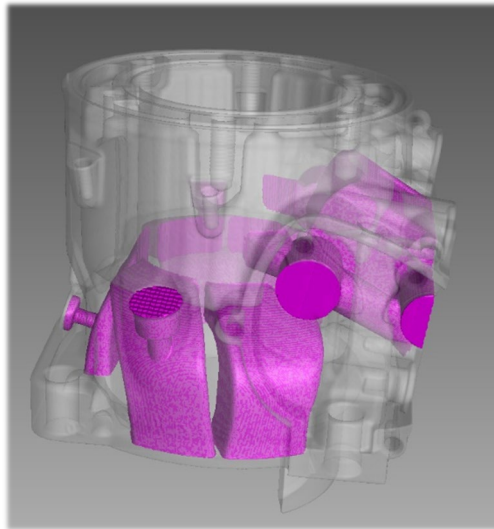


University of Wales Trinity Saint David

School of Engineering



**An Investigation into Controlled Auto-Ignition & its Implementation
to the Modern High-Performance Two-Stroke**



Juan Sebastian Vacas (P138703)

Advanced Engine Design

24/04/2018

Declaration of Authenticity

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

Signed:

Date:

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Abstract

This body of work will begin by conducting a thorough literature review of two-stroke controlled auto-ignition. The focus of this literature review will be to explore design criteria both physical and theoretical, proving the validity of this concept. Having proved concept validity, the focus will then shift to the analysis of physical components and engine pressure data from a current KTM 250cc TPI fuel injected engine. The engine pressure data will be analysed through an Excel combustion model, to better understand the different components of combustion and their relevance to controlled auto-ignition. Combining with the excel model will be a 1D simulation model built in the Engmod2T simulation package, the aim being to predict residual gas values. The aim is to arrive at a point where a clear and confident strategy for the implementation and feasibility of two-stroke controlled auto-ignition can be stated and the theoretical potential of the subject engine reviewed.

Contents

Chapter 1 - Introduction	8
1.1 – Background & Introduction.....	8
1.2 – Outline.....	10
Chapter 2 – Project Planning and Management.....	12
2.1 – Approach	12
2.2 - Resourcing.....	12
2.3 – Time Management.....	12
Chapter 3 – Literature Review and Research.....	13
3.1 – Strategy	13
3.2 – Timeline.....	17
3.3 – Summary	20
Chapter 4 – Geometric Investigation	29
4.1 – Introduction	29
4.2 – Methods	29
4.3 – Comparison	31
4.3.1 – Ports	31
4.3.2 – Cylinder Heads	33
4.3.3 – Exhaust Valves.....	35
Chapter 5 – Excel Model	38
5.1 – Structure & Layout	38
5.2 – Engine Basics	38
5.3 – Displacement & Volume	39
5.4 – Pressure.....	40
5.4.1 – Pressure Graphs	40
5.4.2 – PV Diagrams	43
5.5 – Work & IMEP	45
5.6 – Log Graphs	47
5.6.1 – Graphical Analysis	48
5.6.2 – Sensitivity Analysis (Combustion Chamber Volume)	50
5.7 – Polytropic Index	51
5.7.1 – Slope Calculation.....	51
5.7.2 – Degree Based	51
5.7.4 – Range Selection (Step Change)	55
5.7.5– Range Selection (Pressure Difference).....	55
5.7.6 – Range Selection (Log PV Visual)	59

5.7.7 – KTM Values.....	65
5.8 – Combustion Timing	67
5.8.1 – Ignition Timing	67
5.8.2 – Delay & Duration.....	68
5.9 – Mass Fraction Burn	70
5.9.1 – Calculations	70
5.9.2 – Analysis.....	73
5.10 – Vibe Function	76
5.10.1 – Method & Calculations.....	77
5.10.2 – Graphing & Analysis	78
5.10.3 – Function Retro-Fit	78
Chapter 6 - Excel Model (Combustion Chemistry).....	80
6.1 – Chemistry Basics.....	80
6.2 –Mass Calculations and Purity	81
6.3 – Specific Gas Constants	83
6.4 – Specific Heats	86
Chapter 7 – Auto-Ignition Calculations	89
7.1 – Purity & Mass Comparison.....	89
7.2 – Temperature at Exhaust Port Closure.....	91
7.3 – Auto-Ignition Temperature Calculation Methodology	94
7.4 – Polytropic Index (Expanded Range Selection)	95
7.5 – Final Temperature Calculations	98
Chapter 8 – Conclusion	100
Chapter 9 – Further Work.....	101
Appendices.....	102
Appendix A - 1D Simulation Model.....	102
A.1 - Software Package	102
A.2 – Dat2T (Data Entry)	102
A.2.1 – Introduction	102
A2.2 – Engine	103
A.2.3 – Exhaust Port.....	105
A.2.4 – Transfer Port	107
A.2.5 – Inlet Port	109
A.2.6 – Exhaust System	111
A.2.7 – Inlet System	112
A.2.8 – Combustion.....	113

A.2.9 – Temperatures	114
A.3 – EngMod2T (Simulation)	115
A.3.1 – Start Screen.....	115
A.3.2 – Run Screen	115
A.3.3 – Exit Screen.....	116
A.4 – Post2T (Post Processor)	116
A.4.1 – Plot Options	116
A.4.2 – Plot Display	117
A.4.2 – Excel Data Import.....	117
A.4.3 – Validation.....	118
Appendix B - Excel.....	119
B.1 – Original Data	119
B.2 – Data Sheet.....	120
B.3 – Measurement & Units.....	120
B.4 – Combustion Chemistry.....	121
B.5 – Pivot Charts.....	121
B.6 – Graphs.....	122
B.7 – Combustion Timing Experiment	122
B.8 – Versioning	123
B.9 – Low Load / High Load.....	124
Bibliography & References.....	125
Nomenclature & Abbreviations	133
Figures & Equations	135

Chapter 1 - Introduction

1.1 – Background & Introduction

In normal two-stroke engines there is often misfiring and a high percentage of residual exhaust gas at light loads especially around idle. In the book “*HCCI and CAI Engines for the Automotive Industry*” (Zhao, 2007) it is mentioned how the residual gas amount can be more than 80%. In scenarios such as this it takes several cycles to scavenge the residual gas and allow the gas balance to return to a level suitable for spark ignition. When the load graduates past this low load point the residual gases do start to decrease, but not sufficiently and as such there is still incomplete combustion which creates a cyclic speed variation, or the phenomena known as surging (almost as if the engine is running away with itself, tripping up and then repeating). The engine does not begin to run completely smoothly until load and engine speed increases to a level where the residual gas drops sufficiently to allow constant and complete spark ignition. The majority of two stroke engines would begin to auto ignite if the load was kept low and the engine speed increased but the reality of real world driving conditions dictates that this does not happen.

So, the question is how this inherent disadvantage be turned to an advantage and how can this be implemented in a modern two-stroke engine?

Auto-ignition is referred to with a rather large number of acronyms depending on the author and source, though loosely speaking two different scenarios can be defined;

1. CAI “Controlled Auto-Ignition”- (ATAC, AR, IAPAC);

ATAC = Active Thermo-Atmosphere Combustion (Shingeru Onishi)

AR = Active Radical Combustion (Honda Motor Co Ltd)

IAPAC= Injection Assistee Par Air Comprime (IFP, France)

This category can be described as auto-ignition in the two-stroke engine at low load / part throttle where there is a high percentage of residual gas.

2. HCCI “Homogeneous Charge Compression Ignition”

Described as auto-ignition where the precursory bias or defining catalyst is high compression ratio, generally adopted in four-stroke-engines.

Two-Stroke CAI (Controlled Auto-Ignition)

A number of different development paths have been followed from the early 70's through to present, in terms of CAI for the high performance two-stroke it seems almost all of the work has been conducted by Honda Motor Co Ltd (Ishibashi et al). In all the research papers on CAI Honda have achieved the necessary levels of residual gas using an exhaust trapping valve. Modern high performance two strokes already are fitted with exhaust valves to modulate the exhaust port height / area and thus improve performance. Specifically, in terms of integration it would be necessary that the exhaust valve had a more extreme range of potential closure vs a conventional engine – it would also be necessary to have electronic control of the exhaust valve.

Current Technology:

Currently most two-stroke bikes produced are still carburetted, with KTM only just having begun to adopt Transfer Port Injection (TPI) - though it is generally acknowledged that incremental improvements such as this won't be sufficient in the long term. To enable the longer term commercial viability of two-strokes it is almost certain stratified scavenging / direct injection will need to be adopted to negate the inherent problem of short circuited mixture. If at the same time CAI can be implemented, then certainly it is a subject worth investigating. Interestingly Hondas development path naturally converged, combining CAI and direct injection to bring specific reciprocal as well as individual advantages.

Figure 1 shows direct injected engines developed by Honda from 1997 to 2007, the 1997 engine (Ishibashi, et al., 1997) uses in cylinder charge and a conventional injector to enable air assisted direct injection. The 2007 engine (Nishida, et al., 2007) incorporates a much more complicated system and modified combustion chamber targeting residual gas hot spots and thus enlarging the CAI operational window (almost until idle).

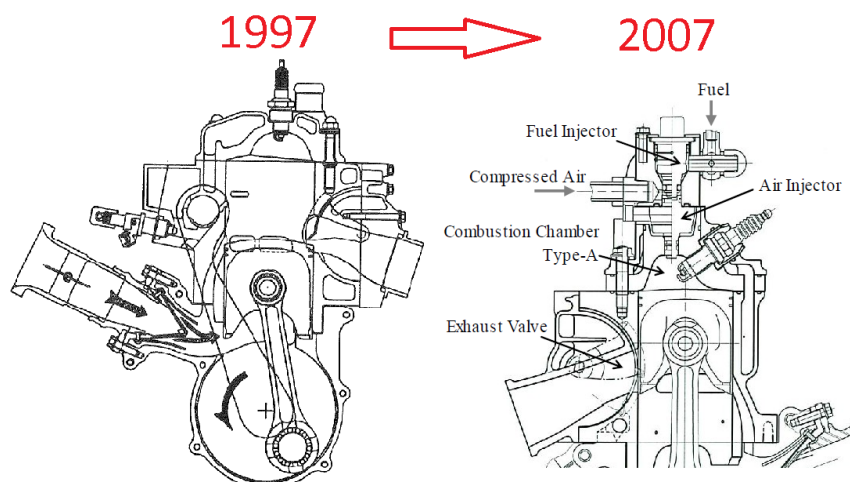


Figure 1 - CAI Direct Injection (Ishibashi, et al., 1997) & (Nishida, et al., 2007)

1.2 – Outline

- **Chapter 1: *Introduction***

This chapter covers a quick review of background information regarding two-stroke CAI and presents the approach and outline of the project.

- **Chapter 2: *Literature Review and Research***

This chapter details the strategy and approach that has been taken to support and validate the project through an extensive literature review, whilst also providing key pieces of information.

- **Chapter 3: *Project Planning and Management***

This chapter details very briefly the approach, acquisition of resources and time management that was necessary to successfully complete the project.

- **Chapter 4: *Geometric Investigation***

This chapter is an investigative exercise reviewing a cylinder and cylinder head from Hondas commercially released 250cc CAI engine and comparing it with those from its closely related SI predecessor and the supplied KTM engine.

- **Chapter 5: *Excel Model***

This chapter focuses on a basic but fundamentally sound analysis of the engine pressure data supplied by KTM and the various linked attributes that can be extrapolated.

- **Chapter 6: *Excel Model (Combustion Chemistry)***

This chapter takes the information from chapters 5&6 and uses it analyse the pre-combustion gas exchange process and estimate a temperature at exhaust port close and subsequently whether the in-cylinder conditions meet the requirement for CAI.

- **Chapter 7: *Auto-Ignition Calculations***

This chapter covers the final considerations and calculations to predict the possibility of auto-ignition.

- **Chapter 8: *Analysis***

This chapter predominantly cover analysis of the work conducted in 4-7, looking at any fundamental issues areas of interest that have arisen and the project as a whole.

- **Chapter 8: *Conclusion & Further Work***

This chapter overlaps slightly with analysis, but the purpose of this chapter is to draw upon the work done and contrast it with research to clearly state the future direction for body of work.

- **Appendix A: *1D Simulation Model***

This appendix shows the creation of 1D simulation model in the software “EngMod2T” with the aim of analysing the residual gas concentration in the KTM engine. This model will be created using three sources of data; supplied data such as ignition timing, combustion duration and vibe constants from the excel model and physical geometry data from the parts supplied by KTM.

- **Appendix B: *Excel***

This Appendix covers a brief view of the structure and content indicative of the excel spreadsheets created for chapters 5 & 6.

- **Bibliography & References: *Citations***

This contains all cited citations, reference list and reading list most relevant reading from the literature review.

- **Nomenclature & Abbreviations: *Supplementary Information***

This section contains all supporting mathematical / scientific nomenclature and any abbreviations used.

- **Figures & Equations: *Supplementary Information***

This section contains 2 lists, one for all figures used throughout and a second for all equations used.

Chapter 2 – Project Planning and Management

2.1 – Approach

Table 1 - Project Approach (Vacas, Authors Own)

Gateway	Task	1-2014	2-2014	3-2014	4-2014	1-2015	2-2015	3-2015	4-2015	1-2016	2-2016	3-2016	4-2016
	General Reading & Research												
	Review of Initial Thoughts												
1*	Academic Guidance (Set & Non-set)												
	Focused Research												
	Resource Planning												

Table 1 shows the very early and graduated approach taken with this project, the project as a defined concept was not present initially, but rather there was the motivation and interest necessary to take pre-emptive approach. This shows the positive impact of building a project around interests and pursuits that are already in some way prevalent in your agenda.

2.2 - Resourcing

An important aspect of project planning covered in tables 1 and 2 was the all-important element of resource planning, whether in academia or business it is essential that resources are available when needed. Resource planning became one of the earliest focal points and without the early and graduated approach the resources and help acquired would have been impossible to obtain. The pre-emptive approach proved successful and meant that throughout the project there were little, or no hold ups due to resources – meaning that the focal bias could be placed upon academic work.

2.3 – Time Management

Table 2 - Project Planning (Vacas, Authors Own)

Gateway	Task	Jan-17	Feb-17	Mar-17	Apr-17	May-17	Jun-17	Jul-17	Aug-17	Sep-17	Oct-17	Nov-17	Dec-17
2*	Interim Proposal (for Industrial Partners)												
	Academic Guidance (Set & Non-set)												
	Focused Research												
3*	Resource Acquisition (Physical/Non Physical)												
	Geometric Analysis of Parts												
	Concept Investigation (CAD)												
	Dissertation Report Write Up												
	Combustion Modelling												
4	Interim Oral Examination (Viva 1)												
5*	Interim Report (for Industrial Partners)												

Table 2 shows what is still a pre-emptive approach, but it also showed how planning changed as the academic year approached, as well as having self-prescribed tasks it was now important to start implementing gateways comprised of tasks such as academic presentations but also including some essential tasks such as combustion data acquisition.

Chapter 3 – Literature Review and Research

3.1 – Strategy

Whether consciously implemented or present by default, strategy is ever important when reviewing literature and conducting research. This chapter aims to display how a broad, diverse and multi-faceted approach was taken to produce a meaningful body of information from which a successful project could be conducted.

Importantly the literature review did not begin because it had to as an academic pre-requisite, rather more the literature review was a process that continued from where the author already was. This is where the merits of motivation and interest become more than just positive adjectives. Having strong motivation in the subject area means that the author did not need to wait for clear objectives before the literature review began and naturally the level of interest dictated that a wide scope and diverse range of sources were explored including fringe material.

Without a rigid structure there was the freedom to explore without pressure – though retrospectively a simplified overview of how strategy & approach evolved can be seen in 3 stages below;

Stage 1: 2016 onwards

Source Bias: *Magazines, Websites, Forums, People*

The objective of this stage was simply “to investigate” - in terms of content nothing was off bounds - websites, magazines, online forums, books, journals and people. Whilst sources such as websites, magazines and forums are not generally dominant as referencing convention in academia, they do importantly act as bridges between sources. Facts or figures from spurious sources may not always be verifiable but they do spark an investigation – the small pieces of unverified information lead to questioning and through credible sources (books & journal papers etc) you can either credit, discredit or clarify these pieces of information.

As an example, (Takashi, 2003) is an enthusiast run website (Japanese Yahoo GeoCities) detailing the history of Honda's 250cc production bike with CAI – this website is a stand out example of why it is important to employ a less rigid approach to research. Contained within this website was a lot of information regarding the practical application of CAI along with the citation of sources (magazines and technical symposiums) - there was also information (Japanese press releases) on this website that currently has not been found anywhere else.

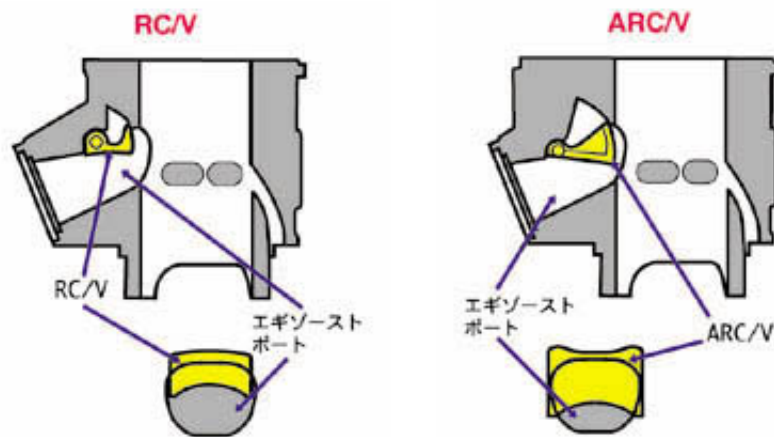


Figure 2 - Exhaust Valve Comparison (Takashi, 2003)

Figure 2 (Takashi, 2003) is a good illustration of the difference between a commercial CAI trapping valve (CRM250AR 1997-1999 **ARC/V**) and that of the traditional exhaust valve (CRM250 1994-1996 **RC/V**). This information would later be verified by geometric investigation of the physical cylinder's / valves.

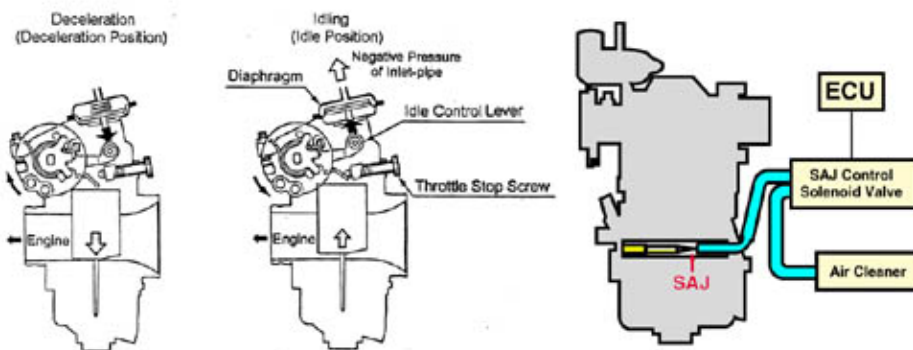


Figure 3 - CRM250AR CAI Deceleration Device (Takashi, 2003)

Furthermore, Figure 3 (Takashi, 2003) shows some aspects of the deceleration device fitted to the Honda's CRM250 AR production bike – this information was of interest as discussed later the control of irregular CAI around idle / deceleration is an important aspect of overall system control and modal transition.

Stage 2 – early 2017 onwards

Source Bias: Technical Papers, Research Journals

The initial broad research in stage 1 (in its own right) procured several pieces of standout information and from the point of progression it gave clear indication of how and where CAI had been implemented commercially into two different production vehicles (Honda CRM250AR and Pantheon 125AR) and one race bike (Honda EXP-2). Looking past this initial information it also provided a number investigative starting points into the specifics of two-

stroke CAI and specifically in relation to motorbikes. Stage one set the scene and gave global focus, stage 2 aimed to dig deeper in terms of mechanisms and a more acute technical focus.

One of the first papers discovered when searching for further technical information was “*Basic Understanding of Activated Radical Combustion and its Two-Stroke Engine Application and Benefits*” (Ishibashi, 2000). Highlighting two key pieces of information from this paper, it gave a target for CAI temperature as well as indicating how changed with AFR Figure 3 (Ishibashi, 2000).

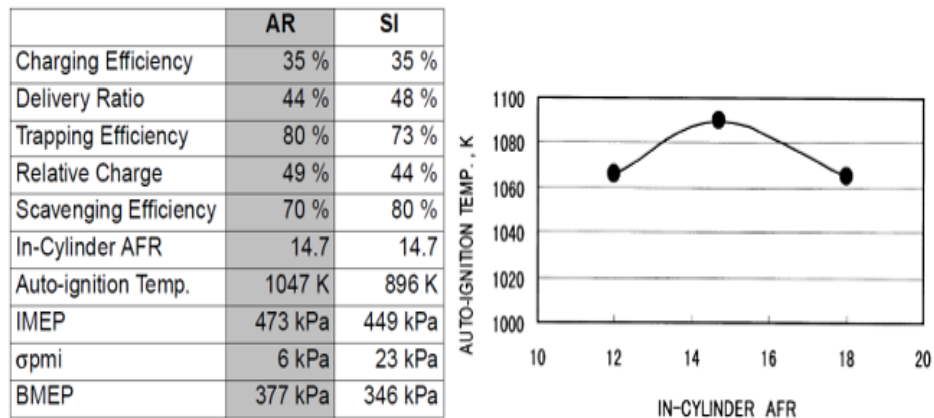


Figure 4 - CAI Temperature (Ishibashi, 2000)

In addition to providing some key technical targets this and other similar papers began to validate one of the core reasons for pursuing this technology. From these technical papers reasons began to resonate and re-affirm the validity of this project. The research was clearly showing that fuel consumption and emissions could be drastically reduced and Figure 5 (Asai, et al., 1995) shows the reduction in BSFC and BSHC of the 400cc EXP-2 Dakar rally bike.

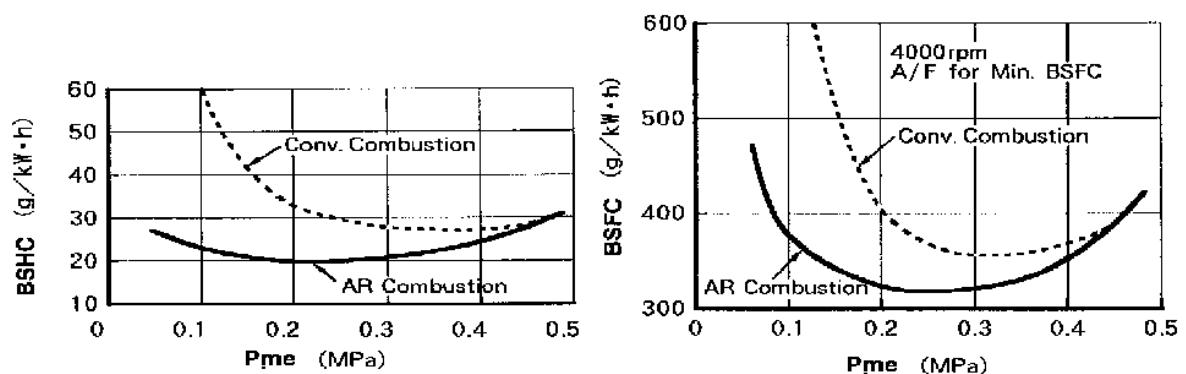


Figure 5 – EXP-2 Emissions & Fuel Consumption (Asai, et al., 1995)

The paper “*Analysis on Fuel Economy Improvement and Exhaust Emission Reduction in a Two-Stroke Engine by Using an exhaust Valve*” (Asai, et al., 1995) not only highlighted the

improvement in BSFC and emissions but also began to highlight how two-stroke CAI could reduce the irregular combustion that is inherent to two-strokes at low load. Figure 6 (Asai, et al., 1995) is a great visual indicator of how this could improve the torque characteristics and thus the drivability of the two-stroke engine,

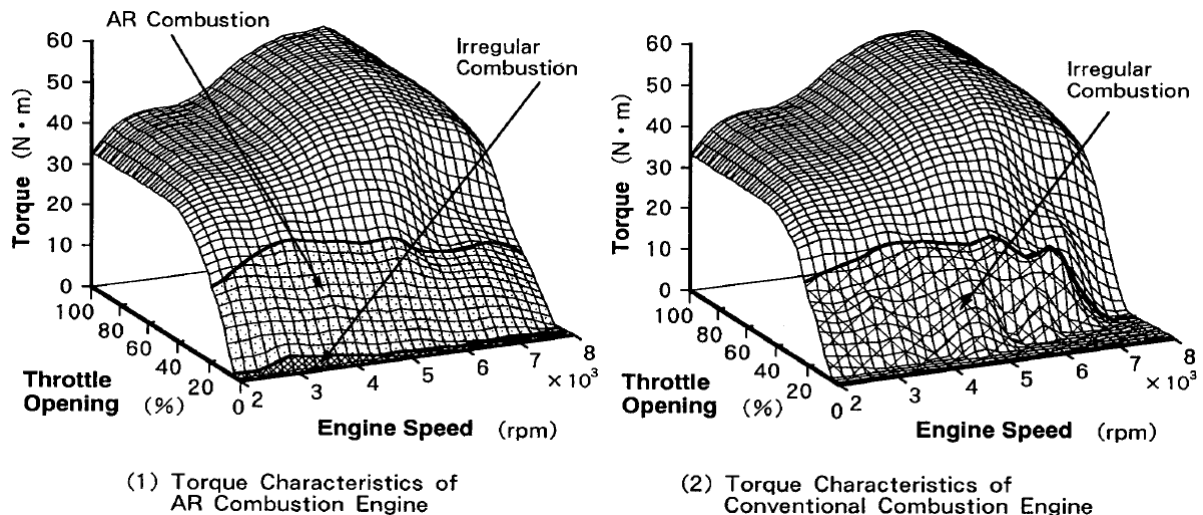


Figure 6 - EXP-2 Torque Characteristics (Asai, et al., 1995)

The characteristics shown in Figure 6 (Asai, et al., 1995) are further displayed in Figure 7 (Asai & Ishibashi, 1996 International Congress & Exposition), here they are shown in terms of cyclic variation of IMEP.

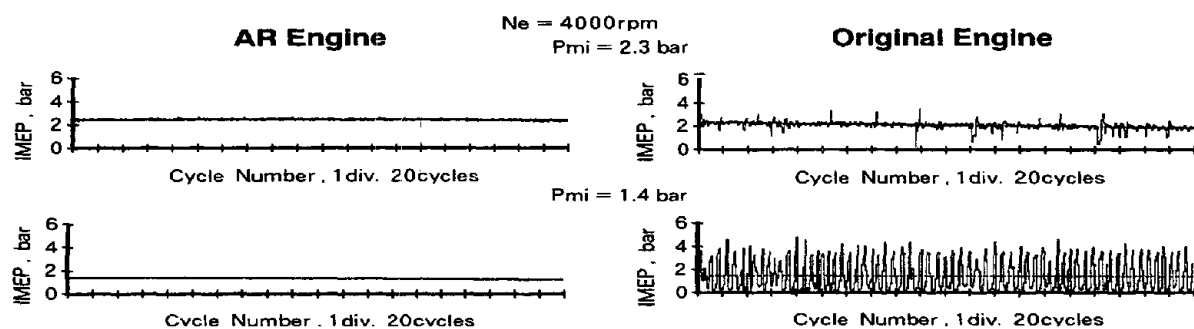


Figure 7 - IMEP Cyclic Variation (Asai & Ishibashi, 1996 International Congress & Exposition)

Stage 3 – mid/late 2017 onwards

Source Bias: *Technical Papers, Research Journals, Patents, Books*

It can be said that the initial research conducted in stages 1 & 2 enabled the author to explore an area of interest and develop and validate the choice of project. Stage 3 continued in much the same manner as Stage 2, but with a greater sense of direction the research could now become more intense and focused.

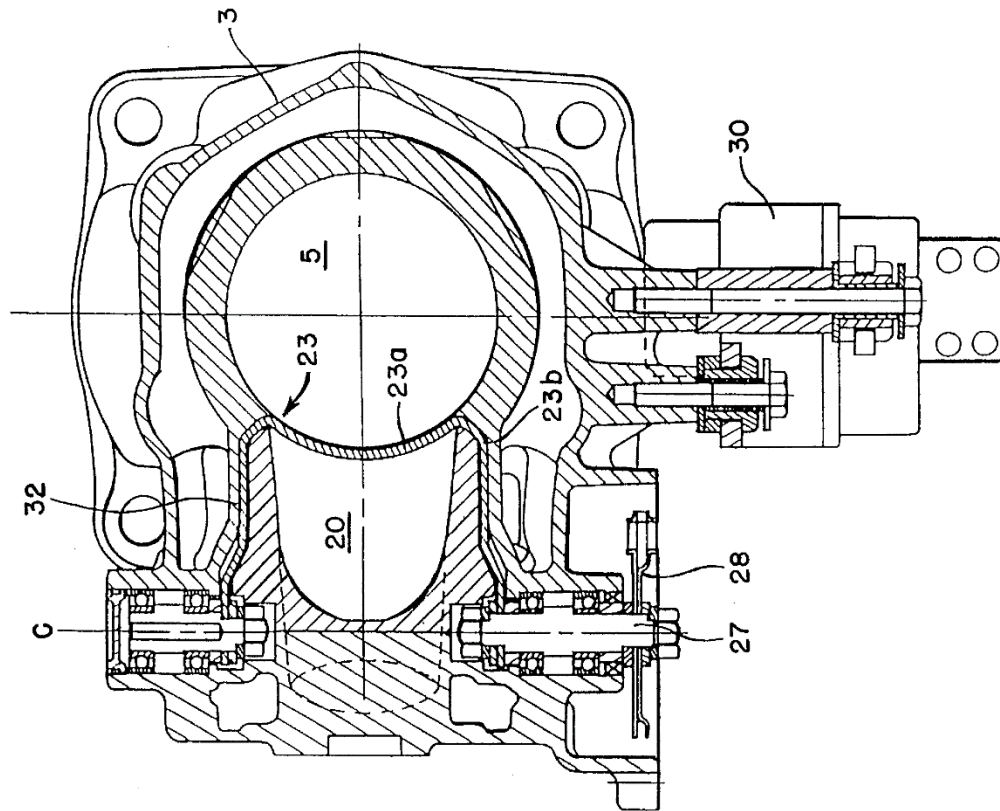


Figure 8 - CAI Exhaust Valve (Wakabayashi, et al., 1994)

One area of interest was the exploration of patents, though the language used in patents is not easy reading, the addition of images / figures can really help understand and link word heavy articles. Often the patents relate directly to journal articles and include much more detailed images of the parts and processes than would be included in typical articles. So, whilst it may seem odd to go from journal papers to looking at pictures, it is in fact a really important tool to gain a better perspective and connect the dots. A good example of this is Figure 8 (Wakabayashi, et al., 1993), from patent “*Throttle-valve control apparatus for spark-ignition two-cycle engines – US5495836*” gives a cross-sectional view of a CAI exhaust throttling valve – showing clearly that certain importance has been put on having the valve seal close to the cylinder.

3.2 – Timeline

The strategy outlined in Chapter 3.1 depicts the quality and breadth of research conducted - but when such a mass of information has been collated it is useful to define the most important content in a logical manner with which it can easily be accessed for future reference. Chapter 3.2 shows a time line of the most important papers & information relating to CAI two-strokes, in-particular the development path taken by Honda from circa 1993-2010.

1993

- “Combustion Controller for a Spark Ignition Type Two-Cycle Engine” (Asai, et al., 1993)

Source: Honda Motor Co Ltd (Patent (US5697332A))

1994

- *“A Trial for Stabilizing Combustion in Two-Stroke Engines at Part Throttle Operation”* (Ishibashi, et al., 1994)

Source: Honda R&D Technical Review Vol.6 (1994)

1995

- *“Analysis on Fuel Economy Improvement and Exhaust Emission Reduction in a Two-Stroke Engine by Using an exhaust Valve”* (Asai, et al., 1995)

Source: SAE

1996

- “Improving the Exhaust Emissions of Two-Stroke Engines by Applying the Activated Radical Combustion (Asai & Ishibashi, 1996 International Congress & Exposition)

Source: SAE

- “Introduction of the Motorcycle “EXP-2” for Granada-Dakar Rally”

Source: Honda R&D Technical Review Vol.8 (1996)

- “Two-cycle internal combustion engine” (Asai, et al., 1996)

Source: Honda Motor Co Ltd Patent (EP0831214B1)

1997

- *“An Experimental Study of Stratified Scavenging Activated Radical Combustion Engine”* (Ishibashi, et al., 1997)

Source: SAE

1998

- *“A Low Pressure Pneumatic Direct Injection Two-Stroke Engine by Activated Radical Combustion Concept”* (Ishibashi & Asai, 1998 International Congress and Exposition)

Source: SAE

2000

- *“Basic Understanding of Activated Radical Combustion and Its Two-Stroke Engine Application and Benefits”* (Ishibashi, 2000)

Source: SAE

2004

- *“An Application Study of the Pneumatic Direct Injection Activated Radical Combustion Two-Stroke Engine to Scooter”* (Ishibashi & Sakuyama, 2004 Fuels & Lubricants Meeting & Exhibition)

Source: SAE

2007

- *“An Experimental Approach to the Controlled Auto-Ignition”* (Morikawa & Ishibashi, 2007 World Congress)

Source: SAE

- *“Enlargement of Auto-Ignition Regions by Applying a Stratified Charge Concept”* (Nishida, et al., 2007)

Source: JSAE

2009

- *“Improvement of Fuel Economy Using a New Concept of Two-Stroke Gasoline Engine Applying Stratified-Charge Auto-Ignition”* (Nishida, et al., 2009)

Source: SAE

2010

- *“A Further Approach to Controlled Auto-Ignition Using a Sequence of Low-Temperature Combustion-States”* (Morikawa, 2010)

Source: SAE

- *“A Macroscopic Understanding of the Controlled Auto-Ignition for Vehicle Engines”* (Ishibashi & Morikawa, 2010)

Source: SAE

3.3 – Summary

Having conducted an extensive literature review it is important to identify the key information that has been gained concerning CAI.

1. Modal Operation (Homogeneous Charge) (Ishibashi, et al., 1994)

A*-B*-C*-D*

A* – HCSI (Homogeneous Charge Spark Ignition), idle ↔ low load

There are low levels of “cold” residual gas, so the engine operates as a normal HCSI two-stroke engine.

B* – HCSAAI (Homogeneous Charge Spark Assisted Auto Ignition), low load

There are reduced levels of residual gas and heat capacity, the engine is able to operate in the auto-ignition region but with spark assistance.

C* – HCAI (Homogeneous Charge Auto-Ignition), low ↔ mid load

The engine operates as a CAI engine, this area of operation encompasses the ideal conditions for auto-ignition (plentiful amount of residual gas and heat capacity).

D* – HCSI (Homogeneous Charge Spark Ignition), mid ↔ high load

Levels of residual gas have drastically decreased, and the engine must operate as a normal HCSI two-stroke engine.

2. Modal Operation (Homogeneous / Stratified Charge) (Nishida, et al., 2007)

A*-B*-C*-D*-E*

A* – HCSI (Homogeneous Charge Spark Ignition), idle ↔ low load

There are low levels of “cold” residual gas, so the engine operates as a normal HCSI two-stroke engine.

B* – HCSAAI (Homogeneous Charge Spark Assisted Auto Ignition), idle ↔ low load

There are reduced levels of residual gas and heat capacity, the engine is able to operate in the auto-ignition region but with spark assistance.

C* – HCAI (Homogeneous Charge Auto-Ignition), low ↔ mid load

The engine operates as a CAI engine, this area of operation encompasses the ideal conditions for auto-ignition (plentiful amount of residual gas and heat capacity).

D* – SCAI (Stratified Charge Auto-Ignition), low ↔ mid load

The engine operates as a SCAI engine, this area of operation encompasses the ideal conditions for auto-ignition (plentiful amounts of residual gas and heat capacity). This engine is enabling SCAI by using an in-cylinder injector to target a residual gas hot spot, but (Nishida, et al., 2007) notes that by changing the injection timing it is possible to run as either SCAI or HCAI, so at discretion the engine can be switched between modes “D* C*”. It is interesting to add that whilst SCAI extends much closer to idle / low load it is also still able to operate in almost all the areas that HCAI does – this though is done with drastically increased cost and complexity.

E* – HCSI (Homogeneous Charge Spark Ignition), mid ↔ high load

Levels of residual gas have drastically decreased, and the engine must operate as a normal HCSI two-stroke engine.

3. Transition: Deceleration & Idle low load (Homogeneous & Stratified Charge) (Nishida, et al., 2007) (Ishibashi, et al., 1994)

The timing and possible onset of auto-ignition can be said to be very well controlled simply by the exhaust trapping valve, this is true for all but one situation. The situation surrounding

“deceleration” (shut throttle / low load / idle) has required the implementation of a further control device in the form of a bypass valve.

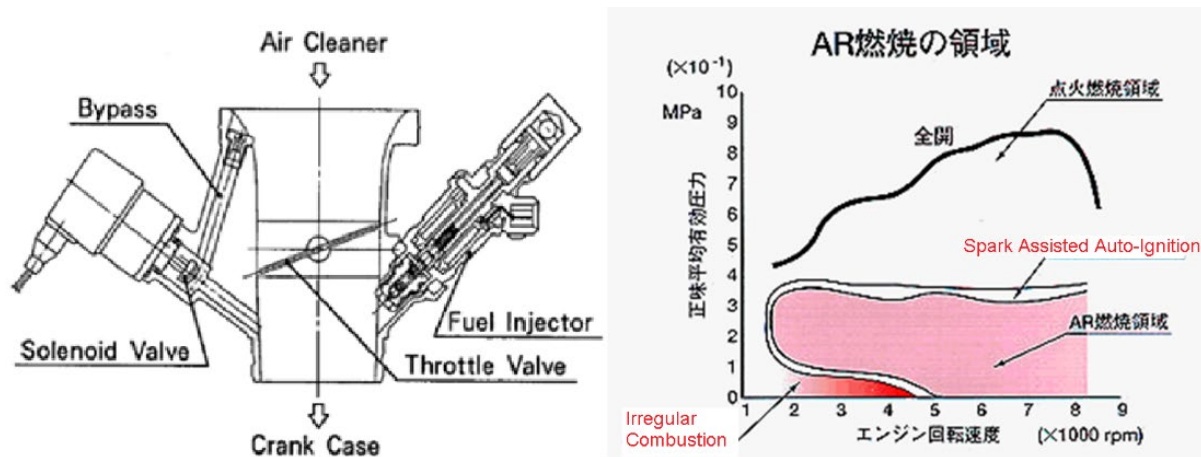


Figure 9 - CAI Deceleration Device & Region (Asai, et al., 1995)

Figure 9 shows a control device (air bypass valve) used by (Asai, et al., 1995) as well as showing a graph of the modular phases and area of irregular combustion. This area of irregular combustion can be described as an area where the HC SAAI becomes uncontrollable, or from another point of view it is the point at extreme low load / shut throttle where a lower load limit of irregular auto-ignition can operate past the delivery ratio limit imposed by the shut throttle. For example, HC SI may have an idle (throttle area) value of 7% whereas the lower load of irregular auto-ignition may be 5%. The bypass valve can in some ways be viewed as an on/off switch for auto-ignition, this is not to say though that the transition would be harsh or create a poor user experience, as noted by (Asai, et al., 1995) this device solves the problem of irregular combustion and poor transition. This can be explained in two ways, firstly when transitioning to this low load / shut throttle state the engine is coming from state of spark assisted auto-ignition and it is believed that because action of shutting the throttle may take a few hundred cycles, this is more than enough time for cyclic variation of the ignition / fuelling to react and create a smooth transition (hypothetically speaking). The air valve whilst described as an on/off switch eliminating the possibility of auto-ignition is not an on/off switch for absolute combustion, relative to the speed and precession of modal transition it is believed the other contributing factors of HC SAAI can shift their bias to controlling combustion completely. Interestingly

Figure 10 (Onishi, et al., 1979) shows the transition between auto-ignition and spark ignition, Onishi discusses how transition (in either direction) is a smooth process.

Regarding transition Honda state ; (Honda, 1997)

“In the new CRM 250 AR, a sharp engine response can be obtained by making use of its advanced AR combustion control system to detect and judge the throttle operation situation of the rider and instantaneously advance the ignition timing when the throttle is rapidly opened “ignition timing warp system” was newly adopted”

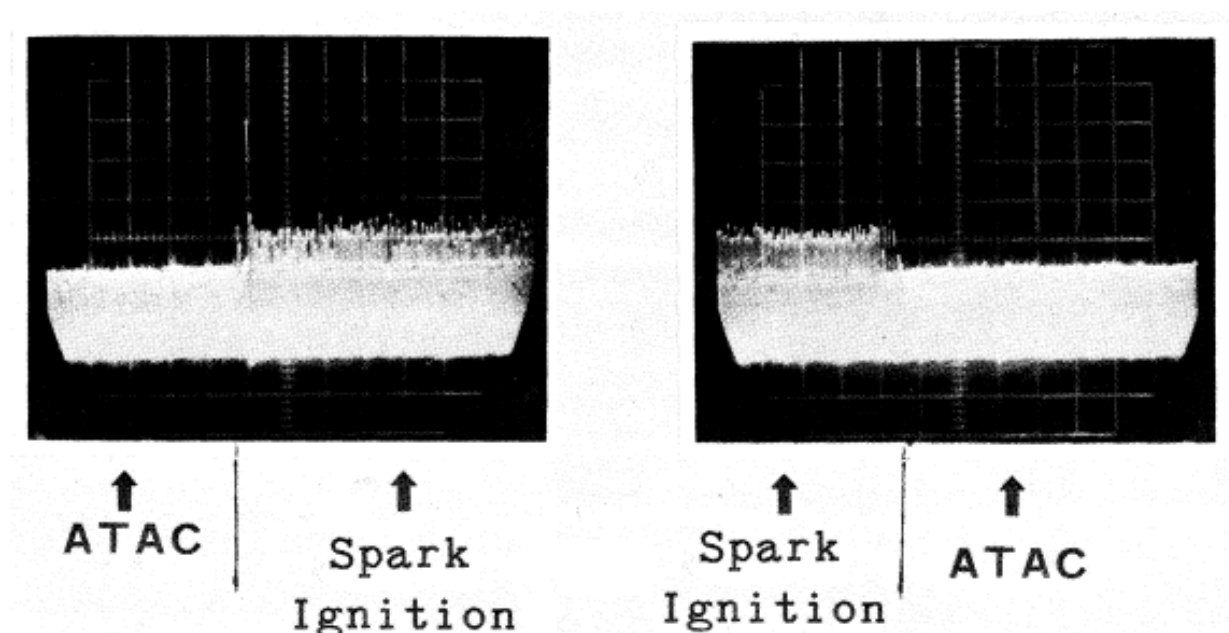


Figure 10 - CAI Transition (Onishi, et al., 1979)

4. AFR

The effect that AFR has on auto-ignition is one discussed by (Ishibashi, 2000) and shown in Figure 11 (Ishibashi, 2000), the effect of AFR was measured at three points using 12 as a rich example and 18 as a lean example and with 14.7 as a midpoint. It was found that the auto-ignition temperature peaked at 14.7 and a shift to either 12 or 18 produced a 25k drop and doing this retards the ignition by around 4 degrees.

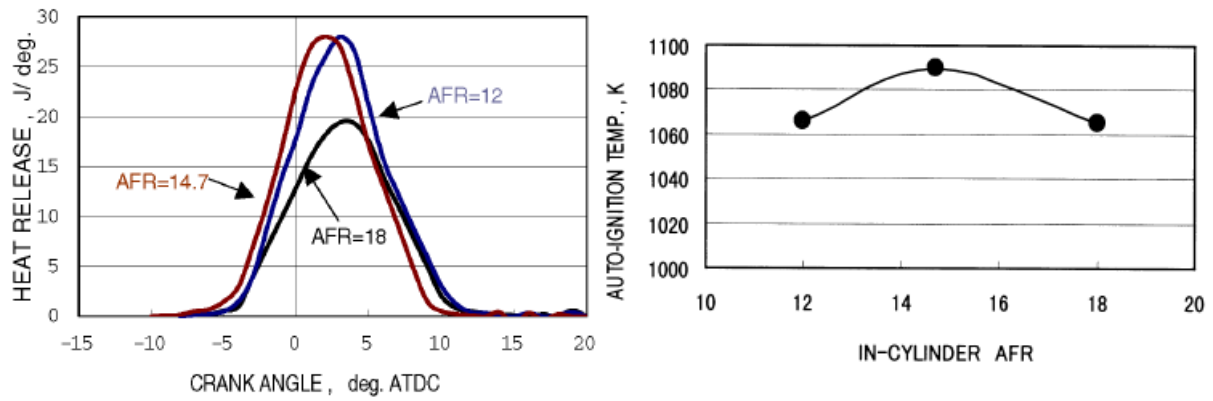


Figure 11 - CAI AFR Effect (Ishibashi, 2000)

5. CAI Temperature and Engine Speed

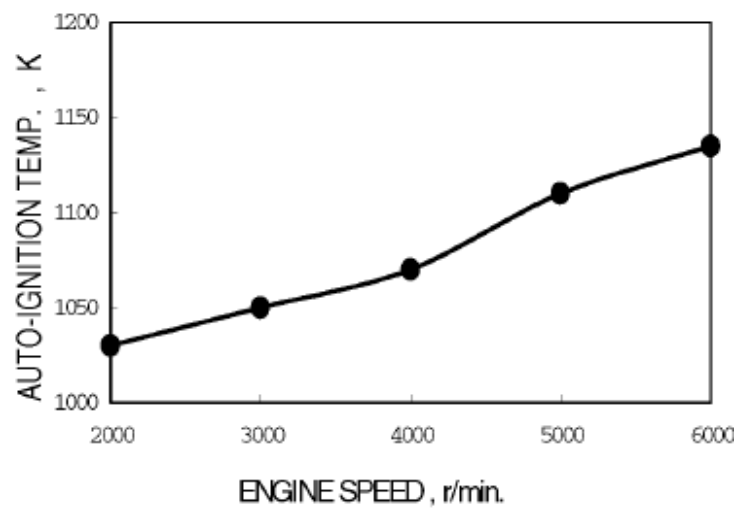


Figure 12 - Auto-Ignition Temperature (Ishibashi, 2000)

Another important point mentioned by (Ishibashi, 2000) and shown in Figure 12, is how the target temperature for auto-ignition changes with engine speed. It was found always to consistently increase with engine speed regardless of conditions imposed.

6. Exhaust Throttling

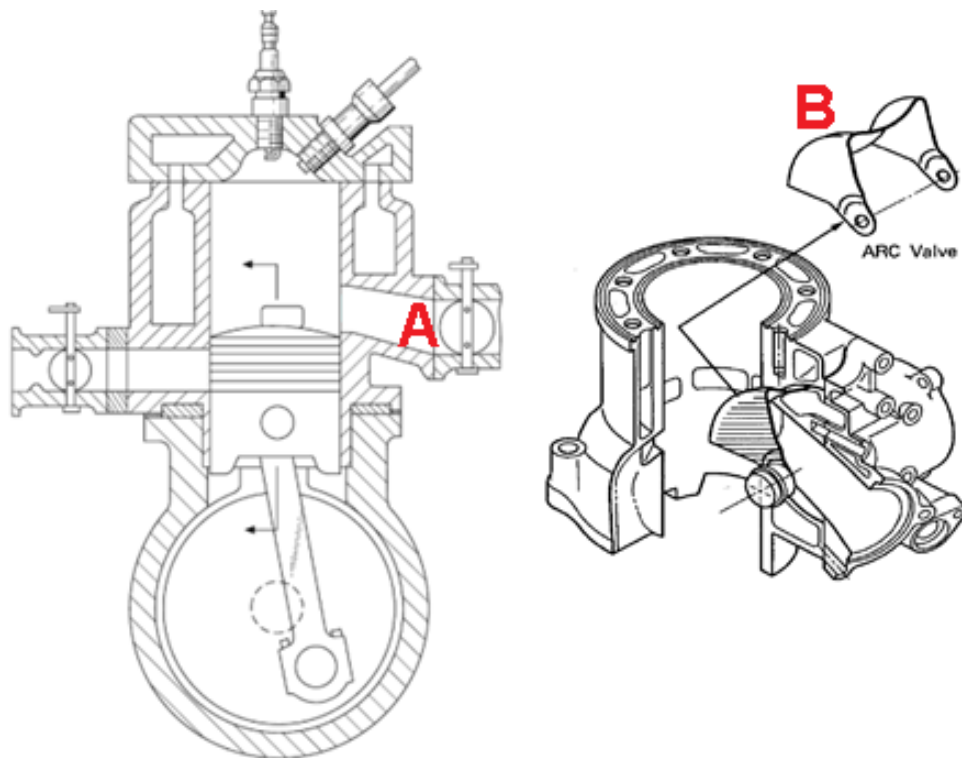


Figure 13 - Exhaust Butterfly Throttle (A) and ARC Valve (B) (Onishi, 1984) & (Ishibashi, 2000)

Figure 13 (Onishi, 1984) & (Ishibashi, 2000), shows two different ways in which the exhaust can be throttled. The first example “A” (Onishi 1984) shows a simple butterfly placed towards the end of the main cylinder exhaust passage, whereas “B” (Ishibashi, 2000) shows what is a much like a traditional exhaust valve that seals up close to the cylinder exhaust port. It can be said that if the main protagonist to auto-ignition is an abundance of residual gas then using a butterfly would certainly throttle the exhaust port in terms of area and achieve this. Residual gas whilst important is not the only factor to be considered, (Onishi, et al., 1979) mention the importance of in cylinder turbulence and that too much turbulence just prior to port closure can have a negative effect by dissipating the hot residual gas too evenly – this is important as the hot spots of gas will form ignition kernels (Ishibashi, et al., 1994).

The exhaust valve shown by (Ishibashi, 2000) not only reduces turbulence but increases the in-cylinder pressure at exhaust port closure. The close-fitting exhaust valve was used on Hondas CRM250AR (production trail bike) but on the Pantheon 125 AR a simple butterfly valve was used in the exhaust port. This indicates that the butterfly valve is a valid tool in prompting CAI, but what disadvantages could there be? Perhaps these disadvantages make little or no difference to a low powered commuter bike, but it would be interesting to investigate what if any effect these two different methods would have on a higher performance bike – the

system could potentially be implemented as an add on to a current engine that already has a conventional exhaust valve.

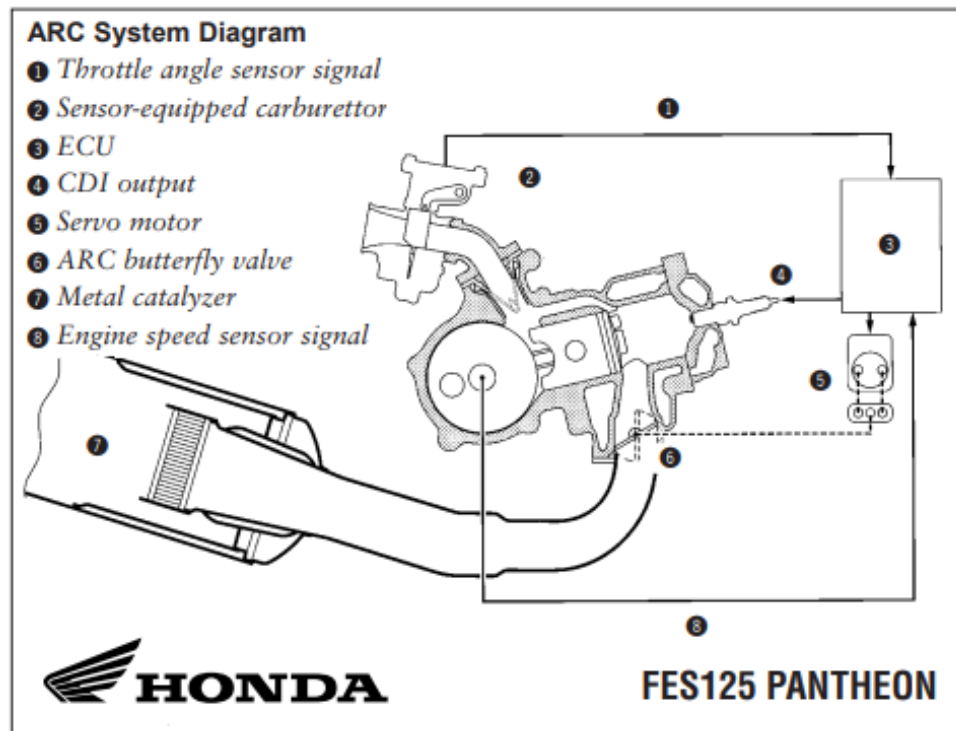


Figure 14 - Honda Pantheon 125 AR (Honda Motor Co Ltd)

7. Water Temperature

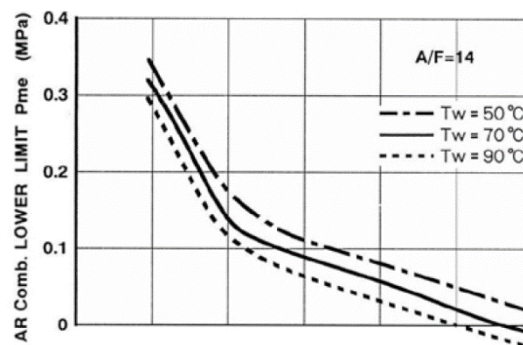


Figure 15 - Water Temperature Effect on CAI (Ishibashi, et al., 1994)

Figure 15 (Ishibashi, et al., 1994) highlights how the temperature of the coolant can have a significant effect on the lower limit of CAI, even just a difference of 20 degrees Celsius can shift the lower limit of CAI by as much as 1000rpm. This information points towards thermal management being another component in CAI system design.

8. Transfer Throttling

Transfer throttling works by heavily restricting the intake of fresh charge and at the same time this restriction of the fresh charge makes the in-cylinder conditions much less turbulent, further prompting residual gas hot spots. This method held back by packaging implications and that it has a negative impact on engine performance – Figure 16.

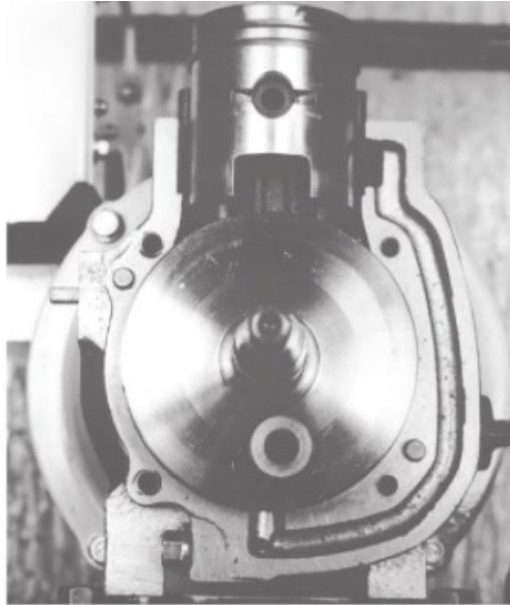


Figure 16 - NiCE Engine IFP (Zhao, 2007)

9. Combustion Chamber Shape and Squish

Controlled auto-ignition in terms of two-strokes and excess residual gas can in its initial propagation stage be considered a slow burn process that starts from circa 3 kernels / hot spots. Almost all the cylinder heads shown by Honda (Asai, et al., 1993) (Asai, et al., 1996) (Ishibashi, 2000) in their development work have an aggressive squish band and globular combustion chamber – signifying increased turbulence close to TDC and spatial concentration of the volume.

10. Direct Injection

Direct injection / charge & fuel separation is naturally the path that needs to be taken to overcome the inherent issues with the two-stroke engine, so naturally it is topic that has appeared when reviewing two-stroke CAI. Direct injection naturally complements CAI in terms of more precise and timed injection strategies – lowering emissions and giving another level of control to combustion timing, and even showing potential to extend CAI down to almost idle through SCAI (stratified charge auto-ignition) (Nishida, et al., 2009).

11. Compression Ratio

Often when reviewing CAI discussion turns towards the term HCCI where the dominant protagonist to combustion is compression ratio, but in terms of two-stroke CAI it can be stated that this is not the case. The two-stroke engine is naturally on the brink of auto-ignition when there is sufficient hot residual gas, so it can be said that in the case of two-strokes that residual gas is the protagonist, though this is not to say that pressure / compression is not important. Most engines discussed in development papers have a trapped compression ratio of around 6.5:1, but it isn't that pressure doesn't play a part in the process. In terms of increasing outright compression ratio, it can be increased but it only changes the timing of auto-ignition, it isn't until it is dangerously high that it has a more dominant effect – research by Honda confirms this (Ishibashi, et al., 1994).

12. CAI Process Flow Chart

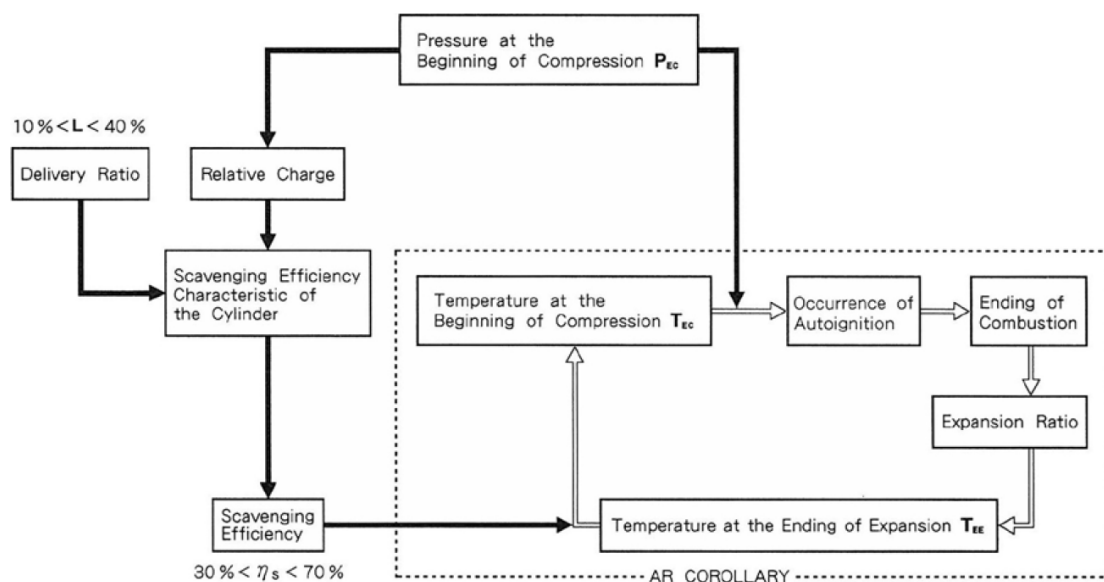


Figure 17 - CAI Process Flow Chart (Asai, et al., 1996)

Figure 17 show a CAI process flow chart produced by (Honda, 1997)

Figure 17 displays the basic layout (theory) of Hondas CAI system.

Chapter 4 – Geometric Investigation

4.1 – Introduction

When reviewing research papers there is often a lack descriptive information on the physical specification. Because Honda had based their CAI 250cc trail bike directly on its longstanding SI predecessor (both circa 40bhp – stated), it was decided it would be a valuable exercise to inspect these parts and see how / if they differed and where possible compare them with the parts supplied by KTM.

4.2 – Methods

Visual:

The first step in any process like this is just to pick the items up and get a feel for them, this step often ignored but when conceptualising or reviewing objects in a digital environment it is a vital first step – at a minimum this process gives a good sense of perspective.

Computed Tomography (CT Scanning)

Global tolerance: >0.2mm

CT scanning was used as many of the parts such as cylinders contain internal cavities which are not line of sight. This can be resolved by cutting the part into multiple sections, though with this method accuracy start to degrade heavily and is simply not an option on complex parts. The machine used was a YXLON CT Compact 450KV scanner, rotation time was circa 45 seconds per slice and each slice was 0.5mm – this is generally the limit when inspecting parts for reverse engineering, if more detail is sort the gains are minimal but the time taken, and file size start to increase dramatically (as does cost). Figure 18 shows the KTM cylinder being setup for scanning, note that there are wooden chucks present as the machine needs an initial line of sight.

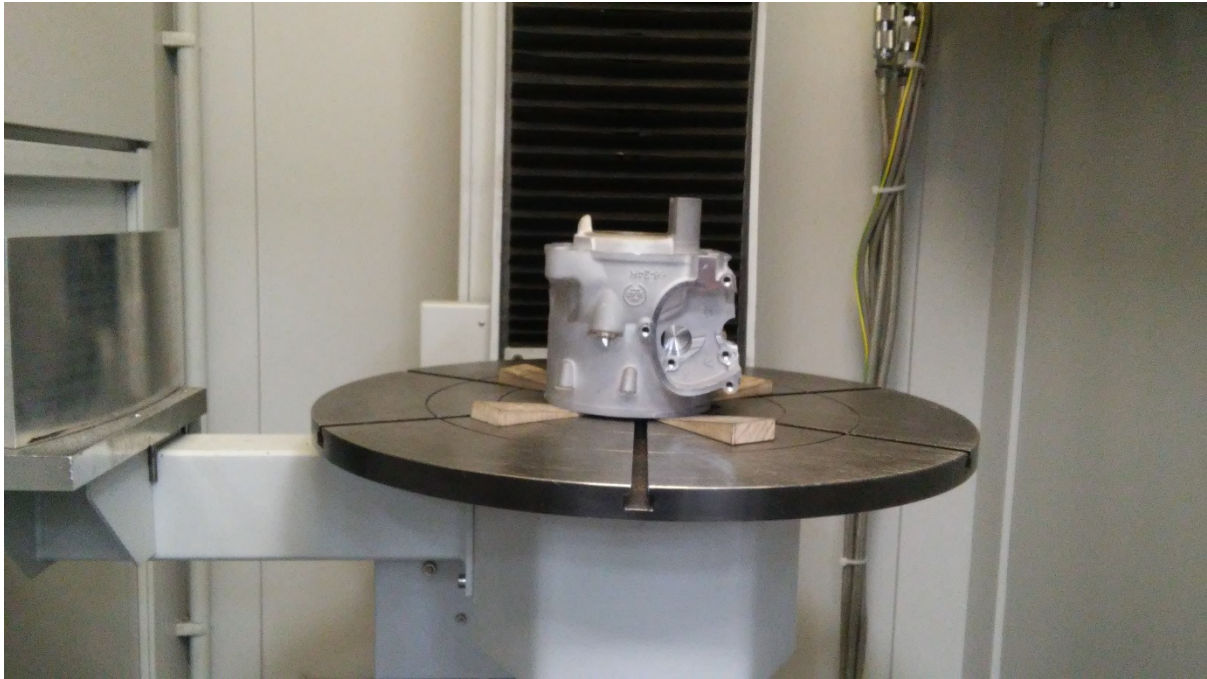


Figure 18 - CT Scan KTM Cylinder (Vacas, Authors Own)

Structured Light Scanning:

Global tolerance: >0.05mm

Two of the exhaust valves were scanned using a GOM white light scanner.

Laser Scanning:

Global tolerance: >0.8mm

Cylinder head “3” was scanned using a desktop laser scanner, this scan was used as the data was already available from the author.

4.3 – Comparison

4.3.1 – Ports

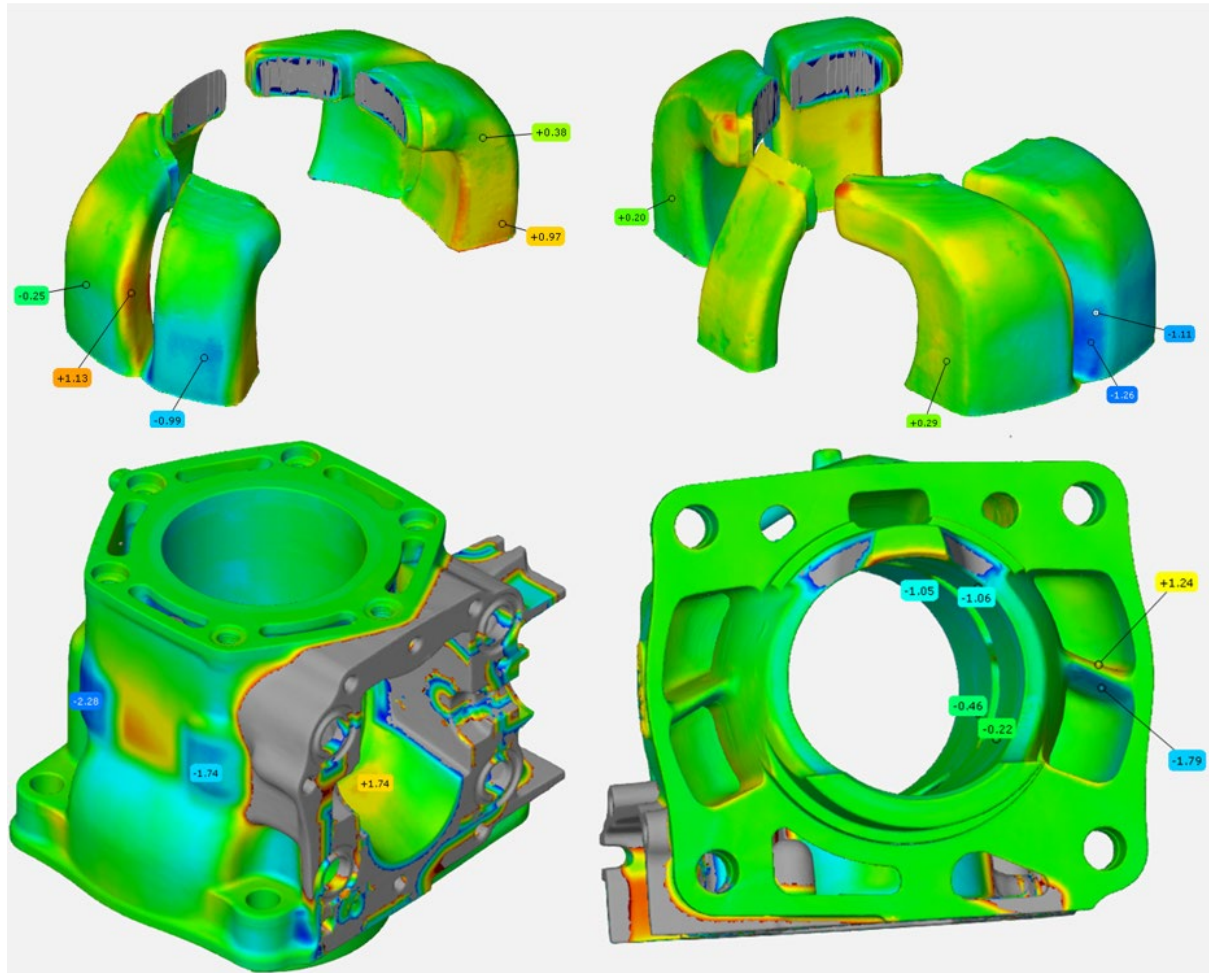


Figure 19 - Port Comparison & Cylinder Honda 250's (Vacas, Authors Own)

Having scanned both Honda cylinders Figure 19 (Vacas, Authors Own) it was obvious that there was next to no difference in the porting arrangements / specifications, with the only visibly different features being the cylinder front face where the CAI bike had a removable cover and a slight wall changes around studs seating areas. The two scans and core extractions were compared in GOM Inspect with a best fit alignment, the worst deviation was minimal and nothing that couldn't be expected from miss-alignment (twist) or simply different tooling iterations / wear.

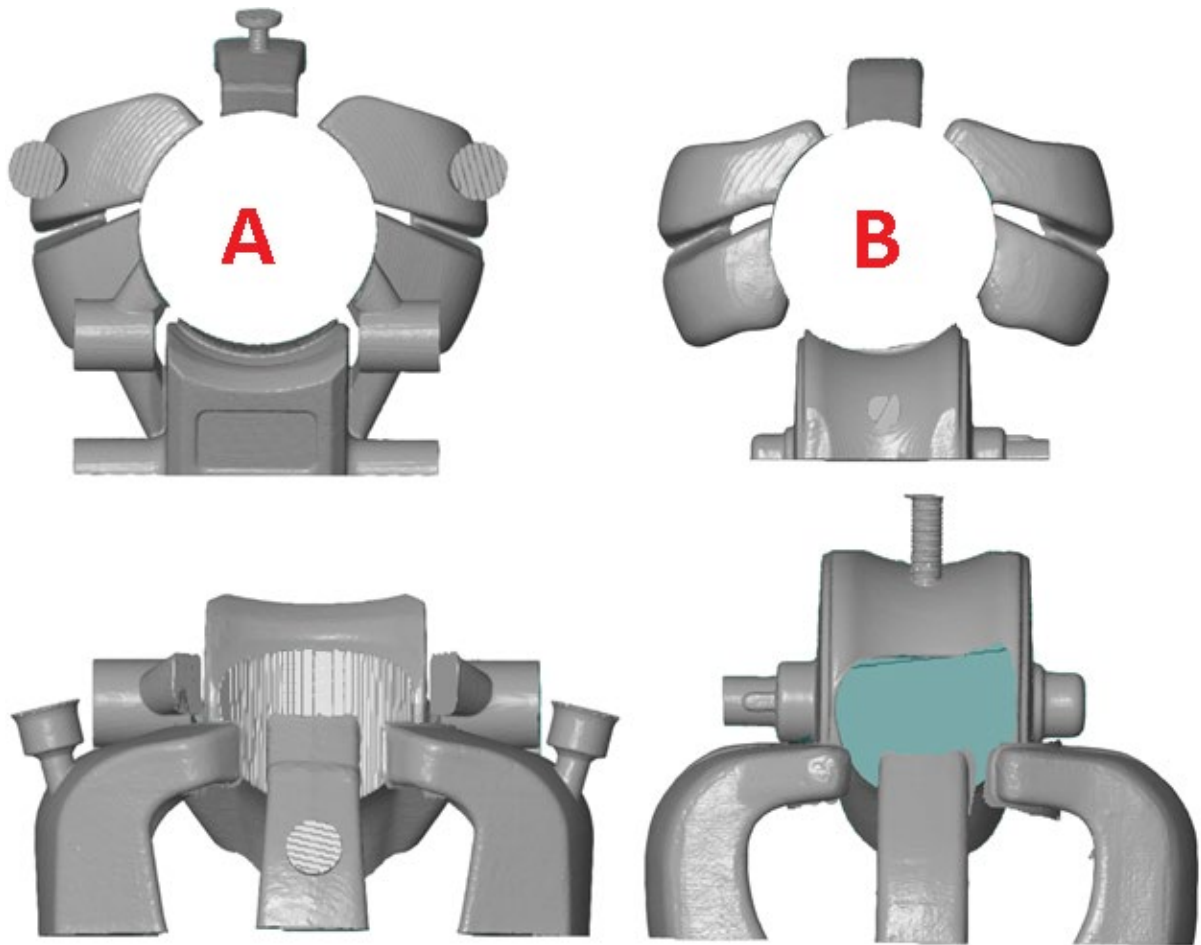


Figure 20 - KTM & Honda Porting Arrangements (Vacas, Authors Own)

Figure 20 shows the comparison between the KTM (A) and Honda CAI (B) port arrangement, although the two images are not quite to scale it is still very clear to see the structure between the two porting arrangements. The KTM porting setup appears to be much less restricted than the Honda setup, whilst it could be discussed that this porting setup is aimed at higher power and performance it is much more important to note that this setup could potentially create a more turbulent scavenging environment. It has been often discussed how low turbulence is a protagonist for two-stroke CAI widely (Zhao, 2007), (Lavy, et al., 2001), (Onishi, et al., 1979) and (Ishibashi, et al., 1994) have all discussed this aspect of two-stroke CAI – though without a practical test it would be naive to comment further.

4.3.2 – Cylinder Heads

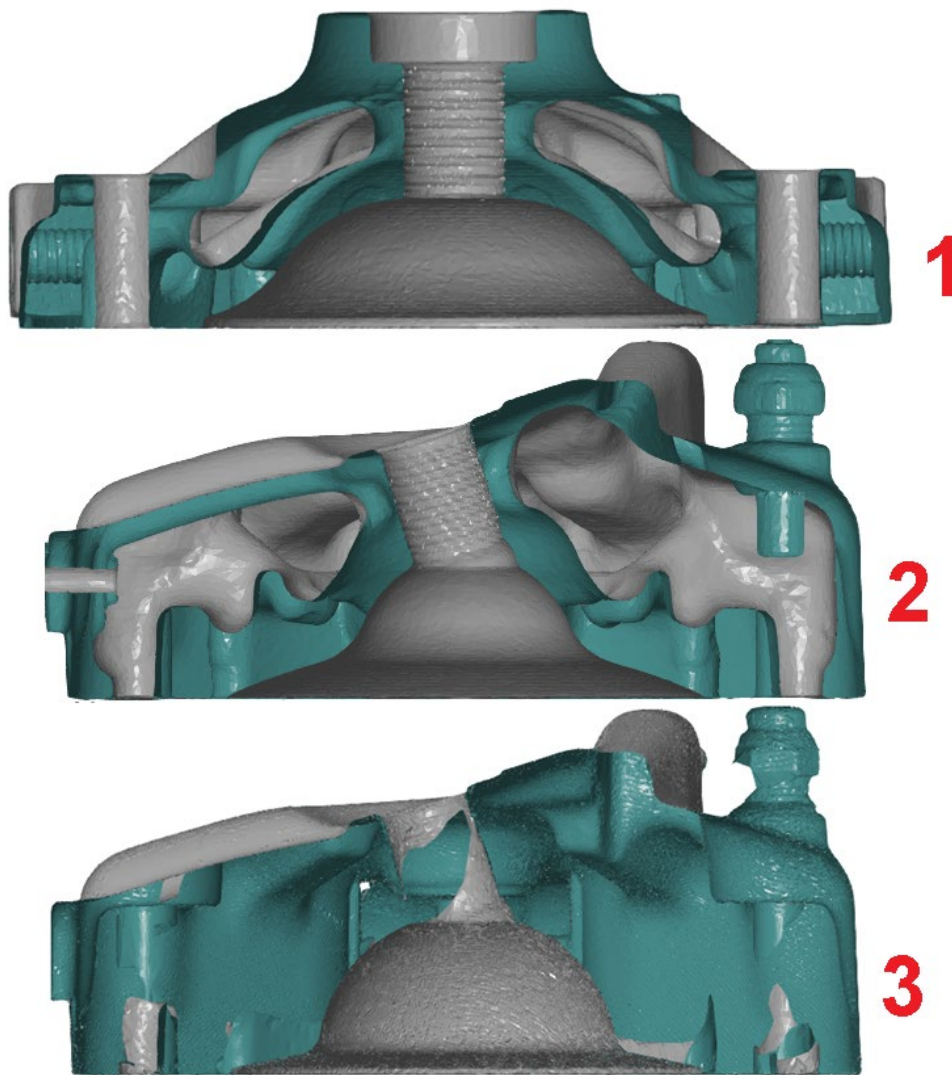


Figure 21 - Cylinder Head Comparison (Vacas, Authors Own)

Cylinder Head 1 & 2: (KTM & Honda 250 SI)

It can be said that cylinder head 1 relates somewhat closely to what (Blair, 1994) describes as a “Bathtub” combustion chamber, though it differs in that there is almost no squish band with such a large blended radius. Cylinder 2 on the other hand appears to use a semi hemispherical chamber and has what may be considered a large (in terms of area ratio) squish band. It may appear at first that the KTM and Honda approach is quite different, but on consideration of discussion by (Blair, 1994) it may not be so, it seems that both KTM and Honda have taken a low squish / slow combustion approach. Whilst high squish velocities can lead to rapid burning characteristics, they can also cause vibration, noise and rapid pressure rises (not good for smooth operation) and if the burning is too quick and too soon it can lead

to high rates of NO_x formation, not to mention slow inefficient burning in the later stages of combustion.

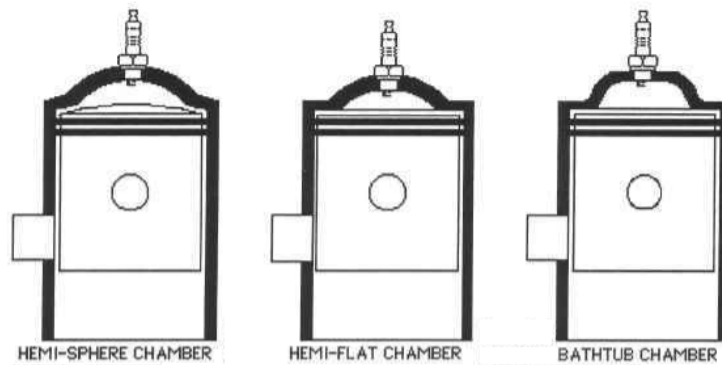


Figure 22 - Combustion Chamber Designs (Blair, 1994)

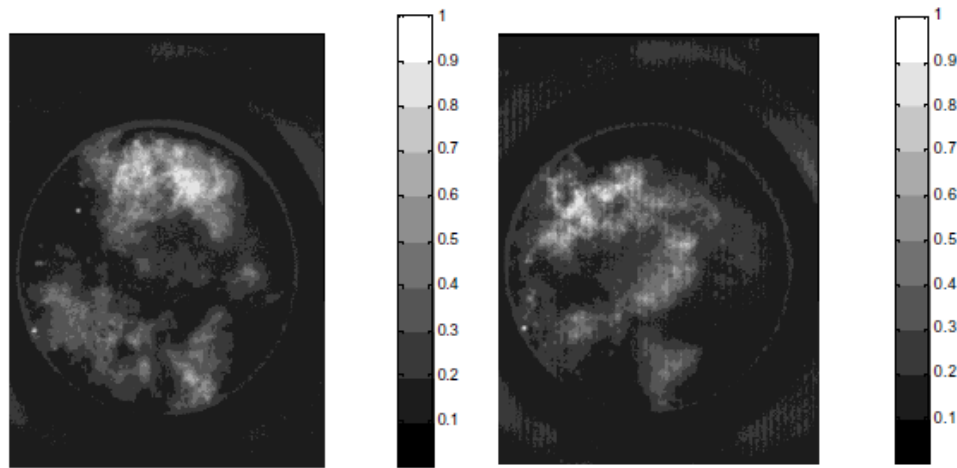
Cylinder Head 3: (Honda 250 CAI)

The Honda CAI cylinder seems to take a different approach using what can be described as a “Hemi-Sphere” chamber Figure 22 (Blair, 1994) and it is interesting to note how the hemi-sphere is close to being an equidistant sphere (if a central point on the piston crown is assumed then the distance to any extremity is almost the same regardless of direction). It is important here to consider how the squish velocity is affecting the speed of combustion and how this is aiding auto-ignition. (Asai, et al., 1993), (Ishibashi, 2000) and (Nishida, et al., 2005) have all discussed how the onset of auto-ignition is influenced by turbulence and how the mixing of fresh charge and residual gas has a negative effect due to it diluting hot spots. Low turbulence is initially sought so that patches of hot residual gases stay concentrated (eventually forming ignition kernels), but when getting close to TDC combustion may have started in many different places but overall is actually taking place rather slowly.

(Ishibashi, et al., 1997) and (Lavy, et al., 2001) Figures 23 & 24 show how auto-ignition is often localised and starts from patches or kernels “K*” in the same area of the combustion chamber, so it could be hypothesised that an equidistant (or close to) central chamber and higher squish velocity may help concentrate and agitate the gases close to TDC, and thus speed up auto-ignition from its initial segmented slow burn.

The Honda press pack states;

“In the current model, slow combustion has been favourable for good drivability during off-road driving, whereas in AR combustion, rapid combustion is required from its characteristics.”



4100 rpm, instantaneous direct visualization
images, 19 °CA BTDC

Figure 23 - CAI Photo Images (Lavy, et al., 2001)

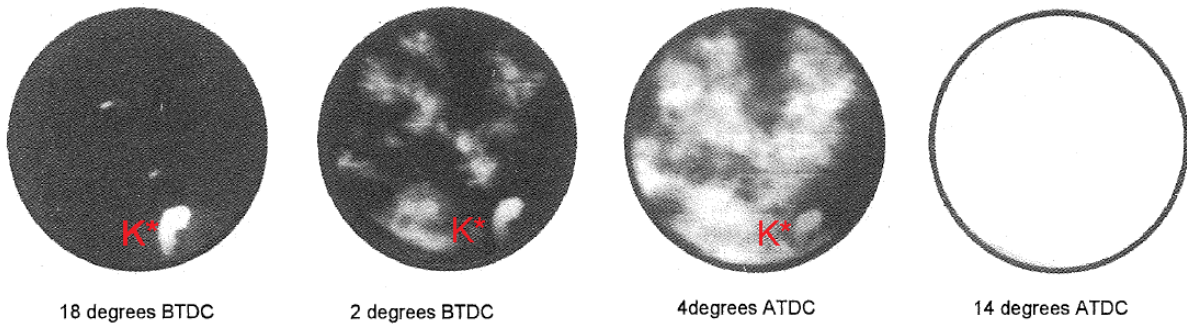


Figure 24 - CAI Photo Images (Ishibashi, et al., 1997)

Easy as it is to speculate (Blair, 1994) reiterates how often the correct combustion chamber is found through theory and practise, a process that he can be further advanced using CFD – which if relevant in 1994 can only be ever more relevant now. Whilst this research may give some indication of design direction, it is by no means a definitive answer.

4.3.3 – Exhaust Valves

The exhaust valve is an important component in the implementation of CAI in a high performance two-stroke engine and whilst the concept of transfer port throttling has been explored, (Zhao, 2007) discusses how it is highly effective for inducing CAI but that it is seriously limiting to the engines performance.

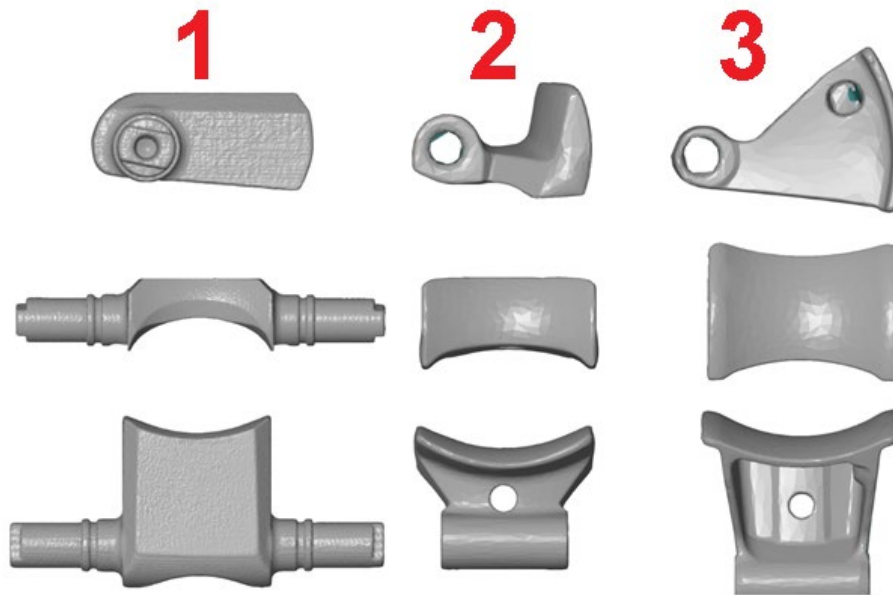


Figure 25 - Exhaust Valve Comparison (Vacas, Authors Own)

The difference in exhaust valve design Figure 25 shows how the KTM valve “1” and the Honda 250(SI) valve “2” have moderate size sealing faces., quite extremely so in compared to valve “3” of the Honda 250 CAI engine – it is obvious just how extreme the exhaust throttling needs to be for CAI.

Honda State (Honda, 1997);

“As a specific requirement, first of all it was to close the exhaust port to about one-third for AR combustion control. Therefore, the valve width is made wider than the exhaust port width, and the operation range is extended from the exhaust port upper surface to the scavenging port upper surface to a large extent”

Note: Both Honda valves (“2” & “3”) have a hole in them, according to the Honda press release this serves 2 purposes with the first simply being a means to release pressure when the valve is being actuated. The second function is a potential performance enhancement, the trapped chamber behind the valves acts to influence the exhaust pulses with the aim of improving low end power – essentially lengthening the pulses so that they force / keep more mixture in the cylinder when at low revs the engine is not in its optimum operating area. Figure 26 (Motorcity, 2010), shows what is known as an “ATAC*” valve, this commercial acronym stands for “Active Torque Amplification Chamber” and is not to be confused with the term “ATAC” that refers to the CAI process “Active Thermo-Atmosphere Combustion”.

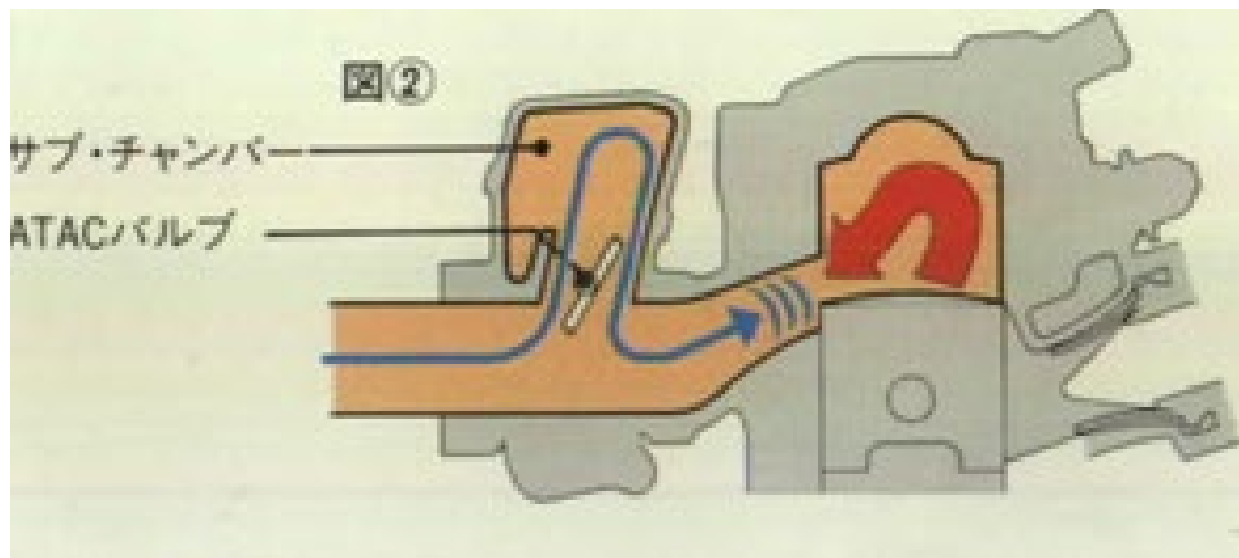


Figure 26 – ATAC* Valve (Motorcity, 2010)

Chapter 5 – Excel Model

This excel model was created with the clear aim of following fundamental steps to analyse the supplied engine data and subsequently the combustion process. Identifying elements such as the start of combustion (SOC), end of combustion (EOC) and the polytropic index “n” was done by combining the application of fundamental convention and visual appraisal. Much of the work conducted in this chapter was iterative (but manual) in its nature and required the constant review of processes and steps, so it was essential to follow a simple but robust structure – meaning that fault finding, and process changes could swiftly be implemented.

5.1 – Structure & Layout

Structure and layout was kept as simple as possible and can be summarised by the following points;

- Simple work flow logic dictated that calculations a horizontal linear pattern
- Colour use was kept to a minimum and only when necessary, low load headings were always “blue” and high load always “red”

5.2 – Engine Basics

Prior to starting this exercise, it is important to make clear some basic supplementary information.

Engine: KTM EXC 250 TPI	
Engine Type	Two-Stroke Case Reed Valve
Geometric Compression Ratio	11.6:1
Capacity (cm3)	248.9
Bore X Stroke (mm)	66.34 X 72
Rod Length (mm)	125
Scavenging Port Opens (degrees)	121 ATDC
Exhaust Port Opens (degrees)	86 to 106 ATDC

Combustion System	Spark Ignition
Fuel System	Transfer Port Injection
Scavenging Type	Schnürle

Abbreviation	Meaning
TDC	Top Dead Centre
BDC	Bottom Dead Centre
BTDC	Before Top Dead Centre
ATDC	After Top Dead Centre
TPO	Transfer Ports Open
TPC	Transfer Ports Closed
EPO	Exhaust Ports Open
EPC	Exhaust Ports Closed
BD	Blowdown
SOC	Start of Combustion
EOC	End of Combustion
20%	Low Load Engine Data
100%	High Load Engine Data
MFB	Mass Fraction Burned

5.3 – Displacement & Volume

The very first step when beginning the excel model was to calculate piston displacement and volume relative to crank angle, shown in Figure 27 is the initial crank slider diagram from which the equation was derived.

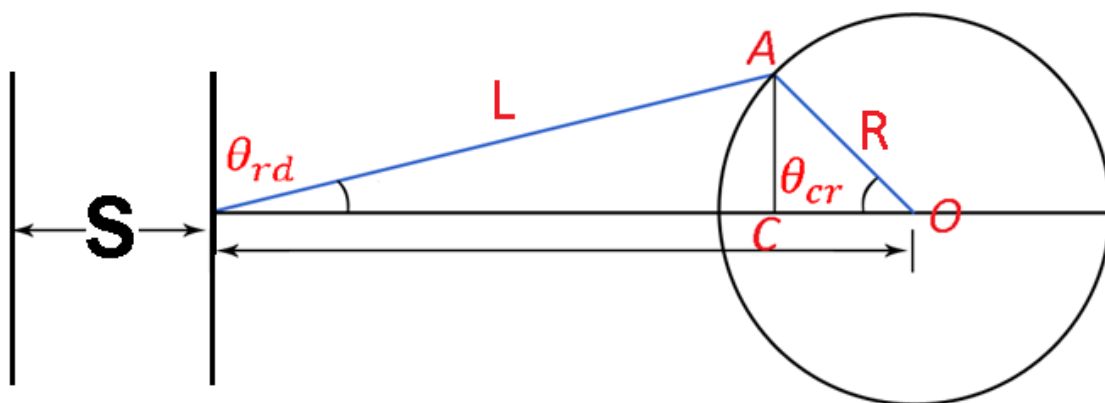


Figure 27 - Crank Slider Diagram (Vacas, Authors Own)

From Figure 27 it can be stated that:

$$S = (R + L) - (R \cos \theta_{cr}) - (L \cos \theta_{rod})$$

Which becomes,

Equation 1 - Piston Displacement

$$S = ((R + L) - (R \cos \theta_{cr})) - (L(0.5(\frac{R^2}{L^2})(\sin^2 \theta_{cr})))$$

The cylinder volume at any crank angle can be described as,

$$V_C = \text{Combustion Chamber Volume} = 23.5cc \text{ (Measured)}$$

$$V_{SV\theta} = \text{Swept Volume Relative to Crank Angle} = S((0.5\phi_{Bore})(\pi^2))$$

Therefore,

Equation 2 - Cylinder Volume

$$V_{CYL} = \text{Cylinder Volume} = V_C + V_{SV\theta}$$

5.4 – Pressure

5.4.1 – Pressure Graphs

The pressure data supplied by KTM came as units of bar, the first step was to convert this to Pascals.

Stated simply,

Equation 3 - Pressure Conversion bar to Pa

$$P_{Pa} = P_{bar} * 10^5$$

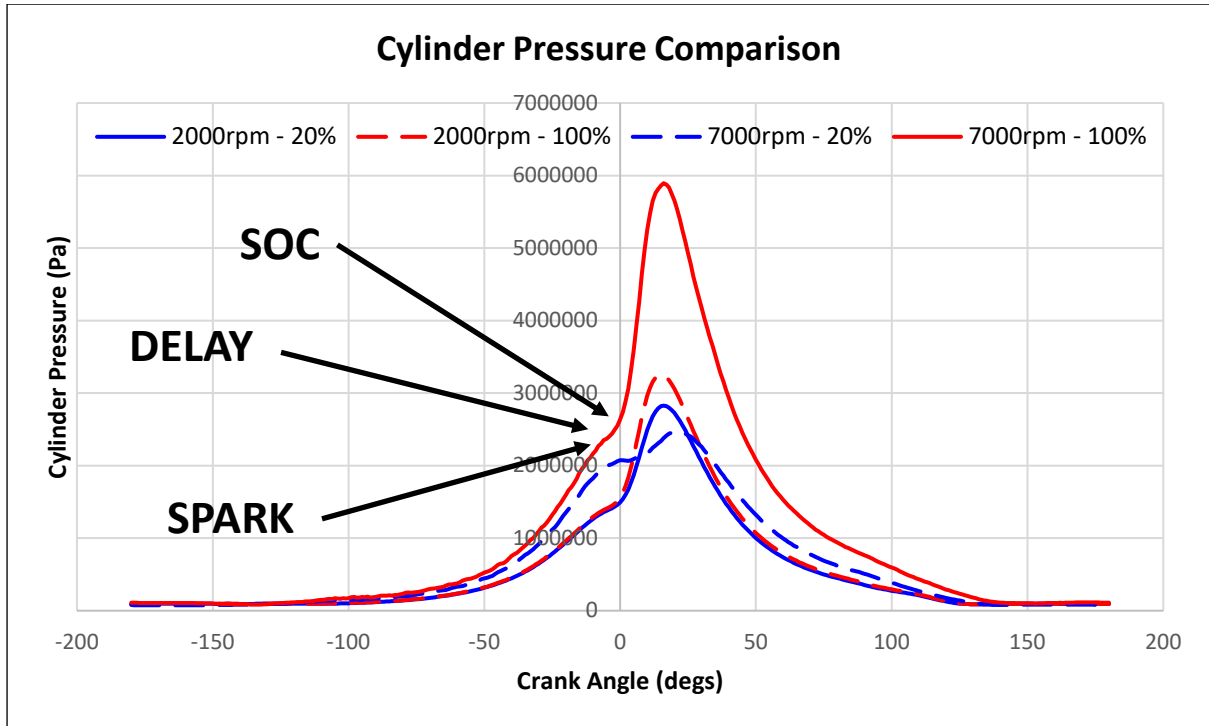


Figure 28 - Cylinder Pressure Comparison (Vacas, Authors Own)

Figure 28 shows a comparison of cylinder pressure data, initially the SOC, delay and EOC are visually identified as a matter of principle. This graph is importantly showing fundamental differences between high and low load, the graph clearly shows that with high load there is a large increase cylinder pressure between 2000rpm & 7000rpm. When looking at this range for 20% throttle peak pressure is lower at 7000rpm and it is obvious that the engine is being choked, even before the spark cylinder pressure is trending slightly lower. Interestingly at 2000rpm cylinder pressure follows the same trend up to ignition, with high load only peaking about 10% higher.

This apparent lack of difference in peak cylinder pressure at low rpm can be explained because at these low rpm points volumetric efficiency is poor no matter the throttle position.

Volumetric Efficiency (Blair, 1994) can be stated simply as;

Equation 4 - Volumetric Efficiency (Blair, 1994)

$$\eta_v = \left(\frac{m_a}{V_{SV} \rho_{at}} \right)$$

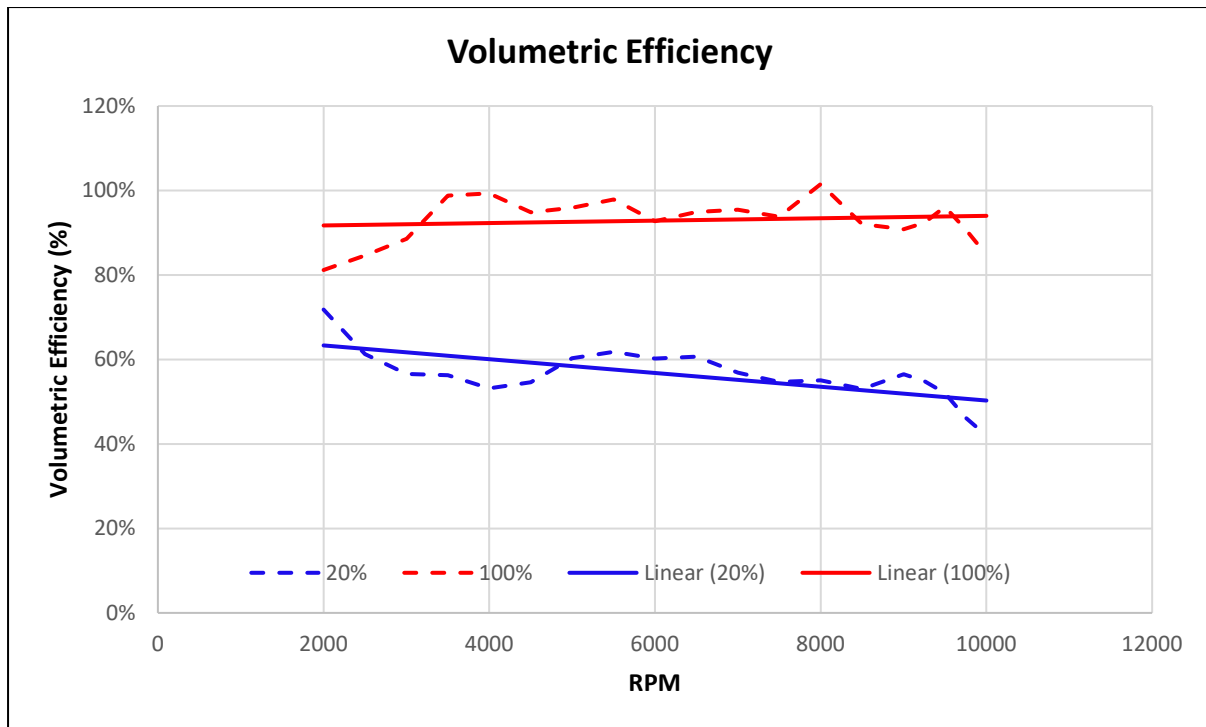


Figure 29 - Volumetric Efficiency (Vacas, Authors Own)

Figure 29 shows the volumetric efficiency for both low and high load data sets, this graph clearly shows how at 2000rpm low and high load are not separated by much, but from 2000rpm to 4000rpm the two data sets move in opposite directions with high load reaching around 100% efficiency and continuing with a neutral trend. Low load though drops to around 50% at 4000rpm and continues a slightly negative trend from there onwards, but it is important to note that both low and high load begin to drop off after 8000rpm. Poor volumetric efficiency is a contributing factor to high residual gas levels which is taken advantage of with CAI, but in a SI engine it will be necessary to start protecting low load / high rpm situations from the risk of detonation – though in reality it is unlikely the user would be operating in these more volatile regions as noted by (Onishi, et al., 1979).

5.4.2 – PV Diagrams

The next logical step in evaluating the engine data was the plotting of PV diagrams, this was done to compare the data against theoretical ideals

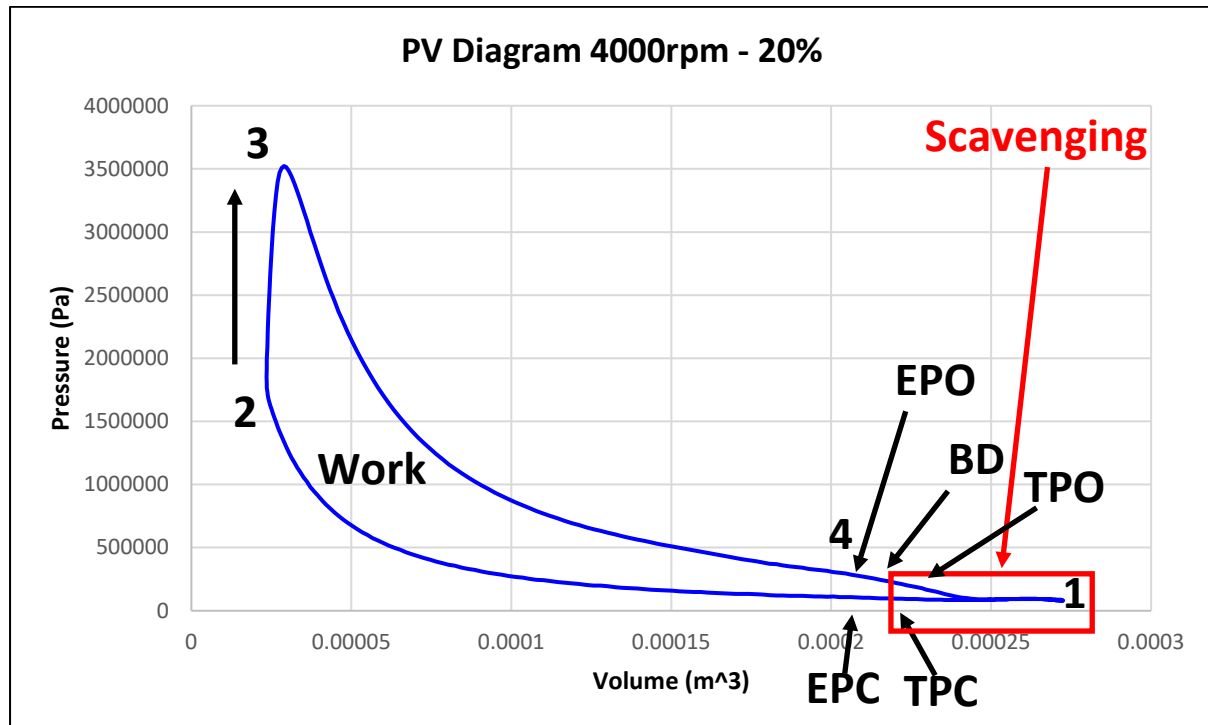


Figure 30 - PV Diagram 4000rpm 20% (Vacas, Authors Own)

The ideal cycle is shown in Figure 36 shows the ideal cycles as described by (Blair, 1994).

The diagram shown in Figure 30 was chosen as it displays a form with similarity to the ideal Otto cycle, the steps can be summarised below;

- 1→2: This is the compression stroke and can be considered both isentropic (reversible & no heat loss)
- 2→3; This is the combustion process and is considered a constant volume process with heat addition.
- 3→4; This is the isentropic expansion process.
- 4→1; This is a constant volume heat rejection process

There was not a lot of analysis done on the PV diagrams but one trend that was noticed was the continuing trend of low vs high load. Figure 31 shows a PV diagram for 2000rpm comparing low and high load, it is quite apparent that that this rpm point there is very little difference in the two curves with only a circa 10% difference in peak pressure.

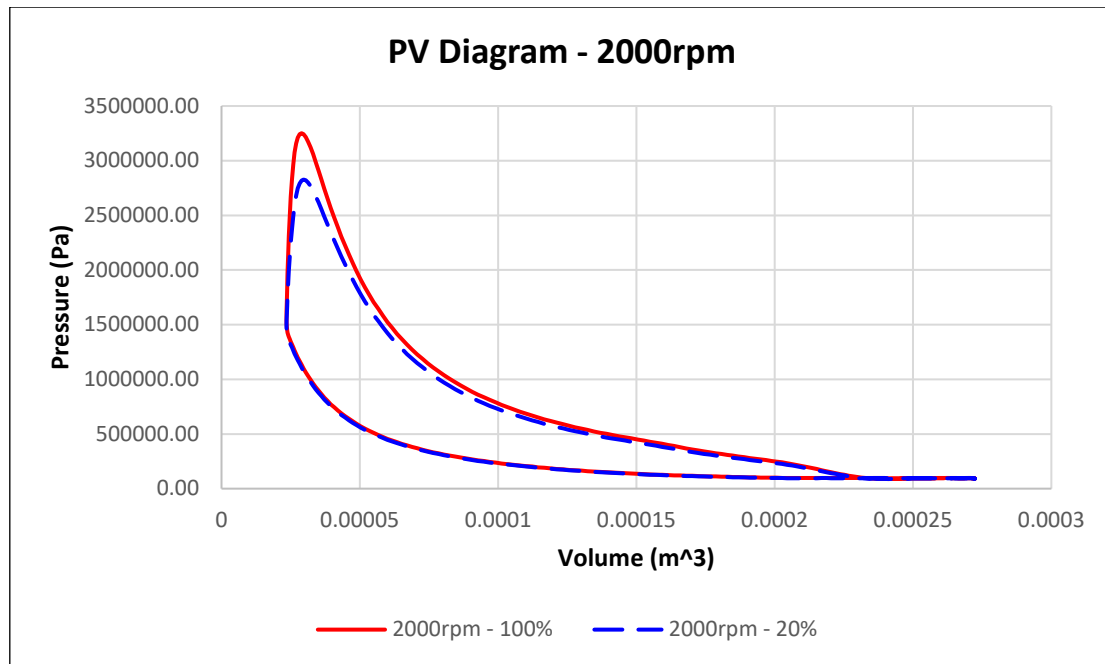


Figure 31 - PV Diagram 2000rpm (Vacas, Authors Own)

Although Figure 31 shows little difference between loads, Figure 32 shows as expected that at high rpm there is a significant difference in the two curves. The difference in peak pressure is obvious, but less obvious is the deviation from the constant volume process during combustion, this is showing the ignition being advanced to account for the inefficient combustion process. The ignition timing shows this, “-8.5” (low) vs “-5.4” (high) but the SOC is “2” vs “-1” revealing that even though the spark timing is earlier for low load it doesn’t start combustion till after high load

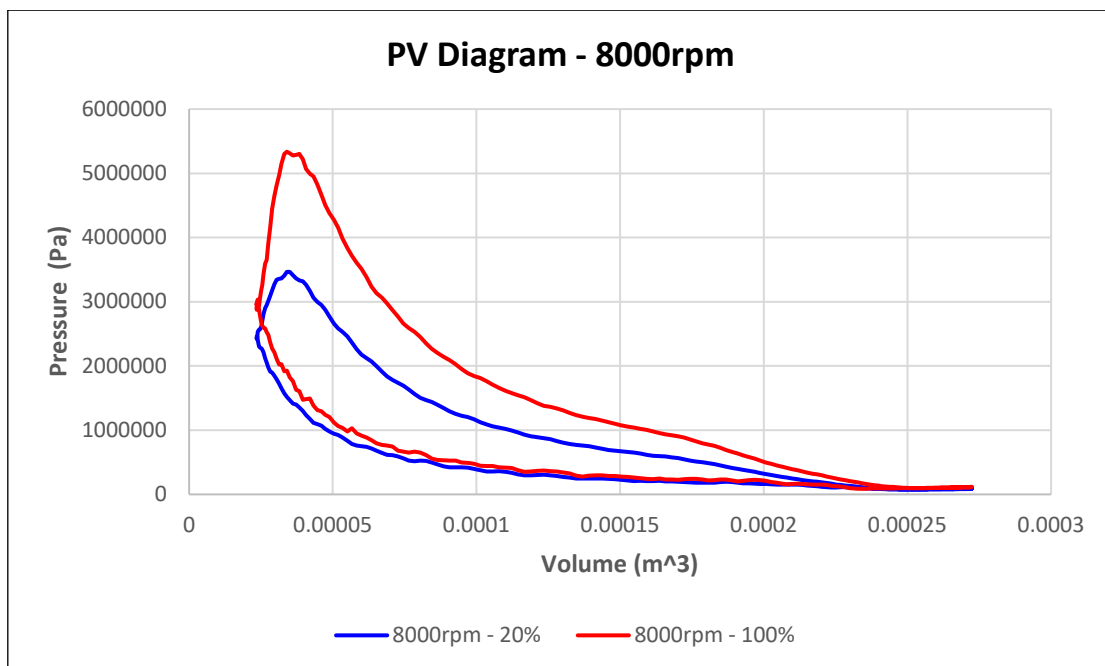


Figure 32 - PV Diagram 8000rpm (Vacas, Authors Own)

5.5 – Work & IMEP

Work is traditionally described as force multiplied by distance

$$W = Fdx$$

Where in an engine the force is cylinder pressure multiplied by the bore area,

$$W = PA dx$$

This can be expressed as,

$$W = PdV$$

When calculating per degree this gives the following equation,

Equation 5 - Work Per Degree Step

$$W_{\theta} = \left(\frac{P_1 + P_2}{2} \right) (V_2 - V_1)$$

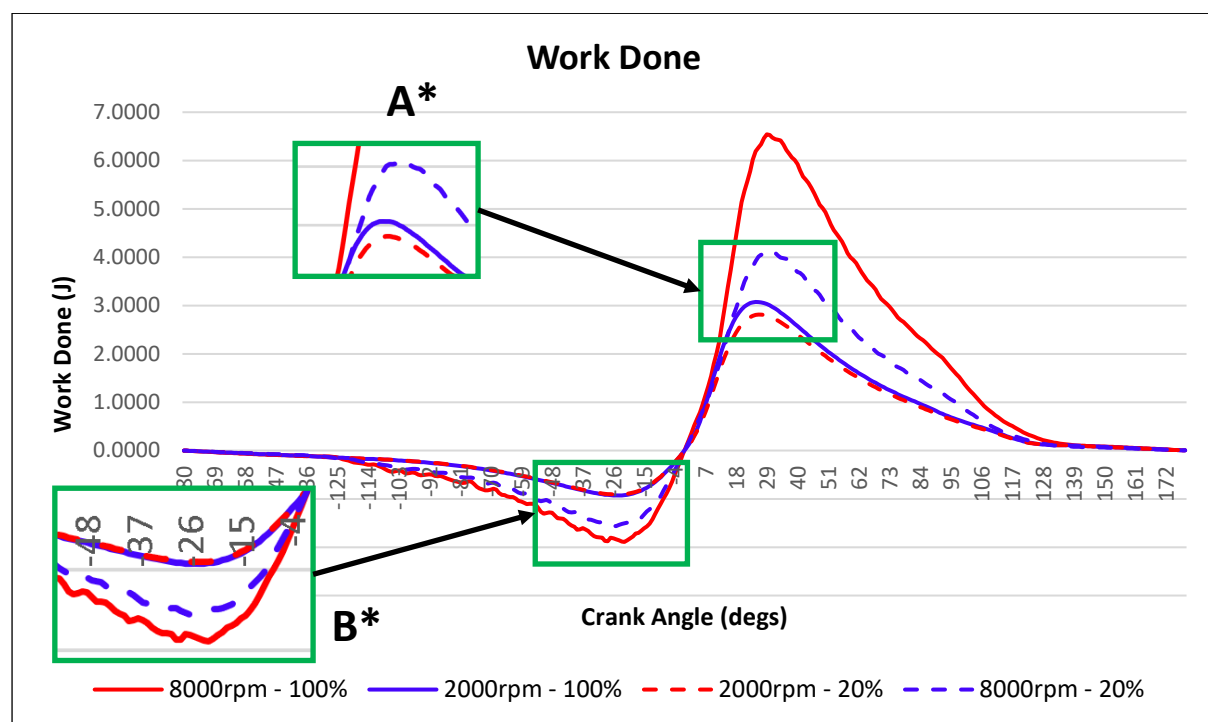


Figure 33 - Work Done (Vacas, Authors Own)

Figure 33 shows a comparison of work done at both low / high rpm and low / high load, the first observation is the area “B*” that shows negative work - with this being a product of inertia and there not being much difference between low / high load respective of rpm. This basic analysis shows the difference between low / high load, at low rpm there is little difference but at high rpm there is a significant difference

Having calculated the work done per degree, the next step was to calculate the IMEP (indicated mean effective pressure), the IMEP is a simple expression to describe an engines output without considering work lost to friction.

IMEP can be calculated by the following equation,

Equation 6 – IMEP (B.Heywood, 1988)

$$\left(\frac{\sum_{-180\theta}^{+180\theta} W_{\theta}}{V_{SV}} \right)$$

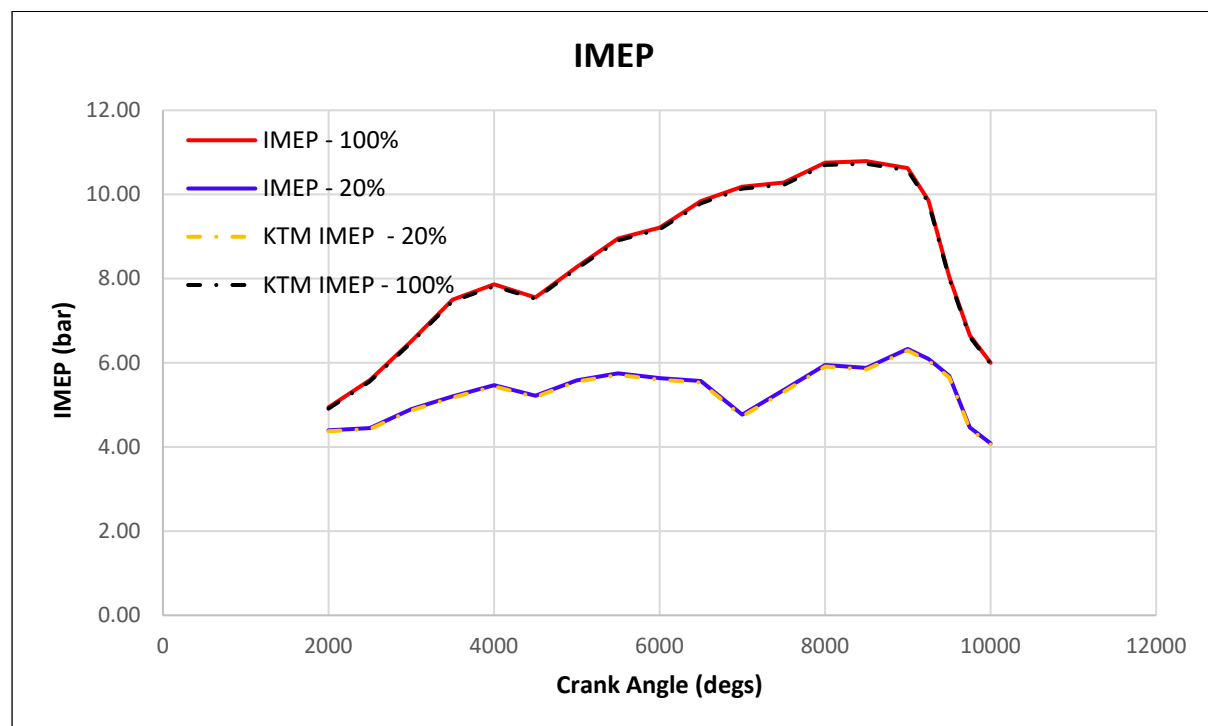


Figure 34 - IMEP (Vacas, Authors Own)

Figure 34 shows a comparison between low / high load IMEP as well as comparing the calculated values with those supplied by KTM. This graph shows a progressive increase in work done by the engine when under high load but at low load the IMEP curve stays flat and rarely peaks above 6 bar IMEP.

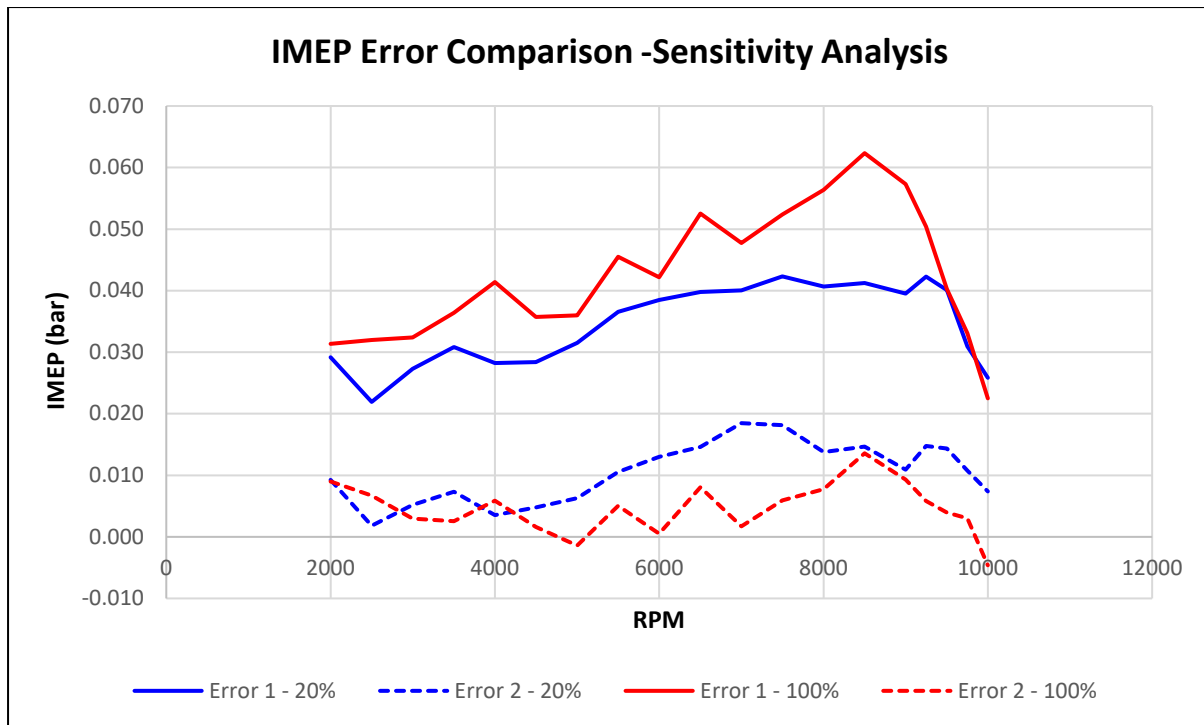


Figure 35 - IMEP Error Analysis (Vacas, Authors Own)

Although the values were very close to those supplied by KTM they were ever so slightly incorrect, and this was initially just assumed to be a rounding error. When calculating the IMEP, the swept volume used was exactly relative to bore and stroke (66.34mmx72mm) giving a swept volume of 248.87cc. It was decided by chance just to try the calculation by using the nominal capacity of 250cc and when using this figure, the error decreased by up to 80% and this is shown in Figure 35. This analysis may seem a little pointless and inconclusive, but it would be good to seek confirmation on this issue, as improving process integrity is always positive however small the improvement.

5.6 – Log Graphs

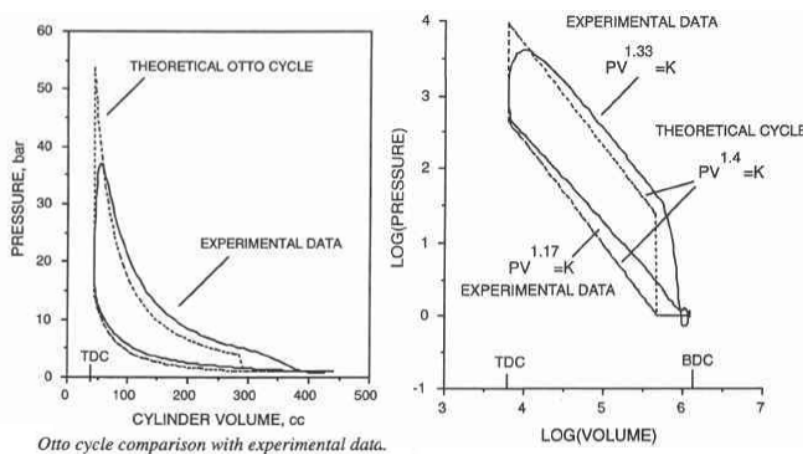


Figure 36 - Ideal Cycles (Blair, 1994)

5.6.1 – Graphical Analysis

It may not initially seem such an important area but as the project has progressed the graphical analysis of Log PV graphs has become an essential element of analysing the prescribed pressure data.

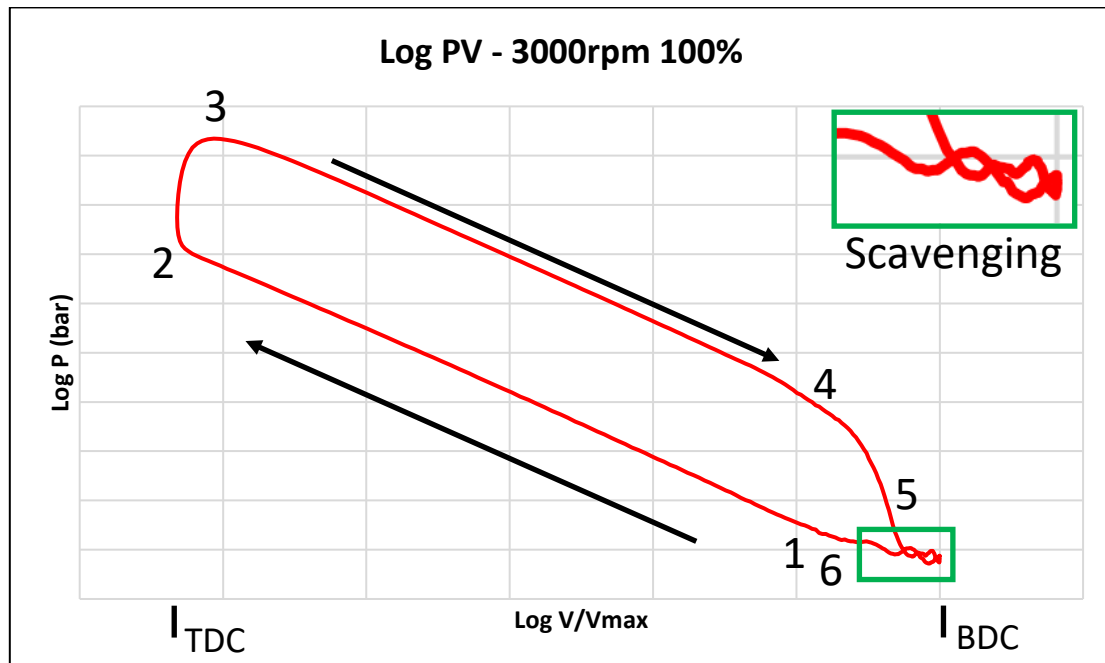


Figure 37 - LOG PV - 3000rpm 100%

Figure 37 shows the Log PV for 3000rpm and 100% throttle, this diagram is very much like the one shown in Figure 36 (Blair, 1994).

The steps can be summarised below;

- 1→2: This is the compression stroke and can be considered both isentropic (reversible & no heat loss) – this process starts when the exhaust port has closed.
- 2→3; This is the combustion process and is considered a constant volume process with heat addition.
- 3→4; This is the isentropic expansion process.
- 4→1; This is a constant volume heat rejection process.

Two extra points have been added (5 & 6) and these can be summarised below;

- 4→5; This is where blowdown (BD) occurs and is from when the exhaust ports open until the transfer ports opens.
- 5→6; This scavenging period, from when the transfer ports open to when the transfer ports close.

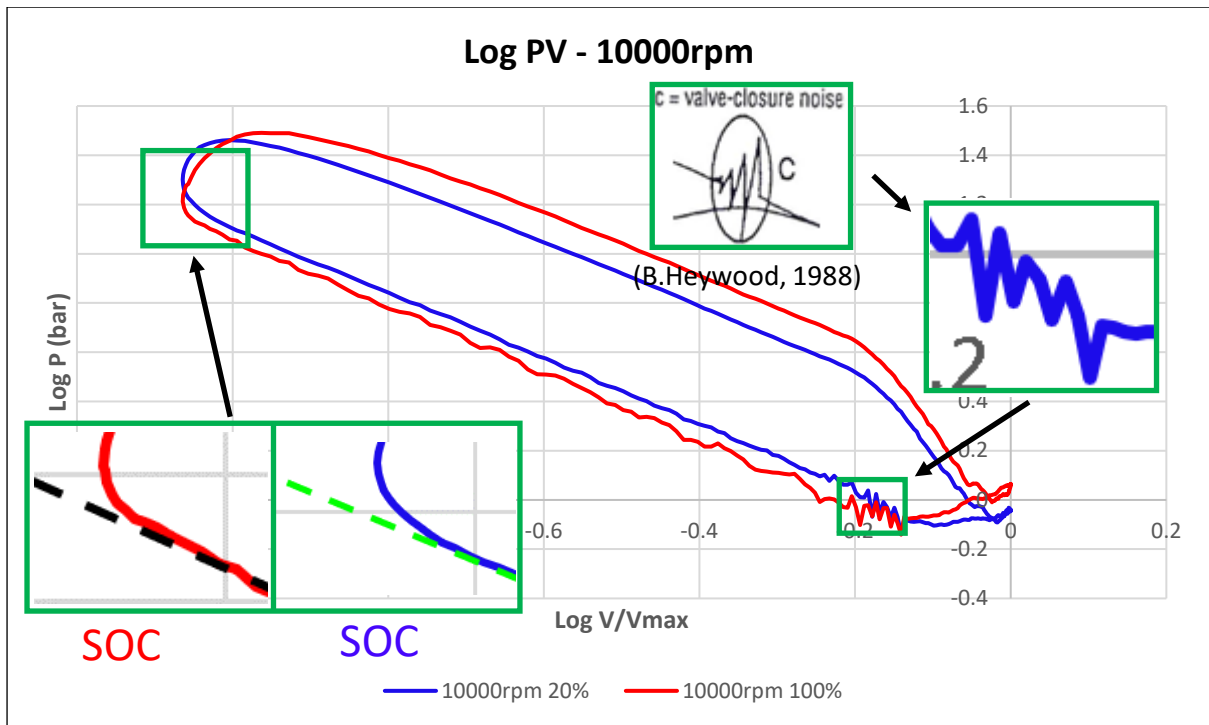


Figure 38 - Log PV 10000rpm (Vacas, Authors Own)

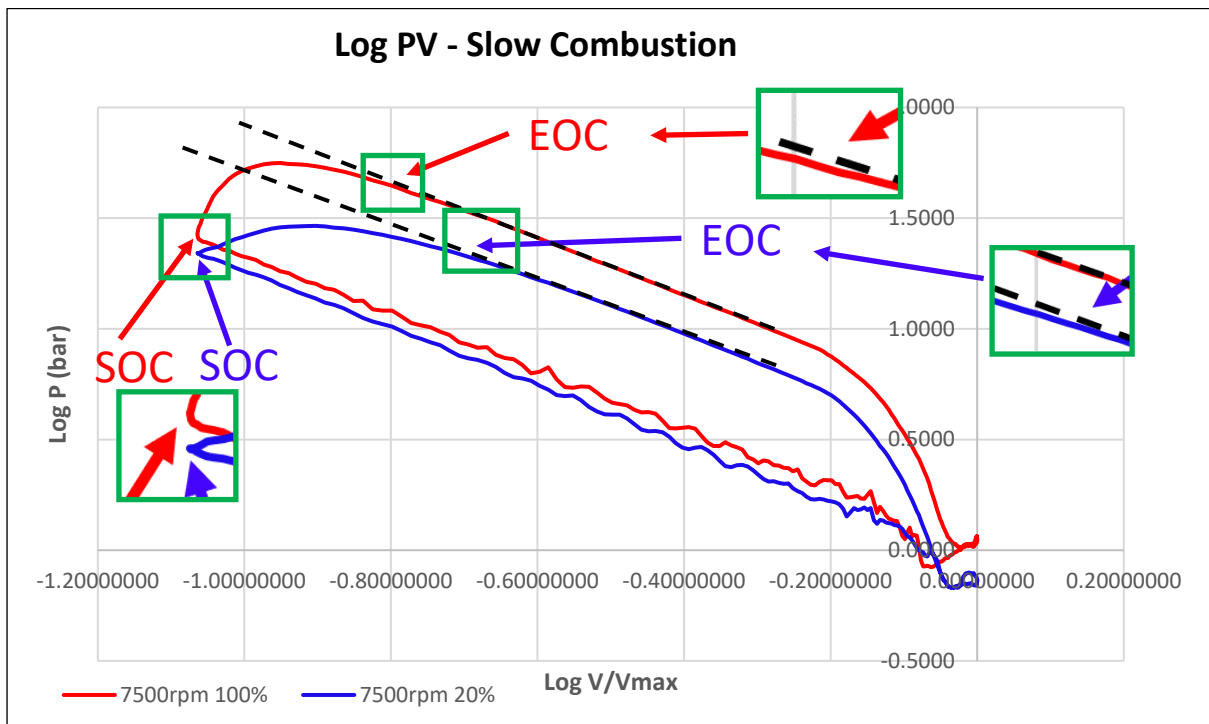


Figure 39 - Log PV 7500rpm Slow Combustion (Vacas, Authors Own)

Figure 38 shows two different subjects, first is the issue of noise around the exhaust port, this noise – this is confirmed as exhaust noise by (B.Heywood, 1988). The more important aspect of combustion that Figure 37 (10000rpm) highlights is protection against detonation, both Log PV profiles show gradual transition in and out of combustion, the gradual transition into combustion signifies SOC prior to TDC – SOC points are “-10” and “-6” low and high load.

Figure 39 shows a Log PV diagram comparing low / high load at 7500rpm, the start of combustion is very close to TDC in both cases (“-2” and “-4” respectively) but the low load plot shows a much different curve after TDC. The low load curve visibly takes longer to reach EOC, EOC in this case is represented by the curve aligning with the dashed guideline – EOC is “43” and “36” degrees ATDC (low and high load).

5.6.2 – Sensitivity Analysis (Combustion Chamber Volume)

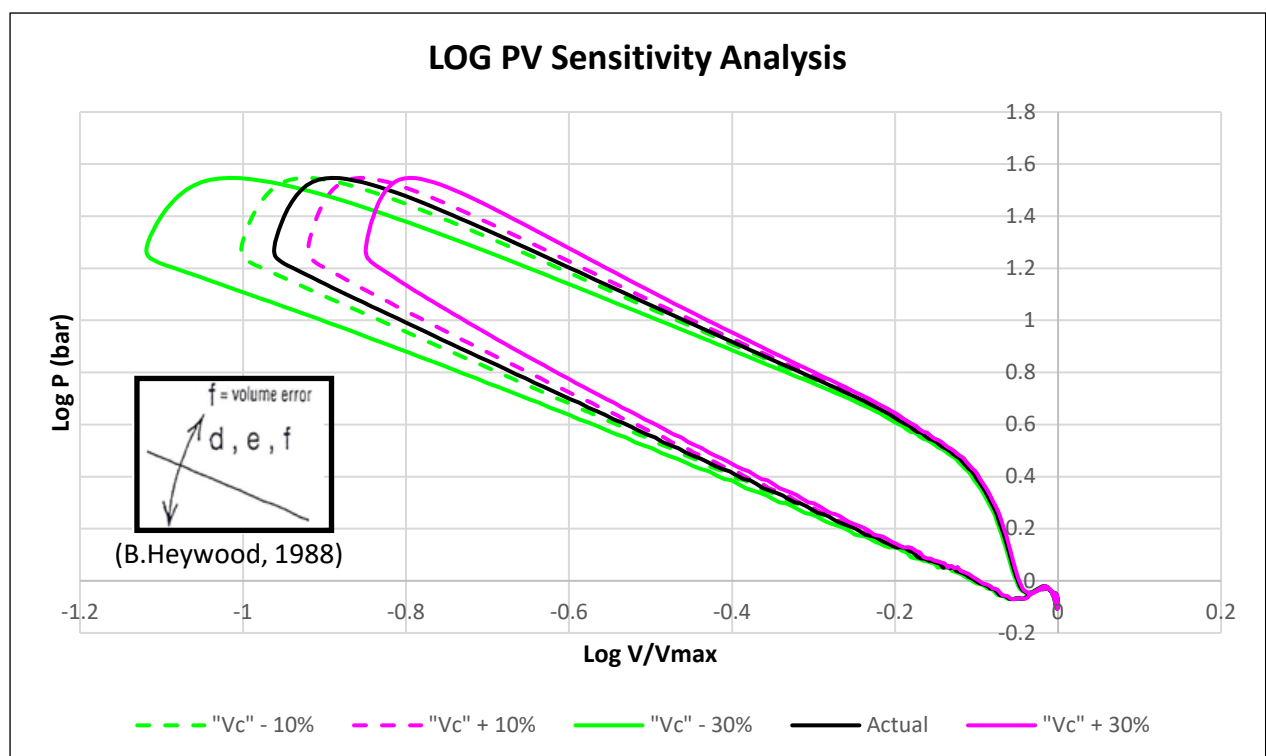


Figure 40 - Log PV Sensitivity Analysis (Vacas, Authors Own)

Figure 40 shows sensitivity analysis of the Log PV graphs in respect to combustion chamber volume, this piece of analysis was conducted due initial doubt in the measurement of combustion chamber volume. The “ideal” situation is that the slope of graph stays constant and does not deviate with curvature, variations of +/- 10% and 30% were plotted and showed that the measurement was in fact correct and showed as a straight slope in the middle, this visualisation of volume error is supported by (B.Heywood, 1988).

5.7 – Polytropic Index

From the ideal relationships shown in Figures 30, 36 and 37, the engine as a system can be considered to follow a polytropic process.

This relationship can be stated as,

$$PV^n = C$$

Where “n” can be considered the index of compression / expansion, or simply the polytropic exponent. The following chapter will discuss several different methods for calculating the polytropic index.

5.7.1 – Slope Calculation

The very first method when considering “n” is simply to consider “n” as the slope of either compression or expansion on the Log PV plots.

Taking the slope of the line can be stated simply as,

$$n = \text{slope} = \frac{\text{change in } y}{\text{change in } x} = \frac{Y_2 - Y_1}{X_2 - X_1}$$

Hence,

Equation 7 - "n" Slope

$$n = \frac{Y_2 - Y_1}{X_2 - X_1}$$

Using this very basic method it was simply a case of picking two points on the slope and performing the calculation. This calculation proved useful in terms of visualising the concept but in practical terms it was very slow to implement and lacked accuracy. These points are taken as close as possible to SOC and EOC whilst avoiding noise as best possible.

5.7.2 – Degree Based

The next logical step was to calculate “n” per degree, considering the polytropic relationship,

$$PV^n = C$$

Stated as,

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

Becomes,

$$P_1 V_1^n = P_2 V_2^n$$

$$\frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^n$$

Taking logs,

$$\text{Log} \left(\frac{P_1}{P_2} \right) = \text{Log} \left(\frac{V_2}{V_1} \right)^n$$

$$\text{Log} \left(\frac{P_1}{P_2} \right) = n \text{Log} \left(\frac{V_2}{V_1} \right)$$

Finally, “n” can be shown as,

Equation 8 - Polytropic Index "n"

$$n = \left(\frac{\text{Log} \left(\frac{P_1}{P_2} \right)}{\text{Log} \left(\frac{V_2}{V_1} \right)} \right)$$

Another representation for “n” can be shown as,

$$n = \frac{-(\text{Log}(P_2) - \text{Log}(P_1)))}{(\text{Log}(V_2) - \text{Log}(V_1))}$$

This equation simply represents the slope, the “-“sign represents the fact that in terms of volume (or fraction of) the slope or product of the equation is negative.

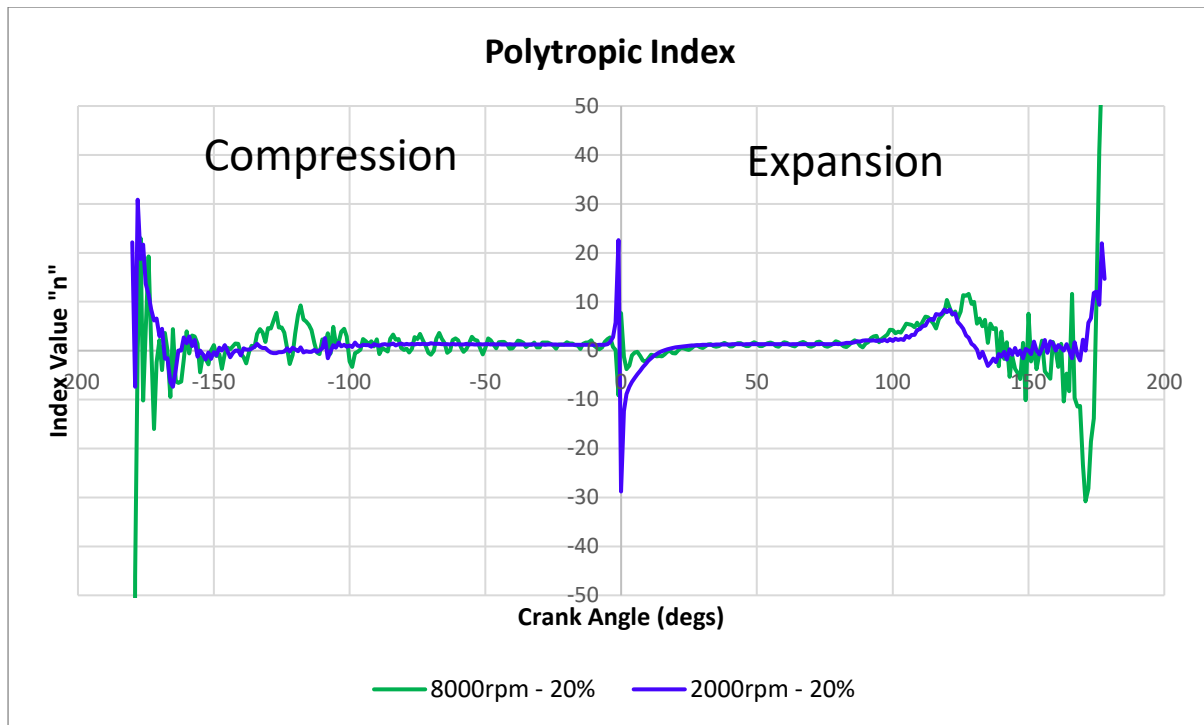


Figure 41 - "n" Per Degree (Vacas, Authors Own)

Figure 41 shows a plot of "n" per degree for the full crank rotation from -180 to + 180 comparing 2000rpm & 8000rpm at 20% it can be seen from this graph that there is a lot of variation and noise, so how can these values be used?

Although this method didn't directly provide any immediate values for "n", it did become a good tool for visualising the two-stroke gas exchange processes. Figure 42 shows clearly where scavenging ends (TPC) and then subsequently where the exhaust ports shut, note that even when the exhaust has "closed" for the 2000rpm trace there is still noise and variation. This is because "closed" means that the piston has reached the height of the exhaust timing valve (power valve), the noise is due to this valve not creating a perfect seal – a true seal is not achieved until around 10 degrees later when the port has ended completely, and the cylinder wall is the only sealing face. The 8000rpm trace does not experience this as by 8000rpm the power valve has fully opened, this is hard to visualise due naturally to the amount of noise at this rpm level.

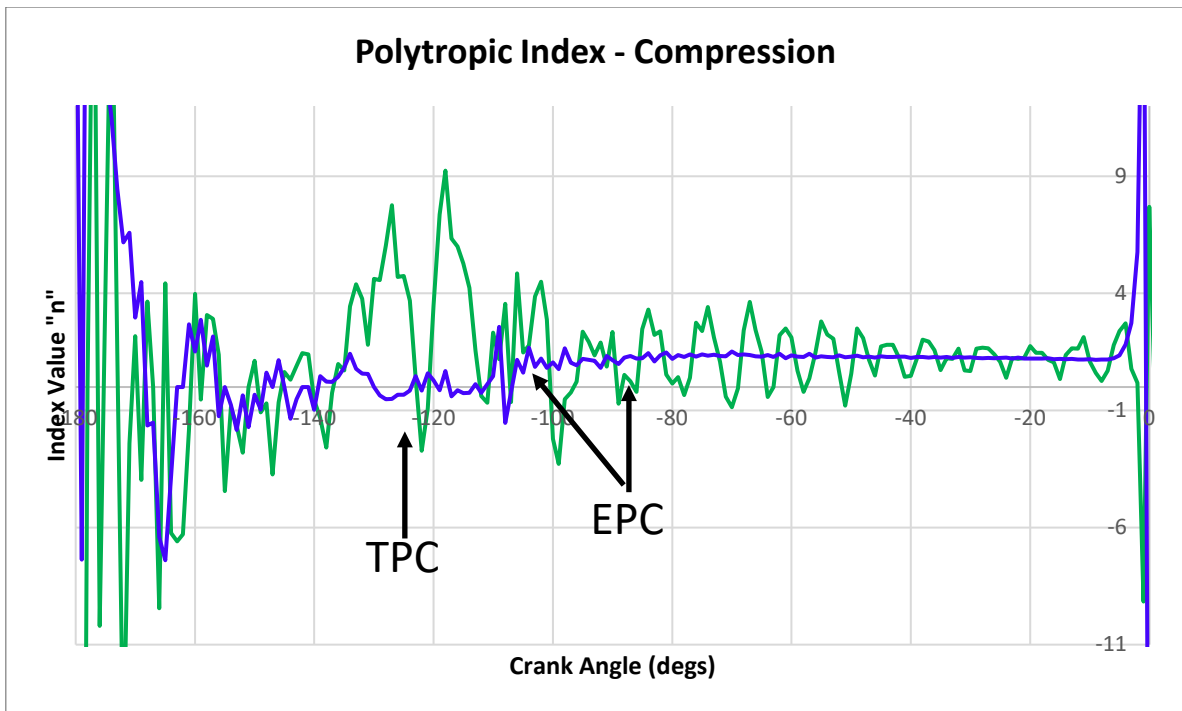


Figure 42 - Polytypic Index Compression (Vacas, Authors Own)

Figure 43 focuses on the expansion process and here the exhaust timing can be seen, the earlier exhaust port timing of the higher rpm trace can be clearly identified, whilst leakage and later opening of the lower rpm trace is also evident. The effect of the transfer ports opening can be seen very clearly.

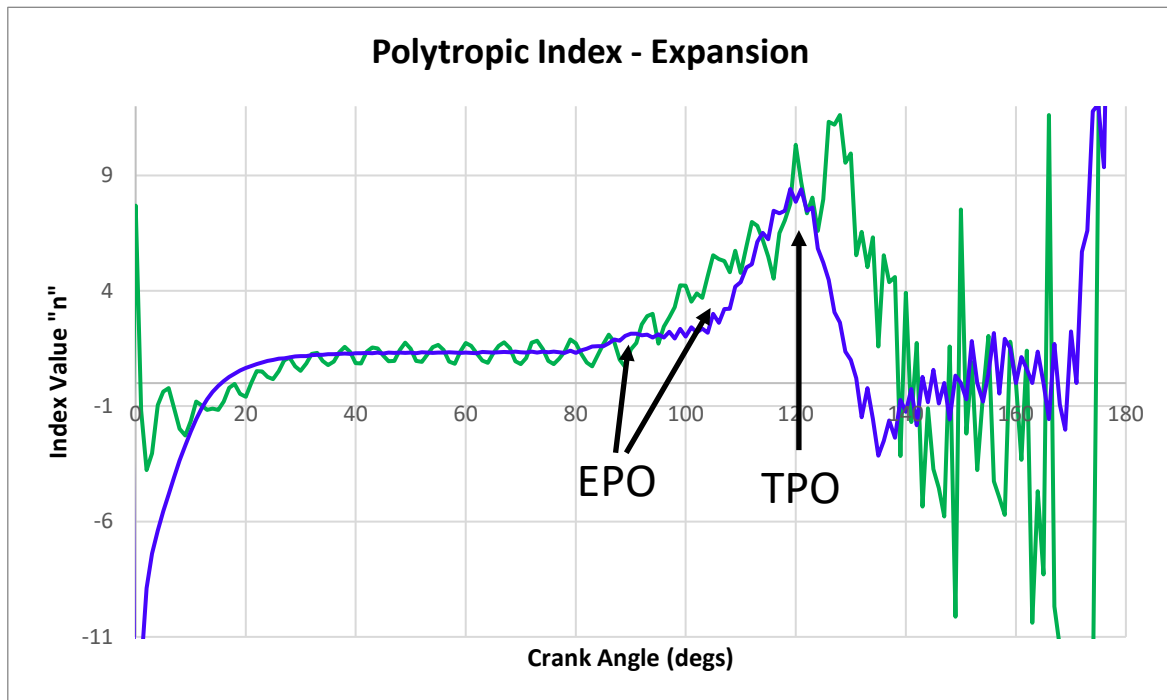


Figure 43 - Polytypic Index Expansion (Vacas, Authors Own)

5.7.4 – Range Selection (Step Change)

As an initial method for identifying SOC and EOC it was possible to look at the “n” per degree values and simply select the point at which a significant step change occurred. This method proved to have some success at low rpm points but there were more problems when looking at higher rpm points (naturally due to noise), Figure 44 shows this scenario very clearly.

2000rpm – 20%	STEP	-11	1.17611	9250rpm – 20%	STEP	-11	1.56809
		-10	1.19343		STEP	-10	0.58508
	CHANGE	-9	1.16553			-9	0.87207
		-8	1.17270		CHANGE	-8	2.32444
		-7	1.17360			-7	-0.58869
		-6	1.23049			-6	4.94014
	SOC	-5	1.34998		SOC	-5	-0.44605
		-4	1.81794			-4	3.38911
		-3	2.73941			-3	2.92973
		-2	5.76380			-2	-4.24943
		-1	22.57611			-1	23.87271
	TDC	0	-28.77978		TDC	0	-16.84592

Figure 44 - "n" Per Degree SOC (Vacas, Authors Own)

With a starting point for identifying SOC and EOC an average value for the polytropic index (compression & expansion) could be calculated by taking the step change identified SOC / EOC points, backing away (from SOC & EOC respectively) 5 degrees and subsequently take a 20-cell average.

5.7.5– Range Selection (Pressure Difference)

The next step having calculated initial “n” values was to calculate the pressure difference due to combustion. The change in pressure in during combustion can be described from,

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

Where,

$$P_1 V_1^n = P_2 V_2^n$$

Giving,

Equation 9 – “P2n” Pressure Due to Change in Volume

$$P_{2n} = P_1 \left(\frac{V_1}{V_2} \right)^n$$

“ P_{2n} ” simply signifies the pressure that would be reached if only piston movement and change of volume were considered, the subscript “n” is not active.

The pressure difference due to combustion can now be stated as,

Equation 10 - Pressure Difference Due to Combustion

$$\Delta P_{comb} = P_2 - P_{2n}$$

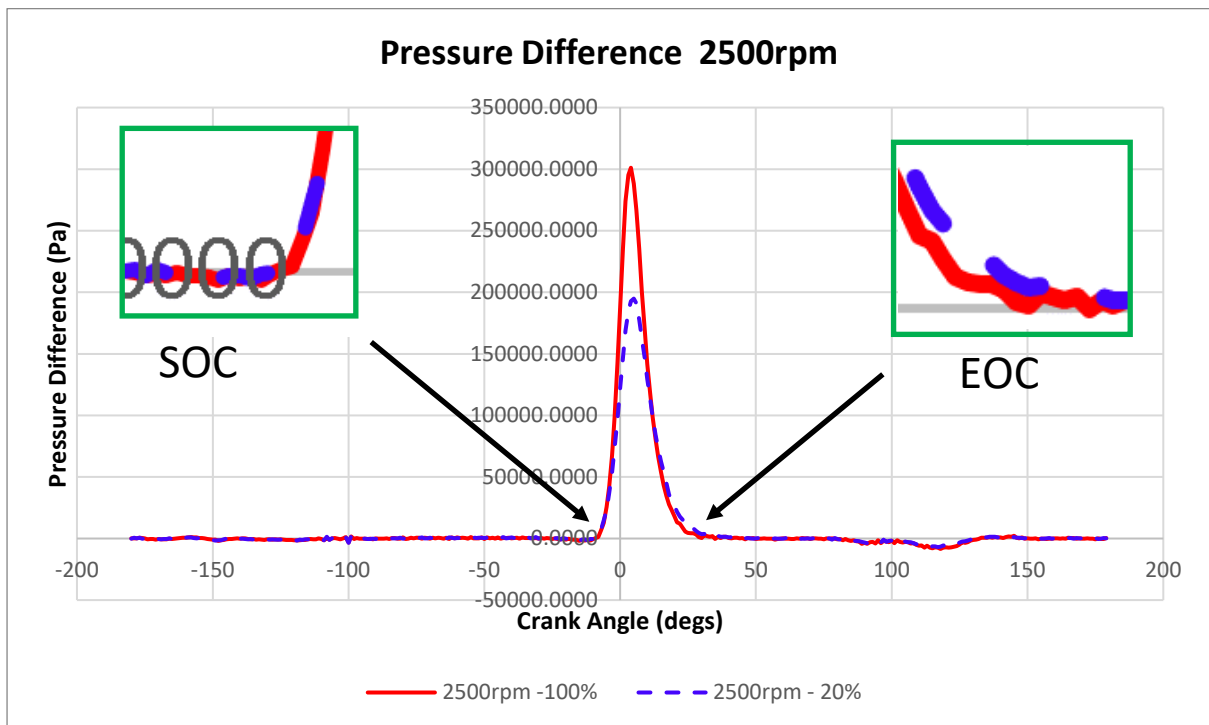


Figure 45 - Pressure Difference 2500rpm (Vacas, Authors Own)

Figure 45 shows pressure difference due to combustion at 2500rpm, as mentioned previously there is not a big difference in peak pressure when comparing low and high load. Figure 45 also shows the areas where “ ΔP_{comb} ” starts to increase / decrease and thus the start and end of combustion can be identified, this method is supported by (Klein & Erikson, 2005).

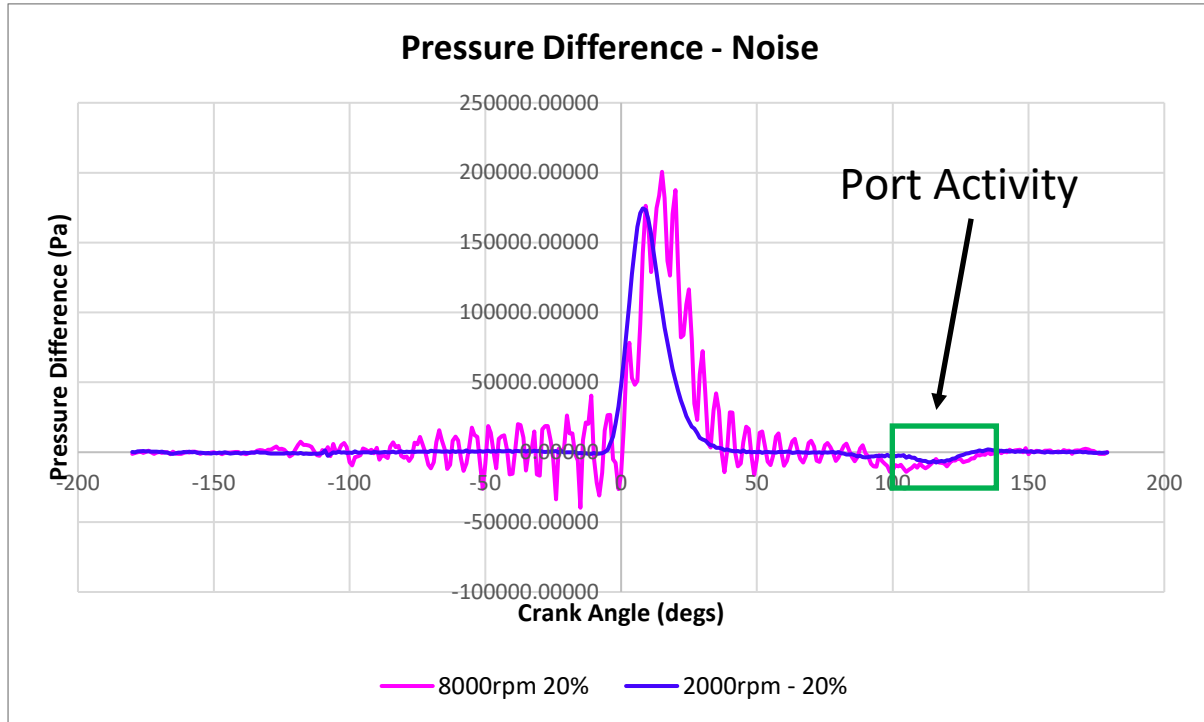


Figure 46 - Pressure Difference Noise (Vacas, Authors Own)

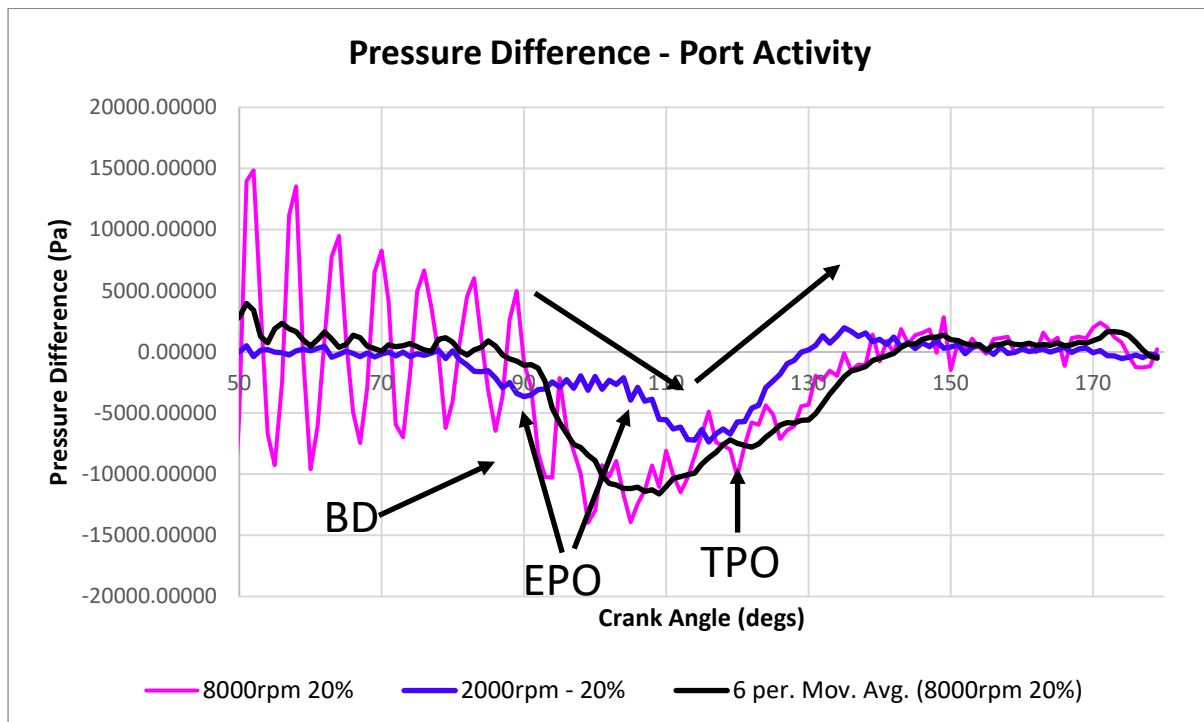


Figure 47 - Pressure Difference Port Activity (Vacas, Authors Own)

Figure 46 shows just how much noise and deviation can be present at the higher rpm points and trying to filter this noise would prove to be a tricky process to accomplish with certainty. It is also important to point out the difference between noise and simply the fluctuations due to port activity – shown in Figure 47 this is most evident on the downstroke where the exhaust and transfer ports open. It would be sensible to assume that the pressure difference at 8000rpm would drop sooner as the exhaust valve would be in its upper position but it can be seen that both traces actually start dropping at the same time although in different fashions – the 8000rpm trace drops drastically and with a higher magnitude, whereas the 2000rpm trace starts dropping but only slightly (showing how the pressure starts to leak before dropping more significantly when the piston starts to pass the exhaust valve).

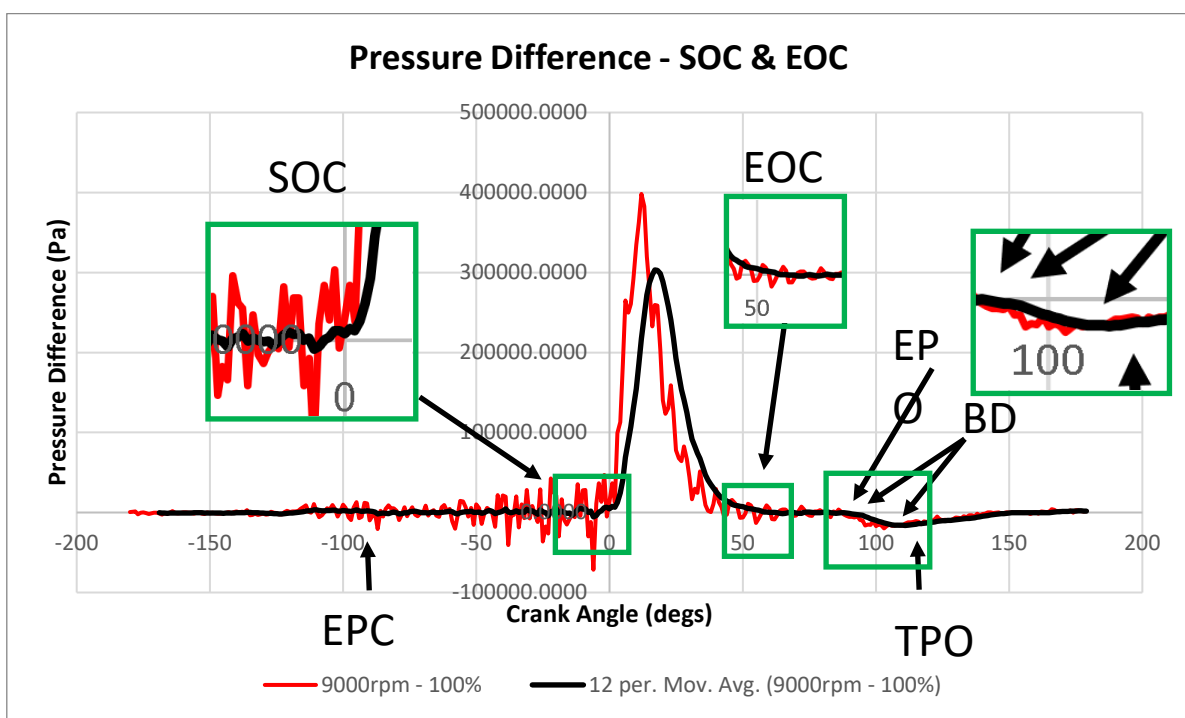


Figure 48 - Pressure Difference SOC & EOC (Vacas, Authors Own)

Figure 48 shows a clear example of how pressure difference was used to identify SOC & EOC, there was no true way to filter out noise and it was decided that it was best not to edit or manipulate the raw data. Figure 48 shows the application of a moving average trend line in excel, this was not used as a strict rule but more importantly it was applied as a visual aid to give a better visual appraisal of where the data would tend away from or to zero. Figure 48 does show though how applying this concept as a visual aid did shift deviation along the crank angle axis (potentially having a significant error), so whilst this method worked well where there wasn't any noise it was questionable how well it really worked with noisy data and if it did work was it being too sensitive? This is where the question began to arise of what exactly should be defined as SOC & EOC and how sensitive should this appraisal be?

5.7.6 – Range Selection (Log PV Visual)

From analysing the pressure difference data, it was questionable how accurate the method was with noisy data and there was the general question of whether this process was being too sensitive. The next strategy reverted to the method of identifying SOC and EOC by the simple appraisal of the Log PV graphs, following the principle of identifying deviation from the ideal slope line.

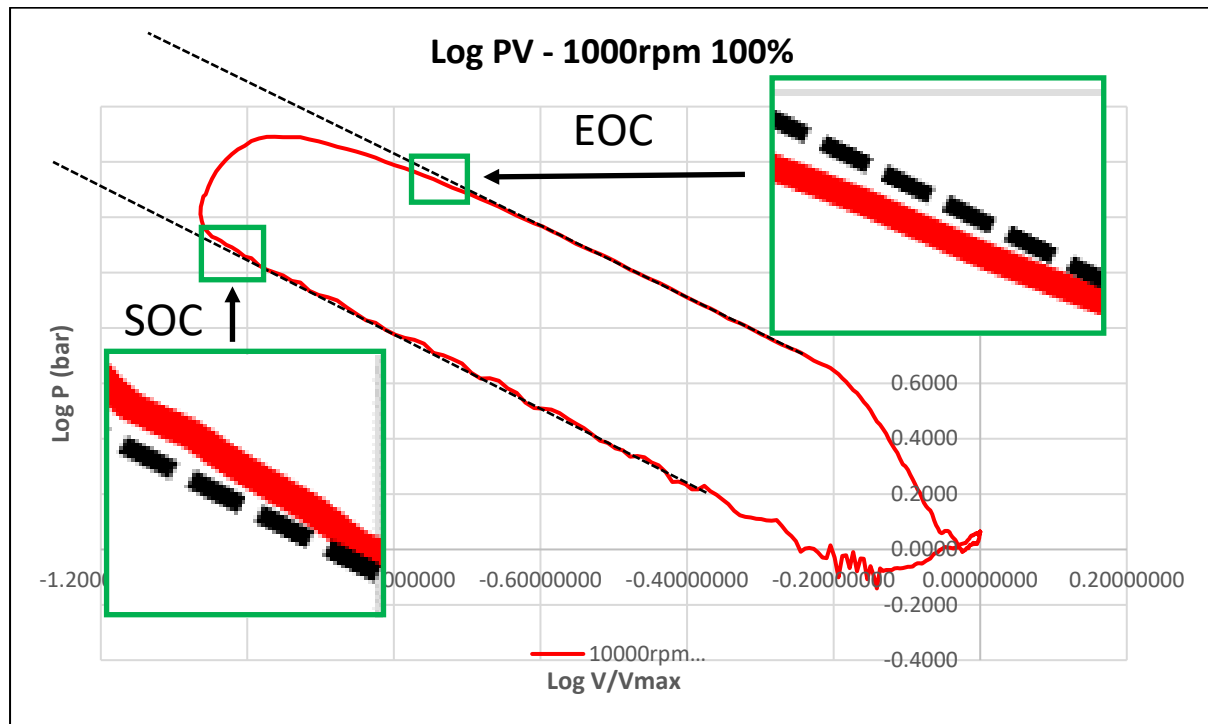


Figure 49 - Log PV (SOC & EOC) (Vacas, Authors Own)

Figure 49 shows the principle of deviation from the ideal slope and it also shows that on the compression slope there is more noise (this is true as a rule), but it is reasoned that when appraising the data in this manner (either compression or expansion) any noise does not have as much of an effect as other methods. Simply stated this can be put down to the fact that the nominal trend although manually / visually implemented, uses a significant portion of data to derive its form. Whilst not perfect, this method has proved to be quick and easy to implement and gives a good view of the combustion process phases in and out, with this method implemented the range selection method was again used respective of the newly identified SOC & EOC points.

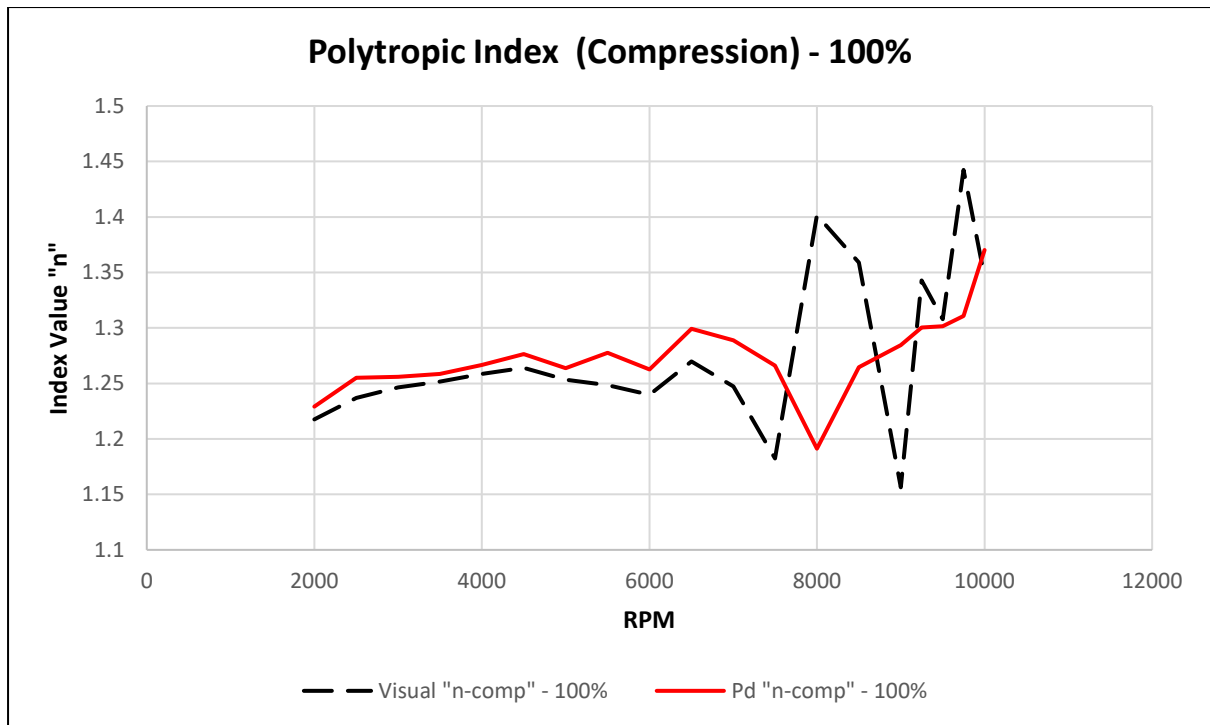


Figure 50 - Polytropic Index Method Comparison (Compression – 100%) (Vacas, Authors Own)

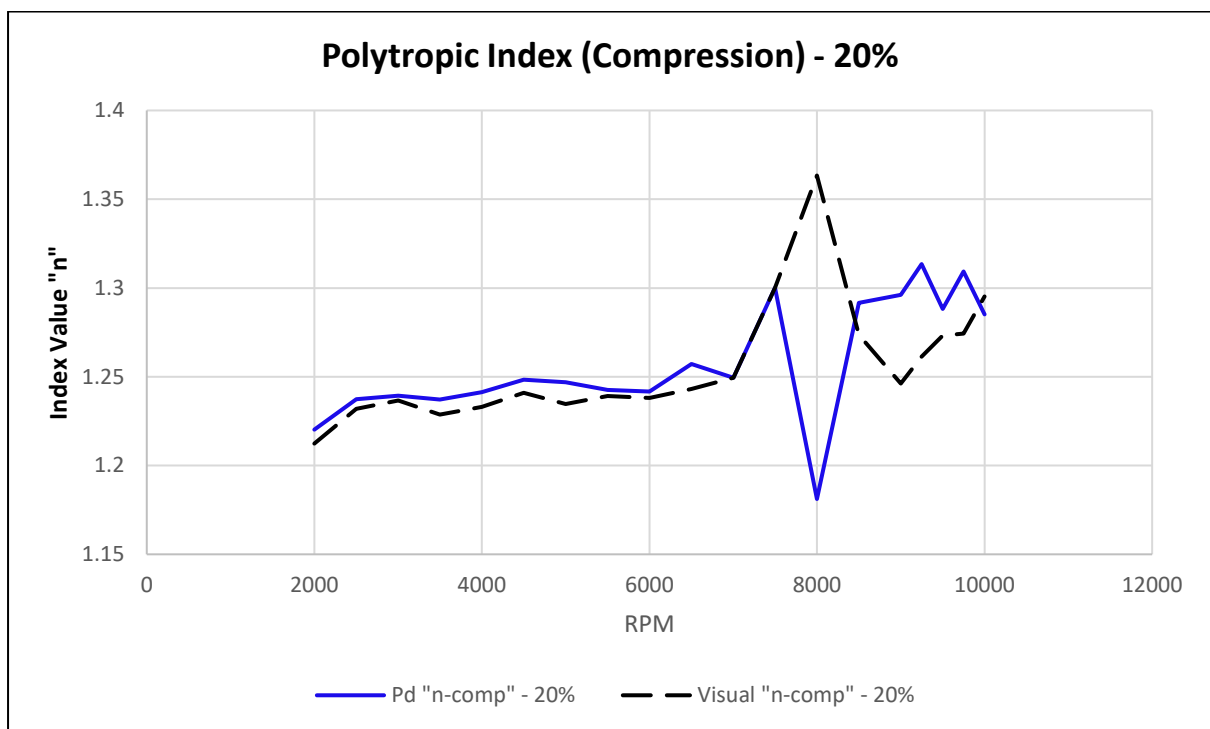


Figure 51 - Polytropic Index Method Comparison (Compression - 20%)

Figures 50 & 51 show the comparison of polytropic index values (compression) calculated for both high and low load and for both the pressure difference and visual combustion identification methods. It can be noted that there isn't actually much difference between the two until high rpm and it can be said that the pressure difference method holds a more stable trend, in particular with the high load data.

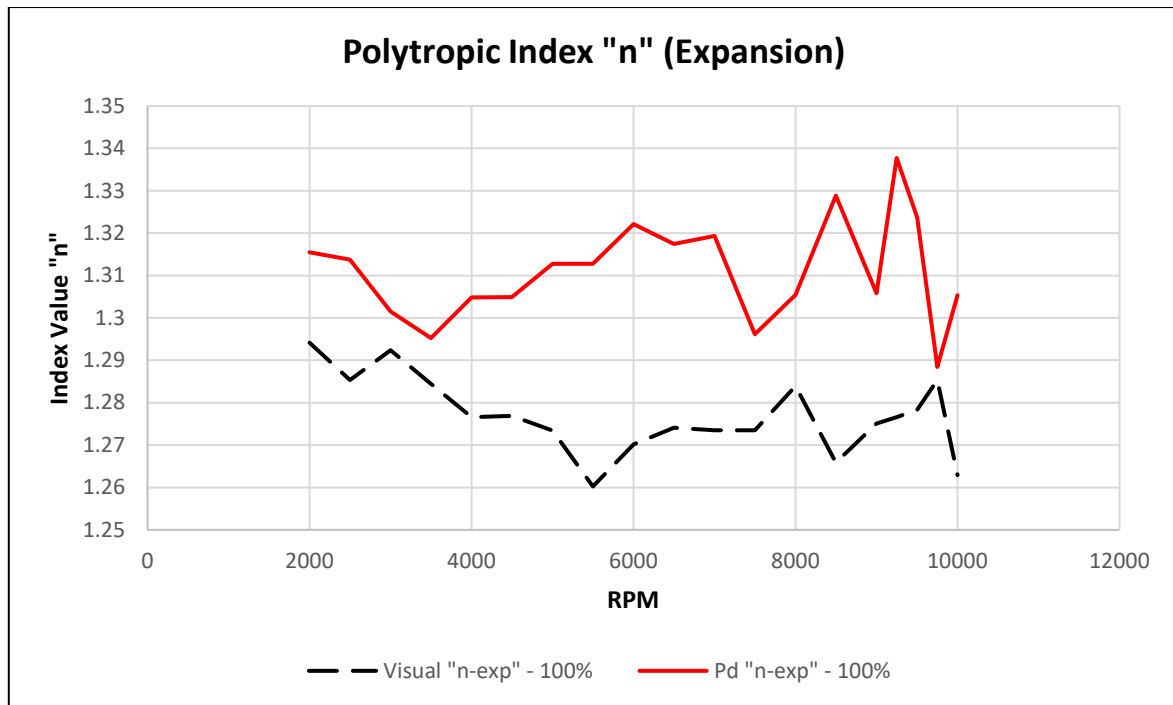


Figure 52 - Polytropic Index Method Comparison (Expansion - 100%) (Vacas, Authors Own)

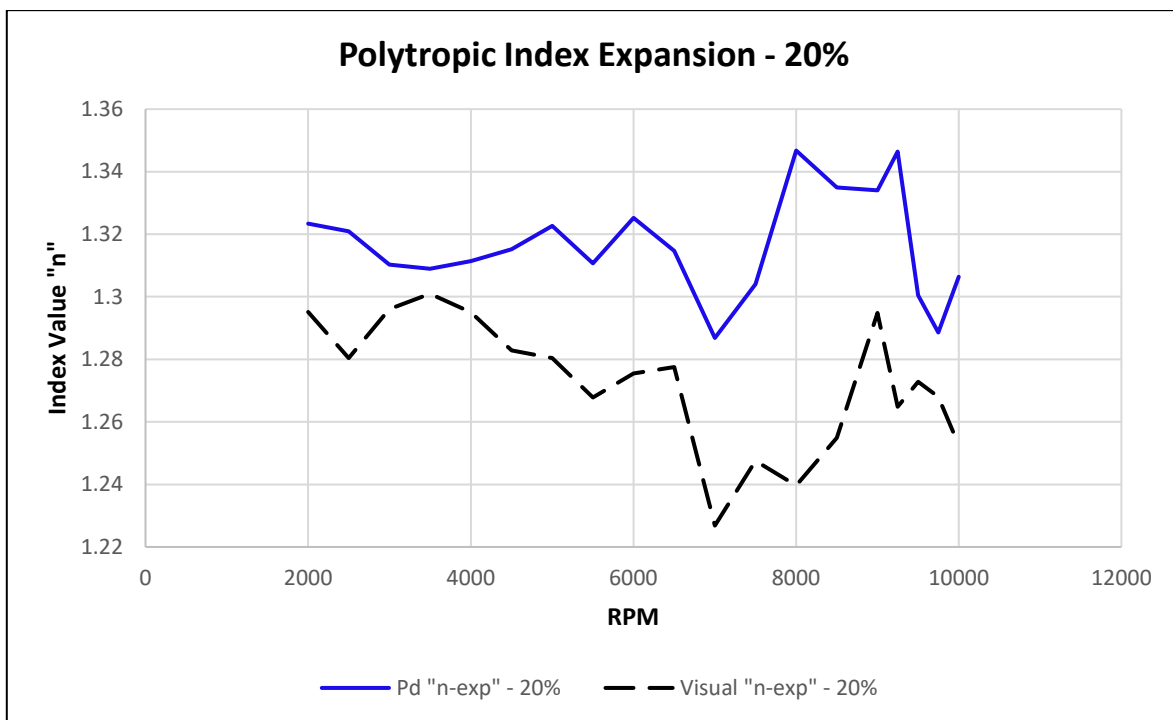


Figure 53 - Polytropic Index Method Comparison (Expansion - 20%) (Vacas, Authors Own)

Figures 52 & 53 show the comparison of high and low load “n” values from both visual and pressure difference methods in the expansion phase, from these graphs there is a distinct difference from around 3000rpm onwards. The expansion phase is a harder area to appraise as EOC is not determined by a spark, naturally at the end of combustion and just after where there is a certain amount of volatility, more so at high rpm.

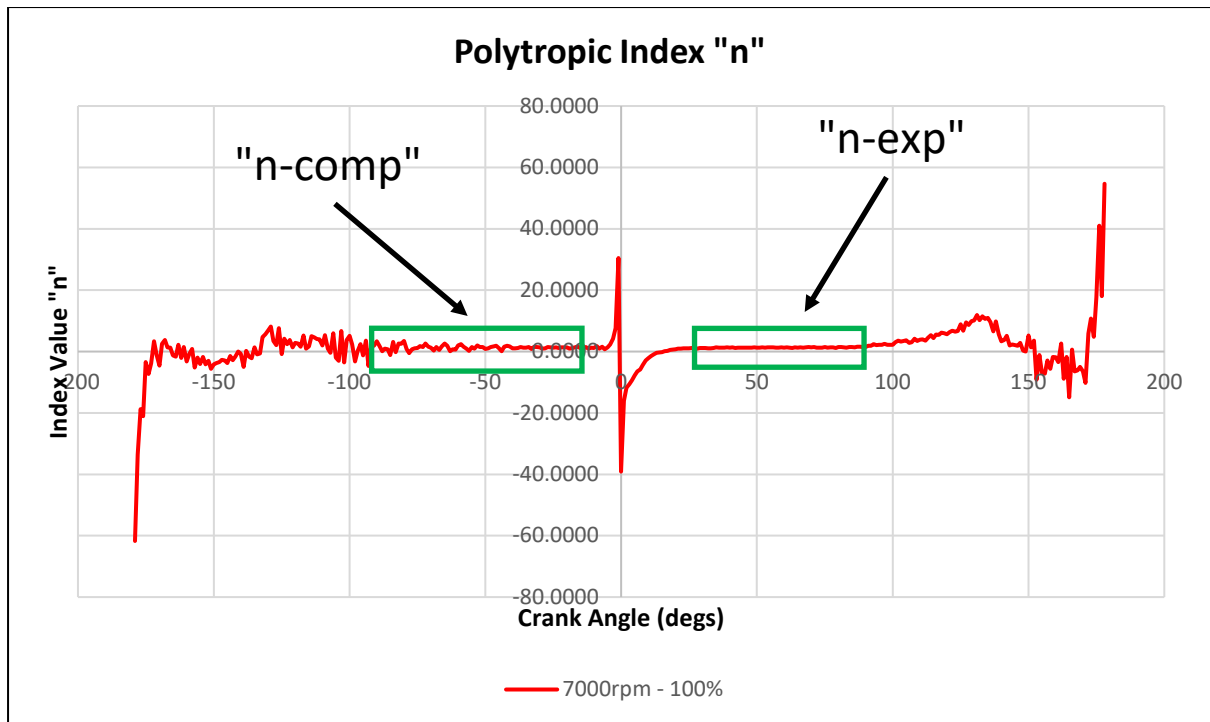


Figure 54 - Polytropic Index - 7000rpm 100% (Vacas, Authors Own)

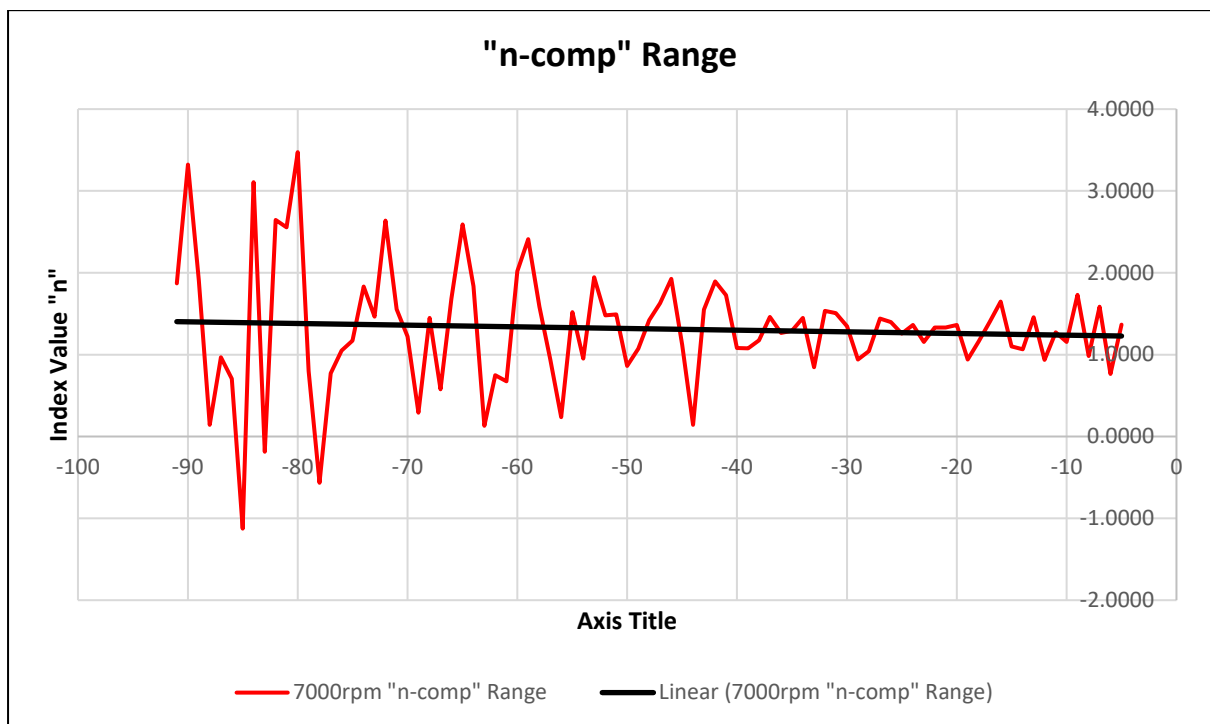


Figure 55 - "n-comp" Range (Vacas, Authors Own)

Figure 54 shows a graph of the degree based "n" values for 7000rpm 100% load, highlighting the areas from which "n-comp" and "n-exp" are identified. Focusing first on the compression phase a sample of data is taken from roughly EPC to just before the ignition spark, it is quickly apparent that this sample of data shows only a slight linear trend tending down towards zero.

It would be fair to say that the range selection close to SOC is not as sensitive as that around EOC, though as shown at high rpm there is more distortion and potentially the range selection could be backed off slightly further, possibly another 5 degrees before SOC - or even just a rule of 5 degrees before the spark. Highlighting this even further is the focused range selection in Figure 56, showing how from “-40” to “-5” there is a very consistent linear trend with only a slight amount of noise occurring after “-10”, note spark timing and SOC are “-11” and “-5” respectively, this further promotes the idea of using spark timing as the defining point from which the range section method is used in compression.

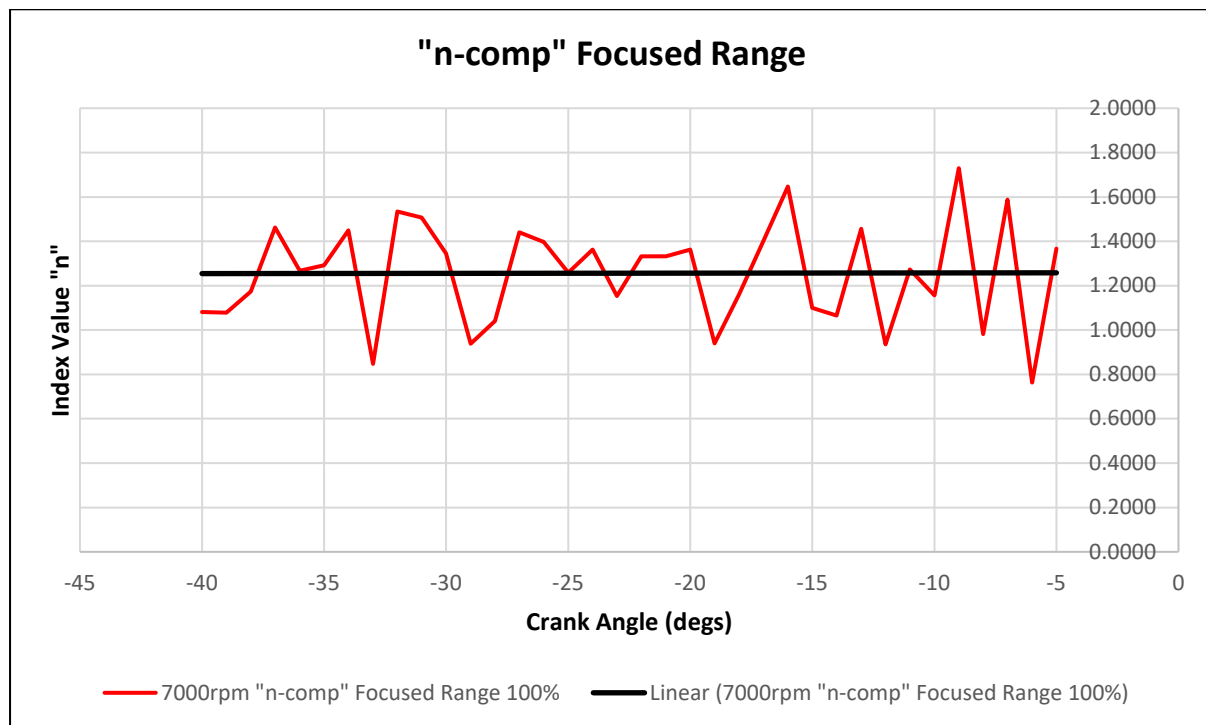


Figure 56 - "n-comp" Focused Range Selection (Vacas, Authors Own)

Figure 57 shows the range selection for expansion which is naturally a smaller range due to the volatility surrounding the end of combustion. Straight away it can be seen that the “n” values for expansion show a clear trend increasing in value. This small range is reduced even further in Figure 58 where the sampling window is reduced even further to just 20 degrees (“41” to “61”), this could be reason to say that whilst calculation of “n” for compression is not that sensitive, the variation shown between methods when calculating “n-exp” indicates that in this situation it may be of benefit to make a more acute appraisal of when combustion ends.

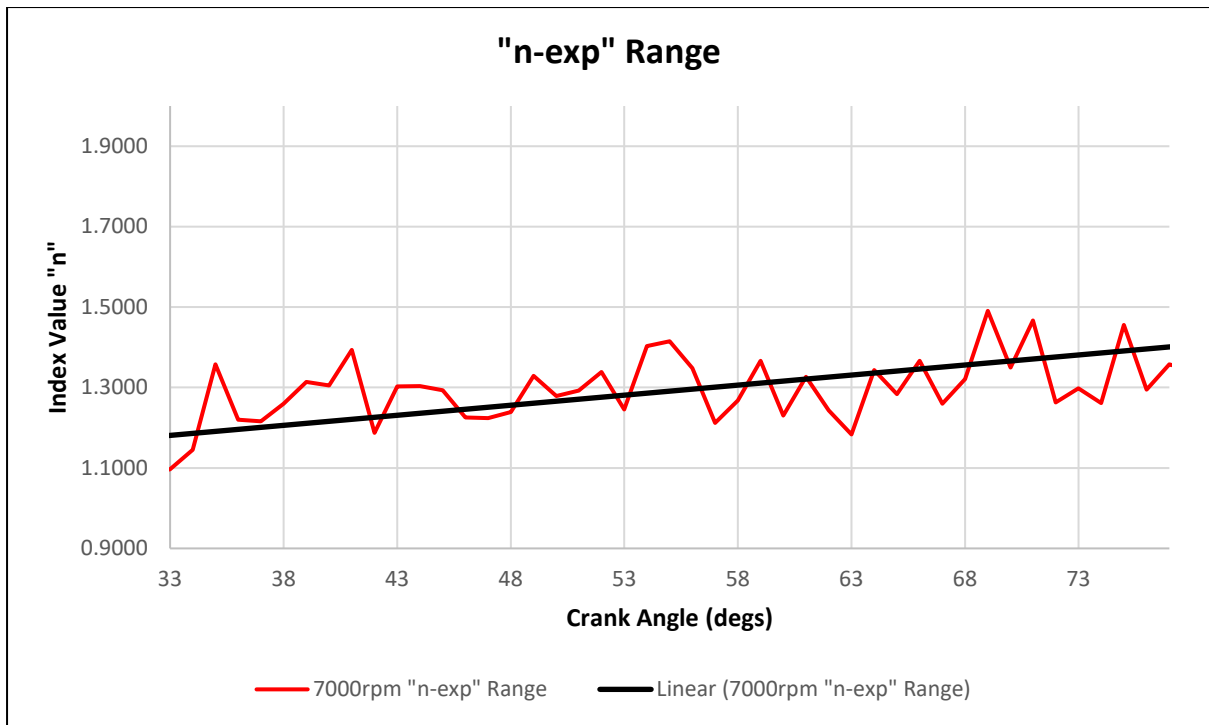


Figure 57 - "n-exp" Range Selection (Vacas, Authors Own)

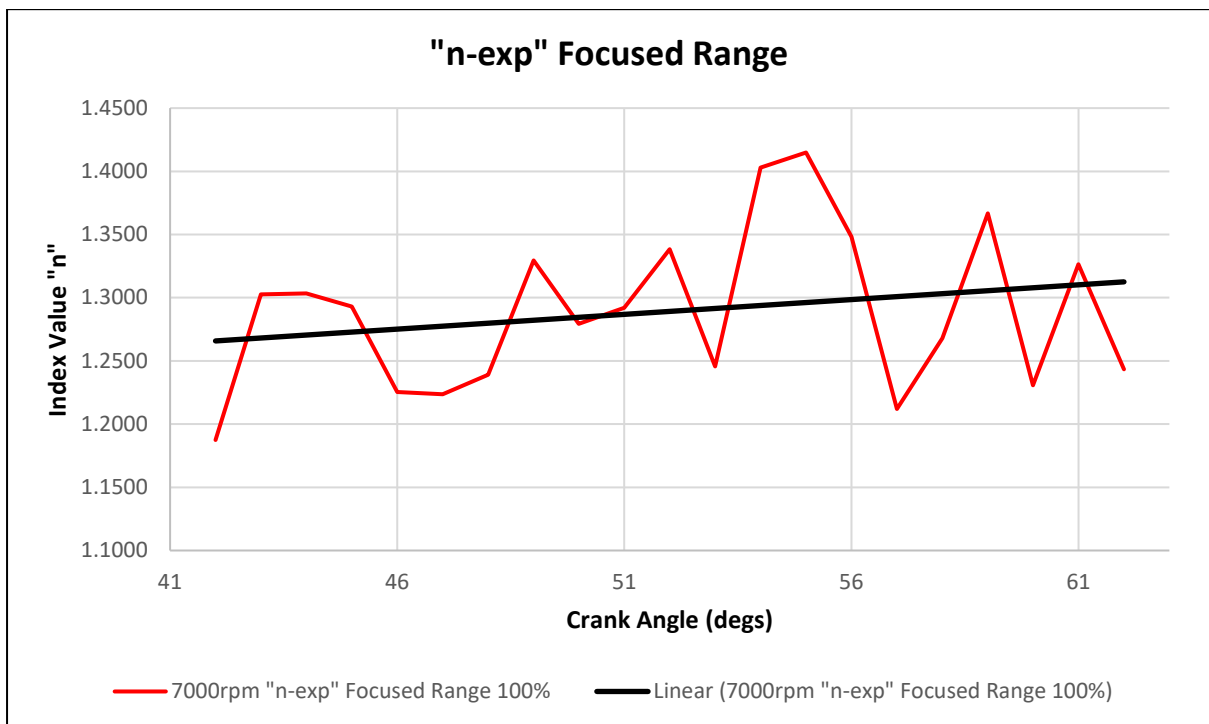


Figure 58 - "n-exp" Focused Range Selection (Vacas, Authors Own)

5.7.7 – KTM Values

KTM quote the indices as 1.32 and 1.27 (compression and expansion), these values are explained as typical values for a port injected engine (Anschuetz, 2018). Though the values are rather the opposite of what has been calculated and contrary to what Blair states;

(Blair, 1994)

“In the absence of better experimental evidence from a particular engine, the modeler could have some confidence in using the polytropic indices 1.25 & 1.35 for the compression and expansion phases, respectively”

The average values calculated were 1.27 & 1.31 (compression & expansion) when identifying SOC & EOC with the pressure difference method, when using the visual Log PV to identify SOC and EOC these values were 1.28 and 1.31, respectively. The KTM discrepancy may at first seem quite significant, which in some sense it is – but when analysing data for power and performance combustion is usually considered between 10% & 90% mass burn (B.Heywood, 1988) & (Zhao, 2007).

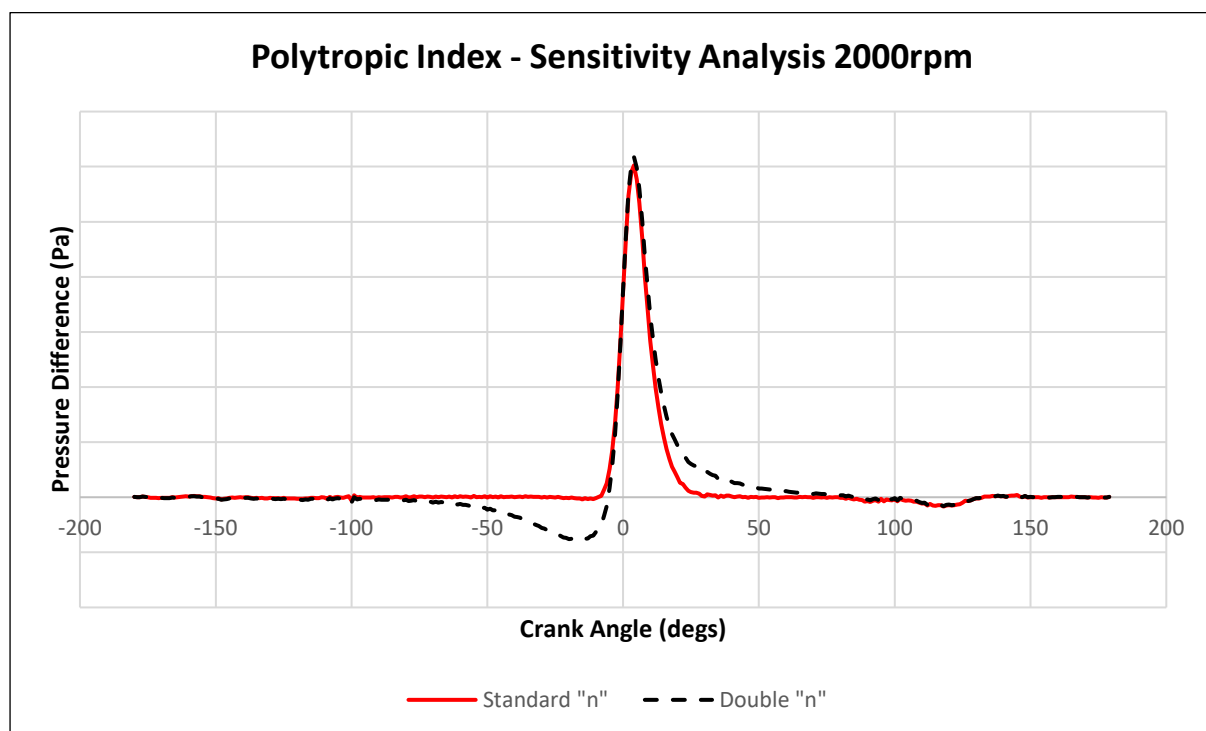


Figure 59 - Polytropic Index Sensitivity Analysis (Vacas, Authors Own)

Figure 59 shows a pressure difference graph for 2000rpm (100%), doubling the “n” value does not cause the pressure difference to deviate during the main portion of combustion, but in the areas surrounding SOC & EOC there is distortion. This indicates that selecting the most accurate “n” values is important when using pressure difference to determine SOC & EOC.

Figures 60 & 61 show the charted polytropic indices (compression & expansion) and compares them against the values supplied recommended by KTM, on average there is roughly a 4% to 5% difference.

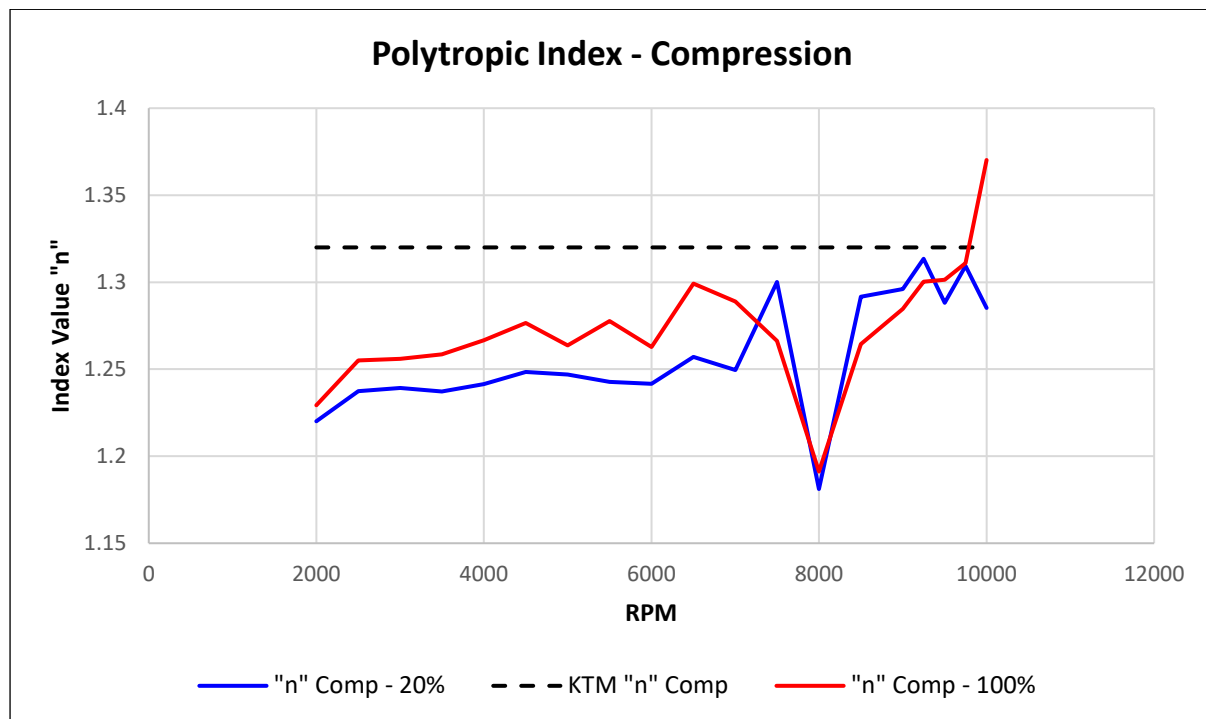


Figure 60 - Polyotropic Index Comparison (compression) (Vacas, Authors Own)

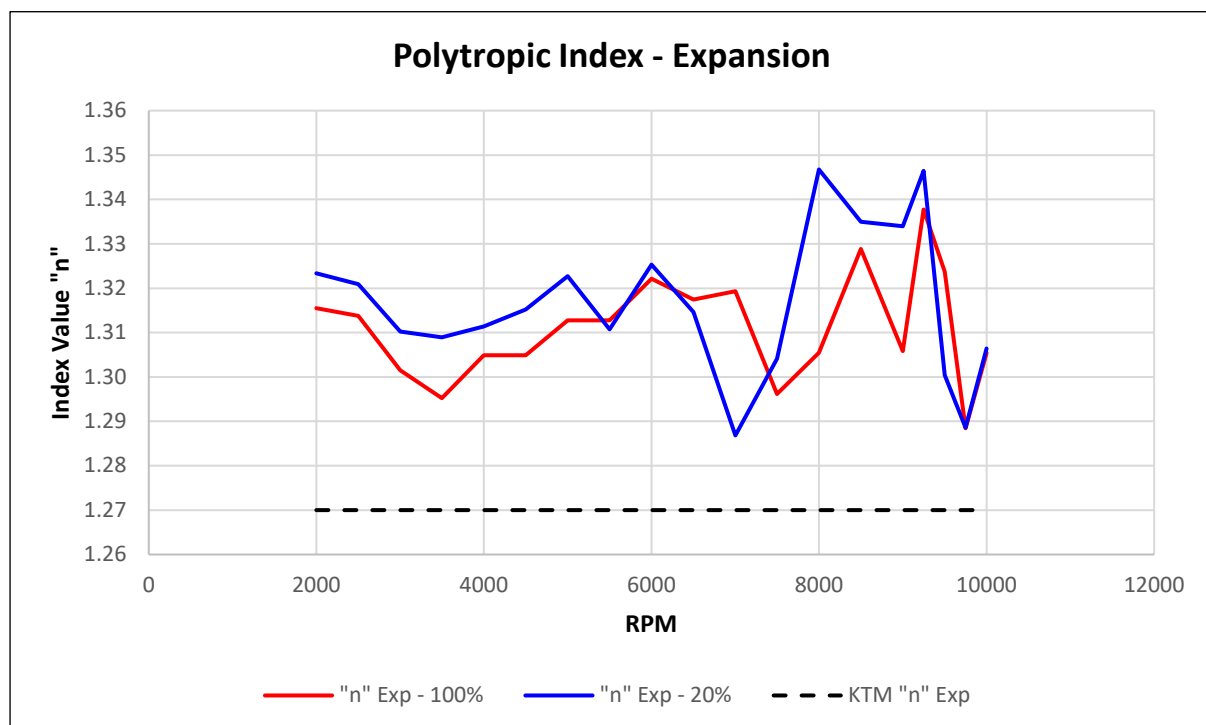


Figure 61 - Polyotropic Index Comparison (expansion) (Vacas, Authors Own)

5.8 – Combustion Timing

Having reviewed several techniques for evaluating SOC and EOC, the topics of delay, duration and timing can be evaluated.

5.8.1 – Ignition Timing

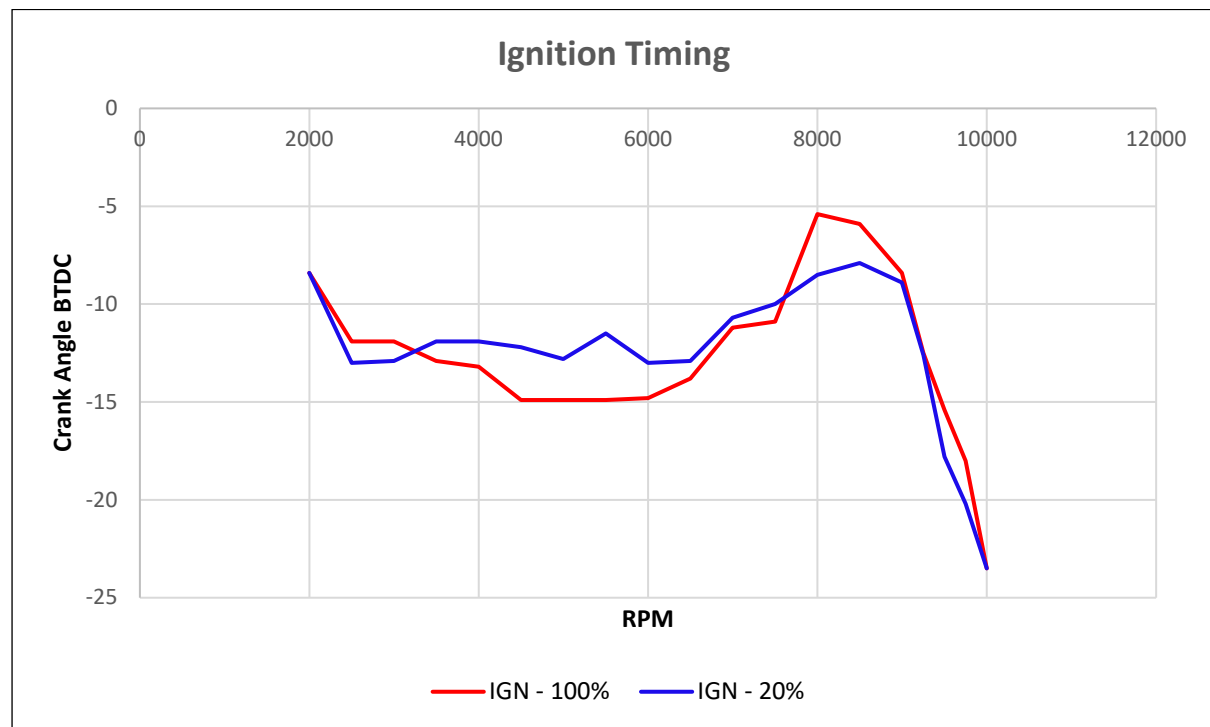


Figure 62 - KTM Ignition Timing (KTM, 2017)

Figure 62 shows how both rpm points initially advance timing to allow for slow combustion and smoother running but when the engine reaches the point at which the exhaust valve starts to open both timing curves start to retard. The retardation around optimum operating rpm is due faster combustion and the way two-strokes work, the hotter the gases entering the exhaust, the faster the speed of sound and the faster the gases travel which results in better scavenging. The extreme advance at high rpm is to account for the natural breakdown in the two-strokes gas exchange process and protect against overrev.

5.8.2 – Delay & Duration

Before calculating delay and duration it must be stated that,

Equation 11 - RPM Per Second

$$RPM \text{ Per Second} = \frac{N}{60}$$

Equation 12 - Second Per Revolution

$$Seconds \text{ Per Revolution} = \frac{1}{\left(\frac{N}{60}\right)}$$

Finally Stating,

$$Seconds \text{ Per Degree} = \frac{Seconds \text{ Per Revolution}}{360}$$

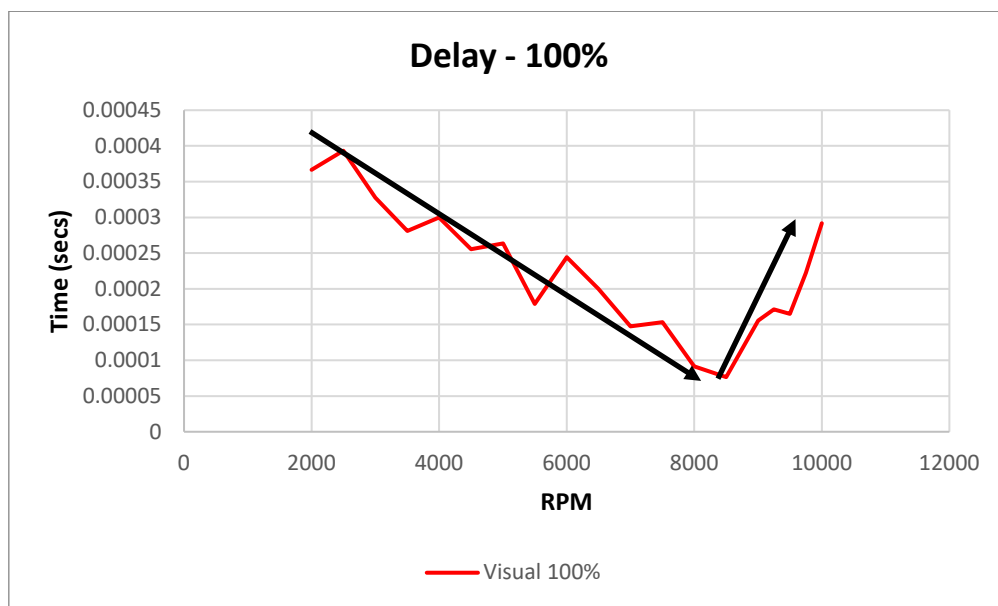


Figure 63 - Delay 100% (Visual) (Vacas, Authors Own)

Figure 63 show the delay time profile calculated for 100% using the visual Log PV technique to identify SOC and EOC - this simple profile is a good illustration of how the delay decreases as the engine gets closer to its optimum operating point (8500rpm) after this the delay increases as inefficient combustion is triggered earlier to protect the engine.

Figures 64 and 65 a comparison of delay and duration, the methods compared being pressure difference and Log PV visual analysis. Figure 65 shows comparable trends in terms of duration time decreasing with rpm, albeit with a difference in scalar quantity. Figure 64 shows a rather different story with the pressure difference method showing only sporadic signs of correlation and clear trend. The two visual methods appear much more successful from this perspective, showing a clear decrease in delay as rpm reaches peak power and then an increase at high rpm – and logically the less efficient low load combustion shows a consistent increase in delay time other than at high rpm where both low and high load combustion processes are faltering.

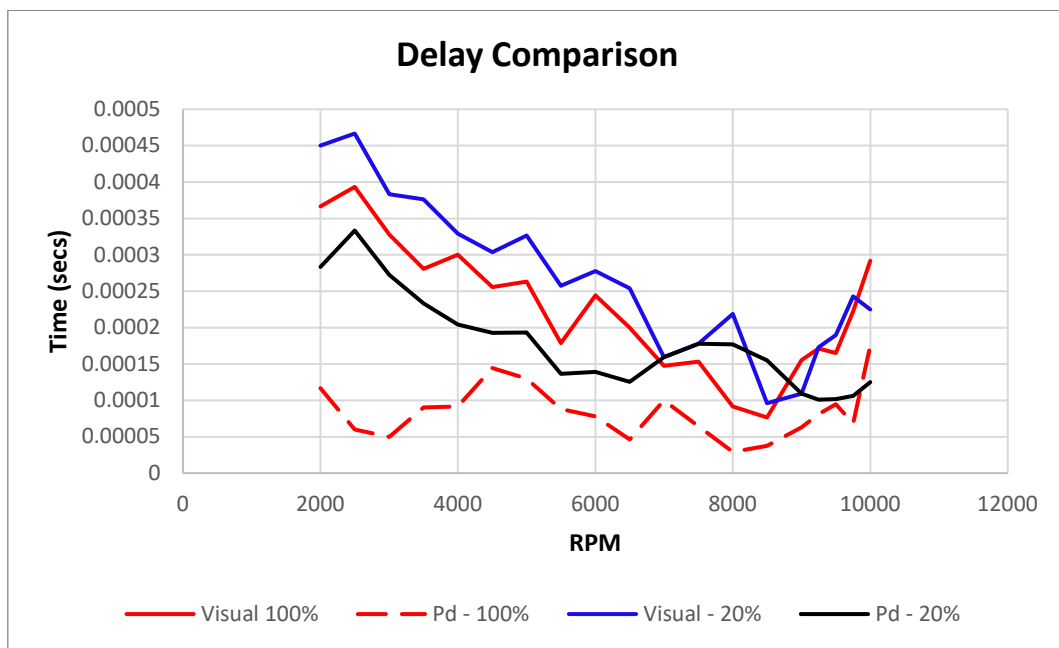


Figure 64 - Delay Comparison (Vacas, Authors Own)

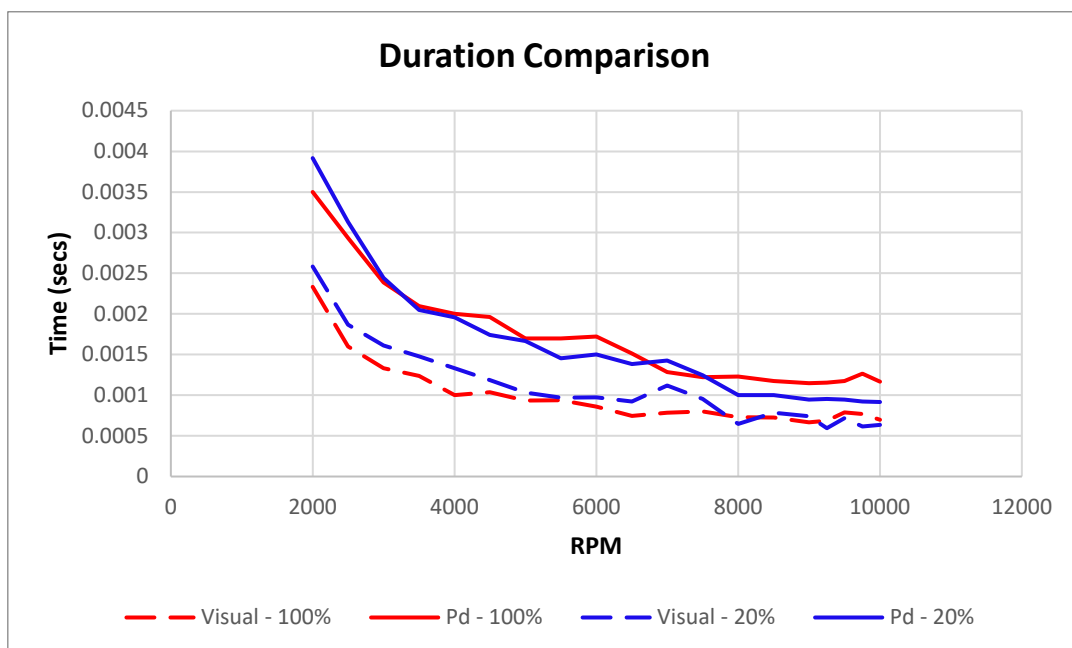


Figure 65 - Duration Comparison (Vacas, Authors Own)

5.9 – Mass Fraction Burn

5.9.1 – Calculations

The next step in evaluating the engine data was to calculate the mass fraction burn values. Mass fraction burned can be described as the percentage of heat released from the fuel burnt, relative of course to the heat realised by the end of combustion. Two different methods were used based on the work of Rassweiler & Withrow.

The technique proposed by Rassweiler & Withrow (Rassweiler & Withrow, 1938) uses the fundamental heat release model to calculate mass fraction burned, where MFB can be defined as,

Equation 13 – MFB (Rassweiler & Withrow, 1938)

$$MFB = \frac{\sum_{SOC}^{\theta} dQ}{\sum_{SOC}^{EOC} dQ}$$

Considering the first law of thermodynamics,

$$dQ = dU + pdV$$

$$dU = m * C_v * dT$$

$$dQ = mC_v + pdV$$

Considering the ideal gas law,

$$pdV + Vdp = m * R * dT$$

Which can be seen as,

$$\frac{pdV + Vdp}{R}$$

If considering a constant volume combustion process, where $dV = 0$,

$$dQ = m * C_v * dT + pdV$$

$$dQ = m * C_v * dT$$

$$dQ = C_v \left(\frac{Vdp}{R} \right)$$

From these statements the mass fraction burned can be defined as,

$$MFB = \frac{\sum_{SOC}^{\theta} \left(C_v \left(\frac{Vdp}{R} \right) \right)}{\sum_{SOC}^{EOC} \left(C_v \left(\frac{Vdp}{R} \right) \right)}$$

Or,

$$MFB = \frac{\sum_{SOC}^{\theta} (C_v * Vdp)}{\sum_{SOC}^{EOC} (C_v * Vdp)}$$

The first method implemented involved the extra pressure due to combustion “ ΔP_{comb} ”, the extra pressure due to combustion can be considered proportionate to the heat added (Asgari, et al., 2012), expressed per degree of course,

Equation 14 - MFB Method 1 (P2n)

$$MFB = \frac{mfb_{\theta}}{mfb_{total}} = \frac{(\sum_{SOC}^{\theta} \Delta P_{comb})}{(\sum_{SOC}^{EOC} \Delta P_{comb})}$$

If the specific heat during combustion is considered constant, then MFB can be defined by,

Equation 15 - MFB Method 2 (P-Comb)

$$MFB = \frac{mfb_{\theta}}{mfb_{total}} = \frac{(\sum_{SOC}^{\theta} V_{cyl} \Delta P_{comb})}{(\sum_{SOC}^{EOC} V_{cyl} \Delta P_{comb})}$$

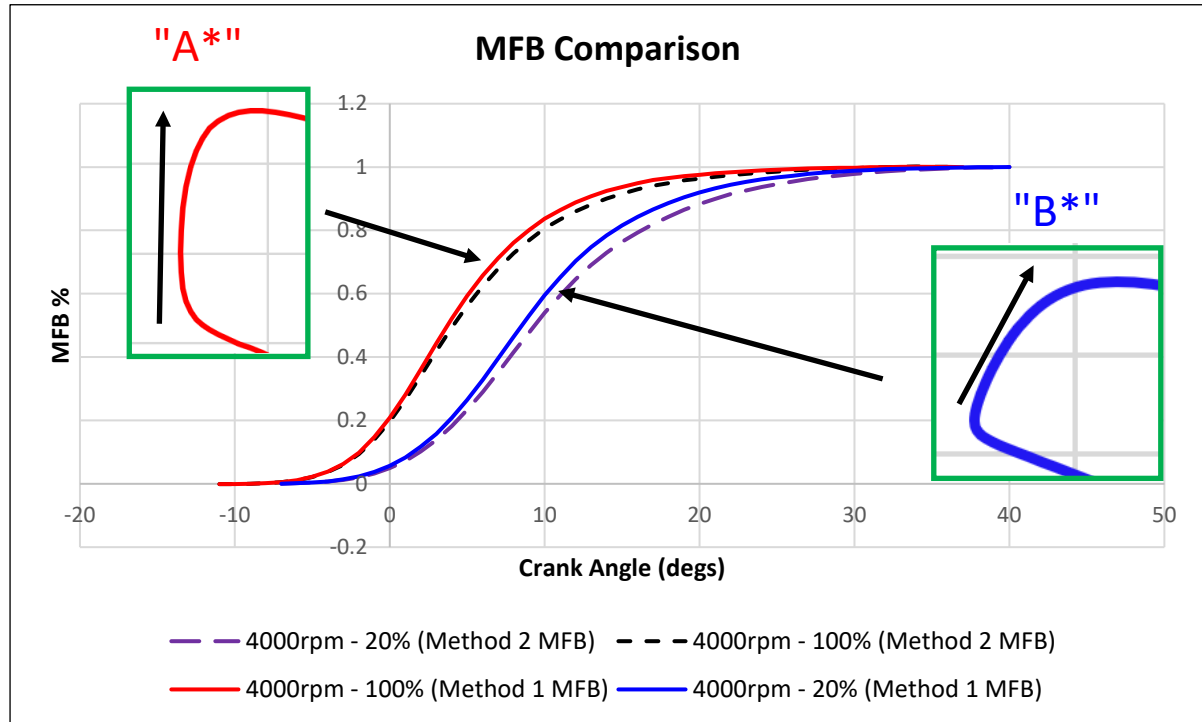


Figure 66 - MFB Method & Load Comparison (Vacas, Authors Own)

Figure 66 produces 2 interesting observations, in the first instance it is clear to see the MFB profile for the full load trace reaches MFB50% around 5 degrees earlier than the low load trace, implying that the peak pressure is closer to MBT. Figure 66 uses durations calculated with the pressure difference method, but when these values were calculated with the visual Log PV method high load duration was seen to be around 6 degrees less.

The second interesting observation is less obvious, method 2 relates to the 2nd MFB method where volume has been accounted for, whilst this doesn't make much difference to the high load trace as it is close to constant volume, it does make a more significant difference to the low load trace – this correlates with the fact that the low load Log PV diagram deviated significantly from the ideal constant volume combustion process, this is shown with "A*" and "B*" (the respective screen shots of the high and low Log PV graphs).

5.9.2 – Analysis

Earlier on the sensitivity of the polytropic index had been evaluated in terms of what effect it had on the pressure difference calculation and whilst it did have an effect (when doubling “n”), this effect only really materialised toward the very beginning and end of combustion. Figure 67 though shows the effects of “n” variation on the MFB curve, the MFB curve was plotted with extremes of (+/-) 10% & 30%. It was clear that these changes in “n” had a significant effect of the MFB curve, whilst this doesn’t have a great effect on the MFB50% value (considering + / - 10%) it does significantly change the curves shape. This was an important parameter to be aware of as MFB values are the precursor to the Vibe constants “a” and “m” being calculated – KTM’s quoted values “n” values are out by roughly 4% which can be considered somewhat significant in light of Figure 67.

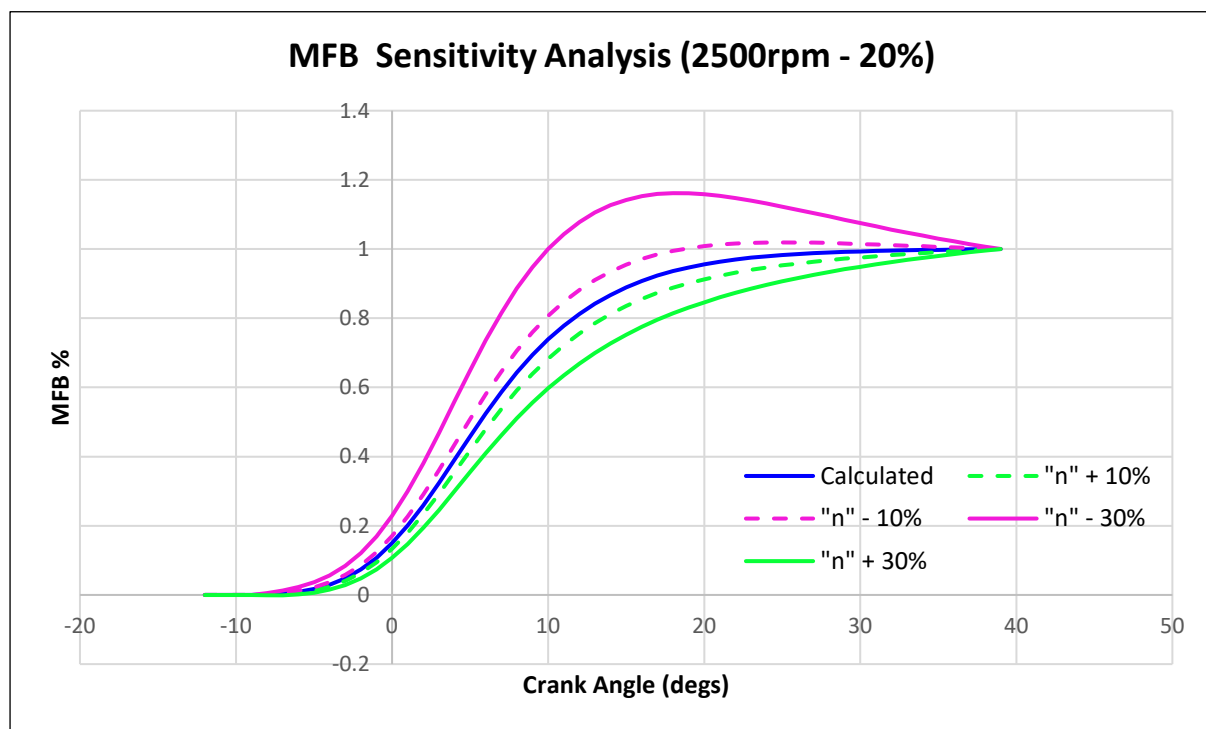


Figure 67 - MFB Sensitivity Analysis (Vacas, Authors Own)

Another step taken to analyse the MFB plots was to overlay an MFB plot and a pressure difference plot – where combustion starts should be where the pressure difference begins and subsequently where combustion ends should be where the pressure difference ends. Figure 68 shows this clearly, the correlation adds an additional sense of affirmation to the overall model’s continuity – if anything conducting simple tests like this may just be a useful tool for spotting simple errors when handling data.

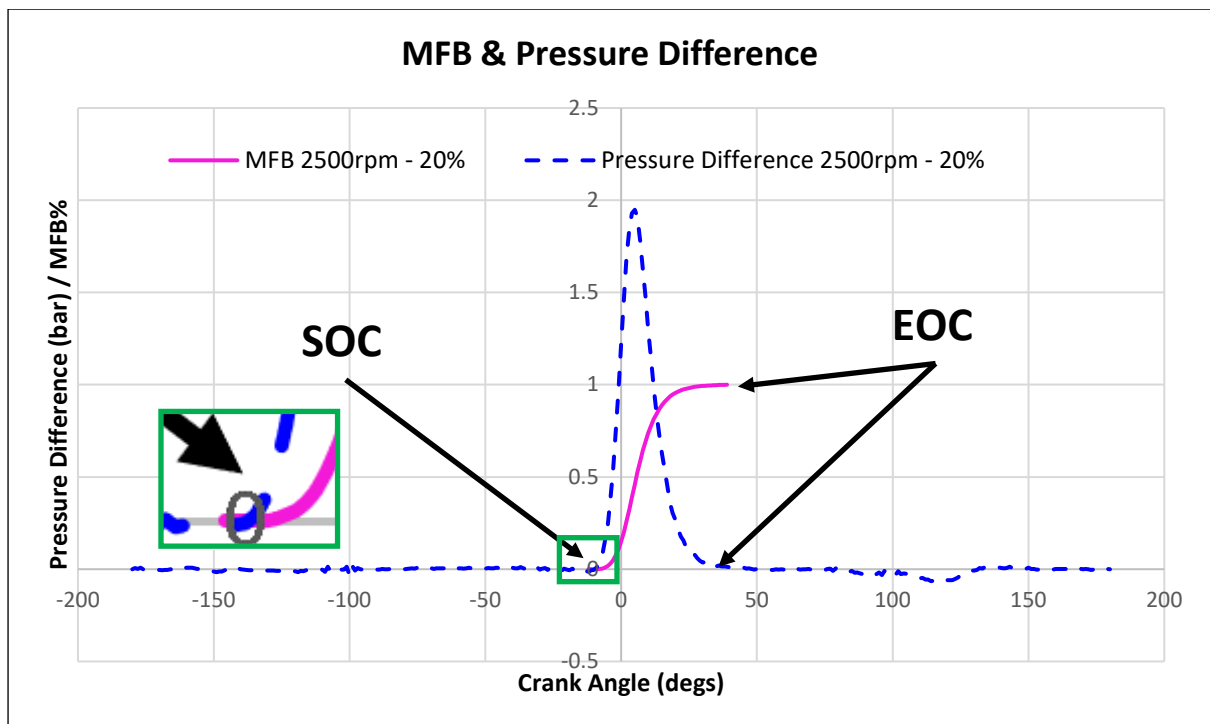


Figure 68 - MFB & Pressure Difference (Vacas, Authors Own)

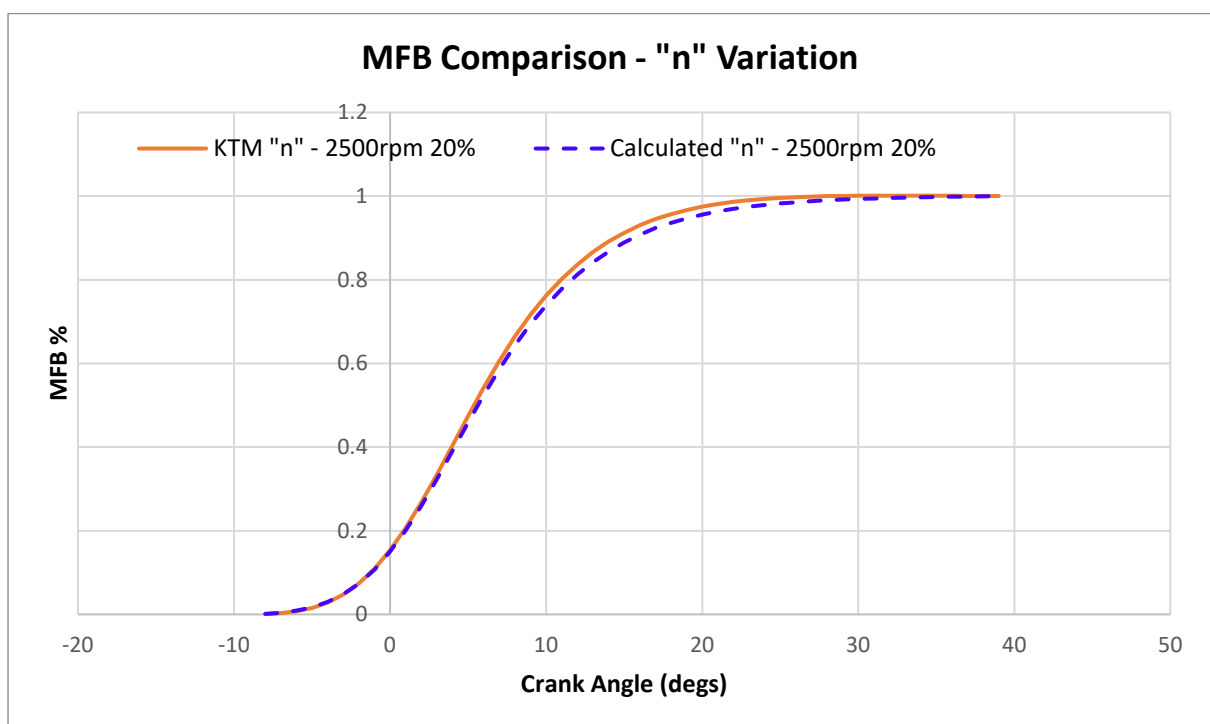


Figure 69 - MFB Comparison "n" Variation (Vacas, Authors Own)

Figure 69 shows the difference that the KTM “n” values have on an MFB curve at 2500rpm & 20%, visually it is not that big a difference and acts as if to say the combustion is happening slightly faster.

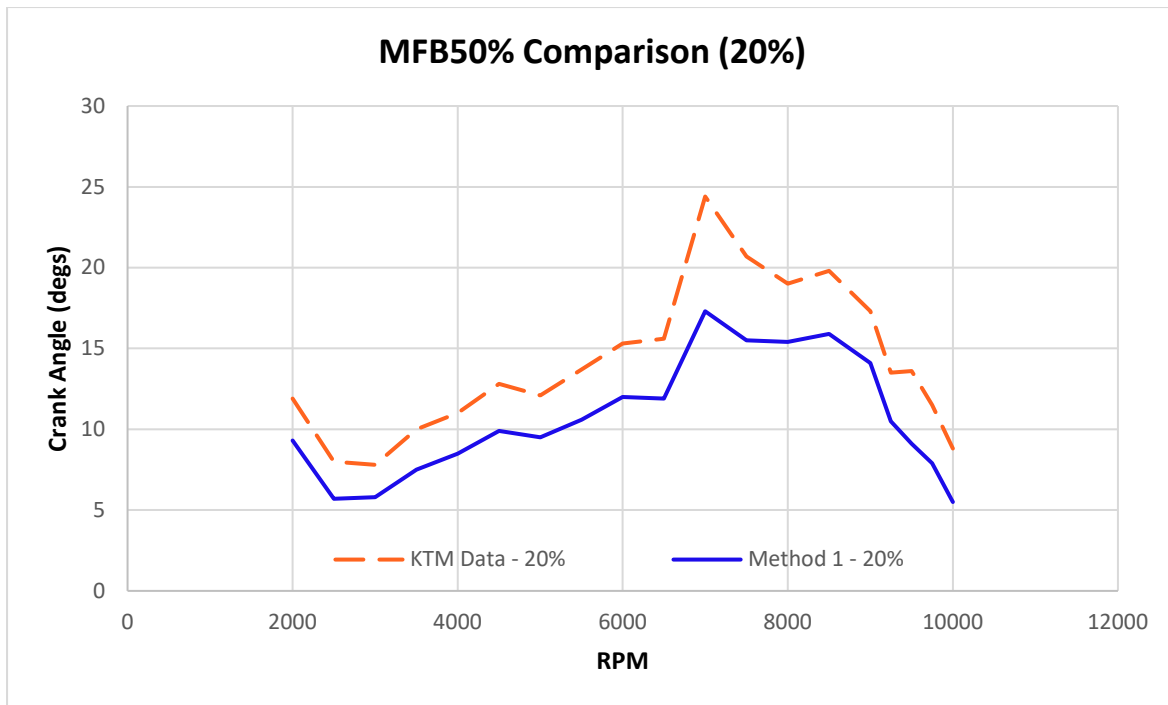


Figure 69 - MFB50% Comparison (20%)

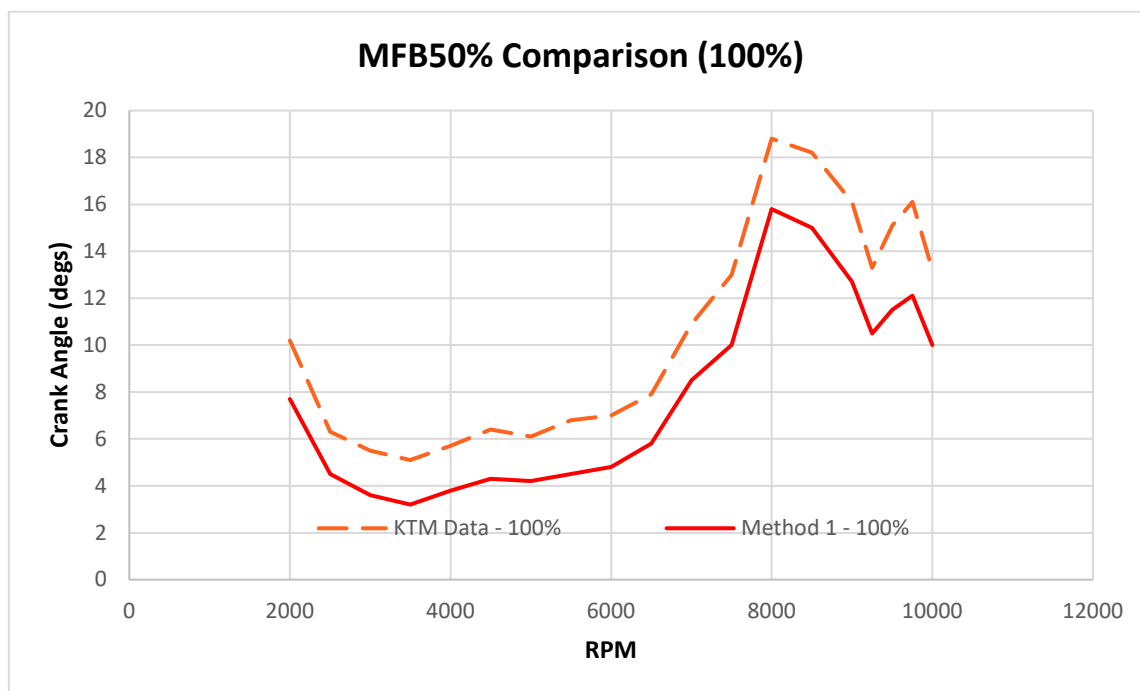


Figure 70 – MFB50% Comparison (100%) (Vacas, Authors Own)

Figures 70 and 71 show the comparison between KTM's MFB50% figures and those calculated with MFB method 1, for low and high load respectively – both comparisons show a clear relationship in terms of having very similar trends but there seems to be a consistent difference of around 2 degrees, although not displayed this does not change much when considering MFB method 2.

5.10 – Vibe Function

The engine simulation model that was being built required the vibe constants of “a” and “m” as it operated on a prescribed burn rate model.

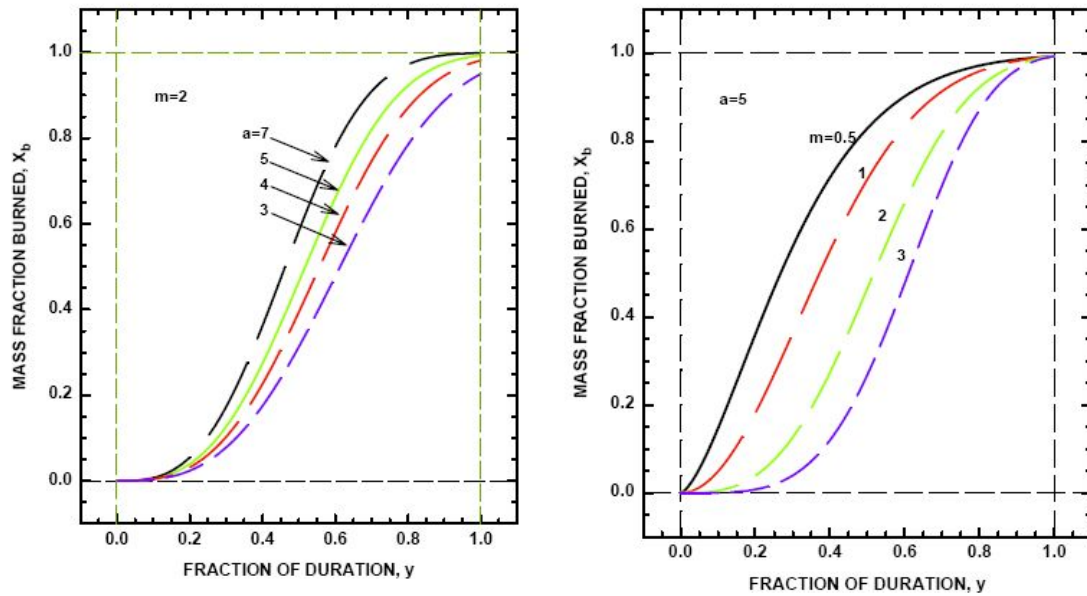


Figure 70 - Vibe Constants (Nierkerk, 2018)

Figure 70 (Nierkerk, 2018) shows two examples of variation in the vibe constants of “m” and “a”, these two constants can be described simply by the following 2 statements (Heywood & Chun, 1987);

- The vibe constant “a” can be considered the slope shape factor, increasing this constant increases the burn rate slope – also referred to as the efficiency parameter.
- The vibe constant “m” can be considered the distribution shape factor, increasing this constant results in more of the mass being burnt later – also referred to as the form factor.

The prescribed burnt rate can be described by,

Equation 16 - MFB Prescribed Burn Rate (van Nierkerk, 2000)

$$MFB = 1 - \exp\left(-a\left(\frac{\theta}{\theta_d}\right)^{m+1}\right)$$

5.10.1 – Method & Calculations

The prescribed burn rate can be re-arranged,

$$\exp\left(-a\left(\frac{\theta}{\theta_d}\right)^{m+1}\right) = 1 - MFB$$

$$\ln\left(\exp\left(-a\left(\frac{\theta}{\theta_d}\right)^{m+1}\right)\right) = \ln(1 - MFB)$$

$$-a\left(\frac{\theta}{\theta_d}\right)^{m+1} = \ln(1 - MFB)$$

$$a\left(\frac{\theta}{\theta_d}\right)^{m+1} = -\ln(1 - MFB)$$

$$\ln\left(a\left(\frac{\theta}{\theta_d}\right)^{m+1}\right) = \ln(-\ln(1 - MFB))$$

$$\ln a + \ln\left(\frac{\theta}{\theta_d}\right)^{m+1} = \ln(-\ln(1 - MFB))$$

$$\ln a + (m + 1) \ln\left(\frac{\theta}{\theta_d}\right) = \ln(-\ln(1 - MFB))$$

This can be now viewed in its final form as the equation of a straight line,

Equation 17 - Vibe Graph Equation

$$\ln(-\ln(1 - MFB)) = (m + 1) \ln\left(\frac{\theta}{\theta_d}\right) + \ln a$$

$$Y = MX + C$$

5.10.2 – Graphing & Analysis

With the prescribed burn rate now in the form of a straight line it could be graphed to calculate the vibe constants “a” and “m”.

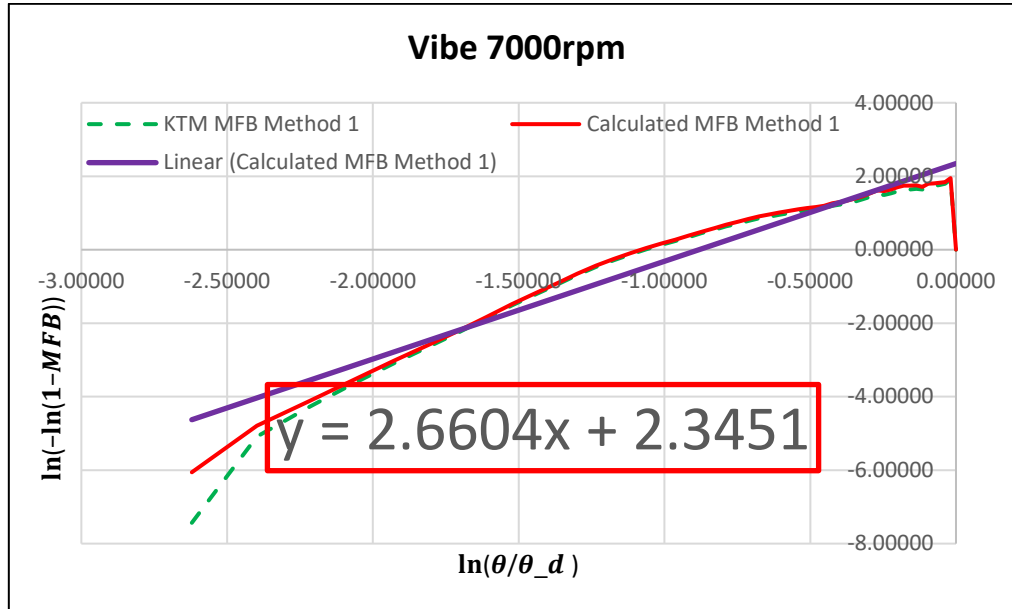


Figure 71 - Vibe Graph (Vacac, Authors Own)

Figure 71 shows a typical graphed trace, to successfully calculate the vibe constants it was necessary fit a linear trend line to the data trace, the equation of which is displayed on the graph. From this equation “a” can be calculated as the exponent of “2.3451” and “m” as “2.6604-1”, in this case “a=10.43” and “m=1.66”, whilst slightly high these values are within the typical range shown by (Blair, 1994).

It can be noted that figure 71 represents the rather ideal scenario where there is little or no noise, several other graphs did contain a certain amount of noise meaning that the range selection had to be shortened or data points taken out – this was avoided wherever possible. It can also be noted that in general the lines graphed using the KTM “n” values tended to degrade or distort towards the extremities of the line.

5.10.3 – Function Retro-Fit

To double check this process it was decided to retro-fit the “a” and “m” values directly back to the prescribed burn rate equation,

Equation 18 - MFB prescribed Burn Rate

$$MFB = 1 - \exp\left(-a\left(\frac{\theta}{\theta_d}\right)^{m+1}\right)$$

Figures 72 and 73 show plots comparing the original KTM values compared with both the initially calculated MFB50% (method 1, 1^2) values and those mfb curve generated from the function retro-fit. The retro-fit values for the high load data went from undershooting the KTM values to matching or overshooting them (whilst generally follow a similar trend – of interest was the low load retro-fit data which up to 6500rpm was an extremely close match to the original KTM data. On a single case basis, it would be naïve to draw a conclusion from this, but it is certainly a future area that could be investigated.

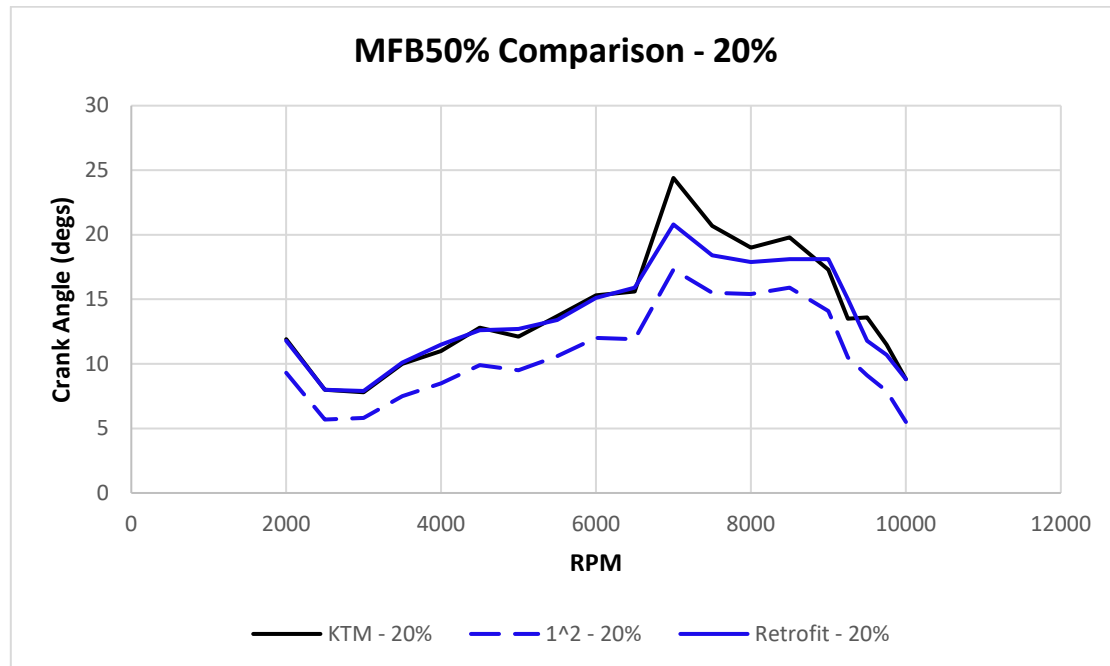


Figure 72 - MFB Retro-Fit 20%

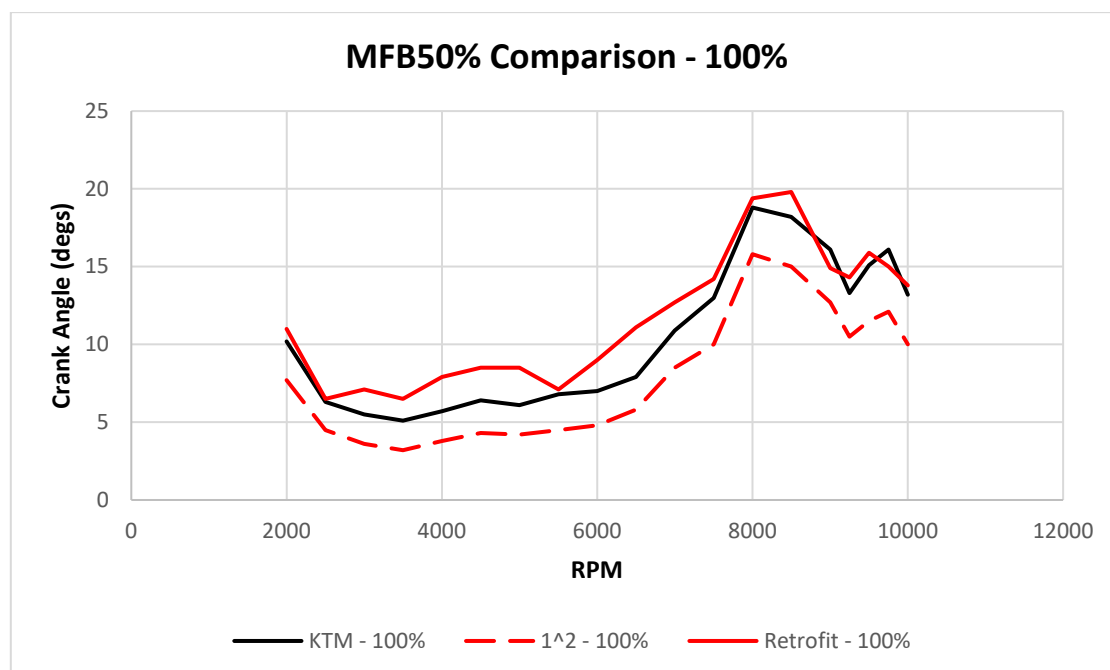


Figure 73 - MFB Retro-Fit 100%

Chapter 6 - Excel Model (Combustion Chemistry)

Having explored the fundamental mechanisms relating to the engine pressure data the next step was to explore the combustion chemistry.

6.1 – Chemistry Basics

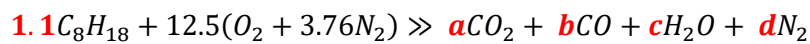
The very first step was to cover the topic of combustion stoichiometry.

Where the fuel equivalence ratio can be stated as the inverse of lambda,

$$\phi = \frac{1}{\lambda}$$

Showing the example condition of 10% rich:

$$\phi = \mathbf{1.1} \text{ (10\% rich)}$$



$$8.8 = \mathbf{a} + \mathbf{b}$$

$$19.8 = \mathbf{2c}$$

$$9.9 = \mathbf{c}$$

$$25 = \mathbf{2a} + \mathbf{b} + \mathbf{c}$$

$$25 = \mathbf{2a} + \mathbf{b} + 9.9$$

$$14.9 = \mathbf{2a} + \mathbf{b}$$

$$(\mathbf{2a} + \mathbf{b}) - (\mathbf{a} + \mathbf{b})$$

$$\mathbf{a} + \mathbf{0} = 4.7$$

$$\mathbf{b} = 8.8 - \mathbf{a}$$

$$\mathbf{b} = 4.1$$

6.2 –Mass Calculations and Purity

The next logical step was to calculate the various in cylinder masses.

Fuel mass “ f_c ” was given in KG/hr, hence fuel mass KG/rev could be given as,

Equation 19 - Fuel Mass Per Revolution

$$m_f = \left(\frac{f_c}{60N} \right)$$

Subsequently the air mass “ m_a ” could be given by,

Equation 20 - Air Mass Per Revolution

$$m_a = \frac{(m_f AFR_{stoich})}{(\phi)}$$

The air and fuel mass could be described as,

Equation 21 - Air & Fuel Mass Per Revolution

$$m_{af} = m_a + m_f$$

Having calculated figures for the air and fuel mass the mass of residual gas could be calculated (based on assuming a purity figure) – Purity values were obtained from Engmod2T or (Zhao, 2007)

Charge purity can be defined as,

Equation 22 - Charge Purity (Blair, 1994)

$$\Pi_{charge} = \frac{m_{ta}}{m_{tr}}$$

(Blair, 1994) defines “ m_{ta} ” as the mass of air trapped in the cylinder, but the assumption is made that,

$$m_{ta} = m_a + m_f$$

Concerning the mass of trapped charge “ m_{tr} ”, it can be stated that,

$$m_{tr} = m_a + m_f + m_{rg}$$

Where “ m_{rg} ” is the mass of residual gas.

Therefore,

$$m_{tr} = \frac{m_{ta}}{\Pi_{charge}}$$

Which becomes,

$$m_a + m_f + m_{rg} = \frac{m_{ta}}{\Pi_{charge}}$$

Defining residual gas “ m_{rg} ” as,

Equation 23 - Residual Gas Mass

$$m_{rg} = \left(\frac{m_{ta}}{\Pi} \right) - (m_a + m_f)$$

Figure 74 shows the various in cylinder masses with a nominal charge purity of 0.95.

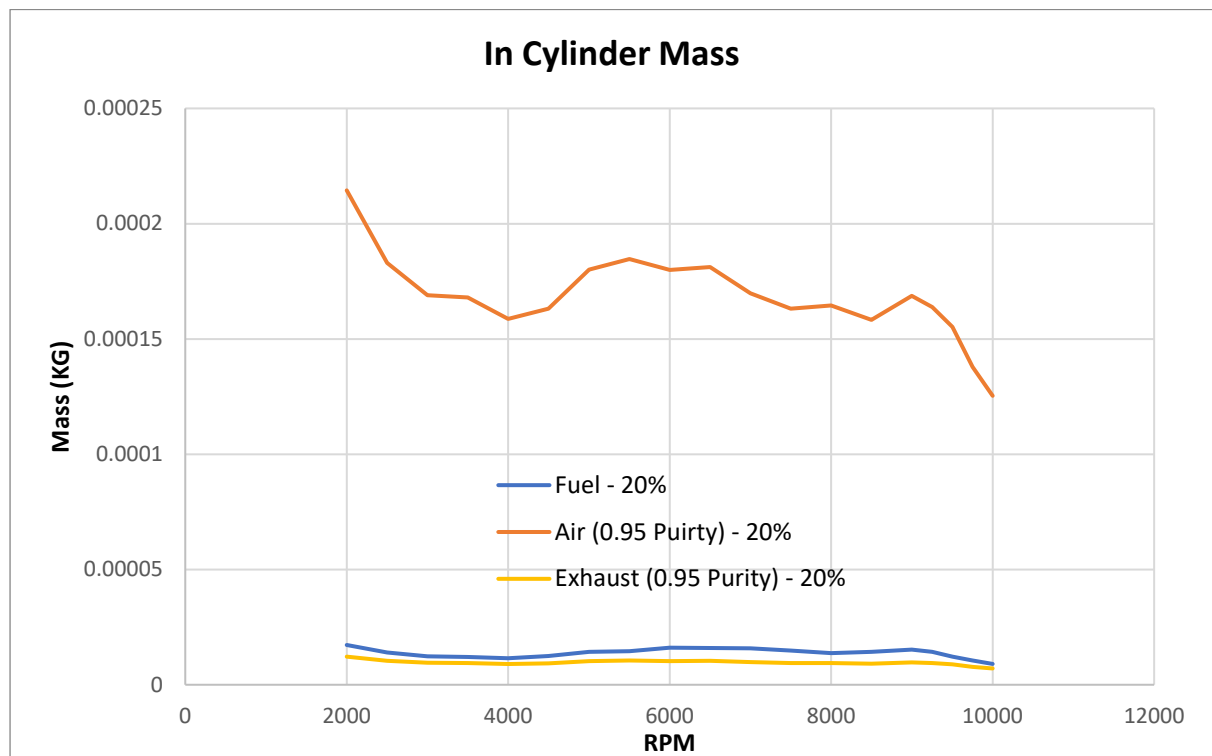


Figure 74 - In Cylinder Mass (Vacas, Authors Own)

6.3 – Specific Gas Constants

The ideal gas law states,

$$PV = n^* R_u T$$

$$n^* = \text{number of moles}$$

$$R_u = \text{Universal Gas Constant J/KmolK}$$

Or,

$$PV = mRT$$

$$m = \text{mass in KG}$$

$$R = \text{Specific Gas Constant Per KG – Dependant upon gas}$$

And,

$$PV = n^* R_u = mR$$

If,

$$n^* = 1 \text{Kmol}$$

It can be said that,

$$M = \text{Molecular weight in KG}$$

Therefore,

$$1R_u = MW(R)$$

Equation 24 - Specific Gas Constant

$$R = \frac{R_u}{MW}$$

$$R = \text{Specific Gas Constant}$$

Prior to calculating the specific gas constants, it was necessary to calculate the average molar weights of the mixture and exhaust – this can be stated simply as,

Equation 25 - Average Molecular Weight

$$\frac{\text{Molar Weight}}{\text{Total Number of Moles}}$$

Hence for the mixture this could be given as,

$$MW_{mix_{avg}} = \frac{MW_{mix}}{mn_{mix}}$$

Equally for the exhaust,

$$MW_{exh_{avg}} = \frac{MW_{exh}}{mn_{exh}}$$

Considering residual gas content, the specific gas constant prior to combustion can be defined as,

Equation 26 - Specific Gas Constant (unburnt)

$$R_{unb} = \Pi_{charge} \left(\frac{R_u}{MW_{mix_{avg}}} \right) + \Pi_{exh} \left(\frac{R_u}{MW_{exh_{avg}}} \right)$$

Where,

$$\Pi_{exh} = 1 - \Pi_{charge}$$

And the universal gas constant,

$$R_u = 8314 \text{ J/K} \cdot \text{K}$$

Subsequently the burnt gas constant can be described as,

$$R_{bun} = \frac{R_u}{MW_{exh_{avg}}}$$

And the average specific gas constant,

$$R_{avg} = \frac{1}{2} \left(\frac{R_u}{MW_{unb} + MW_{bun}} \right)$$

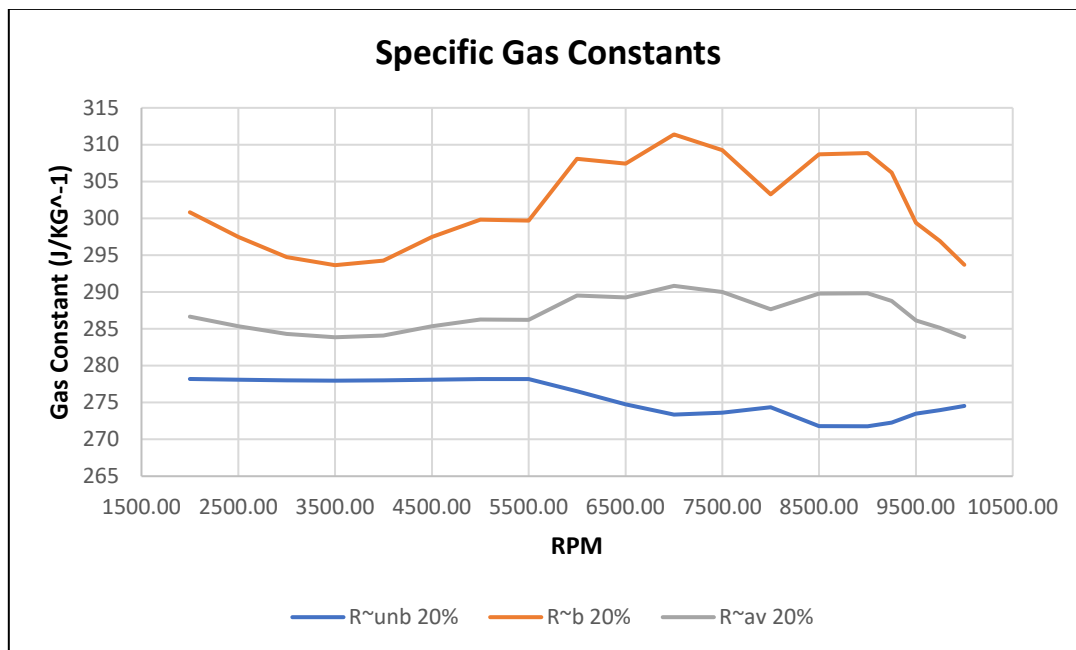


Figure 75 - Specific Gas Constants (Vacas, Authors Own)

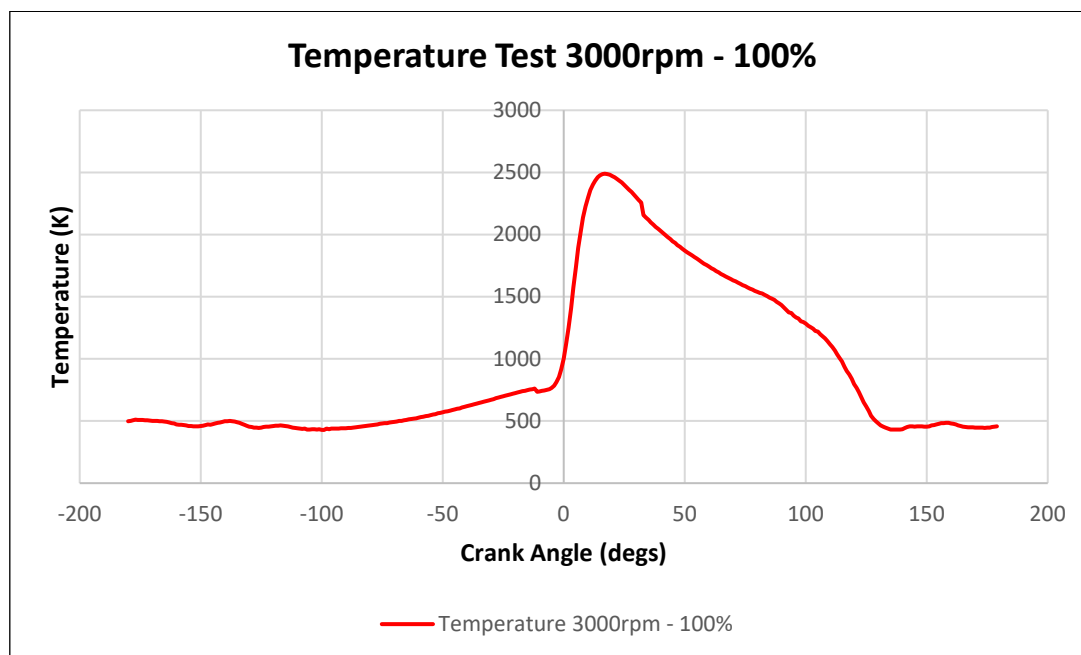


Figure 76 - Temperature Test 3000rpm - 100% (Vacas, Authors Own)

Having calculated values for mass and values for the specific gas constants it was possible to make an initial calculation of temperature.

The equation used for temperature, as derived from the ideal gas law can be stated as,

Equation 27 - Initial Temperature Calculation

$$T = \frac{PV}{mR}$$

6.4 – Specific Heats

The next step was to calculate the specific heats of both the mixture and exhaust, Heywood (B.Heywood, 1988) states,

Equation 28 – (Heywood 1988)

$$\frac{\tilde{C}_{p,i}}{R_U} = a_{i1} + a_{i2}T + a_{i3}T^2 + a_{i4}T^3 + a_{i5}T^4$$

Where “ $\tilde{C}_{p,i}$ ” is the specific heat of a species at constant pressure, with the units “cal/gmol.K”, “ a_i ” is a polynomial coefficient from the JANAF NASA tables, “ T ” is temperature and “ R_u ” the universal gas constant.

Re-arranging this equation gives,

Equation 29 - Specific Heat at Constant Pressure (Heywood 1988)

$$\tilde{C}_{p,i} = R_u(a_{i1} + a_{i2}T + a_{i3}T^2 + a_{i4}T^3 + a_{i5}T^4)$$

And for fuels in a vapour phase,

Equation 30 - Specific Heat at Constant Pressure (fuel) (Heywood 1988)

$$\tilde{C}_{p,f} = A_{f1} + A_{f2}t + A_{f3}t^2 + A_{f4}t^3 + \frac{A_{f5}}{t^2}$$

Where,

$$t = \frac{T}{1000}$$

Converting the units of “cal/K.mol.k” for fuel to “J/K.mol.k”, where “4180” is the conversion factor can be stated as,

$$C_{p,f} = 4180 * \tilde{C}_{p,f}$$

Therefore, converting to constant volume can be shown by,

$$C_{p,f} - R_u = \tilde{C}_{v,f}$$

Converting any of the mixture species to specific heat at constant volume can be shown by,

$$\tilde{C}_{p,i} - R_u = \tilde{C}_{v,i}$$

Hence the average specific heat (J/Kmol.k) at constant volume of the mixture can be stated by,

$$\tilde{C}_{vAV_{mix}} = \frac{((mn_{C_8H_{18}}\tilde{C}_{v,C_8H_{18}}) + (12.5(1\tilde{C}_{v,O_2} + (3.76\tilde{C}_{v,N_2}))))}{(mn_{C_8H_{18}} + (12.5(1 + 3.76)))}$$

Converting to J/KG.K,

Equation 31 - Average Specific Heat at Constant Volume (Mixture) J/KG.K

$$C_{vAV_{mix}} = \frac{\tilde{C}_{vAV_{mix}}}{MW_{mix_{avg}}}$$

Where “ mn ” is the molecular count and “ $MW_{mix\,avg}$ ” the average molecular weight of the mixture.

The average specific heat (J/Kmol.K) at constant volume of the exhaust can be stated by,

$$\tilde{C}_{vAV_{exh}} = \left(\frac{(mn_{CO_2} \tilde{C}_{v,CO_2}) + (mn_{CO} \tilde{C}_{v,CO}) + (mn_{H_2O} \tilde{C}_{v,H_2O}) + (mn_{N_2} \tilde{C}_{v,N_2})}{(mn_{CO_2} + mn_{CO} + mn_{H_2O} + mn_{N_2})} \right)$$

Converting to (J/KG.K),

Equation 32 - Average Specific Heat at Constant Volume (Exhaust) J/KG.K

$$C_{vAV_{exh}} = \frac{\tilde{C}_{vAV_{exh}}}{MW_{exh\,avg}}$$

Chapter 7 – Auto-Ignition Calculations

The aim of this chapter is to bring together the most directly related tangential precursors to calculating the possibility of auto-ignition

The values for purity originates from the work undertaken in the software package “Engmod2T” (shown in appendix “A”) – this value allows the calculation of a theoretical exhaust gas mass and subsequently a temperature EPC. Further to this, calculations are made identify the temperature the charge can reach before TDC, culminating in an appraisal for the possibility of auto-ignition – a short chapter, but the culmination of a far larger body of work. Purity values have not been conclusively validated, but (Zhao, 2007) states how residual gas in two-strokes can be over 80% - values are within this boundary.

7.1 – Purity & Mass Comparison

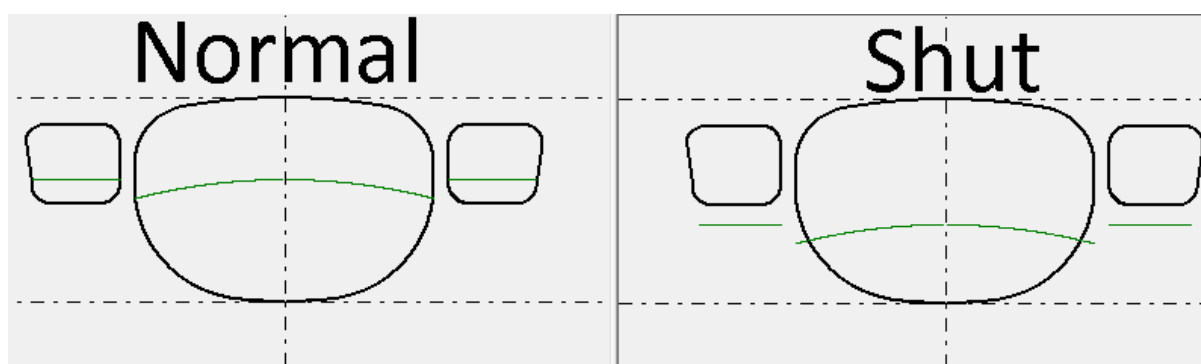


Figure 77 - Exhaust Valve Positions Engmod2T (Vacas, Authors Own)

Figure 77 shows the two positions in which the purity (and thus residual gas mass) were calculated, with “normal” being a close approximation of the standard valve position and “shut” lowering to the point that it intersects the upper roof height of the transfer port (specified by Honda a 2/3rds shut rule) (Honda, 1997).

Figure 78 shows a graph detailing purity for low / high load, and subsequently the respective values of purity for both “normal” and “shut” exhaust valve positions. The graph makes it very clear just how much residual gas there could be when the in the shut position at 20% throttle, note; purity rises rapidly after 5500rpm as this is the point at which the exhaust valve tends towards fully open – it is also useful to add that a purity of “1” signifies nothing but fresh charge in the cylinder and a purity value of “0” signifies nothing but residual gas in the cylinder, both at EPC.

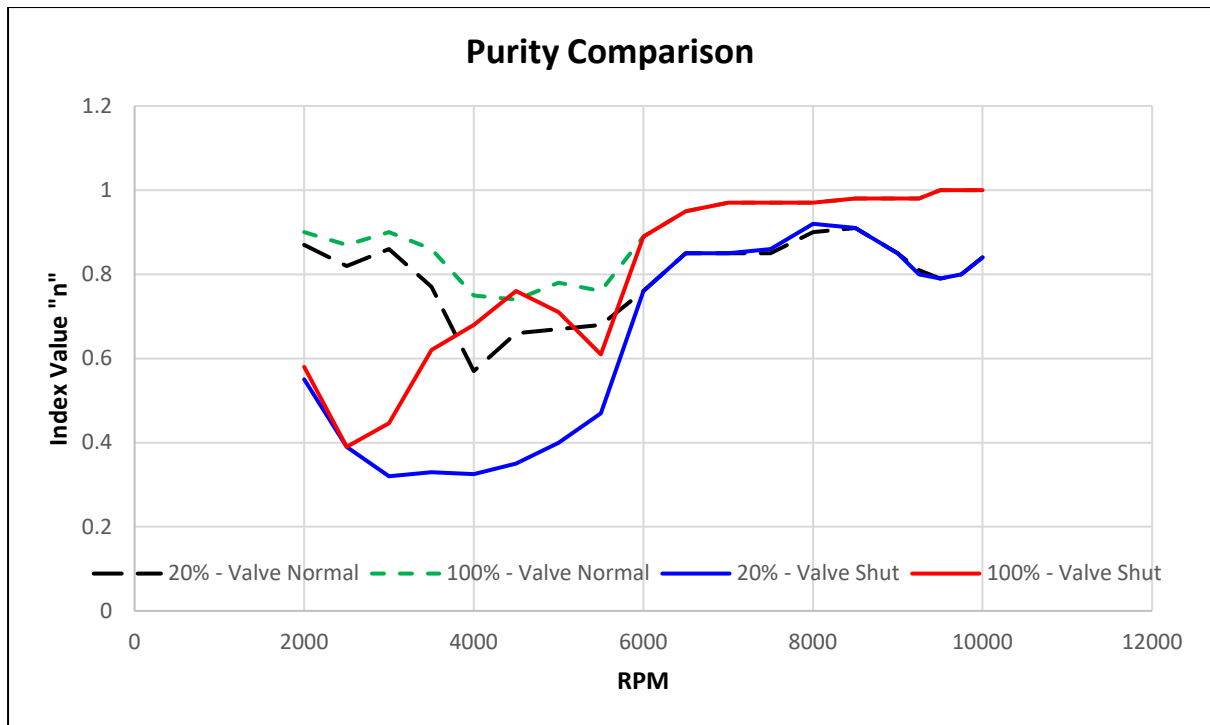


Figure 78 - Purity Comparison (Vacas, Authors Own)

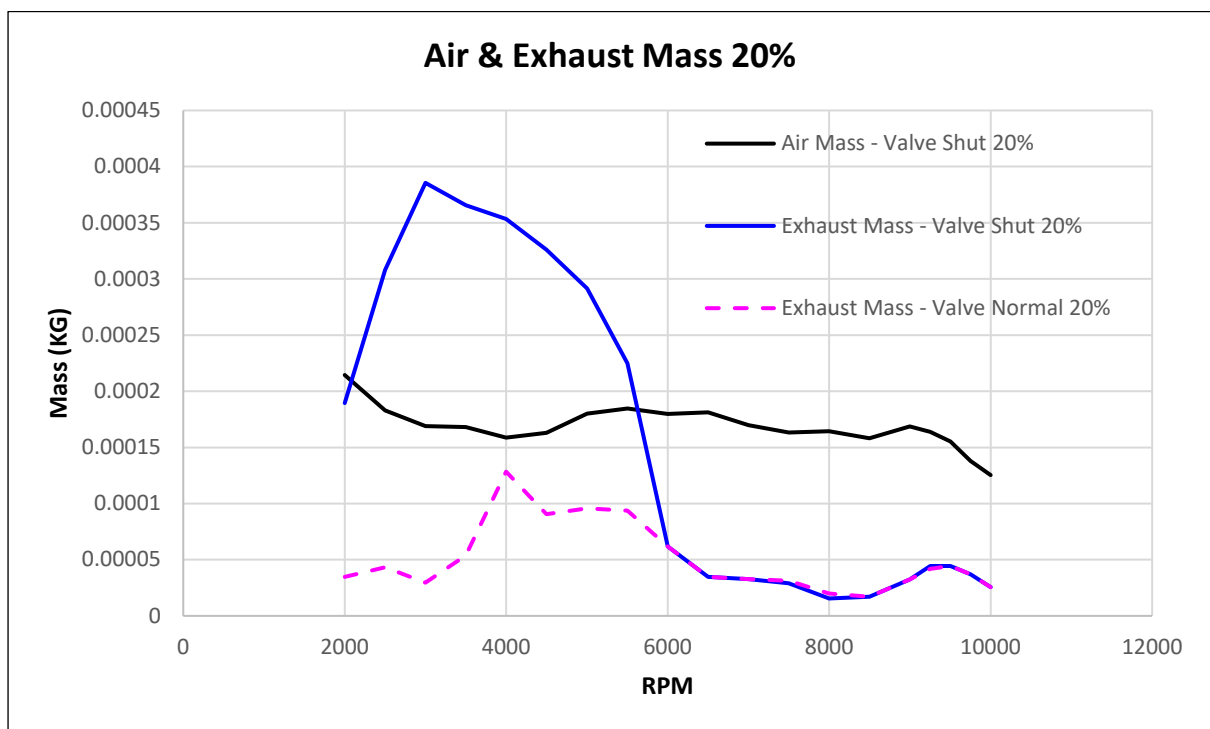


Figure 79 - Air & Exhaust Mass (Vacas, Authors Own)

Figure 79 shows the air & exhaust masses for low load (20%) and can be considered in ways an inverse representation of Figure 78. Apparent is the huge increase in exhaust gas mass, in most cases being double or more – note; air mass stays constant as this value has been calculated from the original KTM fuel and lambda values

7.2 – Temperature at Exhaust Port Closure

Having earlier established the average specific heats at constant volume, the heat energy needed for the mixture to rise 1K could be defined by,

Equation 33 - Heat Energy needed for 1K Temperature Rise

$$Q_{mix} = \left(\frac{C_{vAV_{mix}}}{2} \right) (m_a + m_f)(1)$$

Considering a theoretical target temperature “X”, the heat needed to reach “X” can be defined by,

Equation 34 - Heat Energy Needed to Reach "X" (Mixture)

$$\sum_{T_{mix}}^X Q_{mix}$$

“ T_{mix} ” can be considered the mixture temperature, which has been taken to be 333k (60°C) – roughly the crankcase temperature as supplied by KTM.

Having obtained purity values and thus residual gas mass, heat energy can be calculated needed for the exhaust gas / residual gas to drop 1K,

Equation 35 - Heat Energy Lost Dropping 1K (Exhaust)

$$Q_{exh} = \left(\frac{C_{vAV_{exh}}}{2} \right) (m_{rg})(1)$$

Considering the exhaust gas is dropping to a theoretical target “X”, the heat energy needed can be given as,

Equation 36 - Heat Needed to Reach "X" (Exhaust)

$$\sum_{T_{exh}}^X Q_{exh}$$

With both sets of heat energy defined it can be stated that when,

Equation 37 - Heat Energy Balance

$$\left(\sum_{T_{mix}}^X Q_{mix} \right) - \left(\sum_{T_{exh}}^X Q_{exh} \right) = 0$$

“ T_{epc} ”, the cylinder temperature at exhaust port close is equal to “X”, i.e. the temperature of equilibrium between the two mixtures,

$$T_{epc} = X$$

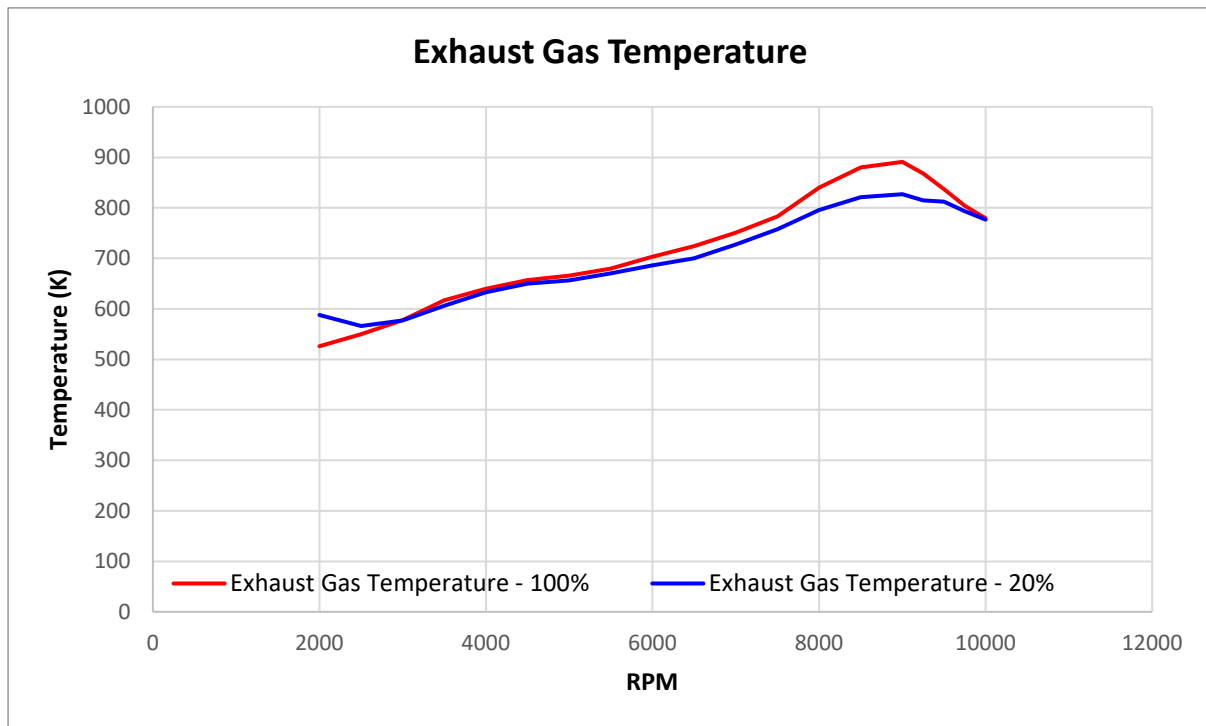


Figure 80 - Exhaust Gas Temperatures (KTM, 2017)

Figure 80 shows a graph of the KTM supplied exhaust gas temperatures, these values will be used for “ T_{exh} ”, subsequently for equation “ T_{mix} ” has been defined as 333K approximately the average crankcase temperature (KTM supplied).

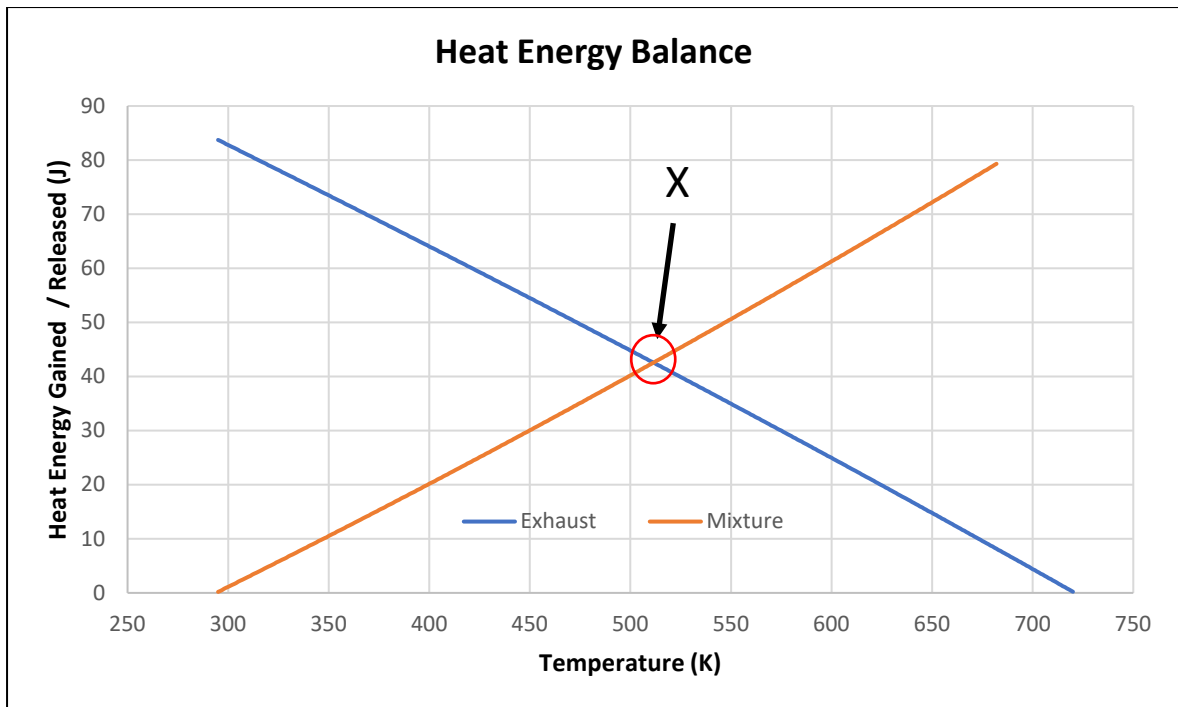


Figure 81 - Heat Energy Balance (Vacas, Authors Own)

Figure 81 is a simple graph illustrating the concept the heat balance equation (36), "X" is identified as the point at which the two heat energy traces cross and subsequently the temperature at this point is considered " T_{epc} ".

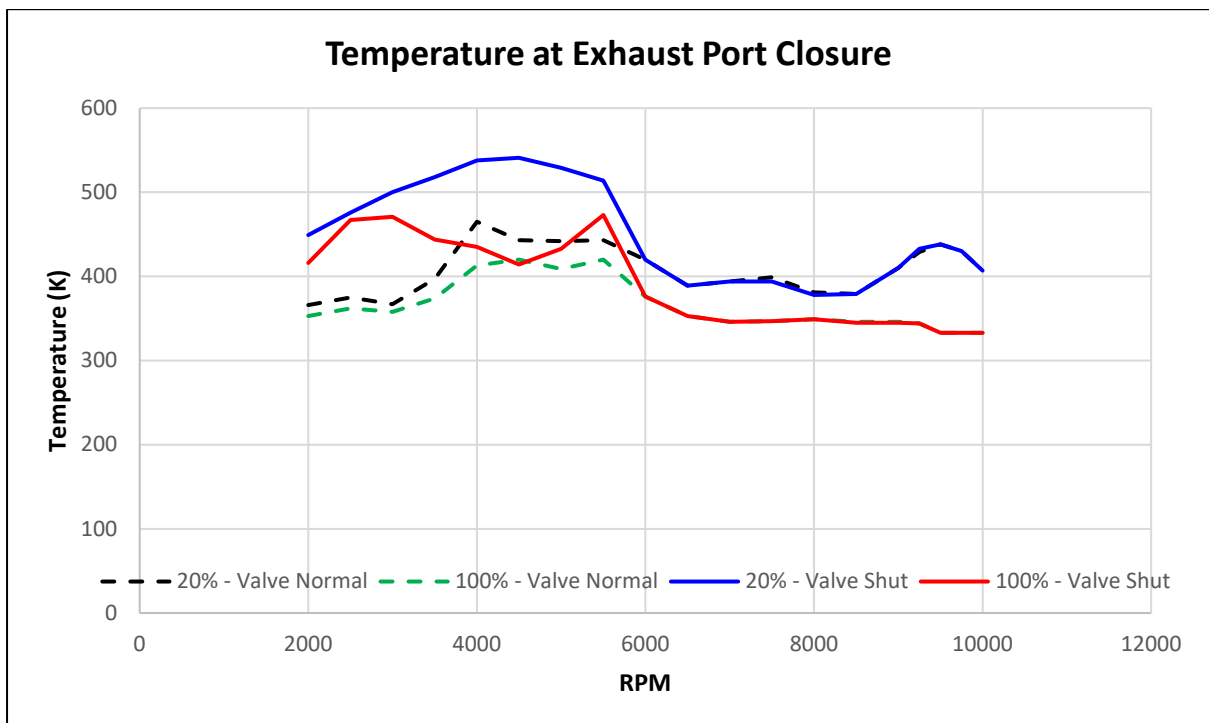


Figure 82 - Temperature at Exhaust Port Closure (Vacas, Authors Own)

Figure 82 shows the calculated " T_{epc} " temperatures for both loads and subsequently both exhaust valve positions, again 20% load shows potential for possible auto-ignition.

7.3 – Auto-Ignition Temperature Calculation Methodology

In the technical paper “Basic Understanding of Activated Radical Combustion and Its Two-Stroke Engine Application and Benefits” (Ishibashi, 2000) states;

“Auto-ignition temperature (T_{θ^*}) is given by the following polytropic change formula:

$$T_{\theta^*} = T_i(\epsilon^*)^{n-1}$$

where,

ϵ^* is the specific gas volume at auto-ignition timing (θ^*). n is the mean polytropic index measured from the indicator diagram in the compression process. The crank angle at 5% mass burn was used for the θ^* , however, the estimation results were not very much affected by the crank angle around the top dead center.”

Citing Hondas above statement concerning the calculation of the autoignition temperature equation 37 was applied on a trailing per degree basis starting from the exhaust port closure point.

Equation 38 - Auto-Ignition Temperature (Vacas, Authors Own)

$$T_2 = T_1 \left(\frac{V_1}{V_2} \right)^{n-1}$$

Importantly the three different exhaust port closure points were defined as follows,

- **1:** $106^\circ \rightarrow 5^\circ$ BTDC – approximately the height of the exhaust valve in its standard lower position (up to 5500rpm).
- **2:** $87^\circ \rightarrow 5^\circ$ BTDC – approximately the height of the exhaust valve when fully open – as a simplification for calculations the exhaust valve is considered fully open from .5500rpm
- **3:** $121^\circ \rightarrow 5^\circ$ BTDC – approximately the height of the exhaust valve if considered to be at the upper height of the transfer ports, an approximation for auto-ignition suggested by the Honda Press Fact Book (Honda, 1997)

7.4 – Polytropic Index (Expanded Range Selection)

Importantly a point discussed by Ishibashi (Ishibashi, 2000), is that when calculating auto-ignition temperature “a mean value of n from the indicator diagram in compression should be used”. The work conducted in chapter 5.7 discussed various methods for calculating n including looking at acute rate selection in compression (prior to SOC) - but it was also clearly noted that there was a trend to the n values, in that at exhaust port closure they started higher and tended down towards SOC. For this reason, it was decided to look at an expanded range selection for compression – selecting a range slightly after EPC and extending this range close to SOC.

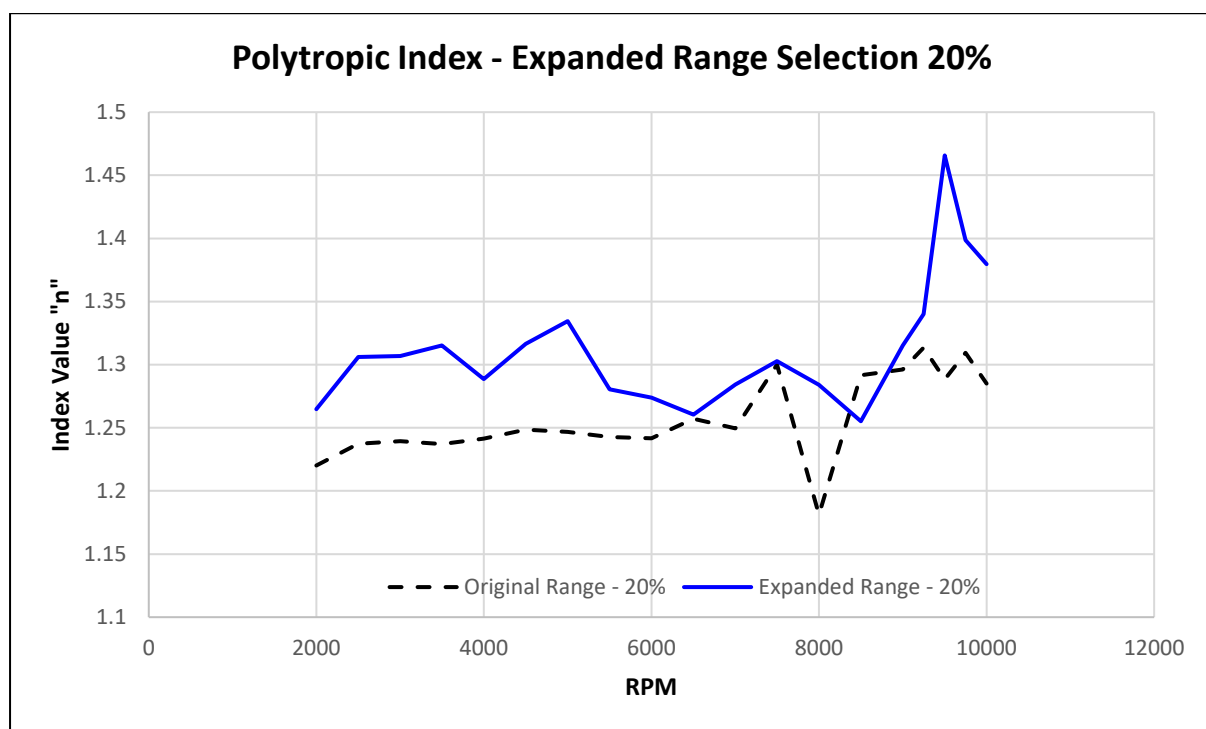


Figure 83 - Polytropic Index (compression) Expanded Range Selection - 20% (Vacas, Authors Own)

Figures 83 & 84 (low and high load) show an apparent and interesting trend, these two graphs plot the “expanded” range selection against the original “acute” range selection values and it becomes apparent that using this new technique brings a consistently higher value of n in compression all around 5500rpm (the point at which the exhaust valve starts to open).

With this information in mind it was decided to conduct a quick analysis of the temperature equations sensitivity to values of n , this comparison is shown in Figure 85 “Temperature - n ” Variation Sensitivity (Vacas, Authors Own)”.

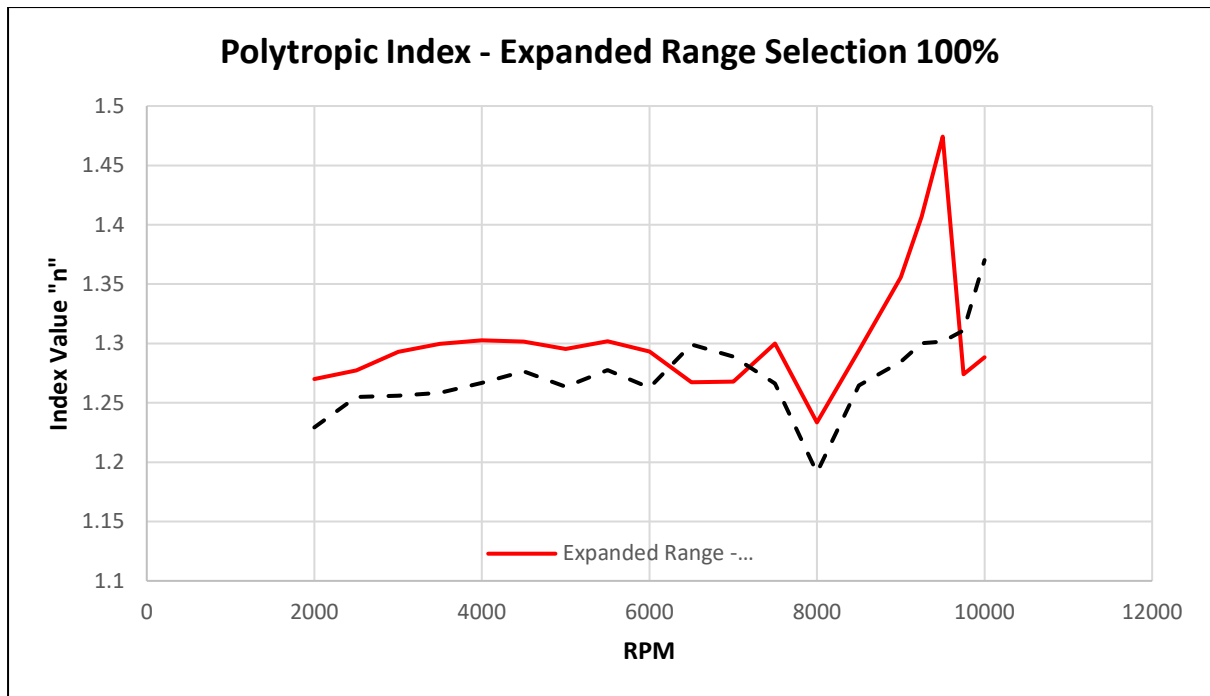


Figure 84 - Polytropic Index (compression) Expanded Range Selection - 100% (Vacas, Authors Own)

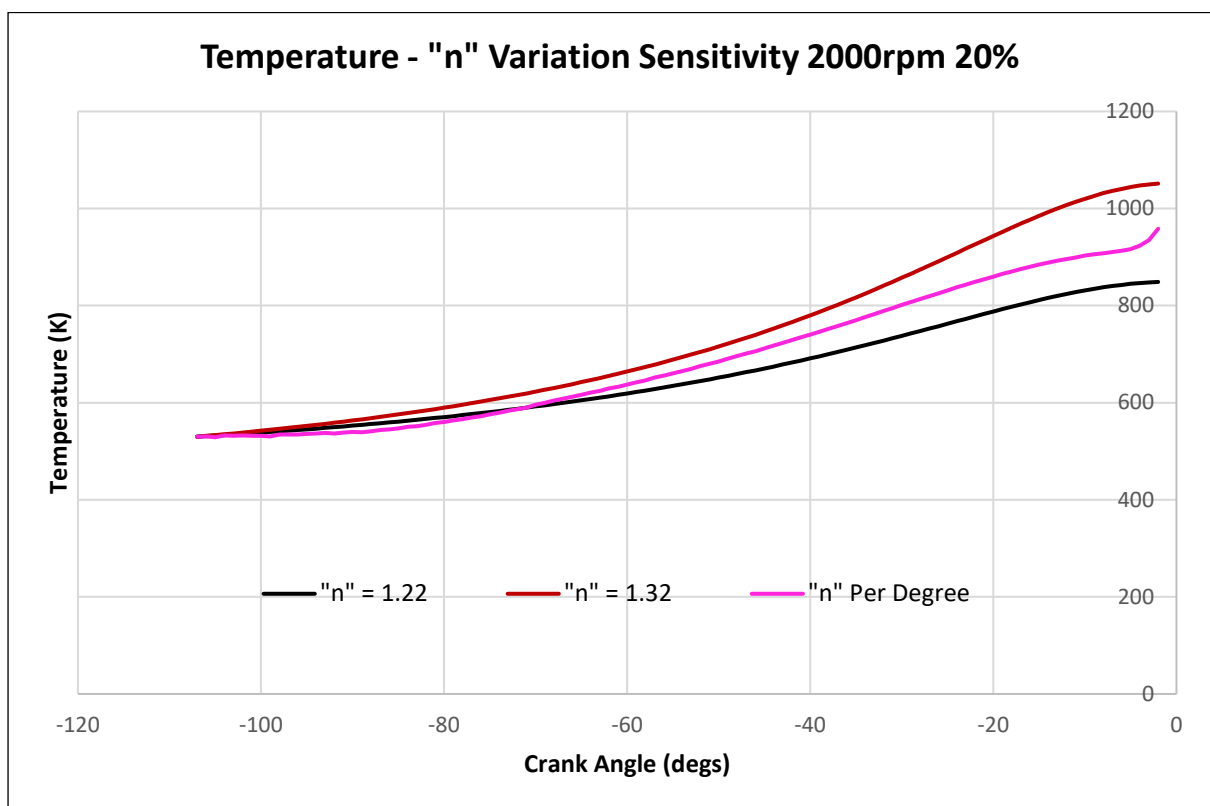


Figure 85 - Temperature - "n" Variation Sensitivity (Vacas, Authors Own)

Figure 85 "Temperature - "n" Variation Sensitivity (Vacas, Authors Own)" shows three different values of n used, as a lower value "1.22" (original acute range calculated value), a high value "1.32" (KTM's supplied value) and a variable value " n " (calculated per degree). The outcome of this analysis was that in terms of auto-ignition temperature (the temperature reached close to TDC) the values of " n " had a significant effect – between the values of "1.22" and "1.32" there was a 200K difference in temperature. The effect seen in this variation was significantly more than any effect " n " variation had shown in prior analysis – considering this it was decided that for accuracy and effect the " n " value used for the temperature equation would be the average " n " calculated from EPC to SOC (circa).

7.5 – Final Temperature Calculations

Figures 86 and 87 show the clear impact of shutting the exhaust valve, both 20% and 100% have had a significant increase in temperature in the sub 5500rpm area.

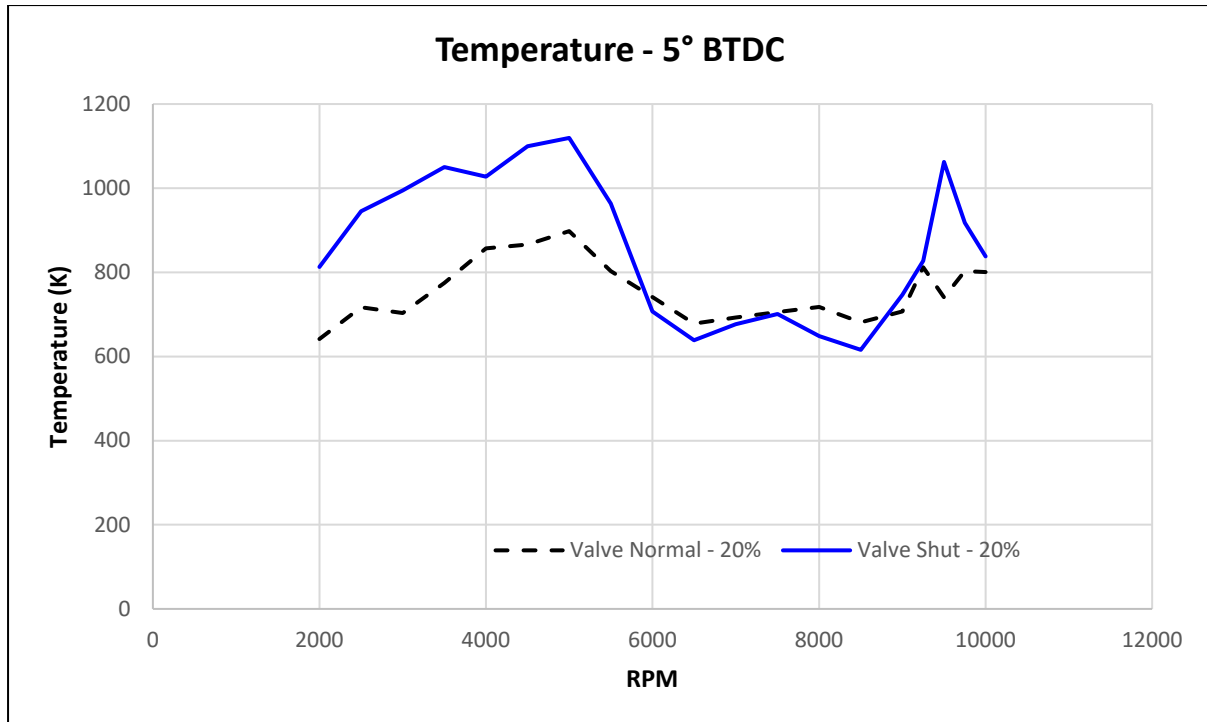


Figure 86 - Temperature - 5° BTDC 20% (Vacas, Authors Own)

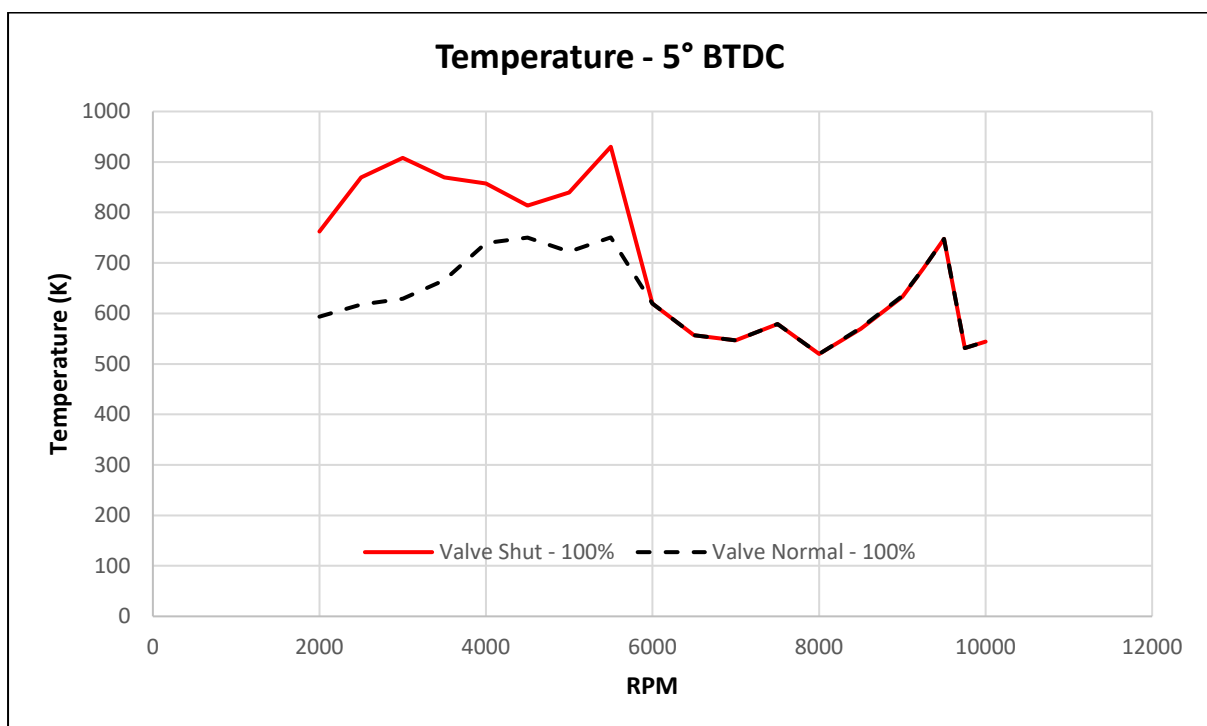


Figure 87 - Temperature - 5° BTDC 100% (Vacas, Authors Own)

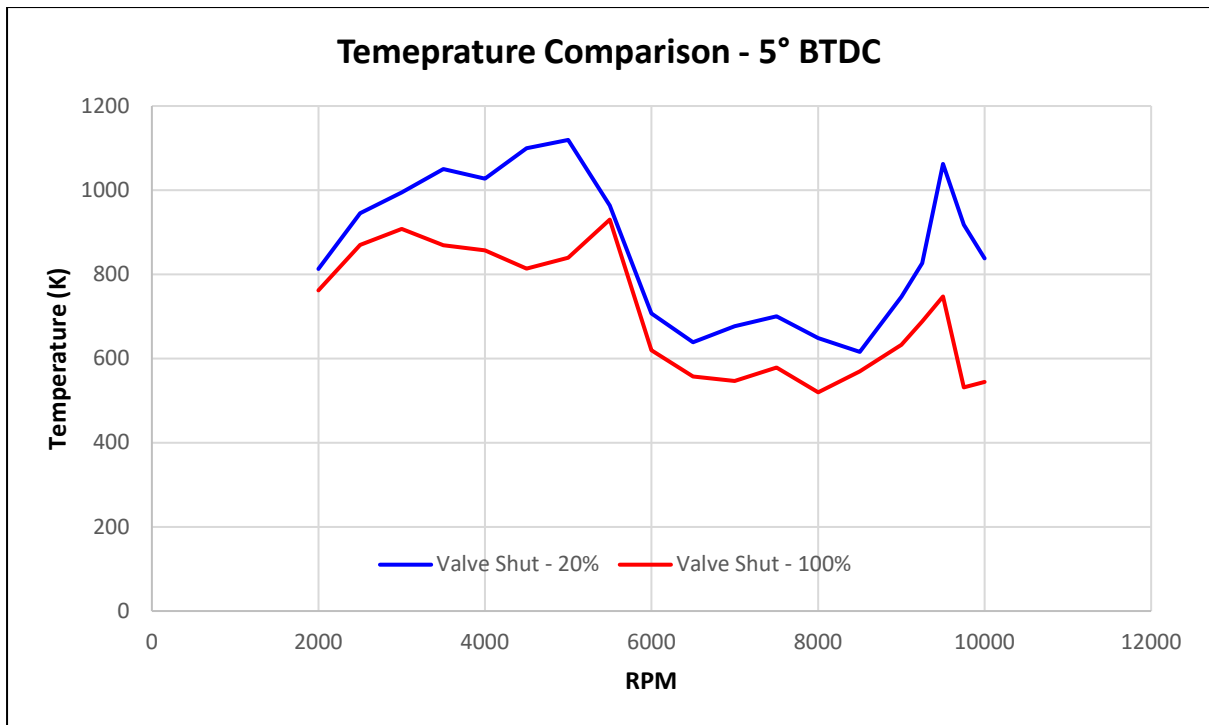


Figure 88 - Temperature - 5° BTDC Comparison (Vacas, Authors Own)

Figure 88 shows the clear difference between low and high load situations, auto-ignition looks likely from 2500rpm to 5500rpm.

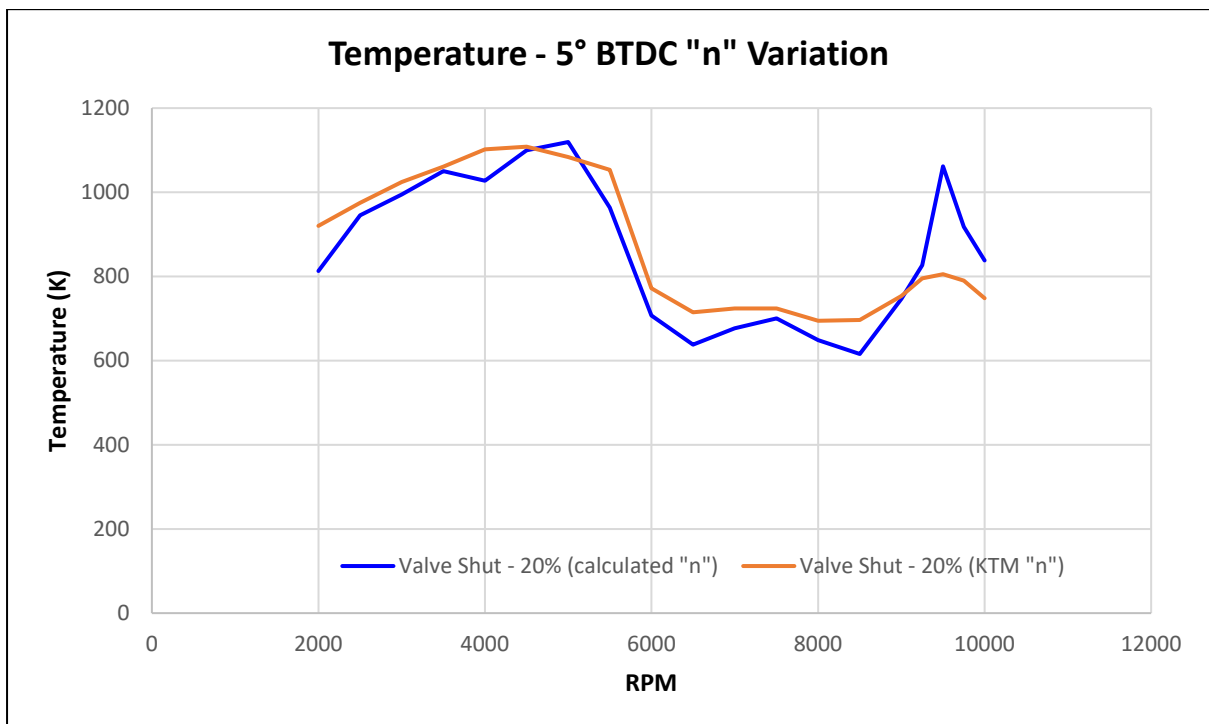


Figure 89 - "n" Variation 20% (Vacas, Authors Own)

These graphs all points towards CAI being an obtainable goal.

Chapter 8 – Conclusion

Investigating two-stroke controlled auto-ignition was a project chosen to combine personal interests with academic enrichment. Originally conceived as a project with CAD in mind, the project soon started to develop a heavy bias towards analysing the fundamentals components of combustion. Initially when examining the combustion process, it was hard to draw conclusion and direction naturally, but the more time spent on the project the more information that was slowly absorbed and something that was originally a weakness soon became a thoroughly enjoyable challenge each step of the way.

Having a project with real industrial potential was a great motivation, and not only being a motivator, it was also a great influence in directing the project towards having a very practical approach. The very initial question was whether auto-ignition had relevance in a modern two-stroke engine, the thorough research conducted answered that question with yes, it is almost certainly possible.

With the question of possibility answered early on, what could the project become and what produce the optimum outcome? The logic soon started to turn towards basics and fundamentals, an excel spreadsheet and several equations is where the project started and ended – but to define the project by that most unassuming description would be a discredit. Keeping to the basics and being unassuming is what gave the project it's real strength.

It must be mentioned that a significant underpinning in the project's success was the extensive literature review, the literature review was conducted months in advance – the aim was simple and that was just to find as much literature as possible on two-stroke controlled auto ignition. Whilst most of the analysis work was based on the exploration of fundamentals, the body of research provided an answer in almost every situation where a question was asked. What temperature is needed for auto-ignition? 1000K was confirmed, did the in-cylinder gases need to be a homogeneous mix? No, in fact it was the opposite, two-stroke auto-ignition relied on a lack of turbulence at EPC – meaning that the mixture and exhaust did not mix creating exhaust hot spots (kernels) that would act as the ignition epicenters.

In terms of whether CAI is possible in the KTM EXC 250 TPI, the answer is without doubt yes – but does this mean that the project provides exact answers and solutions, no it doesn't. What the project has produced though, is a fundamental structure exploring the components and understanding of the process leading up to CAI. The work conducted has a solid structure based in fundamental principles and there is confidence that with further development and improved input data the project could successfully enable the functioning of a modern two-stroke CAI engine, doing so with confidence and understanding.

Chapter 9 – Further Work

1. Further investigation of combustion fundamentals, the polytropic index, MFB and the derivation & application of α and m , with the aim to improve and refine the combustion model
2. Explore methods in deducing in cylinder conditions, one key area of focus is to improve the calculations for in cylinder conditions, two proposed methods could include,

a) Honda EPC temperature & Auto-Ignition Temperature

Equation 39 - Honda Cylinder Temperature (Ishibashi, 2000)

$$Ti = \frac{(Te - Tf)}{(\eta_{s+} + (2Tf - Te))}$$

Equation 40 - Auto-Ignition Temperature (Ishibashi, 2000)

$$T_{\theta^*} = Ti(\varepsilon^*)^{n-1}$$

b) *“A Novel Model for Computing the Trapping Efficiency and Residual Gas Fraction Validated with an Innovative Technique for Measuring the Trapping Efficiency”* (Manente, et al., 2008 Small Engine Technology Conference)– Recent paper exploring and validating new techniques for predicting in cylinder conditions - improvement upon Blair’s methods (Blair, 1994).

3. Get an engine running, work towards a new cylinder design but in the meantime a simple butterfly can be put on the exhaust.
4. Looking further into thermal management of engine components, ask what can be done?!, can cylinder heads be ceramic coated? Could a secondary water jacket be used?
5. Think towards extremes, can auto-ignition be extended through a twin cylinder design to better control and time the reciprocal fluid dynamics.

Appendices

Appendix A - 1D Simulation Model

A.1 - Software Package

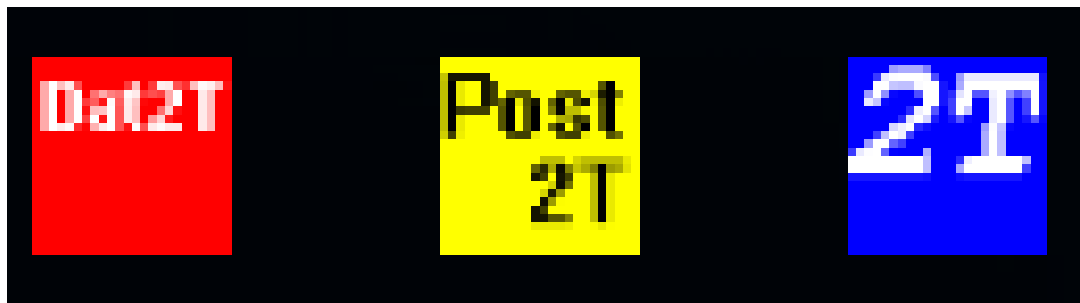


Figure 90 - Software Package

A.2 – Dat2T (Data Entry)

A.2.1 – Introduction

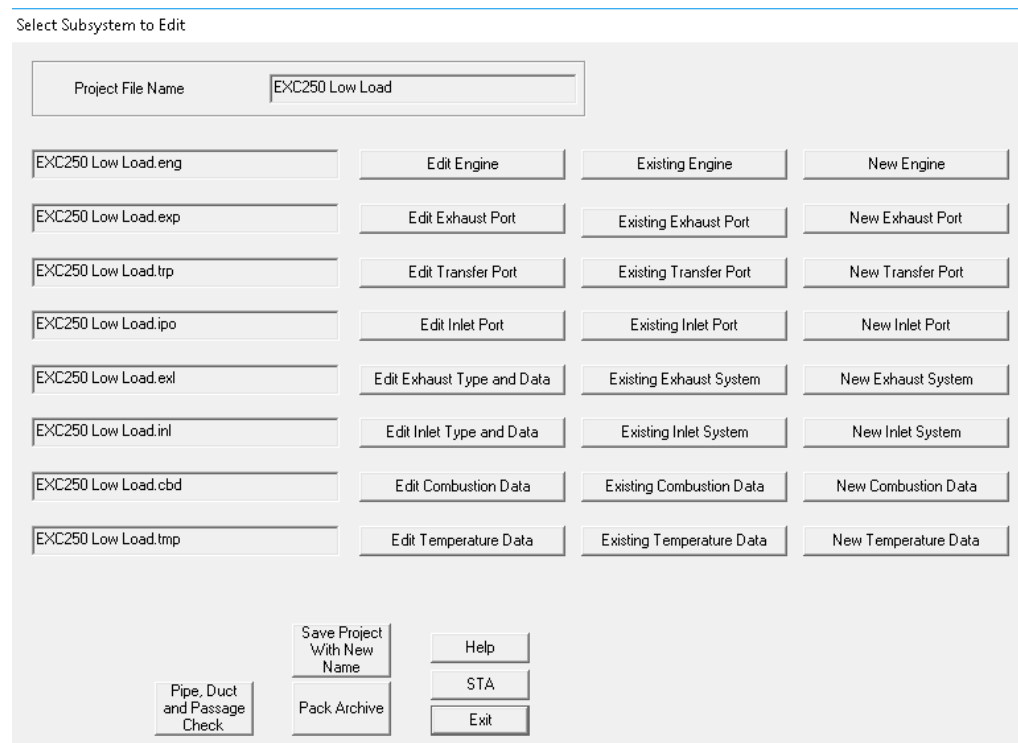


Figure 91 - Dat2T Overview

A2.2 – Engine

Engine Data: EXC250 Low Load.eng

Induction Type

- ☒ Naturally Aspirated Crankcase Scavenged
- ☐ Turbocharged Crankcase Scavenged
- ☐ Supercharged Crankcase Scavenged
- ☐ Turbocharged Externally Scavenged
- ☐ Supercharged Externally Scavenged
- ☐ Turbo Supercharged Externally Scavenged

Basic Engine Type

- ☐ Industrial (Fan Cooled)
- ☒ Performance (Air/Water cooled)

Piston Layout

- ☒ Conventional Parallel Piston
- ☐ Stepped Piston

Friction Loss Model

- ☐ Plain Bearing
- ☒ Roller Bearing
- ☐ User Defined

Loss Coefficient and Slope

Coefficient (0 to 150000)

Loss Slope (-150 to +150)

CrankCase Compression Ratio

Crankcase Compression Ratio: 1.452

- ☐ Transfer volume included in ratio
- ☒ Transfer volume excluded from ratio

CrankCase Compression Calculator

Firing Order

Single cylinder engine

Combustion Type

- ☒ Spark Ignition
- ☐ Compression Ignition
- ☐ Glow Plug

Engine Layout

- ☒ Inline Engine
- ☐ Vee Engine
- ☐ Flat Engine

Basic Engine Data

Number of Cylinders: 1

Cylinder Volume: 248.87

Bore (mm): 66.340

Engine Volume: 248.9

Stroke (mm): 72.000

Bore/Stroke: 0.921

Conrod Length (mm): 125.000

Conrod/Stroke: 1.736

Gudgeon Pin Offset (mm): 1.000

Degree/Height Calc

Combustion Chamber

Compression Ratio: 11.000

- ☐ Trapped Ratio
- ☒ Geometric Ratio

Squish Clearance (mm): 1.800

Squish Area Ratio (0-0.7): 0.300

Piston Dome Radius (mm): 140.00

Combustion Chamber Calculator

Simulation Data

Max Power RPM: 8500.00

Target BMEP (bar): Calculate 10.36

Exhaust Pressure Transducer: 50.000

Inlet Pressure Transducer: 50.000

Number of Iteration Cycles: 12

Throttle Opening (0 to 1): 0.200

Mean Piston Speed

20 m/s

4016 ft/min

16 m/s - Industrial

20 m/s - performance

25 m/s - high perf

30 m/s - extreme drag

Mechanical Design and Balancing

Save Engine in New File

Accept and Save

Exit without Saving

Help

Figure 92 - Dat2T Engine Data

Combustion Chamber Schematic

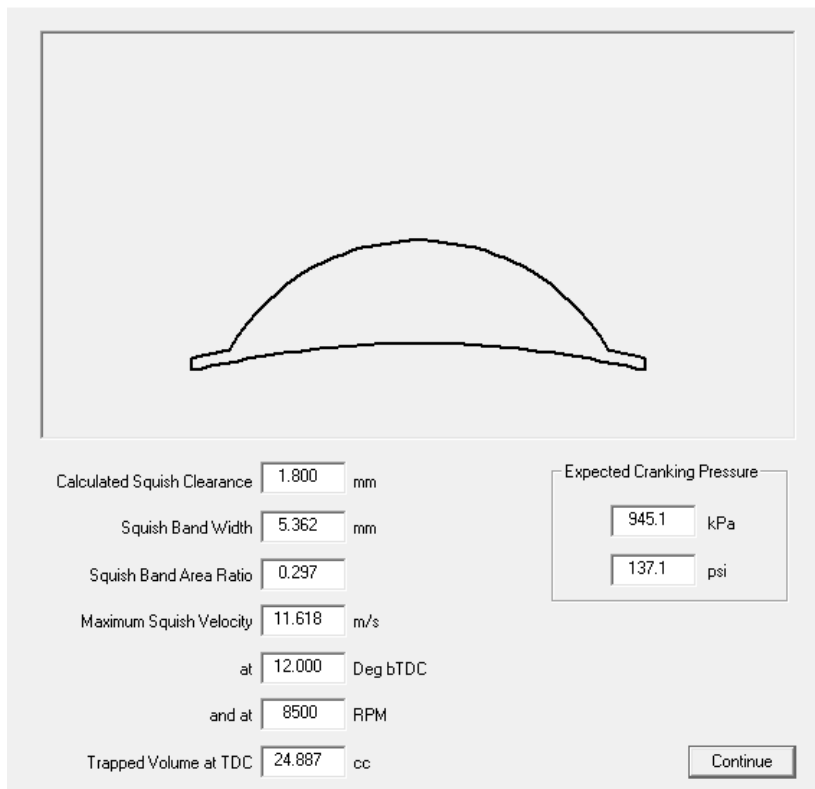


Figure 93 - Dat2T Combustion Chamber 1

Combustion Chamber Schematic

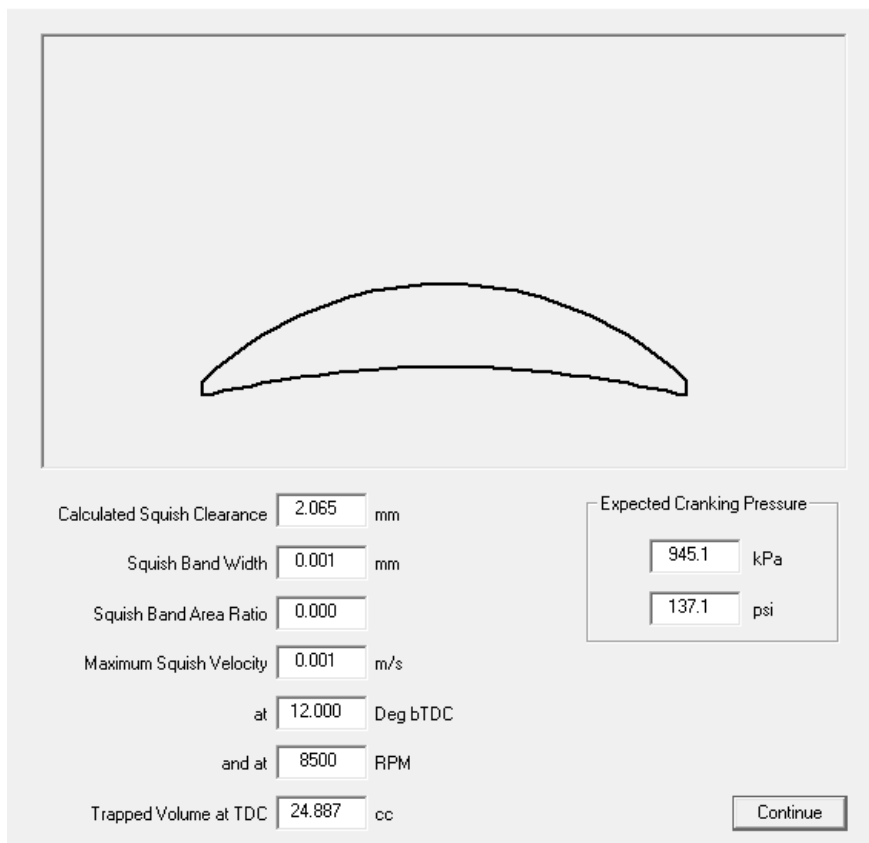


Figure 94 - Dat2T Combustion Chamber 2

A.2.3 – Exhaust Port

Calculate Port Timing from Piston Position

Deck height is amount piston is from cylinder top at TDC

Port Open Height from cylinder top

Port Full Open from cylinder top (this can be full stroke or less)

Port

Positive deck height - piston stops below the surface

Negative deck height - piston sticks out above the surface

Port Edge Height from Cylinder top surface

Deck Height (mm).....

Port Height from top (mm)

or

Port Open Duration (deg)

Degrees after TDC.....

Help

Calculate

Cancel

Accept

Figure 95 - Dat2T Timing Calculator

Exhaust Port Data: EXC250 Low Load.exp

Type of Exhaust Port

- ☒ Single Type
- ☐ Bridged Type
- ☐ Profiled Type

Auxiliary Ports

- ☐ No Auxiliary Ports
- ☒ Auxiliary Ports

Power Valves

- ☐ No Power Valve
- ☒ Controlled Power Valve
- ☐ Pneumatic Power Valve

Number of Port Windows: Number of Auxiliary Port Windows:

Port Window Open (deg ATDC) Calc: Port Window Open (deg ATDC) Calc:

Port Win Full Open (deg ATDC) Calc: Aux Port Full Open (deg ATDC) Calc:

Flow Width at top of window (mm): Flow Width of Top of Window (mm):

Flow Width at bottom of window (mm): Flow Width of Bot of Window (mm):

Top Edge Radius (0 = Straight) (mm): Top Corner Radius of Port (mm):

Bottom Edge Radius (0 = Straight) (mm): Bottom Corner Radius of Port (mm):

Top Corner Radius (mm): Height of Auxiliary Passage (mm):

Bottom Corner Radius (mm): Width of Auxiliary Passage (mm):

Passage Length (mm): Corner Radius of Passage (mm):

Passage End Diameter (mm) Calc: Exit Roof Angle of Passage (deg):

Exit Roof Angle of Main Passage (deg):

Save Data to New File

Help

Display Only

Save and Display

Exit without Saving

Power Valve Data

Auxiliary Ports

- ☐ No Power Valve
- ☒ Also Aux Port Power Valve
- ☐ Only Aux Port Power Valve

Controlled Timing at RPM's

Number of RPM Points:

RPM	Timing (deg)
1: 6000.0	105.000
2: 8500.0	86.725
3: 10000.0	86.725
4:	
5:	
6:	
7:	
8:	
9:	
10:	

Display PowerValve Curve

Pneumatic Power Valve

Powervlve closed (deg)

Valve Shalt Diameter (mm)

Valve Travel (mm)

Diaphragm Diameter (mm)

Diaphragm Stiffness (g/mm)

Diaphragm Preload (g)

Spring Rate (g/mm)

Spring Preload (mm)

Figure 96 - Dat2T Exhaust Port

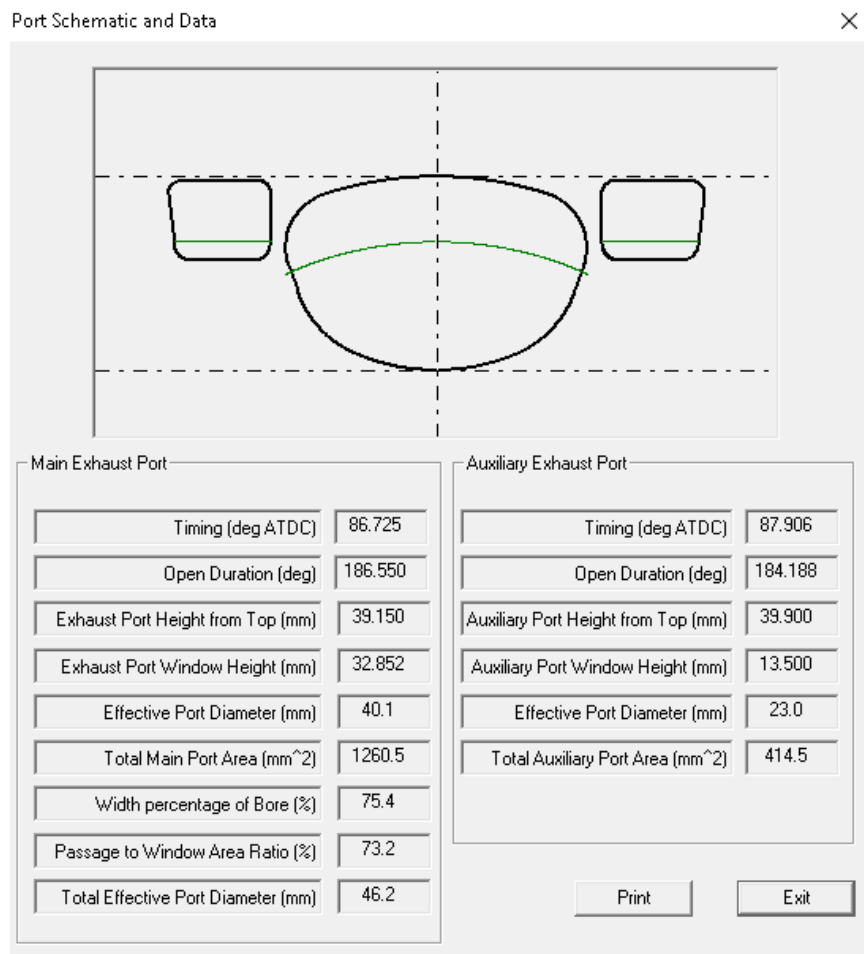


Figure 97 - Dat2T Exhaust Port Display

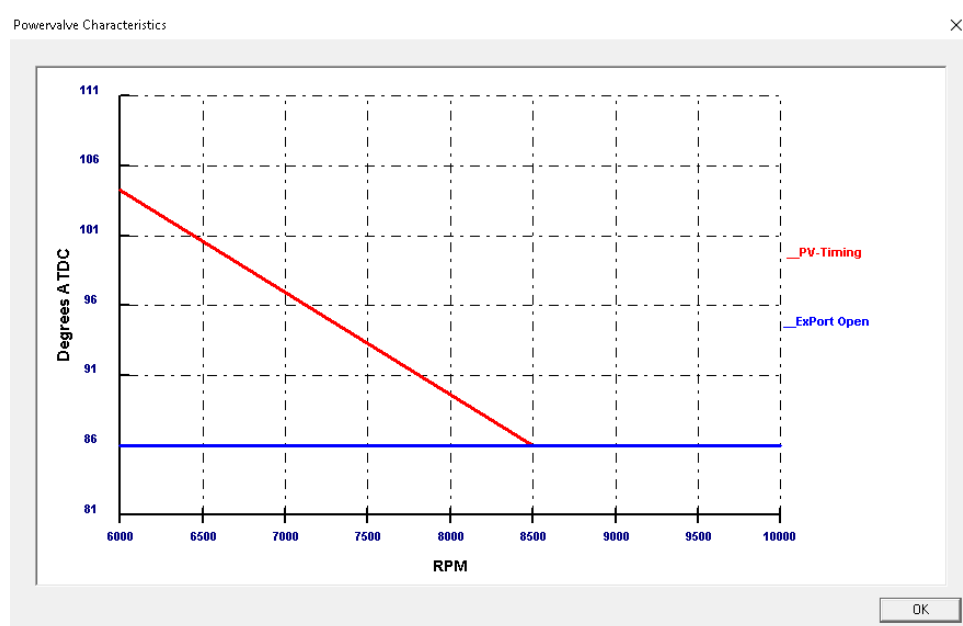


Figure 98 - Dat2T Exhaust Valve Timing

A.2.4 – Transfer Port

Transfer Port Data: EXC250 Low Load.trp

Total Number of Transfer Port Windows (Max=10)

	Port Set 1	Port Set 2	Port Set 3	Port Set 4	Port Set 5
Number of Windows (1 or 2)	2	2	1		
Transfer Port Open ATDC	Calc 121.177	Calc 121.177	Calc 117.129		
Transfer Port Fully Open ATDC	Calc 180.000	Calc 180.000	Calc 180.000		
Flow Width at Top of Window (mm)	24.000	30.000	23.000		
Flow Width at Bottom of Window (mm)	30.000	30.000	23.000		
Top Corner Radius of Window (mm)	3.000	3.000	3.000		
Bottom Corner Radius of Window (mm)	4.000	4.000	3.000		
Roof Angle of Passage from Horizontal (deg)	13.000	13.000	68.500		
Length of Transfer Passage (mm)	87.000	87.000	57.000		
Area Ratio between Window and Passage Entry	1.200	1.200	1.200		
Area Ratio between Window and Passage Exit	1.000	1.000	1.000		
Transfer Passage Inlet Configuration	☒ Sharp ☐ Square ☐ Round	☒ Sharp ☐ Square ☐ Round	☒ Sharp ☐ Square ☐ Round		
Transfer Passage Shape Configuration	☐ Open ☐ Straight ☒ TeaCup	☐ Open ☐ Straight ☒ TeaCup	☐ Open ☐ Straight ☒ TeaCup		

Scavenging Model

☐ Uniflow	☐ Yam6	☐ QUBCR	☐ LoopSaw
☐ SCRE	☐ Yam12	☐ Deflector	☐ Honda RS125 '04
☐ Yam1	☒ Yam14	☐ Small	☐ Aprilia RSW125

Save Data in New File

Help

Passage Layout

Display Only

Save and Display

Exit without saving

Figure 99 - Dat2T Transfer Port Data Entry

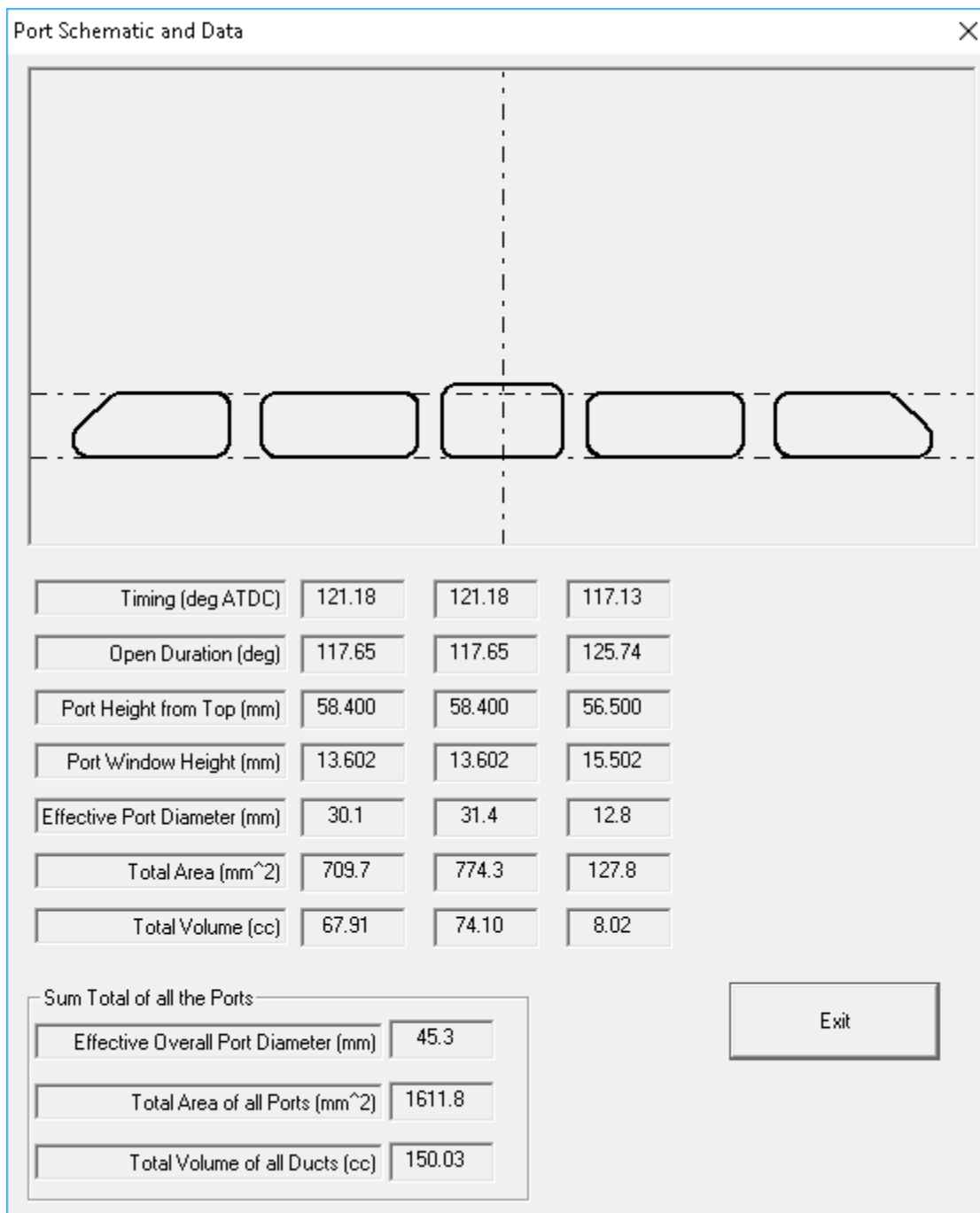


Figure 100 - Dat2T Transfer Port Display

Transfer Port Cylinder Entry Design

Input Data

	Port1	Port2	Port3	Port4	Port5
A					
B					
ThetaA					
ThetaB					
Boost Port	☐ Yes ☒ No	☐ Yes ☒ No	☒ Yes ☐ No		

Window Chordal Width

Flow Width

Help
Calculate
Display
Accept
Cancel

Figure 101 - Dat2T Window / Scavenging Calculator

A.2.5 – Inlet Port

CrankCase Reed Valve controlled Inlet Port: EXC250 Low Load.ipo

Type of Inlet Port

- ☐ Piston Port
- ☒ Case Reed Valve
- ☐ Cylinder Reed Valve
- ☐ Rotary Disc Valve
- ☐ Rotary Shaft Valve
- ☐ Rotary Drum Valve
- ☐ Piston Port Reedvalve
- ☐ No Inlet Port

Curved Stopper Plate

- ☐ Yes
- ☒ No

Number of Sides

- ☐ 1 Sided Block
- ☐ 2 Sided Block
- ☒ 4 Sided Block

Number of Petals: 8

Thickness of Petals (mm): 0.370

Width of Petals (mm): 27.450

Free Length of Petals (mm): 32.000

Young's Modulus of Petal Material (GPa): 20.800

Density of Petal material (kg/m³): 1380.000

Damping factor for first mode: 0.500

Damping factor for second mode: 0.125

Reed block included angle (deg): 25.000

Diameter of rubber block (mm): 38.000

Reed port width in block (mm): 25.300

Reed Port length in block (mm): 28.000

Corner radius of port in block (mm): 4.500

Maximum Tip Displacement (mm): 7.250

Figure 102 - Dat2T Reed Valve Data Entry

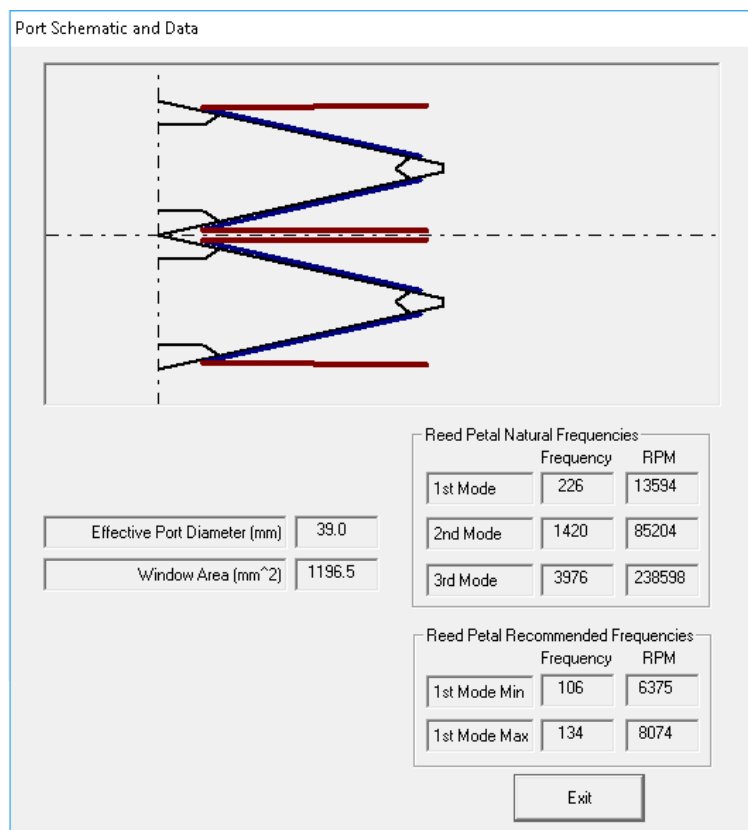


Figure 103 - Dat2T Reed Valve Display

A.2.6 – Exhaust System

Tuned Exhaust Pipe Data: EXC250 Low Load.exl

Tuned Pipe Dimensions

Number Of Segments in Tuned Pipe (Max=15)

Starting Diameter

Length 1	30.000	Diameter 1	44.000
Length 2	500.000	Diameter 2	76.000
Length 3	260.000	Diameter 3	112.000
Length 4	155.000	Diameter 4	112.000
Length 5	340.000	Diameter 5	28.000

Tail Pipe Dimensions

Number Of Segments in Tail Pipe (Max=15)

Starting Diameter

Length 1	295.000	Diameter 1	28.000
----------	--------------------------------------	------------	-------------------------------------

Atmospheric Outlet

☒ Radiused
☐ Sharp

Remove a Segment

Add a Segment

Create new Type

Help

Design Blair Type Pipe

Save as a new File

Display

Save and Display

Exit without saving

Total:

Total:

Figure 104 - Data2T Exhaust Data Entry

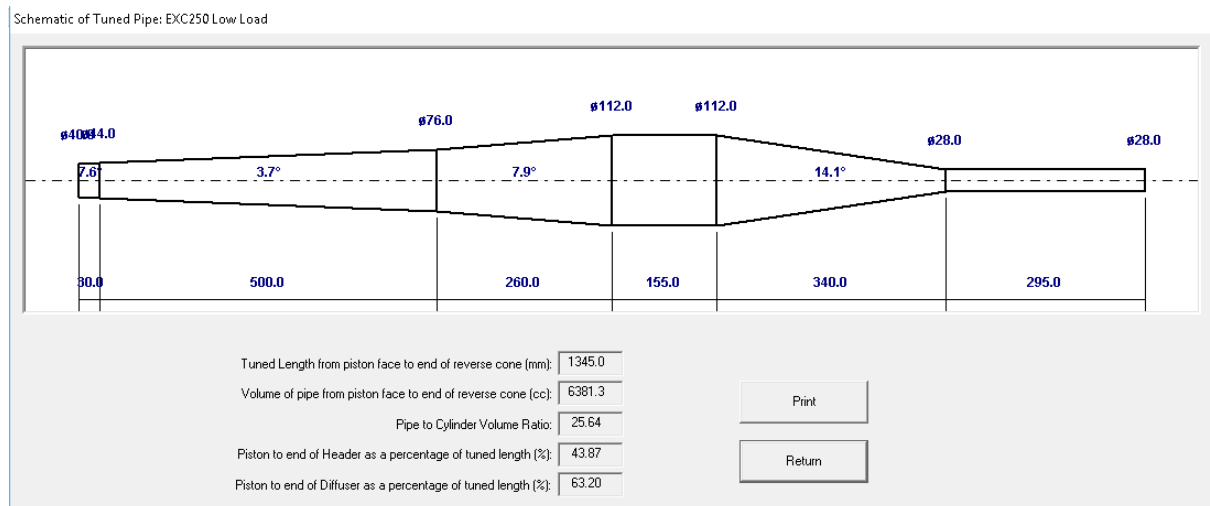


Figure 105 - Dat2T Exhaust Display

A.2.7 – Inlet System

Inlet System Data: EXC 250 High Load.inl

Throttle Pipe Dimensions				Cylinder Pipe Dimensions			
Number Of Segments in Throttle Pipe (Max=11)		5		Number Of Segments in Cylinder Pipe (Max=11)		2	
Starting Diameter		53.000		Starting Diameter		39.000	
Length 1	2.000	Diameter 1	50.000	Length 1	71.500	Diameter 1	38.000
Length 2	2.000	Diameter 2	47.000	Length 2	40.500	Diameter 2	39.000
Length 3	2.000	Diameter 3	44.000				
Length 4	100.000	Diameter 4	41.000				
Length 5	40.500	Diameter 5	39.000				
Total: 146.50				Total: 112.00			

Open End

☐ Radiused

☐ Sharp Edged

☒ Bellmouth

Type of Throttle

☐ Slide

☐ Plate

☒ Butterfly

Spindle Diameter

4.000

Save as a New File Name

Help

Display

Save and Display

Exit without saving

Create New Inlet

Figure 106 - Dat2T Inlet System Data Entry

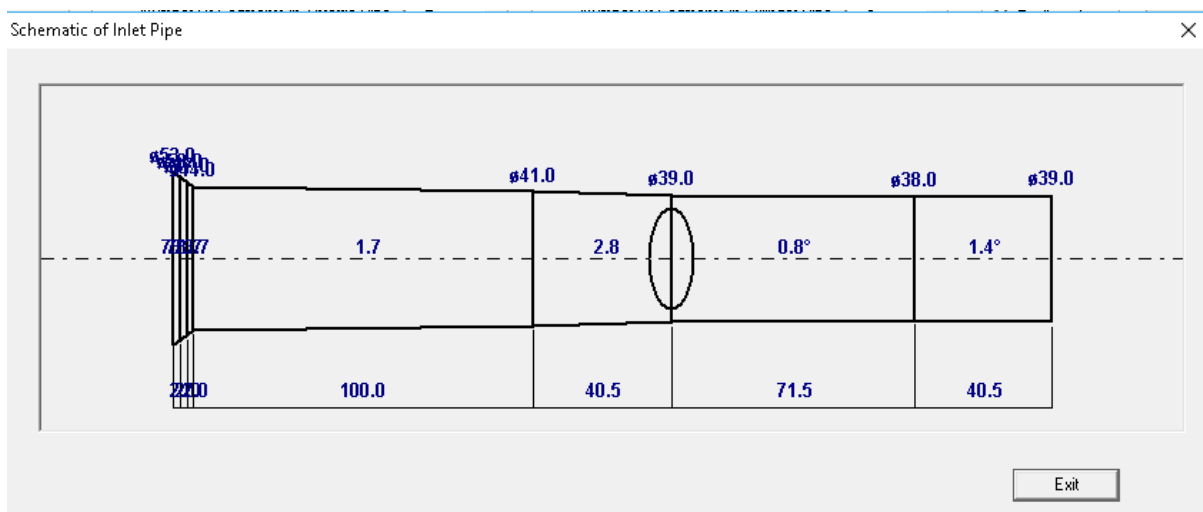


Figure 107 - Dat2T Inlet System Display

A.2.8 – Combustion

Combustion Data: EXC 250 High Load.cbd

Combustion Model

☒ Burnrate Prescribed

☐ Adjusted Wiebe Function

☐ Turbulent Model

Model Parameters

Combustion Efficiency (0-1)

Nitromethane Ratio (0-1)

Number of RPM points in map

Fuel Type

☐ Regular ☐ Methanol ☐ AV-Gas

☐ Premium ☐ Ethanol ☐ Unleaded / Ethanol mix

☒ Unleaded 95 ☐ JP5

☐ Unleaded 100 ☐ JP8

☐ Leaded 110

	RPM	Timing (deg)	AFR	Delay (deg)	Duration (deg)	Vibe A	Vibe M	Stoichiometric AFR
1	2000.00	8.40	16.57	1.40	47.00	11.63	2.24	14.28
2	2500.00	11.90	16.12	1.00	44.00	6.00	1.25	
3	3000.00	11.90	15.82	0.90	44.00	21.70	2.87	
4	3500.00	12.90	15.56	1.90	43.00	18.26	2.47	
5	4000.00	13.20	15.43	2.20	47.00	18.33	2.40	
6	4500.00	14.90	15.55	3.90	47.00	18.54	2.22	
7	5000.00	14.90	15.64	3.90	50.00	13.00	1.99	
8	5500.00	14.90	15.65	2.90	48.00	6.26	0.68	
9	6000.00	14.80	16.12	2.80	54.00	13.11	1.78	
10	6500.00	13.80	15.81	1.80	54.00	17.92	2.41	
11	7000.00	11.20	16.46	4.20	60.00	11.71	1.75	
12	7500.00	10.90	15.99	2.90	56.00	17.16	2.47	
13	8000.00	5.40	15.73	1.40	48.00	14.69	2.27	
14	8500.00	5.90	16.13	1.90	51.00	14.55	2.22	
15	9000.00	8.40	16.19	3.40	51.00	10.33	1.35	
16	9250.00	12.50	15.91	4.50	53.00	11.20	1.64	
17	9500.00	15.40	15.75	5.40	54.00	10.87	1.85	
18	9750.00	18.00	15.65	4.00	54.00	12.03	2.01	
19	10000.00	23.50	15.07	10.50	55.00	13.48	2.03	
20								

Save File with New Name

Help

Remove a row

Plot Timing Curves

Save and Exit

Exit without saving

Figure 108 - Dat2T Combustion Data Entry

A.2.9 – Temperatures

Temperature Data: EXC 250 High Load.tmp

Exhaust Pipe Wall Temperature

☒ Constant Wall Temperature

☐ Temperature RPM function but constant along length

☐ Temperature an RPM function and varying along length

Inlet Port Wall Temperature (C) 20.000

Crankcase Wall Temperature (C) 300.000

Transfer Port Wall Temperature (C) 80.000

Exhaust Port Wall Temperature (C) 250.000

Exhaust Wall Temperature at Max Power RPM (C) 350.000

Piston Crown Temperature (C) 220.000

Combustion Chamber Wall Temperature (C) 220.000

Cylinder Liner temperature (C) 150.000

Atmospheric Temperature (Std=20) (C) 20.000

Atmospheric Pressure (Std=101.325) (kPa) 101.325

Inlet Conditions

☒ Static Inlet Pressure

☐ Dynamic Inlet Pressure

Calculate for Altitude

Altitude (m) 0

Altitude (ft) 0

Exhaust Pipe Wall Temperature at RPM's

Number of RPM Points

	RPM	Temperature1	Temperature2	Temperature3
1				
2				
3				
4				
5				
6				
7				
8				
9				
10				

Save Data to New File

Select Default Values

Accept and Save

Exit without Saving

Help

Figure 109 - Dat2T Temperature Data Entry

A.3 – EngMod2T (Simulation)

A.3.1 – Start Screen

Project Name: EXC250 Low Load EngMod2T Beta, Version:V5.3.0

Run Type ☒ Individual Runs ☐ Batch Run	Beep after run ☐ Yes ☒ No	Output data to new file? ☐ Yes ☒ No	Show Front Page ☒ Yes ☐ No	Show Port Data ☒ Yes ☐ No	Continue Exit Program Help
Display Graphics ☐ Yes ☒ No					
User Controlled Runs ☒ Use Project Files for Timing and AFR data ☐ Manually Input Timing and AFR data Number of RPM points: 10 RPM 1: 2000 RPM 2: 2500 RPM 3: 3000 RPM 4: 3500 RPM 5: 4000 RPM 6: 4500 RPM 7: 5000 RPM 8: 5500 RPM 9: 6000 RPM 10: 6500					

Figure 110 - Engmod2T Start Screen

A.3.2 – Run Screen

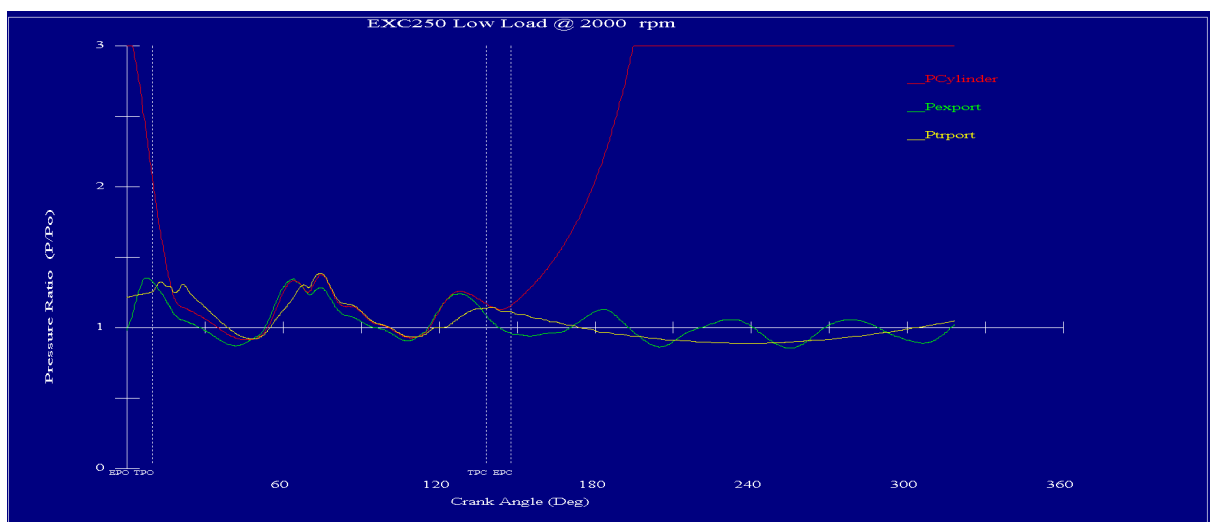


Figure 111 - Engmod2T Run Screen

A.3.3 – Exit Screen

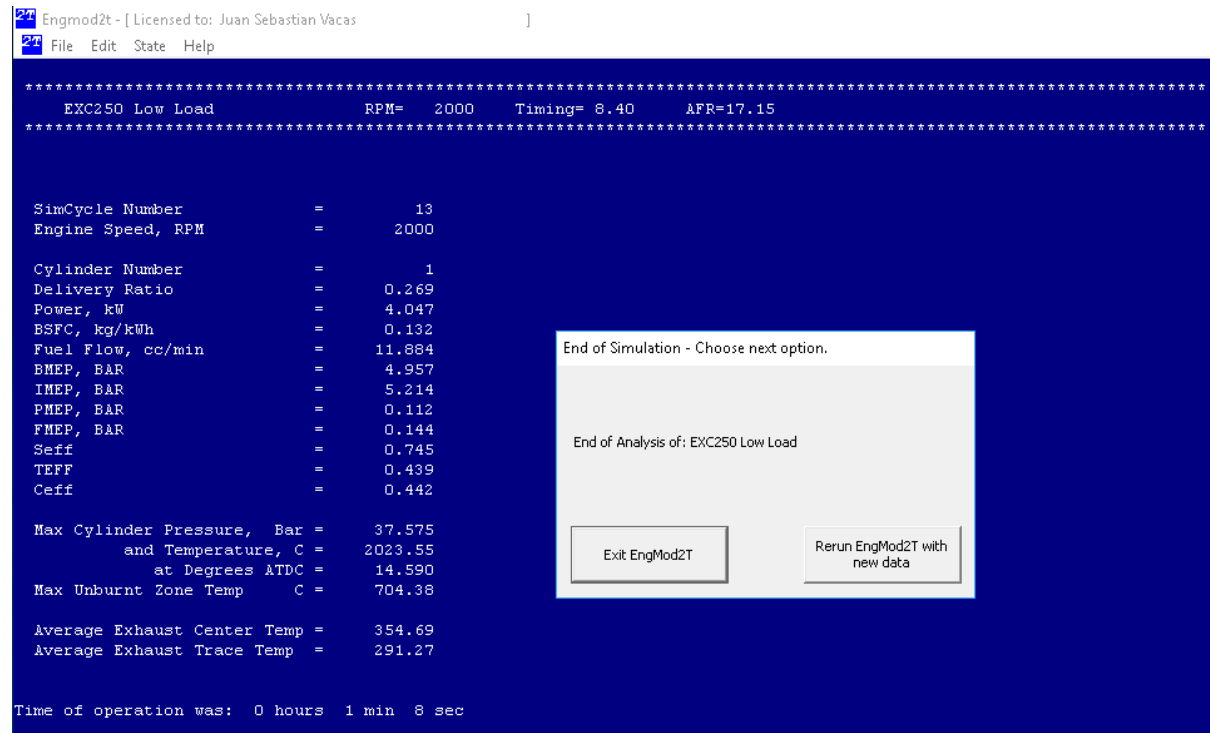


Figure 112 - Engmod2T Exit Screen

A.4 – Post2T (Post Processor)

A.4.1 – Plot Options

Load Result Files and Select Plot Settings

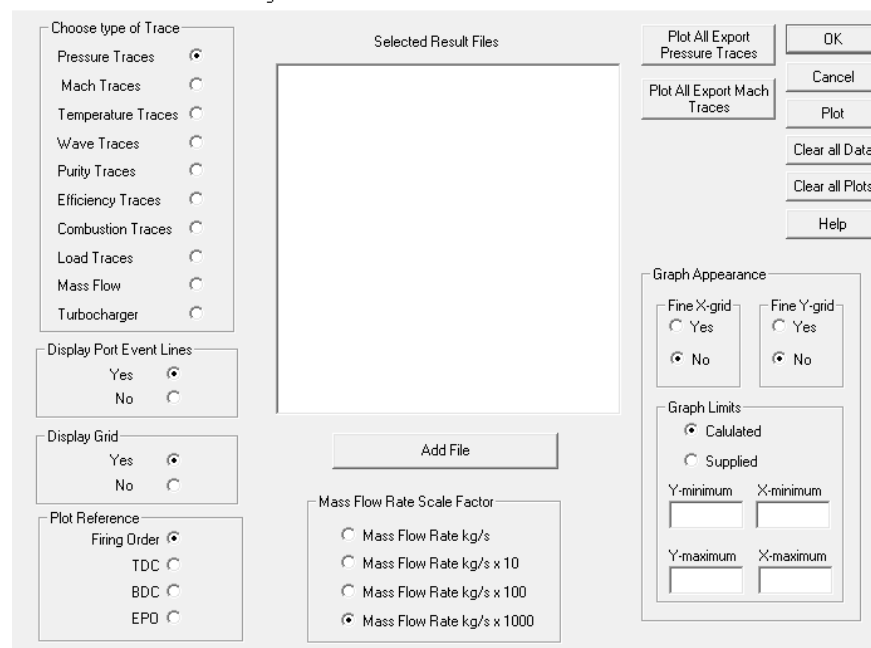


Figure 113 - Post2T Plot Options

A.4.2 – Plot Display

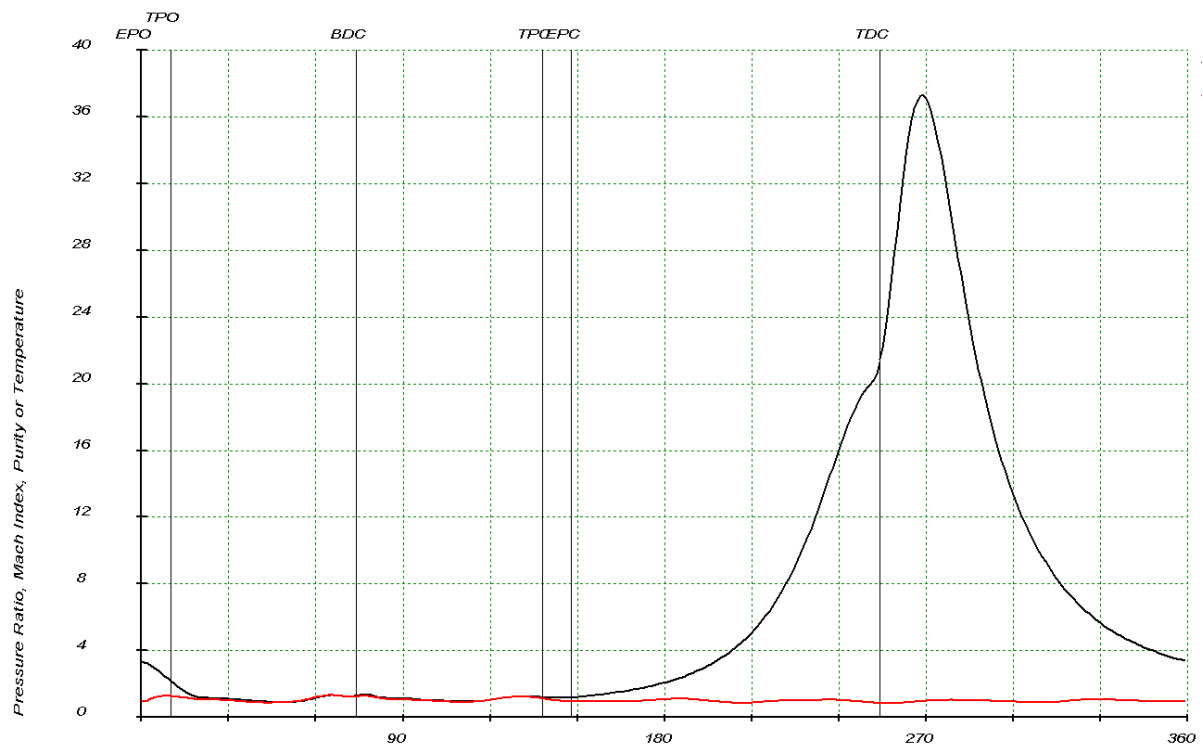


Figure 114 - Post2T Plot Display

A.4.2 – Excel Data Import

A.4.2.1 – File Import

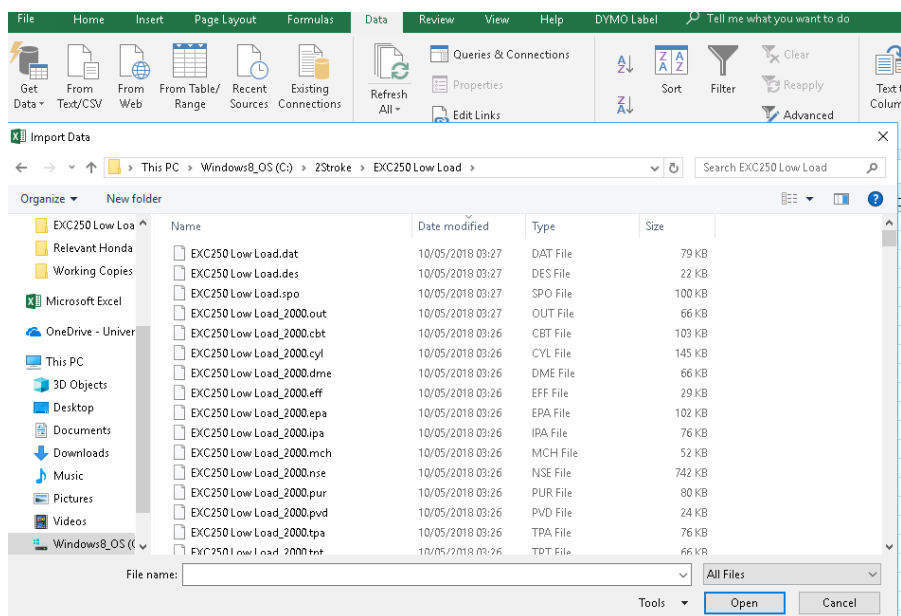


Figure 115 - Post2T Excel File Import (Space De-limited file)

A.4.2.2 – Data Editing

Column1	Column2	Column3	Column4
2000	RPM		
1	0	0	NumCyl NumTurbo NumSuper
1	NumCrankCase		
2	2	NumPipIn	BoostBottle
0	NumColIn		
0	0	NumBoxIn	NumInOrifice
3	NumTrPip		
2	1	NumPipEx	
0	NumColEx		
0	NumBoxEx		
1	NumOutPipEx		
0	NumExSil		
0	NumExSilPlen		
50.00000	ITraceL		
50.00000	ETraceL		
REED			
106.0			
106.0 254.0 116.0 244.0			
Deg	PITrace(1)	PITrace(2)	Pin(1) Displ(1)/10 Pcrank(1) P...
0.000000	0.897570	1.000261	0.898221 0.078643 0.8881...
1.000000	0.898125	1.000115	0.898873 0.078431 0.8889...
2.000000	0.898974	0.999902	0.899765 0.078048 0.8899...
3.000000	0.899870	0.999707	0.900646 0.077550 0.8909...
4.000000	0.900789	0.999543	0.901536 0.076933 0.8920...

Figure 116 - Post2T Excel Data Import Editing

A.4.3 – Validation

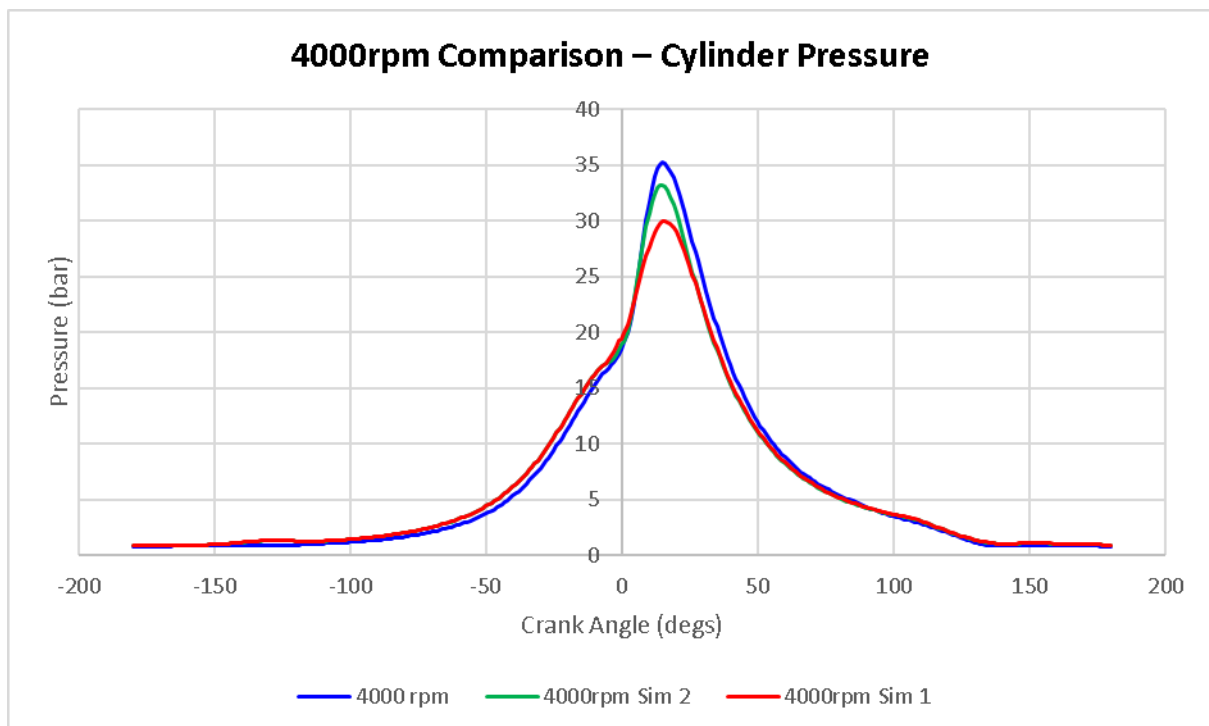


Figure 117 - Post2T Excel Validation (Vacas, Authors Own)

Appendix B - Excel

B.1 – Original Data

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T
1	crank angle [°]	cylinder pressure [bar]																		
2		2000 rpm	2500 rpm	3000 rpm	3500 rpm	4000 rpm	4500 rpm	5000 rpm	5500 rpm	6000 rpm	6500 rpm	7000 rpm	7500 rpm	8000 rpm	8500 rpm	9000 rpm	9250 rpm	9500 rpm	9750 rpm	10000 rpm
3	-180	0.912	0.908	0.885	0.823	0.807	0.896	0.875	0.878	0.845	0.783	0.776	0.734	0.891	0.724	0.72	0.722	0.826	0.778	0.906
4	-179	0.913	0.908	0.889	0.829	0.803	0.889	0.873	0.876	0.847	0.775	0.771	0.727	0.88	0.716	0.718	0.735	0.841	0.76	0.908
5	-178	0.912	0.914	0.894	0.836	0.798	0.876	0.87	0.874	0.844	0.767	0.769	0.722	0.874	0.716	0.716	0.745	0.834	0.766	0.909
6	-177	0.919	0.915	0.9	0.843	0.796	0.87	0.867	0.875	0.844	0.762	0.764	0.715	0.874	0.709	0.72	0.759	0.84	0.778	0.914
7	-176	0.925	0.923	0.9	0.849	0.794	0.863	0.864	0.873	0.847	0.755	0.759	0.709	0.881	0.703	0.72	0.762	0.843	0.772	0.91
8	-175	0.934	0.923	0.899	0.855	0.787	0.858	0.855	0.872	0.844	0.751	0.762	0.701	0.877	0.701	0.717	0.757	0.848	0.763	0.905
9	-174	0.941	0.922	0.899	0.861	0.784	0.852	0.847	0.864	0.846	0.751	0.757	0.701	0.879	0.694	0.715	0.765	0.853	0.766	0.91
10	-173	0.948	0.921	0.896	0.868	0.783	0.847	0.843	0.86	0.844	0.755	0.759	0.698	0.89	0.678	0.716	0.759	0.849	0.768	0.917
11	-172	0.954	0.919	0.9	0.876	0.783	0.843	0.839	0.858	0.843	0.763	0.75	0.693	0.891	0.663	0.714	0.743	0.845	0.769	0.92
12	-171	0.959	0.915	0.897	0.882	0.785	0.841	0.831	0.86	0.838	0.769	0.749	0.692	0.879	0.672	0.714	0.736	0.845	0.767	0.927
13	-170	0.965	0.911	0.9	0.886	0.794	0.841	0.825	0.858	0.83	0.776	0.749	0.693	0.877	0.676	0.708	0.735	0.841	0.765	0.93
14	-169	0.968	0.909	0.903	0.892	0.799	0.844	0.817	0.852	0.824	0.78	0.746	0.692	0.879	0.698	0.7	0.733	0.833	0.769	0.93
15	-168	0.973	0.904	0.903	0.899	0.811	0.84	0.811	0.85	0.823	0.789	0.747	0.698	0.875	0.755	0.692	0.732	0.827	0.759	0.916
16	-167	0.971	0.902	0.9	0.901	0.825	0.84	0.805	0.848	0.815	0.794	0.749	0.697	0.879	0.801	0.678	0.732	0.821	0.757	0.907
17	-166	0.969	0.899	0.895	0.892	0.842	0.836	0.799	0.84	0.809	0.799	0.755	0.695	0.879	0.826	0.67	0.723	0.816	0.758	0.91
18	-165	0.96	0.901	0.895	0.889	0.855	0.836	0.797	0.833	0.797	0.8	0.75	0.694	0.867	0.832	0.667	0.709	0.806	0.749	0.909
19	-164	0.949	0.9	0.893	0.889	0.87	0.834	0.794	0.826	0.801	0.796	0.752	0.696	0.873	0.829	0.664	0.704	0.804	0.738	0.907
20	-163	0.943	0.908	0.889	0.887	0.881	0.846	0.789	0.817	0.796	0.795	0.752	0.701	0.864	0.825	0.658	0.692	0.804	0.733	0.898
21	-162	0.943	0.912	0.889	0.885	0.887	0.841	0.792	0.811	0.791	0.796	0.748	0.701	0.854	0.785	0.655	0.695	0.804	0.735	0.889
22	-161	0.943	0.921	0.882	0.882	0.9	0.85	0.792	0.804	0.789	0.793	0.755	0.704	0.844	0.74	0.651	0.704	0.806	0.728	0.87
23	-160	0.948	0.93	0.878	0.883	0.911	0.859	0.789	0.801	0.78	0.787	0.755	0.71	0.841	0.733	0.666	0.707	0.808	0.721	0.859
24	-159	0.951	0.94	0.875	0.886	0.922	0.869	0.788	0.791	0.773	0.782	0.752	0.709	0.848	0.776	0.662	0.708	0.798	0.721	0.856
25	-158	0.957	0.951	0.874	0.887	0.929	0.872	0.788	0.785	0.767	0.776	0.748	0.706	0.847	0.805	0.654	0.704	0.794	0.723	0.846
26	-157	0.959	0.962	0.874	0.888	0.934	0.876	0.792	0.785	0.764	0.769	0.747	0.706	0.853	0.808	0.651	0.705	0.8	0.719	0.846
27	-156	0.964	0.97	0.875	0.89	0.934	0.882	0.789	0.785	0.766	0.768	0.748	0.705	0.859	0.796	0.661	0.71	0.8	0.718	0.836
28	-155	0.961	0.975	0.874	0.892	0.934	0.889	0.801	0.78	0.759	0.764	0.742	0.7	0.862	0.795	0.664	0.711	0.793	0.705	0.828
29	-154	0.961	0.981	0.873	0.894	0.935	0.895	0.802	0.787	0.752	0.764	0.744	0.703	0.852	0.788	0.663	0.707	0.793	0.702	0.825
30	-153	0.959	0.985	0.874	0.894	0.933	0.904	0.81	0.783	0.749	0.761	0.741	0.705	0.85	0.782	0.655	0.703	0.796	0.7	0.818
31	-152	0.954	0.984	0.88	0.89	0.931	0.91	0.817	0.782	0.752	0.749	0.745	0.707	0.846	0.773	0.654	0.71	0.797	0.707	0.81
32	-151	0.953	0.984	0.884	0.883	0.93	0.917	0.83	0.789	0.754	0.743	0.754	0.705	0.839	0.772	0.661	0.712	0.797	0.71	0.813
33	-150	0.948	0.978	0.891	0.878	0.929	0.927	0.844	0.791	0.763	0.732	0.768	0.701	0.839	0.765	0.676	0.714	0.786	0.717	0.824
34	-149	0.947	0.969	0.895	0.877	0.929	0.931	0.851	0.799	0.777	0.734	0.778	0.702	0.842	0.766	0.676	0.717	0.787	0.715	0.832
35	-148	0.944	0.957	0.91	0.874	0.929	0.932	0.861	0.804	0.789	0.731	0.788	0.693	0.839	0.779	0.681	0.717	0.785	0.712	0.835
36	-147	0.946	0.948	0.92	0.871	0.923	0.93	0.866	0.815	0.795	0.728	0.791	0.686	0.837	0.799	0.733	0.725	0.785	0.705	0.836
		Full load - Cylinder pressure				Full load - Other data				20°THR - Cylinder pressure				20°THR - Other data						

Figure 118 - Original KTM Data (KTM, 2017)

	A	B	C	D	E	F	G	H	I	J	K
1	N	Fuel consumption	lambda, measured	T_environment	T_airbox	T_crankcase	T_exhaustgas	IMEP	MEP	Ignition	MFB50%
2	[rpm]	[kg/h]	[-]	[°C]	[°C]	[°C]	[°C]	[bar]	[bar]	[°CA]	[°CA]
3	2000	2.34	0.881	22.2	27	50.34	253	4.91	4.49	-8.4	10.2
4	2500	2.9	0.906	22.2	27	49.97	277	5.56	5.11	-11.9	6.3
5	3000	3.49	0.923	22.3	27	50.22	305	6.48	5.97	-11.9	5.5
6	3500	4.46	0.938	22.3	27	50.33	344	7.46	6.84	-12.9	5.1
7	4000	5.18	0.946	22.3	27	50.34	367	7.82	7.18	-13.2	5.7
8	4500	5.85	0.939	22.3	27	50.83	384	7.52	6.88	-14.9	6.4
9	5000	6.81	0.934	22.3	27	51.23	393	8.24	7.51	-14.9	6.1
10	5500	7.63	0.951	22.3	27	51.44	407	8.91	8.12	-14.9	6.8
11	6000	8.88	0.906	22.3	27.54	51.95	430	9.17	8.27	-14.8	7
12	6500	9.76	0.923	22.3	28	52.42	451	9.79	8.91	-13.8	7.9
13	7000	11.15	0.887	22.3	28	53.32	478	10.14	9.29	-11.2	10.9
14	7500	11.41	0.913	22.3	28	53.44	510	10.23	9.41	-10.9	13
15	8000	12.13	0.928	22.3	28.2	54.12	567	10.7	9.86	-5.4	18.8
16	8500	12.59	0.905	22.3	29	54.96	607	10.73	9.85	-5.9	18.2
17	9000	13.19	0.902	22.3	29	55.68	618	10.57	9.64	-8.4	16.1
18	9250	13.29	0.918	22.3	29.18	56.67	595	9.81	8.96	-12.5	13.3
19	9500	12.89	0.927	22.3	30	57.7	564	8	7.21	-15.4	15.1
20	9750	12.03	0.933	22.4	30	58.43	531	6.62	5.71	-18	16.1
21	10000	10.95	0.969	22.4	30	59.33	506	5.98	4.95	-23.5	13.2

Figure 119 - Original KTM Data (KTM, 2017)

DM	DN	DO	DP	DQ	DR	DS	DT	DU	DV	DW	DX	DY	DZ	EA	EB	EC	ED	EE	EF	EG	EH	EI
State	Crank Angle (degs)		LOG P (%) - (20%)																			
	LOG VMAX	2500 RPM	3000 RPM	3500 RPM	4000 RPM	4500 RPM	5000 RPM	5500 RPM	6000 RPM	6500 RPM	7000 RPM	7500 RPM	8000 RPM	8500 RPM	9000 RPM	9250 RPM	9500 RPM	9750 RPM	10000 RPM			
Comp	-2	-7.455917200	5.76395	5.77776	5.80038	5.82129	5.84260	5.86426	5.88640	5.90908	5.93180	5.95492	5.97810	5.99155	5.95415	5.93075	5.90778	5.88506	5.86264	5.76804		
Comp	-2	-2.795015811	5.77967	5.79197	5.81261	5.84626	5.88524	5.93126	5.98454	6.04412	6.12017	6.21448	6.32799	6.55654	6.90753	6.98526	6.98000	6.93787	6.84418	5.91411	5.79142	
Comp	-1	-7.721210169	5.79650	5.80585	5.83485	5.86582	5.87674	5.84739	5.75812	5.50057	5.29219	5.14948	5.02269	4.90816	6.02501	6.08801	5.86664	5.50374	5.07919	5.68111	5.79126	
Comp	-2	-2.768357041	5.81268	5.82562	5.85327	5.88156	5.89139	5.88522	5.80424	5.53837	5.30529	5.16979	5.09403	4.98398	6.02693	6.02693	5.95956	5.68860	5.07953	5.88668	5.82126	
Comp	-1	-7.798516041	5.82919	5.82919	5.83919	5.85919	5.87919	5.89919	5.91919	5.93919	5.95919	5.97919	5.99919	6.01919	6.03919	6.05919	6.07919	6.09919	6.11919	6.13919	5.84111	
Comp	-1	-7.811779689	5.84121	5.86424	5.88685	5.91524	5.92593	5.90011	5.82056	5.60376	5.37941	5.26979	6.01213	6.09919	6.09171	6.02214	5.69586	6.01250	5.81622	5.95739		
Comp	-2	-2.642970575	5.86249	5.87121	5.90136	5.92120	5.94236	5.95103	5.94141	5.96785	5.98803	5.99621	6.01218	6.03258	6.04922	6.07405	6.04801	6.01200	6.03995	5.91411	5.87867	
Comp	-1	-5.981156225	5.87897	5.89126	5.91461	5.94812	5.95647	5.93136	5.85609	5.80525	5.60485	5.51511	6.03407	6.04861	6.11217	6.09121	6.05234	6.04411	6.03795	5.94446	5.87660	
Comp	-2	-2.651212729	5.89621	5.90426	5.92436	5.94953	5.95973	5.94936	5.92436	5.87446	5.64446	5.54446	6.06446	6.07446	6.08446	6.09446	6.10446	6.11446	6.12446	6.13446	5.84446	
Comp	-2	-2.684556905	5.91132	5.92428	5.95230	5.96901	5.99176	5.96469	5.90069	5.80739	5.63647	5.49005	6.07771	6.09135	6.14554	6.12014	6.10178	6.07566	6.06963	5.97217	5.92624	
Comp	-2	-2.677322026	5.92578	5.94026	5.96867	5.98905	5.96869	5.90001	5.80595	5.63354	5.49262	6.04905	6.07031	6.10880	6.15063	6.13848	6.11998	6.09503	6.08034	5.99039	5.94304	
Comp	-2	-2.690117005	5.94315	5.95646	5.98485	5.99133	6.02380	5.99612	6.02438	6.04879	6.07122	6.08275	6.09781	6.11899	6.14689	6.15846	6.14236	6.11661	6.11451	6.01606	5.95869	
Comp	-1	-2.920384857	5.95996	5.96996	6.00879	6.02879	6.04139	6.04139	6.04139	6.04139	6.04139	6.04139	6.04139	6.04139	6.04139	6.04139	6.04139	6.04139	6.04139	6.04139	5.84139	
Comp	-2	-2.953326909	5.97419	5.98765	6.01519	6.04407	6.04743	6.02840	6.05779													

B.3 – Measurement & Units

Measurements			Supplied Data																				
1	Bore (m)	0.06634000	N Fuel consumption Torque, measured T_environment T_airbox T_crankcase T_exhaustgas IMEP MEP Ignition MBF50%	[rpm]	2000	2500	3000	3500	4000	4500	5000	5500	6000	6500	7000	7500	8000	8500	9000	9250	9500	9750	10000
2	1/2 Bore (m)	0.03317000		[kg/h]	2.07	2.1	2.23	2.54	2.77	3.37	4.26	4.82	5.77	6.24	6.64	6.66	6.58	7.25	8.2	7.9	6.97	6.12	5.42
3	Bore Area (m²)	0.00345653		[-]	0.852	0.895	0.935	0.951	0.942	0.895	0.644	0.866	0.789	0.776	0.736	0.756	0.822	0.763	0.761	0.789	0.870	0.903	0.950
4	Stroke (m)	0.07200000		[°C]	22.3	22.3	22.3	22.3	22.3	22.3	22.3	22.3	22.3	22.3	22.3	22.3	22.3	22.3	22.3	22.3	22.3	22.3	22.3
5	R (m)	0.03600000		[°C]	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28
6	Rod Length (m)	0.12500000		[°C]	57.62	57.16	57.05	56.63	56.7	56.92	56.96	56.65	57.15	57.39	57.59	57.66	57.94	58.41	58.88	59.35	59.82	60.34	61.19
7	P (l)	3.14159265		[°C]	315	299	304	333	360	377	383	397	413	427	454	485	523	548	542	539	520	504	504
8	Head Volume (V2) (m³)	0.00002350		[bar]	4.37	4.43	4.87	5.17	5.44	5.19	5.55	5.71	5.6	5.53	4.73	5.31	5.91	5.84	6.29	6.05	5.65	4.43	4.06
9	Volume at BDC (V1) (m³)	0.00027237		[bar]	3.99	4.09	4.54	4.81	5.06	4.79	5.13	5.25	5.11	5.05	4.23	4.76	5.31	5.13	5.51	5.24	4.84	3.53	3
10	Geometric Compression Ratio	11.59023141		[°CA]	-8.4	-13	-12.9	-11.9	-11.9	-12.2	-12.8	-11.5	-13	-12.9	-10.7	-10	-8.5	-7.9	-8.9	-12.6	-17.8	-20.2	-23.5
11	Combustion Chamber Volume	0.00003050	[°CA]	11.9	8	7.8	10	11	12.8	12.1	13.7	15.3	15.6	15.4	20.7	19	19.8	17.3	13.6	11.5	8.8	8	
12	Displacement m³	0.00024887	0.00025																				
13	Piston Dome Volume (m³)	0.00000700			0.38	0.34	0.33	0.36		0.38	0.4	0.42	0.46	0.49	0.48	0.5	0.55	0.6	0.71	0.78	0.81	0.81	0.9
14	Piston Position Tolerance m	0.00000010																					
15	Volume Change @ Max Negative Position	0.00000000			4.379	4.432	4.875	5.17728395	5.443471698	5.195	5.556	5.721	5.613	5.545	4.748	5.328	5.924	5.855	6.301	6.065	5.664	4.441	4.067
16	Piston Weight inc Rings (Kg)	0.22400000			2000	2500	3000	3500	4000	4500	5000	5500	6000	6500	7000	7500	8000	8500	9000	9250	9500	9750	10000
17	Gudgeon Pin Weight inc Circlips (KG)	0.05500000			588	566	577	606		633	650	656	670	686	700	727	758	796	821	827	815	812	793
18	"psi"	3.14159265																					
19	Air Density KG/m³	1.20000000																					
20																							
21																							
22																							
23	RPM	RAD/S	RPM/SEC	SEC/RPM	SEC/DEG	IMEP Comparison (bar)																	
24	2000	209.4395102	33.33333333	0.03	8.33333E-05	4.37																	
25	2500	261.7993878	41.66666667	0.024	6.66667E-05	4.399																	
26	3000	314.1592654	50	0.02	5.55556E-05	0.029																	
27	3500	366.5191429	58.33333333	0.017142857	4.7619E-05	0.022																	
28	4000	418.8790205	66.66666667	0.015	4.16667E-05	0.020																	
29	4500	471.238898	75	0.013333333	3.7037E-05	0.009276724																	
30	5000	523.598756	83.33333333	0.012	3.33333E-05	0.009276724																	
31	5500	575.9586532	91.66666667	0.010509091	3.0303E-05	0.005																	
32	6000	628.3185307	100	0.01	2.77778E-05	0.004																	
33	6500	680.6784083	108.3333333	0.009230769	2.5641E-05	0.003																	
34	7000	733.0362858	116.6666667	0.008571429	2.38095E-05	0.002																	
35	7500	785.3981634	125	0.008	2.22222E-05	0.001																	
36	8000	837.7580413	133.3333333	0.0075	2.06333E-05	0.0005																	
37	8500	890.1179185	141.6666667	0.007058824	1.96078E-05	0.00025																	
38	9000	942.4779616	150	0.006666667	1.85185E-05	0.0001																	
39	9250	968.6577349	154.1666667	0.006484846	1.8018E-05	0.00005																	
40	9500	994.8376736	158.3333333	0.006315789	1.75439E-05	0.000025																	
41	9750	1021.017612	162.5	0.006153984	1.7094E-05	0.0000125																	
42	10000	1047.197551	166.6666667	0.006	1.66667E-05	0.00000625																	

120

B.6 – Graphs

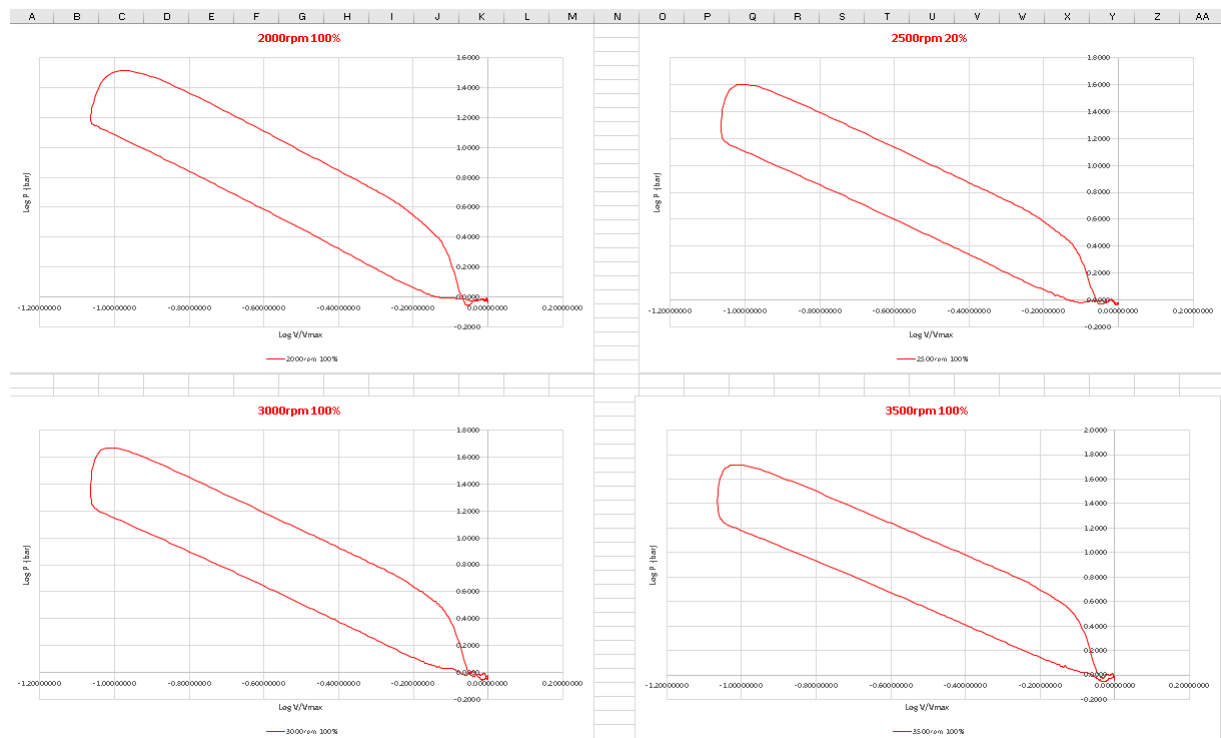


Figure 124 - Excel Log PV Graphs (Vacas, Authors Own)

B.7 – Combustion Timing Experiment

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T
1	Pressure Difference Method																			
2	IGN (BTDC) - Prescribed	2000	2500	3000	3500	4000	4500	5000	5500	6000	6500	7000	7500	8000	8500	9000	9250	9500	9750	10000
3	SOC	-8.4	-8	-13	-8	-7	-7	-7	-7	-8	-13	-8	-10	-2	0	-3	-7	-12	-14	-16
4	EOC	42	39	36	36	40	40	43	41	46	46	56	54	48	51	48	46	42	40	39
5	Duration	47	47	44	43	47	47	50	48	54	54	60	56	48	51	51	53	54	54	55
6	Delay	3.4	5	4.9	4.9	4.9	5.2	5.8	4.5	5	4.9	6.7	8	8.5	7.9	5.9	5.6	5.8	6.2	7.5
7	Time																			
8	Time Elapsed (Delay Period)	0.0002633	0.0003333	0.0002722	0.0002333	0.0002042	0.0001926	0.0001933	0.0001964	0.0001935	0.0001256	0.0001535	0.0001776	0.0001771	0.0001549	0.0001093	0.0001009	0.0001016	0.000106	0.000125
9	Visual Method																			
10	IGN (BTDC) - Prescribed	2000	2500	3000	3500	4000	4500	5000	5500	6000	6500	7000	7500	8000	8500	9000	9250	9500	9750	10000
11	SOC	-8.4	-8	-13	-12.9	-11.9	-12.2	-12.8	-11.5	-13	-12.9	-10.7	-10	-8.5	-7.9	-8.9	-12.6	-17.8	-20.2	-23.5
12	EOC	28	22	23	27	28	28	29	32	33	43	41	33	37	37	30	34	30	28	10
13	Duration	31	28	29	31	32	32	31	32	35	36	47	43	31	40	40	33	41	36	38
14	Delay	5.4	7	6.9	7.9	7.9	8.2	9.8	8.5	10	9.9	6.7	8	10.5	4.9	5.9	9.6	10.8	14.2	13.5
15	Time																			
16	Time Elapsed (Delay Period)	0.00045	0.0004667	0.0003833	0.0003762	0.0003232	0.0003037	0.0003267	0.0002576	0.0002778	0.0002538	0.0001955	0.0001778	0.0002186	9.608E-05	0.0001093	0.000173	0.0001695	0.0002427	0.000225
17	Time																			
18	Time Elapsed (Duration of Combustion)	0.0025633	0.0018667	0.0016111	0.0014762	0.0013333	0.0011852	0.0010333	0.0009697	0.0009722	0.0009231	0.001119	0.0009556	0.0006458	0.0007843	0.0007407	0.0005946	0.0007193	0.0006154	0.0006333
19	"n - comp"																			
20	Pressure Difference	1.2201812	1.2373414	1.23328	1.237231	1.2413856	1.2484464	1.2466723	1.2426739	1.2416412	1.2571487	1.2434567	1.3000805	1.161549	1.2916188	1.2961408	1.3134651	1.2862558	1.3092326	1.2851436
21	Visual	1.2123758	1.2319632	1.2365885	1.2286624	1.2330506	1.2408895	1.2346738	1.2391262	1.2380418	1.2430539	1.2434567	1.3000805	1.3633551	1.2735957	1.2461983	1.2612378	1.2731623	1.2743736	1.2952106
22	"n - exp"																			
23	Pressure Difference	1.3233528	1.3203294	1.3102596	1.3083279	1.3113695	1.3152307	1.3227122	1.3107309	1.3252725	1.3146489	1.2866334	1.3040874	1.3467489	1.3346885	1.3333951	1.3464407	1.3004308	1.28858	1.3063716
24	Visual	1.2951411	1.2803533	1.2960899	1.3009572	1.2950777	1.282647	1.280432	1.2678226	1.2754769	1.2775612	1.2268379	1.2475716	1.2398504	1.2548794	1.2948242	1.2648145	1.2728466	1.2680068	1.2525881

Figure 125 - Excel Combustion Timing Experiment Sheet (Vacas, Authors Own)

B.8 – Versioning

Name	Status	Date modified	Type	Size
 250EXC_Burning_Data v2	✓	29/11/2017 19:47	Microsoft Excel W...	2,620 KB
 250EXC_Burning_Data	✓	26/10/2017 18:39	Microsoft Excel W...	512 KB
 250EXC_Burning_Data_KTM_Original	✓	28/04/2018 20:48	Microsoft Excel W...	118 KB
 250EXC_Combustion_Data	✓	14/12/2017 08:51	Microsoft Excel W...	4,706 KB
 250EXC_Combustion_Data_23_11_17	✓	23/11/2017 17:12	Microsoft Excel W...	3,812 KB
 250EXC_Combustion_Data_29_11_17	✓	29/11/2017 19:18	Microsoft Excel W...	3,829 KB
 250EXC_Combustion_Modelling	✓	23/11/2017 15:57	Microsoft Excel W...	2,609 KB
 High Load	✓	07/12/2017 11:50	Microsoft Excel W...	65 KB
 Low Load Update 070218	✓	07/02/2018 20:38	Microsoft Excel W...	1,967 KB
 Low Load Update160218	✓	16/02/2018 18:17	Microsoft Excel W...	2,996 KB
 Low Load Update200218	✓	20/02/2018 03:01	Microsoft Excel W...	3,306 KB
 Low Load	✓	01/03/2018 03:44	Microsoft Excel W...	429 KB

Figure 126 - Excel File Versioning (before OneDrive) (Vacas, Authors Own)

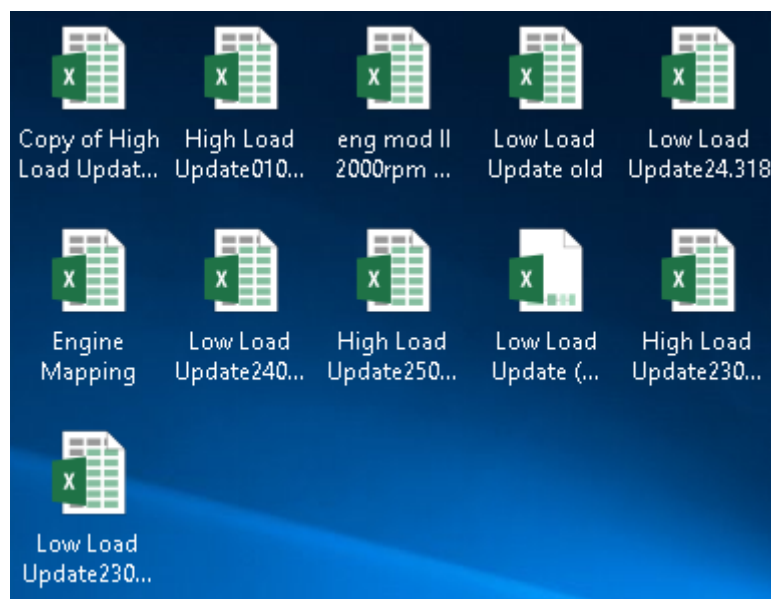


Figure 127 - Excel File Versioning (before OneDrive) (Vacas, Authors Own)

B.9 – Low Load / High Load

P	Q	R	S	T	U	V	W	X	Y	Z	AA	AB	AC	AD	AE	AF	AG	AH	AI
Crank Angle (degs)	Pressure (bar) -20%																		
	2000 rpm	2500 rpm	3000 rpm	3500 rpm	4000 rpm	4500 rpm	5000 rpm	5500 rpm	6000 rpm	6500 rpm	7000 rpm	7500 rpm	8000 rpm	8500 rpm	9000 rpm	9250 rpm	9500 rpm	9750 rpm	10000 rpm
-180	0.912	0.908	0.885	0.823	0.807	0.836	0.875	0.878	0.845	0.783	0.776	0.734	0.691	0.724	0.72	0.722	0.826	0.778	0.906
-179	0.913	0.908	0.889	0.829	0.803	0.889	0.873	0.876	0.847	0.775	0.771	0.727	0.68	0.716	0.718	0.735	0.841	0.76	0.908
-178	0.912	0.914	0.894	0.836	0.798	0.876	0.87	0.874	0.844	0.767	0.769	0.722	0.674	0.716	0.716	0.745	0.834	0.766	0.909
-177	0.919	0.915	0.9	0.843	0.796	0.87	0.867	0.875	0.844	0.762	0.764	0.715	0.674	0.709	0.72	0.759	0.84	0.778	0.914
-176	0.925	0.923	0.9	0.949	0.794	0.863	0.864	0.873	0.847	0.755	0.759	0.709	0.681	0.703	0.72	0.762	0.843	0.772	0.91
-175	0.934	0.923	0.893	0.855	0.787	0.858	0.855	0.872	0.844	0.751	0.762	0.701	0.677	0.701	0.717	0.757	0.846	0.763	0.905
-174	0.941	0.922	0.899	0.861	0.784	0.852	0.847	0.864	0.846	0.751	0.757	0.701	0.679	0.694	0.715	0.765	0.853	0.766	0.91
-173	0.948	0.921	0.896	0.868	0.783	0.847	0.843	0.86	0.844	0.755	0.759	0.698	0.69	0.678	0.716	0.759	0.849	0.768	0.917
-172	0.954	0.919	0.9	0.876	0.783	0.843	0.839	0.858	0.843	0.763	0.75	0.693	0.691	0.683	0.714	0.743	0.845	0.769	0.92
-171	0.953	0.915	0.897	0.882	0.785	0.841	0.831	0.86	0.838	0.769	0.749	0.692	0.679	0.672	0.714	0.736	0.845	0.767	0.927
-170	0.965	0.911	0.9	0.886	0.794	0.841	0.825	0.850	0.83	0.776	0.749	0.693	0.677	0.676	0.706	0.735	0.841	0.765	0.93
-169	0.968	0.909	0.903	0.892	0.799	0.844	0.817	0.852	0.824	0.78	0.746	0.692	0.679	0.698	0.7	0.733	0.833	0.769	0.93
-168	0.973	0.904	0.903	0.899	0.811	0.84	0.811	0.85	0.823	0.789	0.747	0.698	0.675	0.755	0.692	0.732	0.827	0.759	0.916
-167	0.971	0.902	0.9	0.901	0.825	0.84	0.805	0.848	0.815	0.794	0.749	0.697	0.679	0.801	0.678	0.732	0.821	0.757	0.907
-166	0.969	0.899	0.895	0.892	0.842	0.836	0.799	0.84	0.809	0.739	0.755	0.695	0.679	0.626	0.67	0.723	0.816	0.758	0.91
-165	0.96	0.901	0.895	0.889	0.855	0.836	0.797	0.833	0.797	0.8	0.75	0.694	0.667	0.632	0.667	0.709	0.806	0.749	0.909
-164	0.949	0.9	0.893	0.889	0.871	0.834	0.794	0.826	0.801	0.796	0.752	0.696	0.673	0.629	0.664	0.704	0.804	0.738	0.907
-163	0.943	0.908	0.889	0.887	0.881	0.846	0.789	0.817	0.796	0.735	0.752	0.701	0.684	0.625	0.658	0.692	0.804	0.733	0.898
-162	0.943	0.912	0.889	0.885	0.887	0.841	0.792	0.811	0.791	0.736	0.748	0.701	0.654	0.785	0.655	0.695	0.804	0.735	0.889
-161	0.943	0.921	0.882	0.882	0.9	0.85	0.792	0.804	0.789	0.733	0.755	0.704	0.644	0.74	0.651	0.704	0.806	0.728	0.87
-160	0.948	0.933	0.878	0.883	0.911	0.859	0.789	0.801	0.78	0.787	0.755	0.71	0.641	0.733	0.686	0.707	0.808	0.721	0.899
-159	0.951	0.94	0.875	0.886	0.922	0.869	0.788	0.791	0.773	0.762	0.752	0.709	0.646	0.776	0.662	0.708	0.798	0.721	0.898
-158	0.957	0.951	0.874	0.887	0.929	0.872	0.788	0.785	0.767	0.776	0.748	0.706	0.647	0.805	0.654	0.704	0.794	0.723	0.896
-157	0.959	0.962	0.874	0.888	0.934	0.876	0.792	0.785	0.764	0.769	0.747	0.706	0.653	0.808	0.651	0.705	0.8	0.719	0.846
-156	0.964	0.97	0.875	0.89	0.934	0.882	0.789	0.785	0.766	0.768	0.748	0.705	0.659	0.796	0.661	0.71	0.8	0.718	0.836
-155	0.961	0.975	0.874	0.892	0.934	0.889	0.801	0.78	0.759	0.764	0.742	0.7	0.662	0.795	0.664	0.711	0.733	0.705	0.828
-154	0.961	0.981	0.873	0.894	0.935	0.895	0.802	0.787	0.752	0.764	0.744	0.703	0.652	0.788	0.663	0.707	0.733	0.702	0.825
-153	0.959	0.985	0.874	0.894	0.933	0.904	0.81	0.783	0.749	0.751	0.741	0.705	0.65	0.782	0.655	0.703	0.736	0.71	0.842
-152	0.954	0.984	0.88	0.89	0.931	0.91	0.817	0.782	0.752	0.749	0.745	0.707	0.646	0.773	0.654	0.71	0.737	0.707	0.81
-151	0.953	0.984	0.884	0.883	0.93	0.917	0.83	0.789	0.754	0.743	0.74	0.705	0.639	0.772	0.661	0.712	0.737	0.71	0.813
-150	0.948	0.978	0.891	0.878	0.929	0.927	0.844	0.791	0.763	0.732	0.768	0.701	0.639	0.765	0.676	0.714	0.786	0.717	0.824
-149	0.947	0.969	0.895	0.877	0.929	0.931	0.851	0.799	0.777	0.734	0.778	0.702	0.642	0.766	0.676	0.717	0.787	0.715	0.832
-148	0.944	0.957	0.91	0.874	0.929	0.932	0.861	0.804	0.769	0.731	0.768	0.693	0.639	0.779	0.681	0.717	0.785	0.712	0.835
-147	0.946	0.948	0.92	0.871	0.923	0.93	0.866	0.815	0.795	0.728	0.751	0.686	0.637	0.799	0.733	0.725	0.785	0.705	0.836
-146	0.946	0.939	0.921	0.87	0.92	0.931	0.874	0.827	0.813	0.73	0.794	0.678	0.626	0.626	0.801	0.737	0.786	0.697	0.832
-145	0.95	0.937	0.93	0.874	0.918	0.928	0.882	0.842	0.828	0.729	0.739	0.677	0.624	0.834	0.862	0.735	0.789	0.701	0.826
-144	0.95	0.939	0.945	0.869	0.915	0.928	0.895	0.857	0.839	0.732	0.739	0.68	0.626	0.817	0.863	0.731	0.791	0.698	0.832
-143	0.945	0.935	0.955	0.869	0.907	0.927	0.91	0.874	0.845	0.734	0.805	0.682	0.627	0.803	0.868	0.725	0.796	0.702	0.835
-142	0.943	0.941	0.964	0.868	0.9	0.923	0.93	0.889	0.848	0.741	0.816	0.68	0.63	0.797	0.834	0.724	0.803	0.703	0.84
-141	0.943	0.944	0.973	0.869	0.893	0.923	0.945	0.894	0.865	0.741	0.822	0.677	0.635	0.791	0.923	0.727	0.818	0.702	0.84
-140	0.943	0.953	0.981	0.873	0.886	0.923	0.958	0.899	0.873	0.746	0.824	0.678	0.64	0.786	0.861	0.728	0.828	0.714	0.841
-139	0.939	0.957	0.987	0.877	0.875	0.925	0.97	0.906	0.887	0.752	0.825	0.682	0.638	0.788	0.862	0.733	0.836	0.722	0.836
-138	0.941	0.961	0.99	0.884	0.864	0.924	0.968	0.913	0.895	0.77	0.829	0.689	0.633	0.789	0.851	0.736	0.841	0.729	0.84
-137	0.942	0.962	0.988	0.887	0.856	0.923	0.973	0.916	0.897	0.761	0.842	0.707	0.623	0.794	0.925	0.744	0.849	0.728	0.842
-136	0.943	0.962	0.986	0.895	0.854	0.924	0.972	0.935	0.915	0.794	0.853	0.723	0.622	0.793	0.949	0.758	0.863	0.733	0.839
-135	0.945	0.963	0.982	0.908	0.853	0.926	0.975	0.95	0.925	0.803	0.868	0.733	0.626	0.8	0.939	0.744	0.863	0.734	0.84
-134	0.949	0.96	0.979	0.914	0.855	0.925	0.981	0.966	0.931	0.805	0.884	0.76	0.629	0.801	0.923	0.734	0.853	0.728	0.841
-133	0.956	0.96	0.97	0.916	0.856	0.93	0.978	0.974	0.942	0.814	0.9	0.798	0.644	0.802	0.921	0.747	0.863	0.723	0.843

Figure 128 - Excel Low Load Example (20%) (Vacas, Authors Own)

P	Q	R	S	T	U	V	W	X	Y	Z	AA	AB	AC	AD	AE	AF	AG	AH	AI
Crank Angle (degs)	2000 rpm	2500 rpm	3000 rpm	3500 rpm	4000 rpm	4500 rpm	5000 rpm	5500 rpm	6000 rpm	6500 rpm	7000 rpm	7500 rpm	8000 rpm	8500 rpm	9000 rpm	9250 rpm	9500 rpm	9750 rpm	10000 rpm
-180	0.9300	0.9320	0.9320	0.9120	0.9520	1.0370	1.0940	1.1670	1.0880	1.0890	1.0700	1.1110	1.1630	1.0510	1.0930	1.0500	1.0630	1.1470	1.1590
-179	0.9120	0.9340	0.9280	0.9120	0.9400	1.0270	1.0900	1.1670	1.0780	1.0870	1.0950	1.1070	1.1520	1.0600	1.0990	1.0170	1.0990	1.1360	1.1570
-178	0.9110	0.9360	0.9350	0.9100	0.9320	1.0130	1.0770	1.1640	1.0870	1.0770	1.0850	1.0980	1.1480	1.0550	1.0980	1.0260	1.1030	1.1490	1.1590
-177	0.9140	0.9370	0.9450	0.9170	0.9260	1.0020	1.0680	1.1650	1.0960	1.0660	1.0760	1.1010	1.1270	1.0510	1.1210	1.0410	1.1130	1.1530	1.1660
-176	0.9150	0.9430	0.9430	0.9200	0.9190	0.9910	1.0610	1.1660	1.0960	1.0590	1.0690	1.0910	1.1150	1.0540	1.1000	1.0490	1.1230	1.1480	1.1600
-175	0.9210	0.9470	0.9490	0.9290	0.9110	0.9800	1.0490	1.1600	1.0980	1.0540	1.0590	1.0850	1.1110	1.0370	1.0980	1.0440	1.1160	1.1370	1.1600
-174	0.9290	0.9490	0.9420	0.9290	0.9020	0.9740	1.0410	1.1530	1.1040	1.0580	1.0570	1.0820	1.1020	1.0250	1.1010	1.0520	1.1100	1.1440	1.1580
-173	0.9370	0.9490	0.9400	0.9370	0.8950	0.9630	1.0260	1.1460	1.1060	1.0720	1.0520	1.0780	1.0940	1.0150	1.0960	1.0330	1.1070	1.1490	1.1600
-172	0.9480	0.9490	0.9380	0.9410	0.8830	0.9490	1.0190	1.1190	1.1100	1.0840	1.0490	1.0810	1.0990	1.0040	1.0770	1.0230	1.1100	1.1410	1.1480
-171	0.9550	0.9570	0.9380	0.9430	0.8820	0.9440	1.0100	1.1230	1.0960	1.0880	1.0520	1.0730	1.0960	1.0060	1.0690	1.0100	1.1180	1.1290	1.1290
-170	0.9530	0.9540	0.9400	0.9470	0.8840	0.9490	1.0120	1.1250	1.0980	1.0900	1.0540	1.0750	1.0940	1.0040	1.0670	1.0110	1.1190	1.1310	1.1340
-169	0.9700	0.9270	0.9340	0.9570	0.8730	0.9360	0.9810	1.1150	1.0790	1.1000	1.0460	1.0590	1.0900	0.9920	1.0670	1.0150	1.1060	1.1440	1.1410
-168	0.9750	0.9220	0.9330	0.9670	0.8720	0.9300	0.9660	1.1050	1.0750	1.0970	1.0490	1.0530	1.0900	0.9860	1.0590	1.0020	1.1010	1.1340	1.1480
-167	0.9800	0.9180	0.9390	0.9670	0.8810	0.9250	0.9560	1.0980	1.0670	1.0970	1.0540	1.0480	1.0850	0.9860	1.0500	0.9890	1.1080	1.1320	1.1420
-166	0.9780	0.9130	0.9300	0.9680	0.8890	0.9180	0.9410	1.0940	1.0560	1.0890	1.0560	1.0410	1.0830	0.9660	1.0270	0.9790	1.0950	1.1320	1.1400
-165	0.9770	0.9160	0.9210	0.9690	0.8920	0.9140	0.9380	1.0910	1.0570	1.0880	1.0570	1.0400	1.0820	0.9670	1.0260	0.9820	1.1040	1.1310	1.1400
-164	0.9690	0.9130	0.9250	0.9640	0.9080	0.9090	0.9180	1.0660	1.0460	1.0700	1.0500	1.0430	1.0780	0.9620	1.0160	0.9620	1.0790	1.1120	1.1450
-163	0.9550	0.9140	0.9160	0.9620	0.9170	0.9140	0.9090	1.0490	1.0410	1.0610	1.0530	1.0410	1.0740	1.0030	1.0050	0.9600	1.0790	1.1060	1.1390
-162	0.9460	0.9180	0.9110	0.9580	0.9300	0.9020	0.9050	1.0360	1.0350	1.0600	1.0570	1.0400	1.0650	1.0180	1.0310	0.9560	1.0700	1.0980	1.1360
-161	0.9430	0.9250	0.9040	0.9520	0.9410	0.9040	0.8950	1.0230	1.0250	1.0420	1.0530	1.0450	1.0680	1.0150	1.0220	0.9700	1.0620	1.0910	1.1350
-160	0.9420	0.9350	0.8950	0.9450	0.9520	0.9040	0.8870	1.0110	1.0170	1.0340	1.0560	1.0570	1.0670	1.0240	1.0190	0.9760	1.0630	1.0980	1.1290
-159	0.9450	0.9430	0.8920	0.9430	0.9610	0.9000	0.8790	1.0010	1.0200	1.0490	1.0580	1.0660	1.0500	1.0200	1.0260	0.9670	1.0700	1.1050	1.1250
-158	0.9460	0.9580	0.8950	0.9440	0.9750	0.9180	0.8770	0.9980	1.0210	1.0470	1.0610	1.0550	1.0450	1.0210	1.0260	0.9580	1.0920	1.1240	1.1280
-157	0.9520	0.9700	0.8890	0.9420	0.9900	0.9190	0.8680	0.9630	0.9790	1.0200	1.0490	1.0620	1.0700	1.0450	1.0280	0.9650	1.0530	1.0850	1.1240
-156	0.9560	0.9820	0.8860	0.9380	1.0000	0.9260	0.8640	0.9820	1.0040	1.0100	1.0360	1.0640	1.0640	1.0320	1.0270	0.9650	1.0530	1.0820	1.1190
-155	0.9590	0.9930	0.8890	0.9330	1.0090	0.9360	0.8610	0.9640	0.9560	1.0060	1.0310	1.0660	1.0580	1.0250	1.0160	0.9640	1.0450	1.0710	1.1050
-154	0.9620	0.9920	0.8930	0.9300	1.0020	0.9400	0.8620	0.9490	0.9600	1.0070	1.0340	1.0680	1.0680	1.0300	1.0200	0.9650	1.0460	1.0750	1.1080
-153	0.9640	0.9950	0.8970	0.9250	1.0130	0.9510	0.8610	0.9460	0.9520	1.0090	1.0360	1.0700	1.0680	1.0320	1.0200	0.9570	1.0460	1.0760	1.1100
-152	0.9640	1.0100	0.8830	0.9290	1.0190	0.9570	0.8660	0.9470	0.9120	0.9810	1.0300	1.0570	1.0490	0.9670	1.0100	0.9550	1.0410	1.0750	1.1070
-151	0.9590	1.0110	0.8850	0.9160	1.0180	0.9690	0.8720	0.9330	0.9050	0.9720	1.0040	1.0360	1.0450	0.9710	1.0070	0.9500	1.0430	1.0700	1.1050
-150	0.9580	1.0080	0.8890	0.9070	1.0160	0.9780	0.8820	0.9250	0.8970	0.9505	0.9770	1.0380	1.0330	0.9750	1.0020	0.9410	1.0370	1.0650	1.1040
-149	0.9540	1.0020	0.8970	0.9020	1.0210	0.9870	0.8850	0.9190	0.9540	0.9640	1.0200	1.0230	1.0230	0.9690	0.9940	0.9440	1.0340	1.0680	1.1050
-148	0.9570	0.9920	0.9020	0.9020	1.0230	0.9890	0.8920	0.9250	0.9520	0.9630	1.0200	1.0230	1.0230	0.9870	0.9920	0.9490	1.0360	1.0690	1.1060
-147	0.9540	0.9820	0.9240	0.8870	1.0170	1.0090	0.8970	0.9020	0.8790	0.9420	0.9410	1.0020	1.0040	0.9910	0.9910	0.9270	1.0320	1.0460	1.0840
-146	0.9550	0.9720	0.9250	0.8810	1.0140	1.0100	0.9030	0.8980	0.8720	0.9290	0.9320	0.9980	1.0010	0.9820	0.9820	0.9290	1.0260	1.0310	1.0750
-145	0.9550	0.9640	0.9350	0.8840	1.0090	1.0130	0.9220	0.8980	0.8700	0.9240	0.9220	0.9880	0.9940	0.9590	0.9740	0.9190	1.0210	1.0360	1.0640
-144	0.9520	0.9630	0.9520	0.8830	1.0060	1.0130	0.9380	0.9030	0.8630	0.9170	0.9090	0.9810	0.9760	0.9590	0.9710	0.9280	1.0210	1.0310	1.0670
-143	0.9540	0.9610	0.9590	0.8840	1.0060	1.0130	0.9450	0.9080	0.8780	0.9290	0.9200	0.9890	0.9840	0.9660	0.9760	0.9300	1.0240	1.0340	1.0680
-142	0.9530	0.9640	0.9780	0.8800	1.0060	1.0100	0.9550	0.9050	0.8590	0.9090	0.8940	0.9690	0.9700	0.9320	0.9780	0.9390	1.0260	1.0160	1.0400
-141	0.9480	0.9670	0.9970	0.8800	0.9980	1.0120	0.9620	0.9120	0.8600	0.9030	0.8880	0.9560	0.9660	0.9360	0.9730	0.9250	1.0190	1.0150	1.0410

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Nomenclature & Abbreviations

Nomenclature & Abbreviations		
Symbol / Acronym	Definition	Unit / Value
θ_R	<i>Crank Angle</i>	Radians
θ_D	<i>Crank Angle</i>	Degrees
R	<i>Crank Radius</i>	m
L	<i>Rod Length (eye to eye)</i>	m
π	<i>Pi</i>	3.14159265359 *
θ_{rod}	<i>Rod Angle</i>	
V_C	<i>Combustion Chamber Volume TDC</i>	m ³
ϕ_{Bore}	<i>Bore Diameter</i>	m ³
V_{SV}	<i>Swept Volume</i>	m ³
V_{CYL}	<i>Cylinder Volume</i>	m ³
$V_{SV\theta}$	<i>Swept Volume Relative to Crank Angle</i>	m ³
$V_{CYL(BDC)}$	<i>Cylinder Volume BDC</i>	m ³
TDC	<i>Top Dead Centre</i>	
BDC	<i>Bottom Dead Centre</i>	
W	<i>Work</i>	
F	<i>Force</i>	
x	<i>Distance</i>	
P	<i>Pressure</i>	
W_θ	<i>Work Per Degree</i>	J
P_{bar}	<i>Pressure</i>	bar
P_{Pa}	<i>Pressure</i>	Pascals
n	<i>Polytropic Exponent</i>	
SOC	<i>Start of Combustion</i>	
EOC	<i>End of Combustion</i>	
MFB	<i>Mass Fraction Burnt</i>	
m	<i>Vibe Function Exponent</i>	
a	<i>Vibe Function Exponent</i>	
θ_d	<i>Sum of Crank angle from SOC to EOC</i>	
N	<i>Revolutions Per Minute</i>	
m_f	<i>Fuel Mass Per Revolution</i>	KG
f_c	<i>Fuel Consumption</i>	KG/Hr
m_a	<i>Air Mass Per Revolution</i>	KG/Hr
AFR_{stoich}	<i>Stoichiometric Air Fuel Ratio</i>	
ϕ	<i>Fuel Equivalence Ratio</i>	
λ	<i>Lambda</i>	
η_v	<i>Volumetric Efficiency</i>	%
ρ_{at}	<i>Atmospheric Pressure</i>	Pa
R_u	<i>Universal Gas Constant</i>	8314 J/KG. K
m_{ta}	<i>Mass of Air + Fuel</i>	KG
Π_{charge}	<i>Charge Purity</i>	
m_{rg}	<i>Mass of Residual Gas</i>	KG

m_{tr}	Mass of Trapped Charge	KG
$MW_{mix_{avg}}$	Average Molecular Weight of Mixture	g/mol
MW_{mix}	Molecular Weight of Mixture	g.mol
n_{mix}	Molecular Count of Mixture	g.mol
$MW_{exh_{avg}}$	Average Molecular Weight of Exhaust Gas	g.mol
MW_{exh}	Molecular Weight of Exhaust Gas	g.mol
n_{exh}	Molecular Count of Exhaust Gas	g.mol
R_{tr}	Specific Gas Constant of Trapped Charge	J/kg K ⁻¹
Π_{exh}	In Cylinder Exhaust Gas Purity	$1 - \Pi_{charge}$
n	Number of Moles	
M	Molecular Weight	KG
T	Temperature	K
$\tilde{C}_{p,f}$	Specific Heat Constant Pressure (Fuel)	J/Kmol.K
t	JANAF Exponent	$\frac{T}{1000}$
A_f	Coefficients for Fuel Enthalpy and Specific Heat	
ΔP_{comb}	Change in pressure due to combustion	Pa
P_{2n}	Prise rise due to change in volume	Pa
$\tilde{C}_{v,i}$	Specific Heat Constant Volume (Species)	J/kg K ⁻¹
R_{avg}	Average Specific Gas Constant	J/kg K ⁻¹
R_{unb}	Specific Gas Constant Unburnt Mixture	J/kg K ⁻¹
R_{burn}	Specific Gas Constant of Burnt Mixture	J/kg K ⁻¹
$C_{p,f}$	Specific Heat Constant Pressure	j/Kmol.K
$mn_{C_8H_{18}}$	Molecular Count of Isooctane	g.mol
mn_{CO_2}	Molecular Count of Carbon Dioxide	g.mol
mn_{CO}	Molecular Count of Carbon Monoxide	g.mol
mn_{H_2O}	Molecular Count of Water	
mn_{N_2}	Molecular Count of Nitrogen Oxide	
$\tilde{C}_{pAV_{exh}}$	Average Specific Heat Constant Pressure (Exhaust)	
$C_{pAV_{exh}}$	Average Specific Heat Constant Pressure (Exhaust)	
$\tilde{C}_{vAV_{mix}}$	Average Specific Heat Constant Volume (Mixture)	j/Kmol.K
$C_{vAV_{mix}}$	Average Specific Heat Constant Volume (Mixture)	J/KG.K
$\tilde{C}_{vAV_{exh}}$	Average Specific Heat Constant Volume (Exhaust)	j/Kmol.K
$C_{vAV_{exh}}$	Average Specific Heat Constant Volume (Exhaust)	J/KG.K
Q_{mix}	Heat Needed For Temperature Rise (Mixture + 1K)	
T_{mix}	Temperature of Mixture	K
Q_{exh}	Heat Given Out Dropping 1K (Exhaust)	J
T_{exh}	Start Temperature of Exhaust Gas	K
X	Equilibrium Temperature Between Mixture + Exhaust Gas	K
T_{epc}	Temperature of in Cylinder Mixture at Exhaust Port Close	K
20%	Refers to throttle position 20% / low load	
100%	Refers to throttle position 100% / high load	
BTDC	Before Top Dead Centre	
ATDC	After Top Dead Centre	
TPO	Transfer Ports Open	

TPC	Transfer Ports Closed	
EPO	Exhaust Ports Open	
EPC	Exhaust Ports Closed	
BD	Blowdown (From EPO to TPO)	
ATAC*	Active Torque Amplification Chamber	
ATAC	Active Thermo Atmosphere Combustion	
HCCI	Homogeneous Charge Compression Ignition	
CAI	Controlled Auto-Ignition	
HCSI	Homogeneous Charge Spark Ignition	
HCSAAI	Homogeneous Charge Spark Assisted Auto-Ignition	
HCAI	Homogeneous Charge Auto-Ignition	
SCAI	Stratified Charge Auto-Ignition	
AR	Active Radical Combustion (Honda two-stroke CAI)	

Figures & Equations

Figure 1 - CAI Direct Injection (Ishibashi, et al., 1997) & (Nishida, et al., 2007)	9
Figure 2 - Exhaust Valve Comparison (Takashi, 2003)	14
Figure 3 - CRM250AR CAI Deceleration Device (Takashi, 2003)	14
Figure 4 - CAI Temperature (Ishibashi, 2000)	15
Figure 5 – EXP-2 Emissions & Fuel Consumption (Asai, et al., 1995)	15
Figure 6 - EXP-2 Torque Characteristics (Asai, et al., 1995).....	16
Figure 7 - IMEP Cyclic Variation (Asai & Ishibashi, 1996 International Congress & Exposition)	16
Figure 8 - CAI Exhaust Valve (Wakabayashi, et al., 1994)	17
Figure 9 - CAI Deceleration Device & Region (Asai, et al., 1995)	22
Figure 10 - CAI Transition (Onishi, et al., 1979)	23
Figure 11 - CAI AFR Effect (Ishibashi, 2000)	24
Figure 12 - Auto-Ignition Temperature (Ishibashi, 2000)	24
Figure 13 - Exhaust Butterfly Throttle (A) and ARC Valve (B) (Onishi, 1984) & (Ishibashi, 2000).....	25
Figure 14 - Honda Pantheon 125 AR (Honda Motor Co Ltd)	26
Figure 15 - Water Temperature Effect on CAI (Ishibashi, et al., 1994)	26
Figure 16 - NiCE Engine IFP (Zhao, 2007)	27
Figure 17 - CAI Process Flow Chart (Asai, et al., 1996)	28
Figure 18 - CT Scan KTM Cylinder (Vacas, Authors Own)	30
Figure 19 - Port Comparison & Cylinder Honda 250's (Vacas, Authors Own)	31
Figure 20 - KTM & Honda Porting Arrangements (Vacas, Authors Own)	32

Figure 21 - Cylinder Head Comparison (Vacas, Authors Own)	33
Figure 22 - Combustion Chamber Designs (Blair, 1994)	34
Figure 23 - CAI Photo Images (Lavy, et al., 2001).....	35
Figure 24 - CAI Photo Images (Ishibashi, et al., 1997).....	35
Figure 25 - Exhaust Valve Comparison (Vacas, Authors Own).....	36
Figure 26 – ATAC* Valve (Motorcity, 2010).....	37
Figure 27 - Crank Slider Diagram (Vacas, Authors Own).....	39
Figure 28 - Cylinder Pressure Comparison (Vacas, Authors Own).....	41
Figure 29 - Volumetric Efficiency (Vacas, Authors Own)	42
Figure 30 - PV Diagram 4000rpm 20% (Vacas, Authors Own)	43
Figure 31 - PV Diagram 2000rpm (Vacas, Authors Own)	44
Figure 32 - PV Diagram 8000rpm (Vacas, Authors Own)	44
Figure 33 - Work Done (Vacas, Authors Own)	45
Figure 34 - IMEP (Vacas, Authors Own)	46
Figure 35 - IMEP Error Analysis (Vacas, Authors Own)	47
Figure 36 - Ideal Cycles (Blair, 1994)	47
Figure 37 - LOG PV - 3000rpm 100%.....	48
Figure 38 - Log PV 10000rpm (Vacas, Authors Own).....	49
Figure 39 - Log PV 7500rpm Slow Combustion (Vacas, Authors Own).....	49
Figure 40 - Log PV Sensitivity Analysis (Vacas, Authors Own)	50
Figure 41 - "n" Per Degree (Vacas, Authors Own)	53
Figure 42 - Polytropic Index Compression (Vacas, Authors Own)	54
Figure 43 - Polytropic Index Expansion (Vacas, Authors Own).....	54
Figure 44 - "n" Per Degree SOC (Vacas, Authors Own).....	55
Figure 45 - Pressure Difference 2500rpm (Vacas, Authors Own)	56
Figure 46 - Pressure Difference Noise (Vacas, Authors Own).....	57
Figure 47 - Pressure Difference Port Activity (Vacas, Authors Own).....	57
Figure 48 - Pressure Difference SOC & EOC (Vacas, Authors Own).....	58
Figure 49 - Log PV (SOC & EOC) (Vacas, Authors Own)	59
Figure 50 - Polytropic Index Method Comparison (Compression – 100%) (Vacas, Authors Own)	60
Figure 51 - Polytropic Index Method Comparison (Compression - 20%).....	60
Figure 52 - Polytropic Index Method Comparison (Expansion - 100%) (Vacas, Authors Own)	61
Figure 53 - Polytropic Index Method Comparison (Expansion - 20%) (Vacas, Authors Own)	61
Figure 54 - Polytropic Index - 7000rpm 100% (Vacas, Authors Own).....	62
Figure 55 - "n-comp" Range (Vacas, Authors Own).....	62
Figure 56 - "n-comp" Focused Range Selection (Vacas, Authors Own)	63
Figure 57 - "n-exp" Range Selection (Vacas, Authors Own)	64
Figure 58 - "n-exp" Focused Range Selection (Vacas, Authors Own)	64
Figure 59 - Polytropic Index Sensitivity Analysis (Vacas, Authors Own)	65
Figure 60 - Polytropic Index Comparison (compression) (Vacas, Authors Own)	66
Figure 61 - Polytropic Index Comparison (expansion) (Vacas, Authors Own)	66
Figure 62 - KTM Ignition Timing (KTM, 2017)	67
Figure 63 - Delay 100% (Visual) (Vacas, Authors Own).....	68
Figure 64 - Delay Comparison (Vacas, Authors Own)	69
Figure 65 - Duration Comparison (Vacas, Authors Own).....	69
Figure 66 - MFB Method & Load Comparison (Vacas, Authors Own)	72
Figure 67 - MFB Sensitivity Analysis (Vacas, Authors Own).....	73
Figure 68 - MFB & Pressure Difference (Vacas, Authors Own).....	74

Figure 69 - MFB Comparison "n" Variation (Vacas, Authors Own).....	74
Figure 70 - Vibe Constants (Nierkerk, 2018).....	76
Figure 71 - Vibe Graph (Vacas, Authors Own)	78
Figure 72 - MFB Retro-Fit 20%.....	79
Figure 73 - MFB Retro-Fit 100%.....	79
Figure 74 - In Cylinder Mass (Vacas, Authors Own).....	82
Figure 75 - Specific Gas Constants (Vacas, Authors Own)	85
Figure 76 - Temperature Test 3000rpm - 100% (Vacas, Authors Own).....	85
Figure 77 - Exhaust Valve Positions Engmod2T (Vacas, Authors Own)	89
Figure 78 - Purity Comparison (Vacas, Authors Own).....	90
Figure 79 - Air & Exhaust Mass (Vacas, Authors Own)	90
Figure 80 - Exhaust Gas Temperatures (KTM, 2017)	92
Figure 81 - Heat Energy Balance (Vacas, Authors Own)	93
Figure 82 - Temperature at Exhaust Port Closure (Vacas, Authors Own).....	93
Figure 83 - Polytropic Index (compression) Expanded Range Selection - 20% (Vacas, Authors Own).....	95
Figure 84 - Polytropic Index (compression) Expanded Range Selection - 100% (Vacas, Authors Own)	96
Figure 85 - Temperature - "n" Variation Sensitivity (Vacas, Authors Own).....	96
Figure 86 - Temperature - 5° BTDC 20% (Vacas, Authors Own).....	98
Figure 87 - Temperature - 5° BTDC 100% (Vacas, Authors Own)	98
Figure 88 - Temperature - 5° BTDC Comparison (Vacas, Authors Own).....	99
Figure 89 - "n" Variation 20% (Vacas, Authors Own).....	99
Figure 90 - Software Package.....	102
Figure 91 - Dat2T Overview	102
Figure 92 - Dat2T Engine Data	103
Figure 93 - Dat2T Combustion Chamber 1.....	104
Figure 94 - Dat2T Combustion Chamber 2.....	104
Figure 95 - Dat2T Timing Calculator.....	105
Figure 96 - Dat2T Exhaust Port	105
Figure 97 - Dat2T Exhaust Port Display.....	106
Figure 98 - Dat2T Exhaust Valve Timing.....	106
Figure 99 - Dat2T Transfer Port Data Entry.....	107
Figure 100 - Dat2T Transfer Port Display.....	108
Figure 101 - Dat2T Window / Scavenging Calculator.....	109
Figure 102 - Dat2T Reed Valve Data Entry.....	110
Figure 103 - Dat2T Reed Valve Display	110
Figure 104 - Data2T Exhaust Data Entry	111
Figure 105 - Dat2T Exhaust Display.....	111
Figure 106 - Dat2T Inlet System Data Entry	112
Figure 107 - Dat2T Inlet System Display	112
Figure 108 - Dat2T Combustion Data Entry	113
Figure 109 - Dat2T Temperature Data Entry.....	114
Figure 110 - Engmod2T Start Screen.....	115
Figure 111 - Engmod2T Run Screen	115
Figure 112 - Engmod2T Exit Screen	116
Figure 113 - Post2T Plot Options	116
Figure 114 - Post2T Plot Display	117
Figure 115 - Post2T Excel File Import (Space De-limited file).....	117

Figure 116 - Post2T Excel Data Import Editing.....	118
Figure 117 - Post2T Excel Validation (Vacas, Authors Own)	118
Figure 118 - Original KTM Data (KTM, 2017)	119
Figure 119 - Original KTM Data (KTM, 2017)	119
Figure 120 - Excel "Data" Sheet (Vacas, Authors Own)	120
Figure 121 - Excel Measurements & Units Sheet (Vacas, Authors Own).....	120
Figure 122 - Excel Combustion Chemistry Sheet (Vacas, Authors Own)	121
Figure 123 - Excel Pivot Chart (Temperature) (Vacas, Authors Own).....	121
Figure 124 - Excel Log PV Graphs (Vacas, Authors Own)	122
Figure 125 - Excel Combustion Timing Experiment Sheet (Vacas, Authors Own)	122
Figure 126 - Excel File Versioning (before OneDrive) (Vacas, Authors Own)	123
Figure 127 - Excel File Versioning (before OneDrive) (Vacas, Authors Own)	123
Figure 128 - Excel Low Load Example (20%) (Vacas, Authors Own)	124
Figure 129 - Excel Low Load Example (100%) (Vacas, Authors Own)	124

Table 1 - Project Approach (Vacas, Authors Own).....	12
Table 2 - Project Planning (Vacas, Authors Own)	12

Equation 1 - Piston Displacement	40
Equation 2 - Cylinder Volume	40
Equation 3 - Pressure Conversion bar to Pa	40
Equation 4 - Volumetric Efficiency (Blair, 1994)	41
Equation 5 - Work Per Degree Step	45
Equation 6 – IMEP (B.Heywood, 1988)	46
Equation 7 - "n" Slope.....	51
Equation 8 - Polytropic Index "n"	52
Equation 9 – “P2n” Pressure Due to Change in Volume.....	56
Equation 10 - Pressure Difference Due to Combustion.....	56
Equation 11 - RPM Per Second	68
Equation 12 - Second Per Revolution	68
Equation 13 – MFB (Rassweiler & Withrow, 1938).....	70
Equation 14 - MFB Method 1 (P2n)	71
Equation 15 - MFB Method 2 (P-Comb).....	72
Equation 16 - MFB Prescribed Burn Rate (van Niekerk, 2000)	76
Equation 17 - Vibe Graph Equation	77
Equation 18 - MFB prescribed Burn Rate.....	78
Equation 19 - Fuel Mass Per Revolution	81
Equation 20 - Air Mass Per Revolution	81
Equation 21 - Air & Fuel Mass Per Revolution	81
Equation 22 - Charge Purity (Blair, 1994)	81
Equation 23 - Residual Gas Mass	82
Equation 24 - Specific Gas Constant	83
Equation 25 - Average Molecular Weight.....	84
Equation 26 - Specific Gas Constant (unburnt).....	84
Equation 27 - Initial Temperature Calculation.....	86

Equation 28 – (Heywood 1988)	86
Equation 29 - Specific Heat at Constant Pressure (Heywood 1988).....	86
Equation 30 - Specific Heat at Constant Pressure (fuel) (Heywood 1988)	87
Equation 31 - Average Specific Heat at Constant Volume (Mixture) J/KG.K	87
Equation 32 - Average Specific Heat at Constant Volume (Exhaust) J/KG.K	88
Equation 33 - Heat Energy needed for 1K Temperature Rise.....	91
Equation 34 - Heat Energy Needed to Reach "X" (Mixture)	91
Equation 35 - Heat Energy Lost Dropping 1K (Exhaust).....	91
Equation 36 - Heat Needed to Reach "X" (Exhaust)	91
Equation 37 - Heat Energy Balance.....	92
Equation 38 - Auto-Ignition Temperature (Vacas, Authors Own).....	94
Equation 39 - Honda Cylinder Temperature (Ishibashi, 2000)	101
Equation 40 - Auto-Ignition Temperature (Ishibashi, 2000).....	101