

Influence of Injection Duration and Ambient Temperature on the Ignition Delay in a 2.34L Optical Diesel Engine

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ABSTRACT

Non-conventional operating conditions and fuels in diesel engines can produce longer ignition delays compared to conventional diesel combustion. If those extended delays are longer than the injection duration, the ignition and combustion progress can be significantly influenced by the transient following the end of injection (EOI), and especially by the modification of the mixture field. The objective of this paper is to assess how those long ignition delays, obtained by injecting at low in-cylinder temperatures (e.g., 760-800K), are affected by EOI. Two multi-hole diesel fuel injectors with either six 0.20mm orifices or seven 0.14mm orifices have been used in a 2.34L single-cylinder optical diesel engine. We consider a range of ambient top dead center (TDC) temperatures at the start of injection from 760-1000K as well as a range of injection durations from 0.5ms to 3.1ms. Ignition delays are computed through the analysis of both cylinder pressure and chemiluminescence imaging. A simplified one-dimensional (1-d) model of the diesel jet, able to match the behavior of a transient injection and entrainment processes, is used to estimate the ensemble-averaged mixture fraction fields during the injection event and at the ignition kernel locations.

At TDC temperatures of 850K or higher, the injection duration is longer than ignition delay, and thus EOI has no effect on ignition delay. At TDC temperatures of 800K or lower, for short injection durations (<1.3ms), ignition occurs after EOI and ignition delay decreases with decreasing injection duration is observed. In addition, the 1-d spray model predicts a decrease of the mixture fraction at ignition kernels with decreasing ignition delay. This is in contrast to the expected trend of increasing kinetic time with decreasing mixture fraction for well-mixed reactors. This suggests that mixture fraction alone is not the first-order parameter influencing the timing and position of ignition sites. The history of the ignition kernel(s) and/or of the scalar dissipation may also need to be considered.

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INTRODUCTION

The start of combustion is a key parameter in diesel engines that affects efficiency, noise, and pollutant emissions. The duration of the ignition delay (ID) from the start of injection to the start of combustion is controlled not only by the chemical kinetics of ignition, but also by physical processes such as atomization, vaporization, and turbulent mixing.

Because of its prime importance to diesel engine operation, ID has been extensively studied by many authors, including a few key studies in single-cylinder engines [1, 2, 3], rapid compression machines [4, 5, 6], constant volume [7, 8, 9] or continuous flow [10] vessels, and shock tubes [11, 12] for a wide variety of fuels. These experimental results generally show that ID decreases exponentially with increasing in-cylinder gas temperature T_a , consistent with a classical Arrhenius

law using a fixed activation energy. Empirical correlations have been proposed to account for other dependencies, including in-cylinder ambient gas ρ_a , rail pressure p_{rail} , oxygen concentration O_2 or orifice diameter d , such as the following [9, 10, 12]:

$$ID \sim \frac{e^{A/T_a} \cdot d^E}{\rho_a^B \cdot p_{rail}^C \cdot O_2^D}$$

Eq. 1

The expression is constructed such that the exponents A , B , C , D , E are generally taken as positive. These exponents are empirically determined, and thus depend to some degree on various hardware-specific factors, as well as the fuel. Physical properties of the fuel (e.g., volatility, viscosity, etc.) are not explicitly treated, in part because they have only a limited or no discernible effect on ID [11].

Note that Eq. 1 depends on orifice diameter and rail pressure, and thus on the fluid mechanics of the spray/jet development. This fluid-mechanical control of ID has previously been demonstrated in a previous study of starting jets, for which ignition always occurs before EOI [9]. The authors discussed two ways that enhanced mixing shortens auto-ignition delay in these starting-jet non-premixed systems; 1) favorable mixture fractions appear faster and 2) because faster mixing promotes faster heating of the fuel. ID decreases with decreasing orifice diameter [9], and the relative rate of air entrainment (i.e., the mass flow rate of air entrained into the spray normalized by the fuel mass flow rate delivered by the injector) increases when the orifice diameter decreases [13]. Hence, the mixing rate and the resulting mixture fraction (or, local equivalence ratio) field in the penetrating jet impacts ID.

In addition to the development of empirical correlations to capture macroscopic dependencies, the chemical and physical processes occurring within the jet at the local ignition site have also been explored. Diesel-type fuels generally display two-stage ignition chemistry [14, 15, 16]. Although the chemical kinetics of two-stage ignition involve many reaction pathways and intermediate steps, hydrogen peroxide (H_2O_2) is recognized as a key species formed between the two stages. In the first stage, exothermic low-temperature heat release (LTHR) reactions quickly raise local temperatures and shift local equilibria such that the reaction pathways feeding the LTHR are essentially halted. In the dwell between the two stages, a pool of hydrogen peroxide builds as the temperature slowly continues to rise. Hydrogen peroxide becomes unstable at higher temperatures, and its decomposition into hydroxyl (OH) radicals triggers the exothermic high-temperature heat release (HTHR) reactions of the second stage [17, 18]. Most of the heat release occurs in the second-stage ignition, with the first stage ignition accounting for a few percent of the total.

The speed of the two ignition stages is strongly affected by both temperature and local mixture fraction. The mixture fraction with the fastest ignition can be defined as the most reactive mixture fraction ξ_{MR} [19, 20, 21]. In non-premixed turbulent-jet ignition experiments with cold fuel injected into hot air, the value of ξ_{MR} is the result of an inverse relationship between mixture fraction and temperature [21]. Prior to ignition in such jets, local regions of high mixture fraction have low temperature, and vice-versa. Depending on the relationship between mixture fraction and temperature, some mixture fraction value will be the most reactive (ξ_{MR} is typically fuel rich for conventional diesel jets). However, not all of the most reactive mixtures ignite simultaneously. Instead, experiments show that the first ignition sites occur in localized kernels [21]. The distinguishing feature of these kernels compared to other regions of similar mixture fraction that do not ignite as early is the relatively low local scalar dissipation¹ (i.e. the rate at which mixture fraction gradients disappear due to molecular diffusivity) at the site of the kernel.

For conventional diesel jets with ignition before the end of injection (EOI), experiments using optical diagnostics have shown that ignition kernels have equivalence ratios between 2 and 4 [22], which

corresponds to a mixture fraction range of 0.13-0.27. Much like the cold-fuel turbulent jets injected into hot air, the temperature field prior to ignition in diesel jets is also expected to vary inversely with mixture fraction [23, 24, 25]. To date, scalar dissipation has not been measured in the ignition kernels of diesel jets, but presumably the kernels have both the most reactive mixture fraction and the lowest scalar dissipation relative to other regions in the jet.

Although scalar dissipation appears to govern which of the most reactive mixtures will ignite first, it is generally considered to be of secondary importance to processes that create the most reactive mixtures themselves. Hence, much effort has been devoted to understanding air-fuel mixing in diesel jets, including the process of air entrainment [13, 26, 27, 28, 29, 30, 31, 32]. For conventional diesel jets, ignition occurs before EOI, so mixing processes occurring in the transient penetrating jet can be treated to some degree as “quasi-steady,” such that analysis techniques for steady jets can be useful [26, 28, 31].

For advanced low-temperature combustion (LTC) conditions [33], or for low-load conditions with short injection durations and/or fuels/operating-conditions with long IDs, ignition can occur after EOI. In such conditions, the transient mixing processes following EOI can yield air entrainment rates much different than the quasi-steady penetrating jet. For instance, based on a simple one-dimensional (1-d) jet model, it has been proposed that after EOI, a transient wave of temporarily increased entrainment propagates through the diesel jet, greatly affecting mixing [34]. This “entrainment wave” has recently been confirmed experimentally [35].

Whatever the mechanism, any process affecting mixing after EOI will be important for determining ID for jets that don't ignite until after EOI. Hence, EOI itself could be an important mechanism to control ID through its modification of the mixture fraction field. However, to the authors' knowledge, only one study has tried to specifically assess the impact of EOI on ID [36]. Only four values of injection duration, 0.5 1.5 2.5 and 3.0ms were selected in their work and no clear trend was identified for the influence of EOI on ID.

Given the importance of controlling ID for many LTC schemes and for non-standard fuels and/or operating conditions, the aim of the present study is to gain a better understanding of how injection duration might be used to control ID. Through a comprehensive analysis of the influence of injection duration and the timing of EOI on ID, we gain some understanding of the role of the mixture fraction field on ID. Inferences about the role of scalar dissipation on ID are also offered.

Experiments are performed in a heavy-duty single-cylinder optical engine. Injection durations are varied for different intake air temperatures. In-cylinder pressure and broadband chemiluminescence images are acquired to provide metrics for ID and the spatial/temporal locations of ignition.

The general methodology is to quantify ID using conventional in-cylinder pressure analysis, and correlate these observations with variations of injection duration. Then the spatial/temporal locations of the visible ignition kernels are quantified using image stabilization

1. It is preferable to use the scalar dissipation rather than the strain rate since, for partially and fully premixed turbulent flows, these two values are not necessarily proportional to one another.

of ignition chemiluminescence emission. To estimate the ensemble-averaged mixture fraction field at the ignition kernels, we use predictions from a well validated 1-d diesel spray/jet model proposed in these studies [34, 37, 38]. The last sections are dedicated to discussions of 1) the potential roles of various processes including scalar dissipation and future work that could further our understanding of these phenomena, and 2) the limitations and uncertainties associated with this approach.

Table 1. Engine and Injector Specifications

Engine base type	Cummins N-14, DI Diesel
Number of cylinders	1
Number of intake valves	2
Number of exhaust valves	1*
Swirl ratio	0.5
Bore $\times$ Stroke, cm	13.97 $\times$ 15.24
Bowl width, depth, cm	9.78, 1.55
Displacement, liters	2.34
Connecting rod length, cm	30.48
Geometric comp. ratio	9.93:1
Replicated compression ratio [†]	16:1
Fuel injector type	Common rail, solenoid-actuated Bosch CRI2.2
Cup (tip) type	Mini sac
Number of holes	7, equally spaced
Spray pattern included angle	148°
Nozzle orifice diameter, mm	0.141
Nozzle orifice L/D	5
K factor	1.5
Fuel injector type	Common rail, solenoid-actuated Cummins XPI
Cup (tip) type	Mini sac
Number of holes	6 of 8-hole equally spaced pattern, 2 holes missing to isolate one spray
Spray pattern included angle	152°
Nozzle orifice diameter, mm	0.200
Nozzle orifice L/D	4

* In this optically accessible diesel engine, one of the two exhaust valves of the production cylinder head was replaced by a window and periscope.

[†] TDC conditions for a 16:1 compression ratio engine are produced in the optical engine by preheating and boosting the intake stream.

EXPERIMENTAL SETUP

Engine

IDs are measured in a single-cylinder, direct-injection, 4-stroke diesel engine based on a Cummins N-series production engine. A schematic of the engine is shown in Figure 1, and the specifications of the engine are summarized in Table 1. A complete description of the engine and its specifications is available elsewhere [39]. A brief description of the engine follows.

The Cummins N-series engine is typical of heavy-duty size-class diesel engines, with a bore of 140 mm and a stroke of 152 mm, yielding a displacement of 2.34 liters per cylinder. These dimensions are retained in the single-cylinder, optically-accessible engine. The intake port geometry of the production cylinder head also is preserved, yielding a quiescent (low swirl) combustion chamber. The optical engine is designed so that the upper cylinder liner separates and drops down from the head without engine disassembly, allowing rapid cleaning of in-cylinder optical surfaces. To provide optical access, the engine is equipped with an extended piston and flat piston-crown window. One of the exhaust valves has also been replaced with a window, which provides optical access down through the cylinder head and piston-crown windows for schlieren measurements of one of the penetrating sprays that is aligned with the cylinder head window (Figure 1).

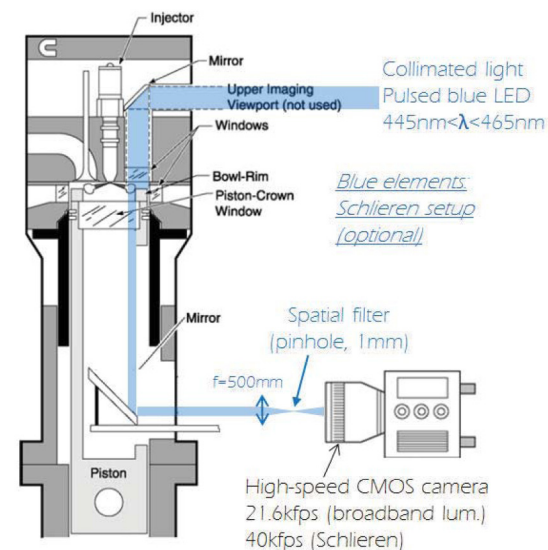


Figure 1. Extended-piston heavy-duty optical engine. Both the schlieren and the broadband luminosity setup are represented, even though the measurements have not been performed simultaneously (for the broadband chemiluminescence measurements, the LED is off and the spatial filter and converging lens in front of the camera are removed).

Fuel Injector

Two different solenoid driven, common-rail fuel injectors are used in this study, a Cummins XPI and a Bosch CRI2.2, whose specifications are included in Table 1. The XPI is equipped with a 6-hole mini-sac injector cup (tip), having diameters of 0.200mm and an included angle of 152° (14° down-angle from the firedeck, i.e., the bottom surface of the cylinder head). Two orifices of the 8-hole pattern are “missing” on either side of one of the sprays, creating one “solo” spray in the six-hole pattern. The Bosch CRI 2.2 is equipped with a 7-hole, mini-sac injector cup (tip), having an included angle of 148° (16° down-angle from the firedeck). The 7 orifices are equally spaced and have nominal diameters of 0.141mm. To minimize thermal loading on the optical engine, the installed injector is skip fired, so that each fired cycle is followed by nine motored cycles.

Fuel

The characteristics of the primary fuel used in this study are summarized in [Table 2](#), with typical values for Diesel #2 and Jet-A included for reference. Fuel 1 is derived from natural gas and was developed to be a replacement for either diesel or jet fuel. Fuel 1 has a higher cetane rating than typical for US pump diesel or jet fuel, with high paraffinic content and H/C ratio. Pump-grade Jet-A is used as a fuel at one operating condition for comparison of ID behavior with Fuel 1.

Table 2. Fuel Properties

	Fuel 1	Jet-A (typ.)	Diesel#2
Derived Cetane Number	60	/	/
Cetane No.	63	/	46
Cetane Index	58	47	46
Viscosity [mm ² /s] at 40 °C	1.0	/	2.4
Density [kg/m ³] at 15 °C	736	810	843
Heating Value [MJ/kg]	47	43.3*	42.97*
Sulfur Content [ppm]	/	496	390
H/C (molar basis)	2.17	1.82	1.8
Aromatics [%]	< 5	14	27
Olefins [%]	0.3	1.6	0.5
Paraffins [%]	> 95	/	72.5

*Net Calorific value, DCN (ASTM D-6890), CN (ASTM D-613), CI (ASTM D-976), Viscosity (ASTM D445) Density (ASTM D4052), Diesel were measured. Jet-A properties are not specific to the fuel used, but rather are generic, typical values from the literature [40].

Engine Performance Diagnostics

For all operating conditions, cylinder pressure, fuel rail pressure, and injector current are digitized and recorded at 1/4 crank-angle-degree increments, simultaneously with the acquisition of the optical data. Other data including fuel return temperature after the injector, engine block coolant temperature, engine RPM, and intake air temperature and mass flow rate, are separately recorded each at 1s intervals. The apparent heat release rate (AHRR) is calculated from ensemble-averaged pressure data using an air-standard first-law analysis (e.g., Heywood [15]). An example of the results from this processing appears in [Figures A1, A2, A3, A4, A5, A6](#). Prior to calculating the AHRR, the pressure data are smoothed using a Fourier series low-pass filter with a Gaussian roll-off function having a transmission of 100 % from 0 to 800 Hz and dropping to 1 % at 3360 Hz. We selected these cut-off frequencies to remove acoustic ringing in the cylinder pressure data, while retaining the general features of the AHRR. Using these filter parameters, the apparent start of combustion (SOC), indicated by the first zero-crossing of AHRR, is typically advanced from that for the unfiltered data up to 70 microseconds, depending on the size and steepness of the premixed burn spike. Additionally, the peak of the premixed burn spike in the filtered AHRR is reduced by about a factor of 2, while its width is

increased by roughly a factor of 2. The apparent energy released during the premixed burn (area under the AHRR curve), however, is virtually unchanged by this filtering technique.

Operating Conditions

Several parametric variations are performed with the two different injectors and summarized in [Table 3](#). To avoid confusion between the timing of the electrical drive pulse to the injector solenoid and the actual fuel-delivery response of the injector, the following nomenclature is adopted:

- Solenoid Energizing: refers to the electric drive pulse for the injector solenoid. The start of solenoid energizing (SSE) is thus the start of the electrical current pulse to the solenoid, and the duration of solenoid energizing (DSE) is its duration.
- Injection: refers to the actual fuel-delivery response at the injector nozzle. The start of injection (SOI), i.e., when fuel first emerges from the nozzle, is delayed from SSE because of the hydraulic delay in the injector and fuel rail system. The duration of injection (DOI) is the *actual* duration of injection, which is evaluated from the visual start to end of injection determined from high-speed imaging of optical Mie scattering (not presented here).

The temperature at top dead center (TDC) is estimated by an isentropic compression calculation using real air thermodynamic properties from the bottom-dead center (BDC) temperature and pressure to the measured TDC pressure on a motoring cycle (with no fuel injection). This is intended to represent the temperature of the “core” gases at TDC, for which heat transfer to the cylinder walls is negligible. Pressure at BDC is assumed to be equal to the average intake manifold pressure between 170 and 190 crank angle degrees (CAD), with TDC of the compression stroke at 360 CAD. A previously developed empirical correlation accounts for heat transfer during induction to give the BDC temperature relative to the intake manifold temperature [41]:

$$T_{bdc} [^{\circ}\text{C}] = 0.68 \cdot (T_{intake} - 95 [^{\circ}\text{C}]) + 95 [^{\circ}\text{C}]$$

Eq. 2

The TDC density is then computed with the perfect gas equation, taking into account a compressibility factor (Z_{fg}) that is about 1.01 at the considered TDC temperatures and pressures.

The first set of experiments (Dataset #1) consists of a TDC temperature sweep at a fixed DSE of 0.5ms. This is intended to highlight the general trend of ID with TDC temperature for Fuel 1 in [Table 2](#). However, at low temperature, the long ID yields a large AHRR peak, especially for longer DSE. To explore longer DSE without excessive peak AHRR that could damage the optical engine, a second set of experiments (Dataset #2) is performed at a lower rail pressure and with an injector having smaller orifices. Dataset #2 consists of a DSE sweep from 0.5ms to as long as 3.1ms at 4 different TDC temperatures (760K, 800K, 850K, 900K) to yield data on the influence of DSE on ID. Results from both datasets are in the “Experimental Results” section.

Optical Setup

Two different optical techniques are employed here. The first, high-speed chemiluminescence imaging, provides the spatial and temporal locations of ignition as evidenced by chemiluminescence emission. Various chemical reactions of ignition yield electronically excited species and/or luminous emission, and these emissions are commonly used as a marker for ignition [42, 43]. The broadband chemiluminescence images are used to compute the position and timing of the first ignition kernel for each injection event. A second optical diagnostic, high-speed schlieren imaging of the penetrating diesel jet, is applied under non-combusting operating conditions, with an intake stream of 100% nitrogen. The purpose of the schlieren diagnostic is to measure the diesel jet penetration rate for calibration of the 1-d jet model for estimating ensemble-averaged mixture fractions. Details of the two diagnostics are provided in the following paragraphs.

Broadband chemiluminescence images are recorded using a high-speed CMOS camera equipped with a Nikkor 50mm f/1.2 lens with no spectral filtering and with the aperture wide open. The frame rate is 21.6 thousand frames per second (kfps), corresponding to 1 image each 0.33CAD at 1200rpm. Exposure duration is 42 microseconds. The resolution of images is 256×256 pixels, corresponding to 116×116mm (2.2 pxl/mm). The images are acquired through the 45° mirror placed under the piston (Figure 1), thus giving access to the whole piston bowl. The camera is triggered at SSE, and 130 images are recorded per cycle, corresponding to approximately 6ms imaging duration per cycle. This long imaging set was recorded mostly as a matter of convenience. In practice, only the first few frames provide useful ignition chemiluminescence before being dominated by soot incandescence. For each experimental condition (i.e. for each combination of a TDC temperature and DSE in Table 3), chemiluminescence images from 20 firing cycles are recorded.

Schlieren images of one penetrating diesel jet passing between the cylinder head and piston window are acquired with the same combination of camera and lens, but with the following additional optics and settings. The schlieren light source is a blue (445-465 nm) light-emitting diode (LED) producing pulses of 3 microseconds at 25kfps. The LED source is imaged onto a 1-mm diameter pinhole by a 50-mm camera lens. The light passing through the pinhole is collimated using a 500 mm spherical lens placed one focal length from the pinhole. The resulting collimated beam is directed through the engine, top to bottom, as depicted in Figure 1. In front of the camera, another 500-mm spherical lens focuses the light transmitted through the engine to the bright-field schlieren stop, another 1-mm diameter pinhole. The camera lens object distance is set to infinity, and the f-stop is set to f/11. The field of view is the intersection of the periscope and piston windows (Figure 1), resulting in imaging of one of seven jets that passes below the cylinder-head window, and from 30mm to 50mm from the injector orifice. The frame rate is 25kfps, corresponding to 1 image each 0.29 CAD at 1200rpm. The resolution of images is 128×256pixels, corresponding to 26×52mm (5 pxl/mm).

Table 3. Experiments Performed

Dataset		1	2
Injector		XPI	Bosch CRI 2.2
P_{rail}	bar	1500	750
O_2	% vol.	21	21
SSE	CAD	355	352, 353.5(late)
DSE	ms	0.5	0.5:0.2:3.1
T_{tdc}	K	800, 850, 900,950,1000	760, 800, 850, 900
ρ_{tdc}	kg/m ³	18.8	18.8
Speed	RPM	1200	1200
Fuel type		Fuel 1	Fuel 1, Jet-A
Comp Ratio		9.93 and 11.22	9.93

EXPERIMENTAL RESULTS

Ignition Detection in an Optical Engine

A number of important considerations must be addressed to use engine pressure data, and in particular on an optical engine for quantification of ignition timing. Rothamer and Murphy [1] provide a review of the influence on different ignition and heat release thresholds, in addition to the effect of filtering the apparent heat release rate (AHRR) for ignition detection in engines. Although the absolute time of ignition is very sensitive to the particular threshold chosen, the sensitivity, or change in ignition time with changing parameters was unaffected, even in the presence of filtering.

For the applicability of optical engines to ignition detection, the variation in material properties, in particular the slower heat transfer through optical surfaces, has a tendency to advance ignition in fired optical engines [44]. But by skip firing, the impact of increased thermal resistance can be nearly compensated in the motored pressure trace [45].

Results presented below first build confidence in our ability to quantify ignition timing reliably, by reproducing the widely observed exponential dependence of diesel ID on temperature of Eq. 1. Then we present measurements of ID with variation of injection duration at TDC temperatures from 760 to 900K.

One objective of the experimental campaign is to assess the non-conventional fuel described in Table 2 as Fuel 1. Even though this fuel has similar properties with diesel #2 and jet-A fuels, a preliminary experiment is performed to assess whether its ID behavior in the engine is characteristic of standard diesel fuels. Intake air temperature is varied to achieve a range of estimated TDC temperatures at a constant TDC charge density of 18.8kg/m³, with a rail pressure of 1500bar and 0.5ms DSE, and with no exhaust gas recirculation (EGR) or other dilution. To extend the lower end of the achievable TDC temperature range to 760K, the piston was equipped with a special top with a large bowl-rim cut-out to lower the compression ratio. With two different compression ratios of 9.93:1 and 11.22:1, the TDC temperature can varied between 760K and 1000K without cooling the intake below room temperature. The

experimental test plan includes an overlap of estimated TDC temperatures with the two piston geometries to confirm that both compression ratios give similar IDs at similar estimated temperatures. The results are presented in Figure 2. In this figure, note that the y-axis is a logarithmic scale.

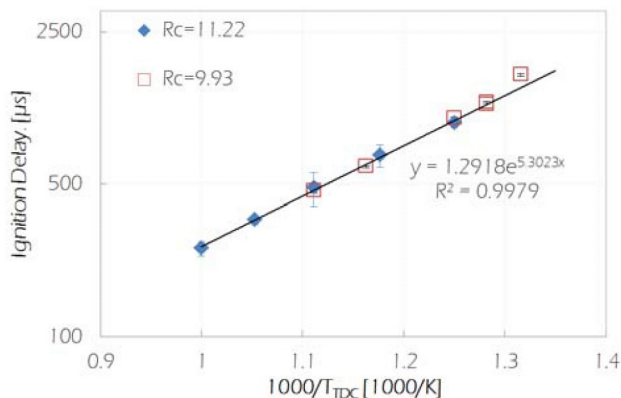


Figure 2. ID with Fuel 1 using a range of intake temperatures at two compression ratios to achieve overlapping estimated TDC temperature ranges. Dataset #1: XPI injector, DSE=0.5ms, and 18.8kg/m³ in-cylinder density at TDC. Error bars represent the standard deviation of ID from 30 fired cycles.

Within the overlap region of TDC temperatures achievable with both piston-top configurations, the compression ratio has no significant effect on ID: at similar TDC temperatures; similar IDs are measured for both compression ratios. In addition, a very linear trend is observed, confirming that ID of the fuel used for our experiments follows the classical exponential dependence of Eq. 1. The constant $C=5.3023$ in the exponential corresponds to an activation energy $E_a=44.1\text{kJ/mol}$. This value is comparable to the values obtained in [46] (E_a spans from 40 to 51.3kJ/mol) and [9] ($E_a=54.3\text{kJ/mol}$) for diesel fuels.

Arrhenius rates are expected for homogeneous reactors, where the thermodynamic boundary conditions are stable. The case of the diesel injection differs greatly from these stable conditions: cold fuel is injected into hot ambient gases and mixes with them. A “parcel” of fuel that ignites thus faces different mixture fractions and temperature in its history from injection to ignition. Yet, ID behavior follows the Arrhenius trend (see Figure 2 or [9]). An inhomogeneous reactor thus shows the same temperature dependence than a perfectly stirred one. This suggests that the factors governing the reaction between the spray and the surrounding air are not the mixing processes, but rather the chemical processes. Another interesting result observed on Figure 2 is the fact that no evidence of a negative temperature coefficient (NTC) region appears. The NTC phenomenon is characterized by a region where ID decreases as the temperature decreases in the range 600-800K. An NTC region has been observed in many shock tube studies of similar jet fuel surrogates (for example [11, 47]). The major difference between the shock-tubes studies -in which the mixture is perfectly stirred before compression ignition-and our experiments is the stratification in mixture fraction and temperature created during direct injection. Thus, the non-existence of a NTC region could be a consequence of physical effects related to the mixing processes. As will be discussed in a later section, fluid mechanical and mixing processes associated with EOI may assist in offsetting the NTC

behavior at the low end of the temperature range in Figure 2, yielding an approximately linear dependence even through the lowest temperatures.

For the data in Figure 2, the DSE was constant and relatively short to avoid excessive premixed burn spikes at the lowest temperatures that have long IDs. As explained in the “Operating Conditions” section, to explore longer DSEs at various TDC temperatures, a lower injection pressure (750bar) and a different injector with a lower fuel delivery rate is used for Dataset #2 in Table 3. This injector and rail pressure combination allows a greater range of DSEs before the peak AHRR becomes excessive.

DSE Variations at Different Temperatures

In Figure 3 we plot ID for DSE sweeps at various TDC temperatures for the conditions of Dataset #2 in Table 3. The dashed black line indicates the visual end of injection as determined from analysis of high-speed Mie-scatter imaging of the liquid fuel (not shown). At TDC temperatures of 900K and 850K, no statistically significant influence of DSE on ID is detected over the range of injection durations used. For these higher TDC temperature conditions, ignition occurs before EOI (i.e., below the dashed black line) for most of the DSEs tested. Hence, these data are representative of conventional diesel combustion with ignition prior to EOI.

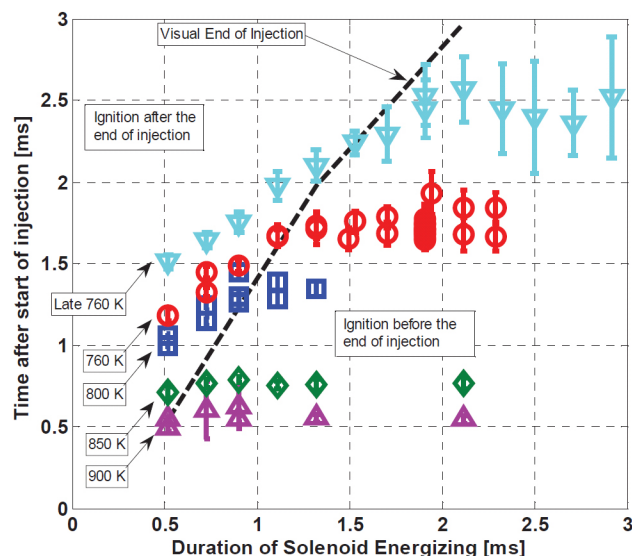


Figure 3. Cylinder-pressure-based IDs for Dataset #2 Fuel 1, DSE sweeps with SSE=352 CAD at TDC temperatures of 900, 850, 800K, and with two SSE timings of 352 and 353.5 CAD at 760 K. The black dashed line shows the EOI timing. Each symbol represents an average of 30 individual measurements. Variation in inlet pressure and temperature between runs causes some scatter in measured average behavior. Error bars represent the standard deviation of ID over 30 fired cycles used to compute the average.

At a TDC temperature of 800K, ID has a statistically significant dependence on DSE when EOI occurs before the start of ignition (i.e., for data above the dashed black line). This dependence will be quantified below. The DSE domain for the 800K condition is limited by excessive AHRR at the longest DSE for this condition of approximately 1.3ms. Nevertheless, the data seen in Figure 3 departures from a linear increase of ID with DSE when ignition occurs

after EOI to a more constant ID when DSE is longer than ID among 0.9, 1.1 and 1.3ms, similar to the higher-temperature conditions. At a TDC temperature of 760K, the AHRR peak becomes large but within acceptable limits (i.e. lower than 500J/CAD), so the full DSE range can be explored. For the 760K data, ID clearly increases as DSE is increased until reaching a somewhat constant level for DSEs greater than about 1.1ms, which corresponds to the visual end of injection.

Because the experimental facility does not have a means to cool the intake stream, the 760K condition with room-temperature intake stream is the coolest motored TDC temperature possible, and thus yields the longest IDs possible at the standard SSE of 352 CAD. To explore conditions with even longer IDs, the DSE sweep at 760K is repeated, but with the SSE shifted slightly later to 353.5 CAD. For these data in Figure 3, labeled “Late 760 K,” ID once more increases with increasing DSE. And once more, within the statistical uncertainty of the data, ID becomes nearly constant when the EOI crosses over the start of combustion, near DSE=1.9ms.

Although the behavior of this fuel has been shown to agree well with diesel-like Arrhenius behaviors with a short injection of fixed duration (Figure 2), we wished to confirm if sensitivity of ID on DSE displayed in Figure 3 would hold for another fuel. Jet-A fuel was used with SSE=353.5 CAD and with a DSE sweep extending over the same range as for the 760K condition with the late SSE of 353.5CAD for the primary fuel. For clarity, the additional Jet-A ID data are plotted alongside the Fuel 1 results in Figure 4. Because of its lower cetane number (Table 2), the jet-A sweep is run at a higher temperature of 800K to achieve comparable IDs as the Fuel 1 data at 760K. The results with jet-A fuel in Figure 4 confirm the injection duration sensitivity for IDs occurring after EOI, as well as the lack of sensitivity when ignition occurs before EOI.

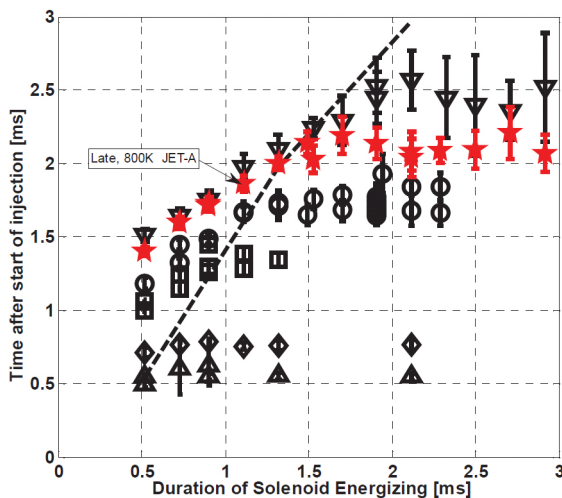


Figure 4. Cylinder-pressure-based ID of Jet-A (red star) for DSE sweep at TDC temperature of 800K and SSE of 353.5CAD. Fuel 1 data from Figure 3 are included as black symbols, and dashed line shows EOI timing. Error bars represent the standard deviation of ID over 30 fired cycles.

From these experiments, it appears that ending the injection can have a significant effect on ID. For the 760K TDC operating condition, the trend in ID advancement appears to be approximately linear with DSE, with a sensitivity of ID to DSE of 0.71 ms/ms in the region

where ID is longer than injection duration. As explained in the introduction, it has been shown [34, 35] that the mixture fraction field in the spray is strongly affected by transient mixing processes occurring as a result of deceleration of the injected fluid as the injection ends. This leads to the hypothesis that, for the shorter DSE, the enhancement of entrainment and subsequent changes to the mixture fraction field induced by EOI are responsible for the decrease of ID. It also suggests the possibility of enhancing control of ignition through a split injection strategy where this sensitivity is used to influence ignition delay through the injection duration of a pilot pulse occurring near TDC in a pilot-main injection strategy.

An effort has therefore been undertaken to determine the mixture fraction at the ignition location and timing. While direct measurements of mixture fraction are possible, they are difficult, especially under high-pressure reacting conditions, and beyond the scope and funding of this project. Instead, we use a well-verified 1-d dense fluid model of the diesel spray to estimate the transient ensemble-averaged mixture fraction field [34, 37, 38]. Although instantaneous measurements of mixture fraction at the ignition sites would be preferable to using ensemble-averaged estimates, these estimates should be sufficient to explain the trends observed in ID data, if caused by changes in the mixture-fraction field.

Because the mixture fraction field evolves in time and space, knowledge of the ignition kernel location is necessary to quantify the mixture fraction at ignition. The average spatial/temporal position of the ignition kernels for each condition would provide an appropriate position to interrogate the model for an estimate. Such data are provided by direct visualization of the chemiluminescence emission from the ignition events. The average temporal and spatial positions of the ignition kernels of each isolated jet are then mapped onto the transient model predictions to yield an estimate of the local ensemble-averaged mixture fraction at the ignition sites as a function of distance from the jet orifice.

The following two sections describe the elements of the estimate of the ensemble-averaged mixture fraction at ignition. First, the experimental ignition data are reported, and then the model is described and the estimated mixture fractions at ignition are presented.

Analysis of Combustion Images

The first step in the estimation of the mixture fraction at the ignition kernels is to locate them spatially and temporally². For this purpose, chemiluminescence emission images are recorded using the high-speed camera setup in Figure 1. For each engine operating condition, images are acquired for 20 fired cycles, and the imaging data are processed as described below to determine the ensemble-average ignition kernel location for comparison with 1-d model predictions.

The broadband luminosity images are processed to detect only the regions presenting chemiluminescence emission from high-temperature (second-stage) ignition, and to exclude background noise. The background noise can consist of shot noise, ambient luminosity (for example, room light through the periscope window),

2. To locate the kernel spatially in the model, the result, we assume the jet is axisymmetric, despite a known swirl of the flow in the particular engine.

cool-flame (first-stage) chemiluminescence emission, and elastic scattering of combustion luminosity off the liquid fuel spray or surfaces of the combustion chamber. The processing scheme must exclude all these interferences while also detecting ignition kernels at their first appearance and detecting the outer borders of the kernels, which are characterized by low absolute intensities.

The image post-processing scheme to locate the ignition kernels is summarized in Figure 5. First, a background intensity level is defined by computing the average intensity over the region of the images where the firedeck is visible (i.e., above the piston bowl). This background is primarily due to elastic scattering of chemiluminescence emission off in-cylinder surfaces, so its value increases with the intensity of the chemiluminescence emission. Then, the threshold defining intensities strong enough to be considered as originating from second-stage ignition chemiluminescence is set to 8 % of the difference between the average background noise level and the maximum intensity in the image. Only intensities higher than that threshold are treated as potential second-stage ignition, and visual inspection of the images shows that this value is low enough to detect early ignition as well as the low-intensity outer borders of ignition kernels.

With only this dynamic threshold, however, some part of the image will always be above the threshold, even before ignition. Typically, pixel noise and other tiny point-like sources that are clearly not ignition kernels rise above the threshold, illustrated as structures 2 and 3 in Figure 5. To reject such non-ignition structures, two additional criteria are applied.

First, any structure rising above the threshold is considered an ignition kernel only if the maximum intensity rises above 200 counts (out of a maximum of 4096 counts) somewhere within the structure. Visual inspection of image sequences show that locally, ignition kernels become very bright very quickly, so that ignition kernels can be characterized as having some bright region. Structure 2 in Figure 5 does not contain sufficient peak brightness within it, so it does not qualify as a kernel. This additional threshold helps distinguish kernels from background noise, but it cannot be used alone because it is too high to detect the outer borders of the ignition kernels. Hence, the 8% dynamic threshold and the 200-count structure threshold are applied together to define ignition kernels and their boundaries.

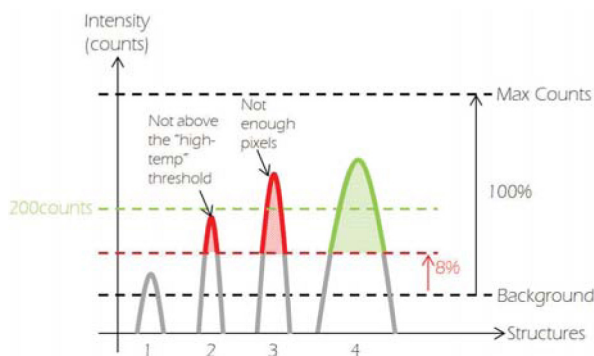


Figure 5. Summary of the methodology used to detect high-temperature regions in broadband luminosity images.

The second criterion to reject structures that are not ignition kernels is a minimum of 20 pixels within the structure that are above the 8% dynamic threshold. This eliminates bright single-pixel noise and tiny point-sources of interference, such as structure 3 in Figure 5.

Together, the 8% dynamic intensity threshold, the 200-count structure threshold, and the 20-pixel size threshold reliably define both the initial appearance of ignition kernels and their outer boundaries. All three of the threshold levels are subjective, but the automated processing scheme using these thresholds is able to define ignition kernels that agree well with visual inspection, for which the ignition sites are quite clear.

Figure 6 shows an example of the temporal evolution of the initial ignition chemiluminescence in a single cycle³. The grayscale in each image has been adjusted for each image to highlight interesting features that appear at different instants. The maximum intensity of the images and the time after start of injection are reported in the upper left corner. Pixels defined as ignition kernels according to the above criteria are colored.

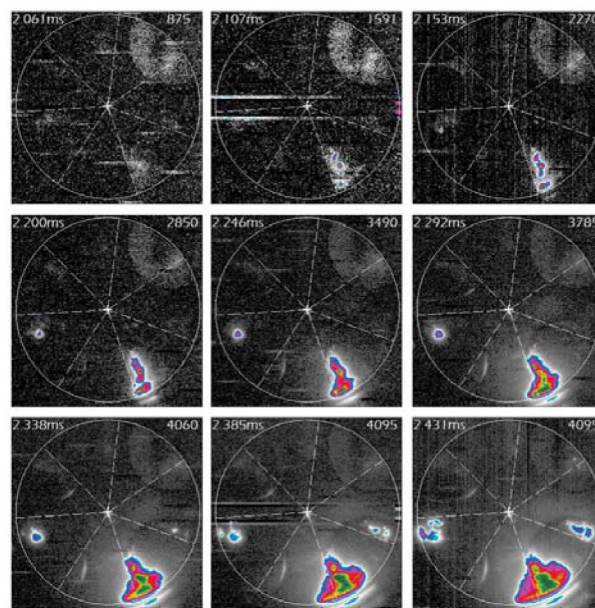


Figure 6. Temporal evolution of a combustion event for $\rho_{TDC}=18.8\text{kg/m}^3$, $T_{TDC}=760\text{K}$ and $DSE=1.1\text{ms}$. The upper left corner contains information about time after start of injection of the images. Colored regions are where ignition kernels are identified. Elastic scattering interference from the liquid fuel spray and other sources is visible when ignition chemiluminescence is bright. The white crosses indicate the position of the injector tip, and the dotted white lines represent the axis of the sprays.

In the upper right corner, for the first images and continuing thereafter, a large low-intensity feature appears. This is due to the upper periscope port in Figure 1, which is open and ambient light from the laboratory is passing through it and the cylinder head window is collected by the camera. Starting at 2.107ms (top middle), the post-processing scheme identifies an ignition kernel in the 5 o'clock position. In the next few frames, the ignition kernel grows,

3. In the example presented in Figure 6, this ignition site appears at 2.017ms after the actual start of injection, and is located in the 5 o'clock spray. In combustion events with another, the spray showing the first ignition kernel is not always the same (the injector is a 7-hole)