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## Closed-Loop Combustion Control of a Multi Cylinder HCCI Engine using Variable Compression Ratio and Fast Thermal Management

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# Balancing cylinder-to-cylinder variations in a Multi-Cylinder VCR-HCCI engine.

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## ABSTRACT

Combustion initiation in an HCCI engine is dependent of several parameters that are not easily controlled like the temperature and pressure history in the cylinder. So achieving the same ignition condition in all the cylinders in a multi-cylinder engine is difficult. Factors as gas exchange, compression ratio, cylinder cooling, fuel supply, and inlet air temperature can differ from cylinder-to-cylinder. These differences cause both combustion phasing and load variations between the cylinders, which in the end affect the engine performance. Operating range in terms of speed and load is also affected by the cylinder imbalance, since misfiring or too fast combustion in the worst cylinders limits the load.

The cylinder-to-cylinder variations are investigated in a multi-cylinder Variable Compression Ratio (VCR) engine, and the effect it has on the engine performance. Different strategies to balance the cylinders are tested, i.e. static balancing of cylinder individual compression ratio and inlet air temperature, and closed loop control of cylinder individual combustion phasing using fuel offsets or inlet air temperature. The best engine performance regarding fuel consumption, combustion stability and emissions was found with cylinder individual combustion phasing control using inlet air temperature.

The test engine is a, 1.6L, four-stroke five-cylinder engine with Port Fuel Injection (PFI), Thermal Management of inlet air temperature, and Variable Compression Ratio (VCR). HCCI is achieved and combustion phasing is controlled with compression ratios up to 21:1 and adjustable inlet air temperature. The test fuel fulfills the specifications of US Regular and has a RON/MON of 92/82.

## INTRODUCTION

Homogeneous Charge Compression Ignition (HCCI) combustion is one of the most promising internal combustion engine concepts for the future. HCCI is however not a recent discovery. Already in the early

twentieth century hot bulb engines operated with HCCI-like combustion [1]. They were superior in terms of brake efficiency compared to the contemporary gasoline engines and at the same level as the diesel engines. The drawback was low specific output and long start up time. These drawbacks still remain today.

The first efforts to characterize HCCI combustion were done on two stroke engines [2, 3] and the primary reasons were to reduce unburned HC at part load and to decrease fuel consumption by stabilizing the combustion of lean mixtures. Shortly after HCCI was also implemented in four stroke engines [4, 5] and the benefits of the concept described, i.e. high efficiency and low NO<sub>x</sub> emissions compared to SI engines. Since then a lot of research effort has been put on understanding HCCI combustion and how to control it.

HCCI combustion is achieved by compressing a fairly homogenous, highly diluted, air/fuel mixture to self-ignition. A highly diluted mixture is needed to decrease burn rate and avoid combustion noise. Controlled auto ignition can be achieved by controlling the temperature and pressure history, the fuel properties, or mixture composition in the cylinder [6-20]. The mixture temperature can be controlled by inlet air heating, efficient compression ratio, or retaining hot residuals in the cylinder. The fuel properties, i.e. ignitability, can be changed with dual fuel systems or fuel reforming, i.e. injecting the fuel during the negative valve overlap. Mixture composition can be affected by stratification in Direct Injection (DI) systems. HCCI combustion in multi-cylinder engines is an additional challenge due to cylinder-to-cylinder variations [6, 7, 13, 18, 22, 23 and 24].

HCCI today is conceived as a part load combustion concept to decrease fuel consumption at low loads. Normal SI or diesel combustion can instead be used at start up and high load. The operating range in terms of speed and load for an HCCI engine is restricted by misfiring at low load, i.e. too much dilution, and fast burn rate that induces noise and NO<sub>x</sub> emissions at high load,

i.e. too little dilution. The fuel/air mixture is leaned out towards low loads and the compression temperature becomes insufficient to complete combustion and combustion efficiency decrease.

In this paper the cylinder-to-cylinder variations in multi cylinder HCCI engines are addressed. To achieve the same combustion performance in all cylinders is important. With an uneven performance the worst cylinders are restricting the whole engines performance and the operating range in terms of speed and load. The worst cylinders are both those that are close to misfiring, i.e. partial burning with poor combustion efficiency, and those with too early combustion phasing producing NOx emissions and noise, i.e. high maximum rate of pressure rise. In this paper an effort is made to investigate and quantify the parameters affecting the cylinder-to-cylinder variations. Different strategies to balance the cylinders are tested, compared, and how they affect the performance of the total engine.

## EXPERIMENTAL APPARATUS

The test engine is a five-cylinder 1.6L Saab Variable Compression (SVC) engine [21], see Figure 1. The pistons have been changed to achieve a Variable Compression Ratio (VCR) range between 9:1 and 21:1. The VCR is achieved by tilting the monohead, i.e. cylinder head and liners, around a pivot so that the compression volume changes [21, 23]. The spark plugs have been removed and replaced by plugs. The mechanical supercharger and the standard inlet manifold have been replaced by an in-house made inlet air manifold. Exhaust gas residuals are not used to promote ignition, i.e. standard and fixed valve timings are used. Some engine specifications can be seen in Table 1.

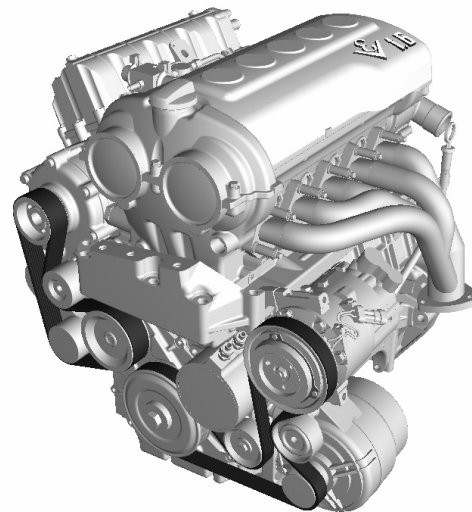
**Table 1, engine specifications.**

Displacement	1598 cm <sup>3</sup> (320 cm <sup>3</sup> /cyl)
Number of cylinders	5
Compression Ratio	Adjustable 9–21:1
Bore x Stroke	68mm x 88mm
Exhaust valve open	45°BBDC at 0.15mm lift
Exhaust valve close	7°ATDC at 0.15mm lift
Inlet valve open	7°BTDC at 0.15mm lift
Inlet valve close	34°ABDC at 0.15mm lift
Combustion chamber	Pent roof / 4 valves DOHC

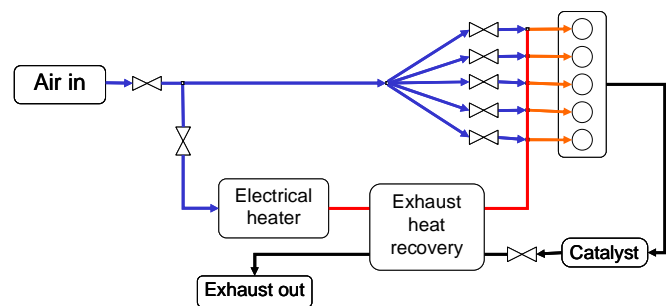
The inlet air and exhaust gas systems have been modified, see the principle lay-out in Figure 2. It consists of an electrical air heater, exhaust to air heat exchanger, oxidizing catalyst, and some throttle valves for controlling the air flow. The electrical air heater is only used for starting up the HCCI combustion with a cold engine and turned off when the engine is hot. Inlet air temperature is controlled by mixing hot and cold air as soon as the engine has warmed up sufficiently. The mixing can either be done globally, i.e. mixing the hot and cold air by one throttle valve on each side for all the

cylinders at the same time, or cylinder individually, i.e. mixing the hot and cold air before each cylinder with its own throttle valve, see Figure 2. By mixing cylinder individually the inlet air temperature can be adjusted separately for each cylinder.

Combustion phasing can be controlled by changing the compression ratio [22] or by controlling the inlet air temperature [25]. The closed-loop combustion phasing control is done by analyzing the cylinder pressure from all cylinders with in-house developed software. Various variables are calculated on-line for control or engine protection, such as net IMEP, coefficient of variation of net IMEP (COV IMEP), Crank Angle at 50% cumulative heat release (CA50), peak cylinder pressure, and maximum rate of pressure rise (dP/dCA). The transient control capabilities have been investigated with this engine in two other papers [22 and 25]. The closed-loop combustion phasing control is fast and has real potential to handle engine transients. A step change in CA50 between 1 and 9°ATDC, at 2000rpm and 2bar BMEP, has a time constant of 8 engine cycles with closed-loop control using inlet air temperature [25]. The corresponding time constant for the closed-loop control using compression ratio is 14 engine cycles [22].



**Figure 1, The Saab Variable Compression test engine.**



**Figure 2, Inlet air and exhaust gas system.**

Water-cooled cylinder pressure sensors from Kistler, model 6043A, are flush mounted in all cylinders. The cylinder pressures are sampled with an A/D-converter, Wavebook 516 with 8-channel simultaneous sample and

hold from IOtech, capable of an aggregate rate of 1Msamples/s. The sample rate is 5 samples per CAD per cylinder for engine speeds below 3000 rpm and 2.5 samples per CAD per cylinder above 3000 rpm.

The exhaust emissions presented in this paper are measured from each cylinder separately, so that cylinder-to-cylinder differences can be quantified. The regulated emissions are measured with a Horiba exhaust gas analyzer MEXA-8120F. The CO and CO<sub>2</sub> analyzer, Horiba model AIA-23, works with Non-Dispersive Infrared (NDIR) technology. The NO<sub>x</sub> (NO+NO<sub>2</sub>) analyzer, Horiba model CLA-53M, uses a Chemiluminescence Light Detector (CLD) to measure the chemiluminescence from the reaction between NO and ozone (O<sub>3</sub>). HC emissions are measured with a High-temperature Flame Ionization Detector (HFID), Horiba model FIA-22-2, as ppm Carbon. The O<sub>2</sub> analyzer, Horiba model MPA-21, uses the paramagnetic property of oxygen and measures the pressure elevation in a disproportional magnetic field. Additionally CO emissions above 0.35vol% are measured with another portable NDIR analyzer Horiba MEXA-324GE. All exhaust gas emissions are measured dry except HC that is measured wet. The relative air/fuel ratio, lambda, is calculated from the exhaust gas emissions by applying carbon, hydrogen, oxygen and nitrogen balances to a chemical reaction between HC and air. Lambda is also measured directly with an ETAS LA3 broadband lambda sensor. The ETAS LA3 consists of a broadband LSU lambda sensor from Robert Bosch and a Lambda Meter, LA3, from ETAS. The lambda sensor works according to the Universal Exhaust Gas Oxygen (UEGO) Sensor principle [26]. The broadband lambda sensor is used for quick screening during test. The regulated emissions in this paper are presented as brake specific emissions [g/kWh].

Fuel consumption is measured by weighing. Parameters that are used for control, like inlet air pressure, inlet air temperature, and compression ratio, are measured with a multi function PCI card NI6052E from National Instruments. The sample rate is one tenth of the sample rate for the cylinder pressure. The compression ratio is measured and calculated from an angular sensor on the eccentric shaft that is used for tilting the engine cylinder head to achieve the different compression ratios. All variables can be saved for post processing, e.g. heat release analysis. Other variables, such as temperatures and pressures of air, exhaust, water, and lubricating oil, are logged with a HP data logger at a rate of approximately 0.33Hz.

## RESULTS AND DISCUSSIONS

### TEST PROCEDURE

#### Fuel

A gasoline with a specification corresponding to U.S. Unleaded Regular is used as fuel in the tests. The fuel has RON between 91 and 92, and MON between 81.5 and 82.5, i.e. a road octane number of 87.

#### Cylinder balancing

Cylinder-to-cylinder variations can be controlled by different strategies. In this paper three different strategies are examined and compared to an uncontrolled case. The engine speed is kept at 2000rpm and the total fuel amount supplied to the engine is kept constant. The achieved load is about 2 bar BMEP and differs slightly from case to case due to different brake efficiency with the control strategies. The compression ratio is set to the maximum 21:1 in all the cases. The three different control strategies and the uncontrolled reference are listed in Table 2 as four different cases.

Case 1 is the uncontrolled case, i.e. the injected fuel amount is the same to all the cylinders and no cylinder individual control of the inlet air temperature or the combustion phasing is done. In case 2 the combustion phasing, CA50, is separately controlled for every cylinder with inlet air temperature. In case 3 the inlet air temperature is adjusted to the same level in all the cylinders. In case 2 and 3 the injected fuel amount is still the same to all the cylinders. In case 4 the combustion phasing, CA50, is separately controlled for every cylinder with the fuel amount and no adjustment of cylinder individual inlet air temperature is done. The cylinder average combustion phasing, CA50, is controlled to 6°ATDC with global inlet air temperature in case 1, 3, and 4. Close-loop control is applied to the combustion phasing control of CA50, i.e. the cylinder pressure is measured and analyzed for every cycle and every cylinder on-line [22, 25].

Another strategy, which is not tested in this paper, to do cylinder balancing, is to control the amount of cylinder individual exhaust residuals. This could be done with e.g. cylinder individually variable valve timings. The effect on combustion phasing should be similar to inlet air temperature.

**Table 2, the compared control strategies to handle cylinder-to-cylinder variations.**

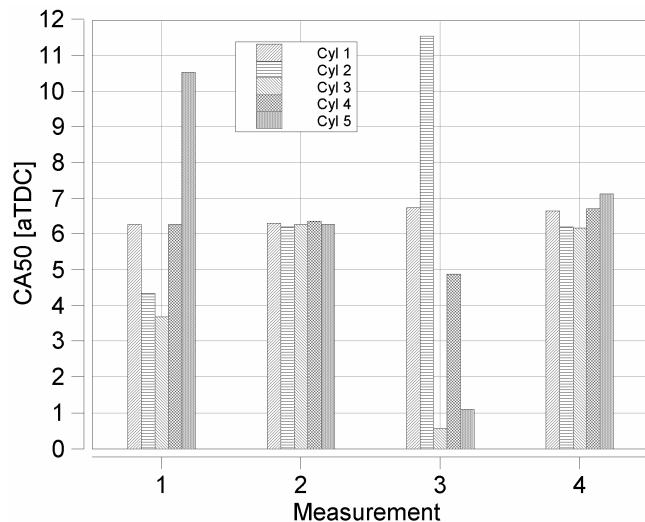
Case	1	2	3	4
Inlet air temperature	no adjust	adjust	even	no adjust
Combustion phasing	no control	control	no control	control
Fuel amount	even	even	even	adjust

The four tested cases are compared to each other by the help of cylinder individual performance data presented in nine different figures from Figure 3 to Figure 11. The compared variables in the figures are: CA50, inlet air temperature, lambda, net IMEP, COV IMEP, specific NO<sub>x</sub>, Emission Mean Effective Pressure1 (Q<sub>emiss</sub>MEP), maximum rate of pressure rise (dP/dCA), and brake efficiency.

## TEST RESULTS FROM CYLINDER BALANCING

### Control of the cylinder-to-cylinder variations

In Figure 3 the cylinder individual combustion phasing is presented for the four cases. In case 2 and 4 the combustion phasing for the different cylinders are almost the same, because they are controlled by cylinder individual inlet air temperature and fuel amount respectively. In case 1 and 3 the combustion phasing differs between the cylinders because of different reasons and they are examined more thoroughly below.



**Figure 3, cylinder individual combustion phasing for the different cases.**

#### Case 1, without any control of cylinder-to-cylinder variations

In this basic case no control is used to adjust the cylinder individual differences and the same fuel amount is supplied to all the cylinders. The combustion phasing, CA50, differs from 3.5 to 10.5°ATDC. The main reason for the late timing in cylinder 5 is low inlet air temperature, see Figure 4. So cylinder 5 is close to misfiring and this can be seen on the high coefficient of variation of net IMEP, which is about 17%, in Figure 7. The late combustion phasing also cause lower load for cylinder 5 compared to the others, see Figure 6, and higher amount of unburned HC and CO which can be seen on the emission mean effective pressure,  $Q_{emisMEP}^1$ , in Figure 9. Despite the poor combustion in cylinder 5 it produce slightly more NO<sub>x</sub> than the other cylinders, see Figure 8. The reason for this is very unstable combustion with some early phased cycles producing a lot of NO<sub>x</sub> while most cycles just produce moderate levels of NO<sub>x</sub> close to misfiring.

Cylinder 3 with the earliest timing has however not the highest inlet air temperature but instead the second lowest, about 7°C higher than cylinder 5, see Figure 4. Cylinder 3 runs though with the second richest combustion as can be seen on the cylinder individual

lambda in Figure 5. Cylinders 3 and 2 have about the same lambda of 3.1, but the combustion phasing is slightly later in cylinder 2 due to lower inlet air temperature.

The conclusion from the first case without any cylinder balancing is that the load is about the same, see Figure 6, but the combustion phasing differs much with one cylinder running with partial burning on the border of misfiring. The poor performance of some cylinders deteriorates the whole engines performance. This can be seen in Figure 11 where the brake efficiency for case 1 is not the highest.

#### Case 2, cylinder individual combustion phasing control by the use of inlet air temperature.

In this case the inlet air temperatures are adjusted cylinder individually to achieve the same combustion phasing in all the cylinders, see Figure 3. This strategy does not mean the same inlet air temperature or the same lambda in the cylinders see Figure 4 and Figure 5. Since the supplied fuel amount and the combustion phasing are the same also about the same load, the same combustion stability, the same NO<sub>x</sub> emissions, the same unburned HC and CO, and the same maximum rate of pressure rise in the different cylinders are achieved, see Figure 6 to Figure 10. Changing cylinder individual inlet air temperature affects the combustion phasing in two ways. First higher inlet air temperature causes the auto ignition temperature to be reached earlier in the cycle and the onset of the chemical reactions. Secondly increasing inlet air temperature decrease the volumetric efficiency, i.e. decrease the amount of air supplied to the cylinder, and consequently lambda. The lower amount of dilution increases the combustion temperature and the chemical kinetics of the chemical reactions. So if a cylinder has a differing combustion phasing due to volumetric efficiency (gas exchange), compression ratio, cooling losses, or inlet air temperature it can effectively be adjusted with inlet air temperature. If the difference in combustion phasing is caused by unevenly supplied fuel amount the timing can still be compensated with inlet air temperature, but the load difference not.

The conclusion from this case is that the cylinder-to-cylinder variations can be adjusted with cylinder individual inlet air temperature to the same combustion phasing and by that equal performance. The highest brake efficiency of 24%, see Figure 11, and lowest NO<sub>x</sub> emissions, see Figure 8, are achieved in this case. The brake efficiency is 7% higher than the second highest case number 4, where the balancing is done using fuel offsets.

#### Case 3, even inlet air temperatures.

In this case the inlet air temperatures are adjusted to the same level for all cylinders, see Figure 4. The combustion phasing is however not the same as can be seen in Figure 3. The earliest and the latest combustion phasing is 0.5 and 11.5°ATDC respectively. Two

<sup>1</sup> See definition in the appendix.

cylinders, 3 and 5, are running much too early and one, cylinder 2, is running much too late. The early combustion phasing cause the maximum rate of pressure rise to be about 11bar/CAD in Figure 10 and this increases the noise level from the engine. Surprisingly the specific NO<sub>x</sub> emissions are not higher from these early cylinders than from the others in Figure 8. The late instable combustion have high coefficient of variation of net IMEP, low combustion efficiency and the lowest load, see Figure 7, Figure 9, and Figure 6. The combustion efficiency is also the worst of all the measured cylinders in all the cases due to partial burning, which can be seen on the emission mean effective pressure, QemisMEP<sup>1</sup>, in Figure 9.

The reason for the early combustion phasing in cylinder 3 and 5 is the lower air-to-fuel ratio compared to the other cylinders, see Figure 5. The reason for the late timing in cylinder 2 can not be found in inlet air temperature or cylinder individual lambda, i.e. volumetric efficiency. Compression ratio is neither differing compared to the other cylinders. The compression pressure from a motored cycle at the same conditions is 43.1bar in cylinder 2 compared to the cylinder average 43.0bar. The cooling losses for cylinder 2 should not differ from the other cylinders since the inlet air temperature and the compression pressure is about the same. If inlet air temperature in cylinder 2 is compared to the other cylinders in the other cases, see Figure 4, it has the highest temperature in all cases and higher than the chosen temperature level in case 3. The temperature level in case 3 is chosen to get a mean CA50 of all cylinders to about 6°ATDC. Another reason for the late combustion phasing in cylinder 2 might be less trapped exhaust gas residuals in cylinder 2 due to uneven gas exchange.

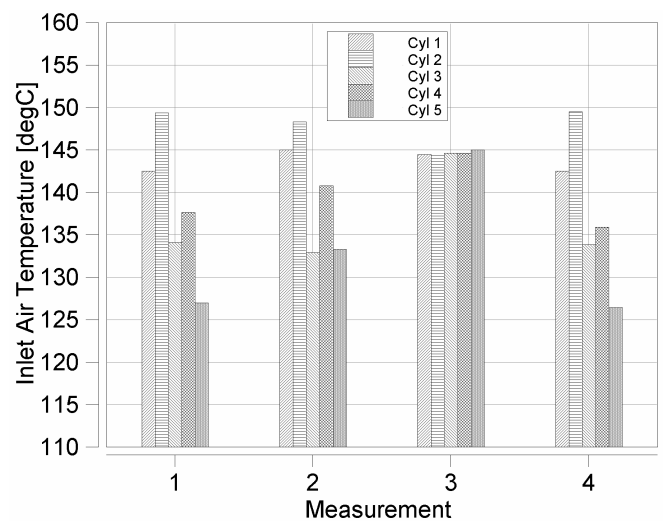
The conclusion from case 3 is that an even inlet air temperature does not give the same combustion phasing or performance in the different cylinders. This can also be seen on the poor brake efficiency, 21% in Figure 11 which is the lowest of all cases.

#### Case 4, cylinder individual combustion phasing control using fuel offsets.

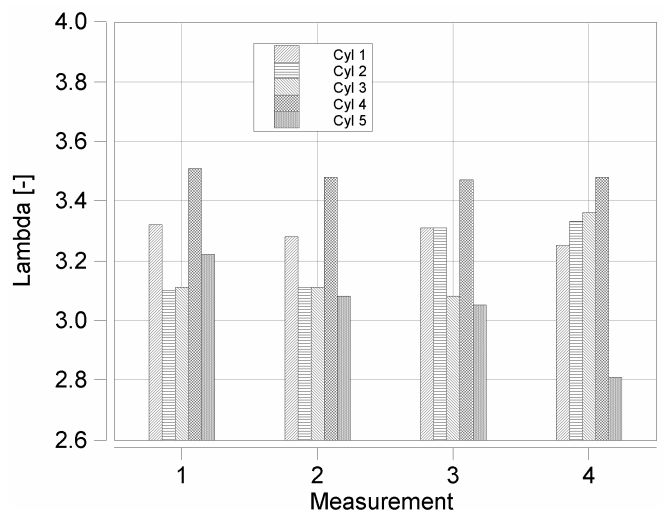
In this last case the combustion phasing is controlled using fuel offsets. So a combustion phasing offset is replaced with a load offset, see Figure 6. The differences in combustion phasing are almost eliminated but the load differences cause the cylinders to run with different performance. The high load in cylinder 5 also gives a high NO<sub>x</sub> level and high maximum rate of pressure rise, see Figure 8 and Figure 10. The reason for the high load in cylinder 5 is low inlet air temperature, see Figure 4. The temperature profile to the different cylinders is the same in case 1 and 4, where no inlet air temperature control is used. So in the uncontrolled case 1 cylinder 5 is running with a late combustion phasing instead. In the same manner cylinder 2 and 3 can be compared for the two cases. In case 1 they are running with an early combustion phasing, but in case 4 the fuel amount is decreased and consequently the load. This also

decreases the combustion efficiency increasing the unburned HC and CO emissions from the cylinders, which can be seen in Figure 9.

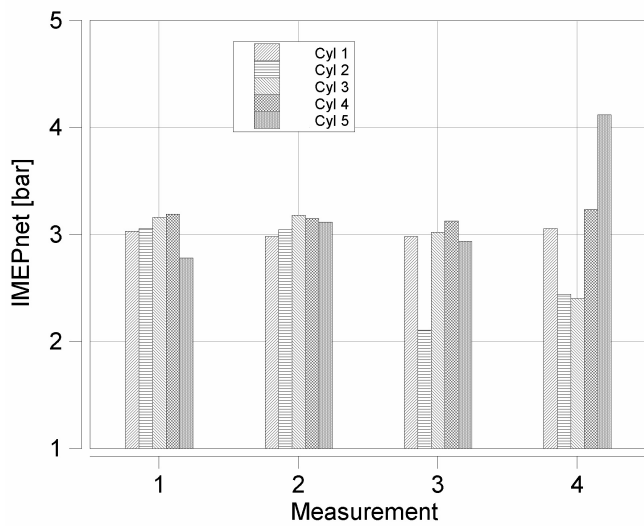
So the conclusion from this case is that fuel offsets are an efficient way to achieve the same combustion phasing in all cylinders but the introduced differences in load decrease the entire engines performance. Engine brake efficiency in this case is 22.4%, see Figure 11. Engine brake efficiency is lower compared to the best case and the high NO<sub>x</sub> emissions and high maximum rate of pressure rise in some cylinders will restrict maximum achievable load.



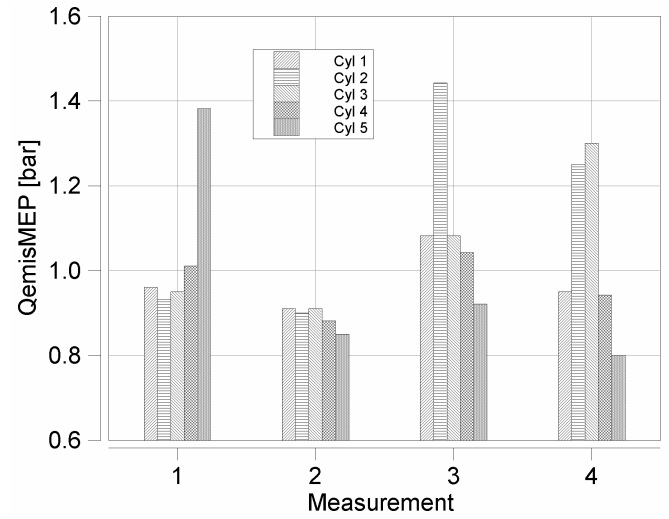
**Figure 4, cylinder individual inlet air temperature for the different cases.**



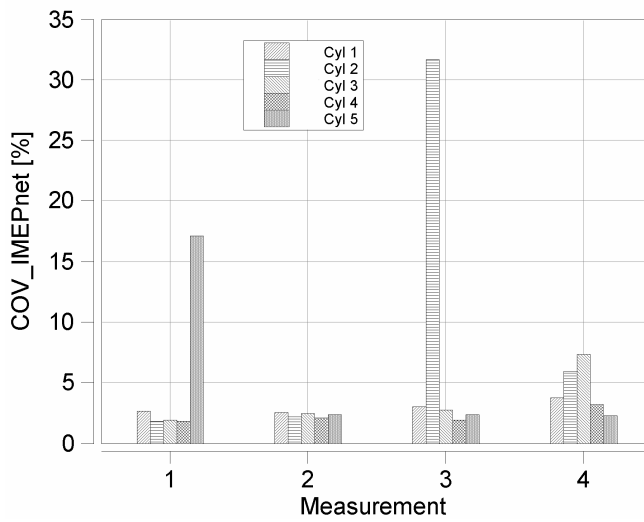
**Figure 5, cylinder individual lambda for the different cases.**



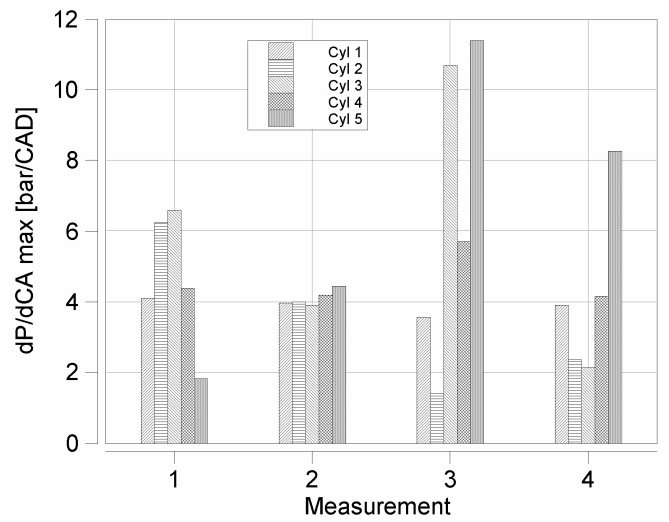
**Figure 6, cylinder individual net IMEP for the different cases.**



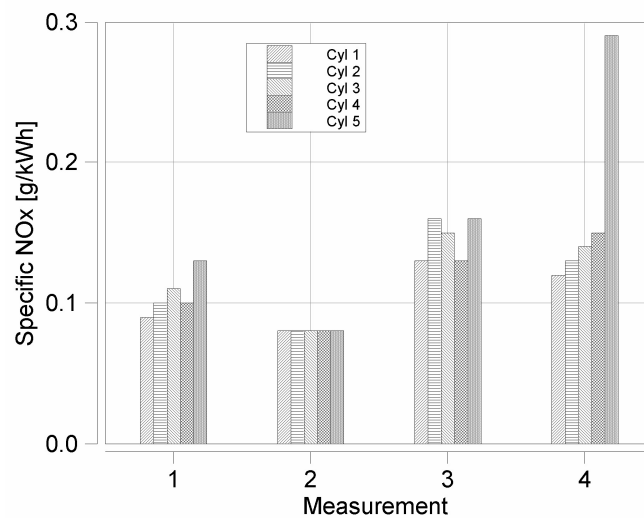
**Figure 9, cylinder individual emission mean effective pressure for the different cases.**



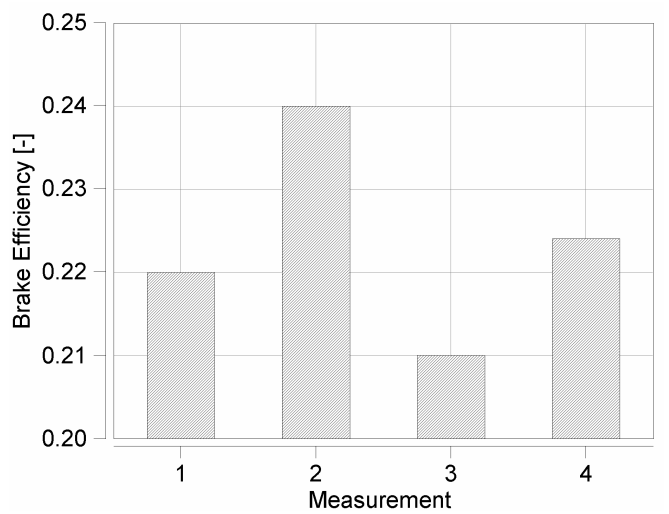
**Figure 7, cylinder individual COV IMEPnet for the different cases.**



**Figure 10, cylinder individual maximum rate of pressure rise for the different cases.**



**Figure 8, cylinder individual specific NOx emissions for the different cases.**



**Figure 11, brake efficiency for the different cases.**



### Operating range in terms of speed and load

Closed-loop cylinder balancing using fuel offsets has been the strategy used in earlier work with this engine [18, 23, and 24] where among other things the operating range in terms of speed and load was investigated. With this strategy some cylinders have been running on much higher load than others restricting the maximum achievable load of the engine. Not only can a better performance of the engine be achieved with closed-loop cylinder balancing using inlet air temperature, but also higher load. With even combustion phasing in all cylinders a later phasing is possible at maximum load decreasing maximum rate of pressure rise, i.e. noise, and exhaust out NO<sub>x</sub> emissions, and consequently making higher load possible. The performance of the closed loop combustion phasing control using inlet air temperature with a hot and a cold air stream has been investigated with this test engine [25]. The phasing control is fast and has the capability of controlling cylinder-to-cylinder variations in engine transients.

The remaining part of this paper is an effort to quantify the parameters affecting the cylinder-to-cylinder variations. All of these variations are easily handled with the chosen strategy, i.e. controlling the combustion phasing (CA50) cylinder individually and not the variation it self. The combustion phasing can be kept within  $\pm 1^\circ\text{CA}$  during steady state and  $\pm 5\text{--}7^\circ\text{CA}$  during transients.

### **PARAMETERS AFFECTING CYLINDER-TO-CYLINDER VARIATIONS**

There are different parameters affecting the HCCI combustion in the individual cylinders and causing cylinder-to-cylinder variations. The parameters are basically everything that affects the HCCI combustion and the onset of the auto ignition. HCCI is affected by the pressure and temperature history of the charge, the fuel properties, and mixture composition. Here are mainly parameters affecting the temperature and pressure history, and mixture composition investigated.

#### Fuel supply

The differences in injected fuel amount to the individual cylinders affect the load, the air-to-fuel ratio, and indirectly the cooling losses to the walls. Fuel injectors on the test engine were selected from a batch of 10 to be as equal as possible. The difference in injected fuel amount, according to the manufacturer's calibration protocol at some nominal injection duration, is about 1% between the selected injectors in the test engine compared to 3% in the whole delivered batch. However, these fuel differences have small effect on combustion phasing. If the change of fuel amount, lambda in Figure 5, and combustion phasing in Figure 3 is compared for case 1 and 4 an estimate of the fuel effect on combustion phasing can be achieved. In case 1 the fuel amount is the same to all cylinders, while in case 4 the combustion phasing is adjusted by changing the fuel amount to the individual cylinders. For example the fuel

amount is increased 25% for cylinder 5 to advance the combustion phasing, CA50, from 10.5 to 7 °ATDC. Lambda decreases 12.5% from 3.2 to 2.8. For cylinder 3 the change is in the same magnitude. Fuel decrease 17%, lambda increases 8% from 3.1 to 3.35, and combustion phasing, CA50, retard from 3.7 to 6.2°ATDC. So a difference in injected fuel amount of 1-3% affects combustion phasing less than 0.5°CA.

### Compression ratio

The maximum geometric compression ratio in the engine is 21:1, but the real efficient compression ratio is lower due to several reasons and not equal in the individual cylinders. The geometric compression ratio is calculated from nominal design values, i.e. using cylinder volume at BDC and TDC respectively. The nominal design values are however not necessarily the real values due to machining tolerances. Some of the manufacturing tolerances that affect compression ratio are listed in Table 3. The compression ratio differences in the table are calculated from the nominal 21:1 to see how much a components tolerance would theoretically affect the compression ratio. In reality the components are somewhere in between the maximum and minimum tolerance, but the table gives a figure of how much the tolerances could affect the compression ratio. A step in compression ratio between 20:1 and 19:1 affects the combustion phasing, CA50, in this engine between 5 and 10°CA, dependent of the direction of the step and the level of compression ratio [22, 25]. So the magnitude of the effect on combustion phasing from tolerances is significant. It should be noted that the trade-off between the combustion phasing change during a step in compression ratio is not linear with the level of compression ratio.

**Table 3, machining tolerances of components affecting the compression ratio.**

	<b>Tolerance max/min</b>	<b>CR min</b>	<b>CR max</b>	<b>CR diff</b>
<b>Component</b>	<b>[mm]</b>	<b>[-]</b>	<b>[-]</b>	<b>[-]</b>
Piston height	0.05/-0.05	20.8	21.2	0.5
Piston pin bearing	0.015/0.01	21.0	21.0	0.1
Conrod length	0.075/-0.075	20.7	21.3	0.7
Big end bearing	0.067/0.021	20.8	21.2	0.3
Crank pin radius	0.1/-0.1	20.6	21.5	0.9
Main bearing	0.071/0.023	20.8	21.2	0.3
Total	0.38/-0.17	19.4	21.8	2.4

The magnitude of how much a tolerance affects compression ratio and combustion phasing increases with increasing nominal compression ratio. The test engine is originally intended for SI combustion and compression ratios between 8:1 and 14:1. SI combustion is not, due to the spark ignition, so sensitive for differences in compression ratio. In this sense a CI engine, with higher nominal compression ratio and with

stricter tolerances, would have been a more favorable engine to modify for HCCI combustion.

The combustion chambers are manufactured by casting and also the casting tolerances affect the combustion chamber volume and geometric compression ratio. The measured combustion volume and compression ratio are listed for the different cylinders in Table 4. The measurement is made by filling out the pent roof of every cylinder separately with a fluid and measuring the volume needed. In practice a cylindrical plexiglas is put towards the machined parts of the cylinder head, i.e. well defined, and the enclosed volume is measured with a titration apparatus during fill up. By knowing the exact shape of the piston the compression volume can be calculated. The nominal piston dimensions and the nominal position in the cylinder are used, not the actual machined dimensions or position of the real pistons. The total accuracy of the volume measurement can be estimated to be maximum  $\pm 0.05 \text{ cm}^3$ , i.e.  $\pm 0.06$  in compression ratio. Regarding accuracy; relative differences between the cylinders and not the absolute values are interesting in this comparison. The highest and the lowest compression ratio in the different cylinders are 22.3:1 and 20.5:1 respectively. These differences definitely affect combustion phasing and originate from the casting tolerances and not from the machining tolerances.

**Table 4, measured geometric compression ratio.**

	Vc	CR	CR diff
Cylinder	[cm <sup>3</sup> ]	[-]	[-]
1	16.0	21.0	0.0
2	16.3	20.6	-0.4
3	16.4	20.5	-0.5
4	15.0	22.3	1.3
5	16.3	20.6	-0.4

To minimize the effect of geometric differences the compression ratio is adjusted with a plug in the spark plug hole, i.e. if the compression ratio is lower than the nominal the plug is inserted deeper or vice versa. The maximum cylinder pressures from motored engine cycles and corresponding compression ratios in the different cylinders are listed in Table 5. The two measurements are done before and after a series of adjustments. The compression ratios have been adjusted at an occasion earlier before these measurements, so the columns noted "before" is not comparable to the measured geometric compression ratios in Table 4. The measurements are performed without inlet and exhaust manifolds to avoid inlet and exhaust gas dynamics affecting the gas exchange. The engine speed is 2000rpm and the cooling water temperature between 82-97°C. The compression ratio is calculated from the maximum cylinder pressure assuming adiabatic compression and with a constant ratio of specific heats of 1.4, i.e. kappa for air.

The level of the maximum cylinder pressure, about 46bar abs, and compression ratio about 15.4:1 should be noted. It is far from the geometric compression ratio of about 21:1 or the maximum adiabatic compression pressure of 71bar abs. If the compression ratio is estimated with a heat release model, by assuming zero heat release from a motored cycle, a value of about 18:1 is achieved. The heat release model is a 1-zone model [28], where top land crevice effects and blow-by are modeled separately. The heat losses are calculated by using a calibrated Woschni heat transfer coefficient [28]. So the efficient compression ratio is dependent on factors as: volumetric efficiency, cylinder volume at inlet valve closing, cylinder heat losses, and blow-by. The difference in compression pressure and compression ratio between the cylinders is decreased to 0.3bar and 0.1 respectively by adjusting the plugs. The cylinder-to-cylinder difference in adiabatic compression ratio of 0.1 is not directly comparable to the earlier mentioned differences in geometric compression ratio or the effect on combustion phasing. The difference is slightly higher in geometric compression ratio and the effect on combustion phasing, CA50, is in the magnitude of 1-2°CA.

The same conclusion as earlier with machining tolerances can be made about casting, i.e. the tolerances are too big. The combustion chamber should be machined completely with strict tolerances to decrease the differences between the cylinders.

**Table 5, maximum cylinder pressures and compression ratios before and after adjustment.**

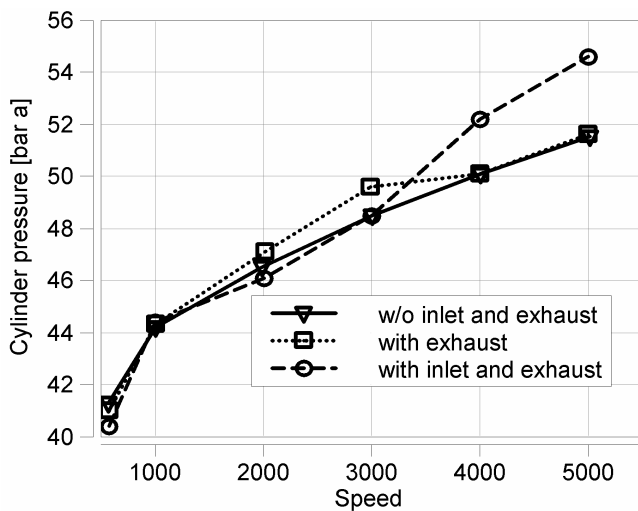
Cylinder	Pmax before	CR before	Pmax after	CR after
[-]	[bar a]	[-]	[bar a]	[-]
1	45.7	15.3	45.6	15.3
2	46.9	15.6	45.4	15.3
3	46.1	15.4	45.5	15.3
4	46.5	15.5	45.3	15.2
5	47.1	15.7	45.3	15.2
difference	1.4	0.3	0.3	0.1

#### Gas exchange, inlet and exhaust

After the adjustments of compression volume motored cylinder pressures are measured versus engine speeds with and without manifolds. This is done to find the origin of gas exchange effects on compression pressure. In Figure 12 compression pressures for cylinder 3 is presented for three cases: without any manifolds, with the exhaust manifold, and with both the inlet and exhaust manifolds. Engine speed has a significant effect on compression pressure, due to intake ramming with late inlet valve closing, decreasing heat losses, and blow-by with increasing engine speed. The cooling water temperature, i.e. the engine block temperature, increases from about 25 to 100°C as the speed increases from 570 to 5000rpm during the tests. This increase in temperature has a decreasing effect on

compression pressure due to thermal expansion of the engine components. With the same cooling water temperature the compression pressure would increase even more with engine speed than shown in Figure 12. The cooling water effect is further discussed later in this paper. Increasing temperature also decreases volumetric efficiency due to decreasing charge density, i.e. less charge is trapped in the cylinder.

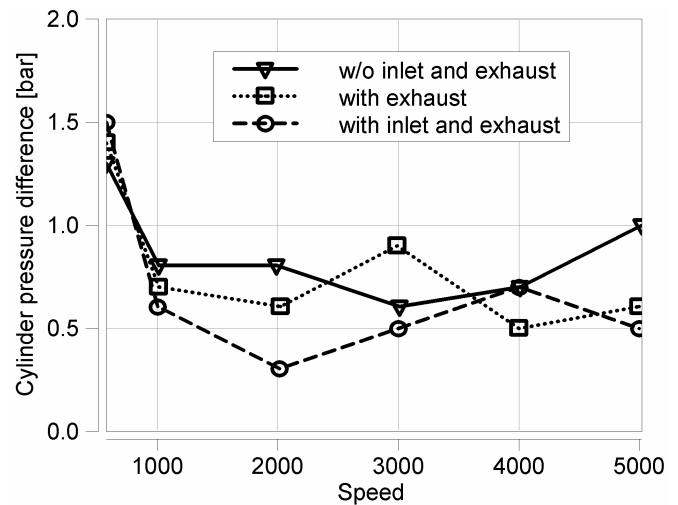
The difference in compression pressure without any manifolds and with only the exhaust manifold is small, except at 2000 and 3000 rpm, see Figure 12. The difference also varies between the cylinders due to different gas exchange dynamics in the exhaust manifold. When also the inlet manifold is added a significant tuning effect is appearing at 4000 and 5000rpm, where the compression pressure increases about 2bar, due to the gas dynamics on the inlet side. If these tuning effects on compression pressure are equal in all the cylinders there are no difficulties to handle them without cylinder individual combustion phasing control.



**Figure 12, maximum compression pressures with and without inlet and exhaust manifolds versus engine speed for cylinder 3.**

The cylinder-to-cylinder differences in compression pressure are below 1 bar for all engine speed except at the lowest measured engine speed, see Figure 13. They vary though with engine speed and the individual cylinder order from the highest to the lowest compression pressure change, i.e. the tuning effect is not equal for all the cylinders when the engine speed changes. There are several reasons to the different gas dynamics between the cylinders [29 and 30]. The reasons are mainly related to differences in geometry of the manifolds and intake port, but also cylinder individual valve events and leakage affect. A difference of 1bar in compression pressure corresponds to about 0.25 units in adiabatic compression ratio. This consequently affects the combustion phasing in the individual cylinders in the magnitude of 1-2°CA, especially during speed transients. Cylinder individual closed loop combustion control is needed.

The accuracy of the cylinder pressure measurements affects these comparisons. The absolute pressure correction of the cylinder pressure trace is done by setting the cylinder pressure trace to an average inlet manifold pressure during intake stroke. Since the gas dynamics differs between the cylinders, i.e. the cylinder pressure at inlet valve closing, the pressure referencing procedure causes an error in maximum compression pressure. However, the measurements done in Figure 13 were done without manifolds and the reference pressure is the ambient pressure. Hence, the gas dynamics can be assumed to be equal. Differences in port geometry, timing of inlet valve closing, and leakage remain. The relative error is assumed to be below  $\pm 0.05$ bar.



**Figure 13, compression pressure differences between the individual cylinders versus engine speed.**

The other effect of the gas dynamics is the amount of trapped air, which affects the air-to-fuel ratio, lambda, in the individual cylinders. Case 1 in Figure 5 shows cylinder individual lambdas with no control or adjustment of the inlet air and with even fuel amount to all the cylinders. Inlet air temperature is however not even due to uneven distribution of cold and hot air, and cooling losses in the manifold; see case 1 in Figure 4. By estimating the effect of inlet air temperature on the air density a corrected lambda value showing only the effect of the trapped air can be calculated. In Table 6 the cylinder individual lambdas are corrected to the same inlet air temperature, hence air density, as in cylinder 3. The maximum difference in lambda between the cylinders, due to the gas dynamics affecting the amount of trapped air, is 14%. As mentioned earlier in the Fuel supply section of this paper lambda has some effect on combustion phasing. The maximum lambda difference between the cylinders of 14% affects combustion phasing, CA50, about 3-4°CA. The same magnitude of effect on combustion phasing can be seen for case 3 in Figure 3 and Figure 5, e.g. comparing cylinders 3-5, with an even inlet air temperature to the cylinders. Hence, the lambda difference is due to the differences in gas dynamics. To avoid timing differences

equal gas dynamics are needed or active cylinder individual combustion phasing control.

**Table 6, the cylinder individual lambda corrected for the affect of inlet air temperature.**

Cylinder	lambda	T <sub>in</sub>	lambda corrected	lambda difference
[-]	[-]	[°C]	[-]	[%]
1	3.3	142	3.4	9
2	3.1	149	3.2	3
3	3.1	134	3.1	0
4	3.5	138	3.5	14
5	3.2	127	3.2	2

#### Inlet air temperature

Inlet air temperature affects the compression temperature and the charge air-to-fuel ratio, lambda. In Figure 3 and Figure 4 the combustion phasing, CA50, and the inlet air temperature are measured for the individual cylinders. The uneven inlet air temperatures in case 1 are due to the uneven distribution of hot and cold air and different heat losses to the walls in the different channels in the inlet air manifold. In case 1 and 2 the fuel amount to the different cylinders is equal, but in case 2 the inlet air temperature is adjusted to get the same combustion phasing in all the cylinders. To quantify the effect of inlet air temperature on combustion phasing and excluding the effect of air-to-fuel ratio, lambda, cylinder 2 and 3 can be compared between case 1 and 2 in Figure 3, Figure 4, and Figure 5. Lambda is about 3.1 in both cases for cylinder 2 and 3. Combustion phasing, CA50, is retarded from about 4 to 6°ATDC by decreasing the inlet air temperature slightly more than 1°C. This seems much. In cylinder 5 the combustion phasing is advanced about 4°CA by increasing the inlet air temperature 6°C. At the same time lambda decreases from 3.2 to 3.1 which also affect the combustion timing in the same direction, i.e. advancing. So the differences in inlet air temperature have the largest effect on combustion phasing but the exact magnitude is difficult to estimate. Clearly some unaccounted effects exist, e.g. the amount of trapped residuals. The inlet air temperature is measured about 150mm before the inlet valves by thermo couples. In this comparison only relative differences in temperature with the same thermo couple are compared, i.e. not absolute values between different thermo couples where measurement accuracy would affect. However, since the hot and cold air is mixed only about 100mm before the thermo couple there is a risk for an uneven temperature distribution in the air which could affect the measurements.

The amount of trapped exhaust gas residuals in the cylinder also affect the charge temperature, i.e. combustion phasing. If the cylinder pressures at the time for exhaust valve closing are compared between the different cases and different cylinders no clear difference in pressure level can be seen. Accurate and fast

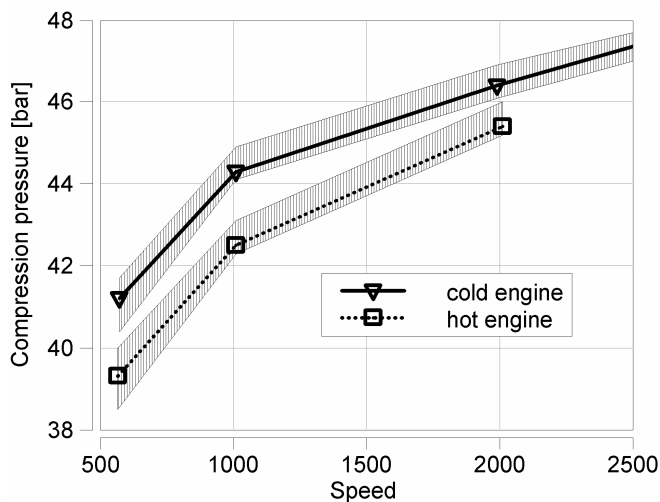
pressure measurements in the inlet and exhaust channels are needed to analyze the gas exchange more thoroughly or 1-dimentional simulation of the gas dynamics.

#### Cooling losses

Compression pressure is also dependent of the heat losses to the walls. The compression pressure for motored engine cycles for a cold and a hot engine can be seen in Figure 14. The compression pressure differences between the cylinders can be seen as a shaded area around the mean value. The cooling water temperature difference is 70°C at 570rpm and 50°C at 2000rpm. The difference in compression pressure between the hot and cold engine is about 2bar at the same engine speed. Other authors [7] have found that cooling water temperature differences between the cylinders, when the cooling water runs longitudinally through the engine, affect the heat transfer differently in each cylinder and by that the combustion phasing. In this engine the cooling water runs also longitudinally through the engine but the cooling water passages to every cylinder were optimized during the engine design to achieve about the same cooling effect in all the cylinders, i.e. the same cylinder wall temperature. The optimization has been done by simulation of the cooling water flow distribution in the engine monohead, i.e. engine block and cylinder head. Cylinder wall temperatures have not been measured in this engine to confirm results of the simulation.

The remaining difference of less than 1 bar in compression pressure between the cylinders is not only due to possible differences in heat losses but also due to gas exchange and compression ratio as discussed earlier. These differences are however easily handled by cylinder balancing with inlet air temperature. Changing cooling water temperature is not a problem with closed loop combustion phasing control since the phasing can be adjusted several orders of magnitude faster than the cooling water temperature can change.

The 2 bar higher compression pressure with the cold engine at the same engine speed can be derived to differences in thermal expansion of the engine components affecting compression pressure. The components are the engine block of aluminum, and crankshaft and connecting rod of steel. If the difference in thermal expansion between aluminum and steel is calculated, and how much it affects the compression volume, it corresponds well to the difference in compression ratio and compression pressure.



**Figure 14, compression pressure versus engine speed, including the span between the cylinders, for cold and warm engine.**

## CONCLUSIONS

An effort to quantify different parameters effect on cylinder-to-cylinder variations in combustion phasing has been done. Inlet air temperature has the largest effect on the cylinder individual combustion phasing. Differences in air-to-fuel ratio have the second largest effect and they are due to differences in inlet air temperature, gas exchange, and fuel supply. Of these, gas exchange has the largest effect on lambda, which varies up to 15% between the cylinders. This lambda difference affects the combustion phasing, CA50, about 3-4°CA. The uneven fuel supply affect combustion phasing about 0.5°CA between the cylinders. Gas exchange also affects the compression pressure and since the cylinder individual gas dynamics in this engine changes with the engine speed it affects combustion phasing about 1-2°CA. Compression ratio differs also between the cylinders due to manufacturing tolerances of engine components including the pent-roof casting. The effect from tolerances can be up to 10-20°CA in combustion phasing. If the differences in compression ratio are adjusted manually in all the cylinders the effect on combustion phasing can be decreased to about 1-2°CA. If the differences in compression ratio due to tolerances can not be removed in the engine design active cylinder balancing is needed.

Three different strategies to control the cylinder-to-cylinder variations have been investigated and compared to an uncontrolled case. The tested strategies are even inlet air temperature, closed-loop combustion phasing control using cylinder individual inlet air temperature, and closed-loop combustion phasing control using cylinder individual fuel amount. Even inlet air temperature to all the cylinders does not give even combustion phasing, since other parameters also affect the phasing. Even inlet air temperatures have the lowest brake efficiency of the compared strategies. Both closed-loop control strategies are able to achieve even

combustion phasing. However, the fuel offset strategy causes a load offset between the cylinders, and consequently different NOx emission and maximum rate of pressure rise in the different cylinders. Closed-loop control using inlet air temperatures not only has equal combustion performance in the different cylinders, it also result in the highest brake efficiency and lowest NOx emissions. Brake efficiency is 7% higher than cylinder balancing using fuel offsets. Cylinder balancing using inlet air temperature to control cylinder individual combustion phasing is an efficient way to improve engine performance in multi-cylinder HCCI engines.

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## DEFINITIONS, ACRONYMS, ABBREVIATIONS

**ATDC:** After Top Dead Center

**A/D:** Analog Digital converter

**BMEP:** Brake Mean Effective Pressure

**BBDC:** Before Bottom Dead Center

**BSFC:** Brake Specific Fuel Consumption

**BTDC:** Before Top Dead Center

**CAD:** Crank Angle Degree

**CA50:** Crank Angle for 50% burned

**CLD:** Chemiluminescence Light Detector

**CO:** Carbon Monoxide

**CO<sub>2</sub>:** Carbon Dioxide

**COV:** Coefficient Of Variation

**CR:** Compression Ratio

**DI:** Direct Injection

**dP/dCA:** Maximum Rate of Pressure Rise

**FMEP:** Friction Mean Effective Pressure

**FuelIMEP:** Fuel Mean Effective Pressure

**HC:** Hydro Carbons

**HCCI:** Homogeneous Charge Compression Ignition

**HFID:** High-temperature Flame Ionization Detector

**IMEP:** Indicated Mean Effective Pressure

**MBT:** Maximum Brake Torque  
**MON:** Motor Octane Number  
**NA:** Naturally Aspirated  
**NDIR:** Non Dispersive Infrared  
**NEDC:** New European Drive Cycle  
**NOx:** Nitrogen Oxides (NO and NO<sub>2</sub>)  
**ON:** Octane Number  
**PFI:** Port Fuel Injection  
**PCI:** Peripheral Component Interconnect local bus  
**PMEP:** Pumping Mean Effective Pressure  
**QemisMEP:** Emission Mean Effective Pressure  
**QhrMEP:** Heat release Mean Effective Pressure  
**QhtMEP:** Heat transfer Mean Effective Pressure  
**QlossMEP:** Heat loss Mean Effective Pressure  
**RON:** Research Octane Number  
**SI:** Spark Ignition  
**SVC:** Saab Variable Compression  
**UEGO:** Universal Exhaust Gas Oxygen sensor  
**VCR:** Variable Compression Ratio

## APPENDIX

Equivalent mean effective pressures [24, 27] can be used in different comparisons for parameters like the fuel heat supplied to the engine, the heat released during

combustion, the heat losses to the walls and exhaust, and the fuel energy losses due to an incomplete combustion. These parameters are converted to mean effective pressures as: Brake Mean Effective Pressure (BMEP), Indicated Mean Effective Pressure (IMEP), Pumping Mean Effective Pressure (PMEP), and Friction Mean Effective Pressure (FMEP). Here are some mean effective pressures used in this paper defined. The Fuel Mean Effective Pressure, FuelMEP, is defined as:

$$FuelMEP = \frac{m_f * Q_{LHV}}{V_d}$$

where  $m_f$  [kg] is the mass of fuel supplied per cycle,  $Q_{LHV}$  [J/kg] is the lower heating value for the fuel, and  $V_d$  [m<sup>3</sup>] is the displacement of the engine. The Heat Release Mean Effective Pressure, QhrMEP, is defined as:

$$QhrMEP = \frac{Q_{HR}}{V_d}$$

where  $Q_{HR}$  [J] is the heat released in the cylinder during an engine cycle. The Emission Mean Effective Pressure, QemisMEP, i.e. the energy lost from the cylinder as unburned in the exhaust or crankcase, is defined as:

$$QemisMEP = FuelMEP - QhrMEP$$