Rapport SGC 134

Hydrogen Addition for Improved Lean Burn Capability on Natural Gas Engine

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SUMMARY IN SWEDISH

S k Lean Burn-motorer är ett intressant alternativ till dieselmotorer speciellt i stadstrafik, där avgasemissioner är ett problem. En av vinsterna är den kraftiga reduktionen av partikelutsläpp. Lean Burn-motorer har god verkningsgrad och relativt låga NO_x emissioner vid fullast. Vid dellast måste man oftast minska λ-värdet pga att motorn går alltför ojämnt, alltså öka andelen bränsle i luft/bränsleblandningen. Ett sätt att kunna köra magert även på dellast kan vara att tillsätta vätgas till huvudbränslet.

Denna rapport behandlar testkörningar gjorda med en 1.6 liter encylindrig naturgasmotor. Mätningarna är här koncentrerade till tre körfall, tomgång, medellast (fullt öppen trottel) och höglast (överladdning). I dessa körfall skedde datainsamling vid driftspunkter från stökiometriskt luft/bränsleförhållande till magergränsen. Dessutom genomfördes tändsvep vid alla driftspunkter för att hitta optimal tändtidpunkt med avseende på maximal effekt. Två olika typer av förbränningsrum har använts, ett med låg turbulens som ger långsam förbränning och ett som ger hög turbulens och snabb förbränning.

Tillsats av vätgas visade på ökad förbränningshastighet för båda förbränningsrumstyperna, som ger jämnare gång vid magerkörning. Effekten av vätgasen var större i de fall där långsamt förbränningsrum använts. Mycket låga NO_x-emissioner mättes upp vid körning nära magergränsen.

ABSTRACT

Lean burn spark ignition (SI) engines powered by natural gas is an attractive alternative to the Diesel engine, especially in urban traffic, where reduction of tailpipe emissions are of great importance. A major benefit is the large reduction in soot (PM). Lean burn spark ignition (SI) engines yield high fuel conversion efficiency and also relatively low NO_x emissions at full load. In order to improve the engine operating characteristics at lower loads, the λ-value is normally reduced to some degree, with increased NO_x emissions and reduced efficiency as a result. This is a drawback for the lean burn engines, especially in urban applications such as in city buses and distribution trucks for urban use. So, it is desirable to find ways to extend the lean limit at low loads. One way to improve these part load properties is to add hydrogen to the natural gas in order to improve the combustion characteristics of the fuel.

It is possible to extend the lean limit of a natural gas engine by addition of hydrogen to the primary fuel. This report presents measurements made on a single cylinder 1.6 liter natural gas engine. Two combustion chambers, one slow and one fast burning, were tested with various amounts of hydrogen (0 to 20 %-vol) added to natural gas. Three operating conditions were investigated for each combustion chamber and each hydrogen content level; idle, wide open throttle (WOT) and a high load condition (simulated turbo charging). For all three operating conditions, the air/fuel ratio was varied between stoichiometric and the lean limit. For each operating point, the ignition timing was swept in order to find maximum brake torque (MBT) timing. In some cases were the ignition timing limited by 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. Hydrogen addition lowers HC emissions and increases NO_x emissions but the improved burn rate gives the possibility to run leaner, which lowers NO_x emissions. In general, the slower combustion chamber design was more affected by the hydrogen addition than the faster one.

ACKNOWLEDGEMENTS

This work has been performed at the Division of Combustion Engines at Lund Institute of Technology. The work was financed by “Svenskt Gastekniskt Center AB, SGC”. Thanks to Owe Jönsson at SGC for initiating this project.

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TABLE OF CONTENTS

1 INTRODUCTION

The aim of this work was to experimentally investigate the effect of addition of hydrogen to the primary fuel in a spark ignited lean burn natural gas engine. Hydrogen is well known for being a very fast burning fuel with a wide operation range in terms of air/fuel ratio. Of primary interest was to study the influence of hydrogen addition on:

- the lean limit (in terms of air/fuel ratio)
- combustion stability (cycle-to-cycle variations)
- emissions
- fuel conversion efficiency

1.1 The Spark Ignition engine

1.1.1 Operating principle

The majority of the medium and large Spark Ignition (SI) engines operate on what is known as the four-stroke cycle. The four strokes are intake, compression, expansion and exhaust, see Figure 1. The combustion process, from which heat is released to the charge, starts late in the compression stroke and continues well into the expansion stroke.
The combustion process is characterized by three different events; combustion initiation by a spark, turbulent flame propagation and flame quenching at the combustion chamber walls.

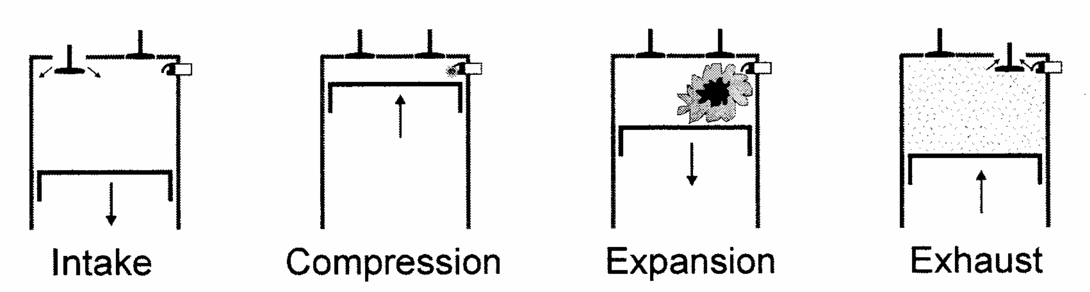


Figure 1. The four-stroke SI cycle [1].

1.1.2 Combustion and flame propagation

The speed of the combustion process is very important for the engine fuel conversion efficiency and the combustion stability. The flame propagation velocity mainly determines the speed of the combustion process. There are many parameters that influence the flame propagation velocity. The most important are:

- in-cylinder turbulence
- air/fuel ratio
- Exhaust Gas Recirculation (EGR)
- fuel type

The effect of turbulence is to wrinkle the flame front. This wrinkling of the flame leads to an increased flame area and thereby to higher propagation velocity. In the literature there exist many models that describe the relationship between the turbulent flame speed, S_T, and the laminar flame speed, S_L. For instance the model given by Damköhler [2]:

$$\frac{S_T}{S_L} = 1 + \frac{u'_{rms}}{S_L}$$

where u'_{rms} is the turbulence intensity.
The air/fuel ratio has a great impact on the flame speed, as the combustion temperature is very dependent on the air/fuel ratio. The highest combustion temperature is obtained at stoichiometric conditions, but due to some chemical effects the flame speed reaches its maximum for slightly rich mixtures. Figure 2 shows the burning velocity (laminar flame speed) for methane versus the fuel/air equivalence ratio, ϕ:

$$\phi = \frac{1}{\lambda} = \frac{\left(\frac{F}{A}\right)_{actual}}{\left(\frac{F}{A}\right)_{stoich.}}$$

where F/A is fuel/air ratio and λ is the relative air/fuel ratio. The flame speed reaches its maximum at around ϕ = 1.05 – 1.10, i.e. slightly on the rich side. The different curves in Figure 2, are from different experiments. The use of EGR (Exhaust Gas Recirculation) reduces emissions of nitrogen oxides, NO_x, due to the lowered combustion temperature and the reduced oxygen concentration. Using EGR has a similar reducing effect on the flame speed as leaning out the fresh mixture. For most hydrocarbons the laminar flame speed lies between 40 – 70 cm/s at stoichiometric and ambient conditions. However for hydrogen the laminar flame speed is over 200 cm/s. In natural gas the main component is methane, which has a maximum laminar flame speed of about 40 cm/s. Hence, adding hydrogen to a natural gas operated SI engine would give faster combustion, which in turn lead to higher fuel conversion efficiency and improved combustion stability [3].

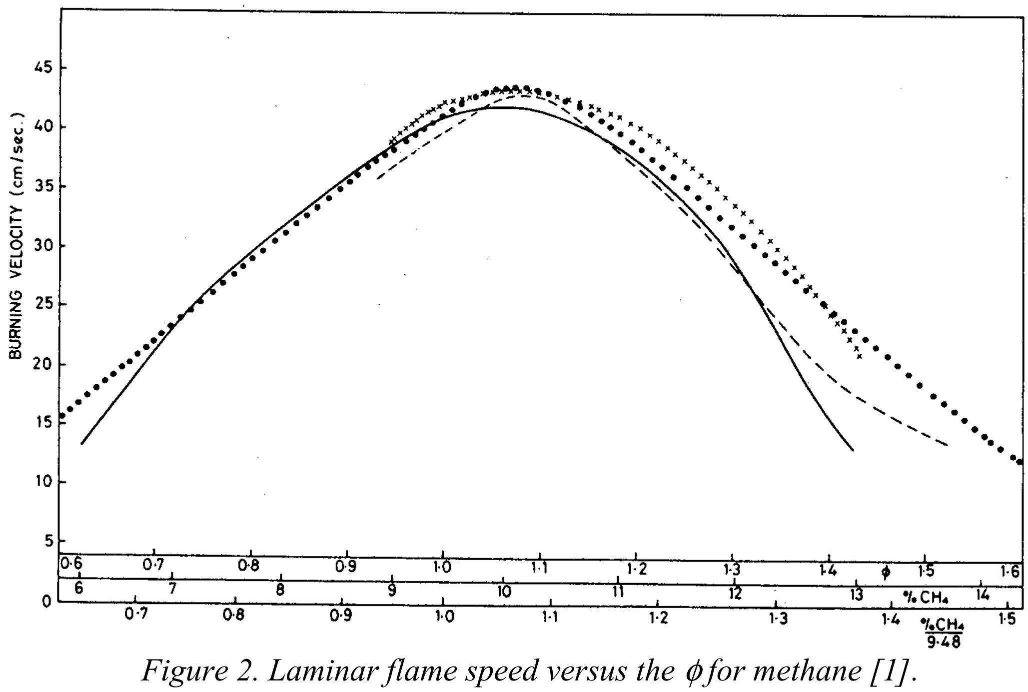


Figure 2. Laminar flame speed versus the ϕ for methane [1].

Table 1. Laminar flame speeds at λ=1.0 and ambient conditions [2].

Fuel	Formula	Laminar flame speeds, S_L (cm/s)
Methane	CH_4	40
Acetylene	C_2H_2	136
Ethylene	C_2H_4	67
Ethane	C_2H_6	43
Propane	C_3H_8	44
Hydrogen	H_2	210

1.1.3 Lean burn natural gas SI engines

Most natural gas engines for commercial vehicles are using converted, truck size, diesel engines. The combustion chambers in these engines are most commonly located in the piston crown and the cylinder head surface is flat. The inlet port is normally designed to generate a swirling in-cylinder gas motion to enhance direct injection diesel combustion. In the conversion to SI operation, the inlet port is kept unchanged. However the diesel combustion chamber is not suitable for SI operation. This means that the piston is changed to a piston with a different combustion chamber and a lower compression ratio. Heavy-duty diesel engines normally use a compression ratio in the range of 16:1 – 19:1. Suitable compression ratio for lean burn SI engines is around 10:1 – 12:1.

There are several demands on the design of the combustion chamber. It should give:

- highest possible fuel conversion efficiency
- lowest possible emissions
- low cycle-to-cycle variations (smooth operation)

This means that the combustion chamber characteristics should be such that the burn rate becomes high and heat losses as low as possible. To achieve lowest possible emissions, the crevice volume should be as small as possible. Normally cycle-to-cycle variations are kept small if the burn rate is high.

In this study we tested two different combustion chambers, one slow burning and one fast burning.

2 EXPERIMENTAL APPARATUS

2.1 Test engine

The test engine was a single cylinder version of an in-line six cylinder *Volvo®* SI natural gas engine. These types of engines are based on *Volvo® TD100* series diesel. Figure 3 shows a photo of the test engine. The engine is operated on cylinder number 6, i.e. the cylinder closest to the flywheel, and the other five cylinders are only 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 (from in-cylinder pressure measurements) have been used. With a multi-cylinder engine modified for single cylinder operation, an electric brake has to be used. Using a conventional starter would be impossible. Table 2 shows the engine specifications.

One drawback with this old style two-valve engine is that the valve area is quite small compared to the cylinder volume. As a result, pumping losses becomes high already at 1200 rpm. These pumping losses reduce the engine output and the fuel conversion efficiency.

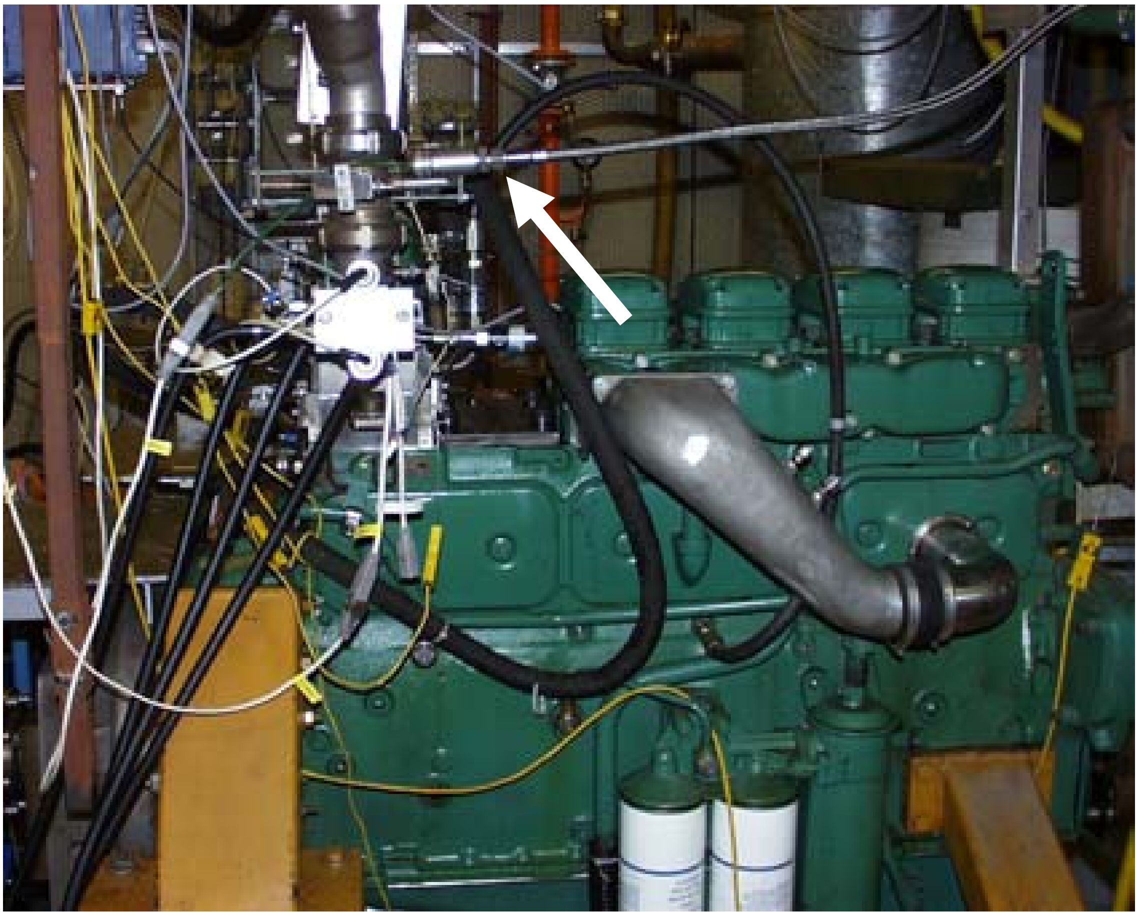


Figure 3. The modified Volvo TD100 engine viewed from the inlet side. The operating cylinder is to the left in the picture.

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

2.1.1 Fuel supply system

The gaseous fuels (natural gas and hydrogen) were added to the intake manifold approximately 400 mm upstream the intake valve (see arrow). Figure 4 shows a schematic picture of the test engine system. The natural gas is supplied to the engine laboratory via pipeline. The natural gas is held at a constant pressure of 4.6 bar (abs). Table 3 shows the natural gas composition. The amount of fuel is then adjusted by the time duration, when a solenoid is open once every engine cycle. The hydrogen fuel was stored on bottles. A Mass Flow Controlled (MFC) gas meter regulated the amount of hydrogen feeded to the engine.

Table 3. Natural Gas Composition.

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

<u>*2.1.2 Ignition system*</u>

A standard Transistorized Coil Ignition (TCI) system was used together with an *NGK® 2330* spark plug. The trigger pulse to the ignition system was supplied from the same triggering device as the cylinder pressure measurement system used.

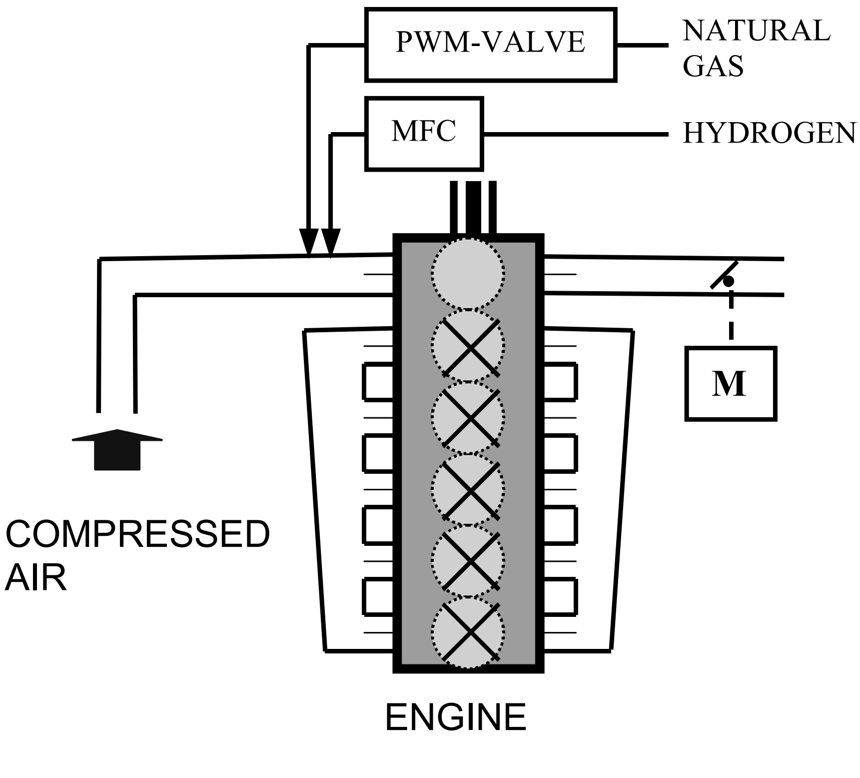


Figure 4. Schematic picture of the test engine system.

2.2 Test rig

2.2.1 Dynamometer

The dynamometer used was an electric AC motor. The user sets a wanted speed and then the control system adjusts the dynamometer torque in order to keep the desired speed independent of the test engine output torque.

2.2.2 Data logging

A *HP® 3852A* data logger connected to a PC was used for data logging. The data logger works like an accurate voltage meter. The voltage values are then transferred to the PC, where the voltage values are converted, displayed and stored. Several different pressures, temperatures and flows are of interest. The temperatures were measured with thermocouples of type K and the pressures with relatively slow piezo-resistive transducers. The gas flows were measured with *Bronkhorst® Hi-Tec* flow meters inserted in the gas line. The air/fuel ratio was measured via exhaust gas analysis.

2.2.3 Exhaust gas analysis

For the emission measurements a complete analyzer system from *Boo Instrument AB®* was used. It was capable of measuring HC, NO_x, CO_2, O_2 and CO. The system is equipped with dual CO_2 instruments, to be able to measure the CO_2 concentration also in the inlet manifold when the engine was operated with EGR. The HC and NO_x were measured wet, while the other gases were measured dry. The instrument is calibrated by the use of nitrogen as zero gas and different span gases with specified concentration.
The unburned HC was measured with an instrument called *Hydrocarbon Analyzer Model 109A* from *J.U.M Engineering®*. It uses a Flame Ionization Detector (FID) and measures in the range 0-100000 ppm. To support the flame a gas mixture of 60 % helium (He) and 40 % hydrogen (H_2) is needed.
The NO_x was measured with the use of an *Eco Physics® CLD 700 EL ht*. It uses a Chemi Luminescence Detector (CLD). The instrument can measure in the range 0-10000 ppm.
The CO_2, O_2, CO was measured with an instrument called *UNOR 611* from *Maihak®* using a Non Dispersive Infrared analyzer (NDIR). The CO_2 could be measured in the range 0-16 % and O_2 in the range 0-25 %. CO was measured in two different ranges, 0-2500 ppm for low concentrations and 0-10 % for high concentration.

2.3 Cylinder pressure measurement

2.3.1 Obtained information

Cylinder pressure changes with crank angle as a result of piston movement, combustion and heat transfer. By measuring the cylinder pressure, indicated work and efficiency can be calculated. But also information of combustion can be obtained. The effect of piston movement (i.e. volume change) on the pressure can easily be accounted for and the pressure increase due to combustion can be discerned.

2.3.2 Pressure transducer

The cylinder pressure was measured with a nowadays-common piezo-electric pressure transducer. The transducer contains a quartz crystal, which is exposed through a diaphragm to the cylinder gas. This type of transducer generates an electric charge, which is proportional to the pressure changes of the cylinder gas. It cannot measure absolute values, only changes in pressure. During an engine cycle, the in-cylinder temperature can vary from room temperature up to over 2500 K in a few milliseconds. Therefore, the transducer has to be compensated for thermal chocks, which otherwise had caused much noise on the output signal. For this work a *Kistler® 7061B* pressure transducer was used. This water-cooled type of transducer is known for being very accurate.

2.3.4 Charge amplifier

The charge amplifier receives the electric charge signal from the pressure transducer. In the charge amplifier, the charge signal is converted to a voltage signal. It is also possible to adjust the amplification of the output voltage signal. A *Kistler® 5011* was used with an output signal of ±10 volt.

2.3.5 A/D converter

The voltage signal from the charge amplifier has to be digitalized before it can be logged in a computer. Here, the A/D-conversion was made with an A/D converter on a board in a PC. The resolution of the conversion is determined by the number bits used. The A/D converter uses a certain voltage span and it divides the span into a number of discrete levels, determined by the number of bits used. A *Data Translation® 2823* A/D-board with 16-bit resolution was used.

2.3.6 Determination of the calibration constant

The entire calibration constant from the pressure transducer through the charge amplifier and to the A/D-converter is determined by using the specifications of the equipment in a few different ways. The specifications of the equipment needed for calculating the total calibration constant are given in Table 3.

Table 3. Specifications of the measurement equipment.

Equipment	Constant	Value/setting	Unity
Pressure transducer	C_{PT}	67.02E-5	pC/Pa
Charge amplifier	T	6.7E+1	pC/Mechanical Unit
	S	10-40	Mechanical Unit/V
A/D-converter	$R_{A/D}$	$20/2^{16}$	V/count

The calibration constant can be calculated as follows:

$$Cal\_const\left[\frac{Pa}{count}\right]=\frac{20}{2^{16}}\left[\frac{V}{count}\right]\bullet S\left[\frac{MU}{V}\right]\bullet T\left[\frac{pC}{MU}\right]\bullet\frac{1}{C_{PT}}\left[\frac{Pa}{pC}\right]$$

2.4 TEST CONDITIONS

2.4.1 Hydrogen addition

Four different hydrogen concentrations were tried (0, 7, 14 and 20 vol%). This corresponds to 0, 2, 4 and 6 % of the total input fuel energy.

2.4.2 Combustion chambers

In this work two different levels of turbulence were used. The in-cylinder turbulence was altered by using pistons with different combustion chambers. Figure 5 shows the two combustion chambers, referred as Turbine and Quartette. The piston type used in production natural gas TD100 engines, Turbine, has a fairly open combustion chamber geometry. The Quartette piston has a more complex combustion chamber shape. Figures 6a and 6b show the turbulence and mean velocity versus crank angle position for Turbine and Quartette. The in-cylinder turbulence was obtained by the use of Laser Doppler Velocimetry, LDV, by Einewall [4]. With Turbine, the turbulence peaks (1.8 m/s) some 15° BTDC. Quartette has its turbulence peak (3 m/s) at around TDC. In the crank angle window where the main combustion process normally takes place, i.e. from about 20° BTDC to 30° ATDC, the turbulence is much higher for Quartette.
Figure 7 displays the heat release rate and the cylinder pressure obtained at similar operating conditions for the both combustion chambers. It can clearly be seen that the heat release rate is much higher for Quartette, due to the higher turbulence, which enhances the flame propagation speed.

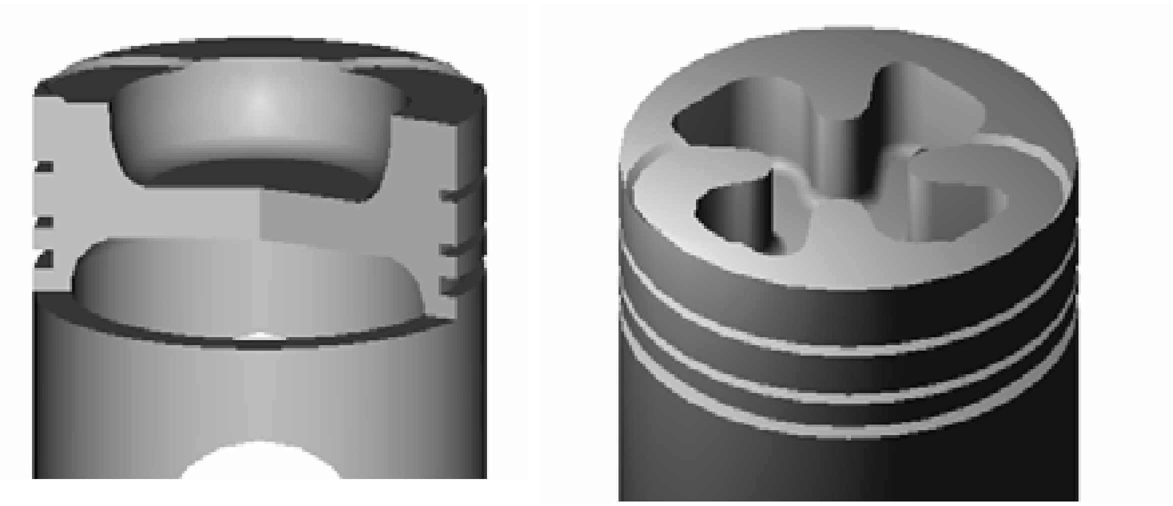


Figure 5. Turbine (left) and Quartette (right) piston.[4]

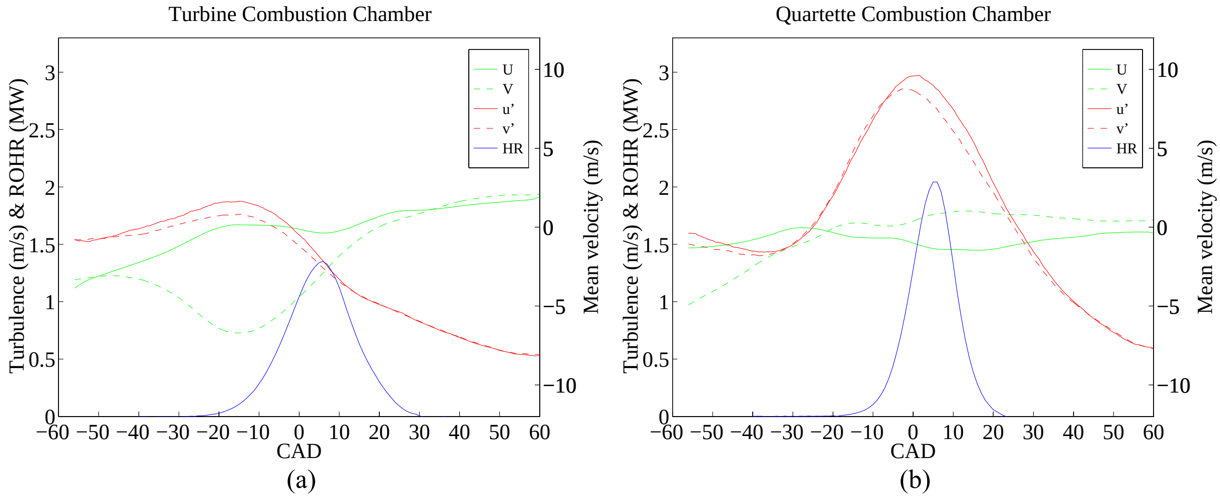


Figure 6. (a) Turbulence and mean velocity for Turbine. (b) Turbulence and mean velocity for Quartette. The rate of heat release curves are from λ = 1.5, WOT and 1200 rpm.[4]

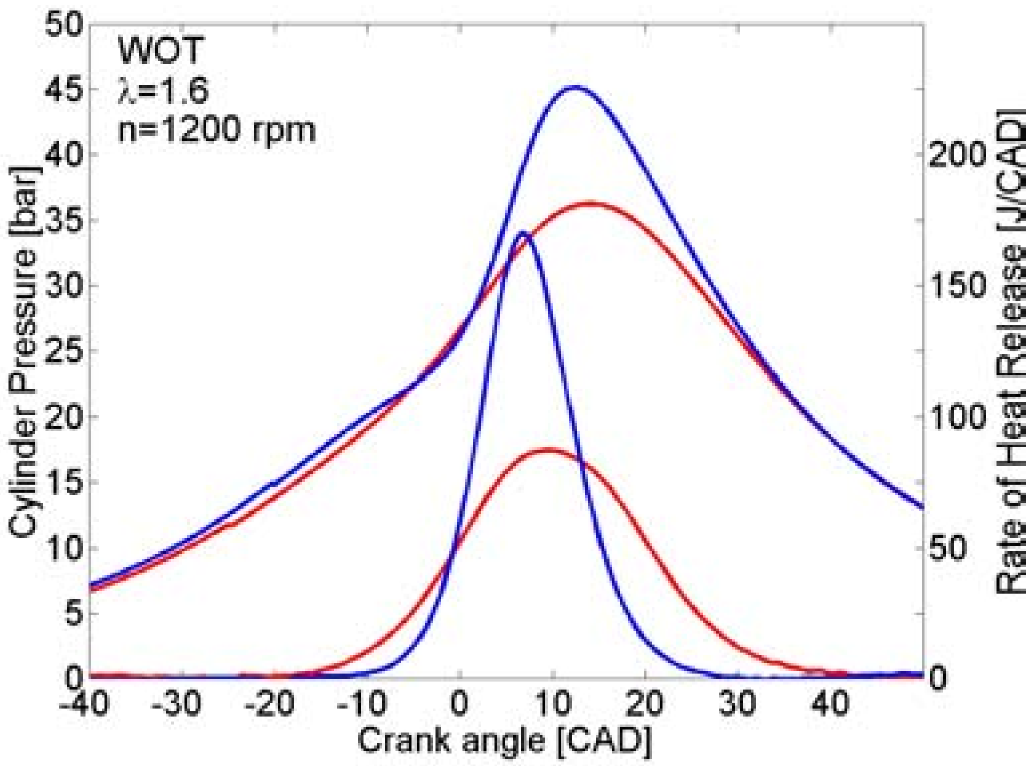


Figure 7. Pressure traces and heat release rates for Turbine and Quartette pistons (MBT ignition timing). The cylinder pressure trace with the highest peak value and heat release curve with the highest peak value corresponds to Quartette.

2.4.3 Test cases

For each combustion chamber, the engine was operated at three different basic test conditions:

- Idle, IMEP = 1.5 bar
- WOT, IMEP = 5-8 bar
- High load, supercharged, IMEP = 13 bar

Table 4 specifies the different test conditions, performed with both combustion chambers.

Table 4. Test cases.

Test case	IMEP [bar]	Inlet pressure [bar]	λ	Hydrogen [%]	Ignition
Idle	1.5	0.32-0.46	1.0-1.7	0-20	MBT timing
WOT	5-8	1	1.0-1.9	0-20	MBT timing
High load	13	1.5-2.1	1.2-1.8	0-20	MBT timing

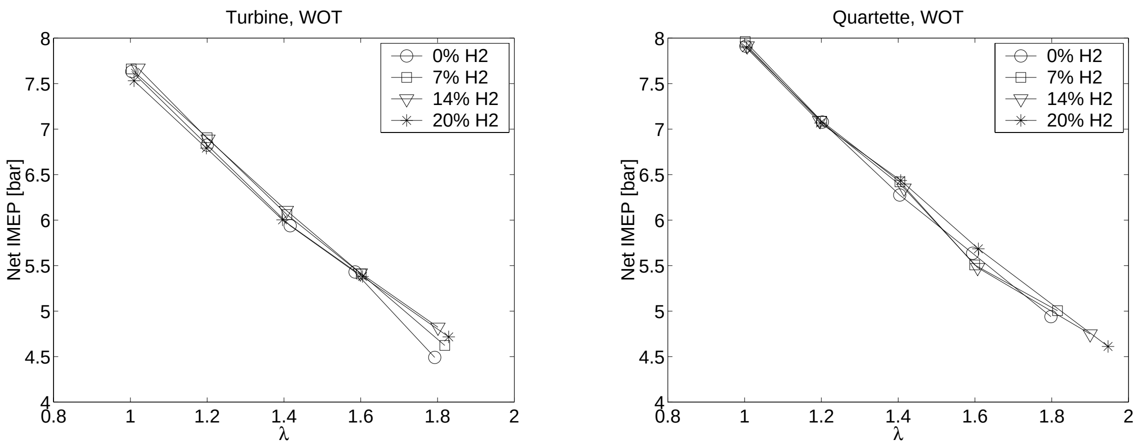
3 TEST RESULTS

The experimental results from the WOT case will be presented first and the most thoroughly discussed. The Idle case and the High Load case are discussed more briefly.

3.1 Results from the WOT case

3.1.1 λ-window

At the Wide Open Throttle conditions, λ was altered by changing the fuel flow. λ was altered from unity (stoichiometric condition) up to the lean limit. As a result of this test strategy, IMEP ranges from about, 5 to 8 bar, see Figure 8. With Turbine it was possible to operate up to λ = 1.8 with reasonable cycle to cycle variations. For the more turbulent Quartette piston it was possible to operate at almost λ = 2.0 when the hydrogen content of the fuel was 20%.



(a) (b)

Figure 8. (a) Net indicated mean effective pressure for Turbine versus λ.
(b) Net indicated mean effective pressure for Quartette versus λ.

3.1.2 Combustion

Figure 9a shows the duration of 10 – 90 % heat released measured in crank angles versus λ for Turbine. This duration is a good measure of how fast the main combustion process is. The effect of hydrogen addition is stronger at the leanest condition (λ =1.6). This depends on that the drop in laminar flame speed with increased λ is greater for natural gas than for hydrogen. Figure 9b shows the combustion duration for the Quartette piston. Generally the combustion duration is shorter here compared to Turbine. This is mainly due to the higher turbulence. The combustion duration is less influenced by the hydrogen addition for Quartette. However, there are some test points that deviates from the general trends. The combustion duration is quite sensitive to ignition timing. Retarded ignition timing from MBT results in longer combustion duration and advanced timing gives shorter combustion duration. In Figure 10 is the duration of 0-10% heat released shown. This duration is usually called the flame development angle in the literature [5]. The early flame development is very dependent on λ and is strongly related to the cycle to cycle variations of the combustion. For both combustion chambers, at the lean side of operation, the flame development angle is quite much reduced by hydrogen addition. However, at stoichiometric conditons, the influence is almost neglectable. The combustion efficiency increases when adding hydrogen, see Figure 11. The combustion efficiency is a measure of how much of the chemical energy in the fuel that is converted into heat:

$$\eta_C = \frac{q}{m_f \cdot Q_{LHV}}$$

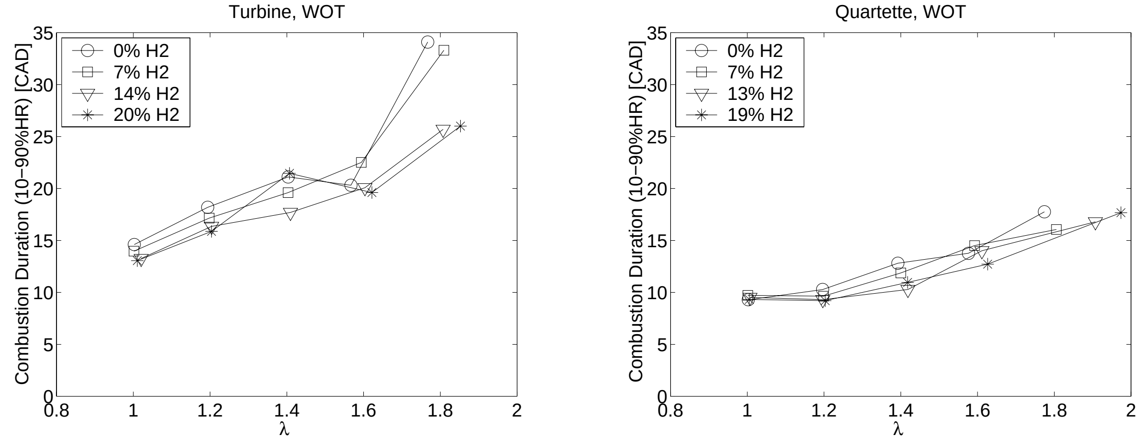
where q is the total heat released per cycle in Joule, m_f is fuel mass per cycle and Q_{LHV} is the lower heating value of the fuel in Joule per kilogram. However, the combustion efficiency presented in Figure 11 is evaluated from the exhaust gas analysis. The combustion inefficiency is related to the amount of unburned hydrocarbons, CO and H_2 in the exhaust gases and can be calculated as follows:

$$1 - \eta_C = \frac{m_{HC} Q_{HC} + m_{CO} Q_{CO} + m_{H2} Q_{H2}}{m_f Q_{LHV}}$$

and rewriting this gives the expression for the combustion efficiency:

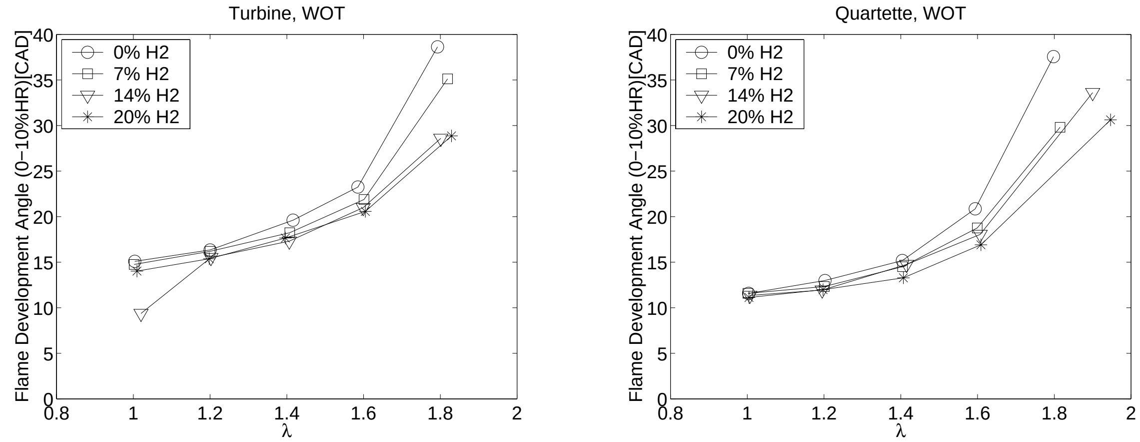
$$\eta_C = 1 - \frac{m_{HC} Q_{HC} + m_{CO} Q_{CO} + m_{H2} Q_{H2}}{m_f Q_{LHV}}$$

where m_{HC}, m_{CO} and m_{H2} are the masses of HC, CO and H_2, respectively, in the exhaust gas. Further, Q_{HC}, Q_{CO} and Q_{H2} are the heating values of these species.
Generally, best combustion efficiency is obtained slightly on the lean side, due to the excess of air. However, it decreases when leaning out from λ =1.2, as the combustion temperature is lowered and the quenching distance gets longer.



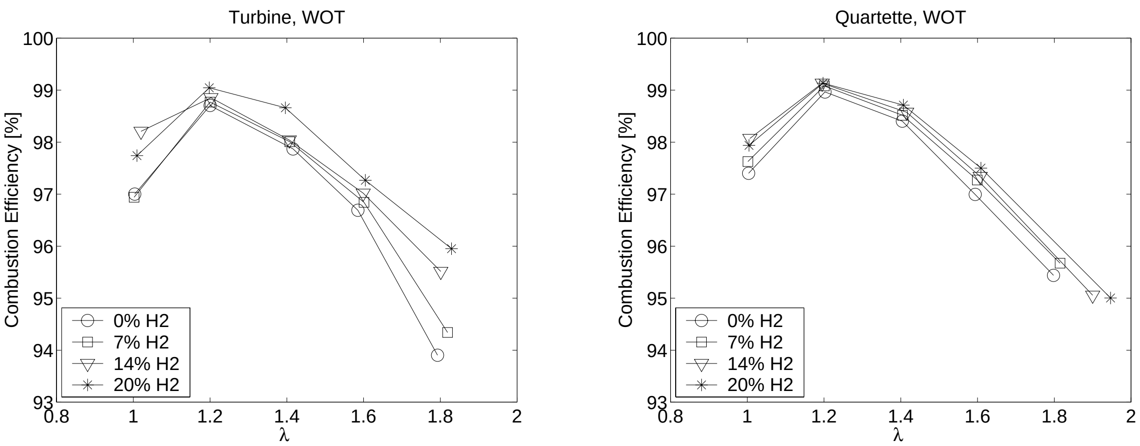
(a) (b)

Figure 9. (a) Combustion duration versus λ for Turbine.
(b) Combustion duration versus λ for Quartette.



(a) (b)

Figure 10. (a) Flame development angle versus λ for Turbine.
(b) Flame development angle versus λ for Quartette.



(a) (b)

Figure 11. (a) Combustion efficiency for Turbine. (b) Combustion efficiency for Quartette.

3.1.3 Engine efficiency (fuel conversion efficiency)

Figure 13 shows the gross indicated efficiency. This efficiency is calculated from the fuel flow and the indicated mean effective pressure during the compression and expansion stroke only (so called gross IMEP). This means that pumping work and engine friction is not considered. 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 ($W_{i,G}$) and the input fuel energy per cycle($m_f Q_{lhv}$):

$$\eta_{i,G} = \frac{W_{i,G}}{m_f Q_{lhv}} = \frac{IMEP_G \cdot V_D}{m_f Q_{lhv}}$$

where V_D is the displacement volume.

With both combustion chambers the gross indicated efficiency improves with increased amount of hydrogen. This depends on several factors. Firstly, increased amount of hydrogen leads to increased burn rate which is beneficial for efficiency. Secondly, adding hydrogen improves the combustion efficiency to some degree. Thirdly, the ratio of specific heats (γ) of the burned gas, goes up with increased amount of hydrogen. The thermal efficiency ($\eta_{i,CV}$) for the ideal constant-volume (CV) cycle is given by:

$$\eta_{i,CV} = 1 - \frac{1}{r_c^{\gamma-1}}$$

where r_c is the compression ratio [5]. Values of $\eta_{i,CV}$ for different values of γ are shown in Figure 12. The efficiency improves with increasing γ. The gross indicated efficiency adjusted with gas exchange efficiency is called the net indicated efficiency. This efficiency is defined on the work delivered to the piston over the entire four-stroke cycle. It is calculated in a similar manner as $\eta_{i,G}$, but $IMEP_N$ is used instead:

$$\eta_{i,G} = \frac{W_N}{m_f Q_{lhv}} = \frac{IMEP_N \cdot V_D}{m_f Q_{lhv}}$$

In Figure 14 is the net indicated efficiency shown. Due to the pumping work this is normally some percent points lower than the gross indicated efficiency.

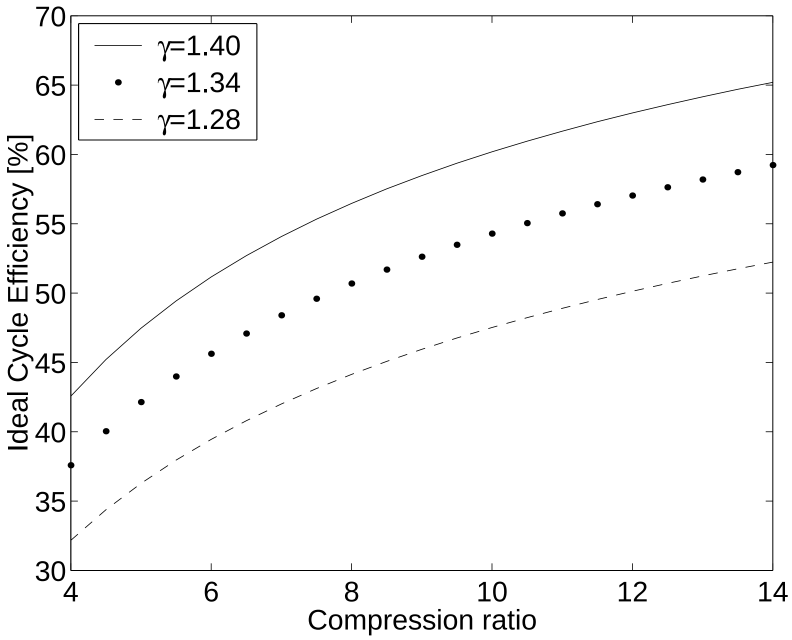
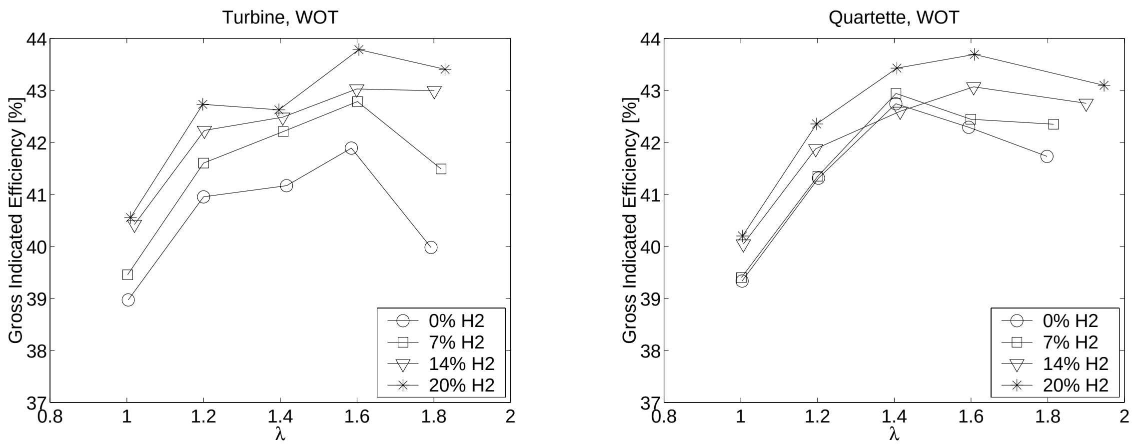
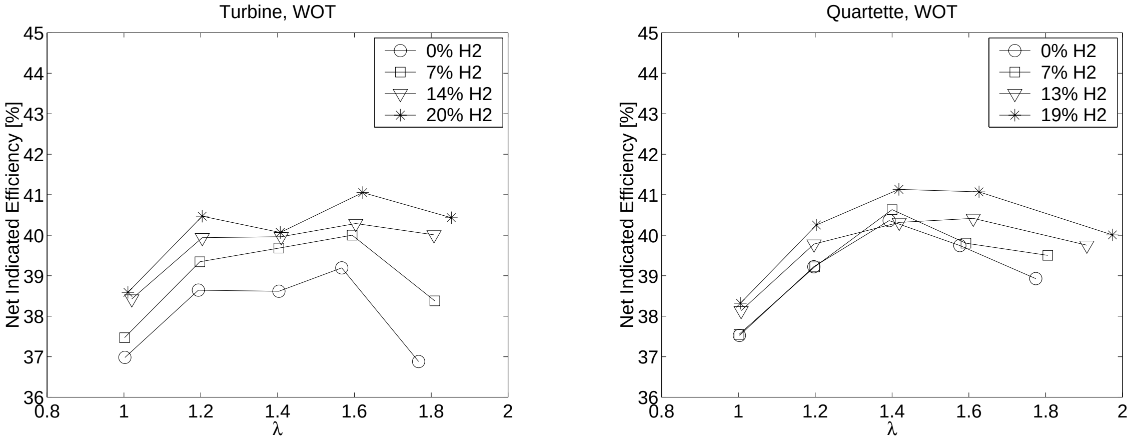


Figure 12. Ideal constant volume cycle efficiency as function of compression ratio for different γ- values.



(a) (b)

Figure 13. (a) Gross indicated efficiency for Turbine.
(b) Gross indicated efficiency for Quartette.



(a) (b)

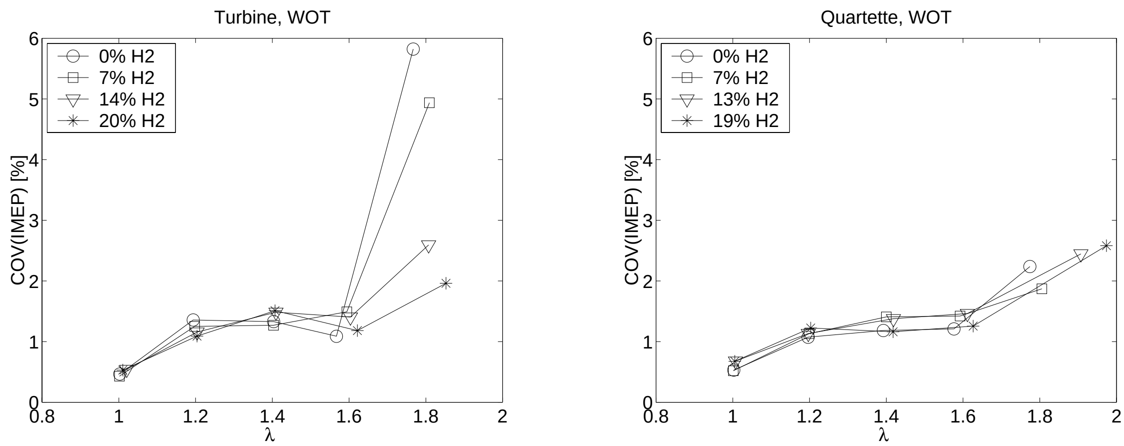
Figure 14. (a) Net indicated efficiency for Turbine.
(b) Net indicated efficiency for Quartette.

<u>*3.1.4 Coefficient of variation (COV)*</u>

A useful measure of combustion stability and cycle by cycle variations is the Coefficient Of Variation (COV) of IMEP and is defined as follows:

$$COV(IMEP) = \frac{\sigma_{IMEP}}{\overline{IMEP}}$$

where σ_{IMEP} is the standard deviation of IMEP and $\overline{IMEP}$ is the mean value of IMEP for 100 cycles. At operation with richer mixtures than $\lambda = 1.6$, COV(IMEP) is very little influenced by the amount of hydrogen content. However, at the lean limit COV(IMEP) is much reduced with increased hydrogen content for the slow combustion chamber. With the fast combustion chamber, hydrogen has a very small effect on COV(IMEP). Figure 15 shows COV (IMEP).

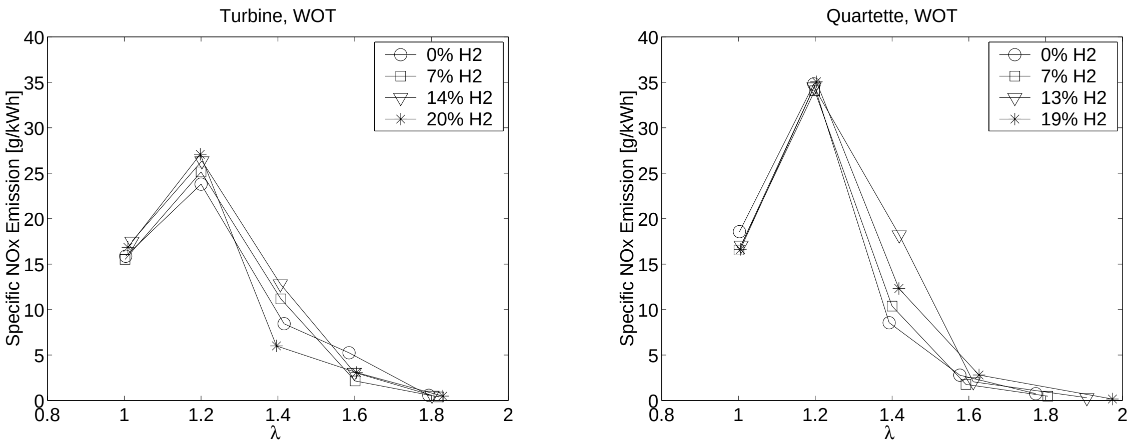


(a) (b)

Figure 15. (a) Coefficient Of Variation (COV) of IMEP for Turbine.
(b) COV of IMEP for Quartette.

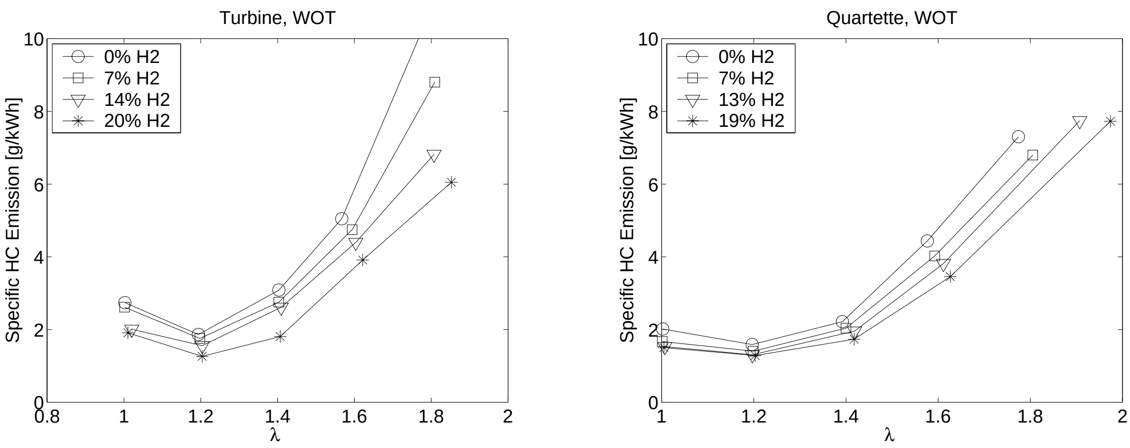
Emissons

Figure 16 shows the (indicated) specific NO_x emission, which is measured in grams per kilowatt hour (g/kWh). The specific NO_x emissions show its maximum at $\lambda = 1.2$ with both chambers. Due to the higher burn rate and thereby higher maximum cycle temperature, NO_x is higher for Quartette. Leaning out from $\lambda = 1.2$, NO_x is strongly reduced. At $\lambda = 1.8$, NO_x is very low. At the lean limit ($\lambda = 1.9$) with Quartette, the volume fraction of NO_x is below 20 ppm.
The emission of unburned hydrocarbons, HC, shows its minimum at $\lambda = 1.2$ for both combustion chambers, see Figure 17. At leaner conditions, HC increases with increased λ due to the lower combustion temperature and longer quenching distance. Higher hydrogen fuel concentration reduces HC, mainly due to lower hydrocarbon concentration in the fresh charge. The reduction of HC is also partly dependent of the higher combustion temperature. Like the emission of unburned hydrocarbons, the CO emission (Figure 18) shows its lowest value at $\lambda = 1.2$. At $\lambda = 1.0$ most CO is formed in fuel rich regions, but at the lean side of operation, CO is a result of incomplete oxidation due to low temperature. The decrease of both HC and CO is slightly more pronounced for Turbine compared to Quartette.



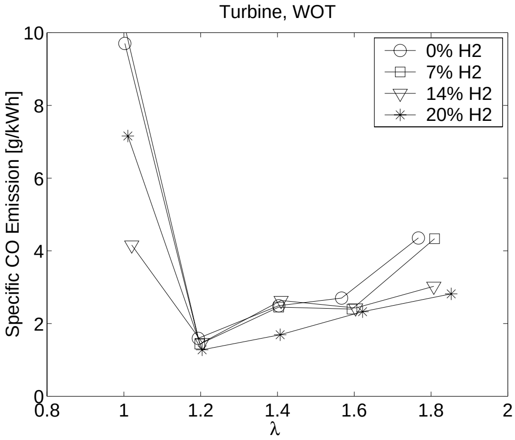
(a) (b)

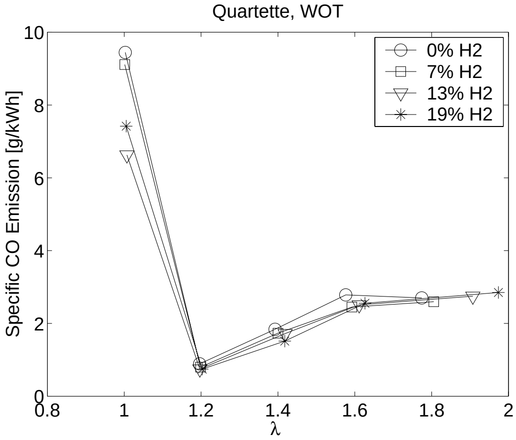
Figure 16. (a) Indicated specific NO_x emission for Turbine.
(b) Indicated specific NO_x emission for Quartette.



(a) (b)

Figure 17. (a) Indicated specific HC emission for Turbine
(b) Indicated specific HC emission for Quartette.





(a) (b)

Figure 18. (a) Indicated specific CO emission for Turbine.
(b) Indicated specific CO emission for Quartette.

<u>MBT ignition timing</u>

The determination of the MBT timing was done by sampling test points with different ignition timings and afterwards evaluate which ignition timing that gave the highest IMEP. As an example Figure 19 shows IMEP versus ignition timing for Turbine at $\lambda = 1.6$. The general trend is that the Maximun Brake Torque, MBT, ignition timing is retarded with increasing hydrogen content. Due to the higher burn rate with Quartette, the MBT ignition timing is retarded compared to Turbine.

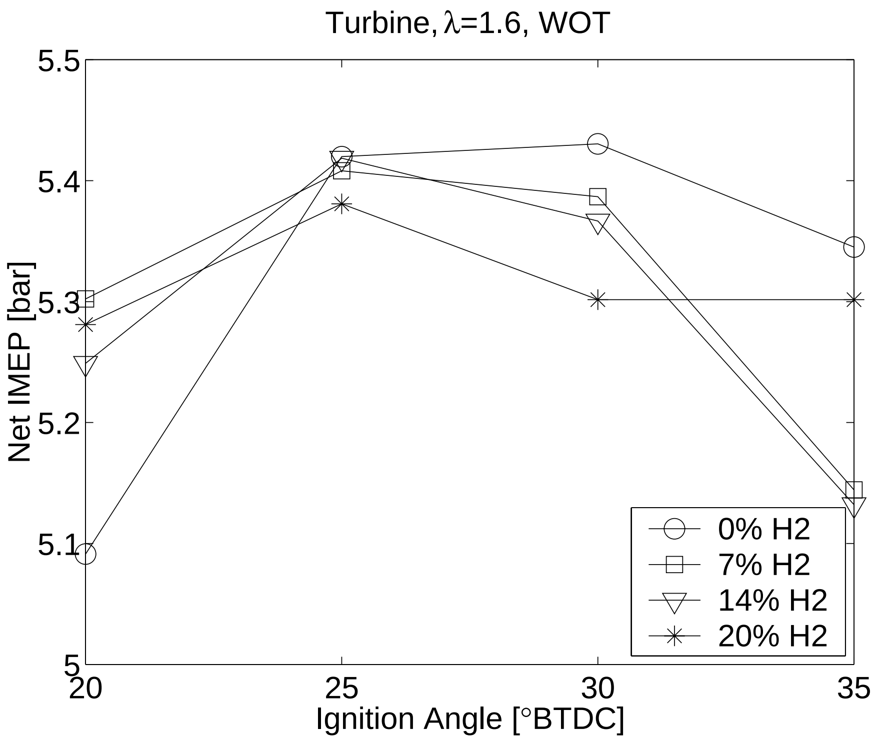
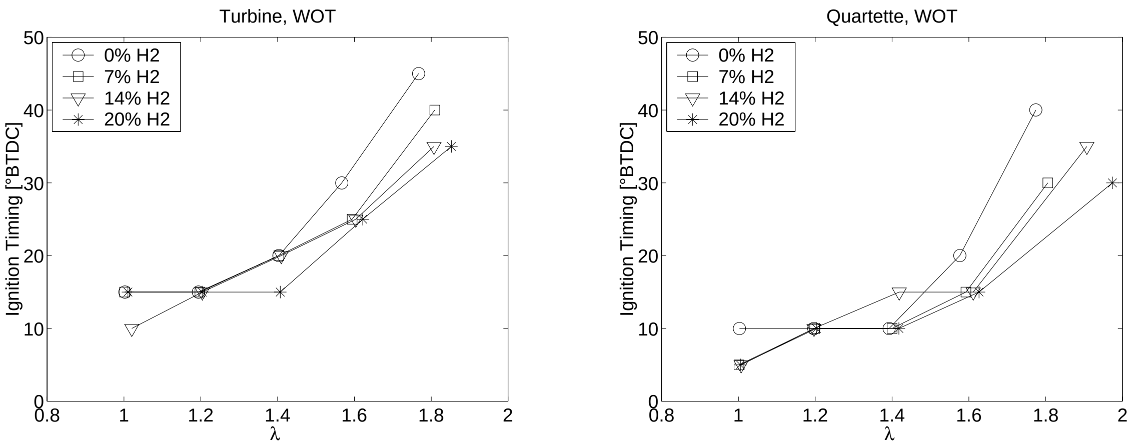


Figure 19. IMEP versus ignition angle for Turbine at $\lambda = 1.6$.



(a) (b)

Figure 20. (a) Ignition angles for Turbine. (b) Ignition angles for Quartette.

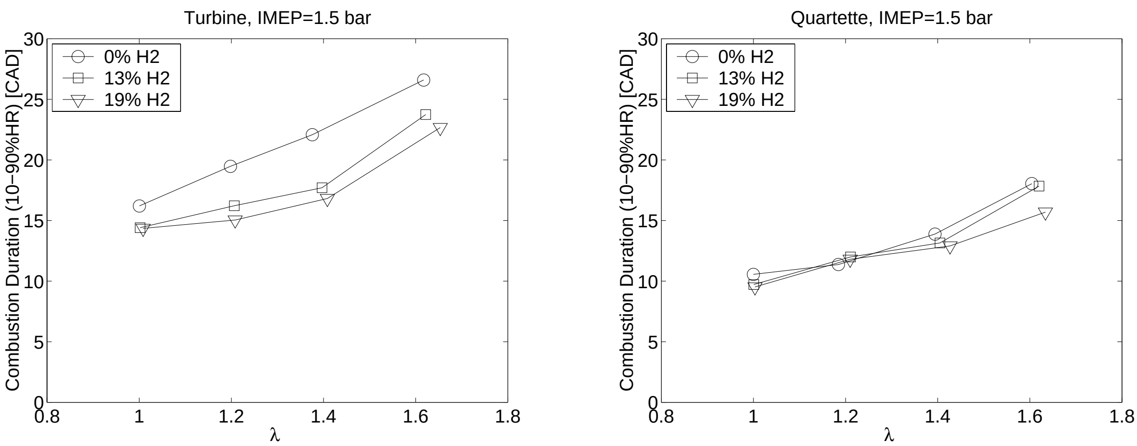
Results from the idle case

<u>λ-window</u>

A constant load of 1.5 bar IMEP was kept by adjusting both throttle and fuel to reach the desired λ-values between λ = 1.0 up to λ = 1.6.

<u>Combustion</u>

As expected shows the case without the use of hydrogen the longest combustion duration and the case with the highest amount of hydrogen (19 vol%) the shortest duration, this for the Turbine see figure 21 (a). Figure 21 (b) shows the combustion duration for Quartette. Here is the influence of hydrogen addition less obvious. One explanation for this can be that the combustion duration is overall much shorter for Quartette. The decrease in combustion duration will thereby be less measured in crank angles. Furthermore, the combustion duration is not only determined by the flame speed, but also by available flame area.

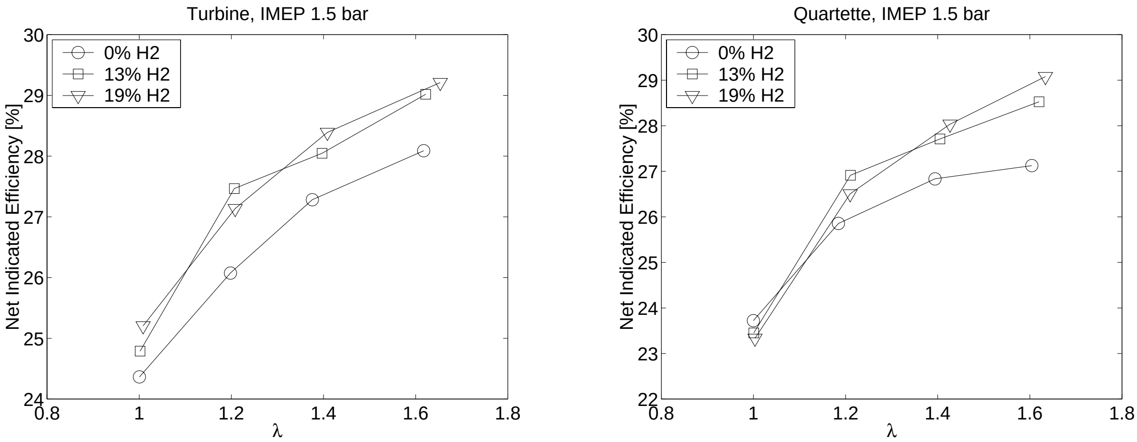


(a) (b)

Figure 21. (a) Combustion duration versus λ for Turbine.
(b) Combustion duration versus λ for Quartette.

Engine efficiency

Figure 22 shows the net indicated efficiency versus λ. The net indicated efficiency improves with increased amount of hydrogen for both Turbine and Quartette, due to the decreased combustion duration (increased burn rate) and the improved combustion efficiency.

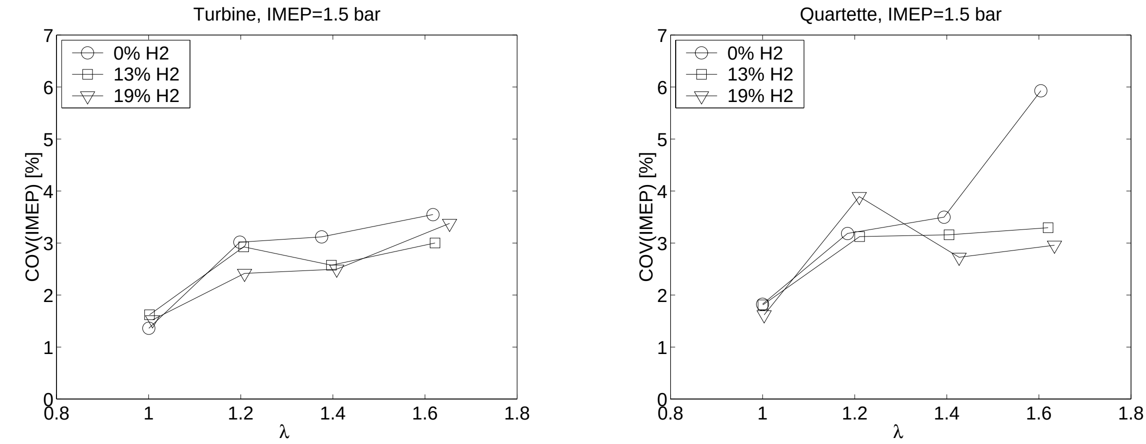


(a) (b)

Figure 22. (a) Net indicated efficiency for Turbine.
(b) Net indicated efficiency for Quartette.

Coefficient of variation (COV)

The Coefficient Of Variation (COV) of IMEP is a good measure of the combustion stability and the cycle-by-cycle variation. The overall trend is that COV(IMEP) decreases with increased amount of hydrogen, see Figure 23. Quite unexpected, COV(IMEP) proved not to be lower for Quartette than for Turbine in these operating conditions. This could be due to other effects than fluid flow dominating the combustion stability. The engine is operated with low inlet manifold pressure and thus less favorable conditions during valve overlap. The large residual gas fraction could then be the major reason for cycle-by-cycle variations.



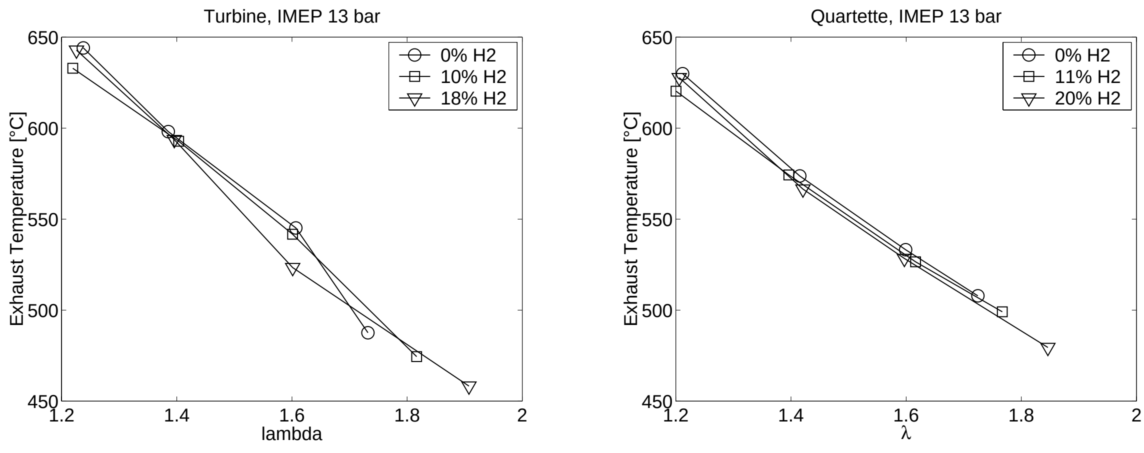
(a) (b)

Figure 23. (a) Coefficient Of Variation (COV) of IMEP for Turbine.
(b) COV of IMEP for Quartette.

Results from the high load case

λ-window

This test case was run at a constant load of 13 bar IMEP. As a consequence of this test strategy, the boost pressure had to be increased with rising λ. Due to too high exhaust temperatures, the test span began at λ=1.2 instead of λ=1.0 as in the previous tests. The temperature reached around 650°C at λ=1.2, and were below 550°C when running at λ=1.6. Figure 24. shows the exhaust temperatures.



(a) (b)

Figure 24. (a) Exhaust temperatures for Turbine.
(b) Exhaust temperatures for Quartette.

Turbo simulation

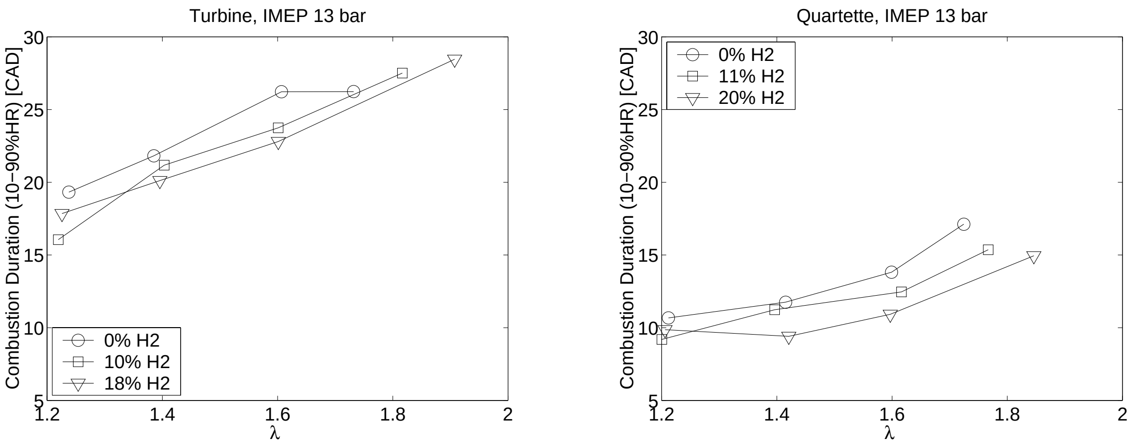
An external air compressor supplied the boost pressure. In order to simulate real turbo machinery, an exhaust throttle was used to adjust the exhaust backpressure. A simple model for turbo efficiency calculation was used in order to adjust the exhaust backpressure to a level corresponding to a turbo efficiency of around 55%. The turbo efficiency model is based on inlet and exhaust conditions. Due to some characteristics of the exhaust throttle and to some small variations in inlet pressure, it was quite hard to adjust the backpressure properly at some test conditions. As a result of this, the turbo efficiency fluctuates between values from about 50-60%.

Combustion

With Turbine the effect of hydrogen on the combustion duration proved to be similar to the previous test cases, i.e. adding hydrogen results in shorter combustion duration. As can be seen in Figure 25(a) some deviations from this trend can be found though. This can depend on several factors:

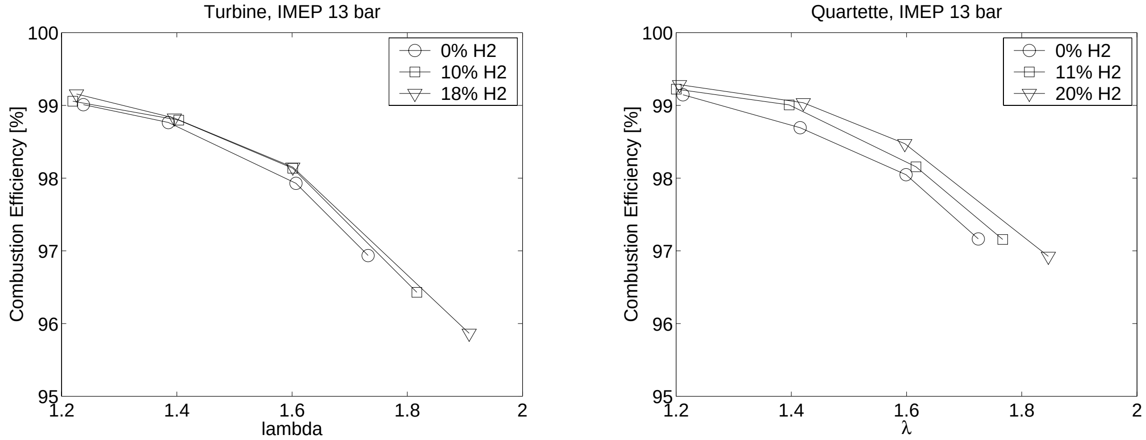
- Deviation from MBT ignition timing
- Variation in ratio of back- and inlet pressure (i.e. variation in internal EGR).

The combustion duration is sensitive to ignition timing. Normally, retarded ignition timing from MBT timing results in longer combustion duration and advanced timing gives shorter duration. Variations in pressure ratio over the engine will affect the amount of internal EGR. Increased amount of EGR will significantly increase the combustion duration. This means that a small deviation in ignition timing and/or variation in internal EGR can suppress the effect from the hydrogen addition. For some reasons, the influence of hydrogen addition is slightly larger with Quartette compared to Turbine, see Figure 25, that presents the combustion duration, and Figure 26, that shows the combustion efficiency.
But, as explained above, these high load tests were more unstable than the previous cases of WOT and idle and this combustion trend can not be totally relied on. It is clear though, that the combustion chamber influences the combustion duration more than the hydrogen addition. Quartettes leanest point, with no hydrogen is comparable to Turbines richest test point with maximum hydrogen addition, regarding combustion duration.



(a) (b)

Figure 25. (a) Combustion duration for Turbine.
(b) Combustion duration for Quartette.

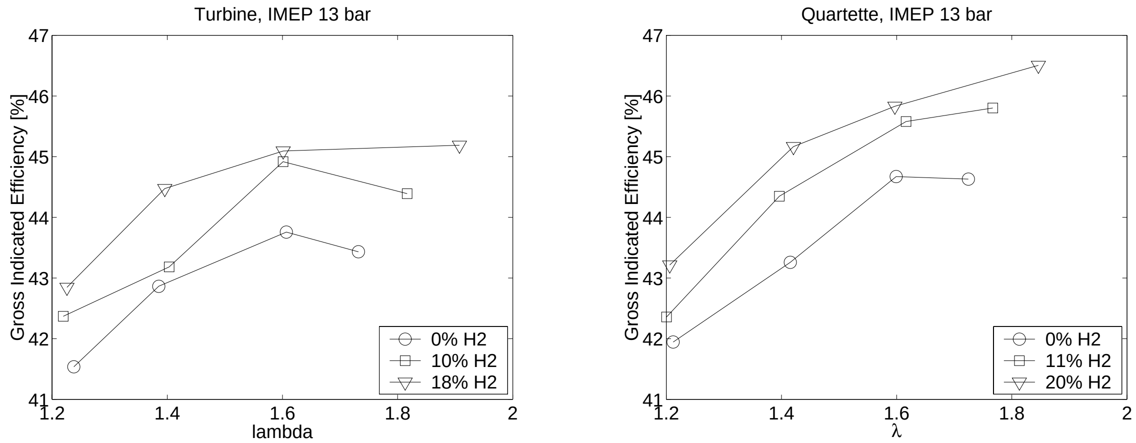


(a) (b)

Figure 26 (a)Combustion efficiency for Turbine versus λ.
(b) Combustion efficiency for Quartette versus λ.

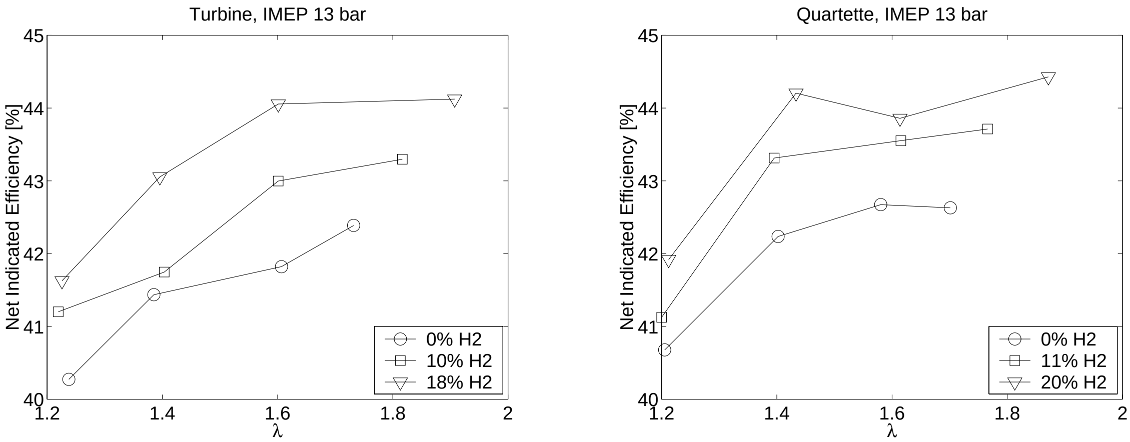
Engine efficiency

Likewise as for the WOT test case, adding more hydrogen leads to improved efficiency. The efficiency increases with the same rate for both combustion chambers and thereby seems to be a pure fuel effect rather than the effect of burn rate. Generally, the net indicated efficiency, see Figure 28, is a few percentage points higher here compared to the WOT case. This depends on the lower pumping losses and to some degree higher combustion efficiency for the supercharged case. Figure 27 presents the gross indicated efficiency, i.e. the efficiency for the compression and expansion stroke only.



(a) (b)

Figure 27. (a) Gross indicated efficiency for Turbine.
(b) Gross indicated efficiency for Quartette.

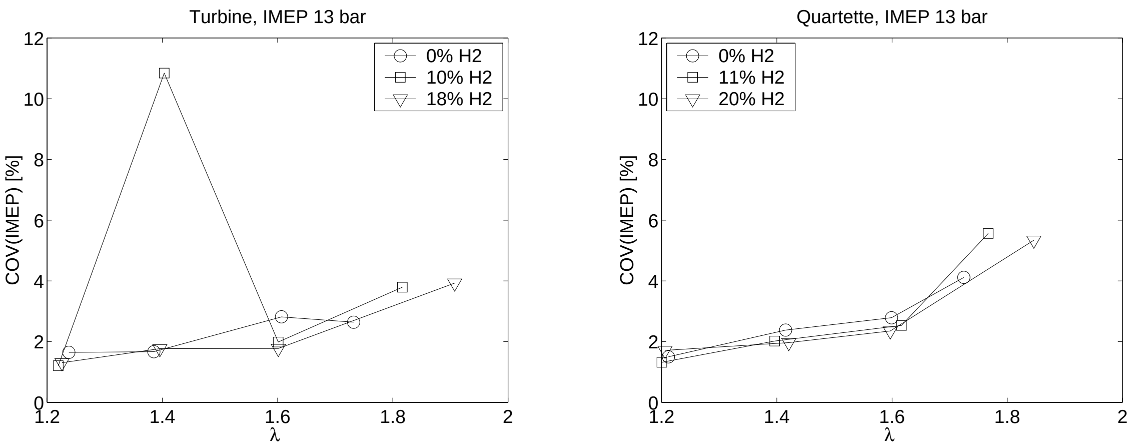


(a) (b)

Figure 28. (a) Net indicated efficiency for Turbine.
(b) Net indicated efficiency for Quartette. Note the different scale on the y-axis.

COV(IMEP)

The combustion stability is a bit more difficult to evaluate from the high load tests. As can be seen in Figure 29 no absolute trends can be found, although for most tests hydrogen addition seems to lower COV(IMEP). Again, it can be the result of variations in backpressure ratio or other deviations that makes the reading difficult. From Figure 29(a) a misfire could be traced, which raises COV(IMEP) severely.



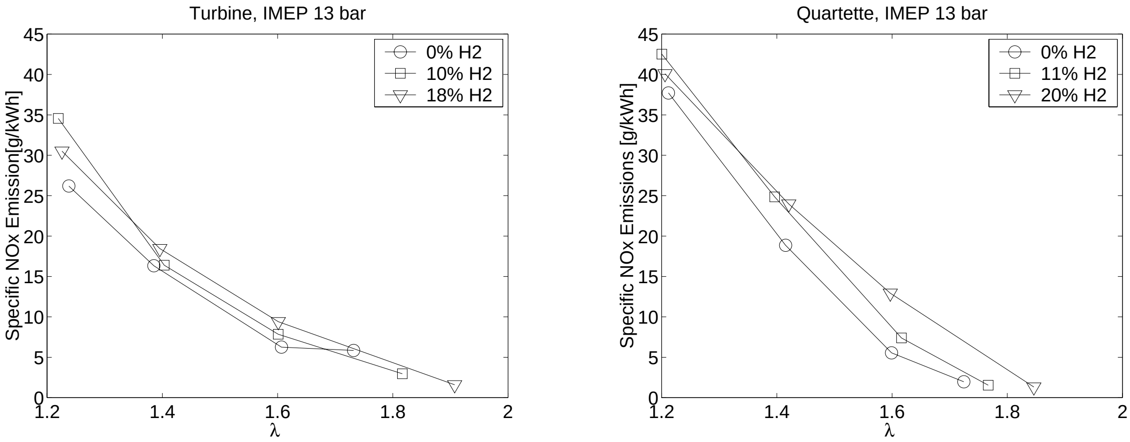
(a) (b)

Figure 29. (a) Coefficient Of Variation (COV) of IMEP for Turbine.
(b) COV of IMEP for Quartette.

Emissions

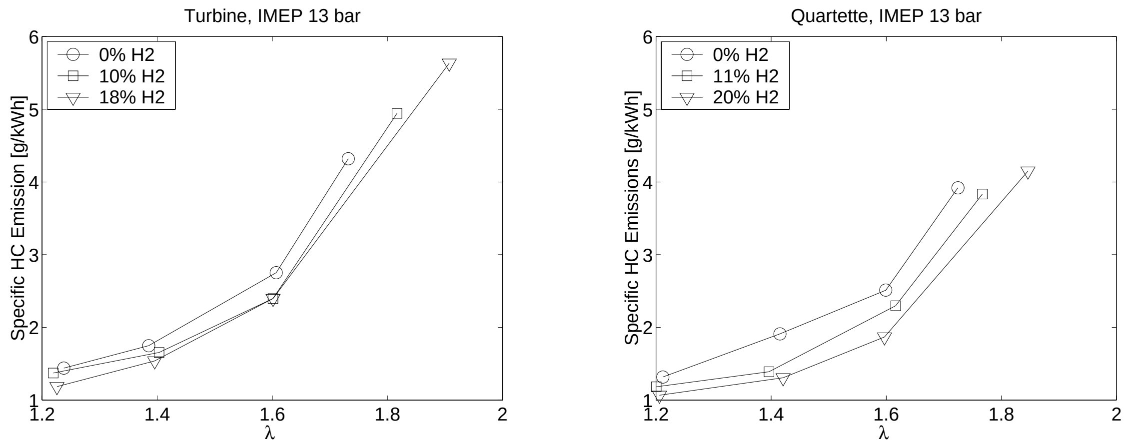
Figure 30 presents specific NO_x versus λ. It can be seen that low NO_x emission can be obtained at the leaner conditions. Hydrogen addition extends the lean operation limit, which results in lower combustion peak temperature resulting in low NO_x formation. The higher turbulence (faster combustion) for the Quartette combustion chamber results in higher combustion temperature and higher NO_x emission. At the lean limit ($\lambda = 1.9$), both Quartette and Turbine shows very low NO_x, both cases with approximately 20% hydrogen.

The emission of unburned hydrocarbons, HC, shows its minimum at $\lambda = 1.2$ for both combustion chambers, see Figure 31. Leaning out from $\lambda = 1.2$ gives rise in HC, due to the lower combustion temperature. The CO emission (Figure 32) shows its lowest value at $\lambda = 1.2$. Generally, when running lean, Quartette shows lower HC and CO emissions. This is due to the higher turbulence which improves the combustion and lowers HC and CO. Hydrogen addition decreases both HC and CO. The reason for the decreased HC can partly be explained by lower hydrocarbon content in the fresh charge.This decrease in HC and CO gives the improved combustion efficiency.



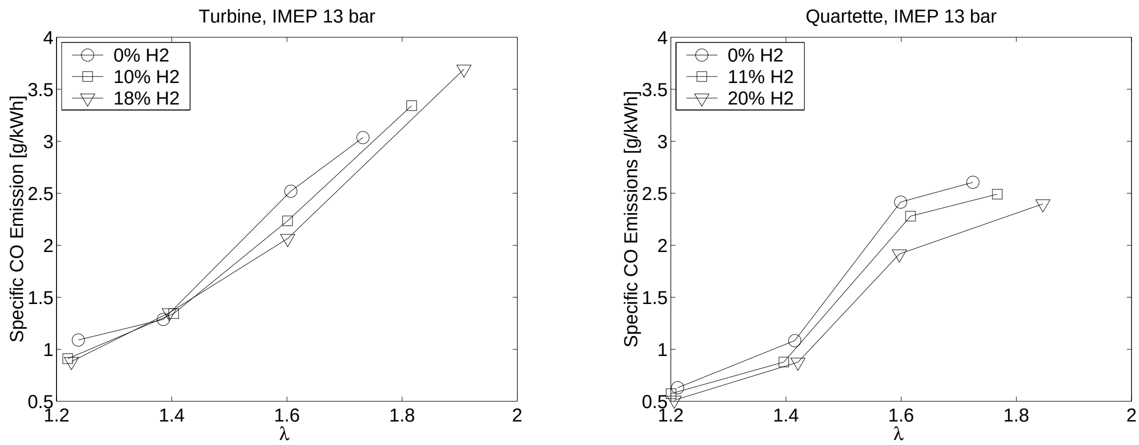
(a) (b)

Figure 30. (a) Indicated specific NO_x emission for Turbine.
(b) Indicated specific NO_x emission for Quartette.



(a) (b)

Figure 31. (a) Indicated specific HC emission for Turbine.
(b) Indicated specific HC emission for Quartette.



(a) (b)

Figure 32. (a) Indicated specific CO emission for Turbine.
(b) Indicated specific CO emission for Quartette.

4 DISCUSSION

Generally, lean burn engines are not equipped with exhaust gas aftertreatment systems for NO_x reduction. Therefore it is vital to operate the engine sufficiently lean that NO_x is kept as low as possible. However, at light loads the combustion stability becomes poor with increasing cycle-by-cycle variations as a result. In order to improve the combustion stability at light loads, fuel enrichment is normally used (reduced λ-value). As a result of this, NO_x emissions raise much. The solid line with circle markers in Figure 33, shows volume fraction of NO_x versus engine load for λ = 1.6. The filled circle by the dotted line shows the NO_x emissions at fuel enrichment, from λ = 1.6 to λ = 1.2. As a consequence of this fuel enrichment NO_x increases by almost two orders of magnitude (from 30 to 925 ppm).

An alternative way to improve the combustion stability is to add hydrogen to the fuel. The filled triangle indicates a test point at λ = 1.6 with hydrogen addition. This point has significantly reduced cycle-by-cycle variations as well as low NO_x emissions.

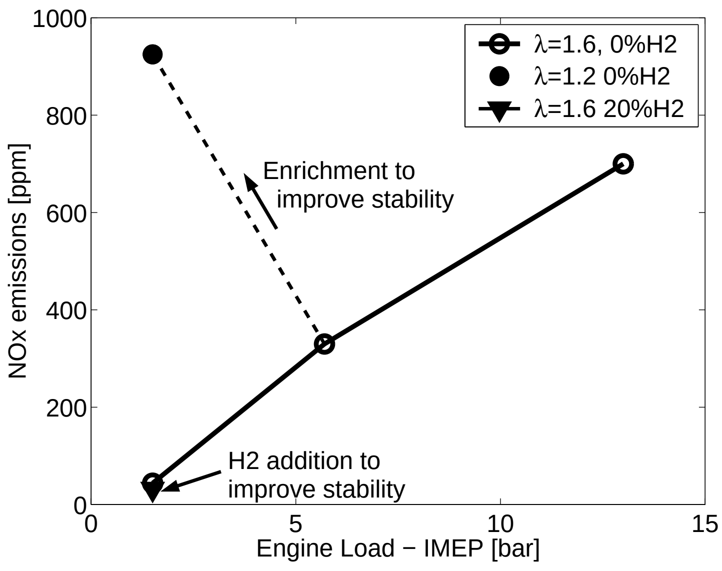


Figure 33. NO_x emissions versus engine load.

5 SUMMARY AND CONCLUSIONS

This experimental investigation was set up to evaluate the effects of hydrogen addition to a lean burn natural gas SI engine. The effect of hydrogen addition was tested for two different combustion chambers. The so called Turbine combustion chamber generates low in-cylinder turbulence and hence has relatively slow burning characteristics, while the so called Quartette combustion chamber generates high turbulence and hence gives fast burning characteristics.

Generally, at light loads for the lean burn engine, stability becomes poor. To avoid this problem, these engines are often operated with fuel enrichment at these light load conditions, which raise NO_x emissions severely. It seems possible that one way to replace this method is by adding hydrogen instead. That way it is possible to keep the engine lean burning with much reduced NO_x emissions.

Addition of hydrogen to the natural gas increases the burn rate and extends the lean limit. Hydrogen addition lowers HC emissions and increases NO_x emissions. However, the increased burn rate makes it possible to run leaner, which lowers NO_x emissions. It indicates that a better trade off between NO_x and HC can be obtained with the help of hydrogen. Results close to lean limit shows very low NO_x emission. The drive ability of a lean burn engine fueled with hydrogen blended natural gas is probably better since the lean burn stability is improved. This could help keeping overall emissions at a minimum for city buses etc.

In comparison, the slower combustion chamber (Turbine) was more affected by the hydrogen addition than the faster (Quartette). Quartette has fast burning characteristics even without hydrogen addition.

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