Applied Energy 98 (2012) 59–68

Contents lists available at SciVerse ScienceDirect

Applied Energy

journal homepage: www.elsevier.com/locate/apenergy

Combustion performance of 2,5-dimethylfuran blends using dual-injection compared to direct-injection in a SI engine

Ritchie Daniel [a], Hongming Xu [a,b,*], Chongming Wang [a], Dave Richardson [c], Shijin Shuai [b]

[a] School of Mechanical Engineering, University of Birmingham, Birmingham, UK
[b] State Key Laboratory of Automotive Safety and Energy, Tsinghua University, China
[c] Powertrain Research & Technology, Jaguar Land Rover, Whitley Engineering Centre, Coventry, UK

ARTICLE INFO

Article history:
Received 30 December 2011
Received in revised form 13 February 2012
Accepted 28 February 2012
Available online 27 April 2012

Keywords:
2,5-Dimethylfuran
DMF
Biofuel
Dual-injection
Cross-over theory

ABSTRACT

Dual-injection strategies in spark-ignition (SI) engines allow the in-cylinder blending of two different fuels at any blend ratio, when simultaneously combining port fuel injection (PFI) and direct-injection (DI). This offers increased flexibility and the potential to optimize the combustion process depending on the engine requirement and fuel availability. Until now, little or no research evidence exists on the experimental comparison of dual-injection to DI in the combustion of gasoline–biofuel blends. Therefore, in this work the authors evaluate this comparison using a new biofuel candidate: 2,5-dimethylfuran (DMF). The differences in performance are examined using D25 (25% DMF in gasoline, by volume) in a single cylinder SI research engine operating at 1500 rpm and varying load (3.5–8.5 bar IMEP). All tests were carried out at stoichiometry ($\lambda = 1$) using the cross-over theory of the carbon monoxide and oxygen emissions concentrations. The current results are promising for dual-injection. The improved mixture preparation compared to DI results in lower combustion durations and higher in-cylinder pressures. This gives rise to higher indicated thermal efficiencies and lower fuel consumption rates compared to the same blend (D25) in DI (up to 4% and 3.2%, respectively). More significantly, the lower fuel consumption rate with dual-injection (on a volumetric basis), is up to 1.2% lower than with gasoline in homogenous DI (GDI) up to 8 bar IMEP, despite the use of a lower energy density biofuel.

© 2012 Elsevier Ltd. All rights reserved.

1. Introduction

There is an increasing need to revolutionize the energy supply chain in order to increase regional energy security and combat rising global CO_2 emissions. 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 important to search for alternative and sustainable 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 for, or substitute to 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, including that of novel alternatives [5]. 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 [6]. In the US, tax incentives have been used to promote the use of ethanol in gasoline [7], in an effort to replicate the success in Brazil [8]. Government policy on energy use is also changing in emerging nations [9]. For large energy consuming nations, such as China and India, which have an increasing gap between energy demand and supply, the biofuel route offers improved energy security and reduced dependency on imported oil. 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.

The success of each biofuel option will largely be determined by the energy density. In general, for oxygen containing fuels, the energy density decreases as the oxygen content increases. Therefore,

Abbreviations: aTDC, after top dead center; bTDC, before top dead center; CAD, crank angle degrees; CI, compression ignition; CID, combustion initiation duration; CO, carbon monoxide; CO_2, carbon dioxide; COV, coefficient of variation; DISI, direct-injection spark-ignition; DMF, 2,5-dimethylfuran; EBS, ethanol boosting systems LLC; HC, hydrocarbon; IMEP, indicated mean effective pressure; KLSA, knock limited spark advance; LCV, lower calorific value; MBT, maximum brake torque; MFB, mass fraction burned; NO_x, nitrogen oxides; PM, particulate matter; RPM, revolutions per minute; SGDI, spray-guided direct-injection; SI, spark-ignition; SOI, start of injection; TDC, top dead center; ULG, unleaded gasoline.

* Corresponding author at: School of Mechanical Engineering, University of Birmingham, Birmingham, UK.

E-mail address: h.m.xu@bham.ac.uk (H. Xu).

0306-2619/$ - see front matter © 2012 Elsevier Ltd. All rights reserved.
http://dx.doi.org/10.1016/j.apenergy.2012.02.073

there is a shift towards fuels with lower oxygen content, like *n*-butanol [10,11]. However, there is also an emerging biofuel candidate – 2,5-dimethylfuran (or DMF, as it has become known) – which has an even closer energy density match to gasoline (within 8%). Since the recent disclosure of its improved manufacturing techniques in 2007 [12,13], research has been prompted into its potential use as a gasoline alternative biofuel [14,15]. Not only does DMF have a higher energy density than other popular oxygenated gasoline alternative fuels (methanol, ethanol and *n*-butanol), it is also insoluble in water (avoiding potential contamination issues). The first publications on the engine combustion and emissions characteristics of DMF were recently reported by the authors of this paper [16,17]. This further adds to the portfolio of research undertaken by the University of Birmingham, UK and Xi'an Jiatong University, China [18–23]. Furthermore, the most recent work discusses the use of DMF using the dual-injection strategy [24]. However, this previous work was evaluated at sub-optimal engine conditions (the throttle openings and ignition timings were fixed, despite the use of different blend ratios) and more importantly, did not compare the effectiveness of blends in dual-injection to DI.

The automotive industry is beginning to focus on the optimal combustion of biofuels with gasoline. Traditionally, in terms of fuel injection, the approach has mirrored that with gasoline; the biofuel, in neat or blended form, is injected using either port fuel injection (PFI) or direct-injection (DI). However, the properties of gasoline–biofuel blends can exhibit non-linear relationships [25] and some fuels, like methanol, require additives to overcome phase separation [26]. In order to cope with this non-linear behavior, the preparation of each externally mixed blend should be well understood. In addition to this, each injection method (PFI and DI) has differing benefits depending on the operating mode [27]. Therefore, in order to avoid phase separation issues and maximize the use of gasoline–biofuel blends, the authors have investigated the possible leveraging of both injection systems simultaneously. This, not only introduces the flexibility of an instantaneously variable blend ratio, it avoids the need for phase separation additives. However, this novel combustion mode is relatively unexplored technology and the effectiveness of blends in dual-injection (PFI + DI) compared to DI only is unknown, especially when using DMF.

The high potential of the dual-injection technology has seen the creation of a Massachusetts Institute of Technology (MIT) spin-off company in 2006: Ethanol Boosting Systems LLC (EBS). The researchers 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 [28–30]. Ford Motor Co. is also investigating the dual-injection technology with 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 by suppressing knock at high loads [31]. Other automotive original equipment manufacturers (OEMs) that have investigated the combination of PFI and DI fuelling (albeit with gasoline in both injectors) include Toyota Motor Co. and more recently, Audi AG. The work by Toyota in 2006 demonstrated the improved engine performance (fuel economy and torque) and reduced emissions at full load using a 3.5 L V6 gasoline engine (2GR-FSE) [32]. For Audi, the dual-injection technique is being used in a turbocharged 1.8 L gasoline engine. As with the 2GR-FSE engine, this combustion mode contributes to higher fuel efficiencies compared to conventional single injection, albeit at part-load only [33].

The authors of this paper take a different approach to utilizing dual-injection technology. The work directly analyses the differences between externally and internally (in-cylinder) mixed gasoline–biofuel blends using a single cylinder, 4-stroke SI engine. The suitability of DMF to dual-injection is the main focus of this work. By fixing the blend ratio, the effect of low blends of DMF (25% DMF in gasoline, by volume) at low loads and low speed (fixed at 1500 rpm) can be investigated. This opposes the approach by the aforementioned leading automotive OEMs (Toyota, Ford and Audi) who have focused on medium to high load performance. The authors of this work believe that these results present a new approach to dual-injection; biofuels with high energy density but low heat of vaporization (unlike ethanol and methanol) are the main beneficiaries. The results for DMF are counterintuitive to the approach made by the OEMs and EBS using ethanol (low energy density but high vaporization). In the following sections, the engine setup, experimental results and finally conclusions are discussed.

2. Experimental setup

2.1. Engine and instrumentation

The experiments were performed on a single-cylinder, 4-stroke SI research engine, as shown in Fig. 1. The 4-valve cylinder head includes the Jaguar spray-guided direct-injection (SGDI) technology used in their V8 production engine (AJ133) [34]. It also includes variable valve timing technology for both intake and exhaust valves, which, for this study, was kept constant. As well as firing under high pressure (150 bar) SGDI conditions, a low pressure (3 bar) PFI system is available. Early start of injection (SOI) timing (280° bTDC) was optimized for the highest volumetric efficiency and homogeneity [25], which occurs close to the location of the highest piston speed. The two fuelling modes (PFI and DI) can be used independently or simultaneously.

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 0.5CAD 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 °C and 95 °C (±3 °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.

2.2. Fuel flow measurement

The fuel consumption rate was calculated (and blend ratios controlled) 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 by inferring the PFI fuel flow (see Eq. (1)). 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 [24].

$$m_{f,PFI} = \frac{m_a - (m_{f,DI})AFR_{s,DI}}{AFR_{s,PFI}} \tag{1}$$

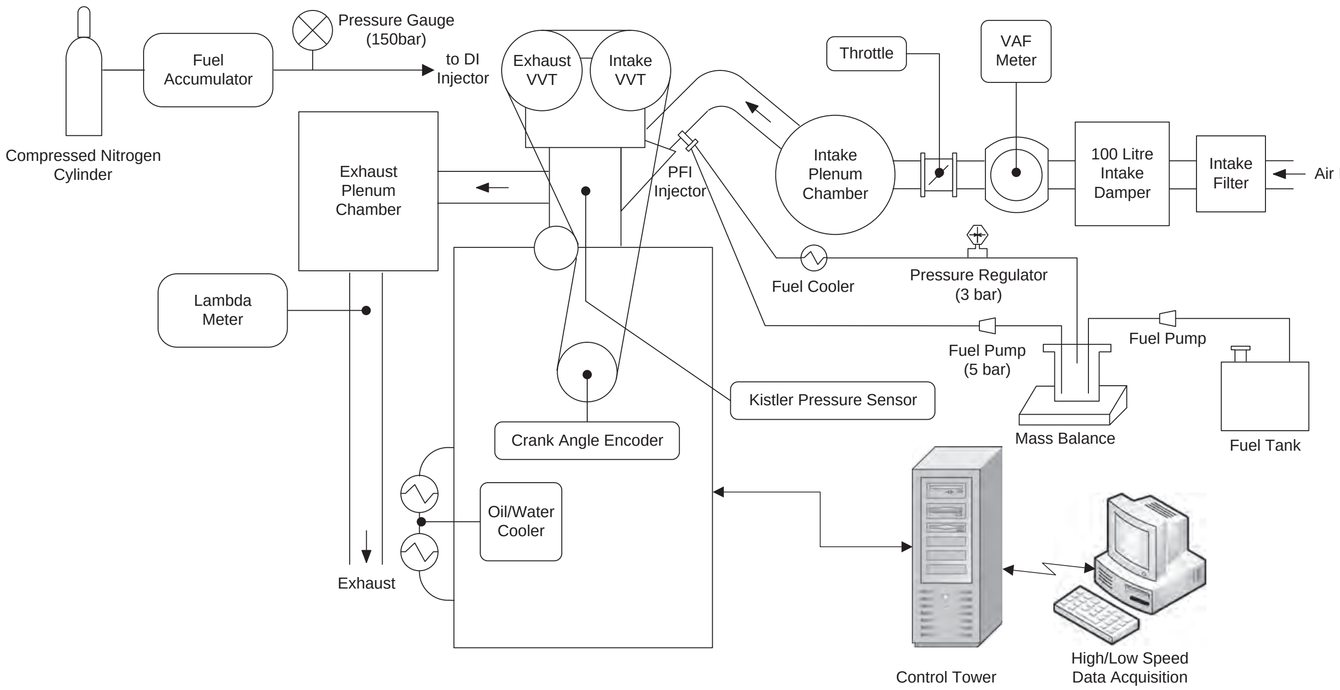

Fig. 1. Schematic of engine and instrumentation setup.

The resulting fuel properties for the D25 blends in dual-injection were accurately calculated based on the final volume from each injector for each case. Therefore, linear interpolation between DMF and gasoline was then used to interpret the individual hydrogen-to-carbon (H/C), oxygen-to-carbon (O/C), stoichiometric air–fuel ratios (AFR_{stoich}) and lower calorific values (LCV). The D25 blends in DI were externally mixed and measured accurately using a graduated measuring cylinder and burette.

2.3. Test fuels

The DMF used in this study was supplied by Beijing LYS Chemicals Co. Ltd., China at 99.8% purity. The 97 RON gasoline was supplied by Shell Global Solutions, UK. A high octane gasoline was chosen as this represents the most competitive characteristics offered by the market and provides a strong benchmark to DMF. The fuel characteristics for gasoline and DMF are shown in Table 1.

Table 1
Test fuel properties.

	DMF	Gasoline
Chemical formula	C_6H_8O	$C_{6.62}$–$C_{11.88}$
H/C ratio	1.333	1.795
O/C ratio	0.167	0
Gravimetric oxygen content (%)	16.67	0
Density @ 20 °C (kg/m^3)	889.7[a]	744.6
Research octane number (RON)	101.3[b]	96.8
Motor octane number (MON)	88.1[b]	85.7
Octane index ([RON + MON]/2)	94.7	91.25
Stoichiometric air fuel ratio	10.72	14.46
LCV (MJ/kg)	32.89[a]	42.9
LCV (MJ/L)	29.26[a]	31.9
Flash point (°C)	1	−40
Heat of vaporization (kJ/kg)	332	373
Stoichiometric heat of vaporization (kJ/kg_{air})	31	25.8
Initial boiling point (°C)	92	32.8

[a] Measured at the University of Birmingham: ASTM D240.
[b] API Research Project 45 (1956) and Phillips data.

2.4. Fuelling variations

For clarity, the fuelling variations used in this work have been abbreviated in the remaining sections. When using gasoline in PFI or DI it is referred to as PFI or GDI, respectively. When using DMF, in neat form in DI, the notation DDI has been used. Therefore, when referring to dual-injection, which comprises PFI and DDI, the notation G-DDI is used. In this work D25 has been used as the only blend in DI and dual-injection. For each case, this blend fraction is subscripted after the DMF element in the previous notations. For instance, D_{25}DI is used for DI and G-D_{25}DI for dual-injection. A summary of this information is found in Table 3.

2.5. 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 AFR_{stoich} ($\lambda = 1$), fixed injection timing (280° bTDC) and engine speed (1500 rpm), ambient air intake conditions (approximately 25 °C ± 2 °C) and constant valve timing (see Table 2). Unlike the previous work on dual-injection by the authors [24], the ignition timing in this work has been optimized. The averaged in-cylinder pressure data (from 300 consecutive cycles) was then analyzed using MATLAB.

Engine load sweeps (3.5–8.5 bar indicated mean effective pressure (IMEP) in 1 bar increments) were performed for all fuel variations described in Table 3 (PFI, GDI, DDI, D_{25}DI and G-D_{25}DI). At each load, the ignition timing was advanced in order to find the minimum advance for best torque or MBT timing [35]. If audible knock occurred, the timing was retarded by 2CAD, an arbitrarily safe margin and is denoted KL-MBT, or knock-limited MBT timing. Repeats were performed and an average was used to present the results.

When changing fuels, the high pressure fuelling system was purged using nitrogen until the lines were considered clean. Each blend was mixed vigorously prior to each test, in order to discourage phase separation (no fuel additives were available). Once the line was re-pressurized to 150 bar using the new fuel, the engine was run for several minutes. This made sure that no previous fuel

Table 2
Engine specification.

Engine type	4-Stroke, 4-valve
Combustion system	Dual-injection: PFI and spray-guided DISI
Swept volume	565.6 cm^3
Bore × stroke	90 × 89 mm
Compression ratio	11.5:1
Engine speed	1500 rpm
Injector	Multi-hole Nozzle
PFI pressure and SOI Timing	3 bar, 50° bTDC
DI pressure and SOI Timing	150 bar, 280° bTDC
Intake valve opening	16° bTDC
Exhaust valve closing	36° aTDC

Table 3
Test variations.

PFI fuel	DI fuel	Notation
Gasoline		PFI
	Gasoline	GDI
	DMF	DDI
DI and dual-injection blends (25% DMF, by vol.)		
	DMF/gasoline	$D_{25}DI$
Gasoline	DMF	$G\text{-}D_{25}DI$

remained on the injector tip or any combustion chamber crevices before any data was acquired.

3. Results and discussion

3.1. PFI compared to GDI

In their detailed technical review, Zhao et al. emphasized the growing presence of DI in modern engines due to the influence of injection timing [27]. However, when employing an early injection or homogenous strategy, PFI retains some clear advantages. In order to highlight the differences between PFI and homogenous DI the authors have analyzed the performance of gasoline in both modes. The results are presented in Fig. 2 and help to explain the attraction of the dual-injection strategy.

At low load (3.5 bar IMEP) there are negligible differences in efficiency between PFI and homogenous DI (Fig. 2a). However, as the load increases the efficiencies begin to separate. Although the part-load efficiency benefits of PFI are greater than homogenous DI, this would be exceeded using DI stratification due to reduced pumping work. Nevertheless, the authors have used homogenous DI as a baseline in this study due to its simplicity to control over stratified combustion. At 7 bar IMEP the two modes cross and the superior PFI mode is superseded by homogenous DI. Until this load PFI operation generates greater efficiencies (higher indicated thermal efficiency and lower fuel consumption) due to higher peak in-cylinder combustion pressures (P_{max}) which helps to lower the combustion duration (Fig. 2b). Above 7 bar IMEP, the benefits of charge-cooling are more apparent with homogenous DI. The ignition timing can be advanced due to greater knock suppression, which results in similar consequences in P_{max} and the resulting combustion duration as found with PFI below 7 bar IMEP. For PFI at high load, the benefits of the lower throttling losses are also offset by the increased knocking susceptibility. The advantage in indicated thermal efficiency for PFI and homogenous DI at their peaks (6 bar and 8.5 bar respectively) is around 1%. Although the efficiency for homogenous DI increases with load, PFI rises and falls with a higher gradient almost symmetrically about 6 bar IMEP. The change in indicated thermal efficiency from 6.5 bar to 8.4 bar IMEP is 1.3% and in terms of fuel consumption is 9.9 g/kWh.

In summary, the difference in efficiency with load is primarily due to the charge-cooling effect with homogenous DI. When using PFI, the fuel evaporation event occurs much earlier (increasing homogeneity) and is largely caused by intake manifold wall and valve wetting. This reduces the cooling effect on the intake temperature as the energy is transferred elsewhere and generates higher combustion temperatures (and pressures) than an equivalent homogenous DI system. This increased physical intensity raises reduces the combustion duration and thus increases efficiency. Although, for homogenous DI, this cooling effect is detrimental at low loads, at higher loads, it helps to suppress knock and allows the ignition timing to be more advanced, which, in turn increases the combustion pressure and resulting efficiency.

Clearly, each mode has benefits in different operating loads. Therefore, the aim of dual-injection is to combine the benefits of the two injections modes in order to maintain a high efficiency across a wider load range. When combining this 'hybrid' injection strategy with two different fuels, the varying benefits from the individual constituents can be further combined. The results from the authors' experimental work are presented in the next few sections and show the promising prospect of this strategy with such gasoline alternative biofuels as DMF.

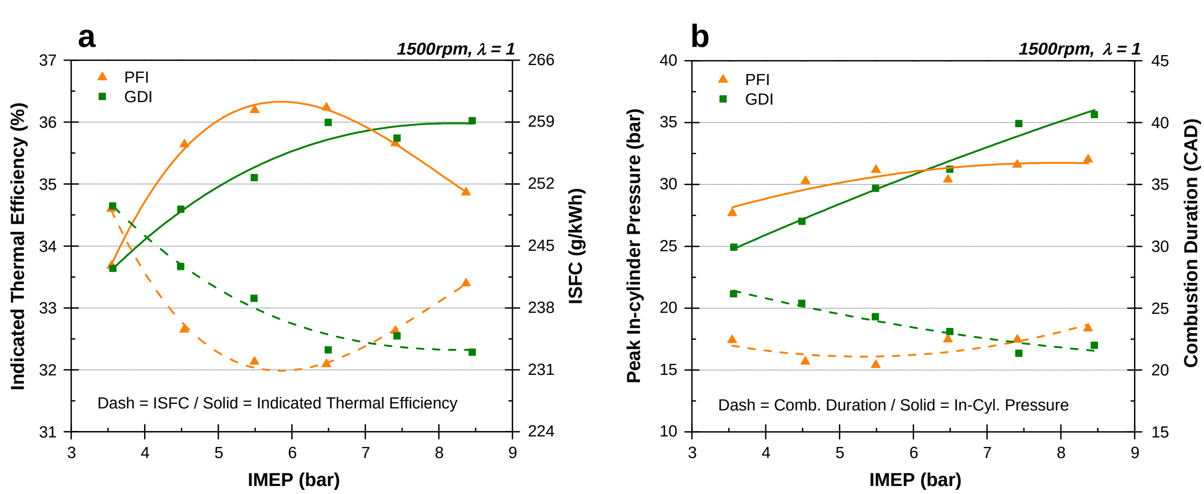

Fig. 2. Indicated thermal efficiency and gravimetric ISFC (a), together with the peak in-cylinder pressure and combustion duration and (b) comparison between gasoline in PFI and GDI with varying load.

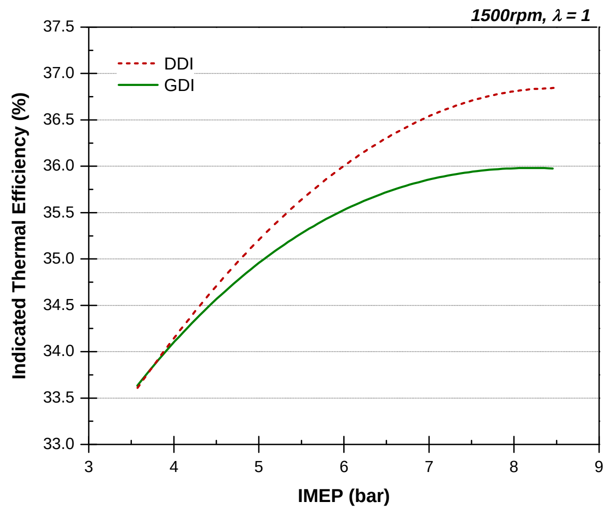

Fig. 3. Indicated thermal efficiency comparison with varying load between DDI and GDI.

3.2. Dual-injection compared to DI

The indicated thermal efficiency has been selected to highlight the varying influence of each element within the dual-injection and DI blends. This is followed by a direct comparison of D25 in DI (D_{25}DI) and dual-injection (G-D_{25}DI) in order to better highlight the differences. Following the indicated thermal efficiency analysis, the impact on fuel consumption and volumetric efficiency is shown. The in-cylinder pressure and resulting mass fraction burned (MFB) data is then analyzed to better explain these trends in efficiency due to combustion.

3.2.1. Indicated thermal efficiency

The improvements in indicated thermal efficiency when using DMF in DI (DDI) compared to gasoline in DI (GDI) are shown in Fig. 3. This is due to greater ignition advance because DMF has a higher octane index (OI) and marginally higher charge-cooling effect, as shown in Table 1. This anti-knock quality is related to its compactness and the reduced branching of hydrocarbons [36]. Even though the two methyl groups are likely to first react during combustion, more energy is required to break the five-ring compound structure of DMF due to the strong oxygen bond. Furthermore, the higher stoichiometric heat of vaporization of DMF (31 kJ/kg_{air}) contributes to greater knock suppression compared to gasoline (25.8 kJ/kg_{air}). However, the lower AFR_{stoich} of DMF (Table 1) suggests that the peak indicated thermal efficiency of DDI is found at a higher IMEP than with gasoline due to the greater throttling requirement and reduced volumetric efficiency with DDI at each load. Nevertheless, this increase in indicated thermal efficiency suggests that if DMF replaces gasoline as the DI element in dual-injection, the indicated thermal efficiency can be theoretically improved over a 'gasoline only' dual-injection strategy.

The indicated thermal efficiency of G-D_{25}DI is compared to its constituent elements (DDI and PFI) in Fig. 4a. For most loads, the indicated thermal efficiency of G-D_{25}DI consistently exceeds that with DDI and PFI. The peak indicated thermal efficiency of 36.2% when using PFI is 0.7% higher (1.8% relative increase) and occurs at a higher load when using G-D_{25}DI (7 bar IMEP, as opposed to 5.7 bar IMEP with PFI). Above 7.8 bar IMEP, the indicated thermal efficiency of G-D_{25}DI then decreases below that with DDI, because of the impact of the PFI element. The efficiency of the PFI element is maintained at the lower loads (3.5–5.5 bar IMEP) and is improved upon, by up to 1.7% (4.8% relatively), at the higher loads. The synergetic reaction has also been seen to benefit compression ignition (CI) engines [37]. Reitz and colleagues claim that the use of pre-mixed early injection homogenous gasoline aids the combustion of late injection stratified diesel in a CI engine. The lower-reactivity gasoline charge helps to advance the start of combustion of the higher-reactivity diesel charge, so that combustion occurs in a smaller volume and results in an increased efficiency. This effect on chemistry for optimal combustion is also supported through the research at the authors' institution on 'Dieseline'; the mixing of gasoline and diesel, rather than through in-cylinder blending [38]. Such combustion benefits with in-cylinder blends are also transferable to G-D_{25}DI in a modern DISI research engine.

In comparison to dual-injection, the results for the equivalent splash blends in DI (D_{25}DI) are presented in Fig. 4b. The constituent elements from Fig. 3 have been added to show the relationship with DMF addition to GDI. Conversely to the positive impact of G-D_{25}DI, the results for D_{25}DI are detrimental to performance compared to DDI and GDI. The indicated thermal efficiency of D_{25}DI reaches a peak at 7 bar IMEP, which is at least 0.3% and 0.8% lower than the peak with GDI and DDI, respectively. Although DMF is immiscible in water [12], its miscibility in gasoline is unknown. However, if like with methanol, DMF requires blending additives to eliminate phase separation problems, the deterioration in performance could be due to the blend separating into DMF-rich phases and hydrocarbon-rich (gasoline) phases [36]. Usually when this occurs, the combustion instability (COV of IMEP) increases. Nevertheless, this non-linear behavior when blending oxygenated fuels with gasoline has been seen in other research [25,39].

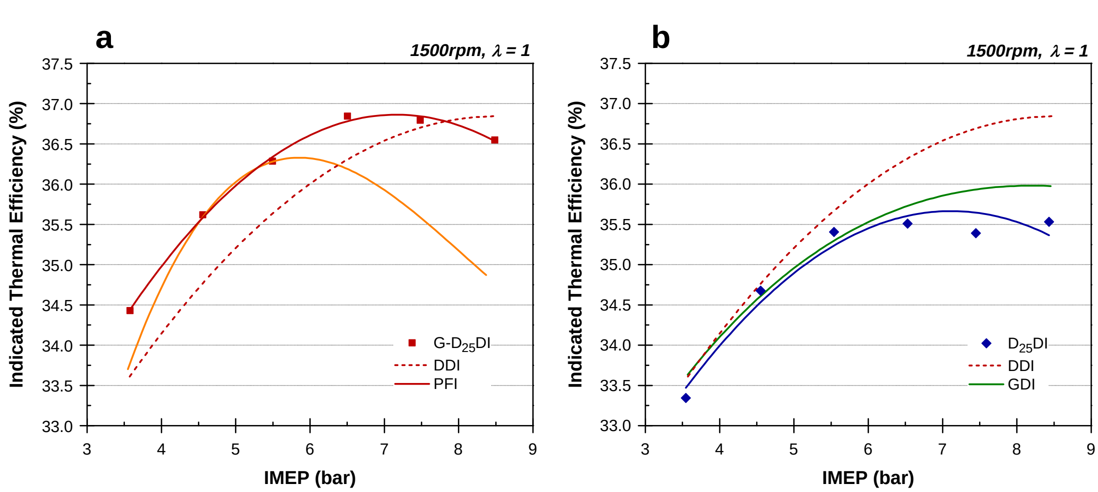

Fig. 4. Indicated thermal efficiency comparison with varying load between G-D_{25}DI and constituents (a) and D_{25}DI and constituents (b).

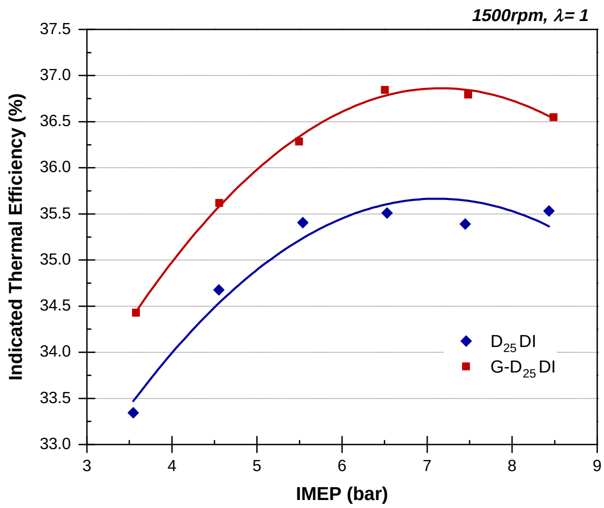

Fig. 5. Indicated thermal efficiency comparison with varying load between D25 in DI (D_{25}DI) and dual-injection (G-D_{25}DI).

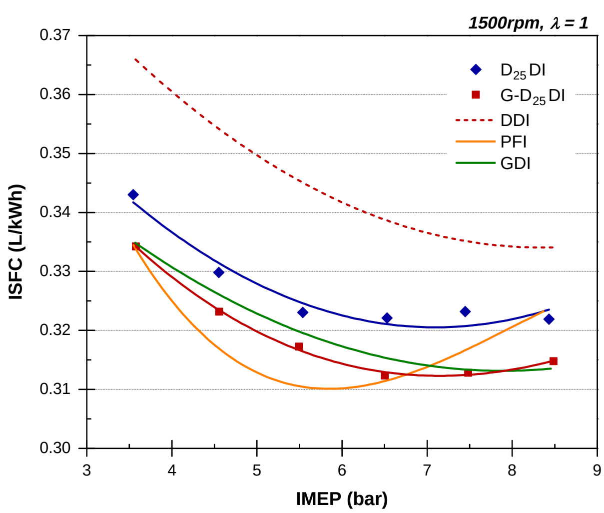

Fig. 6. Volumetric indicated specific fuel consumption comparison with varying load between D25 in DI (D_{25}DI) and dual-injection (G-D_{25}DI) to neat result in DI (DDI) and gasoline in PFI and GDI.

The direct comparison of G-D_{25}DI and D_{25}DI is made in Fig. 5. When using D25, the increase in indicated thermal efficiency from D_{25}DI to G-D_{25}DI is between 0.9% and 1.4% for each load (up to 4% relative increase). Although both modes reach a peak indicated thermal efficiency between 7 and 7.5 bar IMEP, the efficiency at high loads using G-D_{25}DI is affected by the lower efficiency of the PFI component. The rise rate of efficiency from low load (3.5 bar IMEP) to the peak efficiency is also higher with G-D_{25}DI. This reinforces the improved combustion chemistry when using PFI gasoline, a higher volatility fuel, to pre-mix the combustion chamber and help promote the combustion of DMF. This may be either due to the inability of DMF to mix in gasoline, or the synergetic nature of the separate injections when in dual-injection. Nevertheless, thus far, dual-injection shows greater effectiveness in the combustion of D25.

3.2.2. Fuel consumption

The improvements to indicated thermal efficiency, when using D25 in dual-injection, are made more significant when analyzing the effect on the indicated specific fuel consumption, or ISFC. In this case, the volumetric ISFC is presented because this is the usual indicator of fuel consumption, and therefore cost by the end-user. This also promotes the case for DMF because it has a higher density than gasoline (see Table 1). The volumetric ISFC for all test cases is shown in Fig. 6. Immediately, there are two clear advantages with dual-injection (G-D_{25}DI). Firstly, the ISFC is reduced by up to 3.2% (at 7.5 bar IMEP) from the equivalent DI blends (D_{25}DI) and secondly, but also more significantly, the ISFC is reduced by up to 1.2% (at 5.5 bar IMEP) from GDI. This is a very promising result and counterintuitive because the marginally lower volumetric LCV of DMF is offset by the improved efficiency. These results further highlight the synergy of dual-injection and the suitability and usage of low blends of DMF. Although the ISFC with PFI is lower below 6.7 bar IMEP than with G-D_{25}DI, DI is the most widely used injection method and so provides the benchmark in this work. Nevertheless, above 7 bar IMEP, G-D_{25}DI is up to 2.5% more efficient than PFI (at 8.5 bar IMEP).

In order to better understand the reasons for the increases in efficiency with dual-injection, the authors have examined the air flow characteristics and in-cylinder pressure data in the following sections.

3.2.3. Volumetric efficiency

The volumetric efficiency (VE) and intake manifold absolute pressure (MAP_i) for each test case are shown in Fig. 7. The MAP_i is inversely proportional to the pumping work; as the throttle widens with an increase in load, the pumping work is reduced and the MAP_i increases towards ambient pressure. Consistently with load, G-D_{25}DI produces lower VEs than with D_{25}DI, despite both blends (D25) having the same AFR_{stoich}. At 8.5 bar IMEP, the VE of G-D_{25}DI is 2.4% lower than D_{25}DI; less air is consumed because G-D_{25}DI is more effective at converting the fuel energy into useful work, as shown by the indicated thermal efficiency in Fig. 5. The reduced DI injection quantity of G-D_{25}DI results in less charge-cooling and lowers the VE.

The comparable MAP_i between the modes is surprising. This is because the PFI element in G-D_{25}DI reduces the partial pressure of the intake air and necessitates a greater throttle opening. However, in this instance, the throttle demand is similar because the reduced partial pressure is overcome by the increased charge cooling caused by the DMF fuel injection in DI when using G-D_{25}DI. This is due to reduced piston impingement of the DI fuel under G-D_{25}DI which results in a loss of cooling to the surrounding intake air [40]. Therefore, although the overall charge-cooling impact is lower in G-D_{25}DI than in D_{25}DI, the fuel-specific effect is higher. This helps to raise the density of the intake air and partially offsets the need to open the throttle. The reduced piston impingement could also help to explain the improvements in volumetric ISFC of G-D_{25}DI over D_{25}DI as any piston impingement would compromise the mixture quality and result in lower combustion rates.

The benefits of reduced piston impingement with dual-injection are similar to the principle of double pulse or split-injection [41,42]. However, at wide-open throttle (WOT) the overall charge-cooling effect with G-D_{25}DI would be lower, potentially reducing the maximum IMEP achievable (although greatly improved from PFI) compared to D_{25}DI. Therefore, the greatest benefits at WOT with dual-injection would be found when using fuels with a high heat of vaporization but low LCV (increased pulse-width), like ethanol and methanol [43].

3.2.4. In-cylinder pressure

By examining the in-cylinder pressure, the improvements in efficiency seen in Sections 3.2.1 and 3.2.2 can be better explained. The maximum rate of in-cylinder pressure rise or RPR and maximum in-cylinder pressure or P_{max}, are shown in Fig. 8a and b, respectively. Consistently with load, the combustion of G-D_{25}DI

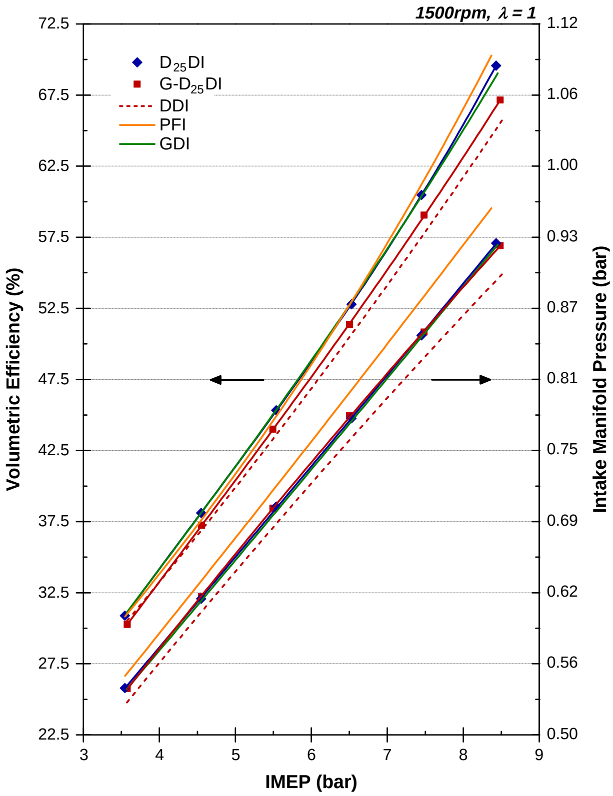

Fig. 7. Volumetric efficiency and intake manifold pressure comparison with varying load between D25 in DI ($D_{25}DI$) and dual-injection ($G\text{-}D_{25}DI$) to neat result in DI (DDI) and gasoline in PFI and GDI.

results in higher RPR and P_{max} values than in PFI, GDI and $D_{25}DI$ modes. Although these values are lower than those in DDI, the addition of only 25% of DMF in DI dramatically improves the combustion of gasoline in PFI. Throughout the load range, the RPR remains high for $G\text{-}D_{25}DI$, which helps to maintain a high indicated thermal efficiency (Fig. 5). However, for $D_{25}DI$, the RPR decreases with loads above 4.5 bar IMEP. At 8.5 bar IMEP, the RPR of $G\text{-}D_{25}DI$ is 0.97 bar/CAD compared to 0.73 bar/CAD with $D_{25}DI$ (33% increase) and 0.65 bar/CAD in PFI (49% increase). The RPR and P_{max} values are primarily related to ignition timing. However, the ignition timings between $G\text{-}D_{25}DI$ and $D_{25}DI$ are similar, as shown in Fig. 9. This suggests that the higher RPR and P_{max} values result from

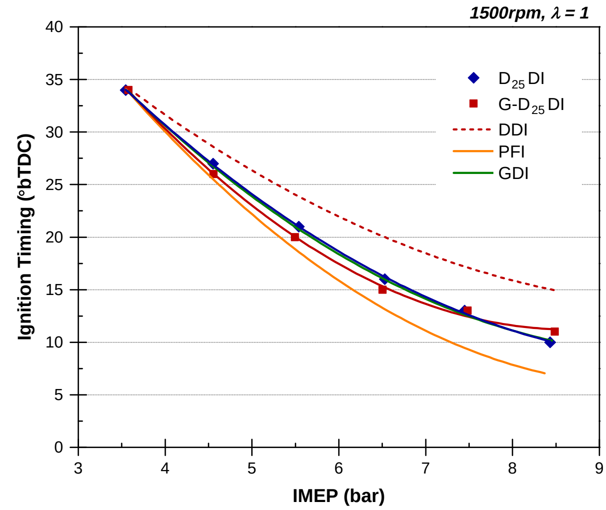

Fig. 9. Ignition timings comparison with varying load between D25 in DI ($D_{25}DI$) and dual-injection ($G\text{-}D_{25}DI$) to neat result in DI (DDI) and gasoline in PFI and GDI.

the higher combustion rates with $G\text{-}D_{25}DI$, as that found with PFI compared to GDI below approximately 6.5 bar IMEP (Fig. 2b). This behavior will be discussed further in Section 3.2.5.

3.2.5. Mass fraction burned

PFI is known to increase the air–fuel mixture homogeneity in the absence of charge-cooling [27]. As seen in Fig. 2b, this results in higher burning rates (lower combustion durations), which increases the combustion pressure (and temperature). In this section, the combustion duration is defined as the 10–90% mass fraction burned (MFB) period in crank angle degrees (CAD) and is herein denoted $CAD_{10–90}$. The MFB is calculated from the heat release analysis using the standard method as described by Stone [35]. For this work, the effect on $CAD_{10–90}$ is shown in Fig. 10. Although DMF, has been shown to have a similar burning velocity to gasoline under laminar, quiescent conditions [15,18,21,22], the high burning speed of DMF during engine combustion has also been shown [16,17,23,24]. Therefore, the combination of PFI and DMF through dual-injection, results in lower $CAD_{10–90}$ than $D_{25}DI$ throughout the entire load range. The largest difference is found at the highest load (8.5 bar IMEP). Here, the $CAD_{10–90}$ for $G\text{-}D_{25}DI$ is 19.7CAD, 13% lower than $D_{25}DI$ (22.8CAD) and 1.7CAD either side of GDI (21.4CAD) and DDI (18CAD). Clearly, the retarded ignition timing between

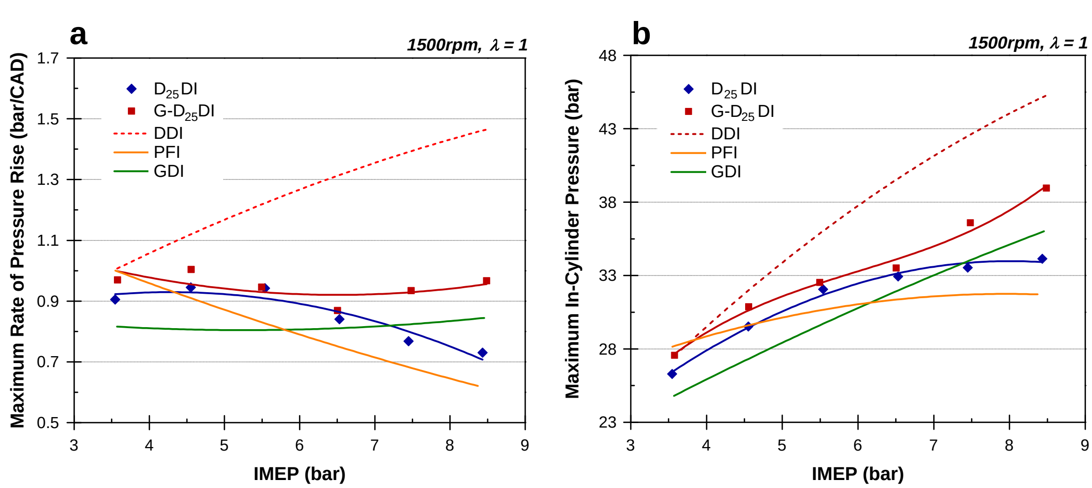

Fig. 8. Maximum rate of pressure rise (a) and maximum in-cylinder pressure and (b) comparison with varying load between D25 in DI ($D_{25}DI$) and dual-injection ($G\text{-}D_{25}DI$) to neat result in DI (DDI) and gasoline in PFI and GDI.

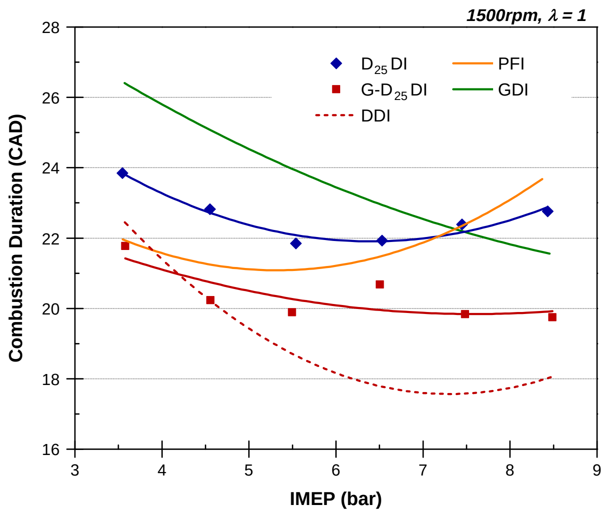

Fig. 10. Combustion duration (10–90% MFB) comparison with varying load between D25 in DI ($D_{25}DI$) and dual-injection ($G\text{-}D_{25}DI$) to neat result in DI (DDI) and gasoline in PFI and GDI.

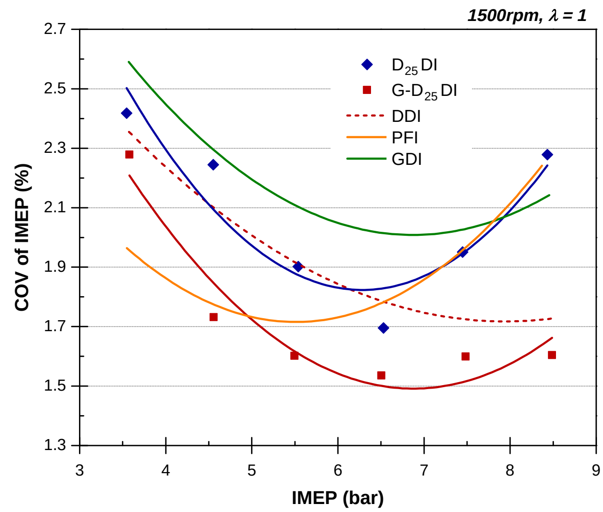

Fig. 12. Combustion stability comparison with varying load between D25 in DI ($D_{25}DI$) and dual-injection ($G\text{-}D_{25}DI$) to neat result in DI (DDI) and gasoline in PFI and GDI.

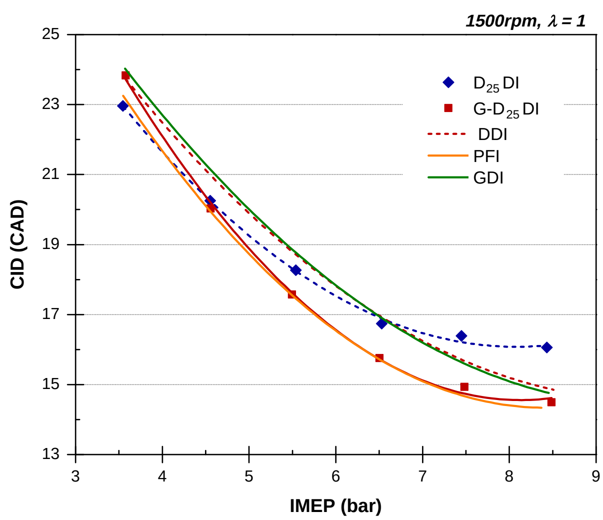

Fig. 11. Combustion initiation duration (CID) comparison with varying load between D25 in DI ($D_{25}DI$) and dual-injection ($G\text{-}D_{25}DI$) to neat result in DI (DDI) and gasoline in PFI and GDI.

4.5 bar and 6.5 bar IMEP with $G\text{-}D_{25}DI$ (see Fig. 9) does not impede the burning rate and the efficiency remains higher than $D_{25}DI$ due to the low CAD_{10-90}. This independence of CAD_{10-90} from ignition timing is also shown by the combustion initiation duration or CID in Fig. 11. This is defined as the CAD between the ignition timing and the location of 5% MFB. Above 4.5 bar IMEP, the CID of $G\text{-}D_{25}DI$ is consistently lower than $D_{25}DI$ (up to 10% at 8.5 bar IMEP) and is comparable to PFI. Therefore, the improved homogeneity of PFI is negligibly affected by the addition of 25% of DMF in DI and helps to combat the usual reliance on ignition timing. In summary, dual-injection is more effective in terms of MFB and these results highlight the advantage of combining different fuelling fuels and modes with high individual burning speeds.

3.2.6. Combustion stability

The combustion stability or coefficient of variation (COV) of IMEP of the various fuelling combinations is shown in Fig. 12. The overall trend between the fuels is similar to the combustion rates in Figs. 10 and 11. Once again, at lower loads (<6.5 bar IMEP) there is a large difference in COV of IMEP between $G\text{-}D_{25}DI$ and $D_{25}DI$. The reduced CAD_{10-90} and CID of $G\text{-}D_{25}DI$ results in up to 20% (relatively) lower COV of IMEP. In fact, above 5 bar IMEP, the COV of IMEP of $G\text{-}D_{25}DI$ is the lowest between all fuelling combinations. Previously in Section 3.2.1, it was suggested that DMF might be immiscible in gasoline as the indicated thermal efficiency above 5.5 bar IMEP was less than that with the constituent elements. Usually, when this occurs (as with fuels like methanol) the COV of IMEP would increase [36]. For $D_{25}DI$ in Fig. 12, the COV of IMEP increases above 6.5 bar IMEP, which is more dramatic than with DDI and GDI. This further suggests that a blending agent might be needed when combining DMF with gasoline rather than using splash blends as is possible with ethanol. Therefore, this will be covered in future investigations by the authors.

4. Conclusions

This study explores the effect of dual-injection – the combination of PFI and DI technology – as a method to better utilize alternative fuel blends with gasoline. In particular, the authors have compared the combustion performance of dual-injection using 2,5-dimethylfuran (DMF); a new biofuel candidate. In this work, dual-injection is defined as the use of gasoline in PFI and DMF in DI. Blends of D25 (25% DMF and 75% gasoline, by volume) have been compared between dual-injection ($G\text{-}D_{25}DI$) to that of DI only ($D_{25}DI$) in order to provide a benchmark for the assessment. Furthermore, the 100% DMF case in DI (DDI) and gasoline in PFI and DI (GDI) have been used to assess the benefits of $G\text{-}D_{25}DI$. The engine tests were performed on a single cylinder DISI engine at various loads, a fixed engine speed of 1500 rpm and at $\lambda = 1$. Based on these experiments, the following conclusions can be drawn:

1. When using gasoline only, PFI produces up to 1% higher indicated thermal efficiencies below 7 bar IMEP compared to homogenous GDI because of improved mixture preparation. However, at higher loads, the greater knock resistance of GDI (due to the charge-cooling effect) allows the indicated thermal efficiency to reach 36%, 1.2% higher than PFI. In both instances, the most proficient fuelling mode produces high in-cylinder pressures and low combustion durations.

2. The indicated thermal efficiency of D25 in dual-injection mode (G-D_{25}DI) exceeds that of its individual components (PFI and DDI) up to 8 bar IMEP. At 6.5 bar IMEP, the indicated thermal efficiency is 0.6% higher than both PFI and DDI, whose efficiencies cross over at this point. Between the two modes, G-D_{25}DI produces consistently higher indicated thermal efficiencies than D_{25}DI (up to 4%; 36.8% at 7.5 bar IMEP).
3. The effectiveness of G-D_{25}DI is reiterated with the volumetric ISFC. There are two clear advantages with dual-injection. First, the ISFC is reduced, by up to 3.2% from D_{25}DI and second, this ISFC is up to 1.2% lower than GDI. The 8% lower LCV of DMF is offset by the higher efficiency of this combustion mode. This behavior is a result of improved mixing and lower piston and wall wetting due to shorter DI injection durations.
4. As found when comparing gasoline in PFI and GDI, the higher volumetric ISFC and indicated thermal efficiencies correlated to higher maximum in-cylinder pressures (P_{max}). At 8.5 bar IMEP, P_{max} for G-D_{25}DI was 4.8 bar higher than D_{25}DI and the maximum rate of pressure rise (RPR) was 0.24 bar/CAD higher. This was the result of higher burning rates (up to 13% lower combustion durations). Also, despite similar ignition timings, the CID with G-D_{25}DI was comparable to PFI and up to 10% lower than in D_{25}DI.
5. The lower combustion durations of G-D_{25}DI produced higher combustion stabilities (lower COV of IMEP) than D_{25}DI. The lowest COV of IMEP was 1.5% at 6.5 bar IMEP, which was lower than its individual components (PFI and DDI).

Ultimately, dual-injection is a very promising technique for optimizing the combustion of low gasoline-DMF blends (D25). As well as providing greater flexibility over external blends using DI only, dual-injection is a more efficient method than homogenous DI, especially at low blend ratios. Clearly, this would benefit economies with a low biofuel supply as the fuel consumption using DMF in dual-injection is, in some instances, more efficient than gasoline in DI.

Future publications intend to disclose the effectiveness of dual-injection with higher gasoline-DMF blends. This will also include the correlation with other oxygenated fuels and the effect of dual-injection blends on gaseous emissions and particulate matter compared to traditional DI.

Acknowledgments

The present work is part of a 3-year research project sponsored by the Engineering and Physical Sciences Research Council (EPSRC) under the Grant EP/F061692/1. The authors would like to acknowledge the support from Jaguar Cars Ltd., Shell Global Solutions and various research assistants and technicians especially Xuesong Wu and Guohong Tian. The authors are also grateful for the financial support from the European Regional Development Fund (EUDF) and Advantage West Midlands (AWM). Finally, the authors would like to acknowledge the support from their international collaborators at Tsinghua University, China.

References

[1] Daniel R, Wang C, Tian G, Xu H, Richardson D. Dual-injection as a knock mitigation strategy using ethanol and methanol. SAE 2012-01-1152; 2012.
[2] Shell, shell energy scenarios to 2050. Signals and signposts. Shell International BV: The Hague; 2011.
[3] Brevitt B. Alternative vehicle fuels, science and environment. London: House of Commons Library; 2002.
[4] White TL. Alcohol as a fuel for the automobile motor. SAE 070002; 1902.
[5] Öner C, Altun S. Biodiesel production from inedible animal tallow and an experimental investigation of its use as alternative fuel in a direct injection diesel engine. Appl Energy 2009;86(10):2114–20.
[6] DIRECTIVE 2009/28/EC. On the promotion of the use of energy from renewable sources. Official Journal of the European Union; 2009.
[7] Curtis B. US Ethanol industry: the next inflection point, 2007 year in review. Energies and Resources Group, Inc., 2008.
[8] Goldemberg J. The challenge of biofuels. Energy Environ Sci 2008;1(5):523–5.
[9] Wang Q, Tian Z. Biofuels and the policy implications for China. Asian-Pacific Econ Lit 2011;25(1):161–8.
[10] AIChE, BP, ABF and DuPont unveil plans for grassroots biofuels plant, chemical engineering progress. American Institute of Chemical Engineers (AIChE): New York; 2007.
[11] Guzman DD. Biobutanol fuel prepares for launch. ICIS Chem Bus 2010;278(7):20–1.
[12] Roman-Leshkov R, Barrett CJ, Liu ZY, Dumesic JA. Production of dimethylfuran for liquid fuels from biomass-derived carbohydrates. Nature 2007;447:982–6.
[13] Dumesic JA, Roman-Leshkov Y, Chheda JN. Catalytic process for producing furan derivatives from carbohydrates in a biphasic reactor. WO 2007/146636. World Intellectual Property: US; 2007.
[14] Wu X, Huang Z, Yuan T, Zhang K, Wei L. Identification of combustion intermediates in a low-pressure premixed laminar 2,5-dimethylfuran/oxygen/argon flame with tunable synchrotron photoionization. Combust Flame 2009;156(7):1365–76.
[15] Wu X, Huang Z, Jin C, Wang X, Zheng B, Zhang Y, et al. Measurements of laminar burning velocities and markstein lengths of 2,5-dimethylfuran-air-diluent premixed flames. Energy Fuels 2009;23(9):4355–62.
[16] Zhong S, Daniel R, Xu H, Zhang J, Turner D, Wyszynski ML, et al. Combustion and emissions of 2,5-dimethylfuran in a direct-injection spark-ignition engine. Energy Fuel 2010;24(5):2891–9.
[17] Daniel R, Tian G, Xu H, Wyszynski ML, Wu Z, Huang Z. Effect of spark timing and load on a DISI engine fuelled with 2,5-dimethylfuran. Fuel 2011;90:449–58.
[18] Tian G, Daniel R, Li H, Xu H, Shuai S, Richards P. Laminar burning velocities of 2,5-dimethylfuran compared with ethanol and gasoline. Energy Fuel 2010;27(7):3898–905.
[19] Tian G, Li H, Xu H, Li Y, Satish MR. Spray characteristics study of DMF using phase doppler particle analyzer. SAE 2010-01-1505; 2010.
[20] Wu X, Daniel R, Tian G, Xu H, Huang Z, Richardson D. Dual-injection: the flexible, bi-fuel concept for spark-ignition engines fuelled with various gasoline and biofuel blends. Appl Energy 2011;88:2305–14.
[21] Wu X, Huang Z, Jin C, Wang X, Wei L. Laminar burning velocities and Markstein lengths of 2,5-dimethylfuran-air premixed flames at elevated temperatures. Combust Sci Technol 2011;158:220–37.
[22] Wu X, Huang Z, Wang X, Jin C, Tang C, Wei L, et al. Laminar burning velocities and flame instabilities of 2,5-dimethylfuran–air mixtures at elevated pressures. Combust Flame 2011;158:539–46.
[23] Tian G, Daniel R, Xu H. DMF – a new biofuel candidate, biofuel production – recent developments and prospects. Bernardes, M.A.S. InTech; 2011.
[24] Wu X, Daniel R, Tian G, Xu H, Huang Z, Richardson D. Dual-injection: the flexible bi-fuel concept for spark-ignition engines fuelled with various gasoline and biofuel blends. Appl Energy 2011;88:2305–14.
[25] Turner D, Xu H, Cracknell RF, Natarajan V, Chen X. Combustion performance of bio-ethanol at various blend ratios in a gasoline direct injection engine. Fuel 2011;90(5):1999–2006.
[26] Kowalewicz A. Methanol as a fuel for spark ignition engines: a review and analysis. Proc IMechE 1993;207(1):43–52.
[27] Zhao F, Harrington DL, Lai MC. Automotive gasoline direct-injection engines. Warrendale: SAE Publications; 2002.
[28] Bromberg L, Cohn DR, Heywood JB. Water based systems for direct injection knock prevention in spark ignition engines. US patent: US 2010121559, United States Patent and Trademark Office; 2010.
[29] Cohn DR, Bromberg L, Heywood JB. Fuel management system for variable ethanol octane enhancement of gasoline engines. US patent: US 2010175659, United States Patent and Trademark Office; 2010.
[30] Cohn DR, Bromberg L, Heywood JB. Direct injection ethanol boosted gasoline engines: biofuel leveraging for cost effective reduction of oil dependence and CO_2 emissions. Cambridge, MA: Massachusetts Institute of Technology; 2005.
[31] Stein RA, House CJ, Leone TG. Optimal use of E85 in a turbocharged direct injection engine. SAE 2009-01-1490; 2009.
[32] Ikoma T, Abe S, Sonoda Y, Suzuki H. Development of V-6 3.5-l engine adopting new direct injection system. SAE 2006-01-1259; 2006.
[33] Wurms R, Jung M, Adam S, Dengler S, Heiduk T, Eiser A. Innovative technologies in current and future TFSI engines from Audi. In: 20th aachen colloquium automobile and engine technology 2011; 2011.
[34] Sandford M, Page G, Crawford P. The all new AJV8. SAE 2009-01-1060; 2009.
[35] Stone R. Introduction to internal combustion engines. 2nd ed. Basingstoke: Macmillan Press Ltd.; 1999.
[36] Hancock EG. Technology of gasoline critical reports on applied chemistry, vol. 10. Blackwell Scientific Publications; 1985.
[37] Reitz RD, Hanson RM, Splitter DA, Kokjohn SL. Engine combustion control via fuel reactivity stratification. United States Patent Office; 2011.
[38] Zhang F, Xu H, Zhang J, Tian G, Kalghatgi G. Investigation into light duty dieseline fuelled partially-premixed compression ignition engine. SAE 2011-01-1411; 2011.

[39] Kar K, Last T, Haywood C, Raine R. Measurement of vapor pressures and enthalpies of vaporization of gasoline and ethanol blends and their effects on mixture preparation in an SI engine. SAE 2008-01-0317; 2008.
[40] Fry M, King J, White C. A comparison of gasoline direct injection systems and discussion of development techniques. SAE 1999-01-0171; 1999.
[41] Yang J, Anderson RW. Fuel injection strategies to increase full-load torque output of a direct-injection si engine. SAE 980495; 1998.
[42] Mittal M, Hung DLS, Zhu G, Schock H. A study of fuel impingement analysis on in-cylinder surfaces in a direct-injection spark-ignition engine with gasoline and ethanol–gasoline blended fuels. SAE 2010-01-2153; 2010.
[43] Daniel R, Wang C, Tian G, Xu H, Richardson D. Dual-injection as a knock mitigation strategy using ethanol and methanol. SAE 2012-01-1152; 2012.