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Published in:
SAE Special Publications

1999

Link to publication

Citation for published version (APA):
Christensen, M., & Johansson, B. (1999). Homogeneous Charge Compression Ignition with Water Injection. In *SAE Special Publications* (Vol. 1999). Society of Automotive Engineers.
http://www.sae.org/technical/papers/1999-01-0182

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SAE TECHNICAL PAPER SERIES

1999-01-0182

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Reprinted From: Processes of Diesel Engine Combustion (SP-1444)

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Detroit, Michigan
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1999-01-0182

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ABSTRACT

The use of water injection in a Homogeneous Charge Compression Ignition (HCCI) engine was experimentally investigated. The purpose of this study was to examine whether it is possible to control the ignition timing and slow down the rate of combustion with the use of water injection. The effects of different water flows, air/fuel ratios and inlet pressures were studied for three different fuels, iso-octane, ethanol and natural gas.

It is possible to control the ignition timing in a narrow range with the use of water injection, but to the prize of an increase in the already high emissions of unburned hydrocarbons. The CO emission also increased. The NOx emissions, which are very low for HCCI, decreased even more when water injection was applied. The amount of water used was of the magnitude of the fuel flow.

INTRODUCTION

THE HCCI CONCEPT – The major advantages with Homogeneous Charge Compression Ignition, HCCI, is high efficiency and low NOx emissions. The major drawback is the problem of controlling the ignition timing over a wide load and speed range. An other drawback compared to the Spark Ignition (SI) engine and the diesel engine is higher emissions of unburned hydrocarbons.

In a HCCI engine, the fuel is injected into the (preheated) air in the intake manifold, to create a homogeneous charge. The charge is then further heated during the compression stroke to get auto-ignition, close to Top Dead Center (TDC). With HCCI, there is no direct control over the ignition timing. The ignition timing can only be controlled indirect. By adjusting the operating parameters correctly, ignition will occur near TDC. In SI engines, large cycle-to-cycle variations occur since early flame development varies significantly [1]. With HCCI, cycle-to-cycle variations of combustion are very small since combustion initiation takes place at many points at the same time. HCCI has no flame progation, instead the whole mixture burns close to homogeneous at the same time [2]. To limit the rate of combustion, very diluted mixtures must be used. Previous studies have used extremly lean mixtures [24, 25] and/or high amounts of with Exhaust Gas Recycling (EGR) [26]. A third alternative is to use water as dilutant. This has been studied in this paper. The water was injected into the intake manifold with a common low pressure fuel injector.

PREVIOUS WORK – Several studies on HCCI have been performed on two stroke engines [2-7] showing that HCCI combustion has great potential for further development. Several studies on four stroke engines have also been carried out [8-23]. High efficiency and low NOx-emissions are reported, but emissions of unburned hydrocarbons are high.

In an earlier research on HCCI by Christensen, et al. the effect of Exhaust Gas Recycling (EGR) on HCCI combustion were studied [26]. The results showed that EGR retards the ignition timing and increases the indicated efficiency. The specific emissions of unburned hudrocarbons were decreased when EGR was used.

PRESENT WORK – The main objective of this study was to experimentally investigate how water injection affect the ignition timing and the combustion process. Three different fuels were used: iso-octane, ethanol and natural gas. The octane numbers are 100, 106 and 120 respectively. The composition of the natural gas used, is given in Table 1. The compression ratio was set at 18:1. The experiments were carried out in both naturally aspirated mode and supercharged mode. When the engine was operated naturally aspirated (NA) no throttling was used and when supercharged (SC), a boost pressure of 1 bar was used. The engine speed was set to 1000 rpm.

Table 1. Composition of the natural gas used.

Component	Vol. %	Mass %
Methane	88.30	75.94
Ethane	6.26	10.09
Propane	2.81	6.64
n-Butane	0.57	1.78
iso-Butane	0.43	1.34
n-Pentane	0.071	0.27
iso-Pentane	0.109	0.42
Hexane	0.049	0.23
Nitrogen	0.333	0.50
Carbon dioxide	1.06	2.50

EXPERIMENTAL APPARATUS

The engine used for the experiments originated from a Volvo TD100 series diesel. The engine is an in-line six cylinder engine, modified to operate on one cylinder only. This arrangement givs less reliable brake specific values. Instead indicated results has to be used. The cylinder pressure was recorded with a pressure transducer. Engine specifications are shown in Table 2. To initiate HCCI combustion high temperature is necessary. Therefore, the inlet air has to be preheated to achieve high enough temperature near TDC, to initiate auto-ignition, at the choosen compression ratio. The inlet air was preheated with an electrical heater. The electrical power needed for the heater has not been included in the efficiency calculation.

Table 2. Geometric properties of the test engine.

Displaced Volume	1600 cm^3
Bore	120.65 mm
Stroke	140 mm
Connecting Rod	260 mm
Inlet Valve Diameter	50 mm
Exhaust Valve Diameter	46 mm
Exhaust Valve Open	39° BBDC (at 1 mm lift)
Exhaust Valve Close	10° BTDC (at 1 mm lift)
Inlet Valve Open	5° ATDC (at 1 mm lift)
Inlet Valve Close	13° ABDC (at 1 mm lift)
Valve Lift Exhaust	13.4 mm
Valve Lift Inlet	11.9 mm
Compression Ratio	18:1
Combustion Chamber	Pancake

RESULTS

MIXTURE QUALITY REQUIREMENTS – In order to slow down the combustion rate, much excess of air and/or EGR can be used. In the present work it has been studied if it is possible to slow down the combustion rate with the use of water injection. Most of the measured results are plotted against the ratio of water flow and fuel flow:

$$WF / FF = \frac{\dot{m}_w}{\dot{m}_f}$$

where $\dot{m}_w$ is the mass flow of injected water and $\dot{m}_f$ is the mass fuel flow. The amount of injected water has been varied for a few different test cases, to study how it affects HCCI. Of especially interest was to study changes in:

- rate of heat release
- onset of combustion
- combustion duration
- efficiency
- emissions

The study has been carried out for six different test conditions, three cases with unthrottled operation and three cases with supercharging. Table 3 givs a short discription of the different test cases. λ is the ratio of the actual air/fuel ratio to the stoichiometric air/fuel ratio. The λ values are calculated from the exhaust gas compositions. The inlet temperatures are measured upstream the fuel and water injection.

Table 3. Test cases used.

Fuel	Inlet condition	λ	Inlet temperature [°C]
iso-octane	NA	2.81	115
iso-octane	SC	3.70	26
ethanol	NA	3.06	105
ethanol	SC	3.44	70
natural gas	NA	2.48	167
natural gas	SC	3.00	105

ENGINE LOAD, IMEP – If the major problem with HCCI is to control the ignition timing over a wide speed and load range, an other problem is the very high rate of combustion. Too fast homogeneous combustion rates can generate knock related problems and very steep pressure rise, with noisy operation and engine damage as results. To slow down the rate of combustion very diluted mixtures have to be used, but the requirement of diluted mixtures limits the attainable upper engine load. Without the use of any EGR, the rich limit for NA HCCI is $\lambda \approx 2$, at least at lower engine speeds. This value may vary, dependent on the fuel type and the compression ratio. The need of very diluted mixtures is a problem if HCCI should be used for the entire load range. If HCCI should be used only at part load, this is not a problem. At part load (1-5 bar of IMEP), the load is changed mainly by adjusting the fuel flow. Figure 1 shows the IMEP for the test cases studied. For the NA cases a load of about 3 bar of IMEP was selected. This load is representative for a passenger car at constant highway speed. The load limit for NA HCCI is about 5 bar of IMEP [26]. For the SC cases a load of about 8 bar of IMEP was selected. This is about the load used during overtaking.

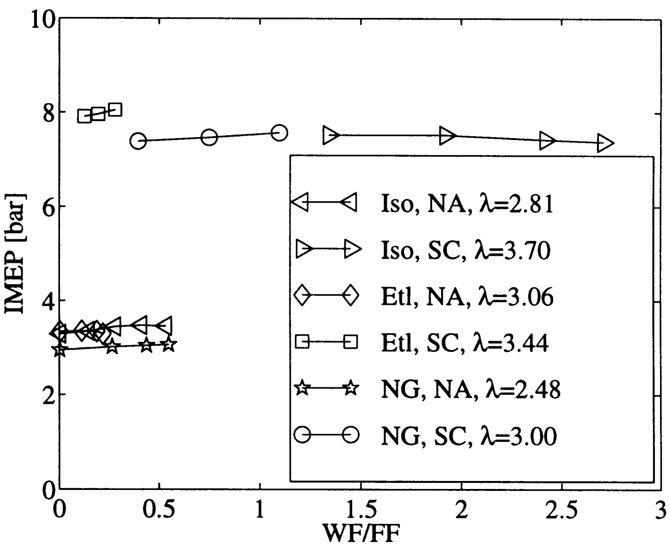

Figure 1. Net indicated mean effective pressures for the operating conditions.

CYLINDER PRESSURE – The cylinder pressure was measured with a pressure transducer for all operating conditions. With the pressure transducer mounted in the cylinder head, the cylinder pressure was recorded for 100 cycles, every 0.2 degrees crank angle. Increased flow of injected water retards the ignition timing, as the inlet mixture temperature becomes lower. Each pressure trace in Figures 2-7 is the mean pressure trace for 100 cycles. In each plot the trace with the highest maximum pressure corresponds to the cases with no or moderate amounts of water used. The maximum rate of pressure rise for each trace is given to the left in the figures. The highest value corresponds to the trace with the highest maximum pressure. In the supercharged case with isooctane the water injection did not affect the cylinder pressure much. This depends probably on the low inlet temperature, which restricts vapourization of the water, and prevents a uniform distribution.

RATE OF COMBUSTION – The cylinder pressure data was analyzed using a single zone heat release model. Details concerning the model can be found in [27]. The rate of heat release is plotted in Figures 2-7. In each plot the heat release curve with the earliest phasing corresponds to the condition with no or moderate amounts of water used. The ignition timing shows a strong dependence on the amount of water injected, except in the case with supercharged operation with isooctane. This was discussed above in the cylinder pressure paragraph. In the supercharged case with isooctane, there is a modest heat release at about 10 CAD before the main heat release starts. With isooctane a two-stage ignition mechanism occurs in some cases. This behaviour is dependent on the temperature and pressure conditions in the end of the compression stroke.

ONSET OF COMBUSTION (IGNITION TIMING) – There are many parameters that more or less affect the ignition timing. The strongest parameters for a certain fuel are the inlet temperature, inlet pressure and the compression ratio. Other parameters that affect the ignition timimg are the air/fuel ratio, EGR rate, coolant temperature and engine speed. Water injection retards the ignition timing as the temperature during the compression stroke becomes lower. Increased water injection has a similar effect on the ignition timing as lowering the inlet temperature. For the NA cases with higher inlet temperatures the effect of water injection is greater on the ignition timing than for the SC cases with lower inlet temperatures. For the cases with low inlet temperature is the cooling effect from the water only modest, probably due to only partial vapourization of the water.

COMBUSTION DURATION – The duration of the main combustion, 10-90 % heat released, is a good measure of how fast the combustion is. The combustion duration increases strongly with increased water flow, see figure 8. Even for the conditions with the longest duration, the combustion is still very fast. With HCCI the combustion duration is also affected by the onset of combustion [24]. If the combustion takes place mainly after TDC, the combustion rate will be slower, due to the increase in volume during the combustion.

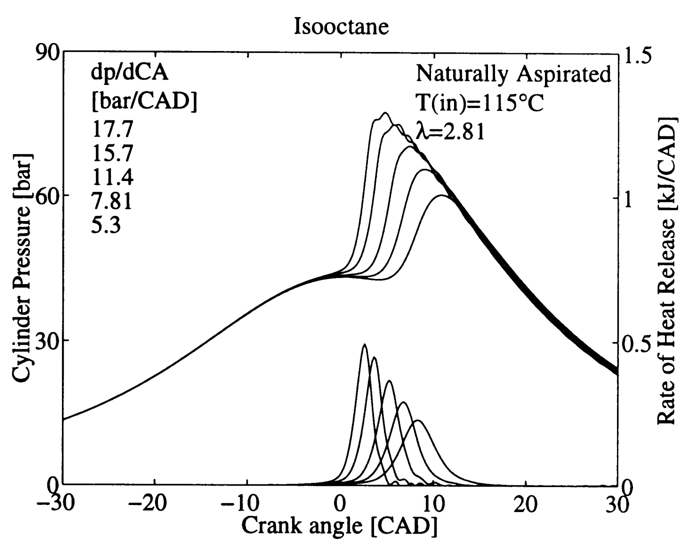

Figure 2. Cylinder pressure and rate of heat release for naturally aspirated operation with isooctane. The trace with the highest maximum pressure corresponds with the test condition with no water injection used.

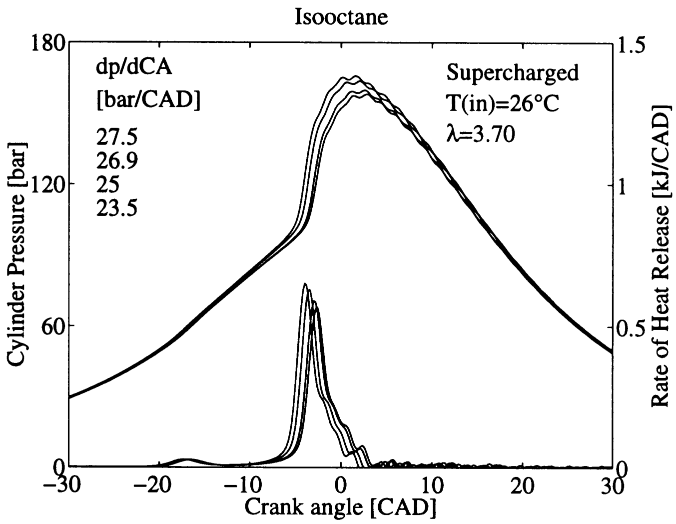

Figure 3. Cylinder pressure and rate of heat release for Supercharged operation with isooctane.

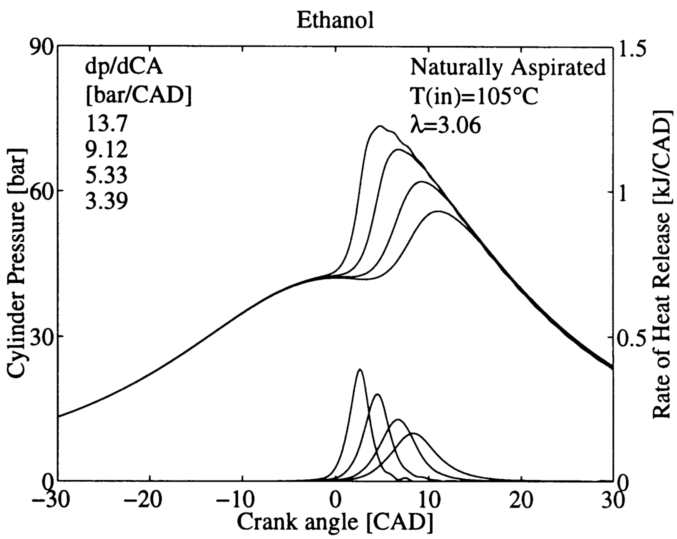

Figure 4. Cylinder pressure and rate of heat release for naturally aspirated operation with ethanol.

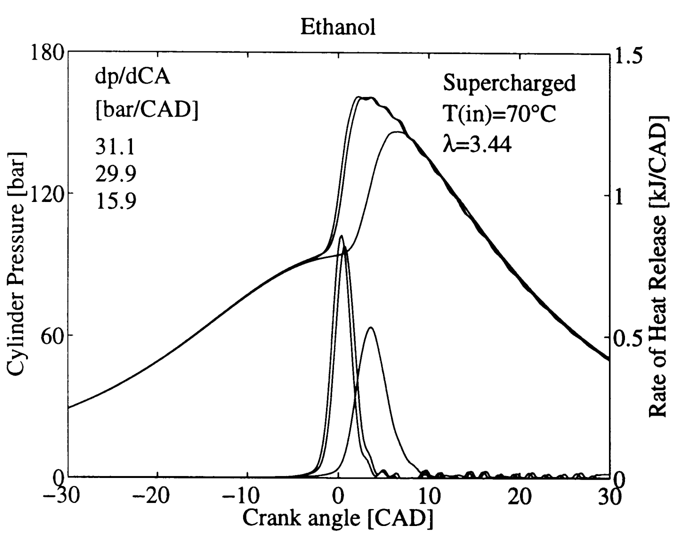

Figure 5. Cylinder pressure and rate of heat release for Supercharged operation with ethanol.

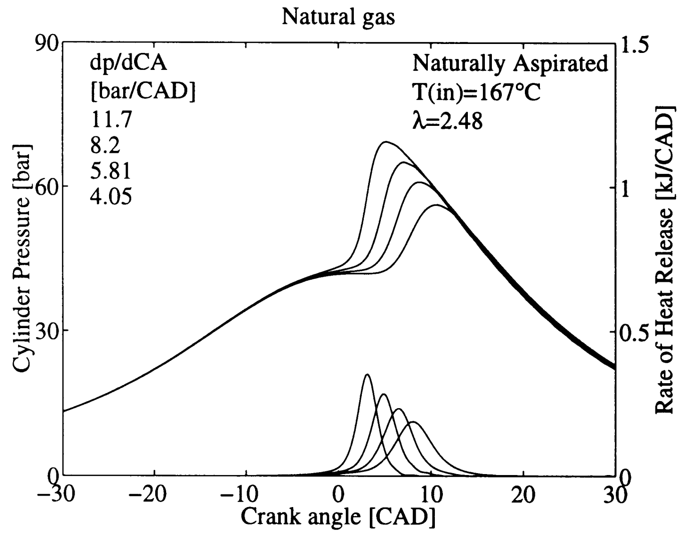

Figure 6. Cylinder pressure and rate of heat release for naturally aspirated operation with natural gas.

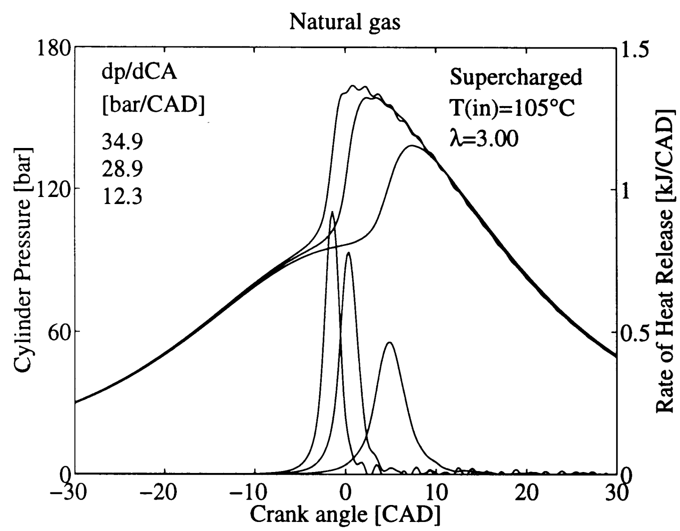

Figure 7. Cylinder pressure and rate of heat release for Supercharged operation with natural gas.

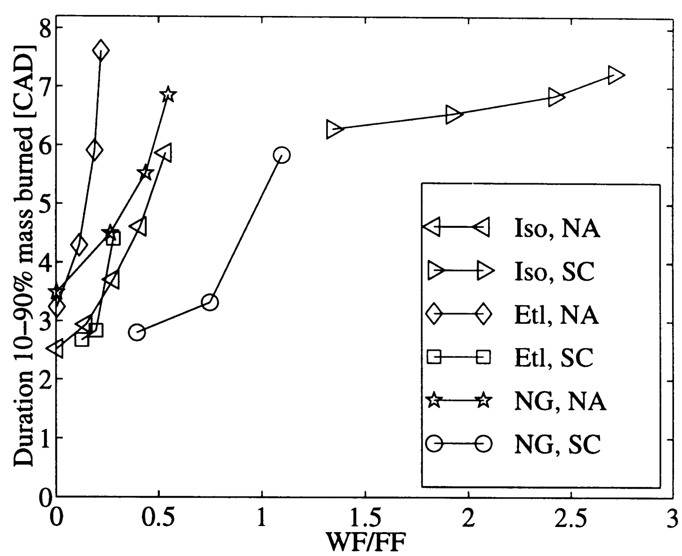

Figure 8. Duration of main combustion.

EFFICIENCYS (theoretical) – As the experiments were carried out with a six cylinder engine, which was converted to operate on one cylinder only, less reliable brake efficiencies can be measured. The other five cylinders were only motored. This means that the engine friction becomes very high compared to the work from the operating cylinder. Therefore only indicated efficiencies can be used. A schematic picture of the relation between the different efficiencys, as function of the onset of combustion (combustion phasing), is showed in Figure 9.

The *combustion efficiency*, η_c, is defined as :

$$\eta_c = \frac{Q}{m_f q_{lhv}}$$

where Q is the total heat released per cycle, m_f is fuel mass per cycle and q_{lhv} is the lower heating value per mass unit fuel.

When the onset of combustion occurs at TDC or later, the combustion takes place during the expansion. If the volume is increased during a homogeneous combustion, the combustion temperature becomes lower compared to combustion at constant volume. Especially at the end of combustion, the temperature becomes too low for complete oxidation of the fuel and much HC and CO is generated. This effect of increased volume during combustion becomes stronger with increased compression ratio, as the change in relative volume around TDC increases with higher compression ratio [22]. This means that increasing the compression ratio reduces the combustion efficiency. Some of the themodynamic benefit with a high compression ratio will be lost in reduced combustion efficiency.

The *gross indicated thermal efficiency*, $\eta_{t,i,g}$, is the ratio between the work on the piston during only the compression and expansion stroke, $W_{i,g}$, and the heat released:

$$\eta_{t,i,g} = \frac{W_{i,g}}{Q}$$

The compression ratio, phasing of the cylinder pressure and heat flux determine the thermal efficiency. Too early onset of combustion results in steep pressure rise with high maximum pressure and increased heat flux to the walls.

The *gross indicated efficiency*, $\eta_{i,g}$, is defined as the ratio between the work on the piston during only the compression and expansion stroke and the input fuel energy:

$$\eta_{i,g} = \frac{W_{i,g}}{m_f q_{lhv}} = \eta_c \eta_{t,,i,g}$$

The product of the combustion efficiency and thermal efficiency is the gross indicated efficiency. The gross indicated efficiency is calculated from gross IMEP and fuel flow.

The *net indicated efficiency*, $\eta_{i,n}$, is the gross indicated efficiency adjusted for pumping work. It is though the ratio between the work on the piston for all the four strokes, $W_{i,n}$, and input fuel energy:

$$\eta_{i,n} = \frac{W_{i,n}}{m_f q_{lhv}}$$

The net indicated efficiency is lower than the gross indicated efficiency when a engine is NA, but can be higher when SC.

The *brake efficiency*, η_b, is the net indicated efficiency adjusted for engine friction:

$$\eta_b = \frac{W_b}{m_f q_{lhv}} = \eta_{i,n}\eta_m$$

where W_b is the engine out work and η_m is the mechanical efficiency. The engine friction increases with increased cylinder pressure, therefore decreases the difference between η_b and η_i when the phasing of the cylinder pressure is retarded. In this paper the brake efficiency has not been measured. The indicated efficiency in figure 9 represents both net and gross indicated efficiency. The only difference between the gross and net is the level of the curve.

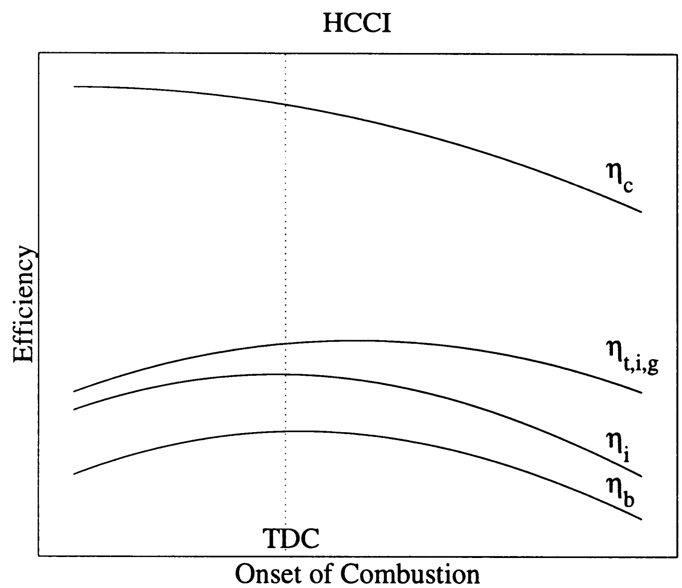

Figure 9. Schematic picture of the influence of ignition timing on the different efficiencys.

COMBUSTION EFFICIENCY – The combustion efficiency is evaluated from the exhaust gas composition and it is a measure of how complete the combustion is. The combustion *inefficiency* is mainly dependent on the concentration of unburned hydrocarbons and CO in the exhaust gases. Figure 10 shows the combustion efficiency. The use of water injection reduces the combustion efficiency. The main reason for this is the decrease in temperature in the combustion chamber and the retarded combustion phasing. The temperature becomes too low to oxidize the fuel, especially near the combustion chamber walls. The lower combustion efficiency can also depend on incomplete vaporizing of the fuel when the humidity of the air becomes higher.

GROSS INDICATED EFFICIENCY – The gross indicated efficiency was evaluated from the fuel flow and the indicated mean effective pressure during only the compression and expansion stroke. Pumping work and engine friction is not included here. For a given compression ratio, the indicated efficiency is mainly dependent on the combustion efficiency and the combustion phasing. With HCCI, the combustion quality is also strongly affected by the timing of the onset of combustion. This means that the most favourable combustion phasing for thermal efficiency does not give the highest combustion efficiency. This is the explaination why the indicated efficiency in some cases increases even though the combustion quality gets poorer. Figure 11 shows the measured efficiency.

NET INDICATED EFFICIENCY – The net indicated efficiency is obtained from the fuel flow and the indicated mean effective pressure for all four strokes. The pumping work is included here. This means that the net indicated efficiency is lower than the gross indicated efficiency when the engine is operated naturally aspirated, see Figure 12. For the supercharged cases, the net indicated efficiency is higher than the gross indicated efficiency. This depends on the use of an external supercharging system, without the usage of exhaust gas backpressure control.

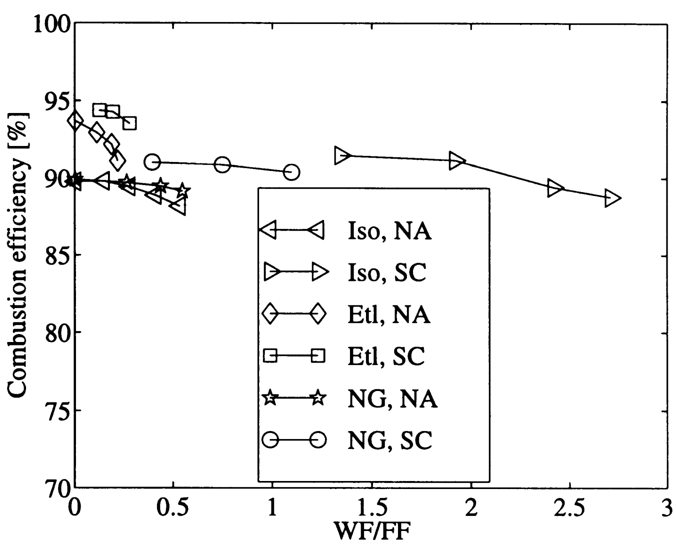

Figure 10. Combustion efficiency evaluated from the exhaust gas analysis.

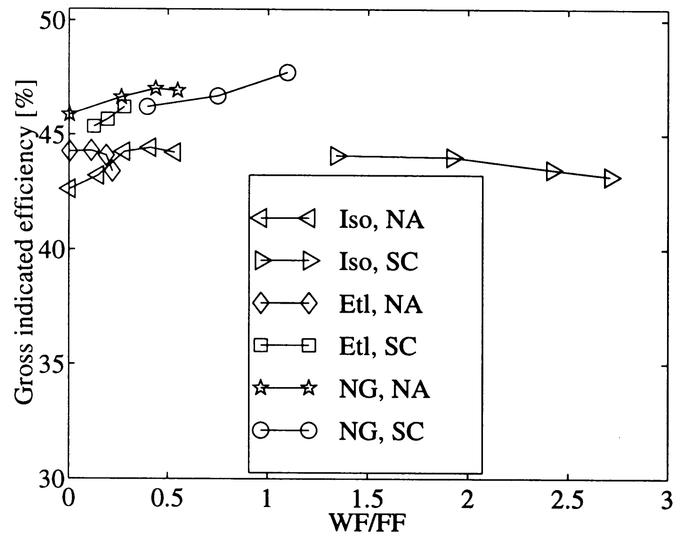

Figure 11. Measured gross indicated efficiency.

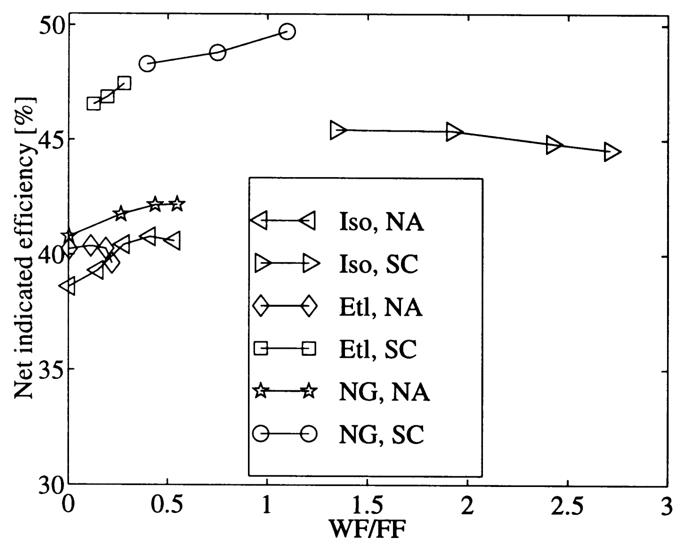

Figure 12. Measured net indicated efficiency.

NOx EMISSIONS – With a homogeneous combustion of a premixed mixture, the temperature is expected to be the same in the entire combustion chamber, except near the walls. This in combination with very lean mixtures, gives a low maximum temperature during the cycle. NOx formation is very sensitive to the temperature history during the cycle. At temperatures over 1800 K, the NOx formation rate increases rapidly with increased temperature. To study the the NOx formation as function of the maximum combustion temperature, the inlet temperature was varied, to get ignition at different CAD positions. The inlet temperature was varied from 150°C to 180°C, other parameters were constant. No water injection was used. Figure 12 shows NOx as function of the maximum temperature during the cycle. The NO formation rate govern by the Zeldovich mechanism [28] is also ploted in the figure. The trend of the measured curve and the Zeldovich curve is similar. Figure 14 shows the specific NOx-emissions and Figure 15 shows the volume fraction of NOx for the current test cases. Natural gas requires higher temperature for auto-ignition than iso-octane and ethanol, due to the higher octane number. This results in a higher maximum temperature, and therefore higher NOx emissions. For the supercharged case with iso-octane only fractions of a ppm NOx is generated. This depends on the low inlet temperature in combination with a very lean mixture. The maximum temperature is here below the temperature limit (about 1800 K), where formation of thermal NOx starts.

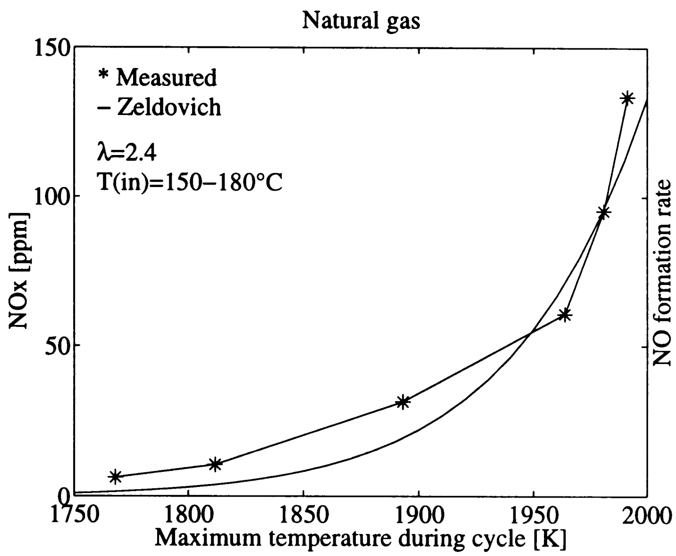

Figure 13. NOx formation as function of maximum temperature (evaluated from the cylinder pressure) during cycle for natural gas. CR=18:1.

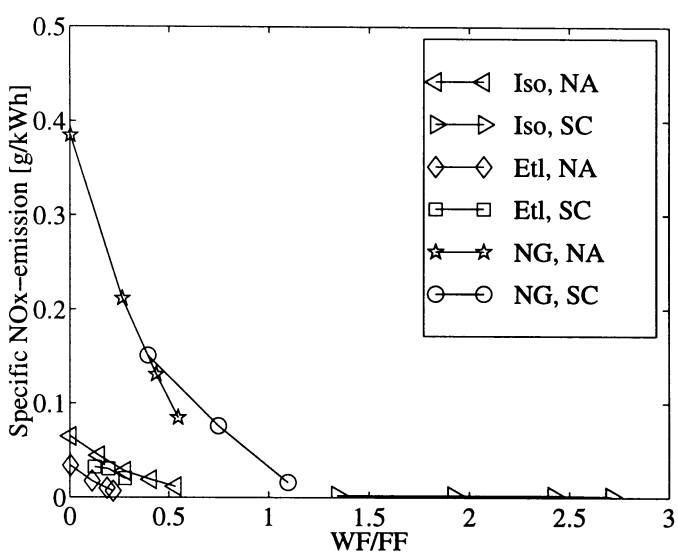

Figure 14. Specific NOx emission.

HC EMISSIONS – The low combustion temperature prevents NOx formation, but the combustion temperature becomes too low to fully oxidize the fuel completely. This low combustion temperature results in high emissions of unburned hydrocarbons. Even if the mixture is well prepared and close to homogeneous, the combustion rate near the walls will probably be slower, due to lower temperature near the walls. If we assume that the combustion rate is slower near the walls, much fuel in this region will not be able to burn completely, as the overall temperature decreases due to the volume increase during the expansion stroke. Much of the HC emissions will probably be descended from the wall regions. Figure 16 shows the specific HC emissions. The main trend is that HC increases when the water injection is increased, as water decreases the already low combustion temperature. The increase in HC can also partly be explained by incomplete vapourization of the fuel when the humidity of the inlet air becomes very high.

CO EMISSIONS – With HCCI, CO is very dependent on λ and on preheating. Close to the rich limit for HCCI and with early combustion phasing, very little CO is generated. But close to lean limit, very much CO can be generated. With increased water injection, CO increases drasticly, as can be seen in Figure 17. This depends mainly on the lower combustion temperature and the later combustion phasing. At the end of combustion, the temperature becomes too low for complete oxidation and much CO is generated. An earlier study on HCCI by the authors, showed that at the lean limit for HCCI combustion, more CO than CO_2 can be generated [24].

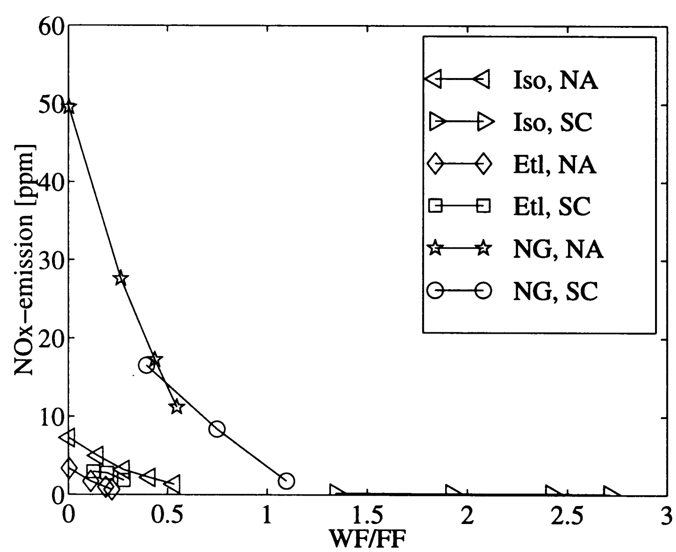

Figure 15. Volume fraction NOx emission.

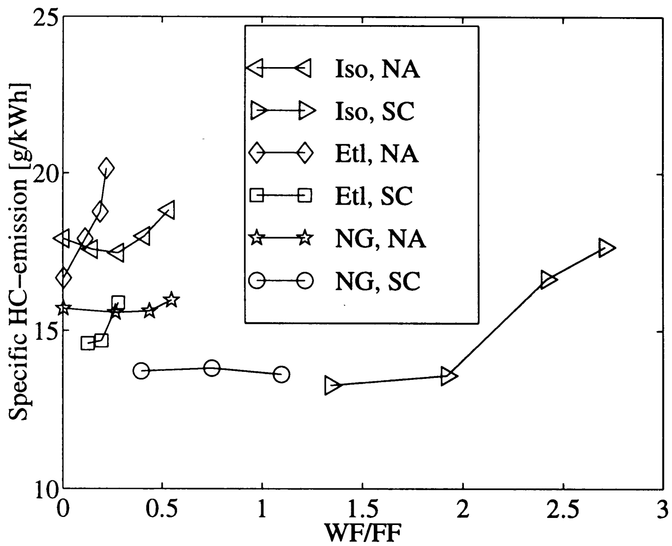

Figure 16. Specific HC emission.

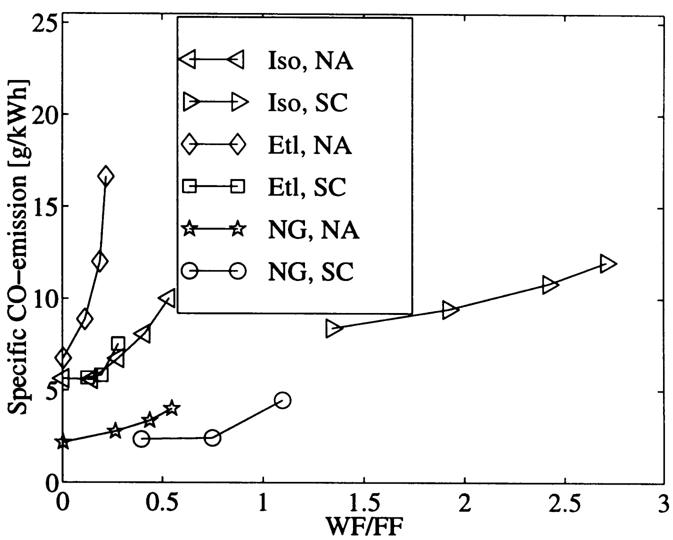

Figure 17. Specific CO emission.

DISCUSSION

The major problem with HCCI is not the high emissions of unburned hydrocarbons. The major problem is controlling the ignition timing over a wide load and speed range, as there is no direct control over the ignition timing as in a SI and diesel engine. There are a lot of parameters that affect the ignition timing in a HCCI engine. By adjusting the these parameters fast and correctly it would possible to control the ignition timing over the entire load and speed range of a HCCI engine.

The major advantages with HCCI is high efficiency and very low NOx emissions. HCCI can achieve efficiencys comparable to values achieved in diesel engines at part load. When the onset of combustion occur at TDC or later only a few ppm NOx is generated. At very light loads only fractions of a ppm NOx can be generated.

The results of the experiments show that water injection can be used in controlling the ignition timing and to slow down the combustion rate. Water injection retards the ignition timing, as the temperature during compression becomes lower. The results also show that water injection has a greater effect on the ignition timing when higher inlet temperatures are used. This depends probably on a better vapourization and a more uniform distribution of the water at higher inlet temperatures. In this study we used a conventional fuel injector for the water injection. When high water flow rates are used in combination with low inlet temperatures, this is probably not the best method for water supply, when complete vapourization and uniform ditribution are desired.

CONCLUSIONS

Water injection can be used to retard the ignition timing and to slow down the rate of combustion for HCCI combustion, but only in a narrow range. Water injection can also be used to increase the upper load limit for HCCI, as the cooling effect from the water slows down the combustion rate and reduces peak pressure. Water mass flows up to three times the fuel mass flow have been used.

There are, however, some drawbacks with the use of water injection. The already high emissions of unburned hydrocarbons increased when water injection was applied. The CO emissions also increased. This indicates that the combustion quality gets poorer with increased water injection. The NOx emissions are overall very low for HCCI. For the richer cases, when NOx is generated to some degree, the use of water injection can reduce NOx to only fractions of a ppm.

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