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Intermediate Alcohol-Gasoline Blends, Fuels for Enabling Increased Engine Efficiency and Powertrain Possibilities

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ABSTRACT

The present study experimentally investigates spark-ignited combustion with 87 AKI E0 gasoline in its neat form and in mid-level alcohol-gasoline blends with 24% vol./vol. iso-butanol-gasoline (IB24) and 30% vol./vol. ethanol-gasoline (E30). A single-cylinder research engine is used with a low and high compression ratio of 9.2:1 and 11.85:1 respectively. The engine is equipped with hydraulically actuated valves, laboratory intake air, and is capable of external exhaust gas recirculation (EGR). All fuels are operated to full-load conditions with λ=1, using both 0% and 15% external cooled EGR. The results demonstrate that higher octane number bio-fuels better utilize higher compression ratios with high stoichiometric torque capability. Specifically, the unique properties of ethanol enabled a doubling of the stoichiometric torque capability with the 11.85:1 compression ratio using E30 as compared to 87 AKI, up to 20 bar $IMEP_g$ at λ=1 (with 15% EGR, 18.5 bar with 0% EGR). EGR was shown to provide thermodynamic advantages with all fuels. The results demonstrate that E30 may further the downsizing and downspeeding of engines by achieving increased low speed torque, even with high compression ratios. The results suggest that at mid-level alcohol-gasoline blends, engine and vehicle optimization can offset the reduced fuel energy content of alcohol-gasoline blends, and likely reduce vehicle fuel consumption and tailpipe CO_2 emissions.

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INTRODUCTION

The Energy Independence and Security Act of 2007 [1], requires that by year 2022, 36 billion gallons per year of bio-derived fuels be consumed in transportation. This uptake in bio-derived fuels is more than a seven-fold increase from the 4.7 billion gallons consumed per year when the law was enacted. The rules for complying with this mandate are specified by the U.S. Environmental Protection Agency (EPA) in the Renewable Fuel Standard II (RFS II) [2].When the total transportation energy consumption is analyzed, it is apparent that this legislation increases the usage of bio-fuels. In 2012 the United States consumed 27.97 quadrillion BTU of energy for transportation, and is projected to consume 29.24 quadrillion BTU in 2022 [3]. Assuming a gasoline equivalent energy of 42.8 MJ/kg, and density of 740 kg/m^3, the RFS II standard will require an increase in the volumetric percentage of transportation energy from biofuels to approximately 14% in 2022 from the approximately 2% in 2007. To date, the RFS II progress has seen more than a doubling of biofuel usage, with the annual recorded share of transportation energy from non-petroleum sources totaling 4.3% in 2012. Although 2012 was the year with the largest biofuel energy share on record [3], there is still an additional threefold increase in biofuel energy share required to comply with the RFS II mandate.

Concurrent with RFS II, legislation by the National Highway and Transportation Safety Administration passed in 2011 requires corporate average fuel economy (CAFE) standards to achieve 54.5 US miles per gallon by 2025 [4], an effective two fold increase compared with present CAFE standards. Ideally, the RFS II and CAFE mandates could be met simultaneously through proper exploration and implementation of high-efficiency bio-fuel engines.

In the United States, the stoichiometrically operated spark ignition (SI) engine has maintained over a 99% market share in the light-duty (LD) vehicle sector (passenger cars and pickup trucks) since 1985 [5] and over a 94% share since the EPA began record keeping in 1975. This LD sector engine dominance is due primarily to the facts that the SI engine has: low production cost, low fuel cost, rugged operation, high torque/power density, low sooting tendency, and can employ known mature catalyst technologies to reduce regulated emissions (oxides of nitrogen (NO_x), hydrocarbon (HC), and carbon monoxide (CO)). The market sector dominance with the SI engine in combination of legislated CAFE and RFS II standards suggests that increases to SI engine efficiency with bio-fuels might offer a very plausible path toward simultaneous CAFE and RFS II compliance.

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Although the SI engine has many beneficial attributes, its efficiency is fundamentally hindered by the throttling of air, and its compression ratio is limited by combustion knock. These two factors result in lower thermal efficiency[1] of SI engines relative to compression-ignited engines (i.e., diesel engines) or lean-burn SI engines. Historically, engine improvements have focused primarily on increasing safety and convenience, yielding increases in performance and vehicle weight, while complying with legislated fuel economy mandates, which have been near-constant since 1985 [5]. The progress in engine performance over the years can clearly be observed when viewed relative to a 1975 baseline (the first year of EPA records): in today's vehicles the industry average power per unit displacement has more than doubled, while vehicle zero to 60 miles per hour acceleration time has halved [5]. This evolutionary increase in performance has resulted in a 50% reduction in the average light-duty engine displacement compared with a 1975 era engine. Although these trends show there has been significant progress in engine performance, to comply with future CAFE requirements, engine and vehicle efficiency must also be addressed and improved.

An evolutionary strategy for achieving CAFE compliance while retaining performance is downsizing and turbocharging with direct injection. These two technologies offer increased engine power/torque density, with similar performance when downsizing and downspeeding engines, a proven efficiency improving strategy [6]. However, the opportunity for downsizing and downspeeding becomes limited by combustion knock from current market available fuels, thereby limiting thermodynamic efficiency.

Unlike distillate fuels, alcohol fuels exhibit some key properties that make them particularly attractive fuels for future engines. Most notably, alcohol fuels tend to have a high octane number and lower carbon intensity.[2] These combinations of properties grant alcohol-based fuels a twofold reduction potential in tailpipe carbon dioxide (CO_2) through molecular advantage and the ability to tolerate higher engine compression ratios. Additionally, alcohol fuels exhibit two other properties that can be favorable for increasing engine efficiency:

First, alcohol fuels exhibit a high latent heat of vaporization (HoV), which, when used in conjunction with direct injection (DI) fueling, can increase the incoming charge density caused by a reduction in charge temperature. When exploited properly, the high HoV improves engine breathing as highlighted Stein et al. [7], and mitigates combustion knock tendency. The effect of HoV has proved to be strong, even in a sparingly used dual-fuel arrangement, where the charge cooling of a small amount of direct injected ethanol prevented knock and extended the torque capability [8], with both benefits enabling higher efficiency engines.

Second, the amount of thermodynamic work that can be extracted from alcohols on a 2nd law basis is higher than is suggested by its lower heating value (LHV) alone (i.e., exergy/LHV). This is attributed to a high yield of molar products for alcohols on both a stoichiometric and energy basis relative to petroleum distillates, increasing expansion pressure [9, 10]. Ford and AVL have shown that ethanol enables efficiency improvements, with several notable works summarized in Stein et al. [11]. Vehicle-specific effects were researched by Jung et al. [12] at light load conditions and also in an additional study by Jung et al. [13] with drive cycle and engine efficiency estimates. The latter study points out that a light-duty pickup truck engine with intermediate ethanol-gasoline blends could be optimized such that the thermal efficiency increase with ethanol-gasoline blends of 20% ethanol vol./vol. are sufficiently high to at least offset the volumetric energy density penalty of alcohol fuels, and achieve even greater tailpipe CO_2 reductions.

These engine experiments and vehicle simulation results demonstrate that reductions in CO_2 emissions without a decrease in miles per gallon (MPG) could be possible with intermediate ethanol-gasoline blends. A major reason for this prediction is the ability of ethanol addition to reduce combustion knock and enable an increased compression ratio. Interestingly, work by Szybist and West [14] demonstrates that blending ethanol, even with very low-octane gasoline blendstocks, offers significant anti-knock resistance and that a high-octane fuel can be produced through blending intermediate levels of ethanol with straight-run gasoline. This is because of the highly nonlinear response of octane number blending with ethanol on a volumetric basis, as previously explained in detail by Anderson et al. [15, 16], and more recently by Foong et al.[17]. These cited studies show that intermediate-level ethanol blends might be promising for the next generation of SI engine fuels. The noted inherent benefits provide the potential to increase the power output and efficiency of the engine through fuel based knock mitigation coupled to engine optimization. However, as pointed out in the 2013 SAE International High Octane Fuel Symposium [18], multiple levels of cooperation from the fuel industry, legislation and regulatory bodies, distribution systems, and point of sale vendors is required if fuel octane is to be increased.

The use of external cooled exhaust gas recirculation (EGR) could be a more direct approach to increasing SI engine efficiency. External EGR is a proven method to reduce the knocking tendency for a given fuel. External-cooled EGR has been employed for years in diesel engines, with recent interest gaining in SI engines. The constituents of EGR in SI engines differ from those in their diesel counterparts. Specifically, λ=1 SI EGR is oxygen deficient, meaning that SI EGR offers the potential to increase charge mass without changing the oxygen content. The lack of oxygen in SI EGR is important when considering both catalyst and throttling requirements of SI engines (i.e., λ=1). In addition to reducing throttling losses, the introduction of EGR into SI engines improves the

1 Defined as the efficiency of converting fuel chemical energy to mechanical output work

2 Defined as the number of moles of carbon per unit of energy (LHV)

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thermodynamic properties of the working fluid (i.e., the ratio of specific heats (γ)), reducing in-cylinder temperatures and improving knock resistance.

Many of these thermodynamic advantages have been documented by others. For example, Alger et al. [19] showed that external-cooled EGR effectively decreased the knocking propensity of distillate fuel which functionally increased the fuel octane number. However, EGR also slowed the flame kernel growth because of slowed reaction rates. These EGR combustion process and anti-knock findings were furthered by the inclusion of intermediate alcohol-gasoline blends in Splitter and Szybist [20], which showed that 15 % external cooled EGR effectively negated the increased burning velocities intrinsic with alcohol fuels [21, 22]. Therefore, regardless of fuel type, higher EGR levels in SI engines might require the incorporation of different higher turbulence combustion chamber flows to increase EGR tolerance, as shown by Wheeler at al. [23], or through high-energy long spark systems as shown by Alger et al. [24]. These previous studies suggest there are technical challenges that need to be addressed if EGR is to be used.

The relations between knock mitigation and cycle difference with external cooled EGR raise several questions:

- What, if any, role does EGR have on engine efficiency in mid-level alcohol blends?
- What are the combustion specific differences between intermediate alcohol-gasoline blends and neat gasoline?
- What, if any, potential performance and fuel economy incentives do mid-level alcohol blends offer both with and without external cooled EGR?
- Can intermediate alcohol-gasoline fuels enable new powertrain possibilities?

The present study explores what engine efficiency and stoichiometric torque capability gains can be enabled through the use of mid-level ethanol and iso-butanol gasoline blends as compared to conventional gasoline. The engine efficiency, load-speed range, and downsizing + downspeeding potential of each strategy and fuel is investigated relative to one another using a higher than stock 11.85 r_c piston and compared to 87AKI gasoline with the production 9.2 r_c piston.

EXPERIMENTAL DETAILS

This study explores SI engine operation at 5 engines speeds (1200, 1600, 2000, 2500, and 3000 r/min) and two different EGR rates (0% and 15%), each with three different fuels (87AKI E0 “regular” gasoline, 30% by vol. ethanol-gasoline, and 24% by vol. iso-butanol-gasoline). A highly modified 2.0 L GM Ecotec SI engine with stock side-mounted direct fuel injector is used. Three cylinders of the production engine are disabled to allow single-cylinder operation with an installed custom-domed piston, which increases the compression ratio to 11.85:1 (stock 9.2:1).

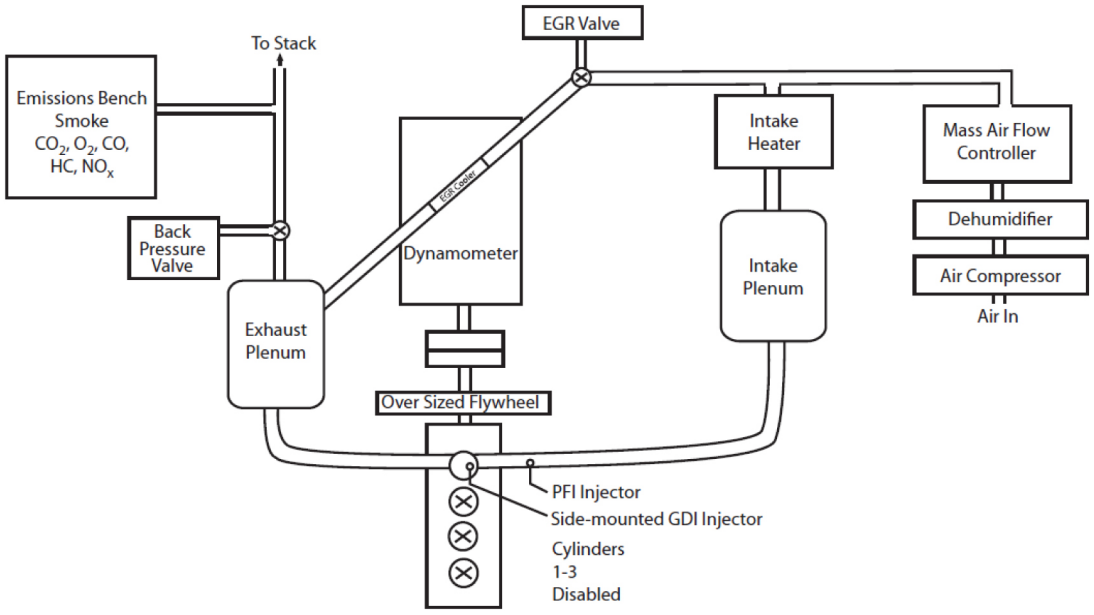


Figure 1. Schematic of the experimental configuration.

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The SI engine is operated with a laboratory air handling system. Pressurized and dried facility air that has less than 5% relative humidity is metered to the engine using a mass air flow controller. The present study omits humidity effects, where added humidity may have an effect of reducing knock. The engine is equipped with separate electro-mechanical valves for backpressure and external EGR, enabling the capability for independent control of intake manifold pressure, exhaust manifold pressure, and EGR. Cooled EGR mixes with fresh air upstream of an air heater, followed by the intake plenum and then the intake manifold. The arrangement results in the intake manifold temperature being controlled to constant a set temperature (52°C) regardless of EGR rate. EGR is measured using an EGR 5230 system from ECM, an instrument that uses pressure-compensated wideband oxygen sensors in both the intake and exhaust to non-intrusively measure EGR. When EGR is used, a constant rate of 15±1% is supplied. A schematic of the laboratory is provided in Figure 1.

The engine is equipped with a hydraulic valve actuation (HVA) system to enable fully variable valve actuation. To accommodate the small research module HVA system from Sturman Industries, the cylinder head has been machined, disabling the functionality of the production cam and fuel pump systems. Details of the VVA system have been published previously [9, 25,26]. The engine geometry is listed in Table 1.

Table 1. Engine geometry

Bore x Stroke	86mm x 86mm
Connecting rod length	145.5mm
Compression ratios	9.2 (stock), 11.85:1 (modified)
Fuel injection system	Direct injection, side-mounted

Crank-angle (CA) resolved data are recorded at 1800 samples per revolution (0.2°CA resolution) for 300 consecutive cycles. Cylinder pressure is measured using a Kistler piezo-electric 6125B pressure transducer coupled to a Kistler 5010 charge amplifier. Additionally, the DI command signal and intake and exhaust valve lift from each of the four HVA valves is recorded on a crank-angle resolved basis. All indicated results presented in this study are for a 300 cycle average.

Engine emissions are measured using a standard emissions bench with instruments manufactured by California Analytical Instruments. NO_x is measured using a heated chemiluminescence analyzer, CO and CO_2 are measured using infrared analyzers, oxygen (O_2) is measured using a paramagnetic analyzer, and HC is measured with a heated flame ionization detector. Smoke measurements are performed using an AVL 415s filter smoke number (FSN) instrument. To measure the fuel flow rate (and thus efficiency), the air-to-fuel ratio (AFR) is measured directly from a Coriolis-effect fuel flow meter and a laminar air flow element. The corresponding fuel flow is then cross referenced to independently calculate air-to-fuel ratios from the engine exhaust using both the emissions bench and automotive wideband oxygen sensor approaches.

The conditions for all fuels, engine speeds, loads, and EGR combinations are listed in Table 2.

Table 2. Constant operating conditions

Exhaust valve open at 0.8 mm lift (°CA $ATDC_f$)	170
Exhaust valve close at 0.8 mm lift (°CA $ATDC_f$)	350
Max exhaust valve lift (mm)	9
Intake valve open at 0.8 mm lift (°CA $ATDC_f$)	-355
Intake valve close at 0.8 mm lift (°CA $ATDC_f$)	-170
Max intake valve lift (mm)	9
Start of DI command (°CA $ATDC_f$)	-280
Intake manifold gas temperature (°C)	52
Engine coolant (°C)	90
Oil (°C)	90
Exhaust λ (-)	1
DI rail pressure (bar)	100

Five engine speeds of 1200, 1600, 2000, 2500, and 3000 r/min are tested with gross load increments ($IMEP_g$ indicating mean effective pressure gross) of 50±5 kPa. The constraints and load range for E30 without EGR can be seen in Figure 2, which illustrates the tested operational map up to the constraints for E30 with 0% EGR. The identical procedure and setup is conducted for each fuel type and both EGR rates.

All fuels and EGR rates are tested at maximum brake torque (MBT) timing until combustion knock is encountered. Once knock limited, combustion is phased through spark timing to maintain a constant level of knock through visual inspection of the indicated pressure trace and by maintaining constant AVL combustion noise, which were empirically found to trend together. The tested load range is from 2 bar $IMEP_g$ to full load, which is defined as the maximum load without enrichment (λ=1) with limits on CA50 (crank angle at 50% mass fraction burned) combustion phasing of 25°CA $ATDC_f$, peak cylinder pressure of 10,000 kPa, and exhaust gas temperature (EGT) of 800°C. (Note: CA50 combustion phasing later than 25°CA $ATDC_f$ is not ideal for high efficiency, which is the focus of the present study. Additionally, 800°C EGT limit is imposed

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because unlike the production engine, the Sturman valvetrain does not use sodium-filled values which can withstand higher exhaust gas temperatures. The maximum cylinder pressure limit is set because the stock engine is rated to 10,000 kPa, although this constraint is not reached by any of the present fuel or experimental condition combinations). Figure 2 displays the constraint limits reached with 0% EGR E30 operation.

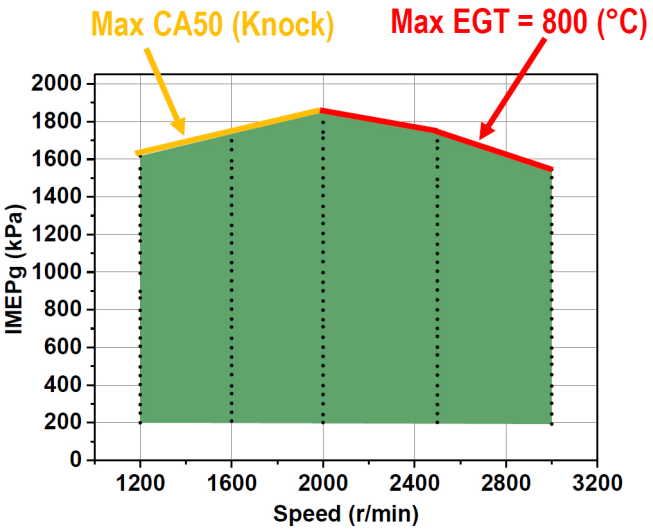


Figure 2. Representative engine load-speed range tested with knock and EGT constraints indicated.

The only exception to the constraints is at the lightest loads with EGR ($IMEP_g \leq 250$ kPa), where combustion is unstable at MBT CA50 phasing, where later than MBT CA50 were found to improve stability. The constraint used to bound the amount of CA50 retard for acceptable operation at these lightest loads is $CA50 \leq 15°CA\ ATDC_f$. Using these limits, operation with each fuel was compared.

The production spark plug heat range and gap is used for all tests. However, the spark energy is generated with an aftermarket MSD DIS6-2 Plus multi-strike ignition system to increase the combustion stability at high EGR levels. The MSD system is capable of up to three consecutive spark discharges per cycle, but the number of discharges is speed dependent, with only one or two discharges at higher engine speeds. The spark coil signal from the MSD system, heat release rate (HRR), and cylinder pressure are indicated in Figure 3, along with the unique valve events provided by the Sturman HVA system. The near square valve lift profile generated by the Sturman HVA system differs from conventional valve lift and duration dynamics, resulting in increased flow area with different engine breathing and charge motion characteristics. The specific differences between the HVA valvetrain and a cam-based valvetrain are discussed in a previous publication [26].

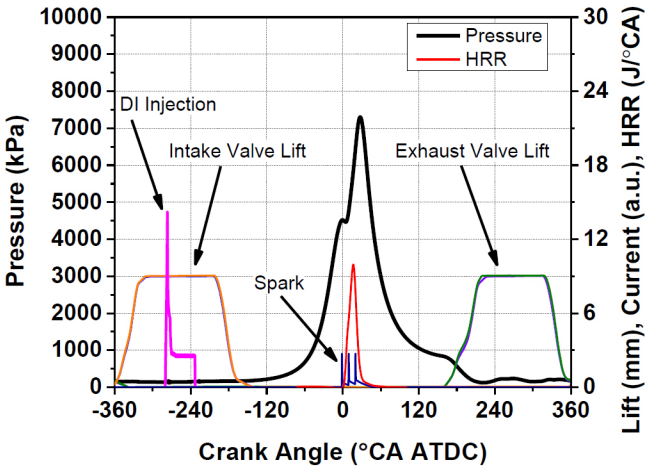


Figure 3. Representative high-load fired case showing indicated cylinder pressure, apparent heat release rate (HRR) with corresponding valve, spark, and injection event schedule.

When operating with higher than atmospheric pressures, a constant overall turbocharger efficiency of 25% with no intake or exhaust pressure restrictions is assumed (i.e., no muffler pipe, catalysis, or air cleaner pressure losses assumed). This assumption is considered valid and conservative because the production turbocharger for this SI engine is capable of over 55% combined overall efficiency. The combined turbocharger efficiency is calculated based on the air standard model as explained in Heywood [27] (Equation1), using intake and exhaust surge tank pressures and temperatures measured in the ports and assuming 100% turbine shaft mechanical efficiency.

$$\eta_{combined} = \frac{\left(\frac{\gamma_{comp}}{\gamma_{comp}-1}\right)}{\left(\frac{\gamma_{turb}}{\gamma_{turb}-1}\right)}\left(\frac{T_{comp,in}}{T_{turb,in}}\right)\frac{\left[\left(\frac{P_{comp,in}}{P_{comp,out}}\right)^{\left(\frac{\gamma_{comp}-1}{\gamma_{comp}}\right)}-1\right]}{\left(1+\frac{1}{AFR}\right)\left[1-\left(\frac{P_{turb,out}}{P_{turb,in}}\right)^{\left(\frac{\gamma_{turb}-1}{\gamma_{turb}}\right)}\right]} \quad (1)$$

When operating with external EGR, the electromechanical valves for backpressure and EGR were modulated as needed to maintain 15% EGR and 25% combined turbocharger efficiency.

Fuels and Fuel Properties

Three fuels are tested: two fuels are alcohol-gasoline blends, and the third is an unblended gasoline. The two alcohol blended fuels were splash blended on site with either 24% neat iso-butanol or 30% neat-ethanol, with alcohol and gasoline volume fractions being measured in un-blended fractions and then combined. All fuels were based on commercially obtained 87 AKI E0 “regular” pump fuel sourced directly from a distribution terminal. It should be noted that splash blending

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these alcohols with finished market E0 gasoline is not likely at an industrial scale if the tested blend ratios were to be market sold. More likely, a blendstock for oxygenate blends (BOB) fuel would be used, which tend to have lower octane numbers. Although using a BOB would decrease the research octane number (RON) of a blended fuel below those tested in the present study, others studies, [7, 14] have shown that the RON difference of a blended fuel based on a BOB vs. a finished E0 is much smaller than the difference in RON between the unblended BOB and finished E0 fuels.

Table 3. Fuel properties.

	87 AKI	IB24	E30
Oxygenates ASTM D5599 (%v)	<0.1 any	23.64 iso-butanol	30.65 ethanol
HoV (kJ/kg)[3]	352[31, 32]	443 [31, 31, 33]	529 [31, 31, 33]
HoV with gasoline energy eq.[4] (kJ/kg)	352 [31, 32]	470 [31, 32, 33]	599 [31, 32, 33]
Reid vapor pressure, ASTM D5191 (psi)	13.13	12.29	13.28
10% distillation point, ASTM D86 (°F)	97	115	111
30% distillation point, ASTM D86 (°F)	144	175	150
50% distillation point, ASTM D86 (°F)	205	208	165
70% distillation point, ASTM D86 (°F)	253	222	170
90% distillation point, ASTM D86 (°F)	316	307	299
RON, ASTM D2699(-)	90.2	96.6	100.3
MON, ASTM D2700(-)	83.9	86.8	88.8
Octane Sensitivity (-)	6.3	9.8	11.5
LHV, ASTM D240 (MJ/kg)	43.454	40.846	38.105
λ=1 AFR (-)	14.70	14.13	12.85
C, ASTM D5391 wt. (%)	86.49	80.63	74.4
H, ASTM D5391 wt. (%)	14.06	13.89	13.73
O, ASTM D5599 wt (%)	<0.1	3.71	11.34
Specific gravity, ASTMD4052(-)	0.729	0.743	0.745
Volumetric energy density (MJ/gal)	119.5	114.5	107.1

3 Calculated through a linear combination of neat alcohol and gasoline HoV.
4 Calculated on a gasoline equivalent energy basis, i.e., fueling required for matched load.

The blending ratios used are based on the following:

- 24% iso-butanol was selected as it has near identical oxygen content as E15, which the EPA has approved for use in 2001 and newer light duty vehicles [28,29].
- The 30% ethanol blend was selected because, as the EPA recently stated, [30] there is no foreseeable issue with higher ethanol-gasoline blends; citing blends as high as E30 would likely be permissible.

These alcohol-gasoline fuel blends are of interest as the neat alcohols used exhibit nearly identical motor octane number (MON) values and similar RON values [31]. However, the energy density of iso-butanol is higher than that of ethanol on both a volume and mass basis, thus making its energy density closer to a gasoline. Additionally, the lower water solubility of iso-butanol as compared to ethanol offers advantages in certain markets such as marine environments where humidity and water contact are more prone. Interestingly, research by Stein et al. [11] has demonstrated that the Reid vapor pressure of intermediate ethanol blends are typically lower than E10 blends for the same blendstock, making such fuels attractive to regulatory bodies, as has been indicated in the recent EPA Tier III notification of proposed rulemaking [30].

In the present study, the alcohols are from non-denatured reagent grade purity and obtained directly from suppliers. Sigma-Aldrich supplied the iso-butanol at a purity of >99%, and Decon Labs supplied the non-denatured ethanol. The three fuels were sent for independent analysis at an ASTM International certified laboratory. The key fuel properties from the analysis are presented in Table 3.

Results

The results section is divided into subsections. The first provides an overview of the fuel-specific and compression ratio differences on the operable speed-load range and combustion process of the engine without EGR. The second provides estimates of brake thermal efficiency differences between the stock engine with 87AKI gasoline 0% EGR and the higher compression ratio piston E30 with 0 or 15% external EGR. The third subsection discusses downsizing and downspeeding considerations.

Fuel and Compression Ratio Specific Effects

All fuels are operated to the maximum load condition defined by the aforementioned combustion knock and EGT constraints in the experimental details section. Figure 4 presents the tested load range of all three fuels with 0% and 15% EGR (bottom and top sub-plots respectively), with the 11.85 r_c piston and 87AKI with the 9.2 r_c piston.

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