

Ethanol Flex-fuel Engine Improvements with Exhaust Gas Recirculation and Hydrogen Enrichment

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ABSTRACT

An investigation was performed to identify the benefits of cooled exhaust gas recirculation (EGR) when applied to a potential ethanol flexible fuelled vehicle (eFFV) engine. The fuels investigated in this study represented the range a flex-fuel engine may be exposed to in the United States; from 85% ethanol/gasoline blend (E85) to regular gasoline. The test engine was a 2.0-L in-line 4 cylinder that was turbocharged and port fuel injected (PFI).

Ethanol blended fuels, including E85, have a higher octane rating and produce lower exhaust temperatures compared to gasoline. EGR has also been shown to decrease engine knock tendency and decrease exhaust temperatures. A natural progression was to take advantage of the superior combustion characteristics of E85 (i.e. increase compression ratio), and then employ EGR to maintain performance with gasoline. When EGR alone could not provide the necessary knock margin, hydrogen (H₂) was added to simulate an on-board fuel reformer. This investigation explored such a strategy at full load, and examined the potential of EGR for ethanol blends at part and full load. This investigation found the base engine torque curve could be matched across the range of fuels at a higher compression ratio. The engine could operate at maximum brake torque (MBT) timing at full load for all but the lowest octane fuel. Fuel enrichment was not needed to control exhaust temperatures, whereby carbon monoxide emissions were drastically reduced. Full load fuel consumption was reduced by 8-10% with regular gasoline (92 RON) and 20-21% with premium (100 RON). Full load brake thermal efficiency (BTE) increased 9.3 percentage points with E85 compared to

the base engine. The full load fuel consumption was only 9% higher than the baseline engine even though E85 has ~25% lower energy content (net heat of combustion) than gasoline.

INTRODUCTION

The Energy Independence and Security Act of 2007 increased the corporate average fuel economy (CAFE) to at least 35 mpg by 2020 and maintained tight emissions standards [1]. The Energy Policy Act of 2005 requires gasoline producers to nearly double the production of renewable fuels by the year 2012, and require federal dual-fuel fleet vehicles to operate exclusively on alternate fuel (typically E85)[2]. The implication of such legislation is that ethanol flex-fuel vehicle production volumes will continue to increase along with domestic production of ethanol. In the 2007 model year there are over 30 flex-fuel vehicle models available, and an estimated 5 million on US roads. The main drawbacks for consumers using E85 are the scarcity of fuelling stations, and the higher fuel costs to operate an eFFV on E85. The number of E85 filling stations has grown exponentially from year 2000 (133) to 2007 (>1200), but are still marginal compared to the ~167,000 retail gasoline stations in the US [3]. The higher cost to operate on E85 will persist until the ratio of gasoline cost per gallon to E85 is approximately 1.35, due to the lower energy content of E85 (based on 2007 EPA fuel economy estimates) [4]. Gasoline and E85 prices are driven by a multitude of market forces and government subsidies, but improving fuel consumption for eFFV is an attainable objective. In general, the strategies for improving fuel consumption for gasoline engines such as turbocharging and EGR can be applied to eFFV engines. Doing so has the potential to provide

eFFV fuel economy on E85 that can approach the fuel economy of the original gasoline counterpart.

Ethanol as an automotive fuel has increasingly become a topic of interest and a focus of research. Ethanol, in many ways, exhibits superior fuel characteristic compared to gasoline:

- Higher latent heat of vaporization ► Provides greater charge cooling
- Higher heat capacity of combustion products ► Provides cooler in-cylinder and exhaust temperatures
- Higher oxygen content ► Decreases potential for incomplete combustion by-products and soot
- Higher octane rating ► Enables higher specific outputs

The main technical issues surrounding ethanol as an automotive fuel are (1) cold startability and subsequent catalyst heating phase, and (2) modifications to fuel system components and lubrication due to the corrosive properties of ethanol. Manufacturers have taken various approaches to resolve these issues. Ford Motor Co. demonstrated that E85 blends can consistently ignite down to approximately -15°C [5]. Direct injection of ethanol has also been shown as a technology that can benefit start-up as well as other operating conditions [6,7].

Typical naturally aspirated FFV engines have been designed to accept alternate fuel blends, but do not take full advantage of the inherent benefits of the alcohol fuel [5, 8, 9]. More advanced eFFV engine architecture includes turbocharging and increased compression ratio to take advantage of ethanol combustion characteristics. However, when operating on gasoline, these engines de-rate maximum power outputs. They are also forced to retard spark timing from MBT and enrich beyond stoichiometric fuel to air proportions to avoid engine knock and exhaust temperature limitations [10, 11]. One of the more modern eFFV engines is the Saab 9-5 Biopower 2.0t. Emissions testing over the US06 drive cycle found carbon monoxide (CO) emissions were at ~82% of the applicable standard (8.0 g/mile) for gasoline while E85 was less than 10%. The researchers concluded that the higher CO emissions were a result of the necessary fuel enrichment to follow the aggressive drive cycle [10]. This study also identified significant spark retard ($10-15^{\circ}$) on gasoline compared to E85 during the first 4 seconds of a 0-60mph acceleration test. Honda's 1.8 liter VTEC engine also required spark retard ($\sim 12^{\circ}$) from MBT at ethanol fuel percentages less than 20% [11]. Researchers investigating a direct injected eFFV engine found that the measured specific fuel consumption for ethanol at full load was only slightly higher than for gasoline, and the full load BTE was 24% better with E85 [7]. Overall, these types of eFFV are capable of providing more torque and power when

operating on E85. Further incentive for eFFV would be created by decreasing the overall fuel consumption. EGR has been shown to provide such a capability.

EGR, in its various embodiments, has been the focus of a large amount of research. Cooled EGR has been investigated as a technique to decrease exhaust temperatures and suppress engine knock to extend the brake mean effective pressure (BMEP) [12,13]. EGR has also been shown to eliminate the need for fuel enrichment at full load and reduce pumping losses at lighter loads [14]. Recent studies have shown similar results wherein EGR allowed savings of up to 17% for carbon dioxide emissions (CO_2), 70% for CO, and 80% for unburned hydrocarbons (HC) compared to conventional fuel dilution [15]. Not only are the engine-out emissions reduced from the conventional gasoline engine, but they retain compatibility with three-way catalysts. By doing so, researchers have shown that EGR strategies can reduce tailpipe emissions to comply with US emissions targets in the near future [16]. EGR has also been investigated for use with neat alcohol fuels, exclusively, as a means to decrease part load throttling losses and suppress engine knock at high loads. Over 40% BTE was achieved with ethanol operating between 10 and 15 bar BMEP at speeds ranging from 1500 to 2500 rpm on a converted 1.9-L Volkswagen TDI automotive diesel engine [17]. These studies all show improved fuel consumption over a range of speed and loads which indicates improved drive cycle fuel economy.

The primary objective of this work was to demonstrate that the target turbocharged PFI gasoline engine could operate at high efficiencies as an eFFV engine at a higher compression ratio by utilizing EGR. The study identified BMEP limits at 2800rpm and verified operation at the target BMEP level at 4000rpm. A secondary objective was to use H₂ addition in situations where full load operation was limited by knock or combustion stability when operating with regular gasoline. The amount of H₂ used was ~10% of the fuel's composite net heat of combustion, or ~3% by mass. The amount of H₂ used during this test was selected to simulate an on-board fuel reformer. However, the benefits of hydrogen have been realized at lower concentrations (<1%) [18]. The modified engine was then compared to the base gasoline engine to identify how EGR could improve the overall performance of the engine

EXPERIMENTAL SETUP

The target engine was a stock Toyota 3S-GTE: 2.0-L, PFI, turbocharged, in-line 4 cylinder with 9:1 compression ratio (CR), fixed valve timing, and stock coil on plug ignition system. For this demonstration, the turbocharger was replaced with an externally driven supercharger and the compression ratio was increased to 11:1. Inlet air temperature and humidity were controlled. The manifold air temperature was controlled to 50°C . Engine back pressure was controlled to simulate the presence of the original turbocharger. Low

pressure EGR was cooled and routed to the intake before the throttle. EGR fraction (EGR_m) was defined as the percentage of EGR mass to the total charge mass (sum of air, fuel, and EGR). The engine description is given in table 1.

Table 1. Test Engine Configuration

Engine	In-line 4 cylinder
Bore X Stroke [mm]	86 X 86
Displacement [cc]	1998
Number of Valves	4
Compression Ratio	11:1 / 9:1
Fuel System	Port Injected
Boost System	Supercharged / Turbocharged

Multiple fuels were tested including E85, E50, premium gasoline (100RON), and regular gasoline (92RON). Selected fuel properties are given in table 2.

Table 2. Fuel Properties

	Ethanol	E85	E50	100 RON	92 RON
Density g/ml @ 60F	0.80	0.79	0.77	0.78	0.75
Carbon Mass %	52.20	57.86	69.23	87.68	86.66
Hydrogen Mass %	13.00	12.70	12.86	12.19	13.24
Oxygen Mass %	34.80	29.44	17.91	0.00	0.00
Net Heat of Combustion (kJ/kg)	26850	32200	37200	42575	43020
A/F Stoich	9.00	9.53	11.08	14.31	14.54
Research Octane Number	~110	>100	>100	100	92

BMEP limits were identified at 2800rpm. This speed represented the original peak torque speed and most susceptible to knock. The limits that constrained this testing are summarized in table 3.

Table 3. Limits of Engine Operation

Manifold Air Pressure – MAP [kPa-a]	200
Peak Cylinder Pressure - PCP [bar]	100
Pre-turbine Temperature – PTT [°C]	900
CoV IMEP [%]	5
Engine Knock	

Cylinder pressure was monitored in cylinder #1 with a flush mounted transducer. The coefficient of variation of the indicated mean effective pressure (CoV IMEP) was monitored to determine combustion stability. The knock limit was determined audibly, with a resonant knock sensor, and by calculating the knock intensity of the cylinder pressure. Three methods of knock detection were used because only one cylinder's pressure was monitored. There was no turbine when using the supercharger, but the back pressure was controlled to simulate one. Pre-turbine temperature was measured at

the same location in both turbocharged and supercharged configurations for continuity. Spark timing was adjusted to MBT by locating average peak cylinder pressure (PCP) at 17° after top dead center (aTDC) when possible per the recommendation of the engine manufacturer. The equivalence ratio (Φ) was held to 1.0 during the testing.

The H₂ measurement and delivery apparatus combined a Coriolis style mass flow meter and electronically controlled gaseous fuel injectors. H₂ was fumigated into the intake upstream of the throttle. H₂ addition was only necessary for the regular gasoline tests. The amount of fuel required to create the hydrogen was estimated based on ideal characteristic of a plasmatron reformer [19]. The required fuel was accounted for in brake specific fuel consumption (BSFC) comparisons.

RESULTS AND DISCUSSION

TEST ENGINE PERFORMANCE

Load sweeps were performed at 2800rpm to identify the limited BMEP for each of the fuels (figure 1). The desired BMEP was determined from the target engine to be 15bar. When knock was encountered timing was retarded and then boost was increase until knock was encountered again. This procedure continued until a second limit was encountered (see table 3). Due to knock, regular gasoline was limited to ~10bar BMEP. Adding H₂ increased the limited BMEP by ~2bar. Premium gasoline increased the limited BMEP to ~13bar. EGR (20%) was required for premium gasoline to meet the target BMEP at MBT. Regular gasoline with EGR (20%) was not able to meet the target BMEP due to knock and combustion stability. Adding H₂ improved combustion stability whereby allowing retarded spark timing to achieve the target BMEP. Both E50 and E85 were able to exceed the target, 17bar and 19bar, respectively. E50 was limited by knock at MBT. E85 was limited by PCP at MBT. This is summarized in figure 2. The large icons in each series of figure 1 are displayed in figure 2.

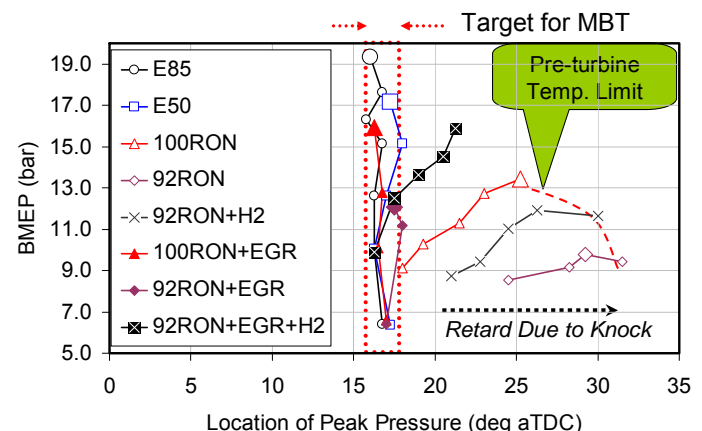


Figure 1. Identifying BMEP Limits at 2800RPM

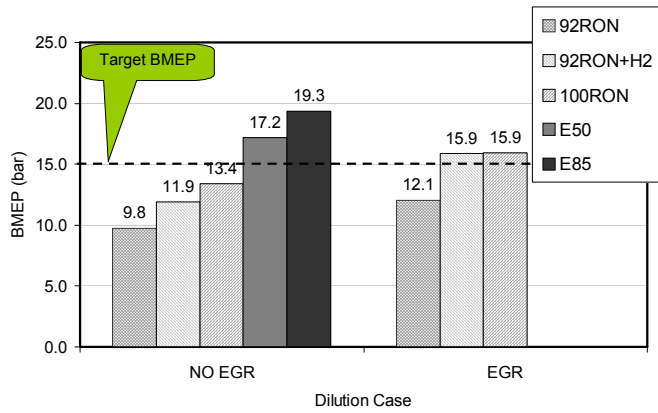


Figure 2. Max BMEP for Diluted and Undiluted Cases at 2800rpm

Flex fuel operation at peak torque was obtained for this engine. Premium gasoline with cooled EGR enabled full load operation with adequate knock margin and eliminated the need for fuel enrichment. Due to the increased knock susceptibility of regular gasoline, hydrogen addition was required to meet the target load. Future work may identify charge motion strategies and enhanced ignition techniques that will improve combustion stability and EGR tolerance so that an on-board fuel reformer is not required.

At 2800rpm the primary limiting factor was engine knock, but at 4000rpm the pre-turbine temperature was primary limiting factor. Figure 3 shows pre-turbine temperature as a function of BMEP at 4000 rpm.

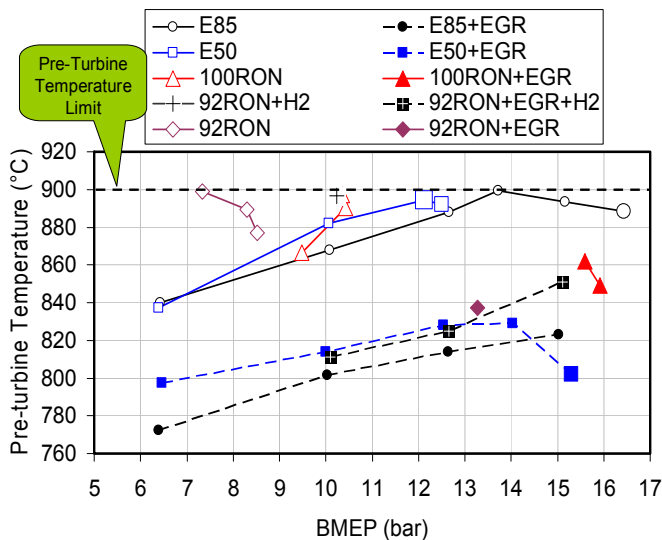


Figure 3. Pre-turbine Temperature as a Function of BMEP at 4000rpm

Undiluted regular gasoline was subject to engine knock and pre-turbine temperature limit. Adding hydrogen decreased knock tendency, allowing timing advance which reduced exhaust temperature. The result was a 2bar increase (from 8 to 10bar) in BMEP. Undiluted premium gasoline was limited to 10 bar BMEP by knock

and pre-turbine temperature. E50 reached a knock and temperature limit at 12.5 bar BMEP. E85 reached the temperature limit at 13.5bar. E85 was advanced from MBT without encountering knock resulting in cooler exhaust temperatures. Doing so allowed E85 to exceed 15bar BMEP. At 2800rpm EGR was needed for all fuels besides E85 to reach 15bar BMEP. In general, EGR cooled the exhaust by $\sim 70^{\circ}\text{C}$ at higher loads. The optimum amount of EGR for the fuels was determined based on BTE improvement. Figure 4 shows BTE and CoV IMEP for the blended fuels as a function of EGRm. The optimum EGR levels were $\sim 15.5\%$ for the ethanol blends and $\sim 18\%$ for regular and premium gasoline at 2800rpm. Similar results were found at 4000rpm. The optimums correlated to the EGR rate where CoV IMEP breaks from the horizontal trend. EGR improved BTE for E50 and E85 by approximately 3-4%.

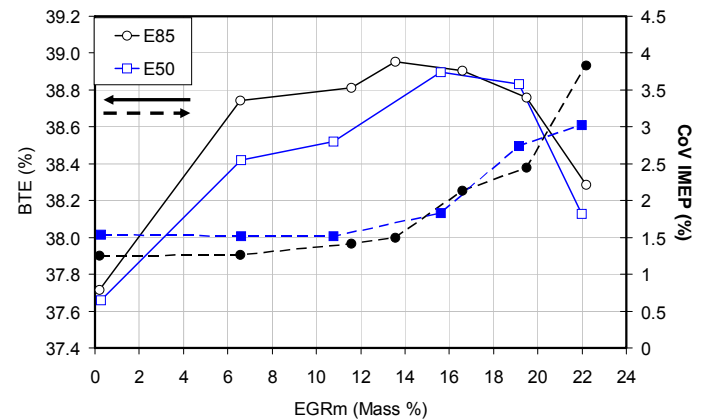


Figure 4. Optimum EGR for E85 and E50 at 2800rpm

Optimum levels of EGR improved full load performance for all the fuels while maintaining adequate temperature and knock margin. A fuel consumption comparison at 4000 (top) and 2800 (bottom) is shown in figure 5. Fuel consumption for the ethanol blends at full load was higher than gasoline on a mass basis. The difference was proportional to the difference in energy content of the fuel (see Table 2). At 15bar, the amount of fuel required to create the H_2 for regular gasoline with EGR increased BSFC to E50 levels. This indicated that the penalty for reforming the fuel to create H_2 is similar to the penalty incurred when using a fuel with 12% lower energy content. Full load fuel consumption with premium gasoline had $\sim 15\%$ lower BSFC than regular gasoline because of MBT operation and no need for H_2 creation.

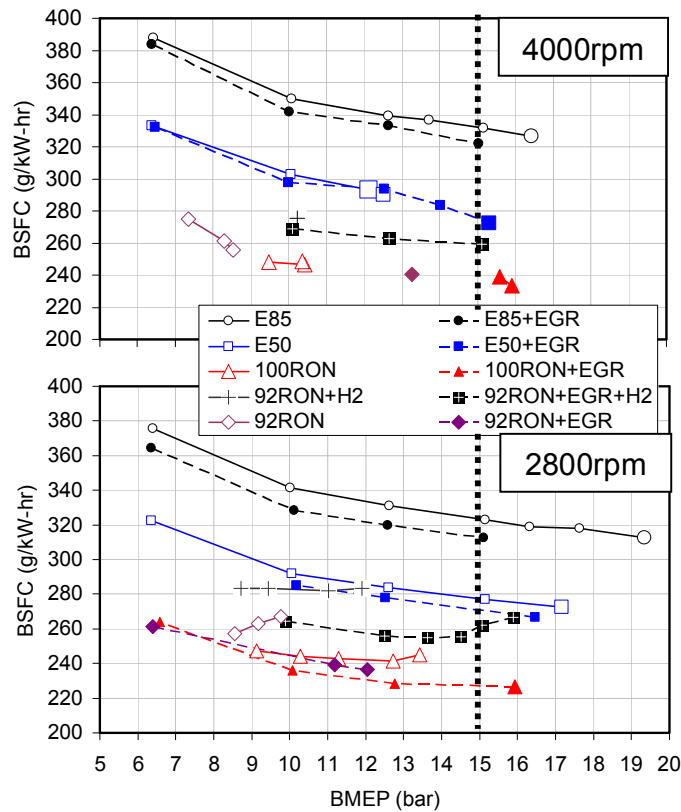


Figure 5. Fuel Consumption Comparison at 4000 and 2800rpm

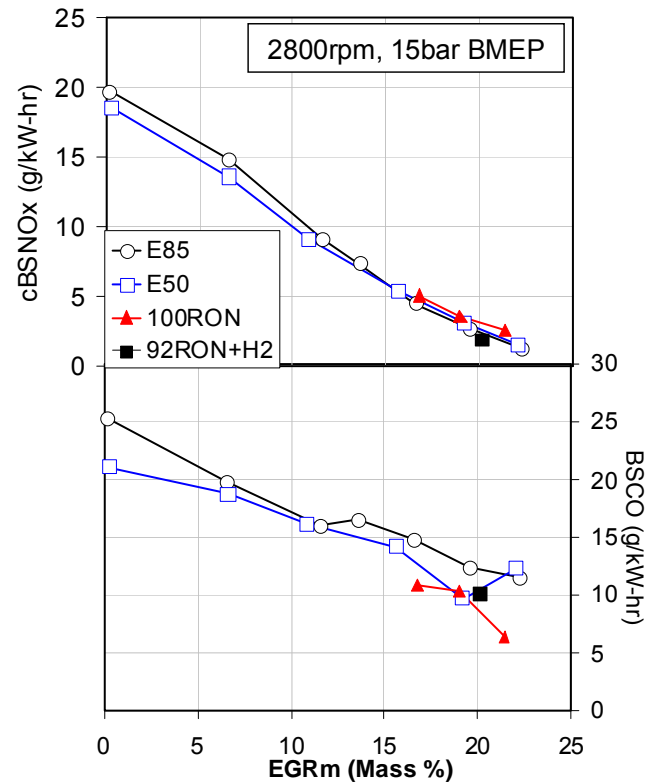


Figure 6. Engine-out Emissions for Selected Fuels with EGR

Engine-out emissions were also improved with EGR. Figure 6 shows brake specific NOx, humidity corrected (cBSNOx), and CO (BSCO) as a function of EGRm. Only data at the target power level is displayed. Increasing EGR to the optimum level decreased engine-out cBSNOx by 60% for E85 and E50. BSCO emissions decreased by ~35%. EGR affected emissions for all fuels in a similar fashion. MBT spark advance was similar for the various fuels at corresponding EGR rates. The linear trend of MBT spark advance at these EGR rates indicated approximately 0.7° advance required for a 1 percentage point change in EGR. Regular gasoline could not operate at MBT as discussed before, but is included in the trend to show the significant impact H₂ addition had on the early burn rate of fuel. Regular gasoline with H₂ had nearly 10° shorter 0-10% mass fraction burn duration (0-10% MFB) compared to the other fuels at 20% EGR. The 0-10% MFB trended with laminar flame speed for the different fuels. Gasoline laminar flame speed (stoichiometric, 1 bar, 300K) is approximately 27cm/s. Ethanol and hydrogen are approximately 42 and 210cm/s, respectively [20].

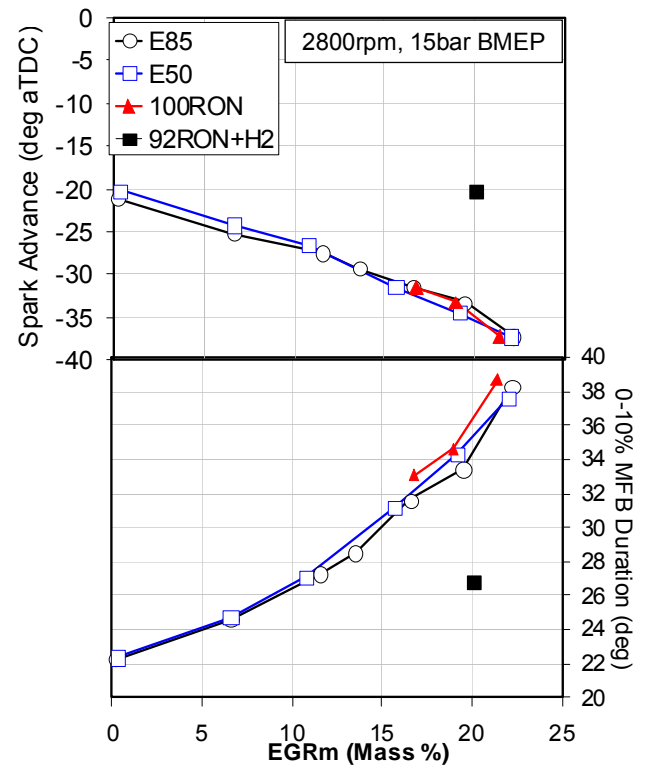


Figure 7. Spark Advance and 0-10MFB Duration for Selected Fuels with EGR

Moderate levels of EGR allowed the target engine to maintain full load capability from E85 to regular gasoline with adequate margin. EGR also improved part and full load fuel consumption.

FULL LOAD COMPARISON TO BASELINE

The primary differences between the flex-fuel engine configuration and baseline was the compression ratio (11:1 / 9:1) and the boost system (supercharged / turbocharged). An empirically derived correction factor was applied to the supercharged data to compare fuel consumption (cBSFC) and thermal efficiency (cBTE) between these two data sets. The correlation data was acquired at 2800rpm, 11:1 compression ratio, and 100RON fuel. The spark timing and MAP followed very similar trends for both boost systems (indicating similar induction dynamics), but there was a step change in the BSFC as seen in figure 8 that was attributed to minor back pressure deviations and the externally driven supercharger. A 3% empirical correction accounted for the testing differences and established a more representative basis for comparison to the stock engine by forcing the supercharged data to match the turbocharged data at 10 bar BMEP while maintain the observed trends.

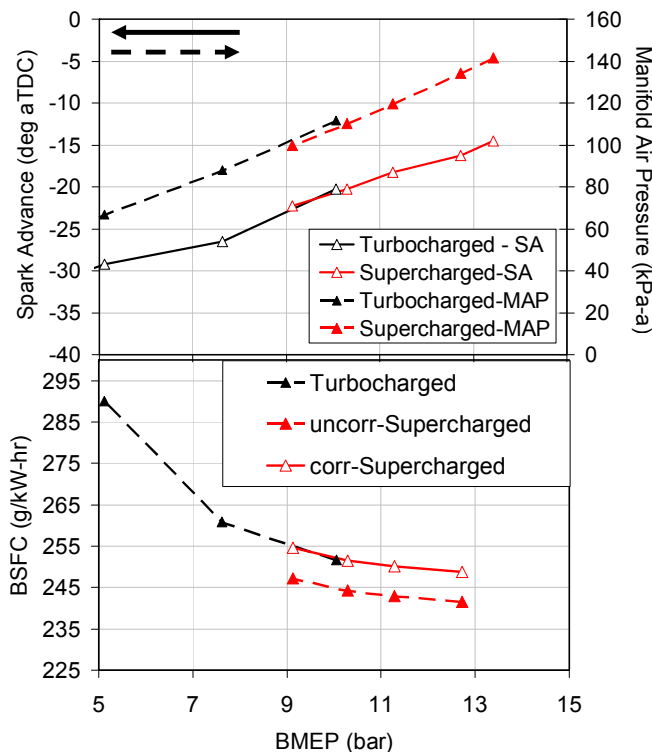


Figure 8. Correction Factor for Comparison to Baseline Performed at 2800rpm and 100RON Gasoline

The baseline data was at wide open throttle from 1200 to 6000rpm with regular gasoline. The equivalence ratio and load for these data sets are shown in figure 9.

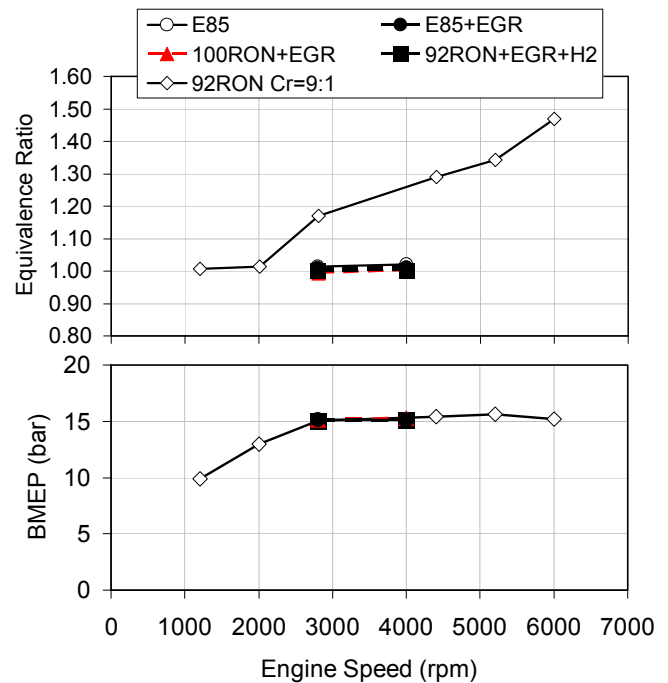


Figure 9. Equivalence Ratio and BMEP Comparison

The target BMEP was achieved at 2800 and 4000rpm. Unlike the base engine, fuel enrichment was not necessary for the eFFV engine developed with EGR. Comparing engine out emissions (figure 10), the eFFV showed a significant decrease in CO due to the ability to maintain stoichiometric AFR. Fuel enrichment coupled with retarded spark timing produced lower NOx emissions for the baseline engine compared to the eFFV which operated at MBT and $\Phi = 1$.

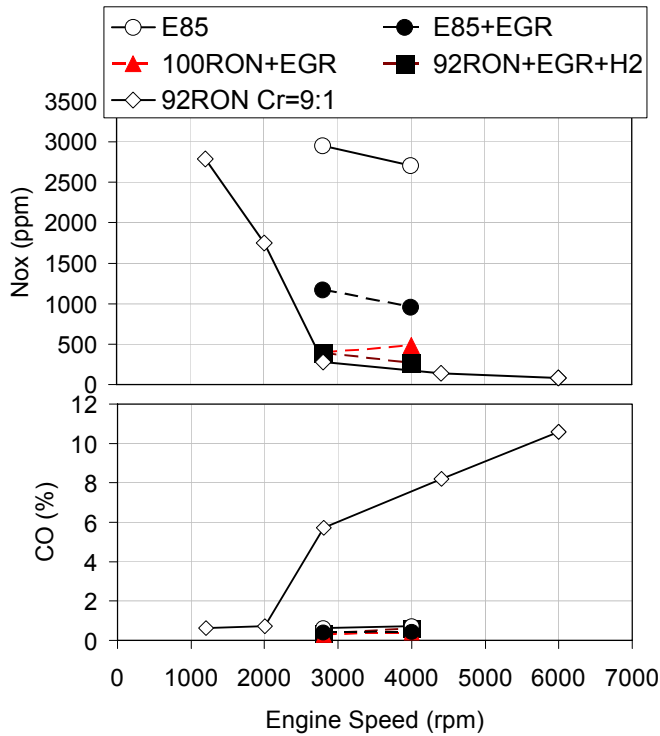


Figure 10. NOx and CO Comparison

Fuel consumption was poor for the baseline engine at full load due to strategies for mitigating knock and exhaust temperatures. Increasing the compression ratio and adding EGR improved full load fuel consumption by nearly 8-10% with regular gasoline, and 20-21% with premium. Fuel consumption of E85 at full load was 9% higher than the baseline engine even though E85 has 25% lower energy content. Thermal efficiency was improved due to the higher compression ratio, MBT timing, and elimination of fuel enrichment. cBTE for E85 and premium gasoline were 9-10 percentage points higher than the baseline data. This data is presented in figure 11. BTE values for the H₂ data was not presented because the heating value of the simulated reformed fuel was not known.

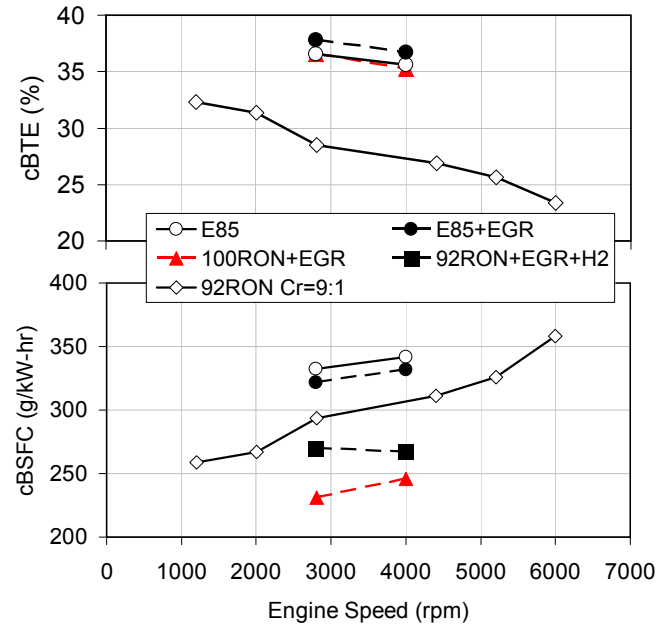


Figure 11. BTE and BSFC Comparison

CONCLUSION

Full and part load eFFV engine performance has been significantly improved by adding cooled EGR to the architecture. Performance was improved by increasing compression ratio, eliminating the need for fuel enrichment, and improved combustion phasing. Performance improvements were identified at 2800 and 4000rpm on fuels ranging from E85 to regular gasoline. Addition of 3% H₂ significantly increased burn rates and stabilized dilute combustion whereby enabling higher specific outputs and improved fuel consumption. Full load BSFC was improved by 20-21% compared to the baseline for premium gasoline, and 8-10% with regular gasoline. Full load BTE improved by 9-10 percentage points with E85, E50, and premium gasoline. Full load fuel consumption was only 9% higher with E85 compared to the baseline engine even though there was 25% difference in the energy content of the fuel. EGR can be used to allow engine architecture to be tuned to the benefits of E85 while maintaining performance, emissions, and fuel economy with gasoline.

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ACRONYMS AND NOTATION

0-10% MFB	0-10% Mass Fraction Burned
AFR	Air Fuel Ratio
aTDC	After Top Dead Center
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
BTE	Brake Thermal Efficiency
CO	Carbon Monoxide
CO ₂	Carbon Dioxide
CoV IMEP	Coeff. Variation of Indicated Mean Effective Pressure
CR	Compression Ratio
E50	50% Ethanol 50% Gasoline
E85	85% Ethanol 15% Gasoline
eFFV	Ethanol Flex Fuel Vehicles
EGR	Exhaust Gas Recirculation
FFV	Generic Alcohol Flex Fuel Vehicle
H ₂	Diatomic Hydrogen
HC	Unburned Hydrocarbons
MBT	Maximum Brake Torque
NO _x	Oxides of Nitrogen
PCP	Peak Cylinder Pressure
PFI	Port Fuel Injection
RON	Research Octane Number
rpm	Revolutions per Minute
Φ	Equivalence Ratio