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# Effects of Gas Density and Vaporization on Penetration and Dispersion of Diesel Sprays

Jeffrey D. Naber\* and Dennis L. Siebers Sandia National Labs.

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# Effects of Gas Density and Vaporization on Penetration and Dispersion of Diesel Sprays

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# ABSTRACT

Ambient gas density and fuel vaporization effects on the penetration and dispersion of diesel sprays were examined over a gas density range spanning nearly two orders of magnitude. This range included gas densities more than a factor of two higher than top-dead-center conditions in current technology heavy-duty diesel engines.

The results show that ambient gas density has a significantly larger effect on spray penetration and a smaller effect on spray dispersion than has been previously reported. The increased dependence of penetration on gas density is shown to be the result of gas density effects on dispersion. In addition, the results show that vaporization decreases penetration and dispersion by as much as 20% relative to non-vaporizing sprays; however, the effects of vaporization decrease with increasing gas density.

Characteristic penetration time and length scales are presented that include a dispersion term that accounts for the increased dependence of penetration on ambient density. These penetration time and length scales collapse the penetration data obtained over the entire range of conditions examined in the experiment into two distinct non-dimensional penetration curves: one for the non-vaporizing conditions and one for the vaporizing conditions. Comparison of the two non-dimensional penetration curves to a theoretical penetration correlation for non-vaporizing sprays helped isolate and explain the effects of droplets and vaporization on penetration. The theoretical penetration correlation was derived using the penetration time and length scales and a simple model for a non-vaporizing spray that has been previously presented in the literature. The correlation is in good agreement with the non-vaporizing data from this experiment and other commonly quoted penetration data sets. It also provides a potential explanation for much of scatter in the penetration predicted by various correlations in the literature.

#### **INTRODUCTION**

Meeting stringent new emissions regulations while maintaining or improving the efficiency of diesel engines is a difficult challenge for diesel engine manufacturers. Some of the design changes used to meet emissions regulations, such as retarding injection timing, result in less efficient engines. To compensate for these efficiency losses and to improve engine performance and power density, engine manufacturers have boosted in-cylinder gas densities through greater use of turbocharging. This approach offers the potential for still greater improvements in diesel engine performance and emissions. However, little is known about the effects of gas densities higher than those in current technology diesel engines on injection, combustion and emissions processes.

Presented in this paper are results from an experiment examining the effects of ambient (*i.e.*, in-cylinder) gas density on the injection, vaporization, and combustion processes in diesel engines with an emphasis on the effects of gas density higher than in current technology engines. The overall goals of the experiment are to determine the effects of gas density on the global development (*e.g.*, the penetration, dispersion, ignition and combustion) and the evolution of the detailed structure (*e.g.*, the liquid, vapor and combustion zones) of diesel sprays. The specific objectives of this paper are to present and discuss results on the effects of ambient gas density and fuel vaporization on the penetration and dispersion of diesel sprays.

The effects of ambient gas density and fuel vaporization on spray penetration and dispersion were explored by comparing injections into non-vaporizing (*i.e.*, ambient temperature) and vaporizing (*i.e.*, high-temperature: 600 K to 1400 K) inert environments over extremely large ranges of ambient gas density. The ambient gas density ranges considered were 3 to 200 kg/m<sup>3</sup> and 3 to 61 kg/m<sup>3</sup> for the nonvaporizing and vaporizing environments, respectively. These density ranges greatly exceed the present data base. The range for the high temperature vaporizing studies covers densities from just above atmospheric to densities more than a

<sup>\*</sup> Now at Motorola Inc., Dearborn, Michigan.

factor of two higher than the 10 to 30  $\text{kg/m}^3$  densities in current technology heavy-duty diesel engines at top-dead-center (TDC).

The experiments were carried out in a constant-volume combustion vessel with line-of-sight optical access using an electronically controlled, common rail diesel fuel injector and a #2 diesel fuel. Injection pressure and orifice diameter were varied from 75 to 160 MPa and 0.19 to 0.30 mm, respectively. Transient spray penetration and dispersion data were obtained by digitization and analysis of time resolved extinction/schlieren movies of the sprays. Additional penetration data were obtained with a high-speed, linear array CCD (linescan) camera. A similar range of experiments was conducted under combusting conditions and will be presented in a follow up paper.

The remainder of the paper is divided up into several sections and appendices. The first section presents a brief review of diesel spray penetration and dispersion that focuses on the effects of ambient gas density and vaporization. The next two sections of the paper discuss the experimental apparatus, procedure and conditions, respectively. The fourth section discusses the image analysis techniques used to obtain the spray penetration and dispersion from the film and linescan cameras. The last two sections present the results and discussion and a summary. The appendices contain details on measurement techniques, the experimental conditions and a derivation of the spray penetration theory used in the Results and Discussion section. Also included after the summary is a nomenclature

#### BACKGROUND

SPRAY PENETRATION — Diesel spray penetration investigations began as early as the 1920s with the majority of the work being conducted at room temperature conditions (*i.e.*, non-vaporizing conditions). A 1972 review by Hay and Jones [1] found that the dependence of penetration on ambient gas density, as well as other parameters, differed significantly from investigation to investigation. The range for the penetration dependence on ambient gas density ( $\rho_a$ ) that they noted was  $\rho_a^{-0.25}$  to  $\rho_a^{-0.5}$ . Since that review several other notable investigations of penetration have appeared in the literature [e.g., 2-6]; however, the overall picture has not changed. The range for the penetration dependence on ambient gas density reported in the more recent investigations is  $\rho_a^{-0.23}$  to  $\rho_a^{-0.5}$  with one exception. That exception occurs early during the injection period, where Hiroyasu and coworkers [2,3] have shown that penetration depends linearly on time, but has no dependence on gas density.

The conditions covered in the investigations cited above are heavily weighted to ambient gas densities less than 25 kg/m<sup>3</sup>, orifice diameters from 0.2 to 0.7 mm, and injection pressures less than 75 MPa. Exceptions are the investigations of Hiroyasu and coworkers [2,3] and Varde and Popa [4]. Hiroyasu and coworkers examined ambient gas densities up to  $33kg/m^3$  and orifice diameters as small as 0.1 mm, but for injection pressures less than 40 MPa. Varde and Popa examined gas densities up to  $40kg/m^3$  with injection pressures from 50 to 150 MPa. The effects of vaporization on spray penetration have not been explored in a systematic manner for typical TDC incylinder temperatures (*i.e.*, 1000 K). Very limited results presented by Hiroyasu and Arai [2] and Kamimoto *et al.* [71] indicate that there is little to no effect of vaporization processes on spray penetration. On the other hand, the penetration correlation of Dent [8] contains a temperature term that predicts a significant reduction in spray penetration for injection into elevated temperatures (for both combusting and noncombusting conditions). Similarly, a correlation by Parks *et al.* [9] contains a temperature term that predicts a reduction in penetration at elevated temperatures, but the correlation gives unrealistically low values for reasonable diesel TDC temperatures and orifice sizes.

In summary, there is little agreement in the literature regarding the effects of ambient gas density and vaporization on spray penetration, especially for conditions relevant to current and future technology engines (i.e., in-cylinder TDC gas densities greater than 25kg/m<sup>3</sup> at a temperature of 1000 K coupled with injection pressure in excess of 100 MPa). The most widely cited penetration data and correlations in text books [e.g., 6,10-12] are those of Hiroyasu and coworkers [2,3] and Dent [8], both of which have a density dependence of  $\rho_a$ .<sup>-0.25</sup> However, recommendations made by Hay and Jones [1] in their review suggest that correlations with a gas density dependence of  $\rho_a$ .<sup>-0.25</sup> will not accurately predict penetration over the entire load range of an engine. Hay and Jones suggested that a correlation presented in graphical form by Wakuri et al. [13] with a gas density dependence of  $\rho_a$ .<sup>0.45</sup> provided more reasonable penetration values over the entire load range. Relating the present uncertainty in the dependence of penetration on gas density to a turbocharged diesel engine in which the TDC gas density varies by a factor of 3 as the load changes, translates to a 35% uncertainty regarding spray location and an 80% uncertainty regarding spray timing with load.\* These uncertainties increase when advanced engines with greater turbocharging are considered.

SPRAY DISPERSION — A common measure of spray dispersion is the cone angle of the outer boundary of a spray, referred to in this paper as the spray dispersion angle. Most investigations of spray dispersion angles at ambient gas densities typical of diesel engine conditions have focused on nonvaporizing sprays. One of the more complete investigations of non-vaporizing spray dispersion angles was presented by Hiroyasu and Arai [2]. The conditions they examined included ambient gas densities up to 30 kg/m<sup>3</sup> and injection pressures up to 80 MPa. They identified a "complete spray' regime in which the spray dispersion angle is mainly dependent on orifice parameters and the ratio of the ambient gas and fuel densities  $(\rho_a/\rho_f)$ . For a fuel with properties typical of #2 diesel fuel, the complete spray regime occurs when injection velocities are greater than 100 m/s (i.e., an injection pressure difference greater than 8 MPa). Injection velocities and pressures in current technology heavy-duty diesel engines are much greater than these, and thus, are well into the complete spray regime. The dependence of the spray dispersion

<sup>\*</sup> The timing uncertainty estimate assumes  $t \propto S^2$ .

angle on arribient gas density found by Hiroyasu and Arai in the complete spray region was  $\rho_a^{0.26}$ .

Two other investigations of non-vaporizing spray dispersion at high injection pressure conditions were those of Varde *et al.* [14] and Wakuri *et al.* [13]. Varde *et al.* reported a dispersion angle dependence on ambient gas density of  $\rho_a^{0.33}$ for gas densities up to 40 kg/m<sup>3</sup> and injection pressures between 50 and 150 MPa. Wakuri *et al.* found a dispersion angle dependence on ambient gas density of  $\rho_a^{0.4}$  for densities less than 22 kg/m<sup>3</sup> and injection pressures between 40 and 75 MPa.

Reitz and Bracco [15] and Wu *et al.* [16] also investigated dispersion of non-vaporizing sprays, but in a region much closer to the injector than others (within ~5 mm of the injector tip). Their research was conducted for conditions under which complete atomization of the fuel spray occurred in the immediate vicinity of the orifice. These conditions correspond to the atomization regime of spray breakup (*e.g.*, see Ref. 11). Similar to the trends observed by Hiroyasu and Arai [2], Reitz and Bracco and Wu *et al.* show that orifice design parameters and the ratio of the ambient gas and fuel densities are the major factors influencing spray dispersion. However, they reported a larger dispersion angle dependence on gas density ( $\rho_a^{0.5}$ ), as well as angles smaller than those measured further from the orifice by others [2-14] for the same density ratio.

Spray dispersion under vaporizing conditions has not been extensively explored. Susuki *et al.* [17] presented data and spray images that indicate there might be an increase in the spray dispersion angle as a result of fuel vaporization at an ambient density of  $12.5 \text{kg/m}^3$  and a temperature of 773 K. However, their conclusions were based on measurements in the transient head of the spray. In other experiments by Yule *et al.* [18] and Fujimoto *et al.* [19], the ambient gas temperature was varied from room temperature to temperatures as high as 900 K, but at a fixed ambient pressure. Their results, therefore, contained the combined effects of ambient gas density and vaporization on dispersion that could not be separated.

Summarizing, as with diesel spray penetration, there is significant scatter in the reported dispersion angle dependence on ambient gas density, and little understanding of the effects of vaporization on dispersion. Reported non-vaporizing spray dispersion angle dependencies on ambient gas density vary from  $\rho_a^{0.25}$  to  $\rho_a^{0.5}$ . Since the dispersion an gle is related to the air entrainment rate of a spray, this uncertainty translates to significant uncertainty in the understanding of the effects of spray dispersion on diesel combustion processes.

#### **EXPERIMENTAL APPARATUS**

The experimental apparatus consists of a constant-volume combustion vessel, a high-pressure fuel injection system, a schlieren imaging system, and data acquisition and control systems. The apparatus is located in a high pressure test cell and is controlled and operated remotely by a computer.

COMBUSITON VESSEL — Figure 1 shows schematic cross-sections of the constant-volume combustion vessel. The vessel has a disk-shaped combustion chamber 114.3 mm in

diameter and 28.6 mm in width. Sapphire windows on either side permit full line-of-sight optical access through the vessel. The walls of the vessel can be electrically heated to temperatures up to 525 K to reproduce engine wall temperatures and prevent water condensation on the window surfaces. Around the circumference of the vessel are five access ports and two injector ports. This vessel is similar to one used previously for diesel combustion studies [20], except that the new vessel is designed for significantly higher pressures (up to 35 MPa).



Figure 1. Schematic cross-sections of the combustion vessel.

Mounted in the five access ports in the positions shown in Fig. 1 are: two solenoid controlled air-operated valves, one for intake and the other for exhaust; a surface gap spark plug; a pressure transducer; and a mixing fan. (The functions of the spark and combustion vessel mixing fan are discussed in detail later.) The pressure transducer used was a Kistler model K-6001 piezoelectric pressure transducer coupled to a Kistler model 504E charge amplifier with a 20 kHz low pass filter. A perforated metal thermal barrier covering the pressure transducer delayed the effects of the thermal pulses from combustion events on the pressure measurements for approximately two seconds. (A typical experiment discussed in the Experimental Procedure and Conditions Section required less than 1.5 seconds to complete.) The pressure transducer was routinely calibrated up to the full pressure of the experiments against a reference transducer with an NBS traceable calibration. The calibrations were conducted with the transducer mounted in place at the operating temperature of the experiment. The accuracy of the calibration is  $\pm 1$  %.

FUEL INJECTION SYSTEM - The fuel injection sys tem consists of a high-pressure liquid pump, accumulators, and an injector. The fuel injector was an electronically controlled, common-rail, solenoid-activated injector designed by Diesel Technology Corporation. It was placed in the left injector port (as shown in Fig. 1) with a single hole orifice oriented to inject fuel through the center of the chamber. Schematics of the injector and injector tip are shown in Fig. 2. The main features of the injector are the solenoid, the control needle, the main needle, and the fuel supply and return lines.

The enlargement on the top left in Fig. 2 can be used to describe the operation of the injector. Activation of the solenoid lifts the control needle, opening orifice **A** directly above



**Figure 2.** Schematics of the common-rail fuel injector and injector tip with 34° of axis orifice.

the main needle. This action allows fuel flow from the supply line through orifice **B** into the small chamber above the main needle, then through orifice **A** to the return line. Since orifice **B** is smaller than **A**, the pressure in the small chamber is much less than the line pressure, creating a force imbalance that unseats the main needle and starts the injection. When the solenoid is deactivated a closure spring reseats the control needle. The pressure in the chamber above the control needle returns to the fuel supply pressure causing the main needle to reseat, ending injection.

Fuel is supplied to the injector from an accumulator located immediately upstream of the injector. The accumulator was sized to limit the pressure loss during injection to less than 2% for injections of up to 100 mg of fuel. The accumulator can be pressurized to 350 MPa by a high-pressure liquid pump; however, pressures were limited to less than 180 MPa by the injector design. The fuel pressure during injection was measured with a Kistler model 607L piezoelectric pressure transducer coupled to a Kistler model 504E charge amplifier with a 60 kHz low pass filter. The transducer was located between the injector and the accumulator. Since the tip orifice area is the most significant flow restriction between the accumulator and the injector tip (by an area ratio of 30:1 for the largest orifice diameter used), the pressure measured at this location is equal to the tip pressure. The pressure transducer was calibrated to the full pressure of the experiments against a reference transducer with an NBS traceable calibration. Calibrations were done with the transducer mounted in place at the operating temperature of the experiment. The accuracy of the calibration is  $\pm 1\%$ .

Fuel injection timing was measured optically via a photointerrupt technique. During injection, the fuel spray blocked a laser beam directed over the injector orifice and onto a photodiode. The signal generated by the photodiode allowed the detection of the start and end of injection.

A Phillips research grade diesel fuel was used for all the experiments. Available properties are given in Table 1.

Table 1. Phillips Research Grade D-2 Fuel Prope	rties.
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Density. Viscosit Cetane N Distillati	Density								
IBP	10%	30%	50%	70%	90%	EBP			
468 K	499 K	518 K	534 K	550 K	577 K	604 K			

SCHLIEREN IMAGING SYSTEM — The schlieren imaging system is shown schematically in Fig. 3. The light source was an argon-ion laser. The laser beam (488 nm) was focused through a pinhole for spatial filtering and expansion, and then collimated by a lens (150 mm  $\phi$ , 1220 mm focal length) to a diameter larger than the vessel. The portion of the beam passing through the combustion vessel (114 mm  $\phi$ ) was refocused by a second lens (150 mm.  $\phi$ , 1220 mm focal length) located one focal length downstream of the vessel. The iris of the imaging lens on the camera was positioned at the downstream focal point of the second lens and functioned as the schlieren stop, except for some of the highest density conditions (discussed later).

Images created with this optical system for the noncombusting conditions reported in this paper were the result of laser light removed from the optical path by two primary effects: a schlieren effect and extinction. The schlieren effect was caused by temperature and fuel concentration gradients (*i.e.*, index of refraction gradients) in the spray, while extinction was primarily the result of scattering from the liquid in the spray.



Figure 3. Schematic of the imaging system setup. The cam era was either a high speed film camera (14,500 frames/s) or a linescan camera (80 kHz line readout rate).

Either of two cameras were used to record the images, a high speed film camera or a linear array CCD camera. The film camera was a 16 mm Hycam camera with a 1/2 frame prism and a 75 mm,f/1.8 imaging lens. The frame rate was nominally 14,500 frames/second, which when coupled with a 1/2.5 shutter, gave an exposure duration of ~28 µs. The film used was Kodak 7250 positive color film. One kilohertz exposure timing marks placed on the film by the camera enabled the interframe timing to be determined to an accuracy better than ±1%. Film exposure was controlled by adjusting the laser power and neutral density filters at the filter location shown in Fig. 3, while the schlieren sensitivity was controlled with the iris of the imaging lens on the camera (*i.e.*, the *f*-stop).

The schlieren sensitivity was set for each experiment to maximize the schlieren signal-to-noise ratio for vaporizing conditions (*i.e.*, the ratio of the spray to the vessel wall boundary layer induced schlieren signals). However, for the highest density conditions of the experiment, the internal f-stop, of the high speed film camera (f-stop=3.3) limited the schlieren sensitivity adjustment. To determine if this affected the results, an intermediate image was formed on ground glass and then reimaged by a secondary lens onto the camera film plane for some high density conditions. This setup eliminated the camera f-stop limit for the schlieren sensitivity adjustment; however, the image quality was reduced. Data obtained with this technique are included in the results, but no differences between indirect and the direct imaging results were noted.

The linear array CCD (linescan) camera used was a Dalsa 128 pixel linear array camera running at a line rate of 80 kHz (12  $\mu$ s) coupled with an 8 bit, 4 MB video digitization and storage board. The linescan camera was positioned at the same location as the film camera with the array aligned along the spray axis to record the penetration in a manner similar to Ahmadi-Befrui *et al.* [21] and Winklhofer *et al.* [22]. The camera was triggered by the TTL pulse controlling the fuel injector.

The spatial resolution for the linescan camera depended on the field of view. The spatial resolution was approximately one millimeter when viewing the entire diameter of the vessel. For reduced fields of view, the spatial resolution improved. The high temporal resolution of this camera relative to the film camera provided a better characterization of the early spray tip penetration. Also, its solid state nature allowed the repeatability of the penetration data to be evaluated more thoroughly.

DATA ACQUISITION AND CONTROL SYSTEMS — Two networked PC's were used to operate the experiment. The main computer operated the experiment hardware (valves, injector, fan, spark, film camera, *etc.*) and acquired, stored and reduced the analog signals. The second computer acquired, stored and displayed the linescan camera images on command from the main computer.

The experiment hardware was controlled via a combination of a 20 channel 1 MHz counter/timer board, a 24 bit digital I/O board, 4 D/A channels, and a solid state relay panel. All steady system pressures, temperatures, etc. (*e.g.*, the gas supply pressures, the fuel supply pressure, the wall temperature) were acquired with a multiplexed 16 channel, single-ended, 12 bit, 70 kHz A/D board. All transient high speed data during an experiment (other than images) were acquired with an 8 channel differential, 12 bit, 250 kHz A/D board. The high speed data recorded for each experiment included the combustion vessel gas pressure, the injector fuel pressure, the injector needle lift, and the start and end of injection via the photo-interrupt. These data were sampled at a 42 kHz per channel rate during the diesel injection event. Slower sampling rates were used before and after the injection event for obtaining initial and final values for each parameter.

#### EXPERIMENTAL PROCEDURE AND CONDITIONS

NON-VAPORIZING SPRAY EXPERIMENTS — For the "non-vaporizing" experiments, the combustion vessel was filled to a pressure corresponding to the desired density with nitrogen, or for three experiments, with a helium/nitrogen (60%/40%) mixture. The fill was done under well stirred conditions (with the vessel mixing fan running) to ensure thermal equilibrium of the gas with the wall. Once the vessel was at the selected conditions, the fuel injection and associated data acquisition were initiated.

The primary parameter varied was the density of the ambient gas in the vessel. The density range considered was 3 to 200 kg/m<sup>3</sup>. Other parameters varied for selected ambient gas density conditions were the tip orifice diameter and the injection pressure. The orifice diameter and the injection pressure ranges covered were 0.198 mm to 0.340 mm and 75 MPa to 160 MPa, respectively. (The injector tips will be discussed in more detail later.)

The ambient gas temperature (*i.e.*, wall temperature) for the "non-vaporizing" experiments was either 300 K or 450 K. Some fuel vaporization does occur at these temperatures, especially at 450 K; however, the rate of vaporization is slow relative to the rate of penetration. The lifetime of an <u>isolated</u> 20  $\mu$ m fuel drop at 450 K was estimated to be ten milliseconds. This lifetime is long compared to penetration times of interest (~1 ms), indicating that the amount of fuel vaporization was small.

The three experiments in the helium/nitrogen environment were conducted to determine if the difference in ambient pressure between the "cold" non-vaporizing experiments in nitrogen and the "hot" vaporizing experiments at the same ambient density had any significant impact on the injector operation or the data. The helium/nitrogen mixture allowed both the ambient pressure and density of a "hot" vaporizing experiment to be simulated at the lower temperatures of the non-vaporizing experiments. No differences were found in the injector performance or the fuel spray using the helium/nitrogen environment. These data are included with the data from the pure nitrogen environment.

VAPORIZING SPRAY EXPERIMENTS — For the vaporizing experiments, an inert high-temperature gas was generated in the combustion vessel. As with the non-vaporizing experiments, the primary parameter varied was the gas density. Densities from 3 to 60 kg/m<sup>3</sup> at an ambient temperature of 1000 K were considered. The other parameters varied for the higher density conditions were the ambient temperature, the orifice diameter and the injection pressure. The ambient temperature range covered was 600 K to 1400 K. The orifice

diameter and injection pressure ranges were the same as for the non-vaporizing experiments.

The method used to generate the high temperature inert gas in the combustion vessel was the same as that previously used by Siebers [20], Oren *et al.* [23] and Durrett *et al.* [24] to simulate in-cylinder diesel engine temperatures and pressures near TDC. This method involves spark ignition and combustion of a premixed combustible gas mixture in the vessel. When the products of this combustion event cool to a preselected pressure and temperature state, the diesel fuel is injected.

The following relationship gives the reactant and product compositions of the premixed combustible gas mixture used to generate the high-pressure, high-temperature environment:

$$3.05 \cdot C_2 H_2 + 0.51 \cdot H_2 + (7.89 + Po_2) \cdot O_2 + (90.33 - Po_2) \cdot N_2 \rightarrow Po_2 \cdot O_2 + (90.33 - Po_2) \cdot N_2 + 6.11 \cdot CO_2 + 3.56 \cdot H_2O.$$
(1)

The reactants consist of acetylene ( $C_2H_2$ ), hydrogen ( $H_2$ ), oxygen ( $O_2$ ) and nitrogen ( $N_2$ ). The reaction is normalized to produce 100 moles of product with the value of  $Po_2$  representing the percentage of oxygen desired in the products.<sup>\*</sup> To generate the inert environment used in these experiments,  $Po_2$  is set to zero resulting in a product gas composed of <u>no</u>  $O_2$ , 90.33% N<sub>2</sub>, 6.11% CO<sub>2</sub>, and 3.56% H<sub>2</sub>O, (Simulation of air, exhaust-gas-recirculation, or oxygen enrichment for investigations of combusting sprays can be achieved by setting  $Po_2$  to values of 21, less than 21, or greater than 21, respectively.)

The premixed gas given by the reactants in EQ (1) is synthesized under computer control by adding the required partial pressure of each pure component isothermally (slowly) to a well stirred 10 liter high-pressure cylinder (the mixer). When full, the mixer contains enough premixed gas for approximately 50-100 experiments in the combustion vessel depending on the operating conditions.

A typical pressure history of a simulation of a combusting diesel spray ( $Po_2 = 21$  in EQ (1)) is shown in Fig. 4 as an example of the simulation process. Prior to time zero in Fig. 4, premixed gas from the mixer is metered into the combustion vessel through the intake valve under well stirred conditions. When the desired combustion vessel pressure is reached (8.2 MPa in Fig. 4), the computer closes the intake valve. The gases in the vessel at this time are at the vessel wall temperature of 453 K and have a gas density that is a constant up to the time of the diesel fuel injection. This gas density can be determined from the fill pressure, the wall temperature, the known premixed gas molecular weight and an equation of state (to be discussed later). For the conditions of Fig. 4, the gas density is 61 kg/m<sup>3</sup>. Just after time zero, a spark ignites the premixed charge which then burns (the first pressure-rise). The premixed burn ends at 0.065 s, after reaching a pressure of 32.9 MPa. This pressure is slightly than the constant-volume adiabatic equilibrium lower pressure of the gas mixture under these conditions due to heat transfer to the walls during the combustion process. The hot product gas (~ 1900 K) from the premixed burn then cools due

to heat transfer. Knowing the density from the fill portion of the experiment, the bulk gas thermodynamic state at any time during the cool down can be obtained from the pressure measurement made with the piezoelectric pressure transducer. When the desired thermodynamic state for the diesel injection is reached (17.3 MPa and 1000 K in Fig. 4), the fuel injector is triggered and the diesel spray autoignites and burns (the second pressure rise at 0.54 s).<sup>\*\*</sup> The pressure and temperature state at the time of fuel injection can be considered nearly constant, since the time constant (0.3-0.4 s) for the pressure decay in the vessel prior to the diesel injection is approximately 100 times longer than the diesel injection event of interest.



Figure 4. An example pressure history of the diesel simulation process for a <u>combusting</u> condition.

Details of the Thermal and Fluid Mechanic Conditions for the Vaporizing Experiments - The combustion-vessel mixing fan (see Fig. 1) runs throughout the entire diesel simulation shown in Fig. 4. The function of the fan is threefold: (a) to insure rapid equilibrium of the gas temperature with the vessel wall temperature during the premixed gas fill process, (b) to increase the flame speed during the premixed burn (the first pressure rise in Fig. 4), and most important, (c) to keep the temperature in the vessel uniform after the premixed burn up to the point of the diesel injection event. A fan speed of 8000 RPM in a direction that pulls gas from the center of the vessel and ejects it along the walls was found sufficient for these purposes, and was used for all the experiments reported in this work. Without the mixing fan the premixed burn takes several hundred milliseconds, and after the burn, severe temperature non-uniformities exist as a result of vertical thermal stratification of the gas in the vessel [20].

<sup>\*</sup> Only products with equilibrium concentrations greater than 100 ppm at the experiment temperatures and pressures of interest are included in the products.

<sup>&</sup>lt;sup>\*\*</sup> The only difference between the pressure history for the combusting condition in Fig. 4 and a pressure history for a vaporizing experiment is that there is no pressure rise as a result of combustion of the injected diesel fuel, only a slight pressure decrease from fuel vaporization.

The primary fluid motion created by the fan motion just described is two counter rotating vortices [20]. Detailed velocity measurements made with LDV at various radial, angular, and axial positions after the premixed burn show that the local mean velocities range from 1.7 m/s directly in front of the fan to approximately zero in the centers of the two vortices. Representative mean and rms velocities are 0.9 m/s and 0.7 m/s, respectively. The measured velocities, which are more than two orders of magnitude less than the injected liquid velocity (~400 m/s), coupled with observations of the sprays in the schlieren movies, indicate that the fluid motion created by the fan has very little impact on the transient spray injection processes in this simulation. For comparison, the velocity magnitudes and rms fluctuations are similar to corresponding values measured near TDC in a quiescent DI diesel engine operating at low speed (300-600 RPM) [25].

The thermal environment created by this diesel simulation technique was measured with fine wire (25 µm diameter) platinum/platinum-rhodium (Type R) thermocouples. The measurements show that there is a core region in the vessel (defined as the inner 90% of the vessel volume) with a temperature uniformity of  $\pm$  2% and an rms fluctuation that is  $\pm$ 6% of the temperature difference between the core gas and the wall. The remaining outer 10% of the vessel volume is occupied by 1-2 mm thick wall boundary layers with steep temperature gradients. The relationship of this overall thermal environment to an engine is unknown at this time, since similar detailed information is not available for engines. (Note: The small  $\pm$  2% spatial temperature nonuniformities observed in the core are repeatable and appear as radial and axial gradients coupled with local nonuniformities that depend on the premixed gas composition. The nonuniformities translate to a  $\pm$  2% uncertainty in core temperatures reported in this paper. See Ref. 26 for more discussion.)

The average temperature in the core region of the vessel  $(T_c)$  is the ambient gas temperature  $(T_a)$  referred to in the paper. It is determined from a simple relationship involving the vessel wall temperature  $(T_w)$  and the mass averaged bulk temperature  $(T_b)$ :

$$\frac{T_c}{T_b} = 1 + a \cdot \left(1 - \frac{T_w}{T_b}\right) + b \cdot \left(\frac{T_b}{T_w} - 1\right)$$
(2)

The bulk temperature is determined from the pressure, density (a constant for an experiment), molecular weight and an equation of state. The second and third terms on the right in EQ (2) account for gases in the boundary layers and crevices, respectively. The form of the equation was derived by Ferguson et al. [27] assuming that the temperature distribution in the boundary layers follow a simple power law relationship and that the gases in crevices are at the wall temperature. The term b is a constant equal to the ratio of the chamber crevice volume to the total chamber volume (i.e., b =1.4 cm<sup>3</sup>/ 293 cm<sup>3</sup> = 0.005). The term *a* is related to the boundary layer thickness and was determined from an empirical fit to the measured temperature data. The value of a was found to be dependent on the gas density, varying with density to the -0.295 power (*i.e.*,  $\rho^{-0.295}$ ). For a wall temperature of 450 K and a density of 20.4kg/m<sup>3</sup>, a in



**INJECTOR TIPS** — Three different injector tips were used in this study, all with sharp edged orifices. (The break on the orifice edges was more than an order of magnitude smaller than the orifice diameter.) Each orifice was tapered with the diameter increasing by approximately 0.015 mm from the minimum diameter near the inlet to the diameter at the outlet. Each tip also had a maximum sac width of 1.02 mm and a total sac volume of approximately 0.7 mm<sup>3</sup>. Table 2 lists the orifice minimum diameter  $(d_i)$ , the orifice exit diameter  $(d_o)$ , the discharge coefficient  $(C_d)$ , the area contraction coefficient ( $C_a$ ) and length-to-diameter ratio ( $l/d_o$ ) for each tip. The orifice diameters were measured with wire gauges to  $\pm 0.003$  mm. The coefficients  $C_d$  and  $C_a$  are based on the exit diameter,  $d_o$ , and are related by the definition  $C_d = C_v C_a$ , where  $C_v$  is the orifice velocity contraction coefficient. Appendix A discusses the measurement of the orifice coefficients.



EQ (2) is 0.108 and the ratio  $T_c/T_b$  ranges from 1.08 at

*ured* core temperature  $(T_{c,meas})$  and  $T_c$ , from EQ (2) as a plot

of the ratio  $T_{c.meas}/T_{c.}$  versus  $T_{c.}$  (symbols near 1.0). The comparison is for a density of 20.4 k g/m<sup>3</sup> and a wall temperature

of 450 K. The results show that the correlation given by

EQ (2) is valid to better than  $\pm 0.5\%$  up to a core temperature

dashed lines) as a function of core temperature. The dashed

lines represent  $\pm$  one standard deviation in the measured

temperature fluctuations. The fluctuations vary from  $\pm 1.0\%$ 

of the core temperature at 600 K to ±4.0% of the core

equation of state" ( $P = Z \cdot \hat{A} \cdot \rho \cdot T/MW$ . The compressibility

factor, Z, is determined from the relationship given by

Reynolds [28] and ranges from 1.0 to 1.09 for conditions in

The equation of state used in this work is the "generalized

Also shown in Fig. 5 are the temperature fluctuations (the

Figure 5 shows a comparison between the average meas-

1500 K to 1.04 at 700 K.

temperature at 1500 K.

of 1500 K.

this paper.

**Table 2.** Injector tip parameters for the three tips used. The discharge and area contraction coefficients are based on the exit diameter,  $d_o$ . The velocity coefficient ( $C_v$ ) can be determined from the definition  $C_d = C_v \cdot C_a$ .

Orifice Diameter(mm) d <sub>i</sub> /d <sub>o</sub>	Discharge Coefficient $C_d$	Contraction Coefficient C <sub>a</sub>	Length-to- Diameter <i>I/d<sub>o</sub></i>
0.185 / 0.198	0.56	0.82	5.1
0.241 / 0.257	0.62	0.81	3.9
0.330 / 0.340	0.62	0.84	2.9

INJECTION RATE PROFILES — Figure 6 shows examples of injection rate profiles from individual injections and the corresponding fuel pressures obtained with the 0.257mm tip (two different pressures) and the 0.198 mm tip in Table 2. The figure shows that the injection profile has a very rapid start and end to injection with a nearly constant rate during the injection (*i.e.*, a "top hat" profile). Repeat experiments show that the injection rate and the time to open and close are repeatable to  $\pm 2\%$  ( $\pm 2\sigma$ ) for a given condition. The injector opening time (*i.e.*, time required to reach the full injection rate) determined from injection rate profiles such as those in Fig. 6 are given in Appendix B for each experimental condition. The rapid opening time of this injector is an important consideration when comparisons are made in the Results and Discussion section to a penetration correlation based on that assumption. The technique used to measure injection rate profiles is described in Appendix A.



Figure 6. Injection rate profiles and corresponding fuel pressures (normalized by the initial fuel pressure) for the 0.257 mm and 0.198 mm orifice diameter tips.

#### ANALYSIS OF THE SPRAY IMAGES

For each experiment, the imaging system setup in Fig. 3 was optimized to provide maximum contrast between the background gas and the spray as described previously. This section describes the analysis techniques developed to locate the spray region in each image from which the global parameters of interest were determined: namely, the spray penetration and dispersion angle from the film images and the spray penetration from the linescan images.

**FILM IMAGE ANALYSIS** — Analysis of the high speed movies of the non-vaporizing and vaporizing diesel injection events required three major steps: (1) digitization of the film, (2) filtering and normalization of the digitized images, and (3) masking and analysis of the images for penetration and dispersion. In the digitization step, each film frame was backlit with a film motion analyzer, imaged onto a monochrome CCD camera, and digitized via an 8 bit, 512 x 480 pixel frame grabber in a manner similar to Refs. 29 and 30. The spatial resolution varied slightly with each film, but was nominally 0.2 mm per pixel. The images on the tops of Figs. 7 and 8 show examples of digitized images for a non-vaporizing and vaporizing spray, respectively.

In the filtering and normalization step, each digitized frame was median filtered using a 5x5 pixel block and normalized with a median filtered image from just prior to the diesel injection event. The median filter removed small scale irregularities while maintaining sharp edges in the image, and the normalization corrected for spatial non-uniformities in the schlieren light source.

The masking and analysis were done in a two-step <u>itera-</u> <u>tion</u>. A mask was applied that eliminated portions of the digitized image from consideration in the analysis that were obviously not part of the spray (*e.g.*, regions outside the vessel window, outside the field of view of the original film frame, or to the left of the known injector tip location). An initial analysis (discussed in the following paragraphs) was then performed providing the location of the spray region in the image. A second more refined mask based on the initial spray location was then applied to the image and the analysis repeated. The second masking step helped eliminate spatially random schlieren effect generated by the boundary layers on the windows during the high temperature vaporizing experiments from being considered as part of the spray.

The spray analysis began with selection of an intensity threshold level to be used to separate the spray region (with low light intensity) from the background ambient gas (with high light intensity) in the unmasked region of an image. The threshold was determined from a cumulative intensity histogram for each film image set. In this histogram two peaks appear, one corresponding to the undisturbed ambient gas region with high intensity levels and another corresponding to the spray region with low intensity levels. A threshold intensity midway between the two peaks in the cumulative histogram was chosen. Pixels with intensity levels below the threshold are defined as the spray region while pixels with values above the threshold are defined as the background ambient gas.

The images on the bottom of Figs. 7 and 8 show the image from the top in each figure after filtering, normalizing, masking and application of the threshold. The black region in the bottom images defines the spray and the white region the background.

Once the pixel resolution and the pixels representing the spray region are defined for each image, the spray is characterized in terms of a number of parameters schematically shown in Fig. 9 that include: the spray axis ( $\mathcal{L}$ ), the penetration distance (*S*), the full cone angle ( $\theta$ ), the projected area ( $A_p$ ), and the "local" spray dispersion angle ( $\theta l(x)$ ). The spray axis is defined as the line from the injector tip through



**Figure 7.** (Top) Digitized image of a non-vaporizing spray 1.14 ms after the start of injection:  $\rho_a = 30.0 \text{ kg/m}^3$  and  $T_{a\nu}=451 \text{ K}$ . The window diameter is 114 mm. (Bottom) The same image normalized, median filtered and masked with the midpoint threshold applied. The dark region defines the spray.

the centroid of the spray region in each image. The centroid location is computed from the pixel locations in the spray region. The spray penetration and cone angle require an <u>iterative</u> process to evaluate, since the definition used for each depends on the other. The penetration is defined as the distance along the spray axis to a location where 1/2 of the pixels on an arc of  $\theta/2$  centered on the spray axis are dark (see the schematic on the left in Fig. 9). The spray angle,  $\theta$ , is defined by the following relationship:

$$\theta = \tan^{-1} \left( \frac{A_{p,S/2}}{\left( S/2 \right)^2} \right), \tag{3}$$

where  $A_{prS/2}$  is the projected spray area of the upstream half of the spray in an image (see the schematic on the left in Fig. 9).

The spray angle defined by EQ (3) gives an angle representative of the outer boundary of the upstream portion of the spray. The Results and Discussion section will show that it excludes the unsteady head of the spray. It also excludes the downstream region where the spray comes in contact with either the side or far walls of the vessel.

"Local" spray dispersion angles were also computed for each image. The local angle ( $\theta_l(x)$ ) definition is given on the right in Fig. 9. It is determined by taking the arctangent of the local spray width (w(x)) divided by the axial location of the spray width. The local spray width was calculated at discrete locations by partitioning the spray into bins of fixed length  $\Delta x$  in the axial direction, as is schematically shown in Fig. 9 on the right. The projected area of the spray in each bin divided by the bin length is defined as w(x), where x is the bin center location. The bin length chosen was  $\Delta x=12$  mm. Partitioning the spray into fairly large bins averaged out some of the turbulent structures along the spray.



**Figure 8.** (Top) Digitized image of a vaporizing spray 1.20 ms after the start of injection:  $\rho_{as}=28.6 \text{ kg/m}^3$  and  $T_a=1000 \text{ K}$ . The window diameter is 114 mm. (Bottom) The same image normalized, median filtered and masked with the midpoint threshold applied. The dark region defines the spray.



**Figure 9.** Definition of the spray axis ( $\mathcal{L}$ ), the penetration distance (*S*), the full cone angle ( $\theta$ ), the projected area (the gray region: *Ap*), and the "local" spray dispersion angle ( $\theta_l(x)$ ).

<u>Sensitivity of the Film Analysis to the Intensity Threshold</u> - As described previously, an intensity value midway between the histogram peaks for a film series was used as the intensity threshold to define the spray region. To examine the sensitivity of the film analysis to this choice of intensity threshold, the analysis was repeated for intensity thresholds of 25% and 75% of the way between the histogram peaks. For this range of intensity thresholds, the penetration determined from an image varied by 1%, while the full cone angle and the local spray angles varied by approximately 3°.

LINEAR ARRAY IMAGE ANALYSIS — The linear array CCD camera with a 128 pixel array was aligned with the axis of the spray. During an injection event, the array was read out every 12  $\mu$ s and stored sequentially as a 128 byte long line in a binary file. Each byte represented the intensity (0-255) of a single pixel on the linear array. Using this data, a *pseudo* image of an injection event was created that was 128 pixels wide and 512 lines long. The image contained the spatial variation of light intensity along the axis of the spray as a function of time for a period of 6.14 ms (0.0 12  $\mu$ s x 512).

Figure 10 shows a linescan image of a non-vaporizing spray. The image is oriented so that the vertical axis is the distance (x) along the spray axis. One vertical line in the image corresponds to one readout of the linescan array. The tip of the injector is located at the bottom (x=0) and the opposite side of the vessel is located at the top (x=109 mm). The horizontal axis is time (t) with the origin at the start of injection at the right. Low intensity (dark) regions in the image are regions along the spray axis with significant light attenuation resulting from the presence of the spray, and high intensity (light) regions are regions with non-attenuated laser light (i.e., no spray). The curved border between the light and dark regions from the lower left to the upper right is the location of the spray tip as a function of time. The light region in the lower right corner of the image is associated with the end of injection, which occurs at 3.5 ms. After this time, light is again transmitted to the CCD array near the injector tip as the tail end of the spray moves away for the tip.

The analysis of linescan images, such as in Fig. 10, involved normalizing the image to remove effects of spatial variations in laser intensity and determination of the spray tip location (*i.e.*, the penetration distance *S*) at each 12  $\mu$ s time interval. The spatial variation in laser light intensity for the normalization step was obtained from the intensities just prior to the start of injection (*e.g.*, the vertical variation in intensity at the far left in Fig. 10). The spray tip location was defined as the location along a vertical line in the image where the intensity gradient was a maximum.

START OF INJECTION — Results show that the early spray penetration for the injector used in these experiments has two distinct periods: an initial very slow penetration period, followed by a sudden transition to a fast penetration period. For all conditions, the fast penetration period was found to start before the spray tip penetrated 6 mm. The open symbols in Fig. 11 show a penetration (S) versus time (t) profile for the early part of an injection. This penetration profile was measured with the linescan camera focused on a magnified field of view near the injector tip, which allowed the early part of the penetration profile (-0.3 ms) to be followed with high resolution. The linescan data clearly show the two distinct periods to the early penetration profile: a slow penetration period before time zero and a fast penetration period after time zero. Injector needle lift measurements indicate that slow penetration period begins when the injector main needle first cracks open. At this time, the fuel flow to the orifice is restricted by the needle, resulting in a slow flow of fuel into the chamber. The fast penetration period (i.e., the main injection period) begins when the force balance on the main needle reaches a condition resulting in a sudden retraction of the needle and unrestricted flow up to the orifice.

For the results presented in this work, we have neglected the time involved in the initial slow penetration period and have defined the start of injection to be the start of the main injection period. Estimates show that quantity of fuel involved in the slow penetration period is small (~0.2 mg). In addition, the penetration profiles show that the fuel involved in the early injection period is quickly overtaken by fuel from the main injection period.



Figure 10. A *pseudo* image of the penetration of a nonvaporizing spray obtained with the linear array CCD camera and the optical setup in Fig. 3. The horizontal axis is time and the vertical axis is penetration distance. The curved boarder between the light and dark regions from the lower left to the upper right is the location of the spray tip as a function of time. The experimental conditions were:  $d_o= 0.257$  mm,  $P_f= 140$  MPa,  $T_a=451$  K,  $P_{ce}=4.08$  MPa and  $\rho_{ce}=30.0$  kg/m<sup>3</sup>.



**Figure 11.** An example of the definition of the "start of injection." Penetration versus time from a film and a high magnification linescan image for the same nominal conditions. The solid curve is the fit of the function used to define t=0 fit to the linescan data.

For high magnification linescan data, such as the open symbols in Fig. 11, the start of the main injection period is found by least squares fitting a function of the form of a penetration correlation (to be discussed later) to the penetration data for distances greater than 6.0 mm, The time at which this function crosses the time axis is used to define the start of injection (t=0). The solid curve in Fig. 11 is an example of the function fit to the high magnification linescan data in the figure. Its intersection with the time axis establishes the beginning of the main injection (t=0) in the figure.

The determination of the start of injection for the film and the lower magnification linescan penetration data taken over the entire injection period (up to 3.5 ms) is slightly more involved. These data have a lower overall resolution than the high magnification linescan data, and as a result, few data points in the early penetration period for a reliable, direct fit to the function used to find the start of injection. This can be seen by comparing the film data (the closed symbols) with the high magnification linescan data in Fig. 11.

For the film data and the lower magnification linescan data, the start of injection is determined instead by adjusting the time base of these data until the initial data points (including those in the initial slow penetration period) overlay a high magnification linescan penetration profile for the same condition. The film data in Fig. 11 shows an example of the overlay. The zero time for the high magnification linescan data then defines the start of injection for the lower resolution data. With this approach, the film and lower magnification linescan penetration data for penetration distances less than 6.0 mm (the slow penetration period) can also be utilized in determining the start of the main injection.

REPEATABILITY OF THE PENETRATION AND DIS-PERSION DATA — Figure 12 gives an indication of the repeatability of the penetration and the dispersion angle measurements from injection to injection. Plotted are the



**Figure 12.** Penetration and dispersion angles for four film data sets and penetration for seven linescan data sets versus time:  $\rho_a=30.0 \text{ kg/m}^3$ ,  $T_a=451 \text{ K}$ ,  $P_f=140 \text{ MPa}$  and  $d_o=0.257\text{ mm}$ . The dashed lines around the penetration and the angle data are  $\pm$  7% and  $\pm$  0.7° repeatability bands, respectively. The average dispersion angles for the four film cases between 0.5 ms and 3.0 ms are: 10.0°, 9.5°, 9.5°, and 9.6°.

penetration (S) and the spray half angles ( $\theta/2$ ) for four injections recorded on film plus the spray penetration for six injections recorded with the linescan camera. The data are for a non-vaporizing condition, an injection pressure of 140±1.0 MPa, an ambient gas density of 30.0±0.05 kg/m<sup>3</sup> and the 0.257 mm orifice diameter tip. The figure shows that the penetration and spray angle data are repeatable to ±7% (±2 $\sigma$ ) and ±0.7° (±2 $\sigma$ ), respectively. These repeatability bands are represented by the dashed lines around the penetration and the spray angle data in Fig. 12. The primary source of these variations is believed to be turbulence.

Also given in the figure title for Fig. 12 are the average dispersion angles for times between 0.5 ms after the start of injection to 0.5 ms before the end of injection for the four film injection events. Such averages, which will be used in the Results and Discussion section, typically have repeatabilities of  $\pm 0.6^{\circ}$  ( $\pm 2\sigma$ ).

# **RESULTS AND DISCUSSION**

The results and discussion are divided into seven sections. Spray dispersion is discussed in the first three in the following order: the spatial and temporal variation of the dispersion angle for non-vaporizing and vaporizing sprays, the overall effects of ambient gas density on dispersion, and the overall effects of vaporization on dispersion. The last four sections discuss penetration. The fourth section presents the general effects of gas density and vaporization on spray penetration. In the fifth and sixth sections, characteristic time and length scales for spray penetration are introduced that correlate the effects of ambient gas density on spray penetration and that help isolate and explain the effects of vaporization. In the last section, the relationship between the spray penetration scaling parameters presented in this work and correlations for penetration in the literature are discussed. In general, the comparisons made to the literature throughout the discussion are mainly for non-vaporizing sprays, since there has been little research on vaporizing sprays under diesel conditions.

Two important considerations to keep in mind regarding the following results are that they were obtained with a #2diesel fuel and with an injector that has a "top hat" injection rate profile (*i.e.*, no "rate shaping"). Controlled investigations of the effects of rate shaping on spray penetration and dispersion have not been conducted.

Detailed conditions for the experiments are given in Appendix B. Listed are the chamber gas density, temperature and compressibility factor at the time of injection; the fuel density; the injection pressure; the orifice diameter; and the time from the start of injection to the full injection rate.

TEMPORAL AND AXIAL VARIATION OF THE SPRAY DISPERSION ANGLE — In the two sections following this one, the effects of ambient gas density and vaporization on spray dispersion will be examined using a single angle to characterize the dispersion of each injection event. The use of a single angle to characterize the dispersion of non-vaporizing sprays is a common practice based on "visual" inspection of photographs of sprays [*e.g.*, 2,14-16]. In this section, the "local" angle  $\theta_l(x)$  defined on the right in Fig. 9 will be used to show in a more quantitative manner the region of the spray and the portion of the injection duration

over which this practice applies for the sprays examined in this work. The results will also provide insight for interpreting the penetration results in later sections.

The temporal and axial variation of local spray angles noted in this experiment are shown in Figs. 13a and 13b. The figures are plots of the tangent of the local spray half angle ( $\theta_l(x)/2$ ) versus time for several axial locations along a non-vaporizing spray ( $\rho_a$ =14.8 kg/m<sup>3</sup>) and a vaporizing spray ( $\rho_a$  =13.9 kg/m<sup>3</sup>). The data were obtained with the 0.257 mm orifice diameter tip (see Table 2) with an injection pressure difference across the orifice of 138 MPa. Only axial locations in the upstream half of the chamber are shown, eliminating the potential for including effects of downstream wall impingement. In addition, the first 4 mm near the tip were not adequately resolved by the film digitization and are not included.

The figures show that there is an initial transient at each axial location along the spray as the spray head reaches that location. This initial transient lasts approximately 0.5 ms and is followed by a period with a relatively constant, "quasisteady" spray angle. The angle during the initial transient can significantly exceed the quasisteady angle (as shown in Figs. 13a and 13b) or gradually approach it (visible for other conditions not shown). By comparing the angle during the quasisteady period at various axial locations, it can also be observed that it is independent of axial location. The small temporal and axial fluctuations during the quasisteady period are the result of the turbulent nature of the spray. Finally, a small increase in the spray angle is sometimes observed at each axial location near the end of injection, such as between 3.0 ms and 3.5 ms in Figs. 13a and 13b. This rise is most likely associated with needle closing processes.

Figures 13a and 13b show that the spray region upstream of the transient spray head develops a dispersion angle that is independent of axial location and time for the conditions of this experiment. The dispersion angle of this quasisteady period will be used to characterize the spray dispersion in the remaining sections. This result also has an important implication if the flow is assumed to be self-preserving during the quasisteady period; namely, the entrainment rate along the spray is a constant (*i.e.*, the mass flow in the spray grows linearly with axial distance similar to fully developed incompressible jets [35]). For the above conditions, the local air entrainment rate along a spray will be proportional to the ambient air density ( $\rho_a$ ), the orifice diameter ( $d_o$ ), the injected fuel velocity ( $U_f$ ), and the tangent of the spray dispersion half angle ( $\theta/2$ ):

Entrainment 
$$\propto \rho_a \cdot d_o \cdot U_f \cdot \tan(\theta / 2).$$
 (4)

This proportionality will help explain the penetration results in later sections.

The constant local entrainment rate implied by the results is supported by recent direct measurements of the entrainment by transient sprays from a diesel injector in <u>atmospheric</u> pressure air [31,32]. It has also been established for the non-head region of transient <u>gas</u> jets [33,34].

EFFECTS OF GAS DENSITY ON NON-VAPORIZING SPRAY DISPERSION — The effects of the various parameters on the dispersion of non-vaporizing sprays noted in this



Figure 13a. The local dispersion angle versus time at four axial locations along a non-vaporizing spray:  $\rho a=14.8 \text{ kg/m}^3$ ,  $T_a=451 \text{ K}$ ,  $P_f=141 \text{ MPa}$ , and  $d_o=0.257 \text{ mm}$ 



**Figure 13b.** The local dispersion angle versus time at four axial locations along a vaporizing spray:  $\rho_a$ =13.9 kg/m<sup>3</sup>,  $T_a$ =1001 K,  $P_f$ =142 MPa, and  $d_a$ =0.257 mm.

experiment are shown in Fig. 14a with an emphasis on the effects of ambient gas density. Figure 14b compares those results to previous data in the literature.

Figure 14a is a plot of the tangent of the spray half angle versus the ratio of the ambient gas and injected fuel densities for all the non-vaporizing experiments (the symbols). For reference, typical current technology diesels fall in the 0.01 to 0.04 density ratio range. The angles plotted are the average angle for the quasisteady, constant angle region of a spray noted in the previous section. They were determined by averaging the angles calculated with EQ (3) for times between 0.5 ms after the start of injection and 0.5 ms before the end of injection. This resulted in an average angle for the upstream portion of the sprav that excludes effects of the transient sprav head, the initial head formation, and the injector closing. The average angles given in the caption for Fig. 10 and those given in Appendix B (as  $tan(\theta/2)$ ) for each condition were (Note: The data for the determined via this method. 0.198 mm orifice diameter tip taken at three different density



Figure 14a. Spray dispersion angle versus the ambient gas to fuel density ratio for all non-vaporizing conditions (see Appendix B, Table B1).



Figure 14b. Comparison of the non-vaporizing spray dispersion angle results with Hiroyasu and Arai [2], Reitz and Bracco [15], and Wu *et al.* [16].

ratios and injection pressures are difficult to see because they fall directly on top of the data for the 0.257 mm orifice diameter tip.)

Also shown in Fig. 14a are least squares fits of the data for the 0.340 mm. orifice diameter tip (the solid line) and the 0.257 mm orifice diameter tip (the broken line) to a power law relationship. The dashed lines on either side of each fit give a band representing the  $\pm 0.6^{\circ}$  repeatability noted for the average angles plotted. Only angles for conditions with density ratios less than 0.1 and the highest injection pressures were used in the least squares fits. The exponent in the fits shown in the figure is the average of two exponents that resulted from fitting the data for each tip individually with the power law relationship.

Four observations can be made from the spray angle data in Fig. 14a. The most significant is that spray angles increase with an increase in the ratio of the ambient gas and fuel densities. Based on the fits to the data for density ratios less than 0.1, the dependence of the tangent of the spray half angle on the density ratio is $(\rho_{a'}/\rho_{f})^{0.19}$ .

The second observation is less definite, but visible in the higher density ratio data. The angles at density ratios greater than 0.1 appear to be diverging from the density ratio dependence noted at lower densities. At higher densities they are approaching angles typically quoted for the outer boundaries of incompressible, fully developed jets with a  $\rho_a/\rho_f$ =1 (*i.e.*, tan ( $\theta/2$ )  $\approx$  0.22 to 0.29). This divergence is most noticeable for the 0.340 orifice diameter data at density ratios above 0.1. It is more evident for the 0.257 orifice diameter tip if angles measured using this tip to inject hydrogen in another set of experiments [36] are considered. The jet half angles measured in the hydrogen experiments at an ambient gas to fuel density ratio of 0.7 were  $13^{\circ}$  (*i.e.*,  $tan(\theta/2)=0.23$ ). This density ratio lies significantly to the right of the data in Fig. 14a, and the angle lies well under the extrapolation of the data for density ratios less than 0.1. The trend at higher density suggests that as the ambient gas density is increased, sprays behave more like gas jets.

The third observation from Fig. 14a is based on the differences between the tips. The dispersion angles for the 0.257 mm diameter orifice (the open symbols) and the 0. 198 mm. (+,  $\times$  and \*) diameter orifice are essentially the same, but the angles for the 0.340 mm diameter orifice (the closed symbols) are consistently about 40% larger than the other two. This implies that tip geometry parameters other than just the orifice diameter are important, since there are differences between tips but no consistent trend with respect to office diameter.

The fourth observation is that there is no significant effect of injection pressure on the spray dispersion angle for the range of injection pressures examined ( $P_{f=}$ 75-160MPa). The data for the various injection pressures are indistinguishable at a given density ratio.

The trends described above for density ratios less than 0.1 in Fig. 14a generally support observations previously made by Hiroyasu and Arai [2] with data obtained for injection pressures less than 80 MPa in a density ratio range of 0.001 to 0.04. Figure 14b compares the data from Fig. 14a to their results. Shown in the figure are the fits from Fig. 14a (the two longer lines in Fig 14b) and lines representing the angles predicted for each of the injector tips in Table 2 with a relationship given by Hiroyasu and Arai [2] (the shorter broken lines in Fig. 14b). The lengths of the lines in the figure correspond to the density ratio range of the data from which the various lines were derived. Keep in mind that the 0.198 mm orifice diameter data.

Figure 14b shows that in the density range for which the Hiroyasu and Arai correlation was developed, the agreement with respect to the effects of density ratio on spray angle is reasonable given the repeatability for this type of data. Over the larger density ratio range of our data, however, we show a smaller dependence on the ambient gas and fuel densities. The fits to our data have a 0.19 power dependence on density ratio compared with a 0.26 power dependence in the Hiroyasu and Arai correlation [2]. Both of these dependencies are less than the 0.4 power dependence given by Wakuri *et al.* [13] and the 0.33 power dependence given by Varde et al. [14].

The agreement with respect to tip geometry effects shown in Fig. 14b is not as good. The total magnitude of the tip geometry effects predicted with the Hiroyasu and Arai correlation agrees with the difference noted in our data, but the tip-

to-tip comparison is not consistent. Even this apparent disagreement with respect to orifice geometry parameters, however, supports the past observations of Hiroyasu and Arai [2], as well as Reitz and Bracco [15]. Both have clearly shown that orifice geometry parameters are the only other non-fuel parameter besides the density ratio to have a significant effect on spray dispersion angles at the high injection pressure conditions of this experiment. The detailed differences in Fig. 14b mainly indicate that all the effects of injector tip design are not understood yet. Differences that exist between the tips in the two experiments, such as the location of the orifice in the tip, could easily cause the trends noted. The tips in the Hiroyasu and Arai [2] experiments have the orifices on the tip centerline axis, while in the our case they are 34° off axis. We noted a fairly substantial difference in some of the orifice coefficients as a result of orifice differences such as this, as is mentioned in Appendix A.

Figure 14b also presents a comparison with the non-vaporizing spray angles measured by Reitz and Bracco (for injector tip IX in Ref. 15) and Wu *et al.* (for injector tip IIb in Ref. 16). The data selected from their papers were for tips similar to those in Table 2, *i.e.*, tips with sharp edged orifices and comparable diameters (0.34 mm) and aspect ratios (4.1). In addition, the data selected from Wu *et al.* was for a high injection pressure (92 MPa).

The comparison of the various data sets at the lower density ratios shows that the angles presented by Reitz and Bracco and Wu *et al.* are generally much less than those presented by Hiroyasu and Arai and those observed in this work. But as the density ratio increases, the various data sets converge by a density ratio of 0.08. In addition, the figure shows that the dependence on the ratio of the ambient gas and fuel density for their data is significantly larger than observed in this work. Reitz and Bracco give a dependence of  $(\rho_a/\rho_f)^{0.5}$ .

However, direct comparison of the angles from Reitz and Bracco and Wu et al. with those of Hiroyasu and Arai and this investigation must be done considering the large differences in the regions of the spray examined and in the parameters controlling atomization and spray mixing. The angles in Reitz and Bracco and Wu et al. experiments were measured in the atomization-dominated region very near the injector tip (the initial 5 mm). In contrast, the angles reported in this work and by Hiroyasu and Arai, are based on a spray region much further from the injector tip. Apparently, as the density increases, either the "atomization" angles measured by Reitz and Bracco and Wu et al. close to the injector become comparable to the angles in downstream spray regions dominated by shear layer mixing effects (*i.e.*, the angles reported in this work) or the mixing processes become important closer to the injector tip.

The results with regard to the effects of ambient gas density on non-vaporizing sprays can be summarized as follows. An increase in the ambient gas density results in an increase in the spray dispersion angle, as has been reported several times previously [*e.g.*, 2,14-16]. However, the dependence of the spray dispersion angle on ambient gas density away from the immediate tip region appears to less than previously reported, based on the data from this experiment obtained over a very large ambient gas density range and at current technology injection pressures. Next, the results of this section suggest that as gas density is increased, sprays will behave more like incompressible jets. The dispersion angles noted at gas densities above 60 kg/m<sup>3</sup> (approximately twice the TDC density in current technology heavy-duty diesels) are already approaching those quoted for incompressible jets with a  $\rho_a/\rho_f=1$ . Finally, referring back to EQ (4) in the previous section, the results indicate there will be an increase in the entrainment rate with an increase in gas density due to two causes: first as a direct result of the increase in the ambient gas density ( $\propto \rho_a$ ), and second, as a result of the increase in the dispersion angle with an increase in gas density ( $\propto \tan(\theta/2)$ ). The second cause will help explain new observations on effects of ambient gas density on penetration to be presented in later sections.

EFFECTS OF VAPORIZATION ON SPRAY DISPER-SION — Figure 15 shows the effects of vaporization on spray dispersion in a plot similar to Fig. 14a. The angles were determined in the same manner as for non-vaporizing sprays and had a similar repeatability. The curves through the vaporizing data show the observed trend. Also plotted in Fig. 15 for comparison are the fits to the non-vaporizing data previously shown in Figs. 14a and 14b for the two tips with larger orifices (the straight lines).

The majority of the vaporizing data were taken at an ambient gas temperature of 1000 K, which is a representative temperature in a heavy-duty diesel engine just prior to combustion. Data for ambient gas temperatures between 600 K and 1400 K were obtained at the higher gas density conditions with the 0.257 mm orifice diameter tip. The variable temperature data are difficult to see, however, because they fall on top of the 1000 K data. Similarly, the data for the 0.198 mm orifice diameter tip are difficult to isolate because they fall on top of the data for the 0.257 mm orifice diameter tip, as was the case for the non-vaporizing data.

Comparison of the vaporizing data to the fits for the non-vaporizing data shows that vaporization reduces the spray dispersion angle, but that the reduction decreases with increasing the density ratio. The reduction at the lowest density ratio is about 30%, but by a density ratio of 0.04 the difference is small. The figure also shows that the effects of tip geometry and injection pressure are the same as were observed for the non-vaporizing sprays, and that temperature has no significant effect at the higher density conditions.

The biggest potential factor contributing to the decrease in the dispersion angle by fuel vaporization is expected to be a contraction of the vaporizing spray as a result of cooling of entrained hot gases by fuel evaporation. The most noticeable indication of a contraction by the evaporative cooling is the in-cylinder pressure decrease typically noted in engines during the ignition delay period [10].

An estimate of the spray angle reduction from the contraction can be made by using the predictions of dodecane and heptane evaporation under conditions of adiabatic saturation in a spray by Kamimoto and Matsuoka [37]. For an ambient gas temperature of 1000 K, a fuel temperature of 300 K, and an ambient gas density of 6.5 kg/m<sup>3</sup> (*i.e.*, a  $\rho_a/\rho_f \approx 6.5/800 \approx$ 0.008), they predict that the fuel/air mass ratios required for complete evaporation of heptane and dodecane are 1.1 and 0.7, respectively. The corresponding fuel/air mixture temperatures for these conditions are 420 K and 530 K, respectively.

72



Figure 15. Spray dispersion angle versus the ambient gas to fuel density ratio for all the vaporizing conditions (see Appendix B, Table B2). The curves show the observed trends in the vaporizing data. The lines are fits to non-vaporizing data from Fig. 14a.

Using these numbers, the fuel and air properties, and the ideal gas law, the estimated net contraction of the spray crosssectional area as a result of contraction of the cooled air and expansion of the vaporized fuel is 40%. This contraction corresponds to a 25% reduction in the spray width, and therefore, the spray angle. The estimate is larger than the measured spray angle reduction of 15% at a density ratio of 0.008 in Fig. 15 since only fuel in the central region of the spray will approach adiabatic saturation conditions. The rest of the fuel in the spray will evaporate under much leaner conditions, lessening the overall contraction.

A second factor, but probably less significant, contributing to the smaller dispersion angles of vaporizing sprays is the evaporation of the larger ballistic drops with high dispersion angles at the periphery of the spray (*i.e.*, droplets not following the gas phase motion) and re-entrainment of the fuel vapor. The visible effect we would detect with our optical setup for this condition would be an apparent reduction in spray dispersion over the non-vaporizing case.

The results indicate that for a density ratio range reflective of current technology heavy-duty diesel engines (0.01 to 0.040), the dispersion angle reduction will range from about 15% at the low density end to a few percent at the high density end. The impact on diesel sprays other than the visible contraction is not clear at this time. Equation (4), if applicable for vaporizing sprays, suggests that a reduction in air entrainment proportional to the reduction in the tangent of the spray half angle could occur as a result of vaporization. However, if vaporization of droplets at the periphery of the spray is responsible for the reduction in the dispersion angle, there may be no significant impact.

EFFECTS OF GAS DENSITY AND VAPORIZATION ON SPRAY PENETRATION -- The general spray penetration trends for the non-vaporizing and vaporizing sprays are shown in Figs. 16a and 16b, respectively. Both figures are plots of penetration versus time. The data represented by the symbols in each figure were obtained with the 0.257 mm orifice diameter tip and an injection pressure difference across the orifice of 137 MPa ( $\pm$ 1.5 MPa). The "non-vaporizing" data in Fig. 16a were obtained in nitrogen at an ambient temperature of 451 K over an ambient density range from 3.6 to 124 kg/m<sup>3</sup>. The vaporizing data in Fig. 16b were obtained in the inert environment discussed in the Experimental Procedure and Conditions section at an ambient temperature of 1000 K over a gas density range from 3.3 to 58.6 kg/m<sup>3</sup>.



Figure 16a. Penetration versus time for non-vaporizing sprays. The data (the symbols) were obtained with the 0.257 mm orifice diameter tip, an injection pressure difference of 137 MPa, and an ambient gas temperature of 451 K. The curves are predictions based on the correlations of Hiroyasu and Arai [2] for each ambient density condition.



Figure 16b. Penetration versus time for vaporizing sprays. The data (the symbols) were obtained with the 0.257 nun orifice diameter tip, an injection pressure difference of 137 MPa, and an ambient gas temperature of 1000 K. The curves are fits to our non-vaporizing data in Fig. 16a.

The most noticeable trends shown by the penetration data in Figs. 16a and 16b are the decrease in penetration with an increase in ambient density and the decreasing rate of penetration with time. These trends have been observed by many others dating back to the 1920s (see Hay and Jones [1]). However, a closer comparison of the non-vaporizing data to data in the literature obtained over more limited and lower gas density ranges, and a direct comparison of the nonvaporizing and vaporizing data obtained in this work reveals new observations.

Figure 16a compares the non-vaporizing penetration data (the symbols) to one of the more quoted penetration data sets, that of Hiroyasu and Arai [2]. The curves in Fig. 16a are the spray penetrations for the various gas density conditions in the figure predicted by the spray penetration correlation of Hiroyasu and Arai. Their correlation was developed with data taken under non-vaporizing conditions, but at much lower injection pressures (< 40 MPa) and over a smaller range of ambient densities (10 to 33kg/m<sup>3</sup>). Comparison of our data with the correlation of Hiroyasu and Arai in the density range of 10 to 33 kg/m<sup>3</sup> indicates that there is reasonable agreement given the turbulent nature of sprays and the repeatability shown in Fig. 12. When the wider density range of our experiment is considered, however, a larger density effect is observed in the data than has been observed previously. At lower densities, the Hiroyasu and Arai correlation consistently underpredicts penetration while overpredicting it at higher densities. Similar observations can be made when comparing to other correlations such as Dent's [8], which was developed with data from the experiments of many researchers for gas densities between I and 23 kg/m<sup>3</sup>. The overprediction at higher gas densities was also observed by Hav and Jones [1] when comparing Dent's correlation to the penetration data available to them.

Figure 16b compares non-vaporizing spray penetration data (the curves) with vaporizing penetration data (the symbols). The curves in Fig. 16b are fits to our non-vaporizing data in Fig. 16a. The slight difference in gas density between the non-vaporizing and the corresponding vaporizing condition results in at most a 2% effect on penetration based on the present understanding of the scaling of spray penetration (to be discussed later). The comparison shows that the vaporization reduces or slows penetration with the effect being most visible at the lower density conditions. The reduction is as much as 20% at the lower densities. However, by an ambient density of 60 kg/m<sup>3</sup> and for longer penetration distances, the effect of vaporization becomes smaller. The magnitude of the effect of vaporization on penetration and the decrease in the effect with increasing gas density are similar to the trends observed for the effects of vaporization on spray dispersion in the previous section.

The results suggest that the practice of using spray penetration data from non-vaporizing sprays to represent vaporizing sprays is only reasonable at the highest density conditions of this experiment. The results also suggest that none of the temperature terms that have been included in penetration correlations apply for vaporizing sprays since they predict reductions in penetration at elevated temperature that are independent of the ambient gas density [8,9]. Penetration data were also obtained for other orifice diameters and injection pressures, and for gas temperatures between 600 K and 1400 K at the higher gas density conditions. The effects of orifice diameter and injection pressure on the non-vaporizing and vaporizing sprays were the same. These effects are consistent with trends noted in the literature and will become apparent in the next section. The effects of ambient temperature between 600 K and 1400 K on the penetration of vaporizing sprays at the higher gas density conditions examined was negligible, consistent with the small difference noted between the high density vaporizing and non-vapor-izing conditions in Fig l6b.

CORRELATION OF THE EFFECTS OF AMBIENT GAS DENSITY ON SPRAYS - In the previous section, gas density was shown to have a larger systematic effect on spray penetration than has been noted before. In this section we will introduce length and time scales for spray penetration that account for and explain those effects, as well as account for the effects of other parameters previously presented in the literature. These length and time scales are simplified forms of the characteristic scales used to non-dimensionalize and simplify a spray penetration analysis presented in Appendix C. The penetration correlation that resulted from that analysis is also introduced in this section. (The analysis approach in Appendix C follows the works of Wakuri et al. [13] and Hays [38].) The characteristic length and time scales and the correlation provide a basis for a more complete examination of the vaporizing and non-vaporizing penetration data and a more thorough comparison with penetration data from the literature.

The penetration length  $(x^+)$  and time  $(t^+)$  scales are:

$$x^{+} = \frac{d_{f} \cdot \sqrt{\tilde{\rho}}}{a \cdot \tan(\theta/2)},$$
(5)

$$t^{+} = \frac{d_{f} \cdot \sqrt{\rho}}{a \cdot \tan(\theta/2) \cdot U_{f}},$$
(6)

where:

$$\overline{\rho} = \rho_f / \rho_a, \tag{7}$$

$$d_f = \sqrt{C_a \cdot d_o}, \tag{8}$$

$$U_f = C_v \cdot \sqrt{2 \frac{\left(P_f - P_a\right)}{\rho_f}}.$$
(9)

The angle  $\theta$  in EQs (5) and (6) is the dispersion angle measured in this work. The term a is relates the tangent of the measured angle  $\theta$  to the tangent of the spray dispersion angle in the model spray used to develop the penetration correlation in Appendix C. For this paper, a value of 0.66 is used for *a* based on a best fit of the correlation to our non-vaporizing data. (The reader should refer to Appendix C for more details on the value selected for *a*.) The other terms are: the fuel ( $\rho_f$ ) and ambient gas ( $\rho_a$ ) density ratio( $\overline{\rho}$ ) defined in EQ (7) , the effective diameter ( $d_f$ )of the fuel stream exiting the orifice with a diameter of  $d_o$  defined by EQ (8), and the fuel velocity at the orifice exit ( $U_f$ ) defined by EQ (9). The area contraction coefficient ( $C_a$ )in EQ (8) accounts for fuel flow area loss as a result of cavitation bubbles reaching the orifice exit [39] or "hydraulic flip" [40]. The velocity coefficient  $C_{\nu}$ , in EQ (9) accounts for head losses through the orifice. Values for  $C_a$  are given in Table 2 for each tip and  $C_{\nu}$  can be determined from  $C_a$  and  $C_d$  in the table. The pressures  $P_f$  and  $P_a$  are the fuel pressure in the injector and the ambient gas pressure, respectively.

The penetration correlation developed in detail in Appendix C is:

$$\widetilde{t} = \frac{\widetilde{S}}{2} + \frac{\widetilde{S}}{4} \cdot \sqrt{1 + 16 \cdot \widetilde{S}^2} + \frac{1}{16} \cdot \ln\left(4 \cdot \widetilde{S} + \sqrt{1 + 16 \cdot \widetilde{S}^2}\right)$$
(10)

The correlation is in the form of dimensionless <u>penetration</u> <u>time</u> ( $\tilde{t} = t/t^+$ ) as a function of the dimensionless <u>penetration</u> <u>distance</u> ( $\tilde{S}=S/x^+$ ).\* Two limits of EQ (10) that will also be referred to in the discussion are the short time limit:

$$\begin{array}{c|c} \text{Limit} & \widetilde{S} = \widetilde{t} \\ \widetilde{t} \to 0 \end{array} \right|$$
 (11)

and the long time limit:

$$\begin{array}{c|c} \text{Limit} \\ \widetilde{t} \to \infty \end{array} \middle| \widetilde{S} = \widetilde{t}^{\frac{1}{2}} .$$
 (12)

Figure 17a is a plot of the correlation (the solid curve) and the short and long time limits (the broken lines) in log-log coordinates to emphasize the nature of the correlation. Both limits appear linear in these coordinates. The dashed lines are  $\pm 10\%$  deviations from the correlation and will be used for data comparisons. The short time limit clearly shows an early period of penetration with a linear dependence on time that has been observed experimentally by several researchers [*e.g.*, 2,15]. The long time limit has the gas jet-like square root dependence on time also noted by several researchers [*e.g.*, 2,8,13]. The transition period between the two limits is the time when the dominant medium in the spray changes from the injected fluid at short times to the entrained gas at long times.

Figure 17b shows the penetration data from <u>all</u> of the nonvaporizing experiments conducted in this investigation, normalized by the length and time scales given by EQs (5) and (6). The figure includes the data for all the conditions listed in Appendix B, Table B1 obtained with the high speed film camera, plus other data obtained at the same conditions with the linescan camera. For the linescan data, the fits to the dispersion angle data from the films (see Fig. 14a) were used to estimate dispersion angles for the length and time scales, since the linescan images did not provide a dispersion angle. The data set plotted includes the wide range of conditions described previously. There are approximately 2000 individual penetration data points on the plot. The solid curve visible at a few locations and the broken and dashed lines correspond to the lines and curves in Fig 17a.

The figure shows that the data is collapsed by the proposed time and length scales to within  $\pm 10\%$ . The scatter in the scaled data is very close to the repeatability noted



Figure 17a. The dimensionless penetration correlation given by EQ (10) and the short and long time limits of the correlation, EQs (1 1) and (12). The dashed lines represent  $\pm 10\%$  deviations from the correlation.



Figure 17b. All of the non-vaporizing penetration data (the symbols) versus time in dimensionless coordinates compared with EQ (10) (the solid curve). The short and long time limits (the broken lines) and the  $\pm 10\%$  deviations from EQ(10) shown in Fig. 17a are also included.

for the condition in Fig. 10. It is most likely due to the turbulent nature of the spray penetration process.

Figure 17b also shows that the form of the correlation given by EQ (10) agrees with the data. Only one small systematic trend was noted in the non-vaporizing data in Fig. 17b that deviates from the correlation. Between  $\tilde{t}=0.5$  and  $\tilde{t}=10$  (or between  $\tilde{S}=0.3$  and  $\tilde{S}=3.0$ ) the data tend to lie above the correlation. We believe that this difference between the theory and the data is the result of ballistic droplets that are not following the gas phase motion.

Figure 18 more clearly shows the effect referred to above. Plotted in Fig. 18 is the average percent difference between the penetration data in Fig. 17b and the penetration given by EQ (10) versus the dimensionless penetration distance (the solid curve). The dashed curves represent  $\pm$  two standard deviations in the difference ( $\pm 10\%$ ).

<sup>\*</sup> An approximate inverse of EQ (10), *i.e.*, distance versus time, is given in the Appendix C.

Figure 18 shows that the non-vaporizing spray penetration data agrees with EQ (10) at short and long penetration distances. However, between  $\tilde{S}$ 's of 0.3 and 3 the data indicates that there is up to an 8% larger penetration in the data than is given by EQ (10). This observation coupled with films showing droplet streaks at the head of the non-vaporizing spray, especially for lower density conditions, supports the argument that the deviation in Fig. 18 is due to larger ballistic droplets leading the spray at early times. The model from which the correlation was developed assumes instantaneous transfer of momentum from the injected fluid to the entrained gas. For larger droplets, this will not happen. A "slip" velocity will exist between the droplets and the gas, with the larger droplets retaining the injected fluid velocity longer, and thus, penetrating faster than the gas initially.

An implication of Fig. 18 is that conditions leading to poor atomization may lead to more significant deviations from EQ (10). Based on the results for the range of conditions covered in this experiment, however, this is not likely to be a significant issue for current technology heavy-duty diesel engine injection pressures, in-cylinder gas densities, and orifice sizes.

Significance of the Terms in the Length and Time Scales -The significance of the various terms in the length and time scales in accounting for the effects of ambient gas density on spray penetration noted in this work are examined in Figs. 19a and 19b. The data in the both figures are the nonvaporizing data from Fig l6a with the same symbols applying for each condition. In Fig. 19a the  $tan(\theta/2)$  term is dropped from the time and length scales in EQs (5) and (6) and is replaced by a constant equal to the tangent of the average angle for the data. The resulting scaled coordinates are given in Fig. 19a. The length scale contains the product  $d_f \tilde{\rho}^{1/2}$ , while the time scale contains the same product divided by the injected fluid velocity. The product  $d_f \tilde{\rho}^{1/2}$  is a term that has long been used to scale the effects of density differences between the injected and ambient fluids in steady jets [41] and transient sprays [42]. The product comes from simple momentum considerations and accounts for the additional momentum of a jet with a density different from the ambient medium. Effectively,  $d_f \tilde{\rho}^{1/2}$  is also the scaling used by Hiroyasu and Arai [2], Dent [8] and others for transient sprays, although they never presented their penetration results in terms of dimensionless penetration time and distances.

Comparing Fig. 16a with Fig. 19a shows that these modified time and length scales account for a large portion of the decrease in penetration with increasing ambient density, but not all of the decrease. In the long time region in Fig. 19a, the remaining unscaled effects of ambient gas density on penetration result in a further decrease in penetration at a given time as the ambient gas density increases. (The direction of increasing ambient density is indicated by the arrow in Fig. 19a.) These remaining unscaled effects are the source of the differences noted in Fig. 16a between the correlations of Hiroyasu and Arai [2] and our data. The reason for this additional effect of ambient gas density on penetration can be understood by considering the increase in air entrainment predicted by EQ (4) with an increase in the dispersion angle. As the ambient density increases, Fig. 14a shows that the



Figure 18. The average percent difference between the data in Fig. 17a and EQ (10) versus dimensionless penetration distance (the solid curve) and  $\pm$  two standard deviations in the difference (the dashed curves). The line at zero percent is for reference.



Figure 19a. Penetration versus time normalized by modified length and times scales as defined in the figure. The data are from Fig 16a.



**Figure 19b.** Penetration versus time normalized by the full length and times scales given by EQs (5)and (6). The data are from Fig 16a.

dispersion angle increases. Equation (4), in turn, states that the air entrainment increases proportionally. The net result is that the fuel momentum is transferred to a larger quantity of air resulting in lower overall spray velocity and a slower penetration. Put more simply, penetration is linked to dispersion.

In Fig. 19b, the full length and time scales given by EQs (5) and (6) have been used to scale the data from Fig. 16a. Clearly the additional indirect effects of ambient density on spray penetration through spray dispersion have been accounted for by the tan ( $\theta/2$ ) term. Wakuri *et al.* [13], following the analysis approach in Appendix C, arrived at a penetration correlation valid for long times with the dispersion angle term in it, but it has largely been overlooked.

The results and discussion above demonstrate that ambient gas density, through dispersion, has a larger effect on spray penetration than previously noted. With the scaling of the additional effect understood, the implications can be estimated from the correlation. For a turbocharged diesel in which TDC densities might vary by as much as a factor of three during operation, the additional density effects correspond to more than a 20% larger variation in the spray timing over the operating range. For advanced engines with greater turbocharging, the effect will be greater. For example, with an intake pressure boost of five atmospheres and a 5:1 variation in TDC gas densities, the additional density effect corresponds to a 35% larger variation in the spray timing.

FURTHER EXAMINATION OF THE EFFECTS OF VAPORIZATION ON SPRAY PENETRATION — All of the vaporizing penetration data obtained in this work at an ambient temperature of 1000 K are plotted in Fig. 20 in the scaled coordinates of Fig. 17a. Also shown in Fig. 20 are the penetration correlation and the  $\pm 10\%$  bands from Fig. 17a, and a mean equivalence ratio relationship that was developed along with the penetration correlation in Appendix C (see Fig. C2). The equivalence ratio curve gives the equivalence ratio,  $\phi$  at the tip of spray located at  $\tilde{S}$  at time  $\tilde{t}$ . The equivalence ratio on spray penetration.

Figure 20 shows that the penetration length and time scales given by EQs (5) and (6) collapse the penetration data from the vaporizing experiments as well as they collapsed the non-vaporizing data in Fig 17b. The scaled penetration data lie in a narrow band near the curve for EQ (10). Comparing Figs. 17b and 20, the only major difference noted between the scaled non-vaporizing data and the vaporizing data is that the vaporizing data lie below the correlation after a  $\tilde{t}$  of 0.5. This difference is the result of the reduction in penetration by vaporization that was noted in the physical coordinates of Fig. 16b.

The trends in Fig. 20 are shown in more detail in Fig. 21. Plotted in Fig. 21 as a function of dimensionless penetration distance is the average percent difference between the data in Fig. 20 and EQ (10). Also shown for comparison is the equivalent difference for the non-vaporizing data previously shown in Fig. 18. The standard deviation of the difference plotted for the vaporizing data is the same as was shown for the non-vaporizing data in Fig. 18, approximately  $\pm 10\%$ .

Figure 21 shows that the percent reduction in penetration relative to EQ (10) as a result of vaporization increases from



Figure 20. All of the vaporizing penetration data taken at 1000 K (the symbols) versus time in dimensionless coordinates. Also shown is EQ (10) (the solid curve) and the  $\pm 10\%$  deviation band (the dashed lines) from Fig. 17a, and the equivalence ratio relationship (the dotted curve) from Fig. C2 in Appendix C.



Figure 21. The average percent difference between the data in Fig. 20 and EQ (10) (the dashed curve) versus dimensionless penetration distance. Also show is the corresponding non-vaporizing data from Fig. 18 (the solid curve). The line at zero percent is for reference.

a negligible value at short penetration distances to a maximum of 18% near an  $\tilde{S}$  of 1.5, then decreases with longer penetration distances. Relative to the non-vaporizing data, the trends are similar, except that the percentage reduction in penetration reaches a maximum of 21% between  $\tilde{S}$ 's of 0.8 and 1.5. The decrease in the <u>percentage</u> reduction in penetration from vaporization effects after an  $\tilde{S}$  of 1.5, however, does not mean that the absolute difference between the nonvaporizing and vaporizing penetration data is decreasing. The rate of the decrease in the percentage reduction with increasing  $\tilde{S}$  (after an  $\tilde{S}$  of 1.5) is such that the absolute difference between the two data sets continues to grow, but at steadily declining rate. These observations strongly imply that vaporization process leading to the reduction in penetration is complete by an  $\tilde{S}$  of 1.5. The length scale given by EQ (5) and the results in Fig. 21 imply two other trends. First, the overall magnitude of the penetration reduction by vaporization will increase with increasing orifice diameter, increasing fuel density, decreasing ambient gas density, or decreasing dispersion angle. Second, for the same variation in the parameters just described, the actual penetration distance corresponding to an  $\tilde{S}$  of 1.5 will move further from the injector tip. Some of these trends with respect to ambient gas density are visible in Fig. 16b.

Potential factors contributing to the reduced penetration of the vaporizing sprays are related to those discussed previously with the spray dispersion results. First, a volumetric contraction of the vaporizing spray as the entrained hot gases are cooled by the evaporating fuel results in a more dense mixture. Newly injected fuel and entrained air must then penetrate through the higher density mixture as they move toward the head of the spray, thus slowing penetration. Second. evaporation of large ballistic drops (i.e., droplets not following the gas phase motion) at the leading edge of the spray and redistribution of their momentum through mixing results in a decrease in the spray penetration rate. The second effect was clearly visible in films at the lower gas densities conditions. For non-vaporizing low density conditions, droplet streaks were visible at the head of the spray, but they were not visible in the films for the corresponding vaporizing conditions.

Support for and explanation of the vaporization results and scalings observed in Figs. 20 and 21 can be derived from recent liquid phase penetration measurements in an engine by Espey and Dec [43] and from recent similar measurements we have made in the combustion vessel (in progress). In both investigations, the liquid phase penetration of the injected fuel reached a maximum length that depended on ambient gas density and temperature. Beyond the maximum length, only vapor was present. When the maximum liquid phase penetration lengths measured at a fixed temperature in each experiment are scaled with the length scale given in EQ (5), the result is a single dimensionless number representing the maximum liquid penetration depth. The number from the Espey and Dec experiment derived from data at a temperature of 1000 K over an ambient gas density range from 11 to 33 kg/m<sup>3</sup> was  $\tilde{S}=1.9\pm0.1$  ( $\pm 2\sigma$ ). The number from our data taken at a temperature of 1000 K over an ambient gas density range from 3.3 to 60 kg/m<sup>3</sup> was  $\tilde{S}=1.5\pm0.1$  ( $\pm 2\sigma$ ). Both locations are marked in Fig. 21 on the line at 0% deviation. They correspond to the location where the percent reduction in penetration from vaporization has reached a maximum. Thus, the vaporizing spray penetration data and liquid phase penetration data are consistent and indicate vaporization is complete by an  $\tilde{S}$  of approximately 1.5 for a #2-like diesel fuel and an ambient temperature of 1000 K. The 25% difference between the dimensionless liquid tip penetration lengths from each of the experiments could be the result of several differences between the experiments that include differences in fuel properties, injectors, and injector tips.

Further support for the results in Figs. 20 and 21 can be derived from an estimation of the mass of hot air that is required to completely evaporate a mass of cold fuel. The estimation can be made either based on a simple energy balance that accounts for the amount of air that must be cooled to the

fuel vaporization temperature to completely vaporize the fuel, or from the fuel/air ratios given by Kamimoto and Matsuoko [37] for complete evaporation of fuel under conditions of adiabatic saturation. Using either technique and expressing the answers in the form of an equivalence ratio, equivalence ratios between 15 and 25 result for most diesel-like fuels. From the equivalence ratio curve in Fig. 20, this equivalence ratio range corresponds to  $\tilde{t}$  between 0.6 and 1 or S's be tween 0.5 and 0.7. This  $\hat{S}$  range for complete fuel evaporation is shorter than that derived from the liquid penetration measurements discussed in the previous paragraph. However, given that the perfect mixing assumption involved in this type of estimate does not apply, a longer measured distance for complete fuel evaporation is expected. As a result, the agreement is reasonable, and therefore, supportive of not only the evaporating spray penetration measurements presented in this work, but liquid penetration measurements discussed in the previous paragraph.

The discussion in this section was based on data taken at an ambient gas temperature of 1000 K with #2 diesel fuel. Ambient temperature and fuel properties are expected to have effects on these trends, especially at the lower density conditions of this experiment. Further research is needed regarding the effects of these parameters.

COMPARISON WITH SPRAY PENETRATION COR-RELATIONS IN THE LITERATURE — Using the short and long time limits of EQ (10) given by EQs (11) and (12), comparisons can be made with spray penetration correlations in the literature and the resulting dependencies of spray penetration on various parameters. However, direct comparison to all the correlations that have been presented would be difficult. We have chosen a few of the correlations commonly cited in text books [*e.g.*, 6,10-12] for direct comparison and will make some general observations concerning others.

In the short time limit, the only correlation presented specifically for short times is by Hiroyasu and Arai [2]:

$$S = 0.39 \cdot \sqrt{2 \frac{(P_f - P_a)}{\rho_f}} \cdot t.$$
(13)

If the short time limit for EQ (10) (*i.e.*, EQ (11)) is recast in dimensional form, the result is:

$$S = C_{v} \cdot \sqrt{2 \frac{(P_f - P_a)}{\rho_f}} \cdot t.$$
(14)

The form of these two equations are the same. They state that the early penetration is proportional to the product of the velocity of the liquid exiting the injector and time, a very intuitive result. However, EQ (14) indicates that initial spray penetration period will be affected by the injector tip design through  $C_{\nu}$ .  $C_{\nu}$  in our work varied from 0.68 to 0.76, which results in almost a factor of two greater penetration than is given by EQ (13). An additional parameter that will impact this comparison is the injector opening time. A long opening time with throttled flow during that time will result in a lower apparent value for  $C_{\nu}$ , if  $C_{\nu}$  is determined by a fit of EQ (14) to the penetration data. (Note: In this work,  $C_{\nu}$ 's were measured independently as described in Appendix A and the injector opening time was short.) In the long time limit, two of the more common correlations cited are the correlation of Hiroyasu and Arai [2] developed from data in the ambient gas density range from 10 to  $33 \text{ kg/m}^3$ ,

$$S = 2.95 \cdot \sqrt{\sqrt{\frac{(P_f - P_a)}{\rho_a}}} \cdot d_o \cdot t.$$
(15)

and Dent's [8], developed with data from many experiments (including engine and combustion vessel studies) in ambient gas densities in the range from 1 to  $23 \text{ kg/m}^3$ ,

$$S = 3.07 \cdot \left(\frac{294}{T_a}\right)^{\frac{1}{4}} \sqrt{\sqrt{\frac{(P_f - P_a)}{\rho_a}}} \cdot d_o \cdot t.$$
(16)

These can be compared to EQ (12), which in dimensional form gives:

$$S = \sqrt{\frac{C_v \cdot \sqrt{(2 \cdot C_a)}}{a \cdot \tan(\theta / 2)}} \cdot \sqrt{\sqrt{\frac{(P_f - P_a)}{\rho_a}}} \cdot d_o \cdot t.$$
(17)

The three correlations show the same dependence of penetration on orifice diameter, time, injection pressure, and ambient gas density and pressure. The major differences between the correlations arise from the  $tan(\theta/2)$  term and the detailed orifice parameters in EQ (17), and the temperature term in Dent's correlation in which  $T_a$  is the ambient gas temperature in Kelvin.

Direct comparisons of the three correlations can be made by assuming representative values for the parameters not common to all three correlations. The temperature  $T_a$  is assumed to be 294 K (non-vaporizing) and the value of *a* is 0.66 as discussed previously. The values selected for  $C_v$  and  $C_a$  were the average values from this experiment: 0.73 and 0.83, respectively. The angles selected were  $\theta/2 = 10^\circ$  for the comparison with the correlation of Hiroyasu and Arai and  $\theta/2 = 9^\circ$  for the comparison with Dent's. These angles are angles measured in this work for the average ambient gas density condition of the data on which each of the other correlations are based (see density ranges cited with EQs (15) and (16)).

With the above values, the leading square root term in EQ (17) becomes 2.9 for the comparison with EQ (15) and 3.0 for the comparison with EQ (16); remarkable agreements given the wide range of conditions on which the three correlations are based. However, EQs (15) and (16) do diverge from EQ (17) for ambient gas densities less than or greater than about 15 to 25 kg/m<sup>3</sup> because of the effects of ambient gas density on dispersion (i.e., the tan( $\theta/2$ ) term in EQ (17)). As shown previously in Figs. 19a and 19b, the tan( $\theta/2$ ) term is needed to account for the effects of ambient gas density on penetration through its effect on dispersion.

Another interesting comparison that can be made to the results of Hiroyasu and Arai [2] is a comparison of the time at which a transition is made from a linear dependence of penetration on time to a square root dependence. Hiroyasu and Arai proposed a transition time they called the "breakup time",  $\tau_b$ :

$$\tau_b = 28.7 \cdot \frac{d_f \cdot \overline{\rho}^{\frac{1}{2}}}{\sqrt{\frac{(P_f - P_a)}{\rho_f}}}$$
(18)

Their "breakup time" can be compared to the transition time in this work at  $\tilde{t} = 1$ ; the time when the spray transitions from being dominated by injected liquid to being dominated by entrained gas (see Fig. 17a). Written in dimensional form,  $\tilde{t} = 1$  gives a transition time,  $t_r$ , of:

$$t_r = \frac{\sqrt{C_{a/2}}}{C_v \mp \cdot a \cdot \tan(\theta)} \cdot \frac{d_f \cdot \overline{\rho}^{1/2}}{\sqrt{\frac{(P_f - P_a)}{\rho_f}}}.$$
 (19)

The correlations again have similar terms. Substitution of the values that were used previously to compare EQs (15) and (17) into the terms in EQ (19) results in a value of 7.6 for the leading group of terms in EQ (19). The difference between the coefficient of 28.7 in EQ (18) and 7.6 in EQ (19) is a factor of 3.8. This difference places the breakup time of Hiroyasu and Arai at a  $\tau$  of 3.8 in Fig. 17b or an  $\tilde{S}$  of 2.0. An  $\tilde{S}$  of 2.0 corresponds closely with location at which the effects of ballistic droplets have dissipated in Fig. 18, based on the results of our experiment.

Some final important observations concerning a number of penetration correlations in the literature relative to the correlation in Fig. 17a can be made. Consider hypothetical penetration data sets that span different regions of dimensionless time in Fig. 17a. A best fit of a straight line to the data in each region in the log-log coordinates of the figure would give a penetration correlation appropriate for each region of time. These correlations would be similar to EQs (11) and (12), except that the exponent would vary between 0.5 and 1.0 depending on the region of dimensionless time covered by the data set. When put in dimensional form (e.g., EQs (14) and (17)), these correlations would include the various parameters in the length and time scales given by EQs (5) and (6). The dependence of penetration on each of the parameters in the correlations would be simple power law dependencies.

The major parameters that would be included in the hypothetical correlations are given in the first column in Table 3. The ranges for the exponents in the power law dependencies of penetration on each of these parameters are given in the second column. The limits for the ranges are given by the dependencies in EQs (14) and (15), where the dependence of  $\tan(\theta/2)$  on  $\rho_a/\rho_f$  in Fig. 14a have been substituted for  $\tan(\theta/2)$ . The arrows in the second column indicate the direction in which the exponent value for each dependence would change as the region of time covered by the data moved from the short time region toward the long time region in Fig 17a.

The exponents in the third column in the table for each parameter dependence are derived from the various correlations in literature [1,2,4-61. Listed in the third column are the ranges for the exponents based on the literature (indicated by the double arrows) and any significant outliers to the range in parentheses. The outliers are exponent values that differed from the average exponent for the corresponding range by more than two standard deviations.

Comparison of the ranges in columns two and three shows that there is good agreement between the ranges derived from Fig. 17a and from the literature with two exceptions shown in parentheses. The exceptions are a 0.82 power dependence on orifice diameter from a work referenced by Hay and Jones [1] and a 0.8 power dependence on fuel density from Ref [4]. Both of these works included fuel viscosity in a Reynolds number term in their penetration correlations, but neither work varied viscosity. The Reynolds number term, which also includes the fuel density and orifice diameter, turned out to be a large term after fitting the respective correlations to data.

Although the set of correlations from the literature on which the third column is based is not a complete set, we believe that it is representative. Overall, the trends in Table 3 suggest that the vast body of data in the literature is in agreement, since the ranges for the parameter dependencies from the various correlations fit to the data are in good agreement. However, the table and Fig. 17a also suggests a reason for much of the scatter noted by Hay and Jones [1] and others in the penetration predicted by various correlations based on that same body of data. Namely, correlations that contain a fixed dependence on each parameter cannot accurately reflect the change in the dependencies that occur as the spray penetrates. Extrapolation of such correlations to conditions different than the data base on which they were developed could easily result in the large variation in penetration predicted by correlations in the literature. To be accurate, a correlation must account for the change in the dependence of penetration on various parameters that occurs between the short time limit and the long time limit. The correlation given by EQ (10) does this very well.

**Table 3.** A comparison of the dependence of penetration on various parameters derived from Fig. 17a with those noted in the literature. The first column lists the parameters with the dependence of penetration on each parameter expressed in the form of a power law. The second column gives the variation of the exponent *n* in the power law derived from Fig. 17a as  $\tilde{t}$  varies from 0 to  $\infty$ . The third column contains the range for each exponent derived from correlations in the literature. The values in parentheses in column three differ from the average exponent for the corresponding range by more than two standard deviations. As a result, they have not been explicitly included in the ranges.

	$n$ based on Fig.17a as <i>t</i> varies from $0 \rightarrow \infty$	<i>n</i> based on literature [1,2,4-6]
$t^n$	1.0→0.5	$1.0 \leftrightarrow 0.48$
$d_o{}^n$	0.0→0.5	$0.0 \leftrightarrow 0.5(0.82)$
$(P_f - P_a)^n$	0.5→0.25	$0.5 \leftrightarrow 0.25$
$\rho_a{}^n$	0.0→0.35	$0.0 \leftrightarrow -0.5$
$ ho_{ m f}^{\ n}$	-0.5→0.1	$-0.5 \leftrightarrow 0.26(0.8)$

#### SUMMARY AND CONCLUSIONS

Ambient gas density (*i.e.*, in-cylinder air density) and fuel vaporization effects on diesel spray penetration and dispersion were investigated over a gas density range that greatly extends the present data base in the literature. The emphasis was on high density and high injection pressure conditions applicable to the next generation of advanced, high efficiency diesel engines. The results and observations provide new insight on the effects of in-cylinder gas density and fuel vaporization on spray penetration and dispersion that will aid the design of these advanced engines. In addition, the results provide a simple well characterized baseline for the global effects of these parameters on sprays that can be used for comparison to multidimensional model predictions and for development of their submodels.

The experiments were carried out in a constant-volume combustion vessel with line-of-sight optical access. Diesel sprays injected into the vessel were recorded with time resolved extinction/schlieren imaging techniques. The recorded images were then analyzed for spray penetration and dispersion information. The injector and fuel used were a high-pressure, common-rail diesel fuel injector and a standard #2 diesel fuel. The injector had a fast opening time and a constant injection rate.

Ambient gas density effects on sprays were examined by comparing injections over a wide range of ambient gas density. Fuel vaporization effects were examined by comparing injections into non-vaporizing (*i.e.*, cold) and vaporizing (*i.e.*, hot) inert environments at the same density. The ambient gas density ranges considered were 3 to 200 kg/m<sup>3</sup> and 3 to 61 kg/m<sup>3</sup> for the non-vaporizing and vaporizing environments, respectively. The vaporizing spray experiments included gas densities more than a factor of two higher than TDC gas densities in current technology heavy-duty diesels (*i.e.*, 10 to 30 kg/m<sup>3</sup>). Other parameters varied were the orifice diameter, the injection pressure and the ambient gas temperature.

As part of the data analysis, characteristic time and length scales for penetration and a theoretical penetration correlation for non-vaporizing sprays were developed. The characteristic time and length scales collapsed all the penetration data taken into two distinct non-dimensional penetration curves: one for the non-vaporizing conditions and one for the vaporizing conditions. The theoretical penetration correlation was derived using the penetration time and length scales and a simple model for a non-vaporizing spray previously presented in the literature. Non-vaporizing data from this experiment, as well as other commonly quoted penetration data sets, are in good agreement with the correlation. The correlation also provided a potential explanation for much of scatter noted in the penetration predicted by various correlations in the literature.

The analysis of the results shows that ambient gas density has a significantly larger effect on spray penetration and a smaller effect on spray dispersion than has been previously noted. The larger effect on penetration is caused by ambient gas density effects on spray dispersion (*i.e.*, air entrainment). As ambient gas density increases, spray dispersion increases, which results in more entrained air in the spray. The larger entrained mass leads to a slower penetration velocity based on conservation of momentum, and therefore, reduced penetration. The penetration time and length scales and the coffelation developed include a term that accounts for this increased effect of ambient gas density on penetration through dispersion. The effect is significant, resulting in a 20% greater variation in spray timing than the current penetration correlations would predict over the range of operating conditions for a turbocharged heavy-duty diesel.

The results also show that vaporization reduces penetration and dispersion by as much as 20% relative to nonvaporizing sprays, but both of these effects decrease with increasing ambient gas density. Comparison of the scaled vaporizing and non-vaporizing data helped isolate and explain the effects of vaporization on penetration. Based on the scaled penetration data, the percent reduction in penetration relative to a non-vaporizing spray was found to be a maximum at the location in the spray where fuel vaporization is complete. This reduction in penetration is hypothesized to be the result of an increase in the density of the gas mixture in the spray as it is cooled by evaporating fuel. The higher density mixture slows newly injected fuel progressing through the spray to the tip more rapidly, thus slowing the tip penetration. The increase in density also leads to a contraction of the spray that explains the reduced spray dispersion angle.

#### NOMENCLATURE:

- a constant with a value of 0.66 or a density dependent term in EQ (2)
  b a constant with a value of 0.005 in EQ (2)
  A area
  A constant area of the "real" spray
- $C_a$ , orifice area contraction coefficient
- $C_d$  orifice discharge coefficient
- $C_{\nu}$ , orifice velocity coefficient
- d diameter
- f fuel mass fraction
- *l* length of the orifice
- *m* mass or a constant equal to 0 or 1
- $\dot{m}$  mass flow rate
- $\dot{M}$  momentum flow rate
- MW molecular weight
- N exponent
- P pressure
- $P_{o_2}$  percent oxygen in the products of the reaction given

by EQ (1)

- *r* radial boundary of the spray
- $\Re$  universal gas constant
- *S* penetration distance
- t time
- *t*<sub>open</sub> time to reach the full injection rate
- $t_r$  transition time (t=1)
- T temperature
- *u* axial velocity
- U axial velocity
- *w* width of the spray
- x axial coordinate of the sprayy radial coordinate of the spray

### Z compressibility factor

#### Greek:

- $\Delta$  incremental change
- φ equivalence ratio or diameter
- $\theta$  full cone angle of the spray
- ρ density
- *s* standard deviation
- $\tau_b$  breakup time [2]
- $\xi$  non-dimensional radial coordinate in the spray *ylr(x)*

# Subscripts:

- *a* ambient gas (*i.e.*, core gas in the combustion vessel)
- *b* bulk (mass averaged)
- *c* core of the combustion vessel (the inner 90%)
- f fuel
- *i* orifice minimum diameter
- *l* local
- *m* centerline
- meas measured values
- *o* orifice exit
- p projected
- *ref* reference condition
- rms root-mean-square
- w wall

#### Superscripts:

- coordinates referenced to the projected origin of the spray
- + characteristic penetration length or time
- non-dimensional coordinates

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#### APPENDIX A: MEASUREMENT OF THE INJECTION RATE PROFILE AND THE ORIFICE COEFFICIENTS

INJECTION RATE PROFILE — The mass flow rate characteristics of the injector under various conditions were obtained by independently measuring (a) the relative momentum of the fuel spray (*i.e.*, a signal proportional to injection velocity squared) and (b) the total mass of fuel injected. The relative spray momentum was measured with a linear force transducer (a Kistler 6121a piezoelectric pressure transducer) placed approximately 2.0 mm in front of the orifice, normal to the orifice centerline [44]. The total mass of fuel per injection was measured by weighing the mass of fuel injected from a number of injections (~50).

The first step in determining a mass flow rate profile from these two measurements was to normalize the square root of the linear force transducer's output by the area under the curve for the square root of the output. The normalization resulted in a profile proportional to the injection velocity (*i.e.*, injection rate). The product of this normalized profile and the independently measured total mass of fuel per injection results in the injection rate profiles shown in Fig. 6.

ORIFICE COEFFICIENTS — To characterize both the momentum and mass flux from an orifice, a discharge coefficient ( $C_d$ ) and an area contraction coefficient ( $C_a$ ) or velocity coefficient ( $C_v$ ) are required. The discharge coefficient characterizes the mass flux from an orifice, while the area contraction or velocity coefficients allow the momentum flux to be characterized as well. The three coefficients are related by the following definition:

$$C_d = C_a \cdot C_v \cdot . \tag{A1}$$

The area contraction coefficient accounts for flow area loss as a result of vapor bubbles generated by cavitation [39] or "hydraulic flip" [40], and for the effects of non-uniform velocity profiles at the orifice exit. A contraction coefficient of less than one results in a higher fluid velocity at the orifice exit than would be given by a mean flow velocity calculated with the discharge coefficient. This later point was shown by recent fuel velocity measurements in an orifice operating under diesel conditions with "super cavitating" flow [40].

The discharge coefficients  $(C_d)$  listed in Table 2 were determined for two different sets of conditions using the standard definition for discharge coefficient:

$$C_d = \Delta m / \left( \Delta t \cdot A_o \cdot \rho_f \cdot \sqrt{2 \cdot (P_f - P_a)} / \rho_f \right).$$
(A2)

The term  $\Delta m$  is the mass of fuel injected over a time  $\Delta t$ ;  $\rho_f$  is the fuel density;  $A_o$ , is the orifice exit area; and  $P_f$  and  $P_a$ , are the fuel and ambient (*i.e.*, in-cylinder) pressures, respectively. The first set of conditions covered actual injection pressures, durations and fuel temperatures, and in a few cases, actual ambient pressures. Under these conditions, the injection rate profile (determined as previously described in this appendix) was used to correct for finite opening and closing time effects on the discharge coefficient. The second set of conditions covered lower injection pressures (8-14 MPa), longer injection durations (-10-60 seconds) and injection into atmospheric pressure air. The longer injection durations made opening and closing time effects negligible. The discharge coefficients for each set of conditions agreed within 2%.

The area contraction coefficients were determined by comparing the momentum flow rate of a spray from an orifice during the steady portion of the spray (see Fig. 6) to the momentum of a spray generated by a reference orifice with a  $C_a$ , value of one. The momentum flow rate was measured with a linear force transducer placed in front of the orifice as previously described in this appendix. The reference orifice selected had a high discharge coefficient (0.8), a long length-todiameter ratio (8.0) and a smooth entrant region into the orifice. This reference orifice was selected because it was least likely to have cavitation, or if cavitation occurred at the vena contracta, least likely to have cavitation bubbles survive to the exit of the orifice.

A relationship for the area contraction coefficient of an orifice can be developed from the relationship between the various orifice coefficients given EQ (Al), and the following definitions for mass flow rate (m), velocity  $(U_f)$ , and momentum flow rate (M) from an orifice,

$$\dot{m} = C_d \cdot A_o \cdot \rho_f \cdot \sqrt{2 \cdot (P_f - P_a) / \rho_f}, \qquad (A3)$$

$$U_f = C_v \cdot \sqrt{2 \cdot (P_f - P_a)} / \rho_f, \qquad (A4)$$

$$\dot{M} = \dot{m} \cdot U_f \,. \tag{A5}$$

Solving EQs. (Al) through (A5) for  $C_a$  and normalizing the result by the reference orifice  $C_{a.ref}$  yields:

$$C_{a} = C_{a.ref} \cdot \frac{\dot{M}_{ref}}{\dot{M}} \cdot \frac{A_{o}}{A_{o.ref}} \cdot \frac{\left(P_{f} - P_{a}\right)}{\left(P_{f} - P_{a}\right)_{ref}} \cdot \left(\frac{C_{d}}{C_{d.ref}}\right)^{2} \cdot$$
(A6)

If  $C_{a.ref}$  is known and the momentum sensor used to measured  $\dot{M}$  is linear with zero offset, EQ (A6) can be applied without calibration of the momentum sensor.

Typical values measured for  $C_a$  for various tips ranged from 0.8 to 1.0. Tips with orifices out the end of the tip (on

#### **APPENDIX B: EXPERIMENTAL CONDITIONS**

Shown in Tables BI and B2 are the experimental conditions and measured spray angles for the non-vaporizing and vaporizing experiments, respectively. The experimental conditions listed in the tables are defined at the right. All the non-vaporizing experiments were conducted with  $N_2$  as the ambient gas except for the cases shown in bold type. Those in bold type were conducted in an ambient gas composed of 60% He and 40%  $N_2$ . The vaporizing experiments were con-

 
 Table B1.
 Non-Vaporizing Experimental Conditions and Measured Spray Angles.

ρ <sub>a</sub>	$T_a$	Ζ	$T_w$	$\rho_f$	$P_f$	$d_o$	topen	$\tan(\theta/2)$
3.57	455	1.002	451	705	138	0.257	70	0.109
4.16	300	1.000	300	834	143	0.340	77	0.153
7.26	451	1.003	451	705	137	0.257	70	0.124
14.8	451	1.024	451	705	141	0.198	70	0.145
14.8	451	1.024	451	705	141	0.198	70	0.143
14.8	452	1.007	451	705	140	0.257	70	0.142
16.3	300	0.999	300	834	145	0.340	77	0.198
27.9	300	1.000	300	834	144	0.340	77	0.212
29.9	451	1.015	451	705	142	0.257	70	0.170
30.0	451	1.015	451	705	139	0.257	70	0.168
30.0	451	1.015	451	705	141	0.257	70	0.168
30.0	451	1.015	452	704	76.1	0.257	120	0.170
30.0	451	1.015	451	705	109	0.257	90	0.168
30.0	451	1.015	451	715	141	0.198	70	0.166
30.0	451	1.015	451	706	142	0.198	70	0.162
30.0	451	1.054	451	705	149	0.198	65	0.169
30.0	452	1.015	451	705	140	0.257	70	0.177
30.0	452	1.015	451	706	77.8	0.198	120	0.165
30.0	452	1.015	451	707	107	0.198	90	0.162
39.6	300	1.000	300	834	142	0.340	77	0.224
51.9	300	1.000	300	834	142	0.340	77	0.228
60.6	451	1.034	451	704	142	0.257	70	0.193
60.9	451	1.034	451	705	144	0.257	70	0.186
60.9	451	1.034	451	705	78.8	0.257	120	0.200
60.9	451	1.034	452	705	111	0.257	90	0.200
61.0	451	1.034	451	706	149	0.198	65	0.189
63.7	300	1.002	300	834	143	0.340	77	0.237
75.3	300	1.005	300	834	144	0.340	77	0.264
86.5	300	1.007	300	834	139	0.340	77	0.250
121	451	1.084	451	705	151	0.257	65	0.218
122	300	1.018	300	834	141	0.340	77	0.258
124	452	1.088	452	704	155	0.257	65	0.212
145	300	1.027	300	834	141	0.340	77	0.288
196	300	1.055	300	834	134	0.340	80	0.273

axis) had  $C_a$  values between 0.9 and 1.0 with values decreasing as orifice length-to-diameter ratio decreased. Tips such as those in Table 2 with the orifice 34° off axis had values closer to 0.8.

ducted in the products of the premixed bum given by EQ (2) with  $P_{o_2} = 0$ .

- $\rho_a$  ambient gas density (kg/m<sup>3</sup>)
- $T_a$  ambient gas temperature (K)
- *Z* compressibility factor
- $T_w$  vessel wall temperature (K)
- $\rho_f$  fuel density (kg/m<sup>3</sup>)
- $P_f$  fuel pressure (MPa)
- $d_0$  orifice exit diameter (mm)
- $t_{open}$  time to full injection rate (µs)
- $\theta/2$  mean spray dispersion half angle

Table B2. Vaporizing Experimental Conditions and Measured Spray Angles

$\rho_a$	$T_a$	Ζ	$T_w$	$\rho_f$	$P_f$	$d_o$	topen	$\tan(\theta/2)$
3.27	995	1.002	451	704	138	0.257	70	0.084
3.30	988	1.002	451	706	137	0.257	70	0.089
3.95	967	1.002	454	705	136	0.340	70	0.116
6.74	1008	1.005	452	704	138	0.257	70	0.104
6.76	1000	1.004	452	704	137	0.257	70	0.114
6.85	1010	1.004	451	705	139	0.257	70	0.107
13.9	1001	1.010	452	703	142	0.257	70	0.130
15.5	950	1.011	454	705	138	0.340	70	0.169
26.7	949	1.021	455	705	130	0.340	75	0.209
28.1	1401	1.023	450	703	147	0.257	65	0.171
28.2	1401	1.023	450	703	148	0.198	65	0.168
28.3	1201	1.023	450	703	145	0.257	65	0.165
28.6	1000	1.022	453	702	147	0.257	65	0.162
28.6	1003	1.022	451	702	77.9	0.257	120	0.160
28.6	1000	1.022	451	703	111	0.257	90	0.156
28.6	1000	1.022	452	701	148	0.198	65	0.155
28.6	1000	1.022	450	704	149	0.198	65	0.170
28.9	803	1.020	449	704	147	0.257	65	0.165
29.4	609	1.015	449	704	141	0.257	70	0.166
37.7	959	1.030	454	705	138	0.340	70	0.219
49.5	946	1.037	451	705	140	0.340	70	0.229
57.6	1405	1.052	451	701	160	0.257	62	0.197
57.9	1204	1.051	450	702	155	0.257	65	0.179
58.4	1004	1.049	451	701	152	0.257	65	0.179
58.4	1004	1.049	452	699	88.4	0.257	110	0.200
58.4	1002	1.049	451	699	121	0.257	80	0.195
58.4	1005	1.049	452	699	157	0.198	62	0.203
59.0	805	1.045	451	699	151	0.257	65	0.187
59.9	607	1.035	450	700	147	0.257	65	0.194
60.3	945	1.051	452	705	146	0.340	65	0.230

#### APPENDIX C: DERIVATION OF THE NON-VAPORIZING SPRAY PENETRATION CORRELATION

**INTRODUCTION** — The non-vaporizing transient spray penetration correlation used in the paper, EQ (10), is derived in this appendix. The derivation follows the spray penetration analyses of Wakuri et al. [13] and Hays [38], but with several modifications. The modifications include: (a) a nondimensionalization of the analysis that simplifies the analysis and that accounts for the effects of the ambient gas density, the fuel density, and various orifice parameters in a more complete manner; (b) an estimate for the arbitrary constant that appears in the penetration correlation based on fuel concentration and velocity profiles for turbulent two-phase jets; and (c) the development of an inverse relationship (i.e., penetration distance versus time) for the time versus penetration distance correlation resulting from the analysis. In addition, a relationship for the mean equivalence ratio as a function of axial distance in the "model" spray used to develop the penetration correlation is derived.

This paper demonstrates that the penetration correlation derived from this type of analysis accounts for the effects of a wide range of conditions on non-vaporizing sprays. The effects accounted for include an extremely wide ambient density range not handled by some of the more quoted correlations, such as those of Hiroyasu and Arai [2] and Dent [8]. The penetration correlation also has the linear dependence on time during the initial penetration period and the square root dependence on time during the later stages of penetration as previously noted by Hiroyasu and Arai [2].

DERIVATION — The method of analysis uses integral control surface techniques. It is applicable to isothermal/incompressible jets and sprays in which the injected and ambient fluids can be any combination of high and low density fluids. However, the derivation below is centered around the injection of a high density liquid fuel into a lower density ambient gas as it occurs under the non-vaporizing conditions described in this paper.

The development of the penetration correlation involves two steps: (1) derivation of a relationship for the spray tip velocity, and (2) integration of the velocity relationship to obtain a correlation for tip penetration time versus penetration distance. Figure Cl shows a schematic of the conceptual spray model used in the analysis to develop the spray tip velocity relationship. The "model" spray is defined as one with a uniform velocity profile at any x-location that has the same mass and momentum fluxes as the equivalent "real" spray at the same x-location.

The major assumptions made in the analysis are that the spray has: (a) a uniform velocity profile, (b) a constant injection velocity with an instantaneous start, (c) no velocity slip between the fuel and the entrained air, and (d) quasisteady flow with a uniform growth rate (*i.e.*, a constant angle  $\alpha$ ). The first assumption is a gross simplification of reality; however, the assumed uniform velocity condition can be related to realistic velocity and concentration profiles in a turbulent spray as will be shown. The second assumption excludes conditions under which the injection rate "ramps up" over a



Figure C1. Schematic of the spray model for the penetration correlation.

significant period of time or does not remain constant during the injection. The validity of the third assumption is doubtful near the tip, as is shown in the this paper, but holds for larger distances. The final assumption is supported by the axially uniform spray angles measured in the non-head portion of a spray in this work. In addition, this has been established for the non-head region of transient gas jets [33,34].

Using the control surface shown in Fig. Cl, fuel mass and overall momentum balances can be written from which the velocity at any x-location in the idealized spray can be determined. At first, the cross-sectional area occupied by the fuel in the spray is retained in the analysis for completeness, following a similar derivation by Wakuri *et al.* [13]. However, we will show that retaining the fuel area leads to significant complexity in the analysis and has only a minor, and possibly erroneous effect on penetration for typical diesel engine conditions. In the end, the area occupied by the fuel will be neglected.

The fuel mass balance and an overall momentum balance in the axial direction on the control surface shown in Fig. Cl are given by:

and

$$\rho_f \cdot A_f(\mathbf{O}) \cdot U_f = \rho_f \cdot A_f(\mathbf{x}) \cdot U(\mathbf{x}), \tag{C1}$$

$$\rho_f \cdot A_f(\mathbf{O}) \cdot U_f^2 = \rho_f \cdot A_f(x) \cdot U(x)^2 + \rho_a \cdot A_a(x) \cdot U(x)^2, \quad (\mathbf{C2})$$

respectively. The terms pf and p,, are the densities of the injected fuel and entrained ambient air, respectively.  $U_f$  and  $A_f(O)$  are the axial velocity and the cross-sectional area of the fuel at the exit of the orifice, respectively. U(x),  $A_f(x)$  and  $A_a(x)$  are the spray velocity and the cross-sectional areas of the fuel and air at any x location in the spray, respectively.

A third equation is added to EQs (C1) and (C2) by defining the cross-sectional area of the jet at x occupied by air as:

$$A_a(x) = A(x) - m \cdot A_f(x). \tag{C3}$$

The area A(x) is the total cross-sectional area of the jet at x and m is a parameter with a value of 0 or 1. Setting m to zero, as will be done later, is equivalent to neglecting the cross-sectional area of the spray occupied by the fuel.<sup>\*</sup>

Eliminating  $A_f(x)$  and  $A_a(x)$  from EQs (CI-C3) and solv ing for U(x) in terms of  $A_f(0)$ , A(x),  $U_f$  and the densities results in:

$$U(x) = \frac{U_f}{2} \cdot \frac{A_f(0)}{A(x)} \cdot \left(\frac{\rho_f}{\rho_a} - m\right)$$

$$\left(\sqrt{\frac{A(x)}{1 + 4} \cdot \frac{A(x)}{A_f(0)} \cdot \frac{\rho_f}{\rho_a}}{1 + 4 \cdot \frac{A(x)}{\left(\frac{\rho_f}{\rho_a} - m\right)^2} - 1}\right).$$
(C4)

This equation for the spray velocity applies at all x in the model" spray. If the following further substitutions and non-dimensionalizations are made,

$$\widetilde{\rho} = \rho_f / \rho_{a_i}, \qquad (C5)$$

$$x_a = \frac{m}{2} \cdot \frac{d_f}{\tan(\alpha/2)},\tag{C6}$$

$$x' = x + x_o , \qquad (C7)$$

$$\frac{dx'}{dt'} = \frac{dx}{dr} = U(x), \qquad (C8)$$

$$A(x) = \pi \cdot [x' \cdot tan(\alpha / 2)]^2, \qquad (C9)$$

$$A_{f}(0) = \frac{\pi}{4} \cdot d_{f}^{2}, \qquad (C10)$$

$$\mathbf{x}^{+} = \mathbf{d}_{f} \cdot \tilde{\rho}^{\frac{1}{2}} \left( \frac{\tilde{\rho} - \mathbf{m}}{\tilde{\rho}} \right) \cdot \frac{1}{\tan(\alpha / 2)}, \quad (C11)$$

$$T = d_f \cdot \tilde{\rho}^{\frac{1}{2}} \left(\frac{\tilde{\rho} - m}{\tilde{\rho}}\right)^2 \cdot \frac{1}{U_f \tan(\alpha/2)}, \qquad (C12)$$

$$\overline{x} = x' / x^+, \qquad (C13)$$

$$\bar{t} = t' / t^+, \qquad (C14)$$

then EQ (C4) simplifies to:

 $t^+$ 

$$\frac{\mathrm{d}\tilde{x}}{\mathrm{d}\tilde{t}} = \frac{2}{\sqrt{1+6\tilde{x}^2 = +1}},\tag{C15}$$

In the above equations, the distance  $x_o$  is the location of the orifice exit relative to the projected origin of the spray. The diameter  $d_f$  is the diameter of the fuel stream exiting the orifice (to be discussed later). The angle  $\alpha/2$  is defined in Fig. C1. The terms t' and x' are the coordinates referenced to the projected origin;  $t^+$  and  $x^+$  are time and length scales used to non-dimensionalize time and distance; and  $\tilde{t}$  and  $\tilde{x}$  are the non-dimensional time and length coordinates. The scaling parameters  $(t^+, x^+)$  are similar to those used by Hays [38] except for the addition of the  $(\tilde{\rho} - m) \sqrt{\rho}$  term and more refined definitions for  $d_f$  and  $U_f$  (to be discussed later). The product  $d_f \cdot \tilde{\rho}^{0.5}$  in  $t^+$  and  $x^+$  has long been used to correlate the effects of density differences between the injected and ambient fluids in steady jets [41] and transient sprays [42].

The final step in the derivation is to assume that U(x) is equal to the velocity of the spray tip and integrate EQ (C15) from  $\tilde{x} = 0$  to  $\tilde{x} = \tilde{S}$ , where  $\tilde{S} = S' / x^+$  and S' is the spray tip location relative to the projected origin. The integration yields the dimensionless penetration time,  $\tilde{r}$ , as a function of the dimensionless penetration distance,  $\tilde{S}$ , both referenced to the projected spray origin in Fig. Cl:

$$\widetilde{t} = \frac{\widetilde{S}}{2} + \frac{\widetilde{S}}{4} \cdot \sqrt{1 + 6 \cdot \widetilde{S}^2} + \frac{1}{16} \cdot \ln\left(4 \cdot \widetilde{S} + \sqrt{1 + 16 \cdot \widetilde{S}^2}\right). \quad (C16)$$

Figure C2 is includes a plot of EQ (C16) in dimensionless logarithmic coordinates.

Also shown in Fig. C2 are the short and long time limits of EQ (C16). The short time limit is derived by integrating



Figure C2. The dimensionless penetration correlation given by EQ (C16) and the short and long time limits of the correlation, EQs (CI7) and (C18). Also shown are the inverse correlation for EQ (C16) given by EQ (C19) and the equivalence ratio correlation given by EQ (C32). The gray region is the equivalence ratio range measured by Espey *et* al. [47] in the head of a spray in an engine.

<sup>\*</sup> Setting *m* to zero does not imply that the  $A_f(x)$  terms in EQs (C1) and (C2) are zero. Those terms result from considering the fuel mass and momentum fluxes in the analysis.

EQ (C15) in the limit of  $\tilde{t}$  approaching zero (*i.e.*  $\tilde{x}$  approaching zero):

$$\begin{array}{c|c} Limit \\ \tilde{t} = 0 \end{array} & \begin{array}{c} \tilde{S} = \tilde{t} \\ \end{array}$$
(C17)

The long time limit is derived by integrating EQ (C15) in the limit of t approaching infinity (*i.e.*, 1 approaching infinity):

$$\begin{array}{c|c} Limit\\ \tilde{t} = \infty \end{array} & \tilde{s} = \tilde{t}^{\frac{1}{2}} \end{array}$$
(C18)

As shown in the figure, the theory predicts a linear dependence of penetration on time for short times and a square root dependence on time for longer times as noted experimentally by Hiroyasu and Arai [2]. The transition region from a linear to a square root dependence occurs in the vicinity of  $\tilde{t} = 1$ . In the transition region, the spray changes from one dominated by the injected fluid to one dominated by entrained air. With the exception of some small deviations in the transition region, the non-vaporizing results in this paper show good agreement with the correlation over nearly two orders of magnitude change in gas density.

AN INVERSE PENETRATION CORRELATION — Penetration time as a function of penetration distance as given by EQ (C16) is not always a convenient form, since it is not possible to explicitly solve for  $\tilde{S}$  in terms of  $\tilde{t}$ . However, a simple combination of the short and long term penetration limits given by EQs (C 17) and (C 18) above,

$$\overline{S} = \left[ \left(\frac{1}{\widetilde{t}}\right)^n + \left(\frac{1}{\widetilde{t}^{1/2}}\right)^n \right]^{-1/n} = \widetilde{t} / \left(1 + \widetilde{t}^{n/2}\right)^{1/n}, \qquad (C19)$$

provides an accurate inverse correlation. With a value of n equal to 2.2, EQ (C19) fits EQ (C16) to within 5% in the transition region. The agreement in the limits of short and long time is exact. A plot of EQ (C19) is included in Fig. C2.

THE FUEL FLOW AREA — Retaining the fuel flow in EQ (C3) in the analysis results in the origin for EQ (C16) being offset from the orifice exit as is shown in Fig. Cl. As a result,  $x_a$ , and  $t_a$ , must be subtracted from S' and t' computed with EQs (C5) through (C19) to reference the penetration distance and time back to the orifice exit. The distance  $x_o$ , is defined by EQ (C6). The time  $t_o$  is the time for the spray to penetrate from the spray projected origin in Fig. Cl to the orifice exit. It is evaluated from EQ (C16) at  $\tilde{s} = x_o / x^+$ . For the conditions of this experiment, typical values for  $x_o$ , and  $t_o$ , are 0.5 mm. and 3 µs.

Another consequence of retaining the fuel flow area in EQ (3) is that the spray tip velocity at the orifice exit is equal to  $U_f$ , as it should be, but it is already decelerating. This incorrect physical outcome results from the fact that "air entrainment" occurs on the part of the model spray inside the orifice (see Fig. Cl). The model spray effectively starts with a velocity at the projected spray origin higher than at the orifice exit to compensate for the ensuing velocity loss by air entrainment up to the orifice exit.

Given the small size of  $x_o$ , and  $t_o$ , relative to the resolution of the data, the incorrect physical outcome of air entrainment inside the nozzle, and the significant complexity that results from retaining the fuel flow area, we have chosen to neglect fuel flow area in EQ (3) from further consideration. This is accomplished by setting m=0 in EQs (C3) through (C15) of the analysis. The terms  $(\overline{\rho} - m)\sqrt{\rho}$  in the penetration length and time scales in EQs (C11) and (C12) become one, and the spray origin distance and time offsets from the orifice exit,  $x_o$ , and  $t_o$  become zero. The net result is that the projected spray origin in Fig. C1 shifts to the orifice exit plane and the injected fuel is treated as a point source of momentum.

THE ORIFICE PARAMETERS — The velocity  $U_f$  and effective orifice diameter  $d_f$  in EQs (C5-C7), (C11-C14), should be determined using the following relationships:

$$U_f = C_v \cdot \sqrt{2 \frac{\left(P_f - P_a\right)}{\rho_f}},\tag{C20}$$

$$d_f = C_a^{\frac{1}{2}} \cdot d_o. \tag{C21}$$

Equation (C20) is the standard relationship for the fluid velocity at the exit of an orifice, where  $P_f$  is the fluid pressure,  $P_a$ , is the ambient gas pressure, and  $C_v$ , is the coefficient of velocity contraction accounting for head losses through the orifice. Equation (C21) relates the effective fluid flow diameter at the orifice exit to the physical orifice diameter  $d_o$ ,  $C_a$ , is the orifice coefficient of area contraction and accounts for flow area loss, such as by cavitation bubbles reaching the orifice exit [39] or by "hydraulic flip" [40]. (See Appendix A for further discussion on the orifice coefficients.)

RELATIONSHIP OF  $\alpha$  TO THE MEASURED SPRAY ANGLES — The angle  $\alpha$  in the "model" must be related to typical angles measured for "real" sprays to utilize the correlation. An angle that is commonly measured in transient spray studies is the angle defining the outer boundary of a spray, such as those measured in this work. To establish a relationship between  $\alpha$  and the outer boundary spray angle ( $\theta$ ), self-preserving flow can be assumed in the spray. With this assumption, the tangent of the two angles are related by a constant:

$$\tan(\alpha) = a \cdot \tan(\theta) \cdot \tag{C22}$$

Equation (C22) introduces an arbitrary constant, a, into the spray penetration correlation given by EQ (C16) via EQs (C11) and (C12). A value for a equal to 0. 66 was found to provide close agreement between the non-vaporizing penetration data and EQ (C16). This is the value used for a in this paper. However, a can also be estimated from steady, turbulent, two-phase jet velocity and fuel concentration profiles. Use of the estimated value for a results in predictions of penetration that are within 5% of those using a value of aequal to 0.66. This close agreement, obtained without direct use of penetration data, adds validity to EQ (C16). The development of the estimate is provided in the next section of this appendix. ESTIMATION OF "a" in EQ (C22) — The constant a in EQ (C22) can be estimated by equating the density, the mass flow and the momentum flow at an x-location in the model spray in Fig. Cl to those of a "realistic" spray as follows:

$$\rho(x) \cdot A(x) = \rho_a \cdot \underline{A}(x) \cdot 2 \cdot \int_{0}^{1} (1 + f(x,\xi)) \cdot \xi d\xi, \qquad (C23)$$

$$\rho(x) \cdot U(x) \cdot A(x) =$$

$$\rho_{a} \cdot \underline{A}(x) \cdot 2 \cdot \int_{0}^{1} (1 + f(x, \xi)) \cdot u(x, \xi) \cdot \xi d\xi, \qquad (C24)$$

$$\rho(x) \cdot U(x)^2 \cdot A(x) =$$

$$\rho_a \cdot \underline{A}(x) \cdot 2 \cdot \int_0^1 (1 + f(x,\xi)) \cdot u(x,\xi)^2 \cdot \xi d\xi. \quad (C25)$$

The cross-sectional area A(x) and the velocity U(x) on the left in EQs (C23-C25) are defined in Fig. Cl. The density  $\rho(x)$ on the left is the average density in the model spray at an *x*-location. The terms  $\underline{A}(x)$ ,  $u(x,\xi)$  and  $f(x, \xi)$  on the right are the cross-sectional area, the velocity profile, and fuel/air mass concentration profile of the "real" spray, respectively. The coordinate  $\xi$  is the dimensionless radial coordinate defined by,

$$\xi = \frac{y}{r(x)},\tag{C26}$$

(C27)

where y is the local radial coordinate and r(x) is the outer boundary of the "real" spray at x.

The integrals on the right in EQs (C23-C25) are over the spray cross-sectional area,  $\underline{A}(x)$ , where:

 $A(x) = \pi \cdot r(x)^2,$ 

and

$$d\underline{\mathbf{A}}(\mathbf{x}) = 2 \cdot \boldsymbol{\pi} \cdot \mathbf{y} \, d\mathbf{y} = 2 \cdot \underline{\mathbf{A}}(\mathbf{x}) \quad \boldsymbol{\xi} d\boldsymbol{\xi}. \tag{C28}$$

These integrals are evaluated by inserting fuel concentration and velocity profiles, such as those used by Abramovich [45] to develop his two-phase spray theory:

$$f(x,\xi) = f_m(x) \cdot (1 - \xi^{1.5}),$$
 (C29)

$$u(x,\xi) = U_m(x) \cdot (1-\xi^{1.5})^2.$$
 (C30)

The terms  $f_m(x)$  and  $U_m(x)$  are the centerline fuel/air mass ratio and the centerline velocity of the "real" spray, respectively. The parameter  $f_m(x)$  varies from infinity at the spray origin to zero far downstream.

Evaluation of the integrals in EQs (C23-C25) after substitution of EQs (C29) and (C30) results in three algebraic equations that can be solved for the ratios  $\rho(x)/\rho_a A(x)/\underline{A}(x)$ , and  $U(x)/U_m(x)$  in terms of constants and  $f_m(x)$ . The square root of the ratio  $A(x)/\underline{A}(x)$  is the ratio of the radius of the "model" spray to the radius of the "real" spray at any *x*, or equivalently, the ratio of the tangent of the "model" angle  $\alpha$ and the tangent of the "real" spray outer boundary angle  $\theta$ (*i.e.*, the constant *a*). The value of *a* determined by this method is a function of the centerline fuel mass fraction  $f_m(x)$ , and therefore, varies with *x*. The variation is small, however. For  $f_m(x)$  varying from infinity to zero, the estimated *a* varies from 0.84 to 0.7. For a reasonable mean value for  $f_m(x)$  of 0.3, *a* is 0.73. The difference in penetration resulting from this value for *a* and the experimentally determined value of 0.66 is 5%. The disagreement between the experimentally determined value for *a* and the estimated value is very close to the disagreement noted for steady gas jets between experimentally measured entrainment rates and entrainment rates computed via similar methods to the above [46]. The disagreement in the steady gas jet case is attributed to the velocity profiles not adequately representing the intermittency of the outer edge of steady jets [46]. Similar arguments are likely to apply for a spray.

EQUIVALENCE RATIO IN THE MODEL SPRAY — The equivalence ratio  $\phi(x)$  at any axial location in the "model" spray in Fig. Cl is given by the stoichiometric air/fuel ratio (assumed to be 15 for diesel fuel) divided by the ratio of the air mass flux to the fuel mass flux at that location:

$$\phi(x) = 15 \left/ \left( \frac{\rho_a \cdot A_a(x) \cdot U(x)}{\rho_f \cdot A_f(0) \cdot U_f} \right).$$
(C31)

The terms in EQ (C31) are the same for EQs (C1-C3).

Substituting EQs (C3-C13) as necessary into EQ (C31) and neglecting the cross-sectional area occupied by fuel in EQ (C3) (*i.e.*, m=0) results in:

$$\phi(\bar{x}) = \frac{30}{\sqrt{1 + 16\tilde{x}^2 - 1}}.$$
 (C32)

If  $\tilde{S}$  is substituted for  $\tilde{x}$ , EQ (C32) gives the equivalence ratio at the leading edge of the spray.

The equivalence ratio is plotted in Fig. C2 as a function of time along with the penetration correlations. Equation (C16) was used to determine  $\tilde{t}$  for each  $\tilde{S}$  or  $\tilde{x}$  in the figure. With respect to  $\tilde{S}$ ,  $\phi$  in Fig. C2 is the equivalence ratio at the leading edge of the spray and  $\tilde{t}$  is the time required for the spray to penetrate a distance  $\tilde{S}$ . With respect to  $\tilde{x}$ ,  $\phi$  is the equivalence ratio at any axial location  $\tilde{x}$  in the spray and  $\tilde{t}$  is the time required for injected fuel to reach  $\tilde{x}$  (The equivalence ratio for any  $\tilde{S}$  or  $\tilde{x}$  in the figure is determined by moving horizontally across the plot at the  $\tilde{S}$  or  $\tilde{x}$  to the penetration curve, then up or down to the equivalence ratio curve.)

The figure shows that equivalence ratio at the leading edge of the spray decreases as  $\tilde{t}$  and  $\tilde{S}$  increase, and that the equivalence ratio in the spray decreases with increasing  $\tilde{x}$ . The gray region in the center of Fig. C2 corresponds to recent equivalence ratios measured by Espey *et al.* [47] in the head of a penetrating fuel jet in an engine. The height of the gray region represents the equivalence ratio range ( $\phi \approx 2$  to 4) measured in the head of the spray. Its width represents the range of times corresponding to the  $\tilde{x}$  values on upstream and downstream sides of the head of the spray ( $\tilde{x} \approx 2.4$  and 2.7, respectively). Given the simplifications made in deriving the EQ (C32), the agreement with the engine results is very good.