AN EXPERIMENTAL STUDY ON COMBUSTION AND EMISSIONS CHARACTERISTICS IN A DUAL-INJECTION SPARK-ASSISTED COMPRESSION IGNITION ENGINE FUELED WITH PODE/GASOLINE

by

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Spark-assisted compression ignition (SACI) is a potential way to improve thermal efficiency for gasoline engine with a relatively low compression ratio. The dual-injection system and spark strategy are considered to be an effective approach to control the combustion of SACI engine. Polyoxymethylene dimethyl ethers (PODE) is a potential fuel for carbon neutral with high oxygen content and unique molecule structure. In this study, the transition of combustion modes with different equivalence ratio and effects of direct injection ratio on SACI combustion and emissions fueled with PODE/gasoline under different loads were investigated. The results showed that SACI combustion could be achieved with the compression ratio of 13 and the brake thermal efficiencies at 2 bar, 3 bar, and 4 bar under the dual-fuel SACI were increased by 49%, 29%, and 27%, respectively, compared with the gasoline spark ignition mode. The increase in direct injection ratio first shortened the combustion duration and then prolonged. An appropriate direct injection ratio was shown to control the combustion process to achieve high efficiency combustion, at which low total HC, CO, and PM emissions were achieved while the NO_x emissions remained at a low level.

Key words: *SACI, combustion and emissions, PODE, dual-injection*

Introduction

The SACI is a combustion mode, in which the in-cylinder temperature and pressure are increased by the spark plug to ignite of the mixture at the end of the compression stroke [1, 2]. The SACI combustion can be achieved under a relatively low compression ratio. According to simulations, the thermal efficiency of the SACI mode is increased by 30% compared to that of the direct injection spark ignition mode fueled with gasoline in an engine with a compression ratio of 12.42 [3]. In addition, SACI engine can be operated under a stoichiometric air/fuel ratio, so the three-way catalysts (TWC) can be used to control the gas emissions [4]. The SACI is a compression ignition mode, that is suitable for gasoline engines.

The proportion of fuel autoignition and flame propagation speed are two main key factors in controlling the combustion phasing and knock [5, 6]. To suppress knock, different strategies such as EGR, injection strategies and spark timing have been proposed to control the combustion phasing. Changes in the combustion cycle such as changes to the Miller cycle, can be used to adjust the ratio of fuel consumed by flame propagation and auto-ignition, which can

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suppress engine knock while maintaining high efficiency [7]. Many studies have shown that the ignition delay of SACI is closely related to the temperature rise caused by flame propagation. The in-cylinder temperature and the ratio of fuel consumed by auto-ignition are mainly decided by the spark timing. An appropriate spark timing can reduce the COV of SACI combustion [8, 9]. A double injection strategy is used to obtain a better air/fuel mixture distribution than the single injection strategy [10]. Zhou *et al.* [11] improved the BMEP by 0.1 MPa and reduced fuel consumption by using a double injection strategy combined with EGR.

The dual fuel SACI could achieve a greater adjustment of the mixture distribution compared to split injection with a single fuel. Chen *et al.* [12] found that the ignition of SACI end-gas mixtures became stable under conditions of high mixture reactivity. The combustion process could be controlled by controlling the differences between the characteristics of the two fuels under partially premixed conditions [13]. A combination of PODE/gasoline has been commonly used in the study of dual fuel compression ignition combustion. The PODE does not have C-C bonds and has nearly 50 oxygen content, while gasoline is composed of long-chain alkane. Late injection of PODE produces a high concentration region in the gasoline mixture; this region promotes complete combustion [14-16]. Research by Leblanc *et al.* [17] has shown that PODE can achieve stable SACI combustion at a compression ratio of 9.2. The SACI combustion fueled by PODE/gasoline with a late injection strategy has the potential to achieve high efficiency at a relatively low compression ratio and control the combustion process.

According to previous studies, PODE also has the potential to reduce PM emissions and improve the combustion performance as a direct injection (DI) fuel [18-20]. According to the reaction mechanism of diesel/$PODE_3$, the addition of PODE promotes the oxidation of particles due to the increase in the premixed combustion process and the oxygen content of fuel [21]. The high reactivity of PODE can improve the reactivity gradient of the fuel [16, 22, 23]. For example, the thermal efficiency of reactivity-controlled compression ignition (RCCI) fueled by methanol/PODE is increased by 3.5% due to the reduction of ignition delay and combustion duration compared to RCCI fueled by methanol/diesel [24]. Moreover, the COV of RCCI fueled by PODE/gasoline is decreased because the reactivity and volatility of PODE are higher than those of RCCI fueled by diesel/gasoline [15]. The high proportion of PODE could prolong the combustion duration and reduce the peak heat release, which was confirmed by the simulation of Wang *et al.* [25].

The review of previous studies [26-29] illustrates that spark timing and the proportion of DI fuel can affect many parameters such as ignition delay and oxygen content, thus controlling the combustion process. In this paper, combustion of SACI engine fueled with PODE/gasoline shows the whole transition process from spark ignition (SI) lean burn to compress ignition (CI) with an increasing air/fuel ratio, so that the trends in the combustion process and thermal efficiency can be further studied. The effects of the DI ratio on the combustion and emissions of SACI engine are further studied. The main goal is to achieve stable SACI combustion under different operation conditions with high thermal efficiency and low emissions.

Experimental set-up

Engine and instrumentation

Figure 1 presents the schematic of the experimental set-up. The experiments were performed on a 2.0 L turbocharged gasoline engine with both DI and port injection, which compression ratio was set as 13. The engine specifications are shown in tab. 1. The test engine was coupled with a DC dynamometer to maintain the desired speed and load. The precise control of the injection timing and injection amount for PFI injectors and DI injectors was used to control the torque of the engine based on the OpenECU manufactured by PI Innovo. The in-cylinder pressure signals were measured by pressure sensor (Kistler 6125C) and were passed to Kibox combustion analyzer, in which the combustion parameters were calculated for 200 consecutive cycles. A Horiba MEXA-7500DEGR was used to measure the gaseous emissions including total HC, NO_x, and CO before TWC. A DMS500 produced by Cambustion company was used to measure the particle size distribution (PSD) and particle number (PN).

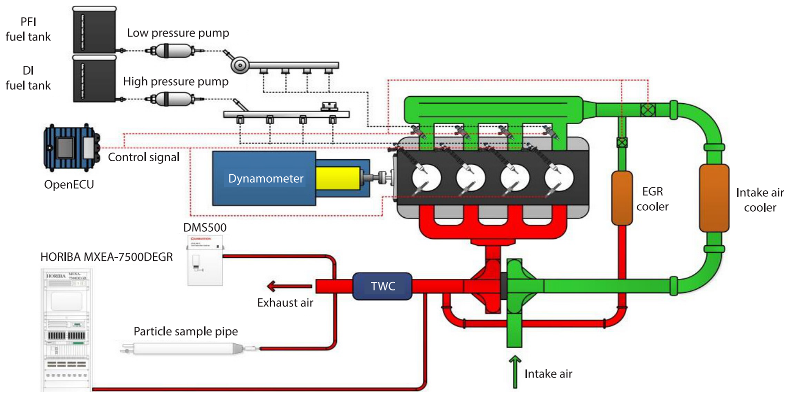

Figure 1. The schematic of the experimental set-up

Table 1. Engine specifications

Parameters	Details
Engine type	In-line four cylinder, turbocharged, dual-injection
Bore × stoke	88 mm × 82 mm
Displacement	1.995 L
Compression ratio	13:1
Intake valve timings	IVO: 20 °CA aTDC/IVC: 100 °CA bTDC
Exhaust valve timings	EVO: 116 °CA aTDC/EVC: 4 °CA bTDC

Experiment procedure

The commercial RON 92 gasoline was used as PFI fuel while the mixture of PODE3 and PODE4 (Volume ratio = 65:35) was used as DI fuel in the experiment. The port injection timing was selected as 350 °CA bTDC while the DI timing was selected as 30 °CA bTDC. The fuel specifications were shown in tab. 2. The intake temperature was set as 60 °C to maintain the stable combustion with the cooling water temperature of 85 ±2°C. The experiment was conducted at a steady-state with the engine speed of 1200 rpm. In this paper, the combustion mode transited from SI to CI was first studied. The load, which was defined as the brake mean effective pressure (BMEP) was set to 4 bar with stable spark timing and DI ratio. The equivalence ratio was increased from 1.2 to 1.6 to change the combustion modes by controlling the throttle opening. On this base, the effect of DI ratio and spark timing was studied at 2 bar and 4 bar BMEP. The DI ratio in this study is mass based, *i.e.*, the fraction of DI fuel mass compared to that of total fuel mass. The DI ratios swept from 100-30% when the spark timing was 12 °CA bTDC. The specific experimental conditions are shown in tab. 3.

Table 2. Fuel specifications

Physical parameter	92# Gasoline	PODE3	PODE4
Cetane number	–	78	90
Research octane number (RON)	92.5	–	–
Boiling point	191.7 °C	156 °C	202 °C
Density	735.2 g/m^3	1020 kg/m^3	1060 kg/m^3
Low heating value (LHV)	44.37 MJ/kg	19.2 MJ/kg	18.4 MJ/kg

Table 3. Experimental conditions

Parameters	Details
Engine speed [rpm]	1200
Load [bar]	BMEP = 2, 4 bar
Equivalence ratio	1.2, 1.3, 1.4, 1.5, 1.6
DI pressure [MPa]	15
DI timing [°CA bTDC]	30
Port fuel injection pressure [MPa]	0.4
Port fuel injection timing [°CA bTDC]	350
DI ratio [%]	30, 40, 50, 60, 70, 80, 90, 100
Spark timing [°CA bTDC]	12

In this study, CA10, CA50 and CA90 are defined as the CA at 10%, 50%, and 90% total heat release, respectively. The rapid combustion duration (defined as the interval between CA10 and CA50), flame development duration (defined as the interval between CA50 and CA90) and combustion duration (defined as the interval between CA10 and CA90) all were calculated from the in-cylinder pressure from 200 cycles. The COV based on indicated mean effective pressure (IMEP) defined:

$$COV = \frac{\sqrt{\frac{1}{n}\sum_{i=1}^{n}\left(IMEP_i - \overline{IMEP}\right)^2}}{\overline{IMEP}} \times 100\%$$

where $IMEP_i$ is the indicated mean effective pressure of every cycle and $\overline{IMER}$ – the average of n cycles.

Results and discussion

Transition of combustion mode for dual-injection SACI engine

Figure 2 presents the in-cylinder pressure and heat release rate curves under different lambda, where the DI ratio and spark timing (ST) were set to 50% and 12 °CA bTDC, respectively. Dual-fuel SACI combustion seems to exhibit two-stage ignition with a weak low temperature heat release (LTHR) before the high temperature heat release (HTHR). It can be observed that the combustion mode changes from SI to CI with increasing lambda. The trend of the heat release rate curve is similar to that of the stratified lean burn in an SI engine when lambda is 1.2. The peak in-cylinder pressure is increased by at least 8.6% and the phase of the peak heat release rate is advanced by at least 4.3 °CA, when lambda is greater than 1.2. This result is mainly due to the combustion characteristics of PODE. The PODE exhibits two-stages of heat release including LTHR and HTHR at lower intake temperatures [30]. The increasing inlet air with higher Lambda, λ, leads to a higher in-cylinder pressure during the compression stroke, which leads that PODE can be ignited easily, and then the combustion mode changes. The peak heat release rate increases and then decreases with increasing Lambda when the combustion mode switches to CI. Meanwhile, HTHR transitions to two-stage mainly because PODE has a higher flame propagation speed and different radicals during combustion compared with gasoline [31]. These properties cause the separation of the ignition processes of the two fuels during the HTHR stage. In addition, the CI mode produces more violent combustion than the SI mode. In addition, the increased Lambda increases the inlet air, which suppresses the in-cylinder temperature and flame propagation speed. These two factors lead to the trend in the peak heat release.

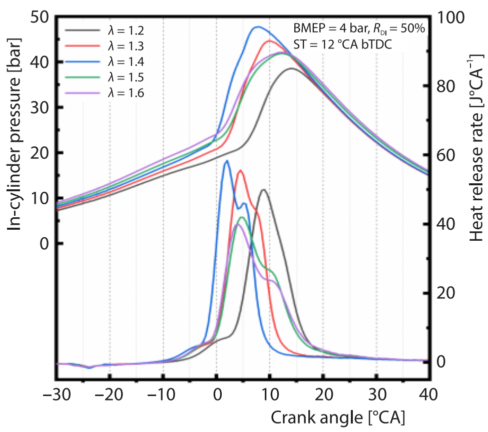

Figure 2. The in-cylinder pressure and heat release rate curves under different λ

The ignition delay of the SI engine is defined as the CA interval from the start of ST to 10% of the heat release, so it is represented by CA10 at the same ST. Figure 3 shows the impacts of Lambda on the combustion phasing. It is noted that the ignition delays of the CI mode are similar under different Lambda and shorter than that of the SI mode. The ignition delay is shortest when Lambda is 1.4. The engine ignition is mainly dominated by ST and the autoignition of PODE. The increase in Lambda enhances the trend in autoignition of PODE and makes SI more difficult. Engine ignition occurs at appropriately 2 °CA bTDC when PODE auto-ignites. The combustion duration is the sum of the rapid combustion duration and flame development duration. With increasing Lambda, the combustion duration decreases, and then increases. The effects of Lambda are mainly due to two factors. With increasing Lambda, the presence of more flame cores under the CI mode accelerates combustion, especially for rapid combustion duration. The flame propagation speed obviously declines in the lean gasoline mixture. These two factors shorten the combustion duration, and then prolong it.

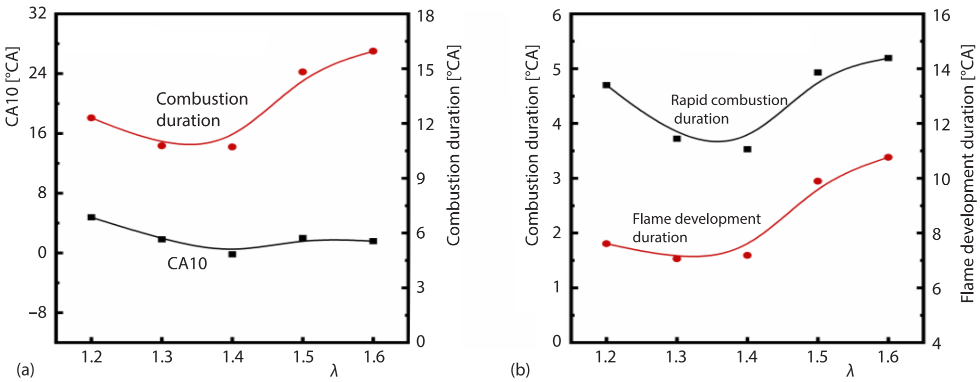

Figure 3. The impacts of Lambda on the combustion phasing

Thermal efficiency is an important index for evaluating the fuel consumption of the engine. Figure 4(a) presents the effect of Lambda on the BTE and COV of dual-fuel SACI combustion. With an increase in Lambda, the BTE decreases, and then increases. The COV is only increased from 1.0-1.5%, which is significantly less than the increment under SI lean burn fueled by gasoline. This result is mainly due to the change in the combustion mode. Although pumping loss can be obviously decreased with increasing Lambda, the proportion of incomplete combustion increases because of the shorter combustion duration. Therefore, the BTE decreases when the combustion mode first turns from SI to CI. The increasing BTE with higher lambda under CI mode illustrates that dual-fuel SACI combustion would achieve optimal thermal efficiency with wide open throttle (WOT). As shown in fig. 4(b), the dual fuel SACI mode with WOT has a higher BTE, and the BTE at 2 bar, 3 bar, and 4 bar under the dual fuel SACI are increased by 49%, 29%, and 27%, respectively, compared with the gasoline SI mode.

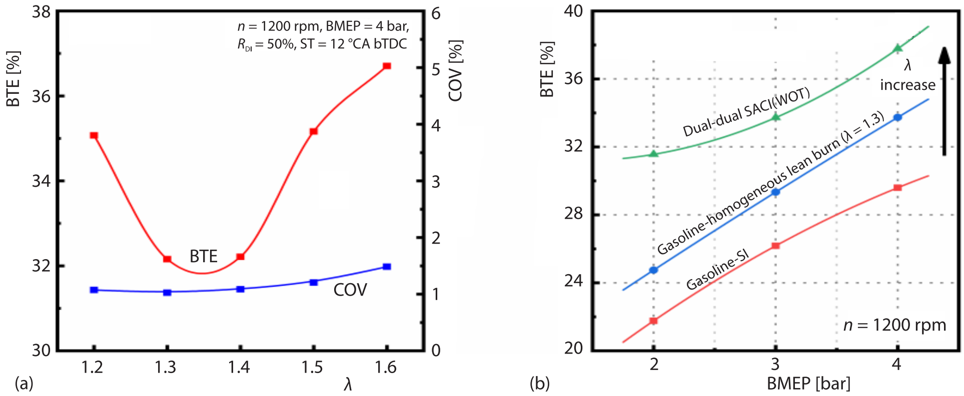

Figure 4. (a) Effects of Lambda on BTE and COV of dual-fuel SACI and (b) BTE under different loads and combustion modes

Effects of direct-injection ratios (R_{DI}) on SACI combustion and emissions

The previous section of this research indicates that the dual fuel SACI mode can significantly increase the thermal efficiency of the engine especially under low loads. In this part, the impacts of the DI ratio on SACI combustion and emission differences of 2 bar and 4 bar, which involved maintaining the wide open throttle and stable ST, were further studied. In this case, the amount of inlet air is constant at the same engine speed and the load is controlled by the fuel injection quantity.

Figure 5 compares the in-cylinder pressure and heat release rate curves under different DI ratios. There would be a large proportion of incomplete combustion even misfire when the DI ratio is less than selected in fig. 5. The phases of peak in-cylinder pressure and peak heat release rate are advanced with the increase in DI ratio under different loads. It is noted that the fuel ignites after 10 °CA bTDC while DI injection timing is set as 30 °CA bTDC, which means that the fuel is partially premixed. When most of the fuel is directly injected into the cylinder, even though higher DI ratio would decrease mixture temperature due to the fuel evaporation, PODE was ignited earlier and easily because of the high concentration at the end of the compression stroke, which leads to rapid combustion. The advancing combustion phase leads to higher in-cylinder pressure and temperature, which is the determinant of in-cylinder temperature. Therefore, the in-cylinder pressure and temperature is significantly increased with increasing DI ratio.

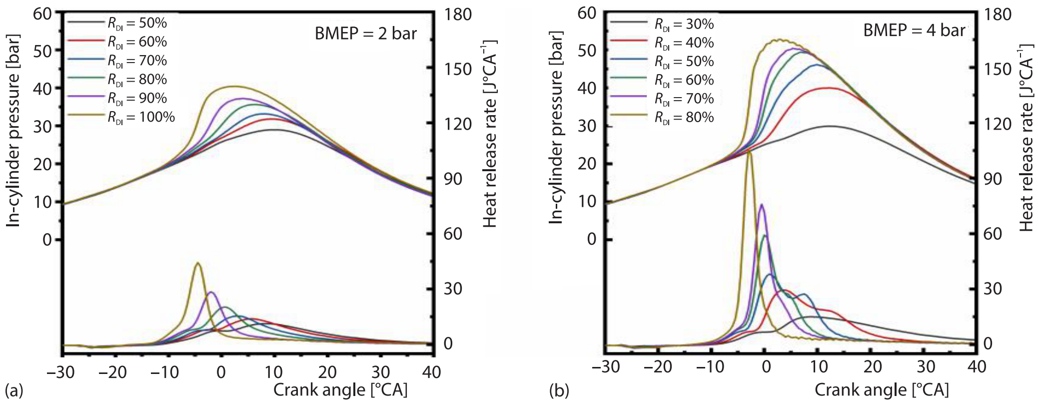

Figure 5. Effects of DI ratios on in-cylinder pressure and heat release rate; (a) BMER = 2 bar and (b) BMER = 4 bar

The peak heat release rate is obviously decreased and the heat release process of engine shifts from single-stage (HTHR) heat release to two-stage (LTHR+HTHR) heat release as the DI ratio decreases as shown in fig. 5. This trend is consistent with the ignition characteristics of PODE because the temperature of the fuel/air mixture is decreased with decreasing loads and DI ratio, which is a key factor influencing the heat release process of PODE. The HTHR stage has variable tendencies under different loads. The HTHR stage exhibits single-stage heat release when the BMEP is 2 bar. However, the HTHR stage shifts from single-stage heat release to two-stage heat release, and finally turns to single-stage heat release with the decrease in DI ratio, when the BMEP is 4 bar. Different concentrations of PODE are distributed in the premixed gasoline mixture in dual-fuel mode, and the ignition of gasoline mainly depends on the diffusion combustion of the PODE flame. When the amount of gasoline increases to a certain level with the decrease in DI ratio and increase in loads, phase deviation occurs between the HTHR of PODE and the heat release of gasoline because of the difference in the flame propagation speed. Therefore, the HTHR of dual-fuel SACI combustion exhibits two-stage heat release. As the DI ratio decreases further, PODE is more homogenously distributed in the gasoline mixture., which leads to a delay in the combustion phase and a low overall flame propagation speed. The combustion duration of the HTHR is significantly prolonged, which causes the overall combustion be similar to the combustion of stratified lean burn under a high air/fuel ratio, as the heat release rate curve of the HTHR exhibits flattened single-stage heat release.

The effects of DI ratio on combustion phases under different loads are shown in fig. 6. The DI ratio has a significantly influence on the combustion phases. With the increase in DI ratio, CA10 is advanced while the rapid combustion duration is nearly linearly shortened. Meanwhile, the ignition delay and rapid combustion duration are shorter under high loads at the same DI ratio. The trend of the overall combustion duration is similar to that of the flame development duration, which is firstly decreased, and then increased with the increase in DI ratio. More PODE can easily produce a high PODE concentration region in the cylinder, which would significantly shorten the ignition delay. It is noted that the engine ignites bTDC when the DI ratio is higher than 50%. For SACI engine, the flame development duration is more than twice as long as that of the rapid combustion duration, so the trend of the overall combustion duration is mainly dominated by the flame development duration. The boundary between the rapid combustion duration and flame development duration is when the heat release reaches 50%. Because the LHV of PODE is only approximately 40% of that of gasoline, the heat release

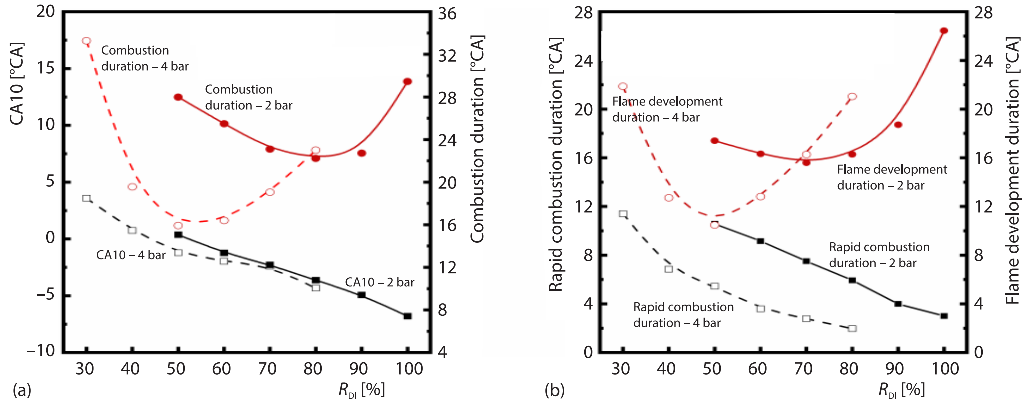

Figure 6. Effects of DI ratios on combustion phases under different loads; (a) CA10 and combustion duration and (b) RCD and FDD

of gasoline is more than half of the total heat release when the DI ratio is less than or equal to 70%. The oxidation activation energy of PODE is lower than that of HC fuel [32] causing the PODE to burn more quickly, which means that the remaining fuel lefts less more under high DI ratio. Thus, the flame development duration is significantly prolonged. With the decrease in DI ratio, the amount of remaining fuel is increased, which causes the flame to propagate more easily among the mixtures. Therefore, the flame development duration is obviously decreased. When DI ratio decreases to a certain level, SACI combustion is similar to gasoline stratified lean burn because there are fewer flame cores produced by PODE. This result means that the flame propagation speed is slower with the decrease in DI ratio, which finally leads to the prolongation of the flame development duration. The DI ratio corresponding to the turning point of the trend of flame development duration is larger under low loads because of the lower amount of fuel and the lower in-cylinder temperature and pressure.

According to the previous study, the decrease in DI ratio can retard the combustion phases, which causes the engine to perform less negative work. The effects of DI ratio on BTE and total HC emissions are shown in fig. 7. For the SI engine, the best CA50 is between 8 and 10 °CA aTDC [33], which can reach higher efficiency. Figure 7(a) shows that the CA50 of peak BTE is approximately 8 °CA aTDC when the BMEP is 4 bar while the CA50 of peak BTE

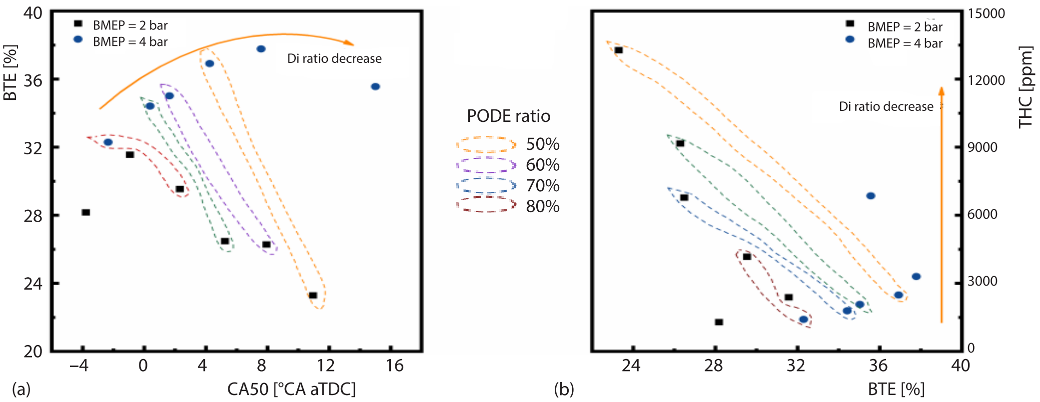

Figure 7. Correlations between BTE and CA50; (a) and total HC emissions and (b) under different DI ratios

is 1 °CA bTDC when the BMEP is 2 bar. With the decrease in DI ratio, the BTE first increases, and then decreases. The engine produces more total HC under a low DI ratio as shown in fig. 7(b). A DI ratio from 50-90% results in an 82% total HC emission decrease and thus higher fuel consumption when the BMEP is 2 bar. As a result, the relative BTE increases from 23.3-31.6%. This result means that the DI ratio must be large enough to ensure the complete combustion of fuel under low loads. When the BMEP is 4 bar, the total HC emissions is slightly increased as the DI ratio below 50%. Thus, the peak BTE is more closely related to the better CA50. The turning point of the effects of DI ratio on BTE is similar to that of the combustion duration.

In general, particles in the range of 5-30 nm are defined as nucleation mode particles while particles in the range of 30-1000 nm are defined as accumulation mode particles [34]. Figure 8 shows the effects of DI ratio on particle size distribution under different loads. The particles are mainly in nucleation mode, and the particle size distribution is unimodal whether the BMEP is 2 bar or 4 bar. Due to the low in-cylinder temperature for SACI engine and the advantages of PODE, it is difficult to produce accumulation particles. Overall, the PM emissions are obviously lower than those of GDI engine and diesel engine, and the PM emissions is decreased with the increase in loads. The nucleation particles consist mainly of unburned fuel droplets, which are significantly affected by combustion. The DI ratio of lower PM emissions is the same as that of higher BTE.

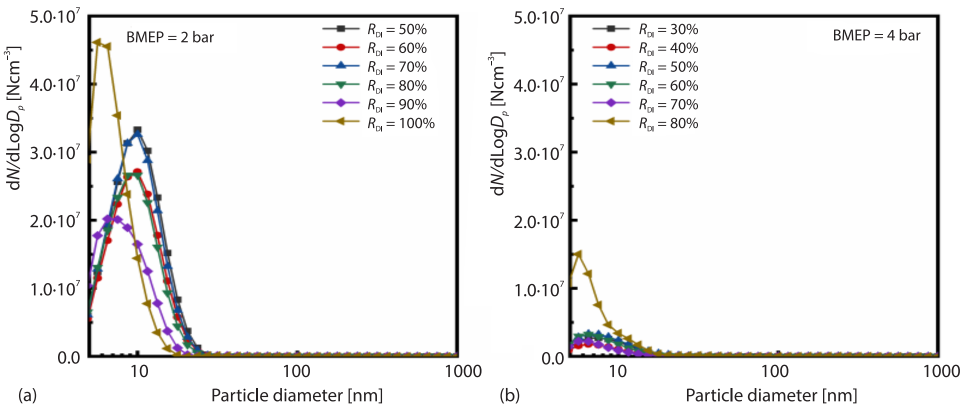

Figure 8. Effects of DI ratios on particle size distribution under different loads; (a) BMER = 2 bar and (b) BMER = 4 bar

The in-cylinder temperature and oxygen content are key factors influencing the CO and NO_x emissions. Figure 9 presents the effects of DI ratios on CO and NO_x emissions under different loads. The NO_x emissions are increased with the increase in DI ratio. When the BMEP increases from 2-4 bar, the NO_x emissions increase by nearly 10 times, however emissions are still less than 1000 ppm, which is significantly lower than the SI mode. On the one hand, the increase in DI ratio increases the in-cylinder oxygen content with the same amount inlet air because PODE is an oxygenated fuel. On the other hand, the in-cylinder temperature increases with higher load and DI ratio as mentioned before. These two factors lead to the increase in NO_x emissions. With the increase in DI ratio, the CO emissions first increase, and then decrease when the BMEP is 2 bar, while the CO emissions first decrease, and then increase when the BMEP is 4 bar. The trend in CO emissions is similar in the same DI ratio region under different loads. According to the combustion characteristics of the SACI engine, the trend in CO emissions is closely related to the transformation of the combustion process. When the heat release process involves single-stage heat release, more gasoline is incompletely burned because of the different flame propagation speeds between PODE and gasoline with decrease in DI ratio, which leads to more CO emissions. When the heat release process turns to two-stage heat release, the decrease in DI ratio improves mixture homogenization due to increased time for fuel evaporation, which is beneficial for CO oxidizing. However, the combustion in the engine is more incomplete when the DI ratio is high, which causes the CO emissions to increase. Overall, total HC and CO emissions of the SACI engine under medium loads are significantly better than those under low loads because of the more complete combustion. Lower total HC, CO and PM emissions are achieved at the most efficient DI ratio while the NO_x emissions remain at a low level.

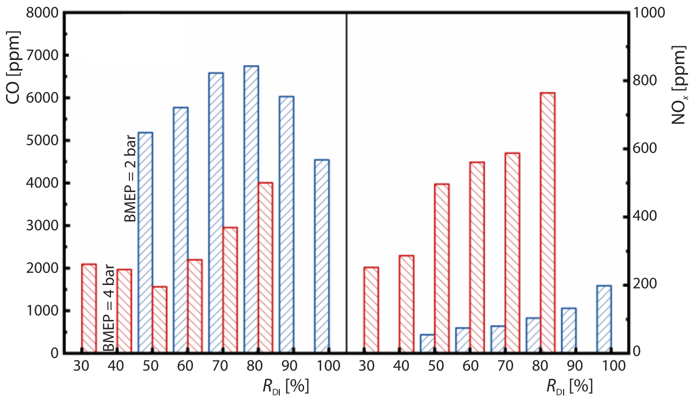

Figure 9. Effects of DI ratios on CO and NO_x emissions under different loads

Conclusions

In this paper, the transition of combustion modes under different equivalence ratio and the effects of DI ratio on combustion and emissions on dual-fuel SACI engine were investigated. The main conclusions are as follows.

- The combustion mode turns from SI to CI with the increasing equivalence with the compression ratio of 13 fueled by PODE/gasoline. The dual-fuel SACI mode has a higher BTE, and the BTE at 2 bar, 3 bar, and 4 bar under the dual-fuel SACI are increased by 49%, 29%, and 27%, respectively, compared with the original SI mode.
- The decrease in DI ratio first increases the BTE, and then decreases. The combustion phasing can be controlled by selected an appropriate DI ratio. The engine under low load needs a higher DI ratio to maintain the high efficiency SACI combustion.
- The emissions of the SACI engine under medium loads (4 bar) are significantly better than those under low loads (2 bar) because of the more complete combustion. Lower total HC, CO, and PM emissions are achieved at the most efficient DI ratio while the NO_x emissions remain at a low level.

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