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Abstract

Biogas is a gas resulting from biomass, with a volumetric content of methane (CH_4) usually ranging between 50% and 70%, and carbon dioxide (CO_2) content between 30 and 50%; it can also contain hydrogen (H_2) depending on the feedstock. Biogas is generally used to generate electricity or produce heat in cogeneration system. Due to its good efficiency through the rapid combustion and lean air-fuel mixture, Homogeneous charge compression ignition (HCCI) engine is a good candidate for such application. However, the engine load must be kept low to contain the high-pressure gradients caused by the simultaneous premixed combustion of the entire in-cylinder charge. The homogenous charge promotes low particulate emissions, and the dilution helps in containing maximum in-cylinder temperature, hence reducing nitrogen oxide emissions. However, HC and CO levels are in general higher than in SI combustion. Moreover, HCCI mode usually requires high intake temperature with values depending on the compression ratio, adopted fuel, equivalence ratio, and intake pressure.

This paper presents the results of an experimental campaign on a diesel internal combustion engine for passenger cars, modified to operate in HCCI mode. The engine was fuelled by mixtures of methane, H_2, and carbon dioxide, with the aim at reproducing the composition of innovative biogas naturally containing H2. The equivalence ratio was kept constant at 0.4 and the intake charge temperature and pressure have been adjusted, depending on fuel composition, to control the combustion process. For each fuel, the intake charge conditions for an optimal combustion phasing have been determined, attaining a maximum indicated engine efficiency of 40% and specific NOx emissions down to 0.1 g/kWh.

Introduction

The Intergovernmental Panel on Climate Change (IPCC) reports [1] has provided a clear insight into climate change with estimated scenarios for global warming. The most optimistic prediction is a global warming of 2°C over the next century. To limit this as much as possible, global greenhouse gas (GHG) emissions will need to be significantly reduced in the coming years. Electricity and heat production, together with agriculture, forestry, and other land use (AFOLU), contribute to half of these global emissions. The agro-industrial feedstock, such as municipal sewage sludge, agricultural waste, manure, etc., is a source of renewable gases, making it an attractive fuel [2]. These gaseous fuels can be produced by various processes such as hydrothermal gasification (375-500°C), gasification (800-1000°C) and methanisation/anaerobic digestion (40°C) [3,4]. The choice between wet and dry processes depends on the nature of the feedstock and its water content. Among these processes, anaerobic digestion can take place in the presence of water and is suitable for the valorization of wet feedstocks such as wet animal manure [5,6] and municipal sewage sludge [7]. Anaerobic digestion produces mainly methane (CH4) and carbon dioxide (CO_2) to form biogas. The production process and the feedstock play an important role in determining the composition of the biogas, with a methane content in the range of 55-65%, 30-40% carbon dioxide and traces of H2 sulfide and sometimes ammonia (especially with nitrogen-rich feedstocks). The production of biogas is particularly beneficial to the environment because it avoids the aerobic decomposition of organic matter, which emits greenhouse gases into the atmosphere, and can replace the use of fossil fuels in several applications. In fact, the use of biogas in internal combustion engines is very attractive, even though the presence of an inert gas such as CO2 has a negative effect on combustion, reducing the combustion rate and lowering the flammability limits [8,9] as well as reducing the range of operation

because it decreases the stability of the engine through the cycle-to-cycle variations (CCVs). However, CO_2 dilutes the mixture resulting in lower reactivity and therefore lower flame temperature and lower NOx levels.

Biogas is suitable for Compression Ignition (CI) engines. Due to its low reactivity, there are two ways to use biogas in a CI engine: either with a pilot injection of a more reactive fuel to trigger ignition in a Dual-Fuel/RCCI (Reactivity Controlled Compression Ignition) combustion mode, or with a high compression ratio and intake temperature and pressure to enable a Homogeneous Charge Compression Ignition (HCCI) combustion mode. The Dual-fuel operation shows potential for noise and vibration reduction compared to natural gas, as well as lower NOx levels but higher CO and HC concentrations [10,11]. Compared to a SI engine running on biogas, a dual-fuel CI engine can offer several advantages such as an increased brake thermal efficiency (up to 20%), a lower fuel consumption and therefore less CO_2 emissions, and less sensitivity to the biogas composition [12]. Compared to a conventional diesel engine, a dual-fuel engine running on biogas is more economical and produces less soot. A major drawback is the need for a catalyst system to limit HC and CO emissions. In addition, this combustion mode still requires a more reactive fuel, which must be renewable to be fully carbon neutral, and an additional injection system. Biogas was recently studied in RCCI or dual-fuel mode experimentally [13,14] and numerically [15] showing good potential in terms of stability, performances and emissions even with Hydrotreated Vegetable Oil as pilot fuel

The HCCI combustion mode is very fuel flexible and is a simpler alternative in terms of design as the combustion is only driven by the auto-ignition of the homogeneous charge depending on the pressure and temperature conditions in the cylinder. However, HCCI requires high compression ratio and often high intake pressure and temperature and is limited in terms of control (this can be addressed by several strategies: reactive species such as H_2, NO, or O_3 [16,17], intake temperature [18], EGR [19]) and maximum cylinder pressure rise rate (MPRR). As a result, biogas is suitable for HCCI-fueled engine because the CO_2 content can reduce the MPRR but on the other hand requires high temperature to ensure auto ignition. It has been shown that using biogas in HCCI mode can lead to very high indicated efficiency (up to 45%) and very low NOx emissions [20]. However, to extend the operating conditions, the addition of more reacting species such as diethyl ether [21,22], solid biomass [23], or H_2 [19,24] is sometimes required.

Adding H_2 in the fuel is a common way of improving the reactivity of biogas, both in terms of increasing the flame speed for spark-ignition engine [25] or reducing the auto-ignition delay [11,17]. For the CI engine, Ibrahim and Ramesh [26] as well as Masurier et al. [17] showed promising results in terms of thermal efficiency and low emissions when H2 is used as fuel in HCCI engines, as it improves fuel reactivity and reduces pure methane auto-ignition temperature. Rather than considering the addition of H_2 from a process external to the anaerobic digestion (e.g., water electrolysis), H_2 is an intermediate product of biomass digestion. Therefore, a properly designed process can maximize H_2 yield to produce an innovative biogas mainly composed by CH_4, CO_2 and H_2. The eubacteria that promote H_2 production are naturally present in water buffalo manure [27], which can be considered as a particularly suitable substrate for the production of such biogas [28,29]. However, in this process, H2 is produced together with CO_2 with a molar ratio of about 1, whereas the CH_4/CO_2 ratio is typically around 2. Hence, the innovative biogas that can be obtained with a well-designed anaerobic digestion is, on the one hand, very appealing because enriched in H_2, and on the other hand, it can have adverse effects on combustion because it is also richer in CO_2. In terms of emissions, the presence of H_2 can favor the appearance of NOx because of higher temperature. However, as H_2 production is associated with CO_2 production in this particular type of biogas, NOx emissions could be mitigated. Such biogas compositions have been numerically investigated by Mariani et al. [19] with a target IMEP of 8.4 bar and an equivalence ratio of 0.4 showing the potential of H_2 in reducing the auto-

ignition temperature and the use of EGR to increase the intake temperature. However, to the best knowledge of the authors, no experimental data is available for such biogas compositions in HCCI engines especially without EGR.

The objective of the present work is therefore to investigate experimentally the composition effect of H_2 enriched biogas on required intake conditions, performances, and emissions of a HCCI engine by:

1. Studying CH_4/H_2 blends (first without CO_2 thus ensuring ignition and engine stability) and varying the CH_4/H_2 ratio to obtain a ratio representative of H_2 enriched biogas from digestion.

2. Adding then CO_2 to the mixture to see its effect on performances and emissions.

3. Finally targeting two types of possible biogas compositions from water buffalo manure digestion to see how they behave in the HCCI engine.

Experimental Methods

Experimental Setup

The experimental setup, Figure 1, consisted of a research PSA DW10 based engine test bench converted to single-cylinder operation. The engine was coupled with an electric motor and tested at 1000 rpm.

The engine characteristics are described in Table 1. The valve timings are given for a 0.1 mm lift.

Table 1. Engine specifications

Engine model	PSA DW10
Displaced volume	0.5 L
Stroke	88 mm
Bore	85 mm
Connecting Rod	145 mm
Compression ratio	18:1
Valves per cylinder	4
Exhaust Valve Open	40° CA BBDC
Exhaust Valve Close	6° CA ATDC
Intake Valve Opening	9° CA BTDC
Intake Valve Closure	23° CA ABDC
Piston type	Flat
Oil and Coolant temperature	94°C

To obtain the desired intake charge compositions a series of mass flow meters, as specified in Table 2**Erreur ! Source du renvoi introuvable.**, were used. The mixing of the biogas/air charge was ensured by the intake plenum (visible in Fig. 1) which also enables to damp pressure oscillations at the intake port. The combustion stability is expressed as the Coefficient of Variation (COV) of IMEP and it was kept below 5% for all operating conditions and blends.

To ensure ignition and stability, the intake heater was set between 220 and 353 °C depending on the operating conditions and to counteract heat losses along the intake line. The intake plenum was maintained at about 220°C using a heating cover. Those temperature settings finally lead to intake temperatures just above the intake valves between 169 and 216 °C ensuring ignition depending on the fuel blend studied. Without CO_2 at the intake

the intake pressure was set at 2 or 2.1 bar depending on the CH_4/H_2 ratio. When CO_2 was added at the intake, the intake pressure was increased up to 2.9 bar to ensure ignition and stability. Details about all the intake conditions depending on the $CH_4/H_2/CO_2$ blend are discussed in the results section.

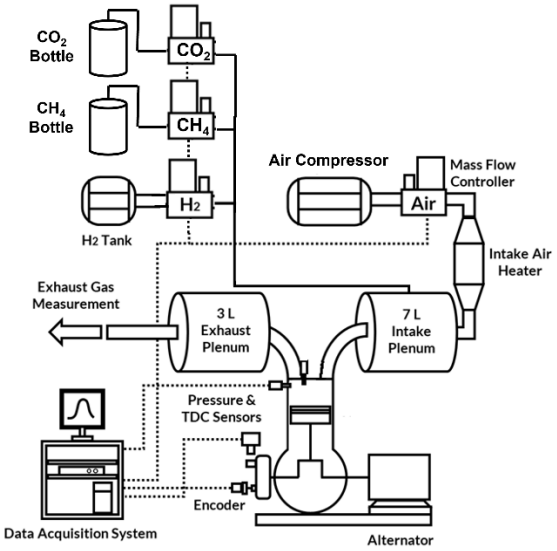

Figure 1. Schematic view of the engine test bench

Table 2. Mass flow Controllers

Gas Type	Flowmeter / Controller	Full Scale (NL/min)	Uncertainty
Air	Emerson F025S	1100	±0.5%
H_2	SLA5850	50	±1,0 %
CO_2	SLA5850	37	±1.0%
CH_4	SLA5850	100	±1.0%

The engine exhaust emissions were measured with the Horiba MEXA 7100DEGR. This exhaust gas analyser measures CO and CO_2 (non-dispersive infrared absorption analyser), NOx (chemiluminescence analyser), O_2 (magneto-pneumatic detector) and unburned hydrocarbon (HC) (flame ion analyzer) with a precision of 1 ppm for all gases. As these results are given on a dry basis (except for HC), a calculation considering the share of water vapour in the exhaust enables to readjust the content of each gas. Results are presented as specific emissions in g/kWh in the following.

Mixture compositions

Fuel properties, namely stoichiometric air/fuel ratio (AFR_{st}), Lower Heating Value (LHV), are calculated based on the CH_4 and H_2 share. For each composition, the equivalence ratio is fixed at 0.4 ± 0.02 by setting the mole fraction of air and fuel components. The stoichiometric air/fuel ratio is calculated based on the H_2 and CH_4 share in the biogas as follows:

$$AFR_{st} = Y_{H_2} \cdot AFR_{st_{H_2}} + Y_{CH_4} \cdot AFR_{st_{CH4}} \quad (1)$$

With $AFR_{st_{H_2}}$ and $AFR_{st_{CH_4}}$ equal to 34.33 and 17.23 respectively. The equivalence ratio is then obtained from the mass flowrates:

$$\phi = \frac{\dot{m}_{H_2+CH_4}}{\dot{m}_{air}} \cdot AFR_{st}$$

(2)

For the present study, different sweeps were carried out to understand the effect of the biogas composition. First, blends of CH_4/H_2 were investigated with different ratios. Then CO_2 was added to the CH_4/H_2 blends to represent H_2 enriched biogas as already studied numerically by Mariani et al. [19]. Table 3 and 4 summarize the different compositions investigated as well as some of their properties. For each blend the share of H_2 or CH_4 is presented as fuel share, as follows:

$$CH_4 or\ H_2\ fuel\ share = \frac{\dot{q}_{H_2\ or\ CH_4}}{\dot{q}_{CH_4}+\dot{q}_{H_2}}$$

(3)

With $\dot{q}$ the volume flowrate. When adding CO_2, its share is set as total share (considering CO_2, CH_4, H_2 and air):

$$CO_2\ total\ share = \frac{\dot{q}_{CO_2}}{\dot{q}_{CO_2} + \dot{q}_{air} + \dot{q}_{CH_4} + \dot{q}_{H_2}}$$

(4)

The share of each component, i.e. CH_4, H_2 or CO_2, in the biogas can then be defined as:

$$"i"\ biogas\ share = \frac{\dot{q}_i}{\dot{q}_{CO_2} + \dot{q}_{CH_4} + \dot{q}_{H_2}}$$

(5)

With *i* being either H_2, CH_4 or CO_2. From this CO_2 sweep, two different H_2 enriched biogas composition can be identified as biogas 3 and biogas 5 as already studied in [19]. Those two compositions are highlighted in grey in Table 4.

Table 3. Effect of the CH_4/H_2 ratio on fuel characteristics, with no CO_2.

CH_4 (% Vol.)	H_2 (% Vol.)	AFR_{st}	LHV ($MJ \cdot kg^{-1}$)	$LHV_{vol\ st\ mix}$ ($MJ \cdot Nm^{-3}$)
30	70	21.10	65.87	3.263
40	60	19.94	61.10	3.284
50	50	19.14	57.82	3.301
60	40	18.55	55.41	3.315
70	30	18.10	53.58	3.326
80	20	17.75	52.13	3.336
90	10	17.47	50.96	3.344

Table 4. Effect of the CO_2 on the LHV of the fuel. In gray, biogas 3 and biogas 5 investigated in [19].

CH_4 (% Vol.)	H_2 (% Vol.)	CH_4/H_2	CO_2 (% Vol.)	LHV ($MJ \cdot kg^{-1}$)
66.7	16.7	4	16.6	31.37
57.2	14.4	4	28.4	22.48
50.1	12.6	4	37.3	19.01
44.6	11.2	4	44.2	14.34
40.1	10.1	4	49.8	12.12
36.5	9.1	4	54.4	13.86
58.0	6.5	9	35.5	19.19
50.8	5.7	9	43.5	15.37
45.1	5.1	9	49.8	12.79
40.6	4.6	9	54.8	10.96

Post-Processing

For each operating condition, 100 consecutive cycles are recorded by a Kistler 6043A piezo-electric pressure transducer (accuracy of ± 2.0 %). All parameters such as maximum pressure gradient, maximum pressure angle, heat release rate, burnt mass fraction (BMF), indicated efficiency, etc. are individually computed for each cycle and then averaged over the 100 cycles. The IMEP is calculated using:

$$IMEP = \frac{1}{V_{cyl}} \int_0^{720\,CAD} -pdV \quad (6)$$

The indicated efficiency is calculated with the following equation:

$$\eta_i = \frac{W_{ind}}{Q_{fuel}} = \frac{IMEP \cdot V_{cyl}}{m_{fuel} \cdot LHV_{fuel}} \quad (7)$$

With fuel meaning the CH_4/H_2 mixture only (without CO_2). The combustion efficiency is computed only based on the CH_4 as follows:

$$\eta_{comb} = 1 - \frac{m_{HC} LHV_{CH_4} + m_{CO} LHV_{CO}}{m_{CH_4} LHV_{CH_4}} \quad (8)$$

Results

Effect of CH_4/H_2 ratio in the fuel

This section presents the results obtained for different fuel composition, i.e. me CH_4/H_2 ratio, at constant intake pressure, varying the intake charge temperature to achieve the autoignition.

Figure 2 shows the imep versus the intake temperature for different fuel compositions, starting from a 30% CH_4 70% H_2 composition, up to 80% methane share in the fuel. As the methane content increases, a higher intake charge temperatue is required to achieve the autoignition. Once the intake temperature attains the thereshold value for autoignition, the combustion was stable and the heat released by the combustion helped in sustaining it. For each fuel composition, the temperature was progresisvely increased to see the effect of the combustion phasing on engine performance. In fact, the autoignition moves closer to the TDC as the intake temperature is increased, for a constant fuel composition.

The imep shows a decreasing trend with temperature for all fuel compositions, with values ranging between 7.5 bar and 6.7 bar. This is the consequence of overlaying effects:

- Temperature increase reduces in-cylinder charge density at constant intake pressure.
- The heating value of the fuel on mass basis reduces as methane content increases, but for internal combustion engines, the heating value of the air-fuel mixture per unit volume must be considered, $LHV_{vol\ st\ mix}$ ($MJ \cdot Nm^{-3}$) in Table 3.

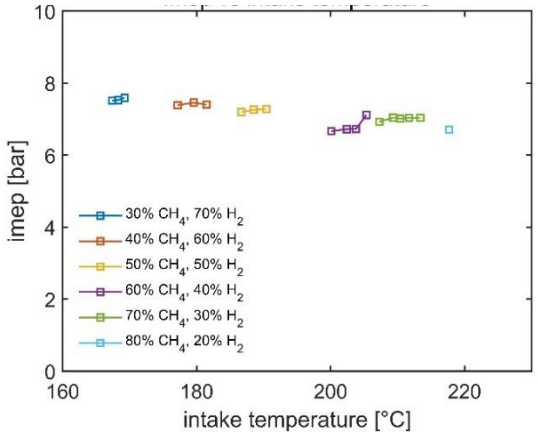

Figure 2 Effect of intake temperature on imep, for different fuel compositions, ER=0.4, p_{in}=2 bar.

The combustion showed very good stability already at the thereshold value for the auto-ignition, Figure 3, with a COV_{imep} always below 1.5%. For each fuel composition, the temperature increase from the thereshold value for auto-ignition caused a remarkable reduction of the COV_{imep}, with values down to 0.5%. The combustion was very stable for any fuel composition, with the 80% CH_4 20% H_2 fuel showing a slightly higher COV_{imep} of 1.5%. This fuel would have required a higher intake temperature, which was not possible to achieve due to an intake temperature limit of 220°C.

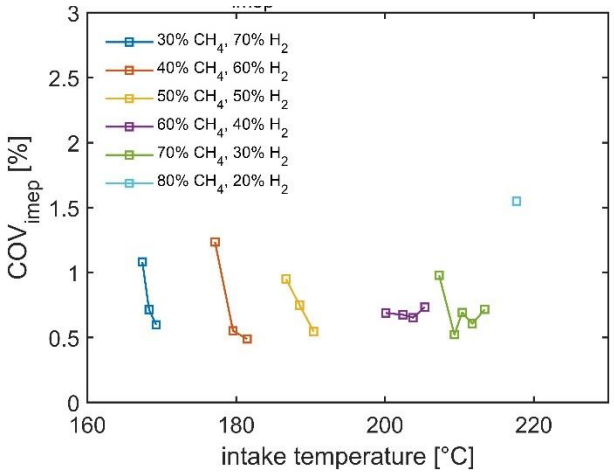

Figure 3 Effect of intake temperature on COV_{imep}, for different fuel compositions, ER=0.4, p_{in}=2 bar.

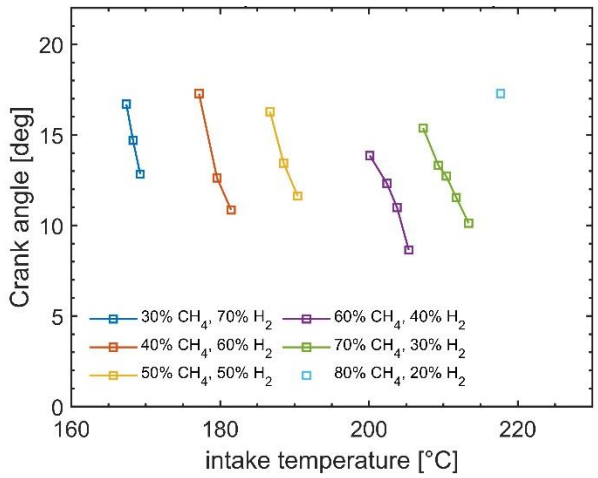

Figure 4 Effect of intake temperature on the crank angle of maximum pressure, for different fuel compositions, ER=0.4, p_{in}=2 bar.

Combustion stability is related to the combustion phasing, in Figure 4 represented by the crank angle at which the maximum in-cylinder pressure is attained. As the intake temperature increases, for each considered fuel, the combustion approaches the TDC and so does the crank angle of maximum pressure (a smaller CA value in Figure 4 indicated a position closer to the TDC). Each fuel composition shows a high-gradient trend, indicating that 1°C variation of the intake temperature strongly impacts combustion, particularly for fuel compositions operated at lower intake temperatures. As the combustion start approaches the TDC, higher in-cylinder pressure is attained, Figure 5. The increase of p_{max} with the intake temperature is remarkable and must be carefully considered for engine design and durability. For example, a variation of p_{max} up to 20 bar has been detected for the fuel composition 40% CH_4 60%H_2, caused by an intake temperature increase of 5°C. As methane content increases and higher intake temperature are required for autoignition, p_{max} also demonstrated an increasing tendency, except for the 80% CH_4 20%H_2 composition, which would have required higher intake temperature as previously described.

Not necessarily higher in-cylinder pressure results in higher engine efficiency, as depicted in Figure 6, where the indicated efficiency η_i is reported as a function of the intake temperature, for all fuel compositions. The indicated efficiency initially increases with the temperature until the optimal combustion phasing is attained, then reduces if the intake temperature is further increased. The trend is quite flat and suggests that it is not convenient to expose the engine to higher thermo-mechanical stresses for marginal gains in engine efficiency. It is sufficient to keep the maximum pressure crank angle around 12 CA degree ATDC to achieve the maximum measured indicated efficiency of 39.5%. The optimal phasing does not depend on fuel composition. In fact, being the combustion evolution controlled by chemical kinetic and not by flame front propagation, the process is very fast and there are no appreciable differences in the combustion speed between fuels in HCCI mode, resulting in a constant optimal combustion phasing.

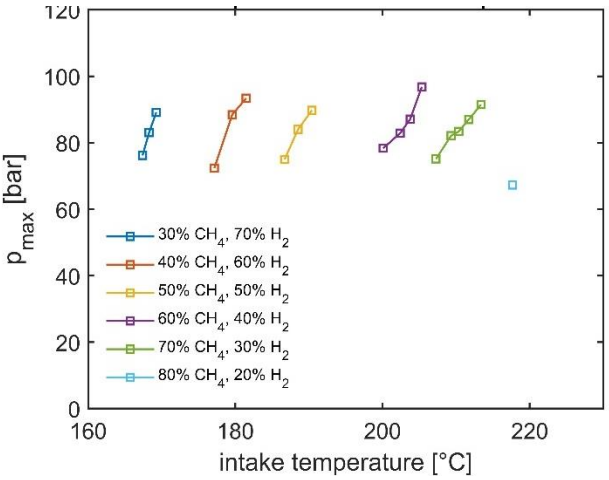

Figure 5 Effect of intake temperature on maximum in-cylinder pressure, for different fuel compositions, ER=0.4, p_{in}=2 bar.

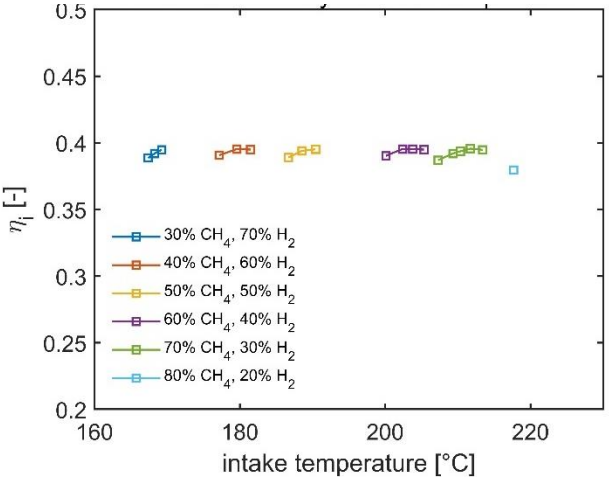

Figure 6 Effect of intake temperature on the indicated efficiency, for different fuel compositions, ER=0.4, p_{in}=2 bar.

Effect of CO_2 in the biogas

This section shows the results obtained for two fixed CH_4/H_2 ratios in the fuel: 4 and 9, adding progressively CO_2 to simulate the composition of biogas containing H_2. Experiments were performed at the constant intake temperature of 218°C, below the temperature limit of 220°C imposed by the experimental setup, varying the intake pressure to achieve the autoignition.

Figure 7 shows the imep versus the intake pressure for different biogas compositions, varying the CO_2 content in the biogas between zero and 55%. As the CO_2 content increases, a higher intake charge pressure is required to achieve the autoignition. For each biogas composition, the pressure was progressively increased to analyse the effect on the combustion process and on engine performance. As previously observed for the increase in the intake temperature, also an increase of the intake pressure promotes the autoignition, which moves closer to the TDC, for a constant fuel composition and intake temperature. The imep increases with the intake pressure because the in-cylinder charge density proportionally increases. There is also an effect on the combustion phasing which in turn affects engine performance. Results demonstrated that also with very high carbon dioxide content in the biogas, the HCCI combustion was possible, achieving up to 10 bar imep at 2.9 bar intake pressure for the biogas containing 55% CO_2. The combustion is stable for all biogas compositions, with COV_{imep} always below 1.5%, Figure 8, with a positive impact of the intake pressure on

combustion stability. In fact, COV_{imep} reduces as intake pressure increases, except for biogas with the highest CO_2 content where the combustion showed more cycle-by-cycle variation, an expected behavior due to the combustion weakening caused by CO_2.

As the CO_2 content increases, the required intake pressure for autoignition also increases for a constant intake temperature. Figure 9 shows the trend of the crank angle of maximum pressure with the intake pressure: as the intake pressure increases, the combustion phasing gets closer to the TDC, with smaller CA values.

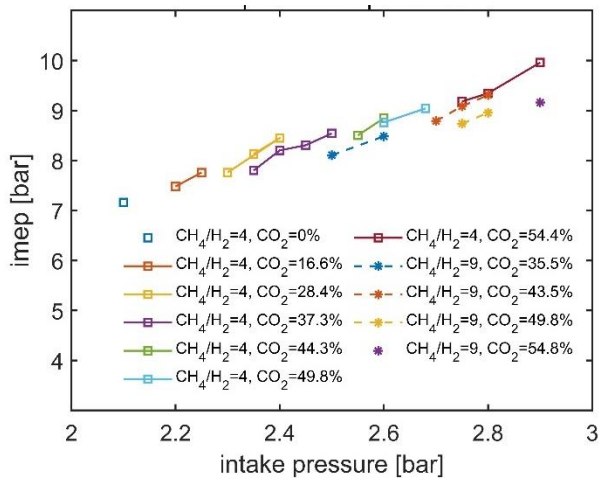

Figure 7 Effect of intake pressure on imep, for different fuel compositions, ER=0.4, T_{in}=218°C.

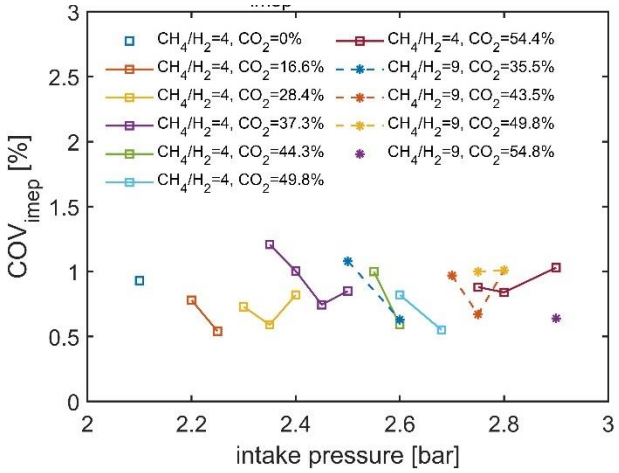

Figure 8 Effect of intake pressure on COV_{imep}, for different fuel compositions, ER=0.4, T_{in}=218°C.

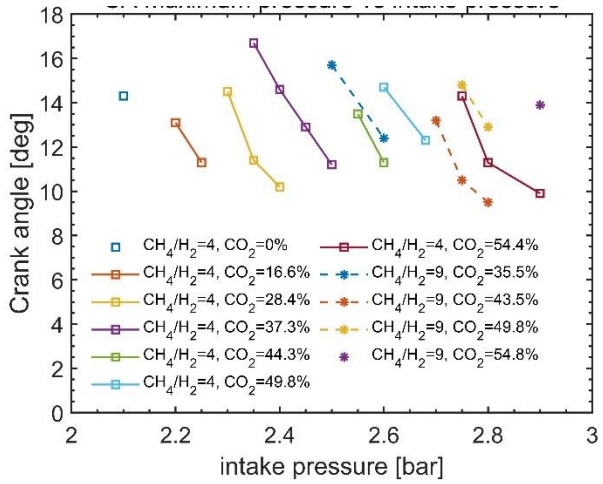

Figure 9 Effect of intake pressure on the crank angle of maximum pressure, for different fuel compositions, ER=0.4, Tin=218°C.

The maximum in-cylinder pressure with biogas, Figure 10, attains higher values than with CH_4/H_2 blends, because of higher intake pressures. In particular, the biogas CH_4/H_2=4 with 54.4% CO_2 shows a p_{max}=127 bar.

The engine efficiency η_i is reported in Figure 11, as a function of the intake pressure, for all biogas compositions. The indicated efficiency increases with the intake pressure, showing that the combustion should be phased closer to the TDC than what was previously obtained with methane-hydrogen blends without CO_2. The trend, in fact, suggests that a combustion phasing closer to the TDC would bring higher engine efficiency, but the pressure sweep was stopped when a maximum pressure gradient of 10bar/CAD was reached. This result is explained by the presence of CO_2 in the fuel, which slows down the reaction rates of fuel oxidation, increasing the combustion duration and requiring an anticipated combustion phasing for maximum engine efficiency. Maximum engine efficiency of 39.6% has been achieved.

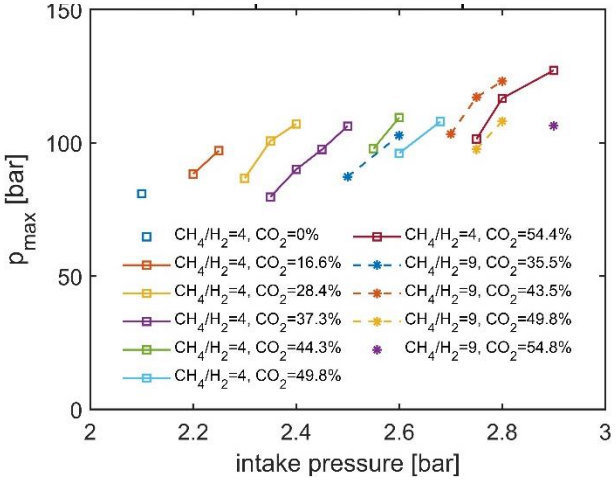

Figure 10 Effect of intake pressure on maximum in-cylinder pressure, for different fuel compositions, ER=0.4, T_{in}=218°C.

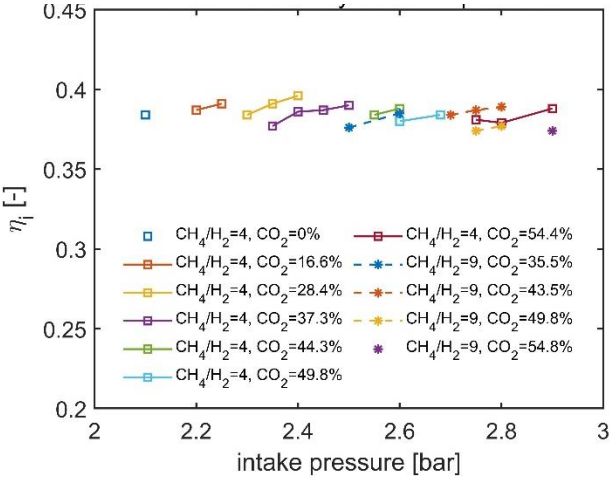

Figure 11 Effect of intake pressure on the indicated efficiency, for different fuel compositions, ER=0.4, T_{in}=218°C.

Emissions

Figure 12 shows the effect of intake temperature (a) and intake pressure (b) on specific NOx emissions, for different fuel compositions. Figure 12 (a) clearly shows that, fuels with higher CH_4/H_2 ratio emit less NOx, despite they require higher intake temperature for the auto-ignition. NOx formation strongly depends on maximum in-cylinder temperatures, and the calculated temperature evolution shows lower values as the methane content increases. This is due to the reduction of the imep delivered by the engine and to the burnt gas composition, which strongly depends on fuel composition. Figure 12 (b) shows a reducing trend of NOx emissions as the CO_2 content increases, despite higher intake pressure is required for the auto-ignition, demonstrating that the presence of carbon dioxide helps in limiting maximum in-cylinder pressure, hence reducing NOx emissions. In general, biogas containing H_2 have low NOx emission levels, below 0.1 g/kWh, much lower than CH_4/H_2 blends, also delivering higher imep.

CO emissions are reported for CH_4/H_2 blends in Figure 13 (a) as a function of the intake temperature and for the biogas in Figure 13 (b) as function of the intake pressure. There is no appreciable trend of CO emissions with the CH_4/H_2 ratio in Figure 13(a). In fact, CO emissions mainly depend on the equivalence ratio, which was constant for all tested fuels. For each fuel composition, as the intake temperature increases, a reduction of CO emission is observed due to a better oxidation of partial/intermediate combustion products promoted by the temperature increase.

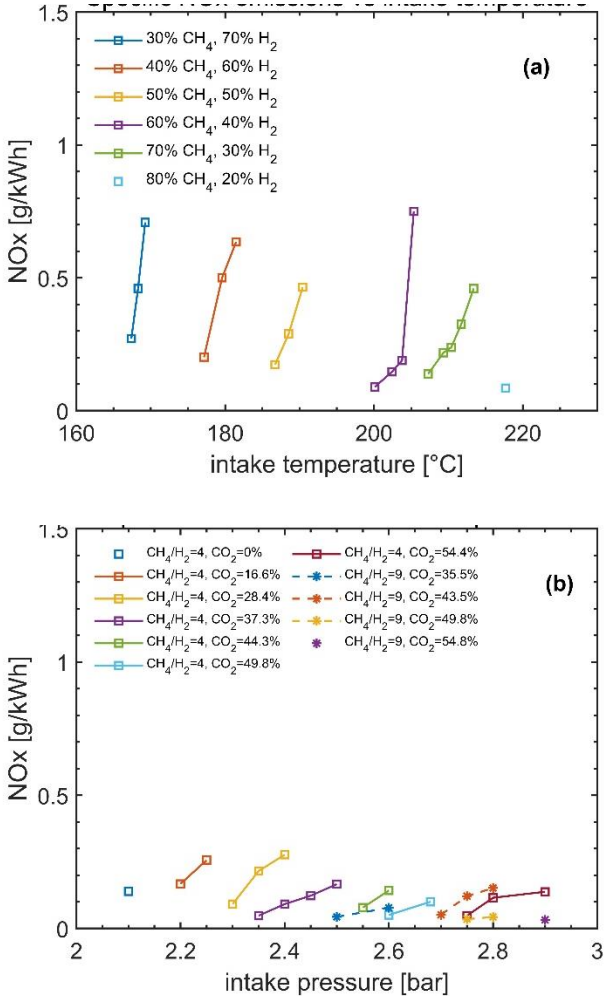

Figure 12 Effect of intake temperature (a) and intake pressure (b) on specific NOx emissions, for different fuel compositions.

There is a visible trend of CO emissions in Figure 13(b), with a reduction of CO as the CO_2 content in the biogas and intake pressure increase. CO exhaust levels are kinetically controlled and the CO concentrations in the immediate post-burned gases are close to equilibrium. However, during the expansion and exhaust strokes, as the burned gas cool, CO oxidation process may not remain locally equilibrated. The presence of CO_2 reduces the heat transfer and contains the cooling, helping the oxidation process, resulting in lower CO emissions.

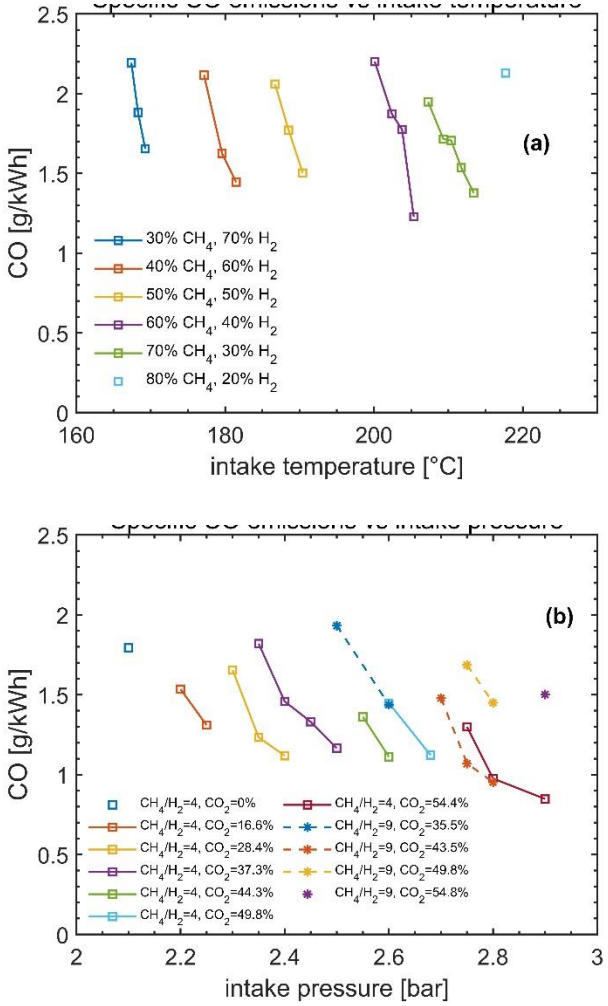

Figure 13 Effect of intake temperature (a) and intake pressure (b) on specific CO emissions, for different fuel compositions.

HC emissions are reported for CH_4/H_2 blends in Figure 14(a) as a function of the intake temperature and for the biogas in Figure 14 (b) as function of the intake pressure. HC emissions slightly increase with the CH_4/H_2 ratio, Figure 14(a). In fact, methane is a non-reactive hydrocarbon, and the amount of it which escape the main combustion process is not oxidated during the expansion and exhaust phases. It is therefore expected to have an influence of the methane content in the fuel on the unburned hydrocarbon emissions. For each fuel composition, as the intake temperature increases, a reduction of HC emission is observed because the amount of methane escaping the combustion process reduces, as the intake temperature increases.

HC emissions are reported for the biogas in Figure 14(b), which shows a reduction of HC emissions as the CO_2 content in the biogas and intake pressure increase. HC emissions, as previously described for CO, depend on the HC burnup in the exhaust. The presence of CO_2 reduces the heat transfer during the expansion and exhaust phases, helping the oxidation process, resulting in lower HC emissions.

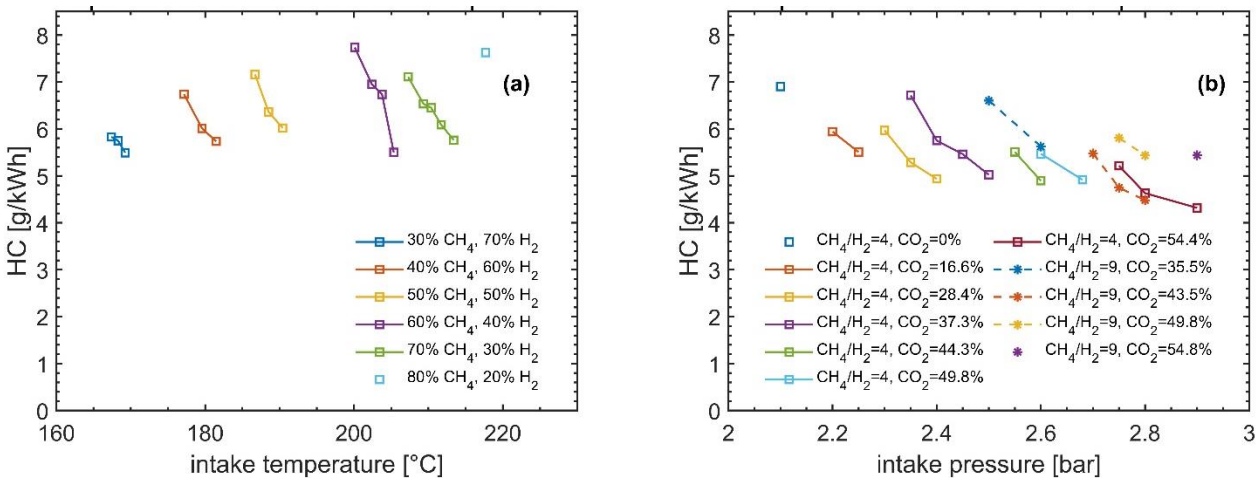

Figure 14 Effect of intake temperature (a) and intake pressure (b) on specific HC emissions, for different fuel compositions.

Summary

In this paper, the effect of the composition of CH_4/H_2 blends and biogas containing H_2 on HCCI combustion was investigated at fixed equivalence ratio. The study first focused on the effect of CH_4/H_2 ratio in the blend. Results showed that relatively high imep (>6.7 bar) with very high engine stability (COV_{imep} <1.5%) with indicated efficiency close to 40%. The main conclusions are:

- Increasing the methane share at constant equivalence ratio leads to lower imep because of higher intake temperature required (lower intake density) and higher fuel-air ratio even if the volume LHV increases with methane.
- The combustion phasing and maximum in-cylinder pressure are very sensitive to the intake temperature whereas the indicated efficiency is not. Thus, this parameter must be carefully set, without looking for optimal combustion phasing at any price to preserve the engine.

The second part of the study focuses on adding CO_2 to the biogas. When CO_2 was added to the mixture, it was necessary to set the intake temperature at 218°C, which was the engine test bench limit, to ensure ignition, as well as increasing the intake pressure. Varying the intake pressure was therefore the only way to affect promote the combustion process. Compared to the blends without CO_2, results show higher imep (up to 10 bar) and maximum in-cylinder pressure (up to 130 bar) due to the higher intake pressure required for auto-ignition and therefore due to higher energetic content in the cylinder. The stability of the engine remained very good with COV_{imep} also below 1.5%. The main conclusions to be drawn are:

- Increasing the CO_2 content in the biogas leads to a higher intake pressure required for autoignition thus leading to higher imep and maximum in-cylinder pressure.
- For a fixed composition, increasing the intake pressure leads to a better combustion phasing, i.e. closer to TDC thus possibly bringing higher indicated efficiency but with some constraints on the maximum pressure rising rate.
- Adding CO_2 slows down the fuel oxidation thus increasing the combustion duration and therefore requires anticipated ignition (through higher intake pressure and temperature) to target maximum efficiency.

NOx emissions results generally very low. CH_4/H_2 blends shows a dependence on the CH_4/H_2 ratio, with less NOx for blends with more methane despite a higher intake temperature required for the auto-ignition. Biogas containing H_2 shows a reducing trend of NOx emissions as the CO_2 content increases.

There is no appreciable trend of CO emissions with the CH_4/H_2 ratio, while the biogas containing H_2 shows a reduction of CO emissions as the CO_2 content in the biogas and intake pressure increase.

HC emissions for CH_4/H_2 blends slightly increase with the CH_4/H_2 ratio, as expected, being unburnt methane difficult to oxidize during the expansion and exhaust phases. The biogas shows a reduction of HC emissions as the CO_2 content in the biogas and intake pressure increase due to the reduction of the heat losses from the exhaust gases during the expansion and exhaust phases promoted by the presence of CO_2.

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