

Synergies between High EGR Operation and GDI Systems

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ABSTRACT

A gasoline direct injection engine was operated at elevated EGR levels over a significant portion of the performance map. The engine was modified to use both cooled and un-cooled EGR in high pressure loop and low pressure loop configurations. The addition of EGR at low and part load was shown to decrease NO and CO emissions and to reduce fuel consumption by up to 4%, primarily through the reduction in pumping losses. At high loads, the addition of EGR resulted in higher fuel consumption benefits of 10-20% as well as the expected NO and CO reductions. The fuel economy benefit at high loads resulted from a decrease in knock tendency and a subsequent improvement in combustion phasing as well as reductions in exhaust temperatures that eliminated the requirement for over-fuelling. The results presented in this paper show that a strategy of cooled EGR at high loads and hot EGR at low loads can enable radical downsizing and high BMEP operation, which can in turn significantly reduce vehicle fuel consumption.

A split injection strategy was investigated at low load conditions to determine if a semi-stratified charge would result in an improvement in EGR tolerance. The results show that, within a narrow window of both the amount of fuel injected and the injection timing, the addition of a second injection event late in the compression stroke leads to a reduction in initial burn duration. The improvement in burn duration leads to an increase in combustion stability and a reduction in fuel consumption and emissions.

INTRODUCTION

Recent concern over fuel economy and emissions for light-duty vehicles, combined with higher gasoline prices, has led to an intensified focus on reducing fuel consumption from gasoline, spark ignited engines. Given the inescapable reality of pumping losses in stoichiometric gasoline engines at low loads, a common proposed solution is to reduce engine displacement and

use a turbocharger to increase the power output. Since the vehicle power requirements will not change, the reduced engine displacement will lead to a shift in the typical engine operating range to higher levels of specific power output, greatly reducing the need to operate the engine at the lowest loads [1]. GDI engines have been recognized as having high potential in this area, as the GDI system shows higher knock tolerance than typical PFI engines [2]. However, at elevated specific power levels, the engines still have higher than desired fuel consumption [3], particularly at engine speeds less than 2000 rpm and greater than 4000 rpm. At low engine speeds, knock is a problem and to combat it spark retard is required. The detrimental effect of spark retard on combustion phasing serves to significantly increase the specific fuel consumption at these conditions. At high engine speeds, knock presents less of a problem but any spark retard at these power levels results in excessive exhaust temperatures, which require enrichment. Enrichment may also be necessary 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. For consumers, downsizing offers a large improvement in CO₂ emissions reduction, 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 [4, 5] and reduce or eliminate the requirement for over-fuelling.

One of the unique aspects of a GDI engine is the ability to form a stratified charge. In normal stratified operating conditions, injection of gasoline is delayed until the piston is near TDC (see e.g. [6-8]). The combustion chamber, piston and injector act together to form a combustible mixture near the spark plug despite the fact that the rest of the charge is outside the flammability limits. This allows GDI engines to operate at very high A/F ratios with good stability and fuel consumption [9].

However, deficiencies in lean after treatment technologies, such as low conversion efficiency and susceptibility to sulfur poisoning [10, 11], have prevented the implementation of this technique in production light-duty engines in the US and limited its usage in the rest of the world. Because of these difficulties, recent work in GDI engines has focused more on implementing stoichiometric operation than on stratified charge operation [12], although lean burn / stratified operation is still seen as having potential in certain markets [13].



Figure 1. 1.6L Turbocharged GDI engine [14] of the Peugeot 207 and Peugeot 308.

The primary purpose of this project was to demonstrate the potential of relatively high levels of EGR to improve fuel economy in GDI engines. In addition to the goal of improving fuel economy, this set of tests was also designed with the goal of demonstrating the emissions benefits of using relatively high levels of EGR, particularly on emissions of oxides of nitrogen (NO_x), as well as the effect of EGR on exhaust temperatures. The exhaust temperature effect is particularly important in downsized, boosted applications, where the engine is expected to operate at load levels higher than traditional engines. High loads tend to produce elevated exhaust temperatures for several reasons. First, the mass flow rate and energy flux through the engine increases, which will necessarily increase the combustion temperatures as well as the temperatures of the engine exhaust hardware. Second, even very well designed engines knock and the requirement for spark retard to prevent knock has a dramatic effect on exhaust temperatures as well as fuel economy and emissions. Typically, the solution to the exhaust temperature problem has been to enrich the charge to reduce combustion temperatures. Previous publications, e.g. [4], have shown that low levels of EGR ($< 10\%$) are more effective at reducing exhaust temperatures than fuel enrichment or dilution with air. This project intends to demonstrate the benefit of higher levels of EGR and show that EGR with GDI hardware has the potential to enable significant

downsizing by allowing operation at high load while maintaining the ability to meet tomorrow's emissions standards.

EXPERIMENTAL SETUP

The engine used in these experiments was provided to Southwest Research Institute (SwRI) by PSA Peugeot Citroen. It is a 4 cylinder, 1.6L engine (Figure 1) equipped with intake cam phasing and a turbocharger. The engine is equipped with a side-mounted injector located under the intake ports and normally operates with early injection timings at $\phi = 1.0$. The relevant hardware characteristics of the engine are shown in Table 1.

Table 1. Hardware specifications [14]

Bore	77 mm
Stroke	85.5 mm
Compression Ratio	10.5:1
Fuel Pressure	120 bar
Maximum Torque	240 N-m @ 1400-4000 rpm
Maximum Power	105 kW @ 5500 rpm
Combustion Mode	Homogenous, early injection, $\phi=1.0$

In stock form, the engine is equipped with a dual scroll turbocharger and, for these tests, the stock turbocharger was used despite the mismatch caused by the addition of EGR. Normally, the engine is also equipped with an air-to-air aftercooler, however for these tests an air to liquid heat exchanger was used and the outlet air temperature was controlled to the same settings as the stock calibration. A test cell oil cooler was used in the same way, with the parameters controlled to match the stock settings provided by the manufacturer.

In stock form, the engine is not equipped with an EGR circuit and the intake cam phaser is used to modulate intake cam timing to reduce NO_x emissions and pumping losses. Previous work at SwRI has shown that different EGR configurations work better in some situations than others. In particular, at low loads a high-pressure loop (HPL) EGR system, where the EGR flows from the high pressure side of the turbine into the intake after the throttle, has been shown to be superior. At higher loads, the engine has a positive pressure gradient, so HPL EGR was impractical unless the backpressure was increased. Increasing the backpressure has a negative effect on turbocharger efficiency and reduces engine knock resistance by increasing the amount of not

residual gases retained in the cylinder. To take full advantage of the turbocharger and maximize efficiency at high loads, the pre-turbine pressure must be reduced below the intake manifold pressure. Because of this, the engine was also equipped with a low pressure loop (LPL) EGR system, where the EGR flows from the exhaust post-turbine and enters the intake before the compressor. Figure 2 shows a schematic of the layout of the EGR systems on this engine.

Figure 2 also shows heat exchangers for both the HPL and LPL EGR circuits. Since the strategy for employing EGR involved using HPL EGR at low loads and LPL EGR at high loads, the HPL EGR cooler required bypass capabilities (Figure 3). These coolers are common on modern diesel engines and this particular cooler (multi-tube, bypassable) is normally used on a low emission, light duty diesel engine currently in production. As has been noted in other publications, one of the benefits of EGR dilution over lean burn applications is the potential to increase manifold temperatures at low loads. At these loads, the cooler valve can bypass the cooling tubes and result in manifold temperatures in excess of 100° C. On the LPL side, the EGR was introduced at the compressor inlet, so cooled EGR was necessary to maintain high mass flow rates through the compressor and to protect compressor materials. Therefore, the heat exchangers on the LPL EGR circuit are not required to be bypassable and are plumbed into the engine coolant circuit. At lower loads and EGR flow rates, the EGR exits the LPL EGR circuit at coolant temperature. During engine operation, it was found that higher LPL EGR temperatures reduced compressor efficiency and mass flow rate through the compressor.

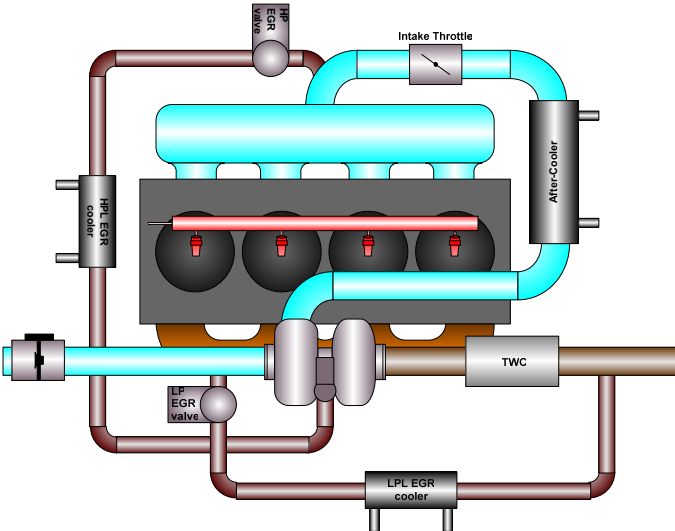


Figure 2. Layout of HPL and LPL EGR systems

Previous studies [15-18] have showed the effectiveness in additional ignition energy or unique spark plug design on improving EGR tolerance and performance in SI engines. However, in this instance, the authors were interested in isolating the effect of adding EGR from other confounding factors. Because of that goal, the stock ignition system was left on the engine and no

modifications were made to the hardware or its method of employment. The ignition system consisted of a 50 mJ coil-on-plug design with a precious metal spark plug with a fine center electrode tip. The stock spark plug gap was maintained at ~ 0.7 mm.

To understand the level of benefit that EGR provides the engine over its performance map, the engine was tested and evaluated at over 15 speed load points, from very low load speed and load to high speeds and loads, including the peak torque and peak power conditions. However, in the interest of space, a test matrix (Table 2) was developed that included representative low, part and high load conditions over the entire engine speed range for detailed discussion. Since the focus of the study was on understanding the level of improvement possible with EGR, other variables besides spark timing and EGR rate were held constant. Intake cam timing, start of injection timing and fuel pressure were maintained at their baseline settings. For future studies, it may be advisable to determine if the optimal engine settings without EGR remain the desired conditions in the presence of EGR. Besides the actuator settings, at high load (> 10 bar bmep) the manifold temperature was maintained at the same level as the stock charge air cooler, which resulted in a range from 30° C to 50°C, depending primarily on engine speed and boost level. Oil temperature was maintained at or below 120°C and the coolant temperature was held constant at 92°C. Spark timing was adjusted to hold the location of 50% MFB at 6° aTDC (representing optimal combustion phasing) or KLSA, whichever was more appropriate.



Figure 3. Bypass HPL EGR cooler (provided by Dayco USA)

Table 2. Test Conditions

Low Load Conditions		High Load Conditions	
Engine Speed [rpm]	BMEP [bar]	Engine Speed [rpm]	BMEP [bar]
1500	3	2000	14
1500	8	2000	18
2000	2	3000	18
2000	5	4000	18
		4500	18
		5500	15

A unique attribute of GDI engines is the ability to perform multiple injection events in cylinder for different purposes, ranging from improved cold start to idle stability to knock resistance. In the case of dilution with EGR, the EGR tolerance at low load is much worse than that at higher loads, due primarily to lower charge and cylinder surface temperatures. To extend the EGR tolerance of the engine, a split injection strategy is employed to form a stratified charge at low loads. The goal is to form a rich mixture around the spark plug to improve initial flame propagation rates and decrease the burn duration of the engine. The split injection test conditions are shown below in Table 3.

Table 3. Split Injection Test Conditions

Engine Speed [rpm]	BMEP [bar]
1500	3
2000	5

RESULTS AND DISCUSSION

As shown in Table 2 and Table 3 above, there were two main tasks performed during this experiment. The first was baseline testing, which consisted of adding the EGR loops to the engine and introducing EGR to the engine in varying amounts. Due to time constraints, there was no attempt to re-match the turbocharger to the engine with new mass flow rates and exhaust energy levels and so the load capacity of the engine at high (greater than 20-25%) EGR levels was necessarily diminished. Once the baseline testing was complete, the split injection conditions were examined.

PART LOAD TESTING

A basic summary of the part load tests is shown below in Figure 4. At these conditions, the engine was operated with HPL EGR and the EGR cooler in bypass mode, in an attempt to increase intake manifold temperature to the greatest extent possible. The addition of EGR at these conditions results in some fuel economy savings, between 1% and 3%, but did not result in the level of improvement that was expected based on prior results [4, 16, 18]. The reasons behind this will be discussed later in this section. The reduction in BSNO was as expected, with NO emissions reduced by 60% and 80% at the best BSFC EGR levels. One point that should be reiterated is that the engine was operated at stock settings for all but spark timing – start of injection (SOI) timing, intake valve opening (IVO) timing and fuel rail pressure were held constant from the stock calibration. After performing more tests, it is obvious that the engine would benefit from re-optimizing the cam phaser and SOI settings to improve EGR tolerance and decrease fuel consumption.

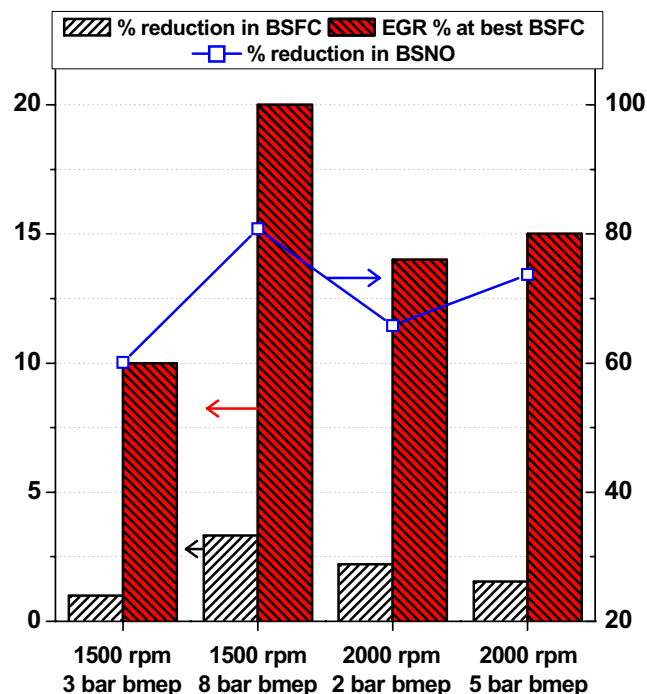


Figure 4. Summary of EGR results at low loads

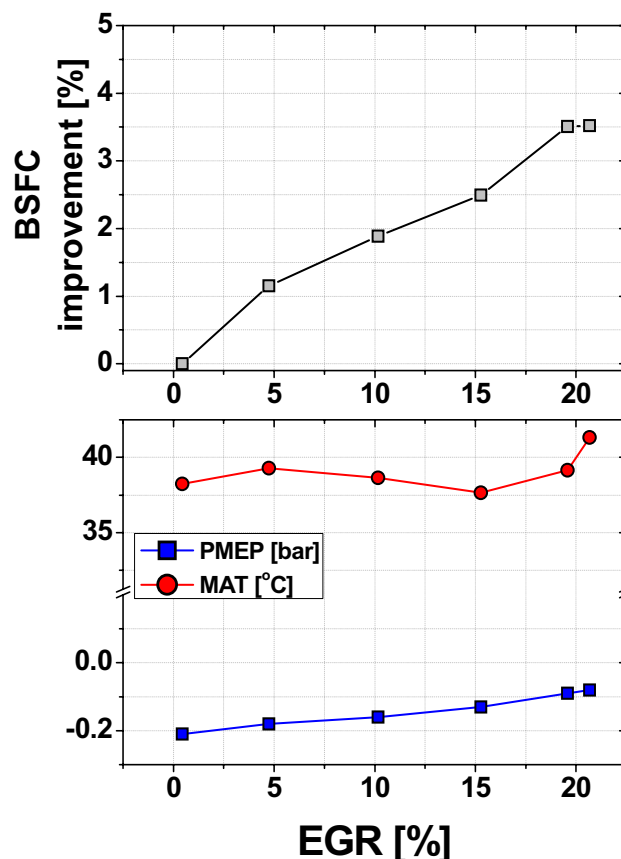


Figure 5. BSFC and pumping losses as a function of EGR at 1500 rpm / 8 bar bmeq

At these part load conditions, the primary cause of the fuel consumption reduction was lower pumping losses. In turn, the reduction in pumping losses was caused by a combination of EGR displacing and heating the intake

air, both of which required the engine to operate at a higher manifold pressure to maintain load. Figure 5 shows the pumping losses and manifold temperatures as a function of EGR for 1500 rpm / 8 bar bmep. At this condition, the engine power was low enough that the EGR temperatures did not begin to heat up the intake manifold to any significant degree until very high EGR rates were reached. However, the pumping losses approached zero despite the lack of temperature increase, due solely to the replacement of intake air with EGR. In addition, the engine was very stable, with CoV of imep < 1.5% even at EGR rates of 20%, so it is conceivable that more EGR could have been added, although its effect on fuel economy after 20% is questionable. In other applications, the addition of more EGR has resulted in intake heating, which, if the intake becomes hot enough, improves burn rate and combustion efficiency.

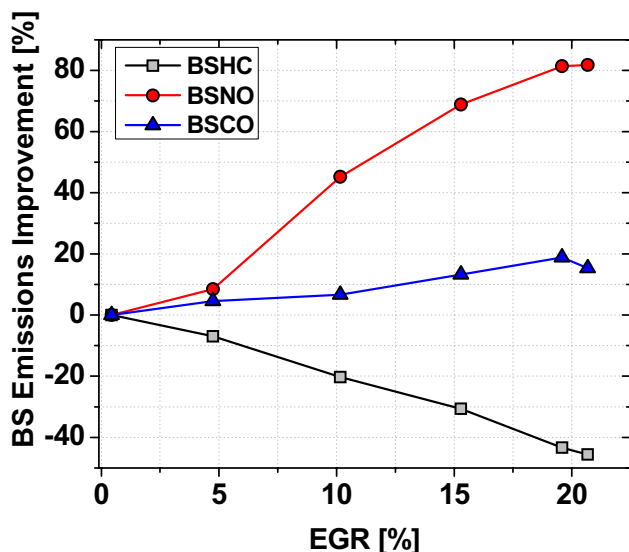


Figure 6. Emissions as a function of EGR at 1500 rpm / 8 bar bmep

In addition to the effects of EGR on fuel economy, it also has a beneficial effect on most emissions. Figure 6 shows the exhaust emissions from the engine at the same conditions as shown in Figure 5 (1500 rpm/8 bar bmep). As expected, the presence of EGR reduces the BSNO emissions by over 80%, but, more interestingly, the BSCO level decreases significantly with increasing EGR rates. In earlier studies of cam phasing [19, 20], it was suggested that increased levels of internal EGR caused reductions in HC and CO emissions by “re-burning” the emissions in the cylinder. A detailed mass flow rate balance indicates that at steady-state conditions, this cannot possibly be the case. It is more likely that the mechanism for CO reduction is more closely related to the changes in equilibrium conditions with EGR. However, the small increase in manifold temperatures may also be helping the injected fuel vaporize more completely and mix well, which is also associated with reduced emissions, in particular CO. A more thorough analysis of the CO reduction will be presented later in this report. Finally, the HC emissions increase by ~ 40%, which was expected due to longer

burn durations and cooler in-cylinder temperatures, both of which are typical results when operating in high dilution conditions.

While the results are quite promising at moderate loads (5 bar < bmep < 8 bar), the results at lower loads are somewhat less impressive. As Figure 4 shows, the BSNO emissions decrease substantially at every condition, but the fuel economy gains are less as the load decreases. In Figure 7, we see that one of the primary reasons for the reduced gains is a decrease in EGR tolerance, as the CoV of imep increases rapidly with EGR addition until the engine becomes unstable. Similar to the conditions at higher loads, EGR decreases BSNO (> 60% reduction at each point) and BSCO (~ 25% reduction at best BSFC points) while slightly increasing BSHC (~ 20% increase at best BSFC points). Unlike at higher loads, at these conditions the manifold vacuum is quite high (MAP = -55 kPa (gauge) / 45 kPa (abs) at 2 bar bmep). This increases the internal residual substantially [21, 22] and, since the engine output power is quite low, the exhaust temperatures are also reduced. The increased but cooler internal residual serves to radically slow the flame propagation rate with the addition of external EGR (Figure 8). As Figure 8 shows, the total (0-90% MFB) burn duration increases rapidly after the EGR level gets above approximately 10%, which has a large effect on stability and therefore fuel consumption (Figure 7).

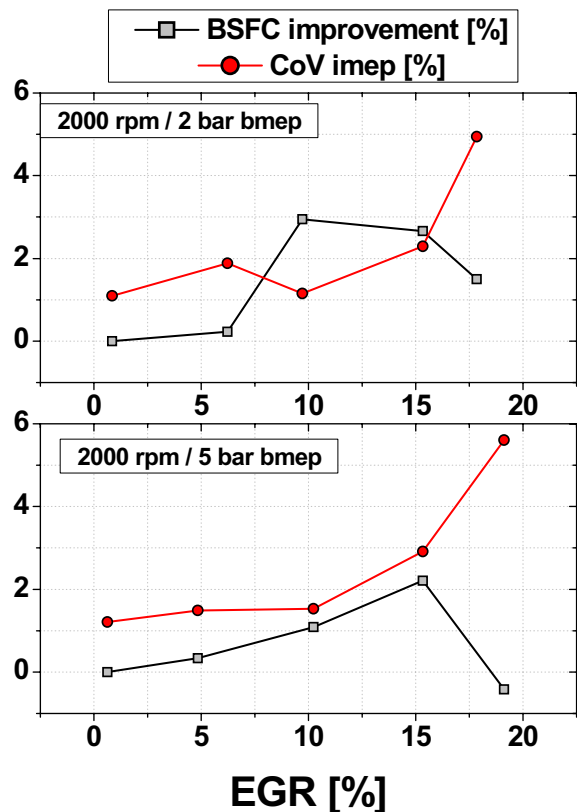


Figure 7. BSFC and stability results at low loads / 2000 rpm

Besides the increase in burn duration, another reason that the inclusion of EGR did not result in a large decrease in fuel consumption was that the engine was