

University of Wales Trinity Saint David

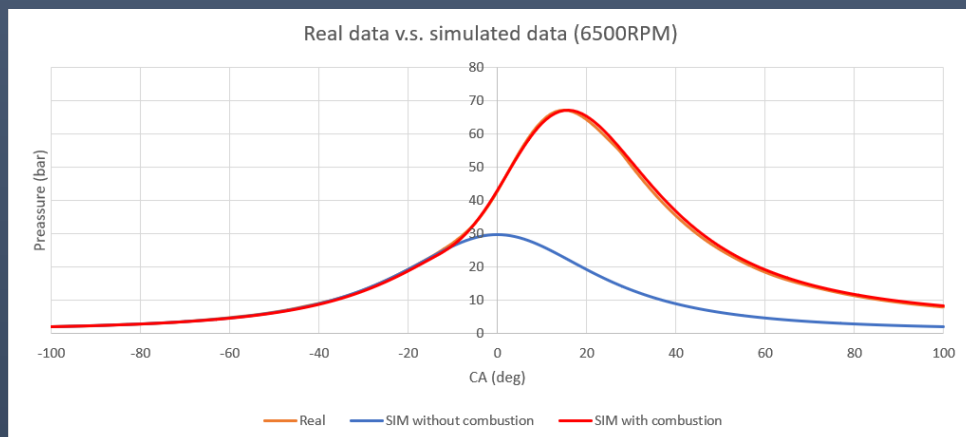
School of Engineering

Powertrain & Control (M5X00404)

Mateusz Daniel Skoczylas 1704045

Last edited: 20/01/2020

Engine Analysis & Simulation



Contents

Figures.....	1
Tables	1
About this report.....	2
1 Part 1 – Analysing Ford Sigma 1.6 Data	3
1.1 Engine identification	3
.....	3
1.2 Pressure data observations.....	4
1.3 Relationship between torque and cylinder pressure.....	5
1.4 Let’s explain the behaviour above – losses.....	6
1.4.1 The Otto cycle & PV graph	6
1.4.2 Work Done (WD).....	7
1.5 Polytropic index	9
1.5.1 Polytropic index over crank angle.....	9
1.5.2 Average polytropic index – Compression & expansion	10
1.6 Piston Velocity.....	11
1.7 Combustion temperature	12
2 Part 2 – Engine Simulator.....	14
2.1 Geometry	15
2.2 Simulation model	16
2.2.1 Non-combustion simulation model	16
2.2.2 MFB & Wiebe Function	17
2.2.3 Combustion simulation model & validation	19
2.3 Observations	20
2.4 Limitations.....	21
3 References	22

Figures

Figure 1 - ZSG 416 engine dyno graph	3
Figure 2 - Pressure over crank angle at various RPM graph	4
Figure 3 - First derivative of pressure vs crank angle	5
Figure 4 - Average pressure vs real torque	6
Figure 5 – PV diagram anomaly	6
Figure 6 – Ideal Otto Cycle (Luc1992, n.d.)	6
Figure 7 - Work Done graph.....	8
Figure 8 - Compression work done	8
Figure 9 - Expansion work done.....	9
Figure 10 - % of negative work in total work done	9
Figure 11 - Polytropic index over CA.....	9
Figure 12 – Average polytropic index (LogP/LogV)	10
Figure 13 - Piston position	11
Figure 14 - Piston velocity	11
Figure 15 - Cylinder temperature over CA	12
Figure 16 - Engine simulator - Main page	14
Figure 17 - Non-combustion simulation model	16
Figure 18 - Basic MFB table.....	17
Figure 19 - MATLAB Wiebe function.....	17
Figure 20 - Basic MFB data vs Wiebe interpolation.....	18
Figure 21 - Burn rate	18
Figure 22 - Combustion simulation model.....	19
Figure 23 - Final combustion model	20

Tables

Table 1 - Zetec-S/SE variants up to 2003 (Wikipedia, n.d.).....	3
Table 2 - Properties for Temperature calculation (part 1).....	12

About this report

In part 1, this report focuses on analysing the engine characteristics of a naturally aspirated Ford Sigma 1.6 engine. Part 2 focuses on simulating this engine, with focus on making the simulation as close to real data as possible.

1 Part 1 – Analysing Ford Sigma 1.6 Data

1.1 Engine identification

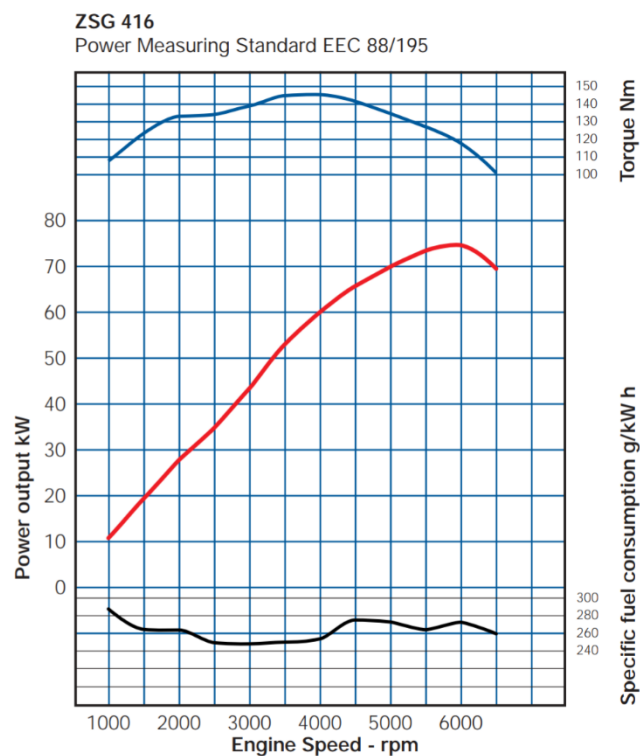
Note: The Sigma family includes many engines. There are many 1.6 Litre engines in the Sigma family, starting with the earlier Zetec S/SE models, and ending with the Duratec Ti-VCT range.

The date of the test on the excel file is specified as 06/02/2003. The only sigma 1.6L engine available at the time was the Zetec S/SE 1.6L (Table 1) (Wikipedia, n.d.), thus it can be assumed that the data supplied in 'Sigma 1.6' excel file refers to this particular engine. It is a relatively simple 4 cylinder 16 valve engine, that is naturally aspirated and has no variable camshaft timing.

Table 1 - Zetec-S/SE variants up to 2003 (Wikipedia, n.d.)

Zetec-S/SE

Code	Years	Displacement	Bore x stroke	Power@rpm	Torque@rpm	Compression
1.25 Zetec-S/SE	1995-	1.2 L; 75.8 cu in (1,242 cc)	71.9 mm × 76.5 mm (2.83 in × 3.01 in)	75 PS (55 kW; 74 hp)@5200	110 N·m (81 lb·ft)@4000	10.0:1
1.4 Zetec-S/SE	1998-	1.4 L; 84.7 cu in (1,388 cc)	76 mm × 76.5 mm (2.99 in × 3.01 in)	75 PS (55 kW; 74 hp)@5000	124 N·m (91 lb·ft)@3500	11.0:1
	1995-			90 PS (66 kW; 89 hp)@5500	123 N·m (91 lb·ft)@4000	10.3:1
1.6 Zetec-S/SE	1998-	1.6 L; 97.4 cu in (1,596 cc)	79 mm × 81.4 mm (3.11 in × 3.20 in)	100 PS (74 kW; 99 hp)@6000	145 N·m (107 lb·ft)@4000	11.0:1



The graph on the left shows the Ford supplied power and torque figures for a ZSG 416 engine (codename for the crate version of 1.6 Zetec-SE).

Figure 1 - ZSG 416 engine dyno graph

1.2 Pressure data observations

Generating a pressure over crank angle graph (Figure 2 - Pressure over crank angle at various RPM graph) shows the characteristic of pressure changes in an engine cycle. It is an easy way to quickly see what happens in the cylinder, however is not the most accurate way to determine when combustion begins [this is best achieved by comparing non combustion simulated model to real data].

Pressure at various RPM

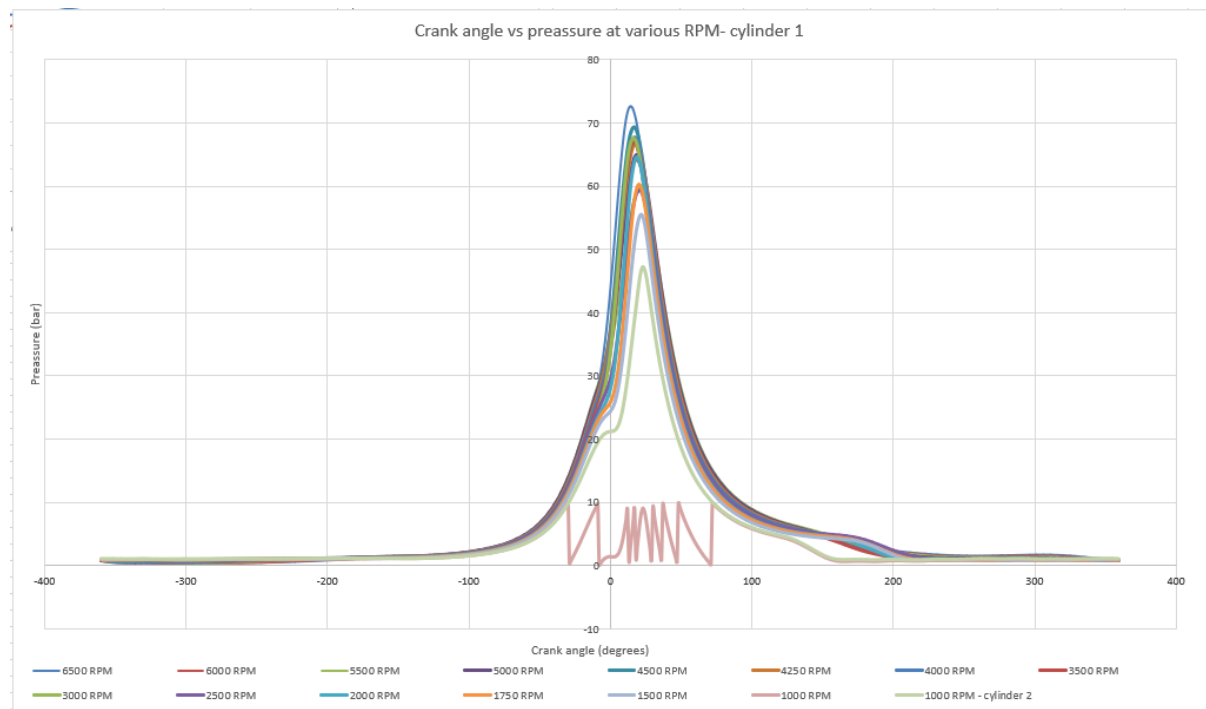


Figure 2 - Pressure over crank angle at various RPM graph

Note: the 1000 RPM data for cylinder 1 is corrupted (random spikes to close to partial vacuum between crank angle of -29 to 70 degrees, hence the data from second cylinder which doesn't contain this anomaly was also plotted. As a general rule of thumb, I avoid using 1000RPM dataset due to the amount of errors it contains.)

Initial observations point to combustion begging before TDC, the energy is released over the end of compression and beginning of expansion cycle. A very sharp peak in pressure peak occurs just after TDC and not before due to charareristics of combustion (as fuel is not burned instantaneously, see section 2.2.2), change in piston displacement being small per degree of crank angle in TDC/BDC areas (see section 2.1), and combustion chamber volume being small (exponential raise in pressure the smaller the volume, which is seen on simulation without combustion, section 2.2.1).

The graph also shows some interesting characteristics when it comes to combustion timing. The flame velocity and MFB won't change much with RPM, however the piston velocity will. This means that spark timing needs to be more advanced at higher RPM to reach MBT.

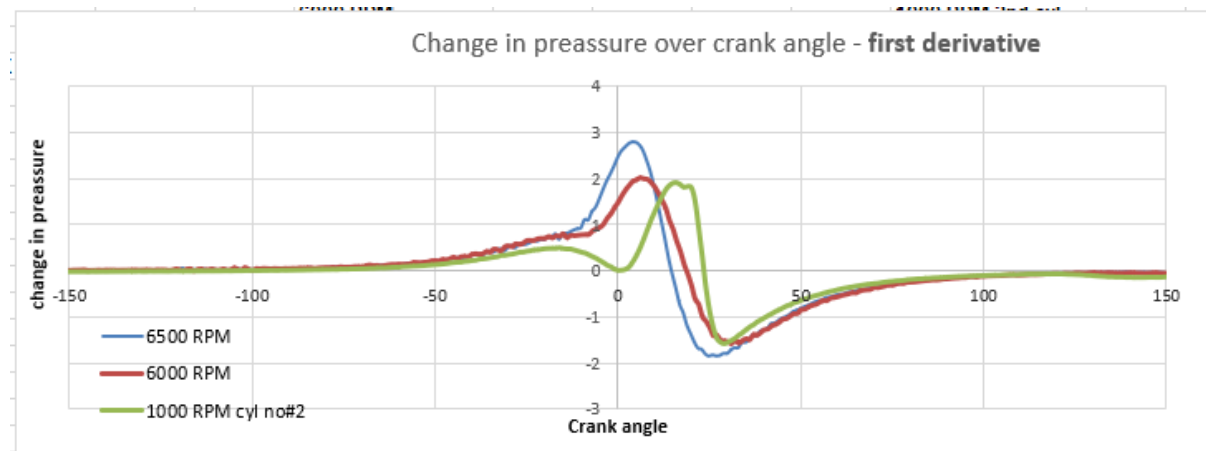


Figure 3 - First derivative of pressure vs crank angle

The first derivative of the graph above shows this behaviour more clearly. (Figure 3 - First derivative of pressure vs crank angle)). As raise in pressure due to combustion starts closer to TDC at low RPM values, less work will be done against the piston as its travelling upwards. This means less work is done against the direction of travel of the piston (**see work done**), indicating higher efficiency at lower RPM.

The general trend also shows a small increase in peak pressure with higher RPM until a certain point, showing that peak pressure does not necessarily need to be higher with higher RPM. This sort of pattern correlates with torque curve (further discussed in section 1.3) as opposed to a power curve.

The derived graph also suggests that at lower rpm there are fewer pumping losses due to less work being done before the piston gets to TDC (retarded timing). I verify this by looking at PMEP (section **Error! Reference source not found.**).

I haven't found a second derivative of P/CA graph to be useful.

1.3 Relationship between torque and cylinder pressure

To gather a basic idea of the correlation between cylinder pressure and torque, I have transferred engine dyno data from the ford into excel and plotted against the university supplied sigma 1.6 pressure data. While the line characteristics show that torque curve is somewhat proportional to the pressure in the cylinder at low RPM, at higher rpm the torque curve starts to deviate from pressure curve. This can be attributed to losses which I explore in further parts of this report.

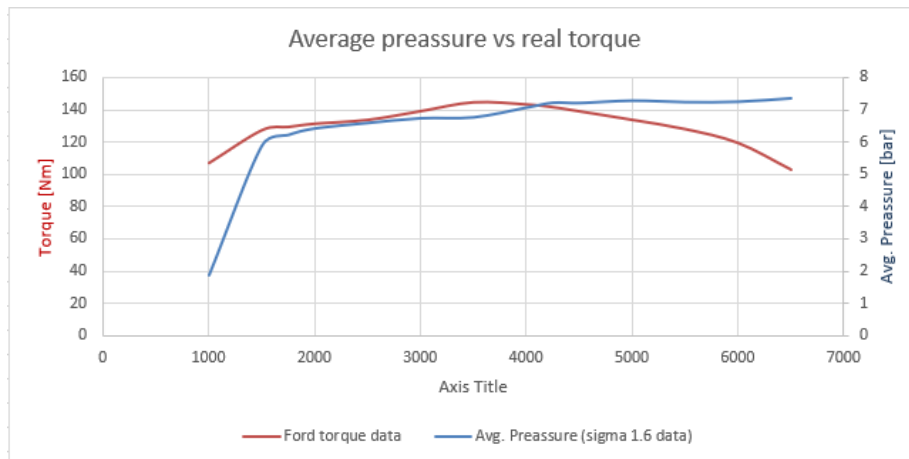


Figure 4 - Average pressure vs real torque

1.4 Let's explain the behaviour above – losses

Thesis: High RPM engines (such as the ones in high performance motorcycles) suffer from higher losses than their low RPM counterparts.

1.4.1 The Otto cycle & PV graph

Looking into the Otto cycle is the first step in determining the reason for the behaviour shown at the beginning of section 1.4. I have used it to gather an understanding of the correlation between MEP and the stages of 4 stroke cycle. The Otto cycle (Figure 6 – Ideal Otto Cycle) is an ideal representation of the 4 stroke cycle. It doesn't take changes in polytropic index (the process stays adiabatic) and heat transfer rate into account, hence I have also generated a PV graphs (Figure 5) from the sigma 1.6 pressure and volume data supplied.

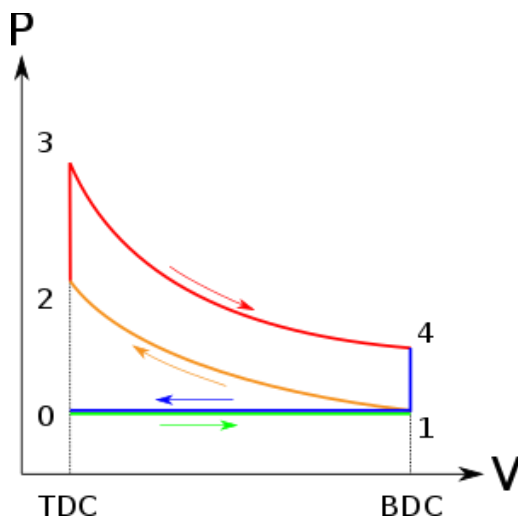


Figure 6 – Ideal Otto Cycle (Luc1992, n.d.)

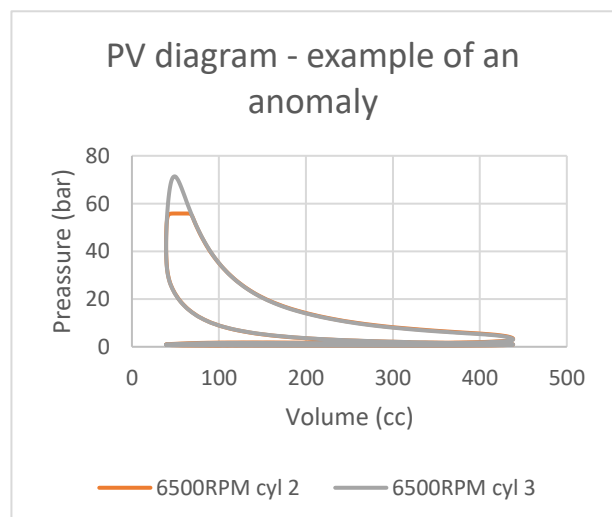


Figure 5 – PV diagram anomaly

It's important to note that during the analysis of PV graphs, I found that there are a lot of anomalies, especially in low RPM range, but some at high rpm too, as shown on Figure 5 (sensor reading going

flat in cylinder 2 at the start of exhaust stroke). For the rest of this section, I have made my best effort to choose pressure data which doesn't contain any anomalies, by comparing pressures between different cylinders, and using mean pressures between cylinders where appropriate

1.4.2 Work Done (WD)

Before finding MEP, I have looked at WD at various RPM values, which gives me a general overview of what the system is doing. Any negative work done indicates losses. Work done is calculated as pressure

Compression Work (Compression Stroke)

Any compression work will be negative, as it will be acting against the piston while it's travelling towards TDC on the compression stroke. Measured from -180 deg (BDC) to 0 deg (TDC) crank angle

Equation 1 - Compression Work (Blair, 1999)

$$W_c = \int_{bdc}^{tdc} p dV$$

Expansion Work (Power Stroke)

Any expansion work will be positive, as the expansion of gasses propel the piston from TDC (0 deg) to BDC (180 deg) on the power stroke.

Equation 2 – Expansion Work (Blair, 1999)

$$W_e = \int_{tdc}^{bdc} p dV$$

Results - Work done over full engine cycle

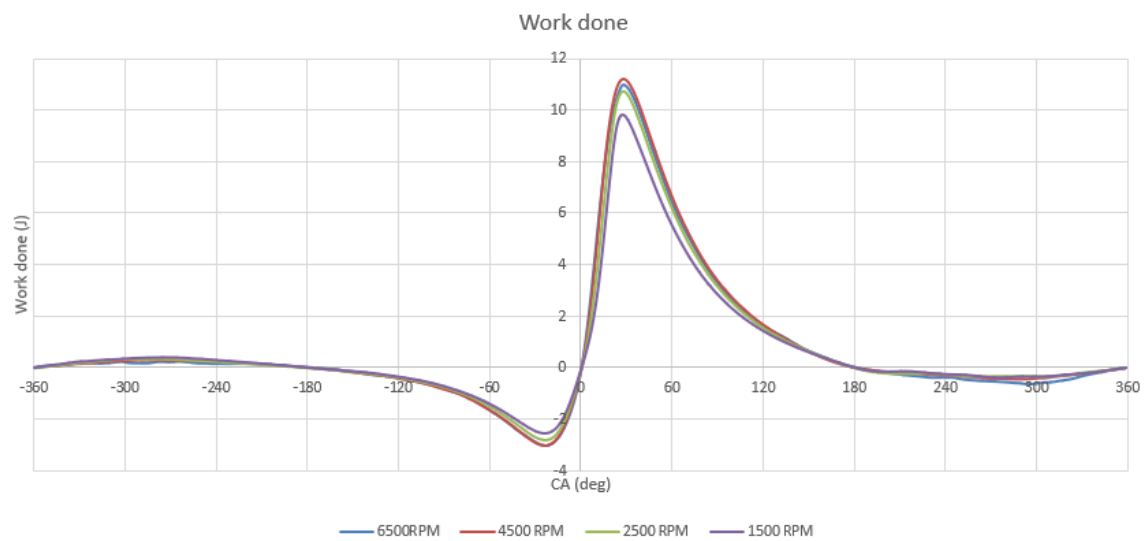


Figure 7 - Work Done graph

Compression work graph

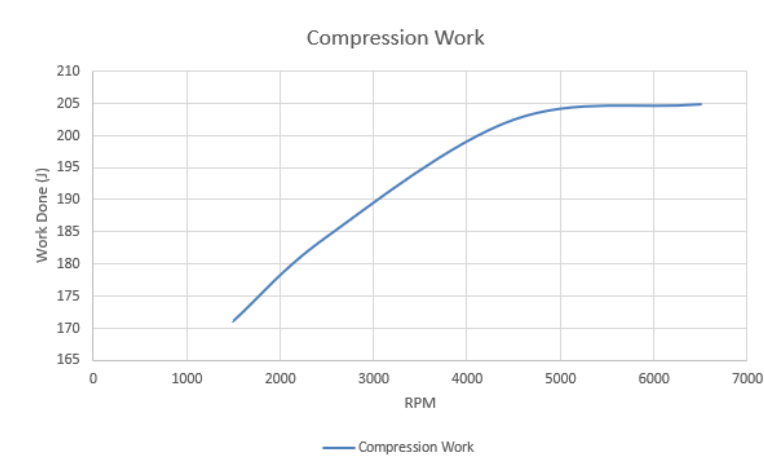


Figure 8 - Compression work done

Note: For the purpose of graph formatting, Work done due to compression has been shown as a positive number. As it is acting against the system, it is considered to be negative in calculations.

Work done against the piston increases with RPM as flame starts to propagate earlier in order to reach MBT, however it is not very significant (delta of 38 J between 1500 and 6500RPM).

Expansion work graph

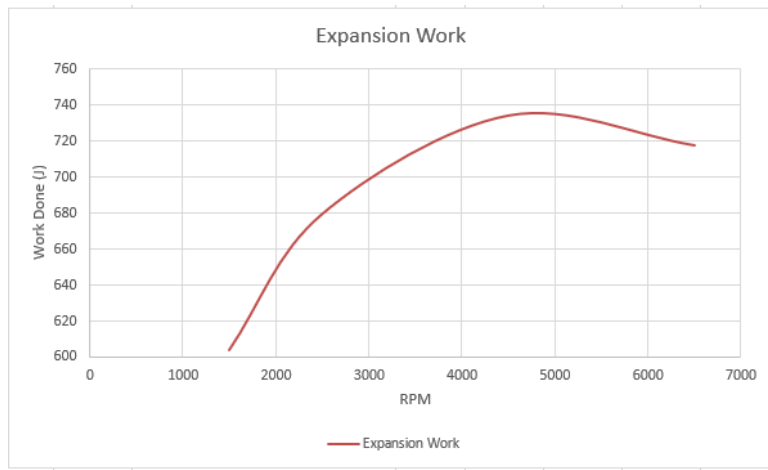


Figure 9 - Expansion work done

Negative work done is fairly significant, making up ~28% of total work done.

% of negative work in total work done	
RPM	WD
6500	28.56%
4500	27.59%
2500	27.12%
1500	28.34%

Figure 10 - % of negative work in total work done

1.5 Polytropic index

1.5.1 Polytropic index over crank angle

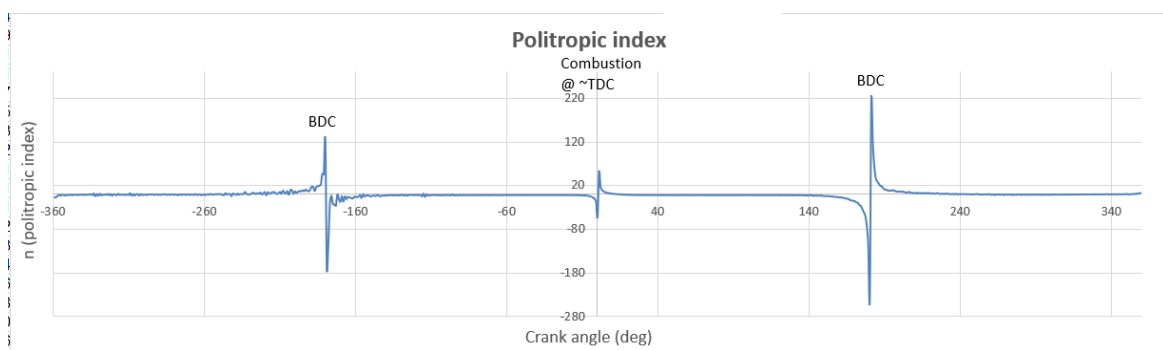


Figure 11 - Polytropic index over CA

1.5.2 Average polytropic index – Compression & expansion

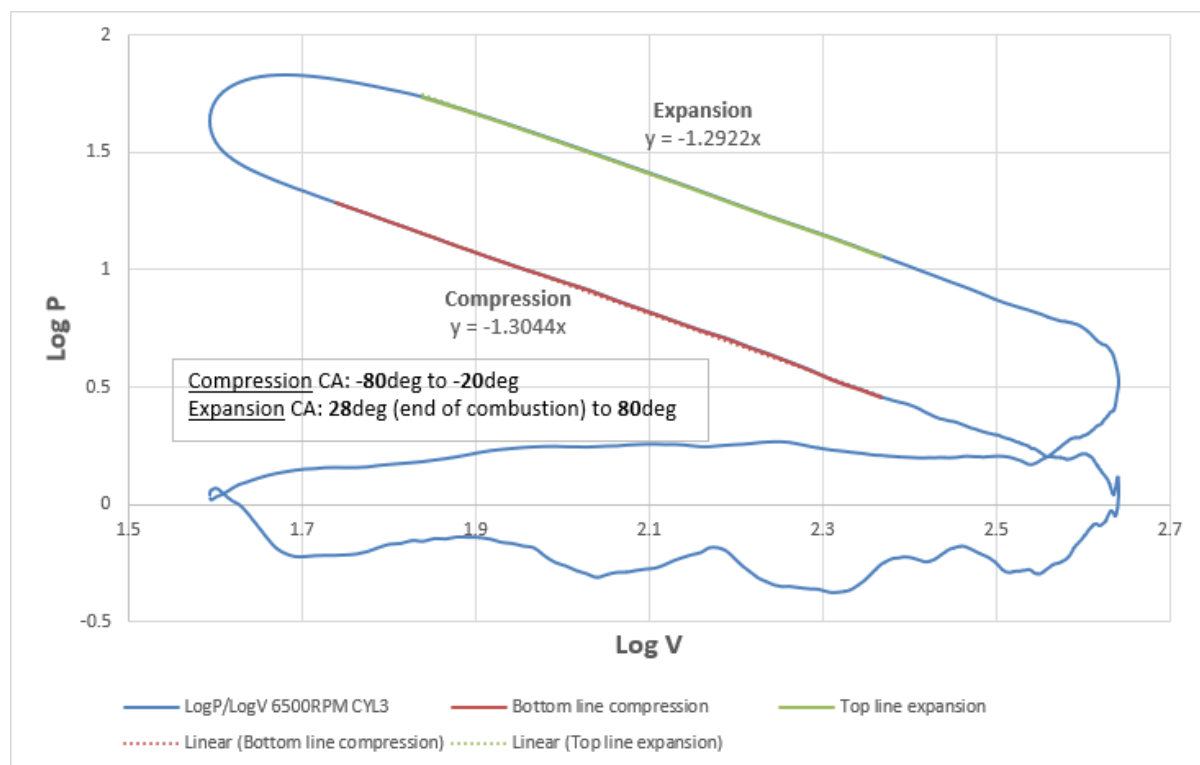


Figure 12 – Average polytropic index (LogP/LogV)

My main method of retrieving polytropic index was to take logs of pressure and volume, and focus on areas where the valves are closed (as indicated by straight line areas). I have ignored areas where combustion occurs, as the polytropic index value goes abnormal during combustion (as well as BDC areas as demonstrated in (Figure 11)). The polytropic index is equal to the gradient of a line. Using the knowledge of adiabatic and isothermal processes, I have determined that those values are within expected range.

I will use the compression value of polytropic index at a later stage (simulation) to produce a non-combustion model.

1.5.2.1 Determining combustion start and end using polytropic index

In addition to using P/CA graph to determine start of combustion, I have also looked at polytropic index at each CA (Figure 11) as a way of determining combustion start and end. According to the polytropic index, combustion effectively starts at around -15deg and ends at around 13deg (for 6500RPM). I believe the combustion start value is correct however I would expect end value to be further into the expansion cycle. I believe I will get a more accurate result by looking at real pressure data vs non-combustion simulation model pressure data.

1.6 Piston Velocity

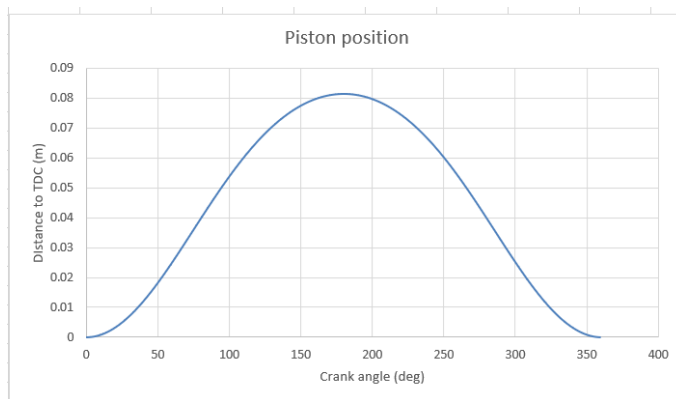
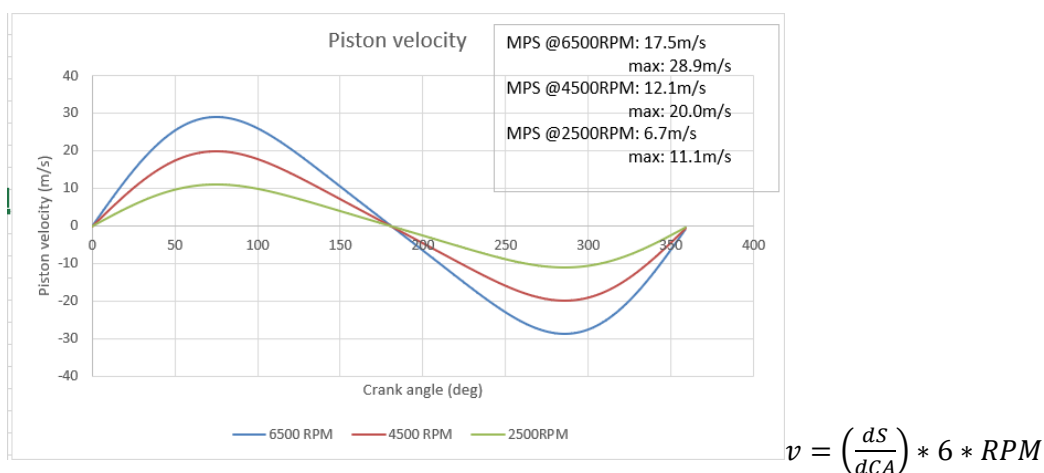


Figure 13 - Piston position



Where: v = velocity (m/s), S = distance, CA = crank angle (deg)

Figure 14 - Piston velocity

To work out velocity, I first had to find piston position at any crank angle. I cover this topic in detail in Part 2 of this assignment report. As RPM doubles, the piston speed will double as well. Due to the design of the crank, piston displacement is not linear as it moves through its stroke, causing the velocity to change quite drastically. For this reason, I have not only looked at mean piston speed (MPS), but also the maximum velocity values.

MPS is dependant of the stroke and RPM of the engine. The typical MPS value for a high revving motorcycle (e.g. Yamaha R6) is 22.6 m/s at maximum rpm (16,000 for R6). The Sigma 1.6L engine only reaches 17.5 m/s at max rpm of 6500. The advantage of having lower max MPS is longer engine life (due to less stress on engine components, but not to be mistaken with lower combustion pressures), lower frictional losses (friction is exponential with velocity), and less work used to lift valves open due to softer valve springs being used. On the other hand, having a high MPS limit allows

for higher RPM and thus more power. Sport bikes sacrifice engine durability & efficiency for more power when compared to typical family cars such as ford focus mk1 with the sigma 1.6 engine.

1.7 Combustion temperature

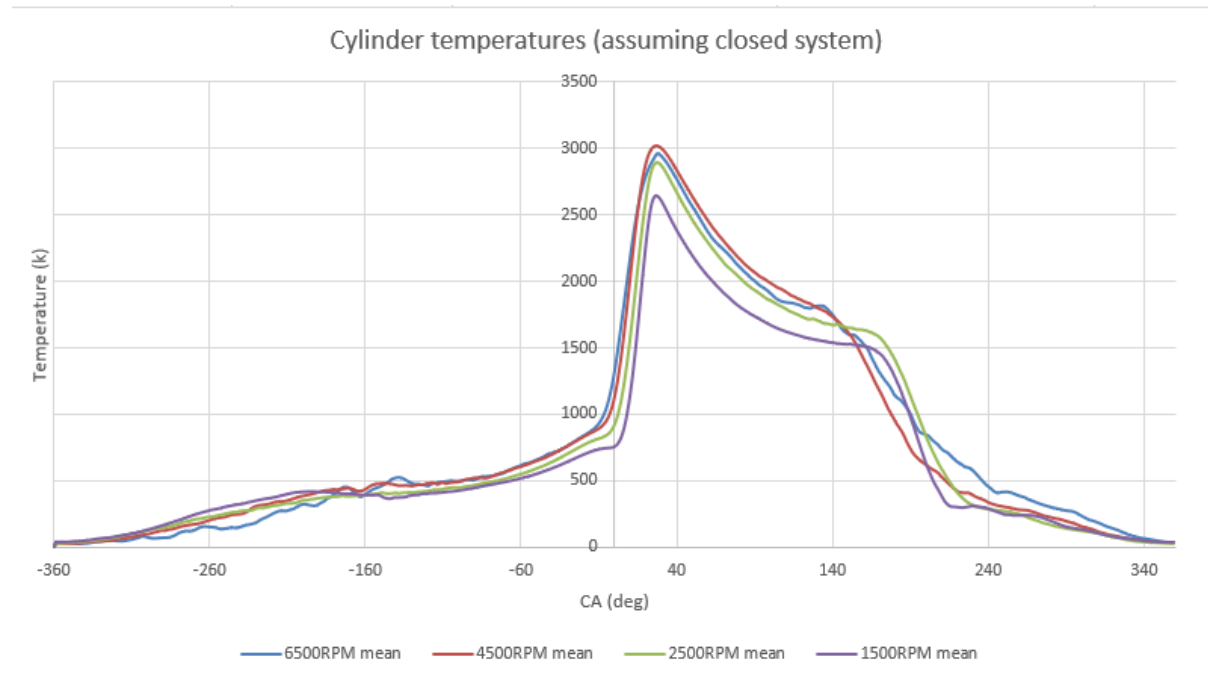


Figure 15 - Cylinder temperature over CA

Table 2 - Properties for Temperature calculation (part 1)

Lambda:	0.9
AFR:	13.23 Ratio
VE	90 %
Density of air + fuel (gas)	1.225 kg/m ³
Bore radius	0.0395 m
Swept volume (TDC:BDC)	0.000399 m ³
Stroke	0.0814 m
Total mass	0.00043989 kg
Mass of air	0.00040898 kg
Mass of fuel	3.0913E-05 kg

$$T = \frac{PV}{(m_f + m_{air}) * R}$$

Where constant $R = 287$

It is important to note that I was not able to find an exact value for the density of petrol in atomised state. I have made an educated guess for the density of fuel. Considering that petrol enters the cylinder in vaporised gas state, I assumed its value to be the same as the density of air at 15C and 1bar (1.225kg/m³).

I have observed the peak combustion temperatures reach between ~2500k and ~3000k depending on RPM. The profiles of the peak temperatures reassemble pressure curves, with higher rpm typically resulting in somewhat higher temperatures (but not always).

2 Part 2 – Engine Simulator

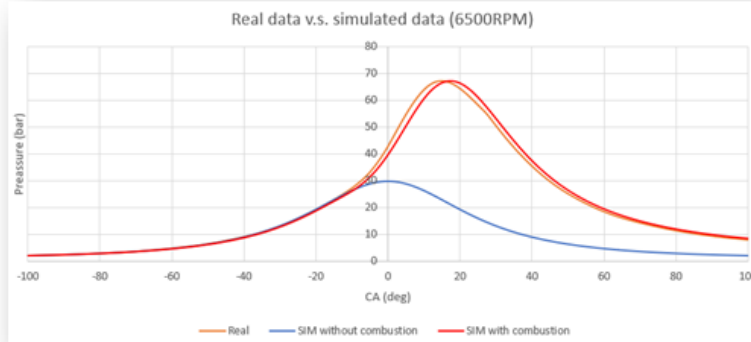
Engine Simulator

Last updated: 20/01/2020 Matt

Calculated values on this worksheet are in orange, values in black need to be entered manually

Real starting point combustion pressure (P1) required for calculation of simulation model, as found in Copy_real_6500RPM s

Other worksheets use data from this one for their calculations



	Value	Units	Abbreviation	Notes
General settings				
Name	Sigma 1.6			
Compression Ratio	11.16	(ratio :1)	CR	
Conrod legth	0.137	m	I	
Stroke	0.0814	m		
Bore	0.079	m		
Lambada	0.9			
VE	90	%		
Air & atomized fuel density	1.225	kg/m^3		air value for 15 deg c at 1 bar
Calorific value of fuel [LHV only]	43.4	MJ/kg		lower heating value
Combustion eff	90	%		
Gamma	1.27			Owen: 1.25. Changes sim peak, adjust alonside 'a'
n compression	1.3044			
n expansion	1.2922			
Basic MFB settings (based on real data and assumptions for 0% and 100%)				
MFB (ratio)	CA (AV deg)	notes		
	0	-15	6500 RPM assumed based on non combustion model vs real data delta	
	0.02	-7.373	6500 RPM	
	0.05	-4.541	6500 RPM	
	0.1	-1.766	6500 RPM	
	0.5	10.802	6500 RPM	
	0.9	24.101	6500 RPM	
	1	36	6500 RPM assumed	
Advanced MFB settings (take values from matlab)				
Weibie function parameters				
m=	1.5	//Start of combustion gradient		
Combustion start (deg)	-15			
Combustion end (deg)	36			
a=	1.77	//End of combustion gradient		
adjust a and m in conjunction with gamma				
Usefull calculated values				
Swept volume	0.000398996	m^3		
Vc	3.92713E-05	m^3		
Crank radius	0.0407	m		

Figure 16 - Engine simulator - Main page

Notes:

- I have assumed the volumetric efficiency to be 90%. The typical volumetric efficiency for a 4 stroke 4 valve engine is between 88% to 95% (HPWizard.com, n.d.)
- The value used for the calorific value of petrol is the lower heating value of 43.4 MJ/kg (engineeringtoolbox.com, n.d.)
- Combustion efficiency is assumed to be 90%
- Compression/expansion polytropic index values are reused from Part 1 of the assignment – logP/logV gradient
- Gamma value was set using trial and error method, to a value that provided a simulation result that was as close to real data set as possible

2.1 Geometry

For part 2 of the assignment, I have used formulas to work out the volumes (V_s , swept volume, V_c) and piston displacement. The simulation model relies on calculated volumes.

Swept volume-

$$Sv = \pi * r^2 * l$$

Where: r = radius of the bore, l = stroke

Piston distance from TDC –

$$s = a + l - a \cos \theta - l \cos(\sin^{-1}(\frac{a}{l} \sin \theta))$$

Where: d = radius of crank, l = conrod length, ϑ = crank angle

V_s

$$Vs = s * \pi * b_r$$

Where: s = piston distance from tdc, br = bore radius

V_c

$$Vc = \frac{Sv}{(CR - 1)}$$

Where: Sv = swept volume, CR = compression ratio

Total volume

$$Vs + Vc$$

2.2 Simulation model

2.2.1 Non-combustion simulation model

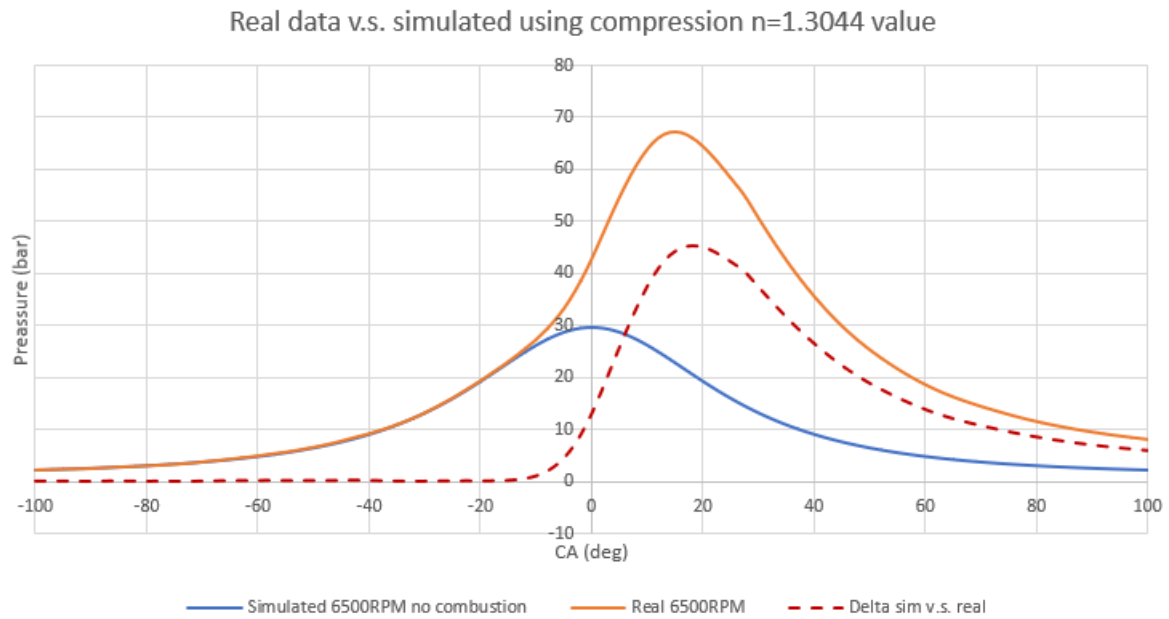


Figure 17 - Non-combustion simulation model

$$P2 = P1 \left(\frac{V1}{V2} \right)^n$$

Where: $P1$ is the previous pressure value, $V1$ is the previous volume, $V2$ is the current volume, n is the polytropic index value.

Note: the $P1$ value for the first $P2$ result was taken from real data. Every other crank angle following the first result uses calculated values

Comparing the real data with non-combustion model shows that the combustion at 6500RPM begins at -15degrees CA, when the two lines start diverging from each other (as represented by the delta line). I believe this method of determining the start of combustion is the most accurate. The delta between the peak pressures of the datasets is 45 bar, showing that the most of the pressure is generated due to combustion.

2.2.2 MFB & Wiebe Function

2.2.2.1 Supplied MFB data

Basic MFB settings (based on real data and assumptions for 0% and 100%)		
MFB (ratio)	CA (AV deg)	notes
0	-15	6500 RPM assumed based on non combustion model vs real data delta
0.02	-7.373	6500 RPM
0.05	-4.541	6500 RPM
0.1	-1.766	6500 RPM
0.5	10.802	6500 RPM
0.9	24.101	6500 RPM
1	36	6500 RPM assumed

Figure 18 - Basic MFB table

Since the supplied 'sigma 1.6' data doesn't contain MFB CA values for 0% and 100% burn, I had to estimate those values. For start of combustion, I used the CA value of -15 DEG CA that I determined from the non-combustion model. For the end of combustion, I noticed that combustion slows down towards the end in terms of CA duration, and thus assumed a CA value that was appropriate for 100% burn.

2.2.2.2 Filling in the blanks – Wiebe Function

$$MFB = 1 - e^{-a\left(\frac{\theta - \phi}{d\theta}\right)^{m+1}}$$

Where: a and m are the variables for the profile of the curve, θ is the crank angle, ϕ is the CA at the start of combustion and $d\theta$ is the burn duration

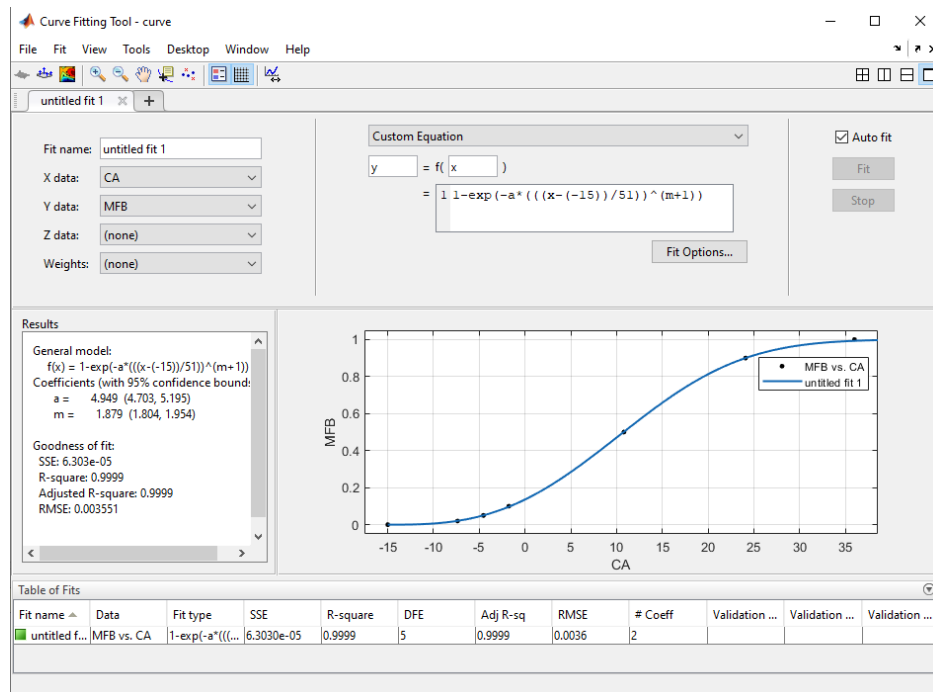


Figure 19 - MATLAB Wiebe function

By inserting the Wiebe function along the MFB values in section 2.2.2.1 into Cftool in MATLAB, I was able to generate 'a' and 'm' values to use with the Wiebe function. I used this function to generate a profile of the MFB curve, to allow me to interpolate MFB values per each degree of crank angle.

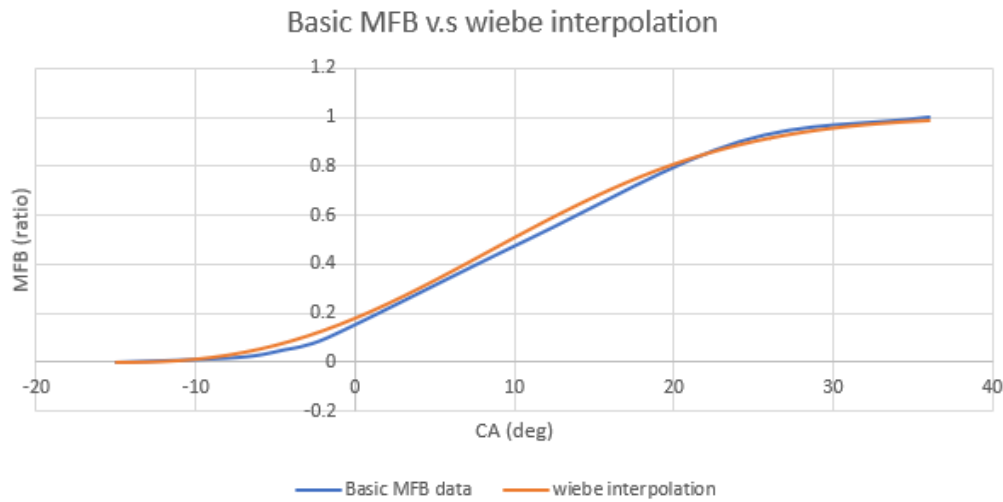


Figure 20 - Basic MFB data vs Wiebe interpolation

Once I had a smooth MFB curve, I also graphed the burn rate to get an idea where it peaks. The peak is around 8 to 9 deg. CA.

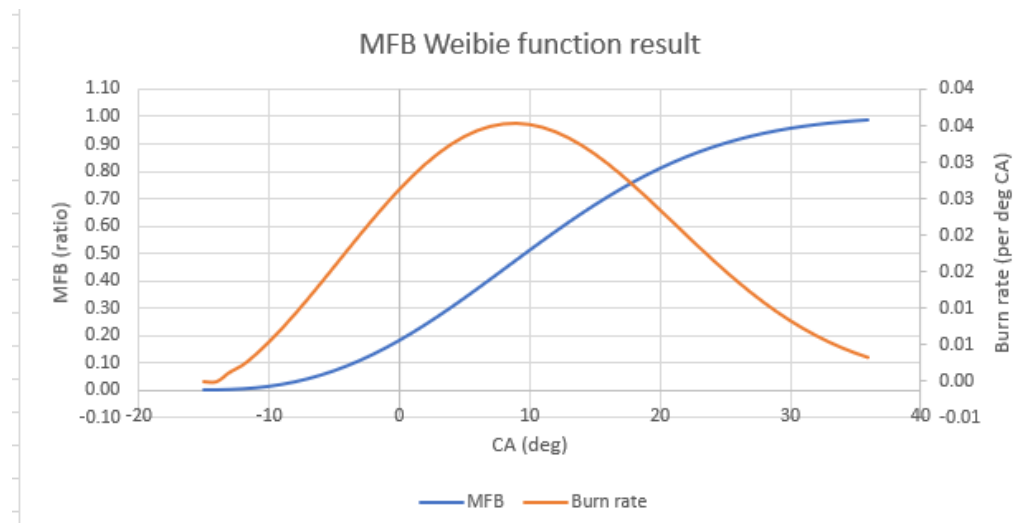


Figure 21 - Burn rate

2.2.3 Combustion simulation model & validation

Amount of energy released per degree of crank angle (dQr)

$$dQr = (\text{total fuel mass} * \text{burn rate}) * \text{calorific value} * \text{combustion efficiency}$$

Combustion pressure:

$$P2 = P1 \left(\frac{V1}{V2} \right)^n + \frac{dQr(\gamma-1)}{V2}$$

Where: P1 is the previous pressure value, V1 is the previous volume, V2 is the current volume, n is the polytropic index value, dQr is amount of energy released per degree of CA, γ is ratio of specific heats

Energy released

$$Q = \text{Mass} * \text{calorific value} * \text{combustion efficiency}$$

I have calculated the total amount of Joules released due to combustion to be 1207.46 J

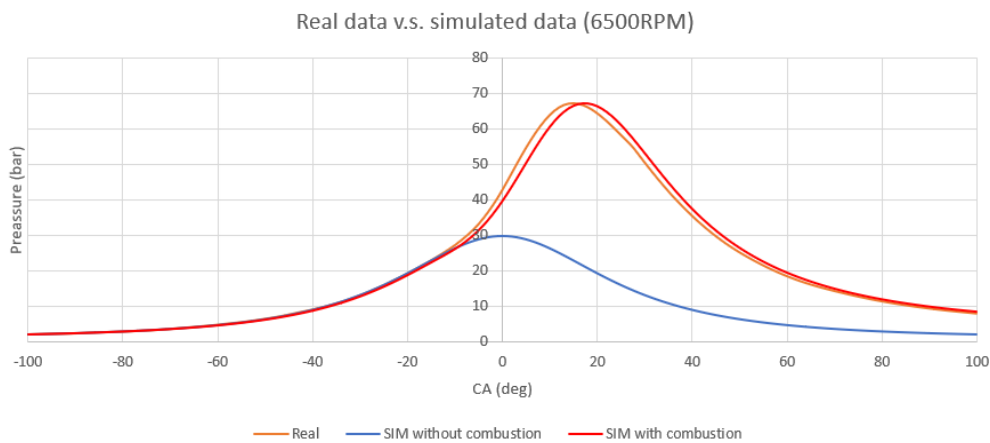


Figure 22 - Combustion simulation model

At this point, my simulation model was very close to real data, however, I notice that my MFB profile is causing the fuel burn to be more delayed than in real life.

2.2.3.1 Adjusting a , m and gamma values for a more precise simulation

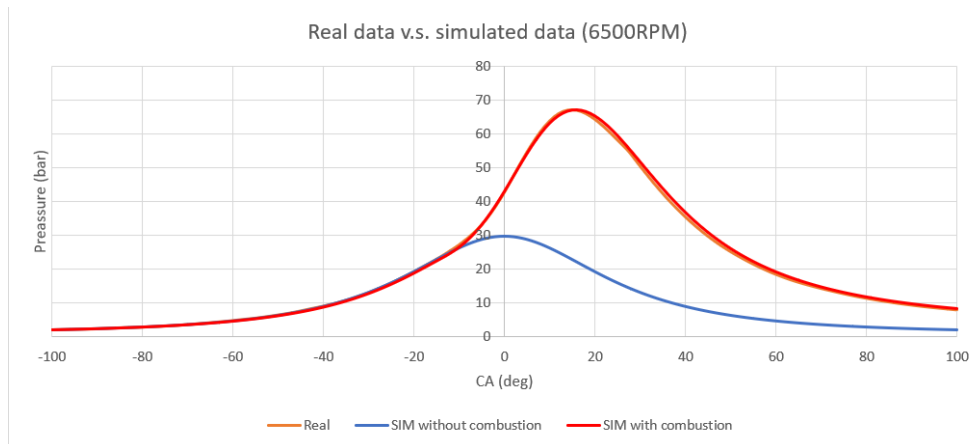


Figure 23 - Final combustion model

I was able to get my own simulation model to be almost identical to real data by modifying the ' a ', ' m ' and gamma values manually.

Final settings:

$n=1.3044$ (switching to expansion n as combustion starts doesn't affect my model in a noticeable way)

$a=1.77$

$m=1.5$

2.3 Observations

By modifying my model, I have observed a few interesting occurrences:

- Increasing the compression ratio will increase the amount of work done against the piston during the compression stroke. Retarding ignition timing with higher compression ratio could possibly yield better results as long as pre-ignition and detonation is not an issue.
- Making the conrod shorter will have a similar effect to starting combustion earlier.
- Going from AFR of 13.23 to Stoichiometric 14.7 would cause a performance decrease (due to lower cylinder pressures). While emissions might possibly become better, the performance will be lower.

2.4 Limitations

- For calculating the initial simulated pressure value, a pressure value from the real data had to be used as P1. This means that if the real data is corrupted, the simulation will not be correct.
- My MFB CA start end point is an educated guess. Because values for 0% and 100% burn were not provided, my simulation might not be as accurate as it could be.
- I had to take an educated guess for the volumetric efficiency value, which might not make the simulation as accurate as it could be
- My combustion efficiency value is an educated guess. If I had the chance to take another attempt at making my simulation, I would calculate it by dividing work done by chemical input.
- The ambient temperature of the system with valves open is not known, thus, the density values I have used might not be exact (I assumed 15 deg c and 1 bar).

3 References

Blair, G. P., 1999. *Design and Simulation of Four-Stroke Engines*. Warrendale, PA: SAE.

Dannana, S., n.d. *What is Mean Effective Pressure? | IMEP, BMEP, FMEP/*. [Online]

Available at: www.extrudesign.com/mean-effective-pressure-imep-bmep-fmep

engineeringtoolbox.com, n.d. *Fuels - Higher and Lower Calorific Values*. [Online]

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

FORD COMPONENT SALES, 2007. *POWERTRAIN PRODUCT GUIDE*. [Online]

Available at: <http://www.ctm.it/pdf/Powertrain%20Product%20Guide%202007.pdf>

[Accessed 11 December 2019].

heeman10, n.d. *Started my university practical work today - dyno*. [Online]

Available at: www.fiestaturbo.com/forums/started-my-university-practical-work-today-dyno-vt118642

HPWizard.com, n.d. *Volumetric Efficiency*. [Online]

Available at: <http://hpwizard.com/volumetric-efficiency.html>

Luc1992, n.d. *Pressure–volume diagram*. [Online]

Available at: https://en.wikipedia.org/wiki/Otto_cycle#/media/File:P-V_Otto_cycle.svg

petrolheadgarage.com, n.d. *ZETEC Engines*. [Online]

Available at: www.petrolheadgarage.com/Posts/zetec-engine/

[Accessed 11 12 2019].

Wikipedia, n.d. *Ford Sigma engine*. [Online]

Available at: https://en.wikipedia.org/wiki/Ford_Sigma_engine

wikiwand.com, n.d. *Ford Ka*. [Online]

Available at: www.wikiwand.com/en/Ford_Ka

ZSG, 2003. *ZSG RANGE*. [Online]

Available at: <https://www.i-m->

[a.de/pdf/Ford/gasoline_gaseous_engines/ZSG_Range_Leaflet_4_2003.pdf](https://www.i-m-a.de/pdf/Ford/gasoline_gaseous_engines/ZSG_Range_Leaflet_4_2003.pdf)

[Accessed 14 December 2019].

