

# A Demonstration of Dedicated EGR on a 2.0 L GDI Engine

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## ABSTRACT

Southwest Research Institute (SwRI) converted a 2012 Buick Regal GS to use an engine with Dedicated EGR™ (D-EGR™). D-EGR is an engine concept that uses fuel reforming and high levels of recirculated exhaust gas (EGR) to achieve very high levels of thermal efficiency [1]. To accomplish reformation of the gasoline in a cost-effective, energy efficient manner, a dedicated cylinder is used for both the production of EGR and reformat. By operating the engine in this manner, many of the sources of losses from traditional reforming technology are eliminated and the engine can take full advantage of the benefits of reformat. The engine in the vehicle was modified to add the following components: the dedicated EGR loop, an additional injector for delivering extra fuel for reformation, a modified boost system that included a supercharger, high energy dual coil offset (DCO) ignition and other actuators used to enable the control of D-EGR combustion. In addition, the compression ratio of the engine was increased to 11.7:1 to take advantage of the improved knock resistance from reformat and EGR. The engine conversion and the development of the control system for the engine are the subject of this paper.

The conversion to D-EGR configuration resulted in an improvement in engine efficiency of at least 10% across the performance map, with some operating conditions seeing substantially higher improvements. For example, the brake specific fuel consumption (BSFC) at 2000 rpm 2 bar BMEP improved from 385 g/kW-h in the series production state to 330 g/kW-h and the lowest BSFC for the engine was 212 g/kWh compared to 236 g/kW-h for the series engine. The addition of 2-stage boosting also allowed the engine to meet its torque targets of at least 17 bar BMEP from 1500-5500 rpm while maintaining good transient response and low engine-out emissions.

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## INTRODUCTION

Concern over energy independence and climate change has led to significant increases in the regulated efficiency of light duty vehicles. In the US, a fleet-average target of 54.5 MPG has been established for MY 2025 vehicles while the European Union has set 95 g/km CO<sub>2</sub> as a requirement for MY 2020. These regulated fuel economy targets, combined with increased consumer demand for high-efficiency vehicles, has led to an intensified focus on reducing light duty vehicle fuel consumption. Competing alternatives including hybridization, increased incorporation of compression ignition (CI) engines in the vehicle fleet, among others, have been offered as potential replacements / supplements for the traditional spark ignited (SI) power plant. However, particularly in the US and other low criteria pollutant regimes, each alternative has significant drawbacks in either their ability to meet modern emissions criteria (e.g. diesel engines) or in customer acceptance due to cost or other concerns (e.g. HEVs and EVs). For the foreseeable future, the traditional SI engine will be the choice

for the primary powerplant of most mass production engines. Therefore, significant improvement in the efficiency of the spark ignition engine is required.

Compared to CI engines, SI engines have lower efficiency at both low and high loads, despite the inherent efficiency advantage of the Otto-cycle over either the Diesel-cycle or combined cycle [2]. At traditional drive-cycle conditions, the SI engine is significantly less efficient than a diesel engine. This difference is due primarily to the gasoline engine's lower compression ratio and higher pumping losses, with pumping losses being the greatest difference at these points. Most of the pumping losses can be traced to the use of a three-way catalyst for emissions compliance. This aftertreatment system is a robust, relatively inexpensive and proven technology to produce ultra-low emissions. However, the three-way catalyst requires a stoichiometric exhaust stream to operate properly, which fixes the engine's air/fuel ratio (AFR). To control load, the engine must therefore be throttled, which results in the aforementioned pumping work. Even if the engine could run lean, the lean limit of gasoline is such that pumping losses

would still be incurred at the lightest loads, unless alternative forms of combustion, such as HCCI or stratified charge, are used [3].

Given the reality of pumping losses in stoichiometric gasoline engines at low loads, a common proposed solution is to reduce engine displacement and use a boosting device to increase the engine's specific power output. Since the vehicle power requirements will not change, the reduced engine displacement will lead to a shift in the drive-cycle engine operating range to higher levels of specific power output, greatly reducing the need to operate the engine at the lowest loads [4]. In addition, shrinking the engine size and operating at higher specific power levels can reduce the relative penalty of frictional losses when the change in engine displacement is accompanied by a shift in operating regime to slower engine speeds.

Pumping losses, however, are not the only significant impediment to high-efficiency gasoline engines. Knock and the measures required to overcome it also reduce efficiency. The first step in addressing knock in a boosted engine is to retard spark timing. Since knock is strongly influenced by the temperature history of the charge, retarding spark timing can reduce knock by lowering the end-gas temperatures and increasing the ignition delay of the end gas during the combustion event. High levels of spark retard are often employed in modern, boosted engines. At low engine speeds, where the challenges from knock are the highest, spark timing can often be retarded by 30° or more from the optimum condition. As engine speed increases, the need for extreme levels of spark retard is reduced but not, however, eliminated. Spark retard has several negative effects on efficiency, including increased specific fuel consumption and higher exhaust gas temperatures. At high speeds (i.e. greater than 2000 rpm) the exhaust temperatures can exceed 900° C, which will require the use of fuel dilution to cool the exhaust.

While spark retard is effective in reducing the knock tendency of the engine, there is a limit to the amount of spark retard that can be employed before the engine becomes unstable. If the specific power requirements of the engine are higher than the load at which the engine can be stably operated, then the only option remaining is to reduce compression ratio. As the downsizing and boosting trend has taken hold, the average compression ratio of a production gasoline engine has fallen significantly. Even the most advanced gasoline engines reduce the compression ratio by at least 1.5 when going from a naturally aspirated (NA) to boosted application [5]. When discussing future powertrains and the requirement for high efficiency, it is worth remarking that, on a fundamental level, the only two factors that influence the ultimate efficiency of an engine operating on the Otto-cycle are the compression ratio and the ratio of specific heats of the working fluid ( $\gamma$ ) [6]. While it is possible to reduce compression ratio and achieve adequate fuel consumption results at some mid-to-high load conditions [7], the fuel consumption at low to part load in these

engines is often compromised from their high compression ratio, NA counterparts. In addition the downsized engine will often encounter reduced efficiencies at high loads [8], due to some combination of knock-related spark retard and, at high engine speeds, the enrichment that accompanies retarded spark timing. Enrichment may also be necessary even at optimum combustion phasing to improve the durability of engine hardware at high specific power levels. However, over-fuelling at high power levels prevents aggressive downsizing by making it difficult or impossible to meet modern emissions requirements and significantly increasing specific fuel consumption. Downsizing in this manner may offer the potential to reduce CO<sub>2</sub> emissions, but low fuel consumption at higher loads is necessary to fully implement a downsizing strategy.

In previous studies of boosted engines, the presence of some EGR has been shown to decrease exhaust temperatures [9, 10, 11] and reduce or eliminate the requirement for over-fuelling [9, 11-12]. In addition, higher levels of EGR (EGR rates > 15%) have been shown to have a significant effect on efficiency by reducing knock, leading to improved combustion phasing [9, 10, 11, 12, 13] and the ability to operate at high loads at compression ratios above 12:1 [14-15]. In addition, elevated EGR levels serve to substantially reduce the emissions of CO [6, 11], NO<sub>x</sub> [6, 9, 11] and particulates (both number and mass) [16-17] through a reduction in combustion temperatures. While the sources of improvement in high dilution engines continue to improve the fuel economy potential as the EGR rate increases, a combination of real-world challenges limits the ultimate level of improvement. The combination of a reduction in the burn rate and high exhaust emissions, caused by unstable operation and low post-flame temperatures, reduces the brake thermal efficiency improvement at EGR rates above 20-30% in most engines [18]. Thus, a major challenge for high dilution engines is to improve their dilution tolerance, through enhanced ignition systems [19, 20, 21, 22] or by supplementing the charge with high reactivity compounds, either pure H<sub>2</sub> [23-24] or reformed fuel, consisting of CO and H<sub>2</sub> in some proportion [25, 26, 27, 28, 29]. The addition of these highly reactive compounds results in significantly faster flame propagation rates, improved engine stability at high dilution levels and better engine performance, e.g. [23, 24, 25, 29, 30, 31].

In previous publications [1, 32], the concept of using one (or more) cylinders dedicated to EGR production was proposed as a method of reducing some of the demerits involved with on-vehicle reformation of hydrocarbon fuels (discussed in [1]). In these applications, however, the work was limited to initial proof of concept testing. The results indicated that, while the concept has some promise as a high efficiency architecture, there are some challenges involved with using a dedicated cylinder [1] and in full-time operation at very high EGR rates [32]. In addition, previous work on high dilution applications indicated that there are considerable challenges in designing a

boosting system that will provide the correct combination of low speed torque, specific power (in terms of kW/L) and transient response for a high dilution engine [33]. The purpose of this study was to address these concerns by designing an engine system that achieves adequate torque, power, transient response and emissions potential while using the Dedicated EGR (D-EGR) concept to improve efficiency across the operating map, including the design of an air handling system that minimizes the issue of EGR pulsations and distribution.

## DEDICATED EGR CONCEPT

An embodiment of the D-EGR concept introduced in the previous section is shown in the schematic in [Figure 1](#). In this schematic one of the cylinders is used to produce the entirety of the EGR used by the engine. The D-EGR cylinder is labeled as such in the schematic. In this case, one cylinder is producing the EGR for all four cylinders of the engine, thus the engine will run with nominally 25% EGR. A feature of the D-EGR concept is that the EGR rate is not proportionately controlled; the engine will continuously run with a nominal 25% EGR rate making control less complex.

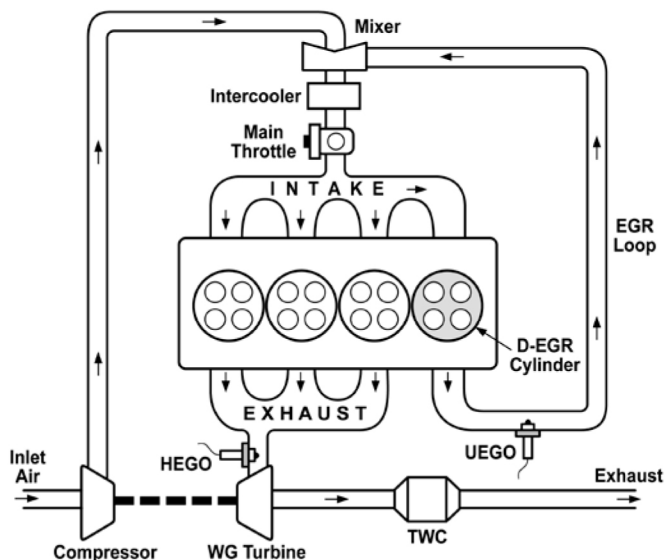


Figure 1. Schematic of a turbocharged 4-cylinder engine in a D-EGR configuration

In addition to the knock tolerance improvement afforded by EGR, the D-EGR concept provided additional knock resistance when in-cylinder reformation was used in the D-EGR cylinder. In-cylinder reformation was realized by running the D-EGR cylinder rich of stoichiometric. A typical operational target was to run an equivalence ratio between 1.3 and 1.4 in the dedicated cylinder. Rich combustion produced reformat consisting of carbon monoxide (CO) and hydrogen ( $H_2$ ), that have Research Octane Numbers (RON) of 106 and 130 respectively. When the D-EGR cylinder was run with an equivalence ratio between 1.3 and 1.4 the knock resistance was similar to an increase in RON of the fuel of approximately

three points based upon a calculation of the mass averaged octane of gasoline (97 RON), CO and  $H_2$ . EGR and reformat have a compounding influence on knock mitigation and therefore engine efficiency.

## APPROACH

The approach for this work was to convert a current production engine and vehicle to a D-EGR configuration. The target vehicle was required to be comparable to vehicles that sell in high volumes in North America. The process started by identifying a modern production sedan with a curb weight of at least 1500 kg that was available in both NA and turbocharged versions. The NA version of the vehicle was to be used as the baseline while the turbocharged version would be the most easily converted to D-EGR. The turbocharged D-EGR version of the converted vehicle will be compared to the baseline NA vehicle for calculation of efficiency improvement.

The target vehicle for this work was a 2012 Buick Regal. The Regal was available with a conventional naturally aspirated SIDI 2.4 L 4-cylinder engine (LEA), and with a turbocharged SIDI 2.0 L 4-cylinder engine (LHU). The LEA NA engine is found in the most efficient version of the vehicle rated by the EPA at 19 MPG city and 31 MPG highway. The LHU engine is found in the higher performance Regal GS and is rated by the EPA at 19 MPG city and 27 MPG highway.

An LHU engine, controller and wiring harness were acquired for the execution of this project. The engine and control system were developed in a dynamometer test cell and were subsequently moved into the vehicle. The dynamometer development is the subject of this paper.

## EXPERIMENTAL SETUP

The engine used for this project was a 2012 LHU produced by GM. An image of the engine is shown in [Figure 2](#). The base engine specifications (before conversion) are shown in [Table 1](#).

Table 1. 2012 LHU engine specifications

|                   |                                   |
|-------------------|-----------------------------------|
| Engine Name       | LHU                               |
| Displaced volume  | 1996 cc                           |
| Stroke            | 86 mm                             |
| Bore              | 86 mm                             |
| Connecting Rod    | 152.5 mm                          |
| Compression ratio | 9.3:1                             |
| Number of Valves  | 4                                 |
| Actuation         | Dual independent cam phasers      |
| Induction         | Twin-Scroll Turbocharged          |
| Maximum Torque    | 400 Nm @ 2400 rpm (25.1 bar BMEP) |
| Maximum Power     | 200 kW                            |



Figure 2. GM LHU 2.0 L turbocharged DISI engine

The engine was installed in a test cell at SwRI. The engine was instrumented with typical engine instrumentation including engine speed, torque, fuel flow, air flow, and various temperature and pressure measurements. In-cylinder indication was performed with Kistler 6052C pressure transducers and a SwRI-developed high speed data acquisition system.

## D-EGR CONVERSION

While the LHU was chosen as the engine to convert, the performance of the converted engine was required to meet or exceed the performance of the LEA that was used as the baseline for fuel economy. The 2.0 L LHU chosen as the conversion engine is 16.7% downsized from the 2.4 L LEA engine, and as such, requires boost to reach the same torque. The LEA engine produces 233 Nm of torque at 4900 rpm and 136 kW at 6700 rpm. The design target for the D-EGR engines was chosen to be 270 Nm from 1500 rpm to 5500 rpm with a maximum power 155 kW at 5500 rpm. The torque and power curves are shown in Figure 3. Note that 270 Nm is 17 bar brake mean effective pressure (BMEP) for the LHU engine.

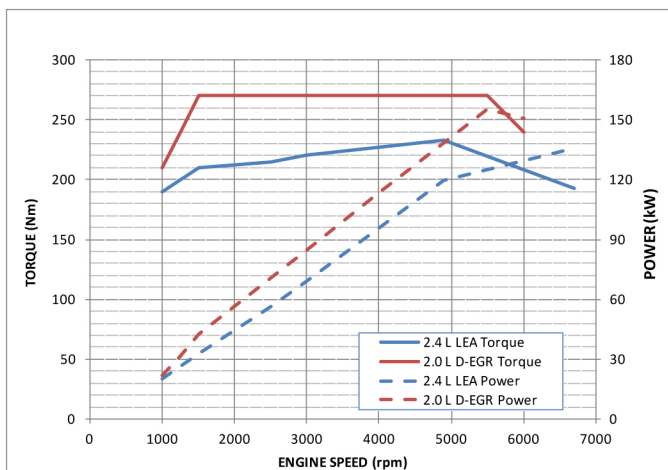


Figure 3. Torque and power curve from the production LEA engine and the design targets for the D-EGR converted LHU engine

It can be seen from Figure 3 that the D-EGR engine design targets are in excess of the LEA engine for torque and power. The rationale for this was to allow down-speeding of the engine where the vehicle could operate with a taller (numerically lower) overall gear ratio for additional fuel economy benefits. To successfully downspeed, the engine must produce elevated torque at lower engine speeds with good transient response and load acceptance. Vehicle modeling was used to support the target selection; however, the vehicle modeling will not be discussed in this paper.

## Air Handling System

The air handling system design consisted of several subsystems: boosting, EGR control and EGR mixing. A schematic of the full air handling system is shown in Figure 4. Each of the subsystems will be discussed separately.

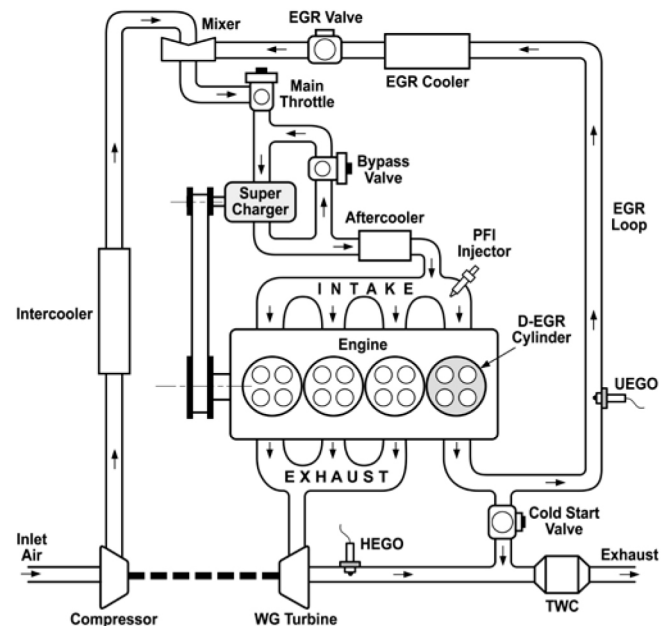


Figure 4. Schematic of the D-EGR engine air handling system

## Boosting System

Boosting the inlet air of an engine is required to reach 17 bar BMEP. Typically boosting is done with an exhaust-driven turbocharger or an engine-driven supercharger. The majority of boosted engines on the market, including the LHU engine, are turbocharged. The choice of turbocharger size is usually a compromise between the ability to run high BMEP at low engine speeds and the ability to run high BMEP at high engine speeds; smaller compressors and turbines perform well at lower flow rates, or lower engine speeds, while larger compressors and turbines perform well at higher flow rates. The addition of EGR dilution to an engine increases the required manifold pressure and compressor pressure ratio for a given load such that the compressor must operate in the higher pressure ratio area of the compressor map [33]. The flow rate range, which is the area between compressor surge and

compressor choke, of the compressor is reduced at elevated pressure ratios which makes boost system matching more challenging.

Gamma Technologies' GT-Power, a commercial 1-D CFD code, was used to select components of the air handling system. The code was used to choose which types of components would be used and the flow rate (size) of each of the components. The allowable devices were limited to production parts or readily available prototypes due to the short duration of the project.

Initial simulation results showed that when a turbocharger was selected to meet the power requirements of the engine, the low speed torque was lower than the design target. Large amounts of overlap could be used to improve the volumetric efficiency, and therefore the BMEP, of the engine, however, the BSFC of the engine at over-scavenged conditions was poor. An example of this is shown in the Steady-State Results section of this paper.

For the D-EGR engine conversion, the choice was made to use a supercharger and turbocharger. The supercharger provided the required boost to meet the low speed BMEP target. The required pressure ratio from the supercharger declined with increasing engine speed until the turbocharger could provide all of the required boost, at which time the supercharger was no longer needed. The supercharger had a clutch installed so that it could be disabled upon reaching said condition, or during light load operation where no boost was required. The supercharger was placed after the turbocharger compressor in the flow path such that the turbocharger provides boosted air to the inlet of the supercharger. The combination of the supercharger and turbocharger was the most conservative option to ensure breadth of high BMEP operation, optimum BSFC and transient response. An additional benefit of the turbocharger-supercharger configuration with D-EGR was that the camshaft phasers could largely stay in the same position. Internal EGR from overlap was not required due to the high concentration of external EGR, and scavenging to produce high BMEP at low engine speeds was not needed due to the supercharger.

The selected turbocharger was a BorgWarner 2074. The turbocharger is used on the production 1.6 L Prince engine used by PSA and BMW/Mini. The turbocharger was smaller than would have been preferred but was immediately available from the local BMW/Mini dealership. An electronic wastegate actuator was added to the turbocharger so that the wastegate could be opened at engine conditions without boost.

The selected supercharger was an Eaton V400 axial outlet Roots-style supercharger. The axial outlet was selected since the supercharger outlet was planned to be a pipe rather than exhausting directly into an intake manifold. Another consideration in the selection of the V400 was the area of peak

efficiency, which is located at low flow rates and low pressure ratio. The location was ideal for a turbocharger-supercharger configuration where the supercharger pressure ratio is highest at low engine speeds and tapers as the turbocharger produces boost at high engine speeds. The pulley ratio was chosen to be 4.2:1, or 4.2 revolutions of the supercharger for every revolution of the engine. The pulley ratio was quite low which suggested that the supercharger may be oversized; however, the larger supercharger allowed more flexibility with pulley ratios to adjust for transient response should it be a problem when the engine is eventually installed in the vehicle.

The GT-Power results from the final configuration are shown in the following figures: [Figure 5](#) shows the predicted engine performance, [Figure 6](#) shows the compressor map with operating points, and [Figure 7](#) shows the supercharger map with operating points.

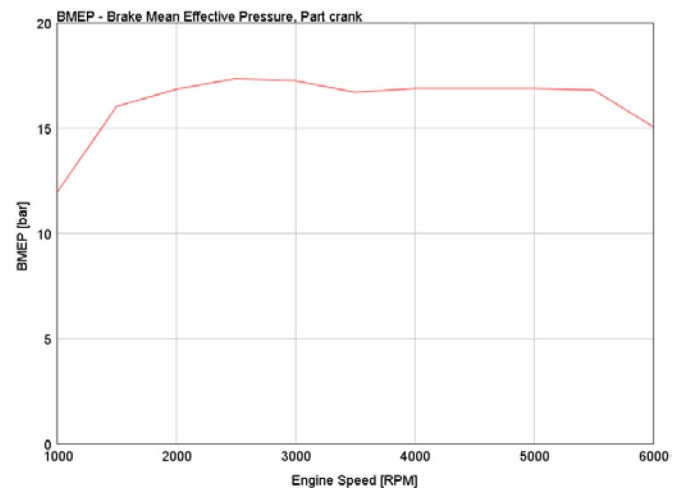


Figure 5. GT-Power predicted engine performance

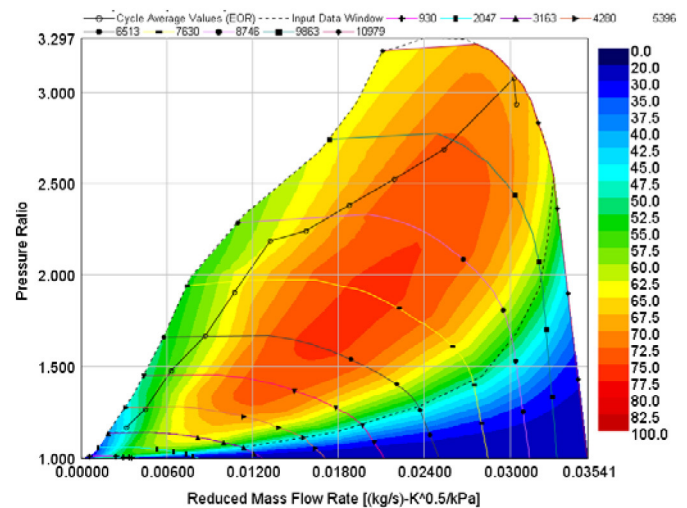


Figure 6. GT-Power predicted compressor operating points shown on an efficiency map for the BorgWarner 2074 turbocharger