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THERMAL ENERGY FORMATION, DISTRIBUTION AND RECOVERY MODELLING

Bachelor of Engineering in Advanced Engine

Abstract

Exploration of formation and distribution of thermal energy during operating conditions of a IC engine, coupled with efficiencies and recoverable energy potential. The construction of models are used to analysis distribution of fuel power and the quantity converted to mechanical work, whilst using combustion models to compensate for anomaly production data.

'The 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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2.0 Introduction

With the ever-increasing demand to improve combustion engine efficiency and tailpipe emissions whilst maintain performance, modeling the operating cycle is becoming a key part in modern development schedules, modelling is used to reduce time and save money. With a computer model, small details can be modified and results quickly refined rather than the construction, testing and analysis of new components, this will prove fundament when attempting to compensate for looming random driving emissions (RDE) test.

The study will predominantly use Microsoft Excel to analysis the combustion process of three Ford engines, 1.6litre naturally aspirated, 1.5litre turbocharged and 1.0litre FOX eco-boost. The combustion model will determine the effects of capacity, fuel equivalence ratio and turbocharging have on burn speed and combustion temperature, and explore the faults and limitations of model methods.

Once a vehicle is developed, it undergoes the New European Drive Cycle (NEDC) which details a complete listing of all engine parameters including, temperatures, mean effect pressure, turbocharger behavior and combustion properties. This test data is used to determine alternate values and to construct functional models which can be validated again the NEDC data to determine accuracy.

With an abundance of operating temperature data available for the FOX engine, the turbocharger (TC) and engine efficiency models will be constructed for it, as there is comparable data to allow for model accuracy analysis.

For the purpose of this study, an engine thermal efficiency model will be formed to analysis distribution of fuel power. This will be forwarded to determine how to harvest the fuel power wasted into the exhaust and auxiliaries. The exploration of three common recovery system will determine the operation, advantages and disadvantages of each. One will be selected that will be best integrated to the Ford 1.0 FOX engine.

Appendix 9.2 will demonstrate a portion of examples of the Excel model.

3.0 Combustion Modelling

Combustion modelling is the term given to calculated data that can be used to demonstrate the combustion process of an ignition combustion engine (ICE). The combustion model highlights the two zones of combustion and their behavior. Although the model consists of some detailed equations it is a simple model, the complexity of turbulent flame propagation modelling is near impossible in simple software packages such as Microsoft Excel. CFD or FEA are the only opportunity to accurately represent the flame demeanor, these also provided dynamic representative unlike Excel which is restricted to static graphs.

The entire model follows the cell management colour scheme to make identification of cells types easier and simple for any user, helping to ensure the spreadsheet can be used in any company in any country by anyone.

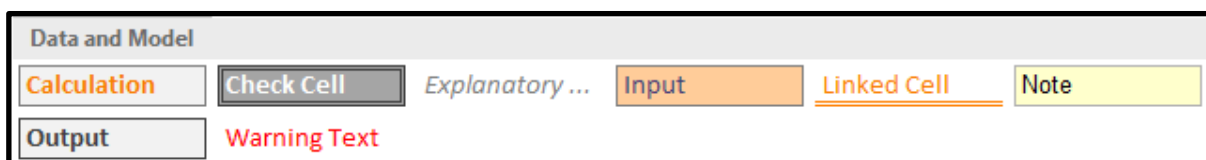


Figure 1 (Cell Management Colour Scheme)

3.1 Pressure/Volume/Crank Angle Analysis

The original data supplied by Ford Motor Company consists of crank angle (CA), cylinder pressure (P) and displacement volume (V), it is this basic data that the bases of the combustion model are built from. The pressure data is gauge pressure, and therefore 1bar above absolute.

This combustion model is aimed to actually predict the start and end of combustion (SOC, EOC), the percentage of fuel burnt and the operating temperature at each degree. Once the base model has been constructed demonstrating combustion behaviour, it allows for further predictions into emissions modelling or knock limitations.

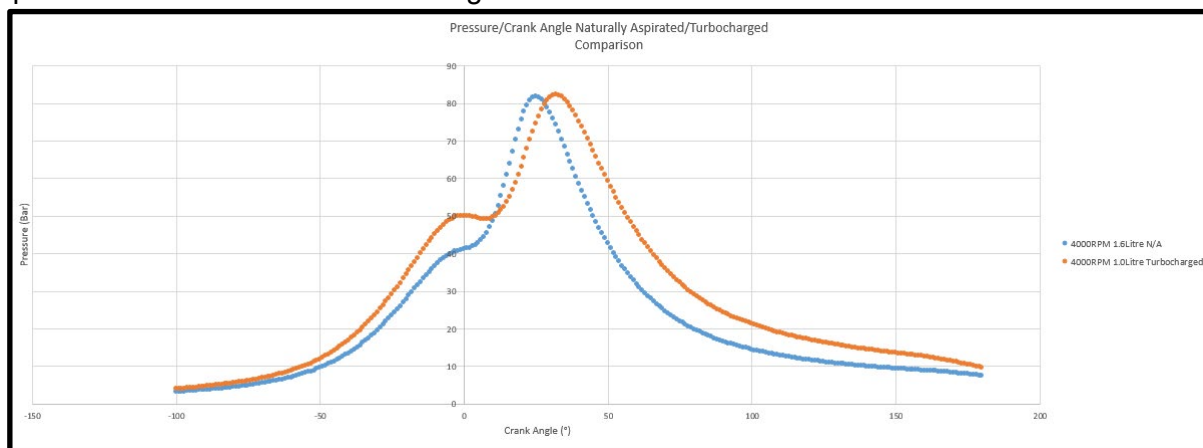


Figure 2 (P/CA Comparison TC vs N/A)

The data can be used show pressure alteration during a cycle, Figure 2 demonstrates the fundamental differences between a turbocharged (TC) and naturally aspirated (N/A) engine.

Although both average peak pressure values are almost the same $\approx 82\text{bar}$, its clear the turbocharger has a considerable affect as the 1.0l TC is producing the same pressure as the larger 1.6l N/A, however the smaller engine peaks later during the cycle. The effect of the turbocharger is clear as the pressure before combustion is much greater. The naturally aspirated engine is constantly increasing in pressure until peak, the turbocharged engine however begins to lose pressure (troughs) as the volume expands just before SOC then rising to peak. Due to the smaller capacity and an increased pressure it can be assumed the temperature is greater, this indicates the later burning is to maintain within the knock limit and assist NOx reduction. Once compressed enough gasoline can react like diesel and self-ignite, the greater the mixture, in a smaller capacity the higher the effective compression ratio (ECR), in effect the 1.0l has been calibrated to allow for the effective compression ratio to drop before combustion. If the combustion begins earlier the unburnt mixture begins to light off as the influence of the additional ECR and temperature transferring from the flame front. Although modern small capacity engines are clearly able achieve the same results as the larger N/A engines there are many trade off's that come as a consequence, NOx and knock being the predominate two.

3.2 Pressure Volume Diagram

Pressure volume diagrams represent the actual Otto cycle during operation.

The Otto cycle is the ideal representative of the combustion process that consists of adiabatic compression and expansion assuming no heat loss from the chamber, there is however thermal runaway through the cylinder walls preventing isentropic compression and expansion. During the Otto cycle an isochoric combustion is optimum as burning in a small steady state volume creates the greatest pressure increase, however during gasoline combustion this is limited by flame speed and RPM. This instant input of thermal energy is transferred ideally into 100% mechanical work through the piston assembly, realistically $\approx 30\%$ is converted, the rest is dispersed by friction loss into the surrounding cylinder block, piston and cylinder head. The most efficient heat cycle is the Carnot cycle, the pressure drop between the end of combustion

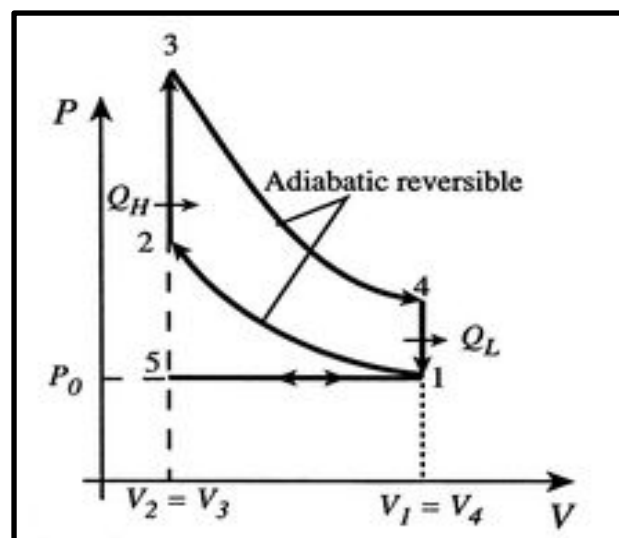


Figure 3 (Ideal Otto Cycle) (Quattrochi, 2006)

is limited by flame speed and RPM. This instant input of thermal energy is transferred ideally into 100% mechanical work through the piston assembly, realistically $\approx 30\%$ is converted, the rest is dispersed by friction loss into the surrounding cylinder block, piston and cylinder head. The most efficient heat cycle is the Carnot cycle, the pressure drop between the end of combustion

(3) and exhaust valve opening (4) indicates work done from the thermal energy, the greater the difference the higher the efficiency.

$$\text{Carnot Cycle Efficiency } C\eta = 1 - \frac{T_4}{T_3}$$

Equation 1 (Carnot Cycle)

During normal operation, the ideal cycle is never met, with the constantly varying displacement and instantaneous combustion is unachievable the energy input is distributed gradually, relative to the flame speed. During the intake and exhaust strokes, the flow restriction of the port creates pressure difference between the respective manifolds and the combustion chamber, this is called 'The Pumping Loop'.

3.2:1 Pumping Loop

During the intake stroke, a descending piston causes negative depression within the cylinder, the maximum flow rate of the port restricts the quantity of air able to enter the cylinder, therefore a vacuum is formed as cylinder pressure drops below that of the crankcase, effectively attempting to pull the piston against its direction of travel, creating negative work.

$$\text{Intake Cylinder Pressure(Pa)} = \text{Manifold Pressure (Pa)} - \text{Pressure Loss Across Port (Pa)}$$

Equation 2 (Intake Cylinder Pressure Pa)

Attempting to force a large body of burned gas through a restrictive port, and against atmospheric pressure in the exhaust forms a positive pressure build-up in the cylinder, which acts on the piston applying resistance to ascending motion forming negative work.

$$\text{Total Negative Work Done} = \text{Pumping Loop Area}$$

Equation 3 (Total Negative Work)

Smaller capacity engines suffer greatly from pumping loss as the manifold pressure is lower, this can be overcome using throttling, opening the butterfly increases manifold pressure increasing port mass flow.

$$\text{Mass Flow} = D_e \times V \times A$$

Equation 4 (Mass Flow)

Where:

D_e = Density

V = Velocity

A = Area

When referring to equation 4 the additional pressure increases the density and velocity of the air in the manifold, and in turn the mass flow through the port, with a greater manifold pressure comes an increase opportunity to maintaining pressure equilibrium. To reduce inlet pumping loss the inlet valve surface area should be as large as practical.

Part loading has the greatest effect on pumping loss as the throttle is restrictive, with recent pressure to comply with emission regulations alternate methods of throttle position maps have been developed to attempt to combat this form of negative work.

The Atkinson cycle is commonly used to assist pumping losses, the reduction in compressive stroke and forcing inlet mixture back into the manifold increases the pressure during the next inlet stroke. Using variable valve timing allows accurate control of quantity of air being fed back, although this comes with a power loss as the compression ratio is smaller than the expansion ratio, during the drive cycle the benefits are great.

Turbochargers have positive effect on pumping loss:

“The pumping work transfer will be from the cylinder gases to the piston if the exhaust stroke pressure is lower than the intake pressure, which is normally the case with highly loaded turbocharged engines.” (Heywood, 1998)

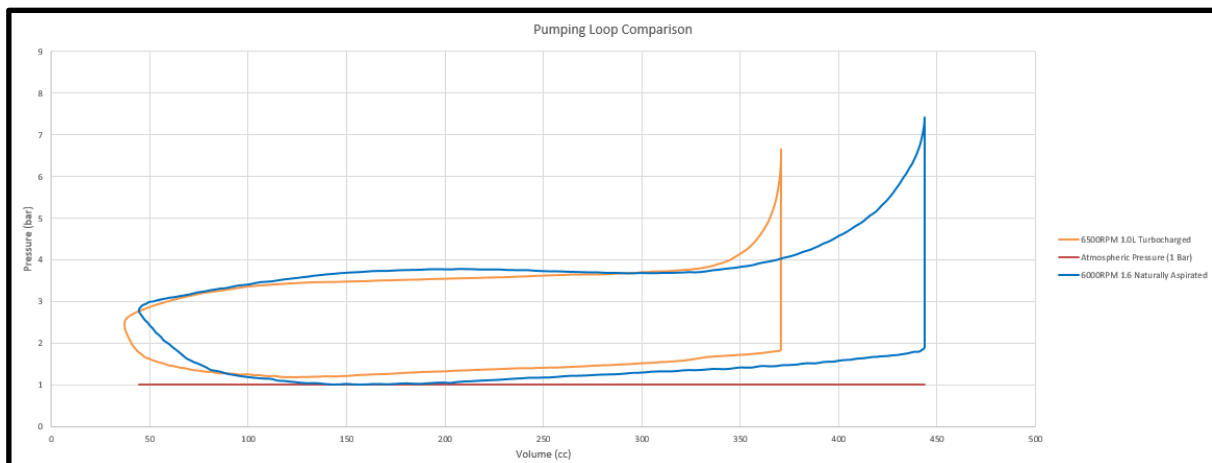


Figure 4 (N/A vs TC Pumping Loop Comparison)

When comparing the pumping loops (Figure 4) the N/A engine is much closer to atmosphere during the inlet. Although is a slightly unfair comparison due to capacity difference it demonstrates the TC engine has a reasonably linear digressive and progressive loop, whereas the N/A engine peaks and troughs slightly. The further above the atmospheric loop is a resultant positive effect of the turbo, the TC engine is also a more modern engine and will consist of more complex throttling strategies compared to the N/A.

3.3 Log P/Log V

The initial use for taking the logarithm of the pressure and volume data is to form relative values that ensure different capacity engines can be compared. The graph resembles that of the ideal Otto cycle and is therefore easier to define SOC and EOC, compared to simply plotting the pressure/crank angle diagrams.

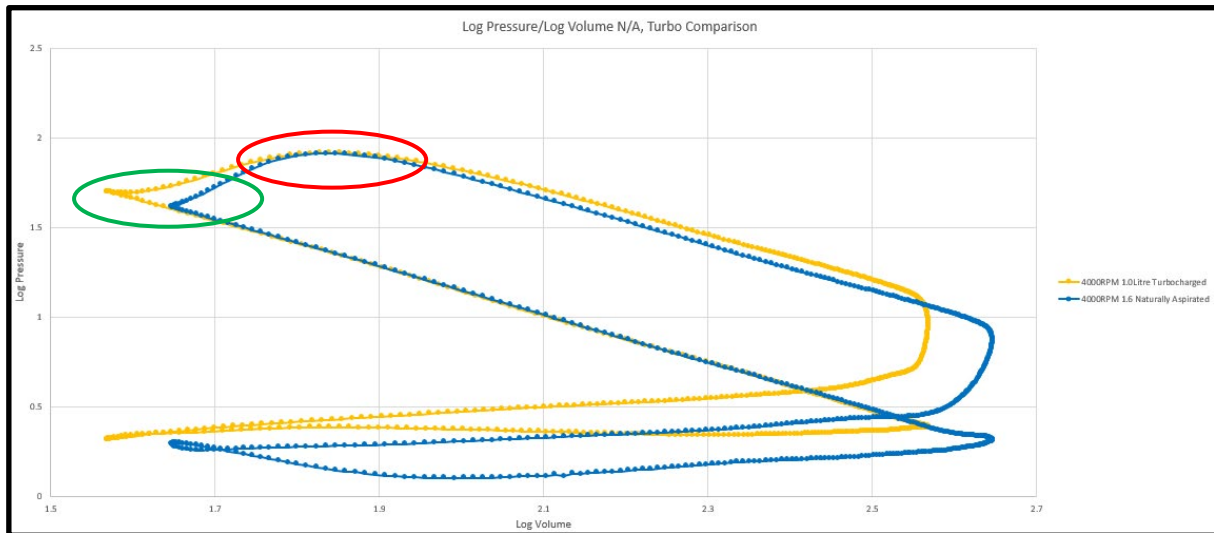


Figure 5 (LogP/V Comparison)

Using logarithms forms both the pressure and volume data into the same effective units, although its main use is to determine the polytropic index, the LogP LogV graph provides a semi accurate method to determine start (green circle) and end (red circle) of combustion.

The comparison of TC and N/A engines clearly indicates that at 4000RPM not only is the pumping loop area smaller but it is at a greater pressure, this is alternate to discussed earlier where smaller capacity engines suffer greater from pumping loss. During the intake stroke the turbocharger is maintaining the pressure further above ambient and greater than that in the crankcase, effectively pushing the piston, however the naturally aspirated doesn't have this benefit and the pressure is clearly a lot lower.

The later combustion of the 1.0l means there is more pressure when the exhaust valve opens, the mass flow is therefore greater, forcing the air through the port rather than relying on the exhaust pulses draw it burnt mixture through.

If the operating combustion process was ideal, pin pointing start and end of combustion would be simple as the graph would consist of sudden changes of direction, however it can only be narrowed to $\approx 15-25$ points as determination of the curve beginning is impossible.

3.4 Polytropic Index

The expansion and compression strokes are considered to fit a polytropic relation, in theory the pressure/volume for the two strokes shares the same curve characteristics however the pressure values are greater when expanding.

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

Equation 5 (Polytropic Index)

Where:

η = Polytropic Index

The polytropic index is deemed to be near, if not the same, as the gamma unburnt 1.3, however differs from gamma burnt due to extensive heat in the mixture.

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

Equation 6 (Polytropic Index)

Calculating the index for each degree of crank angle is used to identify the start of combustion further, however it can be difficult to define from the graph. When using tools such as Excel the values can be analysed and SOC/EOC can be confirmed against the LogP/LogV.

When the values average 1.3 (+/-0.05) before the start of combustion spike this is taken as the average η before TDC and the same average after combustion is taken as the η after TDC. However; after EOC η appears to commonly become 1.25 (+/-0.05).

There are factors that may alter the data being produced, depending on the decimal placing the software is defaulted too, it will create rounding errors. The program also assumes the crank is perfectly aligned, it is possible during cyclic varying gas forces that the crank could twist. The model however averages the pressure data of all the cylinders which allows for this anomaly, however analysing each cylinder individually would require the calculation of crank twist, if any, this is achievable assuming all other parameters are identical.

$$\text{Crank Twist } (^{\circ}) = \text{Last SOC CA } (^{\circ}) - \text{Earliest SOC CA } (^{\circ})$$

Equation 7 (Crank Twist)

3.5 P2N/Diff

The difference in pressure and η indicates the cylinder pressure if the combustion was non-existent.

$$P_{2\eta} = P_1 \times \frac{V_1^\eta}{V_2^\eta}$$

Equation 8 (Pressure Polytropic Index)

$$P_2 \text{ Difference} = \text{Pressure} - P_{2\eta}$$

Equation 9 (Combustion Pressure Difference)

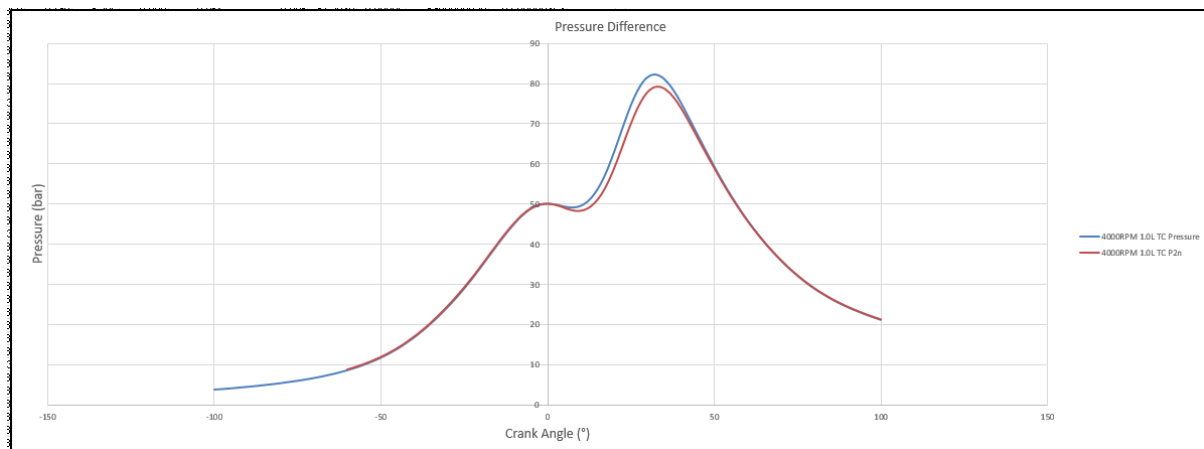


Figure 6 (Combustion Pressure Difference)

This is an accurate way to pin point SOC, when looking at crank angle pressure difference data the value becomes positive and the opposite at end on combustion. The positive value indicated the additional pressure caused by the combustion process. This however is not when the spark plug discharges. After sparking there's a time delay before the pressure begins to rise, the energy release from the initial stages of combustion is insufficient to increase pressure, this is known as ignition delay $\approx 9^\circ$ (Haywood, 1998). The propagation continues creating the pressure difference until the cylinder pressure with and without combustion equalizes.

3.6 Mass Fraction Burned

Mass fraction burned (MFB) is the term given to the percentage of fuel burnt at a given point, in the case of this model, crank angle. There are multiple methods to determine MFB, the two used for this study are polytropic index and thermal energy released from the fuel.

$$\text{Mass Fraction Burn (Polytropic)}(\%) = \left(\frac{\text{SUM}(P2\eta \text{ DiffSOC} \gg \text{Each CA } P2\eta \text{ Diff})}{\text{SUM}(P2\eta \text{ DiffSOC} \gg P2\eta \text{ DiffEND})} \right)$$

Equation 10 (Polytrophic Mass Fraction Burn)

$$\text{Mass Fraction Burn (Thermal Energy)}(\%) = \left(\frac{\text{SUM}(dQ\text{SOC} \gg \text{Each CA } dQ)}{\text{SUM}(dQ\text{SOC} \gg dQ\text{END})} \right)$$

Equation 11 (dQ Mass Fraction Burn)

Where:

$$\text{Thermal Energy Released (dQ)} = \frac{Cv \times \text{Volume} \times P2\eta \text{ Diff}}{R}$$

Equation 12 (Thermal Energy Released)

Referring to Equation 9 this method calculates MFB using the addition of pressure from combustion, although the pressure is directly related to the work done it can still rise although the most the fuel is burnt, at the earlier stages of combustion whilst the volume is still decreasing this will not accurately reflect the fuel burnt.

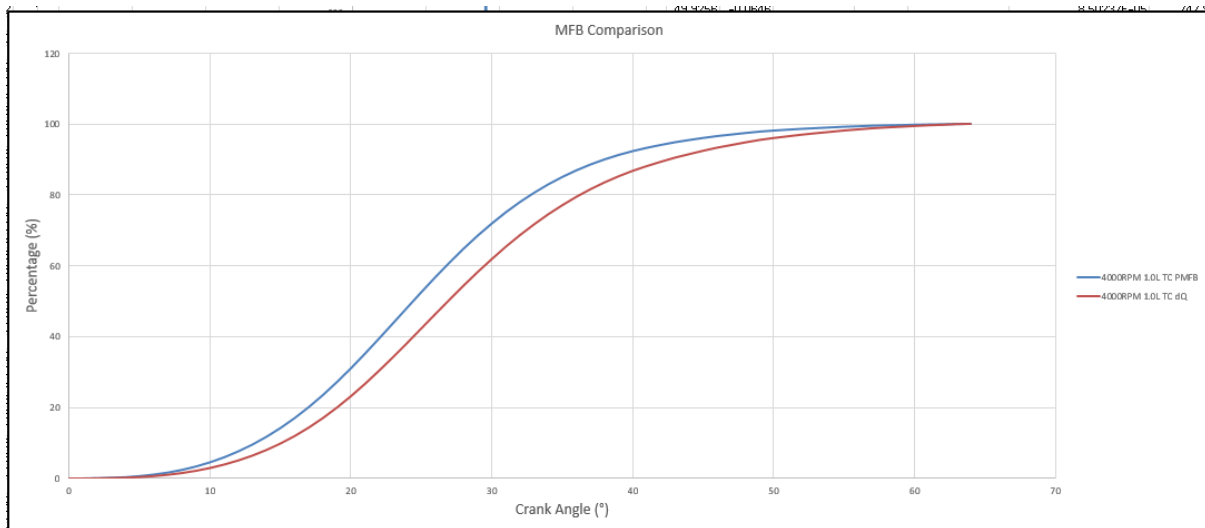


Figure 7 (MFB Method Comparison)

The thermal energy release (dQ) from the fuel is directly relevant, as the fuel begins to the dwindle the energy released will be less, therefore this method more accurate than analysing the polytropic index as it is directly related to all combustion properties.

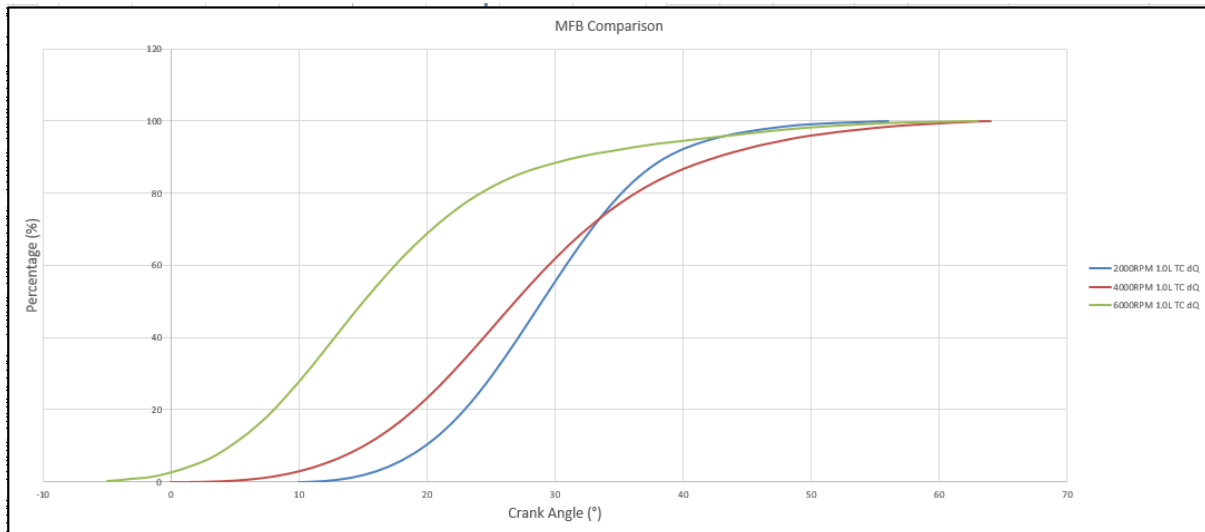


Figure 8 (MFB RPM Comparison)

FOX 1.0Litre Eco Boost – During the low RPM, combustion length is short with the majority burn between 25%-75% combustion length. As RPM rises combustion length increases, majority burn remains along the same percentage period.

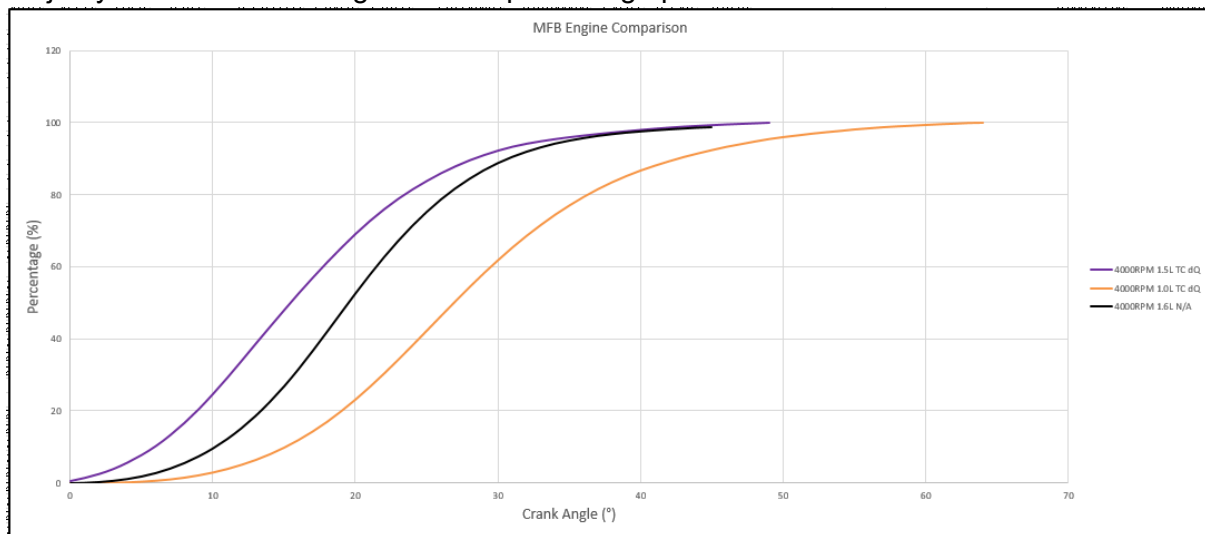


Figure 9 (MFB Engine Comparison)

From the data provided it can be seen the larger N/A has a shorter combustion period at maximum torque compared to the TC engines, however it has a shorter initial period, burning $\approx 65\%$ of the fuel in the middle of combustion. The 1.5L Eco-Boost engine burnt almost all the fuel during the first 30°, the remainder is slowly burnt over the last 24°. This indicates an effective flame speed when the global area is predominantly unburnt. The 1.0L TC FOX has extensive burn duration in comparison to the others, burning the full mixture linearly. With the

excessive pressure in a reduced capacity there is larger resistance to flame propagation, more energy is needed to ignite a compressed mixture.

3.7 Additional Calculations

Up to this point the model uses NEDC data in order to find SOC, EOC and MFB. In order to determine operating temperatures, additional data must be calculated, such as molecular mass and gas constants. These can all be determined using measured dynamometer figures.

3.7:1 Fuel Burn Chemistry

The engines investigated in this study are gasoline, although the equation for determining products after burning remains the same, the outcome varies depending on the composition of fuel and ϕ . Although the molecular construction of gasoline varies depending on the octane rating, additives and where it has been harvested. A certified average used for all modelling purpose is:

$$\text{Gasoline Composition} = C_{8.26}H_{15.5}$$

The composition of air remains the same irrespective of ϕ , the constant quantity of air forces the creation of NO_x and unburnt hydrocarbons, if the mixture is burnt fully the mixture is stoichiometric.

(Haywood, 1998)'s book defines a stoichiometric burn as:

$$C_aH_b + x(O_2 + 3.773N_2) = aCO_2 + \frac{b}{2}H_2O + 3.773(xN_2)$$

$$x = a\frac{b}{4}$$

Equation 13 (Stoichiometric Burn Products)

Where:

a = Number of Carbon atoms

b = Number of Hydrogen atoms

x = Number of parts air to burn stoichiometric

For gasoline:

$$\phi(C_{8.26}H_{15.5}) + 12.135(O_2 + 3.773N_2) = 8.26CO_2 + \frac{15.5}{2}H_2O + 3.773(12.135 \times N_2)$$

Equation 14 (Stoichiometric Burn Products for Gasoline)

Applying this formula to the fuel composition determines the amount of fuel being burnt at each RPM:

	Gasoline	H	O	+	x	O	N ₂	>>	a	b	c	x	O ₂	d
RPM	C								CO ₂	CO	H ₂ O			
	8.26	15.5	0	+	12.135	2	3.76	>>	8.26		7.75	12.135		45.63
1000	7.441414441	13.96396396	0	+	12.135	2	3.76	>>	7.441	6.982	12.135	1.203	45.63	
1250	7.441414441	13.96396396	0	+	12.135	2	3.76	>>	7.441	6.982	12.135	1.203	45.63	
1500	7.509090909	14.09090909	0	+	12.135	2	3.76	>>	7.509	7.045	12.135	1.103	45.63	
1750	8.019417476	15.04854369	0	+	12.135	2	3.76	>>	8.019	7.524	12.135	0.353	45.63	
2000	8.343434343	15.65656566	0	+	12.135	2	3.76	>>	8.343	0.245	7.828	12.135	45.63	
2250	8.343434343	15.65656566	0	+	12.135	2	3.76	>>	8.343	0.245	7.828	12.135	45.63	
2500	8.343434343	15.65656566	0	+	12.135	2	3.76	>>	8.343	0.245	7.828	12.135	45.63	
3000	8.88172043	16.66666667	0	+	12.135	2	3.76	>>	8.882	1.827	8.333	12.135	45.63	
3500	8.787234043	16.4893617	0	+	12.135	2	3.76	>>	8.787	1.549	8.245	12.135	45.63	
4000	8.515463918	15.97938144	0	+	12.135	2	3.76	>>	8.515	0.751	7.99	12.135	45.63	
4500	8.694736842	16.31578947	0	+	12.135	2	3.76	>>	8.695	1.277	8.158	12.135	45.63	
5000	8.97826087	16.84782609	0	+	12.135	2	3.76	>>	8.978	2.11	8.424	12.135	45.63	
5500	8.97826087	16.84782609	0	+	12.135	2	3.76	>>	8.978	2.11	8.424	12.135	45.63	
6000	9.177777778	17.22222222	0	+	12.135	2	3.76	>>	9.178	2.697	8.611	12.135	45.63	
6500	9.177777778	17.22222222	0	+	12.135	2	3.76	>>	9.178	2.697	8.611	12.135	45.63	

Figure 10 (RPM FOX 1.0L Products)

Its clear to see from the calculations that below stoichiometric there is excess O_2 and above there is excess CO.

This method from Heywoods book is highly effective for this computation methods, a second way to calculate 'x' is to determine the amount of O_2 once C and H have been determined, this takes greater time to achieve the identical result.

3.7:2 Total Mass of the Mixture

The total mass of the mixture is defined as the physical amount of premixed fuel/air and is dependant of RPM and ϕ , that is able to enter the cylinder. From the measured load site data provided the fuel/air ratio (AFR) and air/mass per hour (AM/h) (Kg/h) can be used for each RPM individually:

$$\text{Total Mass of the Mixture } T_{mm} \text{ (kg)} = \frac{AM/h}{n_{Cyl} \times 60 \times RPM} + \frac{AM/h}{n_{Cyl} \times 60 \times RPM \times AFR}$$

Equation 15 (Total Mass of the Mixture)

Where:

nCyl = Number of cylinders intaking during one revolution of the crank.

60 = Converts RPM to Revolutions per Second

The changes in AFR make minimal difference on the total mass as the AM/h increases greatly in proportion at full load.

3.7:3 Unburnt Gas Constant

Individual gas constants are based on the characteristics/state of the gas, as the atomic mass or number alters depending on the elements in the gas, its constantly varies. Therefore, having a different value before and after being burnt.

The gas constant for a molecular compound is the dependant on the unburnt molecular mass $\dot{m}_{ub}$.

$$\dot{m}_{ub} (Kmol/s) = \frac{(\phi \times (T_m C \times T_m H \times T_m O_2 \times T_m N_2))}{\phi + (x \times 4.76)}$$

Equation 16 (Molecular Mass Unburnt)

As with most modern small capacity engines as the RPM increases and the load demand rises, ϕ increases. With the demand of small capacity engines increasing due to frictional and fuel economy benefits, at high load overheating is an issue, therefore the calibration is over fuelling to maintain safe operating temperatures. The extra fuel therefore increases the mass of the gasoline at high RPM.

The unburnt gas constant is relationship between R and $\dot{m}_{ub}$.

$$R_{ub} = \frac{R}{\dot{m}_{ub}}$$

Equation 17 (Universal Gas Constant Unburnt)

Governing the molecular mass $\approx 30 \text{ kmol}$ it can be expected for $R_{ub} \approx 270\text{-}280 \text{ J/kg.K}$.

3.7:4 Burnt Gas Constant

As the mixture is burnt the gas state changes and therefore the constant alters, the molecular weight differs dependant on ϕ and the left-over elements O_2 or CO .

$$\dot{m}_b (Kmol/s) = \frac{(\phi \times (T_m CO_2 \times T_m H_2O \times (T_m O_2 \text{ or } T_m CO) \times T_m N_2))}{nCO_2 + nH_2O + (O_2 \text{ or } CO) + (x \times N_2)}$$

Equation 18 (Molecular Mass Burnt)

$$R_b = \frac{R}{\dot{m}_b}$$

Equation 19 (Universal Gas Constant Burnt)

Where:

T_m = Total Mass

n = Number of Molecules

Governing the molecular mass $\approx 30 \text{ kmol}$ it can be expected for $R_b \approx 250\text{--}300 \text{ J/kg.K}$.

The constant drop as ϕ increases O_2 molecularly weighs more than CO therefore the combined molecular mass is greater when burning rich.

Although CO is molecularly lighter than O_2 , more carbon monoxide is produced when rich in comparison to oxygen when lean, this increase in molecular mass and reduces the gas constant as the mixture becomes richer.

3.8 Average Temperature

The average temperature is the calculation of the temperature across both the burnt and unburnt zones creating a single zone model, before combustion this is equal to that of the unburnt mixture, during the combustion it's the assumption of the burnt, burning and unburnt section of the cylinder.

Average temperature is dependent on the molecular mass of the mixture and the burnt gas constant:

$$\text{Total Mass of the Mixture} = \text{Fuel Mass per Cyl (Kg)} + \text{Air Mass per Cyl (Kg)}$$

Equation 20 (Total Mass of the Mixture)

$$\therefore \text{Average Temp (K)} T_{av} = \frac{P_1 (\text{Pa}) \times V_1 (\text{m}^3)}{T_{mm} \times R_{ub}}$$

Equation 21 (Average Temperature)

3.9 Unburnt Temp

The beginning of the two-zone model starts from calculating the unburnt temperatures, as the flame progresses across the cylinder the unburnt zone warms the mixture above that before combustion. The greatest effect is the compressive and expanding cylinder volume and can be expected to rise for the first 1/3 of the combustion before decreasing as the volume change greatens.

$$\text{Unburnt Temp (K)} T_{ub} = T_1 \times \frac{P_2 (\text{Pa})^{\frac{\gamma-1}{\gamma}}}{P_1 (\text{Pa})}$$

Equation 22 (Unburnt Temperature)

Where:

γ = Gamma

P_1 = Previous Crank Angle Pressure

P_2 = Current Crank Angle Pressure

Gamma it the relationship between $\hat{C}_v$ and $\hat{C}_p$, it describes the state of the gas, assuming to be 1.4 as the bunt mixture is predominately air, this was used initially to build the basic model. However variable gamma was used for the final model, and is discussed in section 3.12.

3.10 Specific Heat in a Constant Pressure Unburnt (C_p)

Janaf tables provide enthalpy data for individual molecules, formulating the temperature (K) $\hat{C}_p$ (Kj/kmol.K) graph and using the polynomial allows for calculation of $\hat{C}_p$ at any given temperature.

The polynomial function for a vapour is defined as:

$$\hat{C}_p.f = Af_1 + Af_2t + Af_3t^2 + Af_4t^3 + \frac{Af_5}{t^2}$$

Equation 23 (Polynomial Vapour Function)

Table 1 (Gasoline Polynomial Function)

Fuel	Af_1	Af_2	Af_3	Af_4	Af_5	t
Gasoline	-24.078	256.63	-201.68	64.75	0.5808	T (K)/1000

This varies depending on ϕ and multiplied but 4180 to convert from cal/gmol.K >> J/kmol.K.

$$C_{p.mix} = \frac{\hat{C}_p.f \times \phi \times 4180}{\phi \times (x \times 4.76)}$$

Equation 24 (Specific Heat in a Constant Pressure Mixture)

Where:

n = Number of air molecules

Dividing by the number of moles converts: $\hat{C}_p > C_p$

Simpler models do not follow this method as the polynomial calculation steps consume too much time, however with the correct computing technics, using software such as Excel accelerates the process and assists with model accuracy. This method then allows the use of a polynomial to calculate the exact $\hat{C}_p$ no matter its temperature value.

3.11 Specific Heat in a Constant Pressure Burnt (CP)

The same method was applied to determine the Cp burnt values however the molecular composition of the burnt gas was used to calculate the $\hat{C}_p$ of the mixture. Using the same polynomial method accurately predicts the CP after the mixture is combusted.

$$Cp_b = \frac{(nMols \times \hat{C}_pMol) \text{ \#For each Molecule in the Exhaust}}{Total nMoles}$$

Equation 25 (Specific Heat in a Constant Pressure Burnt Mixture)

3.12 Gamma (γ)

Simple models assume gamma to remain the same, however it can be divulged the alternating pressure effects of a combustion differs the thermal properties of the mixture. Although Universal Gas Constant (R) goes unchanged the Cp differs with temperature and ϕ . As the temperature of the mixture increases and the specific heat in a constant volume (Cv) increases.

$$Cv = Cp_{ub} - R$$

Equation 26 (Specific Heat in a Constant Volume)

where:

$$R = 8314 \text{ (j/kmol)}$$

The constantly varying pressure and volume causes Cp and Cv to differ at each degree of crank rotation, hence varying gamma.

$$\gamma = \frac{Cp_{ub}}{Cv}$$

Equation 27 (Gamma)

Gamma is predominantly:

Table 2 (Engine Stroke Gamma Values)

Engine Stage	Gamma
Intake	≈ 1.4
Compression	≈ 1.3
Combustion	≈ 1.2
Exhaust	$\approx 1.1-1.2$

It can be derived from this table as the fuel is directly injected the mixture mass gets greater and the heat in a sealed volume increases, during combustion the temperature rises rapidly and the quantity of unburnt mixture reduces causing gamma to lower.

3.13 Volume Unburnt/Burnt

The unburnt volume is calculated using:

$$V_{ub}(cc) = \left(\frac{(1 - MFB_{dQ}) \times T_{mm} \times R_{ub} \times T_{ub}}{P_1} \right) \times 10^6$$

Equation 28 (Volume Unburnt)

Where:

$$(1 - MFB_{dQ}) = \text{Quantity of fuel remaining}$$

Subtracting the unburnt volume from the total volume at each degree derives the burnt volume.

$$V_b(cc) = V_{Total} - V_{ub}$$

Equation 29 (Volume Burnt)

3.14 Burn Temperature

The temperature of the burnt zone is practically based on the PV relationship and the amount of mixture burnt.

$$T_b = \frac{P_1 \times V_b}{MFB_{dQ} \times T_{mm} \times R_b}$$

Equation 30 (Temperature Burnt)

It is common to view a spike in the burn graph for the first few degrees:

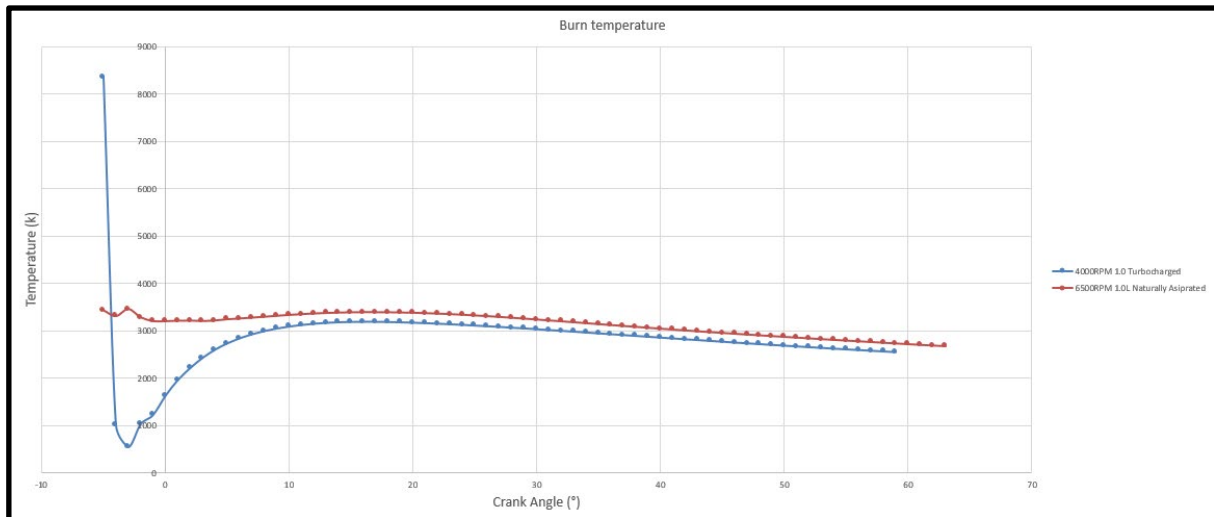


Figure 11 (Burn Temperature Example)

The initial spike in temperature which is an inaccurate function of realistic operation, it is caused by the equation used. When dividing a small number such as the volume burnt which in the beginning stage is around two decimal places, by another small number, MFB, which is also around two decimal places the result answer becomes larger. This is made apparent once MFB and V_b exceed one, the temperature then becomes its realistic value. Although the prediction of temperature is accurate during combustion the initiating temperature is undefinable.

The burn temperatures are current slightly higher than would be expected, the model doesn't include the use of EGR, this would be the next stage of the model.

3.15 Work Done (WD)

Work done for a combustion process is determined by the input of thermal energy, this is then converted to kinetic energy via a pressure increase.

The work done per degree:

$$WD/^\circ(\text{Joules}) = \frac{P_1(\text{pa}) + P_2(\text{pa})}{\Delta V}$$

Equation 31 (Work Done)

Total Work Done over the cycle:

$$\text{Total Negative Work Done} = \text{SUM}(\text{WD Compressive Stroke}) - \text{SUM}(\text{WD Exhaust})$$

Equation 32 (Total Negative Work Done)

$$\text{Total Positive Work Done} = \text{SUM}(\text{WD Power Stroke}) - \text{SUM}(\text{WD Inlet})$$

Equation 33 (Total Positive Work Done)

$$\text{Total WD per Cycle} = \text{SUM}(\text{Positive} + \text{Negative})$$

Equation 34 (Total Work Done Per Cycle)

Positive work is formed during the descending strokes when there is the addition of gravity and a force acting on the piston assembly. When the piston ascends the resistance of at minimum gravity and the compression of the working fluid through the exhaust port.

3.16 Indicated Mean Effective Pressure IMEP

“The Indicated Mean Effective Pressure (IMEP) is a very important engine parameter, giving significant information about the quality of the cycle that transforms heat into mechanical work.” (Taraza, 2000)

The indicated mean effective pressure it's the average pressure from all four strokes;

$$IMEP (bar) = \frac{\frac{Total Work Done (j)}{Total Displacement of one Cyl (m^3)}}{10^{-6}}$$

Equation 35 (Indicated Mean Effective Pressure)

Although an expensive item, cylinder piezoelectric transducer sensors are used to provide data which the ECU calculates the IMEP. Combined with predetermined knock limits, the ECU is able to predict and counter any chance of knock occurring, as modern engines use a small level of knock to attempted to reduce MFB time, this is a key method to prolonging engine life. However currently the computational time is considered slow and could allow for knock for a number of cycles.

3.17 Refining Start/End of Combustion

After analysis of MFB and burn temperature values it was determined the SOC and EOC were close but MFB values should be as minimal as possible without being zero, and the spiking initial temperatures were abnormal. The model was then extended to predict the same values that are already calculated but for every 0.1 of a crank angle. The pressure, volume graphs were plotted for each RPM, this was taken for a smaller section of the complete cycle, -20° >> 60° covered the length of the combustion for all RPM's.

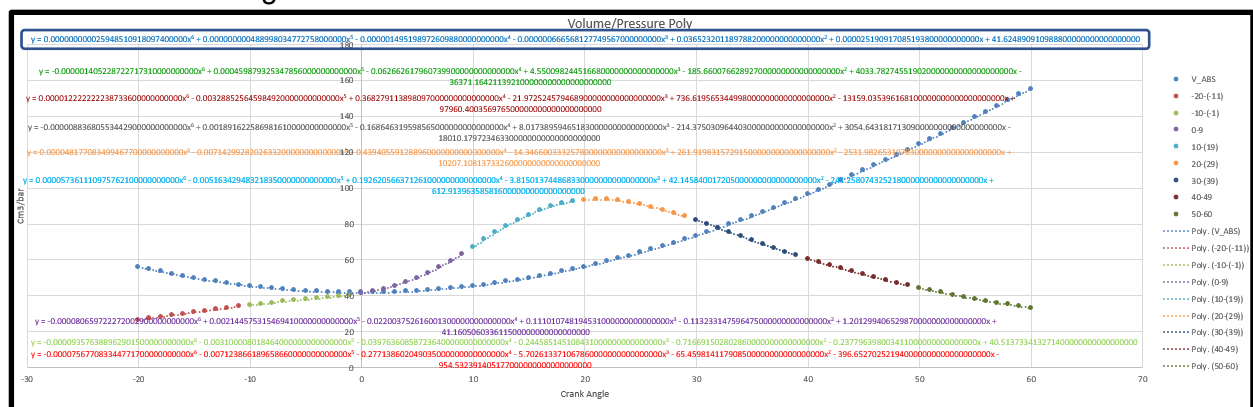


Figure 12 (Refining SOC and EOC Method)

The volume data follows a smooth curve and a sixth order trendline fits perfectly the equation highlighted in the black box, however the pressure data was an imperfect fit. The data was then broken into smaller sections taking the polynomial of each, until the sixth order was identical, dividing the data into 10° segments achieved this. Figure 12 demonstrates the section of the graph to the corresponding equation. SOC and EOC were defined with 0.1 of a degree as well as MFB was found on average to 0.02%,

The refining of SOC is important when calibrating an engine, it allows engineers to determine the ignition delay. MBT timing is therefore more achievable as the timing for peak torque can be accurately pinpointed. However, this process with Excel takes many computing hours and human resources, Excel does not provide the correct tools to automate this method effectively, to obtain the polynomials for each decimal place and to further the corresponding data will take hours by hand. With advanced MatLab techniques it may be possible to create an accurate code to automatically compute this method.

The author purposed after completion, is the refining worth the time spent. Although accuracy is key to a reliable, efficient engine, would one degree either side of SOC and EOC for a modelling purpose oppose a major issue. As most engines are calibrated to be reliable the ignition tolerances will be slightly retarded from MBT timing to reduce chance of knock. Once the engine becomes used regularly, domestic operation will alter tolerances and wear components, varying the results. If this system is automated with a purpose-built code then the refining will become easier, however the human time required for the excel process is deemed unworthy compared to the data gained.

4.0 Turbo Efficiency

There are two components of a turbo, turbine and compressor, the radial turbine converts thermal energy from exhaust gases to kinetic energy. The shaft transfers this energy from the turbine to compressor, in turn pressurising air being drawn in, increasing volumetric efficiency. Turbochargers are the fundamental way to utilize wasted exhaust energy from the high levels of wasted energy in the burnt mixture. With downsized engines, a key method for increasing power is lean burn of an intercooled mixture. This has proven a reduction in CO_2 between 5%-20%. (Alias Mohd Noor, 2014)

The quasi-adiabatic performance of a turbocharger the quantified by the thermal energy into the compressor. It requires a complex model to determine the specific rate of heat exchange during RPM variations, where mass flow, turbine and compressor in temperature the constantly variable.

4.1 Enthalpy Levels (H-H)

JANAF (Join Army Navy Air Force) tables column three $h^\circ(T) - hf^\circ(298)$ refers to the enthalpy difference from zero, this is the heat content at a given temperature. Plotting H for each molecule allowed for polynomial equations to be produced, when using load site test data from Ford this creates an identical table however with calculated enthalpy values.

$$\text{Enthalpy (H)}(KJ/Kg) = \frac{(\phi \times (nMols \times (\Delta H_{PolynoMol}))) \# \text{For each Mole in the Exhaust}}{\text{Total } nMols \times \text{Molecular Weight}}$$

Equation 36 (Enthalpy)

Pro Turbines Enthalpy H (KJ/Kg)																
Full Load	623.37377	708.70108	780.43746	884.43451	991.87644	1025.5043	1075.9427	1064.6695	1036.2324	1143.7921	1124.5248	1146.1691	1146.6904	1117.8171	1073.1745	
200					992.26063	1027.0841	1074.0808	1058.582	1029.2658							
150					973.15416	1004.2077	1051.4775	1039.3129	1008.8014							
100				874.09765	952.25971	987.81395	1030.7714	1039.1837	1059.8983	1127.6326						
170			779.56140	819.73076	932.7232	969.88565	1007.7532	1029.2438	1005.2426	1103.3166	1106.0187					
160			768.30936	805.82774	912.97824	944.96015	979.6353	1003.9974	1035.6335	1081.8323	1073.0523	1118.7117				
150			753.00769	795.48295	874.9007	915.76302	947.63666	978.6384	1009.5033	1054.233	1055.9931	1085.1253	1129.3722			
140			713.15434	745.73937	782.13358	855.47971	888.0102	914.40538	940.471	982.17164	1024.7638	1038.3732	1057.345	1077.4928	1102.7388	
130			700.91568	730.32325	768.43535	820.56568	845.57791	872.74503	903.44917	951.85083	998.65859	1008.9028	1025.5445	1042.5237	1077.7952	
120			698.19331	720.02558	743.082	799.09086	819.92585	838.77064	868.38453	928.82039	980.386	988.45442	987.21687	1020.32	1028.0023	1070.4883
110		623.2519	633.65051	714.94509	724.64889	772.85379	801.38842	819.79754	850.85491	904.4525	958.2304	987.1576	960.29258	931.56581	1010.554	1043.0747
100		588.38414	656.43506	684.42744	714.18584	757.87263	786.08649	806.11365	831.98831	884.10272	922.52882	944.63372	947.04193	977.75997	998.45087	1023.282
90		588.51788	649.63717	667.93601	703.24382	748.8755	779.5236	793.86852	819.3145	862.54505	904.11524	927.94564	929.76255	955.44196	979.31065	1021.3632
80		555.30252	624.34884	655.23108	692.56363	723.97809	756.98502	781.63017	809.79358	845.59957	895.24001	911.29519	918.68331	950.34991	958.63682	994.7647
70		533.56217	600.95378	641.08308	674.98857	704.08385	736.50044	762.56833	783.18387	825.67305	877.33662	895.19337	909.23261	929.50874	948.01492	978.26197
60		508.09507	570.61059	609.60899	659.54588	675.86499	709.49194	742.5472	768.11633	806.18232	860.10894	878.10885	895.07307	917.33829	934.93099	960.90892
50		485.84226	540.51882	581.09489	608.36171	642.92696	673.11302	705.34111	742.29898	787.62559	840.05793	859.32244	879.55522	896.20985	918.64469	943.59562
40	421.56408	465.58108	512.98728	549.36095	573.64621	603.99759	644.7204	673.98951	720.41408	768.24817	816.25219	841.65636	866.83951	893.64426	905.82445	930.72183
30	397.26352	433.99767	484.77379	523.20147	552.07822	579.71339	621.44506	652.38988	693.32278	743.70542	794.17731	824.45108	851.63429	883.08474	892.90173	918.64469
20	338.46717	391.55782	445.52243	502.05994	537.46759	574.05689	602.51824	633.60395	669.90326	722.60421	776.34345	816.62181	840.84948	863.56886	887.00219	896.03615
10	319.1989	387.5917	448.76257	477.92261	514.84924	548.06752	588.9374	607.71391	643.53292	698.49	765.86852	796.96317	822.82701	831.83169	836.84629	868.50627
5	244.37878	314.82434	419.8072	470.21055	525.90919	592.51992	608.81797	622.06477	672.86327	705.11583	762.71879	780.75152	806.72412	809.71703	805.83506	843.1993
2	231.5358	304.12212	403.44316	414.29836	480.15889	585.00471	579.22144	620.08193	699.34608	741.09487	763.09491	777.48898				
0		800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000

Figure 13 (Enthalpy Load Site Table Comparison)

From these tables the formation of enthalpy work at a given point in the exhaust can be defined. As the burnt mixture flows through the exhaust thermal energy will be dissipated through the material and into atmosphere, this is wasted energy. The greater the enthalpy loss across or between components the greater the work done. Therefore, placing the turbo as near to the manifold as possible will increase the enthalpy entering the turbine as the enthalpy in, is as high as possible, there is then the possibility for increased thermal work done. If the turbo is mounted further from the manifold a greater amount of enthalpy will have already dissipated.

The higher the temperature the quicker it is attempting to reach ambient, assuming no addition heat input.

4.2 Exhaust Gas Flow

Work done by the gas is dependent on the mass flow speed, the exhaust mass flow is relative to the proportion of air flow into the cylinder plus the fuel injected during the cycle. As the quantity of fuel rises the air must increase in order to maintain the desired AFR.

$$\text{Exhaust Mass Flow } E_{mf} \text{ (kg/sec)} = \frac{F_{mf}/hr + (\phi \times F_{mf}/hr)}{3600}$$

Figure 14 (Exhaust Mass Flow)

Where:

$$F_{mf}/hr = \text{Fuel Mass (Kg/hr)}$$

$$3600 = \text{Conversion from Kg/hr} > \text{Kg/sec}$$

4.3 Turbine - Work Out

From the load site temperature data maps, matching tables can be calculated to determine the work done. The enthalpy difference (ΔH) across the turbine determines work out as:

$$\Delta H = TuH_{in} - TuH_{out}$$

Equation 37 (Enthalpy Difference Turbine)

$$\therefore \text{Work Out of the Turbine (Kw)} = \dot{m}_{fl} \times \Delta H$$

Equation 38 (Work Out of the Turbine)

The turbine converts the thermal flow energy to kinetic energy along the shaft. As RPM rises, mass flow increases and produces more work. If a propeller is imagined, the harder the user blows the faster it will spin, the exhaust gas has the same effect on the turbine. This increase in velocity greatly increases the kinetic energy produced when:

$$KE = \frac{1}{2}mV^2$$

Equation 39 (Kinetic Energy)

RPM has a direct effect on mass flow, thus greater the RPM increase the work from the turbine.

Work out of the Turbine (Kwatts)																
Full Load	0.9601972	1.5613011	2.2704021	3.8437949	6.6849254	7.5615954	8.7378814	10.243236	11.579366	12.866445	12.661638	15.665279	16.454129	16.237883	12.58236	
200					6.6855226	7.3619313	8.1177018	9.8993075	11.59681							
190					5.9430535	6.2852228	7.4382378	9.2369677	10.651409							
180				3.5434152	5.1193587	5.4700334	6.5220605	7.5648534	9.2563854	11.703335						
170			2.3052015	2.3535354	4.3572872	4.6500012	5.4414865	6.1973537	7.54592	9.4738095	11.409744					
160			2.9170909	2.9257526	2.7629326	3.9304944	4.4895522	5.1636647	6.942958	7.652332	9.3661839	12.229408				
150			1.6629731	1.7824727	12.533699	3.1008134	3.437004	3.9052008	4.9436259	6.1114429	7.4212504	9.8122217	13.741115			
140		1.5405263	1.4409066	1.5002908	2.0577215	2.2901778	2.627265	3.8844141	3.988626	4.7267311	5.5683062	6.9982134	9.6332177	13.700421		
130		1.2643736	1.2579162	1.2387425	1.578312	1.7943242	2.0302715	2.4610161	2.9486289	3.7191839	4.2630954	5.0836202	7.2617455	10.637707		
120		1.1240632	1.2069963	1.971969	1.4317157	1.7325406	1.9928297	2.6234335	2.4423493	2.961796	3.1973312	3.7117035	5.0634046	7.4355939	10.427913	
110	1.0082101	1.0537753	1.1183841	1.1414041	1.3835336	1.6393638	1.6391148	1.7687357	2.0219382	2.4236916	2.5160989	2.8364663	3.5392705	5.8450818	7.5746229	
100	0.8381027	1.1423423	1.0932236	1.0680045	1.2362428	1.4518088	1.4361082	1.6948177	1.3452854	1.3725096	1.3612444	2.2812531	2.664924	3.5392538	5.8245607	
90	0.6461135	1.0005987	0.7091009	0.8522517	0.9771307	1.0588458	1.0430681	0.9951447	1.2498879	1.4575573	1.5384891	1.6393932	1.6865961	2.4553034	3.3458189	
80	0.4301141	0.7029749	0.4852384	0.5675164	0.6250637	0.6932137	0.9366562	1.2511747	1.104247	1.4322878	1.6415317	1.6174714	1.5402715	1.7369915	2.1657369	
70	0.3246721	0.4175341	0.3826331	0.396504	0.5129699	0.7880294	0.8880906	0.8709435	1.1516066	1.4133368	1.6100058	1.6809893	1.6802954	1.7331219	1.9941225	
60	0.3953452	0.456583	0.3818054	-0.1376188	0.543383	0.7358031	0.7638302	0.8982773	1.1100813	1.3583251	1.5532328	1.6959985	1.7975646	1.8330557	2.1353858	
50	0.4430982	0.4834022	0.4368039	0.4319513	0.566671	0.719537	0.7381329	0.8687632	1.1036879	1.3564447	1.5411194	1.6942093	1.824703	1.9626416	2.2683465	
40	0.5260329	0.5777462	0.6189002	0.4897609	0.4871445	0.6084038	0.766781	0.7874083	0.925424	1.1157671	1.3646069	1.567278	1.7247223	1.9416947	1.9589046	2.2701178
30	0.5057509	0.5363191	0.6084531	0.5946108	0.5950825	0.7295969	0.8787881	0.9388589	1.0490437	1.1764065	1.451208	1.6108999	1.7056583	1.9142648	1.9708783	2.2413908
20	0.3553159	0.4354733	0.5642192	0.6635691	0.7413975	0.8970218	0.973612	1.054339	1.1814334	1.2856429	1.5334168	1.7291446	1.8843961	1.9530647	1.9452714	2.1420946
10	0.277908	0.3349032	0.4976121	0.5213106	0.5794993	0.6509223	0.7160398	0.784739	0.9380377	1.1570261	1.4501645	1.5339164	1.6725443	1.66836	1.6326121	1.8488432
5	0.1424002	0.2104103	0.3529775	0.4454575	0.561395	0.7645636	0.8022436	0.8819369	0.920917	1.0639063	1.3100715	1.4091669	1.7006348	1.3774435	1.4800623	1.3589575
2	0.0556241	0.1811074	0.2407565	0.2841021	0.3765903	0.6623714	0.6223385	0.7756523	0.9362508	1.0786748	1.270528	1.3476305				
0		800	1600	2400	3200	4000	4800	5600	6400	7200	8000	8800	9600	10400	11200	12000

Figure 15 (Work Out the Turbine)

Peaking at 5500RPM, the work out is equal to 16.45Kw, a logical assumption for the work out drop after this is due to turbulent intake flow. When mass flow reaches the saturation point governed by the port diameter, the mixture becomes disrupted and laminar flow is lost.

4.4 Compressor Work In

The compressor enthalpy is the opposite to the turbine, as the work is used to draw air in, unlike the turbine which forces air out. The difference in enthalpy is then:

$$\Delta H = CompH_{out} - CompH_{in}$$

Equation 40 (Enthalpy Difference Compressor)

$$\therefore \text{Work Out of the Compressor (Kw)} = \dot{m}_{Fl} \times \Delta H$$

Equation 41 (Work Out the Compressor)

Subtracting enthalpy out from enthalpy in will leave a positive value, it is to be expected as compressing the air will increase the temperature.

4.5 Compressor Efficiency

The greater the compressor efficiency the greater the increase in energy in the air, it can be assumed as the exhaust mass flow increases and the speed of turbine increases overall compressor performance.

With the data provided it is possible to determine the efficiency using the pre and post turbo temperatures.

$$\text{Compressor Efficiency (\%)} = \frac{T_i - CompT_{Pre}}{CompT_{Post} - CompT_{Pre}}$$

Equation 42 (Compressor Efficiency)

5.0 Engine Efficiency (E_{Eff})

When analysing the data provided by Ford it showed the temperature at the manifold was measured lower than the temperature at the turbo, logical assumptions indicate this is incorrect as there is no thermal input between the two. The initial hypothesized reasoning would indicate the sensor was faulty. However, it can be estimated a company such as Ford, calibrate and test all sensors before beginning the test. Therefore, it can be concluded that the positioning of the sensor isn't ideal, depending on the sensor type it may be monitoring the temperature of the manifold material rather than the exhaust gases. The ideal positioning would be at the point the manifold becomes the exhaust pipe, at this point the air has begun to settle after being forced from the port and the temperature will be much more stable and realistic.

This anomaly manifold data prevents the opportunity to calculate the actual recovery potential between the manifold and turbine, this becomes an issue when analysing TEG (Thermal Electric Generator) manifold potential. However, for purpose of exercise the study will continue to use the data to demonstrate the potential energy distribution.

5.1 Mechanical/Engine Efficiency

The E_{Eff} can be determined as the power produced by the engine in comparison to the energy in the fuel. The power table provided by Ford is measured in PS the metric unit of horse power. The power of the fuel is dependant on the quality and species of fuel, various fuels have different calorific values.

$$\therefore \text{Fuel Power for Gasoline (watts)} = \left(\frac{(\text{FuelFlow/hr})}{3600} \right) \times 44 \times 10^6$$

Equation 45 (Fuel Power)

$$\therefore E_{Eff} (\%) = \frac{\text{Engine Power (ps)} \times 735.499}{\text{Fuel Power (w)}}$$

Equation 46 (Engine Efficiency)

Where:

44×10^6 = Calorific Value of Gasoline

735.499 = Conversion from ps > watts

On average, it is apparent only $\approx 28\%$ of the fuel power is converted to mechanical power. The remaining 72% is wasted. This can be confirmed when comparing the calculated table to the power table, multiplying the percentage at a randomly selected load site by the fuel power will determine the theoretical power, if correct the values should match.

Table 3 (Engine Efficiency Validation)

Load Site	NEDC Data Power (Kwatts)	Calculated Answer (Kwatts)
60%_1000RPM	6.25	6.21
40%_2000RPM	8.38	8.382
160%_3500RPM	58.69	59.803
20%_6000RPM	12.65	12.647

From this validation, the engine efficiency model can be deemed as highly accurate.

A portion of the remaining energy is released into the exhaust, this energy lost into the exhaust is determined by the energy in the exhaust manifold in comparison to the fuel power.

$$Energy\ Loss\ Exhaust\ (\%) = \left(\frac{Exhaust\ Manifold\ Energy\ (J/s)}{Fuel\ Power\ (Watts)} \right)$$

Equation 47 (Energy Loss into the Exhaust)

Where:

$$\text{Exhaust Manifold Enegy (j/s)} = E_{mf} \times \text{Manifold Enthaply (j/Kg)}$$

Equation 48 (Exhaust Manifold Energy)

Therefore; the remaining energy is lost into water, oil, friction and auxiliaries:

$$\text{Remaining Energy Loss (\%)} = 1 - E_{eff} - Ex_{loss}$$

Equation 49 (Remaining Energy Loss)

This theory is reinforced using the IMEP data collected, the pressure required to overcome the frictional resistance:

$$\text{Friction Mean Effective Pressure (FMEP)} (Kn/m^3) = \text{IMEP}(Kn/m^3) - \text{BMEP}(Kn/m^3)$$

Equation 50 (Friction Mean Effective Pressure)

Where:

BMEP = is taken from the tablated data from Ford.

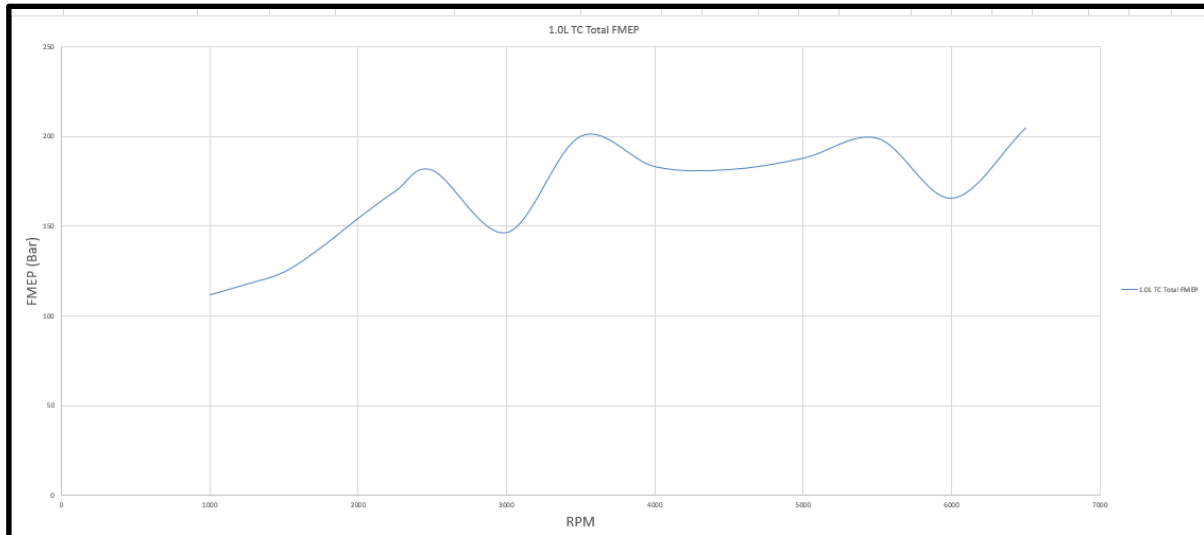


Figure 23 (FMEP RPM)

Figure 23 demonstrates the FMEP for the FOX at full load, reinforcing the escalation in friction as RPM rises.

The Ford data also provides the oil gallery temperatures (Figure 24), this demonstrates as the energy losses increase from low rpm/load to mid rpm/load the oil temperature rises, as the losses decrease to high rpm/load the oil temperature corresponds.

GALL. OIL deg C	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		89	91	90	92	95	98	100	105	108	115	120	120	120	119	117
200						95	97	100	105	110						
190						95	97	100	105	110						
180					92	95	98	100	105	110	115					
170				90	92	95	98	100	105	110	115	121				
160				90	92	95	98	100	105	110	115	120	120			
150				90	92	95	97	100	106	108	113	118	118	120		
140			91	92	93	95	98	100	105	106	110	115	115	121	121	
130			90	91	91	93	95	98	103	103	108	113	113	118	118	
120			90	90	92	92	93	95	100	101	106	111	110	116	116	115
110		90	89	90	91	91	92	92	98	98	103	108	108	113	113	113
100		89	89	90	90	91	91	92	96	96	101	106	105	111	110	110
90		90	90	90	90	90	91	92	93	93	98	103	104	108	108	109
80		90	89	90	90	90	90	91	89	89	96	100	100	106	105	107
70		90	90	89	90	90	90	90	91	91	93	98	98	103	103	106
60		90	89	90	89	90	90	89	91	91	90	95	95	100	101	105
50		90	89	90	90	89	90	89	90	90	90	93	93	98	98	104
40	89	90	89	90	90	90	89	90	90	90	90	90	90	96	96	104
30	89	90	89	90	90	90	90	90	90	89	90	90	90	97	95	104
20	89	89	89	89	89	89	90	90	89	89	90	90	90	98	94	104
10	89	89	90	89	89	89	89	89	90	89	89	89	90	101	94	106
5	88	89	90	89	89	89	89	89	90	89	89	89	89	106	94	111
2	88	89	89	89	89	89	89	89	88	89	90	90	90			

Figure 24 (Oil Gallery Temperature)

Complying the data into a comparable graph demonstrates the predominate quantity of energy loss is into the oil water and auxiliaries.

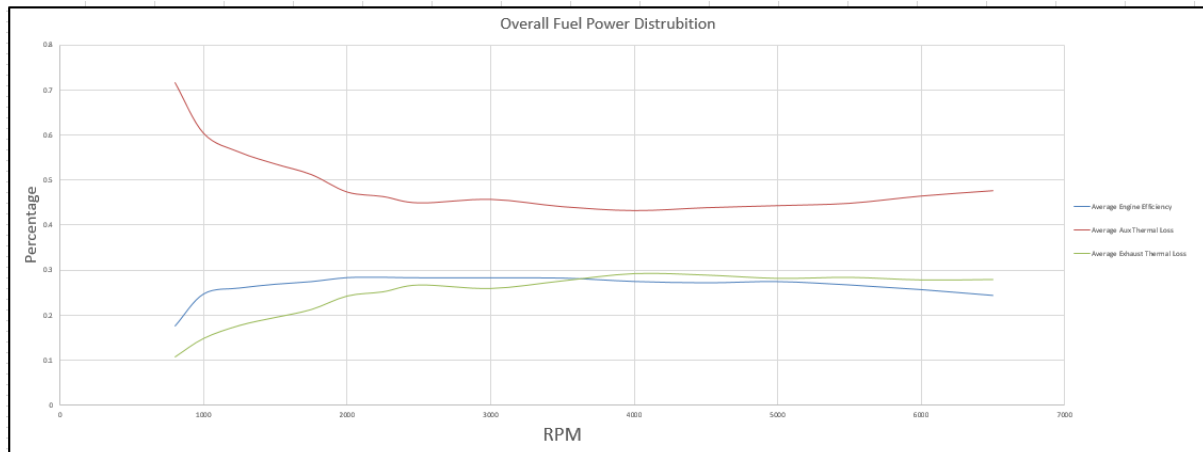


Figure 25 (Overall Fuel Power Distribution)

At 3500RPM the thermal loss into the exhaust exceeds that of the energy converted to work during the combustion. It is believed with the correct manifold data the losses into the exhaust will exceed that of the converted to mechanical work.

5.1 Calculation of Manifold Temperature

From the combustion model it is possible to approximate the mixture temperature when leaving cylinder. Analysing crank angle, the pressure graphs Figure 26 can approximate the point the mixture is vented from the cylinder. Logically it can be assumed the valve will open around $\approx 180^\circ$ ATDC or just before the exhaust stroke begins. The additional dip (red circle) in the data trace can then be matched to the average temperature calculations to provide an estimated exiting mixture temperature.

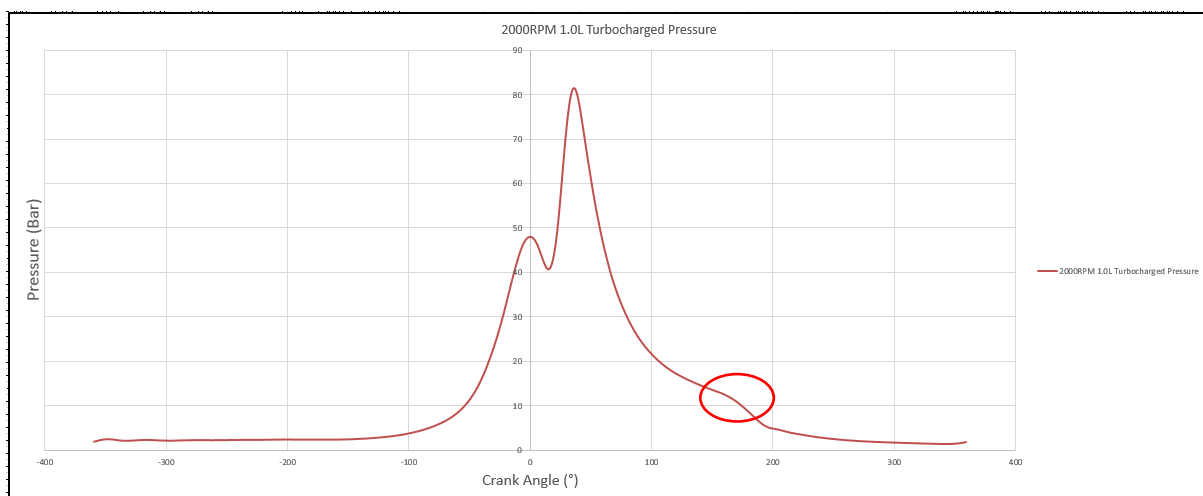


Figure 26 (P/CA Graph)

With these calculated manifold temperatures, at full load for each RPM, the fuel power distribution is as seen below:

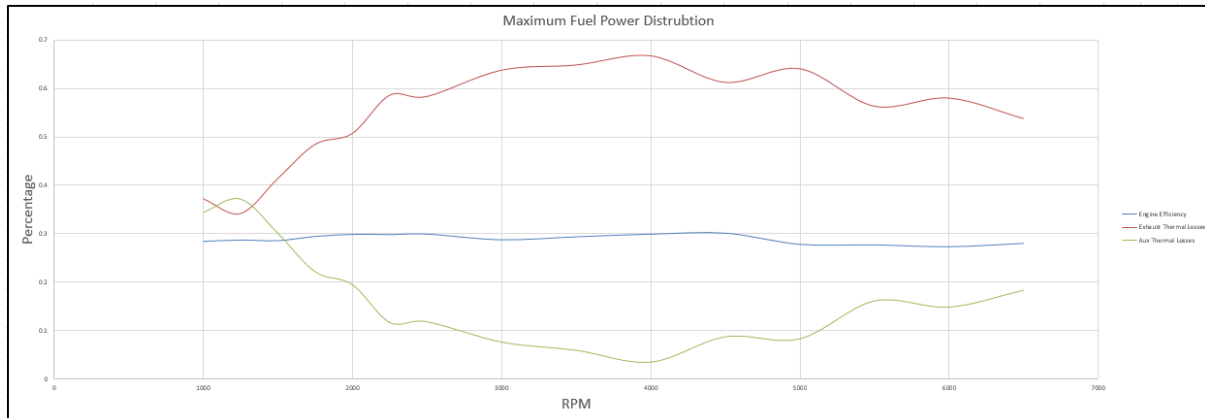


Figure 27 (Maximum Fuel Power Distribution)

At full load with the correct calculated data, the engine efficiency (blue trace) remains linear such as before, however the losses into the exhaust are much greater (red trace). The thermal runaway into the auxiliaries is less than 20% above 2000RPM, and the energy loss into the exhaust is much greater, as the charge in cylinder time get less as the RPM increases, providing less time for the water/oil to absorb the heat. As the water and oil temperature is maintained, the working fluids are unable to withdraw further energy, forcing the remainder into the exhaust.

Although seeming unlikely it can assumed this is correct, the piston assembly can transfer no more than $\approx 33\%$ of thermal energy no matter the combustion or engine parameters, however the energy in the exhaust is greatly increased by mass flow which is quickest at full load. When referring back to Figure 24 it can be seen as the oil temp rises, the auxilliary losses begin to rise accordingly, proving the theory correct. This shows there is great potential for energy recovery at full load.

The next stages of this model would be to analysis ϕ table deeply to accurately calculate the correct exhaust composition for each load site, this model assumed the entirety of each RPM to be the same as full load to save computing time.

6.0 Exhaust Energy Regeneration

Modern IC engines are now verging $\approx 35\%$ efficient and $\approx 25\%$ - 30% is dissipated into coolant and oil, therefore, the rest of the energy goes unutilised. It is possible to recover thermal energy if the exhaust gas is hotter than that of the surrounding environment, the JANAF tables compare the enthalpy values to that of the molecule at 298K (25°C), this is not the absolute enthalpy of the exhaust but the energy that can be recovered. Harvesting this, provides an additional power source which can be used to increase overall engine efficiency.

6.1 Types of Recovery System

6.1:1 Mechanical Turbo-Compound Unit (MT-CU)

For a conventional ICE, the addition of electrical power is unnecessary. Therefore; after the primary turbo, the exhaust gas is fed to a secondary turbine mounted between the gear system and the primary drive shaft. Acting like a mild hybrid motor, this works to reduce the torque needed to rotate the driveshaft. Caterpillar and Scania have applied this system to their large capacity engines, they claim gains $\approx 5\%$ - 6% BSFC at full load and $\approx 3\%$ at part load however during idle and low loading there is a reduction, hence lowering the overall BSFC. The lower gas flow speed at low RPM hinders the transmission efficiency as there is additional friction from bearing and turbine weight, the engine is also having to rotate the turbine to drive the shaft as the gas velocity is too lower to spin it itself. It can be assumed at lower RPM a MT-CU would negatively affect pumping loss however an electrically driven turbine would cause positive effect. (Alias Mohd Noor, 2014)

It has been reported that an axle driven turbine caused Caterpillar a 4.7% BSFC loss during a 50,000mile extra urban test, however Cummins used a radial flow turbine and achieve an increase of 3%-6%. (Alias Mohd Noor, 2014)

A large issue with the secondary turbine is the increased back pressure, although this may benefit in some cases it is another issue that needs to be calibrated and controlled. The author assumed the second turbine must flow at least an identical quantity of air as the primary turbo in order to relinquish the opportunity for “backlog”. If the air does not pass through the second turbine with at least an identical velocity as the primary turbo, the pressure will build and prevent the turbo from functioning effectively. If the same load site trend as shown in section 4.6 for the turbo is matched for the MT-CU then it can be understood it will be only $\approx 20\%$ efficient during the drive cycle, although it will slightly beneficial the gain may be minimal. The author purposes a valve system (like a TC wastegate) that can be used to control the pressure into the secondary turbine, which could be used to release short bursts of high pressure when

needed such as during sudden acceleration, or even to restrict the air flow to increase the resistance from the MT-CU assisting slowing the vehicle.

6.1:2 Rankine Cycle

A Rankine steam engine is a common method to recovery heat energy, consisting of an evaporator, expander, condenser and a pump. The recovery system has a working fluid that heats and cools, there are liquid chemical substitutes that possess enhanced properties at low temperatures forming an organic system (ORC) such as ammonia or variations of hydrochlorofluorocarbons (HFC's), fluorinated alkane fluids with low boiling properties (Kandathil, 2011). The evaporator surrounds the proportion of exhaust that the energy will be extracted. As the working fluid passes through the evaporator it absorbs the thermal energy and is superheated quickly vaporizing it, the greater initial temperature of the vapour the lower the opportunity for the formation of saturated steam. It's important for the steam to be superheated as to remain in a vapour state whilst transferring to the expander. Maximum efficiency is achieved with a large evaporator, positioning this along a tailpipe section may alter the floor plan of the vehicle reducing internal space.

The evaporating pressure makes minimal effect, previous studies have shown that evaporator efficiency is dependent on the mass flow rate of the working fluid, hence the need of the pump in the system.

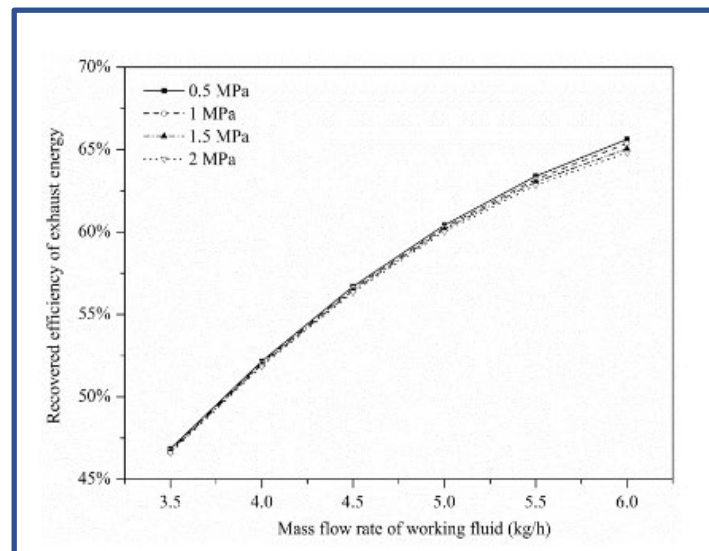


Figure 28 (Recovery Efficiency vs Mass Flow (Tianyou Wang))

When this is the case work into the expander:

$$W_{out_{exh}} \times \eta_{eva} = W_{in_{exp}}$$

Equation 51 Work Out Exhaust)

Where:

$W_{out_{exh}}$ = Work out of the evaporator.

η_{eva} = Evaporator efficiency

$W_{in_{exp}}$ = Work into the expander

The turbine in the expander is spun by the high-pressured steam and creates kinetic energy, commonly in hybrid vehicles an alternator is fixed to the turbine and used to charge the batteries, extending range and helping to maintain the lithium-ion batteries within the 20%-80% capacity range that optimises their life length. A condenser, either air or water cooled, then condenses the steam back to liquid which is then pumped back into the evaporator and the cycle is started again.

BMW reported in 2009 that their “Turbosteamer” salvaged enough thermal waste from the radiator and exhaust system which through a turbine powered by a steam piston, transmitted a further 10Kw and 20Nm or torque back to the crankshaft during peak operation. This addition assistant gave ≈15% more fuel economy. (Alias Mohd Noor, 2014)

6.1:3 Thermoelectric Generator (TEG)

TEG's consist of a thermoelectric element that is sandwiched between two heat absorbers, thermal energy is absorbed one side, passed through the element and expelled the alternate side. This is known as the Seebeck Effect;

“If heat is applied to a circuit at the junction of two different conductors, a current will be generated.” (Alias Mohd Noor, 2014)

In figure 29 type 1 and 2 represent two semiconductor materials, where they connect there is a temperature, T_1 and T_2 . The voltage generated is dependent on ΔT the larger the drop the higher the voltage formed.

$$V = \alpha \times \Delta T$$

Equation 52 (Seebeck Voltage)

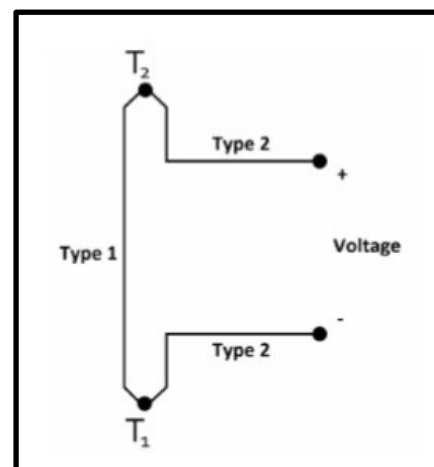


Figure 29 (Seebeck Effect)

Where:

α = Seebeck Coefficient

$\Delta T = T_2 - T_1$

The Seebeck Coefficient is defined as voltage per unit of temperature and alters depending on the characteristics of an electrical field. It is therefore ideal to place the system where the T_1 has the large cooling effect. Although placing in the manifold may provide the largest T_2 , T_1 will have minimal passive cooling potential.

Thermal merit of the setup is determined:

$$ZT = \frac{S^2 T}{k\rho}$$

Equation 53 (Thermal Merit)

Where:

Z = The material thermoelectric of merit

S = Thermopower

T = Absolute Temperature

k = Thermal Conductivity

ρ = Electric Resistance

Z is a dimensionless coefficient of thermal spreading, it is the term given to the ability to uniformly distribute heat along its surface. Materials with larger merit properties assist the electron formation, the thermal management of the material can hinder the effectiveness of the TEG if the heat does not progress through the system quickly. If the Z is low the distribution along the surface will be less uniform creating hot and cold spots, this decreases longevity and reliability.

BMW tested this method changing the position of the material to attempt to improve its effectiveness, the two predominate positions are built in the EGR and incorporated in the exhaust manifold. Both areas resulted in $\approx 2\%$ less fuel consumption and $\approx 5\%$ improvement in BSFC.

The combination of TEG with other components such as maximum power point trackers (MPPT) and Cuk converters provides the maximum recovery possible from the TEG system. These form of voltage enhancers increase the effectiveness of the TEG output.

(Zhang, Chau, Chan, & Gao, 2010) Found that combining TEG with photovoltaic (PV) systems that an MPPT could help produce $\approx 7.5\%$ - 9.4% addition power when the TEG is heated between 100°C - 250°C . During the drive cycle the FOX 1.0litre exceeds $\approx 450^\circ\text{C}$ if the data from the above discussions are correct the potential for addition power using this method of recovery for the Ford engine is great.

6.2 Selection of Recovery System

Below is the table consisting of benefits and disadvantages to each of the discussed systems.

Table 4 (Recovery System Comparison)

Properties	Secondary Turbine	Rankine Cycle	Thermoelectric Generator
Mass	Disadvantage	Disadvantage	Advantage
Packaging	Disadvantage	Disadvantage	Advantage
Price	Moderately Expensive	Moderately Expensive	Expensive
Recovery Efficiency	Low	High	High
Current Development	Modern	Small	Large
Maintenance	Low	High	Low

There are many points in an exhaust system where thermal energy is needed to complete an operation such as turbines and catalyst. Although ideally if the enthalpy drop across the turbine was 100% the turbo would be most efficiency and highly benefit the performance, it is important to maintain temperature to ensure catalyst function. After consistent testing, positioning for a recovery system can be selected, it is important to place it after the turbo in order to transpose the maximum heat through the turbine as possible, from there depending on thermal loss down the remainder of the exhaust will determine whether before or after catalyst will be optimum. From a first principles concept placing before the catalyst could help to reduce NOx tail pipe emissions, as temperature lowers, greater passive NOx dissociation occurs, within the catalyst, therefore it can be said a recovery system will help to reduce tailpipe NOx emitted. However; this may alter the functioning temperature of the catalysts and therefore alternate HC and CO that will vent to atmosphere, it is these trade off's that need to be analysed.

Packaging may also prove a decisive factor; the recovery system consists of many large compounds. With the engine bay already housing a larger quantity of components it may prove space effective to fit around the rear of the vehicle, however if there is an alternator charging the main battery or powering motors additional wiring would be required and therefore time and money will be required.

The author believes that the TEG system would be best suited for the FOX engine, a modern car as a whole focuses on reduction in weight, less overall mass requires less energy to move it, mass is a feature all aspects of the vehicle have in common it is therefore something all designers can work towards. As the mass of the vehicle reduces fuel economy and emission

limitations become easier to meet. TEG can be made small and fitted to multiple sections along the length of the exhaust, working from low temperature compared to the Rankine system which requires the steam to be superheated $\approx 640\text{K}$ which may be unachievable at low load or towards the tail pipe without the use of ORC fluids.. Both the Rankine and MT-CU system also require large packaging variations, the addition of a second turbine mounted to the drive shaft may require a complex exhaust system which can quickly become expensive to build, whilst the Rankine system requires many larger components such as the evaporator and pump. The extra pump and working fluid will also greatly increase the overall weight and work used by the engine.

MT-CU and TEG systems require minimal maintenances, whereas the Rankine system will need water changes as a radiator would, and more components which have a greater chance of malfunction.

The Rankine system was the original commercial system used to attempt to recovery thermal energy, since then the other methods have taken the forefront of development. While truck companies and predominately focusing on MT-CU as packing and weight is less of an issue, car manufactures are concentrating on light weight small designs of TEG which can be manufactured around existing vehicle designs.

6.3 Recovery Modelling

In order to determine the optimum position, the study will calculate the potential for energy governed by the temperature data provided by Ford.

Each load site table provided was copied into excel (manifold example) as shown below:

[illegible]

Figure 30 (Manifold Temperatures)

The recoverable energy at each load site is dependant on the enthalpy (H) and the flow rate of the gas. The third column of the JANAF tables determines the enthalpy level relative to the enthalpy at 298K and is defined as:

$$\Delta H = \int nC_p dT$$

Equation 54 (Enthalpy Difference)

Depending on the exhaust composition and temperature will differ the enthalpy, as when calculating Cp values for the combustion model, polynomial trendlines were used to allow calculation of enthalpy levels, dividing this but the molecular weight coverts to SI units j/kg:

$$\text{Enthaply (H)} = \frac{(\phi \times (nMols \times (\Delta HPolynoMol))) \# \text{For each Molecule in the Exhaust}}{\text{Total nMoles} \times \text{Molecular Weight}}$$

Equation 55 (Enthalpy)

Where:

$\Delta HPolynoMol$ = Polynomial equation for each element plus 273 (Covert to kelvin)

The energy is dependent on the mass flow of the gas, the combusted mixture maintains the same velocity. This is the addition of the fuel mass flow (F_{mf}) and the quantity of air fuel mixture.

Fuel Mass Flow per Hour (kg/hr)																
Full Load	3.24	5.22	7.52	9.39	11.35	12.92	14.34	16.08	18.92	21.54	22.34	25.55	26.61	27.62	25.11	
200					11.34	12.96	14.25	17.78	19.47							
190					10.77	12.04	13.39	16.91	18.46							
180				9.11	10.05	11.26	12.52	15.24	17.76	20.61						
170			7.51	8.77	9.38	10.52	11.72	14.06	17.14	18.91	21.41					
160			7.1	8.08	8.74	9.77	10.88	13.05	15.06	17.46	19.8	23.68				
150			6.52	7.46	8.17	9.12	10.19	11.98	14	16.18	18.15	20.39	24.47			
140		5.19	6.04	6.84	7.63	8.45	9.44	11.01	12.93	14.87	16.65	18.92	21.63	25.58		
130		4.73	5.46	6.16	6.96	7.73	8.55	10.05	11.89	13.84	15.4	17.3	19.71	22.93		
120		4.35	4.93	5.75	6.39	7.07	7.79	9.15	10.95	12.86	14.24	15.88	18	20.45	23.3	
110	3.22	3.91	4.52	5.24	5.78	6.43	7.13	8.42	9.97	11.65	13.06	14.58	16.31	18.88	20.9	
100	2.97	3.43	4.13	4.66	5.2	5.78	6.48	7.71	9.03	10.53	11.88	13.38	15	16.76	19.21	
90	2.56	3	3.6	4.11	4.55	5.13	5.77	6.9	8.05	9.36	10.7	12.11	13.59	15.15	16.95	
80	2.2	2.68	3.15	3.59	4.05	4.55	5.13	6.11	7.13	8.39	9.56	10.95	12.32	13.64	15.25	
70	1.89	2.33	2.77	3.21	3.6	4.07	4.56	5.4	6.32	7.45	8.44	9.71	10.92	12.18	13.69	
60	1.63	2.02	2.42	2.78	3.12	3.54	4.02	4.75	5.54	6.5	7.39	8.51	9.6	10.76	12.04	
50	1.38	1.73	2.08	2.38	2.7	3.03	3.41	4.09	4.8	5.6	6.41	7.31	8.25	9.33	10.42	
40	1.04	1.16	1.46	1.76	2.01	2.28	2.57	2.87	3.46	4.1	4.77	5.48	6.27	7.1	7.94	8.92
30	0.89	0.95	1.19	1.42	1.67	1.88	2.13	2.36	2.85	3.41	3.94	4.55	5.21	5.94	6.62	7.51
20	0.61	0.73	0.93	1.1	1.3	1.48	1.65	1.94	2.21	2.67	3.13	3.64	4.18	4.76	5.37	6.08
10	0.44	0.51	0.65	0.78	0.93	1.05	1.16	1.29	1.63	1.92	2.38	2.72	3.18	3.64	4.18	4.79
5	0.36	0.42	0.53	0.63	0.78	0.89	0.98	1.09	1.32	1.5	1.93	2.33	2.73	3.14	3.56	4.08
2	0.32	0.39	0.46	0.56	0.66	0.78	0.86	0.98	1.22	1.44	1.83	2.09				
0	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500

Figure 31 (Fuel Mass Flow Load Map)

$$\text{Exhaust Mass Flow } E_{mf} \text{ (Kg/sec)} = \frac{F_{mf} + (\phi \times F_{mf})}{3600}$$

Equation 56 (Exhaust Mass Flow)

For the purpose of this model the mass flow is assumed to remain the same no matter the exhaust stage, although cross sectional area varies altering pressure and gas velocity.

Exhaust Mass Flow (kg/sec)																
Full Load	0.01328	0.0213	0.030874	0.03886	0.04761	0.05412	0.061065	0.07242	0.07673	0.090109	0.092525	0.101916	0.10607	0.107565	0.09835	
200					0.04757	0.054288	0.060681	0.071219	0.07896							
190					0.04517	0.050434	0.057019	0.067734	0.07487							
180				0.03771	0.04215	0.047167	0.053314	0.061045	0.07203	0.086219						
170			0.030833	0.0363	0.03934	0.044067	0.049908	0.056318	0.06951	0.079107	0.088673					
160			0.029149	0.03344	0.03666	0.040525	0.046331	0.052273	0.06108	0.073041	0.082005	0.094457				
150			0.026768	0.03099	0.03427	0.038203	0.043392	0.047987	0.05678	0.067698	0.075171	0.083727	0.09754			
140		0.0212	0.024798	0.02831	0.032	0.035396	0.040199	0.044101	0.05244	0.062206	0.069959	0.07547	0.08622	0.09962		
130		0.0193	0.022416	0.0255	0.02919	0.03238	0.036409	0.040256	0.04822	0.057897	0.063782	0.069008	0.078566	0.0893		
120		0.0178	0.02024	0.0238	0.0268	0.029615	0.033172	0.036851	0.04441	0.053798	0.058977	0.063344	0.07175	0.079641	0.09126	
110		0.0132	0.016	0.018557	0.02169	0.02424	0.026935	0.030362	0.033727	0.04043	0.048736	0.05409	0.058158	0.065013	0.07267	0.08186
100		0.01218	0.014	0.016956	0.01929	0.02181	0.024212	0.027594	0.030883	0.03662	0.044051	0.049203	0.053371	0.059792	0.065271	0.07524
90		0.0105	0.0123	0.01478	0.01701	0.01908	0.021489	0.024571	0.027638	0.03265	0.039156	0.044316	0.048305	0.054171	0.059001	0.06639
80		0.00902	0.0109	0.012933	0.01486	0.01699	0.019059	0.021845	0.024474	0.02892	0.035098	0.039594	0.043678	0.049109	0.05312	0.05973
70		0.00775	0.0095	0.011372	0.01329	0.0151	0.017049	0.01948	0.02163	0.02563	0.031166	0.034956	0.038732	0.043528	0.047434	0.05362
60		0.00668	0.0082	0.009935	0.01151	0.01309	0.014829	0.017119	0.019026	0.02247	0.027192	0.030607	0.033945	0.038267	0.041904	0.04716
50		0.00556	0.0071	0.00854	0.00985	0.01133	0.012692	0.014521	0.016363	0.01947	0.023427	0.026548	0.029159	0.032885	0.036335	0.04081
40	0.004	0.00476	0.006	0.007226	0.00832	0.00956	0.010765	0.012221	0.013859	0.01663	0.019955	0.022636	0.02581	0.028901	0.030922	0.03494
30	0.004	0.0039	0.0049	0.00583	0.00691	0.00789	0.008922	0.01005	0.011416	0.01383	0.016482	0.018845	0.020782	0.023678	0.025781	0.02941
20	0.003	0.00299	0.0038	0.004516	0.00538	0.00621	0.006912	0.007835	0.008852	0.01083	0.013094	0.015076	0.016674	0.018974	0.020913	0.02381
10	0.002	0.00209	0.0027	0.003202	0.00385	0.0044	0.004859	0.005433	0.006259	0.00779	0.009356	0.011265	0.012685	0.014509	0.016279	0.01876
5	0.001	0.00172	0.0022	0.002587	0.00327	0.00369	0.004105	0.004642	0.005287	0.00649	0.008325	0.00965	0.01089	0.012516	0.013864	0.01598
2	0.001	0.0016	0.0019	0.002299	0.00273	0.00327	0.003602	0.004173	0.004887	0.00584	0.007656	0.008656				
0	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500

Figure 32 (Exhaust Mass Flow)

With fuel power allows the calculation of engine efficiency, as expected when the power released from the fuel increases the engine efficiency matches.

$$\text{Recoverable Energy (\%)} = \frac{\text{Total Energy at Exhaust Stage (j/s)}}{\text{Fuel Power (watts)}}$$

Equation 57 (Recoverable Energy)

Where:

$$\text{Total Energy at Exhaust Stage (j/s)} = E_{mf} \text{ (kg/sec)} \text{Exhaust Stage Enthaply (j/kg)}$$

Equation 58 (Total Energy at Exhaust Stage)

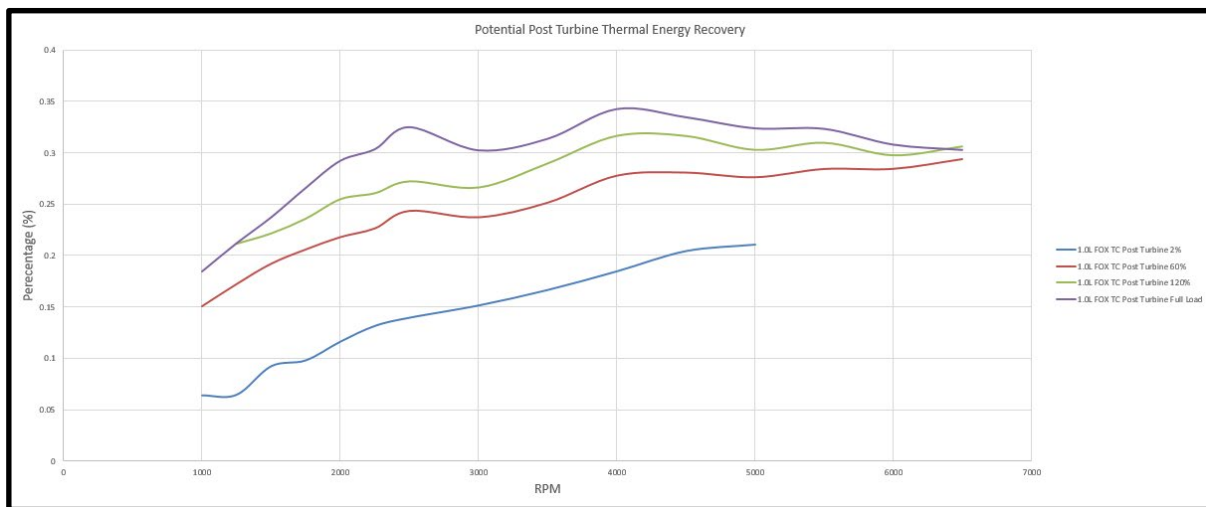


Figure 33 Potential Post Turbine Thermal Energy Recovery)

As discussed post turbine will allow for maximum enthaply into the turbine and will appose the greatest level of recovery possible. During the NEDC 3000RPM 60% load (red trace) cycle there will be ≈23% available, assuming 100% of this is recovered, however most recovery system struggle to exceed 75% of full recovery:

$$\text{Power Available Post Turbine (watts)} = \text{Recoverable Energy} \times \text{Fuel Power}$$

Equation 59 (Power Available Post Turbine)

$$\text{Power Aвалиable Post Turbine (watts)} = (0.23 \times 58055.556) \times 0.75$$

$$\text{Power Aвалиable Post Turbine (watts)} = 10014.6$$

Governing that fact at this load site the power is ≈18Kw, an additional ≈10Kw from the recovered energy will increase the overall power by 55.55%. Figure 33 indicates the effect of the drive cycle on modern engine, there is a clear dip in what is otherwise linear progression, it can be assumed the effects of the emission control is reducing the energy available. This could be due to variation of the Atkinson cycle as discussed in section 3.2:1 or EGR alternating mixture mass and chamber venting temperature.

7.0 Conclusion

The model provides relatively accurate data based on the values provided, the drawback with this form of modeling is assumption of no cyclic variation and, the supplied data, has been found to consist of a number of indiscretions such as low manifold temperatures. Also, the test is commonly undergone in a test cell as the NEDC progress ambient air temperature may increase, when analyzing energy recovery potential this raises the enthalpy relevant datum, above 298K. As with all modeling techniques, mathematical models become sensitive to value inputs. Key junctions of the combustion model such as average η and SOC are user defined, uneducated estimates could form unrealistic results. The model is also limited by the knowledge of the creator, if results are assumed to be correct this could lead to inaccurate results, hence the need for validation.

It is clear that turbochargers are not ideal as during low RPM, low load the compressor is proving to be a hindrance, using energy rather than assisting. But it can be concluded that as the efficiency of the turbo increases the mechanical work produced from the fuel power. The author purposes a concept idea based on TEG system combined with electronic supercharger. With a switchable TEG system before the turbine the extensive energy can be utilized before the turbine requires it, once the turbine is beneficial the pre-turbine TEG would switch off increasing the enthalpy into the turbine. However, this will move the turbo threshold further in the RPM range, but would allow for a larger turbo to be used as the supercharger would support driveability at lower RPM.

It has been made apparent the ICE utilizes $\approx 33\%$ of its combusted power, when considering the efficiency potential of future battery technology, it is becoming important for more studies to be compiled in order to forward the harnessing of the power available within gasoline engines. As the study indicates with the amount of recoverable energy available, up to $\approx 30\%$, hybrid vehicles could almost double the energy used but standard TC gasoline engines.

Load (%)	Manifold (%)	2	60	120	FL	Pre Turbo (%)	2	60	120	FL	Post Turbo (%)	2	60	120	FL
RPM															
1000	0.09957369	0.313807			0.187	0.07768986	0.17			0.209	0.063467782	0.151			0.185
1250	0.13128251	0.31876			0.216	0.10201915	0.191	0.233	0.237	0.064024532	0.172	0.212			0.212
1500	0.13379069	0.320779	0.2189	0.241	0.13478683	0.205	0.242	0.262	0.091964388	0.192	0.222	0.237			
1750	0.15780285	0.322532	0.226	0.283	0.13916659	0.202	0.252	0.3	0.097658162	0.206	0.236	0.266			
2000	0.18172366	0.327889	0.2473	0.327	0.16259926	0.232	0.274	0.34	0.115914506	0.218	0.255	0.292			
2250	0.18583038	0.326385	0.2528	0.341	0.20076298	0.243	0.281	0.351	0.131283457	0.226	0.261	0.304			
2500	0.20382096	0.320346	0.2633	0.366	0.19851499	0.259	0.292	0.375	0.139307309	0.243	0.272	0.325			
3000	0.20659729	0.324323	0.2614	0.339	0.21604218	0.252	0.285	0.349	0.151284567	0.237	0.266	0.303			
3500	0.22302784	0.324783	0.2862	0.351	0.22919478	0.268	0.308	0.384	0.166405982	0.251	0.29	0.314			
4000	0.24203139	0.315699	0.3108	0.377	0.24590875	0.294	0.336	0.391	0.184620412	0.277	0.317	0.343			
4500	0.24686317	0.313508	0.3075	0.363	0.26118658	0.298	0.335	0.381	0.204382048	0.28	0.317	0.335			
5000		0.301947	0.2953	0.355	0.26346274	0.292	0.322	0.374	0.210706438	0.276	0.303	0.324			
5500		0.294617	0.3053	0.352		0.299	0.333	0.374		0.284	0.31	0.323			
6000		0.286904	0.302	0.334		0.298	0.328	0.356		0.284	0.298	0.308			
6500		0.277395	0.3166	0.317		0.308	0.343	0.344		0.293	0.306	0.303			

Load (%)	Manifold (watts)	2	60	120	FL	Pre Turbo (watts)	2	60	120	FL	Post Turbo (watts)	2	60	120	FL
RPM															
1000	355.975947	4688.806			5550	277.741261	2547			5550	226.8973191	2250			5550
1250	553.574601	5302.379	8367.9	10348	430.180736	3530	9301	11329	269.9703649	3188	8451	10158			
1500	686.79222	7115.953	9802.4	16586	691.905021	4543	10930	18071	472.0638581	4256	10029	16368			
1750	954.70725	8219.201	11913	24344	841.957848	5139	13262	25780	590.8318822	5243	12452	22897			
2000	1299.32417	9377.612	14487	34043	1162.58471	6634	16063	35415	828.7887184	6226	14944	30416			
2250	1464.96283	10591.19	16381	40335	1582.68149	7891	18212	41626	1034.951256	7339	16912	35954			
2500	1830.99162	11804.76	18805	48126	1783.32628	9533	20868	49276	1251.443997	8961	19456	42723			
3000	2310.44639	14121.58	21923	56198	2416.07174	10961	23870	57828	1691.865745	10317	22353	50141			
3500	2943.96749	16493.57	28728	60891	3025.37114	13585	30936	63086	2196.558967	12752	29104	54402			
4000	4060.07665	18910.39	36635	74430	4125.11932	17541	39954	77299	3097.00741	16522	37332	67650			
4500	4729.48693	21237.53	40145	74426	5003.8995	20157	43722	78035	3915.619407	18992	41324	68539			
5000		23554.36	42979	83082		22788	46900	87610		21516	44117	75861			
5500		25926.34	50381	85790		26328	54938	91222		24979	51141	78882			
6000		28298.32	56618	84582		29382	61404	90178		28007	55827	78000			
6500		30615.15	67616	73028		33985	73268	79158		32383	65439	69721			

Figure 34 (Recovery Potential Manifold Post and Pre-Turbine)

Further construction of the computational model would be to build a suitable code within MatLab. With the additional precision of MatLab tools, refining of technics will become easier and the model will become much more user friendly. The author began to code the combustion model however time constraints and skill level slowed progress, the beginning can be found in appendix 9.4.

For further progression of the model, the study would form the same calculator to determine the recovery available from the radiator and cooling system. For this next stage of the study the enthalpy of H_2O in its liquid and semi saturated state will need to be match to the corresponding radiator temperature load site map, following the same process as used for exhaust recovery will lead to recoverable data calculation.

The author would like to pay acknowledgement to Malcolm McDonald for continuous support and guidance during the construction, design and analysis of the study, without whom deadlines and completion would have been unachieved.

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9.0 Appendix

9.1 Initial Project Proposal

The aim of the final year project will be to construct a two-zone combustion model, to determine the thermal distribution from a Ford 1.0L FOX GDI TC engine. Starting with production data from the Ford Motor Company production NEDC tests, it will require calculation of polytrophic indices, unburnt fuel zone temperatures, mass fraction burnt, burn time, burnt compression, using a burnt and unburnt two zone model.

Starting with the data from the Ford Sigma 1.6 N/A and 1.5 TC engines to first calculate the polytrophic efficiency and the start of combustion, and comparing them to establish a direct correlation between turbocharged and naturally aspirated engines. From this point the TC data will be compared to the 1.0L Fox engine to establish the how the reduction of engine capacity helps the speed of the burn.

The analysis of temperature data will allow for calculation of TC and engine efficiency. The engine efficiency will then be used to determine the distribution of thermal energy from the combustion process.

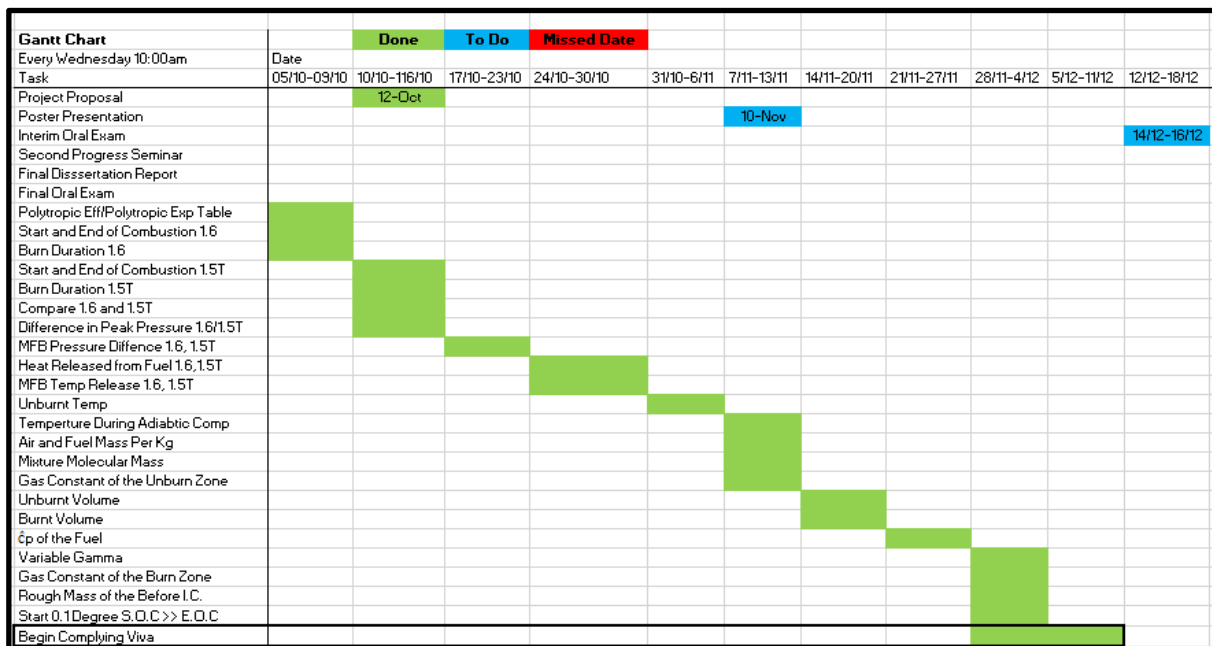


Figure 35 (Initial Gantt Chart)

9.2 Modelling Excel Work

9.2:1 Combustion Model

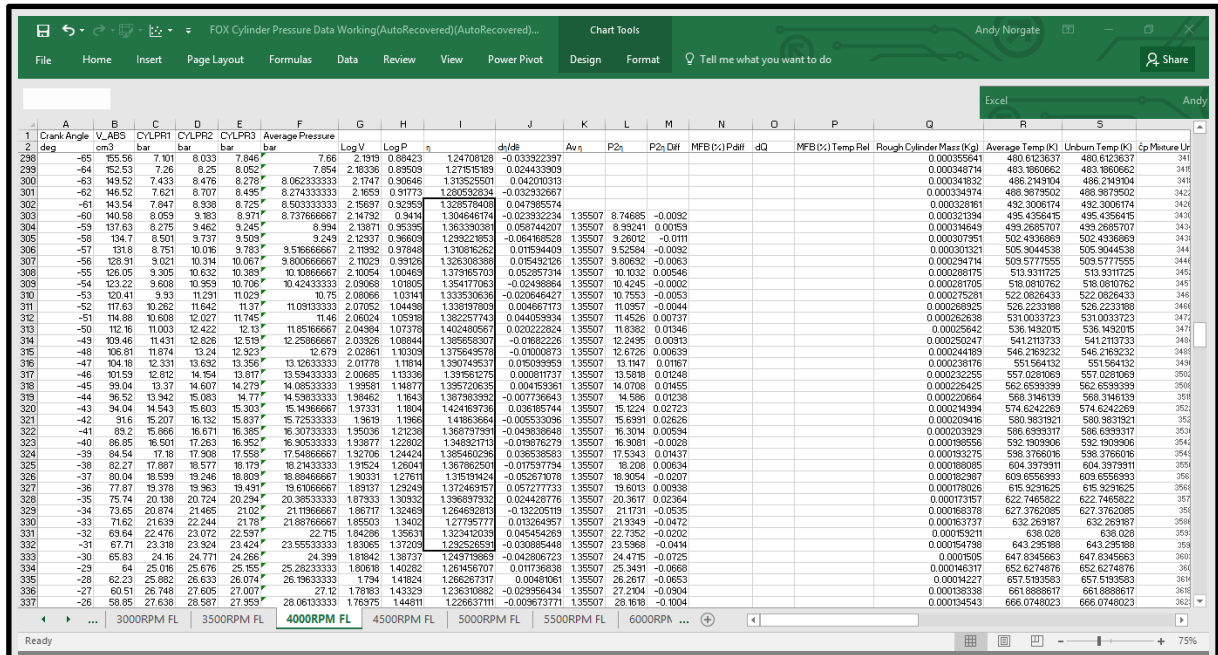


Figure 36 (Selection of Average Polytrp Index)

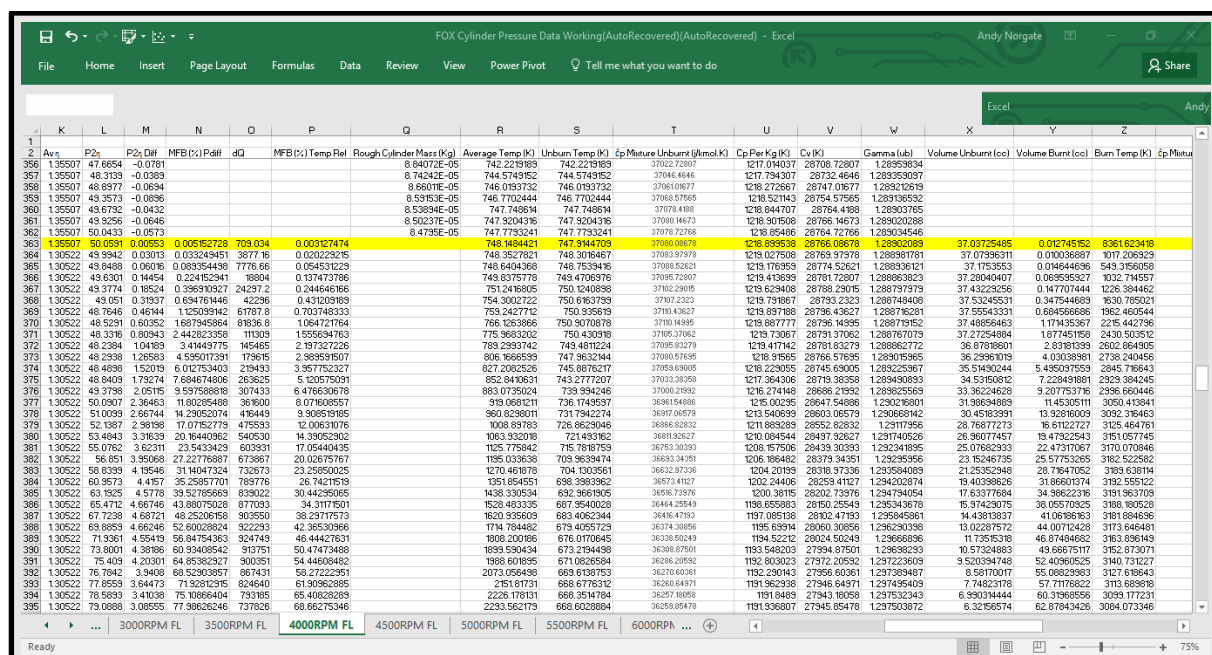


Figure 37 (Excel Model Section)

The yellow highlighted area represents the SOC, purple highlighted cells demonstrate EOC Figure 37 demonstrates the layout of the P2_{tr}, MFB (both methods), all temperature calculations, CP/CV values, gamma and burn temperatures.

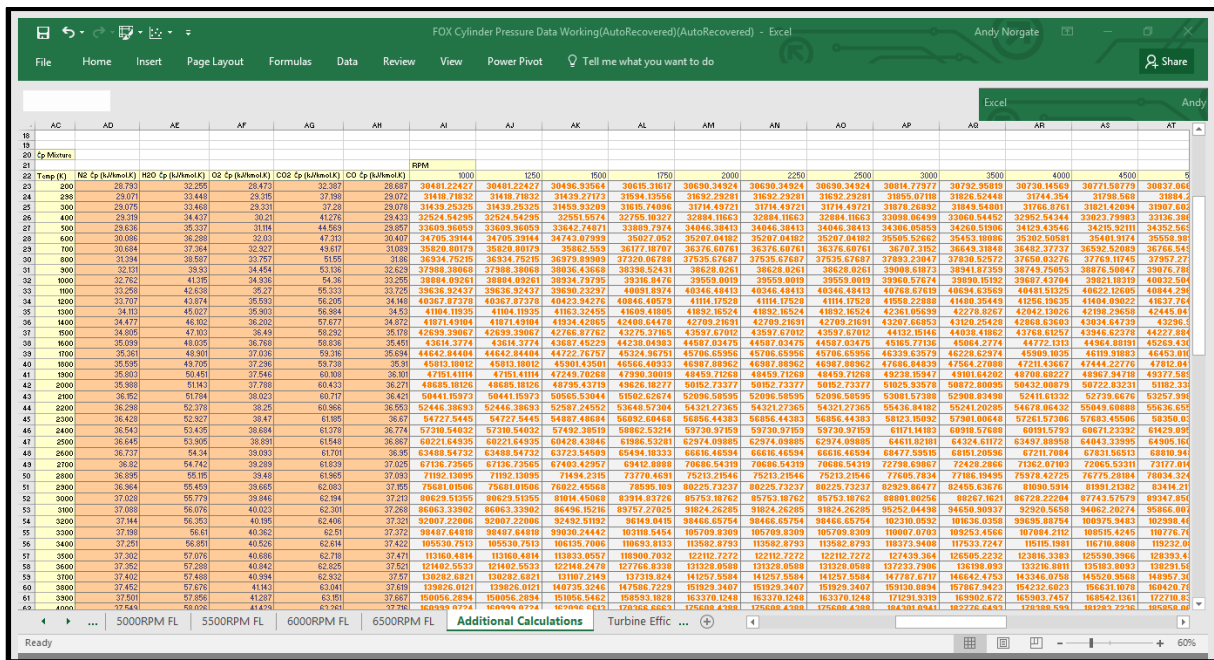


Figure 38 (Cp of the Mixture Excel Model)

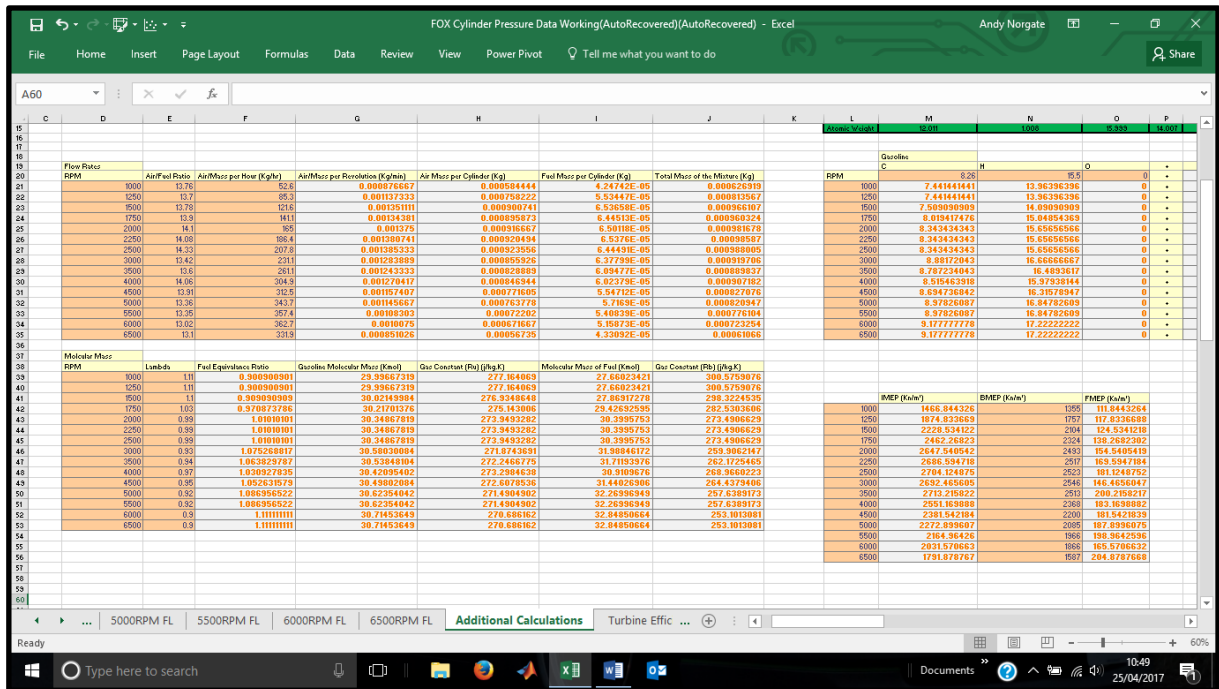


Figure 39 (Chemistry Calculations and IMEP, BEMP, FEMP)

9.2:2 Engine Efficiency Modelling

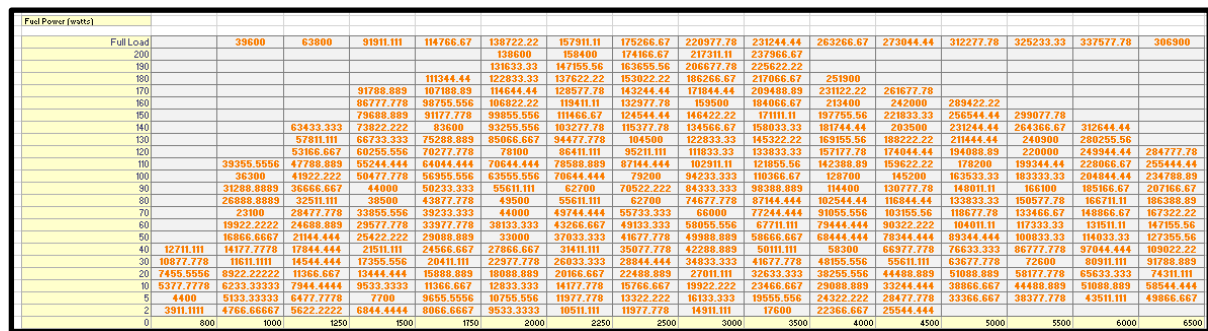


Figure 41 (Fuel Power Load Map)

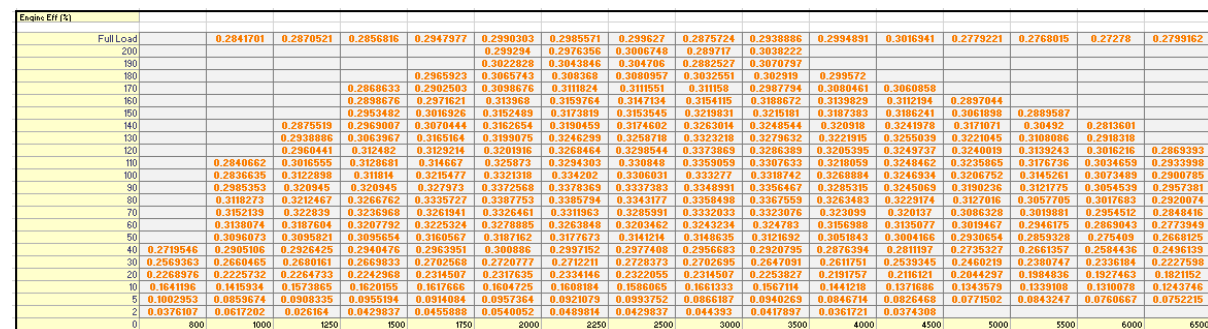


Figure 40 (Engine Efficiency Load Map)

9.2:3 Thermal Losses

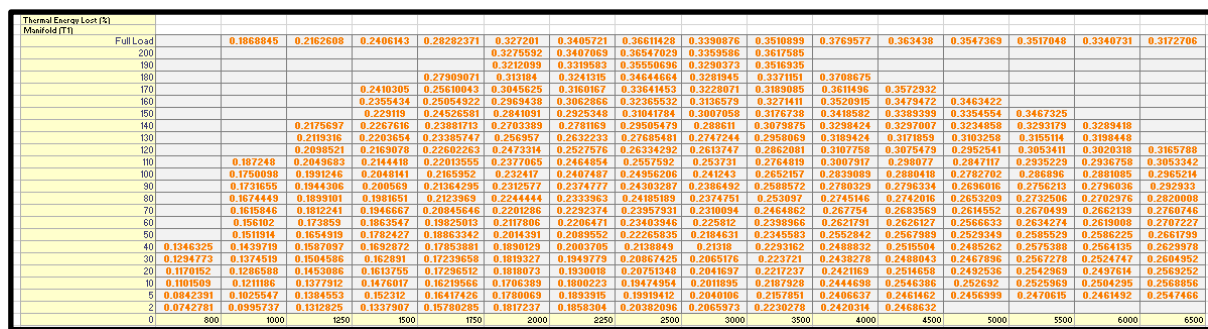


Figure 43 (Thermal Energy Loss/Recoverable in the Exhaust (%))

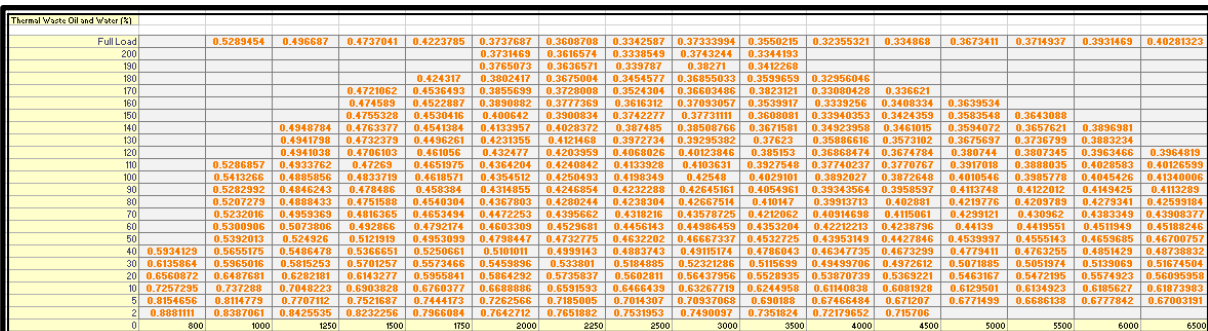


Figure 42 (Auxiliaries Thermal Loss/Recoverable (%))

Recoverable Thermal Energy (%)																	
Pre-Turbine (T2)																	
Full Load		0.2091136	0.2367706	0.262156	0.2995017	0.340394	0.3514683	0.3748682	0.3489212	0.3637498	0.3914889	0.3810606	0.3740679	0.3739774	0.3561772	0.3439037	
200						0.3405289	0.35200973	0.3742223	0.3483262	0.3743791							
180						0.3339688	0.34416938	0.3663443	0.3406112	0.3652659							
160					0.2960012	0.3267982	0.33855078	0.3591301	0.3495689	0.3518335	0.3859579						
170				0.2618618	0.2775906	0.3200936	0.32240626	0.3511004	0.3373113	0.3339578	0.3776352	0.3747895					
150			0.2590821	0.2728826	0.3133175	0.32386362	0.3415347	0.3298373	0.3438442	0.3702817	0.3686516						
140			0.252969	0.2653793	0.4032045	0.3138364	0.3301861	0.3181025	0.3349716	0.3608352	0.3581765	0.3541455	0.3683294				
130			0.2382584	0.2505006	0.2648589	0.2935851	0.30331741	0.3185871	0.3258024	0.3507487	0.3518669	0.345079	0.3514036	0.3513727			
120			0.2341636	0.2453222	0.2602202	0.2816032	0.28980261	0.3040723	0.2968849	0.3159441	0.3418136	0.3418805	0.3347004	0.3400049	0.3434247		
110			0.232362	0.2416631	0.2516075	0.2742335	0.2810109	0.2922353	0.2845833	0.3001995	0.338532	0.3349513	0.3221917	0.3329591	0.3279589	0.3438483	
100	0.2090727	0.2284016	0.2401566	0.2453255	0.2652234	0.27465767	0.2856249	0.2788484	0.3001135	0.3273127	0.3277344	0.3194013	0.3233857	0.3218726	0.334258		
90	0.1937361	0.2226499	0.2299054	0.2418493	0.2608081	0.26941328	0.2808573	0.2726553	0.2933614	0.3158934	0.320102	0.30908	0.3188831	0.3181428	0.3279154		
80	0.1947374	0.2170379	0.2243658	0.2381439	0.2570005	0.26579309	0.2765882	0.2685117	0.2862081	0.309454	0.314447	0.3034407	0.3116044	0.3122352	0.3273024		
70	0.1862788	0.2085853	0.2201182	0.2345293	0.2494565	0.25943941	0.2723271	0.2653915	0.2808853	0.3064165	0.3088048	0.2997987	0.3089437	0.3089558	0.3187765		
60	0.1789853	0.2007886	0.2153456	0.228575	0.2416288	0.25310424	0.2655885	0.2668707	0.2739723	0.3003083	0.3033485	0.2987485	0.3031466	0.302072	0.3134885		
50	0.1704428	0.1906358	0.2047732	0.2161735	0.2318446	0.24316224	0.2587102	0.2517327	0.267506	0.2943918	0.2975592	0.2921193	0.2931767	0.2878903	0.3078276		
40	0.162978	0.1809524	0.1951951	0.2060134	0.2206408	0.23069419	0.2457473	0.2432716	0.2613485	0.2875289	0.2913964	0.2870548	0.2922866	0.2927136	0.3023795		
30	0.1561813	0.171944	0.1845353	0.1942675	0.207261	0.22098828	0.2348237	0.2368953	0.2543167	0.2793889	0.2952069	0.2829046	0.2901453	0.2886206	0.2982571		
20	0.13326385	0.1455865	0.1619585	0.1757491	0.1869538	0.1989471	0.2129817	0.2272988	0.2272208	0.246775	0.2718252	0.2783765	0.2779425	0.286806	0.294511	0.2943835	
10	0.11354035	0.1313499	0.148845	0.1686465	0.1820061	0.1970065	0.20643841	0.2207534	0.2195456	0.2397732	0.2657212	0.2767234	0.2744224	0.2816411	0.2762584	0.2871389	
5	0.10707672	0.1234465	0.1439075	0.1605385	0.1743467	0.1856159	0.19430491	0.2047649	0.2109033	0.2317717	0.2621359	0.2700618	0.2605408	0.2712906	0.2666496	0.2776747	
2	0.08197697	0.1055422	0.1402538	0.1579508	0.1780906	0.2023421	0.20865852	0.2167133	0.2205484	0.2352966	0.2610572	0.2645693	0.2632854	0.2640782	0.2567684	0.2702042	
0	0.07768985	0.1020191	0.1347867	0.1391656	0.1629933	0.200763	0.19851499	0.2160422	0.2239148	0.2459089	0.2611865	0.2634627					

Figure 45 (Recoverable Energy Pre-Turbine (%))

Recoverable Thermal Energy (%)																	
Post-Turbine (T3)																	
Full Load		0.1848662	0.21229879	0.2374539	0.2660094	0.2923488	0.3035832	0.3250146	0.3025399	0.3136757	0.3426166	0.3346885	0.3239033	0.3233857	0.30807602	0.3029054	
200						0.2924371	0.3055329	0.3247427	0.3013726	0.3256462							
180						0.2888203	0.301458	0.3208937	0.2959186	0.3180569							
170				0.2641773	0.2888203	0.301458	0.3208937	0.2959186	0.3180569								
160			0.2367476	0.2556337	0.2820866	0.2962414	0.313123	0.3012475	0.2975372	0.3166447	0.3311873						
150			0.2348378	0.2523698	0.2780988	0.290948	0.3075806	0.2966256	0.3105328	0.3344198	0.3269483	0.3193971					
140			0.2321007	0.2498299	0.2736204	0.2860181	0.3025895	0.2914316	0.3060802	0.3299312	0.3247223	0.3158978	0.3223844				
130			0.2197247	0.2309862	0.2463128	0.2715197	0.2814125	0.2958162	0.2852963	0.3247411	0.3245842	0.3148157	0.3149707	0.30755159			
120			0.21229879	0.2284123	0.2438122	0.2638044	0.2718106	0.2846459	0.276539	0.2895938	0.3198269	0.3182212	0.3106381	0.3088607	0.30525348		
110			0.21193159	0.2219316	0.2362517	0.2551326	0.2609609	0.2724601	0.2664995	0.2899503	0.3166894	0.316577	0.303068	0.3089437	0.29780992	0.3063875	
100	0.1834547	0.20635079	0.2199123	0.2275704	0.2456449	0.2537976	0.2661272	0.2616613	0.2835209	0.310291	0.3119716	0.297484	0.3056311	0.29624372	0.3046893		
90	0.1742879	0.19540081	0.208127	0.2238978	0.2406367	0.2488624	0.2627247	0.2610471	0.2811721	0.305229	0.3065949	0.2951303	0.3043471	0.30088499	0.3031077		
80	0.1740874	0.18974882	0.2082459	0.2211701	0.2394297	0.2483066	0.2617576	0.2667116	0.2735127	0.2967151	0.3022241	0.2918956	0.3014503	0.2983752	0.3111327		
70	0.1702828	0.18696975	0.2076131	0.2215952	0.2358127	0.2438092	0.2573884	0.262153	0.2668811	0.2918542	0.2947559	0.287713	0.2957139	0.2950798	0.3071574		
60	0.1649308	0.18612481	0.2040378	0.2184687	0.2329704	0.2372627	0.2511865	0.2434746	0.2590647	0.2847806	0.2825762	0.2877409	0.2825762	0.290557	0.29042392	0.3015706	
50	0.1509584	0.17214235	0.1918647	0.2057238	0.2176371	0.226186	0.2431641	0.2369489	0.2511116	0.277294	0.2803615	0.2758134	0.2839566	0.28395186	0.2934166		
40	0.1367074	0.15772052	0.1780416	0.1911764	0.203469	0.2113728	0.2280368	0.2288925	0.2423852	0.2677107	0.2717254	0.2608822	0.2741804	0.27550248	0.2845684		
30	0.1000319	0.1154312	0.13669795	0.161814	0.174428	0.1854483	0.1965771	0.2123762	0.2133795	0.2326728	0.2559727	0.2618055	0.2683984	0.26842317	0.2774346		
20	0.0867699	0.0933447	0.12012446	0.1414876	0.1578029	0.1671948	0.1792239	0.1947495	0.1971047	0.2185487	0.2416894	0.2504093	0.2499004	0.2616388	0.26016228	0.2695649	
10	0.0658825	0.0825421	0.09320694	0.1192901	0.1353447	0.1474168	0.1738707	0.1758068	0.2002766	0.2256377	0.2378565	0.237626	0.2480705	0.24661996	0.2583128		
5	0.0555621	0.0637187	0.08728355	0.1059556	0.1233723	0.1348947	0.1438715	0.1649929	0.1633187	0.1824606	0.212283	0.2235213	0.2255079	0.23379	0.2335892	0.2460946	
2	0.0495951	0.0635791	0.08423498	0.09762	0.1199526	0.1222564	0.1412617	0.1502039	0.1634916	0.189192	0.207194	0.2152257	0.2123353	0.2281865	0.22275262	0.2434156	
0	0.0634678	0.0640246	0.09196439	0.0976582	0.1159145	0.1312835	0.1393073	0.1512846	0.166406	0.1846204	0.204382	0.2107064					

Figure 44 (Recoverable Energy Post-Turbine)

9.3 MFB Extended Research

MFB is dependant of many factors;

- Fuel Equivalence Ratio – Flame speed max is at $\approx 1.1\phi$, the distance between fuel atoms is small enough to spread the flame efficiency without being too close as to not create an incombustible localised zone. A lean mixture dissipates the fuel molecules further and each local combustion must grow larger before reaching the adjacent fuel molecules.

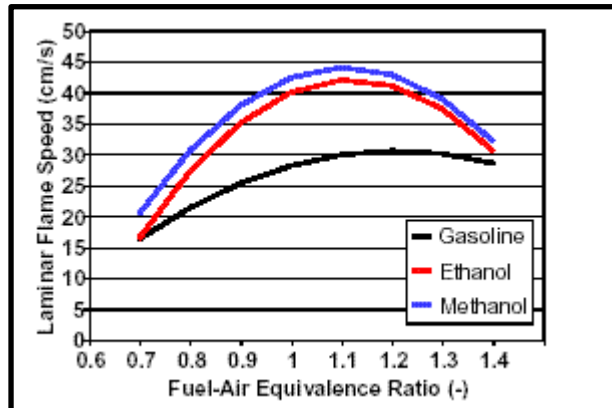


Figure 46 (MFB Effect with FER) (frenchconnector, n.d.)

- Port Geometry – Inducing swirl and turbulence increases flame speed by effectively increasing the surface area, the trajectory of the unburnt mixture helps to “feed” the flame front with fresh fuel. However, too much turbulence could cause flame starvation. As RPM increases and combustion begins earlier the piston travels towards the flame front for longer again “feeding” the burning mixture.
- Inlet Temperature and Pressure – Greater unburnt mixture temperature and pressure forms a homogenous air and fuel mixture by increasing the density of the charge. Ignition timing has a similar effect as MBT timing can be used to get the perfect pressure.
- Molecular Composition – Carbon has a high resistance to combustion, therefore the greater the carbon content in the fuel the more leisurely the flame speed.

9.4 MatLab Code Initial Stages

The code as created to the point of MFB polytrophic index graph formation.

```

clc
% clear

%Data Input
AveragePressure; %Average Pressure Data
CrankAngle; %Crank's Angle
V_ABS; %Volume per Crank Angle
logV = log10(V_ABS); %Log Volume
logP = log10(AveragePressure); %Log Average Pressure

%Polytrophic Index
x = 1; %Columns
y = 720; %Rows
Poly = zeros (y,x);
for j = (2:y-1)
    Poly (j,x) = -((logP(j,x)-logP(j-1,x))/(logV(j,x)-logV(j-1,x)));
end

%DPoly/DCrankAngle
for j = (1:y-1)
    N (j,1) = (Poly(j+1,1) - Poly(j,1));
end

%Average DPoly/DCrankAngle
%Average BSOC
NSOC = mean(Poly(300:335,1));
%Average AEOC
NEOC = mean(Poly(430:460,1));

%P2N
SampleRange = Poly(300:461,1);
AvPP2Sample = AveragePressure(300:461,1);
VolSample = V_ABS(300:461,1);
P2N = zeros (161,1);
for j = (2:60)
    P2N (j,1) = AvPP2Sample(j-1,1)*(VolSample(j-1,1)/VolSample(j,1))^NSOC;
end
for j = (61:161)
    P2N (j,1) = AvPP2Sample(j-1,1)*(VolSample(j-1,1)/VolSample(j,1))^NEOC;
end

%P2N Diff
for j = (1:160)
    P2NDiff (j,1) = AvPP2Sample(j+1,1) - P2N(j+1,1);
end

%Start of Combustion Selection
SOC = 9
% input ('Input Start of Combustion (Degrees from TDC): ')
%End of Combustion Selection
EOC = 62
% input ('Input End of Combustion (Degrees from TDC): ')

%Combustion Length
CLCA = CrankAngle(352:432,1)

```

```
CLAvP = AveragePressure(352:423,1)
CLVol = V_ABS(352:432,1)
CLP2NDiff = P2NDiff(70:132,1)
%Mass Fraction Burnt P2N Diff
for j = (1:53)
MFBPDiff = (sum(CLP2NDiff(1:j,1))/sum(CLP2NDiff))*100
end
plot (MFBPDiff)
```