Combustion characteristics of a direct-injection engine fueled with natural gas–hydrogen blends under different ignition timings

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Abstract

In this paper, combustion characteristics of a direct-injection spark-ignited engine fueled with natural gas–hydrogen blends under various ignition timings and lean mixture condition were investigated. The results show that the ignition timing has significant influence on engine performance, combustion and emissions. The time intervals between the end of fuel injection and ignition timing are very sensitive to direct-injection gas engine combustion. The turbulence in combustion chamber generated by the fuel jet maintains high and relatively strong mixture stratification is presented when decreasing the time intervals between the end of injection and the ignition timing, giving fast burning rate, high brake mean effective pressure, high thermal efficiency and short combustion durations. For specific ignition timing, the brake mean effective pressure and the effective thermal efficiency increase and combustion durations decrease with the increase of hydrogen fraction in natural gas. Exhaust HC concentration decreases and exhaust NOx concentration increase with advancing the ignition timing while the exhaust CO gives little variation under various ignition timings.

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1. Introduction

With increasing concern about energy shortage and environmental protection, research on improving engine fuel economy and reducing exhaust emissions has become the major researching aspect in combustion and engine development. Due to limited reserves of crude oil, development of alternative fuel engines has attracted more and more concern in the engine community. Alternative fuels, usually belong to clean fuels compared to diesel fuel and gasoline fuel in the combustion process of engines. The introduction of these alternative fuels is beneficial to slowing down the fuel shortage and reducing engine exhaust emissions. Natural gas is considered to be one of the favorable fuels for engines, and the natural gas fueled engine has been realized in both the spark-ignited engine and the compression-ignited engine. However, due to the slow burning velocity of natural gas and the poor lean-burn capability, the natural gas spark-ignited engine has the disadvantage of large cycle-by-cycle variations and poor lean-burn capability, and these will decrease the engine power output and increase fuel consumption [1,2]. Due to these restrictions, a natural gas engine is usually operated at the condition of stoichiometric equivalence ratio with relatively low thermal efficiency.

Traditional homogeneous spark-ignited natural gas engine has the disadvantage of low volumetric efficiency since natural gas occupies a fraction of intake charge, and this will decrease the fresh air into the cylinder and reduces the power output compared to that of gasoline engine. Meanwhile, homogeneous charge combustion has the difficult to burn the lean mixture. The introduction of natural gas direct-injection combustion can avoid the loss in volumetric efficiency as natural gas is directly injected into cylinder and has the flexibility in mixture preparation,
Traditionally, to improve the lean-burn capability and flame burning velocity of the natural gas engine under lean-burn conditions, an increase in flow intensity in cylinder is introduced, and this measure always increases the heat loss to the cylinder wall and increases the combustion temperature as well as the NO\textsubscript{x} emission [5]. One effective method to solve the problem of slow burning velocity of natural gas is to mix the natural gas with the fuel that possesses fast burning velocity. Hydrogen is regarded as the best gaseous candidate for natural gas due to its very fast burning velocity, and this combination is expected to improve the lean-burn characteristics and decrease engine emissions [6].

Blarigan and Keller investigated the port-injection engine fueled with natural gas–hydrogen mixtures [7]. Wong and Karim studied engine performance fueled by various hydrogen fractions in natural gas–hydrogen blends [8], and Bauer and Forest conducted an experimental study on natural gas–hydrogen combustion in a CFR engine [9]. Furthermore, studies on lean combustion capability of natural gas–hydrogen combustion and natural gas–hydrogen combustion with turbo-charging and/or exhaust gas recirculation were also conducted [10–12], and these studies showed that the exhaust HC, CO, and CO\textsubscript{2} concentrations could be decreased when exhaust concentration from an engine operated on natural gas–hydrogen blends were compared to those of natural gas engine. However, NO\textsubscript{x} may increase for natural gas–hydrogen combustion at rich mixture condition as the improvement of lean burning ability and increased flame propagation speed. NO\textsubscript{x} concentration can be greatly decreased through lean combustion and retarding of the ignition advance angle. The previous work mainly concentrated on homogeneous mixture fueled from the port and few literatures were reported on direct-injection engine. Shudo et al. investigated the combustion and emissions of an engine with port-injected hydrogen and in-cylinder injection natural gas [13], this type of engine needs two separate fueling systems and this make the system complicated. Huang et al. investigated the influence of the intervals between fuel injection timings and ignition timings on direct-injection natural gas combustion using a compression ignition machine, and revealed the importance of injection/ignition timing on combustion and emissions [14,15]. Since ignition timing is very important to the optimization of direct-injection gas engine, and no previous work was done on direct-injection natural gas–hydrogen engine, thus this study will concentrate on investigation of ignition timings on combustion characteristics of engine fueled with direct-injection natural gas–hydrogen mixtures, and providing practical guidance to engine optimization.

2. Experimental procedures

A single cylinder engine was modified into a natural gas direct-injection engine. The specifications of the engine are listed in Table 1. The injector used in the study is modified version from a gasoline direct-injection engine made by the manufacturer (Hitachi Co.). To increase the flow rate for natural gas application, the swirl near the tip of the nozzle was removed. The calibration of the pulse width with the injection amount was made by the manufacturer as well as by the authors. The flow rate of the injector under

Table 1

<table>
<thead>
<tr>
<th>Engine specifications</th>
<th></th>
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</thead>
<tbody>
<tr>
<td>Bore (mm)</td>
<td>100</td>
<td>Stroke (mm)</td>
<td>115</td>
</tr>
<tr>
<td>Stroke (mm)</td>
<td>115</td>
<td>Displacement (cm\textsuperscript{3})</td>
<td>903</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>8</td>
<td>Combustion chamber</td>
<td>Bowl-in-shape</td>
</tr>
<tr>
<td>Injection pressure (MPa)</td>
<td>8</td>
<td>Ignition source</td>
<td>Spark plug</td>
</tr>
</tbody>
</table>
9 MPa was 193 L/min. In addition to installing the natural gas high-pressure injector, a spark plug was also installed at the center of the combustion chamber as the ignition source. Natural gas was injected into cylinder at a constant pressure of 8 MPa, since the gas velocity from the injector nozzle is kept at a constant value of the sonic velocity due to the condition of choke flow during the fuel injection, thus the amount of injected fuel will keep a constant value determined by the injection duration in this study. Horiba 7100 exhaust analyzer was used to measure exhaust HC, CO, CO\textsubscript{2} and NO\textsubscript{x} concentration, the analyzer has the measuring accuracy of 1 ppm for HC, 0.01% for CO, 0.01% for CO\textsubscript{2} and 1 ppm for NO\textsubscript{x}. In the experiments, the exhaust gases were measured when engine operating parameters were adjusted at the specified conditions, that is, exhaust gases were measured at steady operating conditions.

Hydrogen with purity of 99.995\% was used, and natural gas constitutions are given in Table 2. The fuel properties of natural gas and hydrogen are given in Table 3. Different fractions of natural gas–hydrogen mixtures were prepared in advance in fuel bomb and were supplied to the fuel injector. Sonic flow of the injected gases was presented due to the choke flow during injection. It is estimated that an 18\% volume fraction of hydrogen corresponds to a 2\% mass fraction of hydrogen in mixture, and thus the influence in volumetric flow rate is limited. Thus, the volumetric flow rate of natural gas–hydrogen mixtures in this study is assumed to be unchangeable, and can be regarded as a function of injection duration.

The volumetric heat value of natural gas–hydrogen mixtures will decrease with the increase of hydrogen fraction in fuel blends, and this is due to the low volumetric heat value of hydrogen–air mixture compared to that of natural gas–air mixture at the stoichiometric equivalence ratio condition. Thus, for a given fuel injection duration, the amount of heat release will be decrease with the increase of hydrogen fraction in fuel blends, and in order to maintain the same equivalence ratio, more fuel must be injected for natural gas–hydrogen mixture combustion. Eight percent reduction in volumetric heating value is estimated when hydrogen in volume is 10\% in natural gas–hydrogen blend.

Table 2

<table>
<thead>
<tr>
<th>Items</th>
<th>Volume fraction</th>
<th>Items</th>
<th>Volume fraction</th>
</tr>
</thead>
<tbody>
<tr>
<td>CH\textsubscript{4}</td>
<td>96.16</td>
<td>(\text{C}_2\text{H}_6)</td>
<td>1.096</td>
</tr>
<tr>
<td>(\text{C}_2\text{H}_6)</td>
<td>0.021</td>
<td>(\text{C}<em>4\text{H}</em>{10})</td>
<td>0.023</td>
</tr>
<tr>
<td>(\text{C}<em>3\text{H}</em>{12})</td>
<td>0.005</td>
<td>N\textsubscript{2}</td>
<td>0.001</td>
</tr>
<tr>
<td>H\textsubscript{2}S</td>
<td>0.0002</td>
<td>H\textsubscript{2}O</td>
<td>0.006</td>
</tr>
</tbody>
</table>

Volumetric higher heating value: 36.588 MJ/m\textsuperscript{3}(normal temperature and pressure)

Volumetric lower heating value: 32.970 MJ/m\textsuperscript{3}(normal temperature and pressure)

Table 3

<table>
<thead>
<tr>
<th>Fuel properties</th>
<th>Natural gas</th>
<th>Hydrogen</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density in 1 atm, 300 K (kg/m\textsuperscript{3})</td>
<td>0.754</td>
<td>0.082</td>
</tr>
<tr>
<td>Stoichiometric air-to-fuel ratio (% by volume)</td>
<td>9.396</td>
<td>2.387</td>
</tr>
<tr>
<td>Stoichiometric air-to-fuel ratio (% by weight)</td>
<td>0.062</td>
<td>0.029</td>
</tr>
<tr>
<td>Laminar flame speed (m/s)</td>
<td>0.38</td>
<td>2.9</td>
</tr>
<tr>
<td>Quenching distance (mm)</td>
<td>1.9</td>
<td>0.6</td>
</tr>
<tr>
<td>Mass lower heating value (MJ/kg)</td>
<td>43.726</td>
<td>119.7</td>
</tr>
<tr>
<td>Volumetric heating value (MJ/m\textsuperscript{3})</td>
<td>32.97</td>
<td>10.22</td>
</tr>
<tr>
<td>Octane number</td>
<td>120</td>
<td></td>
</tr>
<tr>
<td>C/H ratio</td>
<td>0.2514</td>
<td>0</td>
</tr>
</tbody>
</table>

Fig. 1. Intervals between ending of injection and ignition at various ignition timings.
of hydrogen into natural gas decreases the stoichiometric equivalence ratio of natural gas–hydrogen fuel blends. Excessive air ratio decreases with the increase of hydrogen fraction in natural gas, indicating the decreasing of excessive air ratio at fixed injection duration in the case of natural gas–hydrogen direct-injection combustion.

3. Instrumentation and method of calculation

The cylinder pressure was recorded by a piezoelectric transducer with the resolution of 10 Pa, and the dynamic top-dead-centre (TDC) was determined by motoring. Crank angle signal was obtained from angle generating device mounted on the main shaft. The signal of cylinder crank angle was obtained from angle generating device mounted on the main shaft. The signal of cylinder crank angle was obtained from angle generating device mounted on the main shaft. The signal of cylinder crank angle was obtained from angle generating device mounted on the main shaft.

Thermodynamic model is used to calculate the thermodynamic parameters. In this paper, the model neglects the leakage through the piston rings [14], thus the energy conservation in cylinder is written as follows:

$$\frac{dQ_B}{d\phi} + \frac{dQ_W}{d\phi} = \frac{d(nu)}{d\phi} + p \frac{dV}{d\phi} = mC_V \frac{dT}{d\phi} + p \frac{dV}{d\phi}$$

(1)

Gas state equation is

$$pV = mRT$$

(2)

The variation of gas state equation with crank angle is given by

$$p \frac{dV}{d\phi} + V \frac{dp}{d\phi} = mR \frac{dT}{d\phi}$$

(3)

Heat release rate $\frac{dQ}{d\phi}$ can be derived from formula (1) and (3) as follows:

$$\frac{dQ_B}{d\phi} = \frac{C_p}{C_v} \frac{dV}{d\phi} + \frac{C_v V}{R} \frac{dp}{d\phi} + \frac{dQ_W}{d\phi}$$

(4)

where heat transfer rate is given by

$$\frac{dQ_W}{d\phi} = h_c \cdot A \cdot (T - T_w)$$

(5)

Heat transfer coefficient $h_c$ is given in Ref. [16].

$C_p$ and $C_v$ are temperature-dependent parameters, their formulae are given in Ref. [16].

The flame development duration is the angle interval from ignition start to the angle that 10% of accumulated heat release is reached; the rapid combustion duration is the angle interval from 10% of accumulated heat release to the angle of 90% of accumulated heat release; the total combustion duration is the angle interval from the beginning of heat release to the ending of heat release. The crank angle of the centre of heat release curve is determined by the following formula:

$$\varphi_c = \frac{\int_{\varphi_s}^{\varphi_e} \frac{dQ_h}{d\varphi} \cdot \phi \cdot d\phi}{\int_{\varphi_s}^{\varphi_e} \frac{dQ_h}{d\varphi} d\varphi}$$

(6)

In which $\varphi_s$ is the crank angle at the beginning of heat release and $\varphi_e$ is the crank angle at the end of heat release.

4. Results and discussion

Fig. 3 gives the brake effective mean pressure (bmeP) versus ignition timings. Bmep increases with advancing ignition timings, as shown in Fig. 3, the time intervals between the end of fuel injection and ignition timing decreases with advancing ignition timing, and this can ensure a high turbulence at the timing of ignition and increases the burning velocity of mixture. Meanwhile, shortening the time intervals between the end of fuel injection and ignition timing can form a better stratified mixture, promoting mixture combustion and increasing brake mean effective pressure. Since the excessive air ratios in this study are set at large values (larger than 1.6), and engine is operated on lean mixture combustion. Thus, mixture stratification is favorable to improve the burning velocity at lean mixture combustion. The results also show that addition of hydrogen into natural gas at lean mixture combustion increases bmep compared with that of natural gas combustion. This is due to the increase of burning velocity by hydrogen addition. Similar trend of bmep to the variation of ignition timings is presented for both natural gas combustion and natural gas–hydrogen combustion, and
this reveals the same sensitivity of bmep to the variation of ignition timings for natural gas direct-injection combustion and natural gas–hydrogen direct-injection combustion. For specific ignition timing, brake mean effective pressure increases with the increase of hydrogen fraction in natural gas. The result also reveals that the variation of bmep give the similar increasing trend with shortening the time interval from the ending of fuel injection to the ignition start regardless of hydrogen fractions, and a linear correlation between bmep and the time interval from the ending of fuel injection to the ignition start.

Fig. 4 illustrates the effective thermal efficiency versus ignition timings. For specific fuel, the effective thermal efficiency increases with advancing ignition timings at lean mixture condition of this study, and this is reasonable since bmep is increased while the amount of heat per cycle maintains a constant value. Addition of hydrogen into natural gas increases the burning velocity of mixture, shortening the combustion duration, increasing the cylinder gas temperature, and this increases the effective thermal efficiency compared to that of natural gas direct-injection combustion under the same lean mixture condition, and this can also be explained by the improvement of lean mixture combustion by adding hydrogen. For specific ignition timing, effective thermal efficiency increases with the increase of hydrogen fraction in natural gas. It can be derived from the results that the variations of effective thermal efficiency give the same increasing trend with shortening the time interval from the ending of fuel injection to the ignition start regardless of hydrogen fractions.

Fig. 5 shows the combustion durations versus the ignition timings. Flame development duration decreases with advancing the ignition timings. Advancing ignition timing will shorten the time duration between the end of fuel injection and ignition timing, and this will form the high mixture stratification in the combustion chamber and relatively rich mixture near the spark plug, making the mixture more easily to be ignited, thus reduces the ignition delay and the flame development duration. For specific ignition timing, the flame development duration decreases with the increase of hydrogen fraction, indicating that addition of hydrogen can promote flame kernel formation and flame propagation at early stage of mixture combustion. Rapid combustion duration and total combustion duration also give the decrease with advancing ignition timings, the improvement of flame propagation speed and the increase in gas temperature of the stratified mixture make quickly the combustion, resulting in the decrease in rapid and total combustion duration. The results also reveal that the flame development duration, the rapid combustion duration and the total combustion duration give a similar increasing trend with the increase of the time interval from the ending of fuel injection to the ignition start regardless of hydrogen fractions.

For lean mixture combustion, the decrease in time intervals between the end of fuel injection and ignition timing makes high mixture stratification and increases the burning velocity. Meanwhile, shortening the time intervals between
the end of fuel injection and ignition timing and can maintain a high turbulence generated by high-speed gas jet, and this will also increase the mixture burning velocity. For specific ignition timing, the rapid combustion duration and total combustion duration decrease with the increase of hydrogen fraction, and this is resulted from burning velocity enhancement by hydrogen addition.

The position of the center of heat release curve can in one aspect reflect the compactness of heat release process. As shown in Fig. 6, the crank angle of the center of heat release curve ($\varphi_c$) moves close to the top-dead-center with advancing the ignition timing, and this reveals the high compactness in heat release process. This is reasonable since combustion durations decrease with advancing the ignition timings, and fast heat release process will occur at early ignition timing. For specific ignition timing, $\varphi_c$ moves close to the top-dead-center with the increase of hydrogen fraction, the increasing of burning velocity of mixture by hydrogen addition will make fast heat release process. The figure also reveals that $\varphi_c$ gives a linear increasing trend with the increase of the time interval from the ending of fuel injection to the ignition start regardless of hydrogen fractions.

Fig. 7 gives the exhaust gases concentrations versus the ignition timings. The exhaust HC concentration decreases linearly and NO$_x$ concentration increases exponentially with advancing the ignition timings. Decreasing the time intervals between the ending of fuel injection and ignition timing will create high mixture stratification and make a rapid combustion, increasing the combustion temperature and NO$_x$ concentration. Postponing ignition timing will weaken the mixture stratification and increase the fraction of lean mixture in the combustion chamber, and thus will increase the fraction of unburned fuel at lean mixture region and decrease the combustion temperature. The decreasing in combustion temperature will reduce HC post-flame oxidation during the expansion stroke. All these make the increase of HC concentration with postponing the ignition timings. Both HC and NO$_x$ concentration decrease with the increase of hydrogen fractions, the decrease in C/H ratio and increase in combustion temperature with the increase of hydrogen fraction are responsible for HC reduction while the dilution of mixtures (increase in excessive air ratio) by hydrogen addition makes the reduction of NO$_x$ concentration. The result also reveals that HC concentration decrease linearly with shortening the time interval from the ending of fuel injection to the ignition start regardless of hydrogen fractions.

5. Conclusions

Combustion characteristics of a direct-injection spark-ignited engine fueled with natural gas–hydrogen blends under various ignition timings and lean mixture condition.

Fig. 6. Crank angle of center of heat release curve versus ignition timings.

Fig. 7. Exhaust gases concentrations of fuel blends versus ignition timings.
were investigated, and the main results are summarized as follows:

1. Ignition timing has significant influence on engine performance, combustion and emissions. The time intervals between the ending of fuel injection and ignition timing are very sensitive to direct-injection gas engine combustion.

2. Brake mean effective pressure and effective thermal efficiency increase with decreasing the time intervals from the ending of fuel injection to the ignition start while combustion durations decrease with decreasing the time intervals from the ending of fuel injection and ignition start.

3. For specific ignition timing, the brake mean effective pressure and the effective thermal efficiency increase while the combustion durations decrease with the increase of hydrogen fraction in natural gas.

4. Exhaust HC concentration decreases and exhaust NO\textsubscript{x} concentration increases with advancing the ignition timing while the exhaust CO gives little variation under various ignition timings.

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