

Transient Liquid Penetration of Early-Injection Diesel Sprays

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ABSTRACT

Diesel low-temperature combustion strategies often rely on early injection timing to allow sufficient fuel-ambient mixing to avoid NO_x and soot-forming combustion. However, these early injection timings permit the spray to penetrate into a low ambient temperature and density environment where vaporization is poor and liquid impingement upon the cylinder liner and piston bowl are more likely to occur. The objective of this study is to measure the transient liquid and vapor penetration at early-injection conditions. High-speed Mie-scatter and shadowgraph imaging are employed in an optically accessible chamber with a free path of 100 mm prior to wall impingement and using a single-spray injector. The ambient temperature and density within the chamber are well-controlled (uniform) and selected to simulate in-cylinder conditions when injection occurs at -40 crank-angle degrees (CAD) or fewer before top-dead center (TDC). Injector parameters such as injection pressure, injection duration, nozzle orifice size, multiple injections, and fuel distillation properties are also varied.

Results show that an injection duration less than one-half of the development time for a steady liquid length will produce liquid penetration distances that are less than that of a steady-state spray. Using multiple, short injections also limits the liquid penetration while permitting the vapor-phase to continue to penetrate into the chamber. Small nozzle orifice diameters are helpful in limiting liquid penetration if sprays reach a steady-state liquid length. However, if the total injected mass is kept constant, requiring a longer injection duration with a small orifice diameter, the maximum liquid penetration is the same as that for a larger orifice diameter (provided injection durations are short enough such that liquid does not reach the steady-state liquid length). Similar findings are also obtained with injection pressure

variation and the same injected mass: short injection durations at high injection pressure produce the same maximum liquid penetration as longer injection durations at low injection pressure. Fuel distillation property variation shows an expected benefit of shorter liquid penetration for low-boiling-point-temperature fuels.

INTRODUCTION

Low-temperature diesel combustion strategies have shown promise in reducing nitrogen oxide (NO_x) and particulate matter (PM) emissions. However, these combustion modes have also been shown to produce higher unburned hydrocarbon (UHC) and carbon monoxide (CO) emissions [1]. As a result, the combustion efficiency may decrease for low-temperature combustion (LTC) compared to traditional diesel combustion.

Researchers are working to understand the source of this inefficiency, hoping to minimize UHC and CO emissions while maintaining low PM and NO_x emissions. One potential source for UHC emissions is liquid fuel that wets in-cylinder surfaces. In contact with cooler, solid surfaces such as the piston bowl and cylinder liner, the liquid fuel film is less likely to vaporize, mix with the charge gas, and burn effectively. Unfortunately, when using a diesel fuel, liquid wall impingement is far more likely for LTC strategies, which often rely on injection timing well before TDC (e.g., [1-21]). This early injection timing allows more time for fuel-ambient mixing prior to ignition and combustion, potentially avoiding fuel-rich, NO_x - and soot-forming combustion, but the spray enters into a low-density, low-temperature environment that offers little resistance to the liquid spray penetration. Figure 1 shows the charge-gas density and temperature typical for good-performance early-injection timings (e.g., [7,11]), along with a schematic showing the spray

targeting for a conventional diesel injector spray angle [4].

As indicated in the schematic, a diesel spray with a conventional, wide spray is likely to impact the cylinder liner with earlier injection timings. Indeed, optical visualization using a wide-angle injector confirms that liner impingement by liquid does occur with -40 crank-angle degree (CAD) injection timing relative to TDC [8]. Although not optically confirmed in other LTC test engines, fuel missing the bowl and possibly wetting the liner is suspected as a major source of high UHC, poor combustion efficiency, and problems such as oil dilution [5,9,10,14]. Conventional wide-angle injectors with timings of -30 to -20 CAD, still considered early compared to conventional diesel, also show liquid impingement upon the piston bowl, as confirmed by optical diagnostics [4,7].

To avoid liquid fuel impingement outside of the bowl, as well as to minimize the formation of liquid films within the bowl, researchers have attempted to use various narrow-angle injectors and bowl shapes for early-injection LTC [2,12-17]. Emissions and efficiency improvements have been noted for some low-load LTC operating conditions with narrow-angle injection, but there is difficulty utilizing available in-cylinder oxygen at high load [19] and performance and emissions suffer compared to conventional wide-angle injection [12]. In addition, although earlier injection timings may be used

to keep fuel within the bowl, liquid films still form once fuel reaches the piston bowl [2,15-17]. In some cases, these liquid films burn as pool fires [16,17], which can be a source of soot and NO_x emissions [17], counter to the goal of early-injection LTC.

Mitigation of liquid impingement upon the piston bowl, and improved mixing to achieve more premixed LTC, can be promoted by use of specialized injectors with small orifice size (e.g., [11,17,18]), however, this injection system is not optimal for high-load conventional diesel combustion [18]. Alternatively, fuels that vaporize more easily than diesel have shown promise for premixed LTC (e.g., [20,21]), in part because liquid impingement is reduced (they also have different ignition characteristics). Other strategies utilize multiple injections as a method for limiting liquid penetration and preventing formation of fuel films [8,16].

The discussion above highlights the importance of understanding spray mixing and vaporization processes, for in-cylinder conditions and fuels that are applicable to early-injection LTC. Suitable diesel LTC performance can be expected only if obstacles surrounding liquid wall wetting can be overcome.

TDC LIQUID-PHASE PENETRATION RESEARCH

One difficulty in the successful application of early-injection LTC is that little quantitative data exist for vaporizing sprays at ambient (charge-gas) conditions typical of early injection (i.e., Fig. 1). Proposed injector hardware, or spray predictions of liquid penetration to prevent wall wetting, are therefore suspect. A simple reason for this deficiency, experimentally, is that liquid impinges upon in-cylinder surfaces, thus preventing further assessment of how far the liquid would have penetrated as a free spray. In contrast to early-injection conditions, there has been significant research about the

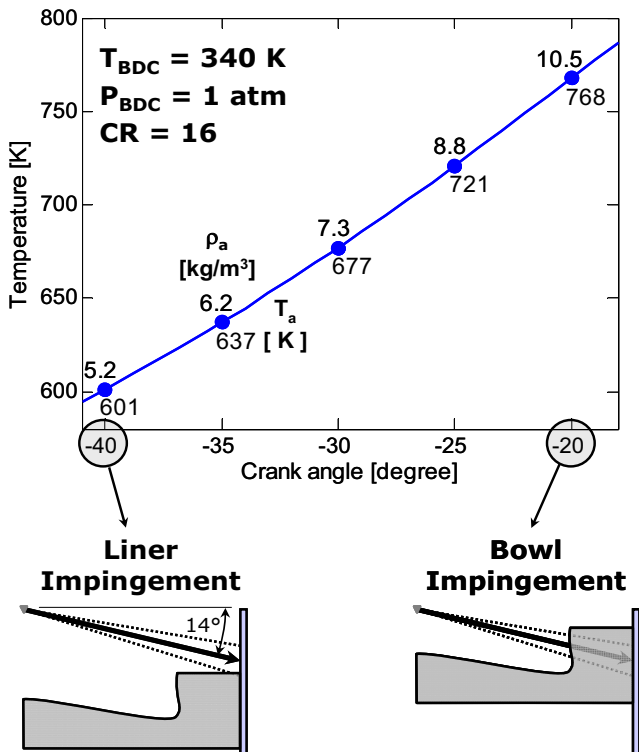


Figure 1. Computed in-cylinder charge ambient gas temperature T_a and density ρ_a at various crank-angles during compression, assuming polytropic compression, $\gamma = 1.35$, compression ratio 16, and BDC conditions given. Schematics showing the piston position, and spray targeting for a 152° included-angle injector [4], are also shown.

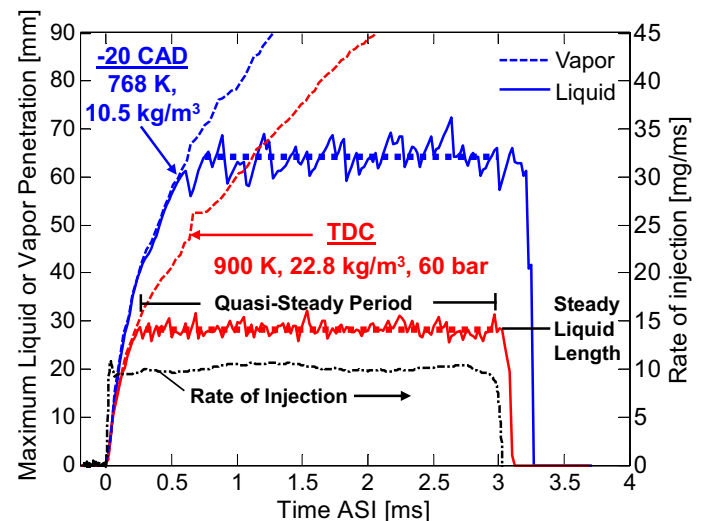


Figure 2. Measured vapor and liquid-phase axial penetration at near-TDC (red) and "early-injection" (blue) conditions. Rate of injection (black) at right axis. Ambient: 0% O_2 at temperature and density given. Injector: 0.18 mm orifice diameter, 110 MPa pressure, #2 diesel fuel. Data obtained in our facility.

Table 1. Parametric effects on quasi-steady liquid length.

Ambient temperature $T_a \uparrow$	Ambient density $\rho_a \uparrow$	Orifice diameter $d \uparrow$	Injection pressure $P_{inj} \uparrow$	Fuel 90% boiling pt. $T_{90} \uparrow$
Liq. length: $\downarrow$	$\downarrow$	$\uparrow$	none	$\uparrow$

governing processes that limit liquid penetration at traditional diesel conditions where fuel is injected near TDC [22-26].

Figure 2 shows spray liquid [22] and vapor [27] penetration results at conditions typical of past research (near-TDC injection), as well as injection at -20 CAD, an early-injection condition of more interest to this study. For both conditions, the rate of injection has a top-hat shape (bottom curve) and the injection duration is the same. With increasing time after the start of injection (ASI), the liquid and vapor phases of the spray penetrate together. The spray penetrates more slowly into a high density environment [27]. As hot gases are entrained into the spray, the liquid phase eventually reaches an axial position where the liquid is vaporized completely [22-26]. The vapor phase continues to penetrate downstream, but the liquid phase fluctuates about a mean "quasi-steady" position throughout the duration of injection, as indicated in the figure. This maximum liquid penetration distance is defined as the quasi-steady liquid length, or simply the liquid length [22].

Measurement and prediction of this quasi-steady liquid length has been the focus of significant research over a wide range of conditions [22-26]. A large part of this dataset is available to download over the internet [28]. Table 1 summarizes the trends evident from the liquid length measurements. These trends are in agreement with the results from Fig. 2. An increased ambient temperature and density, typical of a TDC condition, tends to shorten liquid length. An increased orifice diameter or fuel boiling point tends to increase liquid length [22,24] and injection pressure has no effect [22,25,26]. The reason for each specific parameter trend is discussed in detail in the cited references.

Siebers [23] has shown that the liquid length trends in Table 1 can be explained by assuming mixing-limited vaporization. To develop this claim, Siebers used a model fuel spray [27] to calculate the amount of ambient gas entrained into the spray relative to the fuel mass flow rate, $\dot{m}_a(x)/\dot{m}_f(x)$:

$$\frac{\dot{m}_a(x)}{\dot{m}_f(x)} = \frac{1}{F/A} \propto \tilde{S} = x \cdot \sqrt{\frac{\rho_a}{\rho_f}} \frac{a \cdot \tan(\theta/2)}{\sqrt{C_a} \cdot d} \quad (1)$$

where θ is the jet full spreading angle, x is axial distance from the injector, $\tilde{S}$ is the non-dimensional penetration distance, ρ_f is fuel density, and C_a is the orifice area contraction coefficient. Siebers then calculated the fuel-ambient ratio $(F/A)_{liq}$ by mass where the enthalpy

change in the ambient gases matches the energy required to heat and vaporize the liquid fuel at the local liquid-vapor equilibrium temperature. $(F/A)_{liq}$ depends upon the ambient gas temperature and density, as well as the particular fuel properties and fuel temperature. The full equation to predict the steady liquid length (L) is:

$$L = \frac{b}{a} \cdot \sqrt{\frac{\rho_f}{\rho_a}} \cdot \frac{\sqrt{C_a} \cdot d}{\tan(\theta/2)} \sqrt{\left(\frac{2}{(F/A)_{liq}} + 1 \right)^2 - 1} \quad (2)$$

where b and a are correlation constants with suggested values of 0.41 and 0.66¹, respectively.

The predictions of this model agree well with measurements of the liquid length over wide ranges of parameters given in Table 1. The model's success in predicting the liquid length, along with its independence of injection pressure, implies that diesel fuel vaporization is controlled by mixing, rather than atomization processes. However, the experimental data used for comparison with model predictions only went as low as 700 K, and Siebers notes that departures from model predictions tend to occur at low-temperature, low-density conditions.

As shown in Fig. 1, early-injection LTC conditions can have ambient temperatures of 600 K or lower, depending upon the injection timing. Further investigation of liquid penetration at these conditions is therefore warranted. In addition, the liquid length measurements and predictions are based almost entirely on the quasi-steady period of liquid penetration. For TDC conditions, this may be acceptable because the fuel spray quickly reaches a quasi-steady liquid length (e.g., after only 0.2 ms in Fig. 2). However, for early-injection conditions (-20 CAD), Fig. 2 shows that substantially more time is required for the fuel spray to reach a steady liquid-length condition and, depending upon the in-cylinder geometry, a steady liquid length may not be obtained if liquid impinges upon the wall. Furthermore, given the problems associated with LTC operation at high-load conditions, the injector may close (short injection duration) prior to attainment of a steady liquid length. In this case, the quasi-steady liquid length may have considerably less relevance to the actual liquid spray penetration. Therefore, it is important to understand the transients of liquid penetration and vaporization for early-injection conditions.

OBJECTIVE

Our objective is to study liquid-phase penetration at conditions typical of early-injection LTC, paying specific attention to the injection transients. For this study, we use a constant-volume chamber with optical access and

¹ Siebers later corrected the coefficient a to have a value of 0.75 [27], which changes b to 0.47 in order to maintain the same b/a ratio.

a free path of 100 mm prior to wall impingement. The large dimension allows assessment of steady and transient liquid penetration that would be impossible within a typical engine geometry. The ambient temperature and density of the chamber are well-controlled (uniform) and varied according to expected early-injection timings, like those given in Fig. 1. Although the constant-volume chamber cannot mimic time-varying changes to ambient conditions, as would happen during compression in an engine, the data generated are useful in that each injection parameter effect is decoupled from the uncertainty associated with changing ambient conditions.

Specific measurements of liquid penetration included in this investigation are: (1) the quasi-steady liquid length at early-injection conditions, (2) the effect of orifice diameter, injection pressure, and boost (ambient density), (3) the effect of short, transient injections of the same fuel mass, (4) the effect of multiple injections, and (5) the effect of fuel distillation properties, comparing diesel to a lighter, more-volatile, kerosene fuel.

EXPERIMENTAL APPARATUS

COMBUSTION VESSEL- A cubical-shaped combustion chamber, 108 mm on a side, was used for these experiments. The fuel injector is mounted at the center of a metal side-port such that the diesel spray is directed into the center of the chamber. Sapphire windows located in four other ports of the combustion vessel permit line-of-sight and orthogonal optical access to the injected fuel jet. Further details about the combustion chamber geometry and operation may be found in [28].

The thermodynamic state of the ambient gases within the core of the vessel was varied to simulate different early-injection conditions like those shown in Fig. 1. To eliminate uncertainty associated with heat-release effects on liquid penetration, and for comparison with past steady liquid length research [22], the ambient gas within the vessel was prepared inert, containing 0% O₂, 89.7% N₂, 6.5% CO₂ and 3.8% H₂O [28]. A mixing fan is used to maintain the uniformity of the ambient gases within the vessel, but the velocities of the gases are low (<0.5 m/s) and do not noticeably affect the penetrating spray—the vessel is essentially quiescent.

INJECTION EQUIPMENT- A Bosch second-generation common-rail injector (CRIP 2.2) equipped with a single-spray, axial nozzle was used for this study. Nozzles with different orifice diameters were included in the test matrix, and specifications for these orifices are given in Table 2. The nozzles all have convergent ($K = (d_i - d_o)/10$), hydroground orifices (KS1.5/86 specification by Bosch) and are mini-sac type with a sac volume of 0.12 mm³. The discharge (C_d) and area contraction (C_a) coefficients for these orifices were measured at the fuel injection pressure and temperature using a force sensor similar to that described in [23]. The

orifice coefficients were determined from long-injection events, where the flow is relatively steady, as shown in Fig. 2. Rate of injection measurements were also performed using the force sensor, and cumulative mass measurements, for short- or multiple-injection events to be shown.

FUELS- The distillation properties for the #2 diesel and kerosene fuels used in this study are given in Fig. 3 and further properties are given in Table 3. Note that the fuel properties are given at room temperature, which is lower than the 100 °C fuel temperature used in the experiments. The 90% boiling point (T_{90}) is 75 °C higher

Table 2. Injector orifices and flow coefficients for this study.

Nom. dia. [mm]	K factor	C_d	C_a	P_{inj} [MPa]	Fuel Temperature [K]
0.094	1.5	0.85	1.0	110	373
0.108	1.5	0.91	0.96	110	373
0.181	1.5	0.92	0.98	110	373

Table 3. Diesel and kerosene fuel properties.

	kerosene	#2 diesel
T_{10} [°C]	170	211
T_{90} [°C]	240	315
T_{100} [°C]	266	350
Cetane Index	39	47
Cetane Number	38	46
Low. Heat. Value [MJ/kg]	43.228	42.975
Density at 15°C [kg/m ³]	812	843
^s Density at 100°C [kg/m ³]	736	767
Aromatics Vol. %	11.0	27.0
H ₂ mass %	13.9	13.28
Sulfur [ppm]	1	9
Kin. Visc. (-20°C) [mm ² /s]	4.465	-
Kin. Visc. (40°C) [mm ² /s]	~ 1.4*	2.35
Freeze Point [°C]	-57	-
Flash Point [°C]	47	73

^sUsing density dependency upon temperature from [22].

*Estimated using the measured viscosity at -20°C and viscosity temperature relations for n-dodecane.

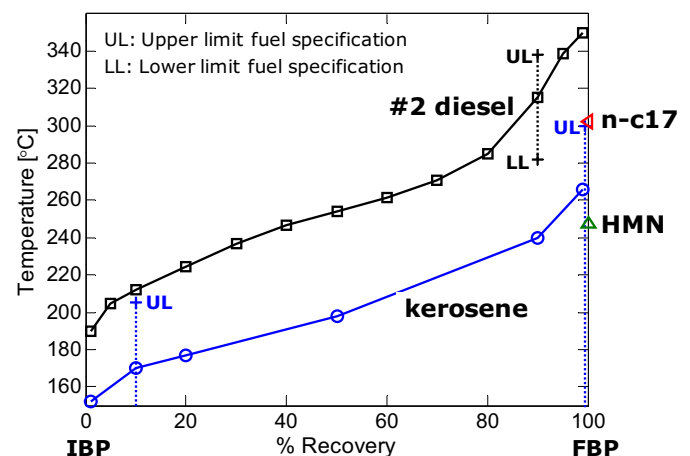


Figure 3. Distillation curves for #2 diesel and kerosene fuel samples used in this study. The atmospheric-pressure boiling points of single-component fuels n-heptadecane (n-c17) and heptamethylnonane (HMN) are also given.

for diesel compared to kerosene. The liquid penetration is therefore expected to be longer for diesel. For reference, the boiling point temperature of two single-component fuels, n-heptadecane (n-c17) and heptamethylnonane (HMN), are also indicated on Fig. 3. The boiling points of these single-component fuels are close to the T_{90} of the real, multi-component fuels, and they will be used later to model the liquid penetration of diesel and kerosene. Siebers [23] found good prediction of #2 diesel liquid lengths when using n-c17 as a surrogate fuel.

LIQUID AND VAPOR HIGH-SPEED IMAGING DIAGNOSTICS

Mie-scatter and shadowgraph imaging were performed simultaneously using high-speed imaging systems to track both the liquid and vapor phases of the spray simultaneously. The results from Fig. 2 were obtained with these techniques. The ability to resolve the transients of a single injection sequence is especially important for sprays that do not attain a steady behavior, or, for example, if there are multiple injections where interaction between successive injections occurs. Also, the extent of fluctuation in liquid penetration, even at steady injection conditions, is important to understand the probability of liquid impingement upon in-cylinder surfaces. The average steady liquid length does not portray this information. For these reasons, results for single injections, rather than ensemble-averages, are shown in this paper.

MIE-SCATTER IMAGING- Scattered light from diesel spray liquid droplets was imaged using a high-speed CMOS camera and a continuous-wave laser. As shown in Fig. 4, a shadowgraph system was also operated simultaneously using a second high-speed CMOS camera. The time-resolved pair of shadowgraph and Mie-scatter images identifies the instantaneous position of both the vapor and liquid phases of the fuel spray.

The continuous-wave laser source was a frequency-doubled Nd:YAG laser (532 nm) operating with 2 W power. The laser was formed into a volume to illuminate only the liquid-phase region of the spray and also to avoid interference with the shadowgraph setup (Fig. 4). A volume illumination method, rather than a laser sheet, was utilized to ensure that all droplets spreading from the nozzle were illuminated to identify the maximum axial and radial distances of any liquid-phase fuel [22].

A high-speed CMOS camera, fitted with a 532-nm bandpass filter (10 nm FWHM) and a 50-mm focal length f/1.2 lens, was used to image Mie-scattered light at a near-right angle to the laser source. The bandpass filter was used to reject background thermal radiation from heated vessel walls. The camera was operated at reduced resolution (48 x 512 pixels) to allow fast framing periods ($\sim 20 \mu\text{s}$) needed to capture dynamics of very

fast diesel sprays. The lens was set to the maximum aperture (f/1.2) and the camera at maximum exposure duration (17 μs) to intentionally saturate the camera in the dense region of the spray while leaving sufficient dynamic range to identify regions of the spray where liquid completely evaporates. However, changes in laser power, exposure duration, or image threshold level (8/256 counts) did not significantly affect the measurable liquid penetration, similar to [22].

SHADOWGRAPH IMAGING- Shadowgraph and schlieren imaging have been used for many years to identify refractive index gradients in various experiments [31]. For vaporizing diesel sprays, these techniques show the boundary between vaporized fuel and background ambient gases.

Figure 4 shows the lighting and imaging arrangement utilized to perform “focused” shadowgraph imaging. The setup and application is described in more detail in Ref. [32]. A 150-W (electrical power) mercury-xenon arc lamp with a small (1.7-mm) arc distance was used as a white light source. Light from the arc was filtered and collimated to pass through the combustion vessel. The collimated beam was then re-focused into a high-speed CMOS camera (Phantom v7.1) equipped with a Nikkor lens and a 532-nm 0°-incidence laser mirror, which acted to block Mie-scattered light from spray droplets.

As discussed in Ref. [32], a software routine was developed to detect the vapor boundary, based on differences between successive images and the gradient of a particular image. This method is needed to account for natural variations in the refractive index of the ambient background, caused by temperature non-uniformities within the boundary layer of the vessel. These non-uniformities of the ambient do not change

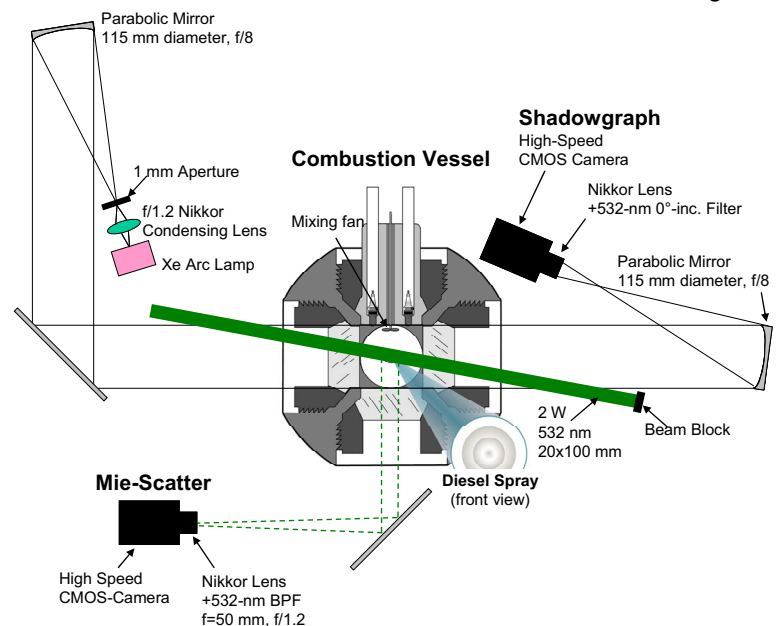


Figure 4. Combustion vessel and high-speed imaging setup. Mirror spacing not to scale.

significantly during injection, however, making detection of the vapor boundary more straightforward. Examples of vapor boundaries will be shown later, but measurements of the axial vapor penetration, already shown in Fig. 2, were the result of this method.

The shadowgraph (vapor boundary) images were also analyzed to determine the jet spreading angle. For consistency with Siebers' method [23], images were time-averaged and analyzed in the region of the jet where liquid was also present, and the spreading angle was defined as the 50% threshold between the ambient and inner-spray intensity of the image (see Appendix A from [23]).

RESULTS AND DISCUSSION

The Results and Discussion is divided into four different sections. First, we present the steady-state liquid length at early-injection conditions for different orifice diameters and fuels. Next, we discuss the use of intake boost (turbocharging) as a method to limit the steady-state liquid penetration. Informed about the steady-state liquid behavior, we then show how liquid penetration can be reduced from the steady-state liquid length by using short transient injections. We discuss the transient liquid penetration for orifice diameters and injection pressures with the same total mass injected. Finally, we show how multiple injections affect the transient liquid penetration.

STEADY-STATE LIQUID LENGTH AT EARLY-INJECTION CONDITIONS- As mentioned previously, there is little steady-state liquid length data at early-injection conditions because wall impingement occurs within typical engine geometries. To begin this study, we therefore examined long injection durations within the 100-mm combustion vessel. The ambient gas temperature and density were varied according to a certain crank-angle degree during compression. Consider the temperature and density plotted in Fig. 1, which are for a BDC temperature of 340 K, and pressure of 1 bar. Obviously, many different BDC conditions exist, but 340 K is reasonable when considering heat transfer from the head, as well as internal and external EGR addition. The effect of boost, which changes the BDC pressure, and ultimately the ambient density, will be considered in the next section.

As representative for this dataset, Fig. 5 shows high-speed Mie-scatter images of liquid penetration during an injection with a relatively small orifice diameter (0.108 mm) and #2 diesel fuel. The ambient temperature and density are the conditions at -30 CAD from Fig. 1. Like the result shown in Fig. 2, the injection duration is long enough to show a quasi-steady period for liquid penetration. Blue border lines are given on the figure to show the liquid penetration (8/256 counts) position. The time between displayed images is not constant and may be larger than the image acquisition speed, in order to highlight various stages of the injection. The image

sequence shows a bright spray cluster growing from the injector orifice. The intensity of the scattered light begins to fade as the spray penetrates across the chamber, and the spray eventually reaches an axial distance where liquid penetrates no further.

The maximum liquid penetration from this injection and for other ambient conditions associated with early-injection CAD (see Fig. 1) is shown in Fig. 6. The liquid penetration history shows attainment of a quasi-steady liquid length after a certain time after the start of injection. However, there is significant fluctuation in the maximum liquid penetration during this quasi-steady period of injection. With 50 kHz acquisition speed, the Mie-scatter image sequence shows that these fluctuations are characterized by large-scale structures evolving at the tip of the jet, which may cause a break in the connected liquid region. Three instances of this disconnected liquid are labeled as **A**, **B**, or **C** in Figs. 5-6. As a downstream cluster of droplets becomes isolated from the rest of the spray, the droplets eventually evaporate. At this instant, the maximum liquid penetration distance reverts to the next upstream parcel of liquid, which causes a sudden decrease in the calculated maximum penetration distance. These types of fluctuations in maximum liquid penetration are also typical for conditions with shorter liquid penetration, for example the near-TDC condition shown in Fig. 2. The length scale of these fluctuations is smaller near the injector, however. By contrast, the vapor penetration of the jet, or liquid penetration prior to reaching the liquid length, does not show the same fluctuation component because the spray continuously penetrates downstream.

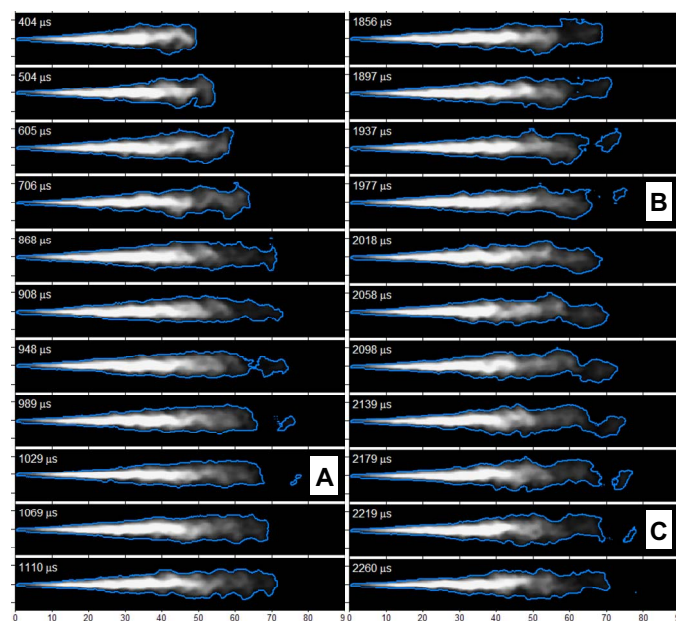


Figure 5. Mie-scatter images of liquid penetration with blue border showing the boundary of the liquid. Simulated ambient conditions corresponding to -30 CAD from Fig. 1: Ambient temperature 677 K, ambient density 7.3 kg/m³. Injector conditions: 0.108 mm orifice, 110 MPa, #2 diesel, with long injection duration. Scale is distance in mm.