

A Study of Gasoline-Alcohol Blended Fuels in an Advanced Turbocharged DISI Engine

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ABSTRACT

This work was concerned with evaluation of the performance and emissions of potential future biofuels during advanced spark ignition engine operation. The fuels prepared included three variants of gasoline, three gasoline-ethanol blends and a gasoline-butanol fuel altogether covering a range of oxygen mass concentrations and octane numbers to identify key influencing parameters.

The combustion of the fuels was evaluated in a turbocharged multi-cylinder direct fuel injection research engine equipped with a standard three-way catalyst and an external EGR circuit that allowed use of either cooled or non-cooled EGR. The engine operating effects studied at both part and boosted high load conditions included fuel injection timing and pressure, excess air tolerance, EGR tolerance and spark retard limits. A number of blends were also mapped at suitable sites across the European drive cycle under downsized engine conditions. Relative in-vehicle fuel economies were then determined via drive cycle simulation and compared to a naturally aspirated gasoline PFI engine.

INTRODUCTION

Evolving emissions legislation and concerns for diminishing fuel reserves continue to prompt the automotive industry to seek alternative modes of engine operation. In recent years, European CO₂ emissions

targets have been met through increased diesel sales. However, the distillation of crude oil results in high proportions of both gasoline and diesel fuel (1) and ultimately this has resulted in Europe being "diesel lean". In order to meet future global emissions goals, in the short term it will be necessary to improve the fuel consumption of the gasoline engine and in the longer term source sustainable alternatives to crude oil.

Alcohol based fuels may present one viable renewable solution, with the potential to be used in a near CO₂-neutral manner through efficient conversion of biomass. The idea of using alcohol as an automotive fuel is not new (2-4), but only recently have such fuels begun to attract significant worldwide attention. First generation biofuels for Spark Ignition (SI) applications have so far largely been based on gasoline-ethanol blends, where current fuel quality standards typically allow between 5-10% inclusion of ethanol within an existing gasoline pool. The main exception to this is Brazil, where alcohol fuels largely produced from sugar cane are widely available in anhydrous "gasohol" up to neat ethanol form (5). Elsewhere, gasoline containing up to 85% ethanol ("E85") has emerged as a passenger car fuel, but so far mainly as a niche product, with flex-fuel operation still required (6). It is still highly questionable as to whether mass conversion to fuels containing high concentrations of biofuel is practical. Production of biofuel from feedstock is limited on a global basis. Next generation processes are therefore currently being investigated that may allow such fuels to be efficiently mass-produced from alternative sources such as cellulose, algae or even recovered waste (7, 8).

The “lower” alcohols of reduced carbon count, such as methanol and ethanol, exhibit high latent heat of vaporization and favourable anti-knock rating. From an engine performance stance, this makes such fuels attractive for many modern SI engine applications. For example, Turner et al. (9) recently demonstrated significant improvement in peak torque (~10% c.f. 95RON gasoline) and power (~14%) when using commercial E85 fuel under supercharged engine conditions. These workers also demonstrated how pre-compressor injection of such fuel could be used to reduce compressor work in such a boosted engine. Nonetheless, the relatively low energy density of the lower alcohols must be considered when blended in high concentrations. Furthermore, gasoline-ethanol blends are known to exhibit azeotropic behaviour, with profound effects on the vaporisation and thermodynamic properties of the blend. Kar and co-workers (10) recently performed cycle resolved in-cylinder temperature measurements and reported that blends with less ethanol content (0-30%) tended to evaporate more readily while higher concentrations (>50%) with reduced vapour pressure did not and hence exhibited reduced “evaporative power”.

Until recently, flex fuel vehicles were powered by Port Fuel Injection (PFI) units. Early concepts focused on methanol usage (11, 12) until problems such as corrosion became clear and ethanol was more widely adopted, albeit with reduced thermodynamic potential for some applications and still not without corrosion and infrastructure issues (13-15). In concentrations higher than 85%, the low volatility of ethanol causes problems under cold starting conditions. Consequently, there is currently significant interest in the application of gasoline Direct Injection (DI) for improved cold start performance. For example, Kapus and co-workers (16) recently performed a comparison of E85 and E100 fuel in an optical single cylinder research engine, operating with side direct injection at a cranking speed of 200rpm and with fluids controlled to 20°C. These workers concluded that improved start on ethanol was best achieved using multiple pulse fuel injections during the induction and compression strokes. Stratification via a single injection late in the compression stroke was observed to result in excessive oil film dilution.

The thermodynamic benefits of combining direct fuel injection with turbocharged SI engine operation are well known (17-19) and the enhanced charge cooling and anti-knock rating of the lower alcohols makes such operation an even more attractive proposition. In addition, the adoption of a downsized unit may, partially at least, offset any apparent fuel economy deficit of such reduced energy density fuel. In recent work, Brewster (20) investigated the benefits of combining E100 fuel with turbocharged direct fuel injection operation using a centrally mounted air assisted injector in a multi-cylinder research engine. It was postulated that such an injector could offer improved low temperature starting with fuels of lower volatility. Furthermore, the fuel metering and

fuel delivery could be de-coupled, allowing relatively high fuel flow rates as required for alcohol direct injection at high specific output. With a start temperature of 25°C, ethanol and gasoline showed no appreciable difference in catalyst light-off performance in terms of exhaust gas temperature. At target torque levels representative of current production turbocharged SI engines, ethanol operated with lower boost pressure, optimum ignition timing, lower exhaust temperature, higher thermal efficiency and reduced emissions of CO₂. Furthermore, significant reductions in excess fuelling were possible at higher speeds and loads. That said, for wider spread flex fuel operation it seems clear that such engines would still benefit from alternative methods of charge dilution to fuel enrichment, such as cooled EGR (21-24) or excess air (25, 26).

In an alternative recent study, Bergström et al. (27) reported on the development of a new 2.0litre turbocharged direct fuel injection flex fuel unit. The effects of E85 direct injection on hardware were compared to observations made previously during the development of a similar production PFI flex fuel engine. During E85 DI operation, it was concluded that peak in-cylinder pressures in excess of 120bar would need to be tolerated. Inlet valve seat recession issues were apparent with either mode of injection, but notably worse in the DI case. At first this perhaps seems counter-intuitive, although possibly at least partially associated with elevated peak in-cylinder pressures under DI conditions. Another observation of E85 DI was increased dilution of the oil, with the need for an upgraded breather system to manage high volume flow rates of fuel suddenly evaporating from the crankcase after cold start. Regardless, in terms of durability one main benefit of DI operation was reduced cracking around the cylinder head intake ports, previously associated with the high fuel flow rates and degree of charge cooling within the ports under PFI conditions.

Another factor that must be considered for alcohol DI operation is difference in fuel injector deposits, of which very little data seems to exist within the public domain. Taniguchi et al. (28) investigated the effects of E100 on part and full load performance, emissions and injector deposits in a 3.0litre V6 direct injection engine operating with an elevated compression ratio of 13:1. In brief, all ethanol blends tested (E20-E100) reduced injector deposit formation due to lower nozzle temperatures and the nature of the single component fuel. Other recent studies on component durability effects of gasoline-ethanol blended fuels have also yielded interesting results. Rovai et al. (29) investigated the wear and corrosion in low-pressure fuel pumps specifically designed for gasoline-alcohol engines. These workers concluded that wear due to abrasion was more pronounced when using E22 whereas corrosion mechanisms were more damaging when using increased alcohol content. The increased wear when using E22 was not related to fuel lubricity, which itself was shown to decrease with yet higher ethanol content. Workers at

Delphi (30) recently reported a similar effect when looking at the effects of E85 corrosion on port fuel injector durability, using different levels of contaminants and corrosion inhibitors. It was concluded that, without corrosion inhibitor, chloride levels greater than 3.5ppm could cause severe corrosion to steel. Both contamination by chloride and organic acids raised the corrosion rating substantially.

While the lower alcohols have previously warranted significant interest, “higher” alcohols such as propanol, butanol and pentanol have also been considered for automotive use. From a thermodynamic stance, higher alcohols generally exhibit higher calorific value (and hence better volumetric fuel consumption), better water tolerance, volatility control and lower Reid vapour pressure. However, benefits in knock resistance and latent heat are typically reduced. Possible differences in source to wheel energy consumption should also be considered if such fuels are ever to be produced on mass. Gautum and co-workers (31, 32) compared a wide range of C_1 - C_5 alcohol-blended fuels in a single cylinder research engine. Interestingly, blends with higher alcohols were reported to show degraded knock resistance when compared with neat gasoline. Aldehyde emissions were higher for all alcohol blends, with formaldehyde as the main constituent regardless of the type of alcohol in the blend. Elsewhere, Bata et al. (33) evaluated the performance of various gasoline/butanol splash blends in a 2.2litre naturally aspirated research engine. These workers observed an increase in specific fuel consumption of 6.5% when using 20% butanol, but under limited test conditions. This fuel was concluded to be superior to both methanol and ethanol in terms of higher thermal efficiency and lower specific fuel consumption. In other recent work (34), some of the currently reporting authors also compared splash blends of butanol and ethanol in an optical SI engine under part-load conditions. Addition of 25% ethanol or 25% butanol to *iso*-octane did not affect appreciably the combustion characteristics of *iso*-octane at fixed ignition timing as observed through flame imaging and thermodynamic analysis of the pressure traces for stoichiometric conditions. For lean conditions though, the effect was to marginally slow down the combustion process.

The currently reported work has been concerned with evaluation of potential next generation biofuels in a turbocharged spark ignition engine, previously optimised for use with 98 RON gasoline (18). One objective of this initial collaborative work was to establish which potential future fuels, if any, exhibited synergy with potential future modes of downsized engine operation. Such modes include homogeneous lean or EGR-boosted operation, studied by the authors previously (35, 36) but using gasoline-only fuel.

EXPERIMENTAL ENGINE

The experiments were performed in a turbocharged four cylinder 2.0litre four-valve per cylinder gasoline direct injection research engine. Some general details of the unit are presented in Table 1. The original pent-roofed engine was of naturally aspirated wall-guided design, with the injector located between the two inlet valves and fuel injected towards a bowl in the piston crown. The injectors used in the current work were of identical hole pattern but 5cc/s lower peak flow rate than those used for previous EGR boost work (35, 36). The original spark plugs were also unavailable, with non-tapered electrode parts used in the current work. Otherwise, the ignition system remained unchanged from that optimized for 98RON gasoline use (18, 35, 36).

In raising the output of the engine, the original pistons were replaced with parts including a flat crown. This modification, illustrated in Figure 1, was considered to be acceptable for the homogeneous operation intended and facilitated the required reduction in compression ratio for gasoline operation. The turbocharger used was a Mitsubishi model 16T-6cm², providing a good compromise between low speed torque and high speed power in an efficient manner. The inlet manifold had been designed to maximise the volumetric efficiency at peak power and hence minimise the compressor work.

No of cylinders	4
Bore (mm)	87.5
Stroke (mm)	83.1
Geometric compression ratio	9:1
Variable valve timing	Dual independent (35°c.a. inlet, 55°c.a. exhaust)
Cam duration	Inlet: 234°c.a. Exhaust: 220°c.a.
Fuel injectors	Bosch (6-hole, 20 cc/s)
Spark plugs	NGK HR7 single electrode
Ignition coils	Bosch coil-on-plug (100mJ)

Table 1: Basic engine characteristics



Figure 1: Illustration of the combustion chamber

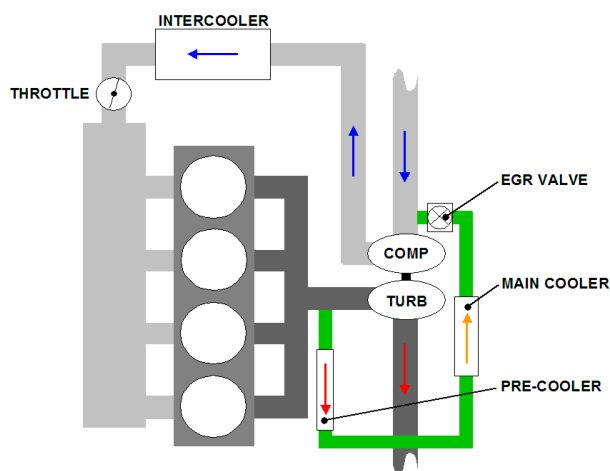


Figure 2: Schematic of the cooled external EGR circuit

A schematic of the external EGR circuit employed is illustrated in Figure 2. This setup was used in earlier work (35) and can be considered to be a “hybrid” version of the conventional low-pressure and high-pressure EGR gas circuits, with EGR withdrawn pre-turbine and passed through the dual cooler and EGR valve assemblies. The gas was then fed back to the intake manifold well upstream (~0.5m) of the turbocharger compressor in an attempt to acquire accurate mixed gas temperature readings. This layout was chosen to ensure high EGR flow rates were obtained but is arguably not the most practical solution (36). Nonetheless, the dual EGR cooler assembly used throughout consisted of a stainless steel pre-cooler (corrugated tube-in-tube design) and main cooler (multi-tube construction more akin to some diesel cooler designs). These coolers could be bypassed if required. In either case, the restricting cross sectional area in the EGR circuit when the EGR control valve was fully open was ~491mm². The gas-to-water intercooler was controlled using a boost temperature control rig, with the air charge controlled to 40°±1°C.

Throughout the part-load tests the engine was fitted with a suitable three-way catalyst. Unfortunately, this catalyst was damaged early on in the high load work and was removed and a suitable orifice restriction fitted further down the exhaust. Pre and post catalyst emissions and inlet manifold CO₂ levels (for external EGR rate) were sampled using a Horiba MEXA 7500/ETR analyser. EGR sampling probes were fitted to each inlet runner, with readings then averaged by simultaneously sampling from all four runners. In order to evaluate combustion, each combustion chamber had been fitted with a Kistler 6125A in-cylinder pressure transducer that mounted flush with the chamber wall. Pressure data acquisition and analysis were performed using AVL Indicom Version 1.2. The engine-averaged thermodynamic parameters expressed herein were evaluated as the average of values compiled for each cylinder over 300 engine cycles. Fuel flow was measured throughout using an alcohol compliant AVL KMA assembly while smoke measurements were obtained using an AVL 415 smoke meter.

EXPERIMENTAL FUELS

Some key properties of the fuels examined are set out in Table 2. In general, these fuels were selected to cover a wide range of oxygen mass concentrations and octane numbers and hence identify key influencing parameters on combustion and emissions in modern turbocharged engines. The matrix was composed of three unleaded gasolines (ULG), the octane ratings of which were varied to match those grades currently available within the UK. The 95RON gasoline was used as the main constituent when producing all other blends. The gasoline-butanol blend (Bu16) was not selected to advocate such fuel for any particular market. Rather, this fuel was included for direct comparison with a “higher” alcohol fuel blend of oxygen mass weighting equivalent to E10. Fuels containing up to 10% ethanol were of significant interest as they are already available within the US and are being considered for the EU. The E22 blend was selected to provide a fuel of moderate ethanol content and also of octane rating similar to the premium 102RON gasoline. Finally, E85 has already emerged in some markets and was required to provide information on the behaviour of a higher ethanol content fuel under downsized, flex-fuelled engine conditions.

	95 ULG	98 ULG	102 ULG	E10	Bu16	E22	E85
Lower Heating Value (MJ/kg)	43.16	43.06	42.34	41.64	41.88	39.96	29.28
RON	95	98	102	98	98	102	108
MON	86	88	91	88	87	88	90
(R+M)/2	91	93	96	93	92	95	99
Density at 15°C (kg/m ³)	0.74	0.75	0.74	0.75	0.75	0.75	0.78
H/C Ratio	1.891	1.796	1.869	1.945	1.943	2.021	2.769
O ₂ (% weight)	0	0	2	3.35	3.35	6.8	29.5

Table 2: Key properties of the test fuels

RESULTS

PART-LOAD RESULTS

Fuel Injection Optimisation – Optimisation of the fuel injection timing and pressure was repeated for each blend at the following three part-load sites:

1. 2000rpm/2bar BMEP
2. 2000rpm/4bar BMEP
3. 4000rpm/2bar BMEP

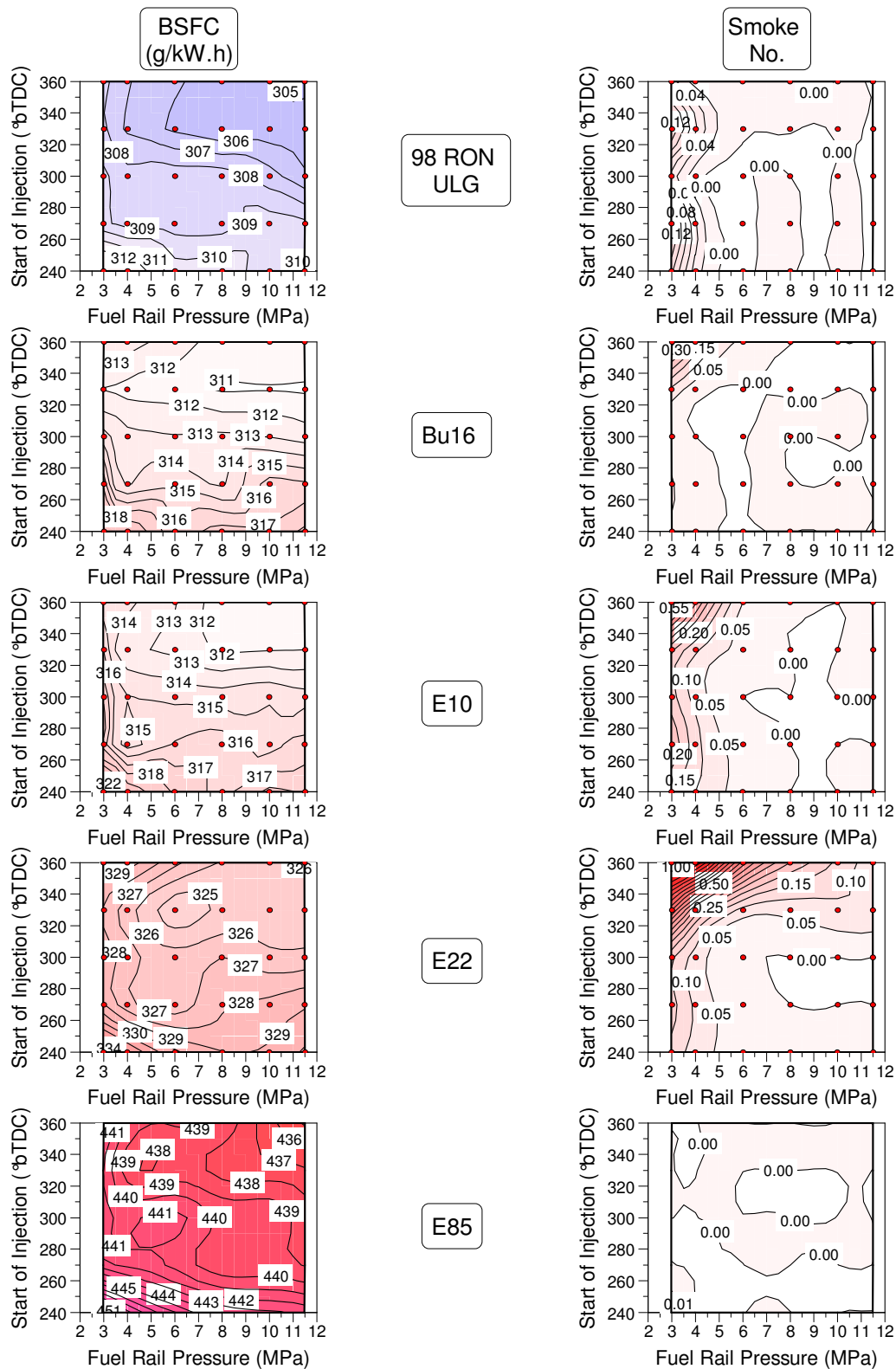


Figure 3: BSFC and post-catalyst smoke maps (2000rpm/4 bar BMEP, MBT spark timing, $\lambda=1$)

The engine was operated with fixed valve timing of low valve overlap (EVC at 10°aTDC and IVO at 3°aTDC) and stoichiometric fuelling throughout. The spark timing was set for Maximum Brake Torque (MBT) at all logged sites. A summary of the injection settings providing good compromise between low fuel consumption and tailpipe emissions is shown in Table 3. Corresponding emissions data is included in Appendix A.

Fuel	Speed (rpm)	BMEP (bar)	SOI (°bTDC)	Fuel Press (MPa)
95 RON	2000	2	360	11.5
95 RON	2000	4	360	11.5
95 RON	4000	2	330	11.5
98 RON	2000	2	330	11.5
98 RON	2000	4	360	11.5
98 RON	4000	2	360	10
102 RON	2000	2	360	8
102 RON	2000	4	330	10
102 RON	4000	2	360	10
Bu16	2000	2	330	6
Bu16	2000	4	330	10
Bu16	4000	2	360	10
E10	2000	2	330	8
E10	2000	4	330	10
E10	4000	2	360	8
E22	2000	2	330	8
E22	2000	4	330	8
E22	4000	2	330	10
E85	2000	2	330	10
E85	2000	4	360	11.5
E85	4000	2	360	10

Table 3: Optimum fuel injection settings

In general, the alcohol blends performed better when injected at slightly later timings and marginally lower fuel pressures. This may have been associated with increased momentum of the air charge at higher intake valve lifts helping to improve the mixing of such fuels. Some of the maps used to determine the optimum injection settings at 2000rpm/4bar BMEP are included in Figure 3. In general, the E22 blend produced the highest levels of smoke with early injection timing at this site. The E85 produced negligible smoke, even at very low fuel pressures. Both these observations also held true at the other speed-load sites. To confirm near-zero smoke with E85, the operation of the smoke meter was double-checked by interrupting the E85 tests and testing another fuel at conditions known to produce smoke. Another general observation, with the exception of the E22 blend, was that reasonable fuel economy and tailpipe emissions were achieved when the injection settings were simply left to those required for optimum gasoline operation. However, given the observations with E22, this would clearly not be feasible as a calibration approach for operation with varying amounts of ethanol. Although it is not expected that higher concentrations of butanol would exhibit similar behaviour, greater volumes of such fuel should be considered in future work.

Dilution Tolerance Effects – In order to evaluate the dilution tolerance of each fuel, excess air and external EGR sweeps were performed at the three aforementioned part-load sites. The cam timings remained fixed at the same low valve overlap as above. The spark timing was set to MBT at all sites. The fuel injection settings were fixed as per Table 3. During the EGR sweeps, the EGR coolers were removed. However, the EGR gas still entered the intake upstream of the compressor and air charge intercooler.

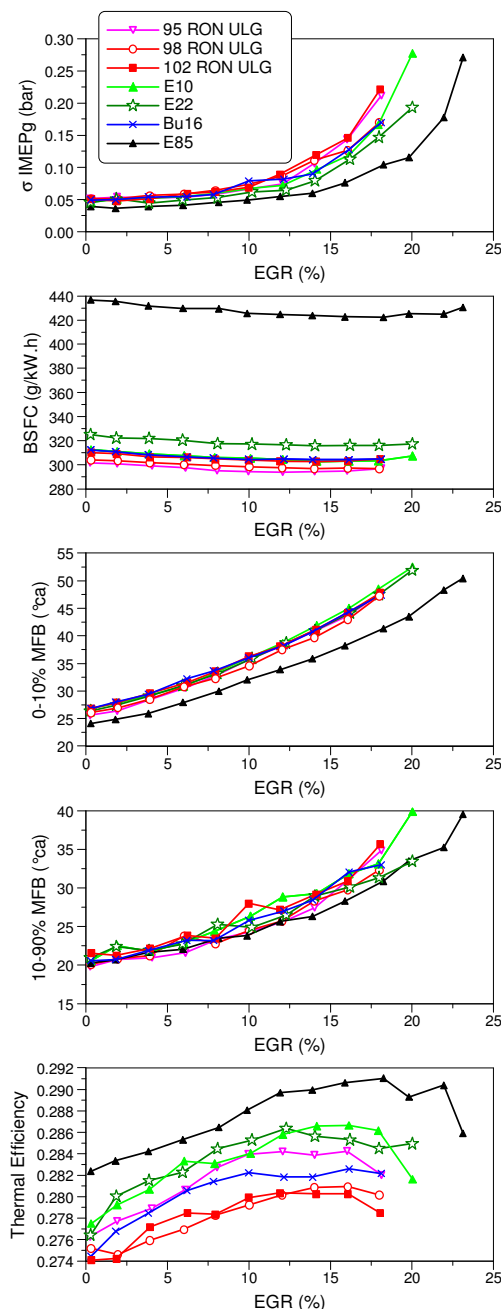


Figure 4: Key measurements during the EGR sweeps at 2000rpm/4bar BMEP (MBT spark timing, $\lambda=1$)