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Hydrogen Addition For Improved Lean Burn Capability of Slow and Fast Burning Natural Gas Combustion Chambers

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ABSTRACT

One way to extend the lean burn limit of a natural gas engine is by addition of hydrogen to the primary fuel. This paper presents measurements made on a one cylinder 1.6 liter natural gas engine. Two combustion chambers, one slow and one fast burning, were tested with various amounts of hydrogen (0, 5, 10 and 15 %-vol) added to natural gas. Three operating points were investigated for each combustion chamber and each hydrogen content level; idle, part load (5 bar IMEP) and 13 bar IMEP (simulated turbocharging). Air/fuel ratio was varied between stoichiometric and the lean limit. For each operating point, a range of ignition timings were tested to find maximum brake torque (MBT) and/or knock. Heat-release rate calculations were made in order to assess the influence of hydrogen addition on burn rate. Addition of hydrogen showed an increase in burn rate for both combustion chambers, resulting in more stable combustion close to the lean limit. This effect was most pronounced for lean operation with the slow combustion chamber.

INTRODUCTION

One of the main sources for air pollution is road transport. There are several different ways of dealing with pollution from road transports. One way is to change transport patterns and to transport more goods and humans by rail instead of by road. Another way is to improve the engine technology and exhaust gas treatment technologies, and a third way is to change to fuels that gives improved combustion characteristics in the engines. One of the main alternatives in this respect is natural gas.

The use of natural has been increasing in recent years and is expected to increase from 300 to more than 500 MTOE (Million Tonnes of Oil Equivalent) in the year 2020 [11]. The expected lifetime of the proven natural gas reserves is more than 60 years and is increasing.

The use of natural gas as a vehicle fuel is also encouraged by an initiative from the European Commission setting up a goal to replace 20% of all fossil transport fuels by alternative fuels by the year 2020. It is expected that natural gas will account for approximately half of this replacement.

Natural gas vehicle emissions are generally regarded as very low and especially emissions of particles that are normally very low compared to emissions from similar diesel vehicles. NO_X and HC emissions from lean burn gas engines have been a concern though, and the focus for many research activities. NO_X emissions are very dependent on the possibility to control the air/fuel ratio, and in the early generation NGV-engines there was no feedback from the oxygen content of the exhaust gases, resulting in increased emissions even at minor fluctuations in fuel quality.

Hydrocarbon emissions from a natural gas engine consist almost only of methane and should thus only be regarded as a greenhouse gas. Methane is a very strong greenhouse gas and the emissions must thus be minimized for lean burn natural gas engines if they are to compete with diesel engines concerning total emissions of greenhouse gases.

Lean-burn operation of natural gas engines provides a means for combining high efficiency with relatively low NO_X-emissions at full load. At reduced load, the air/fuel ratio is normally reduced with increased NO_X emissions and reduced efficiency as a result. This is a major drawback for the lean burn engines, especially in urban applications such as city buses and distribution trucks for urban use. A way of improving these part load properties is to add hydrogen to the natural gas in order to improve the combustion characteristics of the fuel.

Numerous tests have been performed with Hythane ® (a mixture of hydrogen and natural gas with hydrogen content of approximately 20%). Cattelan and Wallace [6] have shown a high efficiency increase at loads below 50%. The emissions of HC and NO_X also decreased dramatically at loads below 50% compared to pure natural gas. Large gains in cold start emissions have also been observed if pure hydrogen is used during the start of the engine. Reduction of cold start emissions between 20 and 30% were observed.

The purpose of this paper is to investigate the benefits of hydrogen addition to a natural gas fueled spark ignition engine. The engine is of truck size meaning that the combustion chamber is located in the piston bowl and a swirling inlet port is used. This configuration is the normal practice for engines in the size range of 4-15 liters swept volume simply because they are based on diesel engine designs. Even if the basic layout is common practice it might not be the best solution for an SI engine. Most smaller SI engines use a four-valve pentroof combustion chamber with a better trade-off between turbulence generation and heat losses.

BASIC EFFECT OF HYDROGEN

It is well known that the laminar flame speed of hydrogen is much higher than that of methane or other hydrocarbons [1]. Adding hydrogen to natural gas thus is a way to increase the laminar flame speed of the charge. Increasing the laminar flame speed is of interest if the engine is operating under conditions resulting in a burn rate that is otherwise too slow. The most common such operating condition is very lean operation. Close to the lean limit, the laminar flame speed is very low and thus also the turbulent flame speed. Small disturbances in fluid flow and/or mixture composition will result in large variations in the combustion process and thus also very poor engine stability. The effect of hydrogen addition close to the lean limit is to increase the laminar flame speed and thus make the initial flame propagation faster and more stable. However, a similar effect can be obtained if the fluid flow in the cylinder is changed. More turbulence results in faster flame propagation for constant laminar flame speed.

A major question then arises: Is it more beneficial to speed up the combustion rate close to the lean limit by adjusting laminar flame speed with hydrogen addition or is it better to adjust the turbulence level? There is also the question if both methods can be used simultaneously to further extend the lean-operation capability. Also of interest are the emissions of unburned hydrocarbons, HC, and nitric oxides, NO_X, with these two ways of extending the lean limit. It is expected that a faster burn would result in higher peak pressure and hence temperature. This higher temperature would then show up in higher emissions of NO_X. On the other hand, a faster burn lowers the probability of slow or partial burn, resulting in lower emissions of HC. However, slow or partial burn are not the only sources of HC with a lean mixture. With a very lean mixture the temperature in the cylinder at the time for post oxidation of fuel hidden in the top land crevice is very low, and the quenching distance also increases.

There are reports in the literature with a special type of hydrogen addition enabling stable operation at λ=5 [2]. Interesting to note is that in the experiments, the hydrocarbon emissions continued to rise as a function of λ from the minimum at λ=1.3 past the normal lean limit at 1.8, all the way up to 5.0. Ultralean mixtures resulted in very low NO_X, but to the price of enormous amounts of HC. The results form the experiments are not directly transferable to a premixed charge of natural gas/hydrogen, since a pure hydrogen feed was applied to a prechamber where the spark plug was located. The spark set fire to the hydrogen in the prechamber, and partially burned products were injected into the main combustion chamber with great speed. Thus the hydrogen can, in that application, be more considered as a very powerful spark plug, and the combustion process in the main combustion chamber is that of natural gas only. Thus all near-wall combustion and post oxidation is left unaltered. This is not the case with premixed natural gas/hydrogen.

[7,4] investigate hydrogen addition to CNG in a large passenger car engine as well as a single-cylinder research engine. It is shown that the relationship between NO_X and unburned hydrocarbons parameterized by equivalence ratio can be affected by addition of various amounts of hydrogen. It is reported that a mixture with 30% hydrogen allows NO_X levels to be kept below 0.05 g/kWh for loads up to 5 bar BMEP and engine speeds above 1700 RPM, with low hydrocarbon emissions and good fuel economy.

In [3], the influence of hydrogen addition to hydrocarbon gas blends on laminar flame speed is investigated. Some engine testing with hydrogen addition to gasoline is also performed.

[5] investigates operation of an automobile engine with partially air-reformed natural gas. Improved emissions, part-load efficiency and lean-operation capability is observed.

[8] presents heat release calculations and emissions from engine operation with steam-reformed natural gas. Steam reformation of natural gas yields methane, hydrogen and carbon dioxide. The burn-rate analysis is conducted on data from experiments with a flat combustion chamber which results in long burn duration. Using reformed natural gas results in improved lean-operation capability compared to pure natural gas.

EXPERIMENTAL SETUP

A Volvo TD100 modified for single cylinder natural gas SI operation is used for the experiments. The fuel system supplies natural gas, see Table 1 for composition, through a Pulse Width Modulated (PWM) valve, and hydrogen through a Mass Flow Controller (MFC). Both fuels are mixed with the air just upstream of the inlet port.

Table 1. The composition of the natural gas used for this study.

Natural Gas Constituents	% Volume
CH_4	88.06
C_2H_6	6.49
C_3H_8	2.81
C_4H_{10}	1.00
C_5H_{12}	0.20
C_6H_{14}	0.06
CO_2	1.05
N_2	0.33

Air can be supplied either at atmospheric pressure from the test cell, or compressed from an external compressor. In both cases the inlet system, outside the port, is somewhat unrealistic for a real engine, consisting of a long pipe. When the engine is boosted the exhausts are throttled to achieve a backpressure simulating a realistic turbo charger. Backpressure is adjusted to correspond to a turbo efficiency of around 55%. In this way the Pumping Mean Effective Pressure (PMEP) is representative and net indicated values can be used for comparison with other engines. Keep in mind though, that the breathing characteristics of this engine could be somewhat offset due to the inlet geometry.

The cylinder is equipped with a cylinder pressure sensor to allow monitoring of the combustion. The pressure trace is used by the combustion control system but also for calculation of indicated parameters.

The cylinder head of the engine is in its original configuration and the camshaft has the same properties as in the natural gas fueled SI engines of the same series. The properties of the engine are summarized in Table 2.

An emission measurement system sampling exhausts is used to measure the exhaust concentrations of O_2, CO, CO_2, HC, NO_X and NO. The heated Flame Ionization Detector (FID), measuring HC, is calibrated using CH_4 and concentration of HC is given as CH_4 equivalent. CO and CO_2 are measured with Non Dispersive Infrared detectors (NDIR). NO_X emissions are measured with a chemiluminescent instrument and O_2 with a Paramagnetic Analyzer (PMA).

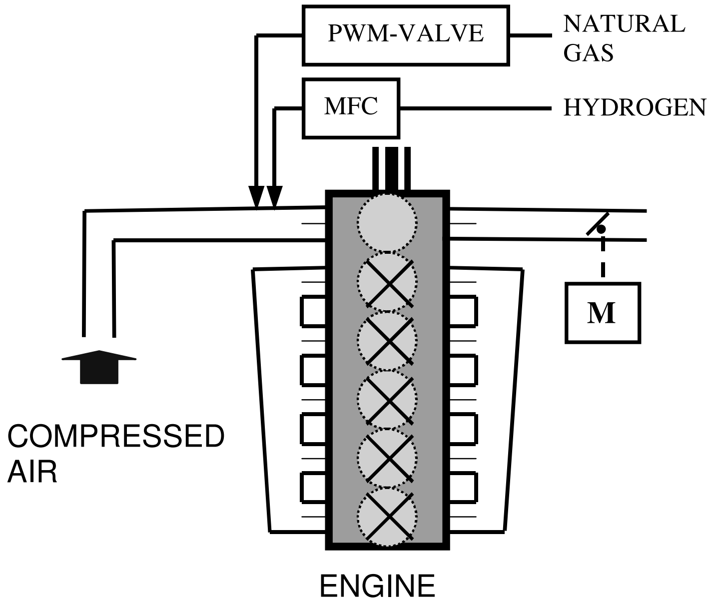

Figure 1: Engine system.

Table 2. Geometric specifications of the engine. Valve timings refer to 1mm lift plus lash.

Displacement volume	1 600 cm^3
Compression Ratio	12.0
Bore	120.65 mm
Stroke	140 mm
Connecting Rod	260 mm
Exhaust Valve Open	39° BBDC
Exhaust Valve Close	10° BTDC
Intake Valve Open	5° ATDC
Intake Valve Close	13° ABDC

The combustion control system is a modified version of the system used previously in [10]. The cylinder pressure trace and the inlet conditions are sampled and some key parameters characterizing the operating condition are calculated in real time. Combustion timing, characterized by the crank angle of 50% burnt, CA50, is calculated through an analysis of the net heat release. IMEP and COV(IMEP) are also computed online. In this study, all experiments are run at 1200 RPM except idle which is at 700 RPM.

EXPERIMENTS

There are two ways to alter the air/fuel ratio for an engine. The engine can be run with a constant fuel flow giving almost fixed load and air flow varied to adjust λ. The benefit of this set-up is that engine efficiency can be studied with some accuracy as the mechanical efficiency is roughly constant. The drawback of this procedure is that the increased amounts of air needed to increase λ means that the conditions at the time of ignition is much different. A leaner mixture results in a higher in-cylinder pressure and thus higher demands on the ignition system. Effects of λ can thus be a effect of ignition system limitations and not actual combustion related effects. Changed pressure also means that the laminar flame speed will change. Thus laminar flame speed will be affected by pressure and λ at the same time. To remove this uncertainty the other strategy can be applied. By keeping the air flow constant and changing the fuel flow to vary λ, the in-cylinder pressure is maintained constant and thus only λ will affect laminar flame speed. On the other hand the engine power output will be proportional to $\phi=1/\lambda$ and thus the engine load will change. This will change amount of heat released per time unit and thus wall temperature. The changed heat transfer will result in a changed gas temperature as well.

Since both constant air flow and constant fuel flow have their merits and drawbacks it was decided to use both methods. One investigation was conducted at the constant engine load of 13 bar IMEP and another at Wide Open Throttle, WOT, changing the IMEP from 8 bar at stoichiometric to 5 bar close to the lean limit. A third sweep was also conducted close to idle to investigate the effects of turbulence and laminar flame speed with a higher amount of residual gas in the cylinder.

The ignition timing was altered from 5 to 45 CAD BTDC in steps of 5 CAD. Ignition timing dependence is not discussed in this paper. Instead all results in this paper are presented at MBT ignition timing. In order to find MBT ignition the timing which resulted in highest IMEP was selected.

COMBUSTION CHAMBERS

The turbulence was altered by using two piston bowl geometries, Turbine and Quartette. Figure 2 shows the piston bowl design for the two combustion chambers. The turbulence levels in these bowls have been measured previously by the use of Laser Doppler Velocimetry, LDV [9], see Figure 3 and Figure 4. The Turbine combustion chamber has a turbulence peak of less than 2 m/s at 15° BTDC whereas the Quartette has a peak of 3 m/s just prior to TDC. Comparing the heat release rates for the two combustion chambers, it can be seen that the Quartette geometry favors a high burn rate.

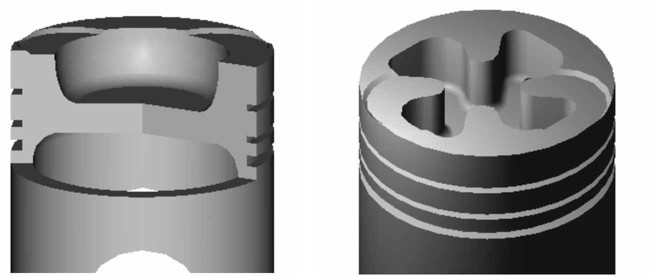

Figure 2: Turbine and Quartette geometries

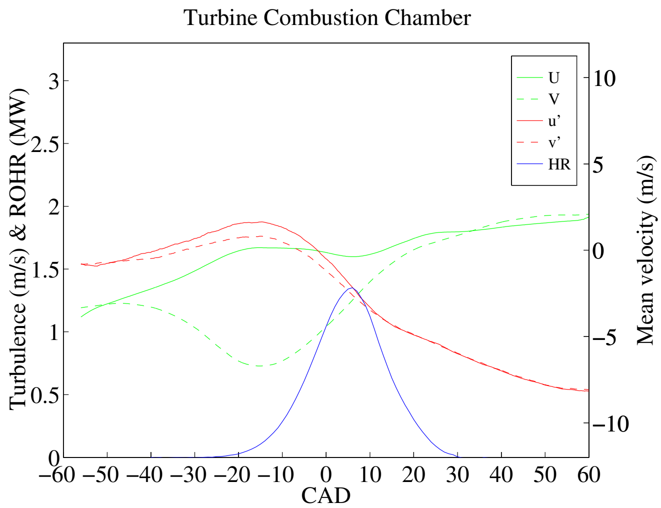

Figure 3: Mean velocity and turbulence data for turbine combustion chamber. Heat release generated at 1200 RPM, WOT and λ=1.5

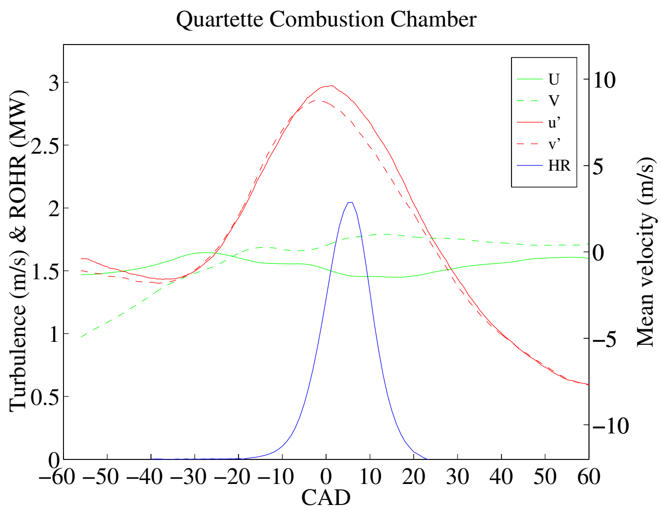

Figure 4: Mean velocity and turbulence data for Quartette combustion chamber. Heat release generated with 1200 RPM, WOT and λ=1.5

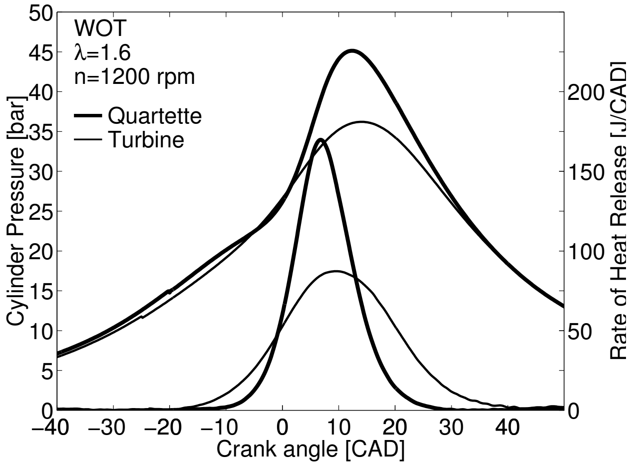

Figure 5: Pressure traces and heat release rates for Turbine and Quartette pistons (MBT ignition timing).

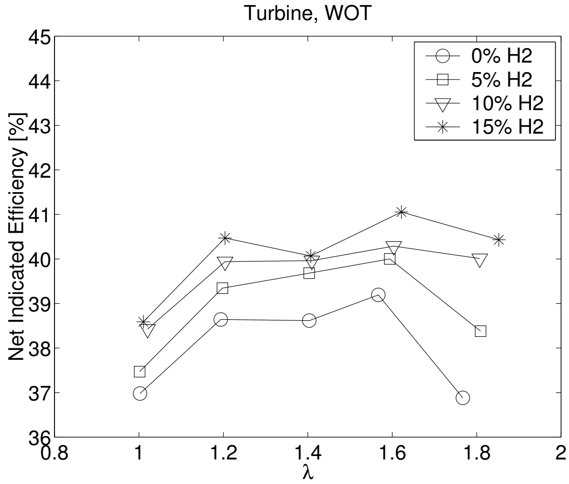

Figure 6: Net indicated efficiency at WOT with various amounts of hydrogen addition (Turbine).

The difference in maximum pressure and burn rate between the two combustion chambers can be seen in Figure 5. The maximum pressure of the Quartette combustion chamber is 10 bar higher than for the Turbine chamber, and the peak heat-release rate is 100 J/CAD higher.

DISCUSSION OF RESULTS

The results of the investigation are presented with the WOT case as the main theme. Similar investigations at idle (1.5 bar IMEP) and simulated turbocharged operation (13 bar IMEP) are presented in the appendix.

EFFICIENCY

Both combustion chambers show increases in indicated efficiency with increasing hydrogen content in the fuel, see Figure 6 and Figure 7, although the increase in efficiency is more pronounced for the Turbine combustion chamber than for the Quartette.

The reason for the increase in efficiency with hydrogen addition is the increase in burn rate and combustion efficiency, particularly for lean operation. The burn duration for the Turbine geometry drops by as much as 10 Crank Angle Degrees (CAD) for the leanest operating points, Figure 8. For the Quartette chamber, the burn duration is less affected by hydrogen addition, and the effect is less than 5 CAD for all operating points, Figure 9. A comparison of the absolute values for the burn duration shows that the burn duration for the Turbine chamber with 15% hydrogen is still somewhat longer than the burn duration for the Quartette chamber operated with pure natural gas. This is, of course, the reason why the efficiency increase with hydrogen addition is more modest for the Quartette chamber.

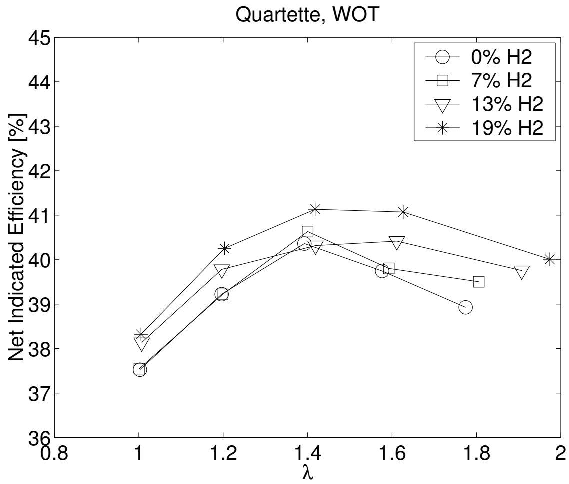

Figure 7: Net indicated efficiency at WOT with various amounts of hydrogen addition (Quartette).

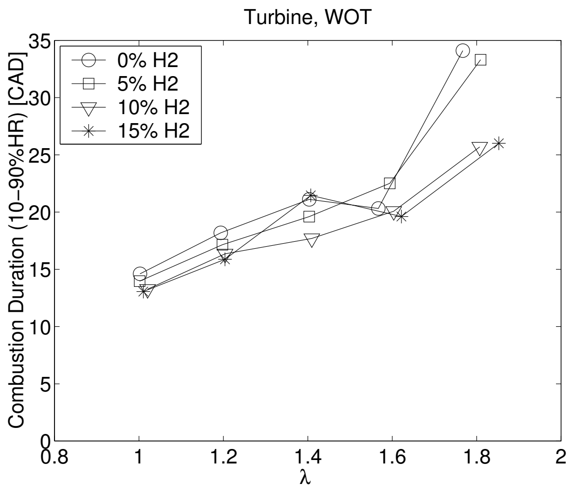

Figure 8: Duration of the main combustion phase at WOT (Turbine).

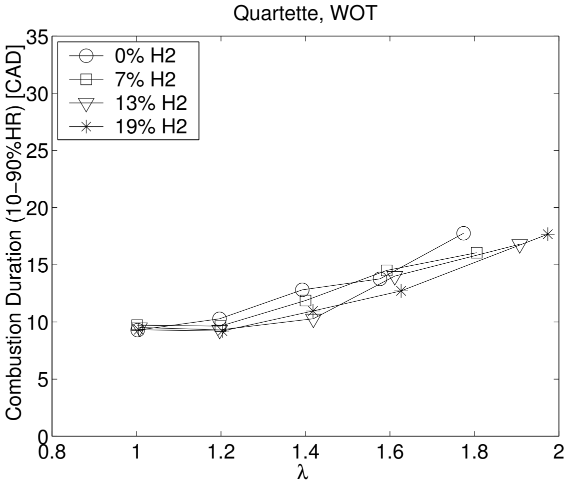

Figure 9: Duration of the main combustion phase at WOT (Quartette).

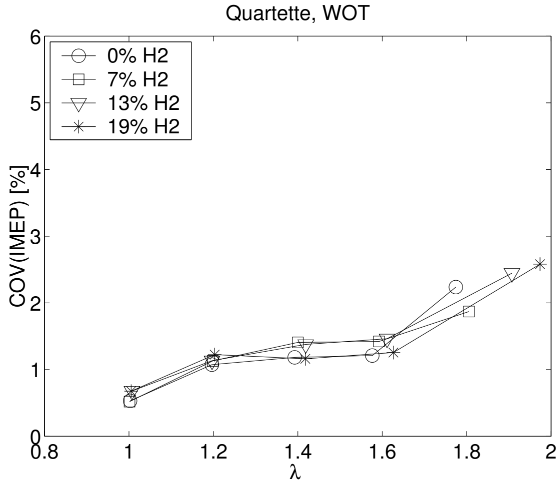

Figure 11: COV(IMEP) at WOT (Quartette).

COMBUSTION STABILITY

Combustion stability is also affected by hydrogen addition. Figure 10 shows how COV(IMEP) for the Turbine combustion chamber is drastically reduced at the leanest operating point when hydrogen is added. Around λ=1.8, COV(IMEP) is reduced from 6% to 2% when the hydrogen content is increased from 0% to 15%. Again, it is seen that the Quartette design is less affected by hydrogen addition, see Figure 11. The value of COV(IMEP) at λ=1.8 is however approximately 2% with or without hydrogen addition, which is the same as for the Turbine chamber with 15% hydrogen.

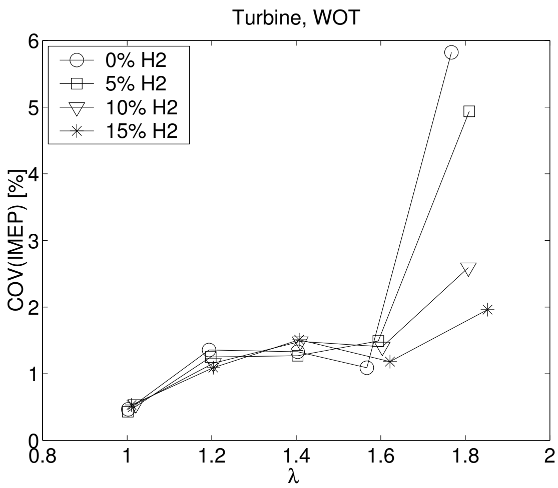

Figure 10: COV(IMEP) at WOT (Turbine).

Figure 12 shows the lean-limit as a function of hydrogen content, and clearly indicates the effectiveness of hydrogen for extending lean-operation capability. The lean-limit is defined as the air excess ratio which results in 5% COV(IMEP).

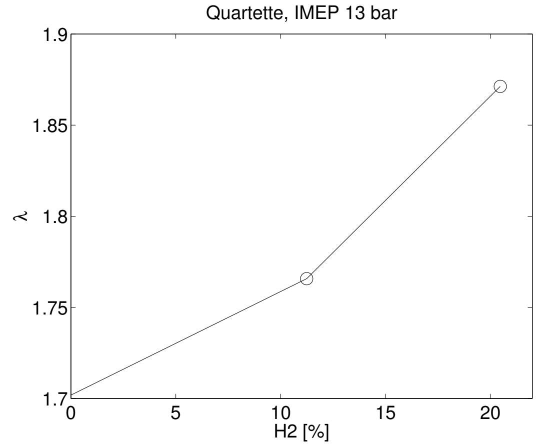

Figure 12: Extension of lean-limit capability (13 bar IMEP).

EMISSIONS

Figure 13 and Figure 14 show HC and NO_X emissions versus air excess ratio for the Turbine and Quartette combustion chambers respectively. It is evident that there is a trade-off between HC and NO_X when selecting the Air/Fuel Ratio (AFR). Increased AFR results in lower NO_X but higher HC. Hydrogen addition affects this trade-off between HC and NO_X emissions, since it allows leaner operation with maintained combustion rate, and thus without the drastic increase in HC which would result without hydrogen addition. For the Turbine geometry it is clearly seen, in Figure 15, how both HC and NO_X can be reduced simultaneously. Since the combustion rate of the Quartette design is less affected by hydrogen addition, the effect on this trade-off too, is modest, see Figure 16.

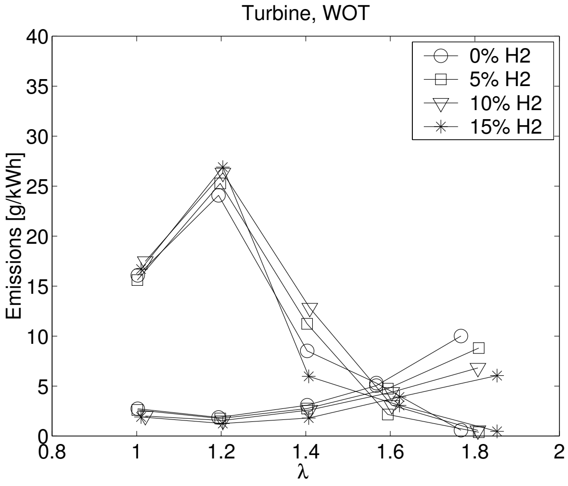

Figure 13: Emissions of NO_X and HC at WOT (Turbine).

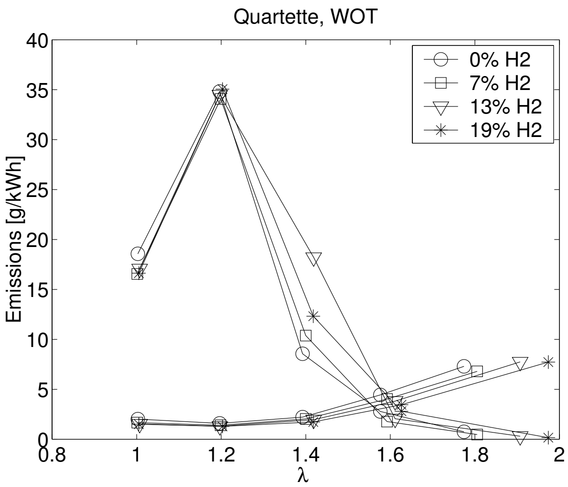

Figure 14: Emissions of NO_X and HC at WOT (Quartette).

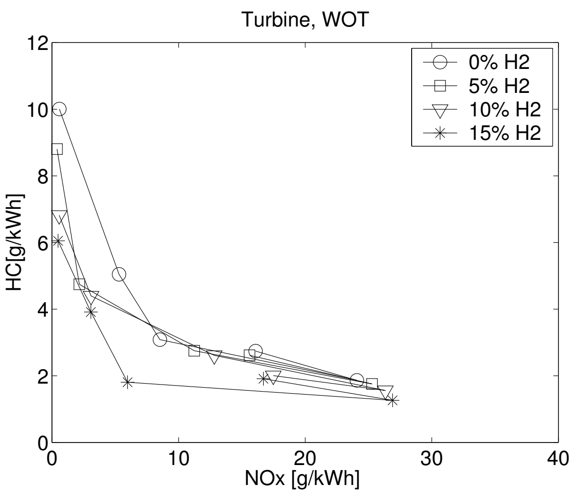

Figure 15: Trade-off between HC and NO_X at WOT (Turbine).

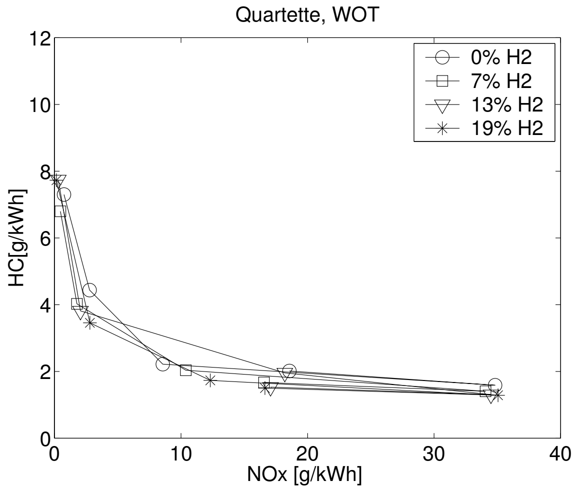

Figure 16: Trade-off between HC and NO_X at WOT (Quartette).

CONCLUSIONS

Addition of hydrogen to natural gas increases the burn rate, and extends the lean-limit. Hydrogen addition lowers HC emissions and increases NO_X emissions for constant air excess ratio and ignition timing. The increased burn rate allows retarded ignition timing which decreases heat losses and results in higher efficiency. The retardation of ignition timing also results in lower maximum temperature and thus lower NO_X emissions. Addition of hydrogen thus allows a trade-off with both lower HC and NO_X emissions compared to operation with pure natural gas.

The effect of hydrogen addition at WOT is most pronounced for the slow combustion chamber (Turbine) close to the lean-limit. This is to be expected, since the faster combustion chamber (Quartette) has very fast combustion even without hydrogen addition.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

ABDC: After Bottom Dead Center

AFR: Air/Fuel Ratio

ATDC: After Top Dead Center

BBDC: Before Bottom Dead Center

BTDC: Before Top Dead Center

CA50: Crank Angle of 50% heat release

CAD: Crank Angle Degrees

COV: Coefficient Of Variation (standard deviation / mean × 100)

FID: (heated) Flame Ionization Detector

IMEP: Indicated Mean Effective Pressure

MBT: Maximum Brake Torque (Ignition Timing)

MFC: Mass Flow Controller

MTOE: Million Tonnes of Oil Equivalent, 11630 GWh

NDIR: Non Dispersive Infra-Red detector

PMA: Paramagnetic Analyzer

PMEP: Pumping Mean Effective Pressure

PWM: Pulse Width Modulation

RPM: Revolutions Per Minute

WOT: Wide Open Throttle

APPENDIX

This appendix contains results from idle operation and (simulated) turbocharged operation which are included for completeness.

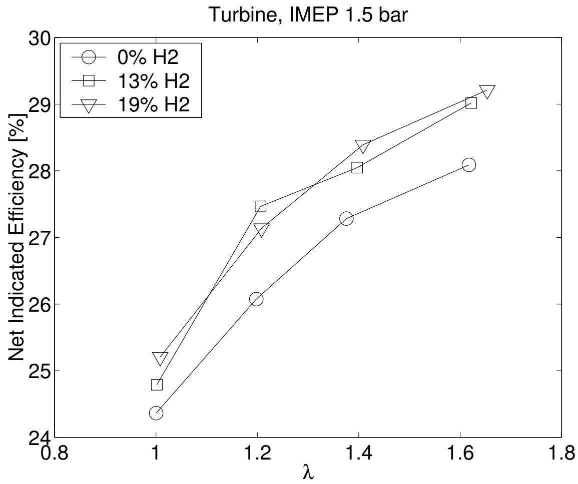

Figure 17: Net indicated efficiency at idle with various amounts of hydrogen addition (Turbine).

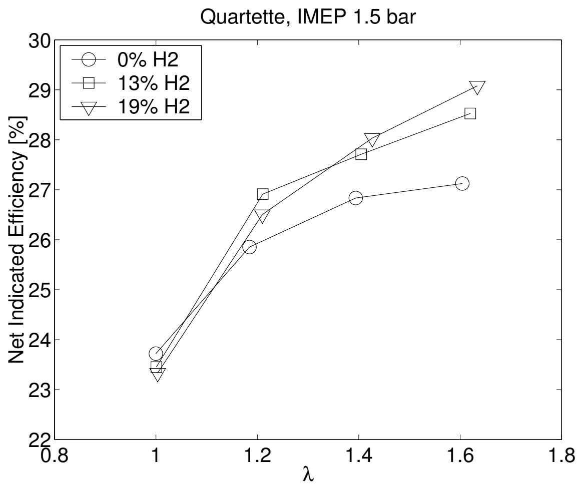

Figure 18: Net indicated efficiency at idle with various amounts of hydrogen addition (Quartette).

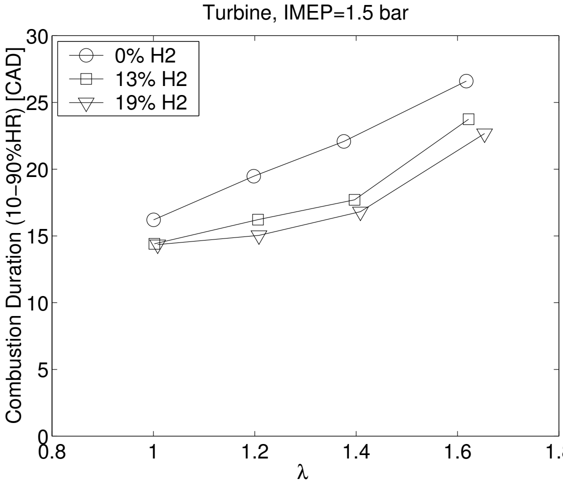

Figure 19: Duration of the main combustion phase at idle (Turbine).

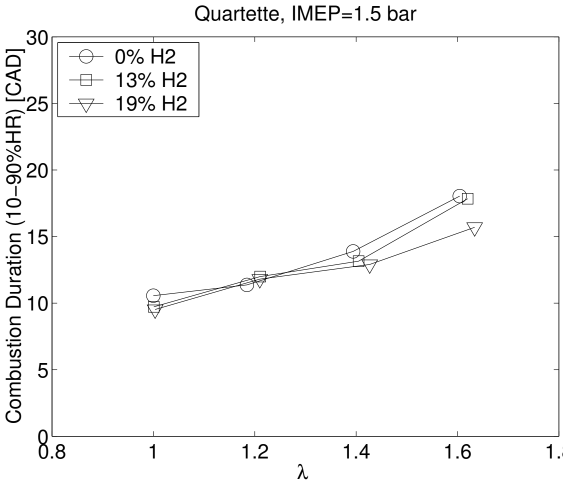

Figure 20: Duration of the main combustion phase at idle (Quartette).

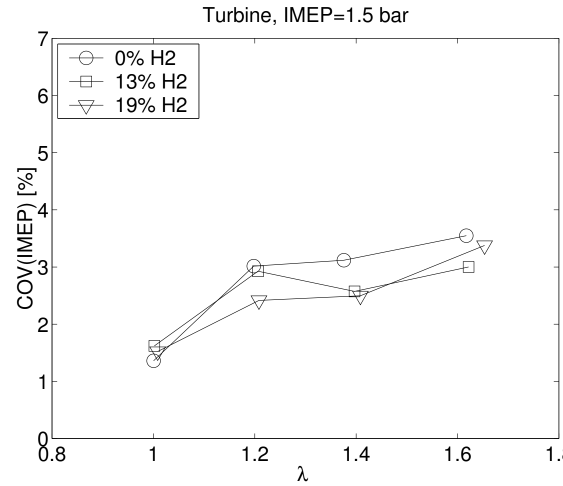

Figure 21: COV(IMEP) at idle (Turbine).

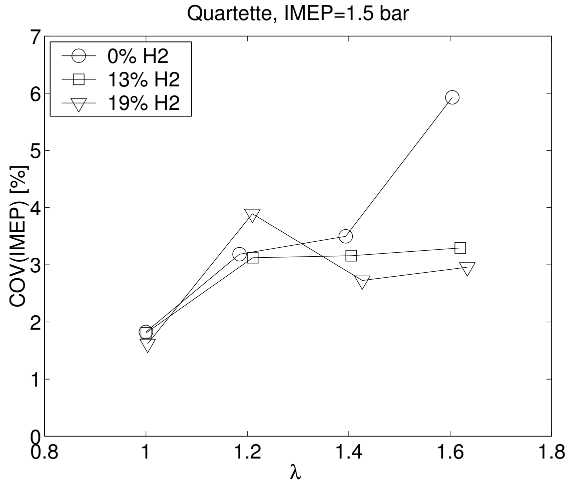

Figure 22: COV(IMEP) at idle (Quartette).

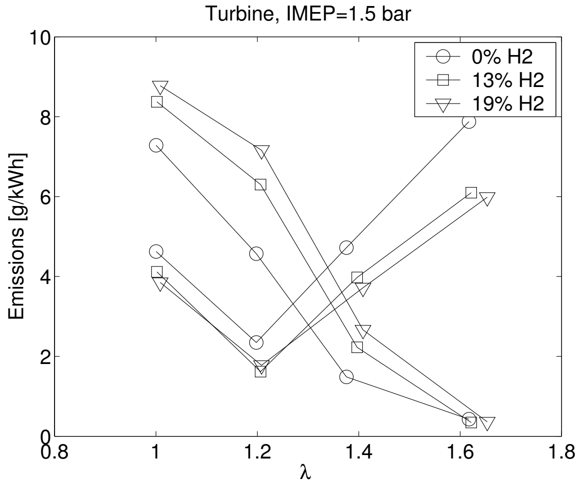

Figure 23: Emissions of NO_X and HC (Turbine).

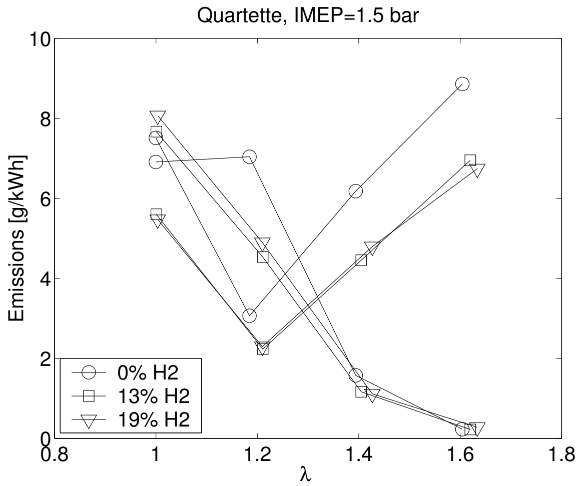

Figure 24: Emissions of NO_X and HC (Quartette).

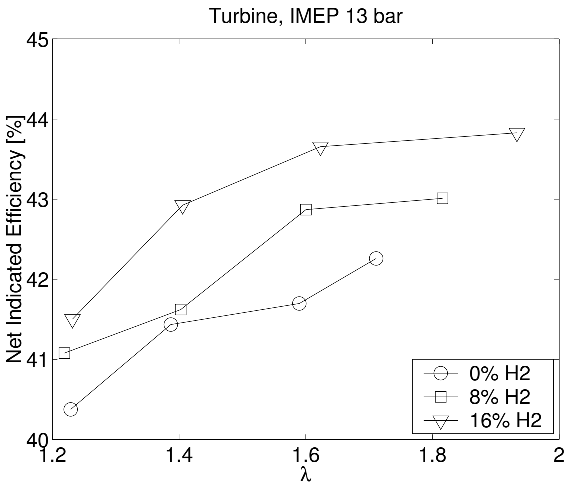

Figure 25: Net indicated efficiency at 13 bar IMEP with various amounts of hydrogen addition (Turbine).

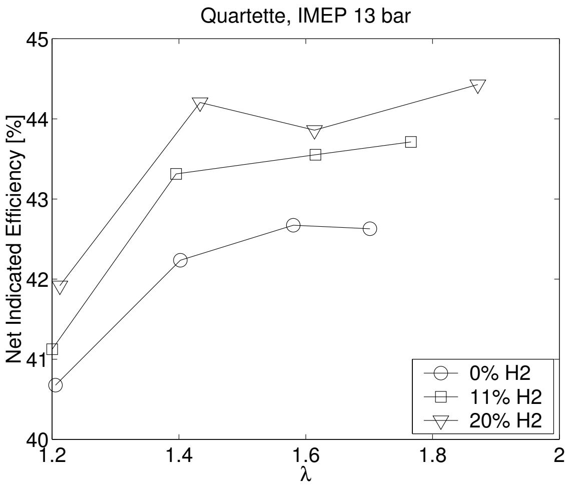

Figure 26: Net indicated efficiency at 13 bar IMEP with various amounts of hydrogen addition (Quartette).

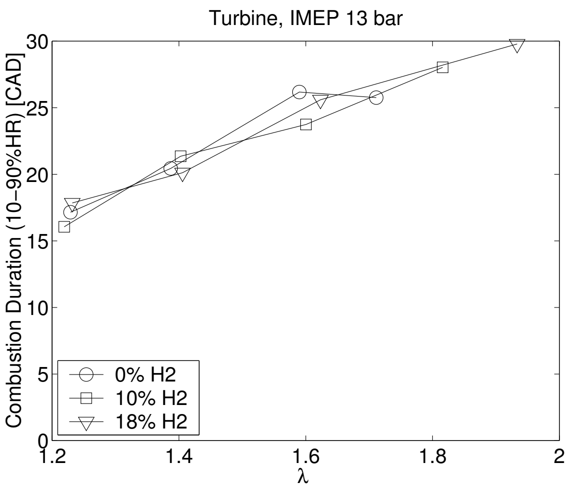

Figure 27: Duration of the main combustion phase at 13 bar IMEP (Turbine).

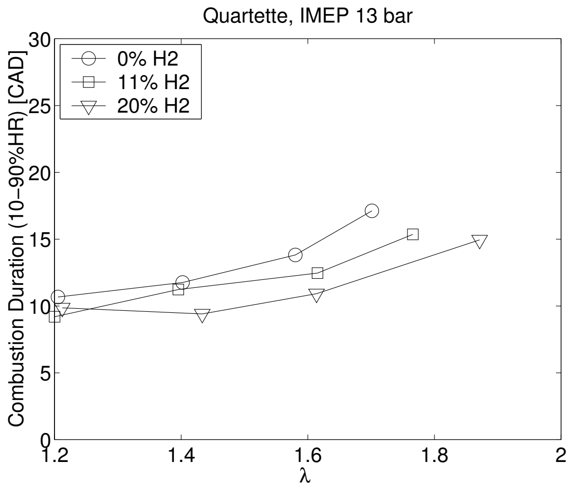

Figure 28: Duration of the main combustion phase at 13 bar IMEP (Quartette).

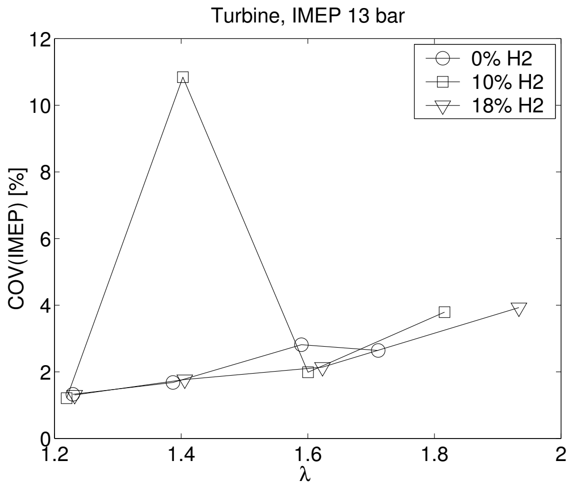

Figure 29: COV(IMEP) at 13 bar IMEP (Turbine).

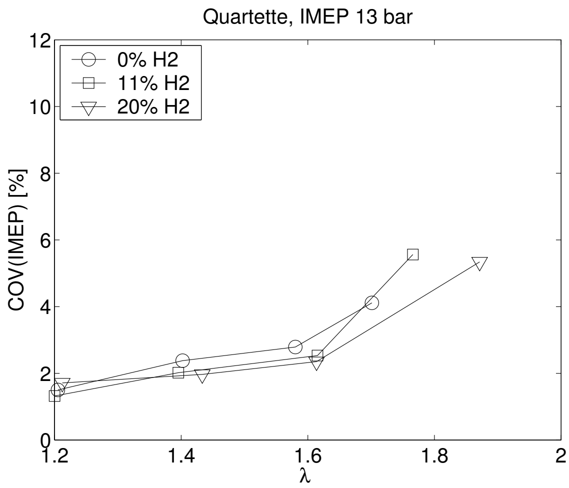

Figure 30: COV(IMEP) at 13 bar IMEP (Quartette).

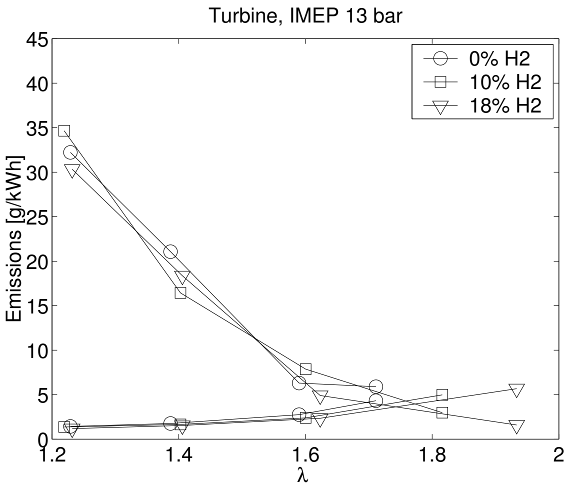

Figure 31: Emissions of NO_X and HC (Turbine).

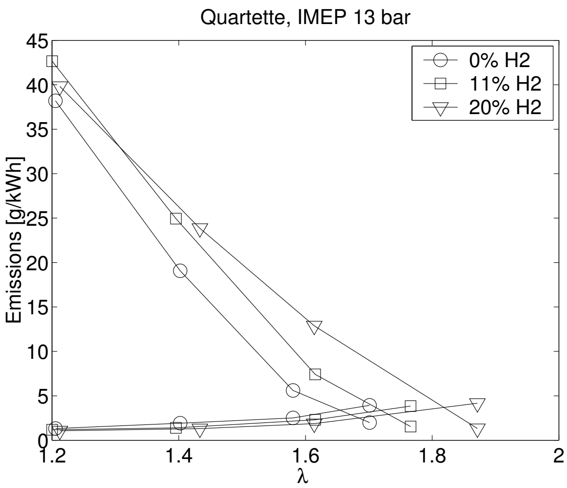

Figure 32: Emissions of NO_X and HC (Quartette).