

Spray Penetrations of Ethanol, Gasoline and Iso-Octane in an Optically Accessible Spark-Ignition Direct-Injection Engine

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ABSTRACT

The spray development of ethanol, gasoline and iso-octane has been studied in an optically accessible, spark-ignition direct-injection (SIDI) engine. The focus is on how fuel properties impact temporal and spatial evolution of sprays at realistic ambient conditions. Two optical facilities were used: (1) a constant-flow spray chamber simulating cold-start conditions and (2) a single-cylinder SIDI engine running at normal, warmed-up operating conditions. In these optical facilities, high-speed Mie-scattering imaging is performed to measure penetrations of spray plumes at various injection pressures of 4, 7, 11 and 15 MPa. The results show that the effect of fuel type on the tip penetration length of the sprays depends on the injection conditions and the level of fuel jet atomisation and droplet breakup. It is observed that at 4 MPa injection pressure, the tip penetration length of ethanol sprays is shorter than that of gasoline sprays, likely due to lower injection velocity and increased nozzle loss associated with higher density and increased viscosity of ethanol, respectively. This assertion is further supported by the longest penetration length of iso-octane that has the lowest density among tested fuels and similar viscosity to gasoline. At higher injection pressure of 7 and 11 MPa, the penetration length difference between ethanol and gasoline sprays decreases and eventually ethanol sprays show a longer penetration length than that of gasoline sprays at the highest injection pressure of 15 MPa. This reversed trend is possibly because the penetration regime is changed such that the tip penetration is limited by aerodynamic drag force applied to fuel droplets, instead of the injection velocity or nozzle loss of the liquid jet. It is suggested that with increasing injection pressure, the fuel jet atomisation and droplet breakup enhance and therefore the lower aerodynamic drag associated with higher droplet size of ethanol sprays than that of gasoline sprays leads to a longer penetration length. The same trends of spray penetrations of ethanol, gasoline, and iso-octane are observed in the warmed optical engine with overall higher tip penetration length than that in the cold spray chamber primarily due to decreased air density and increased fuel temperature.

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INTRODUCTION

Ethanol produced from corn or sugar cane has penetrated into the market. For instance, 10% blend of ethanol in conventional gasoline is widely available in some countries. While the current market expansion has a certain limit due to the issue of food versus fuel, the advancement of production technologies using non-food-based feed stocks such as crop residues, wood residues, algae and other cellulosic materials could introduce ethanol/gasoline fuels with higher blending ratios. Therefore, ethanol utilisation in spark-ignition (SI) engines is of high interest to engine manufactures and research community [1]. The usage of ethanol is known to improve the SI engine performance. For instance, the higher octane number of ethanol compared to conventional gasoline results in higher knock resistance, which allows for a higher compression ratio and thus improved fuel conversion efficiency [2, 3]. Further

advantages of ethanol over gasoline such as lower flame temperature, shorter burning time, broader flammability limits and higher flame speed are also reported in literature [3, 4].

While ethanol is widely used in port-fuel-injection (PFI) engines, it can be better utilised in spark-ignition direct-injection (SIDI) engines because the higher heat of vaporisation of ethanol can maximise the charge cooling effect that leads to increased volumetric efficiency [5]. Indeed, down-sized engines using a SIDI system and turbocharger rapidly takes over PFI engines in a wide range of vehicle categories due to improved fuel economy, better transient response and higher potential for system optimisation [6, 7, 8]. The latest design of SIDI injectors typically includes a multi-hole nozzle and high-pressure fuel rail (~15 MPa) to optimise multi-plume sprays inside the engine cylinder. Fuel sprays in this SIDI system are the result of high pressure driven liquid fuel jets injected through a nozzle orifice into a combustion

chamber. The jet atomises, and then the fuel evaporates and mixes with air to form a reactive charge which, as a consequence of increased gas pressure and temperatures, ignites and thus initiates the combustion process. Therefore, fuel jet atomisation and droplet breakup strongly influence the combustion and pollutants formation [9].

The successful implementation of SIDI fuel injection system requires clear understandings of the temporal and spatial development of spray plumes. For example, in order to improve the engine efficiency at part load, stratified charge and overall lean combustion is needed [8]. Early SIDI systems using swirl-type nozzles relied on the piston bowl to achieve the overall lean stratified charge, which also introduced increased unburnt hydrocarbon (uHC) and soot emissions [6, 8, 10]. In this regards, a multi-hole nozzle has benefits over swirl-type injectors due to better atomisation and droplet breakup [5, 8, 9]. One of key control parameters of a hole-type SIDI injector is the injection pressure. It was reported that better atomisation due to increased injection pressure can improve the combustion stability [8]. Also it is well known that the tip penetration of sprays increase with increasing injection pressure while the droplet size decreases [11, 12].

Another important question raised for SIDI injection systems of today is the effect of ethanol on fuel sprays. Many have studied how ethanol sprays develop differently to gasoline sprays to address this issue; however, existing literature provides virtually no consensus even for a simple question on the spray tip penetration. For example, some studies [10, 13, 14] reported a higher tip penetration of ethanol sprays than that of gasoline sprays in an SIDI fuelling system. It was expected considering higher density, boiling point, latent heat and surface tension of ethanol than those of gasoline. However, others [15, 16] presented the reversed trend despite the use of nearly the same SIDI systems. It was speculated that the heavy components in the conventional gasoline (heavier than ethanol) caused a longer penetration.

The present study aims to clarify potential causes of mixed conclusions on the tip penetration of ethanol and gasoline sprays. To this end, high-speed Mie-scattering imaging of SIDI sprays was performed and ethanol and gasoline fuels were compared in terms of spray tip penetration and penetration rate. Considering the importance of injection pressure on SIDI spray development, a wide range of injection pressures were also tested. The Mie-scattering imaging was first performed in a constant-volume, constant-flow chamber allowing a full optical access for both side-view and bottom-view imaging of fuel sprays. The same high-speed imaging was repeated in a single-cylinder optical SIDI engine to verify whether the findings derived from the spray chamber were relevant to practical engine conditions.

EXPERIMENTS

Constant-Flow, Constant-Volume Spray Chamber Setup

The fuel sprays were first studied in an optically accessible, constant-flow, constant-volume chamber as shown in Fig. 1. The glass windows, located at three sides of the chamber, allowed imaging of both the side- and bottom-view imaging of the sprays. A 45° mirror was used to facilitate the illumination and the imaging of the sprays from different orientations. A list of injection and ambient conditions are shown in Table 1. The fuel supply system consisted of a fuel tank, high-pressure pneumatic pump, custom-made high-pressure rail and an SIDI injector. The SIDI injector (Continental DI XL 2) had 6 holes and was designed to be mounted at the side of the bore and near the intake ports in a base engine. In the chamber, it was mounted vertically at the upper lid, which was controlled using an injection driver (Zenobalti ZB-5100G). An injection duration of 0.5 ms was selected so that the injection ended before the spray tip went out of the field of view, but was sufficiently long to ensure that the injector was operating in a non-ballistic mode. Four different injection pressures (*viz.* 4, 7, 11 and 15 MPa) were tested to investigate the effect of injection pressure on the spray penetration for ethanol, gasoline and iso-octane fuels.

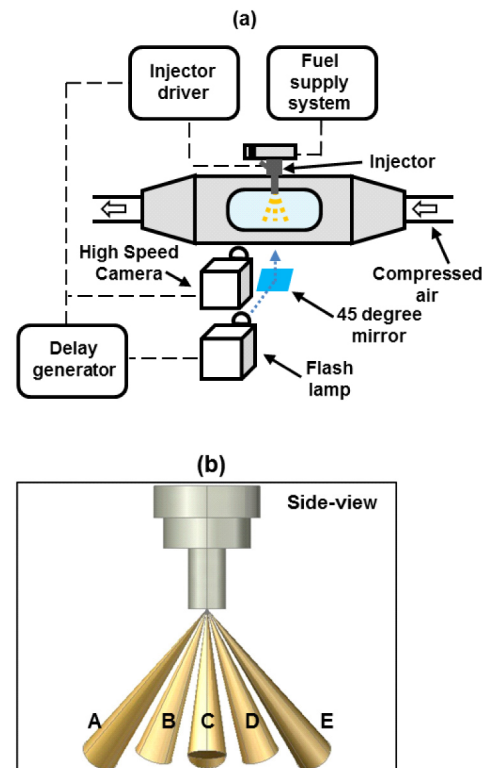


Figure 1. (a) Schematic diagram of the spray chamber and Mie-scattering imaging setup and (b) nozzle hole configuration and spray orientation of side-view images and (c) bottom-view images.

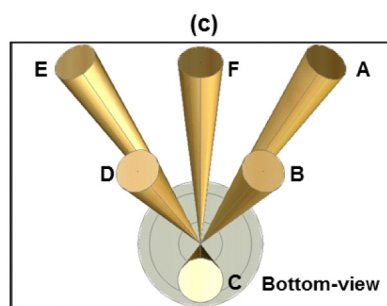


Figure 1. (a) (cont.) Schematic diagram of the spray chamber and Mie-scattering imaging setup and (b) nozzle hole configuration and spray orientation of side-view images and (c) bottom-view images.

Table 1. Spray chamber operating conditions.

Injector	Continental DI XL 2
Number of holes	6
Hole diameter (mm)	0.204
Injection duration (ms)	0.5
Injection pressure (MPa)	4, 7, 11 and 15
Chamber ambient pressure (MPa, abs)	0.2
Chamber ambient temperature (K)	293
Fuel temperature (K)	293

The ambient gas pressure (0.2 MPa) was selected to simulate the in-cylinder gas pressure of early compression stroke of a target SIDI engine. A constant flow of compressed air was supplied to this end. The ambient temperature and chamber wall temperature were fixed at 293 K throughout the experiment and therefore cold fuel jets (293 K) were issued into cold ambient gas.

Single-Cylinder Optical SIDI Engine Setup

While the spray chamber offered an easy optical access for spray imaging, the ambient conditions could not be realistic due to the lack of in-cylinder flows. Also the ambient pressure and temperature were fixed during the injection event, which could not simulate transient in-cylinder pressure and temperature associated with piston movement in a real engine. The chamber used in the present study was also limited to cold conditions as mentioned previously. For the investigation of fuel sprays of ethanol, gasoline and iso-octane at practical conditions, a single-cylinder optical engine was used. The schematic of the optical engine, imaging setup, the field of view through the quartz piston crown, and the orientation of the injector are shown in Fig. 2. A donut-shape quartz window was mounted between the engine cylinder head and liner to provide an optical access to the engine combustion chamber. The in-cylinder spray development could also be visualised through the piston crown that was made of quartz, via a 45° mirror located in the void of the extended piston. The specifications and operating conditions of the optical engine are summarised in Table 2. The engine had a 86 mm bore and 86 mm stroke, which resulted in a displacement volume of 500 cm³. The geometric compression ratio was 10.5. The valve system

consisted of a double overhead camshaft and four valves (two intake and two exhaust valves). The valve timings are as listed in Table 2. It should be noted that the 0 crank angle degree (°CA) corresponding to intake top dead centre (TDC) in this paper, and the crank angle timings in the subsequent sections will be presented with respect to this. The spark plug (not used in this study) was centrally mounted in the engine head and the SIDI injector was side-mounted between the intake valves.

The engine was motored at a fixed speed of 1200 revolutions per minute (rpm) using a 37-kW AC motor. To simulate fully warmed-up running engine conditions, the coolant, which was circulated through the engine head, engine block and cylindrical liner, was heated to 363 K. Therefore, the temperature of the injector that was mounted within the engine head was also heated to a temperature of 363 K. It is noted that a settling time of 30 minutes and an injection frequency of 1 Hz was adopted to ensure that the fuel temperature within the injector corresponds to that of the injector body [6]. The temperature of the intake air was not controlled but measured to be 320 K throughout the experiments. The engine was operated at a wide-open-throttle condition and therefore 0.1 MPa intake pressure was used for data analysis. The stable supply of intake air was possible due to the use of a large volume (60 litres) intake surge tank. The same size exhaust surge tank was also used to minimise pressure fluctuations of the single-cylinder engine. The same SIDI injector and other fuelling system used in the spray chamber were installed in the optical engine. Synchronisation of various control triggers for the SIDI injector, camera and lighting was achieved by using a universal engine controller (Zenobalti ZB-9013P) using signals from a rotary encoder (Autonics E40S8) mounted on the crank shaft. The injection timing was held constant at 270°CA after top dead centre (aTDC). The same injection duration (0.5 ms) and injection pressures (4, 7, 11, and 15 MPa) of the spray chamber experiments were used for the optical engine diagnostics.

The engine in-cylinder pressure was recorded by using a fiber-optic pressure transducer (Optrand C822J6-SP) embedded in the spark plug. The crank angle resolved in-cylinder pressure for all the injection cycles were acquired and plotted as shown in Fig. 3. From the figure, it is evident that there were no significant cycle-to-cycle fluctuations in motored pressures, thus ensuring that the fuel injection events were compared at nominally identical conditions. The peak motored pressure of 1.96 MPa was measured near TDC, which corresponded to 753 K bulk gas temperature. In the figure, in-cylinder pressure and temperature at the time of start of injection at 270°CA aTDC are illustrated. It is shown that the in-cylinder pressure of 0.2 MPa is consistent with the ambient pressure of the spray chamber; however, the in-cylinder temperature of 393 K is 100 K higher than the spray chamber temperature. Therefore, spray imaging in the optical engine was performed at warmed conditions, as opposed to the cold-start conditions of the spray chamber.

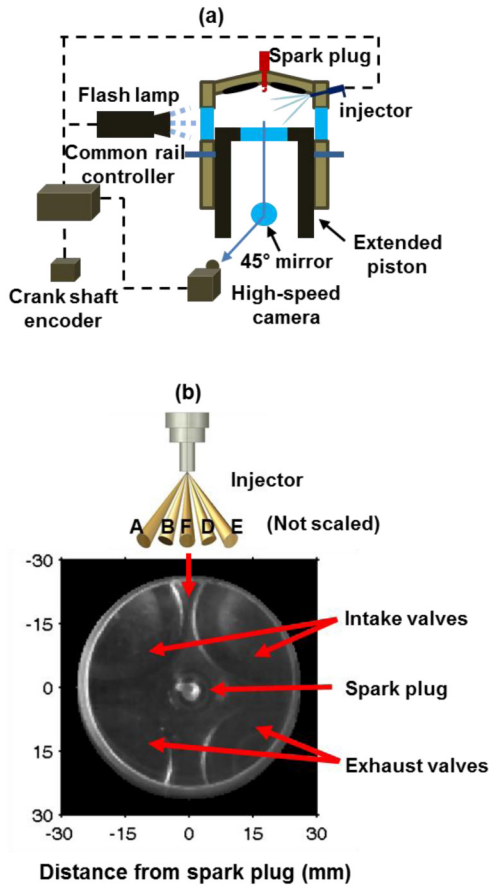


Figure 2. (a) Schematic diagram of the optical SIDI engine and Mie-scattering imaging setup (not to scale) and (b) field of view through the optical piston crown and the orientation of the injector with nominal spray pattern.

Table 2. Engine specifications and operating conditions.

Engine Specifications	
Displacement (cm ³)	500
Stroke (mm) × Bore (mm)	86×86
Compression ratio	10.5
Valve system	Double overhead camshaft
Intake valve open	7°CA aTDC
Intake valve close	247°CA aTDC
Exhaust valve open	496°CA aTDC
Exhaust valve close	0°CA aTDC
Engine Operating Conditions	
Engine speed (rpm)	1200
Engine coolant temperature (K)	363
Temperature in intake manifold (K)	320
Intake pressure (MPa, abs)	0.1
Injection pressure (MPa)	4, 7, 11 and 15
Injection timing (°CA aTDC)	270
Injection duration (electronic, ms)	0.5

Mie-Scattering Imaging Setup

Mie-scattering imaging of fuel sprays was performed using a flash lamp and high-speed camera. The setup details are listed in Table 3. The sprays were illuminated using a 400-W lamp (Multiblitz Profilux Plus 400) with approximately 1 ms of flash duration, which was long enough to record the entire injection event of selected conditions. For homogenous illumination, a semi opaque Perspex sheet was placed in front of the lamp. The scattered light from the liquid jet and fuel droplets was captured by a 14-bit high-speed CMOS camera (VisionResearch Phantom v7.3) through a 50 mm, f/1.4 lens (Nikon AF Nikkor). The frame rate of the camera was set at 10,000 frames per second (fps). In the spray chamber, the images were acquired with the pixel resolution of 0.15 mm per pixel for side-view imaging and 0.25 mm per pixel for bottom-view imaging. The exposure time was set at 0.01 ms and the lens f-number was selected at 2.8 for side-view imaging and 16 for bottom-view imaging considering both the field of view and image clarity. In the optical engine, a slightly different setup was used to reflect thicker piston quartz window that moved upwards during the injection. Specifically, the pixel resolution of 0.33 mm per pixel, the exposure time of 0.001 ms, and the f-number of 16 were used. The same Mie-scattering imaging was repeated for 15 injections to address injection-to-injection variations for all tested conditions of this study. It is noted focus target was imaged at the beginning of the experiments to ensure that the camera was in focus.

Image Processing

All images collected were corrected for background and detector attenuation [17]. Each corrected image was subsequently converted to a binary image using Otsu's boundary detection model - a widely used threshold-based method, prior to the derivation of the spray tip penetration from the spray boundaries. Figure 4 shows an example image of gasoline sprays obtained in the spray chamber at 0.9 ms after the start of injection for (a) bottom-view imaging and (b) side-view imaging. As mentioned previously, the injector nozzle had 6 holes (spray plume A~F) with the same hole diameter of 0.204 mm but various spraying angles (see Fig. 1 for the hole configurations and identification of spray plumes). As a result, in Fig. 4(a), spray plume F appears to be hidden behind spray plume C. Also in Fig. 4(b), spray plumes D and E as well as A and B show overlapping. These complicated the processing of individual spray plumes. Therefore, spray plume C was selected for the measurement of tip penetration. Figure 4 shows that plume C is not interfered by other spray plumes both in side-view and bottom-view images, making it the best candidate for the tip penetration measurement. Therefore, the spray tip penetration is derived from the distance from the tip of plume C to the injector nozzle. The vertical and horizontal tip penetrations of the central plume were first obtained from the side-view and bottom-view images, respectively, as illustrated in Fig. 4. The tip penetration along the central axis of the injector was subsequently derived from these distances using trigonometric function.

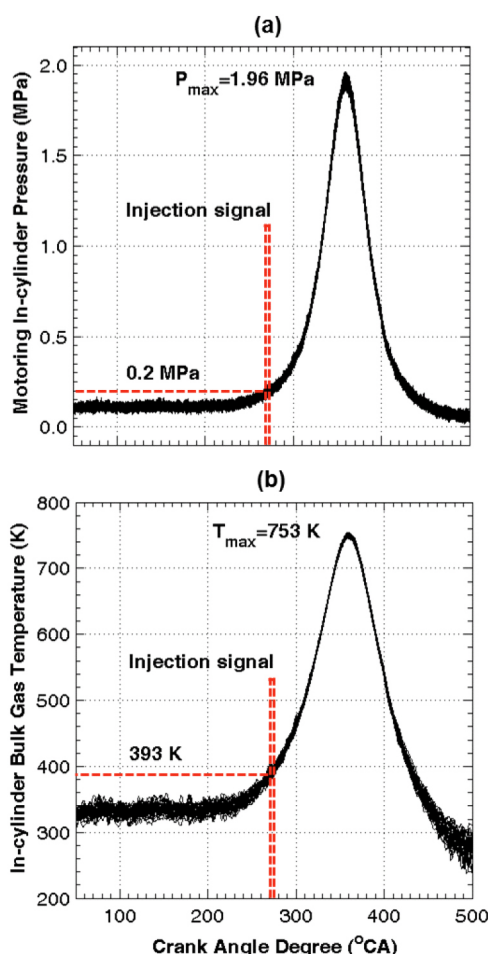


Figure 3. (a) Motored in-cylinder pressure trace versus crank angle and (b) estimated in-cylinder bulk temperature versus crank angle. The in-cylinder pressure and temperature at the time of injection is illustrated by dashed lines.

Table 3. Mie-scattering imaging setup.

Light source	Multiblitz Profilux Plus 400 flash lamp	
Flash duration (ms)	1	
High-speed CMOS camera	Vision Research Phantom v7.3	
Frame rate (frames per second)	10,000	
	Spray chamber	Optical engine
Pixel resolution (mm per pixel)	0.15 (side-view) 0.25 (bottom-view)	0.33
Exposure time (ms)	0.01	0.001
f-number	2.8 (side-view) 16 (bottom-view)	16

Selected Fuels

One multi-component fuel (conventional gasoline) and two single-component fuels (neat ethanol and iso-octane) were selected for spray imaging in this study. Various physical

properties of the investigated fuels are listed in Table 4. The fuel properties are shown for 293 K and 101.3 kPa conditions for fuel-to-fuel comparison purposes. It is noted that fuel density, surface tension, and latent heat are in increasing order of iso-octane < gasoline < ethanol. On the contrary, the viscosity of ethanol shows two-fold higher value than the other fuels while iso-octane and gasoline show similar viscosity. Previous studies [9, 10] reported that fuel viscosity impacts the SIDI spray development significantly. Therefore, viscosity of each fuel is plotted as a function of temperature in Fig. 5. It should be noted that two single-component fuels with low and high carbon numbers (*i.e.* n-pentane and o-xylene) are used to represent the multi-component gasoline. The vertical dashed lines are to indicate fuel temperatures in the spray chamber (293 K) and optical engine (363 K). Figure 5 shows that the viscosity decreases with increasing temperature; however, the trend of higher viscosity of ethanol than other fuels stays the same. How the high viscosity of ethanol impacts the spray penetration compared to gasoline will be one of the main discussion points in the following sections.

For all fuels tested in this study, physical properties summarised in Table 4 together with ambient/injection conditions listed in Tables 1 and 2 can be used to estimate the droplet sizes (*e.g.* SMD: Sauter Mean Diameter) using empirical correlations [18, 19] as in Eq. (1):

$$SMD = 0.38dRe^{0.25}We_L^{-0.32}(\mu_L/\mu_G)^{0.37}(\rho_L/\rho_G)^{-0.47} \quad (1)$$

where d is the diameter of the nozzle, Re and We_L stand for Reynolds number and Weber number, μ_L and μ_G are the liquid viscosity and ambient gas viscosity, ρ_L and ρ_G are the densities for liquid and ambient gas. For example, the droplet sizes of the three fuels at four injection pressures tested in the present study and at 0.2 ms after the start-of-injection (aSOI) are plotted in Fig. 6. From the figure, it is observed that ethanol has a higher droplet size at all injection pressures than gasoline or iso-octane. This was expected given that ethanol had a higher density, viscosity, surface tension value and lower We_L than gasoline or iso-octane [6, 20]. From the figure, it is also observed that the droplet size decreases with increasing injection pressure, consistent with other studies (*e.g.* [11, 21]). It should be noted that the empirical correlation was originally developed for diesel sprays and therefore constants and powers used in Eq. (1) might not be directly relevant to SIDI sprays of this study. However, the predicted values using this correlation have been shown a good qualitative agreement with the measured values of SIDI sprays in the previous study [10]. This, combined with the knowledge that the estimated values follow the expected trend with fuel type and injection pressure, justifies the use of Eq. (1) for fuel-to-fuel comparison purposes. The SMD trends will be referred in the following discussions when jet atomisation and droplet breakup is considered to explain the variations in spray penetration depending on fuel types.

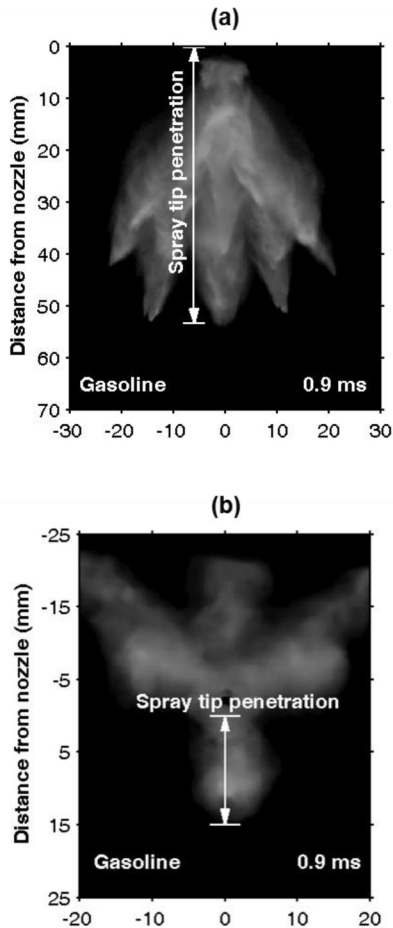


Figure 4. Sample images of gasoline sprays obtained in the spray chamber at 0.9 ms aSOI for (a) side-view and (b) bottom-view.

Table 4. Selected properties of the investigated fuels at 293 K and 101.3 kPa.

Properties/Fuel	Gasoline	Ethanol	Iso-octane
Density (kg m^{-3})	748	789	692
Surface tension (mN.m^{-1})	21.58	22.66	19.35
Latent heat at 298 K (kJ.kg^{-1})	364	902	305
Viscosity (mPa.s)	0.55 (mean)	1.20	0.50
Boiling point (K)	303-473	351.5	372.8

RESULTS AND DISCUSSION

Spray Chamber Measurement

The typical spray images of ethanol, gasoline and iso-octane at four injection pressures (viz. 4, 7, 11 and 15 MPa), as imaged from the side-view (a) and bottom-view (b) of the spray chamber are presented in Figs. 7, 8, 9. For each fuel, the images are presented from 0.3 to 0.9 ms aSOI at a fixed time interval to investigate a temporal evolution of spray plumes. As mentioned previously, the framing rate was fixed at 10,000 fps, resulting in a spray image every 0.1 ms. Therefore, the images in Figs. 7, 8, 9 are examples selected for presentation purposes. Also the images are from an instantaneous cycle out of a total of 15 injections for each fuel and injection pressure.

The corresponding time aSOI is indicated in the bottom-right corner of the first row images for each fuel. From Figs. 7, 8, 9, it is observed that the multiple spray plumes develop distinctly and the spray boundary appears to be clear in the quiescent and cold environment of the spray chamber. The plume-to-plume interactions are minimal although gasoline and iso-octane sprays show a wider spreading of each spray plume than that of ethanol sprays, suggesting a higher degree of plume-to-plume interactions which are attributed to the dispersion of the slow-moving smaller droplets [12]. Close inspection of the images suggest that the spray development could be divided into two stages over the injection event. In the earlier penetration stage (i.e. 0.3 ms aSOI) the sprays are found to show a slightly wider spreading angles than later time aSOIs and the plume structure of the sprays are difficult to identify. In the later stage (i.e. 0.5 to 0.9 ms aSOI), the sprays show smaller spreading angles and the multi-plume structure becomes clearer. It is noted that the spray spreading angle here refers to the spreading angle between the two outer plumes, as the overlapping of the adjacent spray plumes prevented the accurate determination of the spreading angle of a single spray plume.

Spray Tip Penetration

Comparisons of ethanol, gasoline and iso-octane spray tip penetrations at four injection pressures (viz. 4, 7, 11 and 15 MPa) are presented in Fig. 10. The values plotted represent the mean over 15 injections. Error bars (1 standard deviation) are used to show the statistical uncertainty of each data point. Trend lines, which are obtained by fitting second-order polynomial curve fits over the cycle averaged values, are used to give a better representation of the general data trends. The appropriateness of the polynomial was confirmed using residual analysis. The small error ranges confirm that the differences in the tip penetration exceed the cyclic fluctuations. It is noted that the overlapping of adjacent spray plumes (see Figs. 7, 8, 9) prevented the accurate determination of the central plume (plume 'C') penetration at the early injection stage. Hence the data are only presented from 0.2 ms aSOI onwards. Insert plots that represent the close-up view of spray tip penetration from 0.8 to 0.9 ms aSOI are also presented for the purpose of clarity. From the figure, it can be observed that the spray penetration is found to increase with increasing injection pressure, which is well expected. It is worth noting that the end-of-injection (EOI) for the sprays occurs at approximately 0.7 ms aSOI, about 0.2 ms of delay between the electronic signal and actual EOI seen in the images. This EOI delay was found to be insensitive to the injection pressure variation.

Figure 10 shows an interesting trend of tip penetration length of ethanol sprays compared to gasoline sprays, with respect to the injection pressure. It is observed that the spray tip penetration for gasoline is higher than that of ethanol at low injection pressure of 4 MPa. This observation is consistent with our previous study of PFI sprays operating at much lower injection pressure of 250 kPa [3]. However, this trend is changed with increasing injection pressure. At high injection pressures, the tip penetration lengths of ethanol and gasoline sprays are nearly