

Development of the Combustion System for a Flexible Fuel Turbocharged Direct Injection Engine

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ABSTRACT

Gasoline turbocharged direct injection (GTDI) engines, such as EcoBoost™ from Ford, are becoming established as a high value technology solution to improve passenger car and light truck fuel economy. Due to their high specific performance and excellent low-speed torque, improved fuel economy can be realized due to downsizing and downspeeding without sacrificing performance and drivability while meeting the most stringent future emissions standards with an inexpensive three way catalyst.

A logical and synergistic extension of the EcoBoost™ strategy is the use of E85 (approximately 85% ethanol and 15% gasoline) for knock mitigation. Direct injection of E85 is very effective in suppressing knock due to ethanol's high heat of vaporization - which increases the charge cooling benefit of direct injection - and inherently high octane rating. As a result, higher boost levels can be achieved while maintaining optimal combustion phasing giving high thermal efficiency. However, due to their different properties, optimization of a combustion system for both regular grade (91 RON) gasoline and E85 is non-trivial.

This paper describes the initial design and development of a new combustion system for a flexible fuel (gasoline and E85) turbocharged direct injection engine. The development process utilized a boosted single-cylinder engine in conjunction with a second boosted single-cylinder engine with optical access which was used to provide insight into fuel/air interactions and resultant combustion performance.

Using this approach a robust combustion system was developed to meet targets for both gasoline and E85 operation before multi-cylinder hardware was available.

INTRODUCTION

Over the past twenty years, significant progress has been made with gasoline direct injection technology [1,2]. Considerable synergy can be realized by combining gasoline direct injection, turbocharging and variable cam timing. Gasoline turbocharged direct injection (GTDI) engines comprising this combination of technologies have become recognized as one of the most promising options for fuel economy improvement [3,4,5,6,7,8,9,10,11,12,13,14,15].

Downsizing and downspeeding benefits enabled by this technology can provide double digit fuel efficiency gains at significantly lower cost than competing technologies such as diesel or hybridization, therefore interest in GTDI has escalated driven by the current focus on fuel efficiency and fuel economy legislation.

With increased concerns about global warming, energy security and limited availability of conventional fossil fuels, interest in biofuels such as E85 has increased. The 2007 Energy Independence and Security Act (EISA) mandates an increase in US ethanol consumption from the 2007 level of 9 billion gallons per annum to 31 billion gallons per annum by 2022. The Act could provide more widespread supply of E85, but customer demand for E85 must be created as the cost difference between gasoline and E85 currently does not offset the difference in heating value, so the cost per mile of

operating a current flex fuel vehicle on E85 is higher than for gasoline. There are significant benefits to be gained from operating a turbocharged direct injection engine on E85 which can help offset the lower heating value disadvantage [16,17,18].

Direct injection gasoline helps mitigate knock as in-cylinder fuel evaporation cools the charge air. When E85 is used instead of gasoline, the charge cooling benefit is increased proportionately as the latent heat of vaporization (kJ per MJ of heating value) is approximately four times higher than for gasoline. Also the octane number of E85 is higher than that of premium grade gasoline. These two effects make it possible for a turbocharged direct injection engine to operate knock free at high compression ratio and at high BMEP [19]. This provides significant efficiency gains which helps offset the reduced heating value of E85 and provides significant performance increase [20].

In order to realize the maximum fuel economy and performance benefits in a robust and durable turbocharged DI engine, a systematic combustion system development approach is required. Operation on E85 provides further synergies but also adds another dimension to the development challenge. In this paper, development and optimization of the combustion system for a flex fuel engine are presented covering combustion system design and single cylinder and optical engine development. Results are presented showing the optimization of engine operation at part and full load with regular gasoline and E85 and differences in operation and requirements are contrasted for the two fuels.

SPECIFICATIONS

The combustion system is a 4-valve per cylinder pent roof high tumble design with variable cam timing for both intake and exhaust valves. The engine will employ split injection in conjunction with a shallow piston bowl for rapid catalyst heating as a cold start emissions reduction strategy. Each cylinder has both DI and PFI fuel injectors to support a dual fuel concept whereby under most conditions gasoline is delivered through the PFI injectors but then E85 is delivered through the DI injectors at higher load in the quantity required to mitigate knock [7, 19]. This concept was studied during this program in addition to single fuel DI only operation and will be reported in detail in a separate paper to be published at a later date. The DI injector is side mounted for a number of reasons including more favorable thermal conditions for the injector which will not be flowing fuel during part load operation for the dual fuel concept. Turbochargers are utilized to generate boost pressure to improve specific torque and power output, and are appropriately sized for fast transient response.

Figure 1 shows two performance curves for the engine. For gasoline, peak BMEP was 17 bar BMEP for 91 RON US

regular grade gasoline. Peak performance for E85 was 27 bar BMEP.

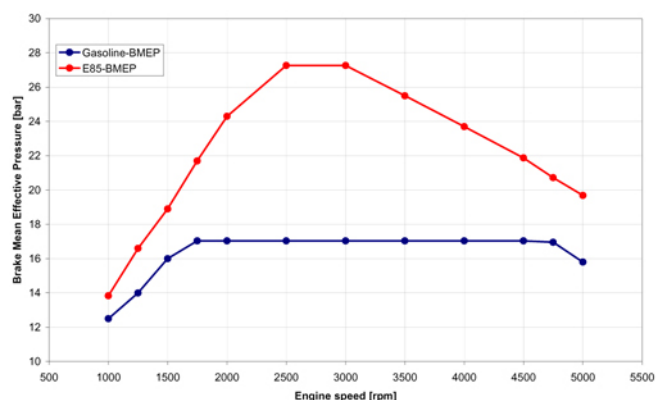


Figure 1. Performance targets

A single-cylinder research engine was used for initial performance and emissions work as this allowed the combustion system to be developed months before the first multi-cylinder engine was available. It was also more cost and time effective to use a single cylinder engine to define the major combustion specifications such as tumble ratio, injector spray, piston bowl design and compression ratio as the number of prototype components required for each build was reduced and rebuilds were quicker. Following single cylinder development, a batch of multi-cylinder engines were built with confidence that further combustion system changes would not be required and multi-cylinder development focused on optimization of valve events, fine tuning of compression ratio and turbocharger matching.

Multi-cylinder engine FMEP was estimated to provide NMEP levels that could be used for single cylinder testing. FMEP for a single cylinder engine is generally higher than for a multi-cylinder engine due to added friction from a balance shaft system and timing drive friction which is carried by a single rather than multiple cylinders. Controlling to NMEP set points also provides consistency for testing and development if friction levels change from one build to the next due to variations in timing belt tension, etc.

Single cylinder development focused on full load operation at low, mid and high speed, part load operation at 1500 rpm 2.62 bar BMEP and the operating point that is typical for catalyst heating. These development points are shown in Table 1.

<table 1 here>

COMBUSTION SYSTEM DEVELOPMENT

COMBUSTION CHAMBER

Figure 2 shows the layout of the combustion system as packaged for the single cylinder engine. The DI injector is side mounted between the intake valves and the spark plug is at the center of the combustion chamber slightly tilting towards the intake side. Also shown is a sleeve on the intake and exhaust bridge for a pressure transducer for cylinder pressure measurement, which is used to calculate NMEP, heat release, and other combustion parameters. The PFI injector is located in the intake manifold targeting the intake valves. The targeting of the injector was optimized using an endoscope and simulated air and fuel delivery conditions on a test rig at AVL. This was done to minimize port wall wetting to improve transient air-fuel ratio control for emissions reduction. Shown in Figure 3 is a CAD model of the piston. The piston has a shallow bowl to support split injection for catalyst heating.

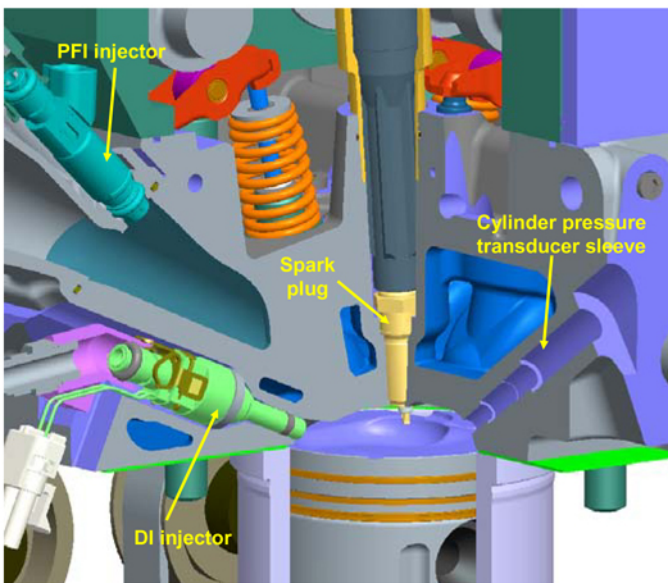


Figure 2. CAD model of the combustion chamber

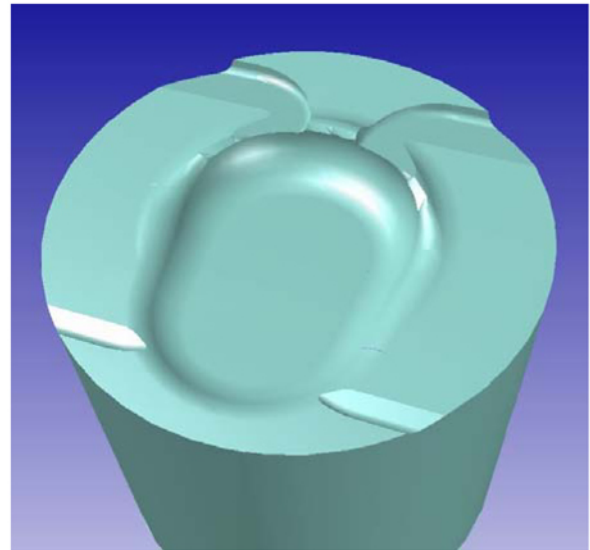


Figure 3. CAD model of the initial piston design

PORT DESIGN

R&D experience has shown that increased tumble can provide significant combustion benefits for a GTDI engine, and in the past two decades of GTDI development, tumble levels have been increasing continuously. High tumble provides a number of benefits for a GTDI engine [21]:

1. High tumble helps fuel/air mixing which helps reduce CO and smoke emissions, improving efficiency and combustion stability.
2. In a side injector engine, strong tumble air motion can help reduce or eliminate liner wetting by deflecting the fuel spray from the liner.
3. High tumble has been shown to provide octane benefits allowing more advanced combustion phasing. This provides efficiency benefits due to more optimum combustion phasing and improves combustion stability and combustion system robustness. Advanced combustion phasing also reduces exhaust temperature reducing fuel enrichment requirement to maintain turbine inlet temperature limits, and improved thermodynamic efficiency reduces boost requirements to achieve a target torque level.
4. High tumble provides increased dilution tolerance due to faster burn rates which allows increased efficiency due to ability to use higher internal or external EGR at part load or cooled external EGR for exhaust temperature and fuel enrichment reduction at full load.
5. Differences in tumble for individual cylinders of a multi-cylinder engine resulting from manufacturing variability can lead to burn rate and knock borderline differences between cylinders causing one cylinder to knock before the others which limits spark advance for the engine even though the other cylinders are operating knock free. With high tumble

ports, manufacturing variability results in lower percentage tumble variation between ports so all cylinders are more closely matched in burn rate and knock borderline ignition timing, which allows more advanced ignition timing.

Figure 4 shows the high tumble intake ports that were designed for this engine. These ports were designed not only to provide a high tumble ratio, but also to provide a centered tumble field within the cylinder that would provide an even turbulence distribution throughout the cylinder for symmetrical flame propagation to mitigate knock. The ports and throat cut were designed to minimize tumble variation from casting core shift to provide a similar tumble level and flow field for all cylinders. The top side of the intake flange has a bulge to accommodate the fuel spray from the PFI injector which is located in the intake manifold, and the bottom side of the flange has a relief to accommodate the DI injector and fuel rail connection.

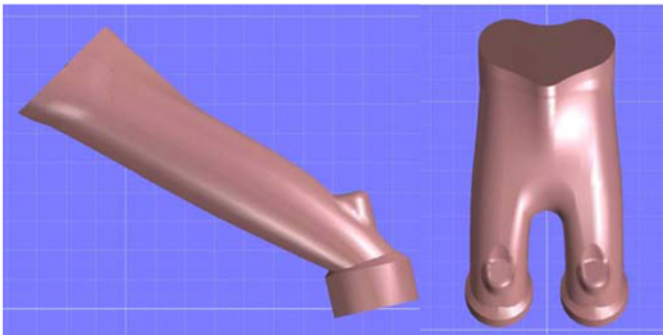


Figure 4. Side and front views of the intake ports

PORT DEVELOPMENT

In order to ensure that the tumble generated by the ports was not only of the magnitude required, but symmetric and centered, port development was conducted in an LDA port development laboratory at AVL. LDA measurement provides velocities of the air stream (seeded with particles) in the vertical direction at 250 points across the bore area, measured at a distance equivalent to half the bore diameter down from the top of the liner [23]. This provides a complete measurement of the air motion field generated by the port, and by taking measurements at various valve lifts, the instantaneous tumble and a cycle average tumble ratio can be derived for a given valve lift curve corresponding to the intake cam profile. This also provides a qualitative and quantitative assessment of the symmetry of the air motion field, determination of the location of the tumble field axis, and an estimate of flow coefficient. A schematic of the AVL Laser Doppler anemometry (LDA) rig is shown in Figure 5.

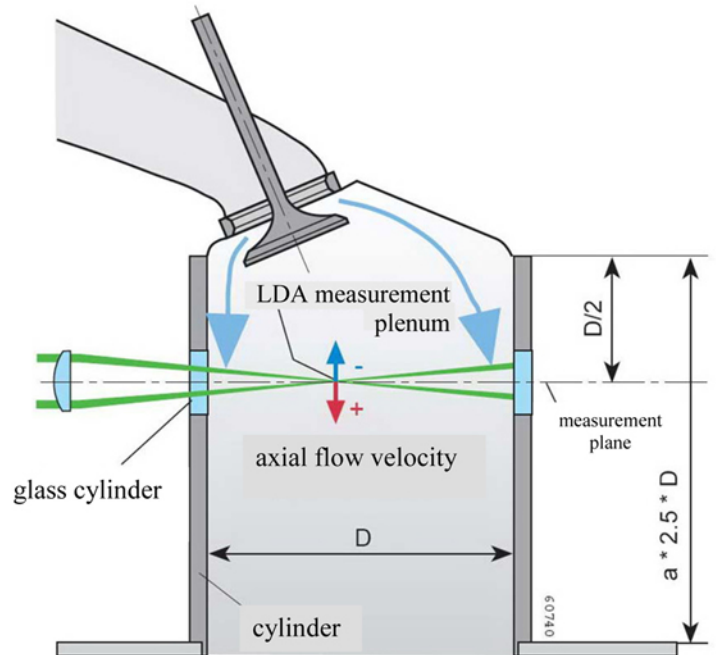


Figure 5. Schematic of AVL Laser Doppler anemometry (LDA) rig

Figure 6 shows LDA measurement results of the in-cylinder tumble flow at different valve lifts from the tumble port with tumble ratio of 1.93. Red dots denote the charge air at the measuring element flowing downwards with high velocity, while blue dots denote air flowing upwards (towards the cylinder head) with high velocity. It is shown that a tumble flow field with good symmetry is established by the time the intake valves are wide open. On the rig, at lower valve lifts small differences in valve lift can produce an asymmetric tumble field pattern. Also shown in the figure are angles of the tumble plane relative to the cylinder center plane between intake and exhaust, which quantifies the symmetry of tumble flow. At the time when intake valves are wide open, the angle of the tumble plane is only 4 degrees rotated clockwise from the cylinder center plan, indicating good symmetry of the flow field.

<figure 6 here>

Tumble ratio is defined as

$$TR = \frac{1}{\pi} \int_{\alpha=0}^{\pi} \frac{\omega_a}{\omega_E} \left(\frac{c(\alpha)}{c_m} \right)^2 d\alpha$$

(Equation 1)

Where:

TR	= Tumble ratio
α	= Crank angle
$c(\alpha)$	= Instantaneous piston speed
c_m	= Mean piston speed
ω_a	= Angular velocity of charge air in the combustion chamber
ω_E	= Angular velocity of engine crankshaft

As shown in [Equation 1](#), tumble ratio is the cycle integral of instantaneous tumble ratios at each crank angle, weighted by the instantaneous piston speed.

Flow coefficients (μ_s) at each valve lift are also measured and shown in [Figure 6](#). At the highest valve lift the flow coefficient is 0.559. Shown in [Figure 7](#) are the mean flow coefficients and tumble ratios of the intake ports tested during the program compared to other engines in the AVL engine database. This shows that the port designs are very efficient as they provide above average flow capacity for the high levels of tumble generated.

<[figure 7](#) here>

INJECTOR SPRAY DESIGN

Injector spray targeting is one of the most challenging design and development tasks for a new DI combustion system. The fuel spray must be designed so that it provides excellent homogenous mixture preparation for part and full load operation while the same spray must target the piston bowl to support split injection for catalyst heating and produce the required level of stratification in this mode. Under part and full load operation, piston impingement must be minimized to maximize charge cooling benefits and to avoid smoke, and under all conditions, bore washing must be avoided to reduce oil dilution, smoke and to avoid cylinder system wear.

Flow rate must be specified so that there is time for the fuel to vaporize and mix under the highest flow rate conditions (usually peak power), however the flow rate must be low enough so that under the lowest fuel flow conditions the injector will continue to operate within its linear flow range. The minimum fuel flow condition is unloaded idle during evaporative canister purge.

Injector flow rate can influence individual spray plume cone angle, droplet size and fuel spray penetration, which all influence spray targeting and mixture preparation.

Two alternative 7-hole spray designs were designed and procured for performance and emissions development. These were designated DT1 and DT2, which had spray patterns as shown in [Figures 8](#) and [9](#) respectively. These designs were based on prior design and development experience with similar combustion system layouts (injector placement, installation angle, tumble level and bore size) from prior

projects and the learning from optical and conventional combustion development on those engines was incorporated into these designs. [Figure 10](#) shows the DT1 spray layout in the combustion chamber.

The difference between DT1 and DT2 injectors is the arrangement of fuel jets. Seven fuel jets for the DT1 injector are arranged in a 2-3-2 pattern, while those for DT2 injector are 1-3-3. Also, the top two rows of sprays are lower for the DT1 injector, compared to those for DT2 injector.

<[figure 8](#) here>

<[figure 9](#) here>

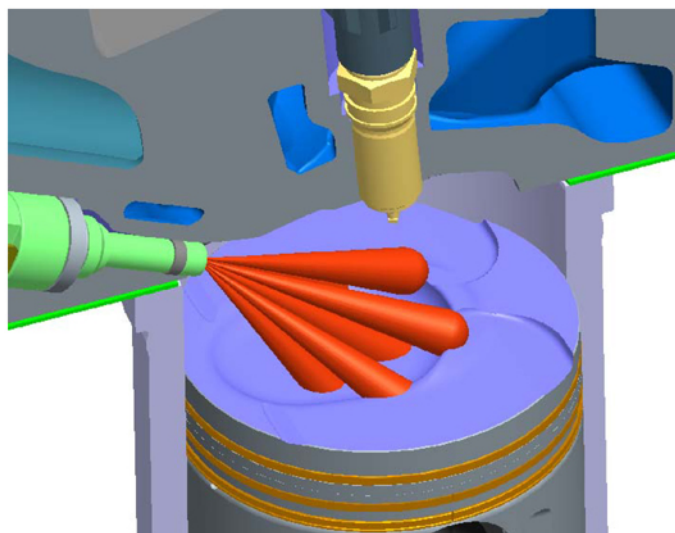


Figure 10. DI fuel spray targeting

SINGLE CYLINDER DEVELOPMENT

TEST ENGINE

Performance and emissions development was conducted on an AVL 580 single-cylinder research engine, shown in [Figure 11](#). Boost air was supplied externally by a test cell system, which was controlled for temperature, pressure, and humidity. The static and dynamic back pressure in the exhaust of the turbocharged multi-cylinder engine was emulated by orifices in the exhaust pipe set to levels defined by 1D simulation.

Three uncooled GU21C piezo-electric high-speed transducers were installed on the engine to monitor intake port, in-cylinder, and exhaust port pressures. All three pressure traces were captured on an Indi-master/Indicom system, and then used as input to AVL Gas exchange and Combustion Analysis (GCA) which among other parameters provided an estimate of residual gas fraction. NMEP and knock intensity were evaluated based on the in-cylinder pressure trace.

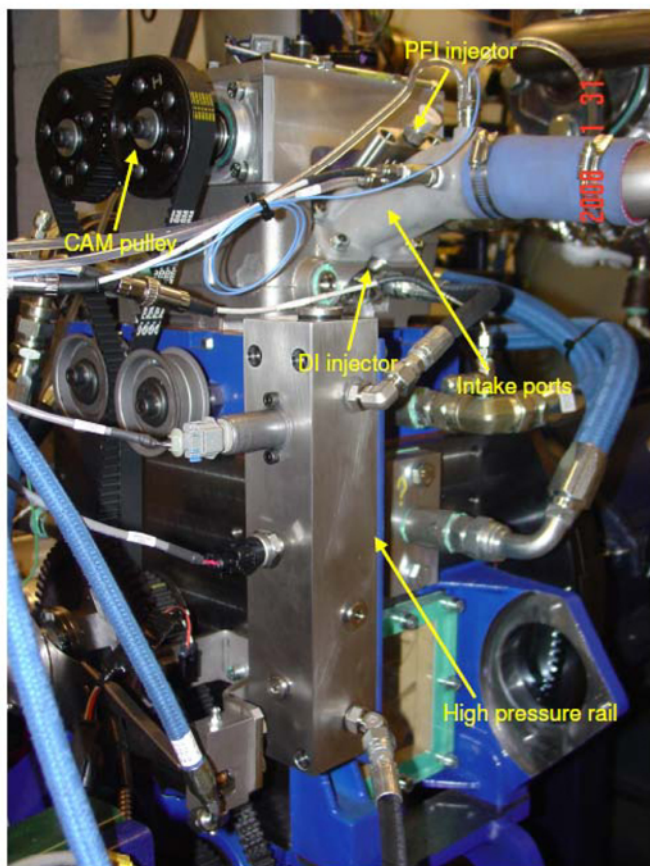


Figure 11. Single cylinder setup in the test cell

TUMBLE RATIO OPTIMIZATION

Fuel economy, emissions, octane requirement, combustion stability and NVH can all be influenced by tumble ratio and all these factors have to be considered when determining the optimum tumble ratio requirement. However, this is one of the areas where different solutions could be reached for gasoline compared to E85, and operation on both fuels had to be considered when selecting tumble ratio.

The target tumble ratio for this engine was between 1.75 and 2.00, however, the initial port as designed and packaged into the single cylinder head had a tumble level of 2.38. Due to concerns over initial data with this high tumble ratio showing very high rates of pressure rise and high combustion noise at high BMEP operation with E85, the intake ports of a second cylinder head were modified to produce much lower tumble for testing. The tumble ratio of this second cylinder head was measured at 1.13. Results for these two tumble ratios are shown in [figure 12](#).

<[figure 12 here](#)>

As shown in [figure 12](#), maximum rate of pressure rise with 2.38 tumble was close to 10 bar/deg at 25.7 bar NMEP. This

is due to a combination of very high load operation at MBT ignition (enabled by knock free operation due to the increased octane number and charge cooling benefits of E85 and high cylinder pressure capability) and high burn rates due to the high tumble ratio. Combustion noise estimated by an AVL combustion noise analyzer (which provides an estimate of combustion noise based on cylinder pressure measurement and a user selectable transfer function) showed a combustion noise level of 91 dB at this condition - which is comparable to a diesel engine. With lower tumble, rates of pressure rise and combustion noise reduced, however CO increased - indicative of poorer mixing - and NSFC increased. Therefore higher tumble appeared beneficial to mixture preparation and fuel economy at high load. High fuel flow rates are required with E85 at high NMEP, so high tumble helps fuel/air mixing especially at lower engine speeds where the effective air motion is lower. As rates of pressure rise are also a function of advanced combustion phasing which increases cylinder pressure and burn rates, the effect of combustion phasing was investigated with the 2.38 tumble ratio head. Ignition timing was retarded from MBT to a timing where rates of pressure rise were similar to those measured with the 1.13 tumble head. This required only 2 degrees of ignition retard leading to less than 1% increase in NSFC which remained much lower than for the low tumble case. This showed that it is more beneficial to run high tumble but retard ignition timing to control noise than to reduce tumble level.

<[figure 13 here](#)>

[Figure 13](#) compares full load operation on 91 RON gasoline with high and low tumble. High tumble provides an octane benefit as combustion phasing is 2 - 3 degrees advanced at detonation borderline (DBL) for tumble of 2.38 compared to the 1.13 tumble case. With the low tumble, COV of NMEP is above the 2.5 % limit, primarily due to retarded combustion phasing. With high tumble, acceptable stability was achieved all the way up to 19.5 bar NMEP. This illustrates a key difference in requirements for E85 and gasoline - while high tumble was beneficial at full load with E85, it is essential at full load with gasoline as combustion stability targets cannot be met with low tumble.

The two other impacts of retarded combustion phasing due to increased octane requirement with low tumble were increased fuel consumption and increased exhaust temperature. Exhaust temperature would lead to a further increase in fuel consumption at high speed full load as more fuel enrichment would be required for turbine inlet temperature control.

<[figure 14 here](#)>

At part load, although high tumble showed a benefit in reduced COV of NMEP, combustion stability was also acceptable with low tumble, even under retarded IVC conditions ([figure 14](#)). 1.13 tumble ratio is still high

compared to NA engine tumble levels. High tumble provided benefits at part load in the form of reduced CO emissions due to improved mixture preparation which also provided reduced fuel consumption. These benefits would appear to more than compensate for increased heat transfer losses due to the higher air motion.

FULL LOAD COMPARISON GASOLINE VERSUS E85 WITH 1.93 TUMBLE RATIO

Following port development for the multi-cylinder engine aimed at reducing tumble to within the 1.75 - 2.0 target range and improving flow capability, a confirmation flow box for the multi-cylinder intake port was manufactured and tested. The tumble level of this port was measured at 1.93. Due to the negative impact on mixture preparation and octane requirement that had been observed when tumble was reduced from 2.38 to 1.13, the intake port of the single cylinder head was modified to match the multi-cylinder port, and performance on 91 RON gasoline and E85 was evaluated.

Figures 15a and 15b compare full load operation on 91 RON gasoline and E85, with 1.93 tumble ratio and 9:1 compression ratio. Ignition timing was set to MBT/DBL or to control cylinder pressure to a limit of 150 bar (mean + 3 σ) and equivalence ratio was set at $\lambda=1$ or rich as required to maintain an exhaust temperature of 950°C. NMEP was set to the target levels for the engine. With 91 RON gasoline, CA50 at 2000 rpm 18.5 bar NMEP was close to 30 degrees and COV of NMEP was close to the limit of 2.5%, any further increase in NMEP would result in further retardation of combustion phasing and a corresponding increase in COV of NMEP, therefore 18.5 bar NMEP was the maximum load that would give acceptable combustion stability at 9:1 compression ratio with 91 RON fuel. This showed that 1.93 tumble was acceptable for 91 RON operation, however higher tumble may have provided a small octane improvement which would improve stability and/or allow increased NMEP. The impact of reduced tumble ratio on mixture preparation was studied in detail on the optical engine and is presented later in this paper.

<figure 15a here>

<figure 15b here>

There is a significant difference between 91 RON (regular grade) and 98 RON (premium gasoline), at 18.5 bar NMEP, 98 RON gasoline would allow several degrees more ignition advance with improved combustion stability, which would therefore allow much higher NMEP operation. Likewise, knock free operation on E85 allows very high NMEP operation as combustion stability is excellent due to advanced

combustion phasing. The limiting factors for performance on E85 are thermal and mechanical loading on the base engine and airflow capability which is defined by the turbocharger match. For this engine, maximum BMEP for the multi-cylinder engine on E85 was set at 27 bar BMEP between 2500 - 3000 rpm, the single cylinder operated at 26, 27 and 22 bar NMEP at 2000, 3500 and 4750 rpm respectively. At all of these conditions the engine ran knock free. At 2000 rpm the engine ran with MBT ignition, but at 3500 rpm ignition timing was retarded slightly to control cylinder pressure.

With E85 the engine operated with an equivalence ratio of $\lambda=1$ but with gasoline, even though the NMEP was significantly lower, enrichment was required at 3500 rpm and above to control exhaust temperatures.

While fuel consumption was higher for E85 than gasoline, the increase was less than the difference in heating value between the two fuels due to combustion phasing and $\lambda=1$ operating benefits with E85. At 3500 rpm E85 NSFC was only 12% higher than for gasoline despite the 33% reduction in heating value. On a thermal efficiency basis E85 showed 34% higher net thermal efficiency than gasoline at 3500 rpm, and at 2000 rpm net thermal efficiency was 41%. This could be increased further by increasing compression ratio if the engine were to be optimized for E85 operation rather than being required to run on either gasoline or E85 as a flex fuel application. There would be some trade-off between thermal efficiency benefit from compression ratio increase and reduced efficiency due to combustion phasing retard for cylinder pressure control at full load as higher compression ratio would increase cylinder pressure, however increased compression ratio would provide fuel efficiency benefits under part load operating conditions which is more critical for most drive cycle and real world conditions.

RESIDUAL FRACTION IMPACT AT FULL LOAD

Trapped residual gas can increase the propensity to knock as it raises unburned gas temperature. Increased valve overlap can help scavenge residual gas from the cylinder as intake pressure is higher than exhaust pressure under boosted conditions at lower engine speeds, however on I4 or V8 engines, overlapping exhaust pulses can lead to backflow from the exhaust to the cylinder and from the cylinder to the intake manifold, as exhaust blowdown from the next cylinder in the firing order sequence will increase exhaust pressure during valve overlap.

Figure 16 shows the impact of residual gas at 2000 rpm, 18.5 bar NMEP. When residual gas fraction was increased (by changing cam timing) from 1.0% to 2.6% (determined from GCA), knock-limited combustion phasing retarded by almost 10 degrees. As a result, combustion stability deteriorated, fuel consumption increased by almost 7% and exhaust

temperature increased by 110°C which would lead to increased fuel enrichment requirements. With E85, due to its anti-knock properties, residual gas content is not expected to lead to knock-limited combustion.

<figure 16 here>

SPLIT INJECTION FOR CATALYST HEATING

Following a cold start, it is important to heat the catalyst rapidly as the majority of drive cycle emissions are produced while the catalyst is warming up and operating at low efficiency. Also, during catalyst heating, engine-out emissions must be reduced to a minimum. Catalyst heating is achieved by using a split injection strategy in a direct injection engine.

Figure 17 shows a comparison of catalyst heating mode operation for DT1 and DT2 injectors. Following optimization of the primary and secondary injection timings for each injector the fraction split between the primary and secondary injection pulse was optimized. These results show improvements in all areas for the DT1 injector with improved combustion stability, reduced emissions and lower fuel consumption. The differences between the injection split fraction for the two injectors relates to poorer stratification with the DT2 injector which favors more fuel injected in the first pulse to produce a richer background homogenous mixture.

<figure 17 here>

OPTICAL ENGINE DEVELOPMENT

DI combustion systems offer a variety of opportunities, but at the same time have to meet a number of requirements, as mentioned previously:

- At part load, a well mixed homogenous charge is required, otherwise fuel energy will be lost in the form of increased CO or HC emissions.
- During catalyst heating a rich stratified mixture must be created at the spark plug with low cyclic variability and low engine out emissions.
- At full load, a well mixed homogeneous charge is required to mitigate knock and fuel impingement on the piston must be minimized to maximize the DI charge cooling benefit.
- Under all conditions, liner impingement must be eliminated or reduced to a minimum to avoid liner wear, smoke and oil dilution.

With conventional engine development some of these issues can be inferred from emissions, fuel economy or cylinder

pressure results, however other potential issues such as liner wetting are difficult to identify with conventional engine testing. AVL routinely uses a single cylinder engine with optical access for GTDI development programs as this helps identify and resolve any potential issues early on in a program. This process has a number of advantages compared to CFD analysis as the results require no correlation, the process does not rely on assumptions or fuel spray models (which are difficult to obtain), and the process accounts for cyclic variability rather than providing a mean cycle result which has limited value for a gasoline engine.

For the current program, E85 operation increased the risk of wall wetting due to increased fuel flow that would be required due to the lower heating value and higher BMEP target. Fuel spray penetration increases in proportion to injection pulse width, so the risk of liner wetting and resultant liner wear and oil dilution increases with higher fuel flow requirements.

The optical engine is a single cylinder engine with the upper part of the liner made from quartz glass, providing valuable insight into the fuel injection and combustion processes. Figure 20 shows the combustion chamber of the optical engine. Laser Induced Fluorescence (LIF) is used to observe the fuel-air mixing process and flame photography is used to observe combustion¹. The engine is operated on iso-octane (with 10% 3-pentanone as a tracer for LIF measurement) rather than regular gasoline in order to reduce deposits on the glass liner which deteriorate visual observation. For LIF an EXCIMER laser with KrF is operated at a wavelength of 248 nm.

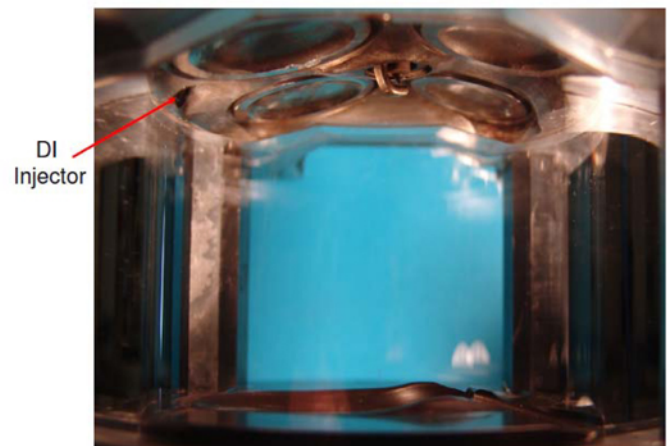


Figure 20. Combustion chamber of the optical engine

At each measurement point, 35 images are captured from consecutive cycles taken at the same crank angle timing. These single images (fuel spray or combustion flame) are

¹The compression ratio on the optical engine is lower than the development engine, however the difference in the compression ratio has minimal impact on the optical engine cycle.

statistically evaluated, replacing the fuel (or flame) intensity by the probability to find fuel of low or high density (or sooting versus premixed (no soot) flames for flame images), see [figure 21](#).

<[figure 21](#) here>

[Table 2](#) lists the operating conditions evaluated on the optical engine. Low speed full load conditions were selected for evaluation as mixing can be more problematic at these conditions due to lower in-cylinder air motion combined with high fuel flow rates. At full load, the engine operated with intake manifold conditions matching boosted multi-cylinder conditions, however ignition timing was retarded to reduce cylinder pressure to protect the glass liner. This has no effect on the air/fuel mixing process which is the focus of the investigation.

<[table 2](#) here>

Optical engine testing was run in parallel with conventional single cylinder engine development so that the optimum hardware and control settings defined by single cylinder testing could be evaluated on the optical engine and vice versa.

Fuel/air interaction and fuel penetration results were similar for both DT1 and DT2 injectors at part and full load, however differences were observed in catalyst heating modes which are described in detail below.

Statistical images of fuel distribution for the first injection in catalyst heating mode at different crank angle degrees are shown in [Figures 22](#) and [23](#) for DT1 and DT2 injectors respectively. The upper spray of the DT2 injector separates from the main spray and results in increased liner wetting. At the start of the second injection (75 degrees BTDC), mixture homogeneity is similar for both injectors.

<[figure 22](#) here>

<[figure 23](#) here>

Statistical images of fuel spray distribution for the second injection at different crank angle degrees with both injectors are shown in [Figure 24](#) and [25](#). The top spray from the DT2 injector - which again separates from the main spray - overshoots the piston bowl and impinges the rear of the piston. Due to this poor targeting, stratification is reduced leading to a leaner mixture at the spark plug with the DT2 injector which resulted in reduced combustion stability. The first two frames of [figure 24](#) (70 and 65 ° CA BTDC) show what looks to be fuel to the right of the spark plug (circled in blue) where fuel would not be expected at these timings. This is an optical reflection of the fluorescing fuel spray onto the

quartz liner and therefore has no meaning in terms of the results.

<[figure 24](#) here>

<[figure 25](#) here>

[Figures 26](#) and [27](#) show the statistical flame images at 40 and 60 degrees ATDC for both injectors in catalyst heating mode. Due to the improved fuel stratification of DT1:

- The engulfed flame volume at the same time is larger
- Cyclic variability is significantly reduced

The slightly increased soot flame probability of DT1 is not significant, as tailpipe smoke measurements showed very low smoke levels.

Based on these findings and similar advantages measured on the single cylinder engine the DT1 injector was chosen.

<[figure 26](#) here>

<[figure 27](#) here>

<[figure 28](#) here>

[Figure 28](#) shows statistical flame images at 30 and 90 deg ATDC in catalyst heating mode on gasoline and E85. With gasoline, sooting flames were measured, especially in the region close to the piston where fuel impingement would have created a fuel film. When the engine was operated on E85 at the same conditions, the sooting flames disappeared and only premixed combustion flames were observed. As ethanol is oxygenated, oxygen contained in the fuel reduces or eliminates soot formation in rich regions. Also ethanol is a single component fuel with a single boiling point of 78.3°C. E85 is a mixture of ethanol and high volatility gasoline components (to aid cold starting). Gasoline is a multi-component fuel with a range of boiling temperatures for its various constituents, typically with boiling points up to 180°C. Therefore it is likely that more complete evaporation of an E85 fuel film on a hot piston crown will occur than for gasoline and that the heavier components that do not evaporate as readily are the source of sooting flames with gasoline.

<[figure 29](#) here>

[Figure 29](#) shows statistical fuel images at full load for 2000, 1500 and 1000 rpm with a tumble ratio of 2.38. At 2000 rpm there is a beneficial interaction of the high tumble air motion with the fuel spray, effectively deflecting the fuel spray away from the exhaust side of the liner and eliminating bore washing. Early injection is preferable as the air motion decays at later crank angles. However injection timing earlier