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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.

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 (CO_2) 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 CO_2 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].

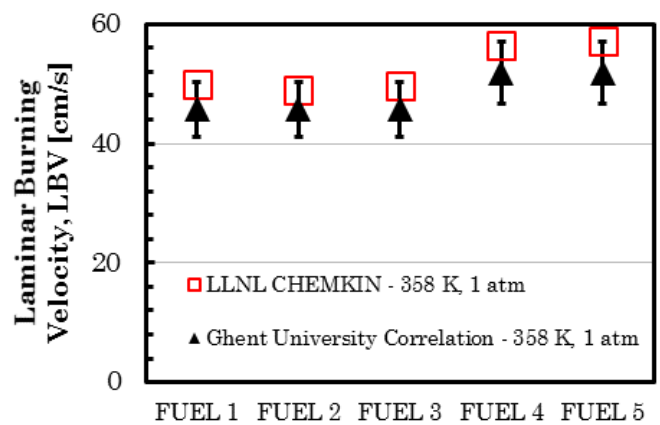

Figure 1. Comparison of fuel laminar burning velocities based on Le Chatelier method and CHEMKIN simulations at 1 atm and 358 K. The uncertainty in the measurements are represented by the error bars.

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].

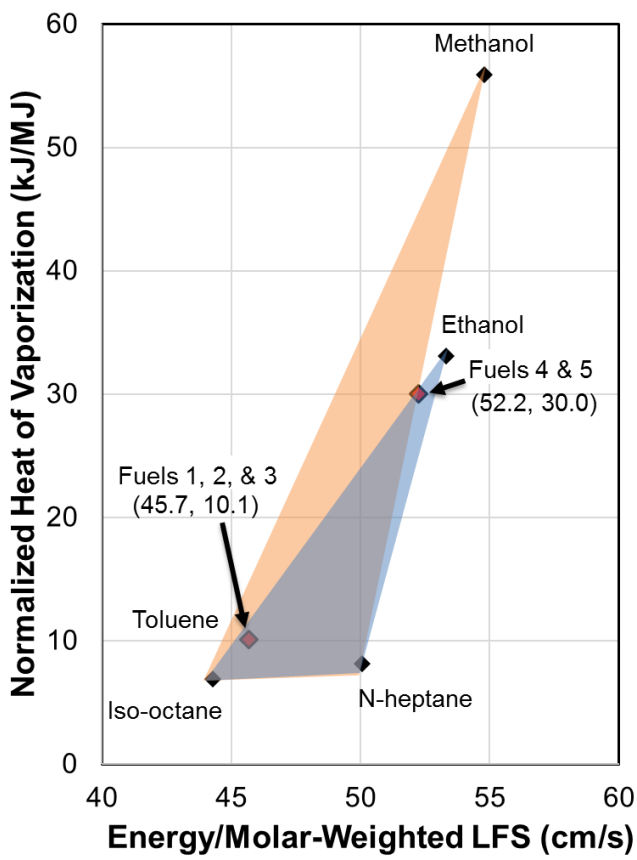

Figure 2. Graphical representation of fuel blending.

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 bar

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.

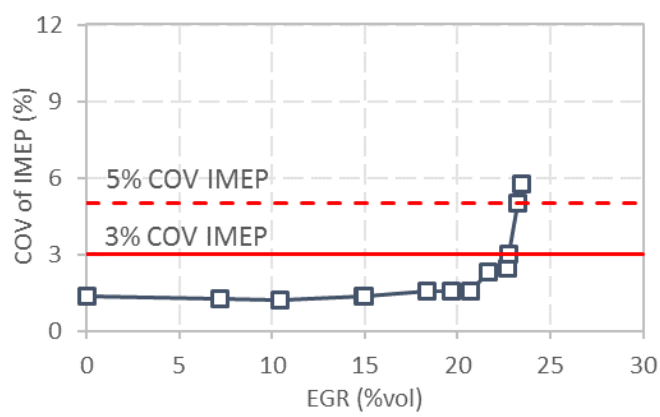

Figure 3. Typical trend of COV of IMEP during an EGR dilution sweep.

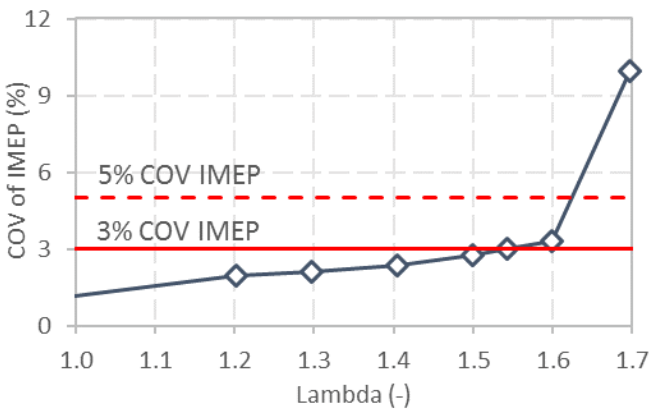

Figure 4. Typical trend of COV of IMEP during a lean lambda sweep.

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%.

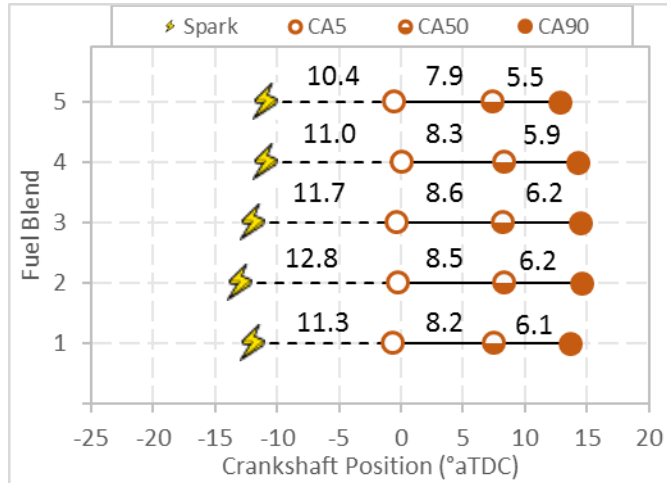

Figure 5. Effects of fuel blends on combustion metrics at baseline conditions (0% EGR, stoichiometry).

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 shorter with higher LFS. The low LFS single component toluene fuel (Fuel 1) had an FDA which was longer than the high LFS Fuel 5 by about 1 CAD, however it was 1.5 CAD shorter than the low LFS Fuel 2. But Fuel 1's FDA was not significantly different (for this work, more than 0.5 CAD) than the high LFS Fuel 4 or low LFS Fuel 3. Similar observations were made for the differences in combustion duration (CD) between the fuels.

EGR Dilution Tolerance

As shown as an example in Figure 3, the EGR dilution tolerance of each fuel blend was tested by an EGR sweep at stoichiometric air-fuel ratio (lambda of unity). The EGR sweep started at 0% EGR and continued until the COV of IMEP exceeded 5%. Along the EGR sweep, there were a few points of interest for comparisons of each fuel blend's tolerance and performance under dilute EGR conditions. The two types of analyses were at fixed EGR level (15% and 20%) and fixed COV of IMEP (3% and 5%). The quantities of 15% and 20% EGR were chosen to represent, at 15%, a relatively higher EGR level where all fuels still had very stable combustion and, at 20%, a level of EGR high enough that not all of the lower LFS fuel blends were able to maintain stable combustion. The second type of analysis was at 3% COV of IMEP, the industry-accepted limit for combustion stability, and 5% COV of IMEP, where comparisons could be made with less-stable combustion. The EGR dilution tolerance achieved of each fuel blend was analyzed at each level of combustion stability.

Constant EGR Rate Analysis

At 15% EGR, all of the fuel blends maintained a very stable level of combustion stability of 1.2-1.4% COV of IMEP. Figure 6 shows the combustion characteristics of each fuel blend at 15% EGR and stoichiometry. In this case, the FDA of the high LFS Fuel 4 and Fuel 5 are at least 0.5 CAD shorter than any of the low LFS fuel blends. The difference in FDA between Fuel 2 and Fuel 4 (ethanol based) and Fuel 3 and Fuel 5 (methanol based) is about 3.5 CAD. Once again, Fuel 1 (100% toluene) had a significantly shorter FDA than Fuel 2 and Fuel 3, but its FDA was still longer than the high LFS Fuel 4 or Fuel 5 by more than 0.5 CAD.

For combustion duration, Fuel 4 and Fuel 5 had shorter combustion duration than Fuel 2 and Fuel 3 by approximately 1-2 CAD. However, Fuel 1 had a combustion duration similar to that of Fuel 4.

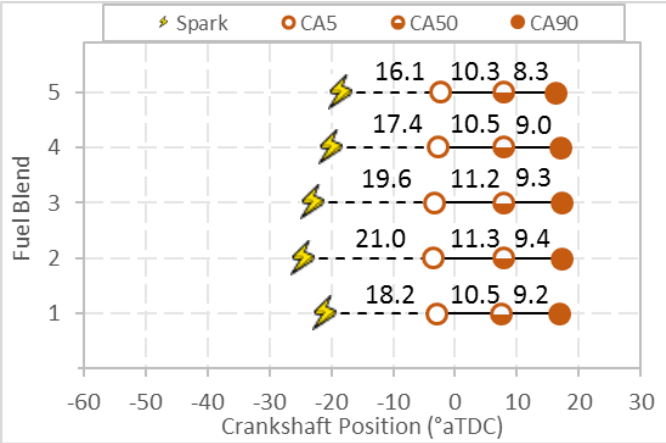

Figure 6. Effects of fuel blend LFS on combustion metrics at 15% EGR and stoichiometry.

At 20% EGR (Figure 7), the trends observed in FDA and CD at 15% EGR were further amplified between low and high LFS fuel blends. Fuel 4 and Fuel 5, with higher LFS, had the shortest FDA and CD, as well as the lowest COV of IMEP. The difference in FDA between the low and high LFS ethanol and methanol fuels (Fuel 2 vs. Fuel 4 and Fuel 3 vs. Fuel 5, respectively) was approximately 8 CAD. This difference in FDA and the subsequent effect on CD caused unacceptable combustion stabilities in Fuel 2 and Fuel 3. As seen at baseline conditions and 15% EGR, Fuel 1 performed somewhere between the high LFS fuel blends and the other low LFS fuel blends. From this work it is not clear why Fuel 1 (100% toluene) would perform better than the other low LFS fuel blends, and further work is needed to discern if fuel dilution tolerance may change, with respect to other fuels, based on the level of EGR or lean dilution.

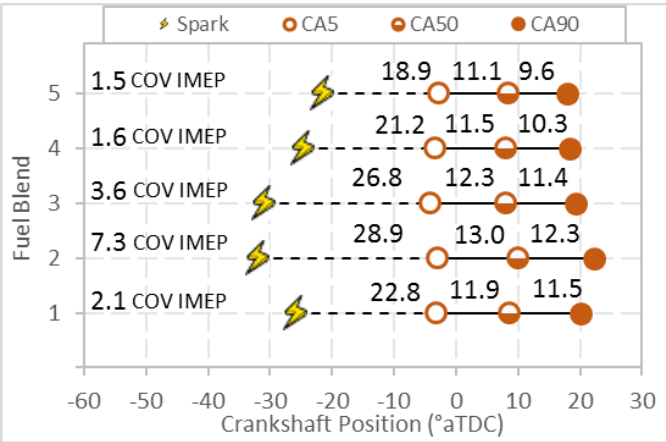

Figure 7. Effects of fuel blend LFS on combustion metrics at 20% EGR and stoichiometry.

Constant COV of IMEP Analysis

Higher EGR tolerance allows for more efficient engine operation through reduced throttling losses, lower heat transfer losses, and earlier knock limited spark advance (KLSA) [4]. The EGR dilution tolerance of each fuel blend at a fixed level of combustion stability (3% COV of IMEP) is shown in Figure 8. As was predicted by fuel blend LFS calculations (using data at 1 atm, 358 K in CHEMKIN), the high LFS Fuel 4 and Fuel 5 had approximately a 4-6% EGR dilution tolerance advantage over the lower LFS Fuel 2 and Fuel 3. Fuel 1, consistent with previous observations, performed a little better than Fuel 2 and Fuel 3, but not as well as Fuel 4 and Fuel 5.

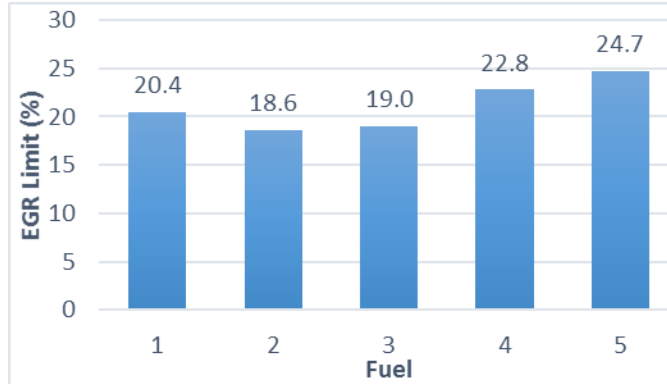

Figure 8. The EGR dilution tolerance at 3% COV of IMEP between all fuel blends.

Extending the combustion stability limit from 3% to 5% COV of IMEP allowed additional EGR for each fuel blend by about 0.5-1%. Between Fuel 2 and Fuel 4, and Fuel 3 and Fuel 5, there was about a 0.5-1% increase in EGR dilution tolerance with higher LFS fuel blends.

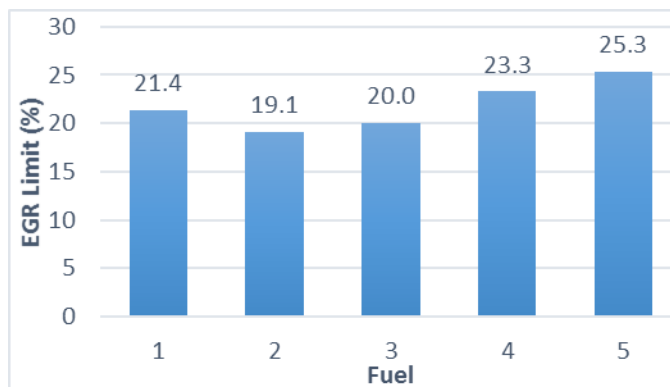

Figure 9. The EGR dilution tolerances of all fuel blends at 5% COV of IMEP.

Lean Tolerance

Constant Lambda Analysis

As shown as an example in Figure 4, a lean sweep was performed of each fuel blend from stoichiometry and 0% EGR at every 0.1 lambda until the combustion stability exceeded 5% COV of IMEP. A comparison is made of all fuels at lambda 1.4, where all fuels were still very stable, and lambda 1.6, where the slowest LFS fuel blends had COV of IMEP exceeding 5% COV of IMEP. The combustion characteristics of each fuel blend at 1.4 lambda are shown in Figure 10. For all tests at 1.4 lambda, combustion stability was very good, at COV of IMEP from 2.1 to 2.6%.

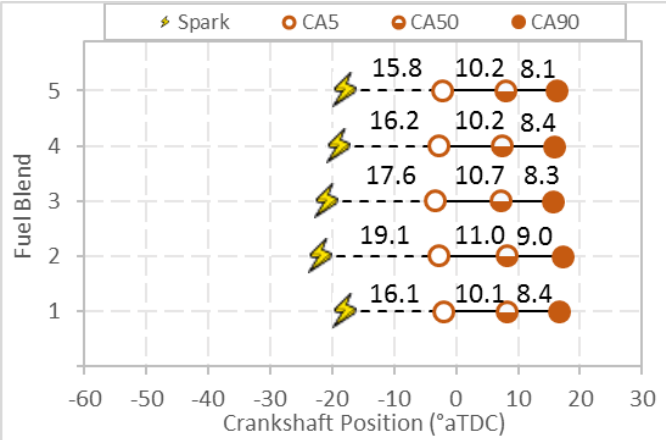

Figure 10. Effects of fuel blend LFS on combustion metrics at lambda 1.4 and 0% EGR.

Among the alcohol fuel blends, Fuel 4 had approximately 3 CAD shorter FDA than Fuel 2, and Fuel 5 approximately 2 CAD shorter FDA than Fuel 3. Similar trends were seen in the CD between these fuel blends. Fuel 1 interestingly had an FDA and CD more similar to Fuel 4 and Fuel 5, despite being calculated to have slower LFS.

Figure 11 shows the combustion characteristics of each fuel blend at lambda 1.6 and 0% EGR. The trends of higher LFS alcohol fuel blends compared to lower LFS alcohol continued at higher lambda, and more noticeably. Fuel 4 and Fuel 5 measured to have lower COV of IMEP, shorter FDA, and shorter CD than the slower LFS Fuel 2 and Fuel 3. Interestingly, Fuel 1 performed slightly better than Fuel 4, however not as well as Fuel 5.

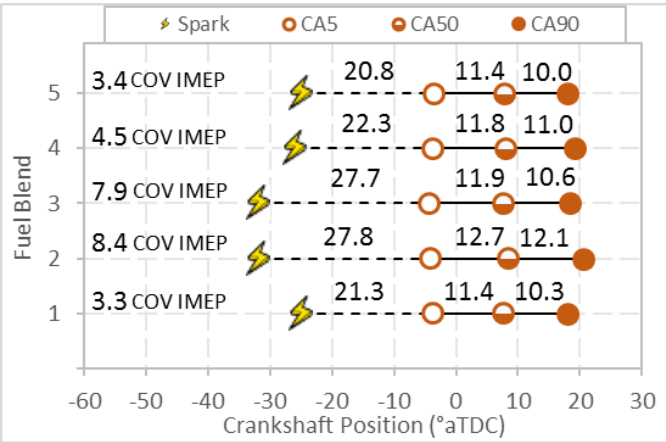

Figure 11. Effects of fuel blend LFS on combustion metrics at lambda 1.6 and 0% EGR.

Constant COV of IMEP Analysis

At 3% COV of IMEP, there were not significant differences in the lean lambda limits between fuel blends with the exception of Fuel 4, which for these tests had an unexpectedly low lean lambda tolerance (Figure 12).

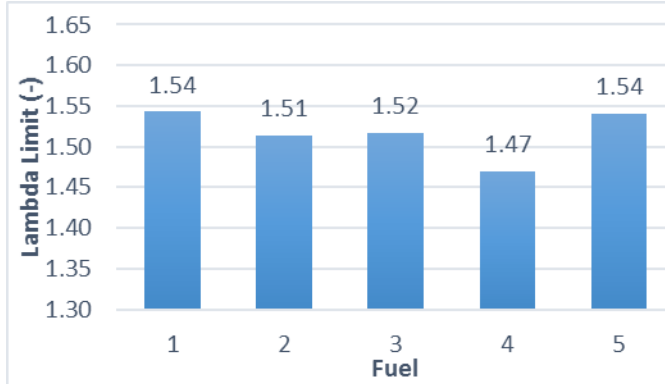

Figure 12. Lean tolerance of each fuel blend at 3% COV of IMEP and 0% EGR.

At 5% COV of IMEP, the expected trends of lean dilution tolerance based on calculated LFS return, and can be seen in Figure 13. In that case, the high LFS Fuel 4 achieved as high of lambda limit as Fuel 5. Interestingly, Fuel 1 continues to perform more like one of the higher LFS fuels than lower LFS fuels. It is proposed that the differences in lean lambda limits were more appreciable as the combustion stability decreased.

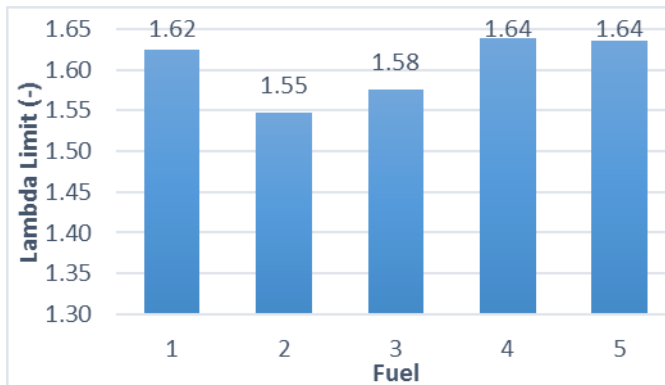

Figure 13. Lean tolerance of each fuel blend at 5% COV of IMEP and 0% EGR.

Effects of Engine Design Parameters

Baseline Conditions

Engine design parameters such as tumble ratio and ignition characteristics can affect EGR dilution and lambda lean tolerance. The effects of low tumble (LT) versus high tumble (HT) and low ignition energy (LIE) versus high ignition energy (HIE) for 0% EGR and stoichiometry on combustion metrics for Fuel 2 can be found in Figure 14 and for Fuel 4 in Figure 15.

For both Fuel 2 and Fuel 4, increasing tumble ratio from 0.6 to 1.5 decreased FDA and CD both by 6-7 CAD. These significant reductions in FDA suggest increased turbulent kinetic energy (TKE) helped to reduce the duration from spark timing to start of flame propagation. The level of impact of tumble ratio on FDA and CD by low and high LFS fuels was not seen at baseline conditions. The effects of LFS on FDA and CD under baseline conditions were more similar to the effects of ignition properties, on the order of 0.5-1.5 CAD. Comparing FDA between Fuel 2 and Fuel 4 across all combinations of engine design parameters, Fuel 4 had slightly shorter FDA by anywhere from no difference to approximately 2 CAD.

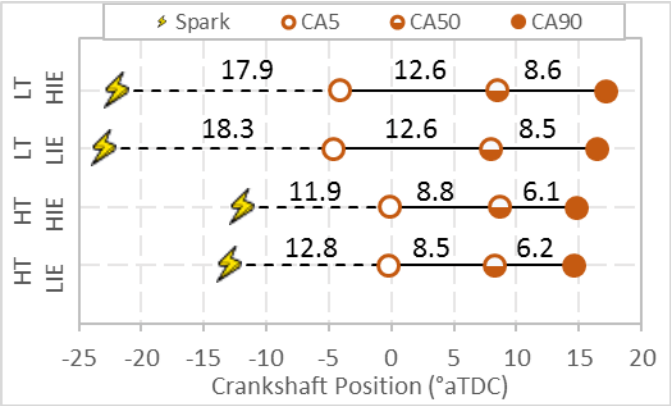

Figure 14. Effects of engine tumble and ignition energy on combustion metrics for Fuel 2 at 0% EGR and stoichiometry (Baseline conditions).

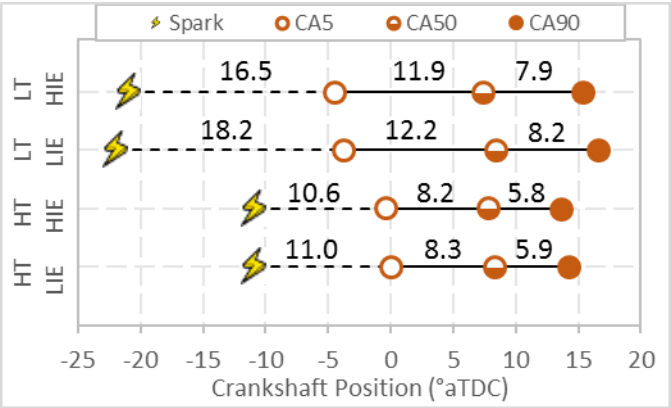

Figure 15. Effects of engine tumble and ignition energy on combustion metrics for Fuel 4 at 0% EGR and stoichiometry (Baseline conditions).

EGR Dilution Tolerance

Constant EGR Rate Analysis

In the same way as before with different fuel blends, EGR sweeps were performed with each configuration of engine design parameters for Fuel 2 and Fuel 4. Figure 16 and Figure 17 show the effects of tumble ratio and ignition energy on Fuel 2 and Fuel 4 combustion metrics, respectively, with 15% EGR. Where increasing tumble ratio at 0% EGR baseline conditions caused a 6-7 CAD reduction in FDA and CD, increased tumble caused a 16-17 CAD reduction in FDA for Fuel 2 and a 12 CAD reduction in FDA for Fuel 4. However, it should be pointed out that the higher LFS Fuel 4 still had an FDA 6-7 CAD shorter than Fuel 2 at low tumble, and 3-4 CAD shorter with high tumble. The CD with each fuel blend was also reduced by approximately 10 CAD with increased tumble. These reductions in FDA and CD were particularly helpful in reducing COV of IMEP, even at only 15% EGR. For Fuel 2, COV of IMEP was reduced by approximately 1% by high tumble, whereas COV of IMEP was reduced by 0.4-6% with high tumble for Fuel 4. The low tumble tests with Fuel 2 were the only ones with COV of IMEP greater than 2%. All other tests had a COV of IMEP between 1.3% and 1.9%.

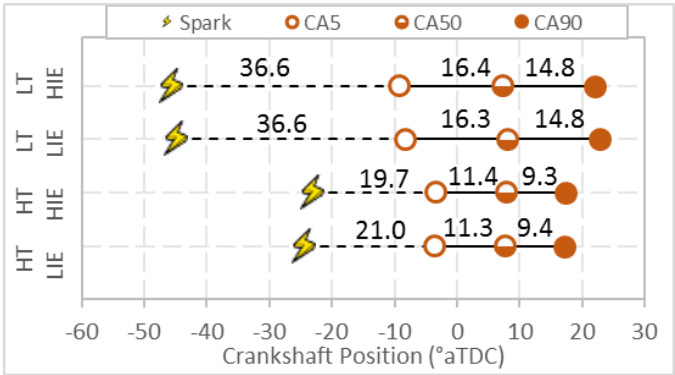

Figure 16. Effect of tumble ratio and ignition energy at 15% EGR and stoichiometry on Fuel 2.

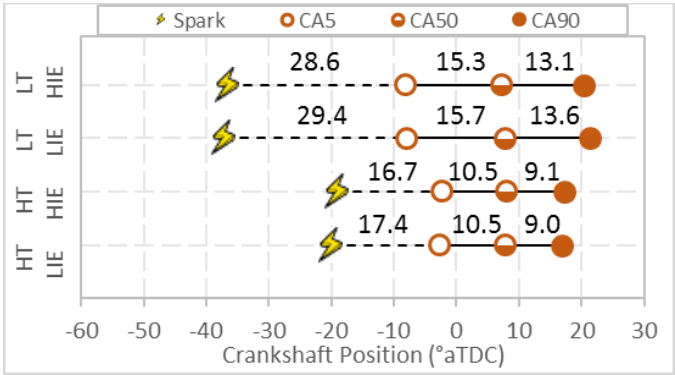

Figure 17. Effect of tumble ratio and ignition energy at 15% EGR and stoichiometry on Fuel 4.

At 20% EGR (Figure 18), only the most stable conditions had a COV of IMEP of approximately 3%. For both Fuel 2 and Fuel 4, the only combination of engine design parameters that gave a 3% COV of IMEP were high tumble and high ignition energy. Increased tumble ratio still had a large impact on FDA (16-26 CAD reduction), CD (11-12 CAD reduction), and combustion stability (0.5-7% reduction). At these higher EGR levels, near the edge of combustion stability, it appeared that ignition energy had more of an effect on combustion metrics and stability than at lower EGR levels.

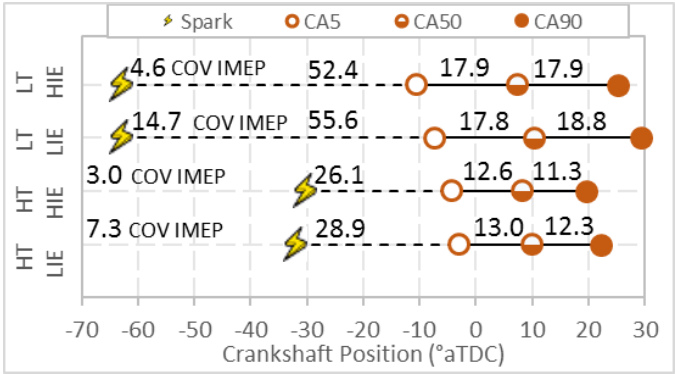

Figure 18. Effect of tumble ratio and ignition energy at 20% EGR and stoichiometry on Fuel 2.

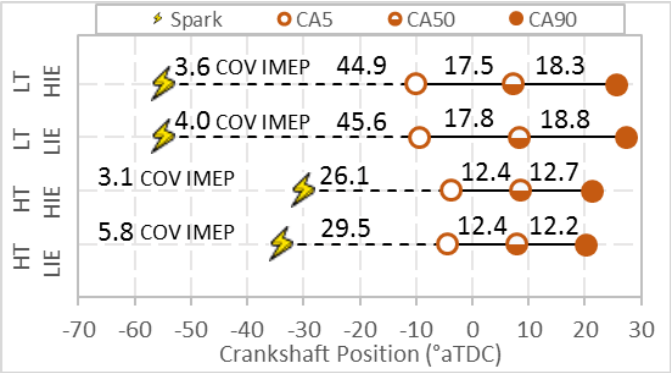

Figure 19. Effect of tumble ratio and ignition energy at 20% EGR and stoichiometry on Fuel 4.

Constant COV of IMEP Analysis

Figure 20 and Figure 21 show the EGR dilution tolerances of Fuel 2 and Fuel 4 under combinations of high and low tumble and ignition energy at 3% COV of IMEP. Increased tumble ratio from 0.6 to 1.5 generally increased EGR dilution tolerance by approximately 2-3% EGR for both Fuel 2 and Fuel 4, while increased ignition energy from 75 to 150 mJ (with constant spark duration) increased EGR dilution tolerance by approximately 0.5-1% EGR. For each tumble and ignition energy condition, increased LFS from Fuel 2 to Fuel 4 generally increased EGR dilution tolerance by approximately 4-5% EGR. This is similar to the LFS-based EGR dilution tolerance improvement observed in Figure 8 by comparing low and high LFS fuels from both ethanol (Fuel 2 and Fuel 4) and methanol (Fuel 2 and Fuel 4), as well as the 4% EGR tolerance seen by Szybist and Splitter [6]. These results suggest that at similar combustion stability (3% COV of IMEP), the improvement to EGR dilution tolerance from fuel LFS is at least equally as important as engine design parameters (certainly tumble).

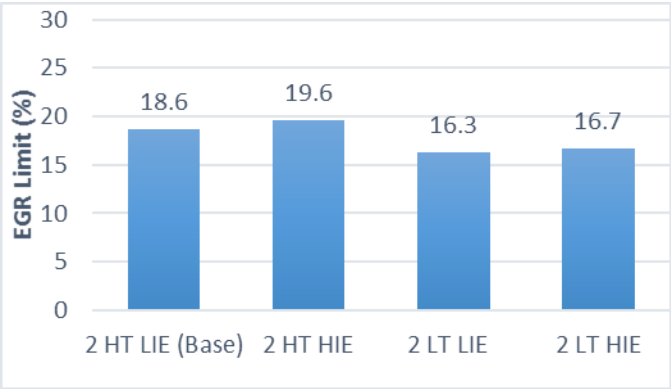

Figure 20. Comparison of engine design parameters on EGR dilution tolerance for Fuel 2 at 3% COV of IMEP.

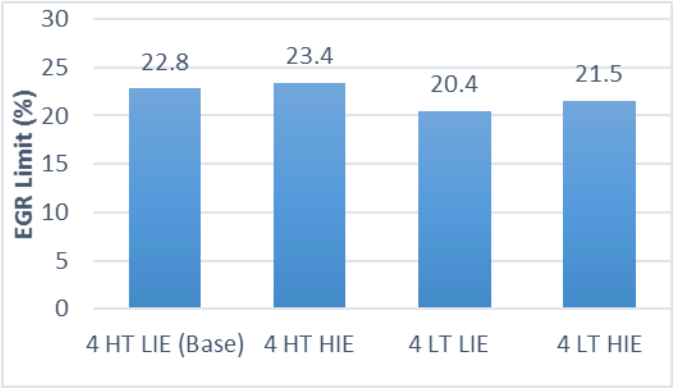

Figure 21. Comparison of engine design parameters on EGR dilution tolerance for Fuel 4 at 3% COV of IMEP.

Figure 22 and Figure 23 show the EGR dilution tolerances of Fuel 2 and Fuel 4 under conditions of high and low tumble and ignition energy at 5% COV of IMEP. These conditions were all less stable than at 3% COV of IMEP due to increased levels of EGR. Allowing the engine to operate at 5% over 3% COV of IMEP provided an increase in EGR levels by approximately 0.5-2% EGR. At 5% COV of IMEP, both increased tumble and increased ignition energy improved EGR dilution tolerance by approximately 0.5-1.5% EGR, which wasn't quite as advantageous as observed at 3% COV of IMEP. On the other hand, increased fuel LFS from Fuel 2 to Fuel 4 did still provide approximately a 4% EGR improvement in EGR dilution tolerance at these less stable combustion conditions.

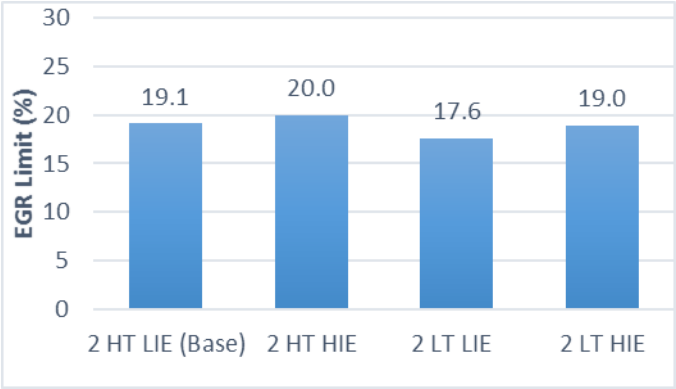

Figure 22. Comparison of engine design parameters on EGR dilution tolerance for Fuel 2 at 5% COV of IMEP.

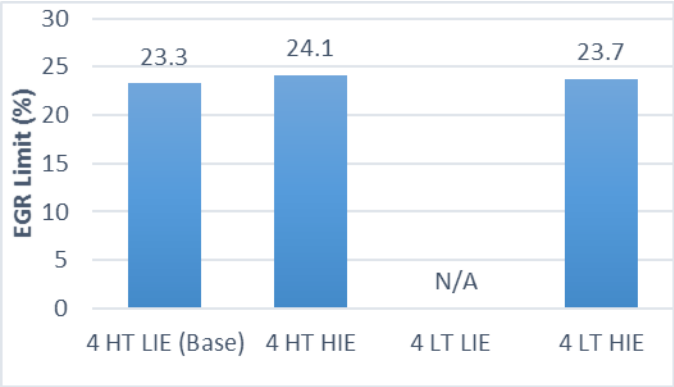

Figure 23. Comparison of engine design parameters on EGR dilution tolerance for Fuel 4 at 5% COV of IMEP.

Lean Tolerance

Constant Lambda Analysis

As in the Effects of Fuel Properties section on lean tolerance, effects of engine design parameters are examined at lambda 1.4 (stable combustion for all conditions) and lambda 1.6 (edge of combustion stability). Increased tumble had almost as big of an impact on lean combustion tolerance as EGR dilute tolerance. Figure 24 and Figure 25 show that increased tumble ratio reduced FDA by 13 CAD for Fuel 2 and by 10 CAD for Fuel 4. The CD was also reduced by 8.5-10 CAD for both Fuel 2 and Fuel 4 with higher tumble ratio. Increased ignition energy reduced FDA by approximately 1 CAD with Fuel 2, but had no noticeable impact with Fuel 4. At these stable conditions, CD was not significantly affected by ignition energy. The combustion stability was between 2% and 3% COV of IMEP for all lambda 1.4 conditions, with the exception of the low tumble conditions with Fuel 4, which had 3.1-3.2% COV of IMEP.

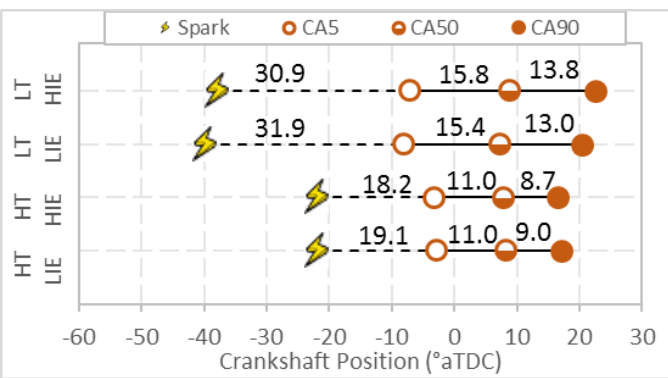

Figure 24. Effects of engine design parameters on combustion metrics for Fuel 2 at lambda 1.4 and 0% EGR.

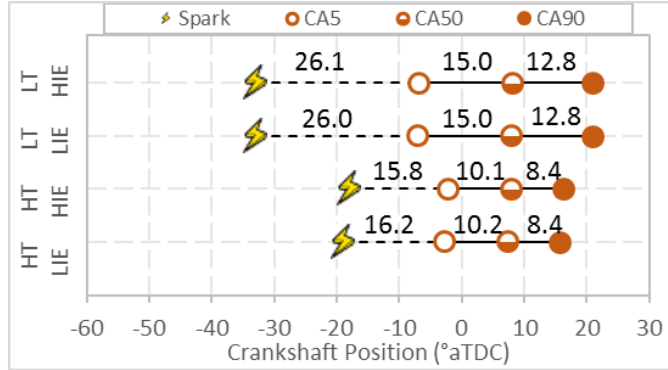

Figure 25. Effects of engine design parameters on combustion metrics for Fuel 4 at lambda 1.4 and 0% EGR.

Figure 26 and Figure 27 show the combustion characteristics of each engine design parameter test condition at lambda 1.6 and 0% EGR for Fuel 2 and Fuel 4, respectively. Lean operation at lambda 1.6 with low tumble and Fuel 2 was not obtained in this study due to high combustion instability. Increased ignition energy resulted in a reduced FDA by approximately 0.5-2.5 CAD for all tests at lambda 1.6. Increased tumble had a much larger effect, reducing FDA by approximately 16 CAD.

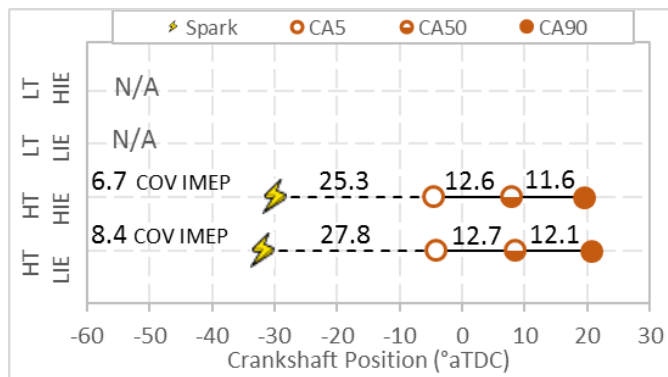

Figure 26. Effects of engine design parameters on combustion metrics for Fuel 2 at lambda 1.6 and 0% EGR.

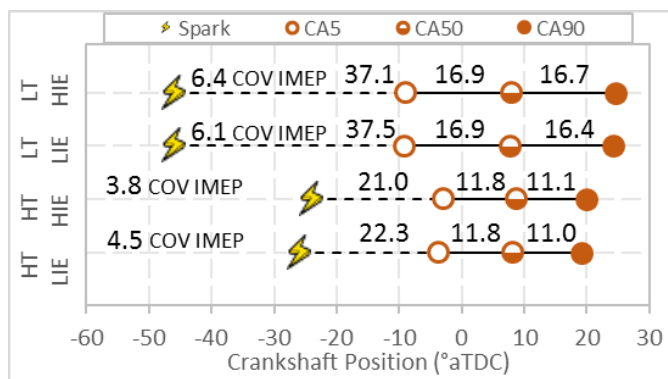

Figure 27. Effects of engine design parameters on combustion metrics for Fuel 4 at lambda 1.6 and 0% EGR.

Constant COV of IMEP Analysis

Figure 28 and Figure 29 show the lambda lean limits of Fuel 2 and Fuel 4 for combinations of high and low tumble and ignition energy at 3% COV of IMEP. Increased tumble allowed approximately 0.1 higher lambda operation, while at this level of combustion stability there was not a clear effect from increased ignition energy.

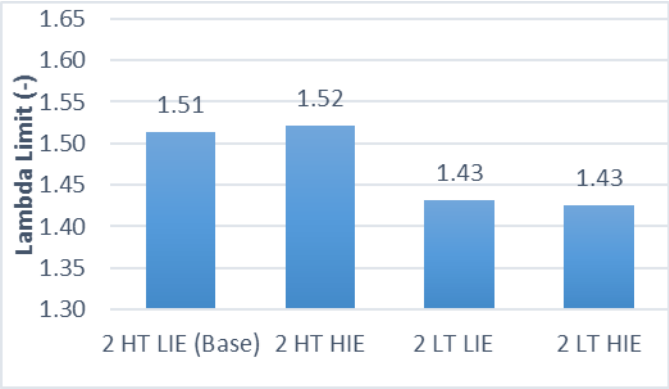

Figure 28. Comparison of engine design parameters on lean tolerance for Fuel 2 at 3% COV of IMEP.

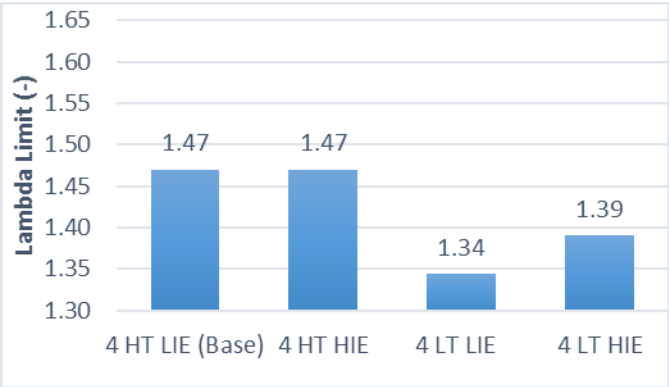

Figure 29. Comparison of engine design parameters on lean tolerance for Fuel 4 at 3% COV of IMEP.

Figure 30 and Figure 31 show the lean tolerances of Fuel 2 and Fuel 4 at 5% COV of IMEP for combinations of high and low tumble and ignition energy. In general, there was a 0.05-0.2 increase in lean tolerance by extending engine operation from 3% to 5% COV of IMEP. At 5% COV of IMEP, ignition energy did not seem to have a significant impact on lean tolerance, however increased tumble did improve lean tolerance by 0.05-0.1 lambda.

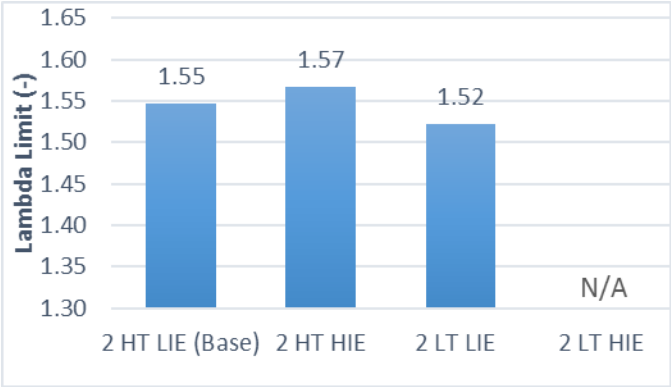

Figure 30. Comparison of engine design parameters on lean tolerance for Fuel 2 at 5% COV of IMEP.

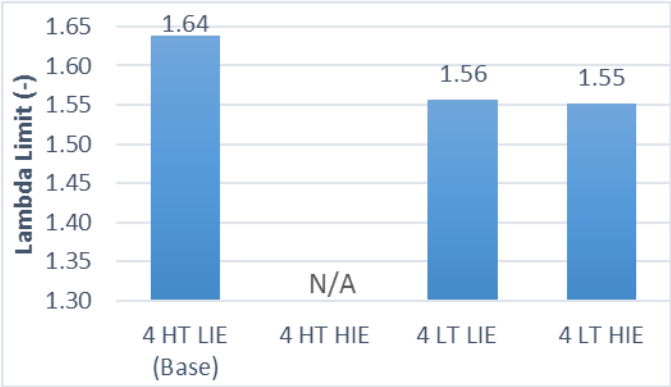

Figure 31. Comparison of engine design parameters on lean tolerance for Fuel 4 at 5% COV of IMEP.

Effect of Direct vs. Port Injection

Seeing the positive influence of increased tumble ratio on both EGR dilution and lean tolerance, direct injection tests were performed with Fuel 1 to examine the effects of increased turbulence due to direct-injected spray momentum. An EGR and lean lambda sweep was performed with Fuel 1 from 0% EGR stoichiometric conditions until the COV of IMEP for the respective sweep exceeded 5% COV of IMEP.

EGR Dilution Tolerance

Figure 32 shows the combustion stability and burn characteristics for PFI and DI test conditions with 15% and 20% EGR. At 15% EGR, the FDA of DI and PFI operation were very similar to each other, likely due to the low values of COV of IMEP and high combustion stability. At 20% EGR, both DI and PFI still had very low COV of IMEP. However, the FDA of the DI case became 1 CAD shorter, suggesting shorter flame kernel development.

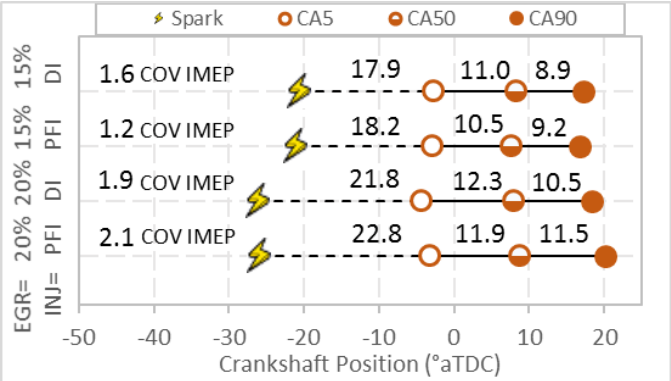

Figure 32. Effect of direct versus port fuel injection on combustion characteristics at 15% and 20% EGR.

Figure 33 shows that at both 3% and 5% COV of IMEP, DI operation obtained a 2% EGR dilution tolerance increase over PFI operation. This was similar to the generally 2% EGR dilution tolerance improvement from 0.6 to 1.5 tumble ratio.

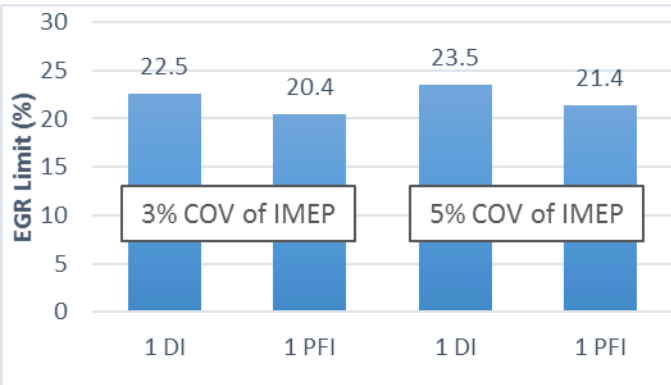

Figure 33. The EGR dilution tolerance of DI and PFI operation analyzed at constant COV of IMEP.

Lean Tolerance

Combustion characteristics and combustion stability are examined at lambda 1.4 and 1.6 in Figure 34. Under these conditions the values of COV of IMEP were higher than 2.0 and it was possible to see some advantage of DI operation. Despite the lower COV of IMEP for lambda 1.4, the FDA between DI and PFI operation was nearly the same. Perhaps this was due to quite stable combustion. At lambda 1.6, the PFI operation had a COV of IMEP just over 3% and almost 1 CAD longer FDA. However, the CD was 1 CAD shorter for PFI operation.

When comparing constant COV of IMEP (3% and 5%) between DI and PFI (Figure 35), it was observed that DI operation had a 0.1 increase in lambda lean operation extension for the same level of combustion stability, at 3% or 5% COV of IMEP. This is similar to the lean tolerance improvement gained by tumble ratio in Figure 28 and Figure 29.

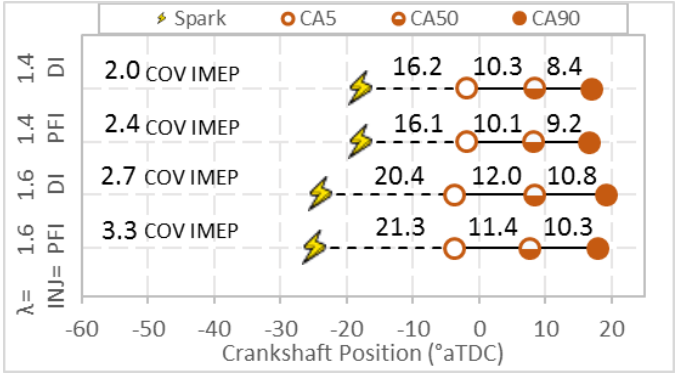

Figure 34. Effect of DI versus PFI on combustion characteristics under a lean sweep at lambda 1.4 and 1.6.

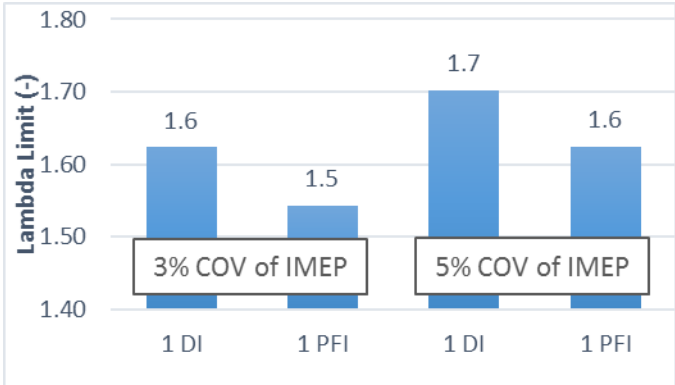

Figure 35. The lean dilution tolerance of DI and PFI operation analyzed at constant COV of IMEP.

In general, it was observed that DI offered increased EGR and lean dilution tolerance than PFI, especially apparent at the highest levels of dilution where combustion stability deteriorated. With the information available from this study, it is not possible to explain the cause for improved dilution tolerance under DI operation. One possibility is that turbulence from the injection could have assisted flame propagation in a similar way as increased tumble. However, it is not clear how much of that spray-induced turbulence main have remained from an injection timing of 300 °bTDC. Another hypothesis is that heterogeneities (rich pockets) in the mixture may have existed from the direct injection to improve EGR and lean dilute operation. However, studies of homogeneous charge compression ignition combustion have not observed appreciable mixture inhomogeneities to affect mixture reactivity when a direct injection timing was used similar to this work [10]. It is also important to point out that mixture inhomogeneities would produce less reactive lean pockets as well. So it is unclear if mixture inhomogeneities under dilute conditions would be beneficial or detrimental.

Overview

Figure 36 shows the benefits of increased EGR dilution or lean operation on engine net indicated thermal efficiency, as assisted by engine design parameters (increased tumble) and fuel properties (increased LFS and HoV). The baseline efficiency for low LFS Fuel 2 was that measured for 0% EGR and stoichiometric operation. Each level of EGR dilution or lean operation corresponds with the highest level before exceeding 3% COV of IMEP combustion stability. Initially the EGR dilution limit and lean limit was 16% EGR and lambda 1.4, respectively, achieved with low tumble. Adding high tumble ratio (1.5) allowed for EGR dilution to expand to 19% EGR

and lean operation to 1.5 lambda. Finally, the highest level of EGR dilution or leanest lambda (as well as highest indicated thermal efficiency) was achieved by increasing the fuel LFS by switching from Fuel 2 to Fuel 4.

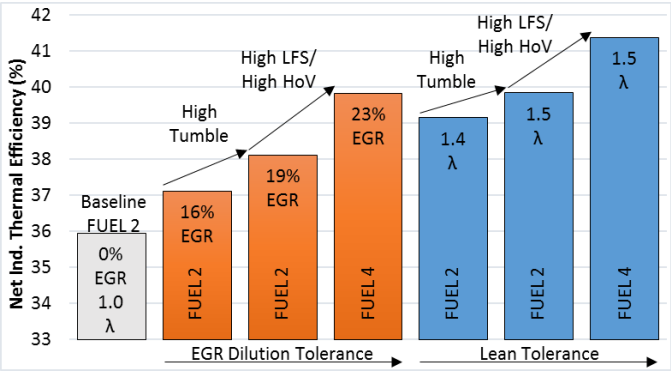

Figure 36. Overview of EGR dilution and lean operation effects on increased engine efficiency with the assistance of increased tumble and fuel laminar flame speed.

Summary

This work discusses the relative benefits of fuel laminar flame speed (LFS), engine design parameters (such as tumble and ignition energy), and direct injection instead of port fuel injection. The objective was to test whether fuel LFS has a high enough impact on dilute EGR and lean lambda tolerance. The following summarizes the findings of this research:

- Among alcohol fuel blends, increased LFS/HOV allowed for a 4-6% added EGR dilution tolerance and 0.05-0.1 lambda lean tolerance
- By increased tumble ratio or direct injection rather than port fuel injection, EGR dilution tolerance was improved by 2-3% EGR and lean tolerance by 0.1 lambda
- Increased ignition energy from 75 to 150 mJ (with constant discharge duration) improved EGR dilution tolerance by 0.5-1% EGR, but no clear effect was measured on lean operation limits
- Combining increased tumble, higher LFS/HOV fuel properties, and higher ignition energy under EGR dilute and lean operation provided a 4 percentage point improvement in ITE_{net} with EGR dilution and a 5 percentage point improvement with lean operation over baseline engine settings and fuel properties.

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Definitions/Abbreviations

°aTDC	Degrees After Top Dead Center (Firing)
°bTDC	Degrees Before Top Dead Center (Firing)
BDC	Bottom Dead Center
CA5	Crank-Angle of 5% Mass Fraction Burned
CA50	Crank-Angle of 50% Mass Fraction Burned

CA90	Crank-Angle of 90% Mass Fraction Burned
CAD	Crank-Angle Degrees
CD	Combustion Duration
CO_2	Carbon Dioxide
COV	Coefficient of Variance
DEIS	Directed Energy Ignition System
DI	Direct Injection
EGR	Exhaust Gas Recirculation
FDA	Flame Development Angle (Spark Timing to CA5)
GDI	Gasoline Direct Injection
HIE	High Ignition Energy
HT	High Tumble
$IMEP_n$	Net Indicated Mean Effective Pressure
KLSA	Knock Limited Spark Advance
LBV	Laminar Burning Velocity
LFS	Laminar Flame Speed
LIE	Low Ignition Energy
LT	Low Tumble
PFI	Port Fuel Injection
RPM	Revolutions Per Minute
TDC	Top Dead Center