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Hydrogen–natural gas blends fuelling passenger car engines: Combustion, emissions and well-to-wheels assessment

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ABSTRACT

In this study, a state of the art passenger car natural gas engine was optimized for hydrogen–natural gas mixtures and high exhaust gas recirculation (EGR) rates in the major part of the engine map. The investigations involved stoichiometric combustion. With optimal combinations of spark timing and EGR rate, the achievements are efficiency increase with substantially lower engine-out NO_x while total unburned hydrocarbons or CO-engine-out emissions are only modestly affected. The efficiency is increased by 3% in the low load and by more than 5% in the medium-load domain. Increasing hydrogen content of the fuel accelerates combustion leading to the efficiency improvements. Combustion analysis showed that the increasing burning rates mainly affected the initial combustion phase (duration for 5% mass-fraction burned). Nevertheless, increase of the hydrogen fraction in the fuel over a certain threshold did not result in any efficiency increase in the medium loads. Loss analysis identified high wall heat losses as the main reason. Dedicated combustion chamber design may be able to avoid these losses and lead to additional efficiency benefits. Well-to-wheel analysis revealed paths for the production of the fuel blends still having overall energy requirements slightly higher than a diesel benchmark vehicle but reducing by 7% overall green house gas emissions.

1. Introduction

In the recent past, our laboratories have demonstrated the potential of a natural gas-optimized engine with the Clean Engine Vehicle (CEV) project. The vehicle used was a production line small sedan (model year 2000) with a curb weight of 1020 kg. The achieved goal was 30% lower CO_2 emissions than the gasoline vehicle while staying in compliance with Euro-4 as well as SULEV emission limits [1]. These low-pollutant levels have been reached mainly through the use of optimized three-way catalysts (TWCs) and improved engine management functions.

In a further development step, an initial study of the effects of CNG–hydrogen blends has been performed [2]. CNG–H_2 blends have the potential to enhance the advantages of the CNG fuel by means of the H_2 combustion characteristics. H_2 combustion is characterized by wide ignition ranges and high flame speed not to mention the self-evident absence of carbon. On the other side, CNG–H_2 blends avoid critical issues associated to pure hydrogen combustion in internal combustion engines, i.e., the high volumetric efficiency losses, the low knocking resistance and the low vehicle range.

In a further perspective, the use of CNG–H_2 blends has the potential to form an intermediate step towards pure

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Nomenclature

Abbreviations

bmep	p_{cyl}
b.s.f.c.	brake-specific fuel consumption
CCGT	combined cycle gas turbine
CNG	compressed natural gas
CR	compression ratio
DoE	design of experiments
EGR	exhaust gas recirculation rate
EVO	exhaust valve opening
GHG	greenhouse gas emissions
HCCI	homogeneous charge compression ignition
IVC	inlet valve closing
SI	spark ignition
ST	spark timing
TDC	top dead center
THC	total unburned hydrocarbons
TTW	tank-to-wheel
TWC	three-way catalyst
WTT	well-to-tank
WTW	well-to-wheel

Latin characters

B	engine bore
s_p	mean piston speed
m_f	fuel mass introduced in the cylinder per engine stroke
cylinder pressure	mean indicated pressure
p_{mi}	mean indicated pressure
$p_{mi,gx}$	brake mean indicated pressure through gas exchange
Q_b	energy introduced by the fuel
Q_{ic}	incomplete combustion losses
Q_{rc}	real combustion losses
Q_{wh}	cylinder wall heat losses
T_{cyl}	mean cylinder Temperature
V_c	cylinder volume at TDC
V_D	cylinder displacement volume
W_{gx}	gas exchange work

Greek characters

α_w	heat transfer coefficient, cylinder gas to walls
γ	ratio of the specific heats of the working fluid
η_{cv}	fuel conversion efficiency of the constant volume process;
η_f	effective (brake) fuel conversion efficiency
η_i	indicated fuel conversion efficiency
η_{rc}	real combustion efficiency
φ	crank angle

hydrogen-based mobility. Using the hydrogen by-product of chemical farms and of biological waste fermentation sites (numerous facilities in operation in Switzerland) in combination with existing, advanced CNG engine technology, the development of the hydrogen infrastructure (production and distribution) as well as dedicated regulations can be promoted.

In the last years, some research activity has been performed concerning the use of CNG–H_2 blends in internal combustion engines. Ref. [3] provides a comprehensive overview of the results so far. The reported results agree in decreased HC and CO, increased NO_x emissions and increased fuel conversion efficiency at modest substitution levels of CNG by H_2. In addition, all reported results demonstrated the ability to achieve leaner combustion with increasing hydrogen amount. The studies referred to in Ref. [3] have been performed on a variety of engines, most of them being research engines, some of them derived from gasoline production engines.

In Ref. [4], the studies mainly focused on full-load behaviour of a 0.61lt displacement, one-cylinder, low-compression ratio research engine. The reported thermal efficiency for stoichiometric operation increased with increasing fraction of hydrogen in the fuel and decreased when the hydrogen fraction was increased beyond the 20 vol.% level. Possible explanation was the higher in-cylinder temperature causing higher wall heat losses as well as the knock-limited spark timing advance. In Ref. [5] similar observations have been reported. The full load indicated power output of the engine increased with increasing hydrogen fraction as long as the H_2 portion was roughly below 20 vol.%. Bade Shrestha and Karim [5] ascribed this behaviour, stronger pronounced at high compression ratios, to the lower volumetric heating value of the fuel blend. However, several studies agree that for stoichiometric combustion only moderate hydrogen enrichment of natural gas results in high efficiencies [3]. Very low H_2 fractions in the fuel do not make enough use of the H_2 potential while too high H_2 fractions exhibit combustion characteristics less suitable for prevalent methane combustion chambers.

In comparison, Ref. [6] identified the CNG–H_2 fuel blend with 30 vol.% H_2 as the one leading to optimal efficiency. Major advantages have been identified in the emissions. The reduction of HC and CO was accompanied by a significant reduction of more than 10% in CO_2 (assuming CO_2-free hydrogen production). Reduced vehicle range by 20% and improved cylinder head cooling requirements have been the disadvantages identified. A demo fleet of urban busses running on CNG–H_2 fuel blends has been announced for the first months of 2008.

In Ref. [7] a systematic investigation of the emissions at different engine speeds, loads and lean mixtures has been performed. The engine emissions have been compared with pure CNG vs. 15 vol.% CNG–H_2 blend. The results obtained demonstrated the potential of the methane–hydrogen mixture in reducing the exhaust concentrations of regulated pollutants while increasing the efficiency. The engine used was a modified gasoline production engine with a quite low compression ration of 8.8:1. A former gasoline engine modified for natural gas operation with a similar low compression ratio was used in Ref. [8]. Again the emissions have been the focus, while a series of fuel blends have been tested. Reaping the benefits of lean operation and keeping the emissions in accordance to the regulation without expensive after-treatment devices resulted in a difficult task. For reducing hydrocarbon emissions λ had to be less than 1.3, whereas NO_x reduction required λ values of at least 1.5. Similarly, Ref. [9] identified HC and CO emissions advantages and NO_x disadvantages in a 10:1 four-cylinder engine compression ratio. The engine's efficiency increased with increasing hydrogen fraction in the fluid blend, the gain, though, was incremental at hydrogen fractions exceeding 20%.

While in Ref. [10] the effects of hydrogen addition to CNG have been evaluated for a stationary power generation unit, in Ref. [11] the hydrogen addition effects have been studied for homogeneous charge compression ignited (HCCI) engine operation. Refs. [12,13] focused on the direct injection of hydrogen-based fuels in the cylinder.

Several studies, to mention Refs. [14–17], showed that addition of reformer gas (mainly consisting of H_2 and CO) to gasoline allows stable operation of the engine under extreme conditions, such as ultra–lean air–fuel mixtures, or very high rates of exhaust gas recirculation (EGR). In addition, they observed crucial advantages during cold start and warm up, reducing substantially unburned hydrocarbons. The advantages have been identified on the impact of the hydrogen compound of the fuel on the combustion process, ascribed to its extremely high reactivity. The hydrogen molecules promptly dissociate, providing active radicals that speed up the methane dissociation.

However, very little information is available concerning combustion analysis and characteristics of CNG–H_2 fuel blends in specific optimized state of the art natural gas engines. Even more scarce is information concerning the combustion of H_2–CNG blends with high EGR. To our knowledge, the only study available is Ref. [18]. Nevertheless the focus of Ref. [18] was combustion stabilization with varying H_2 fuel contents as well as EGR rates. On the other hand, some detailed information concerning combustion of pure H_2 with EGR is available [19].

In a first step [20], we have examined the effects of EGR in a CNG optimized engine with pure CNG fuel. Only a modest part of the EGR benefits could be reaped, due to the ''slow'' pancake-shaped combustion chamber. In Ref. [2], preliminary results have been published confronting efficiency and emissions at one engine speed and load fuelled by pure CNG and two H_2–CNG fuel blends. The amount of EGR and the spark timing have been obtained for optimal fuel conversion efficiency. In the low load examined, the combination of the optimal spark timing and EGR amount with the fuel with the highest hydrogen amount (15 vol.%) resulted in an efficiency increase of 3%, engine-out NO_x decrease of approximately 50%, without affecting CO and unburned hydrocarbons.

In order to gain further understanding, the combustion of H_2–CNG fuel blends without EGR has been studied separately [21]. Apart from the increasing NO_x emission (as expected without EGR), investigations in Ref. [21] showed a new efficiency characteristic at medium loads by increasing the hydrogen fraction in the fuel. At low loads increasing hydrogen fraction lead to increasing engine efficiency [2,21]. At medium loads though, the increase of the hydrogen fraction beyond 10 vol.% resulted in efficiency losses [21]. The present article aims in analyzing and explaining this behaviour also taking into account the effects of EGR. Further, we present a summary of combustion performance and emissions characteristics of H_2–CNG fuel blends in a state of the art, CNG optimized, spark ignition passenger car engine.

Improved engine efficiency though, may be counterbalanced by higher energy effort during the fuel blend production. In addition, CO_2 emission considerations should also take into account the fuel blend production path. In order to investigate these topics, as well as to identify hydrogen production routes leading to an overall energetic as well as green house gas advantage we present a comprehensive well-to wheel analysis.

2. Engine and fuels

2.1. *Engine and testing*

The original engine was a Volkswagen naturally aspirated four-cylinder gasoline engine with a displacement of 1.0 l and an output of 37 kW at 5000 rpm and 86 Nm at 3000 rpm. For optimal CNG fuelling, the compression ratio was increased to 13.5 and turbocharging was implemented in order to surpass the original (gasoline) engine performances [1]. The engine was operated strictly stoichiometric and used one pre-TWC and one main TWC [22]. As reported in Ref. [1], pure CNG operation reduced the CO_2 emission by 30% with respect to the original gasoline configuration, while the vehicle reached the Euro-4 and SULEV emission standards. Cylinder pressure data have been processed by a transient recorder and analyzed by WEG, a heat release software package developed at ETH Zurich [23]. Engine-out exhaust gas was analyzed by a Horiba MEXA-9200DF analyzer.

The EGR in modern SI-engines is used for mainly reducing engine-out NO_x emissions. In addition, EGR has the potential for fuel consumption improvement due to three factors: (1) reduced pumping work, since for the same brake load less throttling is required (higher amount of inert gas as well as higher temperature of the intake gas [24]), (2) reduced heat loss to the walls because of reduced burn gas temperature and (3) reduced dissociation in the high temperature burned gases. According to Refs. [25,26], the fuel conversion efficiency at part load can be increased up to 4.5% when using the appropriate EGR amount. However, EGR also reduces the combustion rate that may fully offset the beneficial effects leading to elongated combustion. In addition, the unburned hydrocarbon emissions increase with increasing EGR.

2.2. *Fuels*

The compressed natural gas (CNG) used consisted of 99.5 vol.% methane. The hydrogen had a 99.995 vol.% purity. The mixing has been performed by the supplier prior to filling the fuels in 200 bar bottles. Table 1 summarizes the most important fuel properties. It should be mentioned that because of the different energy content of the mixtures, we do not consider a direct comparison of b.s.f.c. engine data as appropriate. Instead of fuel consumption, we strictly report fuel conversion efficiencies. For further details, we refer Ref. [2].

3. Experimental procedure

All results without EGR have been obtained by measurements on the engine dynamometer. Principles of Design of

Table 1 – Main properties of the fuels investigated

	CH_4	5 vol.% H_2	10 vol.% H_2	15 vol.% H_2
Volumetric-fraction H_2 (vol.%)	0	5	10	15
Volumetric-fraction CH_4 (vol.%)	100	95	90	85
Mass-fraction H_2 (mass%)	0	0.705	1.377	2.169
Mass-fraction CH_4 (mass%)	100	99.29	98.623	97.831
Energy-substitution H_2 (%)	0	1.652	3.242	5.053
Stoichiometric air ratio	17.19	17.23	17.26	17.284
Low heating value (MJ/kg)	50.02	50.492	50.964	51.519

Experiments (DoE) have been applied for optimizing combustion with EGR. Engine dynamometer tests with specific parameter settings comprise the first step in DoE optimization. Based on the measurement results, models are built for each engine response factor of interest. The models then retrieve the parameters for optimal engine performance. A second series of verification measurements can be performed using the predicted parameters. Specific DoE applications in engine development and optimization are currently an interesting topic [19,27,28]. The interested reader is referred to Ref. [2] for detailed information concerning the procedure chosen in this article.

4. Thermodynamic analysis

Engine energy balance yields the distribution of the fuel energy to the various outputs and is a wide-applied tool mainly used for component layout. Availability (exergy) analysis is more complex and leads to the identification of irreversibilities [29]. Loss analysis aims in identifying, at least theoretically, avoidable losses of the real engine process in respect to an ideal process, which has already taken into account the restrictions of thermodynamics 2nd law. Some recent applications are known [30,31]. However, we are not aware of any loss analysis reported for engines fuelled by hydrogen–natural gas mixtures, apart from our initial work [2].

On the basis of the ideal process, the premixed engine combustion is described by the constant volume process:

$$\eta_{cv} = 1 - CR^{1-\gamma} \tag{1}$$

Multiplying the energy introduced by the fuel Q_b with η_{cv} leads to the part of Q_b, which could be theoretically fully transformed to work at the piston, i.e., to the indicated work. In real engines a series of additional losses is encountered. A brief description of the additional losses follows, for more details the interested reader is referred to Refs. [2,21,30,31].

- Incomplete combustion losses, Q_{ic}: For computing incomplete combustion losses, the measured concentrations of CO and THC in the raw exhaust gas with their associated lower heating values have been used. In addition, we accounted for hydrogen in the exhaust (concentrations computed according to Ref. [23]).
- Real combustion losses, Q_{rc}: The constant volume process assumes instant combustion while the piston is at TDC. In reality, combustion has a finite duration, detailed analysis in Ref. [2].
- Cylinder wall heat losses, Q_{wh}: The computation of the wall heat losses was also performed by WEG [23] using a Woschni [24] correlation for the heat transfer coefficient α_w:

$$\alpha_w \approx s_p^{0.8} \cdot p_{cyl}^{0.8} \cdot B^{-0.2} \cdot T_{cyl}^{-0.53} \tag{2}$$

- Pumping work losses, W_{gx}: The computation of the mean indicated pressure during gas exchange can be achieved by integrating the pressure signal over the associated cycle path [2].

Subtracting all the above-described losses to the energy introduced by the fuel Q_b multiplied by $(1-\eta_{cv})$ results in an amount slightly lower than the indicated work:

$$\eta_{cv} \cdot Q_b - (Q_{ic} + Q_{rc} + Q_{wh} + W_{gx}) > p_{mi} \cdot V_D = \oint p_{cyl} \cdot dV \tag{3}$$

The difference can be attributed to the assumption of pure air as working medium as well as to blow-by losses.

5. Results and discussion

5.1. *Effects of hydrogen–CNG blends with EGR, typical medium engine load*

As typical for the combustion characteristics at medium loads, the operating point at 2000 rpm and 4 bar bmep was chosen. Fig. 1 displays the predicted brake fuel conversion

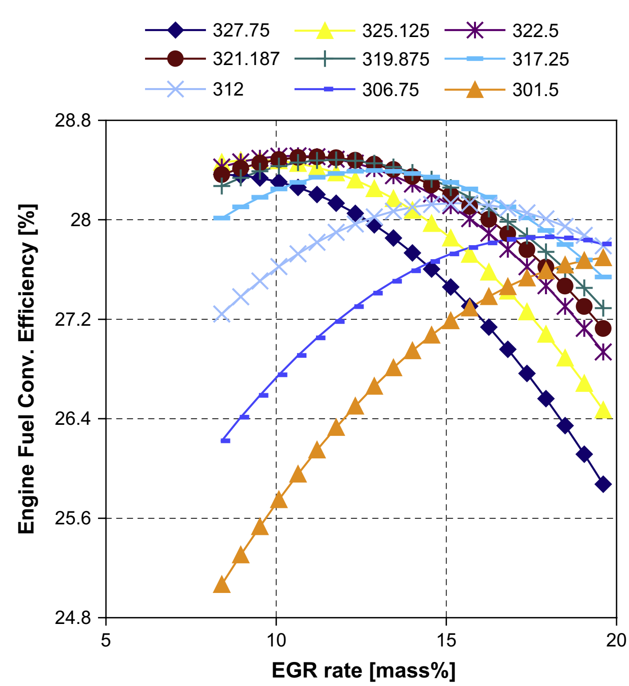

Fig. 1 – Brake fuel conversion efficiency vs. EGR for different spark timings as predicted by the DoE modelling at 2000 rpm and 4 bar bmep using the 10 vol.% H_2–CNG fuel blend.

efficiency vs. EGR rate and selected spark timings with the 10 vol.% H_2–CNG fuel blend. DoE modelling provides the possibility of detailed exploration of the parameter space (the spark timing selection in Fig. 1 is arbitrary and can be chosen upon demand; other views of the parameter–response factor space are also available). The main insight from Fig. 1 is that spark timings from the range of 320 to 325° crankangle lead to highest engine efficiency. The highest efficiency is reached with spark timing at approximately 323° crankangle and moderate EGR (10–12 mass%). Interestingly, the EGR-sensitivity of the optimal efficiency is rather low; the EGR amount can vary from 8 to 15% without detrimental effects on the engine efficiency. This insight becomes important when keeping in mind that the controlling of exact EGR amounts during real engine operation is complex and still not very accurate. Spark timings nearer to TDC reach engine efficiency maxima with less EGR, whereas those further from TDC reach their associated efficiency maxima with more EGR. Nevertheless, it is important to notice that this behaviour is typical only when hydrogen is in the fuel. With pure methane optimal engine efficiencies are decreasing with increasing EGR and are much more sensitive to EGR quantity [2,21].

The comparison of the results from Fig. 1 (corresponding to a medium engine load) with the results from Ref. [2] at lower loads underlines the increased spark timing sensitivity of the maximal achievable engine efficiency. In low loads, we could identify [2] two domains of spark timing and EGR leading to roughly the same maximal efficiency: either low EGR with late spark timing or high EGR with earlier spark timing. With increasing load the second maximum with high EGR is not pronounced.

Combinations of spark timing and EGR quantity have strong impact on the engine-out emissions. Fig. 2 displays the impact of the same engine parameters from Fig. 1 on the engine-out NO_x emission (for the 10 vol.% H_2–CNG fuel blend). Not surprisingly the results are typical for premixed engine combustion systems: early spark timings increase, while increasing EGR quantities decrease raw NO_x emissions. Hydrogen enrichment of the fuel does not affect these well-known trade-offs, although as described further below hydrogen enrichment helps reaching more favourable trade-offs.

The enveloping curve in Fig. 1 can provide all maximal achievable efficiencies associated with the EGR quantities (for each EGR the efficiency optimal spark timing is chosen). In parallel, Fig. 2 delivers the corresponding engine-out NO_x. Similarly, efficiencies and emission levels can be obtained for each of the examined fuel blends. Fig. 3 shows the maximal achievable engine efficiencies with each of the examined fuel blends. Already modest hydrogen addition to the fuel (5 vol.% H_2) leads to significant efficiency increase. With the 10 vol.% H_2–CNG fuel blend the highest efficiency is reached, while further increasing the hydrogen fraction in the fuel leads even to efficiency losses. The issue of the lower efficiency reached with the hydrogen richest fuel is worth of further scrutiny. Experiments performed without EGR revealed a similar behaviour [21]; also without EGR the hydrogen richest fuel leads to decreased efficiency in all higher load engine points. Loss analysis, discussed further below, provides some explanation. On the other side, in combination with very high EGR (over 17%) the hydrogen richest fuel is able to accelerate combustion and achieve high efficiencies, nevertheless lower efficiencies than with low EGR.

Fig. 4 displays the spark timings required to reach the efficiency optimal curves of Fig. 3. With increasing hydrogen fraction in the fuel, optimal spark timings are less advanced from TDC, giving evidence for substantially accelerated

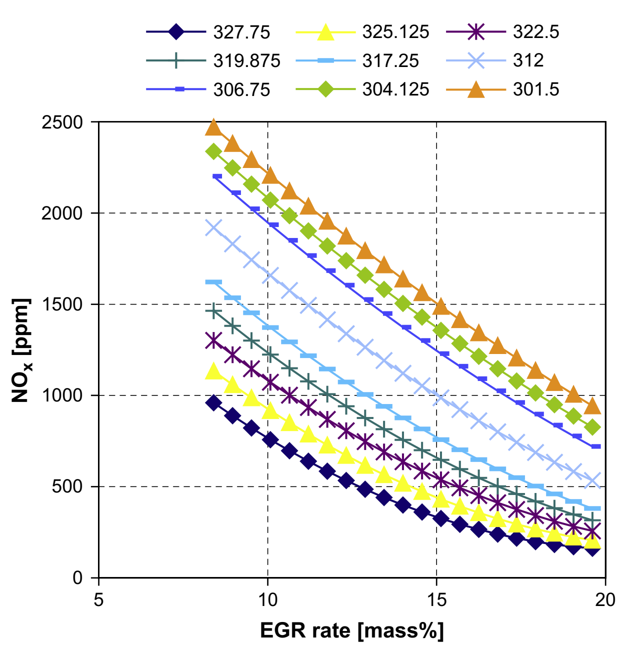

Fig. 2 – Engine-out NO_x emissions vs. EGR for different spark timings as predicted by the DoE modelling at 2000 rpm and 4 bar bmep using the 10 vol.% H_2–CNG fuel blend.

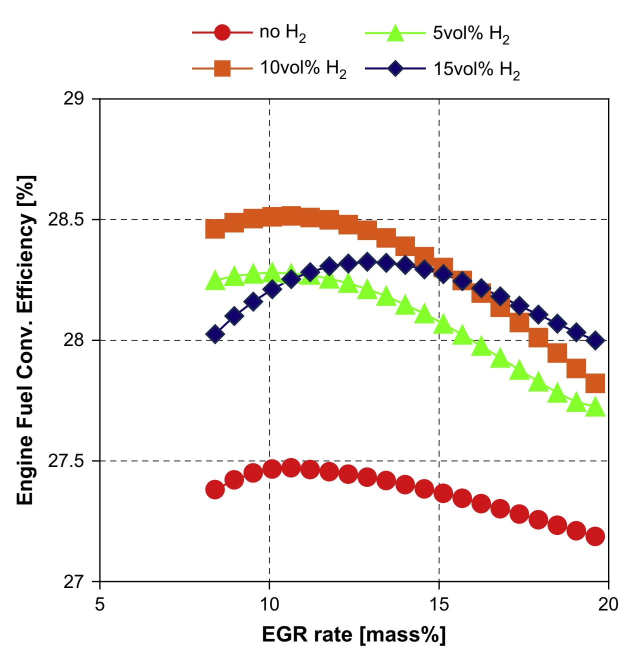

Fig. 3 – Brake fuel conversion efficiency vs. EGR quantity at 2000 rpm and 4 bar bmep engine load (each point with efficiency optimal spark timing).

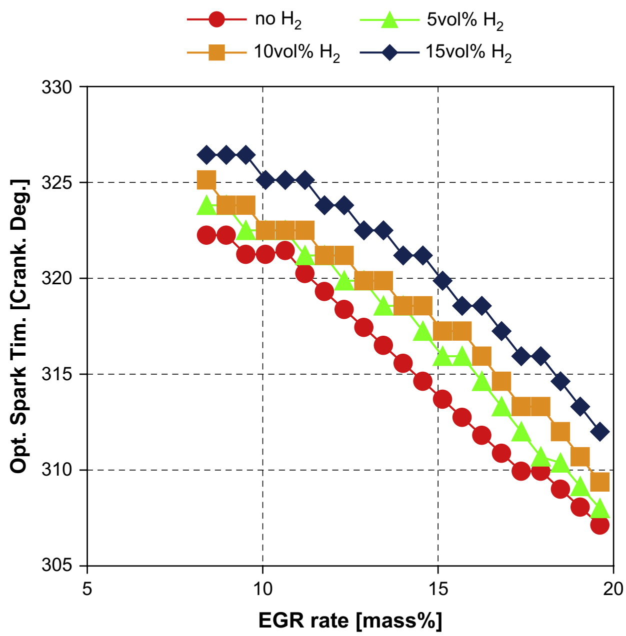

Fig. 4 – Efficiency optimal spark timing vs. EGR quantity at 2000 rpm and 4 bar bmep engine load.

combustion. With increasing EGR combustion slows down, at least partly, increasing the optimal spark timing advance.

In Fig. 5 we plot the engine-out NO_x emission with the engine operating parameters for the optimal efficiencies (Fig. 3). As expected, with increasing EGR the NO_x emission decreases sharply. The NO_x emissions with the 5 vol.% H_2–CNG fuel blend is the lowest, being almost 15% lower in respect to the emissions with pure CNG as fuel and optimal EGR. The NO_x emissions with the hydrogen richer fuels lie in between. As already discussed (Fig. 3), the associated engine efficiencies reached with hydrogen blending are significantly higher. Hydrogen addition in the fuel results in more favourable efficiency-raw NO_x trade-off, the related argumentation can be found in the pressure, temperature and heat release analysis [2]. Simultaneously, hydrogen in the fuel reduces unburned hydrocarbons (THC). According to Fig. 5, the reduction lies between 6 and 20% depending on the fuel and the amount of EGR under consideration.

5.2. *Loss and combustion analysis, typical medium engine load*

Loss analysis provides additional understanding of the hydrogen–CNG fuel effects on the combustion process (Fig. 7). For all four examined fuels the efficiency of the associated constant volume process is, of course, the same, depending only on the compression ratio of the engine and the thermodynamic properties of air (assumed as the working medium, the error by this assumption comprises one part of the ''real gas, blow-by losses'', also displayed in Fig. 8). For all examined fuels, the efficiency of the constant volume process is the same and amounts 64.16%. In this sense, the non-avoidable losses due to the second law of thermodynamics amount to 35.84%.

Incomplete combustion losses decrease with increasing hydrogen fraction in the fuel. Beyond the 10 vol.% hydrogen fraction the incomplete combustion losses increase again. This increase cannot be supported only by the THC emissions, since the THC emissions at optimal EGR values are roughly the same (Fig. 6. Neither CO emissions show a significant rise. H_2 emissions increase, due to higher temperatures in the combustion chamber during the main combustion phase and stronger H_2 dissociation. The increase in the incomplete

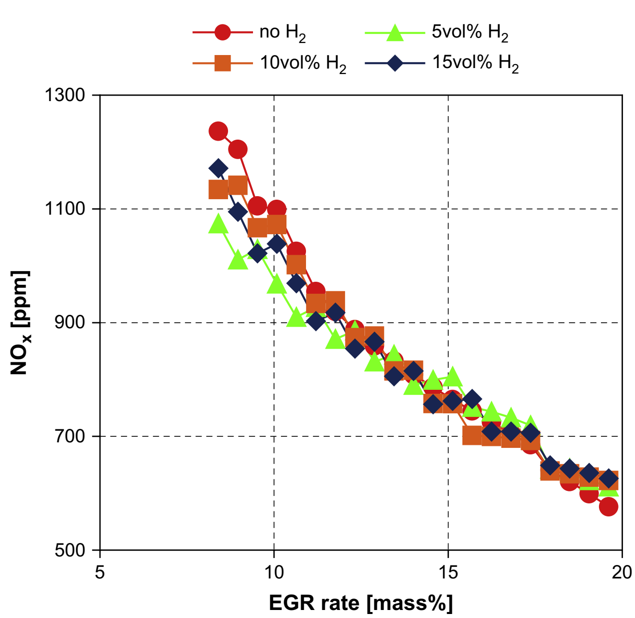

Fig. 5 – Engine-out NO_x vs. EGR quantity at 2000 rpm and 4 bar bmep engine load (each point with efficiency optimal spark timing).

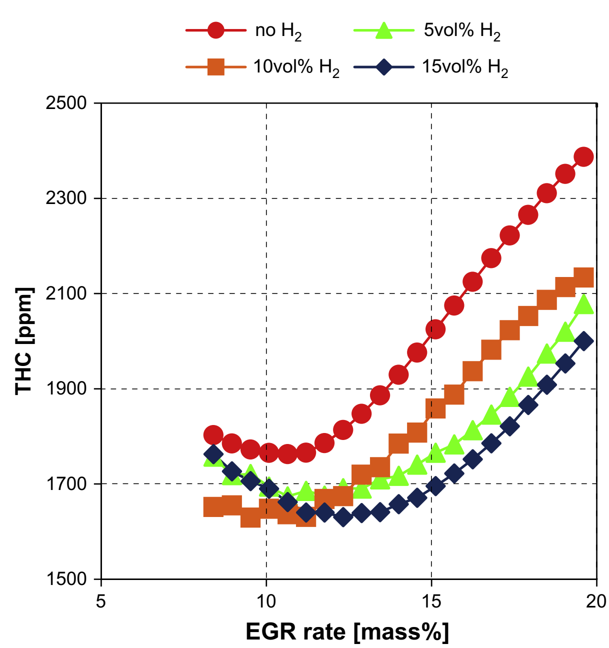

Fig. 6 – Engine-out THC vs. EGR quantity at 2000 rpm and 4 bar bmep engine load (each point with efficiency optimal spark timing).

combustion losses with the hydrogen richest fuel blend was also observed at low load engine operating points [2]. Nevertheless, the incomplete combustion losses are still significantly lower than with pure CNG fuel.

Real combustion losses also decrease with increasing hydrogen fraction in the fuel. Real combustion losses are due to the finite duration of combustion and are only zero in the case of instant combustion at TDC. Decreasing real combustion losses are the evidence of accelerated combustion. Following the evolution of the real combustion losses through the different fuel blends another fact can be observed. While the first 5 vol.% hydrogen addition to CNG results in 10% decrease of the real combustion losses (from 5.5 to 4.0%), the additional 5% accelerates combustion even more and reduces the associated losses for 12.5%. The hydrogen richest fuel blend tested, results also in combustion acceleration, but a weak one. Remarkably, this was also the case in the low loads [2], as well as in all analyzed cases without EGR [21]. An explanation can be provided when taking into account the wall heat losses. The increasing heat losses with increasing hydrogen fraction in the fuel are strongly pronounced in all examined medium loads and are significantly higher in respect to the heat losses with pure CNG fuelling. Similar observations have been made at lower loads, although the increase was not as sharp [2].

There are mainly two reasons for the increased heat losses. Apart from the higher in-cylinder temperatures, the presence of hydrogen in the fuel decreases the flame quenching distance. The significantly warmer flame front approaches closer to the cylinder wall. Dedicated combustion chamber design may reduce the wall heat losses and improve even more the engine efficiency. In a first step, a lowering of the compression ratio may also reduce the heat losses (the flame reaches in a later stage the walls). Ref. [6] reported of increasing engine efficiency with increasing hydrogen fraction in the fuel without finding a reversal of this trend. This may be due to the lower compression ratio and the dedicated combustion chamber design.

The remaining parts of the histograms in Fig. 7 display the proportions of the losses as well as the useful work of the crankshaft. They do not provide any new or additional information. While the fuel conversion efficiency corresponds to the data of Fig. 4, mechanical efficiency is similar for the examined fuel blends (as expected). The gas-exchange losses are anyway low at these loads and do not reveal any hydrogen-related differences as in the lower loads [21]. Finally, the relatively small and positive real gas, blow-by losses are rather a confirmation for the results since, as already discussed in Section 4 they are not directly quantified but exhibit residual values.

Deeper insights in the combustion properties of the examined fuels can be achieved when comparing the time scales (most suitably expressed in crankangle degrees) required for burning a certain percentage of the cylinder mass. In Fig. 8 we show the optimal spark timings as well as the elapsed period thereafter for burning 5% (MFB5), 50% (MFB50) and 90% (MFB90) of the cylinder gases.

Spark timing depends, of course, on flame speed. It is obvious that the fuels containing H_2 accelerate the combustion process requiring less spark timing advance than the pure CNG fuel for achieving optimal efficiency (also Fig. 4). The highest spark timing difference in respect to pure CNG is associated with the hydrogen richest fuel, reaching 8 crankangle degrees. This is in good agreement with the findings of other studies; in particular with the results in Ref. [4], where hydrogen addition up to 60 vol.% decreases spark timing by

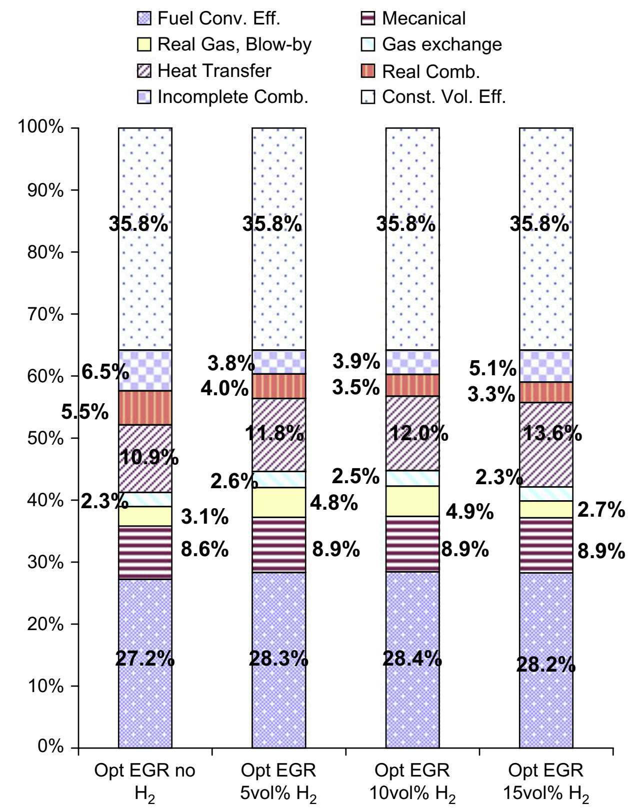

Fig. 7 – Loss analysis with efficiency optimal EGR and spark timing at 2000 rpm and 4 bar bmep engine load.

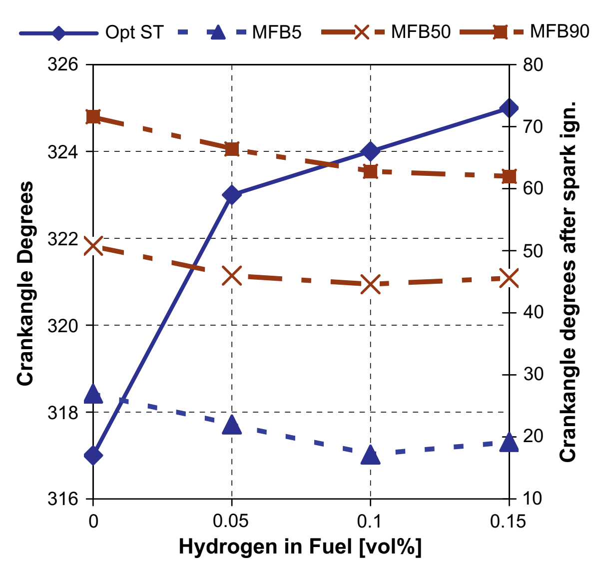

Fig. 8 – Optimal spark timing (Opt ST) and duration for 5, 50 and 90% mass-fraction burned (MFB5-90).

6–14 crankangle degrees at identical equivalence ratios (keeping in mind that in Ref. [4] a larger engine was used at substantially lower engine speeds).

The initial combustion phase (MFB5) is shortened by more than 30% with the 10 vol.% H_2–CNG fuel blend. Interestingly, the lower amount of hydrogen in the fuel (5 vol.% H_2–CNG) resulted in approximately a halved acceleration of the initial combustion phase, whereas the hydrogen richest fuel did not shorten the MFB5 in respect to the 10 vol.% H_2–CNG at all. In contrast to the initial combustion phase (MFB5), the combustion acceleration in the following two phases is rather modest. The 10 vol.% H_2–CNG fuel blend shortens the main combustion part (MFB50) by around 7%. The acceleration of the entire combustion, as expressed by the MFB90, is similar to the shortening the MFB50 part. Noteworthy is that the optimal efficiency spark timings have led to such combustion placements that MFB50 was reached at around 368–371°. CA regardless of the fuel, the engine speed or the load.

We obtained very similar results in the examination of the fuel blend impact on the combustion process without EGR [21], merely all related combustion durations (MFB 5–90) have been around 5–10 crankangle degrees shorter. These observations are in good agreement with findings of Refs. [14–17] where hydrogen–gasoline mixtures have been examined.

5.3. *Overview of the effects of hydrogen–CNG blends at selected engine loads and speeds*

Fig. 9 is summarizing the major effects for typical low load operating points (2 bar bmep). For improved resolution, engine efficiencies at 2000 rpm correspond to the left y-axis, whereas those at 3000 rpm correspond to the right y-axis.

According to the predictions (hollow triangles) at 2000 rpm with pure CNG fuelling, EGR should only slightly increase the efficiency of the engine. The verification measurements showed even a slight deterioration (black triangles). Measurements and predictions show that in combination with the "slow" pancake type combustion chamber, if any potential of EGR in improving the efficiency exists, it is very thin. The addition of hydrogen in the fuel increases the fuel conversion efficiency. With the 5 vol.% H_2 fuel the increase is though negligible, instead, using the 10 vol.% H_2 fuel the increase is substantial. The highest fuel conversion efficiency of almost 20.8% is obtained with the H_2 richest fuel (15 vol.% H_2). Here both measurements (black triangles) as well as predictions (hollow triangles) agree. In comparison, the highest fuel conversion efficiency without EGR and no H_2 is 20.15%, whereas the highest efficiency with the 15 vol.% H_2 fuel and no EGR is 20.45%. So the increase in maximal efficiency without EGR caused by the H_2 richest fuel is 1.5%, while adding optimal EGR increases the maximal efficiency for additional 1.5%. Similar efficiency increases have been found in Refs. [9,10] for stoichiometric combustion ($\lambda = 1$) with and without EGR although the engines are not directly comparable. Even similar efficiency gains have been reported when partly substituting gasoline with H_2, while using EGR [14]. As expected, the maximal efficiency for the same load at higher rpm is lower. The tendencies at 3000 rpm resemble almost exactly those of 2000 rpm.

In medium loads the engine efficiency exhibits different behaviour, as already discussed in Sections 5.1 and 5.2. Fig. 10 provides a comprehensive summary (4 bar bmep). Already the introduction of the correct amount of EGR leads to a 1% efficiency increase in respect to the engine baseline without any hydrogen addition in the fuel. The engine optimization with the 10 vol.% H_2 fuel blend leads to an efficiency increase of 2.5% without EGR. The addition of the optimized EGR amount results in an efficiency increase of 5.2%. Very similar tendencies have been found for 3000 rpm.

Summarizing the results concerning the efficiency improvement over the most important part of the engine map by modest hydrogen addition to CNG, it can be concluded that improvements lie around 2%. In combination with optimized EGR the efficiency increase is enhanced by additional 2% leading to an overall efficiency improvement of at least 4%.

Figs. 11 and 12 show the associated engine-out NO_x emissions. In respect to the baseline (no EGR pure CNG fuelling), the NO_x emissions decreases for at least 30% at both loads. This is though primarily the effect of EGR. As can be seen in both figures the addition of hydrogen in the fuel is accompanied by an NO_x increase. Engine-out NO_x without EGR was studied in detail in our previous work [21]. Hydrogen in the fuel with the appropriate EGR amount is capable of improving the efficiency and significantly reducing NO_x.

A somehow opposite situation can be observed with the behaviour of the total unburnt hydrocarbons (THC) (Figs. 13 and 14). While hydrogen addition in the fuel reduces engine-out THCs, EGR increases them. The latter is responsible for the substantial THC increase at medium loads (Fig. 14). The THC increase with EGR reaches some 15% in respect to the no EGR, pure CNG baseline. The THC reducing impact of the hydrogen in the fuel is also evident when keeping in mind that without hydrogen but with EGR the corresponding THC increase would be more than 20%. Nevertheless in lower loads things are not

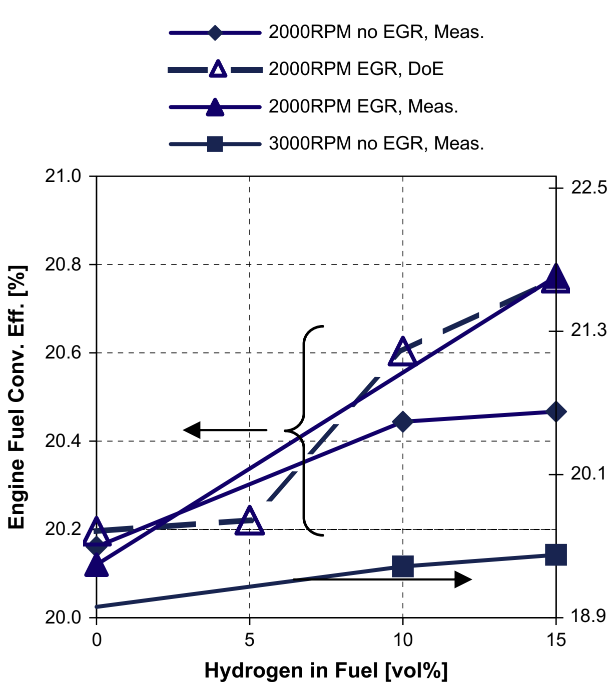

Fig. 9 – Maximal achieved fuel conversion efficiency with the tested fuel blends, 2 bar bmep load.

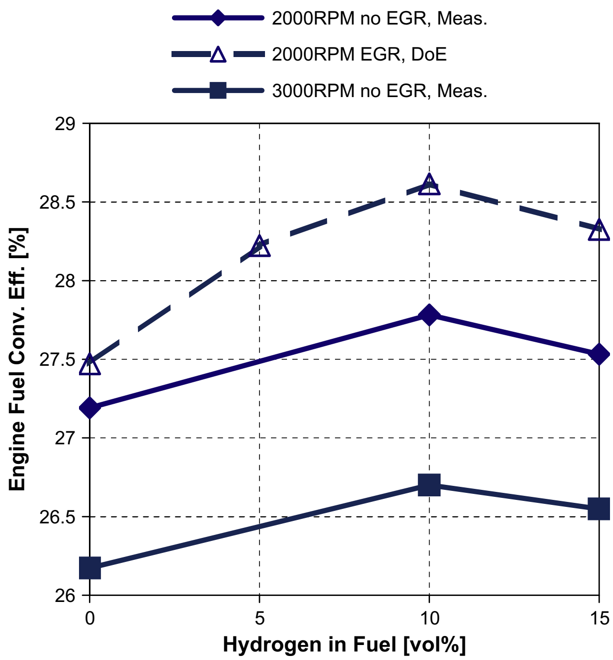

Fig. 10 – Maximal achieved fuel conversion efficiency with the tested fuel blends, 4 bar bmep load.

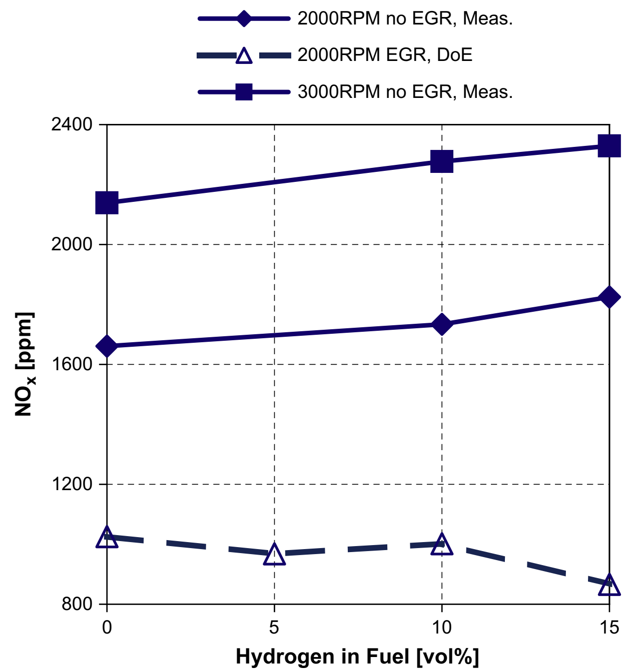

Fig. 12 – Engine-out NO_x with the engine operating parameters leading to the maximal efficiencies 4 bar bmep engine load.

as bad, since EGR and hydrogen together result in even a slight reduction of engine-out THCs in respect to the no EGR pure CNG baseline. The situation is favourable since with increasing load the exhaust temperature increases and THCs oxidation improves in the catalyst.

A last comment concerns the engine-out CO emissions (Fig. 15). It seems that hydrogen in the fuel as well as EGR have a beneficial effect on the CO emissions.

5.4. *Well-to-wheels assessment*

The efficiency and pollutant emission benefits described in the former section raise the question of their sustainability through the whole chain of fuel production, supply and use. Is

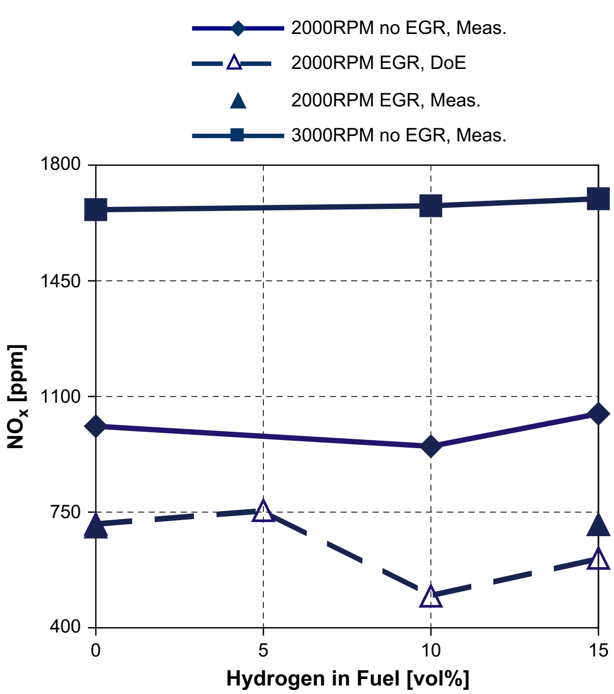

Fig. 11 – Engine-out NO_x with the engine operating parameters leading to the maximal efficiencies 2 bar bmep engine load.

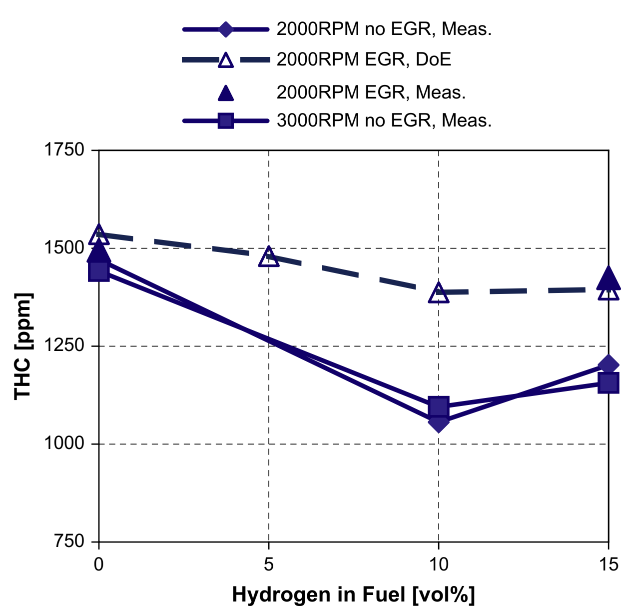

Fig. 13 – Engine-out THC with the engine operating parameters leading to the maximal efficiencies 2 bar bmep engine load.

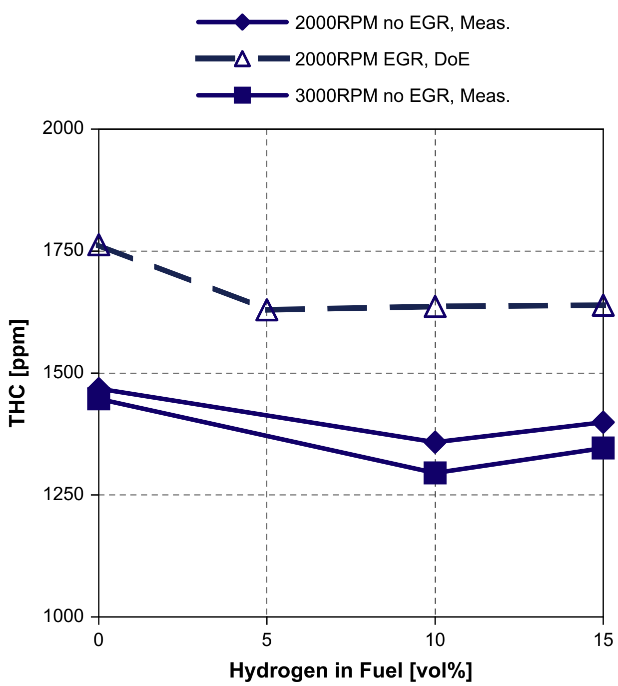

Fig. 14 – Engine-out THC with the engine operating parameters leading to the maximal efficiencies 4 bar bmep engine load.

there a resulting advantage if taking also into account the energy required for producing, transporting and distributing such fuels? An additional and related question concerns the green house gas (GHG) emissions. The efficiency gains

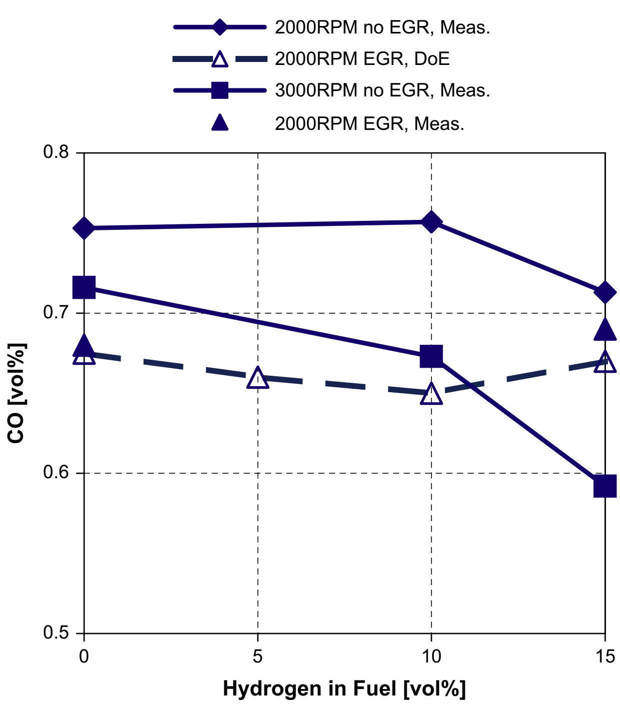

Fig. 15 – Engine-out CO with the engine operating parameters leading to the maximal efficiencies 2 bar bmep engine load.

described in the former section have been accompanied by substantial reduction in CO_2 emissions, given also the substitution of a small portion of CNG by (non-carbon) hydrogen. We did not include a relevant discussion in the former section, notably because GHG emission considerations should be based on well-to-wheel assessments taking also into account relevant emissions of the hydrogen production path.

The following assessments are mainly based on the results reported in Refs. [32,33] and concern current powertrain technologies taking into account their possible optimization potential in the near future (year 2010). Tank-to-wheels (TTW) values represent the energy required in the vehicle tank in order to move a state of the art European mid-sized vehicle, while well-to-tank (WTT) includes the energy required to produce the associated amount of fuel. The sum of the two comprises the well-to-wheels value (WTW).

In a first step, the WTW analysis for CNG has to be considered (Fig. 16). There are three major sources for CNG in Europe today: 7000 km pipeline (from western Siberia), 4000 km pipeline (from south-west Asia) as well as liquefaction and shipping over a distance of 10,000 km (from Middle East). Results of the first and the latter have been very similar and both are represented in Fig. 16 by the bar labelled ''LNG ship.''

An alternative way of producing CNG is based on the anaerobic fermentation of organic matter producing a gaseous mixture known as ''biogas'' consisting mainly of methane and CO_2. Production of automotive fuel requires cleaning and removing various impurities as well as the bulk of the CO_2. Ref. [32] considered three types of upgraded biogas; production from municipal waste, dry and wet manure. In all cases, it was assumed that the upgraded gas joins an existing gas grid to reach the refuelling station.

In Fig. 16, we plot the comparisons having also the state of art gasoline and diesel vehicles as benchmarks. The TTW values are similar for all fuels, notably the diesel vehicle has some advantage based on the better efficiency of the engine. The energy required for the fuel supply is also similar, the ''LNG ship'' path reaching highest values. The energetic disadvantage of the three biogases is not important; the WTT energy required for such fuels is provided by the waste material itself. The green house gas emission profiles of the biogas production are extremely favourable (even negative). The benefits though lie on avoiding emissions associated with livestock rearing or waste. It should be also kept in mind that biogas production is limited and capable of supplying only fractions of mobility's energy demand. Anyhow, GHG emissions of vehicles fuelled with CNG from any source are significantly lower than the diesel or gasoline benchmarks.

For the further analysis, the additional benchmark used (apart from gasoline and diesel) is CNG provided by the 4000 km pipeline and directly used to fuel an IC engine. In addition, this was the CNG production path selected as foundation for the blended fuels. According to the findings described in the former sections, the basic assumption for the following analysis is a 4% efficiency improvement of the engine by 10 vol.% (equivalently 2 mass%) substitution of CNG through hydrogen.

Nowadays the most widespread hydrogen production route is steam reforming of natural gas. Existing reformers are

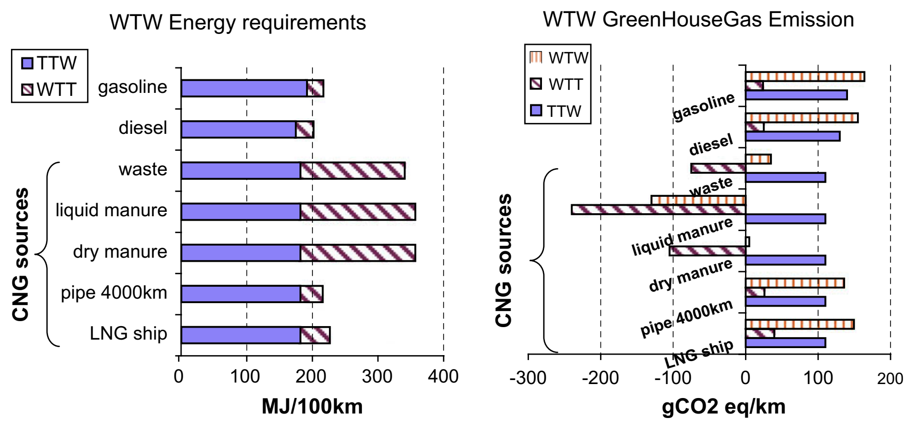

Fig. 16 – Well-to-wheels analysis for different CNG supply paths [32].

large industrial plants but small scale prototypes have also been developed. Although large scale plants tend to be more efficient, a significant disadvantage lies in distribution issues. As an alternative, hydrogen can be generated onboard the vehicle by small reformers using gasoline, CNG or methanol. In this case, the advantages of avoiding the hydrogen distribution issues are counterbalanced by complex vehicle and reformer technology. Methanol (MeOH) reforming is the energetically better option, since the reformer is operating at low temperatures and is more tolerant to intermittent demand. Using methanol raises of course, an additional infrastructure issue.

In Fig. 17 we compare the following hydrogen reforming paths for producing the fuel blends: onsite (e.g., refuelling station) of CNG transported via a 4000-km pipeline. Central reformation of CNG (4000 km pipeline vs. 10,000 km ship transport of liquefied CNG) and onboard processing of CNG to methanol and reforming into hydrogen. Energetic advantages are very small, if any, compared to the pure CNG vehicle. Still the diesel powertrain exhibits here highest advantages. GHG advantages of the hydrogen–CNG blends seem though significant. In particular, onboard reforming as well as CNG reforming in a central plant, lead to overall GHG emissions reduction of 7% in respect to the diesel benchmark vehicle. GHG emission reduction in respect to the benchmark CNG vehicle is approximately half as much. Self-evidently, tank-to-wheel GHG emissions of hydrogen are zero. The non-zero TTW values in Fig. 17 correspond to GHG generated by the combustion of the methane component of the fuel blend. Nevertheless some caution is expedient: the hydrogen quantities in question are very small, whether onboard reforming of such quantities is possible without any additional losses is questionable. The more feasible solution seems to be central reforming of CNG (travelling 4000 km pipeline prior to the reforming station) and then mixing in the existing CNG refuelling network. Central reforming can be relatively easy combined with CO_2 sequestration with a much higher GHG saving potential. Such considerations, though are beyond the scope of the present article, the interested reader is referred to Ref. [34].

A second major hydrogen producing path is electrolytic splitting of the water molecule. This is a well-established technology both at large and small scale. Electricity, being the primary source of energy, allows the consideration of several paths. Fig. 18 displays a comparison among the benchmark fuels and the hydrogen–CNG blends, where the hydrogen is produced by several electrolytic paths. Considering the GHG emissions, nuclear and wind energy electricity exhibit the

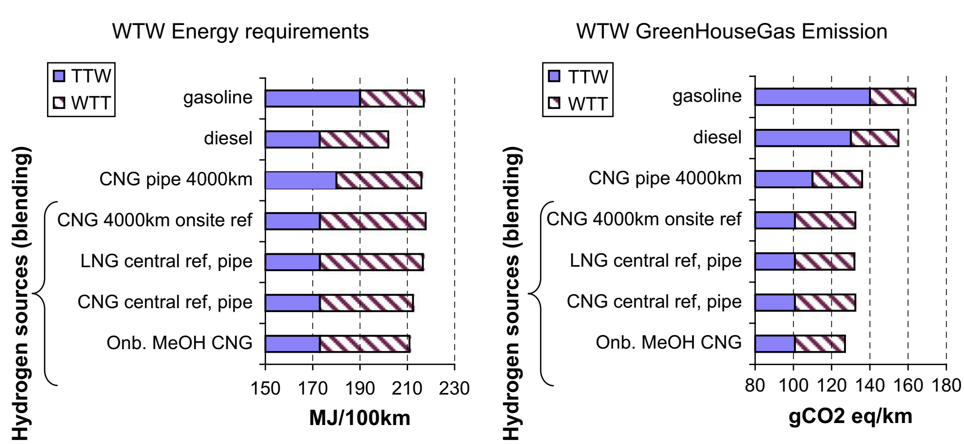

Fig. 17 – Well-to-wheels analysis for different hydrogen reforming production paths.

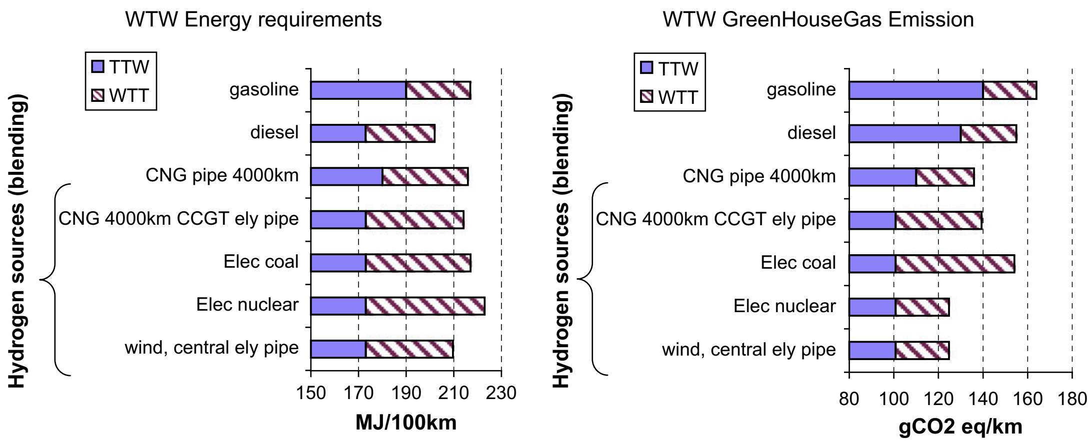

Fig. 18 – Well-to-wheels analysis for different hydrogen electrolytic production paths.

highest advantages. The advantages are not significantly higher than the ones when producing hydrogen through the best reforming paths.

6. Conclusions

We have demonstrated the potential of H_2–CNG mixtures with EGR in increasing the efficiency of a passenger car production engine while substantially reducing engine-out NO_x with only a modest increase in total unburned hydrocarbons.

Increasing the H_2 fraction in the fuel blend not only results in accelerating combustion, improving the effects of and tolerance towards EGR, but also in increased wall heat losses. Increasing EGR rate slows down combustion and reduces pumping work and in-cylinder peak temperatures. The combination of hydrogen in the fuel and moderate EGR leads combustion to significantly more favourable efficiency-(engine-out) NO_x trade-offs. Main conclusions can be summarized as follows:

- The addition of hydrogen in the fuel increases engine efficiency. Beyond a threshold hydrogen fraction in the fuel, the efficiency gains are diminishing, and in higher loads even offset by increasing wall heat losses. Dedicated combustion chamber design has potential in reducing the wall heat losses reaping additional efficiency benefits.
- Hydrogen enrichment of the methane fuel leads to lower sensitivity of the maximal engine efficiency to an exact EGR quantity.
- Depending on the engine load 2–4% efficiency increase can be attributed to the hydrogen in the fuel and additional 1–2% to EGR.
- The optimal EGR quantity leads to a 40% reduction of engine-out NO_x (in respect to the pure methane, no EGR baseline), to a 10% increase of engine-out THC, while practically not affecting engine-out CO levels.
- The hydrogen component of the fuel mainly accelerates the initial combustion phase.
- Well-to-wheel analysis identifies reforming and electrolytic hydrogen production paths leading to overall energy requirements somewhat higher than a benchmark diesel vehicle, concomitantly leading to significant reduction of green house gas emissions by almost 10%.

REFERENCES

[1] Bach C, Lämmle C, Bill R, Soltic P, Dyntar D, Janner P, et al. SAE Paper 2004-01-0645. Clean engine vehicle: a natural gas driven euro-4/SULEV with 30% CO_2-emissions 2004.
[2] Dimopoulos P, Rechsteiner C, Soltic P, Laemmle C, Boulouchos K. Increase of passenger car engine efficiency with low engine-out emissions using hydrogen-natural gas mixtures: a thermodynamic analysis. Int J Hydrogen Energy 2007;32:3073–83.
[3] Akansu SO, Dulger Z, Kahraman N, Veziroglu TN. Internal combustion engines fuelled by natural gas–hydrogen mixtures. Int J Hydrogen Energy 2004;29::1527–39.
[4] Bauer CG, Forest TW. Effect of hydrogen addition on the performance of methane-fuelled vehicles. Part I: Effect on S.I. engine performance. Int J Hydrogen Energy 2001;26::55–70.
[5] Bade Shrestha SO, Karim GA. Hydrogen as an additive to methane for spark ignition engine applications. Int J Hydrogen Energy 1999;24:577–86.
[6] Ferrera M. Natural gas technology a bridge towards hydrogen, Plenary session, 8th International conference on engines for automobile, Capri, Italy; 2007.
[7] Larsen JF, Wallace JS. Comparison of emissions and efficiency of a turbocharged lean-burn natural gas and hythane fuelled engine. J Eng Gas Turbines Power 1997;119: 218–26.
[8] Sierens R, Rosseel E. Variable composition hydrogen/natural gas mixtures for increased engine efficiency and decreased emissions. J Eng Gas Turbines Power 2000;122:135–40.
[9] Akansu SO, Kahraman N, Ceper B. Experimental study on a spark ignition engine fuelled by methane–hydrogen mixtures. Int J Hydrogen Energy. doi: 10.1016/j.jhydene.2007.05.034.
[10] Blarigan PV, Keller JO. A hydrogen-fuelled internal combustion engine designed for single speed/power preparation. Int J Hydrogen Energy 2002;23::603–9.
[11] Wong YK, Karim GA. An analytical examination of the effects of hydrogen addition on cyclic variations in homogeneously charged compression-ignition engines. Int J Hydrogen Energy 2001;25::1217–24.
[12] Huang Z, Wang J, Liu B, Zeng K, Yu J, Jiang D. Combustion characteristics of a direct-injection engine fuelled with

natural gas–hydrogen blends under various injection timings. Energy and Fuels 2006;20:1498–504.
[13] Mohammadi A, Shioji M, Nakai Y, Ishikura W, Tabo E. Performance and combustion characteristics of a direct injection SI hydrogen engine. Int J Hydrogen Energy 2007;32: 296–304.
[14] SAE Paper 2004-01-0972, published in SAE Transactions Conte E, Boulouchos K. Influence of hydrogen-rich gas addition on combustion, pollutant formation and efficiency of an IC-SI engine. Sec. 3. J Fuels Lubricants 2004;113.
[15] Allgeier T, Klenk M, Landenfeld T, Conte E, Boulouchos K. SAE Paper 2004-01-1270. Advanced emission and fuel economy concept using combined injection of gasoline and hydrogen in SI-engines 2004.
[16] Tully EJ, Heywood JB. SAE Paper 2003-01-0630. Lean-burn characteristics of a gasoline engine enriched with hydrogen from a plasmatron fuel reformer 2003.
[17] Topinka JA, Gerty MD, Heywood JB, Keck JC. Knock behaviour of a lean-burn, H_2 and CO enhanced, SI gasoline engine concept, SAE Paper 2004-01-0975, 2004.
[18] Allenby S, Chang WC, Megaritis A, Wyszynski ML. Hydrogen enrichment: a way to maintain combustion stability in a natural gas fuelled engine with exhaust gas recirculation, the potential of fuel reforming. Proc Int Mech Eng Part D 2001;215:405–18.
[19] Heffel JW. NO_x emission and performance data for a hydrogen fuelled internal combustion engine at 1500 rpm using exhaust gas recirculation. Int J Hydrogen Energy 2003; 28:901–8.
[20] Rebecchi P. Untersuchung des Potentials von Abgasrückführung im schnelllaufenden Erdgasmotor mit Verwendung von D.o.E. Methoden. Semesterarbeit, EMPA Dübendorf und ETHZ LAV, Zürich 2005.
[21] Dimopoulos P, Boulouchos K, Rechsteiner C, Soltic P, Hotz R. Combustion characteristics of hydrogen–natural gas mixtures in passenger car engines, SAE 2007-24-0065.
[22] Bach C. Concept of an exhaust gas aftertreatment for methane-gas powered vehicles. In: International conference on low carbon fuels, Empa Akademie, November 2005.
[23] WEG. Rechenprogramm zur Berechnung des Wärmeentwicklungs-gesetzes aus gemessenen Brennraumdruckverläufen von Diesel- und Ottomotoren, Benutzerhandbuch und Programmdokumentation. ETHZ IET-LAV, Zürich September 2005.
[24] van Basshuysen R, Schäfer F. Handbuch Verbrennungsmotor, 2 Auflage; 2002.
[25] Nellen C, Boulouchos K. Aufgeladene Gasmotoren mit AGR und Dreiwege-Katalysator – der Weg zu niedrigsten Emissionen bei hohem Wirkungsgrad und grosser Leistungsdichte, MTZ Motortechnische Zeitschrift. Jahrgang 1/2000;61.
[26] Berner HJ, Bargende M. Ein CO_2-minimales Antriebskonzept auf Basis des Kraftstoffes Erdgas, Haus der Technik Fachbuch Band 37, expert Verlag, Essen 2004.
[27] Edwards SP, Pilley AD, Michon S, Fournier G. The optimisation of common rail FIE equipped engines through the use of statistical experimental design. SAE 971635, Mathematical modelling and genetic algorithms; 1997.
[28] Dimopoulos P, Schöni A, Eggimann A, Sparti C, Vaccarino E, Operti C. SAE 1999-01-1077. Statistical methods for solving the fuel consumption/emission conflict on DI-diesel engines 1999.
[29] Rakopoulos CD, Kyritsis DC. Hydrogen enrichment effects on the second law analysis of natural and landfill gas combustion in engine cylinders. Int J Hydrogen Energy 2006; 31:1384–93.
[30] Golloch R. Downsizing bei Verbrennungsmotoren. Berlin, Heidelberg: Springer Verlag; 2005.
[31] Witt A. Analyse der thermodynamischen Verluste eines Ottomotors unter den Randbedingungen variabler Steuerzeiten, PhD Thesis, Technische Universität Graz, Institut für Verbrennungskraftmaschinen und Thermodynamik, Graz 1999.
[32] EUCAR, Concawe, JRC. Well-to-wheels analysis of future automotive fuels and powertrains in the European context. version 2b, ; 2006.
[33] Muradov NZ, Veziroglu TN. From hydrocarbon to hydrogen–carbon to hydrogen economy. Int J Hydrogen Energy 2005;30: 225–37.
[34] Muradov NZ. Hydrogen via methane decomposition: an application to decarbonisation of fossil fuels. Int J Hydrogen Energy 2001;26:1165–70.