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ABSTRACT

Hydrogen has been proposed as a possible fuel for automotive applications. Methanol is one of the most efficient ways to store and handle hydrogen. By catalytic reformation it is possible to convert methanol into hydrogen and carbon monoxide. This paper reports an experimental investigation of Reformed Methanol Gas as Homogeneous Charge Compression Ignition (HCCI) engine fuel. The aim of the experimental study is to investigate the possibility to run an HCCI engine on a mixture of hydrogen and carbon monoxide, to study the combustion phasing, the efficiency and the formation of emissions.

Reformed Methanol Gas (RMG) was found to be a possible fuel for an HCCI engine. The heat release rate was lower than with pure hydrogen but still high compared to other fuels. The interval of possible start of combustion crank angles was found to be narrow but wider than for hydrogen. The high rate of heat release limited the operating range to lean ($\lambda > 3$) cases as with hydrogen. On the other hand, operation on extremely lean mixtures ($\lambda = 6$) was possible. The operating range was investigated using intake air temperature for control and also this control interval was found to be narrow but more extensive than for pure hydrogen, especially when richer cases were run. The maximal load in HCCI mode was a net Indicated Mean Effective Pressure (IMEP_n) of 3.5 bar for RMG. This is the same maximum IMEP_n as for hydrogen. It is about half the load possible in Spark Ignition (SI) mode and about half the maximal load in HCCI mode with other fuels. For the loads where HCCI operation was possible, indicated thermal efficiency for HCCI was superior to that of SI operation. The indicated overall efficiency of the engine-reformer system is as high for SI as for HCCI operation when RMG is used as fuel. NO_x emissions were, as expected, found to decrease when the equivalence ratio was lowered. High levels of carbon monoxide were found in the exhaust. Emissions of hydrocarbons were detected, probably originating from evaporated and partially oxidized lubrication oil.

INTRODUCTION

The engine manufacturers of today are challenged by the legislative demands of low emissions and the need to decrease the dependency on non-renewable fuels, such as oil. Reformed Methanol Gas (RMG), a mixture of hydrogen and carbon monoxide in the volume ratio 2:1, has been suggested as possible supplement or replacer of the fuels used today [1-3]. Methanol can be produced by sustainable methods but it can also be produced from natural gas if this shows to be a better solution.

A major problem with hydrogen and natural gas, other suggested fuels of the future, is the storage and distribution. Ordinary fuels of today such as petrol or diesel are liquids and thereby relatively easy to handle but hydrogen and natural gas, as gases, will require a new infrastructure and major development of the vehicle gas tank. One of the most efficient (in terms of volume) ways to store hydrogen is as hydrocarbons and the shortest alcohol, methanol, is an excellent hydrogen carrier [4].

After the oil crisis in the 70's the interest in fuels that would minimize the dependence on imported oil exploded. One of the possible candidates was methanol that could be produced from biomass. Thus, numerous studies and fleet tests were conducted with methanol during the 70s and 80s [5-6]. In many cases the goal for these studies was simply to demonstrate that methanol could replace petrol and diesel as Internal Combustion Engine (ICE) fuel. However, a considerable amount of work was invested in evaluating the efficiency gain potential of methanol in SI engines. In this context the potential of RMG was studied thoroughly. [3,7,8].

Today the interest in methanol is rising again as a potential hydrogen carrier for fuel cells. This means that the reformer technology is rapidly improving. Methanol used as engine fuel can be catalytically converted to RMG on demand. The chemical reactions producing RMG from methanol are endothermic and RMG contains 20% more energy than the methanol from which it was

formed [2-3]. The energy needed can be taken from the exhaust gases, assuming they are hot enough to contain the energy. In this way the methanol reformer acts as a chemical turbo [1].

The many studies made with RMG as SI engine fuel, both in premixed and direct injected engines, have identified a number of problems similar to the ones identified for hydrogen. Problems associated with premixed SI operation are ignition in the intake due to hot valves and non-optimal valve timing as well as ignition in the tail pipe after repeated misfires. Other problems identified are increased heat loss and high combustion rate.

But how will RMG act as fuel in an HCCI engine? The HCCI engine can be considered as a hybrid of the SI and CI engines since it uses a premixed charge like the SI engine but combustion is initiated by compression heat like in the CI engine. The major benefit of the HCCI engine compared to the CI engine is the absence of Nitrogen Oxides (NO_x) and Particulate Matter (PM) since the combustion is lean all over the charge. Compared to the SI engine, the HCCI engine presents significant improvement of part load fuel economy. A 100% improvement at 1.5 bar BMEP has been reported [9].

The HCCI engine has been reported to be very fuel flexible [10] with the correct adaptation of the engine configuration. In a previous study [11], HCCI combustion of pure hydrogen was investigated and recent studies of RMG as engine fuel have been made by Shudo et al. [2,12,13]. They performed SI engine tests with RMG mixtures and they also studied RMG–DME mixtures in combination with the new engine concept, HCCI, but no previous experimental study has focused on HCCI combustion of RMG only.

The question raised in this paper is to what extent HCCI engines could benefit from the progress in reformer technology, i.e. running on RMG. One of the most interesting questions concerning RMG as HCCI fuel is whether the exhaust gas reaches a temperature high enough to drive the catalytic reformation of methanol, which is necessary to take full advantage of the concept. Another interesting question is the effect of the carbon present. When RMG is used, the hydrogen is accompanied by carbon monoxide. The chemistry is changed and thus possibly also the combustion behaviour. Will HCCI combustion of RMG be possible in an equivalence ratio interval as wide as the one encountered for pure hydrogen? Will the control possibilities of RMG HCCI by the use of intake temperature be as limited as the one found for hydrogen? RMG contains carbon in the fuel and in which form will this be found in the exhaust gas? Experiments with synthetic RMG are presented. The operating range and the performance of RMG HCCI are compared to SI operation as well as to hydrogen fuelled HCCI.

EXPERIMENTS

EXPERIMENTAL SETUP

The experiments were conducted on a modified Volvo TD100 six cylinder CI engine. The primary modification of the research engine is the deactivation of all cylinders except for one. The pistons in the five deactivated cylinders were connected to the crankshaft and thus generate friction but the pressure in the cylinders was the same as in the crankcase. This modification decreases the displaced volume of the engine to 1.6 liter. Other conversions of this engine include the addition of a pressure transducer to the cylinder head and the possibility to add a spark plug when running the engine in SI mode. In SI mode, NGK sparkplugs (NGK BCP5ES) with 0.8 mm gap and a dwell time of 20 CAD were used. The engine can use a number of different connecting rods, pistons and piston heads allowing a great flexibility to choose the compression ratio and combustion chamber geometry. In this study the focus have been on the changes of compression ratio, keeping the piston top flat to give a pancake combustion chamber. Additional information on the engine is found in Appendix 1.

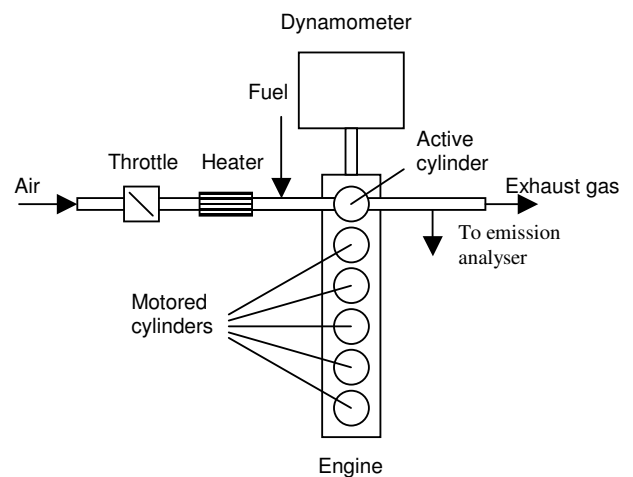


Figure 1. The test engine setup.

The test engine was not equipped with a methanol reformer. The RMG supplied was synthetically produced in the intake by mixing hydrogen and carbon monoxide. The carbon monoxide and hydrogen was mixed as two continuous streams added to each other. The mixture then flowed 1.2 meters in a tube (10 mm diameter) before being added to the air in the intake pipe, a few decimeters upstream the intake valve. The gas was technical grade and supplied from bottles. The amount of gas was controlled by Bronkhorst thermal mass flow controllers.

The cylinder pressure transducer was a Ø14 mm water cooled Kistler 6071B. The pressure was sampled each

0.2 CAD with 16 bit resolution. All pressure traces shown in this publication are averaged over 300 cycles.

A Boo Instrument system for emission analysis was used. The system consists of a UNOR, Non-Dispersive Infra Red detector (NDIR), produced by Maihak. This instrument measures carbon dioxide and carbon monoxide concentrations. Oxygen is handled by a paramagnetic detector (PMD). A CLD 700 EL Chemi Luminescence Detector (CLD) from Eco Physics measures nitric oxide and nitrogen dioxide content. Hydrocarbons were measured by a Flame Ionisation Detector (FID) model 109A produced by J.U.M. Engineering. The Boo Instrument equipment was supplemented by a T3HYE CiTiceL electrochemical hydrogen detector, produced by City Technology, when pure hydrogen was used as fuel.

EXPERIMENTAL PROCEDURE

HCCI engine tests

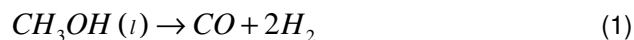
HCCI experiments were mainly conducted as follows: For each combination of compression ratio (15:1, 17:1 and 20:1) and lambda (3 – 6), an intake temperature sweep was conducted to find the intake gas temperature able to sustain HCCI combustion. Limiting factors were lack of autoignition at too low intake temperatures and autoignition too early and/or too fast ($dP/dCAD > 60 \text{ bar}$) at too high intake temperatures.

SI engine tests

The main SI experimental study was conducted as follows: For each combination of compression ratio (9 and 11.2), intake manifold pressure (0.4, 0.6, 0.8 and 1.0 bar) and lambda (1.3 – 2.0), an ignition angle sweep was conducted to find the ignition angles able to sustain SI combustion. Limiting factors were autoignition (knock) and too fast ($dP/dCAD > 60 \text{ bar}$) combustion at too early ignition angles. At too late angles backfire into the intake manifold and misfire were limiting factors.

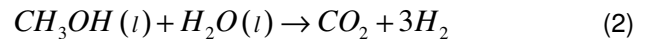
REFORMER CALCULATIONS

Reformation of methanol can be made in a number of ways [1]. Anaerobic decomposition of methanol leading to RMG is shown below (reaction 1).



The reaction above is endothermic and the required energy input is 20% of the Lower Heating Value (LHV) of the supplied methanol. RMG will, when combusted, return this energy and the LHV will thus be 120% of the original methanol.

Methanol can also be steam reformed. In this process the methanol is converted to hydrogen and carbon dioxide according to the reaction below. (reaction 2)



Steam reforming also requires an additional 20% energy input but when it is combusted it only delivers 13.7% more energy compared to the original methanol. The remaining energy is trapped and will not be recovered unless the water is condensed.

No experiments were made with mixtures of hydrogen and carbon dioxide but efficiencies and exhaust temperatures from experiments with pure hydrogen as fuel have been used to estimate the efficiency of this mixture. As the mixtures used are very lean the obliteration of the carbon dioxide in the fuel should not result in a major error.

Calculations of the available exhaust gas energy were made. The energy was calculated from measured exhaust temperature, measured and calculated exhaust composition and from NASA polynomial data. The energy found was compared to the amount theoretically needed to convert liquid methanol into RMG by anaerobic reformation or hydrogen by steam reformation, assuming the converter is operated at 200°C [1] and that all energy above this level can be used in the reformation process.

RESULTS AND DISCUSSION

RATE OF HEAT RELEASE

Figure 2 shows pressure traces from HCCI combustion of RMG. The displayed pressure traces represent five different cases of HCCI combustion when the intake gas temperature is changed, decreasing from left to right. The response of the pressure trace to a small intake temperature change is high.

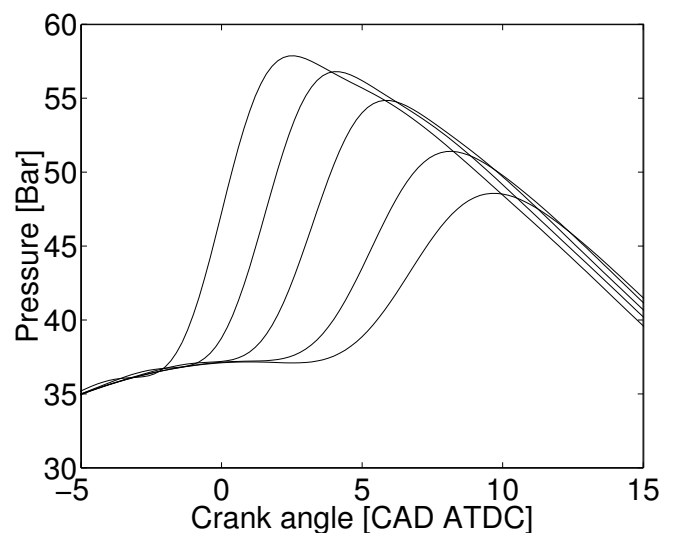


Figure 2. Cylinder pressure as a function of CAD. Engine is run at 1200 RPM, lambda 4.5 and compression ratio 17:1. Five different intake gas temperatures (from left to right 135, 132, 129, 126 & 124°C) giving five different pressure traces.

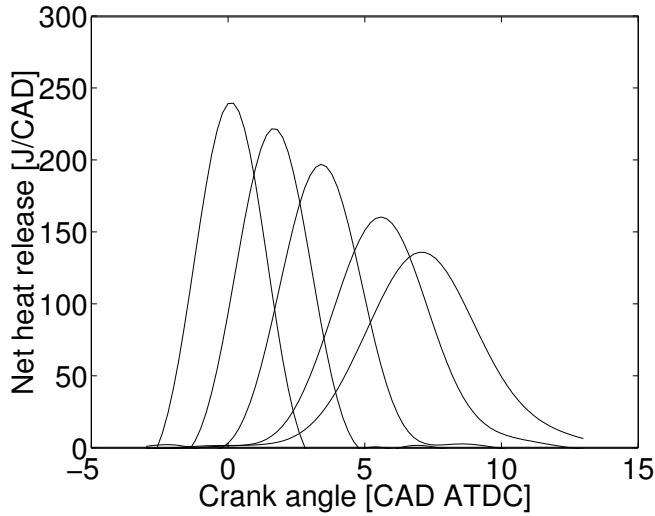


Figure 3. Net heat release as a function of CAD calculated from the pressure traces in Figure 2.

This investigation of RMG shows a very fast heat release (Figure 3) when RMG is combusted in HCCI mode. As can be seen, the start of combustion (SOC) becomes earlier when the intake gas temperature is increased because ignition temperature is reached earlier in the cycle. The rate of heat release changes when the intake temperature and thus the combustion phasing is changed. Figure 4 shows the rapid burn angle (RBA, 10-90% heat release) decreasing when the intake gas temperature increases. Higher intake temperature and a more compressed charge during combustion cause higher temperature and pressure and thus higher burn rate. The intake temperature required to achieve a certain 10-90% heat release angle is higher for RMG than for hydrogen.

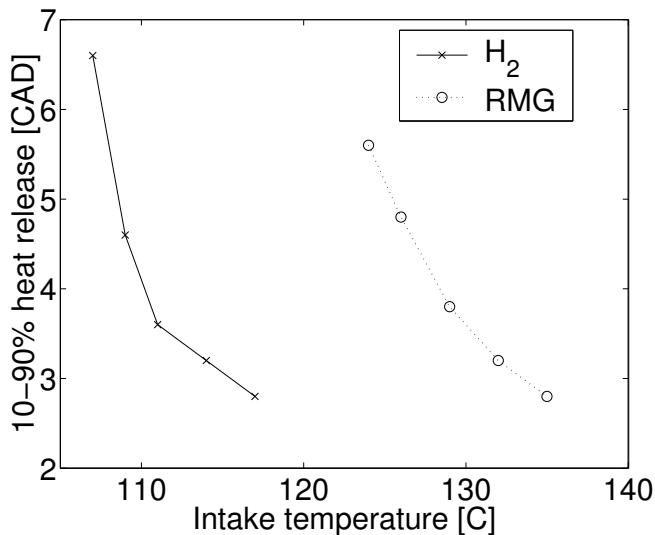


Figure 4. 10-90% heat release angle as a function of intake gas temperature for the RMG cases shown in figures 2-3 and the corresponding 10-90% heat release angles for hydrogen.

Figure 5 shows the required intake temperatures to keep the 10-90% heat release at 4 CAD. The required intake temperature increases as the mixture becomes leaner.

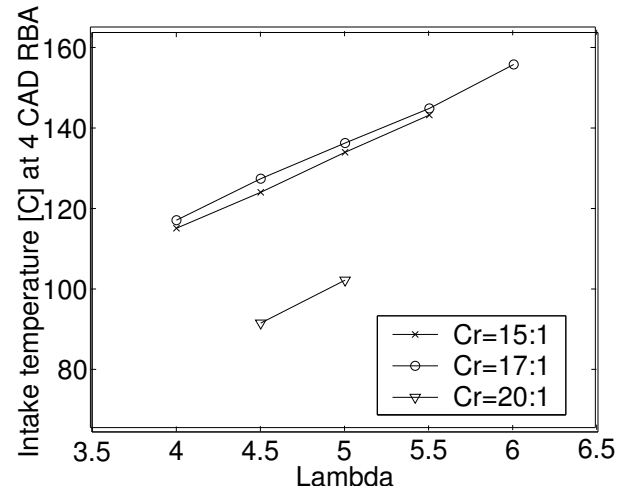


Figure 5. Intake gas temperature required to sustain 4 CAD 10-90% heat release angle as a function of lambda for an engine speed of 800 RPM and compression ratios of 15:1, 17:1 and 20:1.

The required temperature at a 10-90% heat release angle of 4 CAD follows the same increasing trend for both hydrogen and RMG as lambda increases (Figure 6). As in Figure 4 the hydrogen fuelled case requires a lower intake temperature than the RMG does.

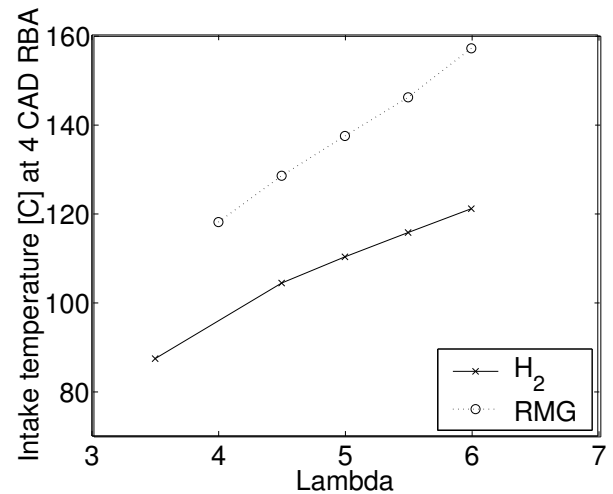


Figure 6. Intake gas temperature required to sustain 4 CAD 10-90% heat release angle as a function of lambda for hydrogen and RMG at an engine speed of 800 RPM and a compression ratio of 17:1.

The required temperature for combustion onset (estimated ignition temperature) as a function of lambda (Figure 7) shows an increase as lambda increases. No trend is seen when engine speed is changed. The SOC temperature is between 940 and 1000 K in the RMG case. That is slightly higher than when hydrogen is used as fuel. A weak increase in SOC temperature when lambda increases can be seen.

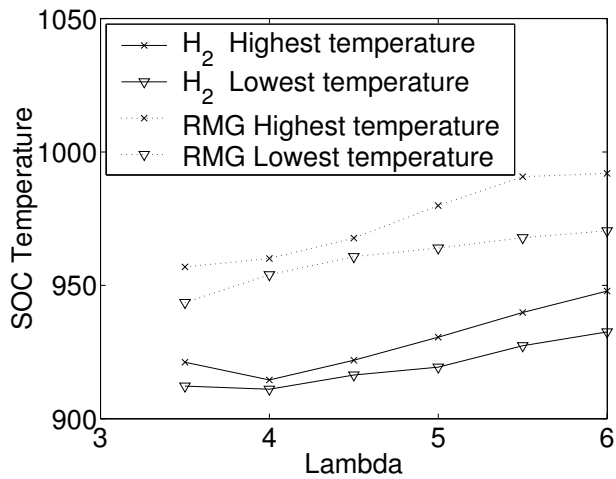


Figure 7. Lambda – SOC temperature area within which operation is possible in HCCI mode with hydrogen and RMG as engine fuel. The engine is operated at compression ratio 17:1 and engine speed of 1200 RPM.

SOC temperature is defined as the temperature at the Start Of Combustion. The temperature is calculated from the ideal gas law starting at the state at Intake Valve Closing (IVC). Since heat transfer from the intake port and the cylinder walls has an impact on the gas temperature at IVC and thus on the trapped mass, used to calculate temperature before the compression, this adiabatic approach will probably over-predict the trapped mass and thereby underestimate the temperature

RANGE OF OPERATION

The HCCI engine was operated with mixtures between $\lambda=3$ and $\lambda=6$. Tests were completed at various compression ratios and engine speeds. Figure 8, 9 and 10 present the required inlet temperature for earliest and latest possible combustion (see HCCI engine tests section above) at engine speeds of 800, 1200 and 1600 RPM (Figure 8), at compression ratios of 15:1, 17:1 and 20:1 (Figure 9) and with hydrogen and RMG as engine fuel (Figure 10). The effect of compression ratio on the required intake gas temperature is seen in Figure 9. The increase of compression ratio lowers both the highest and the lowest SOC temperature. Figure 10 shows the difference in intake temperature range between hydrogen and RMG. The required temperature is higher for RMG but the range is wider.

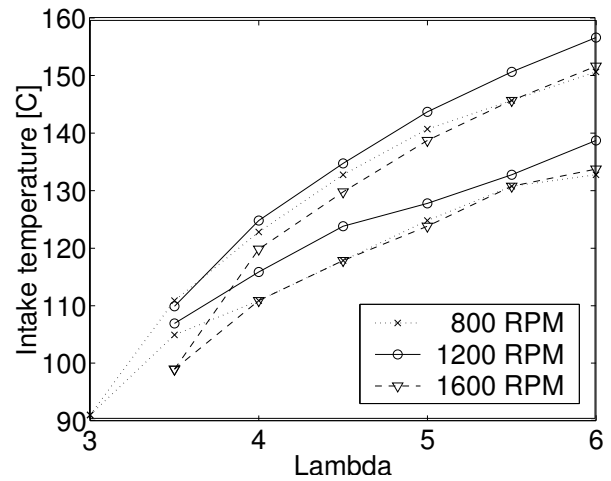


Figure 8. The maximal and minimal intake manifold temperature at which operation is possible in HCCI mode as a function of lambda. Displayed cases have engine speeds of 800, 1200 and 1600 RPM. Compression ratio 17:1.

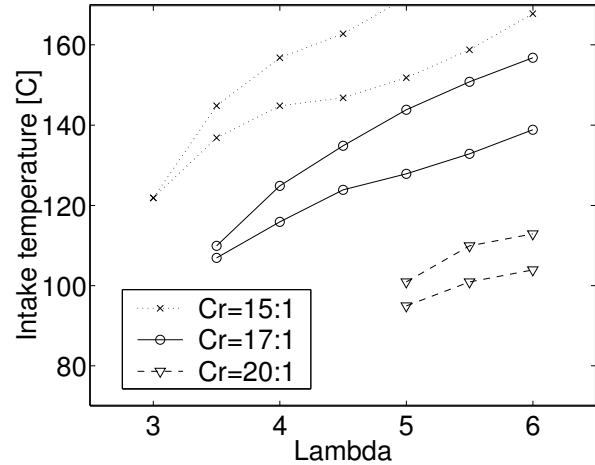


Figure 9. The maximal and minimal Intake manifold temperature at which operation is possible in HCCI mode as a function of lambda. Displayed cases have an engine speed of 1200 RPM and Compression ratios 15:1, 17:1 and 20:1.

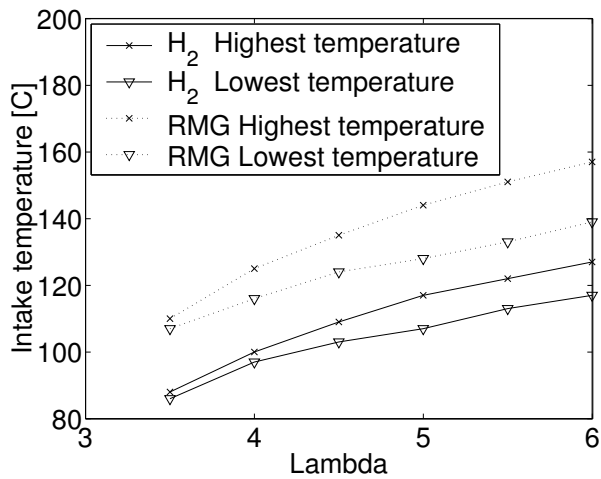


Figure 10. The maximal and minimal intake manifold temperature at which operation is possible in HCCI mode as a function of lambda. Displayed RMG and hydrogen fuelled cases have an engine speed of 1200 RPM and compression ratio 17:1.

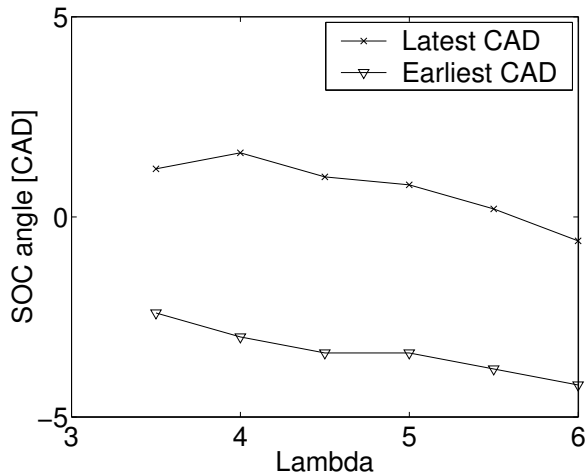


Figure 11. The lambda – Auto ignition angle area within which operation is possible in HCCI mode at 1200 RPM and compression ratio 17:1.

Figure 11 shows the highest and the lowest possible ignition angle as a function of lambda for 1200 RPM and 17:1 compression ratio. As in Figure 8, 9 and 10 intake temperature was used for control of ignition angle. The band of possible ignition angles is narrow. The earliest and the latest possible ignition angles decrease slightly with an increasing lambda but the decrease is small and will probably not have any practical effect on HCCI engine operation with RMG. As the possible ignition angle interval becomes earlier, the combustion must be phased earlier than what is optimal for efficiency.

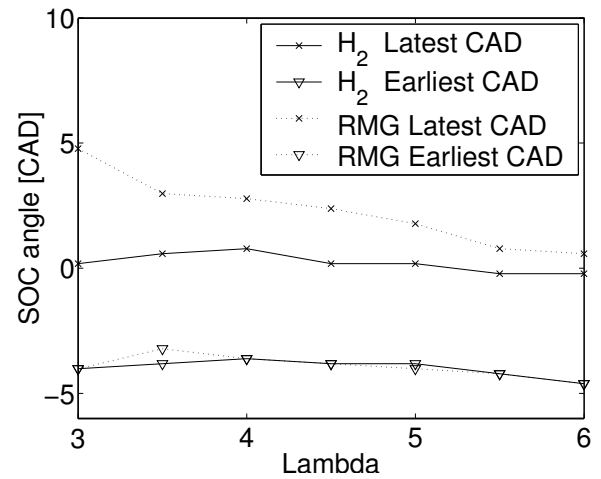


Figure 12. The lambda – SOC angle area within which operation is possible in HCCI mode at engine speeds 800 RPM, 1200 RPM or 1600 RPM with compression ratio 15:1, 17:1 and 20:1.

Figure 12 shows the SOC angle range where operation with HCCI has been successful for at least one studied case as a function of lambda. Worth noting in this figure, compared to Figure 11, is the small effect a change in compression ratio or engine speed has on the available ignition angles (compare Figure 11 and Figure 12). The only major difference seen when comparing hydrogen and RMG in Figure 12 is the slightly later ignition interval at richer mixtures for RMG.

Figures 8-10 show the narrow intake temperature ranges allowed if HCCI combustion should be maintained and the SOC ranges shown in Figure 11 and Figure 12 are much narrower than petrol SI ranges which usually are in the range of 30 CAD (CA50). This, in combination with the limited temperature span available, will probably lead to difficulties to control HCCI combustion when RMG is used as fuel.

EFFICIENCY

For RMG HCCI the net indicated thermal efficiency (ITEn) and the net indicated mean effective pressure (IMEPn) both decrease as the lambda increases (Figure 13). At lambda 4, misfire/partial burn occurs before Maximum Brake Torque timing (MBT) is reached. At lambda 5, MBT is achieved and at lambda 6 the temperature does not get high enough to reach MBT.

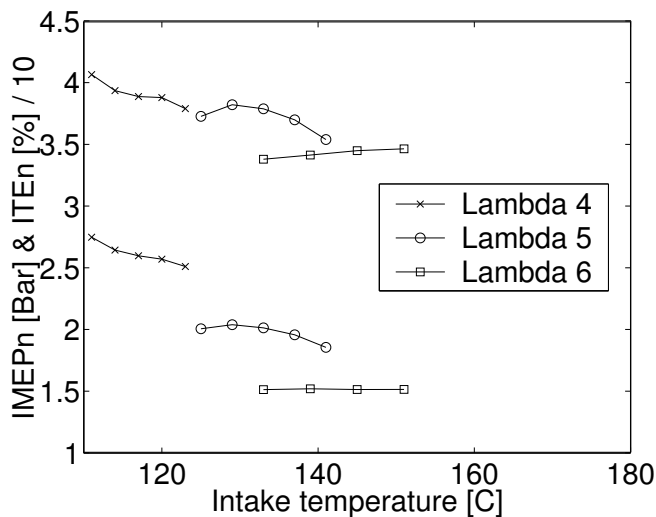


Figure 13. ITEn (curves above) and IMEPn (curves below) as a function of intake gas temperature. Three lambda values (4, 5 & 6) displayed at engine speed 800 RPM and compression ratio 17:1.

Figure 14 shows ITEn and IMEPn as a function of ignition angle instead. At lambda 4 the optimal timing should be later than the latest achievable, at lambda 5 MBT is achieved and at lambda 6 the optimal timing should be earlier than the earliest achievable. This is in line with the results seen in Figure 13.

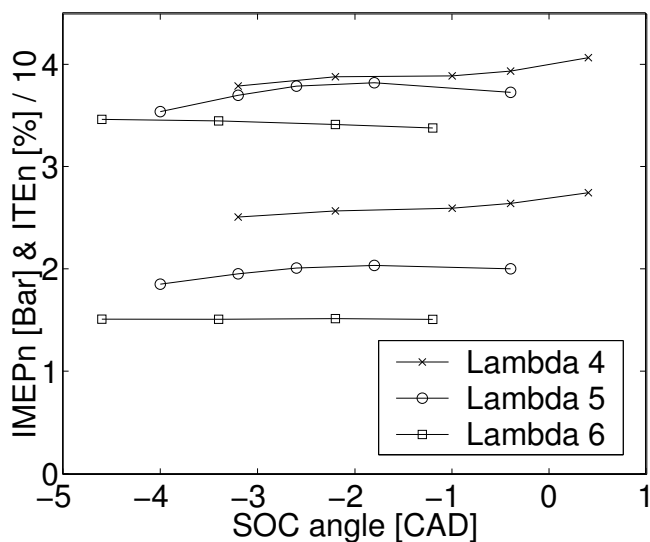


Figure 14. ITEn (curves above) and IMEPn (curves below) as a function of SOC CAD. Three lambda values (4, 5 & 6) displayed at an engine speed of 800 RPM and compression ratio 17:1.

In Figure 15 the effect of lambda on ITEn is shown for three engine speeds. The ITEn of 1200 and 1600 RPM are lower due to the poor gas exchange of the two-valve engine used. Poor gas exchange results, in addition to a higher PMEP, in a higher amount of hot residual gases, advanced ignition and lowered ITEn.

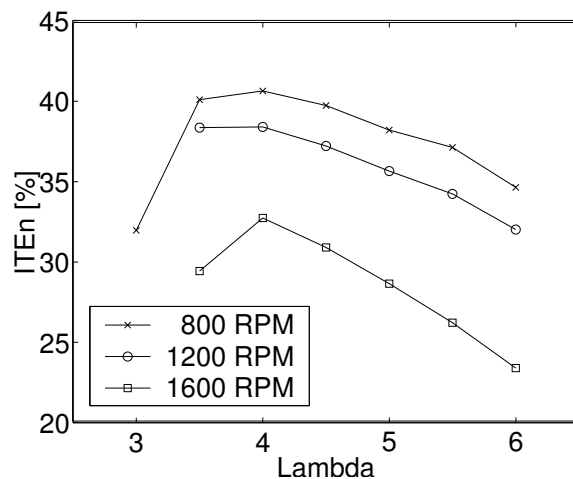


Figure 15. Maximal ITEn as a function of lambda. Three engine speeds (800, 1200 & 1600 RPM) displayed at compression ratio 17:1.

EMISSIONS

In Figure 16 and Figure 17 emission levels from an intake temperature sweep (as a function of SOC CAD in Figure 17) can be seen. HCCI combustion of RMG leaves a high level of carbon monoxide in the exhaust. Exhaust gas concentrations of NOx and HC are low. HC emissions come from unburned or partially burned lubrication oil. Nitric oxide increases with increased maximum temperature. The level of NO increases and the level of CO decreases when the intake temperature increases and the combustion becomes earlier and faster.

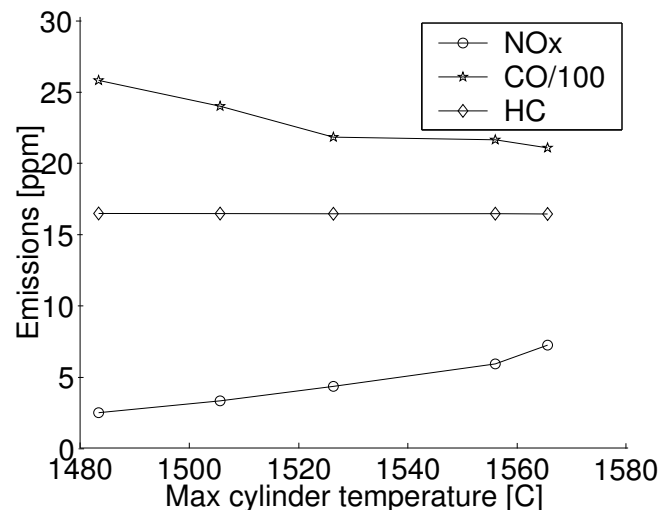


Figure 16. Engine emissions (PPM) as a function of intake gas temperature. Engine speed 800 RPM, compression ratio 17:1 and lambda 4.

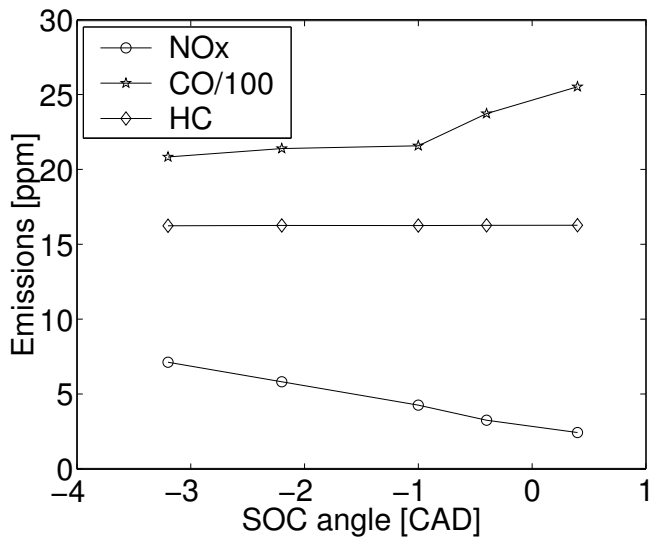


Figure 17. Engine emissions (PPM) as a function of SOC CAD. Engine speed 800 RPM, compression ratio 17:1 and lambda 4.

Figure 18 shows net indicated specific emissions. The specific emissions are influenced by both emission concentrations and efficiency. With higher intake temperature the efficiency is reduced at lambda 4, giving slightly higher specific emission concentrations at high intake temperature.

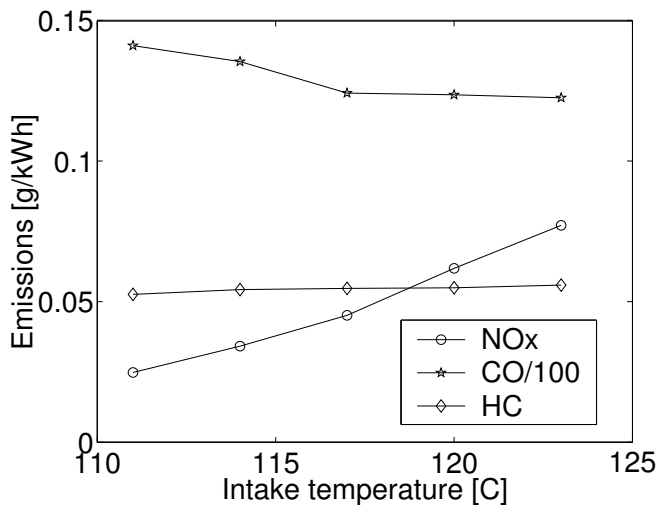


Figure 18. Engine emissions ($\text{g/kWh}_{\text{indicated}}$) as a function of intake gas temperature. Engine speed of 800 RPM, compression ratio 17:1 and lambda 4.

The effect of lambda on engine emissions at maximal ITE is seen in Figure 19. When lambda is increased, the NOx level is decreased and carbon monoxide is increased, due to lower combustion temperature. Hydrocarbon levels are at very low levels for all lambdas.

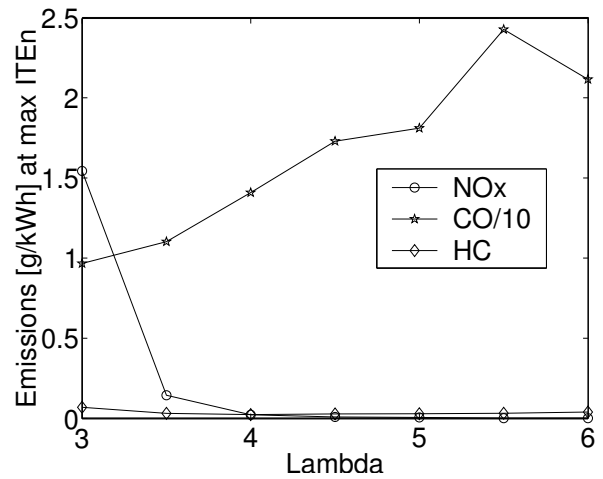


Figure 19. Engine emissions ($\text{g/kWh}_{\text{indicated}}$) at maximum ITE as a function of lambda at an engine speed of 800 RPM and a compression ratio of 17:1.

Figure 20 shows a comparison of hydrogen and RMG fuelled HCCI. Both cases have high levels of unburned fuel. The levels of NOx are comparable but RMG generates more NOx at the richest point of operation. Hydrocarbon levels are very low for both cases.

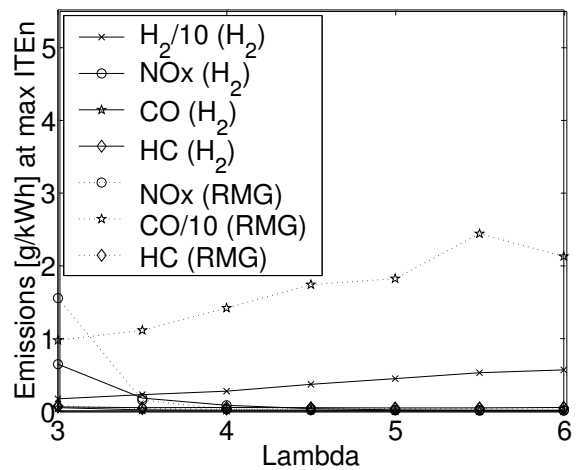


Figure 20. Engine emissions ($\text{g/kWh}_{\text{indicated}}$) at maximum ITE as a function of lambda at an engine speed of 800 RPM and a compression ratio of 17:1 for both hydrogen and RMG fuelled cases.

HCCI COMPARED TO SI

Maximal efficiency at each compression ratio is displayed in Figure 21. SI engine operation at the lower compression ratio and HCCI operation had about the same maximum ITE. From this point the choice of SI or HCCI seems arbitrary but if we have a look at the exhausts and the efficiency-load dependence we will see that this is not the case.

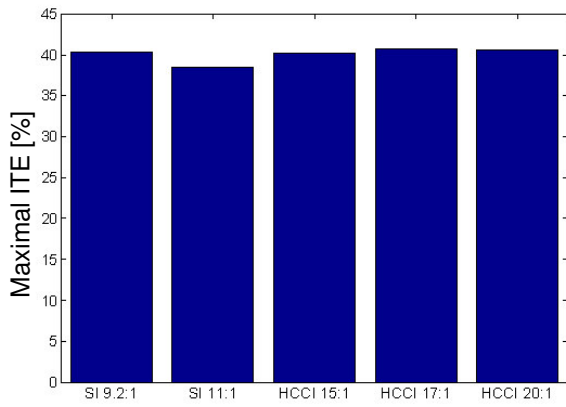


Figure 21. Maximal ITE at each compression ratio. SI operation at compression ratio 9.2 and 11, HCCI at 15:1, 17:1 & 20:1.

Figure 22 and 23 show net (Figure 22) and gross (Figure 23) ITE as a function of corresponding IMEP. Each cross (HCCI) or ring (SI) represents a single point of operation studied. The points of operation represent variations of a number of different operating parameters such as intake gas temperature (HCCI), ignition angle (SI), engine speed, compression ratio and throttling (SI).

From Figure 22 it can be concluded that RMG HCCI can be operated to give better efficiency than RMG SI in the range where HCCI operation is possible. The main reason for the superiority of HCCI is the reduced pumping work of the gas exchange.

The gross efficiencies of RMG HCCI and SI are comparable (Figure 23). Also worth noting is that although SI operation of this engine enables nearly twice the load of HCCI operation, both SI and HCCI operation still are limited in load to about half the load achievable with other fuels [14]. Berckmüller et al. [15] reported IMEPs of up to 18 bar when a DI engine was run on hydrogen with EGR and supercharging. Direct injection and EGR may be the solution if RMG fuelled engines are to be able to run at higher loads.

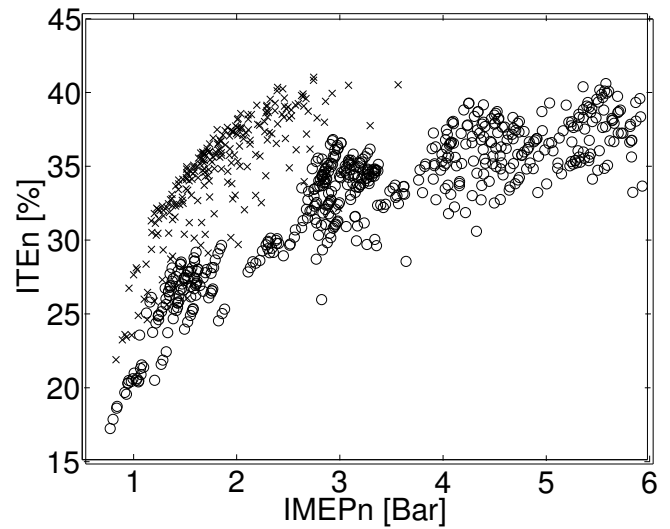


Figure 22. Net ITE – IMEP scatter plot for all investigated RMG fuelled cases. SI displayed as circles and HCCI as crosses.

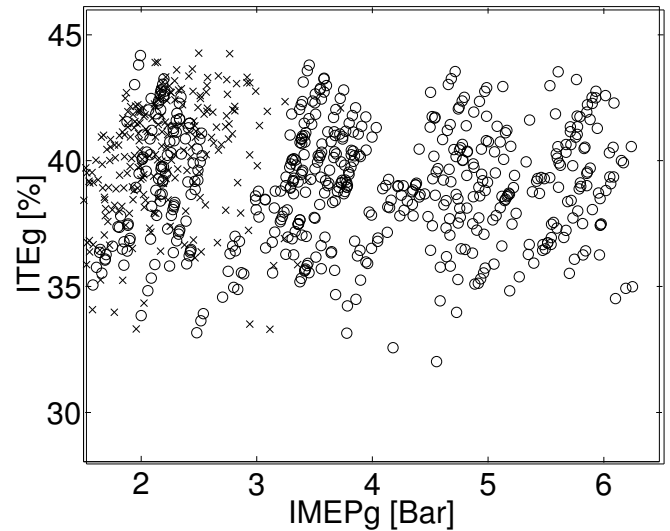


Figure 23. Gross ITE – IMEP scatter plot for all investigated RMG fuelled cases. SI displayed as circles and HCCI as crosses.

Figure 24 shows the upper limit contours of ITEn for hydrogen and RMG fuelled HCCI and SI operation. It compares the four studied cases. HCCI has a higher efficiency than SI for both fuels studied. The figure also shows that RMG HCCI operation is possible at lower IMEPn than hydrogen HCCI. This might however be a result of the experimental design. The study did not investigate operation at lambda values leaner than lambda 6. Since the lean-limit of hydrogen HCCI was not explored, operation at lower IMEPs than the one shown in Figure 24 could be possible. The maximum ITEn of RMG SI is found at a higher load than with hydrogen SI.

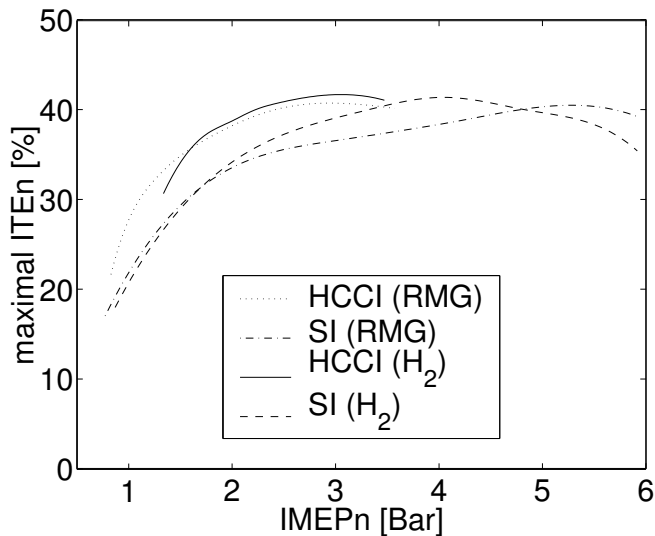


Figure 24. Maximal net ITE – IMEP plot for hydrogen and RMG fuelled HCCI and SI.

REFORMATION OF THE FUEL

The efficiencies displayed in figures 22-24 are all calculated from the heating values of hydrogen or RMG. One of the main advantages of using hydrogen or RMG from reformed methanol is however the potential efficiency gain from the catalytic reformation process which utilizes exhaust energy.

Figures 25 and 26 show the fraction of the required energy available in the exhaust gas for reformation of the fuel. As can be seen in the figures the exhaust gas of the SI cases has a high potential of making full use of the reformer (fraction > 1). The HCCI fuelled cases have too low exhaust temperatures to use the full efficiency gain of the reformation process (fraction < 1). Consequently if fueling an HCCI engine with RMG, additional heat must be added to the reformer in order to produce the required amount of RMG. This heat could be produced by an electrical heater, by a methanol burner or by advanced EVO.

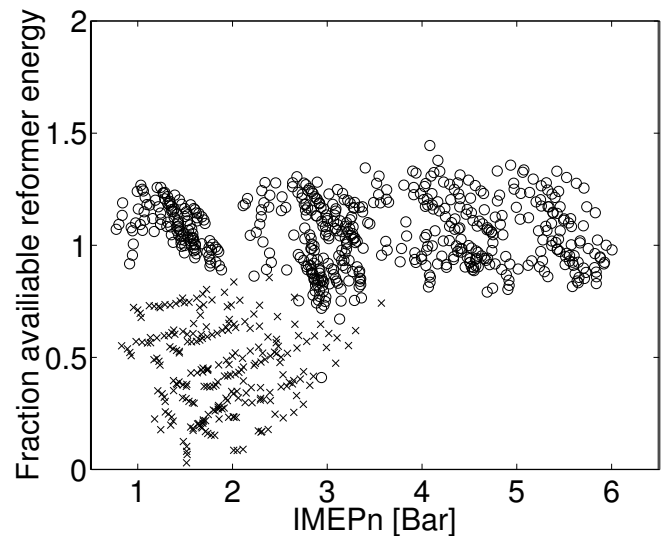


Figure 25. Fraction of required thermal energy for the reformation process as a function of IMEPn for RMG fuelled HCCI (crosses) and SI (rings).

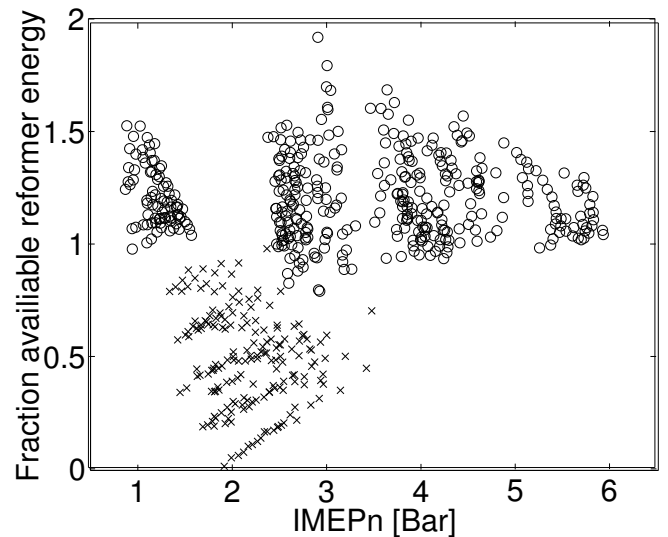


Figure 26. Fraction of required thermal energy for the reformation process as a function of IMEPn for hydrogen fuelled HCCI (crosses) and SI (rings).

The overall efficiency of an engine fuelled by reformed methanol is determined by the efficiency of the engine (seen in figures 22-24), the efficiency of the reformation process and the available exhaust gas energy needed for the reformation process (fractions seen in figures 25-26). If the efficiency of the reformation process is set to 100% the system efficiency can be calculated.

Figure 27-28 shows the efficiency including the reformation process. From the figures it can be concluded that the advantage of reformation is largest in the SI cases.

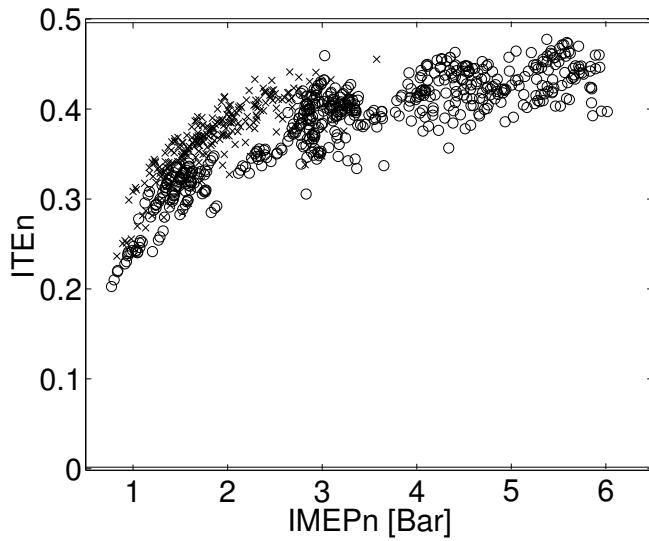


Figure 27. Net ITE – IMEP scatter plot for all investigated RMG fuelled cases. ITE_n calculated from methanol. SI displayed as circles and HCCI as crosses.

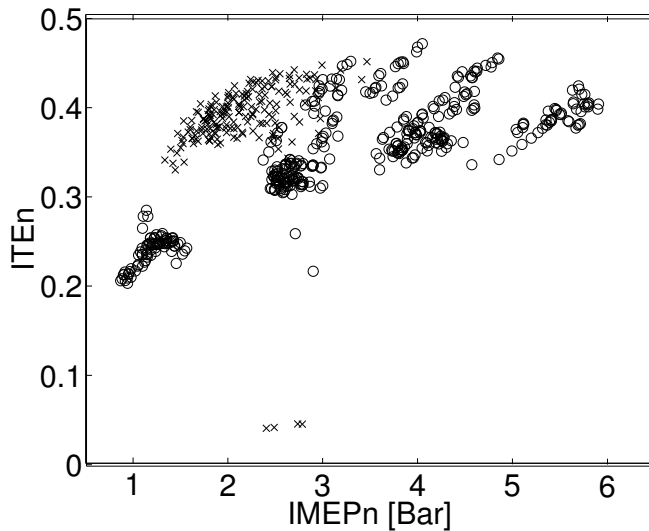


Figure 28. Net ITE – IMEP scatter plot for all investigated hydrogen fuelled cases. ITE_n calculated from methanol. SI displayed as circles and HCCI as crosses.

Figure 29 shows the upper limit contours of the efficiencies displayed in Figure 27-28 i.e. Figure 29 is Figure 24 redrawn with efficiencies calculated from methanol instead of RMG or hydrogen. From Figure 27-29 it can be seen that the overall efficiency is equally high for RMG SI and the two HCCI cases in most of the region where HCCI can be operated. RMG HCCI has the highest efficiency only at very low loads and is comparable to SI for IMEP_n below 2.8 bar. The deep dip of the hydrogen SI curve might be due to an experimental setup with too few throttling levels.

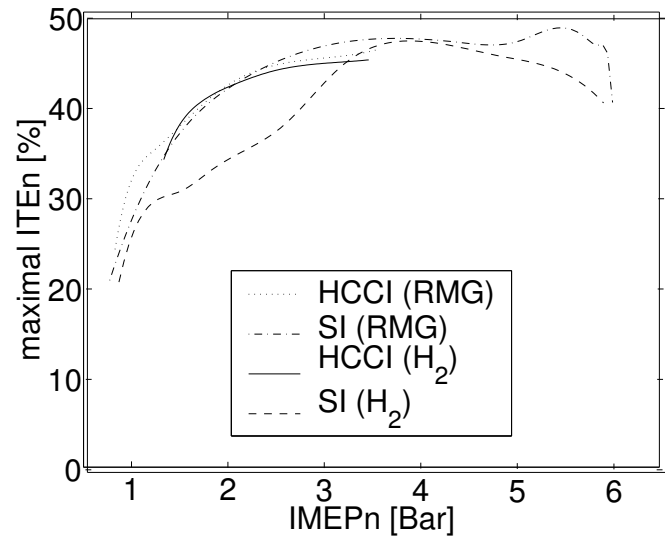


Figure 29. Maximal net ITE – IMEP plot for hydrogen and RMG fuelled HCCI and SI calculated from methanol.

The experiment with the highest ITE_n at IMEP_n ≈ 3 bar has been chosen for each of the four cases studied. They are displayed in Table 1, Table 2, Figure 30 and Figure 31. The exhaust gas energy is highest for the first case (SI H₂) and lowest for the last case (HCCI RMG). The opposite trend is seen for heat loss from the cylinder. This is believed to be attributed to the engine speed, being highest in the first case and lowest in the last. From the pumping losses it can be seen that cases one and two have higher losses than cases three and four. This is because case two is throttled and case one is choked by the engine valve design (at the highest speed). Energy bound to emissions is highest in case two, the richest case. SI cases have a much higher ability to use the reformer as is seen from the RefMEP values.

Case	1	2	3	4
Mode	SI	SI	HCCI	HCCI
Fuel	H ₂	RMG	H ₂	RMG
Speed	1600 RPM	1200 RPM	1200 RPM	800 RPM
IMEP _n	3.00	3.00	3.00	3.00
PMEP	0.48	0.57	0.29	0.15
IMEP _g	3.48	3.57	3.28	3.15
Q _{ex} MEP	3.10	2.15	1.59	1.27
QMEP _n	6.58	5.72	4.88	4.43
Q _{ht} MEP	1.17	2.06	2.27	2.83
QMEP _g	7.75	7.78	7.14	7.25
Q _{em} MEP	0.17	0.49	0.12	0.22
FuelMEP	7.92	8.27	7.26	7.47
RefMEP	0.90	1.39	0.48	0.59

Table 1. MEP data for IMEP_n = 3 cases.

Case	1	2	3	4
μ_c	98%	94%	98%	97%
μ_t	45%	46%	46%	43%
μ_{ge}	86%	84%	91%	95%
μ_{tot}	38%	36%	41%	40%
$\mu_{tot_reformer}$	43%	44%	44%	44%

Table 2. Efficiency data for IMEPn = 3 cases.

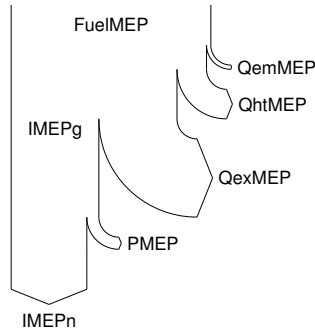


Figure 30. Sankey diagram of hydrogen fuelled SI. IMEPn of 3 bar.

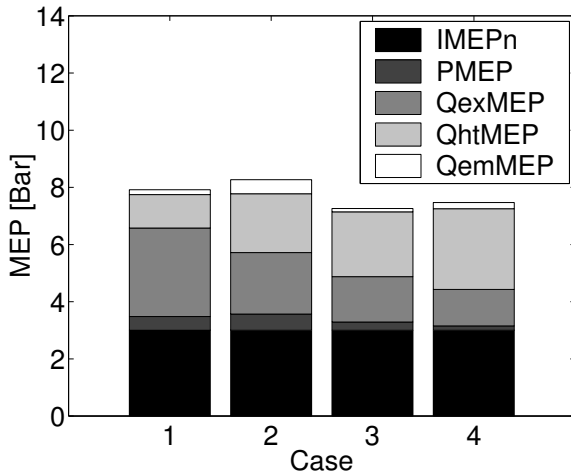


Figure 31. MEP of case 1-4. IMEPn of 3 bar.

CONCLUSIONS

The conclusions of this work are:

1. The HCCI combustion of RMG is faster than HCCI combustion of most other fuels except for hydrogen.
2. The rapid burn angle of RMG HCCI decreases with increasing intake manifold temperature.
3. Comparing hydrogen and RMG, the required intake temperature to achieve the same rapid burn angle is higher for RMG.
4. Possible start of combustion is not affected in any major way by a change of engine speed or compression ratio in RMG HCCI combustion. Lambda also has a weak effect on the earliest possible start of combustion timing but the latest possible timing is delayed when the mixture becomes richer. The latest possible timing of RMG HCCI is observed to be later than for hydrogen HCCI.
5. RMG HCCI is possible at very lean mixtures. Combustion has been studied at lambda six.
6. The estimated SOC temperature decreases slightly with decreasing lambda.
7. The inlet temperature range within which control of RMG HCCI combustion is possible is narrower than the one observed for most other fuels except for hydrogen. It becomes narrower when the mixture becomes richer.
8. Carbon monoxide emissions were high when the engine was run in HCCI mode.
9. Hydrocarbon emissions were present but at extremely low levels in both SI and HCCI mode. They probably originate from vaporisation and partial oxidation of lubricants.
10. RMG HCCI has better efficiency than RMG SI in the range where it is possible to operate in HCCI mode. If the efficiency of the reformer is taken into account the RMG fuelled SI case has a better efficiency in all but the lowest loads.
11. SI operation enables nearly twice the load of HCCI operation when RMG is used as fuel.
12. Both SI and HCCI operation are limited in load to about half the load achievable with other fuels when using premixed air and fuel.
13. The exhaust gas energy of hydrogen and RMG combustion in an SI engine is in most cases high enough to supply dissociative or steam reformation of methanol with the energy needed.
14. HCCI combustion of hydrogen and RMG does not supply the required amount of exhaust energy to use the full potential of reformation.
15. HCCI combustion at maximum ITE has NOx emissions low enough not to require a three way catalytic converter.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

μ_c	Combustion efficiency
μ_{ge}	Gas exchange efficiency
μ_t	Thermal efficiency
μ_{tot}	Total efficiency
$\mu_{tot_reformer}$	Total efficiency with reformer
BTDC	Before Top Dead Centre
CAD	Crank Angle Degree. 0 CAD at TDC combustion.
EGR	Exhaust Gas Recirculation.
EVO	Exhaust Valve Opening
FuelMEP	Fuel Mean Effective Pressure, LHV of the fuel normalized for cylinder volume.
IMEPg	Gross Indicated Mean Effective Pressure, mean effective pressure over the compression and expansion strokes.
IMEPn	Net Indicated Mean Effective Pressure, mean effective pressure over the entire cycle.
ITEg	Gross Indicated Thermal Efficiency, indicated efficiency over the compression and expansion strokes (IMEPg /FuelMEP).
ITEn	Net Indicated Thermal Efficiency, indicated efficiency over the entire cycle (IMEPn /FuelMEP).
Lambda	(mass air/mass fuel)/ (mass air/mass fuel) _{stoichiometric}
PMEP	Pumping Mean Effective Pressure. IMEPg - IMEPn
GemMEP	Emission Mean Effective Pressure, LHV of the emissions normalized for cylinder volume.
QexMEP	Exhaust Mean Effective Pressure, thermal energy in the exhaust gas normalized for cylinder volume.

QhtMEP	Heat transfer Mean Effective Pressure, thermal energy lost through top, piston and cylinder walls normalized for cylinder volume.
RefMEP	Reformer Mean Effective Pressure, exhaust gas energy recovered in the reformer normalized for cylinder volume.
RBA	Rapid Burn Angle, the crank angle between 10 and 90% heat release
RMG	Reformed Methanol Gas
RPM	Revolutions Per Minute
SOC	Start Of Combustion, crank angle of 1% burned – 1°.
TDC	Top Dead Center.

APPENDIX 1 ONE CYLINDER TD100 ENGINE.

Bore	120.65
Compression ratio	9-20
Connecting rod	260 mm
Cylinders	1
Displaced volume	1.6 l
Exhaust valve diameter	46 mm
Exhaust valve lift	13.4 mm
Exhaust valve closing angle (1mm lift)	350 CAD
Exhaust valve opening angle (1mm lift)	141 CAD
Inlet valve diameter	50 mm
Inlet valve lift	11.9 mm
Inlet valve closing angle (1mm lift)	-167
Inlet valve opening angle (1mm lift)	-355
Oil temperature	90° C
Stroke	140 mm