

Effects of Fuel Composition on EGR Dilution Tolerance in Spark Ignited Engines

James P. Szybist and Derek Splitter
Oak Ridge National Laboratory

ABSTRACT

Fuel-specific differences in exhaust gas recirculation (EGR) dilution tolerance are studied in a modern, direct-injection single-cylinder research engine. A total of 6 model fuel blends are examined at a constant research octane number (RON) of 95 using n-heptane, isooctane, toluene, and ethanol. Laminar flame speeds for these mixtures, which are calculated using two different methods (an energy fraction mixing rule and a detailed kinetic simulation), span a range of about 6 cm/s. A nominal load of 350 kPa IMEP_g at 2000 rpm is maintained with constant fueling and varying CA₅₀ from 8-20 CAD aTDC_r. EGR is increased until a COV of IMEP of 5% is reached. The results illustrate that flame speed affects EGR dilution tolerance; fuels with increased flame speeds have increased EGR tolerance. Specifically, flame speed correlates most closely to the initial flame kernel growth, measured as the time of ignition to 5% mass fraction burned. The effect of the latent heat of vaporization (HoV) on the flame speed is taken into account for the ethanol-containing fuels. At a 30 vol% blend level, the increased enthalpy of vaporization of ethanol compared to conventional hydrocarbons can decrease the temperature at the time of ignition by a maximum of 15°C, which can account for up to a 3.5 cm/s decrease in flame speed. The ethanol-containing fuels, however, still exhibit a flame speed advantage, and a dilution tolerance advantage over the slower flame-speed fuels. The fuel-specific differences in dilution tolerance are significant at the condition examined, allowing for a 50% relative increase in EGR (4% absolute difference in EGR) at a constant COV of IMEP of 3%.

CITATION: Szybist, J. and Splitter, D., "Effects of Fuel Composition on EGR Dilution Tolerance in Spark Ignited Engines," *SAE Int. J. Engines* 9(2):2016, doi:10.4271/2016-01-0715.

INTRODUCTION

It is well known that engine efficiency and performance are coupled to vehicle fuel economy. Since the 1980's engine technologies have been deployed and evolved in an effort to increase both vehicle fuel economy and performance [1]. One engine technology being considered as a pathway to increase vehicle fuel economy is cooled external exhaust gas recirculation (EGR). The attractiveness of cooled external EGR is that it offers thermodynamic advantages to the working fluid while maintaining conventional 3-way catalyst aftertreatment compatibility. The overall efficiency advantages for high EGR conditions are summarized nicely in a thermodynamic modeling study by Caton [2]. The additional mass from the EGR causes the manifold pressure to increase without increasing the intake oxygen flow, thereby decreasing engine pumping and increasing engine efficiency at part-load conditions. Additionally, the peak cylinder temperature decreases with EGR, causing the heat transfer to decrease. Finally, due to a combined thermal and composition effect, the ratio of specific heat (γ) increases with EGR.

In addition to these thermodynamic advantages, Alger et al. [3] showed that EGR also reduces the knock propensity, a result of the reduced in-cylinder temperature. This can enable increased engine efficiency as operation approaches full load, through either combustion phasing advance or increased compression ratio. Alger et al. [3] also concluded that every % EGR was equivalent to a 0.5

octane number increase. Splitter et al. [4] confirmed the reduced knock propensity using 15% EGR, which allowed the maximum load limit to be increased by more than 10% in a single-cylinder engine experiment.

The amount of EGR that is implemented in practical engine configurations is limited by combustion stability. Alger et al. [5] reported that the part-load efficiency increase was not as substantial as expected, due primarily to a decrease in combustion stability with EGR. This was due, in part, to high in-cylinder residuals at part-load conditions, due to a combination of the high degree of throttling and due to the cam phasing strategies.

Combustion stability and cycle-to-cycle variability in SI engines has been the subject of extensive research over multiple decades. Ozdor et al. [6] performed a comprehensive literature survey in 1994 that remains extremely relevant. They noted that the initial flame kernel development, which is typically about a third of the total combustion duration, is the most crucial to cycle-to-cycle variations because of its high sensitivity to a large number of factors. During this time, the laminar flame speed of the mixture dominates the combustion rate; the turbulent flame propagation stage dominates only after the flame kernel is sufficiently large to be influenced by multiple turbulent eddies simultaneously.

Ozdor et al. [6] also describe the effects of fuel on cycle-to-cycle variability, in a context not specific to EGR. Fuel effects are directly related to the laminar flame speed and the early flame kernel development. Fuels with high flame speed complete the early flame kernel development process faster, making them less susceptible to the stochastic cycle-to-cycle differences in turbulence and ultimately provide more stable combustion. Ozdor et al. [6] cite Stone et al. [7] to state that, in general, the fuel-specific differences are less than a unity change in air-to-fuel ratio.

Flame speed effects with EGR dilution have been elucidated by Middleton et al. [8], who developed a correlation for the laminar flame speed of iso-octane using air and EGR dilution. It was found that the addition of 20% EGR reduces the laminar flame speed by nearly 50%. This reduction of laminar flame speed can then be expected to slow the early flame kernel development, thereby making it more susceptible to stochastic cycle-to-cycle differences in turbulence. This is confirmed in a study from Weaver and Santavica [9], where the flame kernel growth slowed with 10% EGR, and as a result of stochastic differences the percent variability on the flame kernel radius increased significantly compared to the case without EGR.

In the time since much of the work on cycle-to-cycle variability was conducted, fuel composition has changed significantly and may continue to evolve further. Specifically, nearly all gasoline in the US contains 10 vol% ethanol (E10) [10] and there is a possibility that the ethanol content will increase further. E15 (15 vol% ethanol) is a legal fuel for model year 2001 and newer vehicles [11], but it is not yet widely used. Recent work has demonstrated significant efficiency improvements from the anti-knock benefits of higher-concentration ethanol blends (>15 vol%) [4, 12,13,14,15,16].

The anti-knock benefits of high ethanol blends involve using modern direct-injection fueling systems to leverage the high latent heat of vaporization of the ethanol blends. Stein et al. [16] and Kasseris et al. [17] both found that the charge cooling from ethanol's high latent heat of vaporization (HoV) can account for up to 50% of the reduced knock propensity. Chupka et al. [18] found that the charge cooling from a 50% ethanol blend can be significant, up to 30°C of cooling of additional cooling for a stoichiometric mixture compared to gasoline without ethanol.

Ethanol is a high flame speed component compared to the other hydrocarbons that can be found in gasoline. Farrell et al. [19] measured the laminar flame speed of 45 gasoline-range fuel components, including ethanol, and found a relatively wide variation at the elevated pressure and temperature condition investigated: 450 K and 304 kPa. Among these constituents, there was over a 60% difference in flame speed between the fuels, and that ethanol had the fastest flame speed. However, flame speed is a function of both temperature and pressure, and the extent to which the charge cooling of ethanol fuel blends in direct injection fueling systems reduces the flame speed is not known, particularly in EGR-dilute environments.

The purpose of the present investigation is to study the fuel-specific effects, including ethanol content, on EGR dilution tolerance in a modern direct-injection engine. Fuel injection occurs during the intake stroke at a typical timing for DI engines operating under

homogeneous conditions. Such an injection strategy maximizes mixing and the charge cooling effect from the fuels, while simultaneously minimizing the fuel spray impingement on the combustion chamber surfaces, which can lead to particulate matter. Particular attention is paid to combustion duration through the three stages of the combustion process: early flame kernel development, turbulence-dominated combustion, and the completion of the combustion event. To support engine measurements, laminar flame speed is calculated using two different methods. The findings from both the measurements and simulations are used to determine the effect of fuel composition and flame speed on dilution tolerance in a modern GDI engine.

METHODOLOGY

Experimental Methodology

The engine used in this study is a 2.0 L GM Ecotec LNF engine equipped with the production side-mounted direct injection (DI) fueling system. Engine geometry details are presented in Table 1. The engine is converted to a single-cylinder engine by disabling cylinders 1, 2, and 3. The combustion chamber geometry and camshaft profiles are unchanged from the stock configuration.

Table 1. Engine geometry.

Bore x Stroke [mm]		86 x 86
Conrod length [mm]		145.5
Compression ratio [-]	9.2:1	
Fuel injection system		Direct injection, side-mounted

The engine is operated using laboratory fueling and air handling systems. The fueling system uses a pneumatically-actuated positive displacement pump in conjunction with an electronic pressure regulator to provide fuel rail pressure. A constant fuel rail pressure of 85 bar is used throughout this study. Pressurized and dried facility air having less than 5% relative humidity is metered to the engine using a mass air flow controller. The engine is equipped with an external cooled EGR loop with the flow rate controlled by an electromechanical valve. Cooled EGR mixes with fresh air upstream of an air heater, followed by the intake plenum and then the intake manifold. The EGR used in this study is not treated with an exhaust catalyst before being recirculated to the intake. This arrangement allows the intake manifold temperature to be held at a constant temperature of 52°C regardless of the EGR level. EGR is measured using a non-intrusive method that utilizes pressure-compensated wideband oxygen sensors in both the intake and exhaust.

A DRIVEN engine controller with the Combustion Analysis Toolkit (DCAT) package is used to control the engine and acquire crank-angle resolved data and analyze combustion performance. Cylinder pressure data, heat release, and combustion analysis are collected for 1000 fired cycles for each condition. Cylinder pressure is measured using a flush-mounted piezoelectric pressure transducer from Kistler (6125C). Fuel injection timing occurs during the intake stroke and is held constant at 280 CA bTDC_r. Spark timing is adjusted as-needed

to achieve the desired combustion phasing, and spark dwell is held constant at 1.8ms with the stock ignition coil to maintain constant ignition energy.

Engine fuel flow is measured with a coriolis-based fuel flow meter, and engine coolant temperature is maintained at 90°C. The combustion emissions are monitored using a standard array of analyzers. NO_x is measured using a heated 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 heated flame ionization detector, and smoke emissions are also measured using a reflective filter smoke number (FSN) instrument.

Engine Operating Conditions

The investigation takes place at a single engine operating speed, 2000 rpm, and a single fueling rate for each fuel, corresponding to a gross indicated mean effective pressure (IMEP_g) of 350 kPa at the baseline condition. The baseline condition corresponds to a CA50 combustion phasing of 8 CA aTDC_f without external EGR. The operating space is then mapped out as a function of EGR (0, 5, 10, and 20%) and CA50 combustion phasing (8, 12, 16, and 20 CA aTDC_f) for each fuel, with a combustion stability limit of a COV of IMEP of 5%. With fueling held constant, engine load does vary throughout this CA50 and EGR space, as shown in the results section. The corresponding idealized test matrix is represented in Figure 1, where each fuel is operated at all CA50-EGR conditions for COV of IMEP below ~5%. If a CA50-EGR condition has a COV of IMEP higher than 5%, that condition is typically not operated for a given fuel, though there are several exceptions where post-processing of the data reveals that the COV for the 1000 cycles is higher than expected.

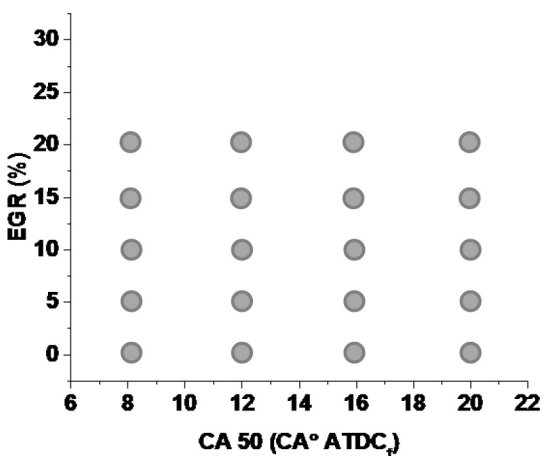


Figure 1. Experimental test matrix of EGR and CA50 conditions

Fuels Matrix

A total of 6 fuels are examined in this study, all of which are blended from four pure components: n-heptane, iso-octane, toluene, and ethanol. All of the fuels are blended to have a target RON of 95 using correlations from the literature [20, 21]. A RON of 95 is chosen to minimize the autoignition differences between these fuels, and the resultant effect that this could have on the combustion duration. The RON relationship between n-heptane, iso-octane, and toluene can be

found in Morgan et al. [20], and the RON relationship for the ethanol-containing blends is determined by interpolating the findings from Foong et al. [21]. The resulting surrogate compositions tested in the present study are shown in Figure 2.

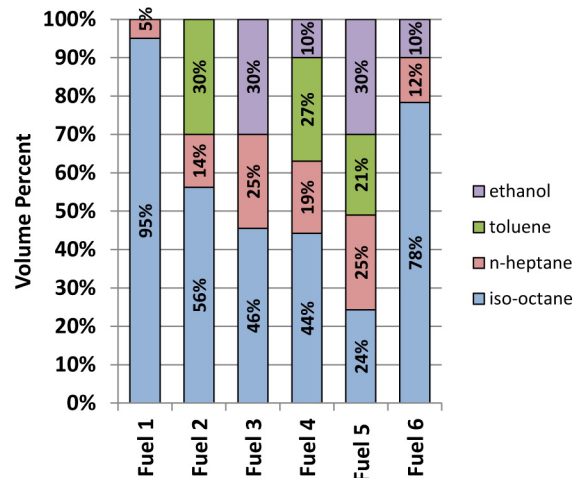


Figure 2. Composition of the 6 fuel blends investigated.

To confirm the fuel properties of the blends in Figure 2, ASTM tests by an independent laboratory are conducted. The corresponding results of the measured RON and motor octane number (MON) values for these fuels are shown in Table 2. All measured values are ± 1.8 of the targeted RON of 95, which is thought to be acceptable given the light load operating conditions being studied.

Table 2. RON, MON and S for the fuels investigated.

	RON	MON	S (RON - MON)
Fuel 1	95.1	94.7	0.4
Fuel 2	93.4	88	5.4
Fuel 3	95.3	88	7.3
Fuel 4	93.5	87.1	6.4
Fuel 5	96.8	87.3	9.5
Fuel 6	95.4	92.2	3.2

Laminar Flame Speed Calculations

The laminar flame speed for the fuels is determined in two ways. The first is an empirical correlation adapted from Sileghem et al. [22], which estimates flame speed from the energy fraction mixing rule shown in Equation 1, where U_l is the laminar flame speed, Φ is the equivalence ratio, and y is the mass fraction of the fuel component. The flame speed inputs used in the present work are from experimental measurements of flame speed found in [22,23], and the calculated flame speed for the fuels in the present study are shown in Table 3. Also in Table 3 is a range of the flame speed of commercial gasoline at the same starting conditions.

$$U_{l,blend}(\Phi) = \sum_{i=1}^n y_i * U_{l,i}(\Phi)$$

Equation 1



Table 3. Laminar flame speed at 1 atm pressure, 358K.

	U_f [cm/s]	Reference
iso-octane	44	[22]
n-heptane	53	[22]
Toluene	46	[22]
Ethanol	55	[23]
Commercial gasoline	45-48	[22]

The second approach to determine flame speed is through kinetic models, using the flame speed calculation function of Chemkin-PRO. This approach is viable across a broader temperature and pressure space, with varying levels of EGR. For this, a reduced version of the gasoline surrogate chemical kinetic mechanism from LLNL is used [24]. The mechanism has been reduced to 679 species and 3479 reactions by Professor Tianfeng Lu at the University of Connecticut and received through a personal communication.

RESULTS

The results are divided into three main sub-sections. The first provides the general engine efficiency and emission trends for the investigated experimental space. The second uses combustion analysis to examine the fuel-specific trends on combustion duration. The third provides flame speed calculation results and puts them in the context of the engine experiments.

General Engine Trends

All fuels exhibited the same qualitative trends with respect to efficiency and emissions over the EGR and combustion phasing space investigated. While fuel-specific trends are highlighted in the next sub-section, results from Fuel 1 are presented in the present subsection as a representative example of engine operation.

Net indicated thermal efficiency from these experiments follows the expected trends, as shown in Figure 3. Maximum efficiency is achieved at the most advanced combustion phasing, 8 CA aTDC_f, and at a moderately high EGR level of approximately 10-15% EGR. Efficiency decreases at retarded phasing due to under-utilized engine expansion, and it decreases at higher EGR levels due to excessively long combustion duration and combustion instabilities. As a result of the variable efficiency at a constant fueling rate, the engine load also changes over a range of approximately 10%, as shown in Figure 4.

Gaseous emissions from the engine also follow the expected trends. Figure 5 shows that NO_x emissions decrease both with increasing EGR and with retarded combustion phasing. Retarding the CA50 combustion phasing from 8 CA aTDC_f to 20 CA aTDC_f in the absence of EGR reduces NO_x emissions by more than a factor of 2, from 9.6 g/kWh to 4.6 g/kWh. Increasing the EGR to 20% reduces the NO_x emissions by about a factor of 7, from 7.4 g/kWh to 1.05 g/kWh, at a constant CA50 combustion phasing of 12 CA aTDC_f.

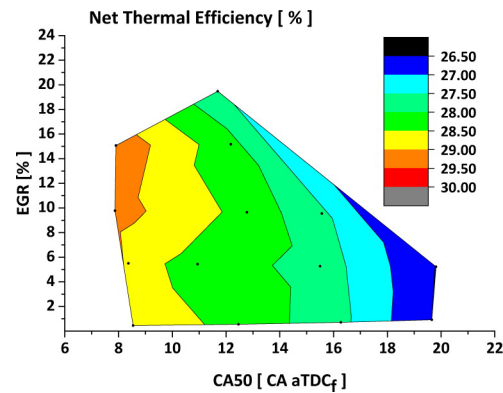


Figure 3. Contour of engine efficiency for Fuel 1 as a function of CA50 combustion phasing and EGR level.

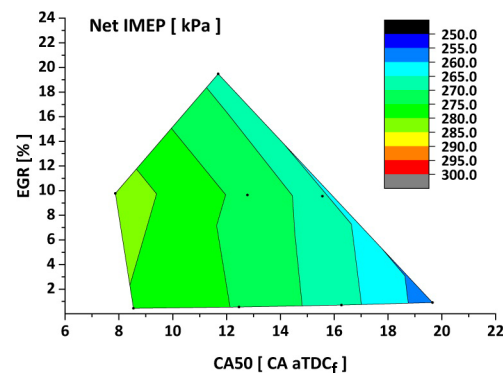


Figure 4. Contour of engine load for Fuel 1 as a function of CA50 combustion phasing and EGR level.

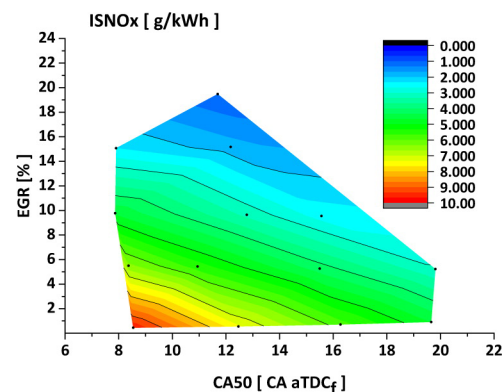


Figure 5. Contour of indicated specific NO_x emissions for Fuel 1 as a function of CA50 combustion phasing and EGR level.

A contour plot of combustion efficiency, which captures the combined trends of CO and unburned HC emissions, is shown in Figure 6. The combustion efficiency varies over a relatively narrow range, from 97% to 98%. Although others have shown that EGR dilution can affect hydrocarbon emissions [25], the present work found that at the tested conditions EGR has only a modest effect on combustion efficiency. However, retarding combustion phasing was found to have a stronger effect, with proportionally increased