

Advanced Combustion Analysis

The development of an advanced combustion analysis tool

Automotive Engineering Project Report

Aijaz Ahmed, Oskar Edholm, Ajanish Lutimath,
Avanish Raj, Filip Tagesson

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Gothenburg, Sweden 2021

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Automotive engineering project report 2021:06

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Abstract

The main purpose of this project is to study and investigate advanced combustion analysis in compression-ignited (CI) and spark-ignited (SI) engines. A MATLAB code will be developed as the major deliverable, which will calculate the (apparent and real) rate of heat release with the provision of different, effective intake valve closing, heat transfer models, blow-by loss and gas properties with automatic polytropic coefficient detection. This code will also be used to compare and validate the different zero level corrections with commercial data acquisition software by Dewesoft (stakeholder for this project).

For the study a 6-cylinder CI engine and an SI engine at Chalmers is used to record the in-cylinder pressure. Measurements of the in-cylinder pressure accurately play an important role in pressure trace analysis. In this project, the experimental data like raw cylinder pressure and crank angle degree is recorded with the Dewesoft data acquisition system. Along with the recorded data, data from other test cells were also collected which was further analyzed and processed through the MATLAB code developed.

After the investigation, the common assumption of constant values of certain parameters, such as IVC effective, constant polytropic coefficient, temperature at IVC, wall temperature, heat transfer loss and blow-by loss was found to skew the data significantly, thus giving incorrect results. Further, in-depth analysis for each of the above parameters is done and function scripts were developed and integrated into the main MATLAB program.

The major findings are that the commercial combustion analyzer software usually considers a constant polytropic coefficient and also performs an apparent rate of heat release, thus neglecting the heat transfer and blow-by losses. But in reality, the polytropic coefficient varies and the heat transfer and blow-by loss do contribute and affect the amount of heat energy liberated during combustion. The results from the analysis of the data's influence on the heat release is presented in the report.

Keywords: Combustion, Analysis, Heat release.

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Aijaz, Oskar, Ajanish, Avanish, Filip

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Nomenclature

W	Mechanical work	J
γ	Specific heat ratio, gamma, c_p/c_v	—
ρ	Lubricating oil type	—
τ	Cycle type	—
θ	Crank angle	°
A	Area	m^2
a	Crank radius	m
B	Bore	m
c_p	Specific heat constant pressure	$JK^{-1}mol^{-1}$
c_v	Specific heat constant volume	$JK^{-1}mol^{-1}$
dz	Shape and clearance piston joint	—
E_{loss}	Wall heat and blowby losses	J
f	Molecular degrees of freedom	—
h_c	Heat transfer coefficient	$W/m^2 * K$
i	Number of cylinders	—
k	Sealing ring type	—
L	Stroke	m
l	Connecting rod length	m
li	Number of piston rings	n
m	Mass	Kg
N	Engine speed	RPM
n	Compound amount	—
p	Pressure	Pa
Q	Heat transfer	J
Q_{ch}	Chemical energy released by combustion	J
Q_{ht}	Heat transfer	J
R	The gas constant	$JK^{-1}mol^{-1}$
r_c	Compression ratio	—
S_p	Mean piston speed	m/s
T	Temperature, mean Temperature	K
U_s	Internal energy	J
V	Volume	m^3
W	Average gas velocity	m/s
w_{BB}	Blow-by gas flow velocity	m/s
z	Wear parameter	—
Q_{BB}	Mass loss (Blow-by)	J

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1

Introduction

Climate change is one of the most serious issues and ongoing that the world needs to address. The effects of this is not only on environment, but also on wildlife as well as the well being and life of humans. Hence it is our responsibility to address this seriously issue that threatens the planet today. One of the ways of doing this is by controlling and lower the emissions from transportation on a global scale. This can be done by imposing new, more stringent emissions regulations. These ever demanding emission regulation has forced automobile manufacturers to study the in cylinder processes in detail. The efficiency, durability, fuel consumption, performance and much more depend on the process that takes place inside the combustion chamber. Hence it is very important to analyze the combustion in depth and understand it thoroughly. This project aims to improve combustion analysis and allow for efficient engine optimization. A MATLAB based models will be developed, calibrated and verified against experimentally collected data. The project will be carried out by five students of the Automotive Engineering Master's programme at Chalmers University and will be supervised by Jonas Sjöblom, Ahfaz Ahmed and Lucien Koopmans from the Combustion and Propulsion System (CaPS) division. The main stakeholder is Dewesoft. The developed tool will also be integrated into ongoing research projects at the division.

1.1 Background

Dewesoft is a company dealing with data acquisition and analysis. They have a suite specifically designed for combustion analysis called "Combustion engine Analyser" (CEA). This analyzer takes the data collected by their data acquisition module and presents a combustion analysis to the user. The accuracy of an combustion analysis is of prime importance as this plays a major role in an engine calibration. Inaccurate calibration may lead to higher emissions and lower efficiency of the engine. Therefore the accuracy of the models is of utmost importance. Dewesoft CEA provides various options and models for processing input pressure data from engine including three modes for zero level correction, suggestion of polytropic coefficient, thermodynamic calculation for heat release etc. Their acquisition devices and software are used globally for engine development. Academia also has a large interest in developing combustion analysis tools that can help them in further understand the combustion process and the heat release. For academia it is important to have full transparency of the workings of the code used to truly understand the analysis and the results.

1.2 Aim

The project aim is to do an in depth analysis of the combustion in a heavy duty diesel engine and a light duty SI engine by developing several model in MATLAB which will be correlated with the experimentally collected data. A new complete code for pressure trace analysis will be developed in MATLAB. This code will be used as point of reference to compare with the commercial software's. Different zero level correction methods, polytropic coefficient determination methods and more, are investigated and if required the models are further optimized. The code written in MATLAB will also contain different models for calculating heat transfer , blow-by, filtering and averaging. A method for input data validation will be created based in the automatic polytropic coefficient determination and optimized.

1.3 Limitations

1. The experimental data is from the different Chalmers test rigs.
2. Only the currently present sensors will be utilized.
3. The project is limited to two study periods.
4. The code is completed in MATLAB.

1.4 Specifications of issues under investigation

1. Different zero level correction models will be investigated. The differences are evaluated and the best method is recommended
2. Improving the accuracy of thermodynamic zero level correction
3. Different methods for determination of polytropic coefficient are evaluated
4. Improve RoHR analysis
5. A model for blow-by is to be implemented and examined.

1.5 Deliverables

Throughout this project a goal is set. This goal is to at the end of the project have created certain results. These products are presented here in the form of project deliverables. The main deliverable is the software that is to be created. The software will be a MATLAB code system that can perform certain tasks. These tasks include the use of different zero level corrections, different ways of calculating and using polytropic coefficient, allow for different models when calculating heat transfer, blow-by, loss angles and data smoothing/filtering and finally visualize the combustion using pressure trace diagrams, P-V diagrams and different heat release diagrams.

2

Theory

An internal combustion engine is a machine which is used for converting chemical energy stored in fuel into mechanical work. The fuel is mixed in appropriate amount with air and burned inside the combustion chamber. This converts the chemical energy of fuel into thermal energy. The thermal energy increases the pressure and temperature of the entrapped gas, which exerts a force on the piston causing it to move. The piston is connected to the crankshaft through a connecting rod which converts the applied force into rotational torque. The basic structure of an ICE is shown in Figure 2.1.

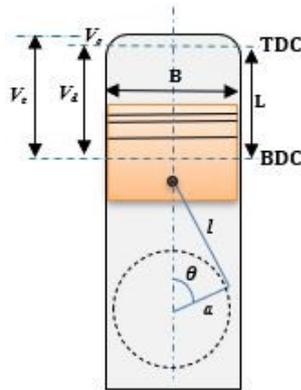


Figure 2.1: Piston-cylinder Model [1]

Modern day engines usually use a four stroke engine cycle in which power is delivered once every two rotations of the crankshaft. The four strokes of the engine are the intake stroke, compression stroke, expansion stroke and the exhaust stroke. Each stroke is equal to 180 deg rotation of the crankshaft and causes the piston to move between top dead centre (TDC) and bottom dead centre (BDC) positions. Based on the type of ignition mechanism used, engines are classified as Spark Ignited (SI) or Compression Ignited (CI) engines. In common SI engines a mixture of air and fuel is injected during the intake stroke (TDC to BDC), which is then compressed (BDC to TDC) and ignited using a spark plug. Combustion energy causes the gases to expand and pushes the piston down. Finally during exhaust stroke the burned charge is expelled out of the chamber. In a CI engine only air is initially inducted and compressed to high pressure and temperature. Ignition is then controlled by injecting highly pressurized fuel into the high pressure, high temperature environment which causes the fuel to ignite.

Ideal four stroke cycle for both SI and CI combustion are shown in figure 2.2.

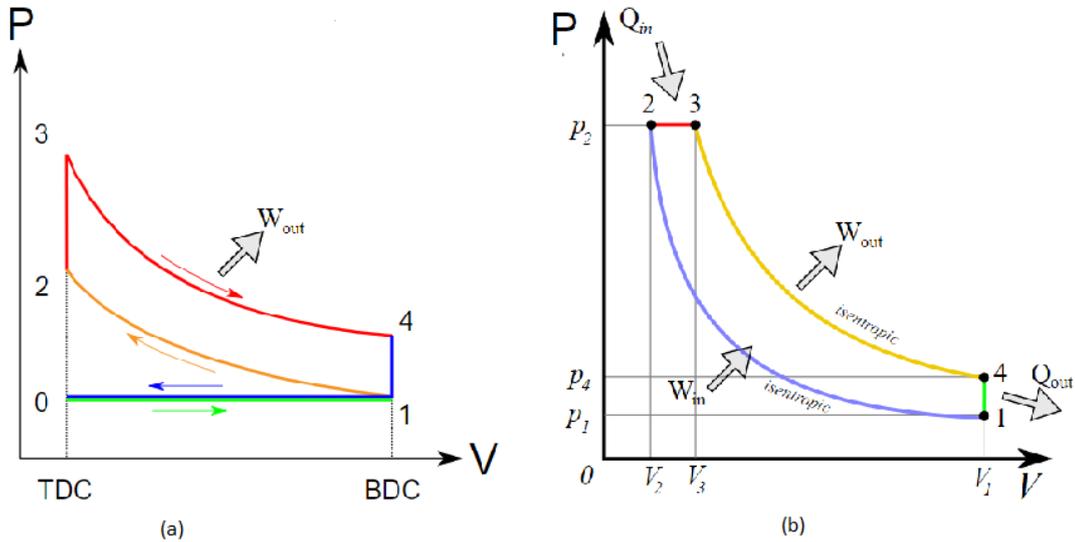


Figure 2.2: (a)Ideal Otto cycle (b)Ideal Diesel cycle [2]

Engine efficiency is dependent on a lot of factors including: the volumetric efficiency of the engine (the actual mass of air trapped inside combustion chamber as compared to theoretical mass), the quality of fuel air mixture, the combustion characteristics including the speed of combustion, the temperature produced during combustion, combustion phasing etc. The energy produced during combustion is divided into heat energy dissipated through chamber walls, the exhaust enthalpy flux, blow-by and the useful mechanical work. Figure 2.3 shows the energy balance of an engine.

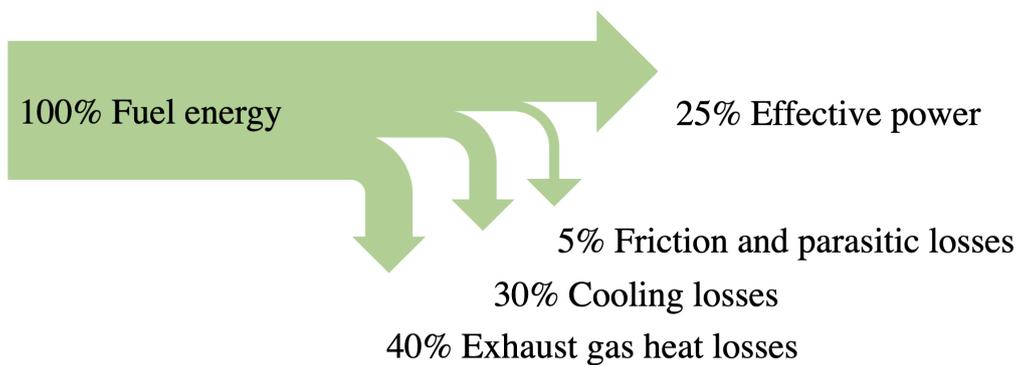


Figure 2.3: Energy balance for an ICE

Monitoring of the in-cylinder combustion and heat release is extremely important in order to increase engine efficiency. In-cylinder pressure is an useful indicator of fuel conversion efficiency and can be used along with crank angle to analyse the combustion [3]. Combustion analysis from pressure data is used to calculate the work, heat transfer, the release of chemical energy, the distribution of energy, etc. In most cases, Mean Effective Pressure (MEP) which is the work done normalised by volume, is used to compare different engines. Gross Mean Effective Pressure (GMEP) is used when work done is considered only for the compression and expansion strokes. Pumping Mean Effective Pressure (PMEP) is used when work done during intake and exhaust stroke is considered. Indicated Mean Effective Pressure is for the whole cycle i.e. the sum of GMEP and PMEP. Pressure data can be also used to estimate parameters like the TDC offset, gamma, wall temperature, blow-by, compression ratio and effective intake valve closing [3]. Experimental setup for measuring in-cylinder data is shown in figure 2.4.

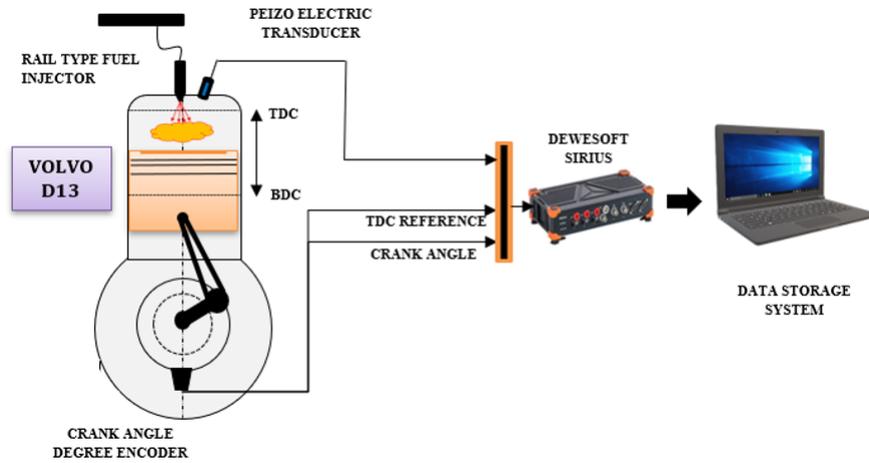


Figure 2.4: Experimental setup for measuring in-cylinder pressure and pressure derivative

The in cylinder pressure is measured using a piezoelectric transducer. The structure of the piezoelectric transducer is shown in figure 2.5. The pressure variation rate experienced by the diaphragm of the pressure transducer is transmitted to the piezoelectric crystal which generates a current proportional to its deformation [4].

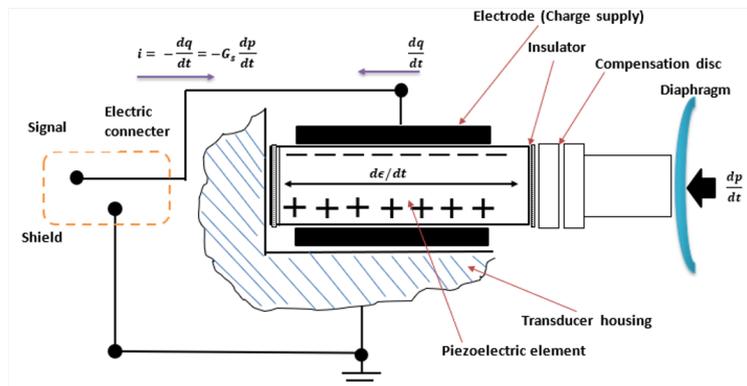


Figure 2.5: Operation of the piezoelectric transducer[4]

The charge produced by the transducer is very weak and thus needs to be amplified by an high impedance charge amplifier. The amplifier outputs a voltage proportional to the input charge, which is the time integral of the input current [4]. Pressure is related to voltage by following Equation 2.1

$$P_{out} - P_{ref} = \frac{V_{ch} * G_a}{G_s} \quad (2.1)$$

where V_{ch} is the voltage output from the amplifier, G_a is the amplifier gain, and G_s is the pressure transducer gain. Figure 2.6 shows the typical signal conditioning applied for a pressure transducer. The accuracy of the transducer is affected by the thermal strain on the sensor diaphragm caused by variable heat flux inside the chamber. This is called thermal drift and is usually divided in short term drift caused by heat flux change during a cycle and long term drift or load change drift caused by changes in transducer temperature due to engine operating points [4].

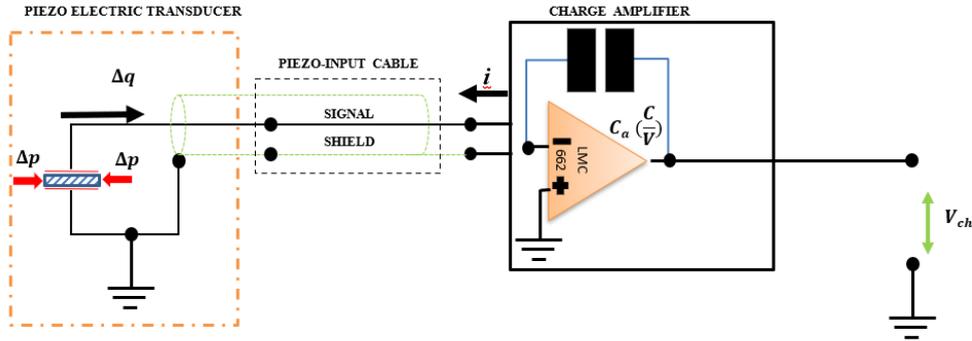


Figure 2.6: Typical signal processing for a pressure transducer [4]

The low current value of the piezoelectric transducer makes the signal prone to noise from the environment. Signal amplification can also act as a source for noise [5]. Filtering of input data is required to remove the high frequency noise content. The relative nature of the piezoelectric sensor along with thermal drift requires the need for correcting the baseline or zero level of the data. This process is called pegging or Zero Level Correction (ZLC). Crank angle encoders are used to measure the angle data of the crankshaft. The resolution of the encoder is dependent on the application and typically combustion analysis require a resolution of 0.1 degrees. Incorrect phasing of cylinder pressure and encoder data can cause errors in combustion analysis especially in IMEP calculation [3]. Piezoelectric pressure transducer and crank angle encoder are the main sensors used for performing combustion analysis. Intake and exhaust pressure sensors, air, fuel mass flow sensors can be used alongside for accurate gas exchange measurements. Fuel line pressure sensor, needle lift sensor, temperature sensor, valve lift sensor, ignition current sensor, blow-by meter and lambda sensors are some of the other sensors which can be used [3]. This project will mainly use data from in-cylinder pressure sensor and crank angle encoder. Intake manifold pressure, temperature sensors, volume flow meter and blow-by meter are also used in some analysis cases.

2.1 Filtering

Most of the data collected in an engine is noisy due to large fluctuations in pressure and temperature. This noise would interfere with calculations and offset the results due to a incorrect signal. This means that the signal will need filtering. The filter type Savitzky-Golay was recommended and used. In short, the Savitzky-Golay filter is based on a series of local low-degree polynomial fits fitted with a least square method. A user selected window is moved across the signal and within the window a least square low degree polynomial curve fit is performed. In the center of the fitted curve the now filtered data is extracted. In Figure 2.7 the process is shown. The original signal in blue is curve fitted by a 2nd order polynomial within the window, here the width is 5, and the mid-point of the polynomial is the new filtered signal. The new values for $n=3-8$ are shown. The interested reader is referred to: [6, 7]. Alternative filters that could be used are low-pass filters but would require a lot of tuning and thus not chosen.

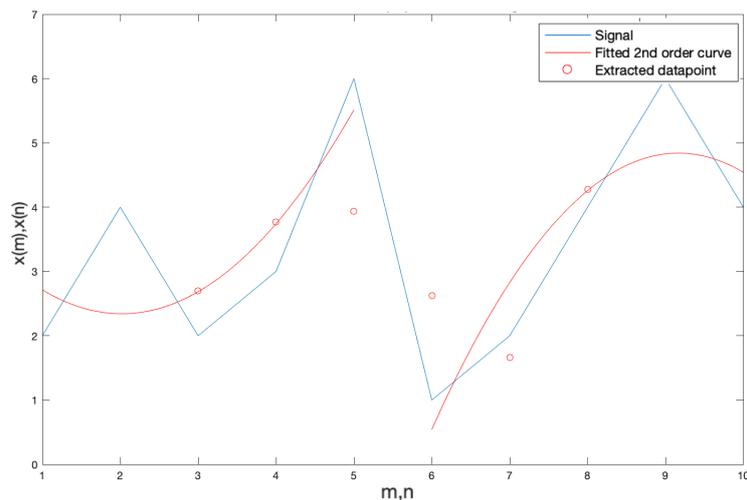


Figure 2.7: Savitzky-Golay filtering polynomial fit for $n=3,8$ is shown.

2.2 Zero Level Correction

Piezoelectric transducers can only measure the relative change in pressure due to their inherent characteristics. In order to get the absolute pressure value, it needs to be referenced to a known pressure at some point. The charge amplifier used to convert charge to voltage is prone to thermal drifts and thus needs continuous resetting. Thus raw pressure data from the sensor amplifier combination needs to be corrected for baseline. The pressure value is referenced to a known absolute pressure value at a reference crank angle. The accuracy of combustion analysis is highly dependent on the selection of correct offset method and the inputs used like the reference crank angle. There are various methods proposed in literature to

estimate this pressure offset and usually falls within two categories:

- Referencing relative pressure to a measured or known pressure at a suitable crank angle. Fixed and Measured correction methods fall under this category.
- Estimation of pressure offset using thermodynamic properties. Thermodynamic 2-point and Least Square method (LSM) are under this category.

2.2.1 Fixed

This is the simplest way of performing pressure correction. Using this method the pressure value is set to a defined (static) value at a specified crank angle position [5]. This method can be used when there is not much pressure drop in the intake side or a good estimate of in-cylinder pressure is already present. It is a simple method which is not always correct.

2.2.2 Measured

In this method the cylinder pressure is referenced to the intake manifold absolute pressure at a certain point usually when the flow can be considered to be balanced. This point is generally taken as intake BDC but can be different for highly tuned engines or engines operating at high speed [3]. This method has the advantage of being simple and fast but require the presence of an additional sensor. The location of the intake manifold sensor, reference timing and sensor noise are the main sources of error for this method [3].

2.2.3 Thermodynamic 2-point

In this method the pressure offset is calculated by measured cylinder pressure data at two points. The points are considered after intake valve closing and before start of combustion so that the compression process can be considered polytropic [3] i.e.

$$p \cdot V^n = constant \quad (2.2)$$

Let P_1 and P_2 be the two points. Then

$$p_2 = p_1 \cdot \left(\frac{V_1}{V_2}\right)^n \quad (2.3)$$

and

$$p_{measured} = p_{actual} + p_{offset} \quad (2.4)$$

$$\Delta p = p_{2,measured} - p_{1,measured} = p_{2,actual} - p_{1,actual} \quad (2.5)$$

From equations 2.3, 2.4 and 2.5 p_{offset} can be calculated as

$$p_{offset} = p_{1,measured} - \frac{p_{2,measured} - p_{1,measured}}{\left(\frac{V_1}{V_2}\right)^n - 1} \quad (2.6)$$

This method assumes the coefficient gamma to be constant which is required as an input for this method. Incorrect estimated of gamma can lead to errors in pressure offset.

2.2.4 Least Square Method

This method uses multiple points between intake valve closing and start of combustion to fit a linear regression curve and estimate pressure offset. Using a linear regression reduces the impact of signal noise and error in individual measurement. The measured pressure is a sum of actual pressure and a pressure offset. The pressure offset remains constant for different measurements.

$$p_{measured} = p_{actual} + p_{offset} \quad (2.7)$$

Considering a reference point, having parameters p_{ref} and V_{ref} , then it can be said that

$$p_{measured} = \left[\frac{V_{ref}}{V_{measured}} \right]^\gamma * p_{ref} + p_{offset} \quad (2.8)$$

For a single point it can be written in a matrix form as shown below

$$p = \begin{pmatrix} 1 & c \end{pmatrix} \cdot \begin{pmatrix} p_{offset} \\ p_{ref} \end{pmatrix} \quad (2.9)$$

For multiple points the above equation can be extended as

$$\begin{pmatrix} p_1 \\ p_2 \\ p_3 \\ p_m \end{pmatrix} = \begin{pmatrix} 1 & c_1 \\ 1 & c_2 \\ 1 & c_3 \\ 1 & c_m \end{pmatrix} \cdot \begin{pmatrix} p_{offset} \\ p_{ref} \end{pmatrix} \quad (2.10)$$

which is a set of linear equation in the form of

$$Y = X \cdot w \quad (2.11)$$

where $Y = \begin{pmatrix} p_1 \\ p_2 \\ p_3 \\ p_m \end{pmatrix}$, $X = \begin{pmatrix} 1 & c_1 \\ 1 & c_2 \\ 1 & c_3 \\ 1 & c_m \end{pmatrix}$ and $w = \begin{pmatrix} p_{offset} \\ p_{ref} \end{pmatrix}$.

w and thus P_{offset} can be calculated by the following equation

$$w = (X'X)^{-1}X'Y \quad (2.12)$$

This method assumes that gamma is constant and is known beforehand. Also matrix inversion for multiple points can be computationally expensive. Around 15 points is a good trade-off between reducing signal noise and computational speed.

2.3 Rate of Heat Release

Combustion of the air-fuel mixture inside the engine cylinder is the main process that controls engine efficiency, power, and emissions. As shown in Figure 2.8 below combustion of the air-fuel mixture is an exothermic reaction for which the outputs are high pressure and temperature inside the engine cylinder. The chemical energy of the fuel is the sum of internal energy(dU_s), mechanical work(δW), heat transfer rate(dQ_{ht}), and energy loss through mass flow(dQ_{BB}) [8]. For an accurate combustion analysis each of the above processes must be studied.

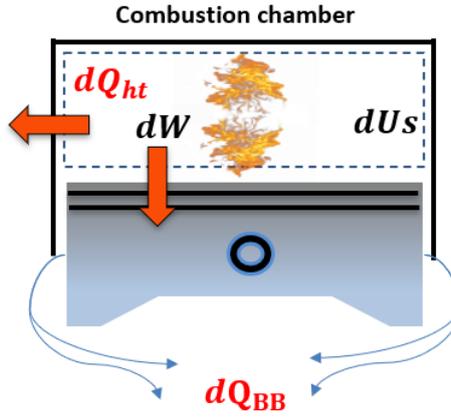


Figure 2.8: Combustion inside the engine cylinder

Rate of Heat Release [8] is a thermodynamic approach based on the first law of thermodynamics. The major advantage of using this approach is the change in pressure can be directly correlated to fuel energy released by combustion. Heat release as a function of crank angle degree can be derived by applying the first law of thermodynamics to a controlled volume and assuming energy loss through mass flow is zero ($dQ_{BB} = 0$). Then heat energy released by the combustion (dQ_{ch}) is given by:

$$dQ_{ch} = dU_s + dW + dQ_{ht} \quad (2.13)$$

This can be represented as:

$$dU_s = mC_v dT \quad (2.14)$$

$$dW = p dV \quad (2.15)$$

$$pV = mRT \implies mT = \frac{pV}{R} \quad (2.16)$$

Differentiating equation 2.16 gives:

$$m dT = d\left(\frac{pV}{R}\right) \quad (2.17)$$

Further:

$$mc_v dT = \frac{c_v}{R}(pdV + Vdp) \quad (2.18)$$

substituting in the equation 2.13 gives:

$$dQ_{ch} = \frac{c_v}{R}(pdV + Vdp) + pdV \quad (2.19)$$

Further, expression for Apparent rate of heat release as function of crank angle degree is given by,

$$\implies \frac{dQ_{ch}}{d\theta} = \frac{c_v}{R} \cdot V \cdot \frac{dp}{d\theta} + \left(\frac{c_v}{R} + 1\right) \cdot p \cdot \frac{dV}{d\theta} \quad (2.20)$$

Now, energy loss by heat transfer is given by the equation,

$$\frac{dQ_{ht}}{dT} = Ah_c(T - T_w) \quad (2.21)$$

Energy loss due to mass transfer i.e. crevice and blow-by losses which is assumed zero due to the fact that it is very small, but in reality cannot be zero and given by,

$$dQ_{BB} = \left(\frac{\gamma}{\gamma - 1}\right) \cdot p \cdot V \cdot \left(\frac{dm}{m}\right) \quad (2.22)$$

Hence, expression for real heat release rate as function of crank angle degree is given by,

$$\implies \frac{dQ_{ch}}{d\theta} = \frac{c_v}{R} \cdot V \cdot \frac{dp}{d\theta} + \left(\frac{c_v}{R} + 1\right) \cdot p \cdot \frac{dV}{d\theta} + A \cdot h_c \cdot (T - T_w) + dQ_{BB} \quad (2.23)$$

Rewriting Equation (2.23) in terms of ploytropic coefficient:

$$c_p = c_v + R \quad (2.24)$$

and

$$\gamma = c_p/c_v \quad (2.25)$$

$$\implies \frac{dQ_{ch}}{d\theta} = \frac{1}{\gamma - 1} \cdot V \cdot \frac{dp}{d\theta} + \left(\frac{\gamma}{\gamma - 1}\right) \cdot p \cdot \frac{dV}{d\theta} + A \cdot h_c \cdot (T - T_w) + dQ_{BB} \quad (2.26)$$

From Figure 2.1 the change in volume with respect to crank angle degree can be calculated by the following relation [1, 8]:

$$V = \frac{V_d}{rc - 1} + \left(\frac{V_d}{2}\right) \cdot [RL + 1 - \cos\theta - (RL^2 - \sin^2\theta)^{\frac{1}{2}}] \quad (2.27)$$

From the geometrical parameters of the engine, the displacement volume (V_d) can be determined by the equation:

$$V_d = \frac{\pi}{4} * B^2 * L \quad (2.28)$$

RL is the ratio of connecting rod length to crank radius and calculated as:

$$RL = \frac{l}{a} \quad (2.29)$$

Differentiating the equation(2.24) with respect to crank angle gives:

$$\frac{dV}{d\theta} = \frac{V_d}{2} \cdot \sin\theta [1 + \cos\theta - (RL^2 - \sin^2\theta)^{\frac{-1}{2}}] \quad (2.30)$$

Further, determination of change in pressure with respect to crank angle degree can be determined by two different methods:

(1)Finite difference method, and

$$\frac{dp}{d\theta_j} = \frac{p_{j+n} - p_{j-n}}{2 * \Delta\theta} \quad (2.31)$$

Where, $\Delta\theta$ is sampling step between two crank angle degrees

(2) Savitzky-Golay filtering

Calculating ($\frac{dP}{d\theta}$) by finite difference method does not give accurate values due to uncertainties i.e. as the pressure transducer is induced to thermal stress the recorded signal is very noisy. But, with Savitzky-Golay filter the pressure signal can be filtered and noise can be reduced to a large extent and proves to be more suitable for this application than the finite difference method is.

To determine the rate of heat release accurately, an in-depth analysis was required for the parameters which have a major effect on the rate of heat release. These parameters are:

- 1) Gas properties
- 2) Effective Intake valve closing (IVC) and Exhaust valve opening (EVO)
- 3) Heat transfer rate
- 4)Wall temperature
- 5) Blow-by

All parameters are investigated and explained in detail below.

2.4 Gamma

In several of the equations governing the operating of an internal combustion engine the polytropic coefficient appears. This coefficient has different names, sometimes being referred to as gamma, kappa, polytropic coefficient or polytropic index. For ease of reading this will from now on be called gamma or γ . This coefficient is a result of the ideal gas law acting in an isentropic process[9]:

$$p * V^\gamma = \text{constant} \quad (2.32)$$

or:

$$p_1 * V_1^\gamma = p_2 * V_2^\gamma \quad (2.33)$$

The coefficient can be calculated in different ways. In this report logarithmic regression and NIST polynomial will be explained in detail. The value of gamma is constantly changing due to several factors, including but not limited to temperature, gas composition and pressure[10]. Thus the common method of using a fix gamma of 1.35 or 1.4 [8, 11] is not optimal and a method that can take these parameters into account is beneficial. A gamma of 1.4 is often used, this is derived from Equation 2.34. As air is mostly a diatomic gas $f=5$ [11].

$$\gamma = \frac{f + 2}{f} = \frac{5 + 2}{5} = 1.4 \quad (2.34)$$

2.4.1 Logarithmic regression

The logarithmic regression method uses thermodynamics to analytically calculate the instantaneous gamma. The regression is based on Equation 2.33 above, and can be rearranged as follows:

$$\ln(p_1) * \ln(V_1^\gamma) = \ln(p_2) * \ln(V_2^\gamma) \quad (2.35)$$

$$\ln(p_1) * \gamma * \ln(V_1) = \ln(p_2) * \gamma * \ln(V_2) \quad (2.36)$$

$$\ln(p_1) - \ln(p_2) = \gamma * \ln(V_2) - \gamma * \ln(V_1) \quad (2.37)$$

$$\gamma = \frac{\ln(p_1) - \ln(p_2)}{\ln(V_2) - \ln(V_1)} \quad (2.38)$$

This equation should give the instantaneous gamma value between two sample points. From the data acquisition all of the input data is known. This equation can be used during the compression and expansion strokes when valves are closed and combustion is not occurring, as the composition changes during combustion.

2.4.2 NIST

A model for finding the c_p of a mixture of compounds as a function of temperature was first introduced by C. Howard Shomate in 1944 [12]. This method is based on experimental measurements of heat change in compounds and defining several coefficients. With these coefficients (A-E) the function describing c_p is:

$$c_p = A + B * T + C * T^2 + D * T^3 + E/T^2 \quad (2.39)$$

The coefficients are available in large quantities for the majority of simpler and more common compounds. In this report the tables made public by The National Institute of Standards and Technology, U.S. Department of Commerce[13] were used. Equation 2.39 determines the c_p of a singular substance [14]. These coefficients can then be added to create a combined c_p for a mixture of compounds as stated in Equation 2.40.

$$c_{p,tot} = \sum_{i=1}^n \frac{mol_i}{mol_{tot}} * c_{p,i} \quad (2.40)$$

The γ can then be calculated by:

$$c_{v,tot} = c_{p,tot} - \frac{R}{M_m} \quad (2.41)$$

Where M_m is the weighted averaged molar mass. And:

$$\gamma = \frac{c_p}{c_v} \quad (2.42)$$

Equation 2.39 requires the instantaneous temperature for accurate calculation. From [9]:

$$pV = nRT \quad (2.43)$$

If constant mass is assumed in the system [9]:

$$\frac{pV}{T} = nR = \frac{p_{ref}V_{ref}}{T_{ref}} \quad (2.44)$$

Equation 2.44 rearranges to:

$$T = T_{ref} * \frac{p * v}{p_{ref} * v_{ref}} \quad (2.45)$$

2.4.2.1 Gas composition

When calculating the NIST polynomial (sometimes referred to as Shomate polynomial) knowing the correct gas-composition is necessary as this is the basis of the calculation. The exact composition of the in cylinder mixture is very hard to model as the introduction of the fuel and its combustion results in a lot of intermediate compounds [8]. Thus, it needs to be simplified. Two modes are taken into account, dry air and exhaust gas. The composition of gas can then be controlled by a weighting factor between the two compositions. These compositions are presented in Figure 2.9 and 2.10 [15, 16]. Due to the complexity and small percentage of the unburned hydrocarbons it is assumed to consist of the most common: C₂H₄ [17].

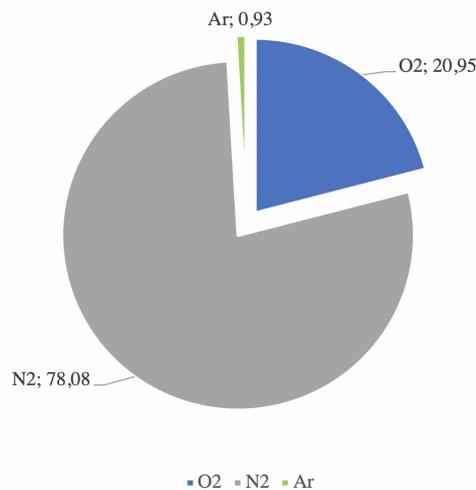


Figure 2.9: Dry air, mole percentage of composition gases.

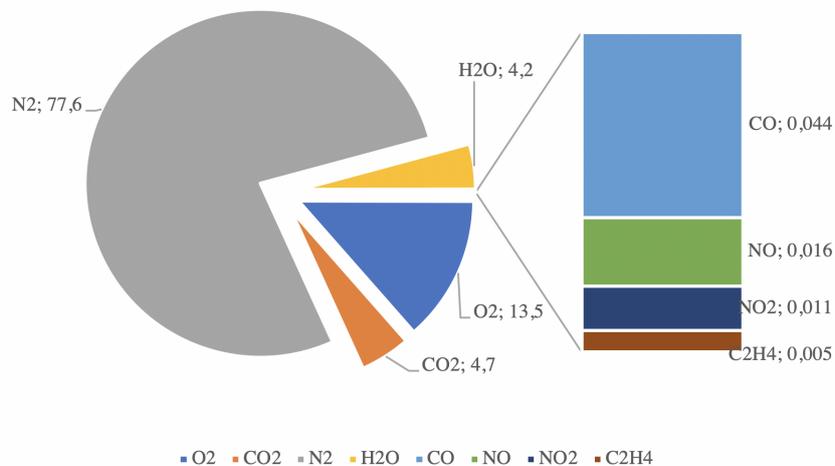


Figure 2.10: Diesel exhaust, mole percentage of composition gases.

2.4.3 Ganippa proposal

To explore alternative methods of estimating gamma a search of literature was conducted. A proposed method presented by G. Abbaszadehmosayebi and L. Ganippa[18] was explored. Based on:

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta} \quad (2.46)$$

the $Ahrr_{50}$ (apparent heat release rate at CA50) can be expressed as:

$$Ahrr_{50} = \frac{\gamma}{\gamma-1} \left(p \frac{dV}{d\theta} \right)_{50} + \frac{1}{\gamma-1} \left(V \frac{dp}{d\theta} \right)_{50} \quad (2.47)$$

Gamma can then be extracted to create:

$$\gamma = \frac{Ahrr_{50} + \left(V \frac{dp}{d\theta} \right)_{50}}{Ahrr_{50} - \left(p \frac{dV}{d\theta} \right)_{50}} \quad (2.48)$$

Substituting γ from 2.48 in 2.46, $Ahrr$ can be expressed as:

$$Ahrr = Ahrr_{50} * K_1 + K_2 \quad (2.49)$$

Where:

$$K_1 = \frac{V \frac{dp}{d\theta} + p \frac{dV}{d\theta}}{\left(V \frac{dp}{d\theta} \right)_{50} + \left(p \frac{dV}{d\theta} \right)_{50}} \quad (2.50)$$

and

$$K_2 = \frac{\left(V \frac{dp}{d\theta} \right)_{50} * p \frac{dV}{d\theta} - \left(p \frac{dV}{d\theta} \right)_{50} * V \frac{dp}{d\theta}}{\left(V \frac{dp}{d\theta} \right)_{50} + \left(p \frac{dV}{d\theta} \right)_{50}} \quad (2.51)$$

This equation can be rewritten to create the Residual Function RF:

$$RF = \left(\frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta} \right) - (Ahrr_{50} * K_1 + K_2) \quad (2.52)$$

Where, according to the authors, the correct gamma is found when RF is minimized. This method was pursued but did not work as presented. For further discussion see section 4.4.3.

2.5 Heat Transfer

During the combustion phase inside the cylinder, the temperature of the burned gasses reach high temperatures[8]. Consequently the differences in heat are large and must be taken into account. The high heat flux that is generated during the combustion phases plays an important role because these high temperatures deteriorates the lubricating oil film, overheating of the exhaust valves etc. The heat transfer plays an important role in the engine's performance, efficiency and emissions. The effect of heat transfer is also significant for the rate of heat release study, because neglecting heat transfer predicts that combustion will takes place faster than it would with it taken into account. In order to predict these heat fluxes in the cylinder various models have been presented over the years. In this project some of these models will be explored to see which model among the proposed models is best suited for CI/SI engines [8].

2.5.1 Woschni

The heat transfer between the charge and the wall takes place by convection and radiation. Since the heat transfer is dominated by the convection, the radiation effect is neglected because the calculation is complex and the additional influence is small [19, 8]. According to Newton's law of cooling the convective heat transfer can be expressed in the following equation as [20]:

$$\frac{dQ_{ht}}{dt} = h_c * A(T - T_w) \quad (2.53)$$

Where h_c is the convective heat transfer coefficient, A is the area, T is cylinder gas temperature and T_w is the wall temperature. In Equation 2.53 the total area of the cylinder wall at any crank angle is calculated using the following equation:

$$A = \frac{\pi}{2} * B^2 + \Pi * B * \frac{L}{2}[l + 1 - \cos\Theta + (l^2 - \sin\Theta)^{\frac{1}{2}}] \quad (2.54)$$

The Woschni heat transfer coefficient, h_c , is taken as the characteristic length and the characteristic velocity is the mean gas velocity in the cylinder [8, 21].

$$h_c = 3.26 * B^{-0.2} * p^{0.8} * T^{-0.55} (2.28 * W)^{0.8} \quad (2.55)$$

In Equation 2.55, W is the average gas velocity and is calculated using:

$$W = [C_1 * S_p + C_2 \frac{V_d * T_r}{p_r * V_r} (p - p_m)] \quad (2.56)$$

In the above equation C_1 and C_2 are constants that vary depending on the combustion period. The values are shown below in Table 2.1 [8, 21].

Table 2.1: Woschni coefficients

Phase	C_1	C_2
Intake-Exhaust	6.18	0
Compression	2.28	0
Combustion-Expansion	2.28	0.00324

The motoring pressure in Equation 2.56 is calculated using Equation 2.57.

$$p_m = p_r(V_r/V)^n \quad (2.57)$$

2.5.2 Hohenberg correlation

Hohenberg suggested a correlation in 1979. He examined Woschni formula and made few changes. There are quite a few differences with Woschni heat transfer coefficient equation and Hohenberg heat transfer coefficient equation. First of which is that, for characteristic length instantaneous cylinder volume is used instead of bore, and the temperature exponent is changed from -0.55 to -0.4. The Hohenberg equation is stated below

$$h_c = 3.26 * V^{0.06} * P^{0.8} * T^{e-0.4}(S_p + 1.4)^{0.8} \quad (2.58)$$

2.5.3 Wall Temperature

The wall temperature is defined as the temperature of the cylinder liner towards the combustion chamber. This is important when a heat analysis is to be carried out, and the analysis will be more accurate if this value is as close to reality as possible.

The first and simplest method is to use a fixed value. This is a fast method and can work if the value is known and close to reality. The problem is that this value often is a guess, or a more or less good estimation based on earlier knowledge. Another problem is that the wall temperature most likely changes depending on load and speed. Therefore, a value that is calculated for the actual load point is more accurate. To do this the cooling water temperatures and flow is used. The temperature of the inlet and outlet water needs to be known as well as the cooling water flow and c_p value for the fluid. Thereafter the energy taken away from the cylinders can be calculated by Equation 2.59.

$$\dot{Q} = \dot{m}c_p\Delta T = \dot{m}c_p(T_{out} - T_{in}) \quad (2.59)$$

$$T_{wall} = \left(\frac{\dot{Q}}{hA}\right) + T_{out} \quad (2.60)$$

The outgoing water can also be assumed to have approximately the same temperature as the water surrounding the cylinders. The simplest way now is to use this value. The next step for an even better assumption is to use the material data of the cylinder liner. This method works best for engine blocks that have an open deck design. This is because most of the cylinder liner is in contact with the water, compared to closed deck design. Now the heat flow can be calculated through the cylinder liner. Since the material is known as well as the outside temperature, cylinder geometry and heat carried away with the cooling water the inside temperature can be calculated. This is done by the help of equation 2.60, and gives the wall temperature. All these calculations are made for steady state cases to make sure the temperatures have stabilized.

2.6 Effective Intake Valve Closing

The intake valve closing is often given from the manufacturer. This IVC however is not always the actual closing. The given value can be a geometrical value, often based on the cam geometry. If this value is used for heat release calculation the answer will most likely differ from reality. The intake valve closing determines how much charge that is trapped in the cylinder. Therefore, this value is vital for knowing how much charge actually is inside of the cylinder during combustion. The value for the IVC that can be used instead is called IVC effective, and is calculated by the help of a logarithmic regression analysis of the in cylinder pressure and volume.

To calculate the IVC effective a logarithmic regression analysis is made. How the logarithmic regression is done can be seen in section 2.4.1 The in cylinder pressure and cylinder volume with associated CAD is used. First the CAD for the given geometrical IVC is used to know when to start looking for the effective IVC. Based on this a value of how many CAD:s after this the analysis should include is estimated, and a maximum limit. Thereafter the logarithm of the volume and pressure is used to make a linear regression, where the pressure trace deviates more than a specified limit is the effective IVC. With other words where the plot makes a “bend”. This is illustrated in figure 2.11.

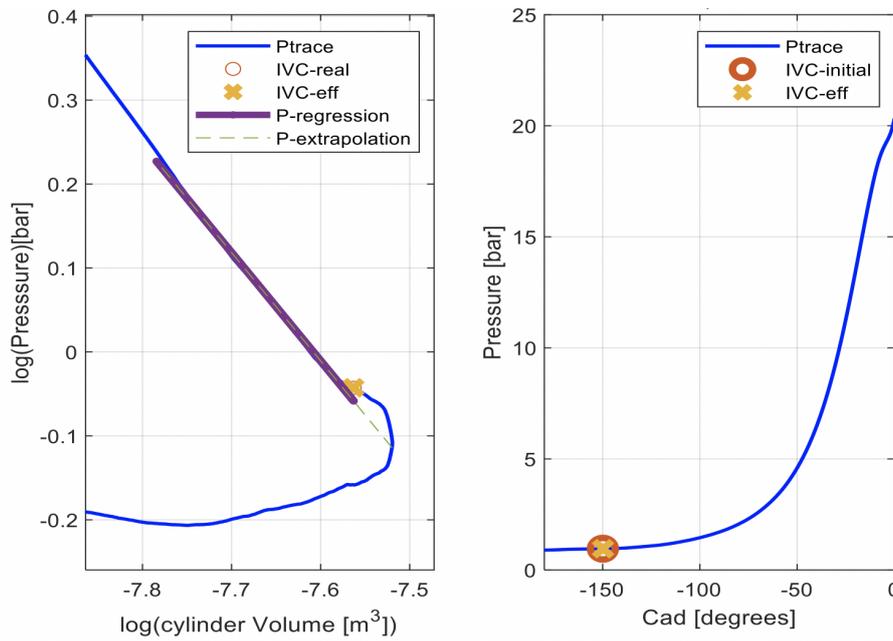


Figure 2.11: Left: logarithmic pv-plot. Right: in-cylinder-pressure vs CAD.

2.7 Blow-by

Despite the accurate construction of piston and cylinder assembly, the engine combustion chamber will have small volumes/ gaps which are known as crevices. The largest crevices are between piston, piston rings, and the cylinder wall. During the engine operating cycles, due to pressure differences, some amount of gas mixture will escape from these volumes into the crankcase and this phenomenon is known as blow-by [8][3].

Most of the blow-by occurs during compression and expansion stroke as shown in the Figure 2.12 below.

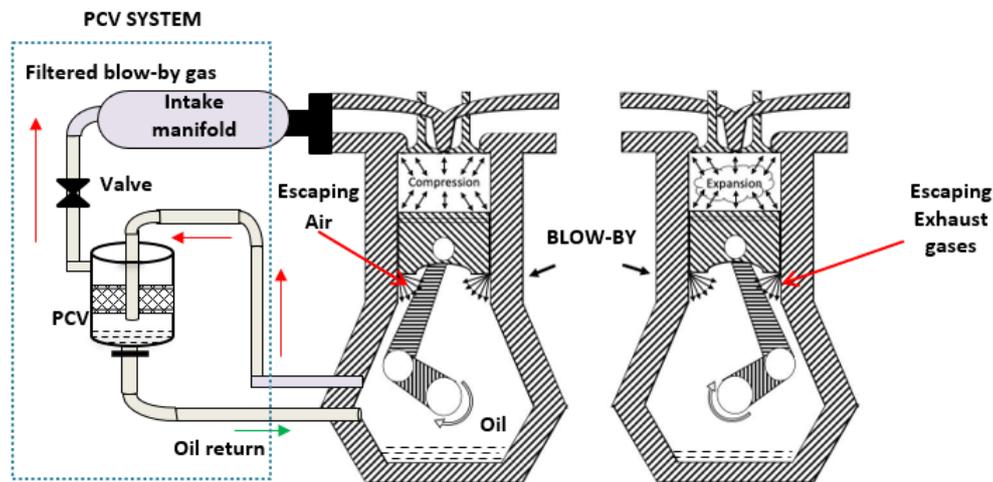


Figure 2.12: Blow-by During compression and expansion stroke with Crankcase ventilation system (PCV)

Even though its not optimal, earlier, the blow-by gas mixture used to be vented in the atmosphere, but now due to more stringent emission regulations, it has to be stopped. Because the blow-by gas consists of unburned hydrocarbons this type of emission needed to be addressed. In addition to this due to blow-by, the power generated and efficiency of the engine is reduced as some amount of air-fuel mixture escapes, also, the fuel vapors escaping the cylinder can mix with the oil in the crankcase, thus compromising the lubrication property of the oil, which causes more friction and frictional wear of engine components.

To avoid venting the gas mixture in the environment a system know as a positive crank case ventilation system (PCV) is introduced as shown in above picture 2.12. The gas mixture accumulating in the crankcase is passed through the PCV, in which the oil and other impurities are separated (i.e.filtered). And then the filtered gas is recirculated back to the intake manifold which contributes to the improvement of engine efficiency. The disadvantage of PCV is the gas mixture filtered is not 100 percent clean, It forms a smudge layer in the intercooler and the intake manifold thus hampering the engine parts[22].

Over the last years a lot of research have been carried out, but there is no established mathematical equation defined from which blow-by mass can be determined accurately, because blow-by mass generated depends on several factors such as engine speed, load and BMEP. It is therefore difficult to generalise for all engine types. It also depends on engine wear properties which is very difficult to predict and it can generally be said that the older the engine the higher the blow by rate [23][24].

In order to determine blow-by accurately, the only way is to install blow by meter and measure the values for that particular engine as blow-by intensity will be different for every engine.

A way to simplify the blow-by phenomenon it can be assumed as a choked flow through a small orifice [25].

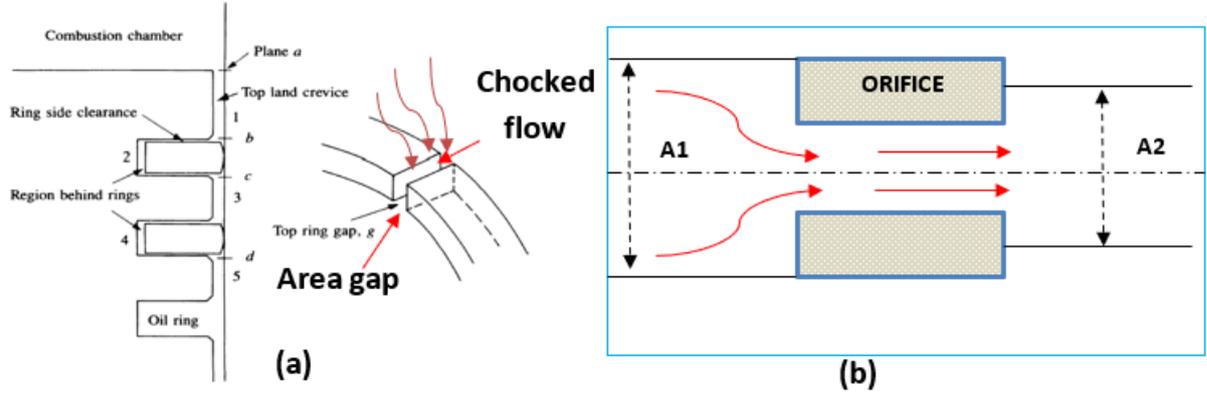


Figure 2.13: (a)Piston and rings assembly, (b) Flow through orifice [8]

The blow-by flow velocity for non-choked flow can be determined by[25]:

$$w_{BB} = \sqrt{2 * \frac{\gamma}{\gamma - 1} * \frac{R}{M} * T [1 - (\frac{p_0}{p})^{(\frac{\gamma-1}{\gamma})}]} \quad (2.61)$$

and blow-by flow velocity for choked flow can be determined by:

$$w_{BB} = \sqrt{2 * \frac{\gamma}{\gamma - 1} * \frac{R}{M} * T} \quad (2.62)$$

The maximum mass flow occurs when the flow velocity at the minimum area (A_{gap}) reaches the speed of sound, and this condition is known as choked flow. which is calculated by introducing the Mach number (M)[8].

$$M = \frac{V}{a}; \text{ where, } a = \sqrt{\gamma * R * T} \quad (2.63)$$

The real mass flow rate can then be calculated by:

$$dm = w_{BB} * A_{gap} * d\theta * N^{-1} \quad (2.64)$$

2.8 Compression Ratio

The compression ratio is defined as the geometrical volume ratio between the volume at BDC and TDC. This is however not always the actual compression ratio. This is due to manufacturing tolerances, piston and valve sealing as well as porosity of the materials. The compression ratio is vital to get a correct heat analysis. Therefore the effective compression ratio is needed.

The effective compression ratio is calculated by using the in-cylinder pressure. The data needs to be collected during motoring of the engine, which means fuel cut and no combustion occurring in the cylinder. This can be made in two ways. Either by letting a motor, for example the starter motor, power the engine, or by running the engine and then do a fuel cut (engine breaking). The benefits by doing it with engine breaking is that the compression ratio can be calculated for different speeds and with different heat levels. This means that for example the piston ring sealing is taken to account, which can vary. The drawback is that the engine is slowing down while the data is collected, which means more unstable data.

Equation 2.65 shows the formula for calculating the compression ratio. The displacement volume is already known from the geometry of the crank. What's unknown is the clearance volume. To calculate this, equation 2.66 and 2.67, must first be analysed. Since there is no combustion occurring the losses are only dependent on losses to the wall and blow-by, hence the equations. When this is known the clearance volume can be calculated with the use of equation 2.68 [26]. γ for air is used at the given temperature. The wall temperature and blowby are already calculated and are therefore not calculated again.

$$r_c = \frac{V_{max}}{V_{min}} = \frac{V_d + V_c}{V_c} \quad (2.65)$$

$$dQ_n = \frac{\gamma}{\gamma - 1} p dV + \frac{1}{\gamma - 1} V dp \quad (2.66)$$

$$dE_{loss} = dQ_{wall} + dH_{blowby} \quad (2.67)$$

$$V_c = \frac{-\gamma p dV - V dp + (\gamma - 1) dE_{loss}}{dp} \quad (2.68)$$

The code gives the user flexibility in choosing different approaches and methods for data processing and analysis purposes. The data coming from Dewesoft is sampled in time domain. The code allows the user to continue the analysis in the time domain or convert it into CAD domain of specific resolution. The user also has the flexibility of choosing to run the analysis for multiple cycles or using an average cycle. Figure 3.1 shows the interaction of the various functions.

3.1 Test engines

Data was collected from three different engines in this project, to develop and validate the code. All those engines had different parameters and sizes, two were compression ignited engines and one spark ignited. The most important data for those engines are stated below.

- A 12.8 liter, in-line, 6 cylinder turbo diesel engine with common rail fuel injection, over head camshaft and 4 valves per cylinder.
- A 2 liter, in-line, 4 cylinder, twin-turbo charged SI engine with direct injection, dual over head camshafts and 4 valves per cylinder.
- A 2 liter, single cylinder, turbo diesel engine with common rail fuel injection, over head camshaft and 4 valves per cylinder.

These engines were then run at different operating points as data was collected. The reason for using three different engines was to validate that the code was working, and to test it for both SI and CI engines.

3.2 Filtering

The filtering of the data is done using a Savitzky-Golay filter. This is implemented in Matlab by using the function `sgolay` to calculate the convolution coefficients and the function `conv` is used to produce the filtered signal and its derivatives.

3.3 Zero level correction

The code `zero_level_correction` takes structs of cylinder pressure and the engine data as inputs along with the correction method and returns a struct containing the offsetted pressure data. Pressure values are offsetted for each individual cycle ensuring that the effects of long term drift are minimised. The code offers four different correction methods. The user is asked to enter specific inputs for the selected method. For the fixed method user needs to input the reference CAD and the static pressure value. The measured value methods requires reference CAD at which the intake manifold pressure will be referenced to cylinder pressure. Thermodynamic 2-point asks the user for two reference points and the polytropic coefficient. The function calculates the pressure offset based on equation 2.6. Least square method asks the user the reference CAD and the gamma value.

3.4 Gamma

The code gasProp was created do to give the user the ability to select between three different types of Gamma calculation. The first method lets the user decide values for bot c_p and c_v to calculate gamma. The second and third applies the methods outlined in Section 2.4.1 and 2.4.2.

3.4.1 Logarithmic regression

The logarithmic regression method applies Equation 2.38 at different indices and calculates gamma. As the pressure and volume of the system is known the equation can be used to calculate mean values for compression by using IVC and SOC as indices and EOC and EVO as indices. The equation can also be applied in a time resolved manner and by a moving window calculate the instantaneous gamma by shifting the indices and evaluating gamma over short change in pressure and volume.

3.4.2 NIST

Prior to calculating c_v for any substance the instantaneous temperature is required. This is calculated using Equation 2.45 where the reference is set to the IVC as the pressure and temperature is known from measurements in the intake manifold. The gas composition at IVC can either be assumed air or a mixture of air and exhaust gas to simulate EGR. The transformation of the gas composition from air/air-EGR mix to pure exhaust gas is best approximated with MFB and 1-MFB respectively. This model does not take in-combustion gases into account.

3.5 Effective Intake Valve Closing

To determine the Effective IVC the code "IVC Effective" was created. This code needs the following inputs: Pressure, volume and CAD. The format this should be inputted in is that the vectors of those data sets needs to be of the same size, and all three needs to be from the same datapoint. Meaning that for every CAD there is corresponding volume and pressure values. When this data is inputted a logarithmic regression analysis can be made to determine the effective IVC, as explained in section 2.6.

3.6 Compression ratio

The code "Effective compression ratio" was created. This code needs motoring data as input. The input from this data is the pressure and volume along with the CAD. Both the actual volume and pressure as well as the deltas for those are needed. Furthermore gamma for air, from the "gamma code" are needed, the wall temperature losses and blowby losses from each accordingly code are also needed. From those inputs the actual compression ratio can be determined, by the help of equation 2.68 & 2.65.

3.7 Heat transfer models

A MATLAB code was created in which, the provision was given to select the heat transfer model among the two models. The input data necessary for calculation of heat flux for both Hohenberg and Woschni were taken from the generated engine data file. For the Woschni model the heat transfer coefficient is calculated using Equation 2.55, where the Woschni correlation constants C1 and C2 takes up different values for different phases of the combustion.

3.7.1 Wall Temperature

To calculate the wall temperature the code "Wall Temp" was created. This code needs the geometry of the cylinder liner (bore & stroke). And material data for this, the heat transfer coefficient value are taken from the Woschni calculations. This is to not build in more uncertainty. It also needs the mass flow of the cooling water, as well as the outlet and inlet temperature to the engine. The last input needed is the c_p value for the cooling liquid. From this the wall temperature can be calculated by the help of equation 2.59 & 2.60

3.8 Blow-by

To determine the blow-by of the engine a code is created based on simplified compressible flow model by using the equations 2.62 and 2.64. The inputs for the code are polytropic coefficient, gas constant, temperature, crank angle degree, piston ring gap and engine speed. For which the resulting output is mass loss due to blow-by, this mass loss is added to the main heat release equation 2.26 as dQ_{BB} because the pressure data recorded is after blow-by losses [3, 25].

3.8.1 Blow-by code validation by using experimental values

Measured blow-by data through a blow-by meter collected from the HD diesel engine. From the data a 3D graph is plotted as shown in Figure 3.2.

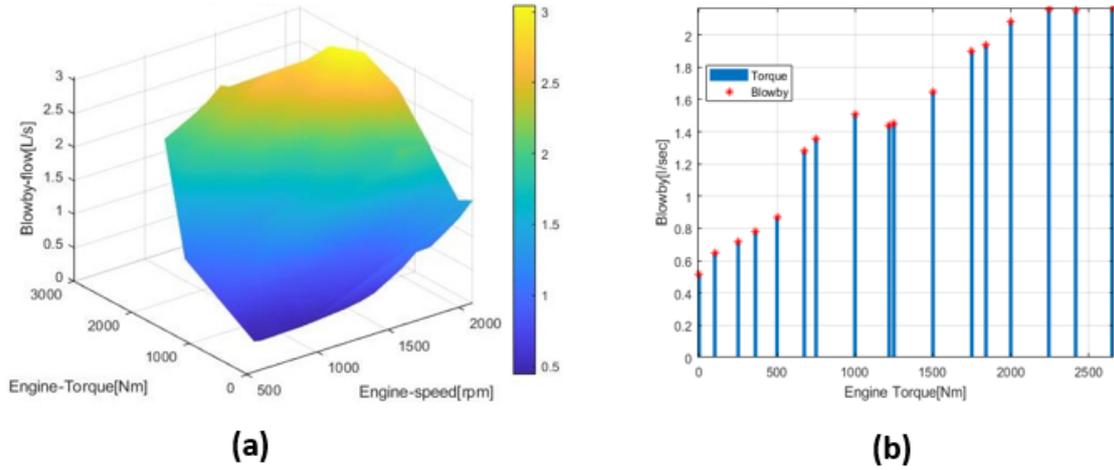


Figure 3.2: (a) Blow by measured values-for different engine speed and Torque,(b) Blow by at engine speed 1200 [RPM] and different Torques

From the above graph it can be interpreted that blow-by intensity changes with change in engine speed and Torque. Blow-by is highest at maximum rpm and torque. Data recorded with Dewesoft acquisition system on a HD diesel engine is at 1200rpm and $T=600[N/m]$ and this can be correlated to the measured data blow-by of 1.13[Ltr/sec]. The blow-by percentage can be calculated [22] for the engine with 4-stroke($n=2$), $V_t=12.8$ litres, at $N= 1200$ [rpm] [22].

$$V = (N * V_t)/n = 128[Litres/sec]. \quad (3.1)$$

$$Blow - by \text{ in percentage} = 1.13/128 * 100 = 0.882\%. \quad (3.2)$$

The Blow-by estimation model is tuned and validated with respect to the experimental values.

3.9 Rate of heat release

The filtered pressure, crank angle degree, and engine data is used as inputs and performs the rate of heat release analysis as described in the section 2.3. The rate of heat release and cumulative mass fraction burned curves are the main outputs from the code and this code will be the base and reference for comparison of the analysis.

3.10 Determination of pressure derivative

In cylinder pressure accuracy plays an important role in performing combustion analysis. A pressure transducer is installed on cylinder-2 of the engine and the experimental set up is as shown in Figure 2.4. As described earlier the pressure transducer is sensitive to outside effects and requires signal conditioning. Signals of pressure transducers can be conditioned by:

- Signal conditioning by charge amplifier
- Signal conditioning by current to voltage converter

Most of the data acquisition systems use signal conditioning by a charge amplifier, which has a major disadvantage of instability and drift [4]. As the data acquisition system previously used by the local researchers was charge amplifier method, signal conditioning by current to voltage converter is investigated further.

3.10.1 Signal conditioning by current to voltage converter

In order to measure the in cylinder pressure derivative accurately the output current from the transducer should be converted to voltage signals[4]. The values will be more precise if this is performed with a low impedance device. Current-to-voltage converter is a low impedance device which eliminates the signal distortions caused by transducer self capacitance. The construction of a current to voltage converter is simple as it only consists of two operational amplifiers and a resistor [4]. The circuit diagram is as shown in the below Figure 3.3:

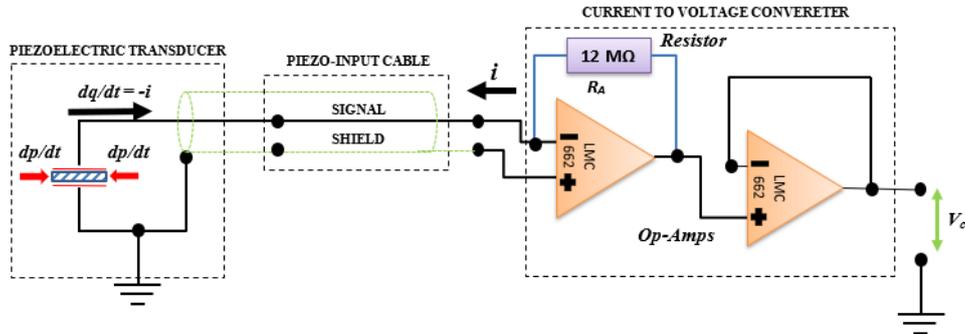


Figure 3.3: Current to voltage converter [4]

The measured pressure variation through the unit converter is given by:

$$\frac{dP}{dt} = \frac{V_{cv}}{G_s * R_a} \quad (3.3)$$

Where V_{cv} = Output voltage, G_s = Transducer gain and R_a =Resistance value. Further, pressure data can be determined by fourth order Runge-Kutta numerical integration method [4]. The current to voltage converter device was built in house calibrated and tested for recording the in-cylinder pressure data.

4

Results and Conclusion

In this chapter results and conclusions for each separate sub-code will be presented. The results will be chosen in a way to show a representative measure of the effect of the process. All measures will be presented in the Appendix for further reference. In conjunction with with the results a short specific conclusion will be presented. For a overall discussion, see Chapter 5.

4.1 Filtering

The filtering of the pressure signal by Savitzky-Golay is carried out using 5th order filter and a window size of 301. This is presented below in Figure 4.1.

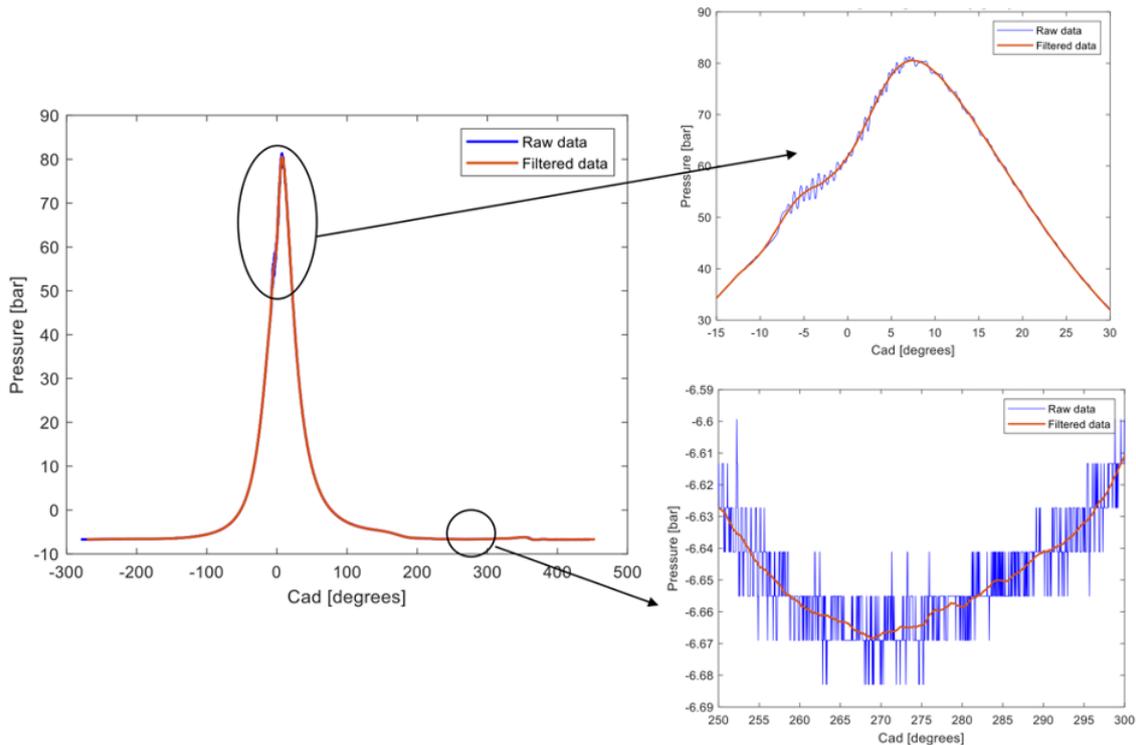


Figure 4.1: The visual effect of the filtering process.

To compare the effectiveness of the filter the difference in IMEP was calculated between unfiltered and filtered pressure data, and the total mean difference of -3.8147×10^{-6} Pa, which represents a percentage difference of $4.8776 \times 10^{-5}\%$.

The effect of the filtering process is advantageous, the signal is smoothed and any high derivative is smoothed down to reasonable levels. When comparing the difference between an unfiltered and filtered IMEP only a small difference is seen. This indicates that the analysis following the filtering isn't affected by the small difference between the signals due to filtering. The difference of $4.8776 \times 10^{-5}\%$ is seen as negligible.

4.2 Interpolation

IMEP was used to justify the effectiveness of interpolation as was done with filtering. The mean IMEP difference with an interpolation of 0.1 deg crank angle came out to be 1.77×10^{-4} , which represents a percentage difference of 0.0015.

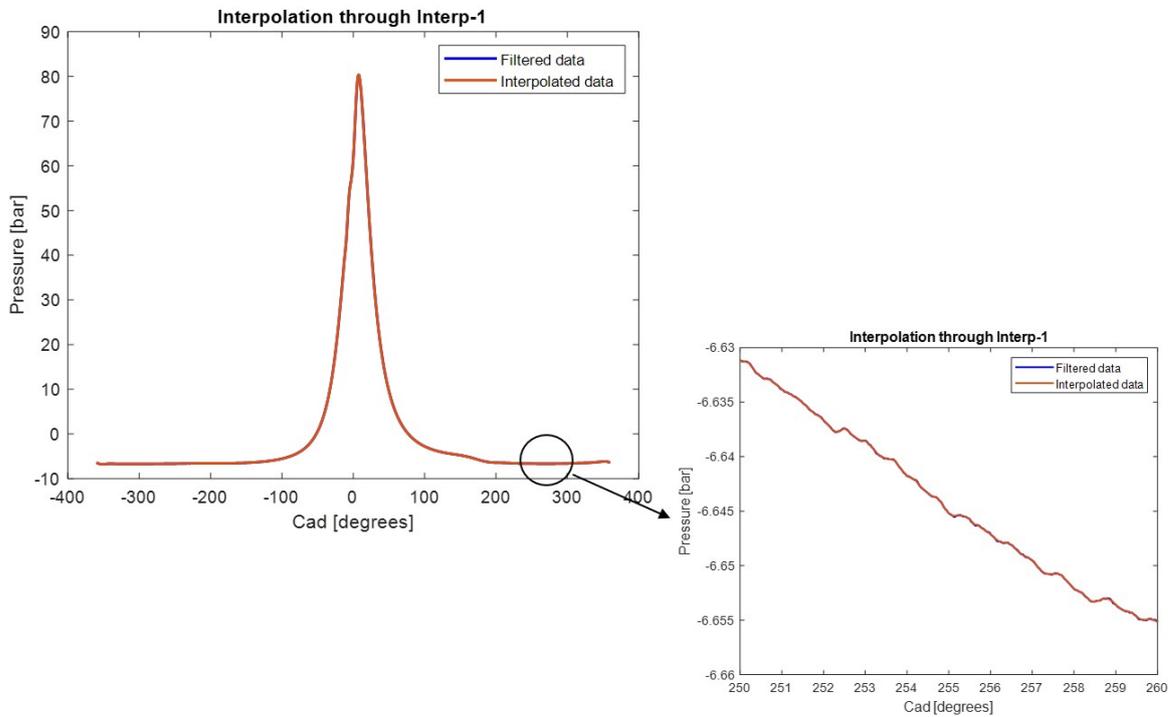


Figure 4.2: The visual effect of the interpolation process.

4.3 Zero level correction

The raw pressure obtained from Dewesoft was corrected for zero level by three different methods.

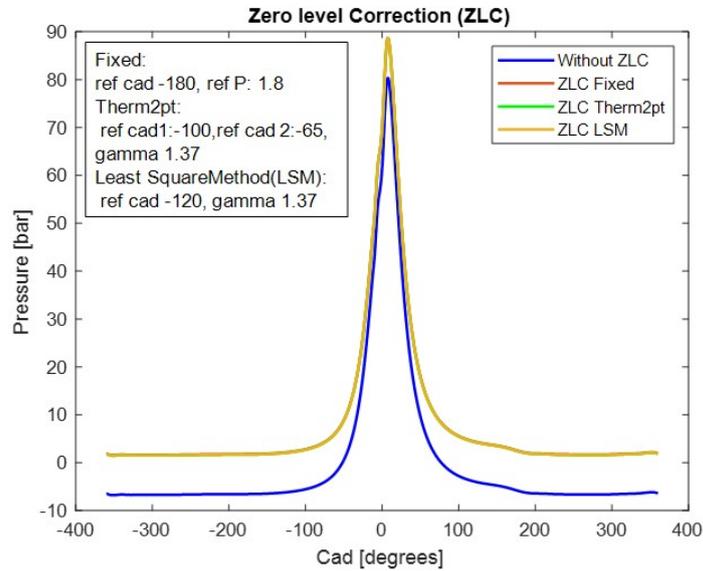


Figure 4.3: Comparison of different Zero level correction methods

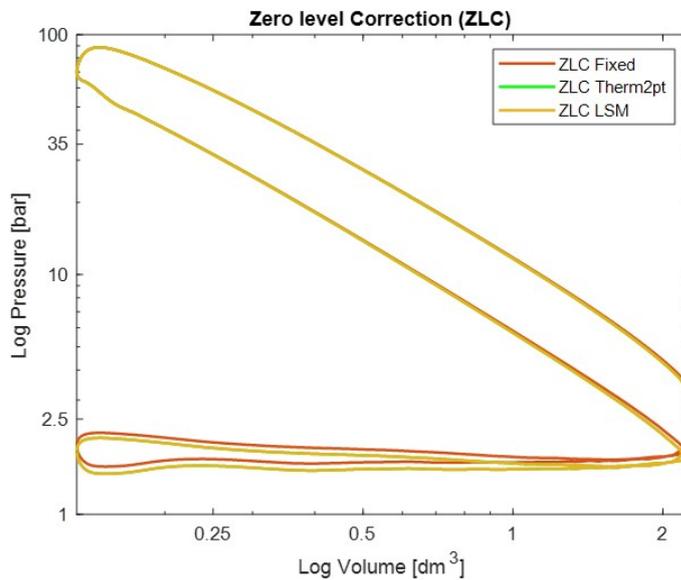


Figure 4.4: Comparison of different Zero level correction methods in log-log scale

The different zero level correction give different offsets. This is more evident in figure 4.4 which shows the logP-logV diagram for different methods. This can lead to differences in gamma, max pressure and combustion identification parameters like CA50.

The choice of best method to use depends on use case, availability and confidence of the available data. If accurate pressure values inside cylinder is known then fixed method should be used. Engines operating at low speeds without having a heavily tuned intake (low ram effect) can use measured value method. Pressure readings for this method is highly influenced by location of intake manifold pressure sensor. Thermodynamic 2pt and Least square methods can be used interchangeably and don't require additional data or sensor. LSM method provides more isolation towards sensor noise at a cost of high computational time.

4.4 Gamma

4.4.1 Logarithmic regression

The method of logarithmic regression is highly dependant on the input data and a general result is difficult to produce. Presented in Figure 4.5 is the resulting output from a data-set acquired from an SI-engine.

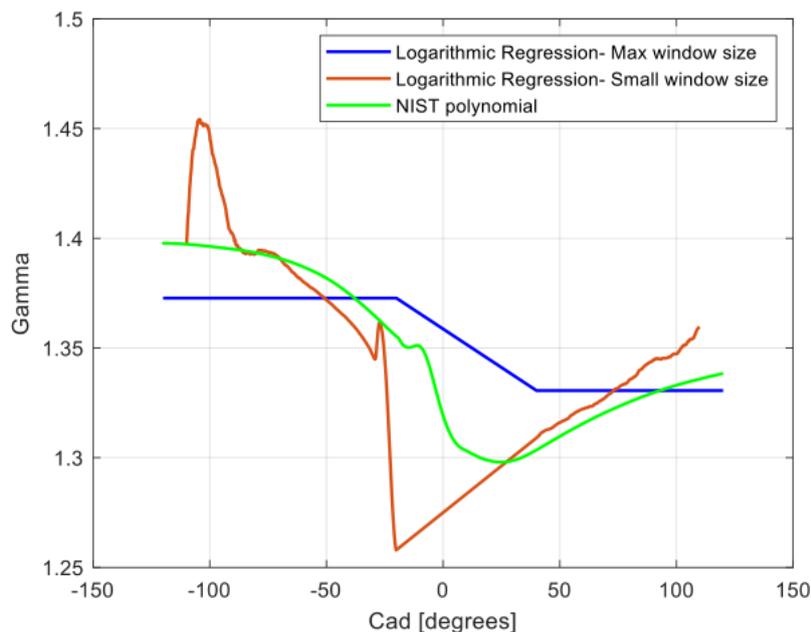


Figure 4.5: Gamma output from Logarithmic regression and NIST.

The method of logarithmic regression has its beginning in a universal thermodynamic truth. This should by logic give it the true value of gamma. The problem with this assumption is that the measurements taken aren't perfect. If the pressure trace collected is incorrect the gamma calculated will be incorrect. The advantage of the logarithmic regression method is its simplicity. As the pressure and volume of the cylinder should be known the gamma can be calculated with only these variables. There is therefore no need to introduce more assumptions, ex. compositions, and the model takes care of the change between SI and CI independently. The

weakness of the method is that it can and will give gamma values that are outside of the expected range if the input data is wrong. The gamma range should be between 1.25-1.4, but this method can give gamma values well outside of this range if any of the inputs are incorrect. Another disadvantage of the logarithmic regression method is the need of adiabatic process, this results in an inability to calculate the coefficient during the actual combustion.

During the project, it was found that using the logarithmic regression method when the pressure sensor and amplifier gain is set incorrectly will result in correct outputs as logarithmic regression seemingly corrects the error of incorrect gains. This has been attributed to that the calculated gamma is made from the same data it is later used to correct, which will correct any inconsistency in data acquisition.

4.4.2 NIST-Polynomial

The output from the NIST-Polynomial is compared to the output from the Logarithmic regression in Figure 4.5. The method of using the NIST-polynomial gives some apparent advantages, mainly that it is a tested method that gives a gamma output that is within the physical range of the coefficient. The second advantage is the academic agreement that the method is correct enough to use for further analysis. The third advantage of the method is that the coefficient can be approximated throughout the combustion which some other methods cannot do. The disadvantage of the method is the need for a proper mixture composition or at least an approximation of the composition which will introduce errors. The use of the NIST method on combustion will always be approximate as there are a large amount of burnt and unburned species reacting throughout the combustion. This was in the code approximated with a pre-ignition and a post-combustion mixture and the composition inbetween these two points approximated as a transfer between the two states with a MFB-curve. It can be seen in Figure 4.5 that the approximation of gamma by NIST-Polynomial gives a gamma that follows the expected curve and at all points fall within the expected range of the coefficient.

4.4.3 Ganippa proposal

The method of calculating gamma presented by G. Abbaszadehmosayebi and L. Ganippa [18] was pursued but could not be made to work. In Equation 2.47 the apparent heat release rate is calculated at CA50, what is not included in this equation is the index for gamma γ_{50} and instead γ is used. Due to the fact that gamma will vary throughout the compression and expansion this index is important. Later in Equation 2.52 the two gamma are assumed the same. This is not correct and the residual function will give an incorrect estimation.

4.5 IVC Effective

The IVC effective differed from the stated value by the manufacture in most of the cases. This was in the range of around 0-7 CAD and in all cases later closing than stated. For some engines tested no values for IVC was stated, and therefore no conclusion on how much it differed can be made. However, by looking at the log-log P-V diagrams it could clearly be seen that the compression started where the code set IVC effective. There was also one engine which the code had problem calculating the actual closing, and it was also hard to determine by looking at the graphs. This was due to fluctuations/ vibrations affecting the pressure. Whether that was due to valve closing or not is difficult to determine. Except from this single case the code seems to deliver accurate values.

The IVC effective will affect a lot of parameters and are in that sense important to get accurate, and to minimise false marginals going further in the code. The IVC effective is used for determining how much charge that is trapped in the engine, and how big the “explosion” will be. But as seen in the sensitivity analysis it does not affect the overall RoHR that much, compared to other factors.

4.6 Compression ratio

The compression ratio differed from the stated from the manufacture by around 0.01-0.1 depending on the tested engines. This seems reasonable and can as said depend on tolerances in the manufacturing process and porosity, as well as valve closing offset. This gives a small different in the overall RoHR, but are far from the most affecting factor, as seen in the sensitivity analysis.

4.7 Heat transfer models

Comparison graphs for heat transfer coefficient and heat transfer for both Woschni and Hohenberg is presented below in Figure 4.6 and 4.7 respectively. From the figures it can be seen that the maximum HTC (heat transfer coefficient) observed for Woschni is around 1250 W/m²-K and for Hohenberg it is approximately 2400 W/m²-K. There is a significant difference between the models for heat transfer curves as well. The maximum heat transfer observed for Woschni is 4 J/deg, whereas the maximum heat transfer observed for Hohenberg is 8 J/deg. The difference in these values is due to the difference in the characteristic velocity and the temperature exponent. The Hohenberg model has the highest heat transfer coefficient compared to the Woschni because of the change in the temperature exponent. Hohenberg assumes constant characteristic velocity across all crank angles, whereas the Woschni equation calculates a different characteristic velocity across all crank angles. It can be seen from the Figure 4.7 that, as the piston is in downward motion, and during the intake process the heat flux is minimum. The heat flux increases when the piston is compressing the gas mixture.

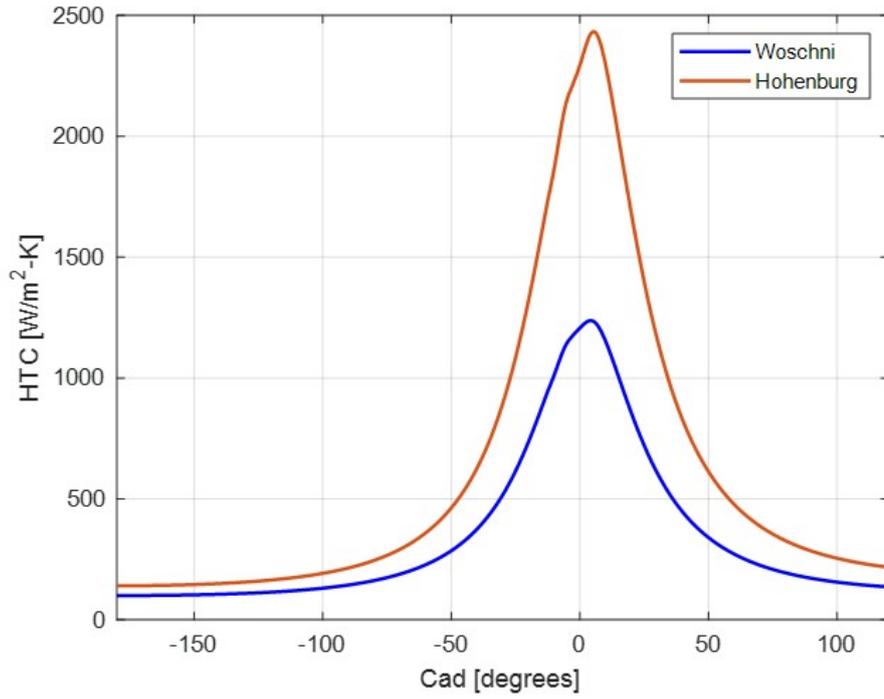


Figure 4.6: Comparison of Heat transfer co-efficient for different heat transfer models

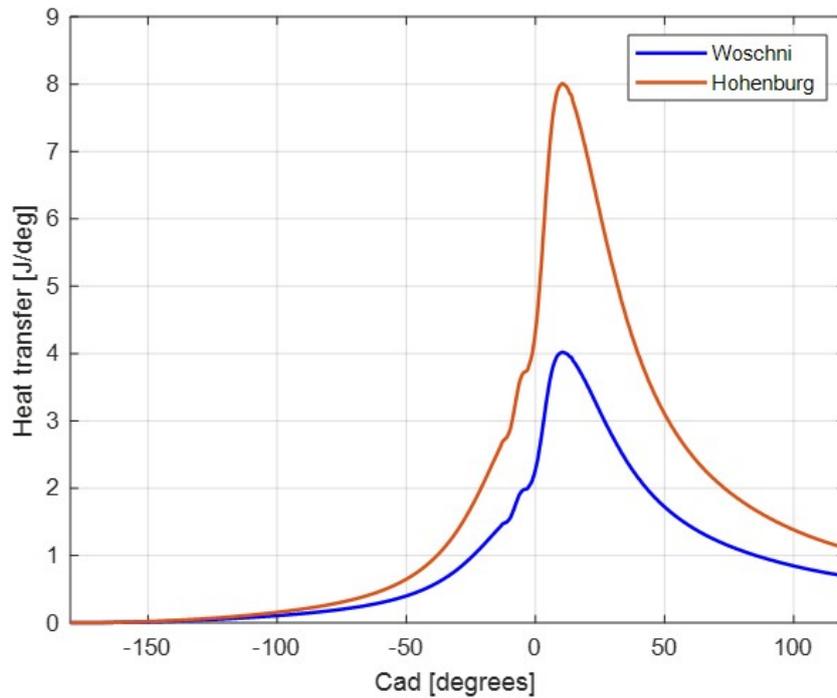


Figure 4.7: Comparison of Heat transfer for different heat transfer models

4.8 Rate of heat release

The major project delivery, the modular rate of heat release analysis software is completed. This software has model based sub functions that can be used with different options for different RoHR analysis. The MATLAB based software is capable of performing both apparent and real rate of heat release calculations. Below are the results of apparent and real rate of heat release of the six cylinder CI engine.

Table 4.1: Engine, operating conditions, and RoHR methods

Engine type	CI Inline Six Turbocharged
Operating condition	RPM: 1200 Torque: 600Nm
ZLC method	Thermodynamic 2pt -100 -65 1.37
Gamma method	Logarithmic Regression 2,3
IVC method	IVC effective
Rohr type	Apparent/Real
Wall temp method	User input
Heat Transfer method	Hohenburg
dP/dCad method	Savitsky golay

In Table 4.1 the details of the engine and the details of different options used to perform the rate of heat release analysis are presented.

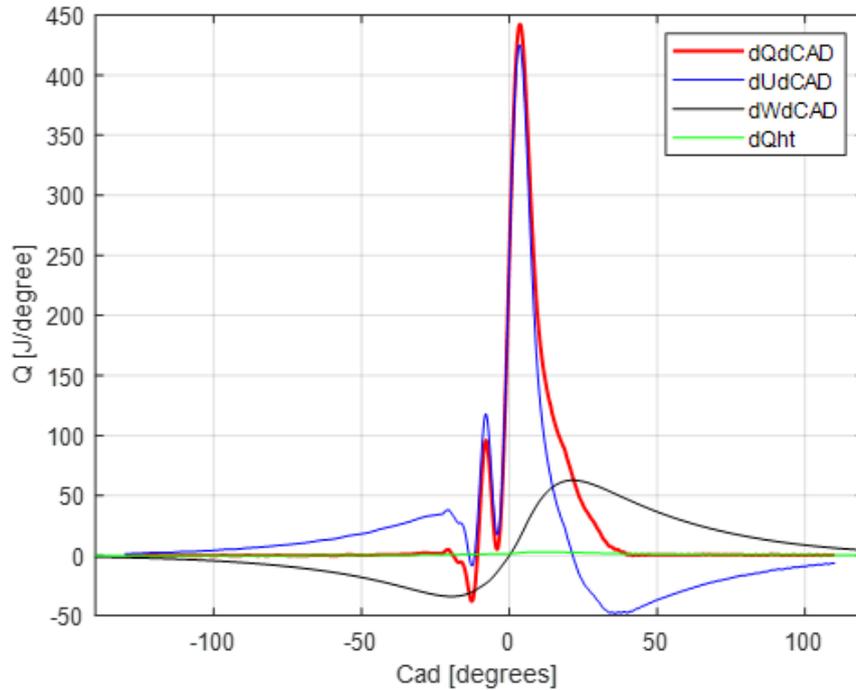


Figure 4.8: Rate of heat release curves of: chemical energy, internal energy, mechanical work and heat transfer rate

Based on the behavior of rate of heat release curves, as seen in figure 4.8 and figure 4.11, several conclusions can be drawn. The internal energy is a function of temperature, mass and calorific value of the fuel. Thus at the end of compression where temperature and pressure are at their maximum, also internal energy will reach its maximum value close to TDC. As the expansion progresses the internal energy will reduce as pressure and temperature decreases. The mechanical work curve will initially be negative as work is required to compress the air-fuel mixture. After TDC and during the expansion phase the work will be positive as pressure from the combustion will power the piston downwards. The heat transfer can be seen to have a small effect on the heat release but can be seen during the compression and expansion strokes as heat transfer will take place to piston and cylinder walls during these phases when the temperature is high [8].

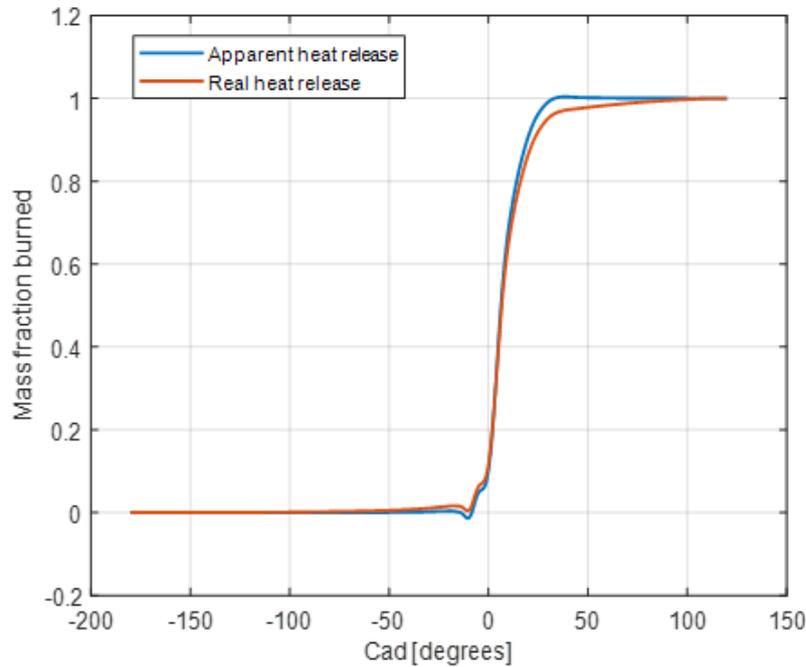


Figure 4.9: Cumulative mass fraction burned curve

The mass fraction burned curves of apparent and real heat release of both CI and SI engine is as shown in the figures 4.9 and 4.12. From the mass fraction burn curve the difference can be seen, especially at the end of the combustion, where the left over mass burns cannot be easily quantified [8]. When considering real rate of heat release, blowby and heat transfer are also added to pressure data obtained from in-cylinder measurement. This increases the cumulative energy released as compared to apparent case. During normalisation, both the curves are normalized by their respective cumulative sum, and this causes the real mass fraction burned curve to appear lower than apparent curve. If instead of the respective cumulative sum, the fuel energy content i.e. $m_f * Q_{LHV}$ (mass of fuel multiplied by its lower heating value) is used as the normalising factor then the results would be different and apparent will show incomplete combustion and unexplained loss of energy. This means the real heat release analysis reflects the actual combustion inside the engine cylinder and there is room for improving the efficiency.

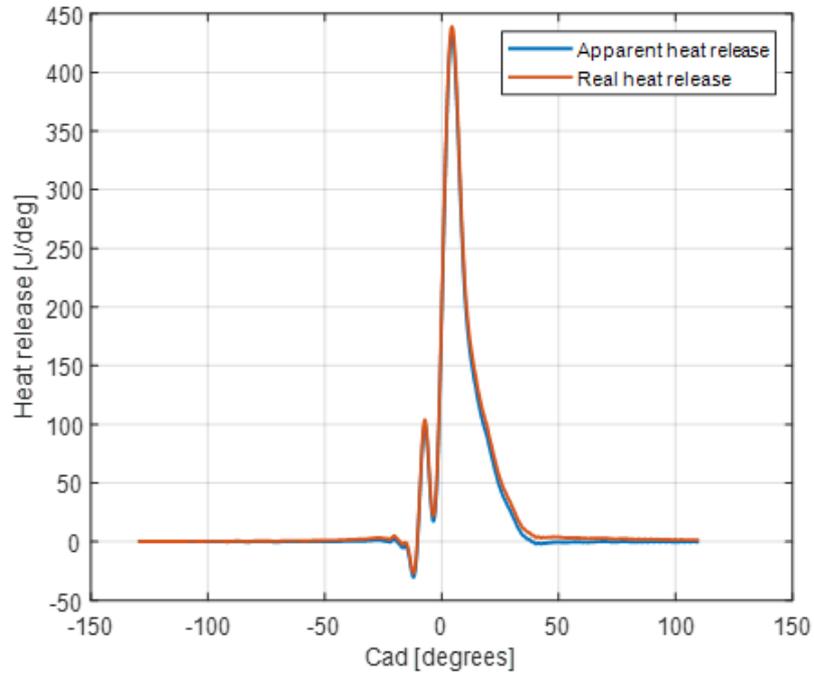


Figure 4.10: Rate of Heat Release

Table 4.2: Results of CI engine at 1200 RPM and 600 Nm heat release analysis

RoHR method	Apparent	Real
$IMEP_{net}$	8.39	8.39
$IMEP_{gross}$	8.12	8.12
PMEP	-0.27	-0.27
P_{max}	88.36	88.36
Gamma	1.37	1.37
hc_{max}	0.00	2384.58
$dQ_{ht_{max}}$	0.00	8.46
T_{max}	1433.02	1433.02
SOC	-4.70	-6.20
CA50	6.30	6.70
CA90	19.50	22.80

The result of CI engine at 1200 RPM and 600 Nm after rate of heat release analysis is presented in table 4.2.

Similarly Apparent and real rate of heat release analysis done for SI engine and following are the results shown below: SI engine at 1500 RPM and BMEP 9 Bar.

Table 4.3: Engine operating conditions, and RoHR methods

Engine type	SI Inline Four Turbocharged
Operating condition	RPM: 1500 BMEP: 9bar
ZLC method	Thermodynamic 2pt -100 -65 1.37
Gamma method	Logarithmic Regression 2,3
IVC method	IVC effective
RoHR type	Apparent/Real
Wall temp method	User input
Heat Transfer method	Woschni
dP/dCad method	Savitsky golay

The above table 4.3 gives the details about the engine and the details of different options used to perform the rate of heat release analysis.

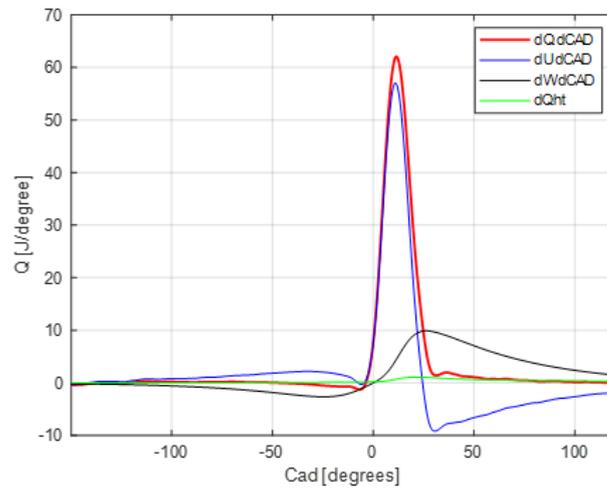


Figure 4.11: Rate of heat release curves of: chemical energy, internal energy, mechanical work and heat transfer

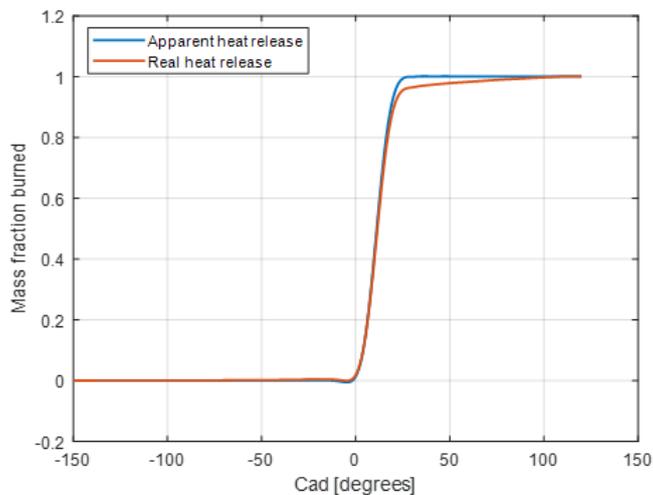


Figure 4.12: Cumulative mass fraction burned

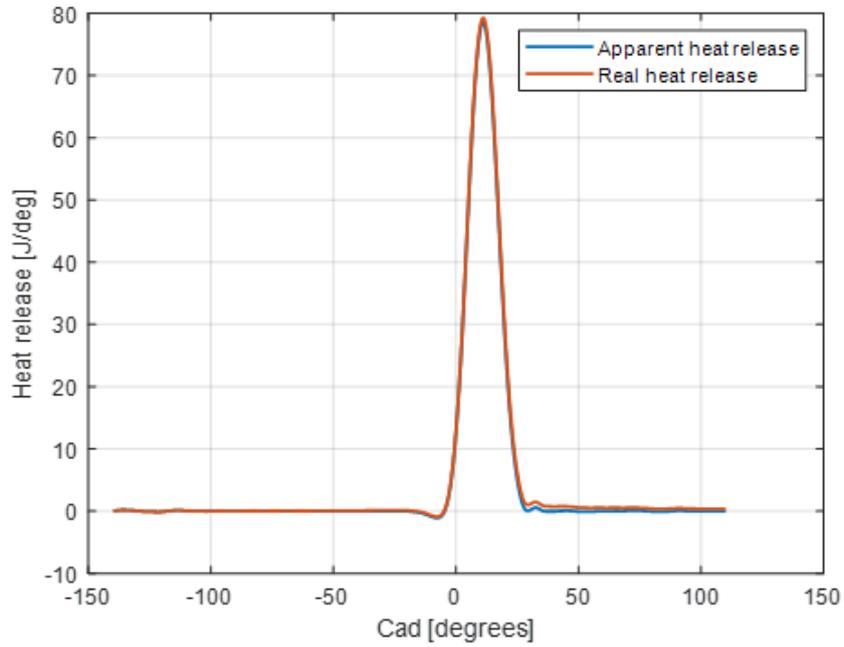


Figure 4.13: RoHR

Table 4.4: Results of CI engine at 1200 RPM and 600 Nm heat release analysis

RoHR method	Apparent	Real
$IMEP_{net}$	9.32	9.32
$IMEP_{gross}$	8.82	8.82
PMEP	-0.50	-0.50
P_{max}	48.53	48.53
Gamma	1.37	1.37
hc_{max}	0.00	471.36
$dQ_{ht_{max}}$	0.00	1.12
T_{max}	2406.77	2406.77
SOC	2.20	2.10
CA50	11.20	11.50
CA90	18.80	20.30

Hence the results of heat release analysis of SI engine at 1500 RPM engine (Speed) and 9Bar (BMEP) are presented in table 4.4.

4.8.1 Validation of RoHR analysis

In motoring case there is no chemical energy involved from fuel and thus its rate of heat release curve should be a flat line. This fact can be used to validate the different RoHR methods and can be used to justify the selection of various parameter sub-models. From the figures 4.14 and 4.15, the chemical heat release curve is approximating a flat behaviour. The values are not exactly zero, but are in range of ± 2 J/deg. This can be considered negligible when it is compared against the values obtained during combustion. This curve can be further tuned by performing data quality checks and ensuring that incoming data is free from small defects like charge leakage etc. The presence of a relatively flat profile can be seen as a validation for the code.

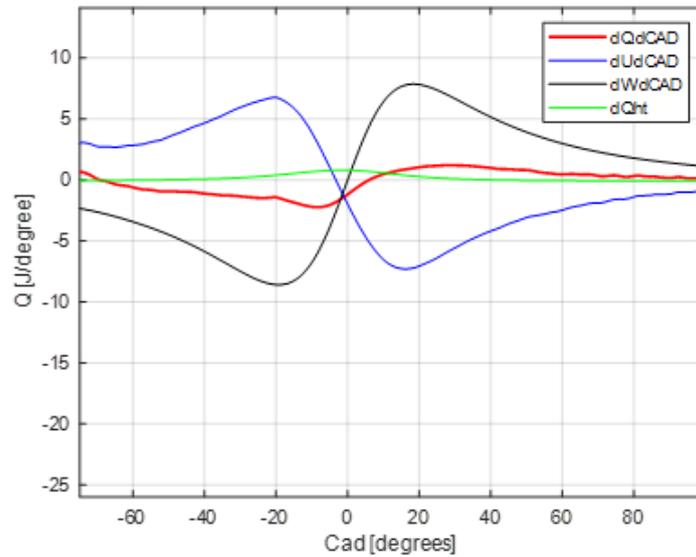


Figure 4.14: Motoring RoHR curves of CI engine

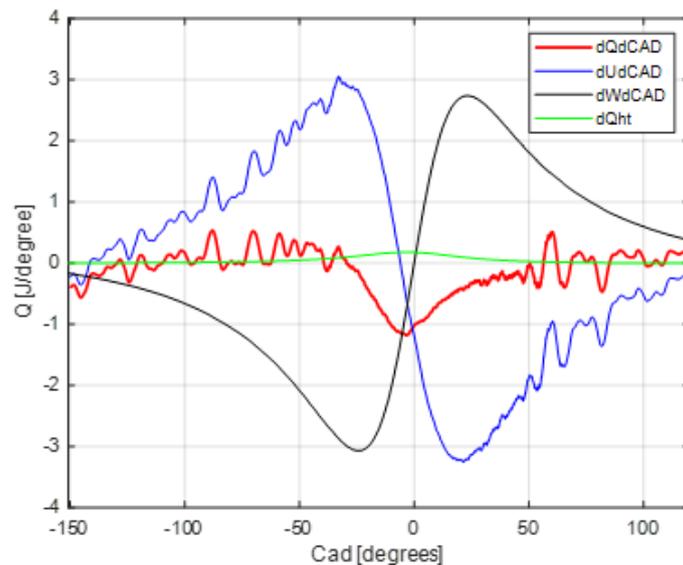


Figure 4.15: Motoring RoHR curves of SI engine

4.8.2 Wall Temperature

The wall temperature gives values of around 80°-95° Celsius. The problem is to validate those numbers, since no actual measurements have been made. However, earlier assumptions on the same engine tested have been 80°C, compared to this it seems to be in the right region. As seen in the sensitivity analysis it does not affect the overall RoHR in that great manner compared to the other factors. This is also a hard parameter to determine since different engine blocks will have different geometries and material properties. The physics should at least give a good assumption/starting point, but to get it more accurate than this actual measurements needs to be made.

4.9 Input data validation

The analysis of in cylinder pressure is as shown sensitive to pressure irregularities and errors. Therefore, it is valuable, to early in the process, be able to identify these problems and thus not rely on incorrect data acquisitions. Inconsistency in the input data can be seen in two ways. Both are based on the logarithmic regression model for calculating gamma, and compared to the known bound NIST polynomial. Therefore if the calculated data is inconsequential in several regards compared to the NIST polynomial the data might be incorrect. First, it can be identified by comparing the computed gamma to the known physical range of gamma. This is shown in Figure 4.16. The left image with "incorrect" input data will have values well above and below the expected margin of gamma, while the right data is more of what should be expected from correctly acquired data.

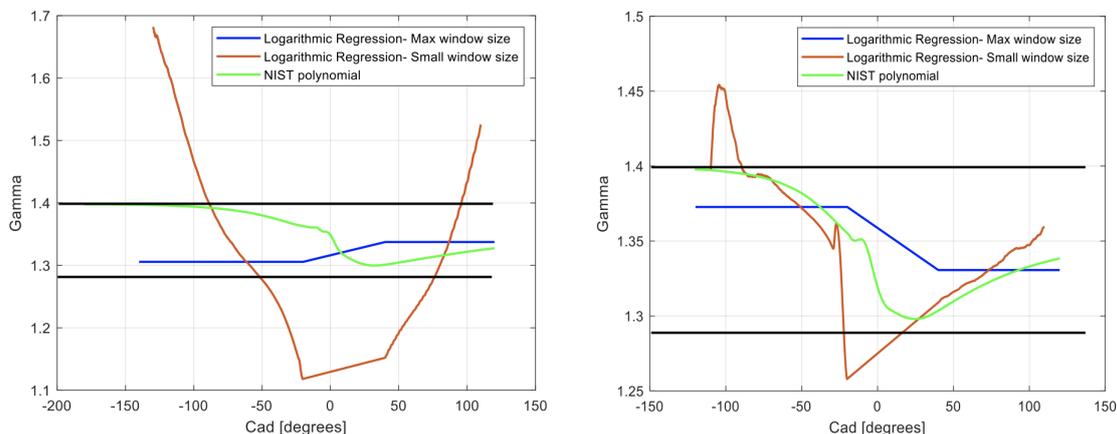


Figure 4.16: Different datasets, out-of-bound and inside-bound

Second it can be checked by the mean gamma slope within the compression and expansion stroke, this can be seen in Figure 4.17. For the left figure it is easy to see the difference in the slope between the NIST polynomial and the logarithmic regression, while in the right figure the mean slope from both -110°- -75° and -75°- -25° follows the mean slope of the NIST polynomial quite well.

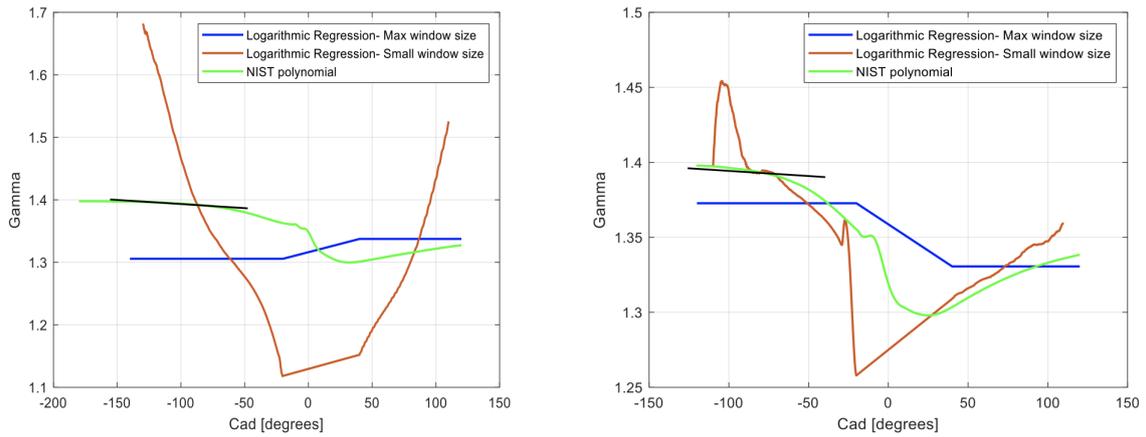


Figure 4.17: Different datasets, not following, and following the NIST-slope

This validation can be done to indicate that there are obvious errors in the data acquired and some further action in cleaning or collecting must be made. The calculated gamma curve and the NIST polynomial results can not be expected to completely match as this would make one of the two models superfluous but the general trend of the two should be close.

4.10 Current to Voltage converter

The current to voltage converter that was built behaved non-linearly during testing and calibration. This uncertainty might be attributed to the soldering as the op-amp is sensitive to temperature, the non-linearity can also be due to current saturation of the op-amp.

5

Discussion

5.1 Sensitivity Analysis

Rate of heat release curves are affected by various parameters such as trigger offset, zero level correction, gamma, heat transfer, compression ratio, IVC, temperature at IVC, wall temperature and RoHR method namely apparent or real. Inclusion of sub models like blowby can also affect the results. A sensitivity analysis was performed to quantify the effect of all these parameters and to establish the greatest contributors. Figure 5.1 represents the inter dependencies of various parameters.

Table 5.1: Sensitivity Analysis dependencies

	IMEPnet	IMEPgross	PMEP	Max P	Gamma*	Max hc	Max dQht	Max T	SOC(CA05)	CA50	CA90
Trigger Offset**	Affected	Affected	Affected								
Pressure Offset**	Not Affected	Affected	Affected	Affected	Affected	Affected	Affected	Affected	Affected	Affected	Affected
Gamma Method	Not Affected	Not Affected	Not Affected	Not Affected	Affected	Affected	Affected	Affected	Affected	Affected	Affected
Heat transfer method	Not Affected	Affected	Affected	Affected	Affected	Affected	Affected				
Compression Ratio	Not Affected	Not Affected	Not Affected	Not Affected	Affected	Affected	Affected	Affected	Affected	Affected	Affected
IVC effective	Not Affected	Affected	Affected	Affected	Affected	Affected	Affected	Affected	Affected	Affected	Affected
Temp@IVC	Not Affected	Affected	Affected								
Twall	Not Affected	Affected	Affected	Affected	Affected	Affected	Affected				
RoHR method	Not Affected	Affected	Affected	Affected	Affected						

Affected
Not Affected
Not Applicable

* Compression Gamma calculated between -100 and -65
 ** For seeing individual effect gamma kept constant

Trigger offset

Trigger offset is used to synchronise the crank angle encoder data with the pressure data. Change in trigger offset changes IMEP, gamma, heat transfer, temperature and rate of heat release.

Pressure offset

Pressure offset or zero level correction affects IMEP gross and PMEP calculation but IMEP net remains constant. Maximum pressure, gamma, heat transfer, temperature and rate of heat release parameters are changed.

Gamma

Gamma affects rate of heat release as shown in equation 2.26. It does not affect IMEP, pressure, temperature and heat transfer properties.

Compression Ratio

Compression ratio changes volume which in turn changes gamma, temperature, heat transfer and rate of heat release.

IVC effective

IVC effective is the reference point chosen to calculate in cylinder temperature. Temperature and heat transfer are affected which leads to a change in rate of heat release parameters.

Temperature at IVC

Temperature at IVC affects temperature calculation which in turn affects heat transfer and rate of heat release calculations.

Wall temperature

Wall temperature affects heat transfer which causes slight change in rate of heat release parameters.

Sensitivity comparison

Figures 5.1 and 5.2 show the change in CA90 in crank angle degrees with a delta change in different parameters. Since parameters have different units and comparison with unity change is not possible (for example-unity change of intake valve temperature is a small quantity, but that of gamma is a very huge quantity) effort has been made to keep the delta change reasonable for each parameter. Comparing the parameters at this scale shows that rate of heat release (as evident by CA90) is most affected by pressure offset and gamma, followed by trigger offset and compression ratio. IVC does not have a significant change in RoHR analysis.

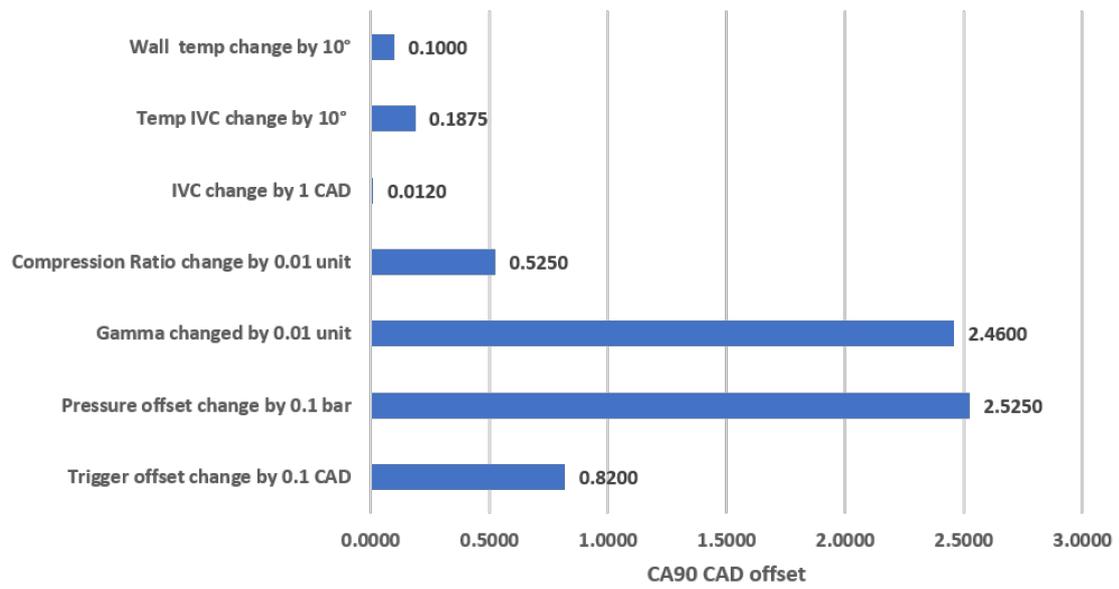


Figure 5.1: Parameter sensitiveness towards RoHR, CI engine

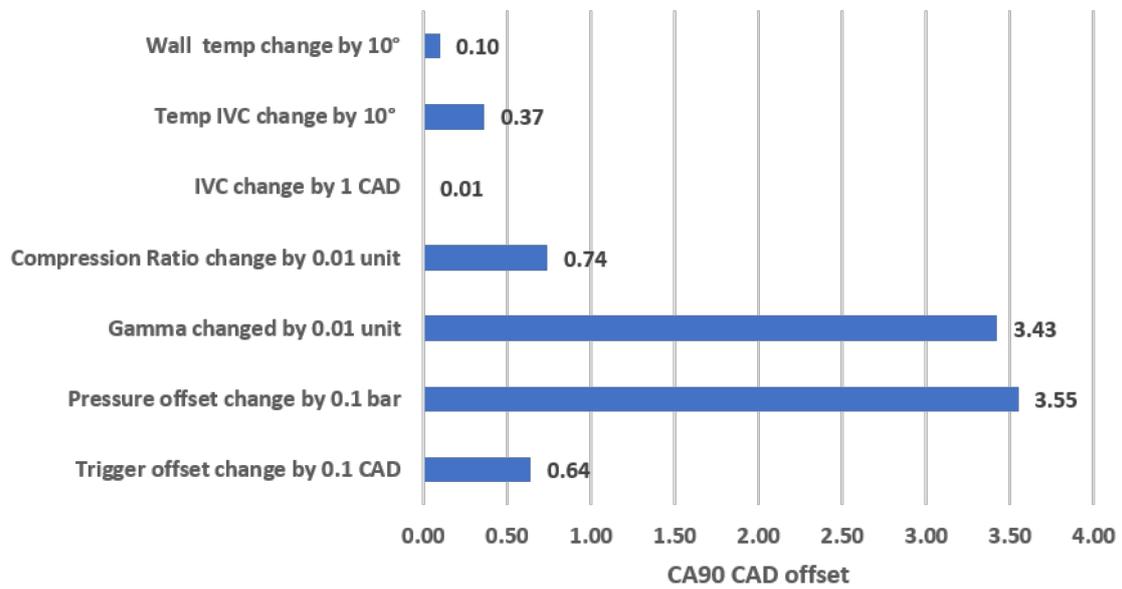


Figure 5.2: Parameter sensitiveness towards RoHR, SI engine

5.2 Concluding remarks

The code developed for calculating combustion analysis parameters investigates different parameters which can affect the heat release calculations. It provides a modular platform to calculate RoHR with different levels of complexity incorporating multiple models to be used according to availability of data and engine type. No model can be completely perfect, but the code tries to give the theoretical best approximation for the combustion parameters. This is important as it will help analyse the split of loss from the engine indicating the efficiency of the engine and major loss areas. Some of the parameters gives a small change in the overall RoHR but all together might add up to a noticeable difference. Also, correct estimation of parameters like combustion duration is important as it is a validation point used for finalising cylinder head design and configurations aimed at providing faster combustion. True estimates will allow better correlations with simulation results and faster and efficient development.

The created code gives the user freedom to choose what models to include and which to omit. It is based on modularity and allows the user to create their own models to use alongside this code. The code is also open for the user to read and there are no embellishments of the results and the user is only given the truth.

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A

Suggested parameters

Table A.1: Recommended options for getting best RoHR results for CI and SI engine

ROHR MATLAB CODE		CI	SI
RoHR method	Apparent	Real	Real
Zero level correction	Fixed/Measured/Thermodynamic 2 point/ Least square method	Thermodynamic 2 point	Thermodynamic 2 point
dPdCAD	Finite difference/ Savitzky Golay	Savitzky Golay	Savitzky Golay
Gamma	NIST/Logarithmic regression/User Input	NIST	NIST
IVC effective	IVC_eff code/User Input	IVC_eff code	IVC_eff code
Heat Transfer	Not applicable	Hohenburg	Woschni
Blow-by	Not applicable	Blow-by_code	Not applicable
Wall Temperature	Wall_Temp Code/User Input	Wall_Temp Code	Wall_Temp Code
Compression ratio	CR_code/User Input	CR_code	CR_code

B

User interface

To make the software usable by the average user a user interface was created to give potential users a simple input to the otherwise complex code and code structure. This UI was coded and designed using the built-in app creator in MATLAB. The UI is capable of loading data-sets and setting engine parameters and then allows the user to select what models that are to be used. The code can save previous setups if the same data or data with the same variable names is to be used. There is a hope to eventually complete a post analysis UI to visualize results in an understandable way but is of yet not completed.

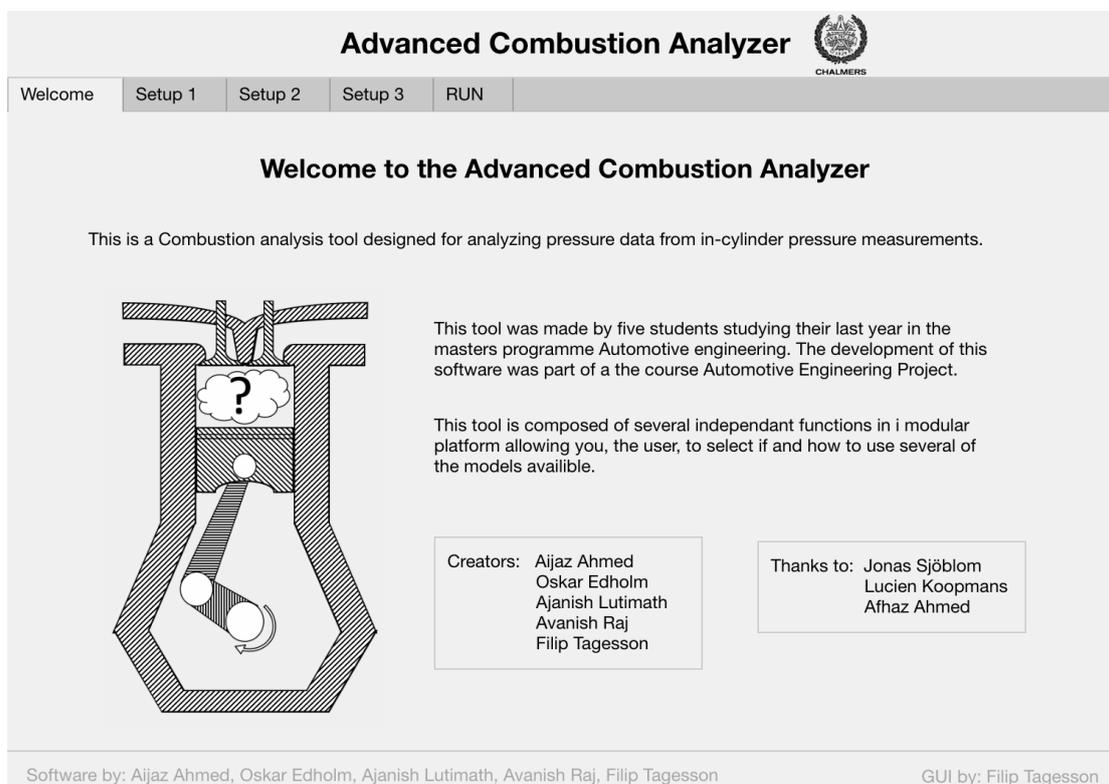


Figure B.1: UI, Mainpage

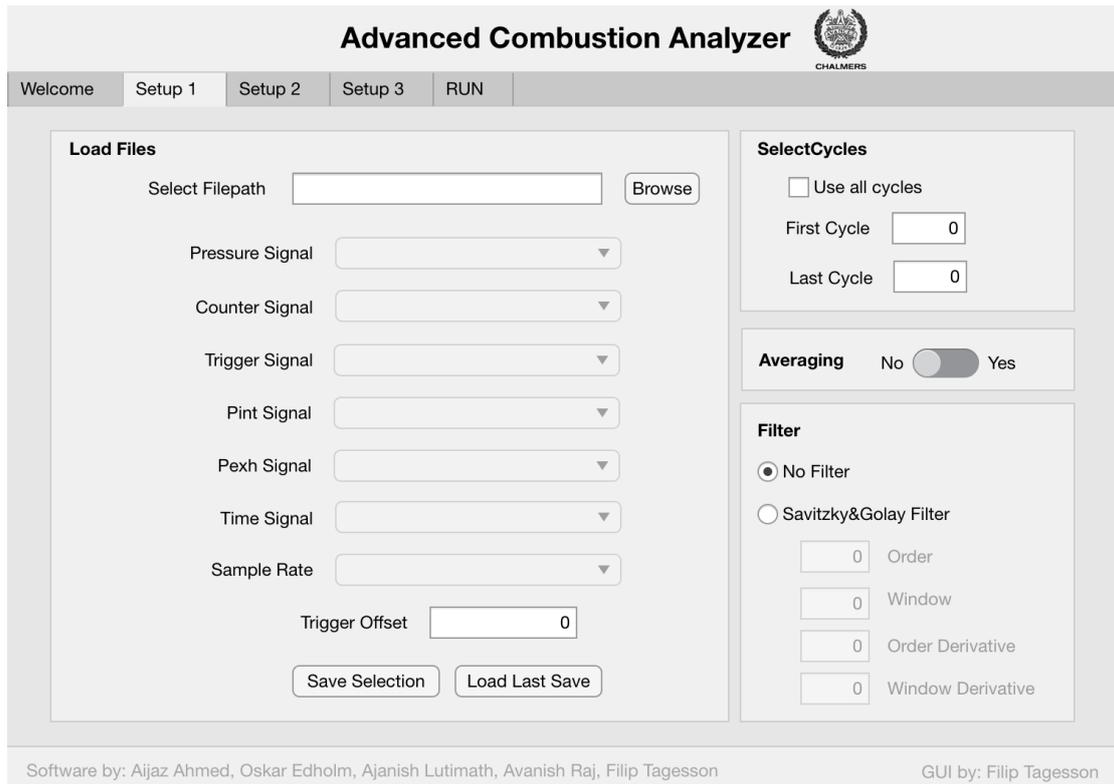


Figure B.2: UI, Setup page 1

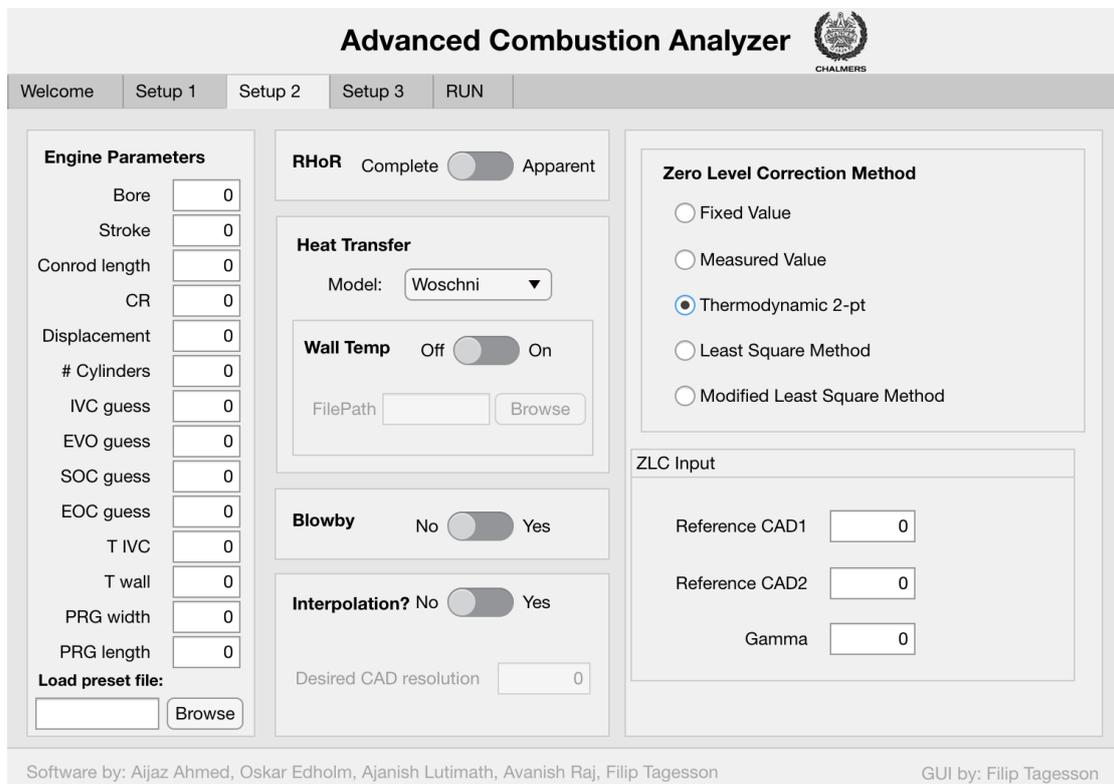


Figure B.3: UI, Setup page 2

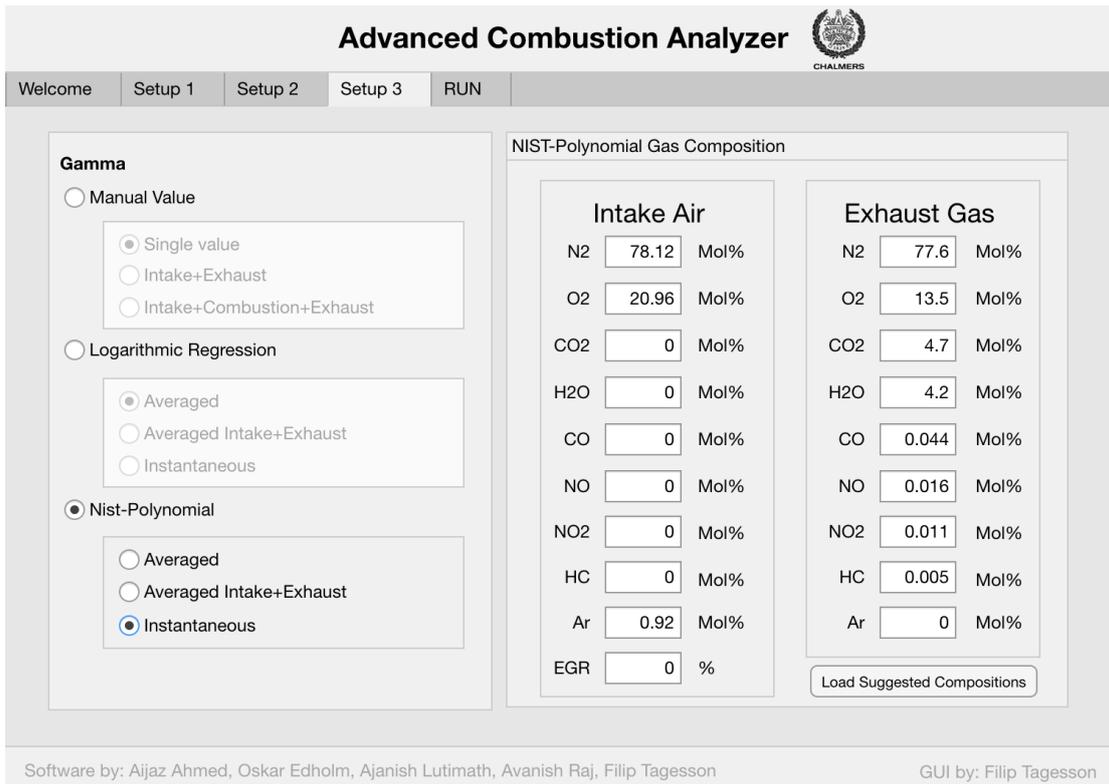


Figure B.4: UI, Setup page 3

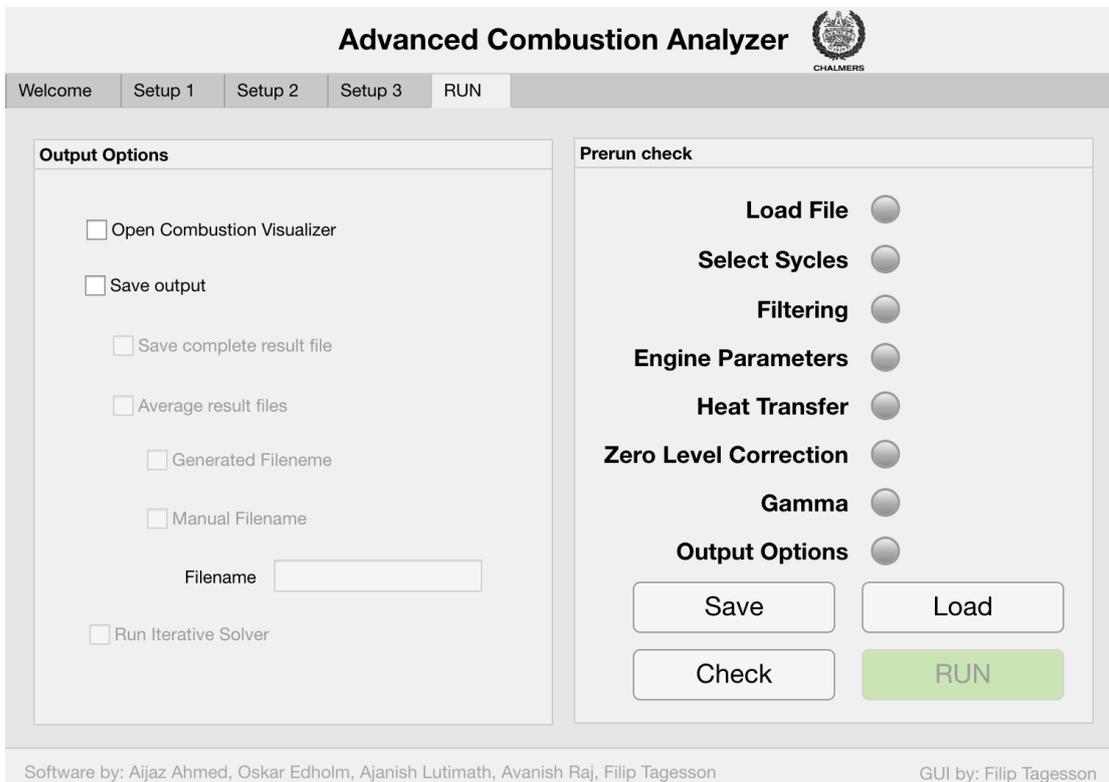


Figure B.5: UI, Run page

