

Experimental Investigation of the Initial Stages of Flame Propagation in a Spark-Ignition Engine: Effects of Fuel, Hydrogen Addition and Nitrogen Dilution

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ABSTRACT

An experimental investigation was conducted in an optical mono-cylinder Spark-Ignition engine in order to explore the influence of the fuel and of the dilution rate on the initial stages of flame propagation. Images of flame radiation were acquired through the transparent piston crown with a high speed CMOS camera operating at 6000 frames/second. Experiments were performed under stoichiometric and lean conditions (0.8 of equivalence ratio), and two engine speeds (1200 rpm and 2000 rpm). The spark ignition timing was set at 30 (iso-octane) and 25 (methane) crank angle degrees before top dead center. Image acquisition was synchronized with in-cylinder pressure to allow simultaneous evaluation of the Indicated Mean Effective Pressure (IMEP) and of the heat release rate. Image post-processing was performed to obtain the temporal evolution of the projected flame area. A flame initiation criterion was defined and a flame initiation delay, based on the optical investigation, was determined for all mixtures and conditions. The effects of the fuel (pure iso-octane, methane and hydrogen blended mixtures) on the initial stages of flame propagation were studied. The effect of dilution, here simulated only by the addition of nitrogen, was also investigated. Results show that the initial flame development duration increases with the dilution rate and decreases with hydrogen addition in the air-fuel mixture. The standard deviation of the crank angle corresponding to a defined projected enflamed area (50% of the total visible area) varies with fuel composition and increases with dilution. The initial stages of flame propagation were found

to correlate very well with the laminar burning velocities obtained at atmospheric pressure and ambient temperature. The positive effect of hydrogen addition is more pronounced under diluted and lean conditions compared to stoichiometric conditions. The addition of hydrogen to avoid the drawbacks of dilution by burned gases (EGR or IGR) and of lean conditions can be one solution to provide sufficiently stable combustion and therefore low thermal NO_x formation in S.I. engines.

INTRODUCTION

Over the last few years, automotive manufacturers have focused on reducing pollutant emissions and on increasing engine efficiency. Optimization of the intake air-fuel mixture composition is one way to achieve these objectives especially in the case of Spark-Ignition (S.I.) engines. Natural gas can also be an attractive alternative fuel and has been widely studied as engine fuel. By operating with a diluted mixture, i.e by recirculating the exhaust gases or increasing the residual gases by changing the valve phasing, NO_x emission is reduced due to the lower combustion temperatures. There are however limitations on the level of dilution and of lean mixture that can ensure the smooth running of SI engines, whatever the fuel considered here, i.e gasoline or natural gas, since diluting the fuel-air mixture or operating with a lean mixture induces an increase in combustion duration and in cyclic variations which impair engine performance. Hydrogen has a wide flammability range, a low ignition energy, high auto ignition temperature and a high laminar burning velocity. Thanks to these properties, adding an

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amount of hydrogen to the fuel can extend the dilution and the lean engine operability limits, which is beneficial in reducing both emission levels and fuel consumption. Hydrogen addition can also improve the knock tendency in SI engines since it reduces the combustion period and inhibits fuel decomposition and hydroxyl radical production [1, 2].

HYDROGEN ADDITION

Numerical and experimental studies on the impact of hydrogen addition on SI engines fuelled by natural gas, methane or gasoline can be found in the literature [3,4,5,6,7,8,9,10,11]. The main results obtained are a reduction in combustion duration and an increase in the thermal efficiency. Apostolescu and Chiriac [7] studied the effect of hydrogen addition in a passenger car engine fuelled with gasoline. Kahraman et al. [8] conducted experiments in a four-cylinder S.I. engine to study the effect of hydrogen addition to methane (0%H₂, 10 %H₂, 20%H₂ and 30%H₂ by volume) on the engine's performance, emission, cylinder pressure and brake thermal efficiency. The cycle-by-cycle variation, HC emissions and the burning stages durations are decreased with the addition of hydrogen while thermal efficiency is increased ([7], [8]). Pana et al. [9], from results obtained with modelling and experimental investigations carried out on a single cylinder SI engine, concluded that the addition of hydrogen increases the thermal efficiency, the NO_x emission level, and the IMEP. The Brake Specific Fuel Consumption and the temporal duration of the initial and main phases of combustion are reduced when hydrogen is added.

HYDROGEN ADDITION COMBINED TO EGR

Dimopoulos et al. [12] investigated the H₂-Natural Gas blend combustion with and without EGR in an S.I. engine. Without EGR, the efficiency increases by approx. 2% when the H₂ content is 10% by volume. However, the engine-out NO_x emissions also increase. An opposite trend is observed for engine-out CO and THC with an increasing hydrogen fraction in the fuel mixture. Less spark timing advance was required for optimal efficiency since the hydrogen component in the fuel accelerates combustion. By combining H₂-Natural Gas blends and EGR, higher efficiencies with early spark timing with also very low engine-out NO_x can be reached. Alger et al. [13] investigated the effect of H₂ addition on EGR tolerance in a single-cylinder engine with two different compression ratio and load settings. They used both natural gas and gasoline as main fuels. They pointed out that a small amount of hydrogen is required to stabilize the engine at its EGR limit, defined as 5% of IMEP variations. At this EGR limit, the engine becomes stable at 0.2% H₂ by volume of the fresh air in the case of gasoline, compared to the 0.4% of H₂ needed for natural gas. Tully and Heywood [14] found that for low dilution levels, the addition of hydrogen has a small

reducing effect on the 0%-10% and 10%-90% combustion durations. This positive effect becomes more and more pronounced with the increase in dilution level. They also showed that hydrogen can extend the lean combustion limit since combustion deterioration occurs at a significantly higher dilution level when hydrogen is added. Similar conclusions on the increasingly positive effect of hydrogen addition with an increase in dilution level were reached in [15,16,17,18,19,20] in SI engines fuelled with different fractions of natural gas/gasoline-hydrogen mixtures and using different rates of EGR. Hu et al. [15, 16] deduced that the brake mean effective pressure decreases with an increase in the EGR rate and at small amounts of hydrogen addition (<20% in the fuel volume) but that it increases with a further increase in the hydrogen fraction. In the case of a high EGR rate (>10%), effective thermal efficiency increases while HC, CO and CO₂ emissions decrease with the increase in hydrogen fraction. Finally they observed that the reducing effect of hydrogen on the combustion duration is more pronounced on the flame development duration (the interval from ignition start to the crank angle degree CAD at which 10% mass fraction burned is reached).

HYDROGEN ADDITION IN LEAN-BURN CONDITIONS

Bauer and Forest [21] tested a one-cylinder S.I. engine with mixtures of hydrogen in methane of 0, 20, 40 and 60% by volume. They found that when 60% of hydrogen in the mixture is used, the partial burn limit decreases from 0.58 for pure methane to 0.34. Akansu et al. [22] experimented mixtures of 0, 10, 20 and 30% of hydrogen by volume in methane in a four-cylinder SI engine and observed an increase in NO emission values with the increase in hydrogen percentage in the mixture. However, in the case of lean burn (equivalence ratio E.R.<0.75) the NO values obtained are low and the brake thermal efficiency values increase with the increase in the hydrogen fraction. Hoekstra et al. [23] investigated the lean limit of natural gas in a SI engine with hydrogen addition. They found that 28% and 36 % of hydrogen in volume yields very low NO_x levels at a 0.625 Equivalence Ratio (E.R.). Moreover, the lean operating limit was significantly extended by increasing the hydrogen amounts in the case of natural gas as fuel [24, 20] or gasoline [25, 26].

All these studies are based on the assessment of global performance and pollutant emissions. Only a few studies have focused on the initial stages of flame propagation, such as Herweg and Maly [27] who developed a 1D model for flame kernel formation in S.I. engines. Conte and Boulouchos [28] carried out experiments by using an optical spark plug and ion sensors to study the flame propagation of gasoline mixtures enriched with reformer gas. Experiments on optical engines were also done by Meier et al. [29] and Rosati and Aleiferis [30] in a hydrogen direct injection SI engine.

Aleiferis et al. [31, 32] investigated the early flame development in a single-cylinder engine equipped with an optical access and fuelled with gasoline in lean-burn conditions. However, there is a lack of optical experimental investigations on the effect of hydrogen addition on diluted or lean conditions in a S.I. engine fuelled with methane or iso-octane.

The objective of this study is to provide new data on the initial stages of flame propagation in a S.I. engine fuelled mainly with iso-octane or methane and operating in diluted (with nitrogen) or lean conditions with different hydrogen fractions in the fuel.

EXPERIMENTAL DETAILS

OPTICAL ENGINE

The ‘transparent’ mono-cylinder S.I. engine used to perform the experiments has a four-valve pent-roof chamber with a displacement volume of 500 cm³ and a compression ratio of 9.5 [33]. The bore, stroke and connecting rod length are respectively 88 mm, 82 mm and 137 mm. To provide optical access inside the combustion chamber, the piston is elongated and equipped with a quartz window (diameter 66 mm). The cylinder head is bored and has two lateral windows. The optical access engine is driven by an electric motor at a fixed regime. A schematic view of the set-up is presented in Figure 1. The engine is equipped with an optical encoder mounted on the main shaft, giving 0.1 Crank Angle Degree (CAD) resolution. A conventional spark plug with an electrode spacing of 1mm was used. The engine is fired every sixth cycle to avoid residual gases and keep the equivalence ratio inside the cylinder constant. A timer card ensures synchronization of the various trigger signals to images and data acquisition systems. The engine sucks the air through a thermal mass flow-meter and all other flows are evaluated from this reference flow. Methane, hydrogen and nitrogen flows are measured using thermal mass flow-meters with an accuracy of $\pm 0.7\%$ for the instantaneous flow. Before the intake pipe, all gases pass through a plenum, to avoid pressure oscillations inside the intake port. To have a premixed mixture in the intake pipe, 4 injections per cycle were done in the case of iso-octane as fuel. The iso-octane quantity was measured by using a 0-5 kg/h Bronkhorst Coriolis mass flow meter with an accuracy of $\pm 0.1\%$ of full scale plus $\pm 0.2\%$ of flow rate.

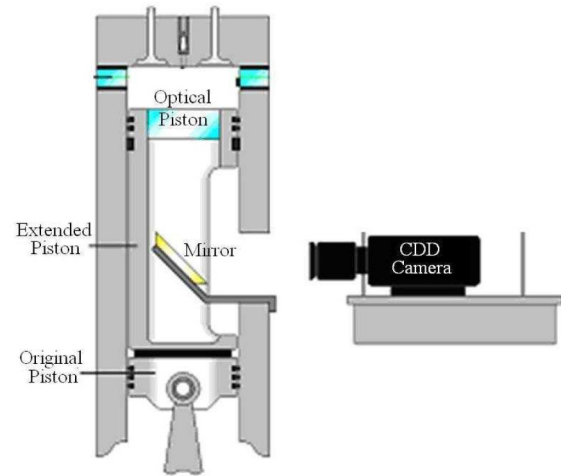


Figure 1. Schematic view of the set-up.

Cylinder pressure was recorded with a water-cooled AVL quartz pressure transducer connected to a charge amplifier at 0.1 CAD resolution. Before experiments, the transducer was calibrated with a Keller high pressure hydraulic calibrator. The linearity ($\leq \pm 0.6\%$) of the transducer was verified starting from a maximum pressure of 10MPa ($>$ maximum pressure in the engine). In this work, the absolute cylinder pressure was deduced by equalizing the in-cylinder pressure at 20 CAD after inlet valve opening timing to the intake mean absolute pressure. Thus, 100 consecutive ‘fired’ cycles of cylinder pressure data are acquired by a PC equipped with National Instruments acquisition board.

HIGH SPEED IMAGE ACQUISITION

Images of flame radiation were acquired through a 45° mirror mounted in the piston extension and with an APX-i2 intensified camera coupled with a 85 mm, f/1.4 Nikon lens. The frame resolution was 512×512 pixels² at 6000 frames per second. The magnification ratio was 0.159 mm/pixel. The gain was fixed at 3.7 V, with a gate width of a few microseconds, which is a compromise between radiation intensity and noise amplification. The initial combustion stage (from the spark to the maximum optical area) capture varies between 15 and 30 images depending on considered conditions (Equivalence Ratio, mixture composition and engine regime). The projected flame area, A_f was determined by using post-processing in Matlab after binarization and thresholding. Images including at least one pixel having 1 as a value (which corresponds to the luminous flame zone) and characterized by a distant ≥ 200 pixel (the radius of the quartz window is 208 pixels which corresponds to 33 mm) from the image center (the center of the quartz window) were not considered (image processing criterion). Thus the maximum luminous area A_{fmax} in this study was 3165 mm².

OPERATING CONDITIONS

The engine regimes were 1200 and 2000 rpm. The oil and coolant temperatures are set at 80°C. The intake pressure was set at 0.1MPa. The spark ignition was set at 30 CAD Before Top Dead Center (BTDC) for iso-octane mixtures and 25 CAD BTDC for methane mixtures in order to compare the initial stages of flame propagation at the same in-cylinder pressure and temperature values. The amount of hydrogen in the fuel was 0%, 20%, 40% and 60% by volume and the amount of dilution in the total mixture was 0% and 20% by volume. Both stoichiometric and lean conditions (E.R. = 0.8) were investigated. The different mixtures investigated are summarized in [Tables 1.a, 1.b, 1.c and 1.d](#). Hydrogen contents in the fuel in term of mass percentage γ and energy percentage E_{H_2} are also reported in [Tables 1.a, 1.b, 1.c and 1.d](#). For all results, the parameter α is defined as the mole fraction of hydrogen in the fuel (iso-octane or methane)-hydrogen blend:

$$\alpha = \frac{X_{H_2}}{X_{C_xH_y} + X_{H_2}} \quad (1)$$

And the parameter of dilution β is the mole fraction of nitrogen used for dilution in the total mixture:

$$\beta = \frac{X_{N_2-dilution}}{X_{C_xH_y} + X_{H_2} + X_{O_2} + X_{N_2-air} + X_{N_2-dilution}} = X_{N_2-dilution} \quad (2)$$

Where $X_{C_xH_y}$, X_{H_2} and X_{O_2} are respectively iso-octane or methane, hydrogen and dioxygen mole fractions in the mixture, X_{N_2-air} the mole fraction of nitrogen in the air mixture ($X_{N_2-air} = 3.78 \cdot X_{O_2}$) and $X_{N_2-dilution}$ the mole fraction of nitrogen used for dilution.

RESULTS AND DISCUSSION

CORRELATIONS

The frame acquisition rate, 6000 frames/second, induces 1 image every 1.2 CAD and 2° CA at respectively 1200 and 2000 rpm. Examples of the flame evolution are given in [Figure 2.a](#) for the C8H18-air mixture (Condition 4) and [Figure 2.b](#) for the C8H18-H2-air-N2 mixture (Condition 11) at stoichiometry. Δt_s is the time after spark ignition.

The flame propagation velocity is higher for the Condition 4 mixture than for Condition 11. The projected flame surface A_f was normalized with respect $A_{f,max}$:

$$A_{norm} = \frac{A_f}{A_{f,max}} \quad (3)$$

The evolution of A_{norm} versus the CAD was fitted with a fourth degree polynomial in order to obtain, for a fixed A_{norm} , the corresponding CAD or Δt_s . The CAD corresponding to a defined value of A_{norm} , for example 0.1 and 0.5, respectively called $\theta_{0.1}$ and $\theta_{0.5}$, was plotted versus the CA50 (corresponding to 50% of the accumulated heat release) in order to verify the image quality, the correct post-processing and the validity of the 2-D flame projected approach. An example for CH4-air-N2 (condition 17) is shown in [Figure 3](#). In this case, $\theta_{0.5}$ corresponds to 6% of total heat release, while for $\theta_{0.1}$ the cumulated heat release was <1% of the total heat release. In [Figure 3](#), the CAD = 0 corresponds to the Top Dead Center position. High correlation coefficients were found for all values of A_{norm} , thus confirming that in S.I. engines, the combustion development depends strongly on the initial stages of flame propagation.

(See [Figure 2a](#) and [Figure 2b](#) after last section of paper)

The correlation coefficients between $\theta_{0.1}$, $\theta_{0.5}$ and CA50 ($Corr_{\theta_{0.1}-CA50}$ and $Corr_{\theta_{0.5}-CA50}$) are reported in [Tables 1.a, 1.b, 1.c and 1.d](#) for all cases. As expected, these correlation coefficients were found to be higher when the defined A_{norm} is 0.5 rather than 0.1 and were not dependent on the mixture composition, thus no effect of hydrogen addition or nitrogen dilution was observed.

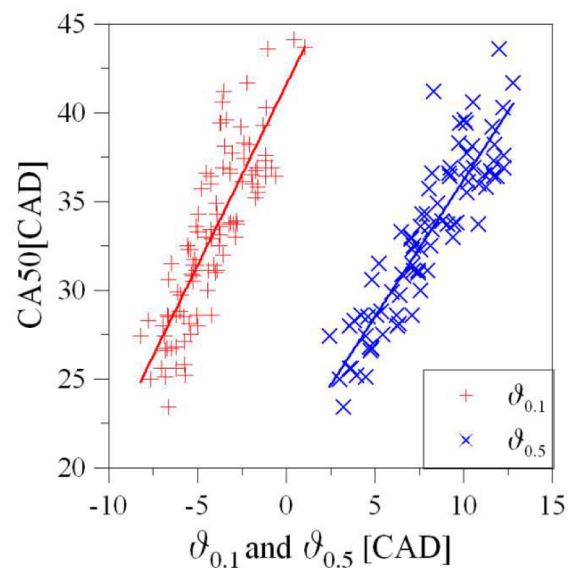


Figure 3. $\theta_{0.1}$ and $\theta_{0.5}$ versus the CA50 (condition 17).

In this study, the minimum of the equivalent flame radius r_f , determined by assuming a spherical shape was 10 mm ($A_{norm} = 0.1$). As Herweg and Maly [27], Natarajan et al. [34] and Bradley et al. [35] indicated, no influence of the spark on the flame growth can be considered: this is confirmed by the strong correlation found between $\theta_{0.1}$ and CA50. The CAD corresponding to the defined value of A_{norm} , from 0.1 to 0.9, i.e. θ_i ($i = 0.1 \dots 0.9$) were found to correlate well with the maximum of the cylinder pressure, P_{max} and the CAD corresponding to this maximum, for all investigated conditions (Tables 1.a, 1.b, 1.c and 1.d). The mean value of the correlation coefficients, called $Corr_{\theta_{0.5}-P_{max}}$ and $Corr_{\theta_{0.5}-\theta P_{max}}$, are respectively 0.86 and 0.88 except for four investigated conditions. For the stoichiometric mixtures, i.e. Conditions 3, 17 and 23, $Corr_{\theta_{0.5}-\theta P_{max}}$ has negative values, and, for condition 5, a low positive value, 0.225. These different values are linked to the spark ignition timing.

The correlation coefficients $Corr_{\theta_{0.5}-IMEP}$ between $\theta_{0.5}$ and the IMEP present both negative and positive values. These values with the corresponding percentage of cumulated heat release of $\theta_{0.5}$ ($Cum_{\theta_{0.5}}$) are reported in Tables 1.a, 1.b, 1.c and 1.d. It can be seen that all conditions with nitrogen dilution or lean mixtures and without hydrogen addition are characterized by negative values of $Corr_{\theta_{0.5}-IMEP}$. This behavior is expected, since for these conditions, the burning velocities are lower than for the other conditions while the spark ignition timing is the same. Finally, ignition occurs close to the spark timing, giving the maximum brake torque for conditions with low absolute values of $Corr_{\theta_{0.5}-IMEP}$ (conditions 2 and 15). This fact reflects the stability of the IMEP since a small fluctuation in the initial stage of flame propagation does not strongly affect IMEP values.

Aleiferis et al. [31] did not find a correlation between CAD5 and IMEP for stoichiometric C8H18-air conditions in a stratified-charge SI engine for two spark advances. However, they found a strong correlation between CAD5 (corresponding to 5% of the accumulated heat release) and IMEP, characterized by a negative value of the correlation coefficient, for lean C8H18-air conditions.

FLAME EVOLUTION

A criterion of the beginning of flame propagation was chosen at $A_{norm} = 0.1$ because the flame is no longer affected by the ignition energy and for all investigated conditions, till $A_{norm} \sim 0.2$, a small heat release was observed (<1%). The maximum value of A_{norm} considered in the following is $A_{norm} = 0.6$ since it is reached in all conditions and is sufficient for our purposes, since this work focuses on the initial stages of flame propagation. As can be seen in Tables 1.a, 1.b, 1.c and 1.d, the value $A_{norm} = 0.5$ corresponds to a percentage of the cumulated heat release higher than 5% in all investigated

conditions. In this study the reported data are averaged data on 100 cycles for each condition.

I. C8H18-H2-air-N2 mixtures, E.R. = 1 at 2000 RPM (conditions 1, 2 and 3)

Figure 4.a presents the temporal evolution of A_{norm} for C8H18-H2-air-N2 mixtures, E.R. = 1 at 2000 rpm. The flame reaches $A_{norm} = 0.1$ after 1.79 ms, 1.69 ms and 2.14 ms for conditions 1, 2 and 3 respectively. The effect of hydrogen addition is evident from the beginning: its accelerating effect increases with the flame growth. Nitrogen dilution has the opposite effect on the flame growth. The difference ϕ_i , at a given A_{norm} is defined as:

$$\phi_i [ms] = \Delta t_{s_i} - \Delta t_{s_ref} \quad (4)$$

Where i refers to the number of the condition (referred in Tables 1.a, 1.b, 1.c and 1.d) and ref , the reference conditions, (condition 1 in this section). ϕ_2 and ϕ_3 , plotted versus A_{norm} in Figure 4.b present an opposite trend. However, the effect of 20% of nitrogen dilution is more pronounced than the effect of 60% of hydrogen addition in the fuel. At $A_{norm} = 0.5$ for example, flame propagation in the reference Condition is 9.7% slower than that in Condition 2, and 24.5% faster than that in Condition 3. Moreover, this difference in terms of time of flame propagation becomes 13.1% slower (than condition 2) and 48.9% faster (than condition 3) at CA50. The enhancement effect of hydrogen on the combustion rate is therefore greater in the initial stages of flame propagation than in the later stages, whereas the effect of dilution remains high in later stages.

(See Figure 4a and Figure 4b after last section of paper)

(See Table 1a after last section of paper)

II. C8H18-H2-air-N2 mixtures, E.R. = 1 at 1200 RPM (conditions 4-11)

Similar to Figure 4.a and Figure 4.b, the temporal evolution of A_{norm} and the evolution of ϕ_i versus A_{norm} , are plotted in Figure 5.a and Figure 5.b respectively. The reference for the calculation of ϕ_i in this section was Condition 4 (C8H18- air). A clear difference between the conditions with and without dilution is observed. For $A_{norm} = 0.5$, $\phi_5 = 1.64$ ms, $\phi_{10} = -0.39$ ms and $\phi_{11} = 0.96$ ms inducing a relative difference in percentage of respectively 52.4%, -12.45% and 30.6% with respect to the reference case (condition 4). These relative differences become 73.32%, -12.1% and 42.1% respectively at CA50. Coherent conclusions are reached in this case for all conditions: the accelerating effect of hydrogen is mainly during the initial stage while dilution has a more prolonged

decelerating effect. In addition, the accelerating effect of hydrogen, at late stages, increases with the dilution amount.

(See [Figure 5a](#) and [Figure 5b](#) after last section of paper)

(See [Table 1b](#) after last section of paper)

III. C8H18-H2-air-N2 mixtures, E.R. = 0.8 at 1200 RPM (conditions 12-15)

The reference condition used to estimate ϕ_i is Condition 4 in this section. For $A_{norm} = 0.5$, the relative difference is 34.5% and 14.4% for conditions 12 and 15 respectively. Thus, [Figure 6.a](#) and [Figure 6.b](#) present the temporal evolutions of A_{norm} and the evolution of ϕ_i versus A_{norm} for these conditions. At 1200 rpm, the effect of 20% of nitrogen dilution is greater than the effect of lean combustion (E.R. = 0.8) on the initial stages of flame propagation. Furthermore the accelerating effect of hydrogen is more pronounced than in the case of dilution at later stages (CA50) since the relative differences become 27.14% and only 2.19% for conditions 12 and 15 respectively, at CA50. Thus for lean conditions, the accelerating effect of hydrogen increases during combustion and proved to be very marked in the late stages of combustion (CA50).

(See [Table 1c](#) after last section of paper)

(See [Figure 6a](#) and [Figure 6b](#) after last section of paper)

IV. CH4-H2-air-N2 mixtures, E.R. = 1 at 1200 RPM (conditions 16-23)

In this section, the reference condition is Condition 16. Unlike the conditions considered in section II, conditions with dilution and hydrogen addition are close to the reference conditions (16) when the temporal evolution of A_{norm} is plotted ([Figure 7.a](#)). This can be explained by the fact that in the case of methane as the main fuel, the same percentage of hydrogen addition by volume is more important in terms of mass or energy than in the case of iso-octane as reported in [Tables 1.a](#), [1.b](#), [1.c](#) and [1.d](#). As shown in [Figure 7.b](#), for a small amount of hydrogen (20%) and without dilution (Condition 18), the effect of hydrogen in terms of ϕ_{18} is almost constant in the initial stages of flame propagation. Thus, even when methane is used as the main fuel, the accelerating effect of hydrogen is almost on the initial stages when there is no dilution and increases in the later stages for conditions with dilution. For example considering Condition 22, the relative difference is -30.95% for $A_{norm} = 0.5$ and becomes -34.2% for CA50, while the effect of dilution remains important even in late stages of flame propagation (CA50).

It is very interesting to consider the behaviour of Condition 23 since the graph of these conditions (with 60% of hydrogen

in the fuel mixture and 20% of nitrogen dilution) is superposed to the graph corresponding to the reference conditions, i.e, without dilution and hydrogen addition. Thus 60% of hydrogen addition is needed to compensate for the effect of 20% of nitrogen dilution, which is in very good agreement with the laminar burning velocity results [36] since 60% of hydrogen addition was needed in the case of 20% of nitrogen dilution to obtain the same burning velocity (32cm/s) as in the case of methane-air mixture without dilution (33.96cm/s).

The main conclusion concerning flame evolution is that the positive (enhancement) effect of hydrogen becomes more noticeable and stronger under lean and diluted conditions, which is in very good agreement with the conclusions obtained in [14,15,16,17,18,19,20,21,22]. Furthermore, it was found that for mixtures with stoichiometric conditions, the effect of hydrogen on the flame development is more pronounced in the initial stages than in the late stages of combustion (CA50), which is consistent with the results reported in [5, 16].

(See [Figure 7a](#) and [Figure 7b](#) after last section of paper)

(See [Table 1d](#) after last section of paper)

CORRELATION BETWEEN FLAME EVOLUTION AND LAMINAR BURNING VELOCITY

I. Methane as main fuel

S_L is the laminar burning velocity at atmospheric pressure and ambient temperature for the mixture conditions 16-23. S_{L-ref} is the S_L for the stoichiometric methane-air conditions. The values of S_L are given by a correlation based on experimental results presented in a recent study [36]. A dimensionless difference ΔS_L is defined as:

$$\Delta S_L = 100 \cdot \left(\frac{S_L - S_{L-ref}}{S_{L-ref}} \right) \quad (5)$$

Another dimensionless parameter is also defined as:

$$\psi_i = \frac{\left[(\phi_i \text{ at } A_{norm} = 0.5) - (\phi_i \text{ at } A_{norm} = 0.1) \right] - \left[(\Delta t_{s-ref} \text{ at } A_{norm} = 0.5) - (\Delta t_{s-ref} \text{ at } A_{norm} = 0.1) \right]}{\left[(\Delta t_{s-ref} \text{ at } A_{norm} = 0.5) - (\Delta t_{s-ref} \text{ at } A_{norm} = 0.1) \right]} \quad (6)$$