

Effects of Combustion Phasing, Injection Timing, Relative Air-Fuel Ratio and Variable Valve Timing on SI Engine Performance and Emissions using 2,5-Dimethylfuran

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ABSTRACT

Ethanol has long been regarded as the optimal gasoline-alternative biofuel for spark-ignition (SI) engines. It is used widely in Latin and North America and is increasingly accepted as an attractive option across Europe. Nevertheless, its low energy density requires a high rate of manufacture; in areas which are deficient of arable land, such rates might prove problematic. Therefore, fuels with higher calorific values, such as butanol or 2,5-dimethylfuran (DMF) deserve consideration; a similar yield to ethanol, in theory, would require much less land. This report addresses the suitability of DMF, to meet the needs as a biofuel substitute for gasoline in SI engines, using ethanol as the biofuel benchmark. Specific attention is given to the sensitivity of DMF to various engine control parameters: combustion phasing (ignition timing), injection timing, relative air-fuel ratio and valve timing (intake and exhaust). Focus is given to the window for optimization; the parameter range which sustains optimal IMEP (within 2%) but provides the largest reduction of emissions (HC or NO_x). The test results using a single cylinder SI research engine at 1500rpm show how DMF is less sensitive to key engine parameters, compared to gasoline. This allows a wider window for emissions optimization because the IMEP remains optimal across a greater parameter range.

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INTRODUCTION

With ever-increasing concerns over fuel security and the problem of global warming, there is a greater need to pursue alternative energy sources. Carbon-free fuels, which do not emit carbon dioxide (CO₂) once consumed, are the long-term ideal in order to eradicate carbon emissions. However, biofuels offer a short- to mid-term solution in reducing the dependence on mineral oil and overall, or life-cycle CO₂ emissions.

One particular biofuel candidate, which can help to reduce life-cycle CO₂ emissions is 2,5-dimethylfuran, otherwise known as DMF. In 2007, bioscientists at the University of Wisconsin-Madison publicized technological breakthroughs in its manufacture and the production of high yields [1, 2]. Since then, these techniques have benefitted from further iterations by other institutions [3,4,5,6,7]. These developments have attracted the attention from automotive

researchers in the potential use of DMF as an alternative energy carrier to gasoline. In comparison to ethanol, DMF offers several improvements, including a higher energy density (approximately 40% higher) and insolubility in water [8].

Currently, relatively few publications exist on DMF as a gasoline-alternative fuel. The first reported engine studies were conducted by the authors' group [9,10,11]. This added to the laboratory studies of the laminar burning velocity [12,13,14,15], spray properties [16] and combustion intermediates of DMF [17] from the authors (summarized in an online book chapter [18]) and collaborating Chinese universities. Evidently, this publication contributes to a series of experiments led by the authors' group to explore the use of DMF as a fuel for automotive applications.

One such area to explore is the sensitivity of DMF to various engine parameters. The sensitivity of a fuel to engine parameters affects the ability to optimize the engine for

various emissions and efficiency trade-offs. The most influential parameter in a spark-ignition (SI) engine is the ignition timing. This significantly affects the combustion process, which determines the fuel economy, torque output and emissions performance [19]. Usually, the ignition timing is optimized using sophisticated mathematical approaches, including advanced Design of Experiment (DoE) methodologies [20,21,22], in order to find the optimum, or minimum advance for best torque (MBT) timing. Minimal advance or retard about this point gives modest variation in power and fuel consumption but can lead to large changes in NO_x and HC emissions. Clearly, an alternative fuel which produces the largest reduction in emissions, whilst achieving competitive performance has the greatest value in a practical application.

Another engine control parameter which dramatically influences the combustion behavior in modern direct-injection (DI) engines is the injection timing. The injection timing can help to reduce the fuel consumption at low and part-load through stratification, as the pumping losses are reduced [23]. Alternatively, at high-load, the engine torque can be increased due to charge-cooling as the knock is suppressed and the ignition timing can be advanced. The much higher injection pressures of DI also encourages rapid fuel atomization and an increased vaporization rate, which also aids combustion stability [24]. However, this can lead to fuel impingement on the piston or cylinder wall [25]. Therefore, as with ignition timing, it is essential to understand the change in engine behavior as the injection timing is varied when introducing new fuels. Each fuel has a different impact on the charge-cooling and the particle size distribution will affect the extent of vaporization and wall wetting. Ethanol is highly sensitive to the injection timing due to the high latent heat of vaporization [26, 27]. Although this can help to increase maximum IMEP, it does narrow the window for emissions optimization for a given performance drop from the optimum.

The use of stratified or lean combustion has become commonly applied to modern direct-injection spark-ignition (DISI) engines [28]. This is a method to simultaneously increase efficiency and reduce emissions but is also very complex to control [29]. At part-load operation the greater throttle opening reduces the pumping losses usually associated with stoichiometric operation. In terms of emissions, lean-burn strategies can also dramatically reduce nitrous oxide (NO_x) emissions, as the production of NO_x is very temperature dependant. This lean burn option becomes more possible because of the development of high conversion efficiency NO_x traps [30]. Nevertheless, as the mixture is leaned, the combustion approaches the limit of stable operation. At this 'lean limit', the ignition becomes unreliable and complete combustion is unsustainable, resulting in engine misfire [31]. Such a lean limit is also fuel dependant because the vaporization rates and combustion speeds vary. Therefore, a fuel which produces a very lean combustion

limit will enable a wide window for optimization and provides an opportunity to greatly reduce NO_x emissions.

With more alternative fuels being investigated in the search for fossil fuel replacements, it is important to fully understand their combustion characteristics. This paper investigates the difference in several parameter sensitivities to engines that are fuelled with ethanol and DMF. Such parameters include: ignition timing, injection timing, air-fuel ratio (AFR) and valve timing (intake and exhaust). Apart from the variation of ignition timing, the parameters were mainly tested at high load (8.5bar IMEP). Sweeps were conducted around the reference operating condition and the change in load and other parameters was observed. At the highest load condition, clearer separations occur between the fuels. Also, where the effect on emissions is analyzed, only the NO_x and total hydrocarbon (HC) emissions have been investigated. In the following sections, the engine setup, experimental results and finally conclusions are discussed.

EXPERIMENTAL SETUP

In this section the experimental setup and procedure are discussed in turn, as similarly found in a previous publication by the authors [11]. However, in this work, more emphasis is placed on the method of performing the individual parameter sweeps.

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 DISI technology used in their V8 production engine (AJ133) [32]. It also includes variable valve timing technology for both intake and exhaust valves.

The engine was coupled to a DC dynamometer to maintain a constant speed of 1500 ± 1 rpm, regardless of the torque output. The in-cylinder pressure was measured using a Kistler 6041A water-cooled pressure transducer which was fitted flush 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 0.5CAD 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.

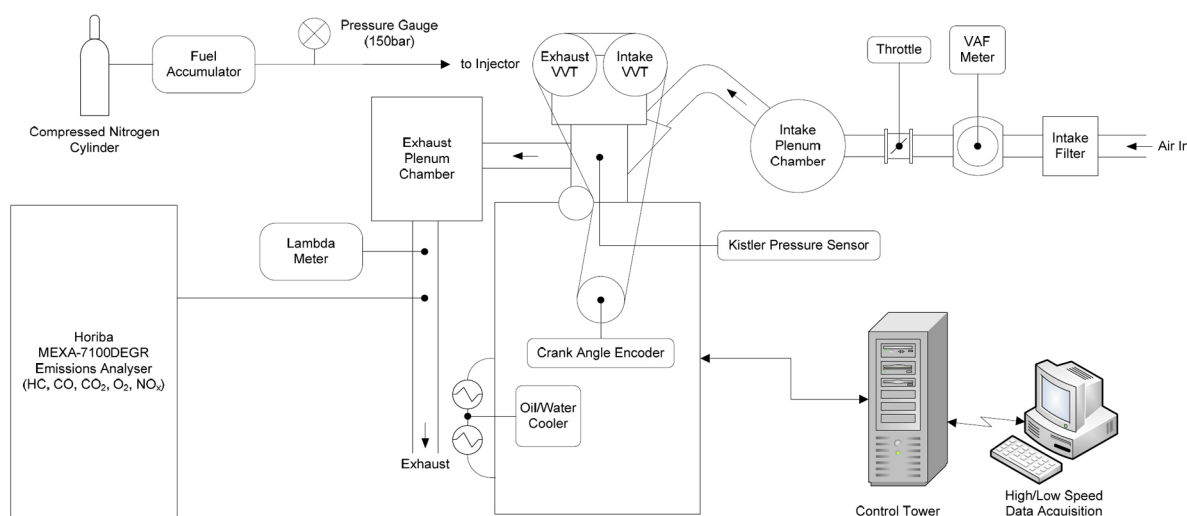


Figure 1. Schematic of engine and instrumentation setup

EMISSIONS AND FUEL MEASUREMENT

The gaseous emissions were quantified using a Horiba MEXA-7100DEGR gas tower. 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 was calculated using the volumetric air flow rate (measured by a positive displacement rotary flow meter) and the actual lambda value (Bosch heated LSU wideband lambda sensor and ETAS LA4 lambda meter). The LA4 lambda meter uses fuel-specific curves to interpret the actual AFR using the oxygen content in the exhaust. Before each test, the user inputs the fuel's hydrogen-to-carbon (H/C) and oxygen-to-carbon (O/C) ratios, as well as the stoichiometric AFR (AFR_{stoich}), so that the fuel composition can be used to characterize the fuel curves.

TEST FUELS

The DMF used in this study was supplied by Shijiazhuang Lida Chemical Co. Ltd, China at 99.8% purity. This was benchmarked against commercial 97RON gasoline and to ethanol, which were both supplied by Shell Global Solutions, UK. The high octane gasoline was chosen as this represents the most favorable characteristics offered by the market and provides a strong benchmark to the two biofuels. The fuel characteristics are shown in Table 1.

Table 1. Test Fuel Properties

	DMF	Ethanol	Gasoline
Chemical Formula	C_6H_8O	C_2H_6O	C_2-C_{14}
H/C Ratio	1.333	3	1.795
O/C Ratio	0.167	0.5	0
Gravimetric Oxygen Content (%)	16.67	34.78	0
Density @ 20°C (kg/m^3)	889.7*	790.9*	744.6
Research Octane Number (RON)	101.3†	107‡	96.8
Motor Octane Number (MON)	88.1†	89‡	85.7
Octane Index, ($K = 0.5$)	94.7	98	91.25
Stoichiometric Air Fuel Ratio	10.72	8.95	14.46
LCV (MJ/kg)	32.9*	26.9*	42.9
LCV (MJ/L)	29.3*	21.3*	31.9
Flash Point (°C)	1	13	-40
Heat of Vaporization (kJ/kg)	332	840‡	373
Initial Boiling Point (°C)	92	78.4	32.8

*Measured at the University of Birmingham

†API Research Project 45 (1956) and Phillips data.

‡Heywood, J.B., Internal Combustion Engine Fundamentals. 1988: McGraw-Hill [19]

EXPERIMENTAL PROCEDURE

The engine was considered warm once the coolant and lubricating oil temperatures had stabilized at $85 \pm 5^\circ C$ and $95 \pm 3^\circ C$, respectively. All tests were carried out at a constant engine speed of 1500rpm and ambient air intake conditions (approximately $25 \pm 2^\circ C$). Sweeps were performed using each sensitivity parameter. During these sweeps the remaining

parameters were fixed at the reference engine operating conditions, as shown in Table 2. Here (regardless of the load), the engine is run at AFR_{stoich} ($\lambda=1$), $280^\circ bTDC_{comb}$ start of injection timing (SOI), $16^\circ bTDC_{intake}$ intake valve open (IVO) timing, $36^\circ aTDC_{intake}$ exhaust valve close (EVC) timing and MBT ignition timing. These MBT timings can be found in a previous publication [11]. The pressure data from 300 consecutive cycles was recorded for each test using the in-house developed LabVIEW code.

Table 2. Engine Specification

Engine Type	4-Stroke, 4-Valve
Combustion System	Spray-Guided DISI
Swept Volume	565.6cm^3
Bore x Stroke	90 x 88.9mm
Compression Ratio	11.5:1
Engine Speed	1500rpm
Injector	Multi-hole Nozzle
Fuel Pressure and Timing	150bar, $280^\circ bTDC_{comb}$
Intake Valve Opening	$16^\circ bTDC_{intake}$
Exhaust Valve Closing	$36^\circ aTDC_{intake}$

When changing fuels, the high pressure fuelling system was purged using nitrogen until the lines were considered clean. Once the supply line was re-pressurized to 150bar using the new fuel, the engine was run for several minutes. This removed any previous fuel from the injector tip and in any combustion chamber crevices before the data was acquired. The ETAS LA4 lambda meter settings were changed for each fuel using the AFR_{stoich} , O/C and H/C ratios in Table 2.

PARAMETER SWEEPS

Sweeps of each of the four key engine parameters were performed, whilst maintaining the other variables at the reference engine condition shown in Table 2. Each of the sweeps were generated between 3.5bar and 8.5bar IMEP (low and high engine loads, respectively), in 1bar IMEP intervals and at a fixed engine speed of 1500rpm. During each sweep, the throttle and ignition timings (except for the ignition timing sweeps) remained constant (generated at the reference condition). This was used in order to increase repeatability. The method for performing each sweep is discussed in turn.

Firstly, ignition timing sweeps were performed to analyze the sensitivity of IMEP when retarding, or advancing (where possible) from the optimum timing at the reference condition. As used in other studies by the authors and commonly employed by other researchers, the MBT, or optimum ignition timing is defined as the ignition timing to produce the maximum IMEP for a fixed throttle position. If audible knock occurred, the MBT timing was retarded by 2CAD, an arbitrarily safe margin, as advised by key engine researchers [19, 33]. Retarding the timing further for emissions preservation was not used, in order to eliminate subjectivity

and better isolate the effect of ignition timing sensitivity. At each load, the ignition timing was advanced to find the knock limit or until a significant drop in performance or stability was seen (IMEP decrease from MBT > 5% or COV of IMEP increase from MBT > 3%). Similarly, the ignition timing was retarded until this aforementioned drop in performance was also seen. At each load the throttle was kept constant so that the drop in performance and emissions would be easily repeatable and the fuel injection pulse-width was adjusted finely ($\pm 1\mu s$) to find stoichiometry.

Secondly, SOI timing sweeps were conducted either side of $270^\circ bTDC_{comb}$. Although this is not the reference SOI timing condition, when using intervals of 30CAD, the behavior at TDC and BDC was neatly included. Firstly, the SOI timing was advanced to $360^\circ bTDC_{comb}$ (TDC) and then retarded to $180^\circ bTDC_{comb}$ (BDC) with two records of $270^\circ bTDC_{comb}$ in the process. This allows the effect of any engine drift during the sweeps to be observed and accounted for.

Thirdly, AFR sweeps were determined using the injector pulse-width, once the desired load was found at stoichiometry. Firstly, the AFR was made increasingly rich ($\lambda < 1$) until $\lambda=0.8$ was reached and then lean until the lean limit was reached; determined by high combustion instability, or when the COV of IMEP exceeded 5%. The results at stoichiometry ($\lambda=1$) were recorded before each rich and lean sweep.

Finally, the IVO and EVC timings were varied either side of their respective middle points ($0^\circ aTDC$, IVO and $20^\circ aTDC$, EVC) in 5CAD intervals with a 40CAD range. Once again, this middle point was recorded twice.

For each set of tests, several repeats were made with each fuel to produce an average and error bars have been shown in the graphs where applicable in order to highlight such variations.

RESULTS AND DISCUSSION

The results from the various sweeps are shown in the following section. The most interesting relationships are discussed in order to better understand and quantify the sensitivity to each parameter.

COMBUSTION PHASING

The effect of ignition timing has a direct effect on the combustion phasing. Therefore the variation of engine load (IMEP) due to the sensitivity of ignition timing is examined in order to quantify differences between the three fuels.

The variation of IMEP using DMF, ethanol and gasoline at the highest load ignition timing sweep (approximately 8.5bar IMEP) is shown in Figure 2a. At this load, there is a clear difference between the three fuels. Ethanol combustion, which is uninhibited by knock at this compression ratio (11.5:1), permits a wide ignition timing sweep and allows the IMEP to be analyzed either side of the MBT timing ($21^\circ bTDC$). DMF and gasoline on the other hand, are much



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