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Experimental investigation on the combustion and emissions characteristics of 2-methylfuran gasoline blend fuel in spark-ignition engine

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HIGHLIGHTS

- 2-Methylfuran gasoline blend and ethanol gasoline blend were compared in a SI engine.
- Combustion duration, thermal efficiency and regulated emissions were studied.
- Compared with E10, BSFC and COV of IMEP can be improved by M10 blend.
- M10 are similar to E10 and superior to gasoline in terms of HC and CO emissions.
- NO_X emissions increase is found by using M10 blend fuel.

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ABSTRACT

Currently, 2,5-dimethylfuran (DMF) has already been extensively studied as a novel potential gasoline substitute. With its improved reaction sequences, another main molecule transformed from fructose has also aroused worldwide interest, which is known as 2-methylfuran (MF). MF has similar energy density and knock suppression ability to DMF. However, little is known about its behavior in spark-ignition (SI) engines, especially when it is used as a gasoline additive. Therefore, focus was given on the combustion and emissions characteristics of 10% volume fraction 2-methylfuran gasoline blend fuel (M10) in this work, which was investigated experimentally in a single-cylinder four-stroke SI engine at various engine speeds (800–1800 rpm in 200 rpm intervals) and wide open throttle (WOT). The in-cylinder combustion process as well as engine performance of M10 were compared with gasoline and the same proportion ethanol gasoline blend fuel (E10) under gasoline maximum brake torque (MBT) spark timing and stoichiometric air-fuel ratio. Results of engine tests show that M10 produces relatively high in-cylinder peak pressure and temperature, which is mainly attributed to its consistently shorter combustion duration. Compared with engine performance of E10, the output torque and brake power increase slightly with less brake specific fuel consumption when M10 is used. Lower regulated gas emissions of hydrocarbons (HC) and carbon monoxide (CO) can be found for both E10 and M10 blend. In addition, more nitrogen oxides (NO_X) emissions are generated from M10 due to its higher combustion temperature.

Abbreviations: aTDC, after top dead center; bTDC, before top dead center; CA50, crank angle of 50% mass fraction burned; CAD, crank angle degree; CD, combustion duration; CO, carbon monoxide; COV, coefficient of variation; DMF, 2,5-dimethylfuran; E10, 10% volume blending ratio ethanol gasoline; GHG, greenhouse gas; HC, hydrocarbon; IMEP, net indicated mean effective pressure (calculated over 720 °CA); LHV, lower heating value; M10, 10% volume blending ratio 2-methylfuran gasoline; MF, 2-methylfuran; MFB, mass fraction burned; MON, motor octane number; NO_X, oxides of nitrogen; PFI, port-fuelinjection; BSFC, brake specific fuel consumption; SI, spark ignition; RON, research octane number; WOT, wide open throttle.

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1. Introduction

Rapid development of industrialization and modernization has led to excessive consumption of petroleum, which is found to be the major source of greenhouse gas (GHG) emissions [1,2]. Today, it is reported that around 62.3% of oil is consumed by transport sector alone [2]. Therefore, it is necessary to search for alternative fuels to reduce the dependency on petroleum-based fuels and to lower GHG emissions. Currently, some substitute fuels have already been used for powering an engine, such as hydrogen, natural gas, alcohols and so on [3–5]. Among these fuel candidates, liquid biofuels turn out to be highly competitive due to their higher energy density, better security, and relatively low cost [6–8]. Incentive policies have also been formulated in many countries to promote the application of biofuels. For instance, all EU members have to realize the mandatory 10% minimum target on use of biofuels in transportation by 2020 [9]. In order to replicate the success in Brazil, tax incentives have also been provided to increase the bioethanol proportion in gasoline in the US [10,11]. By far, the most widely used biofuel seems to be bioethanol, which is mainly attributed to its abundant renewable sources, high octane numbers and good compatibility with internal combustion engines [12–14]. Despite of these benefits, bioethanol has several limitations compared with gasoline, including: low energy density (reducing driving distance), high latent heat of vaporization and low vapor pressure (making engine cold start difficult), and water miscibility [15,16]. Thus, looking for superior gasoline substitutes has become highly pronounced in the present context.

Recently, furan-based fuels have been brought into the sight of fuel researchers since the breakthrough of its production methods, which were reported by the *Nature* and *Science* [17,18]. Dumesic et al. have announced a two-phase reaction sequence to produce DMF from fructose and glucose [17,19]. Then, Shaohua Zhong et al. firstly tested the combustion and emissions of DMF in a direct-injection spark-ignition engine [20]. Experimental results revealed its higher knock resistance and similar engine performance compared with research octane number (RON) 95 gasoline, thus making it an attractive potential gasoline candidate. In order to further improve the transformation of fructose to furans, novel selective catalytic reaction sequences without requiring external hydrogen source have been developed. During this process, another main product is transformed from fructose, known as 2-methylfuran [21,22]. This molecular is even more compact than DMF, whose main fuel properties are presented in Table 1. As shown, the RON number of MF is higher than that of DMF and gasoline, which allows SI engines to operate at higher compression ratios without knock combustion. Compared with conventional gasoline substitute, bioethanol, some properties of MF are more favorable for SI engines. For instance, MF is almost insoluble in water, thus making its gasoline blend more stable. Much lower latent heat of vaporization for MF (358.4 kJ/kg) as against bioethanol (840 kJ/kg) can also avoid the engine cold-start problem. Most of all, the energy density of MF (28.5 MJ/L) is almost 34% higher than that of bioethanol (21.3 MJ/L), which can significantly help to decrease the engine fuel consumption.

Since its superior physicochemical properties, worldwide interest has been triggered in the potential of MF. Matthias Thewes et al. experimentally studied the impact of MF on in-cylinder spray formation and evaporation as well as engine performance in a direct-injection SI engine [25]. The results showed MF had shorter evaporation duration, excellent combustion stability and lower HC emissions compared with ethanol and RON 95 gasoline, all of which indicated that MF was a promising potential biofuel candidate. After that, Chongming Wang et al. compared the performance and emissions of MF with DMF, ethanol and gasoline in a SI engine [24]. Experimental results furthermore highlighted that MF is more competitive than DMF, especially for its excellent combustion stability and knock resistance. Meanwhile, a drawback of higher NO_X emissions for MF was also noticed in their research. At present, biofuels are most often used as additives for gasoline and diesel because of their inadequate yields and fuel supply infrastructures. Ethanol gasoline blend is a typical and successful example, whose combustion and emissions behavior in SI engines have already been reported by many publications [27–29], which are even more than that of pure ethanol. The engine performance of DMF gasoline blends has also been experimentally investigated [30,31]. However, currently researches on using MF as a biofuel are concentrated on its pure substance. Still little has been known about the effect of using it as a gasoline additive on engine performance.

In this paper, the combustion characteristics and emissions of 10% MF gasoline (M10) blend fuel is examined in a single cylinder four-stroke SI engine with port-fuel injection. Experiments were conducted at constant load of WOT as engine speed ranging from 800 to 1800 rpm. The test results were compared with that of RON 97 gasoline and the same proportion ethanol gasoline blend fuel (E10). In-cylinder pressure, engine output torque, fuel consumption as well as regulated gas emissions (CO, NO_X and HC) were mainly measured and analyzed. In the following sections, experimental setup and procedures are explained, and then results are discussed. Finally, major conclusions are summarized in the last section.

Table 1
Base fuel properties [23–26].

	Gasoline	Bioethanol	MF	DMF
Molecular formula	C_2–C_{14}	C_2H_6O	C_5H_6O	C_6H_8O
Lower heating value (MJ/kg)	42.9	26.8	31.2	33.27
Lower heating value (MJ/L)	31.9	21.4	27.63	29.55
Research octane number (RON)	96.8	109	103	101
Motor octane number (MON)	85.7	90	86	88
Stoichiometric air-fuel ratio	14.46	9	10.05	10.75
Heat of vaporization (kJ/kg)	373	919.6	358	330.5
Reid vapor pressure (kPa)	32.8	16	18.5	13.4
Density @ 20 °C (kg/m^3)	744.6	790.9	913.2	889.7
H/C ratio	1.795	3	1.2	1.33
Gravimetric oxygen content (%)	0	34.73	19.49	16.67
Initial boiling point (°C)	32.8	78	64	94
Solubility in water (vol.%)	Negligible	Miscible	0.3	0.26

2. Experimental section

2.1. Engine and instrumentation

The experiments were conducted on a single cylinder, port-fuel injection, four-stroke SI engine, of which specifications are given in Table 2. Fuel injection duration as well as spark timing of the engine was adjusted through MoTeC M400 ECU manager software. The air-fuel mixture equivalence ratio was determined from ECM LambdaCAN Module using a wideband lambda sensor of resolution 0.001, uncertainty ±0.8% and response time within 0.15 s. The engine was coupled with a direct current dynamometer to maintain the desired test speeds with an accuracy of ±1 r/min. Engine load applied through the dynamometer was measured by a ZEMIC H3-C3-200 kg-3B load cell with uncertainty of ±0.5%.

A Kistler 6041A water-cooled pressure transducer was flush mounted in the combustion chamber to measure in-cylinder pressure. The signals were passed to a Kistler 5018 charge amplifier and acquired by a National Instruments PC-6123 data acquisition card. Pressure acquisition was triggered using a 0.5 crank angle degree (CAD) resolution photoelectric encoder coupled to the

Table 2
Engine specifications.

Engine type	Single cylinder, 4-stroke	
Bore × stroke (mm)	80 × 100	
Sweep volume (*L*)	0.5	
Compression ratio	9:1	
Valve mechanism	Dual-overhead camshafts, 2-valve	
Combustion system	Port-fuel injection (3.5 bar)	
Valve	Intake valve	Exhaust valve
Open timing (°bTDC)	10	213
Close timing (°aTDC)	250	17