

Dual-Injection as a Knock Mitigation Strategy Using Pure Ethanol and Methanol

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ABSTRACT

For spark ignition (SI) engines, the optimum spark timing is crucial for maximum efficiency. However, as the spark timing is advanced, so the propensity to knock increases, thus compromising efficiency. One method to suppress knock is to use high octane fuel additives. However, the blend ratio of these additives cannot be varied on demand. Therefore, with the advent of aggressive downsizing, new knock mitigation techniques are required. Fortunately, there are two well-known lower alcohols which exhibit attractive knock mitigation properties: ethanol and methanol. Both not only have high octane ratings, but also result in greater charge-cooling than with gasoline. In the current work, the authors have exploited these attractive properties with the dual-injection, or the dual-fuel concept (gasoline in PFI and fuel additive in DI) using pure ethanol and methanol. The single cylinder engine results at 1500 rpm ($\lambda=1$) show benefits to indicated efficiency and emissions (HC, CO and CO₂) at almost every load (4.5 bar to 8.5 bar IMEP) compared to GDI. This is because the spark timing can be significantly advanced despite the use of relatively low blends ($\leq 50\%$, by volume), which lowers the combustion duration and improves the conversion of fuel energy into useful work. Overall, these results reinforce the potential of the dual-injection concept to provide a platform for aggressive downsizing, whilst contributing to a renewable energy economy.

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INTRODUCTION

In order to increase regional energy security and combat rising global CO₂ emissions, there is an increasing need to revolutionize the energy supply chain. This adds to the underlying concern that, based on current trends, leading energy forecasters expect the world's petroleum to be depleted within the next 40 years [1, 2]. Therefore, it is paramount to search for alternative energy sources in order to alleviate environmental stress and confront the ballooning energy demand. In the UK, it is believed that biofuels offer the most viable mid-term supplement or substitute for gasoline, compared to technologies which are in their infancy (hydrogen fuel cells and full electric platforms) [3].

Although the idea of fuelling internal combustion engines with biofuels is not new [4], its use is receiving increased

worldwide attention. Liquid biomass offers a high energy density option and is compatible with existing combustion systems. In Europe, the promotion of biofuels has led to a legislative approach; by 2020, all EU member states must conform to a 10% minimum target on the use of alternative fuels (biofuels or other renewable fuels) in transportation [5]. In the US, tax incentives have been used to promote the use of ethanol in gasoline [6], in an effort to replicate the success seen in Brazil [7]. Therefore, more emphasis is being placed on the automotive sector to not only design compatible systems with these alternative fuels, but to also optimize their use in neat form and in blends with gasoline.

Currently, ethanol is the most widely adopted biofuel [8, 9]. In 2007, ethanol accounted for 80% of the world's total biofuel production [10]. In Brazil, where its use has dramatically reduced the dependency on petroleum, ethanol is

used as a neat engine fuel or in various blends with gasoline [11, 12]. Alternatively, China has focused on the use of methanol and leads the world as a producer and consumer [13]. This is largely due to the lack of grain and abundance of coal, as opposed to favoring the performance of methanol over ethanol. Nevertheless, low methanol blends with gasoline (up to 15%) have been shown to require only minor engine modifications [14] and yield similar fuel performance to gasoline [15, 16].

In order to maximize the use of these alternative fuel options to gasoline, the automotive industry is beginning to focus on their optimal combustion in spark ignition (SI) engines. An area of keen interest is the method of fuel supply to the engine. Traditionally, in terms of fuel injection, the approach has mirrored that with gasoline; the alternative fuel, in neat or blended form, is injected using either port fuel injection (PFI) or direct-injection (DI). Alternatively, flexible fuelled ethanol vehicles, used ubiquitously in Brazil, can permit a variable blend of ethanol and gasoline (mixed in the fuel tank), as the actual blend can be detected by the diagnostics system [17]. However, the blend ratio cannot be varied in real-time using the engine control unit (ECU), as it is only measured. Therefore, alternative fuelling approaches are being investigated, including that of dual-injection; the combined use of PFI and DI to supply online gasoline-biofuel blends. By leveraging both injection systems simultaneously, instantaneous blend ratios can be supplied to the engine to best suit the duty cycle. In principle, dual-injection combines the advantages of both bi-fuel and flex-fuel approaches.

The potential of dual-injection inspired the creation of Ethanol Boosting Systems (EBS) LLC in 2006. Here, the researchers, who mainly originate from Massachusetts Institute of Technology (MIT), have examined the potential of ethanol (hydrous and anhydrous) boosted direct- and dual-injection engines, to help cool the charge and suppress knock, with only modest hardware modifications [18, 19, 20]. Ford is also investigating the dual-injection technology on their 'Ecoboost' gasoline turbo-charged direct-injection (GTDI) engines. Here, PFI gasoline and DI E85 (15% gasoline and 85% ethanol, by volume) has been used to improve the engine efficiency and to avoid knock at high load [21, 22]. Other original equipment manufacturers (OEMs) that have investigated the combination of PFI and DI fuelling include Toyota and more recently, Audi. The work by Toyota demonstrated the improved engine performance (fuel economy and torque) and reduced emissions at full load using a 3.5 liter V6 gasoline engine (2GR-FSE) [23]. For Audi, the dual-injection technique is being used in a turbocharged 1.8 liter gasoline engine. As with the 2GR-FSE engine, this combustion mode contributes to higher fuel efficiencies at part-load compared to conventional single injection [24]. Clearly, the benefits of dual-injection arise when the engine duty cycle includes frequent medium- to high-load operation.

Most investigations of dual-injection have included ethanol as the DI fuel for knock mitigation purposes. However, the use of methanol can also greatly increase the charge-cooling effect and therefore knock suppression in order to support PFI gasoline. With recent industry focus on engine downsizing, it is more important to mitigate knock. The team from EBS LLC have conducted modeling studies of alcohol fuels in a highly turbo-charged DISI engines which could be used in heavy duty long haul applications [25]. They have suggested that turbo-charged DI alcohol engines could be as, or more efficient, than diesel engines. The team have also produced simulation results comparing E85 and methanol in collaboration with Volvo [26]. Their simulation results showed how methanol might be more effective than E85 (for the same knock suppression, only half the fuel flow of methanol is required). However, these numerical calculations have not been compared with experimental work.

Therefore, it is the aim of this paper to experimentally compare the dual-injection strategy when using ethanol and methanol as knock mitigation fuels in dual-injection using a single cylinder SI research engine. The effectiveness of each alcohol is assessed as a knock mitigation fuel for PFI gasoline. The approach is to use minimum alcohol injections so that the knock limit is raised from PFI gasoline. The engine performance and emissions are compared at various engine loads from 3.5 bar to 8.5 bar IMEP in 1 bar intervals. In the following sections, the engine setup, experimental results and finally conclusions are discussed.

EXPERIMENTAL SETUP

ENGINE AND INSTRUMENTATION

The experiments were performed on a single-cylinder, 4-stroke SI research engine, as shown in [Figure 1](#).

The 4-valve cylinder head includes the Jaguar spray-guided direct-injection (SGDI) technology used in their V8 production engine (AJ133) [27]. It also includes variable valve timing technology for both intake and exhaust valves, which, for this study, was kept constant, as shown in [Table 1](#). As well as firing under high pressure (150 bar) SGDI conditions, a low pressure (3 bar) PFI system is available. The two fuelling modes can be used independently or simultaneously, as in the case for dual-injection.

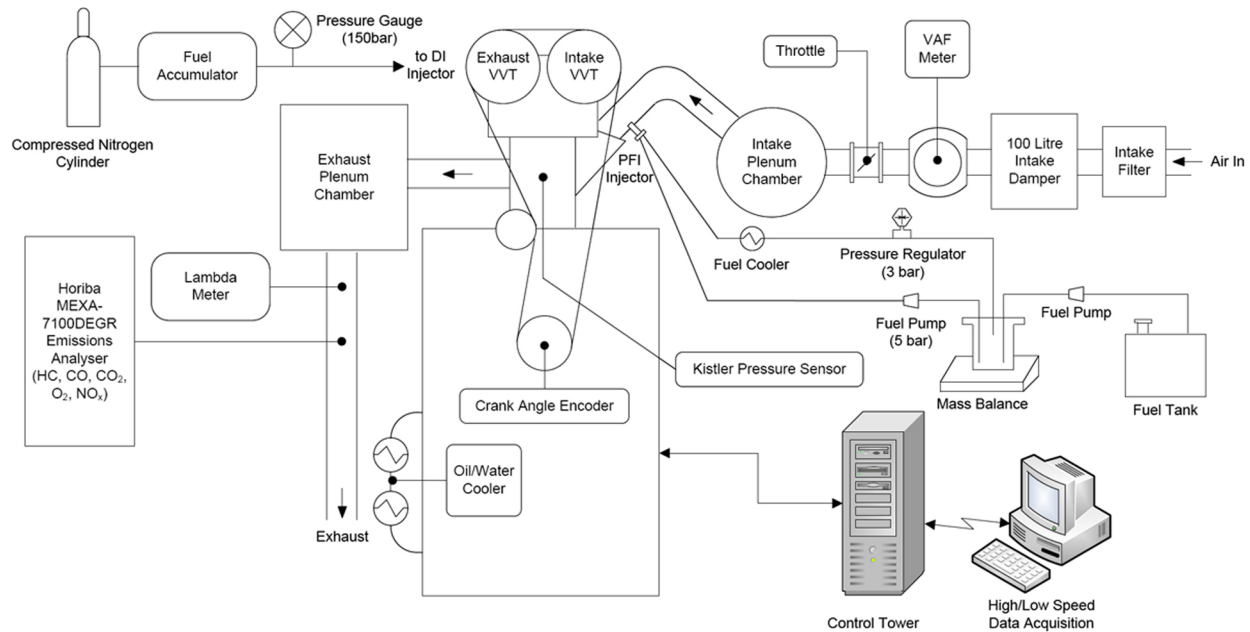


Figure 1. Schematic of engine and instrumentation setup

Table 1. Engine Specification

Engine Type	4-Stroke, 4-Valve
Combustion System	Dual-Injection: PFI and Spray-Guided DISI
Swept Volume	565.6 cm ³
Bore x Stroke	90 x 88.9 mm
Compression Ratio	11.5:1
Engine Speed	1500 rpm
Injector	Multi-Hole Nozzle
PFI Pressure and SOI Timing	3 bar, 50° bTDC _{comb}
DI Pressure and SOI Timing	150 bar, 280° bTDC _{comb}
Intake Valve Opening	16° bTDC _{intake}
Exhaust Valve Closing	36° aTDC _{intake}
Valve Open Duration	250 CAD

The engine was coupled to a DC dynamometer to maintain a constant speed of 1500 rpm (± 1 rpm) regardless of the engine torque output. The in-cylinder pressure was measured using a Kistler 6041A water-cooled pressure transducer which was fitted to the side-wall of the cylinder head. The signal was then passed to a Kistler 5011 charge amplifier and finally to a National Instruments data acquisition card. Samples were taken at 0.5 crank angle degree (CAD) intervals for 300 consecutive cycles, so that an average could be taken. The crankshaft position was measured using a digital shaft encoder mounted on the crankshaft. Coolant and oil temperatures were controlled at $85 \pm 5^\circ\text{C}$ and $95 \pm 3^\circ\text{C}$, respectively, using a Proportional Integral Differential (PID) controller. All temperatures were measured with K-type thermocouples.

The engine was controlled using software developed in-house written in the LabVIEW programming environment. High-speed, crank-angle-resolved and low-speed, time-resolved data was also acquired using LabVIEW. This was then analyzed using MATLAB developed code so that an analysis of the combustion performance could be made.

EMISSIONS AND FUEL MEASUREMENT

The gaseous emissions were quantified using a Horiba MEXA-7100DEGR gas analyzer. Exhaust samples were taken 0.3m downstream of the exhaust valve and were pumped via a heated line (maintained at 191°C) to the analyzer.

The fuel consumption rate and resulting blend ratios were calculated using the volumetric air flow rate (measured by a positive displacement rotary flow meter and stabilized by a 100 L intake plenum), known DI injector calibration curves for each fuel and the lambda (λ) value. All tests were run at stoichiometric conditions ($\lambda=1$), which was controlled using the cross-over of the carbon monoxide (CO) and oxygen (O_2) emissions concentrations, as described in detail in a previous publication by the authors [28].

TEST FUELS

Both 97 RON gasoline and ethanol were supplied by Shell Global Solutions, UK, whereas the methanol was supplied by Fisher Scientific, UK (99.5% purity). A high octane gasoline was chosen as this represents the most favorable characteristics offered by the market and provides a competitive benchmark to the lower alcohols. The fuel characteristics are shown in Table 2.

Table 2. Test Fuel Properties

	Ethanol	Methanol	Gasoline
Chemical Formula	C ₂ H ₆ O	CH ₄ O	C _{6.62} H _{11.88}
H/C Ratio	3	4	1.795
O/C Ratio	0.5	1	0
Gravimetric Oxygen Content (%)	34.78	50	0
Density @ 20°C (kg/m ³)	790.9*	792	744.6
Research Octane Number (RON)	107†	106†	96.8
Motor Octane Number (MON)	89†	92†	85.7
Anti-Knock Index	98	99	91.25
Stoichiometric Air Fuel Ratio	8.95	6.47	14.46
LCV (MJ/kg)	26.9*	19.83*	42.9
LCV (MJ/L)	21.3*	15.7*	31.9
Carbon Intensity (gCO ₂ /MJ)	71	69.4	74.4
Flash Point (°C)	13	12	-40
Heat of Vaporization (kJ/kg)	840†	1103†	373
Stoichiometric Heat of Vaporization (kJ/kg _{air})	93.9	171.5	25.8
Initial Boiling Point (°C)	78.4	65	32.8

*Measured at the University of Birmingham: ASTM D240

†Heywood, J.B., *Internal Combustion Engine Fundamentals*. 1988: McGraw-Hill [29]

FUELLING VARIATIONS

For clarity, the fuelling variations used in this work (3 fuels and 3 injection modes) have been abbreviated in the remaining sections. When gasoline has been used in PFI and DI it is referred to as PFI and GDI, respectively. When referring to either lower alcohol used as a neat DI fuel, the notation EDI and MDI is used for ethanol and methanol, respectively. To indicate ethanol and methanol dual-injection, the first letter of each is positioned after the first letter in GDI (which indicates gasoline) and is separated with a hyphen. For instance, ethanol dual-injection (PFI + EDI) is denoted G-EDI. A summary of this information is found in [Table 3](#).

Table 3. Fuelling Variations

PFI Fuel	DI Fuel	MBT Timing	Notation
Gasoline		PFI	PFI
	Gasoline	GDI	GDI
	Ethanol	EDI	EDI
	Methanol	MDI	MDI
Dual-Injection			
Gasoline	Ethanol	EDI	G-EDI
Gasoline	Methanol	MDI	G-MDI

EXPERIMENTAL PROCEDURE

The engine was considered warm once the coolant and lubricating temperatures had stabilized at 85°C and 95°C, respectively. All the tests were carried out at the stoichiometric air-fuel ratio (AFR_{stoich}), or $\lambda=1$, fixed injection timing (280°bTDC_{comb}) and engine speed (1500 rpm), ambient air intake conditions (approximately 25 ±2°C) and constant valve timing (see [Table 1](#)). The ignition timings for G-EDI and G-MDI were equal to the MBT timings of EDI and MDI, respectively (see [Table 3](#)). This represents the maximum improvement in efficiency (from PFI) with the minimum amount of alcohol injection. Therefore, the authors intend to examine the volume fraction of alcohol required to reach the same level of knock mitigation as EDI and MDI. As a consequence of this effective knock suppression, this method helps to minimize the increase in fuel consumption over PFI gasoline (the lower alcohols have lower LCVs, see [Table 2](#)). The in-cylinder pressure data from 300 consecutive cycles was then averaged and analyzed using the aforementioned MATLAB script.

When changing fuels, the high pressure DI fuelling system was purged using nitrogen until the lines were considered clean. As a further precaution, the new DI fuel was then flushed through the high pressure circuit in order to dilute the effect of any previous fuel. Once the line was re-pressurized to 150 bar using the new DI fuel, the engine was run for several minutes. This made sure that no previous fuel remained on the injector tip or any combustion chamber crevices before any data was acquired. Each test was also run three times for repeatability.

FUEL BLEND CALCULATIONS

In research, the air-fuel ratio (AFR) of a known fuel composition is conventionally measured using an appropriate lambda meter and oxygen sensor. This requires presetting the AFR_{stoich} value for either neat fuel or the known fuel blend. However, in this study, the exact in-cylinder blend ratio of the two fuels from PFI and DI varies as required and the overall composition is therefore unknown prior to testing. Therefore, the authors have used the cross-over theory of the O₂ and CO emissions concentrations, instead of the lambda meter and oxygen sensor combination, to control the excess air ratio under steady-state conditions. For transient engine