EFFECTS OF DIESEL INJECTION TIMING AND METHANOL SUBSTITUTION RATIO ON COMBUSTION AND EXHAUST EMISSION CHARACTERISTICS OF A DUAL FUEL COMPRESSION IGNITION ENGINE

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Based on a diesel-methanol dual-fuel engine, the effects of diesel injection timing and methanol substitution ratio on the combustion, emissions and fuel economy of dual-fuel engines at different loads were investigated. The results showed that the maximum methanol substitution ratio over diesel varied along with load condition. It was disclosed that relatively high methanol substitution ratio, with substitution ratio 75% to be the maximum, could be applied at low and medium loads. However, the results also disclosed that diesel injection timing had significant effects on engine performance. When diesel injection timing advanced, the maximum combustion pressure and thermal efficiency increased with shortened combustion duration. With injection timing -34°CA ATDC, indicated thermal efficiency reaches up to 41.7% when methanol substitution ratio is 55%. While, too advanced injection timing led to premixed combustion, thus sharp heat release rate and high peak pressure were obtained. To prevent mechanical failure, limited methanol substitution ratio was resulted in when too advanced diesel injection timing was adopted. With the increase of methanol substitution ratio, the maximum in-cylinder combustion pressure and the peak heat release rate increased with shortened the ignition delay and combustion duration. The nitrogen oxide emission decreased while the hydrocarbon and carbon monoxide emissions increased. At load BMEP 0.2MPa, the increase of methanol substitution ratio led to decreased indicated thermal efficiency. However, when load was as high as BMEP 1.0MPa, indicated thermal efficiency increased from 43% to 44% when methanol substitution ratio increased from 0% to 40%.

Keywords: *Dual fuel engine, Methanol substitution ratio, Diesel, Combustion, Diesel injection timing, Exhaust emission characteristics*

1. Introduction

Compression ignition engines are widely used in transportation, construction machinery as well as other sectors owing to their advantages of large torque, high thermal efficiency as well as excellent reliability. Widespread use of diesel engines will inevitably consume large amounts of diesel and hence leads to increase of global crude oil consumption[1]. Alternative fuels, which could be used as substitutions of diesel or gasoline, have attracted increased interests as an effective means to reduce petroleum resource consumption[2, 3]. As an oxygen-containing alternative fuel, methanol could not only be produced from fossil fuels like coal, natural gas and oil, but also could be produced from renewable fuels such as biomass as well as clean energy like solar power[4] or wind power, which endows it carbon neutral attribute[5, 6] or even carbon negative attribute as long as carbon dioxide is used as the material[7]. When applied in internal combustion engine, the advantages of methanol are related to high oxygen content, large latent heat of vaporization, good combustion quality as well as storage and transportation convenience[8]. Accordingly, methanol was applied into internal combustion engine in the formats of methanol engine[9, 10], methanol-diesel/gasoline blended fuel engine[11, 12] as well as methanol-diesel/gasoline dual fuel engine[13, 14].

Owing to its low cetane number, methanol could hardly be ignited in conventional compression ignition engine. Hence early investigation over methanol engine was mainly related to spark ignition engine. Such researches show that methanol engine exhibits stable combustion characteristics with low nitrogen oxide (NOx) emissions[15, 16]. To achieve better the exploitation of methanol fuel over internal combustion engine, methanol direct injection spark ignition engine was widely investigated. Researches from C.M. Gong et al.[17, 18] indicated that injection parameter as well as spark timing, which was also indicated by Z.Q. Gao et al.[19], was critical factors that affect engine regulated emissions while compression ratio was the major factor that affected engine efficiency. Apart from regulated emissions, some other investigations also disclosed that formaldehyde as well as unburnt methanol emission massively existed thus should be paid attention to[20].

Except for previous discussion, another drawback of methanol for its application on conventional spark ignition engine lied in its high latent heat value thus glow plug was necessity to ensure engine performances[21, 22]. To avoid such inconvenience, numerous researchers tried investigations over methanol-gasoline blended fuel. Biglin et al. [23, 24] investigated the engine performance and fuel consumption of methanol-gasoline blends with different proportions(M5,M10, M15 and M20), results showed that case M5 presented the highest braking mean effective pressure (BMEP) while M20 exhibited the highest braking thermal efficiency (BTE). Due to the low heating value of methanol, fuel consumption in mass increased due to methanol blending. In addition, researches from Agarwal et al. [25] indicated that methanol addition into gasoline led to higher peak heat release rate, shorter combustion duration as well as lower CO, NOx and particulate matter (PM) emissions.

Nevertheless, compression ratio was one of the critical factors that affected engine efficiency thus exploitation of methanol[26, 27]. Therefore, lots of researcher also investigated the application of methanol on compression ignition engines by blending it with diesel fuel. Investigation from Huang et al. [28] indicated that increasing methanol ratio in blended fuel was conducive to inhibiting diffusion combustion thereby favors reducing engine fuel consumption. Such results were also observed in researches from Canakci’s et al. [29, 30]. However, on the perspective of diesel engine itself, the most important advantage of blending methanol into diesel fuel lies in its low soot emission. A.O. Hasan et al.[12] found that 40% methanol addition could realize roughly 50%~75% smoke emission reduction. Actually, lots of researches disclosed that 5% methanol addition took observable effects while 15% addition led to drastic soot emission reduction[31-33].

Considering such good effects of methanol fuel, some researchers carried out more detailed investigations over application of methanol on compression ignition engine. Yao [34] and Cheung [35] et al. investigated the diesel-methanol dual-fuel combustion in a naturally aspirated diesel engine. The results disclosed that the diesel-methanol compound combustion comprised two combustion stages: diffusion combustion of diesel and ignition of methanol-air premixed mixture by diesel. Diesel-methanol compound combustion caused increase of hydrocarbon (HC) and CO emissions, which entailed diesel oxidation catalyst (DOC) after treatment when stringent emission regulations are considered. In Li's work [36-38], two independent direct injection systems for methanol and diesel injection were designed. According to the fuel injection sequence, the impact of three different injection strategies of methanol-diesel, diesel-methanol, and methanol-diesel-methanol on engine performance was studied. Results indicated that diesel-methanol strategy could effectively prevent detonation at little sacrifice of fuel economy.

However, previous researches actually indicated the necessity of glow plug when methanol fuel, whether blended or not, was applied on spark ignition engines. Besides, application of methanol on spark ignition engines also suffered from confined thermal efficiency owing to limited compression ratio. In compare, application of methanol over compression ignition engine favored more about its energy efficiency. Nevertheless, the main drawback of such an idea involved the cold start difficulty as well as the combustion instability at low-load conditions. Besides, methanol-diesel blended fuel requires emulsifiers to form an emulsified liquid between immiscible methanol and diesel, resulting in a low proportion of methanol substituting diesel[39, 40]. Worse still, fuel blends stratification caused by the hydrophilicity of methanol was another obstacle for its application.

Considering such dilemmas, application of methanol on diesel engine through port injection strategy was promising (diesel was fueled through direct injection). With such manner, numerous researchers have achieved large methanol substitution ratios[41-43]. Q. Wang et al.[44] even achieve as large as 75% methanol substitution ratio. However, diesel injection timing is one of the most important parameters that affect engine combustion and hence engine performances[45, 46]. So far, the injection timing of diesel (without pilot injection) experimentally investigated was mainly within the range of -35~45℃A ATDC. To this end, the main purpose of this paper is to explore the effects of diesel injection timing over wide range on the combustion, emissions, and fuel economy of a dual-fuel compression ignition engine. Besides, the effects of methanol substitution ratio will also be investigated. The novelty of this study is as follow: 1) super advanced diesel injection timing for medium load (-36°CA ATDC, methanol substitution ratio 55%, BMEP 0.4MPa) was tried and found it favored engine thermal efficiency characterized by premixed combustion, 2) port injected methanol favored engine performance greatly under high engine load while worsen that under low engine load condition.

2. Experimental setup and Procedure

2.1 Apparatus

The dual-fuel engine used in the experiment was modified from a WP6.210 supercharged diesel engine batch-produced by Wei Chai Power Group. In addition to diesel direct injection, methanol port injectors have been mounted to achieve fuel stratification through precisely controlled methanol and diesel injection. The dual fuel injection system composed of diesel direct injection and methanol port injection can realize independent and flexible control injection of two fuels. Detailed specifications of the dual-fuel engine are listed in Table 1. Figure 1 exhibits the schematic view of experimental bench.

Table 1. Engine specifications

Description	Specification	Description	Specification
Type	Inline 6-cylinder, 4-strokes, dual-fuel turbocharged	Speed at Maximum Torque（r/min）	1500
Bore×Stroke（mm）	105× 120	Displacement volume (L)	6.23
Rated Power（kW）	132	Maximum Torque（N·m）	650
Compression ratio	18	Rated Speed（r/min）	2300

The experimental platform mainly consists of the dual-fuel engine, fuel supply system, EGR system, exhaust emissions measurement system and operation condition control system (dynamometer, control cabinet, ECU, calibration tool, etc.).

In-cylinder pressure was monitored by a piezoelectric pressure sensor (Kistler 6052C) coupled with a charge amplifier (Kistler 5015A) with a resolution of 0.5°crank angle (°CA). For each test, cylinder pressure of 200 consecutive engine cycles was sampled for combustion analysis. Sampled in-cylinder pressure data were averaged, and then were utilized to calculate indicated mean effective pressure (IMEP), maximum rate of pressure rise, heat release rate, combustion phase and so on. Consumption of diesel and methanol was measured by diesel fuel consumption meter (ToCeiL-CMFG010) and methanol fuel consumption meter (ToCeiL-CMFG025) installed in the fuel supply system respectively. Consumption rate of diesel and methanol was converted to the equivalent brake specific fuel consumption rate of diesel (EBSFC) according to the principle of equal calorific value, as shown in equation 1. Parts of the exhaust gas were introduced into an emission analyzer (HORIBA MEXA 7500) for the gaseous emissions (NO_X、CO、THC、CO_2) analyses. Other parts were imported into a fast particulate analyzer (Cambustion DMS500) for particulate matter (PM) and particle number (PN) measurement.

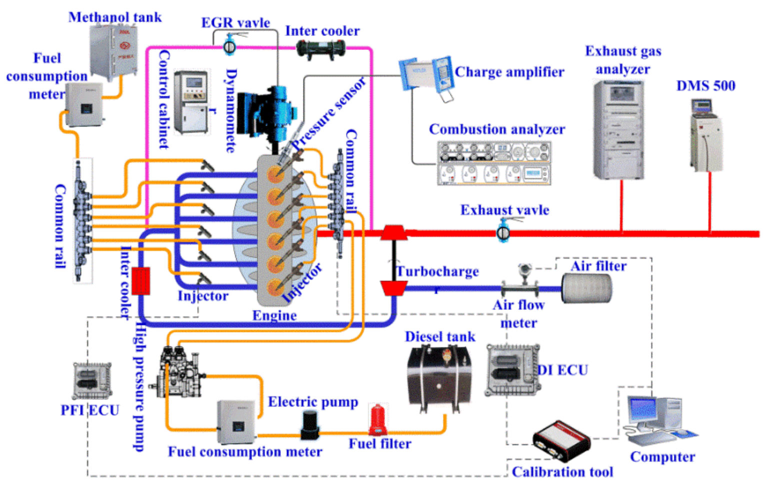


Figure 1. Schematic diagram of experimental setup

$$EBSFC=\frac{B_{dual-diesel}\times H_{u,diesel}+B_{dual-methanol}\times H_{u,methanol}}{H_{u,diesel}} \tag{1}$$

$B_{dual-diesel}$ and $B_{dual-methanol}$ indicate fuel consumption of diesel and methanol in dual fuel mode. $H_{u,diesel}$ and $H_{u,methanol}$ indicate low heating value of diesel and methanol, respectively.

2.2 Test fuels

In order to assess the effect of methanol substitution on engine combustion, single diesel and diesel-methanol dual-fuel with different percentages of methanol substitution were tested in this study. Based on the diesel consumption rate of the single-fuel mode engine, the ratio of the reduced diesel consumption rate of the dual fuel mode to the pure diesel consumption rate of the single-fuel under the same speed and torque is defined as the methanol substitution ratio (MSR), as shown in equation 2**Error! Reference source not found.**. Detailed properties of diesel and methanol are described in Table 2.

$$MSR=\frac{\left(B_{diesel}\text{-}B_{dual\text{-}diesel}\right)}{B_{diesel}}\times 100\% \qquad (2)$$

B_{diesel} indicates diesel consumption in pure diesel mode.

$B_{dual\text{-}diesel}$ indicates diesel consumption in diesel-methanol dual fuel mode.

Table 2.General properties of test fuels

Description	Methanol	Diesel	Description	Methanol	Diesel
Chemical structure	CH_3OH	C_xH_y	Low heating value (MJ/kg)	19.93	42.5
Liquid density (kg/m^3)	792	840	Oxygen content (%)	50	0-0.4
Auto-ignition temperature (K)	773	543-623	Latent heat of vaporization (kJ/kg)	1167	270
Cetane number	3	45-55			

2.3 Test conditions and methods

All engine tests were conducted under engine speed 1500 rpm, the maximum torque speed of stock engine. In order to explore the effect of methanol substitution ratio on the performance of dual fuel engine, engine load (BMEP) was selected as 0.2MPa(low load)，0.6 MPa(medium load) and 1.0MPa(high load). Injection timing and pressure were controlled by the ECU and adjusted by the calibration tool. Diesel direct injection timing in cylinder is 3°CA before the top dead center (BTDC), and the injection timing of methanol is 346°CA after the top dead center (ATDC), which is the opening timing of intake valve. Diesel common rail injection pressure is 100Mpa, and methanol injection pressure is 0.55MPa.

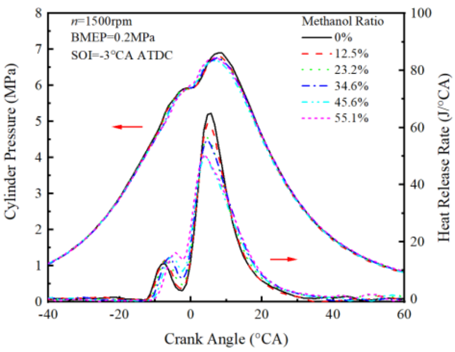
When exploring the influence of diesel injection timing on the performance of dual fuel engine, considering the limit of combustion stability and pressure rise rate, low load (BMEP 0.2MPa, methanol injection 22mg/cycle, approximately 42% substitution ratio) and medium load (BMEP 0.4 MPa, methanol injection 40mg/cycle, approximately 55% substitution ratio) at engine speed 1500r/min were selected as the research condition. Temperatures of the lubricant oil, cooling water and intake air were kept at a range of 85±2°C, 80±2°C and 45±1°C respectively, so as to ensure the repeatability of the tests. Data sampling was conducted after 3 minutes of stable operation.

3. Experimental results and discussion

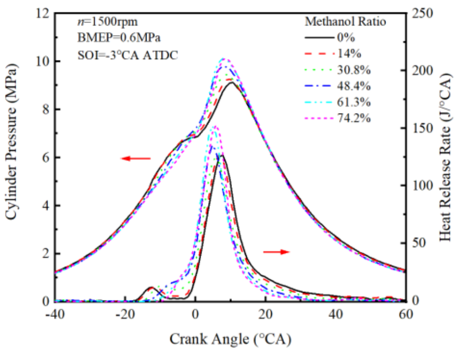
3.1 Effects of methanol substitution ratio

Figure 2 displays the in-cylinder pressure and instantaneous heat release rate curves of engine at 1500 r/min speed, load BMEP 2 bar, 6 bar, 10 bar, and different methanol substitution ratios. At low load, methanol injected from the intake manifold has weak effect on cylinder pressure. With the increase of methanol injection quantity, the first heat release peak produced by pre-injection will be delayed, and the exothermic peak of main combustion will decrease. This phenomenon is due to low cetane number and high latent heat of vaporization of methanol. The heat absorption of methanol vaporization reduces the intake air temperature, and the low-load engine cylinder temperature is low. The addition of methanol further reduces the cylinder temperature before ignition, resulting in prolonged ignition delay. At the same time, the methanol mixture is lean at low load, which inhibits ignition and combustion process, hence the peak heat release rate is reduced.

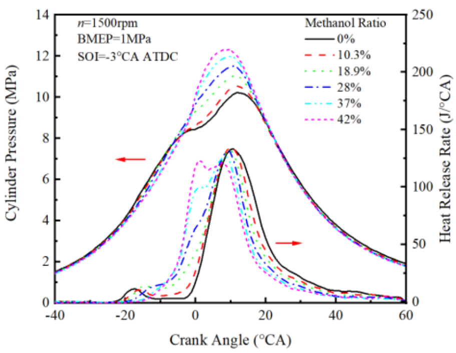
At medium and high loads, with the increase of methanol substitution ratio, in-cylinder maximum combustion pressure and instantaneous heat release rate both increases, the combustion center shifts forward, the ignition delay and the combustion duration are shortened. The reason is that the temperature in the cylinder is higher at high loads, and the increase in the quantity of diesel is conducive to the rapid increase of the ignition source energy, resulting in strengthened premixed combustion. It could also be found that the increase of methanol-air mixture concentration can promote the rapid flame propagation. Such a phenomenon should be ascribed to the reason that more well prepared (with ideal equivalence ratio) fuel-air mixture was formed when more methanol was injected before diesel injection near TDC. Although vaporization of methanol doesn't favor the ignition of fuel-air mixture, factor equivalence ratio controlled by methanol dominates here. Therefore, the ignition delay period at high loads decreases with the increased methanol substitution ratio, which is different from that at low load.



(a) BMEP 0.2MPa



(b) BMEP 0.6MPa

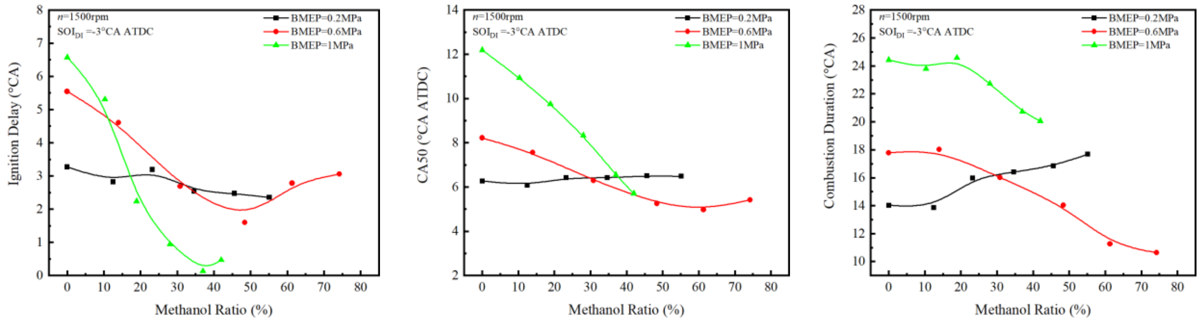


(c) BMEP 1MPa

Figure 2. The effect of methanol ratio on cylinder pressure and heat release rate under different loads

For further understanding of the difference of the combustion characteristics between different fuel ratios, Figure 3 exhibits the ignition delay, CA50 and combustion duration under various BMEP for different fuels. Here, CA10, CA50 and CA90 indicate the crank angle at which 10%, 50% and 90% accumulated heat released. In this paper, CA10, CA90 and CA50 are defined as the start, the end and the barycenter of the combustion, respectively. The ignition delay is defined as the interval of the crank angle between diesel injection timing and CA10, while the combustion duration is defined as the interval between CA90 and CA10.

As shown in Figure 3, at low load, the ignition delay is slightly prolonged while the combustion center (CA50) is little changed due to the influence of methanol heat absorption and lean mixture. With higher methanol substitution ratio, longer combustion duration results. At medium and high load, with the increase of methanol substitution ratio, ignition delay exhibits a decreasing trend while the combustion center gradually moves forward to TDC. From Fig.3, it can be found that the increase of methanol substitution ratio gradually shortens engine combustion duration.



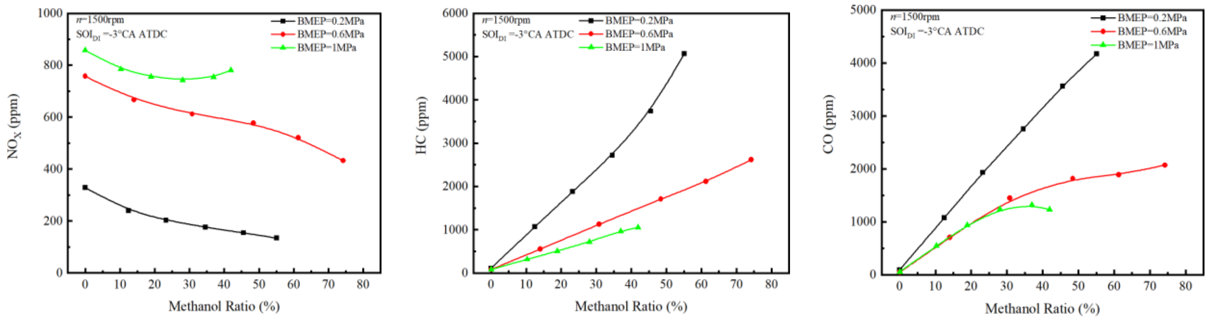
(a) Ignition Delay (b) CA50 (c) Combustion Duration

Figure 3. The effect of methanol ratio on combustion phase under different loads

Except for engine combustion, methanol substitution of diesel also prominently affects engine emission characteristics. **Error! Reference source not found.** displays the NO_X, HC, and CO emissions versus methanol substitution ratio at engine speed 1500rpm, load BMEP 2bar, 6bar, 10bar versus various methanol substitution ratios. Obviously, decreasing trend of NO_X emission could be observed when methanol substitution ratio increases. Especially, at medium and low load, high methanol substitution ratio even leads to about 50% reduction of NO_X emission. According to chemical reaction kinetics, NO_X formation entails high temperature, rich oxygen and long reaction time duration. Due to large latent heat of methanol vaporization, cooling effect caused by methanol evaporation would help to reduce the intake temperature and the maximum combustion temperature. The introduction of methanol can improve the combustion speed and shorten the duration of high temperature. These factors can inhibit the formation of NO_X.

From HC emissions characteristics shown in **Error! Reference source not found.**, it can be seen that with the increase of methanol injection quantity, HC emission exhibits a rising trend. At low load condition, HC emission increases rapidly. After injection of methanol, combustion temperature in cylinder decreases due to methanol evaporation. At low load, the quantity of methanol injection is relatively small, and the formed homogeneous methanol-air mixture is relatively lean, which is difficult to be ignited. In addition, during compression stroke, the methanol mixture enters the cylinder clearance and slot. Excessive lean burn, wall quenching effect and crevice effect strengthened the difficulty of ignition, resulting in a large amount of HC emissions. Under medium and high loads, the concentration of methanol mixture and the quantity of diesel injection increases, thus higher in-cylinder combustion temperature is resulted in, which further leads to the improvement of in-cylinder combustion. Therefore, the growth rate of HC emission slows down, and the overall effect is better than that of low load.

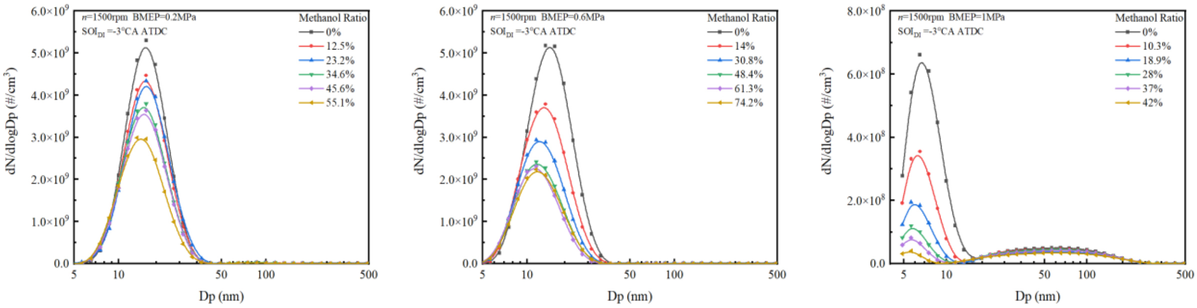
Error! Reference source not found. shows CO emission characteristics versus methanol substitution ratio. CO emission increases when methanol substitution ratio increases. However, the increasing trend of CO emission is more sensitive to methanol substitution ratio under lower engine load condition. Actually, CO is the main intermediate product generated by hydrocarbon fuel during combustion, which can be oxidized to CO_2 at high oxygen concentration and temperature and long reaction time. In the dual-fuel engine, the injection of methanol reduces the combustion temperature in the cylinder, resulting in an increase in the thickness of the quenching layer near the wall and an increase in CO emissions at the initial stage of combustion, which is particularly obvious at low load. On the other hand, the strong cooling effect of methanol vaporization would substantially decrease the temperature of intake mixture, leading to heavier incomplete oxidation of methanol and hence more CO emission.



(a) NO_X (b) HC (c) CO

Figure 4. The effect of methanol ratio on NO_X, HC and CO emissions under different loads

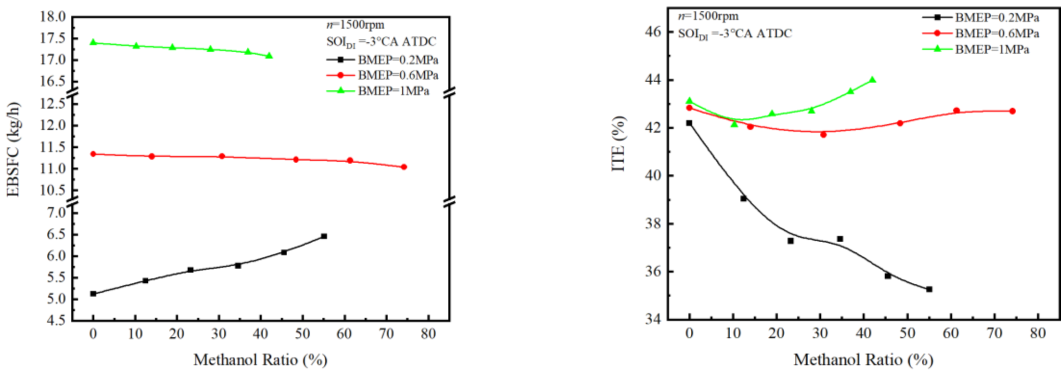
Error! Reference source not found. shows the effect of methanol substitution ratio on the PM emissions of dual-fuel engines under different loads. According to the curves, it is obvious that dual-fuel combustion exhibits lower PM emission than pure diesel combustion does. Such phenomenon is particularly distinct under medium and high engine load. Under conditions BMEP 0.2MPa, with the increase of methanol substitution ratio, PM emission gradually decreases. When methanol substitution ratio exceeds 50%, the peak value of PM decreases by approximately 40%. When BMEP is 0.6MPa, the addition of methanol significantly reduces PM emissions, which can be reduced by more than 55% at most. At higher loads (BMEP=10 bar), methanol substitution has a more significant effect on reducing particulate matter emissions in dual fuel engines. The reduction of PM emissions in dual fuel mode is mainly due to the inhibition of methanol on the formation of soot, which is the main component of PM. As an oxygen-containing fuel, methanol tends to inhibit soot precursor formation among diesel fuel rich region in the combustion chamber ascribed to its C-O bond in molecular structure. Besides, the strong cooling effect of methanol can reduce the combustion temperature and prolong the ignition delay period, which is conducive to the full diffusion of diesel, improving the quality of the mixture, and avoiding the appearance of excessively rich areas. At medium and high loads, the proportion of soot in PM increases, and methanol has a greater effect on prolonging the ignition delay and shortening the combustion duration, resulting in significant reduction of PM.



(a) BMEP 0.2MPa (b) BMEP 0.6MPa (c) BMEP 1MPa

Figure 5. The effect of methanol ratio on PM size distribution under different loads

Figure presents the effects of methanol substitution ratio on equivalent brake specific fuel consumption (EBSFC) and indicated thermal efficiency of methanol-diesel dual-fuel engine. At BMEP 0.2 MPa, with the increase of methanol substitution ratio, the EBSFC of the engine gradually increases. This would be mainly ascribed to the fact that low load combustion is characterized by low temperature and lean methanol-air mixture. When more methanol is injected, heavier cooling effects as well as more unburnt area would lead to more unburnt fuel, which hence leads worse HC emission. So, high methanol substitution ratio under low engine load is not recommended. Excessive substitution ratio will cause incomplete combustion, increase equivalent fuel consumption, and worsen HC emissions. At medium and high loads, as the methanol substitution ratio increases, the equivalent fuel consumption rate decreases slightly, and the indicated thermal efficiency remains at the same level as the diesel engine. When BMEP is 1MPa and the substitution ratio is 42%, thermal efficiency is 1.45% higher than that of the original diesel engine. At medium and high load, the in-cylinder high temperature and the large diesel injection quantity result in the low combustion loss of methanol and high thermal efficiency. The addition of methanol prolongs the ignition delay period and increases the proportion of premixed combustion, which helps to shorten the combustion duration thus leads to the end of combustion near the TDC. Moreover, the oxygen content of methanol can alleviate the local hypoxia in the diesel diffusion combustion, so the thermal efficiency does not decrease, which is better than that under the low load condition.

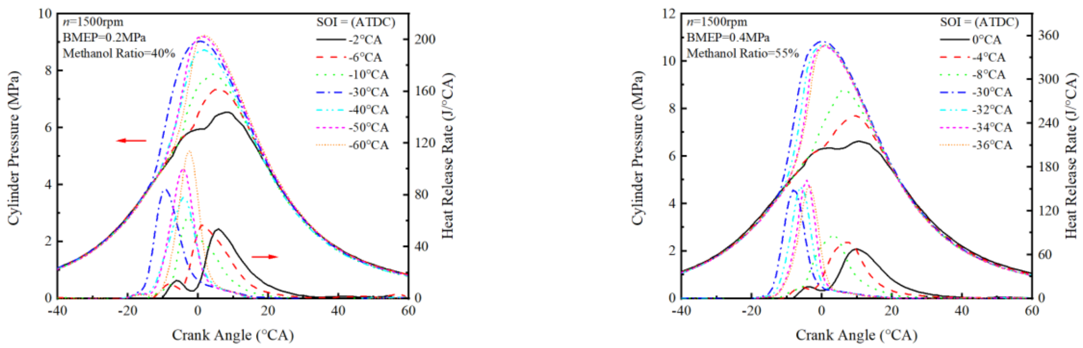


(a) EBSFC (b) ITE

Figure 6. The effect of methanol ratio on indicated thermal efficiency under different loads

3.2 Effects of diesel injection timing

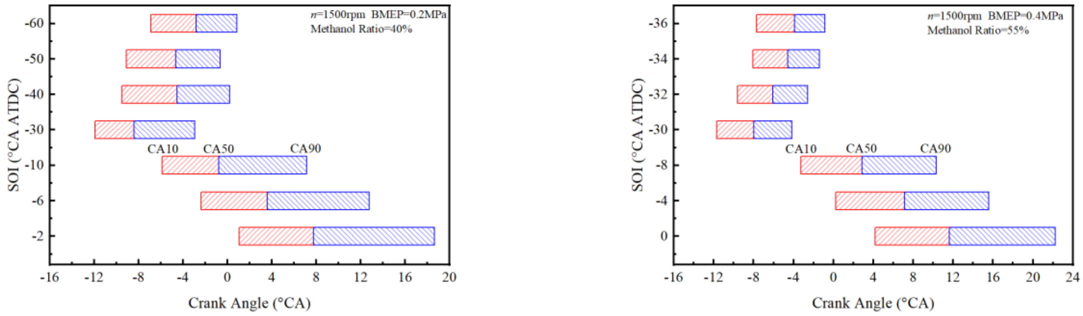
Figure displays the in-cylinder pressure and heat release rate curves of the dual-fuel engine at different diesel injection timings with fixed methanol substitution ratio. Starting from 0°CA ATDC, advancing diesel direct injection timing results in rise of maximum in-cylinder pressure, forward shift of combustion phasing and shorter combustion duration. When diesel injection timing is between -10 and -2 °CA ATDC, methanol premixed mixture would be ignited by injected diesel fuel near TDC, indicating traditional dual-fuel combustion mode. Before ignition, mixture in engine cylinder consists of gaseous methanol, gaseous diesel and diesel spray. Small amount of pre-injected diesel and the diesel-air pre-mixture accumulated during ignition delay period burn and ignite the surrounding methanol-air mixture first, followed by the diffusion combustion of diesel and the flame propagation of methanol. Figure 8. **The effect of injection timing on** shows combustion phase at different diesel injection times. In the range of -10°CA ~ 4°CA ATDC, ignition delay is within 5°CA and it slightly extends with the advance of diesel injection timing. The extension of the ignition delay promotes more diesel-air premixed mixture, which leads to an increase of heat release peak and pressure rise rate of premixed combustion, and gradual shortening of combustion duration. When diesel is injected in the range of -30°CA ~ -10°CA ATDC, the engine is detected to have a cylinder pressure rise rate exceeding 1.3 MPa/°CA, which exceeds the limit of safe engine operation. When the diesel injection timing is earlier than -30°CA ATDC, no diffusion combustion is observed anymore. The reason for this phenomenon is that the early-injected diesel has been sufficiently vaporized before the ignition. Premixed diesel starts to burn and ignites methanol-air mixture when the temperature, pressure and diesel equivalence ratio in the cylinder reach the diesel self-ignition point. Ultimately, both maximum combustion pressure and heat release rate in the cylinder are significantly increased, and the combustion phase shifts forward greatly, which leads to the CA50 appearing before the TDC. The combustion duration is about 10°CA, indicating significantly rapid heat release process. It can be seen from Figure that the CA50 is the most advanced when the diesel injection timing is -30°CA ATDC. Continue advancing the injection timing from -30°CA ATDC, CA50 will move backward and approach the TDC. This is because the low local equivalence ratio caused by the sufficient mixing of early injected diesel and air, which requires higher temperature and pressure to compression ignition,



(a) BMEP 0.2MPa (b) BMEP 0.4MPa

Figure 7. The effect of injection timing on in-cylinder pressure and heat release rate

resulting in delayed ignition timing.



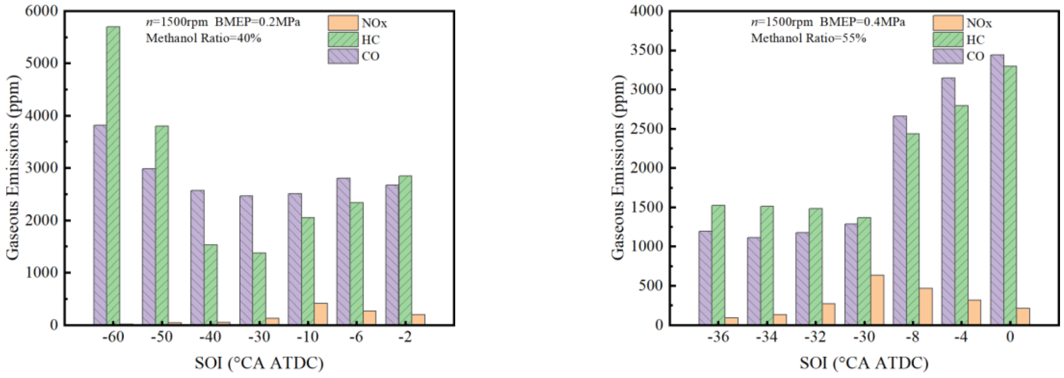
(a)BMEP 0.2MPa (b)BMEP 0.4MPa

Figure 8. The effect of injection timing on combustion phase

Figure shows the effects of diesel injection timings on NO_X, HC and CO emissions of a diesel-methanol dual-fuel engine. When diesel injection timing is within the range of -10°CA ~ 4°CA ATDC, NO_X emission gradually increases with the advance of injection timing. The advance diesel injection causes the combustion to move forward, and the peak heat release of premixed combustion increases. At the end of premixed combustion, high temperature region in cylinder and the average temperature of combustion increase, which promotes the formation of NO_X. However, NO_X gradually decrease when diesel injection timing is advanced to -30°CA ATDC. This should be ascribed to dual-fuel premixed combustion which is characterized with high rapid combustion process and short combustion duration. Sufficient mixing of early-injected diesel and air leads to weak stratification of fuel concentration. As a result, the low average combustion temperature in the cylinder, less local high temperature region, and short high temperature duration are conducive to inhibiting NO_X formation.

From the curves of HC emission shown in Figure , it can be seen that within the interval of -10°CA ~ 4°CA ATDC, HC emission decreases with the advance of injection time. Prolonged ignition delay caused by the advance injection of diesel leads to homogeneous mixing and sufficient combustion of diesel fuel and air, which is beneficial to the oxidation of HC. When diesel injection timing is within -30°CA ~ -40°CA ATDC, HC emission reaches the bottom, and HC emission gradually increases when the injection time is earlier than -40°CA ATDC. The early-injected diesel has a long evaporating time, which causes the diesel to diffuse to the vicinity of the cylinder wall, thus thickens wall quenching layer. In addition, short duration of high temperature caused by the rapid combustion in the cylinder and the low average combustion temperature are also the reasons for the mass generation of HC.

The characteristics of CO emissions curves are similar to that of HC emission. When injection timing is advanced from 4°CA ATDC to -10°CA ATDC, CO emission decreases. Combustion temperature in the cylinder increases gradually with the advance of injection, which is conducive to the oxidation of CO. Moreover, with the increase of ignition delay, evaporation time of diesel becomes longer, which promotes the mixing of fuel and air and reduces the local anoxic zone in the cylinder. When the injection timing of diesel is greatly advanced, subsequent low in-cylinder combustion temperature, lean diesel mixture and incomplete combustion of methanol lead to a large amount of CO.

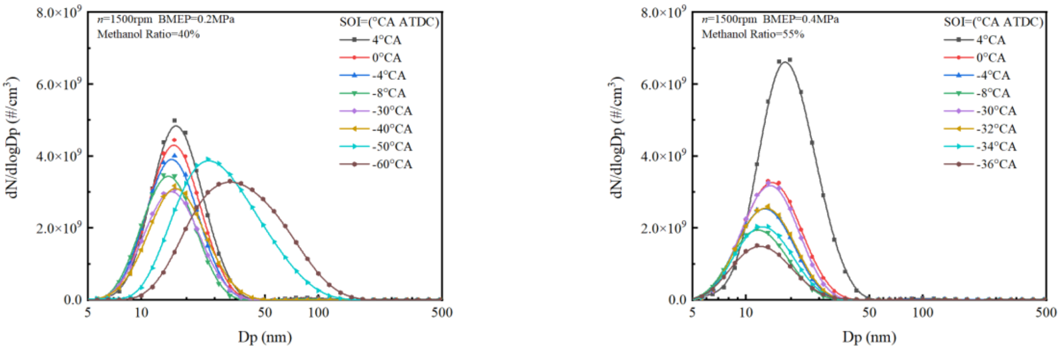


(a) BMEP = 0.2 MPa (b) BMEP = 0.4 MPa

Figure 9. The effect of injection timing on NO_X, HC and CO emissions under different loads

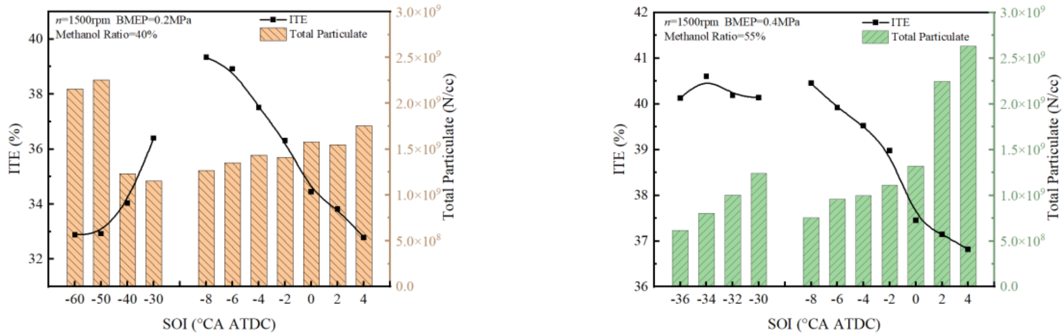
Figure 4 shows the effects of diesel injection timing on PM emission of dual-fuel engines. In the range of -10°CA ~ 4°CA ATDC, with the advance of diesel injection timing, the peak value of PM gradually decreases. When BMEP is 0.4MPa, with the advance of diesel injection timing, PM reduction is more obvious than that when BMEP is 0.2 MPa. Advanced diesel injection timing prolongs the mixing process of diesel and air before ignition happens, which is conducive to improving the uniformity of the mixture and reducing the fuel concentration area prone to PM. In addition, early injection makes combustion closer to the TDC, resulting in high temperature in the cylinder and promoting particulate oxidation. From Figure 4(a), PM emissions are high when diesel injection timing is between -60°CA ATDC and -50°CA ATDC. The main reason is that the pressure in the cylinder is relatively low when the diesel is injected early, which causes the increase of diesel spray penetration distance. Long spray penetration increases the probability of fuel impingement or being adsorbed on the combustion chamber wall, where large amounts of PM are generated during combustion. In addition, low in-cylinder temperature and weak airflow movement intensity at the edge of the cylinder are detrimental to the oxidation of PM, resulting in an increase in PM emissions when diesel is injected too early.

Figure 5 shows the effect of diesel injection timing on indicated thermal efficiency and total particulate number. At low load BMEP 0.2MPa, when diesel injection timing postponed from -10°CA ATDC to 4°CA ATDC, the indicated thermal efficiency decreases. Delaying diesel injection causes advanced (away from TDC) combustion center. However, when diesel is injected before -40°CA ATDC, and CA50 occurs before top dead center. Such a phenomenon indicates that part of the fuel's heat release contributes negative work. In addition, the low combustion temperature in the low-load cylinder combined with the heat absorption and cooling of methanol vaporization leads to low fuel combustion efficiency and incomplete oxidation of hydrocarbon fuels, which is manifested in the production of a large amount of unburned HC and CO. Moreover, the increase of spray penetration caused by early diesel injection at low-load also causes the phenomenon of fuel wet wall and increase fuel consumption. At BMEP 0.4 MPa, the increase of in-cylinder temperature can alleviate the decline of thermal efficiency caused by early injection. When diesel is injected within -40°CA and -30°CA ATDC, the injection time is more appropriate. Diesel and air could be fully pre-mixed. After reaching the ignition condition, diesel fuel spontaneously combusts and ignites the surrounding methanol mixture. During the combustion progress, there is no diesel diffusion combustion in the cylinder, and the combustion releases heat rapidly, which promotes the improvement of indicated thermal efficiency.



(a) BMEP=0.2MPa (b) BMEP=0.4MPa

Figure 4. The effect of injection timing on particulate size distribution under different loads



(a) BMEP 0.2MPa (b) BMEP 0.4MPa

Figure 5. The effect of injection timing on indicated thermal efficiency and total PM number

4. Conclusions

Based on the electronic controlled diesel-methanol dual-fuel engine, the effects of methanol substitution ratio and diesel injection timing on engine combustion, emissions and fuel economy at different loads were investigated in this study. The research results are summarized as follows:

1) Diesel injection timing has a significant effect on the in-cylinder combustion characteristics. When the diesel injection timing is before -30°CA ATDC, in-cylinder combustion includes diesel premixed combustion and flame propagation that ignites methanol mixture, without diffusion combustion stage. When diesel is injected in the -40°CA ~ -30°CA ATDC interval, the emission of gaseous pollutants and PM of the engine is reduced, and high thermal efficiency can be obtained.
2) Diesel injection timing advanced over -30°CA ATDC actually could be applied while it does not favor engine gaseous emission as well as thermal efficiency. Deep analyses indicated that this might be due to too lean mixture prepared before combustion. However, too advanced injection timing also limited the methanol substitution ratio. As indicated by this investigation, only 40% methanol substitution ratio could be achieved when injection timing -60°CA ATDC is adopted. With proper diesel injection timing, methanol substitution ratio could achieve as high as roughly 75% at medium engine load condition.

3) At low loads, with increased methanol injection amount, the peak heat release rate and indicated thermal efficiency decrease. At medium and high loads, with increased methanol injection amount, the maximum combustion pressure and peak heat release rate increase, the equivalent fuel consumption rate decreases slightly, and the indicated thermal efficiency is equivalent to that of diesel engine.
4) For all tested loads, the addition of methanol causes decreased NO_X emissions, and increased HC and CO emission. The PM emissions of dual-fuel engines are lower than that of diesel engines, especially at medium and high loads, as the proportion of methanol increases, the PM decreases significantly.

According to the whole research, it was found that diesel injection timing greatly affected engine performances. Super advanced fuel injection timing favored engine thermal efficiency and characterized engine combustion as premixed feature. However, advanced fuel injection timing led to high pressure rise rate thus challenged engine mechanical performance. So, the investigation over diesel injection timing was limited to medium and low engine load (BMEP 0.2MPa and 0.4MPa) in this article. In the future, our further investigation would focus on implementing super advanced diesel injection timing under high engine load conditions combined with strategies like EGR. Besides, the approaches for improving thermal efficiency of methanol-diesel dual fuel engine would also be explored. Possible approaches may relate to low pressure methanol direct injection, hot EGR strategy and so on.

Nomenclature

ATDC: after the top dead center
BMEP: break mean effective pressure
BTDC: before the top dead center
BTE: break thermal efficiency
CA10: crank angle degree at which 10% accumulative heat released
CA50: crank angle degree at which 50% accumulative heat released
CA90: crank angle degree at which 90% accumulative heat released
°CA: crank angle degree
DOC: diesel oxidation catalyst
EBSFC: equivalent brake specific fuel consumption rate of diesel
ECU: electronic control unit
EGR: exhaust gas recirculation
HC: hydrocarbon
ITE: indicated thermal efficiency
M5: blending fuel with 5% methanol in mass
M10: blending fuel with 10% methanol in mass
M15: blending fuel with 15% methanol in mass
M20: blending fuel with 20% methanol in mass
MSR: methanol substitution ratio
NOx: nitrogen oxide
PM: mass of particulate matter

PN: number of particulate matter
TDC: top dead center
THC: total hydrocarbon

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