

Effects of Fuel Octane Rating and Ethanol Content on Knock, Fuel Economy, and CO₂ for a Turbocharged DI Engine

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ABSTRACT

Engine dynamometer testing was performed comparing fuels having different octane ratings and ethanol content in a Ford 3.5L direct injection turbocharged (EcoBoost) engine at three compression ratios (CRs). The fuels included midlevel ethanol "splash blend" and "octane-matched blend" fuels, E10-98RON (U.S. premium), and E85-108RON.

For the splash blends, denatured ethanol was added to E10-91RON, which resulted in E20-96RON and E30-101 RON. For the octane-matched blends, gasoline blendstocks were formulated to maintain constant RON and MON for E10, E20, and E30.

The match blend E20-91RON and E30-91RON showed no knock benefit compared to the baseline E10-91RON fuel. However, the splash blend E20-96RON and E10-98RON enabled 11.9:1 CR with similar knock performance to E10-91RON at 10:1 CR. The splash blend E30-101RON enabled 13:1 CR with better knock performance than E10-91RON at 10:1 CR. As expected, E85-108RON exhibited dramatically better knock performance than E30-101RON.

The data were used in a vehicle simulation of a 3.5L EcoBoost F150, which showed that E20-96 RON at 11.9:1 CR offers 5% improvement in tailpipe CO₂ emissions and 1% improvement in miles per gallon (MPG) fuel economy relative to E10-91RON at 10:1 CR. E30-101 RON at 13:1 CR in this vehicle yielded 6–9% improvement in CO₂ emissions and 2% worse to 1% better MPG fuel economy, depending on the drive cycle.

The match blend fuels resulted in no opportunity for improved efficiency, and degradation of MPG fuel economy due to reduced heating value per volume.

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INTRODUCTION

At present, most gasoline in the U.S. contains 10%v ethanol (E10). Given that future ethanol availability and its use in U.S. fuel could be greater than at present [2, 3, 4], it is important to quantify the potential effects of increasing ethanol for a range of ethanol-gasoline blends to provide guidance on the following questions:

- Would additional ethanol be best used as E85 or some mid-level blend between E10 and E85?
- If a mid-level blend, should ethanol be "splash blended" into current E10 blendstock for improved octane rating, or should it be "octane-matched blended" so that the finished fuel has the same octane rating as today's regular-grade E10?

- If a mid-level splash blend, what is the potential for using the fuel in vehicles optimized for the increased octane rating rather than in flex-fuel vehicles?

Current E10 fuel is prepared by blending denatured ethanol into a hydrocarbon "blendstock for oxygenate blending" (BOB) formulated such that the resulting E10 fuel meets applicable requirements. Accordingly, the U.S. fuel industry has taken an "octane-matched blend" approach for octane ratings, such that regular-grade E10 fuel meets the same octane rating specifications as the prior regular-grade E0 gasoline. Given that ethanol has high octane ratings (109 RON, 90 MON), this means that the octane ratings for the E10 blendstock are the same as the prior E0 gasoline, which permits the fuel supplier

Blendstock volatility, primarily Reid vapor pressure (RVP), also needs to be managed, though assisted by a 7 kPa (1 psi) waiver for E10 in most regions. Similarly, blendstocks for mid-level blends would be formulated such that the blends meet applicable requirements, with possible differences in octane rating (higher for splash blends or unchanged for match blends).

Knock imposes a fundamental limit on key factors which affect engine efficiency, such as compression ratio (CR) which directly affects engine efficiency, and full load brake mean effective pressure (BMEP) which affects downsizing and downspeeding potential. Several previous studies [3, 4, 5, 6, 7, 8, 9] quantified or estimated knock benefits of various ethanol-gasoline blends, but did not include detailed analysis of changes in vehicle fuel economy and CO₂ emissions.

The goal of this project was to quantify potential vehicle fuel economy and CO₂ benefits for engines where the CR is optimized for mid-level ethanol blends. This paper expands upon a prior paper, SAE 2013-01-1321 [1], which documented changes in vehicle fuel economy and CO₂ emissions enabled by splash blending denatured ethanol with E10-91RON to obtain E20-96RON and E30-101RON. For completeness, certain critical data contained in [1] are repeated in this paper. This paper also includes data obtained at a higher CR (13:1) and using two octane-matched blend fuels (identified simply as “match blends” in this paper), E20-91RON and E30-91RON. For reference, results are also included for E85-108RON and for E10-98 RON premium-grade gasoline.

METHODS

Engine Hardware

A production 3.5L gasoline turbocharged direct injection (GTDI) engine (Table 1) with twin independent variable camshaft timing (TiVCT) was used for this study. Prototype pistons were designed for CRs of 11.9:1 and 13.0:1, in addition to the production baseline CR of 10.0:1. In the first part of the study documented in [1], tests were run with CRs of 10.0:1 and 11.9:1 on the same engine. For the second part of the study tests were run with CRs of 10.0:1, 11.9:1, and 13.0:1 on a second engine.

The production TiVCT calibration was used to control the cam positions for the baseline E10-91RON fuel and 10.0:1 CR. Those positions were noted as a function of engine speed and brake mean effective pressure (BMEP), and then used for all subsequent fuels and CRs at the same speed and BMEP combinations. The implications of not optimizing the TiVCT calibration for each fuel are discussed in a later section of this paper.

The fuel rail pressure and start of injection (SOI) timing from the baseline production calibration were also used and were not varied for the different fuels or CRs. Previous injection timing sweeps at part load showed a range of SOI timing around the production timing where smoke, emissions and efficiency were nearly constant, especially at later SOI timing. Thus, a small change in the end of injection (EOI) due to a change in fuel ethanol content from 10 to 30%v was expected to have only a minor impact on efficiency and emissions because of the robust SOI window. The change in EOI with E85 was more substantial, but did not appear to adversely affect efficiency or emissions.

Table 1. 3.5L engine specifications.

Engine layout	60° V6
Bore	92.5 mm
Stroke	86.7 mm
Compression ratios	10.0, 11.9, 13.0:1
Intake camshaft duration	248°
Exhaust camshaft duration	263°
Direct injector location	Side

Fuels

Seven fuels were evaluated: current regular-grade E10 fuel (E10-91RON), current premium-grade E10 fuel, E20 splash and match blends, E30 splash and match blends, and E85. The E10-91RON had octane ratings typical of U.S. regular grade E10 gasoline (87-88 AKI, 91-92 RON). The two mid-level splash blends were prepared by adding denatured ethanol to the E10-91RON in appropriate quantities and the resulting octane ratings increased to 96 RON for E20 and 101 RON for E30. These fuels are designated as E20-96RON and E30-101RON. The two mid-level match blends were prepared by blending denatured ethanol into hydrocarbon blendstocks that were formulated by BP Products North America Inc. to maintain approximately constant RON and MON after the ethanol was added, requiring lower octane ratings for the blendstock fraction. These match blend fuels are designated as E20-91RON and E30-91RON.

Fuel properties for the splash blends are summarized in Table 2. Fuel properties are summarized in Table 3 for the other fuels used in this study: 91RON match blends, E10-98 RON premium, and E85-108RON. (Additional properties and test method information are provided in Appendix 1.) Fuel properties and compositions were measured by Gage Products Co., Paragon Laboratories, Inc. (Livonia, MI), and/or a BP laboratory (Naperville, IL), with good agreement between laboratories.

Distillation curves for the fuels are shown in Figure 1. E30-91RON included a substantially greater amount of a high boiling range hydrocarbon fraction as part of maintaining 91 RON with 30%v ethanol.

Table 2. Fuel properties of the splash blend test fuels.

Fuel	E10-91RON	E20-96RON	E30-101RON
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
Carbon intensity (gCO ₂ /MJ)	72.7	72.5	72.4
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	-4.4%	-9.1%
Δ NHV (MJ/L basis)	Base	-3.6%	-7.6%

Table 3. Fuel properties of the other test fuels.

Fuel	E10-91RON	E20-91RON	E30-91RON	E10-98RON	E85-108RON
Ethanol (%v)	10.0	20.5	29.5	9.8	84.3
NHV (MJ/kg)	42.0	40.1	38.6	42.5	29.0
HoV (MJ/kg)	0.41	0.48	0.54	0.41	0.861
H/C (mole)	2.11	2.11	2.20	2.18	2.89
O/C (mole)	0.035	0.075	0.111	0.035	0.415
Specific gravity	0.735	0.749	0.760	0.725	0.777
Carbon intensity (gCO ₂ /MJ)	71.3	71.6	71.3	70.2	70.4
RON	91.8	90.6	90.7	99.0	~108
MON	84.1	83.2	82.7	91.4	~90
AKI	87.9	86.9	86.7	95.2	~99
Sensitivity	7.7	7.4	8.0	7.6	~18
Δ NHV (1) (MJ/kg basis)	+1.2%	-3.4%	-7.0%	+2.3%	-30.1%
Δ NHV (1) (MJ/L basis)	0.0%	-2.6%	-5.0%	-0.3%	-26.9%

(1) Relative to E10-91RON in Table 2.

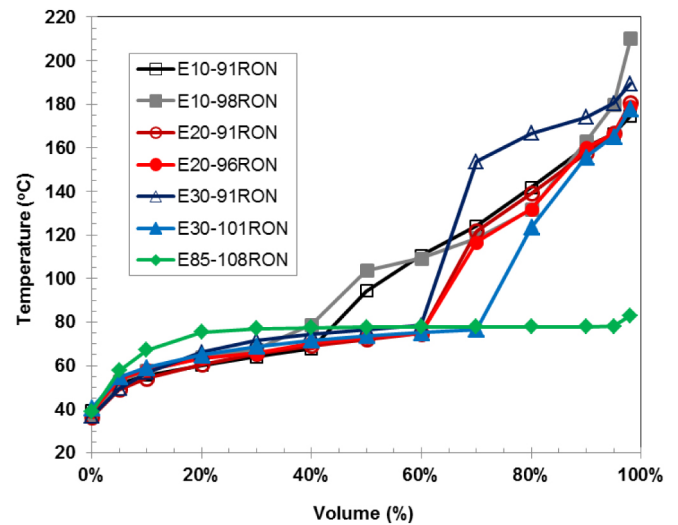


Figure 1. Distillation curves for the test fuels.

Test Plan

The test plan consisted of load sweeps at various engine speeds plus an engine speed sweep. The six sweeps tested with each fuel and CR combination were as follows:

- 1000 rpm maintaining $\lambda = 1$ with BMEP points of 5, 6, 7... $\Delta 1$ bar
- 1500 rpm, 2000 rpm, and 2500 rpm with BMEP points of 2.62, 5, 7, 9, 12, 14... $\Delta 2$ bar
- 2500-5000 rpm with maximum BMEP at $\lambda = 1$ at increments of 500 rpm
- 5000 rpm starting at $\lambda = 1$ and decreasing λ as required to maintain turbine inlet temperature at 950 °C with BMEP points of 9, 11, 13... $\Delta 2$ bar

Where λ is the ratio of the actual air-fuel ratio divided by the stoichiometric air-fuel ratio and BMEP is brake mean effective pressure.

Test points were run at the minimum spark advance for best torque (MBT) spark timing if the engine was not knock-limited or otherwise with the spark timing retarded to borderline knocking conditions. Borderline knocking was determined both by audible detection (with one test technician for the duration of the testing) and by using a proprietary cylinder pressure-based knock intensity metric. Using these two methods in combination has demonstrated very consistent results.

The operational limits that were used during engine testing were as follows:

- CA50 < 30 °aTDC
- $\lambda > 0.75$
- Peak pressure + 3 * σ < 120 bar
- Turbine inlet temperature < 950 °C
- Turbine speed < 205,000 rpm

Where CA50 is the crank angle location of 50% mass fraction burned expressed in degrees after TDC ($^{\circ}$ aTDC), and lambda is the ratio of the actual air-fuel ratio divided by the stoichiometric air-fuel ratio. 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 [5]. For some fuels and engine speeds, the maximum BMEP was also constrained by the available boost pressure supplied by the turbocharger system, compressor surge, pre-ignition (PI), or the flow capability of the direct injection fuel pump.

It should be noted that PI limits were not precisely determined during this study. If a single PI event was observed during a load sweep, then the load sweep was terminated. Due to the stochastic and intermittent nature of low speed PI, the PI limits noted in this paper should not be regarded as definitive.

The boundary conditions during testing were as follows:

- Charge air cooler outlet temperature = 38°C ($\pm 1^{\circ}\text{C}$)
- Engine coolant out temperature = 90°C ($\pm 1^{\circ}\text{C}$)
- Maximum engine oil temperature = 130°C

Figure 2 shows an example of test points using E10-91RON at 10:1 CR, with the dots indicating BMEP points at the various engine speeds and the line indicating the upper limit of stoichiometric operation. As described above, BMEP sweeps were performed at 1000, 1500, 2000, 2500 and 5000 rpm. The maximum attainable BMEP at each speed was different for each combination of fuel and CR.

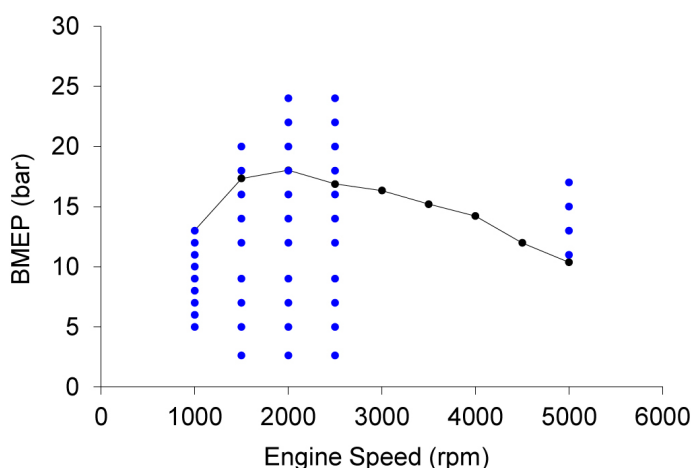


Figure 2. Test points (symbols) and maximum load at stoichiometry (line) for E10 at 10:1 CR.

RESULTS

Engine Dynamometer Test Results

91-RON E10, E20, and E30 Match Blends at 10:1 CR

This section presents results obtained at 10:1 CR with the match blend fuels (E20-91RON and E30-91RON) compared to the baseline E10-91RON of Table 3. Combustion phasing and lambda for load sweeps at 1500 rpm are shown in Figure 3.

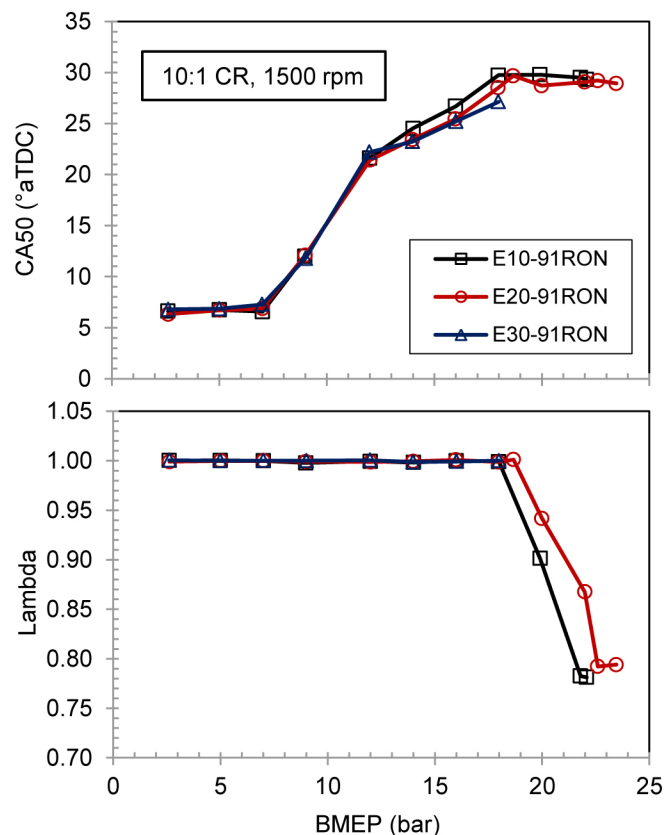


Figure 3. Combustion phasing and lambda for load sweeps at 1500 rpm at 10:1 CR for the match blends.

At light load, the engine was not knock-limited and was operated at MBT combustion phasing, with CA50 approximately 7° aTDC as seen in Figure 3. As the load was increased, the engine became knock-limited and the spark timing was retarded as shown by the retarded CA50 values. Spark retard increased as BMEP was increased until the CA50 limit of 30° aTDC was reached. At that point, enrichment was added as boost was further increased to enable additional 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 (PI, compressor surge, etc.) was encountered. Although peak cylinder pressure and turbine inlet temperature increased as BMEP was increased, their limits were not encountered at 1500 rpm.

As shown in Figure 3, the match blends E20-91RON and E30-91RON resulted in the same knock limits as E10-91RON except at very retarded combustion phasing where there was a slight improvement with higher ethanol content. The data for

E30-91RON were limited by pre-ignition (PI) above 18 bar BMEP. This fuel was more prone to low speed PI than the other fuels, likely due to the greater amount of high boiling point hydrocarbons that were present in the blendstock to maintain 91 RON with 30%v ethanol. As shown in Figure 1, the distillation temperatures of the T70 to T95 fraction were 10°C to 30°C higher than the E10 and E20 fuels and 15°C to 75°C higher than the E30-101 RON fuel. Other E30-91 RON fuel formulations may show different PI behavior.

Results for the load sweeps at 2000 rpm and 2500 rpm are shown in Figure 4 and Figure 5, respectively. Similar to 1500 rpm, the match blends E20-91RON and E30-91RON exhibited no improvement in knock limit compared to the baseline E10-91RON fuel at MBT and moderately retarded combustion phasing. These fuels did exhibit a slight improvement in knock limit at higher loads and very retarded combustion phasing, and a slightly higher achievable load with stoichiometric operation.

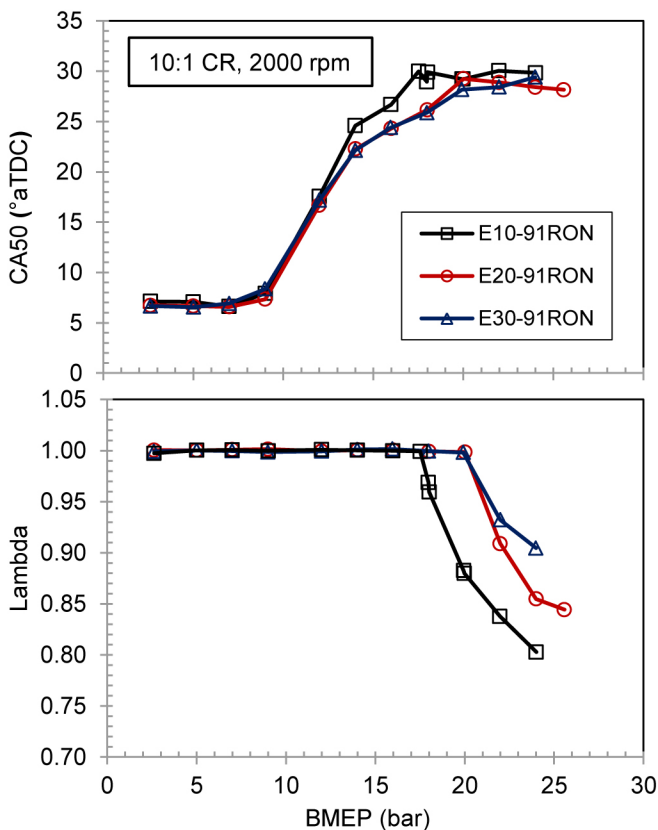


Figure 4. Combustion phasing and lambda for load sweeps at 2000 rpm at 10:1 CR for the match blends.

In a single-cylinder engine study of a large number of ethanol-gasoline blends [5], it was concluded that the knock limit of a fuel is determined by the engine conditions and by the fuel's RON, sensitivity, and HoV. Since both the RON and MON of the fuels were matched in the present study, the sensitivity as measured by RON minus MON was also the same. The HoV of the fuels increased moderately (~30%) as ethanol content was increased from E10 to E30 (Table 3). At least a portion of the charge cooling effect on knock due to fuel evaporation is

included in the measured RON of the fuel. (This occurs because the inlet air temperature is set upstream of the carburetor in the RON test method [10], and any fuel evaporation that occurs as a result of heat transfer from the fresh air results in a decrease in charge temperature.) The similarity in the engine results at MBT and with modest combustion phasing retard indicates that the charge cooling occurring in this DI engine was fully accounted for in the RON values for these match blend fuels. This observation is limited to ethanol content up to E30, and is likely not the case for higher ethanol blends [5, 11].

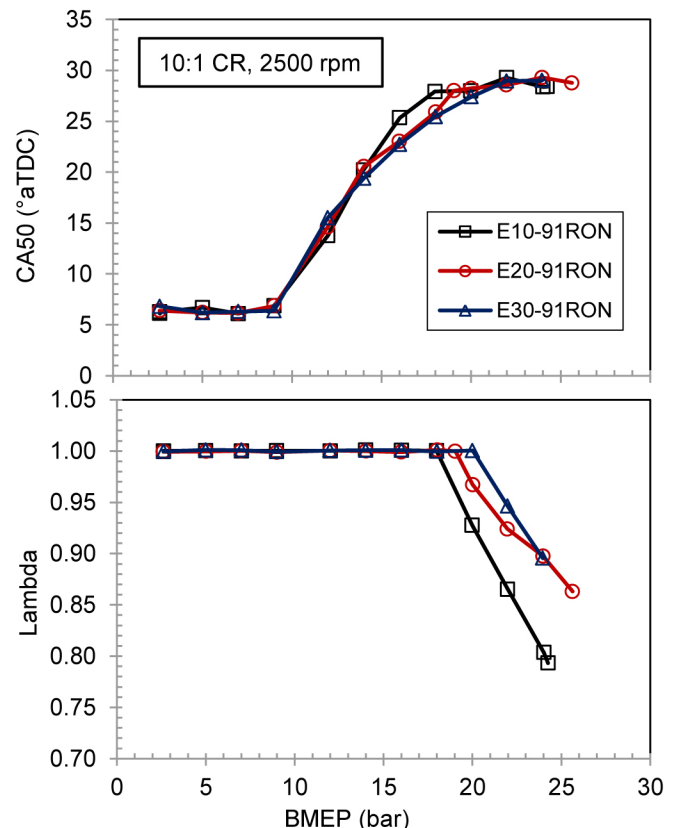


Figure 5. Combustion phasing and lambda for load sweeps at 2500 rpm at 10:1 CR for the match blends.

At very retarded combustion phasing, E20-91RON and E30-91RON exhibited a modest improvement in knock limit compared to E10-91RON. A possible explanation is that unburned gas temperatures are cooler under these conditions than in the RON test, and even though the fuels have the same sensitivity (as measured by RON minus MON), the mid-level ethanol blends could have slower autoignition kinetics at these lower unburned gas temperatures, as illustrated in the hypothetical representation shown in Figure 6 (adapted from [12]).

These results are in general agreement with the match blend testing documented in [5] and [8]. In [8], 95RON E20 fuel was compared to 95RON E0 fuel (RON-matched blends) using a 1.8L GTDI engine with 9.3:1 CR. Sensitivity of the fuel as

measured by RON minus MON increased moderately from 7.3 for E0 to 10.4 for E20. No difference in knock-limited combustion phasing at full load was observed for these fuels.

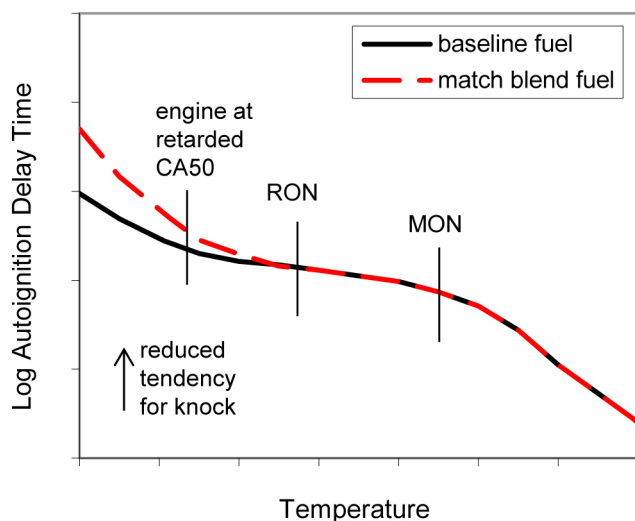


Figure 6. Hypothetical autoignition delay time vs. unburned gas temperature.

In [5], E0, E10, and E20 match blends of approximately 93 RON also exhibited no significant difference in knock limit over the range of combustion phasing. The sensitivity of these blends increased only slightly (7.5 to 8.8) as ethanol content was increased. However, for E0, E10, E20, and E30 match blends of approximately 99 RON, the knock limit at retarded combustion phasing improved significantly, apparently due to increased sensitivity (from 9.5 to 12.4) and HoV associated with higher ethanol content. As discussed in [5], retarded combustion phasing results in a significant decrease in peak unburned gas temperature. Because the dependence of autoignition delay is exponential with temperature, retarded combustion phasing amplifies the HoV charge cooling effect on knock as the sensitivity of the fuel is increased (illustrated in Figure 6).

Thus, the observed knock resistance of mid-level ethanol-gasoline blends with equal RON depends on the sensitivity of the fuels, with two likely cases: 1) If the mid-level match blend has sensitivity approximately equal to that of the baseline fuel (both RON and MON are matched as in the study in this paper), then no difference is expected in the knock limit near MBT and with moderate combustion phasing retard. A slight to moderate improvement at very retarded combustion phasing may occur, depending on the temperature dependence of the auto-ignition reaction kinetics at temperatures below those corresponding to the RON test. 2) If the mid-level match blend has greater sensitivity than the baseline fuel (RON is matched but MON decreases, as in [5]), then no change in the knock limit near MBT is expected, while a substantial improvement in the knock limit will be likely at retarded combustion phasing due to slower auto-ignition reaction kinetics at temperatures below those corresponding to the RON test.

Brake specific mass fuel consumption (BSFC) and brake specific volumetric fuel consumption (BSFC by volume) for the data of Figure 4 are shown in Figure 7. BSFC by volume expressed as liters/kWh provides an indication of relative fuel consumption in liters per 100 kilometers (L/100 km). Since fuel economy in miles per gallon (MPG) is inversely related to fuel consumption, BSFC by volume also provides an inverse indication of relative MPG fuel economy.

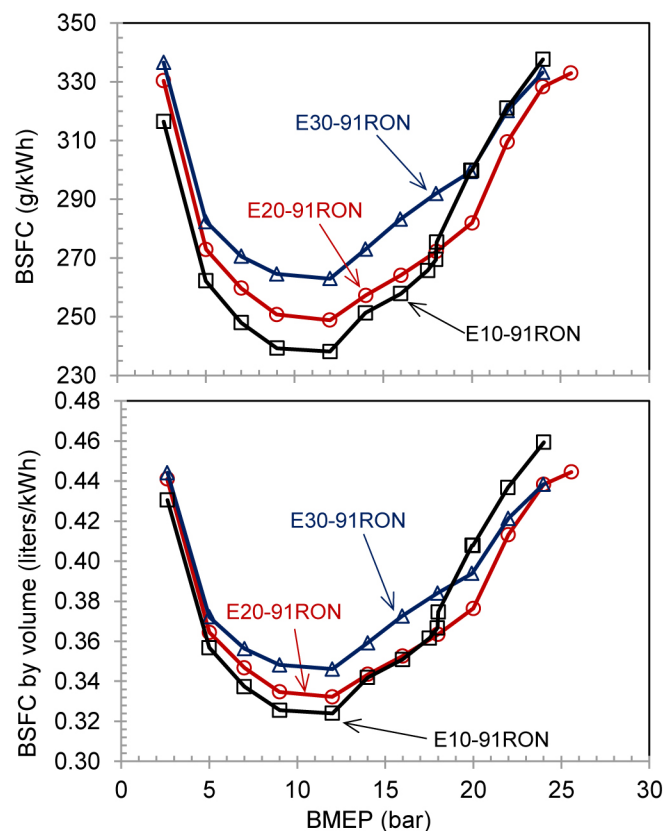


Figure 7. Brake specific fuel consumption by fuel mass and by fuel volume for load sweeps at 2000 rpm at 10:1 CR.

As shown in Figure 7, BSFC increases as ethanol content is increased, because the net heating value (NHV) of ethanol is less than that of gasoline. Since ethanol has slightly higher density than gasoline, the increase in BSFC by volume is slightly less than for conventional BSFC by mass. Nevertheless, there is a NHV-related BSFC by volume degradation over most of the load range with increasing ethanol content for these match blend fuels. Similar results were obtained at 1500 rpm and 2500 rpm.

Tailpipe equivalent brake specific CO_2 (BSCO_2) emissions for the data at 2000 rpm are shown in Figure 8. These values are calculated assuming all of the feedgas CO and HC will be converted to CO_2 in the vehicle's catalytic converter. As shown, these match blend fuels did not result in a significant difference in BSCO_2 emissions except at high loads, where E20-91RON and E30-91RON have lower BSCO_2 than E10-91RON due to more advanced combustion phasing and less enrichment (Figure 4).