

The Impact of Low Octane Hydrocarbon Blending Streams on the Knock Limit of “E85”

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ABSTRACT

Ethanol is a very attractive fuel from an end-use perspective because it has a high chemical octane number and a high latent heat of vaporization. When an engine is optimized to take advantage of these fuel properties, both efficiency and power can be increased through higher compression ratio, direct fuel injection, higher levels of boost, and a reduced need for enrichment to mitigate knock or protect the engine and aftertreatment system from overheating.

The ASTM D5798 specification for high level ethanol blends, commonly called “E85,” underwent a major revision in 2011. The minimum ethanol content was revised downward from 68 vol% to 51 vol%, which combined with the use of low octane blending streams such as natural gasoline introduces the possibility of a lower octane “E85” fuel. While this fuel is suitable for current “ethanol tolerant” flex fuel vehicles, this study experimentally examines whether engines can still be aggressively optimized for the resultant fuel from the revised ASTM D5798 specification.

The performance of six ethanol fuel blends, ranging from 51-85% ethanol, is compared to a premium-grade certification gasoline (UTG-96) in a single-cylinder direct-injection (DI) engine with a compression ratio of 12.87:1 at knock-prone engine conditions. UTG-96 (RON = 96.1), light straight run gasoline (LSRG, RON = 63.6), and n-heptane (RON = 0) are used as the hydrocarbon blending streams for the ethanol-containing fuels in an effort to establish a broad range of knock resistance for high ethanol fuels.

Results show that nearly all ethanol-containing fuels are more resistant to engine knock than UTG-96 (the only exception being the ethanol blend with 49% n-heptane). This allows ethanol blends made with low octane number hydrocarbons to be operated at significantly more advanced combustion phasing for higher efficiency, as well as at higher engine loads.

While experimental results show that the octane number of the hydrocarbon blend stock does impact engine performance, there remains a significant opportunity for engine optimization when considering even the lowest octane fuels that are in compliance with the current revision of ASTM D5798 compared to premium-grade gasoline.

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INTRODUCTION

The Energy Independence and Security Act of 2007 (EISA) mandates significant increases in the use of renewable fuels in the U.S. [1]. The law established a new renewable fuel standard requiring the nation to use 36 billion gallons of renewable fuel per year by 2022. Given that ethanol is the most widely used renewable fuel in the U.S. and production is expected to continue to grow over the next several years, ethanol is likely to make up a significant portion of the renewable fuel mandate. Over 98% of the fuel ethanol consumed in the U.S. is blended with gasoline to create E10-gasoline with up to 10% ethanol. Additional ethanol is exported or used in intermediate ethanol blends or higher level blends for use in flex-fuel vehicles (FFVs). In 2011,

gasoline consumption in the U.S. was just under 140 billion gallons, 13.7 billion gallons of ethanol were produced domestically, over 1 billion gallons of ethanol were exported, and only 0.033 billion gallons of E85 were consumed (<0.2% of ethanol production) [2].

Consumption of E85 is currently limited by the size of the FFV fleet, the number of E85 fueling stations, and unfavorable pricing of E85 (on a cost per unit energy basis). Consumers with FFVs rarely choose E85 due to higher cost per mile with the current pricing structure. In addition there is a reduced mileage range per tank, so consumers need to fill their gas tanks more frequently. While modern FFVs perform adequately on either gasoline or ethanol, or any blend in between, they are often referred to as “ethanol tolerant”



vehicles because there is limited optimization done to take advantage of ethanol's superior antiknock properties [3].

Each FFV sold in the U.S. must be certified on both E85 and gasoline containing no ethanol (E0) for EPA compliance. Data for the city and highway test for model years 2000 through 2012 were mined from the EPA database retained on www.fueleconomy.gov to develop Figure 1. The plot shows the average E85 fuel economy at 73% of the gasoline fuel economy for both the city and highway tests. While the exact properties of the test fuels for these certification tests is unknown, it is common practice to splash blend 85% denatured fuel-grade ethanol in compliance with ASTM D4806 [4] with 15% certification gasoline to net a resulting E81 blend [3] with 70-72% of the energy per unit volume of the base gasoline. As such, the FFV fleet on the whole displays about a 1-3% efficiency advantage with ethanol fueling.

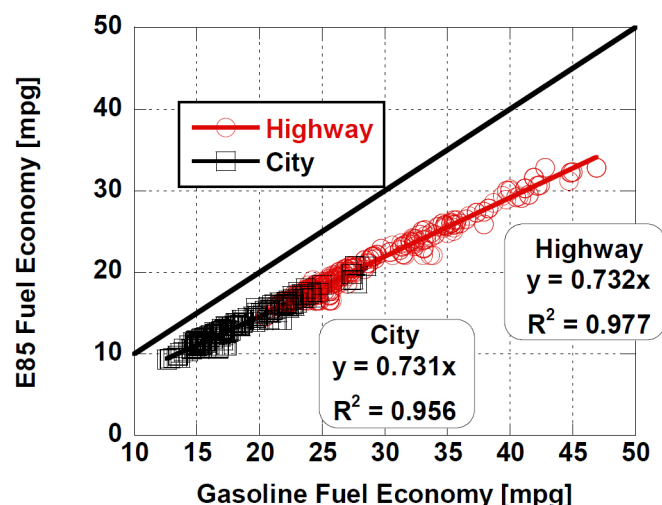


Figure 1. 85 fuel economy versus E0 fuel economy for FFVs (2000-2012) on the city and highway test.

While efficiency increases with E85 are commonly reported, there is not a consensus on the root cause of the efficiency increase. Marriott et al. [5] attributed the efficiency advantage of E85 to a reduction in heat losses due to cooler temperatures, and Szybist et al. [6] found that ethanol led to higher efficiency in a thermodynamic modeling study due to its inherent thermochemical properties. Others attribute the efficiency increase to the way that the engine is operated. For instance, Caton [7] attributed a modest efficiency increase with ethanol to reduced throttling losses on a modeling study of a PFI engine, while on a vehicle level Datta et al. [8] attributed the efficiency increase with E85 primarily to engine calibration differences. Without benchmarking individual vehicles it is not possible to quantify the extent to which the efficiency increases can be attributed to the type of engine calibration differences noted by Dumont et al. [9].

There have been a number of other studies in which engine efficiency has been significantly increased by optimizing the engine hardware to take advantage of the

unique fuel properties of ethanol [10, 11, 12, 13, 14, 15, 16, 17, 18]. While the specifics of the optimization strategies differ somewhat, the premise is to take advantage of the high chemical octane number and high latent heat of vaporization of ethanol by using an engine with high compression ratio, direct injection, and boost. A number of recent studies have highlighted the importance of the latent heat of vaporization of ethanol [14, 15, 16], with Stein et al. [14] finding that the high latent heat of vaporization of ethanol is of equal importance to the overall antiknock behavior as the chemical octane number.

While the optimization potential of high ethanol content fuels is well known, putting an ethanol-optimized vehicle into production necessitates that high quality fuel be widely available. The ASTM D5798 specification for high level ethanol blends, commonly called "E85," underwent a major revision in 2011 [19]. The minimum ethanol content was revised downward from 68 to 51 vol%, and the standard also allows the use of low octane blending streams such as natural gasoline. These changes introduce the possibility of a lower octane "E85" fuel. While this revised fuel is suitable for current "ethanol tolerant" FFVs, this study experimentally examines whether engines can still be aggressively optimized for the resultant fuel from the revised ASTM D5798 specification.

EXPERIMENTAL METHODOLOGY

Single Cylinder Engine

A highly modified 2.0 L GM Ecotec engine with direct fuel injection is used for the study. The engine geometry is listed in Table 1. Three of the cylinders of the production engine are disabled to allow single-cylinder operation. A series of different compression ratio pistons are available for this engine, as was described in an earlier ethanol optimization study with this engine [6], but this study is limited to a single compression ratio of 12.87.

Table 1. Engine geometry

Bore [mm]	86
Stroke [mm]	86
Connecting Rod [mm]	145.5
Compression Ratio [-]	12.87:1
Fuel Injection System	Direct Injection, wall guided

Machining modifications have been made to the cylinder head to accommodate the small research module hydraulic valve actuation (HVA) system from Sturman Industries. Modifications include using custom intake and exhaust valves with extended valve stems. While the valve material is different than the production valves, the combustion chamber geometry is unchanged. Other changes to the engine include a Kistler sparkplug with an integrated piezoelectric pressure transducer. In the production configuration from the original equipment manufacturer (OEM), the high pressure fuel pump for the DI fueling system is driven by the intake camshaft. However, because the camshafts are removed, the fuel is

supplied by an externally powered fuel pump. A picture of the engine research platform used in this investigation is shown in [Figure 2](#).

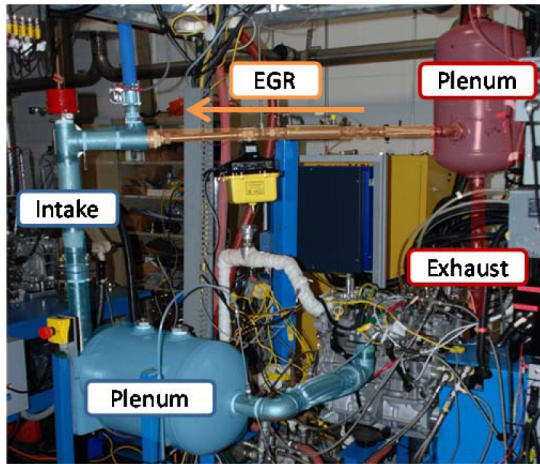


Figure 2. Single cylinder engine equipped with HVA valve train and laboratory air handling system.

The HVA system from Sturman Industries allows independent control of each of the two intake and two exhaust valves, including opening and closing crank angle, opening duration and valve lift, including the option to disable a valve. The intake and exhaust valve timings are held constant at all operating conditions in this study, as shown in [Table 2](#). It should be noted that the valve events produced by this valvetrain are more similar to square waves than a conventional cam-based valve train as was documented in a previous study with this engine [20].

Table 2. Operating conditions.

Valve train	
Intake Valve Opening [CAD] ^{1,2}	-355
Intake Valve Closing [CAD] ^{1,2}	-170
Intake Valve Lift [mm]	9
Exhaust Valve Opening [CAD] ^{1,2}	190
Exhaust Valve Closing [CAD] ^{1,2}	350
Exhaust Valve Lift [mm]	9
Fueling	
Equivalence Ratio	1.0
Fuel injection type	Wall-guided DI
Fuel injection timing [CAD] ¹	-280
Fuel rail pressure [bar]	100
Operating temperatures	
Intake manifold air [deg C]	52
Engine coolant [deg C]	90
Engine oil [deg C]	90

1. 0 CAD represents firing TDC

2. Opening and closing angles at 0.8 mm lift

DRIVEN Combustion Analysis Toolkit (DCAT) performs the crank-angle resolved data acquisition and combustion analysis. These measurements include cylinder

pressure, intake and exhaust valve lift feedback from each of the four valves and the command signal sent to the fuel injector. Crank-angle resolved data is recorded at 0.2 crank angle degree (CAD) intervals.

Engine emissions are measured using a standard emissions bench. NO_x is measured using a chemiluminescence analyzer, CO and CO₂ are measured using infrared analyzers, oxygen is measured using a paramagnetic analyzer and unburned hydrocarbon emissions (HC) are measured with a flame ionization detector. Smoke measurements are performed using a filter smoke number (FSN) instrument. Fuel flow is measured using a coriolis-effect based flow meter.

The engine is operated with a laboratory air handling system. Pressurized air is metered to the engine using a mass air flow controller. Backpressure is applied to the exhaust manifold at the boosted operating conditions using an electromechanically actuated valve so that a constant differential of 10 kPa is maintained between the intake and exhaust manifolds. For example, an intake manifold pressure of 110 kPa has an exhaust manifold pressure of 120 kPa. While this engine platform is equipped with an external exhaust gas recirculation (EGR) system, no EGR is used in this study. The exhaust gas temperature is measured in the exhaust manifold directly after the exhaust port.

Fuel Properties

A total of seven fuel blends are investigated, with full fuel properties listed in [Table 3](#). UTG-96 is a premium-grade certification gasoline procured from CP-Chem. The E85 is a pre-blended product that was also procured from CP-Chem, and as a result, the composition of the hydrocarbon portion of the E85 is unknown. The remaining fuel blends, EB-1 through EB-5, are splash-blended on-site with chemically pure anhydrous 200-proof ethanol and one of three hydrocarbon streams. EB-1 is a blend of UTG-96 with 51 vol % ethanol. EB-2 and EB-3 are ethanol blends with LSRG (antiknock index (AKI) = 63.6) containing 67 and 51 vol% ethanol, respectively. Although it was not investigated in its pure form, the fuel properties of LSRG are included in [Table 3](#). EB-4 and EB-5 are blends of n-heptane (AKI = 0) with 67 and 51 vol% ethanol, respectively. The fuel blends investigated are shown graphically in [Figure 3](#).

LSRG is typically the lowest octane number stream available in high volume at a refinery for gasoline blending [21]. Natural gasoline, the C5-C7 liquid associated with natural gas production, is similar in composition to LSRG and may also be widely available to blend with ethanol given the recent increases in natural gas production in the U.S. It is interesting that the blends made with light LSRG (EB-2 and EB-3) have a resultant research octane number (RON) that is greater than 100. EB-4, which contains 33 vol% n-heptane, also has a RON greater than 100. This result is due to the well-established non-linear response of octane number with ethanol blending on a vol% basis [22, 23], the implications of

Table 3. Fuel properties

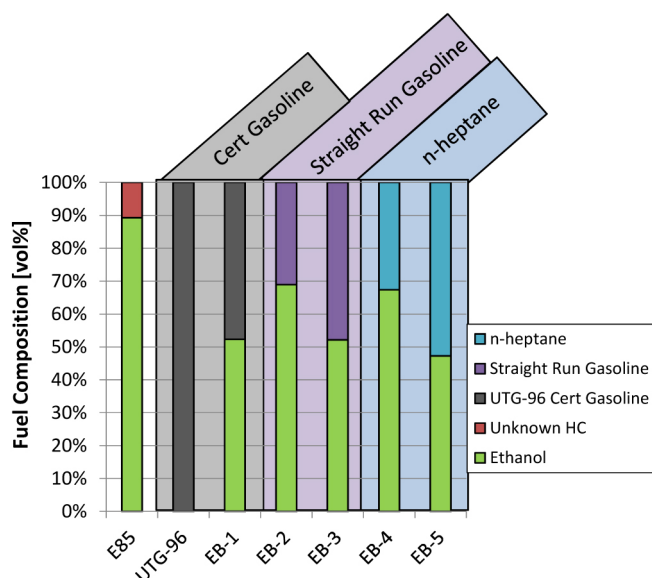
	Spec	UTG-96	LSRG	E85	EB-1	EB-2	EB-3	EB-4	EB-5
Hydrocarbon Type	--	UTG-96	LSRG	Unknown	UTG-96	LSRG	LSRG	n-heptane	n-heptane
Ethanol content [vol%]	ASTM D5599	†	†	89.3	52.3	68.9	52.2	67.4	47.3
Reid Vapor Pressure [psi]	ASTM D5191	8.90	12.40	5.12	8.68	8.09	9.34	3.44	3.68
Oxygen content [wt%]	ASTM D5599	†	†	31.22	18.76	25.24	19.65	24.56	17.67
Carbon wt/%	ASTM D5291	86.83	84.67	56.22	67.83	60.82	65.62	61.45	66.82
Hydrogen wt/%	ASTM D5291	13.20	15.62	12.92	13.10	13.78	14.28	13.92	14.48
Specific Gravity	ASTM D4052	0.739	0.658	0.788	0.768	0.753	0.732	0.756	0.738
Octane nr (R+M)/2	N/A	92.2	63.6	96.9	97.5	96.5	95.0	93.3	81.2
Research Octane Number	ASTM D2699	96.1	64.7	105.0	104.4	104.1	102.0	100.4	84.8
Motor Octane Number	ASTM D2700	88.2	62.4	88.7	90.5	88.9	88.0	86.2	77.6
Octane Sensitivity	N/A	7.9	2.3	16.3	13.9	15.2	14	14.2	7.2
Lower heating value/MJ/kg	ASTM D240	43.286	45.020	29.051	34.186	31.608	34.363	32.103	35.092
Stoichiometric AFR/-	N/A	14.42	14.94	9.45	11.45	10.58	11.57	10.72	11.94
Initial Boiling Point [°C]	ASTM D86	33.9	35.6	†	35	41.1	40.6	70.6	70.6
10% Distillation [°C]	ASTM D86	50.0	47.8	†	57.2	55.0	50.6	71.1	71.1
50% Distillation [°C]	ASTM D86	107.2	57.8	†	76.7	76.7	67.8	73.3	71.1
90% Distillation [°C]	ASTM D86	150.0	76.1	†	89.4	77.8	77.8	77.8	76.1
Final Boiling Point [°C]	ASTM D86	185.0	102.8	†	168.3	78.9	78.9	77.8	78.3
Saturates [vol%]	ASTM D1319	68.0	95.2	†	†	†	†	†	†
Olefins [vol%]	ASTM D1319	2.4	3.3	†	†	†	†	†	†
Aromatics [vol%]	ASTM D1319	29.6	1.4	†	†	†	†	†	†
† Not measured									

which are considered further in the Discussion section. EB-5 is the only ethanol-containing blend that resulted in a lower RON than UTG-96 (RON = 84.8). As a result of its low octane number, EB-5 was extremely prone to engine knock and could only be operated at very light engine loads. EB-5 is therefore not included in the results section. It is being included in the fuel properties table because it remains an interesting data point from a blending perspective.

content requirements of ASTM D5798, no attempt was made to comply with the Reid vapor pressure (RVP) requirements. The minimum RVP for compliance with ASTM D5798 is 5.5 psi [19], a requirement that neither EB-4 nor EB-5 are able to meet. In order to comply with the RVP requirements, more volatile hydrocarbons would need to be used which would necessarily increase the AKI of the fuel blend. It is also interesting to note that the pre-blended E-85 is also unable to meet the minimum RVP specification. Volatility is important from a cold-start and driveability perspective, but RVP would not be expected to affect any of the results reported here.

Operating Conditions

This study is focused on the knock-prone portion of the engine operating map, which occurs at low speed and high load. The experimental space includes engine loads of 800, 1000, 1200, and 1400 kPa IMEP_{net} at engine speeds of 1500, 2000, and 2500 rpm. The maximum load is limited to 1400 kPa IMEP_{net} because higher engine loads approach the 100 bar cylinder pressure limitation for this engine with the high compression ratio configuration used in this study. 1200 kPa IMEP_{net} is nominally the engine load achieved at wide open throttle conditions, although it is dependent of fuel. For fuels capable of achieving 1200 kPa IMEP_{net} with an intake manifold pressure of less than 98 kPa_{abs}, there is no added backpressure. If the intake manifold pressure is 99 kPa absolute or higher, the exhaust manifold pressure is maintained at a pressure of 10 kPa higher than the intake manifold to simulate a turbocharger. It is necessary to boost

**Figure 3. Composition of fuel blends investigated.**

It is also interesting to note that while the fuel blends selected in this study are intended to comply with the ethanol