

Relationship Between Diesel Fuel Spray Vapor Penetration/Dispersion and Local Fuel Mixture Fraction

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ABSTRACT

The fuel-ambient mixture in vaporized fuel jets produced by liquid sprays is fundamental to the performance and operation of engines. Unfortunately, experimental difficulties limit the direct measurement of local fuel-ambient mixture, inhibiting quantitative assessment of mixing. On the other hand, measurement of global quantities, such as the jet penetration rate, is relatively straightforward. Simplified models to predict local fuel-ambient mixture have also been developed, based on these global parameters. However, experimental data to validate these models over a range of conditions is needed. In the current work, we perform measurements of jet global quantities such as vapor-phase penetration, liquid-phase penetration, spreading angle, and nozzle flow coefficients over a range of conditions in a high-temperature, high-pressure vessel. Using this data and other quantitative mixing measurements performed by Rayleigh scattering in the vaporized portion of the jet, we compare to an existing variable-radial-profile model for prediction of fuel mixture fraction during the steady period of injection. Results show that spreading angles based on measurement of the most

sensitive outer boundary of the jet, by schlieren or Rayleigh-scatter imaging, are needed as inputs to the model to obtain a match between modeled and measured fuel jet penetration rates. By adjusting the model (with spreading angle) to match the measured penetration, the model predictions also produce local mixture fractions that are within the Rayleigh scattering experimental uncertainty. Using this same penetration-matching technique, accurate model predictions of mixture fraction are achieved for a range of ambient densities, fuel injector nozzle shapes, injection pressures, and types of fuels. Additionally, extrapolation of the mixing measurements suggests that a fuel spray has a smaller spreading angle in the near-field and transitions to a larger angle in the far-field jet.

INTRODUCTION

Controlled fuel-ambient mixing is key to combustion optimization in engines. For direct-injection engines, an atomized liquid spray evolves into a vaporized fuel jet as it penetrates into the combustion chamber. For clarity, throughout the paper we will refer to the liquid-phase region as the “spray” region and the vapor-phase region as the “jet”

region. However, currently there are many shortcomings in our understanding of sprays and jets and the type of fuel-ambient mixtures produced in engines. For diesel or low-temperature combustion engines, some of the most basic aspects of sprays and jets, including penetration rate and spreading angle, are not fully repeatable from injector-to-injector, or even from hole-to-hole of an individual injector. Causes for these variations likely originate from internal injector and nozzle flows, but these processes are not well understood. As such, computational fluid dynamics (CFD) models for spray breakup and dispersion are not completely predictive. Indeed, a recent review of CFD engine simulation capability identifies spray modeling as the major uncertainty [1].

Because of current limitations in our understanding of sprays and jets, experimentation is required to better characterize them for model validation. Often, CFD spray sub-models are calibrated with measurements of spray penetration and spreading angle. Probably the most common experimental setup for these types of measurements is a pressurized spray chamber operating at room temperature. While many aspects of fuel injection in an operating engine are likely represented, there are several problems with interpretation of data from this type of experiment. First, since a room-temperature spray is essentially non-evaporating, difficulties arise when attempting to derive fuel mass concentration from droplets of different diameter and number density. The relationship of scattered or extinguished light from droplets to fuel concentration is difficult to quantify, particularly in the dense region near the nozzle. Without quantification, the “boundary” of the spray can be difficult to define as it relates to spreading angle or penetration. Second, the mixing processes in a room temperature experiment may be different than those in the high-temperature, high-pressure in-cylinder engine environment. Droplets in diesel fuel sprays injected into gases at high temperature and pressure will vaporize and create a different physical environment in which mixing occurs. For these reasons, spray chambers have been developed that operate at elevated pressures and temperatures typical of an engine. A recent review of these facilities can be found in [2].

Even for chambers that operate at high pressure and temperature, measurements of the spray and jet penetration and spreading angle are ubiquitous in the literature. Simplified penetration and mixing models derived from these data are also fairly common. As such, we will review several experimental observations and existing spray/jet modeling approaches from our own group for which we have the most experience. The general experimental observations and modeling experience are shared, at least in part, by a number of other investigators.

One existing spray/jet modeling approach is the uniform-profile model of Naber and Siebers [3]. Although the model

was derived for non-vaporizing sprays, it was found to be predictive of jet penetration for vaporizing sprays as well. Using high-speed schlieren imaging, they compared the penetration and spreading angle of evaporating (high-temperature) and non-evaporating (low-temperature) sprays over an extensive range of gas densities, even surpassing that typically experienced in an engine (3-60 kg/m³). The parametric study also included data for a range of injection pressures (76-160 MPa), injector nozzle orifice diameters (0.198-0.340 mm), and elevated ambient temperatures (600-1400 K). They showed that penetration slows with increasing ambient gas density, and that the density effect on penetration partly occurs as a result of the effect of ambient density on the spreading angle (*i. e.*, air entrainment and mixing). They also found that the penetration rate and spreading angle of an evaporating spray were both lower than that of a non-evaporating spray, which they attributed to vaporization cooling that contracts the spray.

Using the non-vaporizing and vaporizing data coupled with a simplified model for a fuel jet, Naber and Siebers developed an analytical model describing penetration based on conservation of mass and momentum for a steady fuel injection rate into a developing control volume. For simplicity, the velocity and mass distributions within a conical envelope for the jet were assumed radially uniform at any axial distance from the injector, x . They showed that all of their penetration data collapsed to a single curve predicted by a non-dimensional form of their penetration model [3]. A constant appearing in the model formulation was adjusted to a single value to best fit all the penetration data. The constant effectively accounted for the uniform velocity and mass assumption. The result was a model that accurately predicts penetration when using measured spreading angles from the dataset.

Another existing spray/jet modeling approach is the variable-profile model of Musculus and Kattke [4], which is an extension of the uniform-profile non-vaporizing model of Naber and Siebers. The variable-profile model adds the option for a realistic radial distribution of fuel mass and velocity at any axial distance from the injector. The original purpose of the variable-profile model was to simulate penetration and mixing during and after the end of injection for variable rates of injection. To do so, the variable-profile model used a one-dimensional numerical control-volume solution, discretized along the spray axis [4,5]. However, when the fuel injection rate is steady, the variable-profile model also has an analytical solution that collapses exactly to the “uniform-profile” model under the assumption of uniform velocity and mass profiles. The analytical solution for the variable-profile model is described in more detail here because we use it extensively in the following discussions.

Figure 1 shows a schematic of the model. The mass radial profiles are given by Eq. (1),

$$\frac{\bar{\chi}_f}{\bar{\chi}_{f,c}} = (1 - \xi^\alpha)^2 \quad (1)$$

where $\bar{\chi}_f$ is the turbulent mean fuel liquid volume fraction, the subscript c denotes the value on the jet centerline, and $\xi = r/R$ is the ratio of the radial coordinate r to the jet width, where the jet width is given by $R = \tan(\theta/2)x'$, and θ is the full spreading angle, which is required as an input to the model. As indicated in Fig. 1, the velocity radial profile has the same distribution as the mass profile given by Eq. (1). The exponent α in Eq. (1) is generally variable, and can be adjusted to fit the evolving radial velocity profile as the flow exits the nozzle [6]. For most of the jet downstream of the short transition region near the nozzle, however, the radial profile becomes self-similar, so that α is set to a constant value of 1.5. This is the recommended value for a fully developed profile [7], which produces a radial profile resembling a Gaussian error function as depicted in Fig. 1. Note that the spreading angle input used is for the very outer edge of the jet R where fuel mass is zero, which may be difficult to locate experimentally. However, other definitions for jet spreading angle are possible because of the well-defined jet radial distribution. For example, if one uses a full-width at half-maximum criteria (FWHM) for the spreading angle, this FWHM spreading angle is approximately 45% of the full angle θ , as prescribed by the radial distribution given by Eq. (1).

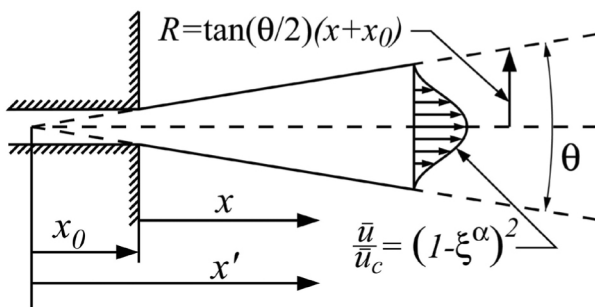


Figure 1. Variable-profile diesel jet model of Musculus and Kattke [4].

The radial profile given by Eq. (1) represents the fuel liquid volume fraction with no explicit consideration of fuel evaporation. In addition, the model assumes no velocity slip between injected fuel and entrained gases. Accordingly, momentum transfer between fuel droplets and entrained gas is assumed to be perfectly complete and the local fuel-ambient mixture is independent of whether there are droplets or vaporized fuel. Essentially, droplet size has no effect and there is no difference between this “spray” model and that of a hypothetical dense gas jet with the same momentum and

mass flow rate. The local fuel-ambient mass ratio F/A can therefore be related to $\bar{\chi}_f$ as

$$\frac{F}{A} = \frac{\rho_f \cdot \bar{\chi}_f}{\rho_a \cdot (1 - \bar{\chi}_f)} \quad (2)$$

where ρ_f and ρ_a are the fuel and ambient densities, respectively. The mixture fraction Z , which is equivalent to the fuel mass fraction Y_f if the system is inert, is then

$$Z = Y_f = \frac{F/A}{(F/A + 1)} \quad (3)$$

The model therefore provides a prediction of the local mixture fraction at any position within the jet, dependent upon model inputs such as the full spreading angle θ , ρ_f , and ρ_a , and the assumed radial fuel concentration profile shape. In addition, the nozzle area contraction coefficient C_a , and discharge coefficient C_d , must be specified to determine the momentum and mass flow rates of the spray at the nozzle [3-4]. These nozzle coefficients typically must be measured experimentally [3,8,9,10,11].

These simplified spray/jet mixing models offer a highly accessible and straightforward method to quantify mixing. These models are widely used as tools to aid the interpretation of combustion measurements, including estimation of mixing to determine liquid-phase penetration length [9] or soot formation trends. They are also used firsthand to guide spray models within more detailed CFD applications [12,13].

While simplified spray or jet mixing models have been developed, even including a variable radial profile for fuel mass and velocity, their representation of mixing has not been validated against experimental data that include direct measurement of mixing in a fuel jet at diesel conditions. Such comparisons would also offer an opportunity to better “calibrate” these simplified models. The major shortcoming for such comparisons is that quantitative measurement of mixture fraction at high temperature and pressure conditions representative of a diesel jet is hindered by many experimental difficulties.

The experimental difficulties result from both spray and diagnostics sources. The dense liquid portion of the spray near the nozzle is inaccessible to quantitative optical diagnostics. Downstream of the dense liquid region, laser-based diagnostics for fuel-oxidizer mixing that have been attempted in evaporating diesel sprays include Raman scattering [14,15,16], laser-induced fluorescence/excited fluorescence (LIF/LIEF) [17,18,19,23], Mie scattering

[20-21], and Rayleigh scattering [22,23,24,25]. Each technique has its own advantages and disadvantages. For example, elastic scattering techniques like Mie and Rayleigh suffer due to interference from scatterers other than the probed species, and Raman scattering is limited by low signal strength and poor spatial resolution (usually 1-D instead of 2-D). In the case of LIF/LIEF techniques, elastic scattering interferences are not as serious, but the fluorescence yield is generally a function of temperature, pressure, and ambient gas composition, making quantitative measurement of transient mixing processes at engine conditions difficult. Exciplex fluorescence has not proven quantitative for liquid concentration [17,18,19], limiting any attempt at quantification to the fully vaporized jet region, rather than the liquid spray region. Also, the injector nozzle coefficients or global parameters such as the liquid spray (or vaporized jet) spreading angle and penetration rate typically have not been reported along with mixing measurements in these studies, making evaluation against a spray/jet mixing model difficult. To date, none of the mixing data from Refs. [14,15,16,17,18,19,20,21,22,23,24,25] has been used to explore the validity of mixing models like that shown in Fig. 1. Data from Ref. [25], available to download from the Engine Combustion Network [26], was recently used for comparison to various multi-dimensional CFD spray models [27].

In addition to these experimental challenges for making detailed mixing measurements at diesel conditions, many widely used techniques for measuring a primary global parameter, the spreading angle, are not fully quantitative or easily reproduced. For example, schlieren techniques used for measuring the jet vapor boundary are not quantitatively relatable to the actual mixture fraction at the vapor boundary. This problem is especially difficult when the ambient gas surrounding the jet is at high temperature and pressure. Non-uniformities in ambient gas temperature create refractive index gradients that appear in the “background” of schlieren imaging [1,3,28], making it difficult to determine the boundary between fuel vapor and unmixed ambient gas. Moreover, increasing the ambient gas pressure (or density) intensifies the refractive index gradients, as the refractive index and its gradients tend to scale with density. The schlieren sensitivity to refractive index gradients can be adjusted optically [29] but this may reduce the ability for detection of the very edge of the jet. Ultimately, it is unknown what mixture fraction corresponds to the jet “border” apparent from variations in the schlieren intensities. As will be shown, however, this challenge is less of an issue for detecting the leading edge of the penetrating jet penetration, where the jet border is marked by sharp gradients that are more easily observed and measured.

Yet another complication is that researchers may not make the distinction between the liquid-phase spray width or penetration, compared to those of the vapor-phase jet at high-

temperature, vaporizing conditions. For example, Mie-scattering may be used to measure the liquid-phase spray angle and penetration, which is expected to be different than that of the vapor-phase. Since spray (or jet) models depend upon the spreading angle derived from these boundaries to determine spray breakup, or the amount of ambient gas entrained into the spray/jet, their calibration is suspect without knowledge of the mixture fraction (or fuel-ambient ratio) that corresponds to a particular spreading angle.

OBJECTIVES

Our objectives are to quantitatively measure fuel-vapor concentration in evaporating diesel sprays, and to use these data for the following:

- To identify the mixture fraction at the edge of the jet that corresponds to the spreading angle detected by schlieren imaging. We will specifically address complications associated with different schlieren imaging setups and different ambient densities.
- To evaluate the accuracy of the mixing prediction for the variable-radial-profile model depicted in Fig. 1, as well to determine if the assumptions employed in deriving this model are justified. We will use measured “global” parameters such as the spreading angle and penetration rate, along with measurements of the nozzle flow coefficients as inputs to the model. We will compare model predictions and mixing measurements for a range of ambient densities, fuels, nozzle sizes, and nozzle shapes, to address the robustness of the model at various conditions.

The comparisons will provide new insights into diesel sprays and jets, validate the general structure of the models, and elucidate the challenges in measuring both global mixing (*i.e.*, the spreading angle) and spatially resolved fuel concentration.

STEADY FLOWRATE MODELS FOR MIXING AND PENETRATION

Analytical solutions for mixing and jet penetration have been developed for both the uniform-profile model (Naber/Siebers) [3], and the variable-profile model (Musculus/Kattke) [4] that is depicted in Fig. 1, provided that the rate of fuel injection is steady. We later show that after a short (less than 0.1 ms) transient, the rate of injection is indeed steady for the injectors used in this study, permitting use of these analytical solutions. In this section, we provide a brief summary of both models and their respective analytical solutions.

UNIFORM RADIAL PROFILE

Mixing

The uniform-profile model of Naber and Siebers has been used frequently by researchers at our laboratory and others as

a means for estimating mixing and penetration in sprays/jets (e.g. [3,9,12,13,30,31,32,33,34]). The amount of ambient oxygen entrained into the jet relative to the fuel injected has typically been expressed in terms of the cross-sectionally averaged quantities, because of the uniform radial distributions for mixing and velocity. That is, the total fuel-ambient mass ratio (F/A) is related to the axial distance from the injector with Eqs. (4),(5),(6),(7):

$$\frac{F}{A} = \frac{2}{\sqrt{1+16 \cdot \tilde{x}^2} - 1} \quad \text{or} \quad \bar{\bar{\phi}} = \frac{2 \cdot (A/F)_{st}}{\sqrt{1+16 \cdot \tilde{x}^2} - 1} \quad (4)$$

The expression on the right of Eq. (4) is the averaged equivalence ratio, $\bar{\bar{\phi}}$, from Eq. (C32) of Ref. [3], where the double overbar signifies both a turbulent mean and an average over the jet cross-section, the subscript st denotes stoichiometric quantities, and:

$$x^+ = \tilde{\rho}^{1/2} \left(\frac{\tilde{\rho} - m}{\tilde{\rho}} \right) \frac{\sqrt{C_a} \cdot d_0}{a \cdot \tan(\theta/2)} \quad (5)$$

$$\tilde{\rho} = \rho_f / \rho_a \quad (6)$$

$$\tilde{x} = x' / x^+ . \quad (7)$$

Eq. (5) provides a characteristic length scale x^+ for the fuel jet, where d_0 is the nominal orifice diameter, C_a is orifice area contraction coefficient, and m is a non-dimensional parameter set to a value of one to represent, or zero to neglect, the liquid fuel area [3]. Typically, for the sake of simplifying the mathematics, m is set to zero for past uniform-profile model considerations [3,9,30,31,32,33,34]. The constant a in Eq. (5) effectively reduces the spreading angle in the model relative to the measured spreading angle to account for the uniform-profile assumption, as will be discussed below.

Penetration

Naber and Siebers [3] derived an analytical solution for penetration, relating a non-dimensional penetration time ($\tilde{t}$) to a non-dimensional axial penetration distance ($\tilde{x} = S$) as:

$$\tilde{t} = \frac{\tilde{S}}{2} + \frac{\tilde{S}}{4} \cdot \sqrt{1+16 \cdot \tilde{S}^2} + \frac{1}{16} \cdot \ln \left(4 \cdot \tilde{S} + \sqrt{1+16 \cdot \tilde{S}^2} \right) \quad (8)$$

$$t^+ = \tilde{\rho}^{1/2} \left(\frac{\tilde{\rho} - m}{\tilde{\rho}} \right)^2 \frac{\sqrt{C_a} \cdot d_0}{\bar{\bar{u}}_0 a \cdot \tan(\theta/2)} \quad (9)$$

$$\tilde{t} = t / t^+ \quad (10)$$

where t^+ is a characteristic time scale for penetration and $\bar{\bar{u}}_0$ is the cross-sectionally average velocity at the nozzle exit. Note that the constant a again appears in Eq. (9), as it did in Eq. (5). As mentioned in the Introduction, the constant a was adjusted to a single value to give the best fit between non-vaporizing spray penetration data and model predictions [3]. The constant effectively reduces a measured spreading angle to a model angle that partially accounts for effects of the uniform-profile assumption compared to a realistic radial profile. Originally, a was assigned a value of 0.66 [3], but this was later changed to a value of 0.75 based on later measurements of the orifice flow coefficients [34].

VARIABLE RADIAL PROFILE

The variable-profile model of Museums and Kattke [4,6] builds upon the uniform-profile model, essentially offering solutions for either uniform-profile or variable-profile constructs of the model. For example, much of the work in Ref [4] shows uniform-profile model solutions, for simplicity aimed at expressing total ambient entrainment, rather than detailed radial distributions, in agreement with the purposes of the uniform-profile model discussed above. The variable-profile model from Ref. [4] also has the capability to accommodate arbitrary injection rates by incorporating multiple control volumes. This modeling form is more general, but it requires numerical analysis for the mass and momentum exchange between control volumes. When the rate of injection is steady, however, the numerical solution is unnecessary and analytical solutions for mixing and penetration are available. The derivation of these solutions is found in [Appendix A](#) and [Appendix B](#), but key simplified expressions are included in this section.

Mixing

Similar to the uniform-profile model, the variable-profile model solution for mixing is based on conservation of mass and momentum in the axial direction of the spray. The shape of the radial profiles for velocity and fuel mass is dictated by Eq. (1). In applying mass and momentum conservation to derive a penetration solution, the terms involving α from Eq. (1) can be collapsed together into a single term, β , as introduced in Eq. (5c) of Ref. [4].

$$\beta = \frac{6(\alpha+1)(\alpha+2)}{(3\alpha+2)(2\alpha+1)} \quad (11)$$

Equation (11) shows that β is a function of α , and hence is a derived metric for the radial profiles of velocity and fuel mass. In the nearfield limit at the nozzle exit, $\beta = 1$ (with $\alpha = \infty$) is prescribed to produce a uniform radial profile. In the

fully developed farfield limit downstream of the nozzle, $\beta \approx 2$ (with $\alpha = 1.5$). In a short transition region between these two limits, Ref. [4] recommends that the value of β (and α) be adjusted between $\beta = 1$ and $\beta \approx 2$ as necessary to prevent unrealistic velocity profiles that exceed the nozzle exit velocity. The transition region typically extends only a few millimeters downstream of the injector, however, so that most of the jet has $\beta \approx 2$ (constant).

Appendix A shows that the average fuel liquid-volume fraction at a given axial distance from the injector is:

$$\overline{\overline{\chi}}_f = \frac{2}{(\tilde{\rho} - 1) \left[\sqrt{1 + 16\beta\tilde{x}^2} - 1 \right]} \quad (A7)$$

where the double overbar again signifies both a turbulent mean and an average over the jet cross-section. Comparing Eq. (A7) to Eq. (5), the constant a is effectively equal to 1 for Eq. (A7), meaning that an empirical modification of the spreading angle is not necessary because of the use of a realistic radial profile of fuel mass and the development of Eq. (A7) takes place without any such constant. Similarly, the constant m to account for the area of liquid is also effectively set to 1 in Eq. (A7). The significant result is that $\tilde{x}$ in Eq. (A7) and Eq. (4) are not equivalent at the same distance from the injector, because of the different values for the constant a , and hence x^+ . Note that the form of Eq. (A7) closely resembles the uniform-profile solution given in Eq. (4), with the addition of the term β , although Eq. (A7) is for fuel liquid volume fraction rather than fuel-to-ambient mass ratio as in Eq. (4).

The centerline and cross-sectionally averaged terms for liquid volume fraction are related using Eq. (4a) of Ref. [4]:

$$\overline{\chi}_{f,c} = \frac{(\alpha + 1)(\alpha + 2)}{\alpha^2} \overline{\overline{\chi}}_f \quad (A8)$$

In Eq. (A8), the single overbar of the term $\overline{\chi}_{f,c}$ signifies a turbulent mean only (the term corresponds to the jet centerline). Using Eqs. (1), (2), (3), combined with Eqs. (A7) and (A8), the fuel liquid-volume-fraction can then be used to provide a local fuel-ambient ratio or mixture fraction prediction at any axial and radial location of the jet. For example, the fuel-ambient ratio by mass can be compared to the uniform-profile solution given in Eq. (4).

This variable-profile mixing solution is used throughout this paper for comparison to experimental results.

Penetration

For the uniform-profile model, an analytic solution for penetration was determined by integrating the jet velocity

function [3]. However, a similar approach is not appropriate for a variable-profile velocity distribution. For instance, the centerline velocity in the steady jet (behind the head of the jet) is typically much faster than the jet penetration rate, due to turning and deceleration of the internal jet flow as it approaches the head vortex. The jet penetration rate should therefore consider the transfer of momentum as flows stagnate and turn at the jet head, which requires analysis of the entire momentum at the jet head. In Appendix B, the jet penetration is derived by equating the spatial integral of momentum over the whole length of the jet to the time integral of momentum flux delivered by the nozzle. In this way, jet penetration is predicted based on momentum, rather than time integration of any specific velocity in the jet.

The full variable-profile analytic solution for jet penetration becomes quite involved, however, because of the non-constant β term within the transition region. The full analytic solution is given in Appendix B in Eqs. (B3) and (B13), which is the solution used to compute jet penetration and to compare with experimental results in this study. A simplified analytical solution is available if β is assumed to be constant throughout the entire jet. For example, $\beta = 105/52$ corresponds to the fully developed jet region. When β is assumed to be constant and the same non-dimensional terms given in Eqs. (5), (6), (7) and (9)-(10) are inserted, Eq. (B3) simplifies to

$$\tilde{t} = \frac{\tilde{S}}{2\beta} + \frac{\tilde{S}}{4\beta} \sqrt{1 + 16\beta\tilde{S}^2} + \frac{1}{16\beta^{3/2}} \ln \left(4\sqrt{\beta}\tilde{S} + \sqrt{1 + 16\beta\tilde{S}^2} \right) \quad (12)$$

with $\tilde{x} = \tilde{S}$. Just like Eq. (A7) above, the constants a and m in Eq. (5) and Eq. (9) are effectively set equal to 1, as this is how Eq. (12) is developed. The implication is that no artificial reduction in spreading angle is required because realistic radial profiles for mass and velocity are already employed. Indeed, Eq. (12) can be considered a general solution that includes Eq. (8), in that if a uniform profile is assumed ($\beta = 1$), Eq. (12) simplifies to the uniform-profile solution of Eq. (8) exactly.

Figure 2 shows that the penetration predictions from either the uniform-profile or variable-profile analytical solutions are also similar, when the constant a for the uniform-profile condition is set to values similar to that found by Naber and Siebers [3]. Five different curves are shown on Fig. 2. The two red curves are for uniform-profile solutions using Eq. (8) with the constant a equal to either 0.66 or 0.68. The green curve is the variable-profile solution from Eq. (12) with $\beta = 105/52$. The blue curve is the variable-profile analytical solution derived in Appendix B, which uses two approximations to admit an analytical solution in the near-injector transition region. The dashed black curve is the numerical integration of Eq. (B1), representing the “exact” solution for the variable-profile penetration. The difference

between this exact solution and the other curves is shown at the top of the figure.

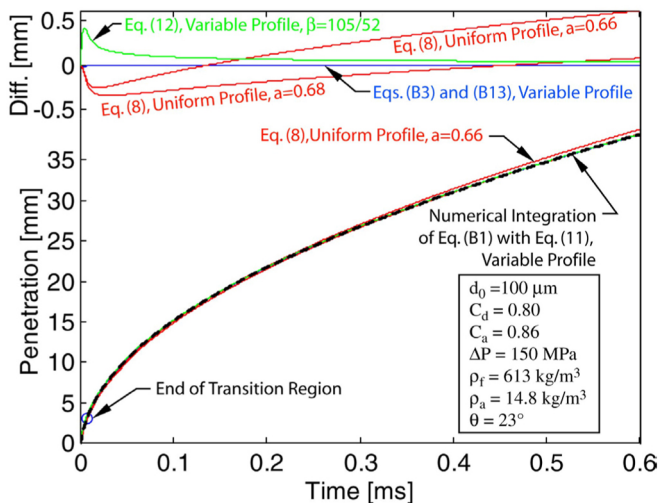


Figure 2. Penetration predictions with either uniform-profile or variable-profile analytic solutions. The top shows the difference between solutions, relative to the exact numerical integration of Eq. (B1) given in dashed black lines. The circle symbol marks the end of the transition region, downstream of which $\beta = 105/52$. Experimental conditions given in legend.

The figure shows that differences from the exact variable-profile solution are less than 0.5 mm for the first 30 mm of penetration at these experimental conditions. The variable-profile solution from Eq. (12) with constant $\beta = 105/52$ differs from the exact solution most significantly near the nozzle in the transition region, but then remains close to the exact solution downstream. Therefore, Eq. (12) could be used as suitable approximation for the penetration when also assuming a variable radial profile velocity distribution. Also, although the full variable- β analytical solution from Appendix B is virtually identical to the exact solution, the constant- β version in Eq. (12) deviates only slightly (less than 0.5 mm) from the exact solution, and is much less cumbersome to use. The uniform-profile solutions have slightly lower penetration near the nozzle, but then faster penetration downstream. At this particular experimental condition, a uniform-profile solution with a value of 0.68 for the constant a is a better match to the variable-profile penetration downstream than when the constant a has a value of 0.66, although either solution is quite close the variable-profile solution.

Note that the constant a is not used in the variable-profile solution, so a comparison between uniform-profile and variable-profile penetration can be used as justification for a particular value of the constant a . As mentioned above, Naber and Siebers first recommended a value of 0.66 for this constant based on tuning the model using measured

penetration and measured spreading angle data [3]. Values of 0.66 to 0.68 appear quite acceptable based on Fig. 2, but agreement with [3] may be partly serendipitous because a was later changed to a value of 0.75 based on later measurements of the area contraction coefficients [34]. However, ultimately one must ask whether the experimental method for determination of full spreading angle in [3], which leads to different values for the empirical constant a depending upon the method, would agree with the full angle for the variable-profile model. For the spreading angle and radial profiles defined here, the variable-profile penetration modeling results suggest that the constant a should be less than a value of 0.70, rather than 0.75. But is the spreading angle and radial profile justified in this variable-profile model? The objectives for this paper are precisely to answer these experimental and modeling questions. We will discuss the experimental approach to address these questions in the next sections.

EXPERIMENTAL APPARATUS

COMBUSTION VESSEL

Experiments were performed in a constant-volume combustion vessel under simulated engine conditions. The combustion chamber is cubical, measuring 108 mm on each side. Figure 3 shows 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. Windows located in three other ports of the combustion vessel permit line-of-sight and orthogonal optical access to the injected fuel jet. The optical diagnostic setups will be discussed later. Further details about the combustion chamber geometry and operation may be found in [26].

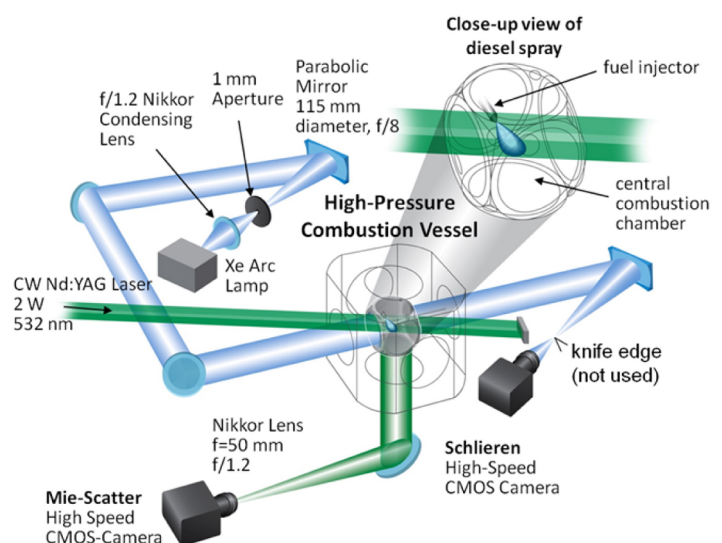


Figure 3. Combustion vessel and high-speed imaging setup (not to scale).

Diesel conditions are simulated by using a premixed, combustible-gases mixture, which is spark-ignited to create a high-temperature, high-pressure environment in the vessel. Following the premixed combustion, the combustion products cool over a relatively long time ($\sim 1\text{--}3$ s) due to heat transfer to the vessel walls and the vessel pressure slowly decreases. When the desired pressure and temperature are reached, the diesel fuel injector is triggered and fuel injection occurs.

The temperature, density, and composition of the ambient gas in the vessel at the time of diesel fuel injection can be varied widely with this simulation procedure. The conditions of the ambient gas at the start of injection are determined by the initial mass and composition of gas within the vessel and the pressure at the time of fuel injection. All Rayleigh and schlieren imaging experiments were performed in an inert environment of 0% O_2 , to evaluate the effects of mixing and spray vaporization without the complication of combustion. Table 1 shows the ambient gas composition and range of conditions that were evaluated in this study, which included several different ambient gas densities and temperatures. Temperature measurements were performed by fine-wire thermometry to establish a relationship between pressure and the temperature during the cool down. The gas temperature prior to injection was uniform to within 1% along the axial direction [35].

Table 1. Ambient gas conditions for this study.

%vol O_2	0.00
%vol CO_2	6.52
%vol N_2	89.71
%vol H_2O	3.77
Molec. Weight MW	28.68
Ambient temperature range	900 – 1400 K
Ambient density range	7.6 – 22.8 kg/m^3
Vessel pressure	3.1 – 6.0 MPa

INJECTION EQUIPMENT

Two different common-rail injectors were used in this study, each with a single-spray, axially-drilled, nozzle. The first is the same injection system that was used in our facility from 1998 [8] to 2007 [25]. The fuel injector was a prototype, electronically controlled, common-rail, solenoid-activated injector designed by Detroit Diesel Corporation. Table 2 shows the specifications for the nozzle used in this study (see DDC). The 0.100 mm orifice is sharp-edged, non-hydroground and designed without taper from the inlet to the exit of the nozzle ($K = (d_i - d_o)/10 \approx 0$, units in μm). The second injector is a Bosch second-generation common-rail injector (CRIP 2.4). This nozzle has a convergent ($K = 1.5$), hydro-ground orifice (KS1.5/86 specification by Bosch) and is a mini-sac type with a sac volume of 0.2 mm. This injector

is now being used for the “Spray A” condition data set of the Engine Combustion Network [26].

Table 2. Injector conditions. Two different injectors were used, with injection pressure and fuel variation.

Inject.	Orifice dia. [mm]	C_d	C_a	K	Fuel	Fuel T^* [K]	P_{inj} [MPa]
DDC	0.100	0.80 ^s	0.86 ^s	0	C7	373	154
Bosch	0.090	0.86	0.95	1.5	C12	363	150
Bosch	0.090	0.86	0.95	1.5	C12	363	100
Bosch	0.090	0.86	0.95	1.5	C12	363	50
Bosch	0.090	0.86 ^t	0.95 ^t	1.5	C7	363	150

^sMeasured for diesel fuel at 300 K [8].

^tassumed to be the same as that measured for n-dodecane

*Calculated fuel density at this temperature and at 1 atm:

n-heptane (C7, 373 K), 613 kg/m^3 ; n-dodecane (C12, 363 K), 699 kg/m^3 ; n-heptane (C7, 363 K), 623 kg/m^3

Figure 4 presents rate-of-injection (ROI) measurements that correspond to the injectors used in the study. ROI measurements were performed using a force sensor, and cumulative mass measurements, with the injector installed in the spray chamber. The method was similar to that described in [9]. The injection profiles for either injector have a rapid start and end of injection, with a steady period in between, forming a top-hat profile. For the 0.100-mm nozzle (DDC injector), we have used the ROI profile measured for this nozzle at a similar condition using diesel fuel, and then applied an appropriate scale factor for n-heptane fuel, which has a lower fuel density. This is necessary because no ROI measurements were made with n-heptane fuel. The ROI measurements for the 0.090-mm nozzle (Bosch injector) are at the stated conditions.

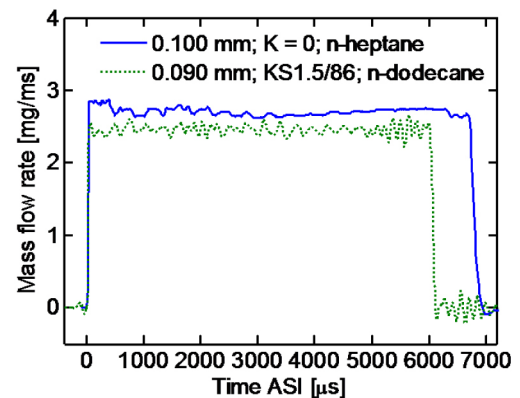


Figure 4. Mass flow rate of injection with 150 MPa fuel injection pressure and 0.1 MPa ambient pressure. The 0.100-mm flow rate is scaled for n-heptane fuel.

The steady fuel injection rate permits use of the analytical solutions for mixing and penetration developed in the

previous section for either uniform-profile or variable-profile models. If the rate of injection had not been steady, or if analysis after the end of injection was desired, the control-volume numerical method of Ref [4] could be utilized. This numerical solution also matches the analytic solution when the flow rate is steady, indicating accurate numerical treatment [4]. But it is unnecessary to use the finite-volume approach in this study because the rate of injection is steady within experimental uncertainty.

The discharge (C_d) and area contraction (C_a) coefficients for the injectors/orifices are shown in Table 2. The orifice coefficients were determined from long-injection events, where the flow is relatively steady, similar to those shown in Fig. 3. Note that the shaped hydroground orifice (0.090-mm) has higher flow coefficients compared to the non-hydroground orifice. While the ROI for the 0.090-mm orifice is expected to be lower than the 0.100-mm orifice, the measured flow rate is only slightly lower because of higher flow coefficients.

OPTICAL DIAGNOSTICS

High-speed imaging systems were utilized to simultaneously track the liquid-phase spray and vapor-phase jet. Rayleigh-scatter imaging was used for a quantitative measurement of mixing in the vapor-phase region. The setup for these diagnostics is discussed below.

MIE-SCATTER

Scattered light from fuel-spray liquid droplets was imaged using a high-speed CMOS camera and a continuous-wave laser. As shown in Fig. 3, a schlieren system was operated simultaneously using a second high-speed CMOS camera, similar to the setup reported in [28]. The time-resolved pair of schlieren and Mie-scatter images identifies the instantaneous position of both the vapor jet and liquid spray, respectively.

The continuous-wave laser beam was expanded to overfill and completely illuminate the liquid-phase spray. This “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. The input beam was directed at a slight angle to avoid interference with the schlieren setup. 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, imaged Mie-scattered light at a near-right angle to the laser source.

SCHLIEREN

Schlieren imaging was performed simultaneously using a second lighting and imaging arrangement. Light from a mercury-xenon arc lamp was collimated to pass through the

combustion vessel. The collimated beam was then re-focused before entering a high-speed CMOS camera equipped with a lens and a 532-nm 0°-incidence laser mirror to block the laser light elastically scattered by spray droplets.

Previous studies utilized several different schlieren stops at the collection focal point shown in Fig. 3, including a knife edge, aperture (bright-field), cutoff disk (dark-field), or no stop [28]. The sensitivity of the system to refractive index gradients can be adjusted by the use of different cutoffs [29]. However, gradients of refractive index caused by heat transfer to the cooler vessel wall, which forms temperature gradients in boundary layers near the walls, create schlieren effects that appear in the background of the images [35]. Note that potential disturbances are possible at two window-ambient interfaces, along with an 108-mm path (the vessel dimensions) through high-temperature ambient gases. To illustrate this effect, we show a raw schlieren image with a computer-algorithm-determined jet boundary in Fig. 5. The disturbances outside of the jet in Fig. 5 show significant light-dark fluctuations with a relatively small length scale. Inside of the jet, the texture of the image appears smoother. The significant schlieren disturbances that persist outside of the jet clearly make it difficult to determine the true vapor boundary, thereby causing uncertainty in the jet penetration rate and spreading angle. If the schlieren sensitivity is too high, the background disturbances of the ambient will appear too prominent in the images, making image processing for detection of the jet boundary impossible. On the other hand, if the schlieren sensitivity is too low, the vapor border of the jet may not be detected. In addition, these schlieren disturbances change with ambient density. Therefore, several different schlieren setups were investigated.

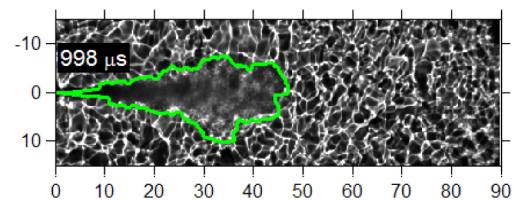


Figure 5. Example schlieren image with processed vapor boundary shown in green. High-sensitivity schlieren setup, 22.8 kg/m³, 6 MPa.

Table 3 shows the imaging parameters for a low-sensitivity setup and a high-sensitivity setup. In either setup, we found adequate sensitivity to refractive index gradients when using no schlieren stop other than the camera aperture. The high-sensitivity setup utilizes a camera lens with 105-mm focal length and set for infinite focus. This setup provides higher pixel resolution at the expense of a slower framing frequency. The low-sensitivity setup uses a 50-mm focal length lens set for 3 m focus, rather than infinite focus. As will be shown, de-focusing the lens and using lower pixel resolution tends to

smear schlieren disturbances from the ambient, making image processing easier. The lower pixel resolution also allows faster camera framing rates to capture the penetration of the jet with higher time resolution.

Table 3. High-speed CMOS camera schlieren imaging setup.

	high-sensitivity setup	lower-sensitivity setup
Lens focal length	105 mm	50 mm
Lens focal setting	∞	3 m
Lens f/#	2.8	1.2
Pixel resolution	512 x 256	256 x 128
Spatial resolution	5.4 pixel/mm	2.7 pixel/mm
Frame period	66.5 μ s	21.2 μ s
Exp. period	3 μ s	2 μ s

In addition to schlieren imaging setup variation, changes to background schlieren disturbances were accomplished by adjusting the vessel mixing fan speed, or by turning the fan completely off. Adjustments to the fan speed tend to change the length scale of the structures in the background [28], potentially making it easier to detect the jet.

RAYLEIGH-SCATTER IMAGING

Quantitative mixing measurements in vaporizing fuel sprays were obtained using Rayleigh-scatter imaging. Most of the experimental approach to Rayleigh-scatter imaging in our facility is covered in Ref. [25]. As only a few modifications to this setup were made for this study, an abbreviated summary of the experimental approach is included here with special attention to the new setup. By avoiding scatter from liquid-phase droplets, Rayleigh-scatter imaging provides a measurement of the mixture fraction within the vaporized jet.

Optical setup

The laser light source for Rayleigh-scatter imaging was the second harmonic of an Nd:YAG laser (532 nm) with pulse energy of 150 mJ and pulse duration of 7 ns. The laser beam was formed into a collimated planar sheet 40 mm wide and 300 μ m thick and passed through the fuel jet center. The laser sheet spanned axial distances from approximately 17 mm to 57 mm. The signal was filtered with a 532-nm narrow bandpass filter (10 run FWHM) and imaged using a high-quantum-efficiency, back-illuminated CCD camera (PIXIS 1024B). Two different imaging setups were utilized, with a comparison of the two setups shown in Table 4. The setup changed between experiments using the DDC injector [25] and newer experiments using the Bosch injector. A more detailed discussion of the theory of Rayleigh scattering and the setup changes is included in Appendix C.

Table 4. Camera and lens configurations for Rayleigh-scatter imaging.

Injector orifice setup	0.100-mm [25]	0.090-mm
PIXIS Camera	1024B-UV	1024B
Q.E. at 532 nm	66%	95%
Gain	1 [cts/e-]	0.25 [cts/e-]
Lens focal length	105 mm	85 mm
Lens f/#	f/1.8	f/1.4
Close-up	10 mm extens.	500 D lens
Pixel resolution	20.09 pix/mm	14.06 pix/mm
Laser trans. windows	Sapphire	Fused-silica slits

Liquid penetration length control

Mie-scatter interferences from liquid-phase droplets can be avoided by positioning the laser sheet downstream of the maximum liquid-phase penetration distance, or the liquid length (see Fig. C1 in Appendix C). However, the liquid length changes depending upon the fuel and operating condition [9]. For example, variation of ambient density is one of the primary objectives of this study, and, as shown in Fig. 6, the liquid length tends to increase with decreasing ambient density. Therefore, avoiding interference from liquid droplets by the Rayleigh diagnostic becomes more problematic. One potential solution is to change the laser sheet position for each density, i.e., to move the laser sheet downstream at lower density. However, this limits the range of data that can be used for calibration of the jet mixing model and it requires a new, repeatable setup for each condition. Another approach is to raise the ambient temperature when decreasing ambient density to prevent liquid droplets from entering the laser sheet interrogation region. For example, Fig. 6 shows that in order to maintain an n-dodecane liquid length of approximately 11 mm, far enough away from the laser sheet at 17 mm, the ambient temperature must be increased from 900 K at 22.8 kg/m³, to above 1400 K at 7.6 kg/m³.

If droplets mix perfectly with the ambient (no velocity slip), the momentum transfer between the injected fuel and the ambient gases depends only upon ambient density, rather than ambient temperature. In fact, there is no temperature input in the uniform- or variable-profile mixing models. Accordingly, we adopted the approach of increasing ambient temperature when decreasing ambient density to keep the liquid length essentially constant.

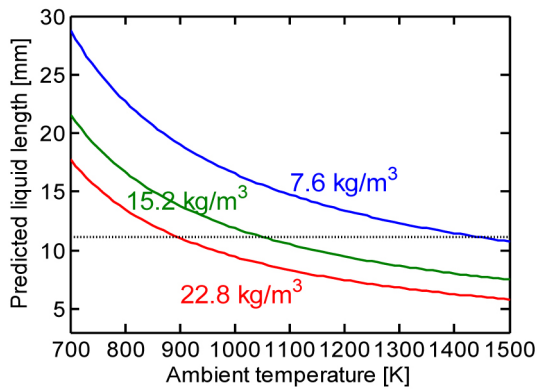


Figure 6. Predicted liquid penetration lengths for *n*-dodecane as a function of ambient density. Model of Siebers [9].

Increasing ambient temperature can adversely affect the Rayleigh-scatter signal, however, if the elevated temperature causes the fuel molecules to decompose. Analysis in [Appendix C](#) shows that fuel decomposition is a concern when the ambient gas temperature is 1400 K. Despite the likelihood for fuel decomposition when the ambient temperature is 1400 K, we performed Rayleigh scattering experiments at this condition to address the effect of lower ambient density (7.6 kg/m³) on mixing. Although fuel decomposition can significantly reduce signal (limiting case is a factor of 2.4, see [Appendix C](#)), the signal is not completely eliminated and Rayleigh scattering could be used to assess such things as the vapor boundary of the jet in connection to that determined by schlieren imaging, for example. We will return to this issue and address fuel-decomposition along with its potential influence on our quantitative mixing measurements later in the results.

Uncertainty analysis

As the purpose of this paper is to compare experimental mixing measurements to the jet mixing model predictions, the ensemble-average of multiple injections will be provided. The uncertainty for the mean of a dataset depends upon the number of data points, the fluctuation (standard deviation), and any bias errors that remain uncorrected. We follow the standard uncertainty analysis method outlined by Moffat [39], where these error terms are combined and uncertainties expressed to a 95% confidence level.

Systematic errors and uncertainties in the Rayleigh diagnostic have been described in Ref. [25] or in [Appendix C](#), along with methods to minimize these errors. However, a major source of uncertainty for this dataset is the limited number of Rayleigh-image realizations at a particular condition. At most, only 40 images were used to calculate the ensemble average, and in some cases, as few as 20. The number of obtained images is limited by the low repetition rate of injection events for the facility, coupled with the fact that

many images were not processed if particle contamination was too severe. A higher uncertainty results when there are few images. We will be careful to present experimental results including uncertainties to qualify how well results match the jet mixing model.

RESULTS

The jet vapor-boundary detection by high-speed schlieren imaging, and the derived jet penetration and spreading angle are shown first. We compare these measurements to penetration predictions by the simplified variable-profile model ([Fig. 1](#)) and provide an initial assessment of methodology for calibration of the model from global spray/jet measurements. We then compare the measured spreading angle to that determined by Rayleigh-scatter imaging by quantitative comparison of Rayleigh mixing measurements and jet model predictions.

VAPOR BOUNDARY

Examples of schlieren images taken at 3 ms after start of injection (ASI) are shown in [Figs. 7,8,9](#) for a range of conditions. The schlieren images are shown with computer-processed boundaries in green to indicate the vapor border of the jet. The boundary is determined by analysis of the difference between successive images ($I_n - I_{n-1}$), which are shown at the bottom of each image pair (raw schlieren at top, I_n). This method for image processing and display attempts to remove schlieren effects that exist in the ambient gases as discussed above. The significance of these background disturbances is clearly manifest. [Figure 7](#) shows how these background disturbances can be changed by switching between an optical setup with high- (top) or low-sensitivity (bottom). [Figure 9](#) shows that the disturbances increase with increasing ambient density, as expected because gradients of refractive index scale with density. [Figure 8](#) shows that the length scale of these disturbances can be increased by turning the mixing fan for the vessel completely off, which can be useful for determination of whether structures originate from the jet or from the ambient [28].