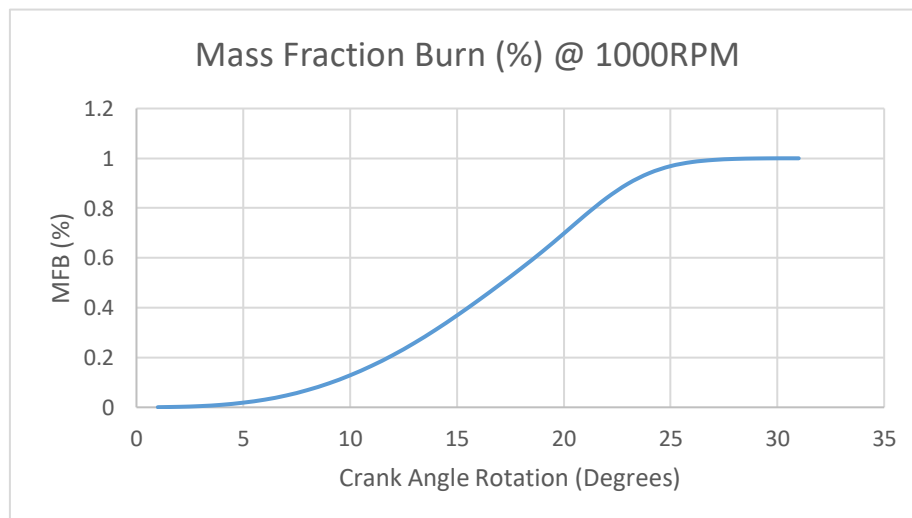


Figure 7.0.11 – MFB (%).

At 1000RPM, combustion duration period is between 0 and 30 degrees ATDC. Consequently, MFB graph in *Figure 7.0.11* is scaled for these degrees of crank angle rotation.

6.0.1.2 – Ford Sigma Excel for 3500RPM:

Graphical results for the Sigma engine at 3500RPM are displayed in the following chapter.

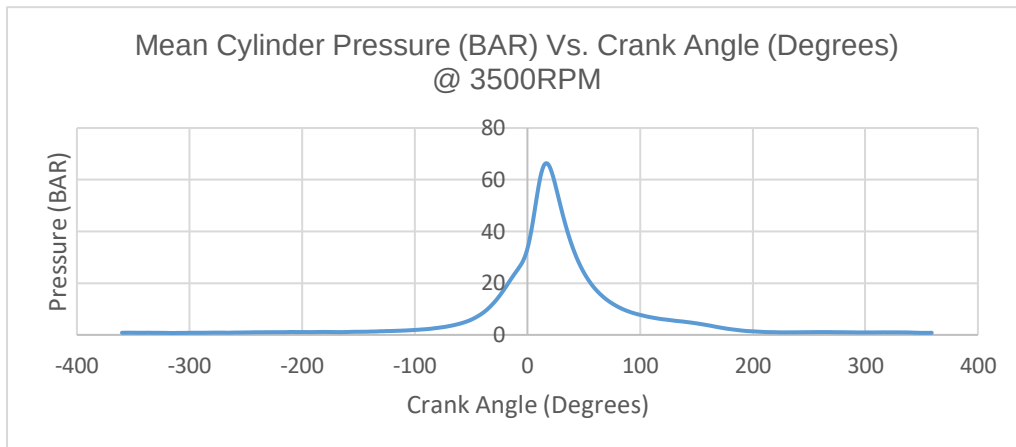


Figure 7.0.11.1 – Mean Cylinder Pressure (BAR) vs Crank Angle (Degrees).

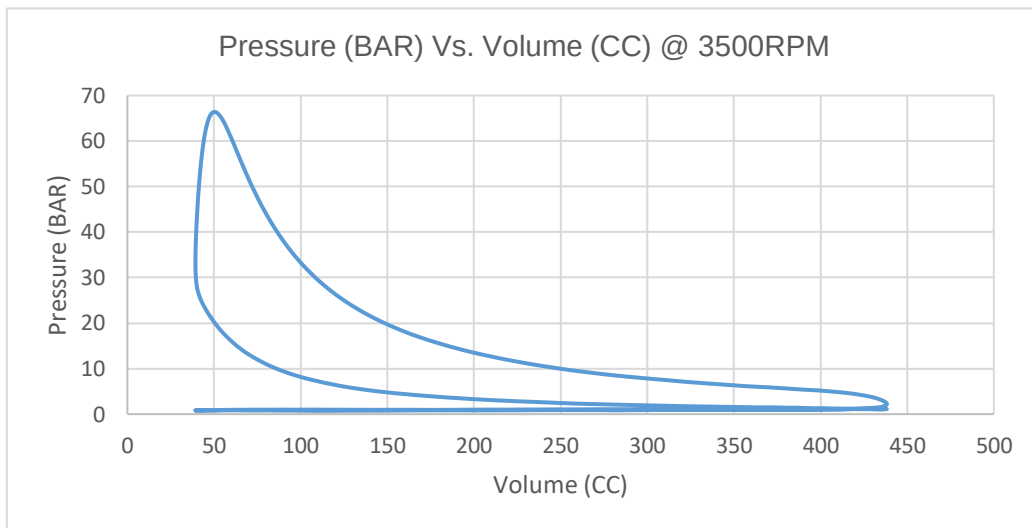


Figure 7.0.11.2 – Pressure vs Volume.

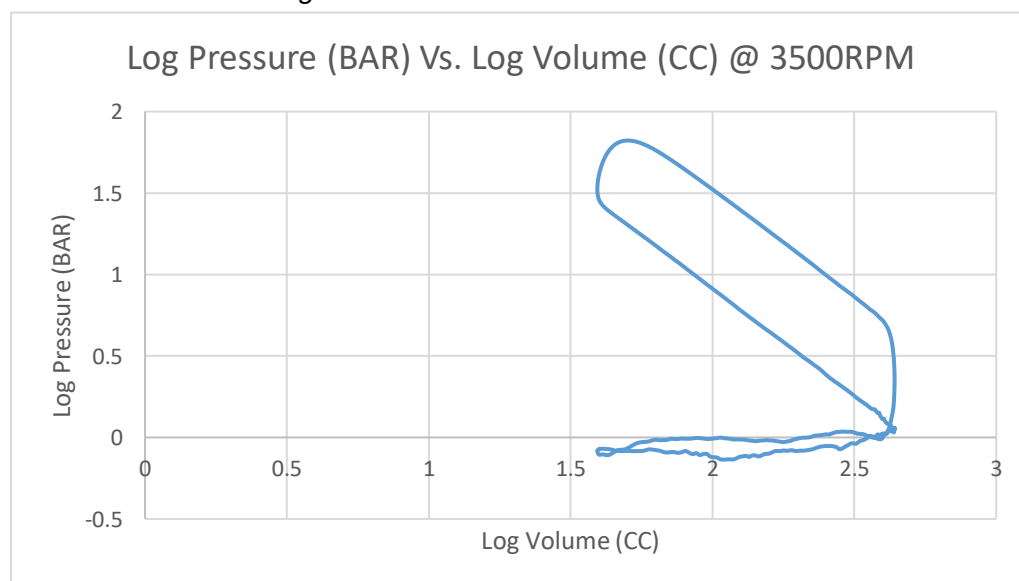


Figure 7.0.11.3 – LogP (BAR) vs LogV (CC).

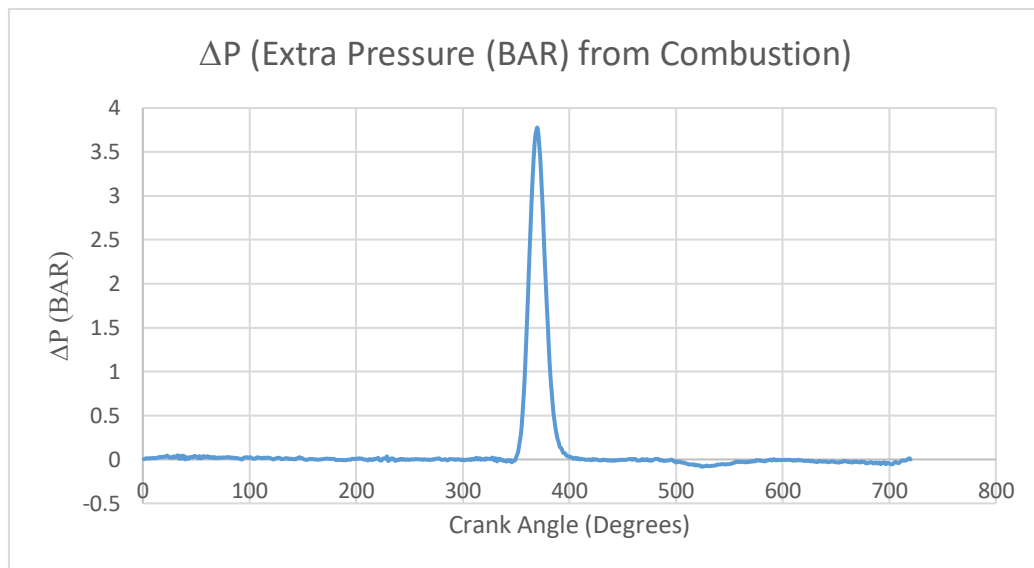


Figure 7.0.11.4 - ΔP (BAR).

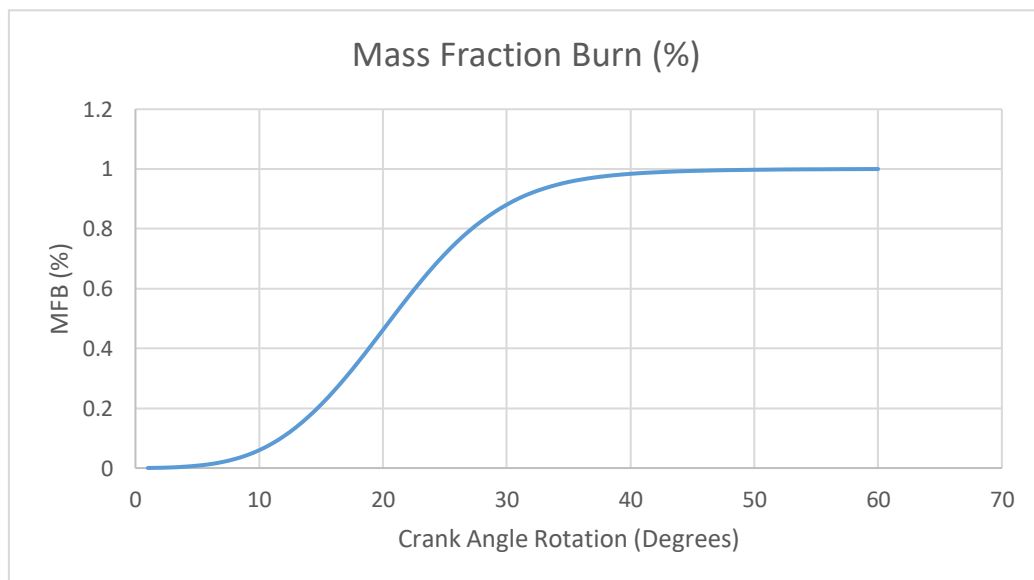


Figure 7.0.11.5 – MFB (%).

This chapter will be concluded in Chapter 6.0.1.4.

6.0.1.3 – Ford Sigma Excel for 6500RPM:

The final data for the Sigma engine is the maximum of 6500RPM and is presented via graphical representation in the corresponding chapter.

Mean Cylinder Pressures Vs. Crank Angle (Degrees) @ 6500RPM

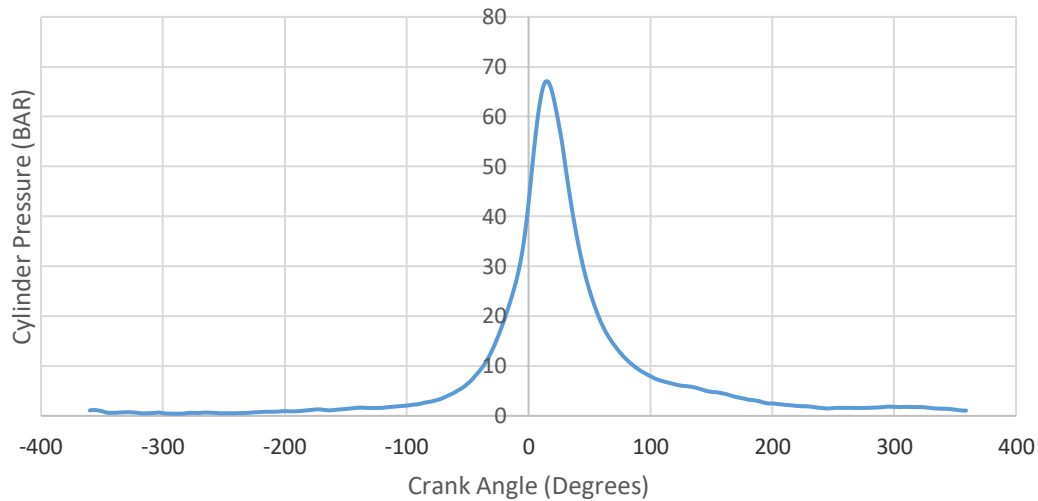


Figure 7.0.12 – Mean Cylinder Pressure (BAR) vs Crank Angle (Degrees).

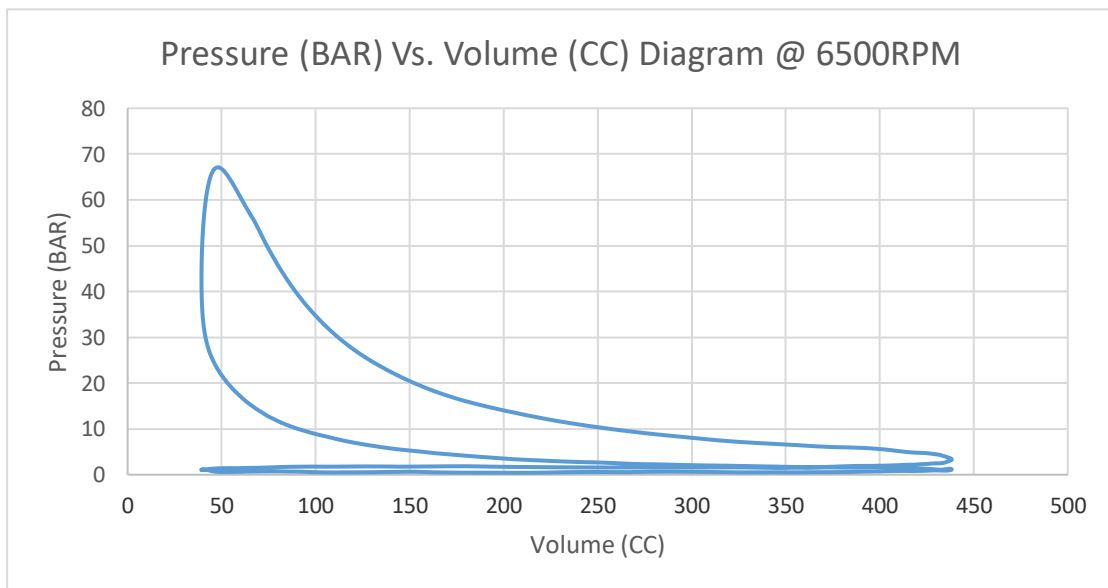


Figure 7.0.12.1 – Pressure (BAR) vs Volume (CC).

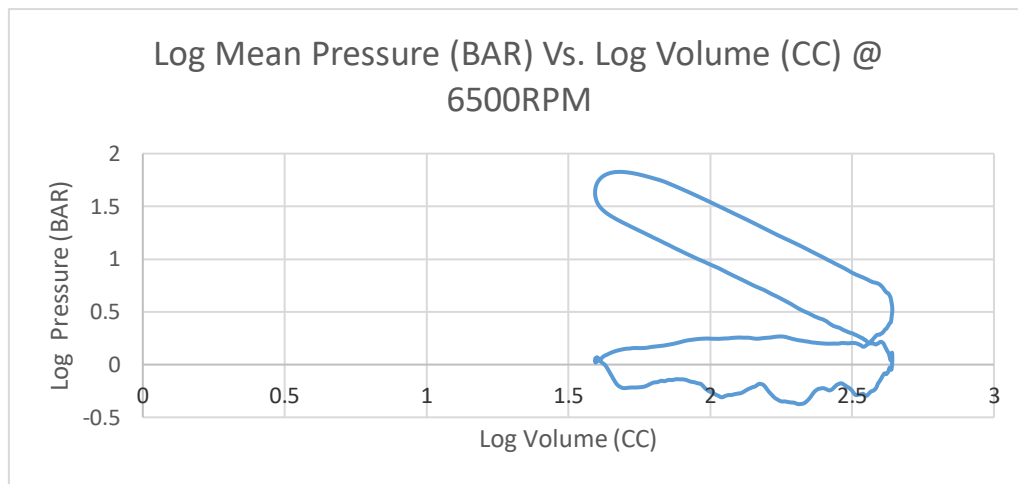


Figure 7.0.12.2 – LogP (BAR) vs LogV (CC).

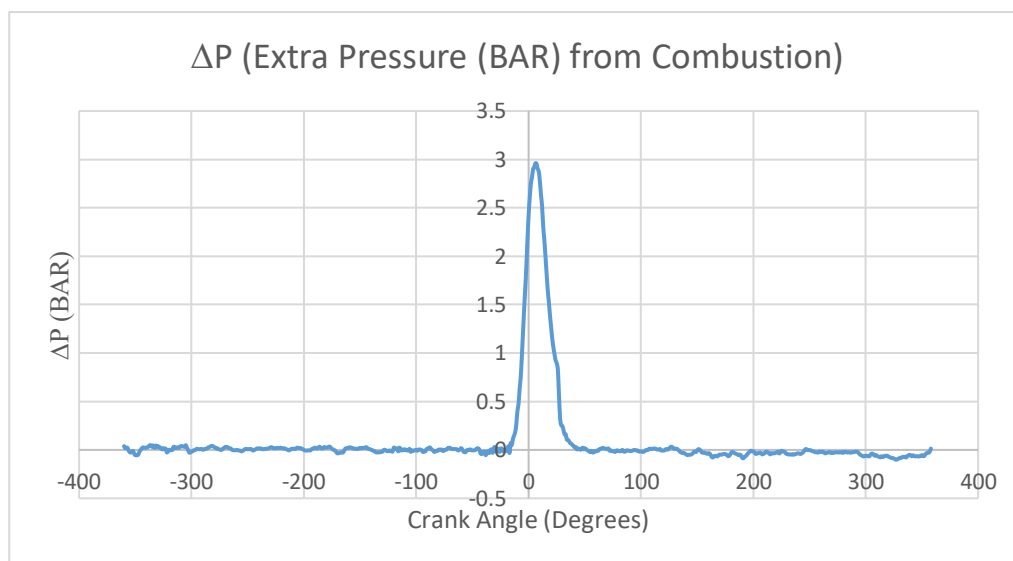


Figure 7.0.12.3 - ΔP (BAR).

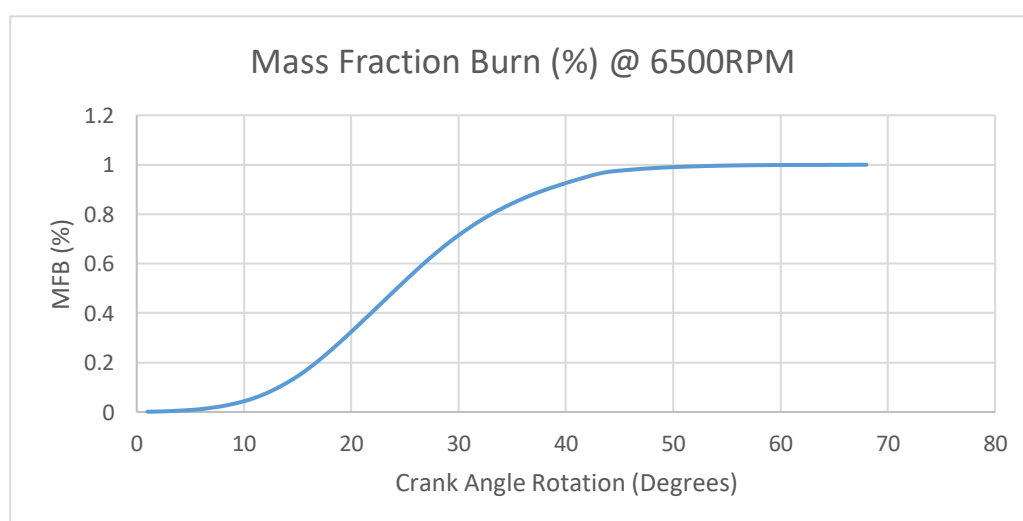


Figure 7.0.12.4 – MFB (%).

6.0.1.4 – Ford Sigma Data Summary:

Fifteen Ford Sigma graphs have been produced for the main RPMs selected.

Cylinder Pressure:

Cylinder pressure reaches a maximum of 47.142 BAR at 24 degrees ATDC for 1000RPM. Next a rapid increase to 66.59 BAR at 19 degrees ATDC for 3500RPM then occurs. Finally, the cylinder pressure plateaus and reaches a maximum of 67.103 BAR at 15 degrees ATDC. Consequently, the relationship of increased RPM causing an increase in maximum cylinder pressure is established.

Pressure vs Volume:

Pressure is dependent on cylinder pressure at each individual RPM and volume has a constant maximum value, as the maximum volume of the cylinder is unable to alter, as the cylinder is unable to expand.

Pressure and Volume Logarithms:

Logarithms of pressure and volume enable SOC, combustion duration and EOC to be visualised. All these aspects can be understood in *Figure 7.0.13*.

The green arrow represents SOC, where pressure rises to a point and then drops and plateaus, the risen point represents SOC. The orange arrow represents EOC, where a reduction in pressure occurs. Finally; combustion duration is the distance between SOC and EOC and is represented by the blue arrow.

At 1000RPM these points are clearly identified, however as RPM increases, these points become more difficult to recognise, as displayed in *Figure 7.0.13.1*.

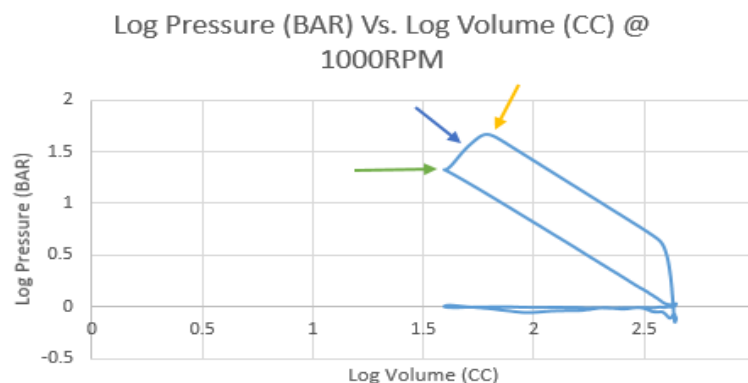


Figure 7.0.13 – LogP (BAR) vs LogV (CC) @ 1000RPM.

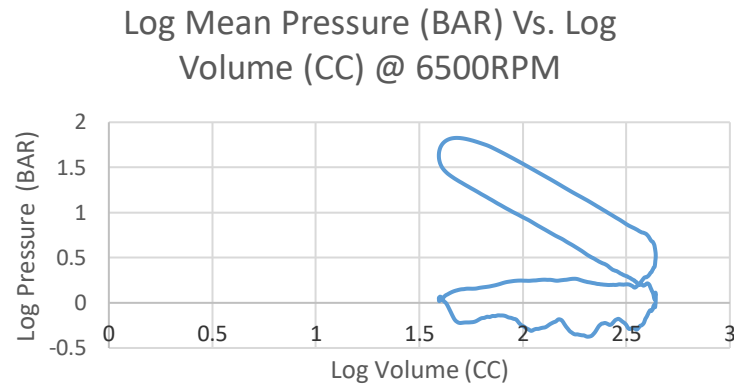


Figure 7.0.13.1 – LogP (BAR) vs LogV (CC) @ 6500RPM.

ΔP :

ΔP is calculated using the equation previously mentioned in Chapter 6.0.1.1. Consequently, ΔP is dependent on volume, pressure and the value of “n”. Therefore, several factors can cause a fluctuation in the numerical value.

ΔP at 1000RPM reaches a maximum of 3.568671BAR at 20 degrees ATDC. ΔP is further increased to 3.777085BAR at 9 degrees ATDC for 3500RPM. However, a reduction to 2.95943394BAR is seen 6 degrees ATDC for 6500RPM.

As previously mentioned, several factors could have caused the reduction of ΔP for 6500RPM. Due to volume staying constant per degree of crank angle rotation independent of change in RPM, either pressure or “n” value are the likely cause of this fluctuation occurrence.

On the other hand, a clear pattern of the maximum value of ΔP occurring closer to TDC as RPM increases is established.

Mass Fraction Burn:

MFB percentage gradually increases until the maximum of 1 (100%) is reached. The curve alters dependant on combustion duration. MFB for 1000RPM last for 31 degrees of crank angle rotation, an increase to 60 degrees’ duration occurs for 3500RPM and 68 degrees for 6500RPM. Consequently; as RPM increases, combustion duration increases, thus MFB duration increases.

6.0.2 – Ford Fox Excel for 1000RPM:

Graphical representation of Ford Fox data was produced for comparison to Ford Sigma data, using data obtained via lecture Malcolm McDonald. Subsequently graphs were produced for 1000RPM.

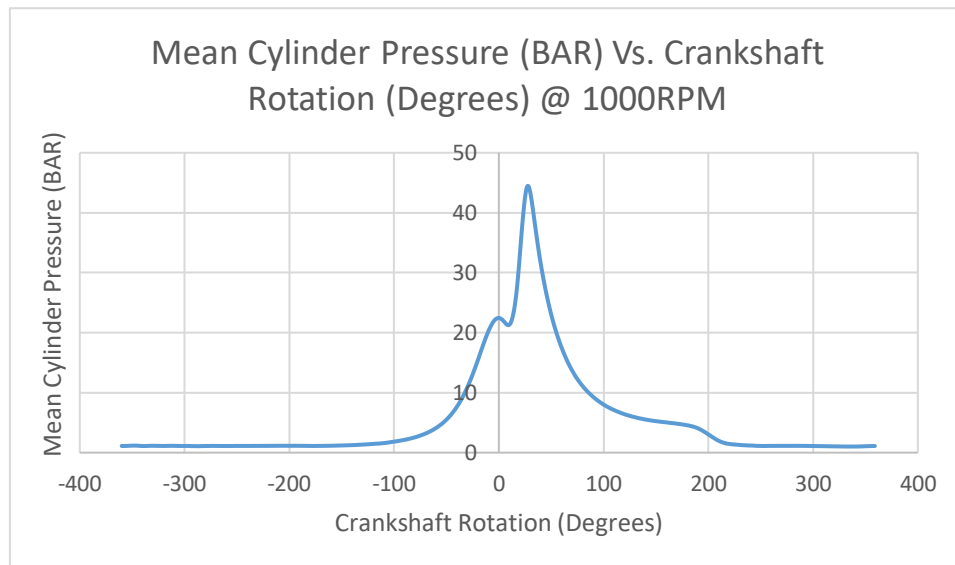


Figure 8.0 - Mean Cylinder Pressure (BAR) vs Crank Angle (Degrees).

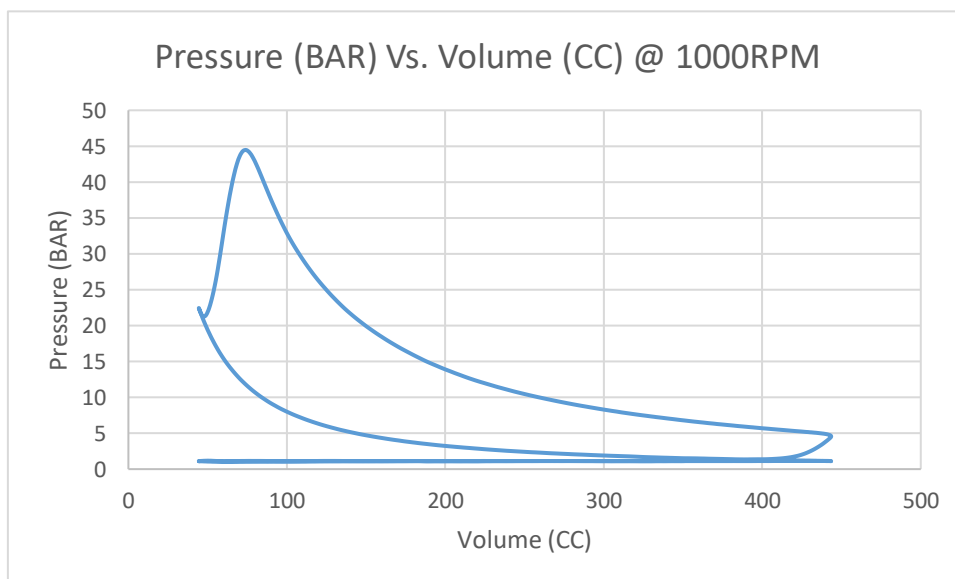


Figure 8.0.1 - Pressure (BAR) vs Volume (CC).

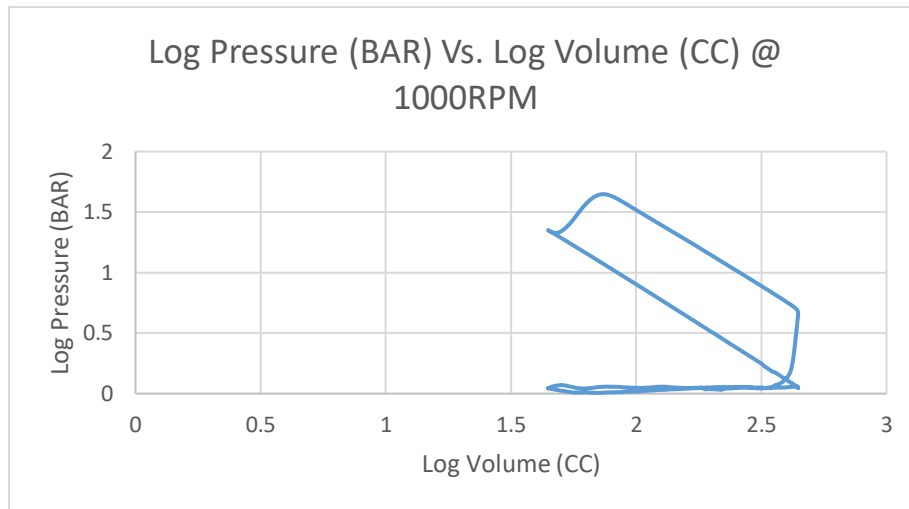


Figure 8.0.2 - LogP (BAR) vs LogV (CC).

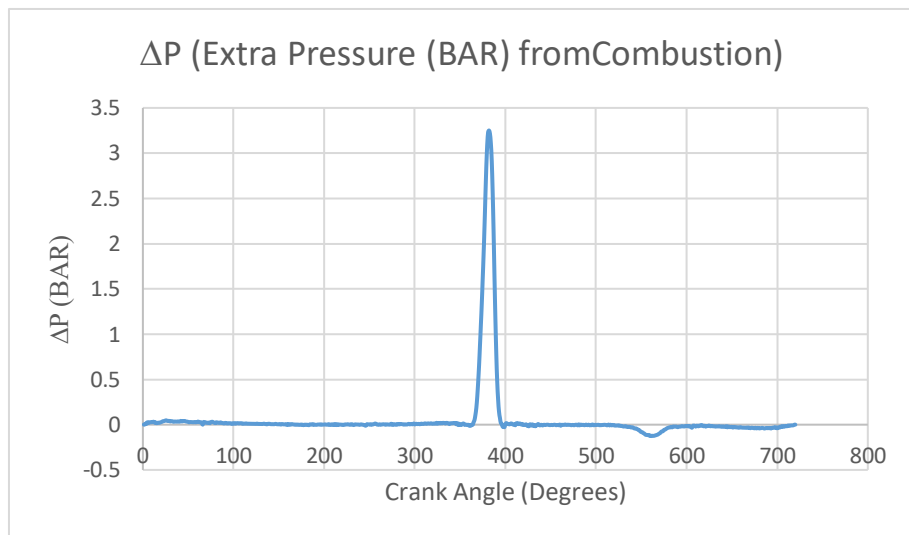


Figure 8.0.3 - ΔP (BAR).

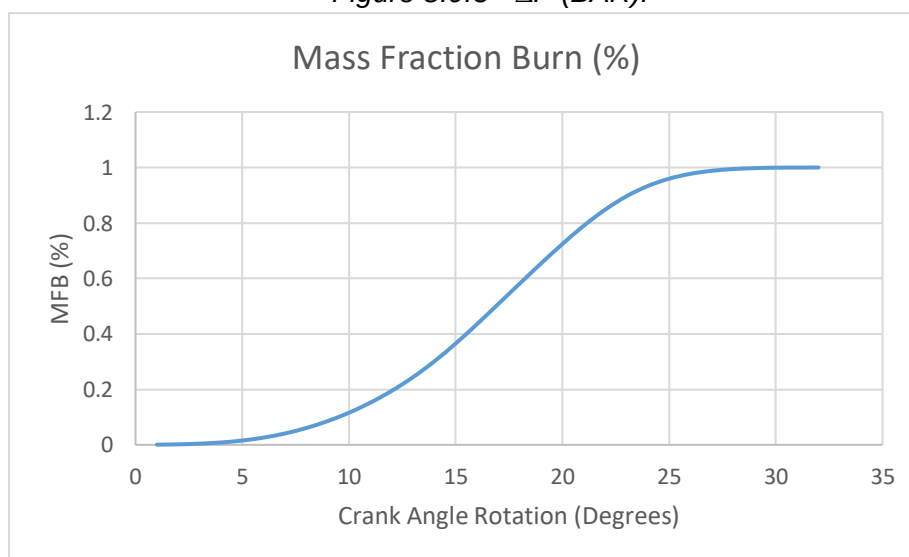


Figure 8.0.4 – MFB (%).

6.0.2.1 – Ford Fox Excel for 3000RPM:

Unlike the Ford Sigma, 3500RPM was selected as the second RPM data of focus, due to the slight varying data for RPM obtained for the two engines.

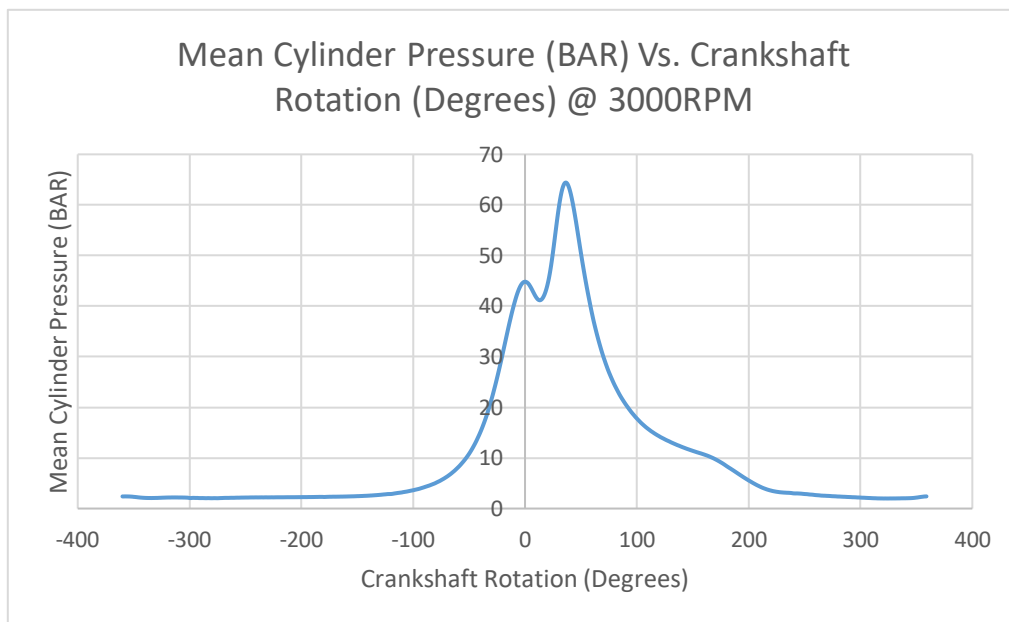


Figure 8.0.5 - Mean Cylinder Pressure (BAR) vs Crank Angle (Degrees).

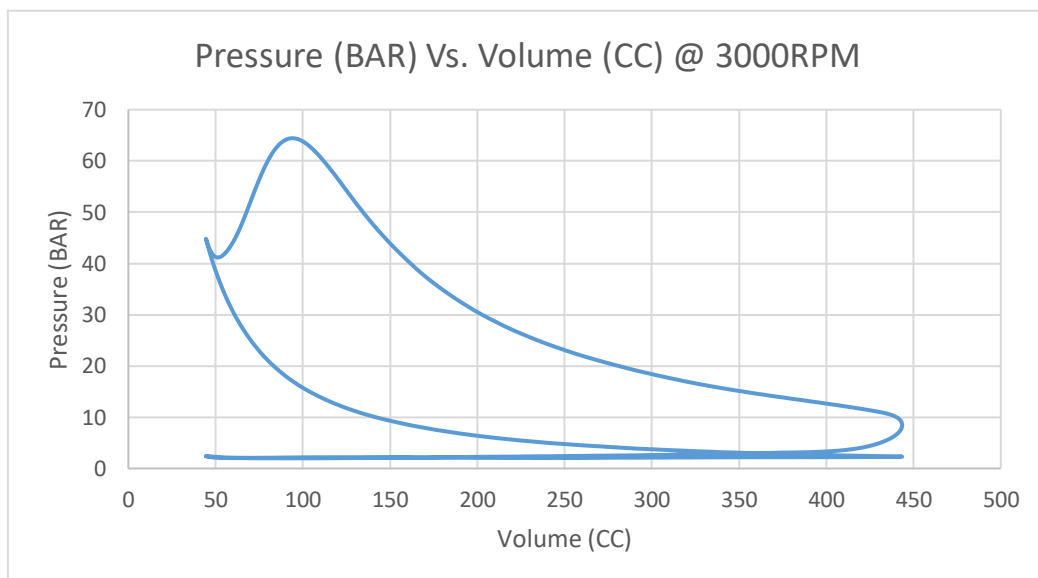


Figure 8.0.6 - Pressure (BAR) vs Volume (CC).

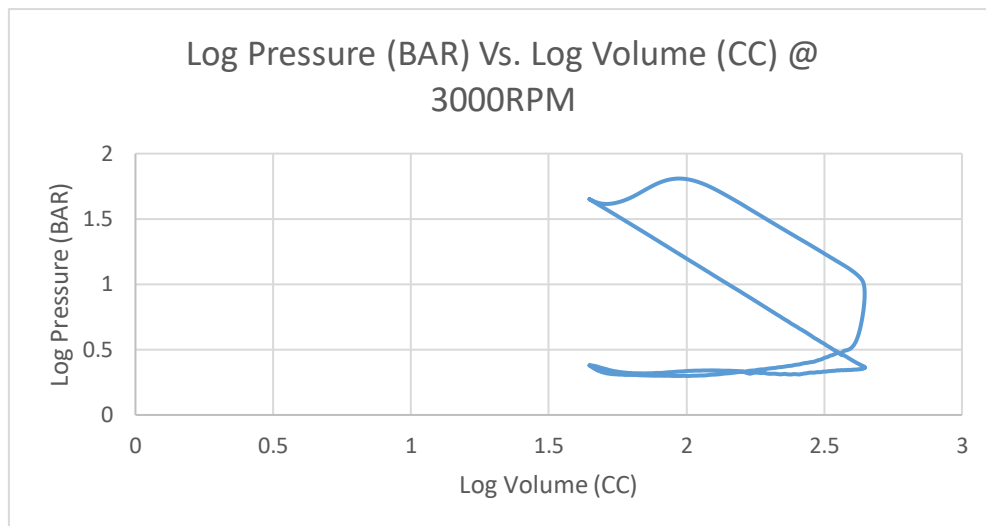


Figure 8.0.7 - LogP (BAR) vs LogV (CC).

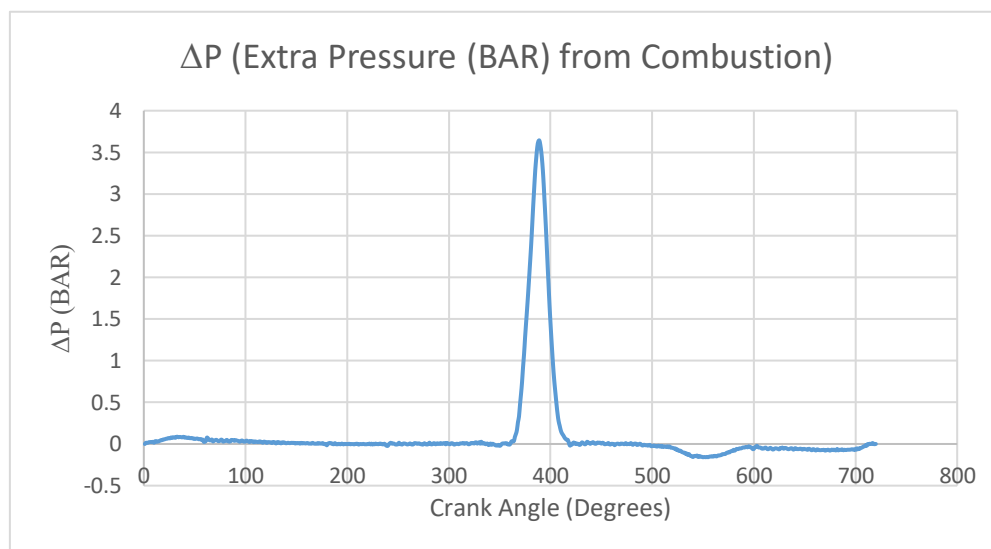


Figure 8.0.8 - ΔP (BAR).

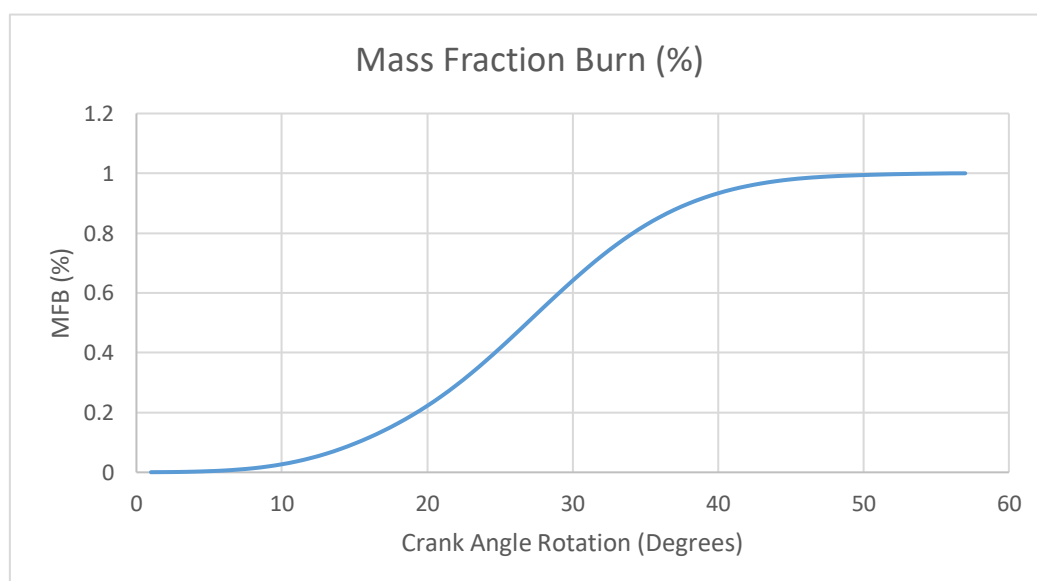


Figure 8.0.9 – MFB (%).

6.0.2.2 – Ford Fox Excel for 6000RPM:

Finally; 6000RPM was selected as the final Ford Fox RPM data to be examined.

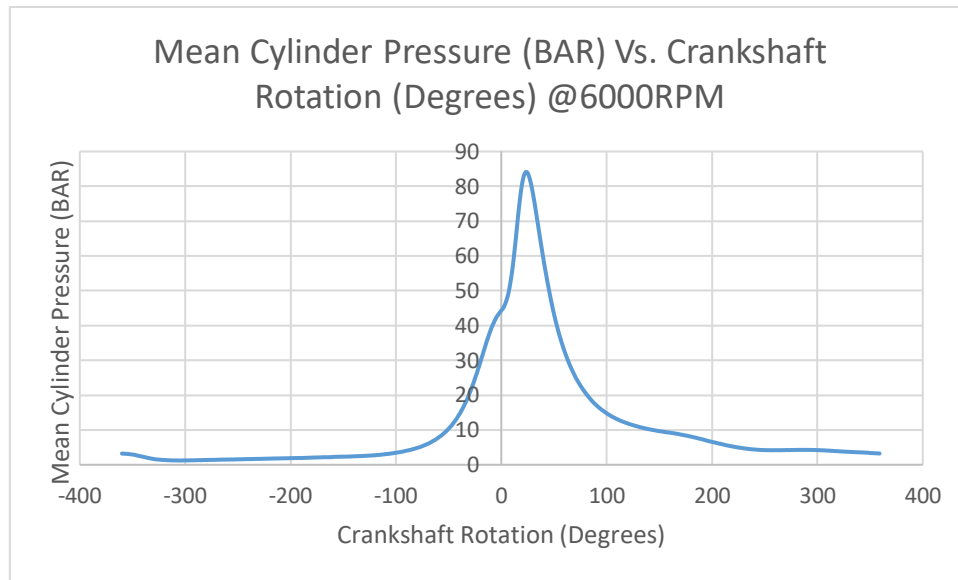


Figure 8.0.10 - Mean Cylinder Pressure (BAR) vs Crank Angle (Degrees).

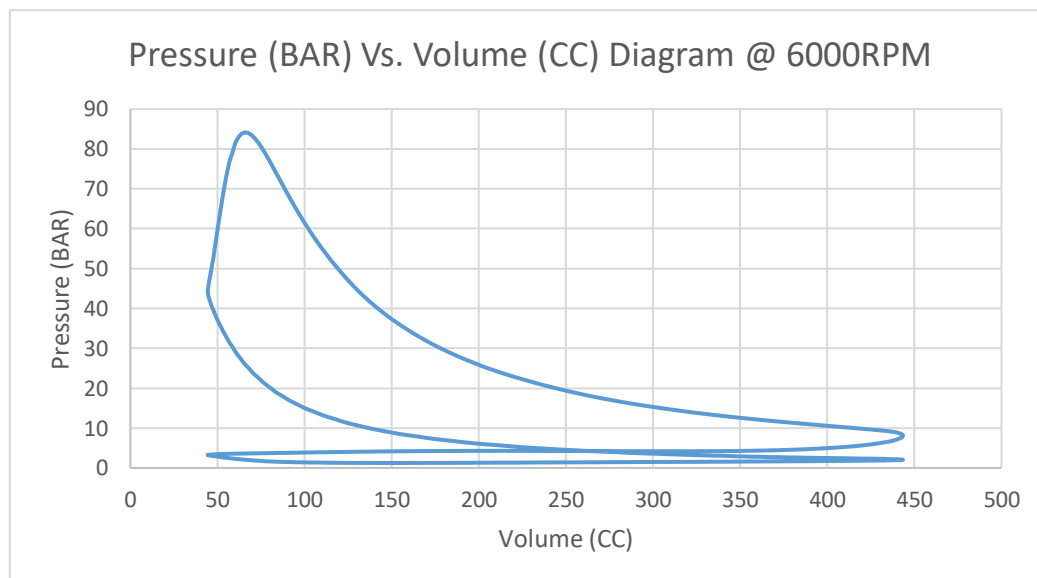


Figure 8.0.11 - Pressure (BAR) vs Volume (CC).

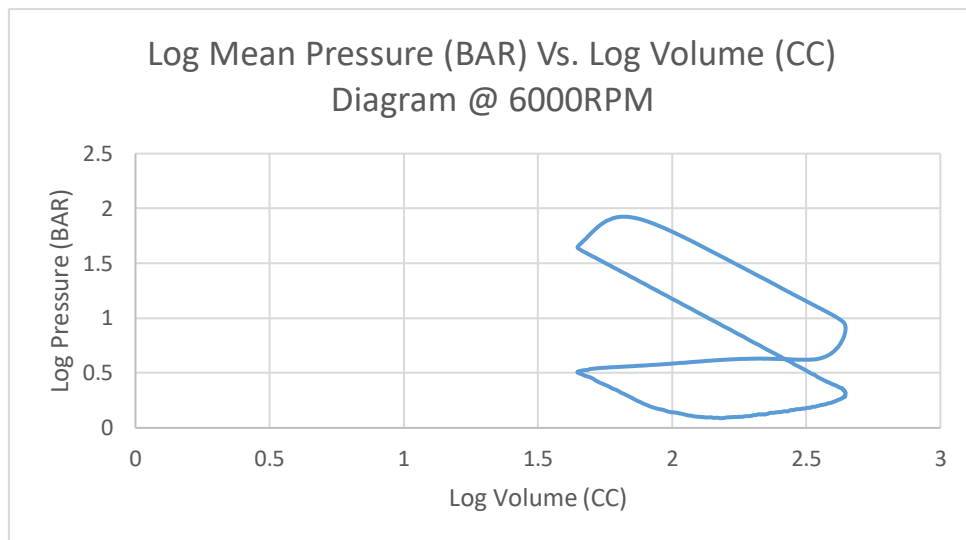


Figure 8.0.12 - LogP (BAR) vs LogV (CC).

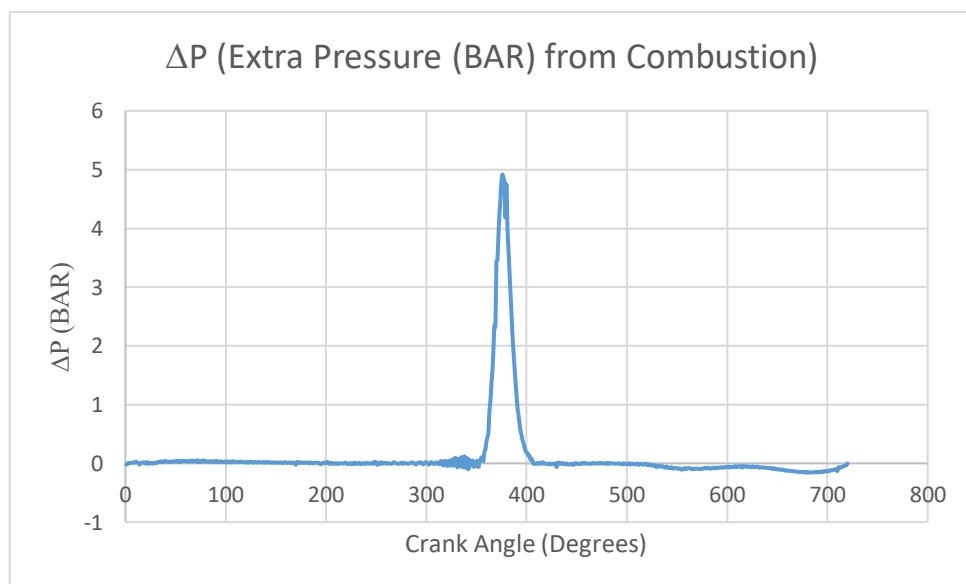


Figure 8.0.13 - ΔP (BAR).

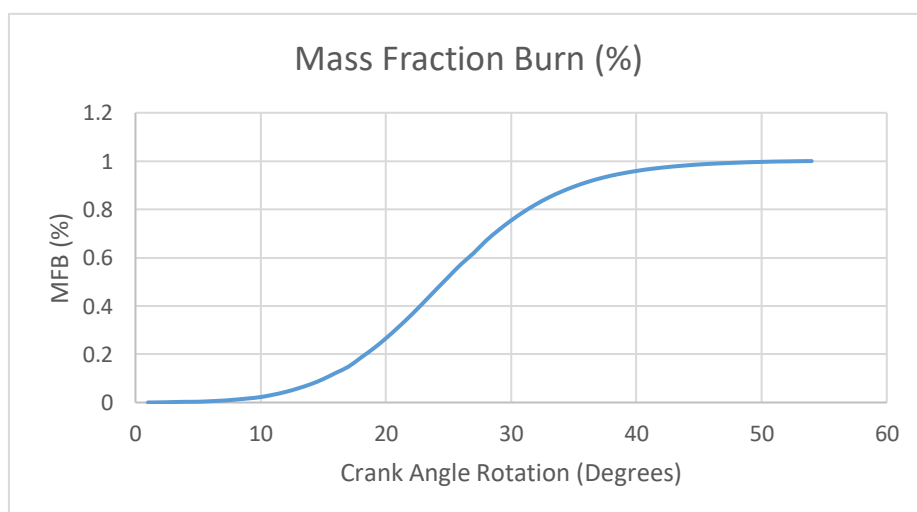


Figure 8.0.14 – MFB (%)

6.0.3 – Mazda Rx8 13b Renesis Rotary Engine Excel for 6646RPM:

Graphical representations of engine data were required for the final engine. However due to the limited engine data available, graphs were only able to be produced for 6646RPM. However multiple calculations and a respectable yield of data was obtained from the limited data available, displayed in *Figure 9.0*.

C.A (Degrees)	Pressure (BAR)	Volume (CC)	logV	LogP	n	Delta P	Sum of Delta P	MFB		
-90	8.963	248.4833746	2.395297	0.952453	1.156739	-0.18493				
-80	10.342	219.5673223	2.341568	1.014605	1.630753	0.781479				
-70	13.789	184.060251	2.26496	1.139533	1.303857	0.008923				
-60	16.547	160.0396192	2.204228	1.218719	2.205975	2.789712				MBT
-50	22.753	138.5241029	2.141525	1.357039	0.991823	-1.69862				117.6965
-40	27.579	114.1029199	2.057297	1.440579	1.305402	0.024963				
-30	33.095	99.2288985	1.996638	1.519762	0.934834	-1.75306				
-20	37.232	87.48191946	1.941918	1.570916	0.817677	-2.65269				
-10	41.369	76.90534184	1.885957	1.616675	2.07095	1.859065	1.859064812	0.015795		
0	46.195	72.91508578	1.862817	1.664595	-0.82196	5.159343	7.018407544	0.059631		
10	48.263	76.90534184	1.885957	1.683614	-1.776	19.85485	26.87325552	0.228327		
20	60.674	87.48191946	1.941918	1.783003	-2.12373	27.78194	54.65519091	0.464374		
30	79.289	99.2288985	1.996638	1.899213	-1.40817	30.39936	85.05454728	0.72266		
40	96.523	114.1029199	2.057297	1.984631	0.303186	15.9985	101.0530461	0.85859		
50	91.011	138.5241029	2.141525	1.959094	0.602647	7.990579	109.0436253	0.926482		
60	83.427	160.0396192	2.204228	1.921307	0.882228	4.184832	113.2284577	0.962038		
70	73.744	184.060251	2.26496	1.867727	0.915944	4.109733	117.3381907	0.996956		
80	62.742	219.5673223	2.341568	1.797558	1.245972	0.358268	117.6964583	1		
90	53.779	248.4833746	2.395297	1.730613	1.198576	0.541198				
100	46.884	278.6204939	2.445013	1.671025	0.871366	2.469871				
110	41.369	321.6517868	2.507386	1.616675	-0.64477					

Figure 9.0 – Calculations produced for the Wankel Rotary Engine.

Consequently; several graphical representations were then produced.

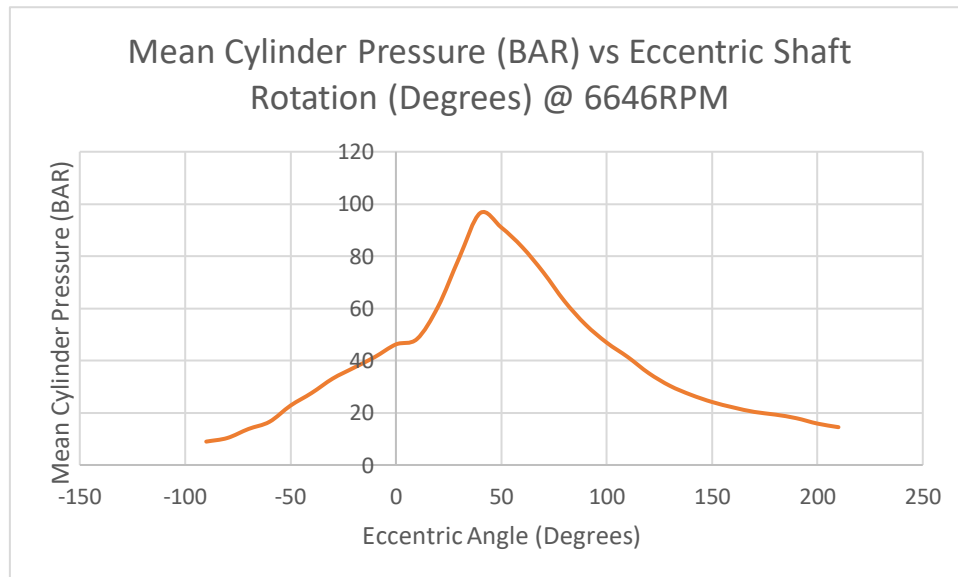


Figure 9.0.1 – Mean Cylinder Pressure (BAR) vs Eccentric Angle (Degrees).

Wankel Rotary engine graphs were dissimilar to graphs produced for the conventional piston engines due to the Rotary engine only measured over 270°, due to no consideration of the intake chamber, which consists of 90°.

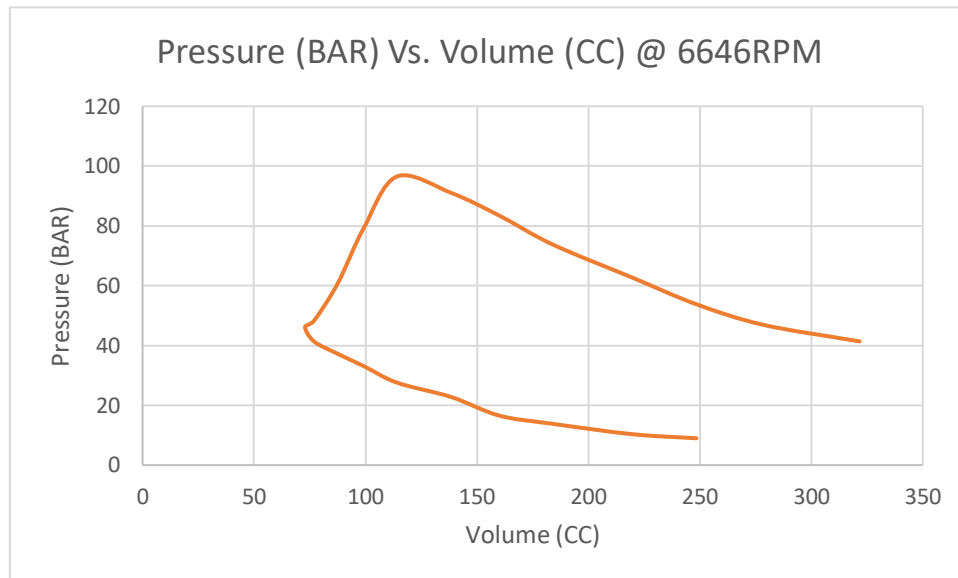


Figure 9.0.2 - Pressure (BAR) vs Volume (CC).

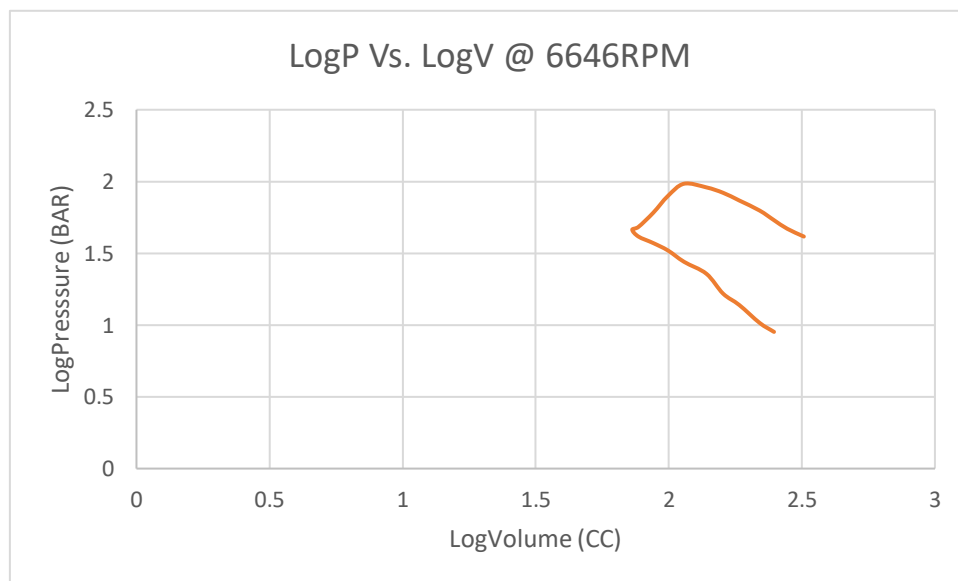


Figure 9.0.3 - LogP (BAR) vs LogV (CC).

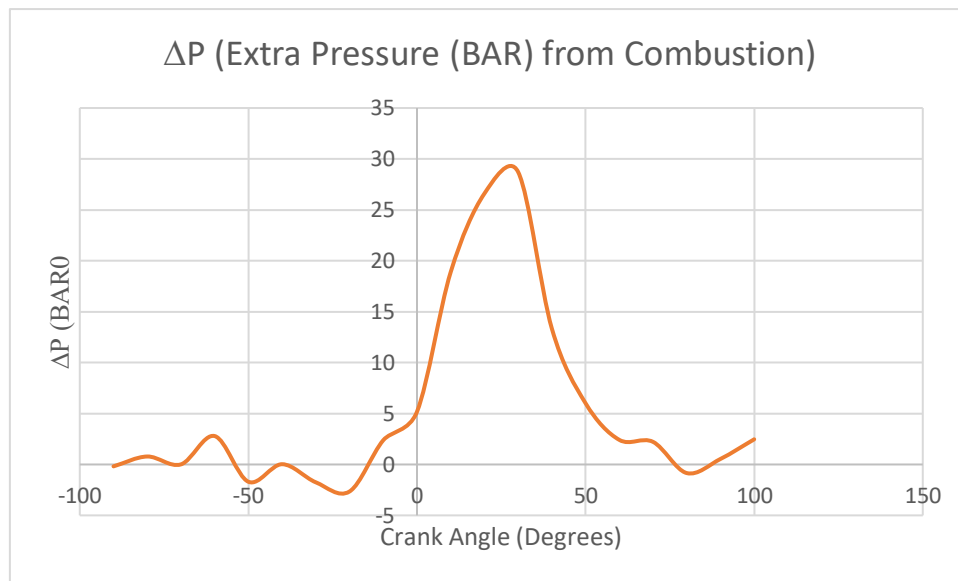


Figure 9.0.4 - ΔP (BAR).

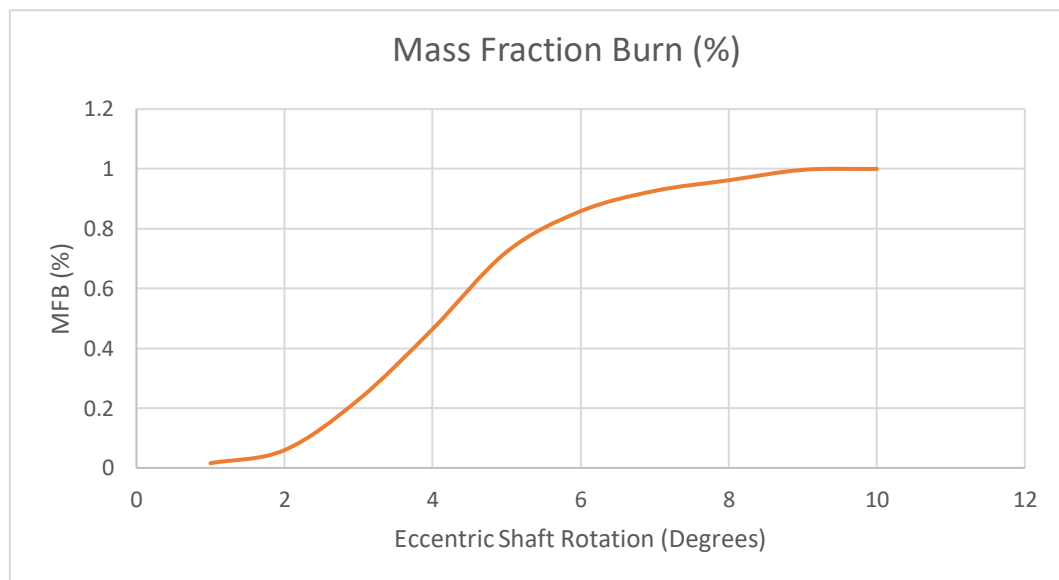


Figure 9.0.5 – MFB (%).

6.0.4 – Chapter References:

1. Malcolm McDonald 2016, Ford Fox Data, Major Project, M6X00417, UWTSD University.
2. Malcolm McDonald 2016, Ford Sigma Data, Powertrain and Sustainability, M6X01206, UWTSD University.
3. Simple Digital Designs 1998, Ignition and Combustion, Viewed 9th April, <http://www.sdsefi.com/techcomb.htm>

6.0.4.1 – Chapter Literature Review:

1. Malcolm McDonald – Obtained Ford Fox engine data.
2. Malcolm McDonalds – Obtained Ford Sigma engine data.
3. Simple Digital Designs – Used for the Sigma data, to confirm 16 Degrees BTDC is approximately the correct period SOC ought to occur and EOC should occur between 20 and 25 Degrees ATDC.

Chapter 7.0 – Engine Disassembly:

Ford Sigma and Ford Fox engine components were easily accessible from the University School of Engineering. However, Wankel Rotary engine components required were unavailable. Consequently, a Wankel Rotary engine was required to be sourced.

7.0.1 – Mazda Rx8 13b Renesis Engine Disassembly:

Once the engine was located and collected, disassembly was required to gain access to specific internal engine components required for examination.

The disassembly process is listed below:

1. Flywheel and Pressure Plate removal required six 14mm bolts
2. A 54mm socket was required to loosen the flywheel nut
3. Once the Flywheel nut was loosened, the specialist stock Flywheel Puller Tool was required to remove the Mazda flywheel from its taper press fit via the eccentric shaft
4. Sump removal required twenty-two 6mm bolts
5. Water Pump removal required eight 10mm bolts
6. Thermostat Housing removal required three 10mm bolts
7. Front Rotor Housing Iron was then able to be removed
8. Sixteen 10mm tension bolts removed from the Rotor Chamber
9. Oil Filter Housing removal required two 10mm bolts
10. Two locating dowels were removed
11. Finally, the Rotor Housing could be removed
12. Engine components were then degreased, cleaned and Vapour Blasted to improve visual appearance for use in the first Interim VIVA.

The Wankel Rotary engines simplistic design, consequently requires minimal effort for disassembly. Thus, disassembly duration may only require two or three hours, compared to a conventional piston engine which may require double or triple the time, dependant on engine.



Figure 10.0 – Engine Disassembly. (Jay, 2017)

7.0.2 – Engine Component Measurements:

Specific engine components required measurement using Digital Vernier Callipers and a protractor to enable the creation of accurate Solid Works models.



Figure 11.0 - Wankel Rotor Measurements. (Jay, 2017)



Figure 11.0.1 – Ford Fox Measurements. (Jay, 2017)

Chapter 8.0 – Alternative Methods of Creating Wankel Rotary Engine Components:

Several alternative methods were investigated to produce Wankel Rotary engine components. However, the following chapter will clarify the alternative methods tested under trial and error.

8.0.1 – Epitrochoid Rotor Housing Creation:

Consequently, production of the Rotor Housing via Microsoft Excel using equations obtained from the Kenichi Yamamoto 1981 Second Edition book were selected.

Firstly; specific measurements were required to complete the equations. Pitch Circle Diameter (PCD) is the diameter of the circle which passes through a specific point, was obtained using the following equations and entering physical measurements.

$$\text{PCD Internal Rotor Gear} = 72\text{mm}$$

$$\text{PCD Secondary Rotor Gear} = 48\text{mm}$$

$$e = \frac{\text{PCD internal Rotor Gear} - \text{PCD Secondary Rotor Gear}}{2}$$

$$R (\text{Radius}) = 90\text{mm}$$

$$K (\text{Trochoid Constant } K \text{ Ratio}) = \frac{R}{e} = 7.5$$

(Yamamoto, 1981)

Finally; “X” and “Y” Coordinates were obtained per degree of eccentric shaft angle rotation, displayed in *Figure 12.0*, using the following equations:

$$X \text{ Coordinate} = e \left(\cos(3(\phi)) \right) + R(\cos(\phi))$$

$$Y \text{ Coordinate} = 3 \left(\sin(3(\phi)) \right) + R(\sin(\phi))$$

(Yamamoto, 1981)

Degrees	Radians	X Coordinate	Y Coordinate
0	0	101	0
1	0.01745	100.9699993	2.181295648
2	0.03491	100.8800463	4.360396766
3	0.05236	100.7302887	6.535113686
4	0.06981	100.5209717	8.703266453
5	0.08727	100.252438	10.86268965
6	0.10472	99.92512688	13.01123716
7	0.12217	99.53957261	15.14678696
8	0.13963	99.09640361	17.2672457
9	0.15708	98.5963406	19.37055339
10	0.17453	98.04019486	21.45468781

Figure 12.0 – X and Y Coordinates per Degree of Crank Angle Rotation.

Once all X and Y Coordinates have been produced for every 360 degrees of eccentric shaft angle rotation, the data was transferred into graphical form to produce the Epitrochoid Shape for the Wankel Rotary Engine Rotor Housing, displayed in *Figure 12.0.1*

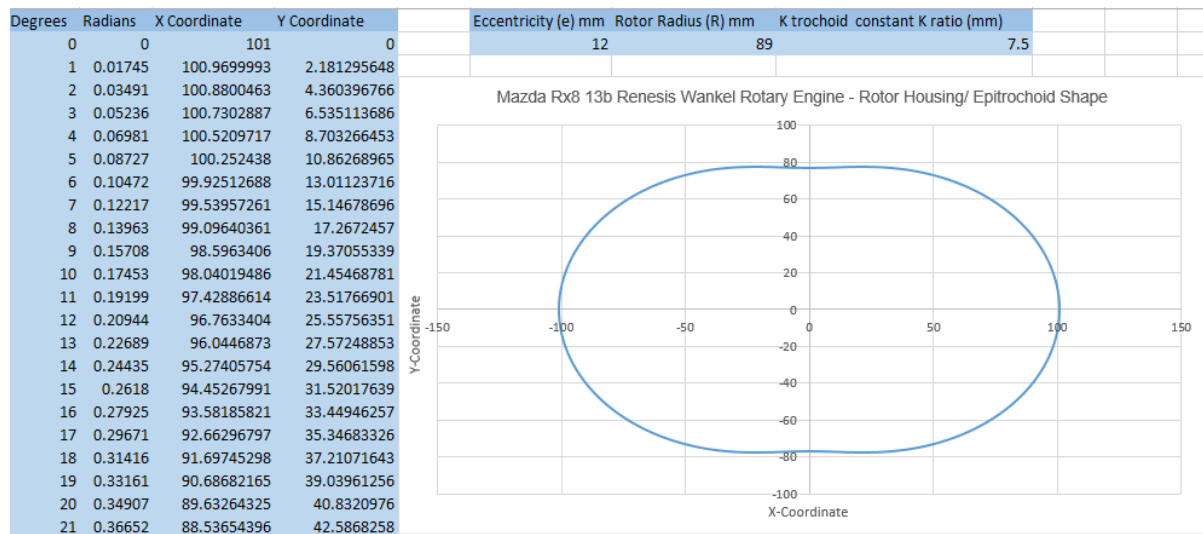


Figure 12.0.1 – Rotor Housing Epitrochoid Shape Production.

8.0.2 – Coordinate Measuring Machine (CMM):

Prior to the production of the rotor via Solid Works 2016, Technician Dave Cooper was approached on the 19th January and the CMM was used to scan the rotor. The rotor was covered in Talcum powder and red dots were placed over the surface to enable the CMM to recognise the surfaces to scan, as displayed in *Figure 12.0.2*. Eight scans of the rotor were produced; the data was then assembled and collected on 23rd January.

This data was transferable to Solid Works 2016, via point clouds produced from the scan, as presented in *Figure 12.0.3*. Using the CMM is beneficial, due to the ability to perform dimensional, positional and surface measurements via a single system. (Nikon, 2017)



Figure 12.0.2 – Red Dots Required for CMM Scan. (Jay, 2017)

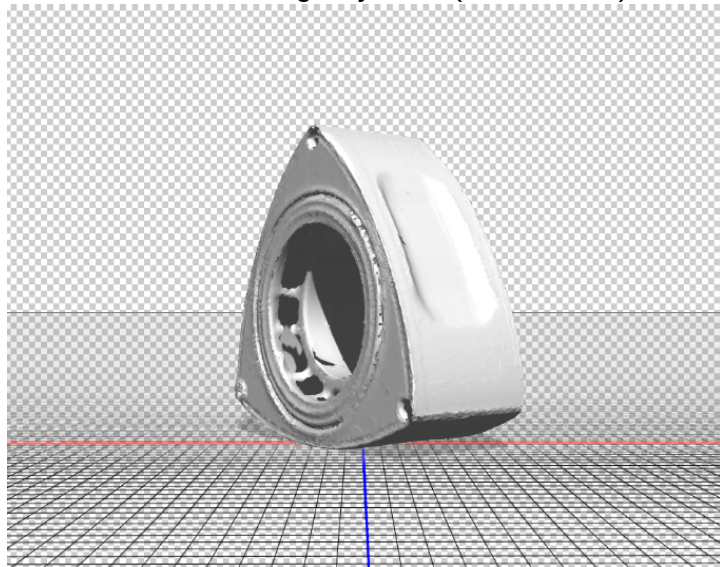


Figure 12.0.3 – CMM Scan Results.

8.0.3 – Chapter References:

1. Nikon 2017, Coordinate Measuring Machine, Viewed 22nd January 2017, http://www.nikonmetrology.com/en_EU/Products/Coordinate-Measuring-Machines
2. Yamamoto, 1981, Automotive, Rotary Engine, Toyo Kogyo CO LTD, Japan

8.0.3.1 – Chapter Literature Review:

1. Nikon 2017, Coordinate Measuring Machine, Viewed 22nd January 2017, http://www.nikonmetrology.com/en_EU/Products/Coordinate-Measuring-Machines
2. Yamamoto – Obtained several equations for the creation of the Epitrochoid Shape for the Rotor Housing, including; X Coordinate, Y Coordinate and PCD equations.

Chapter 9.0 – Solid Works 2016 Component Modelling:

Several alternative methods were researched for production of Wankel Rotary engine components. Two of which were investigated further, as described in Chapter 8.0.2. However, the decision was made to model vital engine components via Solid Works 2016, using engine measurements previously obtained. The Ford Fox Turbo assembly were the first engine components designed using the Solid Works software.

9.0.1 – Ford Fox Turbo Solid Works:

The Connecting Rod (Con-Rod), Gudgeon Pin and Piston assembly were collected from lecturer Malcolm McDonald. Measurements were obtained, consequently enabling component modelling to commence.



Figure 13.0 – Sigma Assembly. (Jay, 2017)

Several sketches and models were produced prior to the final example. Therefore, enabling further understanding of Solid Works, via the use of a variety of built in tools within the software. Consequently, model quality improved over time.

9.0.1.1 – Ford Fox Turbo Connecting Rod:

The first Ford Fox Turbo Solid Works models were presented in the first Interim VIVA, as displayed in *Figure 13.0.1*.

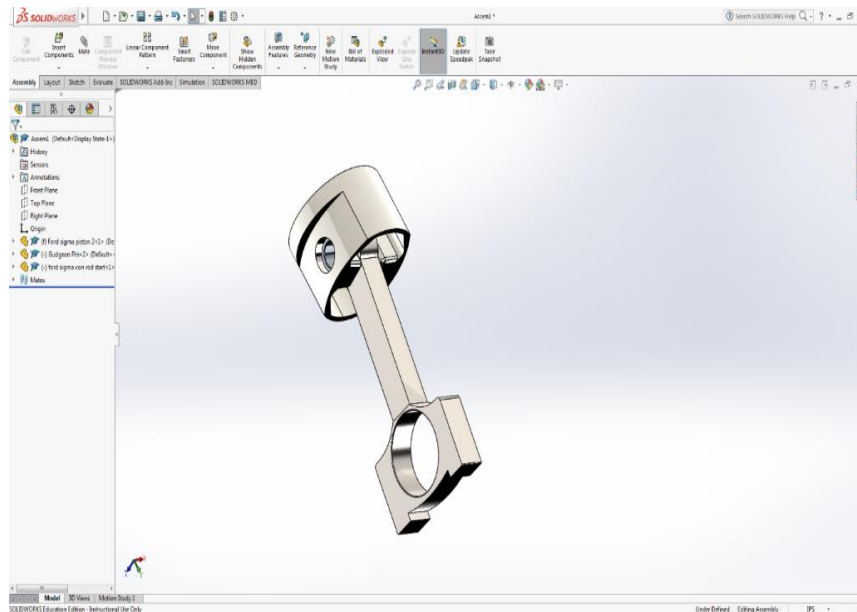


Figure 13.0.1 – First Solid Works Model.

However, further detail of the piston, and re-design of the Con-Rod was required. Therefore, a variety of Solid Works tools, such as the mirror entities, chamfer, fillet, resolved boss base etc, were used to improve design and visual appearance of the engine components.

Con-Rod re-design occurred and additional features were included, as displayed in *Figure 13.0.2*, *13.0.3* and *13.0.4*.

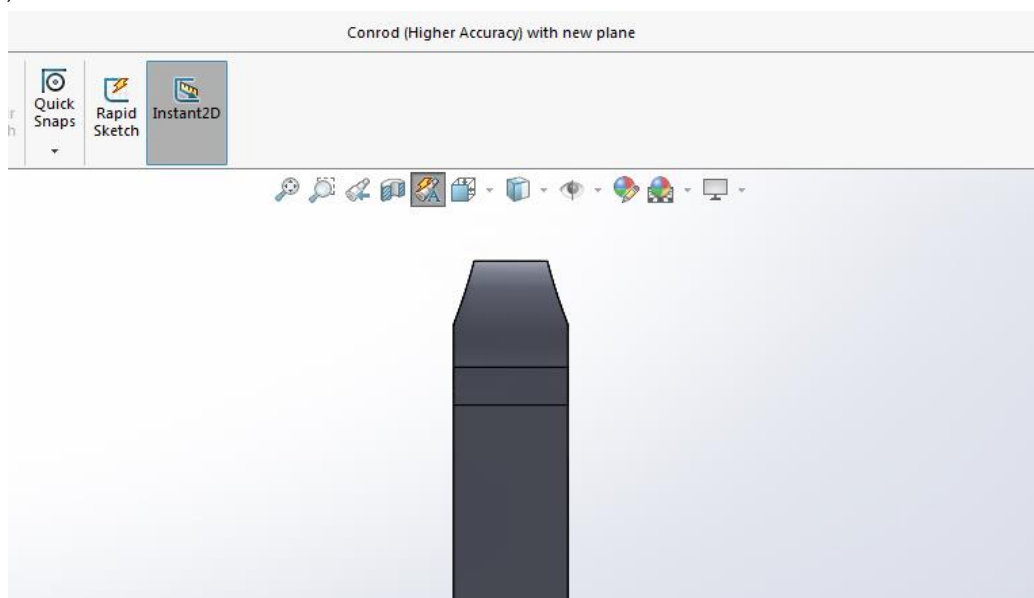


Figure 13.0.2 – Added detail to Re-designed Con-Rod.

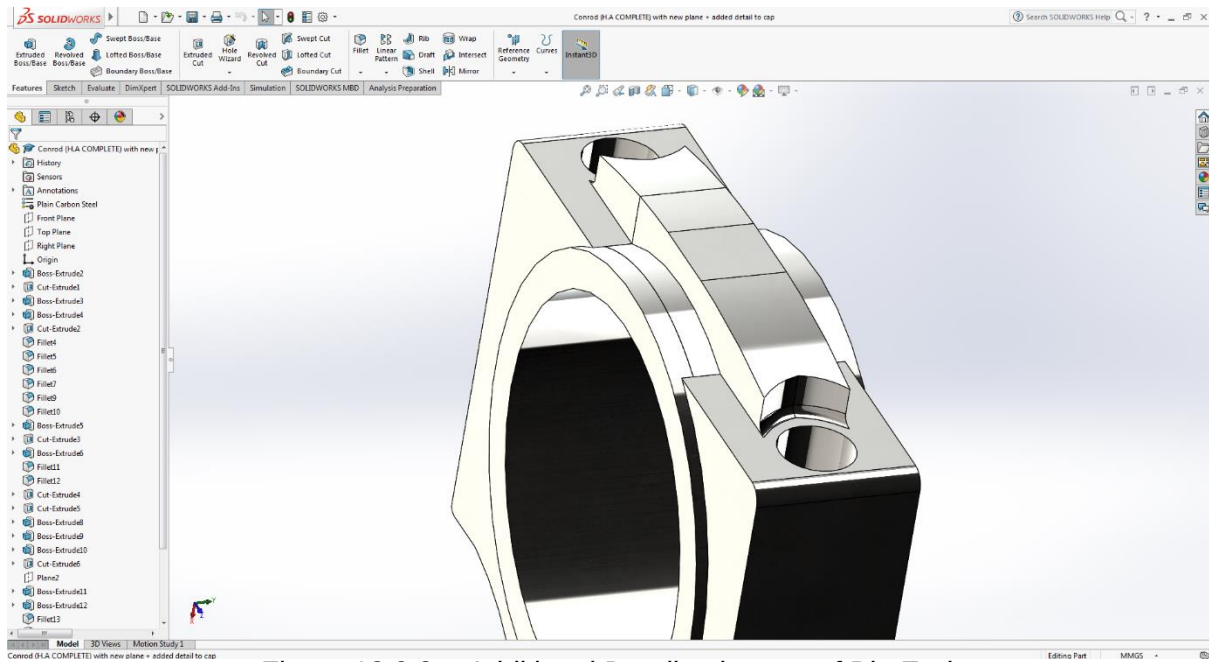


Figure 13.0.3 – Additional Detail to bottom of Big End.

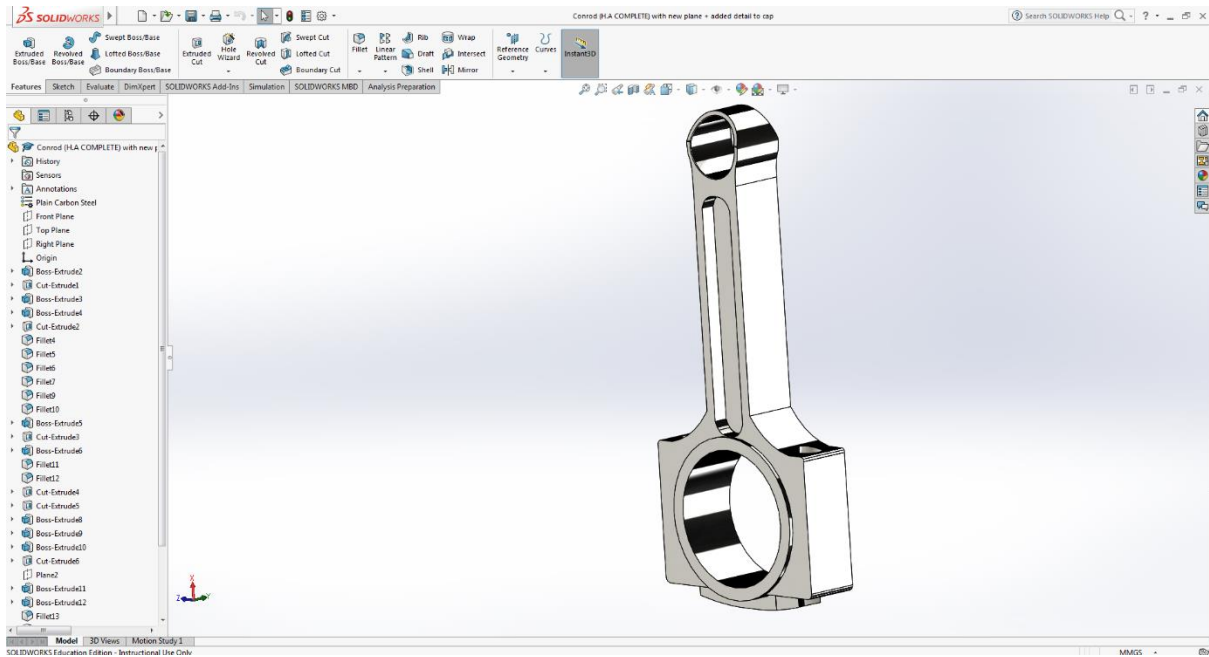


Figure 13.0.4 – Additional Detail to Bottom End.

9.0.1.2 – Ford Fox Turbo Gudgeon Pin:

The Gudgeon pin did not require a re-design, due to having a simplistic design. Therefore only slight areas of additional detail were required, shown in *Figure 13.0.5*.

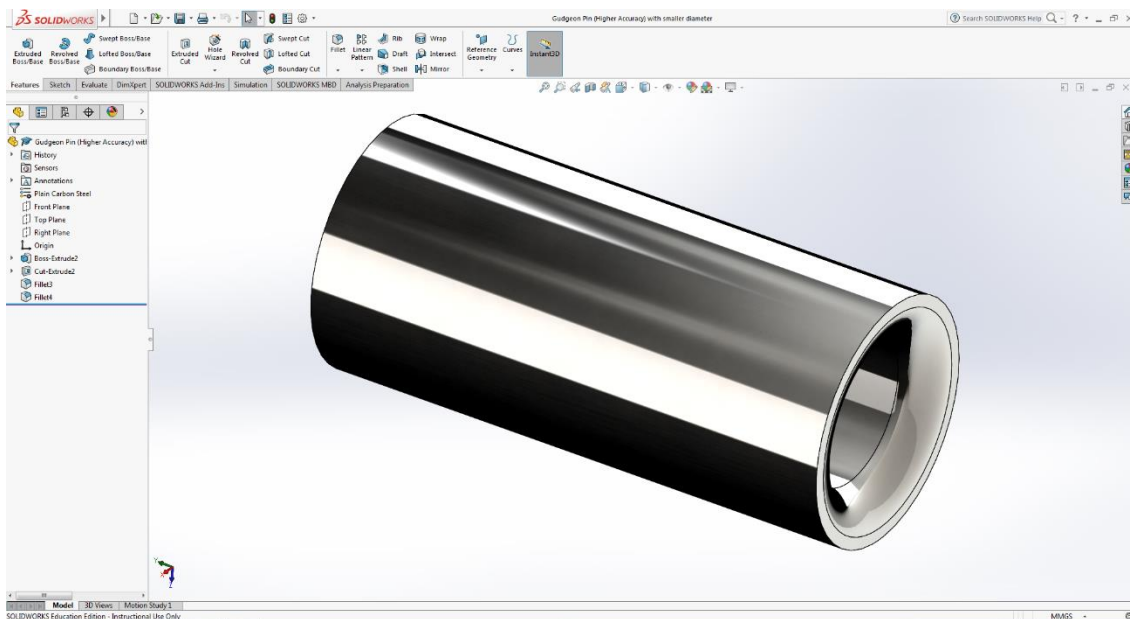


Figure 13.0.5 – Gudgeon Pin.

9.0.1.3 – Ford Fox Piston:

Finally; the piston required a full re-design. Additional features were added to the piston crown. Due to increased familiarity with Solid Works, since the first Ford Fox assembly model was produced for the Interim VIVA, understanding of specific tools and realisation of a simplistic sketch enabling an improved model quality occurred.

Consequently, an improved piston crown was produced, displayed in *Figure 13.0.6*.

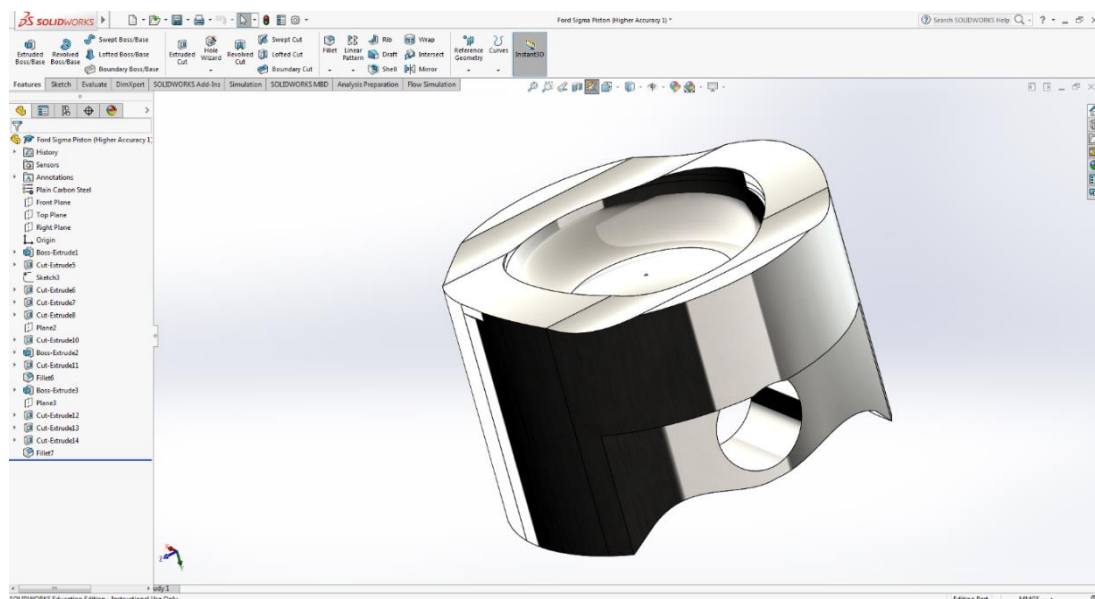


Figure 13.0.6 – Improved Piston Crown.

Further improvements occurred on the underside of the piston, with the production of a further realistic piston skirt and underside design, displayed in *Figure 13.0.6.1*.

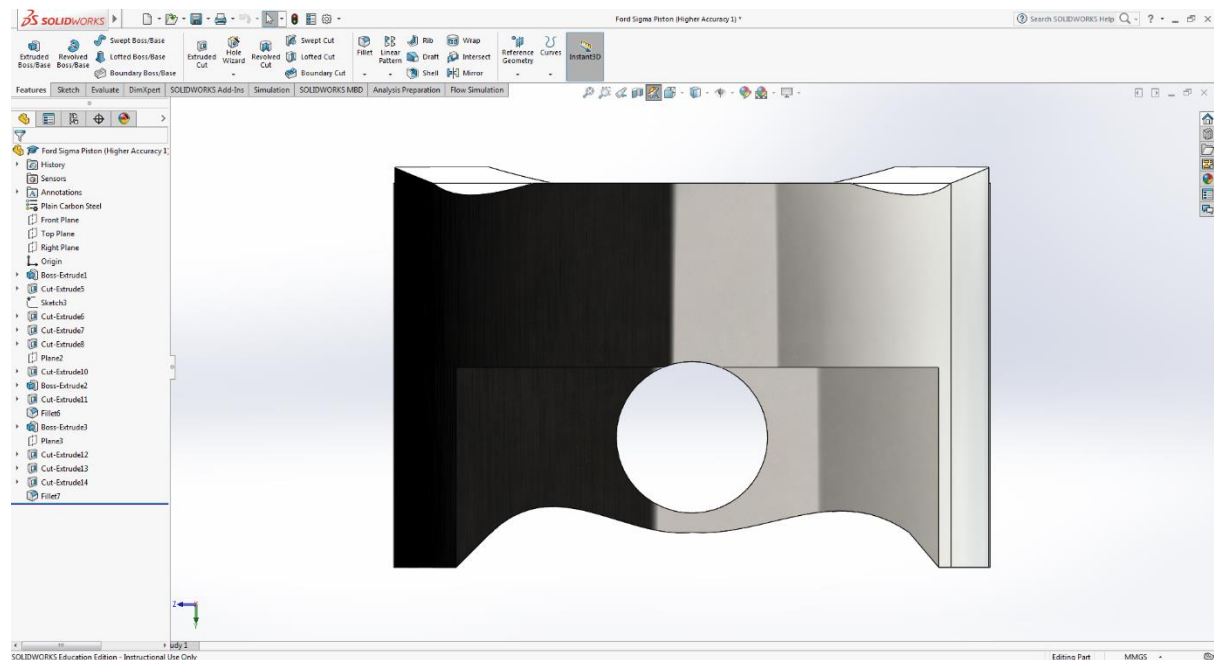


Figure 13.0.6.1 – Improved Piston Skirt and Underside design.

An identical highly detailed design was produced for the underside of the piston, presented in *Figure 13.0.6.2*

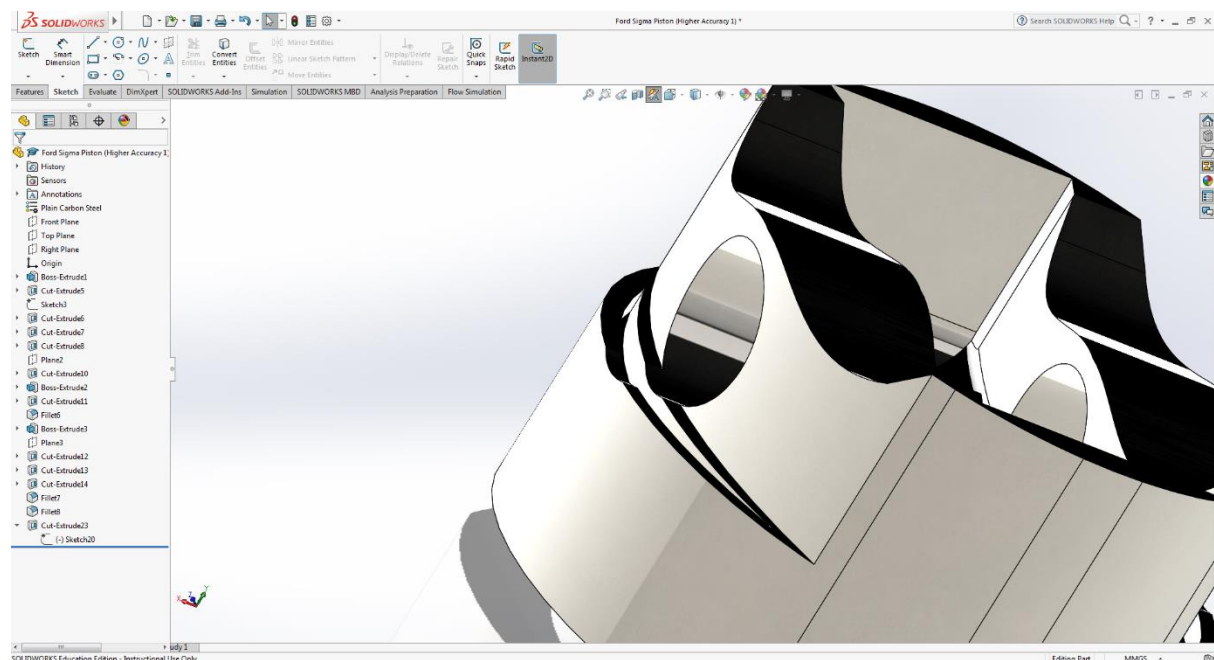


Figure 13.0.6.2 – Highly detailed Underside of Piston.

9.0.1.4 – Assembly:

Finally, all components produced above were inserted into an assembly document and via the use of the mating tool, were assembled into a representative construction, displayed in *Figure 13.0.7*.

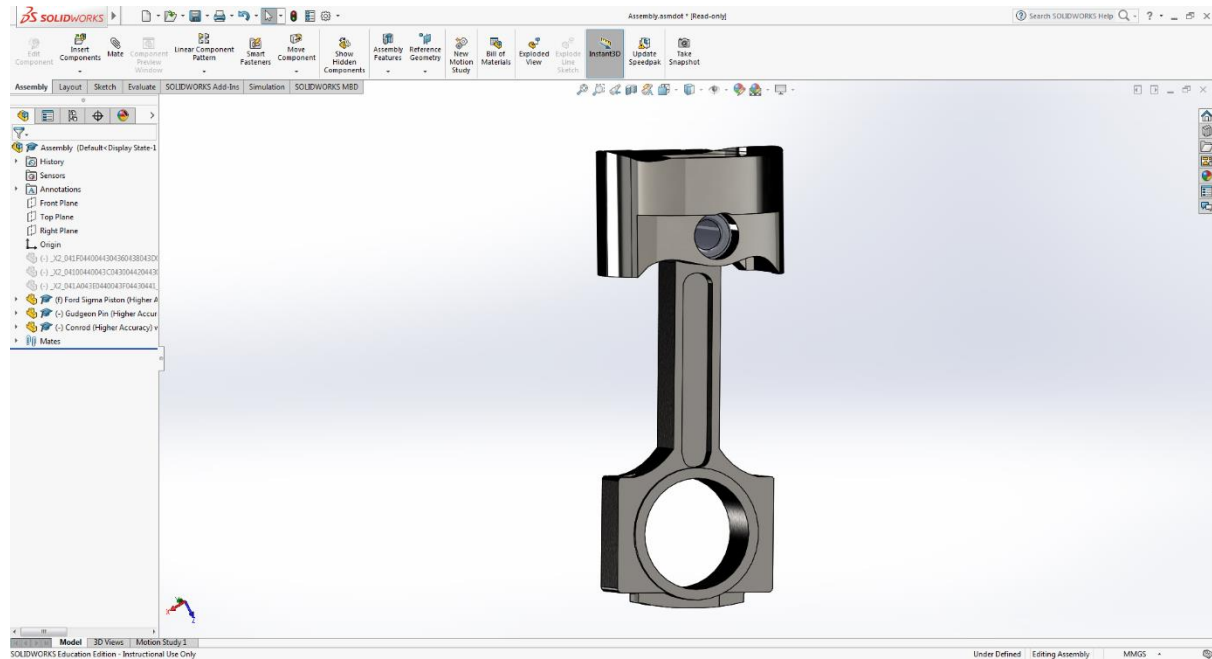


Figure 13.0.7 – Ford Fox Assembly.

9.0.2 – Ford Fox Crankshaft:

Once the Ford Fox piston, Gudgeon pin and con-rod assembly is complete, the crankshaft is now able to be produced.

The crankshaft is created via replicating five main parts and attaching them via the mating tool and consequently producing the final crankshaft assembly. Firstly, the crankshaft flywheel mounting flange was produced, presented in *Figure 13.0.8*.

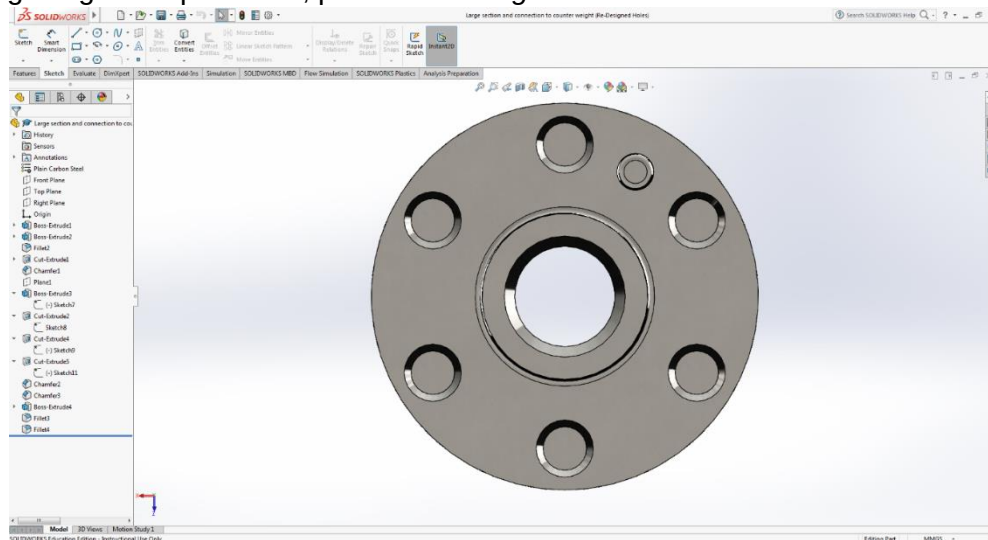


Figure 13.0.8 – Flywheel Mounting Flange.

Two different diameter main journals were then required. The smaller main journal including oil passage is produced in *Figure 13.0.8.1*.

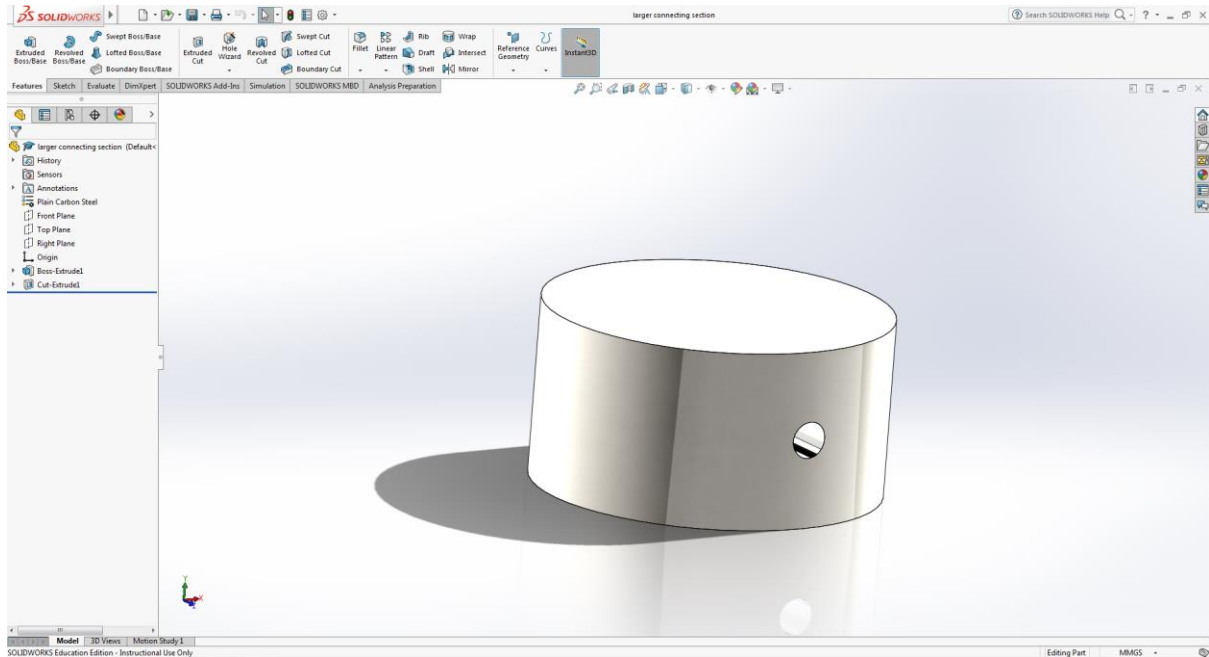


Figure 13.0.8.1 – Small Diameter Main Journal.

The larger diameter main journal including oil passage was created in *Figure 13.0.8.2*.

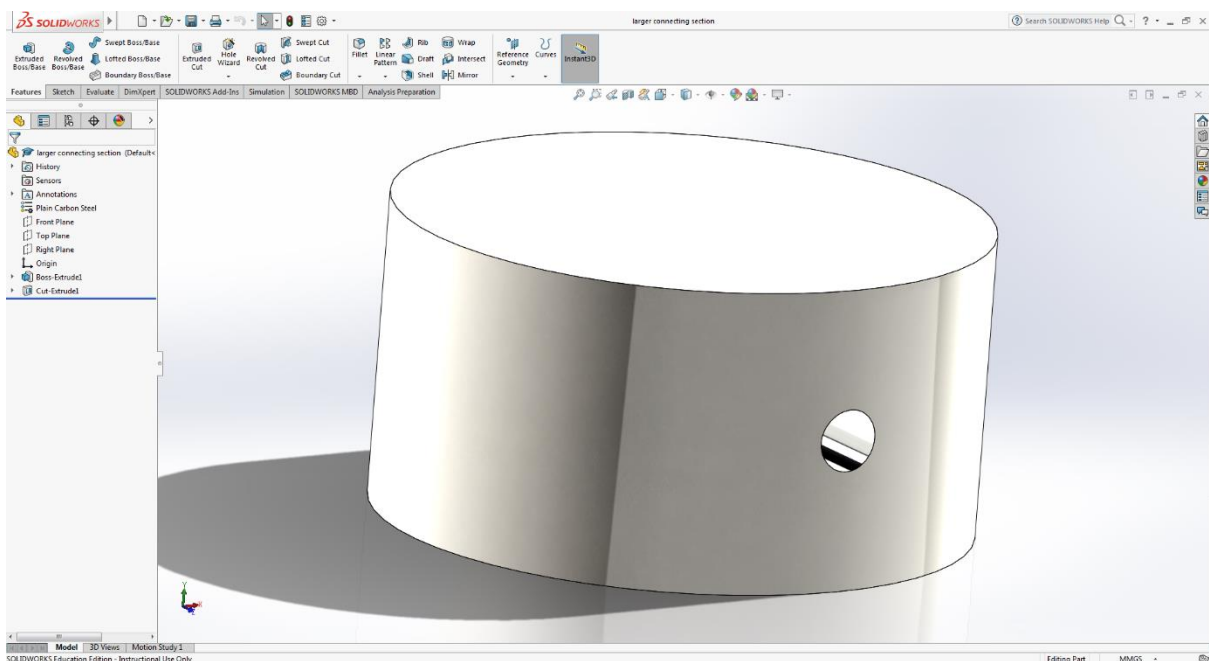


Figure 13.0.8.2 – Large Diameter Main Journal.

The large counter-web was produced, which consisted of a highly detailed complex geometry, production is displayed in Figure 13.0.8.3.

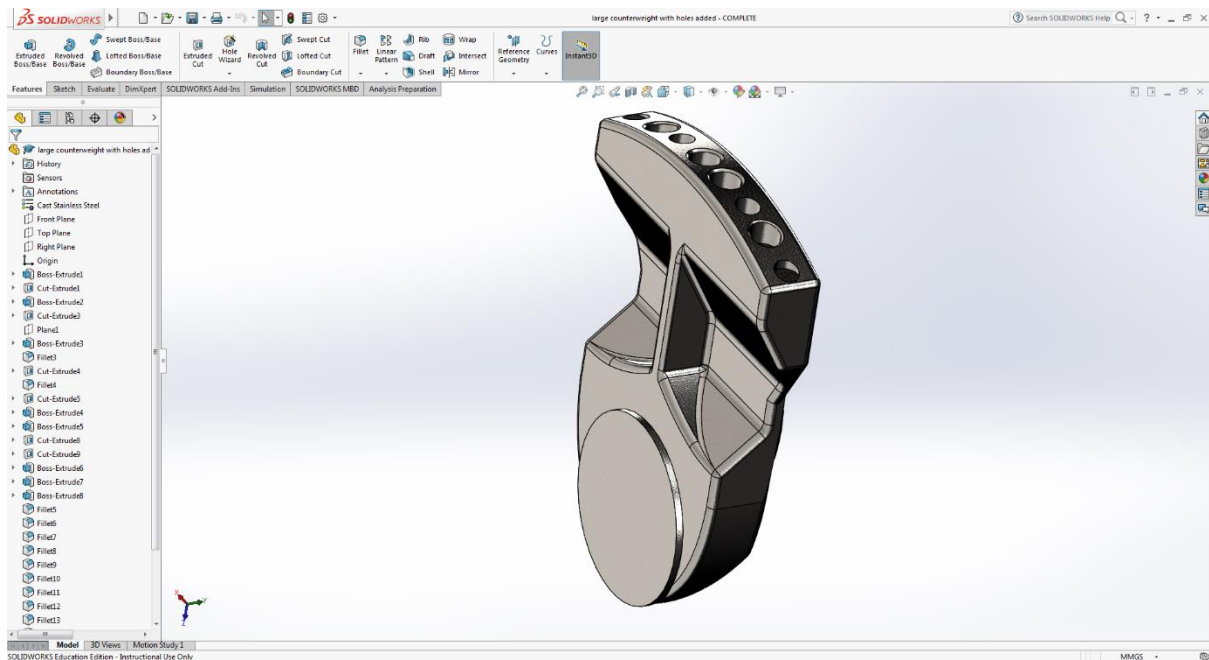


Figure 13.0.8.3 – Large Counter-web.

Finally; the small counter-web was produced next and consisted of a much simpler design, presented in Figure 13.0.8.4.

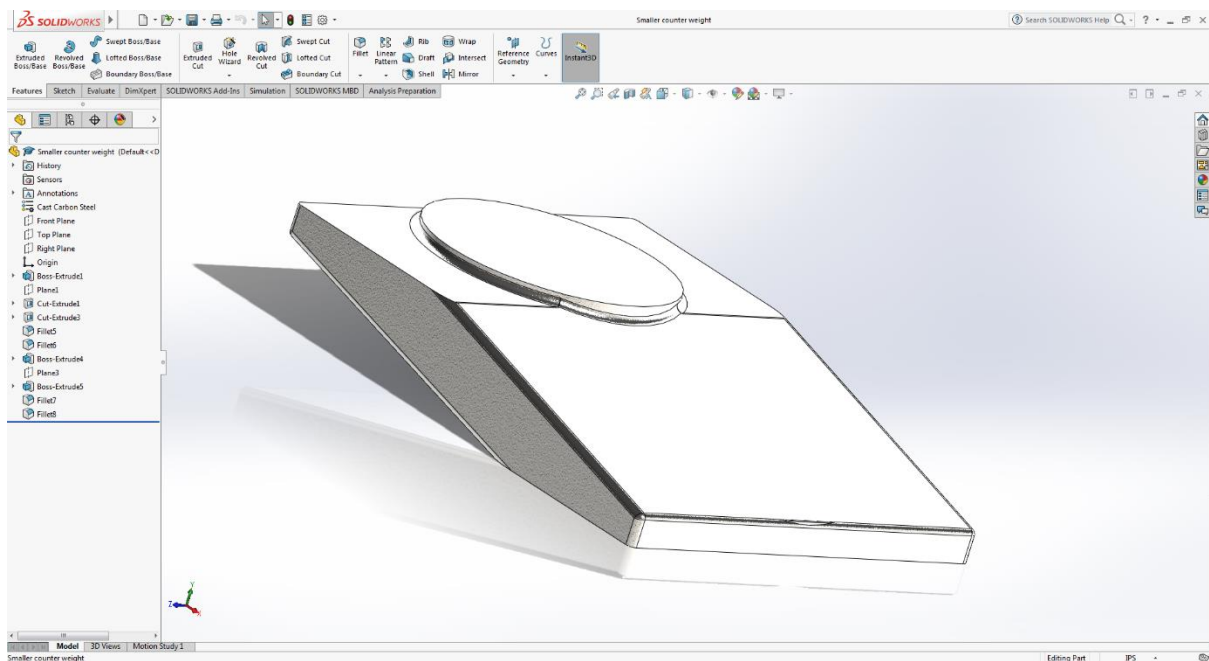


Figure 13.0.8.4 – Small Counter-web.

Consequently; all the individual components for the crankshaft listed above were joined via the use of Solid Works mating tool and assembled. However, replication of each component was required to produce the entire final crankshaft assembly, which is displayed in *Figure 13.0.8.5*.

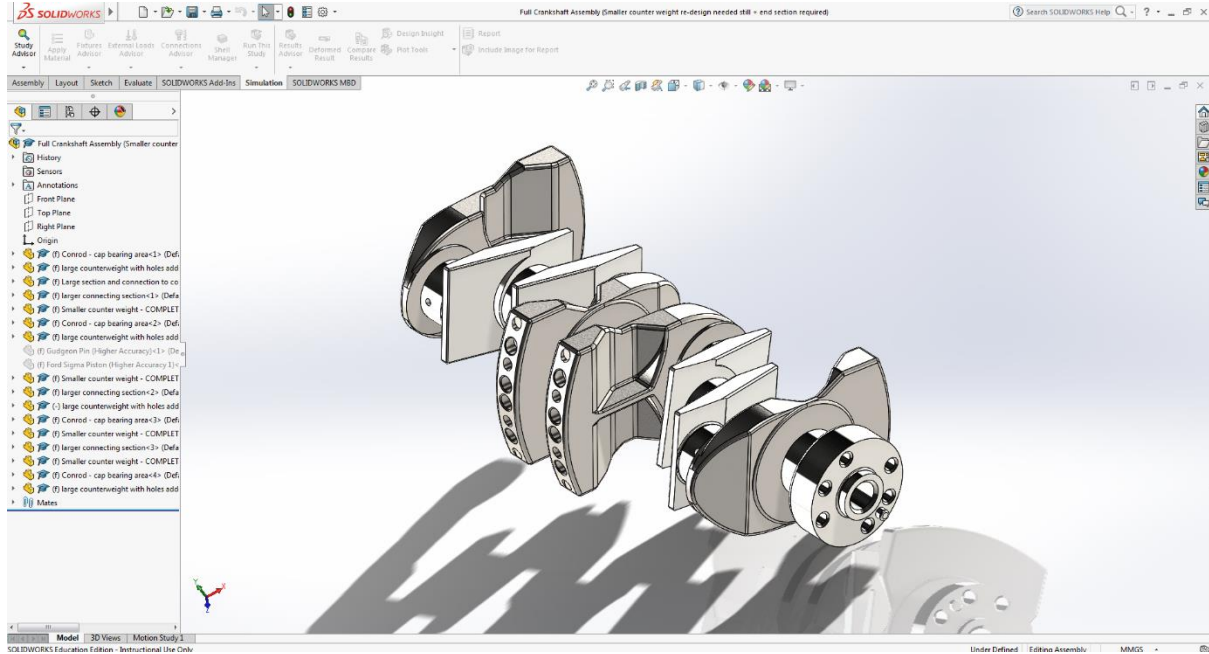


Figure 13.0.8.5. – Final Crankshaft Assembly.

The concluding step consisted of the final completed crankshaft assembly, plus the Ford Fox piston, Gudgeon pin and con-rod construction, inserted into a Solid Works assembly to create the final product, as displayed in *Figure 13.0.8.6 and 13.0.8.7*.

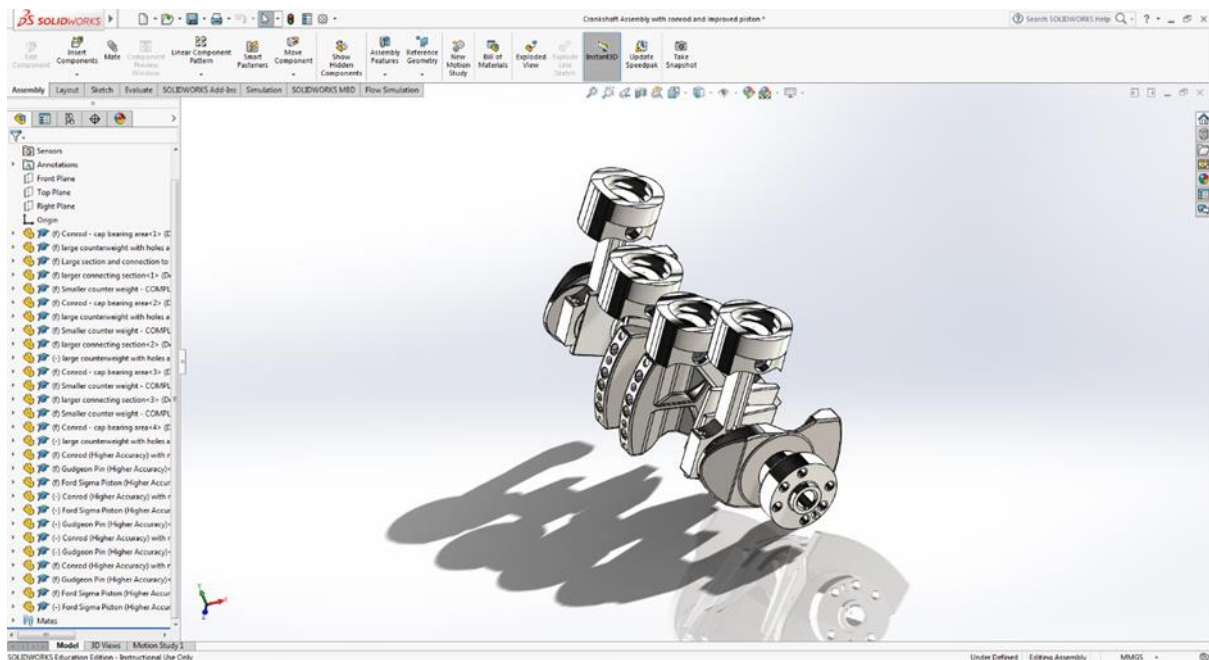


Figure 13.0.8.6 – Ford Fox Final Product.

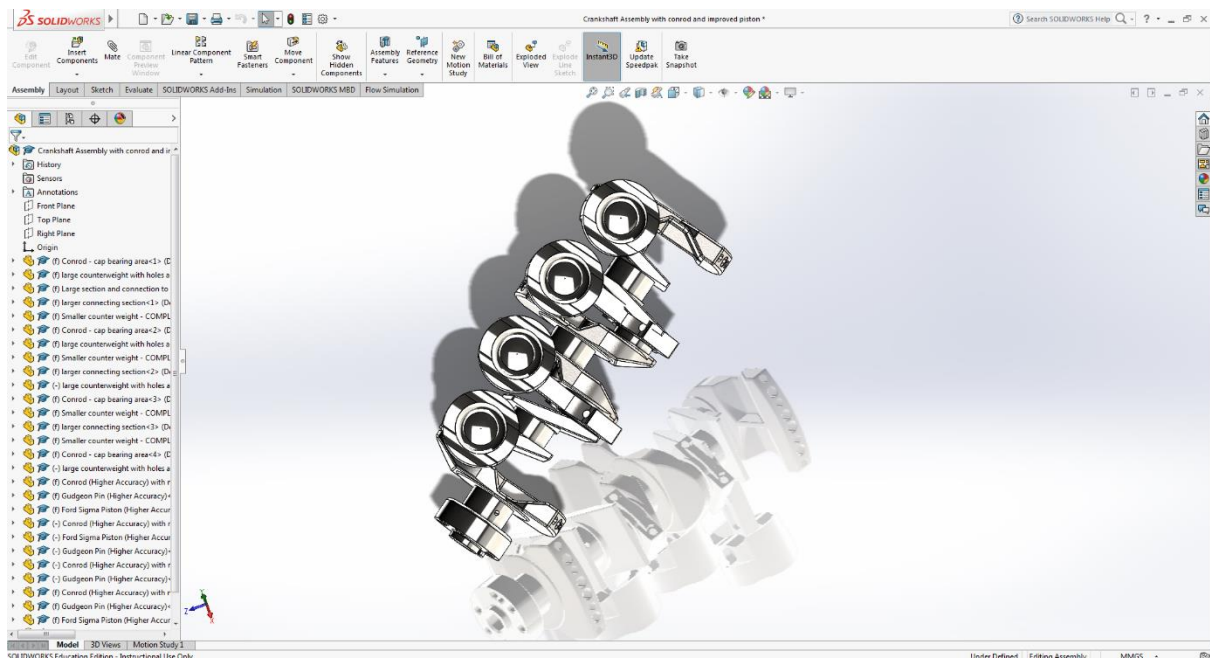


Figure 13.0.8.7 – Ford Fox Final Product (Aerial Perspective).

Consequently, a realistic, highly detailed Ford Fox 1.6 Turbo Crankshaft, piston, Gudgeon pin and con-rod assembly is produced.

9.0.3– Ford Sigma Naturally Aspirated Solid Works:

The Ford Sigma Assembly was collected from Malcolm McDonald and the Solid Works modelling began.



Figure 14.0 – Ford Sigma Assembly (Jay, 2017)

9.0.3.1 – Ford Sigma Connecting Rod:

Once physical measurements were obtained, a highly detail Ford Sigma Con-Rod was produced, displayed in *Figure 14.0.1*.

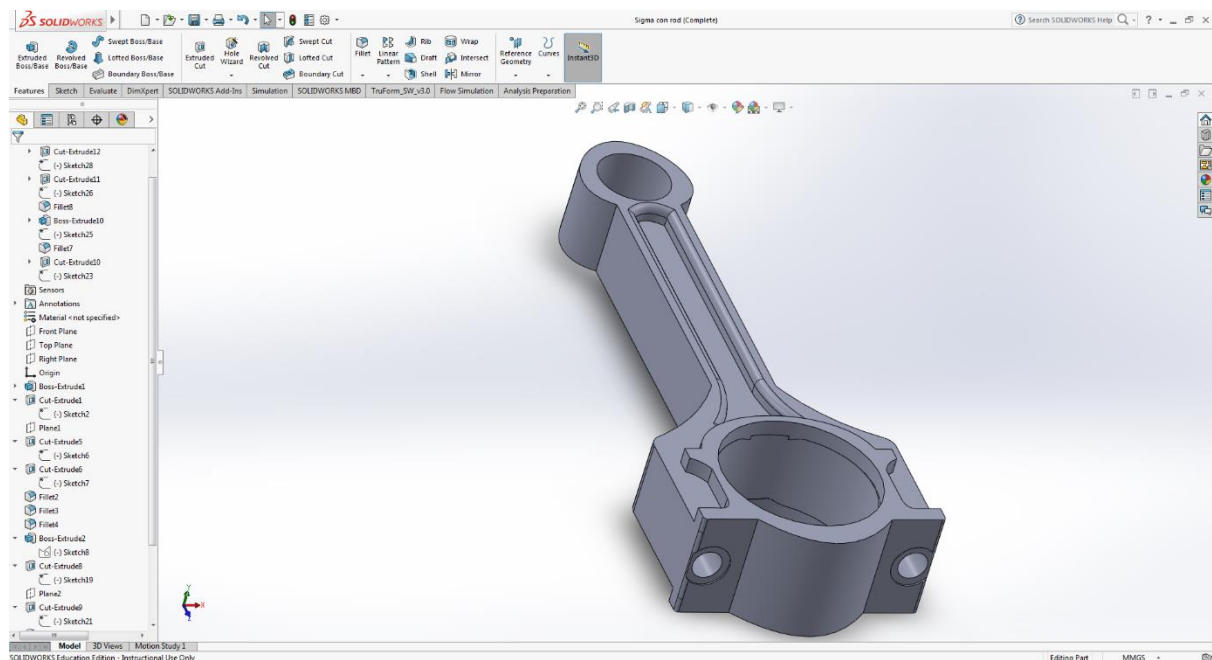


Figure 14.0.1 – Ford Sigma Con-Rod.

9.0.3.2 – Ford Sigma Gudgeon Pin:

The Ford Sigma Gudgeon Pin was then produced, shown below in *Figure 14.0.2*. However due to being an interference fit, the Gudgeon Pin is unable to exit the piston and Con-Rod assembly. An interference fit requires a press to force the Gudgeon pin into place.

Therefore, the digital Vernier callipers were used to measure the Gudgeon pin as accurately as possible. However minor measurement assumptions were produced when creating the Solid Works model.

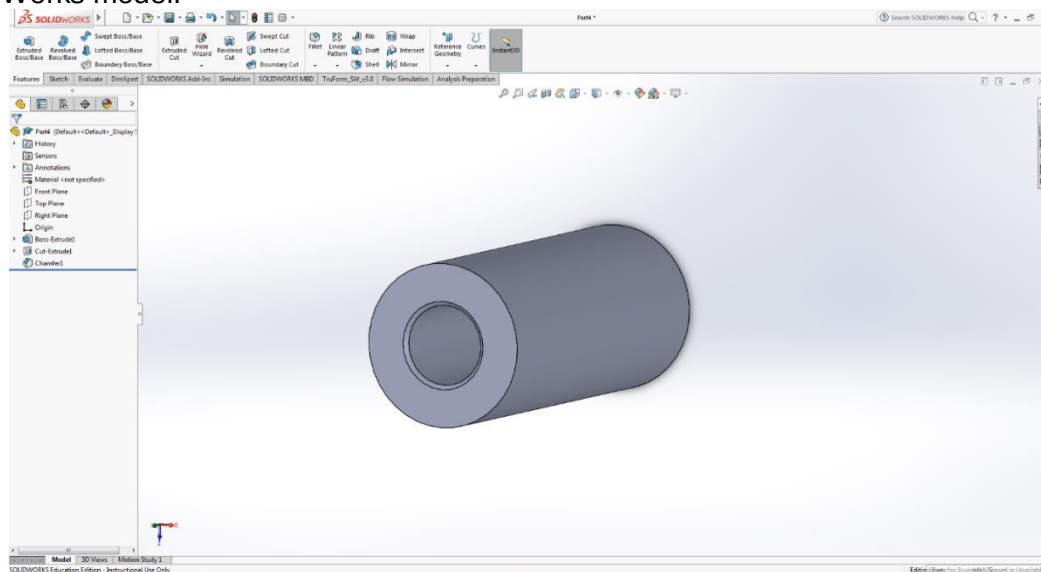


Figure 14.0.2 – Ford Sigma Gudgeon Pin.

9.0.3.4 – Assembly:

The individual engine components produced were then placed into a Solid Works assembly and combined to construct a piston, Gudgeon pin and con-rod assembly, displayed below in *Figure 14.0.5*.

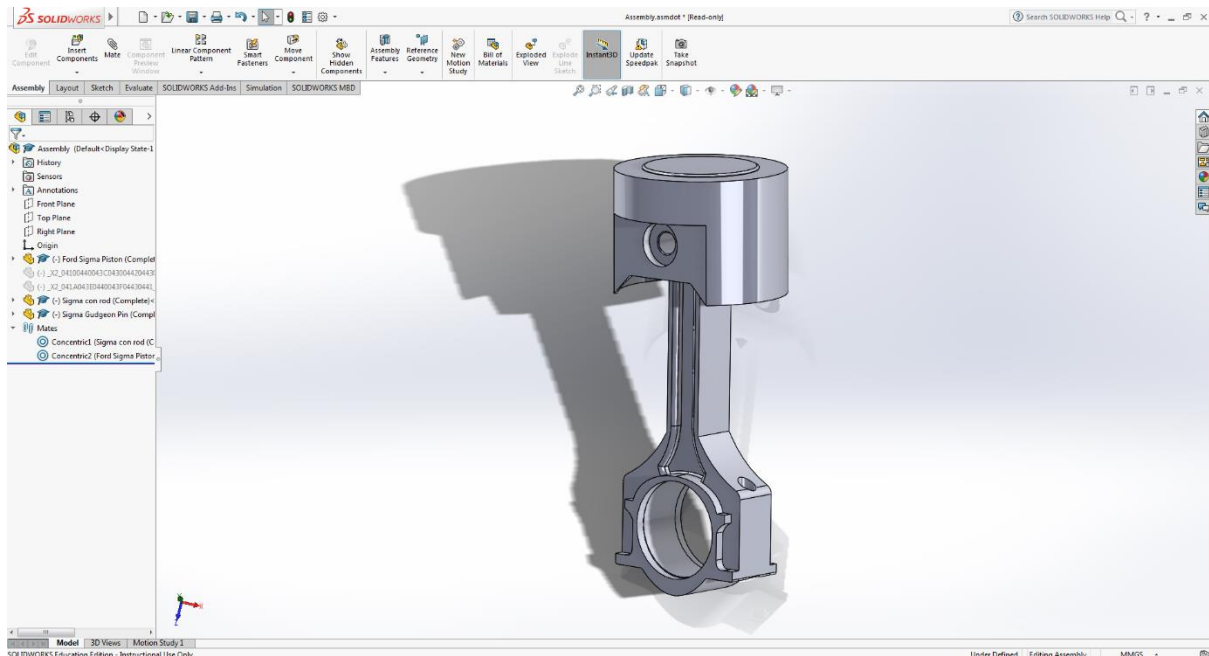


Figure 14.0.5 – Ford Sigma Piston, Gudgeon Pin and Con-Rod Assembly.

9.0.4 – Wankel Rotary 13b Renesis Engine Excel Solid Works:

Finally, a Solid Works model of the Wankel Rotary engine rotor was produced via a basic sketch.

9.0.4.1 – Rotor:

The centre diameter of the rotor and the angle between rotor tips were measured. Therefore, enabling production of the basic shape of the rotor via the Spline tool, presented in *Figure 15.0.1*.



Figure 15.0 – Rotor. (Jay,2017)

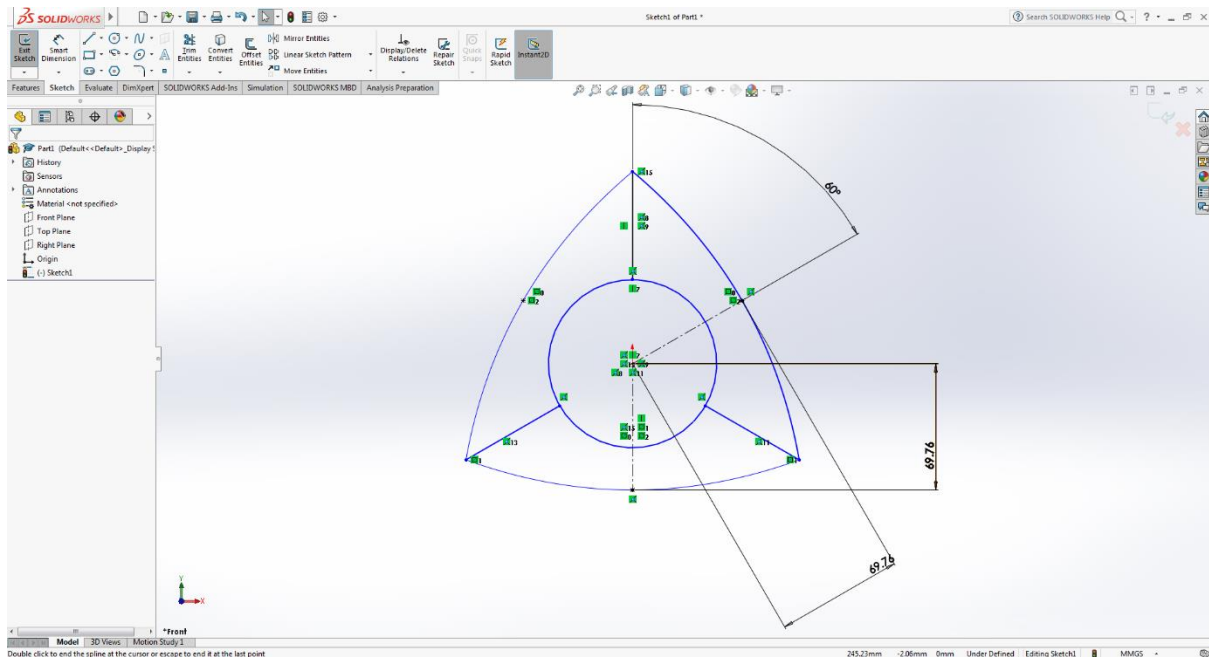


Figure 15.0.1 – Basic Rotor Sketch.

Consequently, the rotor gear was produced via the selection of the circular sketch pattern via the Linear Sketch Pattern tool and extruded, as displayed in Figure 15.0.2.

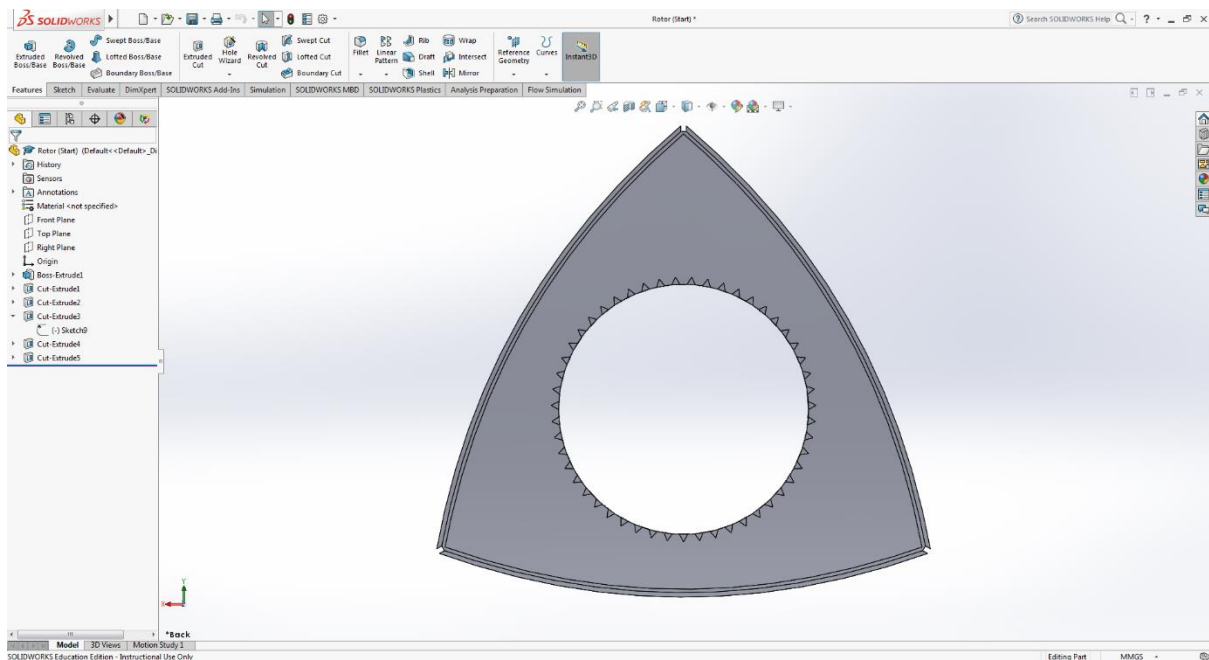


Figure 15.0.2 – Wankel Rotary Engine Rotor.

Sketches were then produced and extruded to support Apex and Face seal slots, shown in *Figure 15.0.3* and *Figure 15.0.4*.

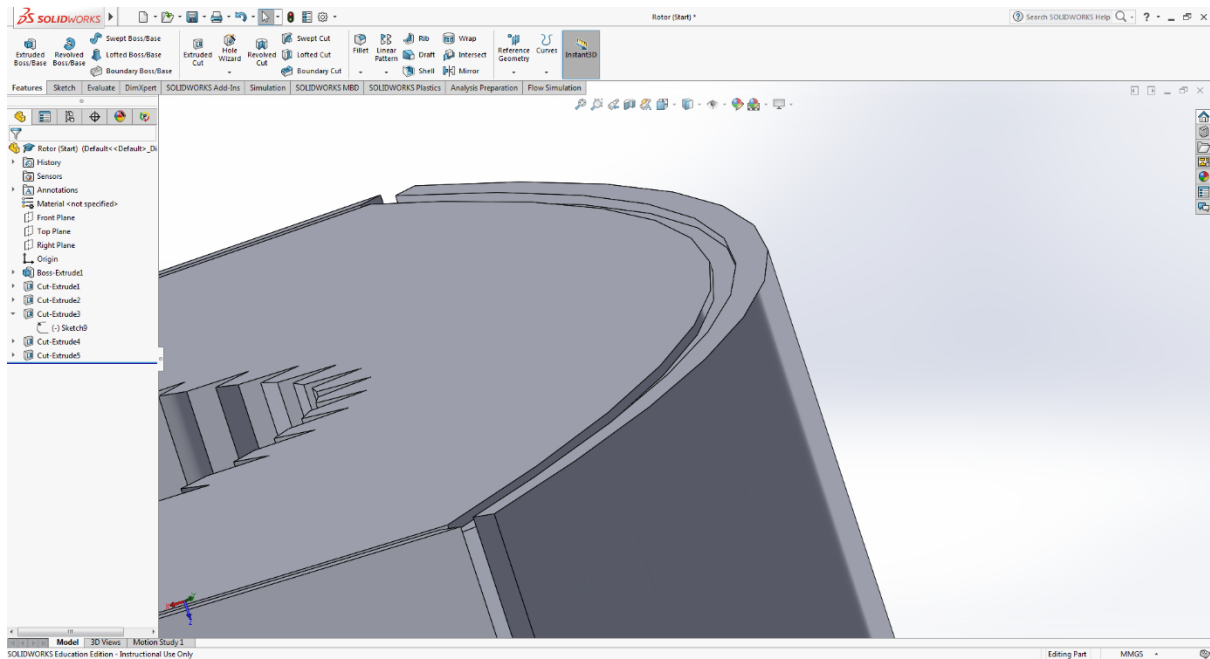


Figure 15.0.3 – Face Seals.

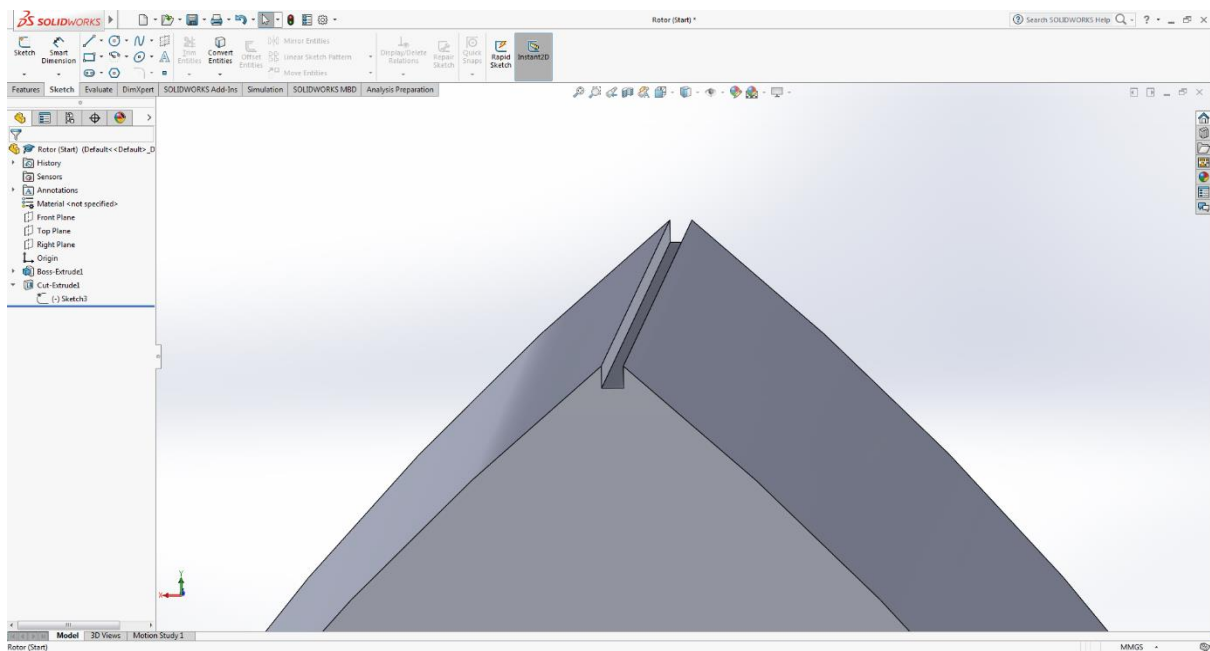


Figure 15.0.4 – Apex Seals.

Further features were then added to the gear and rotor face, shown in *Figure 15.0.5*.

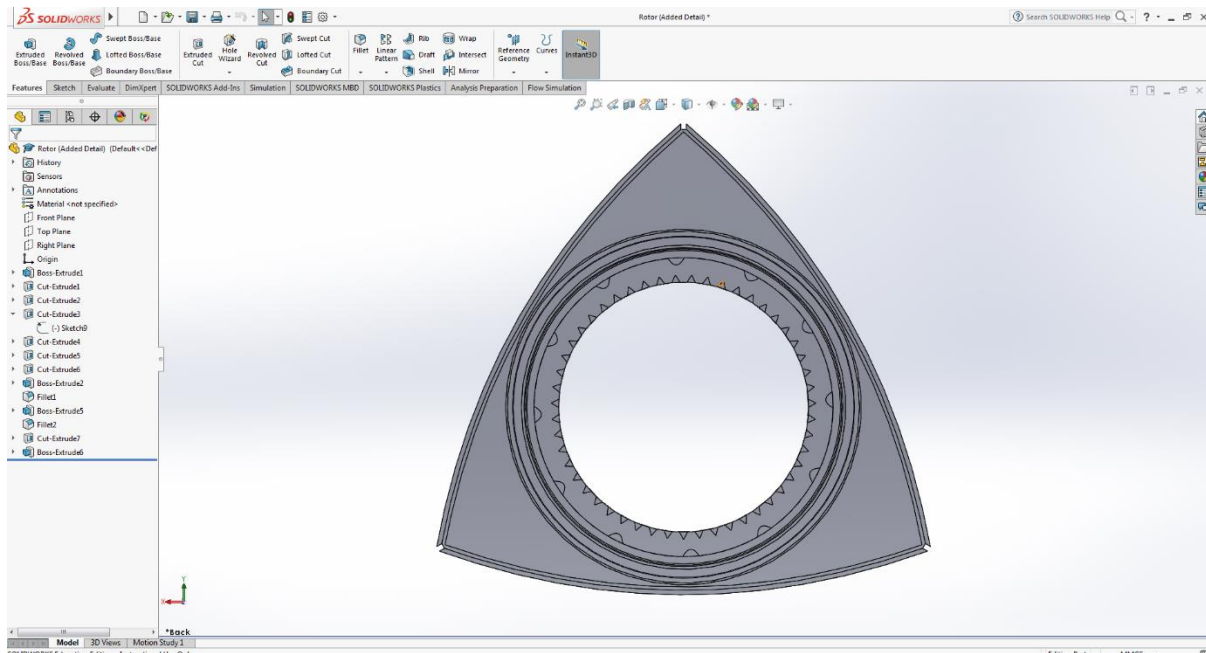


Figure 15.0.5 – Rotor gear and face additional features.

Additional planes were then added to produce the combustion cavities for the rotor, displayed in *Figure 15.0.6*.

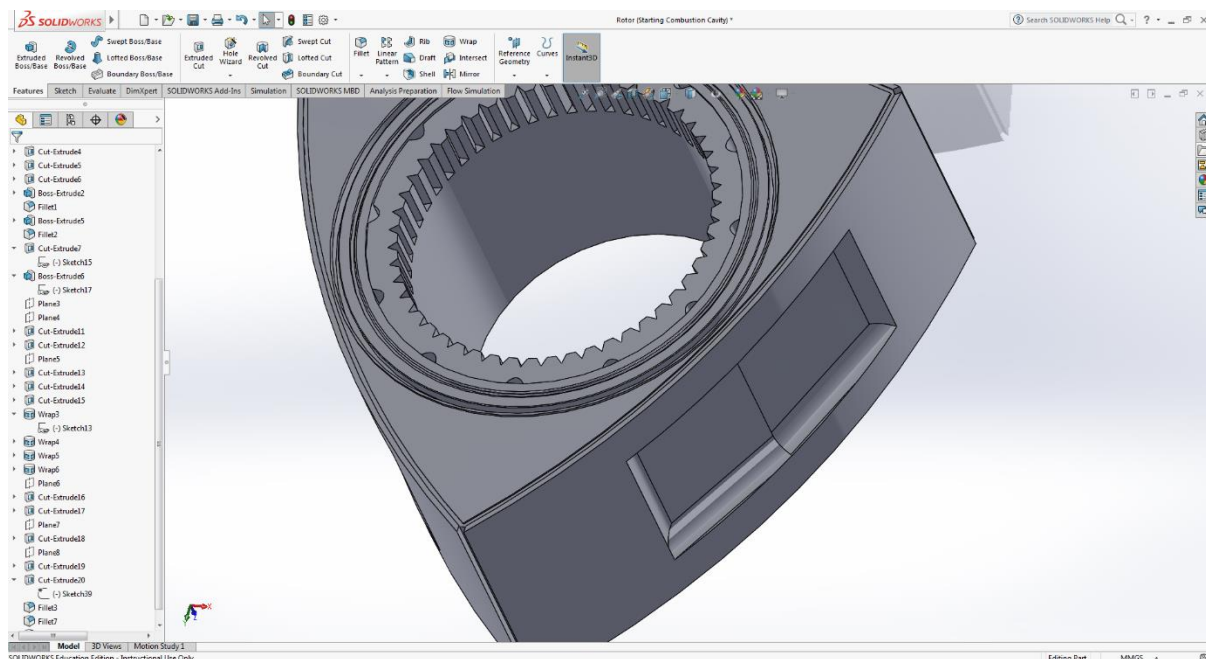


Figure 15.0.6 – Combustion Cavity.

The rotor was replicated, as two rotors were required for the Twin-Rotor 13b Renesis engine design.

9.0.4.2 – Eccentric Shaft:

Finally; the eccentric shaft was required for the assembly. Therefore, several separate models were produced via measurements taken, inserted into an assembly and connected via mates.



Figure 16.0 – Eccentric Shaft. (Jay, 2017)

The eccentric shaft, otherwise known as an output shaft has two lobes mounted eccentrically, pictured in *Figure 16.0*. Consequently, the lobes are offset from the centre line of the shaft. The rotors attach to these lobes and as they rotate, push inwards on the lobes, consequently producing torque. (How Stuff Works Auto, 2017)

Creation of separate models for output shaft then began. Firstly, the front section of the main journals was modelled, including oil passages for lubrication and key way for timing purposes, presented in *Figure 16.0.1*. The front section has components which are attached, including; balance weight, gear for auxiliary drive and V-Belt. (Mazda, 2003)

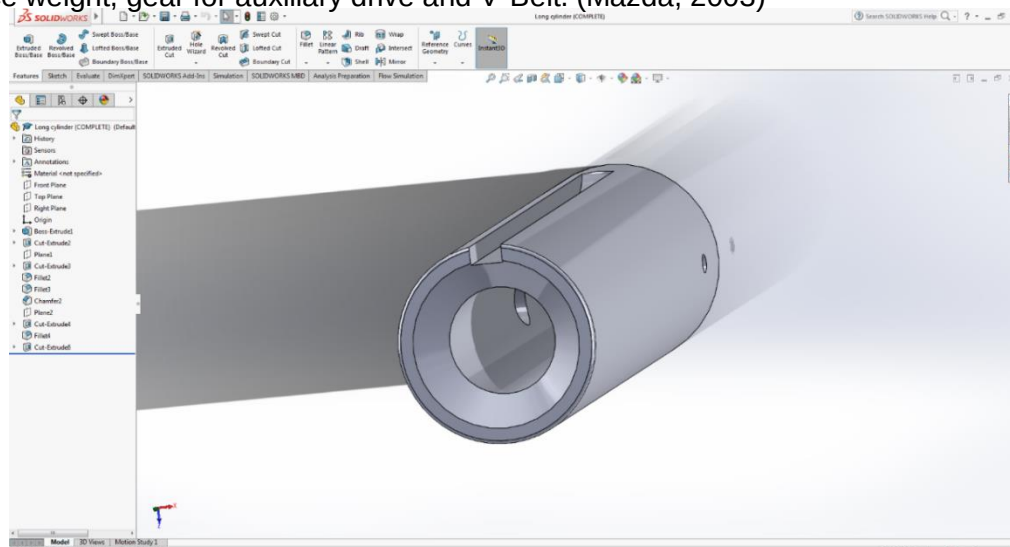


Figure 16.0.1 – Pulley Section of Main Journals.

The connecting cylinder journal was then created in *Figure 16.0.2*, including oil passage.

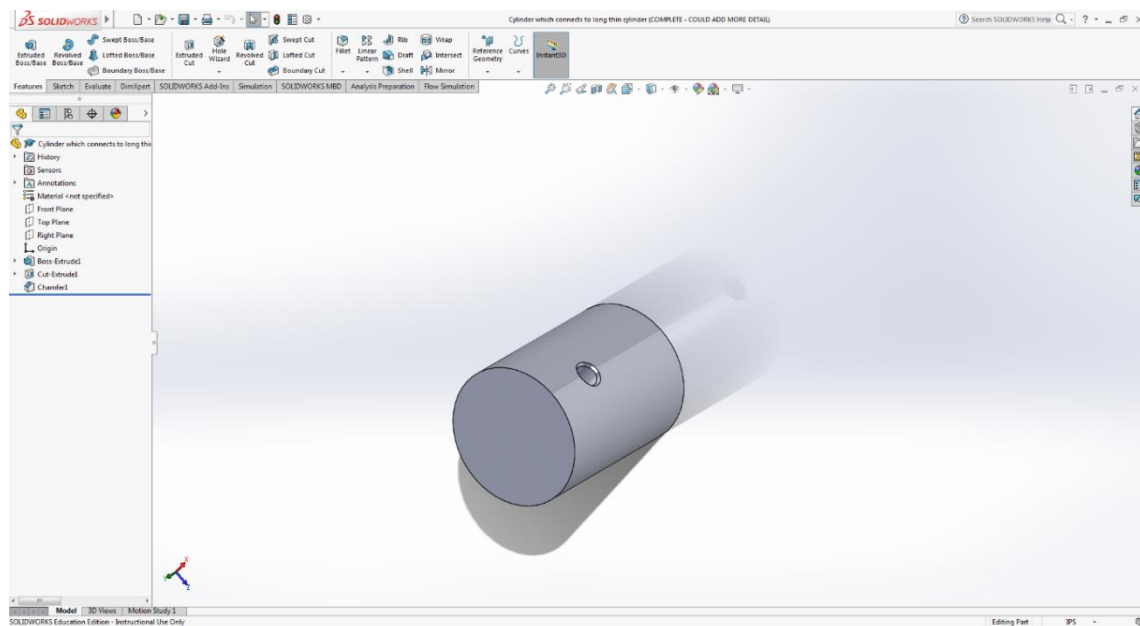


Figure 16.0.2 – Connection Cylinder Journal between Pulley section and Rotor Bearing.

The Rotor Lobe was produced, including all oil passages, shown in *Figure 16.0.3*.

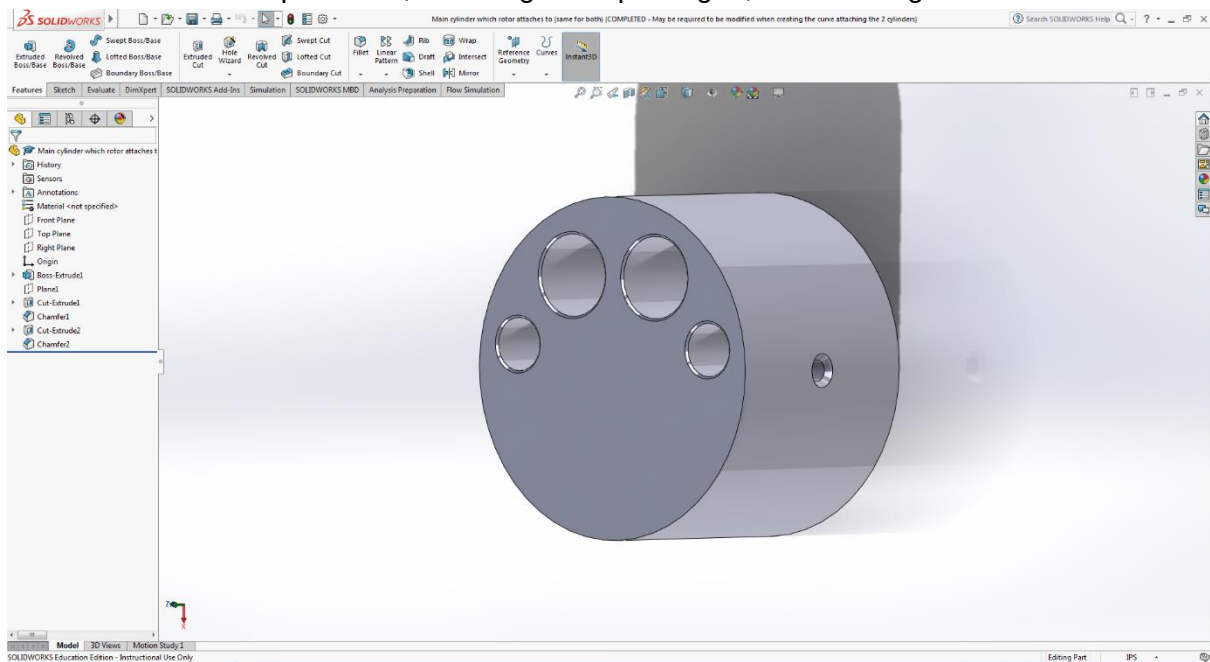


Figure 16.0.3 – Rotor Lobe.

The rear of the eccentric shaft, connects to the flywheel and the flywheel nut is attached to this section. Consequently, the rear of the main journal was produced in Solid Works, using various extrude cut and extrude boss features. Oil passages were also included in this model, as displayed in *Figure 16.0.4*.

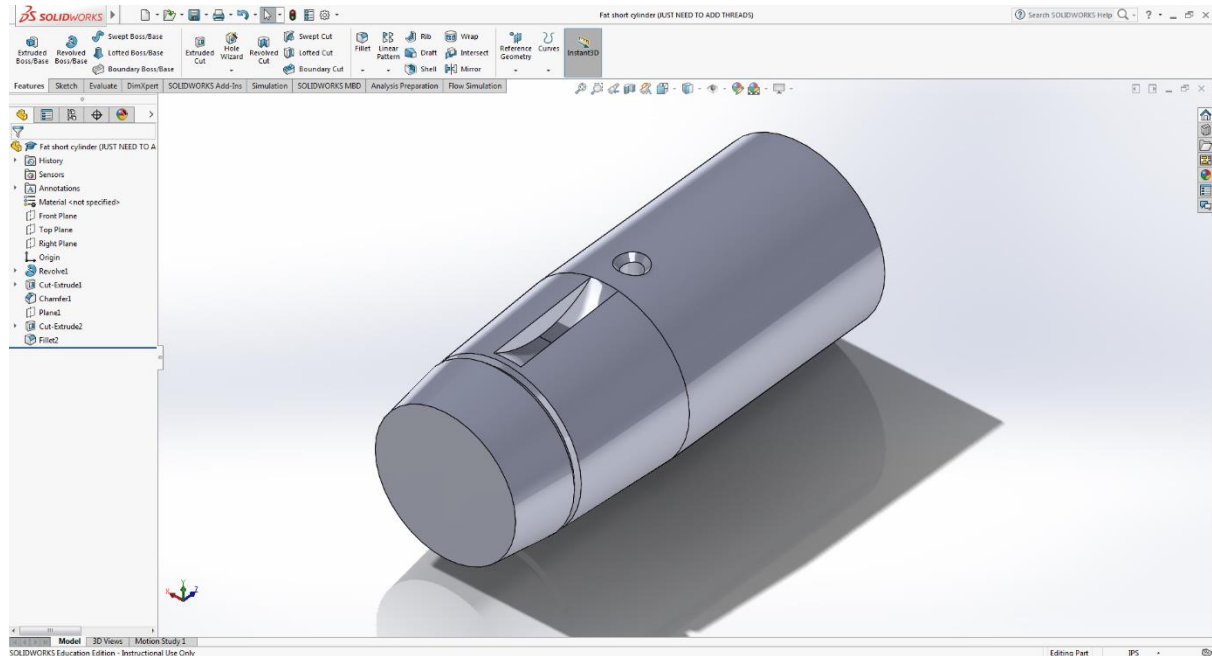


Figure 16.0.4 – Rear Journal.

The Rotor Lobe was then replicated and used in the assembly with all the Solid Works models listed above via the mate selection tool, as presented in *Figure 16.0.5 and 16.0.6*.

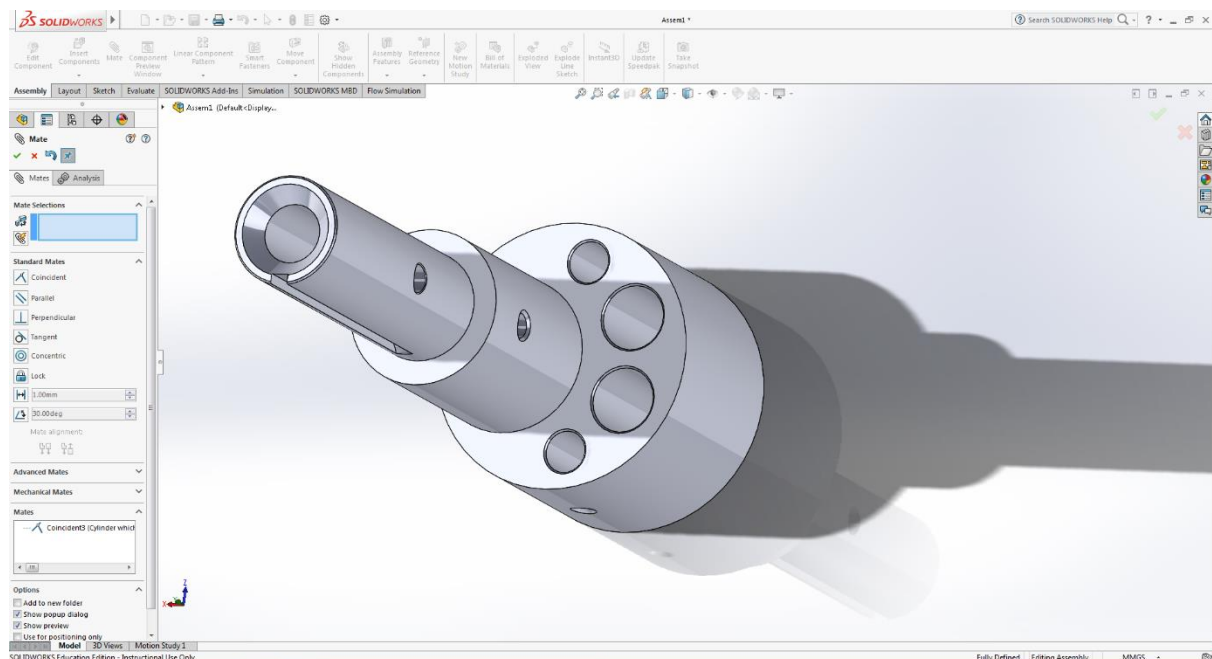


Figure 16.0.5 – First part of Eccentric Shaft Assembly.

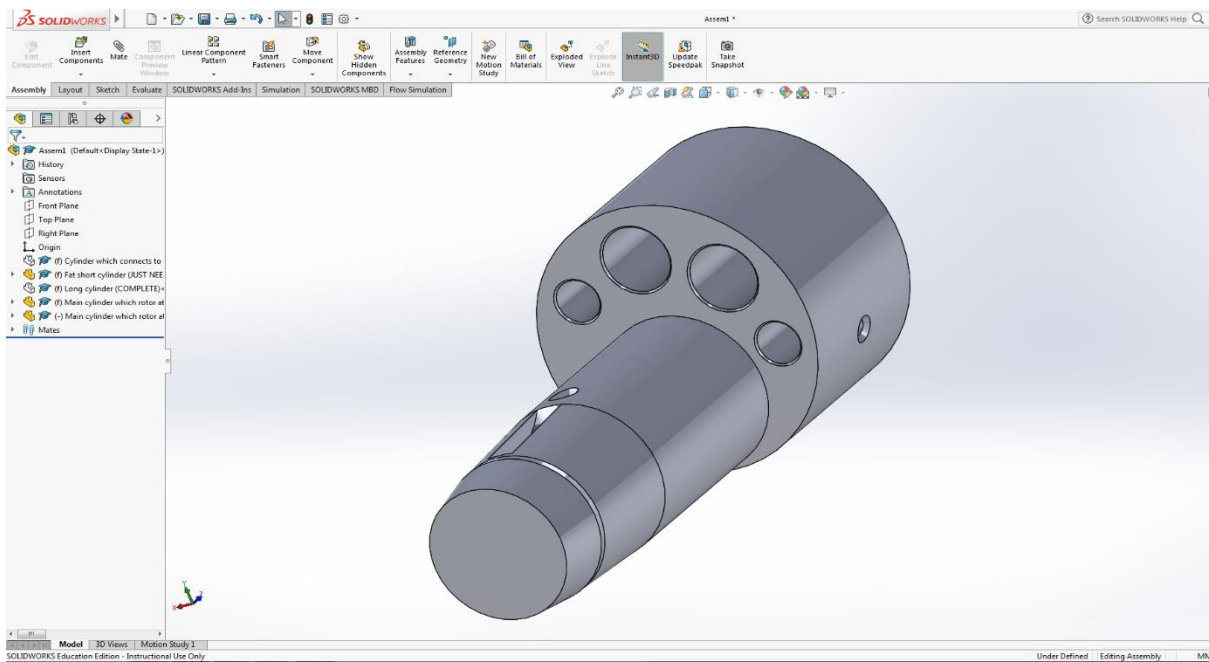


Figure 16.0.6 – Second part of Eccentric Shaft Assembly.

The final section required to be modelled in Solid Works was the centre section of the Eccentric Shaft, as displayed in *Figure 16.0.7*.

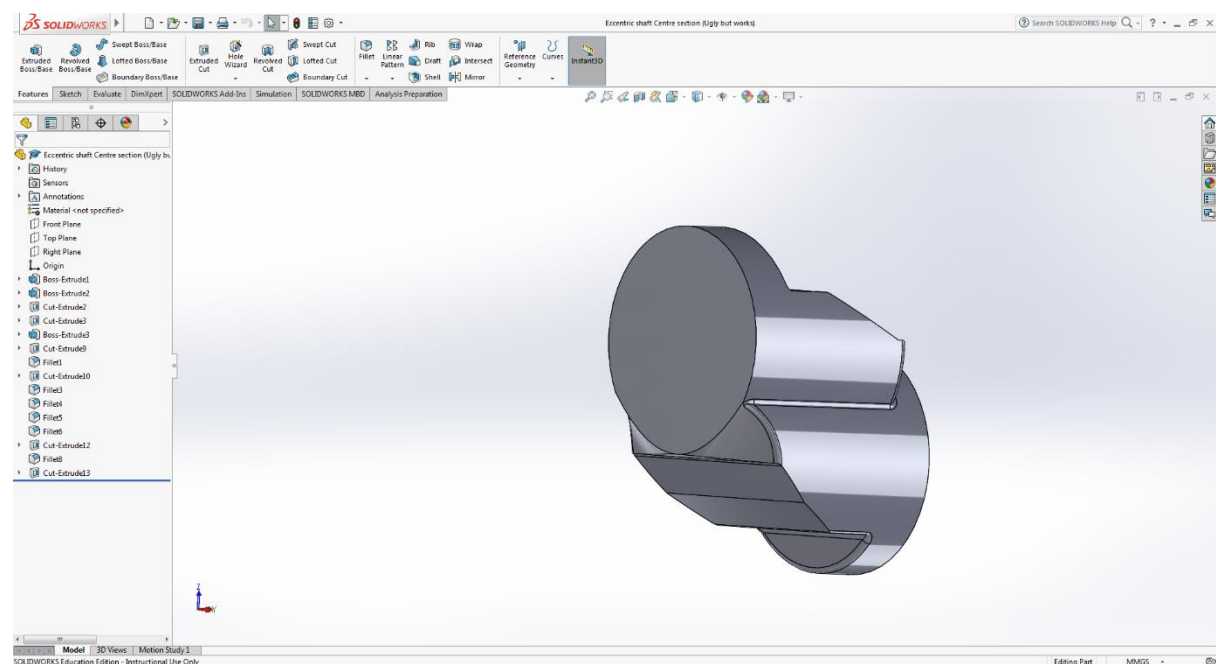


Figure 16.0.7 – Eccentric Shaft Centre Section

Consequently; once all sections of the eccentric shaft were produced and mated together, the following assembly is produced in *Figure 16.0.8*.

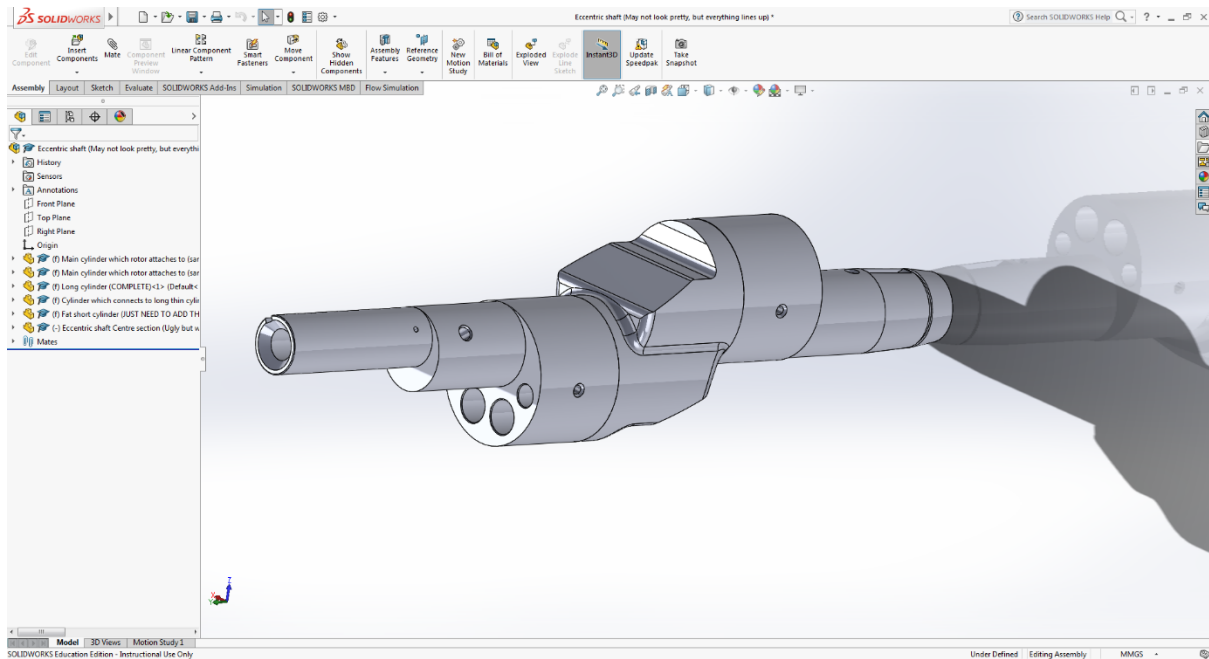


Figure 16.0.8 – Eccentric Shaft Assembly.

9.0.4.3 – Assembly:

Finally; once all components were created and assembled into their required structures, a combined final assembly was produced, including the final model of the eccentric shaft and rotors. As presented below in *Figure 16.0.9*.

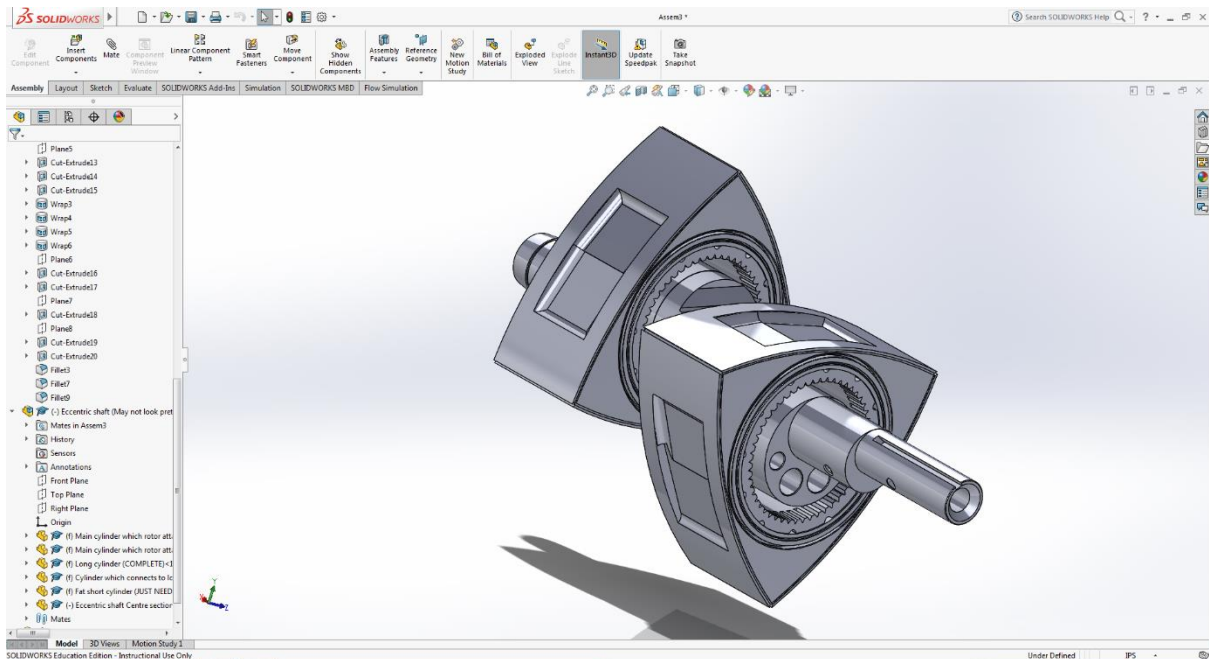


Figure 16.0.9 – Final Wankel Rotary Engine Assembly.

9.0.5 – Chapter References:

1. How Stuff Works Auto 2017, How Rotary Engines Work, Viewed 14th April 2017, <http://auto.howstuffworks.com/rotary-engine2.htm>
2. Mazda Motor of America, 2003, Automotive, Renesis Rotary Engine Fundamentals, USA.

9.0.5.1 – Chapter Literature Review:

1. How Stuff Works Auto – Investigated Eccentric shaft operation.
2. Mazda Motor of America – Obtained information about the Eccentric Shaft Main Journals.

Chapter 10.0 – Finite Element Analysis (FEA):

Solid Works enabled the production of highly accurate engine component models. The models produced in Chapter 9.0 were investigated via the use of Finite Element Analysis (FEA). The use of FEA enabled analysis of structural integrity and performance under load to be completed.

Running a simulation enabled the author to predict areas which have a high likelihood of structural failure, due to excessive stress and strain occurrence. (O'Donnell Consulting Engineers, 2017)

Two separate studies were completed for the Ford Fox piston, con-rod and crankshaft. This was due to an improved simulation with the use of real industrial techniques, with the use of accurate data.

10.0.1 – Ford Fox Crankshaft FEA:

The crankshaft was the first component to undertake a FEA static simulation. Despite the production of an entire crankshaft assembly in Chapter 9.0, half the crankshaft assembly was selected to reduce the likelihood of excessively long run times, as displayed in *Figure 17.0*. The time for a simulation to become completed, is known as the run time. The authors aim is to limit this factor as much as possible, to remain a productive means of testing. Excessively long run times would reduce the effectiveness of this software.

Firstly; the crankshaft assembly was fixed at the flywheel mounting flange and the face of the large counter-web.

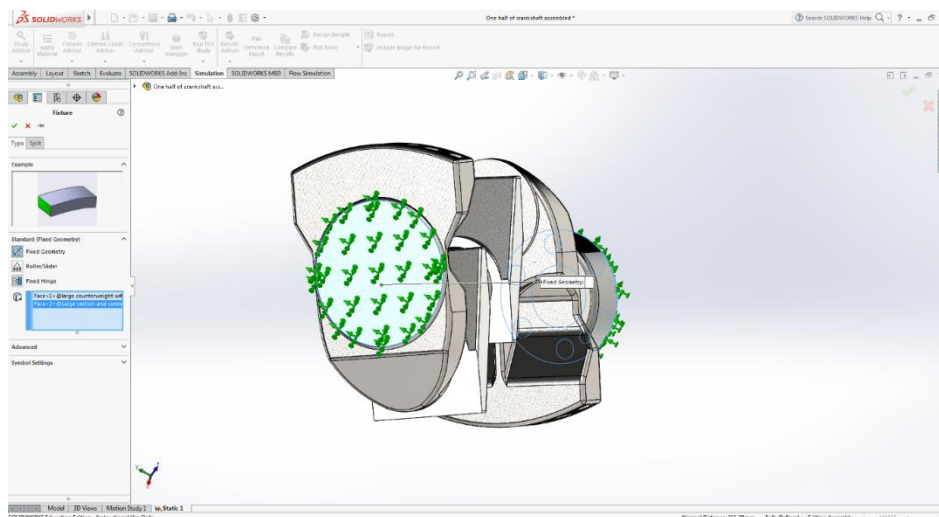


Figure 17.0 – Fixtures Applied to the Crankshaft Assembly.

An estimated force of 4000N was applied to the main journals, of which the con-rods are placed. Subsequently a 50mm curvature based mesh and a material of Plain Carbon Steel was applied to the assembly, to which all is presented in *Figure 17.0.1*. Curvature based mesh are preferred over the standard mesh, due to an increased accuracy for numerical calculations of stress, strain and displacement.

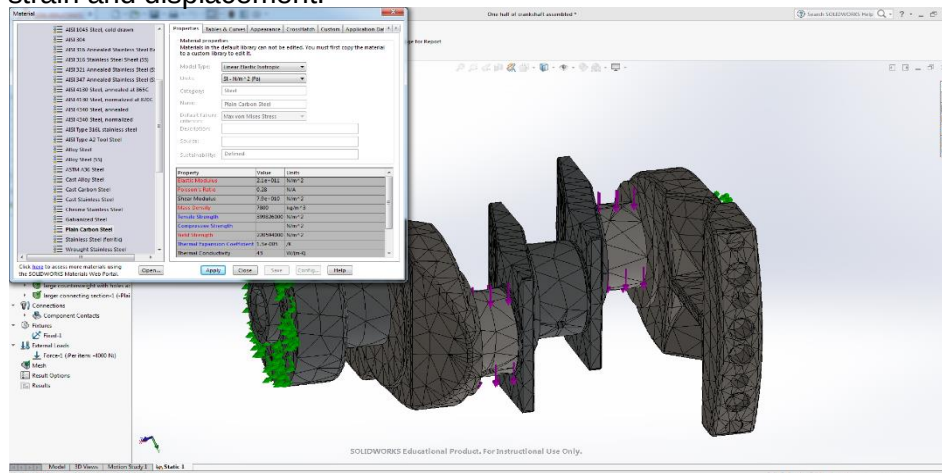


Figure 17.0.1 – Force, Material and Mesh Application.

The static simulation procedure was then run and results for stress, strain and displacement were obtained. Crankshaft deformation is apparent in the figures below.

Figure 17.0.1.1 displays a Maximum Von-Mises stress of 3.289×10^7 N/m² obtained.

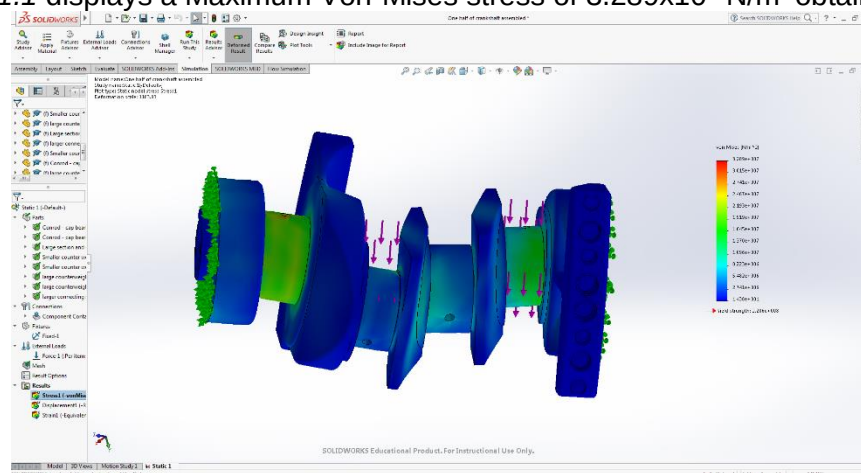


Figure 17.0.1.1 – Maximum Von Mises Stress.

Figure 17.0.1.2 displays a Maximum Displacement of 2.183×10^{-2} mm.

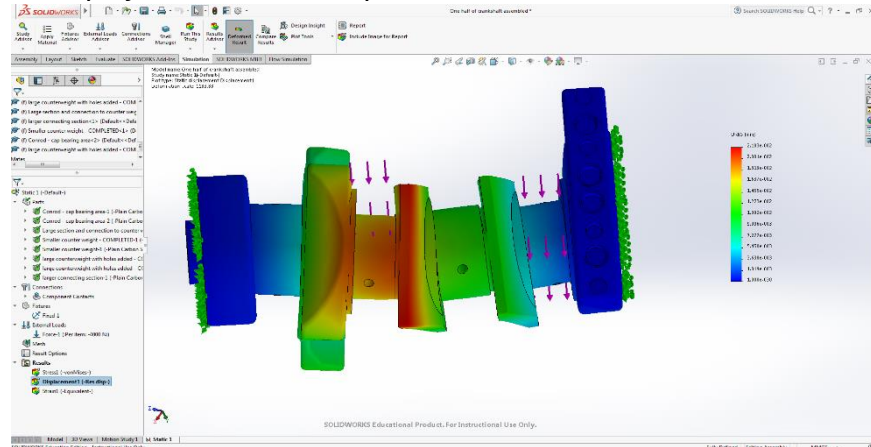


Figure 17.0.1.2 – Maximum Displacement.

Finally; a maximum of strain of 1.286×10^{-4} was obtained in Figure 17.0.1.3.

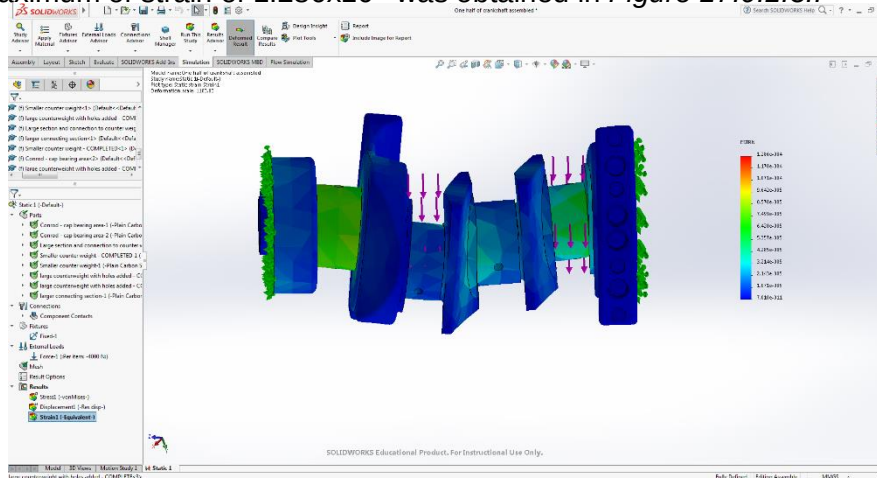


Figure 17.0.1.4- Maximum Strain.

However, the simulations were re-run with a 5mm Curvature Based mesh to further improve accuracy of stress, strain and displacement calculations. Consequently, Figure 17.0.2 displayed the improved mesh, whilst maintaining identical values for force, fixtures and material used in the previous simulation.

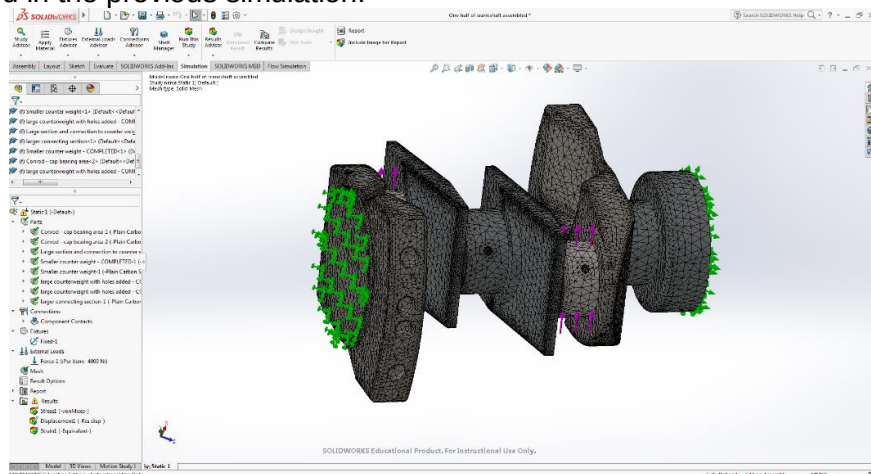


Figure 17.0.2 – 5mm Mesh Applied.

As visualised by Figure 17.0.2, the 5mm mesh produces a finer mesh than that of the 50mm previously used in *Figure 17.0.1*.

Re-running the simulations produced a new Maximum Von Mises Stress of $3.518 \times 10^7 \text{ N/m}^2$.

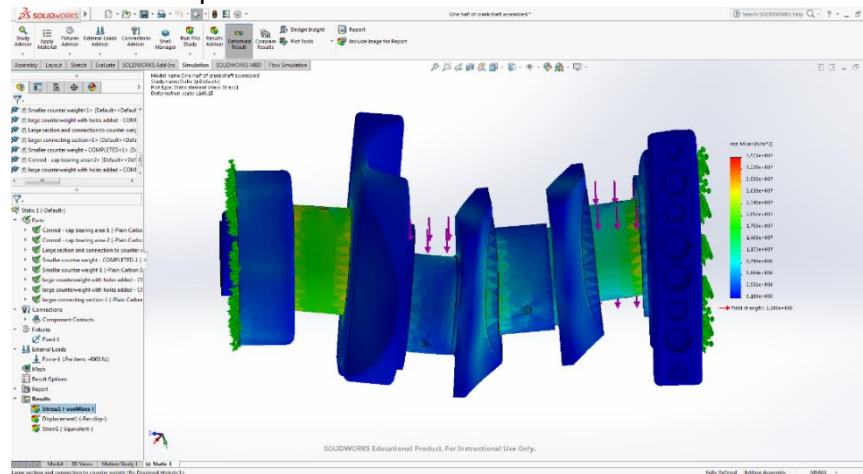


Figure 17.0.2.1 – New Von Mises Stress.

The new displacement of $2.250 \times 10^{-2} \text{ mm}$, was obtained in *Figure 17.0.2.2*.

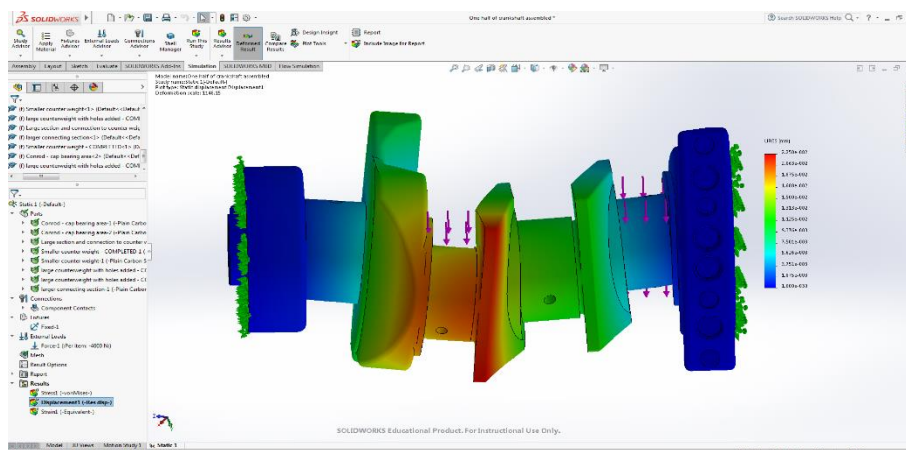


Figure 17.0.2.2 – New Displacement.

Finally, the new strain of 1.430×10^{-4} was obtained in *Figure 17.0.2.3*.

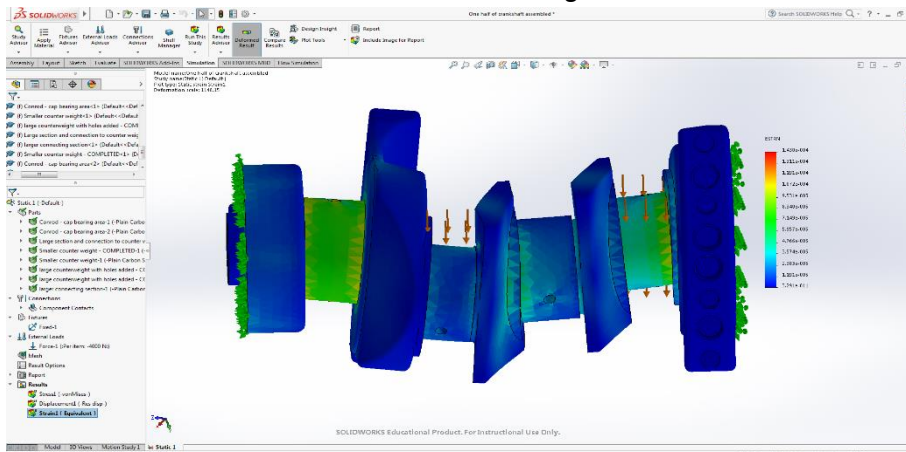


Figure 17.0.2.3 – New Strain.

10.0.1.1 – Ford Fox Crankshaft Single Span FEA:

However, further alterations to the FEA process for the crankshaft were required to produce a truly representative sample. Therefore, a single span of the crankshaft was selected, which included a large counter-web, a small counter-web, small diameter main journal and two large diameter main journals, as displayed in *Figure 17.0.3*.

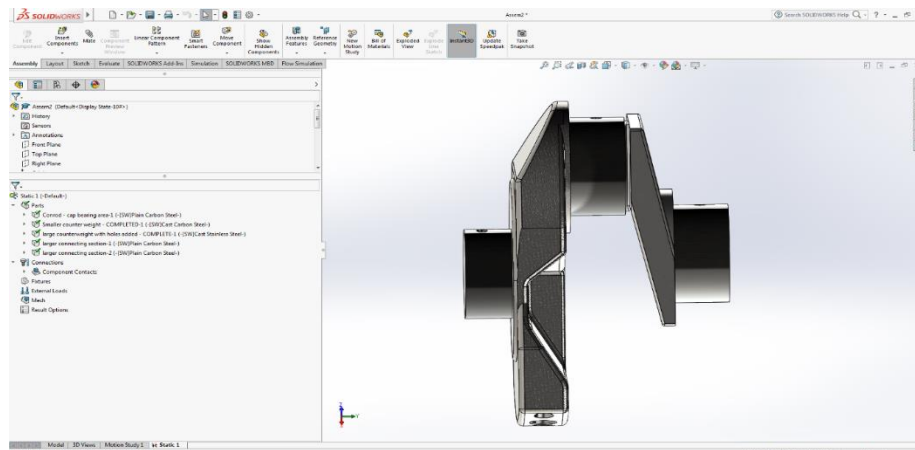


Figure 17.0.3 – Single Crankshaft Span.

This method produced a truly representative sample, due to being an industrial technique to reduce complexity of the analysis. However, the foundation of the analysis technique could be transferred for use with a multi-cylinder engine platform. Therefore, the crankshaft procedure could be implemented for crankshaft from various engines. (Montazersadgh & Fatemi, 2007)

Consequently, the single span of crankshaft was fixed at adjacent main journals to enable the crankshaft to pivot. FNet of 31839.29N was obtained from a previous Advanced Engine Design assignment and applied as a downwards force on the central main bearing. Half of FNet (15919.645N) was applied to the adjacent main bearings and applied as an upwards force. Finally, a further reduced 4mm Curvature Based mesh was applied, to provide additional accuracy and Plain Carbon Steel is applied, as displayed in *Figure 17.0.3.1*. (Jay, 2016)

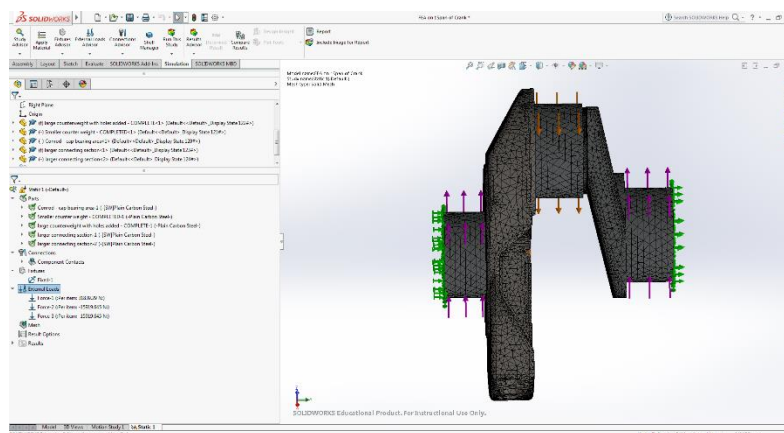


Figure 17.0.3.1 – Force, Fixtures and Mesh Application.

Simulations were then run and Maximum Von Mises Stress of $7.855 \times 10^7 \text{ N/m}^2$ was obtained, as displayed in *Figure 17.0.3.2*. Stress mainly occurred at connection of the central main bearing and the counter-webs.

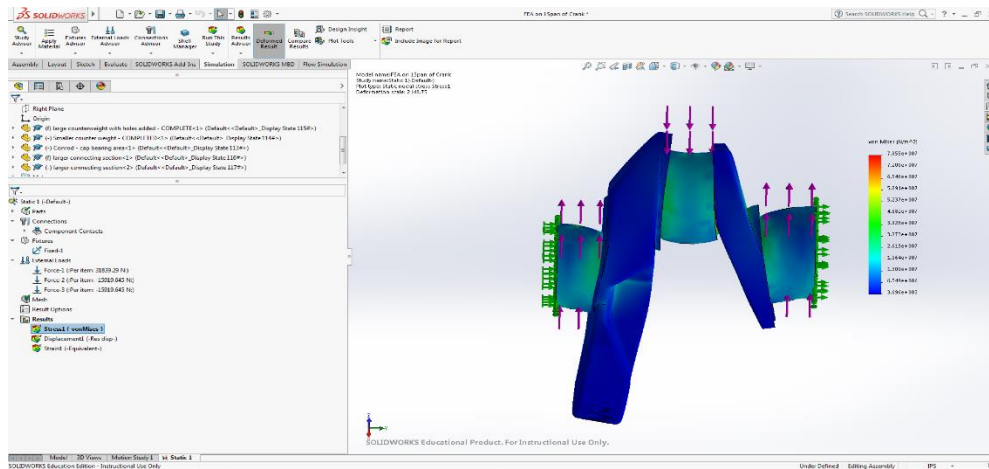


Figure 17.0.3.2 – Maximum Von Mises Stress for Single Span of Crankshaft.

Displacement of $8.77 \times 10^{-3} \text{ mm}$ was obtained and primarily occurred at the central main bearing and base of the large counter-web, as presented in *Figure 17.0.3.3*.

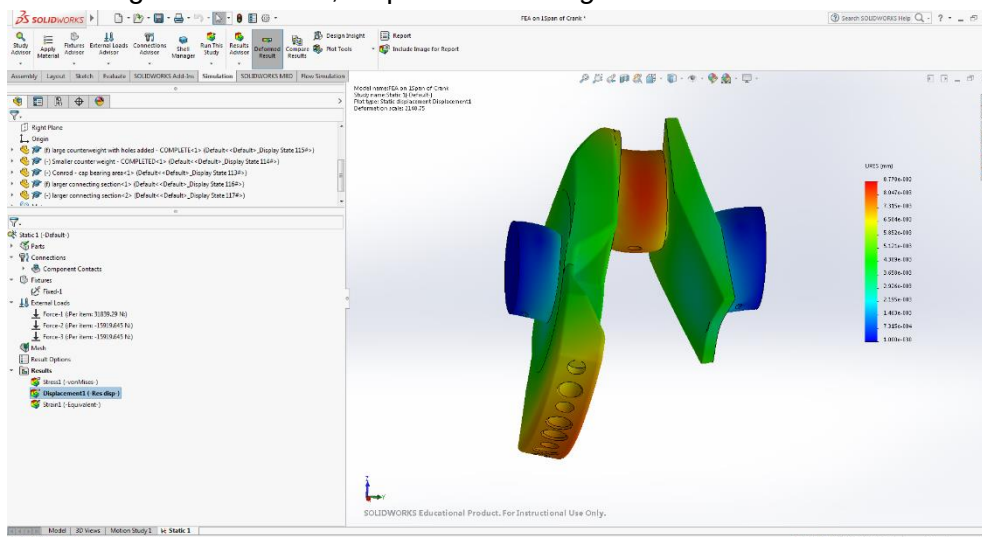


Figure 17.0.3.3 – Maximum Displacement for Single Span of Crankshaft.

Finally, Maximum Stress was obtained at 2.792×10^{-4} and predominantly occurred between the connection of the main journal and the counter-webs, as displayed in *Figure 17.0.3.4*.

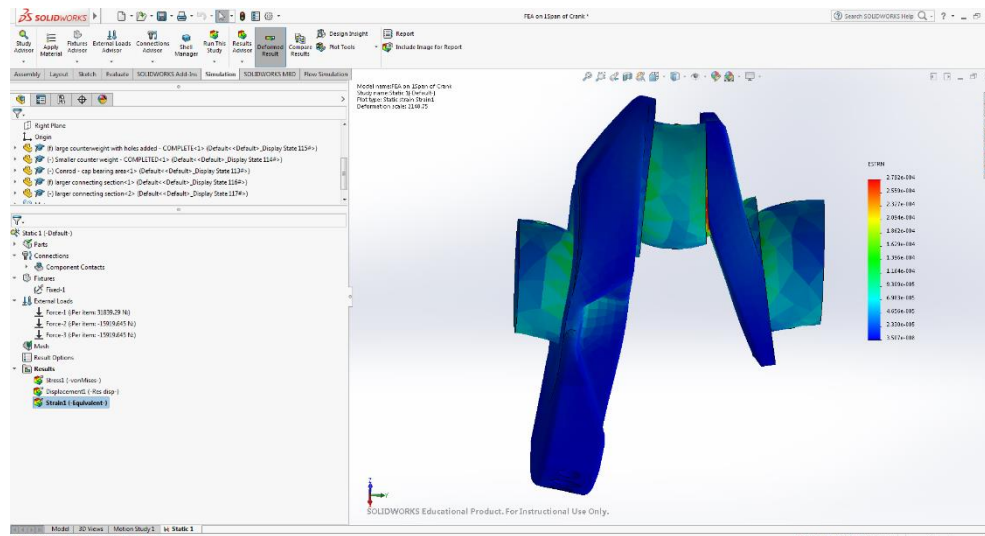


Figure 17.0.3.4 – Maximum Stress for Single Span of Crankshaft.

To further improve accuracy a 1mm mesh refinement was applied to the centre main journal in *Figure 17.0.3.5*.

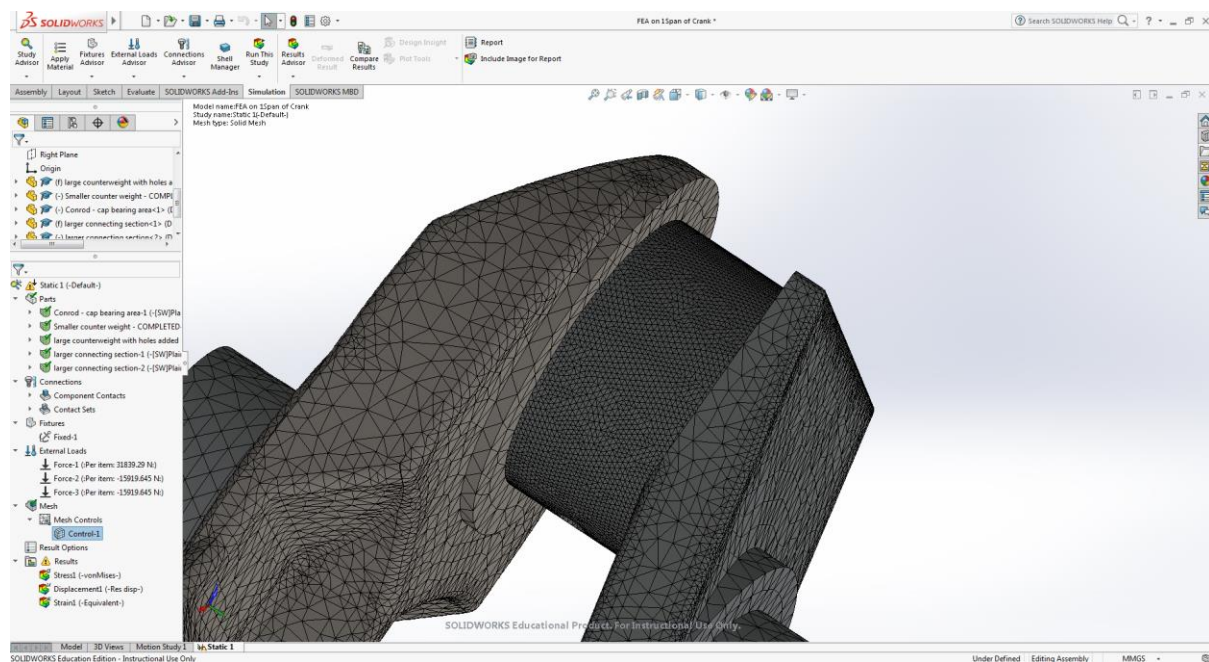


Figure 17.0.3.5 – 1mm Mesh Refinement.

Consequently, the following results were produced. Maximum Von Mises Stress of 9.564×10^7 N/m² was obtained in *Figure 17.0.3.6*.

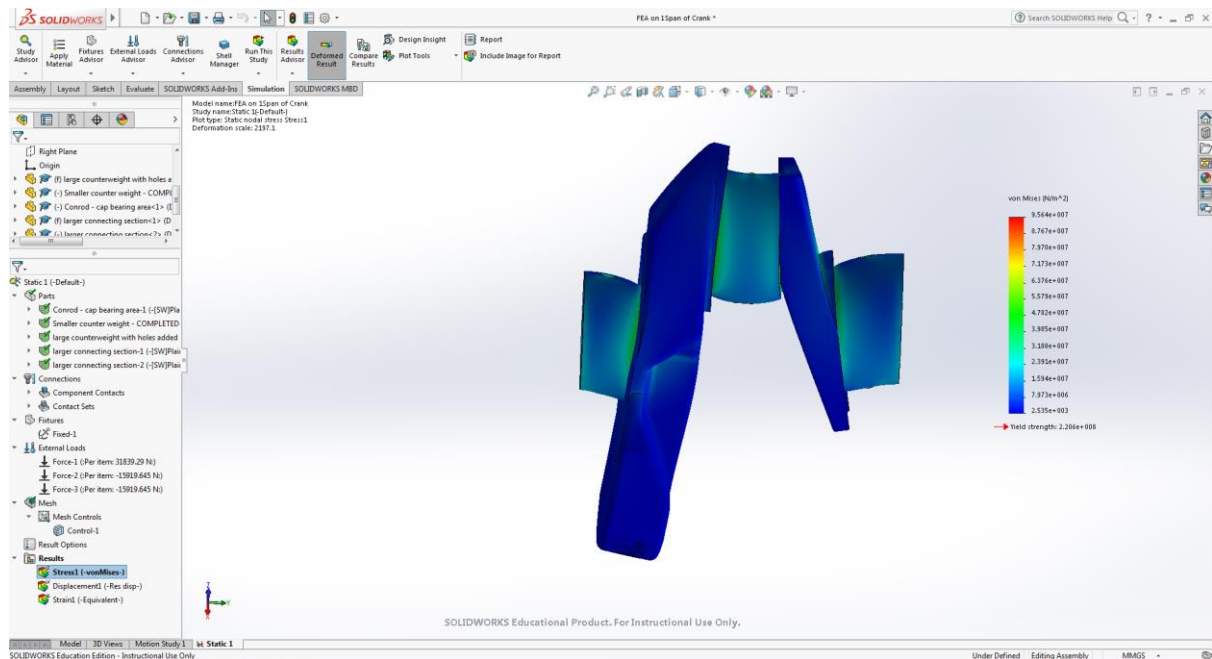


Figure 17.0.3.6 – Maximum Von Mises Stress.

Maximum Displacement of 8.587×10^{-3} mm was obtained in *Figure 17.0.3.7*.

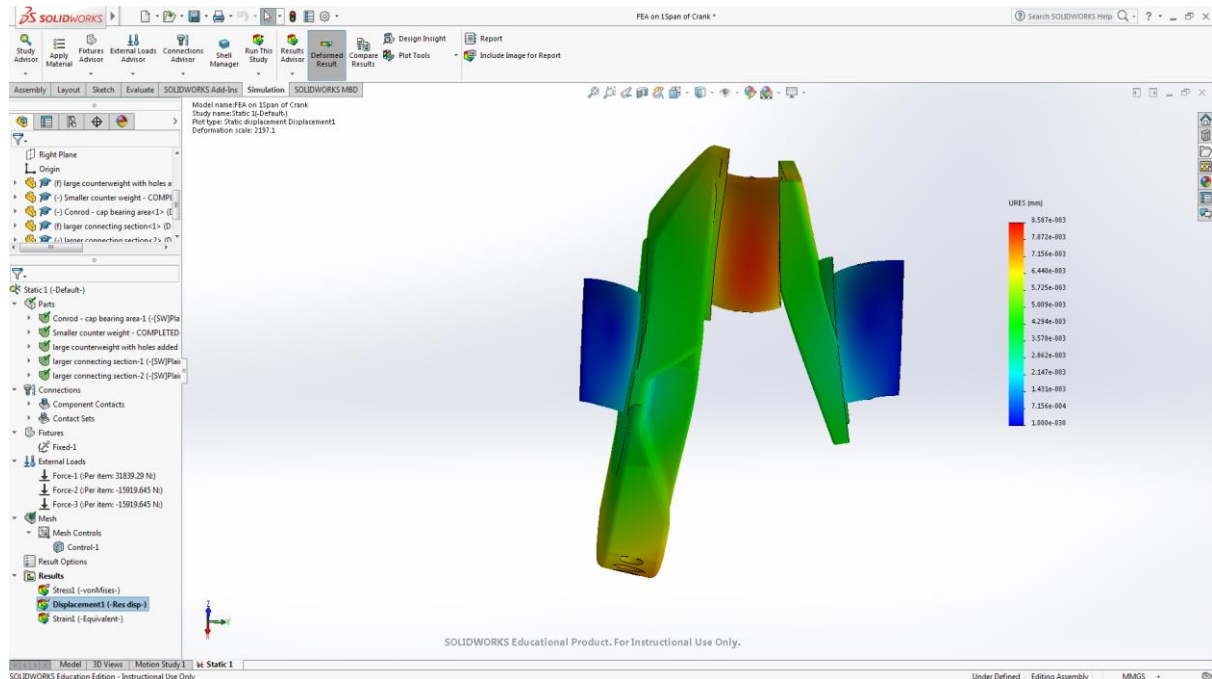


Figure 17.0.3.7 – Maximum Displacement.

Finally, Maximum Strain of 2.985×10^{-4} was obtained in *Figure 17.0.3.8*.

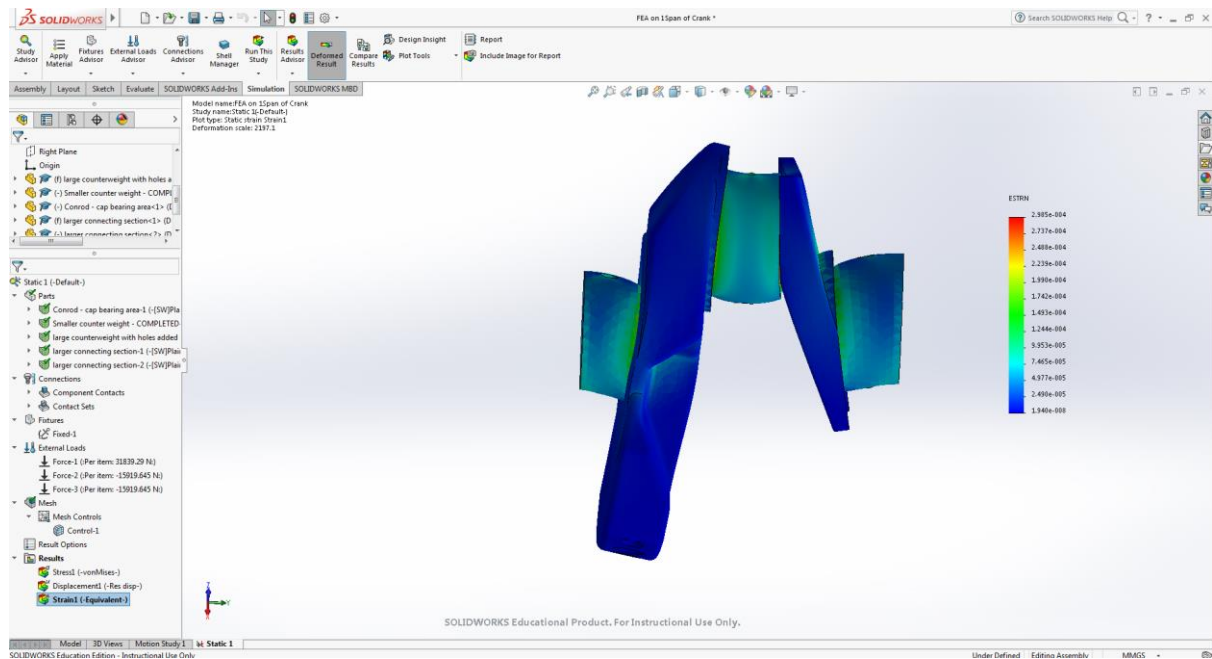


Figure 17.0.3.8 – Maximum Strain.

10.0.2 – Ford Fox Con-Rod FEA:

Once analysis of the Ford Fox Crankshaft was complete, analysis of the Ford Fox con-rod followed.

10.0.2.1 – Ford Fox Big-End of Con-Rod FEA.

An identical process of using a large 50mm mesh and an estimated 4000N force occurred and the same Plain Carbon Steel material was used throughout the simulations, as displayed in Figure 18.0.

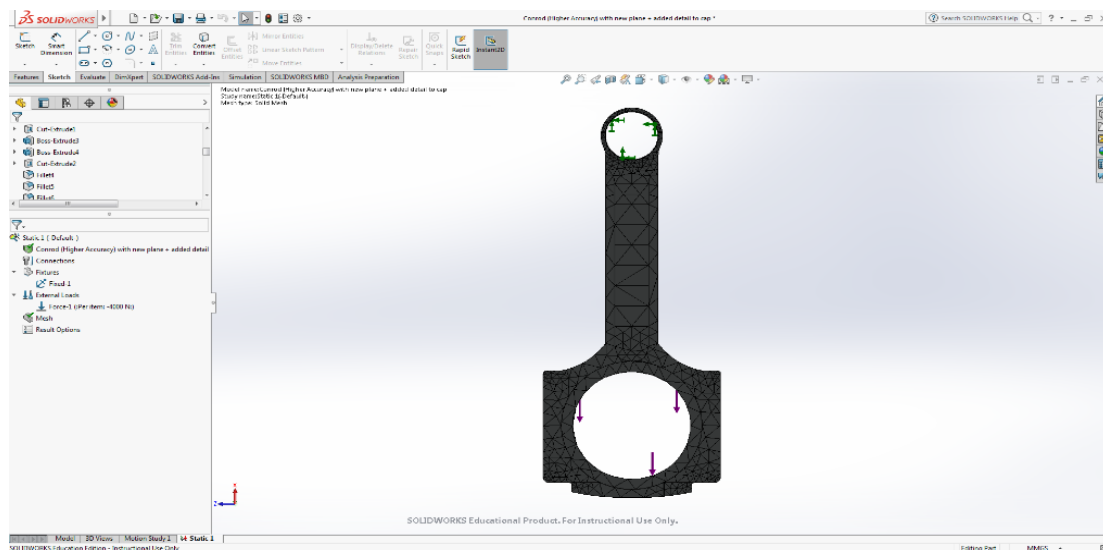


Figure 18.0 – 50mm Mesh, Fixture and Force Application.

Con-rod small and big ends were both subject to investigation. Therefore, a downwards force was applied to the big end and fixed at the adjacent end for the first simulation. The second simulation required the opposite process to investigate the small end.

Consequently, the results are listed below:

- Maximum Von Mises Stress of $5.964 \times 10^7 \text{ N/m}^2$
- Maximum Displacement of $3.852 \times 10^{-2} \text{ mm}$
- Maximum Strain of Max of 1.589×10^{-4}

The simulation was re-run with a 5mm Curvature Based mesh, as displayed in *Figure 18.0.1*.

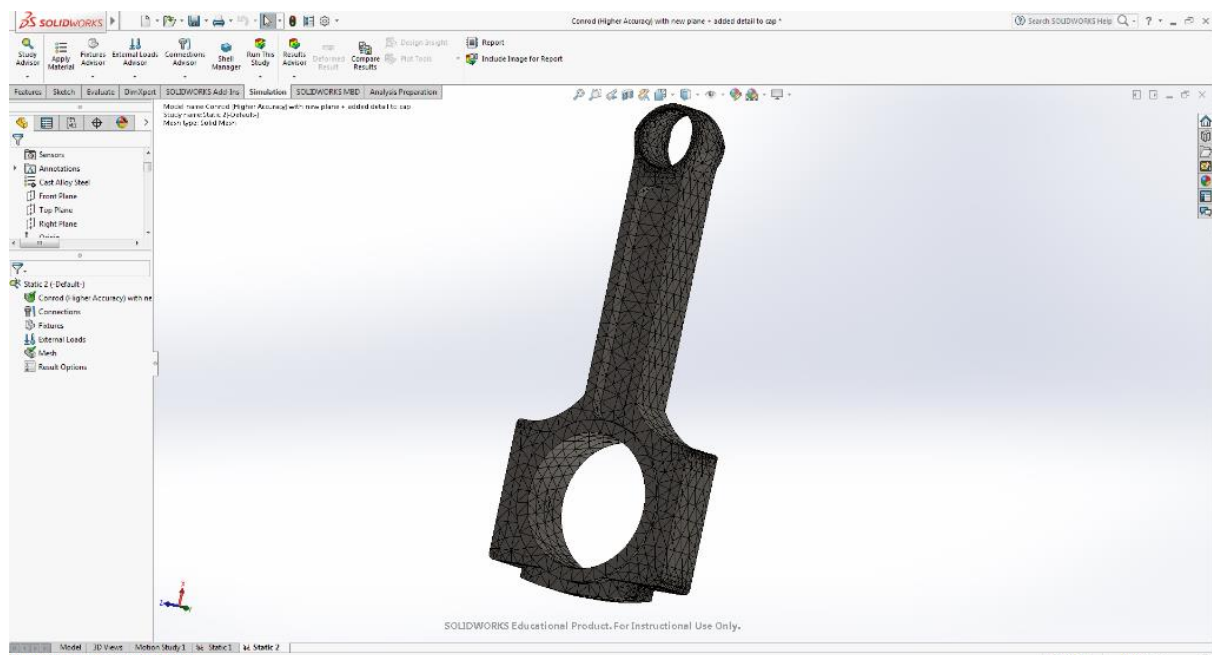


Figure 18.0.1- 5mm Mesh Application.

Identical fixtures and forces were applied to the 5mm mesh simulation. Consequently; results are listed below:

- Maximum Von Mises Stress = $6.374 \times 10^7 \text{ N/m}^2$
- Maximum Displacement = $3.869 \times 10^{-2} \text{ mm}$
- Maximum Strain = 1.666×10^{-4}

Mesh refinements of 1mm were introduced to further increase accuracy of results. However, mesh refinements were only introduced in areas of high stress concentration, as displayed in *Figure 18.0.1.1*.

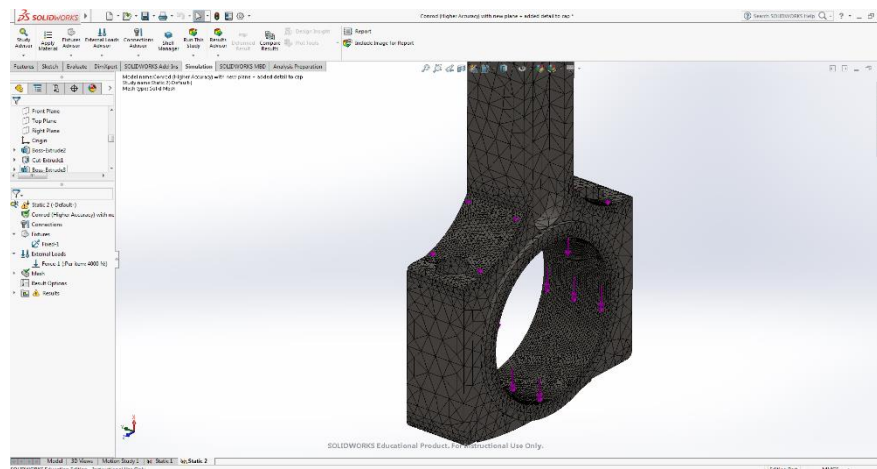


Figure 18.0.1.1 – 1mm Mesh Refinements.

Mesh refinements increase accuracy, due to enabling a finer mesh to be obtained. However, there is negative correlation between mesh refinement and run time. As mesh size reduces, simulation run time increases. Consequently; a compromise between both variables occurs. Therefore, mesh refinement is only produced in areas of high stress concentration to reduce the probability of increased run times. The refined mesh simulation was run and results were obtained.

Consequently, a Maximum Von Mises Stress of $7.328 \times 10^7 \text{ N/m}^2$ was obtained in *Figure 18.0.1.2*.

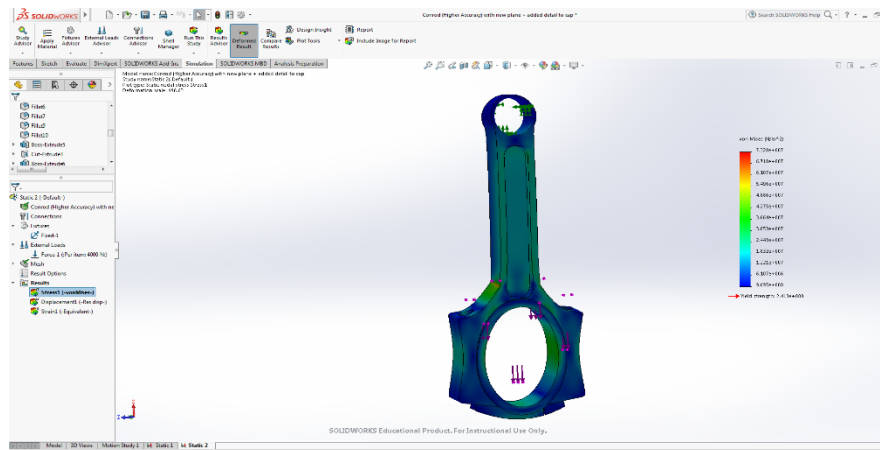


Figure 18.0.1.2 – Maximum Von Mises Stress for Refined Mesh.

Maximum displacement of 4.278×10^{-2} mm was obtained in *Figure 18.0.1.3*.

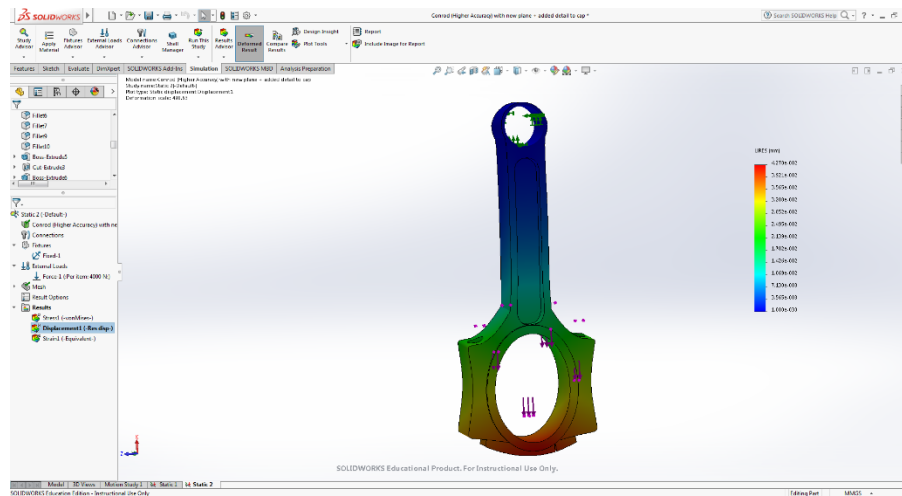


Figure 18.0.1.3 – Maximum Displacement for Refined Mesh.

Maximum Strain of 2.725×10^{-4} was obtained in *Figure 18.0.1.4*.

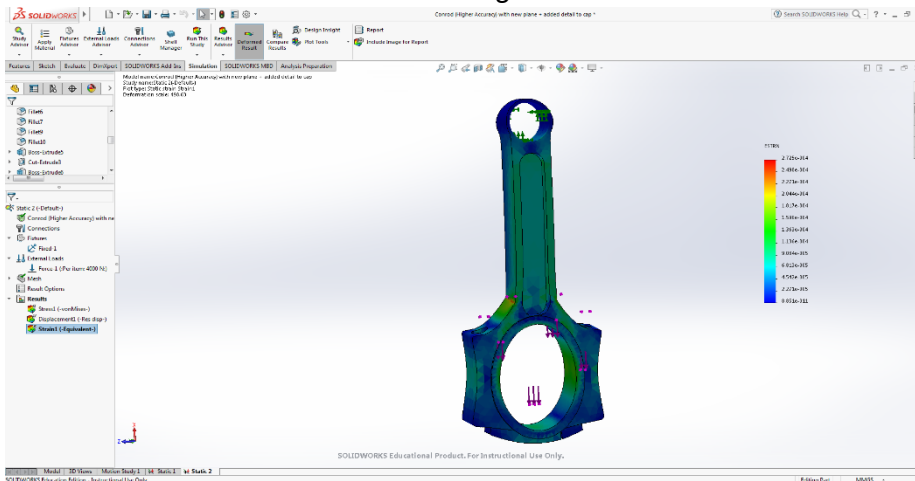


Figure 18.0.1.4 – Maximum Strain for Refined Mesh.

10.0.2.2 – Ford Fox Small-End of Con-Rod FEA:

The higher accuracy 5mm Curvature Based mesh with an estimated 4000N downwards force applied to the Small-End of the con-rod. While the big end of the con-rod was fixed, consequently producing a Maximum Von Mises Stress of $1.397 \times 10^8 \text{ N/m}^2$, as displayed in *Figure 18.0.2*.

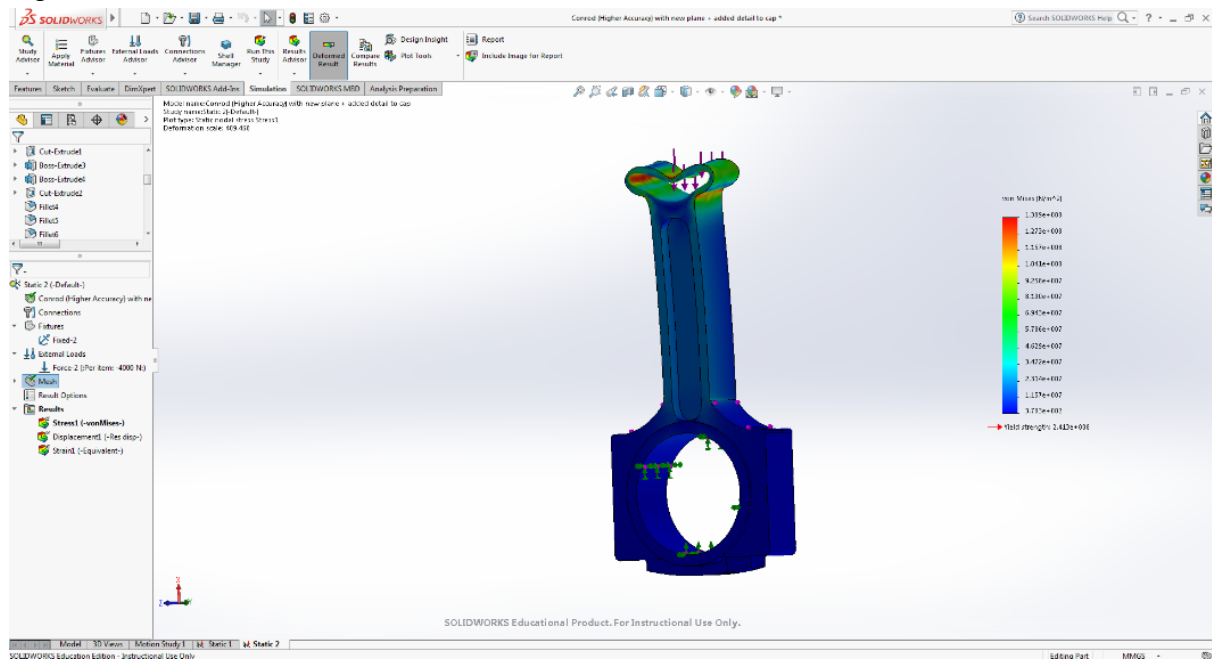


Figure 18.0.2 – Maximum Von Mises Stress.

Maximum Displacement of $4.195 \times 10^{-2} \text{ mm}$ was obtained in *Figure 18.0.2.1*.

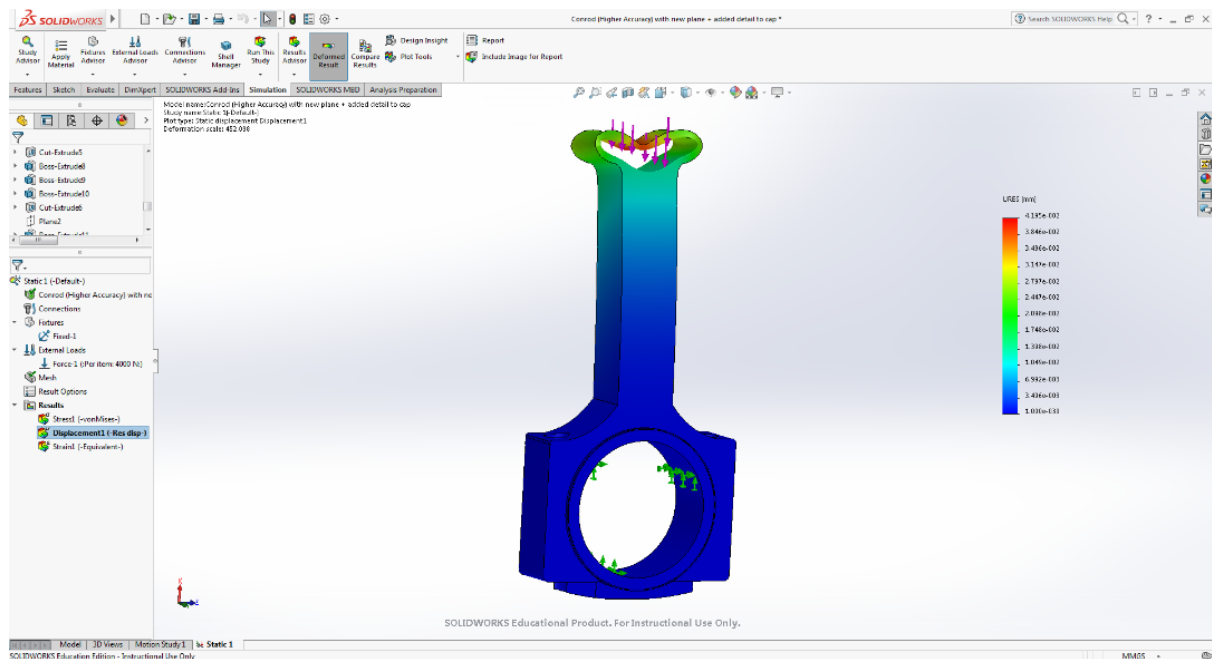


Figure 18.0.2.1 – Maximum Displacement.

Maximum Strain of 4.626×10^{-4} was obtained in *Figure 18.0.2.2*.

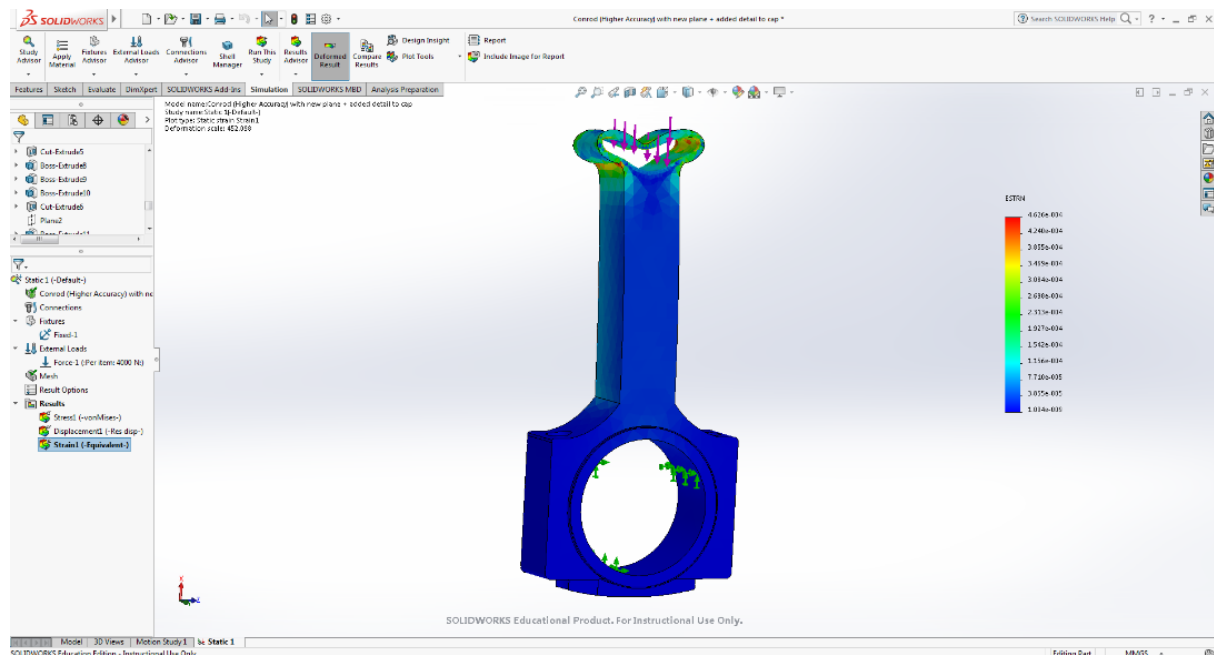


Figure 18.0.2.2 – Maximum Strain.

To improve accuracy the 1mm mesh was applied to areas of high stress concentration, displayed in *Figure 18.0.2.3*.

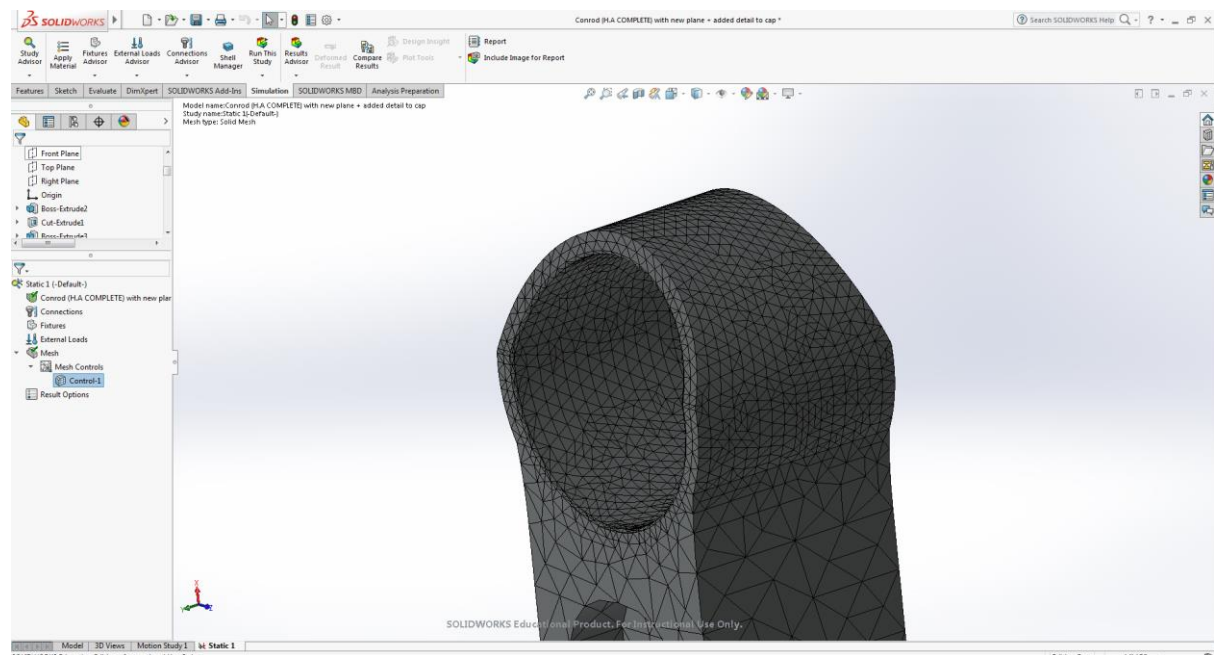


Figure 18.0.2.3 – 1mm Refined Mesh

Consequently, the new Maximum Von Mises Stress of $1.389 \times 10^8 \text{ N/m}^2$ was obtained, in Figure 18.0.2.4.

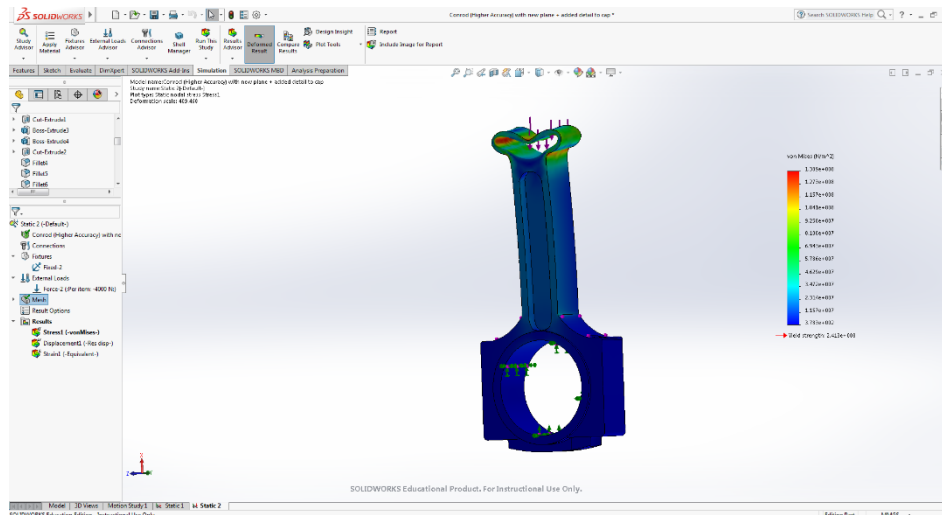


Figure 18.0.2.4 – Maximum Von Mises Stress for Refined Mesh.

The new Maximum Displacement achieved was $4.637 \times 10^{-2} \text{ mm}$, displayed in Figure 18.0.2.5.

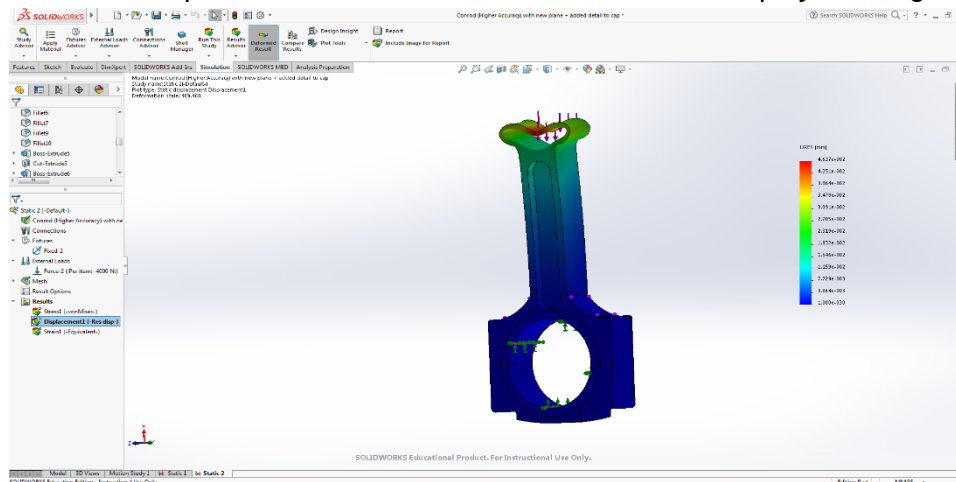


Figure 18.0.2.5 – Maximum Displacement for Refined Mesh

New Maximum Strain of 5.324×10^{-4} was obtained, in Figure 18.0.2.6.

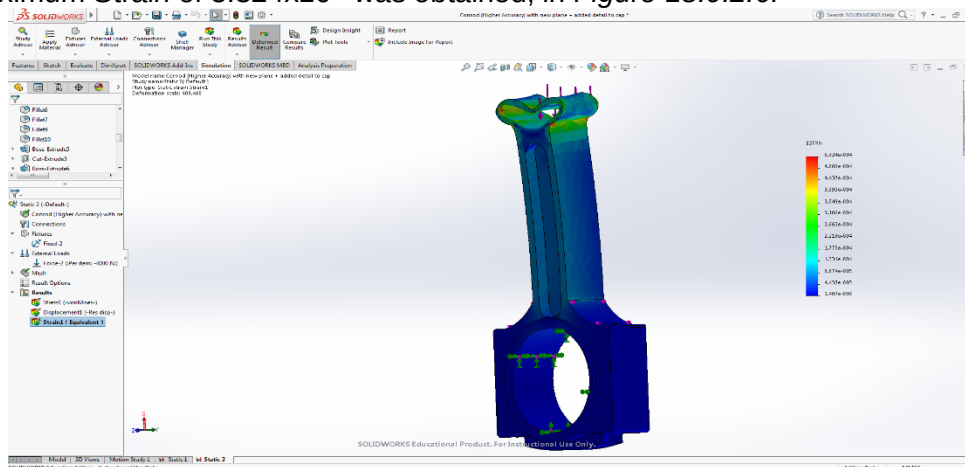


Figure 18.0.2.6 – Maximum Strain for Refined Mesh.

However, the author was not satisfied the simulations produced a realistic representation of the deformation of a con-rod. Realistically the Gudgeon Pin would prohibit the con-rod from deforming in such a way. Therefore, a Gudgeon Pin, Crank Pin and Con-Rod assembly was produced, displayed in *Figure 18.0.3*.

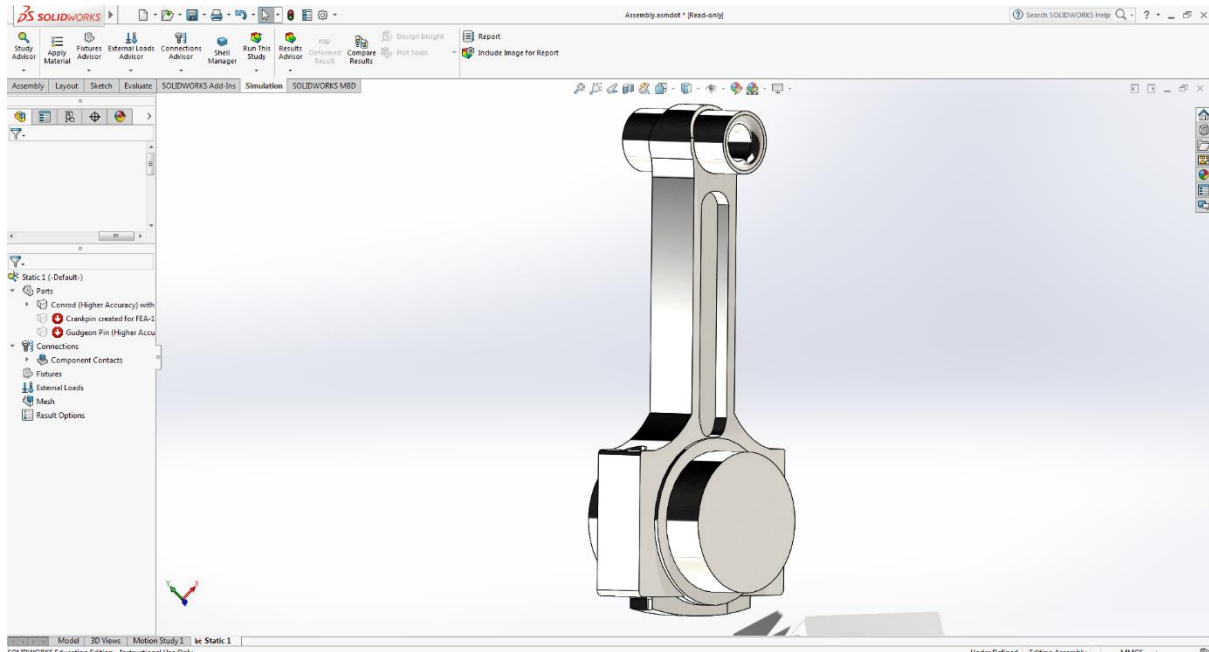


Figure 18.0.3. – Gudgeon Pin, Crank Pin and Con-Rod Assembly.

Previously calculated value of FNet was obtained from a previous Advanced Engine Design assignment of 34415.86N, which was placed downwards as a Uniformly Distributed Load (UDL) on the Gudgeon pin. The assembly was fixed at the crankpin, as displayed in *Figure 18.0.3.1*.

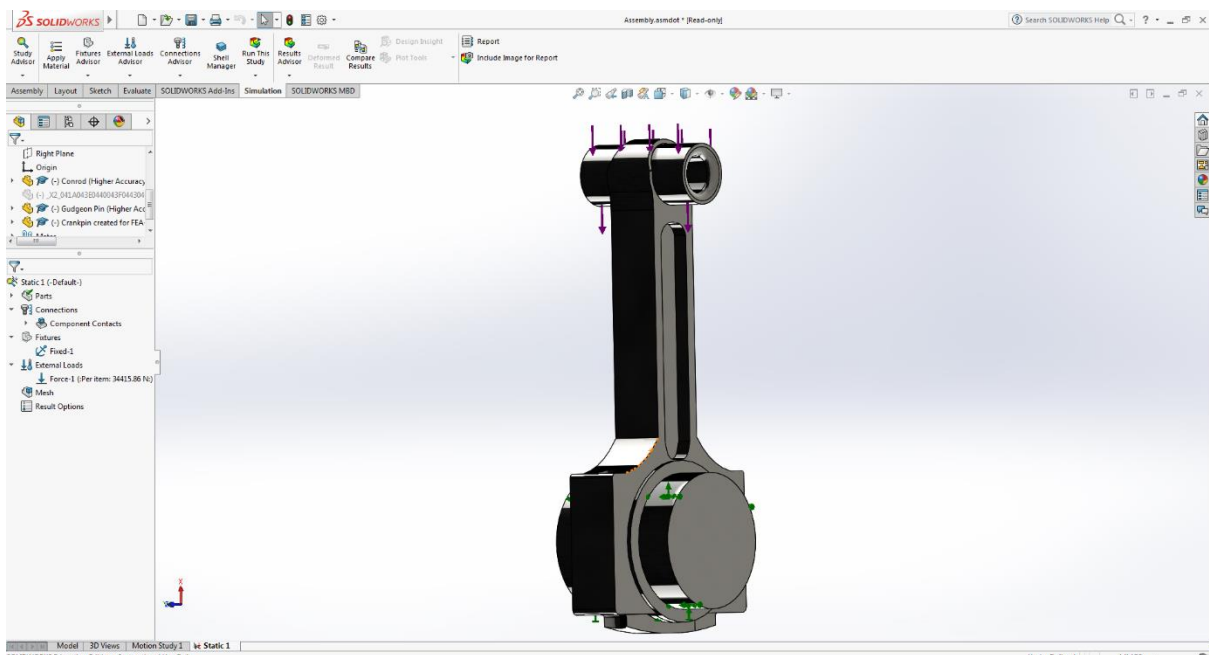


Figure 18.0.3.1 – Force and Fixture Application.

The 4mm mesh and Plain Carbon Steel was applied, to further improve accuracy of data obtained. Displayed in Figure 18.0.3.2.

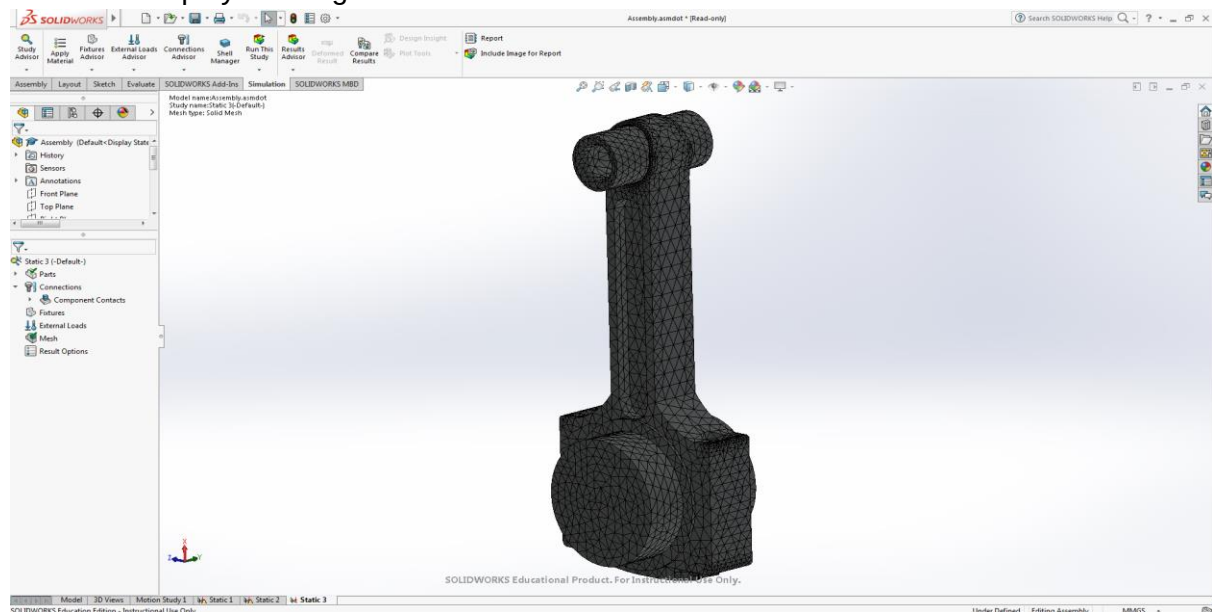


Figure 18.0.3.2 – 4mm Mesh Application.

This was the first FEA simulation produced as part of an assembly. Consequently, Contact Sets were required to reduce the likelihood of component interference causing simulation failure. Component interference causing simulation failure is displayed in Figure 18.0.3.3 via the red highlighted Gudgeon Pin.

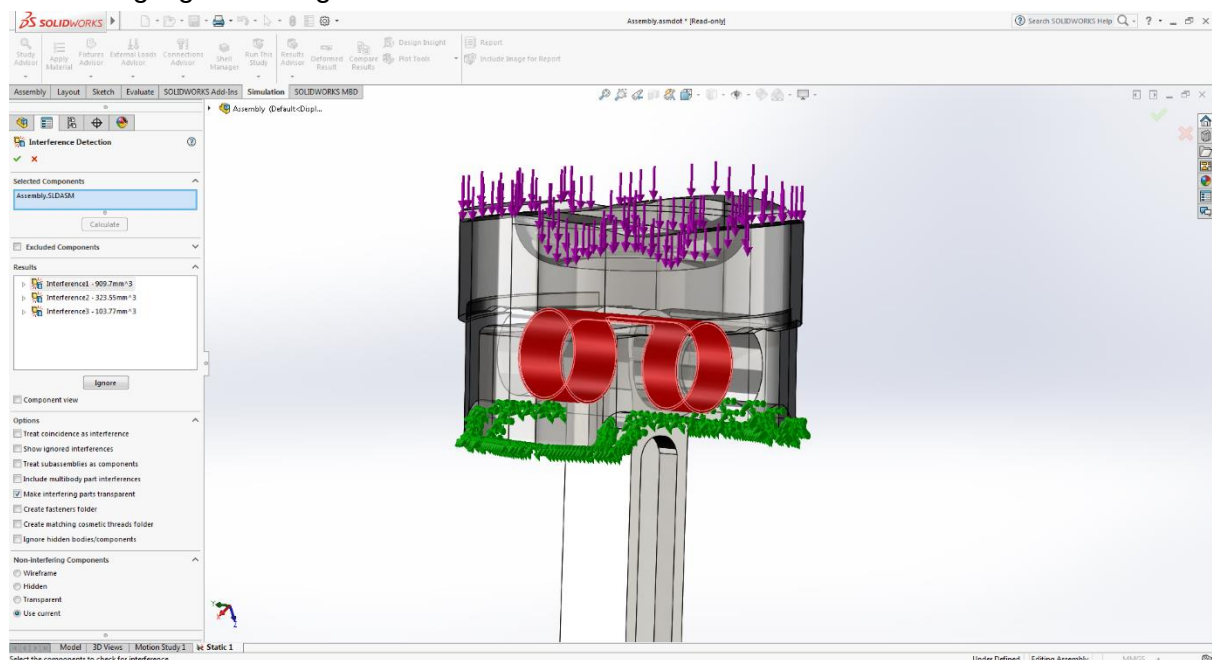


Figure 18.0.3.3 – Component Interference.

Consequently, a Non-Penetration Contact Set was chosen and the components which were likely to produce interference were selected and gap clearances between components were ignored, which could also produce simulation errors, displayed in *Figure 18.0.3.4*.

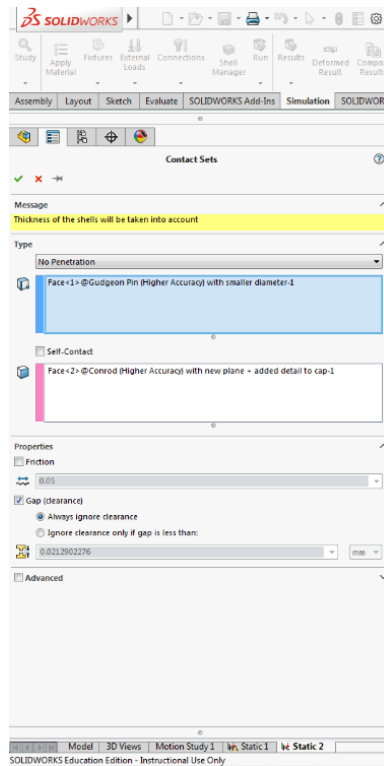


Figure 18.0.3.4 – Contact Set Application.

Simulations with the 34415.86N force applied as a UDL were performed, as presented in *Figure 18.0.3.5*. Where a Maximum Von Mises Stress of 2.612×10^9 N/m² was obtained.

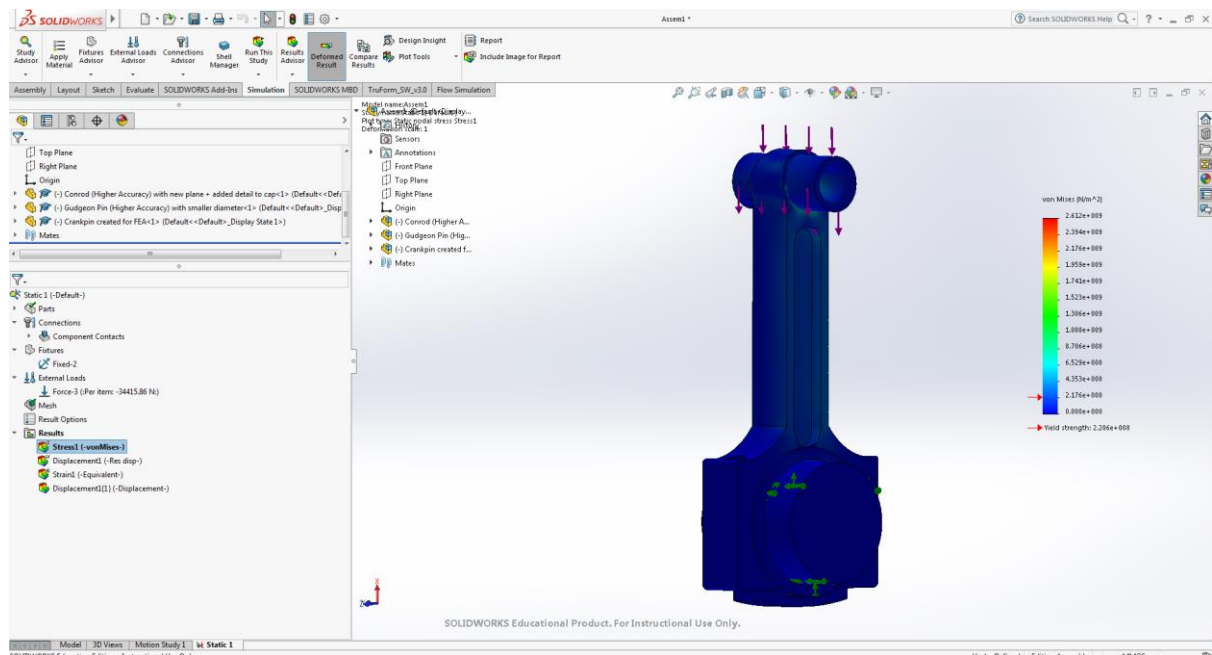


Figure 18.0.3.5 – Maximum Von Mises Stress with force applied as a UDL.

Maximum Displacement of 5.827×10^{-1} mm was obtained in *Figure 18.0.3.6*.

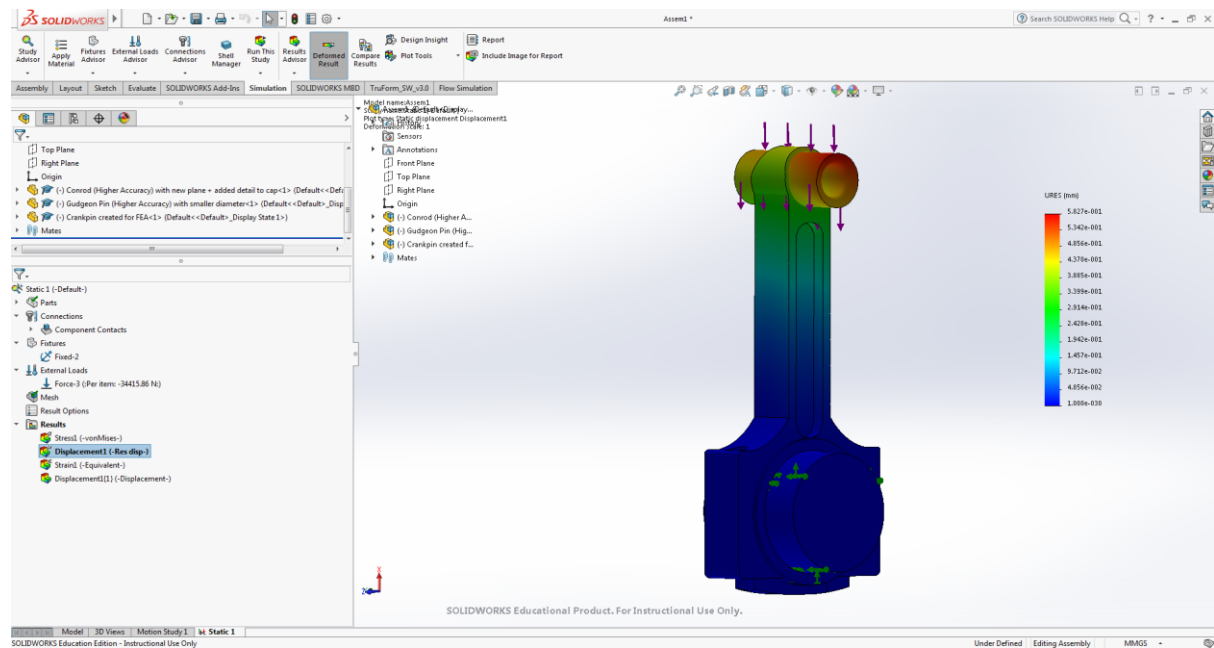


Figure 18.0.3.6 – Maximum Displacement with force applied as a UDL.

Maximum Strain of 9.99×10^{-3} was obtained in *Figure 18.0.3.7*.

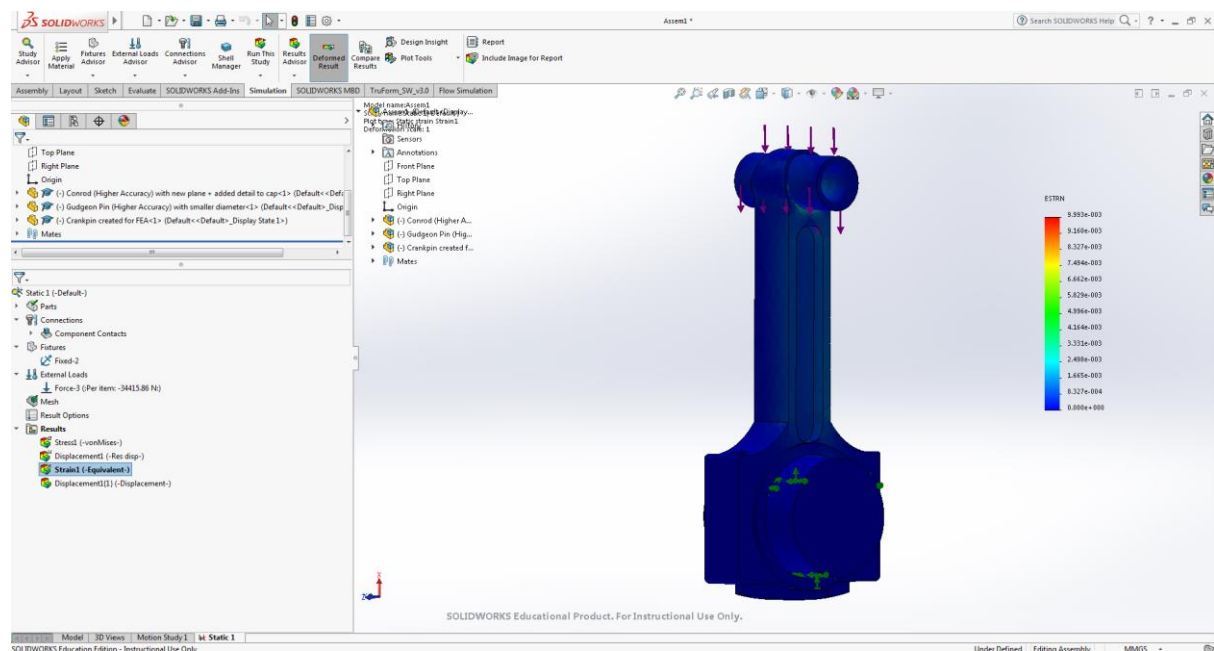


Figure 18.0.3.7 – Maximum Strain with force applied as a UDL.

However, to maintain a truly representative yield of data, point loads were required. This is due to the piston realistically only applying a force at the ends of the Gudgeon Pin.

The application of force via point load required the division of FNet by half. Consequently, a 17207.93N point load was applied to each end of the Gudgeon Pin. The Plain Carbon Steel material, contact sets and fixtures applied remained unchanged. Therefore, Maximum Von Mises Stress of $2.610 \times 10^9 \text{ N/m}^2$, was achieved in *Figure 18.0.3.8*.

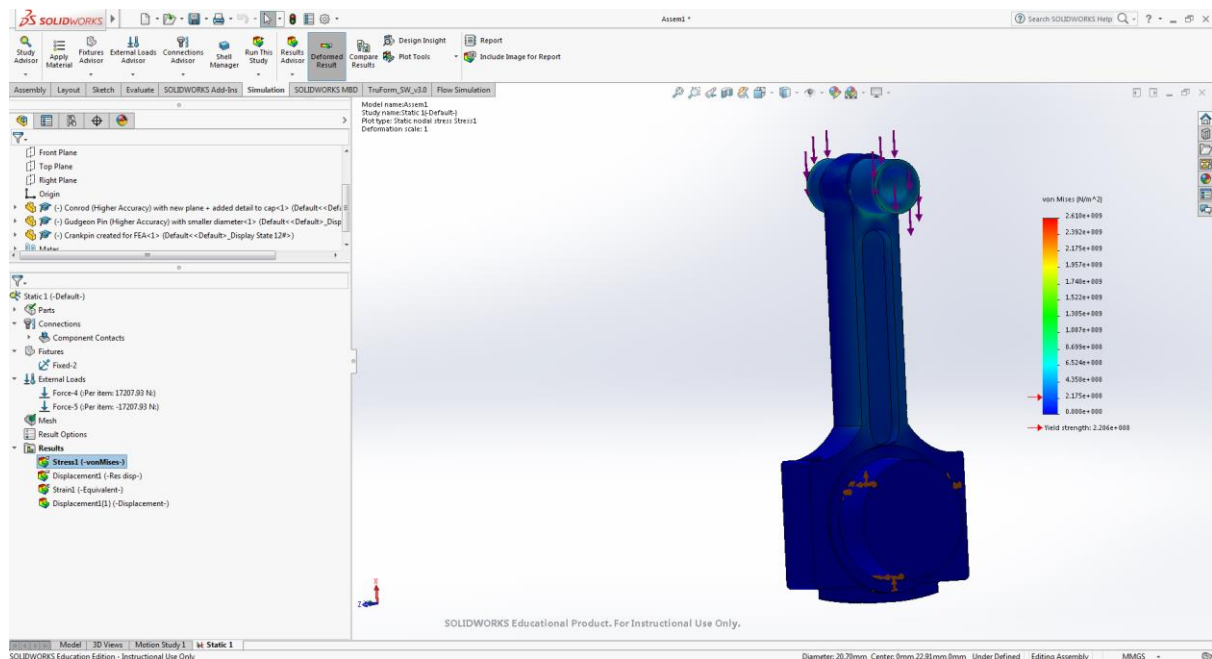


Figure 18.0.3.8 – Maximum Von Mises Stress with force applied as a Point Load.

Maximum Displacement of $7.431 \times 10^{-1} \text{ mm}$ was achieved in *Figure 18.0.3.9*.

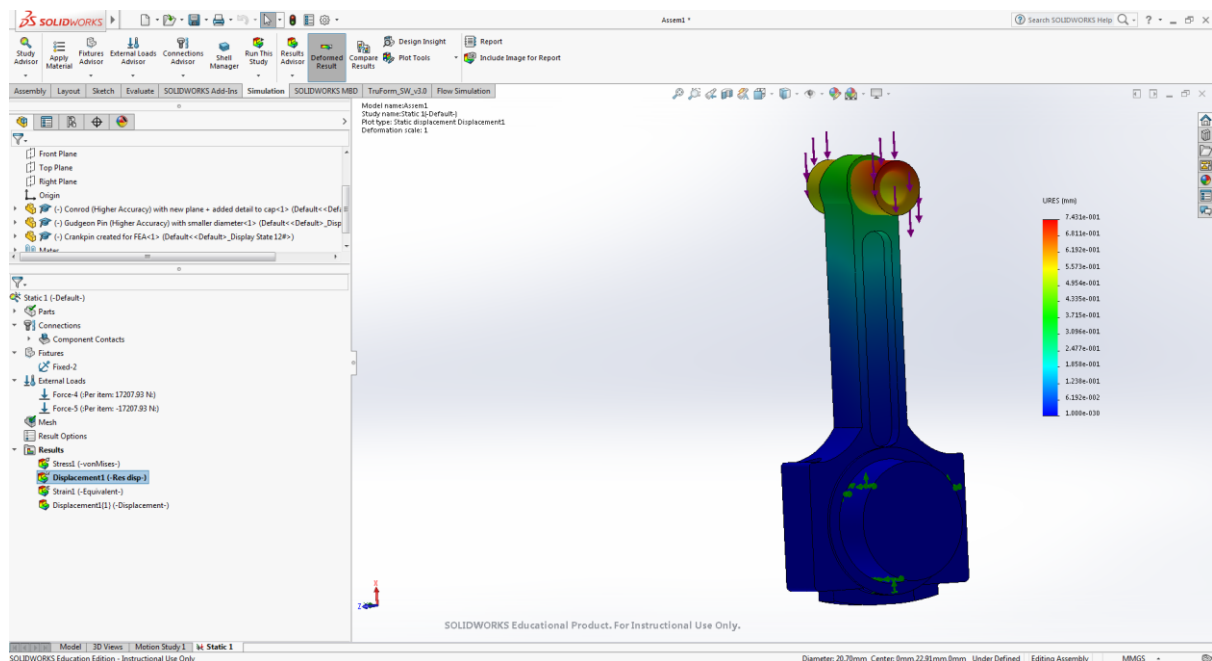


Figure 18.0.3.9 – Maximum Displacement with force applied as a Point Load.

Finally; Maximum Strain of 9.909×10^{-3} was obtained in *Figure 18.0.3.10*.

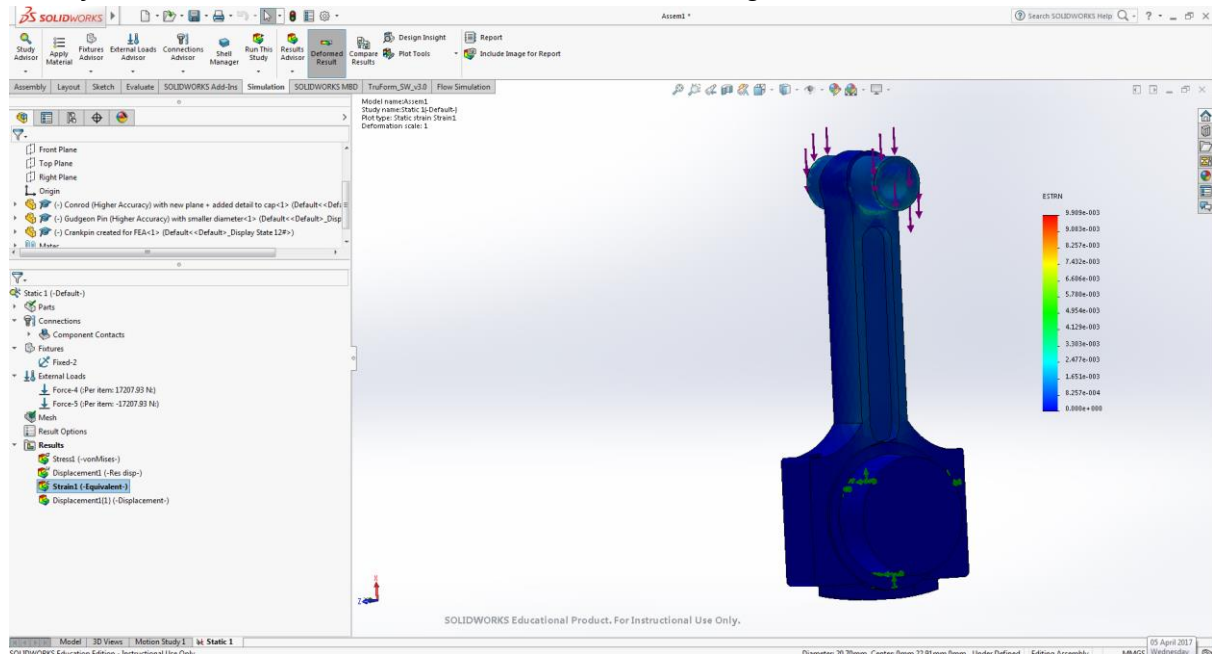


Figure 18.0.3.11 – Maximum Strain with force applied as a Point Load.

Simulation run time of one minute and thirty seconds was obtained for FEA of Con-Rod, Gudgeon Pin and Crank Pin assembly with a 4mm mesh. Displayed in *Figure 18.0.3.12*.

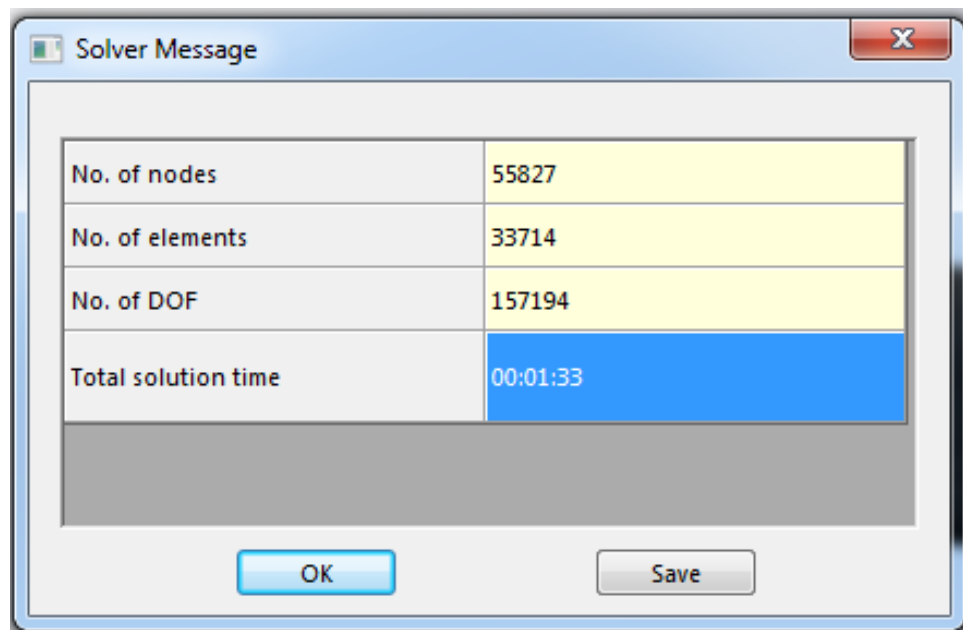


Figure 8.0.3.12 – Simulation Solver Messages.

10.0.3 – Ford Fox Piston FEA:

Ford Fox piston FEA began with the use of a 5mm Curvature Based mesh and an estimated 4000N force applied to the area where the Gudgeon Pin would rest within the piston. Simulations were produced and the following results were obtained. Maximum Von Mises Stress of $1.220 \times 10^7 \text{ N/m}^2$ occurred in *Figure 18.0.4*.

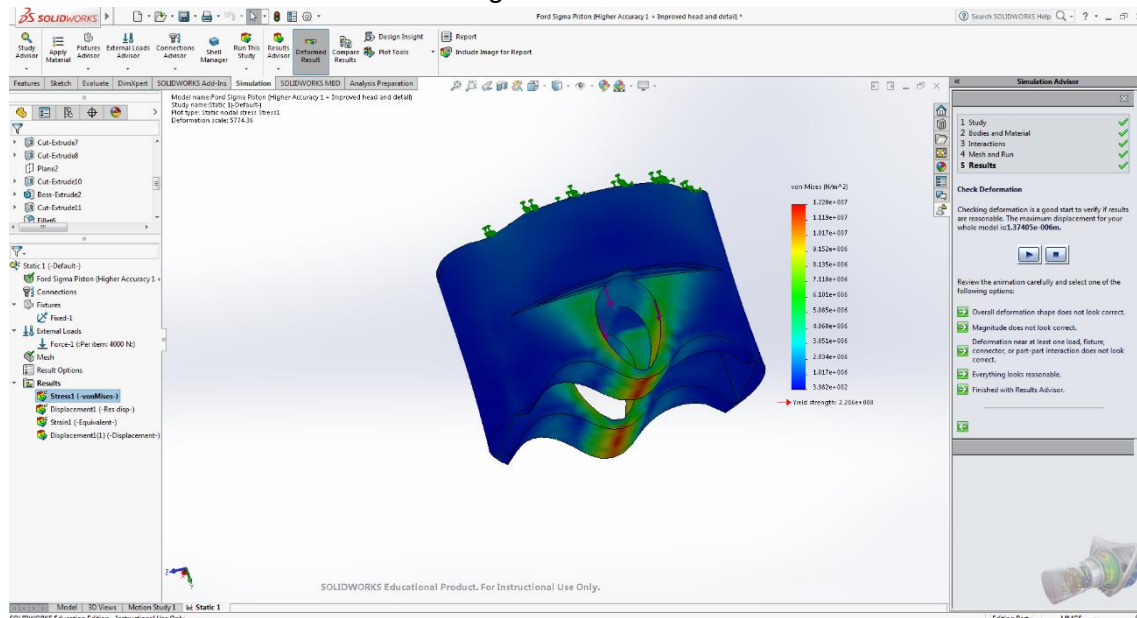


Figure 18.0.4 – Maximum Von Mises Stress.

Maximum Displacement of $1.374 \times 10^{-3} \text{ mm}$ was obtained in *Figure 18.0.4.1*.

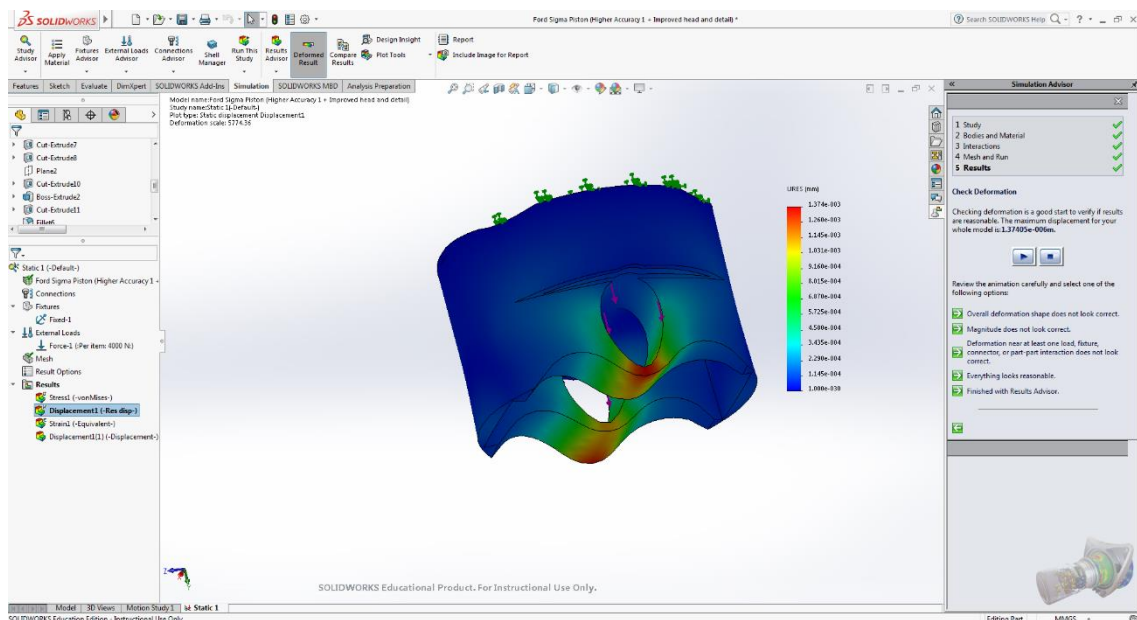


Figure 18.0.4.1 – Maximum Displacement.

Maximum Strain of 3.445×10^{-5} was achieved in *Figure 18.0.4.2*.

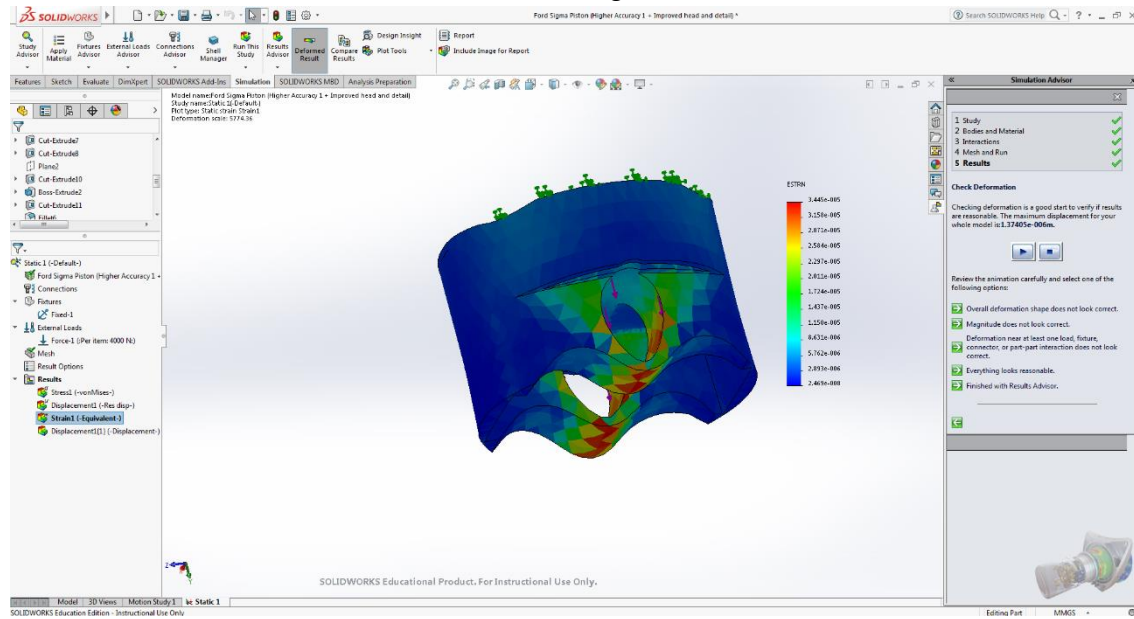


Figure 18.0.4.2 – Maximum Strain.

Mesh refinement of 1mm was positioned within the piston, where high stress concentrations occur at the resting point of the Gudgeon pin location, as presented in *Figure 18.0.4.3*.

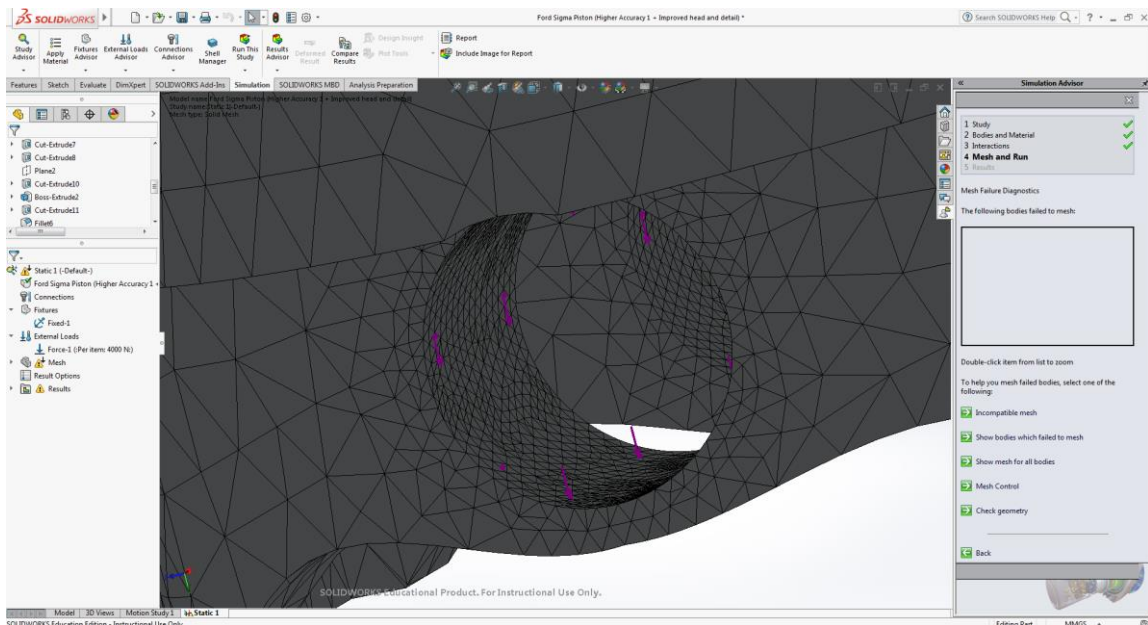


Figure 18.0.4.3 – 1mm Curvature Based Mesh Refinement.

The simulation was re-run with the new mesh refinement and a new Maximum Von Mises Stress of $1.334 \times 10^7 \text{ N/m}^2$ was obtained in Figure 18.0.4.4.

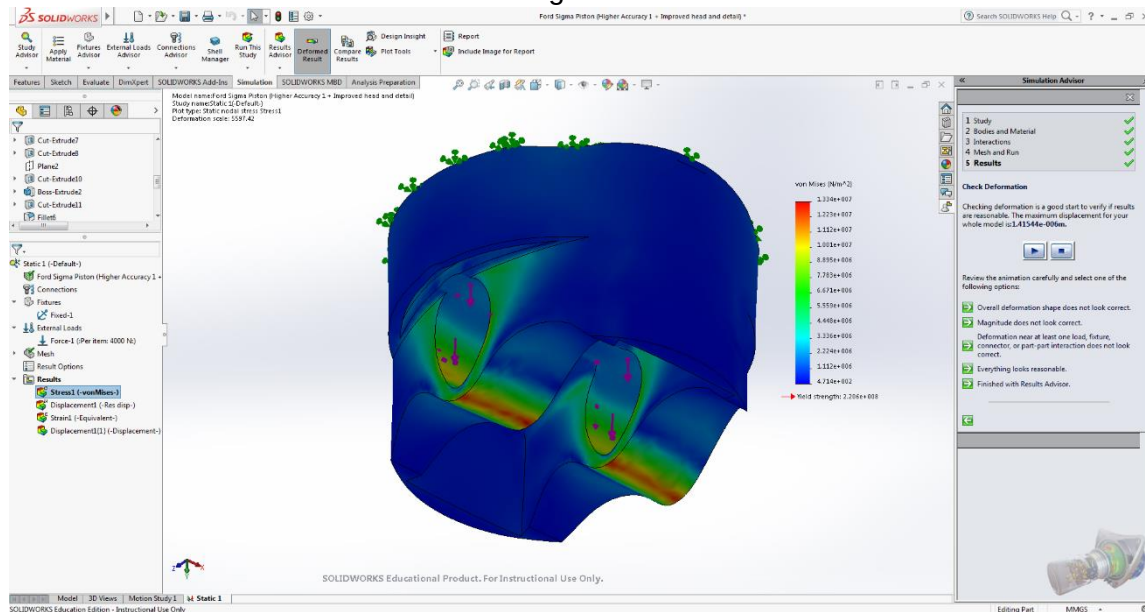


Figure 18.0.4.4 – Maximum Von Mises Stress.

Maximum Displacement of $1.415 \times 10^{-3} \text{ mm}$ seen in Figure 18.0.4.5.

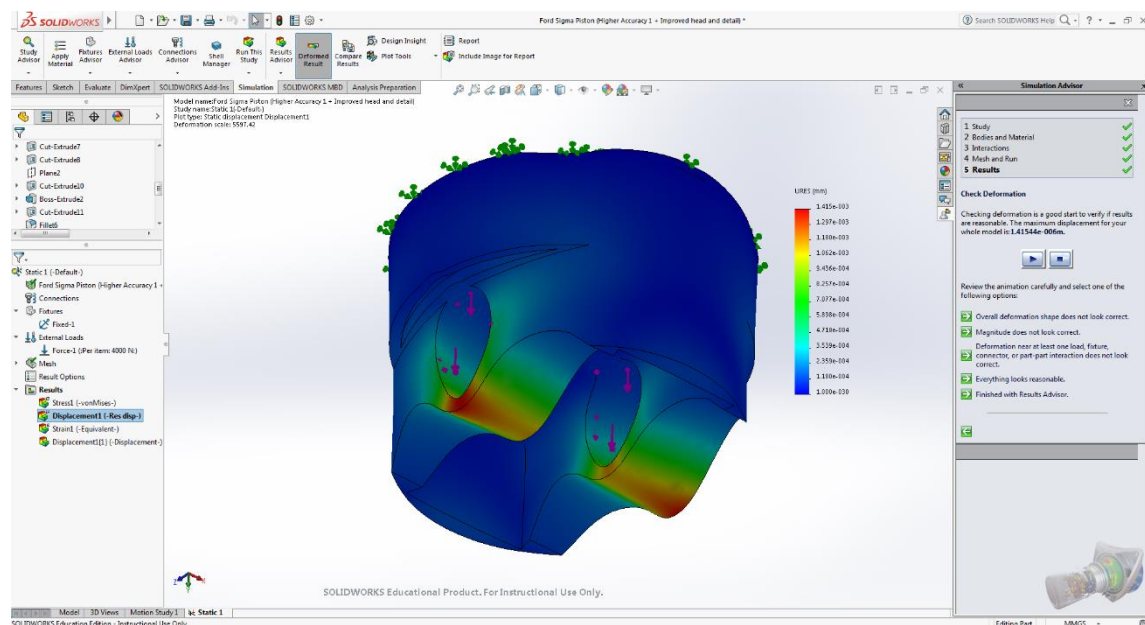


Figure 18.0.4.5 – Maximum Displacement.

Finally; Maximum Strain of 4.367×10^{-6} was obtained in *Figure 18.0.4.6*.

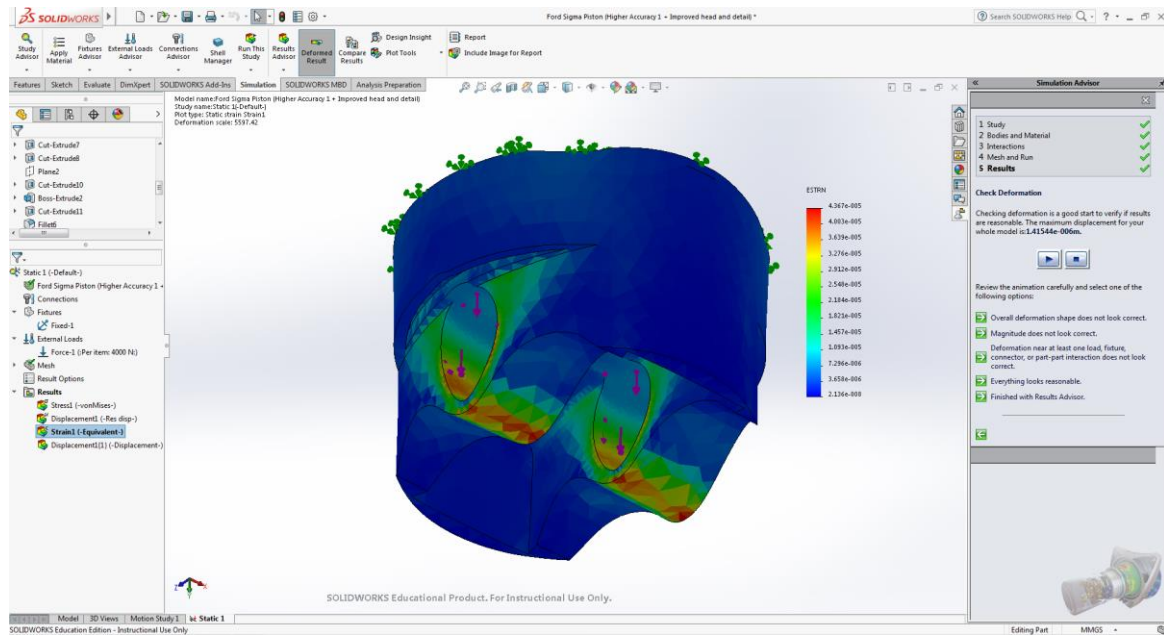


Figure 8.0.4.6 – Maximum Strain.

However, Gas Force (FGas) was required to be applied to the crown of this piston to show a true representation of the pressure applied to the piston during the combustion process.

Therefore, FGas of 84.135BAR was obtained from a previous Advanced Engine Design assignment and converted to PSI using the following equation. (Jay, 2016)

$$84.135 \text{ Bar} (14.503773800722) = 1220.275 \text{ PSI}$$

(Conversion Website, 2017)

Consequently, a 4mm Curvature Based Mesh, Carbon Steel material, fixtures at the base of this piston and FGas of 1220.265 PSI were applied via FEA to the Solid Works model in *Figure 18.0.4.7*. Therefore, Maximum Von Mises Stress of $8.904 \times 10^7 \text{ N/m}^2$ was obtained.

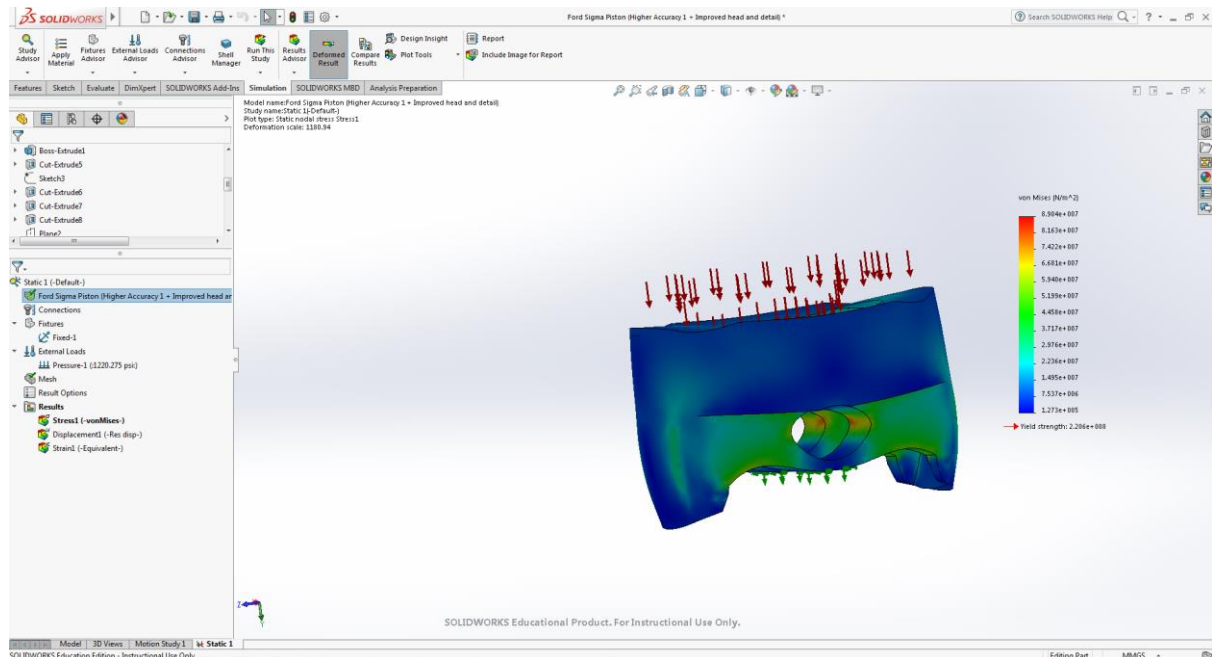


Figure 18.0.4.7 – Maximum Von Mises Stress.

Maximum displacement of $6.687 \times 10^{-3} \text{ mm}$ was obtained in *Figure 18.0.4.8*.

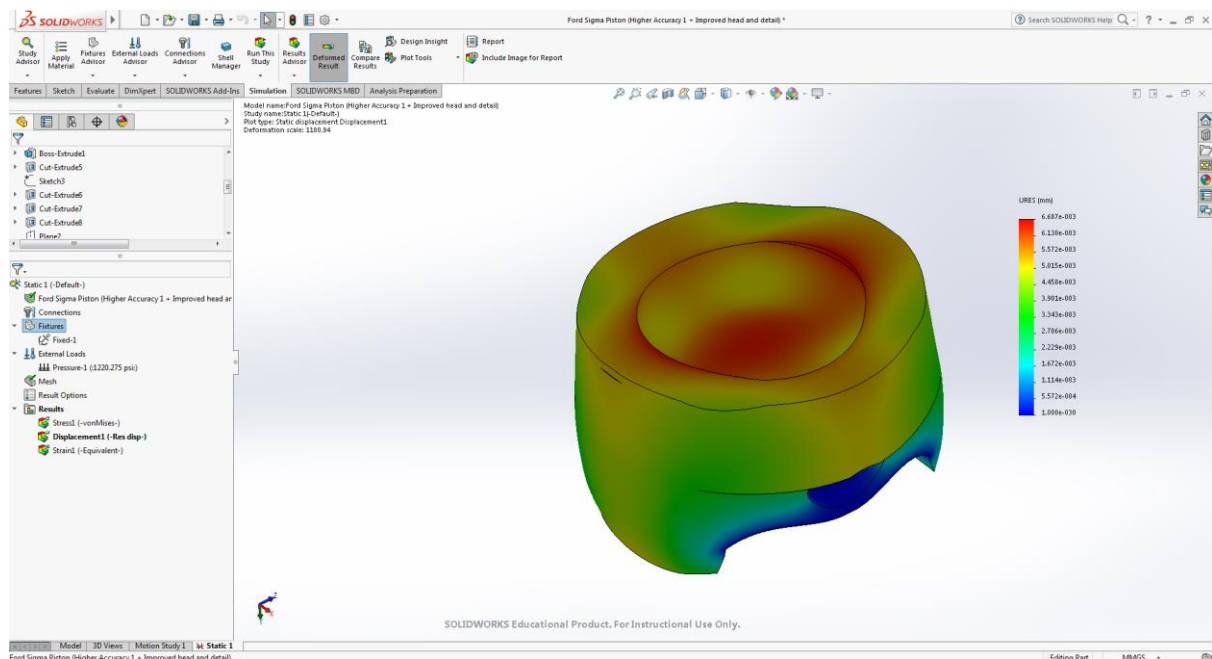


Figure 18.0.4.8 – Maximum Displacement.

Finally; Maximum Strain of 3.411×10^{-4} was obtained in *Figure 18.0.4.9*.

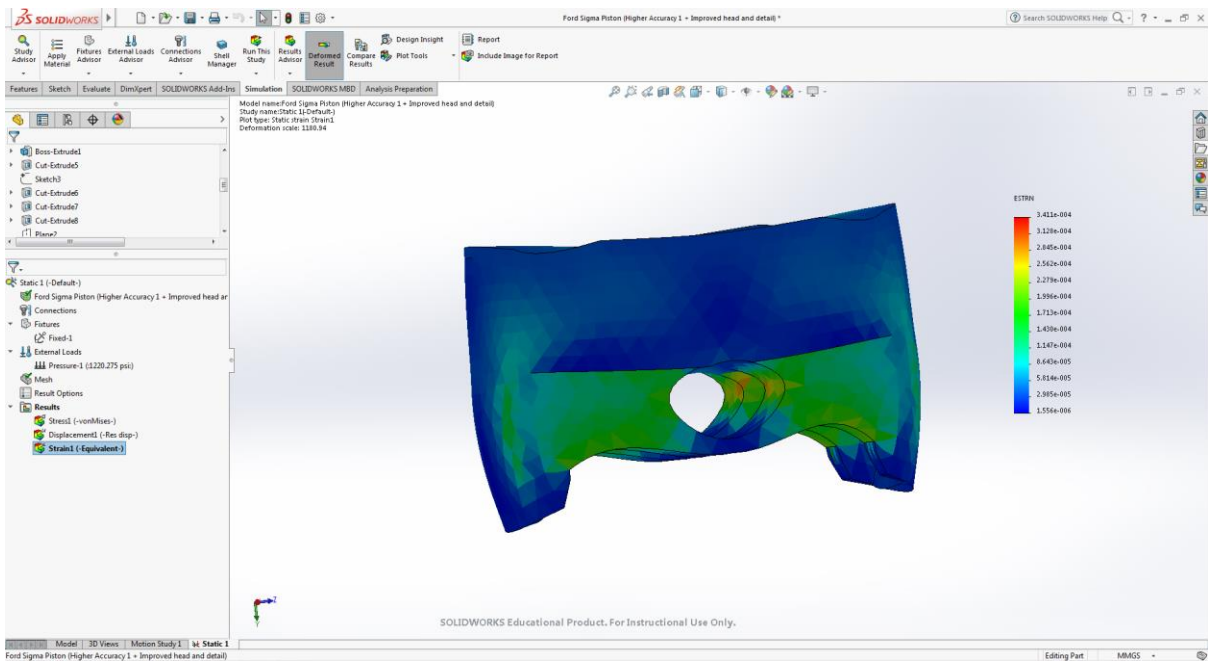


Figure 18.0.4.9 – Maximum Strain.

Run time of 23 seconds was obtained in *Figure 18.0.4.10*.

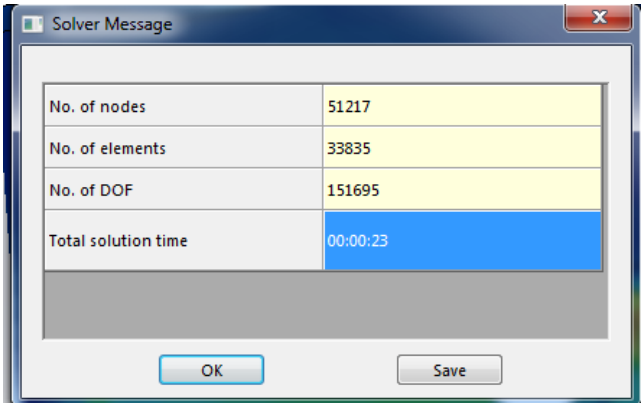


Figure 18.0.4.10 – Run Time.

10.0.4 – Ford Sigma Piston FEA:

Once several methods were demonstrated with the Ford Fox engine components, the optimum method to perform FEA was obtained. Consequently, this method was implemented on the Ford Sigma engine components when placed under FEA.

Therefore, FGas of 77.27BAR was obtained from a previous Advanced Engine Design assignment and converted to PSI using the following equation.
(Jay,2016)

$$77.27\text{Bar}(14.503773800722) = 1120.71\text{PSI}$$

(Conversion Website, 2017)

FGas of 1120.71 PSI was then applied to the crown of the piston, displayed in *Figure 18.0.5*.

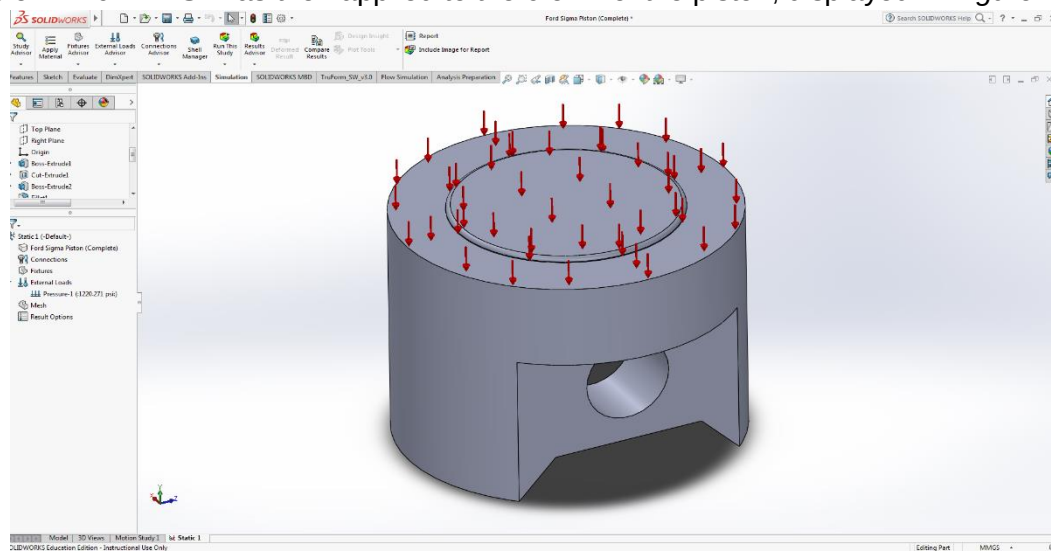


Figure 18.0.5 – Pressure Application.

The base of the piston was fixed, displayed in *Figure 18.0.5.1*.

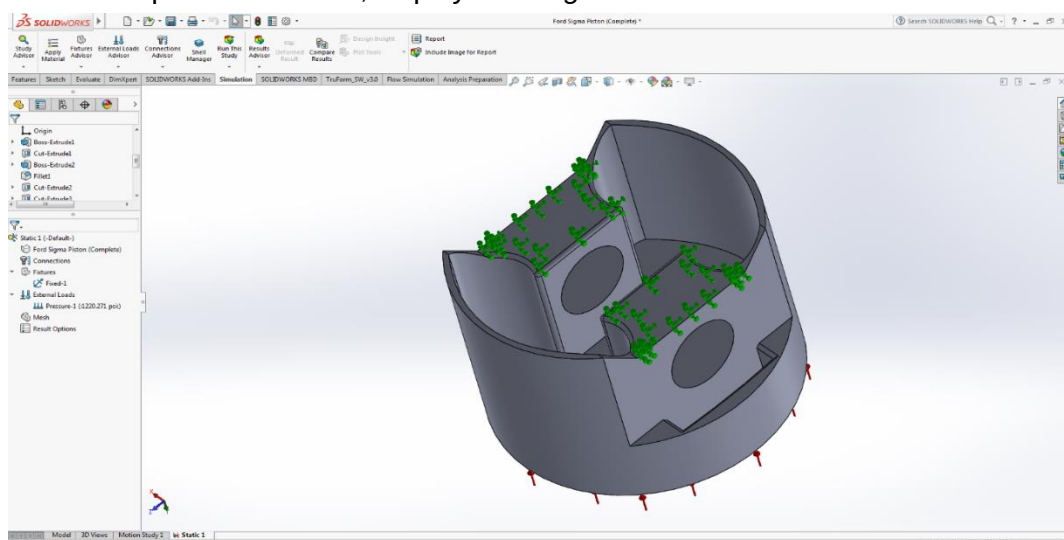


Figure 18.0.5.1 – Fixture Application.

Next the higher accuracy 4mm Curvature Based mesh and Cast Carbon Steel material was applied to the piston, displayed in Figure 18.0.5.2. Cast Carbon Steel was applied due to being listed as the material for production of the Ford Sigma pistons on the Burton Power website. (Burton Power, 2017)

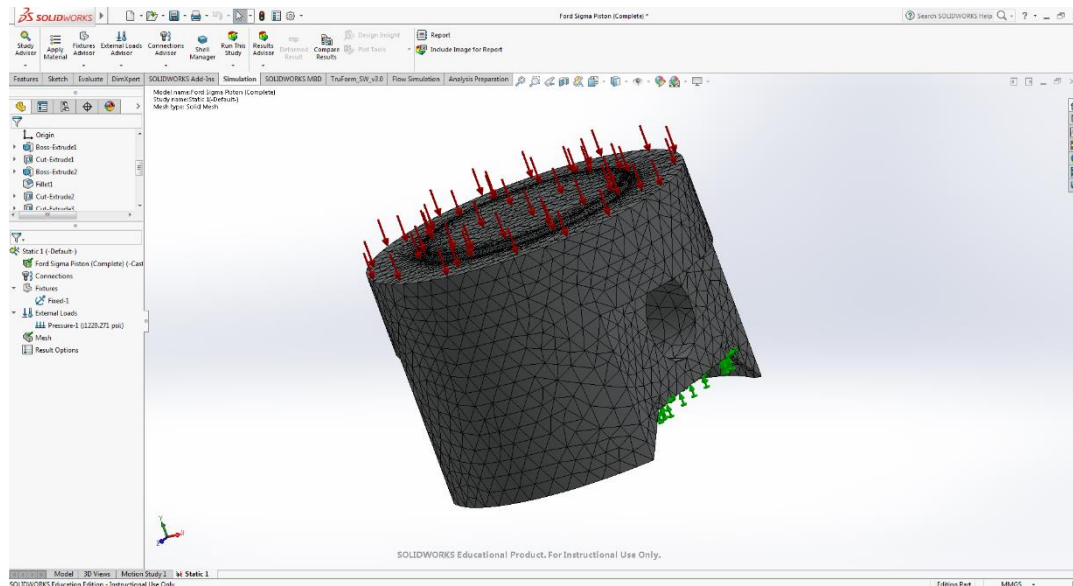


Figure 18.0.5.2 – Mesh and Material Application.

Simulations were then run and the following results were produced. Maximum Von Mises Stress of $3.779 \times 10^8 \text{ N/m}^2$ was obtained in Figure 18.0.5.3.

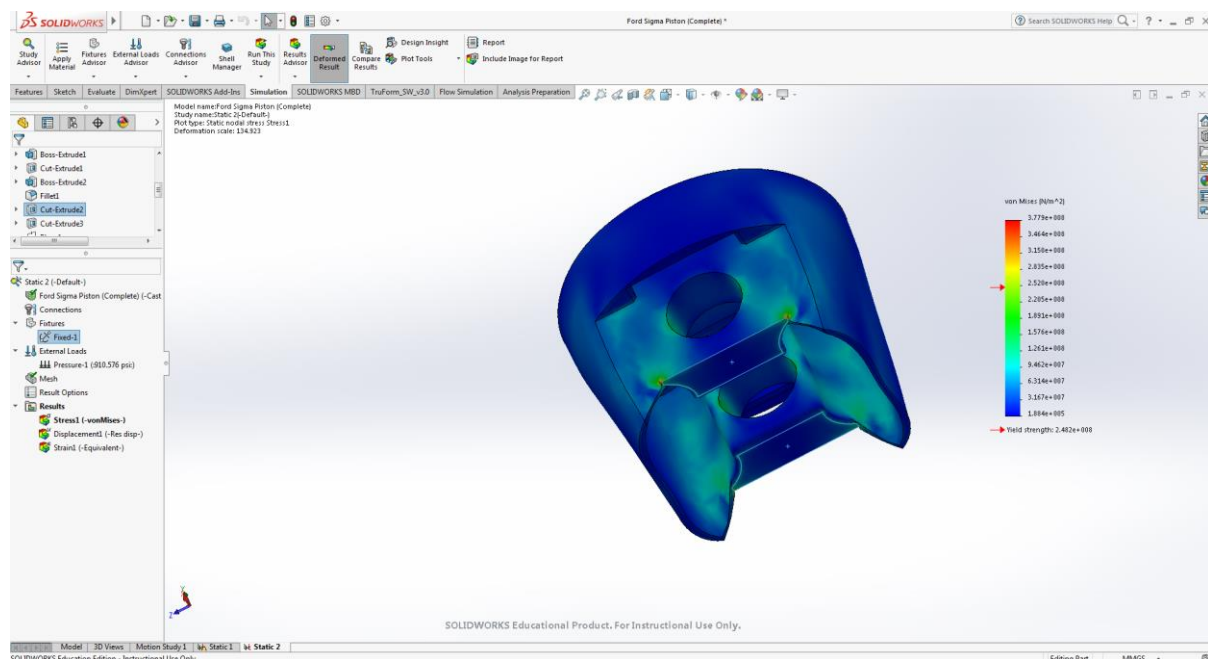


Figure 18.0.5.3 – Maximum Von Mises Stress.

A Maximum Displacement of 6.036×10^{-2} mm was obtained in Figure 18.0.5.4.

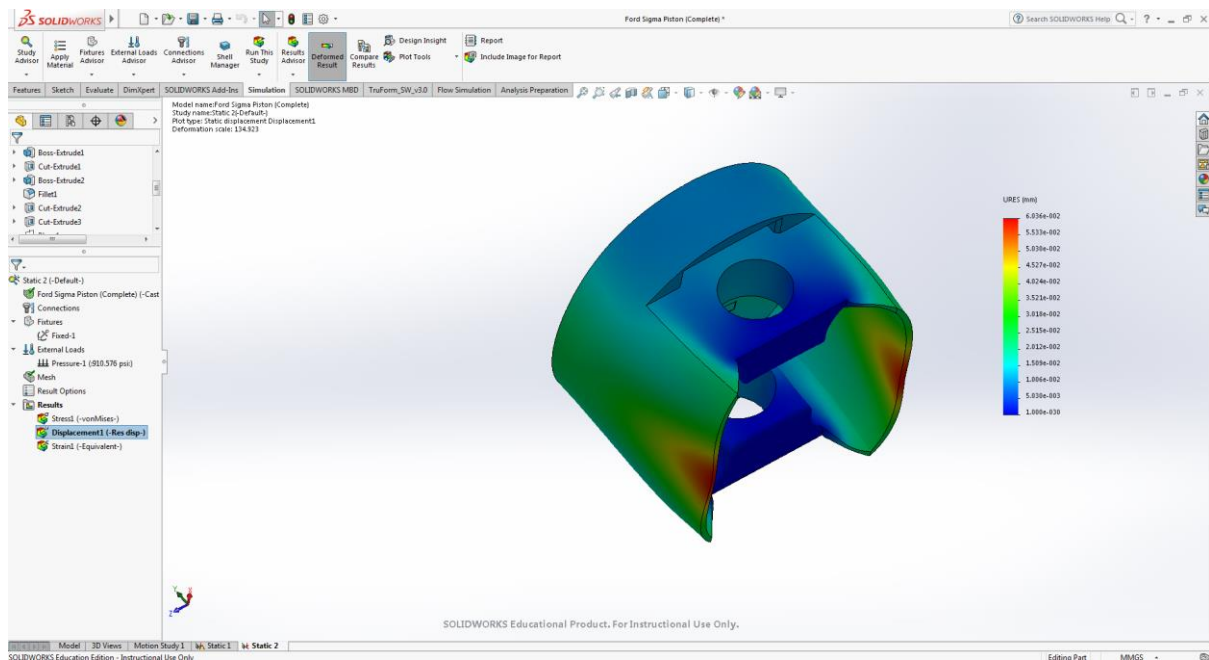


Figure 18.0.5.4 – Maximum Displacement.

Maximum Strain of 1.517×10^{-3} was obtained in Figure 18.0.5.5.

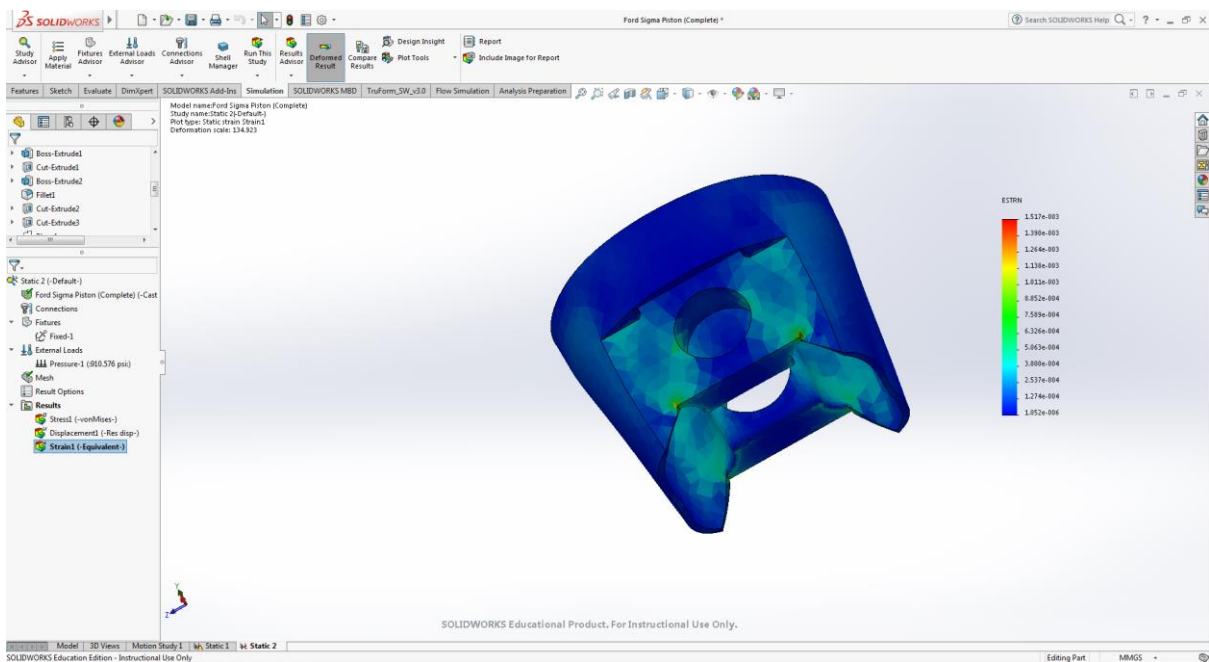
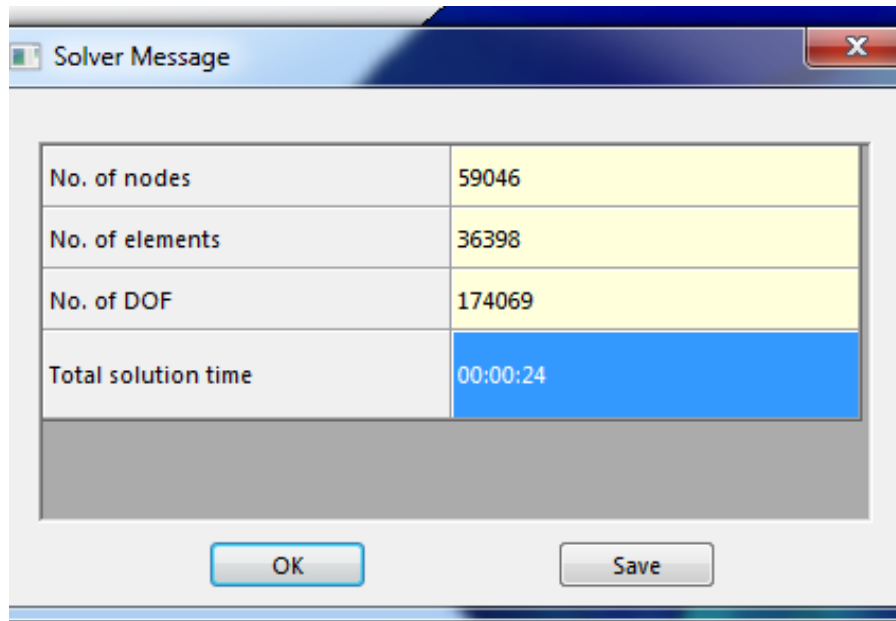


Figure 18.0.5.5 – Maximum Strain.

Finally; Overall run time was obtained from the Solver Message function, displayed in *Figure 18.0.5.6*.



No. of nodes	59046
No. of elements	36398
No. of DOF	174069
Total solution time	00:00:24

Figure 18.0.5.6 – Run Time.

10.0.5 – Ford Sigma Con-Rod FEA:

The 4mm Curvature Based mesh, Plain Carbon Steel and two separate point loads of 17207.5N (half of the 34415N FNet obtained from previous assignment) was applied to adjacent ends of the Gudgeon pin. While a fixture was applied to the Crank Pin, displayed in *Figure 18.0.6*.

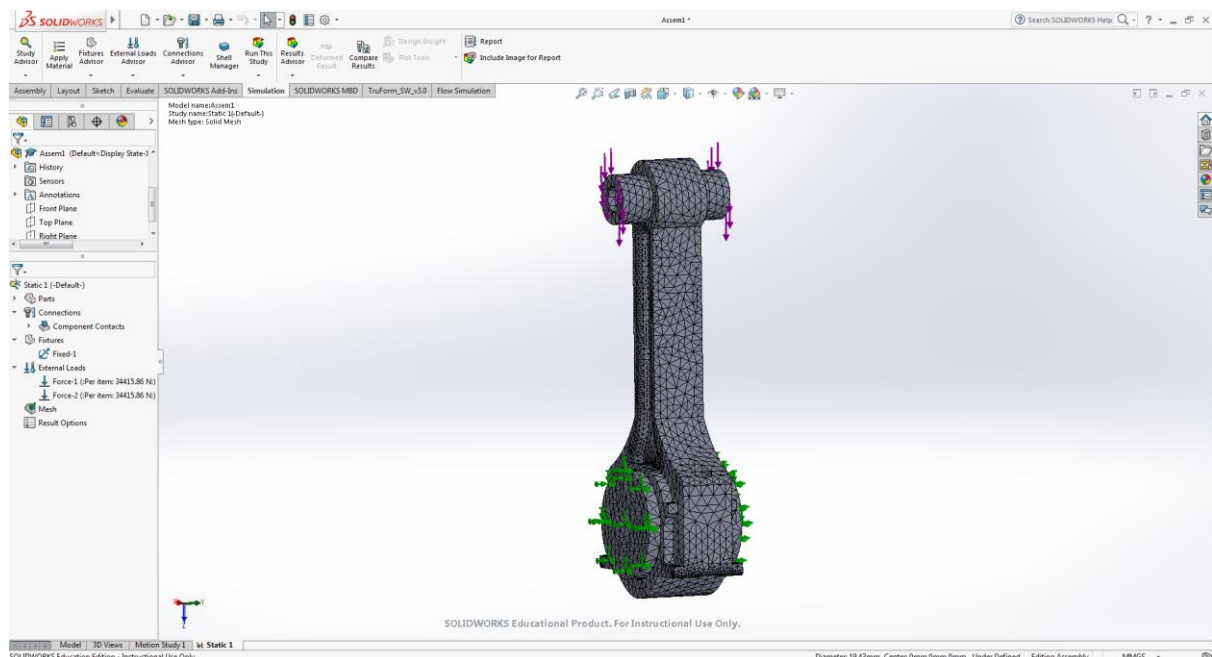


Figure 18.0.6 – Mesh, Force and Fixture Application.

Maximum Von Mises Stress of $8.085 \times 10^8 \text{ N/m}^2$ is achieved in *Figure 18.0.6.1*.

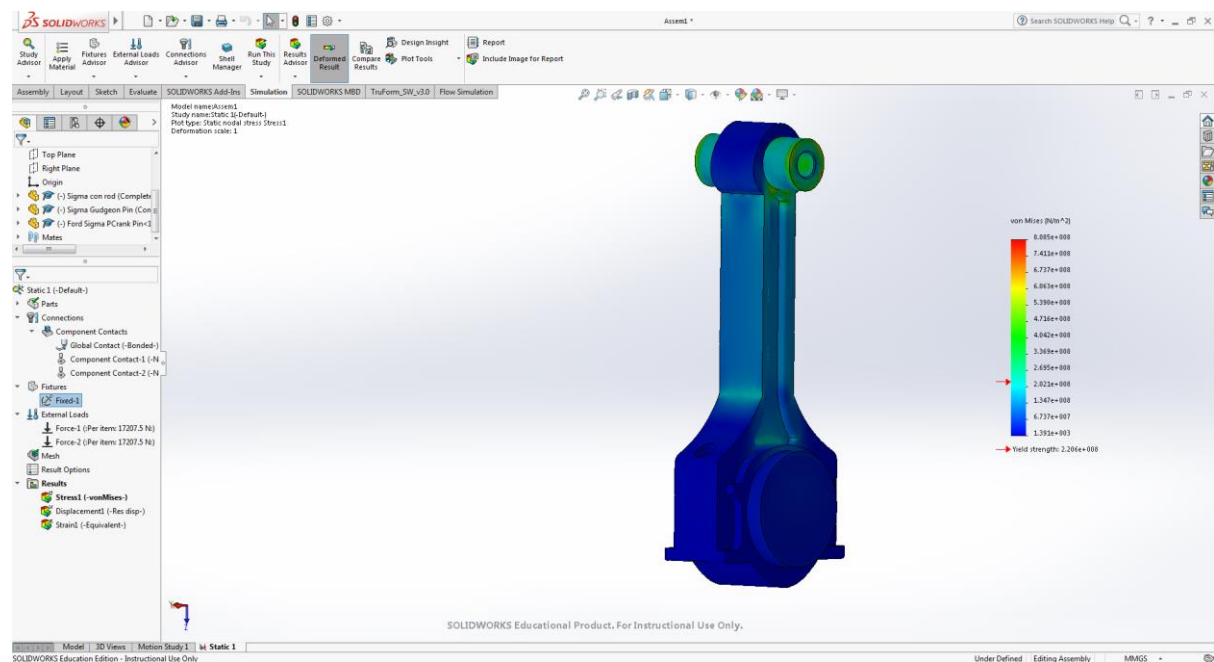


Figure 18.0.6.1 – Maximum Von Mises Stress.

Maximum Displacement of 1.607mm is obtained in *Figure 18.0.6.2*.

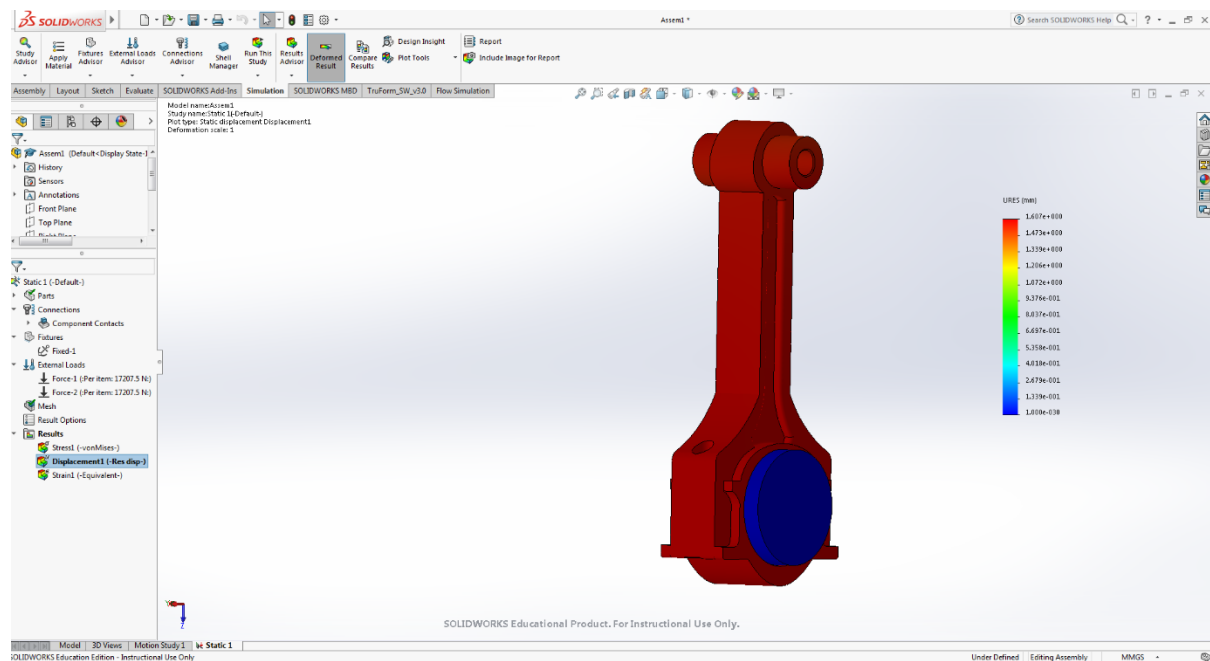


Figure 18.0.6.2 – Maximum Displacement.

Maximum Strain of 2.228×10^{-3} is acquired in Figure 18.0.6.3.

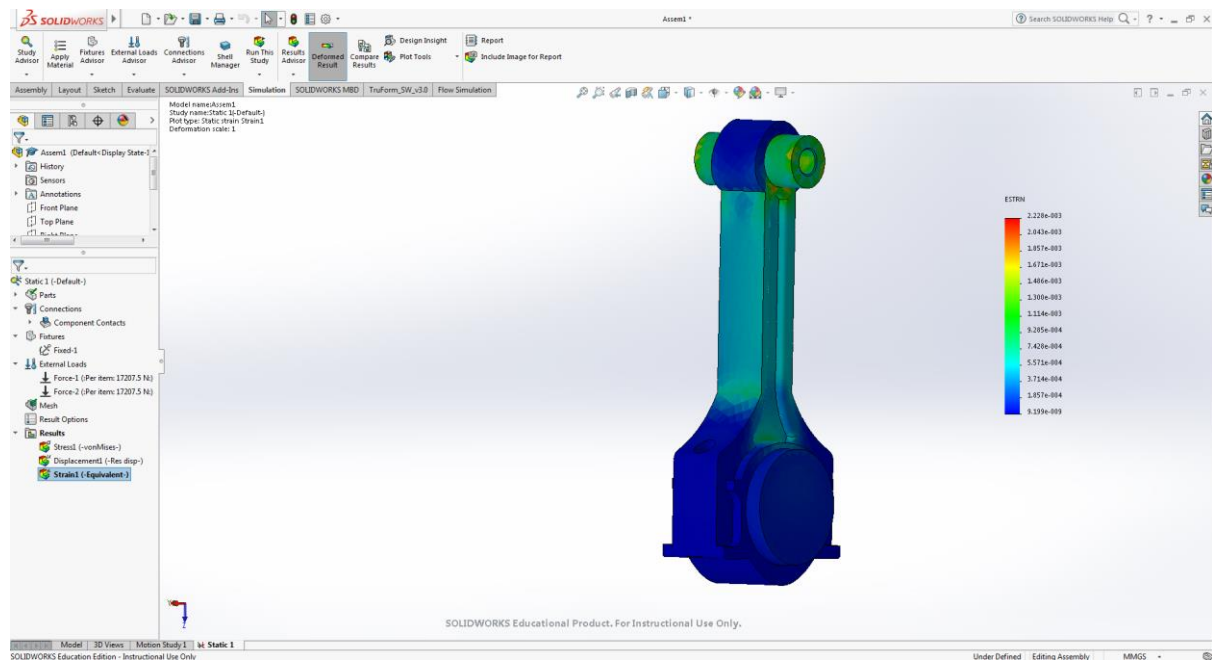


Figure 18.0.6.3 – Maximum Strain.

10.0.6 – Wankel Rotary Rotor FEA:

The final engine to undertake FEA is the Wankel Rotary engine. Force of 32891.7N was applied to the face of the rotor, including the combustion cavity and fixtures were applied to the centre diameter of the rotor. Consequently, Maximum Von Mises Stress of 1.840×10^8 N/m² was obtained in Figure 18.0.7.

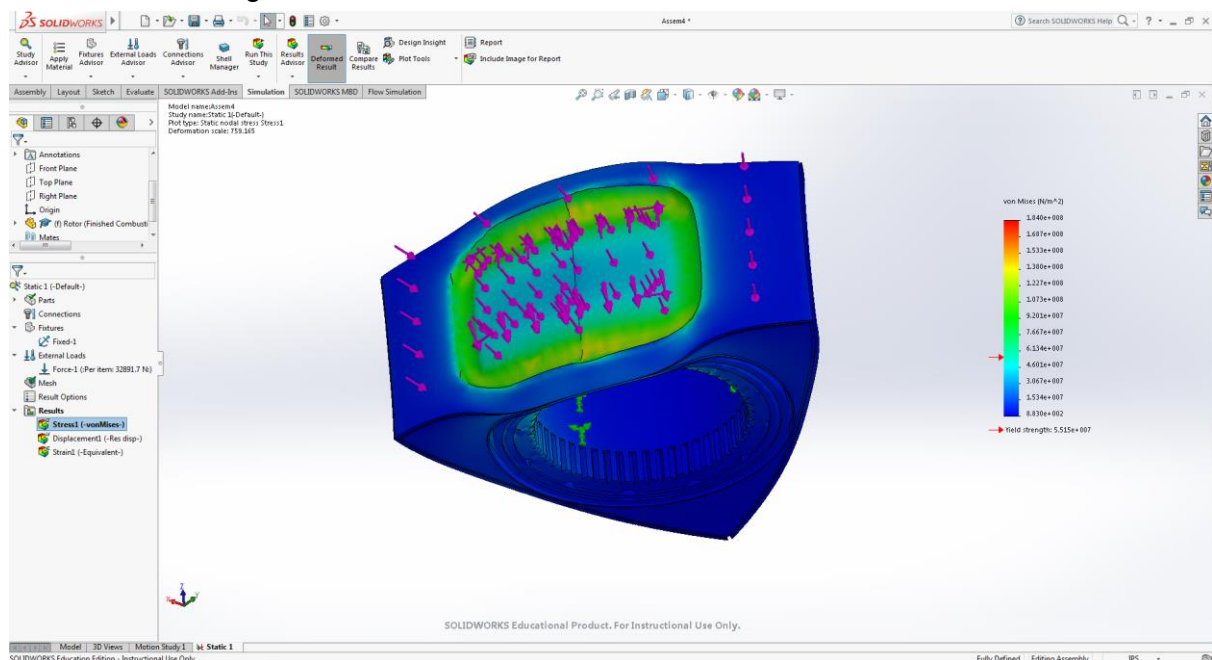


Figure 18.0.7 – Maximum Von Mises Stress.

Maximum Displacement of 2.680×10^{-2} mm was obtained in *Figure 18.0.7.1*.

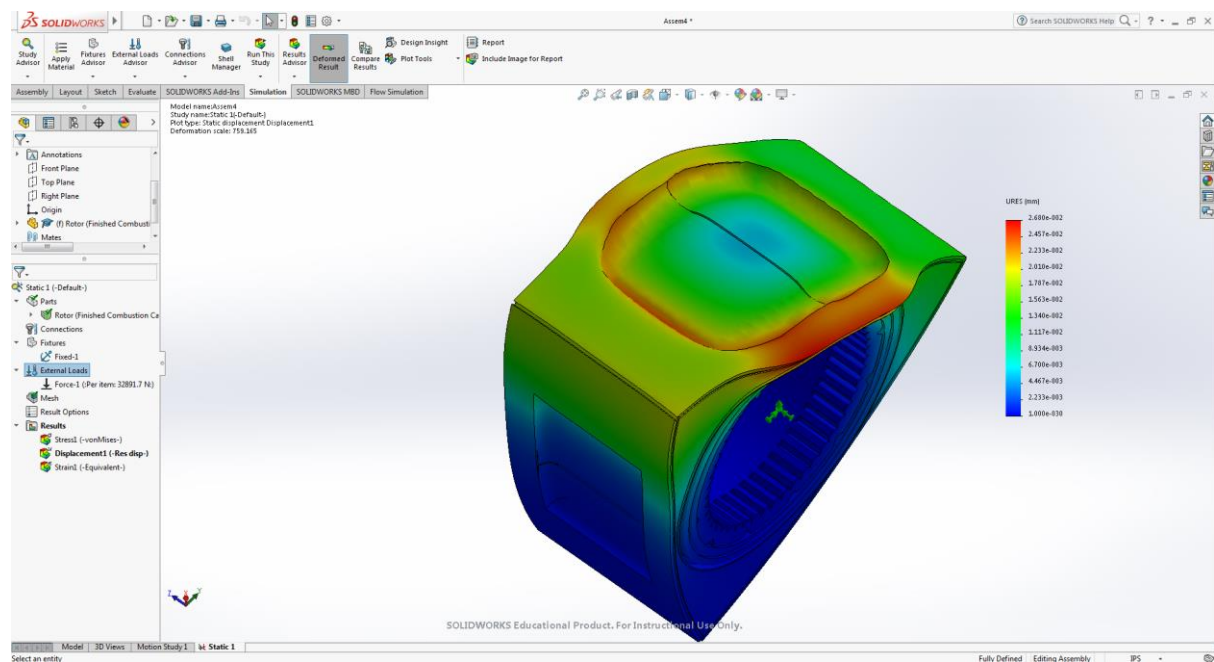


Figure 18.0.7.1 – Maximum Displacement.

Maximum Strain of 1.790×10^{-3} was obtained, as displayed in *Figure 18.0.7.2*.

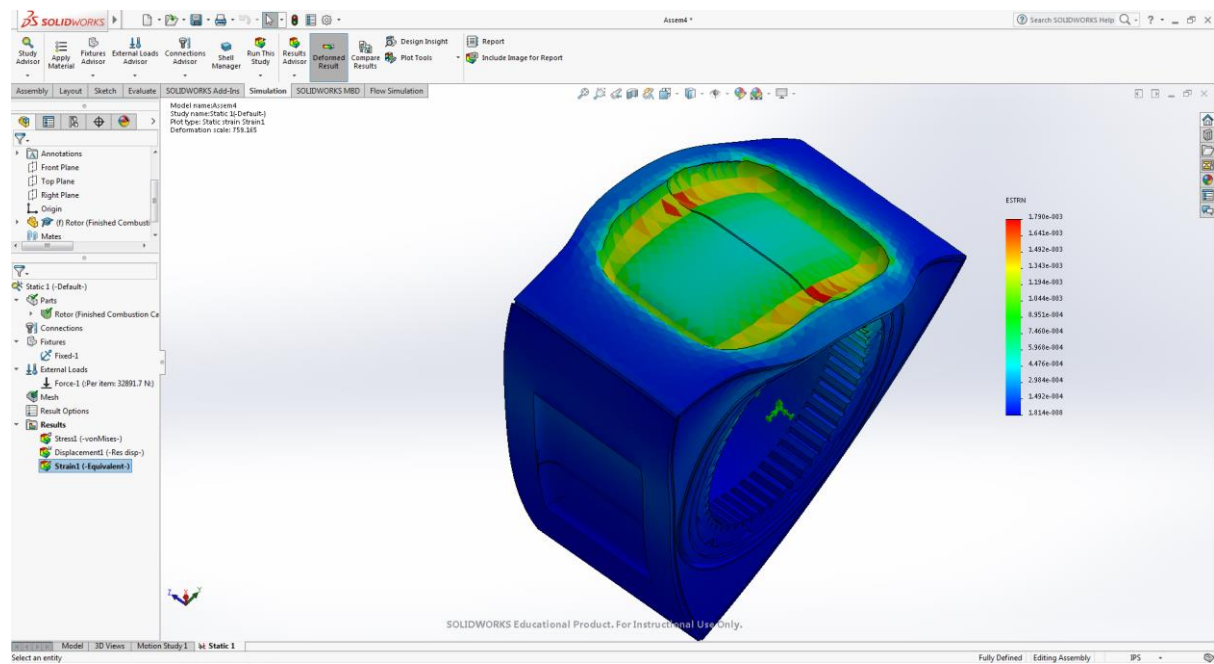


Figure 18.0.7.2 – Maximum Strain.

The 96.523BAR pressure obtained from the Wankel Rotary data produced was then applied and altered using the equation below to produce a pressure in units of PSI.

$$96.523\text{Bar}(14.503773800722) = 1354.965\text{PSI}$$

This pressure and a 4mm Curvature Based Mesh was applied to a Solid Works model of the Rotor, displayed in Figure 18.0.7.3.

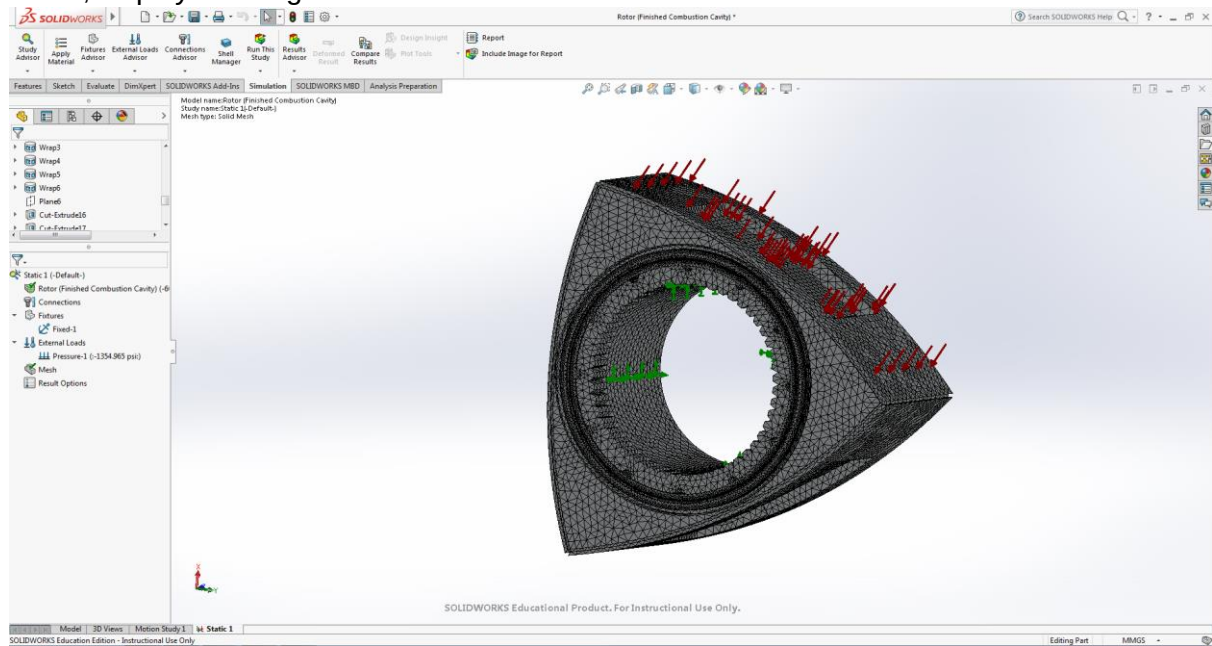


Figure 18.0.7.3 – Application of Fixtures and Pressure.

Subsequently FEA was applied to the rotor and a Maximum Von Mises Stress of $8.610 \times 10^7 \text{ N/m}^2$, displayed in Figure 18.0.7.4.

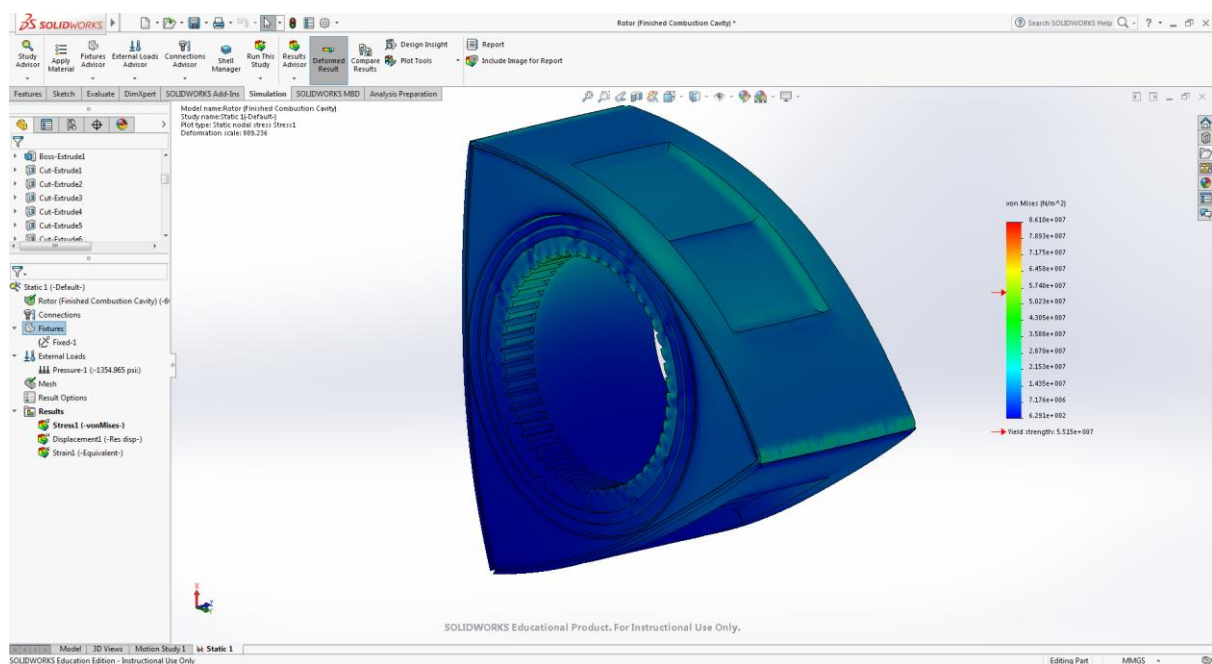


Figure 18.0.7.4 – Maximum Von Mises Stress.

Maximum Displacement of 2.299×10^{-2} mm was obtained in Figure 18.0.7.5.

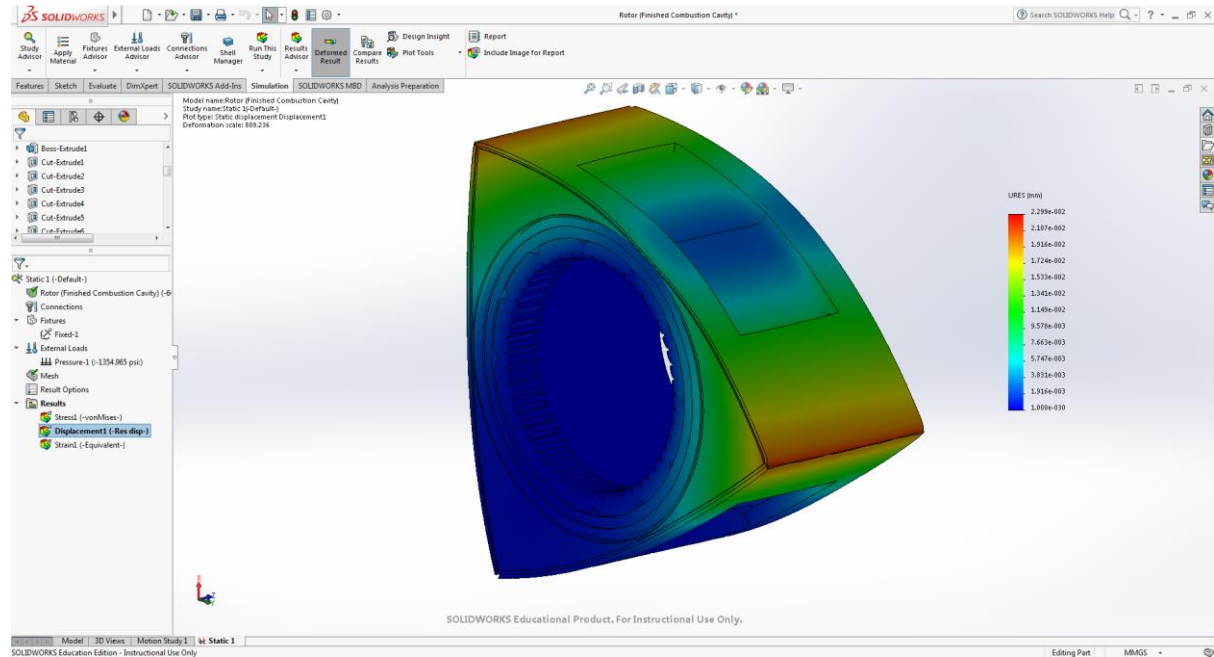


Figure 18.0.7.5 – Maximum Displacement.

Maximum Strain of 5.309×10^{-4} was obtained in Figure 18.0.7.6.

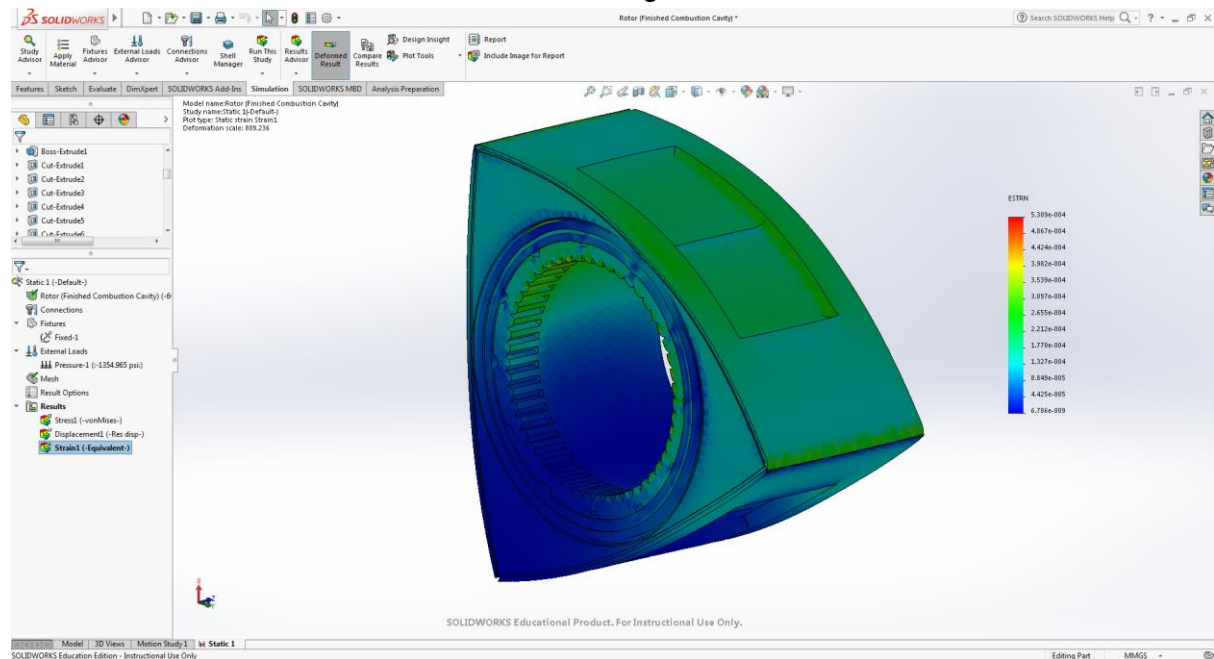


Figure 18.0.7.6 – Maximum Strain.

However; further accuracy was required. Therefore, a 1mm mesh refinement was applied to the Combustion Cavity, presented in *Figure 18.0.7.7*. Where the largest region of stress concentration occurred.

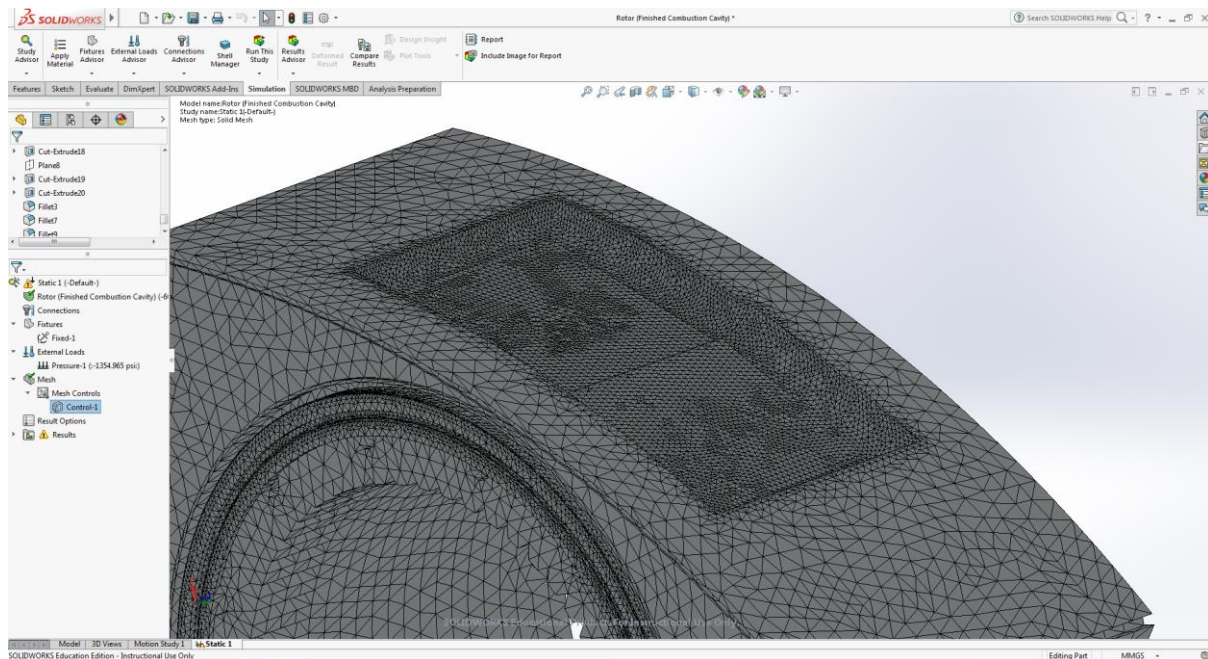


Figure 18.0.7.7 – 1mm Mesh Refinement in Combustion Cavity.

Therefore, FEA was re-run and a new Maximum Von Mises Stress of 4.282×10^7 N/m² was obtained in *Figure 18.0.7.8*.

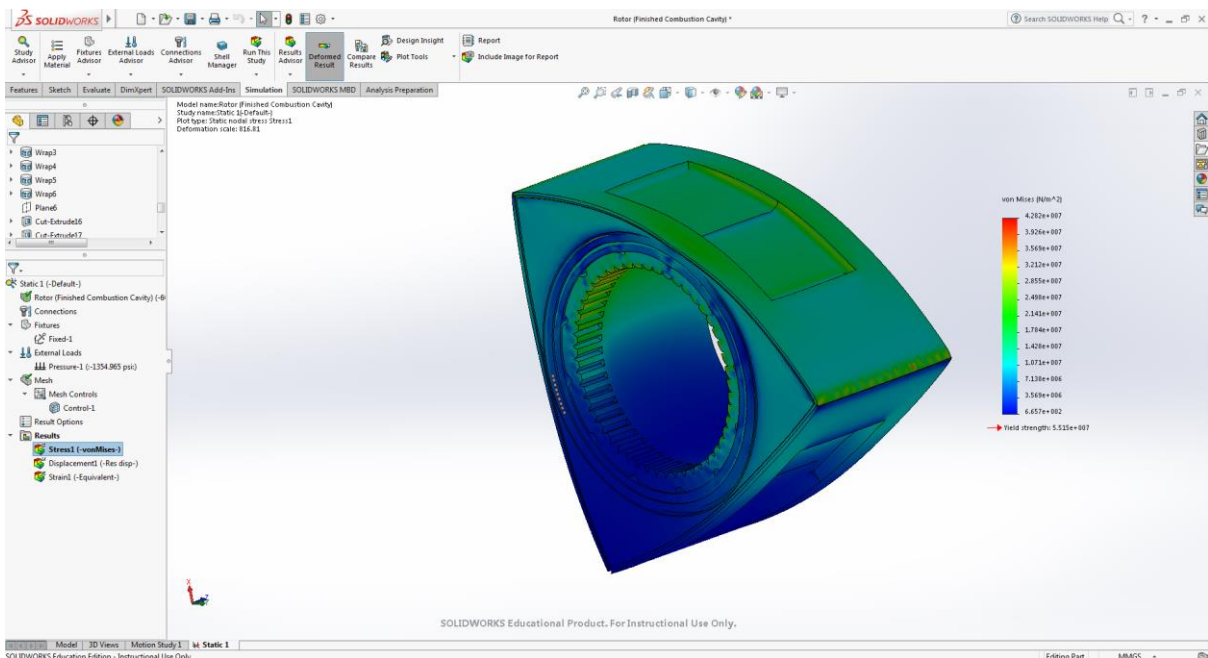


Figure 18.0.7.8 – Maximum Von Mises Stress.

New Maximum Displacement of 2.276×10^{-2} mm was obtained in *Figure 18.0.7.9*.

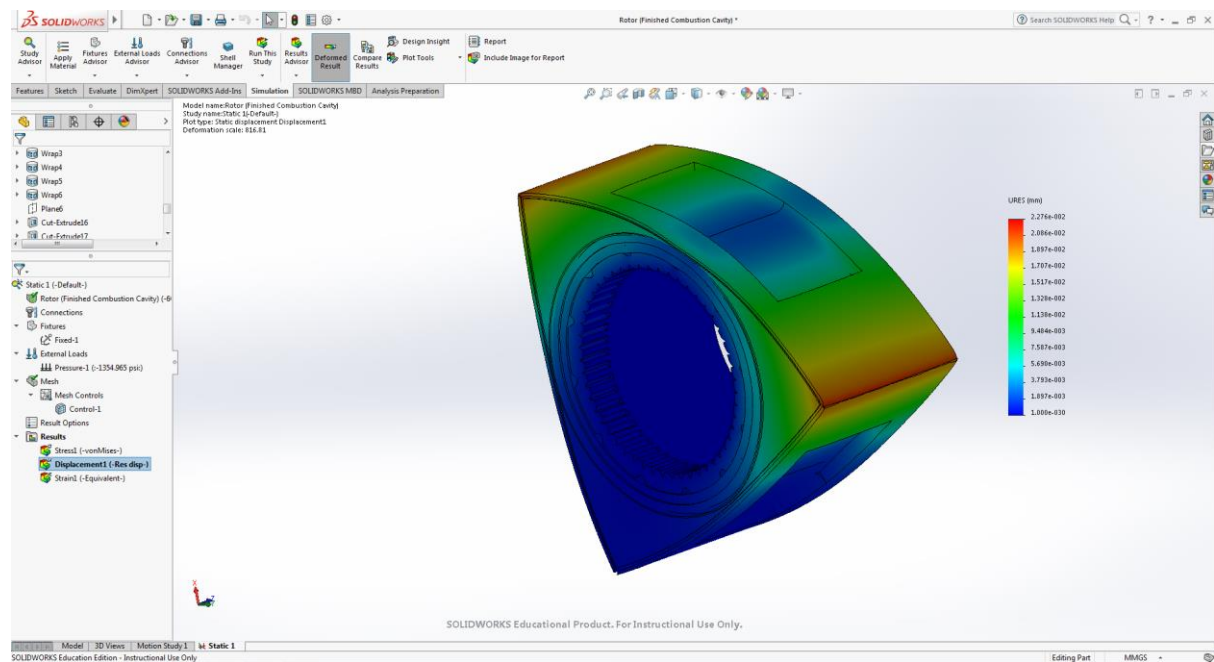


Figure 18.0.7.9 – Maximum Displacement.

Finally; Maximum Strain of 4.438×10^{-4} was obtained in *Figure 18.0.7.10*.

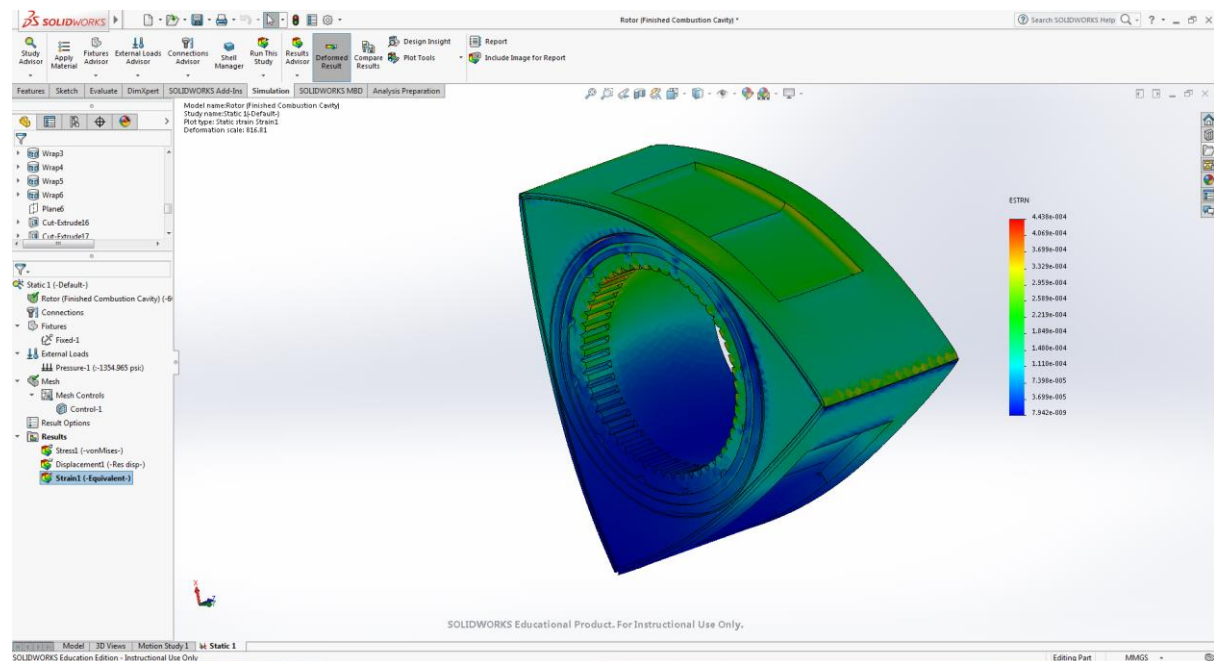
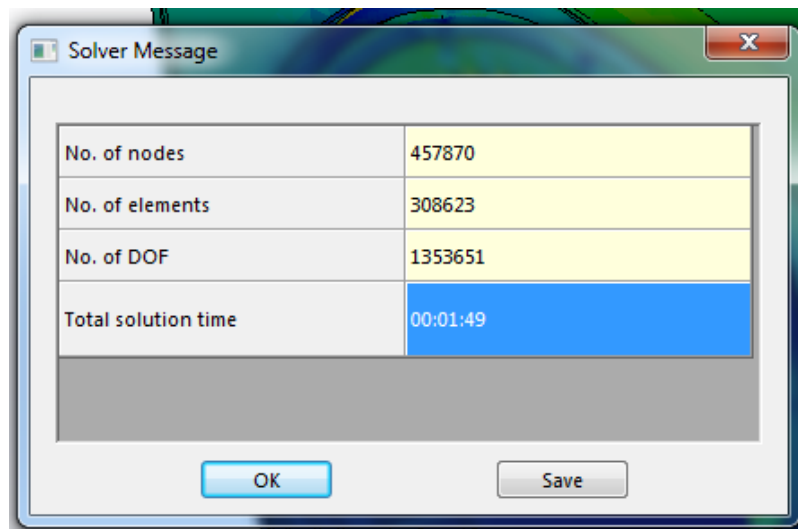


Figure 18.0.7.10 – Maximum Strain.

Run time of the final Wankel Rotary FEA was then obtained in *Figure 18.0.7.11*.



The image shows a 'Solver Message' dialog box with a table of FEA statistics. The table has two columns: a label and a value. The values are 457870 for nodes, 308623 for elements, 1353651 for DOF, and 00:01:49 for total solution time. The 'Total solution time' row is highlighted in blue. At the bottom are 'OK' and 'Save' buttons.

No. of nodes	457870
No. of elements	308623
No. of DOF	1353651
Total solution time	00:01:49

Figure 18.0.7.11 – Run Time.

10.0.7 – Chapter References:

1. Aqua Calc 2017, Pressure Conversion Table, Viewed 22nd April 2017, <http://www.aqua-calc.com/convert/pressure/bar-to-pascal>
2. Burton Power 2017, Nural Piston Ford Zetec SE, Viewed 6th April, <http://www.burtonpower.com/nural-piston-ford-zetec-se-1-4-16v-76mm-10-3-1-cr-24113-std.html>
3. Conversion Website 2017, Bars to Pounds Per Square Inch (BAR to PSI), Viewed 19th April 2017, <http://www.conversion-website.com/pressure/bar-to-pound-per-square-inch.html>
4. D Sport Performance + Tech Magazine 2017, Ford EcoBoost 1.6L Analysis, Viewed 19th April 2017, <http://dsportmag.com/the-tech/ford-ecoboost-1-6l-analysis-turbocharged-direct-injection-ccvtc/>
5. Montazersadgh & Fatemi, 2007, Engineering, Stress Analysis and Optimisation of Crankshafts Subject to Dynamic Loading, University of Toledo, Ohio.
6. O'Donnell Consulting Engineers 2017, Finite Element Analysis, Viewed 19th January 2017, <http://www.odonnellconsulting.com/what-we-do/engineering/mechanical-engineering/finite-element-analysis/?gclid=CJ7cktWXztECFUETGwod2XsNSA>
7. Jay, 2016, Advanced Engine Design Assignment 1A – “Con Rod, Accelerations, Forces and Displacements”, Advanced Engine Design, M6X00410, UWTSD University.

10.0.7.1 – Chapter Literature Review:

1. Aqua Calc – Enabled Bar to Pascal Conversion.
2. Burton Power – Obtaining material used to produce Ford Sigma pistons.
3. Conversion Website – Obtained equation to convert Bar to PSI.
4. D Sport – Obtained Ford EcoBoost cylinder bore.
5. Montazersadgh & Fatemi – Investigated FEA for Industrial application of single span of crankshaft.
6. O'Donnell Consulting Engineers – Investigated Finite Element Analysis.
7. Jay – Obtained engine data.

Chapter 11.0 – Thermal Equations:

The final section of the Major Project required the production of thermal calculations to determine engine component heat transfer.

11.0.1 – Ford Fox Turbocharged Thermal Calculations:

The first engine thermal calculations were produced for was the Ford Fox. The precise power output in PS (Pferdestärke) was obtained via the official Ford website. (What Car, 2011)

182PS (Ford, 2017)

Power output from PS to HP was then produced, using the following equation.

$$\text{Converting PS to HP} = \frac{182}{1.01427772651} = 179.438\text{HP (Convert Units, 2017)}$$

Thermal efficiency of an ICE = 30%. (Energy Education, 2017)

Consequently; 30% ICE efficiency is used in the following equation.

$$\text{Fuel Power} = \frac{179.438}{0.3} = 598.127\text{HP}$$

Approximately 30% of thermal energy is lost via the oil and water. (Physics Forum, 2008)

$$\text{Heat Power into Oil and Water} = \frac{1}{3}(598.127) = 199.376\text{HP}$$

The four-cylinder arrangement of the Ford Fox engine, subsequently required power output to be divided between each cylinder.

Therefore:

$$25\% \text{ into each cylinder} = \frac{25}{100}(179.438) = 44.86\text{HP}$$

HP was converted to KW using the following equation.

$$\text{HP to KW} = 44.86(0.746) = 33.466\text{KW (Rapid Tables, 2017)}$$

The conversion was then divided between the four pistons; therefore:

$$\frac{33.466}{4} = 8.367\text{Kw into each piston}$$

TCold is controlled via the oil and the crankshaft. Therefore:

$$T_{\text{Cold}} = 100^{\circ}\text{C}$$

TCold is then entered into the Heat transfer equation. Thermal Conductivity (K) is dependent on the materials used for engine. Thermal conductivity describes a materials ability to conduct heat. The Ford Fox piston is produced from Plain Carbon Steel. Consequently “K” is obtained for the Ford Fox at the ambient temperature of 25°C.

$$K = 43 \text{ W/m.K}$$

(Engineering Toolbox, 2017)

The distance heat is required to travel, from the piston crown to the skirt is measured with digital Vernier callipers, this is listed as “d”, displayed via the orange arrow in *Figure 19.0*.



Figure 19.0 – Distance “d”. (Jay, 2017)

$$d = 55.107\text{mm} = 0.055107\text{m}$$

Area is the final requirement to enable production of the calculation. Therefore, the area is obtained from the Solid Works model of the Ford Fox piston via the measurement evaluation tool.

Therefore:

$$A = 5505.61\text{mm}^2 = 0.00550561\text{m}^2 = 5505.61 \times 10^{-6}\text{m}^2$$

Heat Flux is then calculated.

$$\text{Heat Flux} = \frac{Kw}{\text{Area}} = \frac{8.367}{5505.61 \times 10^{-6} \text{m}^2} = 1519.723 \text{Kw/m}^2$$

Heat Flux, otherwise known as thermal flux is the degree of heat energy transfer through a specific surface. (Cirimele & D’Elia, 2012)

$$\text{Heat Flux} = \frac{K}{d} (T_{\text{Hot}} - T_{\text{Cold}})$$

Heat flux is converted to Watts.

$$1519723\text{W} = \frac{43}{0.055107} (T_{\text{Hot}} - T_{\text{Cold}})$$

$$T_{Hot} - T_{Cold} = \frac{1519723(0.055107)}{43}$$

The equation is re-arranged to form an equation for T_{Hot}.

$$T_{Hot} = T_{Cold} \left(\frac{1519723(0.055107)}{43} \right)$$

$$T_{Hot} = 100 + 1947.613 = 2047.613^{\circ}\text{C}$$

11.0.2 – Ford Sigma Naturally Aspirated Thermal Calculations:

The Ford Sigma was the next engine for thermal calculations to complete.

115HP (Ford, 2017)

$$\text{Fuel Power} = \frac{115}{0.3} = 383.33\text{HP}$$

$$\text{Heat Power into Oil and Water} = \frac{1}{3}(383.33) = 127.777\text{HP}$$

$$25\% \text{ into each cylinder} = \frac{25}{100}(115) = 28.75\text{HP}$$

$$\text{HP to KW} = 28.75(0.746) = 21.448\text{KW}$$

Four piston assembly; Therefore:

$$\frac{21.448}{4} = 5.362\text{Kw into each piston}$$

$$T_{Cold} = 100^{\circ}\text{C}$$

The Ford Sigma piston is also produced from Carbon Steel. Consequently; “K” is the same as the Ford Fox engine.

$$K = 43 \text{ W/m.K}$$

(Engineering Toolbox, 2017)

$$d = 54.53\text{mm} = 0.05453\text{m}$$

$$A = 4547.01\text{mm}^2 = 0.00454701\text{m}^2 = 4547.01 \times 10^{-6}\text{m}^2$$

$$\text{Heat Flux} = \frac{Kw}{\text{Area}} = \frac{5.362}{4547.01 \times 10^{-6} \text{m}^2} = 1179.236\text{Kw/m}^2$$

$$\text{Heat Flux} = \frac{K}{d} = (T_{Hot} - T_{Cold})$$

Heat flux is converted to Watts.

$$1179236\text{W} = \frac{43}{0.05453}(T_{Hot} - T_{Cold})$$

$$T_{Hot} - T_{Cold} = \frac{1179236(0.05453)}{43}$$

The equation is re-arranged to form an equation for T_{Hot}.

$$T_{Hot} = T_{Cold} \left(\frac{1179236(0.05453)}{43} \right)$$

$$T_{Hot} = 100 + 1495.436 = 1595.436^{\circ}\text{C}$$

11.0.3 – Wankel Rotary Naturally Aspirated Thermal Calculations:

The final set of thermal calculations required are for the Wankel Rotary engine.

Mazda Rx8 = 192PS

$$\text{Converting PS to HP} = \frac{192}{1.01427772651} = 189.297\text{HP}$$

$$\text{Fuel Power} = \frac{189.297}{0.3} = 630.99\text{HP}$$

$$\text{Heat Power into Oil and Water} = \frac{1}{3}(630.99) = 210.33\text{HP}$$

Assume 50% into rotors; Therefore:

$$= \frac{50}{100}(189.287) = 94.644\text{HP}$$

$$\text{Conversion} = 94.644(0.746) = 70.604\text{Kw}$$

Twin-Rotor design, therefore two Rotors required.

$$\text{Each rotor} = \frac{70.604}{2} = 35.302\text{Kw into each rotor}$$

Each rotor has three sides, consequently;

$$\text{Each side of the rotor} = \frac{35.302}{3} = 11.767\text{KW}$$

$$T_{Cold} = 100^{\circ}\text{C}$$

Unlike the previous two engines, the Wankel Rotary engine rotors are produced from Aluminium.

Therefore:

$$K = 205W/m.K$$

(Engineering Toolbox, 2017)

Due to the rotor shape, a minimum dimension represented by the orange arrow and maximum dimension represented by the green arrow was measured, as displayed in *Figure 19.0.1*. An average of the two was then calculated.



Figure 19.0.1 – Measuring value of “d” of the rotor. (Jay, 2017)

$$d = 45.867mm = 0.045867m$$

$$A = 15582.99mm^2 = 0.01558299m^2 = 15582.99 \times 10^{-6}m^2$$

$$Heat\ Flux = \frac{Kw}{Area} = \frac{11.767}{15582.99 \times 10^{-6} m^2} = 755.118Kw/m^2$$

$$Heat\ Flux = \frac{K}{d} = (THot - TCold)$$

Heat flux is converted to Watts.

$$755118W = \frac{205}{0.045867} (THot - TCold)$$

$$THot - TCold = \frac{755118(0.045867)}{205}$$

The equation is re-arranged to form an equation for THot.

$$THot = TCold \left(\frac{755118(0.045867)}{205} \right)$$

$$THot = 100 + 168.951 = 168.951^{\circ}C$$

11.0.4 – Thermal Equations Summary:

Once the thermal equations have been completed for each engine platform, a summary was produced.

11.0.4.1 – Ford Fox:

The four-cylinder assembly requires the power output of the engine to be divided by four to accommodate each cylinder. The conversion from HP to KW then occurs and was divided by four, consequently accommodating for each piston within the four-cylinder assembly.

The assumption of TCold equalling 100°C due to control via the oil and crankshaft was then produced. Thermal Conductivity is required to be obtained dependant on engine component material. The distance heat is required to travel is physically measured and area of the piston crown is obtained from the Solid Works model produced.

Therefore, enabling Heat Flux to be calculated and consequently obtaining THot of 2047.613°C.

11.0.4.2 – Ford Sigma:

The lower power output of the Ford Sigma consequently produces a lower fuel power and thus a reduced quantity of heat power lost into the oil and water. An overall lower power output also produces less HP per cylinder and piston.

The Ford Sigma uses the same Thermal Conductivity as the Ford Fox, due to being produced from the same Carbon Steel material. However, the distance heat is required to travel through the piston is 0.577mm less for the Ford Sigma, compared to the Ford Fox. Also, the Ford Sigma piston has a smaller area by $958.6 \times 10^{-6} \text{m}^2$.

Due to the lower power output, heat flux was reduced by 340.487Kw/m^2 . Finally; THot was obtained at 1595.436°C. Consequently, a reduction of 452.177°C from the Ford Fox occurs, which is likely, due to turbocharged engines suffering from higher temperatures and pressures. (Bell, 1997)

11.0.4.3 – Wankel Rotary:

The final thermal calculations produced were for the Wankel Rotary engine, which has the highest power output of all three engines. Consequently; the rotary engine produces the largest fuel power. However, this also causes the largest heat loss via oil and water by 10.954HP. The assumption was created that power output is equally shared by half the amount into each rotor. Power output was then converted to Kw and divided by three for each face of the rotor.

Due to the rotor design, the face of the rotor has a much larger area compared to a traditional piston crown. Therefore, heat flux is reduced, due to the division of a large area. As previously mentioned in Chapter 10.0.3, the rotors are produced from Aluminium, which has a higher thermal conductivity of 205W/m.K, compared to 43W/m.K of Carbon Steel. Consequently, vastly reducing T_{Hot} to 168.951°C.

11.0.5 – Chapter References:

1. Bell, 1997, Automotive, Maximum Boost – Designing, Testing and Installing Turbocharger Systems, Bentley Publishers, Massachusetts.
2. Convert Units 2017, Convert PS to Horse Power – Conversion to Measurement Units, Viewed 3rd April 2017, <http://www.convertunits.com/from/ps/to/horsepower>
3. Cirimele & D'Elia, 2012, Processes, Measurement Techniques and Applications, Heat Flux, Nova Science Publishers, New York.
4. Energy Education 2017, Thermal Efficiency, Viewed 3rd April 2017, http://energyeducation.ca/encyclopedia/Thermal_efficiency
5. The Engineering Tool Box 2017, Thermal Conductivity of Common Materials and Gases, Viewed 17th April 2017, http://www.engineeringtoolbox.com/thermal-conductivity-d_429.html
6. Ford 2017, Ford Fiesta ST-2, Viewed 4th April 2017, <http://www.ford.co.uk/ConfigureyourCar/Fiesta/ST-2>
7. Ford 2017, Ford Duratec 1.6 Ti-VcT, Viewed 3rd April, [http://www.fordpers.be/content/assets/news-2011/pdf/Ford_tech_TI-VCT_NL\(1\).PDF](http://www.fordpers.be/content/assets/news-2011/pdf/Ford_tech_TI-VCT_NL(1).PDF)
8. Physics Forum 2008, Mechanical Engineering, Viewed 17th April 2017, <https://www.physicsforums.com/threads/internal-combustion-engine-thermal-energy-loss.280572/>
9. Rapid Tables 2017, Horsepower to Kilowatt Conversion, Viewed 3rd April 2017, <http://www.rapidtables.com/convert/power/hp-to-kw.htm>
10. What Car 2011, What Does PS and BHP Mean, Viewed 17th April 2017, <http://www.whatcar.com/news/5-does-ps-bhp-mean/>

11.0.5.1 – Chapter Literature Review:

1. Bell – Investigated Turbocharger process.
2. Convert Units – Obtained equation for calculating the PS to HP conversion.
3. Cirimele & D'Elia – Obtained Heat Flux definition.
4. Energy Education – Obtained ICE thermal efficiency percentage.
5. The Engineering Tool Box – Obtained figure for Aluminium and Carbon Steel Thermal Conductivity and the definition.
6. Ford – Obtained Ford 1.6 EcoBoost official power output.
7. Ford – Obtained 1.6 Duratec official power output.
8. Physics Forum – Obtained thermal energy lost into oil and water figure.
9. Rapid Tables – Obtained HP to KW conversion equation.
10. What Car – Obtained correct term for the power output measurement PS.

Chapter 12.0 - Conclusion:

Production of a conclusion enabled justification of results produced for each individual chapter and a final overall conclusion summarised the Major Project as a whole.

12.0.1 – Combustion Calculations Conclusion:

Consequently, Mean Cylinder Pressure, Pressure vs Volume, Logarithms of Pressure vs Volume, Extra Pressure from Combustion and Mass Fraction Burn graphs for all three engines at specific RPM's selected will be concluded for the Combustion Calculations in Chapter 6.0.

Mean Cylinder Pressure data for the Ford Fox and Sigma were placed into a graph to enable a visual comparison, in Figure 20.0.

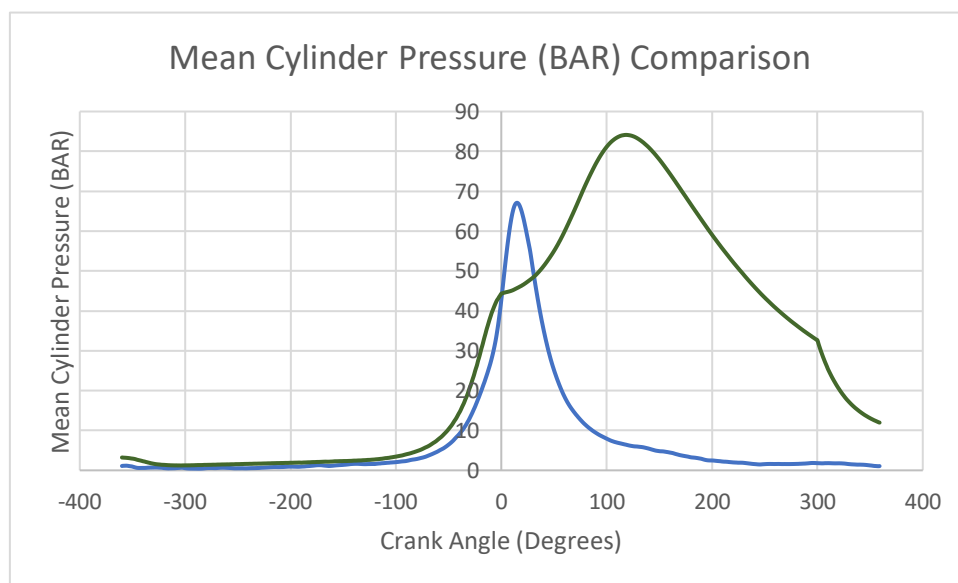


Figure 20.0 – Mean Cylinder Pressure (BAR) Comparison.

Data was obtained from engines at maximum RPM to produce a visual representation of the engines working at the peak of ability under maximum stress. The Sigma engine (Highlighted in blue) achieves a maximum cylinder pressure of 67.103BAR at 6500RPM. However due to the turbocharged nature of the engine, the Fox (Highlighted in green) could achieve 84.135BAR at 6000RPM. Therefore, producing the highest cylinder pressure of the two conventional piston engines.

Nevertheless, the naturally aspirated Mazda Rx8 engine could achieve 96.523BAR at 6646RPM, displayed in *Chapter 6.0.3*. This is due to the high revving nature of the rotary engine and peak power being achieved at the top of the revolution range.

Pressure vs Volume graphs were then produced to display the ideal Otto Cycle of each engine. Logarithms of pressure and volume were then produced for each engine to enable a clear visual representation of the occurrence of SOC, EOC and therefore combustion duration.

The lower the RPM, the easier a visual representation of SOC, EOC and combustion duration could be achieved. However, due to limited data the only revolution range available for the Mazda Rx8 was 6646RPM. Consequently, increasing the difficulty of visual representation of these values.

Next; Extra Pressure (BAR) from Combustion graphs were produced for each engine at selected RPM. The Sigma produced 2.959BAR extra pressure from combustion at 6500RPM. Again, due to the turbocharged nature, the Fox engine produced 4.916BAR extra pressure from combustion at 6000RPM, displayed in *Figure 20.0.1*. This is due to the high-pressure environment of a turbocharged engine.

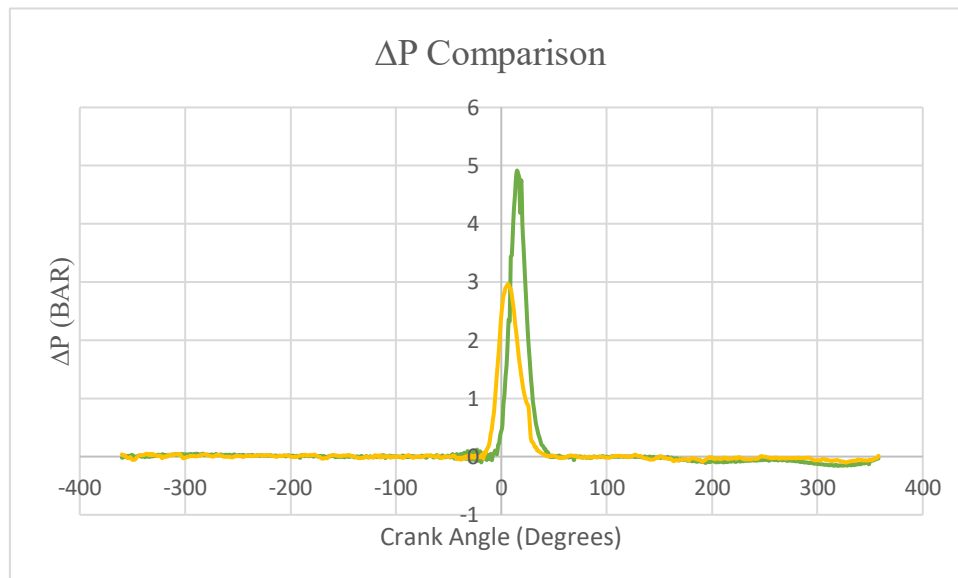


Figure 20.0.1 - ΔP Comparison.

However due to high pressures achieved via the rotary engine, ΔP greatly exceeds what is produced for the conventional piston engines. Consequently; through calculations a ΔP of 28.717BAR is achieved in Chapter 6.0.3.

Calculation of the sum of ΔP enables MFB production. Ford Sigma MFB begins at -16 degrees BTDC and completion occurs at 51 degrees ATDC. MFB begins to increase after 1 degrees BTDC and begins to plateau after 32 degrees of start, before 100% MFB is achieved at 51 degrees ATDC.

The Fox MFB begins later at -8 degrees BTDC and completion occur earlier at 45 degrees ATDC. Therefore, the engine has a faster burn rate, which vastly increases at 15 degrees after start and begins to plateau after 28 degrees of start, until 100% MFB.

Finally; the Renesis rotary engine MFB begins at -10 degrees BTDC and completion occurs at 80 degrees ATDC. Consequently, this engine takes 22 degrees of crank angle rotation longer for MFB completion than the Sigma and 36 degrees longer than the Fox.

12.0.2 – FEA Conclusion:

Models produced in Solid Works 2016 were placed under FEA. Several simulations were produced for each component. This section will conclude final simulations.

Firstly; conventional piston engine pistons were analysed. The Ford Sigma endured the higher Maximum Von Mises Stress of $3.779 \times 10^8 \text{ N/m}^2$, compared to $8.904 \times 10^7 \text{ N/m}^2$ of the Fox engine. This is likely to be due to Plain Carbon Steel material used for the Fox, compared to the lower UTS of Cast Carbon Steel of the Sigma. However, the Fox piston endured a higher Maximum Displacement of $6.687 \times 10^{-3} \text{ mm}$, compared to $6.036 \times 10^{-2} \text{ mm}$ of the Sigma. As well as a higher Maximum Strain of 3.411×10^{-4} , compared to the 1.517×10^{-3} of the Sigma.

Connecting Rods were analysed next. The Fox endured a higher Maximum Von Mises Stress of $2.610 \times 10^9 \text{ N/m}^2$, compared to $8.085 \times 10^8 \text{ N/m}^2$ of the Sigma. The Fox obtained a displacement of $7.431 \times 10^{-1} \text{ mm}$, compared to the Sigma obtaining a positive displacement of 1.607mm via expansion. Finally; the Fox con-rod endured a higher Maximum Strain of 9.909×10^{-3} , compared to 2.228×10^{-3} of the Sigma.

The Single Span of the crankshaft was placed under FEA and obtained a Maximum Von Mises Stress of $7.855 \times 10^7 \text{ N/m}^2$, which was increased to $9.564 \times 10^7 \text{ N/m}^2$ due to a mesh refinement produced in Chapter 10.0.1.1. The refinement also produced a slight increase in Maximum Stress from 2.792×10^{-4} to 2.985×10^{-4} . However, a reduction of displacement from $8.77 \times 10^{-3} \text{ mm}$ to $8.587 \times 10^{-3} \text{ mm}$ occurred.

Once conventional piston FEA was completed, FEA of the Wankel Rotary engine rotor commenced. Maximum Von Mises Stress of $8.610 \times 10^7 \text{ N/m}^2$ was firstly achieved and was reduced to $4.282 \times 10^7 \text{ N/m}^2$ via a refined mesh. Also, a slight increase of displacement from $2.299 \times 10^{-2} \text{ mm}$ from $2.276 \times 10^{-2} \text{ mm}$ occurred. Finally, Maximum Strain was reduced from 5.309×10^{-4} to 4.438×10^{-4} via mesh refinement.

12.0.3 – Thermal Calculations Conclusion:

Thermal equations of each engine are the final section required for conclusion.

The Renesis produced the largest Fuel Power of 630.99HP, followed closely by the Fox with 598.127HP and then the naturally aspirated Sigma with 383.33HP. However, the rotary also had the largest waste amount of power via oil and water of 210.33HP, again closely followed by the Fox with 199.376HP and the Sigma with 127.777HP.

Waste into oil and water was divided via each piston or rotor and converted to KW. Consequently, the Renesis had the largest waste at 11.767Kw, the Fox obtained 8.367Kw and the Sigma 5.362Kw.

Thermal conductivity (K) was dependant on material used for pistons. Consequently the Sigma and Fox obtained the same 43W/m.K, due to both being produced from Carbon Steel, despite being manufactured via a different process. However the aluminium rotor material obtained a 205W/m.K value.

Heat Flux was obtained via the use of area and Kw into rotor or piston. Subsequently producing the largest Heat Flux of $1519.723\text{Kw}/\text{m}^2$ for the Fox, $1179.236\text{Kw}/\text{m}^2$ for the Sigma and lastly $755.118\text{Kw}/\text{m}^2$ for the Renesis, due to the division of the larger surface area of the rotor face.

Finally; THot was acquired for the Fox at 2047.613°C, THot of 1595.436°C for the Sigma and 168.951°C for the Renesis. The large difference of THot for the rotary engine, compared to the conventional piston engines is due to the smaller heat flux multiplied via the smaller average distance for heat to travel (d) of the rotor to reach the eccentric shaft and the division of the much larger thermal conductivity of the aluminium rotor material.

12.0.4 – Final Conclusion:

The final conclusion of all chapters' states that the Renesis required the longest duration for MFB completion, as expected due to the long combustion chamber shape. Consequently, it was the slowest burning engine. Furthermore, the need for dual spark ignition is demonstrated via results produced from this Major Project.

Production of FEA simulations displayed the rotor suffered the minimum Von Mises Stress and minimum Displacement of all three engines. However, the rotor endured the largest Maximum Strain.

Finally, through production of thermal equations, the Wankel Rotary Engine produced the largest fuel power of all engines. However, the rotary engine also suffered the largest wasted power through oil and water. Calculations of heat flux placed the rotary engine last of all three engines, due to the division of the largest surface area of the rotor face, compared to the much smaller piston crown areas. Consequently, producing the minimum transfer of heat energy through the rotor. Finally, the rotary engine produced the lowest temperature due to division of the largest thermal conductivity, due to aluminium rotor material.

In Summary, the Wankel Rotary engines produces some promising results. However further research and development of the rotary engine ignition system is required to further improve engine efficiency.

Chapter 13.0 - Evaluation:

Several areas of improvement have been selected to further improve accuracy and yield of data.

13.0.1 – Assumptions

1. Ford Sigma Gudgeon pin measurement approximation, due to interference fit.
2. 50% wasted power into each rotor for thermal calculations.
3. TCold = 100°C for thermal calculations.

13.0.2 – Improvements:

Firstly; a reduction in assumption listed above, via further investigation would improve precision of findings. Secondly production of Fatigue Analysis via FEA for one million cycles to increase yield of data. Thirdly the production of motion simulation via Solid Works to improve visualisation of component processes.

Also, point load application on main journal of single span crankshaft to reduce stress on connection to counter-webs, displayed in *Figure 21.0*.

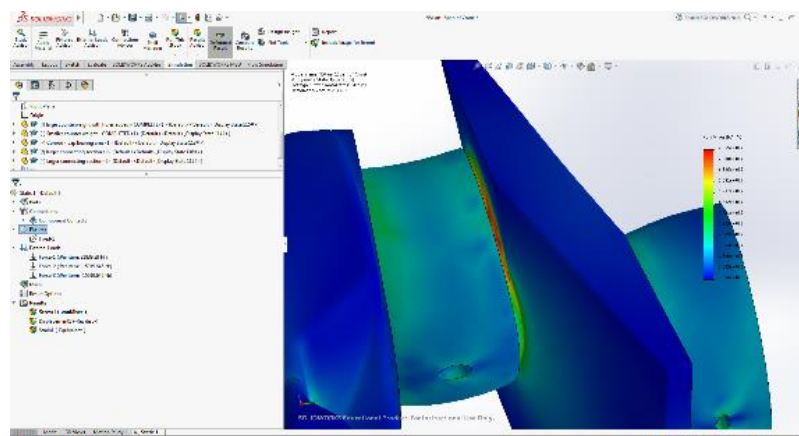


Figure 21.0 – Stress at counter-web connections.

The use of a radius gauge and digital angle finger would have further improved measurement accuracy for Solid Works models.

Finally; the use of suitable computers, would have produced more representative run-times for FEA models, as computers freezing or crashing due to limited power largely increased run-time without the software taking note. Consequently, increasing time frame for completion of FEA chapter, causing other sections of the major project to suffer due to limited time frame available.

Chapter 14.0 – Appendix:

Three readings were obtained for each measurement, to obtain the most accurate measurements possible, while reducing the likelihood of outliers.

14.0.1 – Distance for Heat to Travel Measurements:

Ford Fox “d” Measurements (Crown + Skirt (mm)):

$$24.23 + 30.72 = 54.95$$

$$24.57 + 30.82 = 55.39$$

$$24.24 + 30.74 = 54.98$$

$$\text{Average (A)} = \frac{165.32}{3} = 55.107$$

Ford Sigma “d” Measurements (Crown + Skirt (mm)):

$$15.35 + 39.09 = 54.44$$

$$15.39 + 39.08 = 54.47$$

$$15.49 + 39.2 = 54.69$$

$$A = \frac{163.6}{3} = 54.53$$

Wankel Rotary “d” Measurements (Crown + Skirt (mm)):

$$dMax = 60.43 + 60.21 + 60.28 = 180.92 = \frac{180.92}{3} = 60.307$$

$$dMin = 31.64 + 31.82 + 30.82 = 94.28 = \frac{94.28}{3} = 31.427$$

$$A = 60.307 + 31.427 = \frac{91.734}{2} = 45.867$$

Therefore:

$$d = 45.867 = 0.045867m$$

14.0.2 – Piston Head and Rotor Face Areas:

Areas were measured in mm^2 and obtained in the Evaluation section of Solid Works 2016, using the measurement tool.

Ford Fox Piston (mm^2):

Piston head = 2111.28

Cavity = 3394.33

Ford Sigma Piston (mm^2):

Piston Head = 2434.67

Raised section (Inc Side) = 2112.34

Rotary Face (mm^2):

Rotor Face = 9863.77

Combustion Cavity (Inc Sides) = 5719.22

14.0.3 – Ford Fox Engine Measurements:

Three measurements of each component were extracted to ensure the most accurate measurement was used in the Solid Works model, by calculating the average of the 3 results.

All measurements are calculated in millimetre (mm).

Piston Diameter:

- 78.68
- 78.21
- 78.28
- Average = 78.39

Piston Length (Peaked point of piston head till bottom of side):

- 50.43
- 50.55
- 50.31
- Average = 50.43

Piston Length (Lower point of piston head till bottom of piston skirt):

- 48.01
- 47.74
- 47.86
- Average = 47.87

Oval piston centre shape underneath width:

- 63.3
- 63.12
- 62.99
- Average = 63.137

Indent on each side of piston underneath width:

- $78.38 - 63.137 = 15.253$

$15.253/2 = 7.627$

Length of inside of concave section:

- 66.26
- 66.61
- 66.3
- Average = 66.39

Height of inside of concave section:

- 26.4
- 26.35
- 26
- Average = 26.25

Inner oval shape smaller width:

- 22.58
- 22.94
- 23.01
- Average = 22.843

Inner oval shape larger width:

- 30.17
- 30.66
- 30.55
- Average = 30.46

Piston Skirt Thickness:

- 2.45
- 2.47
- 2.72
- Average = 2.547

Oval Height:

- 36.61
- 36.38
- 36.3
- Average = 36.43

Gudgeon Pin Hole Diameter:

- 19.63
- 19.77
- 20.28
- Average = 19.893

Outer piston skirt to inner Gudgeon pin cylinder:

- 29.68
- 29.94
- 29.58
- Average = 29.733

Top peak of piston to inner Gudgeon pin cylinder:

- 24.61
- 25.23
- 25.18
- Average = 25.007

Small angled line at bottom of piston/side of piston skirt:

- 5.37
- 5.84
- 5.84
- Average = 5.683

Depth of circle on top of Piston Head:

- 11.11
- 11
- 11.67
- Average = 11.26

Diameter of outer circle on top of Piston Head:

- 53.31
- 53.32
- 53.24
- Average = 53.29

Height of lower piston Peak:

- 5.66
- 5.44
- 5.55
- Average = 5.55

Therefore:

$$\text{Higher Peak} - \text{Lower Peak} - 25.007 - 5.55 = 19.457$$

Diameter of inner circle on top of Piston Head:

- 33.94
- 34.08
- 34.11
- Average = 34.043

Height of inner circle on top of piston head:

- 2.88
- 2.69
- 2.67
- Average = 2.747
- Therefore $11.26 - 2.747 = 8.513$ to be cut extruded

Lower Peak at top of piston:

Smaller Width:

- 14.26
- 13.67
- 14.31
- Average: 14.08

Larger Width:

- 45.98
- 45.72
- 46.45
- Average = 46.05

Curve Length:

- 15.42
- 15.01
- 15.41
- Average = 15.28

Higher Peak at Top of Piston:

Height (Taken from previous measurements):

- Average = 11.26

Length:

- 45.07
- 44.74
- 45.09
- Average = 134.9
- =67.45

Width:

- 11.24
- 11.31
- 11.63
- Average = 11.393

Distance from piston head to 1st piston ring:

- 1.33
- 1.42
- 1.67
- Average = 1.473

Gudgeon Pin:

Length:

- 45.83
- 45.85
- 45.8
- Average = 45.827

Outer diameter:

- 20.98
- 20.99
- 20.98
- Average = 20.983

Width:

- 4.64
- 4.71
- 4.67
- Average = 4.673

Inner diameter:

- 11.78
- 11.5
- 11.49
- Average = 12.59

Con-Rod:

Overall Max Length = 175

Overall Max Width:

- 69.99
- 69.96
- 69.97
- Average = 69.973

Overall Max Thickness:

- 20.98
- 20.9
- 20.84
- Average = 20.907

Thickness of bottom of big end:

- 9.37
- 9.02
- 8.71
- Average = 9.033

Length from small end to top of big end:

- 123.5

Diameter of Big End:

- 46.75
- 46.94
- 46.95
- Average = 46.88

Thickness of top of small end:

- 3.53
- 3.54
- 3.62
- Average = 3.563

Small End Diameter:

- 20.93
- 20.58
- 20.88
- Average = 20.797

Max thickness of small end:

- 19.01
- 19.14
- 19.11
- Average = 19.087

Rod shank Thickness:

- 20.67
- 20.44
- 20.44
- Average = 20.517

Rod Length till big end:

- 90.18
- 90.17
- 90.39
- Average = 90.247

Small end minimum Thickness:

- 12.2
- 12.21
- 12.14
- Average = 12.183

Crank Pin Cover Length (Inc Pin):

- 48.31
- 48.51
- 48.8
- Average = 48.54

Thickness of big end at curve from shank to big end

- 10.29
- 10.45
- 10.29
- Average = 10.343

Shank Concave Section Length:

- 89.9
- 90.44
- 90.24
- Average = 90.193

Shank Concave section minimum Width:

- 7.71
- 8.45
- 8
- Average = 8.053

Concave Section wall thickness:

- 3.83
- 3.62
- 3.49
- Average = 3.647

Shank Concave Section Height:

- 5.03
- 5.4
- 4.92
- Average = 5.117

Small End thickness from inner diameter to concave section:

- 5.57
- 5.98
- 5.8
- Average = 5.783

Big End thickness from inner diameter to concave section:

- 4.55
- 4.43
- 4.43
- Average = 4.47

Con-rod bolt holes diameter:

- 8.75
- 8.78
- 8.74
- Average = 8.757

Con-rod bolt holes wall thickness on outer wall on side:

- 1.67
- 1.76
- 1.86
- Average = 1.763

Con-rod bolt holes wall thickness on thicker outer wall (rear/front):

- 3.05
- 3.32
- 3.2
- Average = 3.19

Con-rod bolt holes wall thickness on thicker outer wall (rear/front):

- 2.82
- 2.75
- 2.75
- Average = 2.773

Con-rod bolt depth (till reach con-rod bolt):

- 5.79
- 5.4
- 5.6
- Average = 5.597

Therefore 102.25 (Length from plan to top of where hole needs to be extruded) $+5.597 = 107.848$

However, cut extrude through all to make easier to build bolts.

Con-rod big end minimum thickness on extruded section:

- 4.42
- 4.17
- 4.43
- Average = 4.34

Con-rod big end thickness of centre section underneath:

- 10.41
- 9.64
- 9.97
- Average = 10.007

Con-rod big end minimum height on extruded section:

- 20.907 (Taken from maximum thickness of con-rod) $- 10.007 = 10.9$

Divide by 2 to get the thickness of each side of the centre section

- Therefore $10.9/2 = 5.45$

Con-rod big end section underneath width:

- $10.007/2 = 5.004$

Con-rod big end Section underneath:

- 9.033 (Thickness of bottom of big end) $- 5.45$ (Con-rod big end minimum height on extruded section) $= 3.583$

Distance between con-rod bolt and centre section under big end:

- 2.64
- 2.63
- 2.65
- Average = 2.64

Minimum height of centre section under big end after slope:

- 3.82
- 4.26
- 4.54
- Average = 4.207

Small extruded section edge length attached to centre section underneath big end:

- 6.89
- 6.33
- 6.63
- Average = 6.617

14.0.4 – Ford Sigma Engine Measurements:

Piston:

Diameter:

- 75.33
- 75.49
- 75.16
- Average (A) = 75.327

Length (Inc Skirt):

- 54.35
- 54.24
- 54.41
- A = 54.33

Skirt Width:

- 53.97
- 54.42
- 54.54
- A = 54.31

Piston Crown Minimum Length (Ex Skirt):

- 19.09
- 19.25
- 19.11
- A = 19.15

Diameter of circle on top of piston head:

- 51.06
- 50.39
- 50.76
- A = 50.737

Height of circle:

- 1.22
- 1.2
- 1.17
- A = 1.197

Length of Piston skirt 45degree angle:

- 12.77
- 12.71
- 11.99
- A = 12.49

Gudgeon pin holder gap between each side:

- 20.46
- 20.53
- 20.55
- A = 20.513

Con-Rod:

Big end diameter:

- 42.67
- 42.56
- 42.64
- A = 42.623

Thickness from inner big end to side:

- 11.85
- 11.9
- 11.82
- A = 11.857

Length of side sections:

- 38.18
- 38.44
- 38.15
- A = 38.257

Conrod width:

- 17.55
- 17.49
- 17.53
- A = 17.523

Con Rod body length:

- 94.28
- 93.83
- 94.45
- A = 94.187

Small End:

- Assumed to be same size as Gudgeon pin (Interference fit Gudgeon pin, so unable to measure)

Maximum Width:

- 20.85
- 21
- 20.92
- $A = 20.923$

Indent in rod → Width:

- 8.83
- 9.06
- 8.94
- $A = 8.943$

Length of indent (Ex curved edges):

- $A = 68.41$

Indent Depth:

- 4.25
- 4.34
- 4.46
- $A = 4.35$

Con Rod bolt holes diameter:

- 8.14
- 8.02
- 7.9
- $A = 8.02$

Gudgeon Pin:

Diameter:

- Taken from hole diameter earlier

Length:

- 48.18
- 48
- 48.02
- $A = 48.067$

Inner Diameter:

- 9.91
- 9.26
- 9.2
- $A = 9.457$

14.0.5 – Ford Fox Crankshaft Measurements:

Large End (mm)

Diameter:

- 79.98
- 79.48
- 80.18
- A = 79.88

Width:

- 25.69
- 25.09
- 25.76
- A = 25.513

Fillet:

- 3.67
- 3.63
- 3.35
- A = 3.55

Diameter of 6 holes on top of:

- 9
- 8.91
- 9.06
- A = 8.99

6 holes depth:

- 21.59
- 21.46
- 21.2
- A = 21.417

6 Holes distance from edge:

- 3.79
- 3.9
- 3.8
- A = 3.83

6 holes fillet:

- 1.42
- 1.1
- 1.7
- A = 1.407

Diameter of extruding stud on top:

- 10.05
- 10.05
- 10.05
- A = 10.05

Height of extruding stud on top:

- 5.98
- 5.44
- 5.77
- A = 5.73

Fillet of extruding stud on top:

- 1.34
- 1.28
- 1.09
- A = 1.237

Centre circle outer diameter:

- 34.05
- 34.4
- 34.2
- A = 34.217

Centre circle inner diameter:

- 20.72
- 20.77
- 20.67
- A = 20.72

Centre Circle Height:

- 7.34
- 7.44
- 7.14
- A = 7.307

Centre Circle Depth:

- 34.7
- 35.05
- 34.57
- A = 34.773

Therefore → $34.773 - 7.307 = 27.466$

Centre Circle Fillet:

- 4.22
- 4.22
- 4.35
- $A = 4.263/2 = 2.132$

Section connecting to counter weight:

Diameter:

- 48.97
- 48.02
- 47.99
- $A = 48.327$

Length:

- 24.51
- 24.63
- 25.57
- $A = 24.803$

Small hole on section connecting to counter weight diameter

- 5.09
- 5.06
- 5.08
- $A = 5.077$

Crank shaft area where con-rod/cap bearing sits:

Diameter:

- 43.98
- 43.97
- 44.02
- $A = 43.99$

Width:

- 20.91
- 20.95
- 20.99
- $A = 20.95$

Hole Diameter:

- 4.96
- 4.95
- 4.8
- $A = 4.903$

Con-rod/Cap bearing area connector:

Max Length

- 98.99
- 97.42
- 98
- $A = 98.137$

Max Depth:

- 20.14
- 19.66
- 20.29
- $A = 19.03$

Max Width:

- 74.35
- 74.44
- 74.43
- $A = 74.407$

Width of top:

- 5.76
- 5.65
- 5.65
- $A = 5.687$

Length from top diagonally to midway:

- 56.95
- 56.35
- 57.02
- $A = 56.773$
- Angle used = roughly 15 degrees

Distance from edge to where cylinder required:

- 5.63
- 5.41
- 5.29
- $A = 5.443$

Circle diameter (Not attaching cylinder):

- 64.06
- 64.02
- 63.94
- $A = 64.007$

Large Counter-Weight:

Max Length:

- 150mm

Max Depth:

- 22.04
- 22.45
- 21.74
- A = 22.076

Max width:

- 115mm

Length of flat side face:

- 31.96
- 31.77
- 31.3
- A = 31.677

Angle from flat side to centre circle:

- 24.17
- 24.02
- 24.3
- A = 24.163 @ 150 degrees

Straight line from centre circle to beginning of curve:

- 37.75
- 36.98
- 37.38
- A = 37.37

Length of small straight edge before angle:
(Arrow) @ 156.2 degrees

- 14.2
- 14.4
- 13.9
- A = 14.16

Length of line from straight line before angle to thinnest wall:

- 20.06
- 20.08
- 20.27
- A = 20.137

Thickness of thinnest wall:

- 5.08
- 5.26
- 5.02
- A = 5.12

Length of line upwards from thinnest wall:

- 15.7
- 15.38
- 15.76
- A = 15.613

Circle Diameter:

- 63.26
- 64
- 63.55
- A = 63.603

Larger circle diameter:

- 74.58
- 73.68
- 74.17
- A = 74.143

Depth of thin end:

- 6.17
- 6.34
- 6.23
- A = 6.247

Distance from Large circle to edge:

- 3.52
- 3.25
- 3.75
- A = 3.507

Thickness of thin bridge connecting bottom and top of counter weight:

- 6.61
- 6.26
- 6.67
- $A = 6.513$

Small hole diameter:

- 8.53
- 8.72
- 8.75
- $A = 8.66$

Large hole diameter:

- 10.72
- 10.71
- 10.81
- $A = 10.747$

Distance from first small hole to big hole →
6.29mm

Distance from big hole to 2nd small hole →
3.29mm

Distance from 2nd small hole to big hole →
5.62mm

14.0.6 – Wankel Rotary Engine Measurements:

Wankel Rotary – Rotor Measurements (mm):

Rotor:

Length from rotor tip to opposite base:

- $D1 = 59.43 + 59.61 + 60.14 = 59.73$
- $D2 = 92.92 + 92.89 + 92.76 = 92.86$
- $D3 = 28.12 + 28.75 + 28.13 = 23.33$
- Total Diameter = 175.92mm
- Total Radius = 87.96mm

Rotor Width:

- $79.78 + 79.89 + 79.81$
- $A = 79.83$

Creating Apex Seal Slot

Apex Seal Section Width:

- $2.06 + 2.02 + 2.05$
- $A = 2.04$

Depth of Apex Seal Section:

- $2.34 + 1.94 + 2.04$
- $A = 2.11$

Creating Rotor Gear:

Rotor Gear Tooth top flat width

- $1.83 + 1.77 + 1.81$
- $A = 1.8$

Depth:

- $3.03 + 3.02 + 3.11$
- $A = 3.05$

Tooth Max/Base width:

- $5.1 + 5.16 + 5.09$
- $A = 5.12$

Rotor Gear Extruded Area

Width:

- 3.52

Depth:

- 3.13

Face Seal indent:

Width of indent:

- 1.26
- 1.24
- 1.26
- $A = 1.25$

Distance from edge of indent to edge of rotor:

- 2.26
- 2.37
- 2.34
- $A = 2.32$

Distance from edge of indent to edge of rotor (Minimum distance @ centre):

- 1.7
- 1.67
- 1.67
- $A = 1.68$

Indent Depth:

- 0.5

Circles on face of rotor:

Width of circle with oil holes:

- 6.2
- 6.51
- 6.53
- $A = 6.41$

Oil semi-circle radius:

- 4.37
- 4.39
- 4.24
- $A = 4.33$

Oil semi-circle distance apart:

- 26.61
- 26.64
- 26.54
- $A = 26.6$

Width:

- 6.22
- 6.26
- 6.23
- $A = 6.24$

Depth:

- 8.52
- 9.04
- 8.96
- $A = 8.84$

2nd circle (1st Raised):

Width:

- 3.14
- 3.12
- 3.13
- $A = 3.13$

Height:

- 1.2
- 1.03
- 1.05
- $A = 1.09$

Gap between next raised circle distance:

- 2.02
- 2.01
- 2.05
- $A = 2.03$

3rd Circle (2nd Raised):

Width:

- 3.25
- 3.24
- 3.26
- $A = 3.25$

4th Circle (1st extrude cut circle)

Depth:

- 3.18
- 3.13
- 3.23
- $A = 3.18$

Final Raised Circle:

Height:

- 0.81
- 0.82
- 0.82
- $A = 0.82$

Creating Combustion Cavity:

Distance from Apex Seal slot to
combustion cavity:

- $44.27 + 44.06 + 43.91$
- $A = 44.08$

Distance from Rotor side to combustion
cavity:

- $13.96 + 14.13 + 13.93$
- $A = 14.01$

Combustion cavity width:

- $52.42 + 53.19 + 52.3$
- $A = 52.64$

CC Depth:

- $6.73 + 6.63 + 6.92$
- $A = 6.76$

14.0.6.1 – Eccentric Shaft Measurements:

Long Thin Cylinder:

Cylinder Length (mm)

- 92.34
- 92.27
- 92.23
- Average = 92.28

Cylinder Diameter:

- 30.28
- 30.11
- 30.22
- A = 30.203

Fillet:

- 0.5mm

Inner Diameter:

- 16.36
- 16.15
- 16.05
- A = 16.187

Inner Diameter chamfer:

- 4.89
- 4.44
- 4.55
- A = 4.627

Extruded Long rectangle length:

- 84.17
- 84.29
- 84.35
- A = 84.27

Extruded Long rectangle Width:

- 4.82
- 4.83
- 4.97
- A = 4.873

Extruded Long rectangle Depth:

- 2.94
- 2.94
- 2.95
- A = 2.943

Distance larger hole is away from end of cylinder:

- 35.57
- 35.67
- 35.85
- A = 35.697

Large hole diameter:

- 7.73
- 7.72
- 7.74
- A = 7.73

Fillet:

- 0.25

Distance smaller hole is away from end of cylinder:

- 10.27
- 10.52
- 10.16
- A = 10.317

Smaller hole diameter:

- 2.76
- 2.79
- 2.66
- A = 2.737

Cylinder Connected to Long Thin Cylinder:

Diameter:

- 43.01
- 42.94
- 42.98
- A = 42.977

Length:

- 57.63
- 58.05
- 57.66
- A = 57.78

Distance from end of cylinder (connecting to long thin cylinder) to hole:

- 19.24
- 19.12
- 19.29
- A = 19.217

Diameter:

- 6.76
- 6.89
- 6.91
- A = 6.853

Chamfer:

- 1

Large Cylinder which Rotor Attaches to:

Length:

- 52.15
- 52.27
- 52.22
- A = 52.213

Diameter:

- 73.36
- 74.08
- 73.69
- A = 73.71

Large hole diameters @ 30degrees:

- 17.86
- 17.7
- 17.72
- A = 17.76

Small Hole diameters:

- 11.3
- 11.34
- 11.86
- A = 11.5

Depth of small and large hole:

- 46.34
- 46.56
- 45.71
- A = 46.203

Chamfer:

- 0.5

Hole of side of cylinder Diameter:

- 4.2
- 4.81
- 4.85
- A = 4.62

Chamfer:

- 1.48
- 1.46
- 1.36
- A = 1.433

Short and Fat Cylinder:

Length of 1st section of cylinder:

- 58.36
- 58.82
- 58.93
- A = 58.703

Diameter of 1st cylinder:

- 43.11
- 43.08
- 42.99
- A = 43.06

Length of 2nd section of cylinder:

- 30.66
- 30.75
- 30.78
- A = 30.73

Diameter of 2nd cylinder:

- 40.18
- 40.1
- 39.71
- A = 39.997

Drop from 2nd cylinder to 3rd cylinder:

- 2.81
- 2.7
- 2.14
- A = 2.55

Length of 3rd cylinder/ threaded section:

- 17.17
- 17.54
- 17.3
- A = 17.337

Diameter of 3rd cylinder/ threaded section

- 36.07
- 35.85
- 35.85
- A = 35.923

Length of extruded rectangle:

- 29.02
- 29.4
- 29.54
- A = 29.32

Width of extruded rectangle:

- 7
- 6.87
- 6.97
- 6.947

Depth of extruded rectangle:

- 5.65
- 5.72
- 5.79
- A = 5.72

Distance from extruded rectangle to hole:

- 21.94
- 21.88
- 22.31
- A = 22.043

Hole Diameter:

- 4.92
- 4.91
- 4.84
- A = 4.89

Chamfer:

- 1.74
- 1.9
- 1.39
- A = 1.677

Centre Twist Section Connecting Eccentric Shaft:

Length:

- 74.1
- 76.79
- 76.23
- A = 75.71

Diameter:

- 71.99
- 71.96
- 71.95
- A = 71.97

Distance from main rotor attachment
cylinder to chamfer:

- 5.79
- 5.04
- 5.5
- $A = 5.44$

Drop from chamfer to next edge 95 degrees:

- 30.15
- 30.14
- 30.16
- $A = 30.15$

Next flat surface length:

- 13.03
- 13.13
- 12.8
- $A = 12.99$

Next gradient @ 140 degrees:

- 45
- 39.99
- 45.01
- $A = 45$

Flattens out length:

- 19.31
- 19.83
- 19.47
- $A = 19.54$

Drop From chamfer:

- 20mm

Next flat surface length:

- 13.03
- 13.13
- 12.8
- $A = 12.99$

14.0.7 – Ford Sigma Data:

As previously mentioned in Chapter 6.0.1, data was produced for the Ford Sigma every 500RPM, consequently producing a yield of data, as displayed in Figure 22.0 for every 500RPM.

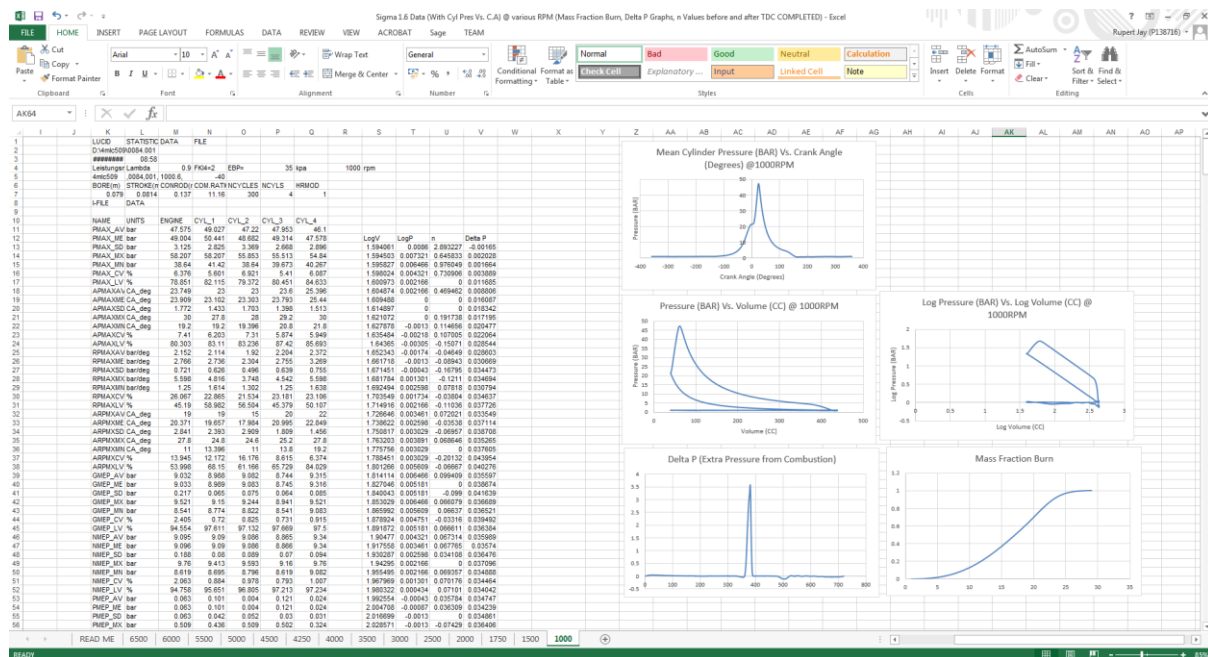


Figure 22.0 – Ford Sigma Engine Data at 1000RPM.

14.0.7.1 – Ford Fox Data:

Data was also produced every 500RPM for the Fox engine. Consequently, producing a yield of data, as displayed in Figure 22.0.1 for every 500RPM.

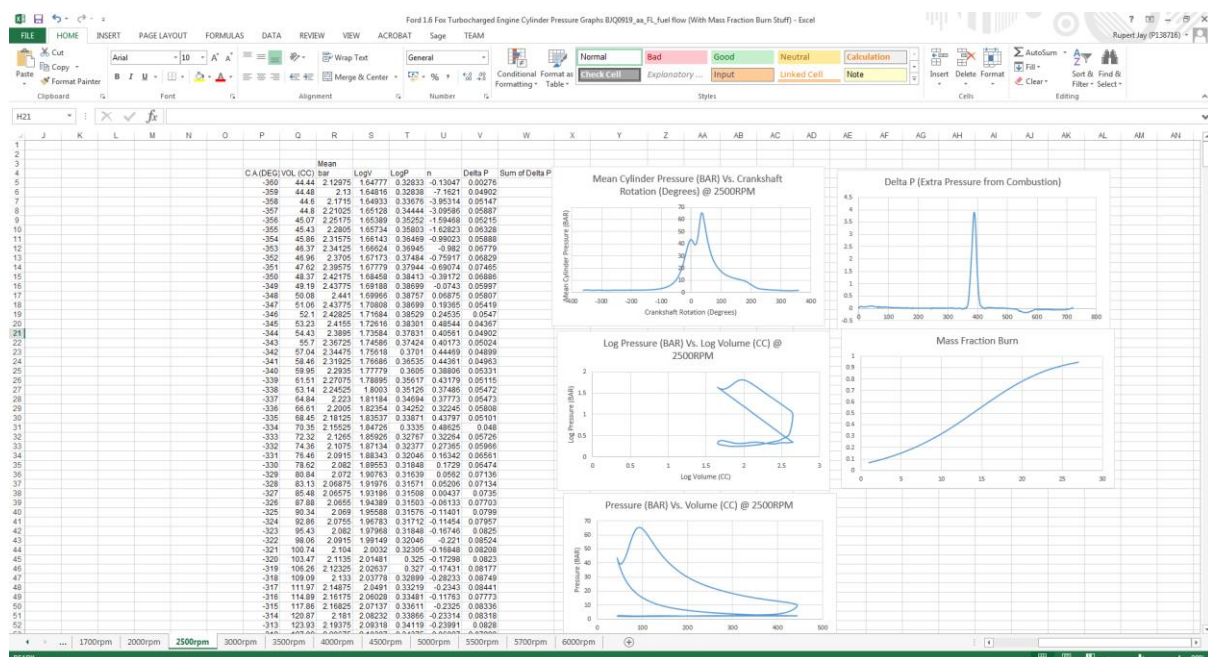


Figure 22.0.1 – Ford Fox Engine Data at 2500RPM.

14.0.7.2 – Mazda Rx8 Data:

Finally, the same process was applied to the Wankel Rotary engine. However due to the limited data available a single set of results was produced at 6646RPM, displayed in Figure 22.0.2.

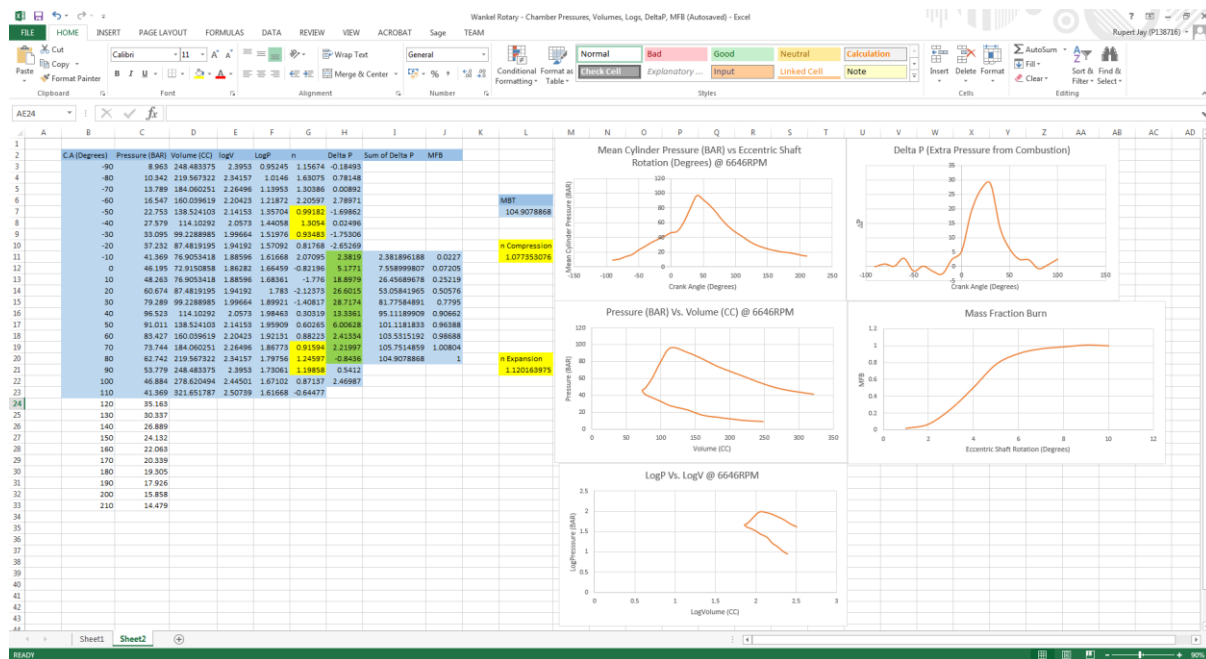


Figure 22.0.2 – Wankel Rotary Engine data at 6646RPM.

14.0.8 – Wankel Rotary Engine Application:

The Wankel Rotary engine was not only produced for the use in production vehicles. The rotary engine was applied to several other sectors, including:

1. Aero industry, within light aircraft, such as the Diamond DA20 (Aircraft Market Place, 2010)
2. Racing, such as the Le Mans 24 Hour winning Mazda 787b.
3. Bikes, such as the Suzuki RE5, multiple Norton motorcycles and the Yamaha RZ201. (Visor Down, 2015)
4. Rotary engine boats, such as the 1996 Rotary boat, powered by a Mazda Rx7 13b REW Twin Turbo engine.
5. Military Application for drone use. (Popular Mechanics, 2015)

14.0.8.1 – Wankel Rotary Engine Inefficiencies:

Several Inefficiencies have been discovered with the Wankel Rotary engine which reduce the likelihood of other manufacturers developing the engine further for production vehicles, which include:

1. Compression loss through spark plug holes
2. Epitrochoid Rotor Housing Chamber Shape
3. Damage to Rotor Housing wall
4. Low Thermal Efficiency (C:R, Chamber Shape, flame travel distance, high emissions)
5. Sealing issues
6. Temperature differences over rotor housing (Expansion)

(Yamamoto, 1981)

14.0.9 – Single Span Eccentric Shaft FEA:

Eccentric shaft FEA was produced with a 4mm mesh. Consequently, obtaining the following results:

- Maximum Von Mises Stress = $5.888 \times 10^{-7} \text{ N/m}^2$
- Maximum Displacement = $3.625 \times 10^{-3} \text{ mm}$
- Maximum Strain = 2.069×10^{-4}

Next a 1mm refined mesh was applied to the Eccentric shaft model main journal, as displayed in Figure 23.0.

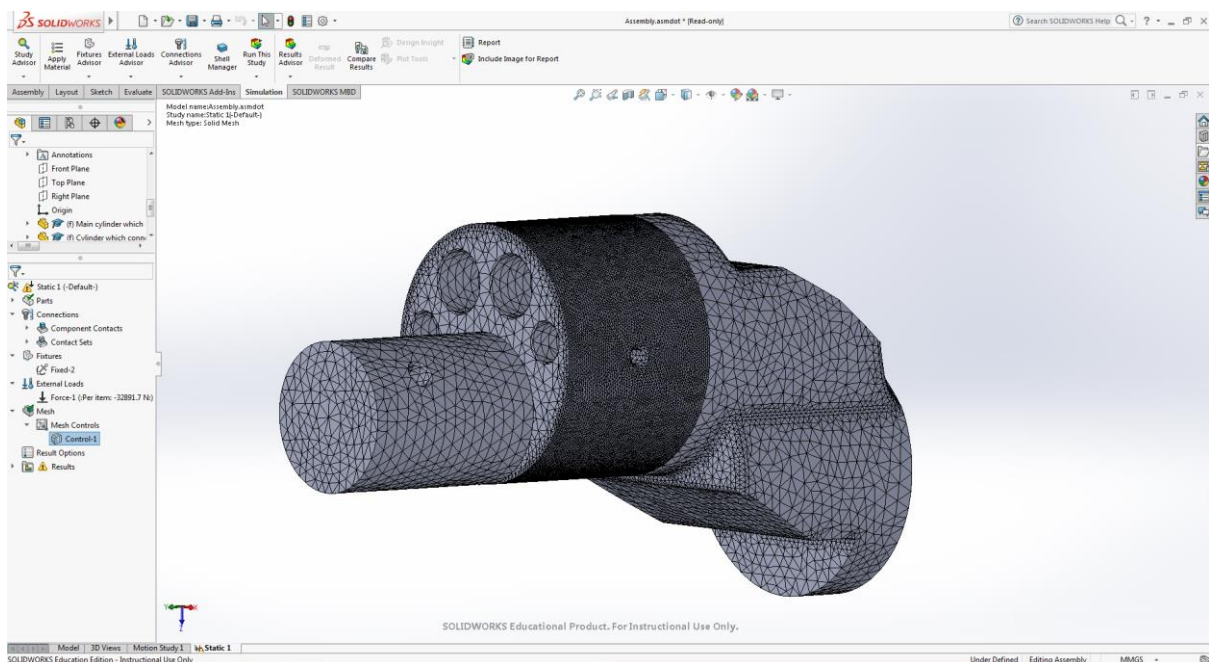


Figure 23.0 – 1mm Mesh refinement.

Therefore, producing a new Maximum Von Mises Stress of $7.802 \times 10^7 \text{ N/m}^2$, displayed in Figure 23.0.1.

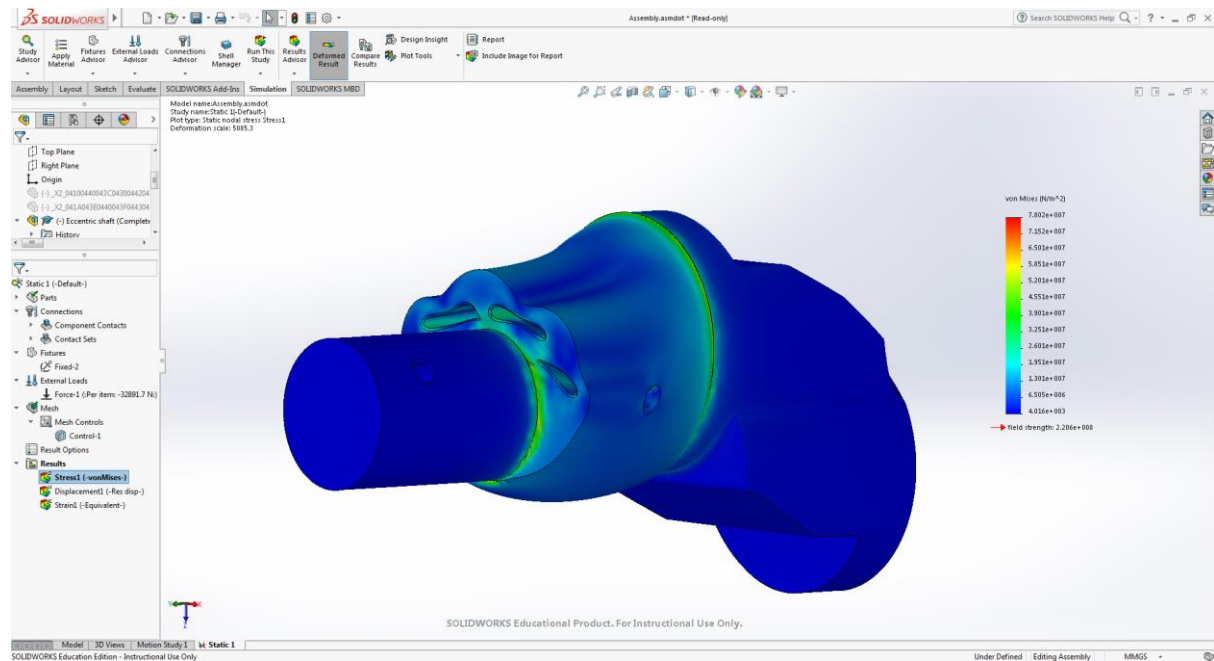


Figure 23.0.1 – Maximum Von Mises Stress.

Maximum Displacement of $3.699 \times 10^{-3} \text{ mm}$ was obtained in Figure 23.0.2.

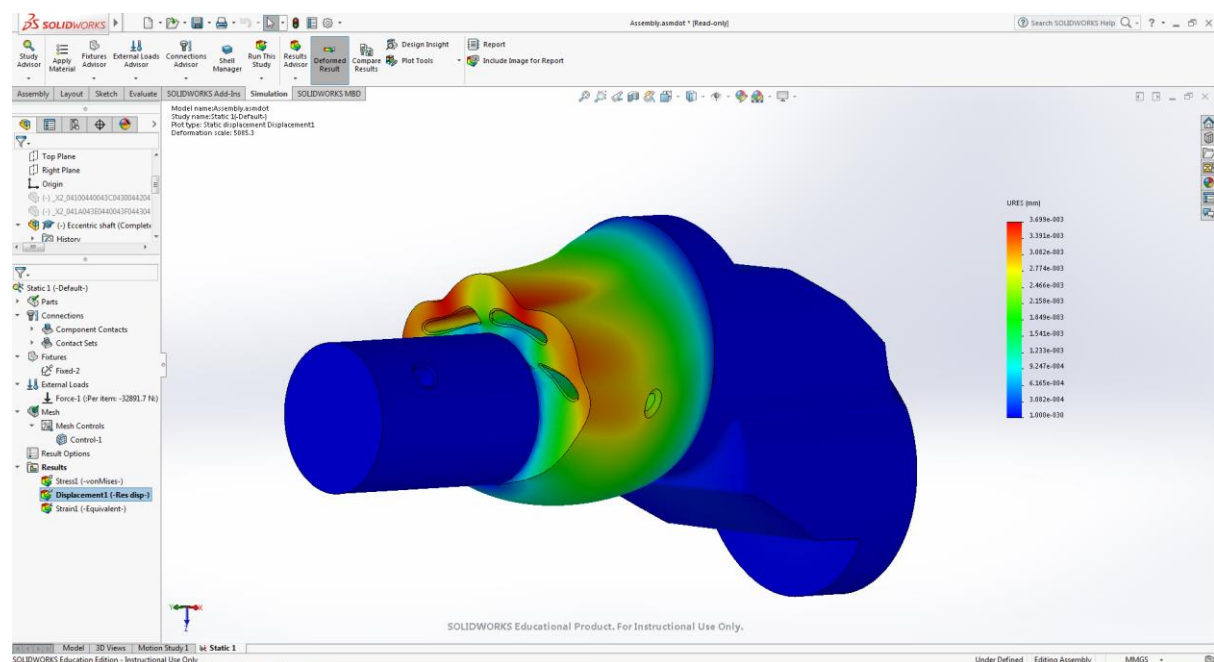


Figure 23.0.2 – Maximum Displacement.

Finally; Maximum Strain of 2.316×10^{-4} was obtained in Figure 23.0.3.

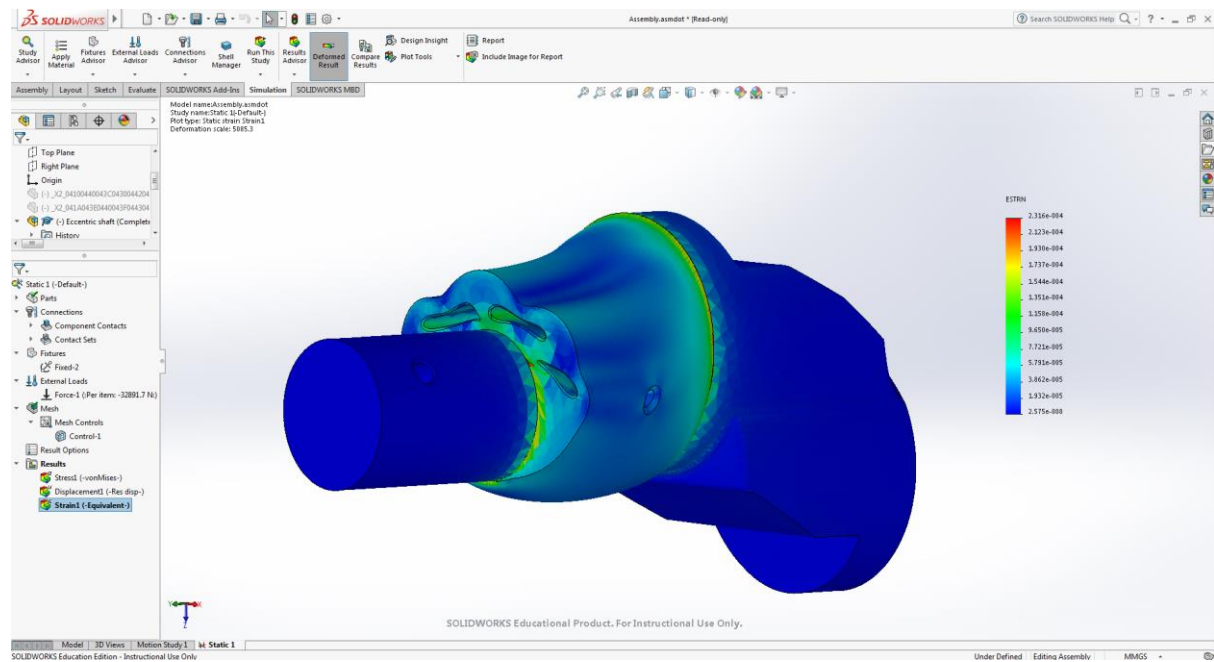


Figure 23.0.3 – Maximum Strain.

Therefore, producing an overall run time of 54 seconds, displayed in Figure 23.0.4.

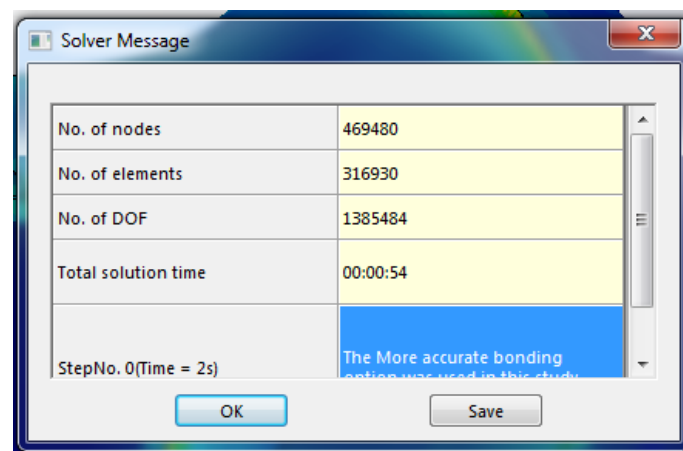


Figure 23.0.4 – Run Time.

14.0.10 – Fuelling Calculations:

Performance and emission calculations were the final calculations produced to finalise the conventional piston versus Wankel rotary engines comparison. Consequently, calculations required were obtained from a previous Powertrain and Sustainability assignment.

14.0.10.1– Ford Focus:

Firstly, calculations for a Ford Focus engine vehicle were produced. The 2011 Ford Focus was selected due to a similar weight to the Mazda Rx8.

FRolling:

Consequently; weight of the Ford Focus was required for the FRolling equation.

Ford Focus = 1333Kg (Parkers, 2017)

Coefficient of friction = 0.015

Gravity (g) = 9.81m/s

$$FRolling = 0.015(M)(g)$$

$$FRolling = 0.015(1333)(9.81)$$

$$FRolling = 196.151N$$

FAero:

Next FAero was required to be calculated, which would later be used to calculate FWheel.

ρ = Air Density = 1.225Kg/m³ (Engineering Tool Box, 2017)

CD = Drag Coefficient of Ford Focus = 0.297 (Car and Driver, 2012)

A = Front Area = Width (Height) = 2.01(1.484) = 2.983m² (Parkers, 2017)

33.35Km/h was selected due to being the average speed, including stops of the New European Drive Cycle (NEDC).

$$V^2 = \text{Average Velocity} = 33.35\text{Km/h} = \frac{33.35(1000)}{3600} = 9.264\text{m/s (Diesel Net, 2017)}$$

Consequently; FAero was obtained.

$$FAero = \frac{1}{2}(\rho)(CD)(A)(V^2)$$

$$FAero = \frac{1}{2}(1.225)(0.297)(2.983)(9.264^2)$$

$$FAero = 46.571N$$

FWheel:

Using data previously calculated for FAero and FRolling.

$$FAero = 46.571N$$

$$FRolling = 196.151N$$

Ford Focus Mass = 1333Kg

Average Acceleration = 0.506m/s²

$$FWheel = FAero + FRolling + M(Accel)$$

$$FWheel = 46.571 + 196.151 + 1333(0.506)$$

$$FWheel = 917.22N$$

Mechanical Energy:

Mechanical energy per second was calculated once FWheel was obtained.

$$FWheel = 917.22N$$

$$V = 33.35Km/h$$

$$Mechanical\ Energy/Sec = FWheel(V)$$

$$Mechanical\ Energy/Sec = (917.22)(9.264) = 8497.126J$$

$$Total\ Mechanical\ Energy = Mechanical\ Energy/Sec(NEDC\ Time)$$

NEDC lasted 1180 seconds, consequently mechanical energy was multiplied by this.

$$Total\ Mechanical\ Energy = (8497.126)(1180 \times 10^{-6}) = 10.027MJ$$

Fuel Energy Required:

To obtain fuel energy required, mechanical energy per second previously calculated is divided by thermal engine efficiency.

Mechanical Energy/Sec 8497.126J

Engine Efficiency = 30% (Energy Education, 2017)

$$\text{Fuel Energy Required} = \frac{\text{Total Mechanical Energy (MJ)}}{\text{Engine Efficiency}} = \frac{10.027}{0.3} = 33.423\text{MJ}$$

NEDC Fuel Mass:

Using the calorific value of gasoline obtained from The Engineering Tool Box and Fuel Energy Required previously calculated. NEDC Fuel Mass for the Ford Focus was calculated.

Gasoline Calorific Value = 48×10^6 J/Kg (The Engineering ToolBox, 2017)

$$\text{Ford Focus NEDC Fuel Mass} = \frac{\text{Fuel Energy Required}}{\text{Gasoline Calorific Value}} = \frac{33.423\text{MJ}}{48\text{MJ}} = 0.696\text{Kg}$$

Litres of Gasoline Used:

Density of Gasoline = 0.75Kg/Litre (Aqua Calc, 2017)

$$\text{Litres of Gasoline Used} = \frac{\text{NEDC Fuel Mass}}{\text{Density of Gasoline}} = \frac{0.696}{0.75} = 0.928\text{Litres}$$

Official NEDC Non-Hybrid (Only using engine) Fuel Consumption:

Litre of gasoline used, previously calculated and NEDC distance obtained was entered into the official NEDC fuel consumption equation.

NEDC = 11.023Km (Car Engineer, 2013)

$$\text{Official NEDC Fuel Consumption} = 0.928\text{Litres}/11\text{Km}$$

$$= \frac{0.928}{11.023} \times 100\text{Litres}/100\text{Km}$$

$$= 8.419\text{Litres}/100\text{Km}$$

$$= \frac{\text{Litres}/100\text{Km}}{4.5} = \frac{8.419}{4.5} = \text{Gallons}/62.5\text{Miles}$$

$$= 1.871 \text{ Gallons}/62.5\text{Miles}$$

Consequently, MPG of the Ford Focus was obtained.

$$MPG = \frac{62.5}{1.871} = 33.405MPG$$

14.0.10.2 – Mazda Rx8:

FRolling:

Mazda Rx8 = 1390Kg (Parkers, 2017)

$$FRolling = 0.015(M)(g)$$

$$FRolling = 0.015(1390)(9.81)$$

$$FRolling = 204.539N$$

FAero:

ρ = Air Density = 1.225Kg/m³ (Engineering Tool Box, 2017)

CD = Drag Coefficient of Mazda Rx8 = 0.3 (Eco Moddler, 2016)

A = Front Area = Width (Height) = 1.846 (1.34) = 2.474m² (Parkers, 2017)

V² = Average Velocity = 33.35Km/h = $\frac{33.35(1000)}{3600}$ = 9.264m/s (Diesel Net, 2017)

$$FAero = \frac{1}{2}(\rho)(CD)(A)(V^2)$$

$$FAero = \frac{1}{2}(1.225)(0.3)(2.474)(9.264^2)$$

$$FAero = 39.014N$$

FWheel:

$$FAero = 39.014N$$

$$FRolling = 204.539N$$

Mazda Rx8 Mass = 1390Kg

Average Acceleration = 0.506m/s²

$$FWheel = FAero + FRolling + M(Accel)$$

$$FWheel = 39.014 + 204.539 + 1390(0.506)$$

$$FWheel = 946.893N$$

Mechanical Energy:

$$FWheel = 946.893N$$

$$V = 33.35Km/h$$

$$Mechanical\ Energy/Sec = FWheel(V)$$

$$Mechanical\ Energy/Sec = (946.893)(9.264) = 8772.017J$$

$$Total\ Mechanical\ Energy = Mechanical\ Energy/Sec(NEDC\ Time)$$

$$Total\ Mechanical\ Energy = (8772.017)(1180 \times 10^{-6}) = 10.351MJ$$

Fuel Energy Required:

Mechanical Energy/Sec 8768.404J

Engine Efficiency = 30% (Energy Education, 2017)

$$Fuel\ Energy\ Required = \frac{Total\ Mechanical\ Energy\ (MJ)}{Engine\ Efficiency} = \frac{10.351}{0.3} = 34.503MJ$$

NEDC Fuel Mass:

Gasoline Calorific Value = 48×10^6 J/Kg (The Engineering ToolBox, 2017)

$$Mazda\ Rx8\ NEDC\ Fuel\ Mass = \frac{Fuel\ Energy\ Required}{Gasoline\ Calorific\ Value} = \frac{34.503MJ}{48MJ} = 0.719Kg$$

Litres of Gasoline Used:

Density of Gasoline = 0.75Kg/Litre (Aqua Calc, 2017)

$$Litres\ of\ Gasoline\ Used = \frac{NEDC\ Fuel\ Mass}{Density\ of\ Gasoline} = \frac{0.719}{0.75} = 0.959Litres$$

Official NEDC Non-Hybrid (Only using engine) Fuel Consumption:

NEDC = 11.023Km (Car Engineer, 2013)

$$Official\ NEDC\ Fuel\ Consumption = 0.959Litres/11Km$$

$$= \frac{0.959}{11.023} \times 100Litres/100Km$$

$$= 8.7Litres/100Km$$

$$= \frac{Litres/100Km}{4.5} = \frac{8.7}{4.5} = Gallons/62.5Miles$$

$$= 1.93 \text{ Gallons}/62.5 \text{ Miles}$$

Consequently, MPG of the Mazda Rx8 was obtained.

$$MPG = \frac{62.5}{1.93} = 32.383 \text{ MPG}$$

Official Litres Used in 11.023Km NEDC:

$$\text{Density of Gasoline} = 0.75 \text{ Kg/Litre (Aqua Calc, 2017)}$$

$$\text{Conventional Combustion Engine Efficiency} = 0.3 = 30\%$$

$$\text{Official NEDC Fuel Consumption} = 0.959 \text{ Litres}/11 \text{ Km}$$

$$\text{Official Litres used in 11.023Km NEDC} = \frac{DE(C1) + 25(C2)}{(D.E + 25)}$$

C2 = Official litres used in NEDC.

DE = Official Miles per Tank Range.

$$= \frac{0 + 25(C2)}{D.E + 25}$$

$$= \frac{0 + 25(0.959)}{(946.893 + 25)} = 0.0247 \text{ Official Litres Used}$$

14.0.11.1 – Chapter References:

1. Aqua Calc 2017, Volume to Weight Conversion, Viewed 21st March 2017, <http://www.aqua-calc.com/page/density-table/substance/gasoline>
2. Car and Driver 2012, Ford Focus US Spec, Viewed 20th April, <http://www.caranddriver.com/reviews/2012-ford-focus-drive-ford-focus-review>
3. Car Engineer 2013, The Different Drive Cycles, Viewed 22nd April, <http://www.car-engineer.com/the-different-driving-cycles/>
4. Diesel Net 2017, Emissions Test Cycle, Viewed 20th March, https://www.dieselnets.com/standards/cycles/ece_eudc.php
5. Eco Moddler 2016, Vehicle Coefficient of Drag List, Viewed 22nd April 2017, http://ecomodder.com/wiki/index.php/Vehicle_Coefficient_of_Drag_List
6. Energy Education 2017, Thermal Efficiency, Viewed 3rd April 2017, http://energyeducation.ca/encyclopedia/Thermal_efficiency
7. The Engineering Tool Box 2017, Air: Density and Specific Weight, Viewed 20th March, http://www.engineeringtoolbox.com/air-density-specific-weight-d_600.html
8. The Engineering Tool Box 2017, Fuels – Higher Calorific Values, Viewed 22nd April 2017, http://www.engineeringtoolbox.com/fuels-higher-calorific-values-d_169.html
9. Jay, 2016, Powertrain and Sustainability Assignment 2B – “Technical Review of Powertrain for 2020”, Powertrain and Sustainability, M6X01206, UWTSD Univfocusersity.
10. Parkers 2017, Ford Focus Hatchback (2011 Onwards) Specs & Dimensions, Viewed 21st April 2017, [http://www.parkers.co.uk/ford/focus/hatchback-2011/16-\(180bhp\)-ecoboost-zetec-s-5d/specs/](http://www.parkers.co.uk/ford/focus/hatchback-2011/16-(180bhp)-ecoboost-zetec-s-5d/specs/)
11. Parkers 2017, Mazda RX-8 (2003 – 2010) Specs & Dimensions, Viewed 21st April, [http://www.parkers.co.uk/mazda/rx-8/coupe-2003/\(192ps\)-4d/specs/](http://www.parkers.co.uk/mazda/rx-8/coupe-2003/(192ps)-4d/specs/)

14.0.11.2 – Chapter Literature Review:

1. Aqua Calc – Obtained density of gasoline.
2. Car and Driver – Ford Focus 2012 Coefficient of Drag obtained.
3. Car Engineer – Obtained distance of NEDC.
4. Diesel Net – Obtained average speed during NEDC.
5. Eco Moddler – Obtained Mazda Rx8 drag coefficient.
6. Energy Education – Obtained ICE thermal efficiency percentage.
7. The Engineering Tool Box – Obtained Air Density.
8. The Engineering Tool Box – Obtained calorific value of gasoline.
9. Jay – Obtained emission calculations required.
10. Parkers – Obtained weight, dimensions, CO₂ and miles per tank of 2011 Ford Focus.
11. Parkers – Obtained weight of Mazda Rx8 192.

14.0.12– Gantt chart:

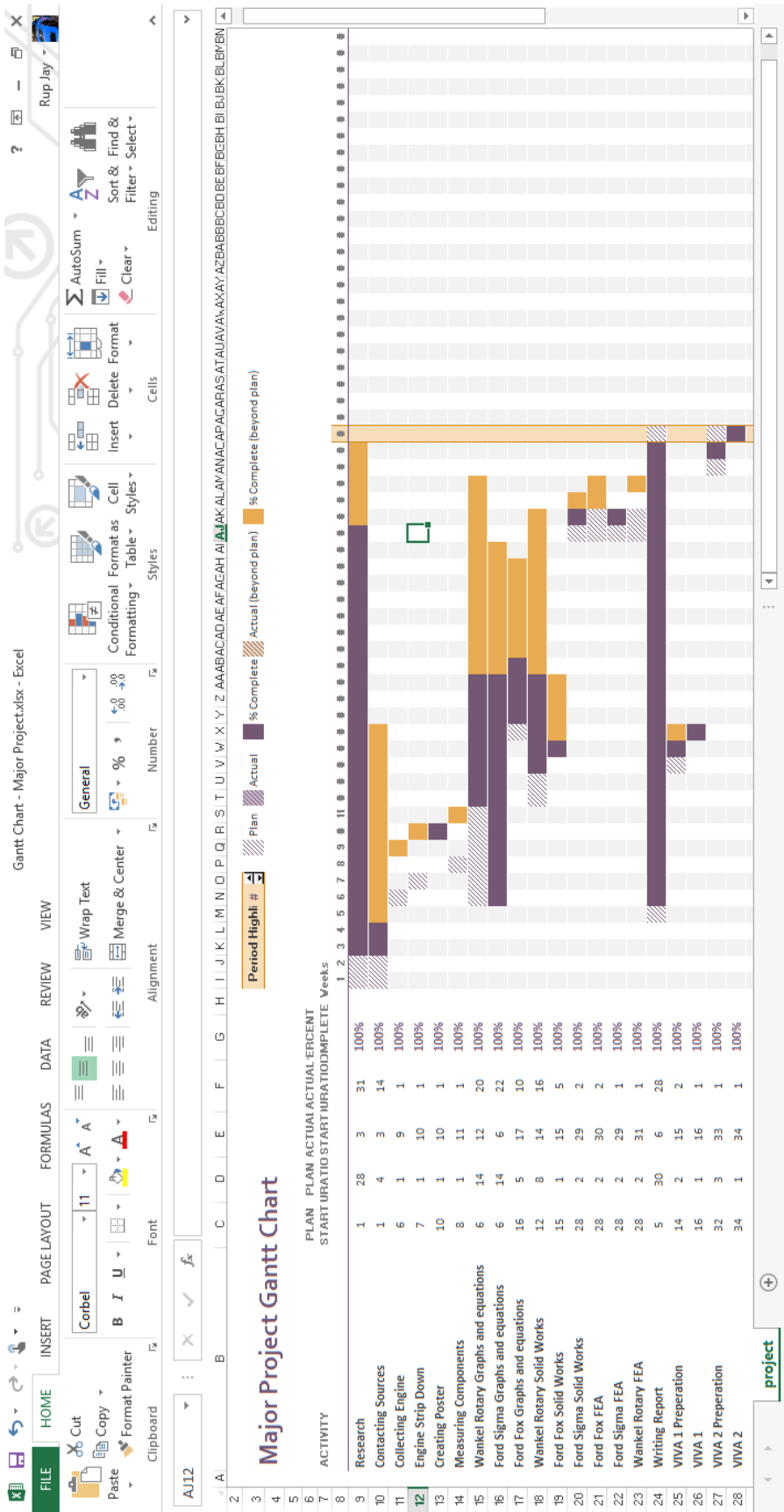


Figure 24.0 – Gantt chart.

Further explanation of Gantt chart will be provided in VIVA 2.

14.0.13 – Chapter Reference:

1. Aircraft Market Place 2010, Why Not Wankels, Viewed 23rd April 2017, <http://www.acmp.com/blog/why-not-wankels.html>
2. Atkins Rotary Marine 1996, Ebb Tide Rotary Boat, Viewed 23rd April 2017, <http://www.atkinsrotarymarine.com/images/EbbTideRotaryBoat.jpg>
3. Popular Mechanics 2015, Military, Viewed 22nd April 2017, <http://www.popularmechanics.com/military/research/a15233/liquidpiston-darpa-contract/>
4. Visor Down 2015, Top Ten Wankel Engine Bikes, Viewed 23rd April 2017, <http://www.visordown.com/features/motorcycle-top-10s/top-10-wankel-engined-bikes>
5. Yamamoto, 1981, Automotive, Rotary Engine, Toyo Kogyo CO LTD, Japan

14.0.13.1 – Chapter Literature Review:

1. Aircraft Market Place – Investigated Wankel engine aeroplanes.
2. Atkins Rotary Marine – Investigated rotary engine boats.
3. Popular Mechanics – Investigated rotary engine military drones.
4. Visor Down – Investigated Wankel engine motorcycles.
5. Yamamoto – Obtained inefficiencies of Wankel Rotary engine.

Chapter 15.0 – Bibliography:

1. Ansdale, 1968, Engineering and Design, The Wankel RC Engine – Design and Performance, Liiffe Books, Frome.
2. Blair, 1999, Technology and Engineering, Design and Simulation of Four Stroke Engines, SAE International, Michigan.
3. Explain That Stuff 2016, How Turbochargers Work, Viewed 18th April 2017, <http://www.explainthatstuff.com/how-turbochargers-work.html>
4. Heywood, J.B, 1998, Internal Combustion Engine Fundamentals, McGraw-Hill Education Publishers, Michigan.
5. Mazda Rx7 Club 2007, Rotary Car Performance, Viewed 8th December 2016, <https://www.rx7club.com/rotary-car-performance-77/solidworks-project-659898/>.
6. Racing Beat 2017, Eccentric Shafts, Stationary Gears and Main Bearings, Viewed 20th April 2017, <http://www.racingbeat.com/mazda/performance/rotary-tech-tips/e-shaft-gears-bearings.html>
7. Stubblefield & Haynes, 1986, Automotive, Mazda Rx-7 Haynes Repair Manual, Haynes Publishing Group, Yeovil.
8. U.S Environmental Protection Agency, 1974, Emissions and Fuel Economy Comparative Study of the Fuel Economy Characteristics of Mazda Rotary Engine Vehicles Versus Conventional Reciprocating Engine Vehicles, USA.
9. Zammit, 1987, Engineering Science, Motor Vehicle Engineering Science for Technicians, Longman Group UK Limited, Malaysia.

15.0.1 – Software and Equipment:

1. CMM
2. Digital Vernier Callipers
3. Microsoft Excel 2013
4. Microsoft Power Point 2013
5. Microsoft Word 2013
6. Scientific Calculator
7. Solid Works 2016
8. Stationary (Inc Protractor)