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

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Operating range in a Multi Cylinder HCCI engine using Variable Compression Ratio

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ABSTRACT

Homogenous Charge Compression Ignition (HCCI) is a promising part load combustion concept for future power train applications. Different approaches to achieve and control HCCI combustion are today investigated and compared, especially concerning operating range. The HCCI operating range for vehicle applications should at least cover contemporary emissions drive cycles.

The operating range in terms of speed and load is investigated with a Naturally Aspirated (NA) four-stroke multi-cylinder engine with Port Fuel Injection (PFI). HCCI combustion control is achieved with Variable Compression Ratio (VCR) and inlet air preheating with exhaust heat. Both primary reference fuels and commercial gasoline are used in the tests.

HCCI combustion with commercial gasoline is achieved over a load range from 0 to 3.6bar BMEP, and over a speed range from 1000 to 5000rpm. Maximum load is at 1000rpm and decreases with an approximately straight slope to zero at 5000rpm. NOx emissions are kept below 15ppm and the rate of pressure rise below 6bar/CAD throughout the operating range. Combustion efficiency is above 90% for all positive loads. High amounts of unburned HC as well as CO emissions decrease combustion efficiency to 70% at zero load. Brake efficiencies up to 33% are obtained.

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 too 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 NOx 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 premixed, 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 [6-19]. The mixture temperature can be controlled by inlet air heating, 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. Mixture composition can be affected by stratification in Direct Injection (DI) systems. HCCI combustion in multi-cylinder engines [6,7,13] is an additional challenge due to cylinder-to-cylinder variations.

HCCI is today conceived as a part load combustion concept to improve drive cycle performance. Normal SI or diesel combustion can instead be used at start up, idle, and high load. The operating range in terms of speed and load for an HCCI engine is restricted by misfiring, i.e. too much dilution, and fast burn rate that induces noise and NOx emissions, i.e. too little dilution. High loads up to 16bar BMEP can, however, be achieved with turbocharging [12]. The fuel/air mixture is

leaned out towards low loads and the compression temperature becomes insufficient to achieve compression ignition. The proportion of diluents compared to fuel decrease at high loads and the burn rate becomes too fast. The available time for initiating compression ignition decreases at high engine speeds and causes misfiring or partial-burn cycles. In a multi cylinder engine also cylinder-to-cylinder variations restrict the operating range.

EXPERIMENTAL APPARATUS

The test engine in Figure 1 is a five-cylinder 1.6L Saab Variable Compression (SVC) engine originally developed to achieve the same performance as a 3.0L NA engine by using supercharging and VCR [20] with conventional SI combustion. The VCR range is between 8:1 and 14:1 in the original engine concept but is increased in this test engine, by changing pistons, to a range between 9:1 and 21:1. These pistons are hereafter called P21 pistons. The spark plugs are removed in the test engine since no SI combustion tests are done in this paper and also to make the piston geometry simpler. Higher compression ratio is needed to achieve HCCI combustion using high-octane fuel without excessive inlet air heating or excessive amounts of hot residuals. The results presented in this paper are compared to earlier HCCI tests done with the same test engine [18], but with different pistons giving a VCR range between 9:1 and 17:1. These pistons are hereafter called P17 pistons. Original valve timings are used during the tests, i.e. no increase in residual fraction is used to achieve HCCI combustion. Some basic engine specifications can be seen in Table 1.

The numbers for compression ratio used in this paper are geometrical compression ratios calculated from minimum and maximum cylinder volume at Bottom Dead Center (BDC) and Top Dead Center (TDC) respectively. The effective compression ratio is however dependent on factors such as timing for inlet valve closing, crevice volumes, blow-by, and manufacturing tolerances in components. The effective compression ratio, in the sense of pressure and temperature in the cylinder at the end of the compression stroke, is also dependent on factors that have to do with gas exchange, i.e. amount of trapped charge, e.g. inlet and exhaust manifold tuning, heat losses, and thermodynamic properties of the charge.

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	49°ABDC at 0.15mm lift
Combustion chamber	Pent roof / 4 valves DOHC

The inlet air and exhaust gas systems are modified compared to the earlier tests [18]. The main change is that exhaust gas heat recovery is used for heating the inlet air. The inlet air and exhaust gas systems, in Figure 2, consist of an air filter, mechanical supercharger, Charge Air Cooler (CAC), electrical air heater, exhaust to air heat exchanger, oxidizing catalyst, and some throttle valves for controlling the air flows. Note that the throttle valves are not used for inlet air pressure control. Inlet air can be routed differently to the engine. First, the air can be supercharged and cooled in a heat exchanger or bypassed directly. Second, the air can be heated with the electrical heater and the heat exchanger or again bypassed directly to the engine. The mechanical supercharger and CAC are not used, i.e. the engine is run NA during the tests presented in this paper.
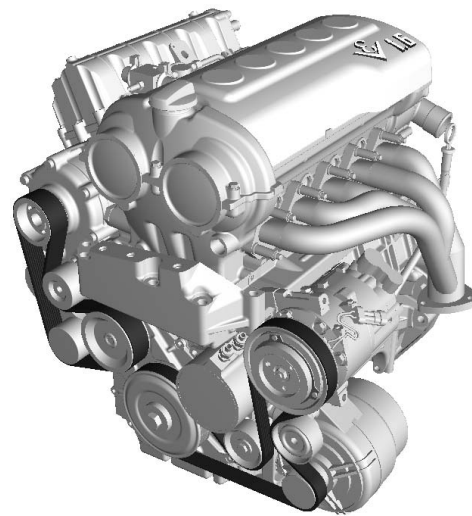

Figure 1, The Saab Variable Compression test engine.

The electrical air heater of 11 kW is only used during start up to achieve HCCI combustion and turned off as the combustion starts. SI combustion could be used instead at start up of the engine before enough heat from the exhaust is available for HCCI combustion. This is however not tested because no spark plugs were fitted in the test engine. Mixing hot and cold air controls the inlet air temperature as soon as the engine has warmed up sufficiently.
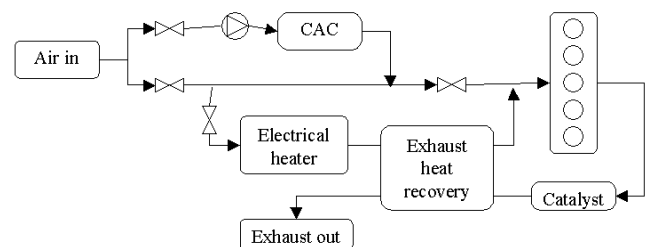

Figure 2, Inlet air heating and exhaust heat recovery.

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, 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 regulated emissions are measured with a Horiba exhaust gas analyzer MEXA-8120F. The CO and CO_2 analyzer, Horiba model AIA-23, works with Non-Dispersive Infrared (NDIR) technology. The NO_X ($NO+NO_2$) analyzer, 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), Horiba model FIA-22-2, as ppmC. The O_2 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 from all cylinders at the same time or from each cylinder individually. 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 [22]. The wideband lambda sensor is used for quick screening during test and to measure relative cylinder individual lambda differences. Fuel consumption is measured by weighing.

For engine control and cylinder pressure monitoring in-house developed software is used. 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). Combustion phasing is controlled through inlet air temperature, compression ratio, and fuel amount. The amount of fuel to the engine can be controlled cylinder-individually for e.g. cylinder-individual combustion phasing control.

Inlet air pressure, inlet air temperature, and CR are used for control, and are measured 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, see Figure 3. 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-logger at a rate of approximately 0.33Hz.
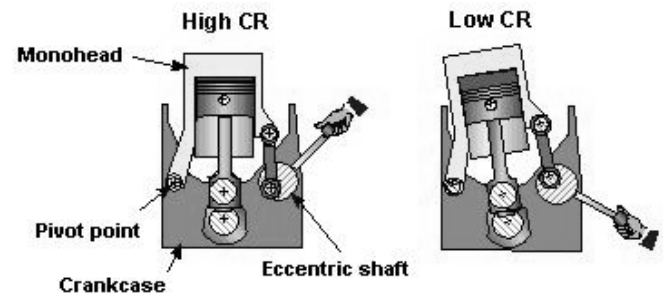

Figure 3, Principle of SVC engine [20].

Besides the difference in maximum compression ratio between P17 and P21 pistons the piston shape is different. The shape of the P17 pistons was designed to allow exhaust cam phasing for more residuals [18], see Figure 4. Most of the compression volume is on the exhaust valve side and the volume on the inlet valve side is narrow, hence a squish effect with heat losses can be expected. The P21 piston shape was modified with exhaust valve pockets instead and the compression volume is more evenly distributed across the combustion chamber, see Figure 4. Squish effects and heat losses to the walls are expected to be smaller with the P21 piston.
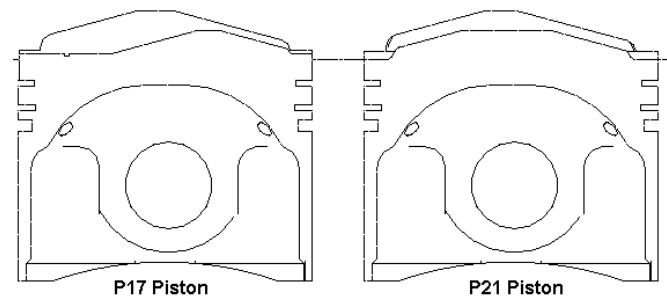

Figure 4, Combustion chamber comparison between P17 (left) and P21 (right) pistons.

TEST PROCEDURE

Two different fuels are used in the tests with different octane numbers, ON60 and RON92. The ON60 fuel is mixed from primary reference fuels, n-heptane and iso-octane. The RON92 fuel is commercial gasoline corresponding to U.S. Unleaded Regular with RON between 91 and 92. The reason for using ON60 fuel is to make comparisons easier with the earlier HCCI tests [18] done on this engine with lower maximum compression ratio, i.e. P17 pistons, and ON60 fuel. The engine is operated unthrottled during the tests, i.e. inlet air pressure is not controlled.

OPERATING RANGE

The operating range in terms of speed and load is tested with both fuels. The measurement matrix is divided into speeds from 1000 to 5000 rpm with 1000 rpm steps and four load points at each speed, i.e. maximum load, minimum load and two intermediate load points. The limits for the operating range are defined by some

chosen variables. The limiting variables at high load are NOx, max 15 ppm-dry at actual O_2 level[1], or maximum rate of pressure rise (dP/dCA), max 10 bar/CAD for an individual cylinder. The limits for 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. However during the tests only NOx and HC were found to be the limiting variables, i.e. the dP/dCA limit is not reached. Lowest tested load in the operating range is 0 bar BMEP, i.e. negative BMEP's are not tested. Combustion phasing is adjusted to Maximum Brake Torque (MBT) timing, CA50 at about 7°aTDC, at each operating point if possible. At the maximum load points combustion phasing is allowed to be later than MBT timing to decrease NOx and at idle or minimum load earlier than MBT timing to decrease HC and avoid misfiring. The means for adjusting combustion phasing are compression ratio and inlet air temperature. At low load inlet air temperature is allowed to be as high as possible using exhaust heat recovery only. The electrical air heater is not used during the tests presented in this paper. The exhaust temperature is increased as it flows through the oxidizing catalyst that converts residual chemical energy into thermal energy.

During the tests cylinder individual combustion phasing is adjusted with the injected fuel amount to achieve the same CA50 in all cylinders, i.e. a cylinder-to-cylinder difference in lambda and IMEP is allowed. The cylinder balancing is done automatically by the engine control system [21].

RESULTS AND DISCUSSIONS

PISTON COMPARISON

The shape of the combustion chamber was changed at the same time as maximum compression ratio was increased from 17:1 to 21:1, as described earlier in this paper. The P21 piston was expected to have less heat losses to the combustion chamber walls since the volume is more evenly distributed with less squish. In the earlier tests [18] the P17 piston was compared to HCCI results from a truck size engine with the same compression ratio. Then it was found that the P17 combustion chamber, with a volume to area ratio of 1.9mm, needed much higher octane number than the larger combustion chamber, with volume to area ratio of 3.2mm, to achieve HCCI combustion.

In this comparison between the P17 and P21 piston inlet air temperature and lambda are kept constant to achieve the same load, and compared at four different engine loads. Brake efficiency in Figure 5 is the same for the two pistons at all tested load points. Combustion phasing is however not the same in the compared points, as was the intention during the test run, see timing for CA50 in Figure 6. The difference in combustion phasing was found during post processing of the heat release. A compression ratio dependent TDC offset was wrong in the engine control program during the P17 test run.

Combustion phasing is earlier with the P21 piston because of higher compression ratio see Figure 5. Combustion efficiency is the same at the two highest load points, but higher for the P21 piston at the two lowest load points, where the combustion is really inefficient. The combustion efficiency can be increased substantially at low load as showed later in this paper.

At the two highest load points there are no clear differences between the two pistons. Compression ratio, combustion phasing, brake efficiency and combustion efficiency are almost the same at 3 and 4.5 bar BMEP. Compression ratio is slightly higher for the P21 piston at 4.5 bar BMEP, but otherwise no difference in performance. At the two lowest load points there is however a difference in performance, but no clear conclusions can be drawn because of the difference in combustion phasing. The compression ratio is higher at 0 and 1.5 bar BMEP with the P21 piston causing earlier combustion phasing and higher combustion efficiency. With a lower compression ratio and later combustion phasing the combustion efficiency would decrease, but what would happen to the brake efficiency is unclear. The lower expansion ratio with lower compression ratio and lower combustion efficiency decrease the brake efficiency, but the later combustion phasing closer to MBT timing increases brake efficiency.

The turbulence level can be expected to be higher with the P17 piston due to the squish effect, and hence the cooling losses to the walls. Combustion is not initiated more easily with the P21 as expected. On the other hand cooling losses might not be so much lower with the P21 pistons with less turbulence, due to the altogether narrow combustion chamber. No clear advantage can be found with the P21 combustion chamber compared to the P17 combustion chamber except the higher available compression ratio.
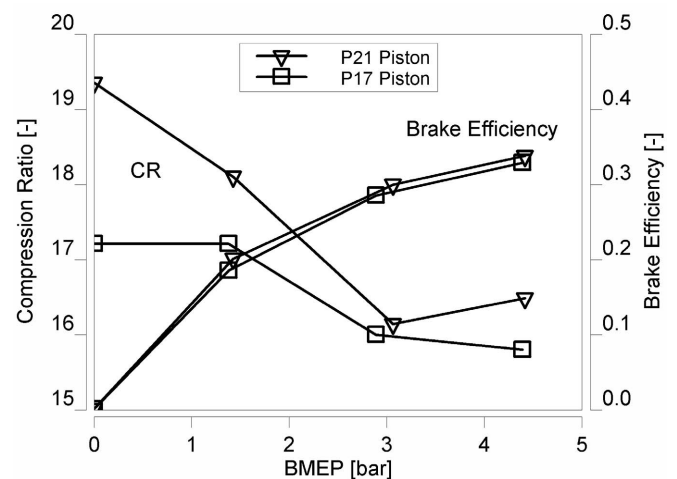

Figure 5, Compression ratio and brake efficiency as function of load at 2000rpm and ON60 fuel

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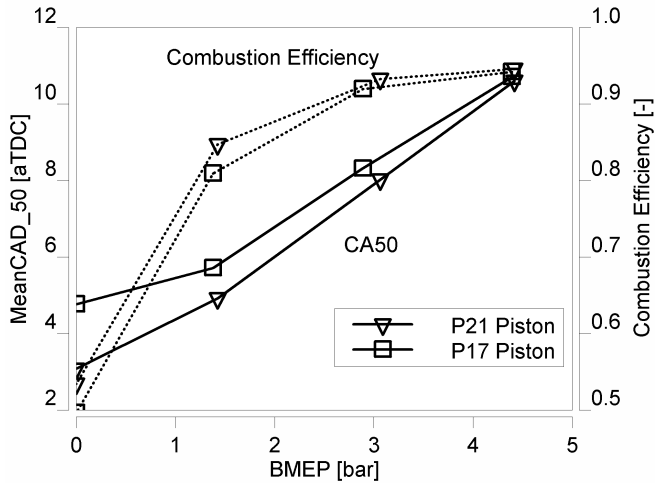[1] Not referenced against a fixed percentage O_2 in the exhaust.

Figure 6, CA50 and combustion efficiency as function of load at 2000rpm and ON60 fuel.

Figure 7, Operating range and compression ratio iso-lines with P21 pistons, ON60 fuel.

OPERATING RANGE WITH P21 PISTONS AND ON60 FUEL

The operating range with P21 pistons is compared with earlier tests made with P17 pistons. Note that negative BMEP is not tested, which would be possible with the P21 piston and ON60 fuel. The operating range with higher available compression ratio, 21:1, in Figure 7 is slightly larger than earlier results with 17:1 in Figure 8. The improvements are mostly found at higher engine speed. The engine can be run unloaded at all engine speeds up to 5000 rpm and higher load is possible at 4000 rpm. The compression ratio is increased as the engine speed increases. At about 2000 rpm the maximum compression ratio is reached with the P17 pistons, but for the P21 pistons maximum compression ratio is reached close to 5000 rpm. About the same compression ratios are used with both pistons at lower engine speeds.

<u>Minimum load</u>

The limiting factor at low load or idle is misfiring and unstable combustion due to lean conditions. At higher engine speeds also the available time for initiating combustion decreases. The possibility to use VCR and increase the compression ratio at increasing engine speed is therefore beneficial. COV (IMEPnet) was between 1.5 and 5% over the whole operating range and below 4% up to 4000rpm. The combustion stability can be improved by phasing the combustion earlier than MBT timing or by decreasing lambda. By phasing the combustion earlier, more time for oxidizing the fuel at high pressure and temperature would be available, and thus combustion efficiency would increase.
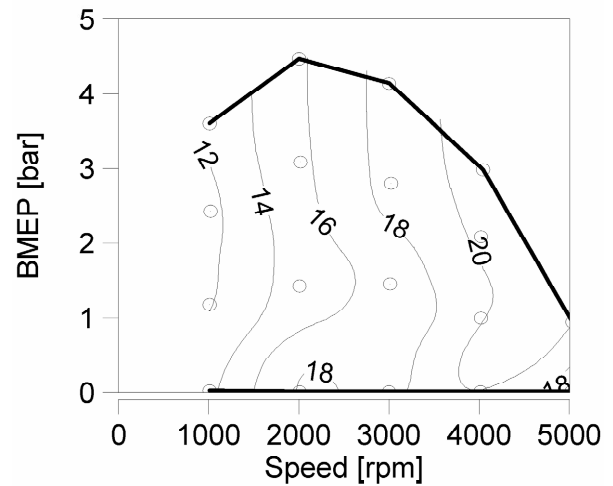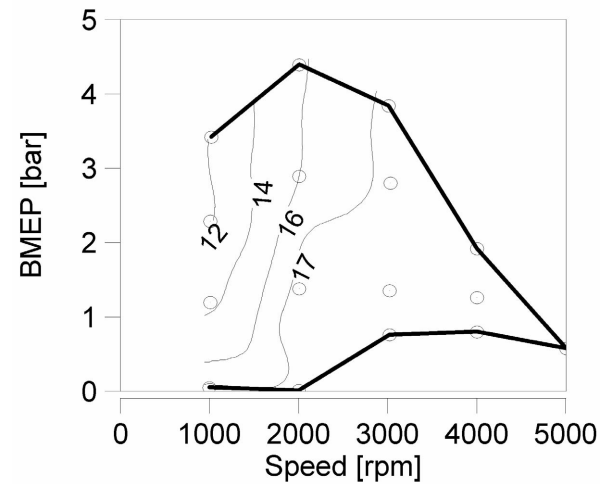

Figure 8, Operating range and compression ratio iso-lines with P17 pistons, ON60 fuel [18].

Thermal throttling at low load, i.e. increasing inlet air temperature, decrease both air excess and ignition time for the air fuel mixture. The earlier ignition can then be used to decrease the compression ratio and improve combustion efficiency. Since the expansion is slower with lower compression ratio the time for fuel oxidation increases [18] and also unburned HC from crevices decreases with lower cylinder pressures. In Figure 9 the inlet air temperature with P21 pistons is compared to what was used with the P17 pistons. The compression ratio changes over load for the two engine speeds, 2000 and 3000rpm, and can be seen in Figure 7 and Figure 8. Inlet air temperature is higher with P21 pistons than was with P17 pistons. Maximum temperature with the P17 piston tests was adjusted and chosen to 150°C with the electrical inlet air heater [18], which was used then for temperature control. However during the present tests with P21 pistons the inlet air heat is recovered from the exhaust and allowed to increase as high as possible whenever needed. So higher inlet air temperature is used at lower load with P21 pistons compared to P17 pistons, see Figure 9, to improve combustion efficiency,

see Figure 10. The combustion efficiency is improved by 12.5% at 1bar BMEP and by 40% at idle, due to less partial burning. Combustion efficiency is above 90% at all loads except unloaded, where it decreases to 70%. With P17 pistons the combustion was poor at 2000rpm and no load with unburned HC in the exhaust corresponding to 50% of the supplied fuel. The combustion efficiency can be increased further at low loads by exhaust gas residuals with exhaust cam phasing, which was earlier tested with the P17 pistons [18].
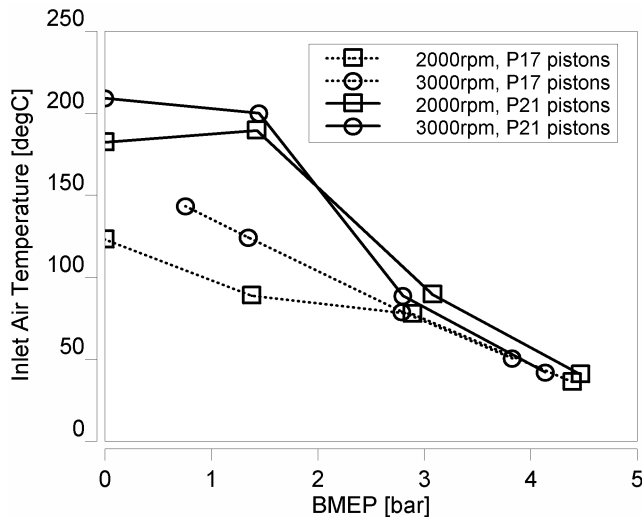
Figure 9, Inlet air temperature with P17 and P21 pistons as function of BMEP, ON60 fuel.

Maximum load

Combustion efficiency at higher load, in Figure 10, is the same for both pistons, even if a slightly higher compression ratio is used with the P21 piston at high load.

The limiting factor at high or maximum load is the lack of dilution resulting in too fast combustion, high dP/dCA, or too high combustion temperature, i.e. high NOx. High NOx is the limit with both pistons. To maximize the amount of dilution at high load inlet air temperature should be as low as possible to increase volumetric efficiency, see Figure 9. With increasing speed, the compression ratio is increased to maintain combustion phasing. Since the P17 piston reaches maximum compression ratio earlier than the P21 piston, inlet air temperature has to be increased to maintain combustion phasing. However, the higher inlet air temperature decreases the amount of dilution and engine load has to be decreased to avoid too high NOx levels. Despite the higher compression ratio with P21 pistons the maximum load is not much higher at the higher engine speeds compared with P17 pistons. The reason is more cylinder-to-cylinder variations with P21 pistons than P17 pistons, discussed later in this paper. The best improvement in attainable load is at 4000rpm where BMEP increased from 1.9 to 3bar, a 55% increase, see Figure 7 and Figure 8.
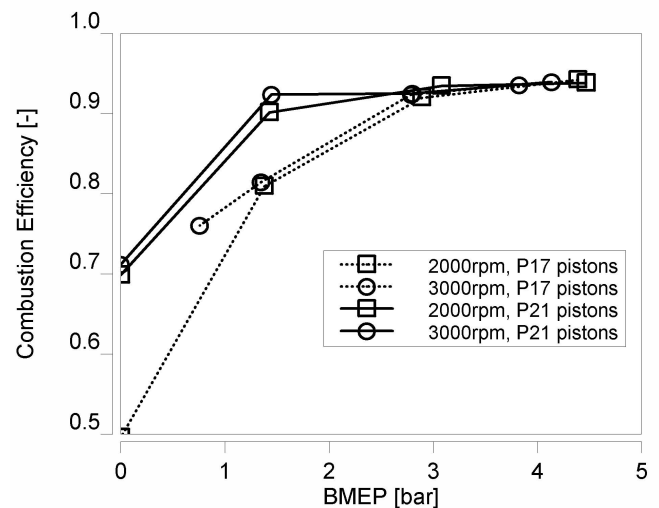
Figure 10, Combustion efficiency with P17 pistons and P21 pistons with heat recovery system as function of BMEP, ON60 fuel.

INLET AIR HEATING

Exhaust heat is transfered to the inlet air through a heat exchanger, shown in Figure 2. The oxidizing catalyst is placed in front of the heat exchanger to increase the exhaust gas temperature by oxidizing residual HC and CO. In Figure 11 the exhaust and inlet air temperatures are presented at different engine speeds and no load. The exhaust gas temperature increases over the catalyst by about 100°C at speeds up to 3000rpm at no load and somewhat less at higher speeds. At 4000 rpm the catalyst has not ignited properly since the temperatures after the catalyst are lower than the general trend, i.e. the measurement is done too soon after start up of the engine. Inlet air temperature after the heat exchanger is above 200°C at all engine speeds without load and about 75°C below the exhaust gas temperature after the catalyst. However the inlet air temperature decreases by another 75°C before entering the inlet ports due to heat losses, especially in the inlet manifold and decreases probably further in the inlet ports before entering the cylinders. The heat losses in the inlet ports can be noticed during warming up of the engine as a need for higher inlet air temperature. Air temperature before the cylinders are measured about 130mm from the inlet valves. Even higher inlet air temperatures are possible with less heat losses and with an optimized heat exchanger. Higher inlet air temperature would then consequently decrease compression ratio at idle and increase combustion efficiency. It should be noted that higher inlet air temperature than exhaust gas temperature is possible with lower heat losses and using the chemical energy released in the catalyst.
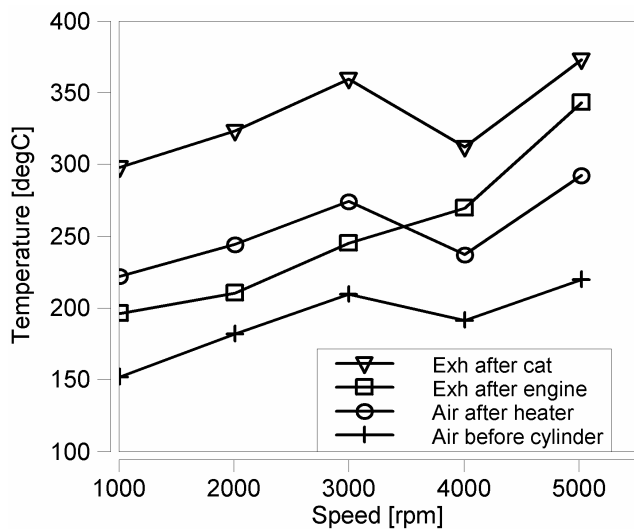
Figure 11, Exhaust gas and inlet air temperatures as function of engine speed at zero load.

Combustion efficiency, in Figure 12, increases since HC and CO emissions decrease with engine speed with no load. The reason is decreasing lambda due to thermal throttling and increased indicated load. Both pumping losses and mechanical losses increase with speed. Lambda iso-lines in the operating range can be seen in Figure 15.
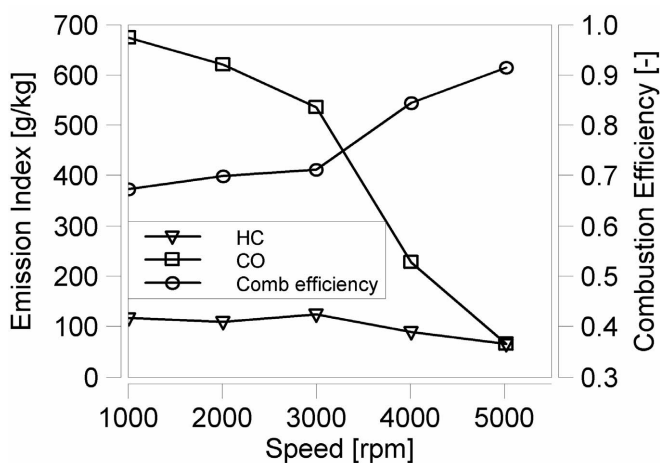

Figure 12, HC and CO emissions and combustion efficiency as function of engine speed at zero load.

The use of VCR and heat recovery makes the combustion initiation robust, compared to HCCI/CAI concepts where the charge ignition relies on exhaust gas residuals from the earlier combustion cycle. The combustion can be turned off for several minutes before restart if desired with VCR and heat recovery. The drawback is the start-up process where an external heat source is needed, unless the heat exchanger and catalyst are preheated.

OPERATING RANGE WITH RON92 FUEL

When fuel octane number is increased, compression ratio and/or inlet air temperature have to be increased to initiate combustion. The operating range in terms of speed and load is presented in Figure 13 with RON92

gasoline. The compression ratios are higher compared to ON60 fuel at lower engine speeds, but lower at some operating points at higher engine speeds. The compression ratio is in fact kept at the maximum at zero load to initiate combustion and keep COV (IMEP) below 5-6%. The compression ratio in the ON60 case is almost only dependent on engine speed, Figure 7, whereas it is also dependent on load in the RON92 case, Figure 13. Lower compression ratio is used at intermediate load at intermediate and high engine speeds to increase combustion efficiency, because higher inlet air temperature is available to keep the preferred CA50. The inlet air temperature at different speeds and loads can be seen in Figure 14. The inlet air temperature is higher because exhaust gas temperature increase with both load and speed. In Figure 15 lambda is indicated throughout the operating range. Combustion lambda is directly connected to engine load since no throttling is used. Lambda decreases towards higher engine speeds since inlet air temperature increases, i.e. thermal throttling.

Compression ratio is close to maximum both at low load and high load. At low and zero loads, maximum compression ratio and maximum available inlet air temperature are needed to initiate combustion and increase combustion efficiency. A lower compression ratio could be used if exhaust gas residuals were used at low loads [18]. At high load the maximum compression ratio is used for reducing inlet air temperature to maximize volumetric efficiency when running NA. Maximum load decreases with increasing engine speed as the inlet air temperature is increased, at the same time decreasing mixture dilution.

The engine can be run unloaded at all tested engine speeds up to 5000rpm with commercial gasoline of RON92. The slight deviations from no load at 3000rpm and 5000rpm in Figure 13 are because of the difficulties in keeping constant load during the measurements without continuously adjusting the injected fuel amount.
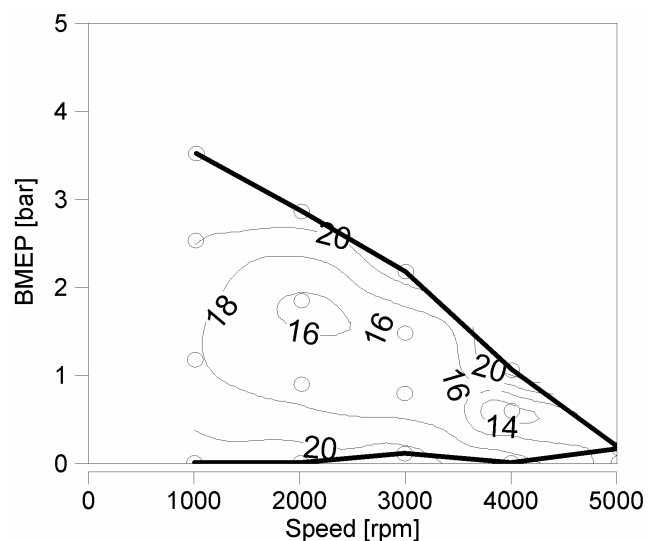

Figure 13, Operating range and compression ratio iso-lines with P21 pistons and RON92.
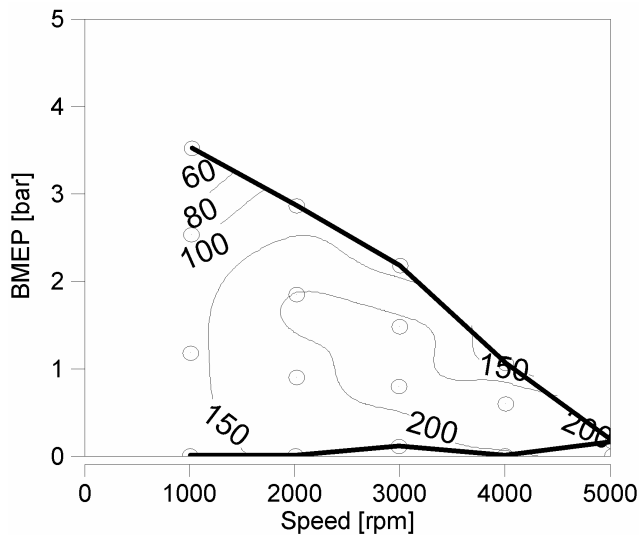
Figure 14, Operating range and inlet air temperature [°C] iso-lines with P21 pistons and RON92.
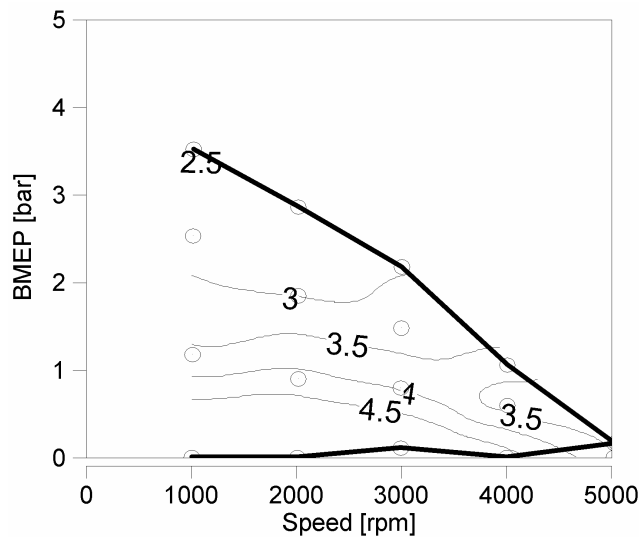

Figure 15, Operating range and Lambda iso-lines with P21 pistons and RON92.

Brake efficiency with HCCI combustion is compared to SI combustion in Figure 16. The SI case is run with the same engine but with the original compression ratio range from 8 to 14. In the SI tests at all the loads presented in Figure 16, i.e. up to 5bar BMEP, compression ratio is at the maximum 14:1. The improvement in brake efficiency from SI combustion to HCCI combustion is about 35% at 3bar BMEP and about 57% at 1.5bar BMEP. The efficiency improvement would have been even larger if compared to a contemporary engine with fixed, lower, compression ratio. Brake efficiency for the HCCI cases are about the same independent of fuel, i.e. ON60 or RON92, and engine speed.
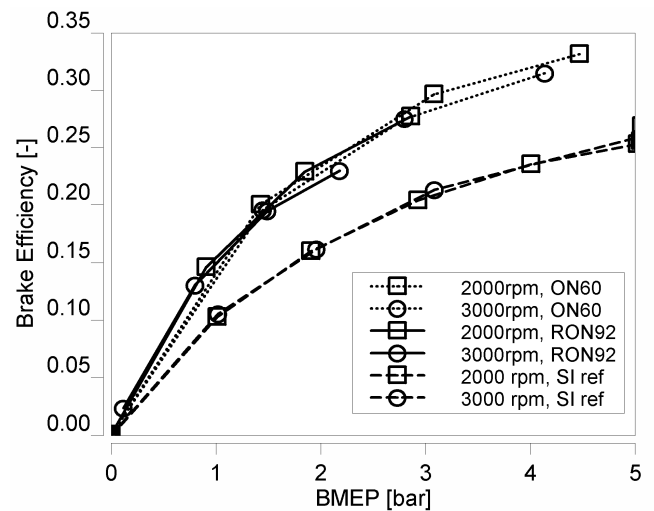

Figure 16, Brake efficiency with HCCI combustion compared to SI combustion, with CR14:1, as function of load.

MAXIMUM BRAKE TORQUE TIMING

Assumed MBT timing is used in all tests presented in this paper if nothing else is stated. The MBT timing for CA50 has been found in earlier tests to be around 7°ATDC at different loads and speeds. The change in brake efficiency seemed to be small for small changes in CA50. In Figure 17 brake efficiency is presented as function of CA50 at about 1.4bar BMEP and 2000rpm. Only three operating points are measured around the expected MBT timing to confirm the assumption. MBT timing for other speeds and loads are not confirmed here. Injected fuel amount is kept constant. To achieve the different combustion timings compression ratio is changed between 20:1 and 19.2:1. Inlet air temperature increased by 6°C during the test and counteracted slightly the change in compression ratio. The MBT timing for CA50 is found to be at about 7°ATDC. The brake efficiency changes about 0.6-0.9 percentage points for 1CAD change in CA50.
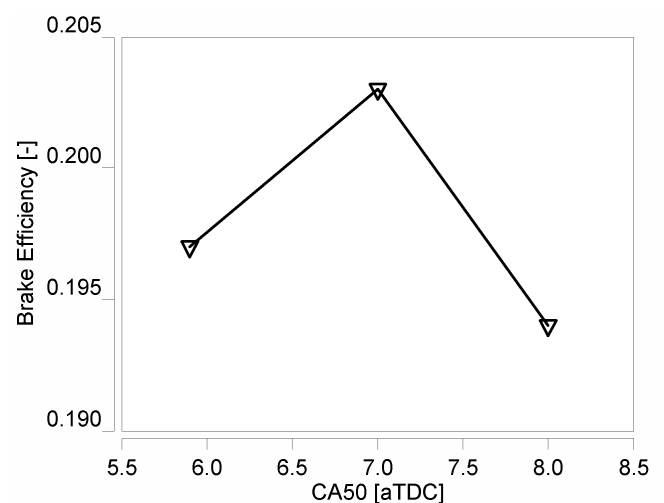

Figure 17, Brake efficiency as function of CA50 at 1.4bar BMEP, 2000rpm, and fuel RON92.

CYLINDER-TO-CYLINDER VARIATIONS

The chosen limit for maximum load is 15 ppm NOx or 10bar/CAD rate of pressure rise during the operating range tests. Cylinder-to-cylinder variations are however an obstacle for higher loads. In the tests cylinder individual combustion phasing is adjusted using the injected fuel amount to achieve the same CA50 in all cylinders, i.e. a cylinder-to-cylinder difference in lambda and IMEP is allowed. The cylinder balancing is done automatically by the engine control system [21]. Without combustion timing adjustment CA50 would differ by up to 10CAD between the earliest and the latest phased cylinder. The engine can be run in this way also but equal CA50 is preferred to avoid misfiring or too fast combustion in individual cylinders. In Figure 18 cylinder individual IMEP net is presented at the maximum load points at 3000 and 4000rpm. The load difference between the cylinders is larger than a factor 2 for 4000rpm. The higher loads in cylinders 3 and 4 are mainly due to slightly lower compression ratio in these cylinders, see cylinder pressure trace in Figure 19. There are also some differences in inlet air temperature and the temperature is highest in cylinder 1 and lowest in cylinder 5, with a difference of about 11-15°C at the engine speeds presented in Figure 18. To compensate for the lower compression ratio more fuel is injected in cylinder 3 and 4 to keep the same combustion timing, hence a higher load is achieved. The load differences are consequently affecting NOx formation and rate of pressure rise. The high NOx formation in cylinder 3 and 4 limits the average load, which defines the HCCI operating range. High load is, however, possible at high engine speeds if not NOx emissions are considered, e.g. IMEP in cylinder 4 is 5bar at 3000rpm and 4.3bar at 4000rpm.
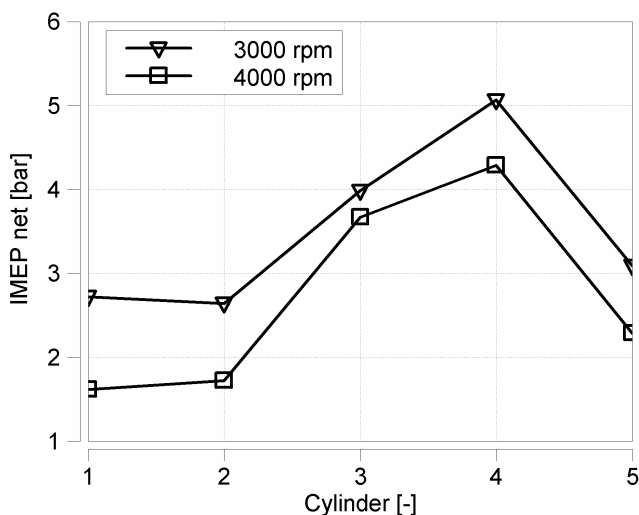

Figure 18, IMEP net cylinder variations at maximum load and RON92 fuel.
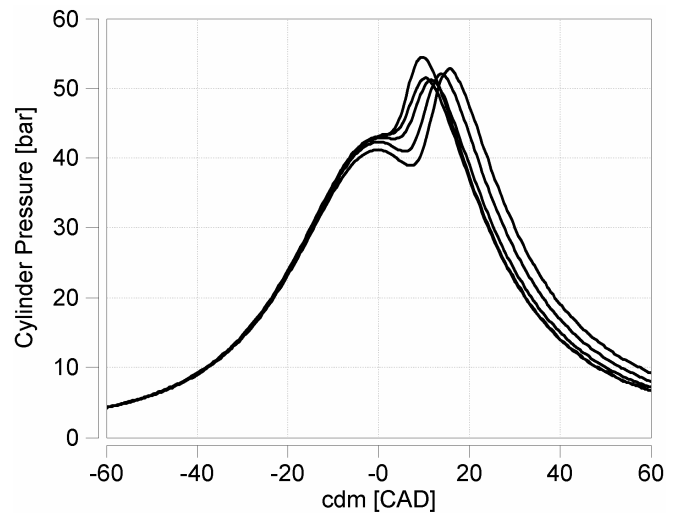

Figure 19, Cylinder pressure traces (300 cycle average) from all 5 cylinders at maximum load, 2000rpm, fuel RON92.

To decrease cylinder-to-cylinder variations, differences in parameters that affect the auto ignition timing should be minimized, e.g. compression ratio, inlet air temperature, cylinder wall temperatures, and mixing between air and fuel [7]. The biggest differences in compression ratio in this test engine derive from the monohead, i.e. cylinder head pent roof, casting. The casting tolerances for normal SI combustion with lower compression ratios are acceptable, but not for high compression ratios in HCCI or DI engines. Fully machined combustion chambers would be an improvement. The differences in inlet air temperature derive from poor mixing in the in-house built inlet air system, where hot and cold air is poorly mixed. An improvement is also needed here.

TRADE-OFF BETWEEN COMPRESSION RATIO AND INLET AIR TEMPERATURE

The same combustion phasing can be achieved with different combinations of compression ratio and inlet air temperature. Trade-off between inlet air temperature and compression ratio is tested at compression ratios above 17:1. Up to 17:1 has already been tested earlier [18]. The same CA50 is kept by adjusting inlet air temperature for three different compression ratios 17,19, and 21. The load, i.e. supplied fuel amount, and speed is kept approximately constant during the test. RON92 is used as fuel.

In earlier tests [18], with maximum compression ratio of 17:1, brake efficiency increased and combustion efficiency decreased with increasing compression ratio. In this test compression ratios up to 21:1 are used. Figure 20 shows the trade-off between compression ratio and inlet air temperature. The load is 2.5bar BMEP and the engine speed 2000rpm. CA50 is kept at about 8 CAD ATDC. A change in compression ratio from 17 to 21 corresponds to a change in inlet air temperature from 174 to 126°C, i.e. 12°C per CR unit.
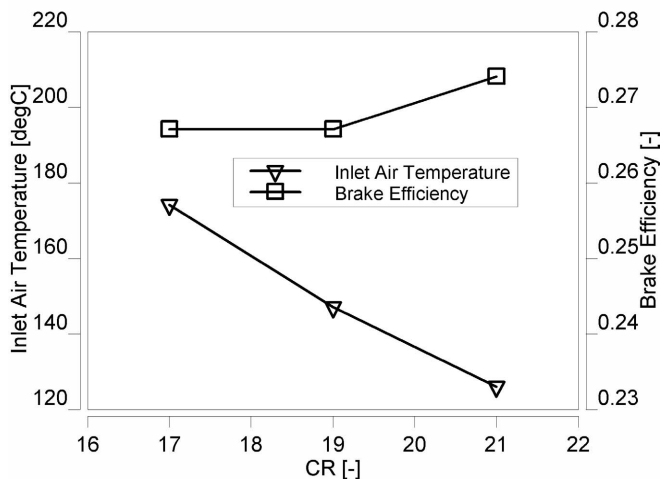
Figure 20, Inlet air temperature and Brake efficiency as function of compression ratio.

Brake efficiency increases slightly between compression ratio 17 and 21 at the same combustion phasing, despite a 2 percentage point decrease in combustion efficiency. The reasons are increased lambda with lower inlet air temperature and increased expansion ratio. Brake efficiency did not increase between CR 17:1 and 19:1 in this test because combustion phasing was slightly closer to MBT timing at CR 17:1. There is no clear disadvantage with using compression ratios up to 21:1. The control strategy at low load would be to use as high inlet air temperature and low compression ratio as possible to maximize combustion efficiency. At intermediate and high load as high compression ratio and as low inlet air temperature as possible should be used to increase brake efficiency, due to compression ratio and increase in volumetric efficiency, and hence increase dilution for slowing down the burn rate.

CONCLUSION

HCCI combustion is achieved in a multi cylinder engine with Variable Compression Ratio, inlet air heating by exhaust heat recovery, and standard valve timings. With commercial RON92 fuel an operating range from 1000 to 5000rpm, and from 0 to 3.5bar BMEP, is achieved. Maximum load is at 1000rpm and decreases with an approximately straight slope to zero at 5000rpm.The load range is limited at all engine speeds by NOx emissions, max 15ppm-dry at actual O_2 level. The engine can be run unloaded at all speeds from 1000 to 5000rpm. Highest average COV (IMEPnet) is 5% measured at 5000rpm and no load. Combustion efficiency is above 90% at all loads above zero. High amounts of unburned HC as well as CO emissions decrease combustion efficiency to 70% at zero loads.

Inlet air preheating is done with an exhaust gas heat exchanger. The exhaust gas temperature is increased before the heat exchanger in an oxidizing catalyst by oxidizing residual HC and CO. The inlet air temperature is between 150 and 200°C at engine speeds from 1000 to 5000rpm and no load.

Trade-off between compression ratio and inlet air temperature is measured at intermediate load. No drawback with higher compression ratio can be seen up to 21:1. At low loads and for all engine speeds, the highest available inlet air temperature is beneficial to increase combustion efficiency. At high load, where the lack of dilution affects the load limit, lowest possible inlet air temperature is beneficial. The compression ratio is adjusted to achieve desired combustion phasing.

Cylinder to cylinder variations due to differences in compression ratio and inlet air temperature prevents higher loads. The differences in combustion timing can be adjusted by cylinder individual fuel amounts, but with the drawback of inducing load differences between the cylinders.

Brake efficiency is increased by 35% at 3bar BMEP and 57% at 1.5bar BMEP compared to conventional SI combustion with the same engine using compression ratio 14:1.

The combustion initiation with Variable Compression Ratio and exhaust heat recovery is robust, since the combustion initiation is not dependent on residuals from the previous combustion cycle. HCCI combustion can be restarted several minutes after a stop from hot engine without external heating.

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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

CR: Compression Ratio

CO: Carbon Monoxide

COV: Coefficient Of Variation

DI: Direct Injection

dP/dCA: Maximum Rate of Pressure Rise

HC: Hydro Carbons

HCCI: Homogeneous Charge Compression Ignition

IMEP: Indicated Mean Effective Pressure

MBT: Maximum Brake Torque

NA: Naturally Aspirated

NOx: Nitrogen Oxides (NO and NO_2)

ON: Octane Number

PFI: Port Fuel Injection

PCI: Peripheral Component Interconnect local bus

RON: Research Octane Number

SI: Spark Ignition

SVC: Saab Variable Compression

VCR: Variable Compression Ratio