

Full-Time Gasoline Direct-Injection Compression Ignition (GDCI) for High Efficiency and Low NO_x and PM

Mark C. Sellnau, James Sinnamon, Kevin Hoyer and Harry Husted
Delphi Corp.

ABSTRACT

A gasoline compression-ignition combustion system is being developed for full-time operation over the speed-load map. Low-temperature combustion was achieved using multiple late injection (MLI), intake boost, and moderate EGR for high efficiency, low NO_x, and low particulate emissions. The relatively long ignition delay and high volatility of RON 91 pump gasoline combined with an advanced injection system and variable valve actuation provided controlled mixture stratification for low combustion noise.

Tests were conducted on a single-cylinder research engine. Design of Experiments and response surface models were used to evaluate injection strategies, injector designs, and various valve lift profiles across the speed-load operating range. At light loads, an exhaust rebreathing strategy was used to promote autoignition and maintain exhaust temperatures. At medium loads, a triple injection strategy produced the best results with high thermal efficiency. Detailed heat release analysis indicated that heat losses were significantly reduced. At higher loads, a late-intake-valve-closing strategy was used to reduce the effective compression ratio. For all tests, intake air temperature was 50 °C.

3D CFD simulations of fuel injection, mixing, and combustion were important to understand the emissions formation processes. With multiple late injections and low-to-moderate fuel pressure, spray penetration was low, mixing was fast, and wall wetting could be avoided. Fuel sprays were characterized in a spray chamber. Injection rate was measured using a rate tube.

Results showed that ISFC was very low. Minimum ISFC of 181 g/kWh was measured at 2000 rpm-11 bar IMEP. For IMEP from 2 to 18 bar, engine-out NO_x and PM emissions were below targets of 0.2 g/kWh and 0.1 FSN, respectively, indicating that aftertreatment for these species may be reduced or eliminated. It was found that combustion noise levels, characterized by several noise metrics, could be effectively controlled by the injection process. Measurements of exhaust particulate size distribution indicated very low particle count, especially for a preferred injector with low levels of in-cylinder swirl.

Collectively, these results demonstrate the potential feasibility of full-time GDCI using RON 91 gasoline at low-to-moderate injection pressures with high fuel efficiency. While more development work is needed, there is good potential for a practical GDCI powertrain system based on these concepts.

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INTRODUCTION

Low temperature combustion (LTC) is known to be a promising approach for simultaneous reduction of NO_x and particulate matter (PM) emissions with high fuel efficiency. Extensive research has been conducted on various LTC combustion concepts for both light duty (LD) and heavy duty (HD) engines operating on diesel fuels, gasoline fuels, and alcohols [1,2]. LTC requires that fuel and air be sufficiently mixed prior to combustion to avoid soot formation in excessively rich pockets. Combustion temperatures must be low enough to avoid NO_x emissions [3]. Dilution of the

mixture with burned gas is one way to reduce combustion temperatures but hydrocarbon (HC) and carbon monoxide (CO) emissions typically increase. Combustion noise, load limits, and combustion phasing control are other problems that have impeded the application of LTC in production engines. However, if LTC could be achieved over the engine operating range and if combustion noise can be controlled, diesel-like fuel efficiency and compliance with stringent emissions regulations might be possible without aftertreatment for NO_x and particulate emissions.

Gasoline partially premixed compression ignition (PPCI) is a relatively new approach to LTC that was first published by Johansson circa 2005 [4]. A high octane fuel (cetane number of 21) was injected late on the compression stroke of a boosted diesel engine operating with 52% EGR. The injection process was complete prior to the start of combustion, which enabled mixing of the fuel and air prior to heat release. At a high load condition (IMEP=15.6 bar), low NOx and PM emissions were measured.

Kalghatgi [5] showed that the best fuels for PPCI are gasoline-like fuels with high resistance to autoignition. These fuels have greater ignition delay relative to high cetane fuels, which gives them more time for mixing. The high volatility and diffusion rates of gasoline also aid in the mixing process. Risberg and Kalghatgi [6] tested a range of fuels with different autoignition quality and volatility on a production diesel engine. They showed that autoignition quality correlated well with cetane number of the fuels.

Kalghatgi [7] performed critical engine tests for PPCI using single injection for gasoline and diesel fuels. It was shown that gasoline fuels produced a much longer ignition delay than diesel fuel, which facilitated premixed combustion and produced much lower NOx and PM. The fuel was said to be “premixed enough but not fully mixed.” Additional experiments were performed using double injection with gasoline [8,9]. It was shown that pilot injection helps reduce maximum heat release rate. The load range was increased to 16 bar IMEP with low NOx and PM emissions and with fuel consumption comparable to conventional diesel. This established that gasoline fuels are well suited for PPCI low-temperature combustion processes.

Researchers at Lund University [10,11,12,13] extensively tested gasoline fuels in diesel engines. They demonstrated high-load capability in both LD and HD engines. An injection strategy was developed similar to UNIBUS that utilizes two injections, high boost, and about 50% EGR. Using high fuel pressure with diesel injectors, they were able to achieve PPCI at 18 bar IMEP with soot levels of 1-2 FSN, NOx below 0.3 g/kW-hr, and maximum pressure rise rates below 15 bar/deg. While not all requirements, such as noise levels, were satisfied and results below 5 bar IMEP were not reported, results were very promising.

Experiments were performed at Delphi [14] on a single-cylinder Hydra engine at part-load operating conditions. Designed experiments (DoE) and response surface models were used to find optimum injection timings and quantities for single, double and triple injection strategies. It was shown that triple injection provided the best results at the lowest injection pressure. Due to favourable mixture stratification, maximum heat release rate (MHRR) and heat losses were both reduced significantly for triple injection compared to single or double injection. This enabled more advanced combustion phasing (within noise constraints) and significantly improved fuel efficiency.

Researchers at the University of Wisconsin have performed detailed CFD studies of PPCI combustion in a HD

diesel engine using KIVA-ERC-Chemkin [15,16]. Both single and double injection was studied with both gasoline and diesel fuels. They showed that injection timing influences HC, CO, and combustion noise levels, and that injection split ratio is an effective way to reduce CO and maximum pressure rise rates. Dempsey [17] extended this work using genetic algorithm optimization. For a medium load condition (11 bar IMEP), it was found that a double injection strategy produced the lowest NOx, PM, ISFC and ringing intensity (RI). NOx and PM emissions were below emissions regulations while maintaining low ringing intensity. For a high load condition (21 bar IMEP), a double injection strategy using 60% of the fuel in the 2nd injection gave greater than 50% thermal efficiency, but PM emissions and RI exceeded targets.

KIVA simulations of gasoline PPCI were also performed for a LD diesel engine [18]. The purpose of this work was to establish feasibility of gasoline PPCI at a high load condition (2500 rpm-14 bar IMEP) and to guide subsequent engine experiments. Tests were conducted using a double injection strategy with diesel injectors at 860 to 1160 bar fuel pressure. It was found that both NOx and PM could be reduced to levels of about 0.1 g/kg-f, while achieving ISFC of about 180 g/kWh. Maximum pressure rise rate and peak pressure were below 10 bar/CAD and 160 bar, respectively. There was an optimal injection pressure that produced the best mixture stratification, while both over mixing and under mixing resulted in incomplete combustion.

In the current work, tests were performed on a LD single-cylinder engine over the speed-load operating map using RON91 pump gasoline. One objective was to determine the best injection strategies for low NOx and PM using low-to-moderate injection pressures. Injector design and fuel spray characteristics play a strong role in controlling mixture stratification. Five injector designs were fabricated and tested.

A second objective was to evaluate an engine concept for full-time operation over the speed-load map from idle to full load. If NOx and PM emissions are low enough, aftertreatment for these species may be reduced or eliminated. Use of variable valve lift profiles was instrumental in enabling full-time GDCI operation. Finally, combustion noise is an important constraint for LTC combustion strategies. Various combustion noise parameters were evaluated and correlations among them were explored.

EXPERIMENTAL METHODS

Experiments were performed on a Ricardo Hydra light-duty single-cylinder engine (Figure 1). The cylinder head has four-valves with double-overhead camshafts and central injection. The aluminum cylinder head is rated at 200 bar peak cylinder pressure (PCP). The engine is very flexible and camshafts and pistons can be interchanged. A removable injector sleeve enables testing of various injector types. An intake port throttle regulates the swirl ratio between 0.6 and 3.0. Engine specifications are listed in Table 1.

Two test fuels were used for experiments. Initial tests were conducted with regular unleaded RON91 gasoline that contained no oxygenates. Later tests were conducted using Shell regular unleaded gasoline with ethanol, as is representative of commercial fuel in the United States. Fuel property data are shown in [Table 2](#).

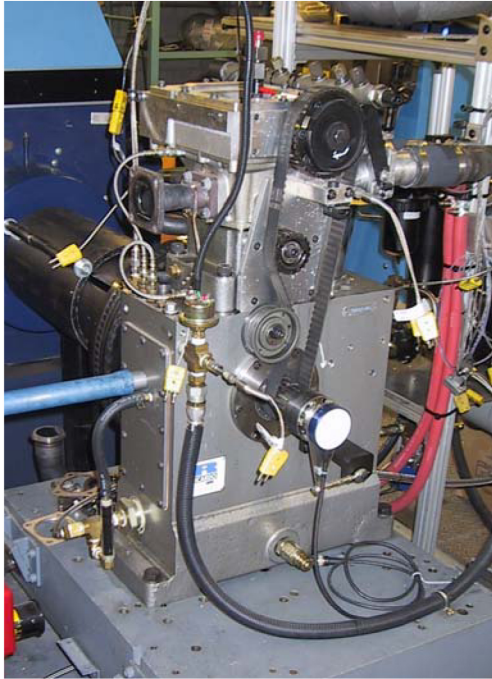


Figure 1. Hydra Test Engine

Table 1. Engine Specifications

Engine Type	4-valve Comp. Ignition
Bore × Stroke	84 × 90 mm
Displacement	499 cc (Light Duty)
Compression Ratio	14 to 16.2 (adjustable)
Max. Cyl. Pressure	200 bar
Injection	Central mount – various injector types
Piston & Bowl	piston blanks & variants
DOHC Type II Valvetrain	Fixed cams (selectable)
Swirl ratio	Variable (0.5 ~ 3.0) PDA

Table 2. Properties of Test Fuels

Parameter	Units	E00	E10
Density	g/cc	0.7426	0.7402
H/C Ratio	atomic	1.871	2.062
O/C Ratio	atomic	0	0.0349
LHV	MJ/kg	43.397	42.03
Octane		86.65	87.6
RON		90.6	91.7
MON		82.7	83.4
T0	C	33.8	40.4
T50	C	99.2	102.4
T100	C	209.2	217.9
Aromatics	vol%	26.1	18.7
Olefins	vol%	2.8	11.2
Saturates	vol%	70.9	70.1
Ethanol	vol%	0	9
Sulfur	ppm	5	27.8

A schematic diagram of the engine test setup is shown in [Figure 2](#). The intake and exhaust systems allowed simulated turbocharged operation. Air flow was controlled and measured by a Flow Systems FC-500 system using a 7-nozzle sonic array. Dry air was supplied from building compressed air and heated by an electric heater. An EGR valve and cooler were located between intake and exhaust surge tanks, which insured that EGR was well mixed prior to entering the engine.

Fuel flow was measured with a high-precision Pierburg PLU103b fuel meter. The maximum total error of this instrument is 0.3 percent of reading down to 0.05 g/s fuel flow. All emissions were measured on a wet basis using heated sampling systems. Carbon species, nitrogen species, and some hydrocarbon species were measured with an AVL Sesam FTIR. Total hydrocarbon emissions were measured with a Horiba Fast FID. Exhaust oxygen and intake CO₂ were measured with a Rosemount MLT-3. Smoke was measured with an AVL 415S smoke meter and particulate size distribution was measured with a TSI 3090 Engine Exhaust Particle Size Spectrometer (EEPS).

Cylinder pressure was measured with a flush-mounted Kistler 6125CU20 pressure transducer. A Kistler 2613B crankshaft encoder provided crank position data and was dynamically aligned with engine TDC using a Kistler 2629B TDC probe. Pressure data was sampled every 0.5 CAD.

A single-cylinder engine controller was developed with real-time heat release analysis capability. The controller is based on National Instruments hardware and Labview software [19], and was built by Drivven, Inc. [20]. Within the controller, a multiple-injection control utility [14] was developed to determine and control the fuel quantity, Q, for each injection. Q was determined using instantaneous rail pressure, cylinder pressure, and an embedded calibration map for the injector. Similarly, closed-loop PID control was used for intake and exhaust pressure, EGR, rail pressure, and intake air temperature.

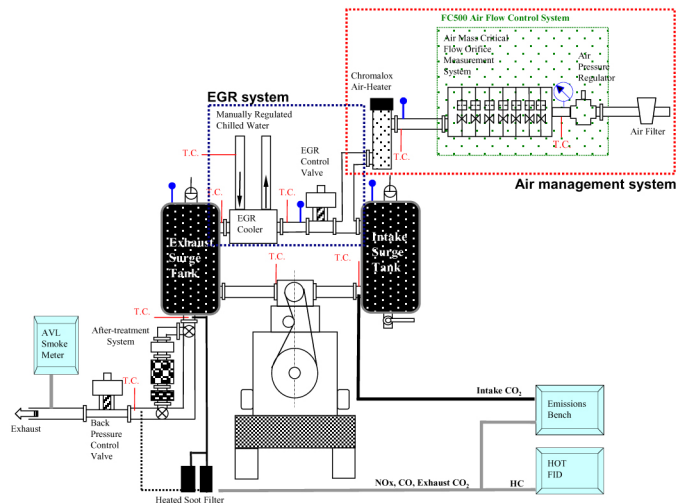


Figure 2. Schematic Diagram of Test Setup

Residual mass concentration in the cylinder was measured by a Delphi Residual Estimator Tool (RET) [21, 22]. The RET provides fast estimates of residuals in the test cell based on measured cylinder pressure, measured airflow, and average temperature and pressure at the intake and exhaust ports.

GDCI combustion involves many design and operating variables with strong interactive effects. The injection quantities and timings for multiple injections need to be optimized, including other system level variables such as swirl ratio (SR), EGR, and boost pressure. A systematic approach is needed to minimize fuel consumption and emissions within constraints such as combustion noise. Low injection pressure is desired to keep cost of the injection system low and reduce fuel pump parasitics. Fuel injection parameters are defined in Figure 3, where SOI1 to SOI5, Q1 to Q5, and PW1 to PW5 are the start-of-injection, injection quantities, and injector driver logic pulse width.

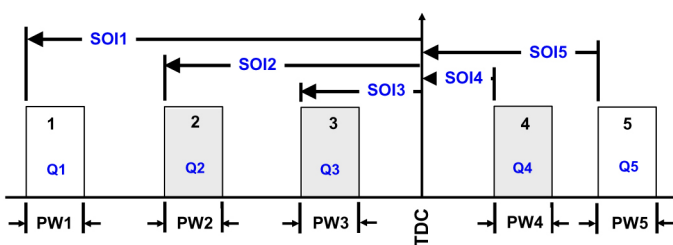


Figure 3. Definitions of Fuel Injection Parameters

Due to the large number of design and operating variables, traditional parametric test methods are not suitable. Design of experiments (DoE), response surface modeling (RSM) and optimization methods were applied using the model-based control (MBC) toolbox from The Mathworks, Inc. [23]. Examples of this overall method can be found in prior work by the authors [14].

INJECTION STRATEGIES

The injection process is critical to controlling mixture stratification for GDCI [14]. Fuel needs to be injected and partially mixed in the period before autoignition occurs. Complete mixing of all the fuel in a homogenous charge must be avoided since this would lead to rapid burning of the whole mixture (i.e., HCCI).

A multiple-late injection (MLI) injection strategy [14] was hypothesized to be advantageous. The intent is to create combustible mixtures in distinct regions of the chamber. The characteristic air-fuel ratio, temperature-pressure history, and ignition delay time may be different for each region. The overall heat release characteristic may be fast enough for high efficiency but long enough for quiet combustion.

Figure 4 shows an example of a triple injection process. The first injection typically occurs at mid-compression stroke and delivers fuel above the piston when cylinder pressure and temperature are relatively low. The second injection typically targets the lower outer bowl. The third injection occurs near top-dead-center (TDC) when cylinder pressure and temperature are higher. Spray penetration is low and this fuel occupies the inner bowl region.

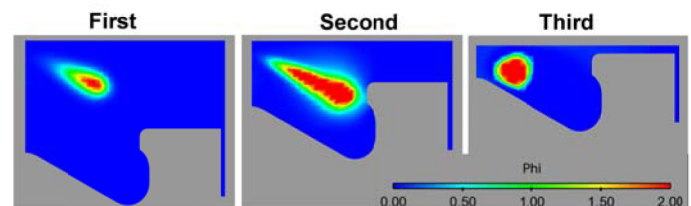


Figure 4. KIVA Simulation of Triple Injection Process.

The injection process can also lead to wall wetting of combustion chamber surfaces. Liquid films on cool cylinder walls may be slow to evaporate and lead to piston and ring lubrication problems. Fuel films including fuel transported into the piston top-land crevice may not oxidize completely. This can be a significant source of HC, CO, and PM emissions and cause reduced engine efficiency.

The injection process was initially examined in previous work [14]. Engine tests were conducted for single injection, double injection, and triple injection at 1500 rpm and 6 bar IMEP. DoE and RSM methods were used to optimize injection quantities, timings, and fuel pressure within constraints for emissions and combustion noise (FSN<0.5, Pinj≤500 bar, CNL≤85 dB). For these tests, intake air temperature was 50 °C, water and oil temperatures were 90 °C, EGR was 45%, and MAP was 1.8 bar.

Results are shown in Figure 5. For single injection, no combination of injection pressure and timing could be found that met targets within constraints. For double injection, NOx near 0.2 g/kWh was measured with ISFC improved over the best single injection data. Injection pressure was reduced and a NOx-ISFC trade-off was apparent. For triple injection, ISFC improved further with 181 g/kWh at NOx of 0.7 g/

kWh. Injection pressure was further reduced. A 7 percent ISFC penalty was needed to achieve the NO_x target of 0.2 g/kWh.

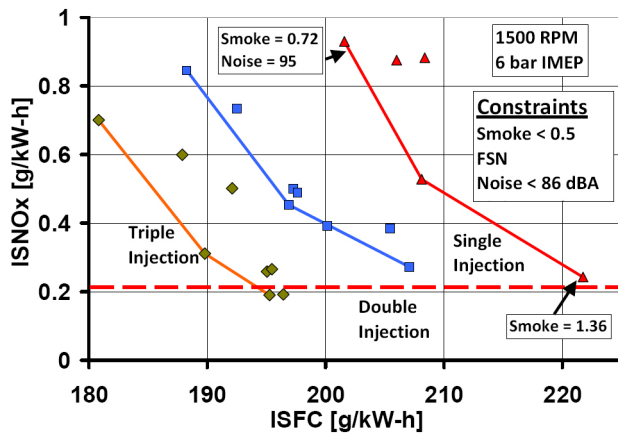


Figure 5. ISNO_x vs. ISFC for Single, Double, Triple Injection (1500rpm-6 bar IMEP) [from ref 14]

Heat release and injection timing events for optimum single, double, and triple injection are shown in Figure 6. Single injection exhibited the highest maximum heat release rate (MHRR) with the latest combustion phasing. Double injection had lower MHRR and more advanced phasing. Triple injection had still lower MHRR with near optimal phasing. It is notable that there was no visible heat release from the first injections until about TDC. Triple injection produced a very desirable HR characteristic with start of combustion (SOC) near TDC and a reasonably abrupt end of combustion (EOC). This provided greater expansion ratio for more of the burned fuel than either single or double injection strategies and contributed to improved fuel consumption for triple injection.

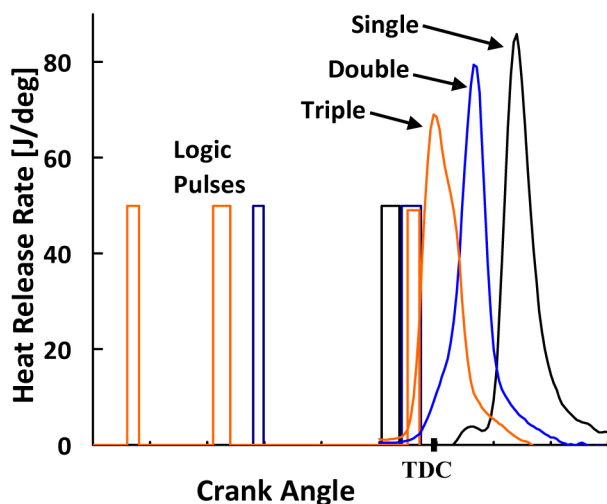


Figure 6. Heat Release and Injection Logic [from Ref 14]

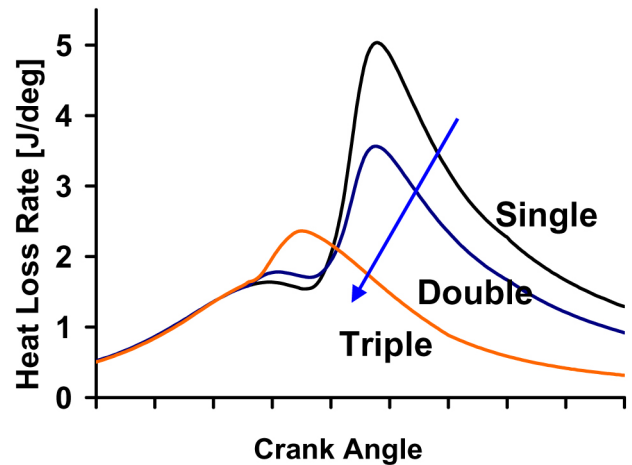


Figure 7. Heat Loss as a Function of Crank Angle [from Ref 14]

Results from an energy balance analysis are shown in Figure 7. While heat losses during the compression stroke are similar for single, double, and triple injections, a large reduction in heat losses during the expansion stroke occurs for triple injection, and less so for double injection. This is attributed to a more favourable distribution of fuel during combustion, which results in less contact between hot combustion gases and the chamber walls. Reduced heat losses contribute to improved fuel consumption for triple injection.

3D CFD simulations were performed using KIVA-3 V-ERC-Chemkin for the test engine operating with the optimum triple injection settings. Phi (equivalence ratio) and temperature contours on the spray centreline at 6 degrees after TDC are shown in Figure 8. The simulation shows that combustion begins in the outer bowl and progresses in distinct regions of the chamber. The third injection has low penetration and burns in the inner bowl region.

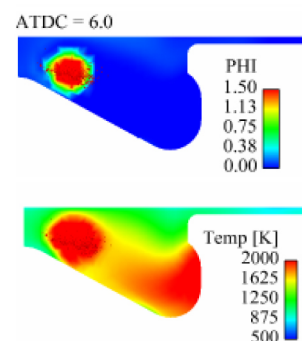


Figure 8. Phi and T Contours on the Spray Axis for Triple Injection at 6 CAD aTDC

The equivalence ratio-temperature (Phi-T) diagram for the optimum triple injection case is shown in Figure 9. Mixture stratification is evident based on separation in the vertical direction in the plot. The Phi-T diagram also shows that fuel