

The Interaction of Fuel Anti-Knock Index and Cooled EGR on Engine Performance and Efficiency

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ABSTRACT

Experiments were performed on a 2.4L boosted, MPI gasoline engine, equipped with a low-pressure loop (LPL) cooled EGR system and an advanced ignition system, using fuels with varying anti-knock indices. The fuels were blends of 87, 93 and 105 Anti-Knock Index (AKI) gasoline. Ignition timing and EGR sweeps were performed at various loads to determine the tradeoff between EGR level and fuel octane rating. The resulting engine data was analyzed to establish the relationship between the octane requirement and the level of cooled EGR used in a given application. In addition, the combustion difference between fuels was examined to determine the effect that fuel reactivity, in the form of anti-knock index (AKI), has on EGR tolerance and burn rate. The results indicate that the improvement in effective AKI of the fuel from using EGR is constant across commercial grade gasolines at about 0.5 ON per % EGR. In addition, there was a detectable difference in burn rates between fuels, the magnitude of which varied with operating condition. At the only condition where all three fuels could be compared on an equal basis, no detectable difference was found in EGR tolerance between fuels. Emissions results confirm the effect of higher fuel reactivity on burn rate, with lower AKI fuels leading to higher NO_x emissions at a constant EGR level and load.

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INTRODUCTION

Concern over CO₂ emissions, combined with higher fuel prices, has led to a focus on reducing the fuel consumption of internal combustion engines. 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 [1]. There are two main sources of this efficiency deficit between spark ignition and compression ignition engines. The most important source of inefficiency, particularly on the lightly loaded test cycles (NEDC, FTP-75) that measure fuel consumption, is pumping work associated with the gas exchange efficiency of the engine. Because of emissions regulations, most gasoline engines employ a three-way catalyst for emissions compliance. This aftertreatment system is robust and relatively inexpensive technology to produce ultra-low emissions. However, the three-way catalyst requires a fixed air/fuel ratio (AFR) to operate properly. To control load, a gasoline engine must therefore restrict airflow, which results in pumping work. Even if the engine was equipped with aftertreatment systems to allow lean operation, the lean limit of gasoline is not low enough to eliminate pumping losses at the lightest loads, unless alternative forms

of combustion, such as HCCI or stratified charge, are used [2]. Variable valve systems - either variable lift and duration systems (e.g. [3-4]) or variable cam phasing systems (e.g. [5-6]) - are often used to throttle the engine using the intake valves, which can significantly reduce the pumping work. However, such systems are expensive and can often add parasitic losses (friction), production challenges or increased warranty concerns. Another commonly proposed solution to reduce pumping losses is to decrease 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 [7]. 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. For ultimate engine efficiency, knock is a larger impediment than the pumping work. Due to the effects of knock, the potential

range of compression ratios for gasoline engines is quite limited. This limit is more severe on downsized engines, since knock tendency becomes worse at high specific power levels. Even the most advanced gasoline engines reduce the compression ratio by at least 1.5 when going from naturally aspirated (NA) to pressure charged operation [8]. While it is still possible to achieve high efficiency at high power levels and reduced compression ratios [9], 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 still encounter knock at high loads [10]. At low engine speeds and high loads, knock is averted with spark retard which leads to sub-optimal combustion phasing and increases specific fuel consumption. At high engine speeds knock is still present, albeit at a lower intensity. However, any spark retard at these conditions often results in excessive exhaust temperatures, which requires enrichment to offset.

The cooling effects of enrichment may also be necessary at high power levels, even at optimum combustion phasing, to improve the durability of engine hardware. This requirement at high specific power levels reduces the application of downsizing by making it difficult to meet modern emissions requirements and significantly increases specific fuel consumption. It also limits the usefulness of the gasoline engine in off-road or medium duty applications, as these applications have duty cycles and test cycles that require operation at high loads. 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.

In previous studies of boosted engines the presence of some EGR has been shown to decrease exhaust temperatures [11, 12, 13] and reduce or eliminate the requirement for over-fuelling [11, 13-14]. 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 [11, 12, 13, 14] and the ability to operate at high loads at compression ratios above 12:1 [15-16]. In addition, elevated EGR levels serve to substantially reduce the emissions of CO [13, 17] and NO_x [11, 13, 17] 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 at extreme conditions. A major challenge for high dilution engines is to improve their dilution tolerance, through enhanced ignition systems [18, 19, 20], or by supplementing the charge with high

reactivity compounds such as pure H₂ [21-22], or reformed fuel consisting of CO and H₂ in some proportion [23, 24, 25, 26, 27]. 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. [27, 28, 29]. In the engine used in this study, a combination of a high energy ignition system, developed by SwRI and detailed in a previous publication [30], was used to promote EGR tolerance along with a prototype, high tumble cylinder head.

As an alternative to using EGR to reduce knock, OEMs also have the option of specifying higher octane fuel for use in their spark ignition vehicles. While high octane fuel does result in a significant improvement in the knock limited performance of the engine, the use of high octane fuels also presents some challenges. In particular, since fuel in many countries is priced according to its octane rating, specifying high octane fuels results in higher fuel costs for the typical consumer. Although the benefit of higher compression ratio or increased downsizing can be measured on the drive cycle, the consumer may not see the additional fuel costs as a valuable trade-off for the improved efficiency. Since there is no way to force the driver to use higher grades of gasoline, many may choose to use lower octane fuels. While this is not a danger to the vehicle, as a modern EMS can adjust spark timing to reduce knock, the resulting change in combustion phasing will reduce or eliminate many of the fuel economy advantages for the engine. In addition to drivers deliberately choosing to use lower octane gasolines, a separate scenario exists in which a global company produces an engine that is intended to be used internationally. In this case, there are some countries with lower octane requirements for their fuels. In this case, the engine hardware may need to be changed to account for fuel differences or it must be designed to accommodate the lowest common denominator octane rating, which will reduce any potential fuel economy benefits in locations using higher octane fuels.

The purpose of this study is to understand the relationship and synergies between the knock suppression effect of EGR and the knock suppression effect of high octane fuels. A baseline, 87 AKI (Anti-Knock Index - defined as (RON + MON)/2) fuel was tested at a variety of conditions and EGR levels to develop a baseline relationship between EGR level and knock. These results were compared to a premium gasoline (93 AKI) and a high octane racing fuel (105 AKI). In addition, since the reactivity levels of the low AKI fuels are by definition higher, a series of tests were performed to determine if low octane fuels promote EGR tolerance and, if they do, how much that higher EGR tolerance offsets the increase in knock tendency due to the low octane rating.

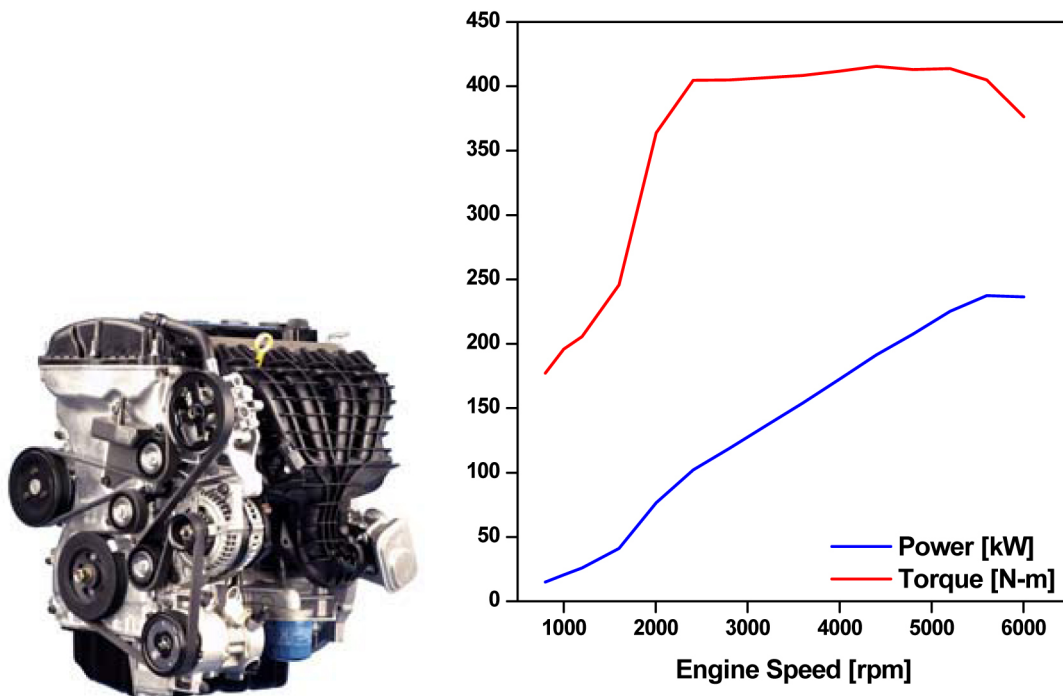


Figure 1. 2.4L engine (left: image; right: stock torque and power curves)

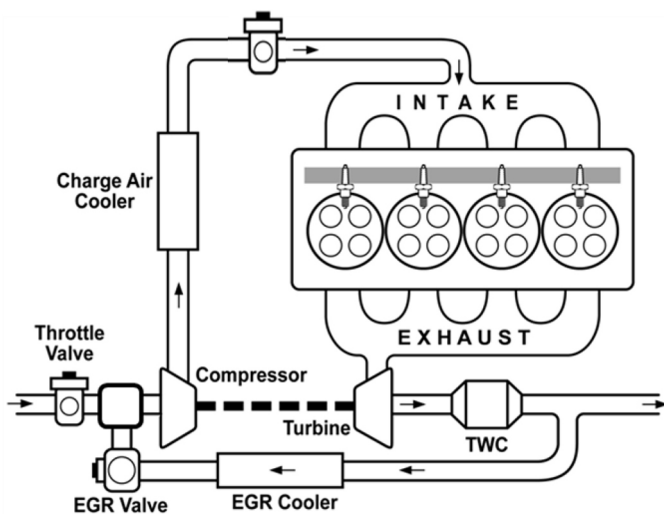


Figure 2. EGR system layout for the 2.4L engine

EXPERIMENTAL SETUP

The engine used in this study was a 2.4L MPI engine (Figure 1) and has been used in previous studies at Southwest Research Institute [13, 31-32]. The 2.4L engine has a peak torque of 400 N-m @ 2400 rpm and a peak power of 240 kW @ 5600 rpm (Figure 1). The engine was port fuel injected and also employed variable camshaft phasing on both the intake and exhaust cams. In stock form, the engine operates at 8.6:1 compression ratio. Previous tests have reported CR from 11.4:1 [32] to as high as 14:1 on this engine [31]. For

this set of tests, the engine was operated at a 11.4:1 CR and with a prototype head with new ports that increased the tumble ratio from ~ 0 to ~ 1.5 , as measured using a flow bench. The ignition system for these tests was SwRI's Dual Coil Offset® (DCO®) ignition system, which has been described in a previous publication [33]. The spark plugs in the engine were upgraded from the stock plugs, which had relatively large, nickel-coated copper center electrodes, small gaps and traditional ground straps. 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 gap of 1 mm.

A prototype EGR system was installed on the test engine, as shown in Figure 2. The system consisted of an EGR cooler and valve, installed in a low pressure loop (LPL) configuration. The cooler was specified to reduce the EGR temperature to ~ 120 - 140° C to control the compressor inlet temperature to less than 60° C. The LPL EGR design was chosen to limit backpressure on the engine at high loads and low engine speeds, reducing the amount of hot internal residual and knock at these conditions. At low loads, the engine has limited tolerance for cooled EGR and the external EGR rate was reduced or eliminated. At these conditions, high levels of dilution were instead achieved with internal EGR by using the dual independent cam phasers.

A Horiba 5 gas exhaust analyzer was used to measure carbon dioxide (CO₂), carbon monoxide (CO), oxygen (O₂), total hydrocarbons (THC), nitric oxide and nitrogen dioxide (NO_x) in undiluted engine exhaust. CO₂ was also measured in the intake manifold. The EGR rate was calculated as the ratio between the CO₂ level in the intake and the CO₂ in the exhaust, correcting for ambient CO₂ levels. Engine operating conditions (engine speed and load) varied during the course of the work and will be detailed in the appropriate discussion section. MBT spark timing and stoichiometric AFR were maintained unless specifically noted. MBT spark timing was assumed when the location of 50% mass fraction burned (50% MFB) was located 6-8 degrees after top dead center (aTDC). Intake and exhaust cam phasings were optimized at each condition for best brake specific fuel consumption (BSFC). Intake manifold temperature was allowed to vary depending on operating condition, but not allowed to exceed 50°C. Fuel injection timing was on a closed intake valve and held constant for all test points at TDC of the compression stroke. All burn rate data was based on a 720° crank encoder and was calculated using the method for net heat release found in [17].

The fuels used in this study were all commercially available fuels purchased from a variety of suppliers. The baseline fuel was a pump-grade, 87 AKI fuel. The premium grade fuel, 93 AKI, was certification grade fuel (Haltermann EEE) and the 105 AKI fuel was an oxygenated, high aromatics racing fuel (VP Racing Fuel MS109). The specifications for each fuel are presented in Table 1 below. Since each fuel has a slightly different lower heating value, the fuel consumption results in this paper are presented in terms of brake thermal efficiency (BTE).

Table 1. Fuel Properties

Parameter	Units	Fuels		
		87 AKI	93 AKI (H-EEE)	105 AKI (MS109)
LHV	MJ/kg	43.61	42.89	41.49
RON		91	97	109
MON		83	88	101
(R+M)/2		87	93	105

RESULTS AND DISCUSSION

The purposes of this study were to: 1. Determine if there was a correlation between the fuel octane rating and EGR tolerance and 2. Investigate the interaction between octane rating and dilution level. In the case of #2, the goal was to determine the level of EGR required for the same effect on knock tendency as an increase in fuel octane rating. In the case of #1, the point of that study was to investigate the effect of fuel octane rating on EGR tolerance to see if a lower AKI (i.e. higher reactivity) fuel may provide improved EGR tolerance and/or burn rates when highly diluted and what effect this improvement would have on engine performance. Unfortunately, the only point at which all three fuels could run at similar dilution levels with similar ignition timings was extremely low loads (less than 3 bar bmep), which limited the usefulness of the study. At these conditions, no effect of fuel

composition or reactivity was seen on EGR rate. To investigate the tradeoffs between the fuel reactivity and EGR rate effects on knock, the engine was run at wide open throttle (WOT) conditions and knock-limited peak BMEP conditions at 1500, 2000, 3000 and 4000 rpm to investigate the knocking behavior of the engine and the effect of EGR and fuel AKI on performance and emissions. As will be seen, the engine was also limited by the turbocharger, in that the achievable WOT loads with high levels of EGR were rather low compared to the stock engine. However, it was possible to achieve loads high enough to enter a knocking condition for all three fuels, particularly at low engine speeds. In addition, at 1500 and 4000 rpm, spark sweeps were conducted at part load conditions of 8 bar and 9.6 bar bmep respectively. These loads were the highest that could accommodate all three fuels from a knock standpoint and that the turbocharger could achieve with 30% EGR. The test conditions are summarized below in Table 2.

Table 2. Operating Conditions for Engine Testing

Speed [rpm]	BMEP [bar]	Notes
1500	WOT	Peak BMEP
2000	WOT	Peak BMEP
3000	WOT	Peak BMEP
4000	WOT	Peak BMEP
1500	Knock-limited peak torque	CA50 ~ 6° aTDC
2000	Knock-limited peak torque	CA50 ~ 6° aTDC
3000	Knock-limited peak torque	CA50 ~ 6° aTDC
4000	Knock-limited peak torque	CA50 ~ 6° aTDC
1500	8	Spark Timing Sweeps
4000	9.6	Spark Timing Sweeps

KNOCK LIMITED PEAK TORQUE CONDITIONS

The results of the testing at knock limited peak torque, which was defined as the highest torque at which the engine could be maintained with MBT timing (~6° aTDC location of 50% MFB) are shown in Figure 3. The results in this figure are shown at a constant EGR level of 25% and are in line with what might be expected from a sweep of fuels in this range of anti-knock properties for both BTE and BMEP trends. However, one interesting point on the plot can be seen at 4000 rpm, where the engine begins to encounter challenges with the boosting system. While the difficulties of boosting highly dilute, high efficiency engines have been discussed in great detail in a previous publication [34] and will not be revisited here, the limitations of the boosting system in this case lead to a considerable reduction in the BMEP potential of the engine and serve to reduce the differentiation between fuels from a knock-limited BMEP standpoint. Of course, once the BMEPs converge, the BTE results for the different fuels converge as well. In this case, the excessive backpressure from the boosting system led to an increase in internal residuals that caused the engine to start knocking at high loads, regardless of fuel composition. This was an outstanding illustration of the need to consider the engine as a system and the limitations that can be imposed by outside parameters. In this case, the boost system limitations reduce