

Effects of Methane/Hydrogen Blends On Engine Operation: Experimental And Numerical Investigation of Different Combustion Modes

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ABSTRACT

The introduction of alternative fuels is crucial to limit greenhouse gases. CNG is regarded as one of the most promising clean fuels given its worldwide availability, its low price and its intrinsic properties (high knocking resistance, low carbon content...). One way to optimize dedicated natural gas engines is to improve the CNG slow burning velocity compared to gasoline fuel and allow lean burn combustion mode. Besides optimization of the combustion chamber design, hydrogen addition to CNG is a promising solution to boost the combustion thanks to its fast burning rate, its wide flammability limits and its low quenching gap.

This paper presents an investigation of different methane/hydrogen blends between 0% and 40 vol. % hydrogen ratio for three different combustion modes: stoichiometric, lean-burn and stoichiometric with EGR. The main objectives are to identify the complex mechanisms involved in the combustion process and to define the optimal hydrogen ratio for each combustion mode.

The study combines engine tests and 0D modeling. Tests were carried out on a spark-ignited single cylinder engine adapted to CNG operation with 2 different compression ratios 9,5:1 and 11,5:1. Computations allow to study separately the different phenomena linked to the progressive addition of hydrogen in the fuel. Hence, using 0D modeling, the effects due only to the combustion speed evolution, as a function of

hydrogen ratio in the fuel, were then quantified using experimental results as comparison basis.

The engine test results reveal that the impact of hydrogen is limited in stoichiometric conditions except for CO₂ savings. A ratio of 10 to 20 vol. % of hydrogen seems to be optimal to reach interesting HC emissions reduction without an excessive lowering of the operation range. The results are more encouraging for lean-burn operation as the lean limit of equivalence ratio is extended for more than 0,125 with 40 vol. % of hydrogen. NO_x emissions and consumption were significantly reduced (–92% for NO_x) while maintaining constant HC and CO emissions compared to methane operation. Hydrogen stabilizes and speeds up combustion especially in very diluted mixtures. Furthermore, hydrogen increases EGR tolerance for the same reasons as in lean-burn mode: in comparison, identical very-low NO_x levels were reached with smaller consumption gains. Finally, the engine test results show that the highest tested ratio of hydrogen (up to 40%) in methane for very diluted mixtures with air or EGR drastically reduce NO_x emissions without penalty on engine performances.

Computations prove that the evolution in terms of thermodynamic properties of the fuel is not the most important contribution to the evolution of combustion. The effect of hydrogen addition contributes mainly to increase the combustion speed. Moreover, lean burn operation results demonstrate that the increase of combustion speed due to

Table 1. Chemical and physical fuel properties

Fuel Characteristics	Unit	Gasoline	Methane	Hydrogen
Formula	-	C _{7.1} H _{12.56}	CH ₄	H ₂
Molar mass	g/mol	98	16	2
Flammability limits	% vol.	1.0 - 7.6	5.3 - 15	4 - 75
Max. laminar burning velocity in air	m/s	0.5	0.4	2.9 – 3.5
Net ignition energy	mJ	0.24	0.29	0.02
Quenching distance	mm	2.84	1.9	0.6
Stoichiometric A/F ratio (mass)	-	14.7	17.2	34.2
Lower Heating Value	MJ/kg	40-45	50	120.5
Energy density (15°C/100kPa)	MJ/m ³	33750	35.82	10.30

hydrogen addition is enhanced. This can only be explained by the increase of thermo-diffusivity which has an opposite and larger effect than the slow down of laminar flame velocity conventionally described in the literature.

INTRODUCTION

CONTEXT

With the increasing concerns in reducing dependence on crude oil given the future come about of shortages, researchers in the automotive industry focus their investigations on the use of alternative fuels. CNG is regarded as one of the most promising clean fuels according to its intrinsic physical and chemical properties: low carbon content, high knocking resistance and absence of fuel evaporation [1, 2]... Besides, CNG is available worldwide and its price is quite low. Natural gas is a variable mixture of gases and its major component is methane. The easiest way to take advantage of natural gas is to burn it in a SI engine easily adapted to this fuel. Nevertheless, CNG combustion is limited in specific conditions and has disadvantages such as slow burning velocity compared to gasoline fuel and poor lean burn capability. Because of these restrictions, natural gas engines are commonly operated at stoichiometric conditions without providing the maximal thermal efficiency.

Different effective ways are available to improve CNG combustion. The first solution is to increase turbulence and flow intensity within the cylinder by optimizing intake ducts and combustion chamber design. But the effect is limited and leads to other penalties such as lowered volumetric efficiency and higher heat losses. Another effective way is to optimize ignition by increasing spark energy or the number of spark plugs. Like the turbulence effect, an important drawback appears since this high ignition energy demand shortens spark plug life. Therefore, there is a need to enhance methane combustion without having the disadvantages mentioned above.

A promising solution is to mix natural gas with an additional fuel with specific chemical properties. Hydrogen is the best

option as it has been recognized as a fuel with unique combustion properties for spark ignited engines [1]. Indeed, hydrogen can accelerate methane combustion as it has a high laminar burning velocity (seven times as much as methane in stoichiometric conditions). Moreover, hydrogen has wide flammability limits, low ignition energy and low quenching gap (see Table 1). All these advantages allow the capability of methane/hydrogen blends to burn with good stability in lean mixtures or with high EGR rates to be extended. This kind of combustion yields to high CO₂ savings by means of self-evident absence of carbon in hydrogen fuel. Furthermore, thanks to these extreme operating modes, emissions can be reduced and thermal efficiency enhanced. Besides, methane/hydrogen blends are a practical route to reduce dependence on non-renewable gaseous fuels. With a limited volume fraction of hydrogen in methane (generally 20%), the existing CNG infrastructure could be used to refuel vehicles.

OBJECTIVES AND METHODOLOGY

This paper presents an overview of the combustion of methane/hydrogen blends. The methodology used combined 3 different tools: literature review, 0D modeling and engine tests. The main objectives of this study were to identify the complex mechanisms involved in the combustion process and to define the optimal hydrogen volume fraction for each combustion mode.

Relevant information found in the literature were used to define the composition of the tested blends and to build the engine test program. Experimental tests were carried out to compare different combustion modes in order to determine the best trade-off for the use of CH₄/H₂ blends. 0D modeling was used to evaluate the impact of fuel thermodynamic properties on combustion process. Then, the contribution of flame speed evolution due to progressive introduction of hydrogen in methane was quantified with a simulator based on specific engine test bench results.

Table 2. Engine specifications

Engine displacement	301 cm ³ / Single cylinder
Bore / stroke	69 mm / 80.5 mm
Connecting rod length	139 mm
Compression ratio	9.5:1 or 11.5 :1
Intake valve opening (IVO)	10° ATDC
Intake valve closing (IVC)	20° ABDC
Exhaust valve opening (EVO)	40° BBDC
Exhaust valve closing (EVC)	10° BTDC
Piston design	Bowl-in-piston
Fuel system	Indirect injection on intake manifold
Ignition system	Delphi Coil (110mJ) NGK spark plug (T2511H)
Cylinder pressure measure	Kistler 6043 flush pressure transducer

TEST PROGRAM AND EXPERIMENTAL SETUP

TEST PROGRAM

Based on the information from literature, a program was built in order to test three different combustion modes: stoichiometry, stoichiometry with EGR and lean-burn. The objectives were to obtain large variations of different parameters: hydrogen volume fraction in methane, engine speed, IMEP, equivalence ratio, EGR rate and compression ratio. In stoichiometric conditions, engine was operated from low IMEP (2 bar) up to full load at 5 different engine speeds: 1000, 1500, 2000, 3500 and 5500 rpm. In stoichiometry with EGR and lean burn modes, 2 specific points were selected: a first case at 3 bar IMEP for 2000 rpm and a second case at 9 bar IMEP for 3500 rpm. Specific tests were conducted at the same spark timing for different methane/hydrogen blends in order to observe the impact of hydrogen introduction without any adaptation of combustion phasing.

ENGINE SPECIFICATIONS

The tests were carried out on a single cylinder research engine with a 301 cm³ displacement (Table 2). Previously designed for gasoline operation, the engine was especially adapted to gaseous fuels by changing valve seats and exhaust valves. Since higher thermal and mechanical constraints in the combustion chamber were expected, 3D modeling was conducted to validate the mechanical structure integrity of the cylinder head, connecting rod, piston and engine block during the tests. Two different compression ratios were tested: 9.5:1 and 11.5:1 in order to evaluate the impact of the compression ratio increase on CH₄/H₂ blend combustion. The variation of this parameter was only obtained by the piston shape modification. The quenching zone in the periphery of the combustion chamber was kept constant for both compression ratios in order to make it possible to evaluate the reduced

quenching distance impact for the hydrogen volume fractions independently from the change of piston shape.

One low pressure gas injector integrated in a Landi Renzo feed ramp was installed and the gas was injected in the intake manifold. The gas injection pressure was controlled at 2 bar above the intake manifold air pressure to ensure injection flow control. Injectors were selected by taking into account the reduction of the volumetric heating value with methane/hydrogen blends compared to methane fuel (see Table 3). Indeed, this effect increases the fuel injection duration to maintain the same heat release for all methane/hydrogen blends. Gas consumption was measured by a MicroMotion CMF010 Coriolis type mass flow meter.

The ignition system is a key technology for combustion with very diluted mixtures such as lean burn or with high EGR rates. Consequently, a specific ignition system was used with a coil drawing approximately 110 mJ of energy on the primary circuit and a spark plug with a 0.7 mm iridium tipped electrode. The coil charging time was kept constant to separate the effect of an improved ignition energy from the effect of H₂. Cylinder pressure was monitored by a flush mounted Kistler pressure transducer. The pressure signal was acquired for every crank angle degree and the acquisition process covered 160 complete cycles. The average value of these cycles was used as pressure data for the calculation of combustion parameters. Engine-out exhaust gases (HC, CH₄, CO, CO₂, O₂, NO_x) were analyzed by a Horiba MEXA-7100DEGR analyzer. A separate line of this analyzer was used to measure CO₂ in the intake for EGR rate evaluation.

FUELS

Four different hydrogen volume fractions were tested: a reference test case with pure methane, an intermediate case

Table 3. Main properties of the investigated fuels

Fuel Characteristics	Unit	CH ₄	10 vol.% H ₂	20 vol.% H ₂	40 vol.% H ₂
Vol. fraction H ₂	vol. %	0	10	20	40
Vol. fraction CH ₄	vol. %	100	90	80	60
LHV	kJ/kg	50 006	50 988	52 172	55 496
VHV	kJ/m ³	35 815	33 322	30 829	25 843
Mass Ratio H/C	-	4	4.22	4.5	5.33
Stoichiometric A/F ratio	(mass)	17.22	17.45	17.74	18.54
Theoretical CO ₂ mass reduction	%	0	3.3	7.1	16.8

for H₂ = 10 vol.%, Hythane[®] fuel H₂ = 20 vol.% and an extreme case for H₂ = 40 vol.% (see Table 3). The two gases were mixed by the supplier prior to filling the fuels into specific high pressure gas bottles (approximately 200 bar).

The variations of thermodynamic properties of the tested fuels are limited: there is a 10% Lower Heating Value (LHV) increase from 0 to 40 vol.% H₂. The H/C ratio increases linearly with the hydrogen fraction in the fuel blends and this is beneficial to the reduction of carbon-related emissions such as HC, CO and CO₂. By the way, theoretical CO₂ mass reduction is evaluated at 16.8% for 40 vol.% H₂. The volumetric heating value is reduced by more than 30% for 40 vol.% H₂ compared to pure methane. This means that hydrogen introduction in methane leads to an exponential penalty on vehicle range.

CRITICAL SETTINGS

The main mechanical and safety limits were identified. The maximum average cylinder pressure takes cycle-to-cycle variations into account and was set to 150 bar. This value constraints the full load operation as there is practically no limitation coming from exhaust temperatures. Besides, intake manifold air temperature, coolant temperature, and oil temperature were respectively regulated at 50°C, 90°C and 100°C. Spark timing was set close to MBT which corresponds to a crank angle position for maximal cylinder pressure of around 12 or 13 CAD (Crank Angle Degrees) ATDC regardless of the fuel used. No knock has been detected during engine tests with both compression ratios. Whatever the test fuel, the ISFC (Indicated Specific Fuel Consumption) was corrected in order to be able to compare the different specific fuel consumptions independently of the fuel's LHV. Therefore, the ISFC was corrected with the gasoline LHV according to the following formula and is referred to as cISFC in this paper:

$$cISFC = ISFC_{\text{test fuel}} \times \frac{LHV_{\text{test fuel}}}{LHV_{\text{gasoline}}}$$

ENGINE TEST RESULTS AND DISCUSSION

STOICHIOMETRIC MODE

Tests at equal spark timing

Impact on cylinder pressure

The first tests were carried out in stoichiometric conditions with the same spark timing for all hydrogen volume fractions. This Spark Timing (ST) is defined as the spark timing for MBT with pure methane minus 10 CAD. This voluntary constraint allows to observe the effect of hydrogen on combustion behavior independently from combustion phasing. The figure 1 depicts the impact of progressive introduction of hydrogen on the cylinder pressure trace at 9 bar IMEP, 3500 rpm and for a 9.5:1 compression ratio. The TDC when combustion occurs was defined at 360 CAD. In this case, spark timing was constant at 18 CAD BTDC.

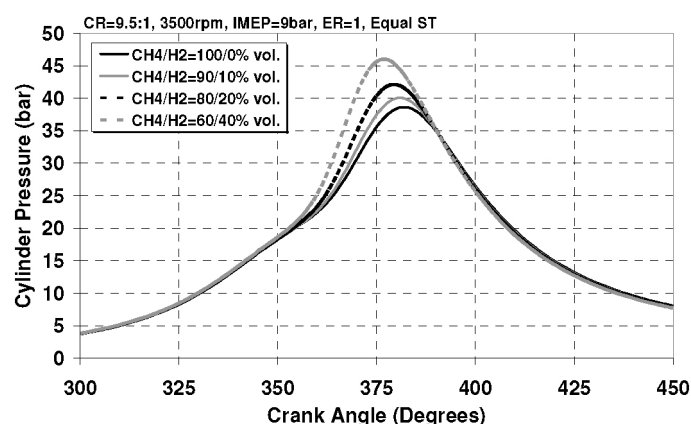


Figure 1. Cylinder pressure in stoichiometric mode at equal ST for different CH₄/H₂ blends

With a constant spark advance, when hydrogen was progressively introduced in the fuel, the maximum cylinder pressure increased linearly: +20% between pure methane and 40 vol.% H₂ and was phased closer to the TDC. Moreover, the maximal positive gradient of cylinder pressure also increased linearly but more rapidly: +60% between pure

methane and 40 vol.% H₂. Reference [3] mentions that the combustion acceleration induced by hydrogen increases the maximum heat release rate, maximal combustion temperature and peak cylinder pressure whatever the combustion mode and confirms these results.

Heat release analysis

Heat release analysis was based on the in-cylinder pressure trace. The impact of enhanced combustion speed caused by hydrogen can also be depicted by the two following parameters: Rate of Heat Release (ROHR) and Mass Fraction Burnt (MFB) in figures 2 and 3.

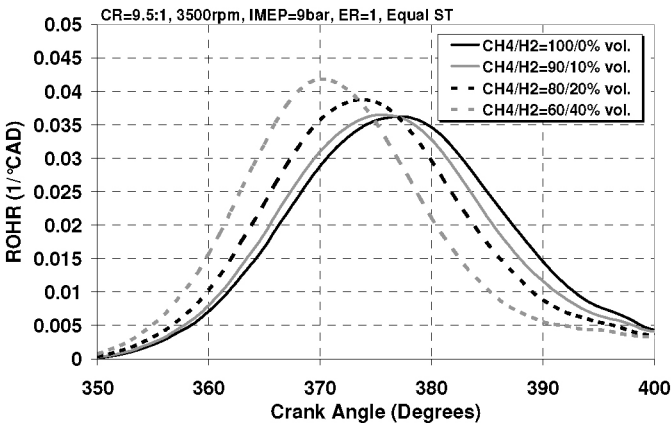


Figure 2. Rate of heat release in stoichiometric mode at equal ST for different CH₄/H₂ blends

When the hydrogen volume fraction increased, the maximum ROHR and its maximal positive gradient increased linearly: +20% for maximum ROHR between pure methane and 40 vol.% H₂. The maximum ROHR was reached closer to the TDC. Besides, deeper insights into combustion behavior can be achieved by comparing the durations required for burning a given quantity of fuel. Figure 3 shows the evolution of MFB50 (which represents the elapsed period for burning 50% of the mass fuel): this parameter was shortened by 2 CAD for every 10 vol.% of hydrogen introduced in methane for a same spark timing. The combustion phasing evolved closer to the TDC.

The acceleration of the combustion flame speed impacted the combustion duration, especially for the initial and intermediate combustion phases respectively defined by the 0-5% and the 10-50% burnt duration parameters (see figure 4). The reduction was similar for these two phases and was estimated at -15% for 40 vol.% H₂ compared to the pure methane case. This phenomenon was observed in the literature but with a combustion phasing meaning that the spark advance is adjusted to MBT for each fuel.

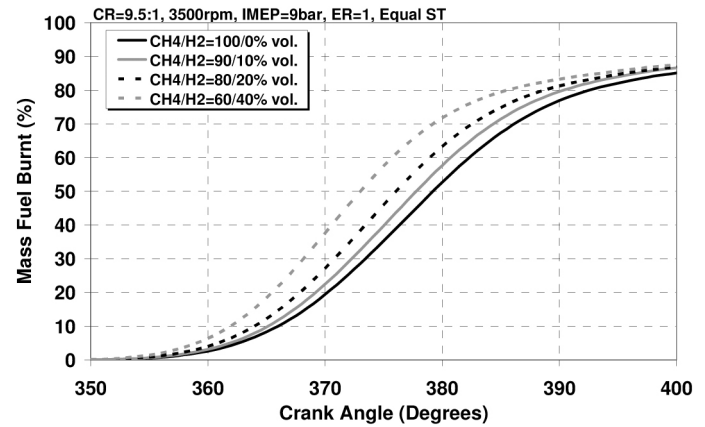


Figure 3. Fuel mass burnt ratio in stoichiometric mode at equal ST for different CH₄/H₂ blends

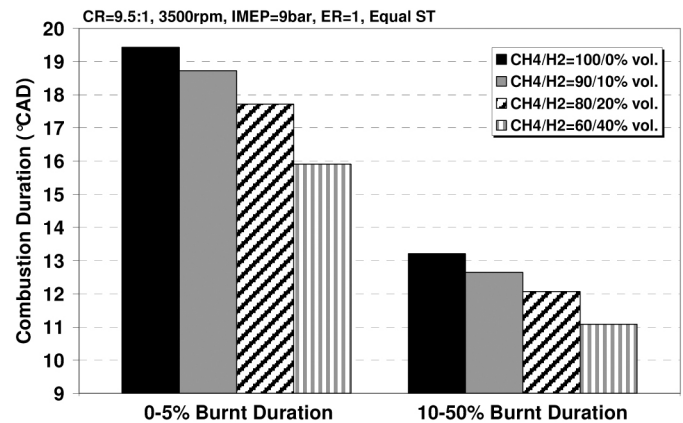


Figure 4. Combustion duration in stoichiometric mode at equal ST for different CH₄/H₂ blends

Test at spark timing set to MBT

Effect of combustion acceleration

In this part of the study, tests were carried out in stoichiometric conditions at an engine speed of 3500 rpm with spark timing set at MBT for each fuel. The use of hydrogen accelerated the combustion process and needed a lower spark advance than for the pure methane. The figure 5 shows the evolution of spark timing versus IMEP for the four different blends. The optimal spark timings at low loads were more advanced than the ones at higher loads. Moreover, the spark advance decreased linearly with the increase of hydrogen volume fraction mixed with methane fuel: there are a 2 or 3 CAD of spark timing decrease at low loads for every 10 vol.% of hydrogen and 1 or 2 CAD of spark timing decrease at high loads for every 10 vol.% of hydrogen. These observations were in good agreement with the results of other publications [4, 5]. According to reference [4], for heavy duty engine with 6.234 liters of displacement in lean burn operation, spark advance set to MBT was closer to TDC by 2° CAD for each H₂ = 10 vol.% introduced in methane.

Given the evolution of flame speed induced by hydrogen, it is necessary to change combustion phasing by setting spark timing closer to TDC.

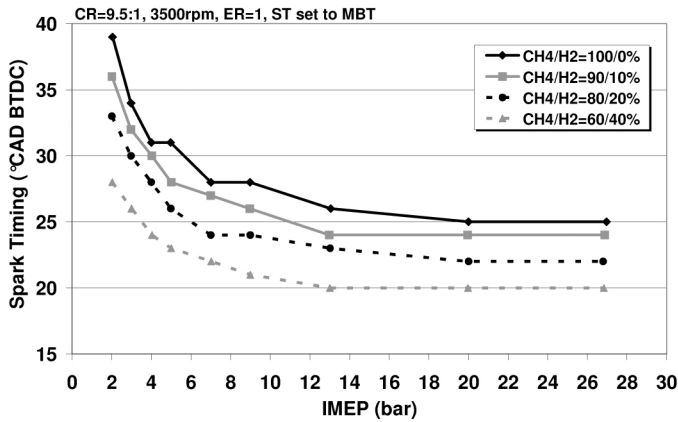


Figure 5. ST versus IMEP at 3500 rpm in stoichiometric mode for different CH₄/H₂ blends

As observed in tests at equal spark timing, the initial combustion phase was drastically shortened: 0-5% burnt duration was reduced by around 1 CAD for every 10 vol.% of hydrogen like described in figure 6. This result is linked to the combustion reactions mechanism. Indeed, hydrogen molecules strongly impact methane combustion development. Reference [6] mentions that methane combustion flame speed is limited by the slow reaction $\text{CH}_4 + \text{O} \rightarrow \text{CH}_3 + \text{OH}$. The other reactions leading to H₂O and CO₂ for completing the combustion are very fast. Radicals obtained through hydrogen dissociation help to remove the first hydrogen atom from methane molecule allowing complete combustion and explaining why this combustion phase is shortened [7].

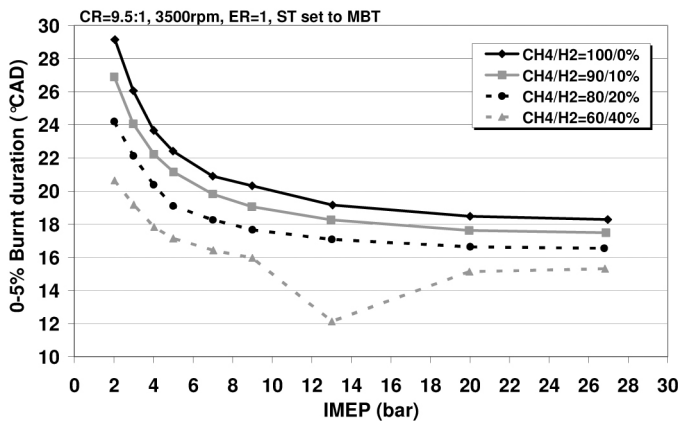


Figure 6. 0-5% burnt duration versus IMEP at 3500 rpm in stoichiometric mode for different CH₄/H₂ blends

Nevertheless, the intermediate combustion phase duration was slightly reduced with 10 and 20 vol. % of hydrogen but it was significantly shortened with 40 vol. % of hydrogen as

described in figure 7. With 10 and 20 vol. % of hydrogen, hydrogen molecules were entirely used for helping the first slow reaction of methane dissociation while at 40 vol. % of hydrogen, the larger quantity of hydrogen was also used for shortening intermediate combustion phase.

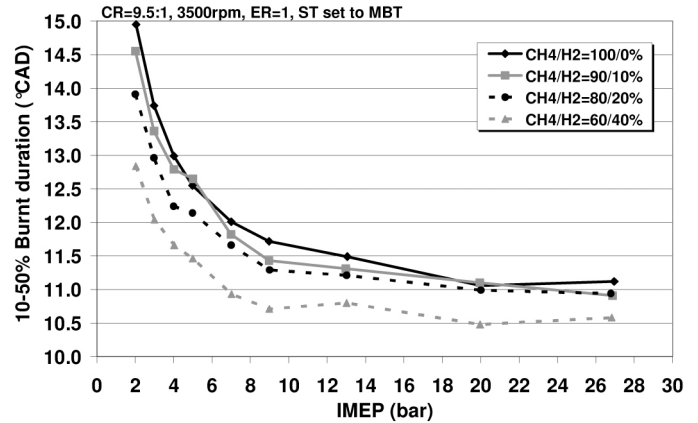


Figure 7. 10-50% burnt duration versus IMEP at 3500 rpm in stoichiometric mode for different CH₄/H₂ blends

Heat release analysis at equal spark timing showed an increase of maximum ROHR with hydrogen. Surprisingly, small ratios of hydrogen did not increase maximum ROHR in great proportions whereas 40 vol. % of hydrogen enhanced it by 4% as mentioned in figure 8. To sum up, hydrogen leads to acceleration of methane combustion causing the reduction of the initiation phase duration. The spark advance needed to be set closer to TDC and heat release was more intense for high hydrogen volume fractions.

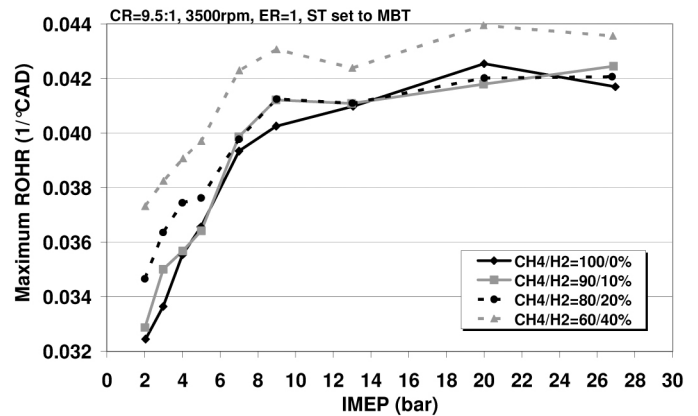


Figure 8. Max. ROHR versus IMEP at 3500 rpm in stoichiometric mode for different CH₄/H₂ blends

Effect of reduced quenching distance

The quenching distance is reduced with the use of hydrogen (five times less than methane) which implies a high decrease in HC emissions. The counterpart is the enhancement of heat losses since the warmer flame front approaches closer to the

cylinder walls when hydrogen is introduced [8]. Figure 9 depicts the HC emissions which were linearly decreased with the amount of hydrogen introduced: there is a reduction of ISHC (Indicated Specific HC) by 0.5 g/kWh for every 10 vol. % of hydrogen independently from the considered IMEP. It is relevant to make the difference between the HC gain due to the lower carbon content of the considered methane/hydrogen blend and the HC gain due to the lower quenching tendency. For instance at 3500 rpm, the reduction of ISFC was evaluated between 35 to 50% while the theoretical mass CO₂ savings were estimated around 16.8% according to Table 3. Consequently, HC reduction was mainly caused by the lower quenching distance of the hydrogen.

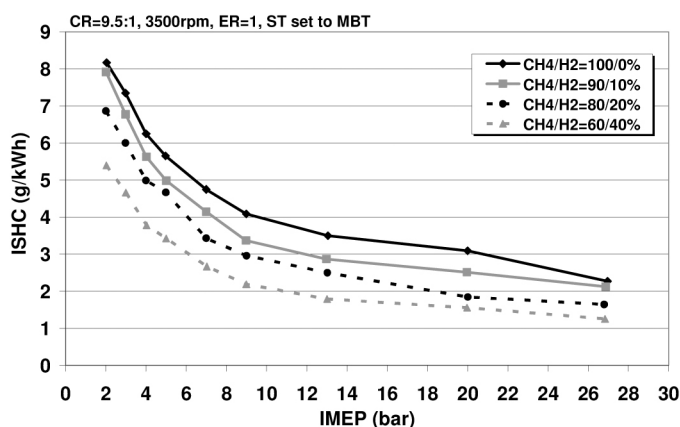


Figure 9. ISHC versus IMEP at 3500 rpm in stoichiometric mode for different CH₄/H₂ blends

Furthermore, this effect can be described by the reduction of O₂ concentration within the exhaust gas as depicted in figure 10. This chart shows that the flame front went closer to cylinder walls and penetrated further into crevices with high hydrogen ratios consuming more oxygen than during pure methane operation. Moreover, it means that hydrogen can be used to compensate the deficiencies of combustion chambers like non adapted designs or low turbulence and flow intensity by enhancing flame propagation through the cylinder. Nevertheless, it can be supposed that HC emission gains could also be achieved for an optimized combustion chamber.

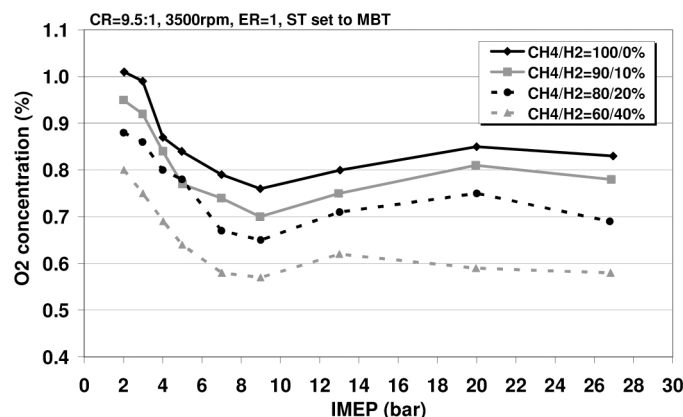


Figure 10. O₂ concentration in exhaust gas versus IMEP at 3500 rpm in stoichiometric mode for different CH₄/H₂ blends

The literature mentions slight gains in engine fuel conversion efficiency in stoichiometric mode [5]. Indeed, there are no physical losses if combustion occurs instantly at TDC. But combustion duration is finite. This is why the H₂ effect on flame acceleration allows a thermal efficiency gain since the real combustion is closer to a theoretical combustion at constant volume. Besides, combustion acceleration implies reduction in heat exchange duration with cylinder walls so engine efficiency is also improved by this phenomenon. In reference [5], with a 4-cylinder Volkswagen engine with 1.0L of displacement, efficiency gain was obtained because of the decrease of real combustion losses from 3.74% to 3.12% with H₂= 10 vol.% at 2000 rpm and 4 bar BMEP. Surprisingly, in this study and for these conditions, no ISFC benefit was observed. This could be explained by the competition between two opposite effects linked to the hydrogen introduction. On one hand, flame speed acceleration for an optimal spark advance enhanced engine efficiency as there were less losses due to the lower duration of combustion (closer to the theoretical case at constant volume). On the other hand, the reduction of quenching distance increased the duration of heat exchange with the cylinder walls of the combustion chamber [5]. For instance, in figure 11, the heat losses at 3500 rpm and 9 bar IMEP were evaluated over a complete thermodynamic cycle, using a Woschni correlation for the heat transfers. This shows that the hydrogen introduction increased heat losses: +15% between pure methane and 40 vol.% and this result cancelled the engine efficiency gain obtained by combustion acceleration for this engine configuration.