

Using PIV Measurements to Determine the Role of the In-Cylinder Flow Field for Stratified DISI Engine Combustion

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ABSTRACT

In a companion study [1], experimental observations in a stratified-charge DISI engine operated with late injection of E70 led to the formation of two hypotheses: (1) For highly stratified spray-guided combustion, the heat-release rate of the main combustion phase is primarily controlled by mixing rates and turbulence level associated with fuel-jet penetration. (2) During the main combustion phase, the role of the in-cylinder flow field generated by the intake and compression strokes is primarily its stochastic disturbance of the mixing and flow associated with the fuel jets, thereby causing cycle-to-cycle variations of the spray-guided stratified combustion. Here, these hypotheses are tested. An optical engine was operated skip fired at 1000 and 2000 rpm, and exhibited the same combustion properties observed in the steady-state all-metal engine tests. High-speed particle image velocimetry (PIV) and spray imaging are used to quantify the intake-generated in-cylinder flow momentum, the spray induced momentum, and the resulting liquid spray variability.

The PIV measurements reveal that the spatially-averaged gas-flow speed (momentum) without injection at 2000 rpm is twice that of 1000 rpm. In contrast, just after injection the gas flow spatial average speed at 2000 rpm is only 24% higher due to the dominance of spray momentum. This is comparable to the 16% increase of the measured ensemble-averaged heat-release rate (in kW). The cyclic variability of the in-cylinder flow speed prior to injection is measured to be considerably higher at 2000 rpm compared to 1000 rpm. Though the injected liquid spray reduced the flow-speed cyclic-variability after injection, the higher variability did persist. The spray imaging reveals that the increased flow-speed variability at 2000 rpm causes increased variability of the spray jet trajectory, jet coalescence, and spray rotation from cycle to cycle. This work supports both the hypotheses that motivated this study.

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1. INTRODUCTION

In the light of constrained petroleum supply and anthropogenic climate change [2], increased engine efficiency and alternative fuels are of interest. Motivated by potential for high thermal efficiency [3], this study focuses on the combustion in a stratified-charge spray-guided (SG) direct-injection spark-ignited (DISI) engine. For lean and stratified operation, low engine-out NO_x and soot emissions must be achieved to avoid expensive aftertreatment technology in markets with stringent exhaust legislation. Recent studies demonstrated the potential of E85 fuel to enable clean and efficient stratified combustion [4, 5] in an engine with a spray-guided combustion system by the use of late injection and ignition of the spray leading edge (head ignition). That study highlighted the challenging trade-off between a reduction of NO_x emissions and the appearance of partial-burn cycles. Furthermore, spark-plasma imaging in [4]

revealed a correlation between flow direction and partial burns, showing that the in-cylinder flow field generated by the intake and compression strokes matters for the stratified combustion.

The in-cylinder flow field generated by the intake and compression stroke interacts with the flow patterns being induced by the fuel-injection process. Current study combined with the companion study [1] aims at understanding the role of the in-cylinder flow field for spray-guided DISI combustion with an oxygenated fuel. A clear understanding of the role of the in-cylinder flow field would provide a cornerstone of an experimentally based conceptual model of spray-guided DISI combustion and has potential to reveal the cause of sporadic partial-burn cycles. As mentioned in Ref. [1], a modest literature survey did not reveal consensus on the role of the in-cylinder flow field for spray-guided DISI combustion. The

work by Honda *et al.* [7] implies that the in-cylinder air flow may not play an essential (required) role for the combustion for that particular spray-guided combustion system and that its effect on the fuel/air mixing should be minimized. The work by Schwarz *et al.* [8] emphasizes that for enabling stratified operation at high engine speeds, it is important to ensure a stable fuel spray despite influence of in-cylinder charge motion. Discussions of the mode of combustion are also rare. The work by Drake *et al.* [6] is the one that shows data of the relative fraction of the combustion taking place in partially premixed regions, and in mixing-controlled zones, respectively. However, that work does not discuss what controls the heat-release rate in these two regions.

However, several important building blocks towards a conceptual model have been laid out in a series of publications by Dahms *et al.* [9,10,11,12]. These publications describe the development of a CFD-modeling framework, with several modeling aspects being closely tied to experimental work in optical engines. Particular focus is given to the spark-plasma development, ignition of the fuel, and the development of the early flame kernel. This body of work focuses on gasoline, or gasoline-surrogate fuels. In summary, the literature survey indicates that several cornerstones are missing for the development of an experimentally based conceptual model of spray-guided engine combustion for highly stratified operation of an oxygenated fuel. To address this, concurrent work in Ref. [1] used systematic all-metal engine performance testing to start unraveling the role of the in-cylinder flow field for a stratified-charge DISI engine, both regarding average heat-release rate (HRR) and combustion stability. The results in Ref. [1] allowed formulation of two hypotheses:

1. The heat-release rate during the main combustion phase of highly stratified operation is controlled by mixing rates and turbulence level in fuel-containing regions, and that these mixing rates are governed by the fuel-injection process.

It should be noted that this hypothesis *does not* stipulate that the HRR is controlled by mixing rates of fuel and air like diesel combustion. That would likely be the case if the dominant combustion mode during the main combustion phase is turbulent non-premixed flame combustion. On the other hand, if the dominant combustion mode is turbulent partially-premixed flame deflagration, the HRR may be controlled by local fuel/air-equivalence ratio and turbulence, with the turbulence level being directly proportional to the mixing rates associated with the fuel jets. The current studies do not allow a determination of the dominant combustion mode. Hence, mixing rates and turbulence level are not distinguished from each other in the hypothesis.

2. The role of the in-cylinder flow field generated by the intake and compression strokes is its stochastic influence on the fuel/air mixing, causing cycle-to-cycle variations of the spray-guided stratified combustion.

The latter hypothesis is in line with the wording found in the gasoline work by Honda *et al.* [7], as discussed above, but it was formulated completely independently of that study. The metal engine results in Ref. [1] show that the cyclic variability increases with increasing engine speed. This demonstrates the need to manage the stochastic processes during stratified combustion, which are expected to be strongly affected by the in-cylinder flow development and its cycle-to-cycle fluctuations. There has been a strong effort within the stratified SG DISI combustion research community to identify potential causes of cyclic variation. Spark emission spectroscopy techniques [13] have revealed that overly lean mixtures near the spark plug at the time of ignition are strongly correlated with the occurrence of partial burn and misfire cycles. Electrical discharge phenomena such as spark plasma extinguishment, multiple spark restrikes, surface sparks, and lower spark energy imparted on the mixture have been identified as causes of partial burn and misfire cycles by Fansler *et al.* [14]. They also utilized light-scattering imaging to reveal that local flow velocity fluctuations can cause unfavorable spark motion [15], which leads to misfire. Peterson *et al.* [16] conducted a more comprehensive study of effects of cyclic variability using high-speed imaging of fuel, flow and flames to investigate the causes of rare misfires and partial burns in a stratified SG DISI engine fueled with gasoline, which requires ignition of the trailing edge of the spray. This work [16] affirmed the work of Fansler that the flow field close to the spark plug is one of the important causes of cyclic variation of the stratified SG DISI combustion. In particular, flow fields that directed the flame kernel away from the fuel cloud prevented sufficient flame propagation into the flammable mixture and showed a tendency to generate misfires or partial burns. This previous investigation [16] in a DISI engine was carried out at relatively low engine speeds (less than 1000 rpm). Hence, further investigations are needed to clarify the role of in-cylinder flows at higher engine speeds and how these flows influence both the heat-release rate and cyclic variability of SG stratified-charge engines. Spray characterizations in a stratified SG DISI engine carried out by Fansler and Drake [14] showed that cyclic fluctuations in spray macroscopic structure are small. Therefore, causality between the spray and the observed combustion variability could not be affirmed for their particular operating conditions. Further investigations need to clarify the spray characteristics and variability in the engine at different operating conditions since the spray atomization and evaporation processes as well as the fuel-air mixing process are highly dependent on the fuel pressure, fuel temperature, ambient density, ambient temperature, fuel properties, and nozzle geometry, as demonstrated in Ref. [17,18,19,20].

To further the understanding of stratified SI combustion, it may be worthwhile to also examine research efforts related to Diesel combustion, which is inherently stratified. In terms of the in-cylinder flow field, studies based on laser diagnostics and modeling have demonstrated that the in-cylinder flow field plays a pivotal role in the mixture preparation, combustion, and pollutant formation processes in direct-injection Diesel engines. For example, recent measurements made by Miles *et al.* [21]

closely coupled with numerical simulations, described well how the flow field affects the combustion under various conditions in diesel engines as injection parameters, swirl and engine speed were varied. They further identified the dominant sources of flow turbulence and the mean flow structures conducive to turbulence generation. However, although cyclic variation is the nature of diffusion combustion, the occurrence of misfire and partial burn is not discussed in detail, perhaps because it is not considered an important challenge for diesel combustion.

The current optical study seeks to determine the role of the in-cylinder flow field on the combustion in a stratified SG DISI engine, regarding both average heat-release rate and cyclic variability. E70 is the fuel chosen by the current authors for both the all-metal study [1] and this parallel optical study. Here, the role of the in-cylinder flow is assessed as the engine speed is doubled from 1000 to 2000 rpm. Of course the flow change due to the change in rpm is confounded by the thermodynamic rate of change during the fixed fuel-injection rate. Thus, particle image velocimetry (PIV) is used to measure properties of the flow, independently. This flow field is captured in a tumble plane in the central part of the piston bowl, which captures the tumble flow generated during the intake stroke and by the late fuel injection. Hence, the swirl flow is not measured in this paper. This optical diagnostic study is conducted at operating conditions determined from continuously-fired all-metal engine testing [1]. The PIV results are complemented by pressure-based heat release analysis, high-speed flame imaging and high-speed light-scattering images of the spray. The diagnostic methods as well as the noise and error of results are discussed in detail. The results include comparisons of the average and cycle-to-cycle velocity without injection and with injection (no ignition). This study will not be able to provide a complete and final assessment of the role of the in-cylinder flow field, but should serve well as a starting point and offer an opportunity to strengthen or reject the two hypotheses presented above.

2. ENGINE AND OPERATIONS

2.1. Single-Cylinder Engine

The engine used for the experiments is a single-cylinder four-valve research engine that can be set up in two configurations; an all-metal configuration for continuously fired performance testing, or in a nearly identical optical piston and head configuration for imaging studies. All tests here were conducted with a water-cooled metal cylinder and oil-lubricated metal piston rings. The piston has a moderately deep piston bowl to aid the stratification of fuel. Fig. 1 shows a cross section of the combustion chamber at top-dead-center (TDC) and Fig. 2 shows a picture of the piston top. The piston bowl has a slight cut-out on one side to facilitate a better view into the bowl when the pent-roof side-window on the right-hand side in Fig. 1 is used. Some engine data are given in Table 1. The engine was fueled using an 8-hole injector with a nominal included angle of 60°, and the distance from the injector tip to

the spark-plug gap was 13 mm. The injector was oriented such that two of the fuel sprays straddled the spark-plug gap. For all data in this study, one of the intake valves was deactivated to increase in-cylinder swirl and tumble levels, see Table 1.

N₂ dilution was used to lower the intake mole fraction of oxygen [O₂] to simulate the dilution effects of exhaust gas recirculation (EGR). This charge dilution is important for controlling NO_x emissions. The intake gas being used here contains 81% N₂ and 19% O₂ which is equivalent to that with 20% external EGR, but of course does not include the exact chemical composition of real EGR. Air and N₂ were metered with sonic nozzles. For each operating point, the in-cylinder pressure, spark current, intake and exhaust pressure, and fuel pressure were acquired with a 0.1°CA resolution. The apparent heat-release rate (AHRR) was computed from the in-cylinder pressure for each individual cycle using a constant ratio of specific heats (γ) following [22]. For optical diagnostics, the engine was equipped with a concave quartz-window in the piston bowl and a flat quartz-window in the end of the pent-roof section closest to the injector, see Figs. 1c & 1d, respectively.

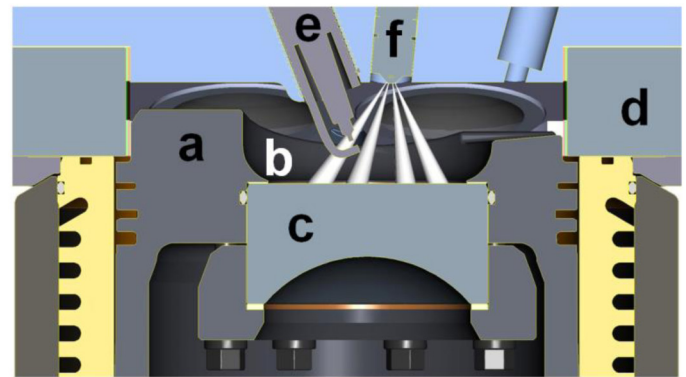


Figure 1. CAD rendering of cross-section of combustion chamber at TDC with a - piston, b - piston bowl, c - piston-bowl window, d - pent-roof window, e - spark plug, and f - injector.



Figure 2. Picture of piston (in all-metal configuration) and cylinder in its dropped-down position.

As discussed above, this study uses E70 fuel. The E70 fuel was prepared in-house by blending the research-grade gasoline with anhydrous high purity grade ethanol in 30%/70% proportions by volume. The gasoline is Chevron-Phillips research-grade gasoline with an antiknock index of 87. It is a typical gasoline and its specifications can be found in [1, 5].

Table 1. Engine specifications

Displacement	0.552 liters
Bore/ Stroke	86.0/95.1 mm
Connecting Rod Length	166.7 mm
Geometric Compression Ratio	12:1
Intake Valve Diameter	35.1 mm
Intake Valve Angle Relative Cylinder Axis	18°
Exhaust Valve Diameter	30.1 mm
Exhaust Valve Angle Relative Cylinder Axis	16°
Swirl / Tumble Index (one intake valve deactivated) ...	2.7 / 0.62
Fuel Injector	Bosch 8-hole solenoid-type
Hole Orientation	Symmetric with 60° included angle
Hole Size	Stepped-hole, min. dia. = 0.125 mm
Injection Pressure	170 bar
Injection Rate @ 170 bar	13.3 mg/ms
Ignition Type	Inductive
Spark Coil	Bosch, ZS-L 1x1E
Spark Plug	Long-reach single-point J-gap
Spark Gap Width	Bosch, ZS-L 1x1E

2.2. Engine Operation with “Head” Ignition and Near-TDC Injection

All-metal engine experiments were conducted in Ref. [1] to establish the test conditions for the skip-fired optical-engine experiments performed here; those tests of 500 consecutive cycles demonstrated that near-TDC injection, with head ignition and E70 can achieve ultra-low emissions of NO and relatively high combustion efficiency and stability. Fig. 3 shows two examples of the head ignition strategy. Fig. 4 presents the injection-retard effect on the trade-off between NO emission and combustion efficiency at 1000 rpm as an example to demonstrate the benefits of near-TDC injection. All four test conditions shown in Fig. 4 were operated stably with head ignition at a constant N_2 dilution level rendering $[O_2]_{in} = 19\%$. It clearly shows that retarding the injection timing decreases engine-out NO emission, while the thermal efficiency increases until a certain start of injection (SOI) before the TDC and then decreases. In this case, SOI = -10°CA ATDC (after top dead center) is considered to offer the best trade-off between NO emission and thermal efficiency. This point has $CA_{50} = 8^\circ\text{CA}$.

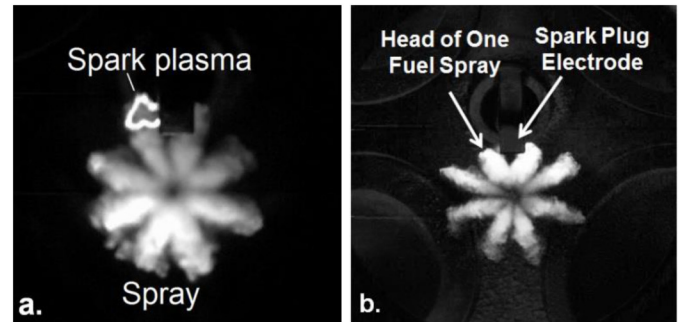


Figure 3. a) An example of “extreme” head ignition from [3]. b) An example of head ignition used in this work (spark plasma exists behind the spark plug electrode); start of injection (SOI) = -23°CA , spark timing (ST) = -22°CA .

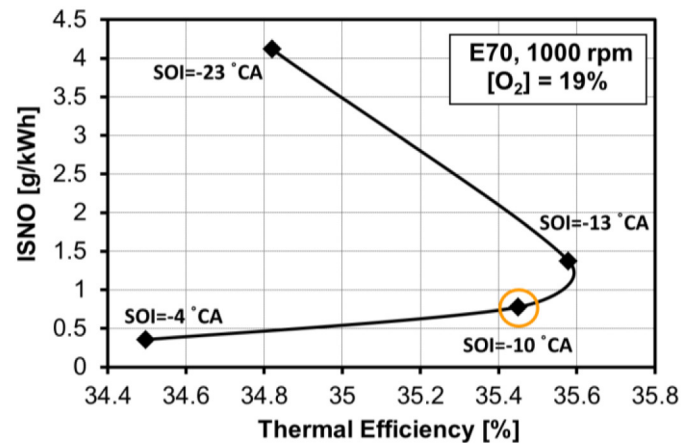


Figure 4. Trade-off between NO and thermal efficiency using near-TDC injection for E70 [1].

The comparison of gasoline and E70 in the metal engine [1] revealed that the strategy to use near-TDC injection with head-ignition for E70 produces low NO emissions also as engine speed is increased. As reported [5], the closely-coupled injection and ignition and more injected fuel, required by the lower heating value of ethanol, leads to a combustion event that visually appears to have a finer structure. It is speculated in [5] that the fine structure is possibly indicative of higher turbulence level during ethanol combustion compared to gasoline with tail ignition. This higher turbulence level could contribute to lower NO formation by enhancing the mixing between hot combustion products and cooler unused air. Figure 5 presents the $IMEP_g$ and the corrected standard deviation (CSD) of $IMEP_g$ as a function of engine speed based on the metal engine experiments [1]. The latter is defined as:

$$CSD \text{ of } IMEP_g [\%] = 100 \cdot \frac{\sigma_{IMEP_{g, \text{fired}}}}{(IMEP_{g, \text{fired}} - IMEP_{g, \text{motored}})} \quad (1)$$

Here, the standard deviation (σ) of $IMEP_g$ for fired operation has been corrected for the heat transfer losses of a motored cycle by being normalized by the difference in average $IMEP_g$ for fired and motored operation. The difference in $IMEP_g$ for fired and motored operation equals the $IMEP_g$ rise caused by

the combustion event. This definition allows the combustion stability to be consistently evaluated even for operation with IMEP_g near zero or negative, (e.g. idle operation or vehicle coasting).

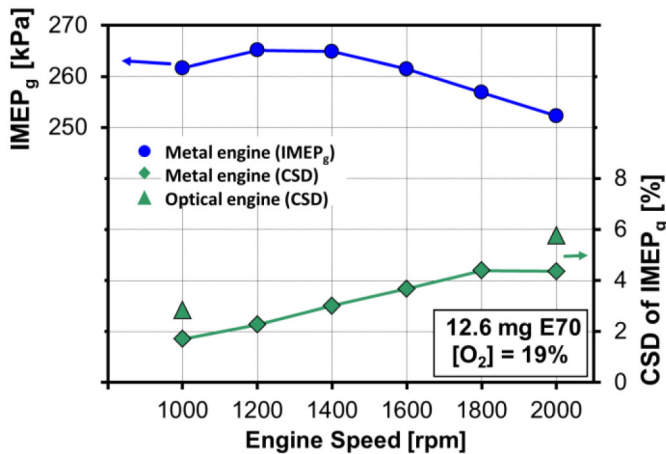


Figure 5. Effect of engine speed on combustion performance and stability.

Table 2. Engine operating conditions

Intake Temperature	26-28°C	
Intake Pressure (abs.)	95 kPa	
Coolant Temperature	60°C	
Fuel Type	E70	
EGR/[O ₂] _{in}	19%	
Injection Duration (command)	0.925 ms	
Fuel Mass	12.6 mg	
Engine Speed	1000 rpm	2000 rpm
Start of Injection	-8°CA	-21°CA
End of Injection	-2°CA	-9°CA
Spark Timing	-7°CA	-19°CA
Average IMEP _g	276 kPa	248 kPa
Std. Deviation of IMEP _g	9.7 kPa	18.5 kPa
CA50	9.2°CA	9.0°CA

For Fig. 5, the fuel mass per cycle, spark energy, intake temperature and pressure, coolant temperature and the N₂ dilution level are kept constant. The spark and injection timings were selected for each speed to achieve head ignition and a 50% burn point (CA50) at 9.0±0.5 °CA ATDC. In this study, SOI is defined to be the crank angle (CA) when the first liquid enters the combustion chamber, as determined by scattered-light imaging via either the piston-bowl window or the pent-roof side window. End of injection (EOI) is the crank angle that the last liquid leaves the injector. Also plotted in Fig. 5 are the data for the two optical-engine-operating conditions shown in Table 2. It can be observed in Fig. 5 that the cyclic variability increases with increasing engine speed. Both the increase of CSD of IMEP_g and the reduction of IMEP_g at the higher speed can be attributed to the appearance of partial burn cycles. It should be noted that the two fired optical data points in Fig. 5 show an increase of CSD of IMEP_g with speed that is quite similar to the all-metal data, providing confidence the optical-engine performance.

2.3. Operating Strategies for Optical Tests

The optical engine experiments were conducted with both non-fired and fired engine operation at each of the two engine speeds that are listed in Table 2. Measurements included PIV and imaging of spark plasma, broad-band flame luminosity, and Mie-scattering of the liquid spray. The four different data acquisition strategies in Table 3 were used in conjunction with the two cycle sequences described in Figs. 6a and 6b. For non-fired operations, a repeated four-cycle sequence is used. Data “a” correspond to the non-fired non-injected operation with spark discharge and PIV acquisition on every second cycle of the four-cycle sequence. Data “a” demonstrate the evolution of flow field generated by the intake stroke and the flow-spark interaction. However, this paper only presents the PIV results for flow evolution, while the data for understanding the effect of flow field on spark motion are not shown. Data “b” on cycle 3 of the four-cycle sequence have injection but no spark and include PIV and laser-based spray imaging. Data “b” reflect the impact of fuel injection on the flow-field development. Data “c” have PIV, spray illumination, and flame acquisition on the second fired cycle of a Fire2-Skip6 sequence. These data are aimed at investigating flow-field effects on combustion. The dilution, injection, ignition are identical to the pre-established all-metal operating points. The Fire2-Skip6 sequence achieves a 25% duty cycle to reduce the thermal load, with data acquisition during the second fired cycle to ensure proper temperature and composition of the residual gases. Data were acquired for 100 cycles for each of “a”, “b”, and “c”, for a total of 300 cycles per test condition. However, due to space constraints, PIV results for cycles with combustion (“c”) are not presented in this paper. Hence, for data “c”, only IMEP, AHRR, and flame imaging are presented.

In addition, data “d” correspond to two injected but non-ignited operating points with spray illumination only. Data “d” are aimed at understanding the flow effect on the cycle-to-cycle variability of the liquid spray. The operating conditions for Data “d” are shown in Table 4. The operating condition with the engine speed of 333 rpm provides a weak in-cylinder flow, while the 2000 rpm condition reflects a relatively strong flow-field impact on the liquid spray development. Here, E85 fuel is used for the two special designed operating points. Ideally, E70 fuel should have been used. On the other hand, the latent heat of vaporization is not radically different for E85. It is expected that E85 requires more heat for vaporization, likely leading to a slightly longer liquid spray penetration. The fuel mass per cycle, injection pressure, coolant temperature, intake temperature and pressure are kept constant for the two operating conditions. The electric signal to trigger the injector was fixed at a same crank angle for both engine speeds. Hence, the liquid SOIs differ slightly. It is observed that the peak pressure is dependent on engine speed due to the differences in heat transfer and blow-by at the two speeds, indicating slightly different in-cylinder gas densities and temperatures. Data were acquired for 200 cycles per test condition using an Inject1-Skip3 sequence with spray illumination on the first cycle.

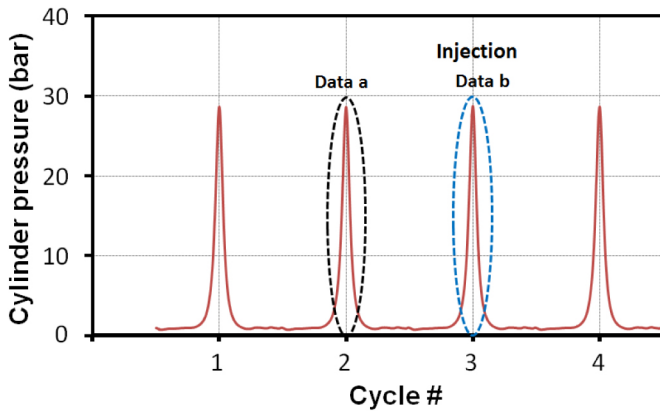


Figure 6a. Data acquisition for non-fired operation; taken at the second ("a") and third ("b") cycles in each sequence.

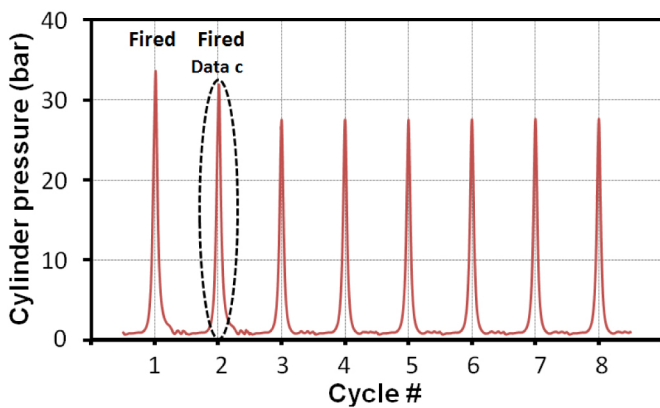


Figure 6b. Fire2-Skip6 sequence for fired operations; data taken at the second cycle (Data "c") in each sequence

Table 3. Data acquisition for four strategies.

Data	Spark	Injection	PIV	Spray	Flame
a (non-fired)	Yes	No	Yes	No	No
b (non-fired)	No	Yes	Yes	Yes	No
c (fired)	Yes	Yes	Yes	Yes	Yes
d (non-fired)	No	Yes	No	Yes	No

Table 4. Non-ignited injection operations at 333 and 2000 rpm

Injection pressure	170 bar	
Intake Temperature	26-28°C	
Intake Pressure (abs.)	95 kPa	
Coolant Temperature	60°C	
Fuel type/ Fuel mass	E85/12.6 mg	
Injection duration	0.925 ms	
Engine Speed	333 rpm	2000 rpm
Start of injection	-22.2°C	-18.6°C
Peak pressure	2285 kPa	2715 kPa

3. OPTICAL MEASUREMENTS

High-speed planar PIV, combustion luminosity, and spray images were acquired to understand in-cylinder mixing and combustion processes within the stratified SG DISI engine. Fig. 7 shows the experimental setup of the single-cylinder optical engine with laser-sheet forming optics and two cameras.

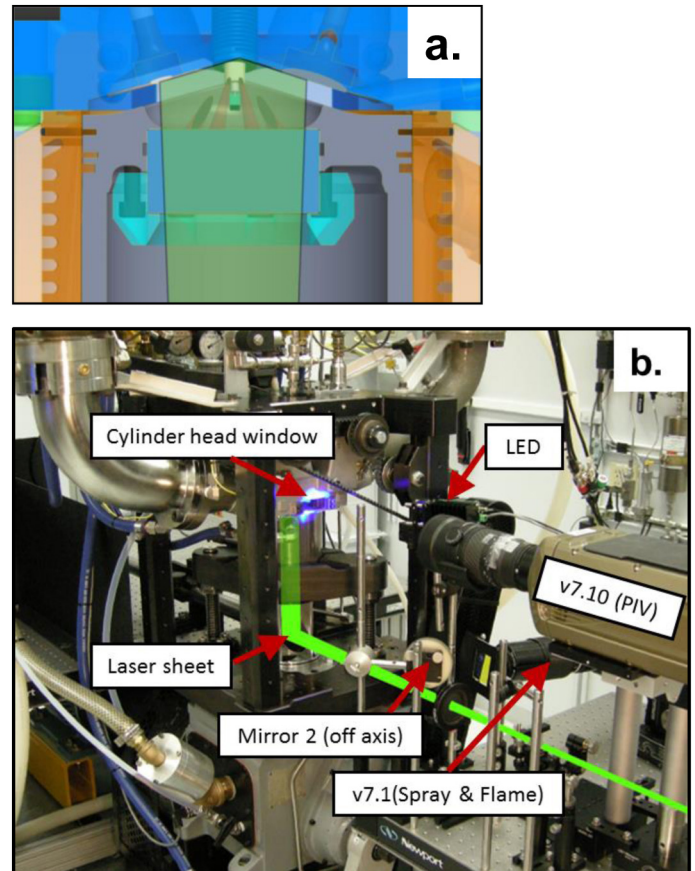


Figure 7. a) CAD rendering of laser sheet entering via piston-bowl window. b) Setup of the single-cylinder optical engine, laser configuration and two camera optical systems.

3.1. Flame and Spray Imaging

Simultaneously with PIV, flame and spray images were acquired through the piston-bowl window using a monochrome high-speed Phantom v7.1 camera. This imaging was performed via two mirrors; the Bowditch-piston mirror was installed just below the piston and a second mirror was located off axis to avoid blocking the laser sheet (refer to Fig. 7b). For orientation, Fig. 8a presents an image of the combustion chamber, but acquired separately using only the Bowditch mirror. Because of the second mirror with its offset position, the view of the flame and spray images is flipped and slightly angled compared to that of Fig. 8a. To allow presenting a consistent view to the reader, all presented flame images are flipped and rotated slightly. Both mirrors were UV enhanced broad-band mirrors for recording broadband luminosity of spark and flame. The spray imaging was recorded simultaneously (prior to the flame images) using a blue high-intensity LED with 2μs pulse duration, with the volume illumination entering via the pent-roof side-window. Notch filters were installed in front of the lens to reject the laser-light scattering at 532±10 nm. The piston bowl window had a concave (R = 25mm) lower surface to enable a wide angle view into the combustion chamber. The spray images for Data "b" & "c" and flame images of Data "c" were taken each crank angle, starting at 10° crank angle before the electric timing of the start of injection. As a separate