

Fuel Economy and CO₂ Emissions of Ethanol-Gasoline Blends in a Turbocharged DI Engine

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ABSTRACT

Engine dynamometer testing was performed comparing E10, E20, and E30 splash-blended fuels in a Ford 3.5L EcoBoost direct injection (DI) turbocharged engine. The engine was tested with compression ratios (CRs) of 10.0:1 (current production) and 11.9:1. In this engine, E20 (96 RON) fuel at 11.9:1 CR gave very similar knock performance to E10 (91 RON) fuel at 10:1 CR. Similarly, E30 (101 RON) fuel at 11.9:1 CR resulted in knock-limited performance equivalent to E20 at 10:1 CR, indicating that E30 could have been run at even higher CR with acceptable knock behavior.

The data was used in a vehicle simulation of a 3.5L EcoBoost pickup truck, which showed that the E20 (96 RON) fuel at 11.9:1 CR offers 5% improvement in U.S. EPA Metro-Highway (M/H) and US06 Highway cycle tank-to-wheels CO₂ emissions over the E10 fuel, with comparable volumetric fuel economy (miles per gallon) and range before refueling. The results also indicated that the E30 (101 RON) fuel at 11.9:1 CR provides improvements in CO₂ emissions of 5% on the EPA M/H cycle and 7.5% on the US06 Highway cycle, while volumetric fuel economy was 3% lower on the M/H cycle and approximately equal on the US06 Highway cycle, compared to the baseline E10 fuel at 10:1 CR.

CITATION: Jung, H., Leone, T., Shelby, M., Anderson, J. et al., "Fuel Economy and CO₂ Emissions of Ethanol-Gasoline Blends in a Turbocharged DI Engine," *SAE Int. J. Engines* 6(1):2013, doi:10.4271/2013-01-1321.

INTRODUCTION

It is well known that ethanol blended into gasoline improves knock behavior, especially for turbocharged direct injection engines. Knock is the main limit on compression ratio, which is a key factor in engine efficiency. Previous studies [1,2,3,4] quantified or estimated knock benefits of various ethanol blends, but did not include detailed analysis of changes in vehicle fuel economy and CO₂ emissions.

At present, most fuel in the U.S. contains 10%v ethanol (E10). Given that future ethanol availability and use in U.S. fuel will likely be greater than at present [1,3,5], future ethanol-gasoline blends with higher ethanol content will probably be required. To guide important decisions about such future fuels, it is critical to quantify potential benefits for a range of ethanol blends. Should additional ethanol be sold as E85 or some intermediate blend between E10 and E85? Should ethanol be "splash blended" into today's E10 blendstock for improved octane rating, or should it be "match blended" so that the final fuel has the same octane rating as today's E10? Should blends higher than E10 be used in flex-fuel vehicles or in vehicles optimized to take advantage of the improved knock behavior of those ethanol blends?

In this project, mid-level ethanol blend fuels (splash-blended E10, E20, and E30) were tested with a turbocharged DI engine at 10.0:1 and 11.9:1 CR. The tests were run with both CRs on the same engine, via a piston change. The boundary conditions and scope of data enabled detailed vehicle simulations to quantify expected changes in fuel economy and tank-to-wheels CO₂ emissions.

The goal of this project was to quantify potential vehicle fuel economy and CO₂ benefits for engines that are optimized for mid-level ethanol splash blends (with higher octane ratings than current regular-grade U.S. E10 gasoline). This study is not intended to advocate a specific ethanol blend, but rather to provide data for comparison to other options, such as E85 and mid-level matched blends. A follow-up study is planned with mid-level matched blends on the same engine.

METHODS

Engine Hardware

A production 3.5L V6 gasoline turbocharged DI engine with twin independent variable camshaft timing was used for this testing. Modified pistons enabling a high CR (11.9:1) were used first. After the 11.9:1 data was taken, the pistons were replaced to provide the baseline 10.0:1 CR data for



comparison. The bore was 92.5mm, the stroke was 86.7mm, and displacement was 3.496L. The variable camshaft timing calibration was unchanged for the 10:1 and 11.9:1 CR at each speed-load combination and was directly from the baseline production 3.5L engine. The intake cam duration was 248° and the exhaust cam duration was 263°. During engine testing, the fuel rail pressure and start of injection (SOI) timing were from the baseline production calibration and were not varied for the different fuels or compression ratios. Previous injection timing sweeps at part load showed that production SOI timing had a range where smoke, emissions and efficiency were nearly constant especially at later SOI timing. A small change in the end of injection due to a change in fuel would have only a minor impact on efficiency and emissions because of the robust SOI window.

Fuels

Three fuels were evaluated: E10, E20, and E30. The E10 had octane ratings typical of U.S. regular-grade E10 gasoline (87 AKI, 91 RON). The two mid-level blends (E20 and E30) were prepared by splash blending denatured ethanol (dE97) into the E10 and their octane ratings increased accordingly.

The fuel properties are summarized in [Table 1](#) (additional properties and test method information are provided in the [Appendix](#)). Fuel properties and compositions were measured by both Paragon Laboratories, Inc. (Livonia, MI) and a BP laboratory (Naperville, IL) with good agreement between laboratories. The measured speciated hydrocarbon composition in the E10 and the measured contents of ethanol (and trace methanol and water) in the E10, E20, E30, and dE97 were used to calculate harmonized values for net heating value (NHV), hydrogen-to-carbon ratio (H/C), oxygen-to-carbon ratio (O/C), and heat of vaporization (HoV) for the three test fuels. This approach provided property values that were internally consistent between the fuels and these interrelated properties. The calculated property values were consistent with the measured property data from the two laboratories within test method reproducibility; the calculated values were used for all further analysis.

Test Plan

The engine limits were as follows:

- CA50 < 30 °ATDC
- Lambda > 0.75
- Peak pressure + 3 * sigma < 120 bar
- Turbine inlet temperature < 950 °C
- Turbine speed < 205,000 rpm

where CA50 is the crank angle location of 50% mass fraction burned in °ATDC, and lambda is the ratio of the actual air/fuel ratio divided by the stoichiometric air/fuel ratio.

Table 1. Fuel Properties

Fuel	E10	E20	E30
Ethanol (%v)	10.2	20.4	31.5
NHV (MJ/kg)	41.5	39.7	37.7
HoV (MJ/kg)	0.41	0.48	0.55
H/C (mole)	2.00	2.08	2.18
O/C (mole)	0.035	0.073	0.119
Specific gravity	0.743	0.749	0.755
RON	90.8	96.2	100.7
MON	84.1	86.1	87.9
AKI	87.4	91.1	94.3
Sensitivity	6.7	10.1	12.8
Δ NHV (MJ/kg basis)	Base	-4.36%	-9.07%
Δ NHV (MJ/L basis)	Base	-3.64%	-7.63%

When testing, data points were taken at a given engine speed with gradually increasing brake mean effective pressure (BMEP) until the BMEP could no longer be increased due to the above engine constraints, as described in [2]. At high BMEP levels for some fuels and engine speeds, engine operation was also constrained by the available boost pressure supplied by the turbocharger system, pre-ignition, compressor surge, or the flow capability of the direct injection fuel pump.

Test points were run at the minimum spark advance for the best torque (MBT) spark timing if the engine was not knock-limited, or at borderline knocking conditions. The test plan consisted of load sweeps at various engine speeds and an engine speed sweep. The six sweeps tested with each fuel were as follows:

- 1500 rpm, 2000 rpm, and 2500 rpm with BMEP points of 2.62, 5, 7, 9, 12, 14... Δ2 bar
- 5000 rpm starting at lambda = 1 and going rich with BMEP points of 9, 11, 13... Δ2 bar
- 1000 rpm maintaining lambda = 1 with BMEP points of 5, 6, 7... Δ1 bar
- 2500-5000 rpm max load maintaining lambda = 1, increasing by Δ500 rpm.

The boundary conditions were as follows:

- Charge air cooler temperature = 38 °C
- Engine coolant out temperature = 90 °C
- Maximum engine oil temperature = 130 °C

[Figure 1](#) shows a plot of planned test points for both 10 and 11.9:1 CR, with the dots indicating BMEP points at the described engine speeds and the line indicating the observed upper limit of stoichiometric operation for the various speeds

using the E10 fuel at 10:1 CR. As described above, BMEP sweeps were performed at 1000, 1500, 2000, 2500 and 5000 rpm. The maximum attainable BMEP point at each speed was different for each combination of fuel and CR.

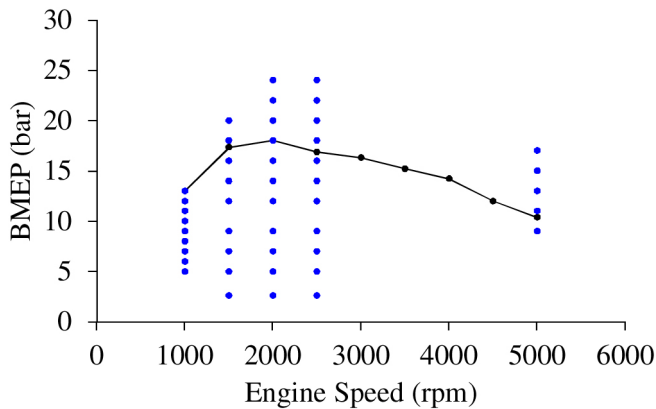


Figure 1. Planned test points for all fuels (symbols) and maximum load at stoichiometry (line) for E10 at 10:1 CR

RESULTS

Engine Dynamometer Test Results

1500, 2000, and 2500 rpm BMEP sweeps

At light load, the engine operated at MBT combustion phasing for each fuel and CR, with CA50 approximately 7-8 °ATDC as seen in Figure 2. As the load was increased, the engine became knock-limited and in response, the spark was retarded as shown by the retarded CA50 values. Spark retard continued as BMEP increased until the CA50 limit of 30 °ATDC was reached. At that point, enrichment was added and boost was increased to enable further increase in BMEP while maintaining CA50 at the 30 °ATDC limit. Enrichment was added until the 0.75 lambda limit was reached or one of the other engine limits (pre-ignition or compressor surge) was encountered. Although peak pressure and turbine inlet temperatures increased as BMEP increased, their limits did not come into effect at 1500 rpm.

As expected, E10 was the first fuel to encounter knock-limited operation, followed by E20 and then E30 as seen in Figure 2 with the last MBT points being 7 bar, 9 bar, and 14 bar BMEP, respectively, at 10:1 CR. This increase in BMEP was due to the increasing RON and charge cooling benefit with higher ethanol content. Under the most retarded spark conditions (CA50 = 30°ATDC), the higher ethanol content fuels achieved progressively higher BMEP without requiring enrichment. As shown in Figure 2, at 10:1 CR the last stoichiometric points before enrichment was added were 9 bar and 18 bar BMEP for E10 and E20, respectively, with E30 reaching 27 bar with a CA50 of 27° ATDC before becoming limited by preignition. This increase was due not only to chemical octane (RON) and charge cooling benefits, but also to the higher sensitivity of the E20 and E30 fuels. As

described in [2], higher sensitivity of the fuel provides additional knock resistance under retarded combustion phasing conditions. At MBT conditions the BMEP was 7 bar with E10 and increased to 14 bar with E30. When the combustion phasing was retarded, the BMEP increased from 9 bar with E10 to higher than 27 bar (at CA50=27, limited by preignition) with E30. For a given CA50 (after the MBT region) the higher sensitivity of the E20 and E30 fuels allowed for higher BMEP at a given speed and a slower increase in CA50 for increasing BMEP (lesser slope in CA50 vs. BMEP), as reported previously [2].

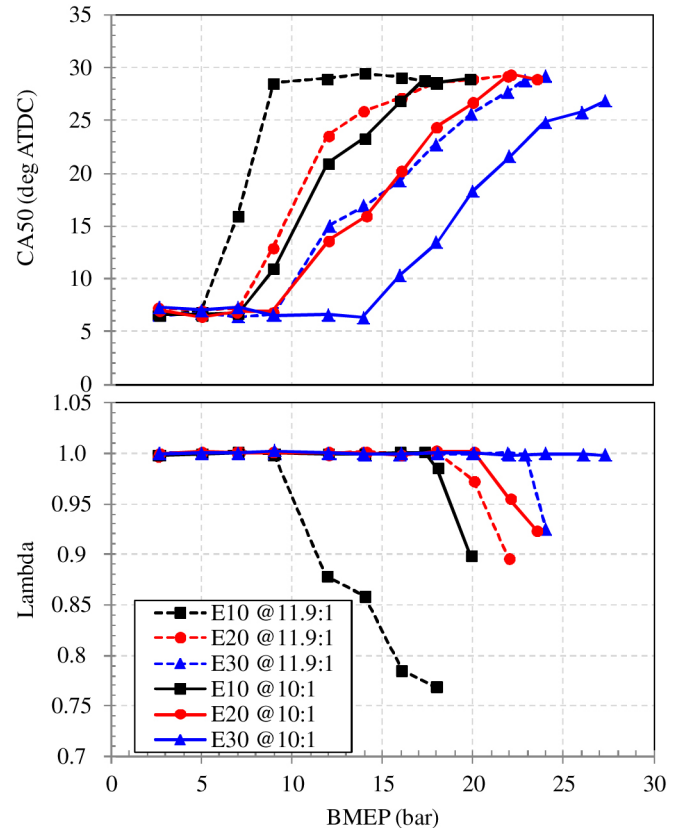


Figure 2. Combustion phasing and enrichment results for load sweeps at 1500 rpm

The use of a higher CR caused the engine to become more knock-limited with each fuel than for the baseline CR, resulting in lower BMEP points for MBT and more retarded combustion phasing prior to enrichment. Notably, the CA50 vs. BMEP data for E20 at 11.9:1 CR were very similar to that for E10 at 10:1 CR in the knock-limited region during stoichiometric operation. Also, the data for E30 at 11.9:1 CR was similar to that of E20 at 10:1 CR. Based on this comparison, the 10%v additional ethanol content (and associated change in fuel properties) provided similar knock relief as reducing CR from 11.9 to 10 in this engine. Stated another way, the additional 10%v ethanol content enables an additional 1.9 CR with similar knock-limited performance.

All the fuel-CR combinations except for E10 at 11.9:1 CR experienced a blow-through scavenging effect at high loads

when intake pressure was higher than the exhaust pressure during high overlap, allowing air to pass through the cylinder during the high valve overlap. This scavenging effect lowered the residual gas content and increased the fresh charge making the engine less knock-limited.

At light loads, it was possible to run each fuel at MBT and thermal efficiency improved as BMEP increased as shown in Figure 3. When spark was retarded at higher loads, thermal efficiency decreased, causing brake specific fuel consumption (BSFC) to increase. After the CA50 limit was reached and enrichment began, thermal efficiencies continued to decrease leading to considerably higher BSFC for the highest loads.

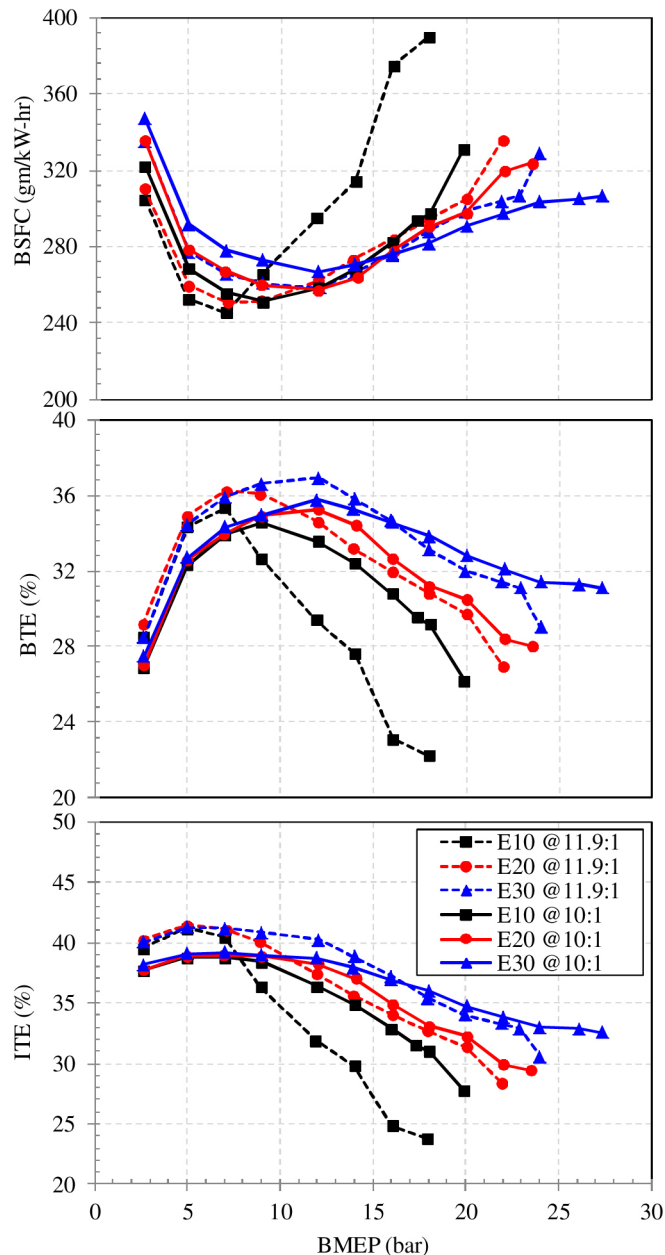


Figure 3. Efficiency results for load sweeps at 1500 rpm

The increased ethanol content (and accompanying higher octane ratings, sensitivity, and charge cooling) allowed MBT and stoichiometric operation to be maintained at higher BMEP, which in turn led to broader load ranges of high BSFC and thermal efficiency (Figure 3). Camshaft timing was not modified for the fuels, so at each MBT point, brake thermal efficiency (BTE) and indicated thermal efficiency (ITE) values were very similar for all fuels (where the thermal efficiency is defined as the ratio of work output to the fuel energy input per cycle). Thermal efficiencies in the MBT region exhibited two distinct groupings associated with the two CRs, with 11.9:1 CR being more efficient than the baseline 10:1 CR. As well, the fuel NHV differences shown in Table 1 are evident in the BSFC values in the MBT regions for the three fuels. By definition, BSFC is fuel mass flow per unit power. The BSFC plot in Figure 3 not only reflects the efficiency difference of the two CRs but also the NHV differences between fuels. Because E30 had the lowest NHV, this fuel showed the highest BSFC within the MBT region. When the spark timing was retarded due to borderline knock and when enrichment was added, BTE decreases and BSFC increases.

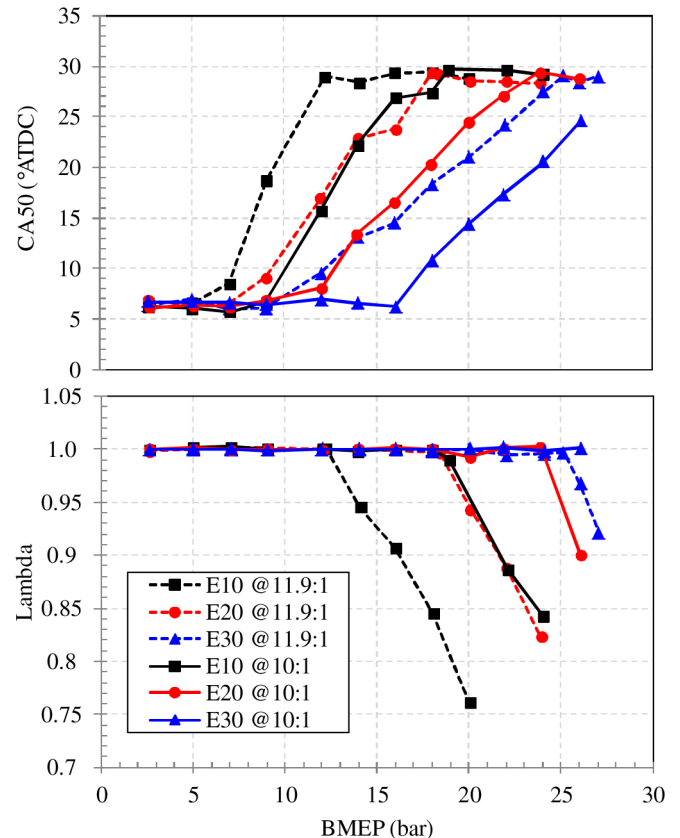


Figure 4. Combustion phasing and enrichment results for load sweeps at 2000 rpm