Development of $2^2$ - an opposed piston HCCI engine

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Sammanfattning

Miljövänliga transporter är ett av de största samtalsämnen i vårt samhälle och i industrin idag. Shell organiserar en årlig tävling som heter Shell Eco Marathon. Målet är att bygga bilen som förbrukar minst bränsle under en tävling. KTH deltog för första gången 2006.


Hittills har principen verifierats, men de flesta planerade testerna återstår. Den indikerade verkningsgraden är 32,5% vid 6500 varv per minut och det indikerade medeltrycket är 4 bar. Projektet kommer drivas vidare av andra studenter och därför är fokus i denna rapport att visa motorns princip och kringsystem snarare än körresultat.
Abstract

Environment friendly transports are one of the main topics in the industry and society today. Shell organizes an annual university competition called Shell Eco Marathon. The goal is to build the car that consumes the least fuel during a race. KTH participated for the first time in 2006.

The engine for this competition, DoubleTwo, has been developed within the school. The principle is a two stroke counter piston HCCI engine with variable compression ratio. HCCI stands for homogenous charge compression ignition. The combustion is quick and simultaneous which leaves a high efficiency and naturally low emissions of NOx and soot. The compression ignition timing is unstable which is the reason for using variable compression ratio. The mechanical system for changing compression ratio is now patent protected.

So far the principle has been verified, but lots of tests remain. The indicated efficiency is 32,5 % at a 6500 rpm and $I_{\text{MEP}}$ (Indicated mean effective pressure) is 4 bar.

The project will be continued by other students and therefore this report is focusing on the different systems around the engine and the function of the engine rather than engine tests.
1 Introduction

In 2006 a faculty team at KTH decided to participate in the annual competition Shell eco-Marathon. The goal is to build a vehicle that will consume as little fuel as possible as it is driven a certain distance around a racing track.

![Figure 1. Spiros competing in 2007](image)

The plan was to use research experience and construct a small HCCI (Homogeneous Charge Compression Ignition.) engine. The result is a two stroke opposed piston flexi fuel HCCI engine with a unique patented variable compression ratio. Most engine manufactures agrees on that HCCI is superior when considering efficiency, besides that the cool combustion is free from NOx emissions. Another advantage is that an opposed piston engine naturally eliminates all vibrations from the pistons and the crank shafts.

The engine originally constructed for Shell Eco Marathon might be usable in other applications than Eco races. Suitable applications are e.g. outboard motor, energy plant and engine source in a hybrid car. The company owning the patent, HCCI Technology AB, has already started looking for industrial partners.

This Master Thesis report summarizes a time span of 18 months development. The focus is on the present engine with a short introduction of the former prototype. The main test plan of the DoubleTwo has still not been carried out. This work is planed to be done in the autumn of 2008.
The first HCCI engine was based on two modified Husqvarna 326L E-techII engines with a cylinder volume of 24.5 cm³ each. The cylinder heads were cut away and the outside of the cylinders was threaded. On each cylinder a ring was mounted. The two cylinder rings were then bolted together with a brass plate in between (See Figure 2). The brass plate contained liquid cooling and holes for spark plug and in-cylinder pressure sensor.

At the first engine setup the phase adjustment mechanism was separately mounted from the engine with only the piston of the linear actuator acting on the axis of one of the middle gear wheels. (See Figure 3). After measurements and calculations of the forces, an integrated phase mechanism was designed [2], see Figure 4. More about the Mechanism can be read in Master of Science thesis MMK 2007:56 MFM 107 by Tobias Granander.
The initially plan was to use one of the two crankcases as scavenging pump. The scavenging didn’t succeed to clear the cylinder from burned gases which resulted in combustions every second revolution as best. After these results booth crankcases were used as scavenging pump via a connecting hose. At this setting the engine run stable with combustion on each revolution for the very first time. Still the net torque was zero. Later the engine was scavenged with compressed air from the central pressure system in the lab. This time the engine gave a net torque of 3 Nm at 3000 rpm, but it required a scavenge pressure of 0.55 bar.

The engine was supposed to start on compression only. (No sparkplug) The consequence of this was that when the engine once started, the compression ratio needed to be reduced from 21 to about 12. Practically this meant a phase difference of 35 degrees, which was acceptable neither for the gearwheels carrying the heavy momentum pulsations, nor for the efficiency.

Because of the poor scavenging the cylinder based on two modified loop-scavenged cylinders was insufficient. Besides that, there were regularly leakages and mechanical breakdowns. However the engine performed a successful result and run as a HCCI engine. Based on that result, the decision to design a custom made cylinder was taken.
3 Doubletwo generation 2

3.1 Mechanical systems
The engine with all surrounding systems is shown in Figure 5. The custom made cylinder and crank case with belonging crank shaft and piston is shown in Figure 6. Below is a engine specification table showing some general data.

Table 1. Engine specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Swept volume</td>
<td>49 cm³</td>
</tr>
<tr>
<td>Trapped volume</td>
<td>~38 cm³</td>
</tr>
<tr>
<td>Stroke/bore</td>
<td>2x27/34</td>
</tr>
<tr>
<td>Aluminum cylinder</td>
<td>With nicasil coated cylinder walls.</td>
</tr>
<tr>
<td>Piston rings</td>
<td>Two compression rings on each piston with 1,1 mm thickness</td>
</tr>
<tr>
<td>Fuel</td>
<td>Gasoline 95 (few tests on DME, and LPG)</td>
</tr>
<tr>
<td>Speed</td>
<td>9500 rpm</td>
</tr>
<tr>
<td>Power</td>
<td>900 W electric DC (6500 rpm)</td>
</tr>
<tr>
<td>Inlet ports</td>
<td>5</td>
</tr>
<tr>
<td>Inlet port area</td>
<td>2,50 cm²</td>
</tr>
<tr>
<td>Exhaust ports</td>
<td>4</td>
</tr>
<tr>
<td>Exhaust port area</td>
<td>2,88 cm²</td>
</tr>
</tbody>
</table>

Figure 5. 2^2 -gen.2 Full system
3.1.1 Compression ratio control system

If the start of combustion in a HCCI engine takes place too early the losses because of heat to cylinder wall and piston ring blow by will increase and there is also a risk for engine damage. On the other hand, a late start of combustion might lead to an incomplete combustion. There are many different methods of controlling the start of combustion in HCCI engines e.g. exhaust gas circulation or variable cam shaft timing. This engine has a simple, unique, patented solution for adjusting the compression ratio continuously. Because of the big range in compression ratio, the engine is not only capable to run as a HCCI, but also as a flexi fuel HCCI.

The compression ratio control system is based on four connected gear wheels and a linear actuator (see Figure 7). One gear wheel is attached to each crank shaft and the two remaining are positioned in between transferring the momentum between the crank shafts. As the linear actuator is in its innermost position the crankshafts rotate in unison and the phase difference is zero. As the linear engine extends its piston, the exhaust crank shaft is forced to rotate ahead of the inlet crank shaft which leads to a phase difference of up to 33 degrees. The resulting compression ratio for one choice of max compression ratio is shown in Figure 8. The compression ratio is defined as:

\[ CR = \frac{V_{\text{min}}}{V_{\text{closed}}} \]

Where \( V_{\text{min}} \) is the minimum cylinder volume and \( V_{\text{closed}} \) is the enclosed cylinder volume just as all the ports have closed.
Except from the possibility to adjust the compression ratio continuously there is a need to change the maximum compression ratio settings to avoid unnecessary big phase angles when testing different fuels. This is solved by combining pistons with different heights. When original pistons are mounted on both inlet and exhaust crank shafts, the compression ratio will be 21. If the processed pistons with a reduced height of 1 mm each are mounted, the compression ratio will decrease to 10,6. In addition to the two settings mentioned above, there are 23 piston combination settings (See Table 2).

Table 2 Maximum compression ratio at different piston height combinations.

<table>
<thead>
<tr>
<th>Reduced piston height [mm]</th>
<th>0,00</th>
<th>0,25</th>
<th>0,50</th>
<th>0,75</th>
<th>1,00</th>
</tr>
</thead>
<tbody>
<tr>
<td>0,00</td>
<td>21,0</td>
<td>18,7</td>
<td>16,9</td>
<td>15,4</td>
<td>14,1</td>
</tr>
<tr>
<td>0,25</td>
<td>18,7</td>
<td><strong>16,9</strong></td>
<td>15,4</td>
<td>14,1</td>
<td>13,0</td>
</tr>
<tr>
<td>0,50</td>
<td>16,9</td>
<td>15,4</td>
<td>14,1</td>
<td>13,0</td>
<td>12,1</td>
</tr>
<tr>
<td>0,75</td>
<td>15,4</td>
<td>14,1</td>
<td>13,0</td>
<td>12,1</td>
<td>11,3</td>
</tr>
<tr>
<td>1,00</td>
<td>14,1</td>
<td>13,0</td>
<td>12,1</td>
<td>11,3</td>
<td><strong>10,6</strong></td>
</tr>
</tbody>
</table>

The most likely CR for HCCI is 13-15. If the CR 14,1 is chosen from there are five different piston settings to achieve this. The most natural choice is to mount two similar pistons with the reduced height of 0,5 mm each crank shaft. If instead an original piston is mounted on the exhaust crank shaft and a processed piston with the reduced height of 1 mm is mounted on the inlet crank shaft, the port opening timing will change. This enables tests with later closing of the inlet ports which opens on to test with a supercharged cylinder scavenged by an exhaust driven turbo. This is a concept used by many two stroke uniflow engines.
The reason why maximum compression ratio in Figure 8 appears at a phase difference of 5 degrees can be explained in Figure 9. The inlet port always closes at CA 245 ATDC but since the exhaust port closes at 250 CA ATDC, it needs a phase adjustment of 5 degrees to close simultaneously, which encloses the most gases and results in the highest compression ratio.

![Port opening diagram at 0 and 30 degrees phase difference.](image)

Figure 9. Port opening diagram at 0 and 30 degrees phase difference.

3.1.2 Scavenging system

One vital function on a two stroke engine is the scavenging. The scavenging takes place as the pistons are positioned close to BDC (Bottom dead center) and has the purpose to clear the cylinder from burned gases and replace with fresh air/fuel mixture.

The fresh air/fuel mixture needs pressure to scavenge the cylinder from burned gases. On the first engine this was achieved with the crankcase pressure, but on the later engine an external displacement pump was installed, see Figure 5. The pump is connected to the inlet crank shaft via a belt with optional gear ratio, which allows running the engine at different Delivery Ratios. The current gear ratio 1.5:1 gives a DR of 46 %.
As the inlet ports opened the plenum pressure decreased from absolute pressure 1.5 to 1.0 bar within 10 crank angle degrees. This quick flow was not ideal for the injection of fuel that often needed more than 20 crank angle degrees to complete the injection. This was solved by mounting a pressure accumulator between the scavenge pump and plenum.

Most two stroke engines are loop scavenged, which means that the fresh air/fuel mixture is forced to flow in a loop from the inlet port(s) on the bottom of the cylinder up to the spark plug and then back to the bottom again where it leaves through the exhaust port(s). The loop scavenging is simple but not to prefer when considering the emissions of hydrocarbons and carbon monoxides. This is because the close location of the inlet and exhaust ports allows some of the fuel/air mixture to short circuit directly to the exhaust port.

The Double Two engine uses a scavenging principle called uniflow. The main difference compared to the loop scavenging is that the inlet port(s) and the exit port(s) are not located in the same end of the cylinder. If the ports and the plenum are designed well, the airflow will create a swirl that moves from the inlet ports towards the exhaust ports with a high trapping efficiency.

The first prototype consisting of two loop scavenged cylinders did not have the swirl needed, despite the separate location of the inlet and exhaust ports like a uniflow scavenged engine. The scavenging was improved as the cylinder was redesigned. The Junker Jumo -205 [1] became the model for the design of plenum, inlet ports and exhaust ports. Plenum and the five ports are marked with light green in Figure 11.
3.1.3 Fuel system
The fuel system consists of an injector, fuel tank, two filters, hoses, pressure relief (3 bar rel.) and a fuel pump. Under the fuel tank is a high resolute scale to measure the fuel consumption. (See Figure 12) The injector is a Bosch 110cc with two jets and an angle of 22 degrees in between. The ECU has a voltage compensating function to avoid disturbance of the effective injection time.

Figure 12. Fuel tank on scale and Bosch injector

Booth direct injection and semi direct injection have been tested with negligible differences. Two of the four cylinders do not have the hole for direct injection, seen as the dark green field in Figure 11. The reason for this is to leave options if the original injector placement is insufficient. The optional ideas were e.g. SDI, DI with a deflector and DI after the ports closed.

After a few hours of engine testing the lubrication level increased and started to smell gasoline. This happened both when running with DI injection and SDI injection. Because there is no emission test performed, it’s uncertain why and at what speeds, loads and injection angles it occurs. Apparently, there is some work to be done to improve mixing of fuel/air and avoid wall/piston wetting.

In Spiros this fuel system was replaced because no electric energy is allowed for fuel pumps in the Shell Eco Marathon. The competition version had a fuel tank pressurized with air at a pressure of 3 bar coming from a 1,5 liter bottle pressurized to 5 bar via a relief valve.
3.1.4 Cooling system
The DoubleTwo engine is liquid cooled. The liquid circulates from the engine to the cooler and back via an electric pump. The pump is supplied by 12 voltages and controlled by Cell4 with PWM to a given engine surface temperature.

The first cooling pump was replaced since when activated it caused a voltage drop in the electric system that increased the injection delay and resulted in a leaner fuel/air mixture.

3.1.5 Startup ignition system
The DoubleTwo engine needs a start up system until there is enough warm residual exhaust gas that helps the HCCI combustion. That is done by an ignition system consisting of ignition coil, ignition cable and an extra small spark plug with ¼ -32 thread (See Figure 13). The small plug size was kept from the first generation where a bigger sparkplug couldn’t fit in the brass plate. The idea was that a smaller spark plug would add less passive volume to the cylinder than a bigger plug, and therefore slightly improve the efficiency. The backside has been trouble to find plugs and cables of good enough quality.

![Figure 13. Ignition coil and ¼ -32 threaded spark plugs](image)

3.1.6 Lubrication system.
The lubrication system consists of a crank shaft mounted oil pump, tank, lube hoses and hoses for the crankcase gases (See Figure 14). The tank has a level indicator. Below the level indicator is the outlet hose to the pump, which gives a over dimensioned flow of 0,5 g/min at 3100 rpm [3]. The lubrication is fed directly to the top of both crank houses. The residual oil returns from bottom of the crank houses to the container together with blow by gases. The lube container is ventilated with a hose that is connected to the inlet of the scavenging pump. This way both the lubrication of the scavenging pump and recirculation of blow by gases is taken care of. At some periods a 2 percentage oil addition to the fuel was used.

The nicasil coated cylinder walls have been honed to improve the lubricating capabilities.
3.2 The electric system.

The electric system is too complex to describe in detail in this report. Figure 15 is illustrating an overview of communication and connections for the engine system.
3.2.1 Cell Control Unit
Each test cell is controlled by a cell control unit (CCU). The interface software, called cell4, has been developed by Professor Hans Erik Ångström. In this case the CCU doesn’t control the engine except for the cooling system. Cell4 with all belonging sensors and microprocessors allows making high/low frequency engine logs. All results in this report have been calculated based on these high/low frequency logs.

![Figure 16. The Cell control unit (CCU)](image)

3.2.2 Description of the electric brake
An electric DC PM motor has been the combined brake and starter, controlled by a hybrid control unit via a servo amplifier, see Figure 17. The produced energy has been stored in a super capacitor, and consumed by an electric power board.

![Figure 17. The generator, Servo amplifier, super capacitor and power board](image)

3.2.3 Description of the sensors
There are a dozen of sensors on the engine system. This chapter describes the two most important sensors on which the ECU bases all control: the two angle sensors and the pressure sensor.

The ECU uses an optic cylinder pressure sensor from OPTRAND to control start of combustion and maximum cylinder pressure, see figure 20. The pressure sensor is dynamic, which means it can only measure pressure differences. The ECU has a calibration function that sets the pressure to atmospheric around BTC every rotation.

If an installation is necessary, make sure that a new copper gasket is correctly positioned on the head of the sensor.
Hall sensor 1 measures the inlet crank shaft angle and hall sensor 2 measures the exhaust crank shaft angle. They are mounted close to the gear wheels and detect the teeth passing, see Figure 19. One both gear wheels one of the 32 teeth is partly missing (missing tooth principle). The missing tooth becomes a reference for the control system on which it bases e.g. phase difference, injection timing and combustion calculation. The missing tooth is located around crank angle 230 ATDC in Figure 18.

![Figure 18. Signal from hall sensor detecting teeth and the missing tooth.](image18)

![Figure 19. Hall sensor and missing tooth.](image19)

![Figure 20. In-cylinder pressure sensor.](image20)

3.2.4 Description of the closed loop engine control system
Because the Doubletwo engine is running as a HCCI engine, it needs a control unit (ECU) to avoid early damaging combustions. The control unit is based on a PIC microprocessor and uses the combustion timing as a closed loop control parameter. The control unit controls the injection, ignition and compression ratio based on two angle sensors and an in-cylinder pressure sensor.
The system has two modes (see Figure 21), one for SI (spark ignition) and one for HCCI (compression ignition). The engine always starts in SI-mode to get the right temperature needed for HCCI mode. In SI-mode the air/fuel mixture is close to stochiometric (lambda = 1) and the compression regulation has a constant phase value. Mode 2 is activated after a certain time. The approach of switching to HCCI from SI is done by changing both the compression ratio and lambda at the same time. The Compression ratio increases with one step per rotation and the air/fuel-mixture is leaned with one step per rotation. As the system detects the first combustion with HCCI ignition, the sparkplug is shut off. The lambda is still increasing one step per rotation until the requested value is reached, but the compression regulation now switches strategy. As long as the combustions are detected as HCCI combustions, the control system is adjusting the compression ratio to achieve start of combustion at the right crank angle. If two repeated misfires are detected, the ECU switches to mode 1 again.

![Diagram of the HCCI control strategy](image)

**Figure 21. Illustration of HCCI-control strategy.**

The ECU can detect the start of combustion angle at all compression ratios and phase differences. To be able to do this, the ECU calculates the compression cylinder pressure, based on interpolation of different values of Kappa and the actual pressure from previous cycle. As the cylinder pressure increases more than the detection level, the ECU saves the angle as start of combustion.

The HCCI detection strategy builds on calculations from the in-cylinder pressure. As the crank angle difference between the start of combustion and maximum pressure is less than a specified angle, The ECU defines the combustion as a HCCI-combustion.

4 Data

4.1 ECU Parameters

The ECU basic strategy is described in 3.2.1. There are several parameters that need to be correct for a proper engine control to take place. Table 3 explains the function of each parameter and what’s necessary to know about them. RV means recommended value. NV means normal value. Table 4 explains the log data from the ECU.
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>RefAngle missing 1 Ca bef. TDC</strong></td>
<td>The calibrated angle between TDC of crank shaft and lost pulse detection.</td>
</tr>
<tr>
<td><strong>Angle between missing 1 &amp; 2</strong></td>
<td>The calibrated angle between lost pulses from the two crank shafts (hall 2- hall 1) at 0 degree phase difference. Hall 2 pulse must always be ahead which means the value typed here must be positive.</td>
</tr>
<tr>
<td><strong>Injection Angle after TDC</strong></td>
<td>A positive number that sets the angle of injection start relative to the inlet crank shaft. This number must be between 90 and 200 degrees. Else the ECU might not function correctly.</td>
</tr>
<tr>
<td><strong>Injection time OTTO start [ms]</strong></td>
<td>Sets the injection duration for Otto mode.</td>
</tr>
<tr>
<td><strong>Ignition current on Angle before TDC</strong></td>
<td>Sets the angle relative the inlet crank shaft where the ignition coil starts to charge. Note that the ignition coil can be overheated if it’s charged to long. RV: 100 (1 ms = 36 degrees at 6000 rpm).</td>
</tr>
<tr>
<td><strong>Ignition Angle before TDC</strong></td>
<td>Sets the angle where the ignition takes place. RV 35</td>
</tr>
<tr>
<td><strong>Phase difference OTTO comb. [deg]</strong></td>
<td>Sets the start up phase difference. RV 15-20.</td>
</tr>
<tr>
<td><strong>Reg. Time OTTO comb. ms/cycle</strong></td>
<td>Sets the phase control motor on time for the start up phase adjustment described in previous point. RV 1-5 (10 ms = one rev at 6000 rpm).</td>
</tr>
<tr>
<td><strong>OTTO starting time [sec]</strong></td>
<td>Sets the time after which the engine goes into mode 2.</td>
</tr>
<tr>
<td><strong>HCCI max angle btw. pMax-CombAng</strong></td>
<td>Sets the definition of a HCCI combustion. When the angle between start of combustion and maximum cylinder pressure is less than this value the ECU defines the combustion as a HCCI (mode 3). RV 5-8</td>
</tr>
<tr>
<td><strong>Injection time step to HCCI-inj ms/cycle</strong></td>
<td>Sets the step in mode 2 of how much less fuel to inject compared to previous cycle. RV 0.1</td>
</tr>
<tr>
<td><strong>Injection time HCCI [ms]</strong></td>
<td>Sets the value of which the previous value is reducing to.</td>
</tr>
<tr>
<td><strong>Compression volume zero phase [cm³]</strong></td>
<td>Input variable for Volume and Kappa calculations. This variable must be changed as the pistons are replaced to adjust the maximum compression ratio.</td>
</tr>
<tr>
<td><strong>Cyl. Pressure bar/volt</strong></td>
<td>Calibration value from pressure sensor. The calibrated value should be divided by three to compensate for the internal gain.</td>
</tr>
<tr>
<td><strong>Pressure difference min comb. [bar]</strong></td>
<td>Sets the minimum pressure difference between the calculated cylinder pressure curve and measured cylinder pressure where the ECU can detect start of combustion.</td>
</tr>
<tr>
<td><strong>Pressure difference for kappa [bar]</strong></td>
<td>Sets the “safety distance” between the calculated cylinder pressure and the measured pressure.</td>
</tr>
<tr>
<td><strong>Pres. Diff. For comb. % of last</strong></td>
<td>Sets the level of combustion detection as the engine has started. The ECU compares the cylinder pressure rise from the calculated pressure curve with the previous combustion total pressure rise. RV 30-50%</td>
</tr>
<tr>
<td><strong>Delay control motor start [ms]</strong></td>
<td>Time delay for phase motor start.</td>
</tr>
<tr>
<td><strong>Reg. Time at no HCCI comb. ms/cycle</strong></td>
<td>Sets the maximum on time for the phase control motor. (10 ms = one rev at 6000 rpm).</td>
</tr>
<tr>
<td><strong>Control factor compression ms/deg</strong></td>
<td>Sets the proportional control factor for the phase control motor.</td>
</tr>
<tr>
<td><strong>Combustion set Angle deg ATDC</strong></td>
<td>Sets the wanted combustion angle. RV 2-6</td>
</tr>
<tr>
<td><strong>Speed limit [rpm]</strong></td>
<td>Sets the upper speed limit of the engine.</td>
</tr>
<tr>
<td><strong>OverSp. Inj. Time decrease ms/cycle</strong></td>
<td>Sets the reduction of fuel injection time when the engine is over speeding.</td>
</tr>
<tr>
<td><strong>Inj. Delay Zero voltage ms</strong></td>
<td>Sets the first parameter in the voltage compensating function for injection time. RV 2.82</td>
</tr>
<tr>
<td><strong>Zero inj. Delay Voltage</strong></td>
<td>Sets the second parameter in the voltage compensating function for injection time. RV 22.32</td>
</tr>
</tbody>
</table>
Table 4 ECU log explanation

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>RevNo</td>
<td>Tells at which revolution the log point is taken. The revolution counter is reset by resetting the power to the ECU.</td>
</tr>
<tr>
<td>CaDeg</td>
<td>Tells at which crank angle the log point is taken.</td>
</tr>
<tr>
<td>Tms</td>
<td>Tells the time from start of log at which the log point is taken.</td>
</tr>
<tr>
<td>RPM</td>
<td>Tells the speed of the engine in rpm, measured from hall sensor 1.</td>
</tr>
<tr>
<td>CombDeg</td>
<td>Tells at which crank angle the combustion is detected.</td>
</tr>
<tr>
<td>DiffDeg</td>
<td>Tells the phase difference between the crankshafts.</td>
</tr>
<tr>
<td>PcMaxBar</td>
<td>Tells the maximum cylinder pressure during last rotation.</td>
</tr>
<tr>
<td>PcMaxDeg</td>
<td>Tells the crank angle where the maximum cylinder pressure took place.</td>
</tr>
<tr>
<td>PCylDecBar</td>
<td>Tell how much the cylinder pressure decreased before combustion compared to calculation. NV:1-5 bar</td>
</tr>
<tr>
<td>PCylIncBar</td>
<td>Tell how much the cylinder pressure increased because of combustion compared to calculation. NV:0-40 bar</td>
</tr>
<tr>
<td>HiKap%</td>
<td>Tells the calculated Kappa estimation. 0 % means Kappa is 1.2 and 100 % means kappa is 1.4</td>
</tr>
<tr>
<td>CtrlMode</td>
<td>Tells what mode the engine is running in. 1 = SI , 2 = transfer SI to HCCI, 3 = HCCI</td>
</tr>
</tbody>
</table>

4.2 Calibration of Heidenheim CA sensor

For accurate crank angle samples in the CCU the Heidenheim angle sensor needs to be calibrated towards the crank shaft on which it is attached.

Calibration procedure:
1. Bring the calibration computer and a metal stick of 5 mm diameter (e.g. screwdriver)
2. Close down the electric system.
3. Remove the two center gear wheels.
4. Remove the spark plug or the inlet/exhaust pipe depending on which shaft to calibrate.
5. Make sure that the Heidenheim sensor is firmly mounted to the crank shaft and plug it in to the calibration computer.
6. Start the computer and load the program CAMLOG by typing it in the DOS-prompt.
7. From the menu, load the setup file named HCCI.CSF
8. Rotate the crank shaft in the correct direction of rotation and control that the angle increases and that one rotation correspond to 360 degrees. If any of these are not correct, make necessary changes in the menu.
9. Insert the metal stick into the spark plug hole or one of the ports. Be careful not to damage the pistons or cylinder wall coating.
10. Slowly turn the crank shaft one direction until it stops. Note the angle given by the calibration computer.
11. Slowly turn the crank shaft the other direction until it stops again. Note this angle as well.
12. The difference of the two angles should be less than 90 degrees. If not, the angle sensor has its 360 degrees interval in between the two angles. This is easy corrected by adding or removing 360 from one of the values so that the mean value of booth
13. The mean value of the two angles corresponds to the angle given by the Heidenheim sensor as the crank shaft is in TDC position.
14. Close down the calibration computer before you unplug the crank angle sensor.
15. Open Cell4 in the CCU. Open the menu: channel setup screen. Open logical channels from the channelgroup menu. Type the mean value with a minus sign before in the field named: CA marker after TDC degrees (normally -).
16. Save the setup file with a new name.
17. Mount the centre gear wheels and the spark plug/inlet pipe/exhaust pipe back in position.
4.3 Calibration of Hall angle sensors
If both hall angle sensors need to be calibrated to the crank shafts, the inlet crank shaft should be calibrated first to avoid unnecessary extra work.

4.3.1 Exhaust crank shaft calibration
The hall angle sensor is not static, which means that it needs rotation to send an accurate angle. To be able to calibrate the hall sensors the Heidenheim sensor must be calibrated first which is described in previous chapter. After that calibration is done connect all sensors to the ECU and CCU (cell4).

Calibration procedure:
1. Unplug Hall 1 from the ECU and replace with Hall 2.
2. Make an estimation of the angle between TDC and the lost tooth passage of the hall sensor. Type this value in the field of the ECU control named: RefAngle missing 1 Ca bef. TDC. (Note the original number somehow)
3. Set the start of injection to 180 degrees ATDC (See Table 3).
4. Motor the engine at a constant speed and make a Pdaq. log of at least the injection timing in the CCU.
5. Compare the sampled start of injection in the CCU with that typed in the ECU and compensate for the difference where the estimated angle was typed in the first step.
6. Plug the hall sensors back in right positions.
7. Change back to the original value in the field of the ECU control named: RefAngle missing 1 Ca bef. TDC.
8. Take the result from the exhaust hall sensor calibration and subtract the result from the inlet hall sensor calibration. Type this angle in the ECU field named: Angle between missing 1 & 2. (This number must be positive.)

4.3.2 Inlet crank shaft calibration
The high resolute heidenheim angle sensor is for the moment mounted on the exhaust crank shaft. Because of this, calibration of the inlet crank shaft is a little more complicated.

Calibration procedure:
1. Remove the Heidenheim sensor from the exhaust crank shaft and instead mount it on the inlet crank shaft. (After removing the lube pump)
2. Make an estimation of the angle between TDC and the lost tooth passage of the hall sensor. Type this value in the field of the ECU control named: RefAngle missing 1 Ca bef. TDC
3. Set the start of injection to 180 degrees ATDC.
4. Motor the engine at a constant speed and make a Pdaq. log of at least the injection timing in the CCU.
5. Compare the sampled start of injection in the CCU with the typed in the ECU and compensate for the difference where the estimated angle was typed in the third step.
6. Keep the final result from previous point. If the exhaust hall sensor is not calibrated, do so before continuing.
7. Take the result from the exhaust hall sensor calibration and subtract the result from the inlet hall sensor calibration. Type this number in the ECU field named: Angle between missing 1 & 2. (This number must be positive)
5 Results

5.1 Cylinder pressure analyzes.
The following three figures (Figure 22, Figure 23 and Figure 24) all contain data from the same test. The test is done with direct injection at 6500 rpm. The Lambda of 1.45 is just a relationship between the measured air and fuel injected. Since no measurements of lambda or tests of trapping efficiency and emissions have yet been carried out, the exact lambda is unknown.

![Figure 22. Cylinder pressure for 40 following HCCI combustions at 6500 rpm.](image)

<table>
<thead>
<tr>
<th>Speed: 6500 rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Phase diff: 11 deg</td>
</tr>
<tr>
<td>CR: ~15.5</td>
</tr>
<tr>
<td>DR: 47 %</td>
</tr>
<tr>
<td>Lambda: 1.45</td>
</tr>
<tr>
<td>El.Power: 780 W</td>
</tr>
<tr>
<td>El work 7.2 J/rev</td>
</tr>
<tr>
<td>Imep: 4 bar</td>
</tr>
<tr>
<td>η teoretic: 32.5 %</td>
</tr>
<tr>
<td>η el: 12 %</td>
</tr>
</tbody>
</table>
20,0 joule is the work from the gas on the pistons but only 7,2 joule is measured electrically (Current times Voltage) out from the servo amplifier. The efficiency of the electric brake is not known. Speculating experts has told it’s between 50 and 80 %. This would mean that the braked power would actually be between 1000 and 1550 W instead of 780 W and the braked efficiency between 15 and 24 % instead of 12%.

As mentioned earlier, the combustion in a HCCI engine is very quick. Figure 24 shows that the whole combustion duration is less than 10 degrees. It also shows a continuing ripple which could be either mechanical vibration after the heavy pressure rise or from pressure oscillations.
5.2 Motored pressure and losses

The engine was motored with two similar pistons with the reduced height of 0.25 mm each (see Figure 25). The maximum compression ratio at these settings is 16.9 according Table 2.

![Motored pressure at 3000 rpm](image)

Figure 25. Motored cylinder pressure at 3000 rpm

During another test when the engine was motored at 5250 rpm the electric motor consumed 900 W. The actual power needed is lower because of losses in the electric motor and in the power amplifier. However the losses corresponds to 10.3 Joule/revolution or a moment of 1.63 Nm. These losses includes piston ring friction, scavenge pump propulsion, gear wheel losses, bearing friction, electric motor losses and electric converting losses in the power amplifier.
6 Conclusions

Unlike before the DoubleTwo engine now runs and produces power. To get here the cylinder has been redesigned to improve the scavenging, the compression control system has been redesigned for mechanical endurance, a start up ignition system has been added and an ECU has been build etc.

The engine produces about 800 W electric braked power at 6500 rpm. The indicated efficiency is 32.5 % and Indicated mean effective pressure is 4 bar. The electric braked efficiency was in the same test 12 %. The problem is that the efficiency of the electric brake and the servo amplifier is unknown.

It’s too early to draw any big conclusions from these few tests. What can be said is that the engine has performed acceptable results considering the relative short time of test and development.

The next suggested step would be to construct a larger engine, i.e. 500 cc. A larger engine would naturally have a smaller proportion of piston ring blow by and friction losses. A larger engine would also be easier to supercharge with an exhaust driven turbo.
7 Future work

The first four suggestions of work were originally planed to be performed during this spring.

1. Design of a stable and reliable brake.
2. Measurements of lambda and emissions with focus on HC.
3. Further tests with purpose of low fuel consumption. CR, DR, speed, lambda
4. Further tests on fuel flexibility.
5. Deeper investigation of wall wetting.
6. Change to thicker piston rings to maintain the high cylinder pressure.
7. Design and optimization of an exhaust system with focus on pulsations.
8. Measurements of vibrations from the crank shafts.

When the engine is considered being “superior enough”, the next step should be to design an engine in the 500 cc size. The reason for this is that small engines always have difficulties to reach high levels of efficiencies. Booth internal losses (such as piston ring friction and blow by) and surrounding engine systems (such as scavenging pump and gears) takes a bigger proportion of the produced power from smaller engines.
8 References


9 Publications
Below are some links to articles about this engine. Except for the following links the engine is mentioned in many reports and articles.

Sweden
Ny teknik, Published 2008.04.16
http://www.nyteknik.se/nyheter/innovation/forskning_utveckling/article331987.ece

Norway
Teknisk Ugeblad, Published 2007.05.08
http://www.tu.no/nettarkiv/article102667.ece

Denmark
Ingeniøren, Published 2008.05.30
http://ing.dk/artikel/88589?highlight=HCCI (Requires a free membership to read article)

World Wide Web
Youtube-video describing the compression ratio control system.
http://www.youtube.com/watch?v=nLHIgucStVA