

# Experimental and Numerical Investigations of Close-Coupled Pilot Injections to Reduce Combustion Noise in a Small-Bore Diesel Engine

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## ABSTRACT

A pilot-main injection strategy is investigated for a part-load operating point in a single cylinder optical Diesel engine. As the energizing dwell between the pilot and main injections decreases below 200  $\mu$ s, combustion noise reaches a minimum and a reduction of 3 dB is possible. This decrease in combustion noise is achieved without increased pollutant emissions. Injection schedules employed in the engine are analyzed with an injection analyzer to provide injection rates for each dwell tested. Two distinct injection events are observed even at the shortest dwell tested; rate shaping of the main injection occurs as the dwell is adjusted. High-speed elastic scattering imaging of liquid fuel is performed in the engine to examine initial liquid penetration rates. The penetration rate data provide evidence that rate shaping of the initial phase of the main injection is occurring in the engine and that this rate shaping is largely consistent with the injection rate data, but the results demonstrate that these changes are not responsible for the observed trend in combustion noise.

A zero-dimensional model is created to investigate the causes of the observed combustion noise behavior. The trend in simulated combustion noise values agree well with the experimentally determined trend, which is associated with two main factors: relative changes in combustion phasing of the pilot and main heat release events and suppression of the pilot apparent heat release for dwell times near the minimum-noise dwell. Two possible mechanisms by which the relative phasing between the pilot and the main heat release events impacts combustion noise are proposed.

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## INTRODUCTION

Pilot injections have long been used in direct injection Diesel engines as a means to reduce combustion noise [1,2]. The mechanism by which noise is reduced is as follows: the heat released by the pilot fuel increases temperatures and radical concentrations within the cylinder and therefore decreases the ignition delay of the fuel injected during the main injection. Consequently, heat release rates during the main portion of the combustion are predominantly mixing-controlled and smaller in magnitude than for premixed combustion. The relatively small amount of pilot fuel, combined with the interruption in the rate of injection, results in a smaller amount of premixed combustion than for a single injection. In this way, combustion noise is suppressed through the use of a pilot. Advances in fuel injection hardware, such as common rail systems and fast response injectors, have made multiple injection strategies robust and effective. Piezo injectors are well known for their precision and short reaction times, but state-of-the-art solenoid injectors can reliably deliver up to eight injections per cycle with very short separation times between injection events [3]. Split armature modules enable closely spaced injections and even zero hydraulic dwell (delay) between injection

events. With this system, injection rate shaping is possible [3, 4]. Various definitions of dwell can be found in the literature, but for this study, dwell refers to the time between the end of the pilot energizing signal and the beginning of the main energizing signal (see [Figure 1](#)).

Dürnholtz *et al.* investigated the effects of a pilot injection on combustion noise, fuel economy, and emissions with various pump-line-nozzle injector configurations in a passenger car Diesel engine of unspecified displacement [2]. Their injection system made it possible to vary the quantity, timing, and rate of the pilot injection. For an engine speed of 2000 rpm and a "medium" load, combustion noise depended strongly on the pilot quantity; minimal combustion noise levels were achieved with a pilot quantity of between 2 and 3 mm<sup>3</sup>/str. The noise minimum corresponds to a minimum in ignition delay for the main injection, and to approximately 8% of the fuel being burned before the start of the main injection. Measured combustion noise was always lower with a pilot than without, but the noise increased for hydraulic dwells (injection interrupt durations) shorter than approximately 12 °CA (1000  $\mu$ s at 2000 rpm).

Badami *et al.* performed investigations using a common rail system with solenoid injectors in a passenger car Diesel engine [5]. They examined three different operating points with pilot-main injection strategies and varied pilot injection quantity and energizing dwell at each point. At a fixed load, pilot injection quantity affected not only the combustion noise at a given energizing dwell, but also the trend in combustion noise with changing dwell. Moreover, trends in combustion noise with changing dwell for different operating points were quite different for a given pilot injection quantity. At moderate engine loads (bmep/n: 5 bar/1500 rpm and 8 bar/2000 rpm), combustion noise sometimes changed dramatically for energizing dwells shorter than 500  $\mu$ s. At the highest load tested (8 bar/2000 rpm) and for the larger pilot injection quantities tested, combustion noise was observed to decrease dramatically at the shortest energizing dwells (near 86  $\mu$ s). Badami *et al.* show that by varying only the dwell for a given pilot injection quantity, combustion noise can vary by as much as 3 dB; this is equivalent to a factor of 2 in acoustic power.

Ricaud and Lavoisier optimized multiple injection settings in a single-cylinder, small-bore engine with a piezoelectric injector for several multiple injection strategies [6]. Dwell and quantity were varied for each injection event, as well as the start of the main injection, the rail pressure, and the EGR rate. For a pilot-main injection strategy, combustion noise varied significantly as dwell and pilot injection quantity were changed. Furthermore, the pilot fuel quantity resulting in the lowest combustion noise level depended on the dwell. The noise-optimal pilot injection quantity was smallest for intermediate dwells. For noise-optimized operation, peak heat release rates are the same for the pilot and the main combustion events. This observation was made for several operating points, and it was suggested that the noise-optimized settings are a balance between excessive pilot heat release and excessive main combustion heat release.

Properly applied multiple injection strategies can be beneficial for combustion noise, fuel consumption, and pollutant emissions, and are therefore a powerful tool. However, with so many calibration parameters, including injection and dwell durations for each injection event; rail pressure; and injection train phasing; developing engine calibrations over the entire operating map is a complicated and time consuming process. A particular injection train may yield minimal combustion noise for a given operating point, but it may not be obvious why these parameters are optimal or why different parameters are necessary for a different operating point. Without a detailed understanding of the mechanisms by which noise can be decreased with multiple injections and closely coupled pilot injections, it will be difficult to reduce the amount of effort necessary to develop and optimize such calibrations. The goal of this work is to investigate the effects of energizing dwell time on combustion noise with a pilot-main injection strategy and to describe the mechanisms responsible for the observed trend in combustion noise.

In this study, a pilot-main injection strategy is investigated at moderate load in an optical small-bore Diesel engine. The preproduction injector used is a fast-acting, pressure-balanced, solenoid-actuated injector, which enables very closely coupled injection events. Engine testing is performed for a dwell sweep at a

moderate load, and trends in exhaust emissions, heat release rates, and combustion noise are measured. The injection schedules used in the engine testing are measured with a hydraulic injection analyzer to provide rate shapes for each of the dwells tested. High speed elastic scattering images are taken in the engine to provide information about liquid fuel penetration and penetration rate behavior for several dwells. A zero-dimensional computational model is implemented to systematically provide insight about the observed trends in combustion noise and their relation to injection behavior and heat release phasing.

## EXPERIMENTAL SETUP AND PROCEDURES

### Engine and Fuel Injector

The single-cylinder optical Diesel engine used in this study is based on a modified General Motors cylinder head with four valves per cylinder. The production version of this engine has four cylinders and a displacement of 1.9 L. The main geometric specifications are provided in Table 1. The engine features a Bowditch-style piston, which is fitted either with a titanium piston top for cylinder pressure and emissions measurements; or with a fused-silica piston top for the optical investigations described below. The geometrically identical piston tops feature a reentrant bowl and valve cut-outs.

Table 1. Engine geometry and fuel properties

|                             |                                                                                                                                              |
|-----------------------------|----------------------------------------------------------------------------------------------------------------------------------------------|
| Valves                      | 4                                                                                                                                            |
| Bore                        | 82 mm                                                                                                                                        |
| Stroke                      | 90.4 mm                                                                                                                                      |
| Displacement volume         | 0.477 L                                                                                                                                      |
| Geometric compression ratio | 16.7 : 1                                                                                                                                     |
|                             |                                                                                                                                              |
| Fuel                        | 58 vol% 2,2,4,4,6,8,8-heptamethylnonane (iso-C <sub>16</sub> H <sub>34</sub> );<br>42 vol% n-hexadecane (n-C <sub>16</sub> H <sub>34</sub> ) |
| Cetane number               | 50.7                                                                                                                                         |

Optical access through the sides of the combustion chamber is provided by fused silica windows (50 mm width, 25 mm height) located at the top of the cylinder liner. These windows are positioned to allow unobstructed optical access into the clearance volume, even when the piston is located at top dead center (TDC). The engine is equipped with a Kistler 6125b pressure transducer, which is mounted in place of the glow plug. The cylinder pressure is digitized every 0.25 crank angle degrees (CAD).

A blend of the diesel reference fuels, *n*-hexadecane and heptamethylnonane, was used in this study. This blend has been chosen to match the ignition delay and physical properties of conventional diesel fuel more closely than the blend of *n*-heptane and isooctane used previously [7], and has been used in more recent studies [8, 9]. Its cetane number of 50.7 is calculated from the relationship given by Heywood [10].

The fuel injector used in this study is a pre-production solenoid-based injector, which is equipped with a pressure-balanced nozzle control valve to enable very closely spaced injection events. The mini sac

nozzle has equally spaced holes with exit diameters of 139  $\mu\text{m}$ ; the included angle of the holes is 149°. Flexible control of solenoid energizing durations (referred to as ET) and energizing dwells was provided by a Genotec injector controller. The dwell was varied between 80 and 1200  $\mu\text{s}$ .

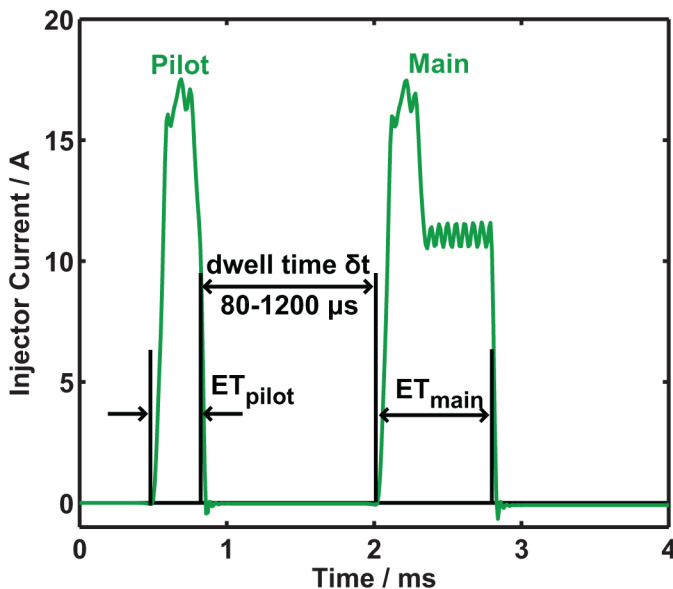


Figure 1. injector solenoid energizing and dwell definition

### Fuel Injection Rate Measuring Device

Rates of injection were measured with the same fuel injection hardware used in engine testing, but with the injector mounted in a Moehwald HDA rate and quantity measuring device [11]. The HDA utilizes the change in hydraulic pressure that occurs when fuel is injected into a constant volume, pressurized, fuel-filled chamber. Additionally, the speed of sound of the fuel in the chamber is measured before and after each injection train. Injection mass flow rates and quantities are then computed from the chamber pressure and speed of sound measurements. The measurement hardware, working principle, and experimental setup are described in more detail in [12].

### Fuel Injection Imaging Setup

An elastic scattering imaging technique was developed to provide temporally and spatially resolved information about the fuel injection process in the motored engine. An evaluation of the dynamic liquid fuel penetration behavior provided a qualitative check of injection rates measured with the HDA to verify trends in main injection rate shapes as dwell changes. Imaging was performed with a Photron SA-X2 monochromatic CMOS camera at a frame rate of 120 kfps; this provides adequate temporal resolution (0.075 CAD at 1500 rpm) of the pilot and main injection events for even the shortest dwells. Images were captured through the bottom of the fused silica piston with a Nikkor lens (focal length: 105 mm, aperture: f/3.5). The cylinder head and valves were airbrushed with a thin coat of matte black heat resistant paint to reduce reflections. A high repetition rate, high intensity, short pulse LED illumination system was employed to provide sufficient image intensities at these high frame rates. A red LED with a 3×3 mm<sup>2</sup> chip proved to be robust and sufficiently bright with a pulse duration of 1.5  $\mu\text{s}$ , using a peak forward current

exceeding 45 A. The LED pulses were timed so that the exposure of each camera frame began approximately 1  $\mu\text{s}$  before the LED pulse and ended several microseconds after the pulse. A system of three plano-convex lenses was developed to increase the amount of light that was passed into the cylinder. The setup in the test bench including the “focusing optics” is depicted in Figure 2 along with an example of a raw image.

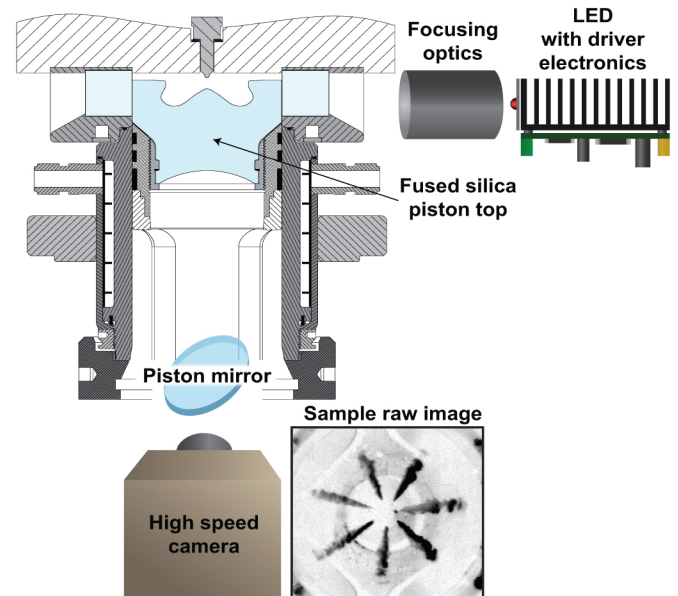


Figure 2. Setup for high speed elastic scattering imaging and sample raw image (inverted grayscale color map, 0-512 counts, gamma = 0.7)

The LED and its lens system were positioned outside of the liner window on the intake side of the engine. The incoming light is scattered by the strong density gradients at the fuel jets, and the angle of incidence relative to the jet axis is different for each jet. This non-axisymmetric illumination can complicate the image analysis, but each jet is treated independently in the image processing. Errors associated with non-uniform illumination are negligible in comparison to the changes observed with changing dwell. The trigger to start acquiring images is the signal used to trigger the injector driver to start injecting, i.e. it occurs at the start of solenoid energizing (SSE). In each injected cycle, 300 images are taken to capture the entire pilot and main injection events. Images are taken for 45 consecutive (skip-fired) cycles for each dwell tested.

The raw images are distortion corrected with an automated routine based on ray tracing; background subtracted using the approximate median method to model the background; and thresholded before liquid penetration lengths and penetration rates are computed from them. These procedures are described in the Appendix, where an example of a processed image is also shown.

### Experimental Procedures

Initial testing was performed with the metal piston to characterize the fired operation of the engine for a variation of dwell. For these tests, the engine was operated with a constant engine speed; IMEP<sub>g</sub>; injection pressure; and pilot injection quantity. The engine was operated in skip-fire mode: fuel was injected in every fifth

cycle. The main injection duration was adjusted to maintain the engine load ( $IMEP_g$ ), and the injection train was block-shifted to achieve a near-constant 50% mass fraction burned angle (MFB50) of 13 CAD ATDC. EGR was simulated by replacing intake air with a mixture of nitrogen and  $CO_2$  and matching the total mass in the cylinder to the mass that would be present with real EGR. The residual fraction that would be expected in a continuously fired metal engine was also simulated in this manner. The mass flow rate of intake gas was held constant throughout the testing and resulted in a gas density of approximately  $21.8 \text{ kg/m}^3$  at TDC for motored operation. Exhaust emissions ( $NO_x$ , smoke number,  $CO$ , and UHC) were measured during skip-fired operation and corrected to represent emissions levels obtained in continuously-fired operation. For each operating point, the engine was motored for approximately 90 seconds to allow the intake and exhaust pressures to stabilize. An emissions measurement was taken just before the engine was skip-fired and subtracted from the subsequent skip-fired measurement; a second emissions measurement was taken after approximately 60 seconds of skip-fired operation and averaged over 20 seconds of operation. This procedure removes background emission levels and mitigates the effects of instrument drift, so the resulting values can be attributed to the combustion. The methodology by which emissions values are computed is briefly discussed in [13].

Table 2. Engine operating parameters

|                                                                                               |                                                                                                                |
|-----------------------------------------------------------------------------------------------|----------------------------------------------------------------------------------------------------------------|
| Engine speed                                                                                  | 1500 rpm                                                                                                       |
| Swirl ratio (Ricardo)                                                                         | 2.2                                                                                                            |
| Intake pressure                                                                               | 1.51 bar                                                                                                       |
| Intake temperature (measured in runner)                                                       | 353 K                                                                                                          |
| EGR rate (simulated)                                                                          | 10.3% (includes 3.3% residual fraction)                                                                        |
| $[O_2]$ (intake)                                                                              | 19.7                                                                                                           |
| Intake gas flowrate                                                                           | 8.5 g/s                                                                                                        |
| TDC temperature                                                                               | 925 K                                                                                                          |
| TDC density                                                                                   | $21.8 \text{ kg/m}^3$                                                                                          |
| Injection pressure                                                                            | 800 bar                                                                                                        |
| $m_{pilot}$                                                                                   | 1.5 mg<br>0 (single)                                                                                           |
| $m_{main}$                                                                                    | 22.1–24.5<br>25.9 (single)                                                                                     |
| Dwell time $\delta t$ for initial testing<br>$\delta t \dots \text{step size} \dots \delta t$ | 0 (no pilot)<br>80...20...200 $\mu\text{s}$<br>300...100...500 $\mu\text{s}$<br>600...200...1200 $\mu\text{s}$ |
| Dwell time $\delta t$ for optical measurements                                                | 0, 90, 140, 300 $\mu\text{s}$                                                                                  |
| $IMEP_g$                                                                                      | $9.0 \pm 0.1$                                                                                                  |
| MFB50                                                                                         | $13 \pm 0.5$ CAD ATDC                                                                                          |

After the initial engine testing, the injector was tested in the HDA. Rates of injection were measured for the injection schedules developed during the engine testing, among others. The HDA chamber pressure was set to be consistent with the cylinder pressure at the time of injection. Fuel and chamber temperatures were held constant at  $90^\circ\text{C}$ , although measured rates of injection are not sensitive to fuel temperature at temperatures above  $60^\circ\text{C}$ .

The injector was then reinstalled in the engine and the fused silica piston top was installed. Intake air mass flow was replaced with pure nitrogen to prevent combustion and the associated window fouling. The  $CO_2$  flow rate was left unchanged. Four of the injection trains developed in fired operation were used for the optical measurements.

For each fired operating point (measured with the metal piston), combustion noise is calculated for each of 50 cycles of unfiltered cylinder pressure data according to the routine provided in [14]. This routine computes a fast Fourier transform of the cylinder pressure trace, filters the resulting spectrum to represent structural attenuation by the engine and the frequency response (A-weighted) of the human ear, and integrates the resulting filtered spectrum. This integrated value is normalized by the threshold of human hearing ( $20 \mu\text{Pa}$ ) to provide combustion noise in dB(A). All combustion noise values reported in this work are computed in this manner. The reader is referred to [14] for more details, but this routine is similar or identical to the routines used by commercial devices to compute combustion noise. The average of these 50 values is reported here for each case.

Apparent heat release rate (AHRR) is computed with an iterative, two-zone model, in which temperature and composition dependent gas properties are taken into account. Heat transfer and crevice flow effects are partially compensated by subtracting the AHRR of a motored pressure trace from the AHRR of the fired trace. The AHRR traces shown in this work are ensemble averaged over 50 fired cycles. A cosine-symmetric low pass FIR filter is used to smooth the fired cylinder pressure traces before the AHRR is calculated. Its pass band edge is located at 5 kHz. This preserves the frequency content of the cylinder pressure traces in the frequency bands most relevant to combustion noise. The cylinder pressure spectral data shown in this work have been filtered with a structural attenuation filter and a A-weighting filter (both are given in [14]), then divided by the threshold of human hearing, and expressed in dB(A). The resulting spectra are referred to as a sound pressure level. The amplitude of the sound pressure level is a measure of each frequency component's contribution to audible combustion noise.

## ZERO-DIMENSIONAL MODELING

A zero-dimensional computational model was constructed to aid in understanding the observed trends in combustion noise. This model makes it possible to decouple changes in the heat release rate from changes in the combustion phasing, which is typically not possible in the fired engine. A heat release rate profile (heat release rate vs. crank angle) is either artificially created or derived from measured data and given as an input to the model. Based on this heat release profile, the resulting cylinder pressure trace (including the gas exchange process) is computed. Combustion noise is computed from the simulated cylinder pressure trace using the routine described above. The computation of the cylinder pressure trace from an arbitrary heat release profile is described below.

The first law of thermodynamics for a control volume in differential form can be expressed as:

$$dU = dQ - dW + \dot{m}_i h_i - \dot{m}_e h_e \quad (1)$$

$dU$  is the differential of the total internal energy of the control volume

$dW$  is the differential boundary work

$\dot{m}_i$  and  $\dot{m}_e$  are the mass flow rates into and out of the control volume, respectively

$h_i$  and  $h_e$  are the specific enthalpies of the intake and exhaust flows, respectively

$dQ$  is given by:

$$dQ = dQ_{HR} - dQ_{wall} \quad (2)$$

$dQ_{HR}$  is the pre-defined heat release rate and the primary simulation input

$dQ_{wall}$  is the heat loss to the walls, and is modeled using Woschni's correlation [10].

Changes in the potential and kinetic energy of the control volume are neglected. The working fluid is assumed to be an ideal gas of unchanging composition:

$$PV = mRT \quad (3)$$

$P$  is the pressure in the control volume

$V$  is the volume of the control volume, taken to be the crank angle-dependent cylinder volume

$m$  is the mass in the control volume

$R$  is the specific gas constant for the working fluid

$T$  is the temperature of the working fluid

Continuity of mass also applies to the control volume:

$$dm = \dot{m}_i - \dot{m}_e \quad (4)$$

$dm$  is the mass differential in the control volume

Equations 2, 3, 4 are used to derive equations for the differential cylinder pressure and the differential temperature:

$$dP = \frac{(\gamma-1)dQ - \gamma PdV + PV \frac{d\gamma}{\gamma-1} + (\gamma-1)(\dot{m}_i h_i - \dot{m}_e h_e)}{V} \quad (5)$$

$$dT = \frac{PdV + VdP}{mR} - T \frac{dm}{m} \quad (6)$$

$dV$  is the differential volume of the control volume

$\gamma$  is the ratio of specific heats, which is assumed to be a function of temperature only [15]:

$$\gamma = 1.35 - 6 \cdot 10^{-5}T + 1 \cdot 10^{-8}T^2 \quad (7)$$

The gas exchange is simulated by computing the mass flow rates through the valves for the cases of choked and non-choked flow [10]:

$$\dot{m} = \frac{C_D A_R P_{hi}}{\sqrt{RT_0}} \sqrt{\gamma} \left( \frac{2}{\gamma+1} \right)^{(\gamma+1)/2(\gamma-1)} \text{ if } \frac{P_{low}}{P_{hi}} \leq \left( \frac{2}{\gamma-1} \right)^{\gamma/\gamma-1} \quad (8)$$

$$\dot{m} = \frac{C_D A_R P_{hi}}{\sqrt{RT_0}} \left( \frac{P_{low}}{P_{hi}} \right)^{1/\gamma} \sqrt{\frac{2\gamma}{\gamma-1} \left[ 1 - \left( \frac{P_{low}}{P_{hi}} \right)^{\gamma-1/\gamma} \right]} \quad (9)$$

$C_D$  is the experimentally measured discharge coefficient, which depends on the valve lift. Valve lift curves have also been measured experimentally and are used to calculate  $C_D$  for each crank angle.

$P_{hi}$  is the greater of the cylinder pressure or the measured static pressure in the exhaust/intake.

$P_{low}$  is the lesser of the cylinder pressure or the measured static pressure in the exhaust/intake.

$T_0$  is the temperature of the medium flowing through the valves in question, be it intake charge, cylinder contents, or hot exhaust gases

$A_R$  is taken to be twice the valve curtain area (since there are two of each kind of valve):

$$A_R = 2\pi D_v L_v \quad (10)$$

$D_v$  is a characteristic valve diameter

$L_v$  is the experimentally measured valve lift

Equations 4, 5, and 6 are evaluated and integrated using a fourth-order Runge-Kutta method and a step size of 0.25 CAD; the results are insensitive to changes in step size at this resolution. A motored pressure trace ( $dQ = 0$ ) is also integrated for use with the Woschni heat transfer correlation. Initial and boundary conditions are taken from test bench data. Wall temperature is estimated to be 150 °C based on the piston surface temperature measurements shown by Kashdan and Bruneaux for a skip-fired small-bore Diesel engine (scaled to account for the higher engine speed in this work) [16]. The resulting cylinder pressure trace represents one full engine cycle; it is

processed using the combustion noise calculation routine described above. In this way, a value of combustion noise can be computed for any arbitrary heat release profile.

For the initial simulations, heat release profiles are derived from the measured results as follows: the pilot and main heat release events are observed to occur with a significant temporal separation for a dwell of 1200  $\mu\text{s}$ . The sharp, narrow heat release profile that is characteristic of the pilot combustion is taken and the negative excursions and waviness surrounding it are removed. The main heat release profile is taken and simplified in a similar manner. Finally, these two basis heat release profiles are added to one another with a variable delay and phasing.

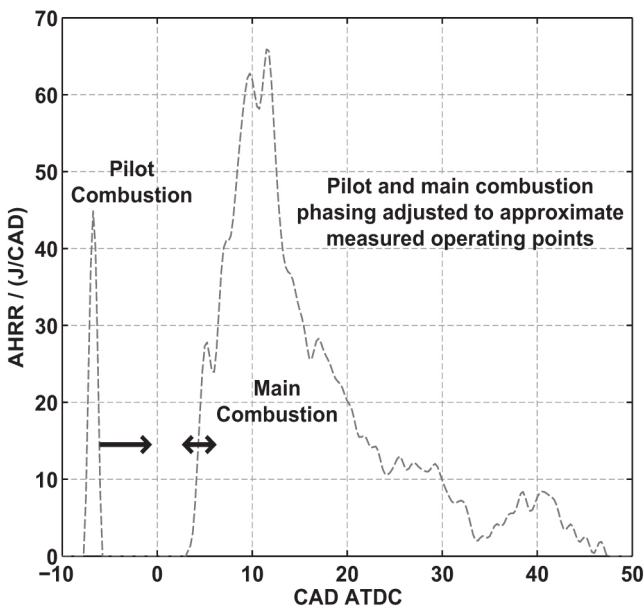


Figure 3. Independent pilot and main heat release events as inputs for initial simulations

The starts of the heat release events are adjusted to mimic the measured starts of combustion for each dwell, and the simulation is run for each dwell. Phasing of the main heat release event changes much less than the phasing of the pilot heat release. Further simulations are performed with different heat release profiles; these profiles will be described as the results are presented and discussed.

## RESULTS

### Measured Trends

As described above, initial investigations are performed with the metal piston top and with fired operation. Dwell is varied from 80  $\mu\text{s}$  to 1200  $\mu\text{s}$ , and a single injection serves as a reference. The pilot and main injections are block shifted to maintain the MFB50 given in Table 2 (with a tolerance of  $\pm 0.5$  CAD). Because the injector solenoid current is measured both in fired engine operation and with the HDA, the measured injection rates can be synchronized with the measurements taken in the engine (engine speed is assumed to remain constant throughout the portion of the cycle during which the injection occurs). Start of injection (SOI) is computed from the rate

of injection data as the last positively sloped zero crossing before an injection event. SOI is shown in Figure 4 for the pilot ( $\text{SOI}_{\text{pilot}}$ ) and main ( $\text{SOI}_{\text{main}}$ ) injections as a function of dwell.

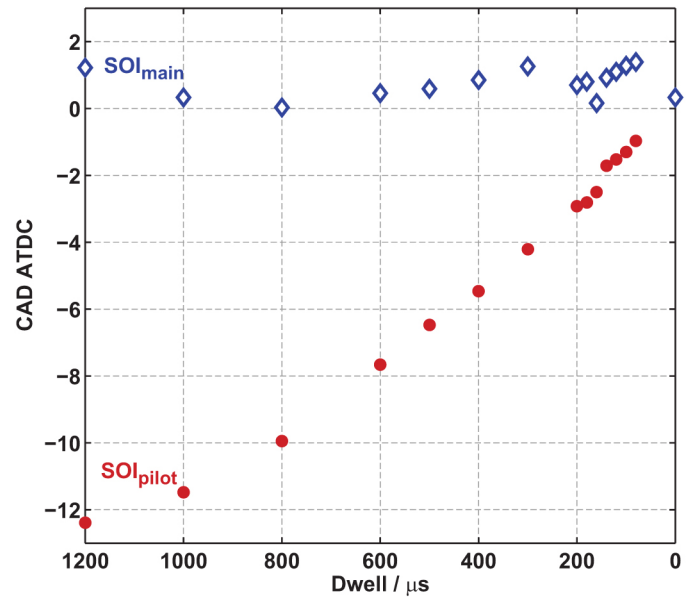


Figure 4. Start of injection (SOI) for pilot and main injections as a function of dwell

The trend in combustion noise is shown in Figure 5 as a function of dwell; the error bars represent two standard deviations (computed or 50 cycles) in each direction. Experiments have also been performed with a second, nominally identical injector, and the results (not shown) are closely repeatable. Standard deviations of combustion noise for a given dwell are typically less than 0.4 dB(A). Although the highest standard deviations typically occur for dwells of less than 200  $\mu\text{s}$ , they are often comparable to the standard deviation for a single injection. The single injection case is shown in this work as a dwell of zero.

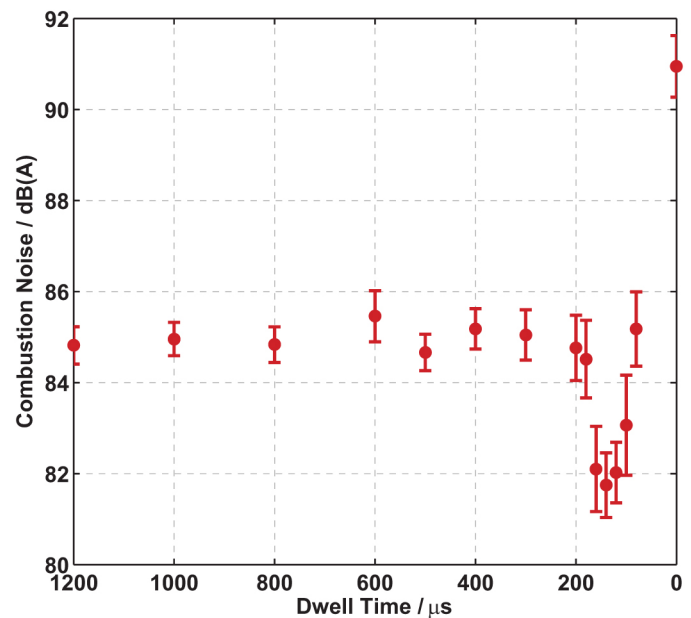


Figure 5. Combustion noise vs. dwell for the pilot-main injection strategy with 2- $\sigma$  error bars (single injection shown as zero dwell)