A Novel Approach for Simultaneous NOx and Smoke Reduction in Diesel Engines: Interacting-Sprays Injection

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ABSTRACT

In the past decade many in-cylinder injection approaches were proposed for simultaneous reduction of nitric oxides (NO_X) and smoke in diesel engines with various degrees of success in operation. In this paper, some results from a novel and promising technique referred to as Interacting-Sprays injection concept is presented. A single-cylinder compression-ignition two-stroke research engine with optically-accessible head mounted on a high-speed CFR (cooperative fuel research) engine crankcase is used to investigate the combustion and emission characteristics of this injection system. The interacting-sprays injection system produces two separate independently-controlled liquid fuel spray injections with a good degree of adjustability with regard to their fuel quantities and injection timings. The impingement schedule of the two sprays on each other at the right time and place inside the combustion chamber is the key to the success of the interacting-sprays injection system. Results are presented that show the effects of the varied injection system characteristics on the combustion and exhaust emissions (NO_X and smoke). The effects of the injection timing and time separation between the first and second injections of the interacting-sprays injection system are explored. Conditions are identified for which a favorable influence on both smoke and NO_X production is observed. A promising and new injection system and strategy are therefore proposed as a result of the data acquired in this study.

INTRODUCTION

Due to ever stricter Federal Environmental Protection Agency (EPA) automotive emission standards and concerns for the adverse health and environmental effects of NO_X and smoke from diesel engines, many manufacturers have been pursuing improvements in traditional means such as exhaust gas recirculation (EGR), injection timing retard, and fuel injection equipment changes. However, as the anticipated even stricter standards come into effect, the quest for new and novel approaches via modifications in injection system design as one important engine component continues. Thus, a complete understanding of in-cylinder processes becomes increasingly important.

In recent years, many investigators have shown potential for NO_X and particulates reductions with minimal effects on the brake specific fuel consumption (bsfc) using a sufficiently high pressure injection system and/or pilot or split injection, see Oblander et al. [1], Sato et al. [2], Shimada et al. [3], Aoyoma et al. [4], Shakal, et al. [5], Shundoh et al. [6] and [7], Uyehara [8], Bower and Faster [9], Bower [10], Tow et al. [11], Durnholz et al. [12], Osuka, et al. [13], and Pierpoint et al. [14]. These are briefly reviewed in Tow et al. [11] and Campbell et al. [15]. Tow et al. [11] in an extensive work using a single injector unit but producing multiple injections (called split injection by momentary interruption of the fuel flow) show that it is easier to favorably affect the universally-observed smoke- NO_X or bsfc- NO_X trade-off curves in diesel engines by split injection at high load as ignition delay is generally shorter than the part or low load conditions. For the part load, Tow et al. [11] demonstrate improvements by triplet injection (i.e. three subinjections in time) and explained to be due to enhancement of the mixing caused by the delayed last subinjection. This elevated mixing is because of a separate ignition and an assumed premixed-burn combustion phase of the last subinjection. They also show that the dwell (i.e. time between the subinjections) is the critical operating parameter for optimization. In their analysis it is clear that global heat release rate analysis is unable to provide any clues as to the nature of the important local fuel, air, and combustion products interactions needed for complete interpretation of their data. Based on limited data, Aoyama, et al. [4] also propose improvements in the mixing (controlling localized high temperature) to be responsible for their "Active Secondary Injection" method to simultaneously lower the smoke, HC, and NO_X with no or minimal effects on the indicated mean effective pressure (imep). They also use a single injector and inject a large amount of fuel followed by a small fraction at the end and thus comes justification for the name active secondary injection.

In this paper a novel in-cylinder injection approach named the Interacting-Sprays injection system is described that has shown promising results for simultaneous reduction of NO_X and smoke from a research engine operating in compression ignition (Diesel) mode. The idea was born by past experiences learned from jet/jet impingement employed in chemical rocket engines for atomization and mixing of the liquid fuel and oxidizer. At first it was just an engineering intuition thinking there might be benefits in such impingement when used in diesel engines. In this paper results from our systematic investigations are presented indicating support for the proposed interacting-sprays injection concept and a need for further investigations. After description of the experimental rig, explaining the engine, fuel injection, and exhaust gas sampling systems, results are presented and discussed.

DESCRIPTION OF EXPERIMENTAL SYSTEM

The experimental setup is briefly described below under four sections: Engine, fuel injection system, and emissions sampling system. For more details refer to Campbell et al. [15], Pushka et al. [16], and Sinko et al. [17].

Engine

The engine used is a single-cylinder two-stroke in operation with a high-speed Cooperative Fuel Research (CFR) crankcase, an elongated flat-top piston, a cylinder head which allows for optical access into the engine. Portions of the basic engine originally designed at Princeton University are reported in many publications including Liou et al. [18], Boulouchos et al. [10], Bardsley et al. [20], and Chehroudi and Schuh [21]. The general experimental setup for the interacting-sprays Injection system is shown in Fig. 1(A) and Fig. 1(B), while major engine dimensions are summarized in Table 1. The engine head is newly designed to produce a cylindrical cup-in-head combustion chamber. The cup axis is the same as the cylinder axis with cup dimensions of 63.5 mm in diameter by 13 mm depth with a resulting squish area of 21.85 cm2. In addition to the cup combustion chamber, the cylinder head piece contains a laser beam entrance slit of 57 mm long by 1.3 mm wide for visualization purposes. The slit is positioned to admit a horizontal sheet of laser light passing through the center lines of the horizontally-injected interacting sprays. The laser sheet travels at a near 90^{O} angle to the spray penetration directions. The beam is admitted through a rectangular flat quartz window mounted outside the head piece covering the slit. The engine head has mounts for two Stanadyne slim-tip pencil fuel injectors. The injectors are positioned with their nozzle tips near the cup side-wall at mid-depth in the cup and 6.4 mm apart from each other. Each injector has a single orifice producing a horizontally-injected spray across the chamber. The sprays are directed to impinge each other. The head piece holds a cylindrical quartz window positioned at the top to permit viewing of the injector tips and penetrating sprays in the chamber. The window axis is at an offset position of 13 mm from the common symmetry axis of the cylinder and the cup-in-head chamber. An aluminum plug replaces this quartz window when engine is fired for emission measurement purposes, being the main focus of this paper.

Rectangular-shaped inlet and exhaust ports are machined into the side walls of the cylinder to provide a cross-scavenged two-stroke engine configuration, see Blair [20]. Table 1 presents details of the port dimensions. The intake ports are equipped with replaceable insert pieces to direct the incoming air upward at 30° to the horizontal and at 40° from a radial line to impart an air swirl motion. The engine is either supplied by air from the laboratory compressed air system for emission measurements and regulated to 120 kPa upstream of a 6 KW electric heater (Sylvania Osram Model 065322) or by nitrogen alone for Exciplex Laser Induced Fluorescence (LIF) studies. For details of the Exciplex LIF approach and work in our engine see Melton [23] and Campbell et al [15] respectively. The gas (air or Nitrogen) flow rate is metered by a laminar flow element (Merriam Model 50MW20-2) located upstream of the heater and the engine's inlet surge tank. The engine instrumentation includes a Kistler Model 7061 pressure transducer for time-resolved cylinder pressure measurements. The dynamometer-drive system is equipped with a shaft-mounted optical encoder to provide control and synchronization pulses.

Fuel Injection System

The fuel injection system consists of two separate single-cylinder injection pumps driven in series from the crankshaft, see Fig. 1(A). Detailed specifications of the Ambac Model APE-1B injection pumps are given in Table 2. With these pumps, control of the injection timing is accomplished by phasing of the pumps drive shafts relative to the engine crankshaft. A Candy Model 5A dynamic differential between the engine and the first injection pump allows continuous adjustment of injection timing (or angle) relative to the TDC crankshaft position. An Ambac TMB splined-shaft timing device, in series with a multi-position Oldham coupling, permits continuous adjustment between the second and the first pump's drive shafts. The use of two separate injection pumps, providing independent and continuous first and second injections with respect to each other and the engine's crankshaft, is a unique feature of this setup. The fuel from each pump passes through thick-walled fuel-injection tubing to a fast-acting high-pressure solenoid valves before arriving at the injectors. A special positioning clamp is used in mounting of the injectors to assure that the undeflected spray centerlines lie in a single horizontal plane.

Figure 1 shows that when the solenoid dump valves are not activated, fuel from the injection pumps returns to the fuel tank through a dump circuit. In this case, fuel pressures in the injection lines do not attain the necessary value of 20.8 MPa to open the injector needle valves and hence injections into the cylinder are prevented. Since pressure must be maintained in the fuel lines to ensure consistent injection when called for, proportional relief valves set at 6.89 MPa are installed downstream of the solenoid valves. When activated, the solenoids close off both dump circuits simultaneously and normal fuel injection commences when the line pressure exceeds the injector needle-valve opening value. These fast-acting solenoid dump valves are adapted from the bodies of solenoid-controlled Detroit Diesel unit injectors modified to act as high-pressure solenoid valves. The quantity of fuel delivered per injection through each injector is controllable. Fuel flow rate is calibrated by collecting the injected fuel for a known period and weighing on a scale.

Instrumentation applied to the fuel injection system includes Kistler Model 6230 pressure transducers installed in each fuel line to measure time-resolved injection pressure. Needle valve lift for each fuel injector is measured by a Wolff Controls Hall-Effect microsensor probe, see Ziemacki and Wolff [24].

Exhaust Gas Sampling System

Engine exhaust gas samples from the fired engine cycles are obtained through a sampling probe in the exhaust pipe as shown in Fig. 2. Since the engine operates in a skip-fired mode explained later, it is necessary to sample only during the exhaust blowdown phase of the fired cycles. A computer-controlled solenoid valve in the sample line ensures acquisition of samples which best represents the contents of the exhaust stream during initial exhaust gas blowdown before major dilution occurs in this two-stroke engine operation. From the solenoid valve, the sampled gases either pass through the Thermo Electron Model 10AS Chemiluminescent NOx analyzer or the Bosch smoke meter EFAW 65A. The two paths are separated by on/off valves. The smoke sampling line has a backflush system and is used to purge the smoke from the sampling system before a new smoke sample is drawn. The NOx sampling system consists of an ice bath/water trap/prefilter, a flip top filter and a Chemiluminescent NOx analyzer.

Ignition delay time in diesel engine is related to the intake temperature in an exponential Arrhenius form and as such decreases with temperature. Cetane number is determined in a standardized engine and procedure. Generally the cetane number of the fuel increase as ignition delay is shortened. Measured ignition delay times are within 0.6 to 3 ms for low compression ratio direct injected (DI) engines and from 0.4 to 1 ms for high compression ratio and turbocharged engines. Addition of the ignition enhancer, therefore, does change the cetane number of the original fuel and consequently the ignition delay period. To promote reliable ignition behavior (when the engine uses air and is fired) with production-engine ignition delay periods and due to the relatively low compression ratio used, the intake air charge was heated to 187^{O} +/- 2^{O} C and isopropyl nitrate (ignition enhancer) was blended at 3% by volume to commercial #2 diesel fuel used throughout the fired tests. These measures produced consistent ignition behavior with ignition delays ranging from 1 and 3 ms, depending on the timing of the first injection. The influence on exhaust emissions of the use of ignition enhancers similar to isopropyl nitrate is recently investigated by Ullman et. al. [25]. They show that addition of 2-ethylhexyl nitrate in a quantity sufficient to produce an improvement of 14.1 cetane rating points elevate exhaust NOx emission levels by no more than 3% compared to the unenhanced fuels of comparable Cetane rating. It is expected that the use of a consistent blend of the fuel and additive throughout our tests yields trends in results and conclusions which retain their validity.

OPERATING PROCEDURE

Operating conditions and variable parameters for the case when the engine is fired are shown in Table 3. The injection system provides the following adjustments: variable timing for the start of the first injection, a variable time or crankangle "separation" between the start of the first and the start of the second injections, and the choice of the fuel quantities injected in the first and the second sprays. The single-injection data set when all the fuel is injected through one injector is produced with the rack of the second injection pump set to deliver 30 mm^3 per injection, see Fig. 2. The timing is then adjusted starting from 30^O to 10^O BTDC (i.e. before TDC) in 5 ^{O} increments. For the interacting-sprays data set when both injectors are used, each pump is set to deliver 15 mm3 for a total of 30 mm^3 per fired cycle. The choice of an equal split of fuel between the injectors for the interacting-sprays series is influenced by results of Pierpont et al. [14] and Tow et al. [11] who reported an optimum distribution with 50%/50% split. Following the interacting-sprays tests and at the end of the experiments, selected single-injection data points are repeated to examine drift and repeatability of the entire system. Accuracy of the results are at most +/- 1.8 % and +/- 3.45 % of the reported data for NOx and BSN (Bosch Smoke Number) measurements, respectively. A single-injection test series using the first pump instead of the second one is also conducted to assess any differences due to the use of the first versus the second fuel injection system, see Fig. 1(A). The results showed good agreement to within 5% between the use of the first and second injectors.

It was necessary to operate this research engine in a skip-fired mode due to the engine's limited cooling capacity and poor scavenging characteristics. In this mode, the engine is motored for a pre-selected number of cycles without fuel injection and then both injectors are activated for one cycle. Cylinder pressure data are acquired on the injected cycle. Operation in the skip-fired mode is accomplished through the use of fast-acting high-pressure solenoid valves installed in the fuel injector supply lines, see Fig. 1(A). For more details refer to Sinko et al. [23]

Before any data are taken, the cold engine is warmed up for nearly 20 minutes at the set conditions until the cylinder head temperature reached 85 ^{O} C. The temperature is then maintained between 85^O to 90^O C. Usually after about 5 minutes the warmed-up engine reaches a new steady state when the operating condition is changed. The NOx analyzer operation is monitored for stability on a strip chart recorder connected to the analyzer. A PC-based acquisition program first acquires cylinder pressure for 25 consecutive fired-engine cycles, before acquiring 1600 measurements from the analog output of the NOx analyzer. Then the smoke lines are purged by the compressed air supply and one smoke sample is drawn followed by a second one. The whole procedure is then repeated for another test condition. During all experiments the engine temperature as well as the oscilloscope displaying cylinder pressure, injector needle lifts, injection line pressures, and the pulse controlling the exhaust sampling solenoid are monitored for any abnormal behavior.

RESULTS and DISCUSSIONS

Before discussion of the emission results it is worthwhile to indicate that a large number of images are collected by Mie scattering (viewing only the liquid phase of the fuel spray) and Exciplex LIF (simultaneously viewing the liquid and vapor phases of the fuel spray) approaches in the non-fired warmed-up engine. Details of our initial image analysis are discussed in Sinko et al. [17], Campbell et al. [15], and Sinko et al. [26]. In this paragraph, important conclusions are briefed and interested readers are referred to these published works for more details. What we find interesting is an observation that the first spray in the interacting-spray injection system apparently establishes a more favorable flow field for less liquid fuel accumulation near the cylinder side walls than the case when single injection is used; see Campbell et al. [15]. Large fuel accumulation is particularly detrimental at high load conditions in diesel engines. To see this effect, Fig. 3 shows the total fluorescence intensity of the liquid phase (proportional to the mass of the liquid phase) within a narrow region near the viewable chamber wall (i.e. 15% of radius in thickness) for 20/40 (i.e. first and second injectors have 20 & 40 mm^3/inj, respectively) and 00/40 cases. In Fig. 3 substantial differences in the amount of the adhered liquid mass exist near the wall. Note that although for the 20/40 case the total amount of injected fuel is 50% more than the 00/40 case (i.e. 20 mm3/inj more), existence of the first injection substantially reduces near-wall liquid-phase fuel accumulation. Similar effect is also observed when stronger spray/spray impingement occurred. This is explained to be due to the nature of the momentum exchange between the two sprays during their impingement. There should, therefore, exist an optimum spray/spray impingement angle that minimizes near-wall liquid fuel accumulation while possessing good fuel penetration and spreading. This condition is currently being sought. Based on these visualization works, Sinko et al. [26] propose that existence of spray/spray impingement can enhance air utilization within the combustion chamber. Also, a suggestion is made that to achieve optimum simultaneous reduction in soot and NO_X, the timing between the first and second injections (i.e. dwell) should be chosen with due consideration of the interaction between the swirl flow and the first injected spray and also the relative magnitudes of the ignition delay period and the dwell. This suggestion is further examined in the following emission measurement part of this paper.

Being the main focus of this paper, emission results for the fired engine are presented for four injection strategies, each represented by a different symbol in Fig. 4, in which the beginning of the first injection pulse is varied for values of 30^O to 15^O BTDC. Figure 4 shows the chronology of the four injection strategies. For the interacting-sprays injection cases the beginning and the end of each injection pulse are clearly indicated in this figure. The position of ignition is denoted by the character "I" placed on the single injection time line. Exceptions are indicated by this same character along their time lines. The terms "beginning of injection" or "injection timing" without any specification refer to the crankangle at which the first injection pulse starts. At each injection timing a family of four time lines presents the scheduling of the first and second fuel injection events for each of the following injection strategies: (a)--Single injection--30 mm^3 of fuel is injected in a single continuous injection through the second injector. (b)--Coincident interacting-sprays injection--about equal amount of fuel, 15 mm^3, is injected through each injector starting simultaneously. (c)--Interacting-sprays injection when 2nd pulse ends at ignition--15 mm3 is injected in each injector. In case (c), the timing of the 2nd injection pulse is set so as it is completed near the end of the ignition delay period. Since ignition delay varies over the range of injection timing studied, in case (c) it was necessary to adjust the separation between the start of the first and the beginning of the second injection pulses at each injection timing. (d)--Interacting-sprays injection at about 12^O CA separation--the start of the 2nd injection is set at about 2.5 ms from the start of the first one resulting in a dwell period of 1.8 ms (i.e. time between the end of the first and start of the second injections.) In this last case and for all the injection timings tested, the second injection starts, with one exception, after ignition occurs. These choices of the injection pulse separation provide a wide range of values for ignition delay period and injection pulse separation period, the two relevant and important characteristic times for the interacting-spray injection system.

The influence of the interacting-sprays injection strategies on the NOx-smoke tradeoff can be seen in Fig. 5. Figure 5 shows the NOx-smoke tradeoff where the exhaust NOx concentration is plotted as a function of BSN. This type of tradeoff for diesel engines was first reported to exist by Yu and Shahed [27]. They show that all measures to decrease NOx cause increase in smoke and vice versa, hence there exists a tradeoff for the designer. Note that data for all the four injection timings of interacting-sprays injection tests are included in Fig. 5. In the interacting-sprays injection tests 15 mm^3 of fuel was introduced through each injector, maintaining the total amount of injected fuel per fired cycle (i.e. 30 mm^3) the same as that used in the single injection case. In Fig. 5, four different line types divide interacting-sprays injection data into four categories. In each category injection timing is fixed, but different than the other categories as indicated in the figure, and separation between the injection pulses is varied. This is how the tests are performed and data collected from the engine. The symbol convention employed in Fig. 5 conforms to that presented in Fig. 4. Extra data points are used to construct these four line categories but not represented by any symbol for clear distinction of the different injection strategies explained earlier (these strategies are distinctively shown by the symbols in Fig. 5). In examining Fig. 5 it is immediately apparent that all interacting-sprays injection cases offer superior NOx-smoke performance than that attainable from the single injection case. Furthermore, and very interestingly, it is generally found, with one exception, that the best NOx-smoke performance within the interacting-sprays injection cases is obtained when the two injections are coincident (i.e. almost all interacting-sprays injection data lie above and to the right of the locus of the coincident injection data points formed by connection of the hollow circles). The finding of this optimum condition with coincident interacting-sprays injection case is in contrast to the findings of Tow et al. [11] who reported an optimum dwell period of about 10^O crankangles between the two or three injection pulses in an engine operating at 1600 RPM and at 75% load. Note that Tow et al. [11] produced two injection pulses through a single injector while two separate injectors are used in interacting-sprays injection system with flexible targeting capabilities.

Figure 6 is a plot in which emission and engine performance data for the interacting-sprays injection cases are superimposed on the single injection data. Single-injection combustion data in our research engine can be examined to establish a baseline for comparison with the interacting-sprays injection tests. Single injection combustion data is acquired at various injection timings ranging between 30^O and 10^O BTDC. Note that NOx emission levels decrease and BSN increases as injection timing is retarded for the single injection case. Hence, the data for this set in Fig. 6 shows the expected trends in combustion parameters and emissions as fuel injection timing is retarded. As timing is retarded from 30^O to 10^O BTDC the ignition delay is found to decline from 2.6 to 1 ms (12.5 to 4.8 crankangles), indicating that in-cylinder air state is highly determinate of the ignition kinetics. The indicated work obtained by integration of the ensemble-averaged cylinder pressure curve reveal (not shown in the figures) a wide maximum somewhere between 15^O to 10^O BTDC, corresponding to MBT (injection timing at which maximum torque is achieved) injection timing for this engine. As expected, peak cylinder pressure is found to continually decrease with retarded injection timing, see Heywood [28]. In Fig. 6, values for the fraction of the total apparent heat released in the premixed phase are seen to continuously decline as injection is retarded from the early positions where long ignition delay periods permit a high degree of mixture preparation prior to ignition. For this series of single injection tests, the duration of injection is found to be 2.29 ms (11 crankangles) indicating that for the 30^O and 25^O BTDC injection timing cases the period of fuel injection is fully confined within the ignition delay period, see Fig. 4. For the more retarded cases, however, the injection extends into the post-ignition active combustion period, similar to practical engines, accounting for the continued decline in the premixed fraction.

In considering Fig. 6 it is most instructive to concentrate on the two most extreme interacting-sprays injection cases, the coincident injection (hollow circles) and the 12^O (i.e. maximum) separation (hollow squares), in relation to the single injection case (filled circles). In the bottom frame of Fig. 6, it is seen that NOx emission is effectively reduced (by 20 to 35 %) below the single injection case when interacting-sprays injection is used at the largest separation tested (2.5 ms or 12^O CA). In the case of coincident interacting-sprays injection, the NOx emission level is found to be somewhat elevated. The reduction in NOx for the maximum separation case may be explained by the introduction of the fuel from the second injection pulse into a burning (gases) environment established by the first injection pulse. This lowers local temperatures, exponentially affecting reaction rates, and consequently reduces local NOx formation rates. Reference to Fig. 4 shows that only for this maximum separation case that the second fuel injection pulse is always active past the time of ignition. This allows for mixing of the cooler fuel with already burned or burning gases hence quenching local NOx chemistry.

Figure 6 also shows the effects of interacting-sprays injection strategies on exhaust smoke levels. Here, and very interestingly, the benefit achievable from interacting-sprays injection strategies is seen to be maximized for the case of coincident injection. BSN is reduced by about 45 %. This benefit declines progressively as one increases the separation between the first and second injections until the maximum separation case which exhibits smoke emission levels comparable to the single injection case. The effectiveness of the coincident interacting-sprays injection may be explained in terms of the unique injection system used in this study and the knowledge of the mechanism of soot formation. In-cylinder visualization by Pushka et al. [16] and Sinko et al. [17], and Campbell et al. [15] in this engine indicate that the two sprays impinge closer to the injectors than to the facing wall. This allows for spray/spray impingement that is maximized in the case of coincident injection. It is known that soot formation is initiated in the dense regions of the diesel fuel spray, see Heywood [28]. He indicates that dense local regions of soot are formed early in the combustion process (during injection period) in the vicinity of the spray centerline. Most of this soot is subsequently oxidized during the later stages of combustion, with tailpipe emissions comprised of the remnant of incompletely oxidized particles. The results presented in this paper suggest that the reduction of soot associated with interacting-sprays injections may be due to interactions between the injection pulses (spray/spray impingement) that disrupt and disperse the dense spray regions which provide nucleation sites for soot particle formation. These interactions are maximized in the case of coincident injection, but also exist in cases where time separation between the first and second injections are short enough to cause sufficient spray/spray impingement. If time (or crankangle) separation between the two injection pulses is too large (such as 12^O CA), our visualization work showed that the two sprays miss each other with minimal interactions in the early phases of injection and combustion. The picture presented here is consistent with in-cylinder visualization works by Pushka et al. [16] and Campbell et al. [15] in this same engine. If the two injection pulses are produced by a single injector (as commonly done), the penetration of the lead spray is impeded by the injection into an undisturbed environment, whereas the trailing spray advances in the wake of the lead spray, eventually overtaking and interacting with it. At large values of dwell, the trailing spray is unable to overtake the lead spray and spray/spray interactions may not be present. At very small values, it is ineffective as shown by Tow et al. [11]. In any case, more spray/spray interactions, disruptions of the rich spray core region, and mixing effects are achieved in our dual-injector interacting-spray system than if they were produced by a single injector through split injection. In fact spray/spray impingement is used for liquid fuel atomization and mixing in rocket engine fuel nozzles. Also, this type of impingement may lower the need for very high injection pressures required for good atomization and mixing hence reducing the bulk weight of the injection system as well. Note that high injection pressures are also used for smoke reduction in diesel engines through its enhances air/fuel mixing reducing the richness of the spray core zone.

In experiments similar to cases (b) and (d), but two sprays are injected to diverge and not impinge each other, results for BSN and NOx are to within 6.4% of the values for the single injection case with NOx predominately lowered. This suggests that the observed trends are due to the proposed interactions mentioned earlier. The results of this study suggest that two distinct mechanisms may account for the observed effects of different interacting-sprays injection strategies in improving the NOx-smoke performance of DI diesel engines. At large time separations between the first and second injections, enhanced mixing of the newly injected liquid fuel and chamber air into the burning and/or burned gases zones reduces NOx emission without benefit (or possibly a degeneration) in smoke emission. At minimal separations, spray/spray interactions destroy the locally-rich spray regions within the spray core and suppress soot formation early in the combustion period. This latter mechanism suggests a new or additional interpretation of the reduced soot emission mechanism observed in split injections. In discussing the effects of dual and triplet injections on the particulate formation, Tow et al. [11] using a single injector suggest that the mechanism of particulate reduction may be increased particulate oxidation later in the cycle rather than a reduction in the amount of particulate formed initially. By using an interacting-sprays injection system which creates strong interaction between the injected sprays early in the cycle, the data of the present study suggests the conclusion that suppression of early cycle soot formation is the mechanism at work in our engine. In assessing the strength of the two mechanisms, reference to Fig. 5 shows that optimum NO_X-smoke benefits are present in the coincident case. This indicates that the greatest benefit would be gained by employing the soot suppression capability of the coincident interacting-sprays injection, while NOx reduction be achieved from conventional injection timing retardation. An attractive approach is to consider an interacting-sprays injection system where one injector produces two injection pulses at large separations for NO_X reduction, and the two injectors are strategically positioned for early impingement of the sprays produced by them for soot formation control. In this proposed design one, therefore, may not need to control NO_X by retarding injection timing. Also, one injector can inject water to reduce NO_X and perhaps smoke as well due to the spray/spray impingement effects shown earlier. Obviously, the actual final design for use in production engines may not be identical to what is used for research purposes, but should incorporate the ideas and the concepts that are introduced in this paper.

It is useful and important here to briefly discuss results from the only-known similar work recently presented by Takeda and Niimura [29] in which two additional independent side injectors (each producing two sprays) are used diametrically-located near the cylinder wall in the head of a four-valved DI-injection diesel engine with conventional centrally-located injector (generating six sprays). Their extensive study consider effects such as orientations of the spray/spray impingement, fuel quantities in the central and side injectors, and time separation between the side and central injections. They clearly show that one needs to carefully optimize these parameters for best results. Orientation and the nature of the impingement is important and when the side sprays are simultaneously initiated 9 degrees after the center injection (happened to be at 3 degrees after TDC) and are directed to interact with the region of the centrally-injected sprays closer to the injector exit holes, maximum smoke reduction and maximum NOx increase are observed compared to when all the fuel are injected through the central injector. The BSN is reduced from 2.2 to 0.25 when side injectors are used. Hence, although BSN of our research engine is high, it is expected that the interacting-sprays injection concept and benefits can be carried to more advanced production engines. Strictly, this prediction is to be examined. Takeda and Niimura [29] also show that 5 to 10 % of the total fuel in each side injector provides the best smoke reduction and larger percentages deteriorates this effect. This effect is being investigated in our injection concept. Note that their spray/spray impingement is not as strong as in our coincident injection case. The smoke reduction of their strategy was explained to be due to the orientation of the side sprays with respect to the centerally-injected ones, being such that the fuel from the side sprays utilizes the unused air in the central region of the combustion chamber. Our interacting-sprays approach, however, proposes early spray/spray impingement for maximum smoke reduction. Finally, they attempt split injection in every injector (both central and sides) and show that one can find an injection schedule that simultaneously reduces NO_X and smoke but there is a fuel economy penalty. Although not fully optimized, our interacting-sprays did also show 5 to 10 % increase in the indicated specific fuel consumption in this research engine.

It is also noteworthy to indicate that because of our low compression ratio research engine, the gas-to-liquid density ratio at the time of injection (estimated to be about 0.013) is at the low end of the typical values in diesel engines ranging from 0.015 to 0.030. This affects spray spreading angle and atomization through their square-root-of-density-ratio dependency. Spray/spray impingement is particularly beneficial for this case to break up the intact core of the two interacting sprays. With recent interest in gasoline direct injection, interacting-sprays injection can also be a useful feature to investigate.

SUMMARY and CONCLUSIONS

1. The key concept in the proposed Interacting-Sprays injection system is the interaction of the two jets (or sprays) at the right time and place inside the combustion chamber to achieve reduction in smoke and NO_X emissions. Although not fully tested, the location of the jets impingement and the impingement angle are expected to be two important parameters for optimization.

2. It appears that the interaction between the two sprays within the combustion chamber can cause a reduction of the near-wall liquid fuel accumulation, which should reduce unburned hydrocarbon emission. Also, from flow visualization works by the authors, it was suggested that spray/spray impingement has enlarged the regions where liquid fuel sprays present and consequently increased the potential for enhancement of air utilization within the combustion chamber.

3. Results indicate that stronger interactions between the two injection pulses (i.e. spray/spray impingement) early within the ignition delay period is critical to a reduction in soot formation. Simultaneous double injection with strong spray/spray impingement produced the lowest soot in our research engine (reduction by about 45 %). This is explained to be due to additional mixing and atomization effects of the impingement which reduced and disrupted soot-producing rich zones within the core of the sprays.

4. For effective NOx reduction the second injection pulse should start near or a little beyond the ignition delay period in order to lower the local burned and burning gases temperatures hence reducing the NOx formation rate (reduction by 20 to 35 %).

5. One strategy using the proposed interacting-sprays injection system to simultaneously reduce NOx and soot would be double near-simultaneous injection (for soot reduction) retarded for NOx reduction.

6. Based on the results presented here a new low-pressure interacting-sprays double-injector injection system is proposed in which the two injectors are strategically located to cause early spray/spray impingement (for example by near-simultaneous injections) for soot reduction, while one of the injectors produces a late second spray to lower the NOx formation rates. Note that a new centrally-located single-injector injection system can possibly be designed to achieve the spray/spray impingement concept and strategy introduced in this paper. Obviously, the actual final design for use in production engines may not be identical to what is used for research purposes, but should incorporate the ideas and the concepts that are introduced in this paper. Due to the added atomization and rich spray-core disruption effects of jet/jet impingement this proposed injection concept may not require a very high injection pressures used to enhance mixing in order to lower exhaust smoke number. This, additionally, can lead to lower weight of the injection system.

7. More data at different percentage of fuel per injector and at different engine speeds along with impingement angle optimization are needed to examine our conclusions farther beyond the region investigated so far .

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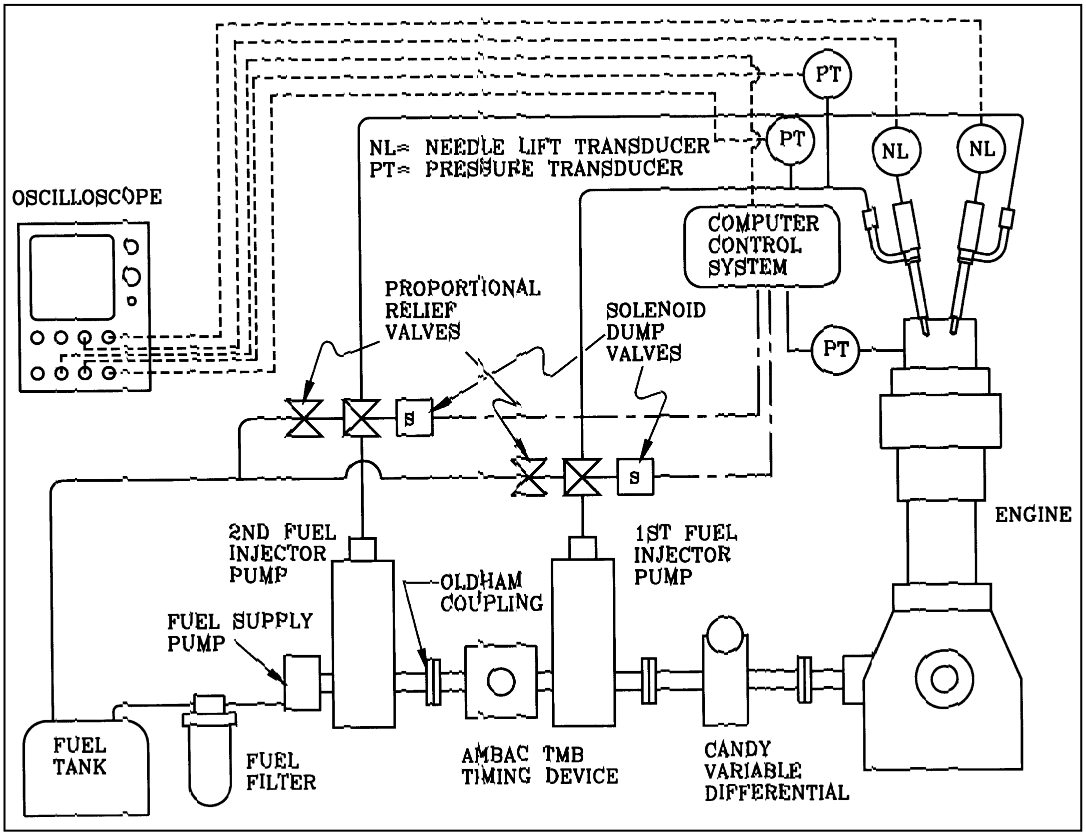


Figure 1(A). Schematic diagram of the setup for the interacting-sprays injection system. Continuous solid lines represent actual fuel lines. Broken lines are to show electrical control wire connections

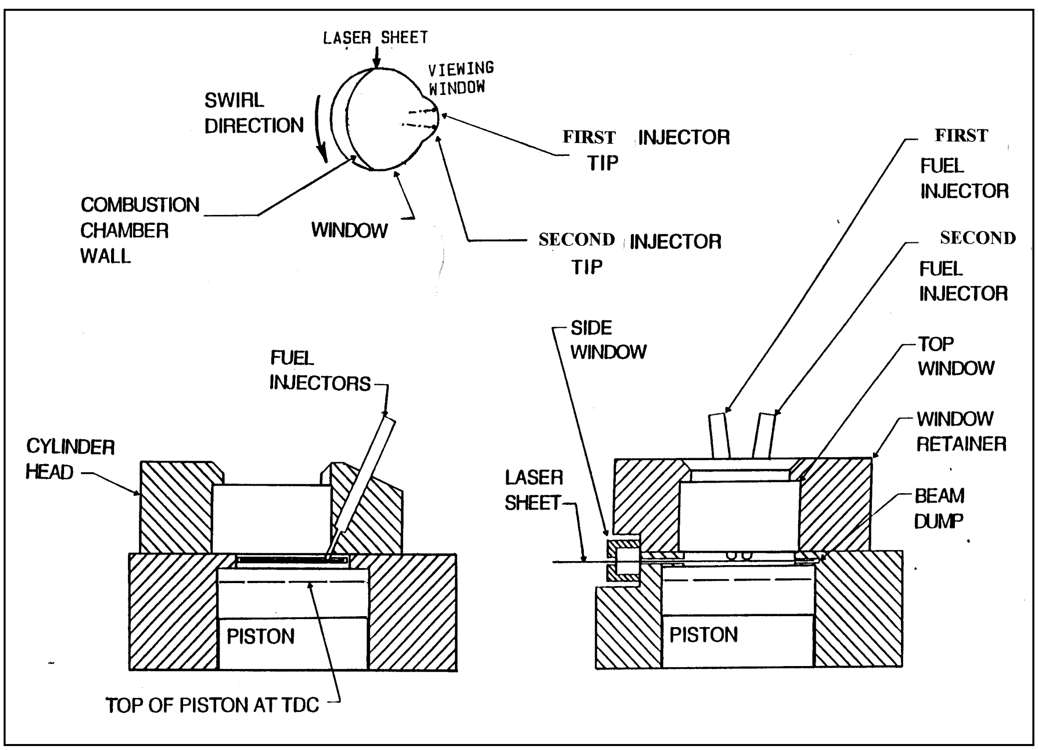


Figure 1(B). Schematic diagram of the engine head showing orientation of the injectors in different views.

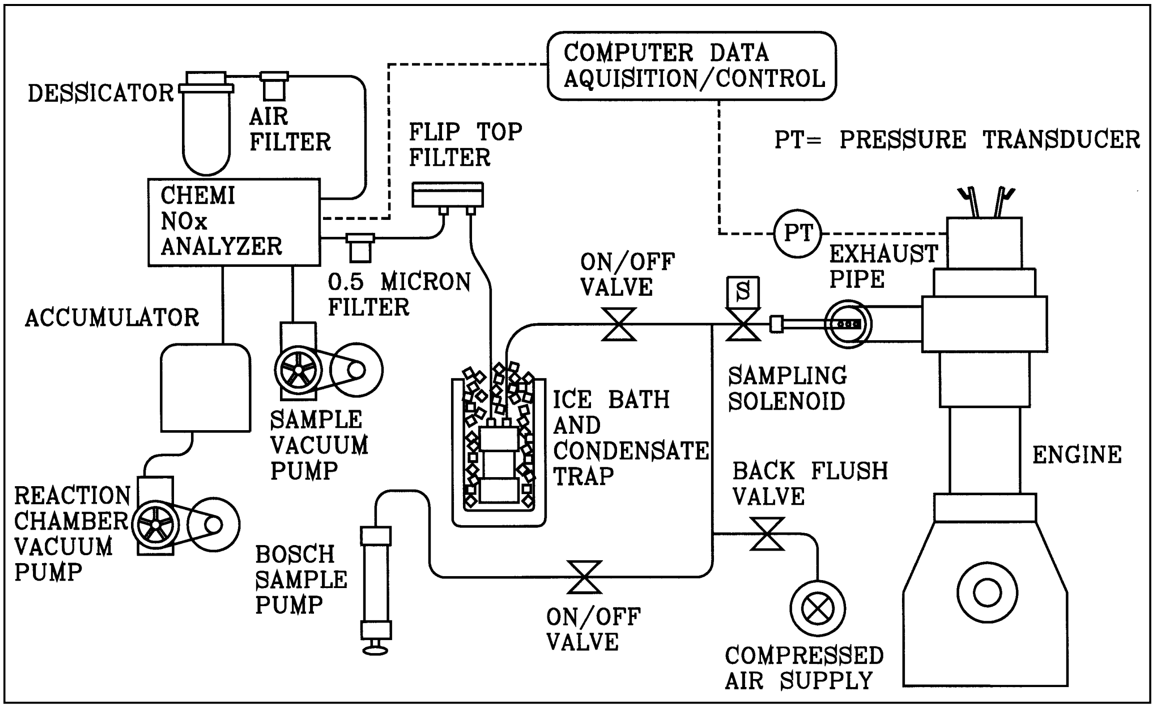


Figure 2. Schematic of the exhaust NO_X and smoke measurement systems.

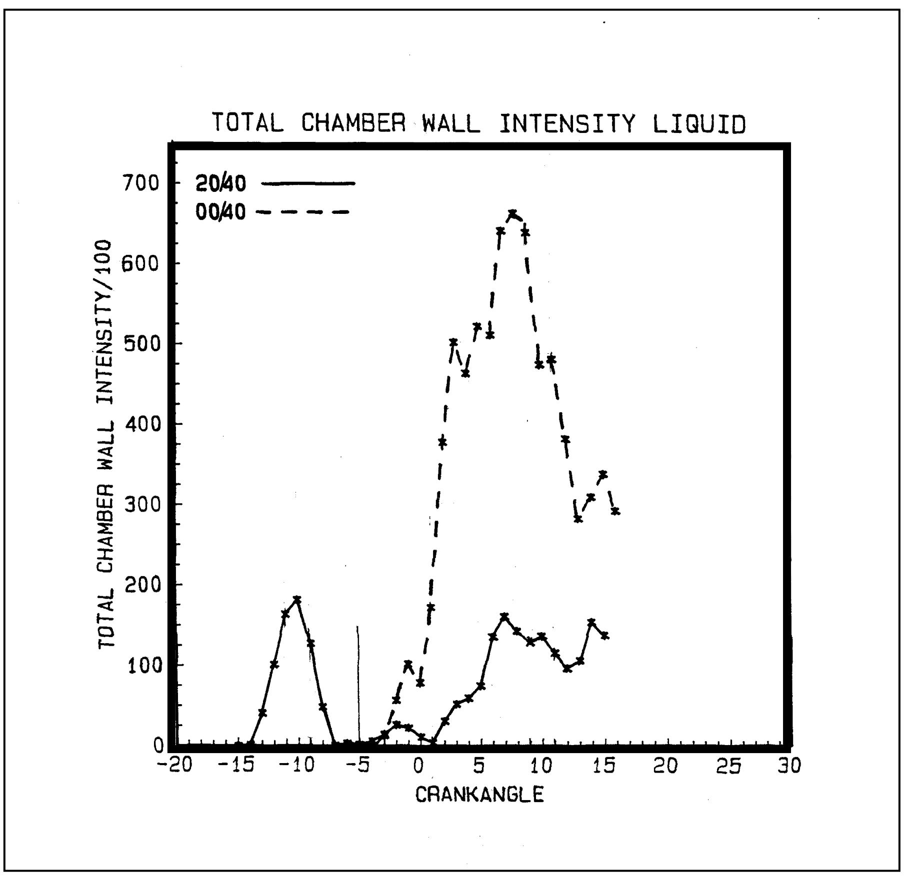


Figure 3. Plots of the total collected fluorescent light intensity from the liquid phase fuel within a narrow crescent-like region near the viewable chamber wall facing the injectors as a function of crankangle degrees.

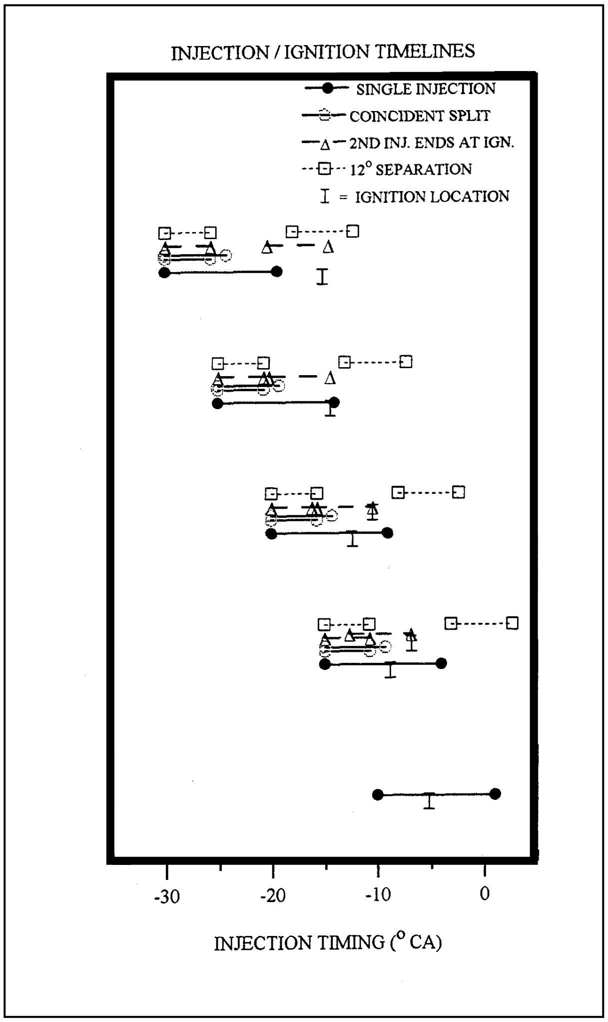


Figure 4. Time lines showing beginning and the end of the first and second injections for the four injection strategies discussed in the text. For horizontal axis, zero and negative numbers are at TDC and before TDC (i.e. BTDC), respectively. Injection scheduling: (a)--Single injection, (b)--Coincident split, (c)--2nd inj. ends at ign., (d)—12° Separation. The ignition location is indicated by the capital letter "I"

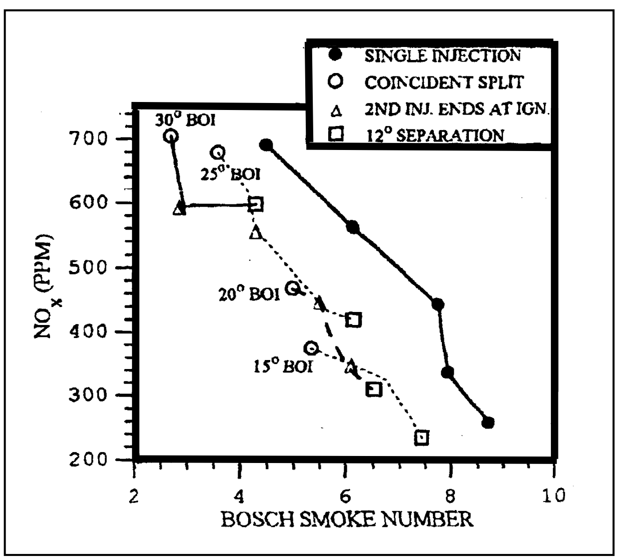
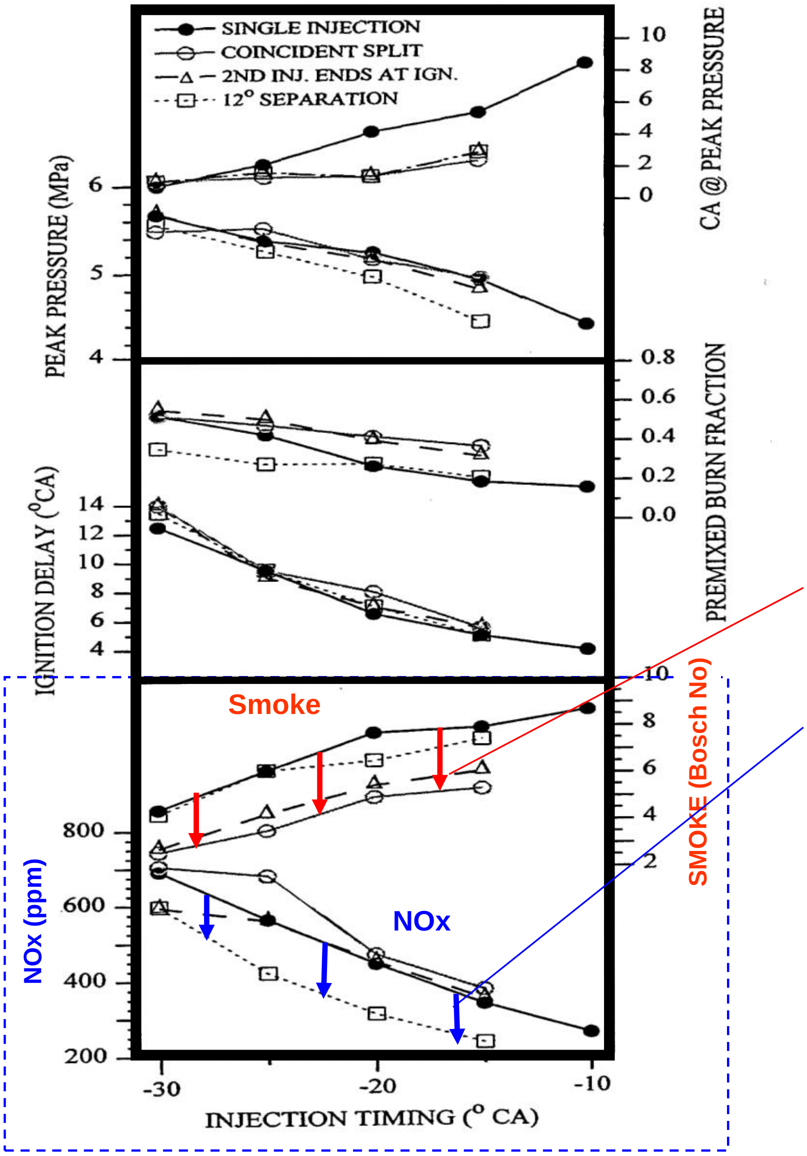


Figure 5. The NOx-smoke tradeoff for the four injection strategies discussed. BOI stands for the beginning of the injection. Measurement uncertainty: NO_X = +/- 1.8 % of the reported data; BSN= +/- 3.45 % of the reported data.



Simultaneous reduction of NOx and Smoke

- Simultaneous injection leads to reduction in smoke
- Injection with time separation leads to reduction in NOx
- Strategies can be envisioned for simultaneous reduction in smoke and NOx

Figure 6. Results for all the interacting-sprays and single injection strategies plotted as functions of crankangle degrees. For horizontal axis, zero and negative numbers are at TDC and before TDC (i.e. BTDC) respectively. Injection scheduling: (a)--Single injection, (b)--Coincident split, (c)--2nd inj. ends at ign., (d)--12 ^{o} Separation. Measurement uncertainty: NO_X = +/- 1.8 % of the reported data; BSN= +/- 3.45 % of the reported data.

TABLE 1
Engine Specifications

Displacement: 612 cm^3 (37.3 in^3)
Bore: 82.6 mm (3.25 in)
Stroke: 114.3 mm (4.50 in)
Clearance Height: 12.78 mm (0.503 in)
Connecting Rod Length: 254.0 mm (10 in)
Compression Ratios:
Geometric: 16.2:1
Trapped: 13.7:1 (Top of Piston)
Trapped: 10.5:1 (Top Ring)
Crankcase: Waukesha CFR High Speed.
Scavenging: 2-Stroke Cross-Scavenged
External Compressor
Port Timing (based on Top of Piston):
Intake: +/- 126° (TDC* chosen to be at zero degree)
Exhaust: +/- 126°
Port Geometry:
Exhaust: 6 ports, 19.5 mm high, 17.5 mm wide.
Intake: 6 ports, 15 mm high, 6.5 mm wide. 30° to the horizontal 40° to the cylinder radius.
Swirl No. 4 (estimated at TDC)
*TDC stands for Top Dead Center when piston is at its highest position.

TABLE 2
Injection System Specifications
Fuel Injector Nozzles:
Stanadyne Slim-Tip Pencil Nozzles
Single Spray Orifice Drilled at 66 degrees to nozzle axis
First: One hole @ 0.43 mm (0.017 in) dia.
L/D Ratio: 1.53
Second: One hole @ 0.38 mm (0.015 in) dia.
L/D Ratio: 1.73
Measured spray impinging angle from images: 12 ^{O} to 14 ^{O}
Valve Opening Pressure: 20.8 MPa (2800 psig)
Injection Timing:
First: Continuously variable w.r.t. Crankangle position
Second: Continuously variable w.r.t. Crankangle position
Fuel Pumps: Ambac APE
Cam Profiles: Ambac # 1 Basic Metric
Fuel Pump Plunger Diameters:
First: 8.0 mm
Second: 5.0 mm
Fuel Pump Delivery Valve Retraction Volumes:
First: 50 mm^3.
Second: 50 mm^3.
Injection Line:
Internal Dia.: 2 mm (0.079 in)
Length:
First: 1.05 m (41.3 in)
Second: 1.23 m (48.4 in)

TABLE 3
Experimental Conditions and Parameters
for the fired case

Engine Speed:	800 RPM
Fuel Rate (First/Second Injections):	30/00 and 15/15
Air mass flow rate:	0.65 kg/min
Intake Manifold Temperature:	187 ^{o} C
Intake Manifold Average Pressure:	120 KPa
Exhaust Manifold Average Pressure:	100 KPa
Cylinder Head Temperature:	~ 85 ^{o} C
Injection Timing (First Injection):	30, 25, 20, 15, 10^{o} BTDC
Double Injection "Separation":	0 - 12 ^{o} CA
	(Separation: Start of First to start of Second injections)
Fuel:	Commercial #2 Diesel + 3% Isopropyl Nitrate

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