Cylinder-Pressure Based Injector Calibration for Diesel Engines

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Abstract

One way of complying with future emission restrictions for diesel engines is to use pressure sensors for improved combustion control. Implementation of pressure sensors into production engines would lead to new possibilities for fuel injection monitoring where one potential use is injector calibration.

The scope of this thesis is to investigate the possibility of using pressure sensors for finding the minimal energizing time necessary for fuel injection. This minimal energizing time varies over the injector's lifetime and therefore requires a re-calibration. The necessary energizing time can be found by estimating the injected fuel mass at different rail-pressures during a calibration state operating under specific engine conditions. Two different approaches based on in-cylinder pressure were used for fuel mass estimation. The result is based on a comparison to a non pressure based production line calibration function.

Both fuel mass estimations show a correlation with convincingly accuracy for calibration use but with the possibility of further improvements. One approach is shown to be less sensitive to signal offsets but more sensitive to noise. The offset sensitiveness can be reduces by changing measurement positions depending on user requirements. Compensations for energy losses depending on engine speed and cylinder differences are shown to be necessary for calibration accuracy. Moreover are both injected fuel mass and rail-pressure shown to influence the combustion.

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Also everyone at the office deserves a big hug for giving me such a joyful stay and for giving me a taste of some lovely German traditions. I hope that I one day will have the opportunity to own a Mercedes; they are truly a piece of art.

Contents

Symbols

A	[m^2]	Heat transfer surface area
c_v	[$\frac{\mathrm{J}}{\mathrm{kg \cdot K}}$]	Specific heat capacity
C	[-]	Arbitrary constant
CoC	[-]	Completeness of Combustion
d	[mm]	Cylinder bore
h	[$\frac{\mathrm{kJ}}{\mathrm{kg}}$],[$\frac{\mathrm{W}}{\mathrm{m^2 \cdot K}}$]	Enthalpy or heat transfer coefficient
H	[mm]	Piston stroke
$IMEP$	[Pa]	Indicated Mean Effective Pressure
l	[m]	Connecting rod length
L_{st}	[°CA]	Crank angle of 50% heat released
LHV	[$\frac{\mathrm{kJ}}{\mathrm{kg}}$]	Lower Heating Value
m	[kg]	Mass
m_{fuel}	[kg]	Injected fuel mass
m_{min}	[kg]	Theoretical minimal injected fuel mass
m_{tot}	[kg]	Total mass in cylinder after injection
p	[Pa]	Pressure
$p_{0,adapt}$	[Pa]	Adaptive pressure compensation
p_{120}	[Pa]	Pressure at 120°CA after TDC
p_{φ}	[Pa]	Pressure at φ°CA from TDC
p_{O1}	[Pa]	Start pressure for pressure offset correction
p_{O2}	[Pa]	End pressure for pressure offset correction
p_N(N)	[Pa]	Pressure compensation for speed
Q	[J]	Energy
Q_{comb}	[J]	Combustion energy
Q_n	[J]	Net heat release
Q_{ht}	[J]	Energy heat transfer
R	[$\frac{\mathrm{J}}{\mathrm{K \cdot mol}}$]	Ideal gas constant

T	[K]	Temperature
T_{120}	[K]	Temperature at 120°CA after TDC
T_{gas}	[K]	Gas temperature
T_{wall}	[K]	Cylinder wall temperature
U	[J]	Internal energy
V_{120}	[m^3]	Volume at 120°CA after TDC
V_{min}	[m^3]	Minimum cylinder volume
V_{O1}	[m^3]	Start volume for pressure offset correction
V_{O2}	[m^3]	End volume for pressure offset correction
V_{piston}	[m^3]	Piston displacement volume
W	[J]	Work
W_{HP}	[J]	Work during high pressure phase
x_b	[-]	Mass portion of combustion gas in cylinder
x_{HR}	[-]	Normalized heat release
λ	[-]	Air-fuel ratio in cylinder
λ_a	[-]	Exhaust gas air-fuel ratio

Abbreviations

BDC	Bottom Dead Center
°CA	Degree Crank Angle
CO	Carbon Monoxide
CRDi	Common Rail Direct injection
ECU	Engine Control Unit
H_2	Hydrogen
HC	Hydrocarbon
HP	High Pressure
IIR	Infinite Impulse Response
LP	Low Pressure
TDC	Top Dead Center
ZFC	Zero Fuel Calibration

1. Introduction

1.1 Background

With an increasing development of injection systems more advanced injection control is made possible. Modern diesel injectors based on a so called "piezo effect" is allowing electronic control of both injection time and fuel amount with high precision. Together with Common Rail Direct injection (CRDi) technology, which allows injection directly into the combustion chamber under constant high pressure, multiple injections are made possible for improving engine performance and for reducing emissions and noise.

Diesel engines are usually associated with knocking and rattling sounds which are caused by the sudden temperature increase in the combustion chamber at combustion. By injecting a tiny amount of fuel, a so called "pilot injection", before starting main injection a small explosion takes place before the main combustion. This cause the temperature in the combustion chamber to gradually increase which reduce the noise almost to the level of gasoline engines.

There is a mechanical manufacturing tolerance for injectors based on the piezo effect. This tolerance limits the precision for each injector which for example results in individual energizing time necessary for start injecting fuel. The system used for controlling the injection is updated with each injector's individual behaviour during the engine assembly process which improves the precision. With injector wear over time its properties change. This wear calls for a re-calibration of the stored information describing the individual injector's behavior for correct injection.

1.2 Problem definition

The scope of this thesis is to investigate the possibility of using in-cylinder pressure measurement for individual injector calibration. The calibration is used to find the smallest energizing time necessary for fuel injection, also called Zero Fuel Calibration (ZFC). This should be made with a focus on using less capacity demanding calculations.

1.3 Thesis outline

To begin with, an introduction is given to fundamental knowledge concerning the topic of this thesis. This chapter is not of interest for readers with knowledge in combustion engines and in-cylinder pressure measurement.

The consecutive chapter describes theory within the field of diesel engine combustion. This chapter is of importance for fully understanding the problematics of injector calibration out of in-cylinder pressure and the complexity of the combustion.

The measurement chapter gives a more detailed view of the system that is used and the concept of in-cylinder pressure measurement. Here, some disturbance factors are described which may cause errors in the pressure measurement.

The calibration chapter describes the calibration concept and its sensitiveness to measurement error. This chapter is based on the information described in the previous chapters.

The implementation chapter describes how the function is implemented and how its functionality is designed for testing and validation.

Last of all, results and conclusions are presented and suggestion on future work is described.

2. Fundamentals

2.1 Diesel engine

There are many different types of diesel engines. In marine and stationary engines, a two-stroke turbo charged configuration is most frequently used, while in smaller engines a four-stroke cycle is more common. Diesel engines are often larger and more rigid built than spark-ignition engines because of higher stress levels due to higher pressure in the combustion cycle.

For injection of the diesel fuel into the combustion chamber there are two main techniques, direct-injection and indirect-injection. In the direct-injection method, fuel is injected directly into the cylinder, where it is mixed with air. For a indirect-injection, the engine has an auxiliary injection combustion chamber where the fuel is mixed with air. This technique is used when a faster fuel-air mix is needed, i.e. small engines operating at high speed. For fuel economy and power density the direct-injection system has a clear advantage and is therefore state of the art in passenger cars.

In contrast to spark-ignition engines, diesel engines do not use spark plugs. Here, the fuel-air mix is compressed by the piston which results in a temperature rise. This temperature rise is large enough for the gas to self-ignite.

2.1.1 Four-stroke process

A four-stroke engine works with an operating cycle of four steps, where each cycle takes place in two revolutions of the crankshaft. This gives that each step is one fourth of a complete cycle, see Figure 2.1.

1st stroke: Intake -360 °CA — -180 °CA

The intake stroke starts with the piston at TDC and ends at BDC. During this step, air is drawn into the cylinder from the inlet valve which is opened.

2nd stroke: Compression -180 °CA — 0 °CA

During compression, both valves are closed and the air in the cylinder is compressed from BDC to TDC. Close to TDC, fuel is injected into the cylinder.

3rd stroke: Expansion 0 °CA — 180 °CA

The highly compressed air-fuel mixture self-ignites at TDC. This increases the pressure in the cylinder and the mechanical work from the piston forces the crankshaft to rotate. At the end of the expansion stroke the exhaust valve opens which reduces the pressure in the cylinder.

4th stroke: Exhaust 180 °CA — 360 °CA

In this stroke, the remaining burned gas exits the cylinder through the exhaust. Just before TDC the inlet valve opens and a new four-stroke cycle starts.

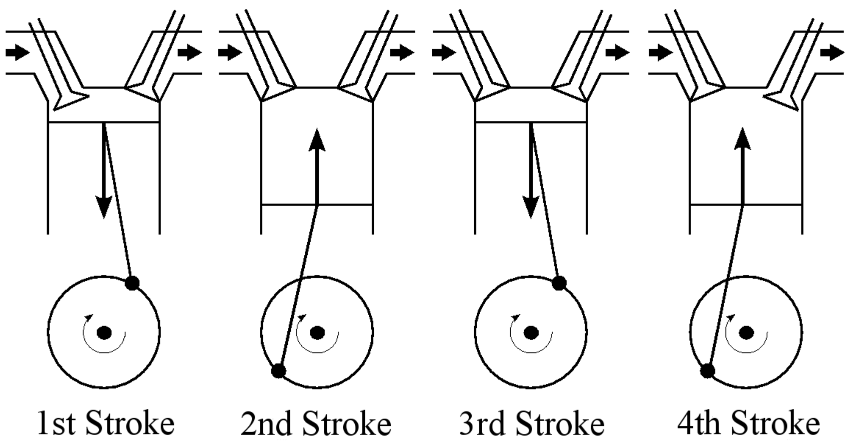

Figure 2.1: Four stroke principle

2.2 Injection system

The injection system has the task to inject the correct amount of fuel at high pressure and at the right time into the combustion chamber. The amount of fuel injected is controlled by changing the rail-pressure and the energizing time. Higher rail-pressures result in an increasing flow of fuel through the injector nozzle, which leads to more fuel injected over time. The energizing time controls how long the nozzle is kept open for fuel injection.

An ECU is used to control the injection parameters and can be described as the "brain" of the injection system. The injection is controlled according to pre-specified maps, curves and constants stored in the ECU.

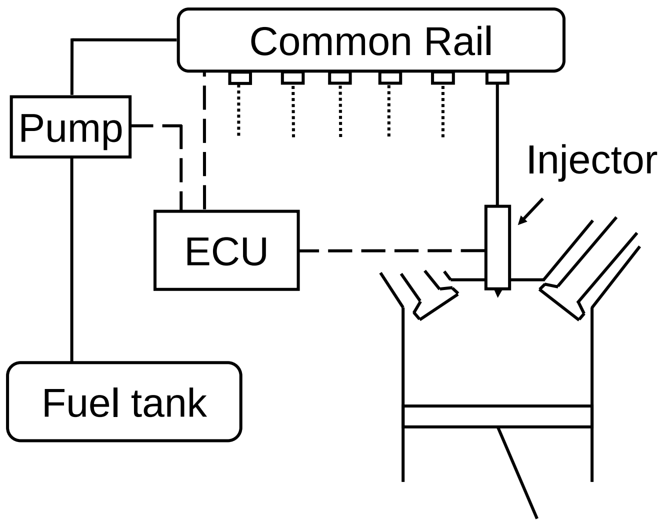

Figure 2.2: Simplified illustration of an injection system

2.2.1 Injector

The injector is used to inject fuel into the cylinder. For optimal injector performance, there are mainly three factors. The first two are injection timing and injection amount, which both are controlled by the injection system. The third factor is fuel atomization, which depends on the nozzle of the injector and its pressure limitation. Fuel atomization means the capability of the injected fuel droplets to be of the correct size and have a correct dispersion in the cylinder which gives a more homogeneous air-fuel mixture. By tuning these factors, engine performance can be improved with e.g. lower emissions and better noise characteristics as a result.

For the injection control system to function properly, the injectors have to meet stringent requirements of quality. In a diesel engine, variations in injection parameters, e.g. quantity and timing, from specified control values will cause unwanted changes in combustion properties. This may be caused by a faulty injector or basically by the mechanical wear of the injector which changes its properties over the lifetime.

Some manufacture's measure each injector after production for its specific properties and programmes these characteristics into the ECU. With the use of in-cylinder pressure sensors more information about the combustion proceedings, for each cylinder individually, is given, which may be used to calibrate the injectors for improved performance.

2.2.1.1 Piezo injector

The piezo injector is based on the so called piezo effect. This means that when an electrical charge is applied to a piezo element, it will stretch out. The stretched displacement is proportional to the applied charge. When the element is discharged it will stretch back to its normal condition. Piezo injectors consist of many small discs separated by metallic electrodes. When these discs are charged, a small displacement occurs. With this technique it is possible to control the injection valve. Since the injection is of great importance to the engine performance, the use of piezo injectors in diesel engines is increasing. The piezo injector has the advantage of having a faster response and can work with higher pressures than its precursor, the solenoid injector. With multiple injections for noise and emission reduction, higher demands are made on the injector capacity.

2.2.2 Common Rail

The common-rail system is frequently used in diesel engines for passenger cars. This technology has the advantage of adapting the correct injection pressure independently from engine speed and fuel quantity. Common-rail injection systems works with a constant fuel pressure which makes it possible to adjust multiple injections at each point for noise and emission reduction. It also gives the advantage of having a constant fuel pressure during the entire injection period which results in higher average pressure than other common injection systems, e.g. cam-driven injection systems. With modern common-rail technology it is possible to execute up to seven separate injections for each cylinder at one combustion cycle.

2.3 Pressure sensors

Pressure sensors are used to measure the pressure in the cylinder. There are variations of techniques for sensoring the pressure which differ from each other in e.g. production cost, electro magnetic sensitiveness and measurement precision. An important issue for implementation in production vehicles is the mounting of the sensor. Here, a concept exists where the sensor is included in the glow-plug which simplifies the implementation in production engines.

2.3.1 Piezoelectric pressure sensor

When piezoelectric materials are affected by pressure, they will get a displacement of charges due to deformation. This causes an electrical output which is highly proportional to the applied pressure. Commonly used material with this type of behavior mono crystalline materials like quartz or certain ceramics. An advantage with the piezoelectric pressure sensor is its linearity in a very wide pressure range and also insensitive to electro magnetic fields. One drawback with this type of sensor is its sensitiveness to temperature where the internal resistance is affected. One other disadvantage is that a steady state is not possible since physical work is required to generate an output signal.

2.3.2 Piezoresistive pressure sensor

Both piezoresistive and piezoelectrial materials show changes in characteristics when they are affected by pressure. For piezoresistive materials the effect is a change in resistance instead of electrical output as for piezoelectrical materials. One drawback with this effect is that it demands a power supply for measuring the change of resistance. On the other hand it is possible to use the piezoresistive effect even in steadystate since the energy source is given by the power supply and not the physical work. Silicon is a commonly used material for piezoresistive pressure sensors.

2.3.3 Optical pressure sensor

Optical pressure sensors use a phase-shift from a light source to measure the pressure. The light is transferred via a optical fiber directly onto a diaphragm, which reflects the light back into the optical fiber. The light that returns is then measured by an optical spectrum analyzer. Since there is a gap between the optical fiber and the diaphragm which changes with the applied pressure, a difference in the return light spectrum is shown. This spectral shift is proportional to the applied pressure.

3. Theory

3.1 Cylinder volume

The cylinder volume at a specific crank position is calculated from geometrical properties of the engine [PKS02].

$$V(\varphi) = V_{\mathrm{min}} + \frac{V_{\mathrm{piston}}}{2 \cdot r} \cdot \left[r \cdot (1 - cos\varphi) + l \cdot \left(1 - \sqrt{1 - \left(\frac{r}{l} \right)^2 \cdot sin^2\varphi} \right) \right] \tag{3.1}$$

Here, r is half the piston stroke H, l the connecting rod length and V_{min} the clearance cylinder volume which is equal to the minimal cylinder volume.

The piston displacement volume V_{piston} is given by the cylinder bore d and the stroke H as

$$V_{\mathrm{piston}} = \frac{\pi \cdot d^2}{4} \cdot H. \tag{3.2}$$

3.2 Combustion

3.2.1 Heat Release

The engine "heat release" is an analyze method based on the first law of thermodynamics and defines the rate of with the chemical energy in the fuel is released in the combustion process. The heat release can be described in relation to time or in reference to crank angle [Cre07]. Using heat release as a way to analyze the engine combustion is most common for diesel engines and has the advantage of describing a direct relation between cylinder pressure and the amount of chemical fuel energy release by combustion [Hey88, BRE98]. The heat release only focus on the part of combustion where both intake and exhaust valve are closed [Kle04].

For the possibility of using the heat release analysis, firstly a model of the engine combustion must be derived. A simple way to create a model of the combustion is to see it as single zoned. In a single zone model the entire combustion chamber is seen as a uniformed single volume with the same heat, pressure and air-fuel composition throughout the entire area [Cre07]. The possibility is also to see the combustion as multi-dimensional with e.g. non-uniform heat and pressure dispersal. This on the other hand gives a more complex description that requires more calculation capacity and often is the result not much better than for the single-zone model with only one pressure measurement [BRE98, AFFS00].

The engine combustion can be described with the first law of thermodynamics, the energy conservation equation, as

$$dU = dQ - dW + \sum_i h_i \cdot dm_i. \tag{3.3}$$

Here, dU is the change of the internal energy of the mass in the system, dQ is the heat transported to the system, dW is the work produced by the system and $\Sigma_i h_i dm_i$ is the mass flux term representing flow across the system boundary [Hey88, Kle04]. The mass flows in the system can be e.g. injected fuel mass, flows from leaking crevice regions and piston ring blow by. The term h_i is used to describe the enthalpy of flow i for the equivalent specific mass [Kle04].

Heat transport dQ, describing the change of heat in the system, can be divided into added heat, dQ_{comb}, from released chemical energy from the fuel and loss of heat, dQ_{ht}, transferred to the chamber walls [Kle04]. This gives a heat representation in the system of

$$dQ = dQ_{comb} - dQ_{ht}. \tag{3.4}$$

Previous work has shown that the effects of mass flow in the system is usually very small for production engines and can therefore be neglected [Hey88, BP99]. The system can now be rewritten as

$$dQ_{comb} = dU + dW + dQ_{ht}. \tag{3.5}$$

The heat release rate can be described in two different ways, net and gross heat release.

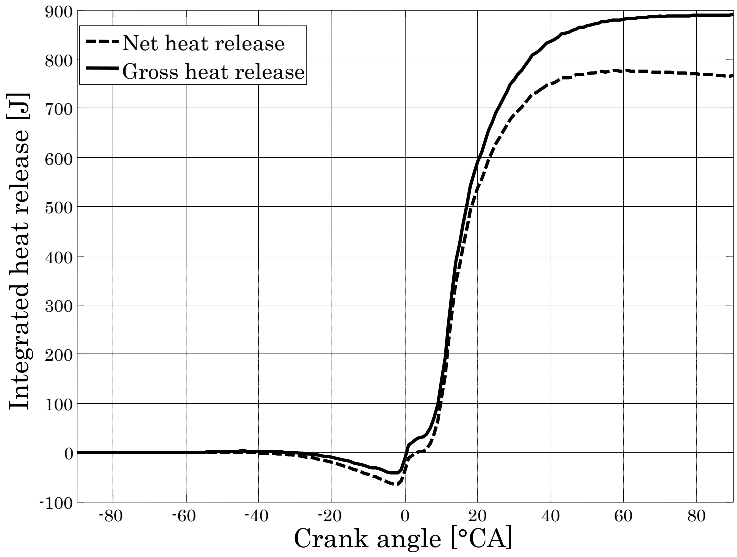

Figure 3.1: Integrated gross and net heat release. The release of energy from the injected fuel approximately begins where the curve reaches its minimum value

The net heat release, dQ_n, is the difference between the gross heat release, dQ_{comb}, and the heat-transfer to the chamber walls, dQ_{ht} [Hey88].

$$dQ_n = dQ_{comb} - dQ_{ht} = dW + dU \tag{3.6}$$

Further dU can be expressed as $dU = m \cdot c_v \cdot dT$, where m is the trapped mass and c_v is the mean specific heat capacity at constant volume. If the contents in the cylinder is assumed an ideal gas [Hey88], then equation (3.6) becomes

$$dQ_n = p \cdot dV + m \cdot c_v \cdot dT \tag{3.7}$$

The ideal gas law,

$$p \cdot V = m \cdot R \cdot T \tag{3.8}$$

with R as constant, gives $d(p, V)/(m \cdot R) = dT$, and can be used in equation (3.7) to eliminate dT, which gives

$$dQ_n = \left(1 + \frac{c_v}{R}\right) \cdot p \cdot dV + \frac{c_v}{R} \cdot V \cdot dp \tag{3.9}$$

where $\frac{c_v}{R} = 2.39 + 0.0008 \cdot \frac{T_{120}}{p_{120} \cdot V_{120}} \cdot p \cdot V$.

This equation was introduced by Hohenberg (1982) and is commonly used in both gasoline and diesel engines for heat release analysis [Mul03].

3.2.1.1 H_{50} position

H_{50} describes the crack angle position where 50% of the injected fuel is burned. It is calculated by normalizing the heat release between its minimum and maximum value.

$$x_{HR}(\varphi) = \frac{Q_n(\varphi) - Q_{n_{min}}}{Q_{n_{max}} - Q_{n_{min}}} \tag{3.10}$$

The crank position φ where the normalized heat release $x_{HR}(\varphi) = 0.5$ describes the H_{50} position.

3.2.2 Heat transfer

When heat is transferred to the cylinder walls the pressure in the cylinder decreases as described by the ideal gas law in equation (3.8). The energy transferred to the combustion chamber walls can be calculated by using Newton's law of cooling

$$\frac{dQ_{ht}}{dt} = h \cdot A \cdot \Delta T = h \cdot A \cdot (T_{gas} - T_{wall}). \tag{3.11}$$

Here, A is the surface area of heat transfer, h is the heat transfer coefficient and ΔT the appropriate temperature difference. The wall heat transfer is typically 15% of the gross heat release [BP99], but varies depending on operating conditions. Normal variation is between 8% at low load and high speed and up to 30% at high load and low speed [MT07].

The heat transfer can not be neglected when estimation the gross heat release.

3.2.3 Mean effective pressure

The indicated mean effective pressure ($IMEP$) calculation is based on the indicated work per cycle and the cylinder volume as

$$IMEP = \frac{W}{V_{piston}}. \tag{3.12}$$

IMEP is a fictitious constant pressure that gives the same work for one complete cycle as the pressure trace [HRB+05].
From the relationship between cylinder volume and pressure, the indicated work per cycle can be calculated by integrating the pressure over volume as

$$W = \int p \cdot dV. \tag{3.13}$$

The indicated work describes the workload performed in one cycle.

Equation (3.12) and (3.13) give the relation

$$IMEP = \frac{\int p \cdot dV}{V_{piston}} \tag{3.14}$$

The mean effective pressure can be divided into two parts, high pressure

$IMEP_{HP}$ and low pressure $IMEP_{LP}$. In this work only $IMEP_{HP}$ is used and includes compression and combustion stroke.

3.2.4 Completeness of Combustion

Completeness of Combustion (CoC) describes the percentage of injected fuel that is burned at combustion. The combustion becomes incomplete if there is not enough available oxygen to burn the injected fuel to chemical equilibrium. The result of a incomplete combustion is an increment of unwanted exhaust gas components like CO, H_2, HC and soot [MSS05].

By measuring the exhaust gas values HC [ppm], CO [ppm] and λ_a, the CoC can be calculated [BBF01] as

$$CO_v = \frac{1 + \lambda_a \cdot L_{st}}{1 + L_{st}} \cdot CO \tag{3.15}$$

$$\eta_{CO} = 1 - 4.5 \cdot \frac{CO_v}{1e6} + 7 \cdot \left(\frac{CO_v}{1e6}\right)^2 \tag{3.16}$$

$$\eta_{HC} = 1 - \frac{HC}{1e6} \cdot (1 + \lambda_a \cdot L_{st}) \tag{3.17}$$

$$CoC = \eta_{CO} \cdot \eta_{HC} \tag{3.18}$$

where CoC is given as a value between 0 and 1.

3.2.5 Combustion energy

At combustion, energy is released from the injected fuel which causes the in-cylinder pressure and the gas temperature to rise. Depending on the quality of the injected fuel and the amount of fuel that is completely burned, the energy release varies. The relation holds [Hey88]

$$Q_{comb} = CoC \cdot m_{fuel} \cdot LHV. \tag{3.19}$$

LHV is a fuel specific value describing the relation between fuel mass and energy release. High quality fuel needs less amounts for the same release of energy which is described by a higher LHV.

3.2.6 Charge efficiency

Charge efficiency is used to describe the relation between the injected fuel and its energy release at combustion. In an ideal combustion all of the injected fuel is burned with maximum efficiency. This gives a theoretical minimum amount of injected fuel necessary for the corresponding release of energy. The theoretical minimum fuel for a specific release of combustion energy is given as

$$m_{min} = \frac{Q_{comb}}{LHV} \tag{3.20}$$

where Q_{comb} is the energy released and LHV the lower heating value of the fuel.

In a normal engine, the efficiency is unfortunately not optimal where the presence of burned gas, variation of air/fuel ratio and incomplete combustion must be taken into consideration. All of these factors reduce the charge efficiency of the injected fuel which gives a larger fuel amount for the corresponding energy released.

$$m_{tot} = \frac{1}{1 - x_b} \cdot \left(\frac{1}{CoC} \cdot \frac{Q_{comb}}{LHV} \cdot \lambda \right) \tag{3.21}$$

The charge efficiency is given as the ratio between the minimum mass and total mass and ranges between 0 and 1 [Mla02].

$$\eta_{eff} = \frac{m_{min}}{m_{tot}} \tag{3.22}$$

4. Measurement

4.1 Equipment

The engine used in this work is a 3.0l V6 diesel engine with a CRDi system and piezo injectors. Piezoelectrical sensors are installed for pressure measurement, both at the testbench and in test-vehicle. The pressure is measured relative to crank angle. In the testbench engine water-cooled sensors are installed for reducing measurement errors. These measurements are expected to have higher accuracy than given in vehicle. The testbench has the possiblity to e.g. control the load, speed and temperature, and to measure the injected fuel mass and exhaust gas particles. Table 4.1 shows some important characteristics for the used engine.

	Symbol	Unit	Value
Displacement volume	V_{piston}	$[cm^3]$	498
Piston stoke	H	$[mm]$	92
Cylinder bore	d	$[mm]$	83
Connecting rod	l	$[mm]$	168
max. load (3600 $\left[\frac{1}{min}\right]$)	P_{max}	$[kW]$	165
max. torque	M_{max}	$[Nm]$	510

Table 4.1: Characteristics for Mercedes-Benz OM642 diesel engine [DFN+05]

4.2 Pressure measurement

In a four-stroke engine, the pressure curve is commonly divided into two parts, high and low pressure. The high pressure part is the 360°CA where compression and expansion occur and the low pressure part is the 360°CA where intake and exhaust of gas take place. For combustion control most common is to focus on the high pressure part and use the crank position in relation to TDC as reference. This part normally starts and ends with pressure around 1 bar because of the opened valves. Figure 4.1 shows a typical pressure curve for the high pressure part.

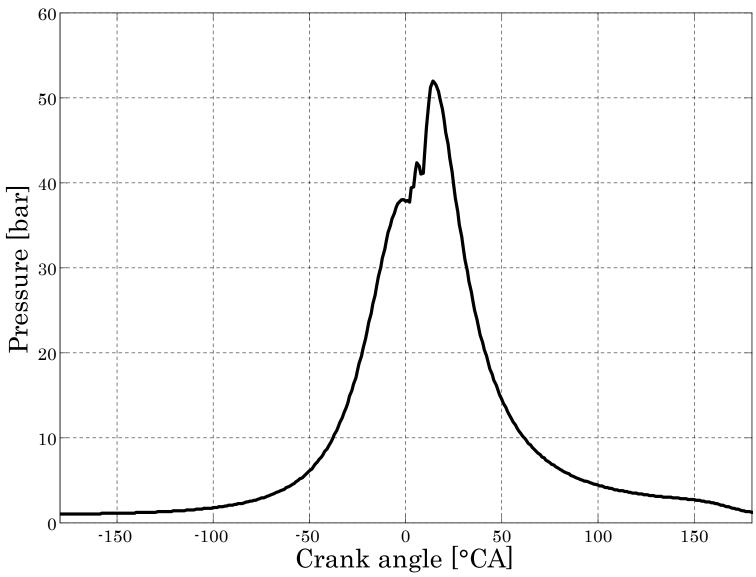

Figure 4.1: Typical pressure curve during high pressure phase

For the first 180°CA before TDC the gas is compressed which causes a pressure rise. At TDC, the change of pressure is normally zero, but can be affected by the pilot injection. In the 180°CA after TDC is where the combustion takes place whereat the change in pressure depends both on the volume change and the combustion.

Some examples of interesting information within the pressure curve characteristics are

- differences between cycles and cylinders
- size and position of max. pressure
- site and position of max. differential pressure
- pressure spikes

4.3 Pressure signal error

Depending on the quality and the sensor type measurement errors differ. The focus has been put on two different errors, pressure offset and crank angle offset, both occurring with the used measurement equipment.

4.3.1 Pressure offset

Pressure offset may occur for all types of pressure sensors that measure the relative pressure. The result is an absolute pressure offset approximately constant for all crank angles, as illustrated in Figure 4.2.

For reducing the effect from this error, an offset correction is implemented. The correction works by using two pressure measurements $p_{O1,meas}$ and $p_{O2,meas}$ within the interval of 100°CA to 65°CA before TDC. With

$$p_{O1} = p_{O2} \left(\frac{V_{O2}}{V_{O1}} \right)^n , \tag{4.1}$$

$$p_{O2} = p_{O1} + \Delta p_{offset,corr} \tag{4.2}$$

and $\Delta p_{offset,corr} = p_{O2,meas} - p_{O1,meas}$, the pressure offset correction is given as

$$p_{offset,Ocorr} = \frac{\Delta p_{offset,corr}}{\left(\frac{V_{O2}}{V_{O1}} \right)^n - 1} - p_{O1,meas}. \tag{4.3}$$

For the diesel engine used in this work the polytropical exponent n can be estimated to $n \approx 1.37$. The offset correction $p_{offset,Ocorr}$ is added to all measured pressures each cycle.

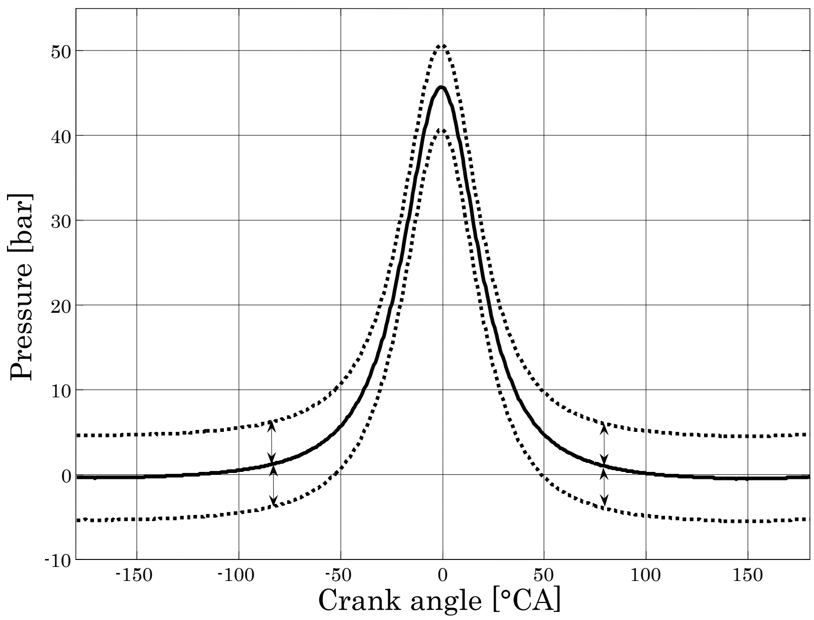

Figure 4.2: Pressure offset

4.3.2 Crank angle offset

A crank angle offset causes an error in the reference position of the pressure measurement, as illustrated in Figure 4.3. This effect will cause errors for all calculations that are referring to a pressure at a specific crank angle position.

Depending on the position where the pressure is measured, the absolute and relative error varies. The size of the absolute error depends on the differential pressure at the same position. High pressure derivative results in a large absolute error and low derivative in a small.

The error relative to pressure will not increase in the same way as the absolute error for increasing pressure derivative, because the pressure is rising parallel to the increasing absolute error.

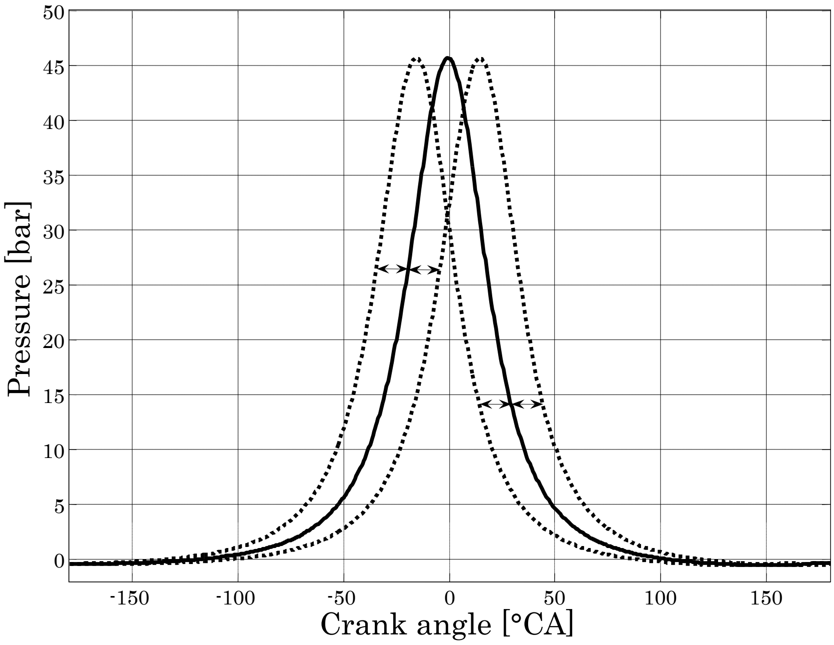

Figure 4.3: Crank angle offset

5. Injector calibration

5.1 Background

Manufacturing piezo injectors requires high precision for correct behavior of the piezo elements. There is a balance here between cost and precision which differs for the specific injector use. To comply with basic requirements each injector's specific properties are measured and later taken into account for when controlling the injection. When implementing the injectors into the engine the ECU is at the same time updated with all the measured injector properties as calibration constants, curves and maps.

During time and use the injector properties change. This leads to incorrect calibration information in the ECU.

5.2 Concept

Between injectors the energizing time differs until the piezo elements have enough charge to open the nozzle for fuel injection. This requires individual energizing time for each injector for equal injection behavior among each other. During time and use this property changes, which makes a re-calibration of these values necessary. The purpose of zero fuel calibration (ZFC) is to find the minimum energizing time necessary to open the nozzle for fuel injection and update the ECU with this information.

For ZFC the engine must be in trailing throttle, which means a state where the engine is driven by the vehicles kinetic energy with no fuel injected into the cylinders. Since ZFC is used to find the energizing time necessary to start

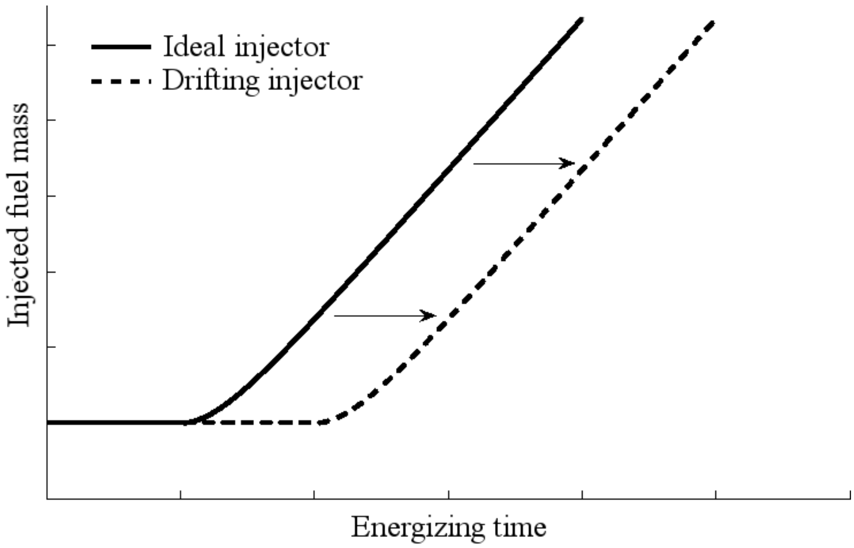

Figure 5.1: Energizing time drift

injecting fuel the injection must be completely controlled by the calibration function. Therefore the engine must work in an overrun state with no influence from the driver.

To find the necessary energizing time for fuel injection, the time step is varied until a specific response is detected. When using pressure sensors for ZFC this specific response can be a pressure rise equivalent to the injected fuel mass.

The energizing time controls the injected fuel mass together with rail-pressure. If the rail-pressure is known and a specific fuel mass can be detected, the equivalent necessary energizing time can be calculated. This calculated "ideal" energizing time can be compared to the time used for injecting the detected fuel mass. The difference between the two energizing times is the injector time drift at that specific rail-pressure. For improving the precision of the calibration, different rail-pressures and fuel mass detection levels are used.

5.2.1 Fuel mass estimation

At ZFC, both energizing time and rail-pressure are know which makes an estimation only necessary for the injected fuel. In-cylinder pressure sensors make it possible to measure the pressure rise caused by the combusted fuel. Measuring in-cylinder pressure can give accurate information concerning cylin-

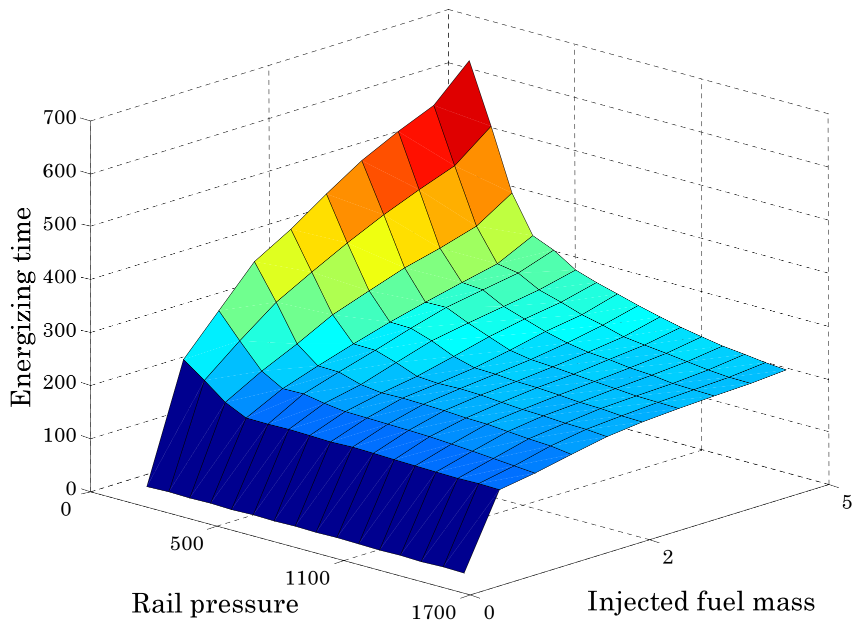

Figure 5.2: Injection map with influencing parameters

der individual combustion without influence from other cylinders nor material strengths. The pressure rise is not always proportional to the injected fuel since the energy losses vary depending on engine conditions e.g. temperature and speed.

The injected fuel mass will be estimated with two different approaches, $IMEP_{HP}$ and Δp_{φ}. Both approaches are designed to correlate with the fuel mass, but the proportional constant for fuel mass estimation is never decided. The proportional constant is not of interest in this project since is cannot be validated. Compensations for losses or other corrections are described in section 5.2.2.

5.2.1.1 Estimation through $IMEP_{HP}$

Fuel estimation through $IMEP$ will only include the high pressure phase, $IMEP_{HP}$. The work output from the injected fuel mass is proportional to the energy released by the combusted fuel minus losses.

$$IMEP_{HP} = \frac{W_{HP}}{V_{piston}} \approx \frac{Q_n}{V_{piston}} \tag{5.1}$$

$$Q_n \approx C \cdot Q_{comb} = C \cdot LHV \cdot m_{fuel} \cdot CoC \tag{5.2}$$

The two most important factors that are affecting the correlation between $IMEP_{HP}$ and m_{fuel} are losses and completeness of combustion. How they affect the relation is described in section 5.2.2.

5.2.1.2 Estimation through Δp_φ

In a dissertation from Mladek (2002) [Mla02] with the aim to estimate the air-flow in a spark ignition engine it has been shown that the charge efficiency can be estimated as a function described by the relation between two pressures, $p(-\varphi)$ and $p(+\varphi)$, at the same volume prior and post combustion and the mean temperature of the gas in the cylinder. This is made possible by estimating the release of energy from the combusted fuel proportional to the pressure difference Δp between the two pressure measurements.

$$\eta_{eff} = \frac{m_{min}}{m_{tot}} = \frac{C \cdot Q_{comb}}{m_{tot}} \approx C \cdot \frac{\Delta p_\varphi}{m_{tot}} \tag{5.3}$$

The measurements are made prior and post combustion at equal volumes at a crank distance $\pm\varphi^o$CA from TDC. In Figure 5.3 the two pressures are shown in a pV-diagram for a cycle with injected fuel.

According to equation (3.19) (section 3.2.5), and the estimation in equation (5.3), the injected fuel mass can be estimated as

$$m_{fuel} = \frac{Q_{comb}}{CoC \cdot LHV} \approx C \cdot \frac{\Delta p_\varphi}{CoC \cdot LHV}. \tag{5.4}$$

Under ideal conditions with no combustion taking place and without losses, e.g. blow-by and heat transfer, the relation between the two pressures hold

$$p(+\varphi) = p(-\varphi). \tag{5.5}$$

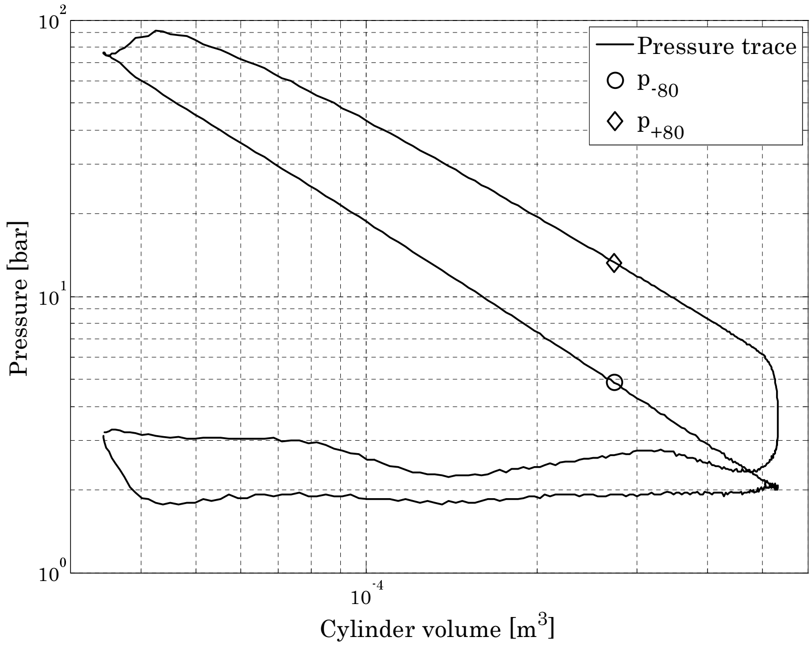

Figure 5.3: pV-diagram with $\varphi = 80°$

In firing operation the difference between the two pressure measurements increases. This pressure difference, Δp_φ, is described as

$$p(+\varphi) = p(-\varphi) + \Delta p_\varphi. \tag{5.6}$$

The pressure before TDC should be measured as close to TDC as possible for reducing the influence from the inlet valve closing, incomplete fuel vaporization and as well as thermal strain to wear off [Mla02]. The relative measured pressure error is also reduced because of the high pressure.

The pressure after TDC should be measured before the exhaust valve opens but late enough for the combustion to be over. In a diesel engine the combustion can be assumed to be over at +80°CA and the exhaust valve opens somewhere between 130-140°CA. According to heat release analysis the combustion is roughly approximated to end between 40-70°CA. The choice of position depends on measurement errors, which are described in section 5.2.3. Throughout this work $\varphi = 80$ is used.

5.2.2 Compensation for losses

By compensating for losses when estimating the injected fuel mass using $IMEP_{HP}$ and Δp_{φ} higher accuracy is given. As described in section 5.2.1.1, fuel estimation through $IMEP_{HP}$ is based on the work given by the injected fuel. This work output is equivalent to the net heat release, Q_n, which is the remaining energy after losses.

Since $IMEP_{HP}$ and Δp_{80} show a clear linear correlation, equivalent corrections are made for both fuel mass estimation approaches, see figure 5.4.

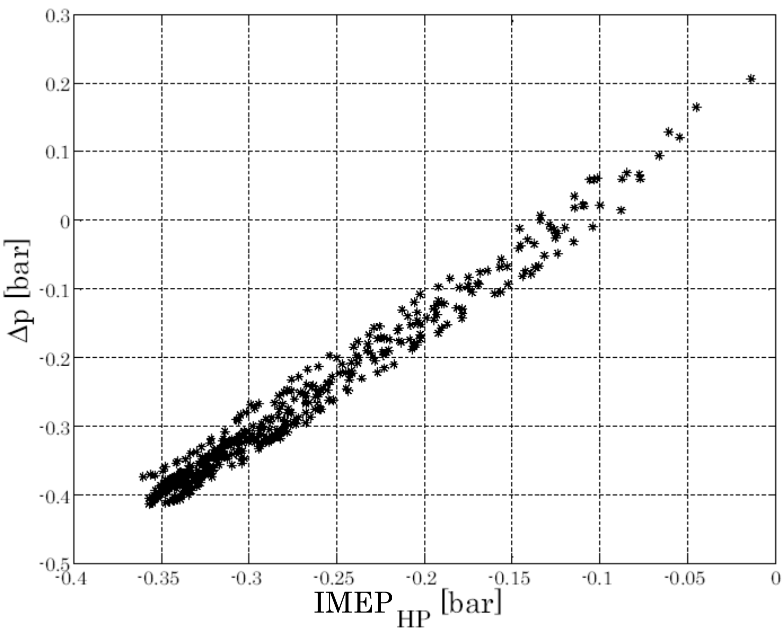

Figure 5.4: Linear relation between Δp_{80} and $IMEP_{HP}$

The deviation is caused by noise, which is affecting Δp_{80} more than $IMEP_{HP}$. Throughout this section only corrections for Δp_{80} are described, but the concept is completely the same for $IMEP_{HP}$.

5.2.2.1 Losses at no fuel injection

For an ideal cylinder with no losses, such as heat transfer and leakage, the pressure is for the same volume equal before and after TDC when no fuel is injected. When including losses, a pressure drop occurs in the expansion part (after TDC) which is mainly influenced by the energy loss caused by heat transfer from the warm compressed gas to the colder cylinder walls. The energy loss causes the pressure after TDC to be less than its equivalent before TDC. The losses from other effects are relatively small e.g. blow-by which causes approximately 1% energy loss [Hey88]. Blow-by is caused by gas flow from crevices between the piston, piston rings and cylinder wall into the crankcase [Hey88].

This pressure drop will affect both $IMEP_{HP}$ and Δp_φ calculations, and will lead to negative estimated fuel mass when no fuel is injected. The correction must compensate for these losses for estimating the fuel mass as zero when no fuel is injected.

$$\Delta p_{\varphi,0corr} = \Delta p_\varphi + p_0 \tag{5.7}$$

Heat transfer has shown to be mainly influenced by speed and load [Hey88]. At ZFC, the load is approximated as constantly low since the engine is driven by the vehicles kinetic energy. The influence of speed per cycle decrease as speed increase. This is caused by the decreasing time for energy transfer for each cycle and less time for the cylinder walls to cool down until the next cycle.

Figure 5.5 shows how Δp_φ depends on the pressure loss occurring relative to engine speed. Also shown is this figure is an offset between each cylinder uninfluenced by speed variations. This might be caused by leakage or pressure sensor difference.

The zero level corrections will be compensated both for speed dependent losses, $p_N(N)$, as well as cylinder individual losses, $p_{0,adapt}$.

$$p_0(N) = p_N(N) + p_{0,adapt} \tag{5.8}$$

The influence from speed shows a linear relation to the pressure loss for all cylinders. The linear relation holds

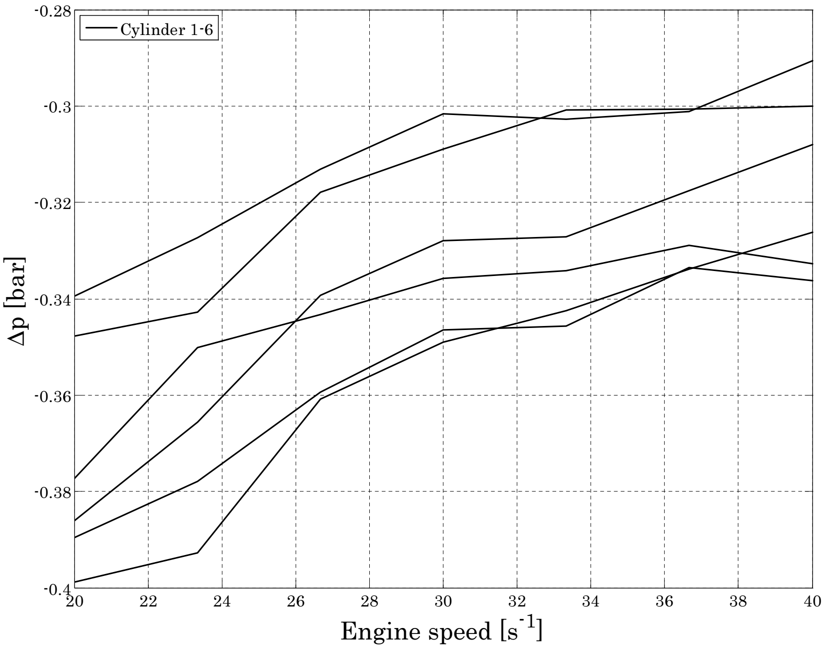

Figure 5.5: Engine speed dependent loss for Δp_{80} at no injected fuel

$$\Delta p_{\varphi}^{-1}(N) = -p_{N}^{-1}(N) = a \cdot \frac{1}{N} + b \tag{5.9}$$

For deciding the linear coefficients a and b a linear fitting was made using MATLAB® function $polyfit$. The data used for finding the linear coefficients is a mean-value for all 6 cylinders over a speed range of 1200-2400 rpm with 200 rpm steps. These speeds are commonly occurring at trailing throttle. The resolved coefficients are only valid for the engine used in this work and will probably differ depending on e.g. cylinder volume and compression ratio.

	$\Delta p_{80,0corr}$	$IMEP_{HP,0corr}$
a	20.30	29.13
b	-3.69	-3.76

Table 5.1: Coefficients for linear fit

The correction from the linear fitting includes compensation for the complete offset from zero level. Since it is made for a mean-value of all 6 cylinders and

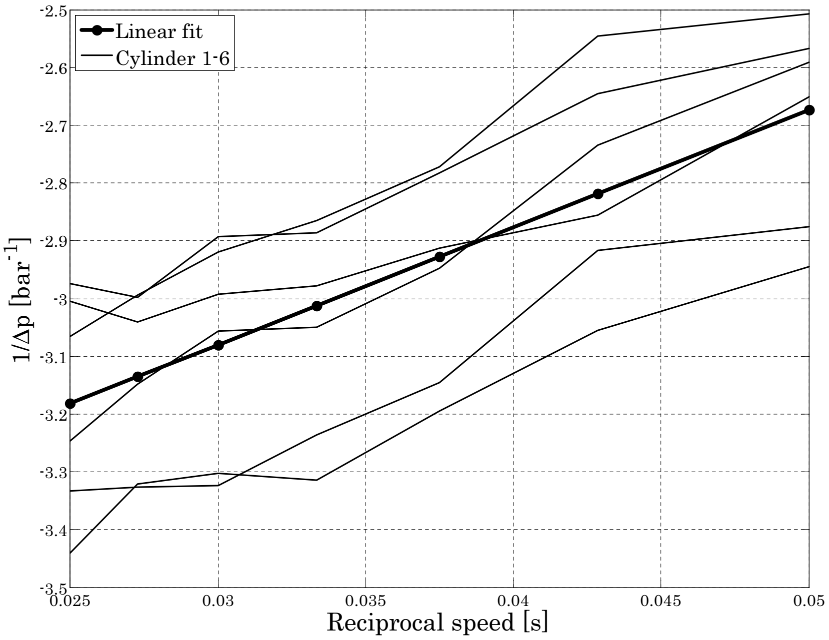

Figure 5.6: Linear fit for engine speed dependent loss

each cylinder differs in offset some will have too much compensation and some too less.

It is not possible to correct the cylinder individual offset from test bench data since it differs for each engine. Instead the correction is decided online during normal engine use. For this, a function is designed for adapting the correction $p_{0,adapt}$, which reduces the cylinder individual zero level offset. The function is designed to continuously correct the offset during engine use.

The adaptation is made by calculating the deviation from zero level as

$$p_{0,adapt} = -\Delta p_{\varphi} - p_N(N). \tag{5.10}$$

When fuel injection takes place the last adaptation value is used for the zero level correction.

To reduce the risk of adapting noise or other disturbances, a IIR low-pass filter is implemented. The filter is designed as a discrete Butterworth low-pass filter of second order.

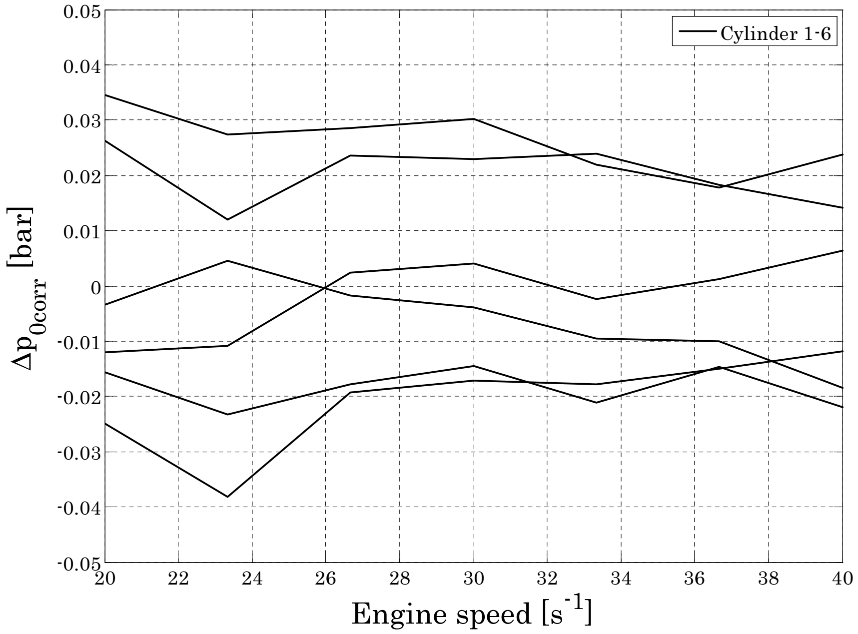

Figure 5.7: Zero level engine speed dependent correction

$$H[z] = \frac{a_0 + a_1 \cdot z^{-1} + a_2 \cdot z^{-2}}{1 + b_1 \cdot z^{-1} + b_2 \cdot z^{-2}} \tag{5.11}$$

The filter is created with MATLAB® function *butter* and is designed with main focus on disturbance reduction. The sampling time is based on the engine speed where an engine speed of 2000 rpm corresponds to a sampling time of $T_s = 60/1000s$. A cut-off frequency of $f_c = 0.05Hz$ is used to get the described behavior. The filter has a rise time of approximately 60 seconds which is too slow for production use. A slow respone is chosen for minimizing the risk of adapting disturbances and to show the concept.

The complete concept of zero level compensation is shown in Figure 5.8.

The influence from engine temperature on the zero level correction for $IMEP_{HP,0corr}$ and $\Delta p_{\varphi,0corr}$ has been investigated. Since the heat transfer depends on the cylinder wall temperature, decreasing engine temperature could result in increasing pressure losses.

For analyzing the influence from engine temperature, three different tempera-

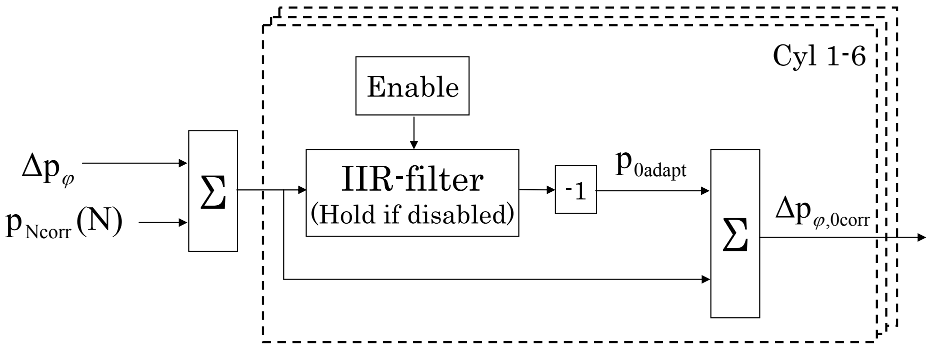

Figure 5.8: Concept for zero level adaptation

tures, 30, 60 and 90°C, are used. For 90°C the measurements are made both after low engine load and directly after high load for which the cylinder walls are expected to have a higher temperature.

These measured temperatures relate to the cooling fluid which may differ from the cylinder wall temperature. The gas entering the cylinder trough the inlet valve is pre-cooled to a constant temperature and should therefore not be effected by the change of engine temperature.

The result from analyzing the losses at different temperatures is that the influence is very small comparing to other effects (e.g. measurement errors and cylinder individual losses), if an influence exists at all.

The only difference between the analyzed temperatures is an increasing zero level offset for one cylinder at 30°C. What this is caused by is unknown but since it is only occurring for one cylinder it is assumed to not be caused by the low temperature. If excluding this cylinder when mean valuing the losses for all cylinders the result shows an equivalent behaviour as for the other temperatures.

A limitation on the lowest engine temperature for ZFC is set to a value of 60°C, which reduces the risk of temperature influences. Normally 3-5 minutes is needed for reaching these temperatures but may vary depending on e.g.

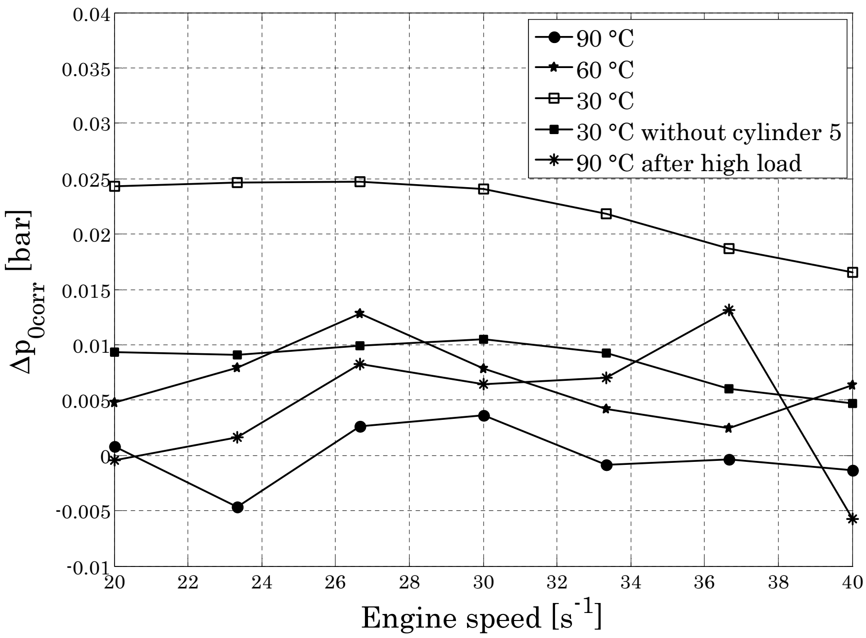

Figure 5.9: Temperature influence on zero level engine speed correction

climate and driving conditions.

5.2.2.2 Losses at fuel injection

During engine operation with fuel injection the relation between the injected fuel mass and $\Delta p_{\varphi,0corr}$ can be influenced by more than constant cylinder offsets and engine speed. Here is also the influence from rail-pressure and injected fuel mass investigated. The different influences are analyzed with data from the test bench with variation in injected fuel mass, rail-pressure and engine speed.

Figure 5.10 shows the influence from the injected fuel mass at different rail-pressures. What can be seen is an influence from both fuel mass and rail-pressure, which may be caused by a change in CoC.

The rail-pressure is in Figure 5.11 shown to have a linear correlation to the relation between the injected fuel mass and $\Delta p_{80,0corr}$.

The influence from engine speed is investigated over a span of different rail-pressures. Figure 5.12 shows that there is no clear influence from engine speed at the investigated rail-pressures.

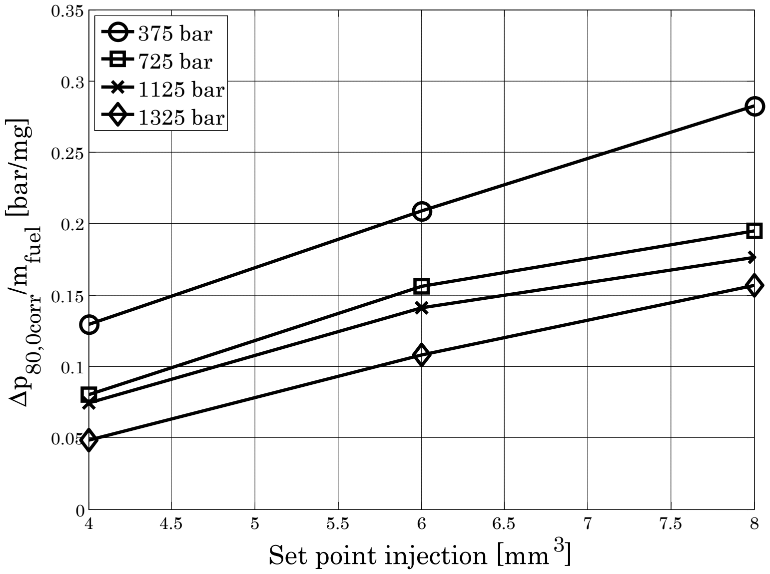

Figure 5.10: Influence from the injected fuel mass

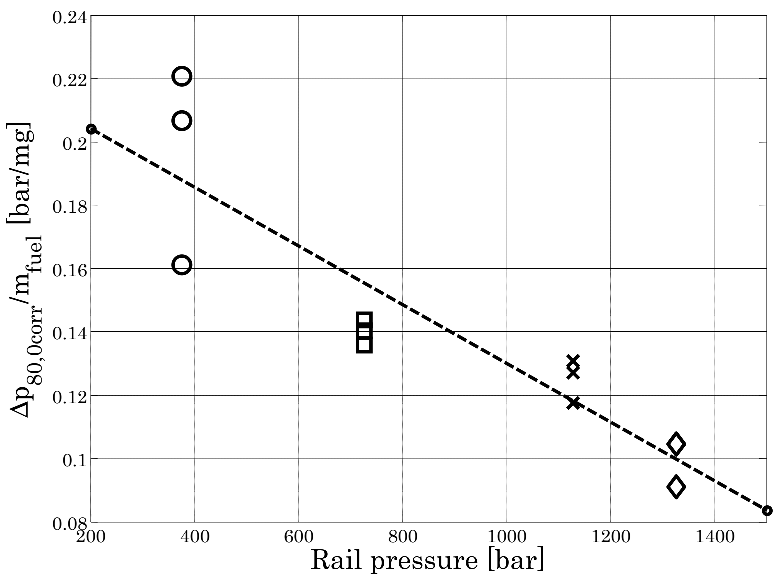

Figure 5.11: Rail-pressure influence

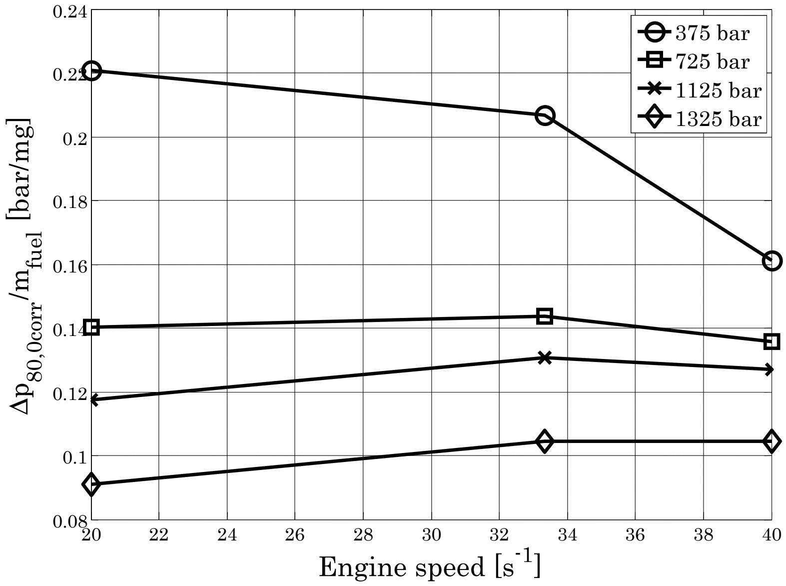

Figure 5.12: Engine speed influence at different rail-pressures

There is a limitation in the measurement accuracy of the test bench for the small fuel masses that are used. Therefore, no corrections are implemented for rail-pressure nor fuel mass variance. Since the rail-pressure is kept constant during injector calibration, the influence on the result is assumed to be relatively small.

5.2.3 Sensitiveness

There is a difference between $IMEP_{HP}$ and Δp_{φ} in how much they are affected by measurement errors. This is mainly caused by the fact that for $IMEP_{HP}$ a pressure zero level is used as reference while for Δp_{φ} a pressure difference is used which gives a variable reference at $p(-\varphi)$.

For pressure offset only $IMEP_{HP}$ is affected due to the use of a zero reference for pressure measurement. This pressure reference is also used for Δp_{φ} but since the reference is constant for all pressures and Δp_{φ} is based on a difference between two measurements it has no influence.

$$\begin{aligned} IMEP &= C \cdot \int (p + p_{offset}) \cdot dV \\ &= C \cdot \int p \cdot dV + C \cdot p_{offset} \cdot V \end{aligned} \tag{5.12}$$

$$\begin{aligned} \Delta p_{\varphi} &= C \cdot ((p(+\varphi) + p_{offset}) - (p(-\varphi) + p_{offset})) \\ &= C \cdot (p(+\varphi) - p(-\varphi)) \end{aligned} \tag{5.13}$$

After correction this error should not have much effect on $IMEP$ calculation but is still counted as a disadvantage comparing Δp_{φ}.

At crank angle offset both $IMEP_{HP}$ and Δp_{φ} are affected. For complete $IMEP_{HP}$ calculation a pressure measurement adjustment for reducing the offset influence is not possible. This is because $IMEP_{HP}$ is calculated over a complete range of pressures which cannot be changed.

For Δp_{φ}, only two pressures are measured which makes is possible to adjust the position of these measurements to reduce the error caused by the crank angle offset. As described in Chapter 4.3.2 the influence from offset is greater at parts of the pressure curve with larger gradient. The choice of measurement positions for maximal reduction of the influence from crank angle offset would be as far away from TDC as possible. For Δp_{φ} this would mean that the pressure after TDC should be measured at a position just before the exhaust valve is opened. At fuel masses used for ZFC, only a very small improvement on the reduction of the influence from the crank angle offset is given when using measurement positions after 80°CA.

Since $IMEP_{HP}$ is integrated over a range of pressures, the influence from high frequency noise is drastically reduced. For Δp_{φ} the influence is far greater because the use of only two measurements.

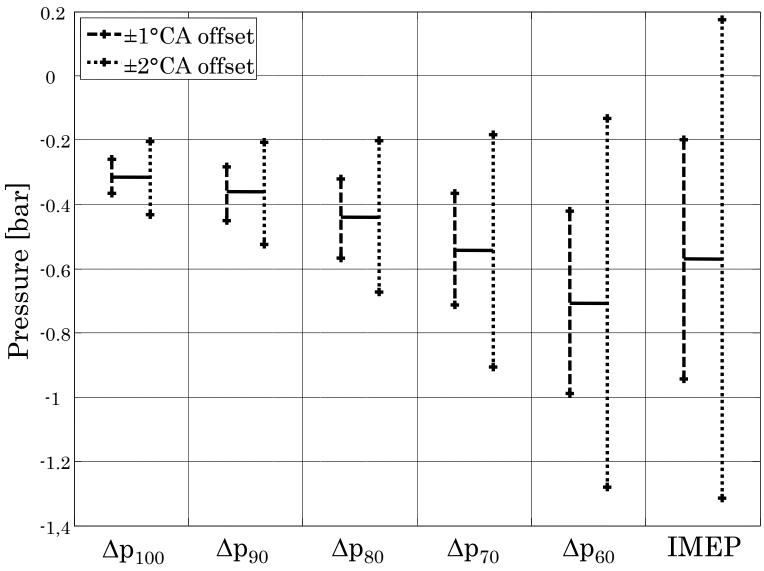

Figure 5.13: Absolute error from crank angle offset

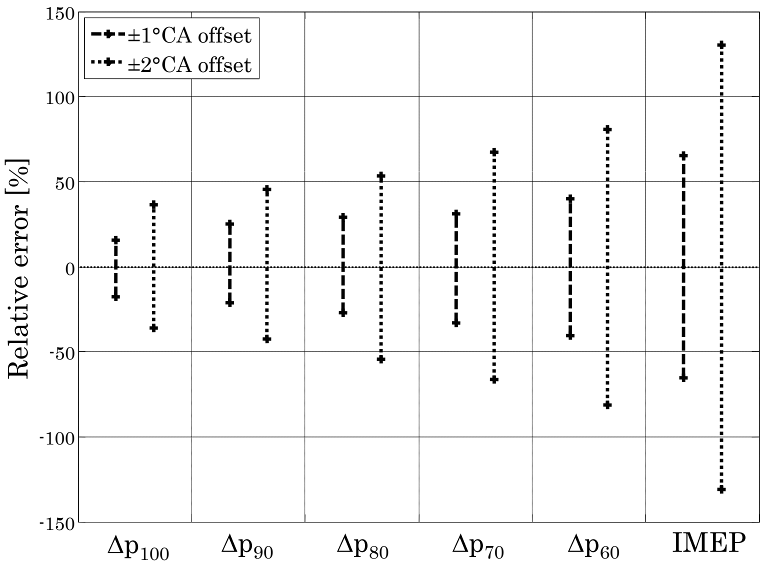

Figure 5.14: Relative error from crank angle offset

6. Implementation

For implementation of the ZFC function into vehicle for testing, MATLAB/Simulink is used. The Simulink model is created as a subsystem into an existing model for combustion control. This gives a possibility of using preprocessed signals from other functions or from the ECU as input signals.

When creating the model an important issue is to get a cylinder individual functionality. Each used input contains the information from all 6 cylinders with a periodic time-based structure ordered in cylinder appearance. This means that for each 120° of crank rotation, the cylinder individual information in the signal changes from one cylinder to another. Information of which cylinder that is currently in use is available and is used for triggering cylinder individual functions.

The Simulink model is not created for a complete zero fuel calibration with an energizing time correction. Instead the model is created with the aim of using it for indicating the possibility of using in-cylinder pressure sensors for ZFC. The model will be run parallel to a production line ZFC function not based on in-cylinder pressure measurement. This parallel running ZFC function will work as a reference when evaluating the created in-cylinder pressure based algorithms.

6.1 Functionality

For beginning ZFC or zero level adaptation ($p_{0,adapt}$) the engine must operate in a trailing throttle state, so called "overrun". When an overrun state is detected, a waiting loop starts which puts all functions on hold for x number of cycles. Just after entering an overrun state the fuel estimation may deviate from a steady zero level but only for some cycles, for what a waiting functionality is added. Directly after these x cycles the zero level adaptation starts if all conditions are fulfilled. The conditions for adaptation are based on engine temperature, speed and if ZFC occur or not. Speed and temperature limitations are based on conditions for which the ZFC function is calibrated and are set for adaptation to work during normal engine use.

The calibration limitation is necessary because zero level adaptation should only occur if no fuel is injected into any cylinder, and during ZFC fuel is injected into one cylinder. When injecting fuel into one cylinder, the other cylinders can be affected by the small change in speed and therefore the adaptations of all cylinders are stopped. All cylinders are adapted parallel by enabling each individual adaptation according to cylinder appearance. The adaptation is reseted everytime the engine is restarted.

ZFC only occur when specific engine conditions are reached. These limitations are balanced between high accuracy and the possibility for ZFC to occur. If the conditions are set too strict ZFC might in some environments never occur which can be worse than calibrating with less accuracy. Examples on conditions are specific engine speed, engine temperature, fuel temperature and battery potential. All conditions except the engine speed will be set according to the values used in the parallel running ZFC.

When all conditions are fulfilled, a triggering signal change its value from non-active to active ZFC state. This signal is used both for activating ZFC and disabling $p_{0,adapt}$ adaptation. When a change in this signal is detected, a loop starts mean-valuing the estimated fuel mass. The loop continues mean-valuing until a change in energizing time is detected which should occur after a specified number of cycles which must appear without disruption. When a change is detected, a reset is made for all cylinders and a new mean-value calculation starts. If the energizing time is held for x cycles, $x - 1$ values are mean-valued since the loop reset takes one cycle. This one cycle reset also reduces the influence from the previous state.

When x cycles have passed by without disruption and the energizing time is changed, the last value for the previous state is stored. This value is overwritten next time the energizing time is changed.

The mean-value calculation runs for all cylinders during ZFC.

7. Experimental results

In short, the production ZFC works by injecting a fuel mass at a constant rail-pressure into one cylinder and see if it differs from a wanted fuel mass. If more fuel is detected than what is wanted, than the energizing time is reduced until the wanted fuel mass is reached. If less is detected, than the energizing time is increased.

The result is based on a comparison between the production ZFC, which also is based on fuel mass estimation, and fuel mass estimation through $\Delta p_{80,0corr}$ and $IMEP_{0corr}$. Four different rail-pressures have been used, with each corresponding to a detection of a specific fuel mass.

The fuel mass estimations are not proportionally calibrated against the correct injected fuel mass. This does not influence the result, since the only thing of interest is to see the correlation with the production function. Also of interest is the correlation between the change in energizing time and the change of the estimated fuel mass.

Figures 7.1-7.4 show the correlation between the three fuel mass estimations during ZFC. Both $\Delta p_{80,0corr}$ and $IMEP_{0corr}$ are shown to correlate well with the production ZFC for all rail-pressures.

Figure 7.5 shows ZFC with all cylinders included, but where only one cylinder is calibrated. Since no fuel is injected into the cylinders that are not being calibrated, the fuel mass should be estimated as zero.

In Figures 7.6-7.8 it is shown that the two approaches based on pressure measurement correlate relatively well with each other. The current ZFC and $\Delta p_{80,0corr}$ have the largest deviation, which might be caused by the sensitiveness to noise.

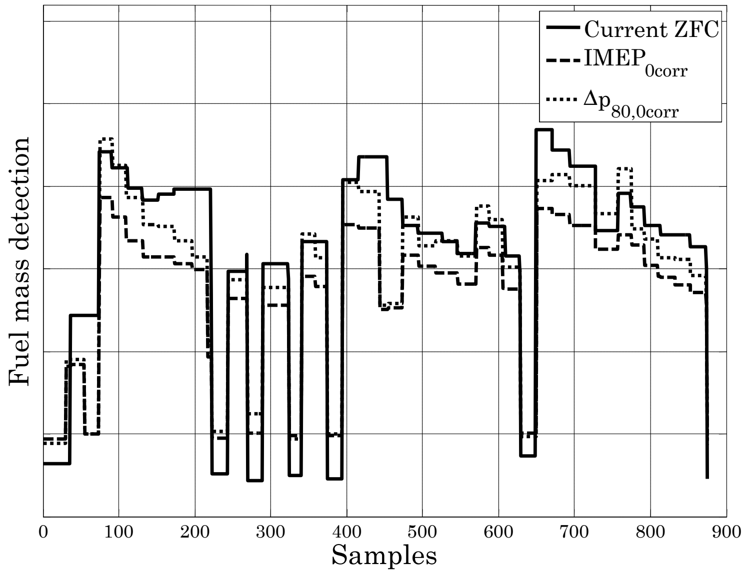

Figure 7.1: Calibration at 250 bar rail-pressure

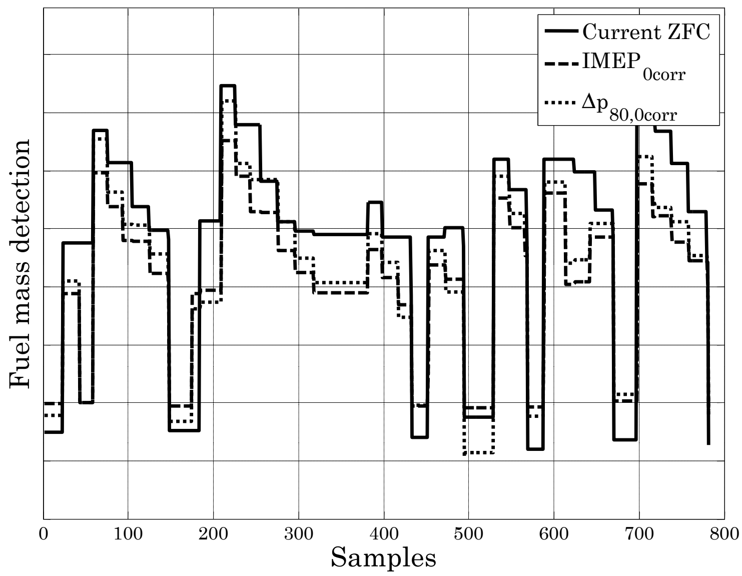

Figure 7.2: Calibration at 800 bar rail-pressure

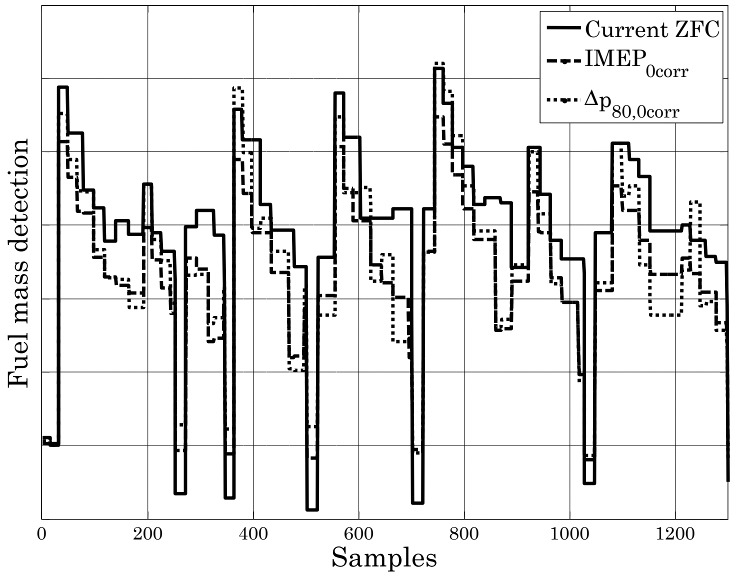

Figure 7.3: Calibration at 1200 bar rail-pressure

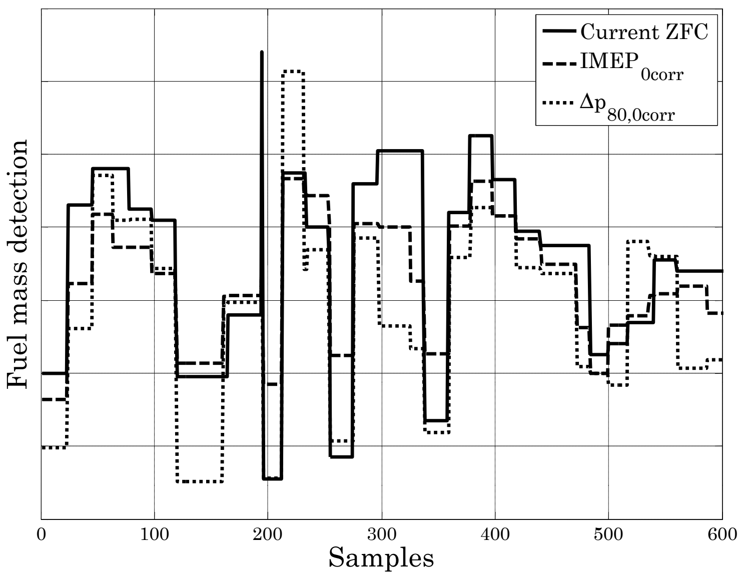

Figure 7.4: Calibration at 1400 bar rail-pressure

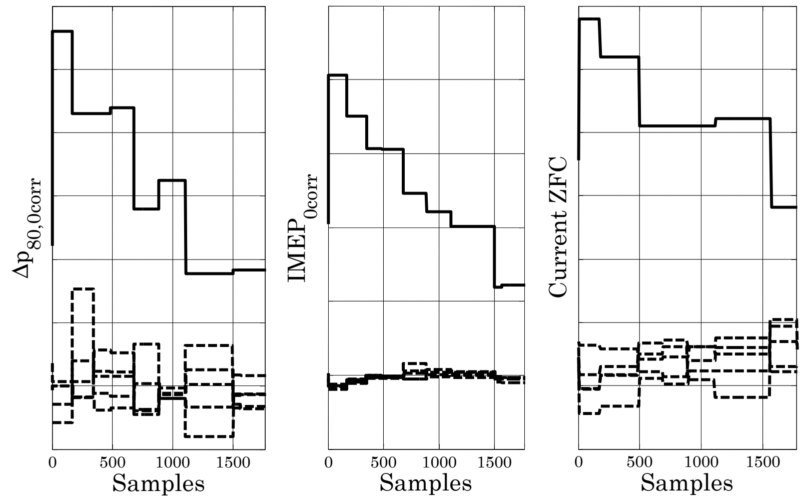

Figure 7.5: Calibration for one cylinder (solid line), with the other cylinders included (dashed lines)

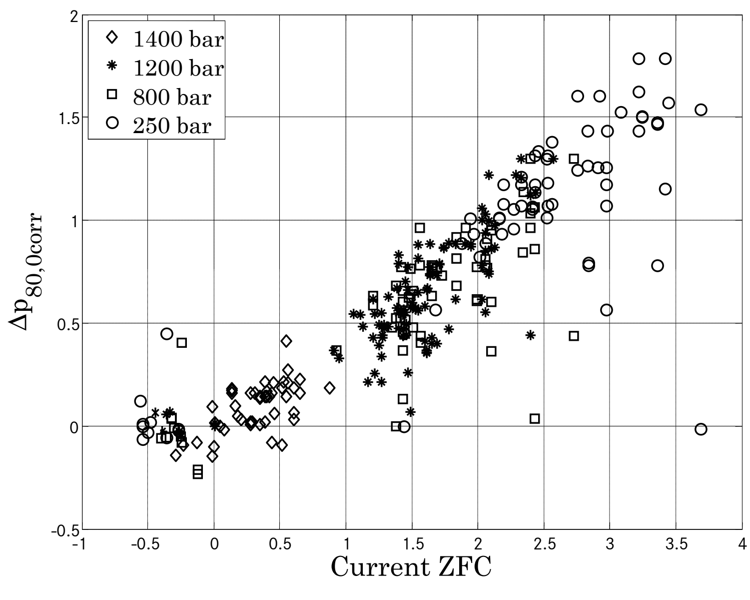

Figure 7.6: Relation between $\Delta p_{80,0corr}$ and current ZFC during calibration

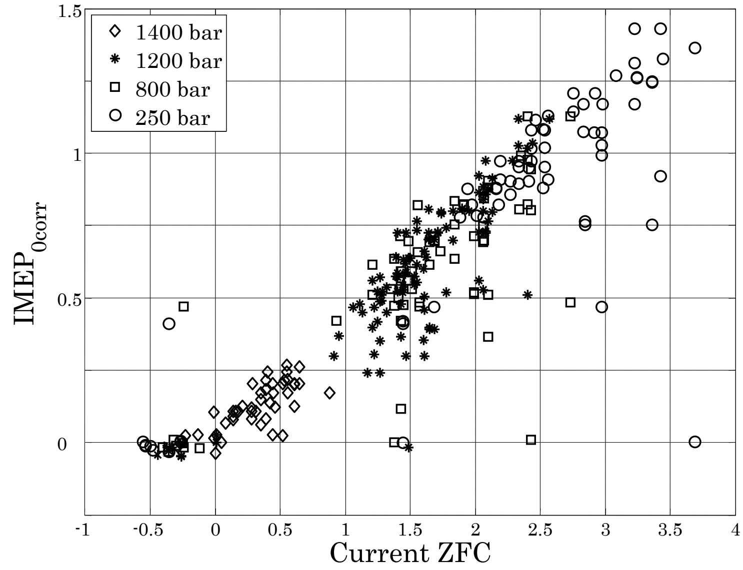

Figure 7.7: Relation between $IMEP_{0corr}$ and current ZFC during calibration

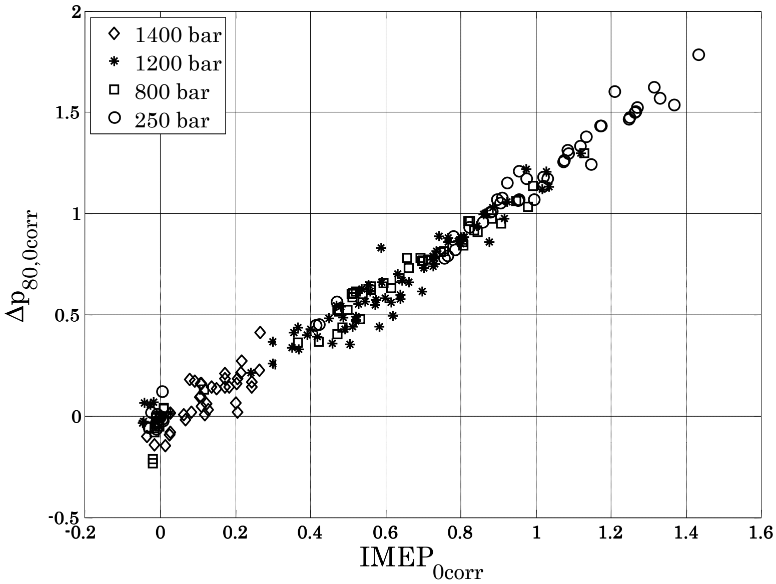

Figure 7.8: Relation between $\Delta p_{80,0corr}$ and $IMEP_{0corr}$ during calibration

The necessary accuracy for calibration is unknown. Therefore a wide range of plots are presented which may give a foundation for a more subjective view. Figure 7.1-7.4 presenting complete calibration for all cylinders are showing a working concept at different fuel mass injections and rail-pressures. One thing of interest in these plots is the sudden signal drop that sometimes occur for the two pressure based methods, but not for the current ZFC. This is especially visible in Figure 7.1 at approximately 450 samples. If this is caused by sensor errors or just the fact that the concepts differ in their design of detecting the injected fuel mass is unknown.

According to Figure 7.5, $IMEP_{0corr}$ gives the most stable result for the cylinders that are not calibrated. This is an indication of less sensitiveness to noise. It also gives the best response to the stepping energizing time which is reduced with constant steps. Both $\Delta p_{80,0corr}$ and the current ZFC show larger variations for the cylinders that are not being calibrated and also more variation from the stepping change in energizing time. This can be explained by the pressure integration in $IMEP_{HP}$, which works as a low-pass filter.

If focusing on Figure 7.6-7.7 it is visible that $\Delta p_{80,0corr}$ shows larger deviations than $IMEP_{0corr}$ at small fuel mass injections. This is assumed to be caused by the noise sensitiveness, which is also visible in Figure 7.5. If assumed that, according to Figure 7.5, $IMEP_{0corr}$ is the most accurate method of all three, than $\Delta p_{80,0corr}$ should be compared according to Figure 7.8. Since it seems that $IMEP_{0corr}$ is most accurate and least noise sensitive, this might be an indication on that $\Delta p_{80,0corr}$ also is more accurate than the current ZFC.

8. Conclusions and future work

8.1 Conclusions

It is hard to draw any direct conclusions from the result since the accuracy necessary for ZFC is unknown. A qualified guess would be that $IMEP_{HP}$ is the best method of all three. This can be motivated by a clear correlation to the energizing time change and the zero level stability for the cylinders without fuel injection. By showing better properties that the current ZFC, less cycles of mean valuing could be a possibility with faster calibration as a result.

One advantage for Δp_{φ} is the possibility of choosing measurement positions for desired response. When it comes to sensitiveness to disturbance $\Delta p_{80,0corr}$ shows more influence to noise. On the other hand $IMEP_{HP}$ is more sensitive to crank angle and pressure offset.

At the highest rail-pressure the detected pressure rise is smaller than the compensated pressure drop at no fuel injection. A conclusion that can be drawn from this is that some kind of pressure correction is definitely necessary. The filter used for zero level adaptation works for evaluation, but will require faster response for production use.

The concept of estimating the injected fuel mass through in-cylinder pressure measurement and use that information for injector calibration is definitely possible. A main conclusion is that the concept works but will require lots of fine tuning and evaluation before a complete product.

8.2 Future work

Interesting for the future would be to investigate the possibility of improving fuel mass estimation through Δp_φ by using more than two pressure measurements. One possibility would be to use a mean value of pressures within a region of $\pm x$°CA from $\pm\varphi$°CA. By using more measurements Δp_φ should become less sensitive to noise.

What could also be of interest is to investigate what is causing the losses to increase at decreasing fuel mass and increasing rail-pressure. A guess would be that the combustion is far from complete at the small fuel masses used for ZFC.

Bibliography

[AFFS00] Dennis N. Assanis, Zoran S. Filipi, Scott B. Fiveland, and Michalis Syrimis. A methodology for cycle-by-cycle transient heat release analysis in a turbocharged direct injection diesel engine. Technical Report 2000-01-1185, SAE Technical Paper, 2000.

[BBF01] Michael Bargende, Christine Burkhardt, and Alfred Frommelt. Besonderheiten der thermodynamischen analyse von deottomotoren. *MTZ Motortechnische Zeitschrift*, 62:56–68, 2001.

[BP99] Michael F. J. Brunt and Kieron C. Platts. Calculation of heat release in direct injection diesel engines. Technical Report 1999-01-0187, SAE Technical Paper, 1999.

[BRE98] Michael F. J. Bruntand, Harjit Rai, and Andrew L. Emtage. The calculation of heat release energy from engine cylinder pressure data. Technical Report 981052, SAE Technical Paper, 1998.

[Cre07] J.J.F.G Cremers. Beginnings for cylinder pressure based control. Master's thesis, Eindhoven University of Technology, 2007.

[DFN+05] Gerhard Doll, Hans Fausten, Roland Noell, Joachim Schommers, Christoph Spengel, and Peter Werner. Der neue v6-dieselmotor von mercedes-benz. *MTZ Motortechnische Zeitschrift*, 09:624–635, 2005.

[Hey88] John B. Heywood. *Internal Combustion Engine Fundamentals*. McGraw-Hill, 1988.

[HRB+05] Haris Hamedovi, Franz Raichle, Joerg Breuninger, Wolfgang Fishcer, Werner Dieterle, Martin Klenk, and Johann F. Boehme.

Imep-estimation and in-cylinder pressure reconstruction for multi-cylinder si engine by combined processing of engine speed and one cylinder pressure. Technical Report 2005-01-0053, SAE Technical Paper, 2005.

[Kle04] Marcus Klein. *A Specific Heat Ratio Model and Compression Ratio Estimation*. PhD thesis, Linkping University, 2004.

[Mla02] Michael Mladek. *Cylinder Pressure for Control Purposes of Spark Ignition Engines*. PhD thesis, ETH Zrich, 2002.

[MSS05] Gunter Merker, Christian Schwarz, and Gunnar Stiesch. *Simulating Combustion*. Springer Berlin Heidelberg, 2005.

[MT07] Klaus Mollenhauer and Helmut Tschoeke. *Handbuch Dieselmotoren*. Springer Berlin Heidelberg, 3rd edition edition, 2007.

[Mul03] Norbert Muller. *Adaptive Motorregelung beim Ottomotor unter Verwendung von Brennraumdrucksensoren*. PhD thesis, TU Darmstadt, 2003.

[PKS02] Rudolf Pischinger, Manfred Klell, and Theodor Sams. *Thermodynamik der Verbrennungskraftmaschine*. Springer, 2nd edition edition, 2002.