

# Development of a Gasoline Direct Injection Compression Ignition (GDCI) Engine

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## ABSTRACT

In previous work, Gasoline Direct Injection Compression Ignition (GDCI) has demonstrated good potential for high fuel efficiency, low NO<sub>x</sub>, and low PM over the speed-load range using RON91 gasoline. In the current work, a four-cylinder, 1.8L engine was designed and built based on extensive simulations and single-cylinder engine tests. The engine features a pent roof combustion chamber, central-mounted injector, 15:1 compression ratio, and zero swirl and squish. A new piston was developed and matched with the injection system. The fuel injection, valvetrain, and boost systems were key technology enablers.

Engine dynamometer tests were conducted at idle, part-load, and full-load operating conditions. For all operating conditions, the engine was operated with partially premixed compression ignition without mode switching or diffusion controlled combustion. At idle and low load, rebreathing of hot exhaust gases provided stable combustion with NO<sub>x</sub> and PM emissions below targets of 0.2g/kWh and FSN 0.1, respectively. The coefficient of variation of IMEP was less than 3 percent and the exhaust temperature at turbocharger inlet was greater than 250 C. BSFC of 280 g/kWh was measured at 2000 rpm-2bar BMEP. At medium-to-higher loads, rebreathing was not used and cooled EGR provided NO<sub>x</sub>, PM, and combustion noise below targets. MAP was reduced to minimize boost parasitics. At full load operating conditions, near stoichiometric mixtures were used with up to 45 percent EGR. Maximum BMEP was about 20 bar at 3000 rpm. For all operating conditions, injection quantities and timings were used to control mixture stratification and combustion phasing.

Transient co-simulations of the engine system were conducted to develop control strategies for boost, EGR, and intake air temperature control. Preliminary transient tests on a real engine with high rate of load increase demonstrated potential for very good control. Cold start simulations were also conducted using an intake air heating strategy. Preliminary cold start tests on a real engine at room temperature demonstrated potential for very good cold starting. More work is needed to calibrate the engine over the full operating map and to further develop the engine control system.

**CITATION:** Sellnau, M., Foster, M., Hoyer, K., Moore, W. et al., "Development of a Gasoline Direct Injection Compression Ignition (GDCI) Engine," *SAE Int. J. Engines* 7(2):2014, doi:10.4271/2014-01-1300.

## INTRODUCTION

Throughout the world, many efforts are being made to improve the thermal efficiency of internal combustion engines. One relatively new approach is gasoline partially-premixed compression ignition (PPCI) that was introduced by Kalghatgi [1] and first tested by Johansson [2]. A high octane fuel was injected late on the compression stroke of a boosted diesel engine operating with high EGR. The injection process was complete prior to the start of combustion enabling partial mixing of the fuel and air prior to heat release. Very low fuel consumption, NO<sub>x</sub>, and PM emissions were measured. This early work established that gasoline-like fuels with high resistance to autoignition are preferred for PPCI. These fuels have greater ignition delay relative to high cetane fuels, which

gives them more time for mixing. The high volatility of gasoline also helps mix the fuel and air quickly after injection. Kalghatgi taught that PPCI requires some stratification of the fuel-air mixture ("partially premixed but not fully mixed") to achieve ignition and controlled burn duration.

Since this early work, many researchers have performed additional engine experiments and simulations, which have contributed to a significant body of PPCI test data and experience. Tests have been performed on diesel engines using various gasoline-like fuels at Shell [3, 4, 5, 6], Lund [7, 8, 9, 10], Cambridge [11], UW-Madison [12, 13, 14, 15, 16], Argonne [17, 18, 19, 20], Tsinghua [21], and RWTH [22]. In practically all cases, diesel-like efficiency was measured. Various injector designs and piston designs have also been

developed with significant improvements. Delphi [23, 24, 25, 26, 27] reported single-cylinder and multi-cylinder engine test results with various injectors using single, double, and triple injection strategies.

Engine tests have also been performed using naphtha fuels on both modified spark-ignited engines [28] and modified diesel engines [29]. Naphtha has significantly lower octane number (RON 70 to 84) and significantly lower processing cost compared to current market gasoline. This work has shown compatibility of the PPCI combustion process with lower octane fuels in a longer term perspective.

PPCI has demonstrated very good potential for very high fuel efficiency with low engine-out NOx and PM emissions using a range of gasoline-like fuels. However, towards a production solution, significant issues remain. Due to the lower exhaust enthalpy of low temperature engines using PPCI, it is difficult to produce intake boost with acceptable boost system parasitics. A practical powertrain system with robust PPCI combustion is needed, including injection, valvetrain, boost, and exhaust subsystems. The engine must also meet vehicle packaging requirements under hood and satisfy cold start and transient response requirements.

As part of a US Department of Energy funded program, Delphi has been developing a multi-cylinder engine concept for PPCI combustion with current US market gasoline (RON91). The engine has four cylinders with a 1.8L displacement and was designed based on extensive simulations and single-cylinder engine tests. A multiple-late-injection (MLI) strategy with GDI-like injection pressures was selected without use of a premixed charge. The absence of classic knock and pre-ignition limits in this process enabled a higher compression ratio of 15. The engine operates “full time” [25] over the entire operating map with partially premixed compression ignition. No combustion mode switching, diffusion controlled combustion, or spark plugs were used. Delphi uses the term Gasoline Direction Injection Compression Ignition (GDICI) in reference to this combustion process.

One program objective was to build a practical GDICI engine that achieves diesel-like fuel efficiency using current market gasoline (RON91) with engine-out NOx and PM emissions below that needed for NOx or PM aftertreatment. Table 1 lists initial targets for engine testing and development. Combustion noise level (CNL) limits are shown in Figure 1. Other program objectives include demonstration of good transient load response and room temperature cold starts.

In the current work, analysis and simulation tools were used to design and fabricate a new multi-cylinder GDICI engine. Design tools were used to package the powertrain in a D-class vehicle. Engine dynamometer tests were conducted over a range of operating conditions and included preliminary calibration mapping. With this data, a competitive assessment of brake specific fuel consumption (BSFC) was performed using published data for gasoline, diesel, and hybrid vehicle engines.

Finally, aggressive transients with high rate of load increase were simulated and then tested on the real engine. Cold starts at various ambient temperatures were also simulated and tested on the real engine.

Table 1. Preliminary Targets for Engine Testing.

|                       |                  |
|-----------------------|------------------|
| Engine Out NOx        | $\leq 0.2$ g/kWh |
| Engine Out PM         | $\leq 0.1$ FSN   |
| CNL                   | load dependent   |
| COV IMEP              | $\leq 3\%$       |
| Combustion Efficiency | $\geq 95\%$      |

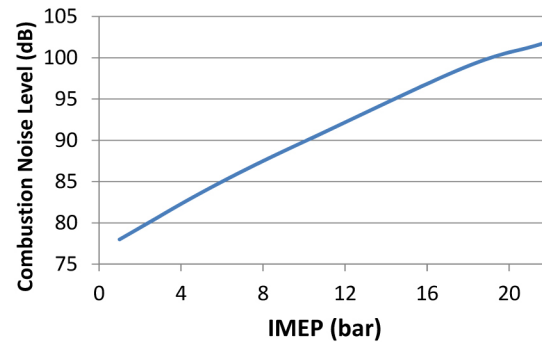


Figure 1. Combustion Noise Level (CNL) Limits used for Testing.

## GDCI CONCEPT AND INJECTION STRATEGY

The GDICI engine concept combines the best of diesel and spark-ignited engine technology. Like diesel engines, the compression ratio is high, there is no intake throttling, and the mixture is lean for improved ratio of specific heats. GDICI uses a new low-temperature combustion process for partially-premixed compression-ignition. Multiple-late-injections of gasoline (RON91) vaporize and mix very quickly at low injection pressure typical of direct injected gasoline engines. Low combustion temperatures combined with low mixture motion and reduced chamber surface area result in reduced heat losses.

A schematic of the GDICI combustion chamber concept is shown in Figure 2. The engine features a shallow pent roof combustion chamber, central-mounted injector, and 15:1 compression ratio. A quiescent, open chamber design was chosen to support injection-controlled mixture stratification. Swirl, tumble, and squish were minimized since excessive mixture motion may destroy mixture stratification created during the injection process. The piston and bowl shape were matched with the injection system and spray characteristics. The bowl is a symmetrical shape and was centered on the cylinder and injector axes.

The GDICI injection strategy is central to the overall GDICI concept and is depicted in the Phi-T (equivalence ratio-temperature) diagram shown in Figure 3. The color contours in Figure 3 show simulated CO emissions for a single injection process involves one, two, or three injections during

the compression stroke and are shown as quantities Q1, Q2, and Q3. Each injection begins in the upper left of the Phi-T diagram (liquid) and quickly vaporizes and mixes to Phi less than 2 by the start of combustion (SOC). At SOC the fuel-air mixture is stratified to achieve stable ignition and controlled heat release. Wall wetting is minimized and fuel is kept away from cold zones such as the piston topland and cylinder liner that may impede complete oxidation.

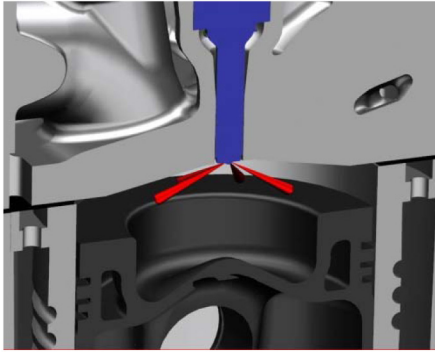


Figure 2. GDCI combustion chamber concept.

The ideal “stratification line” is shown in Figure 3 and represents the ideal injection process. To achieve low NOx and low PM emissions simultaneously, combustion must occur “in the box” shown in Figure 3 (avoiding soot and NOx formation regions). To also minimize CO emissions, which can compromise efficiency, combustion must occur in the region  $0 < \Phi < 1.2$  with  $1300 < T < 2200$  degrees K. Therefore at SOC, which is approximately top-dead-center (TDC), all parcels in the combustion chamber should be no richer than Phi of approximately 1.2. This corresponds to the top of the stratification line in Figure 3. The stratification line is inclined due to charge cooling effects with the richest parcels cooling more than the leanest parcels.

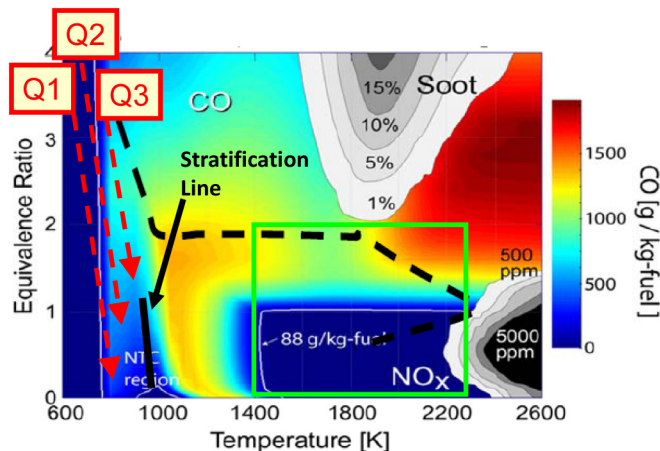


Figure 3. GDCI Injection Strategy depicted on Phi-T Diagram with NOx, Soot, and CO Contours.

KIVA simulations were performed with this combustion system and an injector spray model developed using spray chamber test data. A full 360-degree mesh with a fine grid was necessary to capture the injection, mixing, and combustion

phenomena. Results for a double injection strategy at 2000 rpm and 11 bar indicated mean effective pressure (IMEP) are shown in Figure 4a,b,c,d,e at various crank positions.

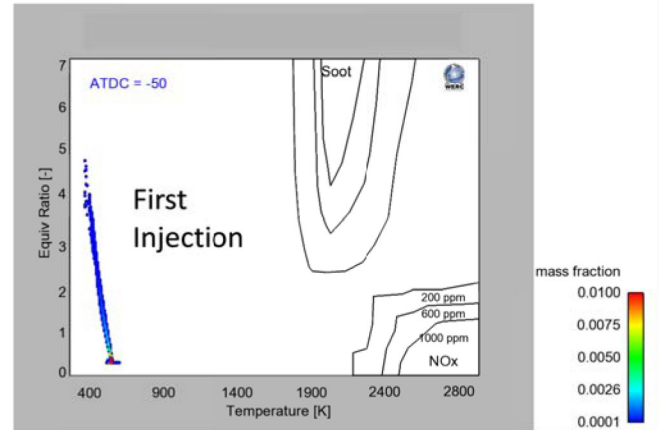


Figure 4a. Phi-T diagram at -50 CAD atdc (during first injection).

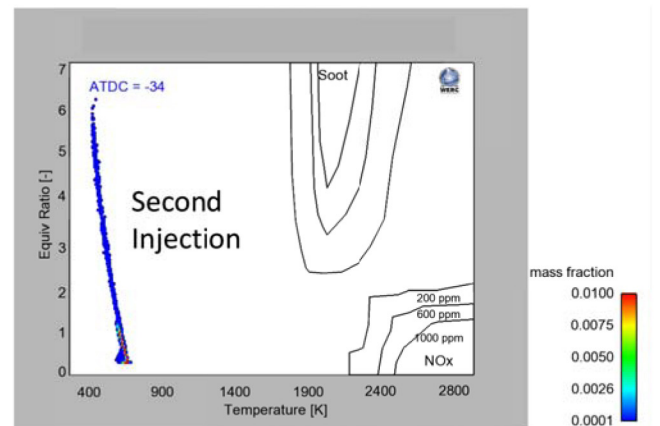


Figure 4b. Phi-T diagram at -34 CAD atdc (start of second injection).

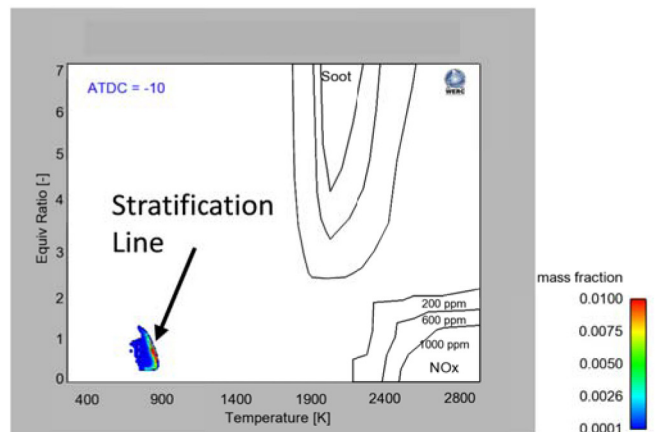


Figure 4c. Phi-T diagram at -10 CAD atdc (prior to start of combustion).

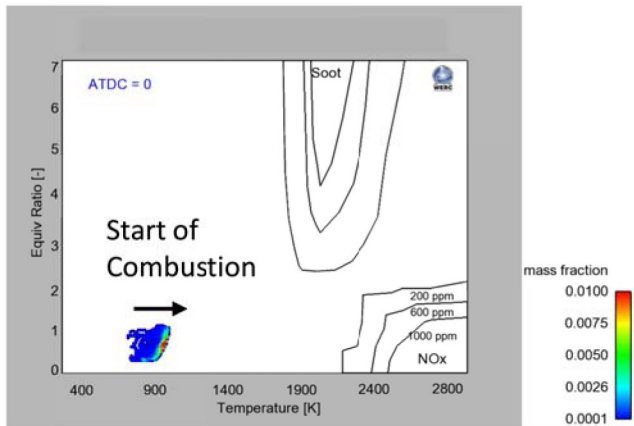


Figure 4d. Phi-T diagram at 0 CAD atdc (near start of combustion).

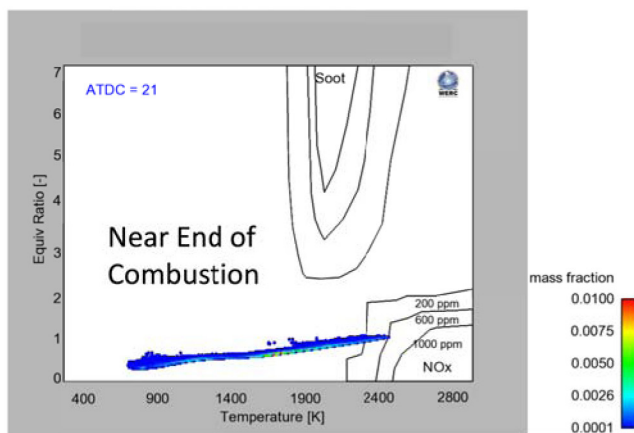


Figure 4e. Phi-T diagram at 21 CAD atdc (near end of combustion).

Figure 4a at -50 crank angle degrees (CAD) after-top-dead-center (atdc) corresponds to the first injection. The fuel spray is entraining air rapidly with very fast atomization and vaporization in the bulk gas. Each data point plotted in Figure 4a corresponds to a different parcel in the combustion chamber with color indicating the fuel mass fraction. Figure 4b at -34 CAD atdc corresponds to the beginning of the second injection. Like the first injection, fuel remains in the bulk gas without contacting cool wall zones. This second and last injection controls the ignition process. Figure 4c at -10 CAD atdc shows the mixture stratification prior to start of low-temperature reactions. As fuel vaporizes and mixes, the equivalence ratio of the richest part of the charge is decreasing. If too lean, the charge will be overmixed and ignition compromised. If too rich, soot and CO emissions will increase. The stratification line is inclined to lower temperatures for the richer elements of the charge due to charge cooling. Figure 4d at TDC, is approximately start of combustion with some temperature rise visible in the figure. Parcels with Phi near stoichiometry are the first to release heat. Figure 4e at 21 CAD atdc corresponds to near end-of-combustion (EOC). CO has burned out across the chamber and small amounts of NOx were produced in the hottest portions of the chamber. The temperature in the Phi-T plot has now shifted to the right due to heat release and temperature rise for each parcel. Richer parcels realize higher temperature rise than the leanest parcels.

This sequence of KIVA simulations represents the intended injection, mixing, and combustion processes for GDCI combustion. Because combustion occurs at relatively low peak gas temperatures, it is referred to as low-temperature combustion. The lower combustion temperatures are a main factor in reducing heat transfer through the cylinder head, liner, and piston and contribute to very low indicated specific fuel consumption and emissions.

## ENGINE DESCRIPTION

The absence of classic combustion knock and preignition for GDCI means that a GDCI engine can be operated on RON91 gasoline at high compression ratio for high efficiency. For this reason, GDCI is also a good candidate for aggressive downsizing, down-speeding, and boosting, which is a proven strategy for high vehicle-level fuel economy. These were two main considerations in base engine design.

Table 2. Base Engine Specifications.

|                    |                      |
|--------------------|----------------------|
| Engine Type        | GDCI                 |
| Configuration      | In Line 4-Cylinder   |
| Displacement       | 1.8 L                |
| Bore-Stroke-Rod    | 82 x 85.2 x 144.7 mm |
| Piston Pin Offset  | 0.5 mm               |
| Compression Ratio  | ~15                  |
| Chamber            | Shallow Pent Roof    |
| Peak Cyl. Pressure | 200 bar              |
| Injection          | Central-Mount GDI    |
| Valvetrain         | 4V DOHC Type II      |
| Piston             | GDCI Design          |
| Swirl              | none                 |



Figure 5. GDCI Multi-cylinder Engine.

Table 2 lists the base engine specifications. The engine is a four-cylinder 1.8L engine with four-valves-per-cylinder. Bore, stroke, and connecting rod length are 82 mm, 85.2 mm, and 144.7 mm, respectively, for B/S of 0.96 and r/l of 0.29. With

geometric compression ratio of 15, this engine has geometry similar to modern light duty diesel engines. A photograph of the first assembled GDCI engine is shown in [Figure 5](#).

### Cylinder Head

The cylinder head is an all new design with four-valve-per-cylinder rated at 200 bar peak cylinder pressure (PCP) for development purposes. Extensive analysis work was conducted to develop the cylinder head using conventional aluminum alloys to meet cylinder sealing requirements. A new multilayer steel head gasket was developed for this application. The injector is central-mounted without offset or inclination relative to the cylinder axis. The combustion chamber features a shallow pent roof with valves flush with the pent roof surface. The low pent roof angle was selected to achieve high compression ratio, while maintaining low chamber surface area and reasonable valve sizes. Intake and exhaust ports were developed for good flow characteristics but without the need for swirl, tumble, or squish. This was a significant advantage and produced good intake flow efficiency as a function of valve lift as measured on a steady flow bench. A double overhead camshaft, Type II valvetrain with roller finger followers was selected due to low friction and compact packaging. No spark plug exists in the cylinder head but there is provision for cylinder pressure sensors. In addition to cylinder pressure sensing, Kistler cylinder pressure transducers were mounted vertically at the cylinder periphery. Overall, the cylinder head is very compact and employs conventional aluminum casting methods, while meeting demanding structural requirements.

### Piston

A new piston was designed for GDCI based on extensive KIVA simulations. The piston bowl shape and injector spray characteristics were matched for typical GDCI injection timings. This resulted in significantly reduced piston surface area and less propensity for wall wetting relative to typical diesel pistons. The bowl is symmetrical and is located on the cylinder axis.

**Table 3. Main Features of GDCI Piston relative to typical LD diesel piston [27].**

| Feature             | Units           | GDCI  | Diesel | % Reduction |
|---------------------|-----------------|-------|--------|-------------|
| Piston Surface Area | cm <sup>2</sup> | 60.67 | 80.53  | 24.7        |
|                     | % of Bore       |       |        |             |
| Squish Area         | Area            | 8     | 67     | 88.1        |
| Topland Volume      | mm <sup>3</sup> | 291   | 1150   | 74.7        |
| Blowby Area         | mm <sup>2</sup> | 0.019 | 0.11   | 82.7        |

The main features of the GDCI piston are listed in [Table 3](#) [27]. The piston, pin, and ring pack were rated at 200 bar peak cylinder pressure. Surface area, squish area and blowby area (defined by ring end gap and second land radial clearance) were significantly reduced relative to typical light-duty diesel piston designs (BMW M57 piston used as reference). Back-to-back tests were conducted on a single-cylinder engine using

the GDCI and baseline diesel pistons. Significant fuel consumption improvements were measured at several speed-load points as shown in [Figure 6](#) [27].

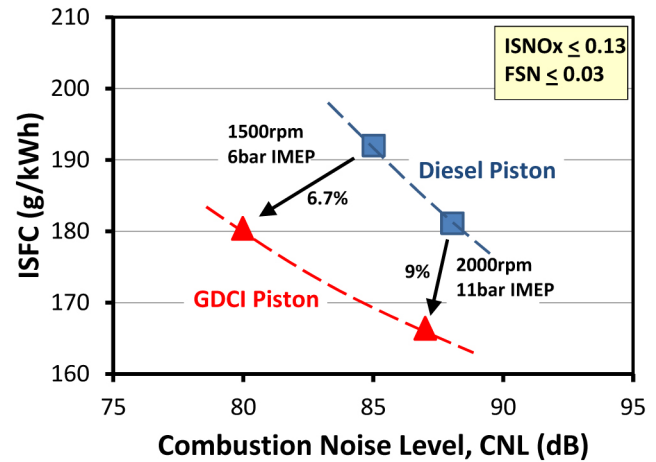


**Figure 6. Fuel consumption and combustion noise for GDCI piston compared to diesel baseline from Delphi [24].**

### Injection System

Previous work by the authors has demonstrated improved GDCI combustion performance through injection system developments [23, 24, 25, 26, 27]. Simulation tools were used to develop injector spray characteristics that provide very fast vaporization, low spray penetration, while also providing the necessary injection rate (flow rate). Designed experiments on a single-cylinder engine were used to determine best injection timings, quantities, and injection pressures for various injector designs. Injection pressures are in the range used by production gasoline direct injection (GDI) engines. Lower injection pressures are desirable to reduce pump parasitics and fuel system cost.

The fuel pump was mounted at the end of the cylinder head and was driven by the exhaust camshaft. The pump shaft has four lobes and exhibits excellent rail pressure control via modulation of the spill valve. The fuel rail has inlet and outlet orifices for dampening of pressure pulsations in the system. Injector drive waveforms were developed to support the multiple late injection strategies used for GDCI. Importantly, the system delivers accurate fuel quantities for multiple injections in all cylinders.

### Valvetrain System

Full-time GDCI combustion over the operating map is achievable by using exhaust rebreathing (RB) at low loads and cooled EGR at medium to higher loads. Rebreathing (RB) is accomplished using the exhaust valvetrain system, which provides a secondary exhaust lift event during the intake stroke. It is an effective method to recuperate heat from hot exhaust gases in order to raise mixture temperatures. This heat promotes autoignition at low loads when boost pressure is zero or low. Rebreathing also collapses the pumping loop (intake and exhaust valves are open at same time) and raises

exhaust temperatures significantly. In this way, a close-coupled oxidation catalyst will remain warm over a wide range of low loads without special maintenance heating.

The valvetrain system is a Type II, continuously-variable mechanical system that provides the rebreath functionality. The valvetrain has good packaging characteristics with low friction. A Delphi electric camshaft phaser is used to actuate the exhaust valvetrain and control the secondary valve lift with very fast response. [Figure 7](#) shows a photograph of the front of the engine with electric camshaft phasers.



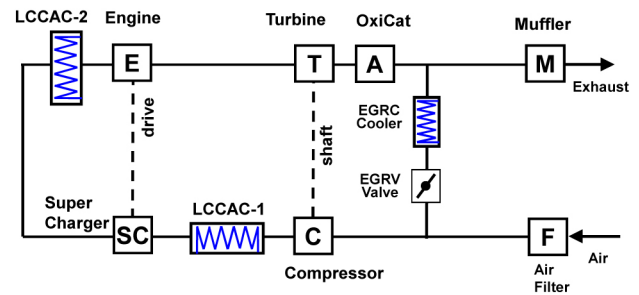
[Figure 7](#). Photograph of front of engine with electric phasers.

## Boost System

Intake boosting is difficult for low-temperature engines because the exhaust enthalpy to drive a turbocharger is very low. For medium-and-high-load operation, GDCI requires increased cooled EGR to dilute the charge and maintain proper combustion phasing. This puts greater demand on the boost system. Extensive engine simulations were performed [26] to develop a system that provides the necessary intake boost with low boost parasitics. Several boost architectures with various boost devices were studied including a 1-stage turbocharger (TC), a 2-stage TC, a 1-stage supercharger (SC), and a 2-stage TC-SC system. The 2-stage system with one variable geometry TC, one SC, and two charge coolers was selected for the most efficient operation over the operating range.

[Figure 8](#) shows a schematic of the boost system. No intake throttle is used. The overall system including liquid-cooled charge air coolers was developed and packaged for minimum system volume, good cylinder-to-cylinder flow uniformity, and low flow restriction on the vehicle. Since the supercharger is a positive displacement device driven by the engine, transient response of the boost system is very fast.

The LPL-EGR system is also shown in [Figure 8](#). The exhaust gases exit the turbine and pass through a close-coupled oxidation catalyst to reduce HC and CO emissions. Exhaust gases at this point are already relatively cool. The EGR flow enters an EGR cooler located upstream of the EGR valve. The EGR cooler operates at engine coolant temperature to prevent water condensate at the TC compressor inlet. The EGR system is designed for high flow in a compact layout for fast response on the vehicle.

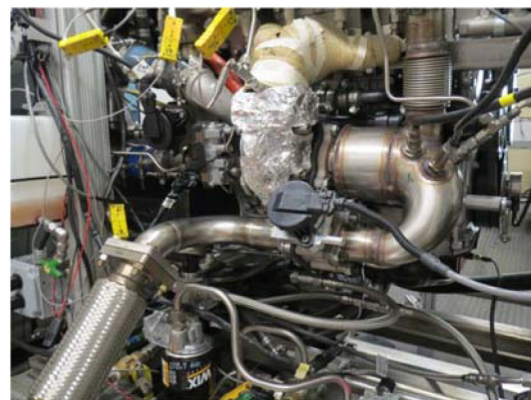


[Figure 8](#). Schematic of boost system with charge air coolers and LPL-EGR.

## Exhaust System and Aftertreatment

[Figure 9](#) shows a photograph of the exhaust system used on multicylinder GDCI engines. The manifold has a 4-into-1 configuration with compact, symmetrical design, and tubular construction. The runner lengths are nearly identical for each cylinder for uniform flow and pressure dynamics. This insures that rebreathing of exhaust gases will also be uniform across the cylinders. The exhaust system is insulated to maintain sufficient exhaust temperatures for aftertreatment.

The aftertreatment system is comprised of only an oxidation catalyst. The intent is to satisfy tailpipe NOx and PM emissions targets using clean GDCI combustion technology. The oxidation catalyst is close-coupled to the turbocharger outlet for heat conservation. The catalyst brick is a round, alumina monolith with high loading of platinum and palladium for low temperature light off of CO and HC species. The entire exhaust system including flex pipes and sensors is packaged for the GDCI vehicle application. An exhaust pipe and muffler were used for dynamometer testing to emulate the restriction of the vehicle exhaust system.



[Figure 9](#). Photograph of GDCI exhaust system on a dynamometer engine.

## EXPERIMENTAL METHODS

Experiments were performed on an engine dynamometer at Delphi. The test fuel was Shell regular unleaded gasoline with approximately 10 percent ethanol and research octane number (RON) of 91, as is representative of common gasoline sold in the United States. Fuel property data are listed in [Table 4](#).