

Experimental Analysis of the Impact of Injected Biofuels on In-Cylinder Flow Structures

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ABSTRACT

The interaction of biofuel sprays from an outward opening hollow cone injector and the flow field inside an internal combustion engine is analyzed by Mie-Scattering Imaging (MSI) and high-speed stereoscopic particle-image velocimetry (stereo-PIV). Two fuels (ethanol and methyl ethyl ketone (MEK)), four injection pressures (50, 100, 150, and 200 bar), three starting points of injection (60°, 277°, and 297° atdc), and two engine speeds (1,500 rpm and 2,000 rpm) define the parameter space of the experiments. The MSI measurements determine the vertical penetration length and the spray cone angle of the ethanol and MEK spray. Stereo-PIV is used to investigate the interaction of the flow field and the ethanol spray after the injection process for a start of injection at 60° atdc. These measurements are compared to stereo-PIV measurements without fuel injection performed in the same engine [19].

The results of the measurements of the vertical penetration length show the flow field to strongly influence the spray behavior. Especially for late injection during the compression phase, a strong interaction of the tumble vortex and the spray is visible, such that liner and piston wetting effects are reduced. Compared to ethanol, MEK shows lower piston and liner wetting effects and a reduced influence of the flow field on the injection behavior. The measurements of the early spray cone angle evidence a decreasing influence of the flow field with increasing spray Reynolds number.

The stereo-PIV measurements show a small change of the tumble trajectory through the fuel injection such that at late crank angles the tumble moves below the intake valves. Furthermore, a strong decrease in the spatially averaged out-of-plane vorticity shortly after injection occurred. The analysis of the kinetic and turbulent kinetic energy reveals lower values during the engine cycle with fuel injection. Moreover, at the end of the compression stroke the turbulent kinetic energy as well as the out-of-plane vorticity level is comparable with the measurements without injection.

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INTRODUCTION

The reduction of pollutant emissions and a decrease in green house gases is one of the main challenges in modern engine development. One approach is the combustion of sustainable biomass-based fuels that ensure a CO₂ neutral energy supply. The design of novel fuels from lignocellulosic biomass, a waste product in agriculture and timber industry [32], is the focus of the Cluster of Excellence (CoE) "Tailor-made Fuels from Biomass" (TMFB). It is important to note that those biofuels do not compete with the food chain. The new fuels have to be tested and special new engine concepts have to be designed to increase efficiency and reduce pollutant emissions. The interaction of the fuel spray and the internal flow field plays an important role in the optimization of the combustion process. As biofuels possess substantially different characteristics with respect to the boiling point, viscosity, or surface tension compared to classical gasoline a detailed analysis of the injection and mixing behavior of these biofuels is necessary. In this study, the influence of the dynamic viscosity, i.e., the impact of the Reynolds and Ohnesorge numbers, is investigated. For this, ethanol as a classical biofuel and Methyl Ethyl

Ketone (MEK) as a second fuel derivate with substantially lower dynamic viscosity are investigated in detail. MEK has been introduced as an excellent biofuel candidate by Hoppe et al. [17] due to its good cold start behavior and its high knock resistance as it possesses a high research octane number (RON). Furthermore, they showed that MEK shows an improved combustion stability, lower particle emissions and a lower auto-ignition tendency compared to classical RON95 gasoline.

The analysis of fuel sprays is well established in modern combustion science. A review on the needs, milestones, challenges, and an overview of common techniques, used for spray analysis if given by Fansler and Parrish [14]. One technique, recommend for the analysis of the liquid penetration length of fuel sprays is shadowgraphy, which has been used by Mathieu et al. [22]. They performed extensive investigations on the spray behavior in pressure chambers for different fuels by analyzing the liquid phase of the fuel spray of an outward opening (A-type) hollow cone fuel injector with shadowgraphy and laser correlation velocimetry. The latter was used

to determine the spray velocity. However, shadowgraphy visualization is not feasible for engines as the cylindrical shape of the liner causes strong optical aberrations. Another possibility for the visualization of liquid fuel spray is Mie-Scattering Imaging. The technique is easily implemented and well known in the spray research community (Bardi et al. [2]). Furthermore, the illumination of the spray through a transparent piston crown allows measurements in optically accessible research engines (Miles [23]).

Particle-image velocimetry (PIV) is a standard tool in the investigation of engine flow fields. Dannemann et al. [12] performed planar PIV measurements (2D/2C) for several crank angles and several planes inside the combustion chamber of an unfired four-valve IC engine at 1,500 rpm. They analyzed the three-dimensional structure of the flow field. It was possible to visualize large scale vortical structures, i.e., the tumble vortex and ring vortices underneath the intake valves, through a quasi three-dimensional reconstruction of the flow field. Furthermore, the highly three-dimensional character of the flow field could be verified.

First stereoscopic PIV (stereo-PIV) measurements, which were capable of resolving all three velocity components in one plane (2D/3C), were performed by Calendini et al. [10]. They recorded 35 double images in the center plane between intake and exhaust valve for different crank angles at 1,000 rpm. However, the ensemble averaged measurements were not able to visualize large scale flow structures. Bucker et al. [8] performed stereo-PIV measurements in a set of planes for several crank angles at 1,500 rpm. They visualized and analyzed the temporal development of the mean kinetic energy in two measurement planes. Furthermore, by analyzing 14 axial planes, a quasi three-dimensional propagation of the large scale vortical structures could be reconstructed.

Volumetric flow measurements are rarely reported in the literature. Van Overbrüggen et al. [24] performed holographic PIV to show the feasibility of this measurement technique for engine flows. They were able to resolve a large part of the flow field at high resolution for 1,500 rpm. However, due to the analog recording on the holographic plate it is difficult to record statistically significant samples. Tomographic PIV at IC engines was first introduced by Baum et al. [3]. They recorded the flow field at two crank angles at 800 rpm with a depth of field of 4 mm as well as 8 mm and showed the feasibility of Tomo-PIV for the three-dimensional measurement of in-cylinder flow fields. Van Overbrüggen et al. [25] recorded the flow field inside an IC engine at 1,500 rpm for three crank angles and resolved nearly the whole combustion chamber. The resulting ensemble averaged velocity fields showed the three-dimensional propagation of the large-scale vortical structures, i.e., the tumble and a vortex ring below the pent roof. Furthermore, the turbulent kinetic energy (TKE) was used for the analysis of the small scale vortical structures.

All those measurements provide a good understanding of the in-cylinder flow field without fuel injection. However, the direct interaction of fuel spray and flow field has to be investigated to understand the mixture formation. Marchi et al. [20] performed Mie scattering measurements of three hollow cone prototype injectors for different injection pressures, needle lifts, and engine speeds. A spray angle analysis showed that a hollow cone injector with a positive-step

inward seal band is the most robust design concerning spray angle, which is desirable for a stratified fuel/air mixture formation. Stiehl et al. [33] investigated a spray-guided direct injection engine with an A-type hollow cone injector operated in stratified charge mode through Mie-Scattering and time resolved PIV. They found the spray to influence the tumble movement. Furthermore, an influence of multi-injections on cycle-to-cycle variations could be revealed. This leads to an increased spray variability. A review on current work on combustion instabilities in spray-guided stratified-charge engines operated in partial load with multihole and A-hole injectors is given by Fansler et al. [15].

Zeng et al. [37] investigated the combustion variability of a stratified direct-injection spark ignited (DISI) engine with simulated exhaust gas recirculation (EGR) and a multihole injector by planar high-speed PIV, spray-, and flame imaging at an engine speed of 2,000 rpm. They showed that a weak gas flow near the fuel spray causes up to 30% of all EGR cycles to develop into partial burn. Further work of Zeng et al. [38] investigated the interaction of spray and intake generated swirl. The results show that the spray has a stabilizing effect on the flow field and combustion. It forces gas with a high angular momentum near the piston bowl rim to flow toward the center-line, which results in a much higher flow stability and reduced cyclic variations.

In this study, two biofuels, ethanol and methyl ethyl ketone (MEK), are investigated concerning liquid penetration length in an optically accessible motored research engine using Mie-Scattering Imaging (MSI) for four injection pressures (50, 100, 150, and 200 bar) and three starting times of injection (60°, 277°, and 297° atdc) corresponding to an in-cylinder pressure of 1, 2, and 3 bar at start of injection. Furthermore, to investigate the influence of a hollow cone fuel spray on the internal combustion engine flow field time-resolved stereo PIV measurements have been performed for ethanol injection to investigate one exemplary biofuel.

The article is organized as follows. First, the optical four-valve research engine is described in detail. Then, the Mie-Scattering Imaging and the stereoscopic PIV setup as well as the processing procedure are explained. The discussion of the results starts with an error estimate for both measurement techniques. Subsequently, the results of the liquid penetration length measurements are analyzed. Subsequently, the impact of the spray on the flow field is investigated by comparing the performed measurements to PIV findings from Karhoff [19] without injection.

EXPERIMENTAL SETUP

The four-valve research engine TINA is a motored model of a direct injection spark-ignition (DISI) engine made by the FEV Motorentechnik GmbH. It is equipped with a tumble port, a pent roof combustion chamber and a flat piston crown. The tumble ports ensure a tumble number of about 5 to 10 depending on the crank angle [7]. The high tumble number has been chosen to increase the combustion stability of tailor made fuels [19]. A flow coefficient of 0.11 has been measured by Adomeit et al. [1]. The bore of the engine is $D = 75$ mm and the stroke is $S = 82.5$ mm. This results in a cylinder displacement of $V = 364$ cm³. A quartz glass liner and quartz glass piston crown

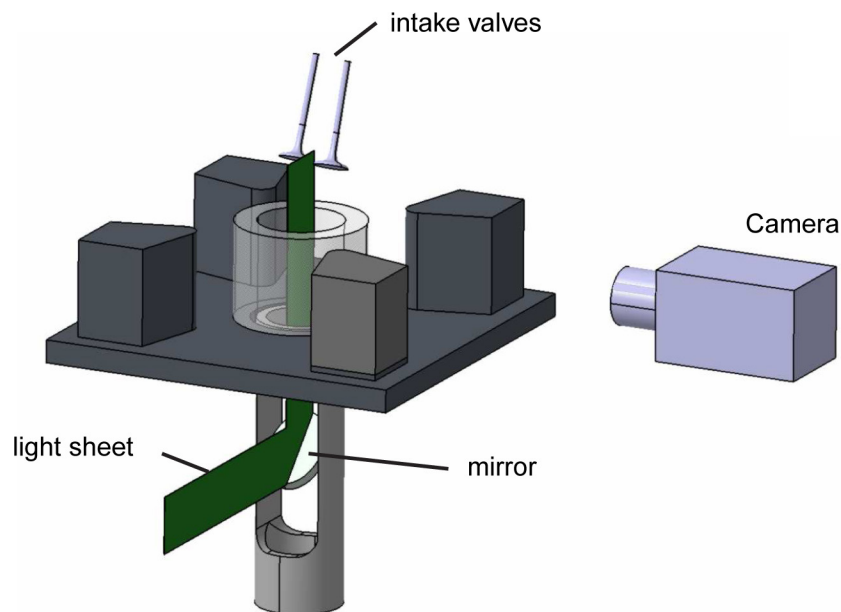


Figure 1. Mie-Scattering Imaging test rig.

enables optical access to the combustion chamber. Free piston movement within the optical liner is ensured by a clearance between piston and liner of 0.4 mm. The quartz glass piston crown is fixed onto the piston by a metal union nut with an inner diameter of 58.5 mm. For this reason, the illumination of the combustion chamber is restricted to its inner part. Moreover, Reuss et al. [30] report that the uncertainty in detecting particles in the outer part of a cylinder increase. They suggest to restrict the field of view to the inner 76% of the cylinder, which can still be achieved. The piston rings are located in the lower part of the engine, i.e., below the displacement volume in the iron liner section.

The optical engine is driven by a 30 kW electrical engine, i.e., the engine is not fired. The electrical engine allows to quickly accelerate and decelerate the test engine. This reduces the starting time and prevents unnecessary contamination of the optical parts of the engine. Furthermore, a mass balancing, which meets reciprocating forces of all orders, and a flywheel, connected to the crank shaft, limit the deviation in engine speed due to varying cylinder pressures during compression and power stroke to about 1.5% of the actual engine speed. The crank shaft is equipped with a shaft encoder with 3600 pulses per revolution. An additional pulse at top dead center combustion is measured to distinguish between top dead center gas exchange and top dead center combustion. The engine is equipped with an oil system to lubricate all essential engine parts. This system is also used to control the engine temperature. A custom made oil conditioning system ensures oil temperatures between 60 and 70 °C. A detailed description of the system can be found in Karhoff [19].

The intake pressure is monitored with a Kistler 4005BA5FA0 piezo-pressure transducer and the combustion chamber pressure with a Kistler 6045A piezo-pressure transducer. The main characteristics of the engine are listed in [table 1](#). The engine is equipped with an intake air conditioning system that enables the control of intake air pressure in the range of 0.7 - 1.5 bar. The intake pressure is measured

by the intake air pressure transducer. This signal is used to control air mass flow into the pressure reservoir via two magnetic valves through a PI control mechanism implemented in Labview.

Table 1. Parameters of the TINA research engine. TDC denotes the gas exchange top dead center.

Bore	75 mm
Stroke	82.5 mm
Displacement Volume	364 cm ³
Compression ratio	7.4:1
Number of valves	4
Valve lift	9 mm
Valve diameter intake/exhaust	27.1 mm / 23 mm
Exhaust valves open (1 mm lift)	206° before TDC
Exhaust valves close (1 mm lift)	20° before TDC
Intake valves open (1 mm lift)	10° after TDC
Intake valves close (1 mm lift)	196° after TDC
Connecting rod length	146 mm
Crank radius	42.2 mm

For the fuel injection, the engine is equipped with a piezo-electric hollow cone A-type fuel injector (Siemens VDO 104480-501). The half-cone angle of the injector is 47°, the maximum needle lift is 35 μ m. The injector is positioned in the symmetry plane between the intake valves with a distance to the liner axis of 6 mm. Furthermore, the injector is tilted with respect to the liner axis. A detailed description of the injector can be found in Mathieu [21]. The injector is driven by a VAPA piezo power amplifier (Vemac GmbH). The piezo power amplifier can be controlled by a TTL signal from the engine control software. The injection pressure is controlled by a custom made hydraulic injection system by the FEV Motorentechnik GmbH. The system consists of a pressure vessel that separates fuel and N₂ via a float lever. Due to safety reasons, the maximum injection pressure is restricted to 200 bar. The fuel is pressurized by an external pressure source, i.e., a N₂ pressure bottle.

Mie-Scattering Imaging Test Rig

Mie-Scattering Imaging (MSI) was used to investigate the interaction of liquid fuel spray and the flow field of the TINA engine. Following the fuel spray analyses of Bardi et al. [2], Marchi et al. [20], and the review of Fansler and Parrish [14] we denote this technique Mie scattering although the droplet sizes are not in the wave length range. The setup can be seen in figure 1. Laser light from a Darwin Duo high-speed laser is used to illuminate the liquid phase of the fuel spray with a frequency of 12,000 Hz. The laser light is expanded using a commercial light sheet optic. The light sheet is redirected by an elliptical 45° mirror and illuminates the fuel spray through the quartz glass piston crown. It is positioned in the x-z symmetry plane between the intake, and exhaust valves commonly referred to as the tumble plane. It covers the complete stroke and about 76% of the bore. Reflections at the liner wall are avoided. The intake air pressure has been controlled with the intake air control system to be at 1.05 bar for all measurements. The engine temperature was set to 60 °C with the oil conditioning system. A Photron Fastcam SA5 and a Tamron 180 mm macro lens with f/8 were used to record the fuel spray with a temporal resolution of 12,000 Hz and a spatial resolution of $[x, z] = 896 \times 704 \text{ px}^2$. To resolve the whole injection process, the image recording was started at start of injection. Then, 40 images were recorded to investigate the whole injection process. To achieve statistically converged data, 100 cycles were recorded for every measurement point. The data were processed using an in-house Matlab code. After recording, a background subtraction was used to reduce noise through reflections at the cylinder head and the cylinder walls. The grayscale images have been binarized using a fixed intensity threshold of 11% of the maximum intensity. Although a threshold of 3% [2, 27] or 13% [26] is suggested in the literature, we found this value of 11% to be the most stable for spray boundary detection. Subsequently, a boundary detection algorithm, already implemented in Matlab, was used to detect the boundary of the binarized spray. The boundary contour allows the detection of the penetration length.

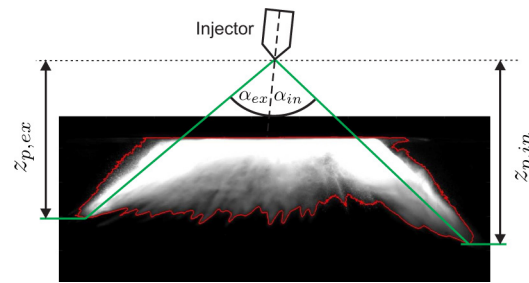


Figure 2. Mie-Scattering Image of the liquid phase. The image shows the extracted contour (red) and the penetration length for intake and exhaust side.

For this study, the 2D sheet of the hollow cone spray has been divided into two parts. The first half is underneath the intake valves whereas the second part is located underneath the exhaust valves. Figure 2 shows the liquid phase and the detected spray boundary as well as the extracted penetration length and half-cone angle for the intake and exhaust side for one exemplary time step. The penetration length $z_{p,i}$ is defined as the maximum distance between the injector and the fuel boundary on both sides. The small filamentary structures that appear in the center along the leading edge of the spray were not considered since they are caused by secondary scattering of the laser light at parts of the spray that are outside of the measurement plane. The maximum half-cone angle is defined by

$$\alpha_i = \arctan \left(\frac{x_{p,i}}{z_{p,i}} \right) \quad (1)$$

with $x_{p,i}$ being the horizontal penetration length corresponding to the vertical penetration length $z_{p,i}$.

The measurements were conducted for ethanol and methyl ethyl ketone (MEK). The injection pressures were set to 50, 100, 150, and 200 bar. For both fuels, three starting of injection (SOI) were investigated (60°, 277°, and 297° after top dead center (atdc)), which correspond to an in-cylinder pressure of 1, 2, and 3 bar at the start of injection. For ethanol, the injection duration was modified such that injection durations of 500 μs , 1000 μs , and 2000 μs were measured. However, at an injection duration of 2000 μs , the highest injection pressure was removed as the high fuel mass caused strong liner

Table 2. Measurement parameters.

Method	Stereo-PIV	Mie-Scattering Imaging
Position	y = 0 mm plane	y = 0 mm plane
RPM	1500, 2000 rpm	1500, 2000 rpm
Start of Injection	60, 277, 297° atdc	60, 277, 297° atdc
Injection Pressure	50, 100, 150, 200 bar	50, 100, 150, 200 bar
Injection Duration	500, 1000 μs	500, 1000, 2000 μs
Fuel	Ethanol	Ethanol, methyl ethyl ketone (MEK)
Crank angles	100°, 104°, 108°, ..., 320° atdc	60°, 61°, 62°, ..., 337° atdc

Table 3. Reynolds and Ohnesorge number for isooctane, ethanol, and MEK for a fuel temperature of 60°C before injection for different injection pressures and a cylinder pressure of 1 bar.

Fuel	Re / Oh (10^{-2})			
	50 bar	100 bar	150 bar	200 bar
Isooctane	9,691 / 1.1219	14,156 / 1.1167	17,671 / 1.1092	20,706 / 1.1021
Ethanol	7,235 / 1.6657	10,707 / 1.6416	13,397 / 1.6277	15,717 / 1.6208
MEK	13,767 / 0.8969	20,016 / 0.8958	24,4857 / 0.8941	28,8998 / 0.8920

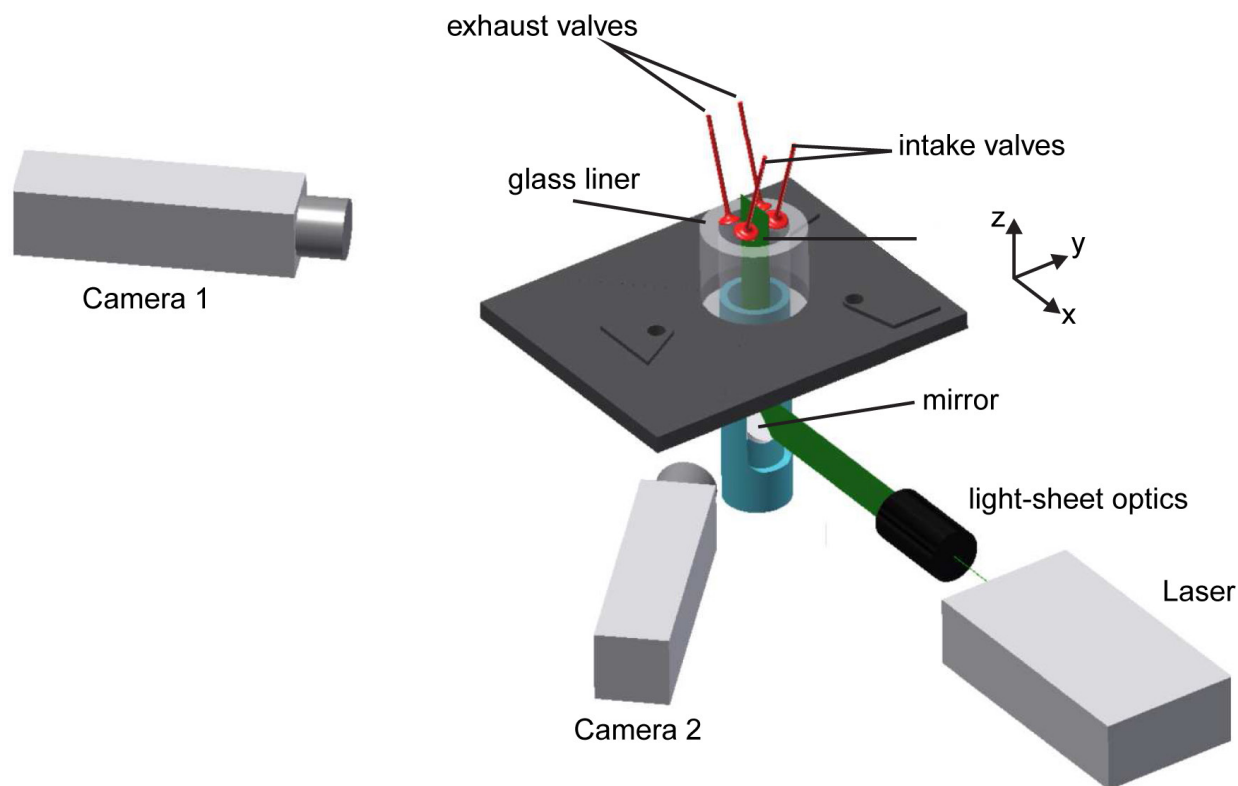


Figure 3. Stereo-PIV test rig.

wetting and therefore problems capturing the injection process occurred. For MEK, only an injection duration of 500 μ s was investigated.

An overview of the performed measurement parameters is given in table 2. Furthermore, table 3 shows the Reynolds and Ohnesorge number for both fuels at the different injection pressures for a cylinder pressure of 1 bar. Additionally, the Reynolds and Ohnesorge numbers for an isooctane spray at the same conditions has been added for reference. The Reynolds number is defined by

$$Re = \frac{\rho_l v_{sp} d}{\eta} \quad (2)$$

where v_{sp} is the nozzle exit velocity, ρ_l the density of the fuel, d the nozzle diameter, and η the dynamic viscosity of the fuel. The Ohnesorge number reads

$$Oh = \frac{\eta}{\sqrt{\rho_l \sigma d}} \quad (3)$$

The quantity σ defines the surface tension of the fuel.

Table 4 gives an overview of the fuel properties of ethanol, MEK, and isooctane at standard conditions. Table 5 shows the mass of injected fuel for all measured parameters. It is evident the injected mass increases for both fuels with increasing injection pressure and injection duration. However, only small differences between ethanol and MEK concerning the injected fuel mass are visible.

Table 4. Physical properties of isooctane, ethanol, and MEK at 298 K and an ambient pressure of 1 bar. All values except research octane number from Daubert and Danner [13].

	unit	isooctane	ethanol	MEK
empirical formula	-	C ₈ H ₁₈	C ₂ H ₆ O	C ₄ H ₈ O
oxygen content	mass-%	<0.5	34.7	22.2
density	Kg/m ³	699	787	799
dynamic viscosity	mPas	0.467	1.073	0.396
surface tension	mN/m	18.3	22.1	23.9
boiling temperature	K	372.15	351.15	353.15
vapor pressure	kPa	5.3	5.8	10.8
lower heating value	MJ/Kg	44.34	26.84	31.45
octane number	-	100	109	117

Table 5. Mass of injected fuel.

	p	0.5 ms	1 ms	2 ms
Ethanol	50 bar	7.7 mg	15.7 mg	32.7 mg
	100 bar	10.2 mg	20.7 mg	42.5 mg
	150 bar	10.6 mg	21.8 mg	43.8 mg
	200 bar	10.9 mg	22.7 mg	44.5 mg
MEK	50 bar	8.3 mg		
	100 bar	10.3 mg		
	150 bar	10.6 mg		
	200 bar	10.9 mg		

Stereoscopic Particle-Image Velocimetry System

Stereo-PIV measurements were performed to investigate the interaction of fuel injection and in-cylinder flow field in higher detail. However, as reflexions from fuel droplets at higher injection durations heavily disturbed the PIV measurements, only measurements at an injection duration of 500 μ s were conducted. Since an investigation after fuel injection for injections with a late start of injection (SOI = 277° atdc and SOI = 297° atdc) is not possible, the focus is on the interaction of fuel and flow field for early injection at a start of injection of 60° atdc.

A schematic of the experimental setup for the stereo-PIV measurements is shown in figure 3. The setup consists of a Darwin Duo dual cavity high speed laser, which is pulsed at a frequency of 3000 Hz per cavity and two Photron Fastcam SA5 high-speed cameras with Tamron 180 mm macro lenses. The light sheet is produced using a commercial light sheet optics and its position matches the position of the MSI measurements. The cameras are positioned in an ideal stereoscopic setup [28] with a symmetric angle of 45° with respect to the normal of the light sheet plane. The cameras and lenses were positioned in Scheimpflug condition [29] with custom made Scheimpflug adapters. Reflexions at the quartz glass liner are avoided by introducing the light sheet through the optical piston crown via an elliptical 45° mirror in the elongated slotted piston. The light sheet thickness is about 1 mm. It covers the complete stroke and about 76% of the bore.

The flow field is seeded with Di-Ethyl-Hexyl-Sebacat (DEHS) at a nominal diameter of 0.5 - 1 μ m. The seeding is introduced by a seeding generator upstream of the intake valves. This ensures a homogeneous distribution of the seeding particles inside the in-cylinder flow field. The intake pressure was controlled by the intake air control system to be at 1.05 bar for all measurements. The intake temperature corresponds to the ambient temperature, which was about 25 °C. The engine temperature was set to 60°C with the oil conditioning system. To calculate ensemble averaged velocity fields, 100 cycles were recorded. For early injection, the measurements start with start of injection. However, as strong reflections from the liquid fuel restrict the PIV evaluation, the analyzed crank angle regime is between 100° - 320° atdc with a resolution of 4° crank angle (CA).

After recording, a background subtraction of a minimum image, calculated out of all images, was used to increase the image quality. Furthermore, a masking algorithm was applied to mask out reflexions at the piston and the pent roof.

The software PivView was used for the flow field evaluation. Before calculating the vector fields, a calibration was performed to map pixel coordinates of both cameras to real coordinates. The calibration is based on a 2D target placed inside the cylinder. The target has equally spaced dots with a dot spacing of 5 mm and a dot diameter of 2 mm. An advanced Tsai-model based on a mixture of calibration approaches by Soloff et al. [31] and Tsai [34] is used to calculate the mapping function. Furthermore, the calibration had to be improved as imaging through transparent cylinders leads to non-linear radial distortions [30]. Additionally, errors through small misalignment between laser light sheet and calibration plane, so-called registration errors [36], have to be removed. Therefore, disparity maps out of the recorded particle images were calculated using the approach explained in Raffel [29]. The technique uses the calibration between images from camera 1 and camera 2 at the same time step to calculate the disparity of the views with respect to each other. As both cameras are stationary, ensemble correlation is used to create statistically more reliable disparity maps. Then, the map is used to correct the calibration factors accordingly. After calibration, a multi-grid interrogation algorithm with a starting window size of 96 x 96 px² and a final window size of 24 x 24 px² with 50% overlap and sub-pixel accuracy was used to calculate the velocity vectors, which leads to a vector spacing of 1 mm. Between each correlation step, five validation passes with a normalized median test [35] were used to determine and filter spurious vectors. The vectors were either replaced by lower order peaks from the cross correlation or interpolated into the velocity field. This leads to a vector field with over 93% valid vectors in the last pass.

PIV Post-Processing

The ensemble-averaged velocity, the turbulent kinetic energy (TKE), the Γ_1 vortex identification function [16], and the absolute averaged vorticity Ω were determined to analyze the velocity fields. The ensemble-averaged mean velocity component $\bar{u}$ in the x-direction is defined by

$$\bar{u} = \frac{1}{n} \sum_{i=1}^n u_i \quad , \quad (4)$$

where u represents the velocity and n the number of vector fields for the defined crank angle. The averaged variance $\overline{u'^2}$ of the velocity component in x-direction is defined by

$$\overline{u'^2} = \sigma_u^2 = \frac{1}{n} \sum_{i=1}^n u_i'^2 = \frac{1}{n} \sum_{i=1}^n (u_i - \bar{u})^2 \quad , \quad (5)$$

with the velocity fluctuation u' and the standard deviation σ_u of u . Note that the overall velocity fluctuations result from cyclic variations and the turbulent fluctuations. The turbulent kinetic energy (TKE) is computed by

$$k(\theta) = \frac{1}{2}(\overline{u'^2} + \overline{v'^2} + \overline{w'^2}) \quad (6)$$

Furthermore, the ensemble and plane-averaged turbulent kinetic energy

$$\overline{k(\theta)} = \frac{1}{lm} \sum_{j=1}^l \sum_{k=1}^m k(\theta)_{j,k} \quad (7)$$

was calculated to gain more information on the propagation of the small scale turbulent fluctuations during intake and compression stroke. The quantities l and m are the number of grid points in the x - and z -direction for one measurement plane. To visualize the tumble vortex and track its propagation, the Γ_1 vortex identification function from Graftieaux et al. [16] was used. The spatially and ensemble averaged mean kinetic energy KE in the measurement plane is defined by

$$KE = \frac{1}{2lm} \sum_{j=1}^l \sum_{k=1}^m (\overline{u_{i,j}^2} + \overline{v_{i,j}^2} + \overline{w_{i,j}^2}) \quad (8)$$

The dimensionless scalar function Γ_1 that has been calculated for every grid point is defined by

$$\Gamma_1 = \frac{1}{S} \sum_{i=1}^S \frac{\vec{r} \times \vec{v}}{||\vec{r}|| ||\vec{v}||} \quad (9)$$

Here, S represents the number of grid points to calculate Γ_1 , $\vec{r}$ the radius vector, and $\vec{v}$ the velocity vector at point i . In this work, $S = 3 \times 3$ grid points are used to calculate Γ_1 as explained by Dannemann et al. [12]. Finally, the absolute value of the normalized, averaged vorticity

$$\Omega(\theta) = \frac{1}{lm\omega_e} \sum_{j=1}^l \sum_{k=1}^m |\overline{\omega_{i,j}}| \quad (10)$$

has been calculated to gain information on the strength of the tumble vortex. The quantity ω_e is the crank shaft rotating frequency and ω_i the out-of-plane component of the vorticity vector that is defined by

$$\overline{\omega_{x,z}(\theta)} = \sum_{i=1}^n \left(\frac{\partial u_i(x,z)}{\partial z} - \frac{\partial w_i(x,z)}{\partial x} \right) \quad (11)$$

RESULTS AND DISCUSSION

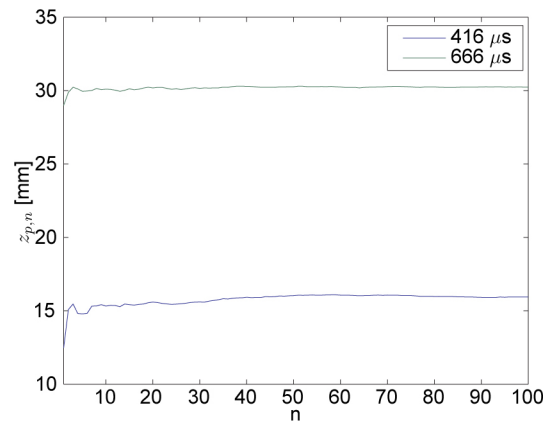
In the following, the results from the MSI and stereoscopic PIV measurements are discussed in detail. In the first subsection, an estimate of the measurement errors is given to assess the reliability of the measurement results. Subsequently, the results of the MSI measurements for the ethanol injection are thoroughly discussed, followed by an analysis of the MSI measurements of the MEK injection. Then, the results of the high-speed stereoscopic PIV measurements are compared to data by Karhoff [19] without fuel injection in the same engine concerning ensemble averaged flow fields, tumble motion, mean vorticity and turbulent kinetic energy. Finally, the influence of higher engine speeds on the mixture formation is presented. Mind that an in depth discussion of the flow field in this engine without fuel injection at an engine speed of 1,500 rpm is given by Bucker et al. [8] and Karhoff [19].

MSI Error Estimate

A convergence check [8] has been used to control whether enough MSI samples have been recorded. Figure 4 shows the running mean penetration length on the intake side at $t = 416 \mu s$ and $t = 666 \mu s$ after start of injection and the convergence

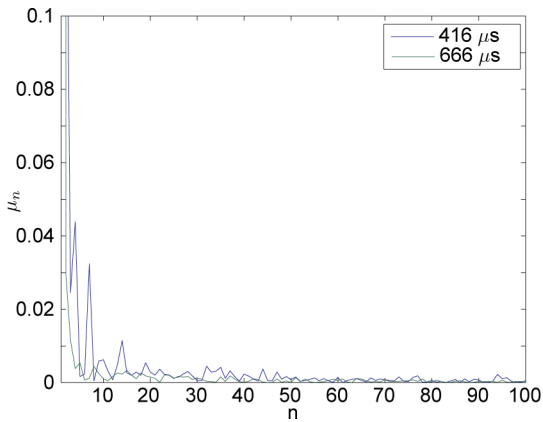
$$\mu_n = \frac{||z_{p,in,n} - z_{p,in,n-1}||}{||z_{p,in,n}||} \quad (12)$$

with $z_{p,in}$ being the penetration length on the intake side. One can see that after the first 20 samples the convergence reaches values below 1%. This shows that the number of recorded cycles is sufficient to determine the mean penetration length. Furthermore, using a sample size of $N = 100$ the absolute error within a confidence interval of 95% of the penetration length has been calculated. The uncertainty of the sample mean Δz_p is given by (Bendat and Piersol [4] and Benedict and Gould [5])



a. Running mean of the vertical penetration length.

Figure 4. Ensemble-averaged penetration length on the intake side for two time steps (a) and convergence μ_n for the two time steps at an increasing number of samples (b).



b. Convergence of the vertical penetration length.

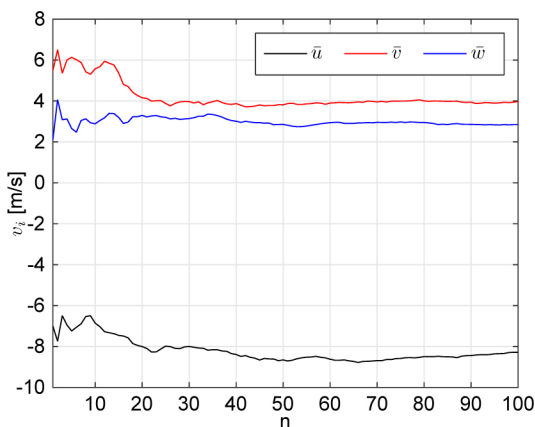
Figure 4. (cont.) Ensemble-averaged penetration length on the intake side for two time steps (a) and convergence μ_n for the two time steps at an increasing number of samples (b).

$$\delta(\bar{z}_p) = \frac{1}{N} \bar{z}_p'^2 \quad (13)$$

The theoretical error for a confidence interval of 95% reads

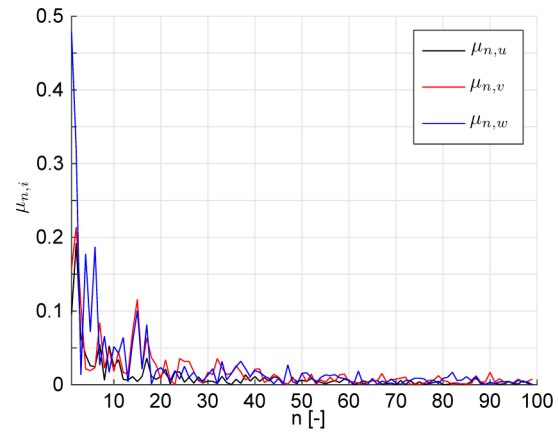
$$\Delta(\bar{z}_p)_{95\%} = 1.96 \sqrt{\delta(\bar{z}_p)} \quad (14)$$

The results show the maximum error for 95% confidence interval for early injection, i.e., for the first two frames to be lower than 2.5% of the maximum penetration length. For more frames, the error drops below 1%. The theoretical error of the injection angle is below 1.25° with a maximum theoretical error for the 95% confidence interval of 1.23 for the MEK injection at 277° atdc, which confirms the excellent quality of the measurement data.



a. Running mean of the velocity components.

Figure 5.



b. Convergence of the velocity components.

Figure 5. (cont.) Ensemble-averaged velocity components $\bar{u}$, $\bar{v}$, and $\bar{w}$ for 180° in the center of the tumble plane near the piston (a) and convergence μ_u , μ_v , and μ_w of the velocity components u, v, and w (b).

Stereo-PIV Error Estimation

The quality of the stereo-PIV measurements was analyzed by a convergence check, the results of which are illustrated in [figure 5](#). [Figure 5\(a\)](#) shows the running mean of all three velocity components at $[x,y,z] = [0,0,10]$ and [figure 5\(b\)](#) the convergence of all three velocity components. It is evident that after about 50 measurements the convergence reaches values below 2% for all measured velocity components. Although the convergence drops even further at higher measurement number, a sample size of $N = 100$ was considered sufficient to calculate a converged ensemble-averaged flow field. Furthermore, using a sample size of $N = 100$, the relative maximum error for all three velocity components for a confidence interval of 95% was calculated. The results show the maximum theoretical error of the mean velocity to be lower than 6%. The theoretical error within the same confidence interval for the variance of the velocity was determined. The theoretical error of the sample variance of the velocity component u is given by (Benedict and Gould [5] and Cho and Cho [11])

$$\delta(\bar{u'^2}) = \frac{1}{N} \left(\bar{u'^4} - (\bar{u'^2})^2 \right) \quad (15)$$

With this, the theoretical error of the variance of u for a confidence interval of 95% reads

$$\Delta(\bar{u'^2})_{95\%} = 1.96 \sqrt{\delta(\bar{u'^2})} \quad (16)$$

The maximum theoretical error of the variance of the velocity is about 10%.

Results of the Injection of Ethanol

The penetration length of ethanol as a function of the crank angle is shown in [figs. 6 - 8](#). For early injection at 60° atdc, it is evident that the vertical penetration length increases with larger injection pressure, which is caused by the higher exit velocities at the nozzle tip. Furthermore, the influence of increasing engine speed can clearly be seen by comparing [figs. 6\(a\) - 6\(b\)](#) and [figs. 6\(c\) - 6\(d\)](#). On the intake side, the spray reaches the piston at earlier crank angles. At 1,500 rpm, the engine speed as well as the general flow velocity is lower. Therefore, the influence of the tumble movement on the spray is reduced. For the exhaust side, two distinct changes in the slope can be clearly seen for the injection at 1,500 rpm. The first change at about 65° atdc is caused by the interaction of the spray with the liner wall. The second variation at about 70° atdc could be caused by the interaction of the spray and the tumble. Karhoff [19] showed for the same engine and engine speed that the tumble forms between 60° and 80° atdc. The tumble movement accelerates the atomized spray. This can not be seen for the measurements at 2,000 rpm. The higher engine speed causes the tumble to form at earlier crank angles. This, and a higher piston and flow velocity results in a smoother propagation of the spray as it is injected into an already formed tumble.

Comparing the injection behavior at 2,000 rpm ([figure 6\(c\) - 6\(h\)](#)) for different injection durations no pronounced differences can be observed. Obviously, the longer injection duration causes the combustion chamber to be filled with more fuel. However, the maximum vertical penetration length is restricted by the piston movement. Therefore, it is comparable for all injection durations. A distinct change of the slope is evident for all measurement at about 67° atdc, which is caused by the impingement of the spray on the liner wall.

[Figure 7](#) shows the vertical penetration length for a SOI at 277° atdc. It is obvious that high differences occur for the spray on the intake and exhaust side. On the exhaust side, the spray behavior is comparable to the injection at 60° atdc. For injection pressures of 100 - 200 bar the spray exits the nozzle with high velocities, until a change of the slope at about 284° atdc indicates the interaction of spray and liner wall. Then, the decelerated spray is directed to the piston. The spray wets the piston at about 300° atdc. At injection pressures of about 50 bar, the interaction of the liner and the spray can be observed for later crank angles of about 287° atdc. Subsequently, the spray is redirected until it wets the piston at about 303° atdc. With increasing injection duration, no real change in penetration length behavior is evident. The general slope of all curves is nearly the same.

On the intake side, the spray is not comparable to the injection at 60° atdc. After a fast first injection process until about 285° atdc, the spray is redirected to the pent roof. This is due to the tumble movement. The momentum of the in-cylinder flow field causes the fuel to be redirected. Therefore, neither liner nor piston wetting occur

on the intake side as the spray droplets are trapped inside the tumble vortex. Only for an injection pressure of 200 bar, the spray interacts with the piston at late crank angles. To highlight this, [figure 9](#) shows the raw Mie scattering images for one exemplary cycle for an injection pressure of $p = 200$ bar and an injection duration of 500 μs . It can be seen that the tumble causes a redirection of the fuel spray on the intake side and the formation of a vortical structure below the intake valves. For higher injection durations, the spray is also redirected through the tumble movement. However, the oscillating amplitude of the vertical penetration length decreases, compared to the measurements with an injection duration of 500 μs .

[Figure 8](#) shows the vertical penetration length for a SOI at 297° atdc. Again, the exhaust side shows a progressing behavior of the spray penetration length. However, the change in the slope is not as distinct as for a SOI at 60° atdc and 277° atdc. The increased pressure inside the combustion chamber as well as the restricted area combined with the tumble movement cause the spray to not interact with the liner wall. Instead, it wets the piston about 15° after SOI for all cases. Again, an increase in the injection duration does not change the penetration length. The slopes on the exhaust side are comparable for all injection durations.

On the intake side, again the interaction of spray and tumble is evident. The redirection of the spray to the pent roof through the tumble vortex can clearly be seen. However, due to the restricted area at later crank angles, the amplitude of the oscillating penetration length is not as distinct as at 277° atdc. Nevertheless, the interaction of spray and tumble movement causes the spray to wet the piston at late crank angles of about 325° for an injection duration of 500 μs . With increasing injection duration, the piston wetting occurs at earlier crank angles. This is due to the increased fuel mass inside the combustion chamber and therefore the higher influence on the momentum of the tumble vortex. This trend can also be observed for an injection duration of 2000 μs .

Results of the Injection of MEK

[Figure 10](#) shows the penetration length for MEK at 60° atdc, 277° atdc, and 297° atdc for the intake and exhaust side at an engine speed of 2,000 rpm. At a SOI of 60° atdc, the vertical penetration increases with increasing injection pressure. This is caused by the higher exit velocities at the nozzle tip as already explained for the ethanol case. On the exhaust side, the spray shows a slower penetration velocity compared to the intake side. Furthermore, the spray seems to accelerate at about 65° atdc. This could be caused by the interaction of tumble and spray and can also be partially observed for the ethanol injection. The injection with 50 bar shows a slow propagation of the vertical penetration length with late interaction of spray and piston. A change in the slope to lower penetration velocities can be observed at about 69° atdc. This suggests an interaction of liner and spray. On the intake side, the spray impinges upon the piston at about 67° atdc. Note that for ethanol piston wetting was reported at about 75° atdc. No direct change in the slope can be observed, which is a sign for missing liner wetting, whereas liner wetting occurred for all ethanol injection cases.