

Effects of Fuel Laminar Flame Speed Compared to Engine Tumble Ratio, Ignition Energy, and Injection Strategy on Lean and EGR Dilute Spark Ignition Combustion

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ABSTRACT

Previous studies have shown that fuels with higher laminar flame speed also have increased tolerance to EGR dilution. In this work, the effects of fuel laminar flame speed on both lean and EGR dilute spark ignition combustion stability were examined. Fuels blends of pure components (iso-octane, n-heptane, toluene, ethanol, and methanol) were derived at two levels of laminar flame speed. Each fuel blend was tested in a single-cylinder spark-ignition engine under both lean-out and EGR dilution sweeps until the coefficient of variance of indicated mean effective pressure increased above thresholds of 3% and 5%. The relative importance of fuel laminar flame speed to changes to engine design parameters (spark ignition energy, tumble ratio, and port vs. direct injection) was also assessed. Results showed that fuel laminar flame speed can have as big an effect on lean or EGR dilute engine operation as engine design parameters, with the largest effects seen during EGR dilute operation and when changes were made to cylinder charge motion.

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INTRODUCTION

For reasons of climate change mitigation and reduced dependence on petroleum, there is increasing interest in improving engine efficiency and introducing secure and low carbon impact alternative fuels. Spark-ignition engines continue to dominate the passenger vehicle fleet in the United States, and are associated with the largest portion of worldwide transportation energy consumption [1,2]. One of the pathways for increased spark-ignition engine efficiency is through charge dilution, either with exhaust gas recirculation (EGR) or by a leaner air-fuel mixture [3]. In both cases, efficiency gains are realized through lower throttling and pumping losses, reduced heat transfer losses from the combustion gases to the combustion chamber surfaces, and increased work extraction during the engine expansion stroke through a higher ratio of specific heats of the working fluid [4]. Dilute EGR operation does not typically achieve as high engine thermodynamic efficiencies as lean operation because of reduced ratio of specific heats and dissociation of carbon dioxide (CO₂) in the hot combustion products near piston top dead center (TDC) [4]. However dilute EGR operation does allow for lower cost exhaust aftertreatment than lean operation in the form of a conventional three-way catalyst.

The level of charge dilution, and engine efficiency improvement, is limited by combustion stability [5]. Previous work has shown that the level of EGR dilution or lean tolerance could be improved by either fuel properties or engine design parameters [5]. Others have shown that EGR dilution tolerance could be extended with higher fuel laminar flame speed (LFS). Research by Szybist and Splitter explained that higher EGR dilution tolerance with increased LFS fuels occurred via shorter flame development angle (FDA), which is the crank angle duration from spark timing to start of combustion [6]. They also found that EGR dilution tolerance increased when ethanol (a high LFS and high heat of vaporization fuel) was added, despite the effects of increased charge cooling.

Engine design parameters, such as tumble and ignition characteristics, can also have important effects on EGR dilution and lean tolerance. Tumble has been shown to reduce the duration of flame kernel development, provided the flame kernel is not extinguished due to excessive stretch rate [7]. Increased ignition energy has also been shown to be beneficial on reducing cycle to cycle variations under EGR dilute and lean conditions [4].

The objective of this work was to first, quantify the EGR dilution and lean operation limits and combustion characteristics of five pure component fuel blends with two fixed levels of LFS and heat of vaporization (HOV). Two levels of HOV were set at the respective LFS level in order to rule out HOV effects on lean or EGR dilution limits within a given fuel grouping. As an additional objective, it will be further tested how representative fuel LFS values calculated at significantly lower temperature and pressure than engine conditions correlate with fuel EGR dilution and lean operating limits in an actual engine. Second, the effects of tumble and ignition energy on EGR dilution and lean operation limits and combustion characteristics were tested with two representative fuels. An extension of these tests was performed with one fuel to examine the effects of charge motion induced by direct injection (DI) versus port fuel injection (PFI). Third, the relative impacts of fuel LFS and engine design parameters on EGR dilution and lean operation tolerances and combustion characteristics were compared.

EXPERIMENTAL SETUP

Engine Description

Experiments were performed on a single-cylinder spark-ignition research engine, capable of DI or PFI operation. The geometries of this engine make it representative of modern gasoline direct injection (GDI) engines. Specifications of the engine setup can be found in [Table 1](#).

Table 1. Engine Specifications.

Displacement (l)	0.63
Bore x Stroke (mm)	89.04 x 100.6
Combustion Chamber	40° pentroof
Compression Ratio (-)	12.1:1
Number of Valves (-)	4
Spark Plug	NGK ILTR6F9, 0.7 mm gap
PFI Injector	Bosch XS4U-AB
DI Injector	6 hole, solenoid, central-mount
Tumble Ratios (-)	0.6, 1.5
Ignition Energy (mJ)	75, 150 (single strike)

The engine head has a 40° pentroof design, with centrally located spark plug and DI injector. The PFI injector was located in the intake runner, sufficiently upstream of the two intake ports to ensure adequate fuel-air mixing. To achieve the lower tumble ratio noted in [Table 1](#), tumble plates were removed from the intake ports to reduce the desired bulk gas motion in the cylinder. Tumble ratio was defined using simulations with the piston at bottom dead center.

The engine may be operated boosted or naturally aspirated. For the purposes of testing in this work, the engine was operated under naturally aspirated conditions. Throttled operation of the engine was performed by a pilot-operated Parker gate valve in the intake air system. A cooled high pressure EGR loop transferred exhaust gases into the intake air system upstream of the intake surge tank (downstream of the Parker gate valve),

which gave a homogeneous EGR mixture with the intake air by induction into the cylinder. The EGR rate was controlled with an automotive production-grade EGR valve. Engine coolant, maintained at 85°C, was used in the EGR cooler. The EGR ratio was measured based on concentrations of CO₂ in the intake and exhaust systems by a Pierburg AMA2000 emissions bench.

A programmable spark-based ignition system, the Directed Energy Ignition System (DEIS) produced by Altronics, Inc., (a capacitive-discharge system) was used in this study. The DEIS allows the user to predefine an amount of voltage and current to the coil for each spark event, even allowing multiple energy pulses. In this work, nominal ignition energy levels of 75 ± 3.5 mJ and 150 ± 6 mJ were programmed into the DEIS with constant duration at approximately 0.5 ms.

Engine combustion characteristics were measured by a flush-mounted Kistler 6125 pressure transducer and AVL INDICOM and Concerto softwares. Cylinder pressure pegging was performed by matching the cylinder pressure with high-speed intake air manifold pressure at piston bottom dead center (BDC) of the intake stroke. High speed intake manifold pressure was measured by a flush-mounted pressure transducer. Apparent heat release rate (AHRR), meaning no cylinder heat transfer was taken into account, was calculated in Concerto, along with the mass fraction burn curve and bulk gas temperature profile. The combustion metrics used in this work included the timings (in engine crankshaft angle degrees) of 5%, 50%, and 90% mass fraction burned (CA5, CA50, and CA90, respectively). Important intervals that were considered included: spark timing to CA5, CA5 to CA50, and CA50 to CA90. Spark timing to CA5 was used as a representative duration of flame kernel development, similar to Szybist and Splitter [6]. The duration from CA5 to CA90 was used as representative of combustion duration (CD). The timing of CA50 is traditionally used as representative of combustion phasing.

Test Fuels

Five test fuel blends were designed from pure components: toluene, iso-octane, n-heptane, ethanol, and methanol. The characteristics of each fuel can be found in [Table 2](#).

Table 2. Fuel blend characteristics.

Fuel #	1	2	3	4	5
LFS (cm/s)	45.7	45.7	45.7	52.2	52.2
HOV (kJ/MJ)	10.1	10.1	10.1	30.0	30.0
Toluene (%_vol)	100	-	-	-	-
Iso-octane (%_vol)	-	75.2	73.6	8.0	-
N-heptane (%_vol)	-	9.0	15.7	-	38.7
Ethanol (%_vol)	-	15.8	-	92	-
Methanol (%_vol)	-	-	10.7	-	61.3

Values of LFS were calculated based on the Le Chatelier energy and molar-weighting method described by Sileghem et. al [8]. These LFS estimates were compared against Laminar Burning Velocities (LBV) performed in CHEMKIN at the same temperature and pressure conditions ([Figure 1](#)), and the trends were found to agree quite well.

The CHEMKIN LFS predictions were on average only 1-2 cm/s higher than the Le Chatelier method. In general, it was encouraging that for both methods of LFS calculation, the fuels designed to have higher LFS (Fuels 4 and 5) were predicted to be faster in both calculations. It has been shown that the LFS of fuels measured at lower pressure and temperature than engine combustion conditions may still be representative, at least trend-wise, of fuel LFS when operated in engines [6].

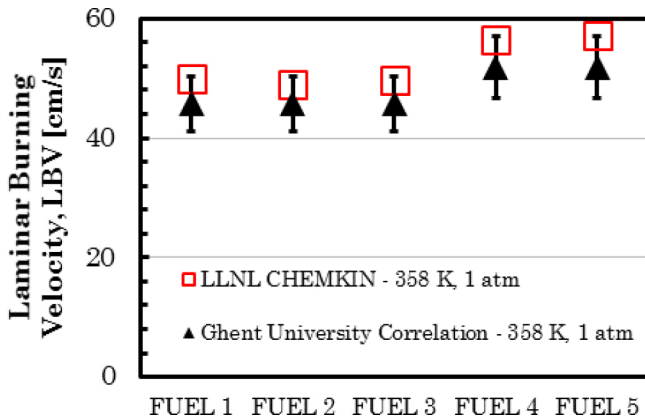


Figure 1. Comparison of fuel laminar burning velocities based on Le Chatelier method and CHEMKIN simulations at 1 atm and 358 K. The uncertainty in the measurements are represented by the error bars.

Szybist and Splitter analyzed the effects of fuel LFS on EGR dilution tolerance for various blends of these components at approximately constant RON 95 [6]. In their work, the blend LFS, HOV, and chemical composition varied amongst the fuels. In the work presented here, the blends of Fuel 1 to Fuel 3 were designed to have a fixed LFS of approximately 45.7 cm/s (at 1 atm, 358 K) and fuel energy weighted HOV of 10.1 kJ/MJ, consistent with pure toluene (Table 2). Fuel 4 and Fuel 5 were designed to be at a higher LFS of approximately 52.2 cm/s and fuel energy weighted HOV of 30.0 kJ/MJ, by utilizing the high LFS and HOV properties of ethanol and methanol. A fuel energy-weighted (lower heating value weighted) HOV was calculated for each fuel to take into account variations in fuel mass between fuel blends at constant engine load. This was calculated for each fuel blend based on pure component values of the HOV at 1 atm and 298 K (rather than 1 atm and boiling point temperature), and weighted by lower heating values measured for the component initially in the gas phase (rather than being initially in the liquid phase) since these are the conditions in the combustion chamber. This is represented graphically in Figure 2. In this way it was possible to analyze several fuels at two distinct levels of LFS and HOV, while allowing the chemical composition to vary. Being that toluene falls inside the triangles made up of iso-octane, n-heptane, and ethanol (or methanol), it was possible to create fuel blends (Fuel 2 and Fuel 3) of those three components which had the same energy and molar-weighted LFS and energy-weighted HOV as toluene. On the higher LFS and HOV side, it was decided to design two-component fuels which would match at the point where the ethanol and methanol triangle edges intersected (Fuels 4 and 5). The total range of fuel LFS covered in this fuels matrix is therefore from 45.7 to 52.2 cm/s, which covers the typical range of commercial gasolines [9].

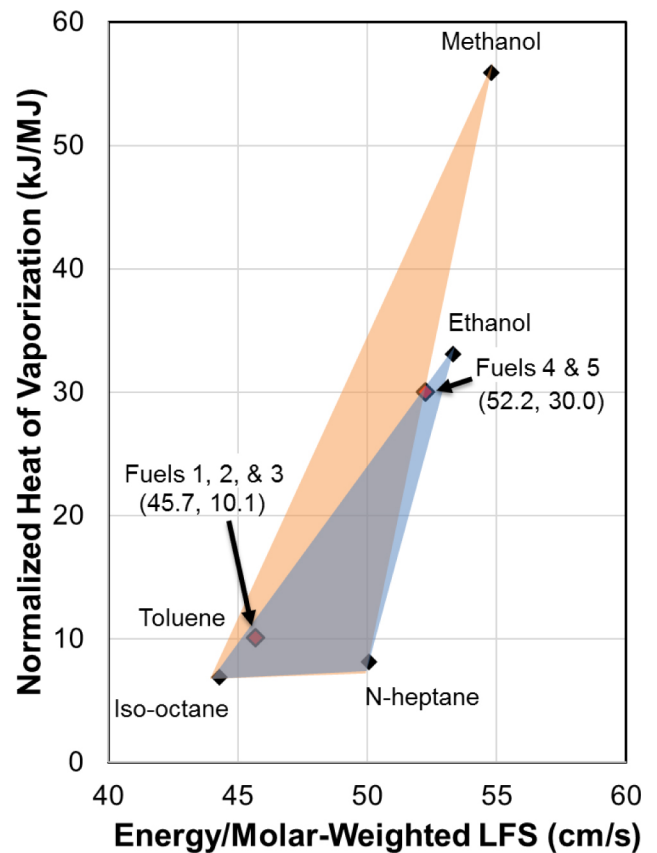


Figure 2. Graphical representation of fuel blending.

Test Methodology

For each test fuel and engine design parameter configuration (tumble and ignition energy) of interest, sweeps for EGR dilution tolerance and lean operation limits were performed at constant engine speed, load, and combustion phasing. Details of the tested engine operating conditions are shown in Table 3. The engine speed and load chosen for these tests were 1500 revolutions per minute (RPM) and 5.6 bar net indicated mean effective pressure (IMEP_n). The combustion phasing was maintained at 8 °aTDC ± 1 crank angle degree (CAD). The comparison of low and high tumble ratio (0.6 and 1.5) and ignition energy (75 and 150 mJ) were only performed on Fuel 2 and Fuel 4. These two fuels were chosen as representative of a low and high LFS and HOV fuel because they both used ethanol, which is more widely used to blend into commercial gasoline than methanol. Most of the testing was performed with PFI operation, but a comparison was also made between PFI and DI using Fuel 1 (100% toluene).

Table 3. Experimental engine operating conditions.

Speed (RPM)	1500
IMEP _n (bar)	5.6
Combustion Phasing (°aTDC)	8 °aTDC ± 1 CAD
Intake Conditions	Atmospheric/Throttled
PFI Operation	SOI = 300 °bTDC P _{inj} = 60 psig (4 bar)
DI Operation (Only Fuel 1)	SOI = 300 °bTDC P _{inj} = 150 psig

The intake conditions were dependent on the target level of EGR or lean operation. Under EGR dilute operation, the air-fuel ratio was maintained at stoichiometry by adjusting intake throttling for a given IMEP. This increased the intake manifold air pressure as EGR was introduced, thus reducing engine pumping work. For lean operation, the intake throttle was similarly opened to supply the excess air required, thereby increasing intake manifold air pressure and reducing pumping work. Both types of sweeps were performed until 5% coefficient of variance (COV) of IMEP_n was exceeded. Figure 3 and Figure 4 show examples of typical EGR dilution and lean lambda sweeps from stoichiometry, and 0% EGR operation, until combustion stability deteriorates past 3% and 5% COV of IMEP.

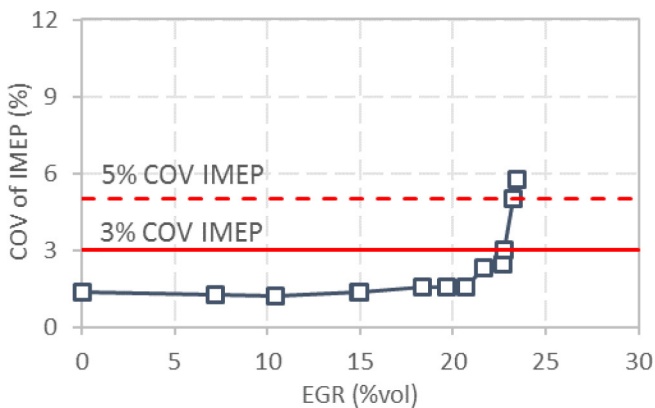


Figure 3. Typical trend of COV of IMEP during an EGR dilution sweep.

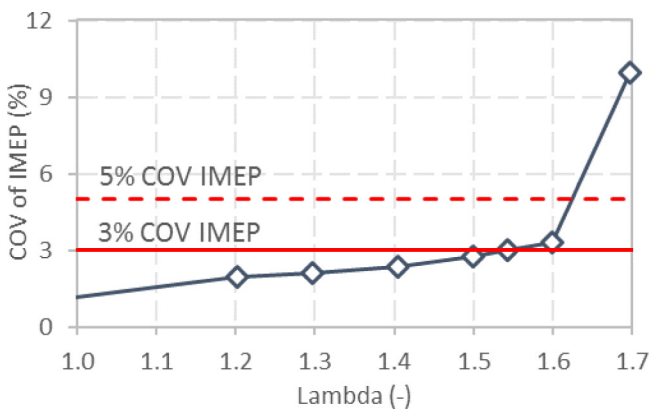


Figure 4. Typical trend of COV of IMEP during a lean lambda sweep.

Analyses of the dilute EGR and lean lambda tolerance of each fuel and engine design parameter were performed in three ways. First, the COV of IMEP and combustion metrics of each test condition (fuel and engine) were evaluated at stoichiometric operation without EGR as a baseline.

Second, the level of COV of IMEP and combustion metrics were analyzed at two fixed levels of EGR dilution or lambda. For this work, levels of 15% and 20% EGR were chosen. At the 15% EGR level, all test conditions exhibited 3% or less COV of IMEP. The 20% EGR level was the highest dilution where all test conditions were still able to operate. At the 20% level of EGR, the most stable conditions had COV of IMEP less than 3%, while the less stable conditions had values of COV of IMEP well over 3% or 5%. For lean conditions,

combustion stability and characteristics were compared at lambda 1.4 (where all conditions were stable) and lambda 1.6 (where only some of the conditions had a stability of 3% COV of IMEP).

The third comparison between fuels and engine design parameters for the dilute EGR and lean lambda sweeps was at fixed levels of 3% and 5% COV of IMEP. In order to assess the combustion characteristics at these set combustion stability targets (3% and 5% COV of IMEP), linear interpolations of the combustion burn curve data were performed for the nearest COV of IMEP points in the dilute EGR or lean lambda sweeps. For example, an interpolation of the mass fraction burn information for test points at 2.5% COV of IMEP and 3.5% COV of IMEP provided the mass fraction burn characteristics of a test point under those conditions with a 3% COV of IMEP. While the accuracy of the EGR dilution tolerance at 5% COV of IMEP could be assessed within 0.5-1% EGR, a lean lambda tolerance greater than 0.05 lambda was required for distinctions in the data at these higher levels of dilution.

RESULTS AND DISCUSSION

Effects of Fuel Properties

Baseline Conditions

Baseline tests at 0% EGR and stoichiometric operation were performed for all fuels, and variations in tumble and ignition energy were performed for two select fuels (Fuel 2 and Fuel 4). Figure 5 shows the combustion metrics of each fuel at baseline conditions. As mentioned in the Test Methodology section, the spark timing was adjusted to maintain CA50 at 8 °aTDC ± 1 CAD. The COV of IMEP for all fuels at baseline conditions was very stable, from 1.0 to 1.3%.

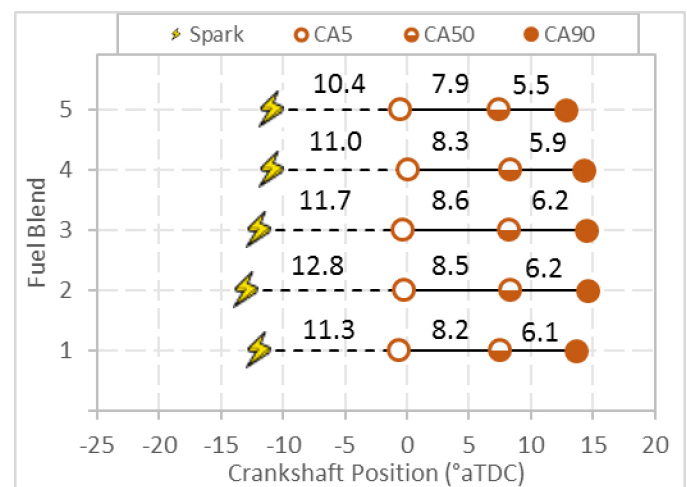


Figure 5. Effects of fuel blends on combustion metrics at baseline conditions (0% EGR, stoichiometry).

As discussed earlier in the paper, FDA is generally shorter for fuels with higher LFS due to a faster flame kernel development. In general, all fuels were very stable and had similar FDA (within approximately 2.5 CAD). Between the low and high LFS ethanol fuel blends (Fuel 2 and Fuel 4, respectively) and methanol fuel blends (Fuel 3 and Fuel 5, respectively), the FDA was about 1-2 CAD. The low LFS single component fuel (Fuel 1) had an