

Influence of the Flow Field on Flame Propagation in a Hydrogen-Fueled Internal Combustion Engine

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ABSTRACT

Flame propagation in an optically accessible hydrogen-fueled internal combustion engine was visualized by high-speed schlieren imaging. Two intake configurations were evaluated: low tumble with a tumble ratio of 0.22, corresponding to unmodified intake ports, and high tumble with a tumble ratio of 0.70, resulting from intake modification. For each intake configuration, fueling was either far upstream of the engine, with presumably no influence on the intake flow, or the fuel was injected directly early during the compression stroke from an angled single-hole injector, adding significant angular momentum to the in-cylinder flow. Crank-angle resolved schlieren imaging during combustion allowed deducing apparent flame location and propagation speed, which were then correlated with in-cylinder pressure measurements on a single-cycle basis.

In a typical cycle, flame shape and convective displacement are strongly affected by the in-cylinder flow. For homogeneous fueling with low tumble, the flame is convected little, growing without significant wrinkling with a shape that is quite symmetric in the vertical plane. In contrast, in the other cases the flame is convected and stretched. Ensemble averaged results show that for fully homogeneous conditions the increase in tumble ratio from 0.22 to 0.70 results in increased flame growth and shorter combustion duration. For the stratified mixture, two regimes were observed: Early in the combustion, the flame grows faster for high intake-induced tumble, while during middle and late combustion low tumble yields a faster burn rate with an overall shortest combustion. On a single-cycle basis, early flame growth strongly correlates with the crank angle at

which 5% of the fuel mass is burned. Convection is characterized by the displacement of the flame's projected area centroid, revealing that the multi-cycle centroid cloud spreads with time and that the cycles follow different paths corresponding to their flame speed: typically the slow cycles stay near the ignition point and at the top of the centroid cloud. For direct injection, the ensemble average centroid speeds are relatively high in the beginning and then slowly decrease. In contrast, with homogeneous fueling the centroids have nearly constant convective speed.

INTRODUCTION

Hydrogen-fueled internal combustion engines (H2ICEs) with near-zero emissions levels and efficiencies higher than modern diesel engines are attractive power plants for automotive applications [1,2,3]. Current research efforts in advanced H2ICEs are devoted to achieving brake thermal efficiencies higher than 45% while keeping levels of NOx emissions low [2, 3]. To accomplish this goal, modern H2ICEs use direct injection (DI) fueling strategies [4,5,6,7]. In typical DI-H2ICE operation the fuel is injected into the cylinder after IVC. This increases the volumetric efficiency and mitigates knock, pre-ignition and backfire, negative effects associated with hydrogen port-fuel injection [4, 8].

Flame propagation in spark ignited (SI) engines fueled by hydrocarbon fuels is well studied and the controlling factors are mostly understood [9,10,11]. Alefeiris et al. [11], for example, report that for stratified lean operation the flame development is affected by the fuel distribution and flow field around the spark plug, residual gas, spark energy, and the heat losses to the electrodes. Among these factors, the community has devoted special attention to the effects of fuel

distribution and flow field on combustion stability, especially when the engine operates near the lean limit [12,13,14,15,16,17,18,19,20]. Lean operation, either in premixed or DI engines, can have significant benefits in efficiency and emissions [9, 21]. However, close to the lean limit, combustion becomes “unstable”, that is, there is large cycle-to-cycle variation (CCV) in performance determining parameters like burn duration. To achieve stable operation near the lean limit, charge stratification by direct injection and enhancement of the in-cylinder turbulence have been used. In the case of charge stratification, mixture preparation enters as an additional process complicating combustion control [16].

In hydrocarbon-fueled SI engines combustion is typically divided into four stages: ignition and flame initiation, initial flame kernel development, fully developed turbulent flame propagation, and flame termination [22]. All stages contribute to the overall combustion process; however, it has been shown that the first two stages are the ones that determine the quality¹ of the cycle and have direct influence on the CCV [23]. For example, several studies show that there is a strong correlation between the crank angle corresponding to 5% mass fraction burned (X_b) and the IMEP [10, 24]. The crank angle of $X_b=5\%$ (CA $X_b=5\%$) choice is subjective: smaller values of X_b are difficult to measure due to the finite accuracy of in-cylinder pressure measurements. Nevertheless, CA $X_b=5\%$ represents a time span corresponding to the initial flame kernel development. The limitations of pressure measurements in determining the earliest crank angle at which X_b defines the quality of a cycle have been overcome by imaging the temporal evolution of the flame kernel. Imaging allows resolving very small X_b and has proven to be very useful in characterizing the combustion process and its associated temporal and spatial CCVs [10, 14, 18, 20, 24, 25, 26, 27].

Concerning the influence of in-cylinder flow, it is well established that in gasoline (SI) engines the intake-induced in-cylinder flow has significant effect on engine performance and emissions [16, 28]. “Tumble”, a coherent rotation of the in-cylinder charge around an axis perpendicular to the cylinder axis, is believed to influence combustion characteristics by breaking up into small-scale turbulence late during the compression stroke [28,29,30]. This increases turbulence intensity and thereby the turbulent flame speed, which promotes faster combustion, extends the lean limit of stable engine operation, lowers knock propensity, and reduces cyclic variability. However, high convective velocity also increases heat transfer to the relatively cold walls of the combustion chamber [9]. In hydrogen engines at low loads, due to the relatively long combustion duration in ultra-lean

operation, tumble break-down may be particularly important for combustion and hence engine performance.

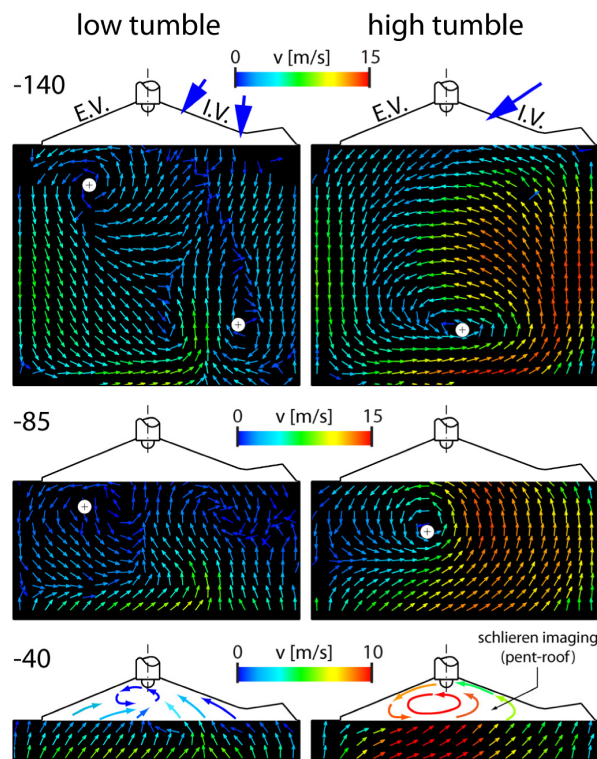
Recent studies in our engine show that in DI-H2ICEs the intake-induced flow field also has a significant influence on the final fuel distribution, even though the injection event completely alters the pre-injection flow [31,32,33]. Due to the relevance of these results for the present study, an abbreviated sequence of flow-field measurements during the compression stroke is shown in Figure 1. Two cases were compared: The unmodified intake-port configuration (“low tumble”), and a tumble-enhanced modified version (“high tumble”). Without injection, at bottom dead center (BDC)² for the low tumble case the flow consists of two weak, counter-rotating vortices. The in-plane tumble ratio (TR), a measure of the coherent rotational motion, is 0.22. During the compression stroke, the flow develops an upward-welling pattern as the tumble ratio gradually approaches zero near top dead center (TDC). Conversely, for the high-tumble configuration, a single, well defined vortex describes the flow field throughout most of the compression stroke. At BDC the TR is 0.70, decreasing to 0.25 at -30°CA . With fuel injection, the fuel jet, issued from an angled single-hole nozzle just after intake-valve closure, impinges on the liner and forms a wall jet. Entraining the cylinder charge during the compression stroke, the wall jet imparts a tumble-like global rotation onto the charge. Due to the injector targeting, injection-induced convection and intake-induced tumble flow counter-act, hence the high tumble case is in general retarded in the development of its circular convection pattern. The mixture distribution before ignition, investigated separately [31], is significantly different between the two cases.

Hydrogen with low ignition energy, high diffusivity, wide flammability limits, and high stoichiometric laminar flame speed [8], behaves very differently from a typical hydrocarbon fuel. Such properties impede the existing knowledge about flame propagation for hydrocarbon fuels to be simply extrapolated unchecked to hydrogen engines. Consequently, since flame propagation is at the heart of efficiency and emissions, we need experimental data on flame propagation in H2ICEs. Visualization of flames in hydrogen-air mixtures in constant volume vessels is reported in the literature for atmospheric and higher pressures [34,35,36]. These studies have focused on the characterization of laminar burning velocities with the main goal of finding correlations that could be used in combustion submodels of multidimensional simulations of H2ICEs. To complement such studies under stationary conditions, the present work investigates flame propagation in an operating engine with a realistic fuel distribution and flow field. Also, while previous studies [6, 31, 32, 33] concerned the mixture-

¹ Here, the quality of a cycle is measured by its relative IMEP, i.e. a good cycle has high IMEP while a poor cycle has low IMEP.

² The crank-angle convention used here assigns 0°CA to TDC, i.e., crank angles throughout the compression stroke are negative.

a) without fuel injection



b) with fuel injection

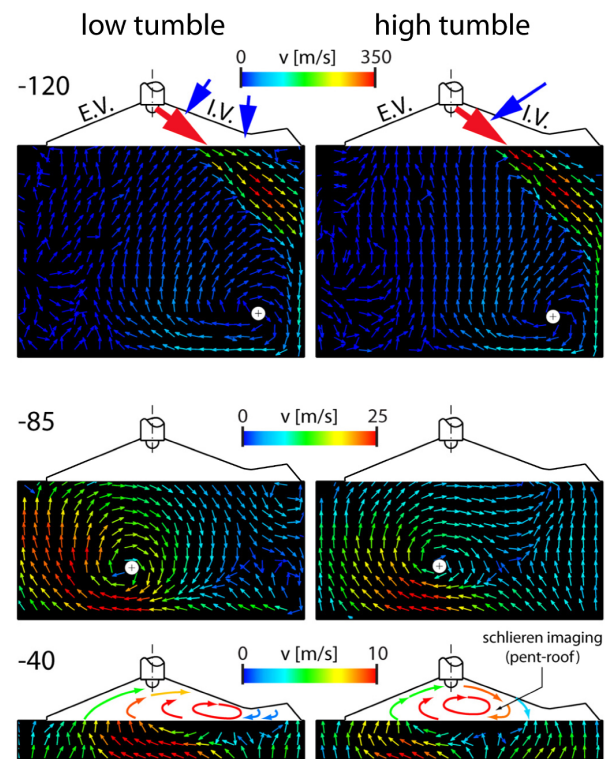


Figure 1. In-cylinder flow in the central symmetry plane during compression, measured by PIV in previous work [33]. Blue arrows indicate the main direction of intake-flow momentum for the two different intake configurations. (a) Without fuel injection, (b) with fuel injection just after IVC, indicated by the red arrows. Note that the velocity color-scale is different for each vector plot. At -40°CA the likely flow pattern in the pent roof, which was not part of the measurement domain, is shown schematically.

preparation phase only, the present work investigates the influence of the in-cylinder flow field on combustion as well. To this end, flame propagation was visualized by high-speed schlieren imaging, complemented by classic fast cylinder-pressure measurements. “Movies” of flame propagation, from ignition to 50°CA after ignition, were recorded for homogeneous and stratified mixtures (from fuel DI) for the same two intake configurations that were investigated in the flow measurements shown above. By correlating these image series with the pressure measurements, the effects of enhanced tumble were evaluated. The present measurements are also very suitable to validate future numerical simulations of the combustion process in DI-H2ICEs and will be added to the Engine Combustion Network [37], an internet data-base for validation of engine simulations.

EXPERIMENT

OPTICAL ENGINE AND HYDROGEN INJECTOR

Experiments were performed using a passenger-car sized optically accessible single-cylinder engine adapted to operate with hydrogen. Schematics of the relevant optical and

mechanical components are shown in Figure 2. The engine uses a four-valve four-stroke single-cylinder GM research head with a pent roof combustion chamber. This head has two intake ports, one for each valve, both straight and parallel to each other forming an angle of 40° with respect to the fire deck (horizontal). The engine head has not been modified for hydrogen fueling other than the injector. A flat-topped piston was used for this study.

Some engine parameters are given in Table 1. Hydrogen was supplied at 100 bar directly into the combustion chamber via a solenoid injector from Westport Inc. As shown in Figure 2a, the injector nozzle has a single hole with a diameter of 1.46 mm at a 50° angle with respect to the injector axis. Figure 2b shows the injector location, centrally located between the four valves, and the nozzle direction, aiming towards the intake between the two valves and downward. Hence the jet was aligned with the mirror-symmetry plane of the combustion chamber.

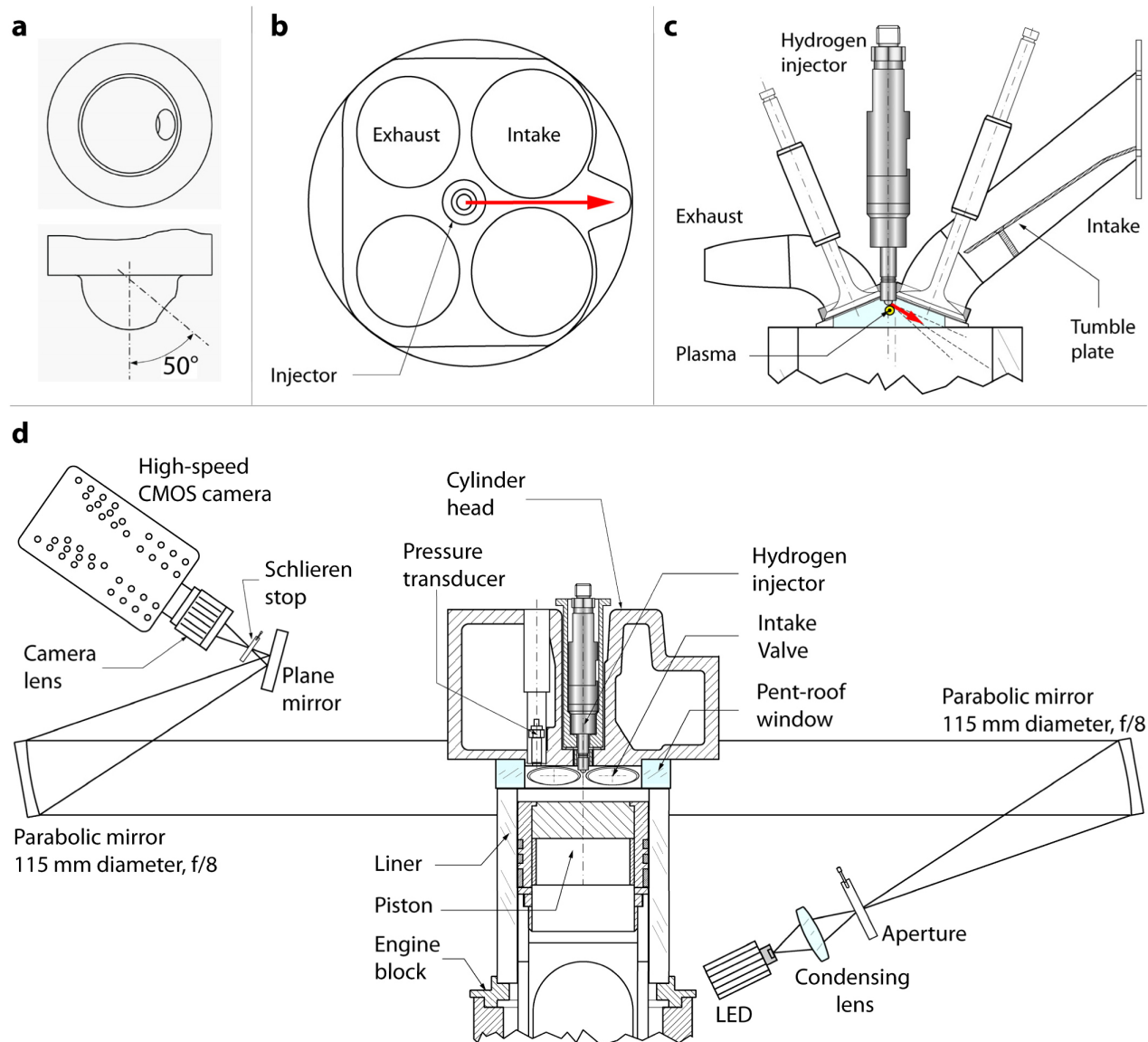


Figure 2. Schematics of engine hardware. (a) Geometry of the single-hole injector nozzle, (b) location and targeting of injector with respect to the combustion chamber, (c) pent roof window, and position of laser-plasma ignition, and (d) upper engine and schlieren-imaging system.

Table 1. Engine Specifications

Bore	92 mm
Stroke	85 mm
Displacement	565 cm ³
Compression ratio (geometric)	11
Speed	1500 rpm
Intake pressure / Temperature	1 bar / 36°C
Intake valve timing	open: 346°CA / close: -140°CA
Exhaust valve timing	open: 130°CA / close: -356°CA
Injection timing	start: -137°CA / end: -119.5°CA

VISUALIZATION OF FLAME PROPAGATION

Flame propagation was visualized by schlieren imaging with a high-speed camera, illustrated schematically in [Figure 2d](#). Since schlieren techniques have been used for many years to visualize density gradients in transparent media [38], only a brief description of the description of the optical system will be given. The arrangement was that of “focused shadowgraphy” in “Z” layout [38] with the engine's pent roof as the imaged object. Light from a green, pulsed, high-power light-emitting diode (LED) was focused through an aperture with 0.7 mm diameter and collimated by a parabolic mirror with a focal length of 900 mm. [Figure 2c](#) shows the region of the combustion chamber that was accessible via flat, fused-silica windows on each end of the pent roof. After passing through the engine's pent roof, the collimated beam was focused by another, identical parabolic mirror through a round aperture with 1.7 mm diameter, which acted as a schlieren stop. A camera lens with focal length $f = 100$ mm projected the resulting schlieren images onto the detector of a high-speed complementary metal-oxide-semiconductor (CMOS) camera (Phantom v7.1). Single-cycle sequences of images were acquired with a temporal frame spacing of 0.5°CA ($= 55 \mu\text{s}$) and an effective exposure time of $5.5 \mu\text{s}$, given by the LED's pulse width. The visual contrast between burnt and unburnt regions in the schlieren images was enhanced by temporal differentiation, *i.e.*, by subtracting the previous frame from each frame [39]. For each condition, simultaneous schlieren sequences and pressure traces of 140 cycles were recorded.

In the current engine configuration the hydrogen injector replaces the spark plug when mounted centrally in the combustion chamber. Therefore, in fired operation, ignition was achieved by creating a laser-induced plasma at the approximate location of the spark gap (as if the spark plug were mounted), as shown in [Figure 2c](#). Omitting the spark plug removes the associated aerodynamic obstruction of the in-cylinder flow such that the movement of the nascent flame kernel can be observed better, but slight differences from the early flame development with a conventional spark plug may result. Laser ignition has previously been used in engines running on gaseous fuels, for example [40, 41], and in liquid-fueled engines [42]. Outside of engines, many studies of laser ignition of flammable gas mixtures have been performed. Most pertinently, in lean hydrogen-air mixtures, the laser-induced plasma and flame kernel formation has been imaged in a constant-volume vessel [43]. A detailed investigation of laser ignition in hydrogen-fueled engines is outside of the scope of the current effort. Instead, given the optical access of the engine, laser ignition simply provided a convenient means of firing the engine to observe the effects of charge motion on flame propagation. In the current experiments, a localized plasma was created by focusing two consecutive pulses from

a dual-head Nd:YAG laser at 1064 nm with a spherical lens with $f = 150$ mm. The pulse separation was $500 \mu\text{s}$ ($= 9^\circ\text{CA}$); each pulse had an energy of 40 mJ. The idea of using two pulses was to make the effective duration of the laser-induced plasma about as long as the duration of the plasma created by a conventional spark plug; however, it was observed that the second pulse had little effect. Hence, the timing of the first laser pulse is taken to be ignition timing. Flame-front visualization was performed for maximum brake-torque (MBT) spark timing.

EXPERIMENTAL CONDITIONS

Two different modes of engine operation are considered in this study: Fired with a fully homogeneous fuel/air mixture, and fired with DI. In both cases, the engine was fired every fifth cycle while the speed was kept constant at 1500 rpm. For DI operation, the pressure in the surge tank upstream of the intake runner was kept at 1 bar and typical intake temperatures, just upstream of the engine head, were around 36°C . In premixed operation, the intake pressure was raised to 1.1 bar to achieve the same time-averaged fuel and air flows as with DI.

With DI, to obtain a global equivalence ratio of 0.25 ($\lambda = 4.0$) or a hydrogen mole-fraction of 0.095, the hydrogen fuel was injected for 17.5°CA at 100 bar. In fired operation at typical efficiencies, this fuel concentration represents a low-load condition with about 2.5 - 2.7 bar indicated mean effective pressure (IMEP). As in previous studies [31,32,33], the fuel injection started shortly after intake-valve closure to yield the maximum time span to interact with the intake-induced flow. With the injection command issued at -140°CA , flow from the nozzle started at -137°CA and ended at -119.5°CA . For premixed operation, fuel was supplied sufficiently far upstream that complete mixing of fuel and air could take place.

As in the previous study summarized by [Figure 1](#), two different intake configurations were used: the unmodified intake and a modified version yielding an approximately three to four times higher tumble ratio. There are several ways to increase the tumble ratio in an engine [14, 28, 44]. Here, we inserted tongue-like plates into the intake ports, gradually reducing the cross-section of each port to its upper half, thereby directing the intake flow more along the pent roof than would be the case without modifications. The engine head with tumble plates is shown in [Figure 2c](#). In the following, when comparing operation with the unmodified head to that with tumble plates inserted, the former will be addressed as “low tumble” and the latter as “high tumble” operation. For each intake-flow condition, flame propagation was visualized for premixed air/fuel supply and with DI, with the former supply option presumably yielding in-cylinder flow identical to that without fueling, investigated as a baseline in previous work.

RESULTS AND DISCUSSION

In this section, a morphological comparison of single-cycle sequences of schlieren images for the different configurations will be described. Then, the growth rate of the flame area and its rate variability as a function of time after ignition are discussed. The section concludes with an extensive study of the flame propagation speed and flame location, which are correlated with engine performance on a single-cycle basis.

VISUALIZATION OF THE EARLY FLAME DEVELOPMENT AND ENSEMBLE AVERAGED HEAT RELEASE

Figure 3 shows typical³ single-cycle sequences of schlieren images, labeled by degrees crank angle after ignition timing (°CA aIT), with the engine operating at maximum brake-torque spark timing (see Table 2). At ignition, 0°CA aIT, for all configurations a well defined round plasma of similar size can be observed. This plasma then develops into a flame kernel, which by 2.5°CA aIT already shows strong influence of the in-cylinder flow. Wrinkled borders, typical of turbulent premixed combustion, become visible in all flames. As time progresses, the effects of the flow becomes more obvious as it drives flame evolution. For the homogeneous mixture with low tumble, the flame evolves radially outwards from the plasma and remains quite symmetric with respect to the pent roof, resembling a growth similar to what is observed in pressurized vessel experiments [35, 36]. Previous velocity measurements [33], summarized in Figure 1, showed that near TDC the flow field does not have a single vortical structure; instead, a weak upwelling flow develops during compression, presumably supported by piston motion. This is consistent with the lack of convection and distortion in the flame shape for this case.

With high tumble, just after ignition the large-scale flow structures convect the flame kernel toward the exhaust valves. Initially, the kernel is small and it is only wrinkled by the smallest flow-scales. However, as the flame grows, the kernel becomes progressively wrinkled by larger flow scales until its size is sufficient to experience the whole spectrum of turbulent length scales. In Figure 3, this takes place around 5°CA aIT; beyond that, the flow also starts to stretch the flame in two directions: along the exhaust side of the pent roof and towards the center of the combustion chamber. The flow measurements in Figure 1 [33] suggest that the intake-induced tumble vortex survives until the end of compression. Based on these two observations (i.e., flow and flame), vortex break-down [14, 28] has not taken place or is at least not complete. Figure 3a shows qualitatively that for high tumble flame propagation is faster. This faster propagation appears to

be enhanced by convection and stretching effects, not only by small-scale turbulence from tumble break-down. However, Figure 3a represents only one single cycle, and indeed there are sometimes cycles in which the flame evolves as it does in the low tumble case, indicating that a strong vortical structure is not always present.

Before we turn our attention to the DI cases, we compare the schlieren-visualized combustion process to the ensemble averaged apparent heat release, derived from the in-cylinder pressure data using a single-zone heat release analysis [9], shown in Figure 4a for homogeneous fueling. Table 2 also lists performance measures derived from pressure traces. Consistent with the morphology of the typical cycles shown in Figure 3a, high tumble yields much faster heat release during the entire combustion, with an overall reduction of the 0-90% burn duration from 95°CA to 77°CA. Engine thermodynamics have a notorious lack of repeatability in optical engines; nevertheless, the measured improvement in indicated thermal efficiency (ITE) and coefficient of variation (COV) in IMEP is qualitatively consistent with the shorter burn duration. Note how small the COV of IMEP is despite ultra-lean, homogeneous operation, which is due to the unique physico-chemical properties of hydrogen as a fuel [8].

With DI, in Figure 3b the effect of flow field on flame kernel development is also clearly visible. Similarly to the homogeneous high tumble case, initial solid-body flame convection is followed by a strong stretch of the flame. Since the injection event causes inversion of the tumble direction [33], the flames are convected and stretched clockwise. Qualitatively, both flames have the same features and propagate at similar speed. However, Figure 4b shows a faster heat-release rate for high tumble during the initial part of combustion (until about 20°CA aIT). Although visualization of the flame front during the intermediate and late combustion was not possible, the heat release analysis in Figure 4b indicates that contrary to the first part of the combustion, after 20°CA aIT the low-tumble case releases heat faster than its high-tumble counterpart. As a result, the total combustion duration is shorter for this case. It is well known [25, 45, 46] that early flame kernel development is affected by the equivalence ratio around the kernel. Therefore, a plausible explanation for the trends observed above could be that while the presence of a richer fuel cloud dominates early combustion, a stronger more turbulent in-cylinder flow overcomes the fuel effect during middle and late stages. In general, the differences between the two tumble configurations are small for DI, and the particular injection geometry appears to create a “super tumble” case in both flow and flame propagation. Consistent with this broad generalization, with DI the burn duration is shorter, while both ITE and COV of IMEP are improved with respect to homogeneous operation with high intake-induced tumble.

³ The issue of what is “typical” is central to cycle-resolved measurements of engine combustion. Statistical analysis, some of which is performed in subsequent sections, is necessary to define “typical” behavior. Here we simply mean that the image sequences were chosen such that their morphological features, as described in the text, can be recognized in the majority of sequences from the same conditions. Exceptions are explicitly mentioned.