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Title of the Coursework

Optimisation of a high speed 3 cylinder using the Ricardo
WAVE engine development software

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Nomenclature

AFR	Air to fuel ratio by mass.
BDC	Bottom dead centre – Crank angle is 180°.
BMEP	Brake Mean Effective Pressure.
CA	Crank angle.
FMEP	Frictional Mean Effective Pressure.
HRR	Heat Release Rate.
ICE	Internal combustion engine.
IMEP	Indicated Mean Effective Pressure.
Q	Heat added into the system.
MBT	Maximum Brake Torque associated with ignition timing.
TDC	Top dead centre – Crank angle is 0°.

1. Introduction.

The purpose of this paper is to outline how interpreting simple data, such as; Cylinder pressures, mass fraction burnt and the crankshaft position, aids in constructing a realistic model to simulate combustion.

The development of the simulation was done using Ricardo's WAVE software. The base model of a Triumph 675cc 3 Cylinder ICE had been provided and required adjustment to improve the predications into more realistic values. Using interpolation techniques and generic equations regarding internal combustion engine construction, the fundamental values governing combustion were found and implemented into the model. The effect of this was then analysed and the implications discussed.

Because combustion simulation concerns many variables and therefore proves difficult to match data gathered in real-world testing, this paper will discuss the key factors that had demonstrated a major role, their exact impacts to the model and then the downfalls of using the idealised equations.

Objectives –

- Develop the data provided into a format that could then be used to obtain engine characteristics.
- Using an appropriate software, interpolate the MFB data via a “curve fitting” method to find the required burn duration [10 – 90%] and the 50% burn point.
- Asses the Ricardo WAVE model and correct any issues found within.
- Adjust the model to correlate with the “real” values governing the combustion within the Triumph Daytona 675cc ICE.
- Configure the model to be realistic in relation to the provided data.
- Outline the validity of the model and complications within simulation.

2. Interpretation of Triumph Data.

The data provided from Triumph Motorcycles required interpretation to enable the return of key data and this was done using both Microsoft Excel & MATLAB, just to check validation of values found when developing a curve fit.

1. Cylinder Pressure.

The cylinder pressures provided were representative of gauge pressure and therefore many points were shown as a negative value, simply because the zero point on the gauge is standard atmospheric pressure of 1 Bar, hence the addition of one bar to correct this.

Using the corrected cylinder pressures, and the graphical representations of both the standard pressure against crank angle, along with the change in pressure through the cycle against crank angle [Appendix A], aided the decision for the positions of the start and end of combustion for each of the three provided engine speeds.

The cylinder pressures were also used to create a Logarithmic version of pressure against volume, again to aid in cross referencing and comparison, along with finding values for the ratio of specific heats under compression and expansion.

2. Volume.

The volume within the cylinder at each degree of crank was provided as a percentage in respect to swept volume, with 0% being TDC and 100% occurring at BDC. This required transposition into a useable format, via an equation shown as;

$$V = \left(\frac{\text{Percentage of Volume}}{100} \cdot V_s \right) + (Cv)$$

With V being the volume within the cylinder, V_s being the swept volume and Cv being the clearance volume of the engine. The Swept and clearance were already known values and therefore were used as a validation test to prove the equation.

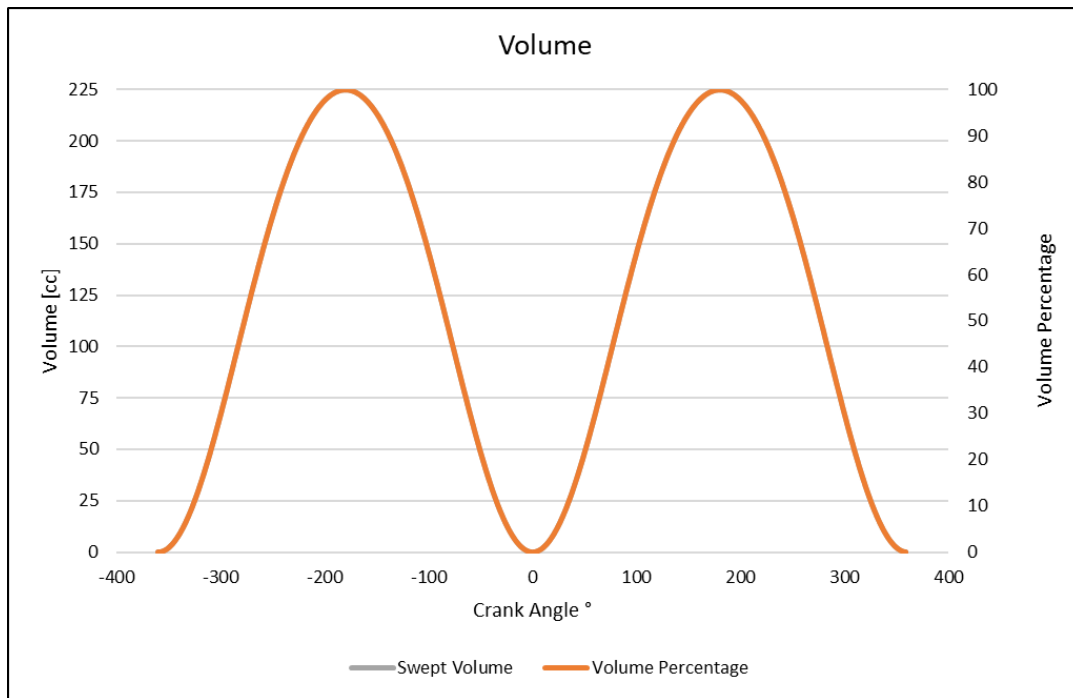


Figure 1 Correlation of the Swept Volume and the given Volume Percentage.

3. Mass Fraction Burnt.

The mass fraction of fuel burnt in the cylinder is the most important factor in predicting and simulating combustion. Using the given trends from the provided data, it seemed to be that measurement of the MFB was started and ended at the same point for each of the engine speeds. Using this plotted against the crank angle position, shows how the burn propagates through the cylinder and the piston moves back down towards BDC.

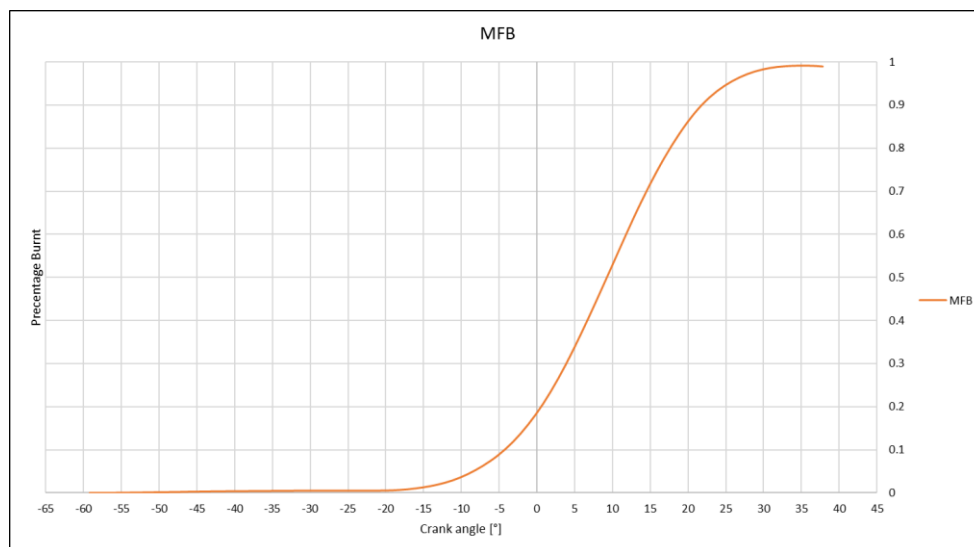


Figure 2 Mass fraction burnt for 10250RPM from given data.

As can be seen in Figure 2, there is a substantial proportion of the data that is essentially useless, being from -60 degrees of crank angle up to around the -25 degrees point. Therefore, it's disregard when using a curve fitting tool to help decide the points to be used for informing Ricardo WAVE, i.e. 10%, 50% and 90% burnt.

1. Excel Curve fitting.

Using Microsoft excel, there is a provided trendline function that contains a polynomial operator that will return an equation that can be used to find values on the Y-axis when using the same X-axis as plotted.

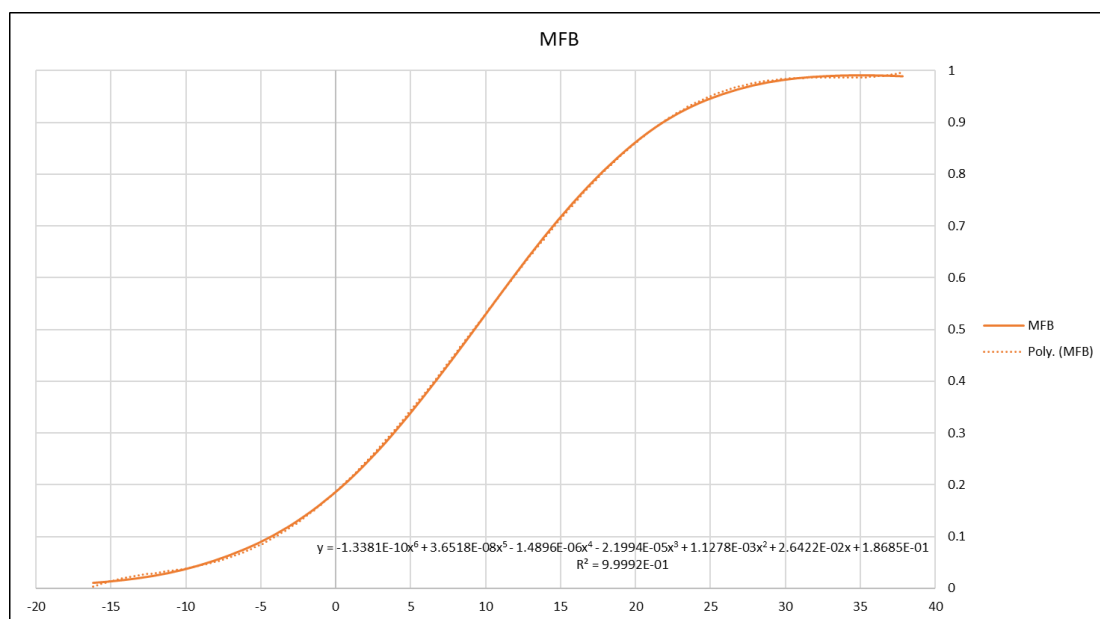


Figure 3 MFB with polynomial trend line and the generated equation.

As can be seen in figure 4, the trendline matches the actual with a rather high degree of accuracy, but there is very clearly discrepancies throughout. Which is one issue with the excel tool, it's limited to the order of fit [6th order max], which hasn't reached the exact fit, proven by the R² equation not being 1 or as close to.

However, using the curve fit equation is simple, export it out with all exponents being to the 15th decimal place to increase accuracy (Again another potential carry through issue with excel if the decimal point isn't enough), and then using it with Y being the MFB and X being the crank angle, it's easy to return values of MFB within a vector of crank angel that is much finer that the actual. In doing this, the points of 10%, 50% and 90% can easily be found.

2. MATLAB Curve fitting.

Using the same principles for the setup of generating a curve fit trend line, the procedure was conducted within MATLAB's curve fitting tool also ("cftool"), primarily for comparison, but also a higher degree of accuracy.

Within MATLAB there is many more functions of curve fitting available, but most importantly is the ability to fit to the 9th order instead of 6th. As can be seen the fitted curve is essentially identical to that of the real;

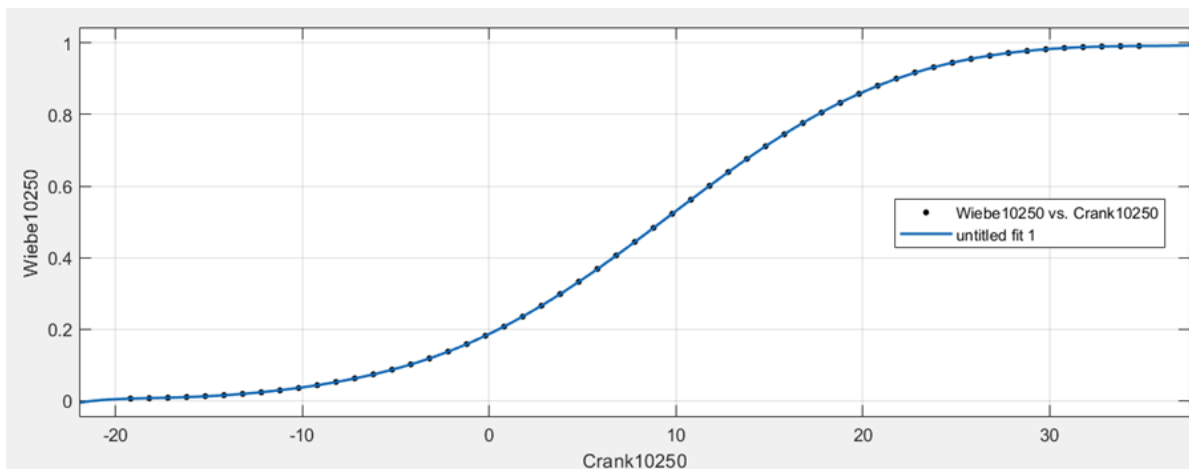


Figure 4 MATLAB curve fitting of MFB for 10250RPM.

Therefor using the equation generated within MATLAB was far more accurate [$R^2 = 1$], and the software also enables much easier manipulation of data.

4. Wiebe Function.

To predict how the combustion process releases heat within an ICE, the function known as Wiebe is to be used. It is shown as;

$$x_b = 1 - \exp \left[-a \left(\frac{\theta - \theta_0}{\Delta\theta} \right)^{m+1} \right]$$

Equation 1 Wiebe Function (Valecha, Mishra, Subbarao, PMV., & Das, 2018).

With the exponents within, “a” & “m”, being the values that characterise the curve it will predict. Then the crank angles concerned being ‘ θ ’ as current crank angle, ‘ θ_0 ’ being the start of combustion and the ‘ $\Delta\theta$ ’ as the combustion duration.

As stated, the values for start of and duration of combustion were chosen via the logarithmic pressure and volume graphs, along with the difference in pressures across the crank angle. It’s understood that “The start of combustion can be identified by the departure of the curve from the straight line” (Heywood J. B., 1988), this is accepted because the straight line areas at which compression or expansion are happening, behave as if they’re adiabatic isentropic processes. With this process defined as $PV^n = \text{constant}$, the increase in pressure and volume is constant, hence the linear increase. So, when combustion happens, this process no longer occurs and leads to the deviation of the curve from the line. As shown in;

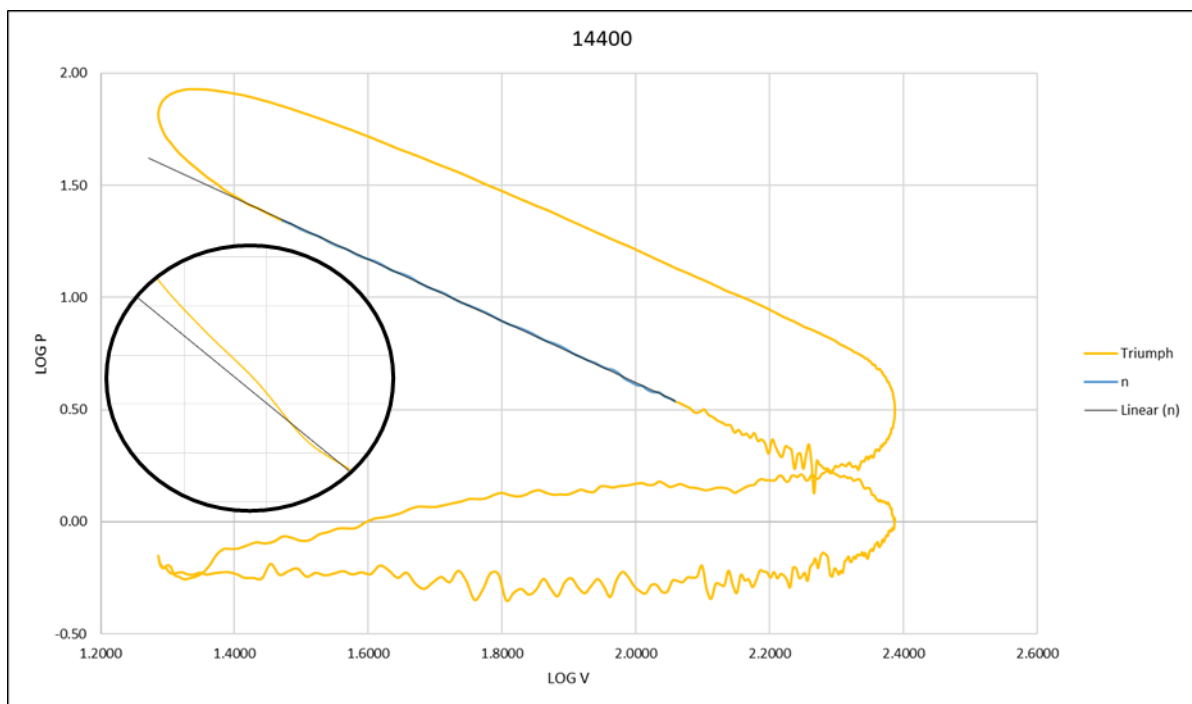


Figure 5 Change in Logarithmic Pressure Vs Volume values, depicting the start of combustion.

The duration can then be found by using the maximum value from the provided data of MFB minus the start of combustion. The values that were chosen for the study can be seen below;

MATLAB				
	a	m	θ	$\Delta\theta$
10250	5.39	1.805	-16	50.81
13000	5.052	2.893	-16.82	51.31
14400	Unable to fit.			

Table 1 MATLAB Curve fitting results for the Wiebe function.

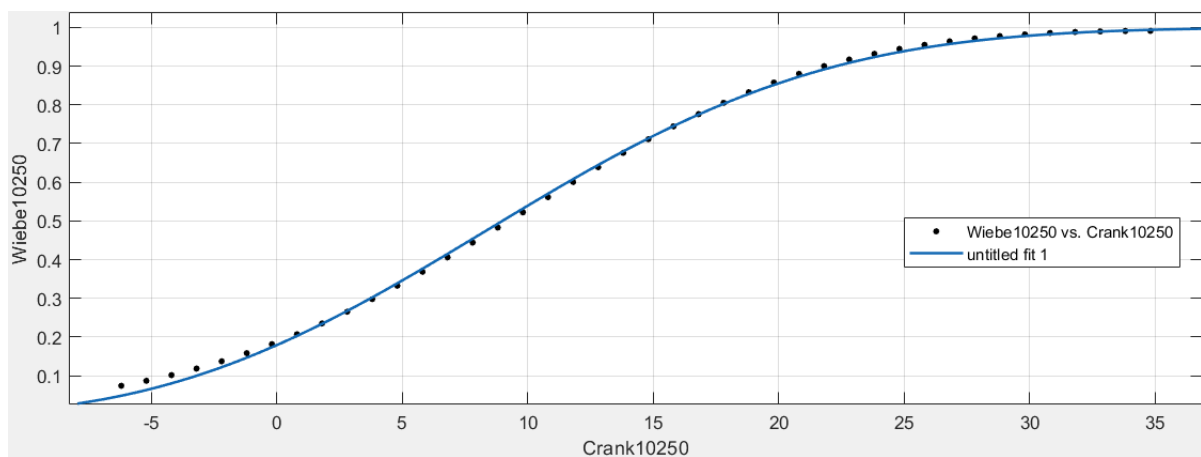


Figure 6 MATLAB Custom equation curve fitting for Wiebe function exponents.

Excel				
	a	m	θ	$\Delta\theta$
10250	5.55	2.00	-16	50.81
13000	5.1	2.89	-16.82	51.31
14400	5.23	3.07	-17.2	56.5

Table 2 Excel Curve fitting results for the Wiebe function.

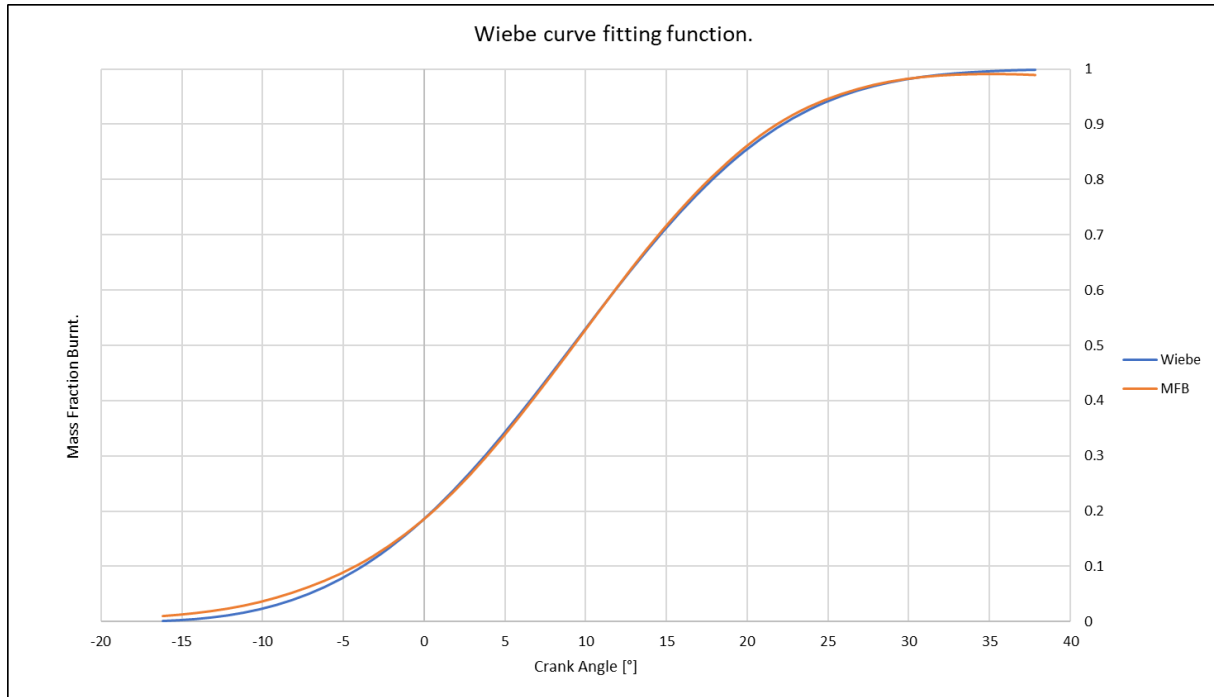


Figure 7 Excel Wiebe function curve fitting against the real MFB for 10250RPM.

The exponent values within the Wiebe function are somewhat disputed by various sources. For example, John. B. Heywood mentions that within his own studies it was found that “a” of 5, “m” of 2, can fit the curve to a satisfactory standard. However, a journal article written by Yeliana et al. suggests there to be two different set values concerning “a”, dependent of percentage of MFB, 0 – 90% “a” fixed at 2.3026 and 0 – 99% is fixed at 6.9078 (Yeliana, Cooney, Worm, Michalek, & Naber, 2008). These results are supported by both Valecha et al. and Merker et al. with a small table (3) describing the percentage of MFB ($n_{conv,total}$), and the associated value of “a”.

$\eta_{conv,total}$	0.999	0.990	0.980	0.950
a	6.908	4.605	3.912	2.995

Table 3 Table presenting values for the exponent “a” used within the Wiebe function (Merker, Schwarz, Stiesch, & Otto, 2006).

With these values being supported by multiple sources, it was taken as a form of validation when considering the used values found in excel, because the given MFB EOC, differed around the mark of 98 – 99.3% burn. Hence the variation within the exponents. (Refer to appendix C to view the effect of “m”, presented by Merker et al.)

5. Burn rate.

The accuracy of modelling the mass fraction burnt is vital when simulating combustion because it then governs the rate of which the fuel mass is burnt. This burn rate will then be used alongside the AFR for the total fuel mass trapped within the cylinder and the heat of combustion to result in the equation;

$$HHR = - \frac{d[fuel]}{dt} \cdot M_{fuel} \cdot Q_c$$

Equation 2 Heat release rate equation (McAllister, Chen, & Fernandez-Pello, 2011)

The heat release is what governs the “power” of the combustion system.

Using the Wiebe function and the actual data to obtain the burn rates for the different engine speeds proved difficult, when trying to develop a definite correlation because of the linear limitations within the prediction, in comparison to the real-life data. As can be seen within the following;

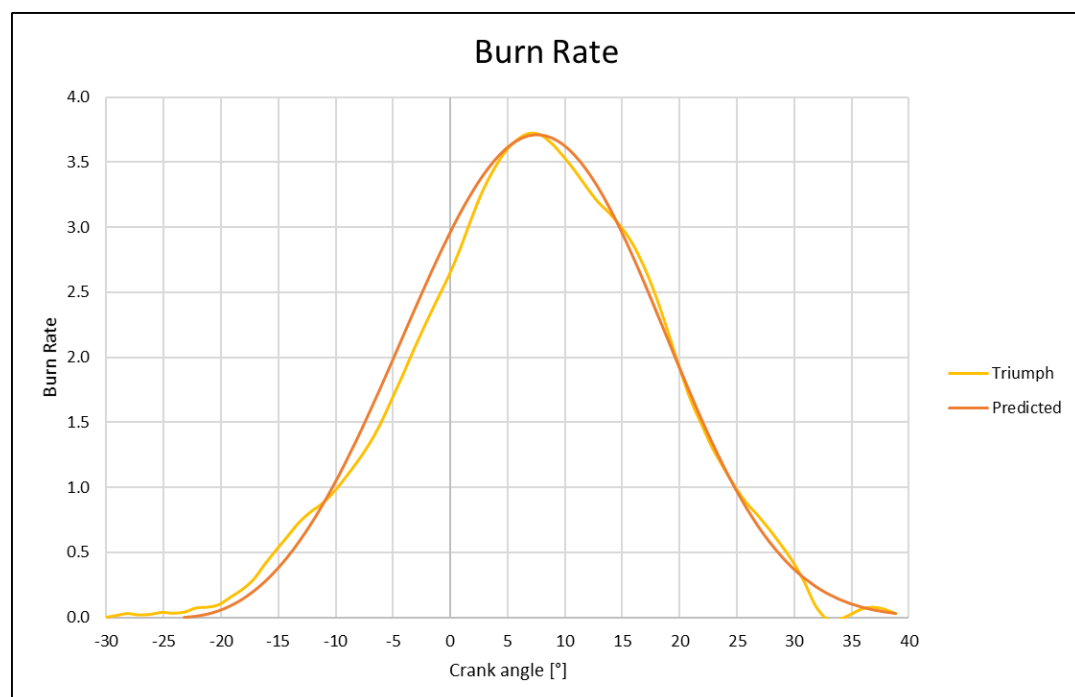


Figure 8 Predicted Vs Real burn rates for 14400RPM.

The figure is clearly demonstrating the potential limitations within the mathematical modelling approach of the Wiebe function because of its linearity; however, the trend has most definitely been predicted, albeit with varying discrepancies and the max burn rate is in the correct position along with the duration.

3. Ricardo Wave Modelling.

1. Model Assessment.

The model of the Triumph Daytona 675cc three-cylinder motorcycle provided contained a few issues throughout the construction itself. Using the inbuilt function for run checks returned five different model geometry points that had what was believed to be “issues”. After correcting these, by either adjusting the opening sizes or simply recalculating the concerned area, these issues disappeared.

The model also requires inputs to be able to run at all, i.e. the profile of valve lift for both the inlet and exhaust, along with their associated flow coefficient profiles for the respective lift. All of which was provided in the correct format to simply load in [appendix D].

There was only one engine geometry change to be made, the connecting rod length. Data provided stated it as 104.6mm and with changing this, it should have reduced the effect of frictional losses of the piston against the cylinder wall lining. It's understood that the shorter the connecting rod, the higher the angularity across the crank angle, creating larger side loads on the piston and “for a given coefficient of friction, a higher load means a higher drag on the piston and increased frictional losses.” (Ward, 2012).

2. Variable Assignment.

Ricardo's WAVE software has been designed with the known reality that more than one engine is going to be studied. Hence, they've incorporated the ability to assign variable names to the major components within the model, due to the reality of many factors changing at each engine speed;

- Ambient temperatures and pressures.
- Different reference temperatures and pressures across the intake, exhaust and within the cylinder.
- Variable AFR's for the injectors.
- Internal temperatures of the cylinder; Piston top, wall lining and cylinder head.
- Wiebe-function exponents, burn durations and the point of 50% MFB.

All of which are done within the constants table and then simply assign the appropriate name where applicable.

3. Noticeable Impacts.

Upon experimenting with the programme, there were many variables that seemingly had a negligible effect on the outcome pressure and effective torque/power, i.e. various temperature ranges down the exhaust sector, the discretization lengths used across the components and original reference temperatures which seems odd given the fundamentals characterised by the ideal gas law.

The model uses the Woschni method type to determine the heat transfer within the cylinder (originally proposed for diesel cycle operation and not the “Otto” based spark ignition petroleum engines). Major impacts were shown by changing the initial pressures the system was experiencing (intake and exhaust), thought to be because of the development using fluid dynamics numbers; Nusselt, Prandtl and the Reynolds numbers. The variation across the cylinder surroundings inputted temperatures made an impact on the results predicted. This is believed to be because of the fixed values for temperature across the combustion cycle, effecting the initial pressures and change in temperature within the system. As well as the heat transfer between the surroundings being limited by the variable values, which were simply speculated and may not have been realistic. Heywood suggests the combustion chamber surface temperatures to be around 200 – 400 °C, which can potentially reduce the heat energy of the working fluid. Again, this is dependent upon the materials used, each of which would have a different thermal conductivity (Heywood J. B., 1988).

In addition to this, the adjustment to “correcting” the air to fuel mass ratio played a key role in the attempt to make the model more realistic. The adjustment of these values proved difficult because of the lack of real data, however there was a given supposed value for 10250RPM as lambda of 0.9 (Φ 1.11). Using the mass of fuel for stoichiometric combustion (114.23Kg/Kmol), where the air to fuel ratio is said to be around 14.7 : 1, meaning 14.7 Kg of air needed to completely combust 1Kg of petroleum (McAllister, Chen, & Fernandez-Pello, 2011).

Multiplying the value to get the effective mass at lambda 0.9 (126.8Kg/Kmol). Same principle for the air mass required to create the 13.23:1 AFR resulted in (1662Kg/Kmol). The required input was a fuel to air mass ratio, hence the division of the fuel mass by air, returning the 0.0763 value used to represent the Phi of 1.11.

It's known that max power occurs at lambda of 0.88 (Φ 1.136) hence the assignment of this with the supposed max power engine speed of 12600RPM and then the AFR was made richer for the higher engine speeds and linearly decreased below 10250.

The finally variable the made a considerable impact in terms of refining the model to more realistic, was the definition of the variables for 50% burn, 10 - 90 % burn duration and the “m” exponent used in the Wiebe function for each individual engine speed. This can be seen in the figure below;

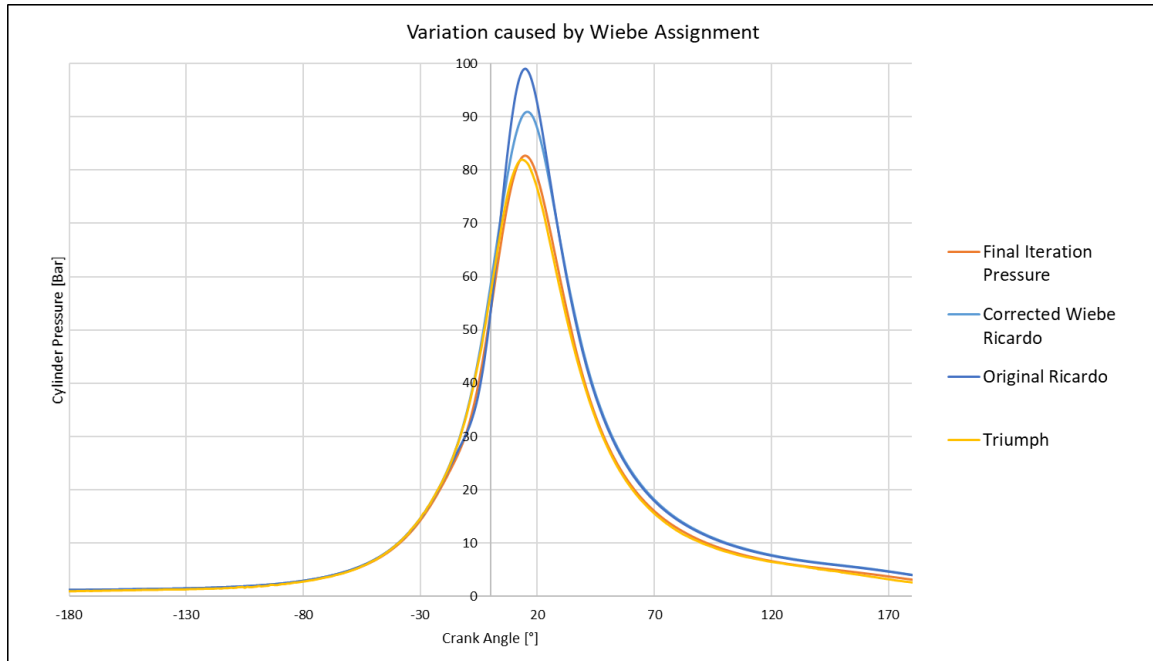


Figure 9 The differences demonstrated by changing Wiebe inputs from the original. Along with the final iteration with all model changes made in comparison to real cylinder data for 10250RPM.

As can be seen, there was a drop of around 8.5 Bar of pressure when the corrected Wiebe function and durations were used. Then the final iteration that includes all the AFR, temperature changes and the Wiebe can be seen to be in correlation with the real provided data collected by Triumph at 10250RPM.

4. Multipliers.

The reality of modelling combustion is that it's very difficult to do. There is simply far too many factors involved within real life processes of combustion, such as “engine speed; engine load; overall equivalence ratio; compression ratio; spark or injection timing; swirl and squish motion; mixture inlet temperature; coolant temperature and composition; wall material; wall deposits. Of these variables, speed and load have the greatest effect.” (Heywood J. B., 2018).

So, when trying to model it mathematically, using idealised situations, it can indefinitely result in inaccurate results. Ricardo has tried to help with adjusting and fine tuning the models results by including the ‘Multipliers’.

Which do exactly as the name suggests.

They are used to “modify” areas of concern in the model, i.e. the surface areas of the cylinders faces, frictional characteristics and many other areas, in a hope to help correlate the results with real life.

4. Comparative Analysis.

After having made the adjustments to the model and assigning seemingly appropriate values across the board, from the ambients of the model, through the intake system and down into the combustion chamber and naturally to the final exit of the exhaust system, having made changes to the associated reference temperatures and pressures, the gas temperatures and the wall temperatures. Primarily because it influences the heat transfer losses within the system, along with pursuing the task of making the model realistic.

From the final iterations of the changes, the gaps between the model and the real data can be seen below;

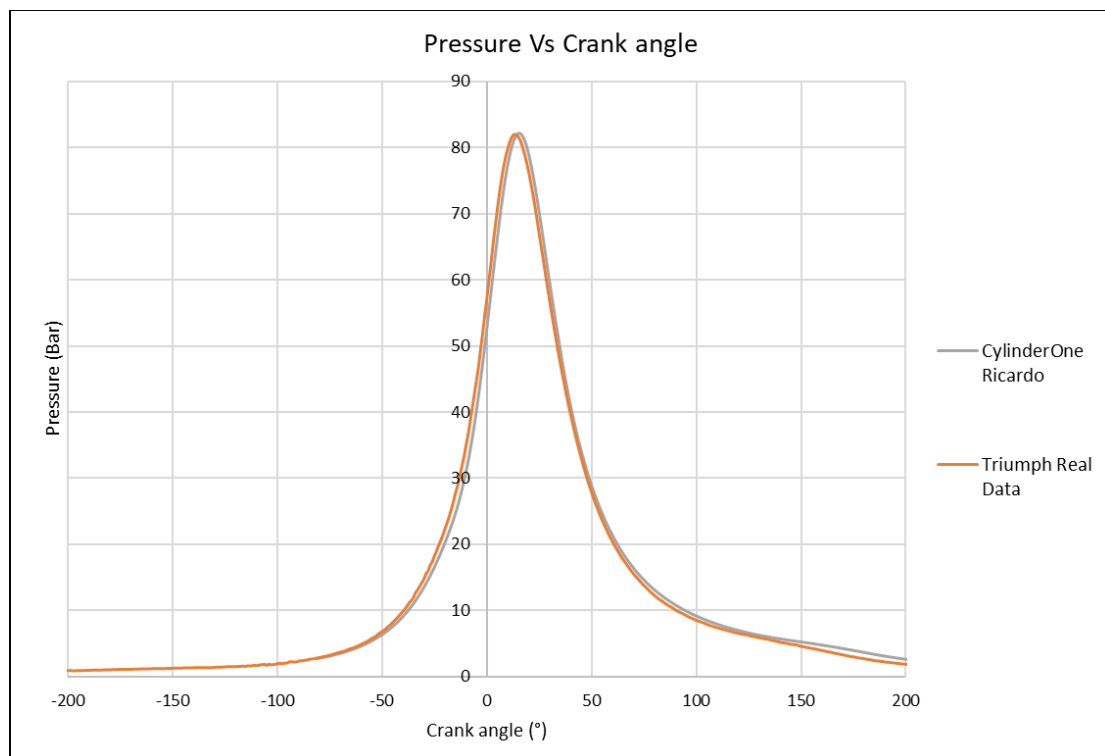


Figure 10 Pressures from the Ricardo WAVE model in comparison with the real data at 10250RPM.

The pressures within the cylinders has been averaged because there was a very slight discrepancies, as they would in the real world, due to both cyclic and cylinder to cylinder variations that occur because of the operating conditions. As can be seen, there is clearly a margin of error within the used values, for example the pressure increase happens slightly earlier and peaks earlier than that of the predicted data, suggesting that there is either an issue with deciding the point of combustion, the 50% burn location or, most likely, the burn rate for this engine speed. The burn rate could be as result of incorrect fuel/air mass ratio, errors within the Wiebe function curve fitting procedure or the combustion duration chosen for the model.

The burn rate is believed to be the cause more than anything else, as a carry through error from the Wiebe function, because when looking the logarithmic pressure against volume graph, there is an obvious issue surrounding the point at which the curve deviates from the “straight line” section of the graph, as previously discussed.

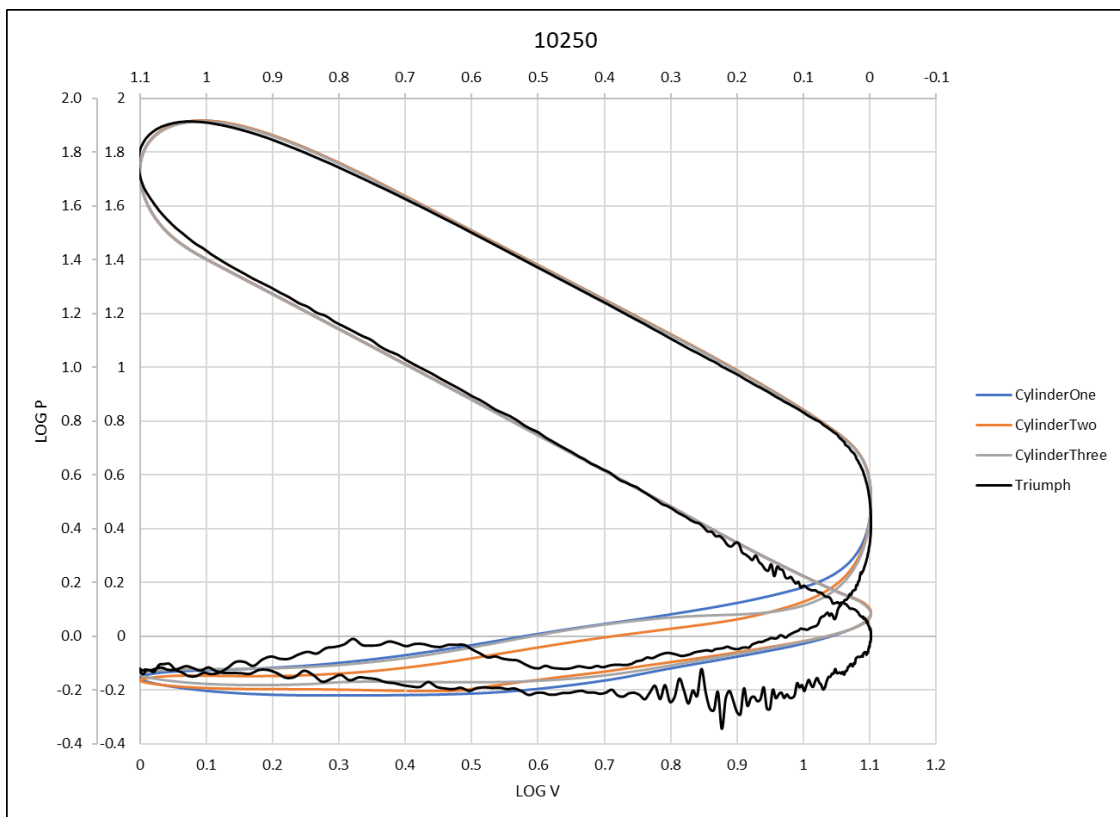


Figure 11 Logarithmic pressure Vs volume graph of the real data and the individual cylinders within the Ricardo WAVE model for 10250RPM.

This graph also proves interesting for the area that depicts the intake and exhaust phases of the cycle. The predicted values are offset by some margin from the real data, although the trend is consistent, the pressures at the volume are too high which could be leading to additional losses from pumping motions or frictional losses.

The reason for this discrepancy at these phases is unclear, but it suggests either incorrect associated values for temperature or reference pressures. But it could also be as a result of poorly correlated constants being used within the Woschni model for heat transfer coefficients, represented below.

$$w = \left[C_1 \bar{S}_p + C_2 \frac{V_d T_r}{p_r V_r} (p - p_m) \right] \quad (12.20)$$

Equation 3 Average gas velocity equation used in Woschni's correlation (Heywood J. B., 2018).

The variable C used within this equation are the constants, identified as;

For the gas exchange period:	$C_1 = 6.18,$	$C_2 = 0$
For the compression period:	$C_1 = 2.28,$	$C_2 = 0$
For the combustion and expansion period:	$C_1 = 2.28,$	$C_2 = 3.24 \times 10^{-3}$

Figure 12 Definition and corresponding value for the constants used in Woschni correlation for standard operating parameters (Heywood J. B., 2018).

However, these values change when at high engine speeds and swirl is concerned, hence the potential for a mistake within model configuration that caused the difference in these phases.

After many iterations and consistent large changes to values, the Wiebe function also began to prove a difficulty. Primarily because the measured data begin at - 60° CA, which is not true (or at least shouldn't be). Therefore, the percentage MFB began to creep up early and caused issues for trying to fit a curve with the correct ignition timing and combustion duration, demonstrated in the following figure;

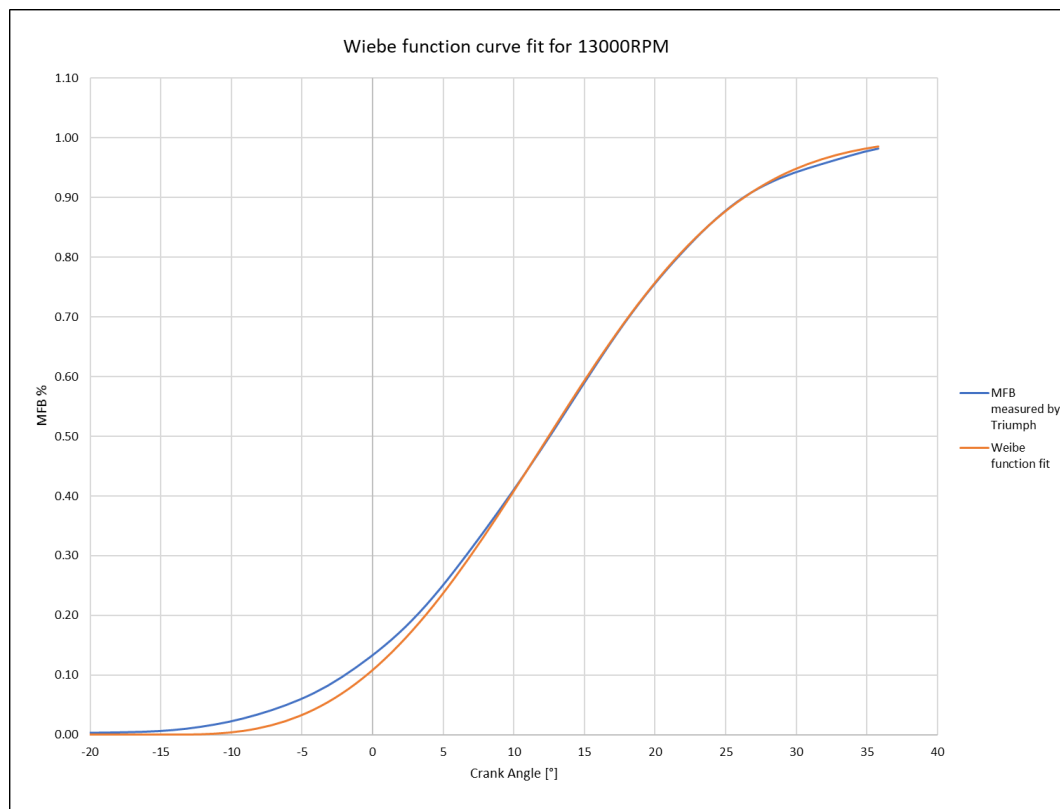


Figure 13 Iteration of the Wiebe function fit at 13000RPM.

This was only an iteration of the fitting, but it didn't get much better regarding the bottom end of the curve around the 0 – 25 % burn area. Which is potential cause for the difference in curvature departure seen in the logarithmic pressure volume graphs.

Because as can be seen in the following, the linear increase before combustion is being predicted as too low. However, at this engine speed, the lower engine of the intake/exhaust phases have been matched in a much more realistic fashion;

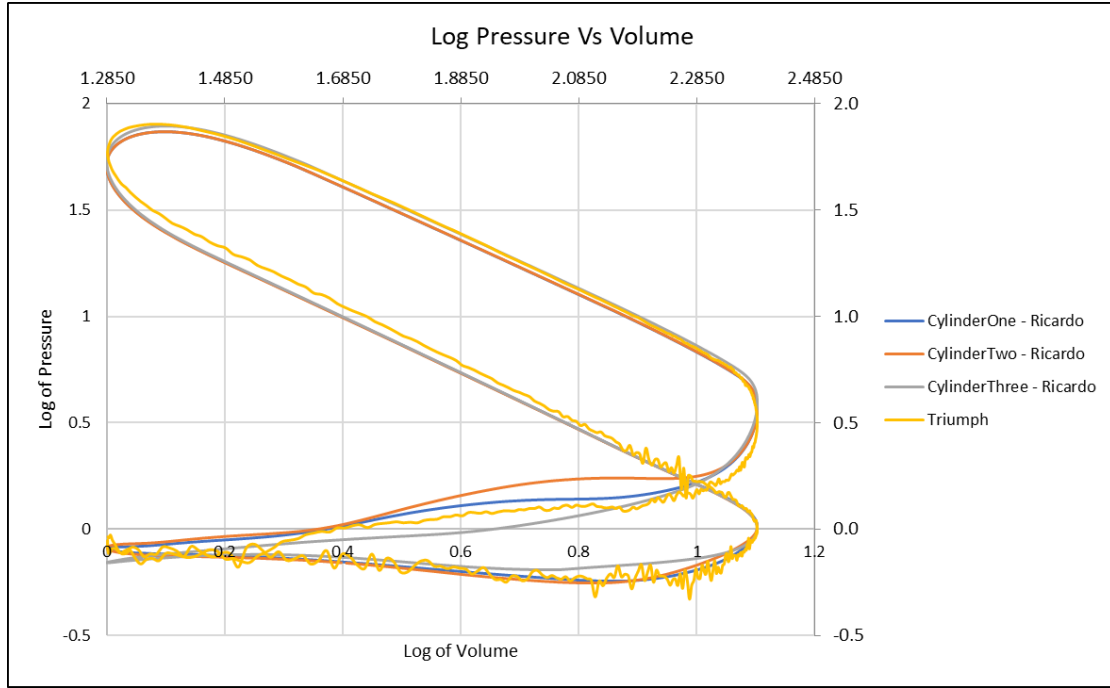


Figure 14 Logarithmic Pressure Vs Volume graph at the final iteration for 13000RPM.

The combustion cycle of the graph is most certainly showing correlation, however all three cylinders modelled in Ricardo WAVE, are not correctly correlating during the adiabatic linear increase. Potentially due to, again, incorrect AFR value and the number of molecules within the system (Air & Fuel) isn't representative of the real values, changing the initial internal enthalpy and therefore the temperature of the model. With specific heat being a function of temperature as demonstrated in the equation;

$$T_P = T_0 + \frac{LHV \cdot N_{fuel} \cdot M_{fuel}}{\sum_i N_{i,p} \hat{c}_{pi}}.$$

Equation 4 Equation showing temperature of products as a sensible enthalpies of reactants (McAllister, Chen, & Fernandez-Pello, 2011)

There's certainly room for error (McAllister, Chen, & Fernandez-Pello, 2011).

The knock-on effect of this, is an unwanted change in the ratio of specific heats, used in the governing equation, $PV^n = constant$. Where the power 'n' is the ratio of specific heats (also denoted as 'γ') and this governs the linearity and increase of the line under the adiabatic isentropic process.

This deviation could have also been a result of incorrect starting reference temperatures and or pressures. When looking at the fundamental equation, deemed as ‘the universal gas law’; $PV = nRT$, along with the AFR causing a different value within this equation, the pressure assumed at the start of the process could be off setting the values, as well as a poorly correlated starting temperature. All of which will change the values carried through the process, leading to potentially large margin of error at combustion.

The last part of the modelling validation, concerns the perceived power and torque produced by the ICE. The values were stated to be; peak power of 94Kw produced at 12600RPM and a peak torque value of 73Nm at 11750RPM. The model did not reflect these values, as seen in the following figure;

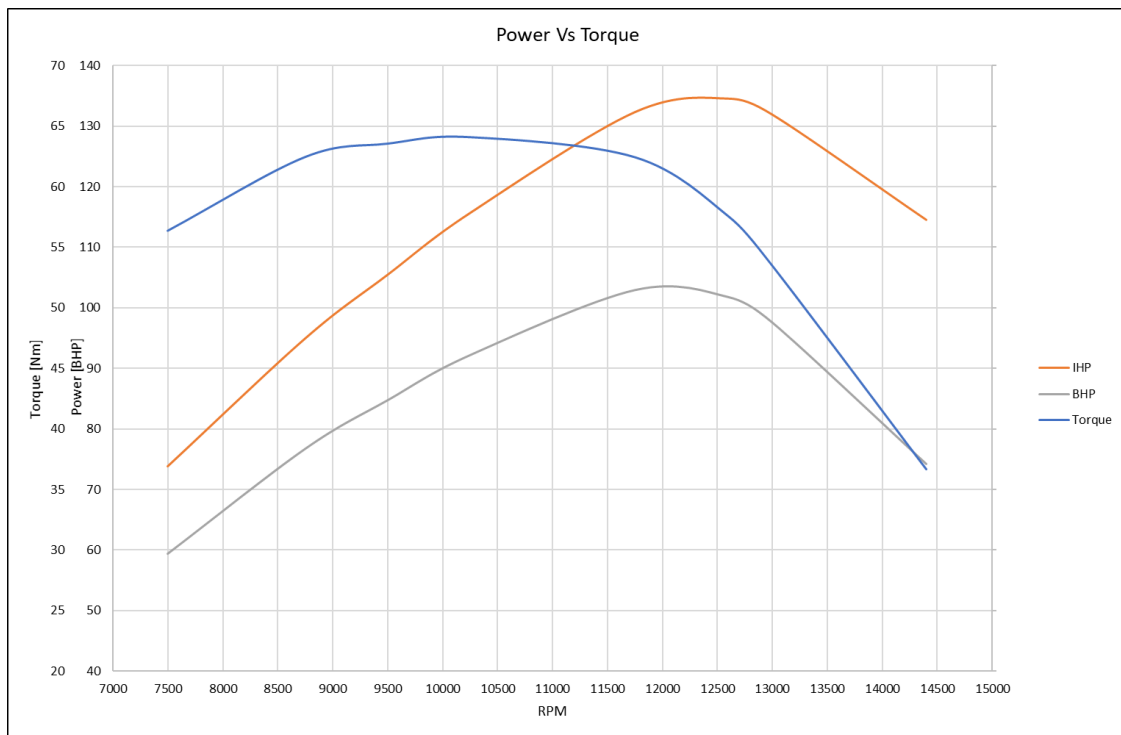


Figure 15 Power and Torque values as a result of Ricardo WAVE modelling.

When looking at the graph, it's clear there is a definite expected trend shown from the results, the climb in power until the point where the restrictive “choking” nature of the ICE is most likely reached. Along with the reality that the frictional losses incurred, rise in an exponential fashion, causing a physical limit.

Also shown, is the rapid decline in torque after around 11000RPM and because power is a known function of torque; $P = T\omega$. Although the engine speed is high, the rapid decline in torque brings the value for power down along with it.

With this being said, upon research into realistic values for the motorcycle, the values that are actually being shown were seemingly close to external dynamometer readings. But the variation in the presented results is too much to draw anything conclusive from.

One potential reason for the power and torque being low is the losses due to friction. This would be presented as the FMEP. Using the equation that describes the relationship between mean effective pressures;

$$FMEP = IMEP - BMEP$$

Equation 5 Mean effective relationship 3 (x-engineer, n.d.).

The IMEP, minus the BMEP, will result in the losses in between. Because the IMEP is a value representative of the real pressure seen inside the cylinder due to combustion, prior to any form of losses in the transfer of the energy. Of course, these values were known because the in-cylinder pressure was provided. But the brake mean is a measurement, consisting of the effective torque measured via a dynamometer, at the flywheel. Therefore, this has then taken the frictional losses into consideration.

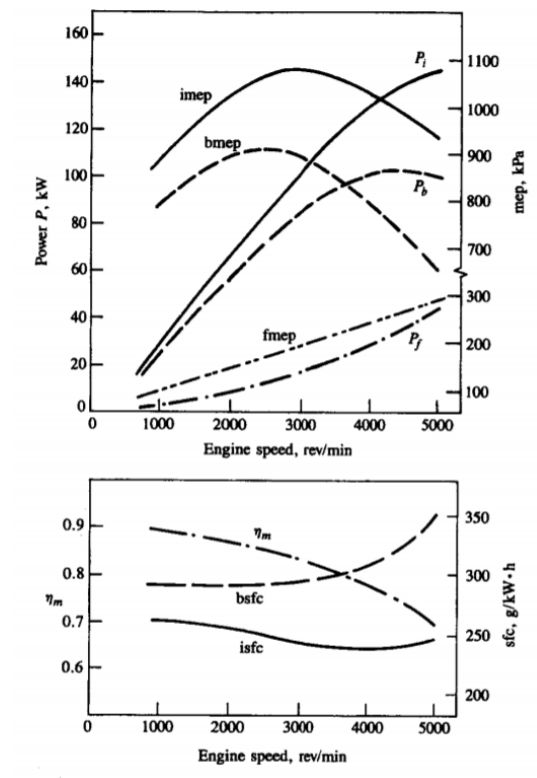


Figure 16 Depiction of the effect of mean effective pressures across lower engine speeds. Along with the effect on fuel consumption and engine efficiency (Heywood J. B., 1988).

This data was not provided and therefore the correlation for frictional losses were simply open to “interpretation” and could quite easily have been dramatically reduced, but that wouldn’t have been realistic. Therefore, the four frictional coefficient input values within Ricardo WAVE; ACF, BCF, CCF and QCF (that would have been different at each engine speed), were left to default.

5. Conclusion.

The conclusion drawn from this study, is such that combustion certainly poses a difficult problem when attempting to simulate. However, with copious experiments conducted across the years of development for the internal combustion engine, there is numerous trends that have become noticeably similar for many variants of ICE's.

Because of these trends, relationships between; fluid flow, the thermodynamic properties of the fluids and materials, initial conditions of temperature & pressure and an understanding of the energy within a system, based upon conservation of mass, can be found and used to quantify the magnitude of variables concerned with simulation.

Upon studying the real data collected and implementation of the found; equations, functions and correlations, it's clear that being able to predict the combustion process within a cylinder is possible. Along with Ricardo's WAVE software as a validation tool is certainly possible.

Although this study hasn't completely correlated the predicted with the measured, the correct and expected trends can be seen even with the limited information available to develop the model. It was clear to see that the main affecting factors of combustion were; the initial conditions for the model because of their impact on the flow characteristics, air to fuel mass ratios and its effect on MFB, joint with the rate at which heat is released, lastly the engine geometry and physical (Heywood J. B., 1988).

Because of this, it is believed that the most effective way to resolve any issues, or at least help in correlating the results, to be realistic and in turn validate the software. Would be to have more real measured data inputs, like; exhaust temperatures and pressures, correct air to fuel mass ratios, definitive operating conditions; ambient temperatures and pressures, load situations and BMEP, and potentially measured points across the intake manifold, all too be put into the software in an attempt to make a significant reduction in errors across the model.

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7. Appendices.

A. Cylinder pressures and the change in pressures for each engine speed.

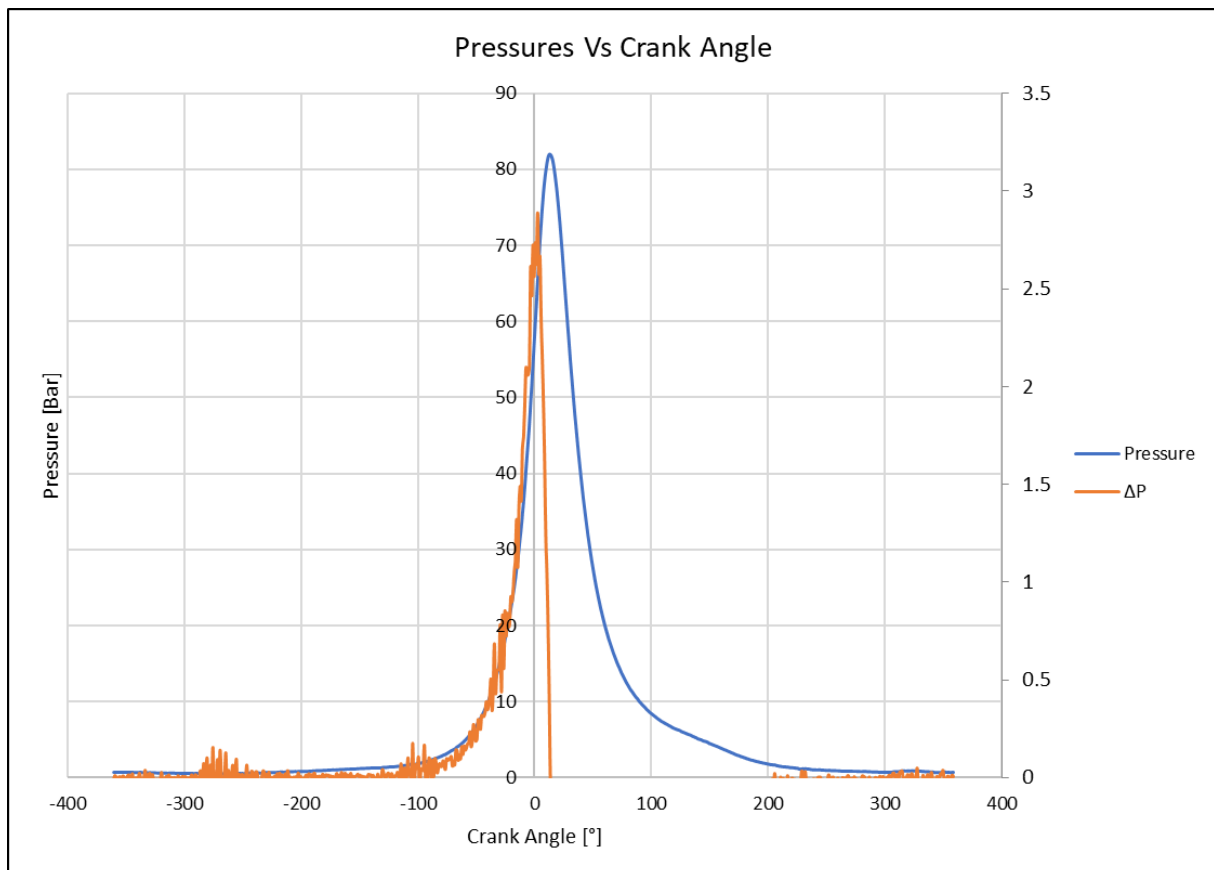


Figure 17 Pressure and difference in pressure Vs Crank angle for 10250RPM.

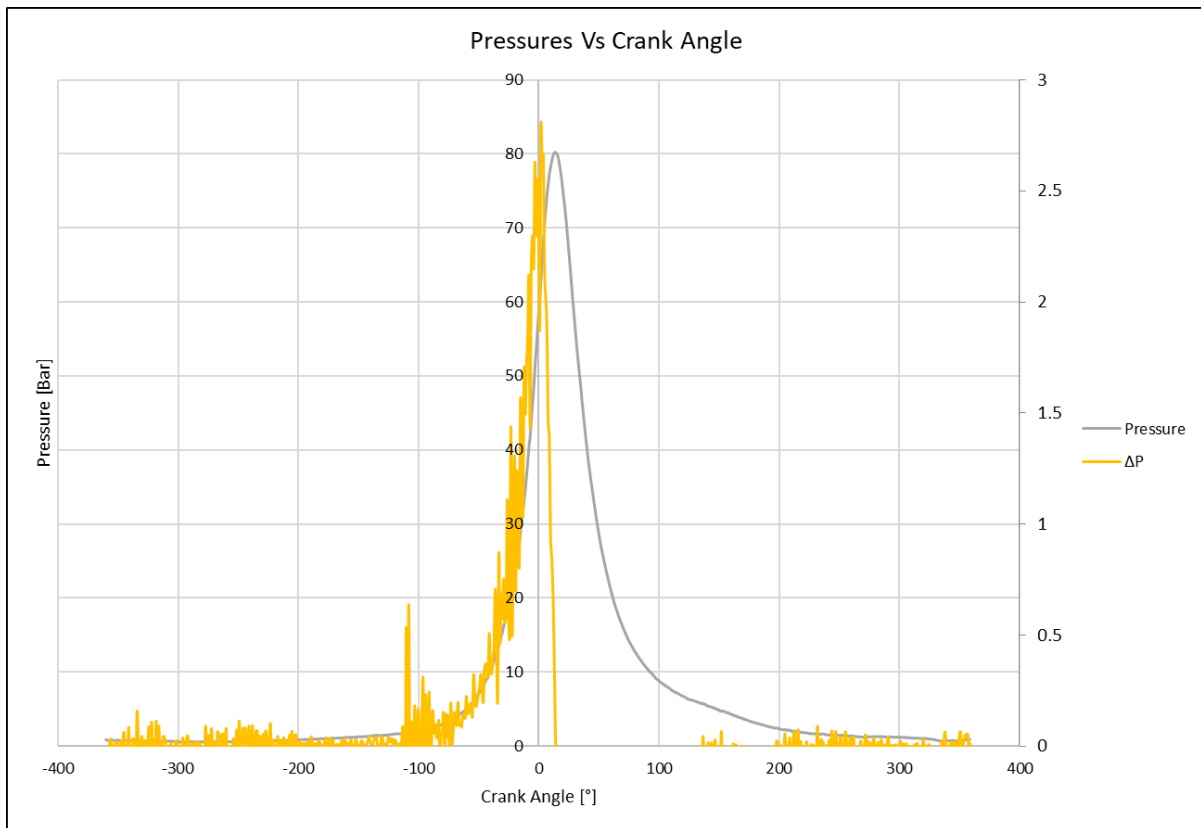


Figure 18 Pressure and difference in pressure Vs Crank angle for 13000RPM.

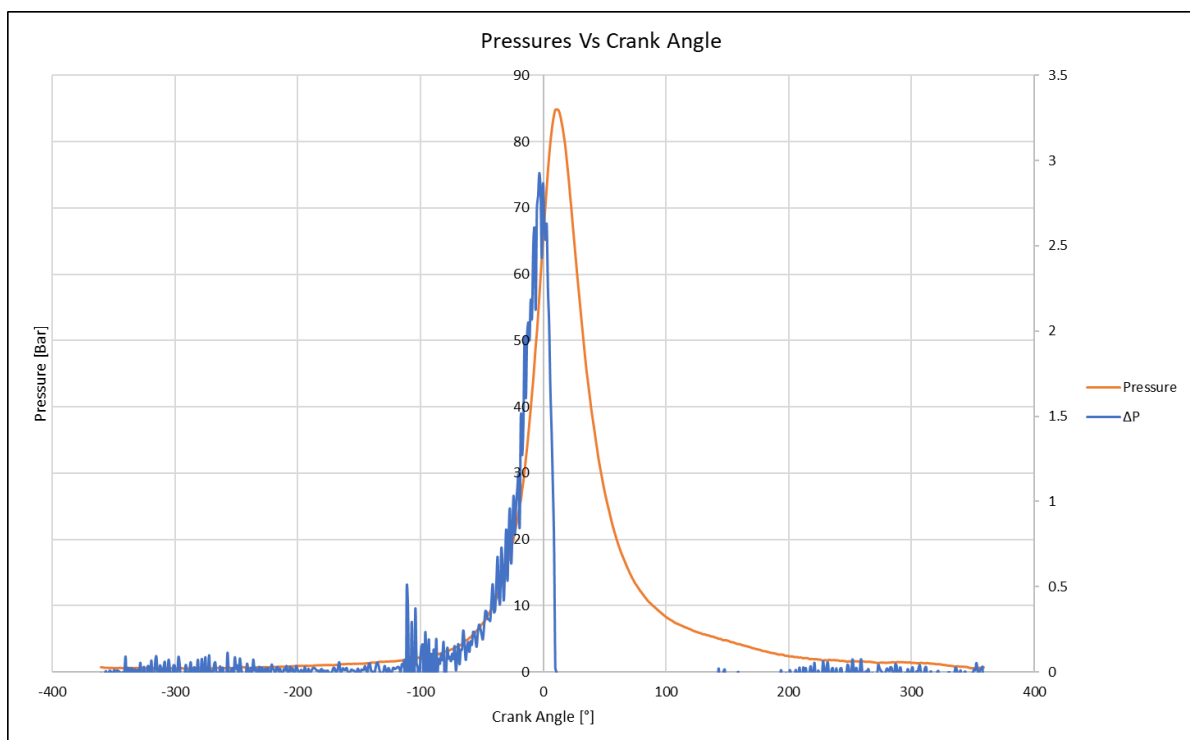


Figure 19 Pressure and difference in pressure Vs Crank angle for 14400RPM.

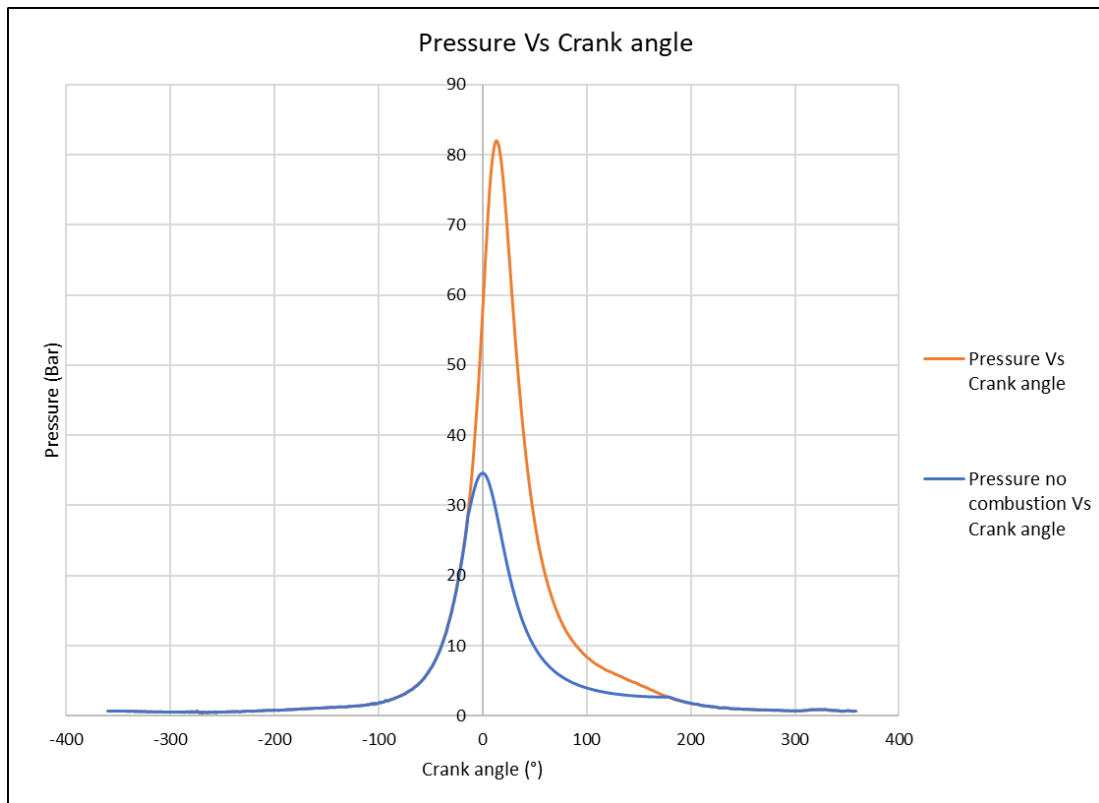


Figure 20 10250RPM Pressures with and without combustion.

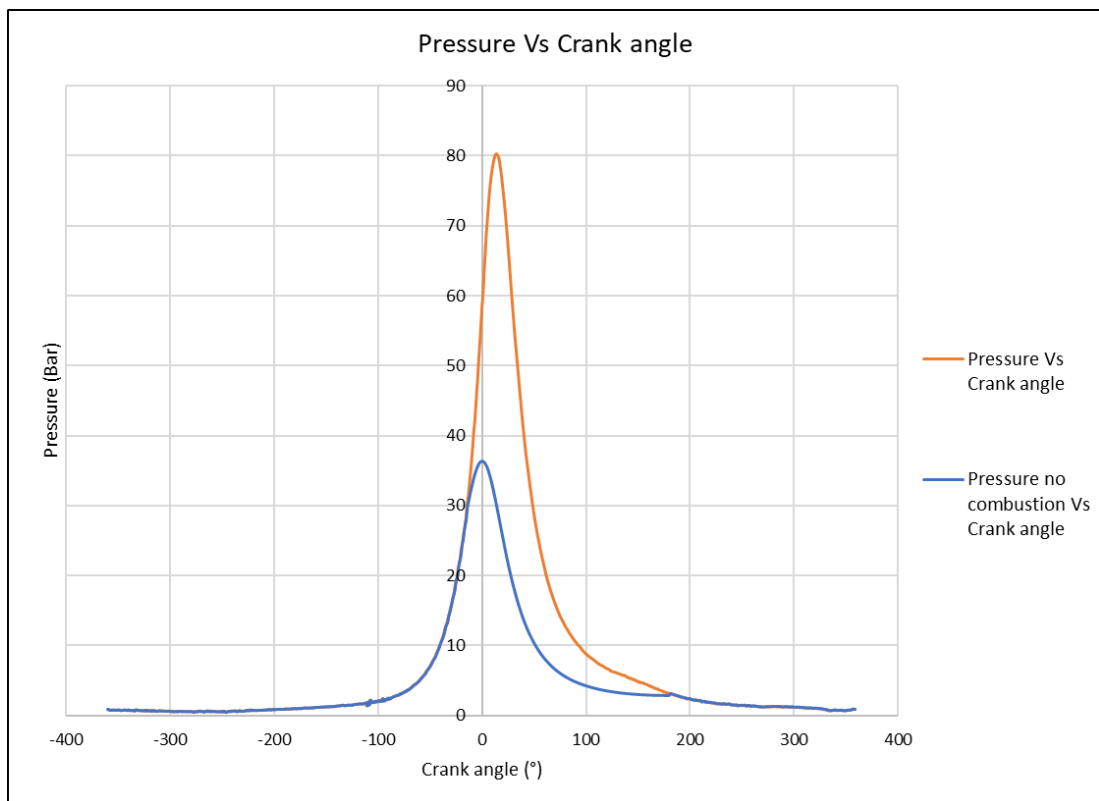


Figure 21 13000RPM Pressures with and without combustion.

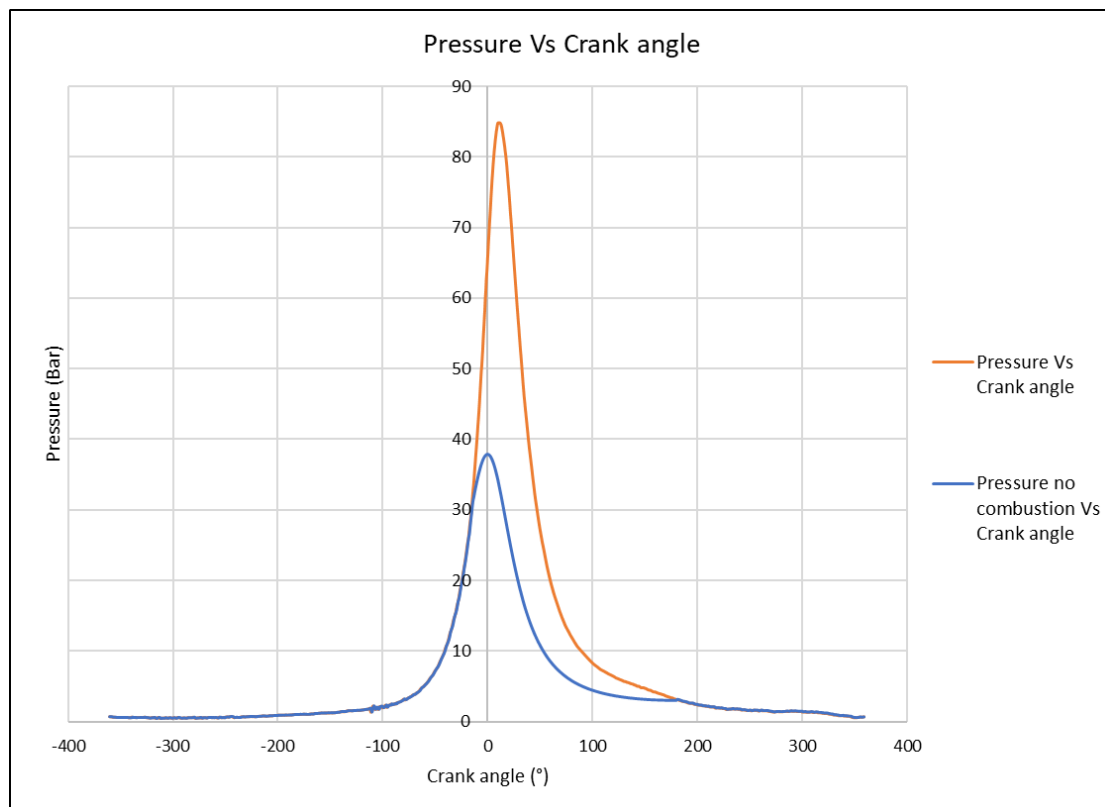


Figure 22 14400RPM Pressures with and without combustion.

B. Volume

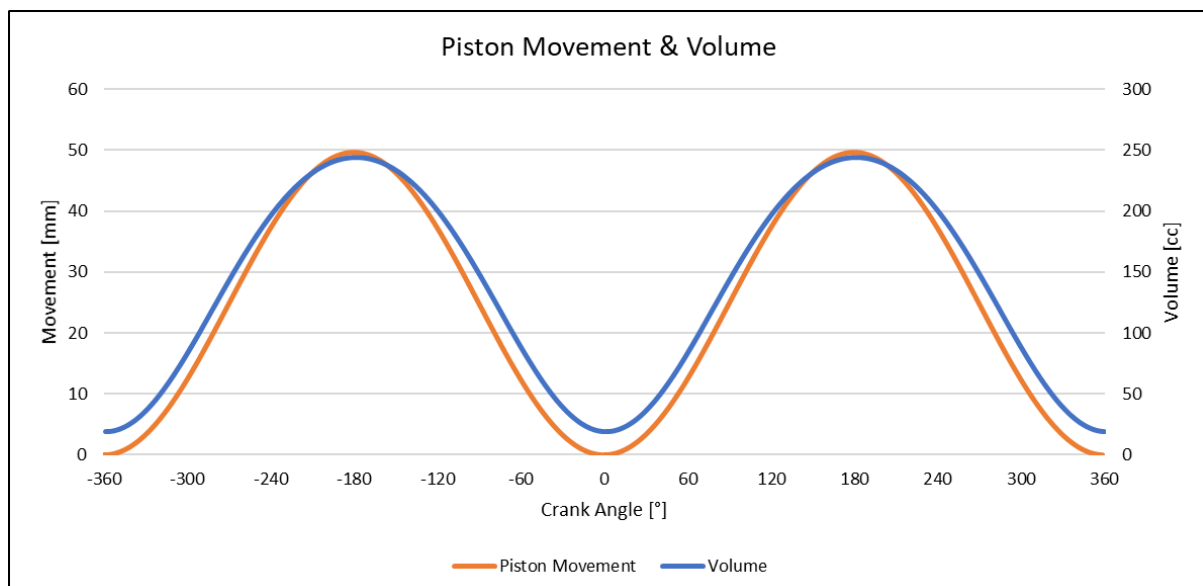


Figure 23 Cylinder Volume and Piston position in correlation to.

C. Graphical representations of the effects of the exponent
“m” used within the Wiebe function and heat release rate.

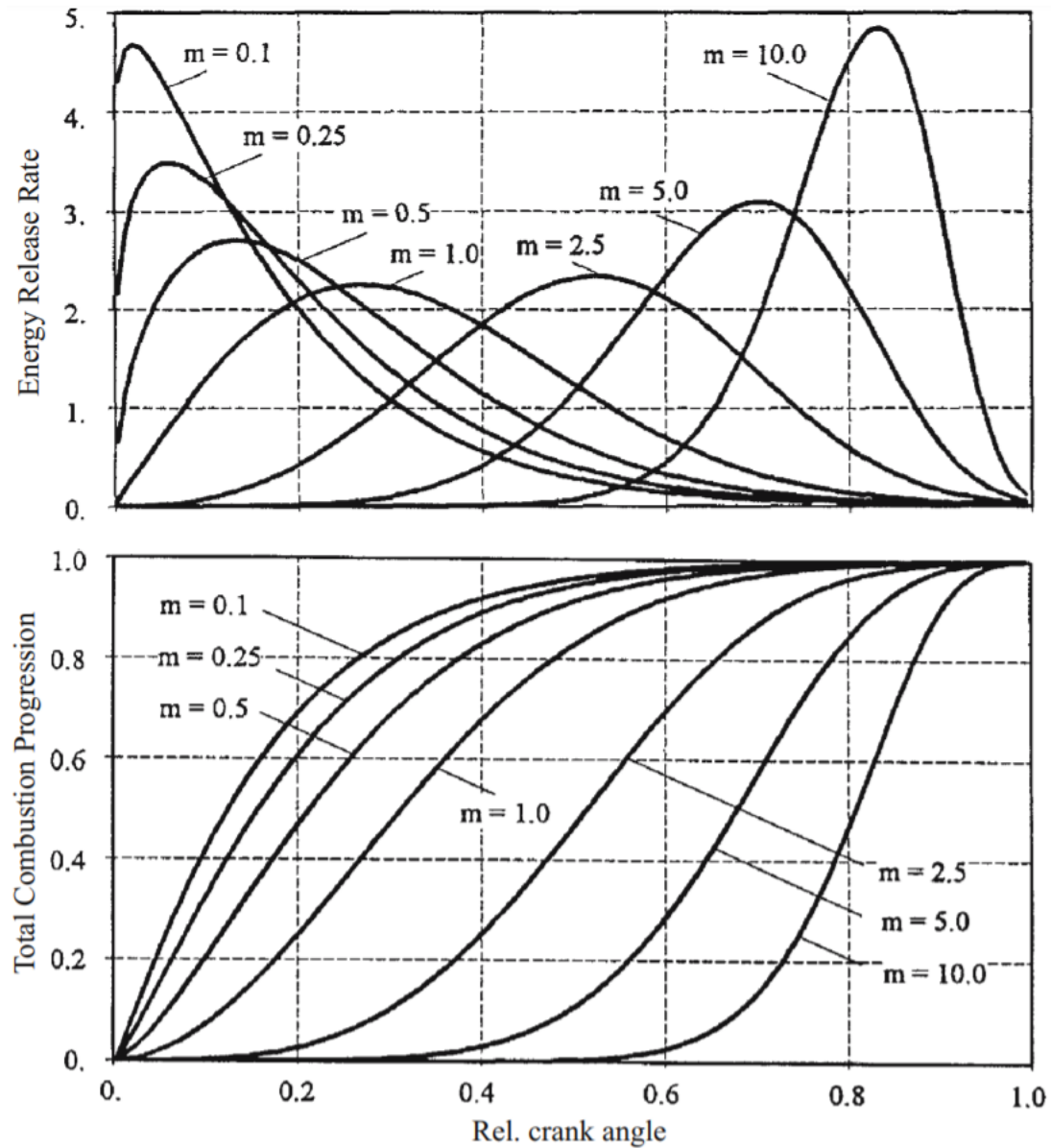


Figure 24 “m” exponent used within the Wiebe function as presented by Merker et al. (Merker, Schwarz, Stiesch, & Otto, 2006)

D. Burnt and Unburnt percentages.

The unburnt is simply shown as $1 - MFB = \text{Unburnt percentage}$.

The unburnt mass fraction can be used to figure out the percentage volume left ahead of the combustion flame. This could then be used as information for when beginning to develop a two-zone model of combustion, not just the simple single zone.

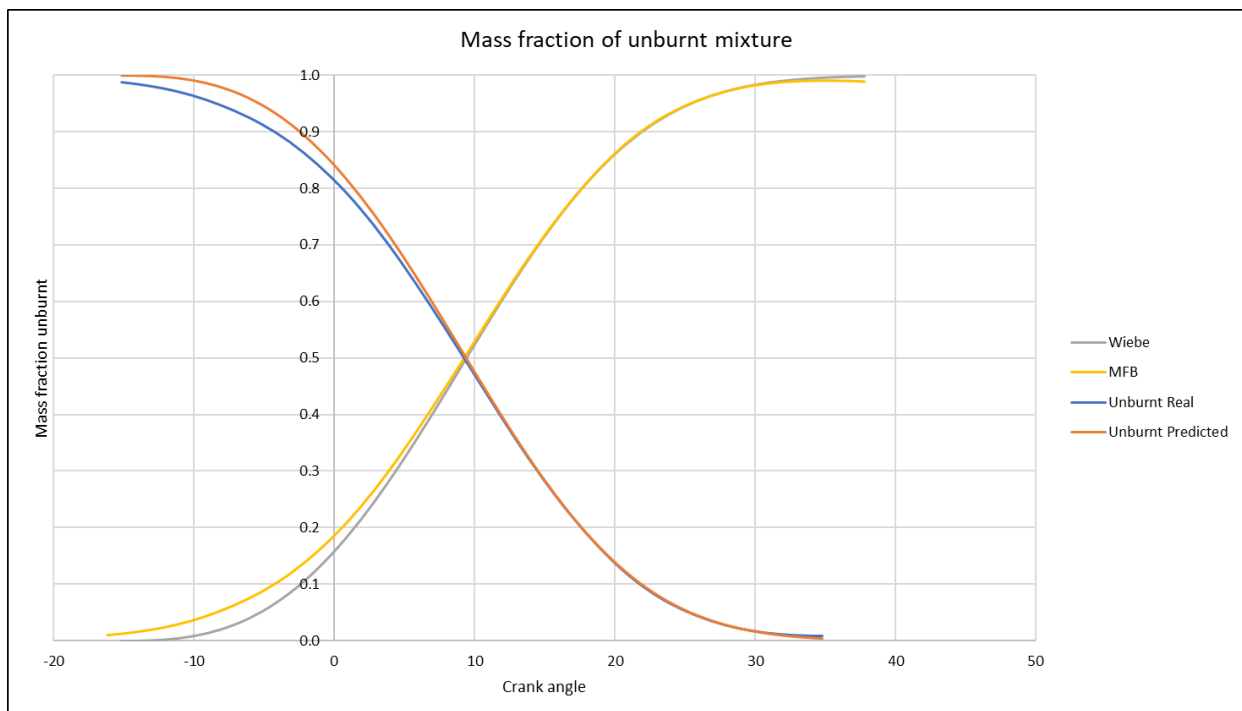


Figure 25 Mass fraction Burnt and Unburnt.

E. Maximum Brake Torque Timing (MBT).

Maximum brake torque timing is known to be achieved through physical testing and is thought of as “Combustion timed to release peak pressure at a point where the engine can mechanically take the best advantage” (Banish, 2007). Meaning that peak pressure will commonly occur at around 7 – 15 degrees ATDC, when the piston is in a good position to make it positive work and the effective volume it’s acting in is still small enough to make a large impact. During this study, when looking at the timing of the peak pressures and 50% MFB, the ignition timing of the Triumph was clearly done, so that it is achieving the MBT across each RPM.

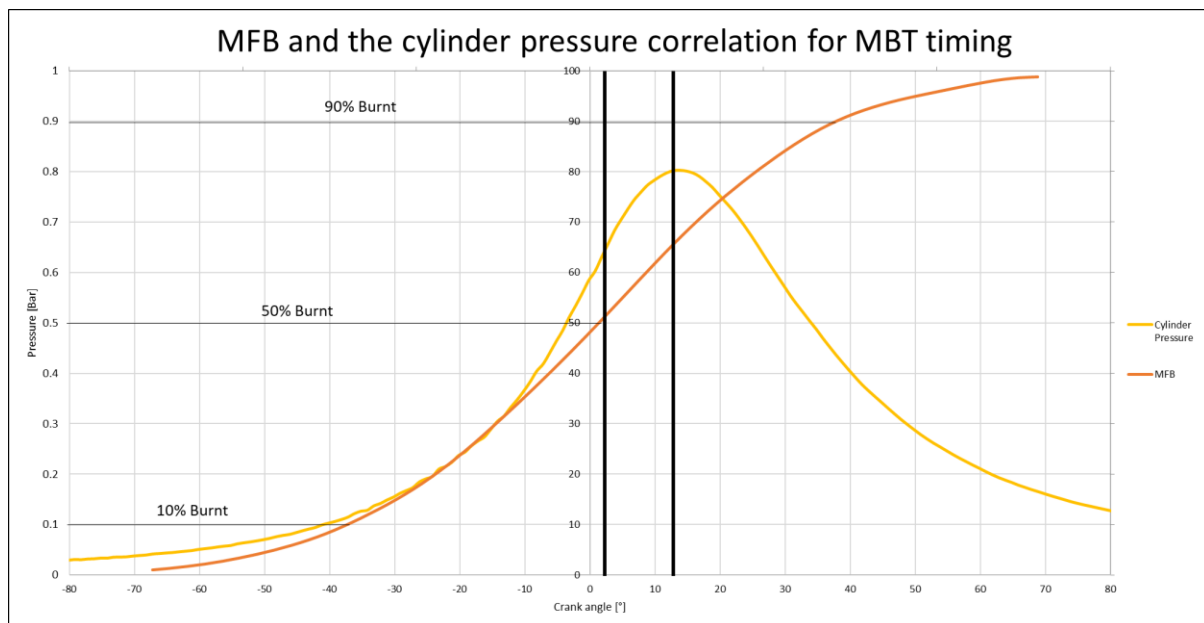


Figure 26 Peak pressure at 13000RPM occurring within the common range of CA, demonstrating the idea of MBT timing.

F. Lift and Flow coefficients used for the inlet and exhaust valves.

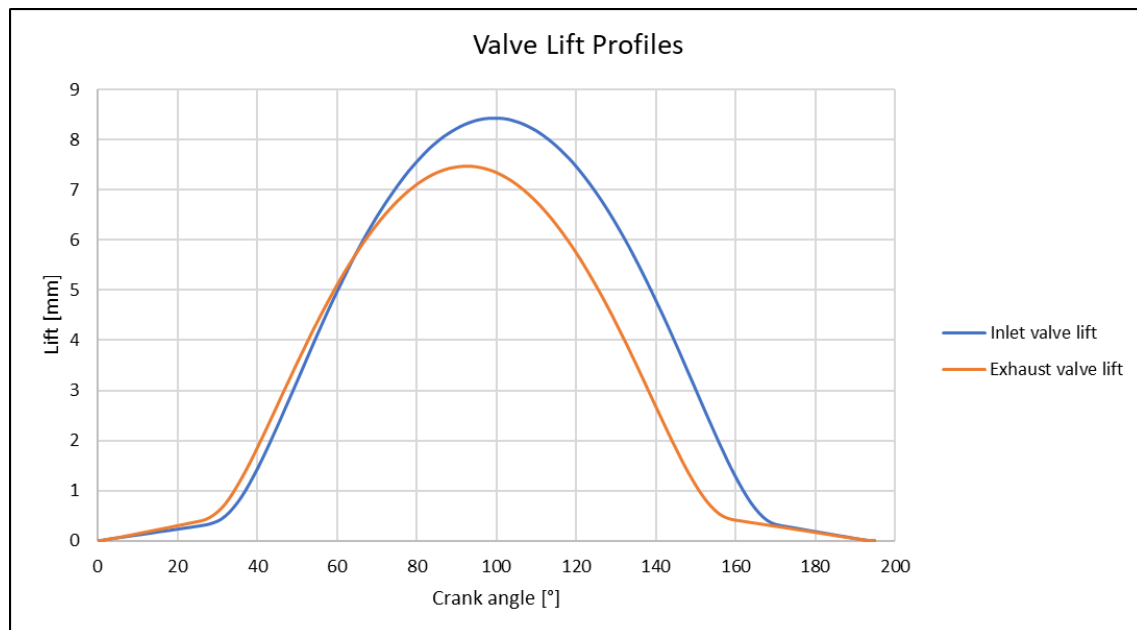


Figure 27 Valve Lift Profiles used within the study.

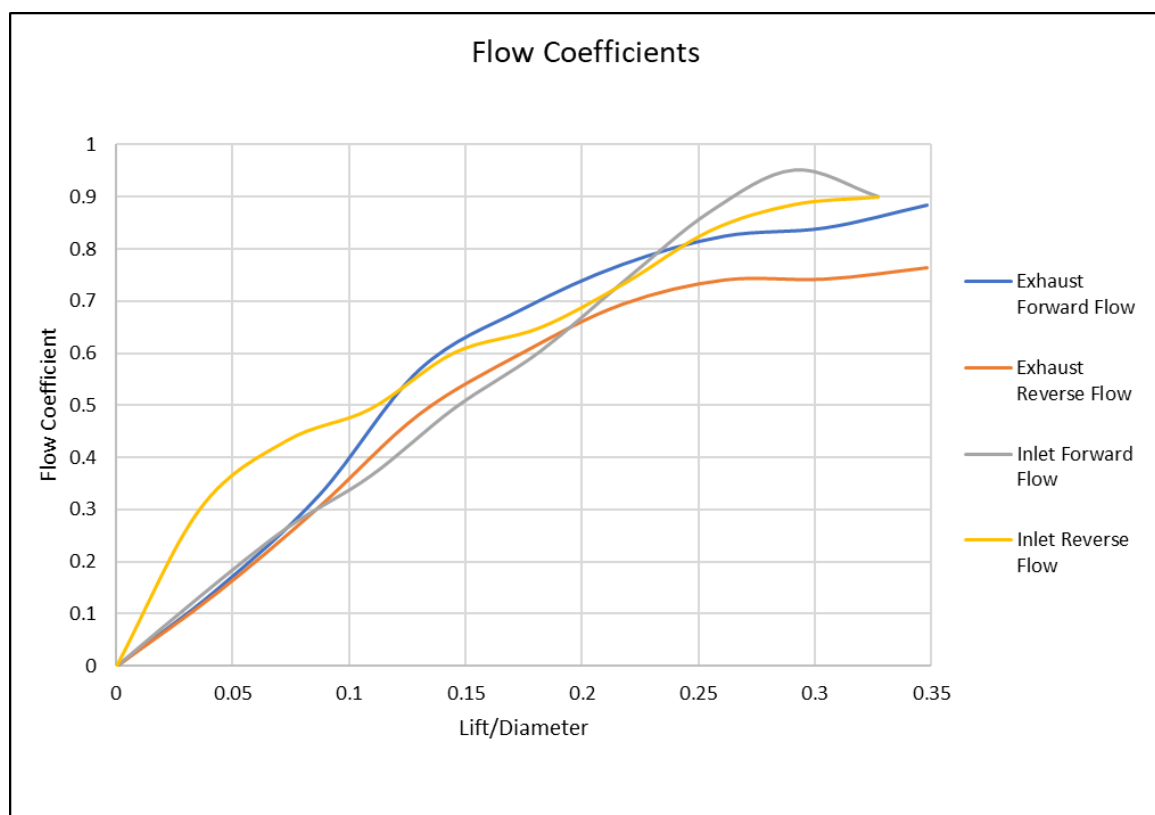


Figure 28 Flow coefficients for the Inlet & Exhaust valves.