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

Haraldsson, Göran

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Supercharging HCCI to extend the operating range in a Multi-Cylinder VCR-HCCI engine.

Jari Hyvönen
Fiat-GM Powertrain Sweden

Göran Haraldsson and Bengt Johansson
Division of Combustion Engines, Lund Institute of Technology

ABSTRACT

The operating range in terms of speed and load for a natural aspirated Homogenous Charge Compression Ignition (HCCI) engine is restricted by a lack of dilution at high load and a poor combustion at low load. Today HCCI is seen as a part load concept, but the operating range for vehicle applications should at least cover contemporary driving cycles, without switching to a conventional Spark Ignited (SI) combustion mode. Dilution with air at high load can be increased by supercharging, but with the drawback of parasitic losses or pumping losses decreasing the brake efficiency and by that one of the benefits of HCCI.

The effect of advancing the combustion phasing and throttling the inlet air on the combustion efficiency at zero loads is investigated. The combustion efficiency increases drastically in both cases. The increase in the combustion efficiency overcomes the drawbacks of the early combustion phasing in the first case and the pumping losses in the second case.

The HCCI operation range with both mechanical supercharging and simulated turbocharging is investigated and compared with a natural aspirated SI with gasoline as fuel. The operating range can be more than doubled with supercharging and higher brake efficiency than with a natural aspirated SI is achieved at the same loads. Mechanical supercharging is however not an option with HCCI combustion, due to the high parasitic losses.

The test engine is a 5 cylinder in-line engine with Variable Compression Ratio (VCR) and displacement of 1.6L.

INTRODUCTION

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

The first efforts to characterize the 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 the 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. the high efficiency and the low NOx emissions compared to SI engines. Since then a lot of research effort has been put on understanding the HCCI combustion and how to control it.

HCCI combustion is achieved by compressing a premixed, highly diluted, air/fuel mixture to self-ignition. A highly diluted mixture is needed to decrease the burn rate and avoid too high combustion noise. A controlled auto ignition can be achieved by controlling the temperature and the pressure history, the fuel properties, or the mixture composition in the cylinder [6-19]. The mixture temperature can be controlled by the inlet air temperature, the compression ratio, or retaining hot residuals in the cylinder. The fuel properties, i.e. ignitibility, can be changed with dual fuel systems or fuel reforming, i.e. injecting the fuel during the negative valve overlap. The mixture composition can be affected by stratification in Direct Injection (DI) systems. The HCCI combustion in multi-cylinder engines [6, 7, 13, 18, 21, and 22] is an additional challenge due to cylinder-to-cylinder variations.

Today HCCI is seen 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. by too much dilution, and by a fast burn rate that induces noise and NOx 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 the combustion efficiency decreases. By advancing the combustion phasing more time is available at high cylinder pressures and temperatures to complete the oxidization, i.e. increasing the combustion efficiency. Inlet air throttling to decrease dilution is another possibility to increase the combustion efficiency, but with the drawback of increasing the pumping losses, i.e. one of the reasons for HCCI in the first place. The effect of the combustion phasing and the inlet air throttling on combustion efficiency is tested in this paper.

High loads up to 16bar BMEP can be achieved with turbocharging [12]. However because of the low exhaust gas temperature a small turbine is needed to achieve a high enough boost pressure. The drawback is an increased turbine backpressure and pumping losses. The option to a supercharged HCCI is a combustion mode switch to natural aspirated SI. Supercharged HCCI, both mechanical and turbocharged, are compared with a natural aspirated SI in this paper.

EXPERIMENTAL APPARATUS

The test engine in Figure 1 is a five-cylinder 1.6L Saab Variable Compression (SVC) engine [20]. The VCR range is between 9:1 and 21:1. The VCR mechanism has been presented in earlier HCCI papers with this engine [18, 21, 22] and by Saab [20]. Exhaust gas residuals are not used to enhance the HCCI combustion in this paper, i.e. SI valve timings are used. Some basic engine specifications can be seen in Table 1.

Table 1, engine specifications.

Displacement	1598 cm^3 (320 cm^3/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, in Figure 2, consist of an air filter, a mechanical supercharger, a Charge Air Cooler (CAC), an electrical air heater, an exhaust to air heat exchanger, an oxidizing catalyst, and some throttle valves for controlling the air flows and the exhaust flow. The engine can be run natural aspirated or mechanically supercharged. In the simulated turbocharger test an external compressor is used for air supply and the turbocharger turbine back pressure is simulated by an exhaust throttle valve. The air temperature can be adjusted by cooling in the CAC or heated by exhaust gases in a heat exchanger. The electrical air heater is only used during the start up to achieve the HCCI combustion and turned off as the combustion starts. The mixing of hot and cold air controls the inlet air temperature as soon as the engine has warmed up sufficiently.

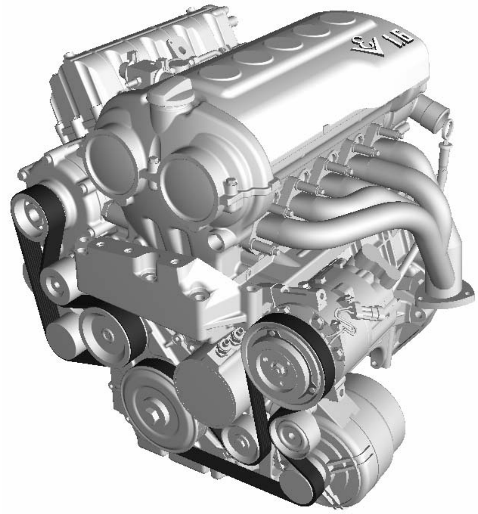

Figure 1, the Saab Variable Compression test engine.

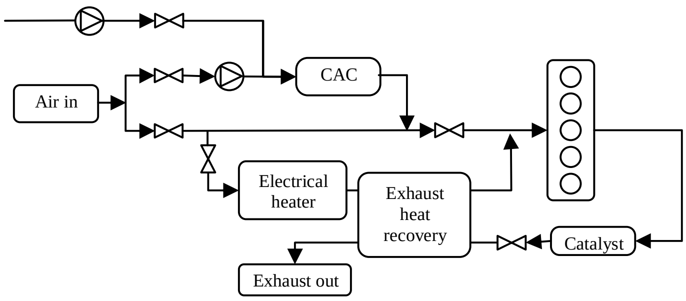

Figure 2, flow chart of the inlet air and the exhaust gas.

Water-cooled cylinder pressure sensors from Kistler, model 6043A, are fitted in all cylinders. The spark plugs are removed and the holes plugged. The cylinder pressures are sampled with an A/D-converter, a Wavebook 516 with a 8-channel simultaneous sample and hold from IOtech, capable of an aggregate rate of 1Msamples/s. The sample rate is 2.5 samples per CAD per cylinder for engine speeds below 3000 rpm and 1.25 samples per CAD per cylinder above 3000 rpm. The regulated emissions are measured with a Horiba exhaust gas analyzer MEXA-8120F. The CO and CO_2 analyzer, a Horiba model AIA-23, works with Non-Dispersive Infrared (NDIR) technology. The NO_X ($NO+NO_2$) analyzer, a Horiba model CLA-53M, uses a Chemiluminescence Light Detector (CLD) to measure the chemiluminescence from the reaction between NO and ozone (O_3). HC emissions are measured with a High-temperature Flame Ionization Detector (HFID), a Horiba model FIA-22-2, as ppm Carbon. The O_2 analyzer, a 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 which 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 [23]. The broadband lambda sensor is used for quick screening during the test and to measure the relative cylinder individual lambda differences. Fuel consumption is measured by weighing. The regulated emissions are presented as emission index [g/kg_{fuel}] for the zero load cases, where brake specific emissions can not be calculated, and as brake specific [g/kWh] for all the other cases.

In-house developed software is used for engine control and cylinder pressure monitoring. Various variables are calculated on-line for control or engine protection, such as net IMEP, COV (IMEP), Crank Angle of 50% heat release (CA50), peak cylinder pressure, and maximum pressure rise rate (dP/dCA). The combustion phasing is controlled with the inlet air temperature, the compression ratio, and the fuel amount. The amount of fuel to the engine can be controlled cylinder-individually by e.g. a cylinder-individual combustion phasing control.

The inlet air pressure, the inlet air temperature, and the CR are used for control, and are measured by using 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 different compression ratios. All variables can be saved for post processing, e.g. for heat release analysis. Other variables, such as the temperatures and pressures of air, exhaust, water, and lubricating oil, are logged with a HP-logger at a rate of approximately 0.33Hz.

RESULTS AND DISCUSSIONS

TEST PROCEDURE

Fuel

A gasoline corresponding to U.S. Unleaded Regular is used as fuel in the HCCI tests. The fuel has a RON value between 91 and 92, and a MON value between 81.5 and 82.5. The gasoline in the SI tests, which are used as comparison, is commercial gasoline with a RON value of 98.

Cylinder-to-cylinder variations

The cylinder-to-cylinder variations in the combustion phasing that exist without any balancing is adjusted with fuel offsets to the different cylinders [21]. Instead a cylinder-to-cylinder difference in load is introduced, e.g. the difference between the maximum to minimum load is about 8% at 5000rpm.

Operating range

The operating range in terms of speed and load is tested. The measurement matrix is divided into speeds from 1000 to 5000 rpm with 1000 rpm steps and a load from a minimum to a maximum with 1bar BMEP steps. Negative loads are not tested, i.e. BMEP below 0bar. The limits for the operating range are defined by some chosen variables. The limiting variable at the high load is NOx. Maximum 15 ppm NOx (dry at actual O_2 level[1]) was chosen as the limit during the test run corresponding to about 1 g/kg_{fuel}. In some of the maximum load points NOx is however higher, because NOx increase rapidly at increasing load at the limit, so a small change in load causes a large change in NOx. The limits for the minimum load are misfiring, defined as COV (IMEPnet) exceeding 10% for an individual cylinder, or an unburned HC level of 5000 ppm-wet at actual O_2 level. The combustion phasing is adjusted with the compression ratio and the inlet air temperature. At low load the inlet air temperature is allowed to be as high as possible using an exhaust heat to inlet air heat exchanger, i.e. the electrical air heater is not used during the tests.

COMBUSTION EFFICIENCY AT LOW LOAD

The combustion efficiency with HCCI at a low load can be poor, i.e. high CO and HC emissions, as has been shown earlier with this test engine [22]. The reason is the lean combustion without throttling. The chemistry is too slow and the supplied fuel is only partly oxidized to CO. A high inlet air temperature is one way to increase the combustion efficiency, affecting as thermal throttling, i.e. decreasing the amount of air supplied to the cylinder and by that air/fuel ratio speeding up the chemistry. Two different approaches to increase the combustion efficiency are presented in this paper. The first is by an advanced combustion phasing and the second is by inlet air throttling. With an advanced combustion phasing the time available for oxidizing the fuel at high pressure and temperature is increased. With inlet air throttling the air/fuel ratio is decreased increasing the speed of chemistry. The combustion efficiency is calculated from the incomplete combustion products in the exhaust, measured by the exhaust gas analyzer, and compared with the supplied fuel heat [27]. The combustion efficiency is defined as:

$$\eta_{combustion} = \frac{\Sigma m_i * Q_{LHV,i}}{m_f * Q_{LHV,f}}$$

where m_i [kg] is the mass of each unburned product in the exhaust, m_f [kg] is the mass of fuel supplied per cycle, $Q_{LHV\ i}$ [J/kg] is the lower heating value for each

[1] Not referenced against a fixed percentage O_2 in the exhaust.

unburned product in the exhaust, and $Q_{LHV\,f}$ [J/kg] is the lower heating value for the fuel.

Advancing the combustion phasing at 0bar BMEP

The combustion phasing is controlled by changing the compression ratio and the inlet air temperature. The inlet air is heated with the exhaust gases after the catalyst, so the available inlet air temperature is dependent on the exhaust gas temperature after the cylinders and the amount of unburned hydrocarbons oxidized in the catalyst. Advancing the combustion timing decreases the exhaust gas temperature decreasing the inlet air temperature. The increasing combustion efficiency decreases the amount of hydrocarbons oxidized in the catalyst decreasing the exhaust heat after the catalyst and the inlet air temperature. So a balance between these parameters is needed at the chosen operating point.

The combustion efficiency is plotted versus the combustion phasing, CA50, in Figure 3. The test is done at two engine speeds, 1000 and 2000rpm, at zero load, i.e. 0bar BMEP. Negative loads are not tested. The combustion efficiency increases with an earlier combustion phasing at 2000rpm, but the trend is unclear at 1000rpm. When the timing is advanced in the 1000rpm case the catalyst eventually go out. The exhaust temperature becomes too low and the mass flow of unburned hydrocarbons is too low to sustain a high enough oxidizing temperature in the catalyst. At 2000 rpm the combustion efficiency increases when combustion timing is advanced. The big step between the combustion phasing 4°ATDC and 1.5°BTDC is due to the difficulties in finding a stable operating point without losing catalyst oxidation.

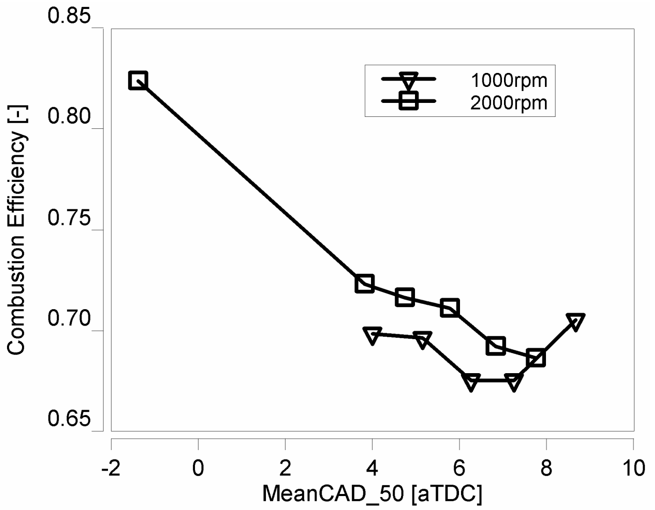

Figure 3, the combustion efficiency versus the combustion phasing at zero load.

Figure 4 shows the exhaust gas temperature versus the combustion phasing. At late phasing exhaust gas temperature increases over the catalyst in both cases, due to HC and CO oxidation, but with early phasing oxidation is taking place in the catalyst only in the 2000rpm case. In both cases fuel consumption decreases slightly as the combustion phasing is advanced, even at the over advanced phasing in the 2000rpm case, indicating that the combustion timing was too late in the first place. The fuel consumption decreases 8% in the 1000rpm case and 16% in the 2000rpm case. The conclusion is that advancing the combustion phasing is beneficial as long as the HC and CO oxidation in the catalyst take place and a zero load is maintained.

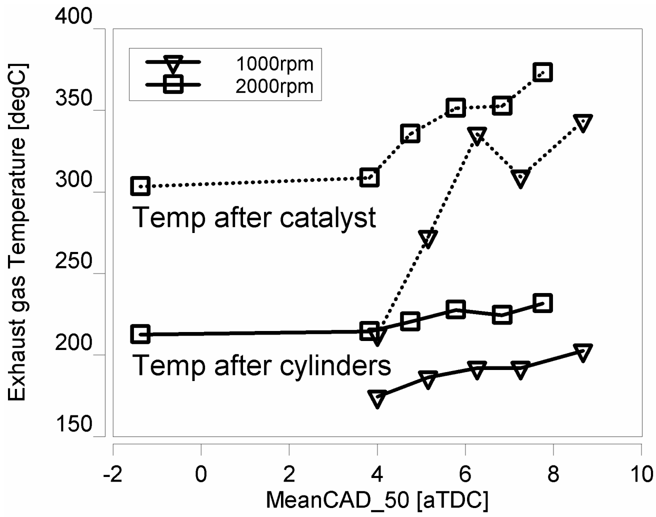

Figure 4, the exhaust gas temperatures after the cylinders and after the catalyst versus the combustion phasing at zero load.

Throttling of inlet air at 0bar BMEP

One of the main benefits with HCCI combustion compared to SI combustion is that the throttling of the inlet air is not needed, hence there are no pumping losses. In this test throttling is used to decrease the dilution of the air/fuel mixture and to increase the kinetics of the chemistry to increase the combustion efficiency. Figure 5 shows the combustion efficiency and the gas exchange efficiency plotted at different inlet air pressures. The gas exchange efficiency in Figure 5 is defined as:

$$\eta_{gasexchange} = 1 - \frac{PMEP}{IMEP_{gross}}$$

where PMEP is Pumping Mean Effective Pressure and IMEPgross is gross Indicated Mean Effective Pressure, i.e. the mean effective pressure during compression and expansion strokes only. Throttling increases the combustion efficiency, but at the same time also the pumping losses increase and the gas exchange efficiency decreases. The increase in the combustion efficiency is 23% in the 1000rpm case and 26% in the 2000rpm case. Both engine out CO and HC decrease, see Figure 6, but mainly CO. NOx increase from about 0.2 g/kg_{fuel} to 0.35 g/kg_{fuel} at 2000rpm, but decrease slightly in the 1000rpm case. Less dilution increases the combustion temperature and decreases the depth of the boundary layers [25] where CO and HC origins.

The limiting factor for how low inlet air pressure can be used is the initiation of the combustion, since compression ignition is used. A lower inlet air pressure decreases compression pressure and has to be compensated with a higher compression ratio to maintain the combustion phasing. The inlet air temperature was again kept as high as possible in these tests, i.e. dependent on the amount of the heat available in the exhaust after the catalyst. The catalyst did not go out in any measurement point. In the 1000rpm case the compression ratio is increased from 16:1 to 21:1 and the combustion phasing, CA50, is kept at about 5°ATDC. In the 2000rpm case a maximum compression ratio of 21:1 is used all the time and the combustion phasing, CA50, retarded from 0°ATDC to 6°ATDC. The limit for throttling is reached when the combustion can not be initiated. The combustion stability, however, increases with more throttling due to less dilution.

An optimum between pumping losses and unburned HC/CO losses can be found. In Figure 7 the product of the combustion efficiency and the gas exchange efficiency, and the fuel flow is plotted versus the inlet air pressure. The maximum for the product of the combustion efficiency and the gas exchange efficiency, and the fuel consumption do not coincide. The trend for the fuel consumption decreases slightly with decreasing the inlet air pressure in both cases. The reason in the 2000rpm case is a more favorable combustion phasing with a decreasing inlet air pressure, i.e. the indicated efficiency increases.

Inlet air throttling is a way to increase the combustion efficiency, by decreasing the level of dilution, without increasing fuel consumption at zero loads. The introduced pumping losses are compensated with the improvement in the combustion efficiency. The limiting factors are the means available for initiating the HCCI combustion, i.e. the compression ratio and the inlet air temperature with this engine.

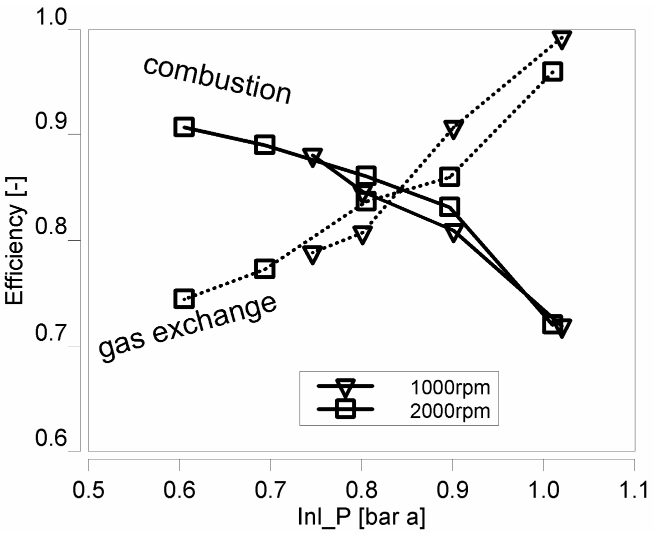

Figure 5, the combustion efficiency and the gas exchange efficiency versus the inlet air pressure at zero load.

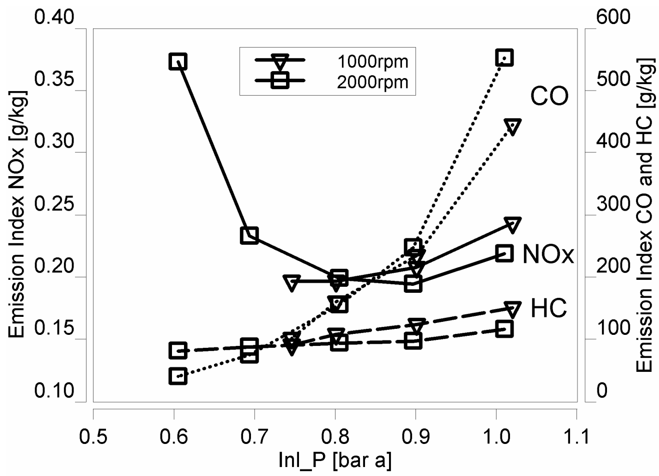

Figure 6, the emission index [g/kg$_{fuel}$] for NOx, CO, and HC versus the inlet air pressure at zero load.

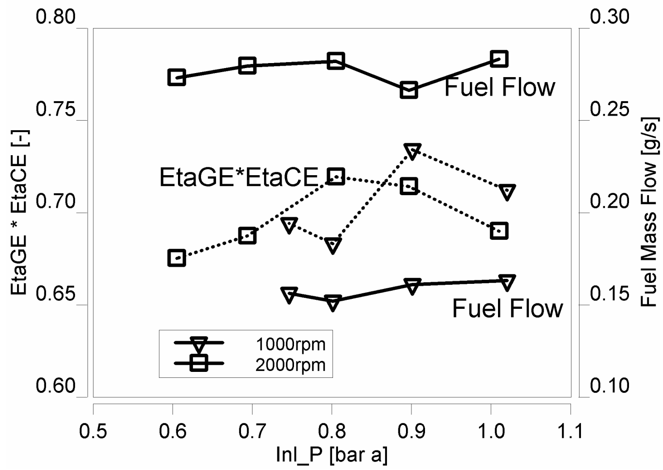

Figure 7, the fuel flow and the product of the combustion efficiency and the gas exchange efficiency versus the inlet air pressure at zero load.

Oxidizing catalyst

The oxidizing catalyst used in the tests presented in this paper is the main catalyst from a Saab 9-5 V6 diesel. A diesel catalyst was chosen because only oxidation of CO and HC take place and because of the low exhaust gas temperatures with HCCI combustion. A diesel catalyst has a lower light off temperature than a three way catalyst for SI engines. Figure 8 and Figure 9 show brake specific CO and HC emissions before and after the catalyst versus engine speed at BMEP 2 bar. The CO conversion efficiency is above 99% for all speeds except 2000rpm where it is 97%, se Figure 8. The HC conversion efficiency decreases from 99% at 1000rpm to 87% at 5000rpm, se Figure 9. The reason for the HC slip through at high exhaust mass flows is because of mass transport limitations of HC to the catalyst surface. The catalyst volume should be increased to decrease the HC slip at high engine speeds, e.g. to use also the pre catalyst from the diesel engine.

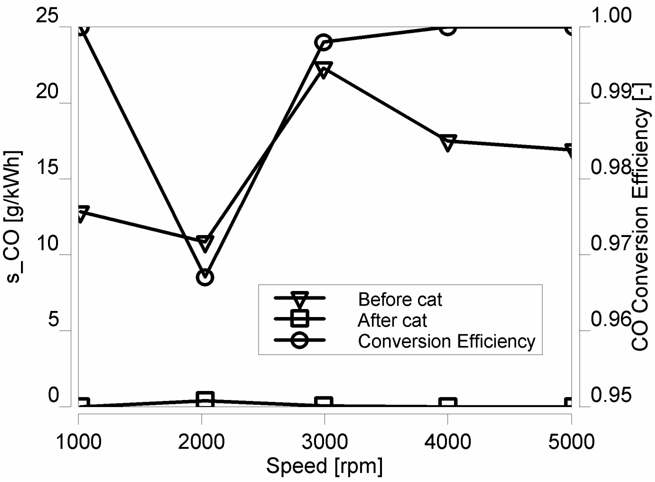

Figure 8, specific CO emissions before and after the catalyst versus the engine speed at 2 bar BMEP.

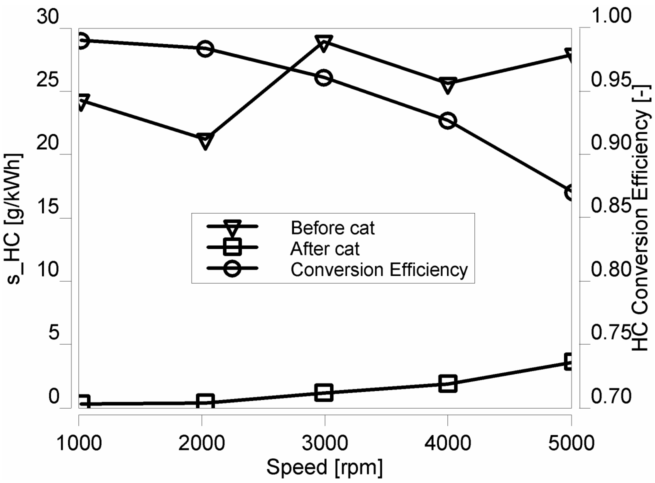

Figure 9, specific HC emissions before and after the catalyst versus the engine speed at 2 bar BMEP.

OPERATING RANGE

The operating range with the HCCI combustion is measured in three different cases. The first is natural aspirated, the second is supercharged with a mechanical Lysholm compressor, and the third is with a simulated turbocharger. The reason to simulate a turbocharger instead of testing a real one is the difficulty to estimate an appropriate turbocharger specification, i.e. the size of the turbine and the compressor. The cold exhaust gas temperatures with the HCCI combustion compared to the SI or CI engines acquire a smaller turbine area.

At earlier HCCI NA tests with this test engine [22] the operating range was limited by NOx, due to the lack of dilution. In this paper the operating range is extended with supercharging and the two different supercharging methods are compared. The test engine is equipped with a mechanical supercharger of Lysholm type, i.e. it has an internal compression, in its original SI setup. It is used for increasing the output in a downsized SI concept with VCR. The drawback with mechanical compressors is the power they need, i.e. the parasitic losses. In the SI concept this can be accepted because of the short time the compressor is used in normal vehicle applications. In the HCCI case the compressor would be used to cover the operating range in a normal drive cycle, i.e. it would be used more than in the SI case. The drawback with turbocharging is the high exhaust back pressures the turbine causes, i.e. the pumping losses. The low exhaust gas temperature from the HCCI combustion is the reason why high exhaust gas pressures are needed by the turbine to drive the compressor [12]. The two supercharging concepts to increase HCCI operating range are compared in the end of this paper with the option of switching the operating mode to SI.

<u>Operating range with a natural aspirated HCCI</u>

The operating range with NA HCCI can be seen in Figure 10. Negative loads below 0 bar BMEP are not tested. NOx emission at the limit for maximum load is about 15 ppm NOx, i.e. about 1g NOx/kg$_{fuel}$. The maximum BMEP achieved decreases with engine speed due to the lack of dilution. The inlet air temperature has to be increased to maintain the combustion phasing at increasing speed, but at the same time the level of dilution decreases and the load has to be decreased. The maximum measured rate of pressure rise is 9bar/CAD. The highest measured COV IMEPnet is about 8%, but generally below 5%. The lowest measured combustion efficiency is 87%.

The natural aspirated operating range does not cover e.g. the European NEDC drive cycle. The degree of the drive cycle that is covered depends on the size of the car used. To cover a larger operating range more dilution at high load is needed or a fuel with a lower octane number [18].

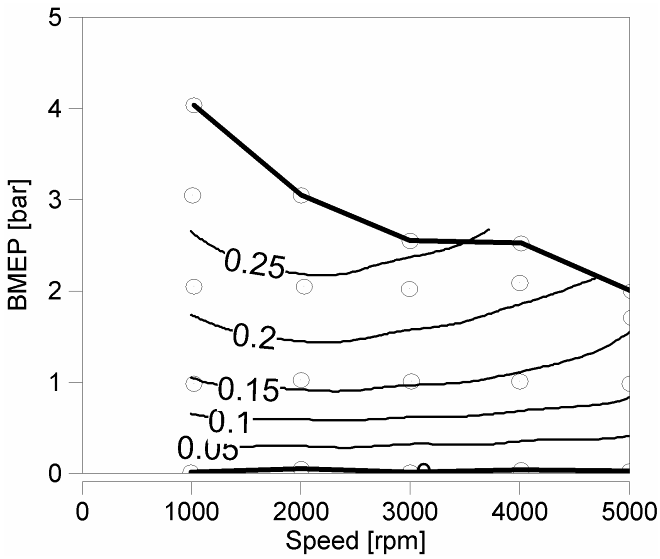

Figure 10, the operating range with a NA HCCI combustion and iso-lines for the brake efficiency.

Operating range with a mechanical supercharger

The Lysholm compressor is belt driven from the engine crankshaft. The compressor also has a gearbox between the belt and the compressor itself. Running the compressor causes a parasitic loss to the engine. Besides the extra air the compressor supplies to the engine it also produces a positive pumping work, since the exhaust back pressure is lower than the boosted inlet air pressure. Figure 11 shows the operating range for the mechanical supercharged HCCI combustion including iso-lines for inlet air pressure. The lower line in the operating range is the full load curve for NA HCCI. A higher engine speed than 4000rpm could not be tested with this test engine setup due to the high exhaust backpressure caused by the exhaust to the inlet air heat exchanger. The heat exchanger is not made for pressurized operation and at high mass flows at high engine speeds the exhaust side of the heat exchanger throttles the flow too much.

The maximum load from 1000 to 3000rpm increases between 50 to 100% with the compressor. The limiting factor for a higher load at these engine speeds is the maximum cylinder pressure. The maximum cylinder pressure allowed for the engine is 125bar and the highest cylinder-individual pressure measured is 118bar. The reason for the high pressures is the need for dilution. Lambda is between 2.4 and 2.8 in the whole operating range. The compression ratio has to be decreased at high load to keep the cylinder pressures below the maximum allowed, but at the same time the inlet air temperature is increased to maintain the combustion phasing. This on the other hand decreases dilution and has to be compensated by higher inlet air pressure that increases compressor work, i.e. parasitic losses. A higher maximum cylinder pressure would decrease the parasitic losses.

Another drawback with supercharging is the noise, but in a supercharged operation the audible noise at a certain rate of pressure rise is lower than at the corresponding natural aspirated operation [26]. The maximum rate of pressure rise is between 8 and 17 bar/CAD in the operating range and the rapid burning angle, 10-90% burned, is about 6-8CAD. Despite the high cylinder pressures and the rate of pressure rise NOx emissions are still low. The maximum measured NOx is 0.63g/kWh, 2.1g/kg_{fuel}.

Figure 12 shows the net IMEP iso-lines in the operating range. A high net IMEP, up to 11.1bar, is measured with supercharging. Most of this work is however lost to drive the mechanical compressor, see the comparison with turbocharging later in this paper.

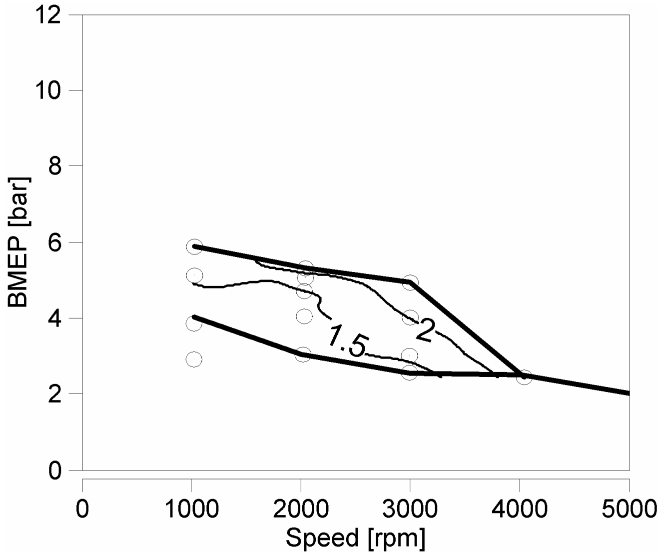

Figure 11, the operating range with the mechanical supercharger and iso-lines for the inlet air pressure [bar a].

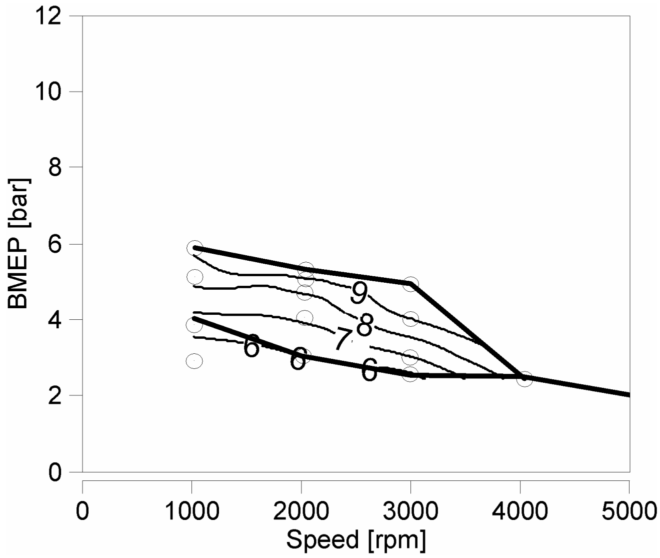

Figure 12, the operating range with the mechanical supercharger and iso-lines for the net IMEP [bar].

Operating range with a turbocharger

The turbocharger simulation is done by an external compressor and by throttling the exhaust. The simulation is done as follows. A desired inlet air pressure for the operating point in question is set with a pressure regulator. The exhaust backpressure from the turbine is calculated from the total turbocharger efficiency assumed, the inlet air pressure and temperature, and the exhaust gas temperature. The exhaust gas pressure is set with a throttle valve. If some of the used parameters change the procedure is repeated. The total turbocharger efficiency is assumed to be 42% in the whole operating range to make it more convenient, i.e. 70% isentropic compressor efficiency and 60% turbine efficiency. Static and not total pressures and temperatures are used in the calculation. The effect of the exhaust blow down pulses is not taken into account, more than by the turbine efficiency itself. The assumed turbocharger efficiency should at least not be overestimated in the calculations.

The maximum load versus engine speed with turbocharging compared to mechanical supercharging and NA mode is shown i Figure 13. Higher engine speeds than 4000rpm could not be tested due to the same reason as with mechanical supercharging, i.e. too high backpressures from the heat exchanger. The engine load with turbocharging increases between 100 and 177% compared to NA operation. Compared with mechanical supercharging the increase in load is between 32 and 97%. The limiting factors for higher load are the same with turbocharging as they are for mechanical supercharging, i.e. maximum cylinder pressure. The compression ratio is also decreased at high load to avoid too high cylinder pressures.

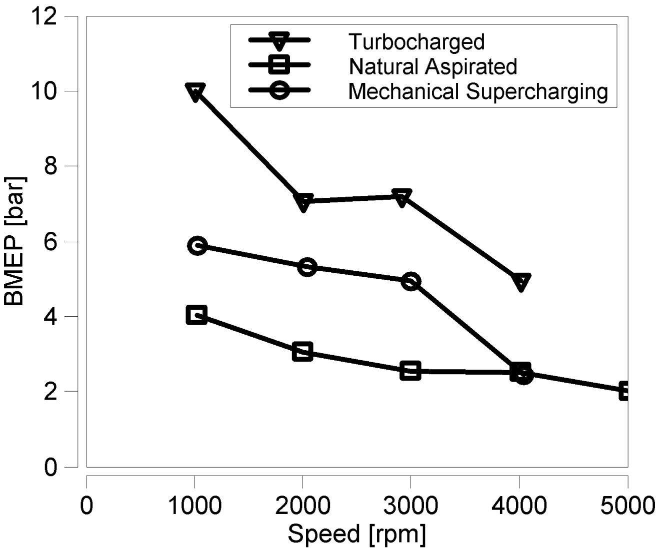

Figure 13, the maximum load versus the engine speed for the turbocharged, the mechanically supercharged, and the natural aspirated HCCI combustion.

The turbocharger operating range and the maximum rate of pressure rise are shown in Figure 14. The highest measured rate of pressure rise is 18.5bar/CAD, causing noise. With a higher maximum cylinder pressure a higher compression ratio could be used to decrease the rate of pressure rise due to a faster expansion. NOx is about 0.1 g/kWh, 0.2-0.5g/kg$_{fuel}$, in the turbocharged operating range except at the maximum load at 4000rpm where 0.57g/kWh, 1.9g/kg$_{fuel}$, is measured. The negative pressure difference between the inlet and the exhaust is between 0.4 and 1.2bar to drive the turbine causing pumping losses, see later in this paper.

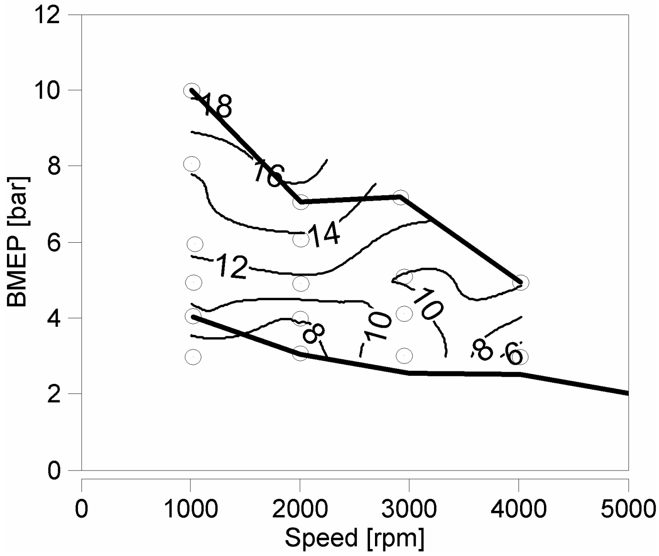

Figure 14, the operating range with the turbocharger and iso-lines for the maximum rate of pressure rise [bar/CAD].

A mechanical supercharger compared with a turbocharger

Equivalent mean effective pressures [24] are used in the comparisons between the mechanical supercharger and the turbocharger for parameters like the fuel heat supplied to the engine, the heat released during combustion, the heat losses, 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). The new mean effective pressures are as follows. 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^3] 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:

$$QemisMEP = FuelMEP - QhrMEP$$

Heat Transfer Mean Effective Pressure, QhtMEP, i.e. the heat transfered to the cylinder walls, is defined as:

$$QhtMEP = \frac{Q_{HT}}{V_d}$$

where Q_{HT} [J] is the heat transfered to the cylinder walls during an engine cycle. The Heat Loss Mean Effective Pressure, QlossMEP, i.e. the apparent heat lost in the exhaust gases, is defined as:

$$QlossMEP = QhrMEP - IMEPgross - QhtMEP$$

QlossMEP and QhtMEP are not calculated separately in this paper. Another way to present these parameters would be as percentage of supplied fuel energy, i.e. like in a heat balance or the Sankey diagram.

BMEP is plotted versus FuelMEP, i.e. the ratio of the two corresponds to the brake efficiency, in Figure 15. A higher brake load is achieved with the turbocharger than with the mechanical supercharger for the same supplied fuel amount, i.e. a higher brake efficiency.

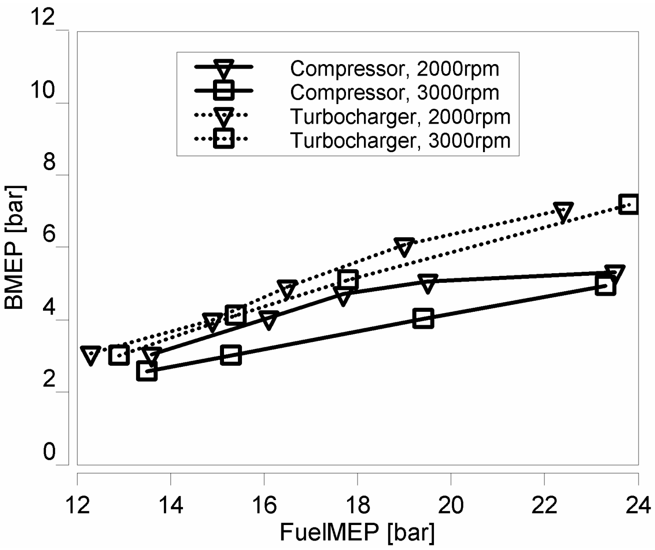

Figure 15, BMEP versus FuelMEP for the mechanical supercharger compared with the turbocharger.

The main reason for the poor performance of the mechanical supercharger is the parasitic loss to drive the compressor, see Figure 16. FMEP is more than 67% higher at 2000rpm and 124% higher at 3000rpm. The higher parasitic loss with mechanical supercharger at 3000rpm is caused by the increased friction. FMEP and IMEPnet with mechanical supercharger increase drastically between the two highest load points at 2000rpm. The indicated load increases with 2.2bar while the brake load only increases 0.25bar. Since the maximum load is limited by the maximum cylinder pressure the compression ratio decreases with the increasing load. The lower compression ratio is compensated with a higher inlet air temperature to keep the right combustion phasing, but at the same time the amount of dilution decreases and the inlet air pressure has to be increased to compensate the higher temperature. In this case almost the entire increase in indicated load goes to drive the compressor at the new operating point, but with poor brake efficiency, as can be seen in Figure 15. FMEP without compressor is about 1.25bar with this engine at 2000rpm and up to 2bar BMEP, so running the compressor always gives poor brake efficiency.

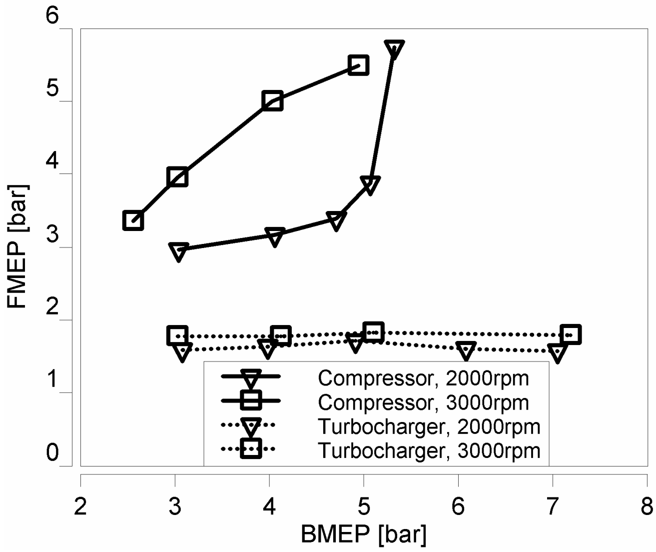

Figure 16, FMEP versus BMEP for the mechanical supercharger compared with the turbocharger.

Pumping losses are presented in Figure 17. For the case with the mechanical supercharger a positive pumping work is produced, i.e. negative PMEP, due to the positive pressure difference between the inlet and the exhaust. Some of the parasitic loss is recovered as positive pumping work. The drawback with turbocharging with poor turbocharger efficiency and low exhaust gas temperature is however seen as a pumping loss.

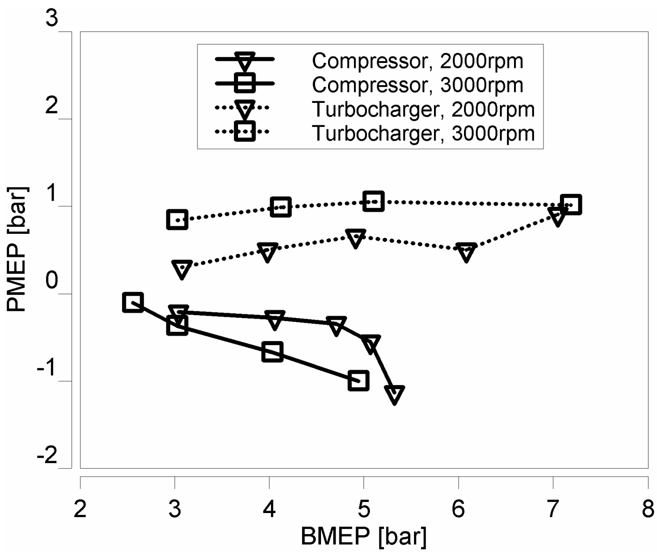

Figure 17, PMEP versus BMEP for the mechanical supercharger compared with the turbocharger.

The total heat losses, both apparent in the exhaust and the heat transfer to the cylinder walls, are presented in Figure 18. The absolute heat losses are larger for the mechanical supercharger than for the turbocharger since QhrMEP and IMEP are higher for the same load. If the heat losses are compared as percentage of supplied fuel energy the loss is about the same, 50-55%, in both cases.

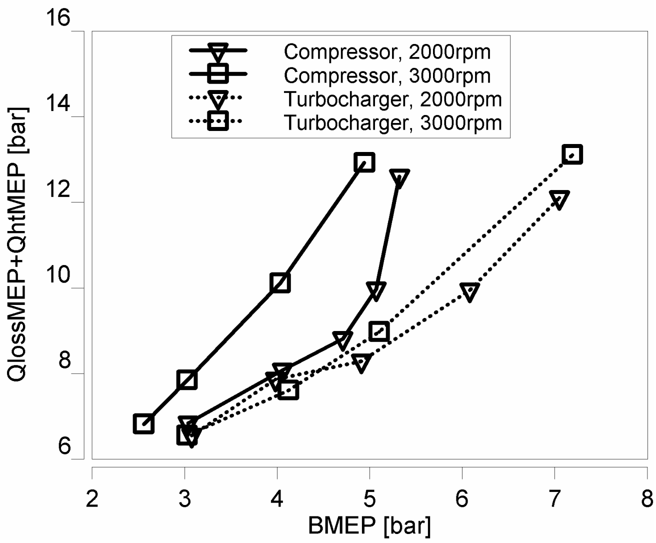

Figure 18, QlossMEP + QhtMEP versus BMEP for the mechanical supercharger compared with the turbocharger.

In Figure 19 the lost chemical energy to the exhaust is presented for the two cases. The absolute values are higher for the mechanical supercharger because of the same reason as earlier, i.e. the higher load - QhrMEP for the supercharger case.

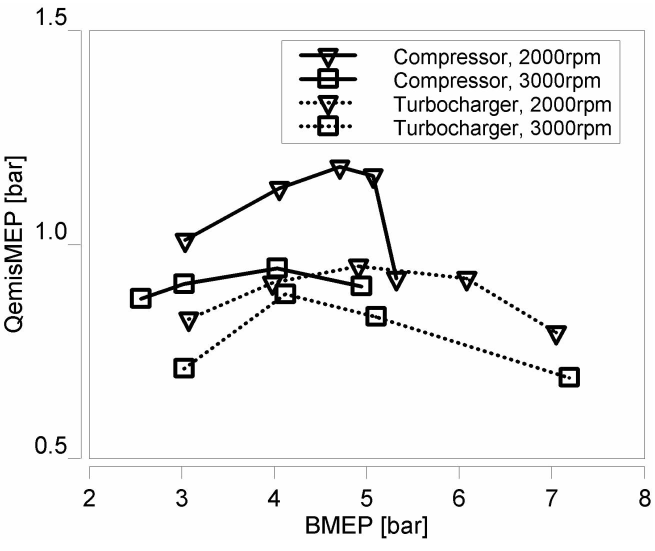

Figure 19, QemisMEP versus BMEP for the mechanical supercharger compared with the turbocharger.

The conclusion from this comparison is that using a mechanical supercharger is a poor way of increasing load on an HCCI engine since the increase in load is low compared with a turbocharged HCCI engine. Turbocharging is the preferred way to increase the operating range in an HCCI engine, despite the high pumping losses. A higher turbocharger efficiency and a higher exhaust gas temperature would increase the load and the brake efficiency.

A supercharged HCCI compared with NA SI

Turbocharging an HCCI engine is a way to increase the load, but the benefit of HCCI compared with SI decreases with an increasing load. Pumping losses increases with the load in a turbocharged HCCI engine, but decrease in a throttled SI. If HCCI is seen as a part load concept the operating mode could be changed to SI instead of turbocharging HCCI. Turbocharging HCCI needs a smaller turbine area than a SI engine due to the low exhaust gas temperature, and the same turbine would be too small for SI operation.

The brake efficiency is compared between the turbocharged HCCI, the natural aspirated SI, and the natural aspirated HCCI operating mode in Figure 20. Note that the compression ratio in the SI case is 14:1, i.e. the maximum in the original SI VCR engine. The brake efficiency is higher for the turbocharged HCCI than for the natural aspirated SI. The difference decreases with load and at about the maximum turbocharged HCCI load the brake efficiency is about the same. The main reasons for the decreasing difference in brake efficiency are the maximum cylinder pressure allowed, as mentioned earlier in the paper, and the decreasing throttle losses in the natural aspirated SI case. The turbocharged HCCI is restricted by maximum cylinder pressures at higher loads and the compression ratio is decreased with load, i.e. expansion work decrease and pumping losses increase. At 3bar BMEP a step in brake efficiency is seen between natural aspirated HCCI and turbocharged HCCI. The brake efficiency decreases from about 29% to 24% when the turbocharger is used. A higher turbocharger efficiency would decrease this step.

Brake specific engine out HC emissions are shown in Figure 21. HC emissions are higher for the HCCI case as expected, due to the lower combustion temperatures and the higher cylinder pressures. The difference decreases with load and the HC emissions are about the same at the maximum turbocharged HCCI load. Most of the HC emissions are oxidized in the catalyst in both cases, se Figure 9 for the HCCI case.

The conclusion is that both natural aspirated and turbocharged HCCI is preferred compared to natural aspirated SI in the entire operating range for a turbocharged HCCI engine. This conclusion is valid with the regard to the need of a different size of turbine in the HCCI and SI case respectively.

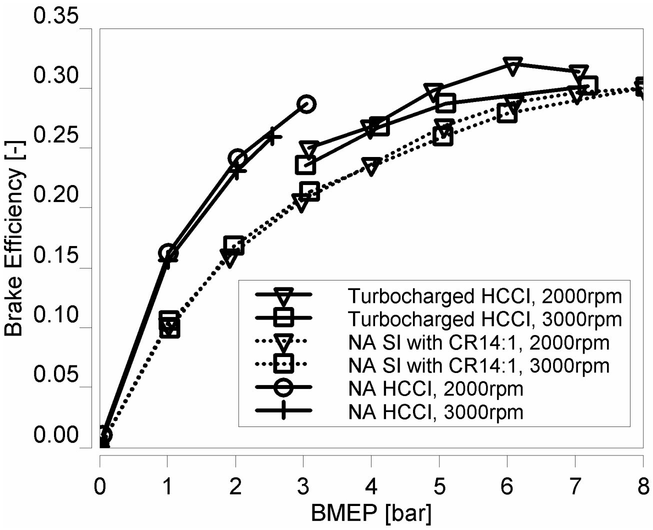

Figure 20, the brake efficiency versus the load for the turbocharged HCCI compared with the NA SI.

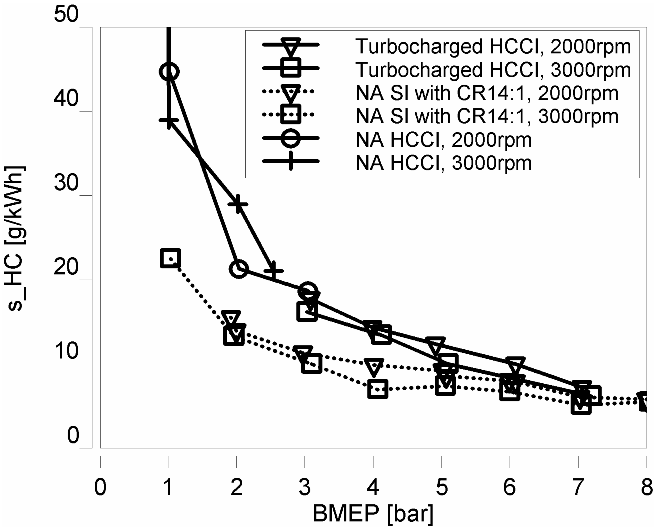

Figure 21, the brake specific engine out HC emissions [g/kWh] versus the load for the turbocharged HCCI compared with the NA SI.

CONCLUSION

The combustion efficiency at zero load is strongly dependent on the combustion phasing due to partial burning. The combustion efficiency increases from 68% to 82% by advancing combustion phasing, CA50, from 8°ATDC to 1.5°BTDC at 2000rpm and 0bar BMEP. At the same time fuel consumption decreases 16%, i.e. the increase in the combustion efficiency compensates the decrease in thermal efficiency. Since an exhaust to inlet air heat exchanger is used advancing combustion phasing is limited by the heat available in the exhaust and by that the oxidation of unburned HC and CO in the catalyst. The earliest combustion phasing possible with this setup was found to be the best at zero loads.

Throttling the inlet air is beneficial at zero loads, because the combustion efficiency increases and fuel consumption decreases. The increased pumping loss with throttling is compensated by the increase in the combustion efficiency. The combustion efficiency is increased from 70% to about 90% at 2000rpm and 0bar BMEP.

The HCCI operating range can be increased with both mechanical supercharging and turbocharging. The maximum load and brake efficiency are higher with turbocharging than with mechanical supercharging, due to the high parasitic losses with supercharging. The total turbocharger efficiency is assumed to be 42% in the turbocharged test. The maximum load achieved with the turbocharged HCCI is 10bar BMEP.

The brake efficiency is higher for the turbocharged HCCI than with the natural aspirated SI in the entire operating range for HCCI. The difference in brake efficiency decreases with load and is almost negligible at the maximum turbocharged HCCI load.

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CONTACT

Jari Hyvönen, MSc M. E.
E-mail: jari.hyvonen@vok.lth.se

Göran Haraldsson, MSc M. E.
E-mail: goran.haraldsson@vok.lth.se

Bengt Johansson, Professor.
E-mail: bengt.johansson@vok.lth.se

Department of Heat and Power Engineering, Division of Combustion Engines, Lund Institute of Technology, P.O. Box 118, SE-221 00 Lund, Sweden.

DEFINITIONS, ACRONYMS, ABBREVIATIONS

ABCD: After Bottom Dead Center

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

CAC: Charge Air Cooler

CAD: Crank Angle Degree

CA50: Crank Angle for 50% burned

CLD: Chemiluminescence Light Detector

CO: Carbon Monoxide

CO_2: Carbon Dioxide

COV: Coefficient Of Variation

CR: Compression Ratio

DI: Direct Injection

dP/dCA: Maximum Rate of Pressure Rise

FMEP: Friction Mean Effective Pressure

FuelMEP: 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: Natural Aspirated

NDIR: Non Dispersive Infrared

NEDC: New European Drive Cycle

NOx: Nitrogen Oxides (NO and NO_2)

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