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Combustion behaviors of a direct-injection engine operating on various fractions of natural gas–hydrogen blends

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Abstract

Combustion behaviors of a direct injection engine operating on various fractions of natural gas–hydrogen blends were investigated. The results showed that the brake effective thermal efficiency increased with the increase of hydrogen fraction at low and medium engine loads and high thermal efficiency was maintained at the high engine load. The phase of the heat release curve advanced with the increase of hydrogen fraction in the blends. The rapid combustion duration decreased and the heat release rate increased with the increase of hydrogen fraction in the blends. This phenomenon was more obviously at the low engine speed, suggesting that the effect of hydrogen addition on the enhancement of burning velocity plays more important role at relatively low cylinder air motion. The maximum mean gas temperature and the maximum rate of pressure rise increased remarkably when the hydrogen volumetric fraction exceeds 20% as the burning velocity increases exponentially with the increase of hydrogen fraction in fuel blends. Exhaust HC and CO_2 concentrations decreased with the increase of the hydrogen fraction in fuel blends. Exhaust NO_x concentration increased with the increase of hydrogen fraction at high engine load. The study suggested that the optimum hydrogen volumetric fraction in natural gas–hydrogen blends is around 20% to get the compromise in both engine performance and emissions.

Keywords: Natural gas; Hydrogen; Combustion; Emissions; Direct-injection engine

1. Introduction

Internal combustion engines are widely used as the transportation and stationary power sources due to its high thermal efficiency and power density. Up to now, most internal combustion engines are operated on the fossil fuels and bring serious pollution problems. The shortage of fossil fuel supplying enhances the investigation of alternative fuels in engines and seeks the way to solve the fuel substitution and emission reduction.

Natural gas is regarded as one of the most promising alternative fuels and probably one of the cleanest fuels in combustion. The use of natural gas as engine fuel has been studied for many years. Nowadays, natural gas engine has entered into the stage of commercial engine. Natural gas is a mixture of different gases where methane is its major component. The combustion of natural gas produces less emission than that of gasoline and diesel fuels due to its simple chemical structure and absence of fuel evaporation. The high octane number of natural gas gives the engine high anti-knocking capability and allows it to operate at even high compression ratio, leading to the further improvement of both power output and thermal efficiency. But natural gas has the penalty of slow burning velocity and poor lean burn ability and this may lead to the incomplete combustion, high misfire ratio and large cycle-by-cycle variation at lean mixture combustion. One effective way to solve the problem is to mix the natural gas with a fuel that possesses the high burning velocity. Hydrogen is an excellent additive into natural gas in improving the burning velocity of mixture due to the high burning velocity of hydrogen. The laminar burning velocity of hydrogen is seven times to that of natural gas. Adding a small amount of hydrogen into natural gas can improve the combustion characteristics and reduce exhaust emissions [1].

Many studies have been carried out on using the natural gas–hydrogen blends in the port-injection spark ignited engines. Blarigan and Keller investigated the port-injection engine

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Nomenclature

A	wall area, m^2	p	cylinder gas pressure
BTDC	before top-dead-center	p_{max}	maximum cylinder gas pressure
C_p	constant pressure specific heat	P_{me}	power output
C_v	constant volume specific heat	R	gas constant
$(\mathrm{d}p.\mathrm{d}\varphi)_{max}$	maximum rate of pressure rise with crank angle	T	mean gas temperature
$\mathrm{d}Q_B/\mathrm{d}\varphi$	heat release rate with crank angle	$T_{max}\,x$	maximum mean gas temperature
$(\mathrm{d}Q_B/\mathrm{d}\varphi)_{max}$	maximum rate of heat release with crank angle	T_w	wall temperature
$\mathrm{d}Q_w/\mathrm{d}\varphi$	heat transfer rate to wall with crank angle	TDC	top-dead-center
h_c	heat transfer coefficient	V	cylinder volume
H_u	lower heating value	WOT	whole opening throttle
m	mass of cylinder gases	φ	crank angle
n_e	engine speed	φ_{ign}	ignition advance angle
		φ_{inj}	injection advance angle
		θ_{td}	fuel injection duration
		λ	excess air ratio

fueled with natural gas–hydrogen mixtures [2]; Wong and Karim studied engine performance fueled by various hydrogen fractions in natural gas–hydrogen blends [3]; and Bauer and Forest conducted an experimental study on natural gas–hydrogen combustion in a CFR engine [4]. Furthermore, studies on lean combustion capability of natural gas–hydrogen combustion and natural gas–hydrogen combustion with turbocharging and/or exhaust gas recirculation were also conducted [5,6], and the studies showed that the exhaust HC, CO, and CO_2 concentrations could be decreased when engine operated on the natural gas–hydrogen blends compared with those of natural gas. However, NO_x may increase for natural gas–hydrogen combustion at rich mixture condition due to the increase of flame propagation speed and combustion temperature. However, NO_x concentration can be decreased through lean mixture combustion and retarding of ignition timing. The previous work mainly concentrated on the homogeneous-charge port injection engine, and few articles were found for the direct-injection engine operating on the natural gas–hydrogen blends.

The unfavorable point of port-injection gas engine is its decreasing in volumetric efficiency as gas fuel will occupy certain portion of intake charge, leading to the decrease of power output. Direct-injection gas engine can avoid the problem of volumetric efficiency decreasing and maintain high volumetric power output. Meanwhile, the direct injection approach can realize the stratified charge combustion and extend the lean mixture combustion capability, leading to the increase in thermal efficiency and the decrease in exhaust emissions. Preliminary studies have been taken by the authors on a direct-injection spark ignited engine fueled with natural gas–hydrogen blends at low compression ratio and hydrogen volumetric fraction less than 20% [7–9]. The results showed that the heat release rate decreased and the combustion duration increased when the hydrogen fraction was less than 10%, while the heat release rate increased and the combustion duration decreased when the hydrogen volumetric fraction was over 10%. This paper investigates the combustion and emissions of a direct-injection engine fueled with natural gas–hydrogen blends at the extended hydrogen fraction and the increased compression ratio. The study is expected to provide a comprehensive evaluation to the direct injection engine fueled with natural gas–hydrogen blends.

2. Experimental setup and procedures

A single cylinder diesel engine was modified into a direct-injection spark ignited natural gas engine. The specifications of the engine are listed in Table 1.

The compositions of natural gas used in this experiment are given in Table 2. Hydrogen with purity of 99.995% was used in this experiment. The fuel properties of natural gas and hydrogen are given in Table 3. Since the laminar burning velocity of hydrogen is seven times to that of natural gas, the addition of hydrogen into the natural gas can increase the burning velocity of the mixtures. It is also noted that the quench distance of hydrogen is much smaller than that of natural gas, the heat loss to chamber wall will be increased by addition of hydrogen into the natural gas [10].

The blends with various fractions of hydrogen into natural gas–hydrogen were prepared in the experiment. Fuel was injected into the cylinder at constant pressure of 8.0 MPa, since the gas velocity from the injector nozzle is kept at the constant value of sonic velocity due to the condition of choke flow during the fuel injection, thus the amount of injected fuel is determined by the injection duration in this study.

Table 1
Engine specifications

Bore × Stroke (mm × mm)	100 × 115
Displacement (L)	0.903
Compression ratio	12
Combustion chamber	Bowl-in-shape
Ignited type	Spark ignited
Injection pressure	8.0 MPa

Table 2
Compositions and properties of natural gas

Items	Volume fraction (%)	Items	Volume fraction (%)	Items	Volume fraction (%)
CH_4	96.16	C_2H_6	1.096	C_3H_8	0.136
iC_4H_{10}	0.021	nC_4H_{10}	0.021	iC_5H_{12}	0.006
nC_5H_{12}	0.005	N_2	0.001	CO_2	2.54
H_2S	0.0002	H_2O	0.006		

Volumetric higher heating value: 36.588 MJ/m^3 (normal temperature and pressure)
Volumetric lower heating value: 32.970 MJ/m^3 (normal temperature and pressure)

Table 3
Fuel properties of natural gas and hydrogen

Fuel properties	Natural gas	Hydrogen
Density at NTP (kg/m^3)	0.754	0.082
Stoichiometric air–fuel ratio (% by volume)	9.396	2.387
Laminar burning velocity (m/s)	0.38	2.9
Quenching distance (mm)	1.9	0.6
Volumetric lower heating value (MJ/m^3)	32.97	10.22
Octane number	120	
C/H ratio	0.2514	0

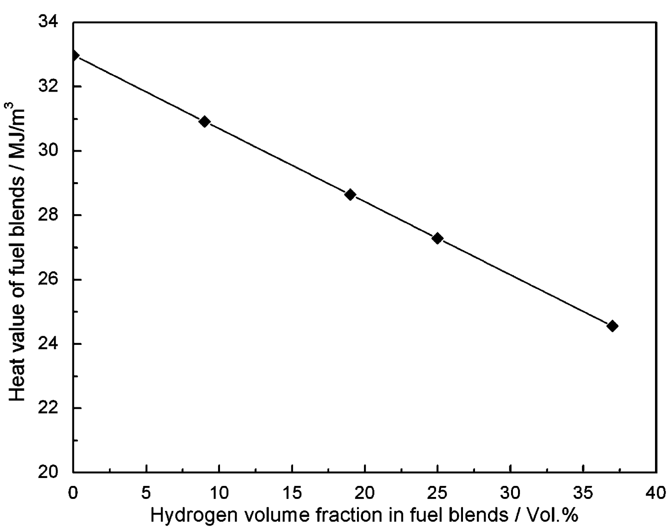

Fig. 1. Lower heating value of the fuel blends versus hydrogen fractions.

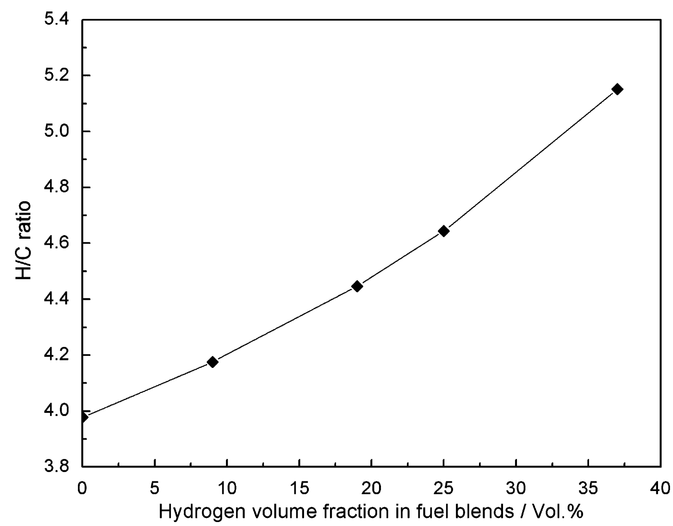

Fig. 2. H/C ratio of the fuel blends versus hydrogen fractions.

Fig. 1 gives the volumetric lower heating values (H_u) of natural gas–hydrogen mixtures at various hydrogen fractions. The volumetric heating value of the mixtures decreases with the increase of hydrogen fraction in the fuel blends since hydrogen has lower volumetric heating value than that of natural gas. The volumetric heating value decreases by 24% when the hydrogen volumetric fraction in fuel blends reaches up to 37%. Thus fuel injection duration needs to be increased to maintain the same amount of heat release in the case of natural gas–hydrogen blends.

Fig. 2 gives the ratio of hydrogen to carbon (H/C) of the natural gas–hydrogen blends at various hydrogen fractions. The H/C ratio increases linearly with the increase of hydrogen fraction in the fuel blends, and this will be beneficial to the reduction of carbon-related emissions such as CO, HC and CO_2.

Experimental study of natural gas–hydrogen fuel blends with hydrogen volumetric fractions of 0%, 9%, 19%, 25% and 37% was investigated under the same engine speed and load by regulating the fuel injection duration (θ_{td}), while the fuel injection advance angle (φ_{inj}) and ignition timing (φ_{ign}) were fixed for each specified hydrogen fraction. The operating parameters are listed in Table 4. High-pressure natural gas injector was used and its flow rate is low for gas fuels, thus the opening of the throttle valve was fixed at 70% WOT (whole opening throttle) to maintain combustion stability in-cylinder. The temperature of cooling water was kept at 70 °C in the experiment.

Cylinder pressure was recorded by a piezoelectric transducer, and the dynamic top-dead-center (TDC) was calibrated in motoring. The crank angle signal was obtained from an angle signal generating device mounted on the main shaft. The signal of cylinder pressure was acquired for every 0.1 crank angle, and the acquisition process covered 100 completed cycles, the average value of these 100 cycles was outputted as the pressure data used for the calculation of combustion parameters. Horiba Lamda sensor was used to detect excess air ratio.

Horiba 7100 exhaust analyzer was used to measure exhaust HC, CO, CO_2 and NO_x concentration, the analyzer has the

Table 4
Parameters of operating modes

n_e (r/min)	P_{me} (MPa)	φ_{inj} CA BTDC	θ_{td} (ms) 0%H_2	θ_{td} (ms) 9%H_2	θ_{td} (ms) 19%H_2	θ_{td} (ms) 25%H_2	θ_{td} (ms) 37%H_2	φ_{ign} CA BTDC
1200	0.14	170	15.36	15.46	15.26	14.46	14.06	28
1200	0.42	180	16.36	16.86	16.06	15.96	16.06	31
1200	0.63	190	18.06	18.66	19.46	19.56	19.86	34
1800	0.14	220	15.66	13.26	14.46	13.96	14.36	37
1800	0.42	224	17.46	16.96	16.96	16.66	17.26	38
1800	0.63	240	18.66	19.06	19.76	20.06	20.56	42

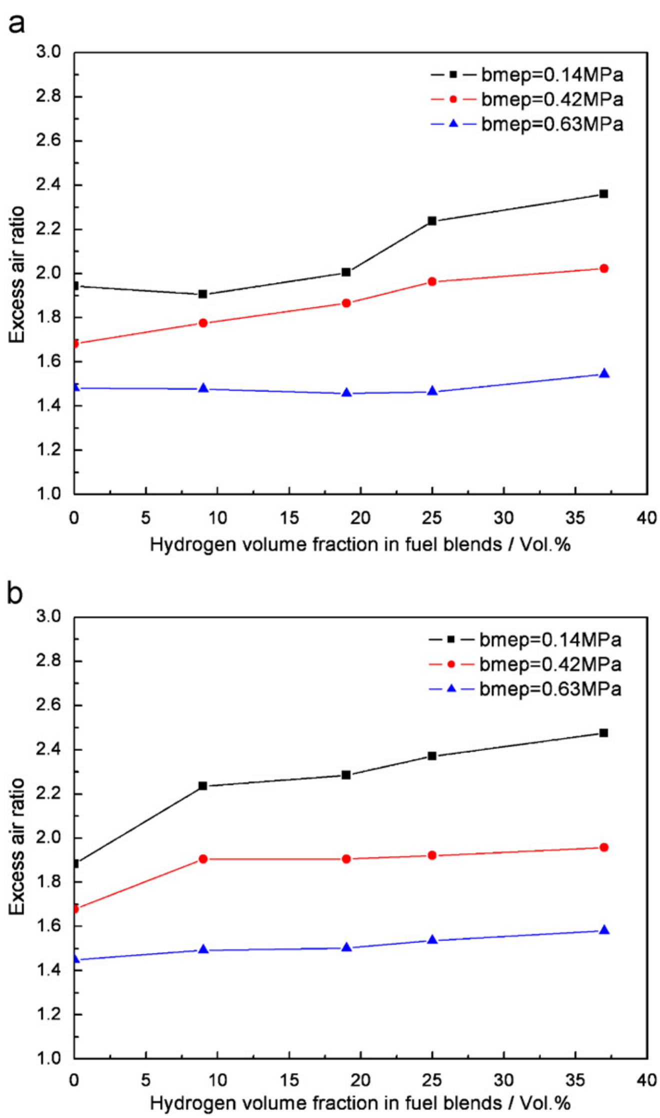

Fig. 3. Overall excess air ratio of the mixtures versus hydrogen fractions. (a) n = 1200r/ min; (b) n = 1800r/ min.

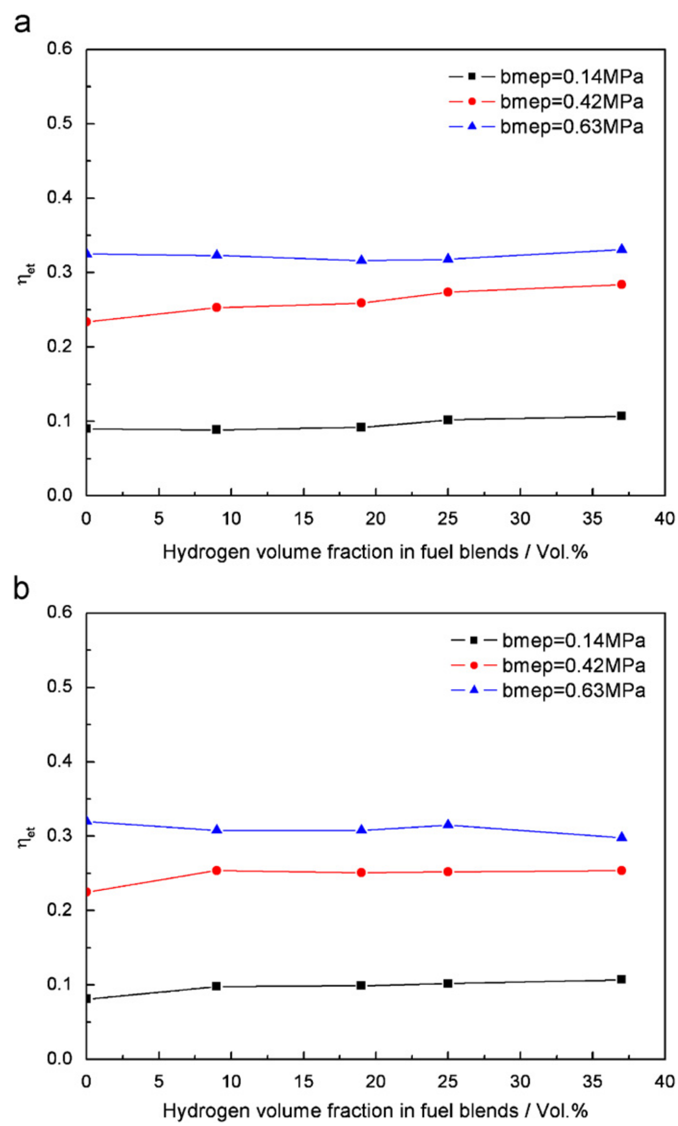

Fig. 4. Brake effective thermal efficiency of fuel blends versus hydrogen fractions. ures versus hydrogen fractions. (a) n = 1200r/ min; (b) n = 1800r/ min.

measuring accuracy of 1 ppm for HC and NO_x, 0.01% for CO and CO_2. In the experiments, the exhaust gases were measured when engine operation parameters were adjusted at the specified conditions, that is, exhaust gases were measured at steady operating conditions.

3. Results and discussions

A zero-dimensional thermodynamic model was used to calculate the heat release rate in the study [11]. The model neglects the leakage through the piston rings, and thus the energy

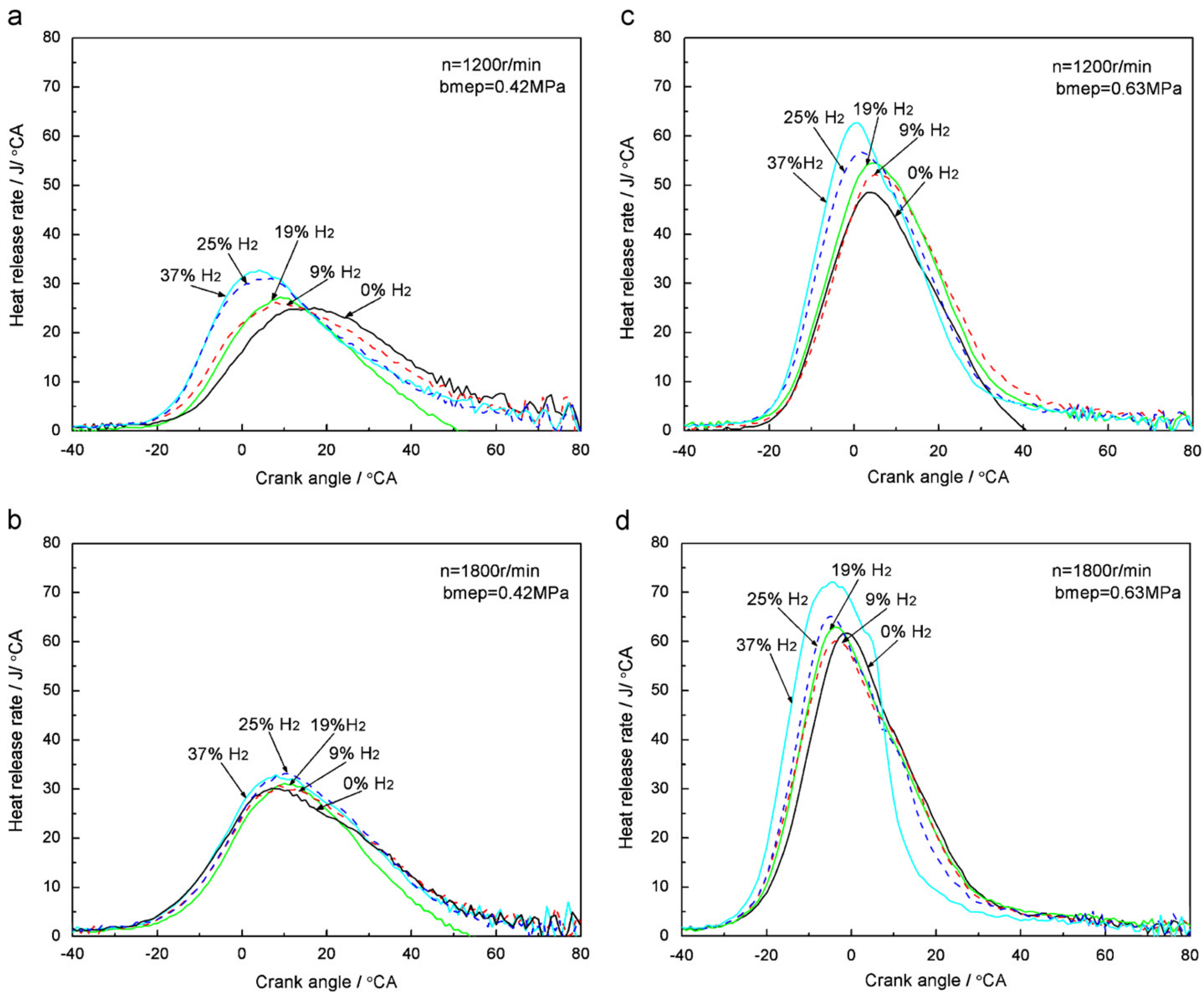

Fig. 5. Heat release rate of the fuel blends.

conservation in the cylinder is written as follows:

$$\frac{\mathrm{d}Q_B}{\mathrm{d}\varphi}=\frac{\mathrm{d}Q_{\mathrm{w}}}{\mathrm{d}\varphi}+\frac{\mathrm{d}(mu)}{\mathrm{d}\varphi}+p\frac{\mathrm{d}V}{\mathrm{d}\varphi}=\frac{\mathrm{d}Q_{\mathrm{w}}}{\mathrm{d}\varphi}+mC_v\frac{\mathrm{d}T}{\mathrm{d}\varphi}+p\frac{\mathrm{d}V}{\mathrm{d}\varphi}. \tag{1}$$

The gas-state equation is

$$pV=mRT. \tag{2}$$

The differential of the gas-state equation with crank angle φ is given by

$$p\frac{\mathrm{d}V}{\mathrm{d}\varphi}+V\frac{\mathrm{d}p}{\mathrm{d}\varphi}=mR\frac{\mathrm{d}T}{\mathrm{d}\varphi}. \tag{3}$$

The heat-release rate $\mathrm{d}Q_B/\mathrm{d}\varphi$ can be derived from Eqs. (1) and (3) as follows:

$$\frac{\mathrm{d}Q_B}{\mathrm{d}\varphi}=p\cdot\frac{C_{\mathrm{p}}}{R}\frac{\mathrm{d}V}{\mathrm{d}\varphi}+\frac{C_{\mathrm{v}}V}{R}\frac{\mathrm{d}p}{\mathrm{d}\varphi}+mT\frac{\mathrm{d}C_{\mathrm{v}}}{\mathrm{d}\varphi}+\frac{\mathrm{d}Q_{\mathrm{w}}}{\mathrm{d}\varphi}, \tag{4}$$

where the heat-transfer rate is given by

$$\mathrm{d}Q_{\mathrm{w}}/\mathrm{d}\varphi=h_{\mathrm{c}}A(T-T_{\mathrm{w}}) \tag{5}$$

The heat-transfer coefficient h_{c} uses the correlation formula given by Woschni from Heywood book [11].

C_{p} and C_{v} are temperature-dependent parameters; their formulas are given in Ref. 11.

Fig. 3 shows the overall excess air ratio of the mixtures versus hydrogen fractions. The overall excess air ratio keeps almost a constant value at high engine load (or bmep) while it increases with the increase of hydrogen fraction at small and medium engine loads. As the opening of throttle valve was fixed

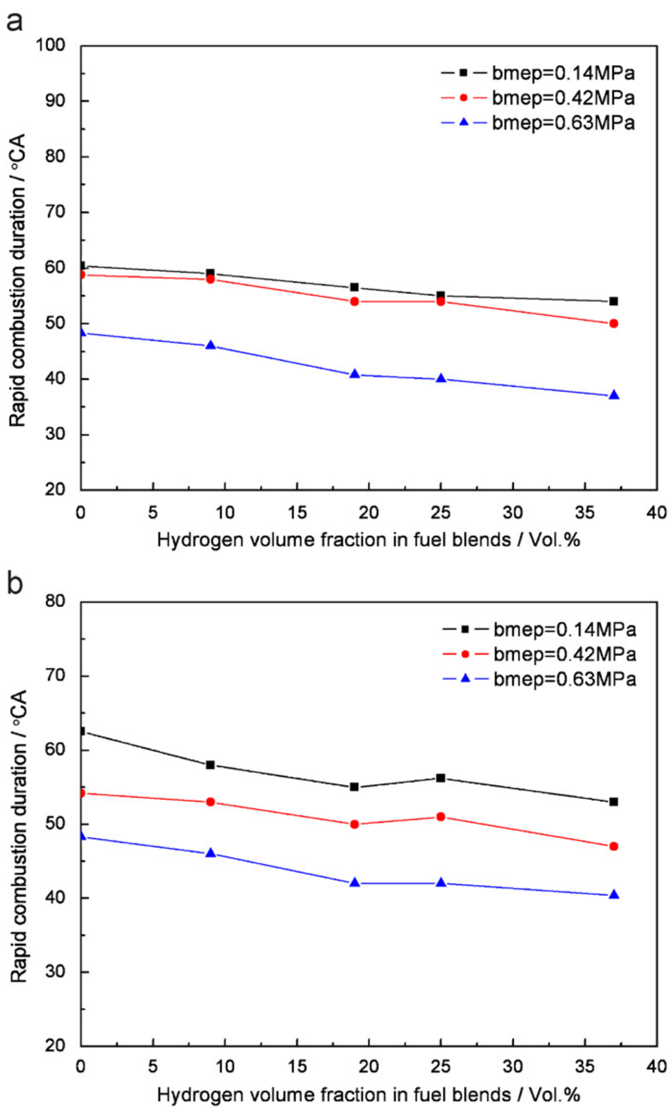

Fig. 6. Rapid combustion duration of fuel blends versus hydrogen fractions. ures versus hydrogen fractions. (a) n = 1200r/ min; (b) n = 1800r/ min.

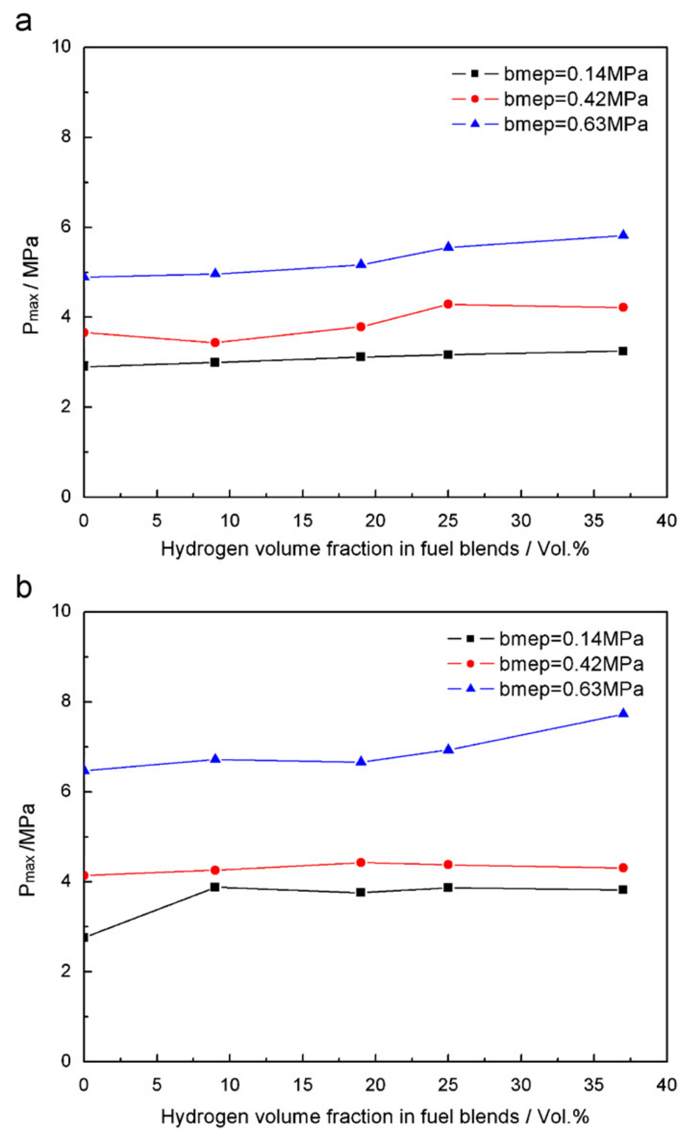

Fig. 7. Maximum cylinder gas pressure of the fuel blends versus hydrogen fractions. ures versus hydrogen fractions. (a) n = 1200r/ min; (b) n = 1800r/ min.

in the experiment, so the fresh air entering into the cylinder will be the same at the same engine speed. The stoichiometric air–fuel ratio of hydrogen is one-fourth of the natural gas, so more volumetric fuel should be injected at high engine load in the case of natural gas–hydrogen blends comparing with that of natural gas as shown in Table 4, and this leads to the little variation of overall excess air ratio at high engine load. The effectiveness of hydrogen addition on the enhancement of burning velocity is more obvious at lean mixture combustion [12], and this to some extent compensates the decrease of lower heat value by hydrogen addition, leading to the increase of excess air ratio with the increase of hydrogen fraction at small and medium engine loads. The overall excess air ratio of the fuel blends versus hydrogen fraction shows a similar phenomenon at high and low engine speeds.

Fig. 4 gives the brake effective thermal efficiency of the fuel blends versus hydrogen fraction. The natural gas–hydrogen blends maintained high thermal efficiency at high engine load and the effective thermal efficiency is almost constant or slightly increases with the increase of hydrogen fraction at the small and medium engine loads; Two factors due to hydrogen addition influence the variation of thermal efficiency; on one hand, the increase of combustion velocity by hydrogen addition shortens the combustion duration and increases the thermal efficiency, and on the other hand, the heat loss to the chamber wall will be increased by hydrogen addition due to the decreased quench distance and increased combustion temperature, and this will decrease the thermal efficiency. The burning velocity at high engine load (or rich mixture) gives high value, and the effectiveness of enhancement on burning velocity by hydrogen addition is relatively low. The increasing in heat loss to the combustion wall increases with the increase of hydrogen fraction. The combined influence

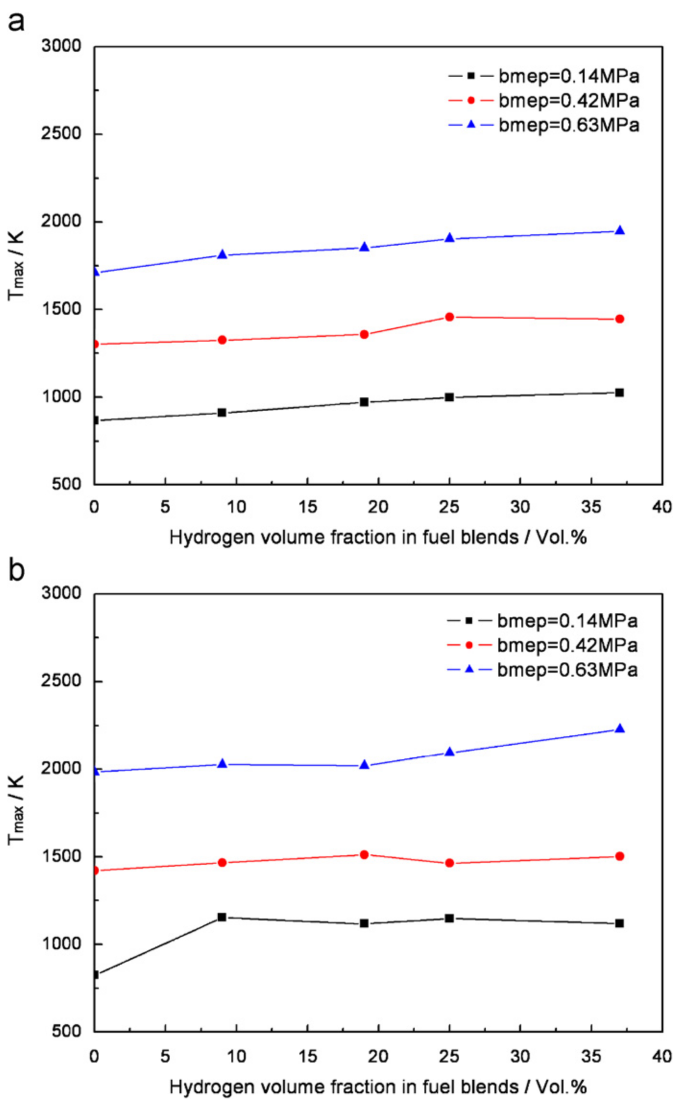

Fig. 8. Maximum mean gas temperature of fuel blends versus hydrogen fractions. ures versus hydrogen fractions. (a) n = 1200r/ min; (b) n = 1800r/ min.

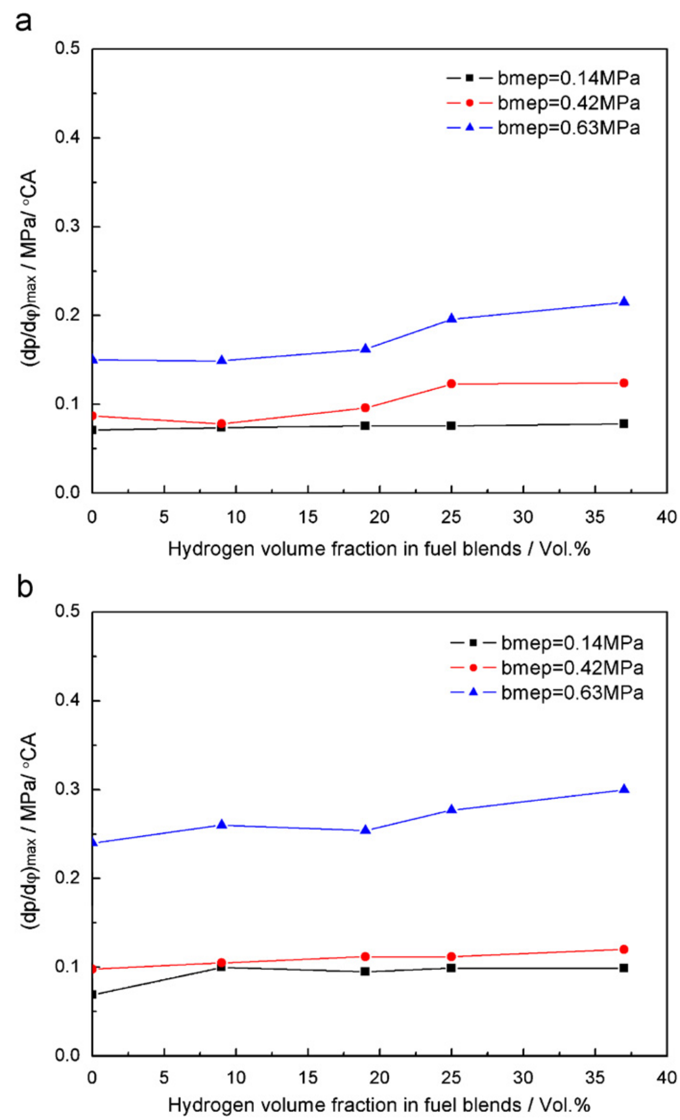

Fig. 9. Maximum rate of pressure rise of fuel blends versus hydrogen fractions. ures versus hydrogen fractions. (a) n = 1200r/ min; (b) n = 1800r/ min.

makes little variation of the thermal efficiency with hydrogen fractions. In contrast to this, the enhancement of burning velocity at lean mixture (small and medium engine loads) by hydrogen addition gives remarkable increase comparing with that of natural gas, resulting in the decrease of combustion duration with the increase of hydrogen addition and increasing in thermal efficiency. The study suggests that the improvement of burning velocity by hydrogen addition is more obvious at lean mixture combustion than that at rich mixture combustion, and this is consistent to the behavior obtained by authors in the combustion vessel study [14].

Fig. 5 gives the heat release rate of the blends versus hydrogen fractions. The beginning of the heat release is advanced with the increase of hydrogen fraction in the fuel blends. This phenomenon is more obvious at low engine speed, as the enhancement of burning velocity is more obviously by hydrogen addition, leading to the maximum heat release moving close to the top-dead center and increasing the thermal efficiency. The effect of hydrogen addition on burning velocity enhancement gives a limited value when hydrogen volumetric fraction in fuel blends is less than 20% while the burning velocity increases remarkably when hydrogen volumetric fraction in natural gas–hydrogen blends exceeds 20%. The experimental study of laminar burning velocities of natural gas–hydrogen–air mixtures in a constant volume bomb gave the consistent results that burning velocity increases remarkably when hydrogen fraction was larger than 20% [12,13].

Fig. 6 gives the rapid combustion duration versus hydrogen fractions in the fuel blends. Rapid combustion duration is defined as the angle interval from the crank angle of 10% accumulated heat release to the angle of 90% accumulated heat release. The rapid combustion duration decreases with the increase of hydrogen fraction in the fuel blends, reflecting the increase of mixture burning velocity with the increase of hydrogen fraction. Although the excess air ratio increases with the increase of hydrogen fraction at specified brake mean

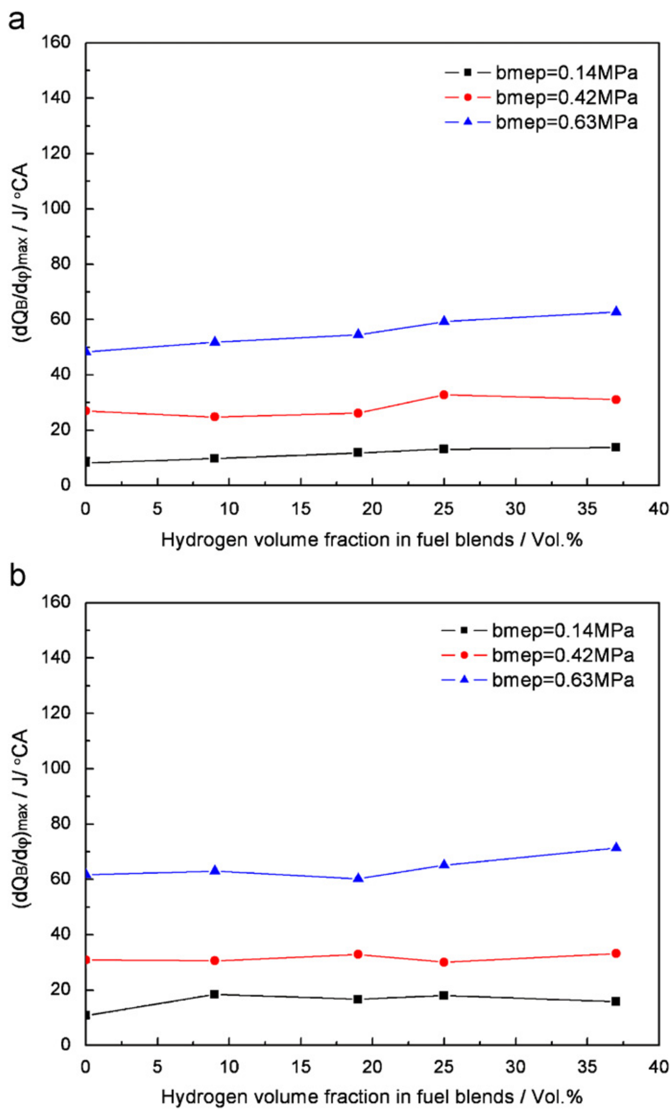

Fig. 10. Maximum heat release rate of fuel blends versus hydrogen fractions. ures versus hydrogen fractions. (a) n = 1200r/ min; (b) n = 1800r/ min.

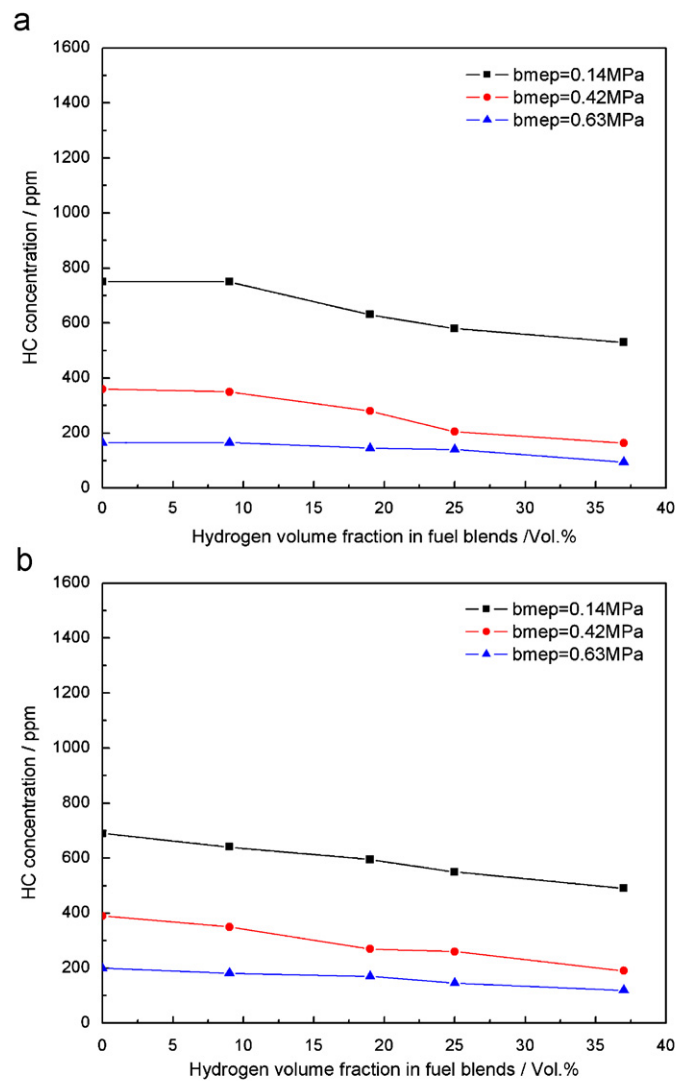

Fig. 11. Exhaust HC concentration of fuel blends versus hydrogen fractions. ures versus hydrogen fractions. (a) n = 1200r/ min; (b) n = 1800r/ min.

effective pressure, the increase in burning velocity and combustion temperature makes the fast combustion process and short combustion duration.

Figs. 7 and 8 give the maximum cylinder gas pressure $p_{\max}$ and the maximum mean gas temperature $T_{\max}$ versus hydrogen fractions at two engine speeds, respectively. $p_{\max}$ and $T_{\max}$ increase with the increase of hydrogen fraction, and fast burning velocity is responsible for this. Fig. 8 indicates a remarkable increase in $T_{\max}$ when hydrogen volumetric fraction is over 20%, especially at high engine load and this will lead to the noticeable increase of NO_x emission. From the limitation of NO_x emission, the study suggests that the addition of hydrogen in natural gas should be controlled to less than 20%.

Figs. 9 and 10 show the maximum rate of pressure rise $(\mathrm{d}p/\mathrm{d}\varphi)_{\max}$ and the maximum heat release rate $(\mathrm{d}Q_B/\mathrm{d}\varphi)_{\max}$ versus hydrogen fraction at two engine speeds, respectively. $(\mathrm{d}p/\mathrm{d}\varphi)_{\max}$ and $(\mathrm{d}Q_B/\mathrm{d}\varphi)_{\max}$ increase with the increase of hydrogen fraction. This can also be explained by the enhancement of burning velocity by hydrogen addition. The increase of $(\mathrm{d}p/\mathrm{d}\varphi)_{\max}$ may increase engine noise comparing to the natural gas operation. $(\mathrm{d}p/\mathrm{d}\varphi)_{\max}$ and $(\mathrm{d}Q_B/\mathrm{d}\varphi)_{\max}$ also increase remarkably when hydrogen volumetric fraction is larger than 20%, especially at high engine load. From the controlling of engine noise, the study suggests that the addition of hydrogen in natural gas should be controlled to less than 20%.

Fig. 11 gives the exhaust HC concentration versus hydrogen fraction at various engine speeds. The concentration of HC decreases with the increase of hydrogen fraction. The main source of the HC exhaust emission in direct-injection stratified charge engine is from the flame quenching during the flame propagation from rich mixture zone near the spark plug to lean mixture zone close to the cool wall of the cylinder. It can be estimated that the tendency of flame quenching for the blends will decrease noticeably by hydrogen addition due to the extension of lean burn ability and decrease the quench distance. Meanwhile, the cylinder gas temperature will increase with the increase of hydrogen fraction and this will promote the post-flame oxidation of the formed HC. Furthermore, the replacement of some

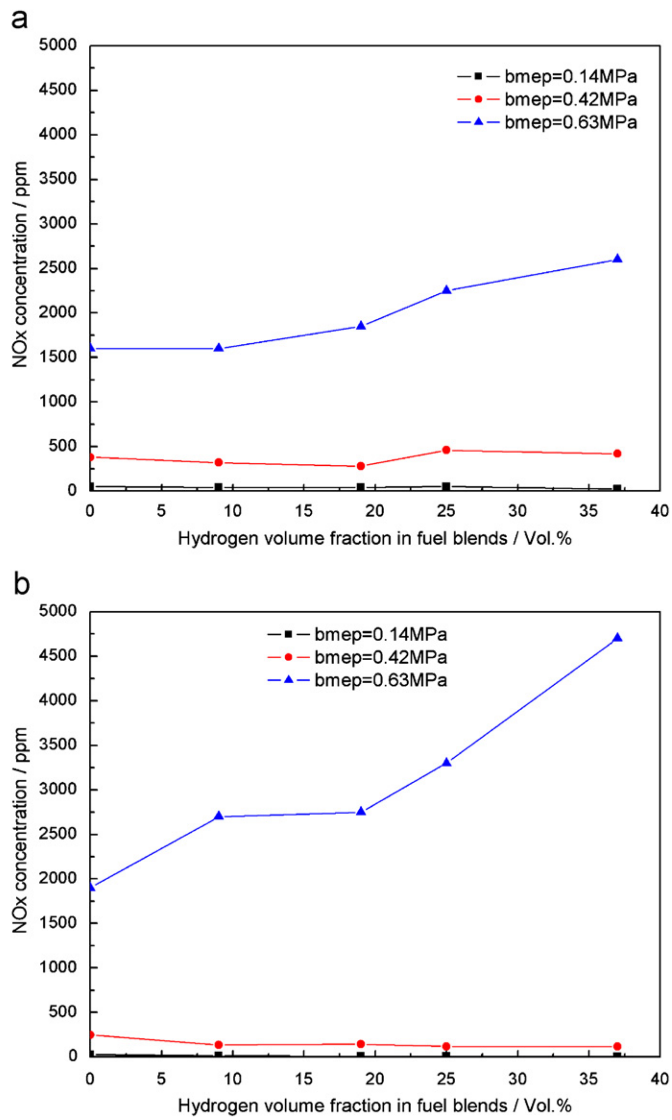

Fig. 12. Exhaust NO_x concentration of fuel blends versus hydrogen fractions. ures versus hydrogen fractions. (a) n = 1200r/ min; (b) n = 1800r/ min.

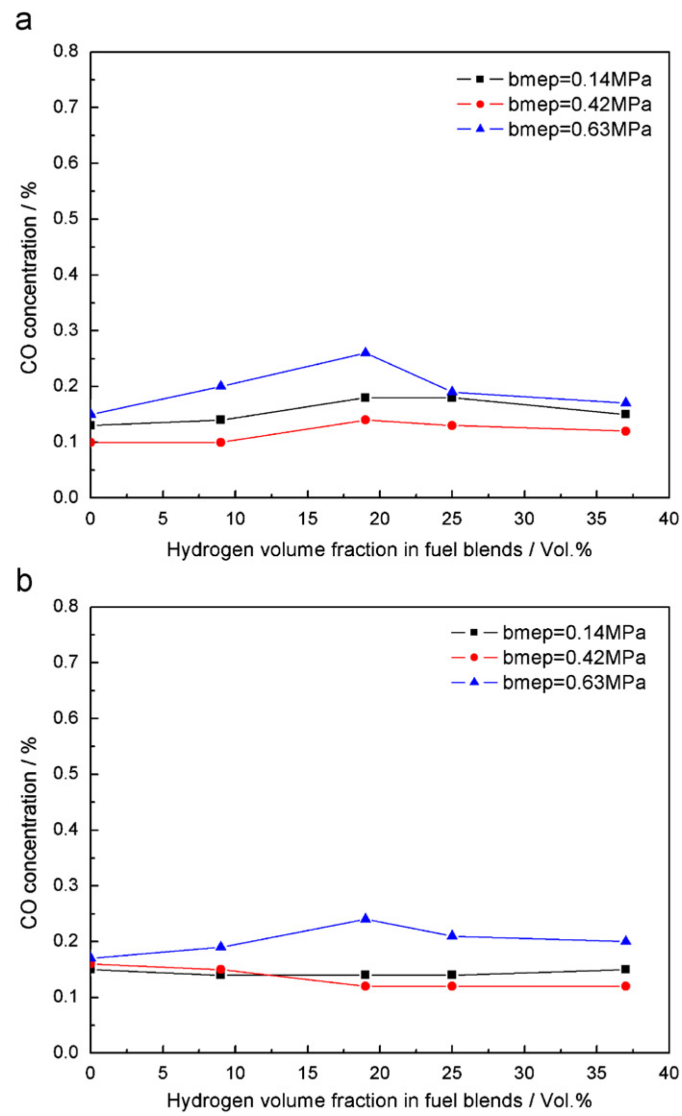

Fig. 13. Exhaust CO concentration of fuel blends versus hydrogen fractions. ures versus hydrogen fractions. (a) n = 1200r/ min; (b) n = 1800r/ min.

natural gas by hydrogen decreases the carbon fraction in the fuel blends. All these make the decrease of HC emission with the increase of hydrogen fraction in the blends.

Fig. 12 gives the exhaust NO_x concentration versus hydrogen fraction at various engine speeds. NO_x emission gives low value at low engine load at various hydrogen fractions. The low gas temperature at low load and large excess air ratio is responsible for this. In contrast to this, the increase in the concentration of NO_x emissions with the increase of hydrogen fraction at high engine load would be due to the increase of gas temperature by hydrogen addition. The concentration of NO_x emission at high engine load increases remarkably when hydrogen volumetric fraction exceeds 20%. This is consistent to the gas temperature in Fig. 8 as gas temperature increases remarkably when hydrogen volumetric fraction is larger than 20%. The experimental study of the laminar burning characteristics of natural gas–hydrogen–air mixtures in a constant volume bomb revealed that the mass burning rate increased exponentially with the increase of hydrogen fraction in the mixtures [14].From the limitation of NO_x emission, the hydrogen fraction of less than 20% is appreciated.

Fig. 13 gives the exhaust CO concentration versus hydrogen fractions in the natural gas–hydrogen blends. Exhaust CO concentration gives low value at all operating modes. As shown in Fig. 3, the overall excess air ratio in cylinder is larger than 1.5 at all operating modes, while CO is strongly related to the air–fuel ratio. The sufficiency of oxygen in the cylinder makes the low CO emission.

Fig. 14 shows the exhaust CO_2 concentration versus hydrogen fraction in the natural gas–hydrogen blends. Exhaust CO_2 concentration decrease with the increase of hydrogen fraction in the mixtures. The increase in H/C ratio of the mixtures with the increase of the hydrogen fraction is responsible for this. H/C ratio increases by 30% in the case of 37% of hydrogen in the blend. Low carbon fraction will produce low CO_2 concentration.

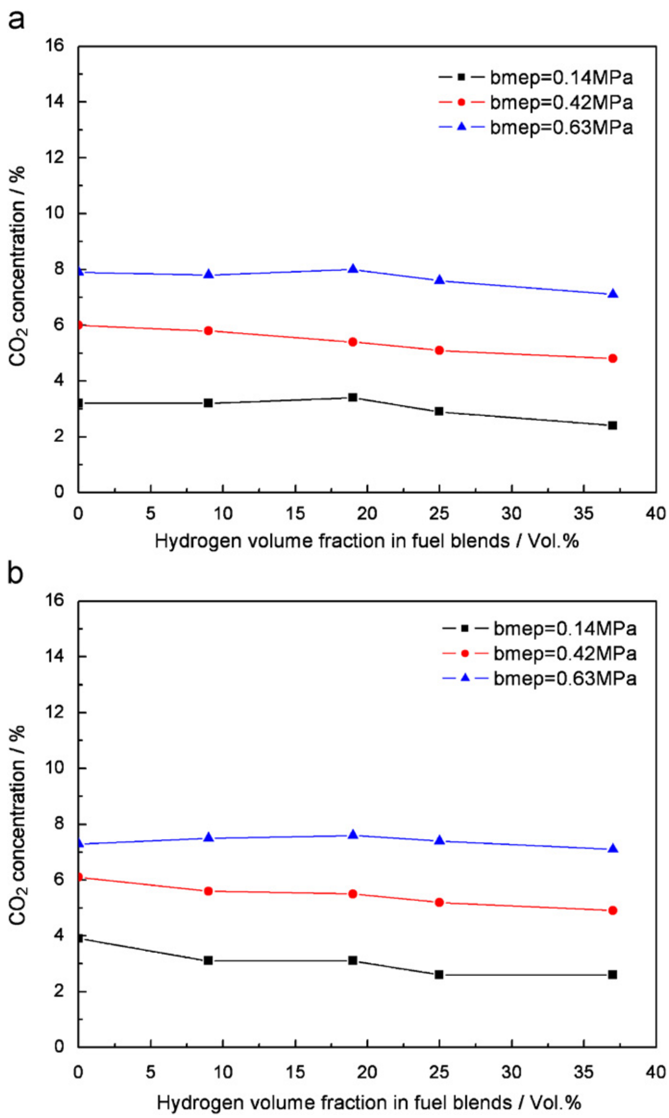

Fig. 14. Exhaust CO_2 concentration of fuel blends versus hydrogen fractions. ures versus hydrogen fractions. (a) n = 1200r/ min; (b) n = 1800r/ min.

4. Conclusions

Combustion behaviors of a direct-injection engine operating on various fractions of natural gas–hydrogen blends were studied experimentally. The main results are summarized as follows:

1. Brake effective thermal efficiency increased with the increase of hydrogen fraction at low and medium engine loads and high thermal efficiency is maintained at high engine load.
2. The beginning of the heat release advanced with the increase of hydrogen fraction. The rapid combustion duration decreased and the heat release rate increased with the increase of hydrogen fraction. This phenomenon was more obvious at low engine speed. The maximum mean gas temperature and the maximum rate of pressure rise increased with the increase of hydrogen fraction.
3. Exhaust HC and CO_2 concentrations decreased with the increase of the hydrogen fraction. Exhaust NO_x concentration increased with the increase of hydrogen fraction at high engine load, and it increased remarkably when the hydrogen volumetric fraction exceeds 20%.
4. From comprehensive evaluation of engine performance and emissions, the study suggests that 20% of hydrogen fraction in natural gas can get the optimum results.

Acknowledgments

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