H₂, CH₄, C₂H₆ Flame Speed Measurement under Firing Engine Conditions

Arturo De Risi and Domenico Laforgia

Università degli Studi di Lecce
Dipartimento di Scienza dei Materiali

Abstract

A procedure to measure the real flame speed in stratified charge engine under working engine conditions has been developed. A single cylinder stratified charge engine with a centered square head-cup, a compression ratio of 10.8, a squish ratio of 75% and a nominal swirl ratio of 4 operating at 800 rpm and propelled with H₂, CH₄ and C₂H₆ has been used. The gaseous fuels were injected via a Ford AFI injector, originally designed for air-forced injection of liquid fuels on purpose modified. Pressure measurements in the engine cylinder and in the injector body coupled with optical measurements of the injector poppet lift and images of the fuel jets and of the combusted products provided both qualitative and quantitative information about in-cylinder processes. Engine cylinder and injector body pressure data, and related images were acquired in the same cycle to be directly compared. All fuels were injected in such a way to obtain an ignitable mixture in the vicinity of the spark plug, and a spark timing of 166° ABDC was used. Combustion was fast and repeatable for all fuels (COVpmax=1.2%, 4.0%, 2.2% for H₂, CH₄ and C₂H₆, respectively). Direct injection, high squish, high swirl and square cup in the engine head produced a fast mixing and a high degree of mixture uniformity within the head-cup at ignition time, thus the assumption of a spherical shape for the flame front was made possible, although it was not essential for the procedure illustrated in this study. Results well agreed with values available from previous experiments.

Introduction

The combustion process in spark-ignition engines takes place in a high pressure, temperature and turbulent flow field generated during the intake stroke and modified during the compression stroke. The importance of the turbulence to the engine combustion process has been recognized long time ago. In fact, experiments where the turbulence originated by the intake stroke was eliminated have shown that the rate of flame propagation decreases substantially [1]. The fact that the propagation rate is related to the engine flow characteristics rises the problem of how to use data obtained at different conditions, which means, for most of the cases, low pressure and temperature, not turbulent flow and uniform composition of the air/fuel mixture. Laminar flames in premixed fuel/air/residual gas mixtures are characterized by a laminar flame speed $S_L$ and by its thickness $δ_L$, commonly defined as $δ_L=\Delta L/\Delta S_L$, where $\Delta L$ is the molecular diffusivity. Yet, turbulent flames can be characterized by the root mean square velocity fluctuation, by the turbulence intensity $u'$ and the integral length scale of the turbulent flow ahead of the flame. Most of the characterizing parameters are not linearly dependent with the species concentration gradient within the flame, mixture
transport and thermodynamic properties, which makes very difficult to find an analytical way to extrapolate flame speed data to other condition from which they have been measured. However, it is possible to find in literature many experimental correlations relating the laminar flame speed to pressure and temperature, as well as the turbulent flame speed to the laminar flame speed and to the turbulence intensity [2,3,4,5,6], but they are significantly different from each other and none has been shown to be reliable over a wide range of operating conditions. The situation has become even more crucial when to better the control on pollutant formation, the engine design moved from premix charge to port injection and direct injection introducing in such a way also a disuniformity in the burning charge. From what has been said, it appears clear the need to develop a procedure for a reliable direct measurement of the turbulent burning velocity in engines. The present study investigates the possibility to measure the turbulent burning velocity in a stratified-charge two-stroke spark-ignition engine.

Experimental setup

The experimental setup was the same used in [7] and it is here reported only for the reader’s convenience. The engine used was a two-stroke single-cylinder engine based on a CRF-48 crankcase. The head of the engine was designed to provide broad optical access to the combustion chamber. This configuration of the engine had been used in previous studies [7,8,9]. An extended Bowditch-type piston, with an oil control ring at the lower end and three bronze-impregnated Teflon sealing rings at the top, was used to provide an almost oil-free environment within the cylinder. The engine was ported with six exhaust ports, equally spaced through 180°, and six equally spaced intake ports on the opposing side of the cylinder. The exhaust port timing was ±67° BDC and the intake port timing was ±54° BDC. The intake ports were fitted with inserts directing the flow 30° from the cylinder radius and 30° upward to provide swirl. The engine parameters are summarized in Table 1. Air was supplied to the engine by an air compressor through a mass flowmeter and a 0.0324 m³ surge tank located two meters upstream of the engine. The engine exhaust gases were discharged to the building extraction system. A sketch of the cylinder head and of the gas injector is shown in Fig. 1. The head of the engine contained a centered cup of square cross section with a side length of 36.6 mm and a height of 26.4 mm. This arrangement provided a squish ratio of 75% and a compression ratio of 14.6 from BDC or 10.8 from the nominal exhaust port closing, i.e. the coincidence of the flat piston plane with the upper point of the exhaust ports. At TDC the piston was within 1.5 mm of the bottom of the head. The injector was mounted on the top of the cup, along the cylinder axis, and the nozzle exit was 1.5 mm below the upper wall of the combustion chamber. The head was equipped with a pressure transducer communicating to the combustion chamber through a passage measuring 4 mm in diameter and length. The spark plug was mounted in the middle plane of the combustion chamber with the spark gap facing directly upward, towards the injector. The spark gap was 2 mm from the cylinder axis. The ignition system consisted of a 12 VDC power supply, a GM LX-301 ignition module, an ignition coil, and a J-gap spark plug (Champion RS15LYC). The total ignition energy supplied by this system was about 60 mJ and the duration of the spark, measured by Schlieren images, was about to 4.2 ms. The in-

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore</td>
<td>82.60 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>114.30 mm</td>
</tr>
<tr>
<td>Connecting Rod Length</td>
<td>254.00 mm</td>
</tr>
<tr>
<td>Intake Port Timing</td>
<td>±54.00 °BDC</td>
</tr>
<tr>
<td>Exhaust Port Timing</td>
<td>±67.00 °BDC</td>
</tr>
<tr>
<td>Squish Area</td>
<td>75%</td>
</tr>
<tr>
<td>Clearance Height</td>
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</tr>
<tr>
<td>Cup Size</td>
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</tr>
<tr>
<td>Cup Depth</td>
<td>26.40 mm</td>
</tr>
<tr>
<td>Compression Ratio (Port Closing)</td>
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</tr>
<tr>
<td>Compression Ratio (Geometric)</td>
<td>14.60 mm</td>
</tr>
</tbody>
</table>

Table 1: Engine parameters
jector was a Ford Air-Forced Injector [10]. It comprises a mixing chamber, referred to as the injector body, a spring mounted poppet assembly and two electronically controlled solenoid valves that originally metered the fuel and the air into the injector body. The poppet is held by a spring and opens when the differential pressure overcomes the preload of the spring. The duration of the air valve opening, the mass of the poppet, and the characteristics of the injected gases determine the duration of the injection and the poppet oscillations. Once the air entrance is closed, the injector body is vented to assure a constant initial differential pressure across the delivery valves. For the purpose of the present work, the injector was used as a gas injector. The fuel path was disabled and gaseous fuels were injected through the air path from high pressure cylinders. Because of the relatively high pressure used for the injection, the injector valve stem was plated with a 13 \(\mu\)m thick Chromium layer to reduce the gap between the valve stem and the injector and avoid undesirable fuel leaks in the cylinder. The operating conditions were 800 rpm, overall fuel-air ratio of 0.398, 0.430, 0.424 for hydrogen, methane and ethane, respectively (using thermodynamic conditions at the exhaust port closing), and the spark timing was 166º ABDC. To assure complete scavenging of the cylinder, each firing cycle was preceded by five motored cycles with no fuel injection. The air flow through the cylinder corresponded to a scavenging ratio of 1.0.

Data Acquisition System

Engine cylinder pressure and injector body pressure were measured with a water-cooled Kistler 7061 and a Kistler 6004 transducers and Kistler model 504 charge amplifiers. Data were acquired using a 16 bit A/D converter at one crank angle degree intervals and stored in a personal computer. Motoring traces were used to verify the integrity of the measurement procedure [11]. Shadowgraph images were acquired to visualize the flame contour. A single pass, recollimated shadowgraph system was employed because of the simplicity with which the sensitivity can be adjusted about the zero point [12]. A schematic of the shadowgraph setup is shown in Fig. 2. A Lexel model 95 Argon Ion Laser operating at a wavelength of 514.5 nm and with an output power of about one watt was used. The laser beam was split by an acousto-optic modulator, and the second order beam was filtered by a spatial filter (10x microscope objective and 10 \(\mu\)m pinhole) and collimated before passing through the engine head. On the opposite side of the engine the beam was recollimated by a system of two lenses with a magnification of 3 to 2 and projected on a diffusing screen. The images of the screen were digitally recorded by an Electim EDC 1000 HR CCD camera (147x213 pixels) and stored in a personal computer. An interference filter centered at 514.5 nm was used in front of the camera to reject the combustion light. The duration of the laser pulse was 30 \(\mu\)s and the camera exposure time was set at 1 ms, the lowest available setting, to prevent the blur effect due to combustion light leaks through the interference filter. The timing signals for the injector, spark, acousto-optic beam modulator, camera and pressure measurement system were controlled by a personal computer equipped with a programmable counter-timer module. The timing pulses were received from a BEI shaft encoder. A schematic of

![Figure 1: Cylinder head: optical access and gas injector.](image)
the data acquisition system with the data flow exchange between devices indicated by arrows, is shown in Fig. 3.

The system synchronization was double-checked using a graduated flywheel with a stroboscope lamp and analyzing all the synchronization signal traces on an oscilloscope. The synchronization between shadowgraph images and pressure traces was ensured by asynchronous serial communication, through the RS-232 interface of the personal computers controlling the image acquisition and the pressure traces acquisition systems. Time delays necessary to drive all the devices involved in data acquisition were generated by a Stanford Research System, Inc. Delay/Pulse Generator. The mass of fuel injected was determined as the difference between the mass of fuel delivered and vented. Fuel was supplied to the injector from a pressurized vessel through a line fitted with a muffler and an accumulator to dampen pressure pulsation present in the line, and a 7 µm filter to remove particles that could foul the injector. The flow rate of the fuel supplied to the injector was measured by a sonic discharge flowmeter. The stagnation temperature and pressure were measured in the line, upstream of the sonic orifice discharge flowmeter, pressure was also measured on downstream side of the flowmeter in such way to be sure that sonic conditions were achieved. The flow rate of the fuel vented from the injector was measured by a Singer 806 positive displacement meter which had been also used for the calibration of the sonic orifice discharge flowmeter. Pressure transducers for both cylinder and injector body pressure measurements, were calibrated with a dead weight tester.

Figure 2: Schematic of the shadowgraph setup.

Figure 3: Schematic of the data acquisition system control and of the data exchange flow.
Flame Speed Calculation

Flame speed can be evaluated by imposing the equivalence between the heat release evaluated from cylinder pressure data analysis and the one obtained from flame photographs analysis. The first time a similar procedure was used by [13] to a premixed charge engine but it has never been applied to DISC engines. Cylinder pressure changes are related to piston motion, heat released by the combustion process, heat transfer to the chamber walls and to the flow into and out of crevice regions. Usually this last contribution is negligible, but for the engine used it represented a significant term, and, therefore, needed to be modeled. The behavior of this system can be represented by the following energy balance:

\[
dQ_{ch} = dL + dU_s + dQ_{ht} + \sum(h_i m_i)
\]

where \(dQ_{ch}\) is the heat due to the chemical energy released by the combustion process, \(dL\) represents the piston work, \(dU_s\) is the change in sensible energy, \(dQ_{ht}\) is the heat transfer to the chamber wall and \(\sum(h_i m_i)\) represents the energy related to the fuel injection and to the crevice flow. The accuracy with which the heat released by combustion can be predicted by this energy balance depends on how adequately each term in equation (1) can be quantified. In the present study the sensible energy has been evaluated using the mean charge temperature given by the ideal gas law, thus, neglecting mass losses, the first two terms in equation (1) can be expressed as:

\[
dL + dU_s = \frac{k}{k-1} \rho dV + \frac{1}{k-1} V d\rho
\]

where \(k=\frac{c_p(T)}{c_v(T)}\) and \(c_p(T)\) and \(c_v(T)\) are the specific heat obtained as mass average of the equilibrium product at the given temperature \(T_g\).

To calculate the heat transfer term convection and radiation have been considered as shown in the following equation:

\[
dQ_{ht} = \left[A h_e \left(T_g - T_w\right) + A \sigma \varepsilon \left(T_g^4 - T_w^4\right)\right] dt
\]

where \(A\) is the chamber surface, \(h_e\) is the heat transfer coefficient given by equation (4), \(T_g\) and \(T_w\) are the mean gas and the walls temperature respectively, \(\sigma\) is the Stefan-Bolzman constant (5.67E-8 W/m²K⁴) and \(\varepsilon\) is the emissivity factor used to take in account the burned gases departure from black-body behavior. In the present study the value of \(\varepsilon = 0.1\) estimated by Hottel and al. [14] for engine combustion gases has been used. To calculate the heat transfer coefficient the Woschni correlation has been used [15]

\[
h_e = 3.26 B^{-0.2} p^{0.8} T^{-0.55} W^{0.8}
\]

where \(B\) is the piston bore in [m], \(p\) is the cylinder pressure in [kPa], \(T\) is the gas temperature given by the ideal gas law in [K] and \(w\) is the average cylinder gas velocity in [m/s] given by the following expression:

\[
w = C_1 \bar{u} + C_2 (p - p_m) \frac{V_d T_r}{V_r p_r}
\]

where \(\bar{u}\) is the mean piston speed, \(V_d\) is the displacement volume, \(p_n\), \(V_r\), \(T_r\) are pressure, volume and temperature of the working gas at exhaust port closing and \(p_m\) is the motored
cylinder pressure at the same crank angle as \( p \). \( C_1=2.28+0.308R_s \) and \( C_2=3.24E-3 \) [m/(sK)] are constant and \( R_s \) is the nominal engine swirl ratio at TDC. To calculate the heat exchange due to crevice flow, a sufficiently accurate model is to consider a single aggregate crevice volume, where the gas is at the same pressure as in the combustion chamber, but at a different temperature. Since crevice regions are quite narrow, the assumption that crevice gas temperature equals walls temperature was made and the following expression was obtained:

\[
dQ_{cr} = \frac{k}{k-1} \frac{T^*}{T_w} V_{cr} dp
\]

where \( V_{cr} \) is the total crevice volume, \( T^* \) is equal to gas temperature if \( dp \) is greater than zero and to walls temperature \( T_w \) otherwise. If the pressure data analysis illustrated above is coupled with flame geometry data, a better insight into the behavior of engine flame is obtained. Schlieren images of the flame well define the front position of the turbulent propagating flame only if the engine flow field does not introduce a substantial distortion of the flame from spherical shape. If flame distortion is present schlieren images have to be replaced by at least two simultaneous and, if possible, orthogonal to each other images of the flame front taken with an appropriate planar imaging technique. In the present study flames were rather spherical, thus schlieren images were used to define the flame front, and the volume of the burned gases was given by the volume of the sphere which radius was equal to the one of the best-fit circle to the flame front silhouette. Defining the burning velocity as the volume of the unburned air-fuel mixture consumed per unit of time divided by the surface area of the flame, i.e. the surface of the sphere defined above, it is possible to write the following equation relating the heat release obtained from pressure data analysis to flame speed:

\[
\frac{dm_u}{dt} = \frac{dm_b}{dt} = \frac{1}{H_i} \frac{dQ_{ch}}{dt} = \rho_u A_f S
\]

where \( m_u \) is the mass of the unburned air-fuel mixture which assuming a combustion efficiency \( \eta_b=1 \) it is also equal to the mass of burned air-fuel mixture \( m_b \), \( H_i \) is the lower heat value of the fuel, \( \rho_u \) is the unburned gas density, \( A_f \) is the effective mean flame surface calculated with the procedure indicated below and \( S \) is the actual flame speed. Schlieren images of the flame kernel were digitally acquired and the effective flame projected area was evaluated. From the average of 10 values of the projected area, the diameter of the equivalent size circle was calculated and \( A_f \) was given as the surface of the sphere having that particular diameter. The value of the flame speed was obtained from equation (7) using the measured values for pressure and volume.

**Results and Discussion**

Typical cylinder pressure plots versus crank angle are shown in Fig.4 for hydrogen, methane and ethane. A comment is necessary about their shape. Since the total energy for the three fuels was the same, because of the early onset of the \( H_2 \) profile it would be expected a significantly higher cylinder pressure than for the other fuels and a later convergence of the three pressure traces after TDC. The departure from the expected profile is due to the greater mass losses through the rings and overall through the head windows of the single cylinder engine because of the faster and earlier growth of the pressure for the \( H_2 \) case. The effect of mass losses can also be seen and quantified in slightly less than 20% of the charge in the integrated heat release plots of Fig.4. Typical schadowgraph images of fuels jets are shown in Fig.5 for the three fuels. Images are acquired at different cycles, but no cycle to cycle variation was observed in the jet structure during the injections. Although in-
jects start and finish at somewhat different crank angles, the calculated [7] instantaneous momentum flow rates for the three fuels are quite different and the interpretation of the shadowgraph images become uncertain when comparing jets of extremely different molecular weights such as those of H\textsubscript{2} versus CH\textsubscript{4} and C\textsubscript{2}H\textsubscript{6}, the analysis of the jets images leads to some understanding. The broading of the H\textsubscript{2} jet as well as the high density ratio avoid it to collaps resulting in a faster mixing with the surrounding air. Yet, shadowgraph images of the mixture after the end of the injection and before ignition for all the three fuels do not show any identifiable structure thus suggesting the presence of a fairly uniform mixture. Mixture uniformity is also suggested by the almost spherical shape of the burned gases as shown in the digitalised images of Fig.5. The flame contour, shown in Fig.6, was identified using a find edges routine after the image has been processed using a median filter with a radius of three pixels. The best fit circle to the flame silhouette has been defined in the present work as the circle which diameter, d, is given by the following relation:

\[ d = \sqrt{\frac{4A_s}{\pi}} \]  

where \( A_s \) is the area of the flame image shown as solid black in Fig.6. The scaling factor from pixels to square meter was obtained by direct measurement of the number of pixels contained in an image of a surface of one square centimeter. Looking at the images of Fig.5 which have been acquired at the same crank angle but at different cycles it is also possible to get an idea of the strong cycle to cycle variation of the early stage of the flame propagation.

Flame speed obtained with the above procedure and its standard deviation are reported in Tab.2 as well as the corresponding experimental value available in literature [16,17,18,19,20]. Data on the laminar flame speed under engine-like conditions are very limited. Eggolopoulos et al. [16] presented measurements of laminar flame speed of H\textsubscript{2}/O\textsubscript{2}/N\textsubscript{2} as a function of stoichiometry, charge dilution and pressure, but the pressure range between 0.02 MPa and 0.225 MPa is below the level of interest in engine applications. This is also the case with data obtained by Andrews et al. [17]

<table>
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<tr>
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<th>Calculated Flame Speed [m/s]</th>
<th>Standard Deviation [m/s]</th>
<th>Laminar Flame Speed [m/s]</th>
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<tbody>
<tr>
<td>Hydrogen</td>
<td>1.1424</td>
<td>0.1737</td>
<td>&lt; 3</td>
</tr>
<tr>
<td>Methane</td>
<td>0.6524</td>
<td>0.1392</td>
<td>0.35-0.60</td>
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<tr>
<td>Ethane</td>
<td>0.8454</td>
<td>0.1895</td>
<td>0.40-0.80</td>
</tr>
</tbody>
</table>

Table 2: Calculated flame speed, standard deviation letterature values [16, 17, 18, 19, 20]
for the laminar flame speed of methane-air mixtures and by Gibbs et al. [18] for the ethane-air mixtures. To obtain at least an estimation of the laminar flame speed under engine conditions, the data were extrapolated using equation (9) suggested by Milton et al. in [19] where $S_u$, $P$ and $T$ are the laminar flame speed (in centimeters per second), at $P$ (in atmospheres) and $T$ (in Kelvin), whereas $S_{ur}$ and $T_r$ are the corresponding values at a known condition. The parameters $\alpha$ and $\beta$ were determined experimentally [19] and fitted, for the purpose of the present work, with a fourth order polynomial. Extrapolated data were in good agreement with those obtained by Halstead [20] under engine conditions. Equation (9) applied to spark conditions ($P=1.172$ MPa, $T=612$ K) and for possible equivalent ratios present around the spark plug at the ignition instant gives the range of laminar flame speeds indicated in table 2.

Considering that all of the fuel but only part of the air is in the head cup at the spark time, the estimated effective equivalence ratio near the spark at the spark time for the cases studied, should be not less than $0.316 \times 1.54 = 0.487$ for $H_2$, $0.363 \times 1.54 = 0.559$ for $CH_4$ and $0.329 \times 1.54 = 0.506$ for $C_2H_6$, but locally it could be higher than these values. From these consideration the corresponding laminar flame speed would be faster than 3 m/s for $H_2$, between 0.35 and 0.60 m/s for $CH_4$ and between 0.4 and 0.8 m/s for $C_2H_6$.

$$lg(S_u) = lg(S_{ur}) + \alpha lg\left(\frac{T}{T_r}\right) + \beta lg(p)$$

In particular, whereas the calculated flame speed for the hydrogen case seems to be slightly underestimate, the values evaluated for methane and ethane well approach the data extrapolated using equation (9). A possible reason of this can be seen in the extremely high hydrogen molecular diffusivity. Thus, it is reasonable to think the hydrogen-air mixture to be leaner than the attempt value proposed above and therefore, to have a lower flame speed than the expected as it turned out. If the calculated flame speed of 1.1424 m/s is assumed to be the real hydrogen burning rate, then the hydrogen-air mixture must have a fuel/air equivalent ratio of about 0.4 which indeed is also a reasonable value. A further comment is necessary on the effect of the turbulence on the calculate flame speed for the

![Figure 5: Schadowgraphs images of hydrogen, methane and ethane jets and of the flame contour at 168° ABDC, 171° ABDC and 169° ABDC for hydrogen, methane and ethane respectively.](image)
three fuels. To make possible a comparison between the burning rate measured in the DISC engine used and the laminar flame speed data, taking also into account the limited optical access to the combustion chamber, determination of flame speeds in the present study have been done only at the beginning of the combustion event (i.e. flame kernel diameters lower than 2 cm) where turbulence does not affect significantly the combustion rate. Nevertheless, because of the high cycle to cycle variation observed at the beginning of the combustion event and of the approximations introduced with the pressure data analysis the uncertainty of the obtained data could be significant.

Summary and Conclusions

The combustion of three different gaseous fuels in a spark ignition stratified-charge engine was investigated under overall lean stoichiometry to have a better insight of the flame speed characteristics. The procedure adopted gave encouraging results been able to evaluate reasonably the burning rate. Although the procedure presented does not present apparent restrains to be applied, the analysis of flame speed characteristics during the main energy release of the combustion event has to be still completed.

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References


