

Characteristics of Ethanol, Butanol, Iso-Octane and Gasoline Sprays and Combustion from a Multi-Hole Injector in a DISI Engine

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ABSTRACT

Recent pressures on vehicle manufacturers to reduce their average fleet levels of CO₂ emissions have resulted in an increased drive to improve fuel economy and enable use of fuels developed from renewable sources that can achieve a net reduction in the CO₂ output of each vehicle. The most popular choice for spark-ignition engines has been the blending of ethanol with gasoline, where the ethanol is derived either from agricultural or cellulosic sources such as sugar cane, corn or decomposed plant matter. However, other fuels, such as butanol, have also arisen as potential candidates due to their similarities to gasoline, e.g. higher energy density than ethanol. To extract the maximum benefits from these new fuels through optimized engine design and calibration, an understanding of the behaviour of these fuels in modern engines is necessary. In particular, the use of direct injection spark-ignition technology requires spray formation and combustion characteristics to be quantified in order to improve both injector design and operating strategies. To this end an optical investigation of spray development and combustion was undertaken in a single-cylinder direct-injection spark-ignition engine with a centrally mounted multi-hole injector. Specifically, crank-angle resolved imaging studies were performed and batches of images from 100 consecutive cycles were acquired with synchronised in-cylinder pressure logging. The engine was motored and fired at 1500 RPM stoichiometrically under part load (0.5 bar intake pressure), with injection timing set early in the intake stroke to promote homogeneous mixture formation. The effects were investigated at engine coolant temperatures of 20 °C and 90 °C using gasoline, *iso*-octane, ethanol and butanol. Projected spray areas as seen through the piston crown were calculated to reveal information about the atomization and evaporation processes for each fuel. Additionally, flame areas and centroids were calculated to analyse the combustion process relative to measured in-cylinder pressure histories.

INTRODUCTION

The application of alcohol fuels in spark ignition engines is not a modern development. As far back as the early 1900s Henry Ford intended his first model T and most other new road vehicles to run on ethanol from renewable sources. The discovery of numerous oil fields and the higher compatibility of gasoline with engine materials, among other factors, meant that ethanol take up was slow and was eventually replaced by gasoline. Today's increasing awareness of the human contribution to global warming, as result of excess CO₂ emissions, and the question marks surrounding the sustainability of an oil-based world economy, have lead to a renewed urgency and increased research efforts to find alternative 'carbon neutral' sources of energy for the transportation sector, which in the UK alone accounted for ~28% of all CO₂ emissions in 2005 [1].

There is significant potential to reduce CO₂ emissions using direct-injection spark-ignition engines by adopting new technologies such as turbocharging and variable-valve actuation with downsizing concepts [2, 3]. However, unless operation with alternative renewable fuels is incorporated, it will not be possible to meet future CO₂ emissions targets. Moreover, unless these fuels are compatible with existing distribution infrastructure and engine components, voluntary take up by vehicle manufacturers is likely to be slow and dictated only by the requirement to meet new waves of legislation. For these reasons the current study investigates the spray development and combustion characteristics of ethanol and butanol in direct comparison to gasoline and *iso*-octane fuels. The former alcohol is currently the most common and widespread alternative to fossil fuels but still faced with significant obstacles for use in concentrations higher than 10–15% with gasoline, whilst the latter alcohol is a more compatible alcohol fuel for use with current vehicle and engine technologies, as well as with existing supply and distribution infrastructure.

In terms of engine performance, the benefits of using ethanol are generally viewed as positive due its higher octane rating – increasing its knocking resistance and allowing the use of higher compression ratios which improves thermal efficiency. This is compounded by significantly higher charge cooling capability compared to gasoline. However, the low energy density of ethanol impacts a vehicle range between re-fuelling periods and therefore is a set-back from the perspective of the consumer. In this respect, the much smaller difference between energy densities of gasoline and butanol, make this an interesting alternative, particularly given its compatibility with current vehicle technologies.

Recent work has concentrated on investigating the potential benefits of ethanol or ethanol/gasoline blends using thermodynamic engines and quantifying the effects in terms of engine performance and particulate, unburned HC and NO_x emissions [4–6]. However, there is little existing literature on the fundamental differences between spray formation of alcohol-based fuels versus standard gasoline, particularly for direct-injection configurations. Moreover, very few reports are available on optical studies of mixture preparation and combustion processes using alcohols and all but none featuring new generation multi-hole injectors or other alcohols than ethanol, especially butanol. Analysis of spray and atomization phenomena with reference to fluid properties is also rare but it is vital in providing assessment and interpretation of the fundamental behaviour of these fuels inside engines. Optical studies of sprays and combustion also provide an essential database for developers and modelers due to the very limited data available on spray break-up and flame speeds of alcohol fuels [7–10].

This work seeks to examine the effect of fuel type on in-cylinder spray development and combustion from a DISI multi-hole injector operating under realistic engine conditions of temperatures and load by:

- Characterising the effects of engine temperature on spray development in a motoring engine for four fuel types, namely, chemically pure ethanol, butanol and *iso*-octane, as well as a standard multi-component commercial gasoline.
- Investigating the behaviour of these different fuels in a firing engine by studying in-cylinder flame growth and motion on a crank-angle resolved basis for different engine temperatures.
- Carrying out analysis of spray and flame parameters relative to in-cylinder pressure parameters to link observed spray effects to flame growth behavior.

In order to study such interactions, a single-cylinder optical DISI engine was used for high-speed imaging of the fuel spray and flame growth through the piston crown, along with acquisition of in-cylinder pressure via a piezoelectric transducer mounting in the cylinder head.

EXPERIMENTAL ARRANGEMENT

OPTICAL ENGINE

Engine Geometry: Experiments were carried out with the same 4-stroke single-cylinder DISI engine and set-up as that used in previously reported work [11, 12] where detailed information about engine and ancillary systems, as well as possible optical arrangements can be found. Therefore only a summary of the engine installation will be given in this paper.

The engine has a prototype cylinder head supplied from Jaguar Advanced Powertrain Engineering with a 4-valve arrangement and has similar bore and stroke geometry to that of a production V8 as outlined in Table 1. It should be noted that in this paper 0° Crank Angle (CA) corresponds typically to intake Top Dead Centre (TDC) and crank-angle timings will be mainly presented with respect to that as °CA After intake TDC (ATDC).

The engine allows for a number of optical access configurations, namely using a piston crown window, a full quartz cylinder liner and a fixed triangular pent-roof window, all of which were used in the current study to set-up lighting or imaging planes so as to study in-cylinder spray formation and flame growth.

Table 1. Optical Engine Specifications.

Engine Base Type	Jaguar V8
Cycle	4-Stroke
Cylinders	1
Valves	2 Intake, 2 Exhaust
Bore	89.0 mm
Stroke	90.3 mm
Compression Ratio	11.15:1
Maximum Speed	2500 RPM
Valve Timings	IVO 24°, IVC 274°, EVO 476°, EVC 6°

Injection System: The fuel injection system comprised a pneumatic pump to produce 150 bar pressure in the fuel line. A 2 µm high-pressure filter element was used on the pressure side with a 150 bar check valve in series to guarantee removal of any impurities and reduce pressure pulsations in the fuel line.

The injector used was vertically positioned and centrally mounted in the combustion chamber with a 6-hole nozzle in a close spacing arrangement with the spark plug. The outer nozzle holes are 0.5 mm in diameter at the exit and essentially consist of two groups of 3 nozzle holes. The spray pattern is illustrated in Figure 1, where plumes 1 and 6 pass around the spark plug.

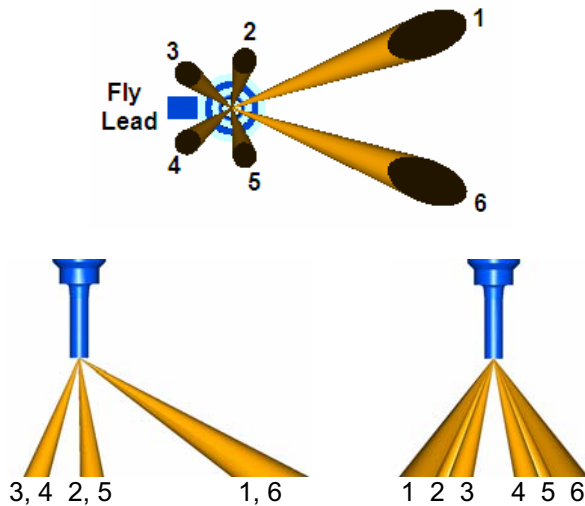


Figure 1. Orientation of the Injector Spray Plumes.

Ignition System: The ignition system used is a single coil-on-plug type, using a shared 12 V, 25 A, DC power supply. A 12 mm diameter NGK-R spark-plug with a platinum-tip central electrode with J-type ground electrode and cone insulator was used. The dwell or charge time was kept constant at 4 ms following manufacturers instructions and previous tests by the authors.

OPERATING CONDITIONS

The baseline operating conditions are given in Table 2. The engine was motored at part load (0.5 bar intake pressure) and fired at the same load continuously, *i.e.* with no skip-firing. The load was defined by setting the throttle position to produce the desired intake pressure. An air-fuel ratio meter was used in the exhaust to measure oxygen content and for setting the correct fuelling for the required engine operating condition. The engine was run with stoichiometric Air/Fuel Ratio (AFR), *i.e.* equivalence ratio $\Phi=1$, for all the tests presented in the current paper. 100 consecutive cycles of in-cylinder pressure data were acquired after the engine had fired up and had been allowed to stabilise for 20–30 s.

Table 2. Engine Operating Conditions.

Engine Speed	1500 RPM
Intake Air Pressure	0.5 bar
Injection Pressure	150 bar
Firing mode	Continuous, $\Phi=1$
Ignition Timing	325° CA ATDC
Engine-Coolant Temperature	20 °C and 90 °C

Synchronization of various control triggers for ignition, injection and camera/laser systems was achieved using an optical encoder mounted on the crankshaft, with 1800 pulses per revolution resolution. This was connected to an AVL Engine Timing Unit (ETU) providing sub-crank angle degree resolution and a LABVIEW program was used for data acquisition at 0.2° CA resolution. In-cylinder pressure data acquisition was synchronised with simultaneous crank-angle resolved imaging of natural flame chemiluminescence.

Pressure data acquisition and processing to infer maximum in-cylinder pressure for each cycle (P_{max}), crank angle of P_{max} , Indicated Mean Effective Pressure (IMEP), Mass Fraction Burned (MFB), crank angle of 10% MFB, 50% MFB, *etc.*, as well as associated Coefficients of Variation (COV), was done using LABVIEW and MATLAB based software.

Fuels: The fuels selected for this study reflect the lack of available published literature on the fundamental behaviour of alcohol fuels inside modern direct injection spark-ignition engines but also the need to compare these with currently used fuels in both commercial and research applications. Table 3 shows a number of selected fuel properties which are relevant to the analysis of spray formation, evaporation characteristics and combustion of these fuels and are given for 1 bar ambient pressure [13–18].

Table 3. Fuel Properties.

Fuel Properties	Ethanol	Butanol	Gasoline	Iso-Octane
Density [kg/m^3] (20 °C)	0.794	0.809	0.729	0.692
Viscosity [cP] (25 °C)	1.08	3.64	0.4–0.8	0.51
Surface Tension [mN/m] (20 °C)	22.4	25.4	25.8	14.7
Latent Heat [MJ/kg] (25 °C)	0.902	0.430	0.364	0.305
Energy Density [MJ/kg], [MJ/l]	29, 23	37, 30	44, 32	45, 31
Boiling Point [°C]	78.5	117.2	30–190	99.8
Reid Vapour Pressure [bar]	0.16	0.02	0.56	0.14
Stoichiometric AFR	9.0	11.2	14.7	15.1
Research Octane Number (RON)	129	96	95	100
H:C, O:C	3, 0.5	2.5, 0.25	1.92, 0	2.25, 0

Injection Strategy: The Injection timing was set for homogeneous mixture preparation mode, *i.e.* early in the intake stroke, with Start of Injection (SOI) set to 80° CA ATDC, in order to maximize the time available for evaporation before ignition. This decision was however balanced by the need to avoid excessive liquid impingement on in-cylinder surfaces, particularly on the piston crown which would affect the imaging arrangement by fouling the windows. It needs to be pointed out here that previous publications by the current authors [11, 12] have demonstrated the viability of using split injections, even under early injection homogeneous operating conditions, and the reduction in wall wetting achieved. For the purposes of this paper, such strategies have not been explored using alcohol fuels, although they are of increased relevance with alcohols due to increased cylinder bore oil film dilution [4] and it is expected that more sophisticated strategies will be explored in upcoming publications by our group.

IMAGE ACQUISITION

High-Speed Camera: A high-speed CMOS camera (Photron APX-RS) was used throughout this study at a frame rate of 9 kHz, corresponding to 1° CA between frames at 1500 RPM. This was possible with an image resolution of 640×480 pixels, giving an optical resolution of ~160 μm per pixel. The camera was coupled to a 60 mm Nikon lens with f2.8-28. At this resolution the on-board camera memory allowed just over 100 cycles of data with 60 frames per cycle to be acquired consecutively before data-download to a PC was necessary.

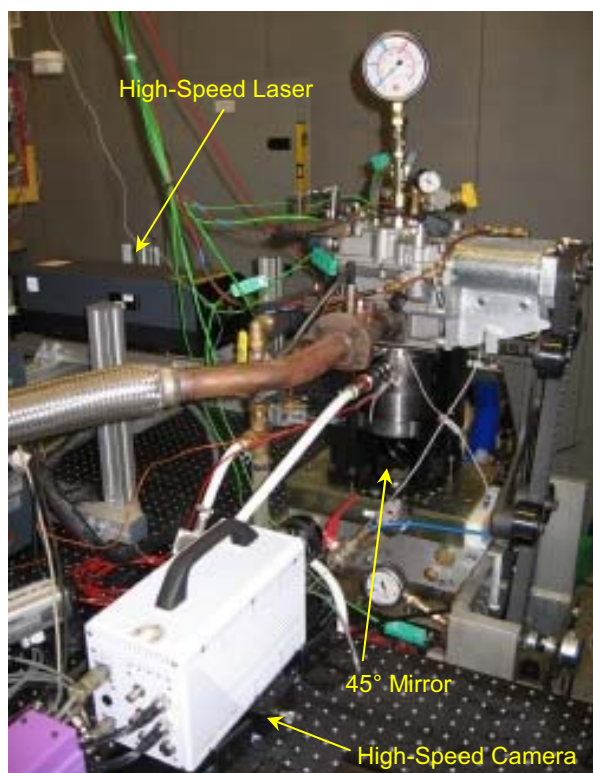


Figure 2. Experimental Set-up for Spray and Flame Imaging through the Piston Crown.

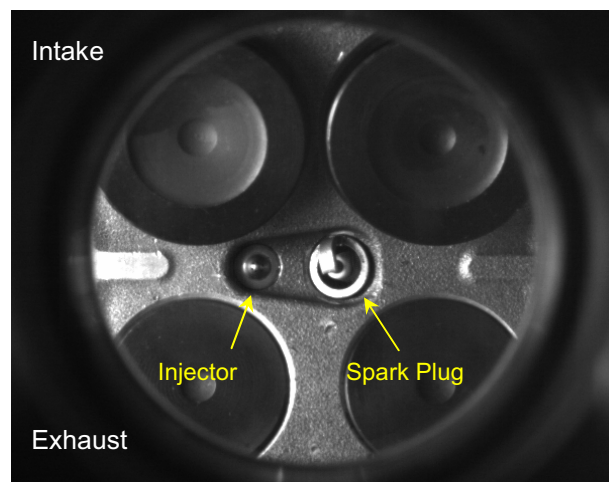


Figure 3. Piston Crown View.

High-Speed Laser: For spray visualization, the combustion chamber was globally illuminated by a New Wave Nd:YLF laser firing at 527 nm wavelength and entering across the engine in the tumble plane, with two coupled divergent spherical lenses, as illustrated in Figure 2. A white semi-opaque diffuser board was also employed to optimize the Mie-scattering intensity captured. For the tests presented and to guarantee uniform light intensity levels during acquisition runs, the laser was run at high power with both heads firing simultaneously at 9 kHz and synchronised with the camera shutter.

The view shown in Figure 3 was selected for this study as it allowed clear and simultaneous observation of all individual six spray plumes, as well as being the same view used to image flame propagation and therefore providing a relevant plane for the visualization of the initial liquid fuel distribution both during the initial and later stages after fuel injection.

SPRAY DEVELOPMENT

GENERAL OBSERVATIONS

Global laser flood-lighting was achieved through the installed pent-roof window and full quartz liner and this allowed illumination of the near-nozzle region, spark-plug region, as well as the length of the engine cylinder and, therefore, the full spray development during and after injection. Spray images were captured through the piston crown window via the 45° mirror installed inside the hollow extended piston (Bowditch design), typical of optical research engine assemblies.

Typical spray imaging sequences for the duration of the injection period are shown in Figures 4 and 5 for all four fuels investigated: ethanol, butanol, *iso*-octane and gasoline, using SOI 80° CA ATDC under motoring conditions at 1500 RPM using 20 °C and 90 °C engine-coolant temperatures; each image is labeled by its crank angle timing following the start of injection, *i.e.* °CA After Start of Injection (ASOI). The images in Figure 4 were

enhanced for printing and reproduction purposes, mainly to make the sprays more easily visible to the reader. There will of course be some shot-to-shot variation of the spray development process on a cycle-by-cycle basis but the contribution of the injector to this variation has been shown to be fairly small with current injector [17] and any variations will be mainly due to flow field related variability, which will not be discussed in detail in this paper; see [11, 12] for more information. It should be noted that for the purposes of this motoring investigation it was decided to use the same fuel pulse-width for all fuels (0.8 ms, corresponding closely to stoichiometric air-fuel ratio for *iso*-octane at the current operating conditions). This allowed both the full injection period and post-injection periods to be analysed directly for all fuels and thereby allow the investigation of the behaviour of the sprays as they interact with – on average – the same flow field. However, for the combustion tests shown later in the paper, the fuel pulse widths were adjusted to achieve correct stoichiometric operation with each respective fuel. From Table 3 it can be seen that for alcohols this results in an increase in the necessary fuel pulse width as a result of the lower AFR of these two fuels.

In general, for the cold operating conditions shown in Figure 4, the four fuels were seen generally to exhibit similar development and break-up behaviour up to the end of injection which was $\sim 11^\circ$ CA ASOI or 91° CA ATDC. The directionality of the spray plumes was close to the injector design requirements as illustrated by the schematic shown on Figure 1, although some spray/flow interactions were seen particularly with respect to spray plumes 1 and 6, which due to their nozzle angles are more susceptible to direct interactions with the flow over one of the intake valves. For the hot operating conditions with the engine head at 90°C , the fuels could generally be grouped into two groups, those which experienced spray collapse (ethanol and gasoline) and those which did not (butanol and *iso*-octane) as described in the following sections.

For the purposes of clarity and in order to facilitate the discussion of the observed characteristics of spray development for all the fuels tested, the text below has been broken down into three main sections, a section on the early injection stage, one on the fully-developed injection stage and a final section on the post-injection mixing stage.

Early Injection Stage: One clear difference between the fuels can be seen in the time taken for fuel to arrive at the injector tip. At 3° CA ASOI in Figure 4 there is no fuel seen at the injector tip for both the alcohol fuels although for *iso*-octane and particularly gasoline there are visible signs that the spray tips are beginning to exit the nozzles with penetrations of ~ 1 mm. The effect of this delay is more clearly seen in the next frame, at 4° CA ASOI. Here the sprays of ethanol and butanol are visibly less developed and exhibit a smaller spray-tip penetration than those of *iso*-octane and gasoline (a result of this

delay which corresponds to $<50\ \mu\text{s}$ from tests conducted with the current injector/driver system). The butanol spray is the slowest to arrive at the injector tip and the effects are replicated to a similar extent at the higher engine coolant temperature of 90°C discussed later. This phenomena is hypothesized to stem from the different fluid dynamic properties of the four fuels (details of which can found in Table 3), in particular, density, viscosity and surface tension. Timing errors were ruled out after extensive and repetitive testing all fuels with different pulse widths and testing the driver system with another similar injector. Confidence in the robustness of the driver system performance is also strong due to the 100% repeatability of the timing of fuel arrival at the injector tip recorded during all the tests and the experience gained from initial hardware performance characterization, details of which can be found in [17].

Figure 5 shows the spray development for all fuels at 90°C engine coolant temperature. The presence of liquid fuel can be seen at the injector tip for all fuels at 3° CA ASOI *i.e.* 1° CA earlier for ethanol and butanol than at 20°C engine coolant temperature. The effect of temperature is clearly affecting internal injector and in-nozzle dynamics to some extent, quantified by a faster time to 'first fuel seen' at the injector tip. At 3° CA ASOI the six individual spray plumes are just visible in the images for ethanol and butanol and are slightly more developed for *iso*-octane and gasoline. Previous work using the same type of gasoline and the same injector in this engine [11, 12] has shown that the gasoline spray collapses in the engine under low pressure and high temperature conditions (namely 0.5 bar ambient pressure and 90°C engine head temperature). In Figure 5 and at 3° CA ASOI the spray plumes are clearly 'swollen' and the emerging spray is already visibly collapsed for gasoline; in contrast, the spray plumes for ethanol are clearly spaced out at 3° CA ASOI, although subsequent spray development also showed a degree of spray collapse. The ethanol spray did not collapse immediately out of the nozzles and this is consistent with previous work by the current authors which showed that in-nozzle boiling and subsequent spray collapse is driven not just by the presence of highly volatile components in a fuel but by the level of superheat experienced by these [18].

The initial spray development process for butanol is not greatly affected at 90°C engine head temperature. The spray does not collapse like gasoline and behaves more like *iso*-octane at 20°C , with wider plumes and faster penetration. Similar trends were observed for the initial behaviour of *iso*-octane sprays when comparing the transition from a cold engine head temperature of 20°C to a fully warm of 90°C , although the effects were accentuated relative to butanol due to the close proximity of *iso*-octane's boiling point to the injector body temperature. No immediate spray collapse was observed for *iso*-octane under the present test conditions, which is again consistent with previous work in the same engine [11, 12].

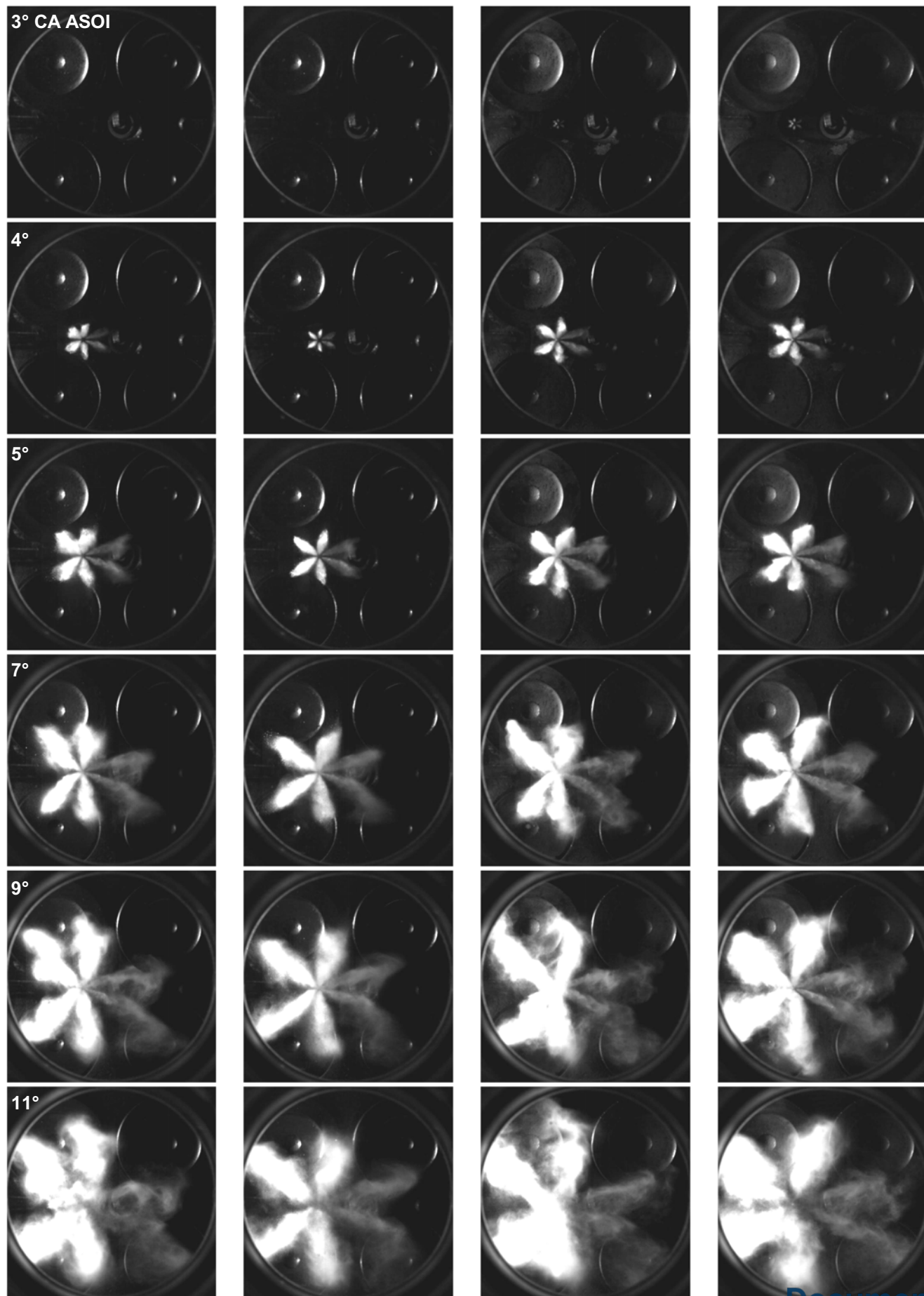


Figure 4. Spray Development: Column1 Ethanol, Column 2 Butanol, Column 3 Iso-Octane, Column 4 Gasoline (Engine Head 20 °C, SOI 80° CA ATDC).