

The Impact of Cooled EGR on Peak Cylinder Pressure in a Turbocharged, Spark Ignited Engine

Terrence Alger, Raphael Gukelberger, Jess Gingrich, and Barrett Mangold
Southwest Research Institute

ABSTRACT

The use of cooled EGR as a knock suppression tool is gaining more acceptance worldwide. As cooled EGR become more prevalent, some challenges are presented for engine designers. In this study, the impact of cooled EGR on peak cylinder pressure was evaluated. A 1.6 L, 4-cylinder engine was operated with and without cooled EGR at several operating conditions. The impact of adding cooled EGR to the engine on peak cylinder pressure was then evaluated with an attempt to separate the effect due to advanced combustion phasing from the effect of increased manifold pressure. The results show that cooled EGR's impact on peak cylinder pressure is primarily due to the knock suppression effect, with the result that an EGR rate of 25% leads to an almost 50% increase in peak cylinder pressure at a mid-load condition if the combustion phasing is advanced to Knock Limited Spark Advance (KLSA). When combustion phasing was held constant, increasing the EGR rate had almost no effect on PCP. In addition, since EGR enables higher compression ratio operation, some data was taken at elevated compression ratios to establish the impact of EGR at a constant load with increased compression ratio.

CITATION: Alger, T., Gukelberger, R., Gingrich, J., and Mangold, B., "The Impact of Cooled EGR on Peak Cylinder Pressure in a Turbocharged, Spark Ignited Engine," *SAE Int. J. Engines* 8(2):2015, doi:10.4271/2015-01-0744.

Introduction

As the desire to reduce fuel consumption, due to either regulatory requirements or consumer demand, continues to be the primary concern of OEMs across the globe, new technologies are being developed to improve engine efficiency. Cooled EGR, combined with hybrid technologies, has been used in several naturally aspirated production applications [1, 2]. In addition, several applications have been considered to use cooled EGR in conjunction with boosting devices to eliminate the need for an electric machine to produce the torque required to satisfy customer expectations [3, 4]. The primary benefit of cooled EGR has been shown to be the reduction in knock in spark ignited engines, which enables higher compression ratio operation and more optimized combustion phasing [5, 6, 7, 8, 9]. Additionally, through its impact on combustion temperatures, cooled EGR has been shown to improve the ratio of specific heats of the working fluid [6], reduce pumping losses and heat transfer losses [6, 7] and improve the emissions of a gasoline engine [10, 11]. Overall, the level of improvement from adding EGR to a gasoline engine, independent of other considerations such as downsizing or hybridization or separate pumping loss mitigation technologies (i.e. variable valve lift and duration, late / early intake valve closing, etc.), has been shown to be approximately 6-8% on cycles such as the New European Driving Cycle (NEDC) [12] and potentially more on more highly loaded cycles such as the Federal Test Procedure (FTP-75). As such, it appears to be an attractive technology for fuel consumption reduction, particularly when combined with other, synergistic technologies.

However, as with any new concept, there are several challenges that must be addressed with this new technology. In previous work, the impact of high levels of dilution on ignition system requirements [13] and the challenge to boosting systems [14] has been examined. One other concern is the impact of cooled EGR on peak cylinder pressure requirements. This is of concern for several reasons. First, the limits of the engine with respect to torque and power need to be understood and planned for appropriately. If the new engine requires an upgrade in peak pressure rating, then the work needs to be considered as part of the cost of the upgrade and may change the value proposition. Second, any work that will be done to increase peak cylinder pressure will involve either increasing the cost of the components or increasing the weight of the engine. In the first case, the value proposition must be understood and, in the second case, the impact of the increased weight of the engine, both as a whole and its reciprocating components in particular, on friction and fuel consumption must be considered. In any event, it is important to understand how and to what extent the use of cooled EGR impacts the peak cylinder pressure constraints of a spark-ignited engine.

In a typical light duty, automotive engine, peak cylinder pressure can be a limit for torque, although typically the peak pressure at peak torque is lower than at other conditions due to spark retard. Conventional spark ignited engine operation uses timing retard as means to reduce engine knock. This coincides with a reduction in peak cylinder pressure even as torque increases. Therefore, conventional engine operation at high load is within the stress limits of the head and linkage to the crankshaft. The downside is

significantly lower efficiency from the retarded combustion phasing. In addition, retarding combustion phasing also leads to high exhaust temperatures and significant inefficiencies associated with fuel enrichment to reduce those temperatures. High combustion dilution has the potential to impact peak cylinder pressure in two ways: First, the added EGR directly increases the mass in the combustion chamber for a given amount of fuel (assuming stoichiometric A/F mixtures). Second, EGR dilution mitigates knock and allows more advanced combustion phasing, raising peak combustion pressures. The objective of this investigation was to de-couple the contribution of the effects EGR has on peak cylinder pressures and to determine the relative contribution of each mechanism.

Experimental Setup



Figure 1. 1.6L turbocharged GDI engine [15]

The engine used in these experiments is a 4 cylinder, 1.6 L 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 [15]

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

The engine was equipped with a low pressure loop EGR system (LPL EGR) and, due to the challenges in boosting high dilution engines [14], a two-stage boosting system. The boosting system in this case was two turbochargers in series, using a variable geometry turbine (VGT) as the low pressure stage (a 2052V TC from Honeywell) and the series production twin-scroll turbocharger for the high pressure stage. A layout of the boosting system is shown in Figure 2. The boosting system was matched to allow the engine to achieve 20 bar BMEP from 1500-4000 rpm while using 25% LPL EGR. The torque curve and BSFC map for the engine when operating with 25% cooled, LPL EGR is shown in Figure 3. Note that the EGR rate is calculated by dividing the CO_2 level in the intake manifold by the exhaust CO_2 level (after correcting for ambient CO_2 levels), so the 25% EGR rate was a volumetric rate and not a mass rate. The BSFC data was generated using a dynamometer test cell equipped with a load cell and a coriolis mass flow meter. Unless specifically remarked upon, all fuel consumption measurements in this publication are brake specific measurements with the series water and oil pumps but without other parasitic loads, such as the alternator.

In addition to modifying the boosting system of the engine, the ignition system was upgraded to a dual coil offset (DCO) ignition system [13] and a production prototype EGR cooler was added. Since the purpose of the experiment was to evaluate the impact of EGR on peak pressure, no attempt was made to optimize the settings on the ignition system and it was run at high power, using approximately 500 mJ per cycle per cylinder to insure adequate combustion stability at elevated EGR levels. The EGR cooler was run as a part of the engine coolant circuit and the EGR coolant flow set to maintain an EGR gas outlet temperature of less than 150° C at rated power. For the testing discussed in this report, an air-to-liquid charge air cooler was employed and the intake manifold temperature was maintained at 45° C for all testing.

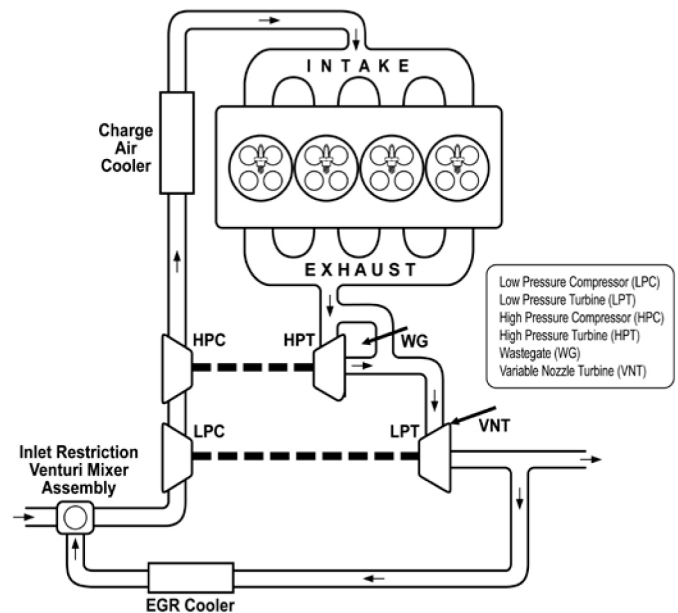


Figure 2. Layout of LPL EGR engine with 2-stage series boosting

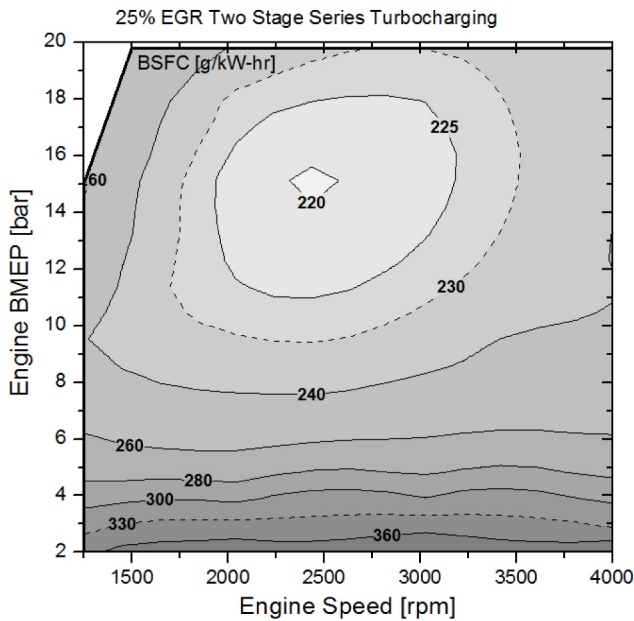


Figure 3. Torque map with BSFC contours of the engine operating with 25% cooled EGR

The indicated pressure data was acquired with a Kistler 6052C pressure transducer and a 720° per revolution shaft encoder. The burn rate data was calculated by performing a heat release analysis and then integrating the instantaneous heat release rate to produce a cumulative heat release that was then normalized to 100%, similar to the method presented in [16]. Based on previous work, the optimized combustion phasing, corresponding to Minimum spark advance for Best Torque (MBT), for the engine was determined to be when the location of 50% mass fraction burned (CA50) was between 6-10° aTDC, which also corresponds to a location of peak pressure of 13-17° aTDC. For results which discuss advancing the ignition timing to knock limited spark advance, the knock limit was defined as the onset of audible knock.

Results and Discussion

The engine was tested at 1500 rpm, with the expectation that the phenomena observed would be applicable, with potentially small differences in the response rate, at all engine speeds. Three different loads with different levels of EGR were identified while maintaining constant intake valve opening timing and GDI fuel parameter settings. The reference combustion phasing at a given load was determined with 0% EGR. At conditions where the engine was not limited by knock, this phasing was matched as EGR was added. At knock limited conditions, two sets of data were generated. The first data set was produced holding combustion phasing constant at the 0% EGR baseline condition. The second set was generated by advancing combustion phasing to the knock limited condition as EGR was added. In this manner, the impact of EGR on peak cylinder pressure due to higher manifold pressure can be separated from the impact due to improved combustion phasing.

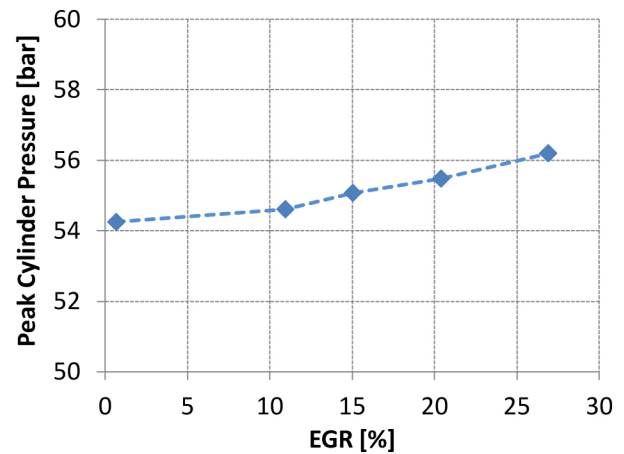


Figure 4. Peak pressure as a function of EGR at 1500 rpm / 8.3 bar BMEP

The first condition is 1500 rpm, 8.3 bar BMEP. This condition was the highest load achievable without knock at optimal combustion phasing. At this condition EGR was increased to 25% incrementally and spark timing was adjusted to maintain constant combustion phasing.

Figure 4 shows the impact of EGR, as dilution only, on peak cylinder pressure.

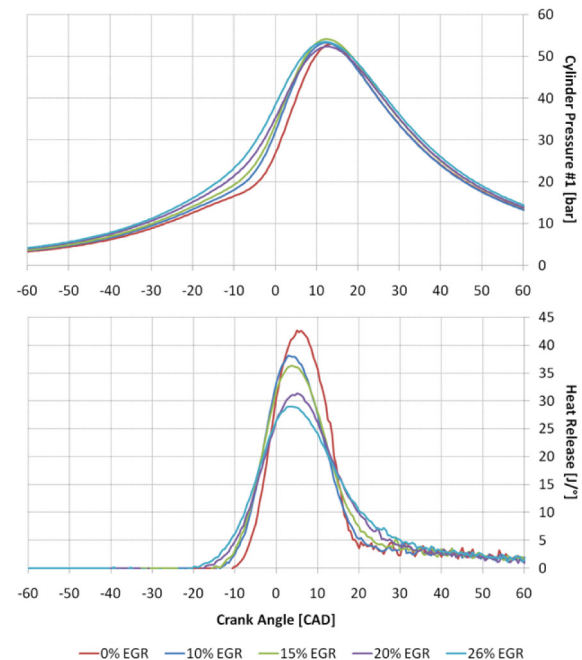


Figure 5. Cylinder pressure and heat release rate with EGR at 1500 rpm / 8.3 bar BMEP

The results shown in Figure 4 show that the increase in peak cylinder pressure was within experimental uncertainties, as 25% EGR increased peak cylinder pressure by ~3.6%, although the trend in cylinder pressure does appear indicate a slight relationship between EGR rate and peak pressure. The cylinder pressure and heat release rates for the different EGR rates are shown to clarify how each individual EGR step impacted combustion. The pressure