

LPL EGR and D-EGR® Engine Concept Comparison Part 1: Part Load Operation

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ABSTRACT

The ongoing pursuit of improved engine efficiency and emissions are driving gasoline low-pressure loop EGR systems into production around the globe. To minimize inevitable downsides of cooled EGR while maintaining its advantages, the Dedicated EGR (D-EGR®) engine was developed. The core of the D-EGR engine development focused on a unique concept that combines the efficiency improvements associated with recirculated exhaust gas and the efficiency improvements associated with fuel reformation. To outline the differences of the new engine concept with a conventional low-pressure loop (LPL) EGR setup, a turbocharged 2.0 L PFI engine was modified to operate in both modes and also compared to the baseline. The first part of the cooled EGR engine concept comparison investigates efficiency, emissions, combustion stability, and robustness at throttled part load conditions. In addition, the LPL EGR configuration was supplemented with bottled H_2 / CO to examine if the D-EGR engine performance can be matched. The results show the D-EGR engine can tolerate higher EGR dilution levels, enable faster burn rates, reduce cycle-to-cycle variations and can further improve efficiency and emissions. Additionally, the D-EGR technology decreases the ignition energy requirements. Moreover, by adding the bottled reformat to the LPL EGR engine, the performance of the D-EGR setup can be simulated.

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INTRODUCTION

The transportation industry is facing significant challenges from the combination of high fuel prices and increasing regulatory pressure caused by the concern over climate change and energy security. By 2025, the automotive industry will be required to have reduced CO_2 emissions by at least 30% and pollutant emissions by a factor of 3 [1]. The continuously more stringent emissions and efficiency legislations are forcing manufacturers and researchers alike to develop new engine concepts and improve current systems. Current technology trends for high efficiency gasoline engines include, but are not limited to, direct injection, downsizing in conjunction with boosting, down-speeding, and hybridization. Alternative routes involve the use of dilution in the form of fresh air or cooled EGR. Lean burn concepts offer great efficiency, however, are challenging with their high NO_x emissions, and often times costly hardware is required to comply with modern emissions criteria. Cooled EGR concepts are becoming more prevalent as is evidenced by several production applications for high efficiency SI engines when maximum power output is not paramount [2, 3, 4, 5, 6]. Efficiency improvements with cooled EGR can be realized by increasing the indicated cycle efficiency and reducing heat transfer and throttling losses. EGR also improves the knock resistance and enables high compression ratio operation and / or an extended operating range with optimal combustion phasing. Heat transfer and knock resistance are improved since cooled EGR has a high specific heat capacity and acts as a diluent in the combustion process. As a result, in-cylinder gas temperatures can be lowered. EGR dilution further improves efficiency by increasing the ratio of specific heats during the cycle. In addition to

improving the engine efficiency, the use of high levels of EGR reduces NO_x , CO, PM, and PN emissions [7, 8, 9]. With relatively minor engine modifications, allowing the utilization of cooled EGR, real-world and NEDC fuel consumption improvements of >5 % can be achieved while reducing emissions by an even larger margin.

The challenges of cooled EGR include misfire, combustion stability, EGR flow control, and engine component durability [10]. EGR dilution decreases burn velocities which can lead to unstable engine operation or even full misfire. Therefore, it is imperative to employ advanced, high energy ignition systems which mitigate stability concerns. Furthermore, in a naturally aspirated (NA) engine or when the boosting system is not altered, EGR also reduces the volumetric efficiency. The reduction in volumetric efficiency, which is also the source of the pumping work improvement, will result in lower specific power if EGR is used at WOT conditions. In addition, EGR tends to increase the HC emissions due to earlier flame quenching and lower post-oxidation temperatures. To address most of the inherent shortcomings of cooled EGR while maintaining or exceeding the efficiency benefits, the D-EGR engine concept was invented.

DEDICATED EGR

The D-EGR engine is an engine configuration in which the exhaust of a sub-group of power cylinders is routed back to the intake of all the cylinders, creating EGR for all cylinders with the exhaust of a few. In a four cylinder application, the exhaust of one cylinder will be used

as EGR for the entire engine, resulting nominally in full time 25% EGR. This minimizes the need for active EGR control across the speed load range. By having dedicated EGR, the D-EGR cylinder(s) can operate at any equivalence ratio since their exhaust will never exit the engine before passing through another cylinder operating at a stoichiometric air fuel ratio (AFR). This offers options to vary the gaseous composition of the EGR. In particular, it allows the D-EGR cylinder to run rich and produce hydrogen (H_2) and carbon monoxide (CO) through in-cylinder fuel reformation. The reformation process recovers some of the waste heat, normally rejected to the atmosphere while still generating power in the dedicated cylinder(s). The presence of reformat enhances the flame speeds, combustion, and knock tolerance of all the cylinders [11, 12, 13] and can hence overcome combustion limitations that have historically limited SI engine efficiency at high EGR dilution levels. Since only stoichiometric exhaust leaves the engine, a conventional three-way catalyst can be utilized to reduce pollutant emission. The net result of this engine architecture has demonstrated very high engine efficiency with a low associated cost. Moreover, the D-EGR engine is capable of operating on a variety of alternative fuels, including ethanol and natural gas, and has shown further efficiency and CO_2 improvements by doing so.

The primary objective of this study was to investigate potential differences between the LPL EGR and D-EGR engine configurations at part load conditions and at different compression ratios. In particular, combustion stability, efficiency, burn rates, and ignition energy requirements were examined at selected operating conditions.

EXPERIMENTAL SETUP

This study has been carried out on a turbocharged, inline four cylinder 2.0 L engine with port fuel injection. The engine was equipped with variable camshaft phasers on the intake side. In its baseline configuration, peak torque of 300 N-m was reached starting at 2250 rpm and maximum power delivery was 180 hp at 5500 rpm. One measure of the engine modifications for this work included the replacement of the production turbo charger with a dual-scroll unit designed for a 1.6 L engine to account for EGR dilution. Additionally, a Dual Coil Offset (DCO[®]) high ignition energy system was employed. The DCO system is a continuous discharge ignition system and can operate in single strike, multi strike, or DCO mode. The level of primary ignition energy, re-strike offset, and amount of re-strikes can be adjusted. A more detailed description can be found in previous publications [14, 15]. Unless otherwise noted, the ignition system was always operated in DCO mode and the ignition energy was controlled to ~ 400 mJ and 5 DCO re-strikes. To take full advantage of the improved knock resistance with high levels of EGR, tests were conducted with various compression ratios. The stock compression ratio of 9.1:1 was increased to 12.5:1 and eventually to 13.5:1. Pictures of the different pistons are depicted in Figure 1. To achieve the higher CR, the piston crown features were significantly modified compared to the 9:1 pistons. The surface area increased and the crevice regions were more pronounced. Both high CR pistons were provided by the OEM. The 13.5:1 CR pistons used the identical piston design as the lower 12.5:1 CR pistons. The only difference was that less of the piston crown was machined.



Figure 1. Tested pistons in 2.0 L D-EGR engine

Table 1 gives a description of the relevant physical parameters of the engine in stock form. To enable a direct comparison between the LPL EGR and D-EGR setups, both configurations used the identical engine long block, boosting, and fueling hardware as well as ignition system. The main discrepancies can be found in the intake and exhaust plumbing.

Table 1. Base engine specification

Displaced volume	1998 cc
Stroke	93 mm
Bore	82.7 mm
Connecting Rod	144 mm
Compression ratio	9.1:1
Number of cylinders / valves	4 in line / 16
Maximum power	180 hp @ 5500 rpm
Maximum torque	300 Nm @ 2250 rpm
Injection	Multipoint sequential

A schematic of the LPL EGR configuration can be seen in Figure 2. The expanded EGR was drawn post turbine, cooled via an EGR cooler, and introduced to the fresh air mass flow by a Venturi mixer. The Venturi mixer was developed for EGR control simplicity. When installed in a LPL EGR system, the Venturi mixer simply eliminates the need for sophisticated and expensive EGR control mechanisms. At low engine speeds and loads, where EGR tolerance is usually inferior, the Venturi mixer inducts small amounts of EGR. As engine speed and load increases, the EGR flow increases naturally. This creates synergy since the engine tolerates more EGR at higher engine loads and speeds and greater EGR dilution is required for knock mitigation. An EGR valve was still required primarily to enable shut-off of EGR during certain conditions - that is cold conditions or at high power levels. For research purposes, it was also used to regulate the amount of EGR. In cases where the inlet pressure was greater than the backpressure of the engine, an inlet throttle was utilized to create the required negative delta pressure to drive the EGR flow.

A configuration of the D-EGR concept is shown in the schematic in Figure 3. In this graphic, only one of the cylinders is used to produce the entirety of the EGR inducted by the engine. An additional feature of the D-EGR setup was the wide-band O₂ sensor in the

dedicated exhaust stream for fueling feedback and control. The prototype EGR mixer was designed to allow good temporal EGR mixing to evenly spread the exhaust gas expelled from one cylinder across all four and minimize cylinder to cylinder EGR imbalance. The mixed gas is cooled through an oversized intercooler. To improve efficiency, a WGS catalyst can be employed to the D-EGR concept. The WGS catalyst should be installed in the dedicated cylinder exhaust runner, downstream of the UEGO sensor. By utilizing the WGS reaction, excessive CO emissions, which are the result of rich combustion and water vapor, can react to form H_2 and CO_2 and further enhance EGR quality. All the testing in this study, however, was performed without a WGS catalyst.

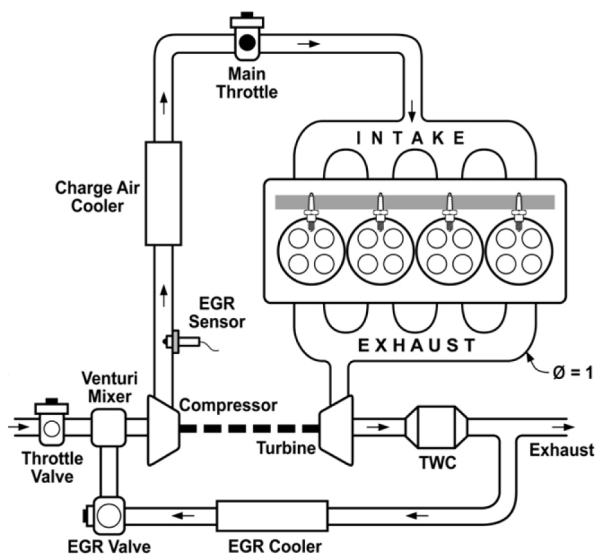


Figure 2. Schematic of a turbocharged 4-cylinder engine in a LPL EGR configuration

The control architecture for both engine configurations included two parallel controllers. The majority of the engine systems, for instance fuel, boost, and timing, was controlled with an open ECU provided by the engine OEM. The unique systems including the EGR valve, inlet throttle valve, DCO ignition for the LPL EGR architecture and dedicated cylinder fuelling, and spark timing in the D-EGR configuration, were controlled with a separate and fully flexible control unit.

Haltermann's EEE certification gasoline was used for the entire testing. The fuel had a lower heating value (LHV) of 42.89 MJ/kg, an Anti knock Index (AKI) of 92 and a Hydrogen to Carbon ratio of 1.862.

For each data point the low-speed data was averaged over 60 s and the high-speed data was acquired over 200 engine cycles, using cylinder pressure transducers in combination with a 0.5 crank angle degree shaft encoder. Combustion phasing was adjusted to knock limited spark advance (KLSA) and / or to yield an optimal location of CA50, which was considered 5-7° aTDC for this engine. Although the engine was equipped with variable intake cam timing, the cam phaser was kept in its parked position resulting in minimal valve overlap. For the large majority of the tested conditions with cooled EGR, this increased the maximum efficiency as determined by minimum BSFC. Fuel flow was measured using a Micro Motion

CMF10M coriolis flow meter. Test-to-test BSFC variability did not exceed 2 g/kWh at stable engine conditions. Exhaust samples were taken with a heated sample line and five gaseous emissions (CO , HC , NO_x , CO_2 , O_2) were quantified using a Horiba MEXA-7100 DEGR gas analyzer.

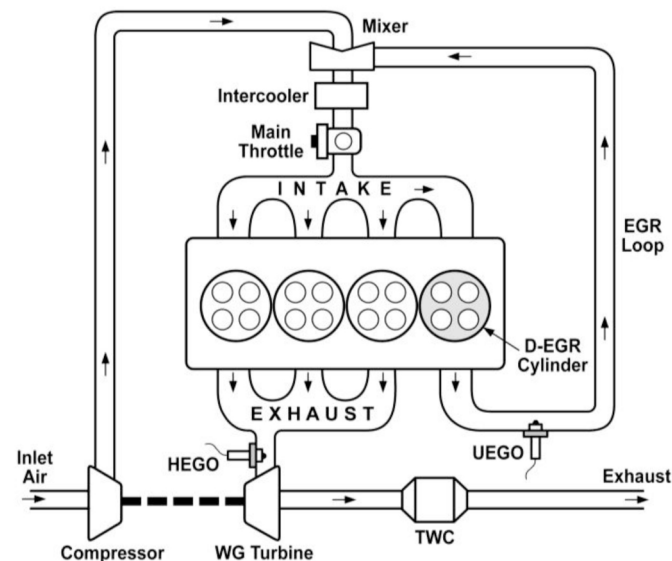


Figure 3. Schematic of a turbocharged 4-cylinder engine in a D-EGR configuration

RESULTS AND DISCUSSION

The following sections are provided to compare LPL EGR and D-EGR directly. All comparisons are at as similar a condition as possible including CR, ignition energy, CAC effectiveness, boost system, and main cylinder's AFR.

Engine Efficiency

The engine baseline was established using the open ECU and the calibrations provided by the OEM. The engine supplied was a prototype engine and the calibration was for test bench use only. It was unknown how much effort the OEM put into optimizing the calibration. However, the results were consistent with an efficiency optimized, boosted, non-EGR dilute MPI engine.

Figure 4 illustrates the BSFC improvement at the 1250 rpm / 3.5 bar BMEP representative point when LPL EGR is applied. At this condition, the stock engine had a BSFC of ~287 g/kWh. The OEM control strategy included some valve overlap for increased internal EGR (44 CAD intake cam advance relative to the parked position). With the conversion into a LPL EGR engine, BSFC was improved by adding cooled EGR until the stability limit was reached at roughly 16% EGR. EGR tolerance was enhanced through the employment of an advanced ignition system and the engine was capable of stable operation with up to 25% EGR. As expected, the D-EGR engine showed similar efficiencies with the production pistons when in-cylinder reformation was not utilized (stoichiometric operation on all cylinders) as shown in Figure 5. However, once the equivalence ratio of the dedicated cylinder was raised, generating H_2 and CO , further BSFC improvements were realized. During the initial stages

of testing, the maximum dedicated cylinder equivalence ratio was 1.3. For most conditions, this was not a combustion limit of the dedicated cylinder but rather a fuel controller limitation. With the CR increase to 12.5:1 and eventually 13.5:1, efficiency continued to improve. Interestingly, the 13.5:1 CR low load efficiency did not significantly differ from the 12.5:1 CR pistons, indicating that there is some practical limit in efficiency improvement due to compression ratio increase. In total, a 28 g/kWh BSFC reduction was found at this condition. The majority of the efficiency improvement came from the ability to increase CR and maintain the stock torque curve with high levels of EGR dilution. The remainder came through improvement in charge properties, burn durations and combustion efficiency enabled by reformat. Besides the further efficiency improvement through the reformat, the superior burn rates and mixture ignitability also allowed up to 25% air dilution at this condition. With the added air dilution further substantial BSFC reductions were achieved. Since lean operation would have necessitated supplemental aftertreatment equipment for NO_x reduction, this option will not be discussed. It is worth mentioning that the part load improvements were solely a result of modifications to charge properties and fuel delivery strategies - all data was at MBT combustion phasing. Technologies like cam phasing or alternative engine cycles (Atkinson/Miller) were not applied but have the potential to improve fuel consumption by reducing pumping losses. These topics will be considered in future work.

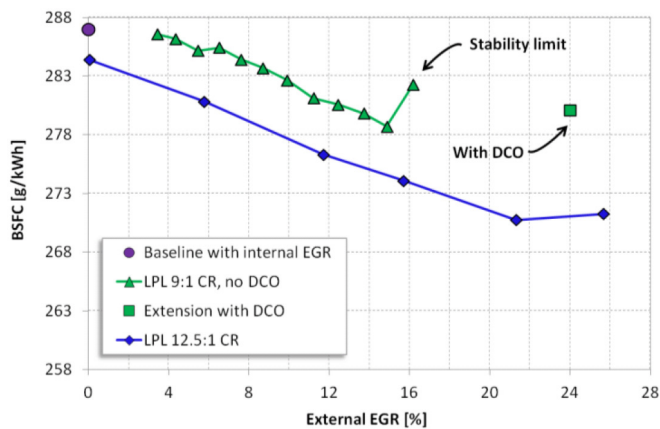


Figure 4. BSFC versus EGR rate for the LPL EGR configuration with different compression ratios at 1250 rpm / 3.5 bar BMEP

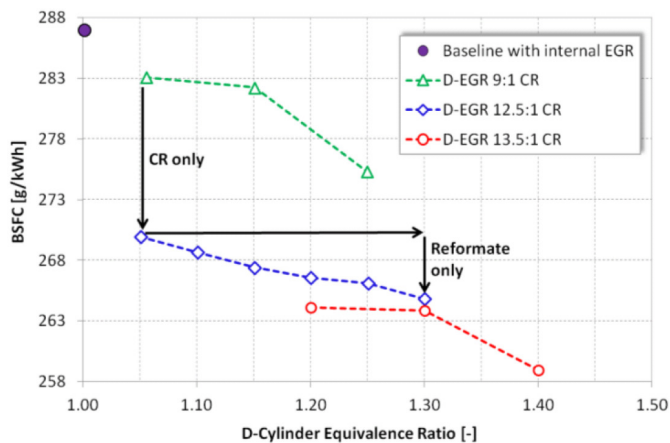


Figure 5. BSFC versus dedicated cylinder equivalence ratio for D-EGR configurations with various compression ratios at 1250 rpm / 3.5 bar BMEP

Since the knock propensity and full load performance of the LPL EGR engine with the 12.5:1 CR pistons was unacceptable [16], the 13.5:1 CR pistons were not tested in this configuration.

Although EGR tolerance was improved with the D-EGR setup, at loads below 2 bar BMEP, 25% EGR dilution still imposed combustion stability challenges. At these conditions, greater H_2 concentrations can aid through a further enhanced combustion process. To show the effect on efficiency, the 2000 rpm / 2 bar BMEP universal point was chosen. Figure 6 compares various engine configurations commencing with the BSFC for the baseline engine. All depicted D-EGR points were operated with $\phi = 1.3$. The first major efficiency improvement can be seen going from the stock engine to the LPL EGR configuration with 25% cooled EGR and then elevated CR. The next large improvement comes from the conversion to D-EGR and significantly improved combustion stability through high quality EGR. For the initial D-EGR test with 9.1:1 CR, an EGR mixer was not installed. As a result, EGR distribution was poor, leading to high instabilities and deteriorated fuel consumption compared to the LPL EGR configuration. This can be seen by the labels inside the BSFC columns which denote CoV of IMEP numbers for each engine configuration.

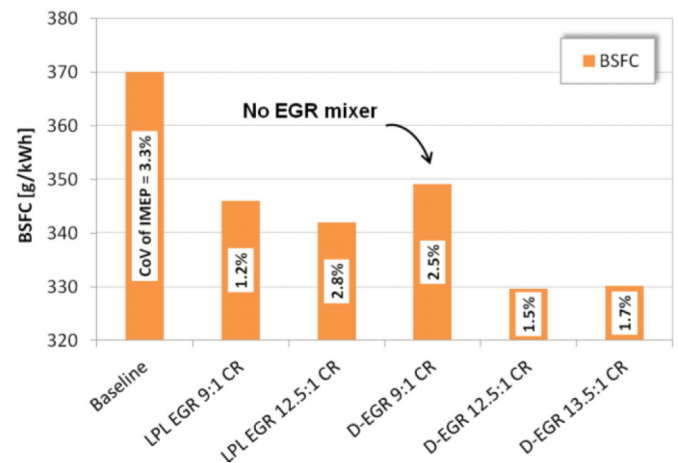


Figure 6. BSFC for various LPL and D-EGR configurations at 2000 rpm / 2 bar BMEP

Engine-Out Emissions

In the following graphs, the engine-out emissions are compared for the two EGR concepts at 1250 rpm / 3.5 bar BMEP. The addition of EGR along with an increased compression ratio had a notable impact on engine-out emissions. The 9.1:1 CR LPL configuration was unable to run more than 16% EGR due to engine instability. With the addition of the DCO system, it was possible to run with 25% EGR. As the LPL EGR rate was increased, there was an elevation in brake specific hydrocarbons (BSHC), while there was also a reduction in brake specific carbon monoxide (BSCO) and brake specific oxides of nitrogen (BSNO_x), as shown in Figure 7. These emissions trends are typical of an engine running traditional cooled EGR [17, 18]. The increases in BSHC were related to the increase in quench distance and reduced post-flame oxidation due to a decrease in flame and cycle temperatures. The gradual and considerable reduction in NO_x emissions proved how effective cooled EGR is at suppressing NO_x formation. CO emissions were reduced owing to less dissociation