

Multiple-Event Fuel Injection Investigations in a Highly-Dilute Diesel Low Temperature Combustion Regime

**Chad P. Koci, Youngchul Ra, Roger Krieger,
Mike Andrie and David E. Foster**

Engine Research Center, University of Wisconsin – Madison

Robert M. Siewert and Russell P. Durrett

Powertrain Systems Research Laboratory, General Motors Research and Development Center

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ABSTRACT

The objective of this research is a detailed investigation of multiple injections in a highly-dilute diesel low temperature combustion (LTC) regime. This research concentrates on understanding the performance and emissions benefits of multiple injections via experiments and simulations in a 0.48L signal cylinder light-duty engine operating at 2000 r/min and 5.5 bar IMEP. Controlled experiments in the single-cylinder engine are then combined with three computational tools, namely heat release analysis of measured cylinder pressure, a phenomenological spray model using in-cylinder thermodynamics [1], and KIVA-3V Chemkin CFD computations recently tested at LTC conditions [2]. This study examines the effects of fuel split distribution, injection event timing, rail pressure, and boost pressure which are each explored within a defined operation range in LTC. This research compliments simultaneous detailed unburned hydrocarbon research which concentrates on the mechanisms that control the formation of UHC during LTC engine operation [3].

Engine operating conditions for low UHC/CO from previous single injection experiments are observed using split (i.e., multiple-event) injections. Fuel split is defined by the fractional percentage of total fuel injected in the first event. For the split fuel injection experiments it is shown that both the timing and proportion of the fuel

split largely impacts emissions and combustion; this is due to the change in injection penetration and momentum flux influencing the fuel distribution in the cylinder. Phenomenological engine spray model and CFD results show how fuel distribution and reactant mixing also strongly influence PM emissions. Non-negligible PM emissions occur when the split-injection fuel distribution produces locally richer regions very close to the start of combustion (SOC). Boost and injection pressure variation also impact the split-injection fuel distribution significantly. There are emissions trade-offs over the entire split-injection timing range, but lower combustion noise trade-offs are only achieved with later (closer to TDC) fuel injection timings.

INTRODUCTION

There is a significant global effort to study low temperature combustion (LTC) as a tool to achieve stringent emission standards with future light duty diesel engines. Highly-dilute LTC utilizes high levels of dilution (i.e., EGR > 60% with ~10%O₂ in the intake charge) to reduce overall combustion temperatures and to lengthen ignition delay. This increased ignition delay provides time for fuel evaporation and reduces in-homogeneities in the reactant mixture, thus reducing NO_x formation from local temperature spikes and soot formation from locally rich mixtures. However, as dilution is increased to the limits, UHC and CO can significantly increase.

Recent research suggests that emissions, namely CO and UHC, during LTC result from the incomplete combustion of under-mixed or over-mixed fuel and charge gas occurring after the premixed burn period [4-7]¹. Much of this diesel LTC research has been constrained to single-event fuel injections, allowing for the initial engine fundamental phenomena to be investigated clearly. Following similar trends with production diesel engines, the transition toward diesel LTC research with more complex fuel injection strategies is a logical path. The objective of the present work is to increase understanding of the emission mechanisms and engine performance with multiple injections in LTC at part load.

The highly-dilute Low Temperature Combustion (LTC) regime (similar to the work in [8]) requires very high EGR (>60%) in order to sufficiently prolong the ignition delay of the mixture and lower the mixture temperatures. This combustion is typically realized at global mixture compositions just lean of stoichiometric, as opposed to much leaner mixtures in conventional diesel combustion. Some of the various names which have been used to describe techniques involving LTC are Premixed Compression Ignition (PCI) [9], Premixed Charge Compression Ignition (PCCI) [10], Homogeneous Charge Compression Ignition (HCCI) [11], and MK [12]. However, HCCI would typically refer to a strategy where the fuel is injected extremely early, resulting in a nearly homogeneous mixture of fuel and air in the combustion chamber. PCI or PCCI, on the other hand, is typically achieved with less advanced injection timing (but more advanced than for conventional diesel) where the fuel is injected in the combustion bowl with the objective of providing a not only fully vaporized fuel cloud but also sufficiently mixed fuel and air within the combustion bowl to avoid significant soot formation. The MK strategy [12] uses relatively retarded injection timings to increase ignition delay and allow for pre-mixing. The strategies cited above can also be combined to enable broader coverage of the engine load and speed range [13].

In the Equivalence Ratio vs. Temperature space shown in Figure 1, it is evident how the requirement to achieve lower equivalence ratios using increased mixing and low temperatures simultaneously can reduce NO_x and Soot.

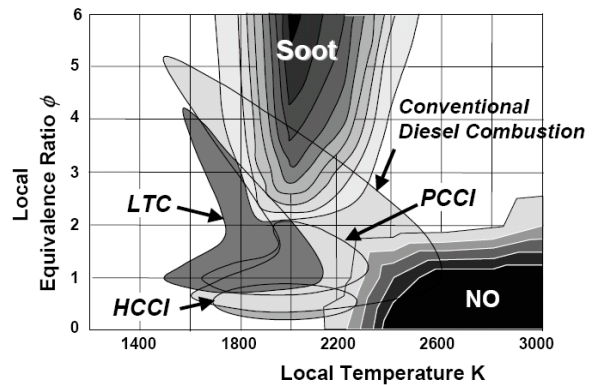


Figure 1: Modern diesel combustion strategies plotted in ϕ -T space [13].

Conventional diesel combustion travels through the Soot and NO_x islands while LTC, PCCI and HCCI all narrowly avoid the high formation regions for these emissions. The shape and location of these islands is a function of the in-cylinder conditions particular to the operating point, but the general behavior remains similar over a range of conditions.

Because of the overlap of these various advanced combustion techniques, for simplicity, the combustion strategies studied in this paper will be referred to simply as LTC. The use of this form of combustion is currently restricted to light loads based on the low cylinder temperature and pressures that are necessary to maintain the decreased NO_x. It is of great interest to push the current limits of LTC by going to higher loads and lower inlet oxygen levels. At these very low inlet O₂ levels, increased amounts of CO and UHC begin to appear along with increased combustion noise. Abundant current research provides specific injection strategy performance and fundamental combustion process analysis [14-17]. By using the current and past research as a guide, much of this paper will focus on how multiple fuel injection events interact and affect emissions mechanisms and performance parameters within LTC. Ideally, a simpler more global view of early (prior to TDC or very near as opposed to significantly after TDC) fuel injection strategies can be formulated to cut through some of the condition specific dependence. If this global view can be achieved, then the transition and application from one design or operation condition to another can be more easily understood.

EXPERIMENT

ENGINE SETUP

The engine used to conduct this research is a single-cylinder version of a GM 1.9L, 4 cylinder light-duty diesel engine; the specifications of which can be seen in Table 1. The production head and valve-train were used in conjunction with a Labeco CLR crankcase. A re-entrant bowl design with lower compression ratio than the current Euro-4 compliant production configuration

¹ Numbers in brackets designate references listed at end of paper

was used along with a slightly longer connecting rod length; all other cylinder geometry was matched to the equivalent 4-cylinder engine. The single-cylinder engine has the capability to run up to 4000 RPM at full-power conditions using pressurized inlet air to simulate the behavior of a turbocharger. Full capability for variation of EGR rate, coolant and oil temperature and respective inlet and exhaust pressures and temperatures exists in the laboratory. Figure 2 shows the schematic of the engine and test cell layout.

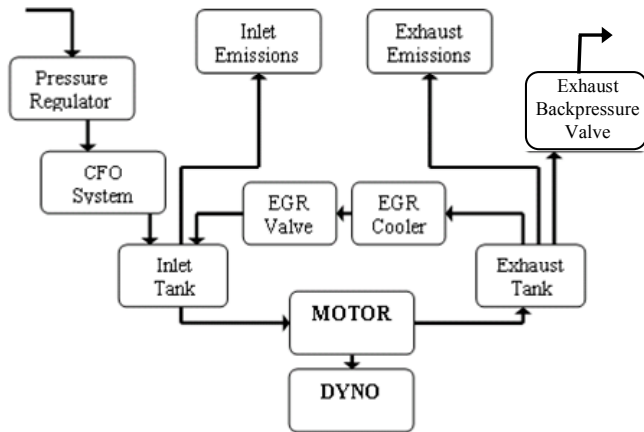


Figure 2: Schematic of test cell layout. CFO stands for Critical Flow Orifice.

Table 1: Engine Specifications

Displacement	0.4774 [L]
Bore	82.0 [mm]
Stroke	90.4 [mm]
CR	16.6:1
IVO	344° ATDC
IVC	-132° ATDC
EVO	112° ATDC
EVC	388° ATDC

The injection system utilizes a BOSCH CP3.3 pump for producing fuel rail pressure. The ETAS INCA calibration software used in conjunction with the production ECU allows for up to 5 injections per cycle. A production CRIP2-MI injector was used with an experimental 7 x 155° nozzle tip with a flow number of 440 cc/30sec. Table 2 summarizes the injection system specifications.

Table 2: Injection System Specifications

Fuel Injection Equipment		
Design	Common Rail	-
Injector	Bosch CRIP 2	-
Included Cone Angle	155	Deg
Nozzle Type	Mini-Sac	-
Nozzle Flow Number	440	cc/30sec
Nozzle Hole Diameter	0.139	mm
Number of Holes	7	-
Operated Rail Pressures	600, 860, 1120	Bar
Peak Rail Pressure	1600	Bar
Fuel Pump	Bosch CP3.3	-

Characterization of multiple-injection rate shapes was needed for modeling efforts and injection-system understanding. A KISTLER 4067A piezoresistive fuel pressure transducer allowed for real-time pressure history, and was located on the hard line from the rail to the injector. Pressure-history and rate-shape characterization, using an injection-rate meter, provided the required multiple-event injection-system data.

The measurement of cylinder pressure data was accomplished via a piezoelectric KISTLER 6125B pressure transducer. The resolution of data was 0.1 CAD, corresponding to 3600 counts per revolution; 200 cycles of data were recorded for all heat release analysis. The combustion noise was recorded using an AVL Combustion Noise Meter (Model A01); meter output was in decibels.

The fuel used in the experiments was intended to be a typical US Ultra Low Sulfur Diesel (US-ULSD) with a Cetane number of 42.9. The general properties of this fuel can be seen in Table 3.

Table 3: General fuel properties for testing

Fuel Properties for the Typical US-ULSD		
Spec Gravity	0.847	-
API Gravity	35.64	API
VISCOSITY @ 40 °C	2.025	cSt
HHV	19571	Btu/lb
HHV	45.52	MJ/kg
NON-AROMATICS	66	% wt
MONO-AROMATICS	26.2	% wt
PAH'S	7.8	% wt
Cetane number	42.9	D613
Temp at 50% Distilled	237.8	°C

Exhaust emissions were measured using a Horiba emissions bench capable of recording CO₂, CO, (Model AIA-220), O₂ (Model FMA-220), THC (Model FIA-236-1, propane span gas) and NO_x (Model CLA-220). Throughout the experiments NO_x emissions remained extremely low near 3-6 ppm for all cases. In addition, a reflectivity-based PM measurement was taken using an AVL Smoke Meter (Model 415S) at each operational point. The emissions sample was extracted downstream of both the exhaust surge tank and the exhaust backpressure valve. In order to obtain the real-time mass percentage of EGR, measurements on the same Horiba emissions bench were done for inlet CO₂ (Model AIA-210) and inlet O₂ (Model MPA-21A). A MKS FTIR (Model 2030HS) was also used for supplemental emissions measurements and hydrocarbon investigations which were part of the parallel research [3]. All emissions equipment used heated lines and filters except the EGR CO₂ and O₂.

TEST CONDITIONS

It is known that there are major NO_x and PM benefits from LTC utilizing a single fuel injection event. Spray-targeting analyses of [6] showed how an optimal split of fuel in the squish and the bowl regions provided lower CO and UHC emissions. The idea of preparing the most uniform charge by spray targeting naturally progressed to multiple injections. If there is an optimal single SOI timing, then there should be benefits of using multiple injections to better prepare the charge. Figure 3 from Opat [6] shows the spray-targeting split of a single injection which produces the most uniform charge. Similarly, by placing the 1st fuel injection in the squish region, then a 2nd injection in the bowl region, perhaps better charge preparation and mixing could be realized.

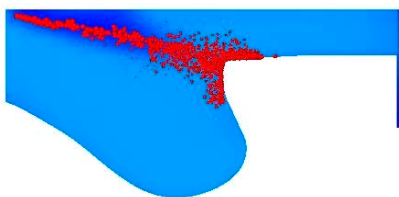


Figure 3: Optimal single fuel injection timing for minimum emissions in LTC from Opat et al. [6].

A limitation of using a wide angle injector is cylinder wall impingement and damage as observed with the work of Dronniou [14]. The avoidance of this was necessary along with the use of a wide-angle injector to be consistent with the overall program goals. This limitation resulted in setting the earliest considered compression stroke injection timings to be near -50° ATDC. Within the general range of -50° to 0° ATDC two injection events were analyzed for a range of fuel-mass splits and timings (both 1st and 2nd injection timings). A convenient

and descriptive way of presenting data is one where the 1st and 2nd fuel injection timings are specified along with the percent of total fuel in the 1st injection. The reader should then be able to visualize where and when the fuel is introduced into the combustion chamber. If a name were to be applied to these types of injections it would be “High-EGR Early Split Injections”, which provides a worthwhile description of the injections in LTC. Other multiple-injection strategies were explored, but the majority of the research was focused on the High-EGR Early Split Injections. Inlet manifold pressure and fuel injection pressure effects were also investigated using this strategy.

Table 4: Baseline Engine Operation Conditions

Engine Speed	2000 rpm
Engine Load	5.5 Bar Net IMEP*
Nozzle Angle	155°
Fuel Rail Pressure	860 Bar
SOIC_1st	-33° ATDC
SOIC_2nd	-17° ATDC
Fuel Mass%_1st Injection	100%-0%
Swirl Ratio	2.2
Inlet Pressure	162 kPa abs
Exhaust Pressure	182 kPa abs
Inlet O2	9.50%
EGR	~67%
Intake Temperature	65° C
EGR Temperature	45° C
Coolant Temperature	90° C
Oil Temperature	90° C

* Net IMEP is defined as gross IMEP minus pumping MEP calculated from pressure-history data.

The baseline split-injection engine operational conditions can be seen in Table 4. The condition is very dilute utilizing a maximum of ~67% EGR based on the controlling parameter of the inlet oxygen concentration of 9.50% +/- .05%. The extremely dilute condition is required to observe mixing and fuel spray-targeting impacts on CO and UHC as documented by [5, 6]. For the baseline conditions the CO minimum, or “sweet-spot”, condition was at a SOIC (start of injection command) timing of -33° ATDC for the 1st injection. The reader should know that at 2000 rpm the listed SOIC timings are 4.8° advanced from the actual SOI timings.

Since the injection strategy sweep starts from 100% of the fuel in the 1st injection, i.e., a single injection, it was decided to “anchor” the sweep with injection timings which were optimum for the single injection case. The optimum single injection timing was -33° ATDC, which was denoted as the “sweet-spot” timing in earlier work on the test engine [6]. Preliminary investigations of 2nd injection SOI sweeps showed interesting and repeatable results in the range of -20° to 15°.

fuel-split injection timings were chosen to be -33° and -17° . A total of eight fuel-split sweeps were performed in addition to the baseline fuel-split sweep. At each experimental condition the percentage of fuel in the 1st injection was varied from 100% to 0% of the total fueling required to maintain the constant load condition. Increments were typically 100, 90, 85, 70, 50, 30, 15, and 0%. Implicit in the fuel-split sweeps are single fuel injections at 100% and 0% with timings of the 1st and 2nd injections respectively. Typical injector current commands and dwell spacing are shown for a similar injection strategy later in Figure 25. Fuel injection pressures were varied from the baseline 860 bar to 600 and 1120 bar along with inlet boost pressures varied from the baseline 162 kPaa to 142 and 182 kPaa. The event timing fuel-split sweeps varied the 1st and 2nd injection timings, maintaining a constant dwell of 16° , such that the entire injection strategy was systematically advanced and retarded from the baseline. A summary of the conditions at each point can be seen in Table 5.

Table 5: Test case matrix. All timings are in $^\circ$ ATDC.

	Base Split	Inlet Boost	Rail Press	Inj Event Timing
SOIC_1 st	-33°	-33°	-33°	$-45^\circ, -40^\circ$ $-25^\circ, -20^\circ$
SOIC_2 nd	-17°	-17°	-17°	$-29^\circ, -24^\circ$ $-9^\circ, -4^\circ$
Injection Dwell	16°	16°	16°	16°
Rail Pressure (bar)	860	860	600 1120	860
Inlet Boost (kPaa)	162	142 182	162	162

Injection Dwell is defined as the interval between the start of the 1st injection pulse and the start of the 2nd pulse. Fuel splits were ranged from 100% to 0% total fuel in the 1st injection unless otherwise specified.

NUMERICAL APPROACH

COMPUTATIONAL GRID

Three dimensional computational grids with crevice volume resolved as an elongated top land region were generated, as shown in Figure 4. To save computation load, a $1/7^{\text{th}}$ sector of the full 360 degree mesh with periodic boundaries (corresponding to the seven-hole injector nozzle) was used. The average cell dimensions were 1.2~1.8 mm and 0.6~4.1 mm in radial and vertical directions, respectively. Twenty cells were generated azimuthally. To resolve the crevice region, i.e., the gap between the piston and cylinder wall above the top ring, two radial cells were used along with three vertical cells. In the computational grid, the bowl region is defined as the area that has radial distances shorter than the bowl lip location including the re-entrant region, while the rest of the region is defined as the squish region.

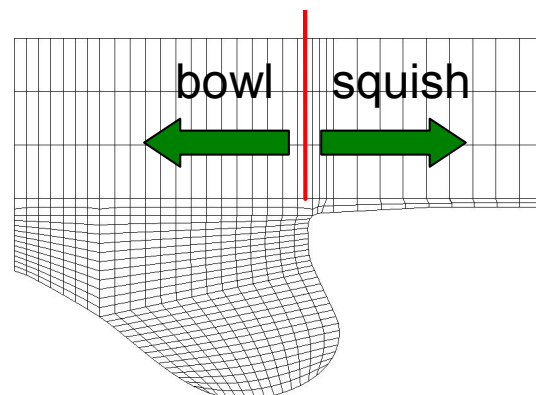


Figure 4: Vertical cross-section view of computational grid with crevice volume resolved as a thin annulus. Azimuthal angle sector span is 51.4 degrees.

SPRAY TREATMENT

Computations were performed using the KIVA3V Release 2 code [18] with an improved droplet vaporization model [19], which considers the droplet temperature range from flash-boiling conditions to normal evaporation. The improved model accounts for variable internal droplet temperature, and considers an unsteady internal heat flux with internal circulation, and a new model for the determination of the droplet surface temperature. The model uses an effective heat transfer coefficient model for the heat flux from the surrounding gas to the droplet surface. Also, the variable density of diesel fuel as a function of temperature is considered in the governing equations and the relevant sub-models.

A hybrid primary break-up model [20] that is computationally effective as well as comprehensive enough to account for the effects of aerodynamics, liquid properties and nozzle flows was employed. In this model, the injected fuel “parcels” are tracked by a Lagrangian method while the break-up of each blob is calculated from considerations of jet stability. For the secondary and further break-up processes, a Kelvin Helmholtz (KH) – Rayleigh Taylor (RT) hybrid model [20] was used.

A droplet collision model based on the stochastic particle method [18] was used, in which the collision frequency is used to calculate the probability that a drop in one parcel will undergo a collision with a drop in another parcel, assuming all drops in each parcel behave in the same manner.

The effects associated with spray/wall interactions including droplet splash, film spreading due to impingement forces, and motion due to film inertia were considered in a wall film submodel [21], in addition to calculations of film transport on complex surfaces, with heating and vaporization of the film, and separation and re-entrainment of films at sharp corners.

The RNG k- model was used for the turbulence [22] and for turbulence parameters needed in the spray model.

In the 2-phase transport equations, droplets are treated as point sources and the wall film fuel flow is not resolved on the computational grid. Therefore, it is assumed that the vaporized fuel in a computational cell where droplets or wall film parcels exist mixes completely with the gaseous mixture within the cell. Thus, stratification of gaseous species within a single cell is not resolved.

COMBUSTION TREATMENT

Detailed chemistry was employed by coupling the KIVA3V (Release 2) with the Chemkin II library [23]. A skeletal reaction mechanism for PRF fuel with 41 species and 130 reactions, which was developed by Ra and Reitz [24], was used to calculate the detailed chemical kinetics of fuel/air mixture burning.

Various mechanisms for n-heptane (or PRF0) oxidation have been widely employed for diesel combustion calculations by many researchers because of its similar ignition characteristics to those of diesel fuel. While the n-heptane mechanism [2] is used for reaction calculations in each cell, the physical properties of the fuel were represented with those of tetra-decane ($C_{14}H_{30}$) in the calculations of spray processes and in gaseous species mixing. For the calculation of NOx formation, a 4 species (N, NO, N₂O and NO₂) and 12 reaction NOx mechanism was used that has been reduced from the GRI NOx mechanism [25] and added to the PRF reaction mechanism.

A phenomenological soot model [26] modified from the Hiroyasu soot model [27] was employed to predict soot emissions. In the modified model, acetylene (C_2H_2) is used as an inception species for soot formation, which not only enables the soot model to be coupled with the detailed chemistry calculation, but also improves the soot emission predictions [26]. The soot oxidation rate was determined by the Nagle-Strickland Constable model [28].

Recent work, as previously referenced, with this code in a number of diesel engines operating at LTC conditions indicated that it performs well predicting not only LTC ignition timing but also combustion evolution. For this reason, it was decided to apply this code to help explain the experimental results presented herein.

COMPUTATIONAL CONDITIONS

Test cases listed in Table 5 were considered for the computations except for the cases with injection timing variation. The operating conditions shown in Table 4 were used with injection split proportion as a parametric variable.

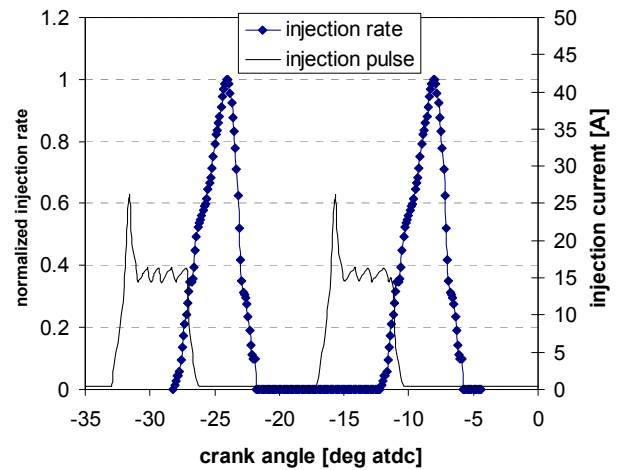


Figure 5: Measured injection rate shape of a double pulse injection. Injection pressure was 860 bar. First pulse split ratio was 50% and the duration of each pulse was 490 μ s. Rates are normalized by the maximum value during the injection.

In Figure 5, the injection rate shape profile of a double-pulse injection is shown for the baseline injection pressure of 860 bar and a split proportion of 50 % in the first injection. The start of injection command of the pulses was -33° and -17° ATDC, for the first and second pulses, respectively. The duration of each pulse was 490 μ s.

RESULTS

BASELINE FUEL SPLIT SWEEP EFFECTS

By holding constant load and sweeping the proportion of fuel between the 1st and the 2nd injections, very smooth changes in the combustion could be realized. Looking at the heat release rates in Figure 6 this smooth transition can be observed. Actual SOI timings will be listed where applicable for clarity, as in Figure 6. Initially all of the fuel is placed early in the cycle. As the fuel is slowly transferred later in the cycle, the HTHR retards and the LTHR from the 1st injection is reduced. Since the injection timings are fixed, the changes in heat release are due to a redistribution of fuel. From previous experiments at these conditions it has been determined that once the timing of the 2nd injection is at or after the midpoint of the LTHR of the 1st injection, increases in the magnitude of the 2nd injection with a fixed 1st injection magnitude will not impact the 1st injections HTHR phasing. In other words, once the 2nd injection is late enough it has little impact on the heat release of the 1st injection. The baseline split 2nd injection location is at the midpoint of the 1st injection's LTHR and there are not believed to be any major HTHR phasing effects due to the 2nd injection. Throughout this baseline fuel sweep a diffusion type part of the HTHR is never observed, indicating that much of the fuel is premixed. Even at the 0% 1st injection fuel split a LTHR is still observed before the HTHR.

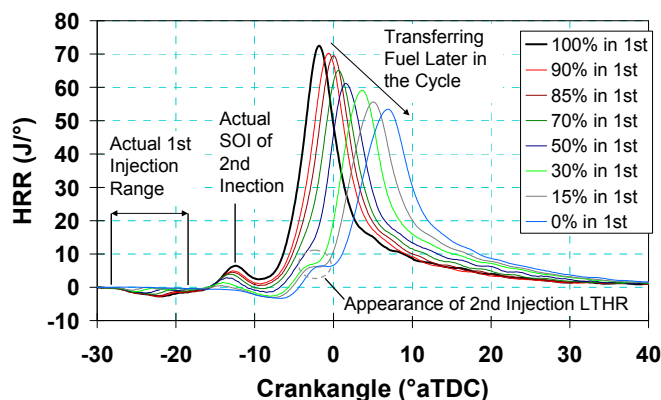


Figure 6: Heat release rates for the baseline fuel split with 1st and 2nd injection SOICs of -33° and -17° ATDC respectively.

As expected from the shapes of the heat release curves, the combustion noise drops off as the fuel is transferred later into the cycle. The location of 50% burn is inversely related to the combustion noise and is moving steadily later as the fuel is placed later in cycle, Figure 7.

The x-axis in Figure 7 is the percent of total fuel in the 1st injection at -33° ATDC. This manner of plotting is useful for combustion performance parameters and emissions because of the incorporation of fuel injection timing and distribution information. In the baseline case peak pressures are also very heavily impacted by the fuel-split variation, and range from near 110 bar at 100% fuel in the 1st injection to 90 bar at 0% fuel in the 1st injection. The noise analysis shows that in order for a certain noise target to be reached, 89 dBa for example, the combustion phasing must be such that the 50% burn location is at least six degrees ATDC. This correlates to at least 70% of the fuel in the 2nd injection, which is not optimal from an emissions standpoint, as outlined next. Injecting as much of the fuel earlier in the cycle allows for increased mixing time and better fuel distribution throughout the combustion chamber.

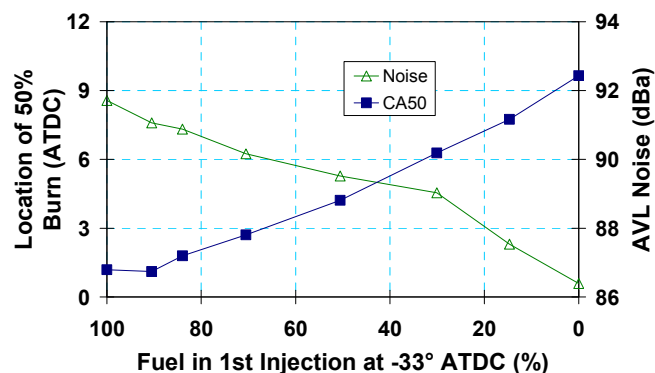


Figure 7: Combustion noise and locations of 50% burn for the baseline fuel-split sweep. SOIC_1st = -33° ATDC and SOIC_2nd = -17° ATDC

Emission for the fuel sweep with fixed 1st and 2nd injection timings are shown in Figure 8 and include single injection “sweet-spot” values (-33° ATDC, 100% fuel). CO emissions are reduced from the “sweet-spot” single injection level when a small percentage of fuel, 15-20%, is injected in the 2nd injection. The addition of the 2nd injection is impacting the CO mixing and oxidation processes; further computational analysis will help to investigate this more thoroughly. As more fuel is swept into the later injection, CO increases to more than three times the minimum multiple-injection value. The calculated combustion efficiency is almost a mirror image of the CO trend. It is well known that the oxidation of CO is the most important energy release step in the hydrocarbon kinetics, and for efficiency sake the CO emissions need to be minimized. Combustion efficiency values are low even at 97% compared to modern DI diesels which typically see 99% or higher.

Despite the dilute nature of the combustion, PM emissions are quite high as the fuel-mass split is moved from 100% to 50%. A peak occurs at 50% fuel split which was rather unexpected. Global equivalence ratios at this condition are near 0.90 and if uniform would be out of the range of the PM peninsula on a typical -T plot, Figure 1. The only way to move into a sooting condition would be to have higher local equivalence ratios. The 1st injection is targeted near the edge of the piston bowl while the 2nd injection is targeted within the piston bowl. If the 2nd injection is injecting into partially vaporized fuel rich regions from the 1st injection, then even richer regions could form which would not have time to mix thoroughly before the onset of combustion. The UHC emissions are an order of magnitude higher than the PM emissions for these conditions. If there were any UHC/PM trade offs occurring, detection of the changes in the measured UHC emissions would be sufficiently small to go unnoticed. Spray targeting and fuel mixing effects are believed to be a dominant player in the UHC trends.