Effect of equivalence ratio on combustion and emissions of a dual-fuel natural gas engine ignited with diesel

Jinbao Zheng\textsuperscript{a,b}, Jinhua Wang\textsuperscript{a,*}, Zhibo Zhao\textsuperscript{b}, Duidui Wang\textsuperscript{b}, Zuohua Huang\textsuperscript{a,*}

\textsuperscript{a} State Key Laboratory of Multiphase Flow in Power Engineering, Xi'an Jiaotong University, Xi'an 710049, China
\textsuperscript{b} Wuxi Fuel Injection Equipment Research Institute, No.15 Qianrong Road, Wuxi 214063, China

\textbf{Highlights}

- Effect of equivalence ratio on combustion and emissions was investigated in natural gas dual fuel engine.
- Comparison between dual-fuel and spark ignition modes was conducted with same IMEP.
- Diesel injection strategy is investigated in dual fuel engine at low load.
- Diesel injector with double layers spray is investigated to decrease CH\textsubscript{4} emissions.
- The dual fuel engine gives lower gas consumption rate compared with spark ignition mode.

\textbf{Abstract}

China is the world largest natural gas engine market, and the natural gas price has a large fluctuation range, Natural gas-diesel dual-fuel engine may be one of the options to cope with gas price fluctuations. Effect of equivalence ratio on combustion and emissions with a low compression ratio of 14.2 in a dual-fuel 6 cylinder engine was experimentally studied. Comparison between dual-fuel and spark ignition engine at stoichiometric combustion and same IMEP was conducted. Results show that the gas consumption rate decreases due to the increase of equivalence ratio regardless of the strategy in nozzle parameter, exhaust gas recirculation (EGR) rate, injection parameter. Peak heat release rate and exhaust gas temperature increased, and decrease of combustion duration is obtained due to the high equivalence ratio in dual-fuel ignition mode. The gas consumption rate trend of spark ignition mode is similar to that of dual-fuel engine with the increase of equivalence ratio. Dual-fuel engine with low compression ratio of 14.2 run as spark ignition (SI) gas engine at stoichiometric ratio, but the difference lies in the combustion process. The peak heat release rate of dual-fuel engine presents higher value, while the combustion duration of dual-fuel engine is shorter, and the gas consumption rate of dual fuel engine is lower than that of SI engine.

1. Introduction

The increasing fuel prices and stringent emission regulation drives the development of high-efficient low-emission internal combustion engines. Compression Ignition engines still remain an efficient option due to their high compression ratio, high thermal efficiency, and un-throttled lean operation. The inherent high torque capability and thermal efficiency make the diesel engines remain an attractive option for the heavy-duty transportation sector. The increase of fuel consumption cost drives the engine sector to increase the engine efficiency or to use alternative fuels with reasonable cost.

In addition to the stringent pollution regulation, engine manufacturers are also required to meet challenging CO\textsubscript{2} emission regulations in the future and natural gas has its advantage in reducing CO\textsubscript{2} emission. The primary constituent of natural gas is CH\textsubscript{4} (methane), and methane has its inherent advantages over other choices. CH\textsubscript{4} is the simplest hydrocarbon molecule, having the highest H to C ratio and the lowest carbon-to-energy ratio among all fossil fuels, as well as a relatively low adiabatic flame temperature. These chemical properties make the combustion of CH\textsubscript{4} much cleaner than diesel in terms of CO\textsubscript{2}, NO\textsubscript{x} and soot. CH\textsubscript{4} produces up to 20\% lower brake specific CO\textsubscript{2} emissions in an engine compared to other typical hydrocarbon fuels [1]. Meanwhile, unburned CH\textsubscript{4} has been reported to be a potent GHG (Green House Gas) and a strong contributor to climate change.

\textsuperscript{*} Corresponding authors.
E-mail addresses: jinhuawang@mail.xjtu.edu.cn (J. Wang), zhhuang@mail.xjtu.edu.cn (Z. Huang).

https://doi.org/10.1016/j.applthermaleng.2018.10.045
Received 28 June 2018; Received in revised form 8 October 2018; Accepted 10 October 2018
Available online 11 October 2018
1359-4311/ © 2018 Published by Elsevier Ltd.
Natural gas is currently an attractive substitute for diesel fuel in the Heavy Duty (HD) diesel transportation sector due to its relatively low cost and abundance. As a result, there has been an emergence of technologies that utilize natural gas in various combustion chamber orientations [4–7]. These technologies are diverse and can range from retrofit options to fully integrated or dedicated natural gas engines, each with their individual sets of advantages and disadvantages [8]. Typically, natural gas is introduced to the combustion chamber using port fuel injection (PFI) or direct injection (DI) systems, and uses an electronic spark plug to ignite the charge. This combustion process is characteristic of the Otto cycle as opposed to the Diesel cycle.

In the past decades, spark ignition natural gas engine usually adopted the lean burn combustion mode to increase engine thermal efficiency. However, the low exhaust temperature inhibits the unburned CH4 further oxidation since most oxidation catalysts require at least 350 °C tailpipe temperature. With latest introduced emissions regulations, particularly Euro VI with a CH4 limit of 0.5 g/kWh [1], it is a challenge to meet tailpipe-out CH4 emissions. Low exhaust gas temperatures during the new World Harmonized Transient Test Cycle (WHTC) [9] do not allow CH4 oxidation catalysts to reach operating temperature, therefore other approaches for reducing engine-out CH4 need to be investigated.

Three Way Catalyst (TWC) is used on natural gas engine for exhaust after-treatment. This design is low-cost due to the simple injection and after-treatment systems. However, with this design, combustion temperatures limit the power density, efficiency, and durability. It also suffers from the decreased efficiency at light load due to throttling to maintain the stoichiometric operation. Lean-burn gas engine is an alternative to spark-ignited stoichiometric gas engine and has been utilized in the HD market. Lean combustion reduces the in-cylinder temperature while improves engine thermal efficiency. It has no light load efficiency penalty since pumping losses can be avoided. Natural gas is proven as an effective mean to reduce NOx and Particulate Matter (PM). However, natural gas engine still has its problem of high unburnt CH4 at lean burn operation. Lean combustion results in relatively low tailpipe temperature which suppresses the further oxidation of CH4.

Reactivity Controlled Compression Ignition (RCCI) [10–12] is a dual-fuel LTC (Low Temperature Combustion) strategy. At least two fuels with high and low reactivity are used to control the combustion. The low reactivity fuel, such as ethanol, gasoline, or natural gas is introduced to the cylinder along with the inducted air, and EGR if applied, to form a well-mixed charge. The high reactivity fuel, such as diesel, is directly injected into the combustion chamber during the compression stroke to ignite the mixture with diesel compression autoignition. RCCI is a distributed auto ignition strategy with the fuel gradient controlling the rate of combustion. RCCI operation can reduce both NOx and PM, but UHC and CO emissions tend to increase compared to the conventional diesel combustion [13,14]. This combustion strategy is also sensitive to intake air temperature and EGR rate [15,16].

In research, on natural gas/diesel dual-fuel engine has received increased attention recent years [17–20]. Premixed Dual-Fuel Combustion (PDFC) [8] is another type of LTC dual-fuel strategy and differs to the conventional dual-fuel and RCCI. PDFC differs to RCCI in that combustion is not controlled by changing fuel substitution ratio but adjusting the timing and quantity of the second diesel injection around top dead center (TDC) [21–24]. The late injected diesel fuel is auto-ignited by compression temperature, and its timing and quantity controls the start of combustion of the premixed fuel-air charge. Results show that the PDFC could increase the peak pressure value at the part load, but CH4 emission still exceeds the regulation limit. Therefore, studies were conducted to optimize the combustion chamber [25,26] and use multi diesel injection [27–30] to decrease CH4 emissions, and Hydrogen is added to promote the combustion process as additive fuel in dual-fuel engine [31,32].

In China, natural gas price fluctuates significantly and may suffer from short supply at heating season, and users hope to use natural gas when its price at low stage, and choose diesel fuel when the gas price at high stage or lack of gas. In other words, the dual-fuel engine should have pure diesel mode. Therefore, OEM (Original Equipment Manufacturer) hopes to develop a dual-fuel engine. Lean burn mode is hard to solve the above problems, while CH4 also features a high resistance to auto-ignition and knocking (RON = 120), which is of particular importance for premixing in compression ignition (CI) engines due to their high compression ratio.

Table 1
Specifications of the dual-fuel engine.

<table>
<thead>
<tr>
<th>No.</th>
<th>Type</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Engine type</td>
<td>4 stroke, SOHC, 4 valves, turbo charge, in-line</td>
</tr>
<tr>
<td>2</td>
<td>Combustion shape</td>
<td>Ω</td>
</tr>
<tr>
<td>3</td>
<td>Compression ratio</td>
<td>14:2:1</td>
</tr>
<tr>
<td>4</td>
<td>Cylinder number</td>
<td>6</td>
</tr>
<tr>
<td>5</td>
<td>Bore and stroke</td>
<td>123 mm × 155 mm</td>
</tr>
<tr>
<td>6</td>
<td>Displacement</td>
<td>11 L</td>
</tr>
<tr>
<td>7</td>
<td>Diesel supply</td>
<td>Bosch CR CP2.2+</td>
</tr>
<tr>
<td>8</td>
<td>Air supply</td>
<td>Holset VGT</td>
</tr>
<tr>
<td>9</td>
<td>Natural gas supply</td>
<td>Port Fuel Injection</td>
</tr>
<tr>
<td>10</td>
<td>Firing order</td>
<td>1-5-3-6-2-4</td>
</tr>
</tbody>
</table>

Table 2
Diesel injection and natural gas injection parameters.

<table>
<thead>
<tr>
<th>No.</th>
<th>Type</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Injector mode</td>
<td>Cylinder Direct Injection</td>
</tr>
<tr>
<td>2</td>
<td>Company</td>
<td>Bosch CP2.2</td>
</tr>
<tr>
<td>3</td>
<td>Injector</td>
<td>Bosch China</td>
</tr>
<tr>
<td>4</td>
<td>Drive</td>
<td>Current peak25A-bond12A</td>
</tr>
<tr>
<td>5</td>
<td>Nozzle</td>
<td>1760 L/min@10 MPa</td>
</tr>
<tr>
<td>6</td>
<td>Natural gas Injection System</td>
<td>Port Fuel Injection</td>
</tr>
<tr>
<td>7</td>
<td>Company</td>
<td>Homemade</td>
</tr>
<tr>
<td>8</td>
<td>Injector</td>
<td>Current peak 5.5A-bond2.5A</td>
</tr>
<tr>
<td>9</td>
<td>Drive</td>
<td>30 kg/h @?bar</td>
</tr>
</tbody>
</table>

Table 3
Test bench and measurements.

<table>
<thead>
<tr>
<th>No.</th>
<th>Instrument</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Dynamometer</td>
<td>AVL DYNOROAD 604/5.5 SX</td>
</tr>
<tr>
<td>2</td>
<td>Diesel flow meter</td>
<td>AVL 735</td>
</tr>
<tr>
<td>3</td>
<td>Natural gas flow meter</td>
<td>EMERSON CF050M Coriolis</td>
</tr>
<tr>
<td>4</td>
<td>Smoke meter</td>
<td>AVL483</td>
</tr>
<tr>
<td>5</td>
<td>Air flow meter</td>
<td>ABB PMT700-P 0-4000 kg/h</td>
</tr>
<tr>
<td>6</td>
<td>Exhaust gas analyzer</td>
<td>HORIBA MEGA-7500 double channel</td>
</tr>
<tr>
<td>7</td>
<td>Combustion analyzer</td>
<td>AVL Indicom 642</td>
</tr>
<tr>
<td>8</td>
<td>Crank encode</td>
<td>AVL 365C</td>
</tr>
<tr>
<td>9</td>
<td>Cylinder pressure sensor</td>
<td>AVL GH14D</td>
</tr>
</tbody>
</table>

Table 4
Natural gas compositions.

<table>
<thead>
<tr>
<th>No.</th>
<th>Components</th>
<th>Volume fraction</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>CH4</td>
<td>92.096</td>
</tr>
<tr>
<td>2</td>
<td>C2H6</td>
<td>3.899</td>
</tr>
<tr>
<td>3</td>
<td>C3H8</td>
<td>0.674</td>
</tr>
<tr>
<td>4</td>
<td>i-C4H10</td>
<td>0.119</td>
</tr>
<tr>
<td>5</td>
<td>n-C4H10</td>
<td>0.124</td>
</tr>
<tr>
<td>6</td>
<td>i-C5H12</td>
<td>0.036</td>
</tr>
<tr>
<td>7</td>
<td>n-C5H12</td>
<td>0.025</td>
</tr>
<tr>
<td>8</td>
<td>C6</td>
<td>0.077</td>
</tr>
<tr>
<td>9</td>
<td>N2</td>
<td>1.272</td>
</tr>
<tr>
<td>10</td>
<td>CO2</td>
<td>0.678</td>
</tr>
<tr>
<td>11</td>
<td>H2S(Sm/g/m²)</td>
<td>1.1477</td>
</tr>
</tbody>
</table>
Combined with the advantages of stoichiometric combustion and compression ignition, the objective of this study is to investigate the effect of equivalence ratio on combustion and emissions in a dual fuel natural gas engine ignited with diesel, especially the combustion characteristics with stoichiometric ratio in order to use the TWC to meet the emission regulations in dual-fuel mode in the further study.

The experiment was conducted in a 6 cylinder engine with low compression ratio piston and an intake throttle. The compression ratio is 14.2, it is higher than the spark ignition gas engine, but lower than the traditional diesel engine. The effect of equivalence ratio on spark ignition engine, and the comparison between dual-fuel and spark ignition at stoichiometric ratio and same IMEP will be conducted.

2. Experimental setup and procedures

2.1. Experimental setup

Experiment was conducted on a 6 cylinder four stroke CI base engine. The diesel engine was re-fitted to a dual-fuel engine. The main specifications of the dual-fuel engine are summarized in Table 1. The combustion chamber of the engine has special open shape with a compression ratio of 14.2.

The common rail direct injection system is an attachment to the original engine. Natural gas is introduced into the intake pipe with a homemade single point injection system and mixes with the cooled EGR. Then, the mixtures goes into the intake manifold of the engine.

### Table 5

<table>
<thead>
<tr>
<th>Case</th>
<th>rpm</th>
<th>IMEP (bar)</th>
<th>Nozzle</th>
<th>Cone angle (°)</th>
<th>pressure (MPa)</th>
<th>GSR (%)</th>
<th>SOI1 (°CA BTDC)</th>
<th>SOI2 (°CA BTDC)</th>
<th>Q Inj1 (mm³)</th>
<th>EGR rate (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>case1</td>
<td>1204</td>
<td>5.6</td>
<td>8-18</td>
<td>151</td>
<td>70</td>
<td>70</td>
<td>40</td>
<td>6</td>
<td>4</td>
<td>0</td>
</tr>
<tr>
<td></td>
<td>1204</td>
<td>16</td>
<td>8-18</td>
<td>151</td>
<td>70</td>
<td>80</td>
<td>40</td>
<td>6</td>
<td>4</td>
<td>0</td>
</tr>
<tr>
<td>case2</td>
<td>1743</td>
<td>6</td>
<td>8-18</td>
<td>151</td>
<td>70</td>
<td>85</td>
<td>30</td>
<td>10</td>
<td>8</td>
<td>20</td>
</tr>
<tr>
<td></td>
<td>1743</td>
<td>6</td>
<td>8-16/4-11</td>
<td>130/90</td>
<td>70</td>
<td>70</td>
<td>52</td>
<td>10</td>
<td>8</td>
<td>20</td>
</tr>
<tr>
<td></td>
<td>1743</td>
<td>6</td>
<td>8-11</td>
<td>70</td>
<td>75</td>
<td>60</td>
<td>8</td>
<td>15</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Fig. 1. Emissions with three strategies at 5.6 bar IMEP with 8-18 nozzle.

Fig. 2. Engine performance with three strategies at 5.6 bar IMEP with 8-18 nozzle.
Specification of natural gas injection system and diesel common rail injection system are provided in Table 2. The engine is capable of running various ratios of diesel and natural gas simultaneously. Dual-fuel injection system is controlled by dedicated electronic control unit (ECU) flexible on diesel injection pulse number, mass, timing and gas substantial rate. Transient operation is not considered in this study as all experimental test points are at the steady-state.

Cylinder pressure was measured using 6 AVL GH14D pressure sensors through 6 AVL Piezo charge amplifiers by AVL 642 Indicom. Engine test bench is equipped with dynamometer of AVL660. A CMF050M Coriolis mass flow meter made by EMERSON is used to measure the natural gas flow rate. Diesel mass flow rate is measured by AVL735 which is also a Coriolis mass flow meter. Engine-out gaseous emissions are measured by Horiba MEXA 7500D EGR. In addition to the standard gaseous emissions (NOx, THC, CO, CO2), the unburned CH4 is also measured. Soot measurements were performed with an AVL 483 smoke meter. All the test apparatus are shown in Table 3. The natural gas was transmitted from the West areas to the East China through the pipeline. Gas density is 0.7626 kg/m³ and heat value is 38.2301 MJ/m³.

Components of natural gas are listed in Table 4. Diesel is low sulfur diesel (≤10 ppm) sold in the market.

2.2. Experimental methodology

CO2 measurement in the intake manifold is used to determine EGR rate by the following equation.

\[
EGR = \frac{CO_2(\text{intake}) - CO_2(\text{ambient})}{CO_2(\text{exhaust}) - CO_2(\text{ambient})} \times 100\%
\]

Gas substitution rate (GSR) is defined as the energy of the gaseous fuel over the total fuel energy input. The GSR is calculated by equation.

\[
GSR = \frac{m_{\text{gas}} \cdot LHV_{\text{gas}}}{m_{\text{gas}} \cdot LHV_{\text{gas}} + m_{\text{diesel}} \cdot LHV_{\text{diesel}}} \times 100\%
\]

where \(m\) is the cyclic fuel mass, LHV is low heat value.

Equivalence ratio, \(\phi\) is defined as the mass ratio of the theoretical air required for the completely combustion of the fuel injected into the cylinder to the actual air volume in the cylinder. For dual-fuel engines, \(\phi\) is calculated by equation

\[
\phi = \frac{m_{\text{gas}} \cdot L_{\text{gas}} + m_{\text{diesel}} \cdot L_{\text{diesel}}}{m_{\text{air}}} \times 100\% \tag{3}
\]

where \(L\) is theoretical air/fuel ratio.

Many variables are included in this study. For diesel injections, the variables include the strategy of injection timing, quantity, and injection pressure. Other variables like EGR and boost pressure are also varied. The injection strategy and nozzle type in the experiment is optimized specifically, but they are not the focus of this study. The effect of equivalence ratio on combustion and emission of dual-fuel engine with low compression ratio was carried out by the ECU with throat valve, VGT and exhaust gas valve. Diesel was directly injected into the cylinder at 70 MPa with 8 × 0.18 mm nozzles. PUMA system collected all the performance and emission data, and Indicom stored 200 cycle combustion analysis data with high-speed and high resolution interval 0.025 °CA. In this study, two engine speeds of 1204 r/min and 1743 r/min are conducted, which refer to ESC test engine speeds \(n_A\) and \(n_C\).

3. Results and discussions

3.1. Effect of equivalence ratio on combustion in dual-fuel engine

3.1.1. Different injection strategies at low load

The effect of equivalence ratio on combustion and emissions at three
strategies as indicated in case1 of Table 5 were studied using two early injection strategy. The engine operates at the 5.6 bar IMEP and 1204 r/min.

Fig. 1 shows the effect of equivalence ratio on emissions at three strategies and low load conditions. It is observed that with the increase of equivalence ratio, both CH4 and CO emissions are decreased, while NOx emission increased slightly. However, the strategy 3 uses 20% EGR, bringing the lowest NOx emission. Natural gas combustion produces almost smoke-free emission.

Fig. 2 illustrates engine performance at three strategies and low load. An increase in engine boost pressure leads to the increase of the intake charge and the decrease of equivalence ratio. With the increase of equivalence ratio, higher thermal efficiency of the engine can be achieved. It is important to note that the change in equivalence ratio will change engine exhaust temperature. For example, equivalence ratio in Fig. 3 changes from 0.4 to 0.6, then exhaust temperature changes from 200 °C to 400 °C.

Fig. 3 presents the cylinder pressure analysis of three injection strategies at low load. Although the injection timing, fuel mass and EGR rate were different, the compression pressure and the peak cylinder pressure decreased significantly with the increase of equivalence ratio, and the start of combustion (SOC) is delayed, the maximum heat release rate is increased. The reduction of charge amount in the cylinder reduces the compression pressure near the TDC and the temperature after the compression. Temperature strongly affects the reaction activity of the mixture, and the decrease in temperature brings the delay of the SOC and also the decrease of the combustion pressure. The temperature after the combustion in cylinder near the TDC is high and the amplitude is shown in Fig. 4. After the expansion, the exhaust temperature before the turbo is also increased as is shown in Fig. 2.
For the same substitution rate, reducing the amount of air through the intake port increases the equivalence ratio. The high equivalence ratio presents higher average temperature in the cylinder. Compared to the lean burn combustion, exhaust gas temperature is increased due to the decrease of air mass, and this is the obvious behavior of the high equivalence ratio combustion. Temperature strongly affects chemical reaction rate, and the increase of equivalence ratio increases the chemical reaction rate of CO and CH₄, and this improves the combustion efficiency, promoting the oxidation of CO and CH₄.

As shown in Fig. 3, with the increase of equivalence ratio, the heat release rate curve moves far away from TDC, and the peak heat release rate is increased. Combustion parameters in Fig. 5 exhibit the variations of heat release rate with equivalence ratio. It can be seen that, with the increase of equivalence ratio, both CA10 and CA50 are delayed, the duration of CA10 to CA90 is shortened and CA90 is reversed. CA90 indicates the location of the combustion end, CA50 of strategy 1 and 2 are advanced with the decrease of equivalence ratio. Since the injection timing of strategy 3 is much close to the TDC, the CA50 of strategy 3 is advanced prior to TDC at larger equivalence ratio. When equivalence ratio decreases, CA50 will postpone and goes after the TDC, but it is still in the ideal range of 5–8 °CA after top dead center (ATDC). CA90 increases with the decrease of equivalence ratio, while the CA90 of strategy 3 is still earlier than those of strategy 1 and strategy 2. This indicates that strategy 3 has the earliest combustion end and potentially higher brake thermal efficiency (BTE).

The start of combustion is also related to the temperature and local equivalence ratio near the ignition position. Decrease of the air mass decreases the temperature after compression and oxygen concentration, contributing to the delay of the ignition time and changing the position of CA10. Peak heat release rate is increased because of the multiple ignition points in the diesel injection spray. The concentration stratification of diesel/air ratio in cylinder is more obvious at lean burn mode, and this is similar to the pure diesel engine. The diesel/air ratio near the diesel spray atomization zone is relatively strong, while the region away from the diesel spray atomization zone is occupied by the natural gas. The equivalence ratio defined in Eq. (3) in this paper is an overall definition, in which natural gas and diesel is weighted as a whole part. Diesel fuel is more easily ignited because of the fuel...

![Fig. 6. Cylinder pressure versus equivalence ratio at 16 bar IMEP with 8-18 nozzle.](image)

![Fig. 7. Emissions versus equivalence ratio at 16 bar IMEP with 8-18 nozzle.](image)
reaction activity. Once the ignition is initialized, the mixture of gas and air around is rapidly ignited to form the flame propagation, and drives the surrounding natural gas to burn with high burning rate because of the premixed mixture. Therefore, at high equivalence ratio, the peak heat release rate increases and the duration of combustion is shortened, as shown in Fig. 5.

It is noted that the strategy 3, with two injections in combination with EGR, presents the high thermal efficiency and low NOx emissions. The higher equivalence ratio also leads to lower CH4 emissions, while engine exhaust temperature will increase. Effect of equivalence ratio on burning rate becomes more obvious in the case of EGR because EGR inhibits the burning rate, as shown in Fig. 3c.

### 3.1.2. Different strategies at high load

The investigation was carried out at high load of IMEP = 16 bar at 1204 r/min as indicated in case 2 of Table 5. With the higher in-cylinder temperature at high load condition, the reactivity of single diesel injection is enough to ignite the natural gas. Considering the high NOx emission at high load, EGR rate of 13% was used to reduce NOx emission.

The effect of equivalence ratio on combustion at high load is shown in Fig. 6. With the increase of equivalence ratio, the SOC is retarded, and the peak heat release rate is increased while combustion duration is shortened. These results are similar to those at the low load operation. When EGR is used, the trend of equivalence ratio effect is similar, but maximum heat release rate is decreased. Fig. 7 shows engine emissions versus equivalence ratio at high load. With the increase of equivalence ratio, unburned CH4 decreases while soot increases slightly, CO emission decreases and then increases, NOx emissions increases slightly. CH4 emissions determines the thermal efficiency of dual-fuel engine, CH4 emissions in Fig. 7 decreases from 22.2 g/kWh to 3.83 g/kWh with the increase of equivalence ratio from 0.56 to 0.78. From the gas consumption and GSR, combustion efficiency of natural gas is obtained on the assumption that diesel is completely burned, combustion efficiency of natural gas is 97.2% in the equivalence ratio of 0.78, while 85.2% is obtained in the equivalence ratio of 0.56. With introduction of EGR, NOx decreases significantly and CH4 emission increases. Soot emission is highly sensitive to the variation of equivalence ratio. Under low equivalence ratio, the soot emission is high, and this presents a typical diesel engine emission behavior. It is noted that we only used 70% energy substantial rate, and if higher substitution rate was introduced, it is predicted that the impact of equivalence ratio on soot emissions will be decreased.

Fig. 8 shows the engine performance versus equivalence ratio at high load. The exhaust gas ambient pressure reduced from 10 kPa to 6 kPa, but the exhaust temperature is increased by 200 °C due to the reduced intake pressure and air mass. Increasing the equivalence ratio decreases the gas consumption and improves engine thermal efficiency. This behavior is similar to that at low load operation. In the absence of EGR, the maximum thermal efficiency can reach to 41.5%. In the case of 13% EGR, the maximum thermal efficiency is 40.7%.

At high load, the efficiency of the dual fuel engine is close to that of diesel engine. High temperature increases the reaction rate at high load. The oxidation of trapped CH4 in the cylinder narrow gap also increases because of the higher temperature. Compared to that at the low load, CH4 emission is decreased at high load. EGR can suppress the reaction rate and reduce the tendency of knocking at high load.

---

**Fig. 8.** Engine performance versus equivalence ratio at 16 bar IMEP with 8-18 nozzle.
Fig. 9. Performance with three type nozzles at IMEP = 6 bar.

Fig. 10. Emissions with three type nozzles at IMEP = 6 bar.
Three types of diesel injection nozzle were used with IMEP = 6 bar at 1743 r/min in this study. Their specifications are shown in case 3 Table 5. The reactivity of injected fuel in the cylinder with different nozzle types are different, thus the control parameters which match the injectors need to be optimized. Even with the new design nozzle which has small hole diameter and narrow cone angle, the thermal efficiency is increased with the increase
of equivalence ratio. Double layers’ orifice 8-16/4-11 nozzle strategy at high equivalence ratio presents the lowest CH₄ and CO emissions and the highest thermal efficiency. NOx emission increases remarkably and thermal efficiency decrease as equivalence ratio is close to the stoichiometric ratio. Thermal efficiency maintains high value (close to 32%) in the broad range of equivalence ratio. This is the unique effect of the double layer design of the nozzle hole. The traditional 8-hole nozzle design fully considers the air inlet swirl intensity to layout the circumferential direction uniformly, but neglects the large conical space below the spray umbrella in dual-fuel engine. The upper layer of the nozzle designed in this experiment can make full use of the upper space of the combustion chamber, and the lower layer has designed 4 holes with narrow cone angle and smaller hole diameter. The spray of this nozzle ignites the natural gas in the core area of the combustion chamber. Therefore, CH₄ emission from incomplete combustion is reduced and higher thermal efficiency is achieved.

As shown in Figs. 9 and 10, the emissions and combustion performances with three types of nozzle are similar. Decreasing equivalence ratio is the result of decreasing the boost pressure and air mass goes into the cylinder which leads to the decrease of exhaust gas ambient pressure when maintains the same output power.

Figs. 11 and 12 show the effect of equivalence ratio on cylinder pressure and heat release rate with three types of nozzle. Decreasing intake pressure and charge mass decreases the peak value of compression and combustion pressure in cylinder at TDC. It presents the same trend with the previous discussion. It is worth noting that 8-16/4-11 nozzle type has relatively higher heat release rate, faster burning rate and shorter combustion duration with high equivalence ratio. This also contributes to the high thermal efficiency as shown in Fig. 10. Other two types nozzle present low heat release rate and long combustion duration. The diameter of the 811 nozzle is less than that of 8-18 nozzle. Under the same injection pressure, the smaller the diameter of the injection hole is, the smaller the droplet diameter is and the better atomization and mixture formation. The shorter evaporation time and higher possibility contacting with the oxygen in the air are realized under the same diesel quantity. Under the same pressure and temperature, the changes of these parameters will greatly affect fuel reactivity, and shorten the ignition delay time, thus the position CA10 is advanced of that with larger hole diameter. The 8-16/4-11 nozzle adopts the double layer holes design, and diesel spray distributes more reasonably in the cylinder, and fast combustion rate is achieved. This contributes to the increase of peak heat release rate and the decrease of combustion duration. The upper nozzle can use smaller hole diameter in dual-fuel engine, but the engine expects to maintain the pure diesel mode. So, the 8-16/4-11 configuration is selected for similar high pressure flow to that of the original diesel engine, which can meet the power demand of the pure diesel mode.

![Fig. 14. Effect of equivalence ratio on combustion.](image)

![Fig. 15. Emissions with equivalence ration in SI mode.](image)
The spark ignition engine adopts the stoichiometric combustion mode and configuration of external cooling EGR and TWC after-treatment which has been widely used in European and American gas engine market. The SI engine used for this work is an inline 6-cylinder 7.2 L heavy duty natural gas engine with cooled EGR and a single point injection system. The compression ratio is 12. The engine is equipped with a TWC, a waste-gate turbocharger and a combination of an EGR valve and intake throttle. Although the spark ignition engine is modified from the diesel engine, the combustion mode of the natural gas engine is more close to that of gasoline engine. Spark plug ignition timing determines the start of combustion. Effect of equivalence ratio on combustion and emissions are studied at the load of IMEP = 5.6 bar and 7.6 bar in 1204 rpm.

3.3. Combustion and emission analysis

The spark ignition engine uses a throttle valve to adjust the intake charge, which is similar to the gasoline engine. This study is based on optimized ignition timing. Fig. 13 shows the heat release rate, cylinder pressure and in-cylinder bulk gas temperature at various equivalence ratios. With the increase of equivalence ratio, the compression pressure at TDC, the overall combustion pressure and maximum heat release rate increase slightly and combustion duration is shorten. The low compression pressure is mainly caused by relatively low compression ratio. In addition, the range of equivalence ratio is very narrow and the minimum value is 0.714. The equivalence ratio of spark ignition engine is limited by the ignition energy and the flame propagation. In the high dilution combustion mode, the flame nuclei of lean mixture is hard to sustain development and may lead to misfire. Spark ignition mode is adopted in natural gas engine, and its heat release rate shows the pattern of the gasoline engine.

3.3.1. Combustion and emission analysis

The natural gas engine uses a throttle valve to adjust the intake charge, which is similar to the gasoline engine. This study is based on optimized ignition timing. Fig. 13 shows the heat release rate, cylinder pressure and in-cylinder bulk gas temperature at various equivalence ratios. With the increase of equivalence ratio, the compression pressure at TDC, the overall combustion pressure and maximum heat release rate increase slightly and combustion duration is shorten. The low compression pressure is mainly caused by relatively low compression ratio. In addition, the range of equivalence ratio is very narrow and the minimum value is 0.714. The equivalence ratio of spark ignition engine is limited by the ignition energy and the flame propagation. In the high dilution combustion mode, the flame nuclei of lean mixture is hard to sustain development and may lead to misfire. Spark ignition mode is adopted in natural gas engine, and its heat release rate shows the pattern of the gasoline engine.
Fig. 14 shows the effect of equivalence ratio on combustion process at IMEP = 5.6 bar and 7.6 bar. With the increase of equivalence ratio, CA10 is retarded and CA90 is advanced because of high heat release rate. Although this can make CA50 in the range of 7–9 °CA, a short combustion duration is achieved. As indicated in Fig. 14, the combustion duration of stoichiometric ratio was shortened to 17 °CA at load of 7.6 bar IMEP. The optimal ignition timing was used to keep the combustion center CA50 in an economy condition, thus the change in combustion parameters at the two loads are in a very small range.

The effect of equivalence ratio on emissions in a spark ignition natural gas engine is shown in Fig. 15. CH4 emission decreases first and then increases, NOx emissions increase first and then decrease, giving its highest value at equivalence ratio of 0.85. CO emissions maintain low value before equivalence ratio of 0.95, and then increase remarkably with further increase of equivalence ratio. TWC has high conversion efficiency at the stoichiometric combustion. It is widely used in both spark ignition gasoline engine and gas engine to reduce the emissions.

A Bosch intake throttle valve was used to control the equivalence ratio by ECU. Closing the valve gradually can quickly reduce the charging pressure and air inflow. This experiment adopted the optimized spark ignition angle, which has a tendency to delay with the increase of equivalence ratio. In Fig. 16, under the condition of IMEP = 7 bar, the spark ignition angle is postponed from 39 °CA before top dead center (BTDC) to 16 °CA BTDC when equivalence ratio increases from 0.53 to stoichiometric ratio. Brake specific fuel consumption (BSFC) increases slightly, while exhaust temperature increases remarkably. As a result, the core issue of gas engine with stoichiometric combustion mode is the high temperature of exhaust gas and the reliability risk caused by high temperature (see Fig. 17).

BSFC of IMEP = 5.6 bar is higher than that of IMEP = 7.6 bar, low load presents higher BSFC than high load because of the decrease of thermal efficiency at low load. Meanwhile, lean combustion mode demonstrates low BSFC than relatively rich combustion. Since TWC requires engine operating on the stoichiometric condition, thus it is hard to simultaneously maintain both high thermal efficiency and low
emission.

3.3.2. Comparision between dual-fuel and SI at IMEP = 5.6 bar

Fig. 18 presents the comparison on combustion and emissions between dual-fuel engine and spark ignition at stoichiometric ratio at IMEP = 5.6 bar. Under the same operating condition, the pressure at TDC, the peak cylinder pressure and average bulk gas temperature of cylinder are close among them, and big difference lies in the combustion process. The peak heat release rate of the spark ignition engine is lower than that of the dual-fuel engine. Combustion duration of the spark ignition engine is longer than that of the dual-fuel engine. Dual-fuel engine has more parameters to be adjusted like gas substitution rate, diesel fuel ratio and injection timing. Although the ignition timing of the dual-fuel engine is later than that of the spark ignition engine, it still demonstrates higher peak heat release rate and shorter combustion duration than those of the spark ignition engine (see Fig. 19).

Fig. 18 shows the combustion parameters of CA10, CA50, CA90, CAPmax, CARmax, CATmax and CAdQmax, of dual-fuel engine and spark ignition engine. CAPmax and CAdQmax of dual-fuel engine is postponed compared with those of spark ignition engine, but the combustion duration of CA10-CA90 is shorter than that of spark ignition engine due to its fast heat release rate and rapid pressure rise. Under the same IMEP and peak combustion pressure, the maximum heat release rate increases from 81.9 kJ/°CA to 118.8 kJ/°CA, and peak rate of pressure rise increases from 1.7 bar/°CA to 2.8 bar/°CA. This is resulted from the burning behavior in the combustion chamber. The natural gas has a higher ignition temperature of 650 °C, and the temperature near the TDC with compression ratio of 12 is not enough to reach that auto-ignition temperature. External spark plug is assisted to ignite the mixture. The dual-fuel combustion mode is different. The diesel spray of the early short injection is mixed with the intake air and forms homogeneous mixture in the cylinder. Although the diesel itself has a strong reactivity, the pressure and temperature in the cylinder are not high enough to support the ignition because of the early injection timing. The second injection timing is set as 20 °CA BTDC. Because the load is not high enough, the temperature does not reach the auto ignition temperature until 10.2 °CA ATDC. Heat release rate of the dual-fuel engine is faster and combustion duration is shorter.

Comparison on performance and emissions of the two engines are shown in Fig. 20. BSFC of the dual-fuel engine is 254.9 g/kW·h, which is slightly lower than that of BSFC of the spark ignition engine. CH₄ emission of the dual-fuel engine is much lower than that of the spark ignition engine. Dual-fuel engine forms soot emission of 0.84 g/kW·h while it is hard to detect soot emission in spark ignition engine with AVL483 analyzer. Dual-fuel engine has gas energy substitute ratio of 85%, which is lower than the pure gas engine. If higher substitution rate was introduced, the dual-fuel engine can reduce the soot emission.

4. Conclusions

Effect of equivalence ratio on performance, combustion and emissions of a natural gas-diesel dual-fuel engine was studied and compared with a spark ignition engine. Combined with the current lean combustion technology, comparison was conducted on stoichiometric ratio at same IMEP and different combustion ignition modes. The main conclusions can be summarized as follows.

1. For dual-fuel engine mode, gas consumption rate decreases with the increase of equivalence ratio regardless of strategy variation such as nozzle parameters, EGR rate, injection parameters. The maximum heat release rate and exhaust gas temperature increase, and shorter combustion duration can be achieved with the increase of equivalence ratio.

2. Thermal efficiency is relative to the CH₄ emissions in dual-fuel engine. Thermal efficiency increases at larger equivalence ratio leads to the lower CH₄ emissions. Decreasing CH₄ emissions is the focus of dual fuel engine in the further research. In order to decrease CH₄ emissions at low load, improving the mixture of diesel and natural gas are useful to control the compression ignition with diesel injection strategy and double layers spray nozzle.

3. For spark ignition mode, the trend of gas consumption rate remains similar to that of dual-fuel engine with the increase of equivalence ratio. CH₄ emission decreases firstly and then increase, while NOx emissions increase firstly and then decrease. CO emissions increases remarkably when equivalence ratio is larger than 0.95 due to the significantly increase of the exhaust gas temperature.

4. Dual-fuel engine with low compression ratio of 14.2 run as SI gas engine at stoichiometric ratio, which is expected to use the TWC to meet the emission regulation. Under the same IMEP, dual-fuel and SI engine have the similar peak cylinder pressure and exhaust gas temperature, but the difference lies in the combustion process. The peak heat release rate of dual-fuel engine presents higher value than
that of the SI engine, and the dual fuel engine demonstrates shorter combustion duration and lower gas consumption rate.

Acknowledgements

This study is supported by National Natural Science Foundation of China (No. 51776164, 91441203). The authors would like to acknowledge the support of Foundation Project and engine test bench of WFIERI.

References


