



University of Wales Trinity Saint David
School of Engineering

Optimisation Of A High Speed 3
Cylinder Using The Ricardo WAVE
Engine Development Software

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Advanced Computational Methods: Ricardo WAVE
M6X00408

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Contents

1) Introduction.....	1
2) Methodology	2
3.1. Excel.....	2
3.2. Ricardo WAVE.....	8
3.3. Changes Made to Ricardo Model.....	10
3.3.1. Con Rod Length.....	10
3.3.2. 10, 50 & 90% MFB Locations & Burn Durations.....	10
3.3.3. Changed Fuel/Air Ratio to Same for All Injectors	11
3.3.4. Recalculated Coolant-Side Area	11
3.3.5. Recalculated Volume & Skin Friction Area of Junctions	12
3.3.6. Changed Induction Components' Air Pressure To "PREF"	12
3.3.7. Changed Induction Components' Air Temperature To "TREF"	13
3.3.8. Changed Junction and Duct Diameters to Stop Computational Errors	13
3.3.9. Reset Heat Transfer Multipliers to Default.....	15
3.3.10. Changed Valve Diameters.....	15
3.3.11. Changed Fuel/Air Mass Ratio For Each RPM.....	16
4) Results & Analysis	17
4.1. Results from Changes to the WAVE Model.....	17
4.1.1. Benchmark Model.....	17
4.1.2. Con Rod Length.....	17
4.1.3. 10%, 50% & 90% MFB Locations & Burn Durations.....	18
4.1.4. Changed Fuel/Air Ratio to Same for All Injectors	18
4.1.5. Recalculated Coolant-Side Area	19
4.1.6. Recalculated Volume & Skin Friction Area of Junctions	19
4.1.7. Changing Induction Components' Air Pressure To "PREF"	20
4.1.8. Changing Induction Components' Air Temperature To "TREF"	21
4.1.9. Changed Junction and Duct Diameters to Stop Computational Errors	22
4.1.10. Reset Heat Transfer Multipliers to Default	22

4.1.11.	Changed Valve Diameters.....	23
4.1.12.	Changed 'FARD' For Each RPM	23
4.2.	Volume	24
4.3.	Pressure	25
4.4.	Logarithmic Pressure vs Volume (Log PV).....	27
4.5.	Power & Torque	29
5)	Conclusion	30
6)	References	31

Table 1 – Standard & Modified Inputs for the Wiebe Equation – 14400 RPM.....	7
Table 2 – Benchmark Max Cylinder Pressures - WAVE vs Excel	9
Table 3 – Mass Fraction Burnt Crank Angles	10
Table 4 – Fuel / Air Ratios Used for Each RPM.....	16
Table 5 – Benchmark Max Cylinder Pressures - WAVE vs Excel	17
Table 6 – Max Cylinder Pressures Due to Updated Connecting Rod Length.....	17
Table 7 – Max Cylinder Pressures Due to Updated Burn Duration	18
Table 8 – Max Cylinder Pressures Due to Updated 50% Burn Locations	18
Table 9 – Max Cylinder Pressures Due to Fuel / Air Ratio.....	18
Table 10 – Max Cylinder Pressures Due to Recalculation of Coolant-Side Area	19
Table 11 – Max Cylinder Pressures Due to Recalculated Junction Volumes & Areas	19
Table 12 – Max Cylinder Pressures When All Induction System Components Used 1 Bar Pressure	20
Table 13 – Max Cylinder Pressures When All Induction Components Except Ambients Used 1 Bar Pressure	20
Table 14 – Max Cylinder Pressures Due to Changing Air Temperature to 300K	21
Table 15 – Max Cylinder Pressures Due to Changing Air Temperature to 315K	21
Table 16 – Max Cylinder Pressures Due to Changing Air Temperature to 285K	21
Table 17 – Max Cylinder Pressures Due After Errors Have Been Removed	22
Table 18 – Max Cylinder Pressures Due to Resetting Heat Transfer Multipliers.....	22
Table 19 – Max Cylinder Pressures Due to Changing Diameter of Valves & Ports.....	23
Table 20 – Max Cylinder Pressures Due to Fuel / Air Ratio Optimisation	23
Table 21 – Power (kW) & Torque (Nm) for Each RPM – Excel & WAVE	29

Figure 1 – Pressure and Volume vs Crank Angle for 10250 RPM	3
Figure 2 – Log P vs Log V for 10250 RPM.....	3
Figure 3 – Difference in Pressure & Pressure vs Crank Angle	5
Figure 4 – Difference in Pressure vs Crank Angle.....	5
Figure 5 – Mass Fraction Burnt & Unburnt Fuel vs Crank Angle	6
Figure 6 – Burn Rate vs Crank Angle.....	6
Figure 7 – Original (Top) & Modified (Bottom) Mass Fraction Burnt vs Crank Angle – 14000 RPM.....	7
Figure 8 – Ricardo WAVE Interface & Triumph 675 Model.....	8
Figure 9 – WAVE Model Constants Table.....	8
Figure 10 – WAVE Injectors Properties – Before & After Adding Fuel/Air Ratio Reference .	11
Figure 11 – WAVE General Engine Conduction Control Parameters – Before & After Recalculating Coolant-Side Area	11
Figure 12 – Complex Y-Junction Parameters Panel – Before & After Recalculating Volume & Skin Friction	12
Figure 13 – Ambient Elements Parameters Panel – Before & After Changing Initial Pressure Condition to "PREF" Constant.....	12
Figure 14 – Ambient Elements Parameters Panel – Before & After Changing Initial Temperature Condition to "TREF" Constant.....	13
Figure 15 – WAVE Computational Interface with Duct & Junction Errors Highlighted.....	13
Figure 16 – WAVE Openings for Y-Junction 20 – Before & After Defining the Diameter	14
Figure 17 – WAVE Parameters for Duct 65 – Before & After Entering the Corrected Diameter	14
Figure 18 – WAVE Engine Heat Transfer Parameters – Before & After Resetting the Multipliers to Default Values.....	15
Figure 19 – WAVE Valve Parameter Editor Panel – Before & After Correcting the Reference Diameter	15
Figure 20 – WAVE Duct Parameters Panel – Before & After Adjusting the Diameter to Match Those of The Valves	16
Figure 21 – Volume vs Crank Angle – Excel vs WAVE Comparison	24
Figure 22 – Pressure vs Crank Angle – 10250 RPM	25
Figure 23 – Pressure vs Crank Angle – 13000 RPM	26
Figure 24 – Pressure vs Crank Angle – 14400 RPM	26
Figure 25 – Logarithmic Pressure vs Volume Comparing – Excel & WAVE	27
Figure 26 – Logarithmic Pressure vs Volume Comparing – Excel & WAVE	28
Figure 27 – Logarithmic Pressure vs Volume Comparing – Excel & WAVE	28
Figure 28 – Power & Torque vs RPM – Excel & WAVE.....	29

Equation 1 – Absolute Pressure Equation	2
Equation 2 – Swept Volume Equation	2
Equation 3 – Clearance Volume Equation.....	2
Equation 4 – Volume Per Degree Equation.....	2
Equation 5 – Work Done Equation	4
Equation 6 – Total Work Done Per Cycle Equation	4
Equation 7 – Revolutions per Second Equation	4
Equation 8 – Time per Revolution Equation	4
Equation 9 – Engine Power Produced Equation.....	4
Equation 10 – Pressure Difference Equation.....	4
Equation 11 – Wiebe Function / Mass Fraction Burnt Equation (Yeliana, et al., 2008)	5
Equation 12 – Unburnt Fuel Equation	6
Equation 13 – Burn Rate Equation	6
Equation 14 – Desired Burn Percentage Crank Angle Linear Interpolation Equation.....	10
Equation 15 – Ricardo Volume Equation.....	14
Equation 16 - Ricardo Volume Equation Rearranged	14
Equation 17 – Calculating Torque from Power Equations	29

1) Introduction

This experiment was conducted to show the correlation between real world measured data and how it is possible to mathematically simulate combustion to a satisfactory degree within the Ricardo WAVE software. The data used was provided from Triumph, for their 675 Daytona motorcycle.

This data was analysed within Microsoft Excel and compared to data extracted from WAVE across multiple iterations in order to determine the most important factors of the software. Along with the hopes of validating the software's ability to predict and simulate the operation using a purely mathematical approach.

Ricardo WAVE is a one-dimensional simulation software that computes the Navier-Stokes fluid dynamics equations which govern the transfer of mass, energy and momentum for gas and liquids. The software allows a user to create a virtual representation of real engine for the analysis of both performance and acoustics for optimisation to suit any application. (Ricardo PLC, 2019)

Some assumptions of both the engine and the surrounding environment are made, this ensures that the data replicated by WAVE is representative of ideal conditions in which to test the engine. This means that the software, in theory, will generate the same data each time it is calculated, if no variables are changed, enabling the ability to correlate this data to a real-world experiment.

2) Methodology

3.1. Excel

When this investigation began, real pressure data collected during a dynamometer run of the 675 Daytona engine was provided by Triumph for three engine speeds: 10250, 13000 & 14400. This data consisted of Crank Angle (°), Gauge Cylinder Pressure (Bar), Burn Rate (%), Volume (%) and Time (μs). It was then imported into Microsoft Excel for analysis before the accompanying Ricardo WAVE model was evaluated.

Two crucial pieces of data required manipulation:

1. Cylinder pressure was measured by a pressure gauge, a tool which at atmospheric pressure, measures zero, when in fact, ambient pressure on Earth is 1 bar at sea level.

$$\text{Absolute Pressure (Bar)} = \text{Gauge Pressure (Bar)} + 1$$

Equation 1 – Absolute Pressure Equation

2. Volume was provided in a percentage form, where TDC = 0% and BDC = 100%. By knowing the basic geometry of the engine (bore, stroke, compression ratio) the volume of the engine at every degree of crank angle could be calculated:

$$\text{Swept Volume (m}^3\text{)} = \left(\frac{\pi * \text{Bore (m)}^2}{4} \right) * \text{Stroke (m)}$$

Equation 2 – Swept Volume Equation

$$\text{Clearence Volume (m}^3\text{)} = \frac{\text{Swept Volume (m}^3\text{)}}{(\text{Compression Ratio} - 1)}$$

Equation 3 – Clearance Volume Equation

$$\text{Volume Per Degree (m}^3\text{)} = \left(\left(\frac{\text{Volume (\%)}}{100} \right) * (\text{Swept Vol (m}^3\text{)}) \right) + (\text{Clearence Vol (m}^3\text{)})$$

Equation 4 – Volume Per Degree Equation

Now the data could be analysed, and graphs plotted to attain characteristics about the engine to determine its performance according to Triumph.

To start with, the pressure and volume were plotted onto a graph to confirm whether both sets of data follow the conventional trends, as expected from a standard spark ignition engine, shown in Figure 1.

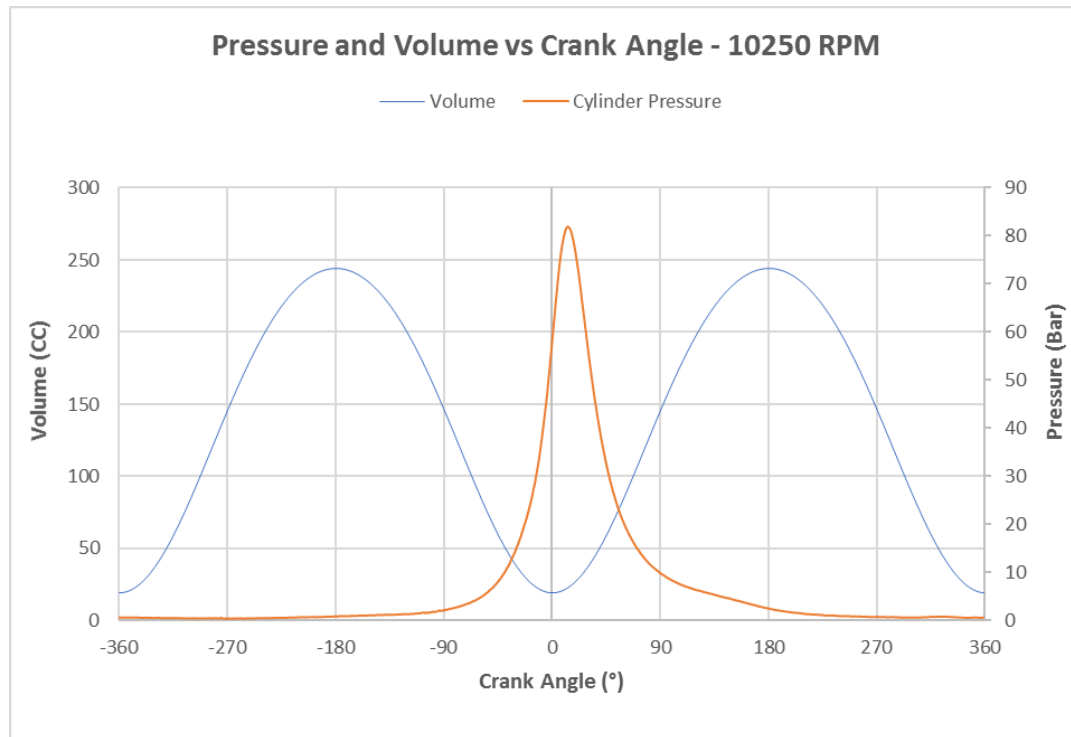


Figure 1 – Pressure and Volume vs Crank Angle for 10250 RPM

The logarithmic values were taken for pressure and volume and plotted to produce a “Log PV” graph, this graph can tell numerous pieces of information about the engine during operation. The main information being taken from the graph at this moment in time, is the polytropic index values during the compression and expansion phases of the engine cycle, as seen in Figure 2, points were chosen along the linear portions, represented by the orange and grey lines, for the start and end of the chosen range, the longer the line, the smoother the average. From the technique described further down, the combustion duration has been highlighted for clarity.

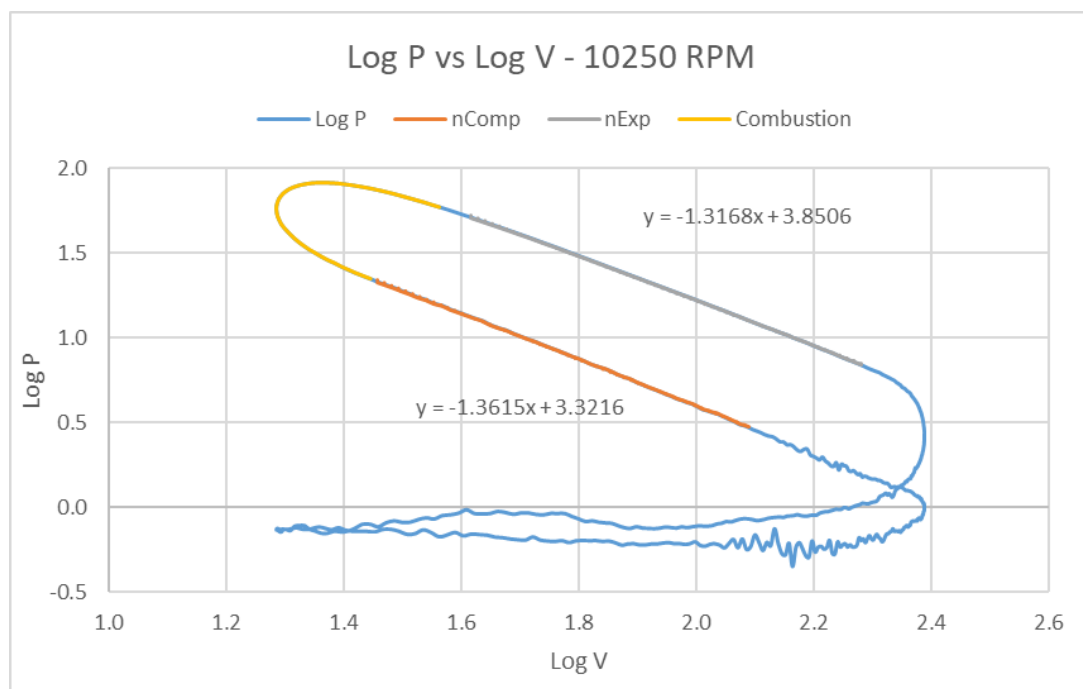


Figure 2 – Log P vs Log V for 10250 RPM

It was possible to calculate the work done per degree of crank angle, throughout each engine speed.

$$\text{Work Done per Degree (J)} = \text{Pressure (Bar)} * (\text{Volume}_2 \text{ (m}^3\text{)} - \text{Volume}_1 \text{ (m}^3\text{)})$$

Equation 5 – Work Done Equation

The sum of the work done across the cycle. This is then used to calculate the power developed from one cylinder and, negating cyclic and cylinder variations, the power of the engine

$$\text{Total Work Done per Cycle (J)} = \sum_{-360}^{360} \text{Work Done Per Degree}$$

Equation 6 – Total Work Done Per Cycle Equation

To further determine the power produced by the engine, the engine speed needs to be converted from Revolutions Per Minute (RPM) to Revolutions Per Second (RPS) to the time per revolution.

$$\text{RPS} = \left(\frac{\text{RPM}}{60} \right)$$

Equation 7 – Revolutions per Second Equation

$$\text{Time per Rev} = \left(\frac{1}{\text{RPS}} \right)$$

Equation 8 – Time per Revolution Equation

Now every component was accumulated to equate the power produced by the engine per cycle. The answer from the equation below is in Watts (W), and therefore, divided by 1000 for the Kilowatts (kW) generated, which is the standard International System of Units (SI) (Check Your Math, 2020). This is multiplied by the number of cylinders to provide to total power of the engine.

$$\text{Power (kW)} = \frac{\left(\frac{\text{Total Work Done per Cycle}}{\text{Time per Rev}} \right)}{1000} * (\text{Number of Cylinders})$$

Equation 9 – Engine Power Produced Equation

The curvature determined by the mass fraction of fuel burnt per degree of crank angle rotation (CA / θ) during combustion can be ascertained by use of the Wiebe function. There are two key variables required for this equation that are unknown: the crank angle that combustion starts, and the duration of the burn.

A column was created which featured the difference in pressures ($P_2 - P_1$), this was plotted onto a graph against crank angle to aid in the deduction of the start of combustion. As seen in Figure 3, the combustion start has been highlighted for clarification, with the entirety of the combustion duration shown within Figure 4. The actual start of combustion (θ_o) is difficult to decide upon, but according to John B. Heywood, referring to the Log PV graph, “the start of combustion can be identified by the departure of the curve from the straight line” (Heywood, 1988) henceforth, 20.19° BTDC was chosen for ignition within the 10250 RPM data.

$$\text{Pressure Difference (Bar)} = (\text{Pressure}_2 - \text{Pressure}_1)$$

Equation 10 – Pressure Difference Equation

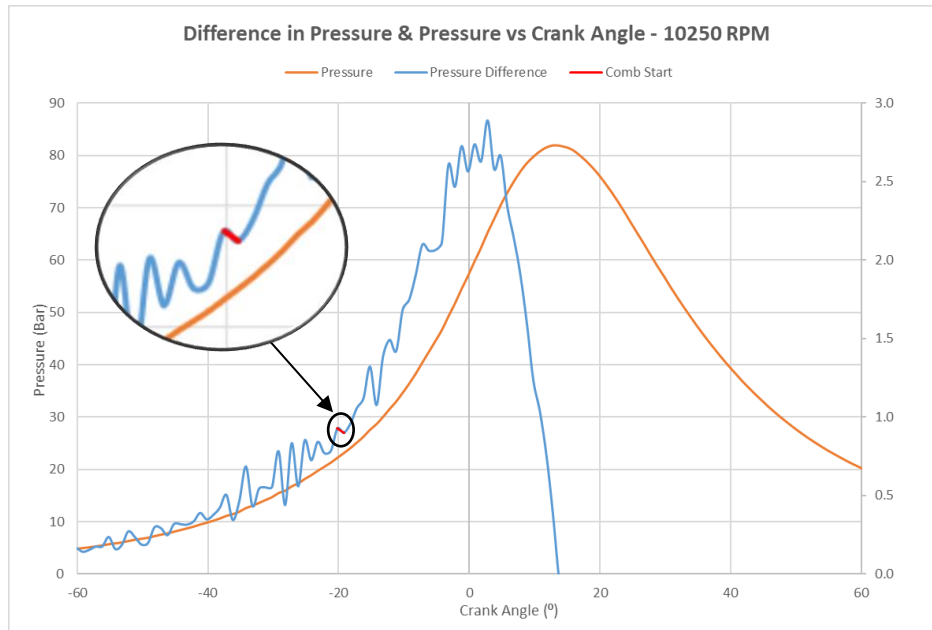


Figure 3 – Difference in Pressure & Pressure vs Crank Angle

As with the start of combustion, the end of combustion is problematic to establish. Using the Triumph data provided to us, the maximum fuel burnt (99.13%) occurs at 34.81° ATDC before then decreasing. The Wiebe function was used to calculate Mass Fraction Burnt (MFB) using the exponents, 'a = 5' & 'm = 2' into the equation, as stated by John B Heywood (Heywood, 1988), and estimating the combustion duration, in the same manner as the start, by choosing the deviating curve from the 'Log PV' graph.

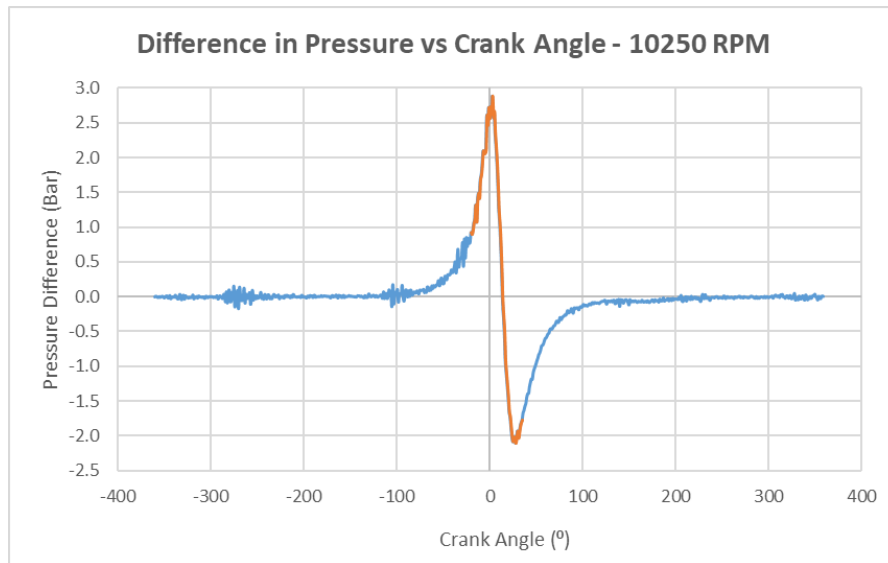


Figure 4 – Difference in Pressure vs Crank Angle

The aim was to match the calculated percentage of fuel burnt to that of the supplied data. By using a burn duration ($\Delta\theta$) of 55.95°, the Wiebe formula managed to match the 99.13% at 34.81° ATDC. This data was entered into Figure 4 to highlight the entire combustion duration.

$$Wiebe = 1 - \exp\left(-a \cdot \left(\frac{\theta - \theta_{\theta}}{\Delta\theta}\right)^{(m+1)}\right)$$

Equation 11 – Wiebe Function / Mass Fraction Burnt Equation (Yeliana, et al., 2008)

The percentage of the unburnt fuel is indirectly proportional to the MFB, decreasing from 100% to 0%, as seen within Figure 5.

$$\text{Unburnt Fuel (\%)} = 1 - \text{MFB}$$

Equation 12 – Unburnt Fuel Equation

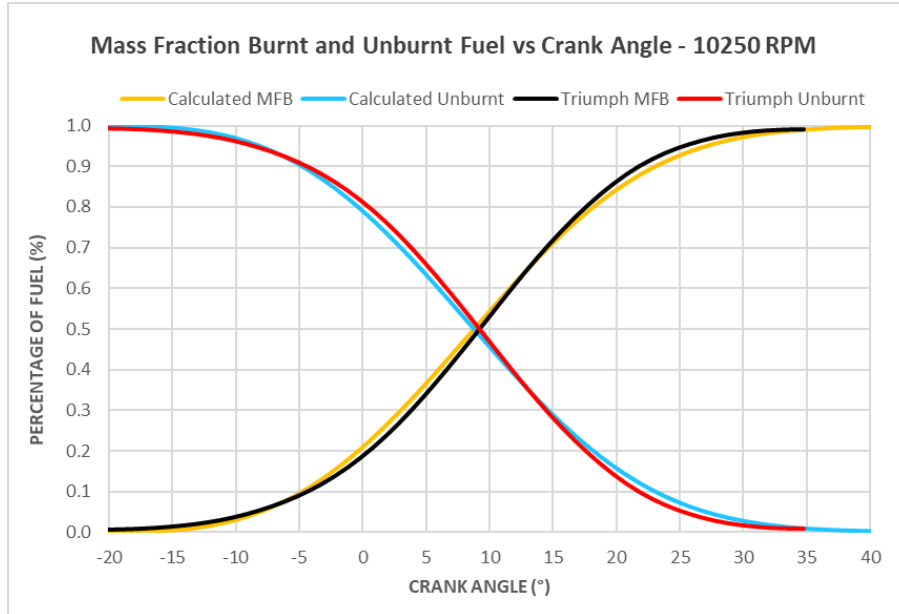


Figure 5 – Mass Fraction Burnt & Unburnt Fuel vs Crank Angle – 10250 RPM

It is also possible to calculate the burn rate of the combustion process by the rate of change in the Wiebe function at each degree of crankshaft rotation during combustion. This gives an indication to the rate the fuel is being consumed within the engine cylinder.

$$\text{Burn Rate (\%)} = \text{Wiebe}_2 - \text{Wiebe}_1$$

Equation 13 – Burn Rate Equation

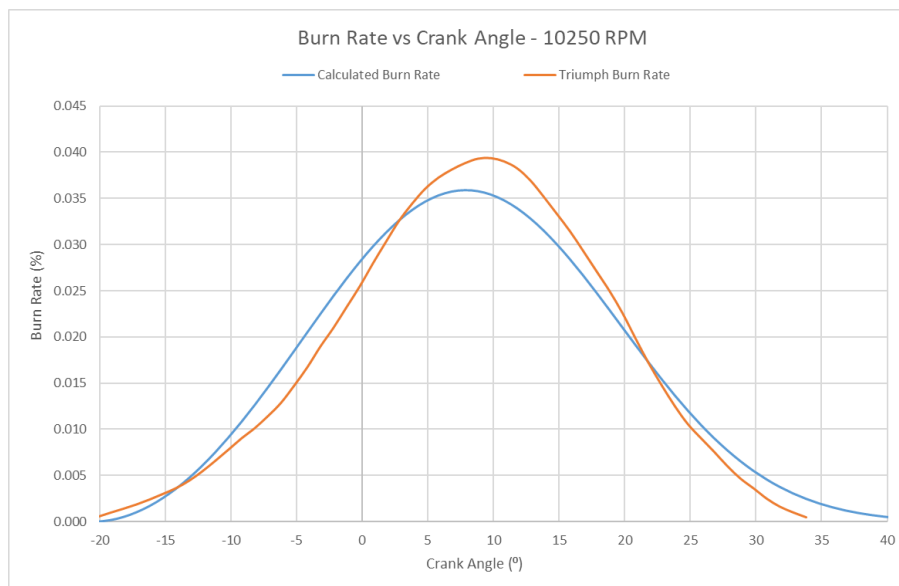


Figure 6 – Burn Rate vs Crank Angle – 10250 RPM

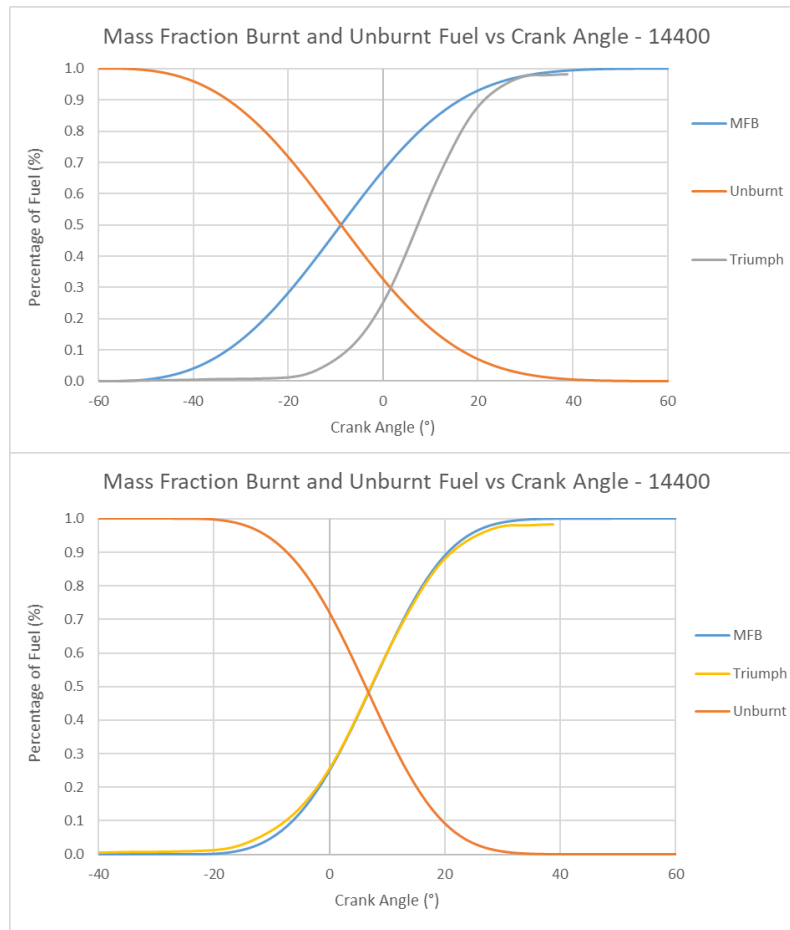


Figure 7 – Original (Top) & Modified (Bottom) Mass Fraction Burnt vs Crank Angle – 14000 RPM

Unfortunately, the method of matching the mass fraction burnt used for 10250 RPM, didn't work for 13000 or 14400 RPM's. This needed different exponents and combustion details in order to match more precisely.

Using the Log PV graphs for each RPM to determine both the combustion start and its duration, then entered into Table 1. The exponent 'm' for the Wiebe equation was then altered until the curve was as close as possible, since WAVE calculates the exponent 'a' based on the input for 'm', so it was kept to the standardised input '5'.

This is shown within Figure 7, where on the top you can see the original distance between the MFB and the Triumph data, by using the claimed burn duration and combustion start from a specification sheet provided from Triumph for this experiment. Whereas, at the bottom of Figure 7 is the matched data by using the modified inputs seen within Table 1. This process was repeated for 13000 RPM.

Standard Inputs		Modified Inputs		
Weibe a	5	Weibe a	5	
Weibe m	2	Weibe m	2.6	
n Comb	1.8	n Comb	1.36	
n Exp	1.15	n Exp	1.29	
Comb Start	-60.19	Comb Start	-26.19	Degrees
Ignition Delay	8	Ignition Delay	8	Degrees
Burn Duration	99	Burn Duration	58	Degrees

Table 1 – Standard & Modified Inputs for the Wiebe Equation – 14400 RPM

3.2. Ricardo WAVE

A WAVE model of the Triumph Daytona 675 was provided for this experiment, shown in Figure 8. The base model featured measured data of diameters and lengths for all ducts, Y-junctions and cylinders, but ‘.txt’ files comprising valve lift profiles and flow coefficient data was required to be loaded into the model for WAVE to be able to complete its calculations. The aim for this was to optimise the engine’s performance to reflect a realistic output from the Triumph.

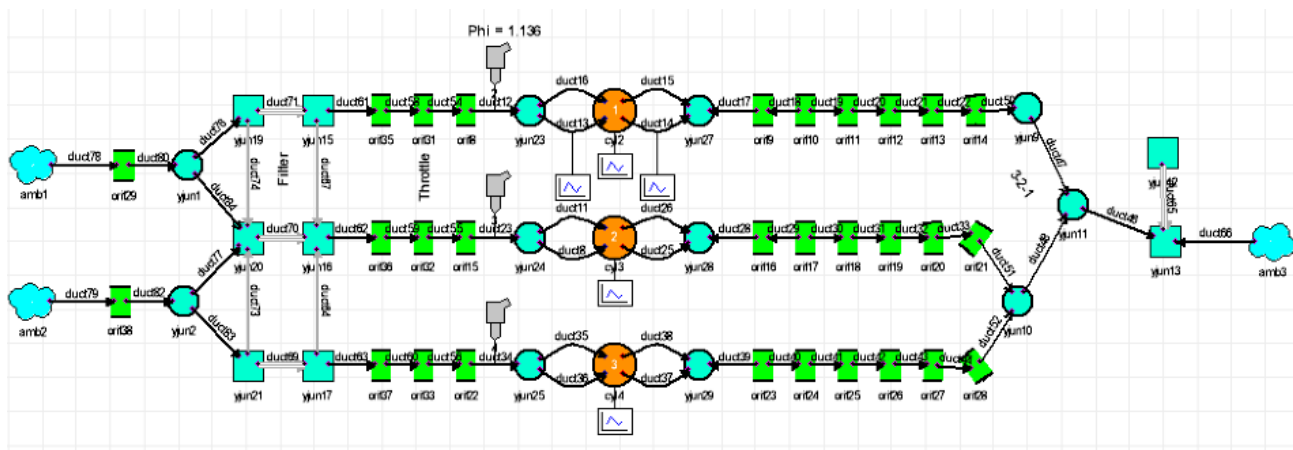


Figure 8 – Ricardo WAVE Interface & Triumph 675 Model

Modifications were made in the hope that the predicted data, produced from WAVES mathematical modelling, correlated to real measured data mapped within Excel. If the cylinder pressures could match, then the values for power and torque would theoretically match, depending on external assumptions made. Within the WAVE software there is a function in which you can assign multiple variables to each engine speed case that is to be simulated (Figure 9). From this table, the correct part of the model has to have the name assigned to it in order to vary the values across the studies. This is done because in reality, all the values would vary as the engine speed and internal temperature increases.

Constants Table

Number of Cases

3

Import...

Export...

Display Profile...

	Name	Units	Case 1	Case 2	Case 3	+
Status			Run	Run	Run	Skip
1	BDUR	deg	20	20	20	
2	eop	deg	85	85	85	
3	exh		180	180	180	
4	FARD		0.08	0.08	0.08	
5	icp	deg	250	250	250	
6	NCYC		20	20	20	
7	PREF		1.01325	1.01325	1.01325	
8	SPEED	rpm	14400	13000	10250	
9	TCYL		575	575	575	
10	THB50	deg	8	8	8	
11	THEAD		600	600	600	
12	thrt	mm	40	40	40	
13	TPIS		550	550	550	
14	TREF		298.15	298.15	298.15	
+						

Sweep Table

	Case 1	Case 2	Case 3
Variable 1			
Variable 2			

Edit Subcases

OK

Cancel

Help

Figure 9 – WAVE Model Constants Table

Using the base model without any changes, a benchmark run was calculated to base and compare all modifications to. The WAVE model results of cylinder pressures to those defined by Triumph (labelled as Excel) provided interesting results, seen in Table 2.

BASE			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	98.89	90.72	76.67
Difference	16.96	10.47	-8.15

Table 2 – Benchmark Max Cylinder Pressures - WAVE vs Excel

Currently the base model was over predicting the pressure seen within 10250 & 13000 RPM, but under predicted those in 14400, shown above. Now a benchmark had been made, optimisations could be carried out to ascertain a true reflection of the real Triumph.

3.3. Changes Made to Ricardo Model

This section details the changes made to the Ricardo WAVE model, the results of these are found in the appendix due to word count, within section 4.1.

3.3.1. Con Rod Length

The original WAVE connecting rod length was set to 98mm, which within the real engine is 104.6mm. As the con rod length increases, the lower its relative angle to the cylinder axis during crankshaft rotation becomes, by lowering the ratio of 'Crank Radius' (R) to 'Con Rod Length' (L) (R/L). The lower this ratio is, the less friction is applied to the cylinder wall by the piston, in turn decreasing drag and frictional losses throughout each cycle. (Ward, 2012)

3.3.2. 10, 50 & 90% MFB Locations & Burn Durations

To make the model realistic, WAVE utilises the Wiebe function in order to predict the MFB and heat release rate during combustion. The inputs required are burn duration, in this case 10 – 90%, not the usual 0 – 90% or 99% MFB, for which, different exponents are required for Equation 11. The Triumph data that was provided for the experiment included information on the percentage of fuel burnt per crank degree of rotation from -360° to 360°. Within Excel, the values above and below 10% burnt were selected for one RPM, then by using a linear interpolation equation to find the crank angle where 10% of fuel is burnt.

This process was repeated for both 50 & 90%, and the other two RPMs, with the values inputted into Table 3 across each engine speed. Equation 14 is only a linear interpolation equation and may not be the exact crank angle (CA) that the desired burn percentage (%) occurs at, but it's a good estimation.

$$\text{Burn CA} = (\text{CA Before}) + \left((\text{CA After} - \text{CA Before}) * \left(\frac{(\text{Required Burn \%} - \text{Burn \% Before})}{(\text{Burn \% After} - \text{Burn \% Before})} \right) \right)$$

Equation 14 – Desired Burn Percentage Crank Angle Linear Interpolation Equation

Mass Fraction Burnt Crank Angles (°)			
RPM	10250	13000	14400
10%	-4.30	-4.01	-7.54
50%	9.24	10.52	7.23
90%	21.80	24.23	21.23
Duration	26.10	28.24	28.77

Table 3 – Mass Fraction Burnt Crank Angles

With the 10 & 90% burn crank angles ascertained, the difference between them equated to the burn duration that WAVE required, so using the constants table (Figure 9), these were inputted for each RPM under "BDUR" and the 50% MFB crank angle was entered as "THB50".

3.3.3. Changed Fuel/Air Ratio to Same for All Injectors

Within the original model, one of the injectors had a slightly different Fuel/Air mass ratio value inputted, this was corrected by referencing a constant “FARD”, as seen within Figure 9. This also meant that the ratio could be altered with each RPM, much like a real ECU would do to optimise performance.

As shown below, the original ratio within the injectors was 0.078, but within the constants table, the default ratio is set to 0.08. For this section, all RPM's and injectors were kept to the same ratio to observe the effect the small difference had upon the pressures within the cylinders. Optimisation would be carried out after other changes had been made to the WAVE model.

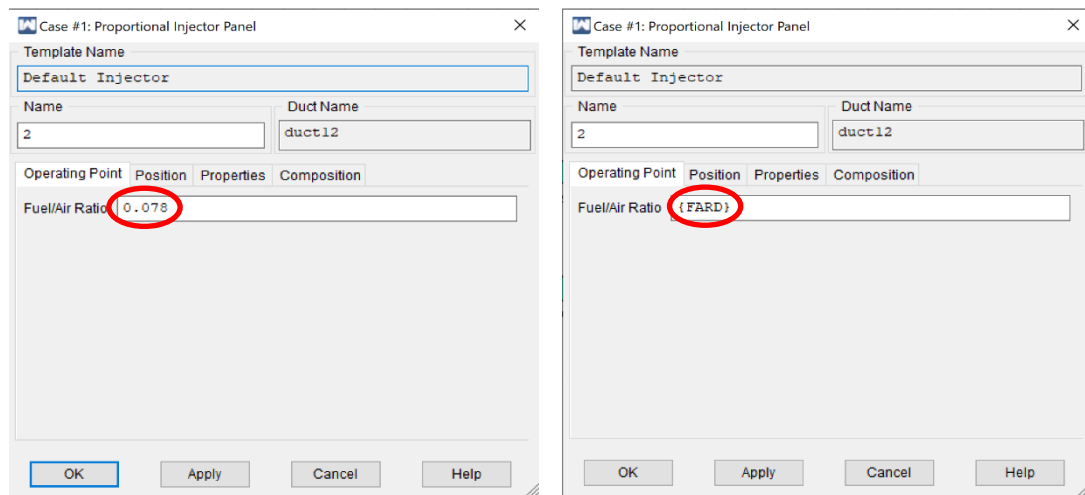


Figure 10 – WAVE Injectors Properties – Before & After Adding Fuel/Air Ratio Reference

3.3.4. Recalculated Coolant-Side Area

Originally, the coolant-side area was at a value of zero, by pressing the calculator buttons to the right of the properties panel, this would automatically be calculated by WAVE to provide accurate and efficient calculations associated with conduction.

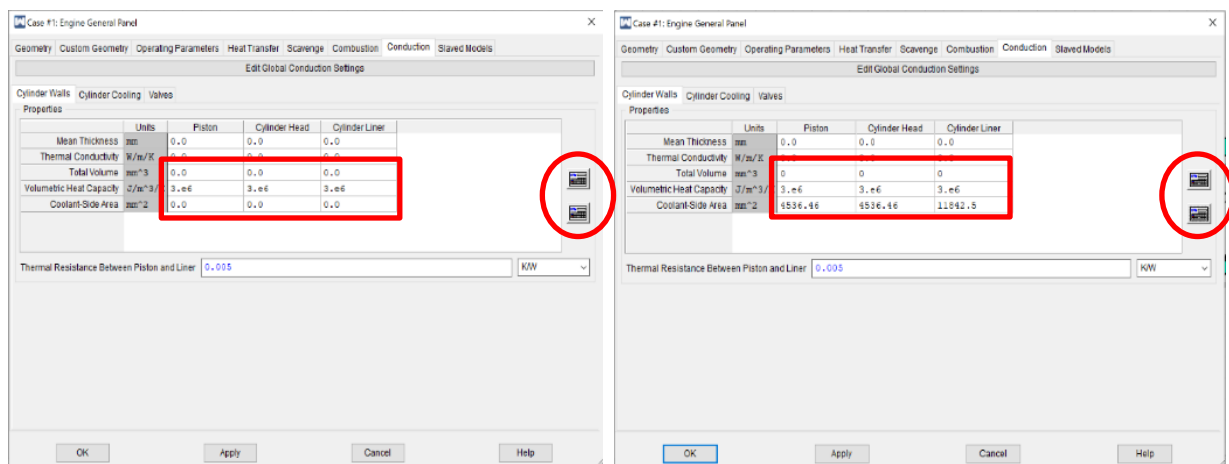


Figure 11 – WAVE General Engine Conduction Control Parameters – Before & After Recalculating Coolant-Side Area

3.3.5. Recalculated Volume & Skin Friction Area of Junctions

Throughout every complex Y-junction, the calculated volume and heat transfer/skin friction area had incorrect values. As previously done, these values needed to be recalculated in order for WAVE to compute a realistic representation of the virtual engine.

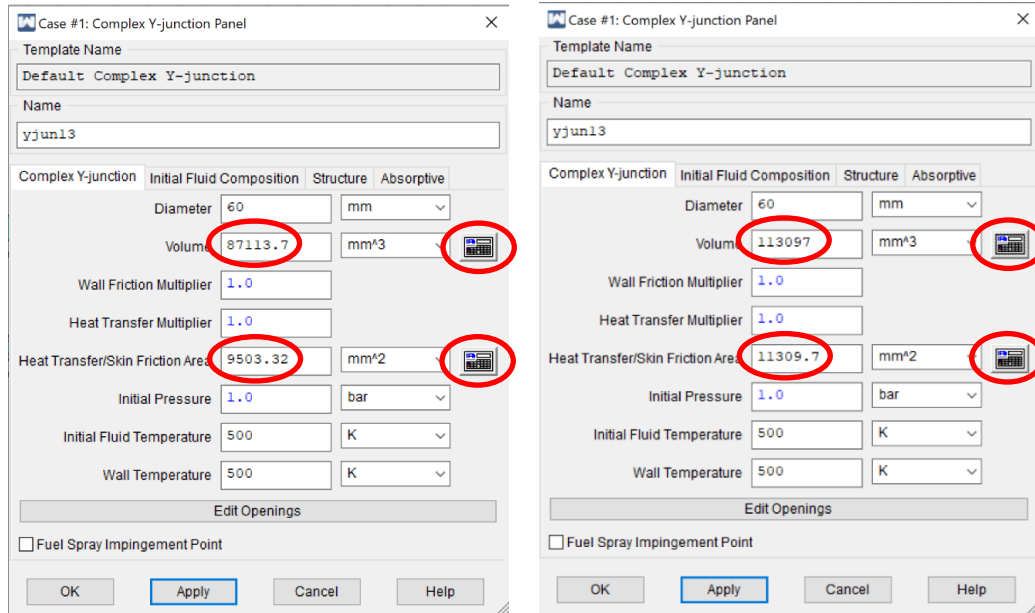


Figure 12 – Complex Y-Junction Parameters Panel – Before & After Recalculating Volume & Skin Friction

3.3.6. Changed Induction Components' Air Pressure To "PREF"

Throughout the WAVE model, every component of the induction system had an initial condition regarding pressure, set to 1 bar. This was changed to a variable "PREF" so that it could be changed and regulated from one variable within the constants table.

Initially, all components were set to 1 bar, but after completing a run, it was noticed that peak cylinder pressures dropped, this resulted in the ambient elements in the model transferred back to the original 1.065 bar, which saw the pressures rise back to their original value (4.1.7).

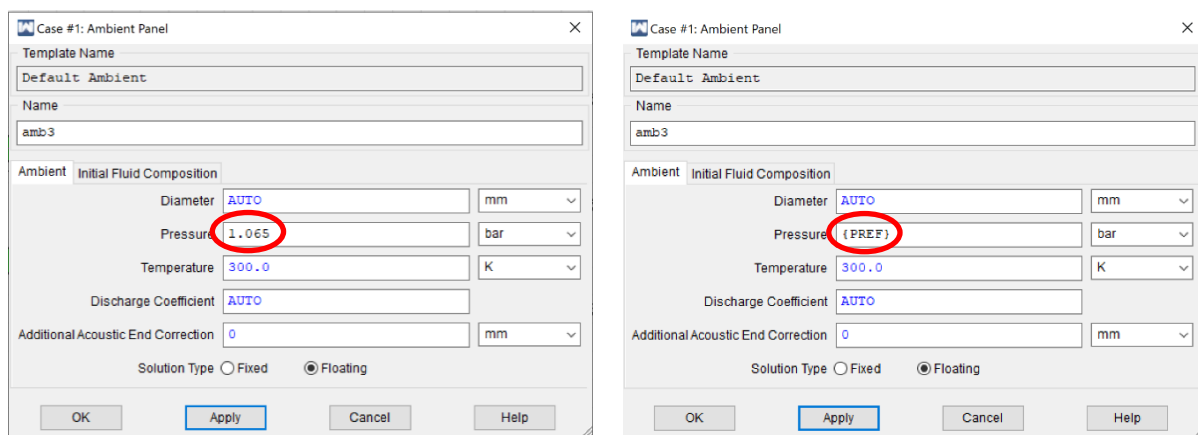


Figure 13 – Ambient Elements Parameters Panel – Before & After Changing Initial Pressure Condition to "PREF" Constant

3.3.7. Changed Induction Components' Air Temperature To "TREF"

As discussed previously, it is significantly easier to refer to and change a variable within the constants table, than to go through and change it within 30 different components. The initial temperature of all the induction components now referred to a variable called "TREF" which was initially set to 298.15K (25°C), where the default value of 300K within each component.

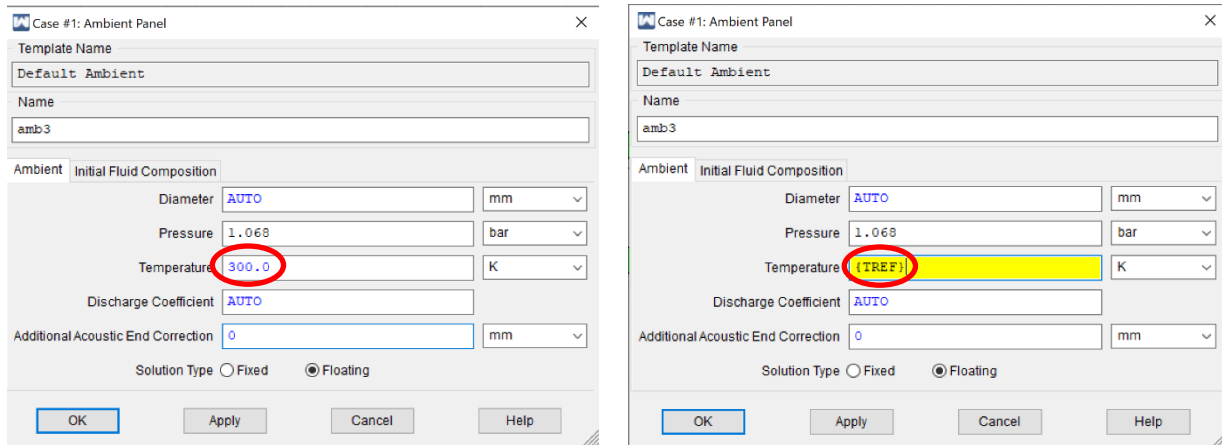


Figure 14 – Ambient Elements Parameters Panel – Before & After Changing Initial Temperature Condition to "TREF" Constant

3.3.8. Changed Junction and Duct Diameters to Stop Computational Errors

During computation of the WAVE model, it was noticed that the errors shown in Figure 15 had appeared since recalculating the volumes of the complex Y-junctions. After identifying the affected components, the steps described below were taken to rectify these errors.

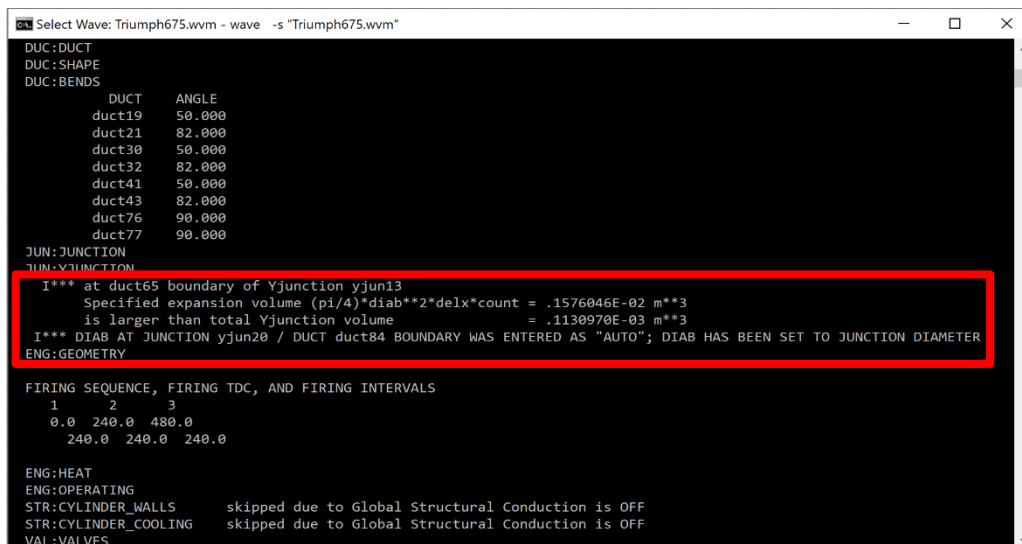


Figure 15 – WAVE Computational Interface with Duct & Junction Errors Highlighted

Beginning with the airbox component 'junction 20', the error defined that the opening connection to 'duct 84' was set to 'auto' and required a defining value. By using the 101.7mm previously set on the other joining ducts, the issue was resolved.

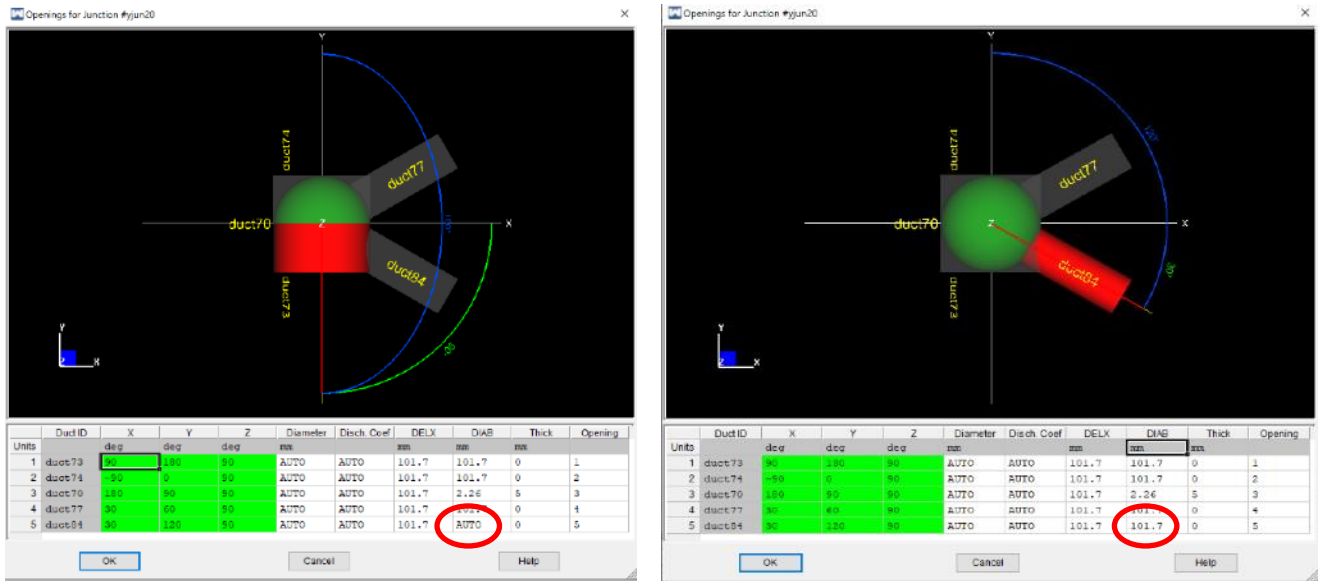


Figure 16 – WAVE Openings for Y-Junction 20 – Before & After Defining the Diameter

The next error dictated that the volume within ‘duct 65’ was greater than that of ‘Y-junction 13’. The duct served as an opening between the two junctions (12 & 13), where ‘junction 12’ was representative of a silencer/expansion chamber for the exhaust gases. By using Equation 15, it was possible to determine what the volume should be to remove the error generated by Ricardo. By changing rearranging the equation, so that diameter was the subject and using the volume WAVE says it should be, the new diameter was calculated for the duct and the error removed.

$$Volume = \left(\frac{\pi}{4}\right) * Diameter^2 * DelX * Count$$

Equation 15 – Ricardo Volume Equation

$$Diameter = \sqrt{\frac{Volume}{\left(\frac{\pi}{4}\right) * DelX * Count}}$$

Equation 16 - Ricardo Volume Equation Rearranged

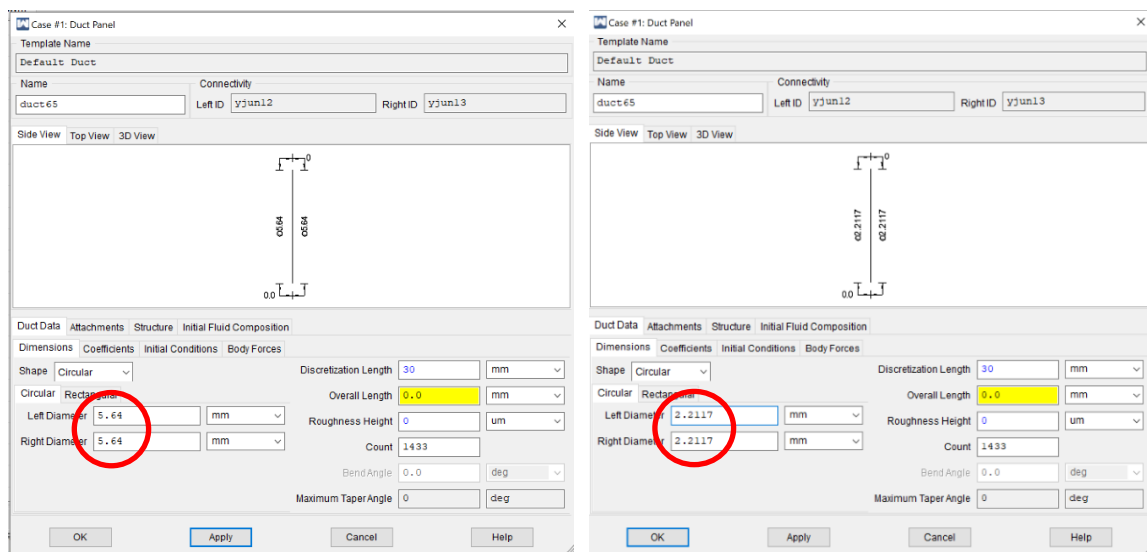


Figure 17 – WAVE Parameters for Duct 65 – Before & After Entering the Corrected Diameter

3.3.9. Reset Heat Transfer Multipliers to Default

Within the supplied WAVE model, altered values of heat transfer multipliers were featured, the exact operation and extent of these remained unknown, so in the attempt to gain a realistic model, these were restored to the default values, as recommended by WAVE as '1', to eliminate any negative or unwanted interference.

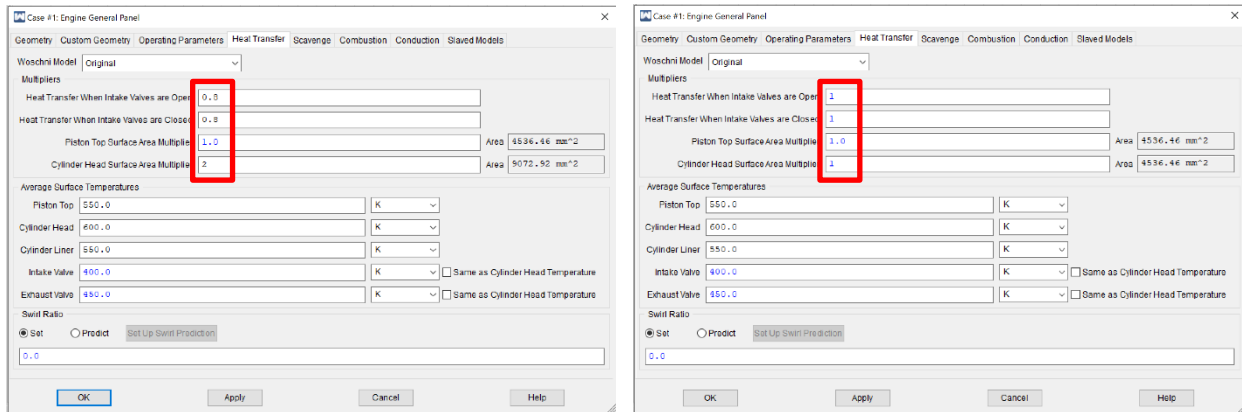


Figure 18 – WAVE Engine Heat Transfer Parameters – Before & After Resetting the Multipliers to Default Values

3.3.10. Changed Valve Diameters

When first conducting this experiment, a data sheet was supplied featuring various information about the Triumph Daytona, one of which was the diameters of the inlet valves. This was stated as 31mm, but inside the WAVE model, the dimension was designated as 29mm. To attain accurate flow simulations, this was altered to the correct value.

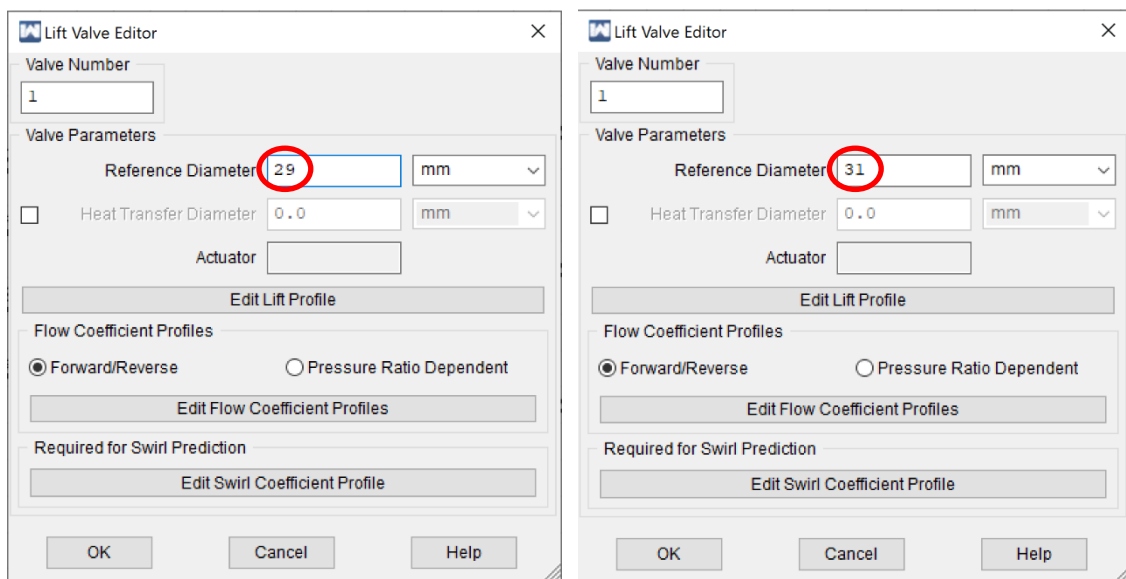


Figure 19 – WAVE Valve Parameter Editor Panel – Before & After Correcting the Reference Diameter

Because the dimensions of the valves had been altered, the diameters of the corresponding inlet ports would also need to be updated, this enabled WAVE to correctly simulate the fluid dynamics within both the induction system and cylinders within the engine.

The original values for the port were 28.3mm, and since the valves' diameters were increased by 2mm, the same increase would be given to the port dimensions, continuing the previous ratio for continuity.

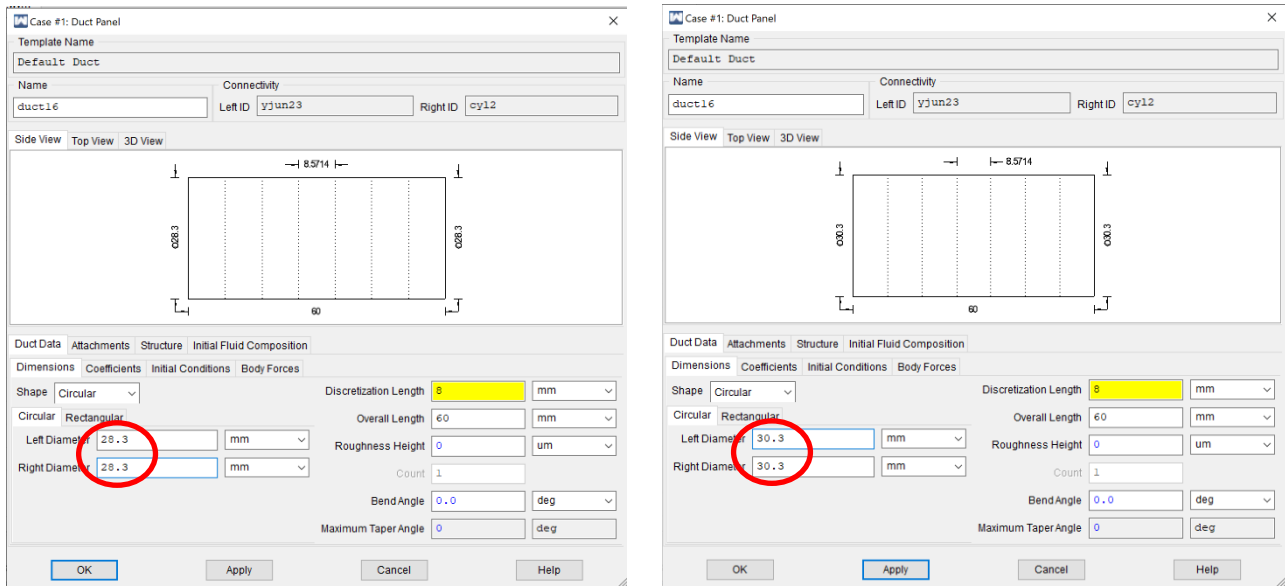


Figure 20 – WAVE Duct Parameters Panel – Before & After Adjusting the Diameter to Match Those of The Valves

3.3.11. Changed Fuel/Air Mass Ratio for Each RPM

Optimisation of the Fuel/Air mass Ratio Definition (FARD) was now undertaken to determine the correct ratio that increased the cylinder pressures to match those determined by Excel. If the pressures calculated by Excel, could validate those within WAVE, it would confirm the performance output could potentially match the power produced by the engine, or at the very least, corroborate the position and amplitude of the Maximum Brake Torque (MBT) produced by the engine.

Although MBT is largely due to ignition timing, the ratio of air and fuel can drastically alter how quickly the flame front within the cylinder travels, significantly defining the maximum brake power and minimum brake specific fuel consumption of the engine (Heywood, 1988).

A constant AFR was used for the three RPM's under analysis, previously defined in WAVE as 0.08. Now, each RPM would receive a separate value to simulate as higher a boost in pressure as realistically achievable. Numerous runs were carried out on each RPM separately, adjusting the ratio from 0.08 in as minute increments as possible to attain gains in peak cylinder pressure.

Fuel / Air Ratio - Optimisation			
Fuel / Air Ratios Used			
RPM	10250	13000	14400
FARD	0.074505	0.06805	0.10

Table 4 – Fuel / Air Ratios Used for Each RPM

4) Results & Analysis

4.1. Results from Changes to the WAVE Model

4.1.1. Benchmark Model

BASE			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	98.89	90.72	76.67
Difference	16.96	10.47	-8.15

Table 5 – Benchmark Max Cylinder Pressures - WAVE vs Excel

The benchmark results have been supplied here for easier comparison between these results and those of the changes made to the WAVE model.

It can be observed that for 10250 & 13000 RPM, the predicted peak pressures are higher than calculated and will need to be decreased throughout the study, but 14400 RPM needs to be increased to match the supplied data.

4.1.2. Con Rod Length

Con Rod Length			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	99.01	90.83	76.82
Difference	17.08	10.58	-8.00

Table 6 – Max Cylinder Pressures Due to Updated Connecting Rod Length

By changing the length of the connecting rod, each RPM's maximum pressure increased slightly. The increase in pressure shown, was proportional to the increase in engine speed, where 10250 RPM saw the smallest increase and 14400 RPM saw the largest.

Although the compression ratio was kept the same as stated (12.65:1) (Motorcycle Specs, 2020) so this should not have interfered in the calculations of the virtual model.

4.1.3. 10%, 50% & 90% MFB Locations & Burn Durations

The locations of 50% burn location and the duration of the burn were entered separately to acknowledge and analyse the effect that change of each component had upon the results determined by WAVE.

Burn Duration			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	92.00	84.87	73.00
Difference	10.07	4.62	-11.82

Table 7 – Max Cylinder Pressures Due to Updated Burn Duration

After entering the burn durations calculated within Excel, the peak pressures dropped drastically. This helped bring 10250 & 13000 RPM's down to a more reasonable number and closer to their target, but unfortunately, it did not help with the endeavour of increasing peak pressure at 14400 RPM.

50% Burn Location			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	88.26	78.67	74.57
Difference	6.33	-1.58	-10.25

Table 8 – Max Cylinder Pressures Due to Updated 50% Burn Locations

The 50% burn locations further aided the pursuit of matching peak pressures. 10250 & 13000 RPM's dropped in pressure, but 13000 RPM went too far and was now lower than those Excel calculated. 14400 RPM increased only slightly, but every little helps.

4.1.4. Changed Fuel/Air Ratio to Same for All Injectors

Fuel / Air Ratio - Same Each RPM			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	88.08	78.35	73.99
Difference	6.15	-1.89	-10.83

Table 9 – Max Cylinder Pressures Due to Fuel / Air Ratio

By changing the FARD in each injector to the same value, each RPM saw a slight decrease in pressure, most likely due to changing the FARD from 0.078 to 0.08. Although this would be optimised further on, it still would've been better to see a step in the right direction when trying to match these values, rather than the other way.

4.1.5. Recalculated Coolant-Side Area

Coolant Side Area			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	85.61	80.69	74.80
Difference	3.68	0.44	-10.01

Table 10 – Max Cylinder Pressures Due to Recalculation of Coolant-Side Area

Through a simple recalculation of the coolant-side area, the peak pressure within 10250 RPM decreased, whereas, 13000 & 14400 RPM's increased. 13000 RPM was now within 1 bar of its goal, and the recalculation put both 10250 & 14400 RPM's closer to their mark.

4.1.6. Recalculated Volume & Skin Friction Area of Junctions

Recalculated Junction Volumes & Areas			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	84.88	78.57	74.79
Difference	2.95	-1.68	-10.03

Table 11 – Max Cylinder Pressures Due to Recalculated Junction Volumes & Areas

Once the recalculations of all the complex Y-junctions had completed, the effect this had on the peak pressure was analysed. 10250 RPM was the one to see a positive reaction, decreasing the pressure slightly to the required value. Both 13000 & 14400 RPM's also saw a decrease in pressure, but with how close the previous iteration of 13000 RPM was, it now decreased too far below target. 14400 RPM only reduced a minute amount which was, unfortunately, further away from the objective.

4.1.7. Changing Induction Components' Air Pressure To "PREF"

Two studies were carried out on changing the inlet pressures to include a reference within the constants table.

1. All components within the induction system used 1 bar of initial pressure – (Ambients, Ducts & Junctions).
2. All components within the induction system used 1 bar of initial pressure – except Ambients.

Reference Pressure For Inlet - All 1 Bar			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	80.88	73.99	68.09
Difference	-1.05	-6.26	-16.73

Table 12 – Max Cylinder Pressures When All Induction System Components Used 1 Bar Pressure

Using a pressure of 1 bar throughout all of the components, saw a significant decrease in cylinder pressures that appear to be proportional to the increase in engine speed. The components all originally had pressure set to 1 bar, except the Ambients.

Reference Pressure For Inlet - Ambient = 1.065			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	85.20	79.10	75.26
Difference	3.27	-1.15	-9.56

Table 13 – Max Cylinder Pressures When All Induction Components Except Ambients Used 1 Bar Pressure

It can be observed that once the Ambient components were returned to their original value of 1.065 Bar, the peak pressures seen throughout each RPM has only slightly altered to what they were before, seen within Table 11.

Both tables above, show the effects caused by slightly altering the ambient pressure. The differences between Tables Table 12 & Table 13 are all due to the change in ambient pressure before the induction elements. This implies a form of ram air effect, where the higher the pressure is, the higher the peak cylinder pressures will be.

But if this was the case, the timing of ignition would need to be adjusted so that 'knock', or pre-ignition, does not occur and to also optimise both the performance and efficiency of the engine.

4.1.8. Changing Induction Components' Air Temperature To "TREF"

Reference Temperature For Inlet - All 300K			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	85.59	80.69	75.40
Difference	3.66	0.45	-9.42

Table 14 – Max Cylinder Pressures Due to Changing Air Temperature to 300K

Previously, every component of the induction system had a default temperature of 300K (27°C), using a variable within the constants table set to 300K, all of the components parameters were changed to use this variable throughout each calculation. This meant that physically, nothing within the engine had changed, but it was noticed, in Table 14, that each RPM actually increased in pressure, albeit marginally.

A small study was conducted to determine the effects made upon the pressures, if the temperature was raised and lowered. The results of which are shown below.

Reference Temperature For Inlet - All 315K			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	88.59	83.09	77.16
Difference	6.66	2.85	-7.66

Table 15 – Max Cylinder Pressures Due to Changing Air Temperature to 315K

Reference Temperature For Inlet - All 285K			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	83.29	78.69	72.64
Difference	1.36	-1.55	-12.18

Table 16 – Max Cylinder Pressures Due to Changing Air Temperature to 285K

As seen in Tables Table 15 & Table 16, cylinder pressures changed sporadically with air temperature. From the original 300K, 13000 RPM had the smallest difference between the Excel and WAVE pressure data. But provided the middle ground for 10250 & 14400 RPM's. Raising the temperature provided the worst result for 10250 but the best for 14400, and lowering the temperature, gave the best result for 10250 but the worst for 14400. This shows the effect that the air temperature has upon its density, and how density can affect pressure.

It was decided that for each case, the temperature would remain at 300K, this was the middle ground for each RPM to provide a good basis on which to fine tune at a later time.

4.1.9. Changed Junction and Duct Diameters to Stop Computational Errors

Changed Duct Diameters to Stop Errors			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	85.39	78.81	76.77
Difference	3.46	-1.44	-8.05

Table 17 – Max Cylinder Pressures Due After Errors Have Been Removed

Since changing the volume and diameters of the duct, the pressures altered differently for each RPM than those in Table 14. 14400 RPM saw an increase in pressure of 1.37 bar, whilst 10250 & 13000 RPM's saw a decrease, where 10250 changed by 0.2 bar, and 13000 decreased by 1.88 bar.

This implies that the exhaust system of the Triumph may be pulse tuned for reducing back pressure, pulling exhaust gasses out with a vacuum effect, which in turn pull fresh air into the cylinder. Because the only junction that saw this change was in the silencer of the exhaust, so it is interesting to see how a minute change of 1mm diameter can affect the pressures within the cylinder during combustion.

4.1.10. Reset Heat Transfer Multipliers to Default

Heat Transfer Multipliers to Default			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	85.72	79.09	76.05
Difference	3.79	-1.16	-8.77

Table 18 – Max Cylinder Pressures Due to Resetting Heat Transfer Multipliers

By manipulating purely the heat transfer multipliers, for open & closed valves, along with the piston and cylinder head surface areas, that govern the Woschni heat transfer model that Ricardo uses to calculate the effective temperatures inside and outside the components of the model, including the wall temperatures of said components. Although the Woschni model is primarily used for Diesel engines, WAVE uses it as it's default method of heat transfer calculation. (Woschni, 1965)

With those multipliers reset to default, the pressures at 10250 & 13000 RPM saw an effective increase, whilst 14400 saw a decrease in pressures.

It appears these numbers refine the model to help validate the results calculated by WAVE to match real data or alter the WAVE model to account for variables within the real world that are hard to predict or model within purely mathematical conditions.

4.1.11. Changed Valve Diameters

Inlet Valve & Port Sizes			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	88.15	77.08	76.28
Difference	6.22	-3.17	-8.54

Table 19 – Max Cylinder Pressures Due to Changing Diameter of Valves & Ports

As stated previously, the valve diameters within the Ricardo model were incorrect. By entering the appropriate values, the 'choke-point' within the valve/port assembly has changed, meaning so too has the amount of air that can flow through the valve and into the cylinder during the engines' operation.

The faster the engine speed, the faster the air is being pulled through the inlet ports. Since the principle governing factor within all aerodynamic equations is velocity, the accumulation and size of the boundary layers formed on the walls of the port are directly related to how fast the air is travelling, i.e. the faster the air travels, the smaller the area it is able to pass through becomes, limiting both volumetric efficiency and engine performance. (NASA, 2015)

As seen above, the difference between Excel and WAVE in 10250 RPM doubled in amplitude, and 14400 increased by 0.2bar. Whereas, 13000 RPM actually dropped in pressure by 2 bar which is surprising as it was expected that with the bigger diameter, more air would be able to enter the cylinder during the induction cycle, and therefore, more pressure during combustion.

4.1.12. Changed 'FARD' For Each RPM

Fuel / Air Ratio - Optimisation			
Max Cylinder Pressures			
RPM	10250	13000	14400
Excel	81.93	80.25	84.82
WAVE	82.06	79.58	81.85
Difference	0.13	-0.67	-2.97

Table 20 – Max Cylinder Pressures Due to Fuel / Air Ratio Optimisation

During a previous iteration, the Fuel/Air Ratio Definition (FARD) was set to a constant variable, regardless of engine speed. Now however, an investigation began into finding the best ratio for each engine speed. By manipulating purely the value of FARD within the constants table, the above pressures were the best that WAVE was able to replicate.

To result in a reduced peak pressure for 10250 RPM, the FARD required to be raised so that it was running a richer ratio, meaning more fuel inside the cylinder, reducing the pressure. This stems from the Ideal Gas law; $PV = mRT$ where if V , R & T are constant, an increase in the fuel moles (m), leads to a decrease in pressure (P) to balance the equation. The opposite occurred for 13000 & 14400 RPM's where the ratio was reduced to increase the pressure.

4.2. Volume

To ensure that the data exported from WAVE was accurate, the volume from each RPM along with the original Triumph volume was exported and plotted against crank angle.

As expected, each line overlays the first, leaving only the top line visible, meaning that the volumes are identical. This validates the volume, showing that during the 'Logarithmic Pressure vs Volume' graph comparisons, the pressure data is the only changing variable, showing the differences throughout each RPM.

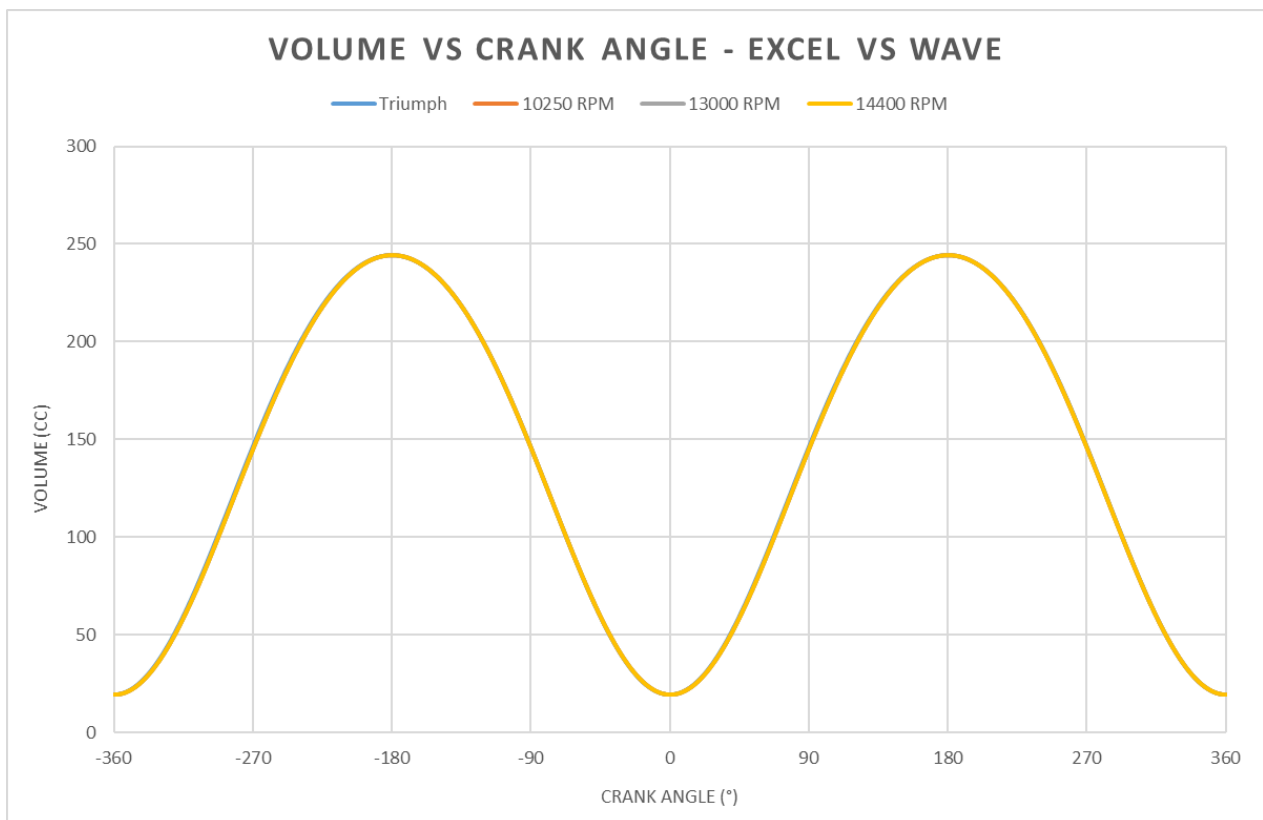


Figure 21 – Volume vs Crank Angle – Excel vs WAVE Comparison

4.3. Pressure

The pressure data from each RPM was plotted, consisting of both the Excel and WAVE data as a visual aid in comparing the differences between the two and determining how far away the predicted pressure data was from that of the original measured data.

By analysing the three graphs below, it is evident how closely the data matched on 10250 & 13000 RPM's, although the Excel data has a slightly steeper gradient during the incline up to peak pressure.

10250 RPM appears as though it is 1 or 2 degrees behind Excel on its entire graph causing a slight offset of the Triumph data, although they are identical otherwise (Figure 22).

13000 RPM follows the same delay as 10250 on the increase but matched the decrease perfectly until 20 bar where the Triumph data has a sharper rate of descent (Figure 23).

14400 RPM features an initial delay during the increase but then matches between 25 & 60 bar. Because it doesn't peak to the same extent as the Triumph, the rate of increase lessens, but then matches nearly perfectly on the descent (Figure 24).

All three graphs feature similar trends, such as the small kink in the WAVE data at around 150° ATDC until around the 225° mark, which is when the exhaust valve is opening. The modelling of this phase is also shown in the Log PV pumping phases.

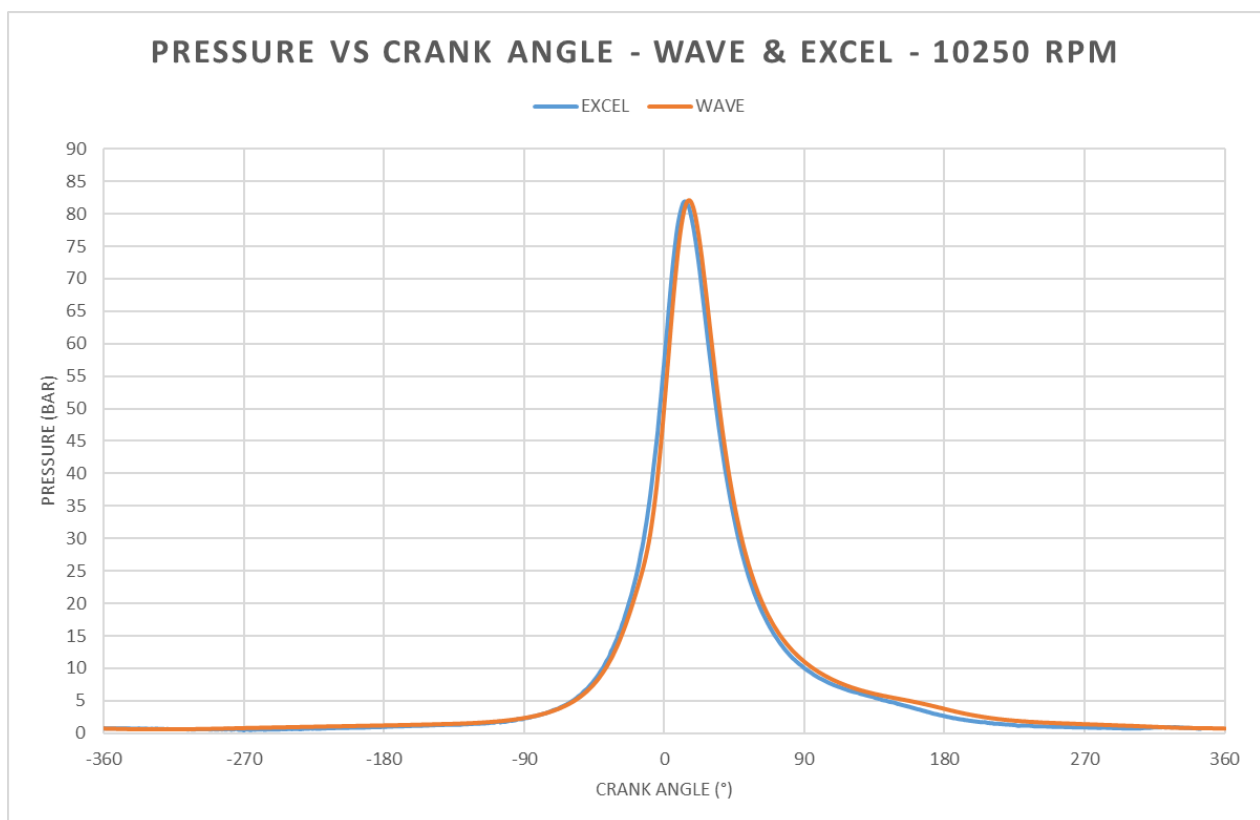


Figure 22 – Pressure vs Crank Angle – 10250 RPM

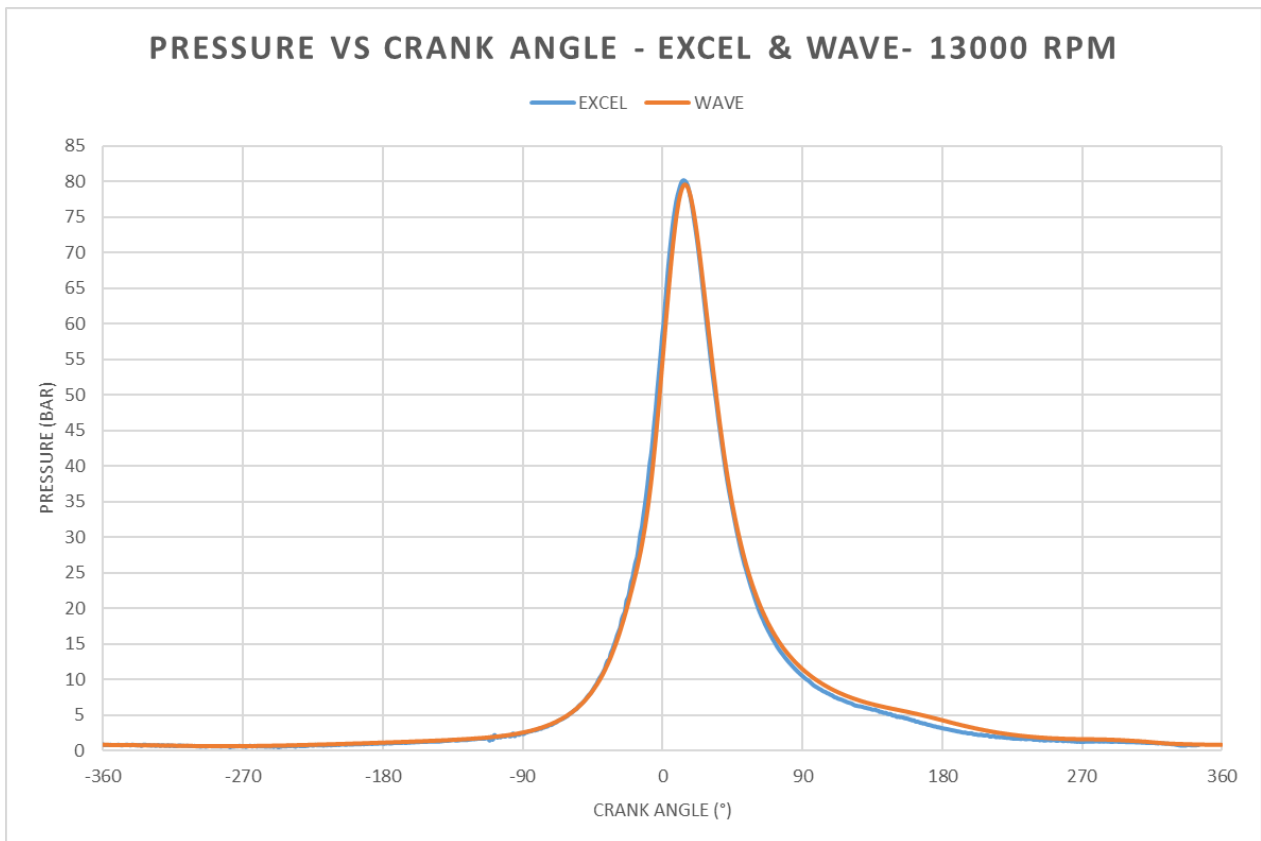


Figure 23 – Pressure vs Crank Angle – 13000 RPM

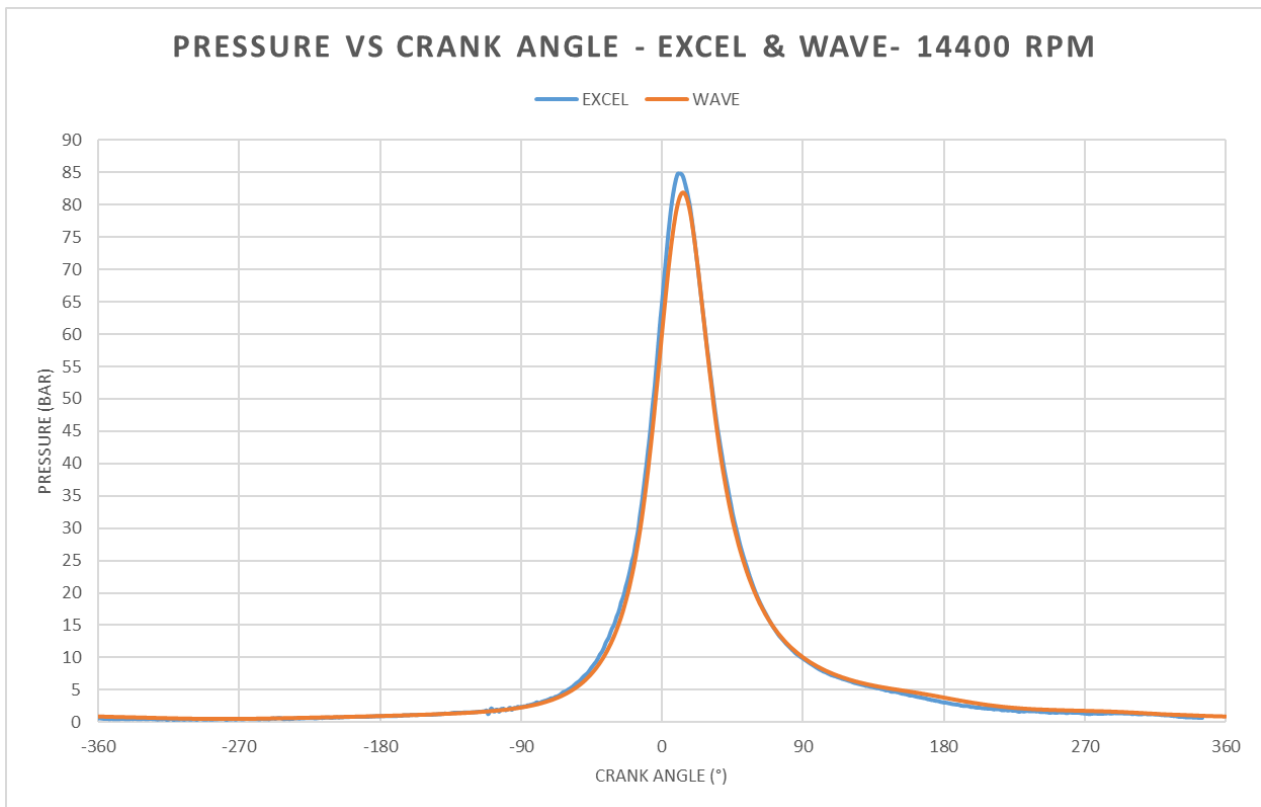


Figure 24 – Pressure vs Crank Angle – 14400 RPM

4.4. Logarithmic Pressure vs Volume (Log PV)

By producing a Log PV graph, the entire engine cycle can be analysed to determine specific characteristics and understand what is happening at each phase of the cycle. It should be noted that WAVE cannot replicate the pumping losses so only replicates to a realistic extent.

Although the pressure vs crank angle graph for 10250 RPM is nearly identical (disregarding the slight offset), the Log PV graph tells a different story. The gradient of the compression stroke is not steep enough to match that of the Triumph and the combustion stroke predicted by WAVE doesn't match the rate of the Triumph, this means that the burn rate is too slow than that of reality. The gradient of the expansion stroke appears to be identical to that of the triumph, showing that both close to an adiabatic isentropic process (Figure 25).

For 13000 RPM, the gradient of compression is closer to the Triumph than that of 10250. The combustion according to WAVE, again, seems to burn at a lower rate, even when entering in correct start combustion and burn duration values. At the end of combustion, the two data sets match until about half-way down the expansion phase, where the difference in gradients increases. The tip of the pumping losses during TDC after the exhaust stroke pinpoints where the Triumph's end, so this graph is not a bad prediction (Figure 26).

With 14400 RPM, seems to match perfectly in some places, and terribly in others. During the compression stroke the gradient of the lines are extremely similar but there's an undeniably big gap between the lines. The burn rates of combustion are similar, but the tip of the curves have the same gap as with the compression, but as the combustion phases into the expansion, these lines are almost identical. If the tip of the pumping losses of WAVE had a lower log P value, the entirety of the lower section would be a good representation, disregarding the noise (Figure 27).

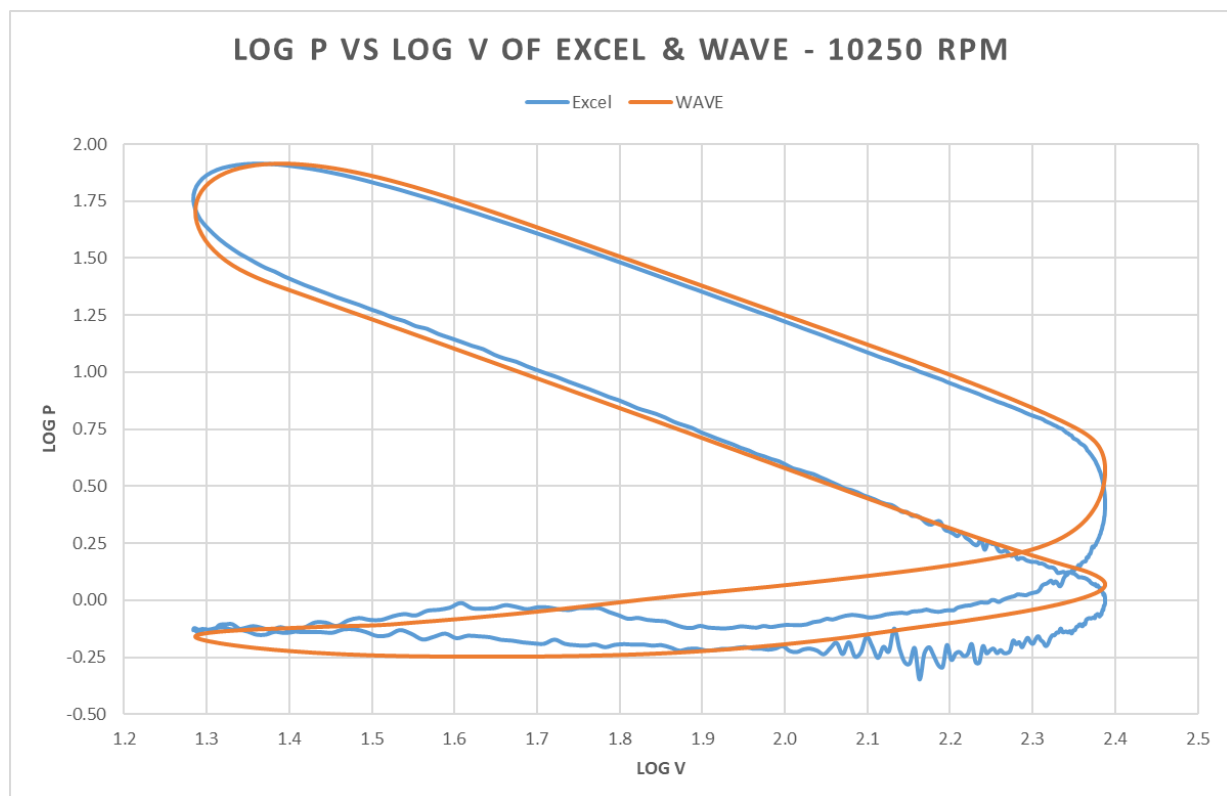


Figure 25 – Logarithmic Pressure vs Volume Comparing – Excel & WAVE

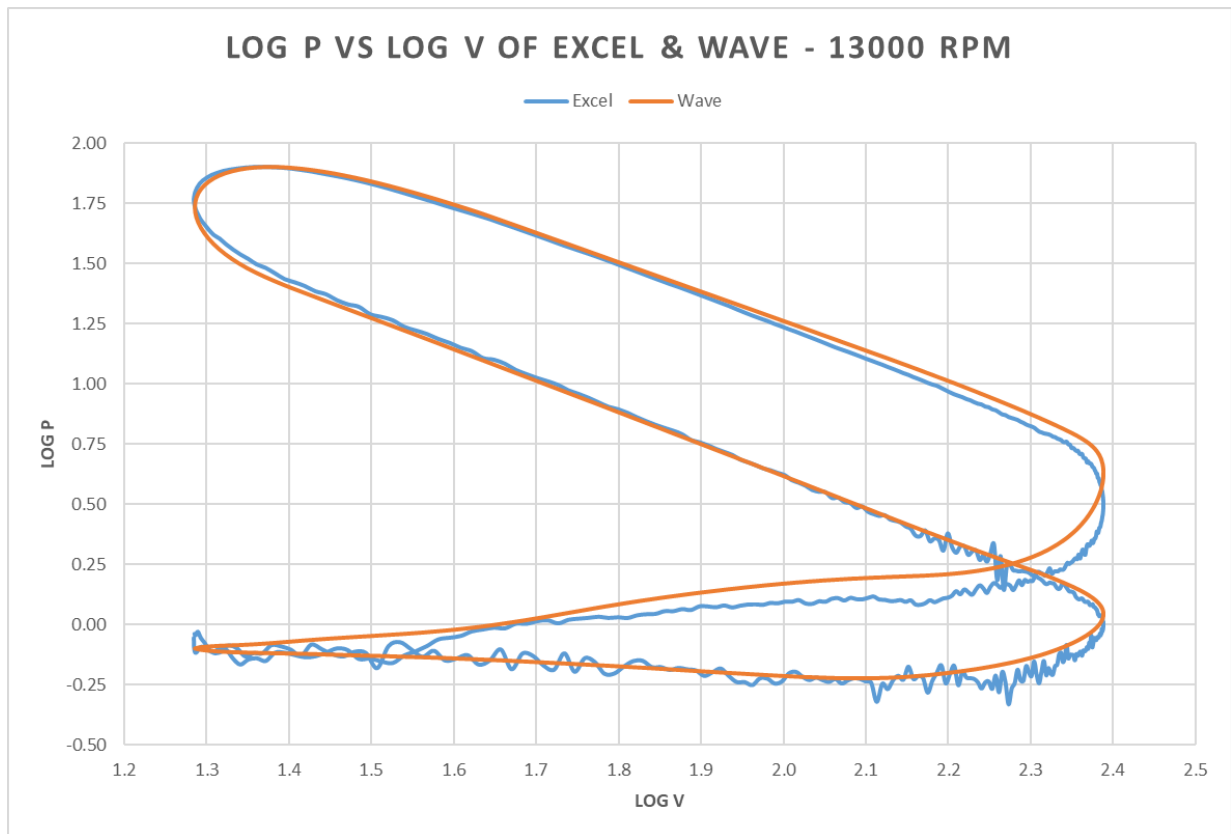


Figure 26 – Logarithmic Pressure vs Volume Comparing – Excel & WAVE

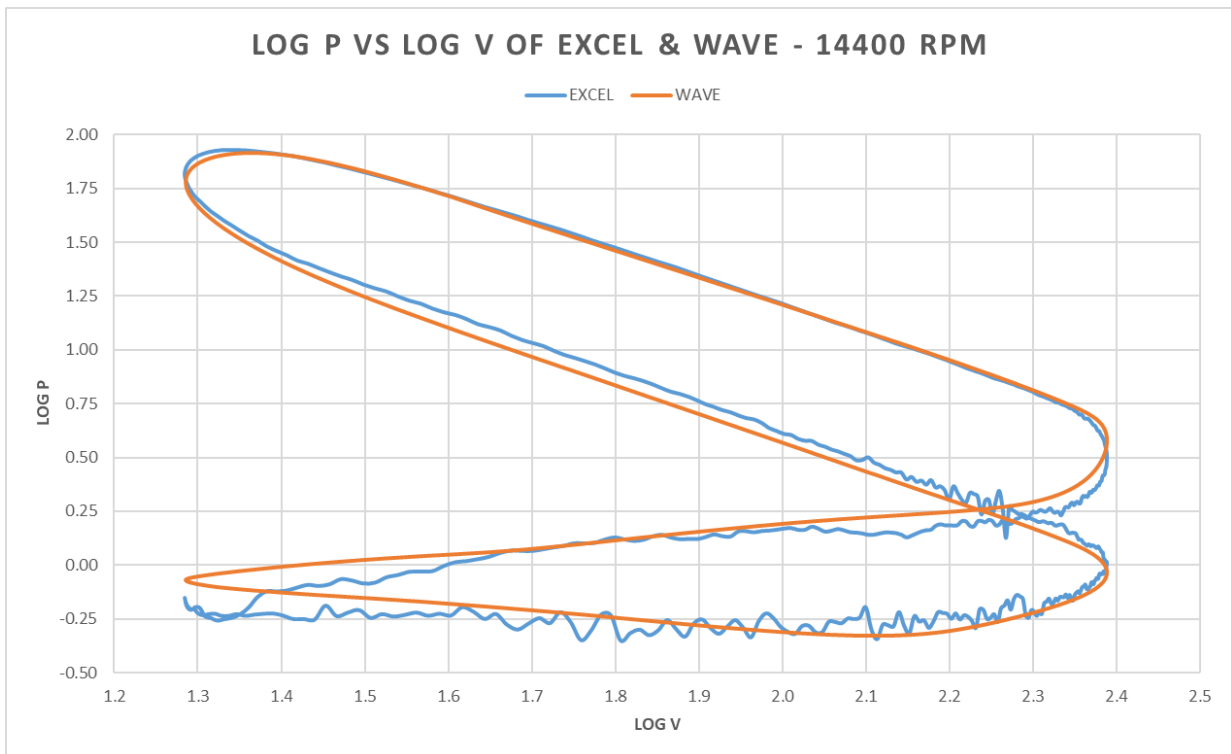


Figure 27 – Logarithmic Pressure vs Volume Comparing – Excel & WAVE

4.5. Power & Torque

From calculating the power produced at each RPM within Excel, and from the exported data from WAVE, it was possible to determine the torque produced by the Triumph data with both programs, using either of the equations below depending on what units for power you input.

$$Torque (Nm) = \frac{Power(kW) * 10^3}{RPM * \frac{2\pi}{60}}$$

Equation 17 – Calculating Torque from Power Equations

Now with both power and torques computed, it was possible to plot these results against crank angle for a visual comparison.

Power (kW)				Torque (Nm)			
RPM	10250	13000	14400	RPM	10250	13000	14400
Excel	83.19	104.31	107.47	Excel	77.50	76.62	71.27
WAVE	82.05	90.02	80.04	WAVE	76.44	66.13	53.08
Difference	1.14	14.29	27.43	Difference	1.06	10.49	18.19

Table 21 – Power (kW) & Torque (Nm) for Each RPM – Excel & WAVE

As seen in Figure 28, Excel & WAVE start off very similar for 10250 RPM, but due to the power calculated within Excel not taking frictional losses into account, the differences between them increases significantly, the faster the speed of the engine. This is most likely due to the Triumph data providing IMEP values, where WAVE supplies BMEP values taking frictional losses into account.

Because of this, Excel claims that the power increases higher at 14400 RPM, which is the bikes RPM limit, even after the stated maximum power 94kW occurring at 12600 RPM. The torques calculated from both software also over predict, the stated maximum torque is 73Nm at 11750 RPM, this figure is only achieved as the torque decreases at 14400 RPM but again, this prediction could be improved by using more detailed data.

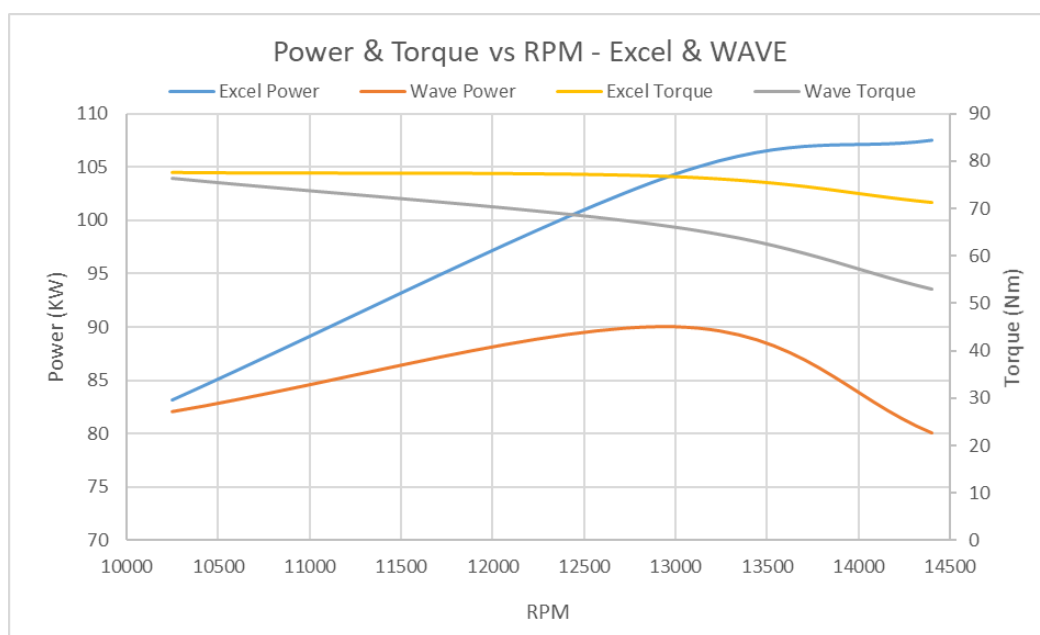


Figure 28 – Power & Torque vs RPM – Excel & WAVE

5) Conclusion

In conclusion, Ricardo WAVE is a powerful piece of software and it has a multitude of uses, it can calculate and compute small changes within variables, enabling the investigation into the impact on important factors, such as combustion & emissions, thermal efficiency, and general engine performance characteristics.

Although some of the data that is generated may not 100% resemble real-world data, the trends and influences are somewhat symbolic of what would happen in real-life, e.g. if an increase in max cylinder pressure is calculated in WAVE, then a similar increase will be witnessed in reality.

But by modelling the combustion within a virtual engine using limited data, this software can also over or under-predict results. If every variable could/has been accounted for, WAVE should provide highly accurate results, but given the limited data supplied for this experiment, it was incredibly hard for the Excel data to be matched.

Even when the data was almost identical within the pressure graph, by viewing the Log PV graph, the differences were exaggerated and clearer to see. This meant that the smallest details about each process could be analysed and alterations to the WAVE model could be undertaken to provide life-like results, should those adjustments be known how to be carried out to supply as accurate data as possible.

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