THE EFFECTS OF EXHAUST GAS RE-CIRCULATION AND INJECTION TIMING ON COMBUSTION PERFORMANCE AND EMISSIONS OF BIODIESEL AND ITS BLENDS WITH 2-METHYLFURAN IN A DIESEL ENGINE

by

Helin XIAO**, Xiaolong YANG, Ru WANG, Shengjun LI, Jie RUAN, and Hongling JU

[a] Hubei Key Laboratory of Advanced Technology for Automotive Component, Wuhan University of Technology, Wuhan, China
[b] Hubei Collaborative Innovation Center for Automotive Components Technology, Wuhan University of Technology, Wuhan, China

Original scientific paper
https://doi.org/10.2298/TSCI190112304X

In this study, the influences of injection timing and exhaust gas re-circulation on combustion and emissions characteristics of biodiesel/2-methylfuran blends are investigated on a modified water-cooled 4-cylinder four-stroke direct injection compression ignition engine. The experimental conditions are, respectively, to adjust injection timing and exhaust gas re-circulation ratio at 0.38 MPa break mean effective pressure with the engine speed at 1800 rpm constantly. With injection timing in advance, the peak cylinder pressure rose while maximum heat release rate first decreased and next slightly raised. Ignition delay and brake specific fuel consumption reduced first and then raised while combustion duration and break thermal efficiency had the opposite trend. The NO_x emissions increased, and HC emissions first reduced significantly and then slightly increased, while 1,3-butadiene and acetaldehyde emissions presented a reduction tendency. As exhaust gas re-circulation ratio increased gradually, ignition delay as well as combustion duration was prolonged. brake specific fuel consumption increased and break thermal efficiency declined. HC, CO, 1,3-butadiene, and acetaldehyde emissions raised while NO_x emissions reduced significantly. Biodiesel could behave well in a Diesel engine and thus a feasible alternative fuel for diesel. Moreover, methylfuran addition into biodiesel could raise break thermal efficiency and the break thermal efficiency of BM20 is higher than BM10. However, both BM10 and BM20 appeared a combustion deterioration when injection timing at 2.5 °CA before top head center.

Key words: *Diesel engine, biodiesel/methylfuran blends, combustion performance, regulated and unregulated emissions*

Introduction

Currently, the alternative fuels applied in a Diesel engine have become an effective way to deal with energy issues caused by global warming and serious shortage of fossil fuel. Several eco-friendly alternative fuels have been investigated in engine so far, such as alcohols, biodiesel and other biomass fuels [1-3].

Biodiesel, the alternative fuel of diesel, composed of alkyl monoesters of fatty acids come from animal fats and plant oil [4]. There are many researches about the application of

* Corresponding author, e-mail: hlxiao_qcxy@whut.edu.cn

biodiesel in Diesel engine because of its renewability, non-toxic, and sulfur-free property. Moreover, because of its analogous properties to diesel, direct injection compression ignition (DICI) engine fueled with biodiesel as well as its blends without modifying the engine structure [5]. Combustion performance of biodiesel and its blends of 70%, 20% as well as 5% with basic diesel were quite analogous to that of pure diesel in a DICI engine, while emission characteristics of biodiesel were better compared with traditional diesel [6]. In comparison with conventional diesel, the fuel-borne oxygen within biodiesel would facilitate the process of combustion, leading to a more complete combustion. Meanwhile, the particular matter (PM), HC as well as CO reduced while the NO_x increased in a Diesel engine [7]. Biodiesel applied in DICI engine produced less CO as well as HC and more NO_x emissions compared with diesel [8]. Therefore, considerable NO_x emissions produced from the blends added into biodiesel might be regarded as an issue using biodiesel on a Diesel engine. However, there are some converse experiment results about NO_x emissions. Antonopoulos *et al.* [9] investigated combustion performance and emissions in a direct injection diesel engine fueled with vegetable oils or biodiesels of various origins. They found that NO_x emissions occurred a slight reduction.

As a kind of clean and efficient alternative fuel, alcohols have already been investigated comprehensively. Among the alcohols, ethanol has been studied widely due to its outstanding properties. Due to high octane number, high-oxygen, and reproducibility, ethanol is usually applied on engines while the properties of water solubility, low energy density, and high latent vaporization heat limited its application [10]. Armas *et al.* [11] investigated emissions characteristics of diesel-ethanol blends on a CI engine. The results indicated PM emissions significantly reduced while the rest gaseous emissions (CO, HC, NO_x) had no substantial increase after fueled diesel-bioethanol blends. Ahmed comparatively investigated combustion performance and emissions characteristics on 15% diesel-ethanol and 10% diesel-ethanol on a DICI engine. He found that compared to conventional diesel, NO_x emissions slightly increased for the two blends, while PM emissions respectively reduced 41% and 27% for 15% and 10% diesel-ethanol [12]. However, there are some problems with the application of diesel-ethanol blends, such as worse lubricity, uneven mixing. Biodiesel might be a suitable additive in improving diesel-ethanol blends. Abhishek *et al.* [13] conducted an experimental study on combustion and emissions characteristics of diesel-biodiesel-ethanol on a CI engine. They concluded that D35B50E15 blend with 15% ethanol behaved better engine performance characteristics with 21.17% increase in brake thermal efficiency (BTE) at full load while most of gaseous emissions were relatively lower.

As furan-based fuels, 2,5-dimethylfuran (DMF) as well as 2-methylfuran (MF) might be promising alternative for CI engines with the development of production technology [14, 15]. Currently, the combustion performance and emission characteristics of DMF have been widely investigated [16-20]. Zheng *et al.* [21, 22] and Chen *et al.* [23] discovered the addition of DMF would significantly decrease soot emissions on a CI engine. Zhang *et al.* [24] compared combustion performance and emissions characteristics of diesel, 20% diesel /n-pentanol and 20% diesel/DMF fuels on diesel engines without exhaust gas re-circulation (EGR). They found that 20% DMF/diesel blends emitted less soot and total hydrocarbon (THC) emissions, while produced more NO_x emissions compared to diesel. Zheng *et al.* [25] investigated engine characteristics of biodiesel/ethanol, biodiesel/DMF as well as biodiesel/n-butanol under dual fuel reactivity controlled compression ignition (RCCI) mode on a CI engine. They concluded that DMF addition into biodiesel prolonged ignition delay while shorten combustion duration. Moreover, compared with neat biodiesel, soot emissions significantly reduced while both THC and NO_x emissions increased for biodiesel/DMF blend.

However, there are little literature researches on the use of MF. Researchers [26, 27] have uncovered and proposed an efficient way of transforming fructose into MF. Moreover, MF may be a kind of suitable fuel output by the approach because the fructose is considerable and reproductive [28]. The MF with better physicochemical properties is more attractive than ethanol. Wei *et al.* [29] investigated combustion performance and emissions characteristics of gasoline-MF blends on a spark ignition (SI) engine and thus disclosed low fraction MF addition into gasoline behaved better than gasoline-MF blends. Wang *et al.* [30] investigated engine performances on a DICI engine fueled with ethanol, DMF, MF as well as gasoline. They found MF behaved better burning performance than DMF or gasoline, but higher NO_x emissions from MF remained to be solved. In this study, the impacts of biodiesel, biodiesel-MF as well as biodiesel-ethanol on combustion characteristics, HC and NO_x at various injection timings and EGR ratios under low engine load on a diesel engine were studied. Unregulated emissions could produce a great threat to human health and thus other living creatures [10]. Therefore, unregulated emissions like1,3-butadiene as well as acetaldehyde was investigated. Based on the aforementioned investigation, suitable injection timing and EGR ratio could be applied to improve the process of combustion and emission for biodiesel/MF blends.

Experiment

Engine layout and instrumentation

The tests were conducted on a modified DICI engine with water-cooled, 4-cylinder, four-stroke, which was equipped with a common rail fuel injection system as shown in fig. 1. The engine main specifications were illustrated in tab. 1. An eddy current (EC) dynamometer

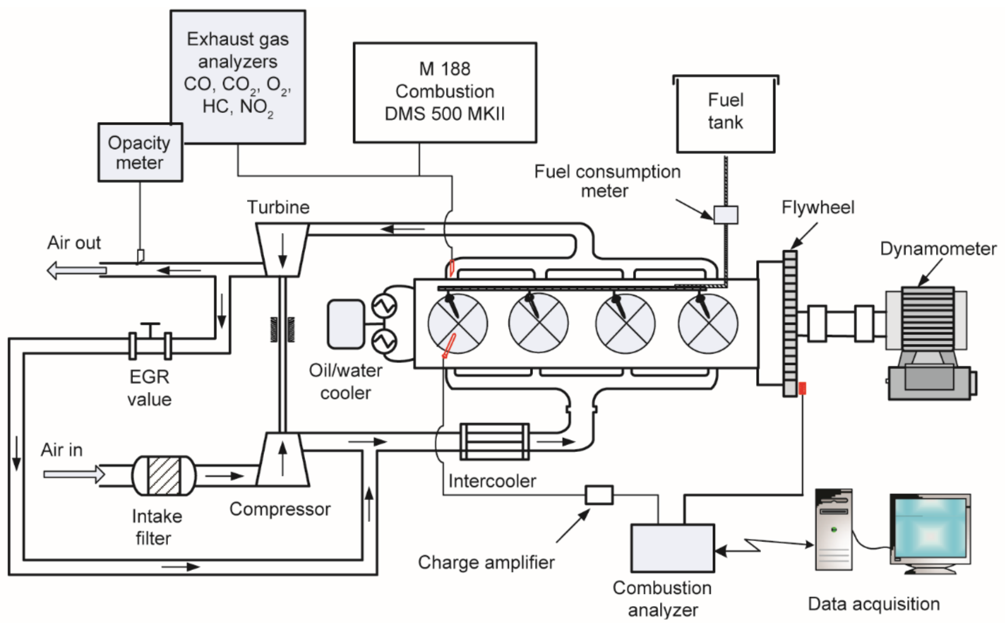

Figure 1. Illustration of engine layout and instrument

Table 1. Engine specification [28]

Type of engine	4-cylinder, four-stroke, water-cooled
Type of ignition	Compression ignition
Method of starting	Electric start
Bore	96 mm
Initial injection	7.5 °CA bTDC
Stroke	103 mm
Displacement	2982 cc
Compression ratio	17.5
Rated power	85 kW
Rated speed	3200 rpm

Table 2. Uncertainties of the acquired quantities [33]

Measurement	Uncertainty [%]
Air-flow meter	±0.5
In-cylinder pressure	±1.0
Torque	±1.0
Fuel-flow meter	±1.0
BTE	±1.72
BSFC	±1.93
EGR	±0.5

was used to keep engine speed constantly at 1800 rpm (±5 rpm) (the engine operated stably at 1800 rpm) and regulate engine torque output. Using an electrical control unit (ECU) to control and monitor the engine operating parameters, such as injection fuel quantity and injection timing. Moreover, high pressure cooled EGR system was applied and EGR ratio was controlled via combination control of air-throttle and EGR valve, while EGR ratio was defined as the percentage of CO_2 concentration in intake air as a percentage of the CO_2 concentration in exhaust gas. In-cylinder pressure signals are collected via a Kistler 6025 C pressure transducer installed in cylinder head. These signals first transmitted to a charge amplifier and next to a CB-466 combustion analyzer. A hundred continuous cycles of samples gauged with an interval of 0.25 °CA so as average measurements of data and then in-cylinder pressure curve could acquire. Moreover, heat release rate (HRR) was calculated according to the first thermodynamic law [31, 32]. The temperature (25±1 °C) and intake air pressure（0.1 MPa）were, respectively, adjusted by an air conditioning system and an additional compressor.

Using an AVL gas analyzer to measure gaseous emissions while the accuracy of HC and NO_x are 1 ppm and 0.1%, respectively. Various unregulated emissions were gauged *via* a gas chromatograph (GC) which were accurate to 0.1 ppm. Furthermore, relative uncertainties about primary measure were shown in tab. 2.

Test fuels and experiment procedures

The fuels used in this experiment include neat biodiesel as well as biodiesel/MF, with relative properties shown in tab. 3. The MF mixed with biodiesel in a fraction of 10% and 20% (by mass), marked as BM10 and BM20, respectively. Meanwhile, pure biodiesel was used for comparison.

This experiment was conducted under 30% engine load corresponding to break mean effective pressure (BMEP) at 0.38 MPa. The thermal value in every cycle should be consistent to better investigate the impact of fuels properties on combustion performance and emissions. Therefore, this experiment needed to adjust injected fuel mass to keep the same energy input due to the different low heating values of various test fuels. In addition, the experiment was operated at various EGR ratios and injection timings. Firstly, the EGR valve should be closed and the start of injection (SOI) time was adjusted from 2.5 to 22.5 °CA bTDC with an increment of 5 °CA. Secondly, the SOI kept at 7.5 °CA bTDC, while EGR ratio were controlled at 0%, 6%, 17%, 23%, and 30%, respectively. In order to ensure test data reliable and repeatable, engine was first warned to a stable situation under each operating condition. Meanwhile, cooling water temperature and lubricating oil were respectively maintained at 86 °C and 85 °C. Moreover, after changing to another fuel, the engine needed to operate around 15 minute before data acquirement to guarantee new added fuel not polluted by the remains of the last operating condition.

Table 3. Properties of diesel, biodiesel, bioethanol, MF, and gasoline [5, 28]

Property	Ethanol	MF	DMF	Diesel	Biodiesel	Gasoline
Motor octane number	90	86	88	–	–	85.7
Research octane number	109	103	101	20-30	–	96.8
Octane number	108	–	119	–	–	90-99
Cetane number	8	9	9	45	51	10-15
Density at 20 °C [kgcm^{-3}]	790.9	913.2	889.7	826	871	744.6
Oxygen content [% mass]	34.78	19.51	16.67	0	10.8	0
Lower heating value [MJkg^{-1}]	26.9	31.2	33.7	42.5	37.5	42.9
Stoichiometric air/fuel ratio	8.95	10.05	10.79	14.3	12.54	14.7
Initial boiling point [°C]	78.4	64.7	92	180-370	–	32.8
Viscosity at 40 °C [mm^{2}s^{-1}]	1.2	–	–	2.86	4.38	–
Evaporation heat at 25 °C [kJkg^{-1}]	919.6	358	332	270-301	300	373
Sulfur content [%]	–	0	0	0.2	$<$0.005	–
Water solubility at 20 °C [wt.%]	Miscible	N	N	N	N	N

Note: N-Negligible

Result and discussion

Combustion performance

Figure 2(a) displayed the variations of ignition delay which was defined as the CA from SOI to10% of total heat release. From 2.5 to 22.5 °CA bTDC, the ignition delay first decreased and then raised with SOI in advance. When injection timing at 2.5 °CA bTDC, the atomization of most test fuels occurred after TDC in the expansion stroke while both in-cylinder pressure and temperature were low under this condition, prolonged ignition delay. With injection timing in advance, the time of fuels atomization increased so as to easily reach the condition of auto-ignition, shorten ignition delay. When injection timing was up to 22.5 °CA bTDC, both in-cylinder pressure and temperature were low so that fuel atomization was suppressed, extended ignition delay. Besides, with MF fraction increasing, ignition delay of biodiesel-MF blends was gradually prolonged, which could be attributed to high latent heat of evaporation and auto-ignition temperature.

Figure 2(b) showed the variations of combustion duration which was defined as crank angle from 10% to 90% of total heat release. With injection timing in advance, combustion duration presented the opposite trend to ignition delay. The ignition delay was prolonged so as to expand premixed combustion ratio while diffusion combustion ratio was significantly reduced, shorten combustion duration. As is known, oxygen in fuels can promote the process of chemical reaction to raise combustion rate. Compared with biodiesel, combustion duration of BM10 and BM20 were shorter because of higher oxygen content. Moreover, combustion duration of BM20 is shorter compared to BM10. On the one hand, higher oxygen content promoted the process of combustion during the combustion duration. on the other hand, the longer ignition delay of BM20 produced more premixed gas so as to make higher combustion temperature and pressure which could boost combustion, shorten combustion duration.

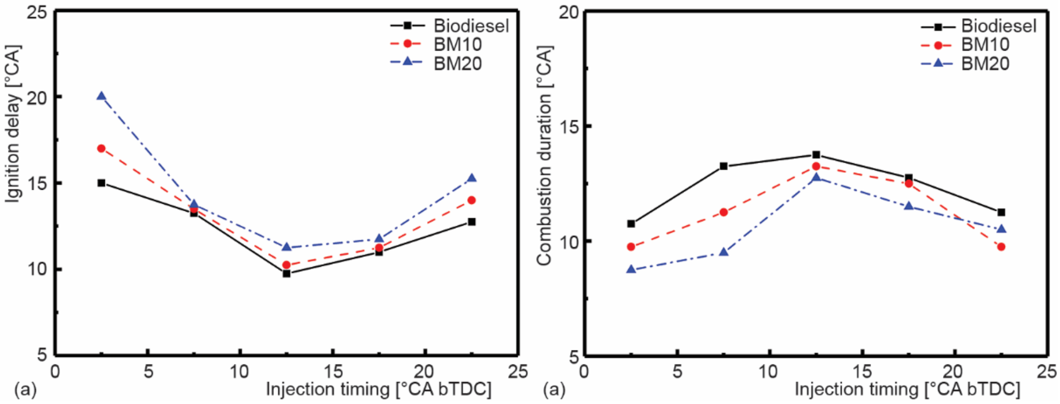

Figure 2. Effects of injection timing on (a) ignition delay and (b) combustion duration

Figure 3(a) displayed the variations of ignition delay with various EGR ratios. With the rise of EGR ratio, ignition delay of the whole test fuels raised. This trend might be explained by following reason. The increasing EGR ratio gradually decreased oxygen concentration in the cylinder, which made it more difficult to reach the condition of auto-ignition, thus prolonged ignition delay. Under all EGR ratios, both BM10 and BM20 had longer ignition delay than biodiesel because low cetane number of MF addition into neat biodiesel raised auto-ignition temperature of the blends.

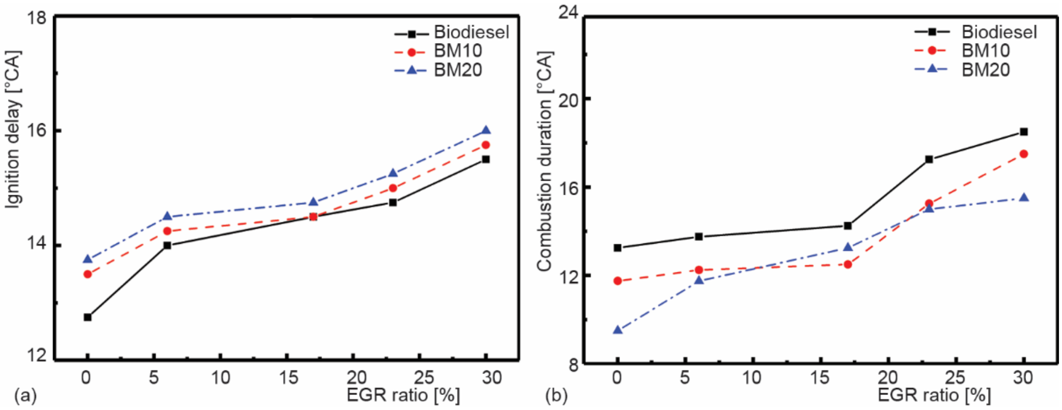

Figure 3. Effects of EGR ratio on (a) ignition delay and (b) combustion duration time

Figure 3(b) showed the variations of combustion duration under various EGR ratios. With the rise of EGR ratio, combustion duration of all the test fuels rose. This tendency could be mainly described as following aspects. With EGR ratio increasing, on the one hand, the premixed combustion ratio raised while diffusion combustion proportion reduced relatively, leading to the reduction of combustion duration. On the other hand, the oxygen concentration in the cylinder decreased and thus reduced combustion rate. In addition, the inert gas from EGR increased and retarded the chemical reaction rate, which prolonged combustion duration. Moreover, the later factor was dominant under the operating condition.

In-cylinder pressure and HRR for three fuels at various injection timings as well as EGR ratios shown in fig. 4. From fig. 4, peak in-cylinder pressure rose and appeared ahead with the advance of injection timing from 2.5 to 22.5 °CA bTDC. The injection timing in advance could make injected fuel adequate time to mix with air. Therefore, there were more premixed gases in the cylinder when condition of auto-ignition was reached, led to an increase of peak value of in-cylinder pressure because of considerable mixture burning together in the cylinder. As for the injection timing at 2.5 °CA bTDC, most of fuels burned aTDC in the expansion stroke, which caused lower proportion of constant volume combustion so as to significantly decline peak value of in-cylinder pressure. Liu *et al.* [34] made a similar experiment conclusion. The maximum HRR first decreased and then raised with injection timing in advance. The advance of injection timing could lead to more mixture burning in premixed phase so that maximum HRR increased. When injection timing at 2.5 °CA bTDC, in-cylinder combustion temperature was slightly low because most fuels burned during expansion stroke. However, due to the too long ignition delay, plenty of premixed gases burned together and instantly released considerable heat. Therefore, the maximum HRR was slightly high at this condition.

With EGR ratio increasing, peak value of in-cylinder pressure gradually decreased. At the same time, the appearance of peak in-cylinder pressure corresponding to the CA was

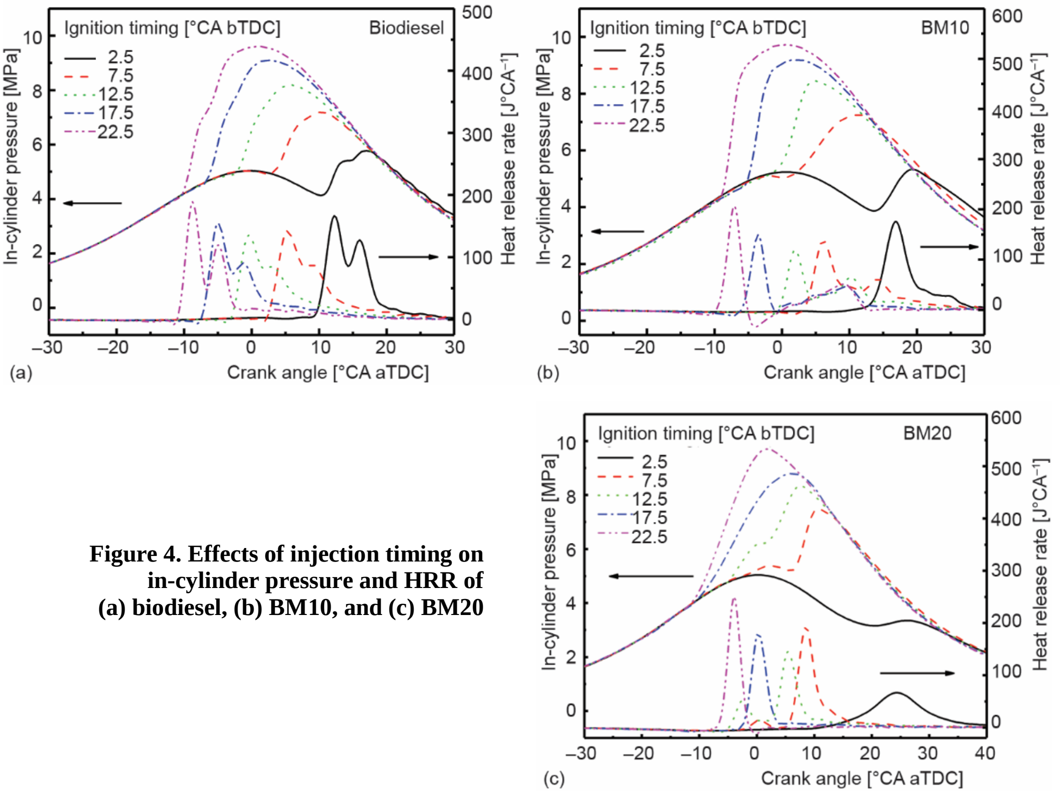

Figure 4. Effects of injection timing on in-cylinder pressure and HRR of (a) biodiesel, (b) BM10, and (c) BM20

gradually deviated from the TDC, as shown in fig. 5. The tendency could be explained by following reasons. Firstly, with EGR ratio increasing, the concentration of in-cylinder oxygen decreased, prolonging ignition delay while declined combustion rate. Meanwhile, the inert gas from EGR increased and thus retarded chemical reaction rate, led to the reduction of peak in-cylinder pressure. Secondly, the exhaust gases raised specific heat capacity of intake gas, which decreased peak value of in-cylinder pressure. As for maximum HRR, with the rise of EGR ratio, the ignition delay was slightly prolonged and more premixed gases burned so that the maximum HRR gradually increased. However, combustion rate was significantly declined with EGR ratio further increasing, led to the reduction of maximum HRR.

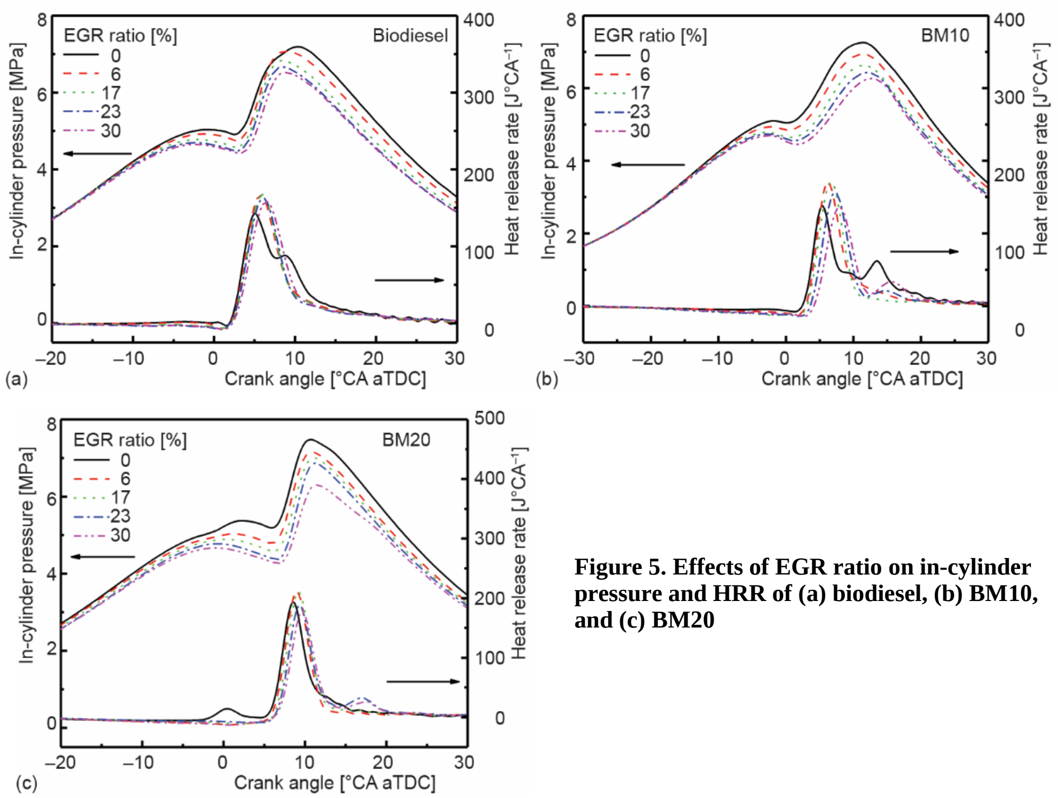

Figure 5. Effects of EGR ratio on in-cylinder pressure and HRR of (a) biodiesel, (b) BM10, and (c) BM20

Brake specific fuel consumption and brake thermal efficiency

Figure 6 presented effects of injection timing on BSFC and BTE for the whole test fuels. With injection timing in advance, the BSFC of the whole fuels decreased first and then increased while BTE had an opposite trend. When injection timing at 2.5 °CA bTDC, most of fuels burned away from TDC and led to burning deterioration, thus increased BSFC while reduced BTE. With injection timing in advance, the combustion process was gradually improved. When injection timing at 22.5 °CA bTDC, most of fuels burned before TDC and thus increased compression negative work, resulted in the rise of BSFC and the reduction of BTE. Besides, compared to biodiesel, the BSFC of biodiesel-MF blends were lower. MF addition into biodiesel improved the atomization effect of blends while raised oxygen content in fuels, which could boost the combustion process and thus reduced fuel consumption. Meanwhile, with rise of MF fraction, biodiesel-MF blends increased gradually.

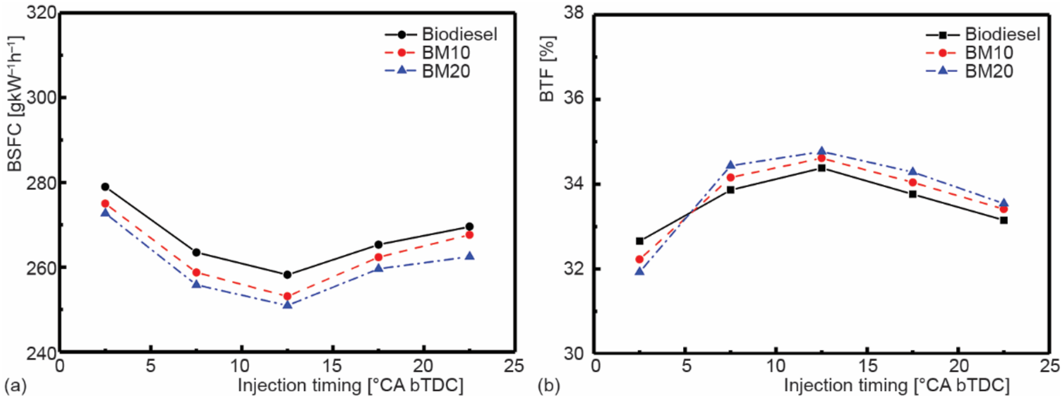

Figure 6. Effects of injection timing on (a) BSFC and (b) BTE

Figure 7 displayed the influences of EGR ratio on BSFC and BTE for all test fuels. With EGR ratio raising, BSFC rose gradually while BTE decreased for the four fuels. The trend could be explained by following reason. The oxygen concentration in cylinder decreased with EGR ratio increasing, thus suppressed the progress of the combustion process. Compared with biodiesel, BM10 and BM20 possessed high oxygen content and thus promoted more fuels complete combustion to reduce the BSFC, while increased the BTE.

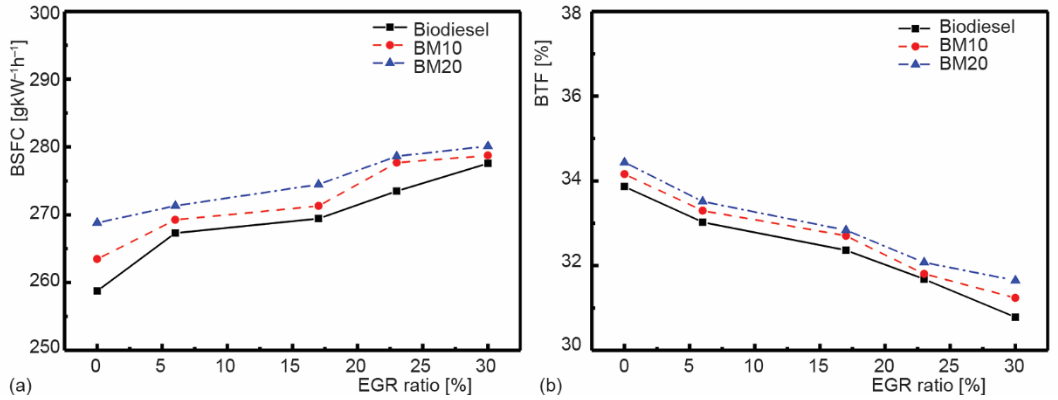

Figure 7. Effects of EGR ratio on (a) BSFC and (b) BTE

Regulated and unregulated emissions characteristics

The NO_x emissions under various injection timings presented in fig. 8(a). With injection timing increasing, NO_x emissions presented an increasing tendency for all test fuels. Fernando [35] *et al.* found the formation of NO_x emissions was mainly influenced by high in-cylinder temperature, oxygen concentration as well as enough reaction time. The advancement of injection timing was able to increase in-cylinder temperature. Meanwhile, the earlier injection of fuels could prolong reaction time in the cylinder, which was beneficial to the rise of NO_x production. Moreover, high oxygen content in fuels could also promote the production of NO_x. Therefore, with a rise of MF fraction, NO_x emissions emitted from blends raised except at 2.5 °CA bTDC.

Figure 8(b) presented the trend of NO_x emissions at various EGR ratios. When EGR ratio increased, the inert gas from EGR increased and thus absorbed more heat in the cylinder while reduced in-cylinder combustion rate, caused the reduction of in-cylinder temperature. In addition, the rise of EGR ratio enhanced the dilution of exhaust gas. The effect caused the reduction of in-cylinder oxygen concentration, which destroyed the conditions for the formation of NO_x. Moreover, compared with biodiesel, both BM10 and BM20 emitted more NO_x emissions due to higher oxygen content.

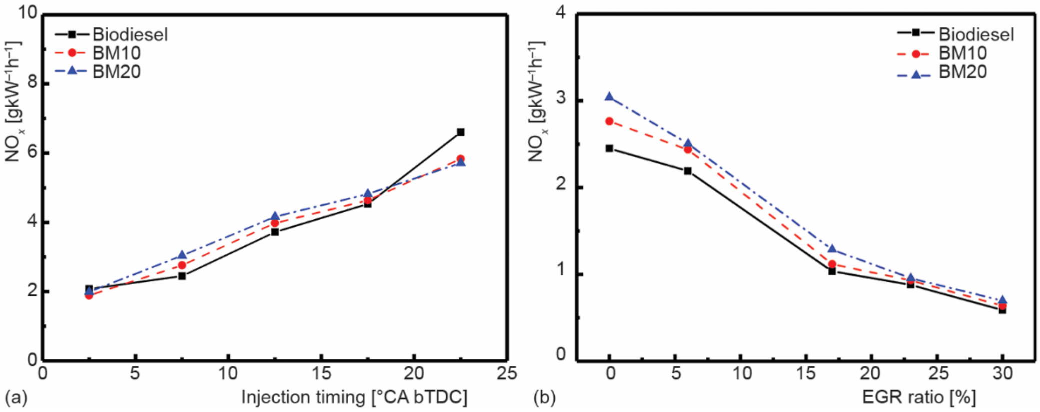

Figure 8. Effects of (a) injection timing and (b) EGR ratio on NO_x emissions

Figure 9 showed CO emissions under various injection timings and EGR ratios. With advance of injection timing, CO emissions reduced gradually. Especially at 2.5 °CA bTDC, BM20 appear combustion deterioration which leading to producing more CO emissions. While CO emissions raised with increasing of EGR ratios. Compared to biodiesel, BM10 and BM20 produced lower CO emissions because higher fuel-oxygen could promote the oxidation of CO.

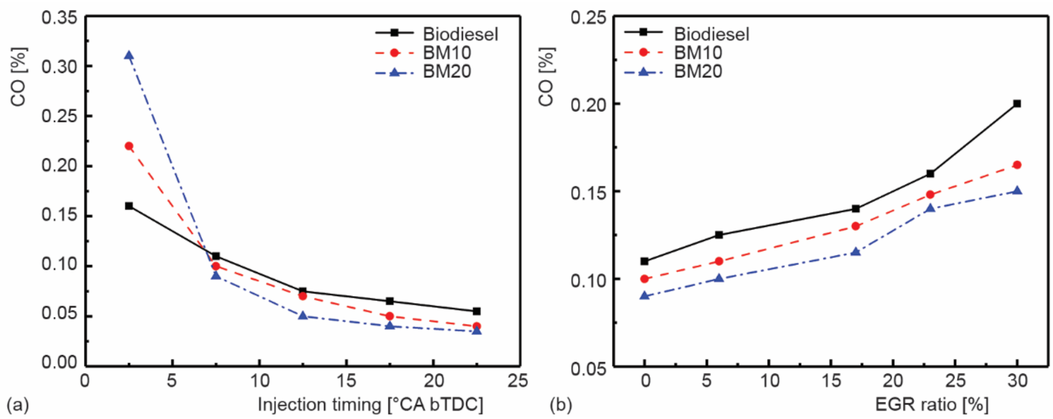

Figure 9. Effects of (a) injection timing and (b) EGR ratio on CO emissions

Figure 10(a) showed HC emissions under various injection timings. With the advance of injection timing, HC emissions reduced first and then increased for all test fuels. With rise of MF fraction, HC emissions reduced except for 2.5 °CA bTDC. It could be explained by following seasons: firstly, the penetration distance of biodiesel was longer than the rest fuels due to higher kinematic viscosity so that injected biodiesel was sprayed into the squeezing zone. Therefore, the wall oil film was easily formed on the cylinder wall or the top surface of piston and thus part of the fuel was not completely burned, resulted in increasing oxidized HC. Secondly, high oxygen content in the blends could promote the oxidation of HC so that BM10 and BM20 emitted less HC emissions.

Figure 10(b) showed HC emissions at different EGR ratios. As EGR ratio increased, HC emissions gradually increased. This could be explained by following reasons. With EGR ratio increasing, ignition delay of fuels was significantly prolonged and in-cylinder temperature was low, thus combustion rate was decreased. Before the flame reached the wall, both in-cylinder temperature and pressure were decreased due to the expansion, and thus combustible mixture was quenched in a large volume to form more HC emissions. In addition, the exhaust gas dilution effect and the heat capacity effect caused oxygen concentration in the cylinder to decrease, resulted in an increase in HC emissions that were incompletely oxidized.

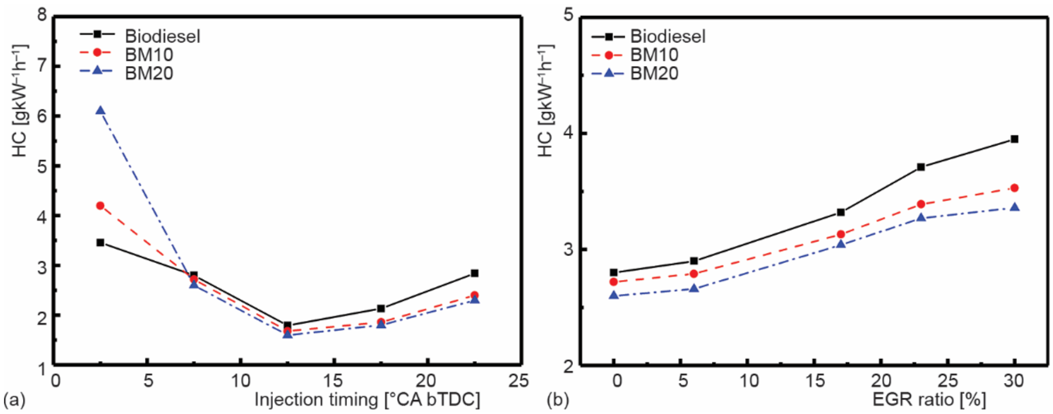

Figure 10. Effects of (a) injection timing and (b) EGR ratio on HC emissions

The 1,3-butadiene emissions were related to a rise of injection timing displayed in fig. 11(a). With injection timing in advance, 1,3-butadiene emissions gradually reduced. It could be attributed to increasing in-cylinder burning temperature which improved the post oxidation of 1,3-butadiene, which helped to decrease 1,3-butadiene emissions. Zervas *et al.* [36] did a detailed research on the formation mechanism of 1,3-butadiene. They found that fuels containing more linear HC were more prone to dehydrogenation and β-cleavage reactions during combustion, which was beneficial to produce more 1,3-butadiene emissions. Meanwhile, high-oxygen additive could be beneficial to improving burning degree and further promoted the oxidation of 1,3-butadiene. Biodiesel contained low linear HC structures and thus emitted less 1,3-butadiene emissions. Moreover, MF addition into biodiesel reduced linear HC contents while raised oxygen contents in fuels. Therefore, BM10 and BM20 produced less 1,3-butadiene emissions than biodiesel.

Besides, with EGR ratio increasing, 1,3-butadiene emissions decreased first and then increased gradually for all test fuels, seen fig. 11(b). The 1,3-butadiene is a product of incomplete burning of fuel. As EGR ratio increased, ignition delay became longer and thus the mixture of fuel formed during the deflagration period was more uniform, which was conducive to promote complete combustion so that 1,3-butadiene emissions reduced. However, with EGR ratio further increasing, the amount of exhaust gas introduced into cylinder per cycle increased and the dilution effect of exhaust gas caused oxygen concentration in the cylinder to decrease, which inhibited complete combustion of fuel and thus increased 1,3-butadiene emissions.

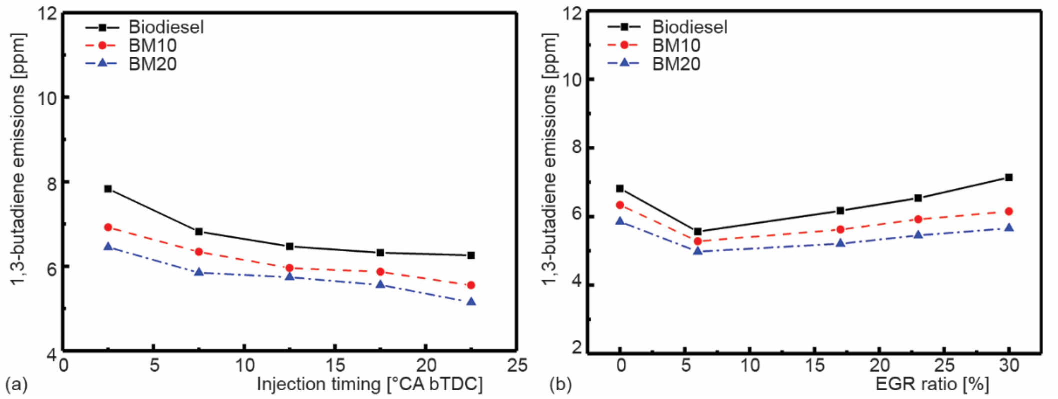

Figure 11. Effects of (a) injection timing and (b) EGR ratio on 1,3-butadiene emissions

Figure 12(a) showed acetaldehyde emissions at various injection timings. With injection timing in advance, acetaldehyde emissions had similar tendency with 1,3-butadiene emissions. Biodiesel emitted a lot of acetaldehyde emissions. Firstly, biodiesel contained a large number of compounds with short carbon chain and linear structure, which were easily converted into aldehyde gases during burning. Secondly, biodiesel was mainly fried and refined from waste animal and vegetable oils. During the frying process, lots of aldehyde gases were produced while these aldehyde gases still existed in the biodiesel. In addition, the main component of biodiesel is fatty acid methyl ester and the carbonyl group contained therein is

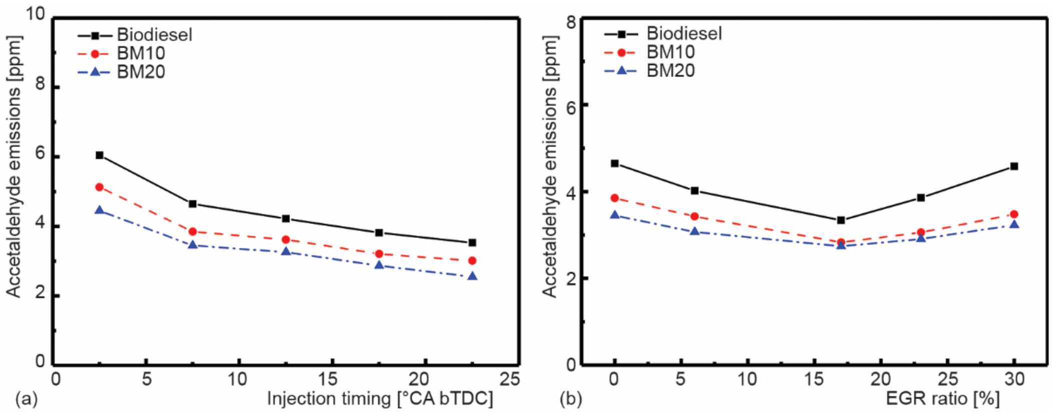

Figure 12. Effects of (a) injection timing and (b) EGR ratio on acetaldehyde emissions

an important source of acetaldehyde production. Besides, the addition of oxygenated fuel to biodiesel reduced the proportion of biodiesel in the blends, thus reduced considerable acetaldehyde emissions. Meanwhile, oxygen atom contained in the fuel also contributed to the complete oxidation of acetaldehyde, thereby further reduced acetaldehyde emissions. With EGR ratio increasing, acetaldehyde emissions presented analogous trend to 1,3-butadiene emissions, seen in fig. 12(b). It could be explained by the effects of EGR just like it to 1,3-butadiene emissions.

Conclusions

The influences of injection timing and EGR on combustion performance and emissions characteristics for four fuels on a Diesel engine studied under the condition of 0.38 MPa BMEP and engine speed constantly at 1800 rpm.

- When injection timing at 2.5 °CA bTDC, both BM10 and BM20 appeared a combustion deterioration.
- The MF addition into biodiesel promotes the combustion process. Compared to biodiesel, BM10 and BM20 have lower BSFC and higher BTE. Meanwhile, the BTE of BM20 is higher than BM10.
- With injection timing in advance, HC emissions first significantly decreased and then slightly increased. The NO_x emissions gradually increased while 1,3-butadiene and acetaldehyde emissions had an opposite trend. The emissions showed a relatively better degree when injection timing at 7.5 or 12.5 °CA bTDC.
- With EGR introduced, NO_x emissions effectively reduced while HC, 1,3-butadiene and acetaldehyde emissions increased. Moreover, EGR ratio within the scope from 6%-17% could be a suitable choice to decrease various emissions.
- Biodiesel could be a suitable alternative applied in Diesel engine. Moreover, MF with low viscosity and high oxygen content addition into biodiesel could show a better combustion performance and emissions characteristics.
- Although BM10 and BM20 emitted more NO_x emissions, introducing EGR could solve this problem.

Acknowledgment

The authors would like to thank the National Natural Science Fund of China (No. 21377101), National Youth Fund of China (No. 51706163) and Wuhan University of Technology for financial support to this study.

Reference

[1] Bessieres, D., *et al.,* Thermophysical Behavior of Three Algal Biodiesels Over Wide Ranges of Pressure and Temperature, *Fuel*, *233* (2018), Dec., pp. 497-503
[2] Coskun, G, *et al.,* An Experimental and Modeling Study to Investigate Effects of Different Injection Parameters on a Direct Injection HCCI Combustion Fueled with Ethanol–Gasoline Fuel Blends, *Fuel*, *215* (2018), Mar., pp. 879-891
[3] Wei, L., *et al.,* Effects of Biodiesel-Ethanol and Biodiesel-Butanol Blends on the Combustion, Performance and Emissions of a Diesel Engine, *Energy*, *155* (2018), Jul., pp. 957-970
[4] Ramadhas, A. S., *et al.,* Use of Vegetable Oils as I. C. Engine Fuels – A Review, *Renew Energy, 29* (2004), 5, pp. 727-742
[5] Zhu, L., *et al.,* Combustion, Performance and Emission Characteristics of a DI Diesel Engine Fueled with Ethanol–Biodiesel Blends, *Fuel*, *90* (2011), 5, pp. 1743-1750
[6] Buyukkaya, E., Effects of Biodiesel on a DI Diesel Engine Performance, Emission and Combustion Characteristics, *Fuel*, *89* (2010), 10, pp. 3099-3105

[7] Zheng, M., *et al.,* Biodiesel Engine Performance and Emissions in Low Temperature Combustion, *Fuel, 87* (2008), 6, pp. 714-722
[8] Lapuerta, M., *et al.,* Effect of Biodiesel Fuels on Diesel Engine Emissions, *Progress in Energy & Combustion Science, 34* (2008), 2, pp. 198-223
[9] Antonopoulos, K. A., *et al.,* Comparative Performance and Emissions Study of a Direct Injection Diesel Engine Using Blends of Diesel Fuel with Vegetable Oils or Bio-Diesels of Various Origins, *Energy Convers Manage, 47* (2006), 18-19, pp. 3272-3287
[10] Xiao, H. L., *et al.,* Combustion and Emission Characteristics of Diesel Engine Fueled with 2,5-Dimethylfuran and Diesel Blends, *Fuel*, *192* (2017), Mar., pp. 53-59
[11] Armas, O., *et al.,*. Emissions from a Diesel–Bioethanol Blend in an Automotive Diesel Engine, *Fuel, 87* (2008), 1, pp. 25-31
[12] Ahmed, I., Oxygenated Diesel: Emissions and Performance Characteristics of Ethanol–Diesel Blends in CI engines, SAE technical paper, 2001-01-2475, 2001
[13] Abhishek, P., *et al.,* An Experimental study of Combustion, Performance, Exergy and Emission characteristics of a CI engine fueled by Diesel-Ethanol-Biodiesel Blends, *Energy*, *141* (2017), Dec., pp. 839-852
[14] Roman-Leshkov, Y., *et al.,* Production of Dimethylfuran for Liquid Fuels from Biomass-Derived Carbohydrates, *Nature, 447* (2007), June, pp. 982-985.
[15] Zhao, H. B., *et al.,* Metal Chlorides in Ionic Liquid Solvents Convert Sugars to 5-Hydroxymethylfurfural, *Science, 316* (2007), 5831, pp. 1597-1600
[16] Wu, X. S., *et al.,* Identification of Combustion Intermediates in a Low-Pressure Premixed Laminar 2,5-Dimethylfuran-Oxygenargon flame with Tunable Synchrotron Photoionization, *Combust Flame, 156* (2009), 7, pp. 1365-1376
[17] Wu, X. S., *et al.,* Measurements of Laminar Burning Velocities and Markstein Lengths of 2,5-Dimethylfuran-Airdiluent Premixed flames, *Energy Fuels*, *23* (2009), 9, pp. 4355-4362
[18] Wu, X. S., *et al.,* Laminar Burning Characteristics of 2,5-Dimethylfuran and Iso-Octane Blend at Elevated Temperatures and Pressures, *Fuel, 95* (2012), 1, pp. 234-240
[19] Zhong, S. H., Daniel R, Xu HM, e*t al.,* Combustion and Emissions of 2,5-Dimethylfuran in a Direct-Injection Spark-Ignition Engine, *Energy Fuels*, *24* (2010), 5, pp. 2891-2899
[20] Daniel, R., *et al.,* Combustion Performance of 2,5-Dimethylfuran Blends Using Dual-Injection Compared to Direct-Injection in a SI Engine, *Appl Energy*, *98* (2012), Oct., pp. 59-68
[21] Zhang, Q. C., *et al.,* Combustion and Emissions of 2,5-Dimethylfuran Addition on a Diesel Engine with Low Temperature Combustion, *Fuel, 103* (2013), Jan., pp. 730-735
[22] Zhang, Q. C., *et al.,* Diesel Engine Combustion and Emissions of 2,5-Dimethylfuran-Diesel Blends with 2-Ethylhexyl Nitrate Addition, *Fuel*, *111* (2013), Sept., pp. 887-891
[23] Chen, G. S., *et al.,* Experimental Study on Combustion and Emission Characteristics of a Diesel Engine Fueled with 2,5-Dimethylfuran-Diesel, n-Butanol-Diesel and Gasoline-Diesel Blends, *Energy, 54* (2013), June, pp. 333-342
[24] Zhang, Q., *et al.* Combustion and Emission Characteristics of Diesel Engines Using Diesel, DMF/Diesel, and N-Pentanol/Diesel Fuel Blends, *Journal of Energy Engineering, 144* (2019), 3
[25] Zheng, Z., *et al.,* Experimental Study on Combustion and Emissions of Dual Fuel RCCI Mode Fueled with Biodiesel/n-Butanol, Biodiesel/2,5-Dimethylfuran and Biodiesel/Ethanol, *Energy, 148* (2018), Apr., pp. 824-838
[26] Chheda, J. N., *et al.,* Production of 5-Hydroxymethylfurfural and Furfural by Dehydration of Biomass-Derived Mono- and Poly-Saccharides, *Green Chem*, *9* (2007), 4, pp. 342-350
[27] Roman-Leshkov, Y., Dumesic, J. A., Production of Furan Derivatives by Dehydration of Biomass-Derived Carbohydrates, Abstracts Papers Am Chem Soc 2007, 234, pp. 352-357
[28] Xiao, H. L., *et al.*, Combustion Performance and Emissions of 2-Methylfuran Diesel Blends in a Diesel Engine, *Fuel*, *175* (2016), July, pp. 157-163
[29] Wei, H. Q., *et al.,* Experimental Investigation on the Combustion and Emissions Characteristics of 2-Methylfuran Gasoline Blend Fuel in Spark-Ignition Engine, *Appl Energy*, *132* (2014), Nov., pp. 317-324
[30] Wang, C. M., *et al.,* Combustion Characteristics and Emissions of 2-Methylfuran Compared to 2,5-Dimethylfuran, Gasoline and Ethanol in a DISI Engine, *Fuel, 103* (2013), Jan., pp. 200-211
[31] Heywood, J. B., *Internal Combustion Engine Fundamentals*, McGrawhill, New York, USA, 1988
[32] Wei, L., *et al.,* Combustion and Emission Characteristics of a Turbocharged Diesel Engine Using HIGH Premixed Ratio of Methanol and Diesel Fuel, *Fuel*, *140* (2015), Jan., pp. 156-163

[33] Mingrui, W., *et al.,* Combustion Performance and Pollutant Emissions Analysis Using Diesel/Gasoline/Iso-Butanol Blends in a Diesel Engine, *Energy Conversion and Management, 149* (2017), Oct., pp. 381-391
[34] Liu, J., *et al.,* The Effects of EGR and Injection Timing on the Engine Combustion and Particulate Matter Emission Performances Fueled with Diesel-Ethanol Blends, *Thermal Science, 22* (2018), 3, pp. 1457-1467
[35] Fernando, S., *et al.,* NO_x Reduction from Biodiesel Fuels, *Energy Fuels, 20* (2006), 1, pp. 376-382
[36] Zervas, E., *et al.,* Influence of Fuel and Air/Fuel Equivalence Ratio on the Emission of Hydrocarbons from a SI Engine. 2. Formation Pathways and Modelling of Combustion Processes, *Fuel*, *83* (2004), 17-18, pp. 2313-2321

Paper submitted: January 12, 2018
Paper revised: June 23, 2019
Paper accepted: July 16, 2019