

University of Wales Trinity Saint David
School of Engineering

Optimization of a High Speed 3 Cylinder Engine Using Ricardo WAVE Development Software

Carl P. Engelbrecht
P150898

Advanced Computational Methods
M6X00408
Motorsport Engineering

27 January 2020



Contents

Introduction.....	2
Engine Parameters	2
Combustion Analysis.....	3
1.1 Cylinder Pressure	3
1.1.1 PV Diagram	4
1.2 Wiebe Function	5
1.2.1 Start and End of Combustion	6
1.2.2 MFB Curve	7
1.2.3 Wiebe Exponent	8
Wave Model Optimisation	9
1.3 Data Table Inputs	9
1.3.1 Bore Stroke, Clearance Height and Conrod Length	9
1.3.2 Valve Diameters & Maximum Lift	9
1.3.3 Throttle Venturi Diameter	10
1.3.4 SI Wiebe	10
1.3.5 Data Table Wave Results	10
1.3.6 Experiments	11
1.4 Final Optimisations.....	17
Conclusion.....	18
References	19
Appendix	19

Introduction

Engine development is an expensive process limited by cost and time and can be greatly aided by computer simulation before any physical changes are made. A software package which has made such development possible is Ricardo wave, a one-dimensional computational fluid dynamics engine simulation package. Through detailed engine parameter inputs the software is able to simulate the engine operation as a whole rather than through individual sub-assembly simulations, however this is possible in conjunction with WAVE to achieve greater development and levels of accuracy. The simulation is achieved by analysing the transfer of mass, momentum and energy for compressible gas flows through a network of modelled ducts, orifices and various other interactions. (Ricardo , 2019) (Cordon, et al., 2007)

The objective of this report will be to evaluate an existing Triumph 675cc engine model developed within Ricardo Wave using real world engine specifications and in-cylinder pressure data. The cylinder pressure data is analysed to understand the tested engine's characteristics such as start and end of combustion and burn durations. Then through an iterative process the existing model is evaluated after each change of information against the real data in hopes to achieve an accurate model in which further development can be based.

Engine Parameters

The quality of the output calculations is only as accurate as the initial inputs given; therefore, it was essential to build up an accurate data table of parameters to insert into the Ricardo Wave model. The engine analysed is a 765cc naturally aspirated three cylinders out of a Triumph Daytona, manufactured between the years 2006 and 2016.

Initially a data table was provided with the Ricardo WAVE model of the engine, see appendix Table 3 - The data table to accompanying the wave model given. The information was cross referenced with that found from (aperaceparts.com, 2011) and (www.motorcyclespecs.co.za, 2009) and it was found some of the data did not correlate. The information from one of the sources matched that of what was found in the data table, it must be mentioned the specification and year of the motorcycle is unknown, and a complete table was compiled as seen in Figure 1 - Table of all the reference data gathered and calculated from the existing engine specifications.

Additional information was calculated such as the clearance height, a known value input within the model. It is done through Equation 1 - Clearance height calculation. which is a product of clearance volume (see appendix Equation 4 - Clearance volume equation.) and bore.

$$Clearance Height = \left(\frac{Clearance Volume}{\pi * \left(\frac{Bore}{2} \right)^2} \right) * 10 \quad mm$$

Clearance Volume (cc), Bore (mm)

Equation 1 - Clearance height calculation.

DATA TABLE		
Engine Displacement	675.0	cc
Cylinders	3	
Max. Swept Vol	225.01	cc
Comp Ratio	12.65	
Bore	76.00	mm
Stroke	49.6	mm
Clearance Height	4.25751	mm
Clearance Vol.	19.31	cc
Con Rod Length	104.60	mm
Crankshaft Radius	24.8	mm

Valves		
Inlet Valve Dia	30.50	mm
Exhaust Valve Dia	25.50	mm
Valves per Cylinder	4.0	
Intake Valve Maximum Lift	9.3	mm
Exhaust Valve Maximum Lift	8.5	mm
Inlet Valve Timing		
Open BTDC	27.49	deg
Close ABDC	51.49	deg
Duration	259.98	deg
Exhaust Valve Timing		
Open BTDC	43.54	deg
Close ABDC	23.54	deg
Duration	247.08	deg
Combustion		
Lamda	0.9	
n Compression	1.320	
n Expansion	1.360	
Wiebe a		
Wiebe m		
T Ambient (K)	298.00	K
Combustion Efficiency	0.90	
Volume Efficiency	0.90	Joules
General		
Throttle Venturi Size	44	mm
Power	94	KW @ 12600

Figure 1 - Table of all the reference data gathered and calculated from the existing engine specifications.

Combustion Analysis

The combustion analysis is undertaken for two reasons, the first is to graph and understand the existing cylinder pressures within the engine being studied in order to validate the calculated data from the WAVE model. The second reason is through understanding the various data inputs the software calculation is built upon, the first one being the Wiebe function and the second the Woschini model, data can be gathered through analysis to add to the model in hopes of improving accuracy.

The cylinder pressure data and volume data are gathered in a text document which is imported into Microsoft excel as a reference point for further analysis conducted within a Matlab script detailed in the following chapter.

1.1 Cylinder Pressure

The cylinder pressures are plotted for each engine speed and used as a reference point during each iterative change within the WAVE model. The data given is from cylinder 1.

The maximum cylinder pressure is experienced at 14400 RPM (83.8 bar), followed by 10250 RPM (80.3 bar) and 13000 RPM (79.2). The shape and positioning as well as maximum pressure figure are what will be strived for.

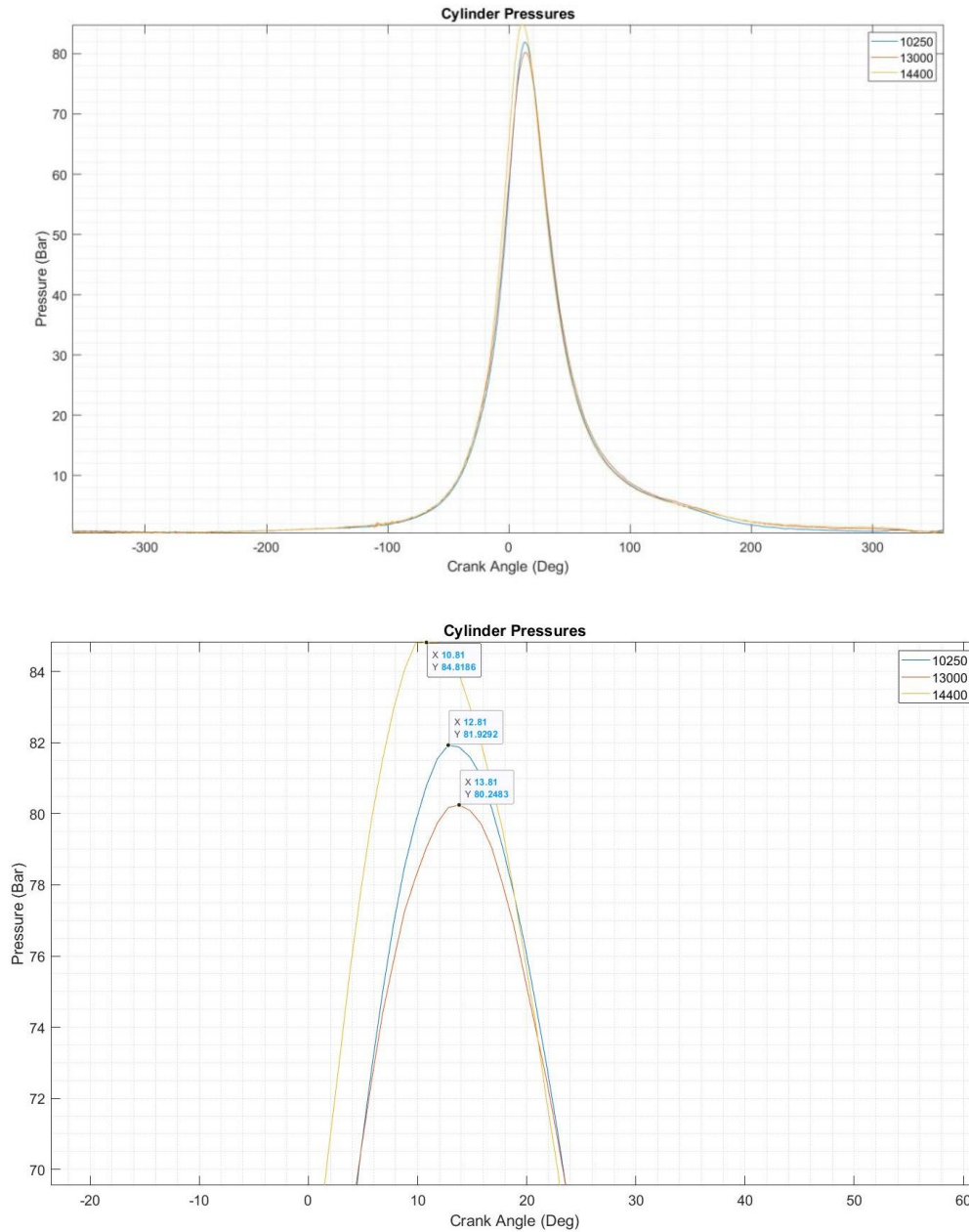


Figure 2 - Cylinder pressures across all measured engine speeds.

1.1.1 PV Diagram

The volume from given data is plotted against cylinder pressures to feature as another point of reference. The given data is in a percentage format which is converted into useable cubic centimetres based off of the values in the data table, through Equation 2 - Volume % to volume cc conversion, required to compare WAVE data.

$$Volume = \left(\left(\frac{Volume \%}{100} \right) * Max. Swept Volume \right) + Clearance Volume \quad cc$$

Equation 2 - Volume % to volume cc conversion, required to compare WAVE data.

The PV diagram will form the bases of the WAVE data validation process as pressure and volume can be simultaneously referenced.

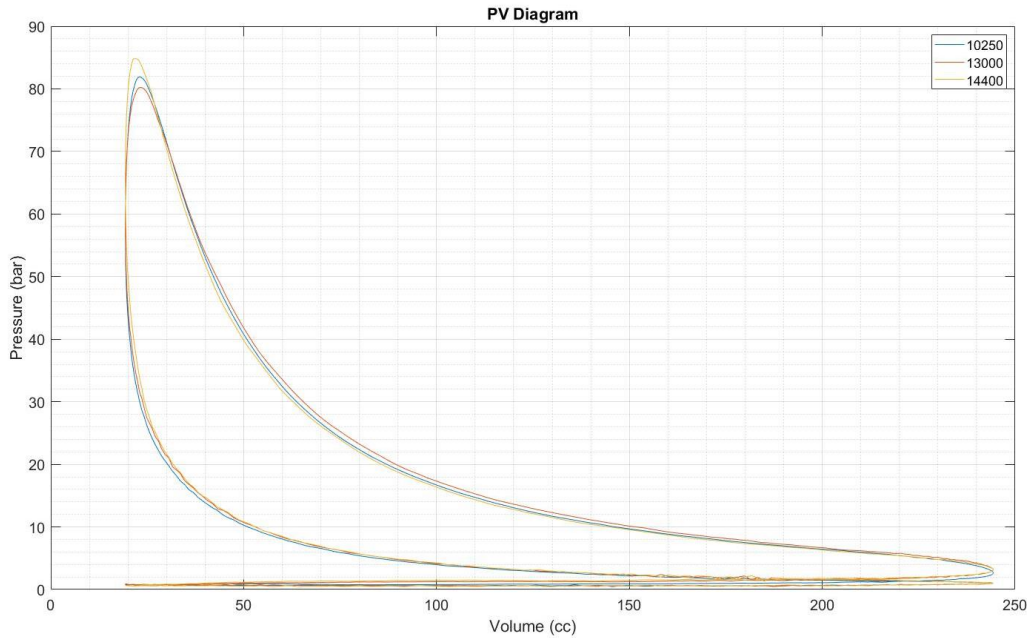


Figure 3 - Pressure Volume diagram of all engine speeds.

1.2 Wiebe Function

Wave allows for the input of three parameters concerning combustion characteristics, 10-90 % burn duration, 50 % burn crank location and a Wiebe exponent, as seen in Figure 4 - Wiebe window within Ricardo WAVE.. These inputs form part of the Wiebe function, Figure 5 - Wiebe equation., used within WAVE to simulate combustion.

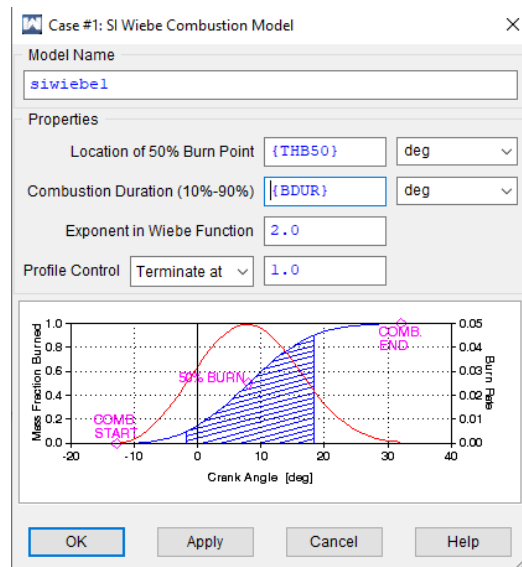


Figure 4 - Wiebe window within Ricardo WAVE.

$$W = 1 - e^{-AWI \left(\frac{\Delta\theta}{BDUR} \right)^{WEXP+1}}$$

Figure 5 - Wiebe equation. (Cordon, et al., 2007)

The Wiebe function is a mathematical equation that describes the known history of combustion through its parametric relationships. As combustion is a product of first law reactions.

Therefore, in order to provide WAVE with the input data, the in-cylinder pressure data is further analysed as detailed below.

1.2.1 Start and End of Combustion

The start and end of combustion (SOC and EOC, respectively) is necessary in calculating the mass fraction burned curve. This is found through the analysis of a difference in extra pressure graph for each engine speed. The extra pressure is found through the method described by (Blair, 1999), using equation Equation 3 - Extra pressure from combustion.

$$P_{2e} = \text{Pressure} * \left(\frac{V_1}{V_2}\right)^n$$

Pressure (bar), V_1 and V_2 (cc), n = Polytropic index either for compression or expansion.

Equation 3 - Extra pressure from combustion.

The polytropic index is essential in predicting in-cylinder extra combustion pressure and is thus a proponent of the extra pressure equation. It is described by the equation,

$$PV^n = \text{Constant}$$

The polytropic index is found through plotting logarithmic pressure and volume against one another. Understanding that before and after combustion, and after the inlet valve closes and before the exhaust valve opens the relationship stated above is satisfied, therefore the slope between those points produce a mean polytropic index. The slope taken for both compression and expansion are seen in Figure 6 - Logarithmic PV diagram at 14400 RPM. Allowing the polytropic index of expansion and compression to be found.. As further calculation is known to be insensitive to minor changes in the polytropic Index an average is created as feature in the table to be used in the further calculations.

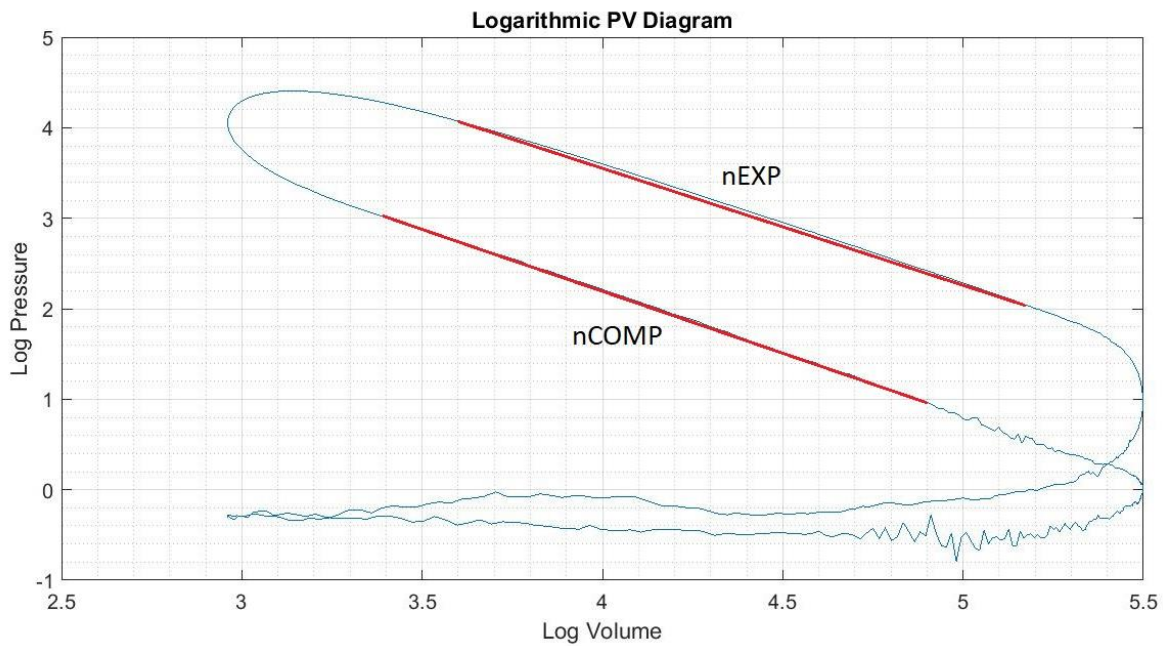


Figure 6 - Logarithmic PV diagram at 14400 RPM. Allowing the polytropic index of expansion and compression to be found.

Examining the difference between real in-cylinder pressure and extra pressure, $P_2 - P_{2e}$, it produces a graph where the start of combustion can be identified by the sudden spike in graphical data. The end of combustion can be identified by the return of the graphical spike to more constant pattern. This is seen in all three of the engine cases in figure x.

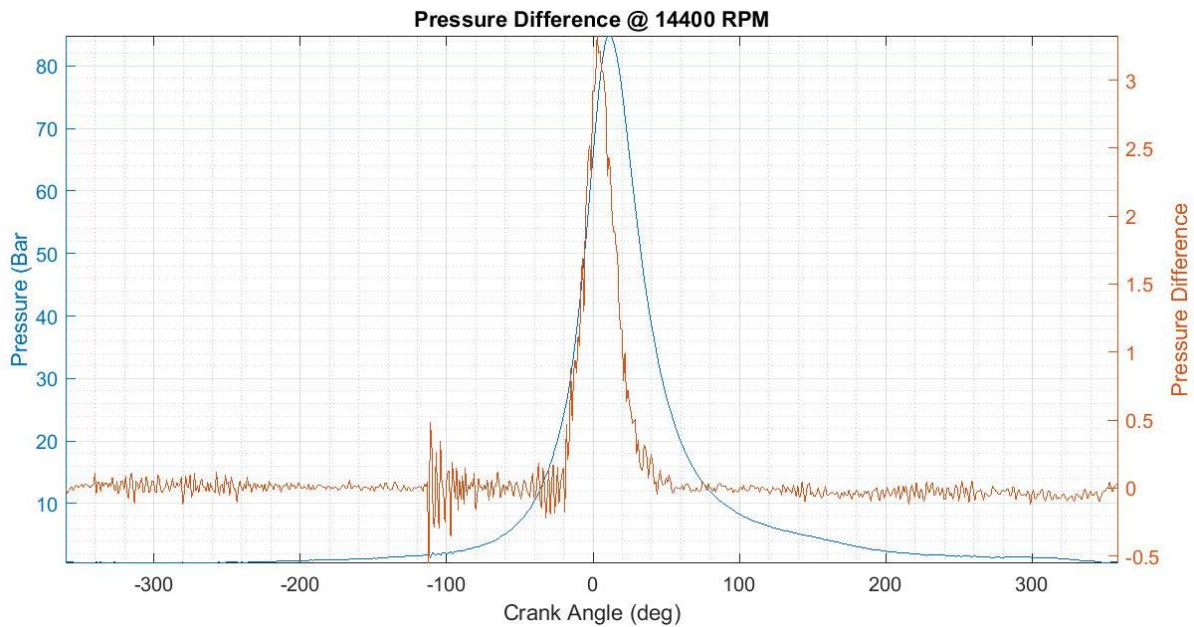


Figure 7 - Combustion pressure overlaid by pressure difference from extra combustion pressure, allowing the SOC and EOC to be found.

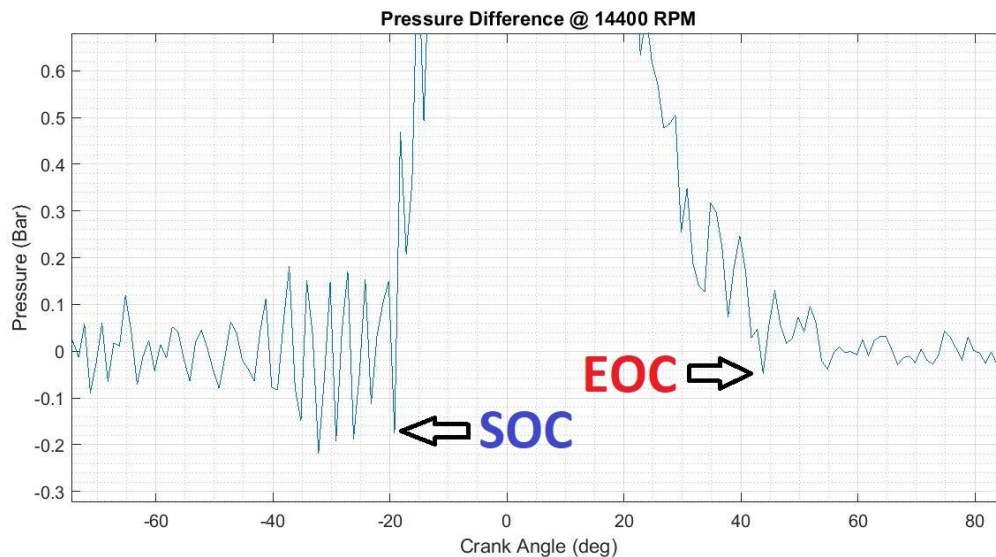
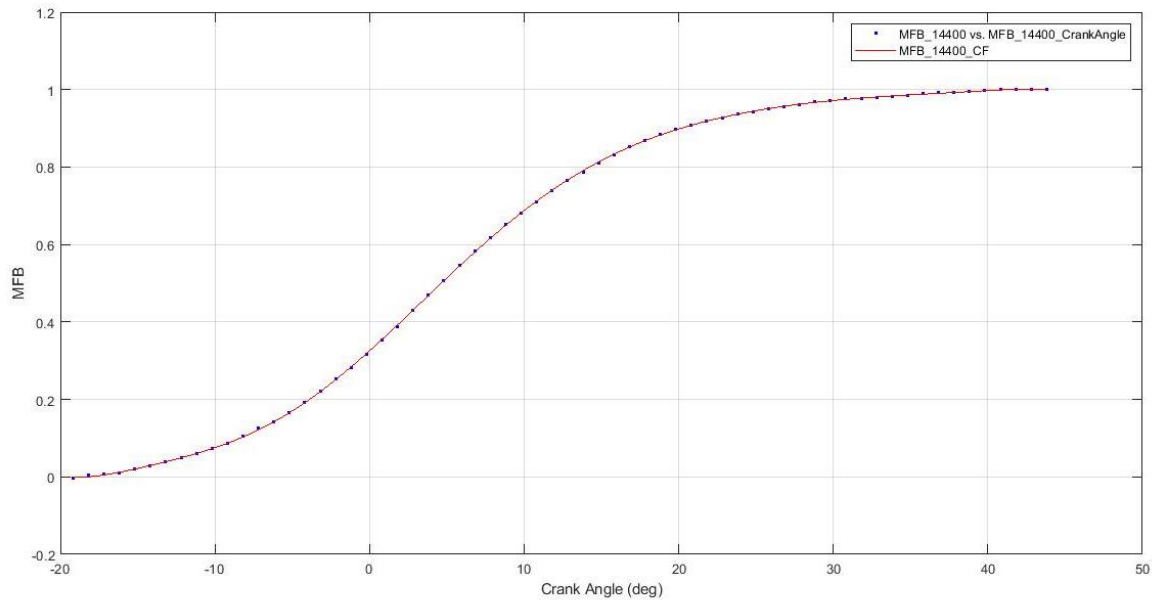


Figure 8 - Identifying points for the SOC and EOC.

In all engine speed cases there is a large amount of noise present at SOC and EOC which means it is difficult to definitively identify at which point either occur. According to John B. Heywood the start of combustion is defined as the separation from a linear line within a logarithmic P-V diagram. Using this information, a more educated point can be selected. The SOC and EOC as well as burn duration are classified in the Table 1 - Data table of analysed combustion parameters such as nEXP, nCOMP, SOC, EOC, 10-90% burn and 50% burn position.. These values were used for the calculation of mass fraction burned curve.

1.2.2 MFB Curve

The mass fraction burned curve is calculated through a cumulative summary taken from that start of combustion until the end and converted into a percentage up to 1 (complete burn).



In order to specifically identify 10-90% burn and 50% a curve fit had to be made to the MFB data as not enough data points were available. This was done through the curve fitting tool within the MATLAB software package and fit to a polynomial of the 9th degree, as this featured the best fit. The data was interpolated to find the 10%, 50% and 90% points.

The data points for burn duration and 50% burn are identified for each case, showcased in Table 1 - Data table of analysed combustion parameters such as n_{EXP} , n_{COMP} , SOC, EOC, 10-90% burn and 50% burn position..

Engine Speed	n_{EXP}	n_{COMP}	SOC	EOC	10-90% Burn	50% Burn
10250	1.3243	1.3661	-22.19	44.81	35	6.22
13000	1.3221	1.3671	-23.19	46.81	28	7.05
14400	1.3384	1.3717	-19.19	43.81	28	4.69

Table 1 - Data table of analysed combustion parameters such as n_{EXP} , n_{COMP} , SOC, EOC, 10-90% burn and 50% burn position.

1.2.3 Wiebe Exponent

The Wiebe exponent is another component of the equation. The values used are that based off the findings in (Cordon, et al., 2007). It is seen that the value above 9000 RPM consistently remains at 1.45 and is assumed to remain at this value at higher RPM.

Data Group	Engine Speed (RPM)	BDUR (Degrees)	WEXP	CA50 (ATDC)
Recommended NA* Values [Blair, 1999]	1200	14	1.72	20
	2400	15	1.93	25
	3600	17	1.82	25
	6000	21	1.64	32
NA* Input Values	3000	16	1.85	25
	4000	17	1.75	27
	5000	19	1.70	29
	6000	21	1.60	32
	7000	22	1.55	33
	8000	23	1.50	34
	9000	23	1.45	35
	10000	23.5	1.45	35
	11000	24	1.45	35

Figure 9 - Wiebe exponents. (Cordon, et al., 2007)

Wave Model Optimisation

Through an iterative process changes to the bare wave model is evaluated against the real data. The model evaluation is approached into three steps, initially the known data specifications are entered and cases run. This sets the mechanical settings and calculated data which will remain unchanged for the second approach. Secondly experiments will be done in order understand the affect certain changes in elements have on the Pressure-Volume curve. Following this, educated changes can be made to the PV diagrams from the first step, and final best fit can be achieved.

1.3 Data Table Inputs

The pre-built wave model is first opened and the valve flow data and lift data are inserted into the model. This is the bare minimums required for it to complete a run.

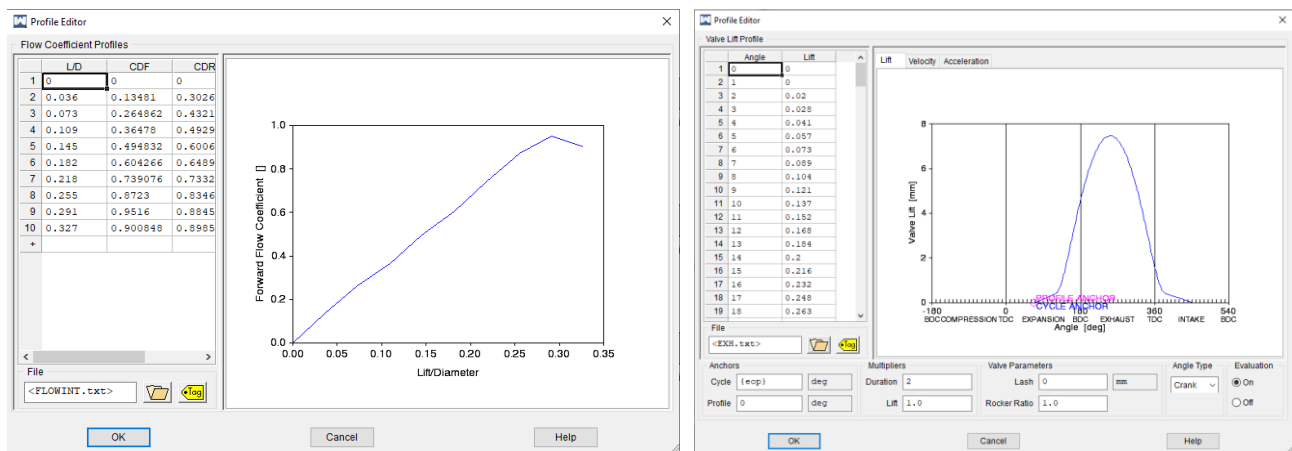


Figure 10 - Valve flow coefficients and valve lift profiles inputted at the start of the WAVE model iterations.

Once the initial run is done the results are viewed in wave post and then exported to Excel for further analysis and comparison as seen in Appendix Figure 27 - Iteration analysis window within Microsoft Excel., and the changes commence. The change made is each iteration is featured in the Table 2 – Specified iteration changes.. Only data from cylinder one is taken to be analysed against that of the real engine.

Iteration	Change
0	Only valve coefficients added and lift profiles.
1	Bore and stroke confirmed and clearance height added.

2	Con-Rod length changed.
3	Valve diameters altered.
4	Maximum lift multiplier.
5	Wiebe Variables.

Table 2 – Specified iteration changes.

1.3.1 Bore Stroke, Clearance Height and Conrod Length

The logical starting point was to input the values from the data table. For the first iteration bore and stroke were confirmed to be correct. Clearance height, calculated previously, was also added.

Iteration 2 saw a change in the conrod length from 98 to 104.6 mm. Both changes saw a maximum pressure increase of 0.1 bar.

1.3.2 Valve Diameters & Maximum Lift

For the following iteration the valve diameters were adjusted to that featured in the table, a minor increase was apparent in each instance. Once the adjustment was made an error occurred when the model was run, indicating the duct diameters did not allow for the increased diameters, Figure 11 - Valve diameter change error.. Therefore, the duct diameters were made 0.7mm smaller than the valve diameters. This is based off of the duct-to-valve diameter ratio of the previous configuration. The model run without fail and the results showed a minor variation in pressures.

For maximum valve lift a data sheet was referenced for the real maximum lift for both inlet and exhaust. A percentage difference is calculated and input into the lift multiplier to achieve the right value. Reviewing Figure 28 - Iterational differences present seen at 14400 rpm. a pressure increase of 3.48 bar in total from previous iteration is present.

```

ENG: VALVES
F*** MAXIMUM VALVE EFFECTIVE AREA IS LARGER THAN CONNECTING DUCT; VALVE # 1 AT DUCT duct13
* AEFFMAX = 672.9 (mm2), ABIG = 629.0 (mm2)
F*** MAXIMUM VALVE EFFECTIVE AREA IS LARGER THAN CONNECTING DUCT; VALVE # 1 AT DUCT duct16
* AEFFMAX = 672.9 (mm2), ABIG = 629.0 (mm2)
F*** MAXIMUM VALVE EFFECTIVE AREA IS LARGER THAN CONNECTING DUCT; VALVE # 1 AT DUCT duct8
* AEFFMAX = 672.9 (mm2), ABIG = 629.0 (mm2)
F*** MAXIMUM VALVE EFFECTIVE AREA IS LARGER THAN CONNECTING DUCT; VALVE # 1 AT DUCT duct11
* AEFFMAX = 672.9 (mm2), ABIG = 629.0 (mm2)
F*** MAXIMUM VALVE EFFECTIVE AREA IS LARGER THAN CONNECTING DUCT; VALVE # 1 AT DUCT duct36
* AEFFMAX = 672.9 (mm2), ABIG = 629.0 (mm2)
F*** MAXIMUM VALVE EFFECTIVE AREA IS LARGER THAN CONNECTING DUCT; VALVE # 1 AT DUCT duct35
* AEFFMAX = 672.9 (mm2), ABIG = 629.0 (mm2)
W*** Skipping subcase due to fatal input error
I*** Remaining cases skipped

```

Figure 11 - Valve diameter change error.

1.3.3 Throttle Venturi Diameter

Although the throttle venturi was not altered moving forward it has to be mentioned that some specifications on the motorcycle set the throttle venturi size at 44mm as opposed to that of 40mm in the wave model. Initially it was altered to that of the specification sheet however the caused the pressures to dramatically rise in all engine speed cases. A aspect which could be contracted with the medication of other elements, however it was decided that the original venturi size is the most appropriate.

1.3.4 SI Wiebe

The result from the Wiebe investigations concerning burn duration between 10-90% and crank angle at 50% burn were added in last. The Wiebe exponent was considered from the literature produced by (Cordon, et al., 2007), which resulted in a value of 1.45. The Wiebe components further increase the pressure by 2 bar and altered the shape of the graph. The changes from individual Wiebe components are further investigated in the next chapter.

1.3.5 Data Table Wave Results

The result from the five iterations are showcased below, graphing the bare run and the final iteration. The best match is seen at 14400 rpm where the curve has moved to suit that of the real engine in both pressure and gradient. The other two engine speed exhibit very small changes from the first iterations. Therefore, only using the inputs from the data table does not prove sufficient enough to match that of the real engine.

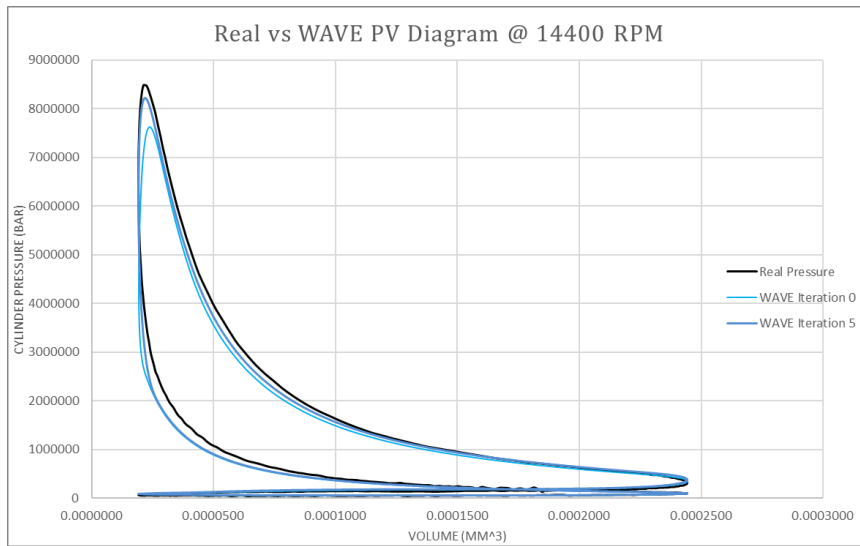


Figure 12 - Final PV diagram results for 14400 from data table inputs.

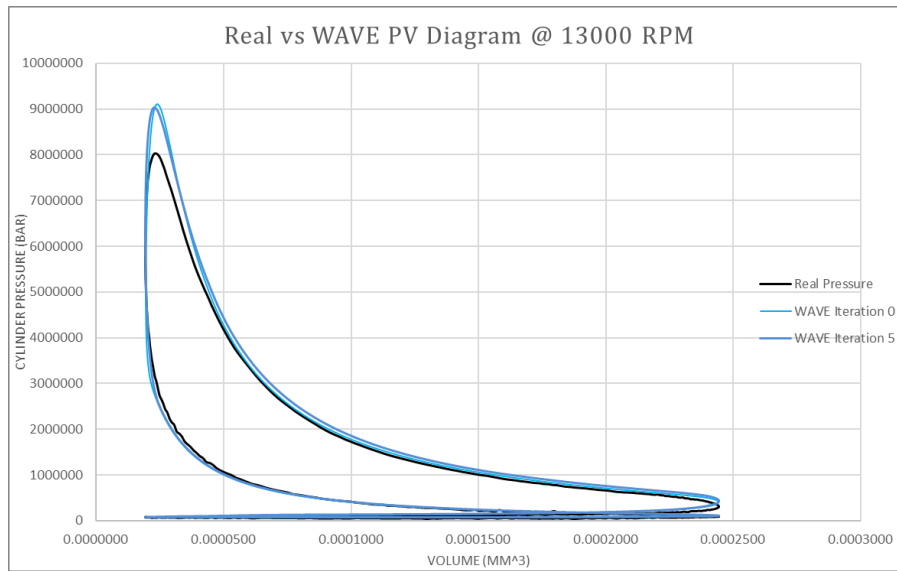


Figure 13 - Final PV diagram for 13000 from data table inputs.

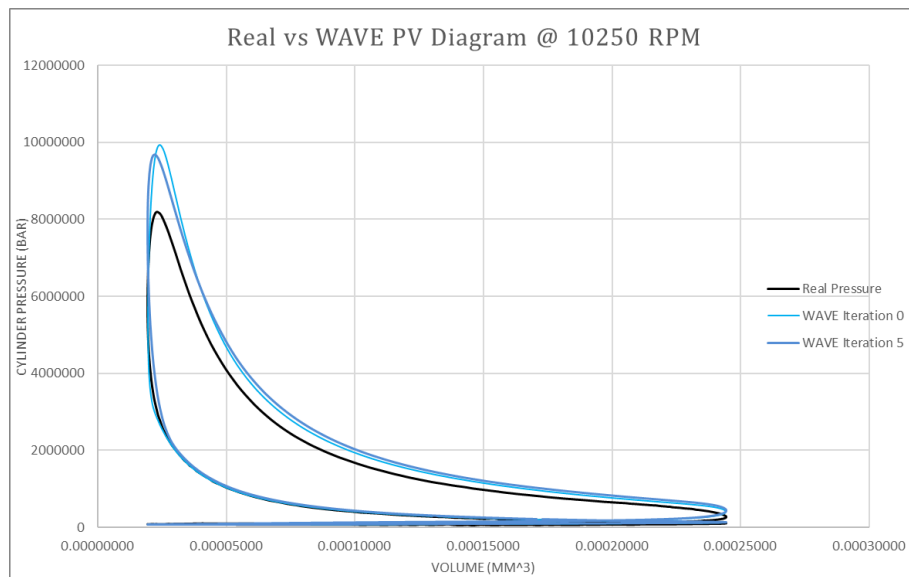


Figure 14 - Final PV diagram for 10250 from data table inputs.

1.3.6 Experiments

In order to understand the impact input changes has within WAVE on the model and corresponding outputs experiments are conducted. Using the final iteration input data from the previous chapter numerous cases are run altering on of the input parameters. The constants used in the experiments are illustrated bellow.

	Name	Units	Case 1
Status			Run
Title			Case 1
1	BDUR	deg	20
2	eop	deg	85
3	exh		180
4	FARD		0.08
5	iop	deg	250
6	NCYC		20
7	PREF		1.01325
8	SPEED	rpm	13000
9	TCYL		575
10	THB50	deg	8
11	THEAD		600
12	thrt	mm	40
13	TPIS		550
14	TREF		298.15
15	WEXP		2
+			

Figure 15 - Constant variables used in the experiments.

1.3.6.1 Burn Duration, 50% Burn Location and Wiebe Exponent

The first experiment was done looking at the components of the Wiebe function. Firstly, the burn duration is swept from 20° to 50° in 2° increments, the result is seen in **Error! Reference source not found.** Increasing burn duration lowers the peak pressure as well as increase the gradient of the curve during compression. The gradient decreases on expansion with longer burn durations. The reason for this being is that a longer burn duration causes the charge to be burnt in a bigger volume.

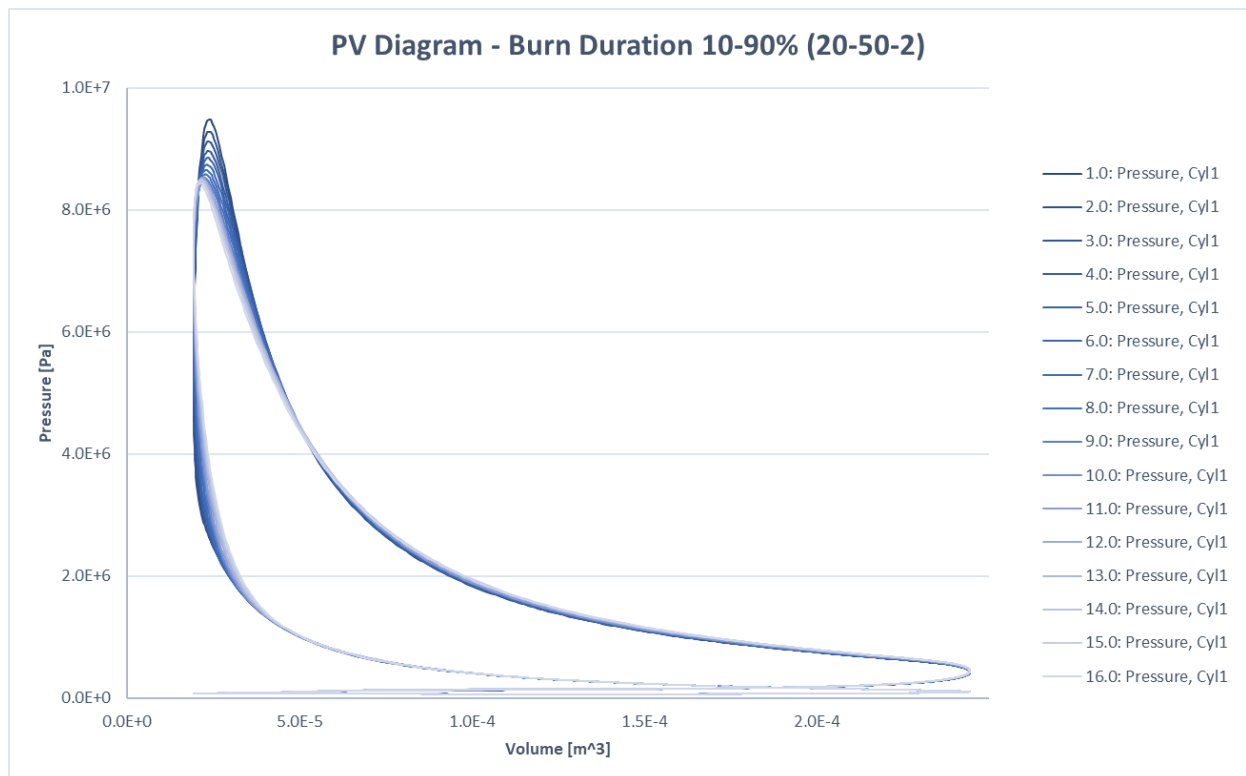


Figure 16 - Burn duration 10-90% sweep. The sweep is from 20° to 50° in 2° steps, the legend represents these steps in values of 1 to 16.

Analysing the burn duration sweep plot a clear influence on peak pressure is noticed. As the burn location gets closer to that of TDC it increases and vice versa. The 50% burn location anchors the mass fraction burned curve and dictates the percentage burned at the smallest volume, lending to the higher pressures as it nears TDC. However,

inlet valve timing must be considered in connection to an alteration in 50% burn location as they both control the timing of the combustion phenomenon.

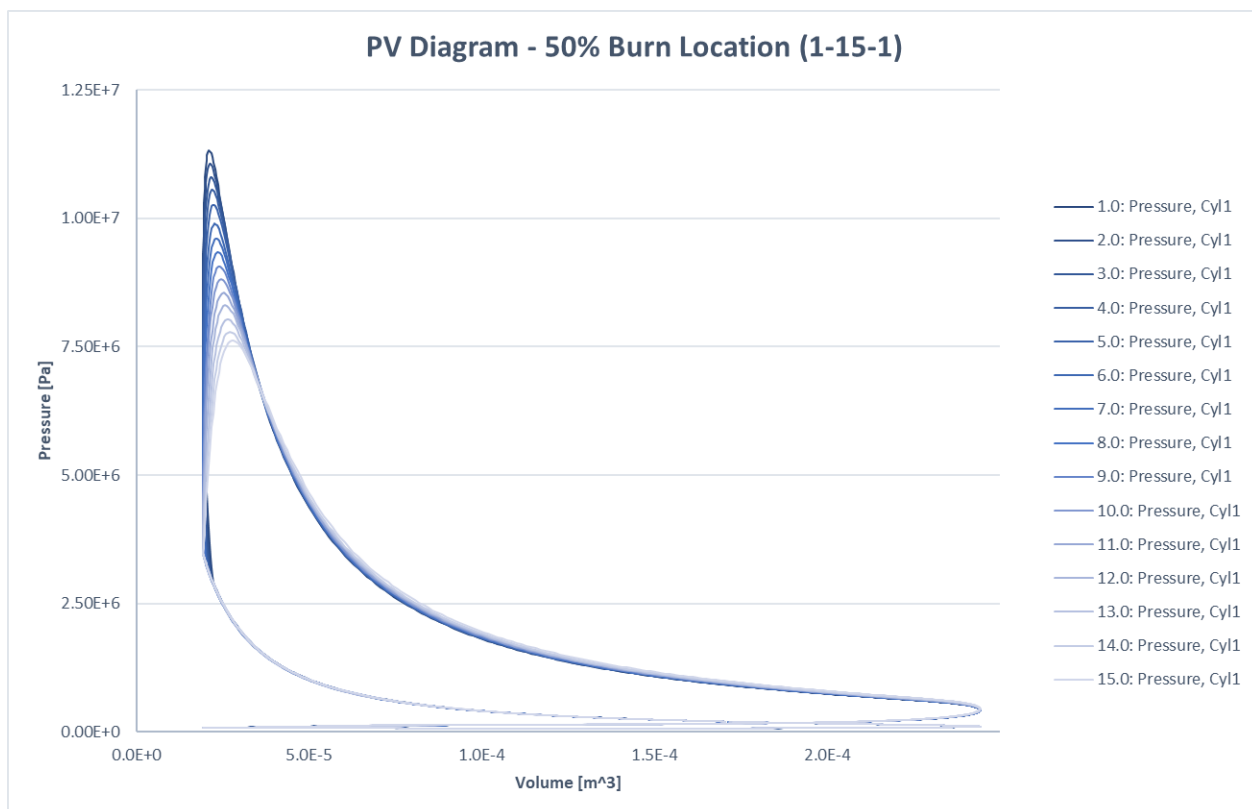


Figure 17 - 50% burn crank angle location sweep from 1° to 15°.

Very minor differences were seen with change of Wiebe exponent (WEXP) across the range studied.

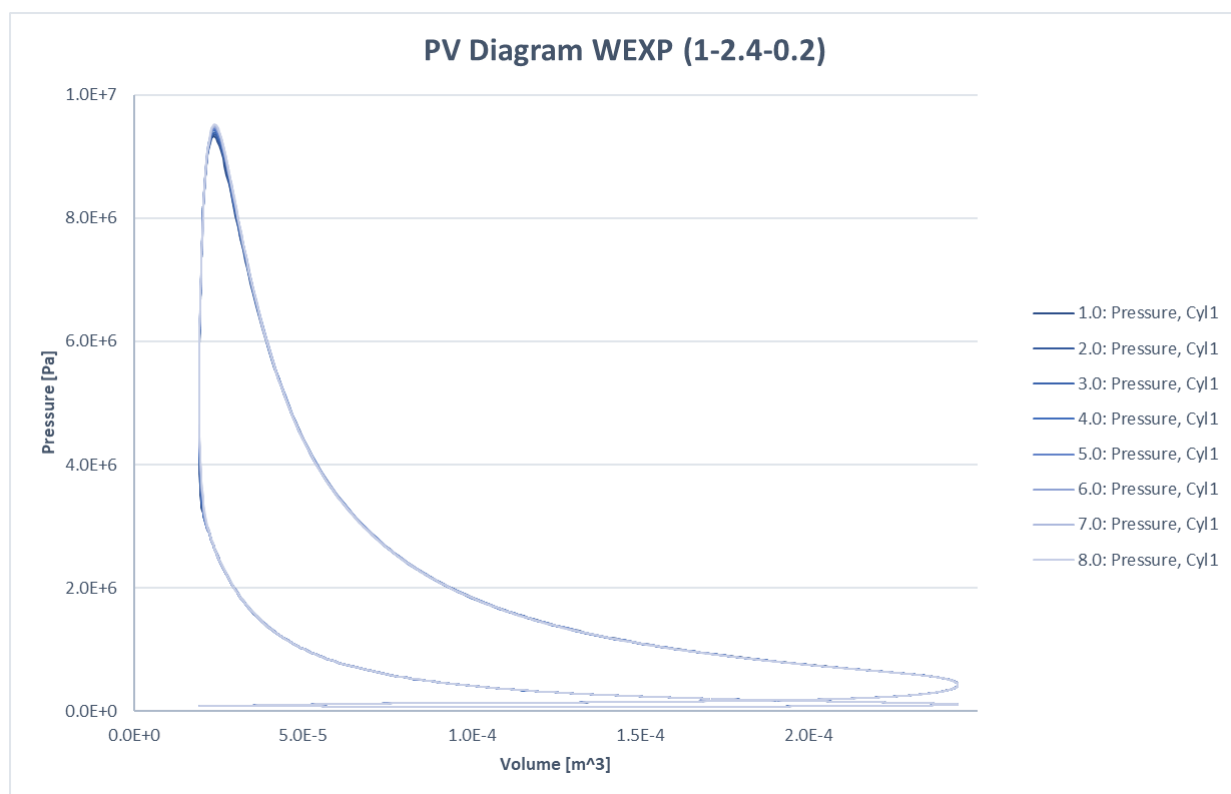


Figure 18 - Wiebe exponent sweep from 1 to 2.4.

1.3.6.2 FARD

The FARD is the fuel air ratio, the inverse of air fuel ratio. Added to each of the injectors for all of the cylinders through the use of a variable {FARD}. A sweep is done from 0.09 (rich) to 0.06 (lean), stoichiometric falls at 0.068. The gradients of the curve are not affected through the range, only the peak pressures. Peak pressure is seen at roughly stoichiometric, as either side the pressures are lower. When the combustion becomes leaner the pressure drops to a greater degree than when rich.

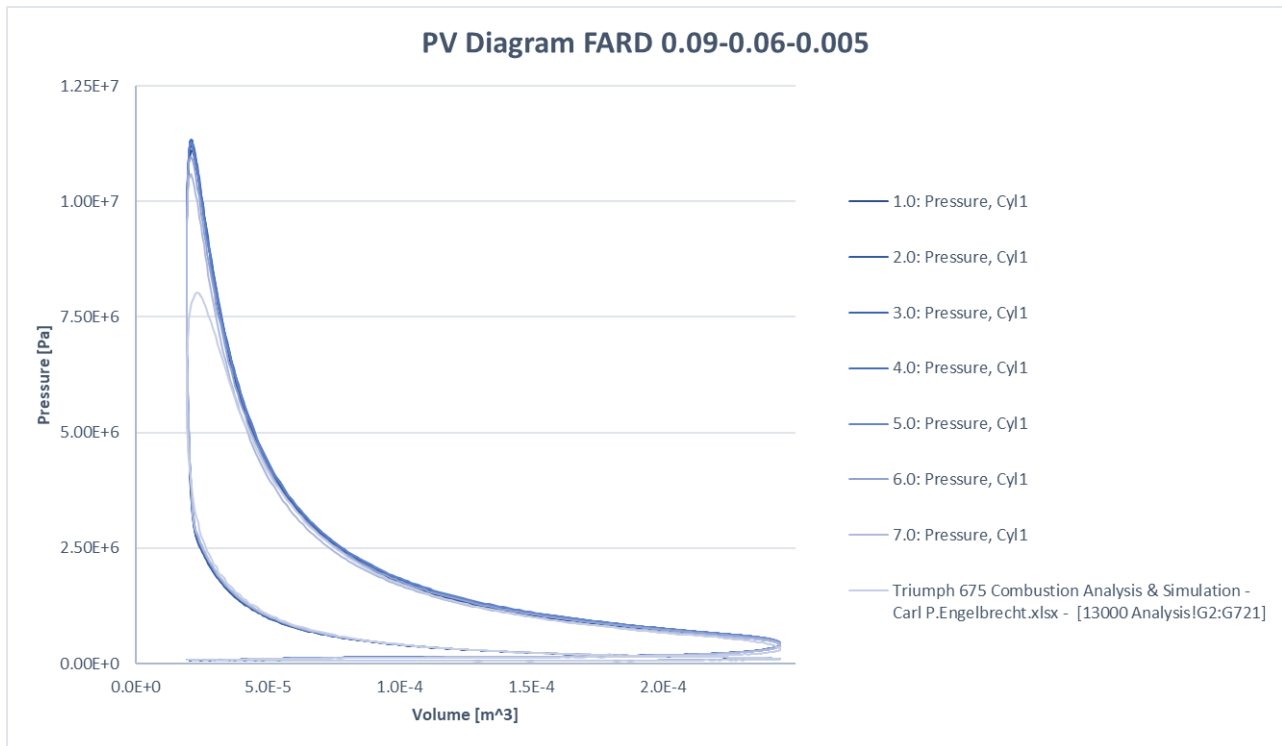


Figure 19 - Fuel Air Ratio (FARD) sweep from 0.09 to 0.06. the latter being lean and the former being rich , either side of stoichiometric at 0.068.

1.3.6.3 Valve Timing

As mentioned previously the inlet valve timing controls the ignition timing in effect, showcased through the affect it has on peak pressures as well as the expansion curve. The anchor within the valve timing window is changed to the maximum opening point of the valve, hence the values ranging from 400 to 500. The gradients before TDC vary greatly according to the timing.

On the other hand, the sweep done on the exhaust timing showed no affect on the expansion stroke BTDC but rather had an effect on the end of the expansion curve. As both inlet and exhaust timing will be done in conjunction to on another the affect of EOP will be very insignificant compared to that of EOP.

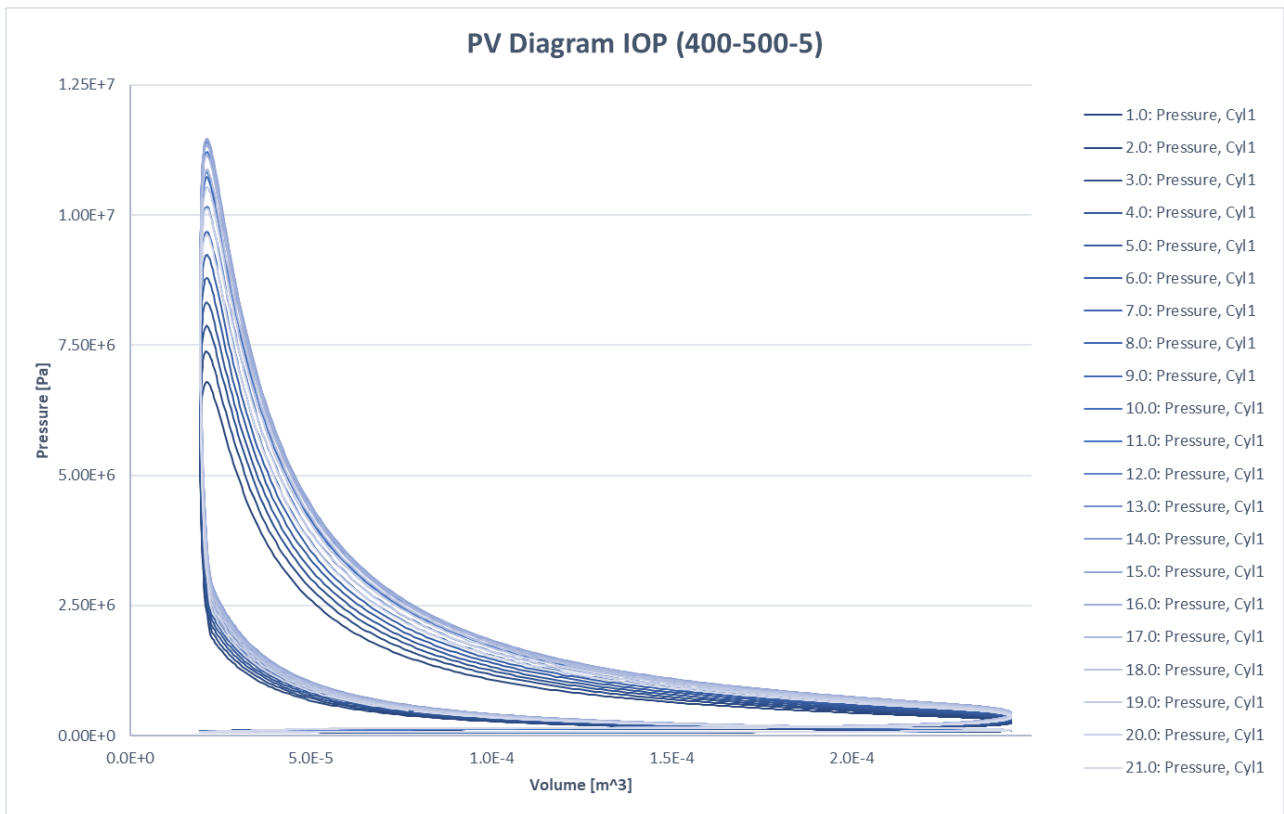


Figure 20 - Inlet valve opening timing sweep.

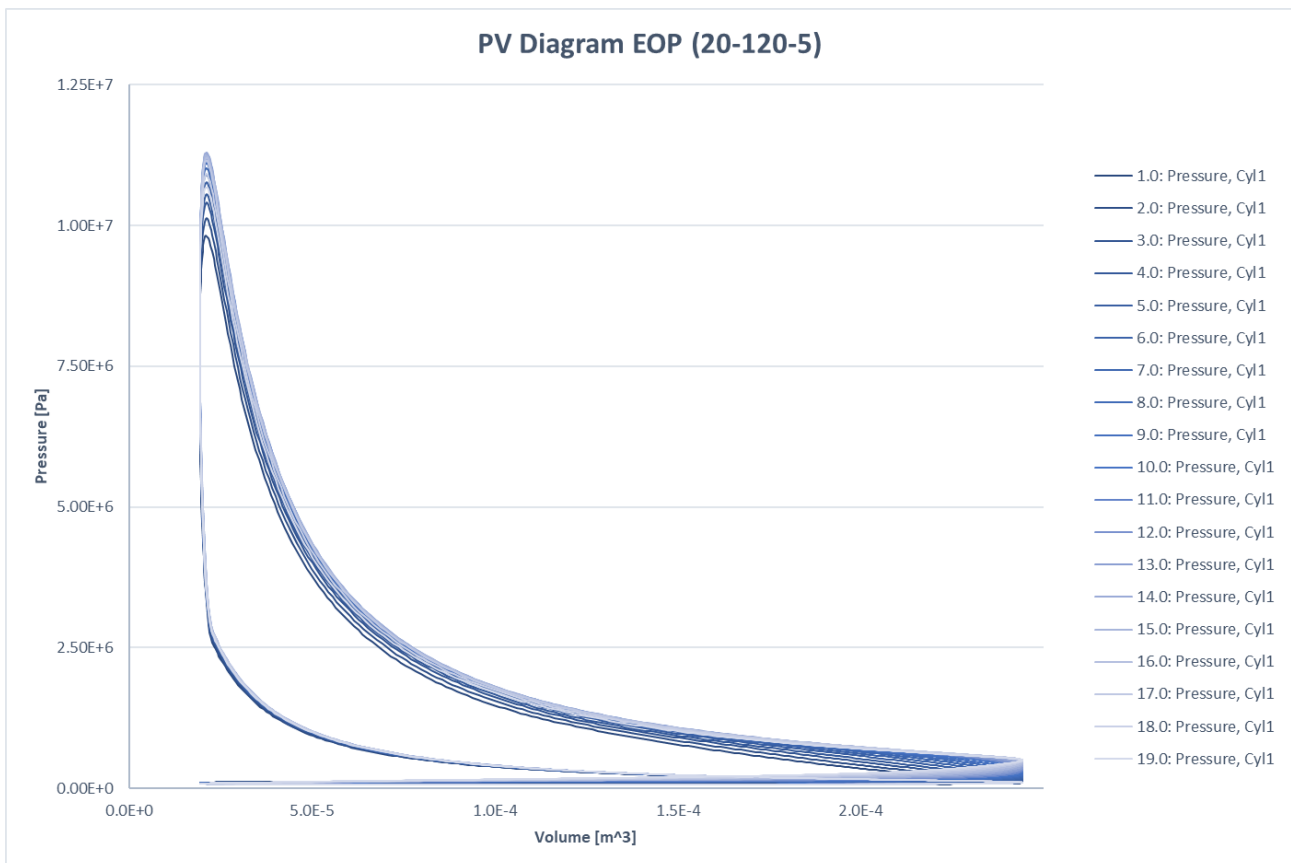


Figure 21 - Exhaust valve opening timing sweep.

1.3.6.4 *Woschni*

The energy entering the cylinder in the form of fuel leaves as mechanical work, exhaust gas and heat transfer through the cylinder liner, piston and other components. (Woschni, 1965) The Woschni heat transfer model emulates this within the WAVE model. The available inputs are heat transfer multipliers, as seen in Figure 22 - The Woschni heat transfer model featured within WAVE.

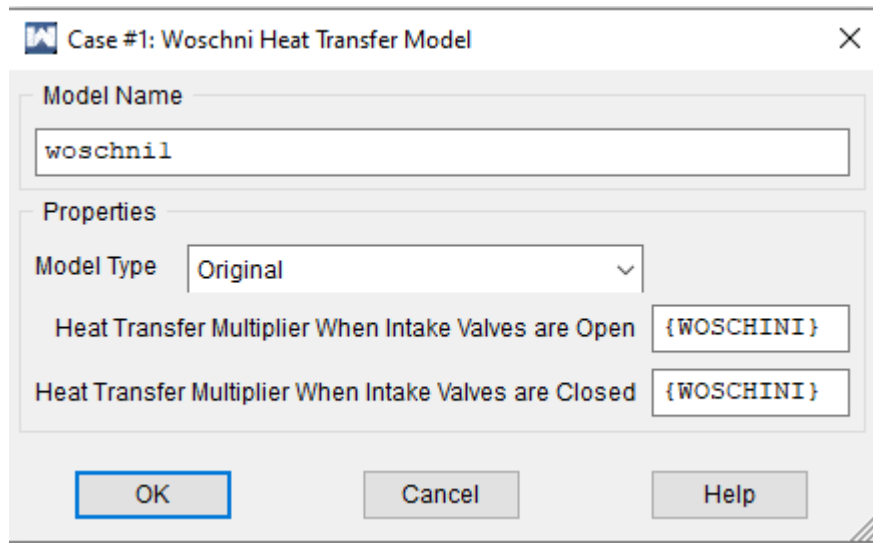


Figure 22 - The Woschni heat transfer model featured within WAVE.

An increase in the heat transfer multiplier in both cases lowers the peak combustion pressures and decreases the gradient during expansion. This is expected as a greater amount of heat transfer means more heat energy is lost into the surroundings.

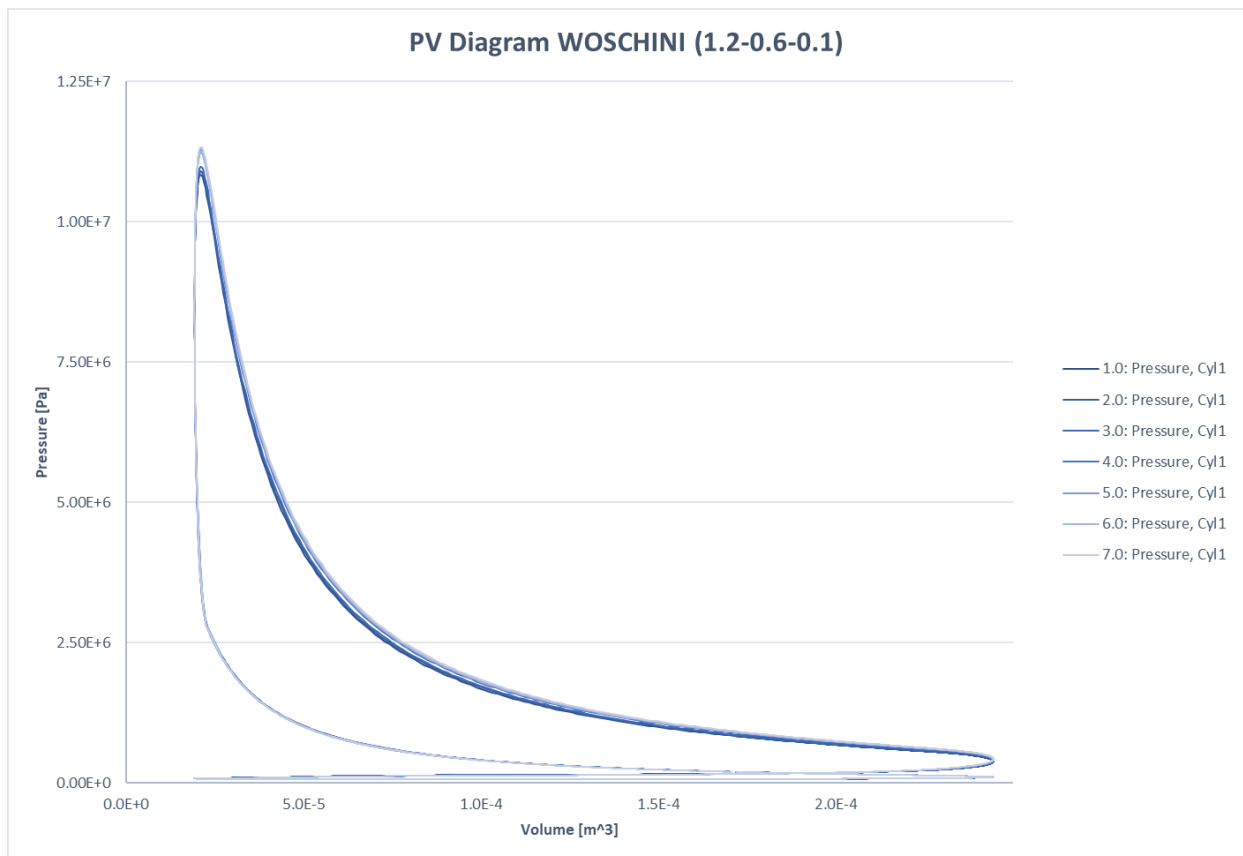


Figure 23 - Woschini variable sweep from 1.2 to 0.6.

1.4 Final Optimisations

Understanding the impact each variable on the curve from the previous experiments allow the curves of each engine to be fit through experimentation to that of the real engine. The constant from the last iteration run is taken and numerous runs are done altering different constant in order to match the PV diagram.

The following are the ideal constants for each engine speed in order to best fit the P-V curve of the real engine.

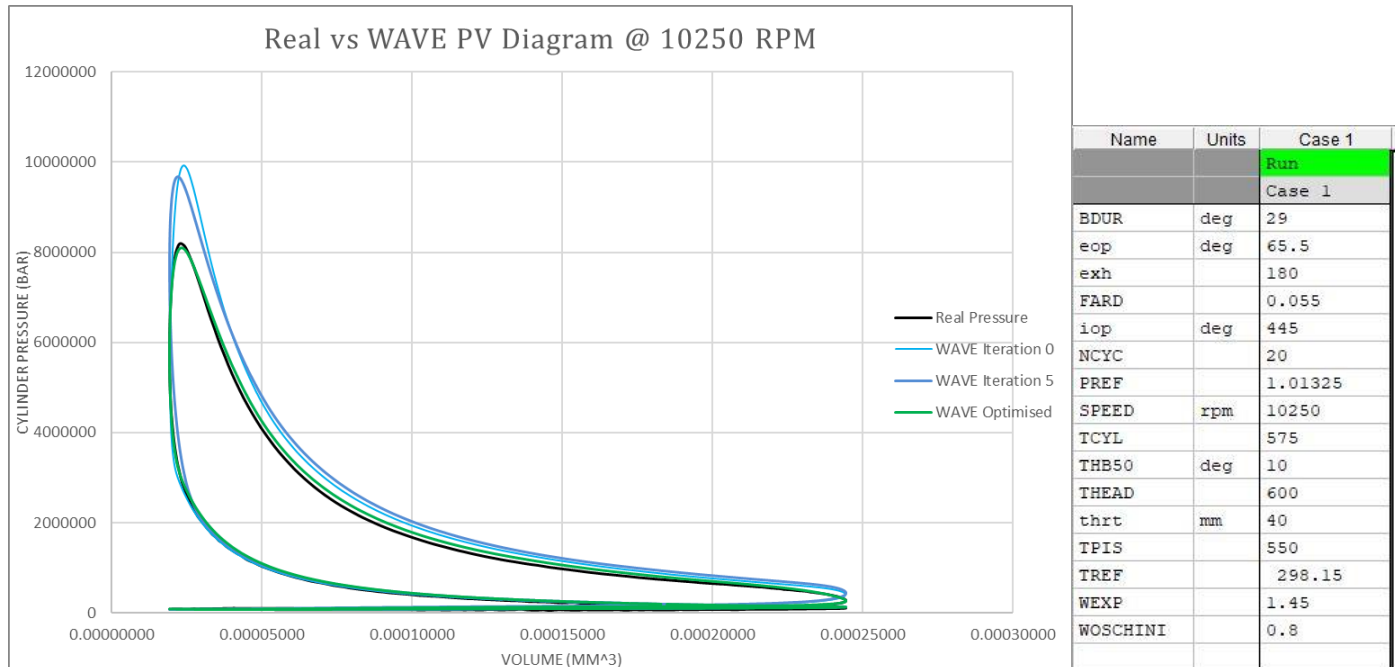


Figure 24 - Optimised WAVE constants for 10250 rpm compared to that of the real data.

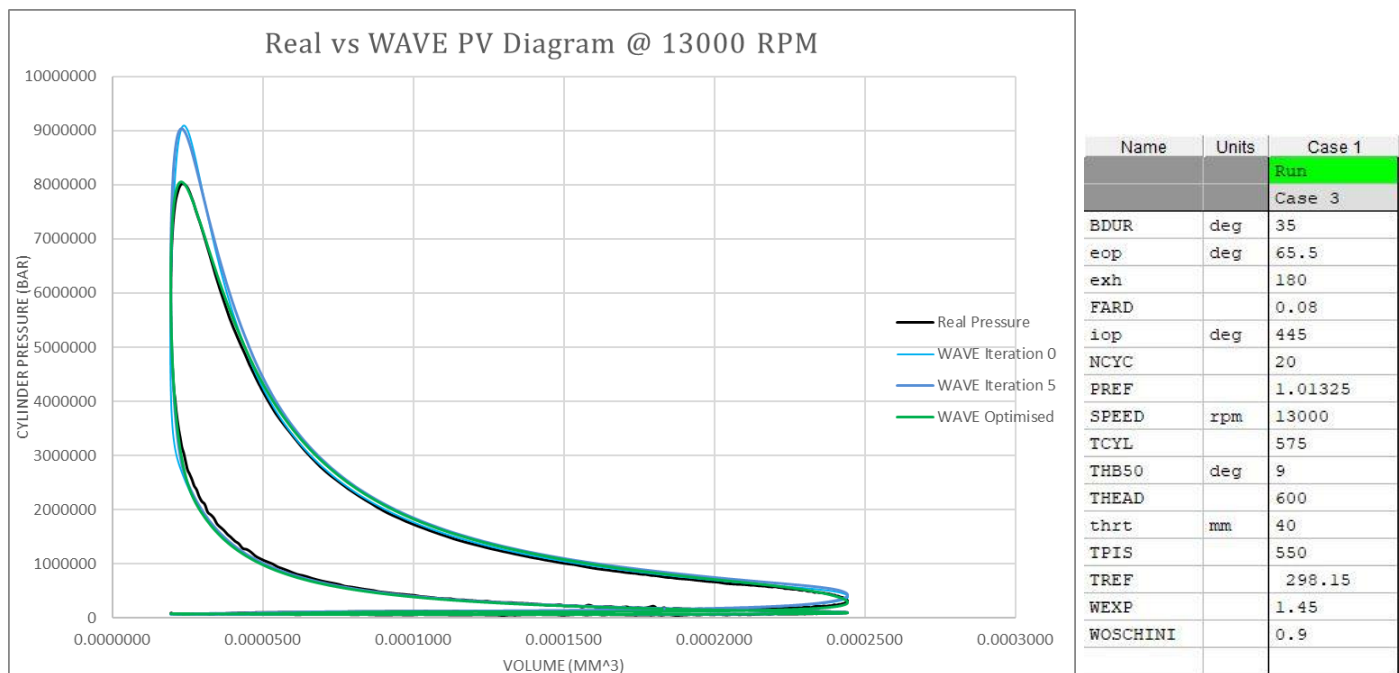


Figure 25 - Optimised WAVE constants for 13000 rpm matching that of the real data.

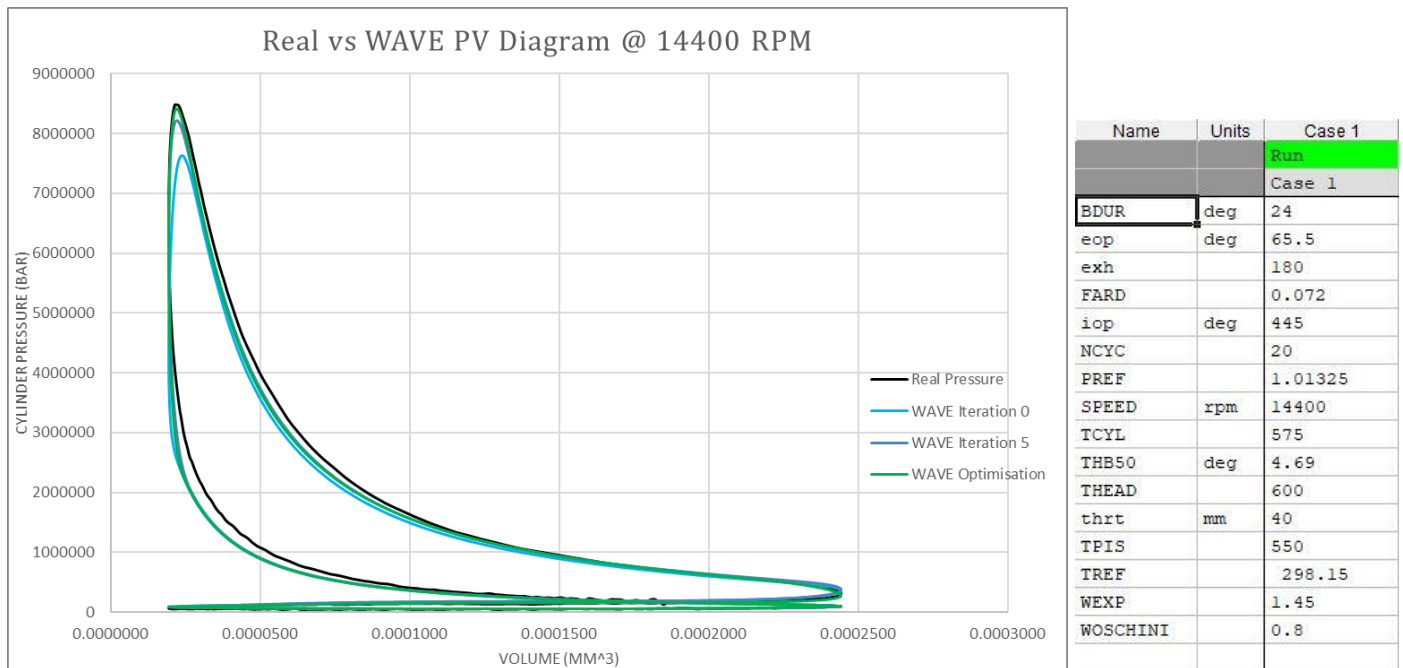


Figure 26 Figure 24 - Optimised WAVE constants for 14400 rpm compared to that of the real data.

Conclusion

One dimensional fluid dynamics simulation has proven to be a useful tool in running rapid engine simulation featuring numerous cases. Allowing for valuable engine prediction before production modifications, an affordable solution. In this case it has been successfully used to evaluate a Triumph 675 engine WAVE model against real combustion data.

Initially a combustion analysis was undertaken in Matlab to find the essential data inputs for the SI Wiebe model. Along side the combustion analysis general parameters concerning the engines mechanical constraints were found. The initial iterative process produced promising results for 14400 rpm however did not come close to matching that of the other speeds.

Thus, through an experimental process using the final iterations' constants, variables were altered one at a time using a sweep method. This led to the discovery of the large impact burn duration and its position at 50% burn had on the shape of the resultant PV diagrams. This was also true for inlet valve opening degree and FARD. Interestingly it was found that the Wiebe exponent had little effect on the final shape of the PV diagram.

Through these initial iterations and experiments conclusive constants that produced that output data to fit that of the real engine were developed, achieving the aim of this investigation. However, a few coherent short comings are evident. Firstly, the lack of a physical test element meant all data had to be assumed correct. Incoherent model and engine specifications were present as the year and model was not verified. Therefore, a mismatch to the combustion data may be present. Overall a definitive understanding of the Ricardo WAVE package is developed and implemented.

References

- aperaceparts.com, 2011. *2011 Triumph Daytona 675R*. [Online]
Available at: <https://www.aperaceparts.com/tech/2011triumphdaytona675r.html>
[Accessed 16 January 2020].
- Cordon, D., Dean, C., Steciak, J. & Beyerlein, S., 2007. *ONE-DIMENSIONAL ENGINE MODELING*, Idaho: National Institute for Advanced Transportation Technology.
- Hughes, M., 2016. *Ferrari's Formula 1 jet ignition*. [Online]
Available at: <https://www.motorsportmagazine.com/opinion/f1/ferraris-formula-1-jet-ignition>
[Accessed 07 October 2019].
- Ricardo , 2019. *WAVE*. [Online]
Available at: <https://software.ricardo.com/products/wave>
[Accessed 16 January 2020].
- Woschni, G., 1965. *Computer Programs to Determine the Relationship Between Pressure Flow, Heat Release, and Thermal Load in Diesel Engines*, Augsburg: SAE International.
- www.motorcyclespecs.co.za, 2009. *Triumph Daytona 675*. [Online]
Available at: https://www.motorcyclespecs.co.za/model/triu/triumph_daytona_675%2009.htm
[Accessed 25 January 2020].

Appendix

Engine	0.675025217	Litre
Cylinders	3	
Swept Volume (+Base Gasket)	0.000225008	225.00840567247 cc
Bore	0.076	Metre
Stroke	0.0496	Metre
Conrod Length	0.1046	Metre
Crank Radius	0.0248	Metre
Head Gasket Thickness	0	Metre
Base Gasket Thickness	0	Metre
Inlet Valve Diametre	0.031	Metre
Inlet Valve Area	0.000754768	
Number of Valves	2	
Inlet:Bore Area Ratio	3.005202914	
Distance Piston to Trumpet	0.2	Metre
Clearance Volume (+Head Gasket)	1.92315E-05	Metre ³
Compression Ratio	12.7000000	12.7
S-Swept	0.1294000	
Lambda	0.9000000	1
Air Temp	20	Celcius
Air Density	1.167	Kg/m ³
Air Pressure	1	Bar
Calorific Fuel Value	44000000	J/Kg
Air/Fuel Ratio	12	12 calculated
Vol Eff	0.9	
Com Eff	0.85	
nCom	1.8	
nExp	1.15	
R	286.9	J/Kg.K

Qr	0	J
Word Done	0	J
RPM	10250	1073.37748997651 rad/s
Power	0	J/s
BHP	0	
Specific Heat Ration-γ	1.25	
Wiebe a	5	
Wiebe m	2	
Ignition Crankshaft Degree	-20	Degree
Ignition Delay Degree	8	Degree
Burn Duration	37	Degree
Crankshaft Burn Duration	205	Degree

Table 3 - The data table to accompanying the wave model given.

Q

Air Mass per Cylinder	0.00023632 6	Kg
Fuel Mass per Cylinder	1.96939E-05	Kg
Heat Output per Cylinder	736.550390 4	Joules
ΔT	2876.92307 7	Kelvin

Table 4 - Addition initial data given with the wave model.

$$\text{Clearance Volume} = \frac{\frac{\text{Displacement}}{\text{No. of Cylinders}}}{\text{Compression Ratio} - 1} \quad \text{cc}$$

Displacement of engine (cc)

Equation 4 - Clearance volume equation.

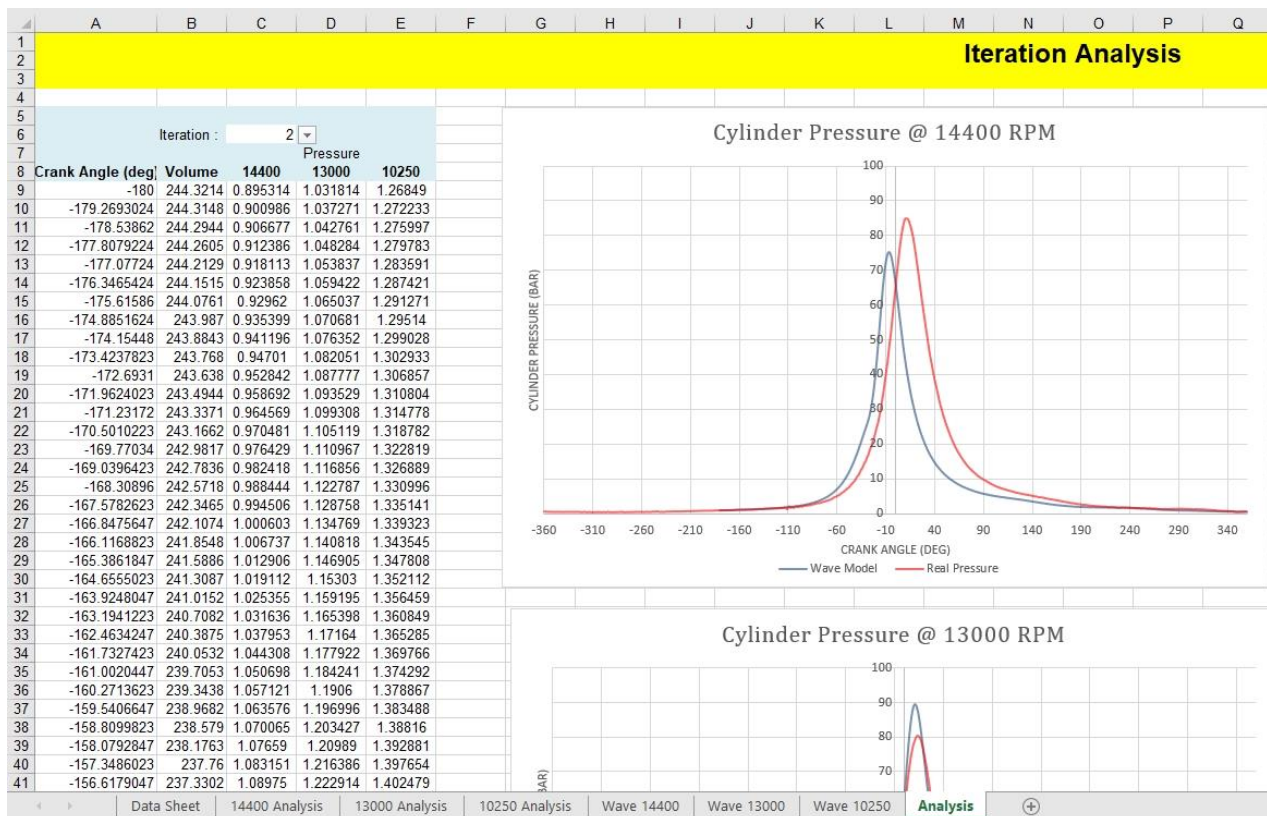


Figure 27 - Iteration analysis window within Microsoft Excel.

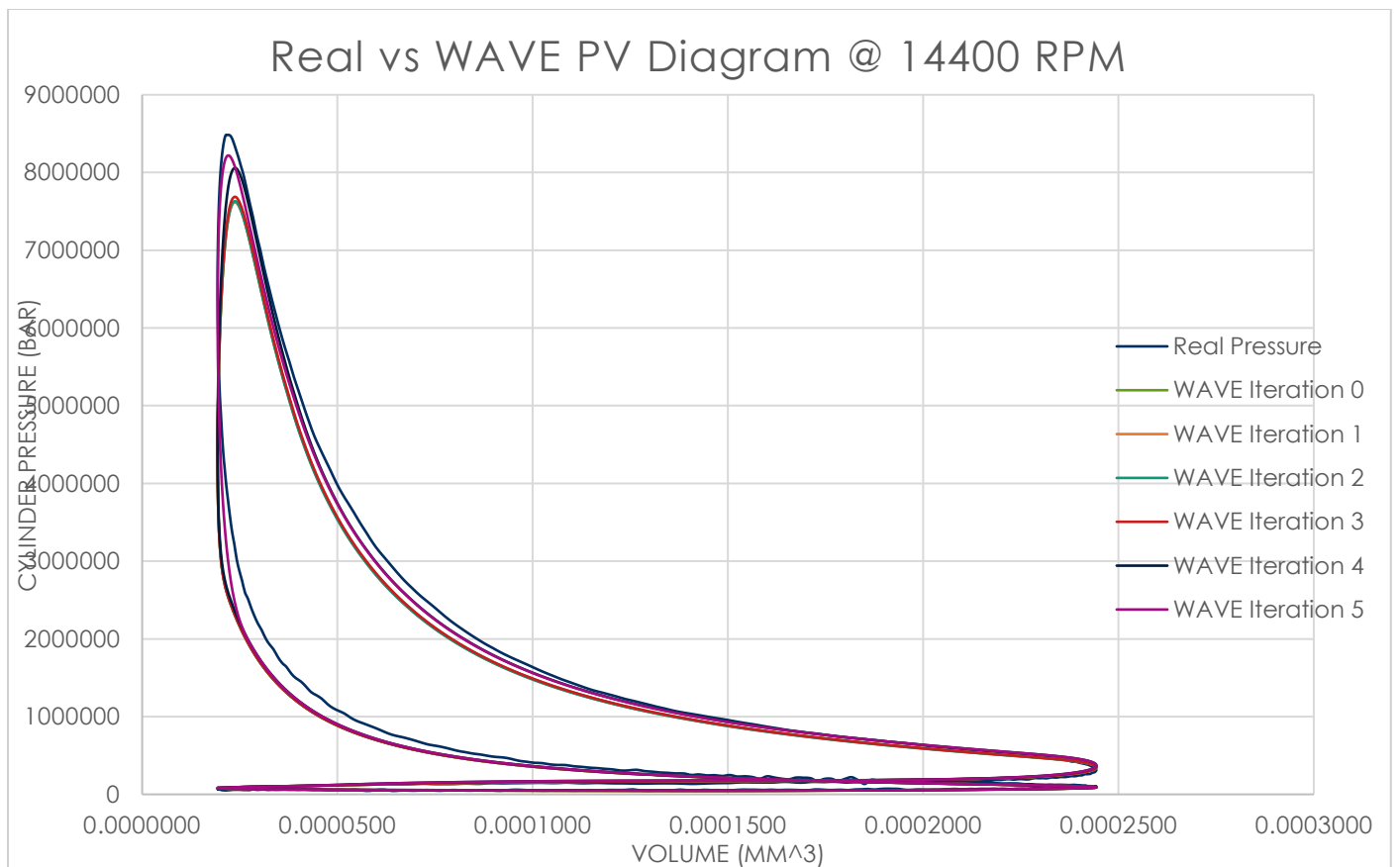


Figure 28 - Iterational differences present seen at 14400 rpm.