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Demonstrating the Multi Fuel Capability of a Homogeneous Charge Compression Ignition Engine with Variable Compression Ratio

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ABSTRACT

The potential of a Homogeneous Charge Compression Ignition (HCCI) engine with variable compression ratio has been experimentally investigated. The experiments were carried out in a single cylinder engine, equipped with a modified cylinder head. Altering the position of a secondary piston in the cylinder head enabled a change of the compression ratio. The secondary piston was controlled by a hydraulic system, which was operated from the control room. Dual port injection systems were used, which made it possible to change the ratio of two different fuels with the engine running. By mixing iso-octane with octane number 100 and normal heptane with octane number 0, it was possible to obtain any octane rating between 0 and 100. By using an electrical heater for the inlet air, it was possible to adjust the inlet air temperature to a selected value. In this way it was possible to study the relationship between the fuel's octane number, the inlet air temperature and the compression ratio needed to get auto-ignition close to Top Dead Center (TDC). Different fuel mixture ratios of gasoline and diesel fuel have also been tested in the same manner. All tests were carried out with a constant air/fuel equivalence ratio (λ) of 3.0.

The test results show that almost any liquid fuel can be used in an HCCI engine using a variable compression ratio. The indicated efficiency did not increase with increased compression ratio as much as expected. This was mainly due to a decrease in combustion efficiency with increased compression ratio. NOx emissions were overall very low, and did not increase much with increased compression ratio. With diesel fuel, smoke was generated in some cases. Emissions of unburned hydrocarbons were quite high for all cases and they increased with increased compression ratio.

INTRODUCTION

THE PRINCIPLE OF HCCI – The major advantages with Homogeneous Charge Compression Ignition, HCCI, compared to the diesel engine are low NOx emissions and less problem with smoke. Diesel engines are widely used in heavy-duty vehicles and in other commercial applications, due to its high efficiency and durability. In the diesel engine soot is formed in the fuel rich regions and NOx in the hot stoichiometric regions. Due to these mechanisms, it is difficult to reduce both NOx and soot simultaneously through combustion improvement. To eliminate the problems with fuel rich regions and stoichiometric regions, a homogeneous charge can be used instead.

The benefit with HCCI compared to the Spark Ignition (SI) engine is the much higher part load efficiency. The conventional (stoichiometric charge) SI engine fitted with a three-way catalyst can be seen as a very clean engine compared to the diesel engine. But it suffer from poor part load efficiency.

The major drawback with HCCI is the problem of controlling the ignition timing over a wide load and speed range. Another drawback compared to the Spark Ignition (SI) engine and the diesel engine is higher emissions of unburned hydrocarbons.

In an HCCI engine the fuel is injected into the (preheated) air in the intake manifold to create a homogeneous charge. During the compression stroke the charge is further heated to attain auto-ignition close to Top Dead Center (TDC). With HCCI, there is no direct control of the onset of combustion, as the ignition process relies on a spontaneous auto-ignition. The ignition timing can only be controlled indirectly. 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

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since combustion initiation takes place at many points at the same time. HCCI has no flame propagation, instead the whole mixture burns close to homogeneous at the same time [2]. As the whole bulk burns almost simultaneously the combustion rate becomes very high. Therefore highly diluted mixtures have to be used to limit the rate of combustion. This can be achieved by much excess air [30, 31] and/or with Exhaust Gas Recycling (EGR) [32]. Water injection can also be used to slow down the combustion rate [33].

PREVIOUS WORK – Several studies on HCCI have been performed on two stroke engines [2-7] showing that HCCI combustion has great potential to reduce cycle to cycle variations and make engine operation smoother. Regarding the great problem with HC emissions from two stroke engines, it is reported that switching to HCCI operation will drastically reduce HC.

A free piston engine concept running on HCCI has also been presented by researchers at Sandia National Laboratories [8-9].

Most recent studies on HCCI have been focusing on four stroke engines [10-33]. Three different main types of HCCI concepts are being investigated at several laboratories.

In the *first* type the premixed charge is created in the intake system. The homogeneous charge is then compressed to auto-ignition late in the compression stroke. This is the HCCI concept of the present work.

The *second* type also uses a premixed charge, created in the intake system. But the charge is here ignited by a directly injected spray at the end of the compression stroke [16, 27]. This concept is named Homogeneous Charge Diesel Combustion (HCDC). With this concept NOx and soot emissions are relatively proportional to the amount of fuel being injected directly.

In the *third* type the fuel is injected at a very early stage of the compression stroke. A partly homogeneous mixture is then formed during the compression stroke and combustion starts close to TDC. This concept is called PREmixed lean DIesel Combustion (PREDIC) [22, 25, 26] or Premixed Compression - Ignited (PCI) combustion [28].

Regarding NOx emissions, the first type of combustion system is to be prefered, due to the lack of any close to stoichiometric combustion zones. In general for HCCI, high efficiency and low NOx emissions are reported, but emissions of unburned hydrocarbons are high.

PRESENT WORK – In this work it has been studied whether a low octane (high cetane) fuel or a high octane (low cetane) fuel or a medium octane fuel is the most suitable for HCCI operation, regarding fuel efficiency and emissions. The relationship between the fuel's octane number, inlet air temperature and the compression ratio needed to get auto-ignition close to TDC, has also been investigated.

Firstly, different mixes of the reference fuel's iso-octane (octane number 100) and normal heptane (octane number 0) have been used. By mixing these two fuels, any octane number betweeen 0 and 100 can be achieved. For different fuel octane numbers, the inlet air temperature has been varied and the compression ratio was adjusted to get ignition close to TDC.

Secondly, different mixture ratios of gasoline (98 RON) and diesel fuel were tested from pure gasoline to pure diesel fuel in the same procedure as described above.

All tests were carried out with an air/fuel equivalence ratio (λ) of 3.0. The engine speed was set to 1000 rpm, the same as was used in previous papers [30-33]. The engine was operated unthrottled, giving an inlet pressure of about 1 atm. No external Exhaust Gas Recycling (EGR) was used in these tests.

Table 1 summerizes the test conditions. This work can be divided into two different parts, regarding the fuels studied. The change in IMEP depends on the change in inlet air temperature. With increased inlet temperature, the mass flow through the engine decreases and so does the IMEP.

Table 1. Test conditions.

	Part I (reference fuels)	Part II (commercial fuels)
Fuels	iso-octane & n-heptane	gasoline & diesel fuel
λ	3.0	3.0
CR	9.6:1 - 21.5:1	9.6:1 - 22.5:1
T(in)	30 - 130°C	30 -130°C
Speed	1000 rpm	1000 rpm
IMEP	3.0 - 4.0 bar	2.5 - 3.9 bar
p(in)	1 atm	1 atm

EXPERIMENTAL APPARATUS

The engine used for the experiments originated from a Volvo TD100 series diesel. This engine has been a reliable work horse for many years in our lab [30-34]. The engine is an in-line six cylinder engine, modified to operate on one cylinder only. The other five cylinders were motored. This arrangement gives less reliable brake specific values, as the total engine friction is high compared to the output torque from the running cylinder. Instead, only indicated results have been used. The running cylinder was equipped with a cylinder head, modified to achieve Variable Compression Ratio (VCR). The compression ratio could be changed by changing the position of a secondary piston in the head, see Figure 1. The secondary piston was controlled by a hydraulic system, which could be operated during engine operation. The compression ratio could be varied in a wide span, from about 10:1 to 28:1. To be able to change the fuel mixture ratio of two different fuels with the engine running, dual port injection systems were used. The fuel was injected at about 300 millimeters upstream the inlet valve.

Acquired by experience from earlier experiments, this setup gives an almost fully homogeneous mixture [34]. The inlet air was preheated with an electrical heater. The electrical power needed for the heater has not been taken into consideration in the efficiency calculations. The cylinder pressure was recorded with a pressure transducer, mounted in the secondary piston unit. 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
Inlet Valve Diameter	50 mm
Exhaust Valve Diameter	46 mm
Swirl number	2.8
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
Secondary Piston Diameter	40 mm
Secondary Piston Stroke	100 mm

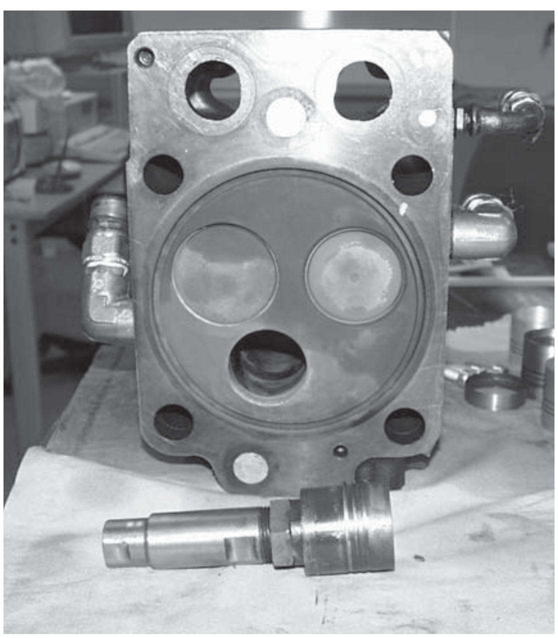

Figure 1. The modified cylinder head and the secondary piston unit. Note the lack of spark plug and fuel injector. The pressure transducer was placed in the secondary piston.

RESULTS

COMPRESSION RATIO AND INLET TEMPERATURE – For constant fuel octane numbers the inlet temperature was varied and the compression ratio was adjusted to get auto-ignition close to TDC. Figure 2 shows the compression ratio versus the inlet temperature. The legend in the figure represents the octane number. The different octane numbers have been obtained by mixing iso-octane and n-heptane. The octane number (ON) of a fuel mixture of iso-octane and n-heptane can be calculated as follows:

$$ON = \frac{\dot{m}_{iso}}{\dot{m}_{iso} + \dot{m}_{hep}} \cdot 100$$

where $\dot{m}_{iso}$ ıass flow iso-octane and is m $\dot{m}_{hep}$ flo n-heptane. Table 3 shows a brief specification of the fuels used. For each constant ON, the inlet air temperature was varied from 30°C to 130°C in steps of 20°C. For each test point the compression ratio was adjusted to get auto-ignition at TDC. Figure 3 shows the same results as Figure 2 though in the shape of a three dimensional map between the compression ratio, the inlet air temperature and the octane number. For the two cases with lower ON it was not possible to run the engine at the higher inlet temperatures, limited by the demand of lower compression ratios than achievable with the test engine.

In the same manner, different fuel ratios of gasoline and diesel fuel have also been tested. Figure 4 shows the compression ratio used versus the inlet air temperature for different fuel mixture ratios. Figure 5 shows the corresponding three dimensional map. The labels in the legend state the fraction of gasoline. 100 % means pure gasoline and 0 % means pure diesel fuel. With diesel fuel only and a low inlet temperature (below 90°C), the combustion quality became very low, due to the poor vaporization of the diesel fuel. Diesel fuel has a much higher boiling-point than the other fuels used. It also contains some very heavy fractions with boiling points above 300 °C. Therefore, it was of no interest to run the engine at the lower inlet temperatures.

Table 3. Fuel specifications [36].

	Iso-octane	Normal heptane	Gasoline unleaded	Diesel fuel
Octane no.	100	0	98 (RON)	-
Cetane no.	-	-	-	54
T_{Boil} [°C]	118	98.4	195	280 (95%)
ρ [kg/m^3]	700	686	750	814
$(A/F)_{Stoich}$	15.13	15.18	14.6	14.5

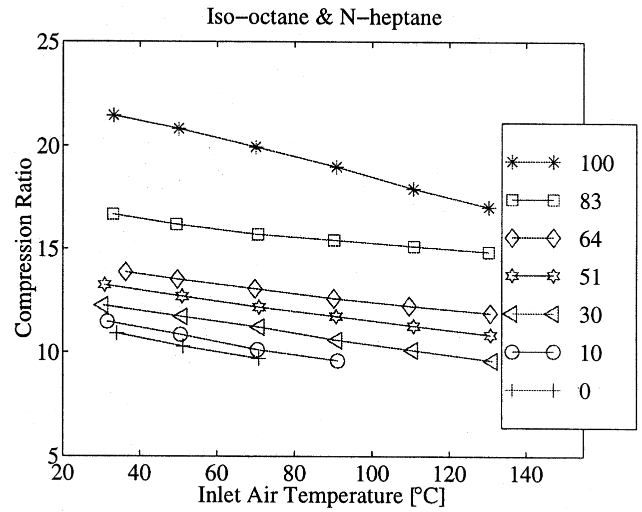

Figure 2. Compression ratio as function of inlet air temperature for various fuel octane numbers. The legend states the octane number.

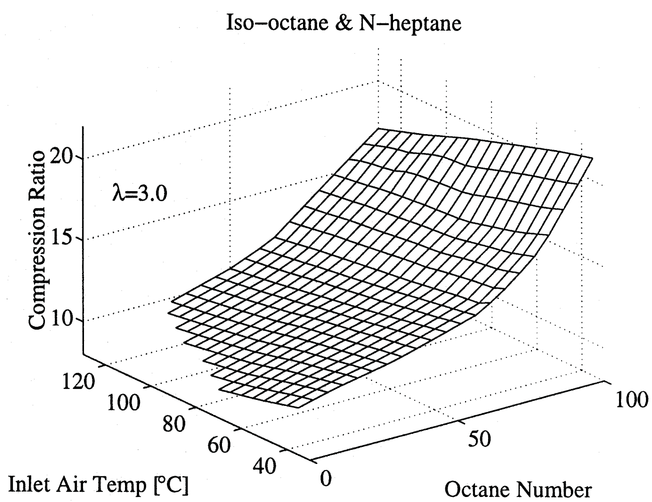

Figure 3. The three dimensional map between the compression ratio, the inlet air temperature and the octane number.

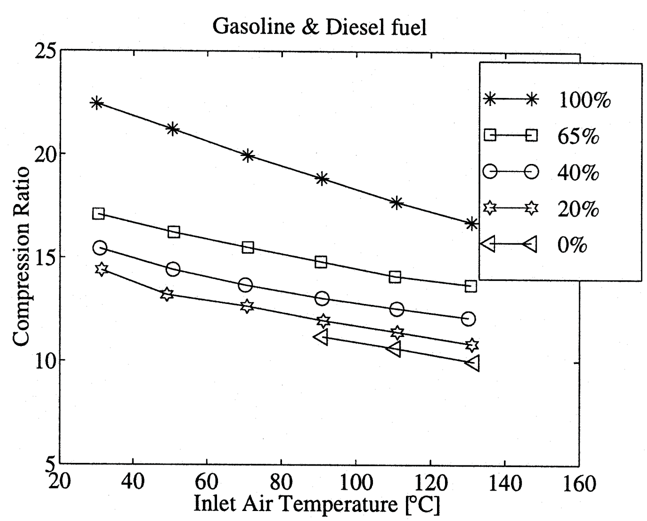

Figure 4. Compression ratio as function of inlet air temperature for different fuel mixture ratios of gasoline and diesel fuel. 100 % being gasoline only.

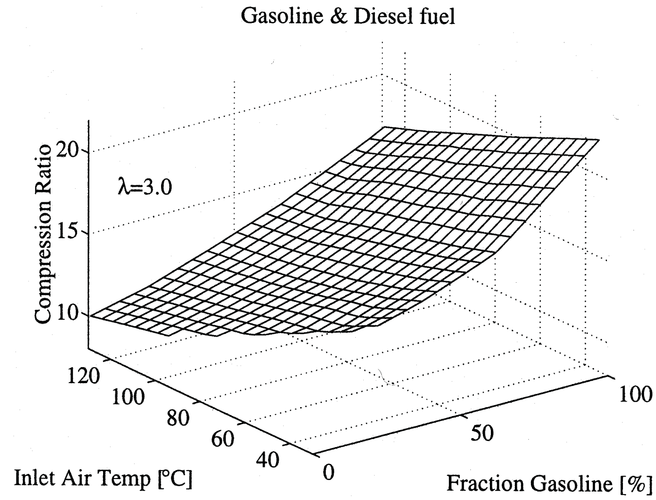

Figure 5. The three dimensional map between the compression ratio, the inlet air temperature and the fuel mixture ratio of gasoline and diesel fuel.

CYLINDER PRESSURE MEASUREMENTS – The cylinder pressure was measured with a pressure transducer for all operating conditions. For each test point, the cylinder pressure was recorded for 100 cycles, every 0.2 degrees crank angle. Each pressure trace in the figures is the mean pressure trace for 100 cycles. For each test point the compression ratio has been adjusted to get auto-ignition at around TDC. As a result of this ignition timing, the maximum cylinder pressure will occur at approximately 6 to 8 crank angles ATDC. Figures 6-11 show the test cases with iso-octane and n-heptane, and Figures 12-16 show the pressure traces with gasoline and diesel fuel mixtures. In each plot the trace with the highest maximum pressure corresponds to the highest compression ratio and lowest inlet air temperature used. When the inlet temperature is increased the compression ratio has to be lowered to get auto-ignition at the same crank angle. Each figure shows the pressure traces for different inlet temperatures and compression ratios at a constant fuel mixture ratio of iso-octane and n-heptane.

With the pressure transducer mounted in the secondary piston, strong pressure oscilliations on the pressure trace occured in some cases. The amplitude and frequency varied with the position of the secondary piston. The pressure oscilliations were probably a result of pressure waves in the secondary cylinder unit, generated by the rapid pressure rise during the combustion. The frequencies of the pressure oscilliations were related with the length of the secondary cylinder.

RATE OF COMBUSTION – The cylinder pressure data was analyzed using a single zone heat release model. With a single zone heat release model, it is assumed that the temperature and the gas composition are the same in the whole bulk. With HCCI, this is a quite good assumption as combustion is expected to occur simultaneously in the whole combustion chamber, with a homogeneous temperature, except for near the walls where the temperature is lower due to heat losses. Details concerning the model can be found in [34]. With HCCI, the combustion rate is very high as combustion occur simultaneously in the whole combustion chamber. The overall combustion rate is very dependent on λ (species concentrations), combustion temperature and combustion timing. In this study λ was kept constant. The combustion temperature and the combustion timing are of course related to each other. Advanced combustion timing gives higher combustion temperature.

Figures 6-11 show the combustion rate for the different fuel mixtures. In general, the combustion rate becomes higher with increased compression ratio (increased ON). With iso-octane (ON=100) no detectable cool flames (pre-reactions) appear before the main combustion. When n-heptane is used, cool flames appear at some 10 to 20 CAD before the main combustion starts. The amount of heat released in the cool flames increases with increased amount of n-heptane (lower ON). If we study a particular case with a constant fuel octane rating, the cool flames appear earlier in the compression stroke with higher inlet temperature and lower compression ratio. For all test points, the compression ratio was adjusted to initiate the main combustion at TDC.

Regarding the cool flames a similar trend was observed with diesel fuel and gasoline. With an increased amount of diesel fuel the amount of heat released in the cool flames increases. When running on pure diesel fuel, the ignition timing has to be advanced to about 5 CAD BTDC to obtain improved combustion quality and stable operation, see Figure 17.

As the rate of heat release is evaluated from the cylinder pressure data, the oscilliations on the pressure traces will generate oscilliations on the rate of heat release curves as well. The rate of heat release curves plotted in the figures have been filtered from the oscilliations. In this way it is easier to read the plots. It is the mean values of the rate of heat release curves that have been filtered. The filters were designed to influence the overall shape as little as possible. This means that the accumulated heat release is unchanged.

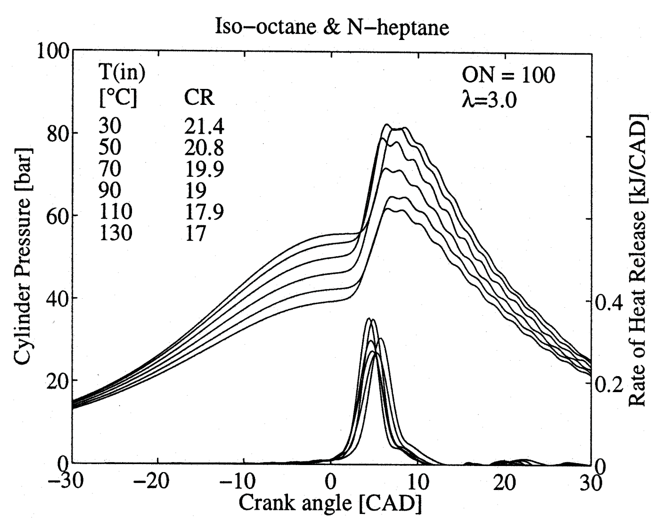

Figure 6. Cylinder pressure and rate of heat release for the case with a fuel octane number of 100, pure iso-octane.

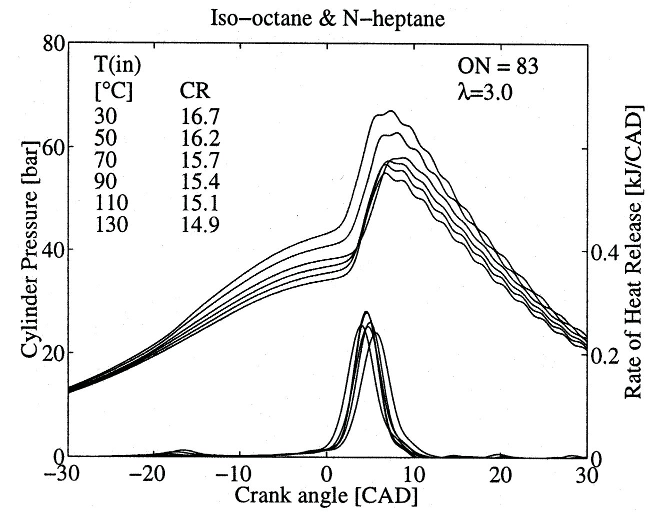

Figure 7. Cylinder pressure and rate of heat release for the case with a fuel octane number of 83.

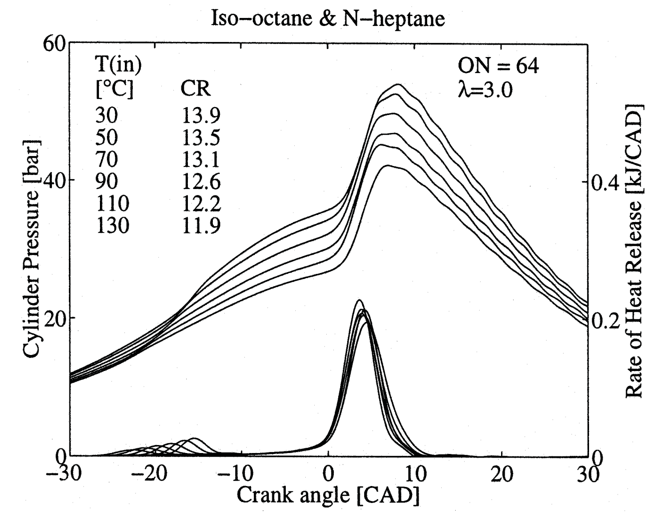

Figure 8. Cylinder pressure and rate of heat release for the case with a fuel octane number of 64.

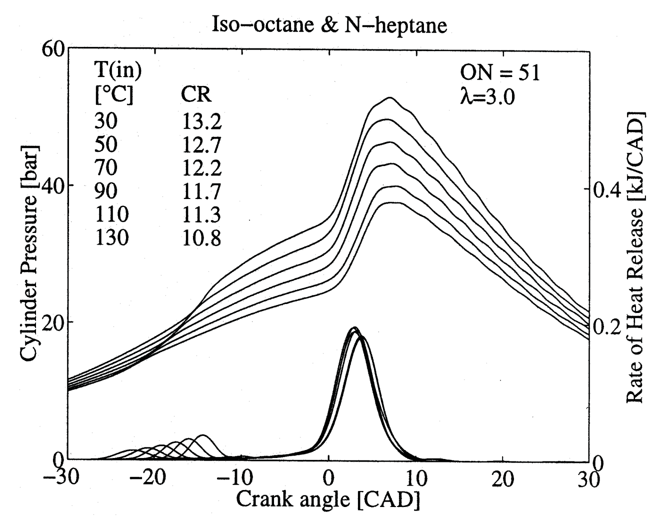

Figure 9. Cylinder pressure and rate of heat release for the case with a fuel octane number of 51.

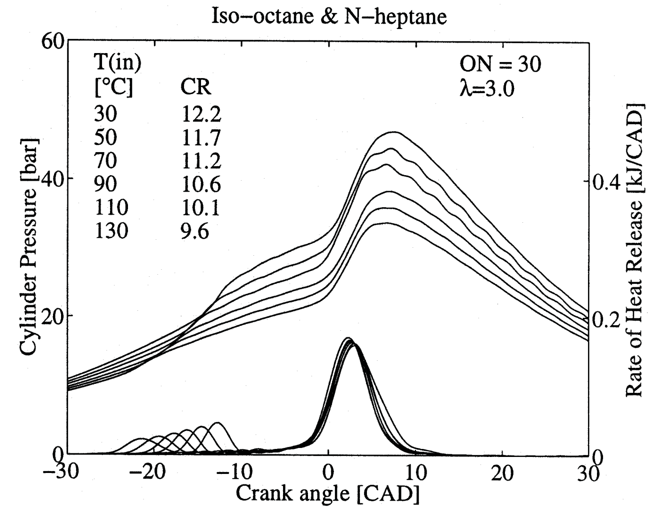

Figure 10. Cylinder pressure and rate of heat release for the case with a fuel octane number of 30.

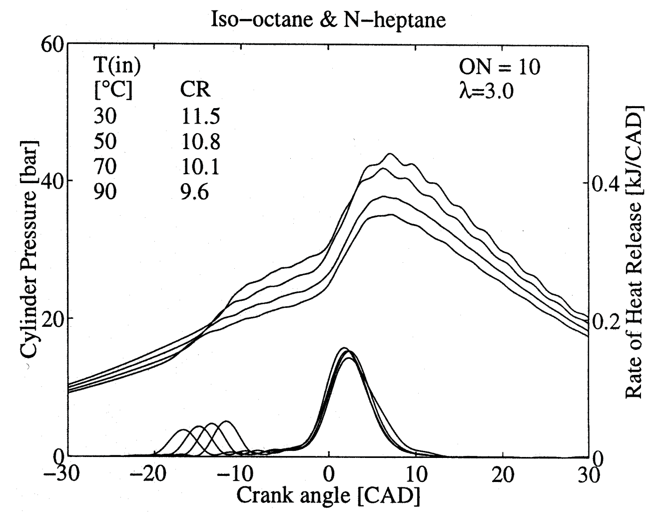

Figure 11. Cylinder pressure and rate of heat release for the case with a fuel octane number of 10.

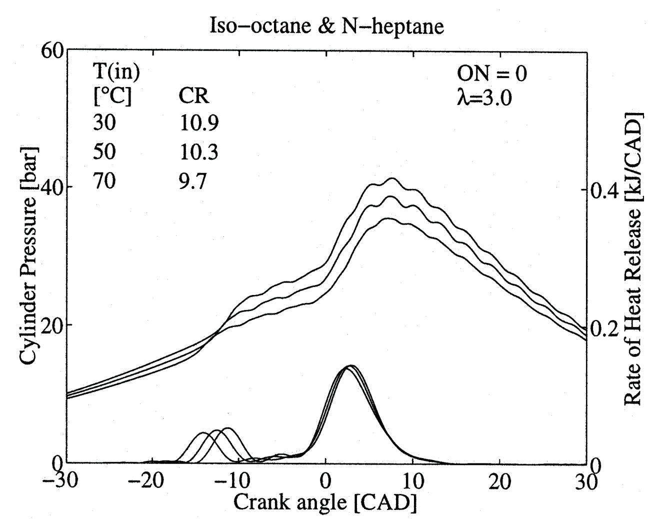

Figure 12. Cylinder pressure and rate of heat release for the case with a fuel octane number of 0, pure n-heptane.

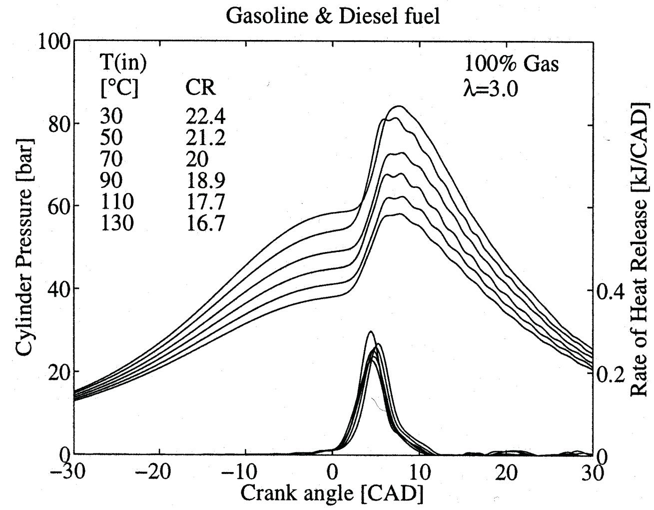

Figure 13. Cylinder pressure and rate of heat release for the case with pure gasoline.

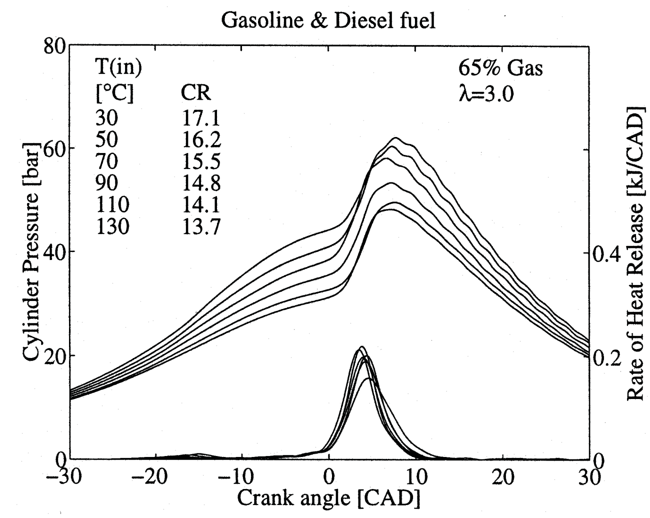

Figure 14. Cylinder pressure and rate of heat release for the case with 65 % gasoline and 35 % diesel fuel.

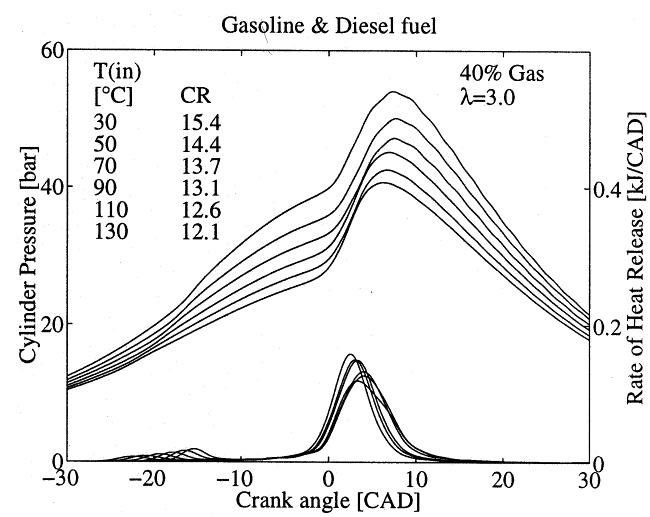

Figure 15. Cylinder pressure and rate of heat release for the case with 40 % gasoline and 60 % diesel fuel.

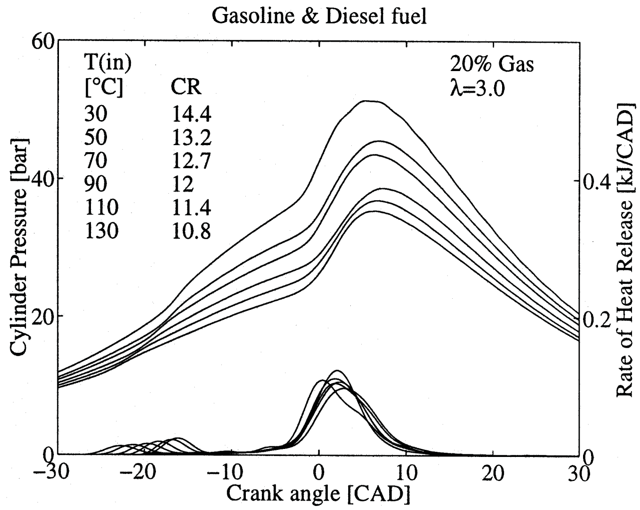

Figure 16. Cylinder pressure and rate of heat release for the case with 20 % gasoline and 80 % diesel fuel.

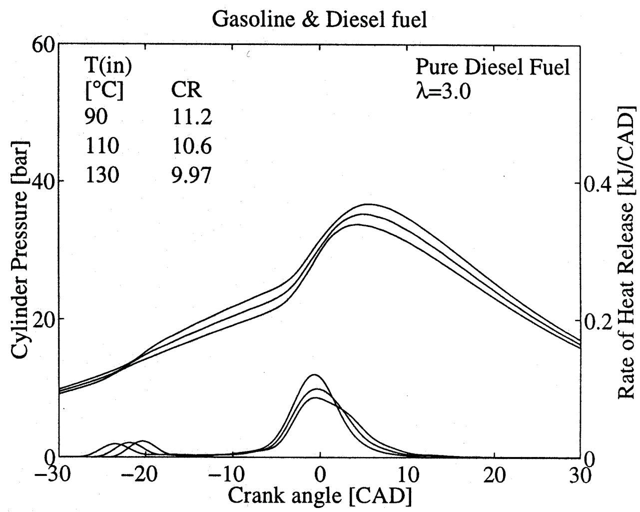

Figure 17. Cylinder pressure and rate of heat release for the case with diesel fuel only.

COMBUSTION DURATION – The duration of the main combustion, 10-90 % heat released, is a good measure of how fast the combustion is. Turbulence has a very modest influence or none at all on the combustion rate for HCCI. Instead λ and temperature strongly affect the combustion rate, as homogeneous combustion is controlled mainly by chemical kinetics. Turbulence is of course important for the mixing process to create a homogeneous charge.

The combustion duration decreases with increased fuel ON as the overall combustion temperature becomes higher, see Figure 18. Higher ON means higher auto-ignition temperature, which gives a higher overall combustion temperature. Figures 21-23 shows the temperature traces for cases with ON of 100, 51 and 0, respectively. The increased duration with lower ON also depends on the cool flames. Increasing the amount of n-heptane increases the amount of heat released in the cool flames. Therefore the combustion duration presented here will include the delay time between the cool flames and the main combustion.

With constant ON the trend is different. Here the main trend is that the combustion duration increases with increased compression ratio and reduced inlet temperature. This depends on that the combustion temperature becomes higher with increased inlet air temperature. The change in volume around TDC also is more rapid with a higher compression ratio. As a result the temperature becomes lower, see Figure 20.

With gasoline and diesel fuel the trend is similar. The combustion duration decreases with increased amount of gasoline, see Figure 19. With a constant mixture ratio the combustion duration becomes longer with increased compression ratio and reduced inlet temperature, due to the lower overall combustion temperature. Compare with the reference fuel case

CYLINDER BULK TEMPERATURE – The cylinder bulk temperature was calculated from the cylinder pressure trace. It was assumed that the temperature and the gas composition is the same in the whole bulk. As mentioned in the paragraph on rate of combustion, this is a quite good assumption with HCCI. The initial temperature for the compression stroke was estimated by using simple heat transfer calculations.

The bulk temperatures for three different ON:s, 100, 51 and 0, are plotted in Figures 21-23. With constant ON and combustion phasing, the overall cycle temperature becomes higher when a higher inlet temperaure and a lower compression ratio is used.

Increasing the ON means higher auto-ignition temperature, and thereby an increased compression ratio is demanded. As a result, the overall cycle temperature increases with increased fuel ON.

The cylinder pressure has a quite strong influence on the auto-ignition temperature, due to the change in species concentrations. If we study the case with pure iso-octane (Figure 21) the difference in auto-ignition temperature is about 100 K between the two extreme cases.

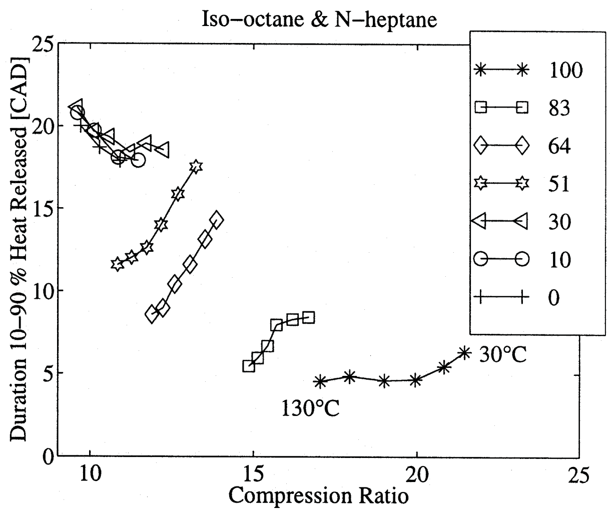

Figure 18. Duration of the main combustion (10-90 % heat released) for the test cases with iso-octane and n-heptane. For each of the seven test cases with constant ON, the inlet temperature has been varied and the compression ratio has been adjusted to get auto-igntion at TDC.

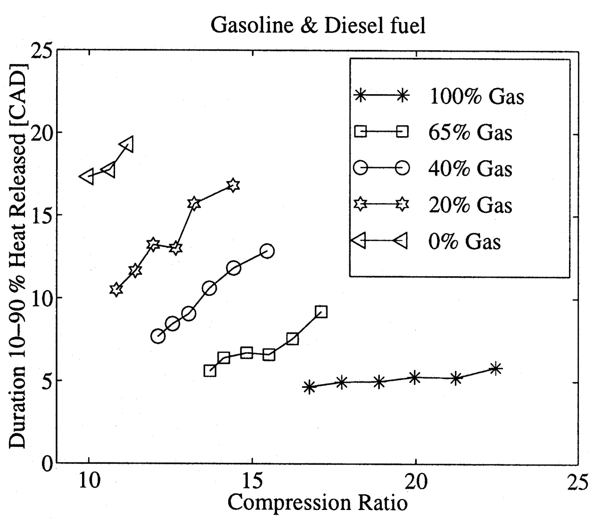

Figure 19. Duration of the main combustion (10-90 % heat released) for the test cases with gasoline and diesel fuel.

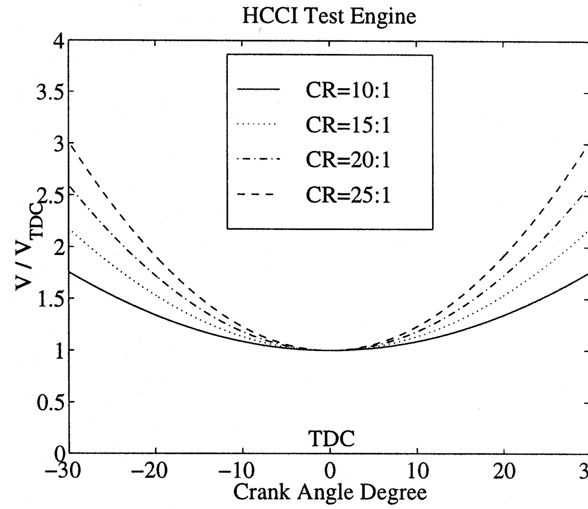

Figure 20. Instantaneous volume in relation to volume at TDC.

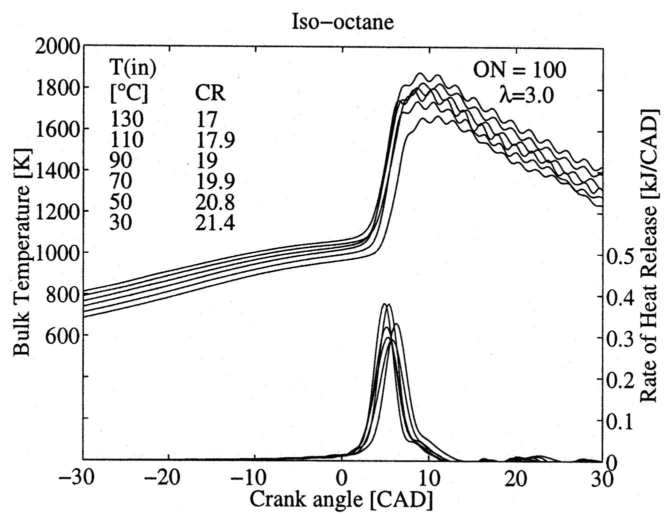

Figure 21. Calculated bulk temperature for the case with pure iso-octane.

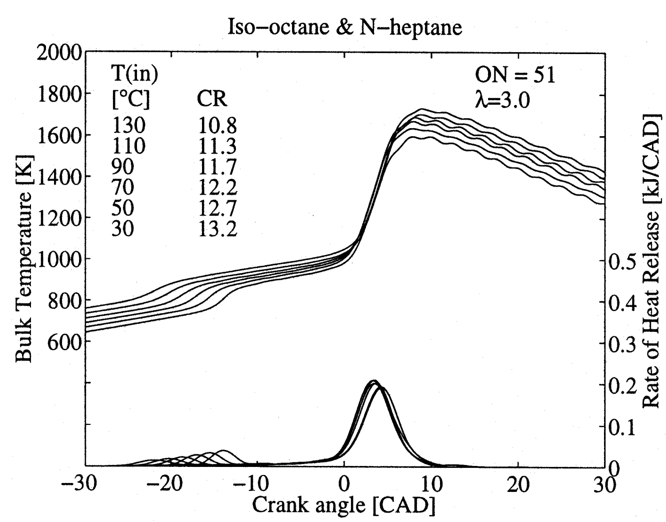

Figure 22. Bulk temperature for the case with a fuel octane number of 51.

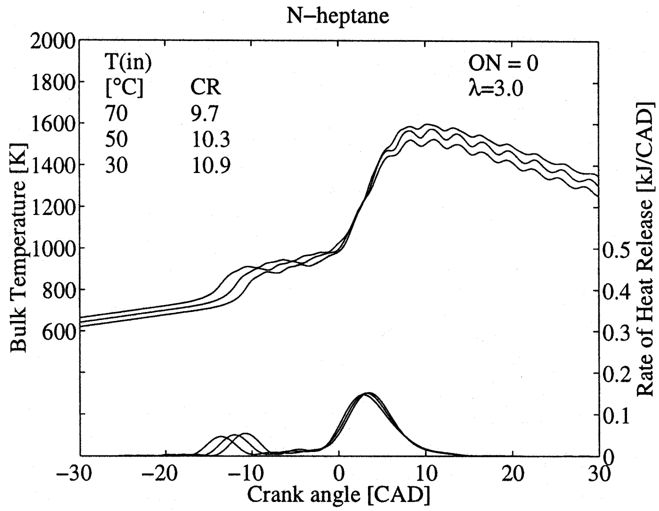

Figure 23. Bulk temperature for the case with pure n-heptane.

GROSS INDICATED EFFICIENCY – The gross indicated efficiency was evaluated from the fuel flow and the indicated mean effective pressure during the compression and expansion stroke only. Pumping work and engine friction is not included here. Defined on one cycle only, this efficiency can be expressed as the ratio between the work on the piston during the compression and expansion stroke only, $W_{i,g}$, and the input fuel energy:

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

where m_f is fuel mass per cycle and q_{lhv} is the lower heating value per mass unit fuel.

Figure 24 shows the results from the tests with iso-octane and n-heptane. The highest efficiency obtained here is close to 43 %. It was with iso-octane only, a compression ratio of 21.5:1 and an inlet air temperature of 30 °C. With a compression ratio of around 10:1, the efficiencies vary between 36 % to 37 %. The efficiency also decreases with increased inlet temperature, due to increased heat transfer.

Figure 27 shows the efficiency for the cases with gasoline and diesel fuel. The highest efficiency obtained here is about 41 %. This is lower than with iso-octane at a lower compression ratio. The compression ratio used was 22.5:1. There is no simple explanation for this lower efficiency. If one compare the combustion phasing and the pressure traces for the two different test points, the phasing is almost equal, see Figures 6 and 13. With diesel fuel only, the efficiency becomes very low due to the poor combustion quality.

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, which means that the net indicated efficiency is lower than the gross indicated efficiency. It can be witten as the ratio between the work on the piston for all the four strokes, $W_{i,n}$, and input fuel energy:

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

Figure 25 and 28 show the measured results. The pumping work is quite high with this old style two valve cylinder head. Figure 30 shows pressure versus volume for the gas exchange process. The mean pressure difference between the exhaust stroke and the intake stroke is about 0.3 atm. Therefore the net indicated efficiency becomes much lower than the gross indicated efficiency. Using a four valve cylinder head instead, the pumping losses would be lower and the differences between gross and net efficiency would be lower. The low exhaust gas temperature related with HCCI givs less blowdown, and more charge is thereby left within the cylinder for the exhaust stroke.

GROSS INDICATED THERMAL EFFICIENCY – This efficiency is defined as the ratio between the work on the piston during the compression and expansion stroke only, $W_{i,g}$, and the heat released, Q:

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

This efficiency is also equal to the gross indicated efficiency adjusted with the combustion efficiency, h_c. This efficiency is higher than the gross indicated efficiency, see Figure 26 and 29. This is the indicated efficiency, which could be obtained if the combustion efficiency had been 100 %. One way to increase the combustion efficiency for HCCI is to use EGR. In a previous study on HCCI, combustion quality was strongly improved when much EGR was used [32]. When studying the thermodynamical benefits with increased compression ratio, this is the efficiency to look at.

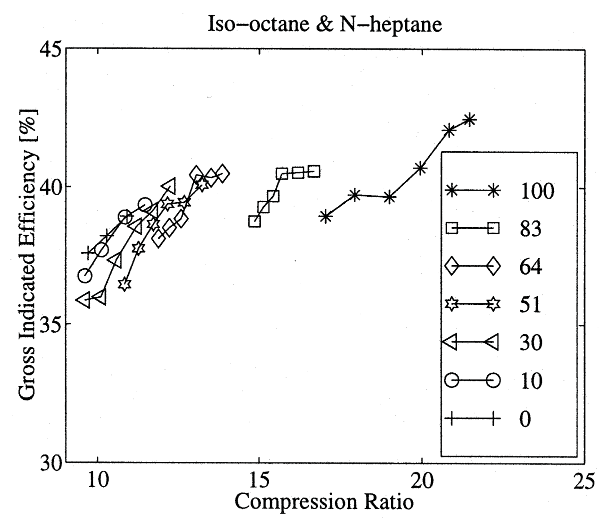

Figure 24. Gross indicated efficiency with iso-octane and n-heptane.

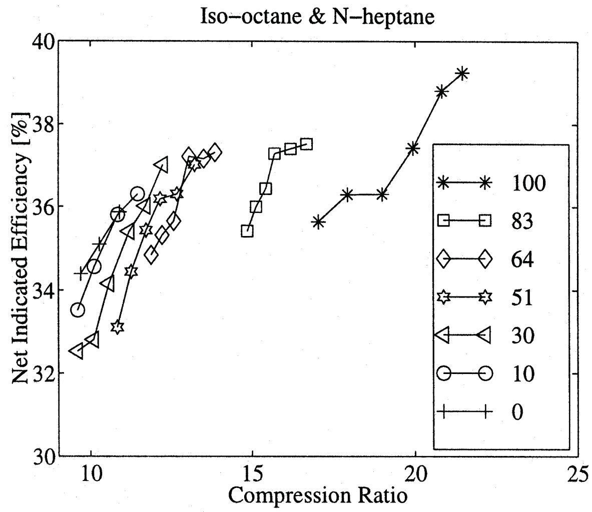

Figure 25. Net indicated efficiency with iso-octane and n-heptane.

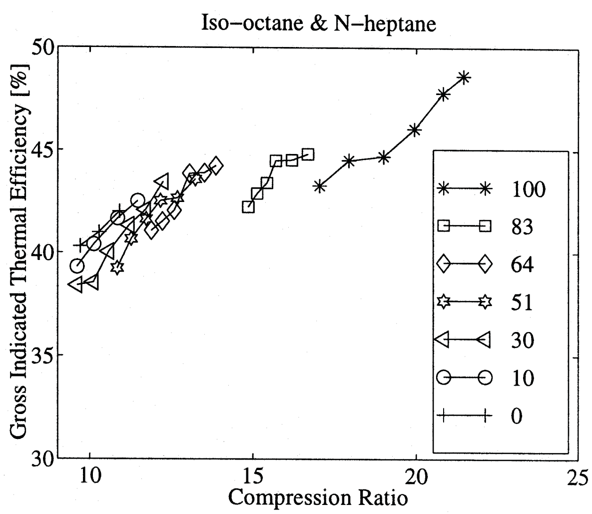

Figure 26. Gross indicated thermal efficiency with iso-octane and n-heptane.

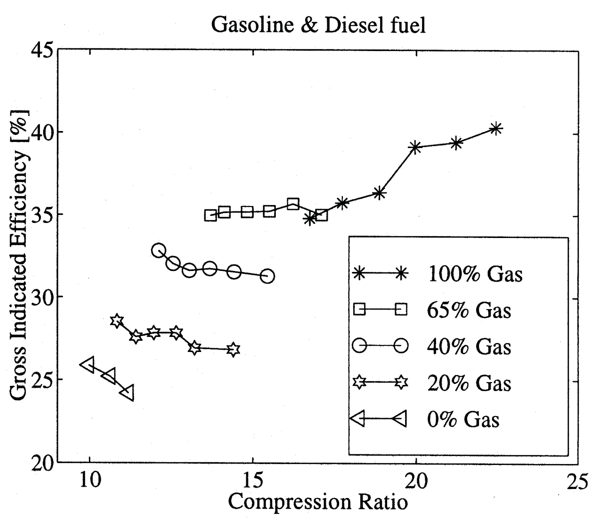

Figure 27. Gross indicated efficiency with gasoline and diesel fuel.

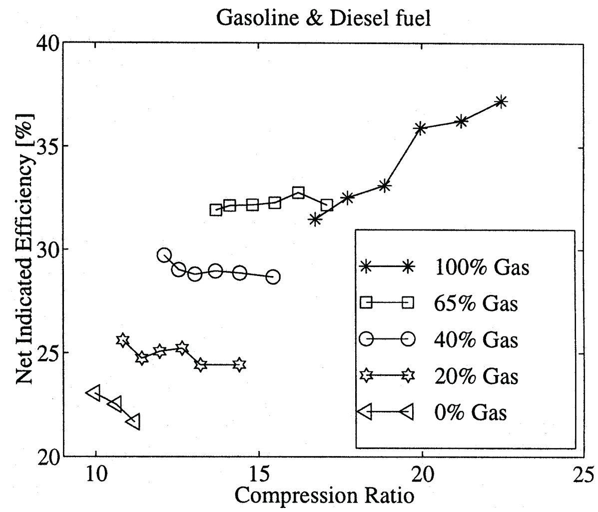

Figure 28. Net indicated efficiency with gasoline and diesel fuel.

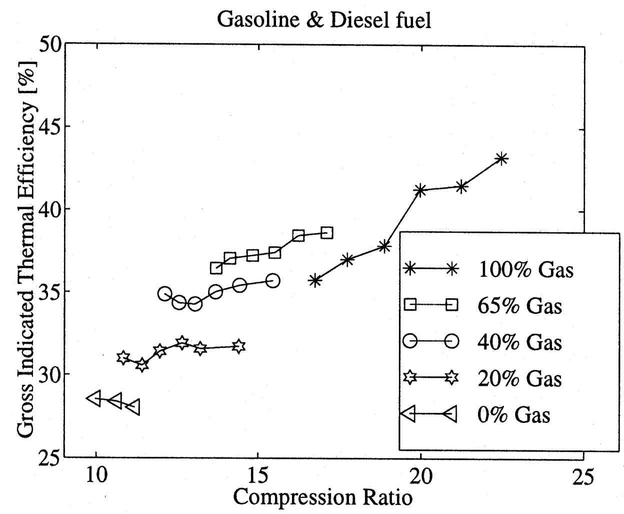

Figure 29. Gross indicated thermal efficiency with gasoline and diesel fuel.

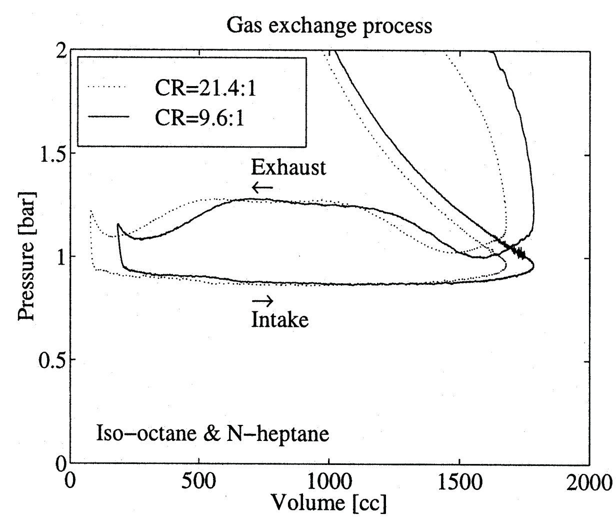

Figure 30. Pressure versus volume for the gas exchange process. Note the relatively high pressure difference between the exhaust stroke and the intake stroke.

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. Figure 31 shows the specific NOx emissions for the test cases with iso-octane and n-heptane. The specific emissions are calculated by using the gross indicated power. NOx emissions are increased with increased fuel octane number since a higher temperature is required to get auto-ignition. But for each test case with a constant octane number, NOx is reduced with increased compression ratio and lowered inlet temperature. This depends on that the combustion temperature becomes higher with increased inlet temperarure. Figures 21-23 shows the temperature traces for the ON of 100, 51 and 0. The total amount of NOx generated is also dependent on how long the gases are exposed to high temperature. With higher inlet temperature and lower compression ratio the temperature is also higher for a longer period of time. This results in increased NOx with increased inlet temperature.

With gasoline and diesel fuel the trend is less obvious, see Figure 32. More NOx is generated with pure diesel than with pure gasoline. This may depend on the poor vaporization of diesel fuel and thus a more inhomogeneous fuel distribution. In the rich zones, combustion temperature will locally be higher and more NOx is thereby generated.

HC EMISSIONS – The low homogeneous combustion temperature related with HCCI prevents NOx formation, but the combustion temperature becomes too low to oxidize the fuel completely. This low combustion temperature results in high emissions of unburned hydrocarbons. The combustion temperature near the walls will be even lower, due to heat losses. Combustion may be quenched or not occur at all close to the walls. With very much excess air (λ=3) which have been used in this study the quenching distance is expected to be quite long.

When the onset of combustion occurs at TDC, the main combustion takes place during the early 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, the temperature becomes too low for complete oxidation of the fuel and much HC 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. Figure 20 shows how fast the volume changes around TDC for different compression ratios.

Another source for HC emissions is the crevice volumes. The mixture trapped in crevices will probably be too cold to ignite at all. HC emissions descended from crevice volumes also increase with increased compression ratio. When the compression ratio is increased, the effect of crevice volumes becomes greater as the clearence volume is decreased. The size of the crevice volumes does not change with changed compression ratio. Therefore more of the charge will be trapped in crevice volumes around TDC. As the cylinder pressure also increases with increased compression ratio, more unburned mixture will be compressed into crevices.

With gasoline and diesel fuel the trend is reversed. Here HC increases with decreased compression ratio. The reason for this is not dependent on the change in compression ratio at all, instead it depends on the poor vaporization, see Figure 34. In these experiments we used conventional low pressure fuel injectors. With diesel fuel, one cannot expect sufficient atomization of the fuel to create a homogeneous mixture, especially with a low inlet temperature.

CO EMISSIONS – With HCCI, CO is very dependent on the combustion temperature. Close to the rich limit for HCCI and/or with early combustion phasing, very little CO is generated. But close to the lean limit and/or with late combustion phasing very much CO can be a result.

In Figure 35 the specific CO emission is plotted against the compression ratio used. If we look at CO as function of the fuel's octane number and compare the levels of the different test cases, it is obvious that CO decreases with increased fuel octane rating. This depends on the higher overall combustion temperature related to a high octane fuel. As the auto-ignition temperature is higher for a high octane fuel, the combustion temperature also becomes higher for a given λ (and equal fuel heating value).

If instead we study a particular case with constant fuel octane rating, CO increases with increased compression ratio and lowered inlet temperature. With a lower compression ratio there is more time for constant volume combustion than with a high compression ratio. This gives a flatter combustion temperature profile versus time (CAD). With a higher compression ratio there is less time available for complete oxidation of CO.

In the test series with gasoline and diesel fuel, CO increases along with an increased amount of diesel. With a more inhomogeneous mixture related to diesel fuel, combustion will partially occur in richer zones. In these rich zones much CO is generated. Figure 36 shows the specific CO emission.

SMOKE EMISSION – With iso-octane and n-heptane almost no smoke was generated at all. The level was below 0.05 Bosch Smoke Number for all test points.

When diesel fuel was used much smoke was generated in some cases. No obvious trend regarding the amount of smoke generated could be drawn from the test results with diesel fuel. The overall trend though is that the smoke emission decreases when the inlet temperature is increased. Figure 37 shows the Bosch Smoke Number.

Running on pure gasoline almost no smoke was generated. This indicates that the diesel fuel does not vaporize completely and combustion partly takes place under rich conditions. Combustion may partly occur around fuel droplets.

During the tests with diesel fuel an interesting behaviour, regarding the soot formation was found. When the engine was operated with advanced combustion timing (increased compression ratio) the soot emission increased strongly. Figure 38 shows this behaviour. The compression ratio has here been changed from 12.2:1 down to 10.6:1.

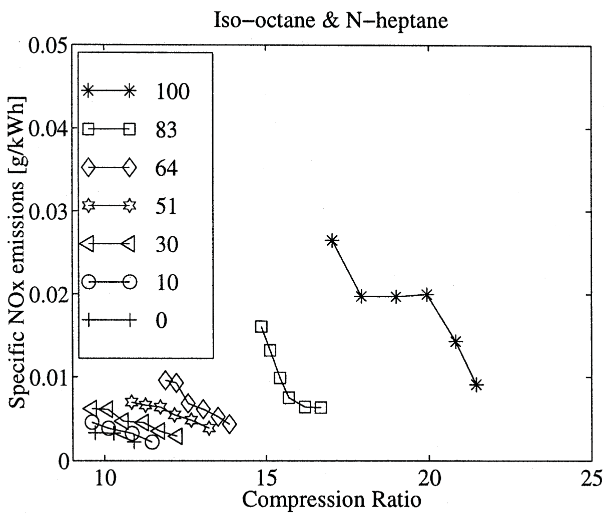

Figure 31. Specific NOx emission plotted versus the compression ratio used. Inlet temperature was varied for each test case with a constant fuel octane number.

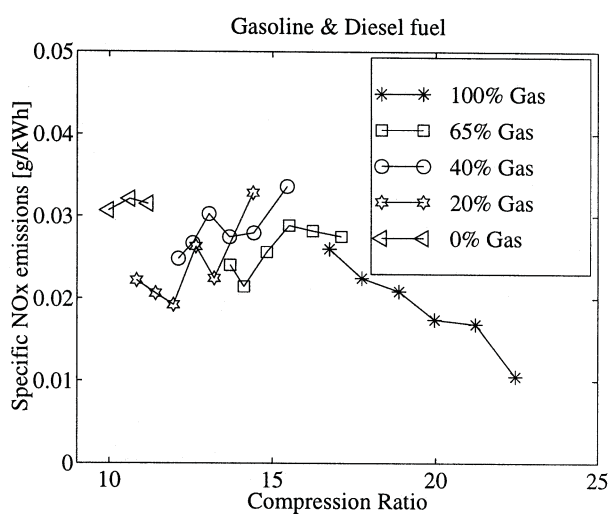

Figure 32. Specific NOx emission plotted versus the compression ratio used. The inlet temperature was varied for each test case with a constant fraction of gasoline.

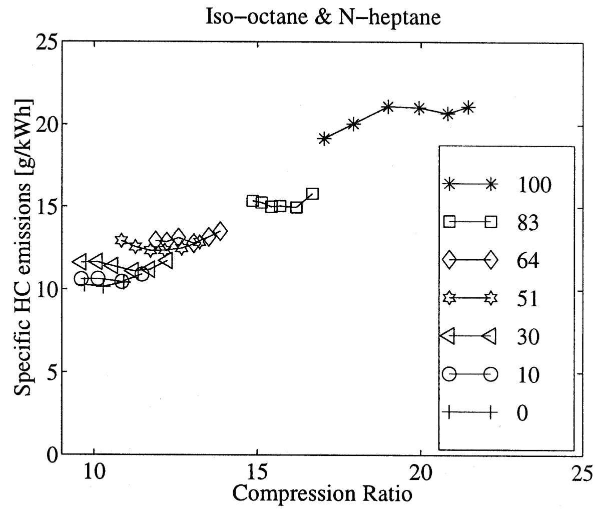

Figure 33. Specific HC emission plotted as function of the compression ratio used.

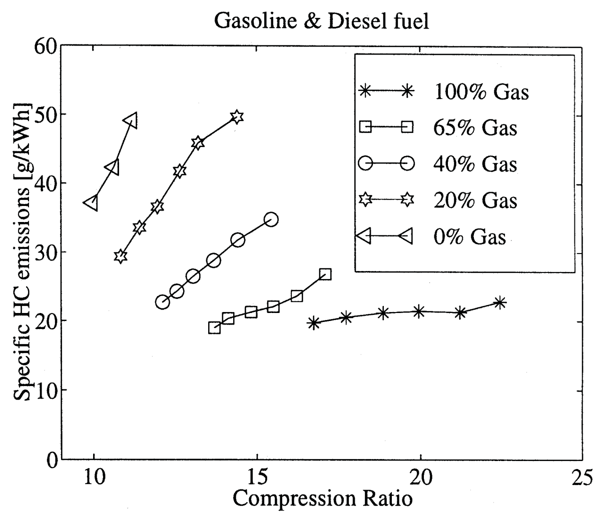

Figure 34. Specific HC emission plotted as function of the compression ratio used for different mixture ratios of gasoline and diesel fuel.

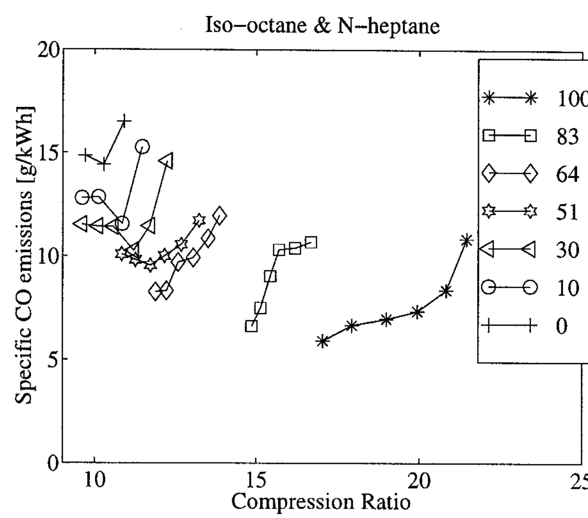

Figure 35. Specific CO emissions plotted versus the compression ratios used.

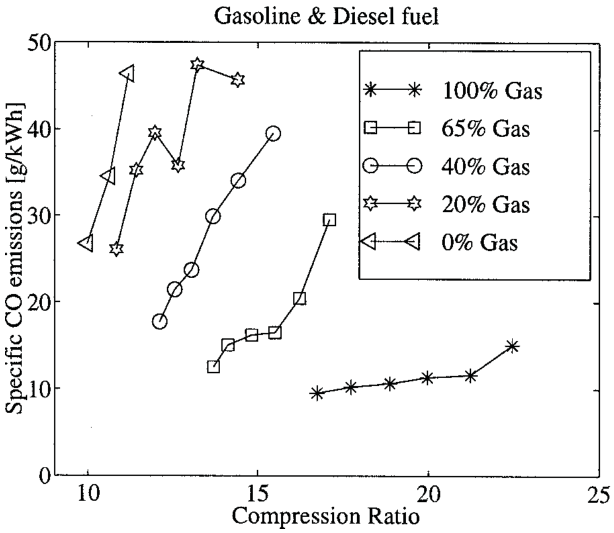

Figure 36. Specific CO emissions plotted versus the compression ratios used for different mixture ratios of gasoline and diesel fuel.

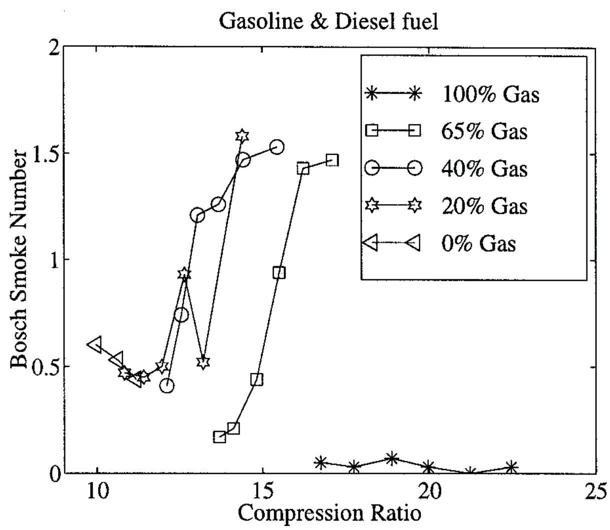

Figure 37. Measured smoke emission for the test serie with gasoline and diesel fuel.

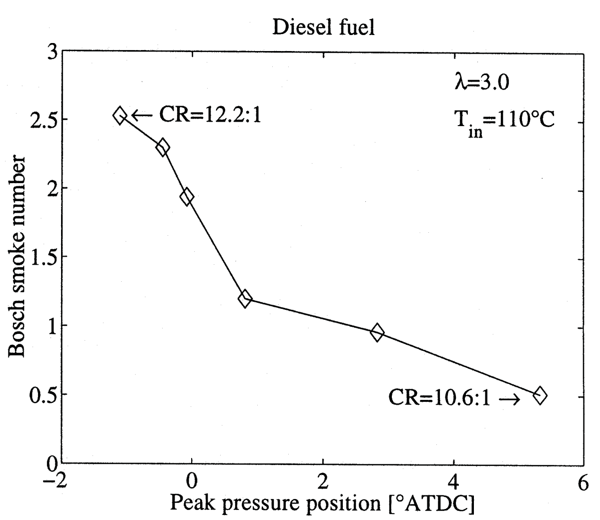

Figure 38. Smoke emission as function of peak pressure position.

COMBUSTION EFFICIENCY – The combustion efficiency was evaluated from the exhaust gas composition and it is a measure of how complete the combustion is. The calculated combustion *inefficiency* is mainly dependent on the concentration of unburned hydrocarbons and CO in the exhaust gases. NOx formation also affects the combustion efficiency. But as the NOx emissions are extremely low for HCCI at part load operation, they have an insignificant effect on the combustion efficiency.

In the case with the reference fuels, the combustion efficiency is strongly affected by the compression ratio. The combustion efficiencies are plotted in Figure 39. For the span of compression ratios used in this study, the combustion efficiency improves almost linearly with reduced compression ratio. This behavior can be explained by the reasons discussed in the HC emission and CO emission paragraphs above. The combustion efficiency is overall quite low. The mixture near walls and trapped in crevice volumes will probably not burn completely or do not ignite at all. With the very lean mixtures used here, quenching distance is expected to be quite long. The best combustion efficiency is around 93 %, and this in the cases with low compression ratios.

In the case with gasoline and diesel fuel the trend is not clear. Here the effect of diesel fuel's poor vaporization quality is very strong. Instead the combustion efficiecy is very dependent on the inlet temperature, see Figure 40. With increased amount of diesel fuel the combustion efficiency becomes as low as 80 % in some cases. This will reduce the overall efficiency much.

A decrease in combustion efficiency will lead to a proportional decrease in the overall efficiency. This means that some of what gained in thermal efficiency with an increased compression ratio will be lost in reduced combustion efficiency.

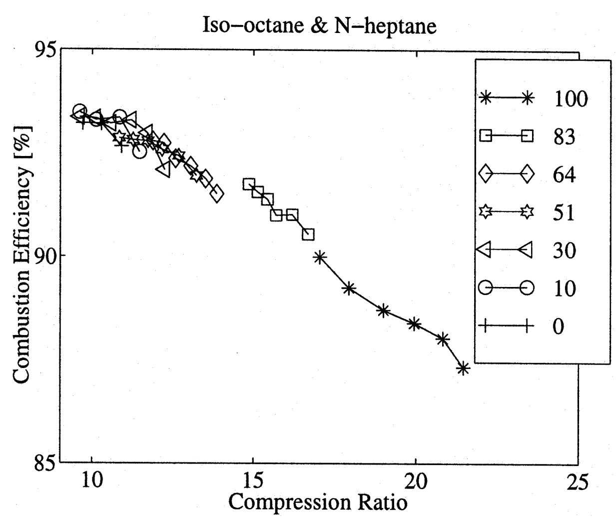

Figure 39. Combustion efficiency evaluated from the exhaust gas analysis.

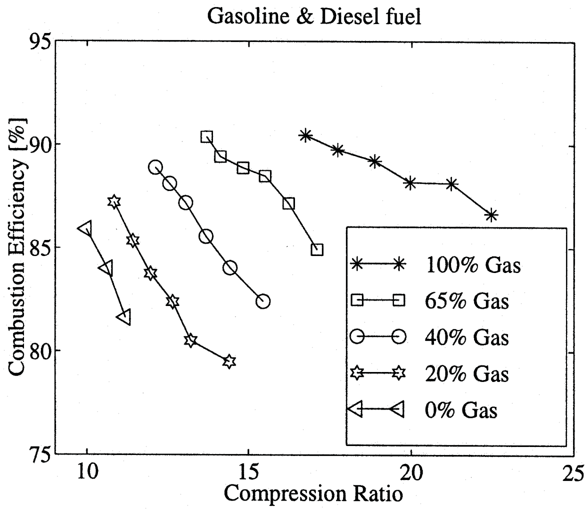

Figure 40. Combustion efficiency evaluated from the exhaust gas analysis. Gasoline and diesel fuel used.

DISCUSSION

The major advantages with HCCI are high efficiency and very low NOx emissions. HCCI can achieve indicated efficiencies comparable to values achieved in diesel engines. At light and medium loads extremely low NOx levels are achieved, about 1 to 2 ppm.

A serious problem with HCCI is high emissions of unburned hydrocarbons. High HC emissions means poor combustion quality, which reduces the overall engine efficiency.

The major problem with HCCI is not the high emissions of unburned hydrocarbons. This problem can surely be solved, at least with exhaust gas aftertreatment. The major problem is controlling the ignition timing over a wide load and speed range, since there is no such direct control over the ignition timing as there is in an SI and diesel engine. There are a lot of parameters that affect the ignition timing in a HCCI engine. The strongest ones are the inlet temperature and the compression ratio. By adjusting these parameters fast and correctly it would possible to control the ignition timing at any condition.

In this paper it has been demonstrated that almost any liquid fuel can be used in a HCCI engine with variable compression ratio. It was possible to run the engine on both pure n-heptane and on pure iso-octane, which have very low and very high auto-ignition temperatures respectively. It was possible to switch from pure n-heptane to pure iso-octane with the engine running. The results also show that when port injection is used, the fuel has to be quite easy to atomize, otherwise one will not get a homogeneous fuel/air mixture. An inhomogeneous mixture will lead to increased emissions of HC, CO and to soot formation. This was the case when diesel fuel was used.

CONCLUSIONS

The test results show that almost any liquid fuel can be used in a HCCI engine using a variable compression ratio. Operation with pure n-heptane required a compression ratio of about 11:1 to get auto-ignition at TDC, without the use of inlet air preheating. Under the same conditions, iso-octane required 21.5:1, gasoline (98 RON) required 22.5:1. It was expected that iso-octane would require a higher compression ratio than gasoline, but this was not the case. With pure diesel fuel and low inlet temperature (below 90°C) the combustion quality became very low, due to the poor atomization and vaporization. Therefore operation with diesel fuel without the use of inlet air preheating was of no interest. But with an inlet air temperature of 90°C, diesel fuel required about 11:1 to ignite at TDC.

The NOx emissions were generally very low. They did not increase much with increased fuel octane number and incresed compression ratio.

With diesel fuel much smoke was generated in some cases. But with gasoline almost no smoke was generated at all. With iso-octane and n-heptane no detectable smoke was generated.

The combustion efficiency for iso-octane and n-heptane decreased with increased compresion ratio. With mixtures of gasoline and diesel fuel the combustion efficiency is so strongly affected by the poor vaporization of the diesel fuel that no other trends can be detected, regarding the compression ratio.

The indicated efficiency did not improve with increased compression ratio as much as expected. This was a result of the decrease in combustion efficiency. The best gross indicated efficiency obtained here is close to 43 %. This value was achieved with a compression ratio of 21.5:1, and running on iso-octane.

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