SAE International

Engine Efficiency Improvements Enabled by Ethanol Fuel Blends in a GDi VVA Flex Fuel Engine

2011-01-0900
Published
04/12/2011

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doi:10.4271/2011-01-0900

ABSTRACT

Advances in engine technology including Gasoline Direct injection (GDi), Dual Independent Cam Phasing (DICP), advanced valvetrain and boosting have allowed the simultaneous reductions of fuel consumption and emissions with increased engine power density. The utilization of fuels containing ethanol provides additional improvements in power density and potential for lower emissions due to the high octane rating and evaporative cooling of ethanol in the fuel. In this paper results are presented from a flexible fuel engine capable of operating with blends from E0-E85. The increased geometric compression ratio, (from 9.2 to 11.85) can be reduced to a lower effective compression ratio using advanced valvetrain operating on an Early Intake Valve Closing (EIVC) or Late Intake Valve Closing (LIVC) strategy. DICP with a high authority intake phaser is used to enable compression ratio management. The advanced valvetrain also provides significantly reduced throttling losses by efficient control of intake air and residuals. Increased ethanol blends provide improvements in power density due to knock resistance. Knock resistance also provides a significant potential for reduced NOx since higher dilution without knock is enabled at moderate loads typical of normal driving. E85 also shows significant advantages for particulate emissions that enable broader authority in selection of optimal injection timings for improving efficiency. An increase in the ethanol content improves low end torque providing an addition opportunity for improved fuel economy by using down-speeding for more efficient vehicle operation

INTRODUCTION

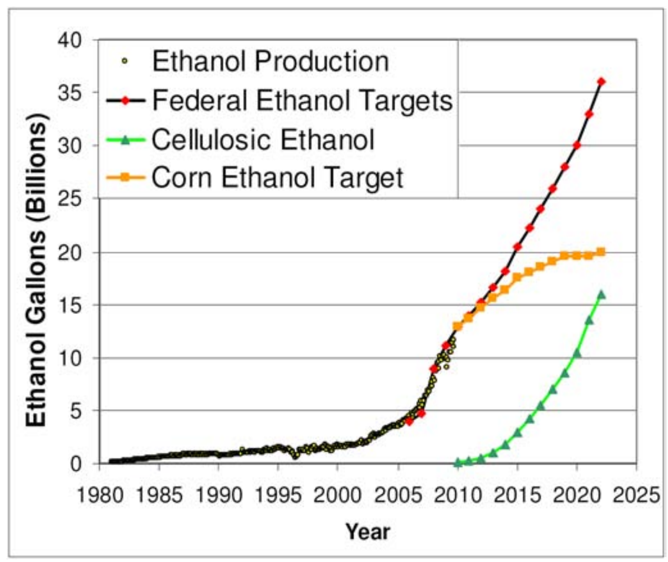

Figure 1. Ethanol production and targets as outline by the EISA.

The production of ethanol for fuel has risen dramatically in the last decade since to a level of 12 Billion gallons/year in 2009[1], as shown in figure 1. The passage of the Energy Independence and Security Act (EISA) of 2007 [2] has set a target for ethanol production of 36 billion gallons by 2022. The United States approves the use of blends up to E10, for use in all vehicles, E15 for Model year 2007 and newer vehicles and E85 for use in flex fuel vehicles. The increased production of ethanol is rapidly approaching the point where even if all the gasoline is blended to E10 the goals of the EISA can not be met. Current acceptance of E85 is hindered due to its reduced energy density relative to gasoline, which results in reduced MPG and vehicle range. The energy content is about 28% lower on a volumetric basis and 32%

lower on a mass basis. Despite its lower energy density ethanol blends offer significant benefits with respect to increased power density due to its high octane rating and latent heat of vaporization. E85 can operate more efficiently and produces lower warmed up exhaust emissions. Difficulties include cold starting, primarily at low ambient temperatures due to poor vaporization. Corrosion and compatibility with materials are also critical such that flex fuel vehicles must be specifically designed and calibrated to operate on E85 and intermediate blends. These designed modifications are well outlined in [3,4,5] and include issues of pump flow capability, injector flow capability and dynamic range, injector deposits, valve seat, ring, liner and piston durability. Ethanol usage in other markets is also driving development including E85 in Sweden and E100 in Brazil [5].

The investigation of ethanol as a fuel to leverage its ability to increase power density due to its high octane and latent heat of vaporization has been studied in various platforms. Evaluation on a CFR engine was able increase the knock limited Compression Ratio (CR) to 16.5:1 at 900 RPM with E85 [6]. Evaluation with a port injected engine with the CR increased to 13 showed demonstrated knock free operation above E50 blends but low end torque suffered with conventional gasoline and E10-20 blends[7]. Researchers investigated the effect of valvetrain modifications for improving cold start ability by using Late Intake Valve Opening (LIVO) with closing near Bottom Dead Center (BDC) to increase mixing and maximize compression heating [7,8]. These evaluations were able to reduce cold start temperatures to −35 C with a 10.5 CR port fueled engine. Ethanol blends up to E85 were evaluated in a Direct Injection (DI) Engine with Variable Valve Actuation (VVA) with compression ratios up to 12.87. E50 and E85 blends were not knock limited [9]. The electro-hydraulic VVA was used to evaluate EIVC and LIVC strategies to reduce the compression ratio when operating on gasoline or low ethanol blends. This reduced the knocking tendency but with reduced output due to lower displacement. A naturally aspirated DI Flex Fuel application with CR increased to 11.9 showed improvement in specific power, even over knock free gasoline reaching 13 Bar BMEP on E85 at 4000 RPM[3].

The use of ethanol blends with boosted engines provides significant opportunity for increased power density and efficiency. High ethanol blends such as E85 can provide knock free operation at high loads. Cooler exhaust temperatures also reduce the need for Power Enrichment (PE) to limit turbine inlet temperatures. Work to optimize for E85 and Flex fuel operation cover a spectrum of technologies including calibration optimization and algorithm development of a boosted MPFI [5] and of a boosted DI engine with DICP [10]. Research results [4,10] conclude that power density using E85 could be further increased if the engine was designed to allow peak cylinder pressures of 140-150 Bar. An evaluation to determine the relative benefits of the increased RON and the latent heat of evaporation has been documented showing the RON provides 70% of the knock resistance at 1000 RPM but only 40% at 3000 RPM [11].

Technical approaches to address the inherent power density discrepancy between E85 and gasoline operation have been investigated to develop strategies to reduce the knocking tendency with gasoline. This enables some of the power density and efficiency losses from spark retard and PE to be reduced. These include cooled EGR, [12,13] cooled EGR with hydrogen addition to improve dilution tolerance [14] and the use of an Atkinson cycle (LIVC) to reduce the effective compression ratio to limit knock [15]. Lean boosted systems have also been evaluated with E85 to improve efficiency [15]. The use of a duel fuel system with gasoline Port Fuel Injection (PFI) and DI E85 has demonstrated the ability to leverage E85 for high load efficient operation while providing increased efficiency with gasoline at low loads [4]. The 12:1 CR boosted engine uses the required quantity of E85 to limit knock as load increases effectively providing the required ethanol “blend”. This application did not require exclusive E85 operation since peak cylinder pressures limits required spark retard which reduced the octane requirement. They concluded the development of engines permitting increased peak cylinder temperatures would enable increased power densities with E85 without the need for spark retard or PE.

CONCEPT DESCRIPTION

To leverage the high octane potential of ethanol a turbocharged GDi Engine with DICP was chosen. Engine simulation was carried out previously [16] to identify strategies for improved engine efficiency and allow operation as a flex fuel vehicle with gasoline to E85 blends. To improve engine efficiency the CR was increased from 9.2 to 11.85 by changing out the pistons. The valvetrain was also modified to accept a 2 step VVA system that employed both EIVC and LIVC strategies to control the effective displacement and compression ratio of the engine. The phasing authority of the intake phaser was increased to 80 crank angle degrees (cad) based on results of the engine simulation. Table 1 outlines the base engine specifications and calls out the modifications that took place.

Table 1. Engine Specifications

Base Engine Specification	GDi Turbocharged Engine with DICP
Displacement	2.0 L
Bore	86 mm
Stroke	86 mm
Compression Ratio	11.85 increased from 9.2
Intake cam phaser	80 cad increased from 50 cad
Exhaust cam phaser	50 cad
Intake valve lift / Duration (high)	10.3 mm / 277 cad @0.15mm lift
Intake valve lift / Duration (low)	5.6 mm / 132 cad @0.15mm lift
Exhaust valve lift / Duration	10.3 mm / 230 cad @0.15mm lift
Intake Valve (High) Opening / Closing	Opening -41 to 39 cad atdc Closing 136 to 56 cad abdc
Intake Valve (Low) Opening / Closing	Opening -42 to 38 cad atdc Closing 6 to 86 cad bbdc
Exhaust Valve Opening / Closing	Opening 10 to 60 cad bbdc Closing -10 to 40 cad atdc
Fuel Injector	6 hole, 16.4 g/s @10 MPa
Turbocharger	Borg Warner K04-025 *Wastegate and Compressor bypass fully open for testing*
Fuel injection pressure	10 MPa (100 Bar) for all testing

The CR of the engine was limited by valve clearance constraints to allow valve phasing. A picture of the modified piston is shown in figure 2. The design includes a small feature to aid cold starting with late stratified injection timing. The feature is smaller than desirable for stratified operation but was a compromise for increased compression ratio.

Figure 2. Modified geometry of 11.85 CR piston.

The modifications to the valvetrain include a Delphi 2-step mechanism, 3 lobe camshaft and associated oil control valves and passages. The 2-Step Roller Finger Follower (RFF), Figure 3, is designed for a type II valvetrain. The 2-step RFF operates on the trilobe camshaft, Figure 4, by operating on the outer rollers in low lift mode. A central slider integrates a loss motion spring to retain contact on the center high lift profile. The high lift is activated by increasing oil pressure through the Hydraulic Lash Adjuster (HLA) to engage a locking pin, via an oil control valve to the HLA passages. Development of this system is documented in reference [17].

Figure 3. Delphi 2-step Roller Finger follower

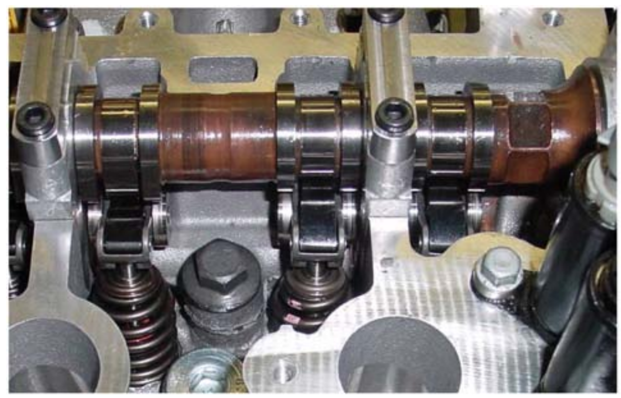

Figure 4. Tri-lobe camshaft installed with 2-step RFF

The use of EIVC has been documented [9, 18,19,20,21] to improve fuel efficiency at low load by a reduction of pumping losses. Typical challenges involve reduced in-cylinder charge motion from reduced valve lift and more time for turbulence dissipation. An alternative is to use LIVC [9, 22] which has better charge motion but is slightly less efficient than EIVC at low loads. The cam profiles selected for this engine allow both of these strategies to be utilized to provide a variable displacement system controlled by valve closing time. The relative cam positions, valve lift and phasing authority is shown in figure 5. The dashed lines indicate the cam positions in the park condition at cold start, since oil pressure is required to provide valve phasing. The engine starts in low lift at the condition of the dashed low lift cam and dashed exhaust cam. This start configuration provides low overlap for minimal internal EGR and Late Intake Valve Opening (LIVO) which has been documented to aid cold starting due to high intake velocities and high effective compression at low speeds, [8].

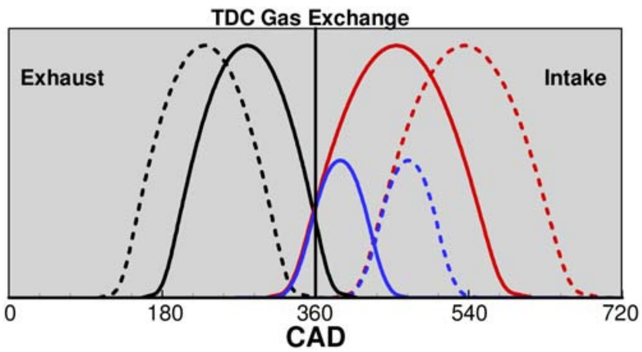

Figure 5. Cam profiles and cam phasing authority.

INSTRUMENTATION

The modified engine was installed on an engine dynamometer. Table 2 lists instrumentation and sampling locations on the engine.

Table 2. Instrumentation Description

In-cylinder Pressure Transducers	Kistler 6117BCD36
Shaft Encoder	Kistler 2614A @ 0.5 cad
Combustion Analysis	A and D Redline II
UEGO Sensor	Post Turbo before converter
Emission analyzer	AVL GEM 110 HC, CO, NOx, O2, CO2
Emission Sample tap	Exhaust plenum prior to TC
Particulate	AVL 415S
Particulate sample	Post converter
Fuel Measurement	AVL 735
Fuel Conditioner	AVL 753 (20 C)
Intake Temperature TC	Inlet port Cyl. 1-4
Exhaust Temperature TC	Exhaust runners Cyl. 1-4, Turbine inlet 1+3,2+4, Converter inlet, 25 mm bed
Manifold Pressure	Intake manifold
Exhaust Backpressure	Turbocharger inlet

FUELS

A variety of fuels were evaluated during testing, and with the exception of the section evaluating fuel blends, the test fuels consisted of 91 RON E0 and a commercial grade E85 designated as E85C. Tables 3 and 4 show the measured and calculated fuel properties. The intermediate blends E10-E50 were prepared by splash blending the E0 and E85 to the targeted concentration E100 was not evaluated. The data is only shown for reference from the stock used for making the E85 blend.

Table 3. Test Fuel Properties (Measured)

Fuel Blend	Vol. % Ethanol	LHV KJ/g	Density g/cc	Wt.% Water	RON
ASTM	D5501	D240	D4052	E203	D2699
E0*	0	43.397	.7426	0.013	90.8
EEE#	0	42.890	.742	0	96.8
E10	10.46	41.47	.7449	.1289	95.6
E20	21	39.53	.7512	.2373	99.7
E50	49.7	34.38	.7666	.4947	104
E85	82.2	29.2	.7854	.7653	106
E85C	82.8	29.16	.7856	.8179	(-)
E100 x	96.6	26.7	.794	0.9	(-)
* 91 RON (87 (R+M)/2) Test fuel M52642 -Gage products					
# EPA TIER II EEE – Haltermann Products					
x Ethanol feed stock includes 2% denaturant					

Table 4. Test Fuel Properties (Calculated)

Fuel Blend	Stoich. AFR	H/C	%O2 wt.	J/cc Rel E0
E0*	14.67	1.92	.012	1
EEE#	14.61	1.879	0	.994
E10	14.08	2.04	3.971	.964
E20	13.48	2.16	7.89	.927
E50	11.83	2.47	18.219	.825
E85	9.94	2.82	29.389	.716
E85C	9.89	2.82	29.613	.715
E100	8.99	3	34.78	.657

CONCEPT EVALUATION

The fundamental evaluation on the engine in this report focuses on un-boosted operation to maximize fuel consumption during loads typical of the Federal Test Procedure (FTP) city and highway drive cycles. The engine retained the OEM turbocharger configuration but all testing was completed with the waste gate fully open and the compressor bypass fully open to eliminate boost pressure. The advanced valvetrain allows the effective compression ratio to be managed to compensate for the ethanol variation in the fuel. Initial testing focused on the limit fuels of E0 91 RON gasoline and E85 fuel. The concept evaluation involved the following phases which will be reviewed:

- Engine valvetrain evaluation for load and compression ratio control
- Injection timing evaluation
- Evaluation of valve deactivation
- Valvetrain cam timing optimization for efficiency
- Evaluation of Ethanol Blends E0, E10, E20, E50 and E85
- Fuel consumption optimization of E85
- GT power simulation using speed-load maps optimized for fuel efficiency on E85

Valvetrain Evaluation

A primary feature of this engine concept is its ability to modify the effective displacement and hence compression ratio of the engine using valve timing. It still retains its full geometric expansion ratio of 11.85 independent of the effective compression ratio. The effective displacement is controlled by the intake phaser. By adjusting the intake valve closing time the trapped air mass is controlled as shown in figure 6. The system provides a smooth transition between cams by matching airflow at the switch point. The system is limited at low displacements by poor combustion stability due to increased residual fraction and slow combustion. At high loads, tuning allows volumetric efficiency to exceed 100% this providing increased effective displacement.

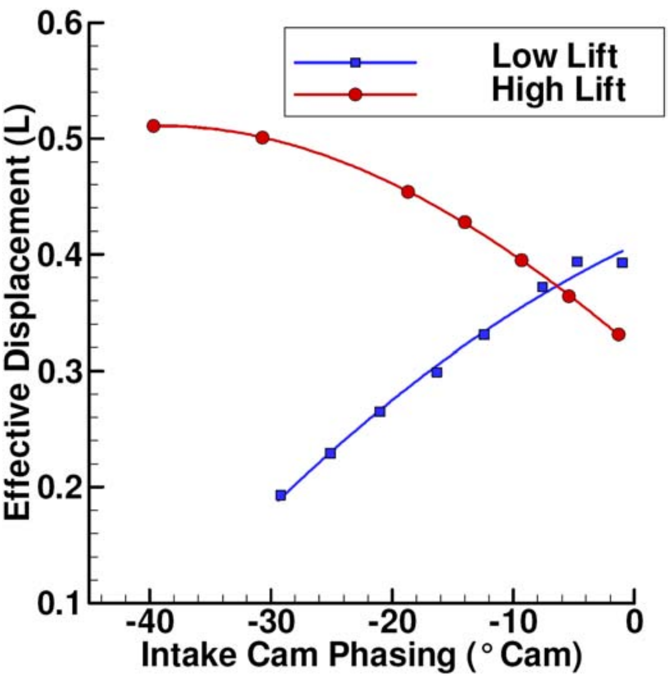

Figure 6. Effective displacement of engine controlled by lift selection and intake cam phasing, 2000 RPM, un-throttled operation, E85C fuel.

Cylinder pressure analysis is used to define the effective displacement and compression ratio. When operating on EIVC the valve closes prior to Bottom Dead Center (BDC) and the gas is expanded and recompressed in a nearly isentropic process. Conventional pegging of cylinder pressure is not possible at bottom dead center since the cylinder pressure is different than intake pressure. To resolve this issue alternative times were evaluated in the cycle both earlier in the intake stroke for intake pegging and late in the exhaust stroke with exhaust pegging. A third technique was also developed which involved adjustment to maximize linearity of the polytropic recompression process. Depending on the cam phasing, at higher speeds and loads, the intake and exhaust pegging were not always reliable due to transient flows across the valves. In these cases the polytropic technique was used. The definition of effective displacement and CR is illustrated in Figure 7 which shows an un-throttled EIVC condition. To define the effective displacement the volume where the cylinder pressure crosses MAP during polytropic compression is used as a definition. This effective volume can then be used to calculate the effective CR. The effective displacement is calculated by correcting for the geometric clearance volume. For the LIVC strategy the polytropic compression is extrapolated to MAP to define effective CR.

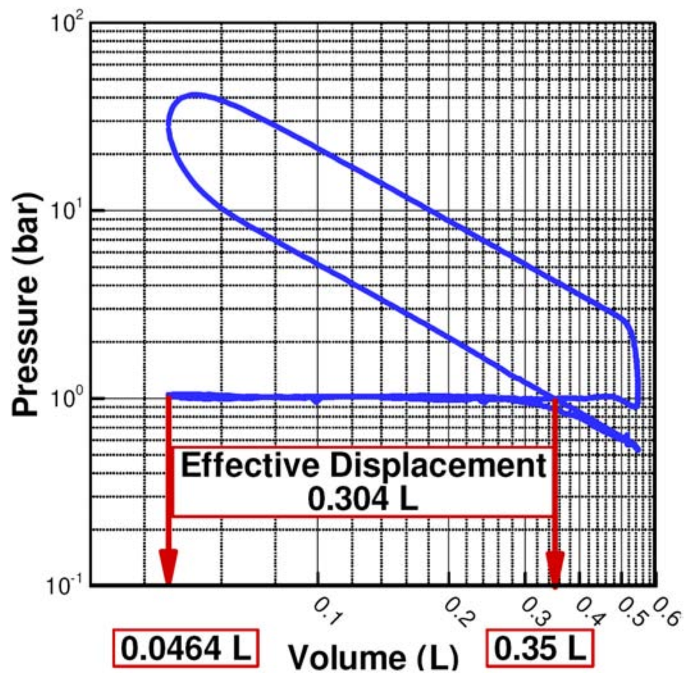

Figure 7. Definition of effective displacement, effective CR=7.6

The range of load control enabled by the EIVC/LIVC strategy is shown in figure 8. The low lift cam is speed limited to 4000 RPM but testing was limited to 3500 RPM due to deterioration of combustion stability. Operation of the EIVC strategy in an un-throttled condition revealed problems with long burn durations at poor combustion stability as the speed was increased, see figure 9. The increase in burn duration indicates poor turbulent mixing and near laminar flame propagation during initial flame development as shown in the 0-10 burn durations. Conversely when operating with an LIVC strategy the burn duration in crank angle degrees did not deteriorate significantly with speed maintaining short burn durations and good combustion stability.

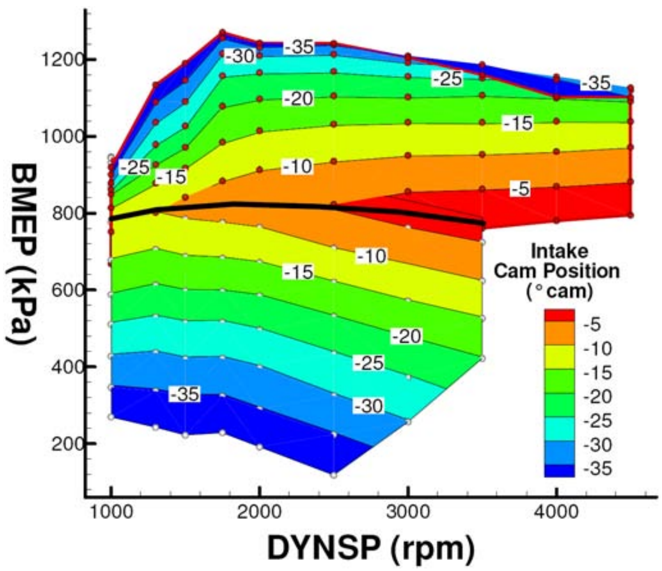

Figure 8. Un-throttled load control domain, contours indicate cam phasing location, 0=park, 40 =max phasing in cam degrees, Black Line Switch point

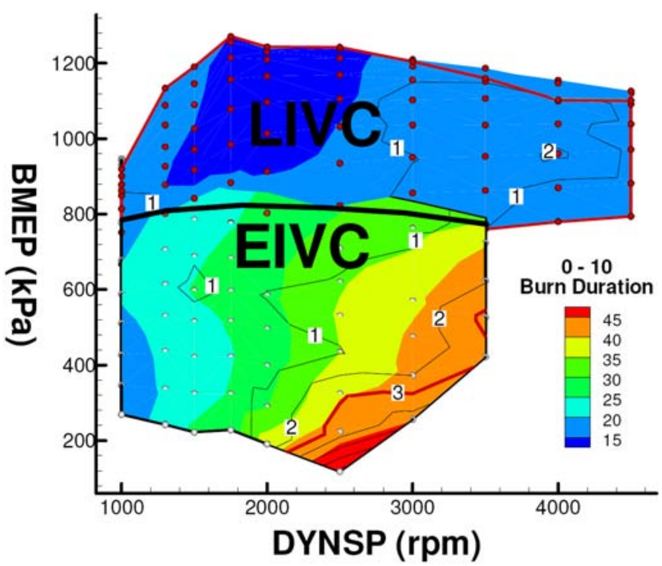

Figure 9. 0-10 cad burn duration (colors) with %COV contour lines for EIVC and LIVC

Injection Timing Evaluation

Another issue that was identified was a region prone to producing smoke at high loads and at speeds below 2000 RPM. This was only apparent when testing with the 91 RON E0 gasoline and not observed with E85 due to its resistance to particulate formation. This difficulty actually arises with all of the fuel blends tested other than E85 as will be discussed in the fuel blend section. To illustrate the issue a stoichiometric injection timing sweep at 1500 RPM, 8 bar BMEP, on gasoline is shown in figure 10. Of notice is the plateau in smoke level near 0.4 Filter Smoke Number (FSN) regardless of the injection timing. The FSN tends to trend sharply higher if injection is too early due to development of fuel films on the piston likely producing diffusion flames. The FSN also tends to increase for timings later than 280 cad bTDC likely the result of fuel on the cylinder liner which has not fully evaporated and mixed. An accompanying increase in HC is also shown for the later injection timings. During engine mapping injection timing optimization was conducted with the primary goal of minimizing fuel consumption, subject to acceptable combustion stability and FSN. The minimum fuel consumption was not limited by combustion stability since both are related, however at some conditions injection timing from minimum BSFC needed to be retarded due to high soot.

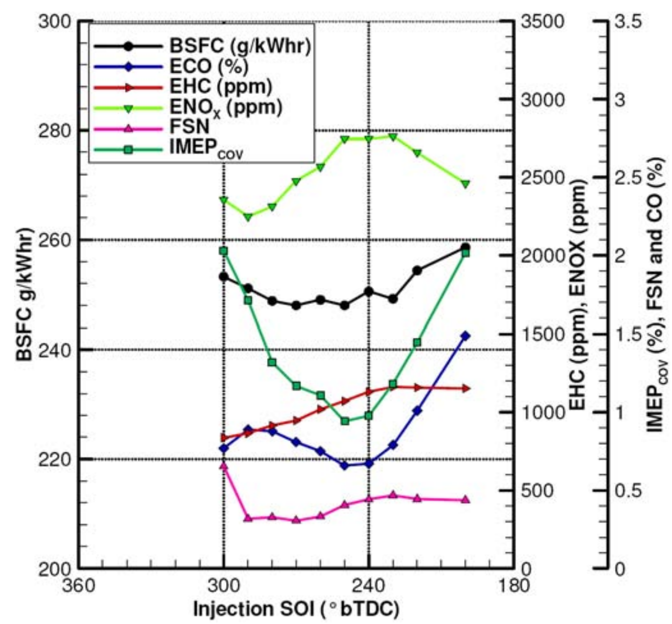

Figure 10. Injection timing sweep at 1500 RPM, 8 bar BMEP 91 RON E0 Gasoline. (2 Valves)

Valve Deactivation Evaluation

To address the issues identified with combustion stability with EIVC and particulates at high load valve deactivation was evaluated. By deactivating a single intake valve swirl is significantly increased and the measured tumble index approximately doubles. Valve deactivation to address these issues with improved charge motion with EIVC and LIVC has been effective. [23] Charge motion enhances the increased in-cylinder charge motion which can minimize liquid impingement that produces wall films that can lead to inhomogenieties and diffusion flames. Work by other researchers [24, 25] also shows similar findings for 2-3 mm lift valves. Bulk cylinder motion is significantly enhanced. Figure 11 shows flow bench results quantifying in-cylinder swirl and tumble at the peak lifts of the 2 cams for this application. The base engine is swirl neutral but does include a tumble feature in the intake port. To evaluate the effects on engine performance the valvetrain was reconfigured to allow deactivation of one of the intake valve. Comparisons were made at a series of operating points. These points are listed in [23] showing improved combustion stability with EIVC and reduced soot at higher loads.

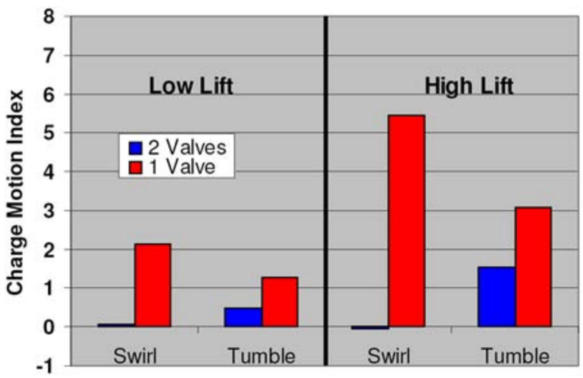

Figure 11. Swirl and tumble comparison with a 2 valve and 1 valve configuration, (single valve deactivation).

The effects of valve deactivation at high loads on the injection timing widow are shown in Figure 12, In contrast to Figure 10 the soot and hydrocarbon levels are reduced significantly at later injection timings. The best injection timing shifted earlier with a slightly reduced injection timing window.

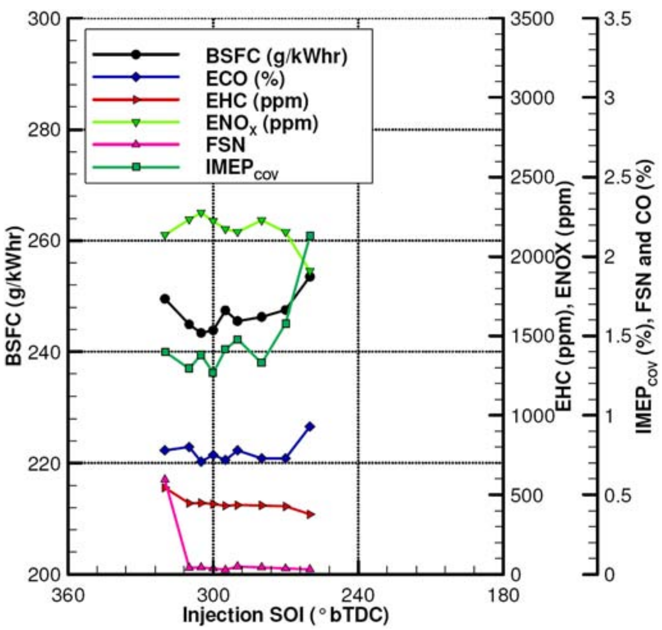

Figure 12. Injection timing sweep at 1500 RPM, 91 RON E0 Gasoline. (With valve deactivation).

The effect of valve deactivation on power density was evaluated by measuring peak torque from 1000 - 4000 RPM, Figure 13. The testing was conducted on E85 and all conditions are free of knock with MBT timing and stoichiometric fueling. Results show a shift to peak torque to slightly lower speeds and a falloff of peak torque at higher speeds. Particulate formation was primarily an issue at speeds under 2500 RPM, where engine breathing was not compromised significantly. Results when testing with E0 and E20 blends, which were knock limited, at peak torque did show an additional small reduction in peak torque due to an increased knocking tendency. [23] The increased swirl and charge motion is likely to increase the mixture temperature during the intake process thus leading to higher end gas temperatures. Additional spark retard was needed with the low ethanol fuels which compromised peak torque and reduced volumetric efficiency since these fuels are not as effective at charge cooling. Further valvetrain optimization to moderate the in-cylinder motion may provide a better compromise between volumetric efficiency and mixing, but is beyond the scope of work for this paper.

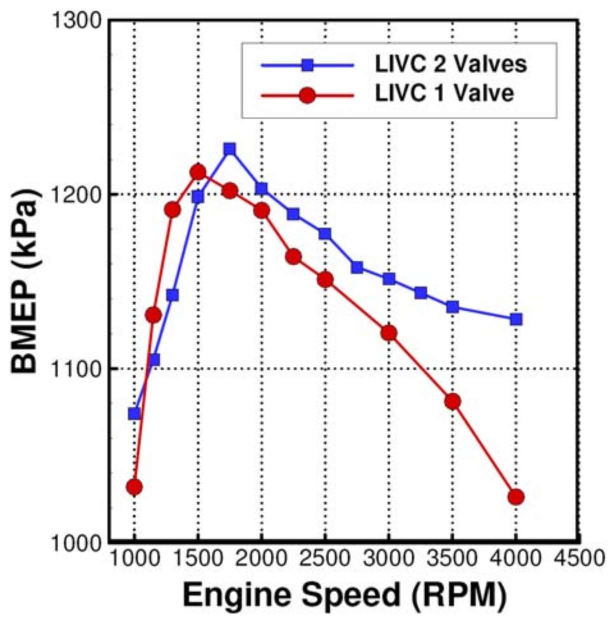

Figure 13. Peak torque curves, 2 Valve (Normal) vs. 1 Valve (valve deactivation) E85C Fuel.

Valvetrain Control Optimization

The use of DICP with the addition of the 2-step VVA system on a DI engine provides a high degree of freedom system for optimization of fuel consumption, emissions and performance. To gain a better understanding of the tradeoffs resulting from cam phasing, selected operating conditions were mapped across the allowable cam phasing domain with simultaneous optimization of injection timing for minimal BSFC. To illustrate these tradeoffs a cam optimization map is shown in Figure 14, showing the cam timing effects on fuel consumption, engine stability and MAP for a 2000 RPM, nominal 2 Bar BMEP operating condition. The testing was actually conducted with a fixed fueling rate which enabled more efficient testing. The BSFC is therefore based on the maximum performance for a given fuel quantity. The data presented is for an EIVC strategy with one valve deactivated and E0 Gasoline. When operating with valve deactivation and EIVC, combustion stability was excellent at 2 bar BMEP with the exception of small region with high valve overlap and some manifold vacuum to drive excess internal EGR. For loads above 2 bar combustion stability (COV) was typically less than 1% at all cam phasings, at lower loads the region of excessive EGR with poor COV increased. As shown in the plot MAP varies from 50 KPA to un-throttled conditions. Of interest, minimum BSFC is not under un-throttled conditions but with light throttle which enables capture of internal EGR. This also results in reduced NOx Emissions.

Cam phasing and injection timing optimization was also completed for the 2 valve EIVC configurations as well as the LIVC strategy with and without valve deactivation. A comparison between these strategies is shown in figure 15 for E85. Evaluations were also conducted with E0 showing similar trends. Data in Figure 15a (Top) shows the fuel consumption benefit resulting from the strategies evaluated. All improvements are relative to the base engine gasoline thermal efficiency. All of the strategies show an improvement near 8% at higher loads. This is a combination of benefits from the increased compression ratio and thermodynamic benefits inherent to E85. [9] When operating on gasoline the improvement is about 5% as a result of the increased compression ratio. A significant fuel consumption improvement using EIVC with valve deactivation is provided below 5 bar BMEP, resulting from reduced pumping losses. The base engine configuration and calibration already was providing very good fuel consumption by use of DICP for internal EGR management to reduce throttling. The reduction in throttling is apparent in figure 15c (Bottom) where at loads above 2 Bar BMEP the MAP was above 90 KPA for the EIVC with deactivation strategy. Figure 15b (Mid) shows the NOx emissions. With E85 it was possible to introduce significant internal EGR without EGR induced knock at higher loads. The LIVC strategy, with high overlap, provides minimal NOX emissions with E85. High residual levels can be introduced and excellent charge motion and combustion stability is maintained. This benefit is limited with gasoline or low ethanol blends since high internal EGR results in knock requiring spark retard which leads to deteriorating combustion stability and efficiency. This will be discussed further in the section on ethanol blend testing.

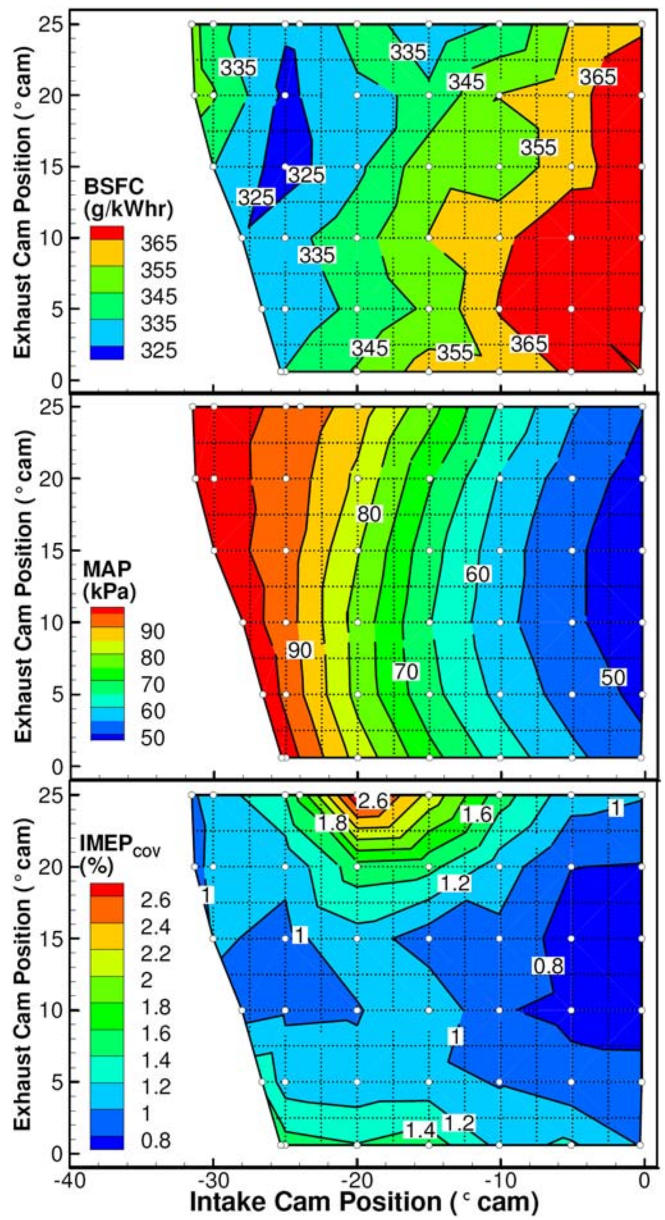

Figure 14. Cam phasing optimization map with EIVC with valve deactivation, 2000 RPM, Fixed fuel 9,95 mg/cyl, Nominal load 2 Bar BMEP, 91 RON E0 gasoline

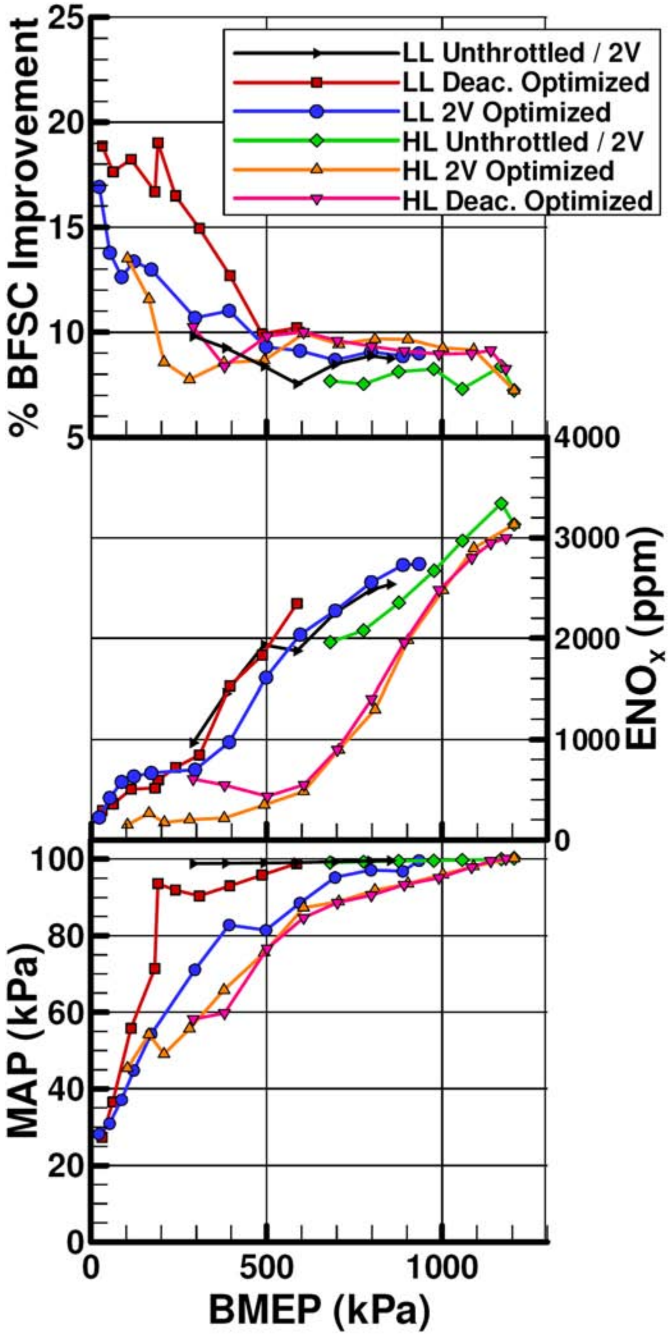

Figure 15. 2000 RPM load sweeps, E85C Fuel, Evaluation of valvetrain control strategies. (a) Improvement in fuel consumption (thermal efficiency) over base engine, (b) NOx Emissions, (c) Map showing reduced throttling.

FUEL BLEND EVALUATION

Fuel blends from E0 gasoline to E85 were evaluated; fuel properties are shown in Tables 3 and 4. A 97 RON E0 Gasoline was also tested for reference. A series of test were conducted to evaluate the benefit of ethanol content for knock control and soot reduction, which were the primary benefits observed in the previous phases of work.

The test consisted of the following evaluations:

- Start of Injection (SOI) timing sweeps (EIVC and LIVC) 2250 RPM, 6 Bar BMEP
- EGR tolerance evaluation LIVC, 2000 RPM, 6 Bar BMEP
- Load sweeps to identify knock limited load and compression ratio LIVC, 1500 RPM, 2000RPM
- Knock limited torque vs. speed LIVC, 1000-4000 RPM

EIVC 2250 RPM SOI Evaluation

The primary interest in evaluating injection timing windows was to determine if the ethanol content significantly changed the allowable injection window or optimal timing with the EIVC and LIVC strategies. Testing with E0 and E85 had demonstrated a significant difference resulting from the E85 blends resistance to soot. When evaluating the EIVC strategy a small acceptable injection timing window was typical. Narrow injection windows with EIVC has also been documented in another evaluation.[25] E85 provided a wider window resulting from elimination of the soot constraint there was no significant benefit identified from the intermediate blends tested. Soot levels were generally lower for later injection timings with increasing ethanol content. Figure 16a. The combustion stability limited timing was not changed for any of the blends evaluated, Figure 16b. Optimal timing to minimize fuel consumption was limited by these two constraints and was similar for all of the fuel blends, Figure 16c. Hydrocarbons and NOx emissions did tend to trend lower with increasing ethanol content, Figures 17 a,b. This was also typical for all of the blend testing. Lower NOx, should result from reduced combustion temperatures due to charge cooling and a lower adiabatic flame temperature. Hydrocarbon reductions likely result from the reduced fraction of higher molecular weight components in the fuel blend. The effect of ethanol concentration on aldehydes was not measured in this study but has been shown [27] to increase with increasing ethanol content

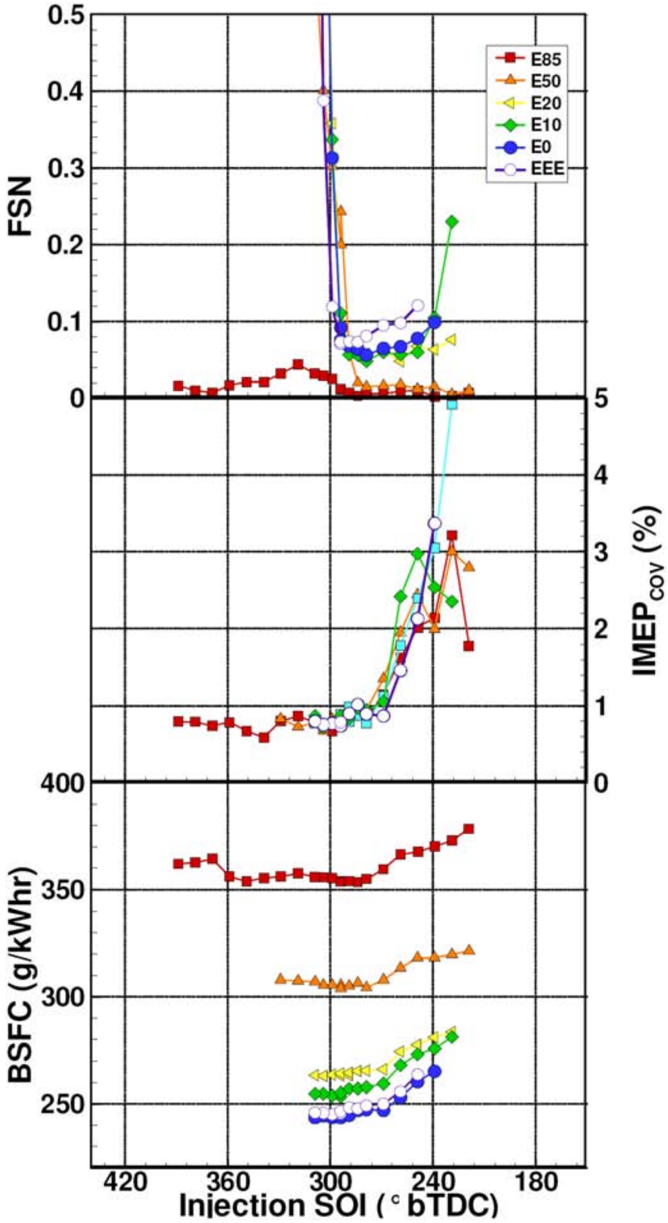

Figure 16. (a) Soot (FSN), (b) Combustion stability (COV%), (c) Fuel consumption (BSFC g/KW Hr), 2250 RPM, 6 Bar BMEP, EIVC cam

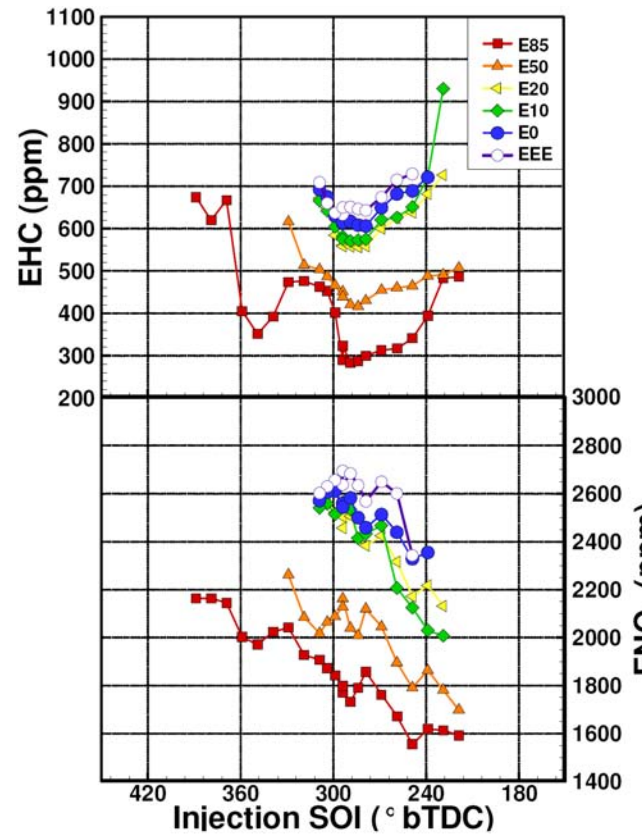

Figure 17. (a) Engine out Hydrocarbons (ppm), (b) Engine out NOx (ppm), 2250 RPM 6 Bar BMEP, EIVC cam

LIVC 2250 RPM 6 Bar Evaluation

Evaluation of injection timing with the LIVC strategy provided similar results to the EIVC strategy but combustion stability was better allowing later timings, Figure 18 a (FSN), b (%COV of IMEP). Optimal injection timings were similar between the blends and emission trends also trended lower with ethanol content.

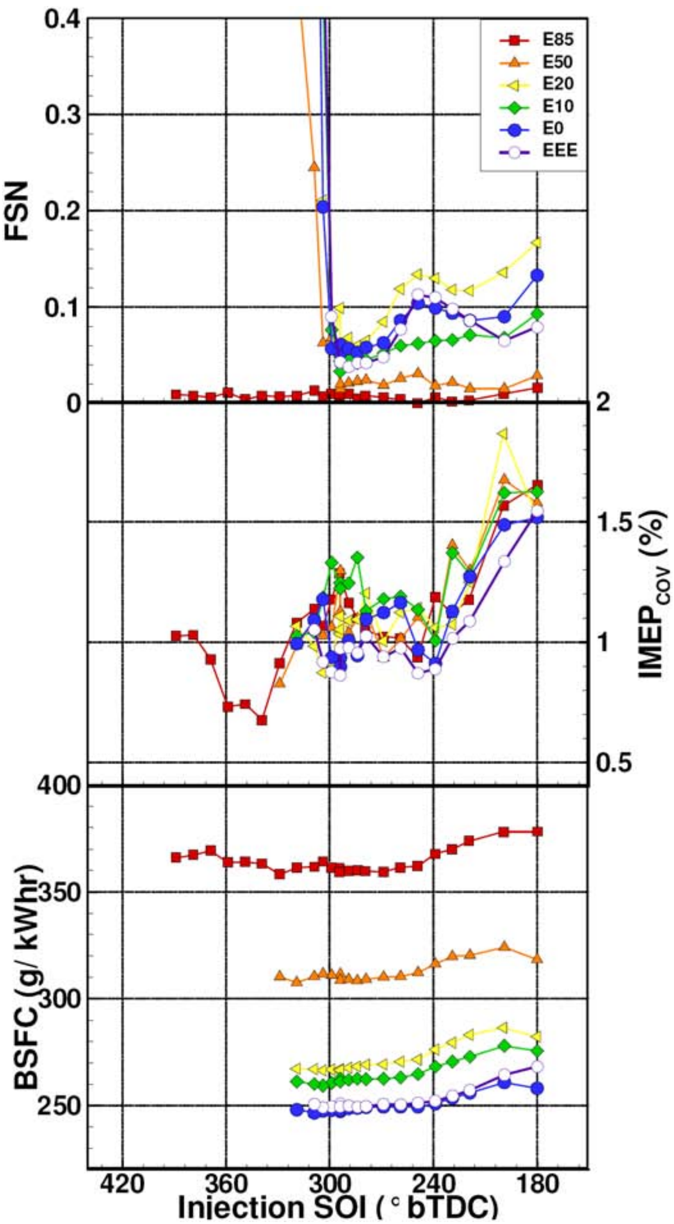

Figure 18. (a) Soot (FSN), (b) Combustion stability (COV%), (c) BSFC (g/kWhr), 2250 RPM 6 Bar BMEP, LIVC cam

Internal EGR Tolerance

E85 showed high resistance to EGR-induced-knock at high loads allowing internal EGR optimization up to peak torque. During E85 cam optimization it was determined that a small operating window existed that allowed high levels of internal EGR, high compression and high MAP where E85 was susceptible to EGR induced trace knock. Slight retard (2-3 cad) to eliminate knock was required for E85. To evaluate the effect of ethanol content this test condition was repeated for the fuel blends to determine the relative knock resistance of the test fuels. The test was conducted as an intake cam phasing sweep which produced an increase in the effective CR and an increase in valve overlap to allow more internal EGR. The condition of 0 degrees of cam phasing corresponds to nearly unthrottled operation with very low residual. All conditions are stoichiometric fueling and MBT or knock limited spark. The effect on NOX and BSFC is presented in terms of intake cam phasing Figure 19 (a) and (b) respectively. Once the onset of knock was detected spark retard was used to keep knock at an acceptable level. The use of spark retard also results in a reduction in NOx, but both combustion stability (COV) and BSFC deteriorate. Figure 20 (a) and (b) show the required retard in combustion phasing of the 50% burn duration (CA50) and the combustion stability. For this test condition, E50 and E85 ethanol blends enable a significant reduction of NOx with a reduction in fuel consumption. For gasoline and lower ethanol blends there is a tradeoff because of the reduced knock resistance. The fuel consumption reduction is also partially the result of a reduction of manifold vacuum as overlap is increased resulting in reduced pumping work for cam phasings over 20 degrees, Figure 21 (a). The effect of ethanol content on burn duration was small under low EGR conditions Figure 21 (b) but difficult to distinguish at higher EGR levels as the effect of spark retard confounds the results. Adjusting for the variation in energy content of the fuels highlights the advantage of higher ethanol blends to improve thermal efficiency by allowing higher internal residual before knock is induced.

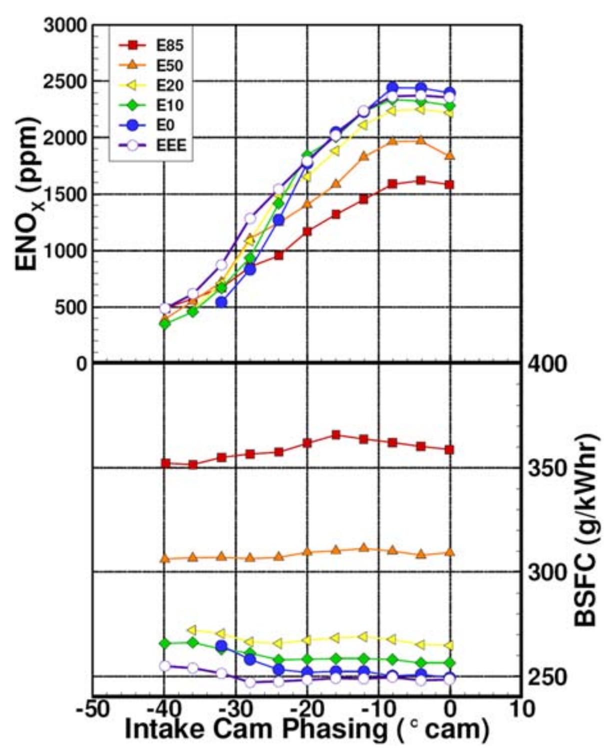

Figure 19. (a) Engine out NOx (ppm) (b) Fuel Consumption BSFC (g/KW Hr), 2000 RPM 6 bar BMEP, LIVC cam

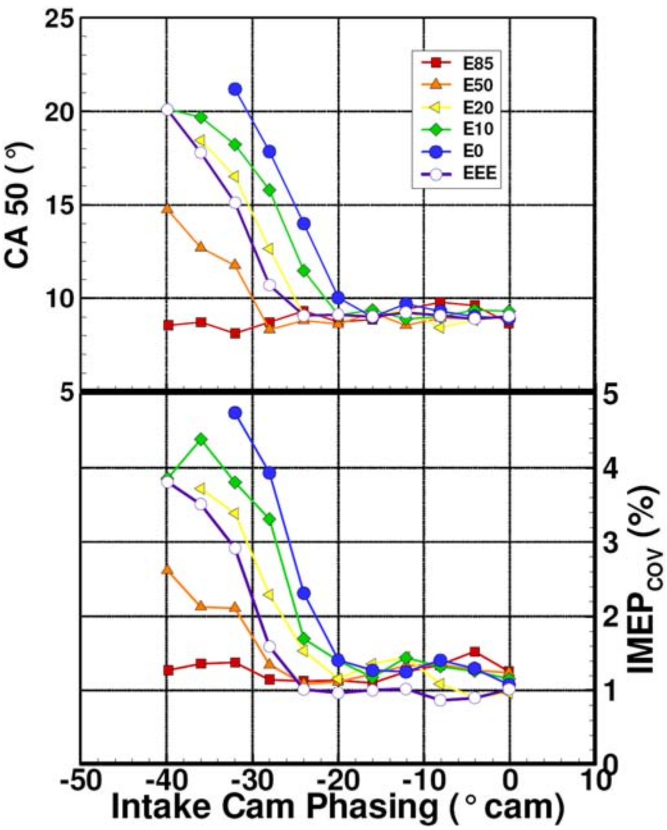

Figure 20. (a) Combustion phasing CA50 (cad aTDC) (b) Combustion Stability (COV%), 2000 RPM 6 bar BMEP, LIVC cam

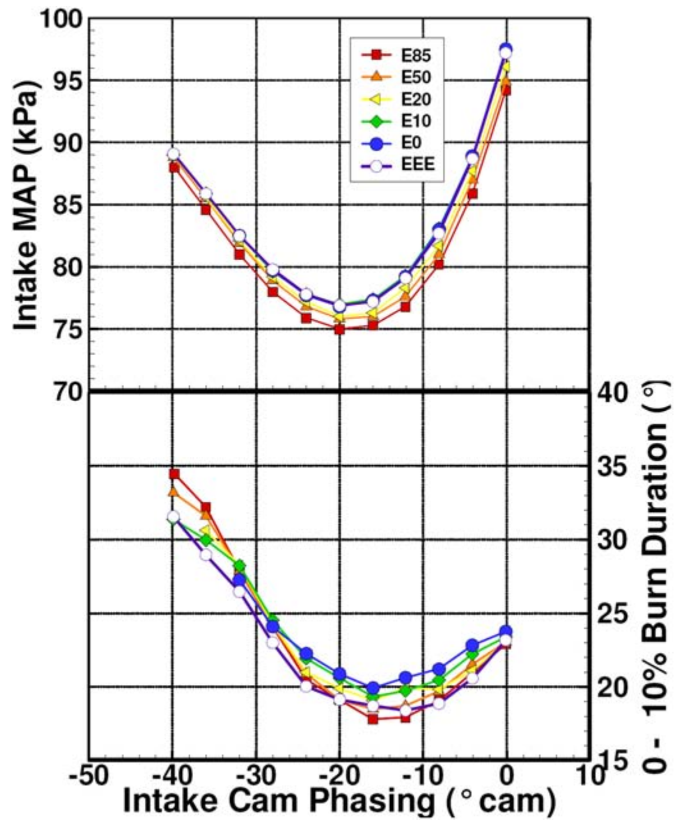

Figure 21. (a) MAP (KPa), (b) 0-10% burn durations (cad), 2000 RPM 6 bar BMEP, LIVC cam

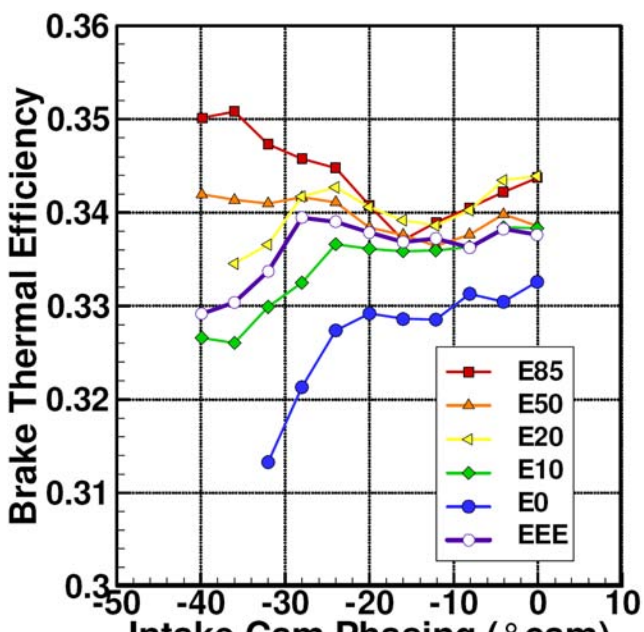

Figure 22. Brake thermal efficiency, 2000 RPM 6 Bar BMEP, LIVC cam

Load - Effective Compression Ratio Sweep (2000 RPM)

To determine the effectiveness of ethanol content for suppressing knock an effective compression ratio sweep of the engine was run at 1500 and 2000 RPM. By adjusting the intake valve closing time the effective compression ratio can be varied from 8-12. The exhaust cam phasing and injection timing were fixed for all cases, to focus on the fuel effects. Injection timing was chosen to avoid the FSN increase resulting from injections being too early. The valvetrain allows the engine to maintain MBT spark without knock for all of the fuels including the E0 91 RON gasoline. As the compression ratio is increased the trace knock limit at MBT or knock limited spark was identified for each fuel. Figure 23(a) shows ignition timing as a function of load. To provide CR specific data the MBT or knock limited combustion phasing with respect to effective compression ratio is shown in figure 24. Of interest is the similar performance of the E10 / 91 RON blend to the 97 RON EEE gasoline, showing the benefits of small quantities of ethanol for increased knock performance. As the ethanol content increased higher effective CR was possible, for the E50 and E85 blend no significant spark retard or performance penalty was apparent. E0 gasoline provides minimal fuel consumption up to 9 bar BMEP while E20 provides minimum fuel consumption at peak power, Figure 23 (b). The effect of increasing load via effective compression ratio resulted in an increase in hydrocarbons, NOx and FSN. The emissions were however strongly related to the ethanol content with higher ethanol blends reducing emissions for all 3 constituents, Figure 25 (,a,b,c) As compression ratio and load is increased the maximum pressure rise rate increases, which may be undesirable from the standpoint of combustion noise. For reference this information is provided in Figure 26. Spark retard can be used to limit combustion noise independent of knock, the reduction of pressure rise rate with low ethanol fuels is the result of spark retard for knock control.

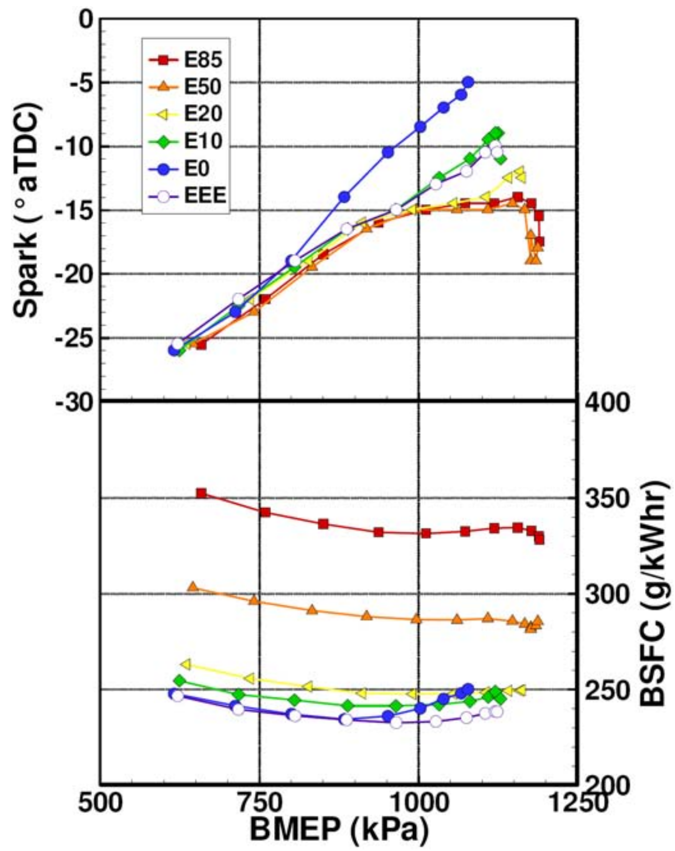

Figure 23. (a) Knock limited load and spark timing, (b) Fuel consumption BSFC (g/KW Hr), 2000 RPM Un-throttled LIVC

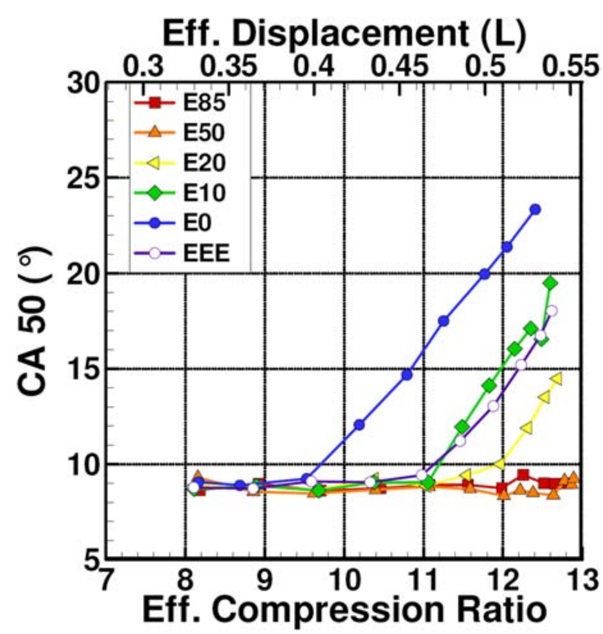

Figure 24. Knock limited CR and combustion phasing (CA50, cad aTDC), 2000 RPM un-throttled LIVC.

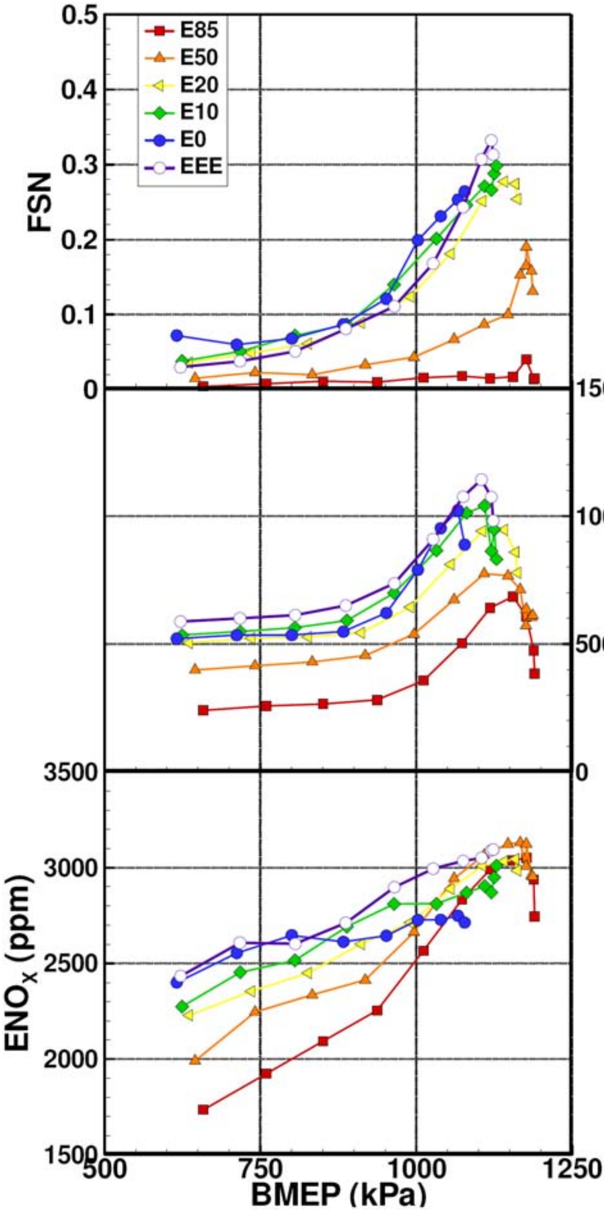

Figure 25. (a) Soot (FSN), (b) Engine out Hydrocarbons (ppm), (c) Engine out NOx (ppm), 2000 RPM un-throttled LIVC.

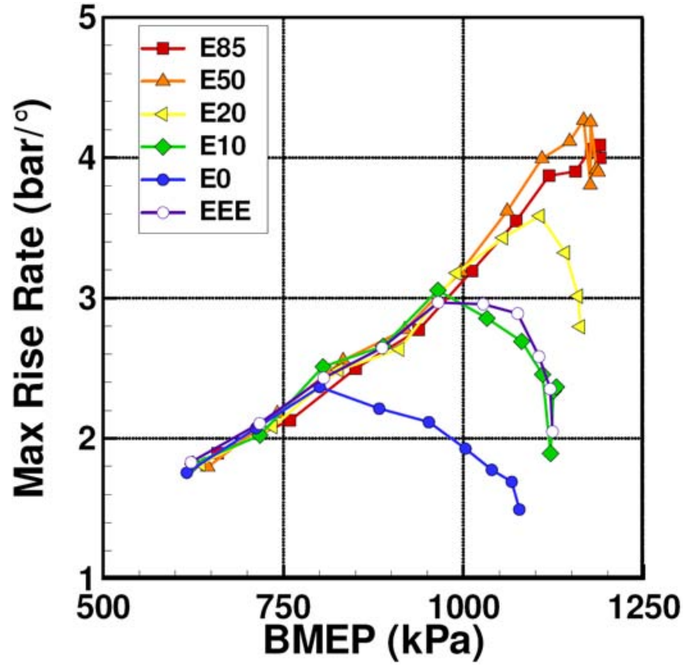

Figure 26. Maximum pressure rise rate (Bar/deg) for ethanol blends, 2000 RPM un-throttled LIVC.

Load - Effective Compression Ratio Sweep (1500 RPM)

Evaluation of the knock limited load and compression ratio was also conducted at 1500 providing a more knock sensitive condition. The testing was conducted at stoichiometric conditions and MBT or knock limited spark. The fuels are more knock prone requiring another 10% ethanol for similar knock resistance compared to 2000 RPM, Figures 27 (a). 1500 RPM is near the peak torque with E85, which results from tuning producing some scavenging. Due to the scavenging an increase in BSFC results as load is increased. Even though the net air fuel ratio is stoichiometric if air is scavenged into the exhaust the in-cylinder charge will be rich, producing additional torque and increasing fuel consumption, Figure 27 (b). This explanation is also supported by an increase in engine out CO and O2 at peak load, which would result when a rich in-cylinder mixture is mixed with air that was over scavenged. Figure 28 shows the required combustion phasing retard to limit knock as the effective compression ratio is increased. The knock limited compression ratio is increased about 1 point for each 10% increase in ethanol up to E20.

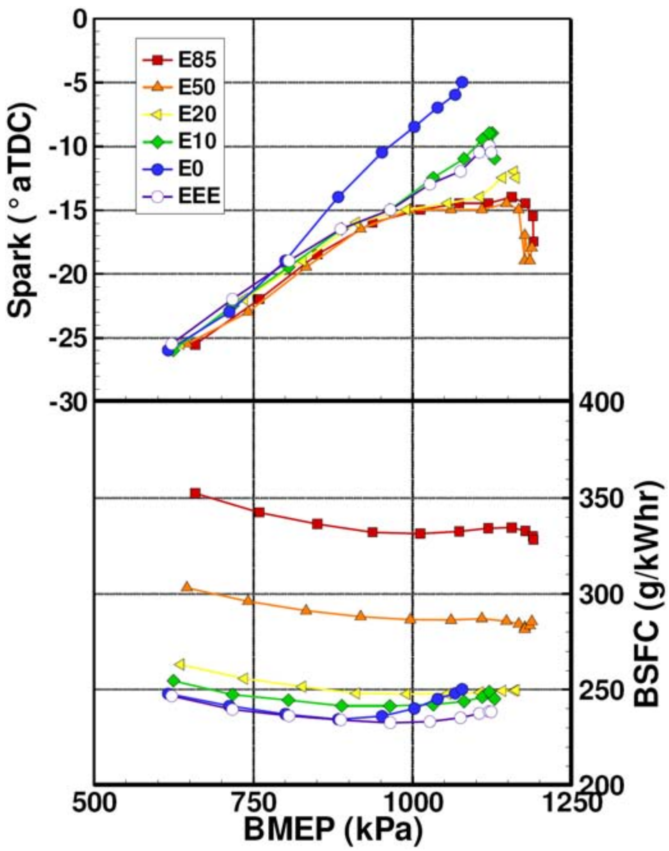

Figure 27. (a) Knock limited load and spark timing (b) Fuel consumption BSFC (g/KW Hr), 1500 RPM Un-throttled LIVC

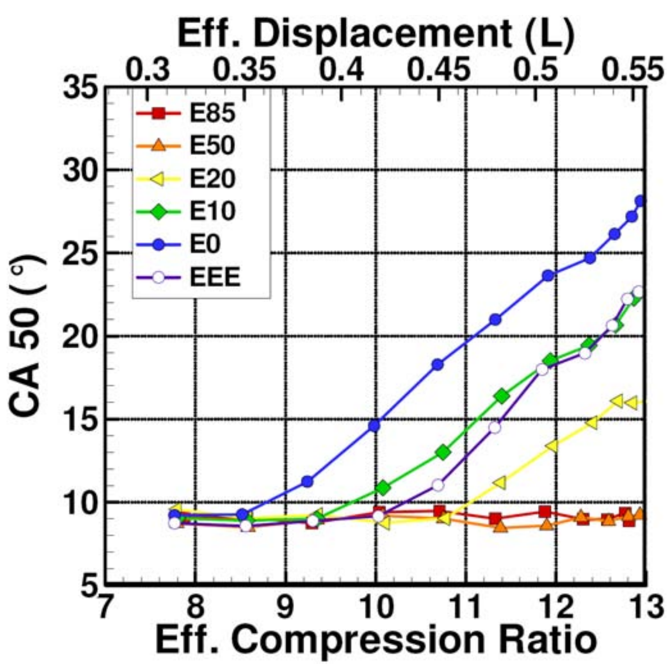

Figure 28.) Knock limited CR and combustion phasing (CA50, cad aTDC), 1500 RPM un-throttled LIVC

Knock Limited Torque - RPM Sweep

Testing was done over the speed range of 1000- 4000 RPM to identify the knock limited load and compression ratio. Testing was done in a similar fashion to the 2000 and 1500 RPM load sweeps by increasing the effective compression ratio until trace knock was detected. Data was taken with an MBT combustion phasing (CA50; 8-10 cad aTDC). Figure 29 (a,b,c) shows the knock limited BMEP, Combustion phasing and effective compression ratio over the speed range. There is a significant difference in the knock limited CR and associated MBT torque for the ethanol blends. Knock limited torque at MBT is limited to an effective CR of 7.6 at 1000 RPM to 10.5 at 4000 RPM. Increasing the ethanol content shows a consistent effect of allowing a 1 point increase per 10% ethanol addition until the geometric compression ratio of the engine is reached.

For fuel blends that were knock limited, spark retard was used to retard combustion phasing until maximum torque was achieved. As the effective compression ratio was increased spark retard allowed knock free operation. For the low ethanol blends the effective displacement and CR was limited at low speeds since excessive spark retard was needed as the CR increased. The peak CR was limited to a point in which further increases resulted in a loss of torque. Figure 30 (a,b,c) shows the knock limited BMEP, combustion phasing and effective compression ratio over the speed range. The use of spark retard allowed the effective CR to be increased about 2.5 points before the efficiency penalties associated with spark retard were more significant than the increased displacement. This level of spark retard was typically at a combustion phasing near 24 cad aTDC. Unlike a fixed cam system the use of VVA with LIVC allows cam phasing selection to limit the efficiency loss that results from very late combustion phasing which produces lower torque with increased fuel consumption. For fuel blends above E20 the maximum torque curve was not significantly limited. E20 can provide 97% of the peak torque of E85.

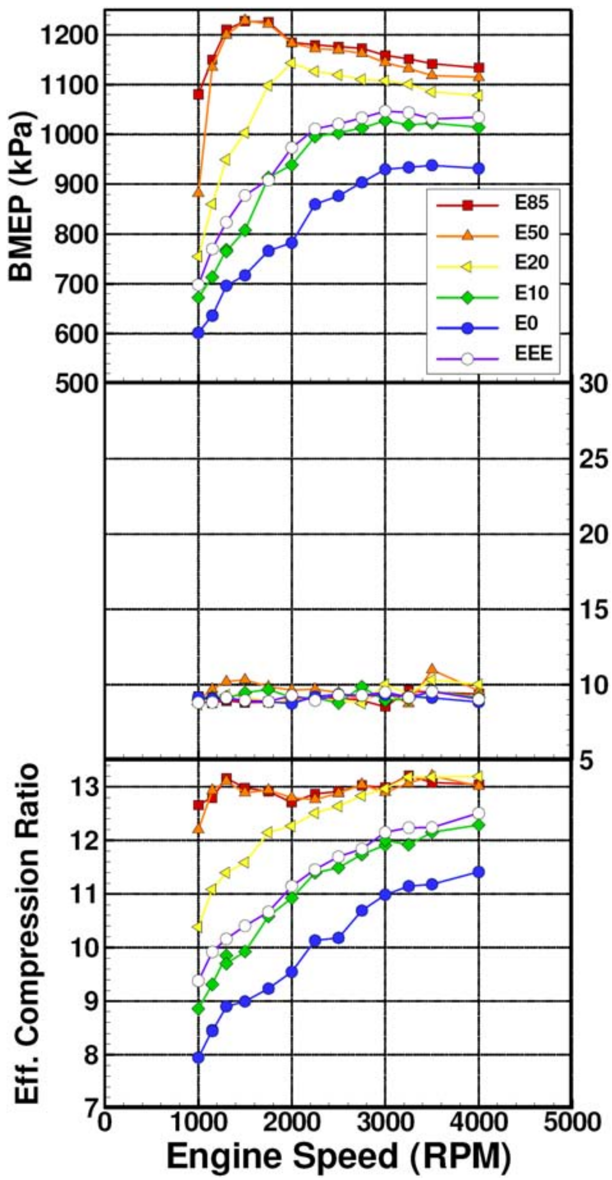

Figure 29. (a) Knock limited load,(b) Combustion phasing(CA50), (c) CR, 1000-4000 RPM LIVC Cam

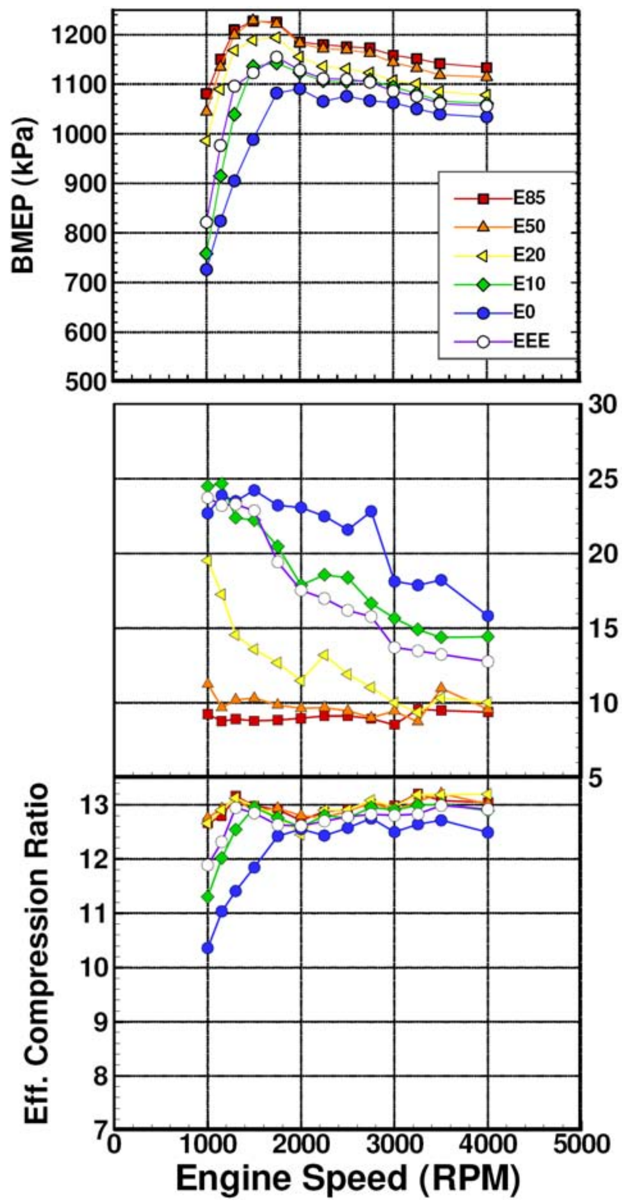

Figure 30. (a) Peak Torque, (b) CA50, (c) CR of peak torque vs RPM, MBT or knock limited torque, Stoichiometric operation. 1000-4000 RPM, LIVC Cam

ENGINE - VEHICLE OPTIMIZATION

To optimize vehicle fuel economy, improvements in both the base engine performance and how the engine is efficiently utilized in the vehicle are important. The improvements made to the base engine resulted in improvements in engine efficiency from 5% at high loads to over 20% at low loads. This is over and above the base engine which had already taken advantage of GDi technology with DICP to produce a very competitive baseline. The relative improvement in efficiency is shown in Figure 31 for E85. Peak thermal efficiency on E85 reached 38% at 2250 RPM, 11.9 bar BMEP. The use of boost will allow an increase in power density and a further increase in peak efficiency. However for typical drive cycles the range of engine operation focused on in this study is sufficient.

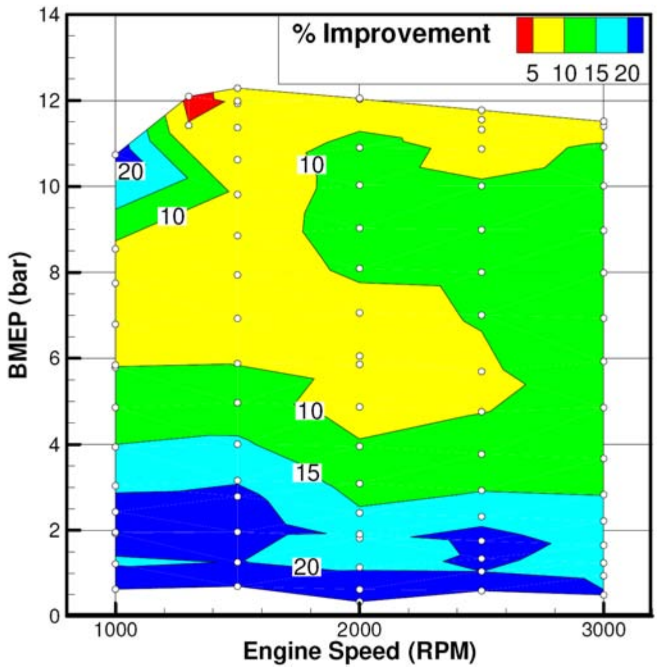

Figure 31. E85 Speed load map showing relative thermal efficiency improvement over base engine data.

Fuel Consumption Optimization

The intention of this work is to identify opportunities to improve overall vehicle efficiency when operating on E85. A significant part of this involves identifying operating conditions that allow more efficient operation of the engine. During many operating conditions with mild acceleration and moderate vehicle speeds the engine power requirements are significantly less than the engine's capacity. The Federal Test Procedure (FTP) city and highway cycle are examples of operating modes that place the engine under inefficient operating conditions. With the development of improved transmissions with 5, 6 or more speeds a significant potential exists to down-speed the engine to significantly improve performance. To analyze this potential Figure 32 is introduced. In addition to showing the BSFC curves an analysis of preferred operation conditions is presented. With the ability to select between different gear ratios, it would be desirable to operate the engine in the most efficient operating point for the desired power to supply the power demanded by the driver. To evaluate this, a line of constant power is shown by the blue dashed line, in this case 10KW. If we compare the locus of points the most efficient operating condition would be to operate at a low speed and high load. This point is shown by the Yellow line which is the locus of points of most efficient operation as a function of power. While it may not be possible due to transmission capability or even desirable to operate at this load due to Noise, Vibration and Harshness (NVH) issues it is useful as a reference. For comparison the lines with the red labels show the relative fuel economy penalty by operating at different conditions for the same power. For example at the demanded power of 10 KW this can be achieved at 1500 RPM, 4 Bar BMEP which suffers a 5% penalty, 2000 RPM 3 Bar which has a 17% penalty or 3000 RPM 2 Bar which suffers a significant 45% penalty. Figure 32 thus provides a useful tool to identify regions that proper selection of the transmission gear and shift schedules can significantly aid vehicle fuel economy. This must be balanced with needs for good drivability; however the combination of good low end torque and improved transmissions offers the potential for both good fuel economy and performance. To leverage this potential, shift strategies to minimize the amount of time at high speed low load conditions with high fuel penalties were evaluated in vehicle drive simulations.

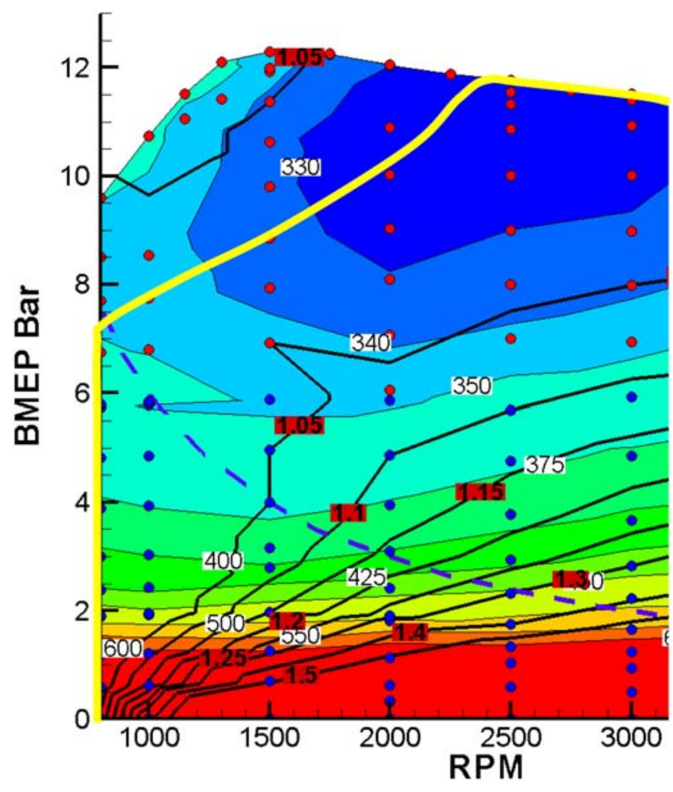

Figure 32. Speed Load map of E85 BSFC (Contours) showing relative fuel consumption (Red Labels) at equivalent power (Blue Dashed Line), Yellow line indicates most efficient path. Red Points LIVC, Blue Points EIVC (Deac).

Vehicle Simulation

A vehicle simulation using GT Drive was conducted to evaluate the potential for fuel consumption reduction from engine improvements, hardware selection and transmission calibration. A production Chevrolet Cobalt with the base engine was used as a reference. Baseline fuel consumption data was adjusted for the lower energy content of E85. This produced an E85 baseline with equivalent thermal efficiency to the base engine over the speed load domain. The system was then evaluated incrementally to determine the relative benefit of engine improvements, more aggressive shift schedules, improved transmission range and reduced final drive ratio. To reflect the engine modifications the measured fuel consumption for the E85 optimized engine was used. A shift schedule was developed which stayed within the unboosted operating window and avoided higher speed low load conditions that could be more efficiently provided by upshifting to more favorable conditions. The final drive ratio was reduced from 3.73 to 3.23 to provide additional down-speeding potential. Integration of a 6 speed transmission to offset the loss of launch torque with the lower axle ratio was also included in the evaluation, see table 5 for a tabulation of gear ratios.

Table 5. Transmission gear ratios

Gear	5 speed	6 speed
1	3.75	4.48
2	2.26	2.87
3	1.51	1.84
4	1	1.41
5	0.73	1
6		0.74

The operating points on the FTP city cycle for the base case and the final case are presented in Figure 33. A significant reduction in the amount of time spent below 4 bar BMEP above 2000 RPM is apparent. This is the result of the upshift strategy.

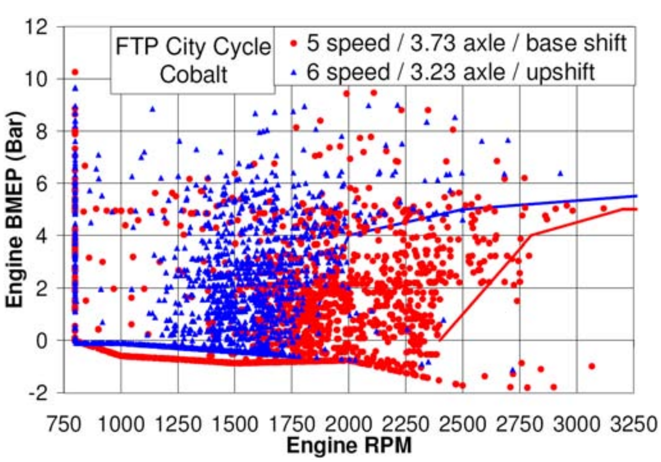

Figure 33. FTP City cycle showing operating points with base and proposed transmission, axel and shift schedule. Up-shift line, Red (Baseline) Blue (up-shifted)

Results of the drive cycle evaluation are shown in Figure 34. The benefit of the improved strategies for reducing the disparity between fuel consumption with gasoline and E85 is almost entirely offset on the FTP city cycle but is less effective as the demands of the driving conditions increase. At highway cruise speeds the shift schedule has no effect since the vehicle is in overdrive in all cases, only the benefits of the lower final drive ratio and the engine modifications are evident. The 6 speed transmission's final drive ratio is similar to the 5 speed so its advantage will primarily show up in launch performance not fuel economy.

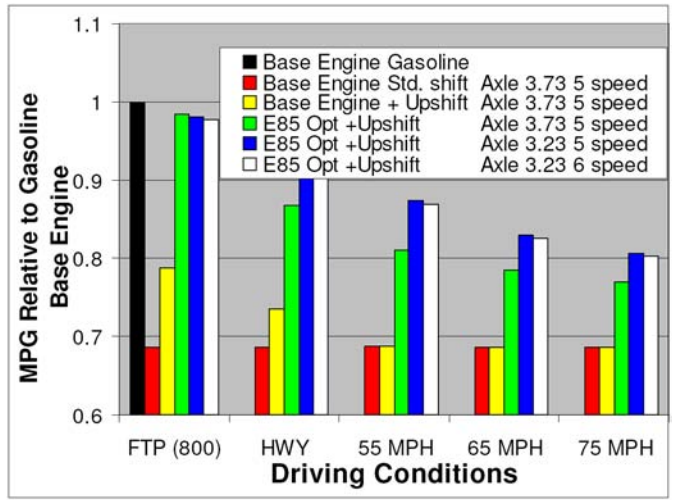

Figure 34. Relative fuel economy to base engine operating on E85 for various operating modes and effect of transmission axel ratio and shift strategy.

It is also important to consider that many of the techniques used to improve performance on E85 would also improve fuel consumption with gasoline or lower ethanol blends. Differences will show up more in performance and may need a shift schedule dependent on the ethanol blends torque capability. Ethanol blends from near E20 provide a good compromise, enabling most of the performance of an E85 blend with a significantly reduced energy density penalty. Blends in this range would likely be able to offset the fuel density penalties with improved efficiency while providing superior performance to gasoline.

Fuel Blend Variation Issues

To utilize ethanol blends effectively relies on consistent fuel properties of the E85 gasoline blend stock to produce reliable intermediate blends. A fuel specification for E85 for use in ethanol blend pumps would allow the benefits of ethanol to be consistently leveraged. If ethanol is instead used to upgrade a low quality gasoline fuel stock these benefits may be limited. A survey of the reported RON, [3, 7, 9, 10, 11, 12, 14, 26,28] of ethanol blends is shown in Figure 35. The large degree of reported variation in RON is partially the result of different gasoline blend stocks but may also indicate variation in testing with ethanol fuels or a high degree of sensitivity to fuel composition. The influence of ethanol content on RON has been shown to blend in nearly a linear response to the mole fraction of ethanol [28], this is in contrast to the non-linear response on a volumetric blend ratio

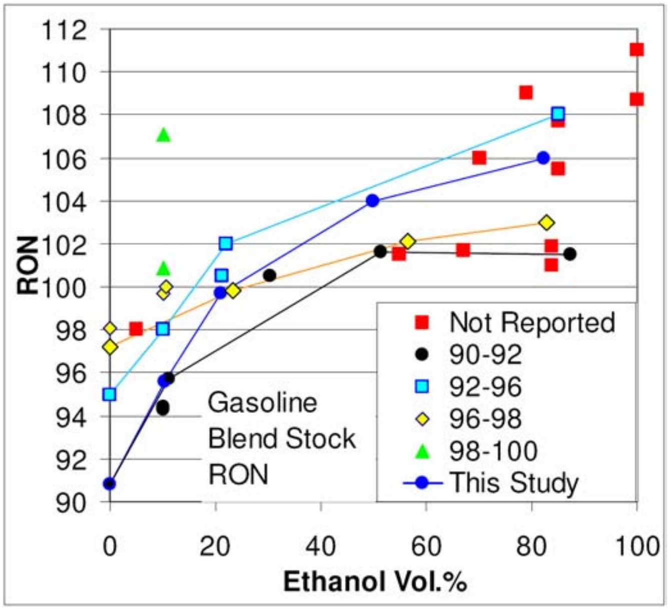

Figure 35. Variation of RON for ethanol blends

SUMMARY

A 2.0 L GDi Engine with DICP was modified for flex fuel operation with increased compression ratio and 2 step VVA to control effective CR and load with valvetrain phasing.

Effective CR could be controlled allowing MBT spark with cam phasing with an LIVC strategy for 91 RON gasoline E0-E85 fuels. Increased load could be achieved with spark retard. Excessive retard could be limited with valve phasing control.

Valve deactivation was used to improve combustion stability at low loads

Cam phasing and injection timing were optimized for E85 to minimize fuel consumption and emissions

Gasoline ethanol blends E0, E10, E20, E50 and E85 were evaluated at selected operating conditions where E0 and E85 differed significantly to understand the blending response.

Vehicle level simulation was carried out to leverage the improved low end torque with E85 to improve fuel economy by down-speeding the engine.

Future work will include evaluation of E30 and E40 blends. The engine will also be operated with boost for E0-E85 fuel blends.

CONCLUSIONS

High low end torque, (11-12 bar BMEP) under 2000 RPM could be achieved without knock for E50 and E85 blends.

Load could be managed efficiently down to 2 bar BMEP with an EIVC strategy providing improved fuel economy. The use of valve deactivation significantly improved performance.

Lightly throttled performance for internal residual control was more efficient than unthrottled operation.

Valve deactivation at high loads under 2500 RPM was effective at reducing soot.

Valve deactivation did not significantly affect peak torque under 2500 RPM with E85.

Intermediate fuel blends up to E50 were still prone to soot formation with early injection timing.

Engine out HC, NOx and soot emissions were reduced with increasing ethanol content.

Resistance to EGR induced knock enabled reduced NOx emissions for higher ethanol blends, using high valve overlap for internal EGR.

The improvement in low end torque with E20 -E85 blends should enable better launch performance and give an opportunity to operate more efficiently with down-speeding.

For the FTP city cycle much of the energy density gain from the base configuration can be made up with a down-speeding strategy and hardware leveraging the benefits of E85.

Intermediate blends near E20 can provide the majority of the performance benefit of E85 and enable strategies that offset their lower energy penalty.

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ACKNOWLEDGMENTS

The authors wish to acknowledge project support from:

Delphi Powertrain; Tim Kunz, Cindy Tawaf, Ray Parker, Steve Crossman, Tom Verstraete, Tim Coha, Eunjoo Hopkins, David Yen, Randy Okenka, Keith Confer. Carl McQuillen Racing., Wayne State University; Dr. Ming-Chia Lai, Dr. Xingbin Xie, Atsushi Matsumoto, Yi Zheng

This material is based upon work supported by the Department of Energy under Award Number DE-FC26-07NT43270.

DEFINITIONS/ABBREVIATIONS

abdc
after bottom dead center

ASTM
American Society for Testing and Materials

atdc
after top dead center

BDC
Bottom Dead Center

bbdc
before bottom dead center

BMEP
Brake Mean Effective Pressure (KPa)

BSFC
Brake Specific Fuel Consumption (g/KW Hr)

btdc
before top dead center

cad
crank angle degrees

CA50
Crank Angle of 50% Burn duration

CFR
Cooperative Fuels Research

COV
Coefficient of Variation (IMEP)

CR
Compression Ratio

DI
Direct Injection

DICP
Dual Independent Cam Phasing

EA
Engine Averaged

ECO
Engine out Carbon Monoxide (%)

EGR
Exhaust Gas Recirculation

EHC
Engine out Hydrocarbons (ppm)

EIVC
Early Intake Valve Closing

ENOx
Engine out Nitrogen Oxide (ppm)

E02
Engine out Oxygen (%)

FMEP
Friction Mean Effective Pressure (KPa)

FSN
Filter Smoke Number

FTP
Federal Test Procedure

GDi
: **Gasoline Direct Injection**

HLA
: **Hydraulic Lash Adjuster**

IMEP
: **Indicated Mean Effective pressure (KPa)**

LHV
: **Lower Heating Value (KJ/g)**

LIVC
: **Late Intake Valve Closing**

LIVO
: **Late Intake Valve Opening**

MAP
: **Manifold Absolute Pressure (KPa)**

MBT
: **Minimum spark advance for Best Torque**

MPFI
: **Multi -Port (Point) Fuel Injection**

MPG
: **Miles Per Gallon**

NMEP
: **Net Mean Effective Pressure (KPa)**

NVH
: **Noise, Vibration, Harshness**

PE
: **Power Enrichment**

PFI
: **Port Fuel Injection**

RFF
: **Roller Finger Follower**

RON
: **Research Octane Number**

SOI
: **Start of Injection**

SI
: **Spark Ignited**

TDC
: **Top Dead Center**

VVA
: **Variable Valve Actuation**

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ISSN 0148-7191

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