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The Mixture Response of a Stratified Charge Gasoline Engine with Independent, Twin, Port-Fuel Injector Control

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ABSTRACT

An experimental study of the mixture response performance of novel, port-fuel injection strategies upon combustion stability in a gasoline engine was undertaken at low engine load and speed conditions in the range of 1.0 bar to 1.8 bar GIMEP and 1000 rpm to 1800 rpm. The aim was to improve the thermal efficiency of the engine, by extending the lean limit of combustion stability, through promotion of stable charge stratification. The investigation was carried out using a modified 4-valve single cylinder head, derived from a 4-cylinder, pent-roof, production, gasoline engine. The cylinder head was modified by dividing the intake tract into two, separate and isolated passages; each incorporating a production fuel injector. The fuel injection timing and duration were controlled independently for each injector. The performance effects of a single or multiple fuel injection event on a single-sided injector were compared to simultaneous and phased fuel injection for the pair of injectors, with both open valve or closed valve fuel injection timings. A model of the engine, implemented in the Ricardo WAVE software and refined using in-cylinder pressure data, was used to support the findings. The initial experimental results showed good agreement with the model's prediction and baseline data obtained in a previous study. Analysis of the experimental results for the alternative injection strategies showed that the engine could be operated with far leaner mixtures at low speeds and loads. Combustion stability, defined for a single-cylinder engine as 10% CoV_{GIMEP}, was improved for each engine condition tested. At 1000 rpm and 1.0 bar GIMEP, the lean combustion limit was extended from 14:1 air-to-fuel ratio (AFR) to 17.5:1. At 1500rpm and 1.5 bar GIMEP, the lean combustion limit was extended from 17.5:1 to approximately 21:1 AFR. At 1800 rpm and 1.8 bar GIMEP, the lean combustion limit was improved from 21:1 AFR to 22:1. The improved tolerance of the combustion system to charge dilution, due to the optimised injection strategies, was evaluated for high levels of trapped residuals. The relevance to conditions required for controlled auto-ignition combustion is discussed. Finally, the influence of the new strategies upon the rates of heat release and the combustion duration were evaluated and compared to cycle-resolved measurements of the concentration of unburnt hydrocarbons in the exhaust gases.

INTRODUCTION

There is an ever increasing demand upon internal combustion engines for lower emissions and increased fuel efficiency. Although direct fuel injection has become more prevalent in recent times, port-fuel injected engines still present an economically viable option for smaller A and B segment vehicles, particularly suited to the emerging markets. Any performance gain from the use of relatively 'cheap' technology is of great interest to automotive engineers. Such engines currently perform close to the European Euro 5 legislation limits. They are relatively inexpensive power plants to make, suitable for both high volume production and hybrid development. To meet these aims, future PFI engines must be both fuel efficient and demonstrate low levels of emissions of oxides of nitrogen (NOx). A comprehensive review of PFI engines can be found in [1].

One approach has been to develop combustion systems that have a high tolerance to dilution with exhaust gas, whether achieved externally or internally by manipulation of the valve events [2, 3]. A very lean or diluted mixture can be used to improve fuel consumption (reduced pumping work), prevent knock at higher loads and achieve low combustion temperatures, thereby reducing thermal NOx formation; in some cases by more than 90% [2]. Combustion of a lean air-fuel mixture results in a higher ratio of specific heats, a reduction in heat losses and improved thermal efficiency. However, charge dilution results in slower and incomplete combustion and consequently increased unburnt hydrocarbon (ubHC) emissions. The poorest combustion stability occurs at idle and low-load operation where due to heavy throttling, exhaust gas residual levels can reach 30% and the charge velocity is at a minimum. Both effects compromise the flame burning speed. One means of improving the tolerance to residuals dilution is through the generation of structured air motions that can promote stratification of the charge into layers of trapped residual gases and air-fuel mixture [4, 5]. The stratification can be controlled by phasing of the valve opening periods, timing of the fuel injection and by increasing the strength of the dominant air motions that exist in the cylinder [e.g. 6]. This method aims to maintain a stoichiometric air-fuel mixture in the vicinity of the spark plug at ignition, whilst the overall air to fuel ratio (AFR) is lean. However it also compromises the volumetric efficiency of the engine at high-speeds and loads [7-9]. Researchers have sought to overcome these restrictions by using the high temperatures within the EGR gases to initiate compression ignition of a homogeneous or stratified mixture of air, fuel and residual gases; Controlled auto-ignition (CAI). The rapid rate of heat release must be controlled by dilution of the mixture [10-12]. In a review of the literature there is some general agreement that for CAI, the stratification of temperature and air-fuel mixture with the residuals in the combustion chamber at the beginning of the compression stroke has the greatest influence upon the combustion phasing (auto-ignition timing), rate of heat release and combustion stability [13]. The influence of air motion has less of an effect on combustion phasing than that generally reported for lean-burn spark ignition engines [e.g. 14]. In [15], researchers reported that combustion could be controlled effectively by split injection in divided intake ports using fuels with different knock characteristics.

In this study, new port-fuel injection strategies were investigated with the aim to promoting charge stratification with high residual levels in a lean, PFI combustion system. The concept was based upon two, independent fuel injectors per cylinder injecting into two separate intake ports; the variation in injection settings (a combination of number, side and phasing) between the two were utilised to control the distribution of fuel within the cylinder and the initiation, duration and stability of combustion at lean AFR's. A 1-D gas dynamics model implemented in the WAVE model was used to predict the residual fractions and gas properties. In this paper, only the results of the preliminary study are presented for production engine valve timings and valve lifts on each of the 4 valves of the single cylinder engine.

EXPERIMENTAL SETUP

The experimental investigation was carried out in a 4-valve, port-fuel injection, single-cylinder engine derived from a 4 cylinder, pent-roof, Volvo production engine (model B234). The engine was chosen because extensive data had been gathered and analysed over a period of 20 years [e.g. 16-18]. The data obtained from firing and optical research engines related to combustion performance, mean and turbulent air motion characteristics, fuel spray characteristics and exhaust emissions. A forward tumble motion was the predominant in-cylinder bulk air flow motion. The Ricardo tumble ratio was 0.88 (defined as the ratio of the angular velocity of the intake charge for a solid body rotation, after intake valve closure, to the rotational speed of the crankshaft). The specifications of the modified engine and the standard valve timings (no valve deactivation or lift and phasing variations) are given in table 1. The historical data provided the basis of the Ricardo WAVE software base model.

The 4-valve per cylinder, production, donor engine had a typical twin intake port arrangement with a single, central fuel injector, directed towards the split in the port wall and angled at the back of the intake valves. In this study the cylinder head and intake manifold were modified by dividing the intake port into two separate passages between the throttle body and the bifurcation of the intake passages. The intake manifold was also modified to accommodate two, production, fuel injectors. The injectors were installed within each intake passage observing the same angle and location from the valves as in the production geometry. The original and modified set-ups are shown in Figures 1 and 2 respectively.

Table 1: Specification of the Dual Port Fuel Injection Research Engine

Engine Type	single cylinder, 4 valve, pent-roof with flat piston
Displaced Volume [cm3]	575
Stroke [mm]	86.6
Bore [mm]	92.0
Compression Ratio (nominal)	10.1:1
Connecting Rod Length [mm]	159
Exhaust Valve Open [CAD]	68º BBDC
Exhaust Valve Close [CAD]	12 º ATDC
Intake Valve Open [CAD]	10 º BTDC
Intake Valve Close [CAD]	70 º ABDC
Maximum valve lift [mm]	9.8
Ricardo Tumble ratio	0.88
Fuel Type / Pressure [bar]	95 RON pump grade gasoline / 3.5
Fuel injector	four hole Bosch
Ignition System	Spark Plug NGK BP8EVX Coil on Plug (Diamond FK0138)

The engine was installed in the Sir Harry Ricardo Laboratories at the University of Brighton. A customised engine management system was used for open loop operation of the engine. The control of the twin fuel injectors and the ignition system could be varied independently of engine speed and top-dead centre timing. The engine was instrumented to record air, fuel, exhaust, water and oil temperatures (type K thermocouples) and pressures. Oil and water pressures and temperatures were regulated to 3 bar and 70 °C respectively in the usual manner with PID control, three-way mixing valves and shell-in-tube heat exchangers. A thermal mass flow meter (type Endress and Hauser, T-MASS) was used to measure mass flow of air through the engine. The intake manifold pressure was varied using a throttle butterfly valve controlled independently by means of a stepper motor drive. Absolute pressures in the intake and exhaust manifolds were recorded by two transducers (type Kistler 4045 and Drück, water-cooled). The exhaust was fitted with a wide-band lambda sensor (type Bosch ETAS LA4). The ignition System consisted of a spark plug (type NGK BP8EVX) and a standard Mitsubishi coil on plug (type Diamond FK0138).

A commercial low pressure fuel pump and regulator (type Weber) and a shell-in-tube heat exchanger were used to maintain a fuel pressure of 3.5 bar and temperature of 20 °C. The fuel used was pump grade BP 95 RON unleaded gasoline. Two, Bosch, four-hole, PFI injectors and two, separate, configurable injector drivers were fitted to the engine using a shortened Bosch production fuel rail. These are denoted as Injector A and B in Figure 2. Synchronisation and control of the fuel and ignition systems with the engine was performed using a Z8 processor interface box controlled with a Visual basic Serial PC software interface. A programmable arbitrary waveform generator and logic gate were used to generate the second fuel injection timing pulse.

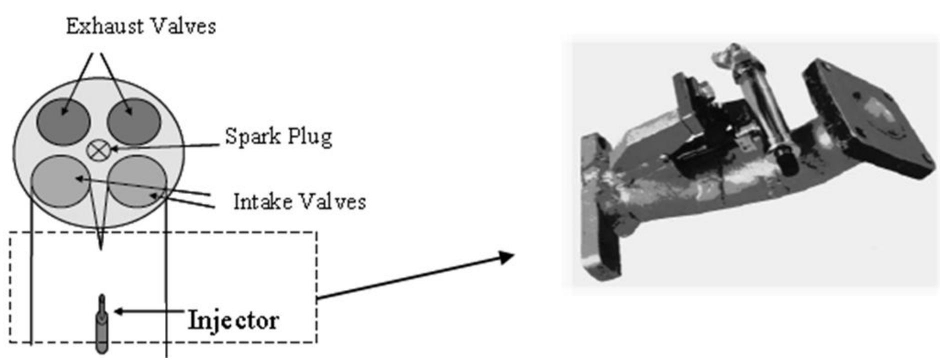

Figure 1: Single cylinder intake manifold with original configuration of a single fuel injector.

In-cylinder gauge pressure was measured with a pressure transducer (type Kistler 6125). Indication was provided by an AVL INDISET 620 data logger. Engine speed was recorded using a 720 +1 pulse rotary crankshaft encoder (type Leine & Linde) with a resolution of 0.5 CAD. Data was recorded over 300 cycles at each test condition. The individual cycles were then averaged to produce ensemble-averaged statistics.

Exhaust hydrocarbon concentration was measured with a fast flame ionization detector (type Cambustion FastFID HFR 400). The sample probe was positioned in the exhaust port, as close as possible to the exhaust valves. All test points were recorded when the engine had reached a thermal equilibrium and for minimum advance for best torque ignition timings (MBT).

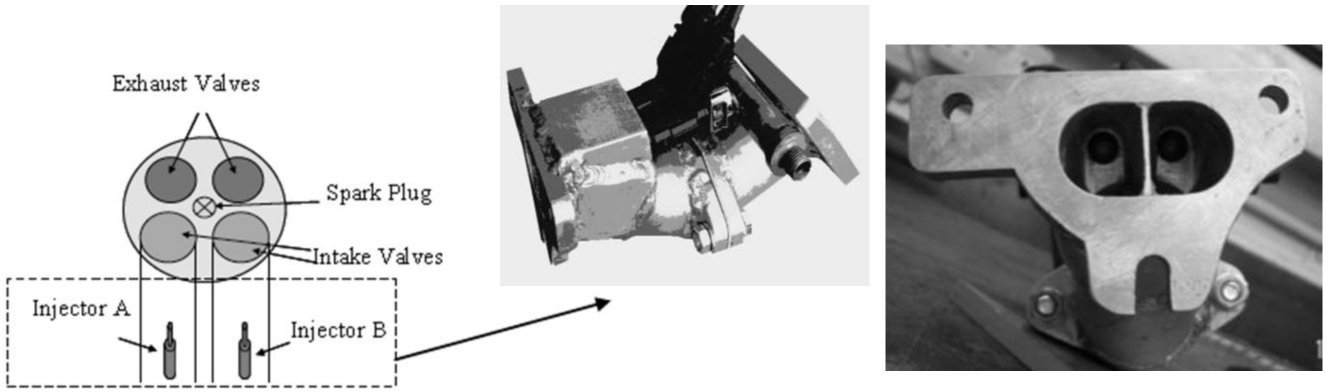

Figure 2: Single cylinder intake manifold with modified configuration incorporating twin fuel injectors and internally divided separate passages.

KEY ENGINE OPERATING POINTS

The key engine test points (KP) were chosen to evaluate the influence of the differing injection strategies upon low load, lean operation compared to the production engine. In the first series of tests, an ignition timing swing was used to determine MBT at 1500 rpm and 1.5 bar gross indicated mean effective pressure (GIMEP) using the original, single injector, production engine configuration. The results compared favourably with prior data from the same engine recorded during an engine development programme [16].

The dual injector configuration was used to evaluate low load engine combustion performance at three different operating conditions; 1000 rpm and 1.0 bar GIMEP (KP1), 1500 rpm and 1.5 bar GIMEP (KP2) and 1800 rpm and 1.8 bar GIMEP (KP3). A mixture response swing was used to determine the optimum spark gap and to rank the different injection strategies against increasing combustion stability. In parallel, 1-D a gas dynamics model of the engine was developed in the Ricardo WAVE software. The model was tuned using in-cylinder pressure data. A detailed description of the model is beyond the scope of this paper but similar approaches can be found elsewhere [e.g. 19]. The results of the model were then used to determine parameters relating to engine efficiency and exhaust emissions that could not be measured directly.

The following injection strategies were investigated with the modified intake system at each of the three key operating points:

• Simultaneous twin port injection with closed intake valves (CVI) and equal injected fuel quantities.

• Stepped (phased) twin port injection using both open valve timing (OVI) and CVI.

• Single-sided injection in one port only with OVI timing.

• Single-sided injection in one port only with CVI timing.

• Single-sided multiple fuel injections (CVI and OVI) in one port only.

RESULTS AND DISCUSSION

The preliminary results presented in this paper are focused upon combustion stability and its response to variations in air-to-fuel ratio at a constant load. In this case, the combustion performance of the single cylinder engine was considered to be stable for a coefficient of variation in gross indicated mean effective pressure (CoV_{GIMEP}) of ≤ 10% over 300 recorded cycles. A single-zone heat release model [following 20] was used to determine the combustion burn angles of the different injection and dilution strategies. The accuracy of the model to predict the heat release was highly dependent upon the value of the ratio of specific heats, γ. Therefore the WAVE model was used to calculate the range of values of γ, in the period between intake valve closure and exhaust valve opening, at different combustion temperatures resulting from varying AFR and residual fraction.

Effect of Spark Plug Gap upon Combustion Stability

To explore the effect of the spark plug gap on the lean combustion limits, three different spark plug gaps were tested; 0.82mm, 0.85mm and 0.92mm, using the modified intake configuration, with equal quantities of fuel injected in each port. The investigation was performed by varying the AFR whilst maintaining the engine conditions at 1500 rpm and 1.5 bar GIMEP. The results are summarised in Figure 3.

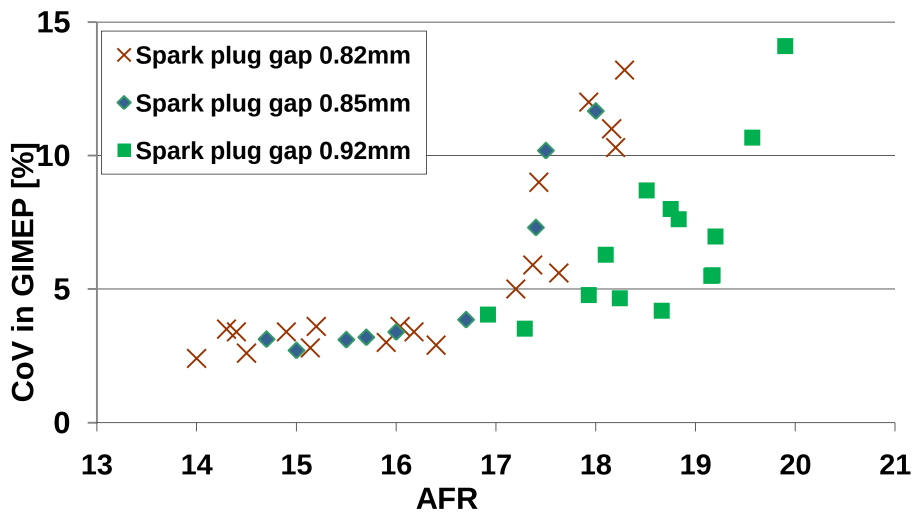

Figure 3: AFR mixture response swing using different spark plug gaps at KP 2.

At the low load and speed condition, it was not unexpected that a similar behaviour was observed between the 0.82 mm and the 0.85 mm gap setting. However, noticeable differences were found in the combustion stability limits when the spark plug gap was increased to 0.92 mm. Up to an AFR of 17.5:1, all points were within the 5% threshold. For the 0.92 mm setting, the lean limit for stable combustion was extended to approximately 19.5:1. These results are in agreement with previous data obtained with this same engine [16-18] in the same spark plug gap range from 0.65 mm to 1.0 mm. Of all the spark plugs tested, the 0.92 mm gap exhibited the best compromise between ignition system durability and lean combustion stability at low engine speeds and weak in-cylinder gas motion.

A set of experiments were devised to principally investigate the differences in performance between open valve injection (OVI) and closed valve injection (CVI) timing on the lean combustion limit. Injection timing for the CVI cases was 90 CAD BTDC in the firing stroke. In the OVI case, start of injection timing was close to exhaust valve closure, at the beginning of the intake stroke. In the 'dual' case, both injectors were triggered with synchronised pulses. In the 'dual phased' cases, one injector was triggered during OVI and the other during CVI. In the multiple injection case, a single injector was consecutively triggered twice in the same cycle; firstly during OVI and then followed by the CVI timing. For all cases, the air-to-fuel ratio was varied with constant load and speed, whilst recording the CoV_{GIMEP}. For each dual injection strategy, the mass of fuel was split in half equally between each port injector, regardless of the injection timing.

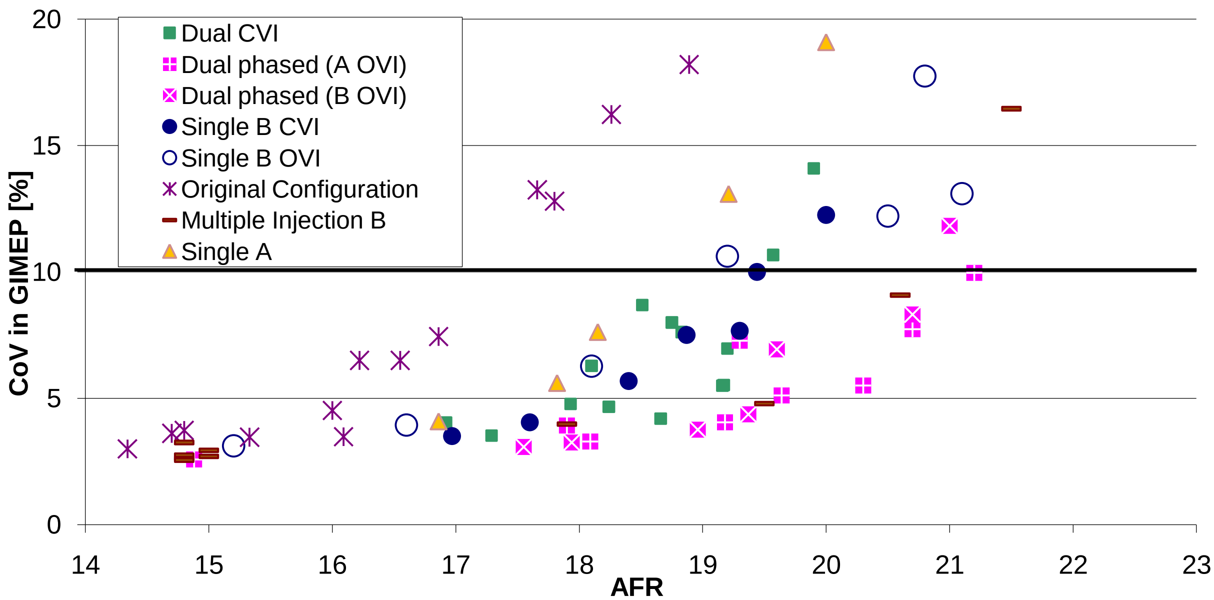

Figure 4a: Comparison of mixture response combustion sensitivity to injection strategy at KP2.

The WAVE simulation at these conditions predicted that up to 25% of the cylinder mass was composed of exhaust backflow which occurred during the valve overlap period due to the heavily throttled condition. The results of the mixture response swing for KP2, KP1 and KP3 are shown in figs 4a, b and c respectively for each of the injection strategies. The results of the original configuration are also plotted to provide a baseline for comparison.

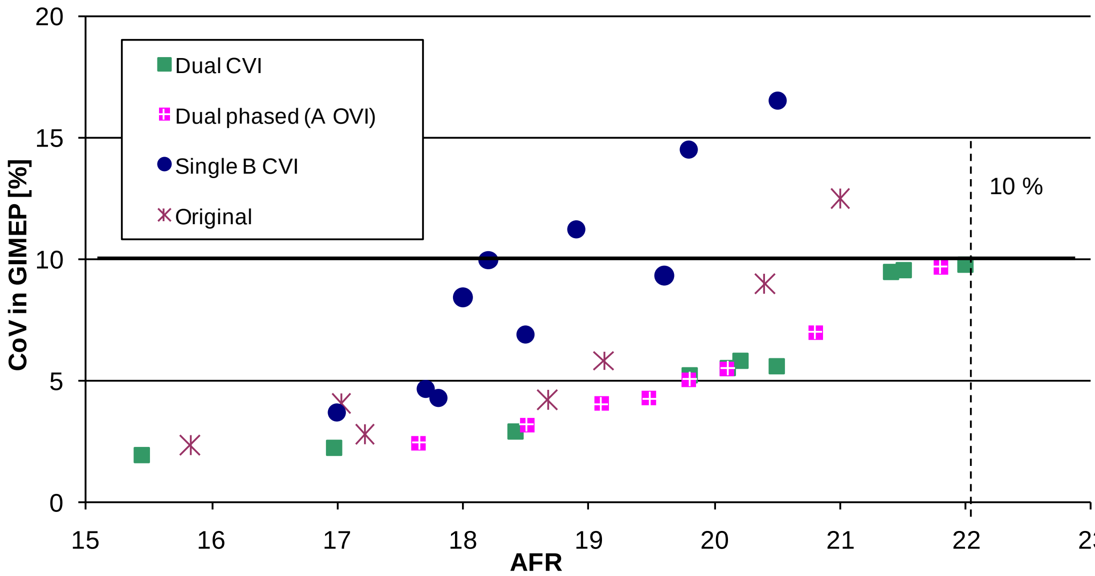

Figure 4b: Comparison of mixture response combustion sensitivity to injection strategy at KP3.

It was observed that a significant improvement in combustion stability at lean AFR's could be achieved in comparison to the original configuration. The split, phased injection strategy showed the best combustion stability at KP2 and KP3. The 5% level was extended up to 19:1 and 20:1 AFR and the 10% limit up to approximately 21:1 and 22:1 AFR for KP2 and 3 respectively. This represented a 4 times ratio improvement over the original combustion stability limit of 17:1 AFR at KP2. The phased injection strategy showed similar results regardless of which port performed the open valve injection. In single injector mode, combustion stability was preferred with the injector/port B configuration at KP2 but not KP3; the injectors were swapped over to confirm the result. The worst combustion stability was observed for single injection in port A for KP2. Multiple injections on port B for KP2 showed a similar level of improvement in terms of combustion stability, with a 10% CoV_{GIMEP} for an AFR of 21:1. The common factor identified for the two best injection strategies at KP2 and KP3 was the fact that half of the fuel was delivered under closed valve conditions and the other half during the open valve period. However, at KP1, the dual and dual phased fuel injection strategies performed comparably with the original configuration. The greatest spread in results was observed at KP1. The best improvement in combustion stability was achieved with the multiple injection strategy at KP1 for AFR's up to approximately 17:1.

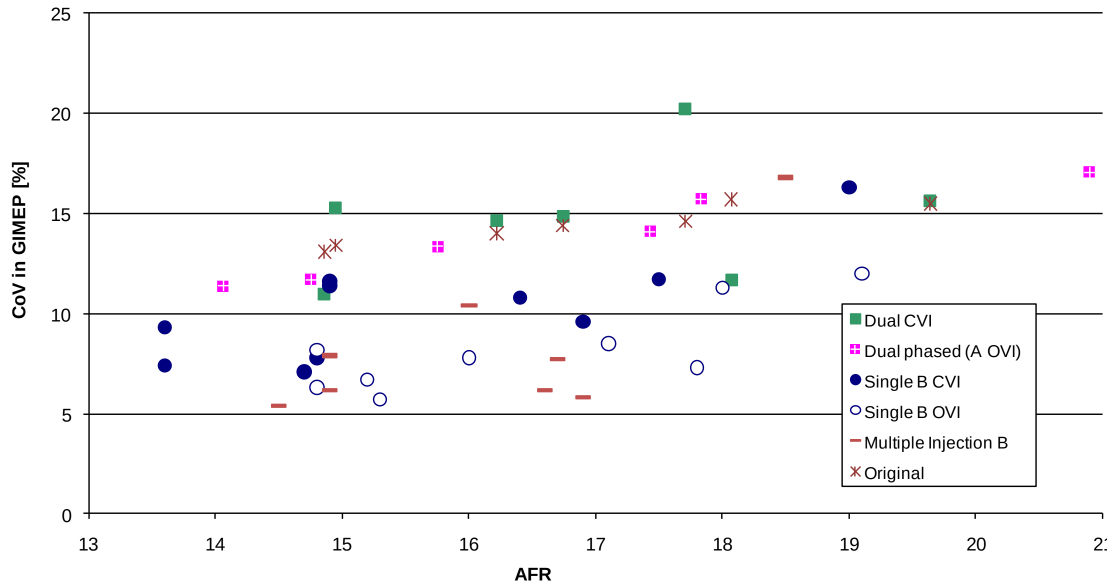

Figure 4c: Comparison of mixture response combustion sensitivity to injection strategy at KP1.

Combustion Burn Duration

A burn angle analysis was carried out for the different test conditions recorded. As expected, the combustion duration increased with increasing AFR in all cases. The ignition delay angle (defined as ignition to10% mass fraction burned duration, (MFB)) is shown in Figure 5 for KP2. The main combustion duration (defined as 10-90% MFB) is plotted in Figure 6 for each of the injection strategies.

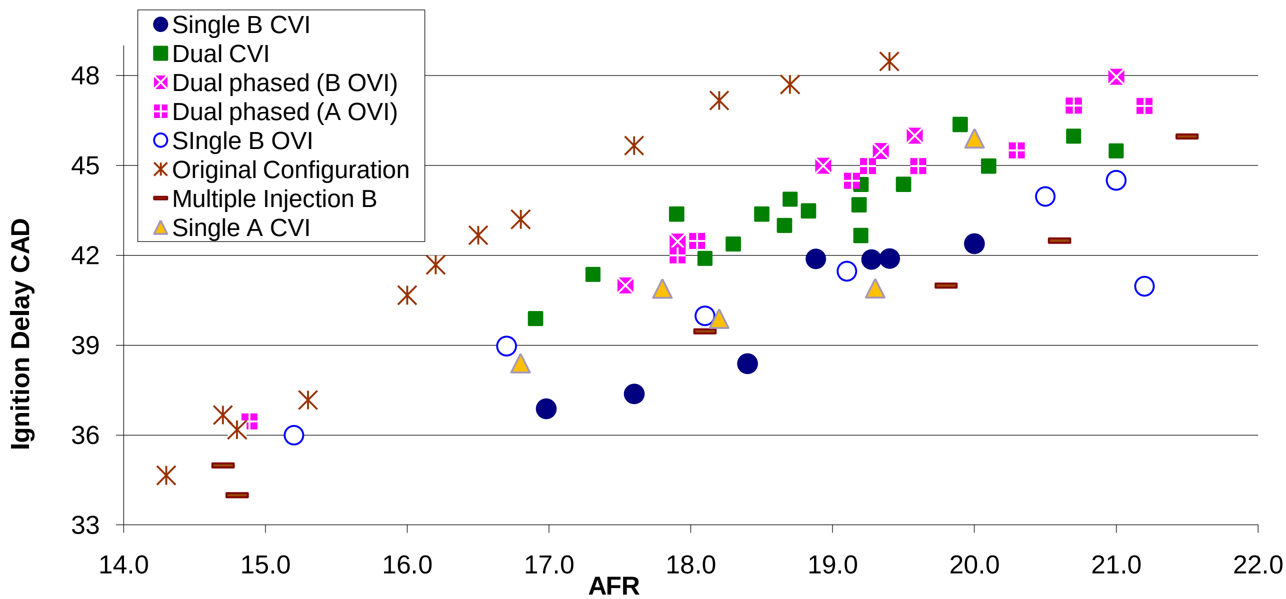

Figure 5: Ignition delay angle, defined as ignition to 10% MFB, response to AFR at KP2.

In all cases, the duration of early combustion increased almost linearly with AFR. The shorter ignition delays were recorded for single injection on port B (particularly for CVI) and multiple injections on port B. The longest ignition delay recorded has been added for reference. This was observed for the original configuration with a 0.82 mm spark plug gap. The shorter ignition delays did not correlate with the lower CoV_{GIMEP}. On the contrary, phased injection at several test points exhibited the longest duration in early flame development.

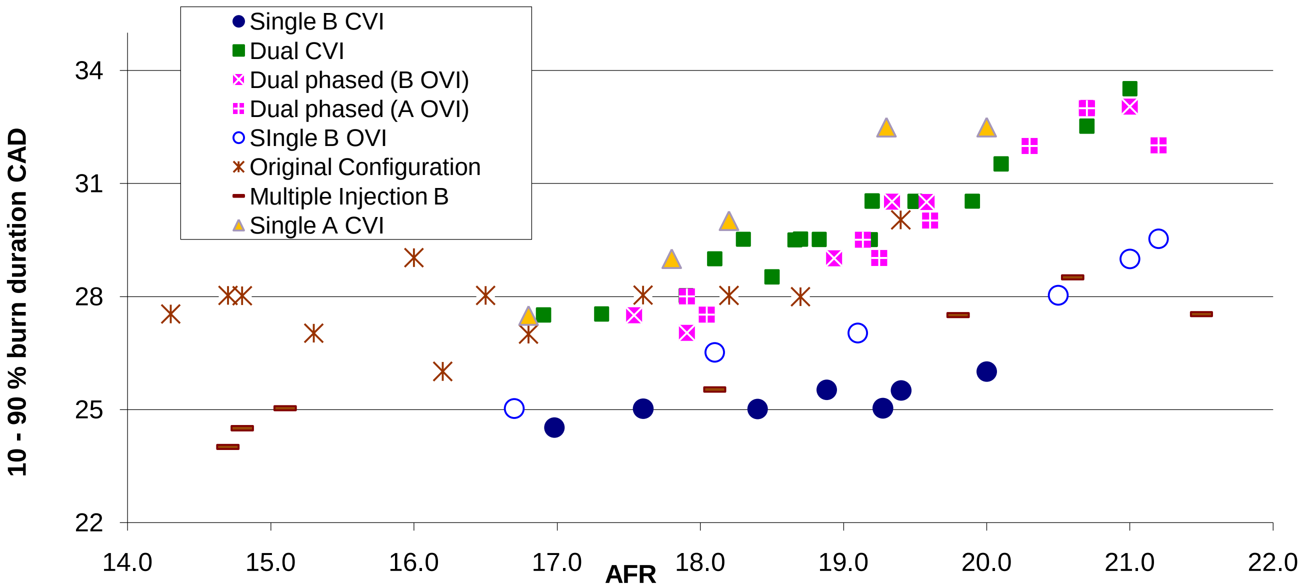

Figure 6: Main burn duration, defined as 10% to 90% MFB, response to AFR at KP2.

The main combustion duration was faster for single injection on port B, particularly for CVI. Injection on port B (single or multiple) showed a lower overall data scatter in the 10-90% duration over the range of AFR's investigated. The 10-90% MFB duration of the 3 dual injection strategies showed very similar durations and similar variations with AFR, irrespective of injection phasing. For AFR's between 17:1 and 20:1, the duration of combustion for the dual configuration increased approximately by 4 CAD. Over the same AFR range, the combustion duration for the single injection on port B varied by only 1.5 CAD. It was also noted that the fastest flame propagation did not generally correspond to the lowest recorded $CoV_{GIMEP.}$ The slowest burn duration was recorded for single injection in port A which corresponded to the poorest combustion stability. However, for this modified configuration with dual injection, shortening the combustion duration did not necessarily lead to improved combustion stability.

The difference in the main burn period between single injectors A and B suggested that the lean, in-cylinder charge conditions, following the ignition delay, had been affected by the separation of the intake port and fuel injection. The conditions meant that the shorter combustion durations were shown by single injection in port B, where fuel was consumed at a greater rate. In a previous optical engine study by the authors [18] and others [16, 17], it was reported that the plane of the in-cylinder mean gas tumble motion was inclined towards port B, during the intake and compression strokes. Other factors, such as differing injector performance, unequal gas and fuel flow characteristics in the ports and the spark plug tip orientation were also likely to influence the results.

Prediction of In-cylinder Residual Concentrations

At a fully throttled condition, the reduced pressure and density in the intake manifold resulted in high levels of in-cylinder residuals caused by exhaust gas re-induction during the valve overlap period. The Ricardo WAVE simulation was used to determine the in-cylinder gas concentrations. A summary of the results of the simulation are shown in Table 2. As the engine speed and load were increased, the volume of in-cylinder trapped residual gas decreased. For increasing AFR, the concentration of residual gas was reduced, due to a wider throttle position. The greatest variation in residual concentration was recorded at the lowest engine speed.

Table 2: Prediction of In-cylinder Residual Gas Fraction with AFR for KP1-3.

AFR range	KP1 residual	AFR range	KP2 residual	AFR range	KP3 residual
13.3	31 %	14.7	25 %	14.7	23 %
14.7	29 %	16.2	23 %	17.6	21 %
16.2	27 %	17.6	22 %	19.1	20 %
17.6	25 %	19.1	21 %	20.6	19 %

Hydrocarbon Emissions

The effect of the different injection configurations upon ubHC emissions was investigated using a Cambustion flame ionisation detector. The probe was located close to the exhaust valve. The results are presented in Figure 7 for the hydrocarbon emissions averaged over 300 engine cycles. The uncertainty in the results was ± 2% as indicated by the error bars.

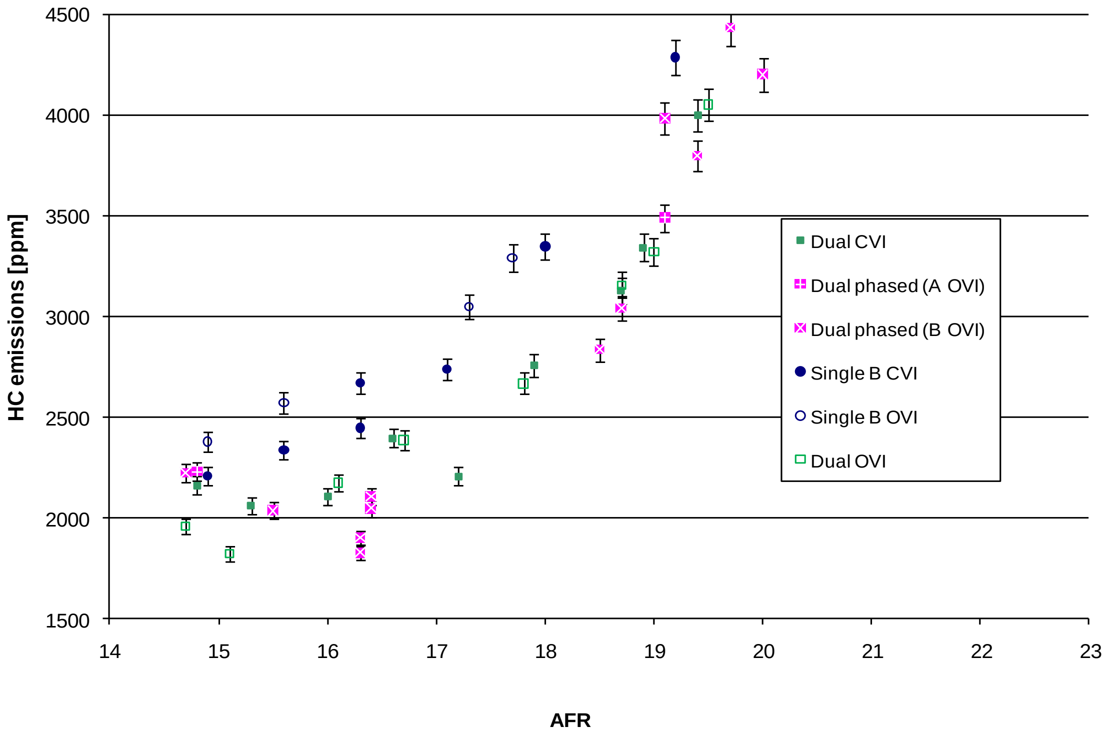

Figure 7: Comparison of ubHC emissions with air-fuel ratio for dual and single injection cases at KP2.

The results in Figure 7 show that for a given AFR, the single-sided injection cases resulted in higher levels of ubHC emissions compared to the dual injection at lean AFR's. This occurred irrespective of OVI or CVI injection timings. In the dual injection strategies, lower ubHC's were present in all cases, consistent with the combustion stability findings. However, it is interesting to note that the most stable combustion configuration, using single-sided injector B with CVI, results in the highest levels of ubHC across the AFR range. In the CVI cases, evaporation of fuel in the intake port displaced fresh charge. For the OVI cases, fuel injection induced charge cooling, resulting in higher peak and end of combustion temperatures and lower ubHC's. However, measurement of the differences was made difficult due to the very small quantities of fuel injected at the chosen engine conditions. In addition, at the lower engine speeds and loads, the weaker in-cylinder gas motions lead to increased fuel impingement and poor mixing and evaporation during OVI operation. For CVI, the lower wall temperatures and gas velocities lead to reduced evaporation and thicker wall films.

CONCLUSIONS

The performance of a port-fuel injected gasoline engine was evaluated and modeled with high levels of exhaust gas dilution. Multiple fuel injectors (dual, separate ports) and multiple fuel injection strategies (single-sided and dual) were introduced to promote in-cylinder charge stratification of air, fuel and exhaust gas. The fuel injection strategy was utilised to maintain combustion stability and improve burn durations under lean, low speed and low load operation. The lean combustion limits were extended at three different engine conditions. Combined induction of liquid fuel (OVI timing) and fuel vapour (CVI timing), generated by multiple, single-sided injections and phased, twin, multiple injection showed an improvement in combustion stability for the three conditions tested as well as injection in a single, isolated port. The results, confirmed by simulation showed that combustion stability was maintained even when relatively high levels of residuals were present. At 1000 rpm and 1.0 bar GIMEP, single-sided injection proved to be the best solution to improve combustion stability. At 1500 rpm and 1.5 bar GIMEP, the lean combustion limit was considerably improved. The greatest improvement was achieved by phased split injection which extended the lean combustion limits by approximately 4 AFR’s from 17:1 to 21:1. Similar combustion stability was obtained with multiple injections in one port only. These two strategies had in common the fact that half of the fuel was delivered during open valve and the other half injected against a closed valve. However, the shorter combustion durations observed for a single-sided injection did not necessarily correspond to improved combustion stability. In addition, one of the ports showed far shorter 10-90% MFB burn duration angles with a single CVI or OVI injection. This suggested that the role of an inclined, in-cylinder tumble plane gas motion (identified in previous work [16-18]) had led to lateral combustion stratification. However, the same port showed the highest concentration of ubHC emissions, due to lower peak and end combustion temperatures. The dual fuel injection, whether synchronised, phase shifted (CVI or OVI timed) resulted in significantly reduced ubHC emissions due to the reduced thermodynamic effects resulting from the far smaller fuel quantities delivered in each port.

It is proposed that a combination of the port fuel injection strategies, that have successfully demonstrated stable combustion at higher AFR's, be used to investigate the transition to controlled auto-ignition operation at low speed and low load regimes. It is envisaged that closed-loop control of SI and CAI combustion could be controlled by manipulating the correct injection strategies; OVI, CVI or both, along with a suitable valvetrain method for trapping of residuals. This is the subject of current work in progress along with varying the injected fuel quantities and phasing between each injector, varying valve timing and de-activation of intake and exhaust valves.

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DEFINITIONS/ABBREVIATIONS

AFR	Air-to-Fuel Ratio
ATDC	After Top Dead Centre
BTDC	Before Top Dead Centre
CAD	Crank Angle Degree
CoV_{GIMEP}	Coefficient of Variability in gross indicated mean effective pressure
CVI/OVI	Closed/Open Valve Injection
EGR	Exhaust Gas Recirculation
IMEP	Indicated Mean Effective Pressure
MBT	Minimum Advance for Best Torque
MFB	Mass Fraction Burned
NOx	Oxides of Nitrogen
PFI	Port Fuel Injection
RON	Research Octane Number
TDC	Top Dead Centre
ubHC	Unburned Hydrocarbons