

Boost System Development for Gasoline Direct-Injection Compression-Ignition (GDCI)

Kevin S. Hoyer, Mark Sellnau, James Sinnamon and Harry Husted
Delphi Powertrain

ABSTRACT

Intake boosting is an important method to improve fuel economy of internal combustion engines. Engines can be down-sized, down-speeded, and up-loaded to reduce friction losses, parasitic losses, and pumping losses, and operate at speed-load conditions that are thermodynamically more efficient. Low-temperature combustion engines (LTE) also benefit from down-sizing, down-speeding, and up-loading, but these engines exhibit very low exhaust enthalpy to drive conventional turbochargers. This paper describes modeling, evaluation, and selection of an efficient boost system for a 1.8L four-cylinder Gasoline Direct-Injection Compression-Ignition (GDCI) engine.

After a preliminary concept selection phase the model was used to develop the boost system parameters to achieve full-load and part-load engine operation objectives. The simulation was used to demonstrate that a practical boost system can provide the boost necessary at reasonable brake efficiency levels over the entire engine operating range. A comprehensive simulation based calibration was performed to determine the most efficient steady operation settings. Also a step change in speed/load engine operation was simulated to demonstrate the system transient response.

CITATION: Hoyer, K., Sellnau, M., Sinnamon, J. and Husted, H., "Boost System Development for Gasoline Direct-Injection Compression-Ignition (GDCI)," *SAE Int. J. Engines* 6(2):2013, doi:10.4271/2013-01-0928.

INTRODUCTION

There is ever increasing pressure in the global auto industry to increase vehicle efficiency measured by either CO₂ emissions or fuel consumption. The market requires that improvements in efficiency do not compromise vehicle performance, increase cost excessively, or cause drivers undue burden or inconvenience. There have been proposed powertrains that require new fuel infrastructure development, such as natural gas, hydrogen, naphtha, etc. that will require more consensus and more time before being adopted. Some of the proposed high efficiency concepts are homogeneous charge compression ignition (HCCI), reactivity controlled compression ignition (RCCI), high exhaust gas recirculation dilute combustion (as being developed by the High Efficiency Dilute Gasoline Engine HEDGE consortium, e.g.), partially premixed compression ignition (PPCI), and GDCI. The GDCI concept being developed uses compression ignition under lean to near stoichiometric fueling conditions over the complete engine operating range. The GDCI concept has been shown capable of delivering clean, quiet, and efficient operation which is more completely described in the references [1, 2, 3, 4]

A similarity among these proposed high efficiency concepts is they use low temperature combustion. Low

temperature combustion has the advantage of permitting lean or dilute combustion which can be low in particulate generation and can avoid excessive NO_x pollutant formation. These high efficiency concepts derive some improvement in indicated efficiency from less heat losses due to the lower combustion temperatures [4]. Boosting has been shown to be necessary for these alternative combustion mode engines. Efficient boosting is critical in order for these concepts to succeed.

Boost systems have evolved significantly over the years. Design of turbine and compressor blades for turbochargers, inlet and outlet flow geometries and features such as variable geometry turbines (VGT) have led to more efficient devices [5,6]. Superchargers have also become more efficient as more optimization is integrated into the designs of drives, compressor geometry and flow paths [7].

New, specialized boost systems for high exhaust gas recirculation (EGR) engines are being developed. For example, Roth, et al, [8] described a turbo system that uses divided exhaust period to drive a turbocharger. The system improves fuel efficiency and provides for significantly increased boost. The system requires a special exhaust manifold and special valvetrain.

Recently it has been suggested [9] that a two-stage boost system is essential for fast response while maintaining rated power for diesel engines. A system with a supercharger and turbocharger was shown to provide improved steady-state, transient performance with improved fuel economy.

Other boost systems consisting of turbochargers and superchargers have been developed to provide low speed torque for downsized and down-speeded gasoline SI engines [10].

Gasoline SI engines are limited in the amount of boost that can be used because of preignition and knock. Turbochargers for gasoline SI engines may also be subjected to high exhaust temperatures, which can limit their operation, require higher cost materials, or require cooled exhaust manifolds. Boost system requirements for low temperature combustion engines are somewhat different than those for conventional SI and CI engines. High efficiency powertrains typically have reduced exhaust temperatures and require higher amounts of EGR. Chadwell, et al, [11] performed an analysis showing that both HCCI and RCCI required boost systems comprised of both a supercharger and a turbocharger and that these systems can result in a significant reduction in brake efficiency when properly accounted for.

Gasoline PPCI studies by Manente [12] showed the need for higher boost and higher EGR levels depending on operating conditions. This work did not address the specifics of efficiently delivering the necessary air but would likely require a two-stage system.

A study of boosting strategies for HCCI performed by Mamalis, et al, [14] concluded that a staged turbocharger system extends the load range at 2000 rev/m. The additional pumping work of the boost system reduced the brake efficiency, however. A single supercharger allowed the engine to achieve high indicated efficiency but was shown to have low brake efficiency due to the mechanical energy required to drive the supercharger.

Similar to other high efficiency concepts the GDCI engine has low exhaust temperatures. The GDCI concept being developed differs in key ways to the other high efficiency concepts. GDCI relies heavily on fuel injection strategy to ensure proper combustion. Since the injection strategies depend on the in-cylinder pressure, temperature, composition conditions the injection is coupled to the boost system. The fuel injection system is undergoing simultaneous development with the boost system, and for the purposes of this study it is assumed that the injection strategy will permit proper combustion provided the boost system can deliver the correct air and EGR mixture at the correct pressure and temperature. GDCI has provisions to use a late intake valve closing strategy when it is necessary to reduce the engine effective compression ratio at high loads. Using LIVC has an effect on the engine volumetric efficiency and air flow.

The objective of this paper is to demonstrate a practical and efficient boost system for a low-temperature 1.8L GDCI engine. Simulation tools are used to select the boost architecture, develop the selected system for full-load and

part-load requirements, and evaluate transient response. Finally, a methodology was developed for simulation-based calibration to determine the best settings for efficient operation and potential for low fuel consumption over the full speed-load map.

BOOST SYSTEM ARCHITECTURE

The engine being used in development is a 1.8L total displacement, 4-cylinder engine with four valves per cylinder. Simulations were performed using the multicylinder engine GT Power [13] model and supercharger and turbocharger steady state maps obtained from suppliers.

Flow from boost devices is inherently coupled to the temperature of the gases. This requires that the intake air temperature (IAT) system is developed in conjunction with the boost system. Coolers are modeled using flow data from representative liquid cooled charge air coolers (LCCAC) in use or nearly production ready. Air flow pressure drop across the LCCAC is based on flow measurements for actual coolers. The EGR cooler is based on a production design currently in use and a known steady flow restriction relationship and heat transfer characterization.

Each of the boost devices considered in this evaluation was modeled using GT power models of the engine, associated sub-systems and steady-state component performance maps of production-viable boost devices. The performance maps used in the models are based on actual flow test data. Since the boost system development occurs ahead of the multicylinder engine built it is not possible to validate this model. Where appropriate, single cylinder engine test data is used to support model development. Heat release curves are modeled based on single cylinder test results in addition to matching the cylinder engine indicated specific fuel consumption values. The systems evaluated ranged from a single fixed-geometry device, to a subsystem of variable geometry compound devices. The ability of each proposed system to provide maximum steady-state performance was considered at engine operation of 50% EGR and the appropriate valvetrain IVC at each speed.

Initial performance targets were established for the GDCI engine. A full load brake mean effective pressure (BMEP) expectation was established. The engine is expected to have a top speed of 4500 rev/m. The boost system is not permitted to exceed an intake manifold absolute pressure of 4 bar.

Preliminary simulations were performed with various alternative boost systems considered. The following is a summary of a few of the systems mechanisms and performance results when modeled together with the development engine.

Supercharger System

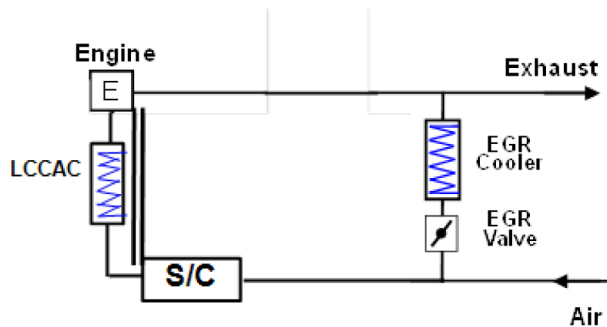


Figure 1. Mechanization of single supercharger with single liquid charge air cooler (LCCAC) at supercharger outlet to control IAT.

The simplest system to provide boost is a single supercharger driven from engine power. Many types of devices have been developed, mostly for diesel and SI applications. Available devices range from centrifugal compressor designs to screw-type and roots type displacement compressors which can be electrically or mechanically driven. A mechanically driven roots type supercharger device is modeled as shown in Figure 1 with simulated full load performance results shown in Figure 2. This mechanization is appealing because it is the simplest to implement with the least impact on exhaust temperatures. The power to operate the supercharger is supplied from the engine crankshaft and the speed of the supercharger depends on the mechanical drive ratio. There is a single liquid cooled cooler at the supercharger exit maintaining the IAT at the desired temperature. A range of supercharger devices were modeled ranging from designs for smaller through larger engines (<1.0L up to 2.5L total displacement). The full load (near stoichiometric fueling, at 50% external EGR) manifold pressure and engine performance resulting from operating a supercharger over a range of drive ratios from 3 to 4.6 times the engine crank speed is plotted in Figure 2. The MAP available from the device shown is limited at about 2 bar absolute which only results in a gross indicated mean effective pressure (IMEP) around 11 bar at the highest simulated 4.6 drive ratio. This shows relatively low BMEP at high speed where the supercharger input power requirements are high. A device could provide higher than 2 bar absolute MAP, which would result in a higher IMEP value but would also be expected to have even higher input power requirements and thus too low BMEP. BMEP is improved at the low range of engine speed, but the resulting BMEP does not meet requirements for full load BMEP approximately 20 bar for the 1.8L GDCI engine.

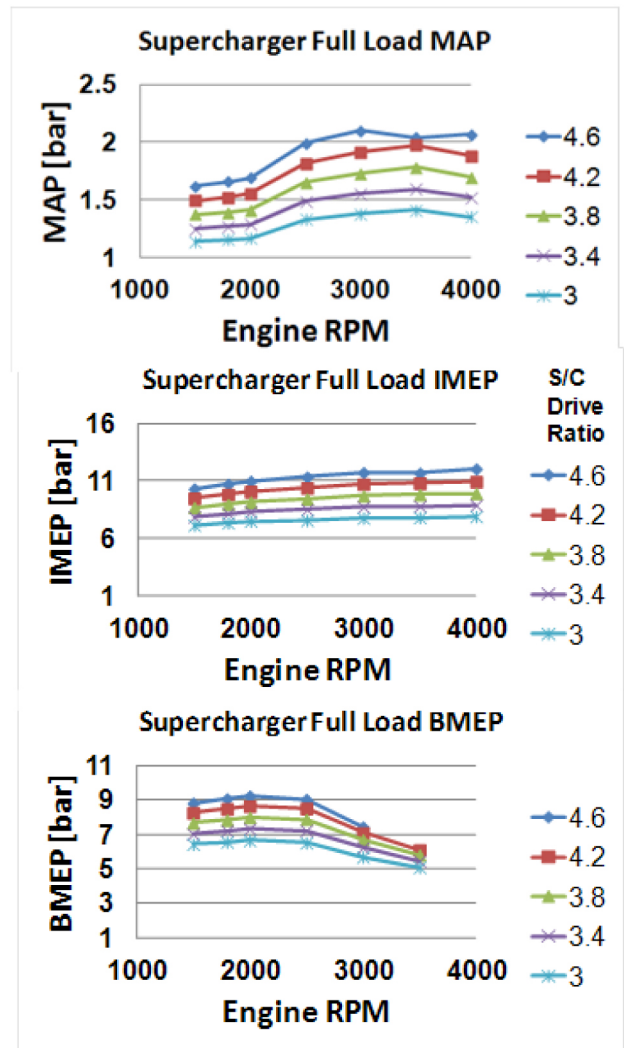


Figure 2. Full load simulation of single supercharger at various drive ratios. Individual curves indicate the fixed drive ratio that is simulated as a multiple of engine speed.

Turbocharger System

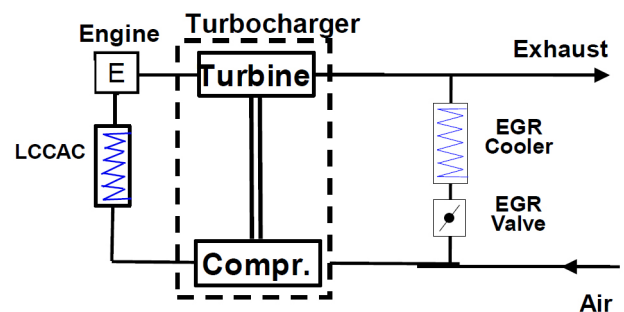


Figure 3. Mechanization of single turbocharger with single liquid charge air cooler (LCCAC) at turbo compressor outlet to control IAT.

A single turbocharger mechanization driven by exhaust gas is shown in [Figure 3](#) with a LCCAC at the turbo compressor outlet to control IAT. Various types and sizes of turbo devices have been developed for light-duty diesel applications and a range of sizes has been considered in this study. A variable turbine geometry with a fixed geometry compressor is modeled here. An external EGR path is shown with a cooler. In this and all of the mechanizations shown this cooled low pressure EGR system is shown after the turbocharger devices. The GDCI engine is capable of rebreathing the exhaust gases with a variable exhaust valve strategy when high temperature EGR is required to promote autoignition. Downstream of the EGR flow split is an oxidation catalyst, assumed to be the only aftertreatment device required for the GDCI engine. An exhaust silencer is included, modeled from the one used on the production vehicle. There is also a backpressure valve simulated to increase pressure at the EGR cooler inlet when needed to meet the cooled EGR flow requirements. These devices are present in all mechanizations but not shown in the figures.

Simulation results of intake manifold pressures and engine performance for full load are shown in [Figure 4](#). These points are at 50% external EGR rate and near stoichiometric fuel levels.

It is obvious from these results the low speed performance of the turbocharger device is not adequate in this configuration. With the rack in the 0 or fully closed position there is an appreciable amount of MAP generated, however the IMEP and BMEP full load values are well below performance targets for this engine.

The amount of heat rejected to the exhaust affects the turbocharger performance. [Figure 5](#) shows that different amounts of in-cylinder heat transfer causes exhaust temperature variations over the speed range. In-cylinder heat transfer variations can result from a variety of sources in the GDCI engine, including gas mixture motion and piston surface temperatures that are not entirely known. A reduction of about 50 K in exhaust temperature can result in a MAP decrease of about 0.5 bar for the device shown at the same rack setting. The GDCI combustion process is still being developed, so it is not certain exactly how much heat will be available for the turbo devices to extract. This plot indicates the amount of uncertainty in the model in being able to predict the temperature and resulting boost generated.

GDCI also has provisions to use a late intake valve closing (LIVC) strategy when it is necessary to reduce the engine effective compression ratio at high loads. Use of LIVC has an effect on the engine volumetric efficiency and air flow. [Figure 6](#) shows that when the intake valve closing is retarded by 40 crank degrees (effective compression ratio reduction from 13.5 to 10.2) the resulting reduction in airflow is around 25 percent which can result in about 0.4 bar lower MAP produced from the turbocharger at the same rack position.

The amount of EGR also has an effect on the turbocharger performance. [Figure 7](#) shows that reduction of 20% in EGR

gases can make over a 1 bar difference in the MAP for a given turbocharger. This is because the EGR used must flow through the turbocharger compressor the way the low pressure EGR system is designed. Although cooled, the EGR is hotter than the ambient air temperature and the compressor flows lower density gas at the inlet as the EGR increases.

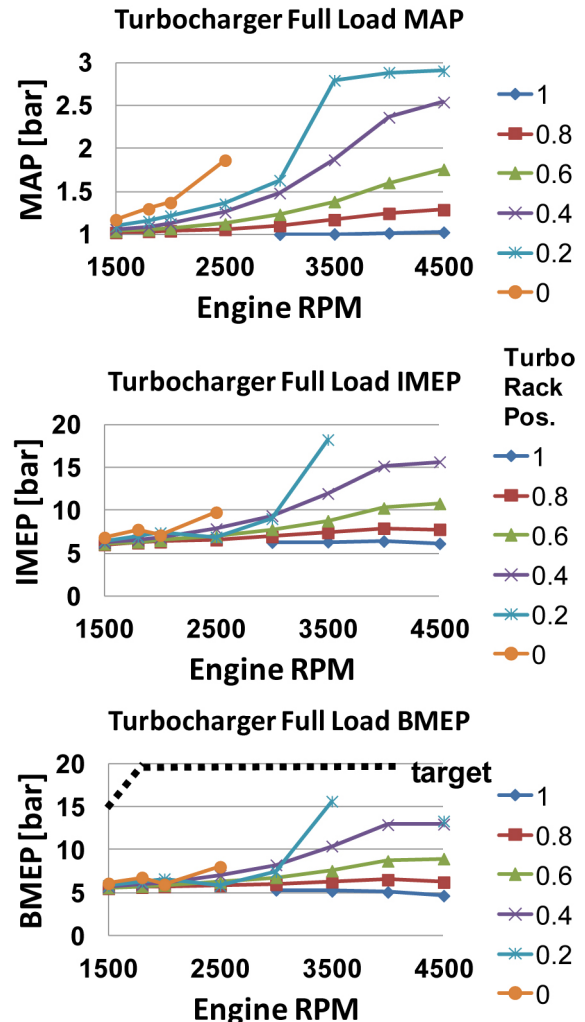


Figure 4. Full load simulation of single-stage turbocharger with various turbine geometry settings. Individual curves show results with turbine rack simulated at fixed position ranging from 1 (fully open) to 0 (closed.)

Sensitivities to the exhaust temperature, LIVC, and EGR demonstrate that typical turbocharger devices cannot simply be expected to be a good match for this GDCI application. Many combinations of superchargers and turbochargers were considered. Components that could deliver higher pressure ratios at lower flow rates were compared to higher flowing devices. [Figure 8](#) shows that devices that can deliver more flow at higher pressure ratios at the low end of the flow range typically suffer from limited mass flow capacity. This shows