A comparison of non-reactive fuel sprays under realistic but quiescent engine conditions for SGDI

P. Marti-Aldaravi[*,1], T. Rogers[2], E. de Benito[3], S. Hemdal[3], P. Dahlander[3]
[1]CMT - Motores Térmicos, Universitat Politècnica de València, Valencia, Spain
[2]School of Aerospace, Mechanical & Manufacturing Engineering, RMIT University, Melbourne, Australia
[3]Department of Applied Mechanics, Chalmers University of Technology, Gothenburg, Sweden

Abstract

A comparative study on the two most commonly found gasoline direct injectors is presented, where a solenoid driven multi-hole (6 horseshoe hole) and piezo driven outward opening injector were evaluated on liquid penetration, spray width and spray structure within a constant volume chamber. These three variables have been investigated for three typical fuels (iso-octane, gasoline 95 and e100 ethanol) at a fixed calorific delivery value of 389.4 J, typical combustion required value for stratified road-load operation. A first series of tests allowed correlating mass flow and injection duration for each injector and fuel. The chemical properties of the three fuels were used to calculate the injection duration for the target calorific value. This energy value was determined previously by tests in a single cylinder research engine at stratified operation. The non-impinging, non-combusting spray was visualized using back-lit high speed photography. The pressure and temperature values set on the chamber correspond to SOI of 20, 30 and 40 CAD bTDC during engine testing at 2000 rpm and an IMEP of 2.5 bar for an overall lean operation of $\lambda = 4$. The spray visualization was also carried out at ambient conditions ($25°$C of temperature and 1 and 6 bar of back pressure). The results show that the penetration length is function of ambient temperature and pressure, fuel and injector type. The solenoid driven multi-hole injector produces longer penetration lengths and at a faster rate than the piezo unit under all test parameters. Moreover, there is greater variance in penetration curves for the fuels tested using the solenoid multi-hole injector compared to the piezo actuated outward opening injector. Despite wall-wetting aspects have not been tested, larger variance in penetration curves for different fuels in the solenoid multi-hole injector indicate that it is less suitable for bi-fuel engines. Thus, spray targeting differences will lead to potential increase in combustion stability (COVimep), increase in emissions and increase in wetting of surfaces i.e. sparks plug, bore or piston.

Introduction

Lean burn gasoline direct injection in combination with downsizing and turbocharging is regarded as one of the key solutions to reduce fuel consumption, hence reaching future CO_2 emission targets [1, 2]. This paper focuses on the application of the spray guided approached to stratified combustion, which has already proven to reduce CO_2 emissions and increase efficiencies even further [3, 4]. At stratified operation the fuel is injected late during the compression stroke. A centrally located fuel cloud is formed close to the spark plug permitting ignition and combustion at globally lean conditions ($\lambda = 3 - 6$). Stratified operation significantly reduces the heat losses and the possibility to operate the engine with wide open throttle minimizes the pumping losses. The main drawbacks are that the concept suffers from high cycle-to-cycle variations and stratification of the fuel leads to locally fuel rich areas which results in higher particulate emissions. The short timing between the fuel injection and the time of ignition makes the SGDI concept highly dependent on the fuel delivery strategy. Spray guided systems utilize either a piezo or solenoid driven outward-opening unit which is mounted centrally or tilted within the cylinder, alternatively a (more cost effective) inwardly-opening solenoid driven unit can be employed.

Outward-opening units distribute the fuel via an annulus giving a largely uniform, hollow cone spray. This uniformity insures an even and largely predictable spray structure. Alternatively, the inward-opening unit is configured with multiple holes in a variety of configurations; typically arranged axisymmetrically with 5-7 holes. The unit investigated is a six-hole horseshoe configuration where largely solid 'plumes' of fuel are expelled with an even distribution of fluid in each of the plumes.

The industry is ultimately heading toward stratified operation in the low-medium load operating regions. The choice of piezo or solenoid driven injectors still remains uncertain. Constant volume chambers are a proven tool

*Corresponding author: pedmar15@mot.upv.es

for studying fuel delivery and combustion without the cost and complexity of an optical engine [5]. This method provides a fundamental understanding of the fuel delivery, one which typically ignores impingement and flow-field effects, and is often used as a bridge to optical engine tests.

For late injection strategies under stratified SI conditions, the wetting of a surface is of concern for Soot/PM and HC emissions [6, 7]. The main wetted surface is the rising piston-crown which may lead to a diffusive (locally rich) combustion, giving rise to emissions. These pool fires are of particular concern at these late injection timings (SOI at 20 CAD bTDC) which would locate the piston 18.3 mm (for the engine baselined) below the injector tip with an incoming fuel spray with a substantial high momentum. By understanding the development of the axial and radial penetrations and overall fuel structure, conclusions can be drawn on the propensity of wetting. These aspects are of particular importance as the fuel injection pressure levels are continually increasing. Hence, the rational for using the current gasoline ceiling pressure of 200 bar. This increasing pressure trend is largely due to increase the need for better liquid breakup leading to better mixing, greater combustion propensity and reduced emissions [8, 9]. It has been shown for similar experiments albeit not at conditions corresponding to stratified operation that outward-opening piezo-driven injectors produce smaller droplets (SMD) [10].

Experimental Methods

Initial engine tests as outlined in [11] provided operational conditions. The engine was operated using an outward-opening piezo injector with a gasoline fuel delivery pressure of 200 bar. Operating the engine with fully-opened throttle at 2000 rpm and an IMEP of 2.5 bar resulted in a globally lean air-fuel ratio of $\lambda = 4$. Multiple SOI points of 20, 30 and 40 CAD bTDC were investigated. These parameters, alongside two ambient temperature conditions form the basis for the current investigation; Table 1 displays the measurement points for every of the tested fuels (Table 2).

Test Series	Engine-like	Ambient
SOI (deg bTDC)	N/A	N/A, 40, 30, 20
Back pressure (bar abs)	7.5, 11, 15	1, 6
Temperature (K)	476, 520, 559	298
Piston axial distance Z (mm)	N/A	N/A, 28.4, 22.7, 18.3
Injection angle (deg)	N/A	15

Table 1: Measurement parameters.

Injecting three fuels with two injectors, which have different transport mechanisms, in a quiescent chamber allows to correlate fuel properties, spray macroscopic parameters and the mechanism, all at once. The constant volume chamber provided an adequate means to reach the relatively high pressures and temperatures required for realistic environmental conditions. The flow rate of the air was kept at considerably low flow rates, several orders of magnitude lower than the spray velocity, to ensure no flow field influences.

Backlit photography was employed as the imaging technique to visualize the liquid fractions of the fuel sprays. Two Dedocool 250W lamps illuminated an opaque Perspex imaging plane with a monochrome Vision Research Phantom V7.1 high speed camera coupled to a 50 mm Nikkor lens capturing the spray. A 256x265 spatial resolution was chosen for an externally triggered frame rate of 24000 fps, which provided sufficient detail in the time domain. The experimental set-up is depicted in Figure 1.

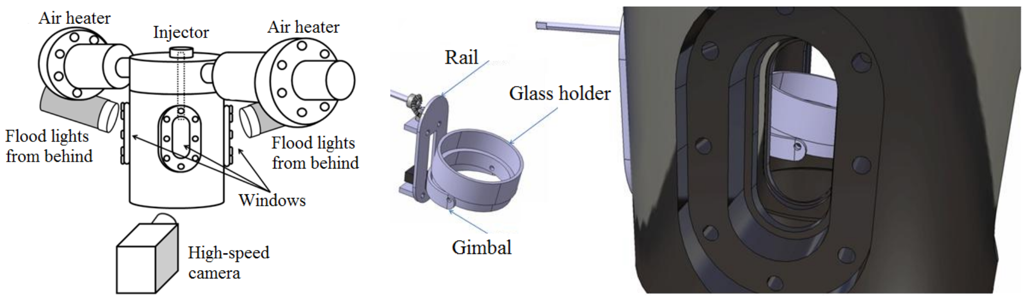


Figure 1: Experimental apparatus: complete arrangement (left), internal dummy piston configuration (middle) and dummy piston assembled in chamber (right).

The energy delivery value per injection event was calculated for the engine operational conditions and was used as the normalization parameter for both injector and fuel types (described below). Using an energy value rather than a constant delivery time allows to study the spray behaviour/shape when changing fuel. This approach gives a more representative outlook on fuel delivery phenomena in a quiescent environment relative to the experimental conditions carried out in [11]. Traditional mass delivery normalization could be used for further analysis of i.e momentum affects but is not included in the current work. The specifications of each of the fuels are given below in Table 2.

Fuel type	**Iso-Octane**	**Gasoline 95**	**e100 Ethanol**
Fuel brand	Fischer Scientific	Preem (Pump Grade)	SAAB Automotive
Density (kg/m^3) @ ambient	690	745	790
Viscosity (10^{-3} Pa·s) @ ambient	0.466	0.292	1.200
Surface tension (10^{-3} N/m)	18.18	21.60	22.80
Heat of Vaporization (kJ/kg)	300.2	350.0	840.0
Boiling point (K)	372.45	343.00	354.15
Stoichiometric AFR (:1)	15.09	14.70	8.98
Lower Heating Value (kJ/kg)	43.95	44.00	26.90
Research octane number	100	95	107

Table 2: Fuel specifications.

Results and Discussion

Determination of injection durations at constant calorific value

Determination of the injection pulse timing was performed using a mass balance approach. Each fuel for each injector was subjected to duration range testing. Range testing provided a fuel delivery characteristic curve where the benchmark delivery energy was extrapolated and matched to the target 389.4 J.

Tests were conducted where the injector was held above a conical flask which sat upon a mass-scale (balance), 1000 repeats were conducted for each fuel with 200 pre-injections performed to avoid any transient effects. The mass for each test was record every 100 delivery sequences. Statistical hypothesis testing with a confidence of 95% (alpha value of 0.05) between repeats were carried out for checking the repeatability. In all cases, no evidence of variation were found, so the average value of the repeats was taken as the correct value for each injector, fuel and injection duration. A graph of mass per injection versus injection duration is plotted in Figure 2. The relationship between these parameters must be linear, so a linear regression could be implemented. It can be seen that the outward-opening piezo driven injector has higher injected mass rate for the same injection duration, an expected result considering that the effective fuel exit area is higher than in the multi-hole solenoid driven injector. Also this result is in concordance with the results obtained by Smith, J. et al (2011) [12] who obtained 3% greater fuel consumption for the outward opening injector. The normalized injection signal durations can be viewed in Table 3.

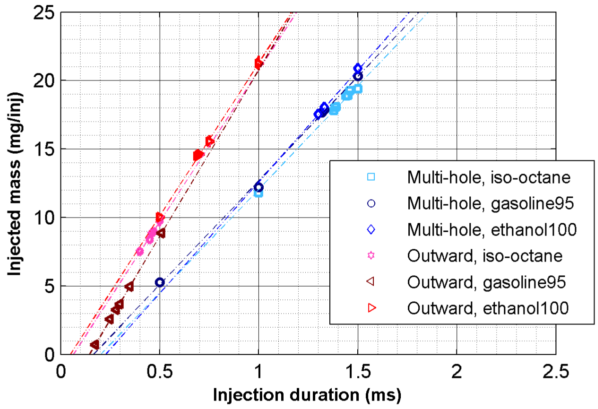


Figure 2: Mass per injection versus injection duration for the multi-hole injector and the outward-opening injector. Injection pressure of 200 bar under ambient conditions.

Injector type	Multi-hole			Outward opening		
Brand	Bosch			Bosch		
Actuation	Solenoid inward-opening			Piezo outward-opening		
Pintle style	6 hole horseshoe			Annulus		
Fuel pressure (bar abs)	200			200		
Fuel calorific value (J/inj)	389.4			389.4		
	Iso-Octane	Gasoline 95	e100 Ethanol	Iso-Octane	Gasoline 95	e100 Ethanol
Opening signal duration (ms)	0.788	0.748	1.115	0.463	0.510	0.696

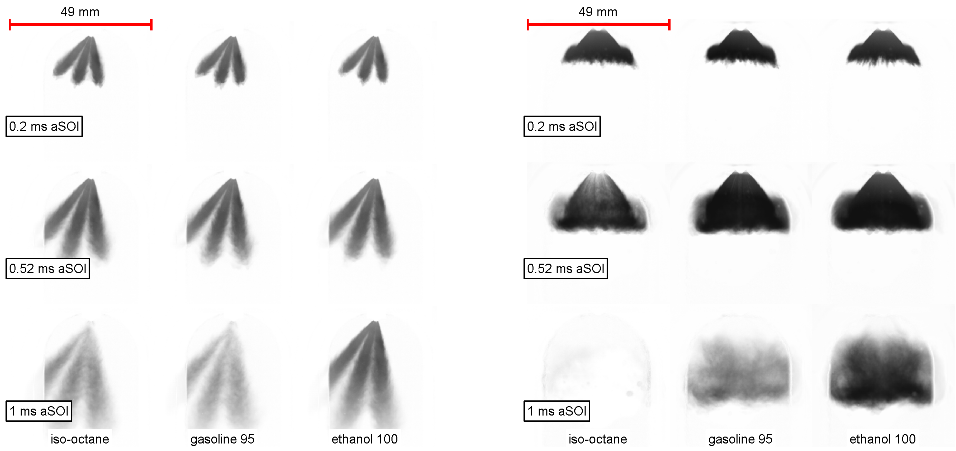
Table 3: Fuel delivery parameters for the multi-hole injector and the outward opening injector.

Figure 2 also shows that the slope of the curve is not the same for every fuel. This is mainly due to fuel viscosity and density (see Table 3). For short injection duration, the friction losses are important, so a lower mass flow rate is expected for fuels with higher viscosity. However, for long injection duration the transient effects of the friction losses could be neglected and the higher the density the higher the mass flow rate. This reasoning matches with the results obtained for the multi-hole injector, but not for the outward-opening injector. This is explained again with the higher fuel exit area, which allows neglecting the transient effects for all injection durations. This clearly points out that the effects of the fuel properties cannot be directly compared between the two injectors due to their inherently different delivery mechanism, however, the inter relationships of the fuel properties for each specific injector type are concluded.

Spray structure (width and vortex formation)

Backlighting technique allows us to see the evolution of the spray and a 2D slice of the macroscopic vortex structures that are generated by the outward opening injector. Processing of the images was performed using an in-house Matlab script, where each image was subjected to background removal and noise reduction via grayscale pixel thresh-holding and binary analysis [13]. All figures presented in this section are ensemble averaged images of five injection events, where all images were taken and processed under identical conditions.

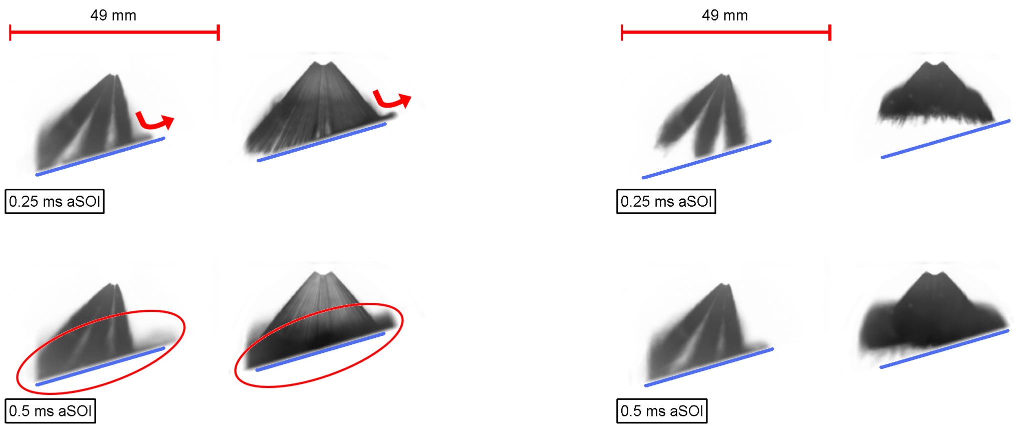
Figure 3 compares the spray structure for the three different fuels. There are no major differences, for example, inner and outer vortices of the outward opening injector are always formed and they have apparently the same size in all cases. It is observed that every fuel has a different distribution density and slightly different spray width and vertical size. This is explained by the different evaporating rate for every fuel which is higher for e100 and lower for Iso-Octane, as predicted by [14].



(a) Multi-hole injector. (b) Outward opening injector.

Figure 3: Averaged images showing the spatial distribution of the fuel with different working fluids at 7.5 bar of back pressure and 476 K of vessel temperature.

A dummy piston with a flat upper surface was built and placed inside the constant volume chamber with an angle of 15° with respect to the horizontal plane to reproduce the geometry of the engine tested by [11]. The distance between the piston and the nozzle tip was set to reproduce stratified operating conditions (see Table 1). Figure 4 displays how the fuel is distributed inside the vessel with the presence of the piston. In both injectors, the fuel impacts the piston and moves outwards along the surface generating a small traveling vortex. Although the multi-hole spray reaches the piston earlier, the density of the spray near the wall is greater for the outward opening injector (Figure 4a), meaning that this injector drives more fuel close to the piston. However, increasing the back pressure modifies the spray structure of the hollow cone reducing the amount of fuel surrounding the piston and leading to more uniform distribution of fuel inside the combustion chamber (Figure 4b). As shown also next in Figure 6a, a further increase of back pressure will reduce even more the spray penetration, allowing the piston to move closer to the nozzle, in other words, allowing a later injection strategy.



(a) 1 bar of back pressure.

(b) 6 bar of back pressure.

Figure 4: Averaged images showing the spatial distribution of the fuel (Gasoline 95) with two different distances between the nozzle tip and the dummy piston, for the multi-hole and outward opening injectors under ambient temperature.

Spray penetration and spray angle

Also spray penetration and spray angle can be calculated from the images obtained by the high speed camera. In this case, for allowing comparison of the two kinds of injectors, the spray penetration is defined as the maximum vertical distance of the spray measured from the nozzle tip, and the spray angle is the total angle between the two lateral edges of the spray [13] as depicted in Figure 5. Both injectors were aligned in the vessel with the fuel inlet at 90° of the viewing axis.

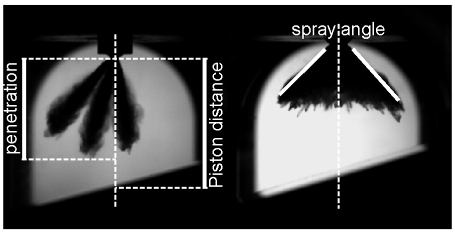
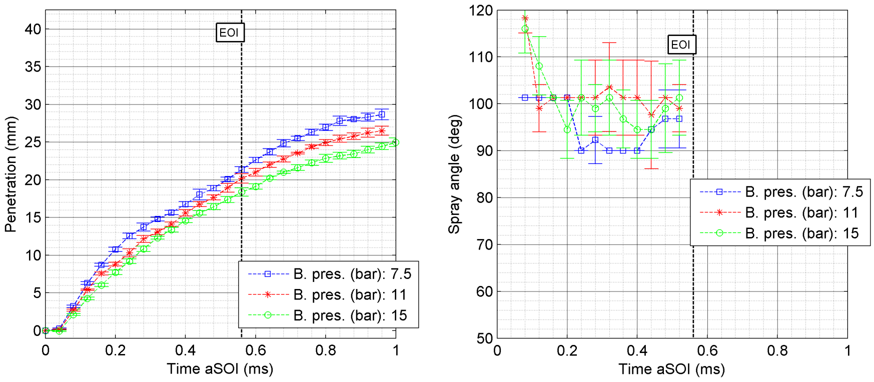


Figure 5: Raw images showing the definition of axial penetration, spray angle and piston distance.

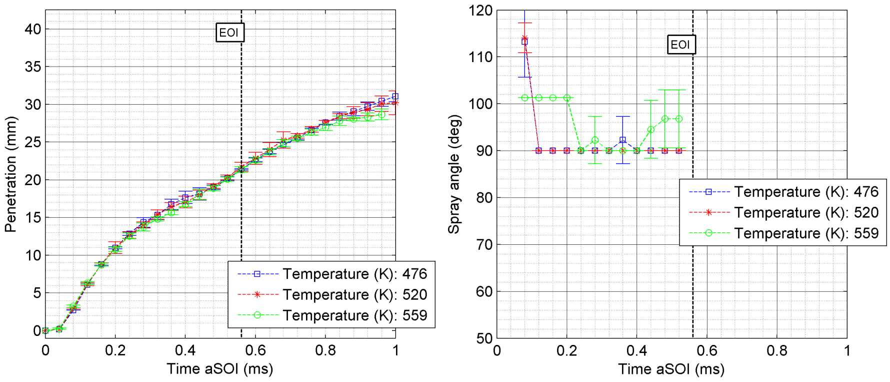
Figure 6a shows the effect of back pressure on the spray penetration. As expected, the greater the vessel pressure the smaller the penetration. Figure 7a shows the effect of the vessel temperature. Apparently this parameter has no significant effect on the spray penetration in the tested range. Although the figure only exhibits the penetration for Gasoline 95, this trend is also observed for the other two tested fuels.

Figures 6b and 7b show the spray angle. It has been calculated only until the EOI, where liquid is constantly injected. These figures show how the spray angle for the tested outward opening injector is around 110° shortly aSOI, and then it decreases to a constant value around 90°-100°. The oscillations observed in Figure 6b are caused by a high evaporating ratio. As it can be seen in Figure 7b, the angle remains constant for 476 and 520 K, however its value oscillates significantly in a range of 10° for 559 K of vessel temperature. This behavior is related with the heat of vaporization of the fuel, for Iso-Octane which has the lowest heat of vaporization the spray angle oscillates



(a) Spray vertical penetration. (b) Spray angle.

Figure 6: (a) Spray vertical penetration and (b) spray angle for the outward-opening injector with Gasoline 95 as working fluid, 559 K as vessel temperature and without piston. The vertical line represents the end of injection.



(a) Spray vertical penetration. (b) Spray angle.

Figure 7: (a) Spray vertical penetration and (b) spray angle for the outward-opening injector with Gasoline 95 as working fluid and 7.5 bar of back pressure. The vertical line represents the end of injection.

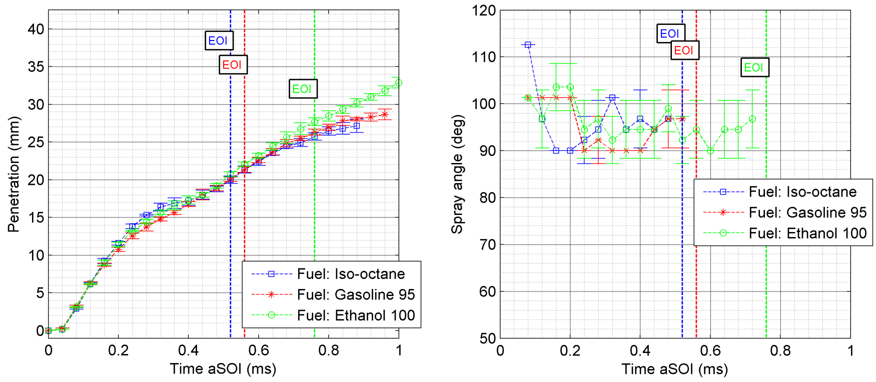
under all tested conditions, for Gasoline 95 large oscillations are obtained at 559 K, and for e100 Ethanol smaller oscillations are obtained only at 559 K.

Figure 8a shows the penetration for different fuels. The penetration values converge at 0.44 ms and diverge somewhat thereafter. This collapse and the change in the penetration slope are due to inner and outer vortices of the outward opening injector are fully developed and, as it has been commented before, they have approximately the same size for the three tested fuels. The variance in penetration distance between 0.16 and 0.44 ms is attributed to a change in momentum based on the individual fluidic properties, whilst it can be said that the divergence to EOI is attributed to the influence of evaporation rate. This is evidenced by the penetration value order of e100 Ethanol followed by Gasoline and Iso-Octane. Furthermore, aEOI the weight of the fuel appears to become important and the higher the density the higher the penetration.

It can be observed that in Figure 8b that no conclusive results on the variation in spray angle can be made, uncertainties are too large.

All figures show that the first point of the penetration curve for the outward opening injector has a value close to zero. This is simply because the origin of the penetration has been defined, as usually, at the nozzle tip, and the hollow-cone spray starts upstream from this point. Although other definitions are possible [8], for comparisson purposes this one is still the best.

If we now compare the penetration and the spray angle for both injectors, Figure 9a shows that the gradient of the penetration curve is steeper for the multi-hole injector, which has longer penetration. Regarding the spray angle, Figure 9b shows that the multi-hole injector has a cone angle around 25° lower. Increasing the back pressure

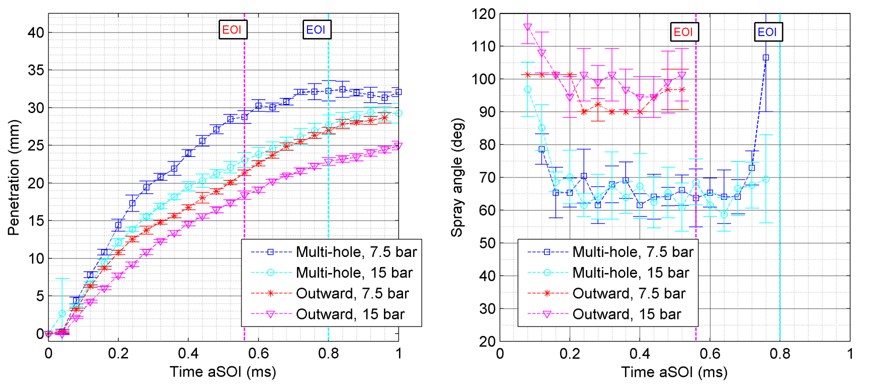


(a) Spray vertical penetration. (b) Spray angle.

Figure 8: (a) Spray vertical penetration and (b) spray angle for the outward-opening injector with 7.5 bar of back pressure, 559 K of vessel temperature and without piston. The vertical lines represent the end of injection; the first one for Iso-Octane, second one is for Gasoline 95 and third one for e100 Ethanol.

changes the structure of the hollow cone spray [10] leading to a vortex formation, and so, to a reduction of spray penetration (see Figure 4). It is shown that the spray angle of this kind of injector remains the same; however the spray angle of the multi-hole injector is affected by the back pressure and reduced around 10°, as depicted in Figure 9b. This collapsing phenomena at elevated temperatures is in good agreement with for example [14]. Furthermore, given the spark plug is located adjacently to the injector, increased variability in the spray angle (or radial penetration) will inevitably lead to spray targeting and ignition quality challenges.

The smaller cone angle and longer penetration of the solenoid driven multi-hole injector conclude that this kind of injector will lead to more wetting of the rising piston-crown on its compression stroke which will lead to a rise in HC emissions, and also makes it more prone to knock [12]



(a) Spray vertical penetration. (b) Spray angle.

Figure 9: (a) Spray vertical penetration and (b) spray angle under 7.5 and 15 bar of back pressure and 559 K of vessel temperature, and Gasoline 95 as working fluid. The vertical lines represent the end of injection; the first one for outward opening injector, and the second one for multi-hole injector.

Summary and Conclusions

A comparative study from an engine test point of view of two current possible injectors for SGDI engines has been carried out successfully. The presented results deliver quantitative and qualitative data on the characteristics of liquid fuel sprays, which play an important role in defining which injector drive, control and delivery strategy, is used in the real world for a variety of fuels. The understanding of the macroscopic changes in fuel structure under a quiescent and realistic environment allows for insight to aid the development of operational spray targeting techniques.

As expected, the piezo driven outward opening injector supplies more fuel than the solenoid driven multi-hole injector for the same injection period. This is of benefit, as when injecting a normalized (mass or calorific value) the injection duration time can be decreased allowing potentially later injection timing.

No significant differences have been found using one fuel or another at the tested conditions and macroscopic analysis. This is due to normalize the fuel delivery to energy value rather than duration as commonly found in literature. This also allowed the subjective analysis of the two injector types. Differences found in this analysis are due to the different transport mechanism of the injectors.

The spray angle variability, stronger for the multi-hole injector, under increased temperatures is a concern for spray targeting and ignition quality given the spark plug is located adjacently to the injector.

Despite risk of wall-wetting aspects having not been tested quantitatively, conclusions on wettability have been draw from the qualitative data, whereby the amount of fuel surrounding the piston with the multi-hole injector is greater under high pressure (real) conditions due to longer axial penetration and the vortex formation phenomena which limits the vertical penetration of sprays from outward opening injectors.

The larger variance in penetration curves for different fuels in the solenoid multi-hole injector indicate that it is less suitable for bi-fuel engines. Thus, spray targeting differences will lead to potential increase in combustion stability (COVimep) and increase in emissions.

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