

Dedicated EGR: A New Concept in High Efficiency Engines

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ABSTRACT

The use of high levels of EGR has been documented to increase fuel efficiency and reduce emissions of spark ignition engines [1-5]. However, these engines typically face challenges in EGR control and tolerance, which can reduce the expected efficiency improvement. A concept developed by Southwest Research Institute explores the potential of an engine with individual cylinders dedicated to EGR production to overcome the challenges associated with EGR tolerance and control. In this study, a 4-cylinder engine was run with cylinder 1 exhausting directly to the intake manifold, leading to a constant 25% EGR level. The engine was run naturally aspirated over a large portion of the performance map at an ultra-high (14:1) compression ratio. As a part of the study, air-to-fuel ratio in cylinder 1 was varied from stoichiometric to rich to determine the effect of the products of partial combustion on EGR tolerance and fuel consumption. The results show that this concept results in large gains in fuel efficiency over the baseline, low compression ratio engine and significantly reduces the exhaust emissions from the other 3 cylinders.

INTRODUCTION

Concern over CO₂ emissions for light-duty vehicles, combined with higher fuel prices, has led to an intensified focus on reducing their fuel consumption. Compared to compression ignition engines, spark ignited 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 [6]. Because of increasingly strict emissions regulations, most gasoline engines employ 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 stoichiometric conditions in the 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 pumping work at low loads. 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 [7]. 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 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 [8]. In addition, shrinking the engine size and operating at higher specific power levels also reduces the relative penalty of frictional losses, especially 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 reduce compression ratio. 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 [9].

While it is possible to achieve good fuel consumption results at high power levels and reduced compression ratios [10], the fuel consumption at low to part load in these engines is often compromised from their high compression ratio, NA counterparts. Despite a reduced compression ratio, the downsized engine will often encounter reduced efficiencies at high loads [11], particularly at engine speeds less than 2000 rpm and greater than 4000 rpm. At low engine speeds, knock is reduced with spark retard, which leads to sub-optimal combustion phasing and increases specific fuel consumption. At high engine speeds, knock presents less of a problem but any spark retard at high power levels results in excessive exhaust temperatures, which require enrichment. 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. 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 [1, 4, 5] and reduce or eliminate the requirement for over-fuelling [1, 4, 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 [1, 4, 5, 12] and the ability to operate at high loads at compression ratios above 12:1 [3, 13]. In addition, elevated EGR levels serve to substantially reduce the emissions of CO [1, 14] and NO_x [1, 4, 14] through a reduction in combustion temperatures. A difficulty presented by high EGR operation, however, is a reduction in the burn rate, which can lead to engine instability, increased HC emissions and poor fuel consumption. A major challenge of high dilution engines is to improve their dilution tolerance, through enhanced ignition systems [15-17] or by supplementing the charge with high reactivity compounds, either pure H₂ [2, 18] or reformed fuel, consisting of CO and H₂ in some proportion [19-23]. 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. [2, 18, 19, 23-25].

One drawback of the methods detailed in [19-23] is the requirement for a device to convert the fuel into the reformat. This has traditionally been accomplished in several ways. In [21], the incoming fuel was reformed using a low (600° and 650° C) temperature reformer to

produce CO and H₂. In [19, 20, 22-25], “simulated” reformer gas was used to mimic the output of a plasma-based reformer system. Due to the relatively low H to C ratio of gasoline, any fuel that is reformed in the presence of oxygen will contain a fairly large percentage of CO, which also has an improved lean limit compared to gasoline ($\phi=0.345$ for CO vs. $\phi=0.667$ for a typical gasoline) and a higher octane rating (RON = 106). The presence of CO may also increase the positive effects of the H₂. Also, in the studies referenced above, a fairly significant percentage of the fuel was reformed into H₂ and CO – in some cases representing up to 30% of the gasoline [22]. While the proponents of reformation of gasoline outside the engine acknowledge that there is a loss of approximately 23% of the Lower Heating Value (LHV) of the fuel through reformer heat losses [22], they often fail to account for the energy requirements of the reformer, which is another source of parasitic losses on the engine. In [21], the reformer uses exhaust gas energy and, as long as the heat is not needed to help a 3-way catalyst operate, should not incur an additional fuel economy penalty from the reformer besides the loss of LHV from the reformation process. Clearly, the loss of LHV and additional parasitic losses from the method of reformation are significant drawbacks to the use of H₂ enrichment in a gasoline engine.

The objective of this project was to revisit the idea of reforming gasoline for use in a stoichiometric, high EGR engine but to use a more efficient and cost effective method of reforming the fuel. An alternative method, based on previous experiments on large-bore natural gas engines [26] was proposed in a previous publication [2]. This method involves running one (or more) cylinders rich of stoichiometric and routing 100% of that cylinder's exhaust directly into the intake manifold – establishing a dedicated source of EGR that is augmented with significant amounts of CO and H₂. The goal of the tests was to show that operating an engine with a dedicated EGR loop and other modifications to accommodate high EGR operation (high compression ratio, EGR cooler) can significantly improve fuel consumption and emissions.

EXPERIMENTAL SETUP

The engine used in this experiment was a 2.4-L World Engine manufactured by Chrysler, LLC (Figure 1). The engine was naturally aspirated, with a peak torque of 220 N-m @ 4400 rpm and a peak power of 125 kW @ 6000 rpm (Figure 2). The engine was port fuel injected and also employs variable camshaft phasing on both the intake and exhaust cams. In stock form, the engine operates at 10.5:1 compression ratio, which was increased for these tests to 14:1 for high EGR operation.



Figure 1. 2.4L NA World engine from Chrysler

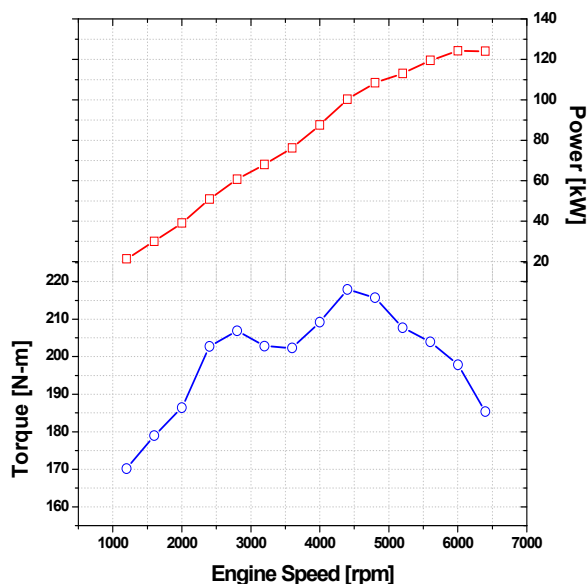


Figure 2. Stock torque and power levels of the 2.4L engine

Bore [mm]	88
Stroke [mm]	97
Compression Ratio [-]	14:1
Fuel	Haltermann-EEE
Octane rating (R+M)/2	93
H-to-C ratio	1.862
Ignition system	Multi-strike Inductive
Coil energy	50 mJ
Spark Plug Type	Dual Fine Electrode
Gap Width	1.6 mm

Table 1. Engine specifications

One of the most important parameters in improving the dilution tolerance of the engine is the ignition system. For these tests, the spark plugs in the engine were upgraded from the stock plugs, which had a relatively large, nickel-coated copper center electrode, small gap

and traditional ground strap. The new plugs had a very fine (< 0.6 mm) iridium center electrode and a similar, precious metal electrode on the ground strap. To promote dilution tolerance, the plugs were deployed with a very large gap of 1.65 mm.

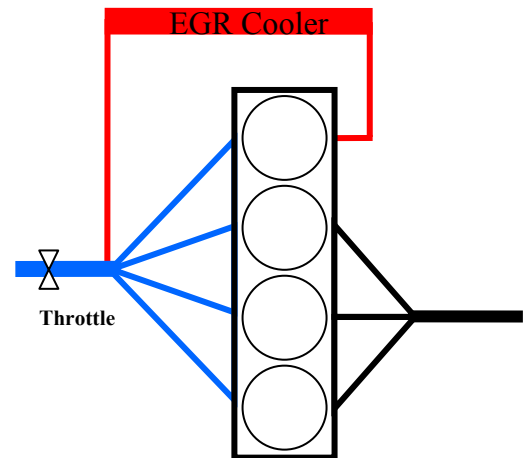


Figure 3. Layout of 25% dedicated EGR engine

As has been described previously [2], the concept of in-cylinder fuel reformation involves using one or more cylinders dedicated to EGR production. In this instance, the engine was re-configured such that the exhaust from cylinder 1 was segregated from the exhaust manifold and plumbed directly to the intake manifold, passing through an EGR cooler (Figure 3). With the engine configured in this manner, cylinder 1 can be run at air/fuel ratios much different than the rest of the engine without compromising the efficiency of the three-way catalyst or increasing the emissions of the vehicle. In addition, directly plumbing the cylinder exhaust into the intake manifold eliminates any uncertainty in the engine's EGR rate as well as the requirement for an EGR valve, which should make transient control and operation much easier. As part of a variation on this concept, a partial oxidation catalyst (PO_x) was placed in the exhaust runner of cylinder 1 in an attempt to increase the mass fraction of H_2 by converting some of the unburned hydrocarbons (HC) in the exhaust to CO and H_2 .

The engine was run at two part load operating conditions, 2000 rpm / 2 bar BMEP and 2000 rpm / 5 bar BMEP as well as along the full load curve up to 4000 rpm. The equivalence ratio in cylinder 1 was varied between 1 and 1.3 to determine the optimal level of enrichment for maximum fuel economy benefit. For the majority of the experiment, cylinders 2-4 were run stoichiometric, although a short experiment running them lean at low loads was performed. A description of the operating conditions is shown in Table 2. For these tests, spark timing was controlled so that the location of

50% mass fraction burned was between 6° and 10° aTDC when possible (primarily at part load). If the engine was knock limited, then spark timing was set at the knock limited spark advance. The engine was equipped with cam phasers, but since the purpose of the project was to perform a preliminary investigation of the D-EGR concept, no effort was made to optimize the valve events. Instead, the settings that yielded the best BSFC with EGR from the “normal” EGR configuration and 10.5:1 CR were used. These settings were very similar to the cam timings that were used in the stock calibration of the engine. In future studies, intake and exhaust valve opening timing will be an important attribute for study.

Test Condition	Cylinder 1	Cylinder 2-4
2000 rpm / 2 bar BMEP	$1 < \phi < 1.2$	$\phi = 1.0$
2000 rpm / 2 bar BMEP	$\phi = 1.2$	$0.85 < \phi < 1.0$
2000 rpm / 5 bar BMEP	$1 < \phi < 1.35$	$\phi = 1.0$
1500 rpm / Full Load	$1 < \phi < 1.2$	$\phi = 1.0$
Full Load (1500, 2000, 3000 and 4000 rpm)	$\phi = 1.2$	$\phi = 1.0$

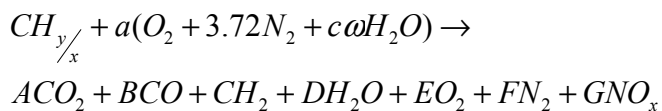
Table 2. Test conditions

RESULTS AND DISCUSSION

This section will detail the results of the testing and examine the reasons behind any improvement / degradation in engine performance. Prior to discussing those results, however, an analysis of the amount of reformed gas available from different levels of enrichment will be presented.

IN-CYLINDER REFORMATION OF GASOLINE

Since the purpose of re-configuring the engine to run one cylinder rich was to generate H_2 and CO, the first step in the experiment was to run a series of rich A/F ratio tests. From the emissions data collected during these tests, the level of H_2 in the exhaust was calculated by performing a chemical balance in the exhaust. Equation 1 was solved to determine the amount of H_2 created, where the fuel properties (H-to-C ratio, A/F_{stoich}) and mass flows (fuel and air) are known and the level of CO, CO_2 , O_2 , NO_x and HC emissions are measured with a 5-gas emissions analyzer (Horiba MEXA-7100DEGR).



Equation 1. Exhaust mass balance

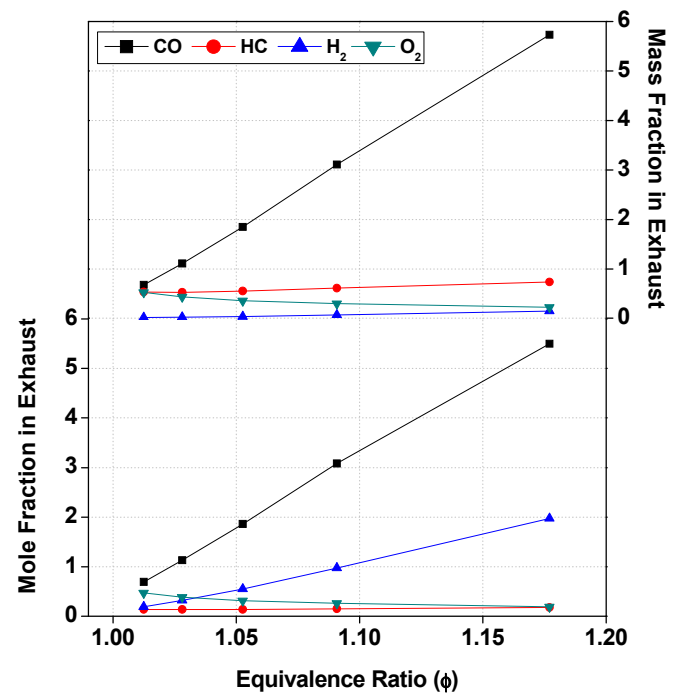


Figure 4. Mole and mass fractions in the exhaust from rich combustion

The results of applying Equation 1 to a rich air/fuel (A/F) ratio sweep are shown in Figure 4. This figure shows that the mole fraction of CO and H_2 in the exhaust products increase rapidly with a decrease in A/F ratio, reaching ~ 6% and 2% respectively by the time the equivalence ratio is 1.2. The level of unburned HC stays low and relatively constant, while the amount of available O_2 declines, but is not eliminated. On a mass basis, CO continues to make up about 6% of the charge at the lowest A/F ratio, but the H_2 concentration by mass falls significantly, due to the low molecular weight of hydrogen (2 kg/kmol). However, in terms of energy, the H_2 contributes a significant amount to the overall heating value of the exhaust, as its lower heating value (LHV) of 120 MJ/kg is more than 11 times the LHV of CO (10.9 MJ/kg). Figure 5 shows the results of calculating the LHV of the exhaust gas based on the LHV of the combustible components (CO, HC and H_2) and the relative contribution of each component to the overall LHV. From this, one sees that CO was also the primary source of energy in the exhaust but hydrogen's contribution is not as small as its mass fraction might indicate. In addition, Figure 5 also shows that the LHV of the exhaust is small (< 1 MJ/kg) when compared to gasoline (42.8 MJ/kg), primarily due to the relatively low mass fractions of the combustible components (Figure 4).