

A red motorcycle, likely a Triumph Daytona 770, is shown from a side profile, parked on a light-colored gravel surface. The motorcycle is the background for the title text.

MODEL OPTIMISATION OF A HIGH SPEED 3 CYLINDER USING RICARDO WAVE

Advanced Computational Methods M6X00408



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1 Introduction

This investigation focuses on the Ricardo Wave model optimisation of a prototype Triumph Daytona 675cc inline 3 engine, with an aim of improving the simulation results accuracy.

Author uses the supplied pressure data to define important engine characteristics which describe the engine performance.

In the first step of model validation, author examines the supplied Ricardo Wave model in detail and explains all of the found problems.

Later, author addresses all found problems in 5 model validation sections: engine specification, combustion, heat transfer, flow and mechanical friction. Every section is validated either against the previously defined engine characteristics or by additional information which is specified in the section.

2 Pressure Data Analysis

Author has decided to analyse the supplied cylinder pressure data from the “41hrs Endurance” test, as to find out the real engine characteristics so that certain ‘targets’ could be created and used to validate the accuracy of the simulation.

2.1 Combustion Characteristics

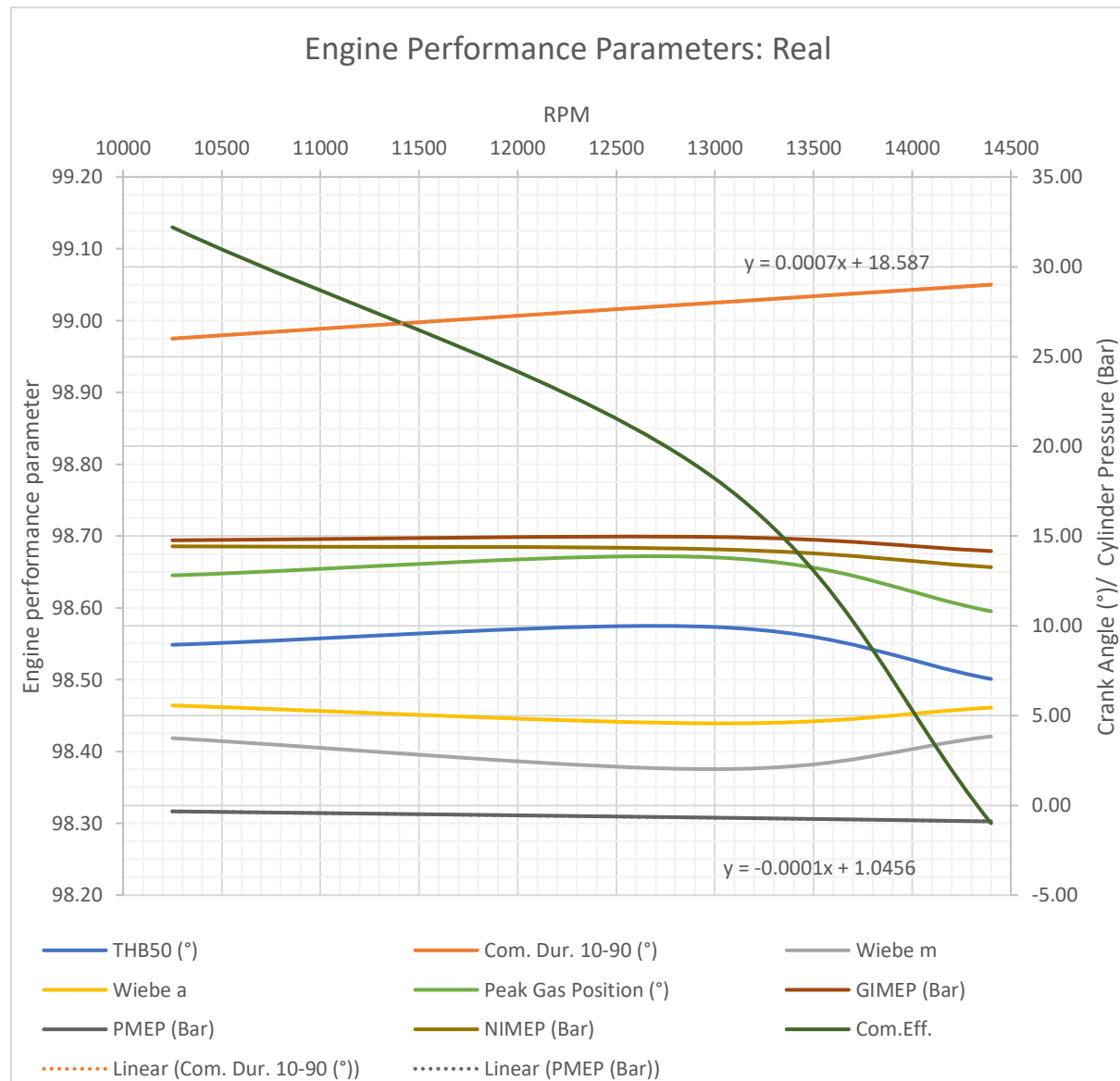


Figure 2-1: Graph of Engine Performance Parameters - Real

Figure 2-1 shows the calculated relative engine performance parameters. Author has immediately concluded the scope of the supplied data to be insufficient, mainly due to the inability to conclude the linear trends in Wiebe m, Wiebe a and GIMEP (gross indicated mean effective pressure) between 10,250rpm and 13,000rpm as well as a rapid change in gradient between 13,000rpm and 14,400rpm, to be all solely caused by the retardation of THB50.

However, the Com. Dur. (Combustion Duration) has shown to be very linear. Author believes that the Com. Dur. is mainly governed by the engine speed and that it is relatively unaffected by dynamic environment inside the ICE (internal combustion engine). This is why Author believes that extrapolating more Com. Dur. points at different rpms is accurate enough, considering the limited scope.

Author also believes that extrapolating further data points from GIMEP, PMEP and NIMEP is not wise, as mean effective pressures are affected very much by 'outside' factors such as scavenging. For example, the PMEP in Figure 2-1 is relatively straight, which is why the equation of the line has been generated. The equation suggests that at all rpms before 10,456 the PMEP>0, in fact according to that equation the PMEP would reach +0.95 Bar at 1,000rpm, which is highly unrealistic.

According to the supplied data, the combustion efficiency is between 98.3% and 99.13%. This might be caused by high cylinder turbulence due to the high rpm, or, author believes it to be possible that the supplied MFB is overly optimistic.

2.2 Polytopic Index

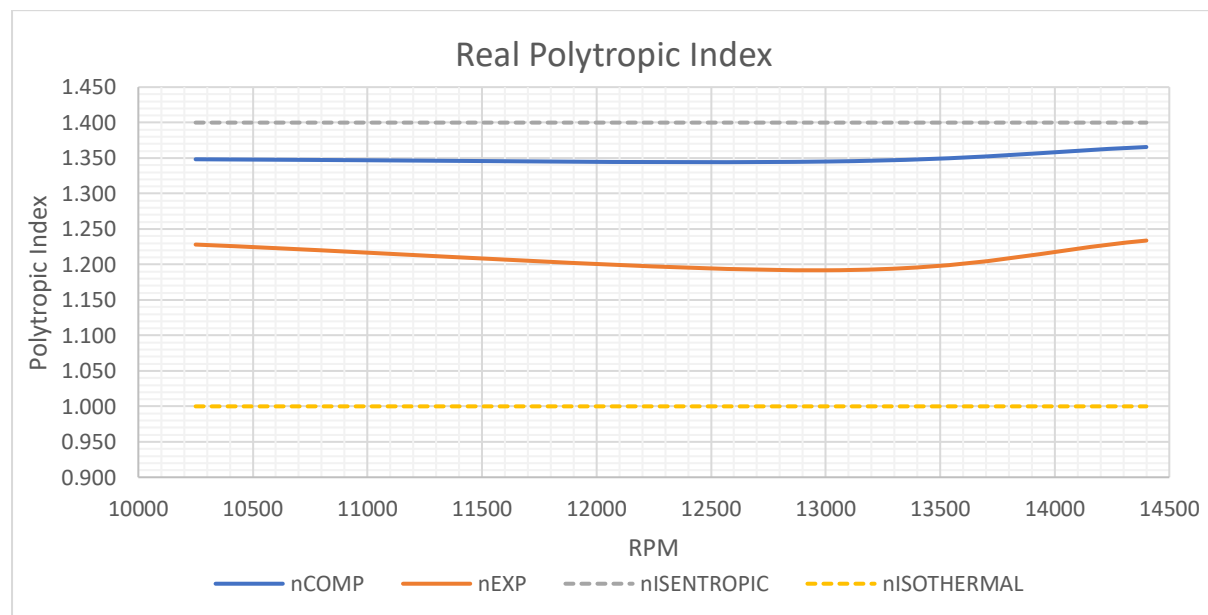


Figure 2-2: Real Polytopic Index

Figure 2-2 shows the real polytopic index of the engine. As expected, the nCOMP is close to the adiabatic value of 1.4, due to the high rpm and therefore low time duration at which the heat transfer occurs. However, author didn't expect the nCOMP to increase at 14,400rpm that suddenly, Author believes it's not valid to make a conclusion, due to deficient rpm data. The relatively small nCOMP difference of 0.22% between 10,250rpm and 13,000rpm is believed to be caused by the data collection method.

Unexpectedly nEXP has a trough at 13,000rpm. As Figure 2-3 shows, the combustion at that rpm has the lowest combustion rate. This could have been caused by poor scavenging optimisation at that rpm which would decrease the charge purity and as a result decrease the combustion rate, as there would be more mass absorbing the heat instead of producing it. This would have caused the heat rate output due to combustion to be lower. Since the supplied data was conducted on a pre-production engine, it is possible that the cooling system hasn't yet been finalised, as such, it is possible that the engine was being overcooled despite lower heat output. As a result, the temperature difference between the gas and the cylinder head and liner could have increased, despite the lower gas temperature due to poorer combustion. This would have resulted in higher heat transfer and as a result, lower nEXP.

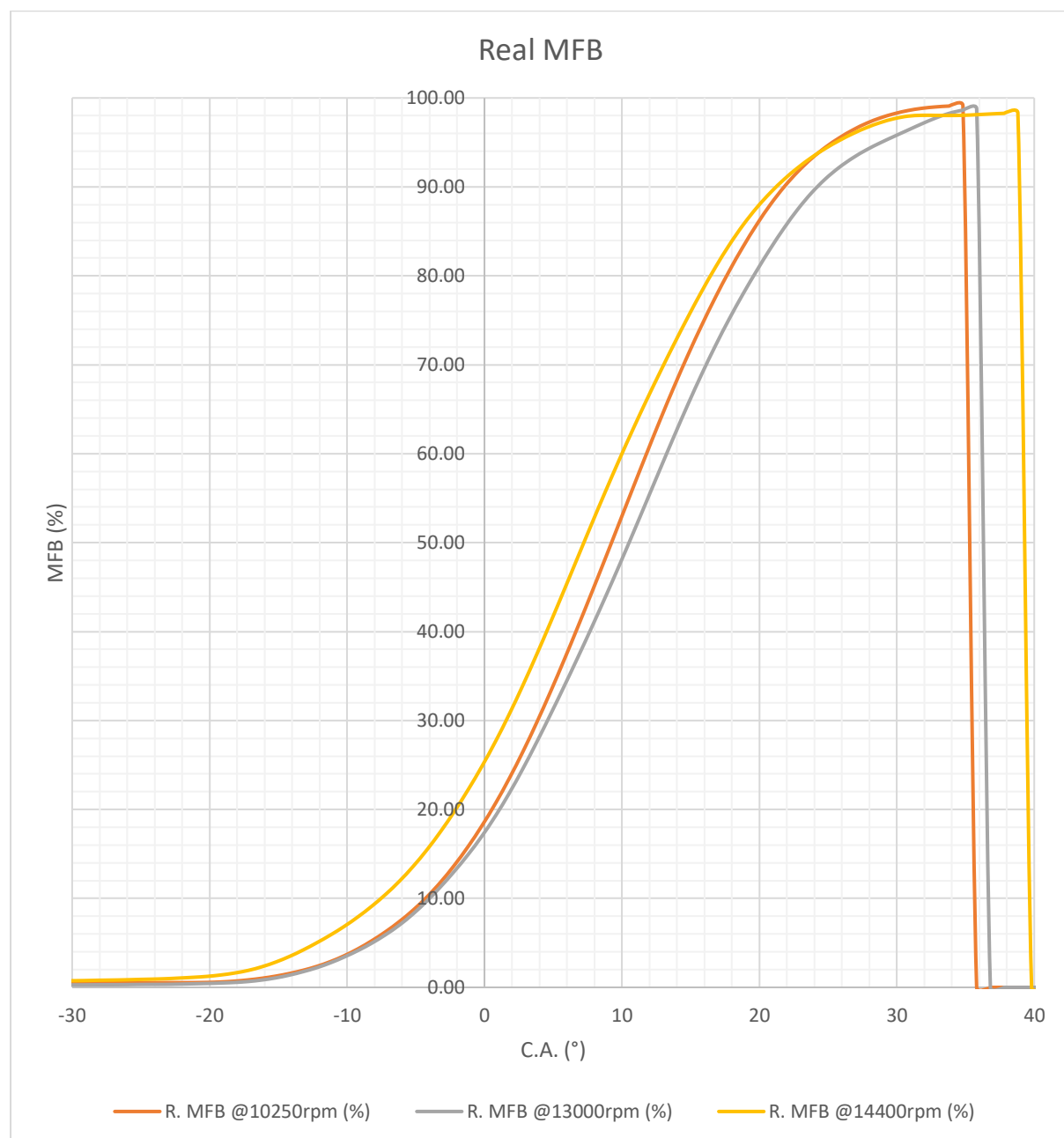


Figure 2-3: Real MFB

3 Initial Model Analysis & Comparison

3.1 Original Model Comparison Against Real Cylinder Pressure Data

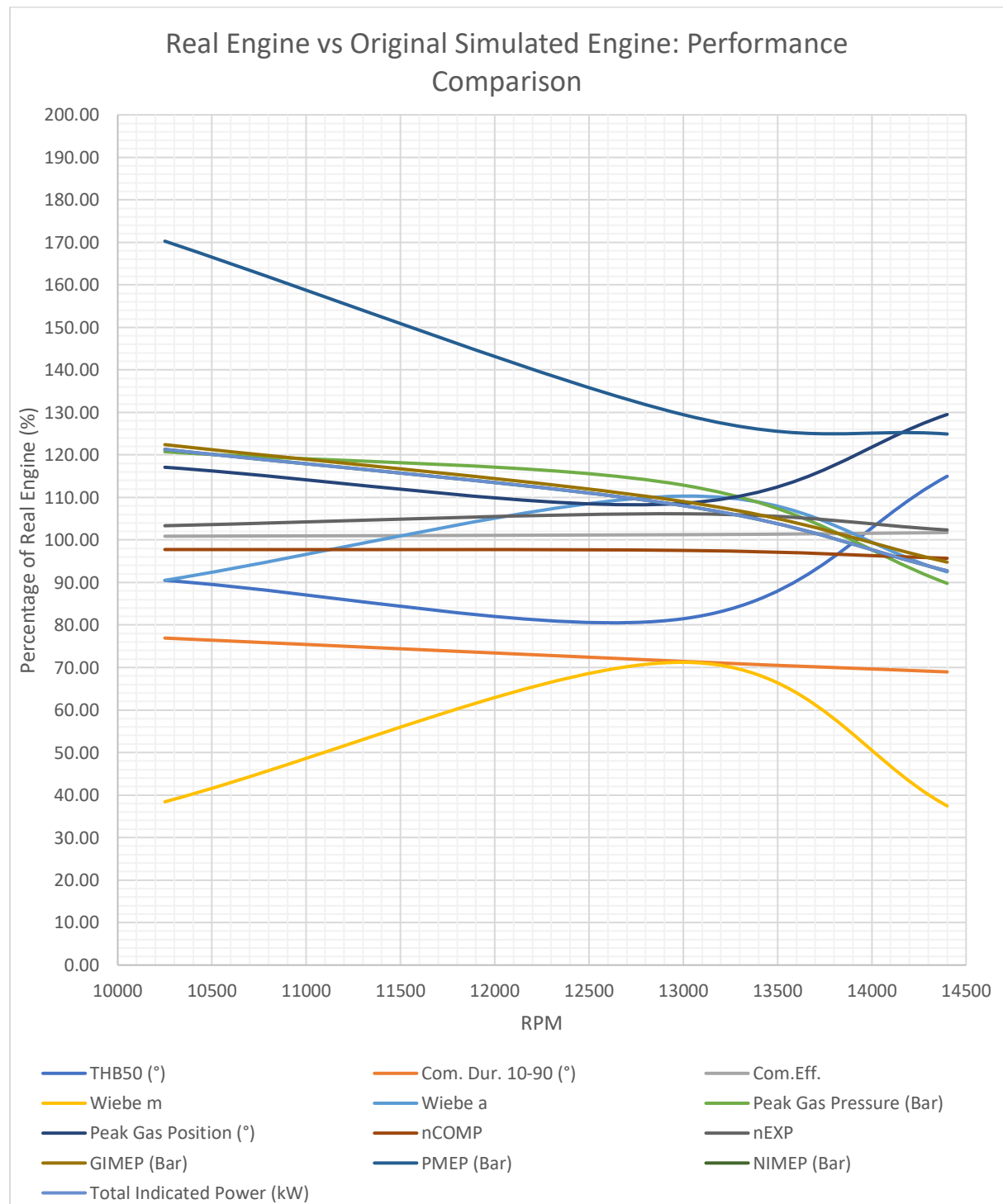


Figure 3-1: Real Engine vs Original Simulated Engine - Performance Comparison

Figure 3-1 shows the comparison between the real engine values and the values from the simulation in its original state. The majority of the parameters have an inaccuracy range of $\pm 20\%$. However, the extremes are -65.59% and $+70.28\%$ respectively for Wiebe m and PMEP.

Author has decided to have an average target accuracy range of +/- 1% as this would theoretically yield in a peak power range between 93.06kW and 94.94kW. Given that the real peak power output is 94kW (Williams, 2019), author believes the yield to be acceptable. The majority of parameters from Figure 3-1 such as Com. Dur. or THB50, can be easily put to 100% accuracy as the values are known and they can be directly used in the simulation. What author is expecting to cause the biggest challenge are the FMEP and PMEP simulation.

3.2 Original Model Comparison Against Real Engine Specification

Engine Specification Sheet Comparison		
	Real Engine	Original Simulated
Conrod Length (mm)	104.6	98
CR	12.7	12.65
λ	0.9	0.88
Ambient Pressure	1	1.065
Ambient Temperature	20	27

Figure 3-2: Engine Specification Sheet Comparison

Figure 3-2 shows the comparison between parameters that author found to be different in the original simulation compared to the supplied cylinder pressure data.

Camshaft Specification Comparison				
Bike:	Real		Original	
Specification:	Intake	Exhaust	Intake	Exhaust
Duration @ 1mm lift	258.5°	247°	250°	234°
Full lift @ deg. (ATDC/BTDC)	102°	100°	84°	81°
max lift (mm)	9.25	8.50	8.43	7.46
Valve Diameter	30.5	25.5	31.0	unkown
Reference Diameter	28.1	23.0	29.0	24.5

Figure 3-3: Camshaft Specification Comparison (Triumph Designs Ltd., 2008) (Ape Raceparts, 2018)

Figure 3-3 shows that the camshaft specification was very different to the production engine. It also shows that the “reference diameter” in the valve editor was also incorrect for intake and exhaust by 0.9mm and 1.5mm respectively. To calculate the reference diameter for the real engine author has taken away the 4mm valve stem area (Ape Raceparts, 2018) as well as an assumed valve seat width of 1.5mm at 45° angle since author wasn’t able to find this information. The assumption was based on author’s empirical knowledge of similar valves and their applications.

The end and beginning diameter of intake and exhaust ports are 28.3mm and 23.8mm respectively. Those values are very similar to the calculated reference diameter values from Figure 3-3. As a result, author believes these to be the correct reference diameters values.

Lastly, author has noticed the unrealistic C_d graphs for the intake and exhaust valves.

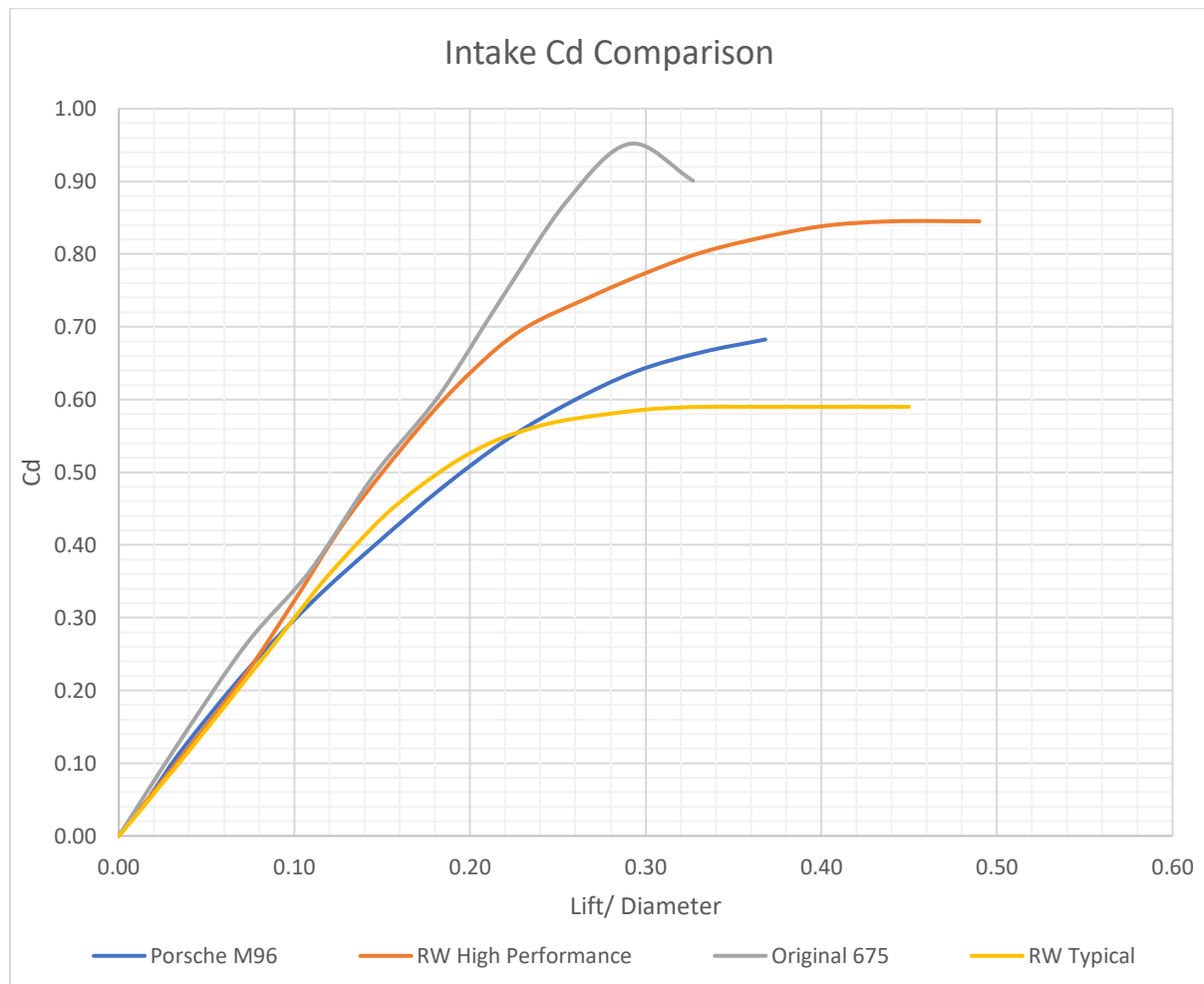


Figure 3-4: Intake C_d Comparison

Figure 3-4 shows the comparison of intake discharge coefficients between supplied Triumph 675, Ricardo Wave typical and high performance as well as OEM Porsche M96 (Racetechniques, 2019). Author believes the supplied Daytona graph is incorrect due to:

- Relatively constant gradient up until 0.29 L/D
- Peak C_d value of 0.95
- The C_d decreases after 0.29 L/D

Author used the OEM Porsche M96 graph to showcase a typical 'high performance' OEM values one might expect. Author believes this is applicable in this case due to similar Valve diameter of 35mm and peak valve lift of 0.29 L/D compared to Daytona's 31mm valve diameter and peak L/D of 0.298.

3.3 Model Quality Examination

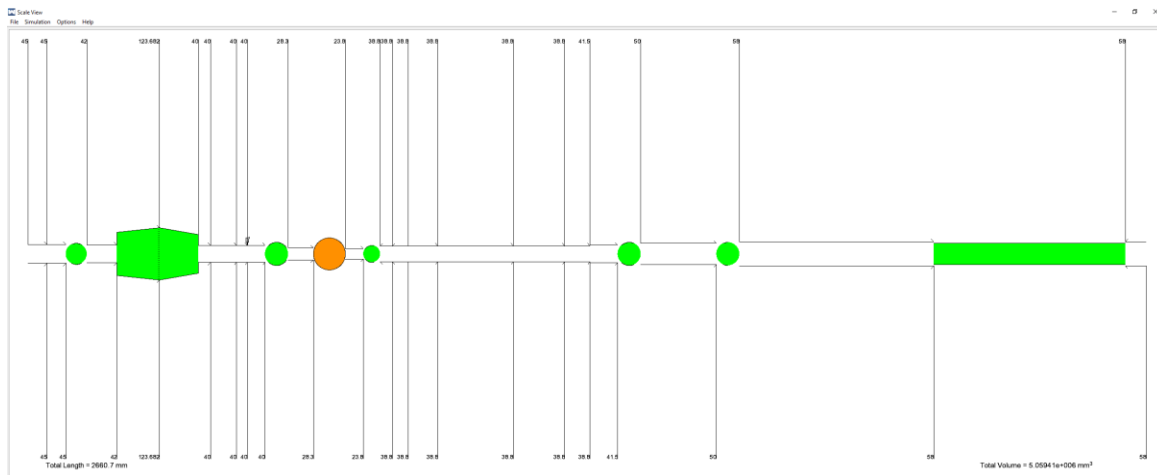


Figure 3-5: Cylinder 1 Flow Track Scale View

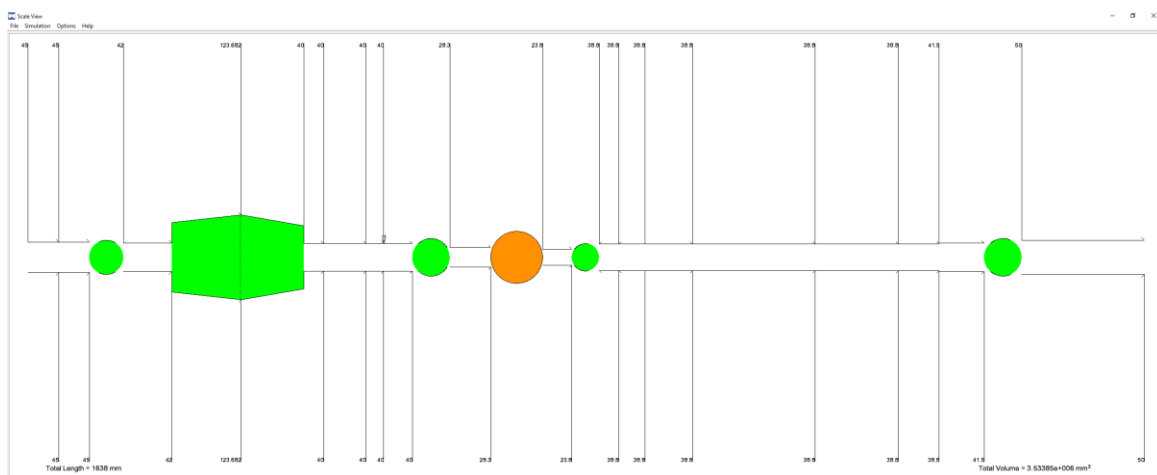


Figure 3-6: Cylinder 2 Flow Track Scale View

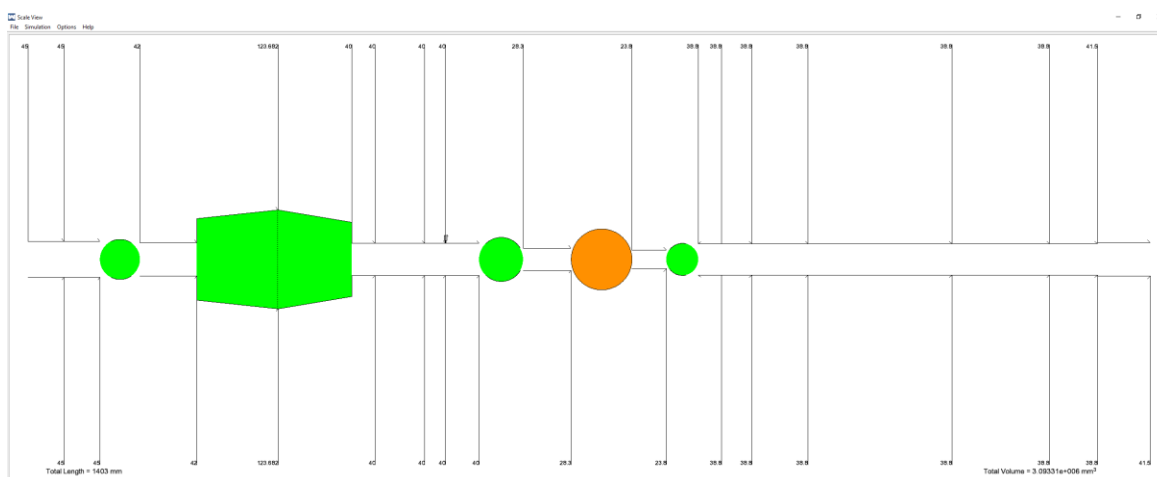


Figure 3-7: Cylinder 3 Flow Track Scale View

Figure 3-5, Figure 3-6 and Figure 3-7 show the Scale View of the flow tract through the entire engine for cylinders 1, 2 and 3 respectively. Author has used this to make sure that no bottlenecks or

abnormal lengths and volumes were created in the model by mistake. Author couldn't find anything what he believes is out of ordinary.

However, author has managed to find two incorrect design choices:

1. The exhaust system type in the original simulation is 3-2-1. However, as Figure 3-8 shows, the real OEM exhaust type is 3-1.
2. The throttle body diameter in the original simulation is 40.0mm whereas throttle bodies used in the real engine are 44mm (Ape Raceparts, 2018).

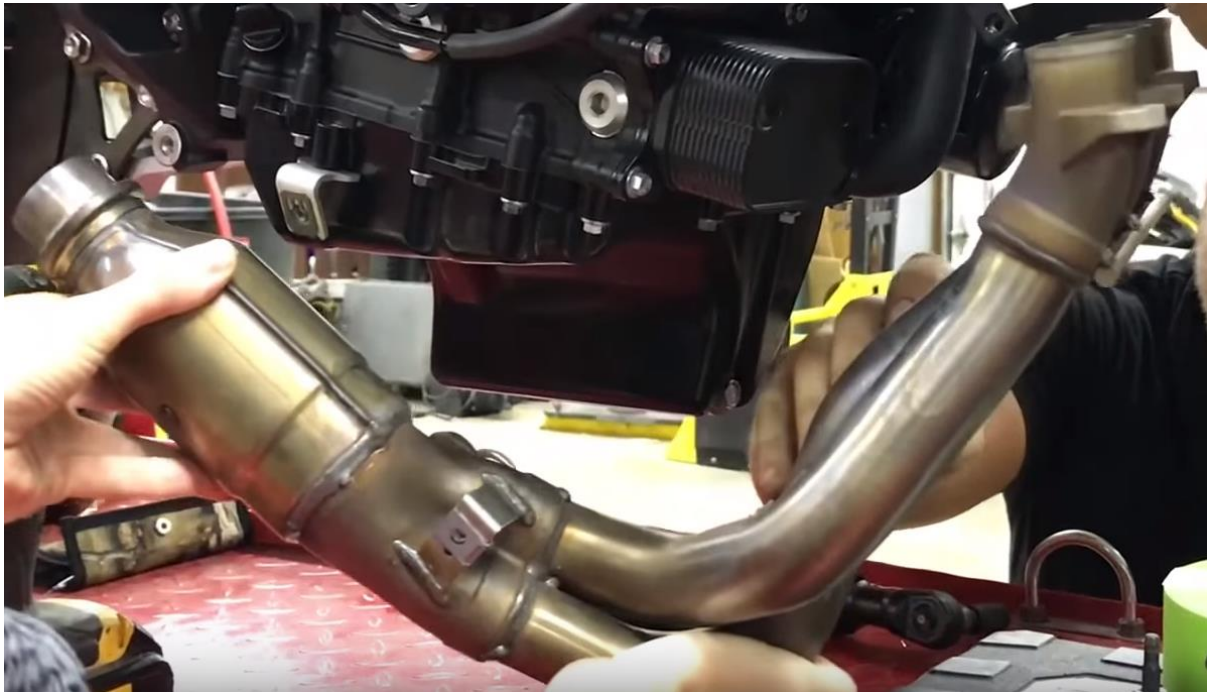


Figure 3-8: Picture of the OEM Triumph 675 3-1 exhaust (Campisi, 2017)

As a result, author believes that all discrepancies observed in section 3 are due to the model coming from relatively early stages of the powertrain development for the 675 Daytona project.

4 Model Optimisation

4.1 Engine Sheet Specification Utilisation

In this section, the original simulated values from Figure 3-2 have been replaced by those in “Real Engine” column.

To calculate the required Fuel/ Air Ratio, Equation 1 has been used.

$$\varphi = \frac{1}{AFR}$$

Equation 1: Equivalence Ratio AFR relation

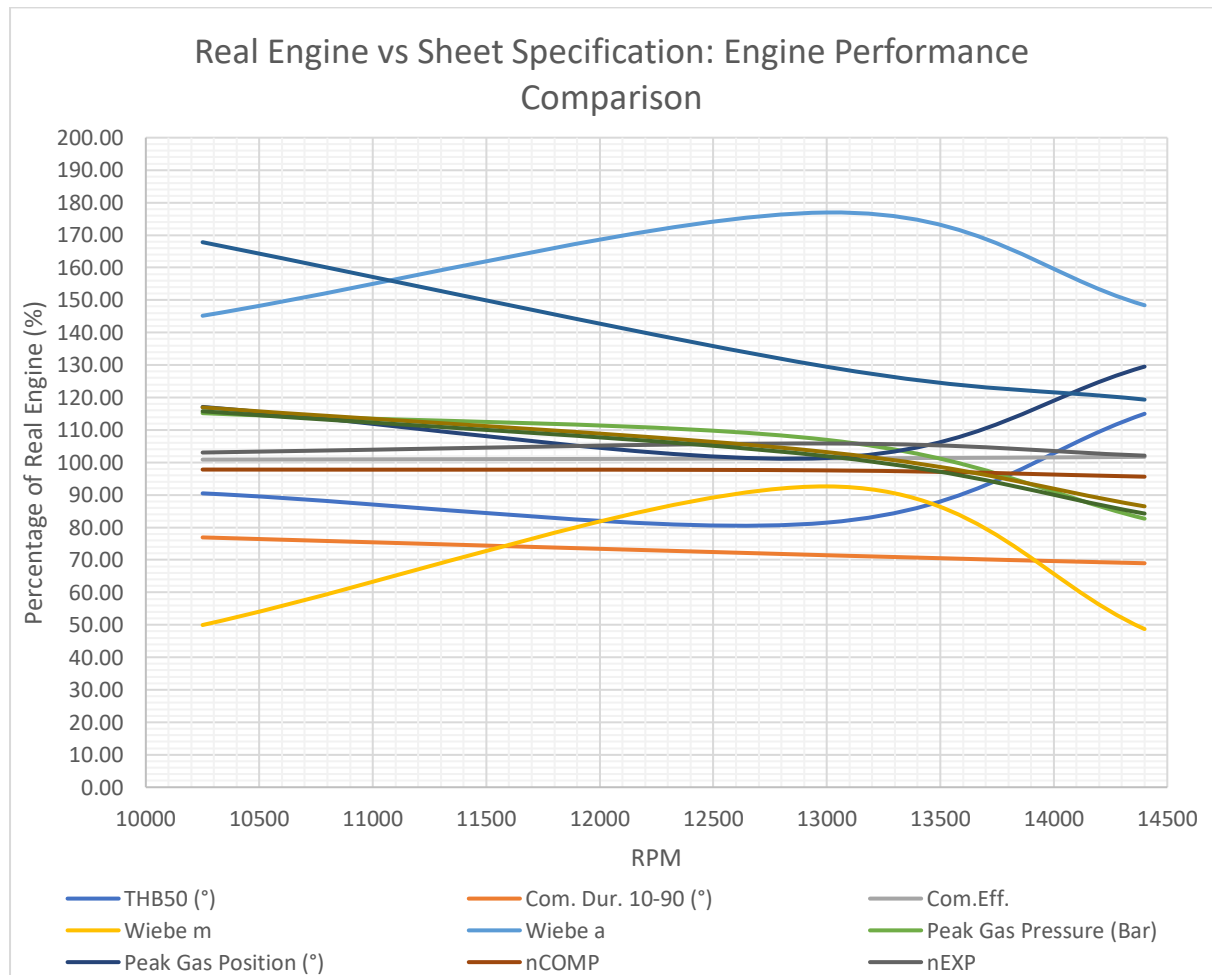


Figure 4-1: Real Engine vs Sheet Specification - Engine Performance Comparison

Figure 4-1 shows that the implemented changes caused the Wiebe variables to increase across the board, the peak cylinder pressures lowered, causing avg. accuracy increase of 6.2% and all other inspected values stayed practically the same. As a result, the avg. accuracy of GIMEP was improved by 6.54%.

4.2 Combustion Characteristics Utilisation

In this section, in addition to changes done in section in 4.1, the parameters of THB50, Com. Dur. 10-90, Com.Eff. and Wiebe m from Figure 2-1, have been used to improve the relative combustion accuracy of the model.

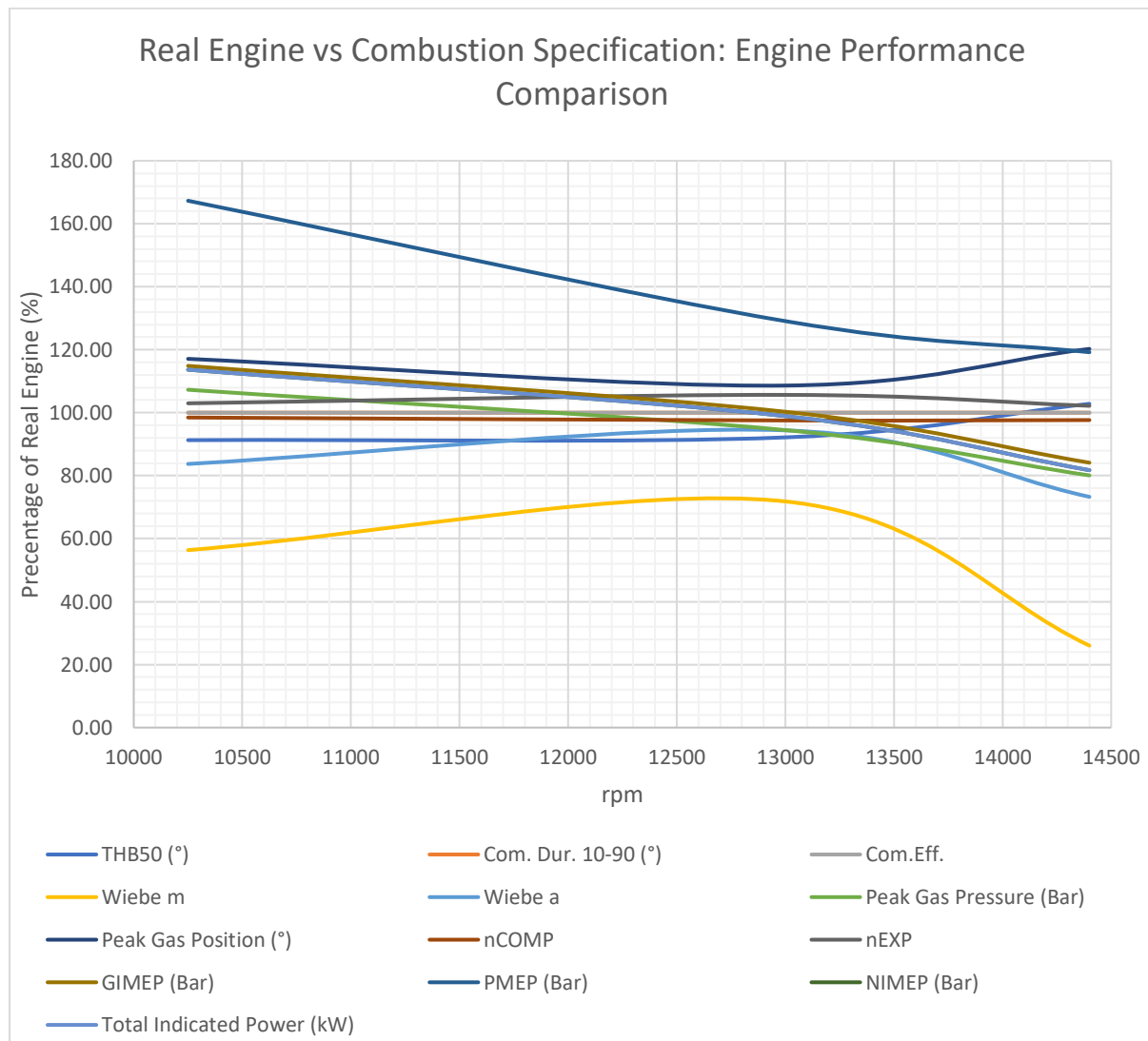


Figure 4-2: Real Engine vs Combustion Specification - Engine Performance Comparison

Figure 4-2 shows that the implemented changes to combustion characteristics parameters caused combustion duration and efficiency to be 100% accurate. However, despite inputting the exact THB50 as well as the Wiebe parameters, the avg. accuracy of those variables didn't noticeably improve. The avg. accuracy of the peak cylinder pressure has been lowered by 4.44%. The GIMEP avg. accuracy has been improved by 1.96% to the value of 99.76%. Despite this relatively good result author wasn't satisfied due to the GIMEP being only accurate at 13,000rpm with accuracy of 100.3%, since at 10,250rpm and 14,400rpm, the accuracy decreased to 114.87% and 84.11% respectively, resulting in standard deviation of 12.56%.

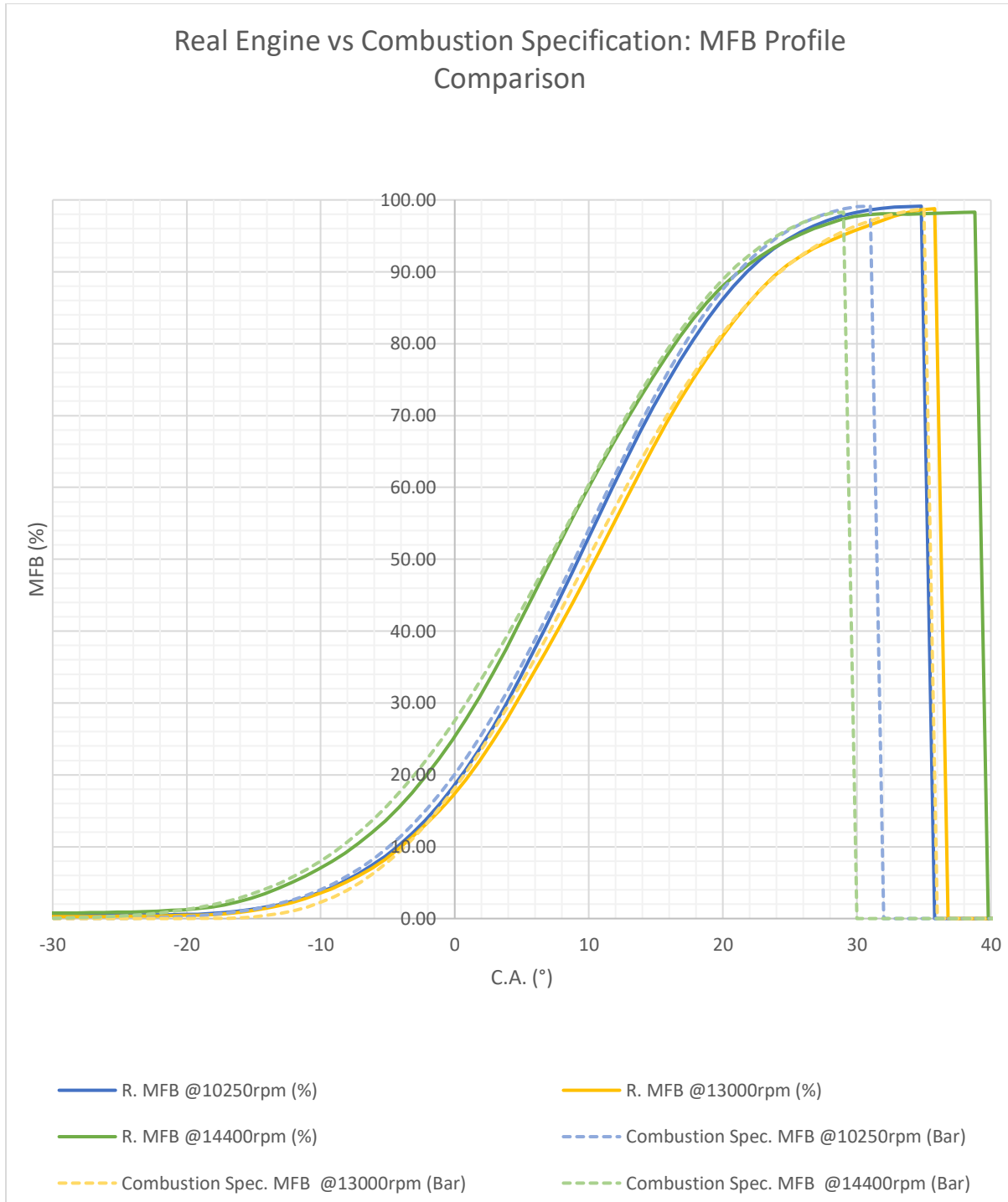


Figure 4-3: Real Engine vs Combustion Specification - MFB Profile Comparison

Figure 4-3 shows that despite no practical accuracy improvement of THB50, Wiebe a and m, the MFB profiles for all three known rpms match their real engine counterparts. This suggests that the difference in Wiebe a and m values are due to the inaccuracy in the curve fit method. The only practical discrepancy showed by Figure 4-3 are the different points at which MFB value falls to 0%. This however is not an issue, as it has already been mentioned, the peak MFB values (Comb. Eff.) are the same.

4.2.1 Fuel and Air property editor

Author is aware of the ability to create user defined fuels. However, this requires accurate data such as: specific heat capacity, vapour pressure, viscosity and surface tension. Author doesn't have access to this data. Furthermore, author doesn't have the fuel used in the endurance testing from which the data comes from. This is important since the substance qualities (iso-Octane, iso-Pentane etc.) alter depending on the time of the year, factory and the specific batch.

Thus, author has decided to use the "Indolene" fuel, supplied and recommended by Ricardo Wave. Its chemical composition consists of 7.3kmol of carbon and 13.9kmol of hydrogen. This is done so that the "Indolene" fuel can be used to accurately represent gasoline, which consists of hydrocarbons normally between C4 and C12. It has an LHV of 43.1 MJkg^{-1} , which is in the ballpark of typical petrol, normally around 43.4 MJkg^{-1} (Engineering ToolBox, 2019).

Lastly and most importantly, it already has all of the required substances property values, which are accurate, precise and tested by Ricardo Wave themselves.

4.2.2 The Gas Equation

Author is aware of the ability to replace ideal gas law with real gas equation in Ricardo Wave simulation. However, since author is not expecting cylinder pressures above 100bar and mean cylinder temperatures above 3000K, author believes such implementation would have practically meaningless effect on the accuracy while increasing the computational time required to run the simulation.

4.3 Heat Transfer

Polytropic index (n) describes the thermodynamic process of a system. As a consequence, it is directly related to the heat transfer of the system. To set this value efficiently author tried to use the "experiment" function. However, Ricardo Wave software doesn't output polytropic index. Therefore, author was forced to use other parameters which are related to polytropic index such as peak cylinder pressure and GIMEP.

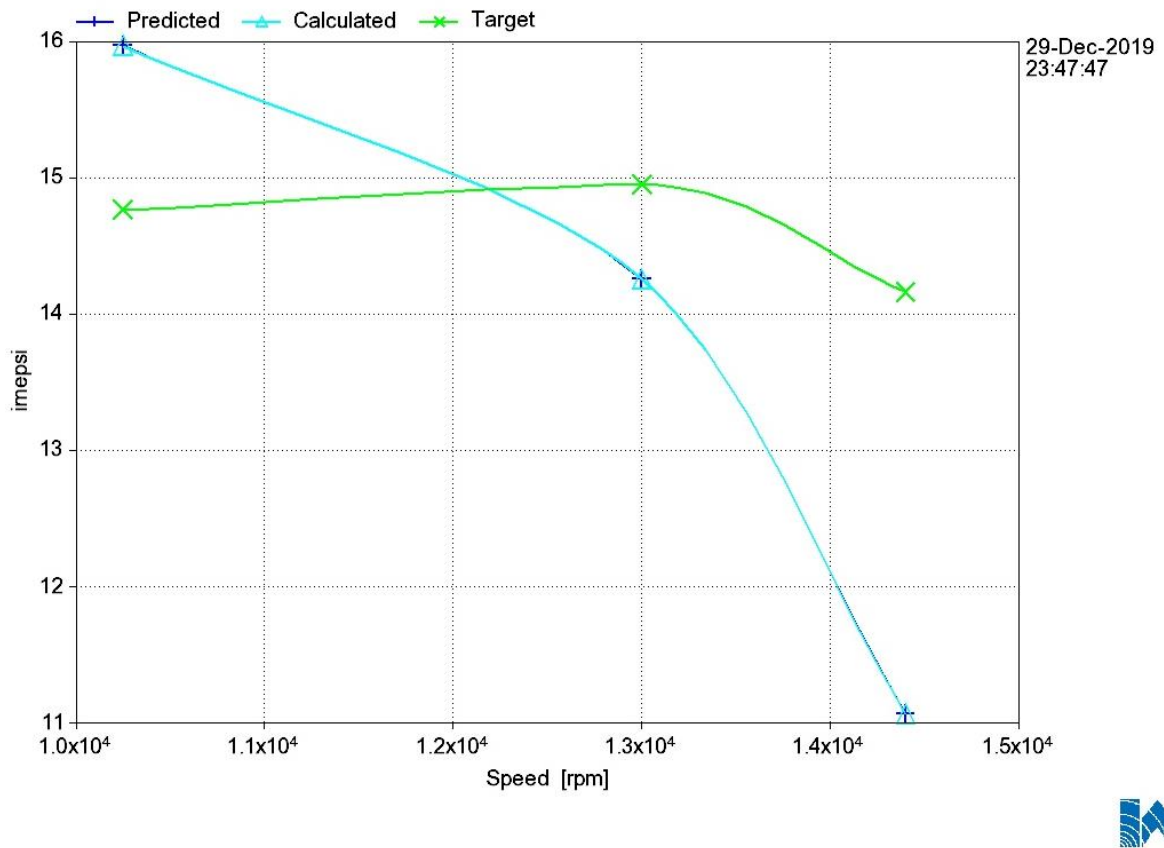


Figure 4-4: Initial IMEP Match

Figure 4-4 shows a comparison between the GIMEP of real engine (target) and the simulated GIMEP which has been fitted to the target by adjusting the heat transfer and area multipliers. It is clear that the enormous discrepancy is caused by something other than polytropic index, especially since no combination of the multiplier adjustment would transform the simulated GIMEP line to even resemble the target line.

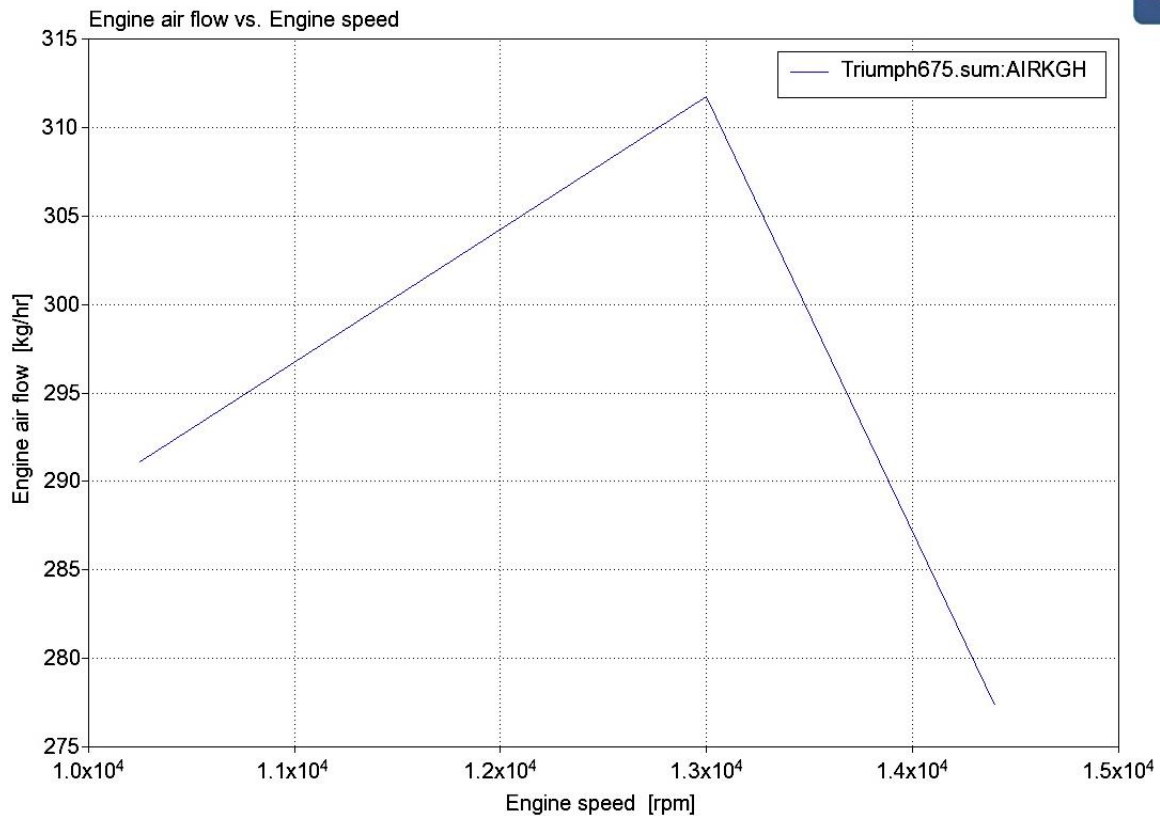


Figure 4-5: Initial Airflow

Figure 4-5 shows that the issue is clearly flow related. Since author believes that polytropic index should be set before attempting to alter the flow, author was forced to carry out the experiment manually.

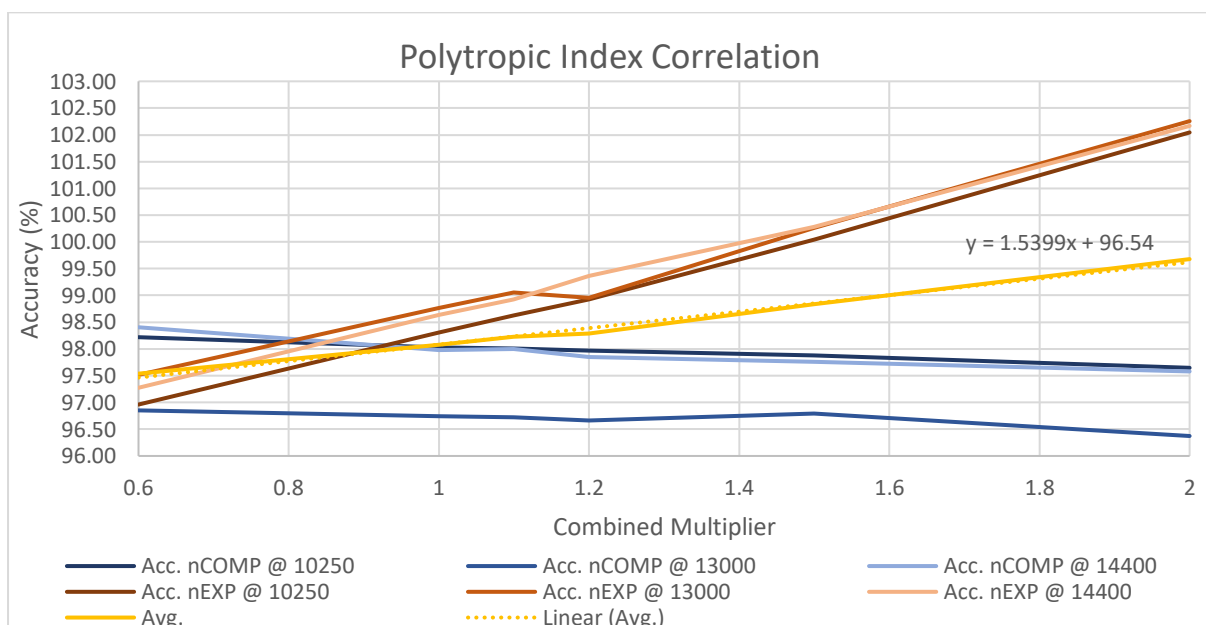


Figure 4-6: Polytropic Index Correlation

Figure 4-6 shows the relation between the accuracy of the polytropic index against the combined multiplier (heat transfer multiplier when valves are closed and the cylinder-head area multiplier). It is clear that the relations of nCOMP and nEXP are opposite: accuracy of nEXP increases and accuracy of nCOMP decreases as multiplier is increased. Therefore, author decided to conduct the same experiment. However, this time the multipliers weren't combined, results are shown by Figure 4-7 and Figure 4-8.

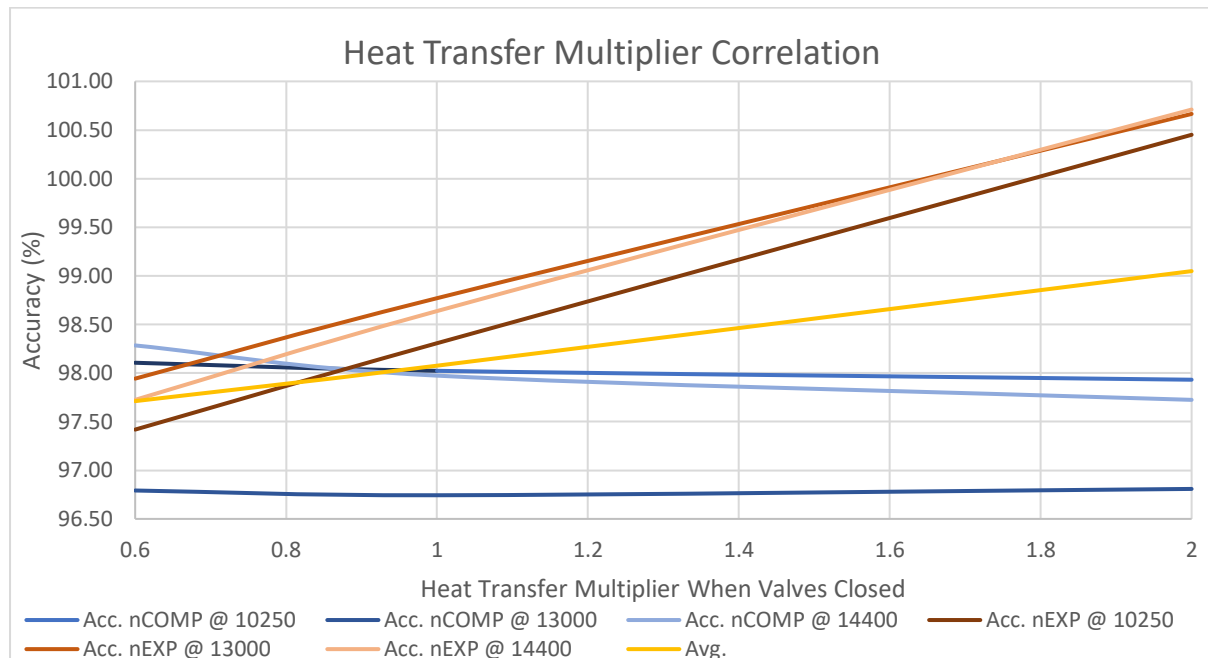


Figure 4-7: Heat Transfer Multiplier Correlation

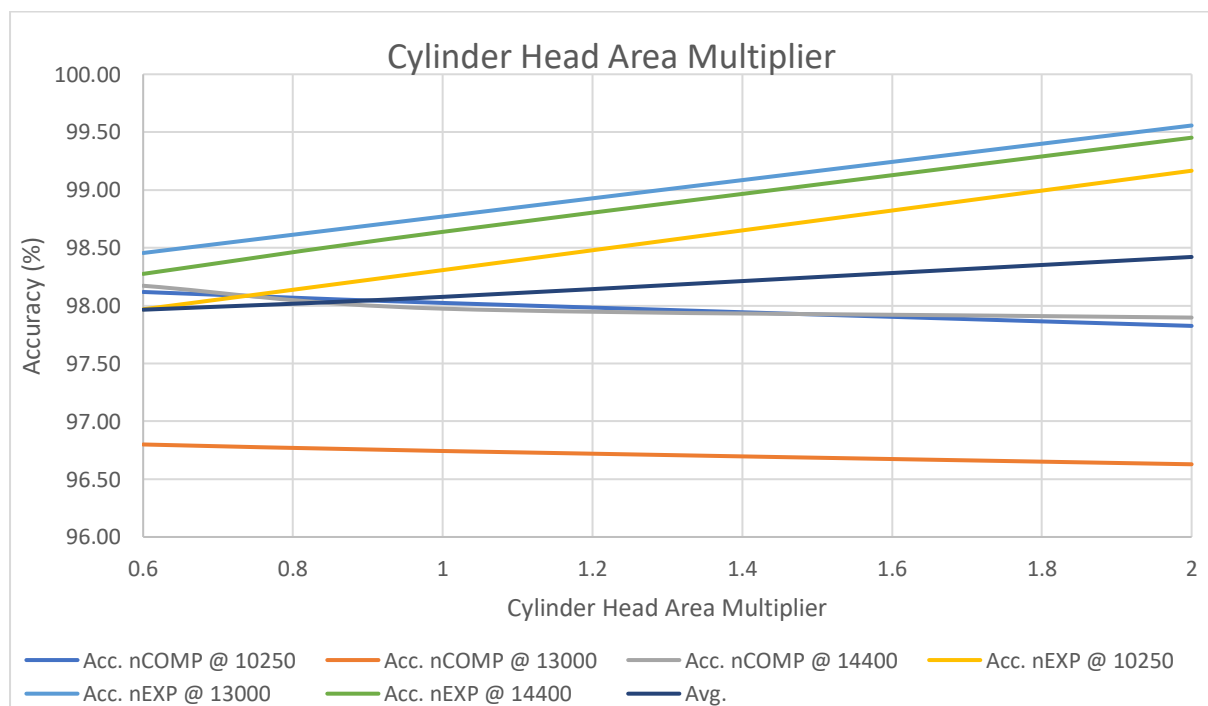


Figure 4-8: Cylinder Head Area Multiplier

Figure 4-7 and Figure 4-8 both show the same relation as Figure 4-6. Therefore, author calculated the mean and standard deviation for all of the experimental results, this is shown by Figure 4-9.

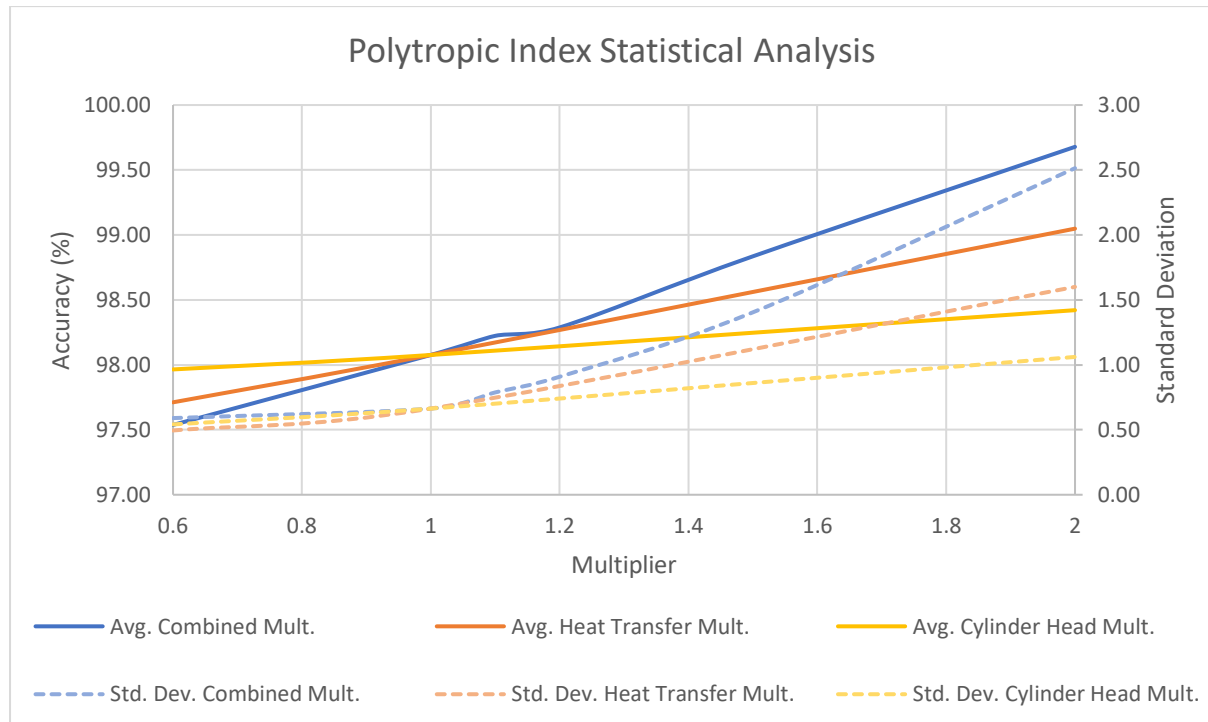


Figure 4-9: *Polytropic Index Statistical Analysis*

However, Figure 4-9 doesn't clearly show what is the appropriate multiplier/s. As a consequence, author plotted Figure 4-10.

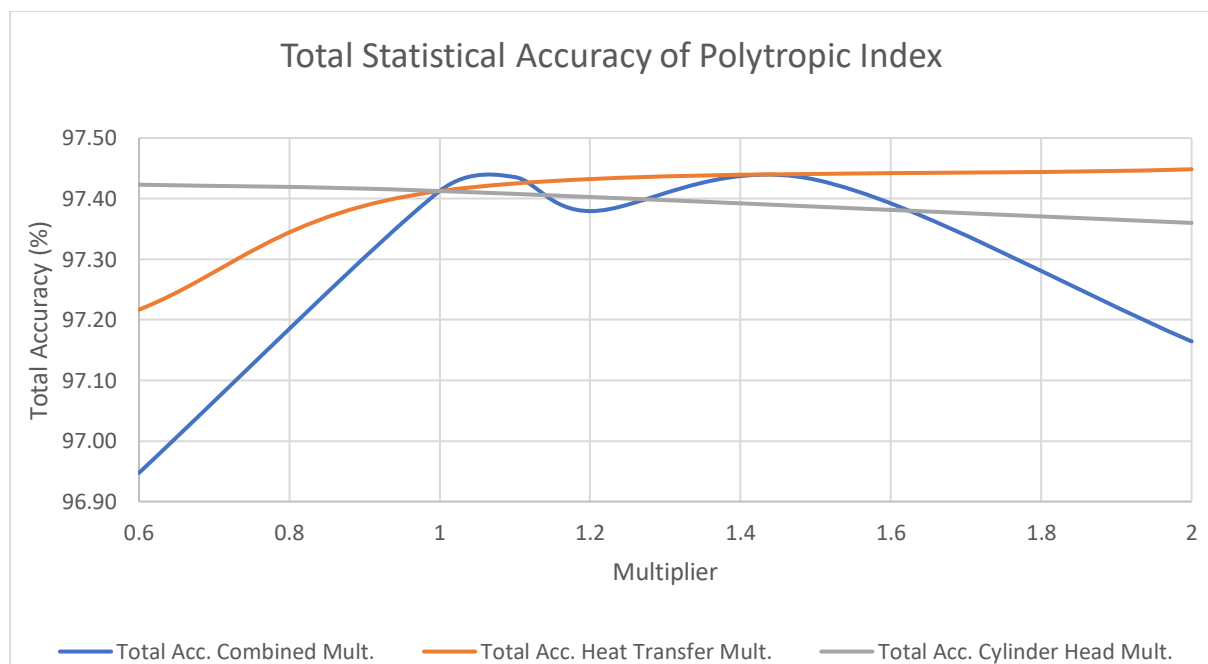


Figure 4-10: *Total Statistical Accuracy of Polytropic Index*

Figure 4-10 shows the total statistical accuracy of the simulated polytropic index. These values were calculated by taking away standard deviation from the mean accuracy of polytropic index. This method was used since all of the mean values were below 100%, as shown by Figure 4-9. This ultimately shows that the variables which have been used aren't enough to perfectly match the polytropic index at all known rpm. This matter is even more complicated since the multipliers change the rate at which heat is lost, this means that if any other parameters change, such as the volumetric efficiency, then the total heat release will change and as a result, the used multipliers will no longer be valid. Therefore, author has decided that the best choice from Figure 4-10, that is the heat transfer multiplier when valves are closed and cylinder-head area multiplier set to 2 and 1 respectively, is good enough and any further work on the polytropic index would be wasted in the later stages of the model development.

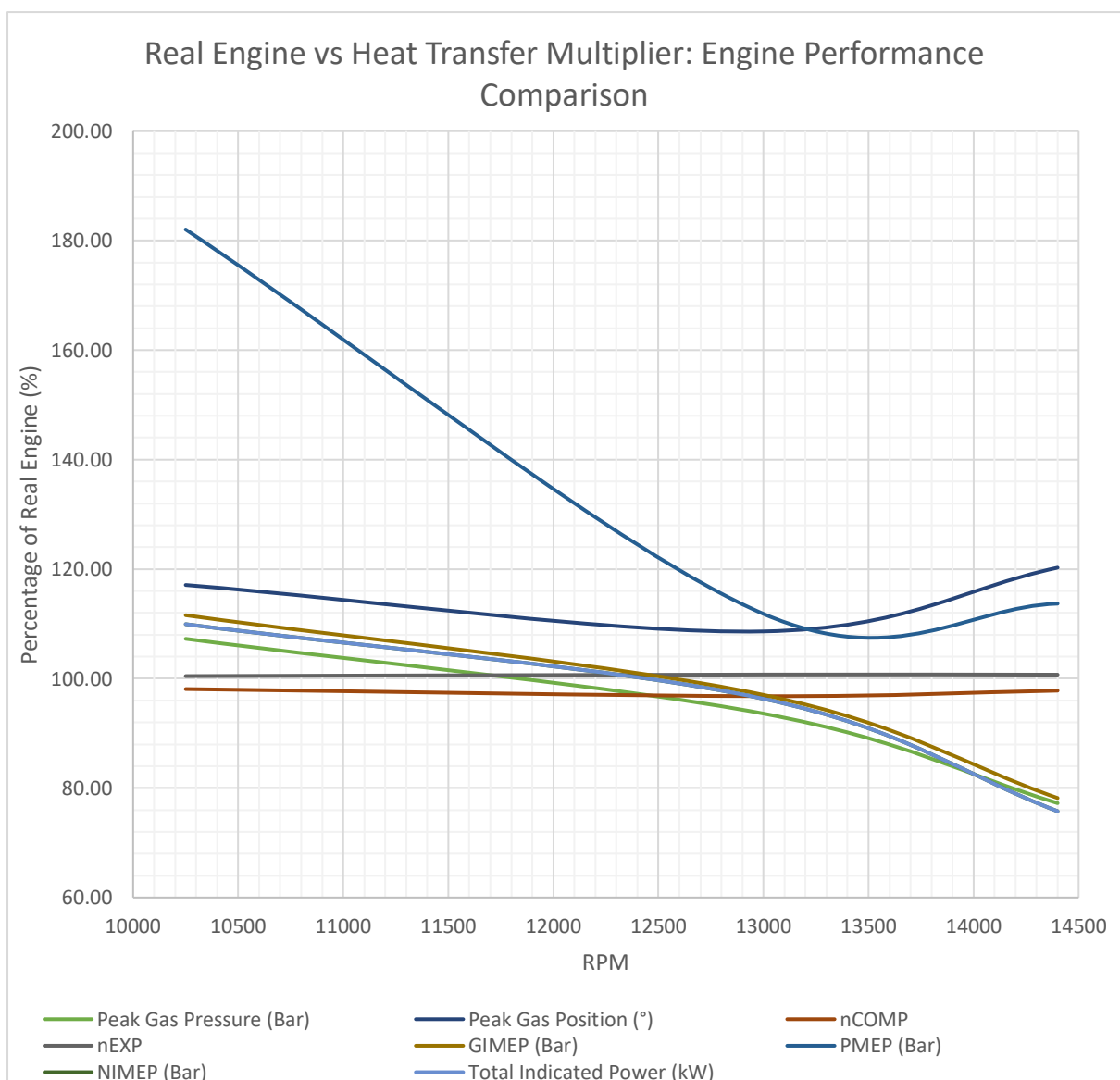


Figure 4-11: Real Engine vs Heat Transfer Multiplier - Engine Performance Comparison

Figure 4-11 shows the final accuracy of the engine performance parameters. Variables such as combustion efficiency or duration are directly defined and as a result they won't change, therefore these variables aren't displayed. The nCOMP was practically unchanged, however, the nEXP mean accuracy was increased by 0.55% to 100.65% and the standard deviation was lowered by 0.06% to 0.13%. This relatively small change has caused the NIMEP at 14,400rpm to decrease by 0.84 Bar, this has caused 5.94% decrease of NIMEP accuracy at the corresponding rpm. This occurred at other rpm as well but with smaller effect. As a result, the mean accuracy of NIMEP has dropped by 4.17% to 95.59%. It is very clear that the volumetric efficiency, especially at 14,400rpm is invalid, since all of the other parameters which ultimately affect the NIMEP have been validated to good enough accuracy already. This had a direct effect on the GIMEP and the indicated power. This occurrence has clearly shown that the flow through the original model must be altered and validated.

4.4 Flow Alterations

Since no real air consumption data was given, to validate the intake charge mass and as a result, the total heat release, author has designed an experiment which varies the valvetrain specification, the used values can be seen in Table 1.

Valvetrain GIMEP Experiment		
	Intake	Exhaust
Duration Min	250	230
Duration Max	260	250
Timing Min	80	80
Timing Max	105	105
Lift Min	8.4	7.5
Lift Max	9.25	8.5

Table 1: Valvetrain GIMEP Experiment

Experiment tool in Ricardo Wave isn't able to show the total heat release, author decided to use the IMEP, NMEP and PMEP plots instead. Author was able to directly use the mean effective pressures as for the validation of the flow through the engine, since all other parameters which may affect mean effective pressures have already been validated.

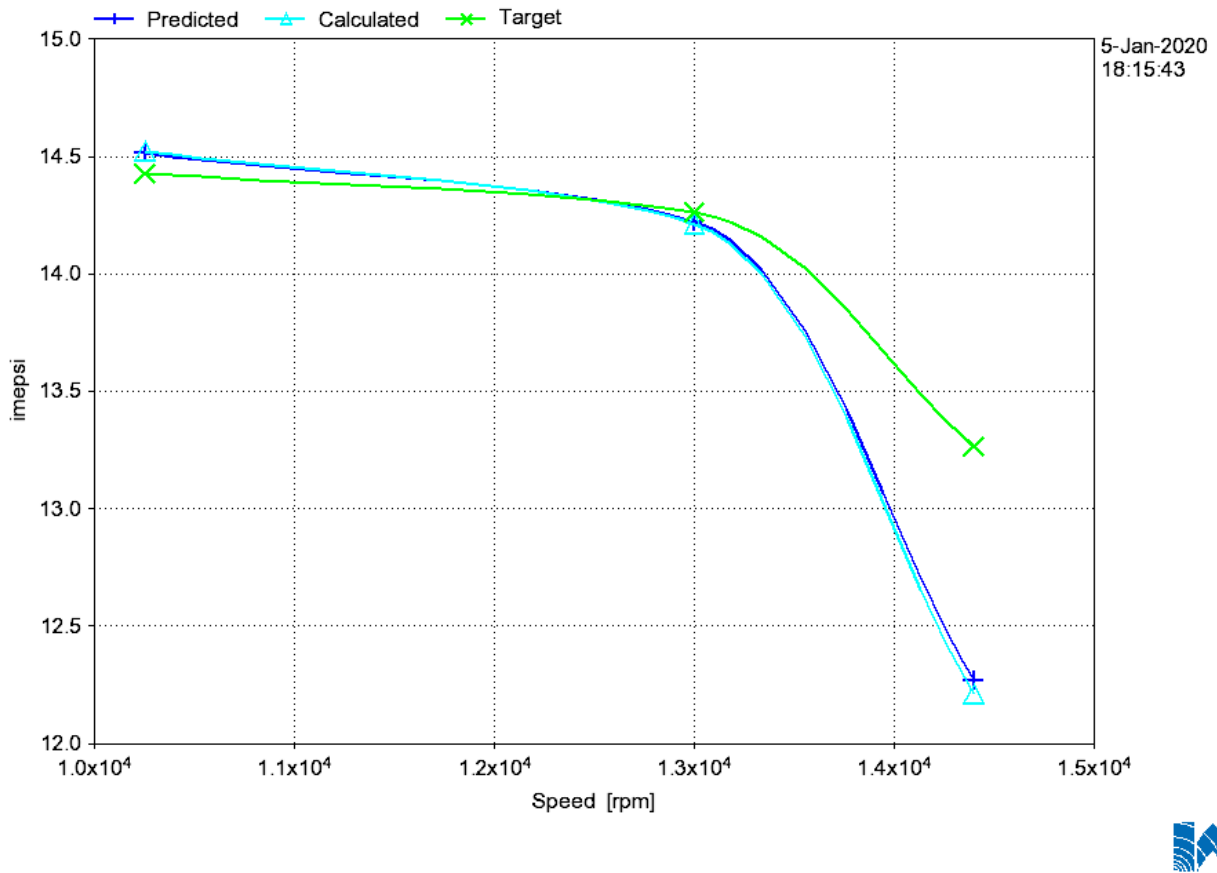


Figure 4-12: Initial valvetrain IMEP experiment results

Figure 4-12 shows the results of the initial experiment using the data from Table 1. Please note that Ricardo Wave has mistakenly named IMEP in bar as “imepsi”. The green “Target” shows the real IMEP. The match function was used to vary all 6 variables in an attempt to matching the simulated IMEP to the real Target values. This has failed, especially at 14,400rpm.

Author has run many more experiments with more variables such as the throttle diameter, as the section 3.3 showed it to be different from a production engine, to keep the same diameter ratio between the intake system constituents, the throttle diameter was multiplied by the original ratios, example is shown by Figure 4-13. However, the work didn’t yield improvements in accuracy of NIMEP.

Left Diameter {thr_diam*1.05}

Figure 4-13: Duct 76, left diameter function

Since author believes the original valve flow coefficient profiles to be incorrect, author has decided to run more tests while varying original, high performance and typical intake and exhaust valve C_d profiles, as shown by Figure 4-14 and Figure 4-15.

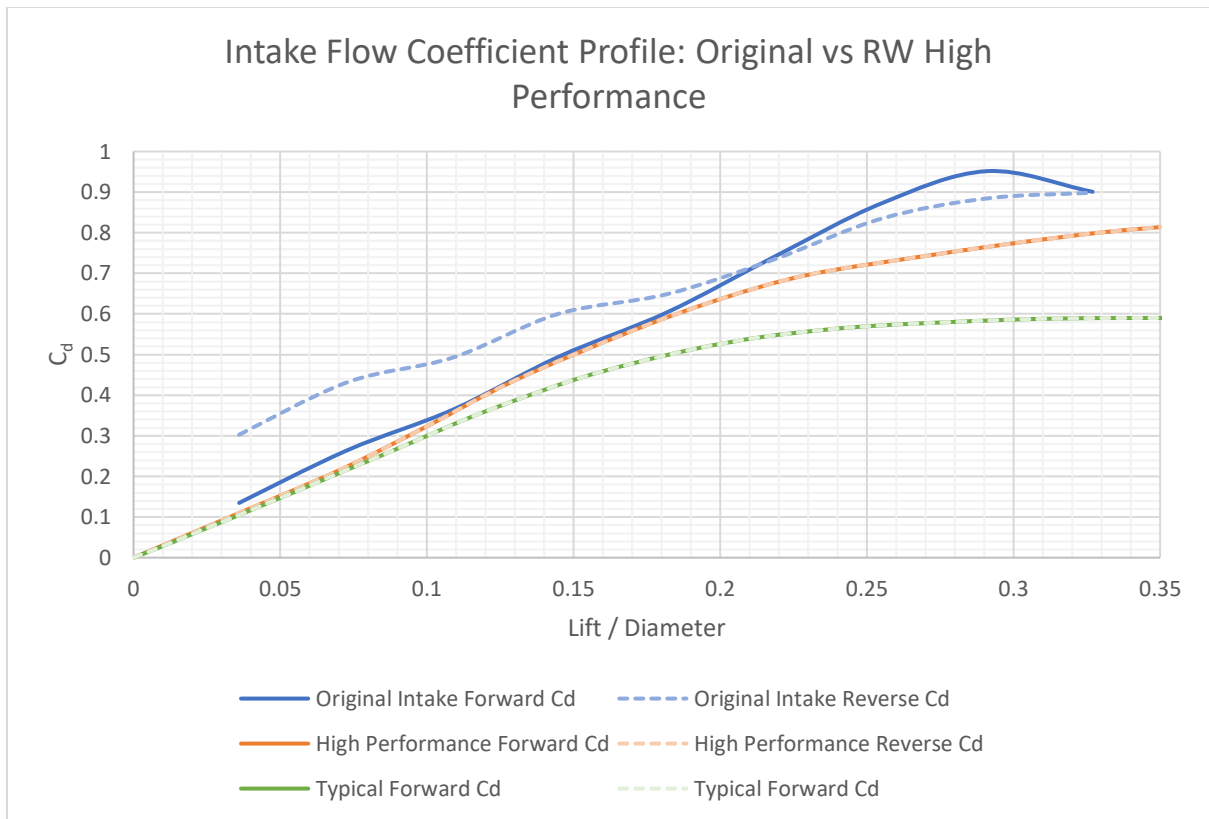


Figure 4-14: Intake Flow Coefficient Profile – Original vs RW High Performance

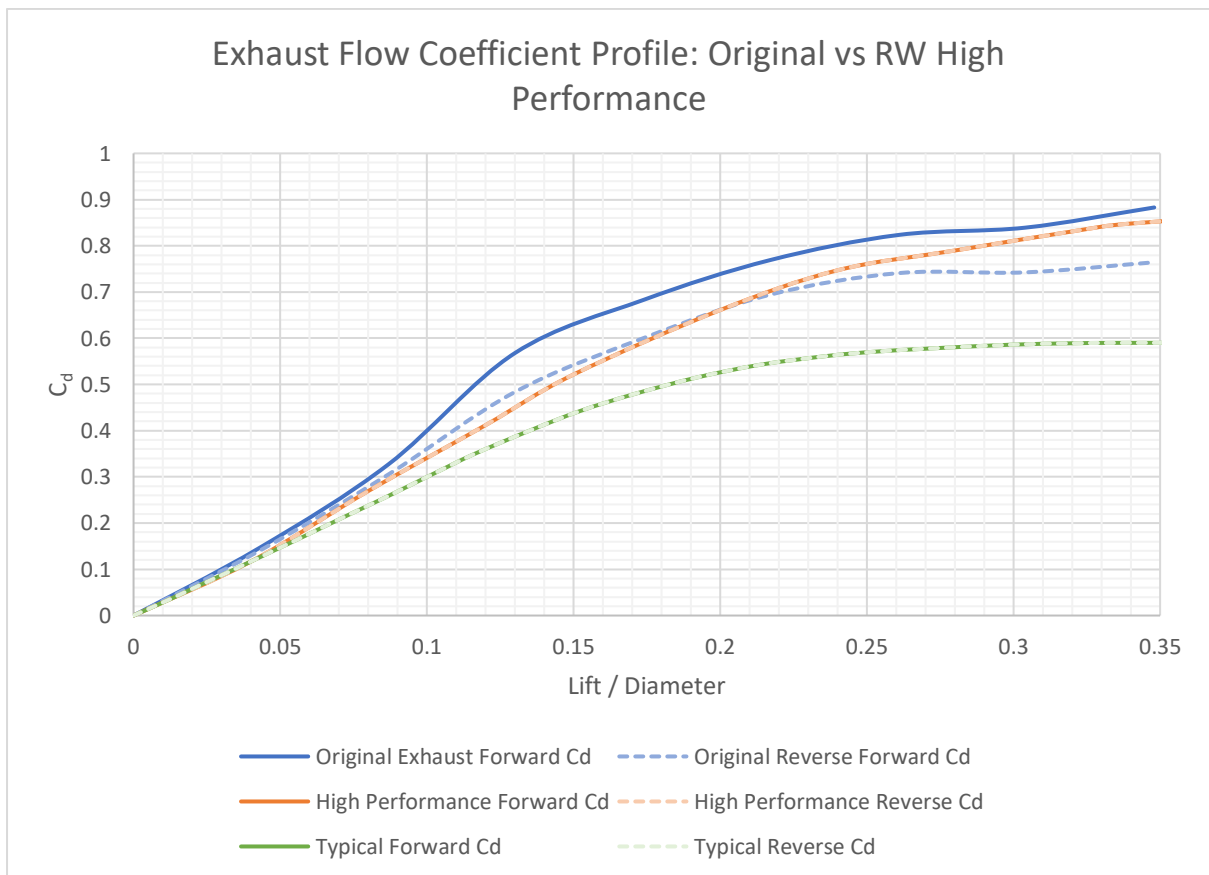


Figure 4-15: Exhaust Flow Coefficient Profile – Original vs RW High Performance

The most accurate result was achieved with typical intake and original exhaust C_d profiles. While running the experiments author has decided to use either the original corresponding value or the value from Table 1, since author had no data on any other values which Triumph may have used.

The final values used by the author were: original duration and lift for both camshafts as well as real timing for both camshafts, throttle diameter and the heat transfer multiplier when valves are open was reverted back to 1.0. Author believes this combination to be possibly used in the endurance dyno test since all of the altered parameters occurred on both camshafts and most importantly, all of those changes are relatively easy to implement, unlike the new camshaft profiles.

Author had to alter the method of results acquisition in this section, please refer to Appendix 1 for further details.

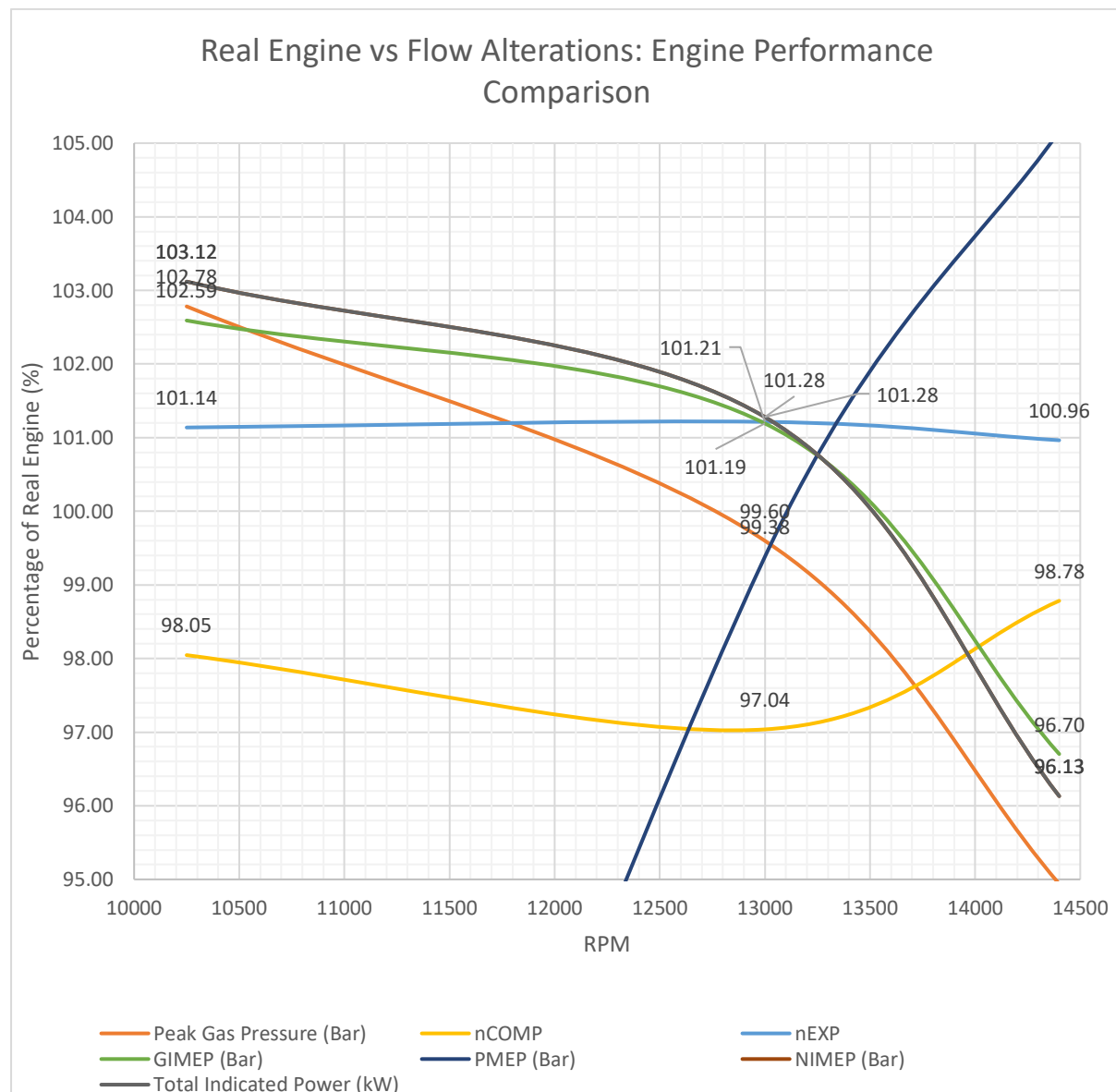


Figure 4-16: Real Engine vs Flow Alterations: Engine Performance Comparison

Figure 4-16 shows the accuracy of the most relevant engine performance parameters within the +/- 5% inaccuracy. The only parameter which author wasn't able to achieve the set accuracy was PMEP @ 10,250 since that value was 20.16% inaccurate. However, due to the relatively low absolute target value of -0.34 Bar, the mentioned inaccuracy had minuscule effect on the total accuracy of the model. Author has deemed these results to be good enough primarily due to the appalling lack of data which is required to validate model components in more detail. As Figure 4-17 shows, the only parameters which average inaccuracy was worse than 5% were: Wiebe m, Wiebe a (values which don't correspond well to what they represent, as shown in section 4.2) and peak gas position, majority of which inaccuracy was due to the lack of precision in the plot of crank angle. For those reasons the effect of the inaccuracy is miniscule. Ultimately, what matters the most is the accuracy of the NIMEP, which average inaccuracy was 0.18%. However, author doesn't acknowledge the model to be accurate since the standard deviation of NIMEP is nearly 3%, which is at least 2% too many. As it has already been mentioned, this is primarily due to lack of data. However, author believes it is crucial to also acknowledge the fact that the simulated values are being compared to real cylinder pressures, of which only 1 set was supplied. Therefore, it is impossible to determine whether or not the real cylinder pressures used as the base of the validation procedure are average and not 'special', since cycle to cycle variation will naturally occur in any combustion engine due to the complexity and as a result, lack of control of all variables which untimely impact the measured cylinder pressures.

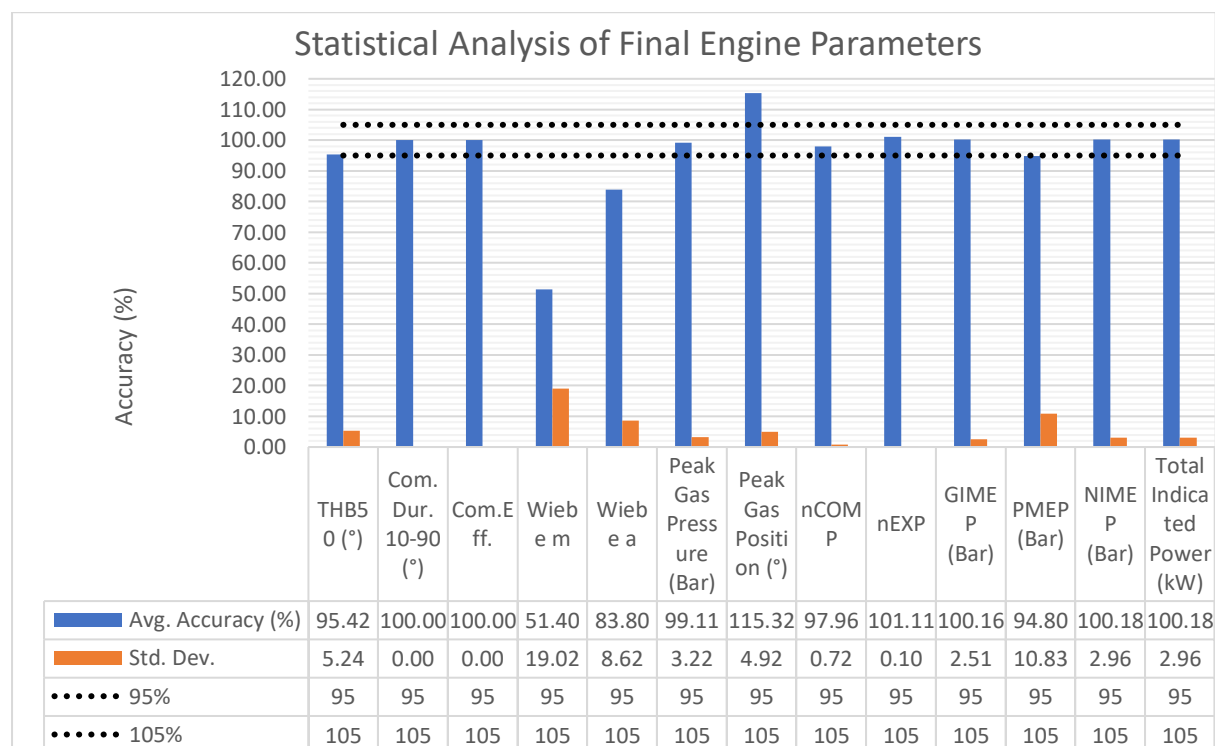


Figure 4-17: Statistical Analysis of Final Engine Parameters

Figure 4-17 shows the statistical summary of the final engine performance parameter accuracies. Out of all mean effective pressure types, the PMEP has been modelled the worst, due to average inaccuracy of 5.2% and the standard deviation of astonishing 10.83%. Despite the poor results, as it already has been mentioned the effect of PMEP accuracy has relatively small effect on the total model accuracy. Furthermore, half dimensional models such as Ricardo Wave (half dimensional due to 1D modelling of the flow components but 0D modelling of the combustion chamber) are supposed to be used as an estimated comparison tool once ‘satisfactory accuracy’ have been achieved, since the change in the model will not affect the nature of combustion process. The natural consequence is that it is expected to compute much more inaccurate results compared to more advanced modelling techniques, making the results from Figure 4-17 acceptable despite being inaccurate.

Lastly, author has plotted comparison of indicated power between the real and the simulated engines, due to much better concept of power compared to % of power, this can be viewed in Appendix 2. It further shows that the accuracy is acceptable. However, the relatively big drop in power and increase of inaccuracy at 14,400rpm suggests that the model is likely to produce unreliable results outside of the validated rpm range, especially since author doesn’t have the data to directly retrieve important model parameters such as combustion duration.

4.5 Friction Correlation & Final Model Accuracy

Break performance parameters are typically the ones used to describe the engine performance since it considers all of the engine losses. To achieve valid results, author had to validate the friction model. Ricardo Wave employs Chen-Flynn empirical polynomial type friction model, as shown by Equation 2, which was modified to consider the entire engine cycle, as shown by Equation 3.

$$FMEP = A_{cf} + B_{cf} * P_{max} + C_{cf} * S_f + Q_{cf} * S_f^2$$

Equation 2: Chen-Flynn friction model (x-engineer, 2020)

$$FMEP = A_{cf} + \frac{1}{ncyl} * \sum_{i=1}^{ncyl} [B_{cf} * (P_{cyl})_i + C_{cf} * (S_f)_i + Q_{cf} * (S_f)_i^2]$$

Equation 3: Ricardo Wave friction model (Gilani, 2012)

The user inputs required by Ricardo Wave are A_{cf} , B_{cf} , C_{cf} and Q_{cf} , which respectfully correspond to accessory friction, cylinder pressure coefficient, piston speed coefficient and quadratic speed coefficient (x-engineer, 2020).

To validate friction, author has used the provided power and torque (which was converted into power) of 94kW @ 12,600rpm and 73Nm @ 11,700rpm. Author wasn't able to acquire engine dyno data as the model was of a development engine, not of the production unit.

To calculate the required input parameters at those rpm, author has used the known values at known parameters to create an equation, which was then used to calculate the values at required rpm, as shown by Figure 4-18 and Table 2.

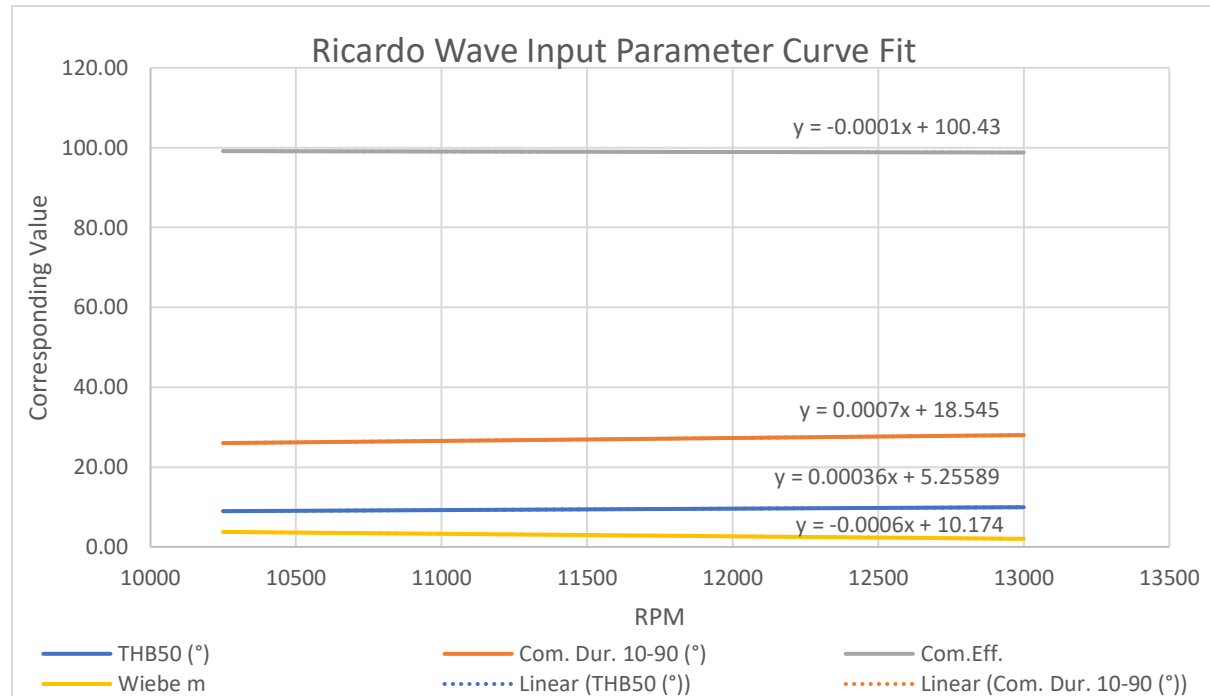


Figure 4-18: Ricardo Wave Input Parameter Curve Fit

Table of Input Values		
RPM	11750	12600
Burn Duration (°)	26.77	27.37
Comb. Eff. (%)	99.26	99.17
Wiebe m	3.12	2.61
THB 50 (°)	9.49	9.79

Table 2: Input Values

The same method was used for all other rpm points. To validate the entire rpm range author had to use dynamometer data, which measured power at the wheel. To accurately read the plotted values from a picture of a graph, author used ImageJ software, as shown in Figure 4-19. Once data was acquired, author calculated the power ratio between the newly acquired power at the wheel to the advertised power at the crank, as shown in Figure 4-20. The equation of the line was then used to calculate the estimated power ratio at every rpm step, which was then multiplied by the corresponding wheel power, the result of which can be seen in Figure 4-21.

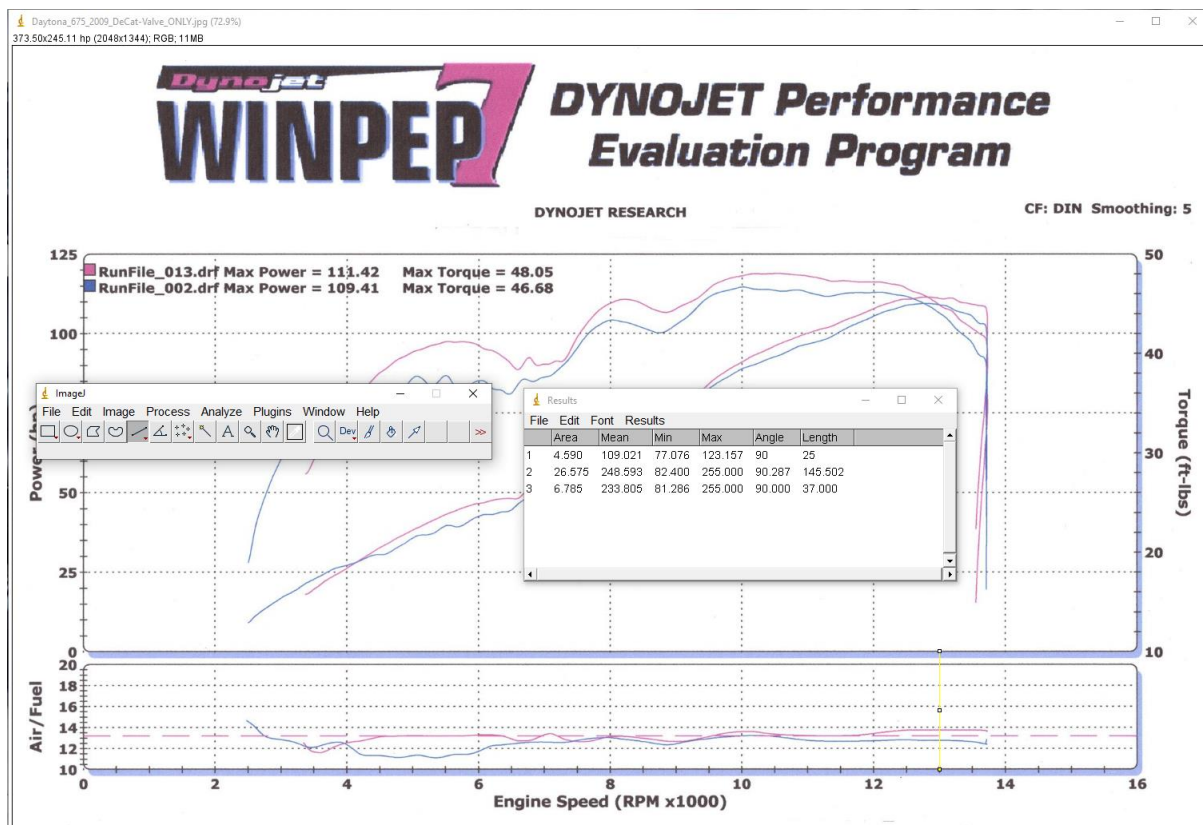


Figure 4-19: ImageJ Software screenshot (ukracesupport, 2020)

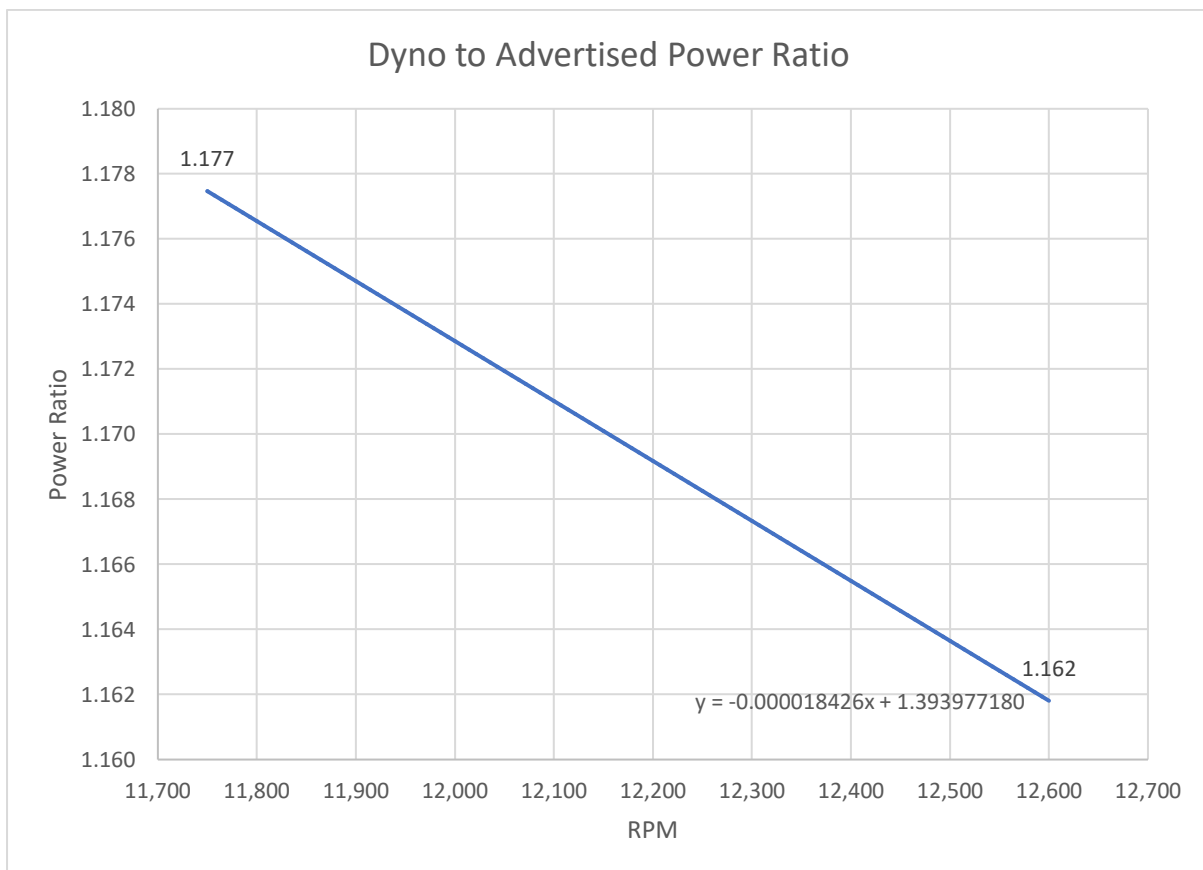


Figure 4-20: Dyno to Advertised Power Ratio

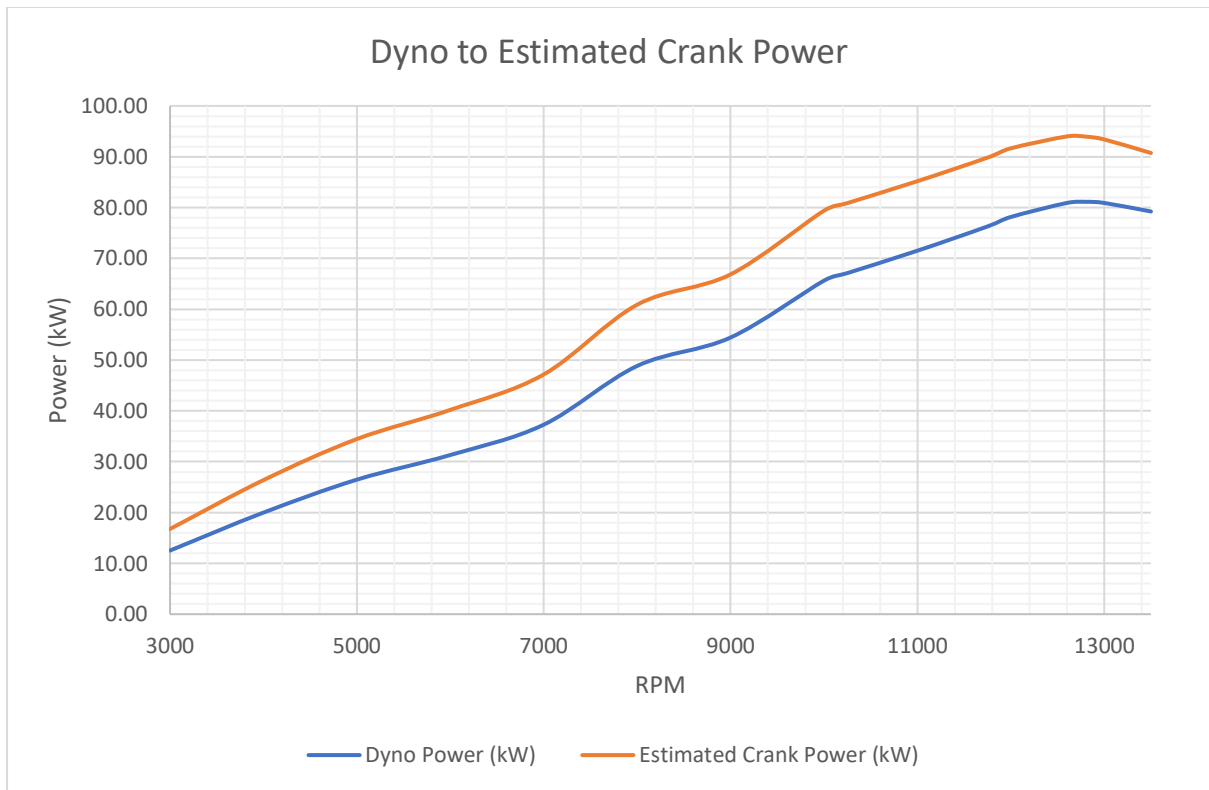


Figure 4-21: Dyno to Estimated Crank Power

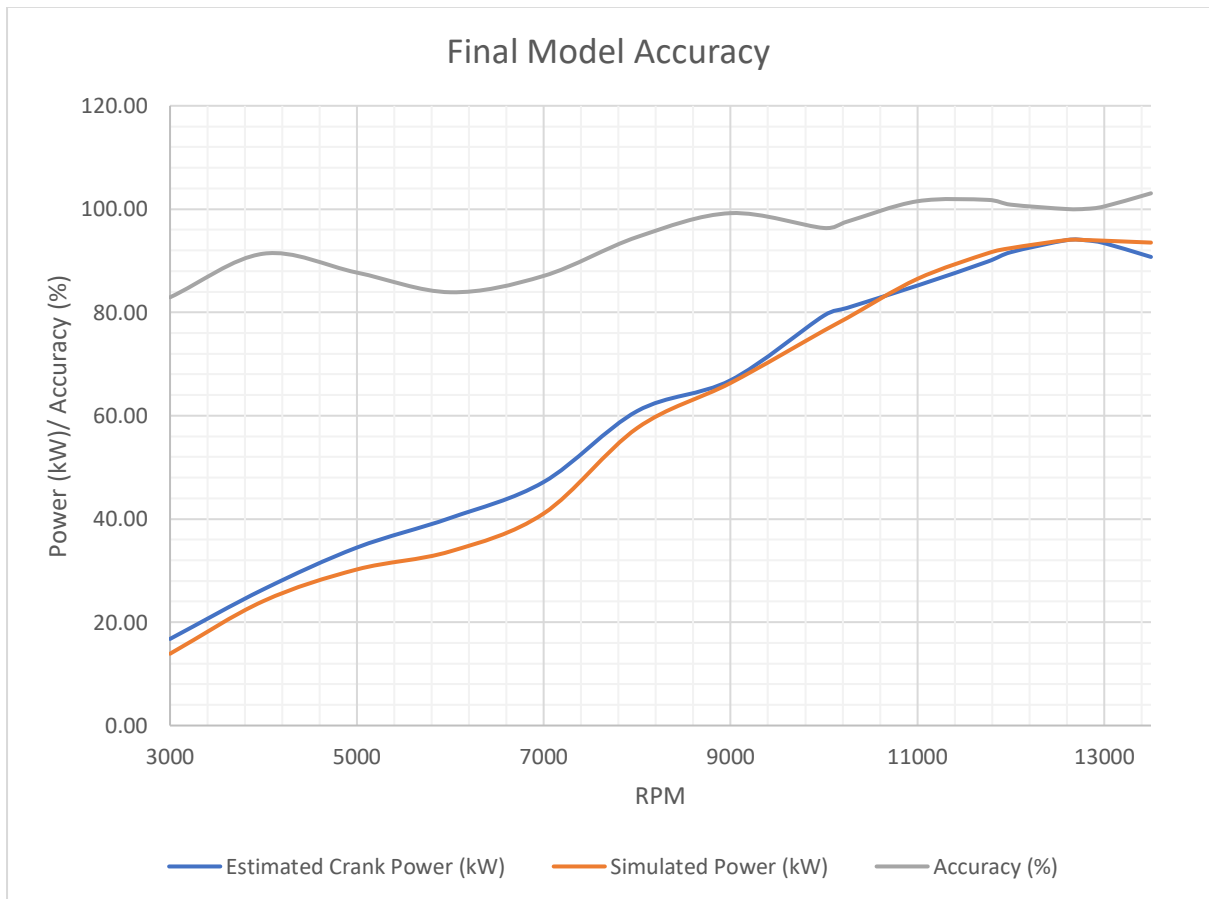


Figure 4-22: Final Model Accuracy

Figure 4-22 shows the final model accuracy, including the friction model validation. Author used the experiment tool to vary the four friction model coefficients to match the target power values, as shown in Table 3. Considering the amount of estimation carried out, author was satisfied with the results, the mean accuracy over the entire rpm range was 95.53% and the standard deviation was 6.56%. However, the mean and standard deviation of accuracy between 9,000 and 13,500rpm increased to 100.09% and 1.89% respectively. Author believes this to be mainly caused by a design alternation which was carried out for the production engine, author suspected the 3-2-1 exhaust system.

However, due to lack of data, author was not able to increase the accuracy of the model or the accuracy of the base comparison values, used to validate the model, any further. Considering the plethora of unknown information, author has deemed the validation to be complete since +/- 5% inaccuracy has been achieved.

Friction Model Experiment		
	RW recommended	Used
Acf (Bar)	0.5	0.0
Bcf	0.006	0.0
Ccf (Pa min m ⁻¹)	600	450
Qcf (Pa min ² m ⁻²)	0.2	0.2

Table 3: Friction Model Experiment

5 Conclusion

Author believes the Ricardo Wave software to be a powerful tool which, if properly applied, can decrease the engine development time as well as improve the engine performance.

The results found in this document show that an inaccuracy of $\pm 5\%$ can be expected for the majority of the simulated results. This was achieved with relatively little data and knowledge of the real engine, especially due to the prototype nature.

Furthermore, the improvements applied to the model were done in relatively short time, especially compared to more advanced software such as Converge, making Ricardo Wave a tool which can be realistically used in competitive professional environments.

As Figure 4-22 has shown, the achieved total model accuracy throughout the entire rpm range was impressive and most likely good enough to be used in further development: altering engine design and comparing the % of change of power output.

However, these results may be misleading since the friction model was adjusted and compared against the converted power at the wheel using Dynojet dynamometer, which from author's experience always measures inflated figures. Furthermore, author used the official Triumph release figures, of which there are only 2. These could have been inflated as means of advertisement and more importantly, author believes the Ricardo Wave model to be of earlier prototype engine as mentioned in previous sections, making the 'calibration' values to be incorrect as well.

The displayed work has also shown that the simulated results should only be used to define the ballpark in which the real-life values might be expected. Author believes that the ballpark of the particular model discussed in this document could be greatly improved with relatively simple information such as the data of air consumption or schematics of the engine's components.

Lastly, author has only validated the engine power output as well as any characteristics which might impact it. Therefore, more validation work would have to be carried out to determine if other aspects of the Ricardo Wave software correspond to real life, such as the simulation of the NO_x production.

6 References

Ape Raceparts, 2018. *TECH/ SPECS*. [Online]

Available at: <http://aperaceparts.com/tech/2009%20Triumph%20Daytona%20675.html>

[Accessed 2 October 2019].

Campisi, J., 2017. *Episode 6 - Removing a Daytona 675R Stock Exhaust!*. [Online]

Available at: <https://www.youtube.com/watch?v=Mz-DMpomdQI>

[Accessed 10 November 2019].

Engineering ToolBox, 2019. *Fuels - Higher and Lower Calorific Values*. [Online]

Available at: https://www.engineeringtoolbox.com/fuels-higher-calorific-values-d_169.html

[Accessed 29 Decemeber 2019].

Gilani, R., 2012. Engine Simulation Model for a Formula. *Applied Design, Development, Correlation and Optimization*.

Racetechinques, 2019. *Porshce M96 Summer Internship* [Interview] (August 2019).

Triumph Designs Ltd., 2008. Daytona 675 Motorcycle Race Kit Manual. Issue 2.

ukracesupport, 2020. *Triumph Daytona 675/R & Street Triple/R 2006-16 Venturi System Performance Velocity Stacks*. [Online]

Available at: <https://www.ukracesupport.com/products/triumph-daytona-675-r-street-triple-r-2006-12-ukrs-performance-velocity-stacks>

[Accessed 9 January 2020].

Williams, D. O., 2019. Optimisation of a high speed 3 cylinder using the Ricardo WAVE engine development software. *LEVEL 6 ASSESSMENT SPECIFICATION*.

x-engineer, 2020. *Mechanical efficiency and friction mean effective pressure (FMEP)*. [Online]

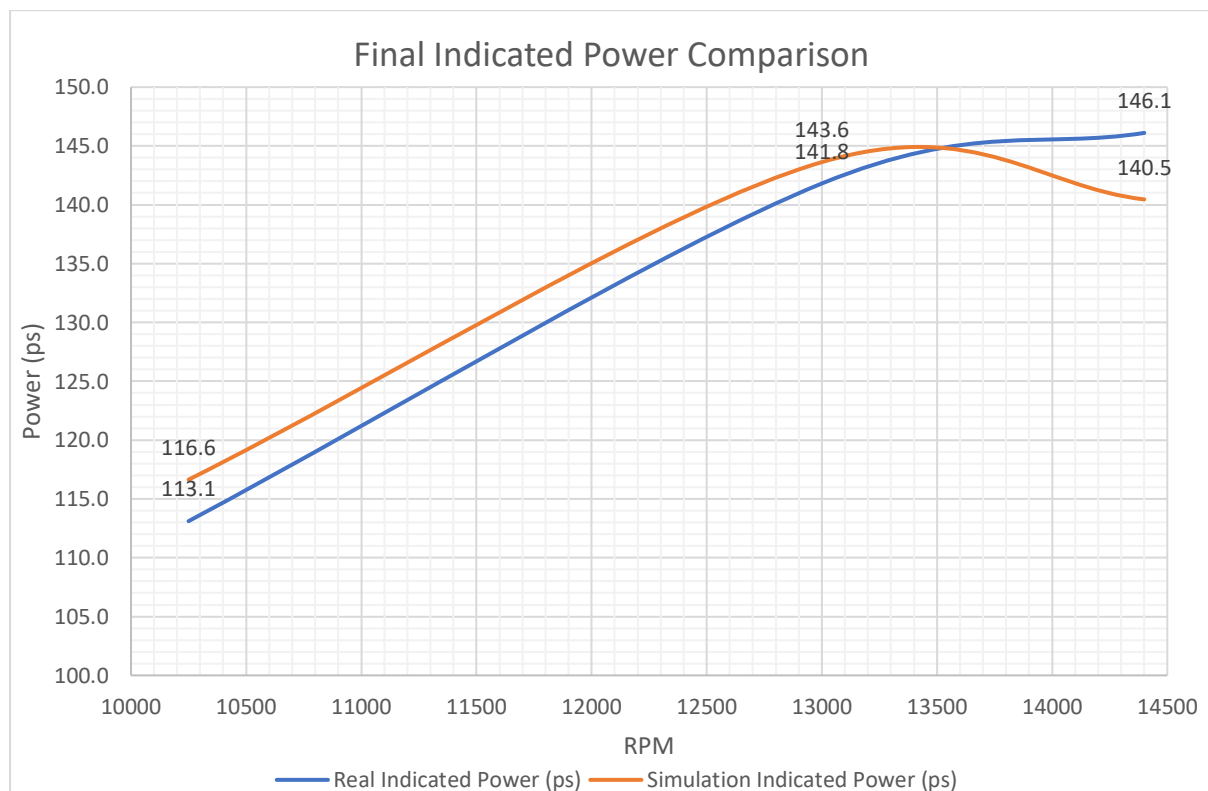
Available at: <https://x-engineer.org/automotive-engineering/internal-combustion-engines/performance/mechanical-efficiency-friction-mean-effective-pressure-fmep/>

[Accessed 8 Januray 2020].

7 Appendix

Up until section 4.4: Flow Alterations, all final engine performance parameter values were retrieved from the cylinder 1 to cut the model development time. From author's experience this method is almost always acceptable up until the very last steps of the model development cycle. However, due to high cylinder to cylinder variation in the model, author had to take all 3 cylinders into consideration in section 4.4, since this is the very last section where validation of the NMEP takes place. As a result, all of the corresponding values apart from nCOMP and nEXP, in Figure 4-16, have been directly taken from the Ricardo WavePost software, as oppose to calculating them from the simulated cylinder pressures. This means that direct comparison of the engine performance parameter accuracies in the previous sections isn't valid.

Appendix 1: further insight on the result acquisition method



Appendix 2: Final Indicated Power Comparison