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ABSTRACT

The major advantages with Homogeneous Charge Compression Ignition, HCCI, is high efficiency in combination with low NOx-emissions. The major drawback with HCCI is the problem to control the ignition timing over a wide load and speed range. Other drawbacks are the limitation in attainable IMEP and relativly high emissions of unburned hydrocarbons. But the use of Exhaust Gas Recycling (EGR) instead of only air, slows down the rate of combustion and makes it possible to use lower air/fuel ratio, which increases the attainable upper load limit. The influence of mixture quality was therefore experimentally investigated. The effects of different EGR rates, air/fuel ratios and inlet mixture temperatures were studied. The compression ratio was set to 18:1. The fuels used were iso-octane, ethanol and commercially available natural gas. The engine was operated naturally aspirated mode for all tests. Stable operation could be achieved with ethanol at about 5 bar of IMEP with a gross indicated efficiency close to 50 %. Stable operation of the engine was achieved with over 60 % EGR.

INTRODUCTION

The Homogeneous Charge Compression Ignition, HCCI, is a promising alternative for combustion in the internal combustion engine. The HCCI concept is a hybrid of the successful Spark Ignition (SI) and Compression Ignition (CI) engine concepts. As in a diesel engine, the fuel is exposed to high enough temperature for autoignition, but in contradiction to the diesel engine type, a homogeneous fuel/air-mixture is used. The homogeneous mixture is created in the intake system, using a low pressure injection system. To limit the rate of combustion, very diluted mixtures must be used.

The high dilution can be achieved by a high air/fuel-ratio and/or with Exhaust Gas Recycling (EGR). If the mixture is too rich, the rate of combustion becomes too fast and will generate knock related problems. A too lean mixture will lead to incomplete combustion or misfire. In SI engines, large cycle-to-cycle variations occur since the early flame development varies significantly, due to mixture inhomogeneities in the vicinity of the spark plug and variations in flow conditions [1]. With HCCI, cycle-to-cycle variations of combustion are very small, since combustion initiation takes place at many points at the same time. The whole mixture burns close to homogeneous at the same time. In this way, unstable flame propagation is avoided.

PREVIOUS WORK – Several studies on HCCI have been performed on two stroke engines [2, 3, 4, 5, 6, 7] showing that HCCI combustion has great potential for further development. Several studies on four stroke engines have also been carried out and a briefly description follows:

Najt et al. also studied the effect of EGR on HCCI combustion. They showed that, increasing the oxygen concentration by decreasing the amount of EGR, advanced the ignition [8].

Thring investigated possible combinations of equivalence ratio and EGR [9]. He used EGR rates in the range of 13 to 33 per cent. He also suggested an interesting engine concept, an engine that uses conventional SI operation at high loads and HCCI operation at part load.

The first to test HCCI on an real production engine was Stockinger and et al. in 1992 [10]. They used a standard 1.6 litre VW engine which was converted to HCCI operation with preheated intake air. The part load efficiency increased from 14 to 34 %.

Interesting research on HCCI was made by Aoyama et al. in 1996 [11], they called it PCCI (Premixed Charge Compression Ignition). They made comparisons between PCCI, diesel and Gasoline Direct Injection (GDI). At optimum λ, PCCI had the lowest fuel consumption. NOx emissions were much lower with PCCI, but HC emissions were higher.

Gray and Ryan studied HCCI combustion with diesel fuel [12]. Succesful operation was achieved with EGR rates in the range from 0 to 50 %. They also tested ceramic coating of the combustion chamber.

Different combustion methods, which partly uses compression ignition of a homogeneous charge, have recently been studied by several different authors [13, 14, 15, 16, 17, 18, 19, 20, 21].

Earlier research on HCCI in comparison with SI operation by the authors showed that the indicated efficiency was much higher with HCCI compared to SI operation at part load [22]. Very little NOx was generated with HCCI, but HCCI generated more HC and CO.

The effect of supercharging on HCCI was studied to investigate the how much the attainable load could be increased and how supercharging affected the emissions of unburned hudrocarbons [23]. The attainable load for HCCI was dramatically increased and the emissions of unburned hydrocarbons decreased, when supercharging was used.

In summary, previous studies on HCCI show that it can be used in both two-stroke and four-stroke engines. The usable operating conditions are, however, limited due to the requirements of diluted mixtures to slow down the combustion rate. In two-stroke engines, this results in a requirement of high residual gas concentration, which is found at part load. In four-stroke engines, that normally have much less residual gas, ultra-lean mixtures or large amounts of EGR must be used. The emissions of NOx have been reported to be very low, in most cases only a few ppm. The emissions of unburned hydrocarbons are, however, not low. HCCI produces in general two to three times more unburned hydrocarbons compared to SI operation [22].

PRESENT WORK – The aim of this work was to invetigate the influence of mixture quality, especially the use of EGR, on HCCI combustion. The main objective of the investigation was to explore how different EGR rates affect the ignition and combustion process. The fuels used were iso-octane, ethanol and natural gas (table 1), with octane numbers of 100, 106 and 120 respectively. A high compression ratio, 18:1, was selected to achieve auto-ignition near Top Dead Center (TDC) with resonable amount of inlet air preheating. This compression ratio was selected as comprimize to be suitable for all three fuels. With natural gas, a higher compression ratio could be used, to reduce the preheating requirement. It was decided to use a four-stroke engine operated at low engine speed. The engine speed was set to 1000 rpm during all experiments.

Table 1. Composition of the natural gas used.

Component	Vol. %	Mass %
Methane	91.1	81.0
Ethane	4.7	7.9
Propane	1.7	4.2
n-Butane	1.4	4.7
Nitrogen	0.6	0.9
Carbon dioxide	0.5	1.2

EXPERIMENTAL APPARATUS

THE ENGINE – A diesel engine was converted to run in HCCI mode. The engine originates from a Volvo TD100 series diesel. The in-line six was, however, modified to operate on one cylinder only. This arrangement gives a robust and inexpensive single-cylinder engine, but at the cost of less reliable brake specific results. With a pressure transducer, indicated results can be used instead. The major engine specifications are shown in table 2.

Table 2. Geometric properties of the test engine.

Displaced Volume	1600 cm^3
Bore	120.65 mm
Stroke	140 mm
Connecting Rod	260 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)

To initiate HCCI combustion, high temperature is necessary. HCCI is not dependent on turbulence for flame propagation or swirling flow for diffusion combustion. As a consequence, the simplest possible combustion chamber geometry was selected, a flat piston crown giving a pancake combustion chamber. The inlet mixture was preheated with an electrical heater. The energy needed for preheating has not been included in the efficiency calculation.

THE EGR SYSTEM – To make the use of EGR possible, when operating the engine unthrottled, it is necessary to increase the pressure in the exhaust manifold. This was done by slightly throttling the exhaust side. One drawback with throttling the exhaust is increased pumping work, which reduces the efficiency. The amount of EGR was regulated by a valve in the pipe, which connected the exhaust and inlet system. The amount of EGR was evaluted by measuring the CO_2 concentration in the inlet and exhaust. The amount of EGR is defined as:

$$EGR = \frac{\dot{m}_e}{\left(\dot{m}_a + \dot{m}_f + \dot{m}_e\right)}$$

where $\dot{m}_e$ is the mass flow recycled exhaust gases, $\dot{m}_a$ is the mass air flow and $\dot{m}_f$ is the mass fuel flow.

RESULTS

AIR/FUEL EQUIVALENCE RATIO, λ – With HCCI, highly diluted mixtures must be used, in order to slow down the rate of the chemical reactions enough, to slow down the rate of combustion. The dilution can be made with air and/or EGR. Figures 1-3 shows usable combinations of λ and EGR for the three fuels used. λ is the ratio of the actual air/fuel ratio to the stoichiometric air/fuel ratio. The values are calculated from the exhaust gas composition. The amount of EGR was varied for different constant fuel flows, as indicated by the legends in the figures. This means that when the EGR is increased, λ decreases.

INLET AIR TEMPERATURE – With HCCI, the inlet air has to be preheated to achieve auto-ignition with the selected fuels and compression ratio. The degree of preheating is dependent on the compression ratio, manifold pressure, fuel type, λ and EGR rate. With higher amount of EGR, higher inlet temperature is necessary to initiate combustion, as the chemical reactions become slower. The inlet temperature was adjusted to get combustion initiation near TDC for each operation point. Figures 4-6 shows the inlet temperatures used.

START OF COMBUSTION – In SI engines and in diesel engines the Start Of Combustion, SOC, is determined by the ignition timing and the injection timing, respectively. In a HCCI engine there is no direct control of the start of combustion, as the ignition process relies on a spontaneous auto-ignition of a homogeneous charge. The SOC is dependent on many parameters. The most important parameters that affect the SOC is the:

- compression ratio
- inlet mixture temperature
- inlet manifold pressure
- fuel
- air/fuel ratio
- amount of EGR
- engine speed
- coolant temperature

ENGINE LOAD, IMEP – The high dilution grade used to slow down the rate of combustion give a limitation in attainable engine load. Stable operation could be achieved with ethanol at about 5 bar of IMEP with high amount of EGR. With natural gas, it was hard to achieve stable operation with high amounts of EGR at proper ignition timings. With iso-octane, stable operation could be achieved at about 4.5 bar of IMEP with proper ignition timing. The engine load versus EGR is showed in figures 7-9. One way to increase the attainable IMEP is with supercharging, but the low exhaust gas temperatures, due to the diluted mixtures used with HCCI, is a problem if a turbo charger should be used. But the exhaust gas temperature increases slightly with increased EGR. The increase in exhaust gas temperature can partly be explained by the later combustion phasing. The exhaust gas temperatures is showed in figures 4-6.

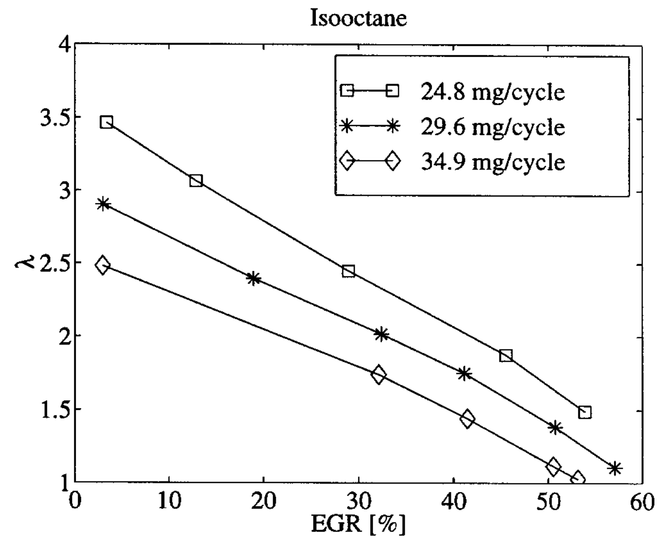

Figure 1. λ versus EGR for iso-octane.

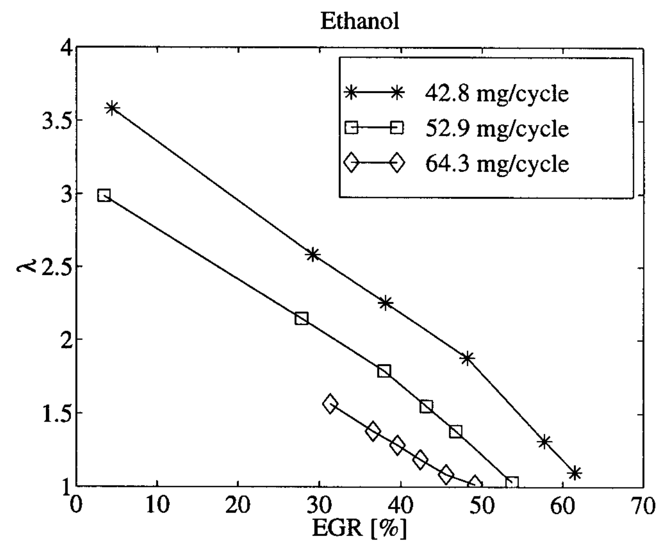

Figure 2. λ versus EGR for ethanol.

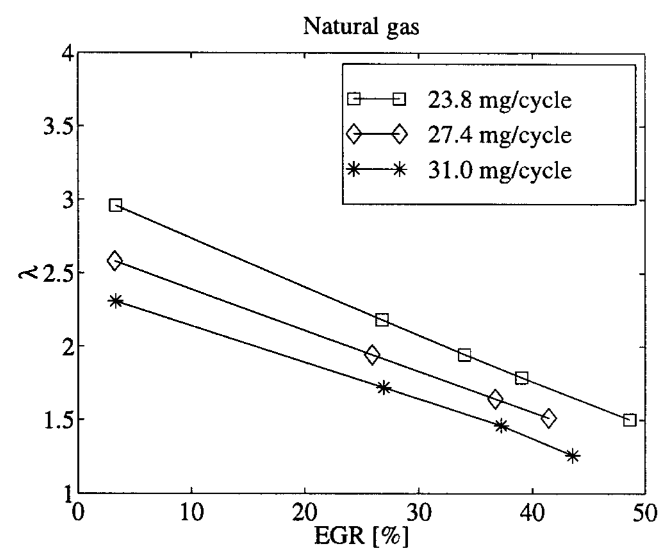

Figure 3. λ versus EGR for natural gas.

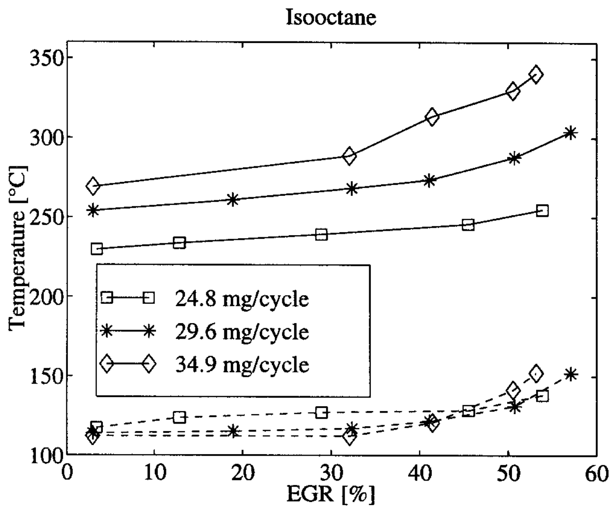

Figure 4. Inlet mixture temperature (dashed lines) and exhaust gas temperature (solid lines) running on iso-octane.

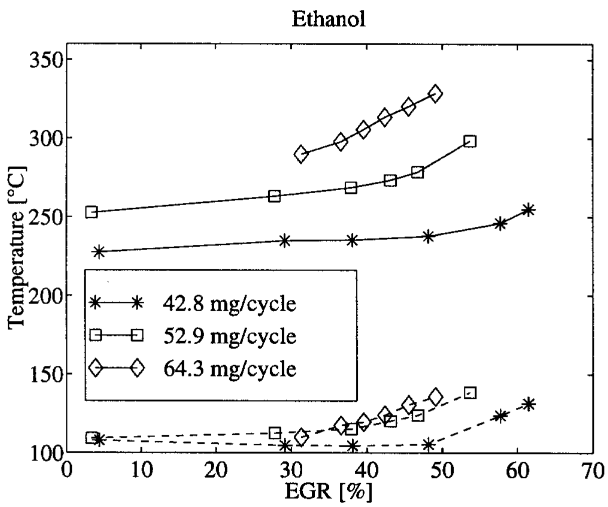

Figure 5. Inlet mixture temperature (dashed lines) and exhaust gas temperature (solid lines) running on ethanol.

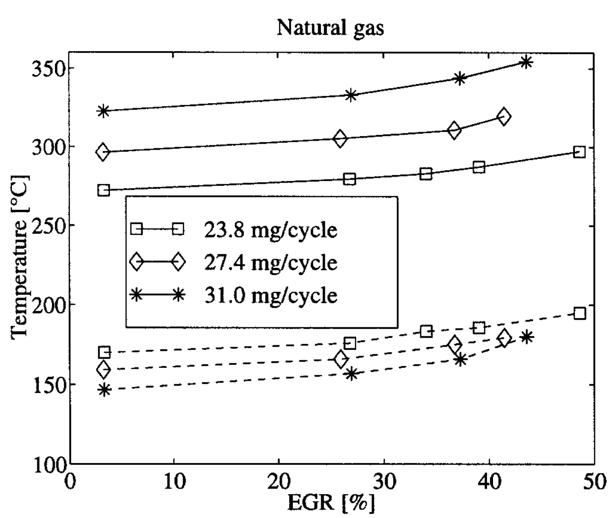

Figure 6. Inlet mixture temperature (dashed lines) and exhaust gas temperature (solid lines) running on natural gas.

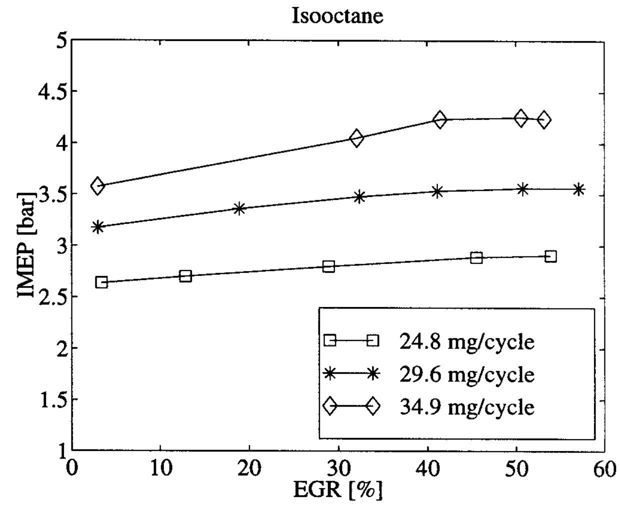

Figure 7. Net indicated mean effective pressure for iso-octane.

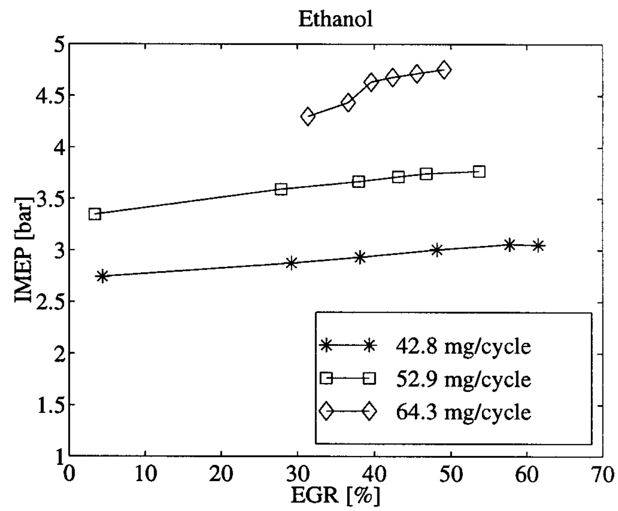

Figure 8. Net indicated mean effective pressure for ethanol.

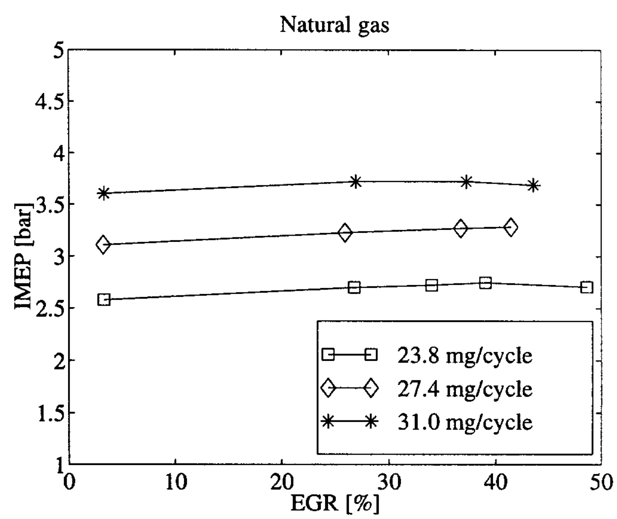

Figure 9. Net indicated mean effective pressure for natural gas.

CYLINDER PRESSURE – The cylinder pressure was measured for all operating conditions. The cylinder pressure was recorded for 100 cycles, every 0.2 degrees crank angle with a pressure transducer mounted in the cylinder head. With higher amounts of EGR the combustion becomes slower, resulting in lower maximum pressure and later phasing of the peak pressure. Figures 10-18 shows the mean pressure traces for 100 cycles. In the figures the curve with highest peak pressure corresponds to test conditions with no EGR used. The maximum rate of pressure rise for each trace is given to the left in the figures. The highest value corresponds with the test conditions with no EGR used. For the cases with none or moderate amounts of EGR, the rate of pressure rise is extremly fast, which generates much combustion noise. The maximum rate of pressure rise shows a very strong dependence on the combustion phasing.

RATE OF COMBUSTION – The cylinder pressure was analysed using a single zone heat release model, which gave the rate of heat release. Details concerning the model can be found in [24]. The phasing of the heat release is later in the cycle for the cases with high amount of EGR, which can be seen in figures 10-18. The rate of heat release is also slower with higher amounts of EGR. This leads to lower maximum pressure and smoother engine operation. The slower rate of heat release, with higher EGR amounts, is mainly dependent on the later combustion phasing. Later combustion phasing gives slower combustion rate.

COMBUSTION DURATION – The duration of 10-90% heat released, main combustion, is a very good parameter to use for discussions on fast or slow combustion. Analysis of the cylinder pressure gave the combustion duration. In figures 19-21 one can see that the combustion duration increases with higher amount of EGR, though the combustion duration is very short for all cases tested. With no EGR used, the combustion duration is extremly short, 2 to 3 crank angle degrees. This very fast combustion generates knock related problems, and the engine operation becomes very noisy. This can also affect the wear of the engine, but for the relavivly short periods that we have operated the engine at this hard conditions, we have not been able to detect any abnormal wear of the engine or any damages to the piston.

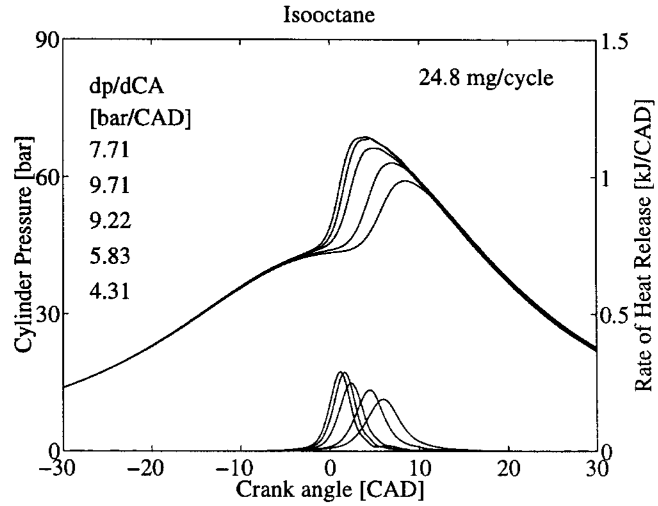

Figure 10. Cylinder pressure for iso-octane, fuel flow 24.8 mg/cycle.

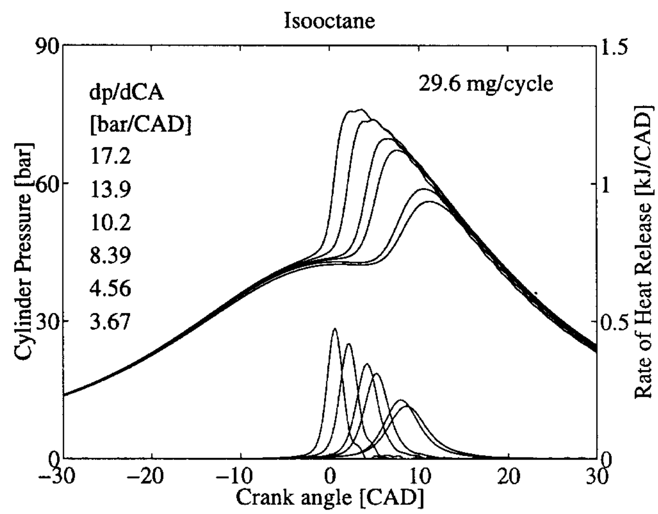

Figure 11. Cylinder pressure for iso-octane, fuel flow 29.6 mg/cycle.

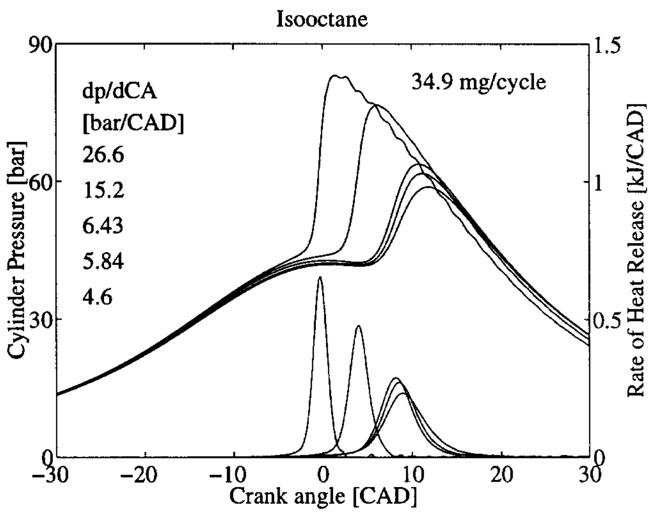

Figure 12. Cylinder pressure for iso-octane, fuel flow 34.9 mg/cycle.

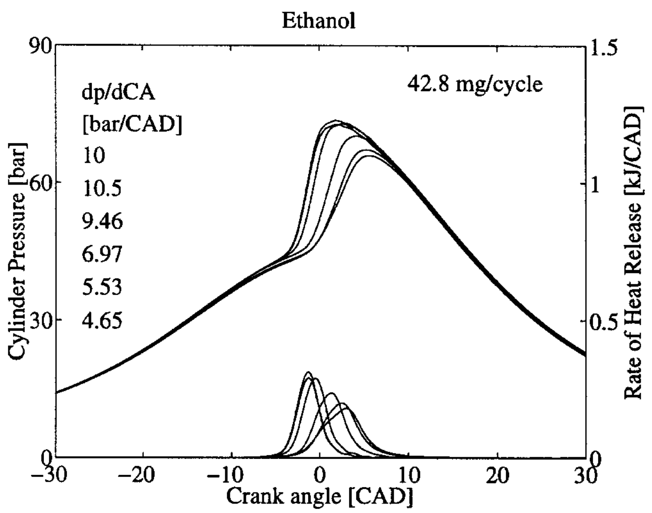

Figure 13. Cylinder pressure for ethanol, fuel flow 42.8 mg/cycle.

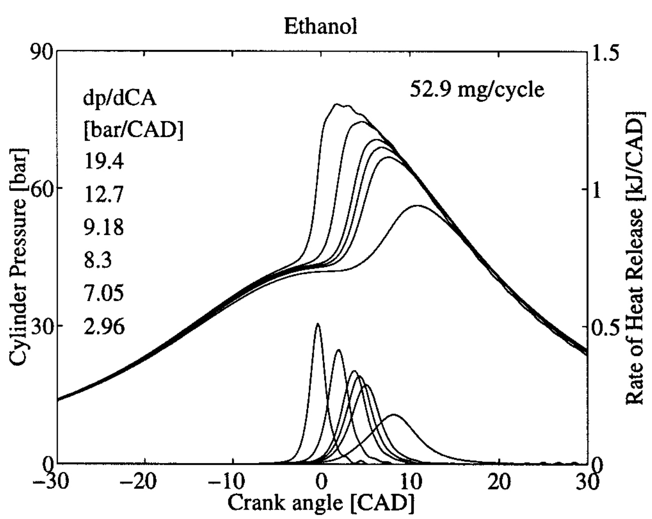

Figure 14. Cylinder pressure for ethanol, fuel flow 52.9 mg/cycle.

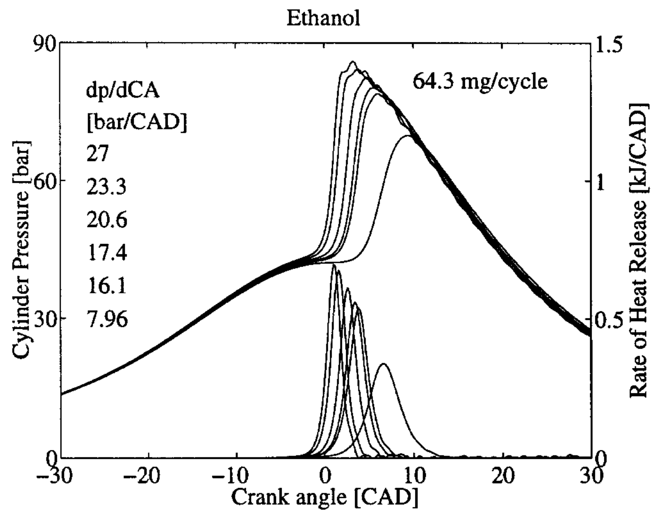

Figure 15. Cylinder pressure for ethanol, fuel flow 64.9 mg/cycle.

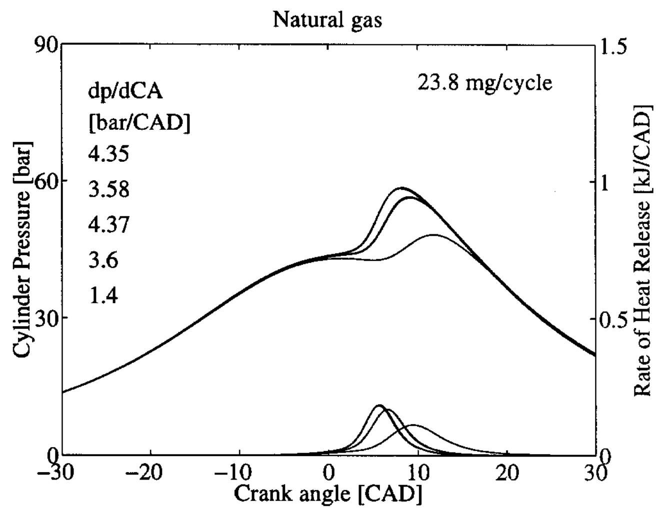

Figure 16. Cylinder pressure for natural gas, fuel flow 23.8 mg/cycle.

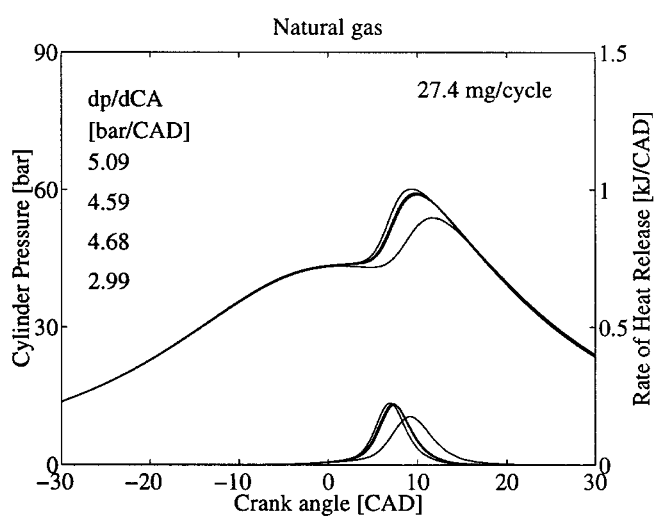

Figure 17. Cylinder pressure for natural gas, fuel flow 27.4 mg/cycle.

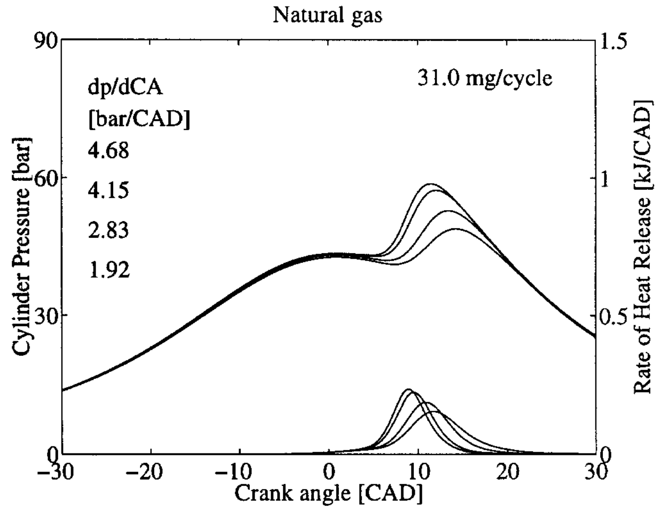

Figure 18. Cylinder pressure for natural gas, fuel flow 31.0 mg/cycle.

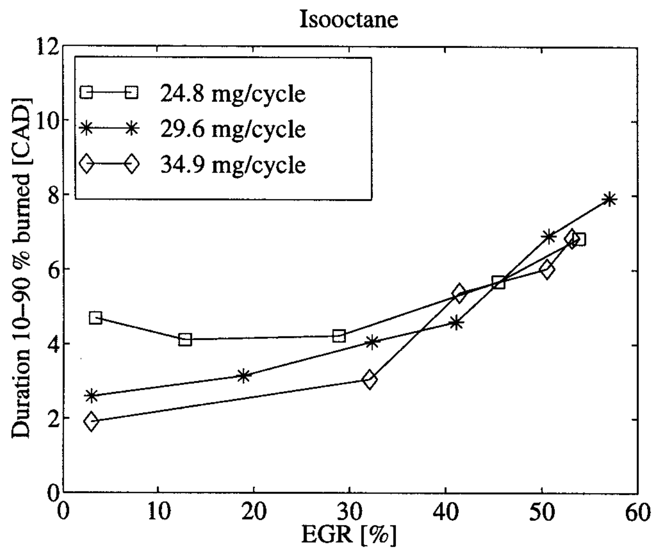

Figure 19. Combustion duration for iso-octane.

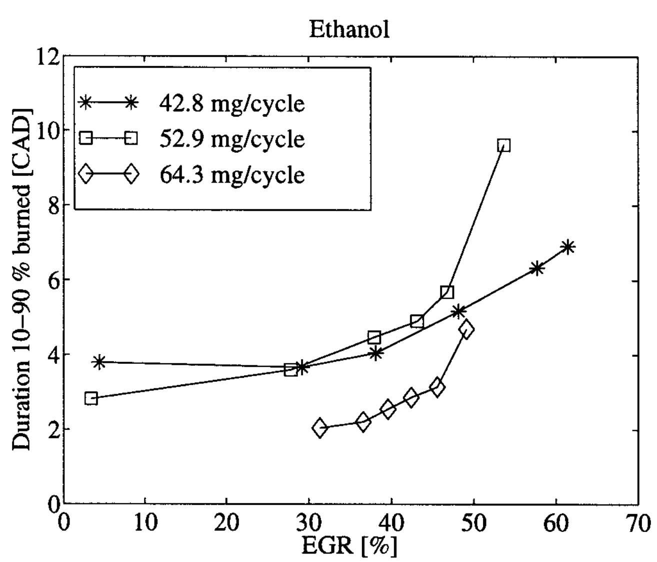

Figure 20. Combustion duration for ethanol.

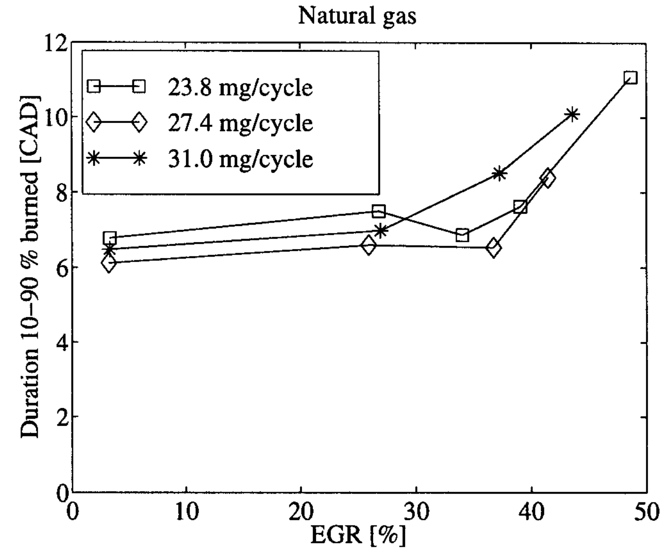

Figure 21. Combustion duration for natural gas.

COMBUSTION EFFICIENCY – The combustion efficiency is evaluated from the exhaust gas analysis. The efficiency is mainly dependent on the amount of unburned hydrocarbons and CO in the exhaust gases. The combustion efficiency can be seen in figures 22-24. HCCI combustion with ethanol shows the highest efficiency, nearly 98 % at a few test conditions. With iso-octane the combustion efficiency is a bit lower, targeting 95 % as best. Natural gas shows an even lower efficiency. The overall trend is that the combustion efficiency increases with increasing EGR. For very high EGR rates (over 50 %), the combustion efficiency shows a slightly decrease, in some cases. HCCI combustion is very dependent on the inlet temperature. This means that the inlet temperature setting also affect the combustion efficiency. This should be considered, as the inlet temperature was increased with increased EGR, to get SOC near TDC and to get satisfying combustion.

GROSS INDICATED EFFICIENCY – The gross indicated efficiency was evaluated by measuring the fuel flow and the indicated mean effective pressure during the compression and expansion strokes. The pumping work and engine friction are not included here. Figures 25-27 shows the evaluated efficiencies. With ethanol the highest efficiency is obtained, nearly 50 % in one case. With iso-octane the efficiency is overall lower than with ethanol. The best obtained value with iso-octane is about 47 %. With Natural gas the lowest efficiency is obtained here. The lower efficiency obtained with natural gas is mainly dependent on the lower combustion efficiency. The higher inlet temperature used with natural gas, giving an overall higher cycle temperature, increases the heat losses, which also reduces efficiency. The overall trend is that the efficiency increases with increasing EGR. This depends on the increasing combustion efficiency.

NET INDICATED EFFICIENCY – The net indicated efficiency is obtained from the fuel flow and the indicated mean effective pressure during all four strokes. The gas exchange process is included here. This means that the net indicated efficiency is lower than the gross indicated efficiency, as can be seen in figures 28-30. The exhaust pipe was slightly throttled to increase the pressure of the exhaust gases. This was done to create a pressure difference between the exhaust gas and the inlet air, to feed the exhaust gases into the intake manifold. Throttling at the exhaust side increases the pumping work, which reduces the net efficiency. But the backpressure in the exhaust pipe was only increased by about 5 kPa.

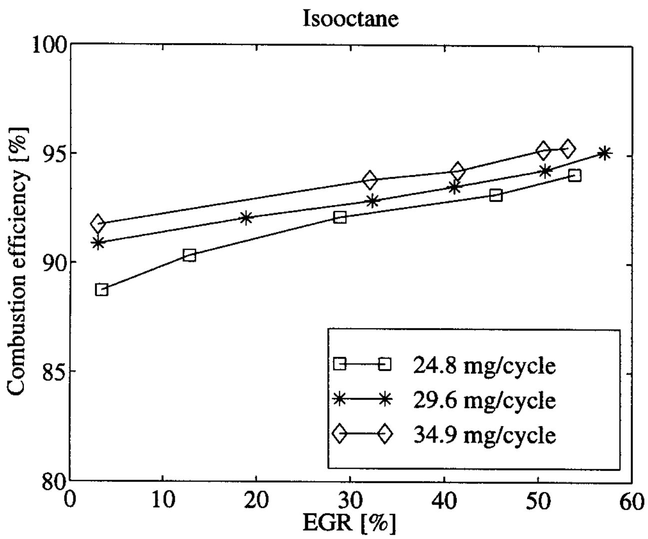

Figure 22. Combustion efficiency with iso-octane.

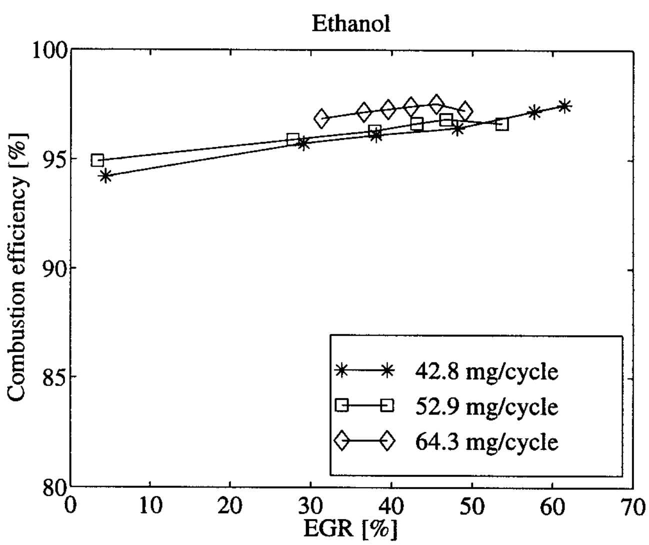

Figure 23. Combustion efficiency with ethanol.

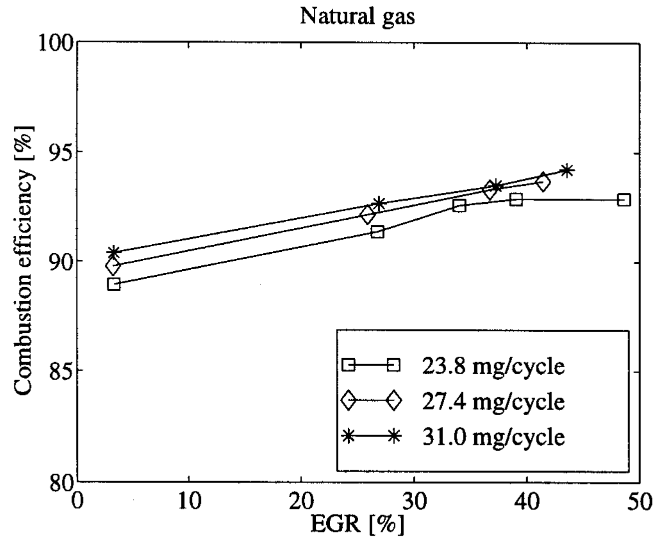

Figure 24. Combustion efficiency with natural gas.

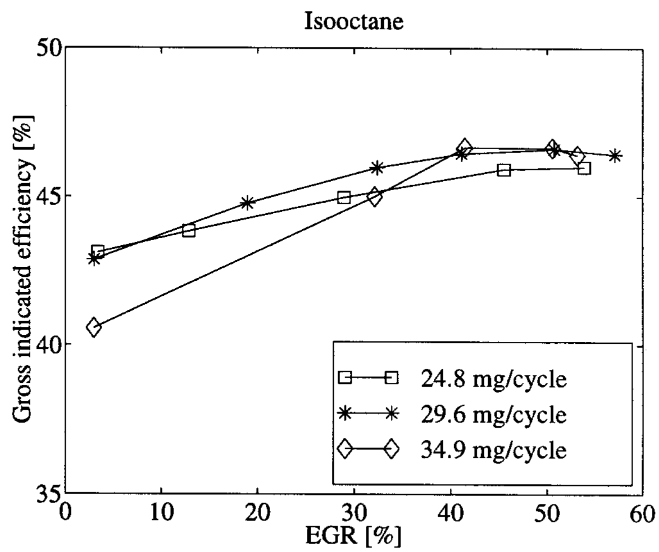

Figure 25. Gross indicated efficiency for iso-octane.

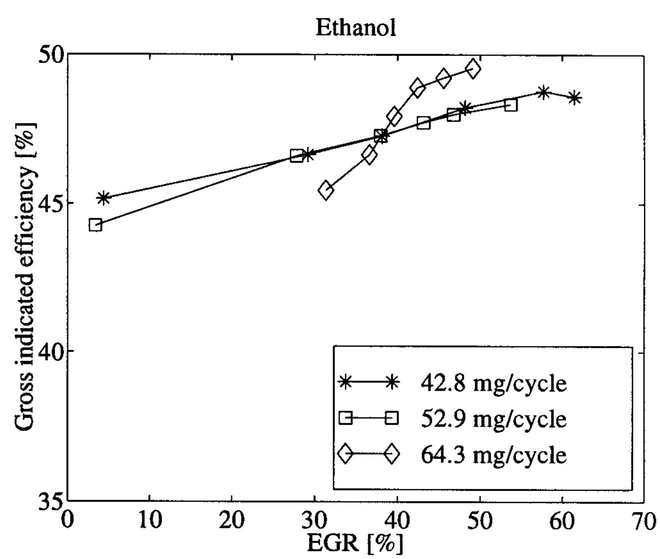

Figure 26. Gross indicated efficiency for ethanol.

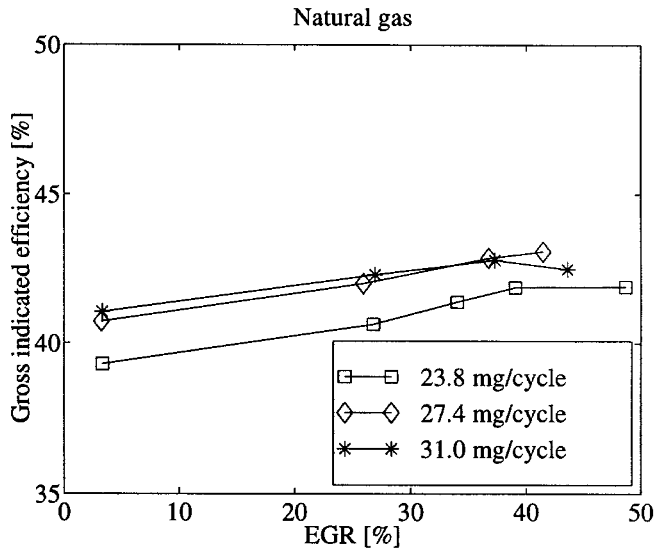

Figure 27. Gross indicated efficiency for natural gas.

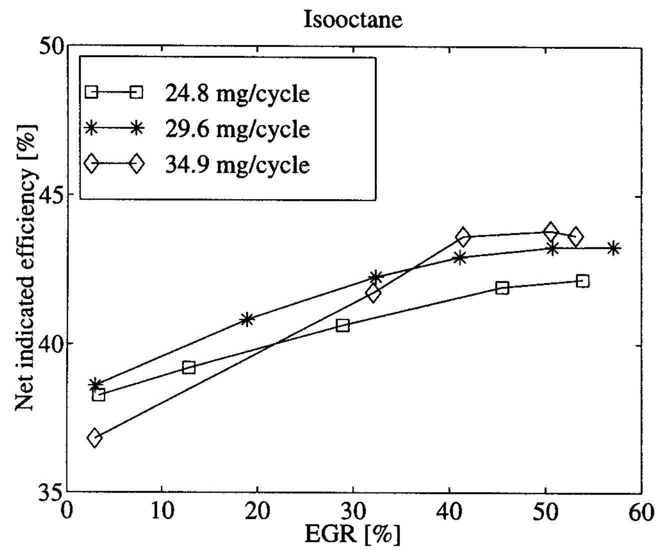

Figure 28. Net indicated efficiency for iso-octane.

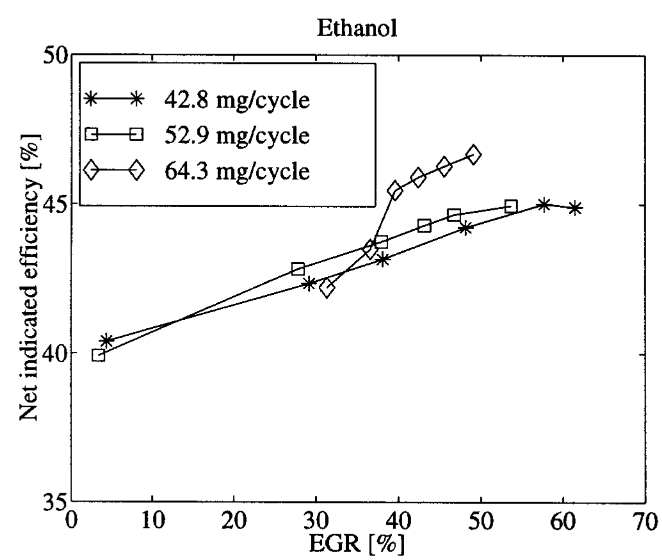

Figure 29. Net indicated efficiency for ethanol.

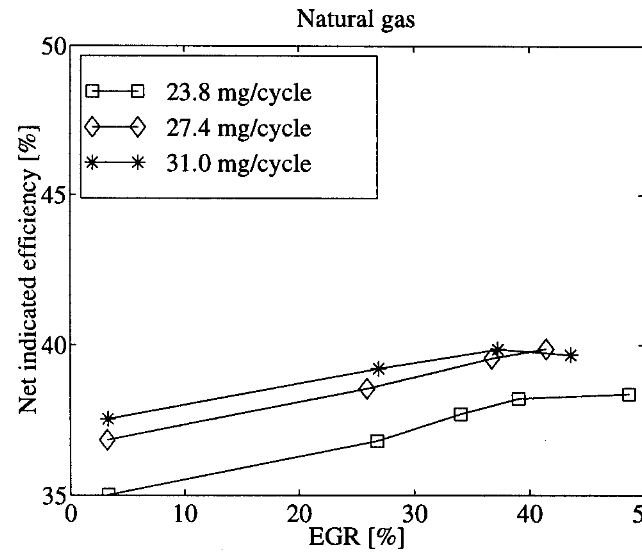

Figure 30. Net indicated efficiency for natural gas.

NOx – One of the most appealing attributes of HCCI is the homogeneous combustion. This means that the temperature is expected to be nearly the same in the entire combustion chamber. This in combination with highly diluted mixtures, required to control combustion rate, gives a low maximum temperature during the cycle. NOx formation is very temperature sensitive at temperatures over 1800 K. If combustion initiation takes place too early before TDC, the maximum temperature will increase and NOx will be generated to some degree. The NOx levels are overall low, and decrease with increasing EGR. This decrease depends on the slower combustion rate and the later combustion phasing, which leads to lower peak pressure, and hence lower maximum temperature. The levels are overall higher with natural gas than with the other two fuels used, at a comparable load. This depends on the higher inlet temperature used, resulting in higher temperature at TDC, necessary to initiate combustion using natural gas. Figures 31-33 show the specific NOx emissions and figures 34-36 show the NOx emissions in ppm.

HC – One drawback with HCCI is the high emissions of unburned hydrocarbons. When the engine is operated ultra lean without any EGR, up to ten per cent of the fuel can escape from the combustion [17, 18]. The volume fraction of unburned hydrocarbons does not show any obvious dependence on the EGR. But the flow of exhaust gases is reduced as the amount of EGR is increased. Therefore the specific emissions of unburned hydrocarbons decrease with increasing EGR, as can be seen in figures 37-39. The low exhaust gas temperature with HCCI can a problem if an oxidizing catalyst should be used. But the use of EGR also increases the exhaust gas temperature, as shown in figure 4-6. This increases the possibility to use an oxidizing catalyst, to oxidize the unburned hydrocarbons.

CO – Emission of carbon monoxide is generally an indication of incomplete oxidation of fuel (low air/fuel ratio in SI engines). With a stoichiometric mixture, CO is generally formed to some degree. With HCCI, CO is very dependent on preheating and engine load. With high load and sufficient hot inlet air, very little CO is generated. Close to the rich limit for HCCI, almost no CO is generated. But on the other hand, close to the lean limit for HCCI very much CO is generated. The specific CO emission is showed in figures 40-42. The amount of EGR does not seem to affect the specific emission of CO.

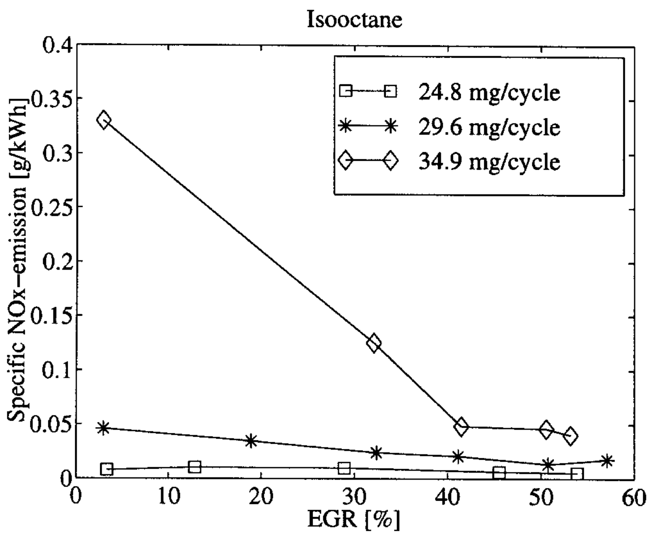

Figure 31. Specific NOx emissions for iso-octane.

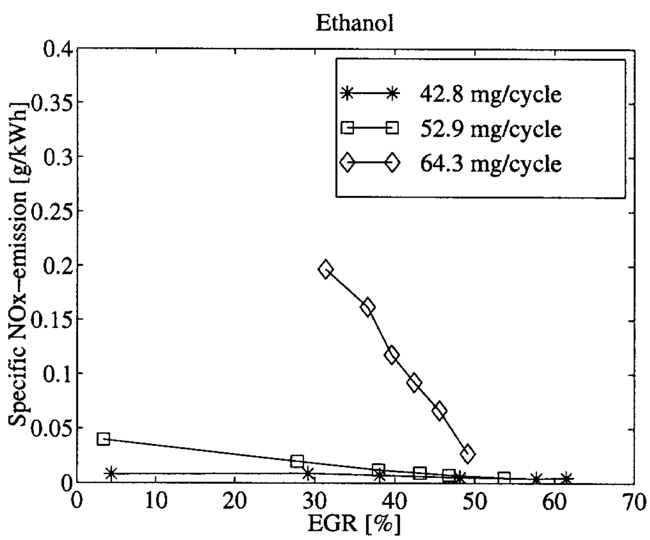

Figure 32. Specific NOx emissions for ethanol.

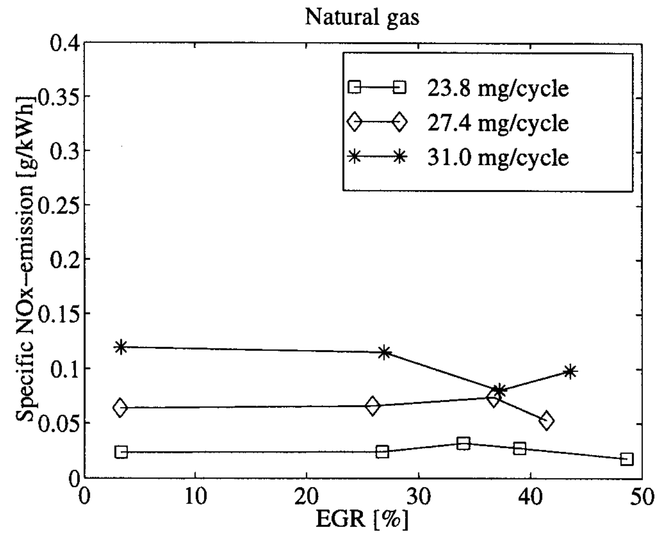

Figure 33. Specific NOx emissions for natural gas.

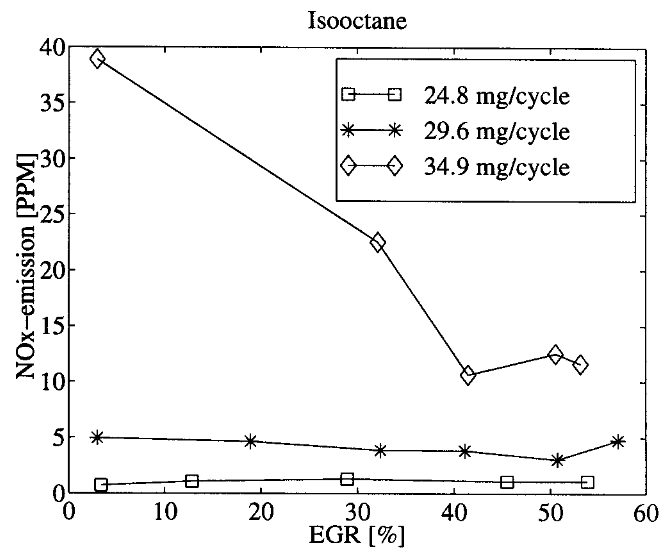

Figure 34. NOx emissions for iso-octane.

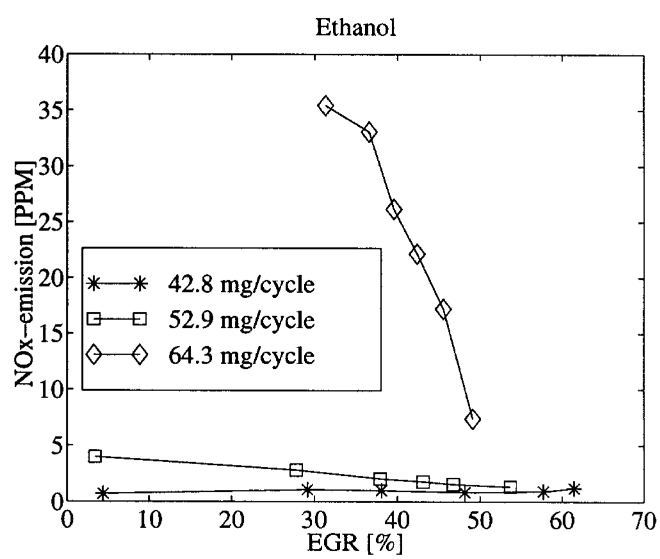

Figure 35. NOx emissions for ethanol.

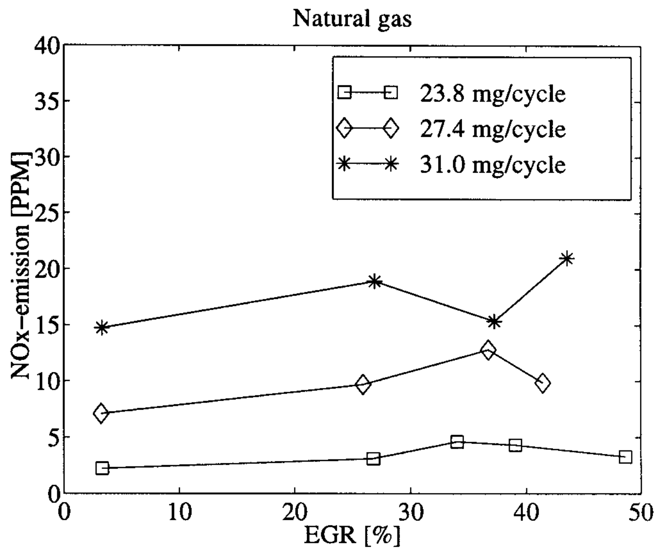

Figure 36. NOx emissions for natural gas.

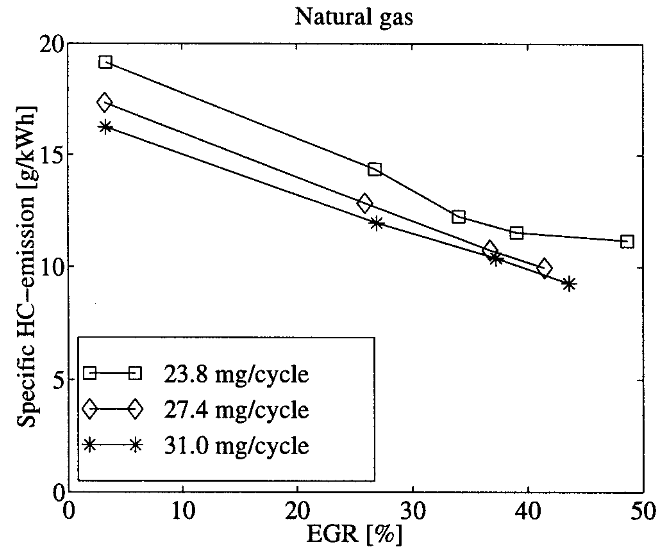

Figure 39. Specific HC emissions for natural gas.

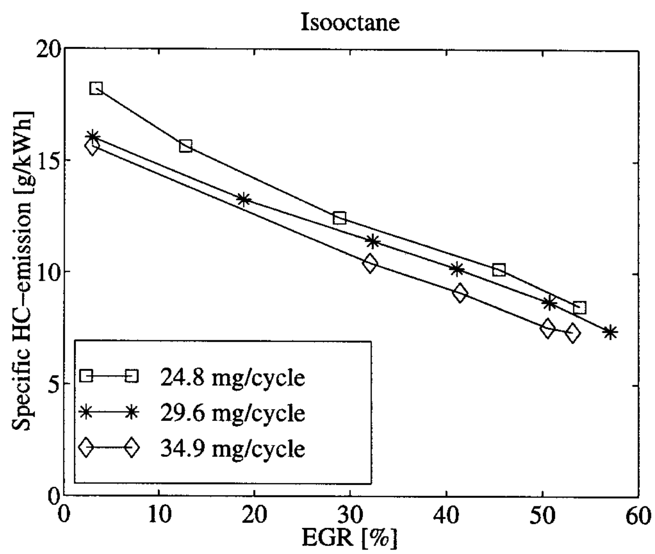

Figure 37. Specific HC emissions for iso-octane.

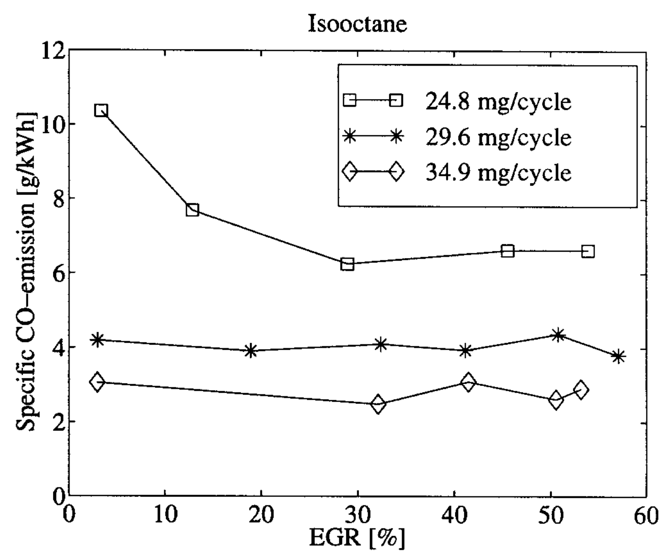

Figure 40. Specific CO emissions for iso-octane.

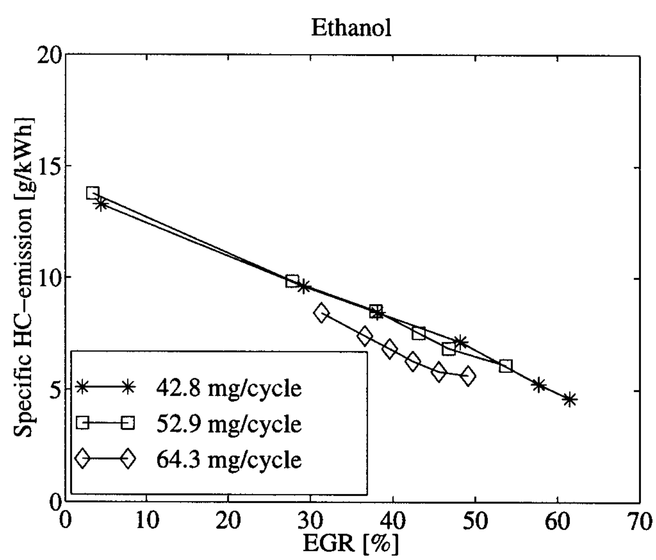

Figure 38. Specific HC emissions for ethanol.

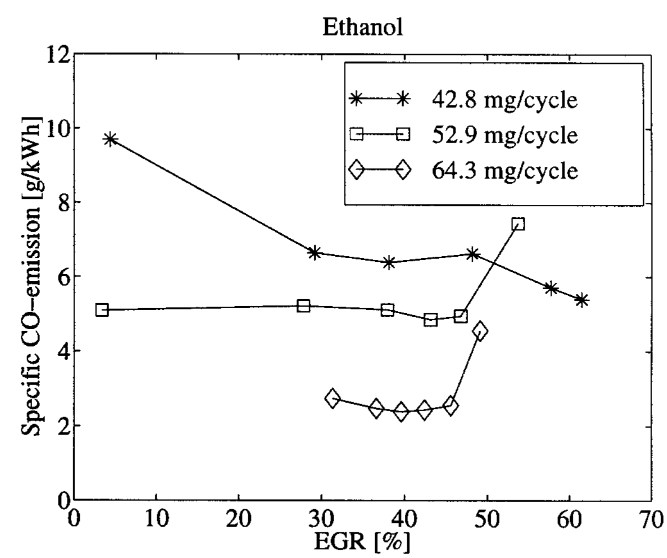

Figure 41. Specific CO emissions for ethanol.

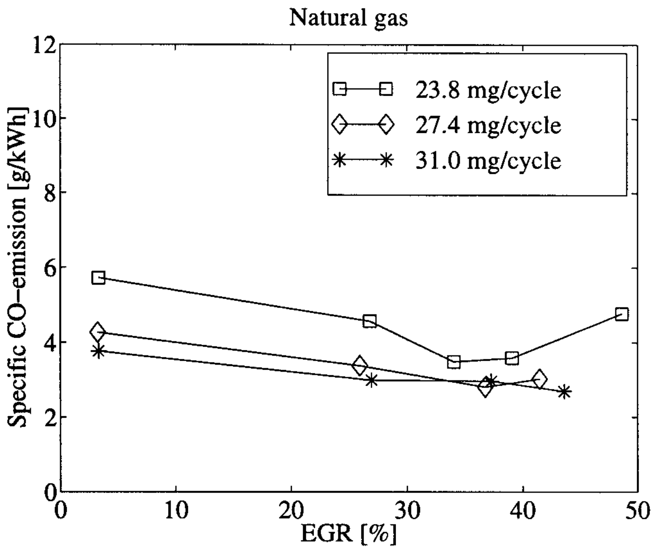

Figure 42. Specific CO emissions for natural gas.

DISCUSSION

The results of the experiments show that the mixture quality has great influence on the HCCI combustion process. The use of EGR slows down the speed of the chemical reactions, which retards the SOC and reduces the rate of combustion. This makes the engine operate with less combustion noise. When very much EGR is used, more preheating is necessary to initiate combustion.

The influence of mixture quality on HCCI combustion was here experimentally investigated for three different fuels. With ethanol it was possible to achieve stable HCCI operation with 62 % of EGR. Iso-octane tolerated 57 % of EGR and natural gas 48 %. With natural gas it was impossible to achieve stable operation, at higher loads, with high amounts of EGR, with proper combustion phasing. At least with the test equipment used in this study.

The best net indicated efficiency obtained here is 47 %. This was achieved with ethanol at about 5 bar of IMEP. This efficiency has to be adjusted for engine friction. If we assume that the engine friction mean effective pressure, FMEP, is 1 bar we obtain

$$\eta_b = \eta_{i,n} \frac{\mathrm{IMEP} - \mathrm{FMEP}}{\mathrm{IMEP}} = 47 \times \frac{4.75 - 1}{4.75} = 37\%$$

This efficiency is comparable to values achieved in diesel engines at part load.

The major advantage with HCCI is not only high efficiency. When combustion initiation is adjusted to occur at a proper crank angle, very little NOx is generated, only a few ppm or less.

The emissions of HC and CO are, however, not low. But the specific HC emissions decreases when EGR is increased (λ is decreased). The use of EGR also increases the exhaust gas temperature. This makes it possible to use an oxidizing catalyst, to oxidize the unburned hydrocarbons.

The major problem with HCCI is not the high emissions of unburned hydrocarbons. The major problem is to control the ignition and combustion process over a wide load and speed range. Futher work is needed on to controling combustion rate, especially for richer mixtures.

CONCLUSIONS

The use of EGR retards the start of combustion and slows down the rate of combustion, which makes the engine operate with less combustion noise. If the combustion takes place mainly after TDC, maximum cylinder pressure and temperature becomes lower, almost no NOx is generated, only about 1 to 5 ppm. The specific HC emissions, which is a drawback with HCCI, decrease when EGR is used. The CO emissions are mainly dependent on preheating and load. High load and sufficient preheating will give low emission of CO. The engine efficiency increases when EGR is used, depending on the increase in combustion efficiency and decrease in pumping work. The highest gross indicated efficiency reported here is close to 50 %. This was achieved with ethanol at 4.75 bar of IMEP.

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