

Role of Engine Speed and In-Cylinder Flow Field for Stratified and Well-Mixed DISI Engine Combustion Using E70

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ABSTRACT

This study compares the role of the in-cylinder flow field for spray-guided stratified-charge combustion and for traditional well-mixed stoichiometric operation, both using E70 fuel. The in-cylinder flow field is altered by changing the engine speed between 1000 and 2000 rpm. The stratified operation with the ethanol blend enabled "head ignition" of the fuel sprays, thus minimizing the available fuel/air-mixing time prior to combustion, creating a highly stratified combustion event.

For well-mixed stoichiometric operation, the heat-release rate (HRR) scales proportionally with engine speed due to increased in-cylinder turbulence, as is well-known from literature. In contrast, increasing the engine speed influences the stratified combustion process very differently. Ensemble-averaged over 500 cycles, the time-based HRR in kW remains comparatively unchanged as the engine speed increases. However, cyclic variability of the stratified combustion increases substantially with engine speed.

These observations lead to the development of two hypotheses; a) 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. b) 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.

These hypotheses are supported by cycle-resolved heat-release analysis, and also by PIV measurements in a companion paper [1], which show that the magnitude of the flow variations increases with engine speed. This increased variability may explain the occurrence of partial-burn cycles at high engine speed. In particular, the in-cylinder flow frequently becomes sufficiently strong to disturb both fuel/air-mixing and flame spread, both being critically important for the heat-release rate of the highly stratified charge.

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INTRODUCTION

In the light of constrained petroleum supply and anthropogenic climate change [2], increased engine efficiency and alternative fuels are of interest. Because of its potential for high thermal efficiency [3], this study deals with combustion in a stratified-charge 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. This can be challenging since the application of EGR to reduce NO_x also makes the early flame spread less robust, potentially causing partial burns and misfires [4]. A recent study demonstrated the potential of E85 fuel to enable clean,

efficient, and comparatively stable stratified combustion [5] in an engine with a spray-guided combustion system. In a follow-up study, the reasons for the low engine-out NO and soot were studied in more detail. The low smoke was mostly attributed to the high oxygen content of the E85 fuel, but also to two effects of ethanol's strong vaporization cooling: suppressed formation of ultra-rich zones, and suppressed combustion in moderately rich zones [6]. Because of these beneficial soot-reducing effects of ethanol, it is of interest to study how soot emissions are affected for other fuel blends that are available as alternatives to gasoline. In particular, E70 is the fuel chosen for the current study. This fuel blends is defined by ASTM specifications of Ethanol-Flex Fuel (aka E85) [7], and

US DOE's definition of alternative fuels [8]. The latter requires an ethanol-based alternative fuel to contain at least 70% alcohol. However, this particular study does not focus on soot emissions. Instead, the focus is on the role of engine speed and the in-cylinder flow field for the heat-release rate and stability of the main combustion phase when the E70 fuel is employed.

There are several reasons why a clear understanding of the role of the in-cylinder flow field is important for stratified SI combustion; First, it would provide a cornerstone of a conceptual model of spray-guided SI combustion that can guide both experimentalists and developers of computer models. Second, it has potential to clarify the appearance of sporadic partial-burn cycles that can occur when the engine speed is increased.

In this work, we define "in-cylinder flow field" as the flow generated by the intake and compression strokes. This in-cylinder flow field interacts with the flow patterns induced by the fuel-injection process. There is no consensus in the literature on the role of the in-cylinder flow field for spray-guided SI combustion.

In work by Honda *et al.* [9], it is concluded that it is important to have a high penetration strength of the fuel jet so that the mixture formation is less affected by the air flow in the cylinder. This conclusion implies that for that particular spray-guided combustion system, the in-cylinder air flow may not play an essential (required) role for the combustion, and its effect on the fuel/air mixing should therefore be minimized to avoid combustion problems. Along the same lines, the work by Schwarz *et al.* [10] 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.

Another required cornerstone for a conceptual model of stratified combustion is a clear understanding of what factors govern the heat-release rate (HRR). Directly related to this is a need to also understand the relative importance of various combustion modes that are present during the main combustion phase. In particular, the main combustion phase of stratified-charge SI operation could be a combination of:

A. Turbulent flame propagation (deflagration) through partially premixed regions that are either rich ($\phi > 1$), stoichiometric ($\phi = 1$), or lean ($\phi < 1$), depending on the amount of fuel/air-mixing prior to flame arrival. Flame speed and HRR may be controlled by turbulence level or mixing rates.

B. Turbulent non-premixed flame, with HRR controlled by mixing rates of fuel, air, and hot combustion products.

The only significant combustion mode for the main phase of well-mixed SI operation is:

C. Turbulent flame propagation (deflagration) through fully premixed regions that have a homogeneous fuel/air-mixture. Flame speed and HRR are controlled by turbulence level.

A modest literature survey did not encounter publications that use experimental data to explicitly show what factors govern the heat-release rate for spray-guided stratified SI combustion. Discussions of the mode of combustion are also rare. The work by Drake *et al.* [11] is the only paper found that shows data of the relative fraction of the combustion taking place in premixed regions, and in mixing-controlled zones, respectively. However, that work does not discuss what controls the heat-release rate in these two regions.

This paper does not aspire to a full review of the available literature on spray-guided stratified combustion. Nonetheless, the literature survey also revealed that several important building blocks towards a conceptual model have been laid out in a series of publications by Dahms *et al.* [12, 13, 14, 15]. These publications describe the development of a CFD-modeling framework, with several modeling aspects being closely tied to experimental observations in optical engines. Particular focus is given to the spark-plasma development, ignition of the fuel, the development of the early flame kernel, and turbulent-flame propagation. Their work focuses on gasoline, or gasoline-surrogate fuels. Even so, a high-speed imaging study of E85 shows that the early flame-kernel development remains very important for stable combustion, even for a highly oxygenated fuel [5].

In summary, several cornerstones still are missing for the development of an experimentally based conceptual model of spray-guided engine combustion, in particular for highly stratified operation with an oxygenated fuel. To address this need, the current work seeks to clarify the role of the in-cylinder flow field for a stratified-charge SI engine, both regarding heat-release rate and combustion stability. This study is also one step toward identifying factors that control the heat-release rate.

This study aims to provide a starting point from which hypotheses can be developed and tested. This will be accomplished by systematically examining the effects of engine speed on both stratified and well-mixed SI combustion. Since it is known that the combustion mode of well-mixed SI combustion is turbulent flame propagation through a fully premixed charge, this comparison will naturally lead to a discussion of the mode of combustion for the highly stratified operation. Logically, the study is divided into three main parts:

1. Discussion of the injection and spark timing strategy used for stratified combustion.
2. A comparison of stratified and well-mixed combustion across a range of engine speed. Focus is on:
 - a. IMEP stability.
 - b. Average heat-release rate.
 - c. Cyclic variability of the heat-release rate.
3. Development of hypotheses based on the results in #2 and testing of these hypotheses by detailed heat-release analysis and a brief comparison with a concurrent PIV study in Ref. [1].

EXPERIMENTAL SETUP

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 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. Figure 1 shows a cross section of the combustion chamber 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 via the pent-roof side-window on the right-hand side in Fig. 1. Engine data are given in Table 1. The engine was fueled using an 8-hole injector and the distance from the injector tip to 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, one of the intake valves was deactivated to increase in-cylinder swirl and tumble levels, see Table 1. Moreover, the phasings of the cam shafts were chosen to provide both low residual levels and high volumetric efficiency. The resulting valve lifts are plotted in Fig. 3.

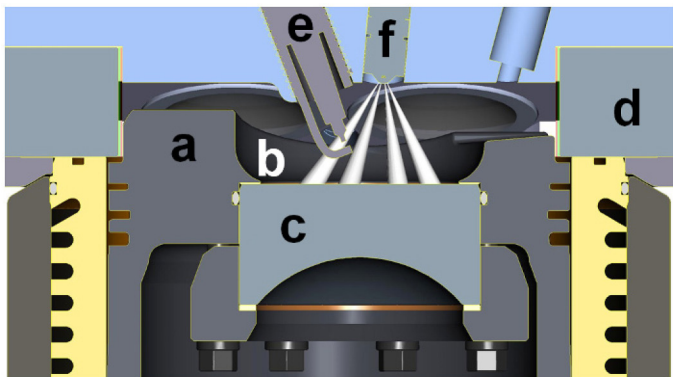


Figure 1. CAD rendering of cross-section of combustion chamber at TDC with a - piston, b - piston bowl, c - piston-bowl window, d - pent-roof window, e - spark plug, and f - fuel injector.

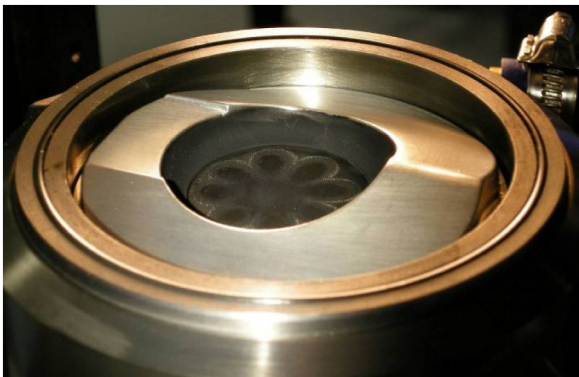


Figure 2. Picture of all-metal piston and cylinder in its dropped-down position.

TABLE 1. Engine Specifications

Displacement.....	0.552 liters
Bore.....	86.0 mm
Stroke.....	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.....	1.0 mm

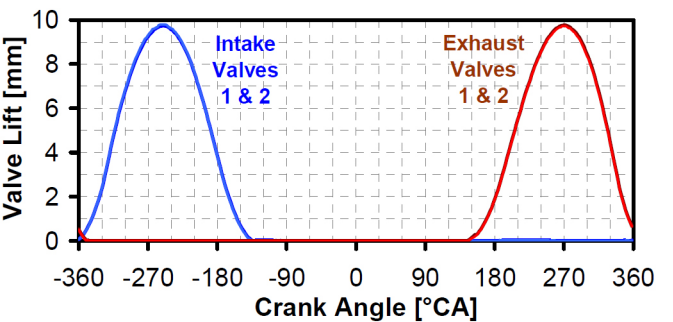


Figure 3. Valve lift profiles measured dynamically using high-speed imaging combined with automatic image processing.

N₂ dilution was used for all stratified-charge operation to lower the intake mole fraction of oxygen [O₂] to 19%. This simulates the dilution effects of EGR, which is important for controlling NO_x emissions. Both air and N₂ were metered with sonic nozzles. Exhaust gas emissions were measured using a Horiba MEXA-584L emission analyzer. Its NO sensor was calibrated at 70 or 1000 ppm, depending on the exhaust concentration to be measured. It should be noted that this instrument does not measure NO₂, so its contribution to engine-out NO_x is not accounted for. Exhaust smoke was measured with an AVL 415S smoke meter.

For each operating point with the all-metal engine configuration, the engine was allowed to run for several minutes until all measured parameters were stable, at which point the data were acquired. Fuel-flow rate and thermocouple readouts were averaged over one minute. The in-cylinder pressure, spark current, intake and exhaust pressure, and fuel pressure were acquired for 500 consecutive cycles using 0.1°CA resolution. For the in-cylinder pressure, an uncooled Kistler 6125C piezoelectric sensor was used in combination with a Kistler 5010B charge amplifier. Day-to-day repeatability

is very good, with typical variations of average IMEP of less than 2%. The apparent heat-release rate (AHRR) is computed from the in-cylinder pressure for each individual cycle using a constant ratio of specific heats (γ) following [16]. For computing combustion-phasing metrics like the 50% burn point (CA50), the AHRR is integrated over the crank-angle range for which AHRR is positive. Unless noted explicitly, this computation of burn points is performed in a conventional way whereby the integral of AHRR is scaled so that rises from 0 to 100% for every cycle, irrespective of the actual combustion efficiency. An exception to this is Fig. 22 where the final burn-point for each cycle varies according to an estimate of the combustion efficiency for each cycle. The latter is computed from the integral of the AHRR and the average combustion efficiency derived from exhaust-emissions measurements. To quantify combustion instability, the standard deviation (SD) of the pressure-derived indicated mean effective pressure (IMEP) is first computed for all 500 individual cycles. As a measure of the *relative* combustion instability, the corrected standard deviation (CSD) of IMEP is computed:

$$\text{CSD of IMEP [\%]} = 100 \cdot \frac{SD_{IMEP_{fired}}}{(IMEP_{fired} - IMEP_{motored})} \quad (1)$$

This is identical to the traditionally used coefficient of variation (COV), except that SD of IMEP for fired operation has been corrected for the heat-transfer losses of a motored cycle by being normalized by the difference in average IMEP for fired and motored operation. The difference in IMEP for fired and motored operation equals the IMEP rise caused by the combustion event. This definition allows the combustion stability to be consistently evaluated even for operation with IMEP near zero or negative, see Ref. [17] (e.g. idle operation or vehicle coasting.) For operation at moderate loads, CSD values are only slightly lower than COV values, as demonstrated in Fig. 12b. Furthermore, the value of CSD of IMEP does not depend on the definition of IMEP (*i.e.* compression-expansion only, or all four strokes), provided that the gas-exchange strokes are repeatable.

Optical Diagnostics Setup

High-speed imaging is presented in this paper to justify the hypotheses being formulated. For this imaging, the engine was equipped with a quartz window in the piston bowl, see Fig. 1c. This window had a concave ($R = 25\text{mm}$) lower surface to enable a wide-angle view into the combustion chamber. High-speed spray and flame imaging through the piston-bowl window was accomplished using a Phantom v7.1 or Phantom v7.3 camera via a 45° Bowditch mirror, see Fig. 4¹.

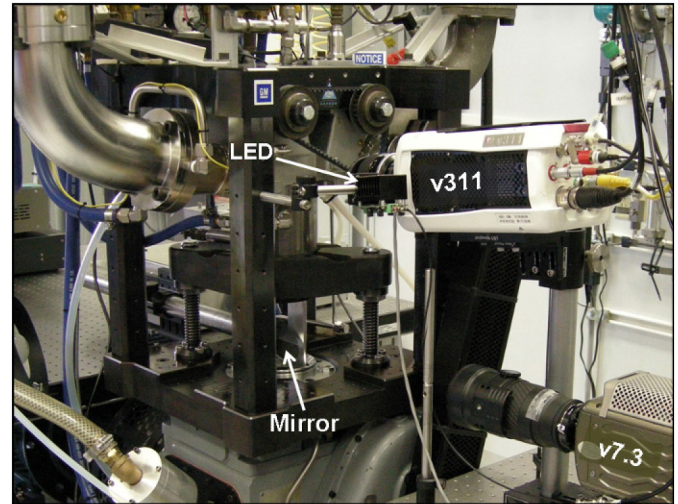


Figure 4. Picture of single-cylinder research engine in its optical configuration.

Both cameras have monochrome sensors that are sensitive in the 350 - 900 nm range, so broadband² flame luminosity was collected. Liquid fuel was visualized using illumination from a blue high-intensity LED with 2μs pulse length, with the light entering via a quartz window installed in the end of the pen-trough section closest to the injector; see Figs. 1d and 4. In this optical configuration, the engine was operated identically to the pre-established all-metal operating points, but with a shorter run duration using a Fire2-Skip6 sequence to achieve a 25% duty cycle and thereby reduce the heat load on the non-cooled quartz windows. All images were acquired on the 2nd fired cycle of each sequence to ensure proper temperature and composition of the residual gases. To limit run times, typically only 100 cycles were acquired for each optical operating point.

Fuel

This study uses primarily E70, which was prepared in-house by blending research-grade gasoline with anhydrous high-purity grade ethanol in 30%/70% proportions by volume. The gasoline fuel is Chevron-Phillips research-grade gasoline with an antiknock index of 87. It is a typical gasoline and its specifications are given in Table 2. With the assumption of a linear variation of properties upon blending, some estimated properties of the E70 fuel are given in Table 3.

E85 was used for the well-mixed combustion imaging shown in Fig. 19a. However, the well-mixed flame spread is expected to be identical for E70, so the combustion-mode comparison in Fig. 19 is still valid [18].

1. Images from the Phantom v311 camera depicted in Fig. 4 are not presented in this study.

2. For the stratified operation in Fig. 19b, a notch filter was used to suppress laser light at 532±10 nm, see Ref. [1]. This imaging also involved a 45° mounted second mirror between the Bowditch mirror and the 7.1 Phantom camera, causing a slight image rotation in Fig. 19b.

TABLE 2. Chevron-Phillips Research-Grade Gasoline

Antiknock Index (R+M)/2	86.8
RON.....	91.0
MON	82.7
Specific gravity.....	0.7456
Carbon, wt%	86.37
Hydrogen, wt%	13.55
Oxygen, wt%	0.08
A/F Stoichiometric.....	14.60
Lower Heating Value, gas-phase fuel (MJ/kg).....	43.10
LHV for stoichiometric charge (MJ/kg).....	2.763
Hydrocarbon Type, vol%	
Aromatics	24.5
Olefins	5.1
Saturates	70.1
Distillation, °C	
5%	56.9
10%	63.4
30%	78.9
50%	96.1
70%	114.1
90%	147.3
95%	164.9

TABLE 3. Estimated Properties of E70 Fuel

Specific gravity.....	0.776
A/F Stoichiometric.....	10.62
Lower Heating Value, gas-phase fuel (MJ/kg).....	31.58
LHV for stoichiometric charge (MJ/kg).....	2.718
Hydrocarbon Type, vol%	
Aromatics	7.4
Olefins	1.5
Saturates	21.0
Oxygenates	70

RESULTS

For all data in this study, the engine was operated in a manner that provides relatively low in-cylinder gas temperatures. First, the temperature of the fresh gases was close to room temperature, see Table 4. Second, the valve timings provide low residual levels³ (≈ 4 -6% by mass). Third, a somewhat late intake valve closing makes the effective CR ≈ 11 . Fourth, a coolant temperature of 60°C was selected for the all-metal tests. This lower-than-normal coolant temperature allows the use of elevated coolant temperature for the skip-fired optical tests to achieve in-cylinder surface temperatures that are similar to those of the all-metal tests.

3. Residual gas mass and temperature were estimated from the pressure-derived mass-averaged charge temperature just prior to exhaust-valve opening, taking into account blow-down cooling and pressures at valve-overlap and during the intake stroke, following Ref. [20].

Swirling in-cylinder flow was used throughout the study since it generally was found to reduce the cyclic variability of IMEP for stratified operation. This stabilization is presumed to be the result of a more consistent flow structure from cycle to cycle, resulting from a more directed intake flow [19]. The swirling flow was accomplished by deactivating one of the intake valves, as noted previously in conjunction with Table 1.

In this study, the crank angles are referenced as after top dead center of the combustion stroke, aTDC. The start of injection (SOI) is the crank angle when the first liquid enters the combustion chamber, as determined by Mie-imaging via either the piston-bowl window or the pent-roof side window. Likewise, end of injection (EOI) is when the last liquid leaves the injector, as determined by Mie-imaging via the side window.

TABLE 4. Operating Conditions

Fuel	E70	
Engine Speed	1000 to 2000 rpm	
Injection Pressure	170 bar	
Spark Energy	106 mJ	
Gas Temperature in Intake Runner	25-28°C	
Coolant Temperature	60°C	
Operational Mode	Highly Stratified	Well-mixed
IMEP _g	252 to 265 kPa	601 to 681 kPa
Intake O ₂ Mole Fraction	19%	21%
Fuel/Air Equivalence Ratio [ϕ]	0.27	1.0
Intake Pressure	95 kPa	59 kPa
Fuel per Cycle	12.6 mg	29.0 to 30.0 mg
Start of Injection	-4 to -23°C A	-303 to -306°C A
Spark Timing	-3 to -22°C A	-13 to -17°C A

Table 4 shows the operating conditions covered in this study. Two modes of operation are compared side-by-side: lean stratified and stoichiometric well-mixed operation. Most of the attention is paid to the stratified operation. The selected operating conditions warrant a brief discussion. For the stratified operation, an intake pressure of 95 kPa is combined with an intake [O₂] of 19%. This is a realistic low load, operated nearly unthrottled with EGR to control NO_x. In this case, 9% of the air mass was replaced with N₂ to simulate EGR. For the