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Analytical Investigation into Engine efficiency

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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.’

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Abbreviations

TDC = Top Dead Centre
BDC = Bottom Dead Centre
NA = Naturally Aspirated
MFB = Mass Fraction Burned
FL = Full Load
PL = Part Load
SL = Smaller Load
HL = Higher Load
C.A = Crank Angle
HCCI = Homogeneous-Charge Compression Ignition
WLTP = World-Harmonized Light-Duty Vehicles Test Procedure

Nomenclature

° = Degrees
RPM = Revolutions per minute
CC = Cubic Centimetre
L = Litre
 Σ = Change in
 ΔP = Difference pressure

1. Introduction

The purpose of the internal combustion engine is to produce mechanical power from the chemical energy stored in the fuel being used by the vehicle. This power is converted into torque which is then used to propel the vehicle.

The aim of this project is to determine the combustion characteristics of different engines from cylinder pressure data. By understanding what is actually happening in the engine during the combustion process.

Looking at cylinder pressure data is done because it is one of the only practical ways to understand and measure what is really happening inside of an engine.

Using the cylinder pressure data for the selected engines, the techniques needed for creating a combustion model for each engine will be developed. Data for all engines used in this study came directly from Ford Motor Company.

List of Engines:

- Naturally Aspirated 1600cc four-cylinder engine, known as the Sigma.
- Turbocharged 1600cc four-cylinder engine, known as Ford 1.6L EcoBoost
- Turbocharged 1000cc three-cylinder engine, known as Ford Fox or Ford 1.0L EcoBoost

For the Fox engine, there are three sets of data that have been provided. One contains the pressure data for a whole range of different RPMs from 1000rpm to 6500rpm. One contains cylinder pressure data recorded whilst the engine is under full load for different log points, which are just specific RPMs. The last consists of pressure data also recorded at set log points but the engine is only at part loads.

2. Literature Review

The combustion process is directly linked to the fuel economy and power produced from the engine. The better the combustion that more the will fuel economy and power of the engine will increase. The Ideal combustion process is where the combustion process/flame speed is as quick as possible.

There are two types of engines being looked at, naturally aspirated (NA) and turbocharged. An NA engine is an engine which relies on atmospheric pressure and changes in pressure the drawn in air and expel exhaust gases. A turbocharged is an internal combustion engine that uses a turbocharger to help force air during induction.

A turbocharger is a compressor that is used to increase the amount of air that is being drawn

into the engine. Approximately a third of energy released during combustion is wasted as thermal energy in the exhaust gas. A turbocharger uses some of this otherwise wasted energy to drive itself. – (Bell, C. 1997)

The two factors that limit the amount of pressure that can be archived in a cylinder, is the knock and hardware limit. Knock, also know Denotation, is where the mixture in the cylinder is prematurely igniting before the desired time. This can occur if the temperature or pressure in the cylinder is high enough to ignite the mixture before the spark. It is known as knocking because when this occurs it produces a 'knocking' or 'pinking' sound. – (Blair, 1999)

The hardware limit is where the pressure is so much that parts of the engine start to fail or fatigue because of the forces they are being put under.

With an ever-increasing importance of improving engine efficiency and reducing of emissions the legislation regarding production vehicles is getting increasing more demanding. The latest change to be bought in, is the World-Harmonized Light-Duty Vehicles Test Procedure (WLTP). This is the latest drive cycle test that any vehicle being sold by a manufacturer will be tested on and it will have to meet strict targets.

'The World-Harmonized Light-Duty Vehicles Test Procedure (WLTP) will replace the NEDC from September 2017 and will be used for more realistic mapping of a vehicle's exhaust emissions and fuel consumption. The procedure uses a modified driving cycle and more stringent test specifications, and is aimed at making it possible to compare results worldwide. In future, the WLTP will form the legal basis of the Euro emissions legislation The WLTP has three power-to-weight ratio classes:

Class 1: up to and including 22 kilowatts per 1000 kg of vehicle weight

Class 2: up to and including 34 kilowatts per 1000 kg of vehicle weight

Class 3: from 35 kilowatts per 1000 kg of vehicle weight

Vehicles typically found in the EU often have a power-to-weight ratio above 34 kW/t and thus fall under class three'. -(En.volkswagen.com, 2017)

The average road car engine at this moment in time, wastes between 70 to 75 percent, meaning its thermal efficiency is around 25 to 30 percent.

The current era v6 hybrid f1 cars which has boosted thermal efficiency to an almost unheard of 47 percent. - (Gitlin, 2017)

If F1 Engine Technology could be integrated into everyday road cars and a thermal efficiency of somewhere near 50% could be achieved, then this would complete change the landscape for internal combustion engine because that degree of efficiency could not be

ignored. A technology that is currently being used in Formula one to reach this level of efficiency is Jet Ignition systems.

'The MAHLE Jet Ignition® system is a new combustion technology which replaces the standard spark plug in SI engines with a jet ignition chamber assembly. MAHLE Jet Ignition® facilitates the implementation of ultra lean-burn operation in gasoline engines, improving their efficiency and reducing the formation of pollutants such as nitrogen oxide and particulates.' - (Mahle-powertrain.com, n.d.)

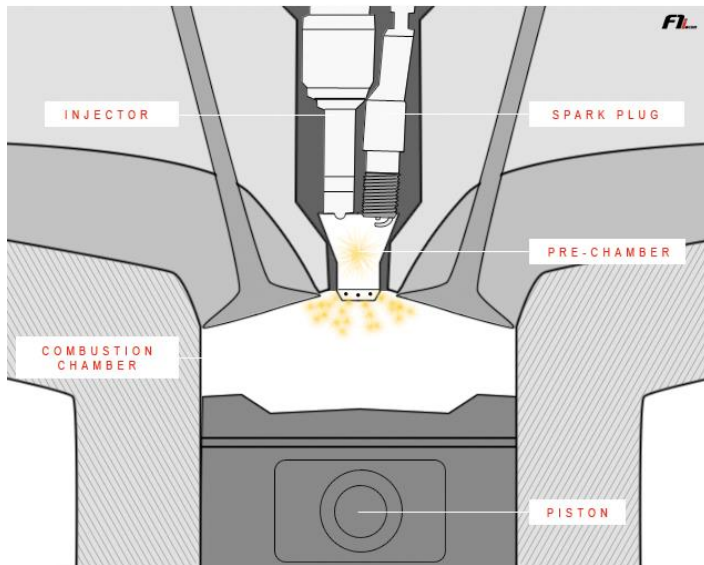


Figure 1 Jet ignition diagram - (En.f1i.com, 2017)

Jet ignition works per using a pre-chamber where the injector squirts a small amount of fuel. The amount is so small that if it was used in a regular cylinder, using a traditional spark ignition combustion the mixture would be so lean it would struggle to ignite. When confined in the small area of the pre-chamber the fuel is enough to make the mixture in the pre-chamber rich. This mixture is ignited using a spark and the flame rapidly spreads and expands. It is forced through very small holes connecting the pre-chamber to the rest of the cylinder. The mixture in the rest of the cylinder is very lean but because the flame has been forced out of these small openings, it jets and reaches throughout the cylinder. This means the flame has a very large surface area and doesn't struggle to burn because it is in contact with most, if not all the mixture as it enters the cylinder. This means that all the fuel being used is being optimised and that all the benefits of running lean are achieved whilst also have a high flame speed and a complete combustion thus minimizing exhaust emissions.

Another New ignition method that has emerged is Homogeneous-Charge Compression Ignition (HCCI)

'HCCI is the ultimate combustion method for achieving both CO₂ reduction and clean exhaust using auto-ignition of gasoline, as in a diesel engine. The system features lower-temperature combustion compared to ordinary gasoline engines, resulting in nearly no NO_x emissions. It offers the highest potential heat efficiency among internal-combustion engine systems, leading to projections of major cuts in CO₂ emissions. - (NISSAN MOTOR Co., n.d.)

HCCI is another technique that could increase the efficiency ordinary production cars and could greatly improve the potential of internal combustion engines for the near future.

3. Methodology

3.1. Finding Pressure over Crank Angle

The data that had been provided for each engine, contained the values measured for the pressure inside each cylinder for each degree of crank angle rotation during a four-stroke cycle. This means that each set of data consisted of a cylinder pressure value for every crank angle from -360° to 360.

Each engine was measured at a different set of RPM values, meaning that a range of different could be look at and an investigation to see how changing the engine RPM could affect combustion.

Taking the data and then putting it into an excel spreadsheet, a graph can be created that has all the cylinder pressure values plotted over the crank angle they occur. This creates a visual representation of how the cylinder pressure changes throughout a four-stroke cycle. Because the pressure inside a cylinder changes when the engine is at different speeds, this process is repeated for each RPM that has been recorded in the data. The result of this is that for each RPM in the data, a graph of the cylinder pressure over crank angle is now produced. Depending on how many cylinders the specific engine in question has and what cylinder is being looked at, the cylinder pressure will be different so the previous process will have to be repeated. The engines used for this project have either three or four cylinder so this means that each cylinder of engine had to be taken into account. The mean pressure across all cylinders will be taken as it is a good representation of pressure across all the cylinders.

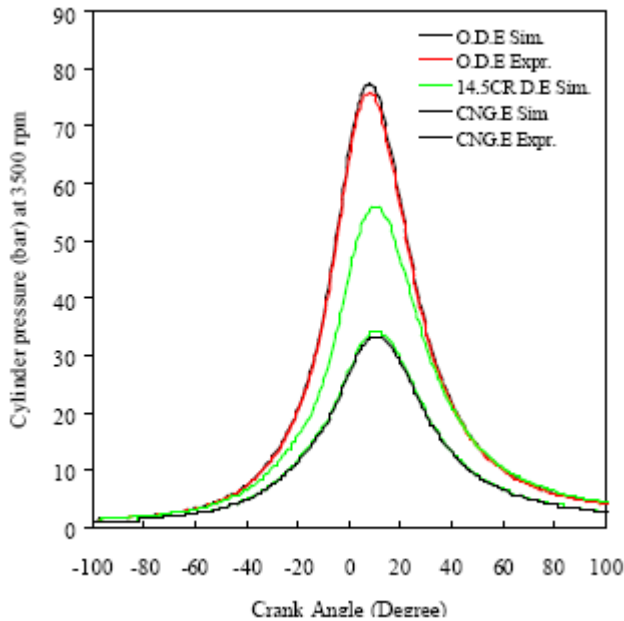


Figure 2 Example of Pressure over Crank Angle graph - (Semin , A.R. Ismail and T.F. Nugroho, (2010)

3.2. Finding Pressure over Volume

In order to understand of how the combustion is taking place in each engine, pressure data for the mixture in the cylinder over the operating cycle of the engine can be used in order to find the amount of work transferred from the mixture to piston - (Heywood, p46)

A change in volume of the cylinder occurs as the piston is moved up or down in the cylinder so the effective length of the cylinder will constantly changing thus the overall volume of the cylinder will also be constantly changing. If the piston is moved up then the effective volume of the cylinder is reduced and if the piston is moved down it will increase.

This is known as the compression stroke and during this time the mixture inside the cylinder is compressed which leads to an increase in pressure of the mixture.

For the 1.6 litre turbo engine, the Volume was not given as part of the data but because the engine is basically a turbocharged version of the 1.6 Naturally Aspirated engine, the volume is calculated using the volume data for the Naturally Aspirated engine and using this equation.

With all the cylinder pressure and volume data available, an excel spreadsheet is made having all the pressure data put on a graph over the corresponding cylinder volume for the same point in time. This means the pressure and volume values from the same crank angle degree are put over each other and this is done for every crank angle from -360° to +360° generate the graph.

6500rpm	C.A.(DEG)	VOL(CC)	N	Mean(P)
	-360	39.27		1.097
	-359	39.31		1.135
	-358	39.43		1.149
	-357	39.63		1.161
	-356	39.9		1.173
	-355	40.26		1.153
	-354	40.69		1.115
	-353	41.2		1.069
	-352	41.79		1.03
	-351	42.45		0.999
	-350	43.2		0.938

Figure 3 Data for PV Diagram

The pressure over volume graphs are known as PV Diagrams (Look for Source). PV diagrams are used to show the different strokes done by the engine.

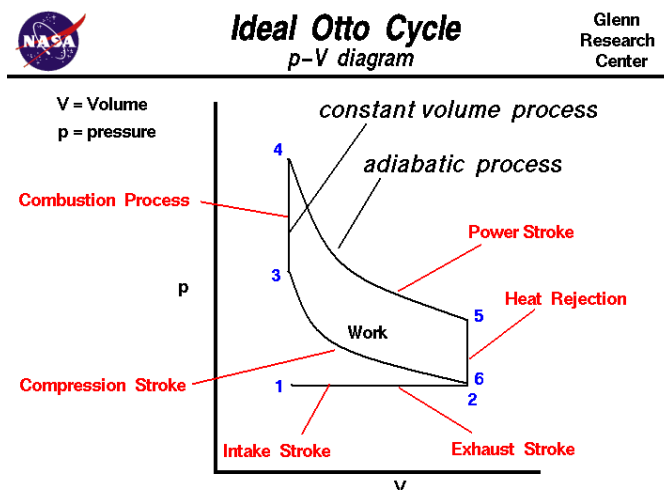


Figure 4 Example of Ideal Otto Cycle – (Hall.N)

An ideal Otto, also known as the thermodynamic cycle for the four-stroke spark ignition engine, is a gas power cycle that is used in all spark ignition internal combustion engines. Almost all modern internal combustion engines use a four-stroke cycle. For the ideal Otto cycle, the combustion is instantaneous and there are adiabatic processes that take place meaning there is no heat transfer to the surroundings – (Heywood, J. 2000).

It is used was introduced by Dr. Nikolaus August Otto, a German Engineer.

'An adiabatic process is one in which no heat is gained or lost by the system. The first law of thermodynamics with $Q=0$ shows that all the change in internal energy is in the form of work done.' – (Nave.R)

Q in thermodynamics is used to represent heat energy and in this case, it represents heat energy added to the system.

An Ideal Otto consists of six different stages which applies to all internal combustion engines. Each stage is numbered in the figure above the cycle can be illustrated from the figure. For Clarity stages 1-2 is the intake stroke and stages 6-1 is the exhaust stroke.

3.2.1. Stage 1 of Ideal Otto Cycle

The cycle starts with stage 1 which is beginning of the intake stroke of the engine. At this point the pressure inside the cylinder is around atmospheric and the volume of the cylinder is at its smallest because the piston is at TDC. From stage 1 to stage 2 the piston is pulled down the cylinder which means the cylinder volume is increased until the piston reaches BDC. The intake valve remains open while this happens which allows the air/fuel mixture to be drawn into the cylinder if the engine uses port injection. If the engine uses direct injection, then air is drawn in through the intake valve but the fuel is direct injected into the cylinder which requires the air to already be in the cylinder before they mix. During this the pressure stays around atmospheric.

3.2.2. Stage 2 of Ideal Otto Cycle

Stage 2 is the start of the engine's compression stroke. The intake valve is closed and then the piston starts to move up the cylinder. This takes place during stage 2 and stage 3, whilst this is happening the cylinder volume is decreasing thus decreasing the volume of the air/fuel mixture. Because there is work done on the mixture by the piston to reduce its volume, the pressure of the mixture is increased.

3.2.3. Stage 3 of the Ideal Otto Cycle

Stage 3 is where the combustion of the mixture starts to occur. The combustion process will start using one of several methods depending on what type of fuel the engine is designed to use. For an ideal Otto cycle, it is assumed that the combustion is instantaneous meaning there is no time for any heat transfer to occur. During combustion, the volume remains the same and heat is released from the combustion process which leads to an increase in both temperature and pressure.

3.2.4. Stage 4 of the Ideal Otto Cycle

Stage 4 is the power stroke of the engine. During this stage the piston is forced down the cylinder towards the crankshaft and the volume is increased. The pressure of the mixture is decreased as work is done to move the piston down and the volume increases.

3.2.5. Stage 5 of the Ideal Otto Cycle

Stage 5 starts when the exhaust valve is opened. The volume of the mixture remains constant with the pressure returning to being around atmospheric like it was at the start of the cycle as well as heat being transferred from the mixture to the cylinder walls and surroundings.

3.2.6. Stage 6 of the ideal Otto Cycle

Stage 6 of the cycle is the exhaust stroke of the engine. This involves the piston moving back up to cylinder to be at TDC and the volume decreasing but pressure remains the same because the exhaust gasses remaining from combustion are forced out. After the exhaust stroke has finished the engine is in the same state as it was at the beginning of stage 1 thus allowing the process to start again.

3.2.7. Work done during the cycle

Throughout different stages of the cycle work is done either by the mixture or is work done on the mixture. Between stages 2 and 3, the compression stroke, work is done by the piston to compress the mixture, while between stages 4 and 5, the power stroke, work is done by the combusting mixture to force the piston down. The amount of work done and the rate that it happens can be seen by the two curves in the PV-diagram. The difference between the work done on the mixture and the work done by the mixture is the area inside the diagram and this is the work produced by the cycle. The power produced from an engine is equal to the work produced from the cycle multiplied by the rate of cycles (cycles per seconds).

3.3. Finding Log pressure over Volume

With the PV-Diagrams now plotted for each RPM, for each engine, the next step is to generate a log p-log V (Log PV) diagram in order to find an approximate start and end of combustion. In a Log PV diagram the compression and expansion strokes are essentially straight lines on the diagram. Because of these straight lines, the start and the end of the combustion process can be located at where these straight lines start to change direction and start to curve.

To generate the Log pressure over volume diagrams the values used in the spreadsheet used to find the PV-diagrams are copied into a new spreadsheet and then the logarithms of these pressure values are calculated using the LOG function in excel. The loop that is underneath the graph is known as the pumping loop or pumping losses. This loop shows the amount of work required to draw in air during the intake stroke and expel the exhaust gases out of the cylinder during the exhaust stroke.

3.4. Finding polytrophic index

In order to understand the different characteristics of the combustion, such as start and end of combustion, The Polytrophic index needs to be calculated. The polytrophic index is the gradient of the line 1-2 for compression or line 3-4 for expansion

Equation used to calculate N:

$$N = \left(\frac{\log p_2 - \log p_1}{\log v_1 - \log v_2} \right)$$

3.4.1. Polytrophic index for compression

When finding the polytrophic index, it needs to be considered that the polytrophic index will be different depending on whether the mixture is put under compression or if the mixture is expanding. Because the compression stroke happens before the expansion the polytrophic index for compression needs to be found first. The Polytrophic index needs to be calculated and this is done by calculating an average polytrophic value from the polytrophic values previous calculated. For compression, a range of values were chosen that occur at a crank angle that is known to be before TDC and any potential crank angles where combustion could possibly start. An example of this would be that for the turbocharged 1.6 litre engine a range from -60° to -40° of crank angle was chosen because this range will be before any combustion will occur.

The average is just the polytrophic values from -60° to -40° added together and then divided by the number of values in the range. In this done in excel by using the AVERAGE function which calculates the mean average of a range entered after. The equation entered in the cell is like so.

$$= AVERAGE(G303:G323)$$

3.4.2. Polytrophic Index for Compression and Expansion

The polytrophic index for expansion is found using the same method as the one used to find the polytrophic index for compression. For an appropriate value for expansion, requires for the select range of crank angles to be changed. In order for the range to occur while expansion is taking place, the range must be after combustion has started. Because the exact crank angle that combustion starts is unknown at this point the Log P-V diagrams have to be used to gain an approximate start of combustion. An example would be that the range selected for the turbocharged 1.6 litre engine was 50° to 70° of crank angle. The mean average of polytrophic values is calculated using the same equation in excel just with the new selected range.

3.5. Finding Pressure Change from Piston Movement (P2n)

P2n is the Pressure it would have got to at the end of each degree from piston motion only. This can be plotted as a graph but doesn't provide any useful information by doing so, however the values for P2n need to be calculated for every crank angle in order to find ΔP which is then used to find the Mass Fraction Burned

$$P2n = P1 \times \left(\frac{V1}{V2}\right)^{(N_{compression} \text{ or } N_{expansion})}$$

3.6. Finding Pressure Change from Combustion (ΔP , P2-P2n)

ΔP is the extra pressure produced from combustion or the difference between total pressure (P2) and the pressure from just piston movement (P2n). When the total pressure from the piston movement (P2n) is subtracted from the total cylinder pressure, the value remaining is the amount of pressure produced by the combustion. This is because the total cylinder is made up from the pressure from the piston compression and the increase in pressure when the mixture is combustion.

$$\Delta P = P2 - P2n$$

3.7. Finding Mass Fraction Burned

The Mass Fraction burned was calculated so that the exact degree of the crank angle in which the start and end of combustion happen are now found by going through the values of ΔP and finding where a noticeable rise starts and continues (for compression). Where they stop decreasing gradually and start to become random values that rise and fall and don't follow any trend or pattern is the end of combustion is. For expansion, the N values need to be taken after combustion has finished.

$$MFB = \frac{\sum \Delta P (S.O.C \text{ to } \theta)}{\sum \Delta P (S.O.C \text{ to } E.O.C)}$$

Where θ = previous ΔP value.

4. Results

This section will just be the results gathered by using the techniques and methods mentioned in the section above.

Full Load (FL) Log points:

- log 14 = 6000rpm
- log 10 = 4500rpm
- log 7 = 3000rpm

- log 5 = 2000rpm

- log 1 = 1000rpm

Part Load (PL) Log points:

- log 19 = 4000rpm

- log 18 = 3000rpm

- log 14 = 2500rpm

- log 9 = 2000rpm

- log 4 = 1500rpm Higher Load (HL)

- log 3 = 1500rpm Smaller Load (SL)

- log 1 = 800rpm

4.1. Pressure over Crank Angle

For each engine, a pressure over crank angle graph was plotted for each cylinder of the engine. Using pressure data for cylinders 1, 2, 3 for the 1.0 litre fox engine and cylinders 1, 2, 3 and 4 for the 1.6 NA and turbocharged. All engines have the mean cylinder pressure across all cylinders of the engine used because to include the results for each individual cylinder would be hard to present and isn't necessary. For all the Pressure over Crank Angle graphs the X axis is the Crank angle in degrees (°) and the Y axis is the Cylinder pressure measured in Bar.

4.1.1. Sigma 1.6 Litre Naturally Aspirated

When all the cylinder pressure was plotted over the corresponding crank angle degree the following graphs were produced. Mean cylinder pressure is calculated from the pressure of each of the four cylinders added together and then divided by four.

- Engine at 6500rpm

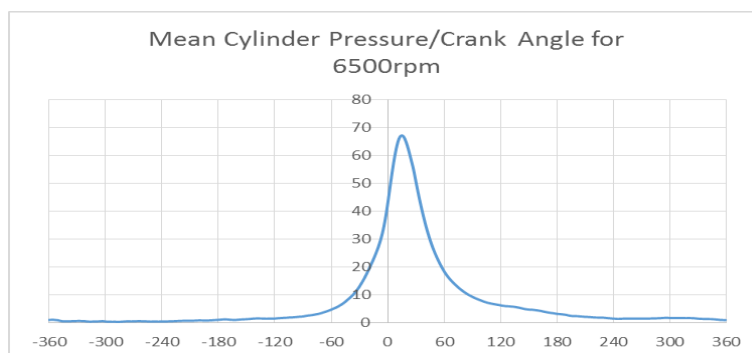


Figure 5 6500rpm P/C.A Graph

- Engine at 6000rpm

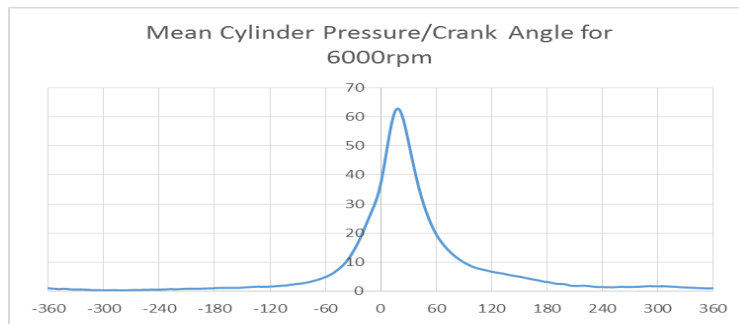


Figure 6 6000rpm P/C.A Graph

- Engine at 5500rpm

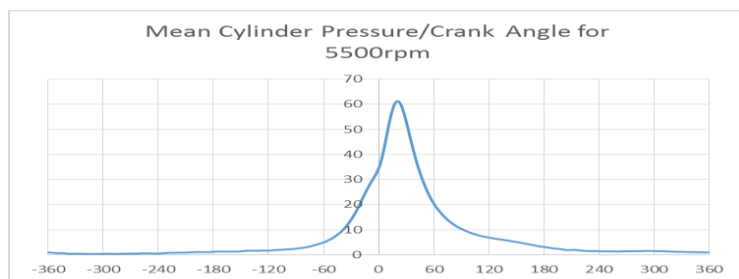


Figure 7 5500rpm P/C.A Graph

- Engine at 5000rpm

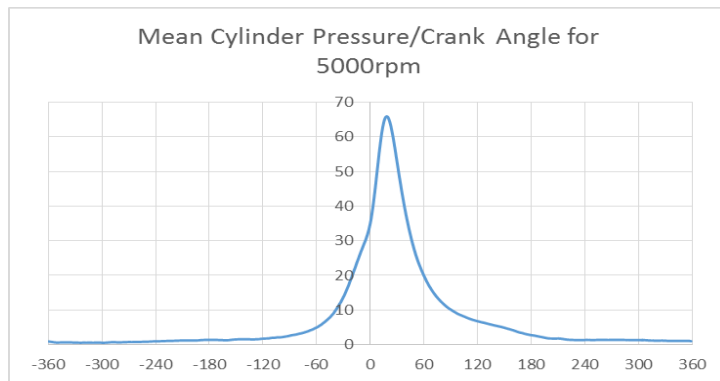


Figure 8 5000rpm P/C.A Graph

- Engine at 4500rpm

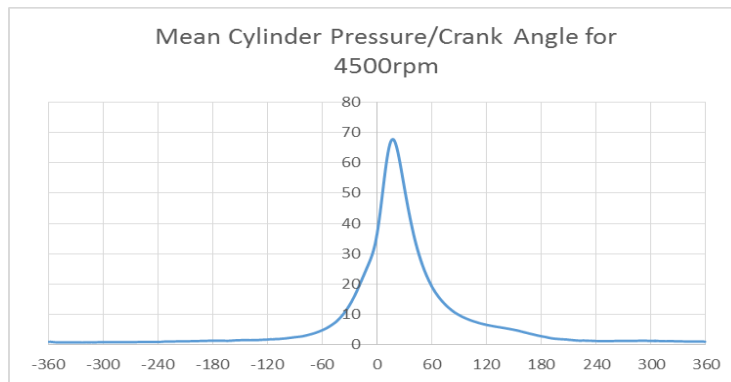


Figure 9 4500rpm P/C.A Graph

- Engine at 4250rpm

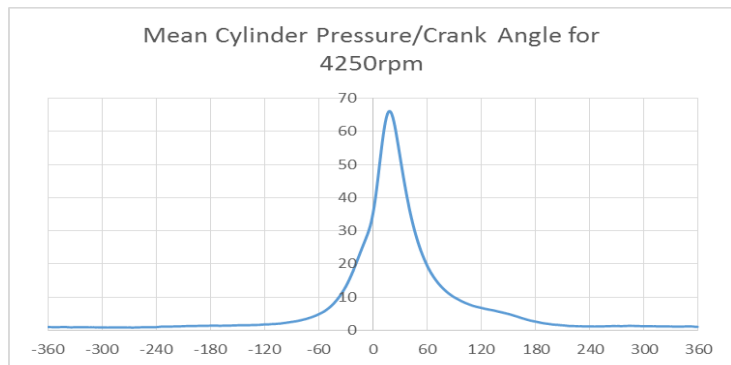


Figure 10 4250rpm P/C.A Graph

- Engine at 4000rpm

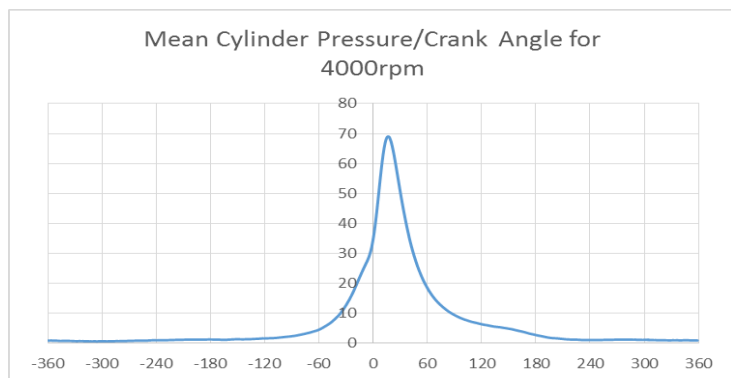


Figure 11 4000rpm P/C.A Graph

- Engine at 3500rpm

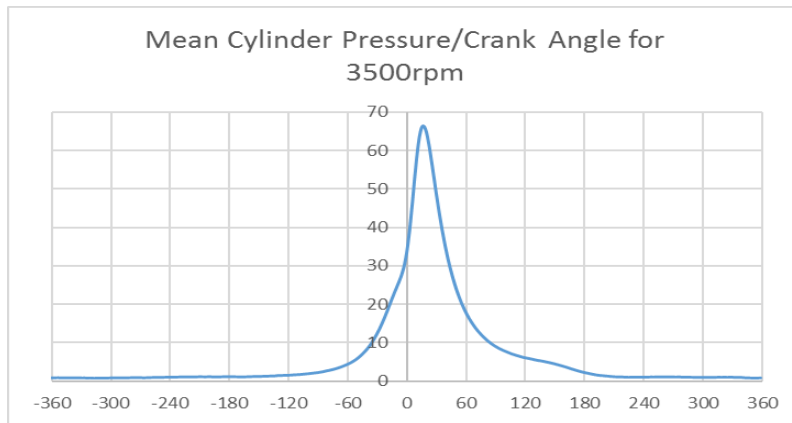


Figure 12 3500rpm P/C.A Graph

- Engine at 3000rpm

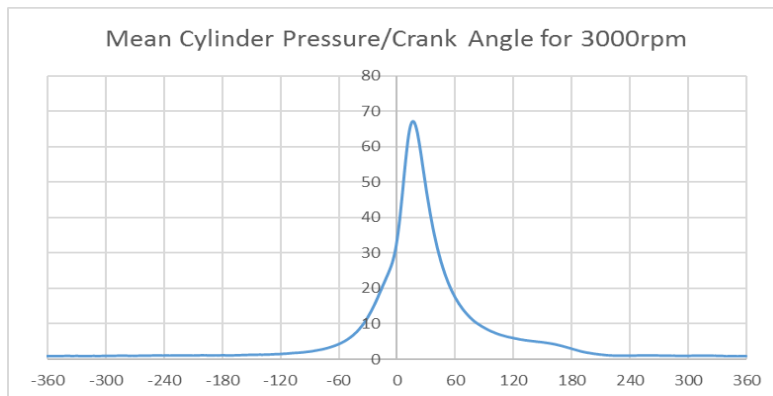


Figure 13 3000rpm P/C.A Graph

- Engine at 2500rpm

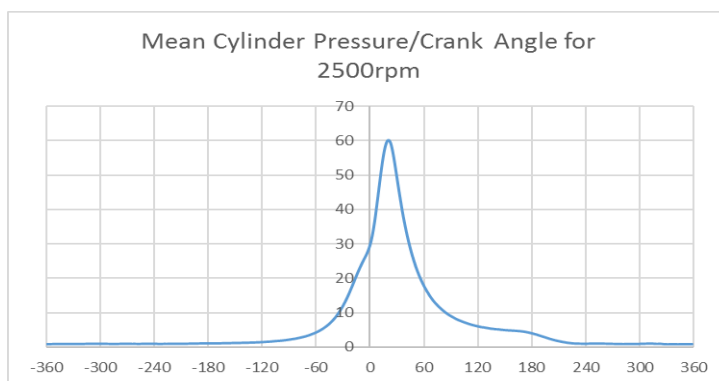


Figure 14 2500rpm P/C.A Graph

- Engine at 2000rpm

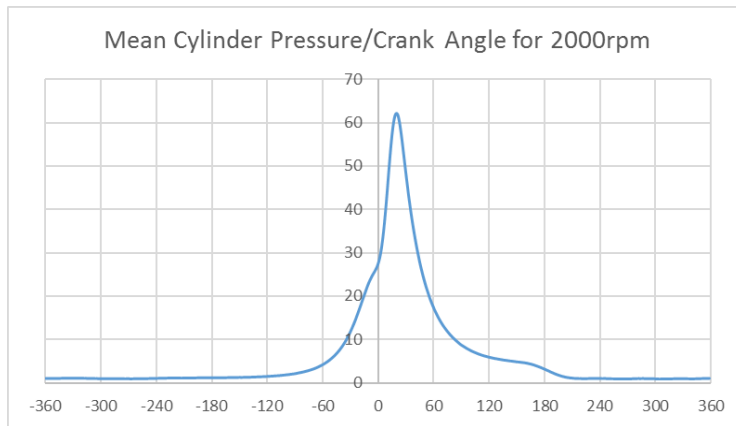


Figure 15 2000rpm P/C.A Graph

- Engine at 1750rpm

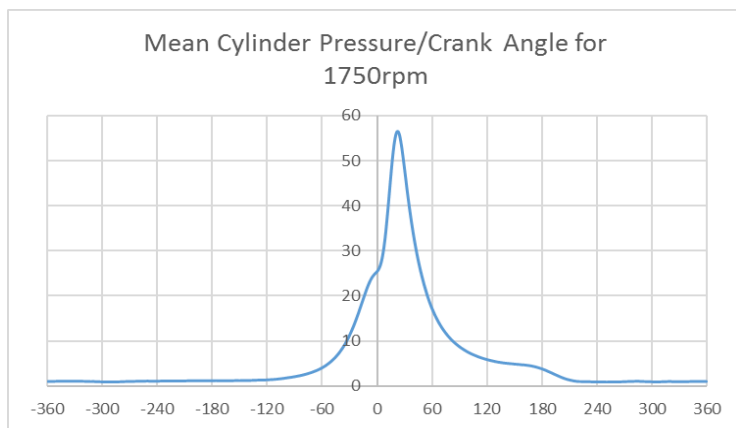


Figure 16 1750rpm P/C.A Graph

- Engine at 1500rpm

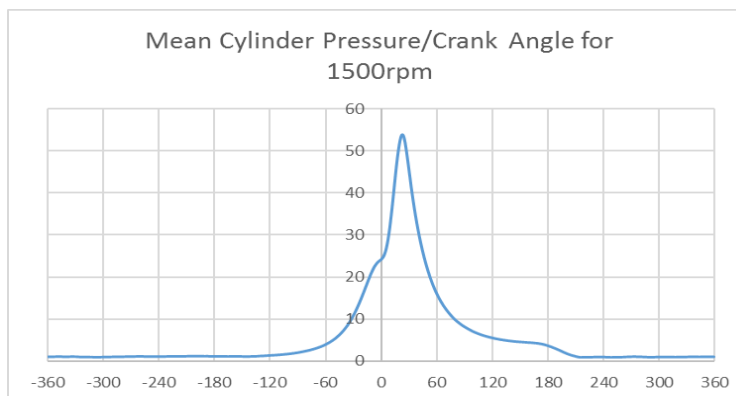


Figure 17 1500rpm P/C.A Graph

- Engine at 1000rpm

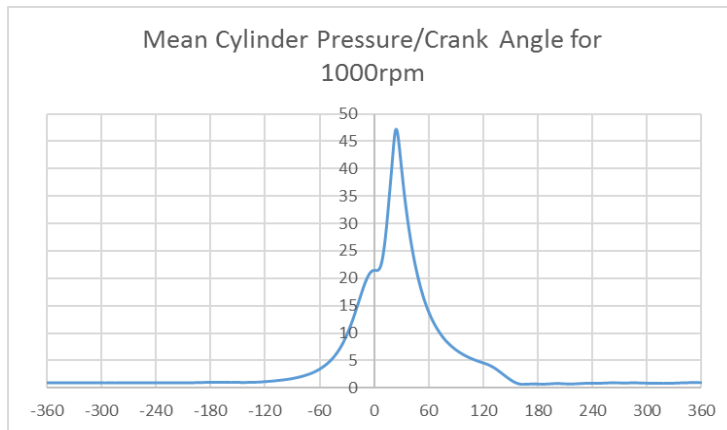


Figure 18 1000rpm P/C.A Graph

4.1.2. 1.6 Litre Turbocharged

For the 1.6 turbocharged engine. This engine has four cylinders so the mean pressure is calculated in the same way.

- Engine at 6000rpm:

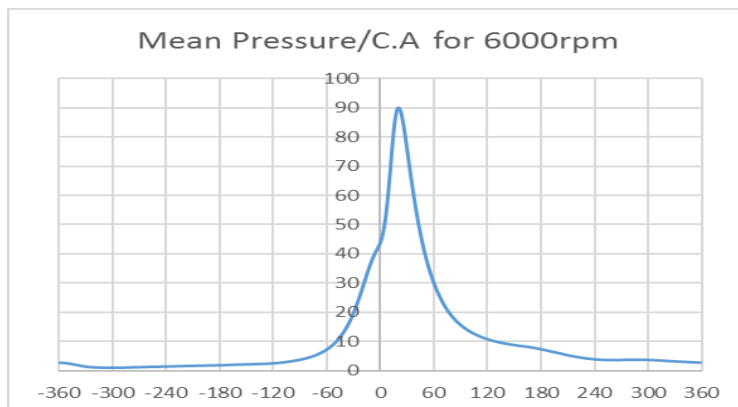


Figure 19 6000rpm P/C.A Graph

- Engine at 5700rpm

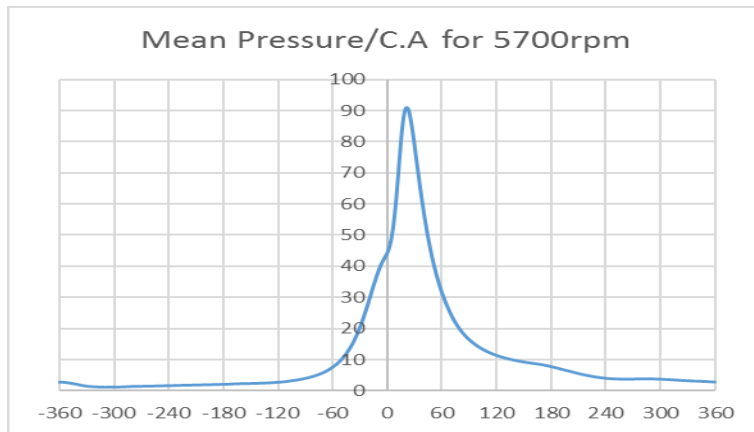


Figure 20 5500rpm P/C.A Graph

- Engine at 5500rpm

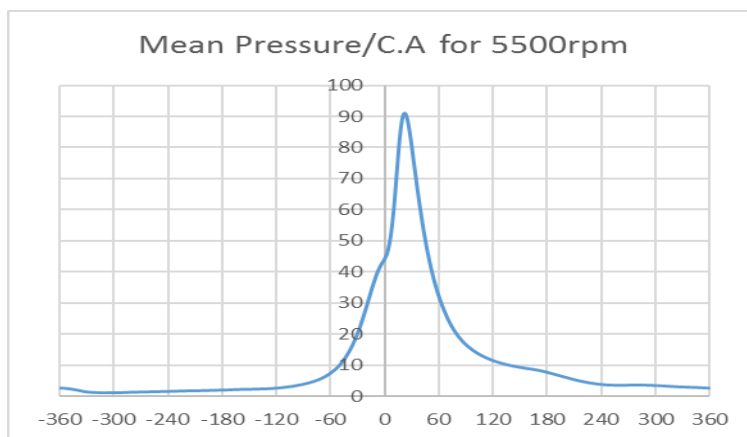


Figure 21 5500rpm P/C.A Graph

- Engine at 5000rpm

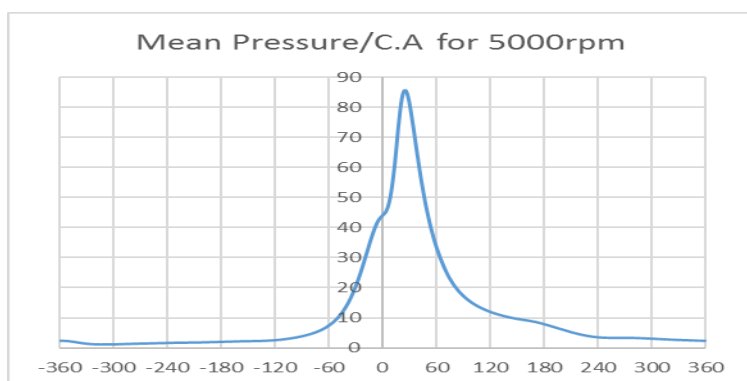


Figure 22 5000rpm P/C.A Graph

- Engine at 4500rpm

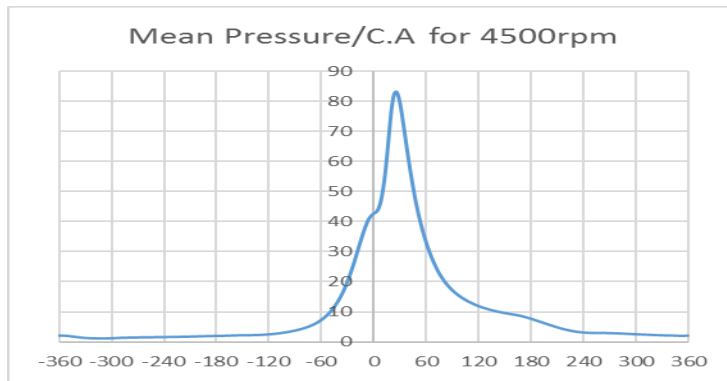


Figure 23 4500rpm P/C.A Graph

- Engine at 4000rpm

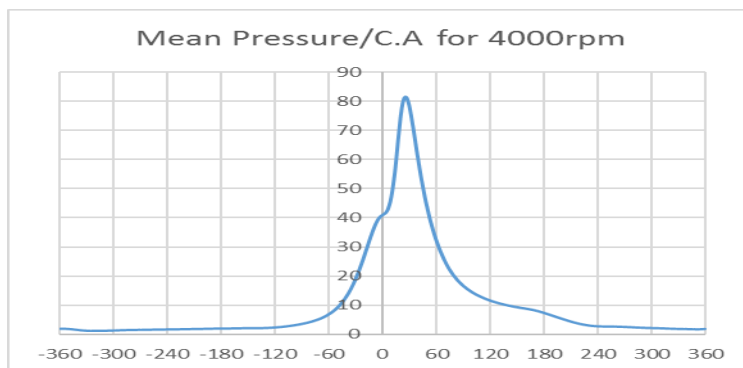


Figure 24 4000rpm P/C.A Graph

- Engine at 3500rpm

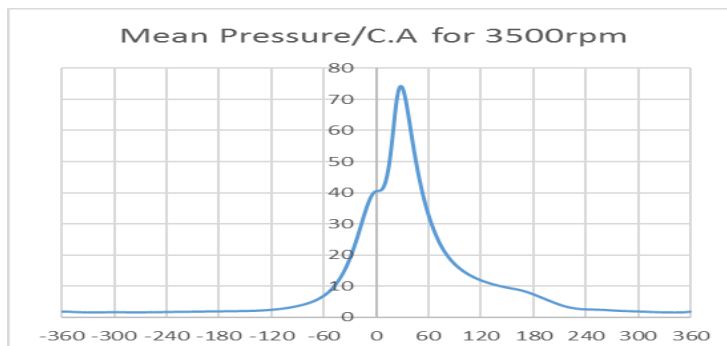


Figure 25 3500rpm P/C.A Graph

- Engine at 3000rpm

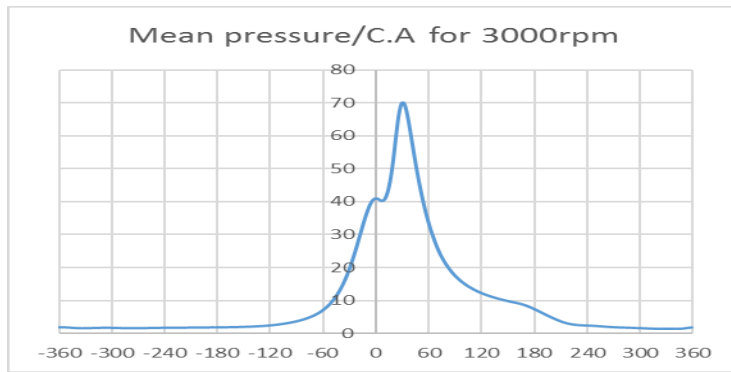


Figure 26 3000rpm P/C.A Graph

- Engine at 2500rpm

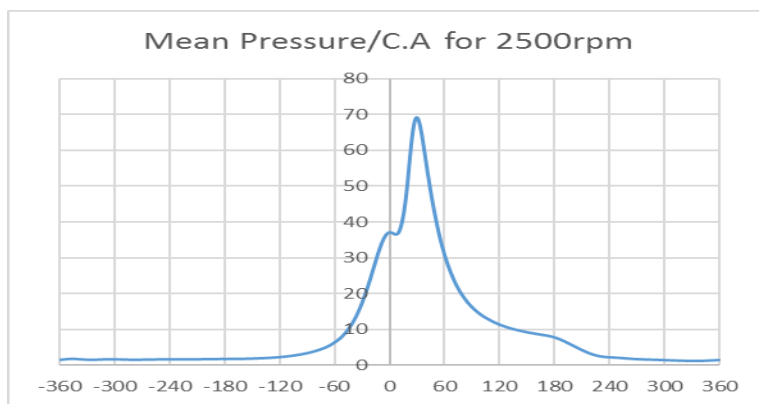


Figure 27 3000rpm P/C.A Graph

- Engine at 2000rpm

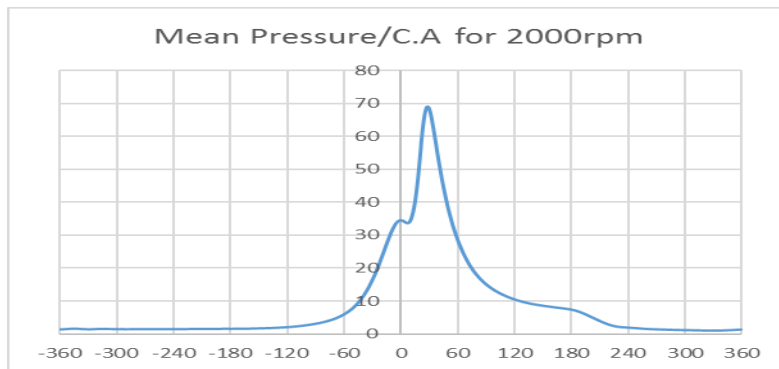


Figure 28 2000rpm P/C.A Graph

- Engine at 1700rpm

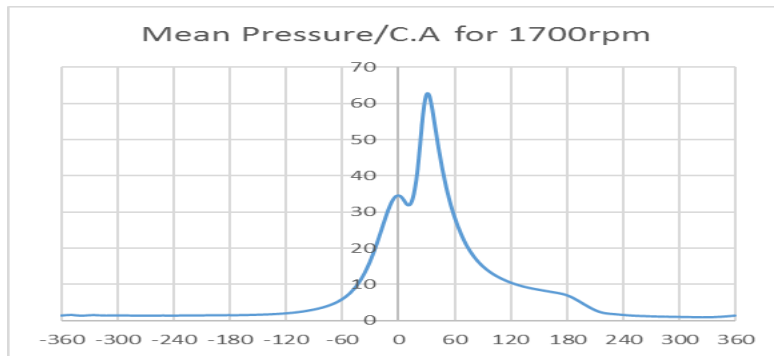


Figure 29 1700rpm P/C.A Graph

- Engine at 1500rpm

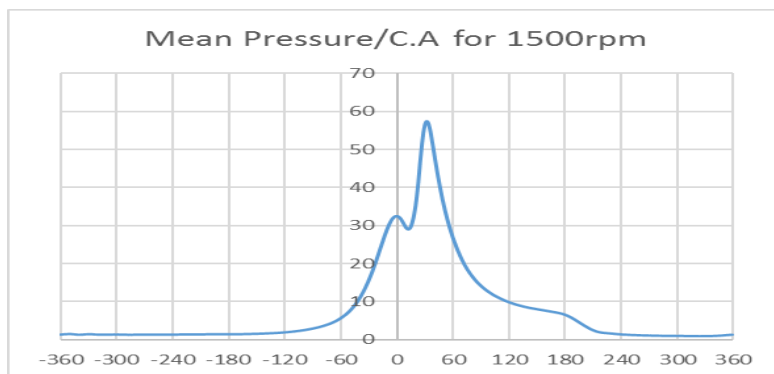


Figure 30 1500rpm P/C.A Graph

- Engine at 1000rpm

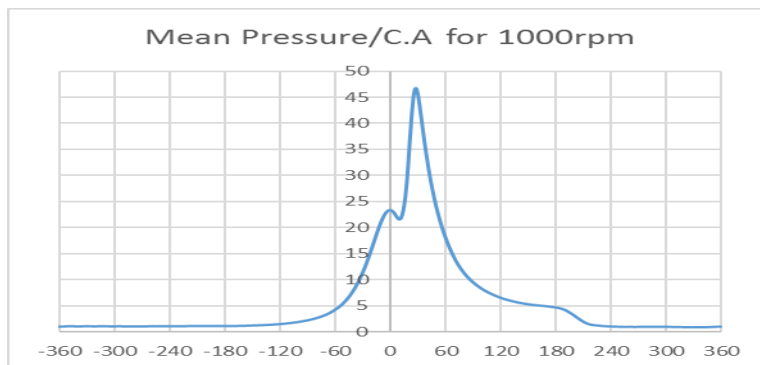


Figure 31 1000rpm P/C.A Graph

4.1.3. Fox 1.0 Litre

Pressure over crank angle for fox 1.0 litre. The Mean pressure is only calculated across the three cylinders in the fox engine because it is a three-cylinder engine.

- Engine at 6500rpm

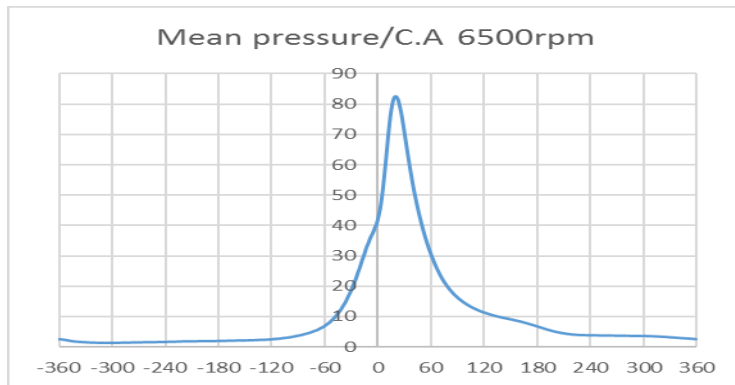


Figure 32 6500 rpm P/C.A Graph

- Engine at 6000rpm

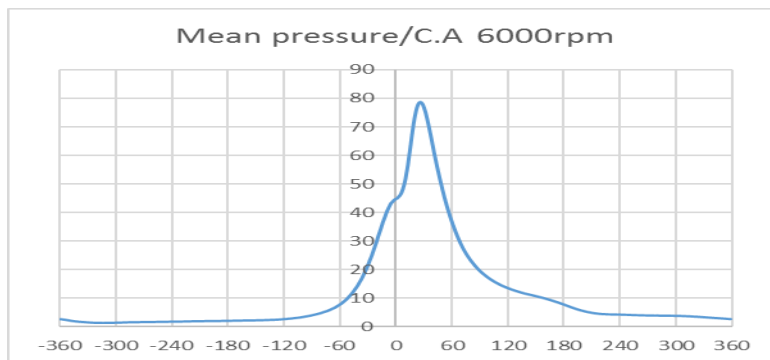


Figure 33 6000rpm P/C.A Graph

- Engine at 5500rpm

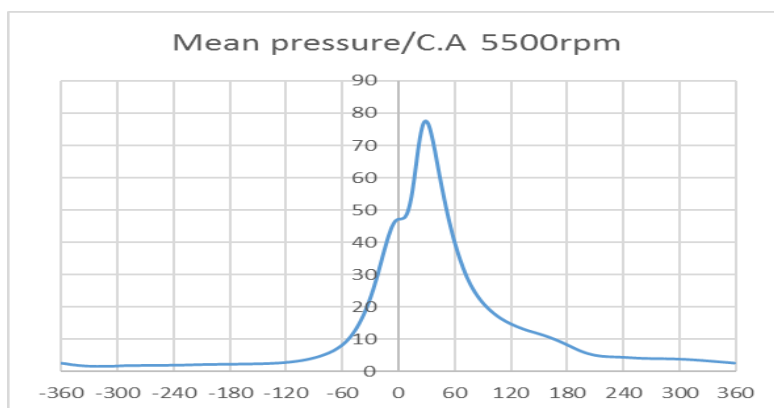


Figure 34 5500rpm P/C.A Graph

- Engine at 5000rpm

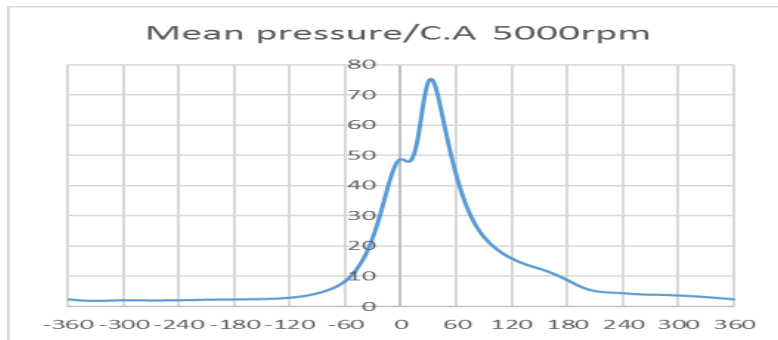


Figure 35 5000rpm P/C.A Graph

- Engine at 4500rpm

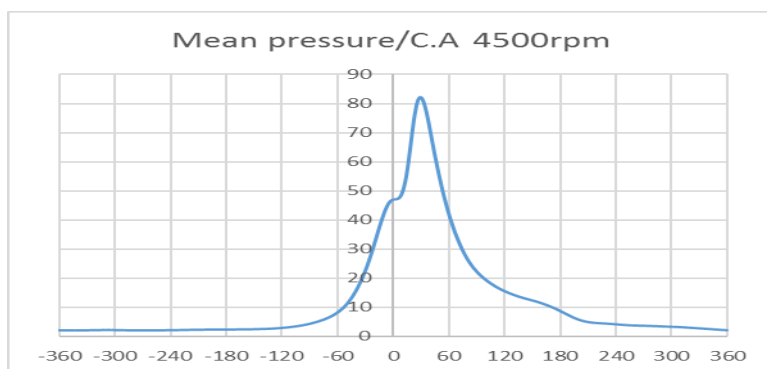


Figure 36 4500rpm P/C.A Graph

- Engine at 4000rpm

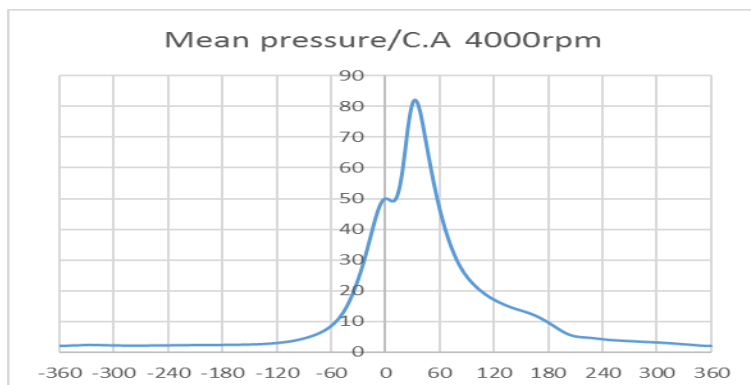


Figure 37 4000rpm P/C.A Graph

- Engine at 3500rpm

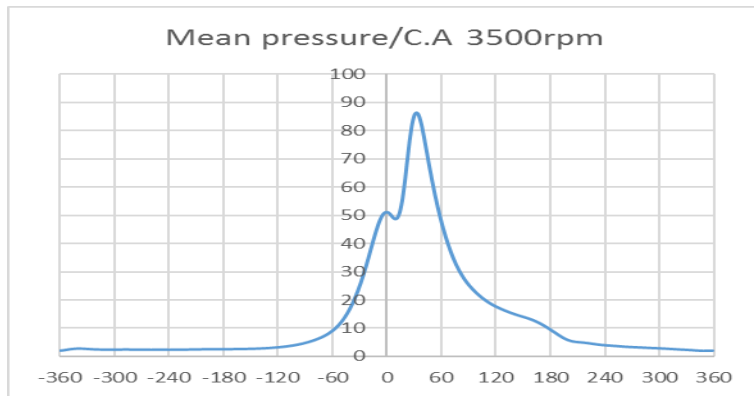


Figure 38 3500rpm P/C.A Graph

- Engine at 3000rpm

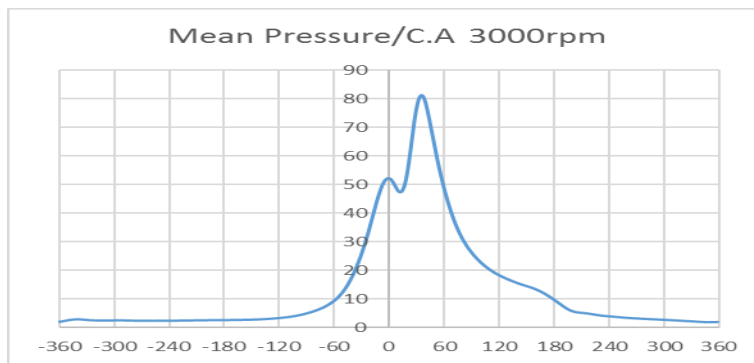


Figure 39 3000rpm P/C.A Graph

- Engine at 2500rpm

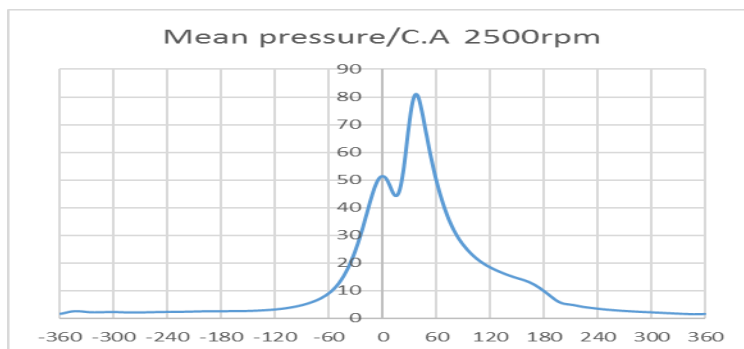


Figure 40 2500rpm P/C.A Graph

- Engine at 2000rpm

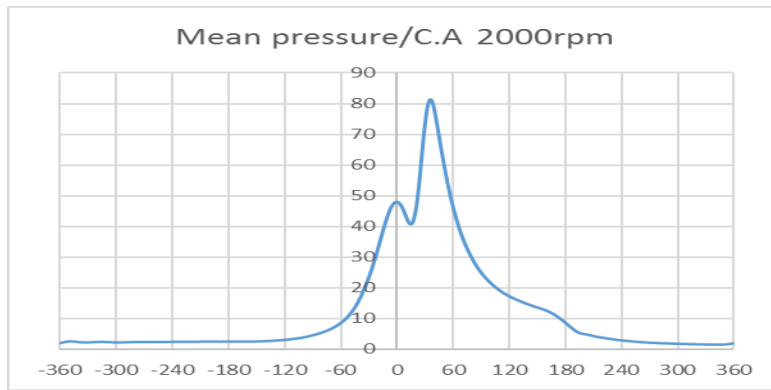


Figure 41 2000rpm P/C.A Graph

- Engine at 1750rpm

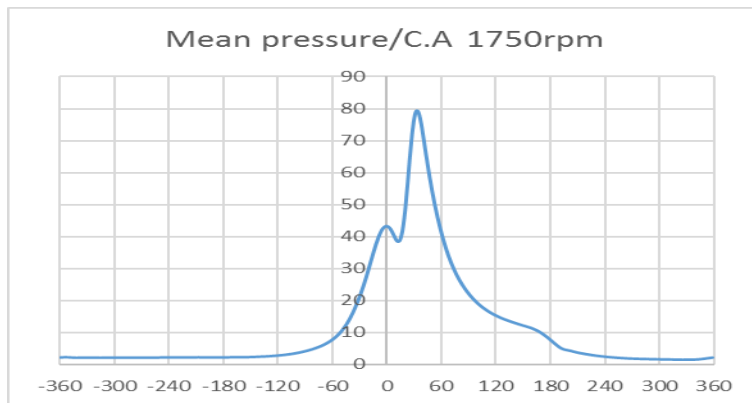


Figure 42 1750rpm P/C.A Graph

- Engine at 1500rpm

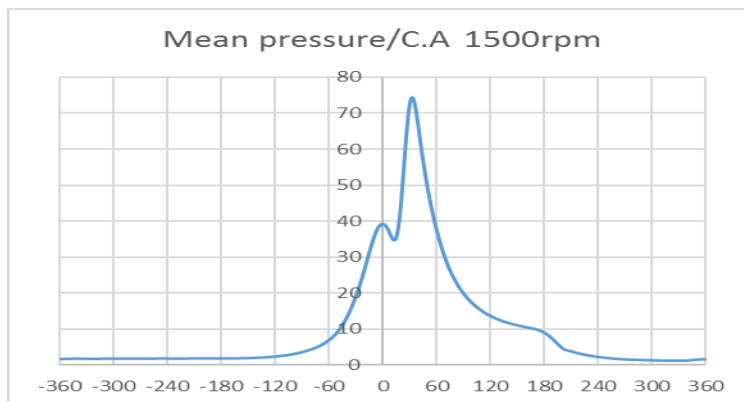


Figure 43 1500rpm P/C.A Graph

- Engine at 1000rpm

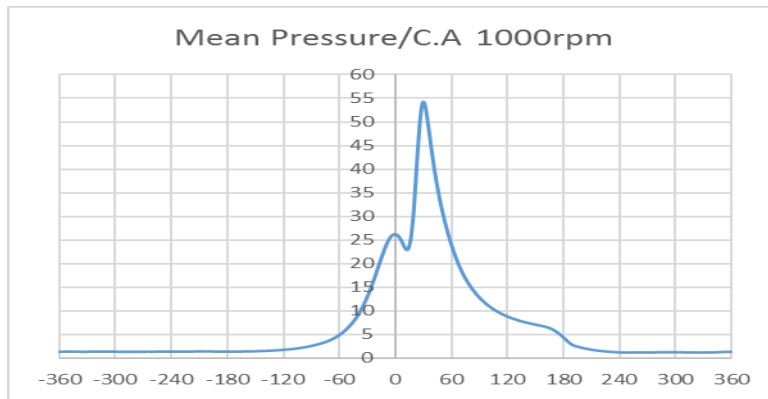


Figure 44 1000rpm P/C.A Graph

4.1.4. Fox 1.0 Litre Full Load

- Log Point 14

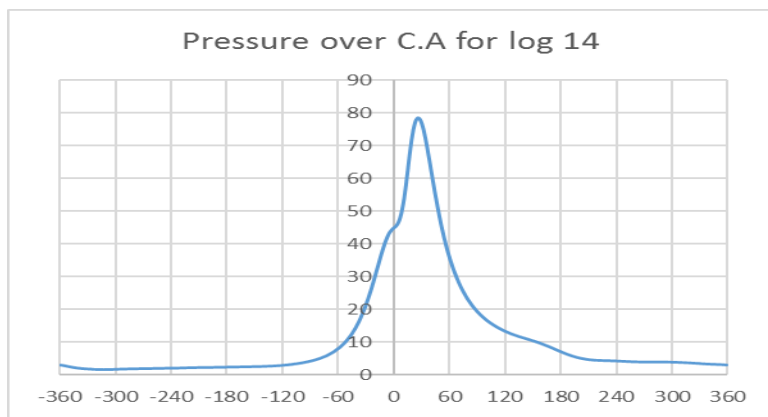


Figure 45 Log point 14 P/C.A Graph

- Log point 10

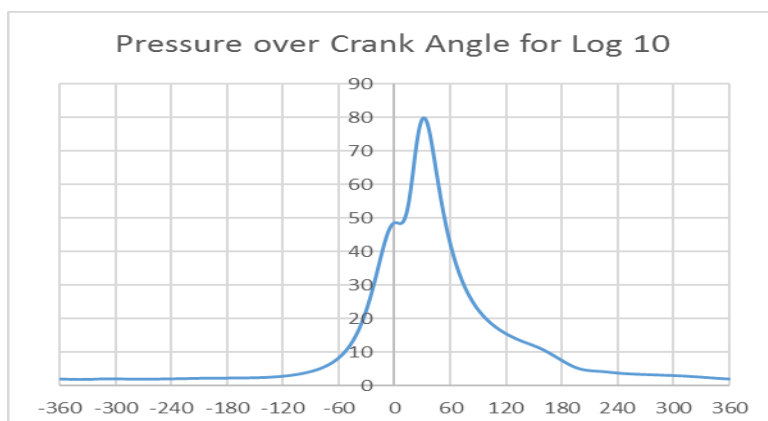


Figure 46 Log point 10 P/C.A Graph

- Log point 7

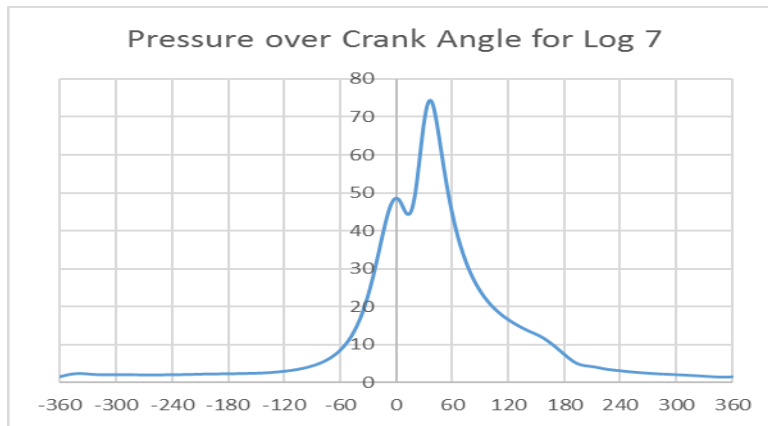


Figure 47 Log point 7 P/C.A Graph

- Log point 5

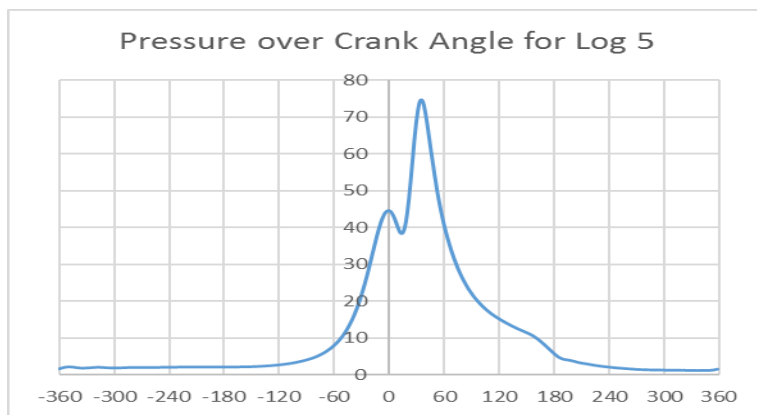


Figure 48 Log point 5 P/C.A Graph

- Log point 1

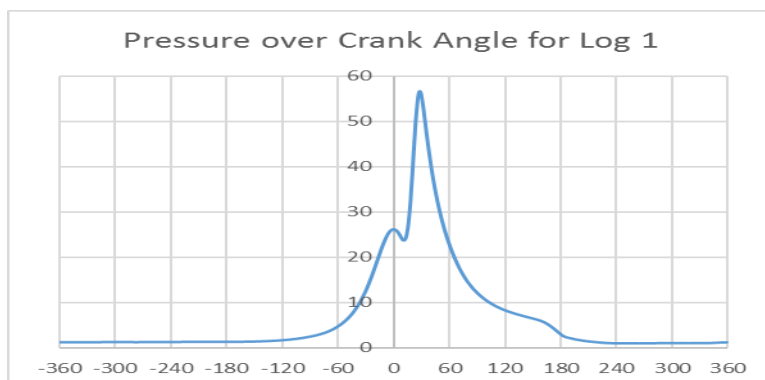
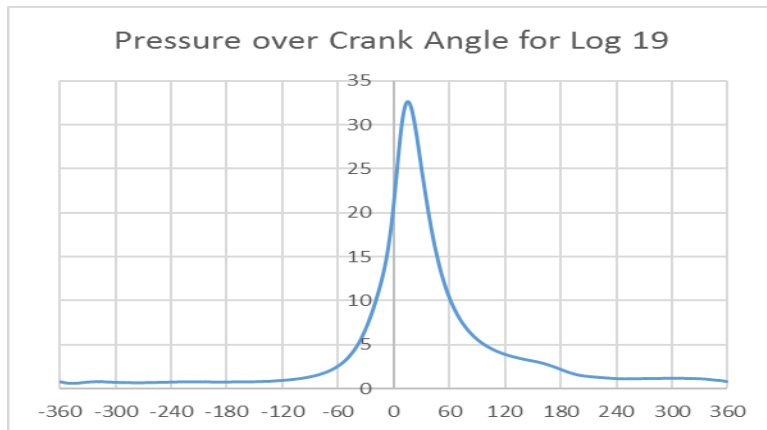


Figure 49 Log point 1 P/C.A Graph

4.1.5. Fox 1.0 Litre Part Load

- Log point 19



- Log point 18

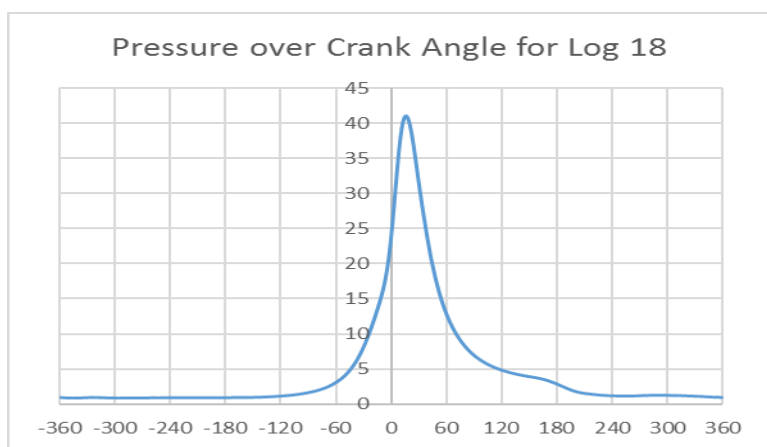


Figure 50 Log point 18 P/C.A Graph

- Log point 14

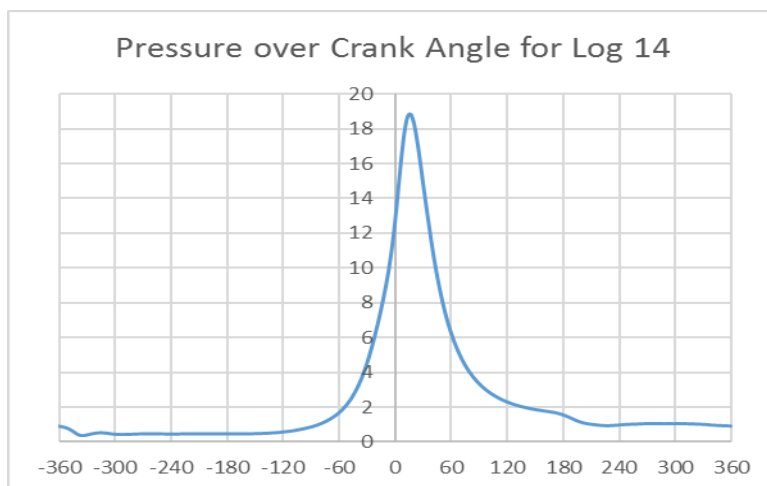


Figure 51 Log point 14 P/C.A Graph

- Log point 9

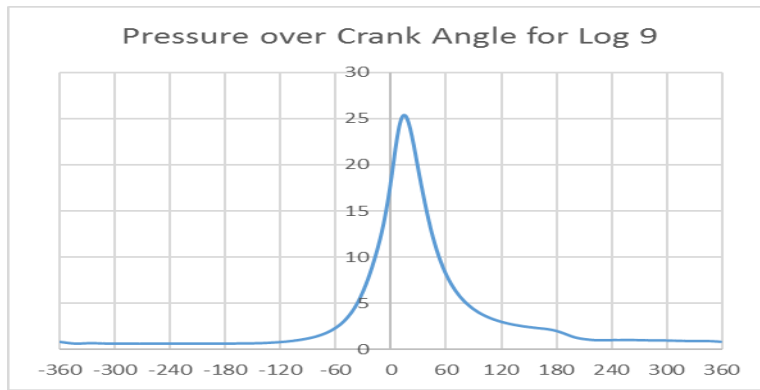


Figure 52 Log point 9 P/C.A Graph

- Log point 4

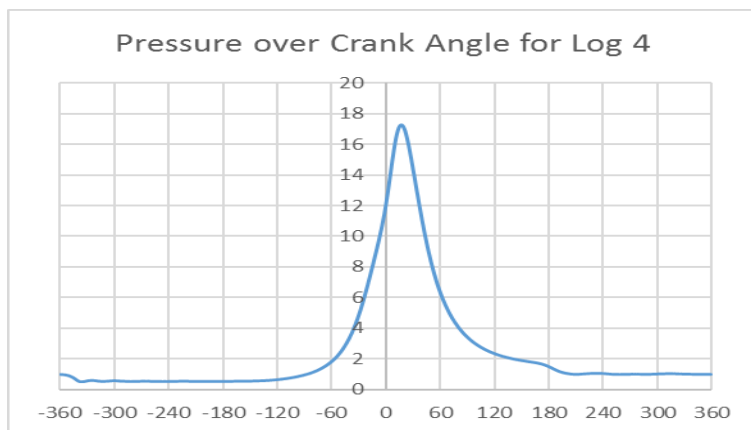


Figure 53 Log point 4 P/C.A Graph

- Log point 3

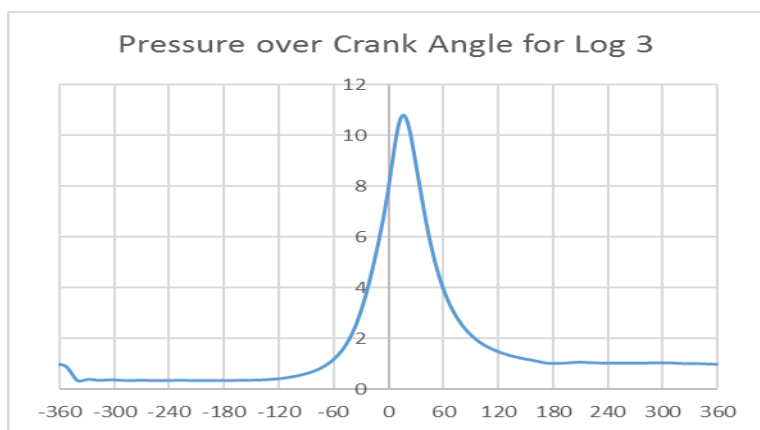


Figure 54 Log point 3 P/C.A Graph

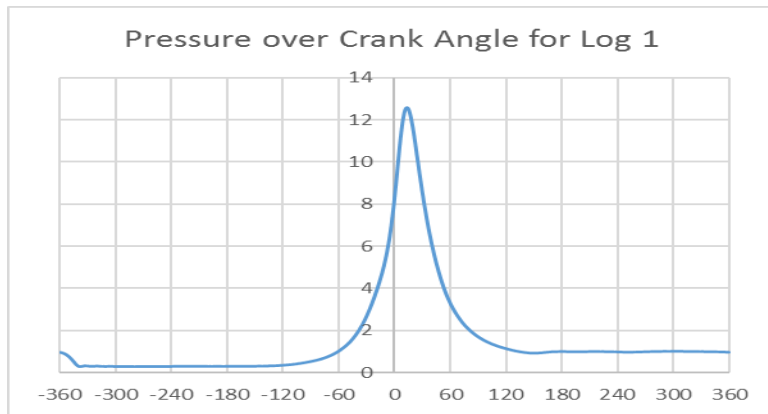


Figure 55 Log point 1 P/C.A Graph

4.2. Pressure over Volume

4.2.1. Sigma 1.6 Litre Naturally Aspirated

The following figures show the PV-diagrams produced for the Sigma engine.

- Engine at 6500rpm

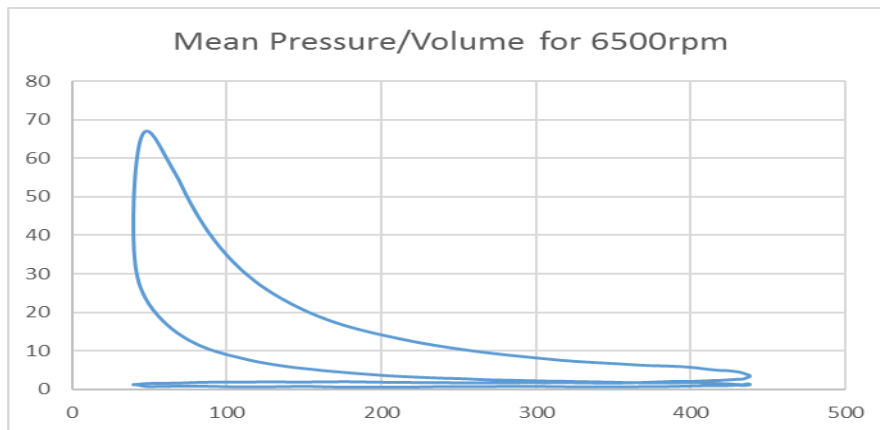


Figure 56 PV-Diagram for 6500rpm

- Engine at 6000rpm

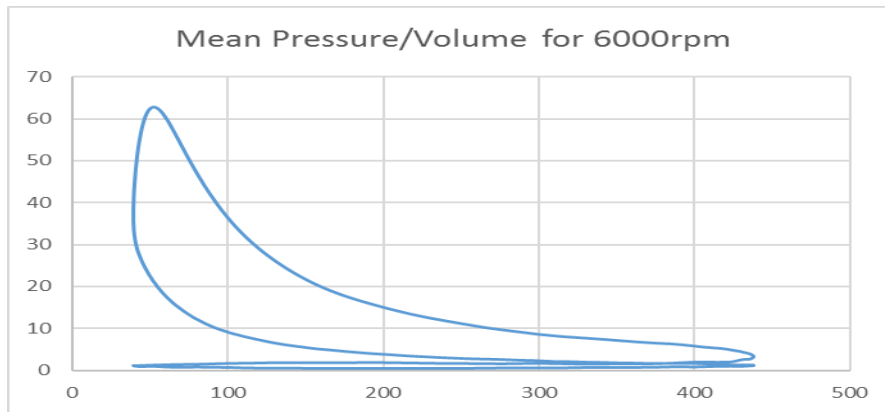


Figure 57 PV-Diagram for 6000rpm

- Engine at 5500rpm

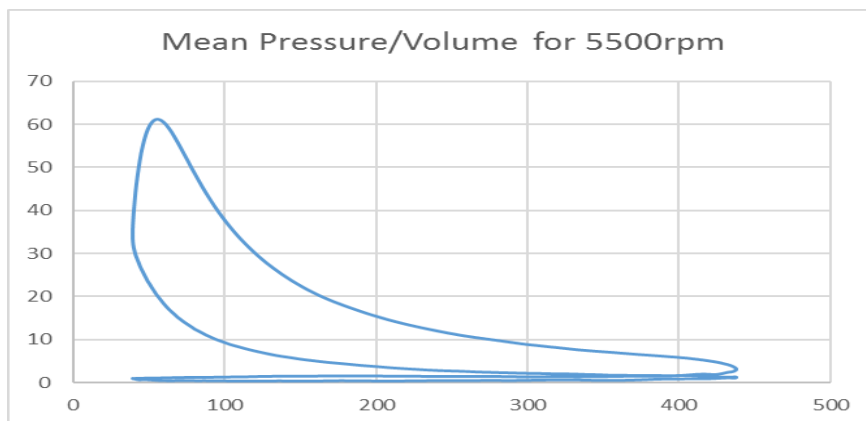


Figure 58 PV-Diagram for 5500rpm

- Engine at 5000rpm

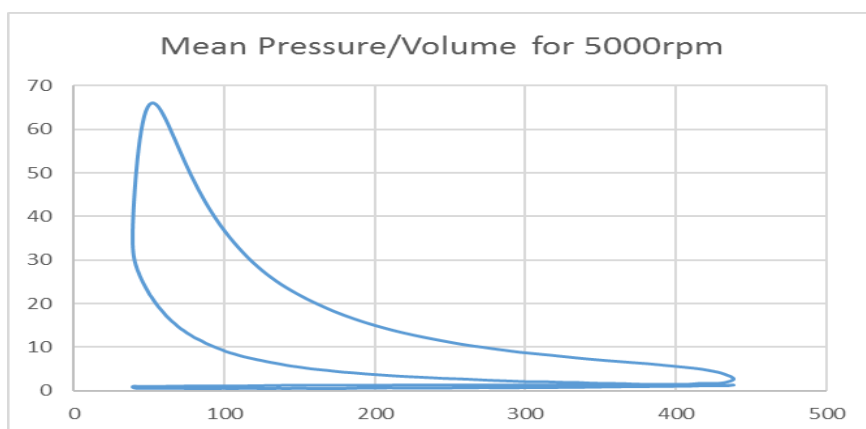


Figure 59 PV-Diagram for 5000rpm.

- Engine at 4500rpm

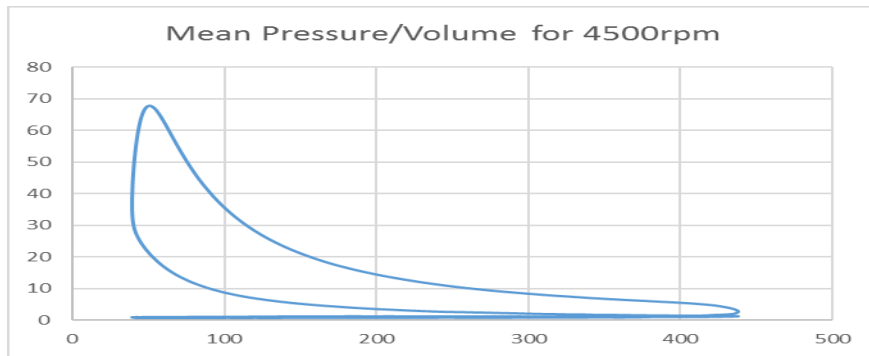


Figure 60 PV-Diagram for 4500rpm

- Engine at 4250rpm

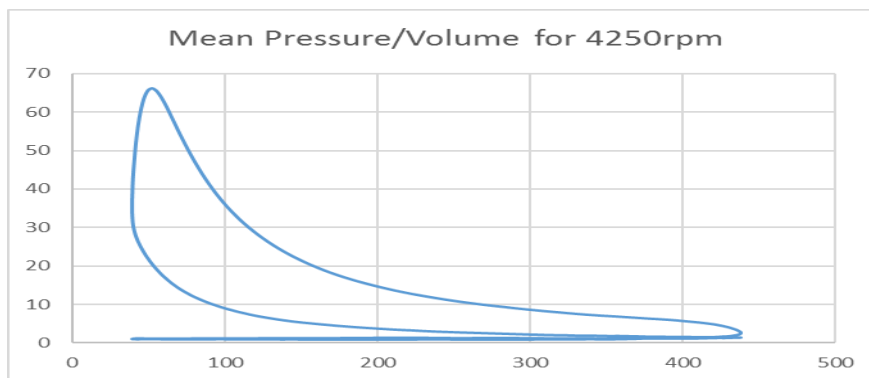


Figure 61 PV-Diagram for 4250rpm

- Engine at 4000rpm

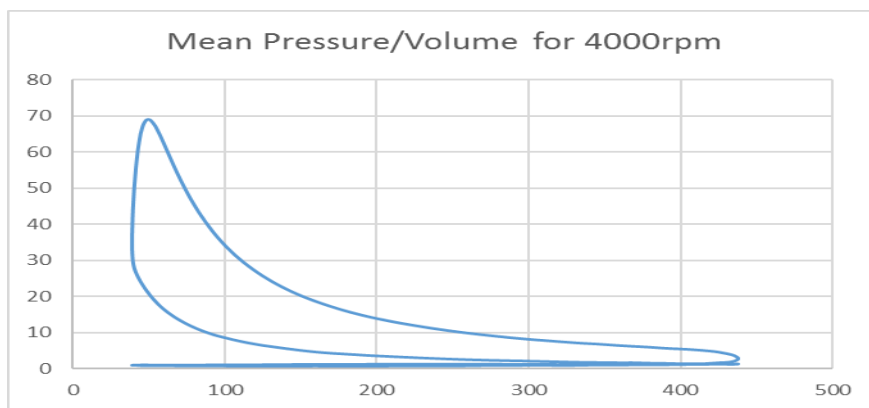


Figure 62 PV-Diagram for 4000rpm

- Engine at 3500rpm

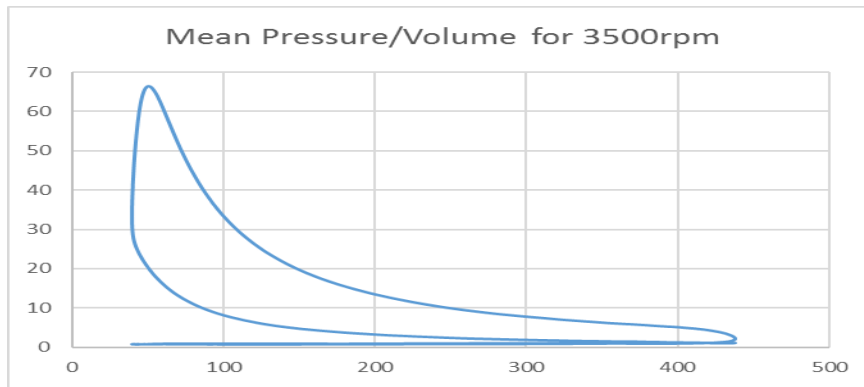


Figure 63 PV-Diagram for 3500rpm

- Engine at 3000rpm

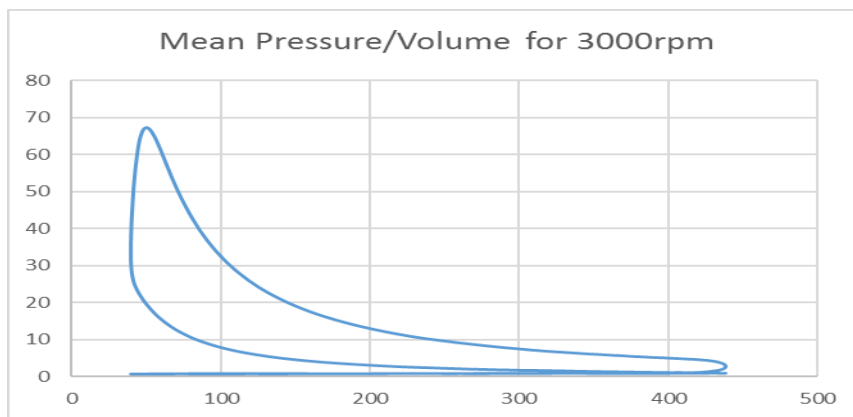


Figure 64 PV-Diagram for 3000rpm

- Engine at 2500rpm

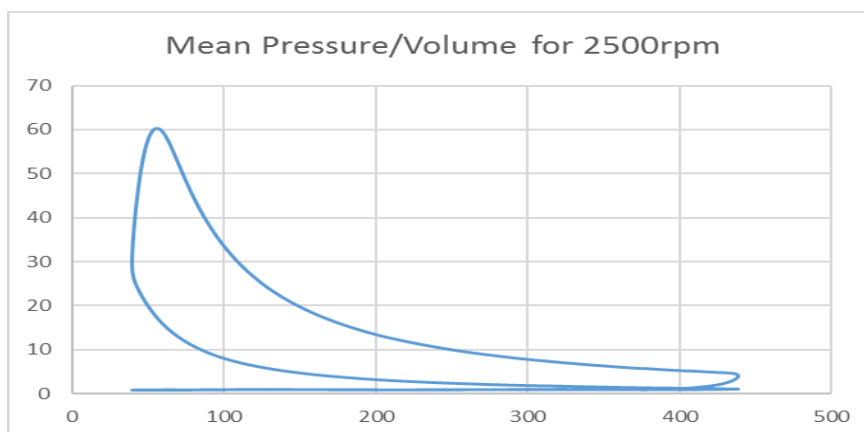


Figure 65 PV-Diagram for 2500rpm

- Engine at 2000rpm

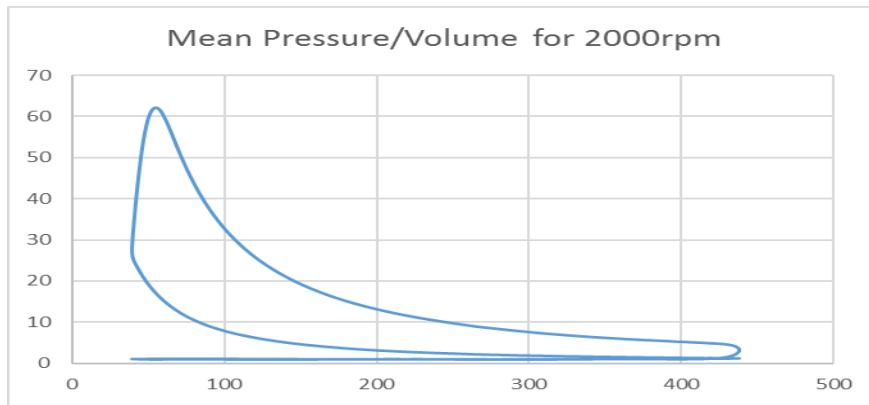


Figure 66 PV-Diagram for 2000rpm

- Engine at 1750rpm

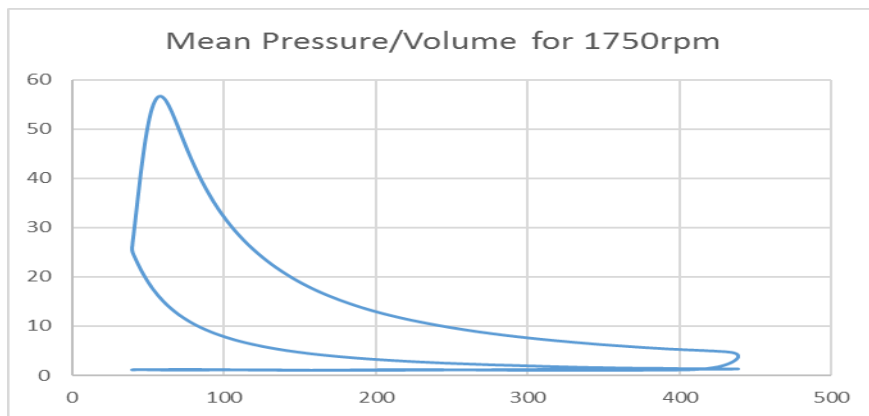


Figure 67 PV-Diagram for 1750rpm

- Engine at 1500rpm

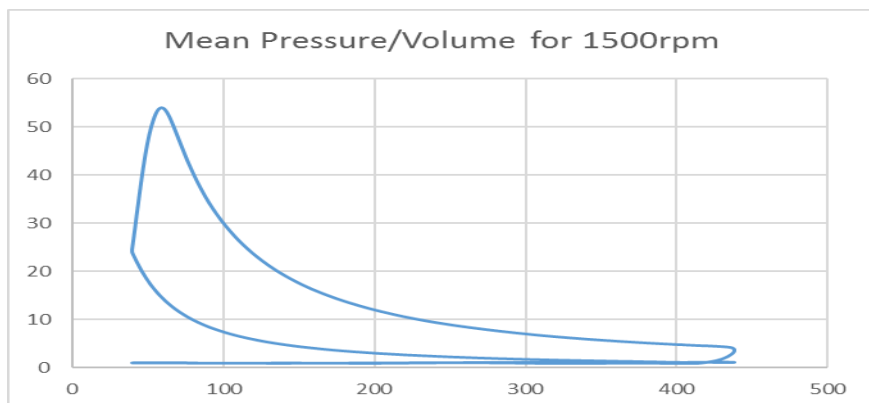


Figure 68 PV-Diagram for 1500rpm

- Engine at 1000rpm

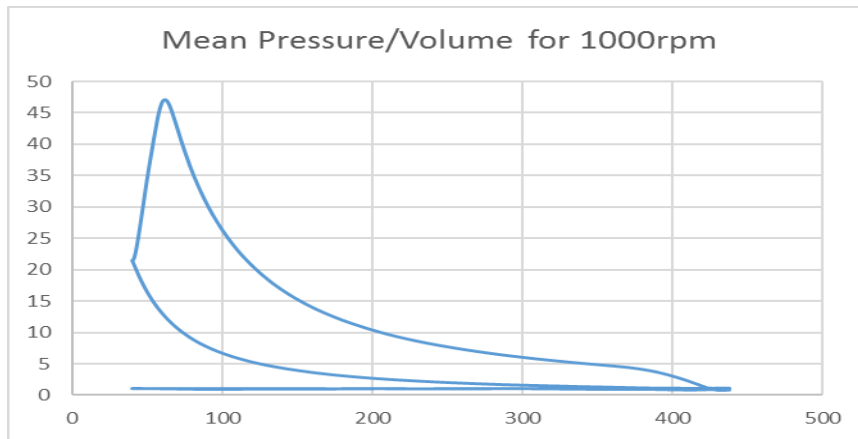


Figure 69 PV-Diagram for 1000rpm

4.2.2. 1.6 Litre Turbocharged

- Engine at 6000rpm

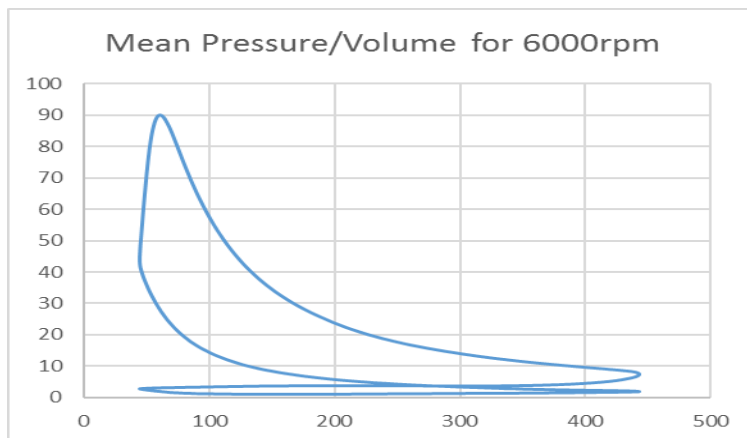


Figure 70 PV-Diagram for 6000rpm

- Engine at 5700rpm

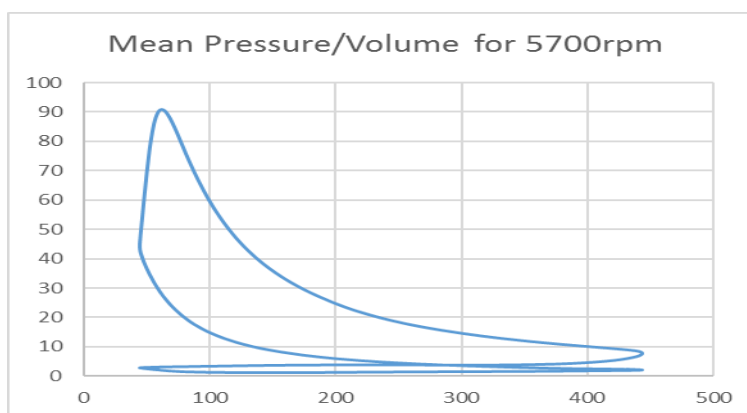


Figure 71 PV-Diagram for 5700rpm

- Engine at 5500rpm

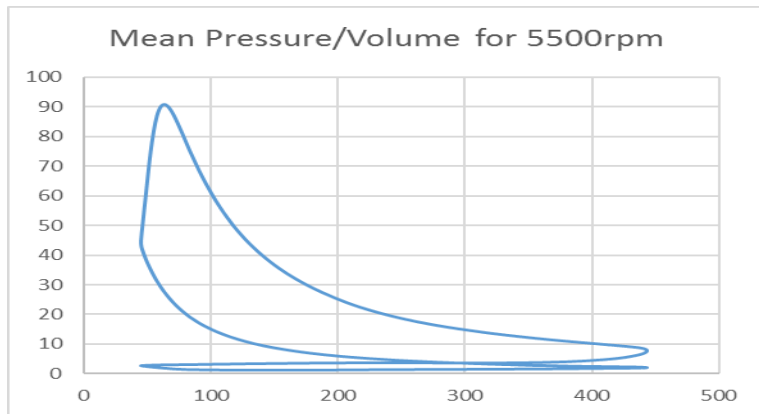


Figure 72 PV-Diagram for 5500rpm

- Engine at 5000rpm

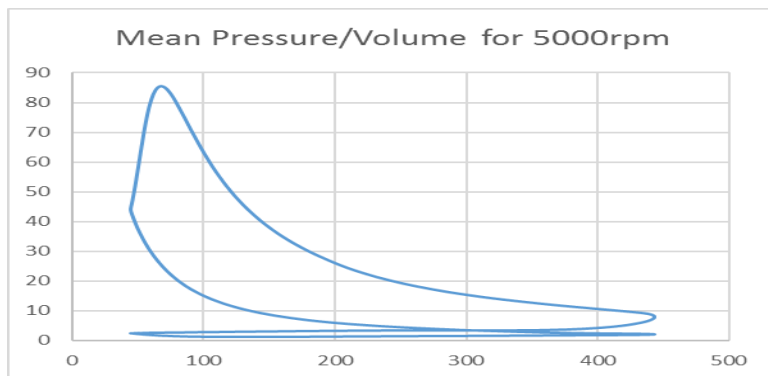


Figure 73 PV-Diagram for 5000rpm

- Engine is at 4500rpm

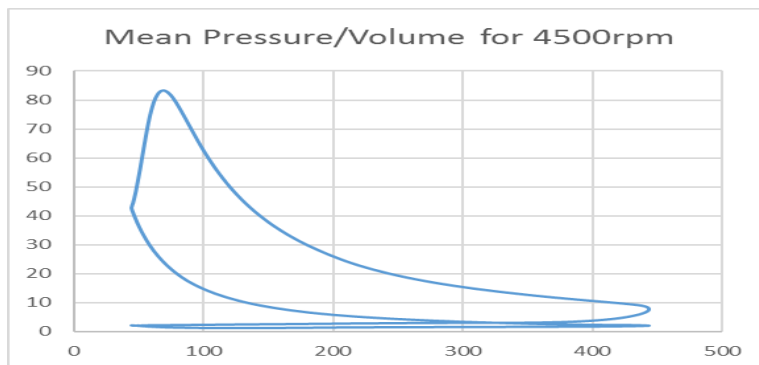


Figure 74 PV-Diagram for 4500rpm

- Engine is at 4000rpm

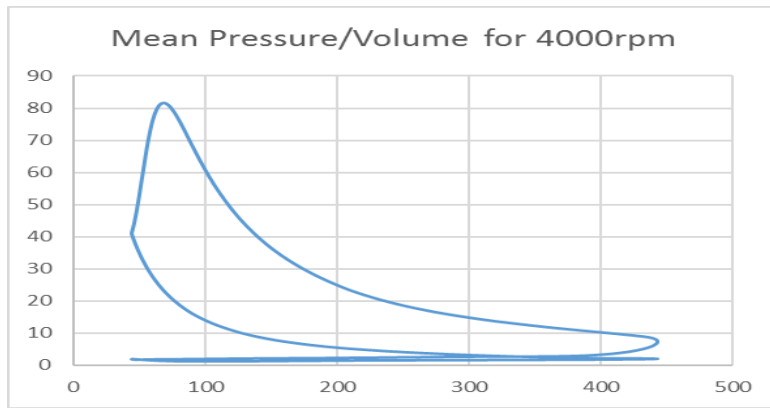


Figure 75 PV-Diagram for 4000rpm.

- Engine at 3500rpm

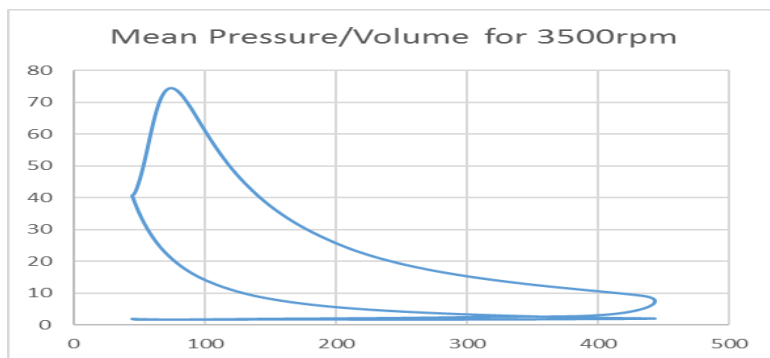


Figure 76 PV-Diagram for 3500rpm

- Engine at 3000rpm

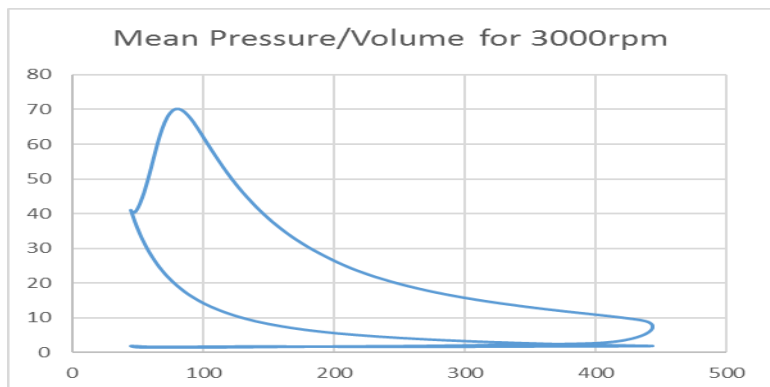


Figure 77 PV-Diagram for 3000rpm.

- Engine at 2500rpm

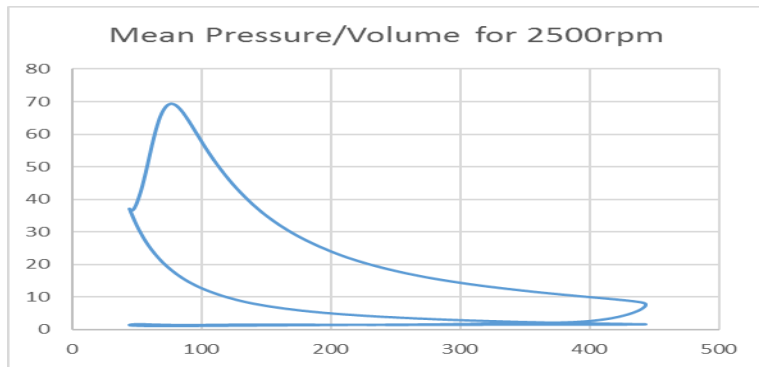


Figure 78 PV-Diagram for 2500rpm

- Engine at 2000rpm

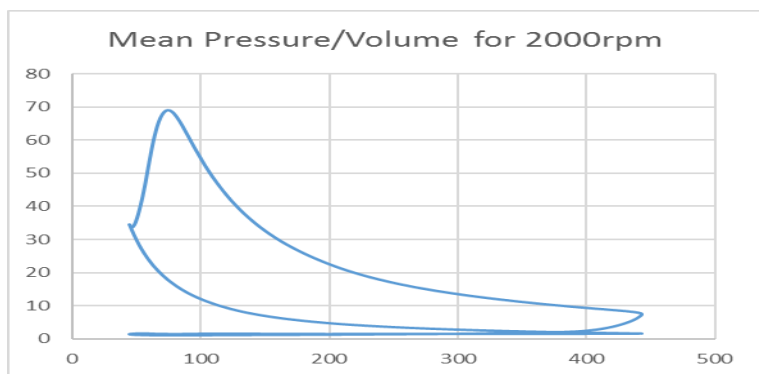


Figure 79 PV-Diagram for 2000rpm

- Engine at 1700rpm

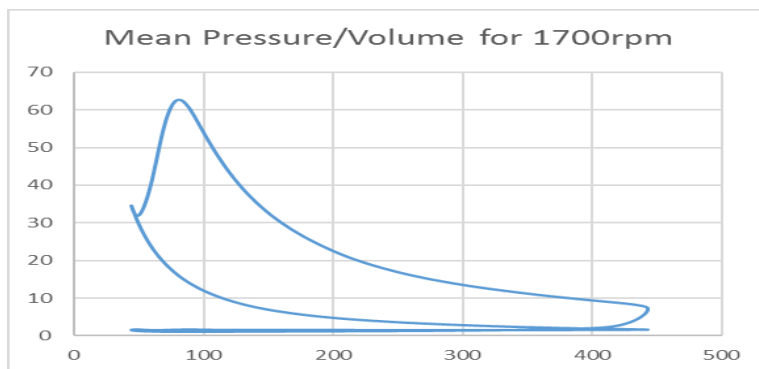


Figure 80 PV-Diagram for 1700rpm

- Engine at 1500rpm

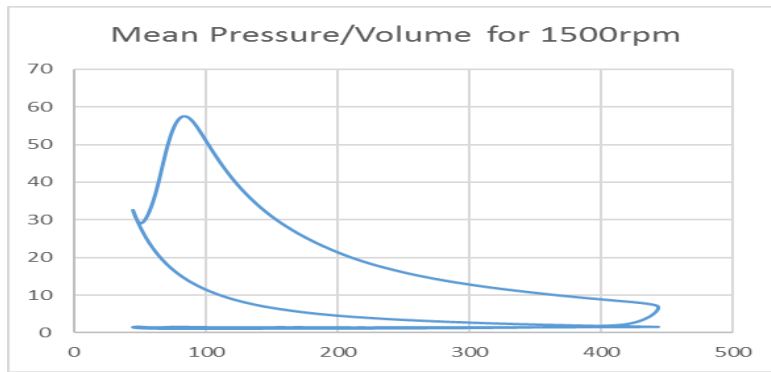


Figure 81 PV-Diagram for 1500rpm

- Engine at 1000rpm

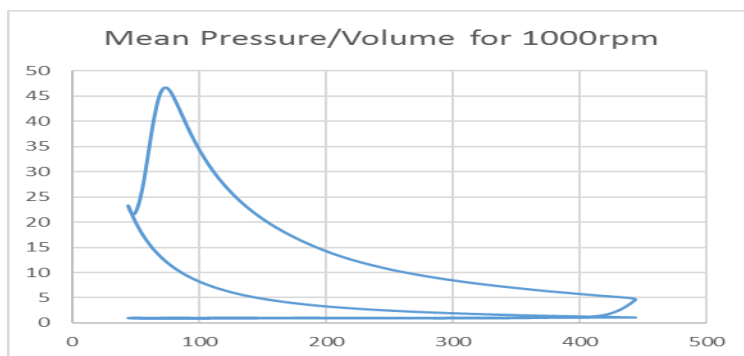


Figure 82 PV-Diagram for 1000rpm

4.2.3. Fox 1.0 Litre

- Engine at 6500rpm

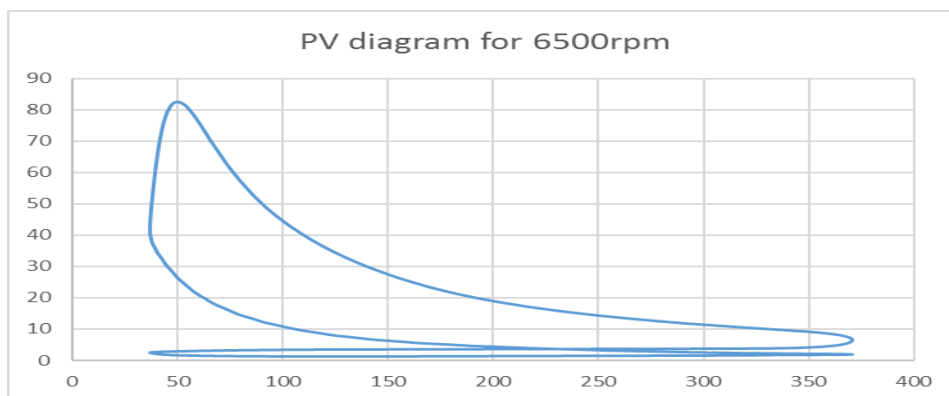


Figure 83 PV-Diagram for 6500rpm

- Engine at 6000rpm

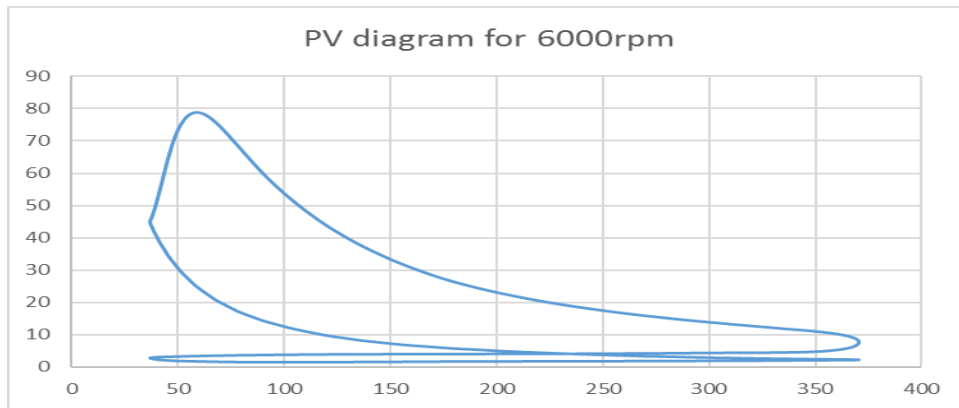


Figure 84 PV-Diagram for 6000rpm

- Engine at 5500rpm

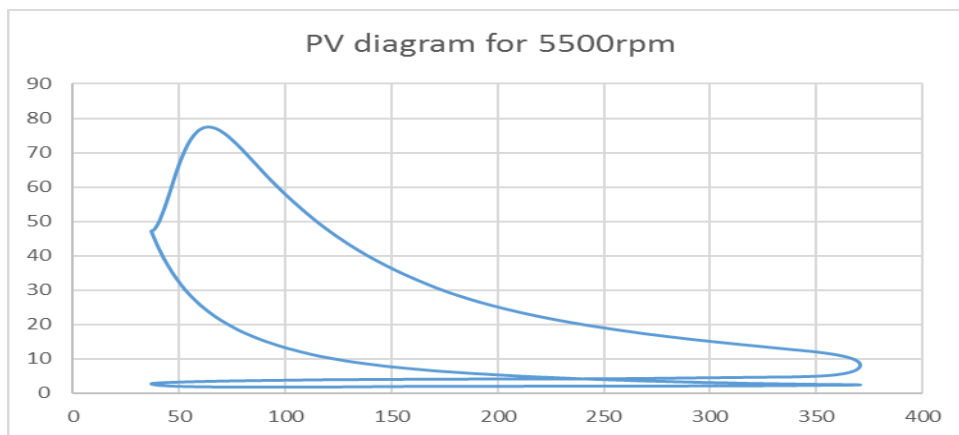


Figure 85 PV-Diagram for 5500rpm

- Engine at 5000rpm

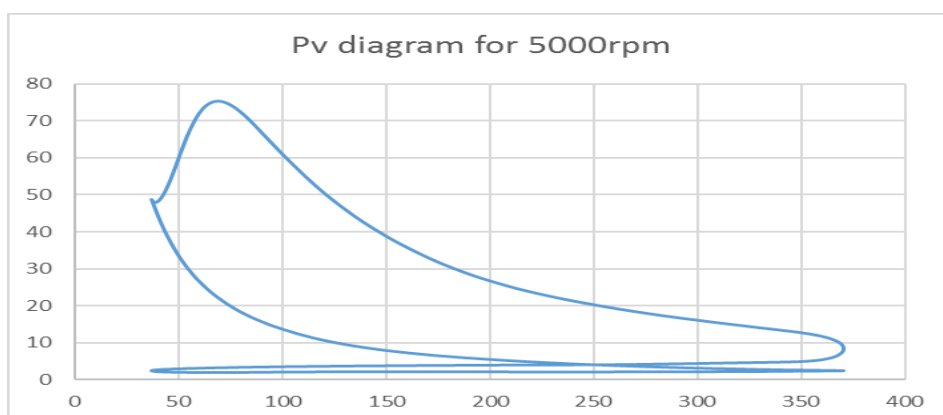


Figure 86 PV-Diagram for 5000rpm

- Engine at 4500rpm

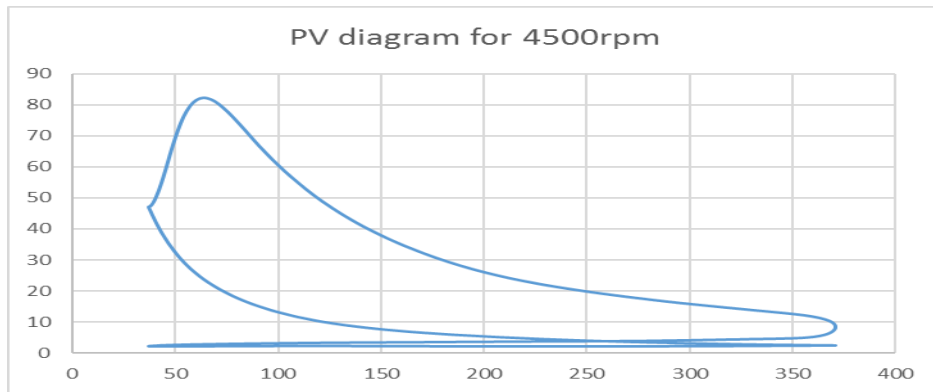


Figure 87 PV-Diagram for 4500rpm

- Engine at 4000rpm

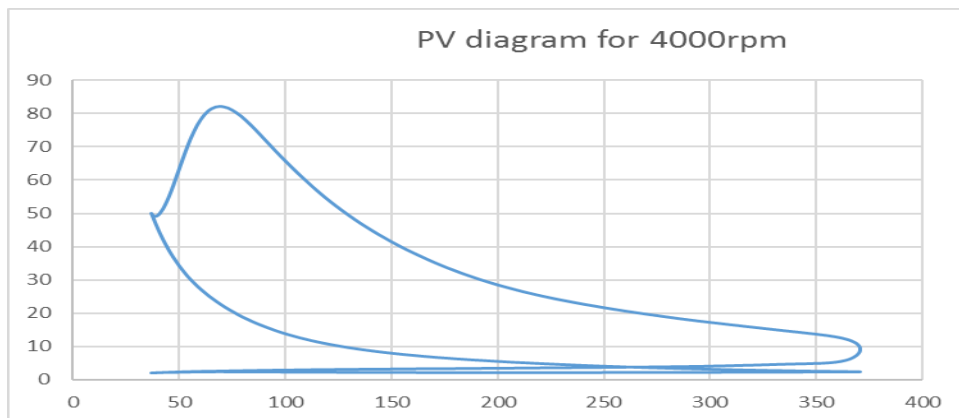


Figure 88 PV-Diagram for 4000rpm

- Engine at 3500rpm

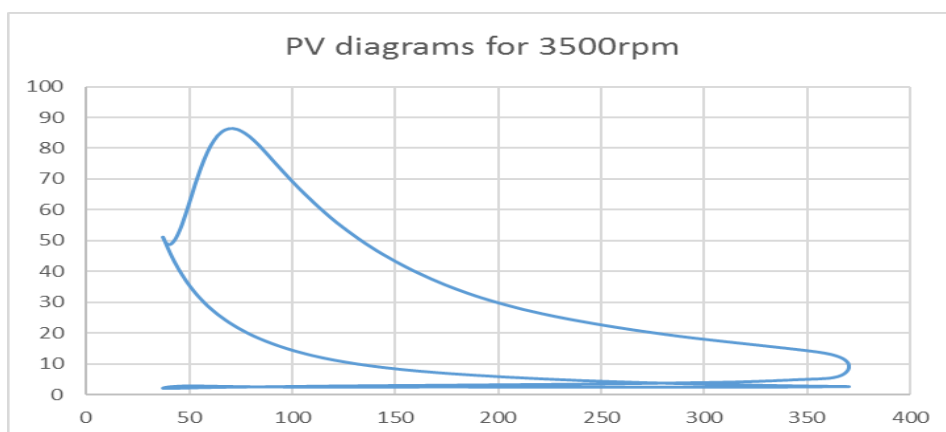


Figure 89 PV-Diagram for 3500rpm

- Engine at 3000rpm

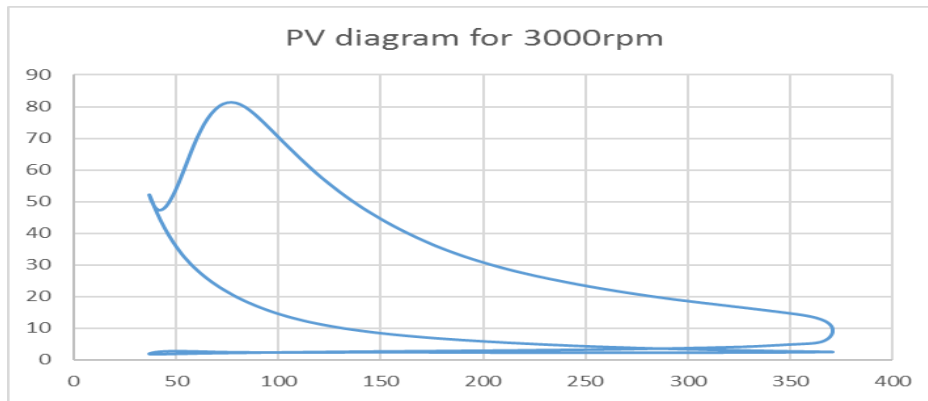


Figure 90 PV-Diagram for 3000rpm

- Engine at 2500rpm

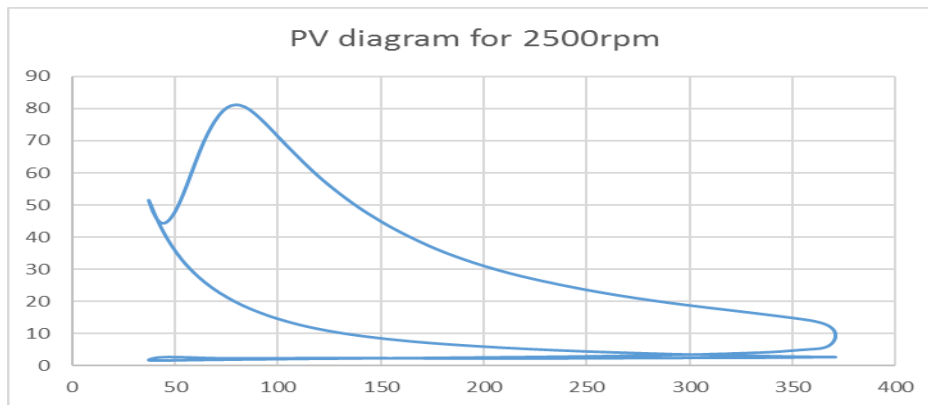


Figure 91 PV-Diagram for 2500rpm

- Engine at 2000rpm

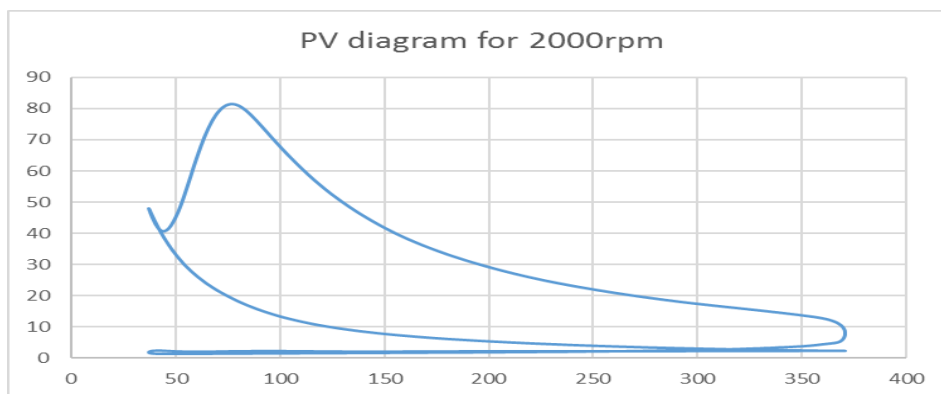


Figure 92 PV-Diagram for 2000rpm

- Engine at 1750rpm

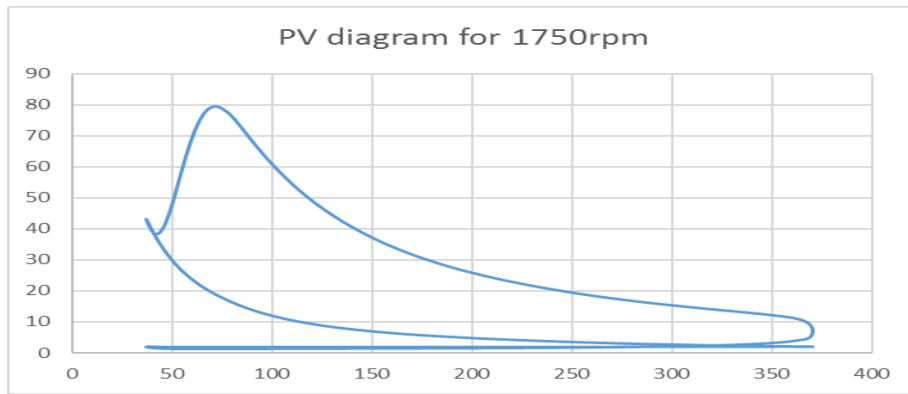


Figure 93 PV-Diagram for 1750rpm

- Engine is at 1500rpm

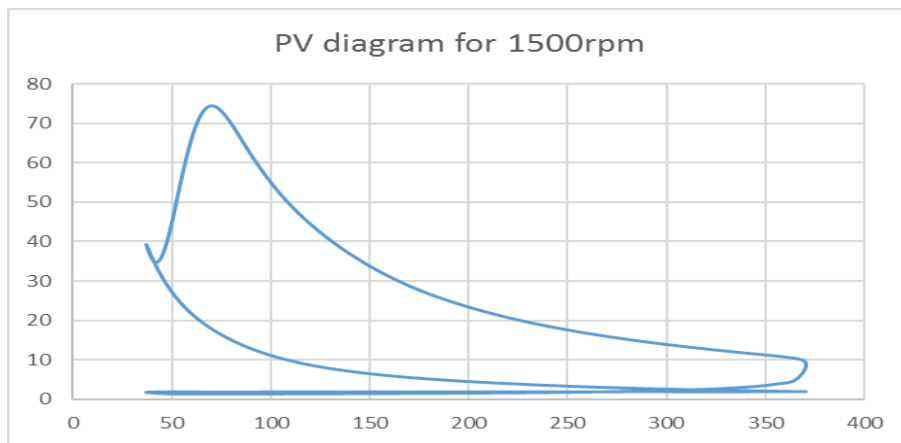


Figure 94 PV-Diagram for 1500rpm

- Engine at 1000rpm

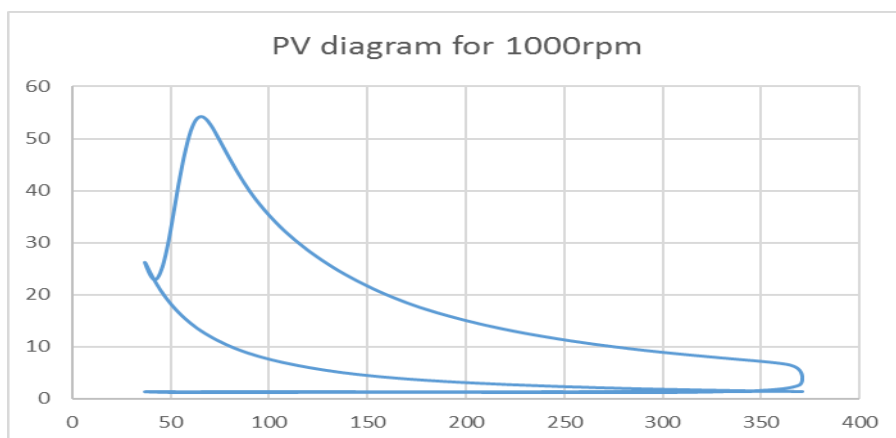


Figure 95 PV-Diagram for 1000rpm

4.2.4. Fox 1.0 Litre Full Load

Full load is recorded in log points.

- Log point 14

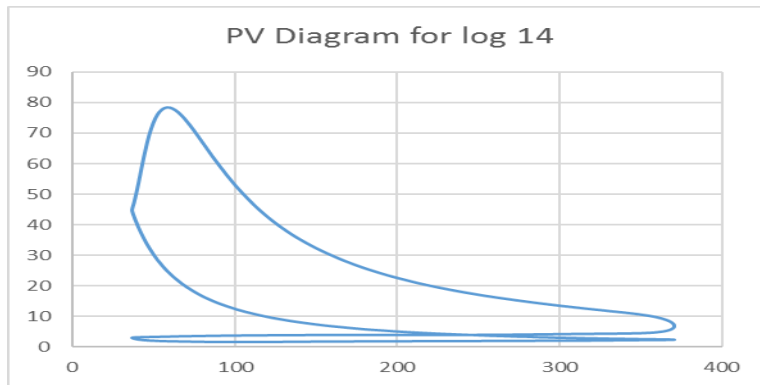


Figure 96 PV-Diagram for log 14

- Log point 10

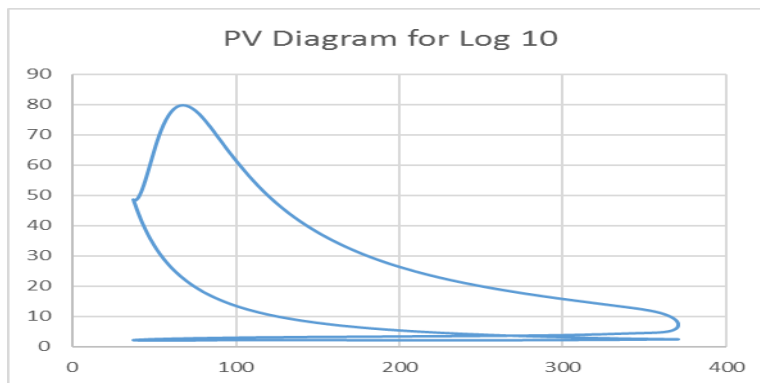


Figure 97 PV-Diagram for log 10

- Log point 7

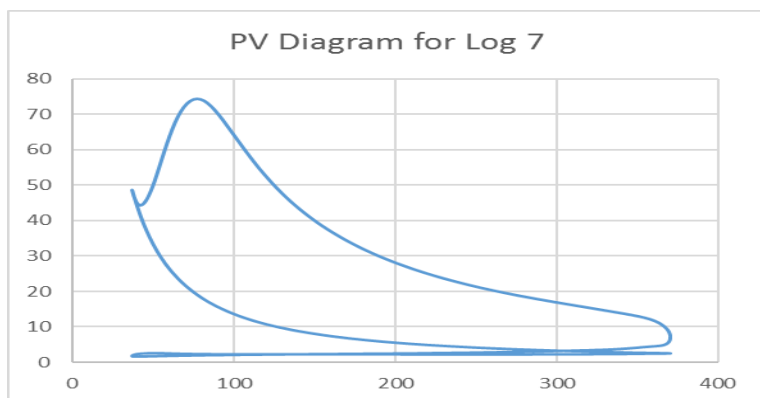


Figure 98 PV-Diagram for log 7

- Log point 5

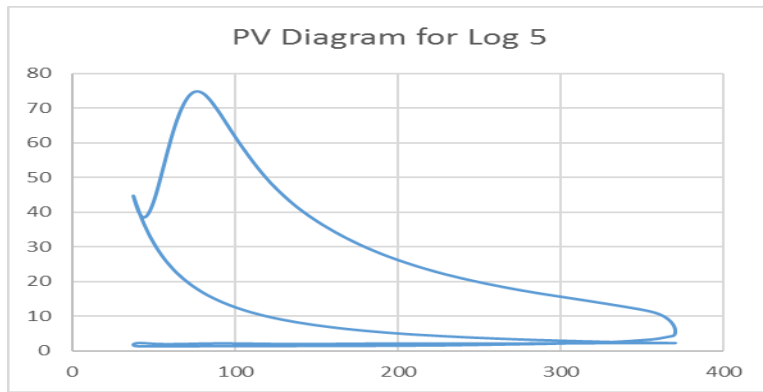


Figure 99 PV-Diagram for log 5

- Log point 1

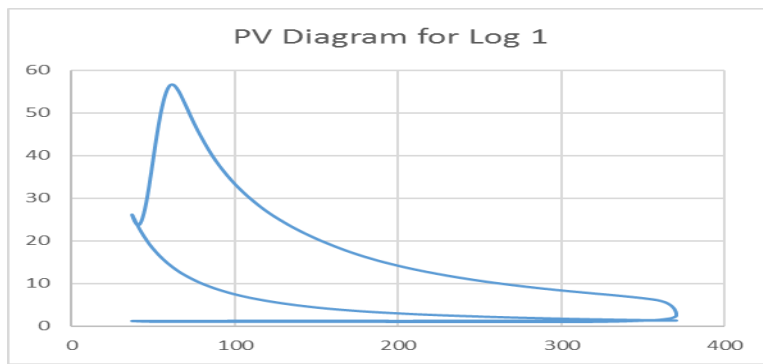


Figure 100 PV-Diagram for log 1

4.2.5. Fox 1.0 Litre Part Load

- Log point 19

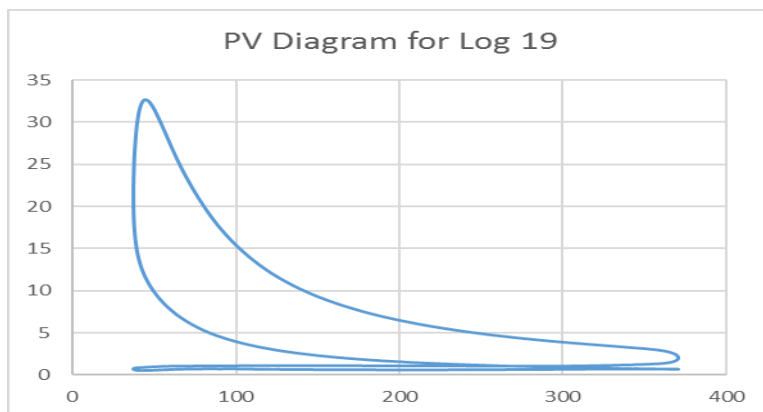


Figure 101 PV-Diagram for log 19

- Log point 18

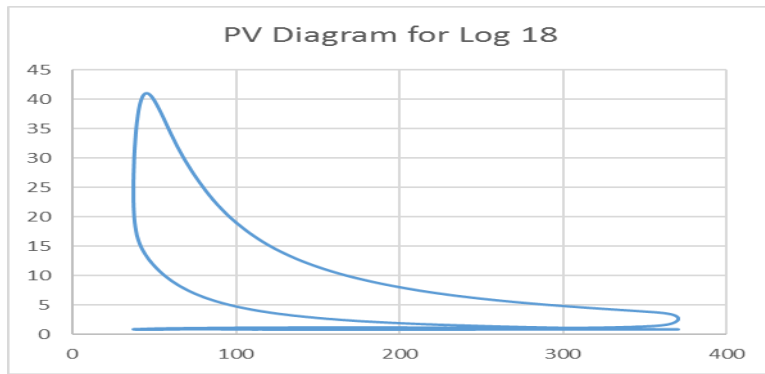


Figure 102 PV-Diagram for log 18

- Log point 14

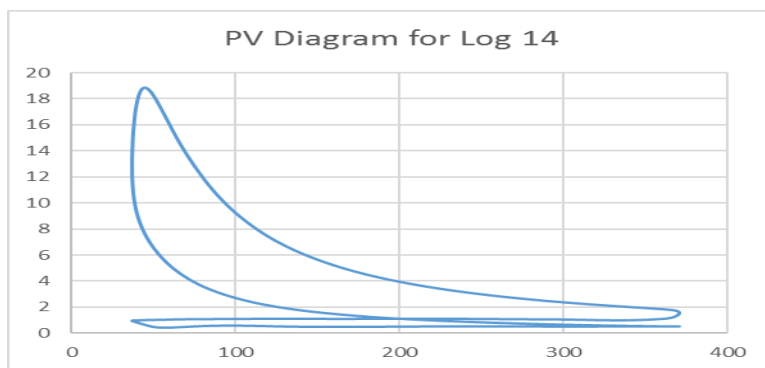


Figure 103 PV-Diagram for 14

- Log point 9

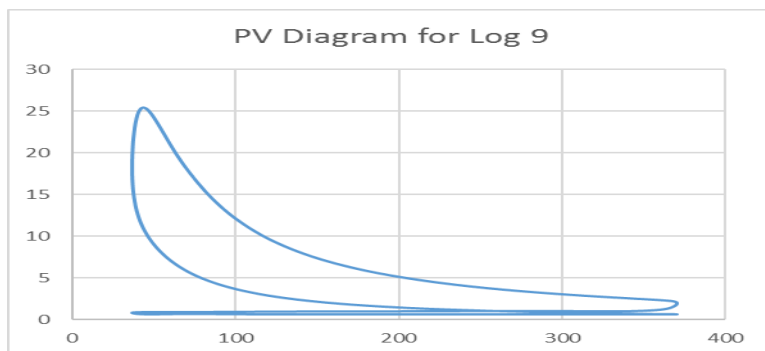


Figure 104 PV-Diagram for log 9

- Log point 4

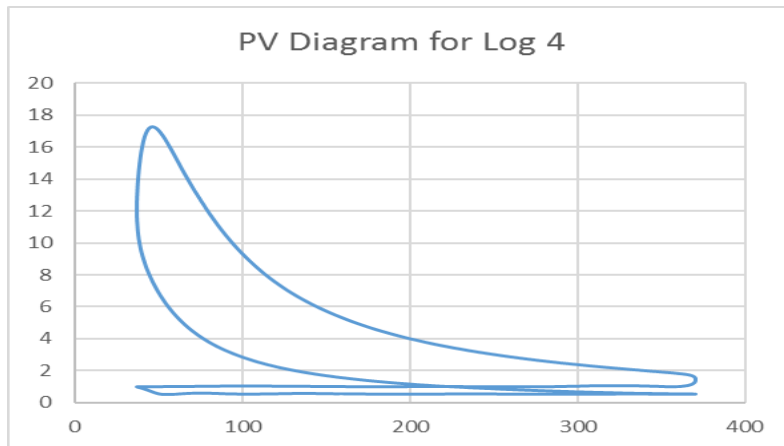


Figure 105 PV-Diagram for log 4

- Log point 3

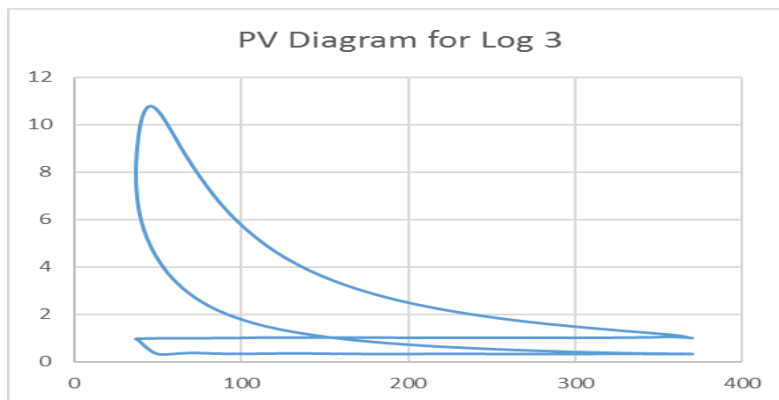


Figure 106 PV-Diagram for log 3

- Log point 1

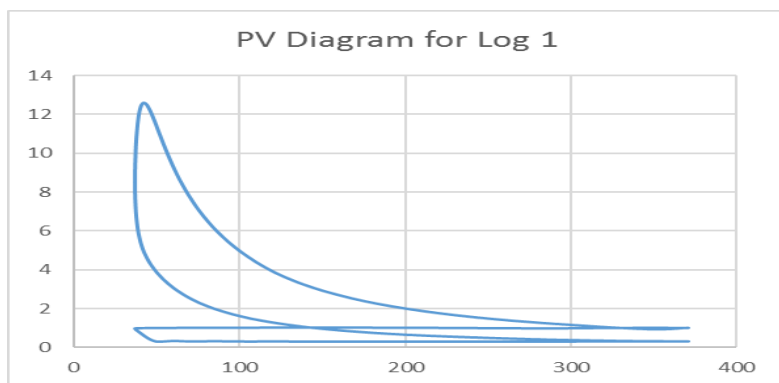


Figure 107 PV-Diagram for log 1

4.3. Log Pressure over Log Volume

Log Pressure over Log Volume

4.3.1. Sigma 1.6 Litre Naturally Aspirated

- Engine at 6500rpm

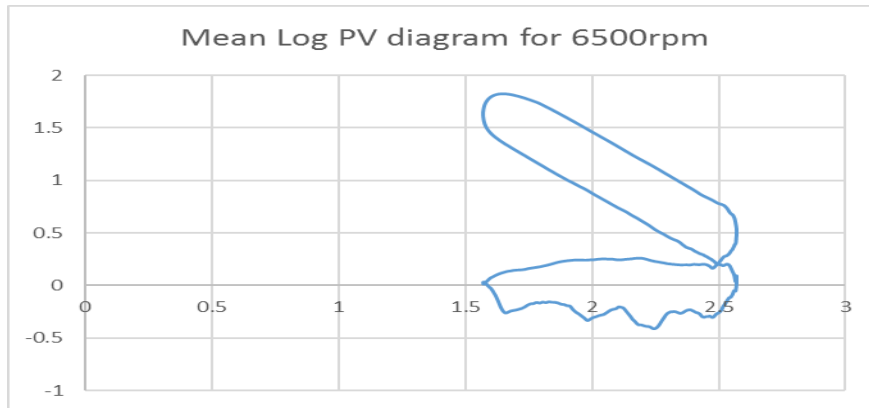


Figure 108 log PV-diagram for 6500rpm

- Engine at 6000rpm

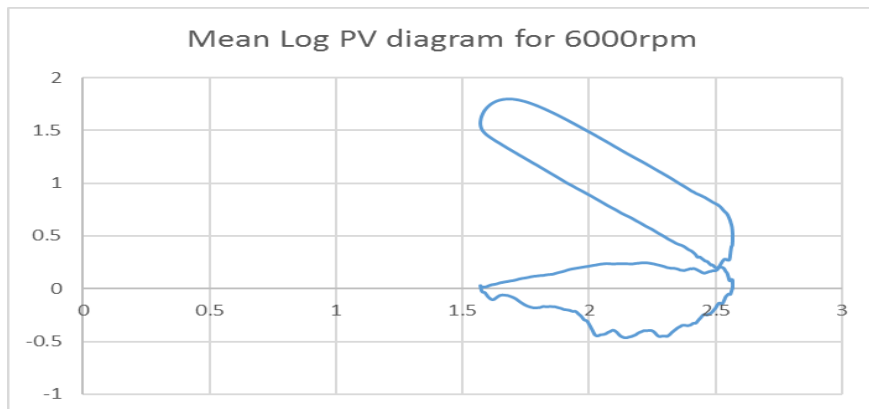


Figure 109 log PV-diagram for 6000rpm

- Engine at 5500rpm

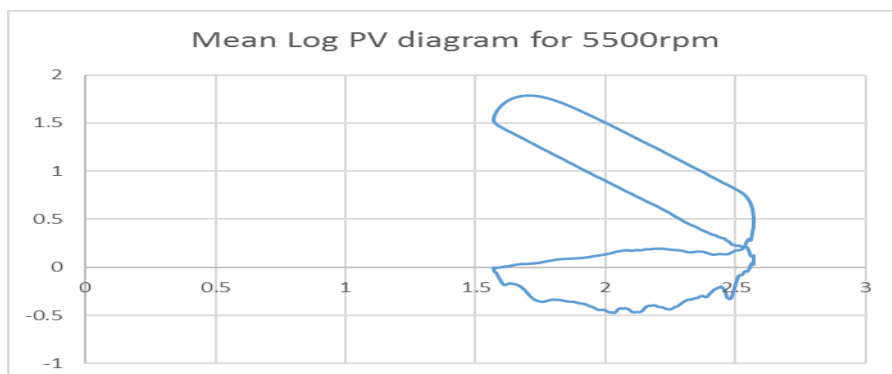


Figure 110 log PV-diagram for 5500rpm

- Engine at 5000rpm

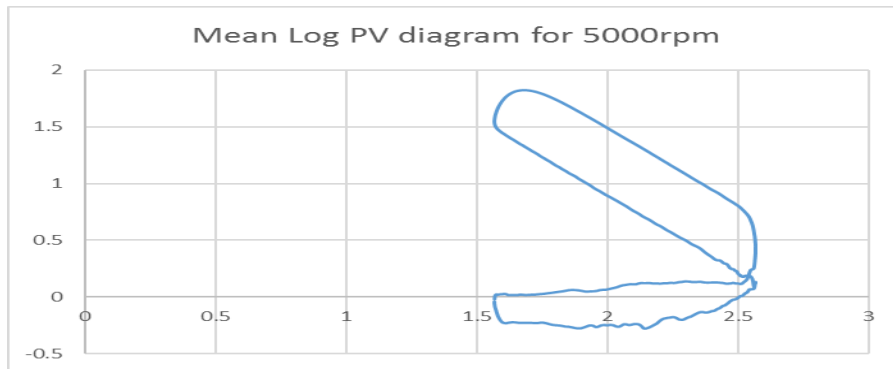


Figure 111 log PV-diagram for 5000rpm

- Engine at 4500rpm

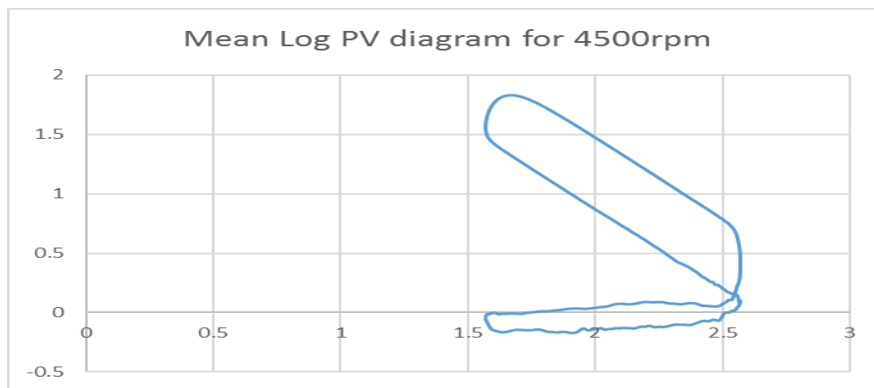


Figure 112 log PV-diagram for 4500rpm

- Engine at 4250rpm

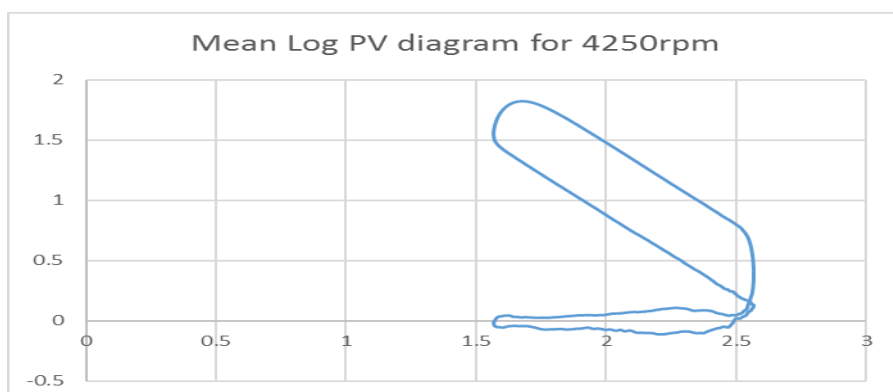


Figure 113 log PV-diagram for 4250rpm

- Engine at 4000rpm

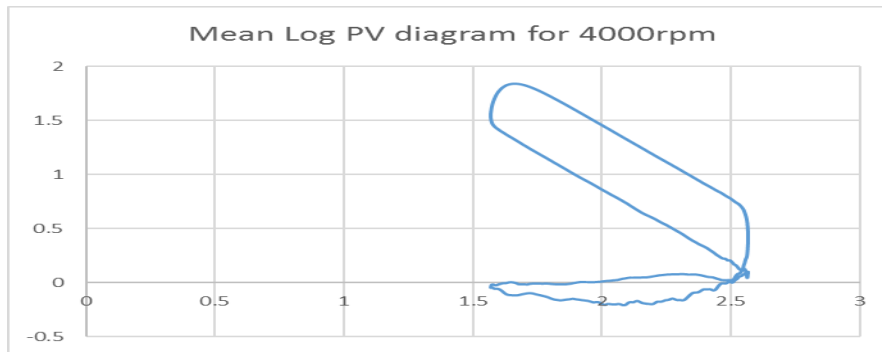


Figure 114 log PV-diagram for 4000rpm

- Engine at 3500rpm

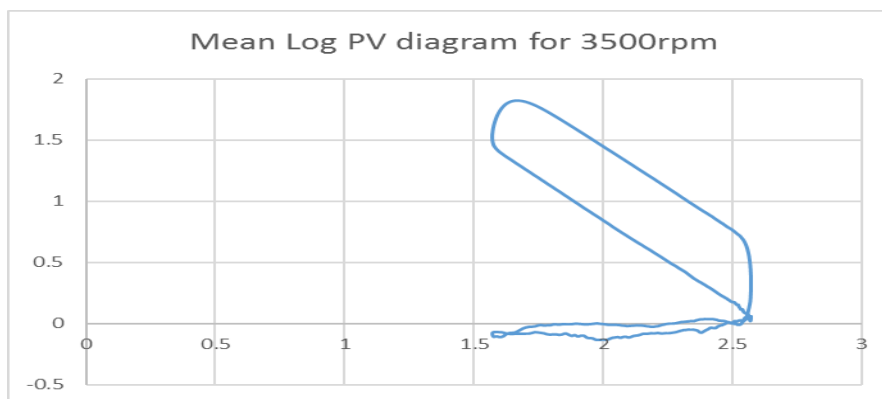


Figure 115 log PV-diagram for 3500rpm

- Engine at 3000rpm

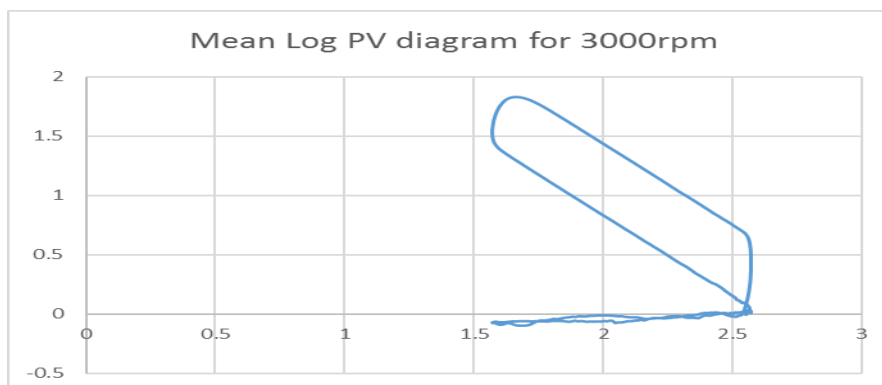


Figure 116 log PV-diagram for 3000rpm

- Engine at 2500rpm

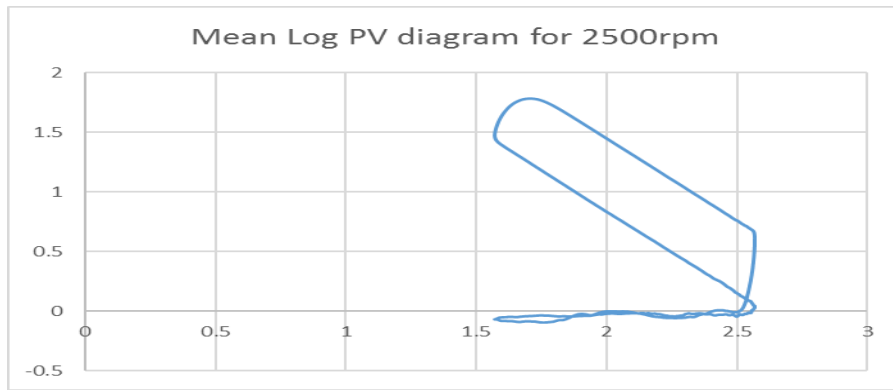


Figure 117 log PV-diagram for 2500rpm

- Engine at 2000rpm

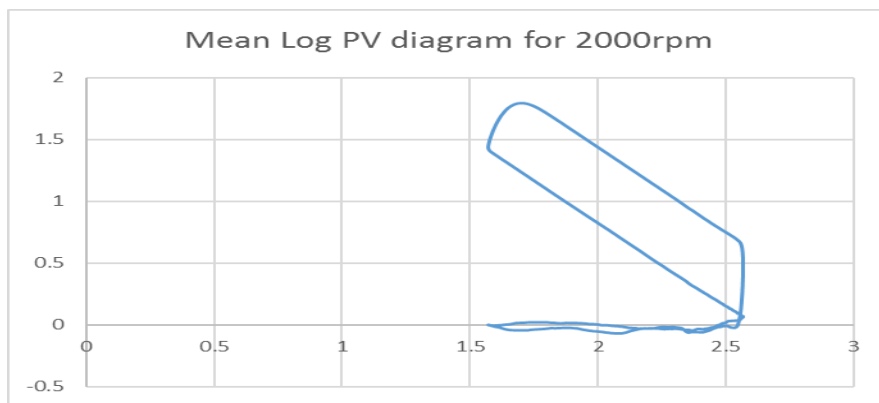


Figure 118 log PV-diagram for 2000rpm

- Engine at 1750rpm

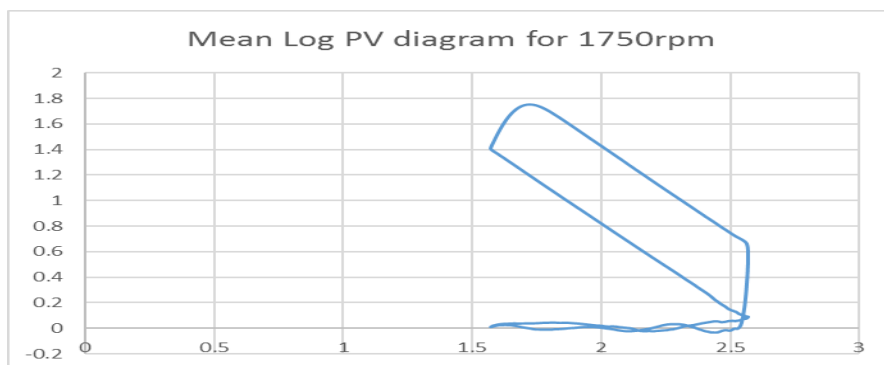


Figure 119 log PV-diagram for 1750rpm

- Engine at 1500rpm

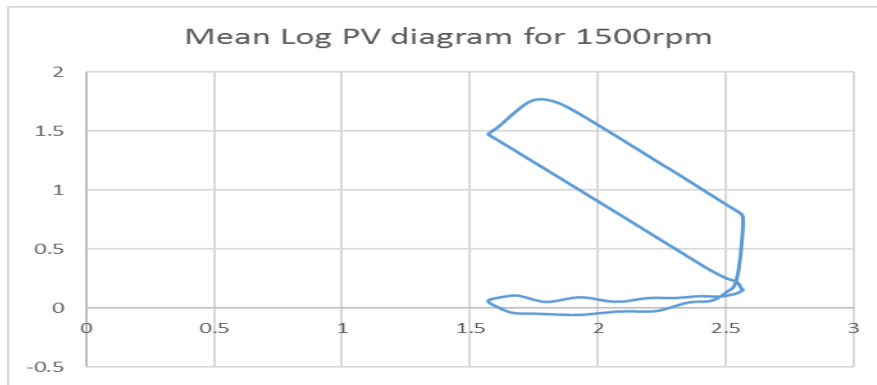


Figure 120 log PV-diagram for 1500rpm

- Engine at 1000rpm

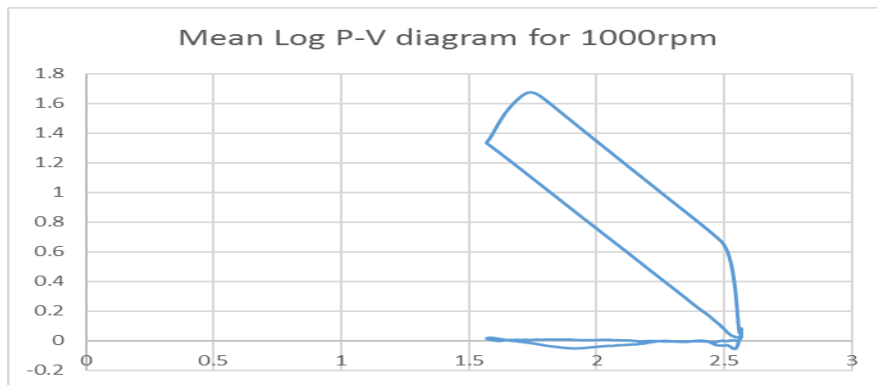


Figure 121 log PV-diagram for 1000rpm

4.3.2. 1.6 Litre Turbocharged

For the 1.6 Turbocharged engine.

- Engine at 6000rpm

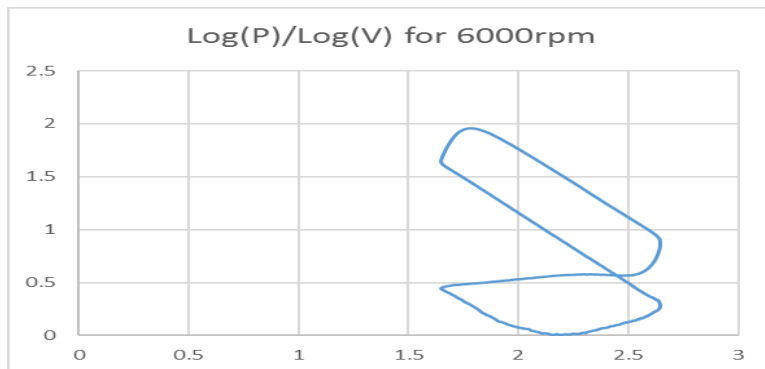


Figure 122 log PV-diagram for 6000rpm

- Engine at 5700rpm

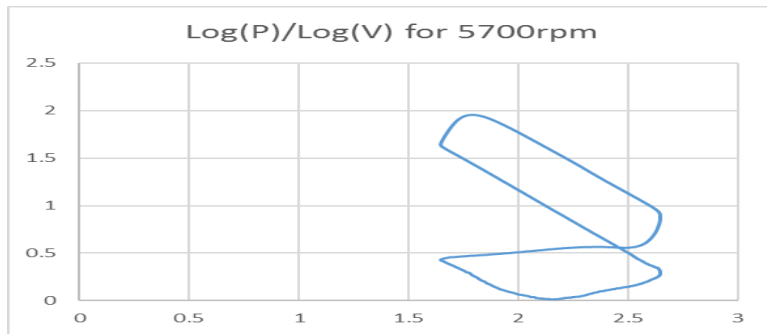


Figure 123 log PV-diagram for 5700rpm

- Engine at 5500rpm

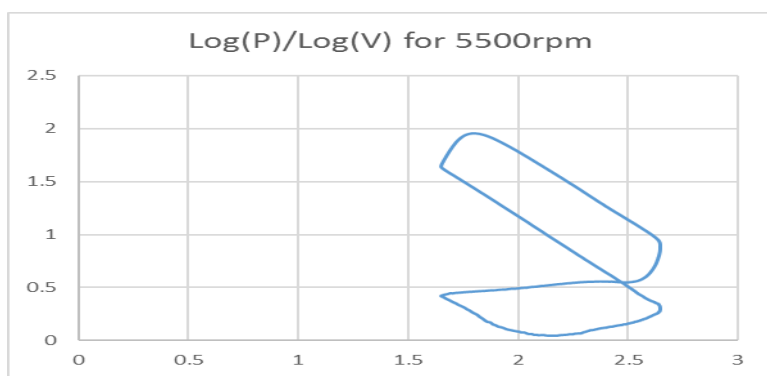


Figure 124 log PV-diagram for 5500rpm

- Engine at 5000rpm

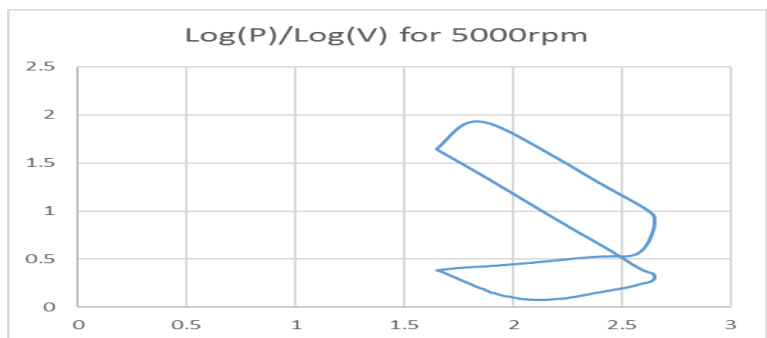


Figure 125 log PV-diagram for 5000rpm

- Engine at 4500rpm

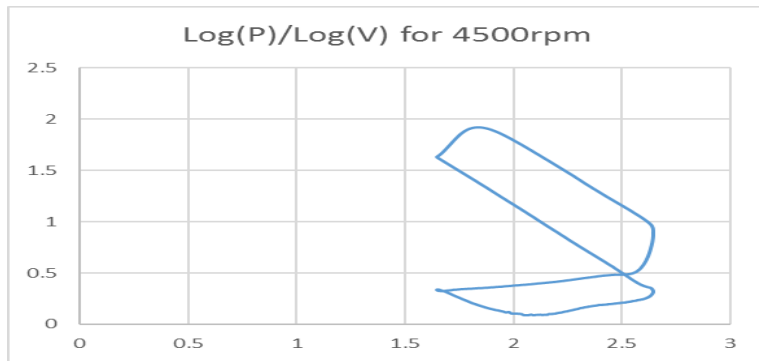


Figure 126 log PV-diagram for 4500rpm

- Engine at 4000rpm

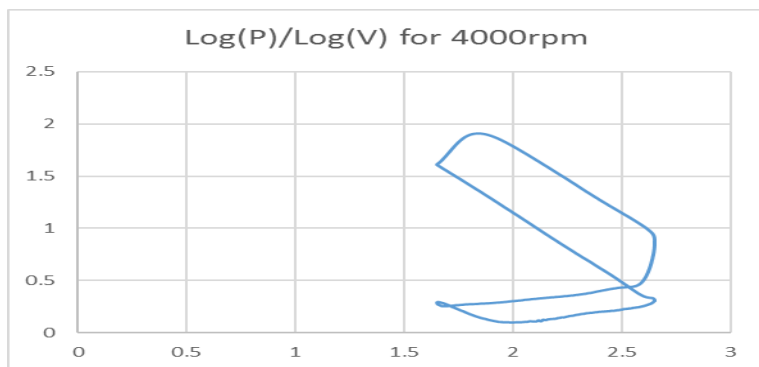


Figure 127 log PV-diagram for 4000rpm

- Engine at 3500rpm

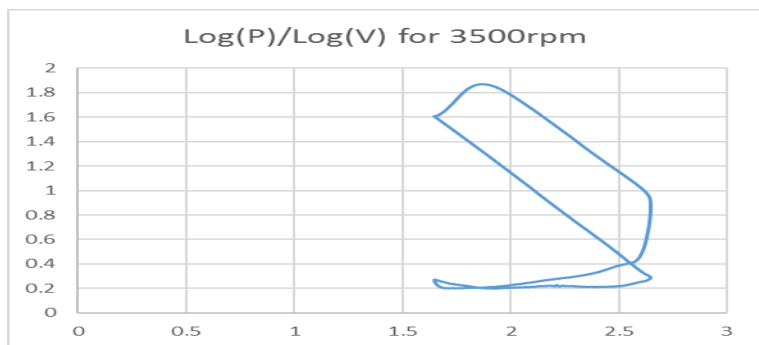


Figure 128 log PV-diagram for 3500rpm

- Engine at 3000rpm

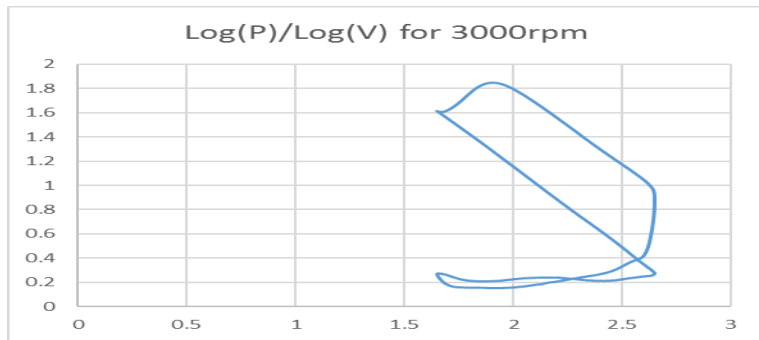


Figure 129 log PV-diagram for 3000rpm

- Engine at 2500rpm

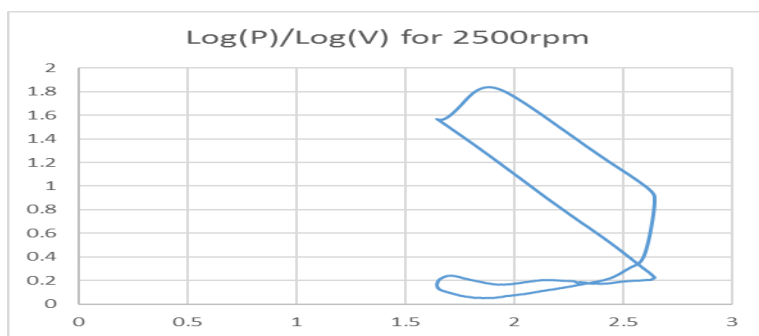


Figure 130 log PV-diagram for 2500rpm

- Engine at 2000rpm

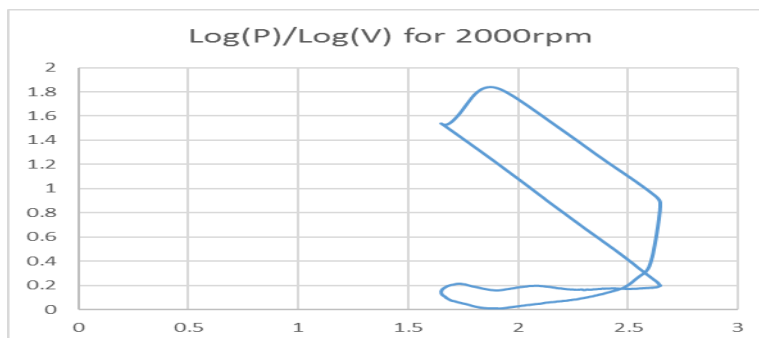


Figure 131 log PV-diagram for 2000rpm

- Engine at 1700rpm

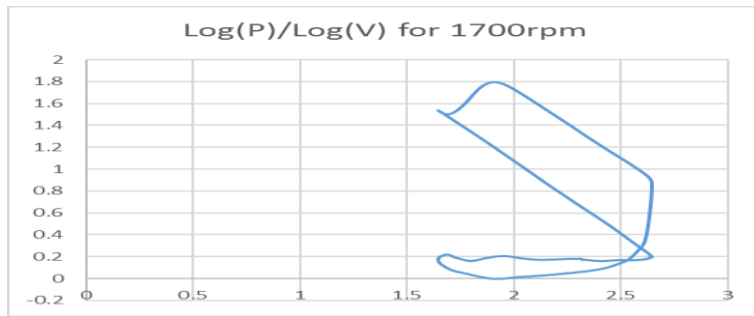


Figure 132 log PV-diagram for 1700rpm

- Engine at 1500rpm

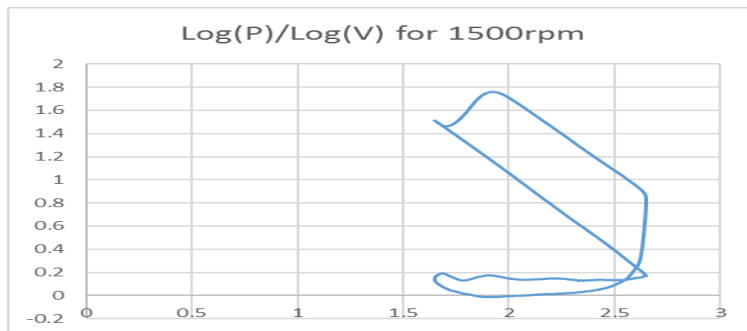


Figure 133 log PV-diagram for 1500rpm

- Engine at 1000rpm

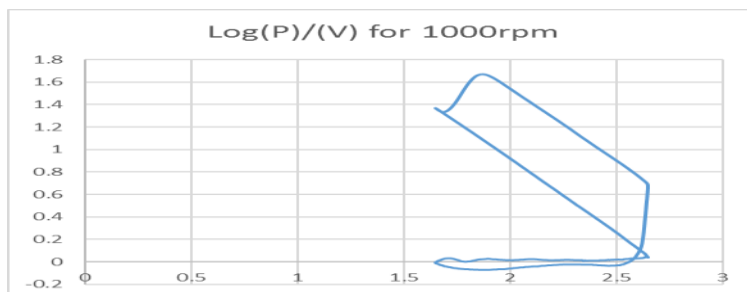


Figure 134 log PV-diagram for 1000rpm

4.3.3. Fox 1.0 Litre

- Engine at 6500rpm

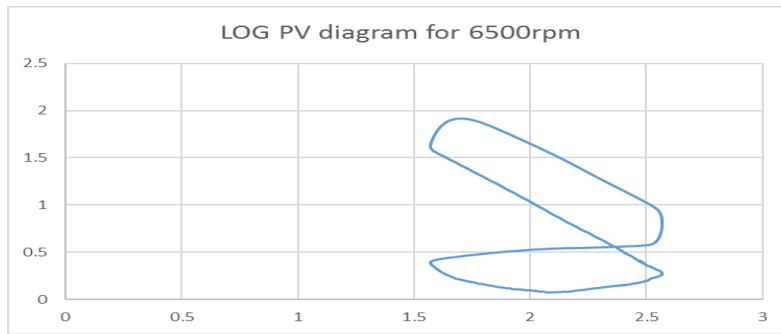


Figure 135 log PV-diagram for 6500rpm

- Engine at 6000rpm

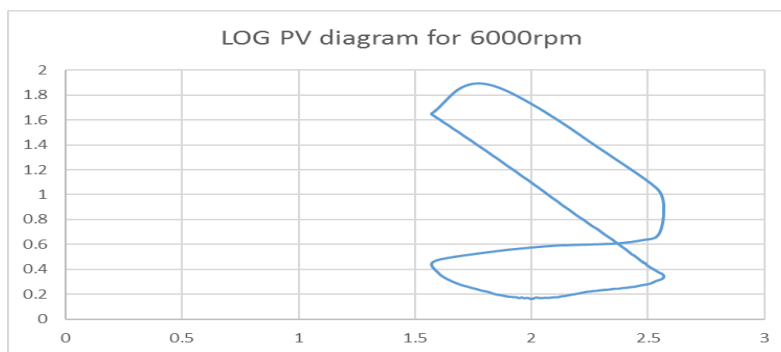


Figure 136 log PV-diagram for 6000rpm

- Engine at 5500rpm

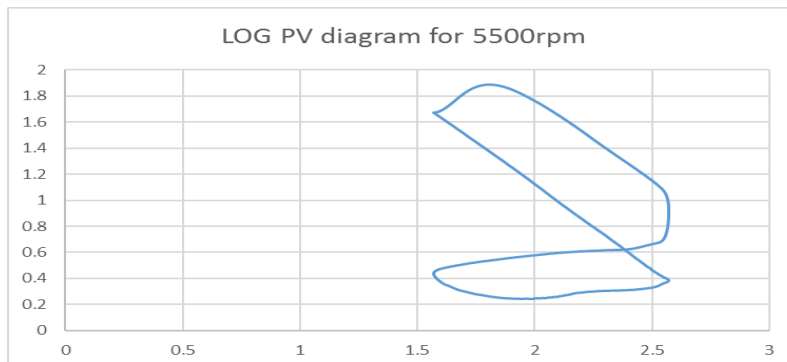


Figure 137 log PV-diagram for 5500rpm

- Engine at 5000rpm

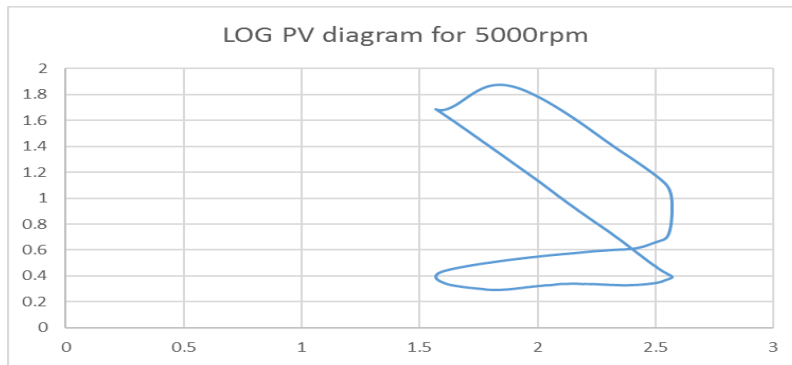


Figure 138 log PV-diagram for 5000rpm

- Engine at 4500rpm

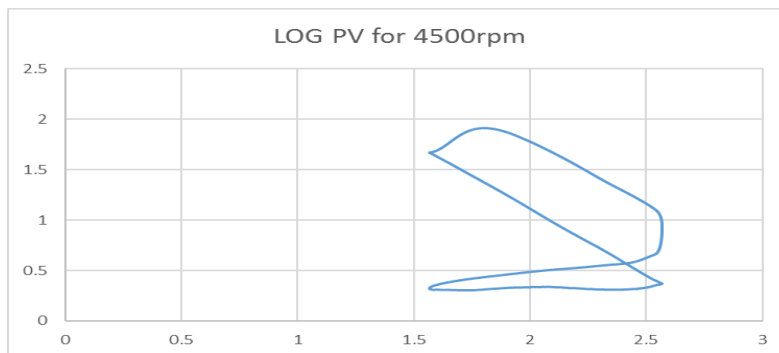


Figure 139 log PV-diagram for 4500rpm

- Engine at 4000rpm

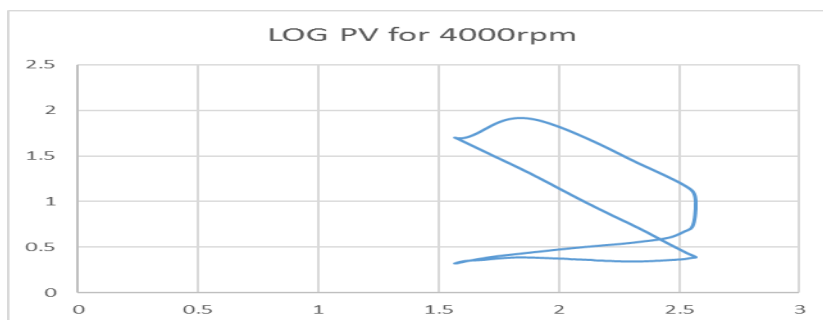


Figure 140 log PV-diagram for 4000rpm

- Engine at 3500rpm

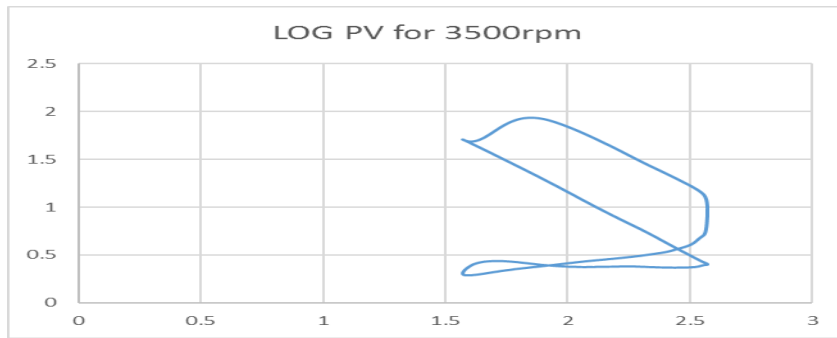


Figure 141 log PV-diagram for 3500rpm

- Engine at 3000rpm

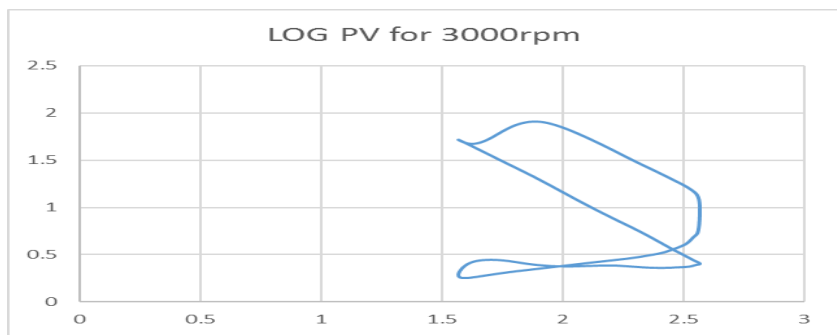


Figure 142 log PV-diagram for 3000rpm

- Engine at 2500rpm

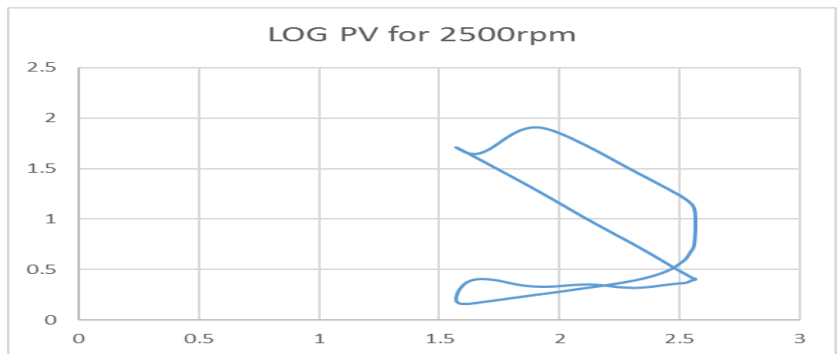


Figure 143 log PV-diagram for 2500rpm

- Engine at 2000rpm

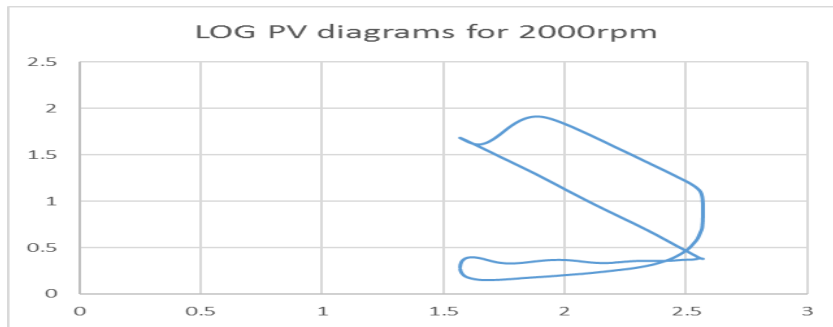


Figure 144 log PV-diagram for 2000rpm

- Engine at 1750rpm

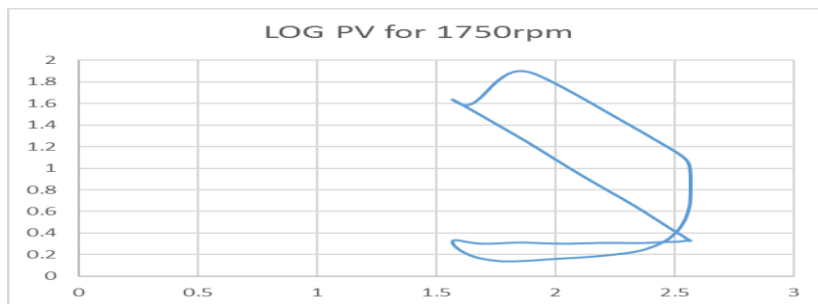


Figure 145 log PV-diagram for 1750rpm

- Engine at 1500rpm

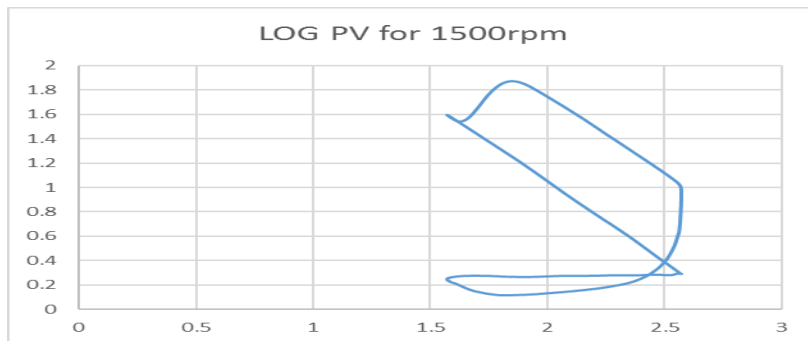


Figure 146 log PV-diagram for 1500rpm

- Engine at 1000rpm

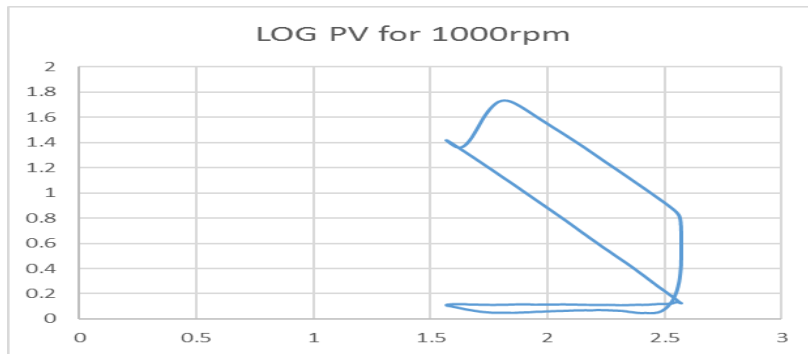


Figure 147 log PV-diagram for 1000rpm

4.3.4. Fox 1.0 Litre Full Load

- Log point 14

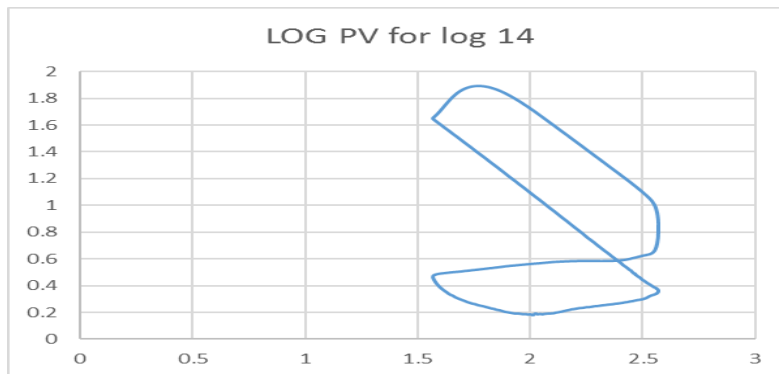


Figure 148 log PV-diagram for log 14

- Log point 10

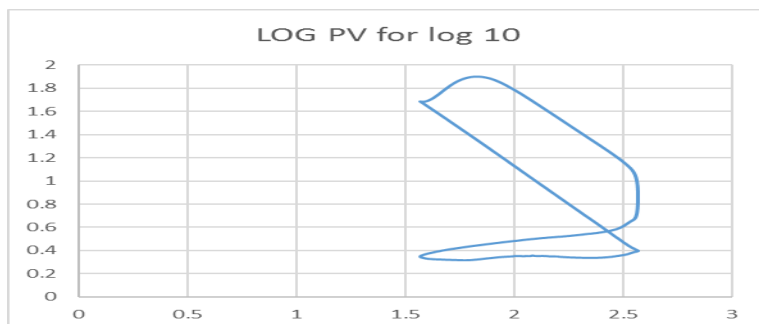


Figure 149 log PV-diagram for log 10

- Log point 7

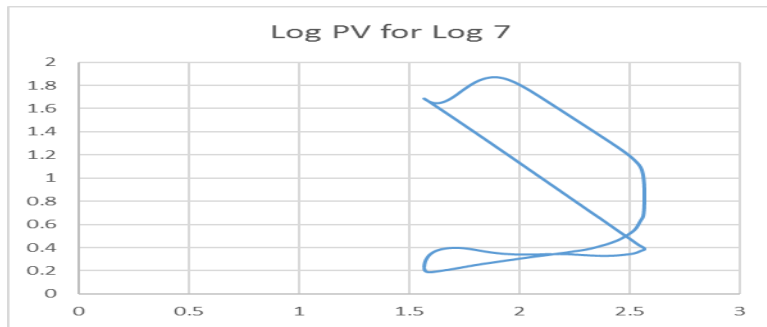


Figure 150 log PV-diagram for log 7

- Log point 5

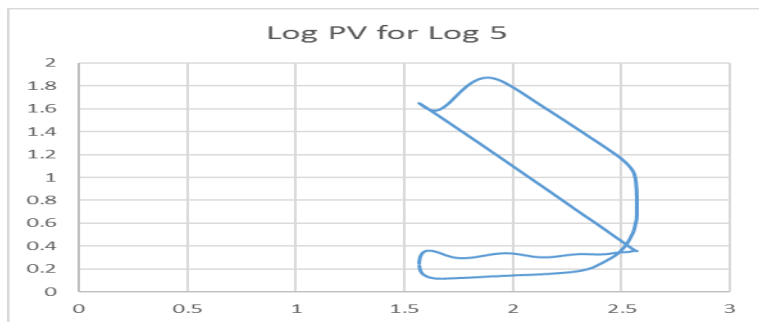


Figure 151 log PV-diagram for log 5

- Log point 1

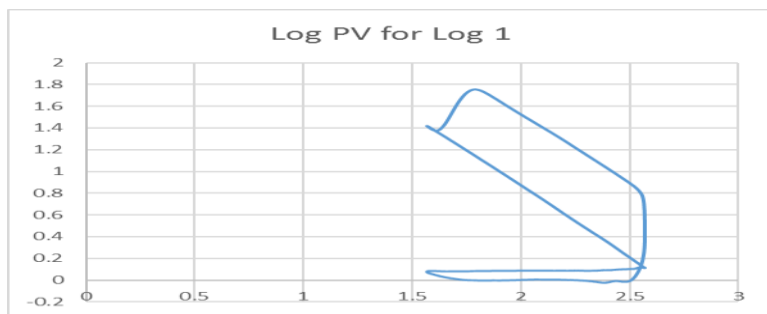


Figure 152 log PV-diagram for log 1

4.3.5. Fox 1.0 Litre Part Load

- Log point 19

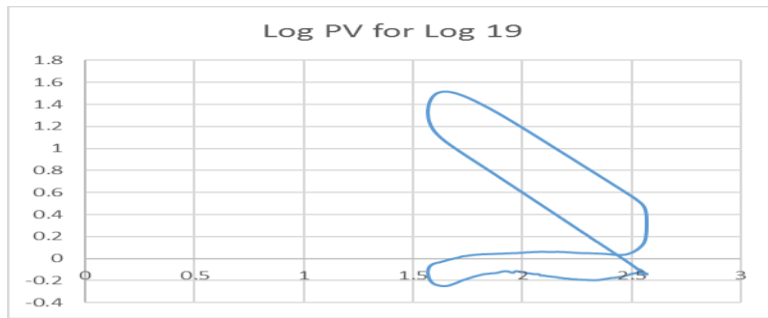


Figure 153 log PV-diagram for log 19

- Log point 18

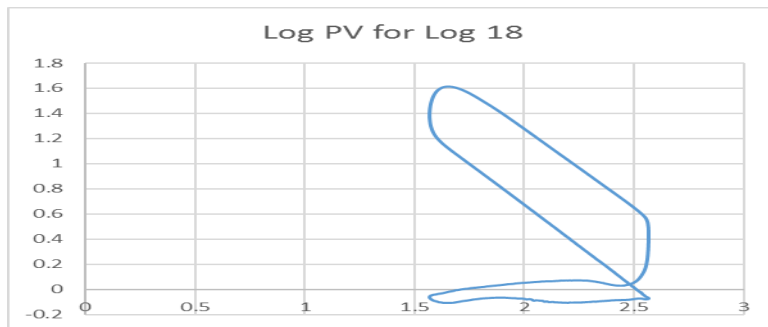


Figure 154 log PV-diagram for log 18

Log point 14

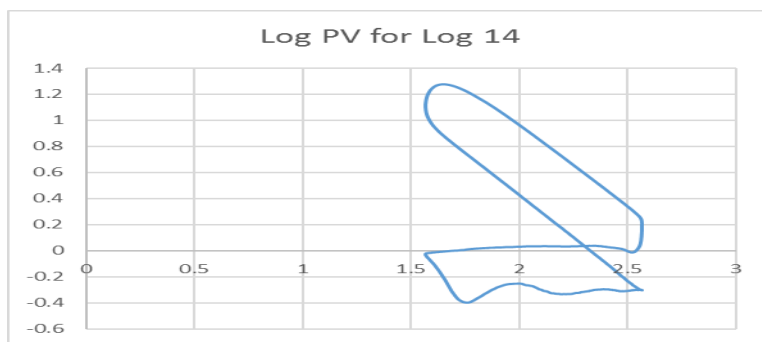


Figure 155 log PV-diagram for log 14

- Log point 9

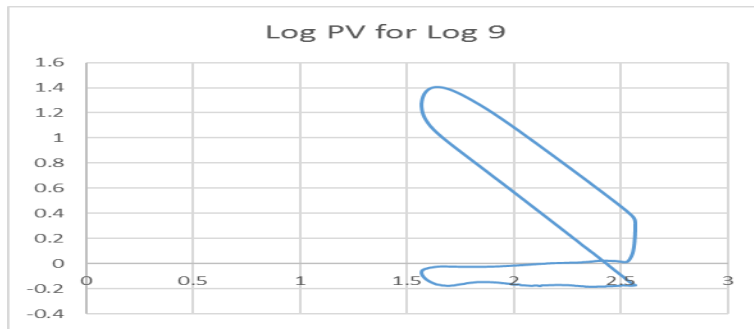


Figure 156 log PV-diagram for log 9

- Log point 4

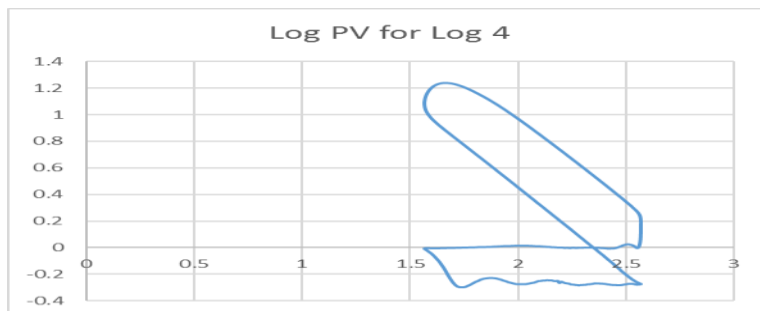


Figure 157 log PV-diagram for log 4

- Log point 3

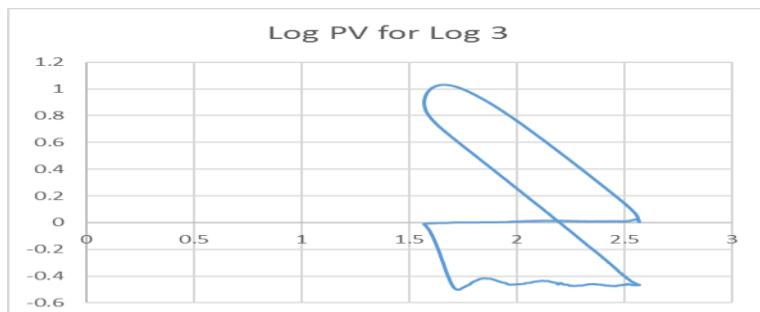


Figure 158 log PV-diagram for log 3

- Log point 1

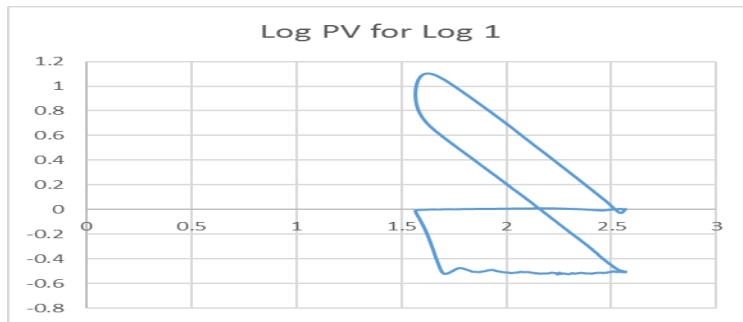


Figure 159 log PV-diagram for log 1

4.4. Finding Pressure Change from Combustion (ΔP , $P_2 - P_{2n}$)

The Pressure change that's occurs just because of the piston movement is subtracted from the overall cylinder pressure to find the amount of pressure change that occurs due to combustion taking place. This is also used to find the start and end of the combustion process.

4.4.1. Sigma 1.6 Litre Naturally Aspirated

- Engine at 6500rpm

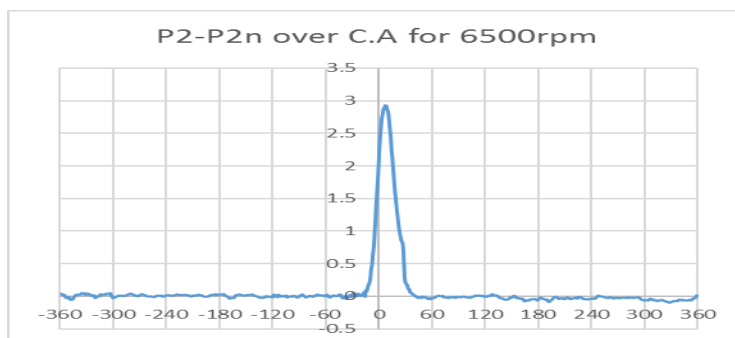


Figure 160 ΔP for 6500rpm

- Engine at 6000rpm

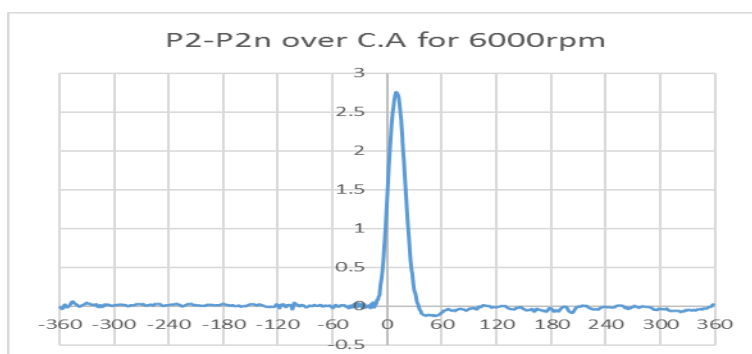


Figure 161 ΔP for 6000rpm

- Engine at 5500rpm:

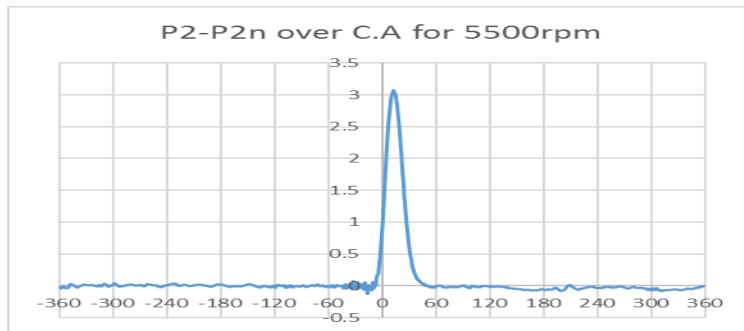


Figure 162 ΔP for 5500rpm

- Engine at 5000rpm

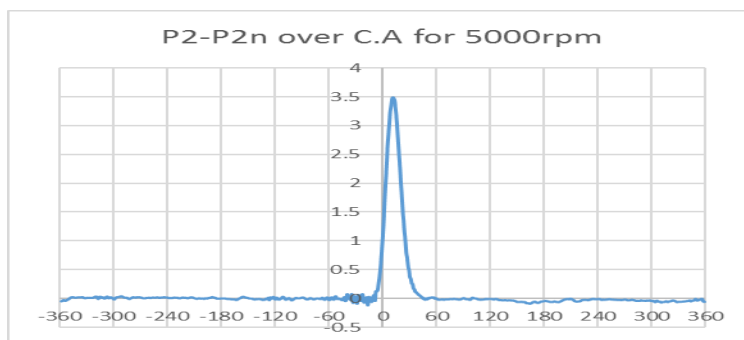


Figure 163 ΔP for 5000rpm

- Engine at 4500rpm:

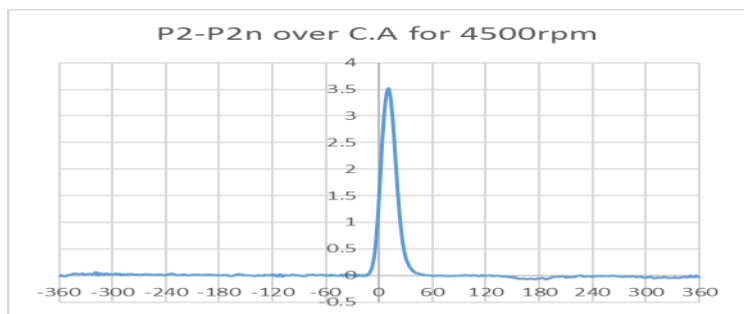


Figure 164 ΔP for 4500rpm

- Engine at 4250rpm

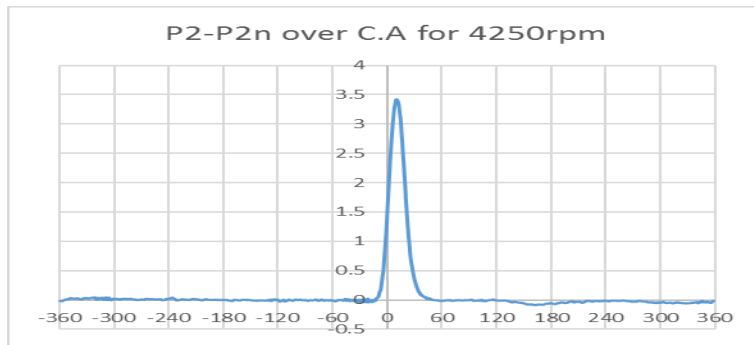


Figure 165 ΔP for 4250rpm

- Engine at 4000rpm

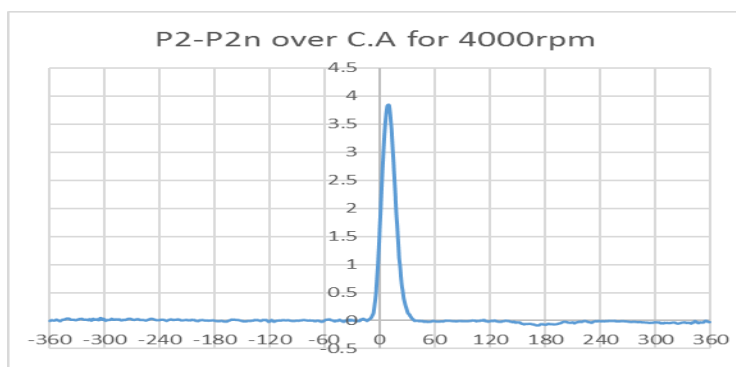


Figure 166 ΔP for 4000rpm

- Engine at 3500rpm

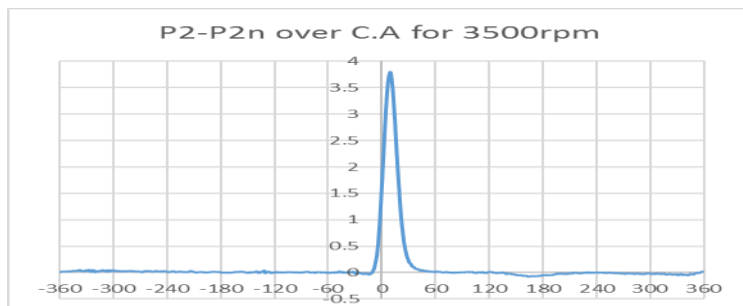


Figure 167 ΔP for 3500rpm

- Engine at 3000rpm

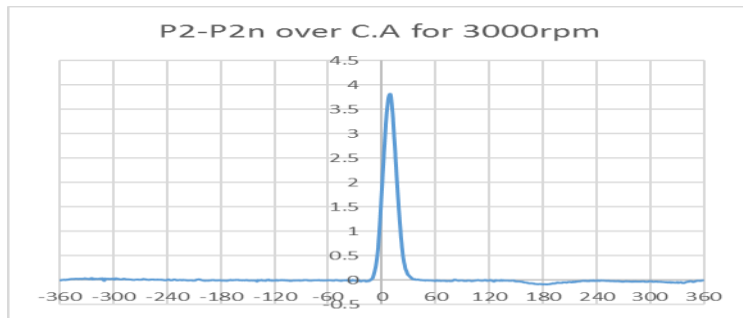


Figure 168 ΔP for 3000rpm

- Engine at 2500rpm

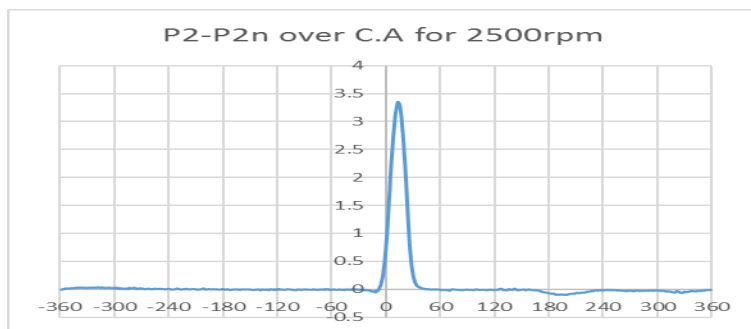


Figure 169 ΔP for 2500rpm

- Engine at 2000rpm

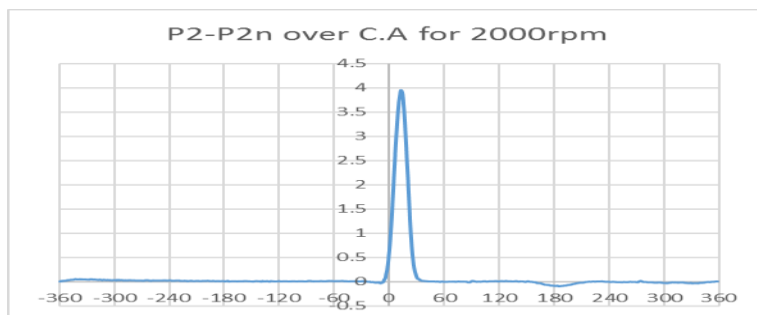


Figure 170 ΔP for 2000rpm

- Engine at 1750rpm

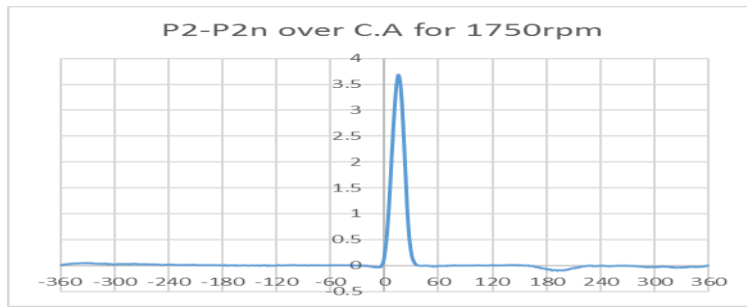


Figure 171 ΔP for 1750rpm

- Engine at 1500rpm

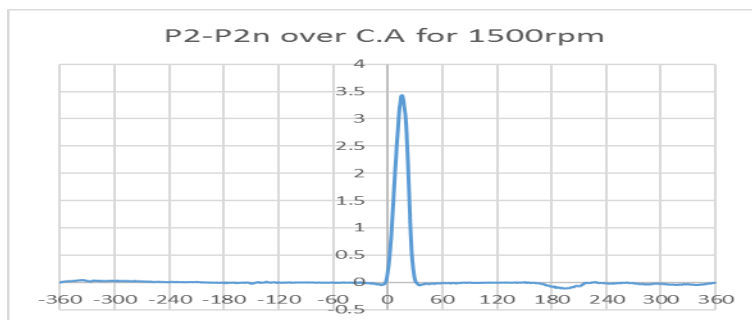


Figure 172 ΔP for 1500rpm

- Engine at 1000rpm

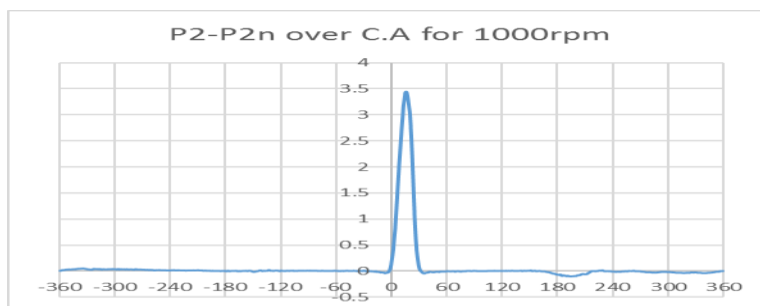


Figure 173 ΔP for 1000rpm

4.4.2. 1.6 Litre Turbocharged

- Engine at 6000rpm

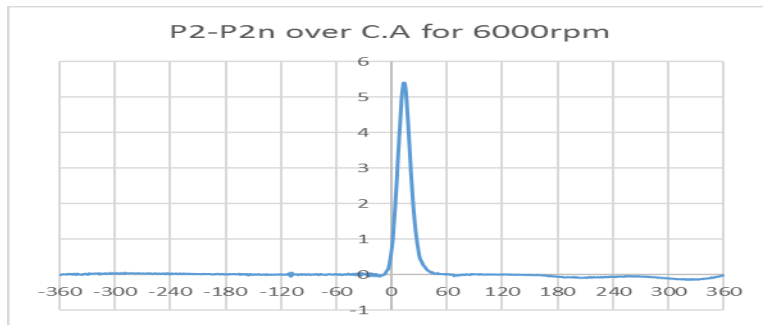


Figure 174 ΔP for 6000rpm

- Engine at 5700rpm

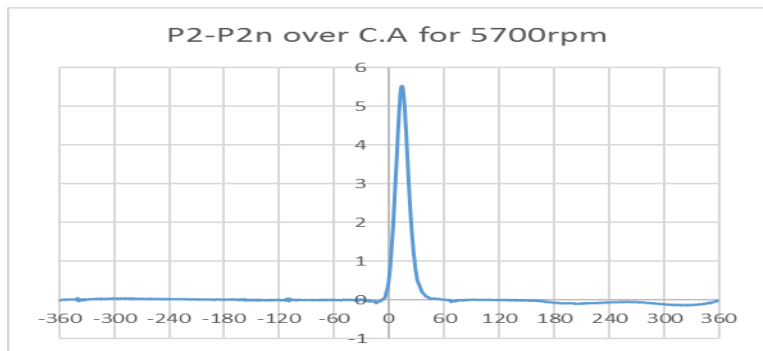


Figure 175 ΔP for 5700rpm

- Engine at 5500rpm

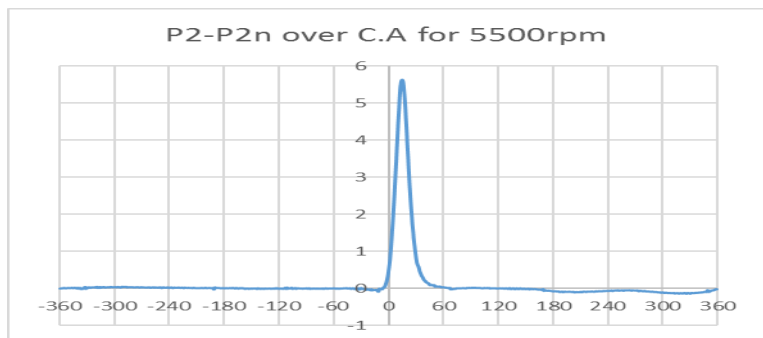


Figure 176 ΔP for 5500rpm

- Engine at 5000rpm

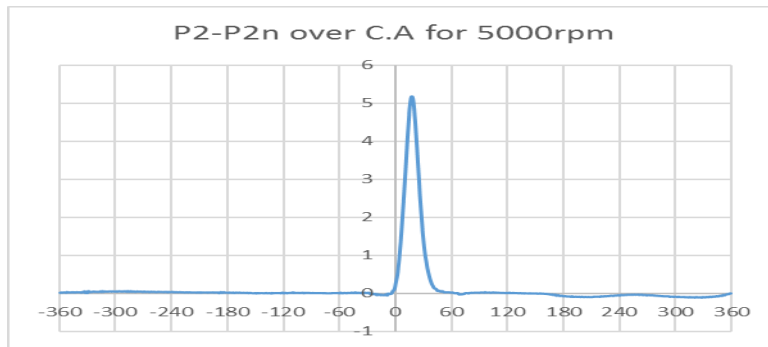


Figure 177 ΔP for 5000rpm

- Engine at 4500rpm

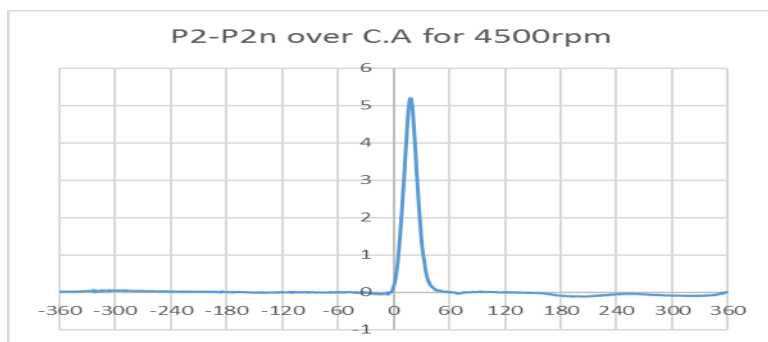


Figure 178 ΔP for 4500rpm

- Engine at 4000rpm

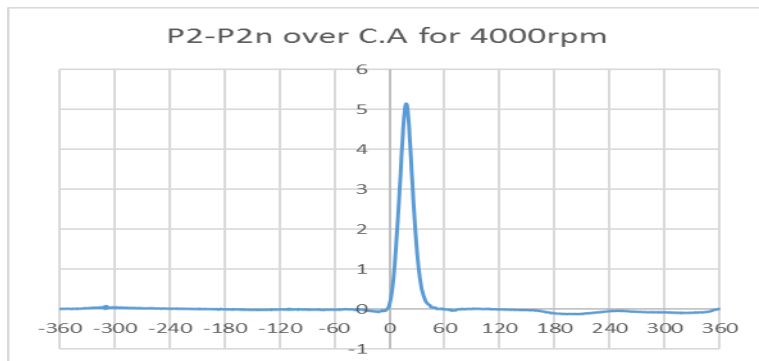


Figure 179 ΔP for 4000rpm

- Engine at 3500rpm

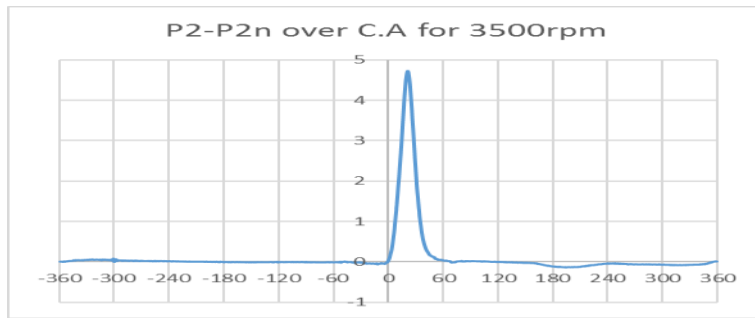


Figure 180 ΔP for 3500rpm

- Engine at 3000rpm

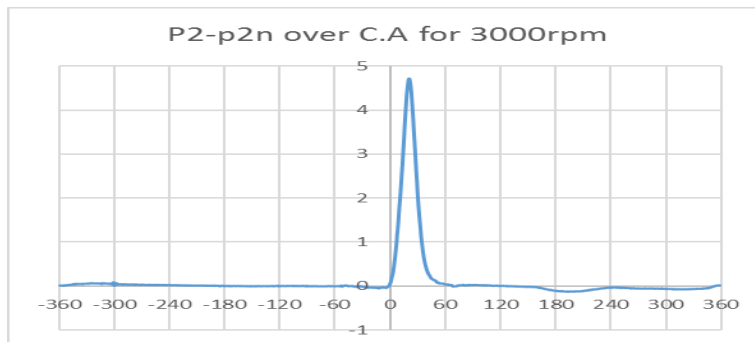


Figure 181 ΔP for 3000rpm

- Engine at 2500rpm

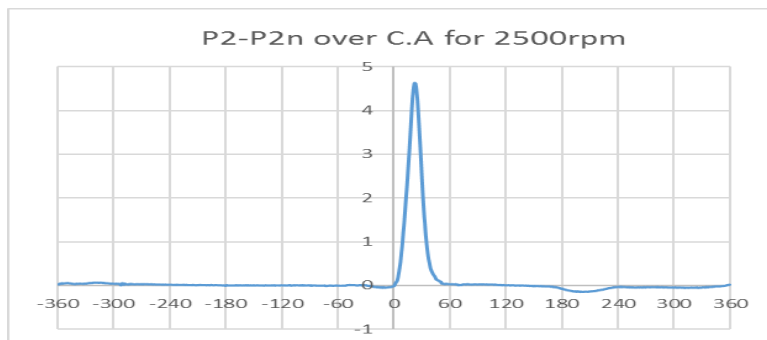


Figure 182 ΔP for 2500rpm

- Engine at 2000rpm

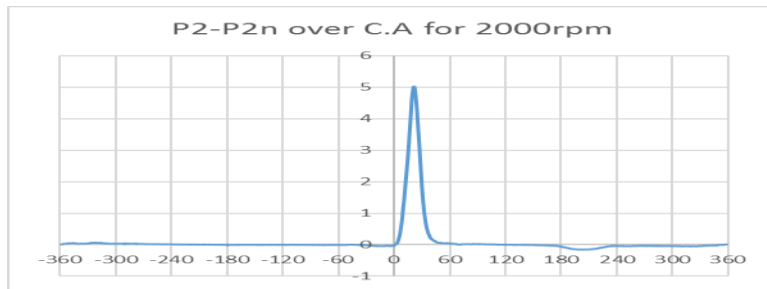


Figure 183 ΔP for 2000rpm

- Engine at 1500rpm

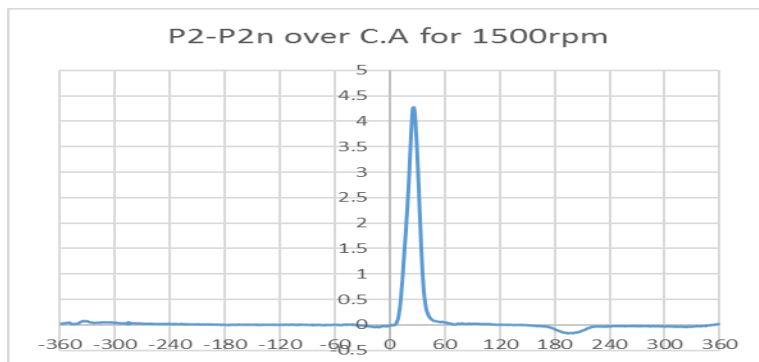


Figure 184 ΔP for 1500rpm

- Engine at 1000rpm

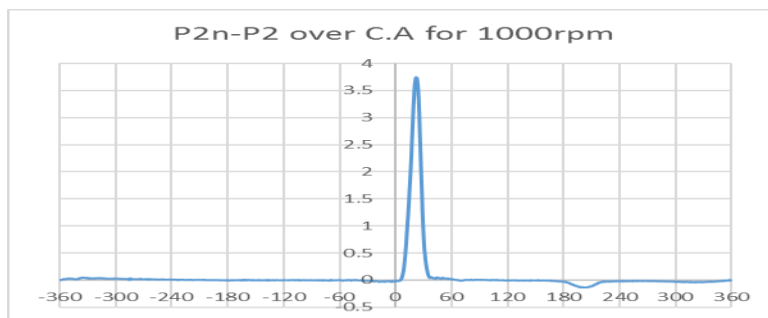


Figure 185 ΔP for 1000rpm

4.4.3. Fox 1.0 Litre

- Engine at 6500rpm

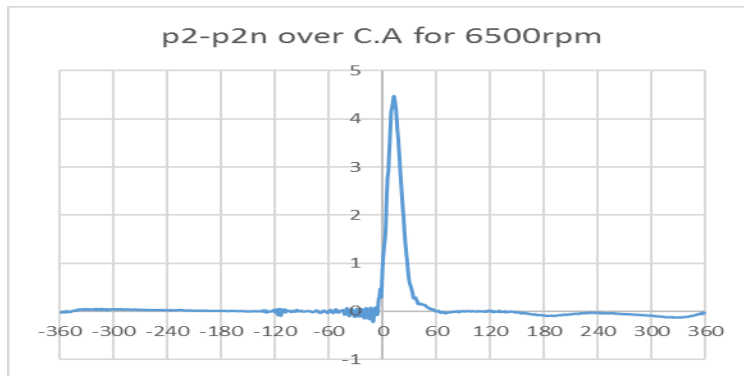


Figure 186 ΔP for 6500rpm

- Engine at 6000rpm

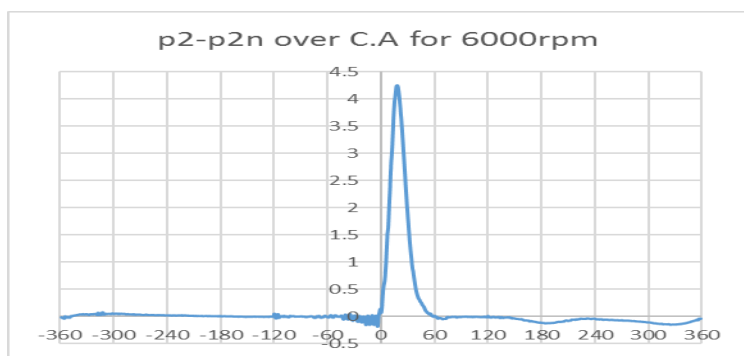


Figure 187 ΔP for 6000rpm

- Engine at 5500rpm

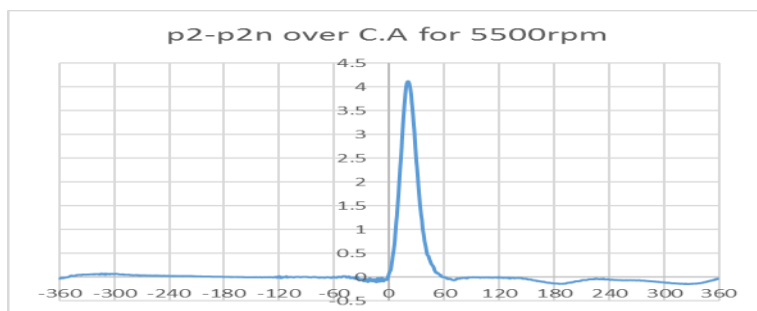


Figure 188 ΔP for 5500rpm

- Engine at 5000rpm

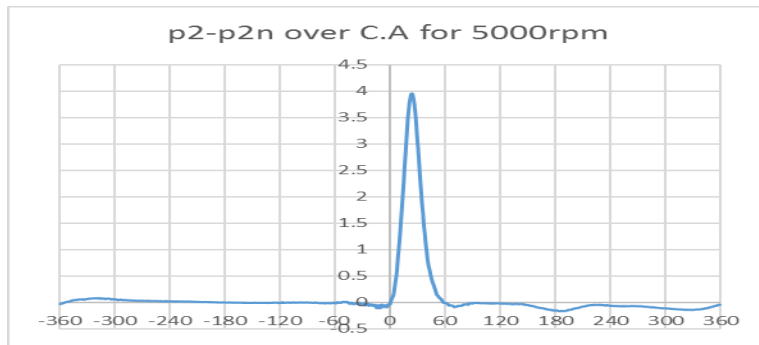


Figure 189 ΔP for 5000rpm

- Engine at 4500rpm

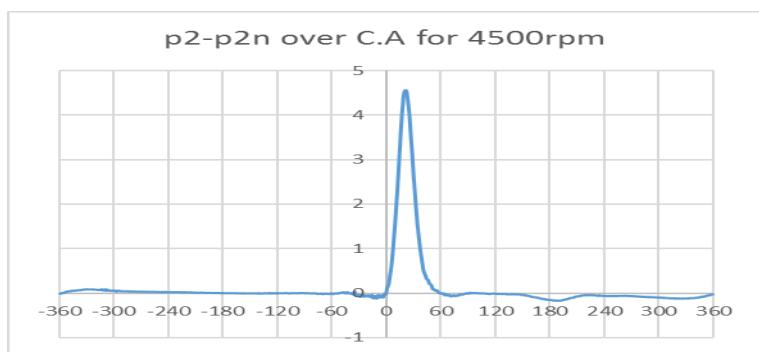


Figure 190 ΔP for 4500rpm

- Engine at 4000rpm

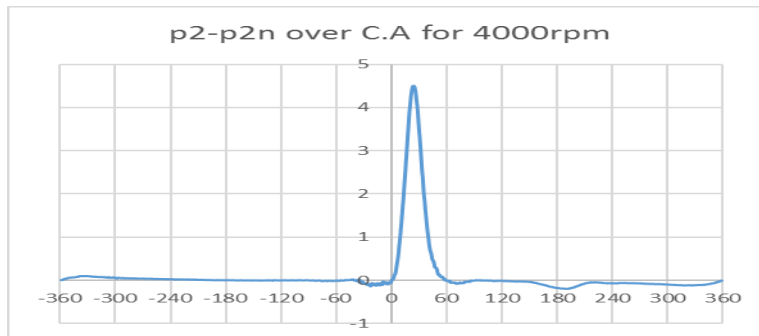


Figure 191 ΔP for 4000rpm

- Engine at 3500rpm

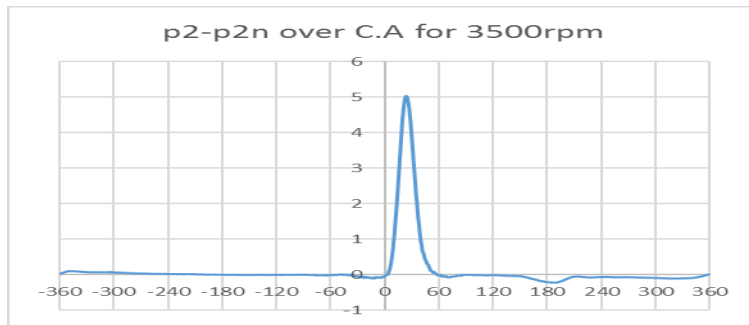


Figure 192 ΔP for 3500rpm

- Engine at 3000rpm

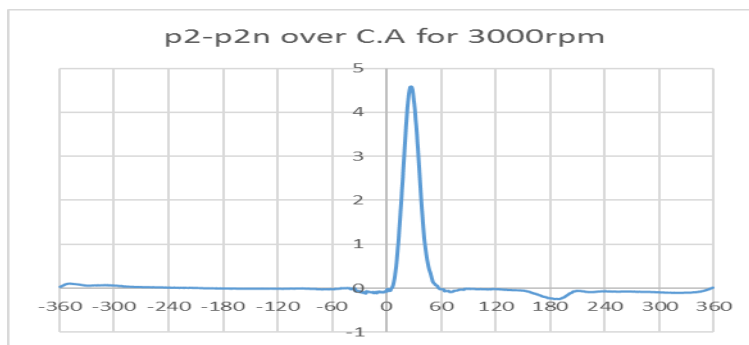


Figure 193 ΔP for 3000rpm

- Engine at 2500rpm

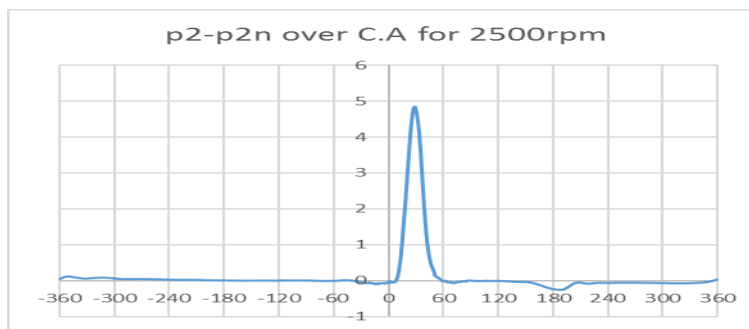


Figure 194 ΔP for 2500rpm

- Engine at 2000rpm

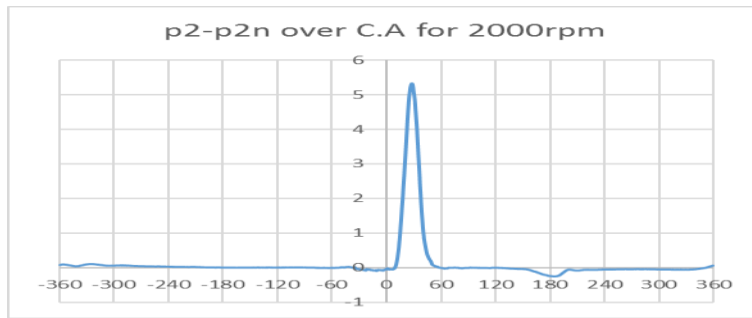


Figure 195 ΔP for 2000rpm

- Engine at 1750rpm

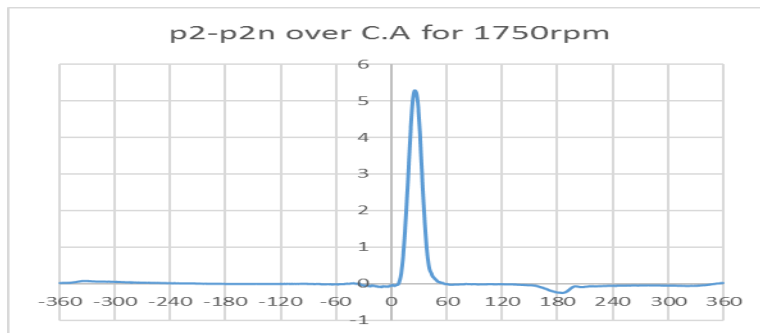


Figure 196 ΔP for 1750rpm

- Engine at 1500rpm

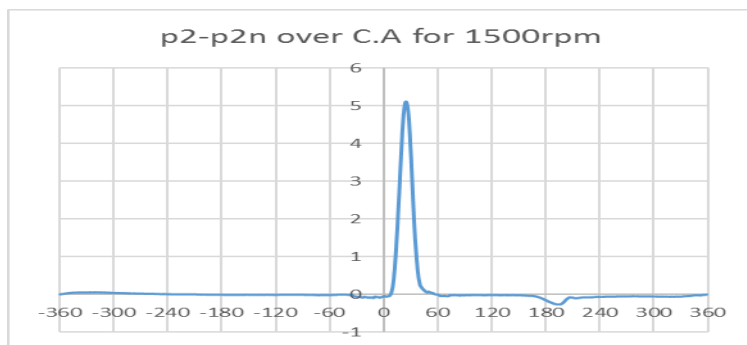


Figure 197 ΔP for 1500rpm

- Engine at 1000rpm

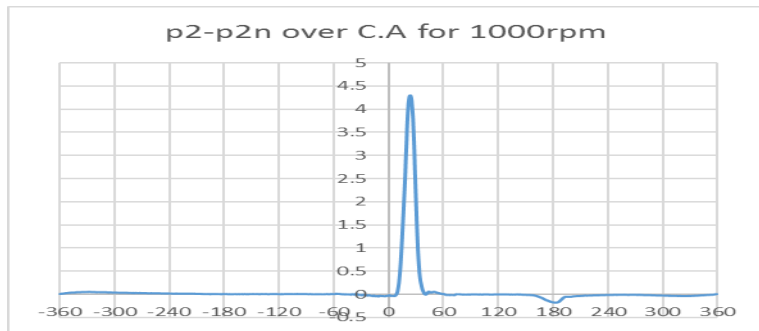


Figure 198 ΔP for 1000rpm

4.4.4. Fox 1.0 Litre Full Load

- Log point 14

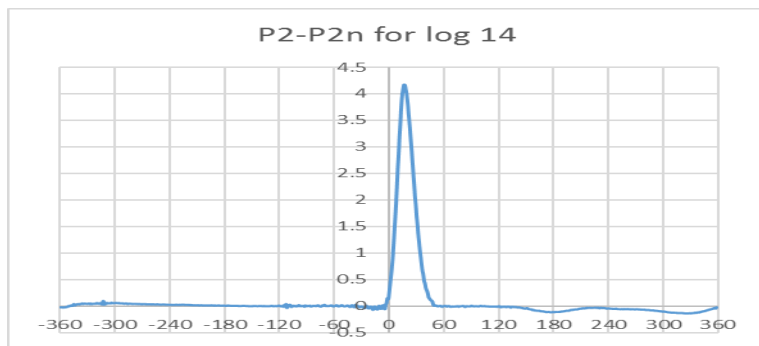


Figure 199 ΔP for log 14

- Log point 10

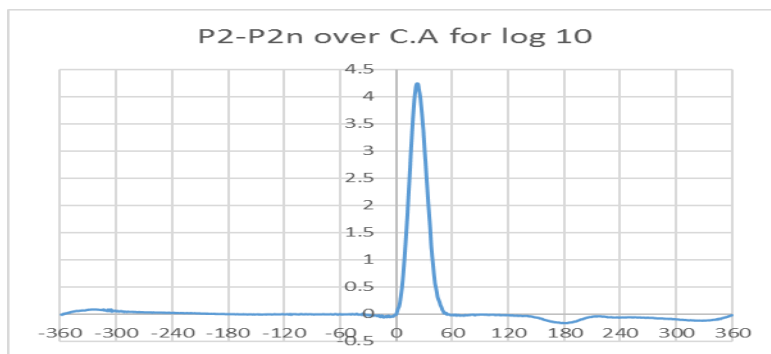


Figure 200 ΔP for log 10

- Log point 7

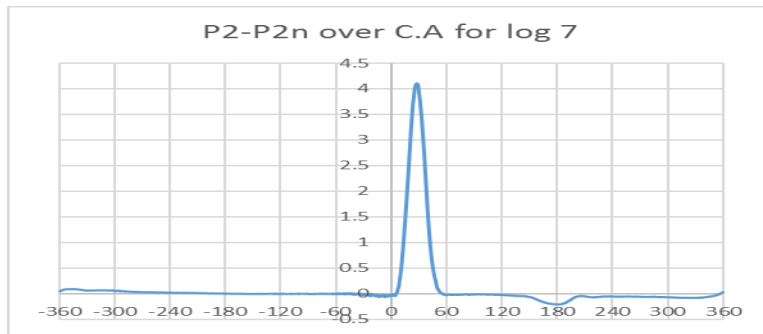


Figure 201 ΔP for log 7

- Log point 5

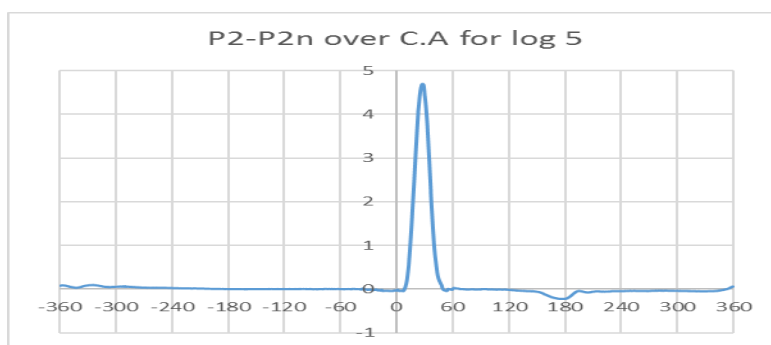


Figure 202 ΔP for log 5

- Log point 1

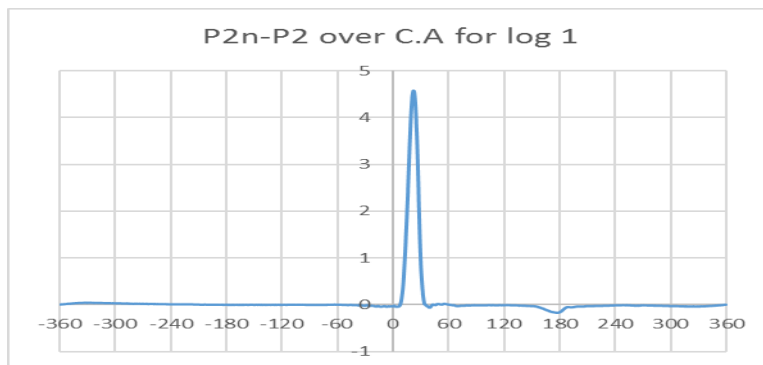


Figure 203 ΔP for log 1

4.4.5. Fox 1.0 Litre Part Load

- Log point 19

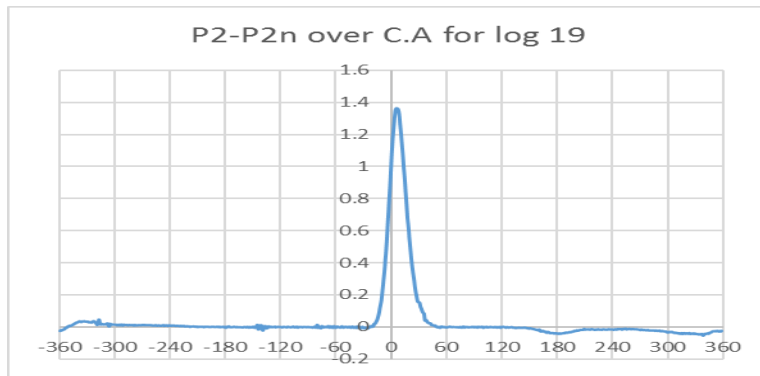


Figure 204 ΔP for log 19

- Log point 18

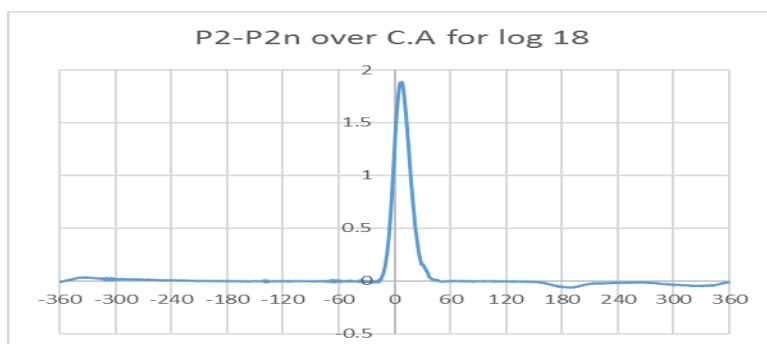


Figure 205 ΔP for log 18

Log point 14

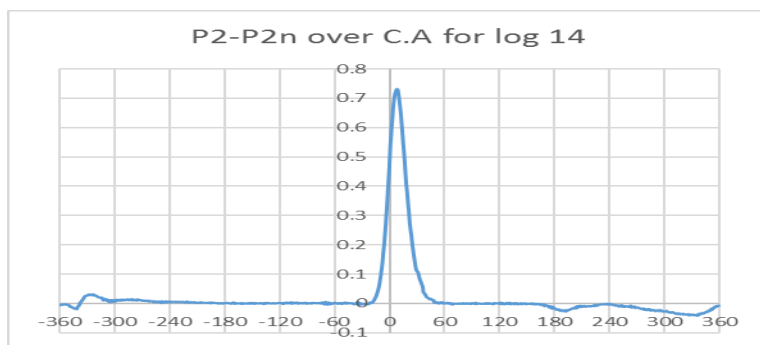


Figure 206 ΔP for log 14

- Log point 9

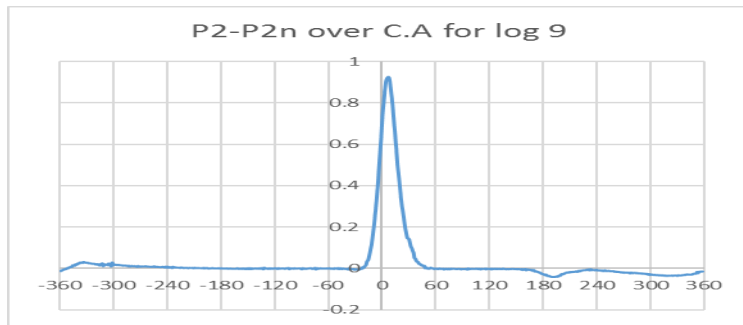


Figure 207 ΔP for log 9

- Log point 4

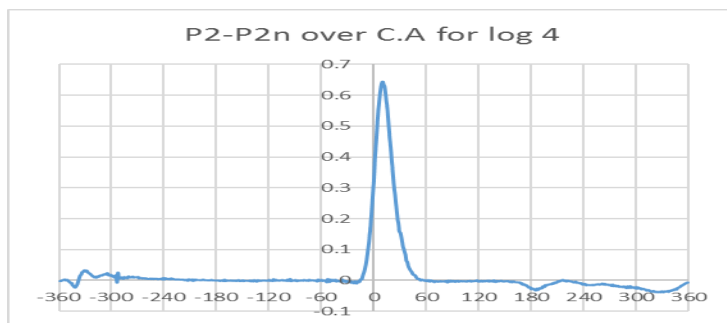


Figure 208 ΔP for log 4

- Log point 3

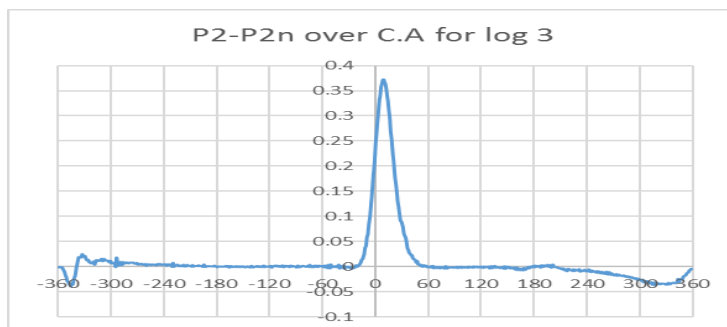


Figure 209 ΔP for log 3

- Log point 1

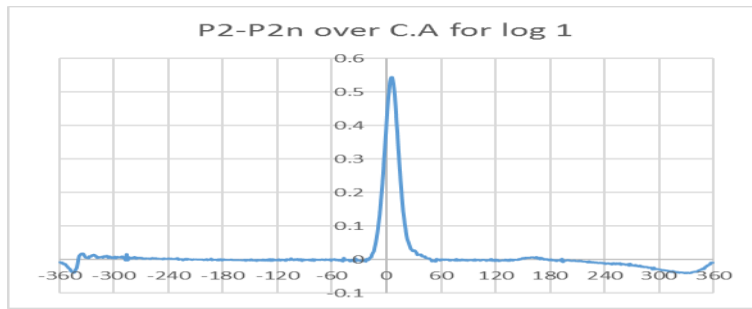


Figure 210 ΔP for log 1

4.5. Mass Fraction Burned

The Mass Fraction Burned (MFB) is used to give a visual representation of how the mixture is being burned. The MFB is given a value between 1 and 0 with 1 being equal to 100% and 0 being 0%. This shows how much of the mixture has been burned.

The time it has taken to burn a specific amount of fuel is measured in crank angle rotation, so the X axis is the crank angle degrees in which the combustion has taken place during.

4.5.1. Sigma 1.6 Litre Naturally Aspirated

- Engine speed at 6500rpm

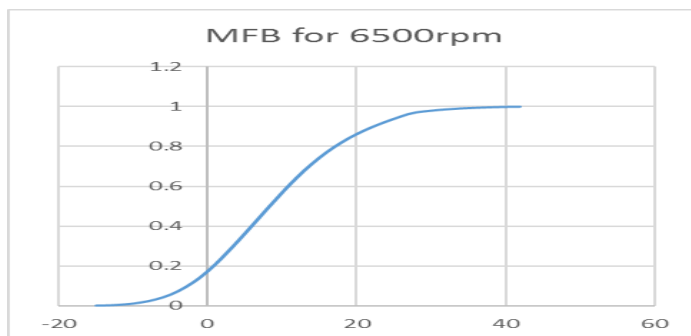


Figure 211 MFB for 6500rpm

- Engine speed at 6000rpm

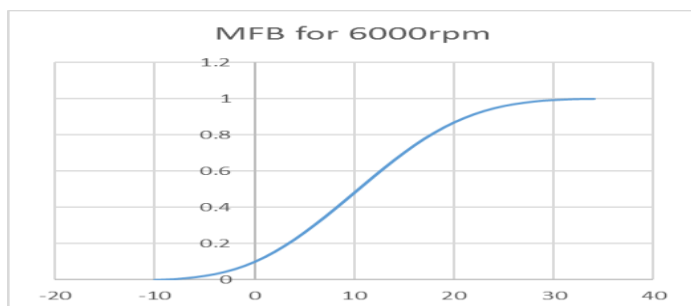


Figure 212 MFB for 6000rpm

- Engine speed at 5500rpm

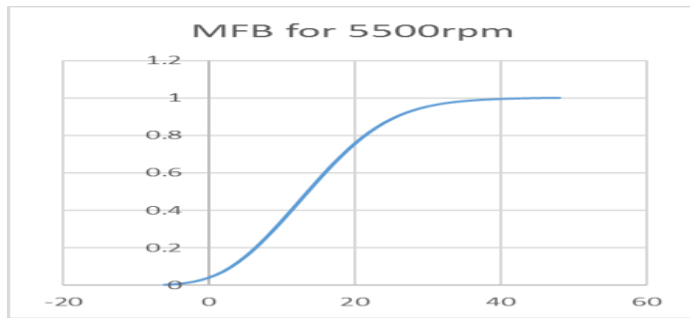


Figure 213 MFB for 5500rpm

- Engine speed at 5000rpm

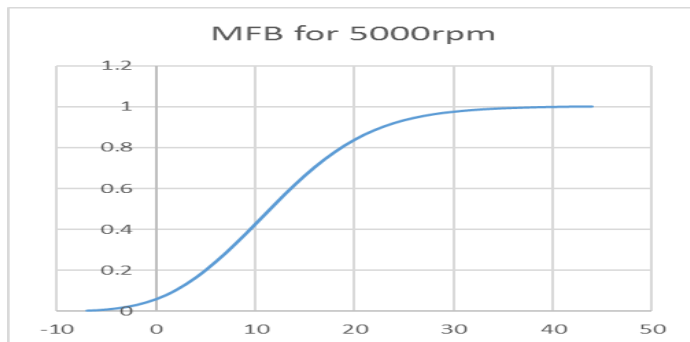


Figure 214 MFB for 5000rpm

- Engine speed at 4500rpm

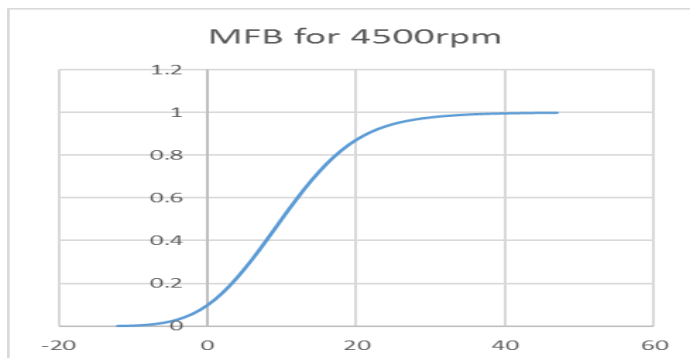


Figure 215 MFB for 4500rpm

- Engine speed at 4250rpm

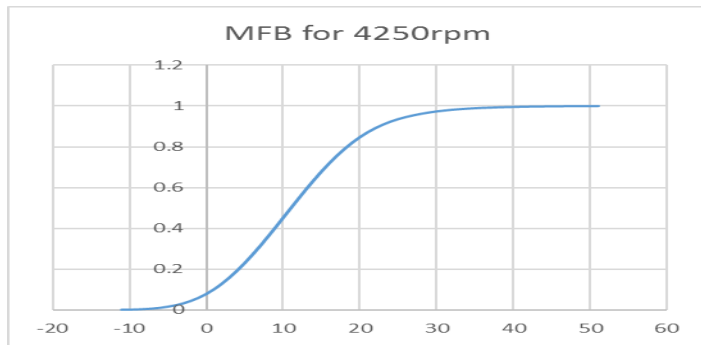


Figure 216 MFB for 4250rpm

- Engine speed at 4000rpm

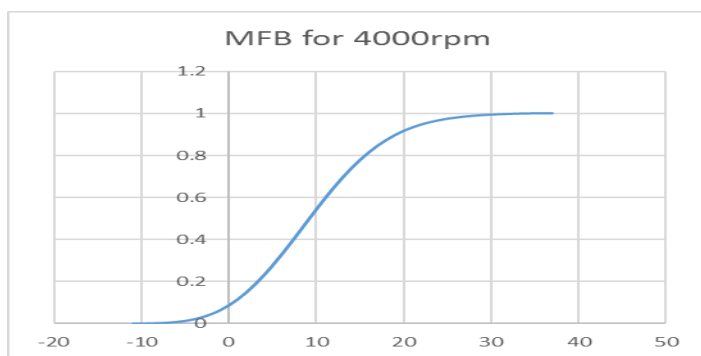


Figure 217 MFB for 4000rpm

- Engine speed at 3500rpm

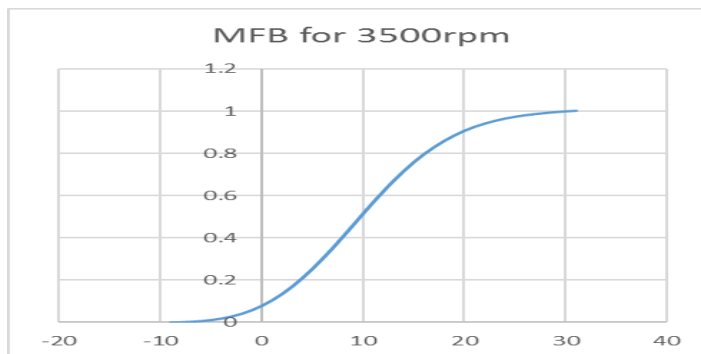


Figure 218 MFB for 3500rpm

- Engine speed at 3000rpm

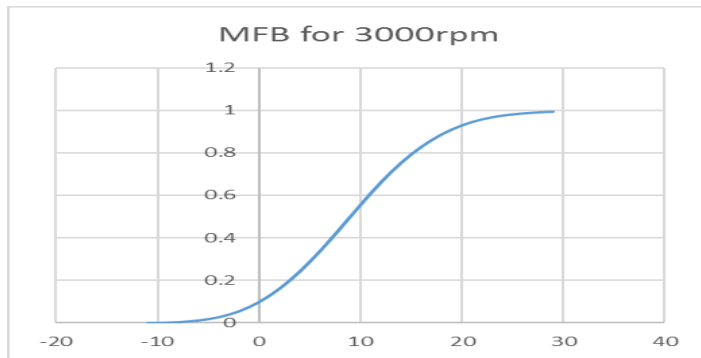


Figure 219 MFB for 3000rpm

- Engine speed at 2500rpm

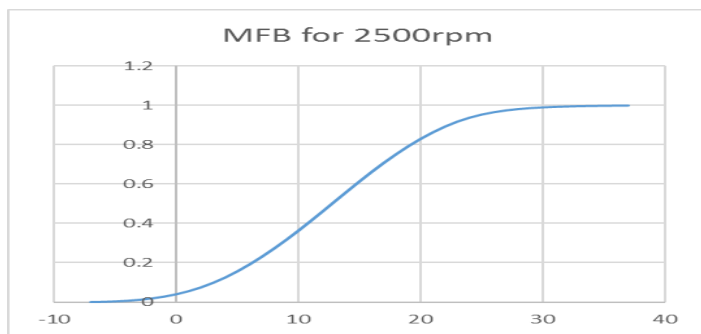


Figure 220 MFB for 2500rpm

- Engine speed at 2000rpm

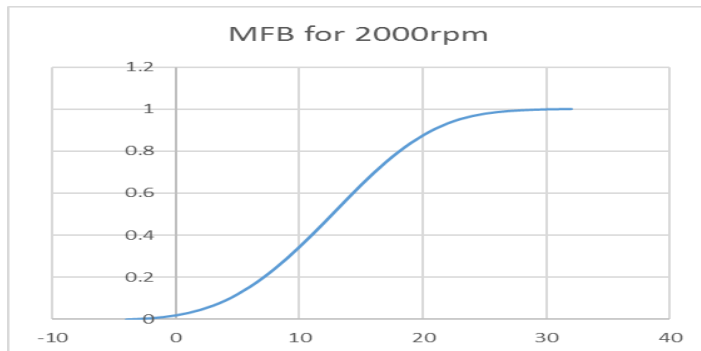


Figure 221 MFB for 2000rpm

- Engine speed at 1750rpm

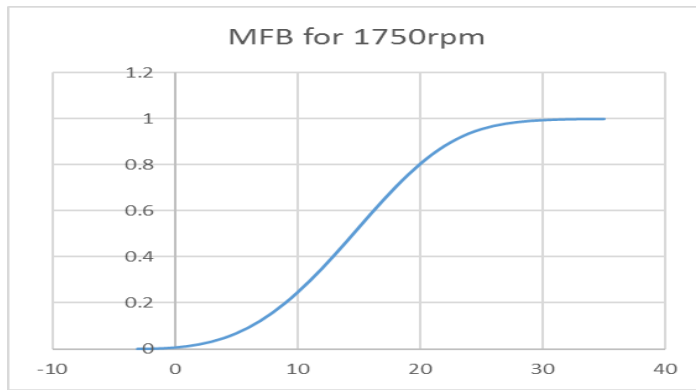


Figure 222 MFB for 1750rpm

- Engine speed at 1500rpm

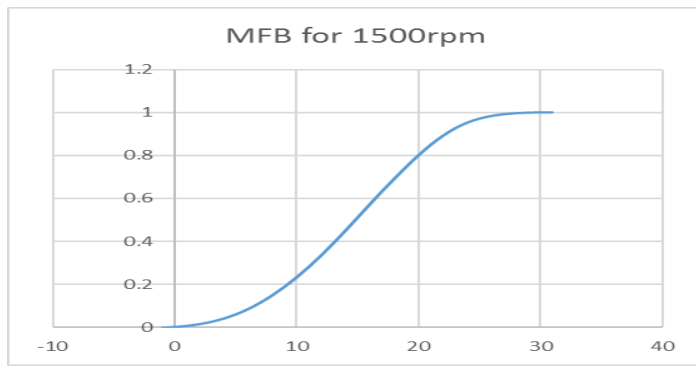


Figure 223 MFB for 1500rpm

- Engine speed at 1000rpm

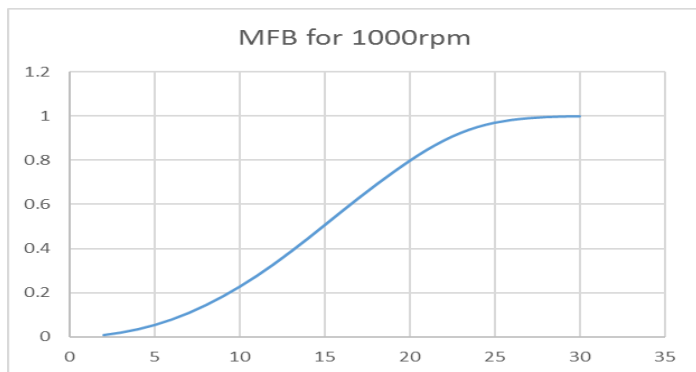


Figure 224 MFB for 1000rpm

4.5.2. 1.6 Litre Turbocharged

- Engine speed at 6000rpm

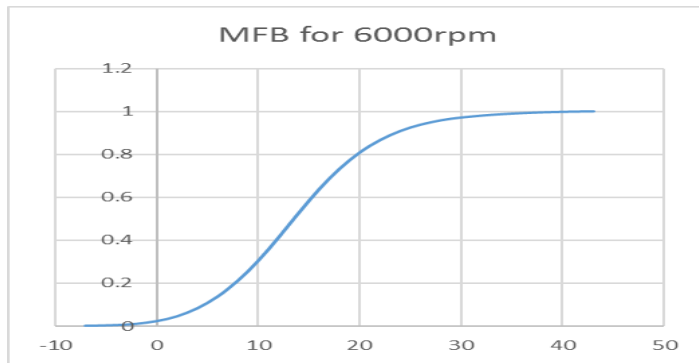


Figure 225 MFB for 6000rpm

- Engine speed at 5700rpm

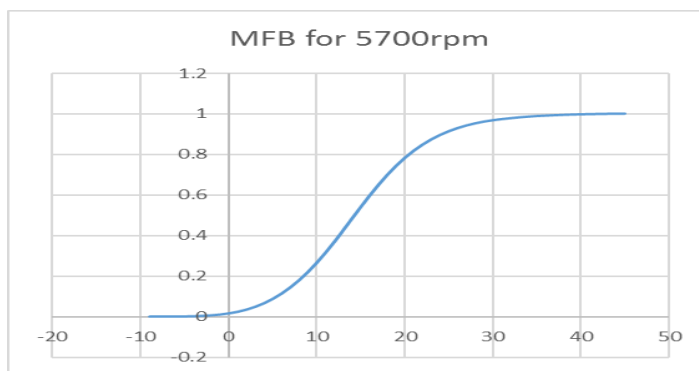


Figure 226 MFB for 5700rpm

- Engine speed at 5500rpm

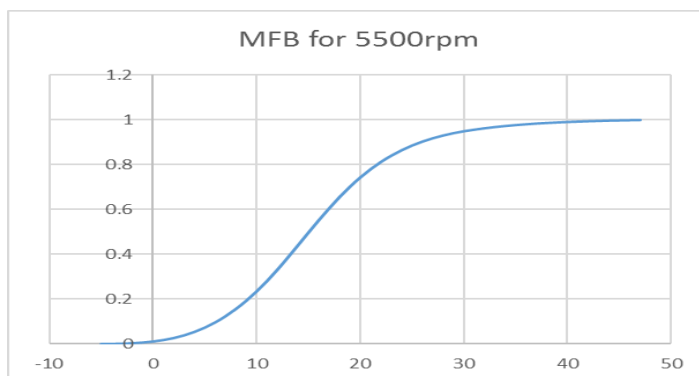


Figure 227 MFB for 5500rpm

- Engine speed at 5000rpm

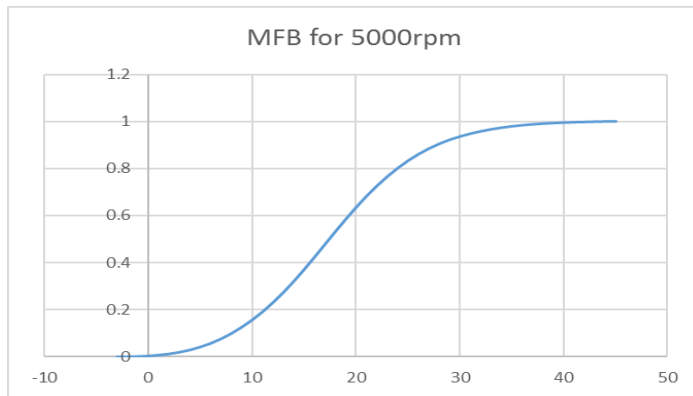


Figure 228 MFB for 5000rpm

- Engine speed at 4500rpm

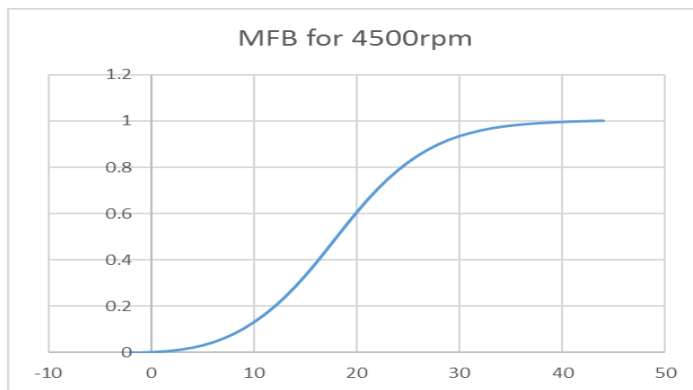


Figure 229 MFB for 4500rpm

- Engine speed at 4000rpm

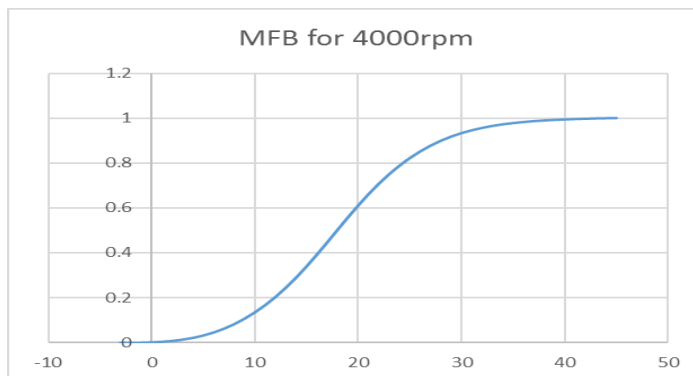


Figure 230 MFB for 4000rpm

- Engine speed at 3500rpm

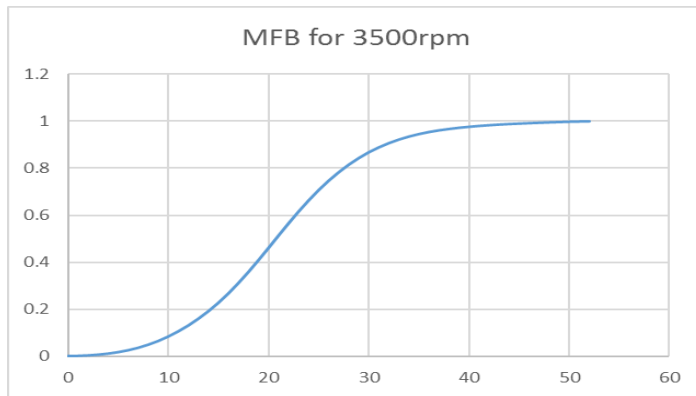


Figure 231 MFB for 3500rpm

- Engine speed at 3000rpm

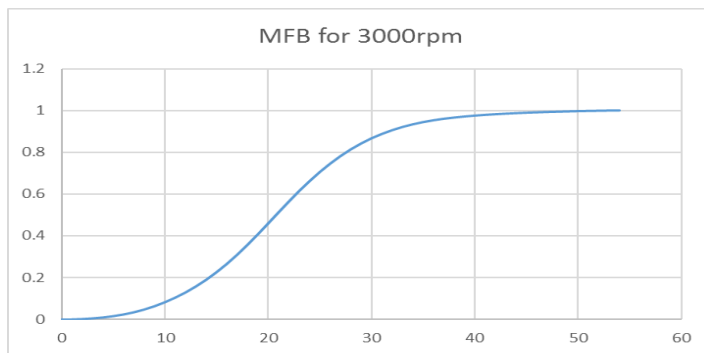


Figure 232 MFB for 3000rpm

- Engine speed at 2500rpm

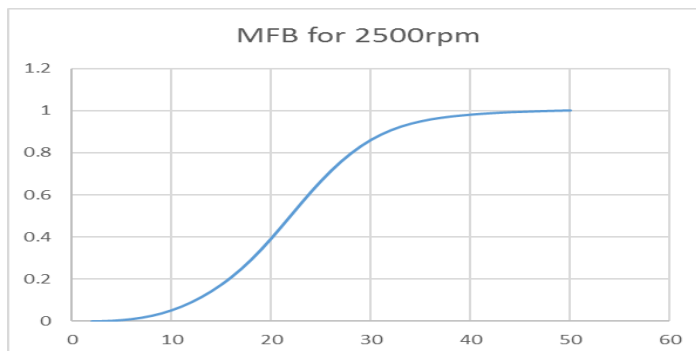


Figure 233 MFB for 2500rpm

- Engine speed at 2000rpm

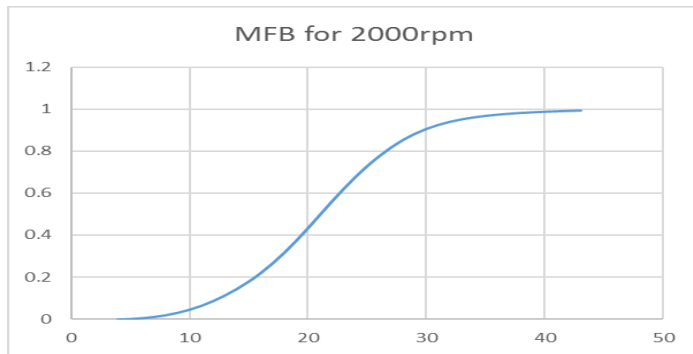


Figure 234 MFB for 2000rpm

- Engine speed at 1500rpm

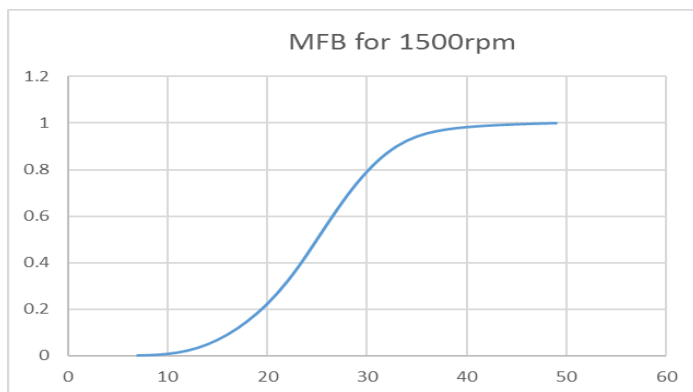


Figure 235 MFB for 1500rpm

- Engine speed at 1000rpm

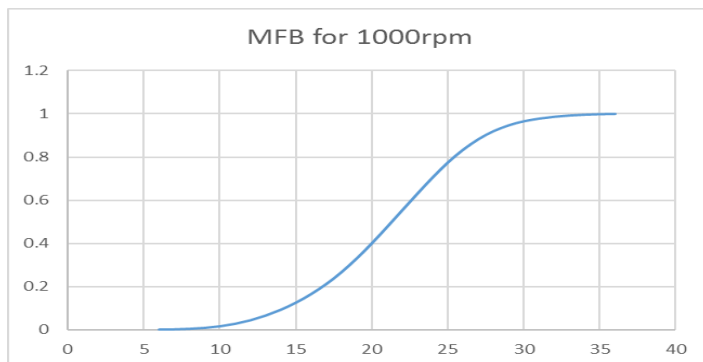


Figure 236 MFB for 1000rpm

4.5.3. Fox 1.0 Litre

- Engine speed at 6500rpm

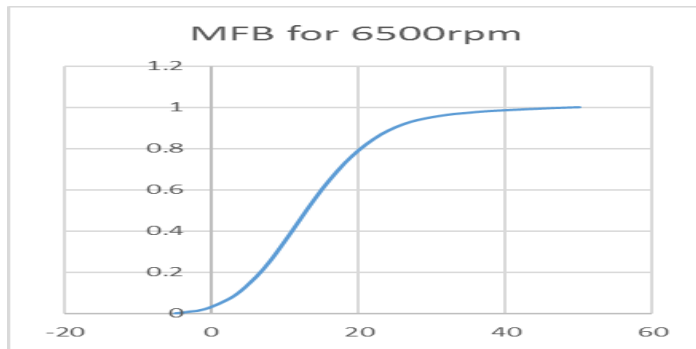


Figure 237 MFB for 6500rpm

- Engine speed at 6000rpm

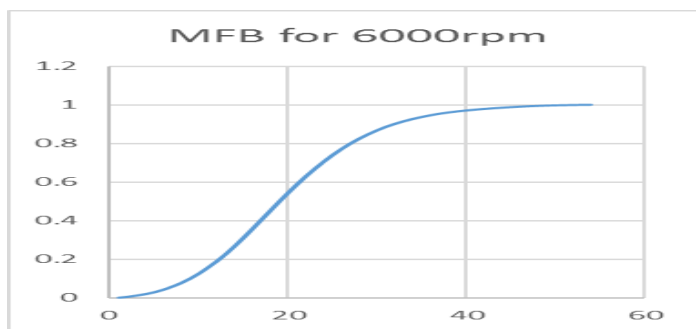


Figure 238 MFB for 6000rpm

- Engine speed at 5500rpm

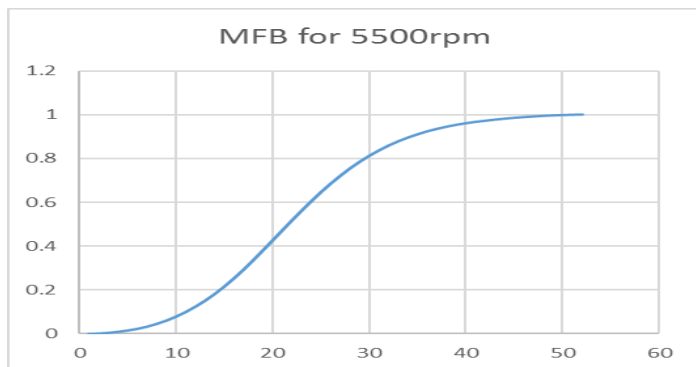


Figure 239 MFB for 5500rpm

- Engine speed at 5000rpm

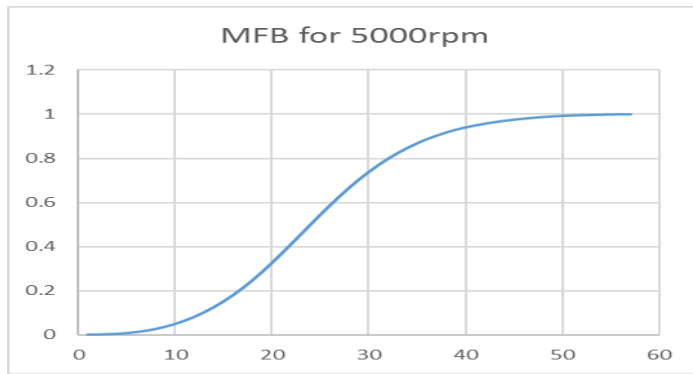


Figure 240 MFB for 5000rpm

- Engine speed at 4500rpm

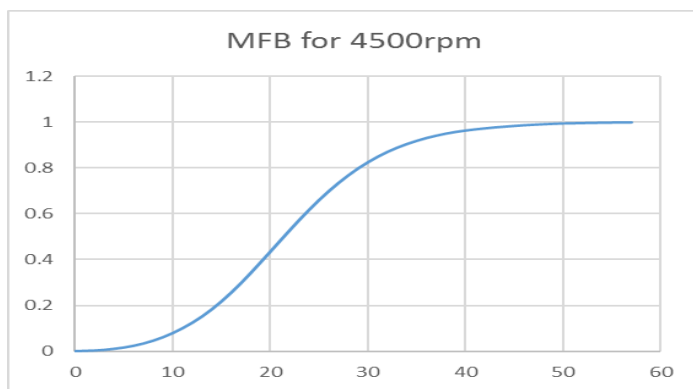


Figure 241 MFB for 4500rpm

- Engine speed at 4000rpm

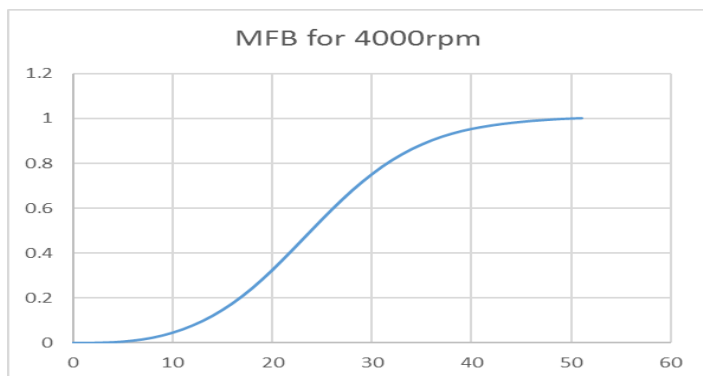


Figure 242 MFB for 4000rpm

- Engine speed at 3500rpm

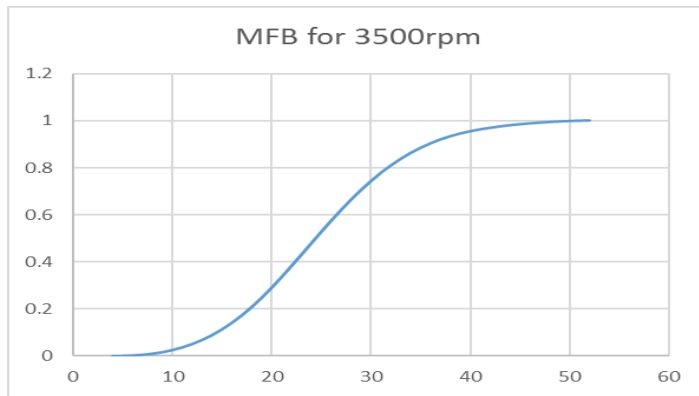


Figure 243 MFB for 3500rpm

- Engine speed at 3000rpm

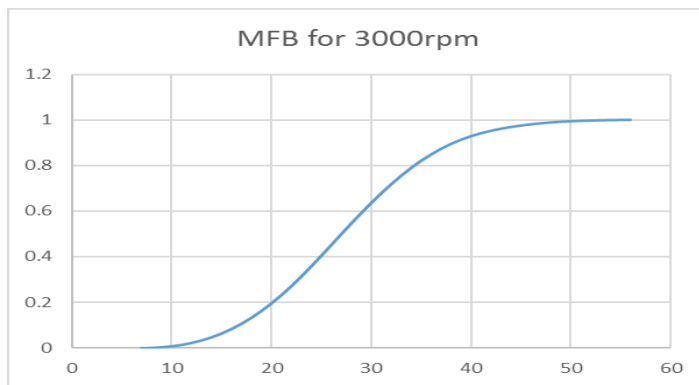


Figure 244 MFB for 3000rpm

- Engine speed at 2500rpm

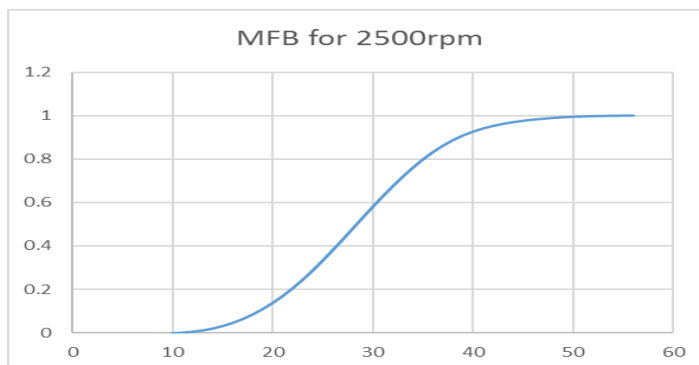


Figure 245 MFB for 2500rpm

- Engine speed at 2000rpm

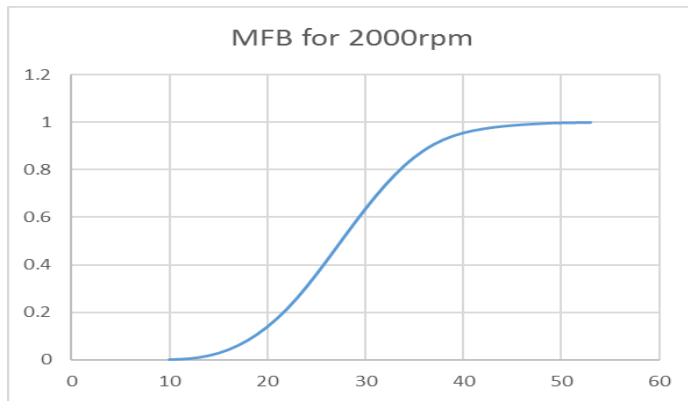


Figure 246 MFB for 2000rpm

- Engine speed at 1750rpm

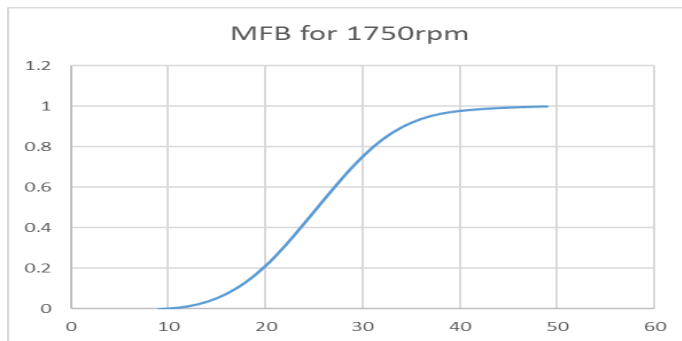


Figure 247 MFB for 1700rpm

- Engine speed at 1500rpm

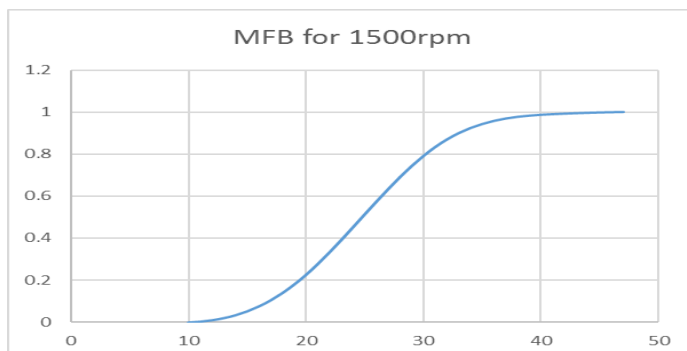


Figure 248 MFB for 1500rpm

- Engine speed at 1000rpm

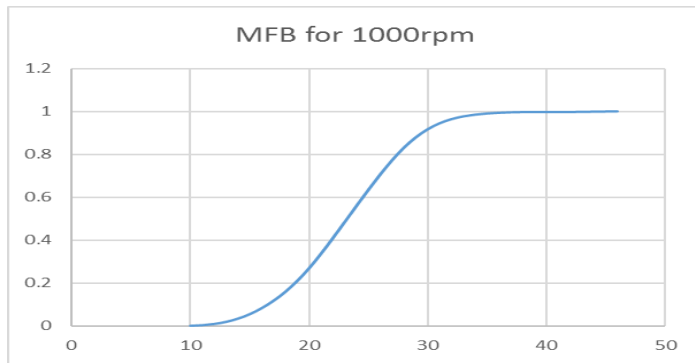


Figure 249 MFB for 1000rpm

4.5.4. Fox 1.0 Litre Full Load

- Log point 14

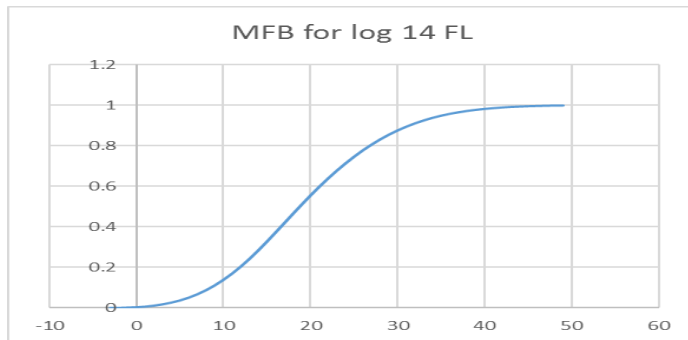


Figure 250 MFB for log 14

- Log point 10

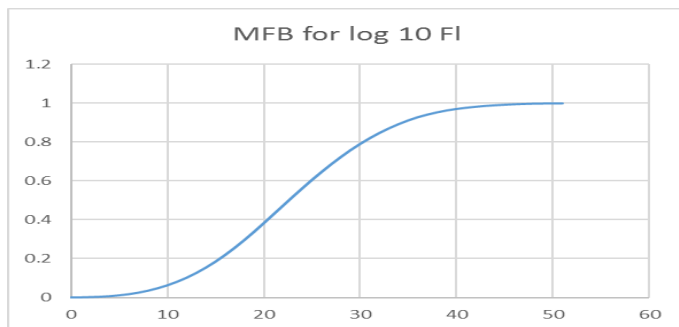


Figure 251 MFB for log 10

- Log point 7

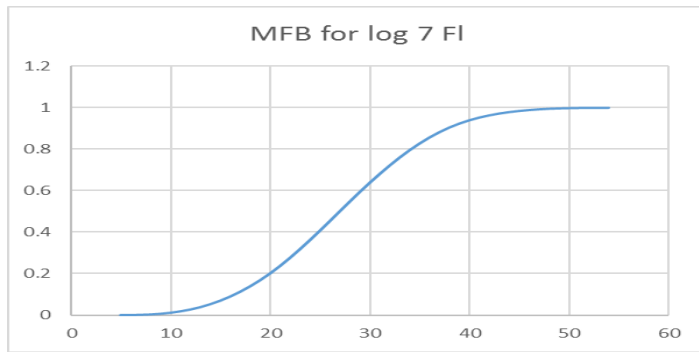


Figure 252 MFB for log 7

- Log point 5

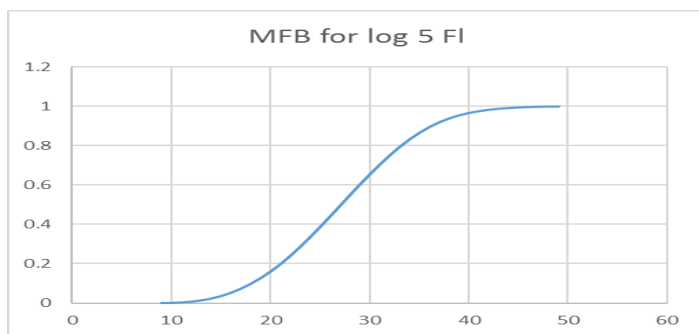


Figure 253 MFB for log 5

- Log point 1

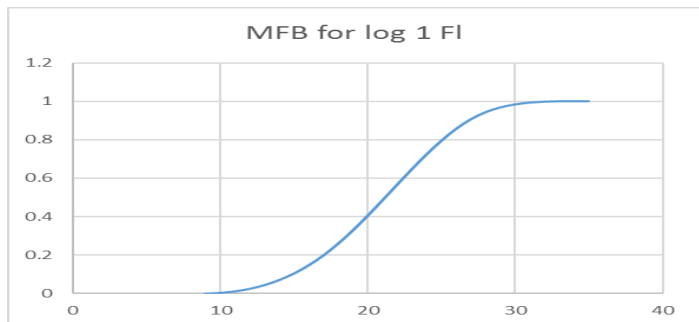


Figure 254 MFB for log 1

4.5.5. Fox 1.0 Litre Part Load

- Log point 19

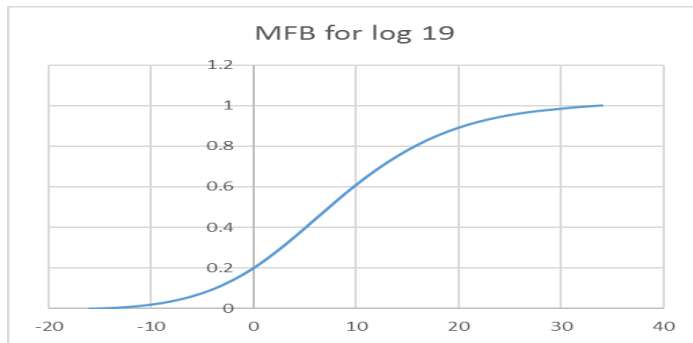


Figure 255 MFB for log 19

- Log point 18

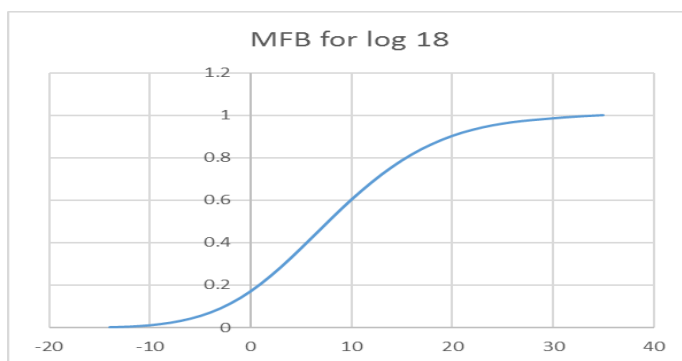


Figure 256 MFB for log 18

- Log point 14

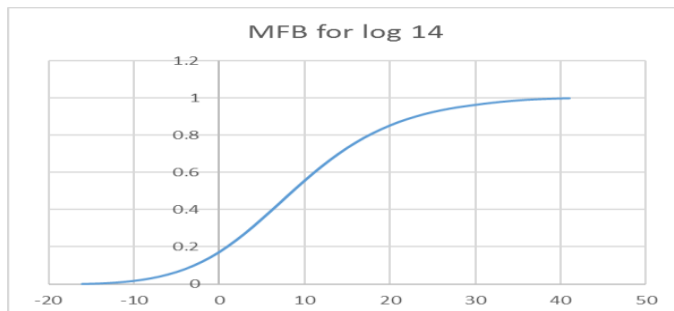


Figure 257 MFB for log 14

- Log point 9

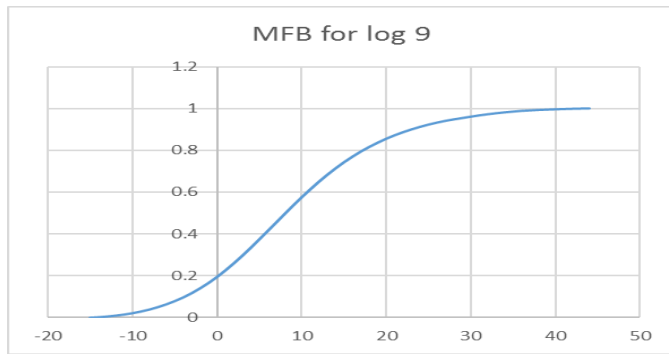


Figure 258 MFB for log 9

- Log point 4

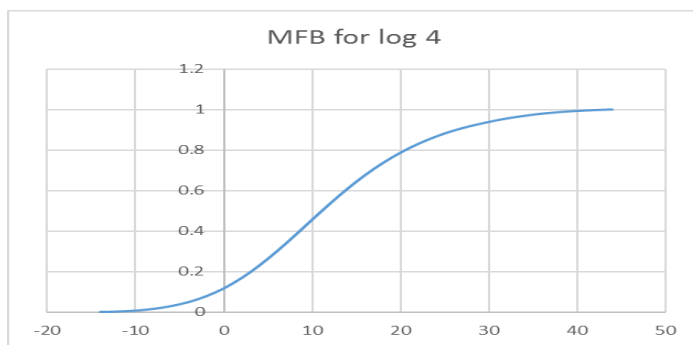


Figure 259 MFB for log 4

- Log point 3

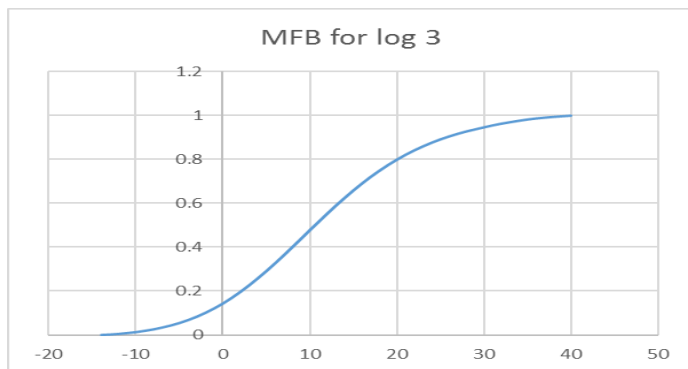


Figure 260 MFB for log 3

- Log point 1

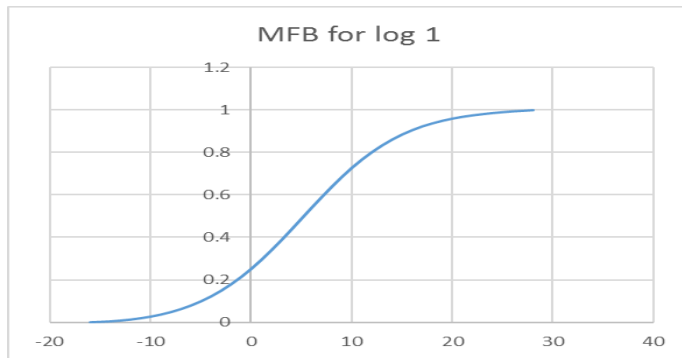


Figure 261 MFB for log 1

5. Analysis of Results

With all the MFB's calculated for each rpm or log point they can now be compared to see how different factors affect the combustion process taking place.

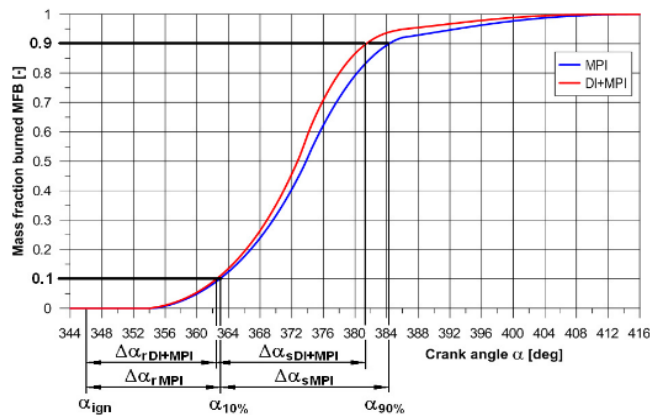


Figure 262 Example of MFB graph (Sendyka and Noga, 2013)

0.1 and 0.9 are commonly used to split the diagram into three stages; 0 – 0.1 is the Slow initial burn zone, 0.1 – 0.9 is the Rapid burn zone and 0.9 – 1 is the Slow late combustion zone. 0.5 is sometimes used to represent when half the mixture has burned and also this is usually when the speed of the flame is at its fastest.

During the initial burn phrase the flame is starting to combust and it starts in a small part of the mixture. As the combustion starts to take place, the flame starts to spread outwards in all directions as it starts to grow and take up a big volume. In the rapid burn zone the flame continues to increase and the flame speed gets faster because of the increase in the flame's perimeter means the flame is encountering more unburned mixture. This continues until around 0.5 when roughly 50% of the mixture has been burned. At this point the gradient of the of the MFB graph will be at its steepest because the combustion is at its fastest. The flame then starts to reach either the cylinder walls meaning and/or the top of the piston crown. From this point on the amount of unburned mixture the flame meets will reduce and the

speed in the which the flame is spreading start to reduce. As the flame continues to spread down the rest of the cylinder it will hit more cylinder wall and piston crown which further reduces the flame area. The flame will continue to reduce until the flame dies due to not being able to burn enough mixture to sustain itself. Just because the flame dies, it does not mean that 100% of the mixture has been burned.

5.1. Comparing MFB's for all RPMs

The MFB's are compared to see what is happening during the combustion process at different rpms and see what effect changing the rpm has. The first three diagrams below start at 1000rpm with each line representing another thousand-rpm increase. The last two diagrams have all the log points included, so each line represents a different log point and because of this the full load has less lines due to there being less log point points recorded.

For the diagrams below the following colour code has been used:

Blue = 1000rpm

Orange = 2000rpm

Yellow = 3000rpm

Green = 4000rpm

Red = 5000rpm

Black = 6000rpm

Purpled = 6500rpm

- NA 1.6

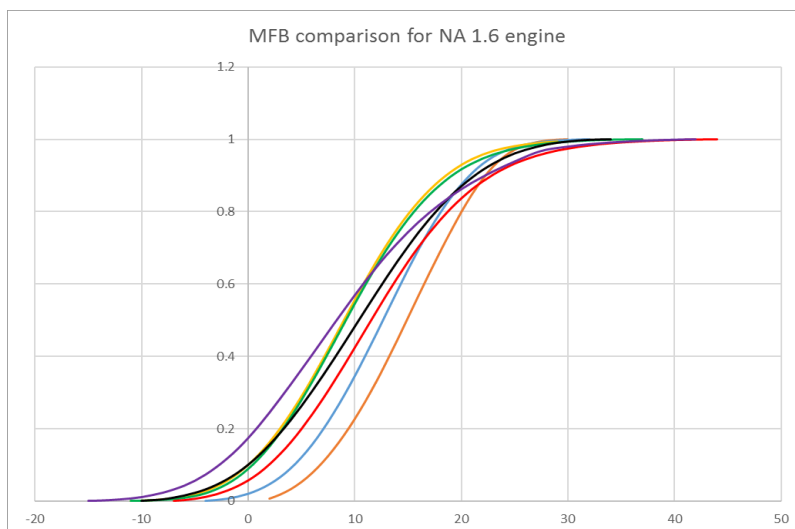


Figure 263 MFB comparison NA 1.6

Looking at the NA engine it can be seen that the gradient of lines are lowest at high rpms and are at their steepest around the middle of the rpm range with 3000-4000 being the

steepest and very similar. The gradient at low RPMs is higher than at high rpms. The start of combustion seems to roughly start earlier, the higher the rpm the engine is at. The black and purple lines are starting the earliest and the blue and orange starting the latest. However, the orange line shows that at 2000rpm combustion starts later than at it does at 1000rpm and for 5000rpm the start is earlier than both 3000 and 4000rpm.

- Turbocharged 1.6

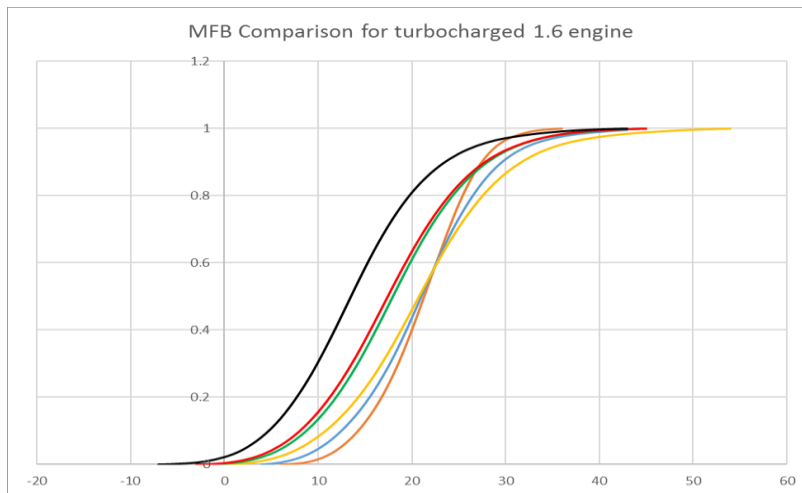


Figure 264 MFB comparison Turbo 1.6

The data supplied only went up to 6000rpm which suggest the turbo can't rev as high as its NA equivalent. Because of this there is no purple line on this diagram.

Most of the lines have a very similar gradient and with 2000 and 3000 being the only rpms with a noticeable difference. 2000rpm has the steepest gradient out of all of them with 3000rpm being the shallowest. As with the NA, the general trend is that as rpm increases the combustion is started earlier with the only outlier being the orange 2000rpm line starting later than the blue 1000rpm line.

- Fox 1.0 turbocharged

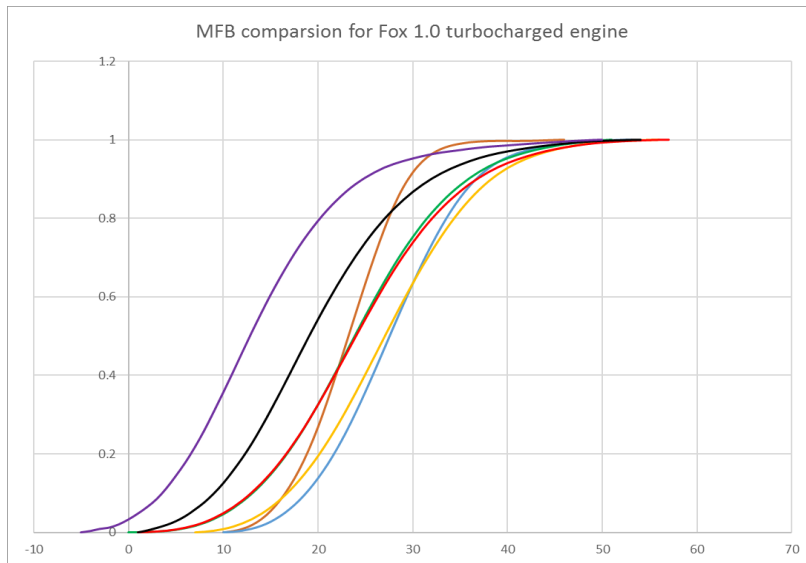


Figure 265 MFB comparison Fox 1.0

The orange 2000rpm line has a much steeper gradient when compared to any of the other lines. The Higher and low end of the rpm range have the steepest gradients with the 5000 and 4000rpm having the lowest gradient.

Again, the start of combustion generally starts later, the lower the rpm.

- Fox Full Load

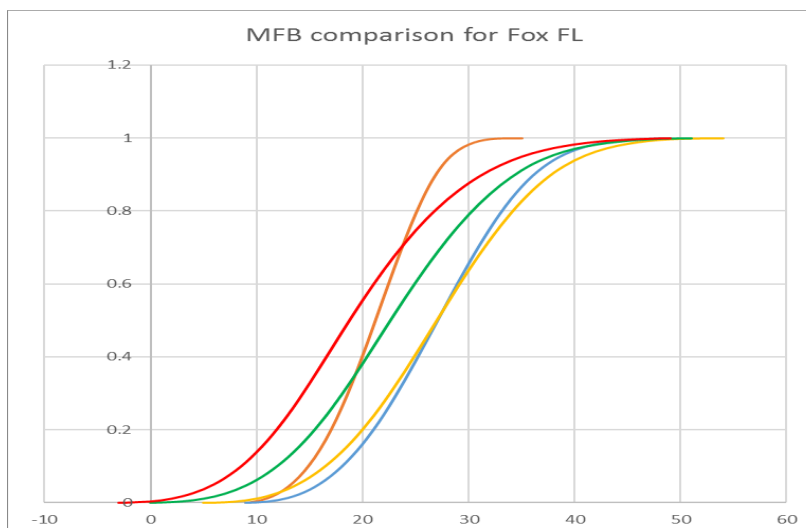


Figure 266 MFB comparison for Fox FL

For the Full load data

- Blue = log 1 (1000rpm)

- Orange = log 5 (2000rpm)
- Yellow = log 7 (3000rpm)
- Green = log 10 (4500rpm)
- Red = log 14 (6000rpm)

2000rpm clearly has the steepest gradient. The first and last log points are the next steepest with the least steep logs points being the log 7 and log 10 which are in the middle of the rpm range.

- Fox Part Load

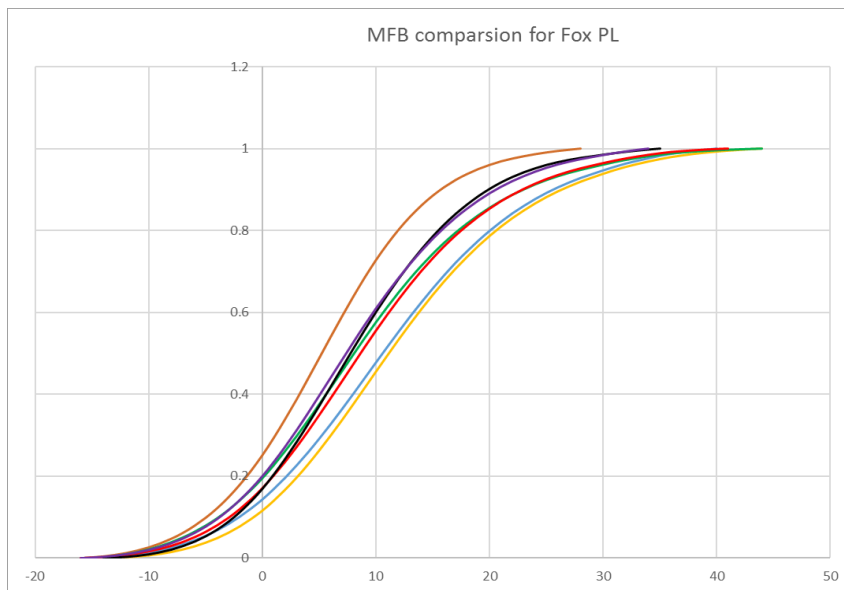


Figure 267 MFB comparison for Fox PL

For the part load

- Blue = log 1 (800rpm)
- Orange = log 3 (1500rpm SL)
- Yellow = log 4 (1500rpm HL)
- Green = log 9 (2000rpm)
- Red = log 14 (2500rpm)
- Black = log 18 (3000rpm)
- Purple = Log 19 (4000rpm)

The Part load data shows that the yet again 2000rpm seems to have the steepest gradient compared with the rest. 6500 and 6000rpm have the next highest gradients while 3000rpm and 1000rpm look the lowest.

The start of combustion looks to be at similar not matter what log point is being considered.

5.2. Comparing naturally aspirated to turbocharged

The introducing of forced induction means that the amount of air being drawn into the cylinder is increased because the turbocharger is essentially an air compressor that forces more air into the cylinders. This is done by increasing the pressure of the air from atmospheric pressure which it is in a NA engine. - (Davis and Perkins-Davis, 2001)

The data starts at 6000rpm because turbo engine only revs to 6000 so even though the NA revs higher, it can't be compared to the turbo engine. There are also some recorded rpms that aren't common across both engines so they can't be used either

Series1 represents the NA engine and is the blue line shown in the graphs below. Series2 represents the Turbocharged engine and is the orange line.

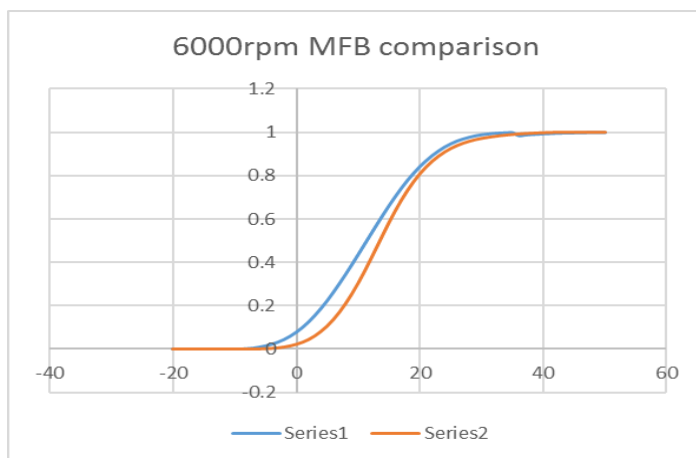


Figure 268 NA/Turbo MFB's at 6000rpm

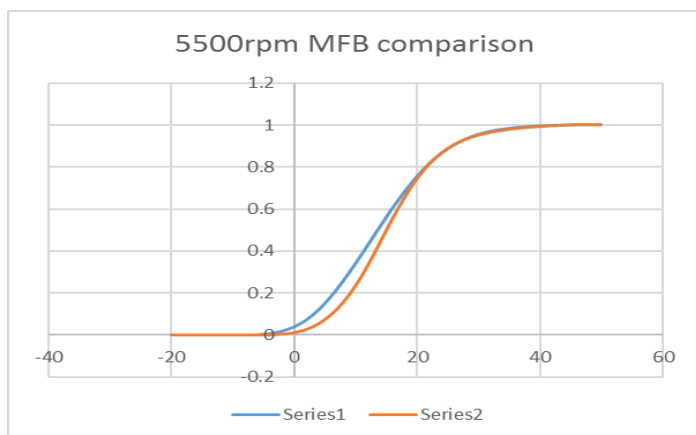


Figure 269 NA/Turbo MFB's at 5500rpm

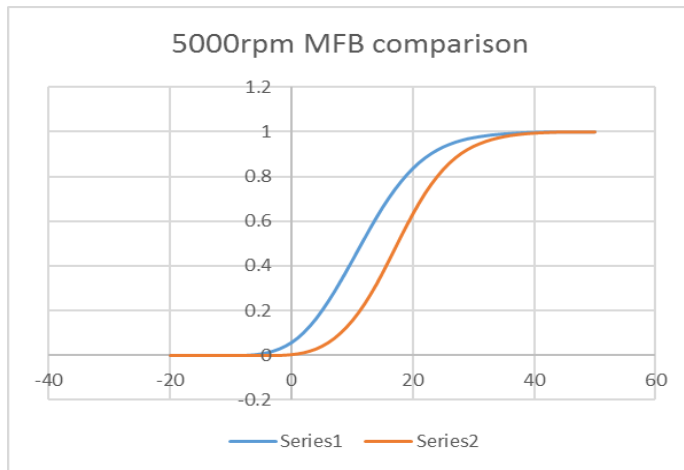


Figure 270 NA/Turbo MFB's at 5000rpm

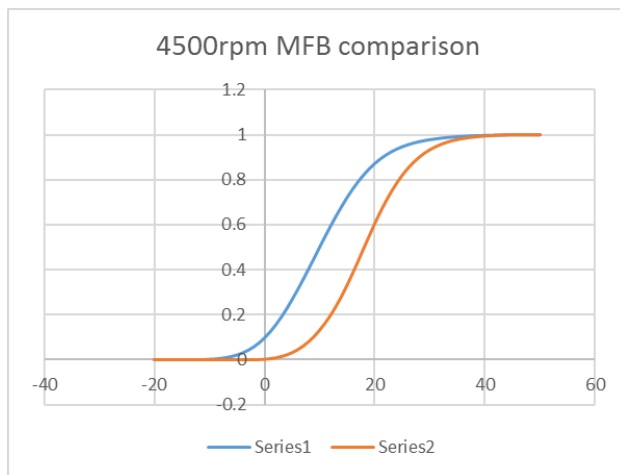


Figure 271 NA/Turbo MFB's at 4500rpm

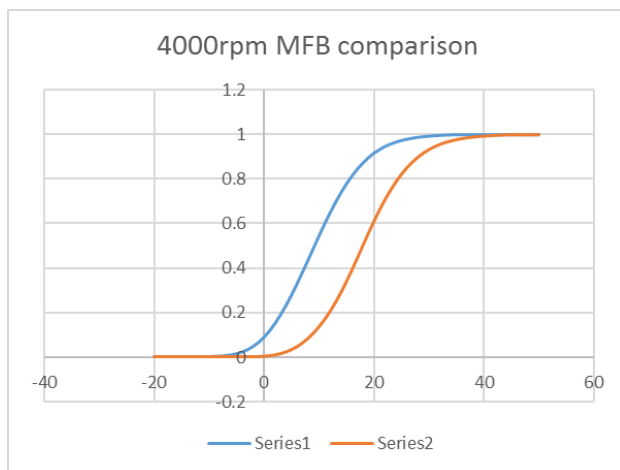


Figure 272 NA/Turbo MFB's at 4000rpm

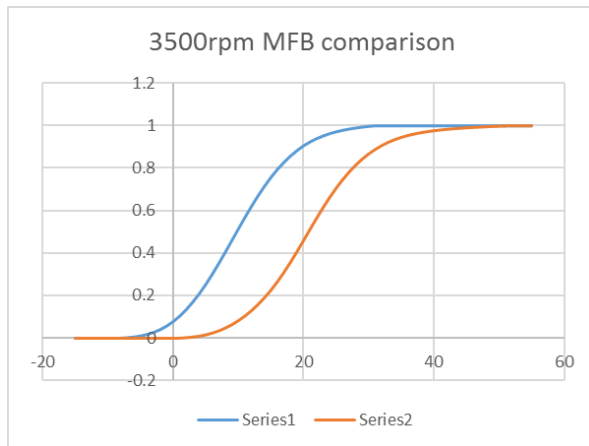


Figure 273 NA/Turbo MFB's at 4000rpm

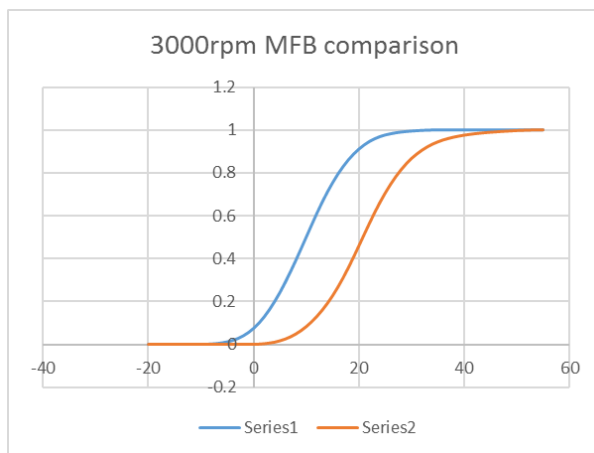


Figure 274 NA/Turbo MFB's at 3000rpm

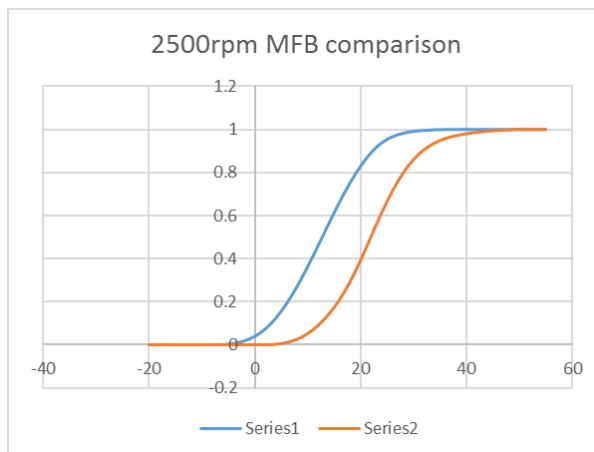


Figure 275 NA/Turbo MFB's at 2500rpm

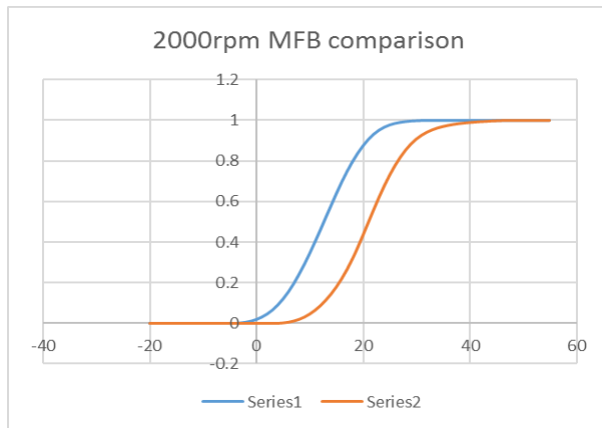


Figure 276 NA/Turbo MFB's at 2000rpm

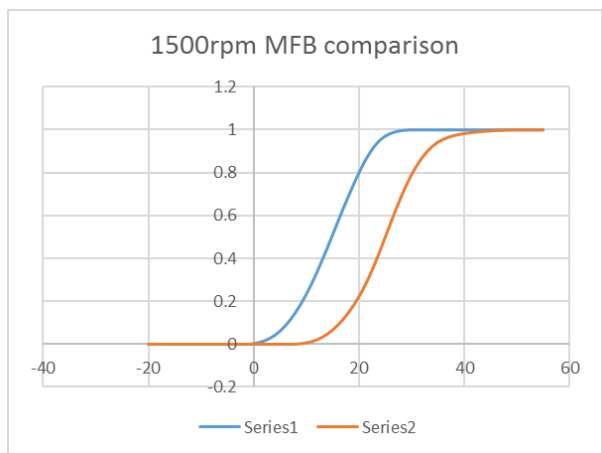


Figure 277 NA/Turbo MFB's at 1500rpm

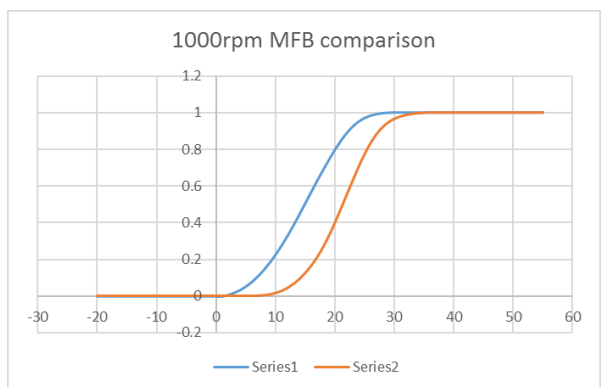


Figure 278 NA/Turbo MFB's at 1000rpm

Overall the NA and turbo engines have a similar gradient for the most part until around 5500rpm, where after that the turbo starts to have a steeper gradient than the NA. this means at high rpms the turbo is burning quicker than the NA and the flame speed is quicker. The turbo engine always starts combusting later than the NA but the turbo ends later at low rpms except 1000rpm. For

1000rpm and 4500rpm+ the end of combustion starts to get closer to the NA's end of combustion. This shows that the duration of combustion at these rpm's is less for the turbo engine than the NA.

From these results, the turbo engine has a faster flame speed and a quicker combustion than the NA at 1000rpm and higher rpm but between 1500 to 4000rpm the NA and turbo engine have very similar combustion characteristics.

5.3. Comparing different engine capacities

For these graphs the aim to compare how different engine capacities effect the combustion. The NA graph has been included to show how similar or different these engines might be a an NA engine.

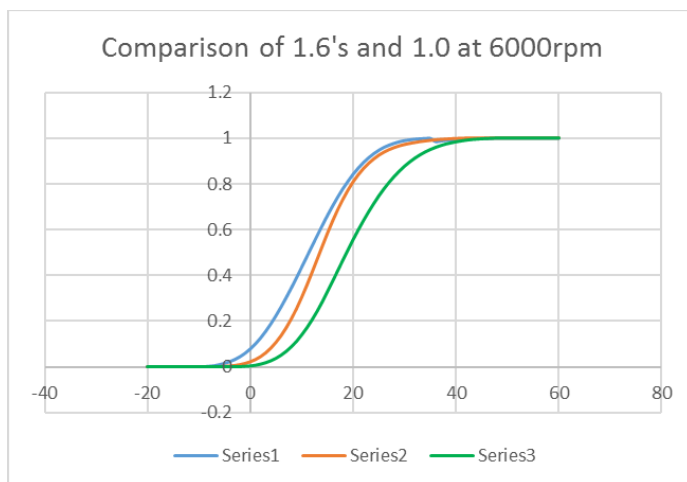


Figure 279 1.6 and 1.0 MFB comparison for 6000rpm

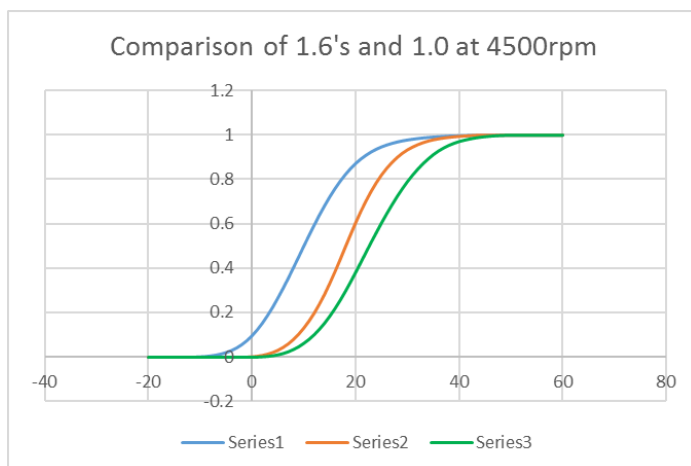


Figure 280 1.6 and 1.0 MFB comparison for 4500rpm

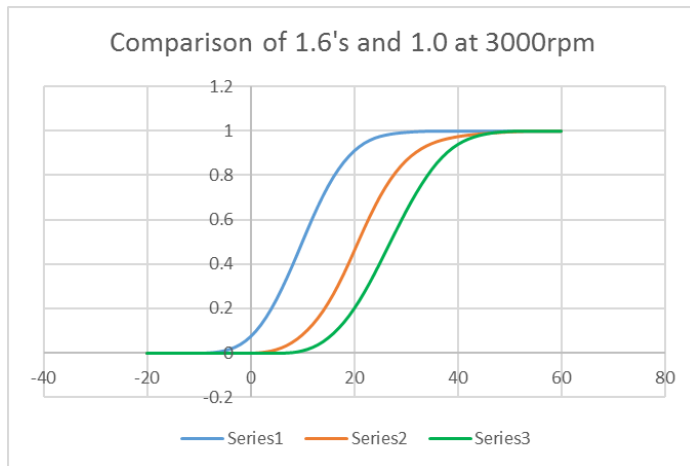


Figure 281 1.6 and 1.0 MFB comparison for 3000rpm

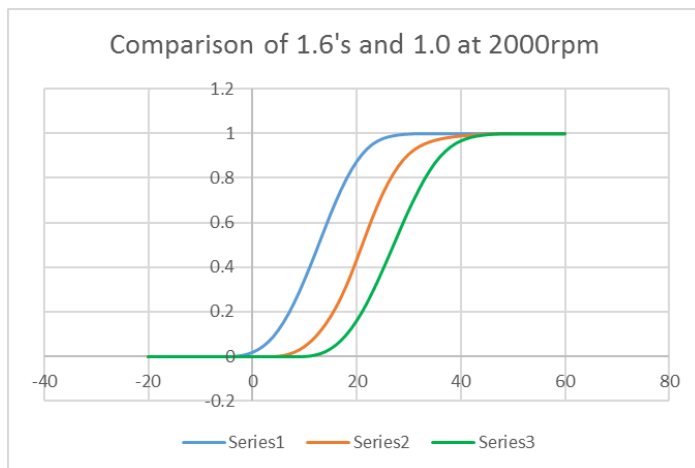


Figure 282 1.6 and 1.0 MFB comparison for 2000rpm

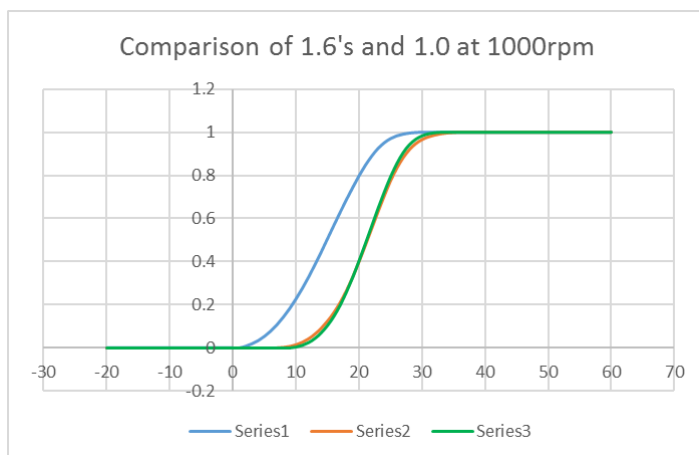


Figure 283 1.6 and 1.0 MFB comparison for 1000rpm

When comparing the different size engines, the 1.6 and 1.0 turbocharged engines are the main focus but the NA has been included just as a reference to see which of the two turbo engines is most like an NA engine. At 1000rpm the size seems to have little to no affect with both turbo engines having

very little differences. Once the rpm starts the increase however the 1.6 starts to have a steeper gradient than the 1.0 In terms of gradient the 1.0 seems to look like the NA 1.6 especially at the higher rpms, it just starts combustion later and ends slightly later too. Both the 1.0 and NA seem to start combustion quicker but stays at a more stable speed where the 1.6 turbo seems to be slow at the start of combustion but starts to burn quicker during the rapid burn zone.

5.4. Comparing 1.0 at Full load to Part Load

With having the data for the same engine just for one test the engine was at full load and for the second set of data the amount/percentage of load was changed. When the MFB graphs for the FL and PL data for the same rpm are compared, it shows how the load of the engine affects combustion. The only time the two sets of data were recorded at the same rpm was at 3000rpm (Log point 7 for FL and Log point 18 for PL) and 2000rpm (Log 5 for FL and log 9 for PL).

Blue = Full Load

Orange = Part Load

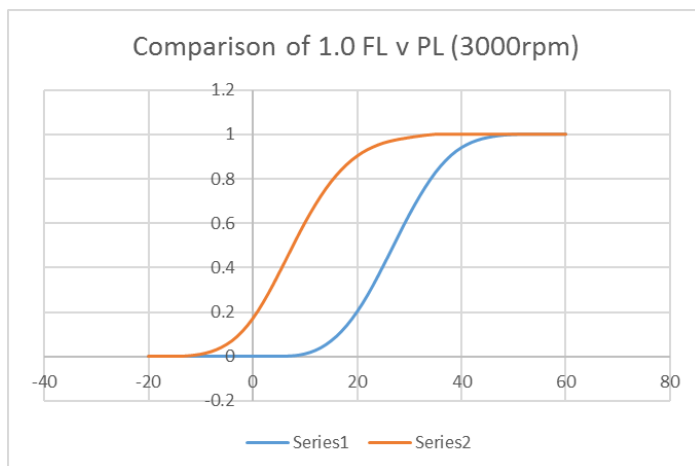


Figure 284 PL and FL MFB for 3000rpm

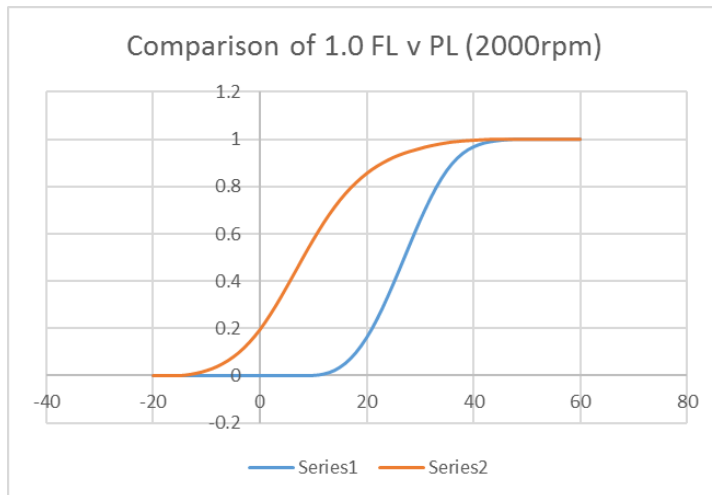


Figure 285 PL and FL MFB for 2000rpm

Because there are only two rpms that appear on both sets of data, the opportunities for comparison are very limited. Using the two diagrams, when the engine is under part load the combustion is starting a lot earlier. The Full load starts later which could be because of it having a much faster flame speed and a much shorter combustion duration.

6. Conclusion

From this study a good understanding of how to measure and understand different characteristics that effect and occur during combustion, has been achieved.

6.1. How RPM affects combustion

With the results produced, it can be seen that the NA has the fastest combustion near the middle of its rpm range and that high and low rpms seem have a slow combustion. This is to be excepted because the faster combustion is, the higher pressure that occurs in the cylinder and this is when maximum torque is being produced. The start of combustion is earlier as the rpms increase because the speed of each cycle is increasing so the actual time between each crank angle gets smaller. To make sure the ignition is combustion at the right point in time compared to the piston, it needs to be earlier to allow for enough time for the combustion to be at the ideal point even though the piston speed is increased.

The 1.6 turbo engine had similar characteristics to the NA engine, whilst the fox engine did not. The Fox engine was the opposite to both the 1.6 engines with the fastest combustion happening at the highest and lowest rpms. 2000rpm was constantly a lot faster than any other RPM so it could be possible that some error has occurred whilst either recording the data or carrying out the calculations to get to the MFB. The rest of the rpm data would suggest that the fox engine could possibly be starting combustion either slightly later to avoid the pressure reaching a point where they could reach a knock or hardware limit.

6.2. How are NA and Turbo different

When comparing the both the 1.6 engines, it was found that from 1500rpm to 4500rpm both engines had a similar flame speed and combustion duration. The turbo engine started later but also ended later. This changed for 1000rpm and any RPM above 4500rpm. At these rpms, the turbo has a steeper gradient meaning the flame is spreading faster. This leads to the turbo engine still starting later but the end of combustion starts to get close to the end for the NA.

6.3. Impact of engine capacity

At 1000rpm there is very little difference between the two sizes. However, as the rpm increases the 1.6 gets a steeper gradient during the rapid burning phrase. At high RPMs, the 1.0 is very similar to the NA 1.6 just starting and ending later. The 1.0 seems to start combustion in the initial zone faster than the 1.6 but in the rapid burn zone the 1.6 starts to catch up and overtake the 1.0. This could be because of the extra size of the cylinder. The flame has more room to grow meaning when it's at its largest, just before coming into contact with any boundary, the flame is spreading its quickest. The extra volume might allow for the flame to spread fast enough that it is faster than the flame in the 1.0 can get before hitting the boundaries of the cylinder walls and piston.

6.4. Effect of load on combustion

Even though there are only two situations to compare Full load and Part load, it can be seen that the full load is igniting later than the part load in both cases. The full load is also burning quicker because it has a higher gradient in both diagrams. For 2000rpm the difference is quite significant but for 3000rpm both lines start to look similar in terms of gradient with the full load only having a slightly steeper gradient. The difference in burn speeds could be because of the pressure being so low in the part load there will be big gaps between fuel particles so the flame struggles to spread. There is also more time for heat to transfer away from the flame and the flame isn't not getting the fuel needed to sustain the itself.

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