

Second Generation GDCI Multi-Cylinder Engine for High Fuel Efficiency and US Tier 3 Emissions

Mark Sellnau, Matthew Foster, Wayne Moore, James Sinnamon,
Kevin Hoyer, and William Klemm
Delphi Powertrain

ABSTRACT

The second generation 1.8L Gasoline Direct Injection Compression Ignition (GDCI) engine was built and tested using RON91 gasoline. The engine is intended to meet stringent US Tier 3 emissions standards with diesel-like fuel efficiency. The engine utilizes a fulltime, partially premixed combustion process without combustion mode switching.

The second generation engine features a pentroof combustion chamber, 400 bar central-mounted injector, 15:1 compression ratio, and low swirl and squish. Improvements were made to all engine subsystems including fuel injection, valve train, thermal management, piston and ring pack, lubrication, EGR, boost, and aftertreatment. Low firing friction was a major engine design objective.

Preliminary test results indicated good improvement in brake specific fuel consumption (BSFC) over the first generation GDCI engines, while meeting targets for engine out emissions, combustion noise and stability. BSFC at the 2000 RPM-2 bar BMEP global test point was 248 g/kWh. At 1500 RPM, low BSFC of 210 to 215 g/kWh was measured over a wide load range from 6 to 12 bar BMEP. Friction levels were reduced considerably over Gen 1 engine friction. Tests at 800 RPM idle with zero brake torque indicated very stable combustion with very low NOx and PM emissions.

Initial warmed up tailpipe CO emissions were very low with good potential to meet US Tier 3 targets. Tailpipe non-methane HC were also low at low to medium loads, but increased at higher loads. Improved HC aftertreatment will be needed for Tier 3 targets. NOx aftertreatment was not used on this engine but is expected to be needed for Tier 3-Bin 30 emissions standards.

A detailed thermodynamic loss analysis was performed at various operating conditions. Indicated thermal efficiency was high due to low heat losses and an efficient combustion process. A mechanical loss analysis was also performed. While firing friction was low, boost parasitics were significant at higher loads. Reduction of boost parasitics is an opportunity for further brake efficiency improvements.

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INTRODUCTION

Near term fuel economy and CO₂ regulations throughout the world are becoming increasingly stringent. In the United States (US) by 2025, standards for corporate average fuel economy (CAFE) and vehicle CO₂ emissions are greatly reduced (Table 1) [1] and will require large reductions in passenger car fuel consumption. Simultaneously, new Tier 3 emissions standards established by the US Environmental Protection Agency (EPA) are also very stringent [2]. Tier 3- Bin 30 emissions targets for 2025 are shown in Table 2.

Table 1. Corporate Average Fuel Economy (CAFE) and vehicle CO₂ emissions standards for the United States through 2025.

Year	CAFE Target (MPG)	CO ₂ Target (gCO ₂ /mile)
2011	27.6	322
2016	35.5	250
2025	54.5	163

Table 2. US Tier 3- Bin 30 emissions standards for 2025.

Species	Drive Cycle (g/mile)		
	City	Hwy	US06
NMOG + NOx	0.030	0.030	0.140
CO	1	-	8
PM	0.003	-	0.06
HCHO	0.004	-	-

As well, shifting world demand for diesel fuel, jet fuel, and heavier distillates, will require changes to crude oil refineries that will negatively impact the refining process, fuel cost and energy efficiency [3]. There is a need for clean and efficient engines operating on gasoline-like fuels for passenger cars and trucks in light, medium, and heavy duty segments.

One approach to achieve an ultra-efficient and inherently clean internal combustion engine operating on gasoline fuels is gasoline partially-premixed compression ignition (PPCI). In this combustion process, fuel is directly injected into the cylinder and partially but not fully premixed prior to the start of heat release. Initial studies of PPCI were conducted in 2005 by Kalghatgi [4] and Johansson [5]. Since then, widespread interest in PPCI has led to numerous studies and a significant body of PPCI test data and experience. Tests have been performed on diesel engines using various gasoline-like fuels at Shell [6, 7, 8, 9], Lund [10, 11, 12, 13], Cambridge [14], UW-Madison [15, 16, 17, 18, 19], MIT [20], Argonne [21, 22, 23, 24], Tsinghua [25], and RWTH [26]. In practically all cases, diesel-like efficiency was measured.

Various injector designs and piston designs have also been developed with significant improvements. Delphi [27, 28, 29] 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 [30, 31, 32, 33, 34]. Naphtha has significantly lower octane number (RON 60 to 84) and 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.

As part of a US Department of Energy funded program, Delphi and partners have been developing a multi-cylinder engine concept for PPCI combustion with current US market gasoline (RON91). Delphi uses the term Gasoline Direct Injection Compression Ignition in reference to this combustion process. A description of the first-generation 1.8L four-cylinder GDCI engine, including injection, valve train, boost, and exhaust subsystems was reported [35, 36, 37]. The engine was operated “full time” [28] over the operating map with partially premixed compression ignition. No combustion mode switching, diffusion controlled combustion, or spark plugs were used. The engine was packaged in a D-class passenger vehicle. The electrical and electronic engine control system for the first generation engine is presented in [38]. Initial vehicle test results are presented in [39]. While engine dynamometer test results were encouraging and the vehicle demonstrated excellent fuel economy, not all program goals were achieved.

A second US DOE funded program beginning in 2014 lead to development of the second generation GDCI multi-cylinder engines. Key program objectives were to achieve diesel-like fuel efficiency with tailpipe (TP) emissions that meet US Tier 3- Bin 30 standards. This brought renewed focus on aftertreatment technologies that were expected to be needed. Other program objectives included demonstration of good transient response, increased operating speed-load range, and robust idles and cold starts with low emissions.

Preliminary targets for engine-out emissions, combustion noise levels (CNL), and combustion stability to be used during engine development and testing are shown in Table 3. An alternative engine out (EO) NOx target of 1 g/kWh was also established to understand possible fuel economy tradeoffs.

Table 3. Preliminary targets for engine testing (engine out emissions).

Engine Out NOx	≤ 0.2 g/kWh
Engine Out PM	≤ 0.1 FSN
CNL	load dependent
COV IMEP	≤ 3%

In the current work, the second generation 1.8L GDCI engine was designed and built, and underwent preliminary dynamometer tests. Improvements were made to all engine subsystems including fuel injection, valve train, thermal management, piston and ring pack, lubrication, EGR, boost, and aftertreatment. A top design objective was reduced firing friction. Another top design objective was an improved thermal management system for cold start and warm up with variable intake air temperature control.

The engine was tested for a range of load including 800 RPM idle with zero brake torque. For low loads, an exhaust rebreathing strategy was used with elevated intake air temperature. At medium and higher loads, cooled EGR was used. An efficiency loss analysis was conducted over a range of operating conditions to understand opportunities for further design improvements.

GDCI CONCEPT AND INJECTION STRATEGY

The GDCI engine concept combines concepts from both diesel and spark-ignited engine technology [37]. Like diesel engines, the compression ratio is high, there is no intake throttling, and the mixture is lean for improved ratio of specific heats. GDCI uses a low-temperature combustion process for partially-premixed compression-ignition. Multiple 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 significantly reduce heat losses.

The GDCI injection strategy is central to the overall GDCI concept and is depicted in the Phi-T (equivalence ratio-temperature) diagram shown in Figure 1. The color contours in Figure 1 show simulated CO emissions concentration. The injection process involves one, two, or three injections during the compression stroke, which 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 top land and cylinder liner that may impede complete oxidation.

The ideal “stratification line” is shown in Figure 1 and represents the ideal injection process. To achieve low NOx and low PM emissions simultaneously, combustion must occur “in the box” shown in Figure 1 (avoiding soot and NOx formation regions). To also minimize CO emissions, which can compromise efficiency, combustion must occur in the smaller region defined by $0 < \Phi < 1.2$ with $1300 < T < 2200$ degrees K. Therefore at SOC, which is approximately top-deadcenter (TDC), all parcels in the combustion chamber should be no richer than Φ of approximately 1.2. This corresponds to the top of the stratification line in Figure 1. The stratification line is inclined due to charge cooling effects with the richest parcels cooling more than the leanest parcels. Extensive KIVA simulations with this combustion system and an injector spray model were reported in 2014 [36].

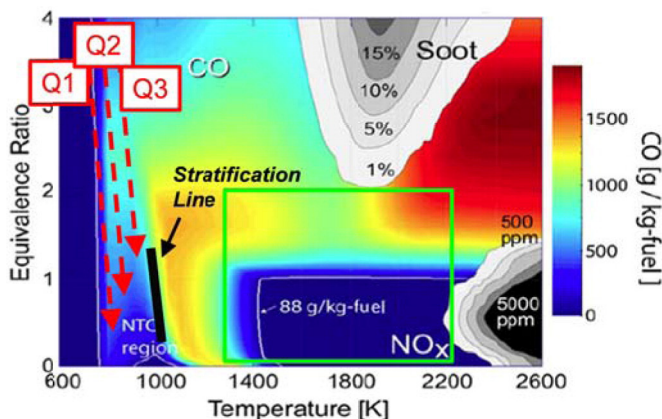


Figure 1. GDCI injection strategy depicted on Phi-T Diagram with NOx, Soot, and CO Contours.

ENGINE DESCRIPTION

The second generation GDCI engine was based on the first generation GDCI engine [35, 36, and 37] and shares a common platform. Engine specifications are listed in Table 3. Photos of the Gen 2 GDCI engine are shown in Figures 2 and 3. The first Gen 2 engine was built in late 2014 and tested on a dynamometer in 2015.

Table 4. Engine specifications - Gen 2 GDCI engine.

Engine Type	4-valve Compression Ignition
Fuel	US Gasoline RON 91 E10
Displacement	1.8 L
Bore x Stroke x Rod	82 x 85.2 x 148.2 mm
Compression Ratio	~ 15:1
Chamber	Shallow Pentroof 10 deg
Swirl and Squish	Minimal Mixture Motion
Piston	GDCI Omega
Oil Jets	Enhanced Individual-Cylinder
Cylinder Pressure Rating	200 bar
Injection System	Central Mount, 400 bar GDi Multi-hole
DOHC Type II Valvetrain	Exhaust Only – Exhaust Rebreathing
Boost	2-Stage (Turbo and Super)
EGR	Low Pressure EGR Cooled
Thermal Management	Low and High Temperature Loops
Aftertreatment	Gasoline Oxidation Catalyst
Lube Oil	Full Synthetic 0W-30

Low base engine friction is required in automotive engines to achieve world class fuel economy. For the Gen 2 GDCI engines, low engine friction was a key design objective. About ten base engine design changes were implemented. The changes included piston, piston pin, and ring pack; accessory drive, valve train, and lubricant properties. The piston-to-cylinder bore clearance was increased and the main bearing bores were precision machined with a deck plate. The mechanical water pump was also replaced with an electric water pump. Measured engine friction is presented in a later section of this paper.



Figure 2. Photo of the Gen 2 GDCI 1.8L multi-cylinder engine (front view).



Figure 3. Photo of the Gen 2 GDCI 1.8L multi-cylinder engine (top view).

Cylinder Head

The cylinder head has four-valves with double-overhead camshafts and central injection (Figures 4 and 5). The injector is mounted vertically 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. The intake and exhaust ports were developed for high flow efficiency but without the need for swirl, tumble, or squish. High intake and exhaust flow efficiency was measured as a function of valve lift on a steady flow bench. The cylinder head features a Type II valve train with roller-finger followers with low friction and very compact packaging. No spark

plug exists in the cylinder head but an automotive grade cylinder pressure sensor (CPS) was mounted nearly flush with the chamber surface under the intake valves in each cylinder. In addition to the automotive grade cylinder pressure sensors, Kistler cylinder pressure transducers were mounted vertically at the cylinder periphery in each cylinder. Figure 6 shows the chamber layout. The aluminum cylinder head and 5-layer MLS head gasket are rated at 200 bar peak cylinder pressure. Overall, the cylinder head is very compact and employs conventional aluminum casting methods and alloys, while meeting demanding structural requirements.

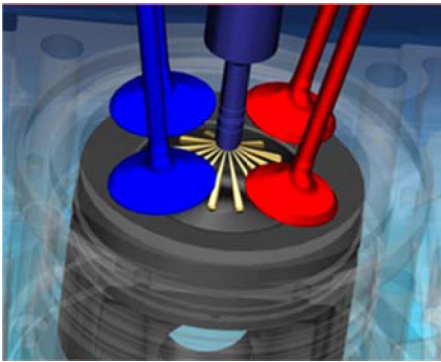


Figure 4. Packaging of central injector and valves in Gen 2 GDCI engine. Fuel Spray is not representative as shown.

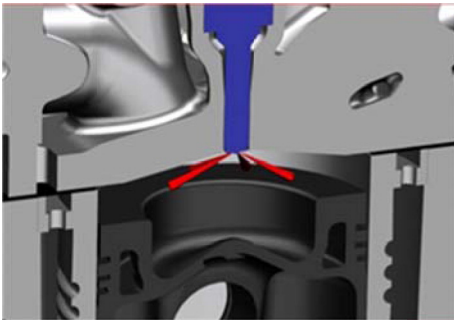


Figure 5. Cross section of Gen 2 cylinder head showing GDI injector and piston. Piston bowl design is not representative as shown.

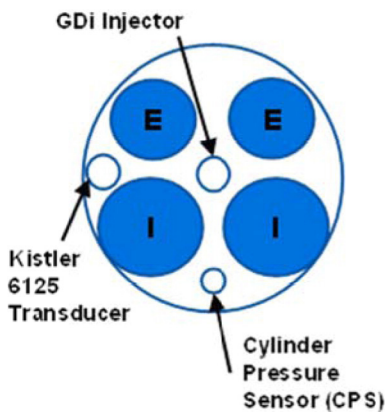


Figure 6. Layout of combustion chamber showing positions of GDI injector, cylinder pressure sensor, and Kistler pressure transducer.

Piston

A new piston and ring pack were redesigned for the Gen 2 GDCI engines. The piston bowl shape was matched to injector spray characteristics and features an open symmetric bowl located on the cylinder axis. The propensity for wall wetting was greatly reduced and bowl surface area was greatly reduced relative to typical diesel pistons [35]. When combined with low mixture motion and low combustion temperatures, the combustion system has fundamentally very low heat losses.

Injection System

The GDCI injection system has evolved considerably. Simulation tools were used to develop injector spray characteristics that provide very fast vaporization and low spray penetration, while also providing the necessary high injection rate (flow rate). The multi-hole GDI injector features a symmetric spray pattern with the intent to minimize piston and liner wetting for all injections. A variety of GDCI injectors have been built and tested on single-cylinder engines and in a high pressure, high-temperature spray chamber.

The GDCI injection system delivers accurate fuel quantities for multiple injections over a wide range of engine speeds and injection timings. In general, low injection pressures are desirable to reduce pump parasitics and fuel system cost. The GDI fuel pump was mounted at the end of the cylinder head and was driven by the exhaust camshaft (Figure 3). The pump shaft has four lobes and maintains excellent rail pressure control via modulation of the spill valve. The fuel rail has a log shape and seals to the injectors with Orings. An orifice at the fuel rail inlet and orifices at each injector inlet were tuned to dampen pressure pulsations in the system. Injector drive waveforms were developed to support the multiple injection strategies used for GDCI.

Valve Train System

Full-time GDCI combustion over the full operating map is achieved by using exhaust rebreathing (RB) at low loads and cooled EGR at medium to higher loads. Exhaust rebreathing is accomplished using the exhaust valve train 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 reduces pumping work (intake and exhaust valves are open at the same time) and raises exhaust temperatures significantly [37]. In this way, a close-coupled catalyst will remain warm over a wide range of low loads without special maintenance heating [37].

The rebreath functionality is provided by a Type II, continuously-variable mechanical valve train system. The valve train has compact packaging characteristics with low friction. A Delphi electric camshaft phaser is used to actuate the exhaust valve train and control the secondary valve lift with very fast response.

Boost and EGR Systems

Intake boosting is difficult for low-temperature engines because the exhaust enthalpy to drive a turbocharger (TC) is very low. Extensive engine simulations were performed [40] to develop a boost system that provides the necessary intake boost with low boost parasitics. The system includes a variable geometry turbocharger, a supercharger (SC) with clutch, and two charge coolers. No intake throttle is used. Figure 7 shows a schematic of the boost system.

The low pressure loop exhaust gas recirculation (LPL-EGR) system is also shown in Figure 7. The exhaust gases exit the turbine and pass through a close-coupled oxidation catalyst to reduce HC and CO emissions. The EGR system is designed for high flow in a compact layout for fast response on the vehicle. The EGR valve is located upstream of the EGR cooler to maintain higher valve operating temperatures and to minimize deposit accumulation on the valve.

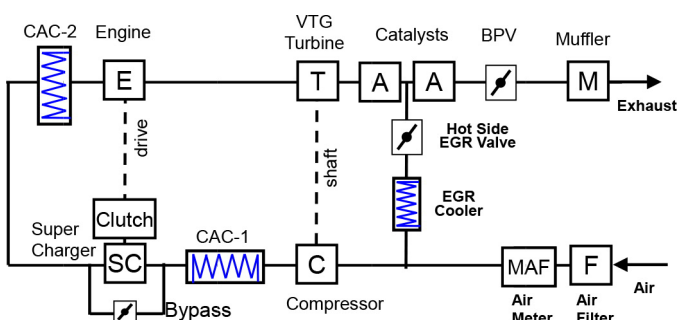


Figure 7. Schematic of two-stage boost system with two charge air coolers and LPL-EGR.

Aftertreatment and Exhaust Systems

Figure 8 shows a photograph of the aftertreatment system (ATS) used on Gen 2 GDCI engines for initial dynamometer testing. The aftertreatment system is comprised of only an oxidation catalyst, which was close-coupled to the turbocharger outlet for heat conservation. While NOx aftertreatment is likely to be needed for stringent US Tier 3- Bin 30 emissions standards, no NOx aftertreatment was used on this engine. The intent was to satisfy PM emissions targets using clean GDCI combustion technology without a gasoline particulate filter (GPF). The entire exhaust system, including flex pipes, sensors, and back pressure valve, was packaged for GDCI vehicle applications. An exhaust pipe and muffler were used for dynamometer testing to emulate the restriction of the vehicle exhaust system.

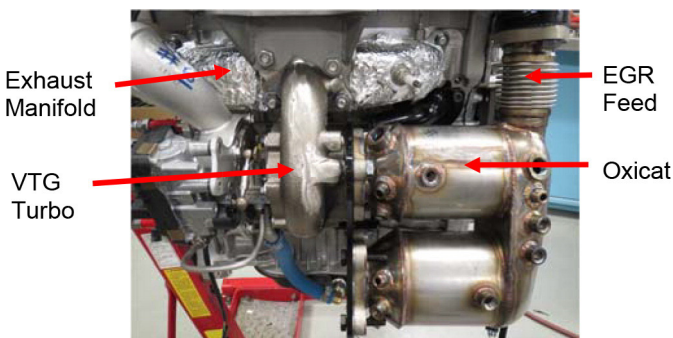


Figure 8. Photograph of GDCI exhaust system for initial engine dynamometer testing (insulation removed for photo).

THERMAL MANAGEMENT SYSTEM

Autoignition of the fuel-air mixture for GDCI depends on cylinder gas temperature during the compression process. Cylinder gas temperature depends on the coolant temperature, the engine oil temperature, and the intake air temperature. For the Gen 2 GDCI engine, all of these temperatures are controlled by the thermal management system. The system provides variable intake air temperature control and maintains minimum gas temperature to avoid water condensate in the boost system. A schematic of the Gen 2 GDCI thermal management system is shown in Figure 9.

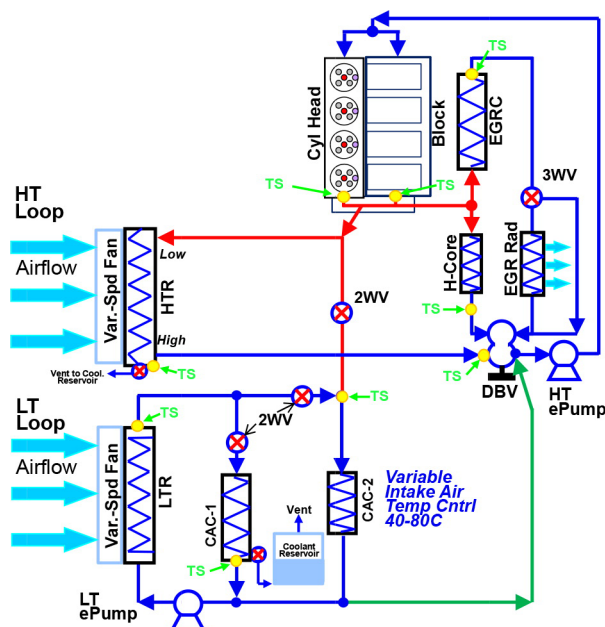


Figure 9. Schematic of Gen 2 GDCI thermal management system.

The system consists of two main sub-systems, a high-temperature (HT) loop which manages engine coolant flows, and a low-temperature (LT) loop which manages charge air cooling. The HT loop features a variable-speed electric pump and a dual-ball-valve (DBV) electronic thermostat. Pump speed control provides sufficient coolant flow at all engine operating conditions with minimum parasitic loss, while the DBV controls coolant flow distribution between cabin heater, EGR cooler and the high temperature radiator (HTR). This allows optimal control over the full range of conditions, including engine warmup and stop-start operation.

The two loops are connected through the blend valve. This valve is closed for most ordinary operating conditions. But during warmup after cold start, or long periods of idle, or light load operation at low ambient temperatures, the boost devices may not provide enough heating to maintain the desired intake air temperature. In these cases the blend valve controls delivery of hot engine coolant to CAC-2 over a short distance.

The LT loop features a low temperature radiator (LTR) and a variable-speed pump to provide intake air temperature control with minimum parasitic loss. The two charge air coolers (CAC-1 and CAC-2) are arranged in parallel for independent control of their outlet air temperatures using a coolant flow control valve at the inlet

of each CAC. In this way acceptable supercharger inlet temperature can be maintained while also avoiding water condensation at the CAC-1 outlet at high engine loads when EGR can reach 50%. The disadvantage of the parallel configuration (rather than series) is that air temperature control is more difficult because of interactions between the two coolant circuits.

Another important architecture option is the air flow configuration of the low temperature and high temperature radiators. The radiators may be configured side-by-side (parallel) or front-to-back (series). Side-by-side has the advantage that interaction between HTR and LTR systems are eliminated, which simplifies control. Simulation showed that: a) radiator capacity is the primary limitation on thermal control at peak engine loads, and b) the series airflow arrangement provides the highest heat rejection capacity taking into account packaging limitations on the vehicle. The series arrangement was therefore chosen with the LT radiator positioned ahead of the HT radiator with two independently controlled air side-by-side fans (note that Figure 9 shows the radiators side-by-side only for clarity in the schematic). Two disadvantages of the series radiator configuration are that heat rejection from the HTR is affected by that of the LTR, and changes in fan speed required by the HT system also affect the LT system.

A model-based control (MBC) strategy has been developed for the thermal management system with the aid of co-simulation. A detailed GT Suite model [41] of the complete thermal management system was developed from test data for the radiators, charge coolers, and coolant control valves. This system model was coupled to Matlab Simulink [42] for transient control simulation.

Figure 10 shows simulated response of engine intake temperature, IAT, control during a tip-in transient from a light load condition of 1500 RPM- 3 bar IMEP to a medium load condition of 2000 RPM- 11 bar IMEP, followed by a tip-out back to light load. A boost system response time of about one second gives a load application rate of 8 bar/s, which is common during a FTP test. The peak IAT excursion at the intake ports was +4/- 2 degrees from the target nominal value of 55 C, but the excursion was somewhat smaller at the CAC-2 exit. The difference is due to compression heating (or cooling during tip-out) of the gas downstream of the cooler. While these preliminary results show good control is feasible, further improvements in control are expected through the development process.

The GT-Suite model of the thermal management system was also used to evaluate thermal capacity of the thermal management system at full load. Two engine operating conditions were chosen, 2500 RPM- 24 bar IMEP and 4000 RPM- 18 bar IMEP. The first represents a peak torque condition, and the second peak power. Each condition was run at two vehicle speeds, 80 and 130 kph, for ambient air temperature of 25 C. The results are summarized in Figures 11 and 12. The solid and patterned bars are for speeds of 130 and 80 kph, respectively.

Three conclusions can be drawn from Figure 11. First note that in all cases air temperature out of CAC-1 is very close to coolant temperature at the LTR outlet. This implies that CAC-1 has more than adequate capacity and a large margin for higher ambient temperatures. Second, in all cases, the air temperature at CAC-2 outlet is only 5 C above coolant temperature at the LTR outlet. This represents very good cooler performance considering the rather severe packaging constraints for CAC-2. During engine dynamometer

tests on the Gen 1 engine, a peak IMEP of 24 bar was obtained using an intake air temperature of 40 C. This IAT is lower than simulation predictions for both vehicle speeds and quite low for a highly boosted engine. Therefore, the third conclusion is that engine output at engine speeds at or below 2500 RPM is limited primarily by inadequate LTR capacity, which results from vehicle packaging constraints.

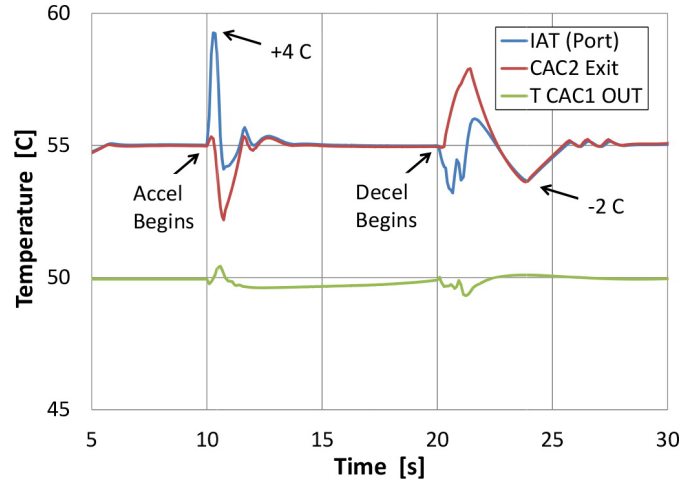


Figure 10. Simulated performance of the Gen 2 thermal management system for IAT control.

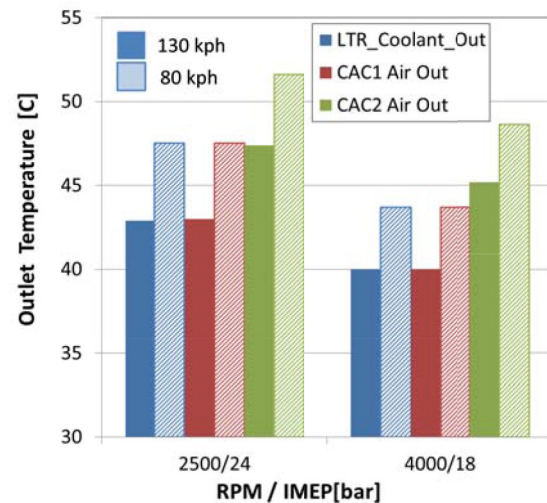


Figure 11. Simulated charge air cooling performance at 2500 RPM peak torque and 4000 RPM peak power.

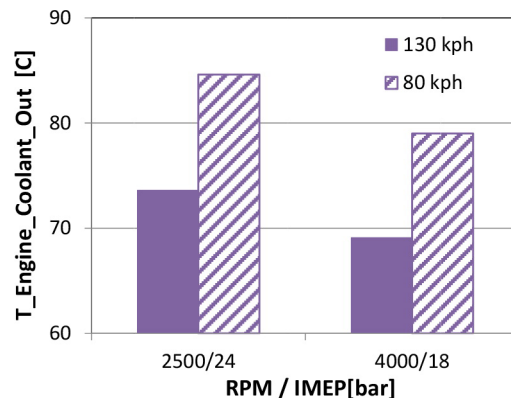


Figure 12. Simulated engine cooling performance at 2500 RPM peak torque and 4000 RPM peak power.