

Spray Characterization of Ethanol Gasoline Blends and Comparison to a CFD Model for a Gasoline Direct Injector

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ABSTRACT

Operation of flex fuel vehicles requires operation with a range of fuel properties. The significant differences in the heat of vaporization and energy density of E0-E100 fuels and the effect on spray development need to be fully comprehended when developing engine control strategies. Limited enthalpy for fuel vaporization needs to be accounted for when developing injection strategies for cold start, homogeneous and stratified operation.

Spray imaging of multi-hole gasoline injectors with fuels ranging from E0 to E100 and environmental conditions that represent engine operating points from ambient cold start to hot conditions was performed in a spray chamber. Schlieren visualization technique was used to characterize the sprays and the results were compared with Laser Mie scattering and Back-lighting technique. Open chamber experiments were utilized to provide input and validation of a CFD model. In addition to the fuel variation and operating conditions three

different injectors were investigated with differences in static flow rate and internal nozzle geometry. This empirical data was used in the initial development of a robust CFD model for predicting spray behaviors from a multi-hole gasoline direct injector. Comparisons of spray penetration were made between the optimized CFD model and the empirical results. Image processing techniques were presented for characterizing the spray images to quantify the penetration and the vapor cloud development.

3D-CFD numerical simulation is commonly used in order to make selections of engine injection in the early design period. In this article, the spray behavior was predicted by CONVERGE. The characteristics of the spray tip penetration and mass of the liquid and vapor phases were calculated under different temperature and pressure conditions,

Testing was also conducted at realistic chamber operating conditions for stratified operation. In these test the piston crown geometry was included in the spray chamber to

evaluate fuel impingement effects. Schlieren imaging provided insight into the effect of spray impingement resulting in spray bounce and fuel puddles.

INTRODUCTION

Ethanol has been identified as alternative fuel to displace gasoline in automotive applications. Two significant advantages of ethanol are its high Octane number providing excellent knock resistance, and high latent heat of vaporization which promotes charge cooling for increased power density [1, 2]. However, ethanol has 33% reduction in Lower Heating Value (LHV) on a volume basis in comparison with gasoline. It is also a preferred fuel for a turbo charged direct injection engine, which is able to increase compression ratio and achieve further improve of fuel economy by engine downsizing because of its anti-knock characteristic [1]. Not much difference has been reported between ethanol and gasoline spray [3], but detail study is needed for better understanding of spray formation for developing optimized injection strategies.

Gasoline Direct injection (GDi) has potentials for enhanced power output and better fuel economy, improved transient response, and reduced cold-start hydrocarbon emission levels if it comes with precise control of fuel-air mixture formation. In a side-mount GDi engine, an injector is mounted on the cylinder wall and injects fuel directly into the combustion chamber. GDi engines are characterized by less pumping loss and higher compression ratio, and better volumetric efficiency comparing to Port Fuel Injection (PFI) engines. Thus GDi engines with well optimized spray structure and injection strategy are required to meet the ever-tightening emission standards and fuel economy regulations [4, 5]. Since a GDi engine delivers fuel at compression stroke when it runs in a stratified mode, piston wetting and resultant harmful emissions are major issues [6]. Details of GDi engines are discussed in the [reference](#) [7]. Combination of ethanol-gasoline blended fuel and GDi engine has been studied widely [1, 3, 8,9,10].

In a GDi engine, spray development is critically related to mixture formation and the injection strategy must be well-optimized in order to exploit potential of the engine. Although swirl injectors had widely studied for GDi engine [11], multi-hole injectors were found to be more suitable for spray-guided GDi engines because they offer advantages of stable spray pattern and flexibility in spray plume targeting [12]. Especially, independence of spray cone angle on ambient pressure is preferred for DI operation [13].

If sprays hit a piston bowl during stratified operation, this impingement should be studied more for better understanding of mixture formation. It has been reported that the piston impingement must be taken into account for a better simulation result [3]. Over-estimation of volumetric

efficiency for ethanol fuel was found and a suggestion was presented to create a sub-routine to accurately simulate impingement, vaporization, and heat transfer on a piston surface. The previous work on wall impingement discovered that the wall temperature strongly affected the vapor phase propagation after the spray hit the surface [14].

Schlieren visualization method is one of the most effective techniques to visualize non-homogeneous transparent flow fields, such as a vapor phase of sprays. Using the Schlieren technique, it is able to visualize changes of the refraction indexes and density gradient in an object caused by material and temperature difference [15]. [Fig. 1](#) shows a basic principle of the Schlieren method. Suppose parallel light is coming through the object. If the object is entirely homogeneous, the light fluxes are accumulated at the focal point by the magnifying lens and project an image on the screen. However, if there is inhomogeneity in the object, the light flux passing through the inhomogeneity area may be refracted as the red solid line in the figure shows and off the focus of the lens. Therefore the intensity change in the resultant image can be observed, and its difference can be clearer by cutting the refracted light by a knife edge [16, 17].

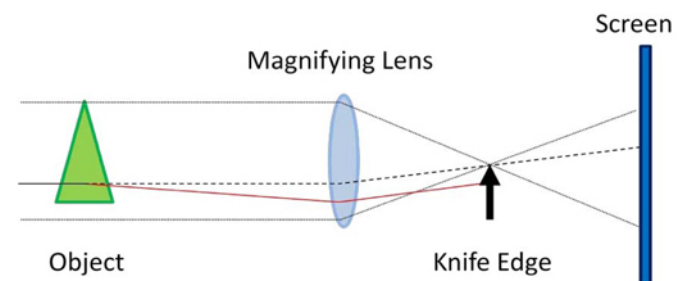


Fig. 1. Basic scheme of Schlieren method

3D-CFD numerical simulation is commonly used in order to make selections of injection strategy in the early period of engine design. Many studies have been done in numerical simulation of spray development and mixture formation, but it needs to be more accurate to understand mixture formation of DI engine which is very critical for better fuel efficiency and emission [6, 18,19,20,21]. In spray simulation, spray breakup is a critical event and studies have been conducted to model the discrete phase of spray. The Taylor Analogy Breakup (TAB) model [22] is a classic method for calculating drop breakup. The Kelvin-Helmholtz (KH) model which represents the primary breakup, and Rayleigh-Taylor (RT) model representing the secondary breakup are presented by Ricart and Reitz [23] and suggested to adjust the model constant with respect to experimental data from spray experiments [24]. The KH model and RT model are combined to simulate the spray [25] which showed good agreement with the experiment results. O'Rourke collision and coalescence model has been widely used [26]. An alternative to the O'Rourke numerical collision scheme is the

No Time Counter (NTC) method of Schmidt and Rutland [27]. In this article, the spray behavior was predicted by CONVERGE, which is a KIVA-based commercial code.

In this study, high-speed spray visualization was performed to understand spray behavior in various fuels, injection strategies, and surrounding conditions. Initial numerical simulations were also carried out and the results were compared with the experimental results.

EXPERIMENTAL SETUP

The experiment was performed using a high-speed digital camera to characterize the spray structures under typical fuel injection conditions for a DI engine.

Three fuels were tested to examine the influence of ethanol content in gasoline fuel. They were 100% pure ethanol (E100), RON-91 gasoline, and the mixture of two in 50% of volume ratio (E50). The specifications for the fuel are listed in Table 1. The differentiating properties of ethanol are its LHV and higher latent heat of vaporization. The stoichiometric Air/Fuel ratio is lower than gasoline because ethanol has oxygen in its molecular structure. Since ethanol is single component fuel, it has the constant boiling point of 78.4C at the standard state. Therefore the distillation curve of ethanol is nearly flat and very different from gasoline's.

Table 1. Fuel properties of E100, E50, and gasoline

	E100	E50	Gasoline
Density [kg/m3]	789	764	739
Lower Heating Value [MJ/kg]	26.8	34.7	43.1
Stoichiometric Air/Fuel ratio	9	13	14.7
Latent Heat of Vaporization [kJ/kg]	904	-	380-500
Boiling Temperature [C]	78.4	-	25-215
Research Octane Number	129	-	91

<table 2 here>

The experiment was performed with three different injectors. Injector A was a production DI injectors and also called “baseline” injector. Two other injectors were prototype injectors. The specifications are summarized in Table 2. All three injectors had different hole geometries, mass flow rate, and spray targeting. The pictures taken by the camera were the projected side view image. The 3D structure of spray A is shown in the NUMERICAL CONDITIONS section.

Fig 2 shows the experimental setup. The injector was fixed at the top of the test chamber, which can be heated up to 250C by the air transmitting through the circulation heater. The chamber pressure can be controlled from 0.4bar with the vacuum pump, to 4bar to simulate typical engine operating conditions. At the time of injection, two valves at upstream and downstream of the chamber were closed to produce

quiescent ambient. The fuel temperature was controlled by a water jacket surrounding the injector which was capable to maintain the injector temperature from 25C to 80C with some exceptions. Since the injector tip was exposed to hot air at the heated conditions, it was impossible to maintain the injector cool at low temperature in the extreme conditions such as 200C chamber temperature with 25C fuel temperature. The injection pressure can be changed by regulating the nitrogen pressure at the fuel tank. An injection signal was generated at the signal generator and transferred to the Delphi GDi Injector Driver to fire the injector. The injection signal was also transferred to the high-speed digital camera as a trigger in order to synchronize the camera to the injection. For the piston impingement testing, a dummy piston head was placed in the chamber with angle of 23 deg to replicate the piston injector orientation of a side-mounted GDi engine.

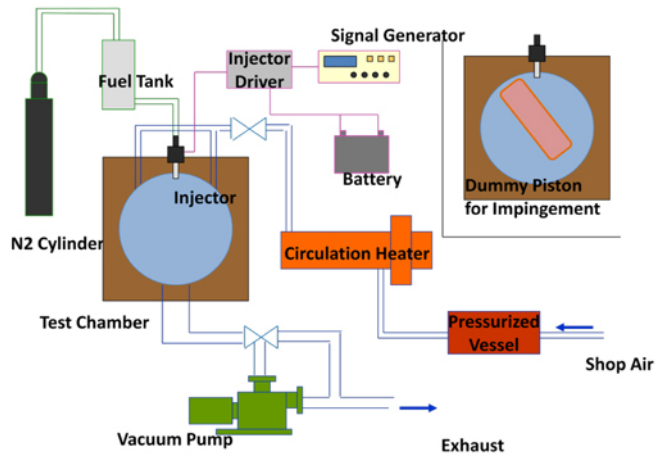


Fig. 2. The experimental setup

Fig. 3 shows the basic scheme of the Schlieren setup. The light source was a projection lamp. The light coming from the source passed the pin hole, which was placed on the focal length of the first magnifying lens. As a result, the beam after the lens became parallel. Then the parallel light proceeded through the chamber and reached the other magnifying lens. On the focal point of second lens, there was a knife edge placed to cut the refracted beam. And finally, the beam came into the high speed digital camera, which speed was set to 8213 frames per second (FPS) with a resolution of 512 × 512 pixels (76mm × 76mm). This “straight” layout of Schlieren takes space to set up compared with complex layout such as in Ref [15], but the simplest optical passage guarantees good quality of the resultant image.

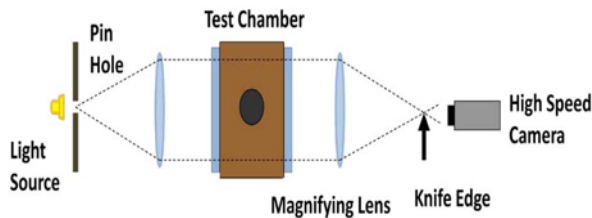


Fig. 3. Schlieren visualization setup

IMAGE PROCESSING

The images taken by the high speed camera were processed for a quantitative study. In order to represent a spray shape, spray penetration, spray angle, area of projected image, and position of the centroid of the spray, were measured. Three injections were recorded with the identical conditions and processed to evaluate consistency. All scripts were written and run by MATLAB.

First, a raw image was subtracted from a background image to reduce the effect of inhomogeneity of the light source. This also eliminated black spots caused by dusts. The obtained image was filtered through a Low-Pass Filter (LPF) by the Fourier Transform to make the background more uniform. The MATLAB built-in function of Fast Fourier Transform (FFT) was used for the calculation. The process and the background intensity change are illustrated in Fig. 4. Then threshold was applied to convert the figure to a binary image. However, simple thresholding could not pick up the vapor area for most cases as shown in Fig. 5. The circles indicate the point of insufficiency. In order to catch the vapor phase, 2-step thresholding is proposed which is a combination of simple thresholding and thresholded image of variance of the image. Variance indicates the degree of variety of the subjects, also known as a square root of the standard deviation, which is expressed as;

$$Var = s^2 = \frac{1}{n} \sum_{i=1}^n (x_i - \bar{x})^2$$

where n is a number of subjects and $\bar{x}$ is a mean value of the subjects. The idea is to consider the vapor area as a sort of wave motion of lean and rich (black and white) regions, where the partial (5×5 pixels for example) variance must be larger than other relatively uniform areas. After calculation of partial variance, the value was returned in the center of the small square and threshold was applied on it to make another binary image. As shown in Fig. 5, thresholded variance image was able to capture the vapor area. The center of the spray was filtered out because the thick spray cloud area had relatively uniform gray level. Then two binary images were added together and after applying Median filter for smoothing, the final binary image of spray was obtained (Fig. 5).

<figure 4 here>

The final binary images are processed to measure all four parameters; spray area, position of the centroid, penetration, and angle. The area of projected image was obtained simply counting the number of white pixels and converted to metric unit. The position of the centroid of the spray was calculated by;

$$(X, Y) = \left(\frac{\sum_{i=1}^N \sqrt{(x_i - x_0)^2}}{N}, \frac{\sum_{i=1}^N \sqrt{(y_i - y_0)^2}}{N} \right)$$

where (x_0, y_0) is a reference point, and N is a number of white pixels. The gray level of the sprays was ignored because it did not indicate the density of the spray directly.

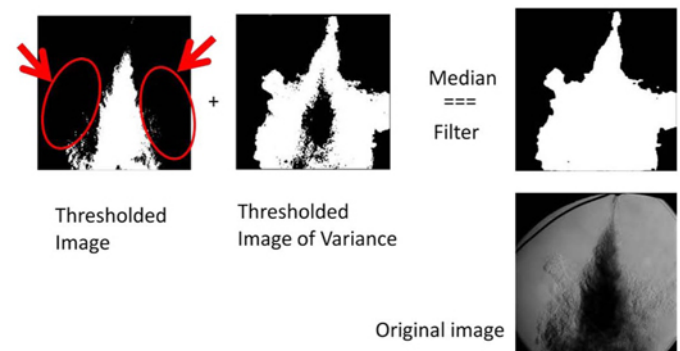


Fig. 5. Example of resultant image of a 2-step thresholding

Fig. 6 illustrates the definition of penetration and spray angle. Spray penetration was defined as a length from nozzle tip to the tip of the spray. It indicates how far the spray travels. In this paper, averaged penetrations of all plumes are presented in the RESULTS section unless distinction of plumes is impossible. Three spray angles were computed to quantify how the spray is widely distributed. Spray angle was defined as an angle between two lines (yellow lines in Fig. 6) which connect two intersections and the nozzle tip at 5, 10, and 20mm downward from the injector tip.

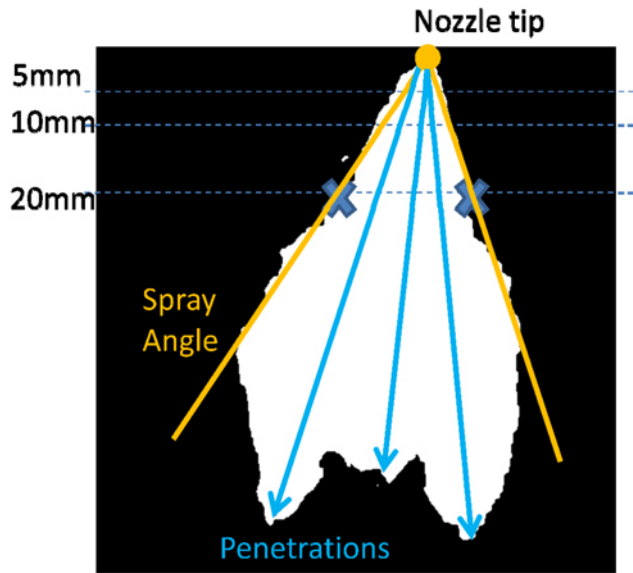


Fig. 6. Definition of penetration and spray angle

NUMERICAL ANALYSIS METHOD

Simulations of multi-hole spray were carried out with CONVERGE, a commercial KIVA-based 3 Dimensions CFD software [28]. In CONVERGE, drop “parcels” are applied in the computational domain, which represent a number of identical drops with same radius, velocity, temperature, etc. The droplet drag coefficient was determined by the dynamic drag model for accurate spray modeling [29]. The rapid distortion Reynolds Averaged Navier-Stokes (RNG) k-epsilon model is used [30]. The collision and coalescence of droplets was simulated by the No Time Counter (NTC) method which was suggested by Schmidt and Rutland [27]. The method is based on the techniques used in gas dynamics for Direct Simulation Monte Carlo (DSMC) calculations. The time rate of change of droplet radius due to ethanol vaporization is calculated from the Frossling correlation [31].

The prediction of the ethanol spray characteristics was using Kelvin-Helmholtz (KH) model and Rayleigh-Taylor (RT) model together. The KH model simulated the primary aerodynamic instabilities breakup and the RT model calculated the secondary decelerative instabilities breakup. In the KH breakup model, the initial parcel diameters were set equal to the nozzle hole diameter [28], and the breakup of the parcels and resulting drops of radius was calculated by;

$$r = B_0 \Lambda_{KH}$$

where B_0 is a model constant which typically set to 0.61 based on the work of Reitz [26]. The maximum growth rate (Ω_{KH}) and the corresponding wavelength (Λ_{KH}) were also given by Reitz.

In the RT breakup model, the scaled wavelength given by $C_{RT} \Lambda_{RT}$ was calculated to be smaller than the droplet diameter. RT waves were assumed to be growing on the surface of the drop. When the RT waves have been growing for a sufficient time (C_τ / Ω_{RT}), the drop is broken up according to the RT mechanism [32]. The breakup constants C_{RT} and C_τ in this study are set to the optimal values of 0.1 and 1.0, respectively.

For KH-RT breakup length model, the breakup length L_b can be specified as

$$L_b = \frac{B_1}{2} \sqrt{\frac{\rho_l}{\rho_g}} d_0$$

where ρ_l and ρ_g are the density of the fuel liquid and the ambient gas, d_0 is the diameter of the orifice, and B_1 is the KH breakup time constant which was determined to be 8 after the comparison between the experimental and calculated results. Only KH instabilities are responsible for drop breakup inside of the breakup length, while both KH and RT mechanisms are activated beyond the breakup length [28].

NUMERICAL CONDITIONS

The numerical grid used in this research is show in Fig. 7. The computational domain was a cylinder of Φ 150mm×180mm, which represented a constant volume vessel. The number of cells was in the range of 75,000 and 270,000, and the cell size was: 2mm for the central region, 1mm for the each nozzle direction area, and 8mm for the other area. In addition to the embedded grid control, CONVERGE is able to use Adaptive Mesh Refinement (AMR) automatically to enhance the mesh around the spray edge as shown in Fig. 7. The level of embedding for velocity, temperature, and mass fraction in this study was set to 3, which made the mesh size 1mm where AMR was turned on. The calculation was performed only for Injector A. The spray targeting and the numerical conditions are shown in Fig. 8 and Table 3 respectively.

The injection pressure and duration were 10MPa and 1.5ms respectively. Four ambient conditions were tested, which is a matrix of 25C/200C with 1bar/3bar.

<figure 7 here>

<figure 8 here>

Table 3. Numerical conditions

CFD code	CONVERGE
Coordinate system	Cartesian
Numerical space (mm)	$\Phi 150 \times 180$
Number of cell	300,000
Cell size (mm)	Basic Region: 8.0mm Embedded Region: 2.0mm Embedded Nozzles Direction: 1.0mm Adaptive Mesh Refinement (AMR): 1.0mm

RESULTS AND DISCUSSIONS

FREE SPRAY TESTING

Three different imaging techniques were compared for better understanding of spray visualization. Only injector A and E100 as fuel were used through this test. Mie scattering and back-lighting visualization, or also known as shadow photography, have been widely adopted for spray visualization [14, 33,34,35]. However, these techniques are not good for vapor visualization as Schlieren is. Throughout the Schlieren testing, the fuel temperature was fixed at 90C, while it was set to 60C for the Back-lit testing. Since the fuel temperature was not controlled during the Mie scattering visualization testing, it was assumed to be more than 90C. The injection pulse width was 1.5ms for all the cases. Some examples of raw images are shown in Fig. 9. The images of Schlieren were able to visualize the vapor cloud around the spray, which could not be seen at all in Mie scattering images. Back-lit images showed some dark area where the vapor was supposed to exist, but it was not clear enough as Schlieren. It is confirmed here that the Schlieren visualization is very effective to see a vapor envelope of a spray as well as a dense core.

Then Schlieren images were examined to evaluate the process of vaporization. The injected fuel started evaporating from the spray surface after a few hundred microsecond delay in which spray breakup occurred. Once it was vaporized, the vapor envelope lost its momentum and was overtaken by the following liquid spray. Therefore most of vapor was observed at the side of sprays at the early stage of the injection event. At the later stage, slower liquid penetration as well as accelerated evaporation due to heat/mass transfer resulted in no more overlap. This process can be seen in Fig. 10. By comparing 5MPa and 10MPa injection pressures in Fig. 9, better evaporation at the spray tip region was observed for higher injection pressure. It is believed that this was caused by better mixing due to smaller droplets and larger amount of air entrainment that higher injection pressure could produce.

<figure 9 here>

<figure 10 here>

The effect of the ambient temperature is shown in Fig. 11. Comparing 25C and 100C of chamber temperature at 1bar of ambient pressure, the degree of vaporization increased with the ambient temperature due to enhancement of heat and mass transfer. The overall spray shape did not change much. Far above the boiling temperature, “Flash Boiling” phenomena instantly increased the volume of the spray and could collapse the multiple-hole sprays into a coherent spray, with a resultant spray image resembling that of the air-assisted DI spray. The Flash Boiling of spray jet has also been discussed in references [36,37,38]. After a fuel injection, the injected liquid is exposed to huge pressure drop as well as increased temperature. Flash boiling occurs when the pressure of fuel drops instantly below the saturation pressure at certain temperature as Fig.12 shows. The saturation pressure and temperature data were taken from the Ref [39]. It is believed that the flash boiling was not observed at higher ambient pressure because the pressure drop was not large enough to end up with fuel pressure less than the saturation pressure. The complexity of flash boiling sprays include effects not only of ambient temperature and pressure, but of fuel temperature and pressure, and a combination of other effects, including injector design, cavitation, in addition to other multiple-component effects (Fig. 13). For example in Fig. 9, the reason of longer penetration of Mie scattering at 1.5ms after start of injection (ASOI) than the 5MPa Schlieren can be the existence of flash boiling due to its higher fuel temperature. If the injection pressure increases, the flash boiling effect was less noticeable because larger momentum of the spray overcame the effect of the flash boiling which was induced by sudden volume increase.

Vapor was observed at the edge of the sprays as illustrated in Fig. 11 at the chamber temperature and the pressure were 100C and 3bar respectively. The left plume of the image had more vapor on its left. And more vapor are shown in the right side of the right plume. The interior spray surface exhibit less vapor, this may be the results of both individual plume bending to the injector axis and limited air-entrainment inside the “cage” of the sprays.

<figure 11 here>

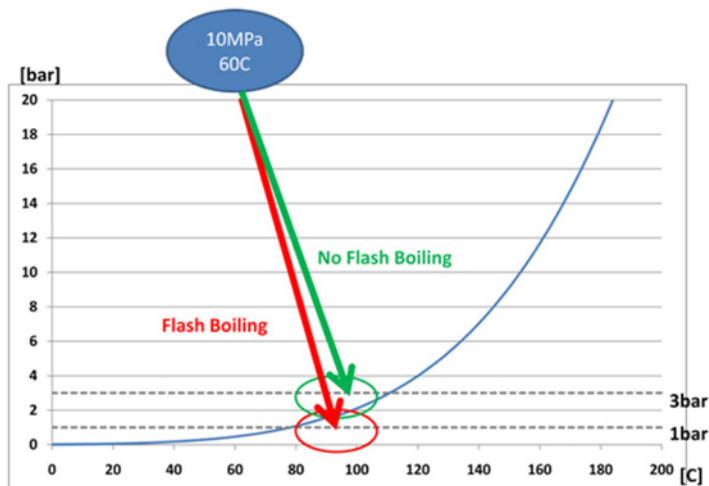


Fig. 12. Saturation pressure curve of ethanol and flash boiling

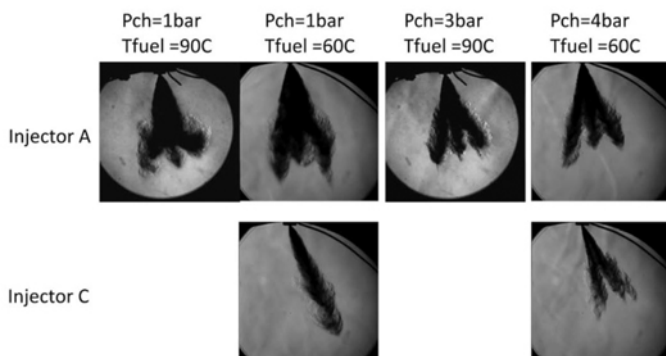


Fig. 13. Effect of chamber pressure and fuel temperature at 1.0ms ASOI and 200C of ambient temperature.

For the injector B and C, only Schlieren testing was performed (Fig. 14). In order to simulate an in-cylinder environment with a late injection timing, the temperature and pressure of the chamber were set high, 200C and 3bar respectively, and the mass of injected ethanol was fixed at 5mg for the test. The fuel temperature was 60C. It was obvious that the plumes of smaller L/D (Injector B) develop slower and wider. Even if its individual plume had wider cone angle, the sprays of the injector B remain separated from each other better than injector A due to its larger spray angle. This enlarged the available surface of the sprays and made the evaporation faster. Because of the faster evaporation and larger diameter of the holes, it was able to keep the mixture cloud near the nozzle exit, which is important to avoid excess wall wetting during engine operation.

<figure 14 here>

The effect of fuel composition of the spray structure was examined with injector A. The ambient condition was set to

simulate a warmed up homogeneous condition with high Exhaust Gas Recirculation (EGR). The ambient temperature and pressure were 200C and 1bar, fuel temperature was 60C. The injection pulse width (PW) was fixed to 1.5ms. The images are displayed in Fig. 15.

At the early stage of injection, separation of plumes in E50 and E100 spray was observed, while the plumes of gasoline collapsed. As the time elapsed, the difference became more clear. Although the vapor of gasoline was found to be widely distributed in the middle to bottom of the image, E100 vapor was detected mainly at the bottom of the image at 3ms ASOI. These results indicate that slow evaporation of E100 fuel. The liquid fuel kept penetrating and traveled farther while the evaporated fuel lost its momentum and slowed. The slow vaporization of ethanol can be considered as a result of relatively higher Initial Boiling Point (IBP), which is 78.4 degree C for 1bar. On the other hand, gasoline generally contains lighter Hydrocarbons which boiling points are lower than ethanol.

<figure 15 here>

The effect of fuel was studied quantitatively. Fig. 16 shows averaged penetration of plumes of spray with different fuel. The time started with the injection command signal and actual injection begun after 0.3ms roughly for the rest of the paper. The figure shows the results from three identical testing for each fuel, and the results demonstrated very good consistency. By 0.3ms ASOI, spray penetration was identical for all fuels. After that, gasoline spray started evaporating and lost its momentum to slow down. Although the penetration of gasoline was slower at the middle of the figure, it decelerated slower than ethanol sprays as time elapsed. This may result from the heavier components of gasoline have more resistance to complete evaporation and penetrating more than the single component ethanol spray. The shape of E50 penetration resembled E100, but it slightly slower because of evaporation of gasoline portion.

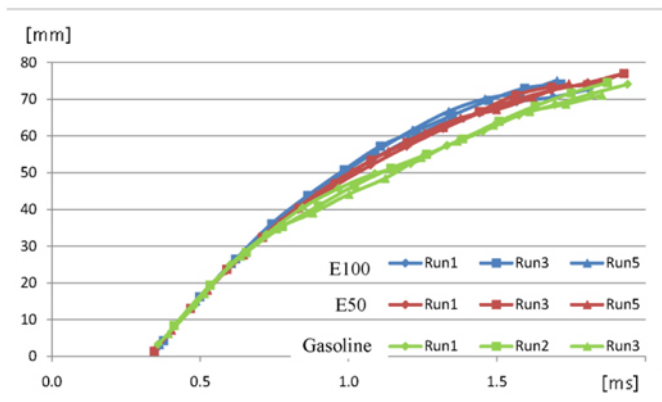


Fig. 16. Averaged penetration of spray for different fuels with fixed PW

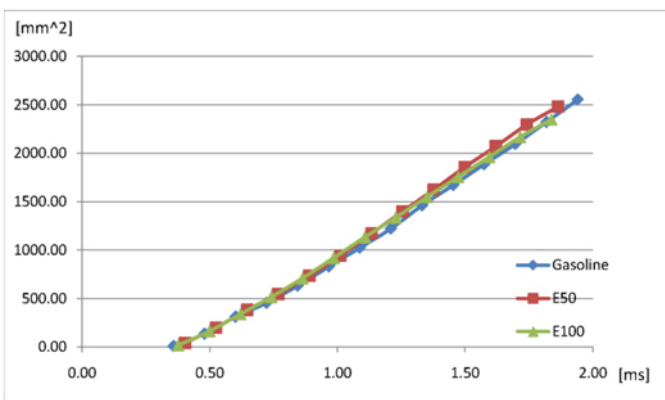


Fig. 17. Area of projected sprays for different fuels with fixed PW

The areas of projected spray and the spray angles were almost same for all fuels (Fig 17-18). Stable sprays are supposed to be an advantage of multi-hole injector, and it is confirmed experimentally. The position of the centroid of the sprays can tell the difference of fuels in Fig 19. At 1.5ms ASOI, the position of the centroid of E50 and E100 were 7.1% and 9.3% farther from the tip compared to the gasoline spray. Large differences of the position of the centroid was not observed because it was impossible to calculate the centroid after 1.8ms ASOI, due to sprays exiting the field of view as shown in Fig 15.

<figure 18 here>

<figure 19 here>

To see the effect of fuels in realistic engine operating condition, another injection study was conducted. In this evaluation, the energy content of injected fuel was kept constant to be equivalent to 5mg of gasoline. The injected mass and corresponding pulse width of the injector command signal for Gasoline, E50 and E100 were 5mg (0.38ms),

6.2mg (0.46ms), and 8.0mg (0.59ms) respectively. As a result of varying injection durations, spray shapes of different fuels would be expected to change dramatically. Fig. 20 shows the spray images of the test and Fig. 21,22,23,24 show the measured data. The difference of fuels is clearly seen in the figure qualitatively. Both low IBP and longer injection duration enhance the spray penetration. E50 had 7.9% longer penetration and 22.4% longer for E100 (Fig. 21). Increased penetration may lead to piston impingement and bore wetting resulting in increased hydrocarbon emissions and soot formation during engine operation. To minimize wall wetting, the injection timing must compensate for the effect of the fuel composition. The projected area is shown in Fig. 24, and illustrates 11.9% and 26.5% of increase for E50 and E100 with respect to gasoline at 1.5ms ASOI. And there are no significant difference in the spray angle and the position of the centroid.

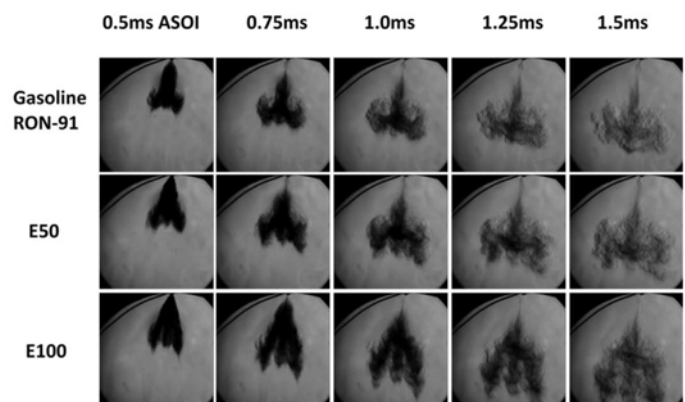


Fig 20. Effect of fuel composition. Tch=200C, Pch=1bar, Tfuel=60C, Energy content=5mg of gasoline.

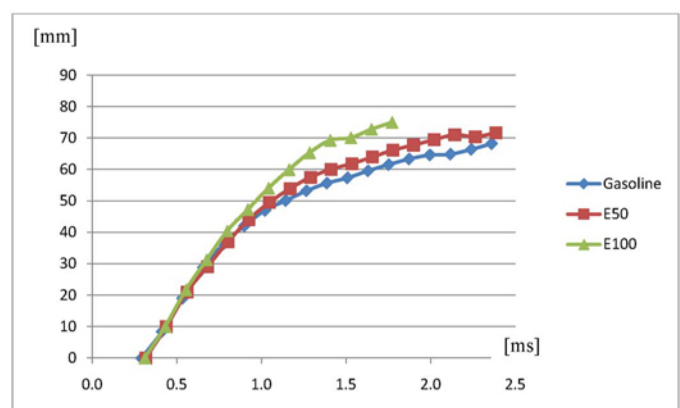


Fig 21. Averaged penetration of spray for different fuels with fixed energy input