

Development of a Naturally Aspirated Spark Ignition Direct-Injection Flex-Fuel Engine

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ABSTRACT

Companion empirical and analytical studies were conducted to assess the feasibility and constraints (hardware and combustion perspectives) associated with operating a Spark Ignition Direct-Injection (SIDI) engine on high ethanol and gasoline mixtures ranging from 0 to 85% by volume. Cold start, part and full-load performance aspects were explored.

Analytical experiments were performed to correlate with the empirical data using a commercially available single dimensional engine simulation code. Under WOT operating conditions it was found that the engine's simulated output was overestimated with E85 fuel which was caused by the over prediction of volumetric efficiency. It was necessary to create a sub-routine to accurately model the impingement, vaporization, and heat transfer of fuel on the piston surface. Results could only be correlated after taking fuel impingement into account.

Spray measurements were conducted including white light illumination and phase Doppler interferometry to study the differences in the spray shape, impingement, and the droplet formation regime between gasoline and E85 fuels. Efforts were taken to study the effects of temperature and back pressure on the spray. The results suggested that the spray properties were quite similar between the fuels, that the penetration lengths and droplet sizes of the fuels were comparable for like conditions, however E85 was more resistant to flash boiling effects for typical engine operating conditions.

Selected empirical part-load optimization suggested efficiency improvements of 3-6% were possible over the optimized gasoline baseline. These gains were primarily due to a reduction in heat rejection and increased Exhaust Gas Residual (EGR) tolerance. These efficiency gains mildly offset the inherent energy density deficiency of ethanol based gasoline mixtures.

Full-load empirical data suggested a potential for 13-15% increase in specific output with E85, as compared to a customer intent gasoline baseline fuel. These performance gains were enabled by the anti-knock qualities of the ethanol blends and an increase in volumetric and indicated efficiencies. It was also found that optimized full-load E85 operation was possible with injector hardware that was sized for gasoline (E00) operation. Ultimately, strategies were developed that enabled E85 operation at all operating conditions, as enabled by low smoke operation, improved knock resistance, increased charge cooling, and increased combustion stability.

Cold startability tests with representative Class 3 E85 fuel were conducted at extreme ambient conditions to assess the cold start performance of the SIDI engine. It was found that a Low Pressure Start (LPS) operating strategy enabled engine starts at ambient conditions as cold as -20C, however, High Pressure Stratified Starts (HPSS) were necessary to meet colder startability requirements. Ultimately, a HPSS strategy improved startability and reduced enrichment requirements for E85.

INTRODUCTION

Bio-ethanol fuels offer the potential for the reduction of Green House Gas (GHG) emissions due to their sustainable and renewable processes. High ethanol content fuels also offer the opportunity to significantly reduce the consumption of non-renewable fuels. When compared to other alternative spark ignited fuels, they have a relatively high energy density value and are relatively compatible with the automotive energy infrastructure. Significant usage of these fuels requires Flex Fuel Vehicles (FFV) to be a significant portion of the consumer fleet. To support bio-ethanol initiatives, the US-based automobile manufacturers have committed to making half of their North American product portfolio flex-fuel capable by 2012 [1]. Therefore it is prudent to comprehend the effects of E85 usage on future engine technologies such as SIDI.

The purpose of this study is to define, document, and resolve the technical challenges associated with implementation of SIDI powertrains and flex-fuel technology. Major risks include component wear, Positive Crankcase Ventilation (PCV) system capacity, pre-ignition, and fuel system dynamic range, all of which have implications on long term engine durability, cold startability, emissions, and full-load capability. Solutions to these challenges will enable future engine designs that include direct injection, and related technologies, to be ethanol compatible and further enable the maturity of the bio-fuel usage.

EXPERIMENTAL FACILITY

ENGINE-The engine selected for these experiments was an advanced development inline four cylinder. The engine was naturally aspirated and featured direct in-cylinder fuel injection. It was equipped with dual independent camshaft phasing (DICP) with 50 CAD of phasing authority each, which allowed up to 22500 mm² * deg/L of overlap area for the trapping of internal Exhaust Gas Residual (iEGR)

The geometric compression ratio represents a tradeoff between part-load efficiency and full-load knock-limited output. In this engine, the geometric compression ratio was 11.9:1. This compression ratio was not optimized for E85 operation since the engine was intended for a flex-fuel application with a customer intent gasoline (87 Pump Octane Number - PON) as the base fuel. Optimizing the compression ratio for E85 fuel would have unacceptably compromised full-load performance of customer intent gasoline operation.

The fuel injectors were located at an angle approximately 22 degrees from the fire deck and were positioned under the intake port of the cylinder head. The fuel injectors were a multi-hole design with 6 spray jets forming a free-oriented spray pattern positioned to avoid impingement on the intake valves and optimized for homogeneous operation. A high pressure positive displacement fuel pump was driven off of the intake camshaft with a three event cam lobe per mechanical cycle. Additional characteristics of these SIDI and base engine hardware are subsequently discussed and are presented in Table 1.

E85 Base Engine Hardware Considerations - The test engine was specifically altered from a typical gasoline configuration to address some of the issues that are currently known about E85 operation. The fuel system components were specified to comprehend the anticipated additional fuel flow, including the fuel injectors and engine driven high-pressure fuel pump. The fuel injectors were specified to include static flow rate variants of 17.5, 20.0, and 22.5 cc/sec (as measured at 10 MPa fuel pressure), to be evaluated during the engine development process. A gasoline-only version of the test engine would typically require an injector with a static flow rate of 17.5 cc/sec to meet peak-power enrichment requirements at 15 MPa of fuel pressure while

maintaining the capability to perform during idle canister purge operation at 4 MPa. While it is possible to increase the flow rate of the injectors, it is recognized that such a change would likely compromise the minimum injection quantity capability and associated fuel control. Therefore, there was a strong desire to develop methods to minimize the flow rate specification of the fuel injectors. The high-pressure fuel pump displacement was maximized within its architectural constraints. This led to a geometric displacement of 1.1 cc/engine cycle, which was more than the calculated requirement for worst case WOT operation. However, the pump specifications necessary for high-pressure cold starts were unknown at the time of engine construction but were empirically determined, as discussed in the COLD START results section.

For the vehicle tests, it was necessary to reduce the temperature specification of the spark plugs by one heat range to avoid pre-ignition. It has been documented in the literature that ethanol is more susceptible to surface based pre-ignition [2, 3]. More specifically, the spark plug metal temperatures should remain below 700 C to avoid the risk of pre-ignition.

The valves, valve seats, piston rings, and piston were also modified to resist additional component wear that is traditionally experienced with E85 operation. In particular, the upper compression ring was replaced with a proprietary plasma coated variant that is intended to increase the surface hardness. Some of the ring and bore wear is traditionally thought to occur during cold starting in fumigated engine applications where the additional enrichment requirements of E85 can lead to washing of the lubrication oil off of the cylinder bores. This can also occur during direct-injected Low-Pressure Starts (LPS). However, High fuel Pressure Stratified Starts (HPSS), as discussed later in the results section, are much less likely to contribute to such wear due to improved fuel containment and reduced enrichment. The piston geometry was also modified to include a fuel containment bowl feature to further enable HPSS, as shown in Figure 1. The valves and valve seats were specified to increase hardness and reduce galvanic potential via proprietary materials and heat treatment processes.

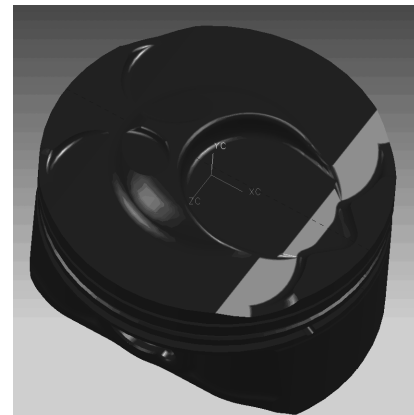


Figure 1. Piston with fuel containment bowl feature to enable high-pressure stratified starts.

Table 1. Test engine specifications.

Manufacturer	GMPT Advanced Engineering
Engine Configuration	Naturally Aspirated, Inline 4-cylinder, Direct Injection, Variable Valve Timing
Cycle	4-stroke
Advertised Displacement	2.4 liters
Valves per cylinder	4
Bore	88.0 mm
Stroke	98.0 mm
Nominal Compression Ratio	11.9
Intake Valve Peak Lift @ Park	131 Crank Deg aTDC
Intake Camshaft Duration	267 Crank Deg
Intake Cam Timing Range	50 Crank Deg
Intake Cam Park Position	Full Retard
Exhaust Valve Peak Lift @ Park	-125 Crank Deg aTDC
Exhaust Camshaft Duration	241 Crank Deg
Exhaust Camshaft Timing Range	50 Crank Deg
Exhaust Cam Park Position	Full Advance
Dynamic Injector Flow rate	17.50-22.5 cc/sec @ 10Mpa
Fuel Pump displacement	1.1 cc/engine cycle
Oil	Mineral-based 5W-30

INSTRUMENTATION – A 300 kW AC dynamometer was used to absorb the engine load. A pre-production engine controller was utilized in the typical manner with ETAS equipment to view and modify the engine control parameters. A Pierburg Model P11609 Ultra-Low Flow fuel meter was used to measure fuel flow rate to the engine at gravimetric fueling rates below 6 g/s, while a Pierburg Model P11401-308B Low –Flow fuel meter was sufficient for fueling rates above 6 g/s.

An AVL model MZZ230000 emissions bench was used to sample exhaust gas constituents, which was performed on a dry basis. All major engine out emissions, (CO₂, CO, NO, NO_x, HC) were measured. Air flow rate metrics were calculated based on the measured fuel flow rate and the mean (carbon and oxygen) air fuel ratio balance (AFR) according to exhaust gas analysis equations developed by Stivender [4]. However, direct comparisons of air flow for the test fuels were performed using the engine Mass Air Flow (MAF) sensor due to the differences in HC response from the Flame Ionization Detector (FID).

It can be found in the reference literature that unburned hydroxyl bound hydrocarbons as measured with a FID will lead to error resulting in a reduced sensitivity [5]. Absolute error associated with detecting these bounded hydrocarbons are dependant on the method of sampling (wet vs. dry), FID burner design, and the test fuel. The emissions data presented in this study are not corrected for these effects. Correction of the emissions based airflow analysis with ethanol-based fuels may be the topic of a future publication.

Engine-out smoke emissions were measured with an AVL 415S smoke meter and are reported herein with the units of Filter Smoke Number (FSN) [6, 7].

Data Acquisition – An ACAP 6.0B combustion data acquisition unit was used to gather cycle-resolved pressure based statistics and was utilized in conjunction with a cycle averaged data acquisition system [8]. Cycle averaged data were cross-referenced with the cycle-resolved pressure data using a GMPT proprietary software toolset. All calculations used in the analysis adhere to GMPT guidelines and state of the art practices. In-cylinder pressure data for all four cylinders were sampled with a resolution of 1 CAD utilizing AVL GH12D miniature pressure transducers, in conjunction with AVL PH01 flame arrestors, mounted in 304-925 and 304-1063 Champion spark plug transducer adapters. The appropriate heat range was used during testing to avoid spark plug fouling and pre-ignition. The spark plug transducer mount passages were modified to give high frequency resolution required for spark knock assessment.

COLD START VEHICLE – A model year 2007 Chevrolet Malibu was built specifically with the aforementioned engine to support vehicle-based cold startability and emissions testing. The vehicle was equipped with ETAS instrumentation for calibration access and data collection. Cold chambers were used to soak the vehicle and replicate extreme cold ambient conditions.

TEST FUELS - The fuels used for the engine tests of this study were selected based on their knock limited properties and availability in the North American market. They consisted of a representative customer-intent “regular” gasoline with 87 PON (CP Baseline), a high octane gasoline variant (B25), and an E85 fuel (ED85). Pertinent specifications of the fuels are listed in Table 2. For comparison, all mass specific fuel consumption values were normalized to a Lower Heating Value (LHV) standard of 42.94 MJ/kg.

Table 2. Test fuel specifications. *Estimates based on known base components [9, 10].

	H/C	O/C	Stoich. mass air/fuel ratio	RON	MON	LHV	Molecular Weight	Specific Gravity	Heat of Vapor.
Fuel Type	[-]	[-]				[MJ/kg]	[kg/kmol]		[kJ/kg]
Chevron Philips Baseline	2.002	0.000	14.79	91.5	85.6	43.60	14.03	0.712	58.9
Chevron Philips B-25	1.908	0.000	14.65	104.3	96.5	43.25	13.93	0.736	56.6
ED85	2.703	0.382	9.85	107.7	89.5	29.62	20.85	0.785	129.3
E70*	2.721	0.360	10.11	105.3	89.0	31.46	20.52	0.773	117.6
E50*	2.526	0.263	11.06	102.2	88.3	34.75	18.76	0.758	101.4
E25*	2.271	0.135	12.61	98.3	87.4	39.06	16.45	0.739	80.3



Documents

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The three intermediate ethanol content fuels (E25, E50, and E70) between E00 and E85 were used to study the effects that might be encountered during partial fuel tank re-fill conditions. These test fuels were mixed on a molar basis from known base component fuels. Fuel properties of the commercially available fuels were analyzed empirically, while the properties of the intermediate blends of ethanol and gasoline (E25, E50, and E70) were estimated based the molar fraction of the characterized base components [9, 10].

In addition to the representative Class 1 E85 fuel (ED85) and other intermediate ethanol blends listed in Table 2, a Class 3 E85 fuel was used in the cold start portions of this study as discussed in the COLD START RESULTS section.

In the spray experiments, Indolene, a commercially available controlled multi-component gasoline, was used as the representative gasoline reference fuel. The relevant properties of this fuel are discussed in the SPRAY RESULTS section.

RESULTS

SIMULATION RESULTS - Simulation of the subject engine was performed using GT Power® software by Gamma Technologies, Inc. The simulation was conducted to understand the fueling and engine management system requirements of E85 compared to the customer intent gasoline. The parameters of interest for component sizing and engine management strategies were WOT performance, volumetric efficiency, and exhaust gas temperatures, all of which have a significant effect on the control strategy employed in a flex-fuel application.

The fuel properties used for the baseline gasoline model were extracted from the Indolene model within GT Power with modifications to match the parameters shown in Table 2 for CP Baseline. The fuel properties for E85 were represented by mixing two base fuels from the GT Power® library, specifically Indolene and ethanol. This combined fuel required further adjustments to match the composite LHV, heat of vaporization, specific gravity, as well as carbon, hydrogen, and oxygen content as shown in Table 2 for ED85.

The Weibe function based non-dimensional combustion model used by GT Power does not have any predictive capability for heat release rates nor knock limitations. Therefore, the baseline gasoline model was correlated to previously known empirical measurements of combustion duration, as characterized the 10-90% Mass Fraction Burn (MFB), and the location of 50% of the charge MFB (CA50). This correlation to empirical data considered knock limitations that were encountered with the test engine. Since the combustion model was not predictive, any change in these parameters associated with E85 had to be determined independently.

The knock limitations were assumed to be absent for E85 engine operation because of the relatively high octane rating, heat of vaporization, and a gasoline-based compression ratio of the test engine (11.9:1). Therefore the CA50 parameters were assumed to be near 8.5 deg aTDC, which is generally considered to be near-optimum combustion phasing. Combustion durations and burn rates were determined from empirical correlations to laminar flame speed measurements available in the literature [11], as well as a priori combustion information from WOT operation of a port fuel injected flex-fuel engine. Both sources suggested that the burn rates and associated combustion durations were similar between the two fuels at WOT operating conditions, within the accuracy of cylinder pressure based measurements. However, this empirical information also suggested that E85 burns at a faster rate for lower pressure and temperature conditions associated with part-load engine operation.

The WOT simulation results shown in Figure 2 indicate a significant improvement in performance associated with E85 fueled operation, both at stoichiometric and enriched equivalence ratios (EQR=1.00 and 1.15). Some of this improvement is associated with the assumption that the combustion phasing can be fully optimized for E85. Specifically, the relative significance of this combustion phasing effect can be seen in Figure 2 by comparing the baseline gasoline fuel simulation to the academic assessment of gasoline with Minimum Spark for Best Torque (MBT) combustion phasing, both of which were run at enriched conditions near Leanest fuel for Best Torque (LBT), which was a fuel/air equivalence ratio (EQR) of 1.15. Yet the stoichiometric conditions with E85 fuel yield even higher engine output than MBT gasoline operation. Much of this difference is attributable to an increase in volumetric efficiency of the engine, also shown in Figure 2. Additionally, the engine output performance can be further improved by operating the engine on E85 fuel at an enriched condition (EQR=1.15). This is expected since this EQR represents a near peak flame temperature, or LBT, condition for both fuels. The LBT condition of the fuels is discussed in more detail in the FULL-LOAD test results section.

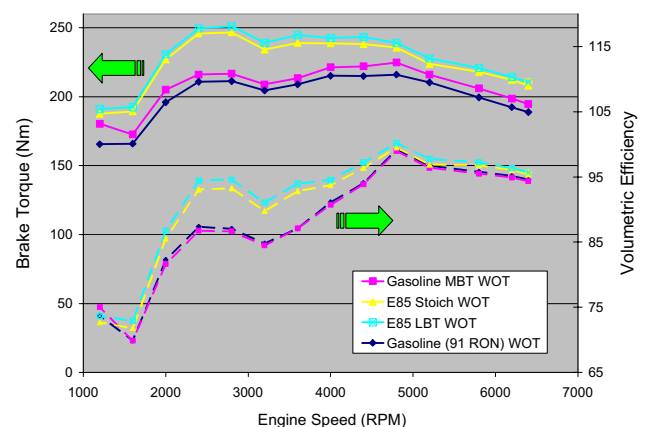


Figure 2. Simulated engine performance for WOT gasoline and E85 operation.

This increase in volumetric efficiency with E85 operation is generally attributable to the increased heat of vaporization of the fuel, as shown in Table 2. The simulation software assumes that all of the heat associated with vaporization of the liquid fuel is extracted from the air charge leading to a significant cooling effect, thus increasing the charge density and volumetric efficiency. This evaporation model assumption does not account for the practical effects of fuel impingement on the metal surfaces of the cylinder [12], which leads to reduced charge cooling and less overall heat rejection of the engine. This effect is largely ignored when simulating gasoline operation with little significance due to the relatively low heat of vaporization. However, not considering impingement effects with E85 can lead to exaggerated performance due to the over-prediction of charge cooling. This observation becomes obvious when comparing the simulation to the experimental results and the effect of Start of Injection (SOI) on volumetric efficiency, as discussed later in the FULL-LOAD results section. This realization led to the development and implementation of an impingement sub-model for future engine simulation. The content and results of this sub-model are not included in this subject matter, but may be documented in future publications.

The Exhaust Gas Temperatures (EGT) results of the simulations were evaluated to determine the extent to which exhaust component protection enrichment may be effected by E85 operation. All of the fuels were simulated at a constant equivalence ratio with the exception of 91 RON gasoline, where the Fuel/Air ratio was enriched above 4800 RPM to maintain an EGT below 850 C. The EGTs for each simulation are plotted in Figure 3, where it can be observed that the exhaust of E85 fueled operation is significantly cooler than gasoline operation for equivalent enrichment conditions. The EGT of E85 is approximately 35 C cooler than gasoline operation for equivalent enrichment. Some portion of this exhaust cooling is attributable to the improved combustion phasing, thus leading to improved engine efficiency and less exhaust waste heat. The extent of the combustion phasing effect can be observed by comparing the baseline gasoline operation to the academic run of gasoline operation with MBT combustion phasing at speeds below 4800 RPM where no component enrichment was required. However, further EGT reduction is accomplished by using E85 at equivalently enriched conditions (EQR=1.15). This effect is presumably due to the lower flame temperature of ethanol combustion compared to gasoline, and the relative differences in enthalpies of vaporization [13]. In the case of E85 operation, it may be possible to avoid component protection enrichment at high speeds due to the inherently low EGT and reduce the injection requirements. The analysis suggests a reduction of 0.12 EQR may be possible with E85 optimum performance operation.

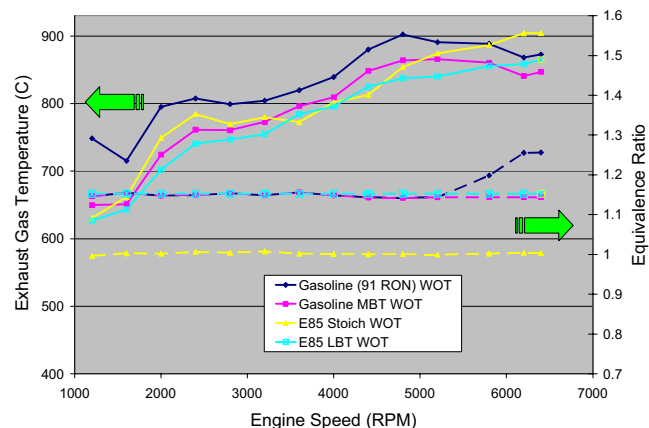
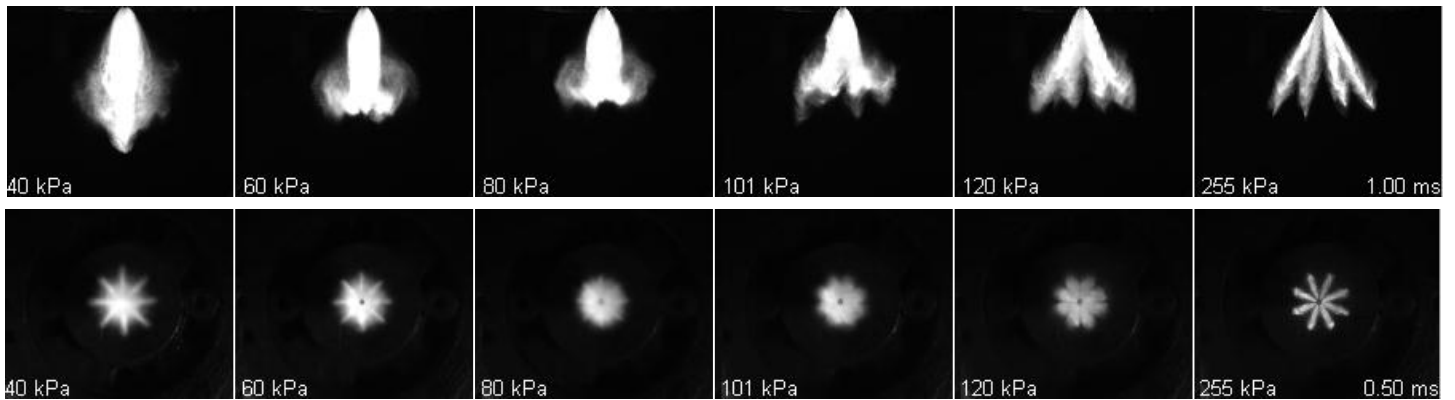


Figure 3. Simulated engine exhaust gas temperature and equivalence ratio for WOT gasoline and E85 operation.

FUEL SPRAY COMPARISON RESULTS - Spray measurements were performed prior to engine testing to assess the influence of fuel composition on the spray characteristics. An 8-hole Valve Covered Orifice (VCO) injector, different from the injector specified for the engine experiments (6-hole VCO), was utilized for the spray measurements. Although the injectors were different, the liquid fuel break-up regime is similar for the two injectors and the findings are expected to apply directly to the engine injector. White light volume illumination was used to characterize the spray geometry and phase Doppler interferometry measurements were performed to assess particle size characteristics. Measurements were performed at various fuel temperatures and ambient pressures.

The white light volume illumination images shown in Figure 4 compare spray images of E85 and gasoline, specifically Indolene. The images represent tests conducted at a fixed fuel temperature and pressure that is representative of naturally aspirated WOT engine operation (80°C and 11 MPa) over a range of back-pressures that are representative of part-load cylinder conditions (low pressure) and late injection that can occur during the compression stroke (high pressure). It can be observed from the images that the spray geometry of E85 is less affected by ambient pressure than that of gasoline. In both cases, the change in spray geometry at reduced ambient pressure is the result of flash boiling. In the case of gasoline, this behavior is more prevalent due to the more significant amount of light-end fuel components. Thus the effects of fuel composition on the spray shape are related to its distillation properties. In the case of gasoline, the distillation curve represents a relatively continuous transition of boiling temperatures due to the multi-component nature of the fuel, as shown in Figure 5. However, E85 is made up largely of (~82 % by mass) single boiling point ethanol, leading to a significant vaporized fraction at 78°C, also shown in Figure 5.

Indolene



E85

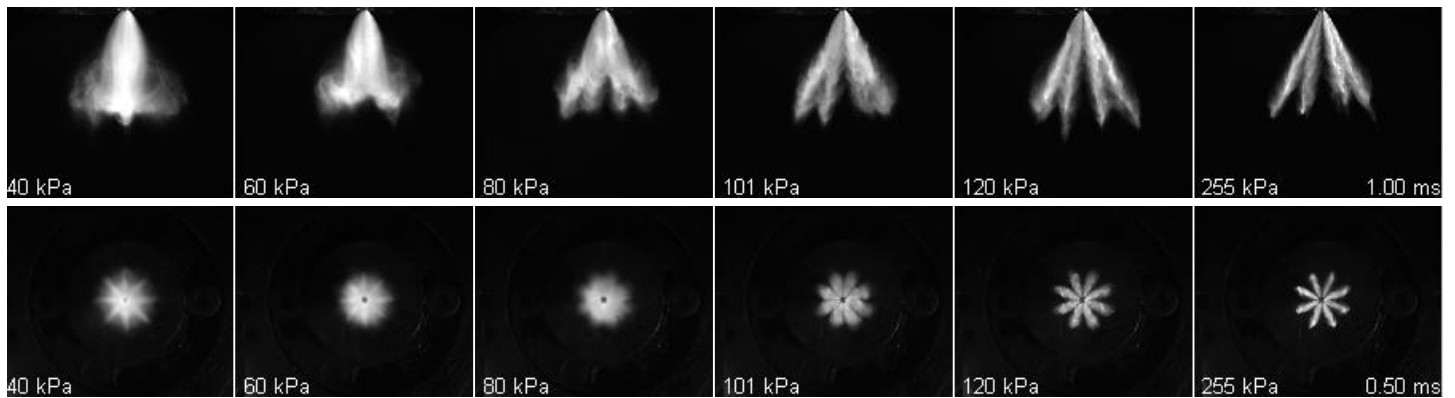


Figure 4. White light volume illumination comparisons of Indolene and E85 at 80°C fuel temperature, 11 MPa fuel rail pressure, and various ambient pressures (upper-side view; lower-end view).

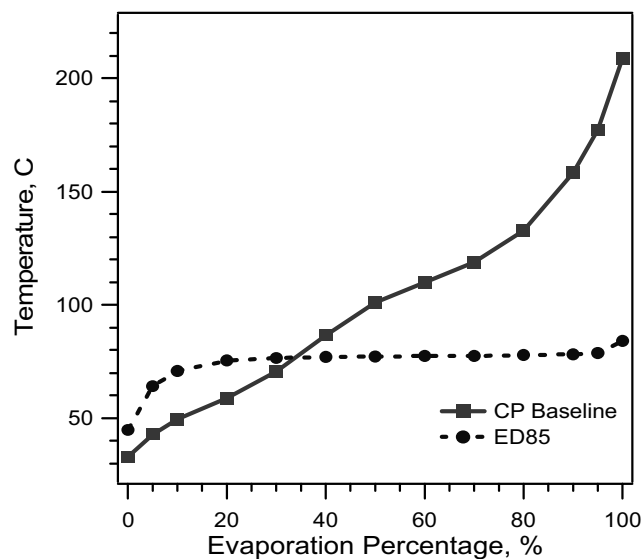


Figure 5. Distillation curves for CP Baseline and ED85.

It can be concluded from the spray images that E85 has more consistent behavior over typical engine operating conditions, as shown in Appendix A.1. However, E85 does demonstrate the effects of flash-boiling at fuel temperatures elevated beyond 80°C. This is further illustrated by the change in spray shape as viewed on end where, when exposed to an ambient pressures of 60 kPa and below, the individual plumes expand at such a

rate that adjacent plumes impinge against one another and form plumes between plumes, as also shown in Figure 4 and Appendix A.1. Yet the two fuels show largely the same characteristics when injected into relatively high ambient pressures. It can be presumed that the two fuels will have similar behavior in such conditions of late injection when fuel stratification is desired for some operating strategies.

Phase Doppler interferometry measurements were also conducted to assess the effect of fuel composition on atomization. In particular, the Sauter Mean Diameter (SMD) was measured to quantify the fuel droplet size. The measurements were made on the centerline of one fuel plume of the multi-hole spray, as shown in Figure 6. The injection duration was set such that 10 mg of fuel was delivered per injection event.

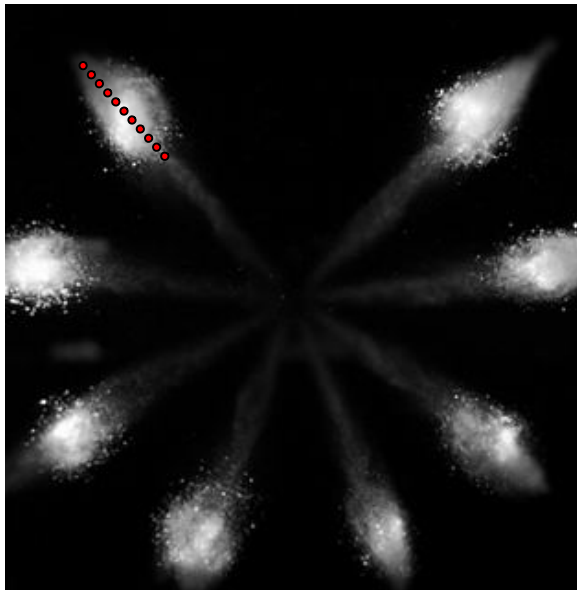


Figure 6. Phase Doppler interferometry measurement plane definition.

The SMD was measured at two fuel temperatures and three ambient pressures, which are shown in Appendix A.2. For convenience, shown in Figure 7 is a comparison of SMD as a function of time and space for the two fuels at 90°C and 60 kPa ambient pressure when injected with 20 MPa of fuel pressure. Droplets contained within the contour lines indicate velocity magnitudes greater than 10 m/s. These results indicate that the differences between the two fuels are insignificant for conditions of typical part-load stoichiometric engine operation with a slight trend of larger droplets for E85 sprays. In Appendix A.2, it can be observed that the droplet size differences become more apparent at higher charge pressures. However, these differences are expected to have a negligible impact on the combustion system characteristics.

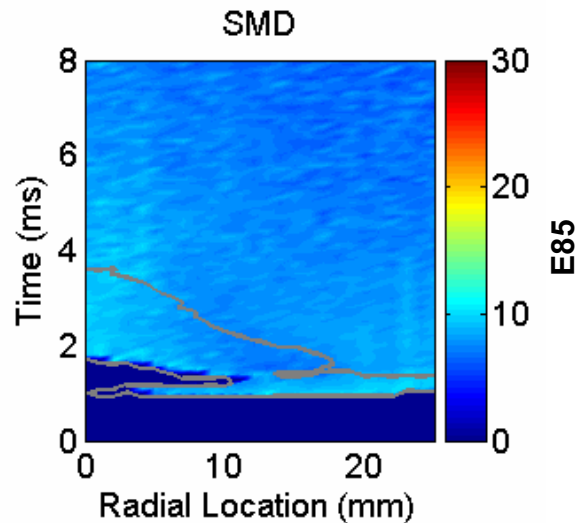
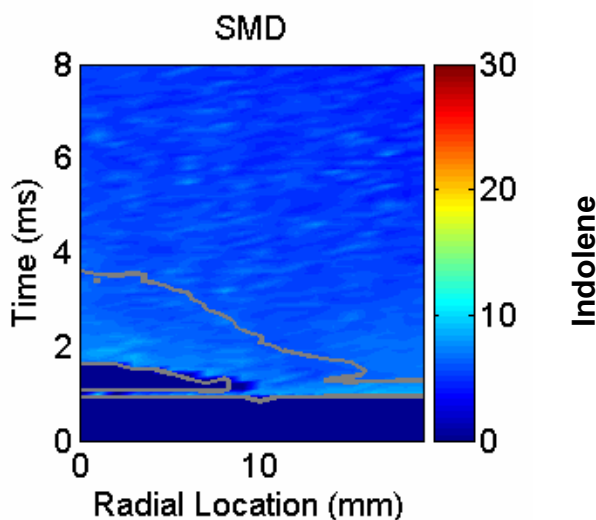


Figure 7. Sauter Mean Diameter (SMD) measurements vs. time and radial location at 90°C fuel temperature and 60 kPa ambient pressure.

PART-LOAD ENGINE RESULTS

Efficiency Improvements -Three speed/load points were used to characterize the part-load behavior of E85 fuel as compared to the customer intent gasoline baseline.

- 1300 RPM 330 kPa NMEP
- 2000 RPM 200 kPa BMEP
- 2200 RPM 455 kPa NMEP

These speed and load test points were selected for partially heuristic reasons. While they indicate trends associated with part-load vehicle operation, they are not completely representative of US EPA test cycle engine operation.

Systematic DICP timing sweeps were conducted to assess the tradeoff of trapped internal gas residual, effective compression ratio, and effective expansion ratio where minimized specific fuel consumption was desired while observing combustion stability limitations. In the study combustion stability was limited to values at or below 3% Coefficient Of Variation (COV) of Indicated Mean Effective Pressure (IMEP), however during data collection the DICP sweeps were continued until misfire occurred to insure that the limits of acceptable combustion variation were comprehended.

Internal Exhaust Gas Residual (iEGR) is the major part-load efficiency benefit mechanism of VVT powertrains and does so by reducing heat rejection and displacing inducted fresh air, which ultimately reduces pumping losses. Due to the difficulty of making direct measurements of iEGR, it is customary to use specific NO_x emissions as a surrogate. However, this is not accurate when comparing the effects of high ethanol content fuels to gasoline due to the inherent differences in adiabatic flame temperature and enthalpy of vaporization [13]. Therefore, the GMPT Residual Fraction (GMPTRF) estimate model, which is a modification of a popular simple residual fraction model that considers the overlapping valve curtain areas and

the pressure differential across the engine, was used as an indicator of the relative amount of iEGR [14]. Figure 8 illustrates the correlation of GMPTRF to a DICP sweep in terms of peak valve lift timing at 1300 RPM 330 kPa Net Mean Effective Pressure (NMEP) Load.

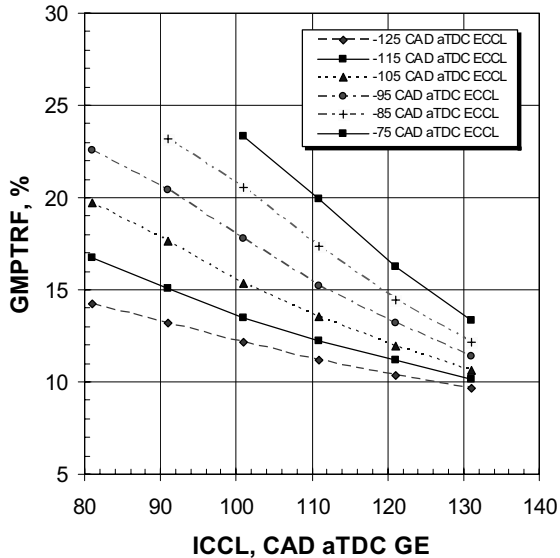


Figure 8. Relationship of GMPTRF as a function of peak valve lift timing.

At all of the engine operating conditions referenced, the combustion stability decreased as GMPTRF was increased, as shown for 1300 RPM in Figure 9-A. One can observe that for a fixed GMPTRF, combustion stability with E85 fuel was better than the baseline fuel. This correlates well with faster 0-10% MFB duration as indicated by Figure 9-B. This faster flame initialization period is due to the faster laminar flame speed enabled by the chemistry of E85 fuels. Note that at comparable GMPTRF the bulk burn durations, as denoted by the 10-75% MFB (Figure 9-C), were comparable, suggesting that the laminar flame speeds of the fuels had little effect on the main burn duration.

At a comparable GMPTRF, the indicated engine efficiency with E85 was improved, as shown in Figure 9-E, due in part to the decreased flame initialization period. As a result, it was necessary to throttle the engine more aggressively in order to maintain the target load. This is reflected in the Pumping Mean Effective Pressure (PMEP) measurements shown in Figure 9-D. For a given indicated efficiency, this effect reduced the net (Figure 9-F) and related brake efficiencies.

Since the engine had a lower COV of IMEP at a given GMPTRF with E85, the amount of iEGR could be increased before reaching the combustion stability limit. It can be seen in Figure 9-A that the engine tolerated an additional 3% of GMPTRF, or approximately 10 CAD of valve overlap duration, before the stability limit was reached. This had several implications. First, it can be observed in Figure 9-B that the flame initialization periods are similar between the two fuels when considering the same combustion stability limit. This suggests that total time losses are similar for both fuels.

Second, the pumping losses (PMEP) are also similar for both fuels when compared at the same stability limit. Since the combustion phasing for both fuels was maintained at MBT, the indicated efficiencies should be comparable. However, when one looks at the engine efficiency on an indicated basis, as illustrated in Figure 9-E there is a significant difference between the fuels, approximately 5% when evaluated at the same combustion stability limit. Since the pressure measurements are a direct indicator of the apparent heat release, the indicated efficiency benefit of E85 operation must be due to implicit phenomena.

A thermodynamic analysis using a Woschni heat transfer model supported the production of Figure 10 for two comparable operating conditions for gasoline and E85 fuels and includes cumulative heat released and cumulative heat rejection statistics [15]. When accounting for the stoichiometric energy content differences of the two fuels as operated at equivalent Net load, this analysis suggested that the differences in the total heat energy released was 42.6 J. This difference in heat energy released is equal to the reduced heat energy loss for E85, which is thought to be due to the increased enthalpy of vaporization and reduced flame temperatures for high ethanol content fuels.

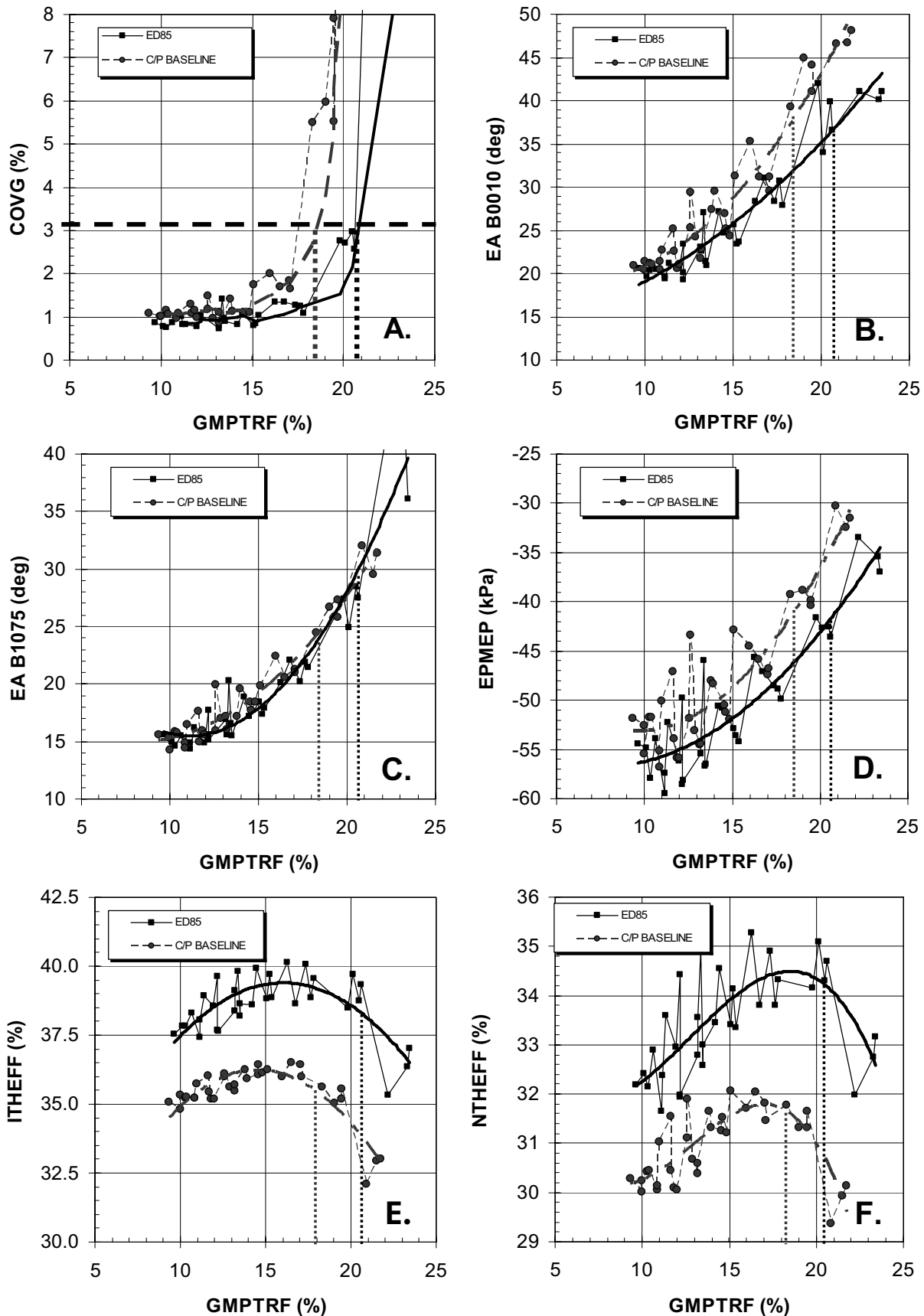


Figure 9. Response parameters of interest for DICP sweeps at 1300 RPM 330 kPa NMEP with CP Baseline and ED85. **A)** Engine Global COV of IMEP **B)** Engine Average 0-10% MFB Duration **C)** Engine Average 10-75% MFB Duration **D)** Engine Average PMEP **E)** Indicated Thermal Efficiency **F)** Net Thermal Efficiency