Estimation of the Residual Gas Fraction in an HCCI-engine using Cylinder Pressure

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Sammanfattning
Abstract

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The goal in this thesis is to formulate an algorithm for estimation of the residual gas fraction based on the cylinder pressure. The estimation is improved if also the exhaust gas temperature is used together with the cylinder pressure.

The formulated algorithm has then been tested on data from a single cylinder engine running in HCCI-mode during steady state conditions. An error of ±4% was noted compared with the residual gas fraction obtained from simulations.

The thesis also investigates the effects of some possible error sources.

Nyckelord
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Habo, May 2003

Niklas Ivansson

Notation

Symbols

θ	Crank angle
λ	Normalized air/fuel ratio
p	Pressure
T	Temperature
V	Volume
A	Area
m	Mass
B	Cylinder bore
l	Connecting rod length
a	Crank radius
c_v	Specific heat at constant volume
c_p	Specific heat at constant pressure
γ	Ratio of specific heats
Q_{LHV}	Lower heating value
η_c	Charge efficiency
CoC	Completeness of Combustion
Q_{wall}	Heat losses to the wall
Q_{fuel}	Energy released by the fuel
W	Work
n	Engine speed

Abbreviations

EGR	Exhaust Gas Recirculation
TDC	Top Dead Center
BDC	Bottom Dead Center
HCCI	Homogeneous Charge Compression Ignition
SI	Spark Ignition
CI	Compression Ignition
EVC	Exhaust Valve Close

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1 Introduction

The main goal in this thesis is to find an algorithm that uses the cylinder pressure as input to estimate the residual gas fraction in an HCCI-engine. It should be possible to use the estimation algorithm in a real time control system and it should therefore be fast and have good accuracy. The method used for the estimation is an extension of the method developed by Michael Mladek, originally used for air mass estimation [1]. HCCI stands for Homogeneous Charge Compression Ignition and is a new type of engine which has shown good efficiency in combination with low emissions.

1.1 Background

Measuring the cylinder pressure is one way to get important information about the combustion from the inside of the cylinder.

Previously it has only been used for research and development of new engines. But when the demands for more efficient and cleaner engines increase it will be interesting to use this information to control the different engine parameters to get optimal combustion. Some articles have described how to use the cylinder pressure for example air/fuel-estimation and to get optimal spark-timing [2]-[3].

Two important parameters in the HCCI-engine is the position of autoignition and the combustion rate. Both of these parameters are affected by the amount of residual gas. Higher amounts of residual gas will also give lower emissions of Nitrogen Oxides, NO_x.

There is no sensor available to measure the amount of residual gas or residual gas fraction. The only methods present today to get the residual gas fraction is through simulations or an estimation.

This report is about estimation of residual gas fraction and it can be interesting to mention something about the difference between simulation and estimation. The goal of the simulation is to produce as good result as possible. To get a more accurate result more relations and equations are added, which often, but not necessary, results in long simulation time. The estimation on the other hand needs just to produce reasonable result but has to be fast so it is possible to use it in real-time applications to control the engine.

The estimation model should consist of physical relations and equation rather than a black-box model.

1.2 Methods

Data from previous measurements have been used in this work. The measurements were done on a single cylinder engine running in HCCI-mode at different speeds and loads in the interesting operation area. The residual gas fraction comes from a simulation program called "GT Power" and is here assumed to be correct.

The measured and simulated data is then used to investigate the possibility to use the cylinder pressure for residual gas estimation.

1.3 Thesis outline

The work done during this thesis and the concepts contained in the thesis are described in the following chapters.

Chapter 2, The engine Some necessary background information about the engine is explained to make it easier to understand the following chapters.

Chapter 3, HCCI-engine The differences between an ordinary and an HCCI-engine and why the HCCI-Engine is of interest.

Chapter 4, Cylinder pressure This chapter gives some knowledge about cylinder pressure and how to use it to calculate the mass fraction burned.

Chapter 5, The Algorithm This chapter explain how it is possible to use the cylinder pressure for residual gas fraction estimation.

Chapter 6, Sensitivity Analysis Some possible sources of errors and how they affect the result is described in this chapter.

Chapter 7, Summary The last chapter summarize the work and give some ideas for further investigations.

2 Engine

2.1 The Four Stroke Engine Cycle

The normal gasoline engine used in modern cars is the four stroke Otto-engine. During one cycle the crank-shaft has two revolutions (720°). The different strokes is shortly described below. The numbers 1-4 refers to the numbers in the pressure-volume diagram, fig. 2.1.

Intake, TDC-BDC, 0-180°, 1-2 The intake valve is open and the cylinder is filled with new charge of fuel and air.

Compression, BDC-TDC, 180-360°, 2-3 During the compression stroke the piston moves up and compress the charge. Shortly before the piston reach the TDC (Top Dead Center) the spark plug ignites the charge and the combustion starts.

Expansion,TDC-BDC,360-540°, 3-4 During the combustion the gas expands and pushes the piston down to BDC (Bottom Dead Center).

Exhaust, BDC-TDC, 540-720°, 4-1 During the last stroke the exhaust valve opens and the piston pushes the burned gases out of the cylinder.

The angles given here are valid for the ideal Otto cycle. The valves open at slightly different angles in a real engine. (See the section about valve timing 3.1.)

2.2 The Diesel Engine

The Diesel engine or compression ignited (CI) engine has the same strokes as described above but there is some differences. During the intake stroke only air is inducted to the cylinder. The air is then compressed and instead of ignition by a spark, fuel is injected. The temperature of the compressed air is so high that the fuel ignites starts the combustion.

2.3 Air to fuel ratio, Lambda

The ratio between mass of air and mass of fuel which gives complete combustion without excess air is called the stoichiometry ratio, L_{st}. This ratio is about 14.7 for gasoline. The air/fuel ratio is often expressed using a relative measure called lambda. Lambda is the ratio between the actual air/fuel ratio divided with the stoichiometric ratio.

$$\lambda = \frac{m_{air}/m_{fuel}}{L_{st}} \tag{2.1}$$

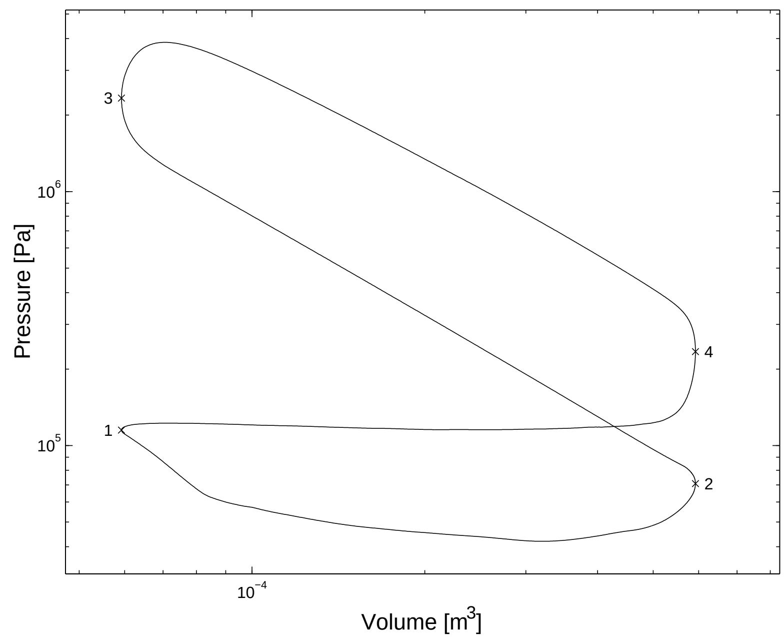

Figure 2.1: Volume vs Pressure in a conventional spark ignited engine in a log-log diagram

2.4 First law of thermodynamics

The engine produce work by transforming the chemical energy of the fuel into mechanical energy. One equation that describes this is the first law of thermodynamics that is based on the conversation of energy.

$$dU = \delta Q_{fuel} - \delta Q_{wall} - \delta W - \sum_i dm_i h_i \tag{2.2}$$

δQ_{fuel} is the chemical energy released from the fuel. dU is the change in internal energy of the mass in the cylinder and δQ_{wall} is the heat loss to the cylinder wall. δW is the work from the fluid to the piston. The last term is the changes in the enthalpy due to the flows of mass into and out of the cylinder. If we consider just when the valves are closed and assumes that there are no crevice effects the last term can be skipped. Crevice effects are when mass in the cylinder blows by the piston and cylinder wall. The crevice effects are small compered to the other terms.

2.5 Residual Gas

Residual gas is already burned gas from previous engine cycles that is left in the cylinder. The main reason to use residual gas is to lower the amount of NO_x that is formed. The temperature in the cylinder is so high that the nitrogen and oxygen in the air react and form NO_x. By using residual gas the total mass in the cylinder is increased which means that also the heat capacity of the charge is increased. This results in lower maximum temperature and in the end that less NO_x is formed. There are two ways to increase the amount of residual gas in the cylinder. Either through Exhaust Gas Recycling, EGR, which is to lead back the exhaust gases to the cylinder, or to close the exhaust valve before all exhaust gas has left the cylinder. The second alternative, when residual the residual gas stays in the cylinder, is sometimes called internal EGR.

The amount of residual gas is often measured as a fraction of the total mass and the definition of residual gas fraction is the ratio between the mass of residual gas and the total mass:

$$x_{rg} = \frac{m_{rg}}{m_{tot}} \tag{2.3}$$

3 HCCI-Engine

Today the commonly used engines in modern cars today are either Spark-Ignited (SI) Otto-engines or Compression-Ignited (CI) Diesel-engines. In the spark ignited engine the fuel and air mixture is compressed and the combustion starts when the spark-plug ignites the mixture. The flame then propagate through the cylinder to the cylinder walls. In the CI-engine only air is in the cylinder during the compression. The combustion starts when the fuel is injected to the cylinder. The late injection of the fuel in Diesel-engines makes the mixture inhomogeneous, which result in high emissions of unburned particles, soot.

The higher demands for cleaner engines with lower fuel consumption drive the manufacturer to find alternatives. An alternative is the HCCI-engine, which has high efficiency and low emissions of NO_x. The principle of the engine is something in between the SI and CI engine. The fuel and air are mixed together during the intake like in an SI-engine. The homogeneous charge is then compressed and when the temperature is high enough the charge auto-ignites. The temperature at the auto-ignition is about 1000K.

To reach this temperature there is need to increase the initial temperature of the fuel and air. This is done by using residual gas which is warm from previous cycles and already in the cylinder. The initial temperature affects the position of auto-ignition. By controlling the amount of residual gas, the temperature and position of auto-ignition can be controlled. The latter parameter is important to control for high efficiency.

Since the residual gas is already burned, it also slows down the combustion, which avoids problems with knock. If the combustion is too fast it causes pressure oscillations which can damage the engine.

A third function of the residual gas is to lower the emission of NO_x. The presence of residual gas increases the total mass of the cylinder charge, which results in increased heat capacity. This will lower the temperature after the combustion and a lower amount of NO_x emissions is formed. In an HCCI-engine the emissions are so low that no aftertreatment to reduce the NO_x emissions is needed. In the HCCI-engine it is possible to have up to 80% residual gas compared to an SI-engine which can tolerate up to 20% residual gas.

The HCCI-engine has high efficiency during low load conditions compared to an ordinary SI-engine. And since the engine spends most of the time in the car at low load conditions, it is also possible to reduce the fuel consumption by using an HCCI-engine instead of the SI-engine.

3.1 Valve timing

The valve timing for this HCCI-engine is used to control the amount of residual gas and it differs from an ordinary spark ignited engine. In an ordinary engine the intake valve opens before the exhaust valve closes. Some of the exhaust gases will then flow back to the cylinder because the pressure drop during the intake. This valve overlap, which can be seen in figure 3.1 is an important factor that will affect the amount of residual gas in the SI-engine[4]. In the HCCI engine on the other hand the the exhaust valve closes earlier and is totally closed before the intake valve opens. The reason is to trap even more residual gas in the cylinder. This also the causes the second top seen in the pressure-crankangle diagram, fig. 4.1.

4 Cylinder Pressure

The cylinder pressure can give much information about the combustion process. Former the cylinder pressure has only been used as an important tool in research and development of new engines. But when the requirement of better engine control increases it will also be interesting to use pressure sensors in production engines. Figure 4.1 shows a typical pressure vs crankangle

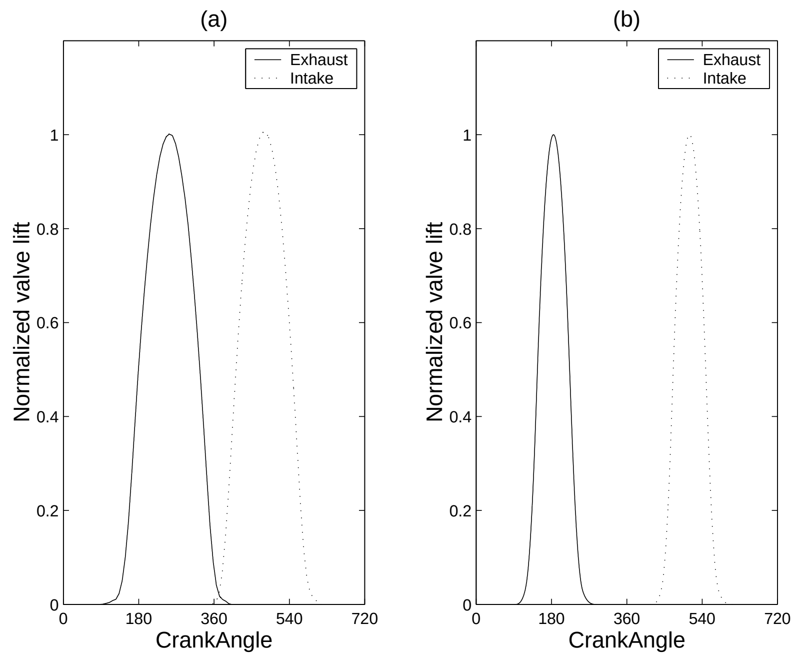

Figure 3.1: Normalized valve lift for an SI-engine(a) and an HCCI-engine(b)

diagram and a pressure vs volume diagram for the HCCI-engine. The first top in the left diagram shows the pressure rise during the combustion and the second top comes from the pressure rise, caused by the trapped residual gas, during the exhaust and intake stroke. The right diagram shows the pressure as a function of the cylinder volume.

4.1 Cycle to cycle variations

The cylinder pressure varies from cycle to cycle even if the engine speed and load are the same. The variation is mainly caused by inhomogeneities in mixture and temperature. This affects for example the burning speed and completeness of combustion, which results in variations in the cylinder pressure. The largest variations occur during low load and high dilution [5]. To avoid the problems caused by these variations, the pressure is averaged over 64 cycles in this report.

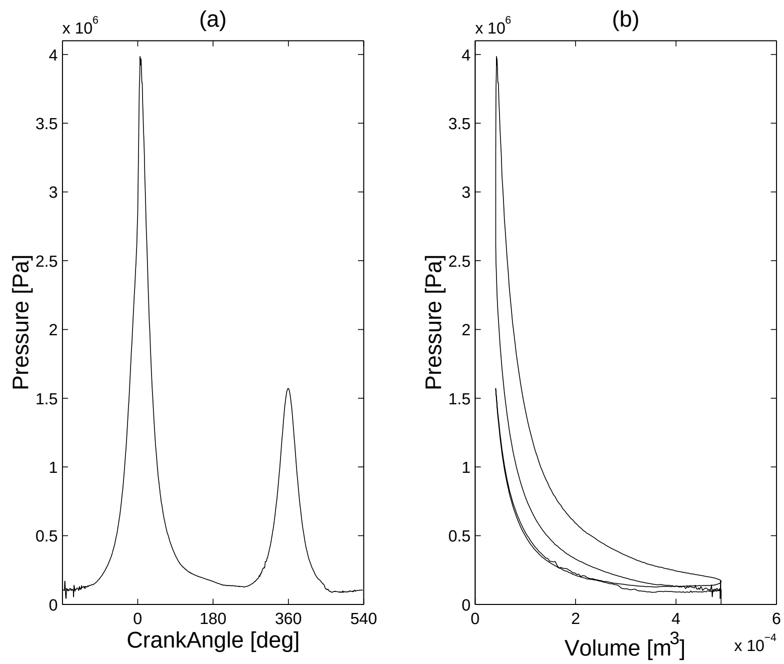

Figure 4.1: Crankangle vs Pressure (a) and Volume vs Pressure (b)

4.2 Offset correction

The cylinder pressure is measured with a piezoelectric transducer, which only measures the dynamic pressure rather than the absolute pressure. To get the absolute pressure there is need for pressure correction. Brunt has investigated some methods to correct the pressure to get the absolute pressure[6]. There are two main alternatives. The first one is to compare the cylinder pressure with the inlet manifold pressure when the intake valve is open at around BDC. Then the pressure in the cylinder should be equal to the inlet manifold pressure. The second method is to use the assumption that the polytropic index is constant during some points in the compression stroke. In this thesis the second approach is used.

The absolute pressure at the first position is p_1 and due to the offset we measure $p_{1m} = p_1 + p_0$ where p_0 is the pressure offset. At the second position we measure $p_{2m} = p_2 + p_0$. With the assumption that the process is polytropic it means that $pV^n = const$, n is the polytropic index and is about 1.32 during the compression. Since the pressure has the same offset at both positions it is then possible to calculate the offset[6].

$$(p_{1m} + p_0)\, V_1^n = (p_{2m} + p_0)\, V_2^n \tag{4.1}$$

$$p_0 = \frac{p_{2m}\, V_2^n - p_{1m}\, V_1^n}{V_1^n - V_2^n} \tag{4.2}$$

4.3 Mass fraction Burned

With the pressure trace it is possible to estimate the fraction of the fuel that have burned so far in the cycle. The method is formulated by Marvin and is originally described in [7].

During the compression and expansion stroke the processes can be described as polytropic, which can be seen as straight lines in the log-log pressure volume diagram. The lines are then extended to the minimum cylinder volume, p_1 and p_3 in fig 4.2. Then also the pressure at the interesting point is extended to the minimum volume, eq.4.3. Before TDC the polytropic index, $n(\theta)$, is the same as during the compression (1.32) and after TDC it is 1.27, the polytropic index for expansion.

$$p_2(\theta) = p(\theta) \left(\frac{V(\theta)}{V_c} \right)^{n(\theta)} \tag{4.3}$$

Then it is possible to calculate the mass fraction burned at p with,

$$mfb(\theta) = \frac{p_2(\theta) - p_1}{p_3 - p_1} \tag{4.4}$$

The result can be seen in fig 4.3. As it is formulated the result always goes from 0 to 1.

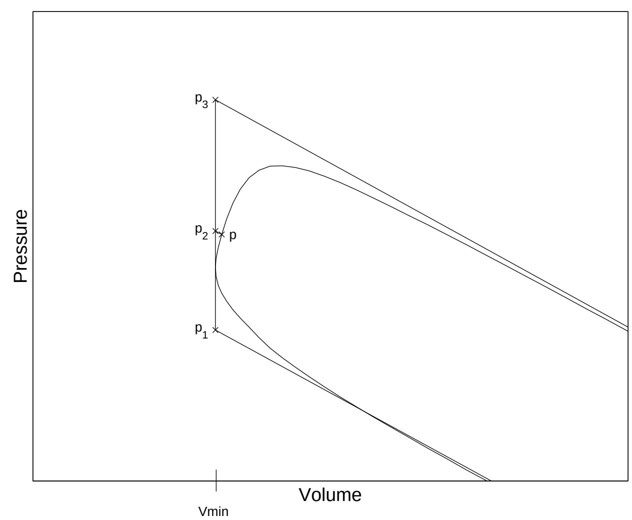

Figure 4.2: Definition of the points used for mass fraction burned calculation

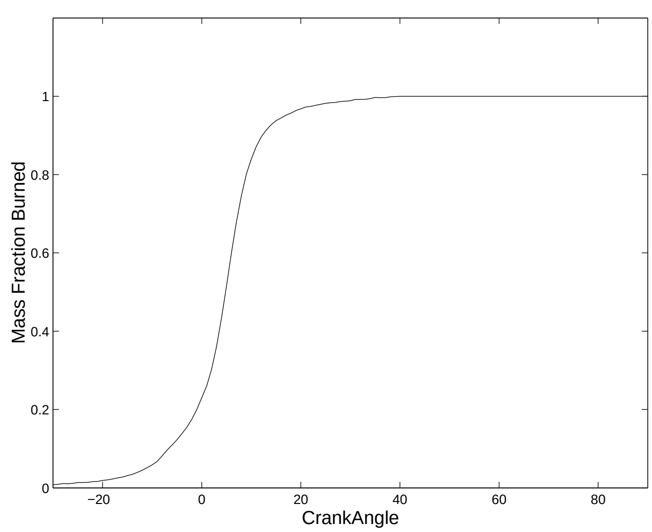

Figure 4.3: Result of mass fraction burned calculation

5 Algorithm

This chapter will first describe the different parts of the algorithm and in the end the whole algorithm is described step by step. The total mass in the cylinder consists of fuel, air and residual gas.

$$m_{tot} = m_{fuel} + m_{air} + m_{rg} \tag{5.1}$$

To estimate the residual gas fraction,(x_{rg}), we are using the fact that the residual gas effects the total mass in the cylinder but do not take part in the combustion process. It works as dilution to the fuel in the same way as the excessive air. It is possible to calculate the residual gas fraction if we know the total mass and the amount of air and fuel in the cylinder. The algorithm should use the cylinder pressure as input.

5.1 Pressure ratio

The pressure is measured before and after the combustion at the same volume, see figure 5.1. The pressure after combustion will be higher than before combustion because the energy in the fuel is transformed to heat in the combustion that will raise the temperature and pressure. The magnitude of the pressure rise is proportional to the combustion energy that is released. The correlation between pressure rise, Δp_{comb}, and combustion energy released is shown in figure 5.2.

$$\Delta p_{comb} = p_2 - p_1 \tag{5.2}$$

If there is no combustion the pressure will be almost the same at the two points.

$$Q_{fuel} = C\Delta p_{comb} \tag{5.3}$$

5.2 Correction function

The process is not totally adiabatic and reversible even if there is no combustion. Without combustion the pressure p_2 will always be slightly lower than p_1 due to heat losses to the cylinder wall and crevice effects. Therefore p_0 is added to p_{comb} in eq. 5.2 to compensate for this.

$$\Delta p_{comb} = p_2 - p_1 + p_0 \tag{5.4}$$

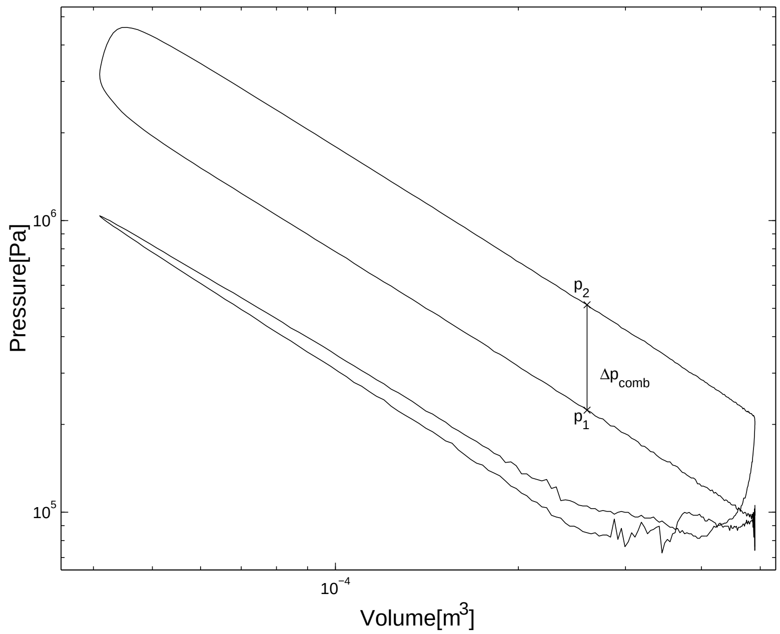

Figure 5.1: Definition of p_1,p_2 and Δp_{comb}

p_0 is in [1] approximately 0.1 bar. One way to check this is to study the pressure difference between the points before and after TDC in a motored cycle, i.e. a cycle without combustion. Another way is to use the second top in the $p-\theta$-diagram instead of a motored cycle. This gives a p_0 between $0.1 - 0.4$ bar. In this work a constant value of 0.2 bar is used. The adding of p_0 also makes the line in fig 5.2 to go closer to origo. The line is a linear least square fit to the data.

5.3 Charge efficiency

The charge efficiency, η_c, is defined to be the ratio between m_{min} and m_{tot}. m_{tot} is the total mass in the cylinder and m_{min} is the theoretical minimum mass of air and fuel that is required to get the combustion energy that is released from the fuel.

$$\eta_c = \frac{m_{min}}{m_{tot}} \tag{5.5}$$

The lower heating value, Q_{LHV}, is the amount of energy that the fuel can release. The minimum mass of fuel will then be Q_{fuel} divided with

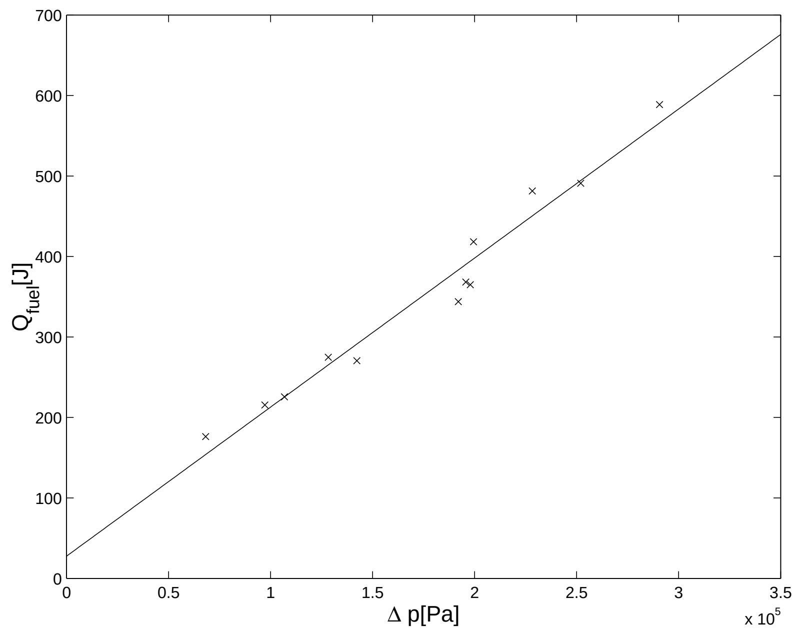

Figure 5.2: Correlation between Δp_{comb} and Q_{fuel}

Q_{LHV}. The amount of air will be the amount of fuel multiplied with the stochiometric ratio, L_{st}, since in the minimum mass there is no excessive air or fuel. This gives the following expression for m_{min}:

$$m_{min} = \frac{Q_{fuel}}{Q_{LHV}} \left(L_{st} + 1\right) \tag{5.6}$$

The main difference between the total mass, m_{tot}, and minimum mass is the presence of residual gas and excessive air. The HCCI-engine operates under very lean conditions, which means that there is more air in the cylinder than is consumed by the fuel ($\lambda > 1$). The actual amount of air in the cylinder will then be expressed:

$$m_{air} = \lambda L_{st} m_{fuel} \tag{5.7}$$

There is also residual gas in the cylinder which is expressed as a part of the total mass.

$$m_{rg} = x_{rg} m_{tot} \tag{5.8}$$

Another difference is that not all the fuel injected in the cylinder will burn. The amount that actually is burned is dependent of several parameters. For example: The homogeneousness of the mixture, temperature differences between the cylinder and cylinder wall and the magnitude of the crevice flows. The amount of fuel in the minimum mass should be divided with the Completeness of Combustion, CoC.

$$m_{fuel} = \frac{1}{CoC}\frac{Q_{fuel}}{Q_{LHV}} \tag{5.9}$$

If the mixture is lean the Completeness of Combustion is high and about 96-98% [8]. The completeness of combustion (CoC) will be more important when single cycle data is treated. In this thesis it is handled as constant 96%.

By combining eq. 5.7-5.9 we get the total mass:

$$m_{tot} = \frac{1}{1 - x_{rg}}\left(\frac{1}{CoC}\frac{Q_{fuel}}{Q_{LHV}}(\lambda L_{st} + 1)\right) \tag{5.10}$$

The differences between m_{min} and m_{tot} are also summarized in table 5.1.

If we put eq. 5.6 and 5.10 in the definition of charge efficiency we can get the following expression for the residual gas fraction:

$$x_{rg} = 1 - \frac{\eta_c}{CoC}\left(\frac{\lambda\, L_{st} + 1}{L_{st} + 1}\right) \tag{5.11}$$

The Completeness of Combustion is set as a constant and lambda is measured with a lambda sensor. The cylinder pressure is used to estimate the charge efficiency, η_c.

$$\eta_c = \frac{m_{min}}{m_{tot}} = \frac{C\Delta p_{comb}}{\frac{p_1 V_1}{R_1 T1}} = C'\, T_1 \frac{\Delta p_{comb}}{p_1} \tag{5.12}$$

The m_{min} is calculated with 5.3 and 5.6 and m_{tot} is calculated with the ideal gas law at 80°BTDC. The gas constant, R_1, and volume, V_1 is included in a new constant, C'. The only parameter still unknown is the temperature, T_1.

5.4 Temperature and total mass in the cylinder

By using the first law of thermodynamics it is possible to estimate the temperature and total mass in the cylinder. The principle of the law is described in 2.4. The internal energy difference between 80° BTDC and

Table 5.1: The difference between m_{min} and m_{tot}

-	Minimum mass	Total mass
Fuel:	$\frac{Q_{fuel}}{Q_{LHV}}$	$\frac{1}{CoC}\frac{Q_{fuel}}{Q_{LHV}}$
Air:	$L_{st}m_{fuel}$	$\lambda L_{st}m_{fuel}$
Res. gas.:	$-$	$x_{rg}m_{tot}$
	$\frac{Q_{fuel}}{Q_{LHV}}(L_{st}+1)$	$\frac{1}{1-x_{rg}}\left(\frac{1}{CoC}\frac{Q_{fuel}}{Q_{LHV}}(\lambda Lst+1)\right)$

50% mfb consists of heat transformed from the fuel, Q_{fuel}, mechanical work, W, and heat losses to the cylinder wall, Q_{wall}. The energy from the fuel can be expressed with combustion efficiency and total mass.

$$Q_{fuel} = \frac{m_{tot}\eta_c Q_{LHV}}{L_{st}+1} \tag{5.13}$$

The mechanical work can be calculated by integrating the pressure from the volume at 80° BTDC to the volume at 50% mfb position.

$$W = \int_{V_1}^{V_{50\%}} pdV \tag{5.14}$$

The heat losses, Q_{wall}, is described to be dependent of the combustion energy and engine speed and the relation comes from simulation data done by Mladek [1]. The relation seems to be suitable also for the HCCI-engine according to the verification done in appendix B.

$$Q_{wall} = 2\cdot\left(\left(\frac{-0.2161}{n} - 0.0049\right)\cdot 2\cdot Q_{fuel} + \frac{-90.78}{n} - 3.91\right) \tag{5.15}$$

The internal energy at 50%mfb will then be

$$U_{50} = U_1 + Q_{fuel} - Q_{wall} - W \tag{5.16}$$

, where U_1 and U_{50} refers to the same datum. The relation between the internal energy and the temperature is

$$dU = mc_v(T)dT \tag{5.17}$$

Since the heat capacity is dependent of the temperature we need to integrate

$$\int dU = m\int c_v(T)dT \tag{5.18}$$

It can be rewritten to

$$U = m\,T\frac{1}{T}\int_0^T c_v(T)dT \tag{5.19}$$

If we set

$$\bar{c}_v = \frac{1}{T}\int_0^T c_v(T)dT \tag{5.20}$$

which then is estimated with a linear approximation. See the next section. Then the temperature can then be calculated.

$$T_{50} = \frac{U_{50}}{\bar{c}_{v50}\,m_{tot}} \tag{5.21}$$

5.5 Iteration and constants

The calculation of temperature and mass in the last section requires knowledge of the heat capacity, c_v, and gas constant, R. These parameters are however dependent on the gas composition and cylinder temperature.

Since the calculation of the temperature and total mass requires these parameters and the parameters are dependent on the temperature, the algorithm is iterated to achieve better result.

The gas constant is only dependent of the gas composition. Only with very high temperatures there is a need to count for dissociation. Dissociation is the breakdown of molecules that can occur at high temperature. For example: carbondioxide breaks down to carbonmonoxide and oxygen. The heat capacity is dependent of both the composition and the temperature. In this thesis the following linear approximations of $\bar{c}_v$ are used[1].

$$cv_{rg50} = 0.2426T_{50} + 615.44[J/kgK] \tag{5.22}$$

$$cv_{fg50} = 0.00482T_{50} + 738.4[J/kgK] \tag{5.23}$$

$$\bar{c}_{v50} = (1 - x_{rg})cv_{fg} + x_{rg}cv_{rg} \tag{5.24}$$

The gas constants for the residual gas and the fresh charge are:

$$R_{fg} = 274[J/kgK] \tag{5.25}$$

$$R_{rg} = 287[J/kgK] \tag{5.26}$$

During the iterations the heat capacity and gas constant is updated when a new T_{50} and x_{rg} is estimated.

5.6 The Algorithm - Step by Step

We now have all the equations to formulate the iterative algorithm. An overview over the algorithm can be seen in fig. 5.6.

Offset Correction Since the pressure sensors only are able to measure pressure changes, the first step in the algorithm will be to calculate the absolute pressure. It is done with the method described in sec. 4.2.

Mass fraction burned The algorithm uses the pressure at 50% mass fraction burned position so the next step is to find the that position with the method described in sec.4.3

Pressure ratio and Mechanical work In the next step the pressure difference between 80°BTDC and 80°ATDC is calculated. Also the mechanical work from 80°BTDC to 50% mass fraction burned position is calculated.

First estimations T_1 is set to 500K and a first estimation of total mass is calculated with the ideal gas law and pressure at 80°BTDC, (p_1).

Charge efficiency and residual gas fraction The pressure ratio is used to estimate the charge efficiency. It is also compensated as described in the section about correction function 5.2. A first estimation of the residual gas fraction is also made.

First law of thermodynamics The mechanical work, estimated Q_{fuel}, Q_{wall} together with the first law of thermodynamics are used to calculate the internal energy. The internal energy is then used to calculate a new total mass.

Iterative loop The calculation of total mass and temperature requires knowledge of heat capacity and gas constant. These are then dependent on the gas composition and temperature. Therefore is the algorithm iterated until the temperature stabilizes.

5.7 Result

The algorithm is tested on a single cylinder engine with the dimensions presented in table 5.2. The test is done during steady state conditions at

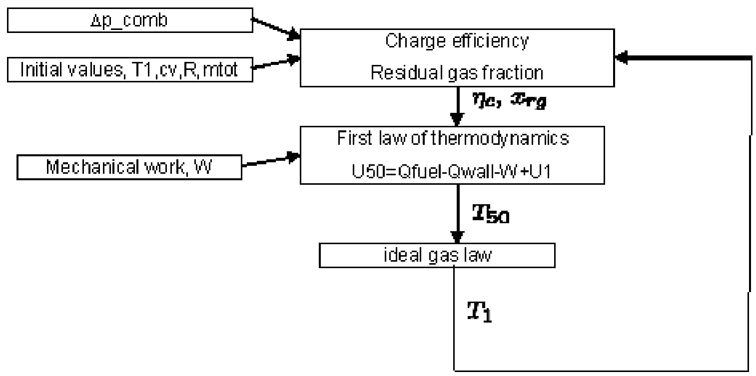

Figure 5.3: Flowchart over the algorithm

speeds and loads that covers the interesting operation area for the HCCI-engine. About 7-13 iterations is needed before the algorithm stops. In fig. 5.4 the result from the algorithm is compared with the simulated value and the first estimation done with constant temperature. In many operation points the iteration does not improve the estimation at all and the result is worse than the first estimation. The absolute error after the iterations is up to 10% compared with the first estimation with an error of 5.5%. The first estimation is of course dependent on which value the initial temperature is set.

Table 5.2: The dimensions of the HCCI-engine

Bore, mm	82
Stroke, mm	85
Connecting rod length, mm	143.5
Compression ratio	11.96

5.8 Residual gas mass

The algorithm does not converge to the right value. A way to solve this problem is to add more information to the algorithm.

Unlike an ordinary SI-engine the exhaust valve closes before the intake valve opens in the HCCI-engine. This means that the mass trapped in the cylinder between exhaust valve close and intake valve open is the mass of the residual gas.

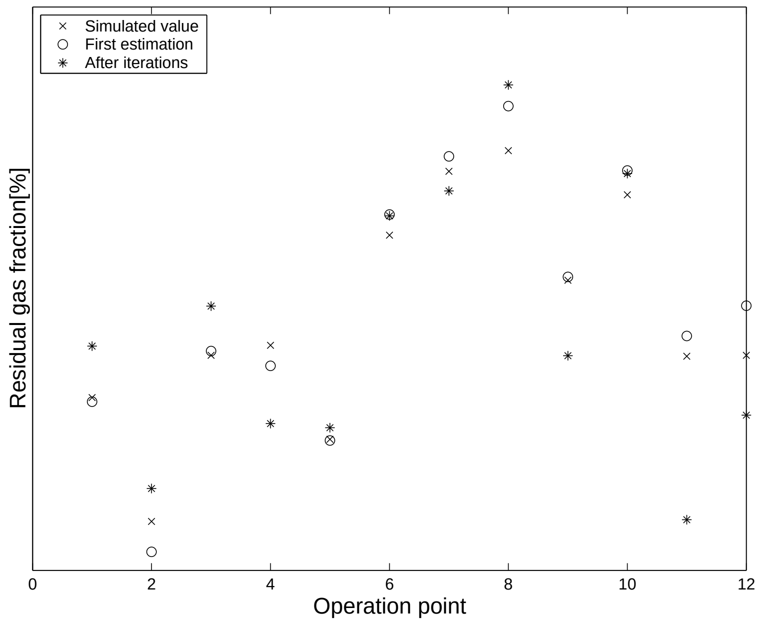

Figure 5.4: Result from the algorithm.

The mass of residual gas is then estimated by using the exhaust gas temperature and the pressure and volume at exhaust valve closing. The closing of the valve causes oscillations on the pressure signal. To avoid some of the disturbances a mean value over 5 samples is used.

$$m_{rg} = \frac{p_{evc}V_{evc}}{R_{rg}T_{exhaust}} \tag{5.27}$$

The residual gas mass can be used in two different ways. Either to get the residual gas fraction directly by dividing the mass with the total mass calculated above. This does not give a good result since a reason to that the algorithm did not work before was that the total mass did not converge to the right value. Another and better way to is to calculate a new total mass with the m_{tot} and x_{rg} from the last iteration.

$$m_{tot} = m_{fuel} + m_{air} + m_{rg} = m_{tot}(1 - x_{rg}) + m_{rg} \tag{5.28}$$

The estimation of the residual gas mass has a relative error up to about 15%. Some of the error is cancelled out because if m_{rg} is too high the total mass will also be too high.

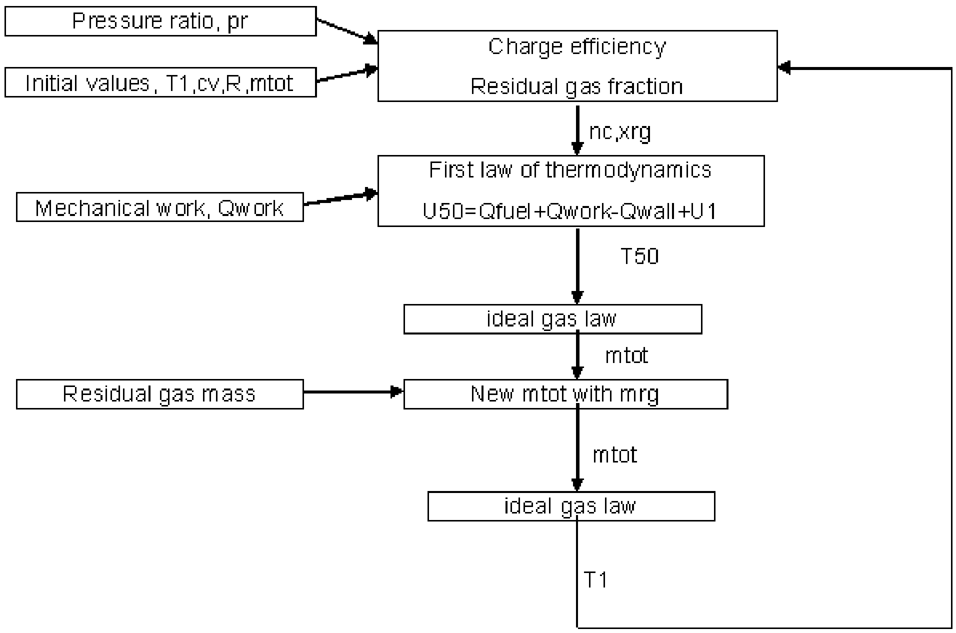

Figure 5.5: Flowchart over the algorithm with m_{rg}

5.9 Result with m_{rg}

A new flowchart over the algorithm with the addition of exhaust gas temperature can be seen in fig. 5.5. The algorithm has been tested at the same engine speeds and loads as above. The result was compared with simulated values and the first estimation done with constant temperature. The result from the algorithm can be seen in fig. 5.9. The use of residual gas mass improved the result. At most operation points the result after the iteration is better than the first estimation. Another advantage of this approach is that it needs fewer iterations, about 4-7, to converge. The first estimation has a max error of 5.5%(absolute) and in average 2.8%. The max absolute error for the algorithm with m_{rg} is 4% and in average 1.5%. It is probably good enough to be used in a control system.

5.10 Alternative formula

The pressure difference, $p_2 - p_1$, was earlier used to estimate the energy released by the fuel. The mass of the fuel and air can then be expressed

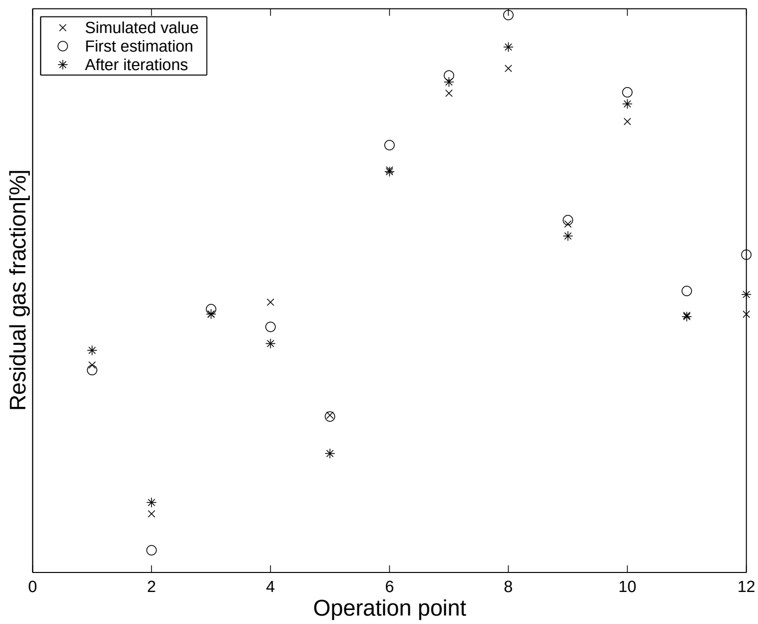

Figure 5.6: Result from the algorithm with m_{rg}

$$m_{fuel} = \frac{Q_{fuel}}{Q_{LHV}\, CoC} \tag{5.29}$$

$$m_{air} = \lambda\, L_{st} m_{fuel} \tag{5.30}$$

When the mass of residual gas is calculated (eq. 5.27) it is possible to calculate the residual gas fraction with the expression

$$x_{rg} = \frac{m_{rg}}{m_{tot}} = \frac{m_{rg}}{m_{fuel}(1 + \lambda L_{st}) + m_{rg}} \tag{5.31}$$

The result from this estimation is presented in fig. 5.7. The absolute error is in the range of $\pm 8\%$.

6 Sensitivity Analysis

An algorithm for estimation of residual gas fraction has been formulated in the preceding chapter. The algorithm uses many variables and it is interesting to investigate how each of them affects the result. What happens

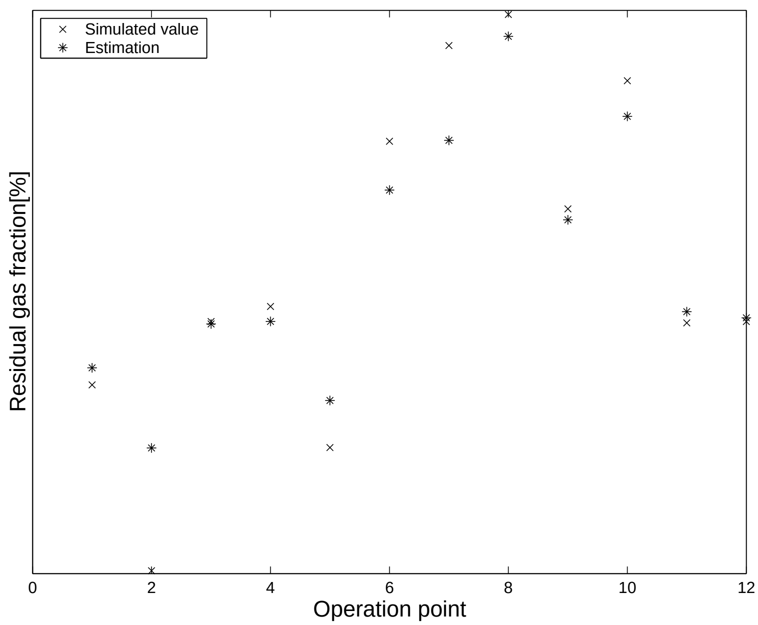

Figure 5.7: Result from the alternative formula

with the result if the pressure is 10% higher or what happens if we assumes that the completeness of combustion is 100% instead of 96%?

Since the algorithm works iteratively and the variables interact with each other, it is not always easy to answer these questions. To see in what direction and how much a variable affects the result, the variable is changed 10% and the changes in the estimated x_{rg} is noticed. The results for the different variables are discussed below and summarized in table 6.1

6.1 Input parameters

The parameters, which are measured are here called input parameters. These are for example the cylinder pressure, exhaust gas temperature, and lambda. The change of 10% in these parameters could correspond to a measurement error.

Cylinder Pressure The cylinder pressure is maybe the most important parameter since the task during this thesis is to use the cylinder pressure to estimate the residual gas fraction. As described in section 4.2 only the dynamic pressure is measured and then offset correction is done before it is used in the algorithm. Therefore a change of 10% in the pressure will make no difference. To see what happens if the offset correction fails is the change of the pressure done after the offset correction. A 10% higher pressure results in about 2% higher residual gas fraction. The change comes mostly from eq 5.27. The pressure is also used in eq.5.12, but here the difference is of interest, which does not change very much.

Exhaust Gas Temperature The temperature is measured after the exhaust valve. The temperature is about 500K and a 10% higher temperature corresponds consequently to 50K and leads to about 2% lower x_{rg}. In this thesis only steady state engine operation is considered. But it can be good to have in mind when looking at dynamic changes between the operation points because it takes some time for the temperature to stabilize. The decrease in x_{rg} comes from eq. 5.27 there a lower mass of residual gas will be calculated when the temperature is higher.

Lambda Maybe it is not realistic to have such a big change in lambda as 10%. But it can anyway be interesting to see in what direction the residual gas fraction changes with a higher lambda. The residual gas fraction will be about 2.3% lower when the value of lambda is increased 10%. The explanation is that the algorithm then assumes that the difference between m_{min} and m_{tot} consists of a bigger fraction excessive air and lower fraction of residual gas.

Crank angle and pressure Not only the pressure offset is important when talking about the cylinder pressure. The phasing between pressure and crankangle can also cause problems, since it is the difference in pressure before and after combustion that is considered. Before combustion the pressure is increasing and after the combustion the pressure is decreasing with the result that an error with crankangle phasing causes a big change in the pressure difference. A change of one degree between the pressure and crankangle results in a change in x_{rg} of 1-3%.

6.2 Approximations

Some approximations are done in the algorithm to simplify the reality and to make the estimation faster. These are the heat transfer, heat capacity and Completeness of Combustion.

Table 6.1: Result of the sensitivity analysis

Variable	Change	Result on x_{rg} % mean	max
Lambda, λ	+10%	-2.3	-2.8
Pressure, p	+10%	+2	+2.3
Theta phasing	+1°	-1.97	-3.37
Exhaust gas temperature	+10%	-2	-2.8
Completeness of Combustion	+4%	+1.03	+1.27
Wall heat losses, Q_{wall}	+10%	+0.1	+0.2

Wall heat losses A change of 10% of the wall heat losses will result only in a very small change. The reason is that the dominating part in the equation (5.16) there this losses is used is Q_{fuel} and W.

Completeness of Combustion This parameter is set constant 96% in the algorithm. If it is set to 100% the algorithm will give a result which is about 1.5% higher. It is because the difference between m_{min} and m_{tot} will consist of a smaller fraction of unburned fuel and higher fraction of residual gas.

Other approximations that is done is the linear approximation of the heat capacity and the fuel dependent constants, Q_{LHV} and L_{st}. But changes to these parameters do not change the result that much($< \pm 1\%$).

7 Summary

The HCCI-engine is a new kind of engine where the charge is homogenously mixed during the intake and then compressed to auto-ignition. The engine has shown high efficiency in combination with low emissions. But to achieve this there is need to control the position of auto-ignition and the combustion speed. Both of these parameters are affected by the residual gas fraction. The residual gas is already burned gas left in the cylinder from previous engine cycles. In this report an algorithm for estimating the residual gas fraction with the cylinder pressure has been formulated. By measuring the cylinder pressure information is obtained about the combustion from the inside of the cylinder.

The residual gas fraction can be calculated from knowledge about the amount of fuel, air and the total mass in the cylinder. The pressure difference between a point before and another point after the combustion correlates well with the energy released by the fuel, which gives the amount of fuel in the cylinder. The amount of air is then related to the fuel via lambda.

The total mass is related to the cylinder pressure via the ideal gas law. But it requires knowledge about the temperature in the cylinder. An attempt was made to use the first thermodynamic law together with a simple approximation of the wall heat losses to determine the temperature. Since the temperature is dependent of the mass, iterations between ideal gas law and first law of thermodynamics were done. The iterations converged but to wrong temperature with the result that it did not improve the result. A constant cylinder temperature at a certain point produces better result. The reason to the iteration to fail is probably that the input, the pressure in this case, does not contain enough information or that the used model of first thermodynamic law is too simple.

To improve the total mass estimation the exhaust gas temperature was introduced. The result was much better. Probably because the algorithm gets more information.

Since there is no method available to measure the residual gas fraction the result from the algorithm was compared with simulated values. Different engine speed and load is used and an error of $\pm 4\%$ was noted compared with the simulated value. A problem due is that same data has been used both to develop the algorithm as to test it. This will probably result in a poorer result with other data. Further testes have to take place before the margin of error can be determined.

The influence of changes in the parameters has also been investigated.

The most parameters, if changed individually, only change the result with a few percent. But if several parameters are changed at the same time it may cause a big errors in the result.

The algorithm is not mathematical complex and it requires only a small number of iterations to converge when the exhaust temperature was used. Therefore is it possible to assume that there are no problems to implement and get it running in a car. An implementation of the algorithm also requires, besides the pressure sensors, the exhaust gas temperature to be measured or estimated in some way. There is of course need for calibration of the algorithm for every new kind of engine.

7.1 Further investigations

Dynamic estimation In this work only steady state behavior have been studied. It needs to be further investigated if there is possible to use same approach when the engine is changing between the operation points.

Multi cylinder A data used in this thesis comes from a single cylinder engine. A multi cylinder engine will be more complicated since there can be difference in temperature between the different cylinders. This may also affect the exhaust gas temperature and cause disturbance on the algorithm used here.

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Appendix A: Engine Data

The dimensions of the engine can be seen in table A.

Table A.1: Engine dimensions

Bore, mm	82
Stroke, mm	85
Connecting rod length, mm	143.5
Compression ratio	11.96

The pressure has been measured at different engine speeds and loads together with the corresponding crankangle at a resolution of 1 degree. The crankangle has then been used to calculate the corresponding cylinder volume. Several other parameters has been measured but only lambda and exhaust gas temperature are used in the algorithm. For the investigation also the measured mass of fuel and air has been used together with the simulated residual gas fraction.

Appendix B: Heat transfer

The first law of thermodynamics

$$dU = \delta Q_{fuel} - \delta Q_{wall} - \delta W - \sum_i dm_i h_i \tag{B.1}$$

δQ_{fuel} is the chemical energy that is released from the fuel. dU is the change in internal energy of the mass in the cylinder and δQ_{wall} is the heat losses to the cylinder wall. W is the work from the fluid to the piston. The last term is the changes in the enthalpy due to the flows of mass into and out from the cylinder. If we consider just when the valves are closed and assumes that there are no crevice effects the last term can be skipped. Crevice effects are when mass in the cylinder blows by the piston and cylinder wall. The crevice effects are small compered to the other terms.

The internal energy can be expressed:

$$dU = mc_v dT \tag{B.2}$$

and with the ideal gas law it is possible to rewrite it as

$$dU = m\, c_v\, d\left(\frac{pV}{mR}\right) = m\, c_v \left(\frac{V}{mR}dp + \frac{p}{mR}dV\right) \tag{B.3}$$

,since there is no change in the mass.

δW is as before

$$\delta W = pdV \tag{B.4}$$

The relations between heat capacity, gas constant and γ are:

$$\gamma = \frac{c_p}{c_v} \tag{B.5}$$

$$R = c_p - c_v \tag{B.6}$$

Equation B.1 can then be written

$$Q_{wall} = Q_{fuel} - \int_{\theta_1}^{\theta_{50}} \left(\frac{1}{\gamma - 1} V \frac{dp}{d\theta} + \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta}\right) d\theta \tag{B.7}$$

Q_{fuel} is calculated with the lower heating value, Q_{LHV} and CoC here set to 98%.

$$Q_{fuel} = 0.98\, m_{fuel} Q_{LHV} \tag{B.8}$$

The other term require knowledge of γ. It can be expressed as a linear approximation dependent of the temperature [9].

$$\gamma = 1.4 - 7.18 \cdot 10^{-4} T(K) \tag{B.9}$$

The temperature is calculated with the ideal gas law and total mass in the cylinder and the total mass in the cylinder is calculated using the measured mass of fuel and air together with the simulated residual gas fraction.

Equation B.7 is used on the data from the HCCI-Engine and then compared with the approximation of Q_{wall} used in sec. 5.4. See fig. B.1

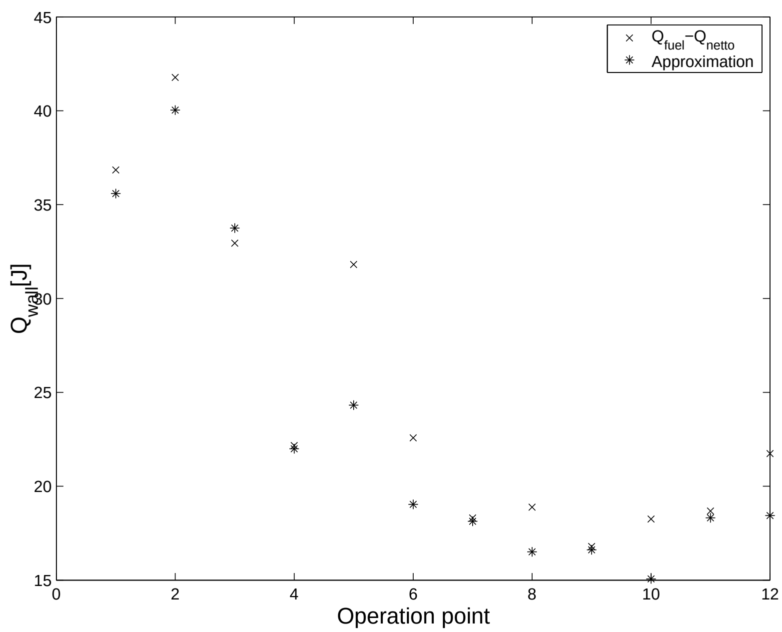

Figure B.1: Comparison between $(Q_{fuel} - Q_{netto})$ and the Approximation