Experimental Study on Engine Performance and Emissions for an Engine Fueled with Natural Gas−Hydrogen Mixtures

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An experimental study on the performance and emissions of a spark ignition engine operating on the natural gas−hydrogen mixtures was conducted. The results show that the engine lean-burn limit is extended by the addition of hydrogen into natural gas. For a specific excessive air ratio, engine power output and thermal efficiency decrease with the increase of hydrogen fraction in natural gas when the hydrogen fraction is less than a certain value (20%) whereas engine power output and thermal efficiency increase with further increasing hydrogen fraction when hydrogen fraction is larger than a certain value (20%). Addition of hydrogen into natural gas decreases the exhaust hydrocarbon (HC) concentrations. However, addition of hydrogen into natural gas will increase NO_x concentration. Thus, an engine operating on lean-burn natural gas−hydrogen combustion is favorable for getting higher thermal efficiency and lower emissions.

Introduction

With increasing concern about energy shortage and environmental protection, research on improving engine fuel economy and reducing exhaust emissions has become the major research aspect in combustion and engine development. As a result of 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 engine combustion process. 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, because of 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] As a result of these restrictions, a natural gas engine is usually operated at the condition of stoichiometric equivalence ratio with relatively low thermal efficiency. 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 the 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_x emission.[3] To overcome the demerit of the homogeneous charge gaseous fuel engine, Huang et al. used the natural-gas direct-injection method to increase the volumetric efficiency and flame propagation speed.[4] Another effective method to solve the problem of the 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.[5,6]

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 the 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 hydrocarbon (HC), CO, and

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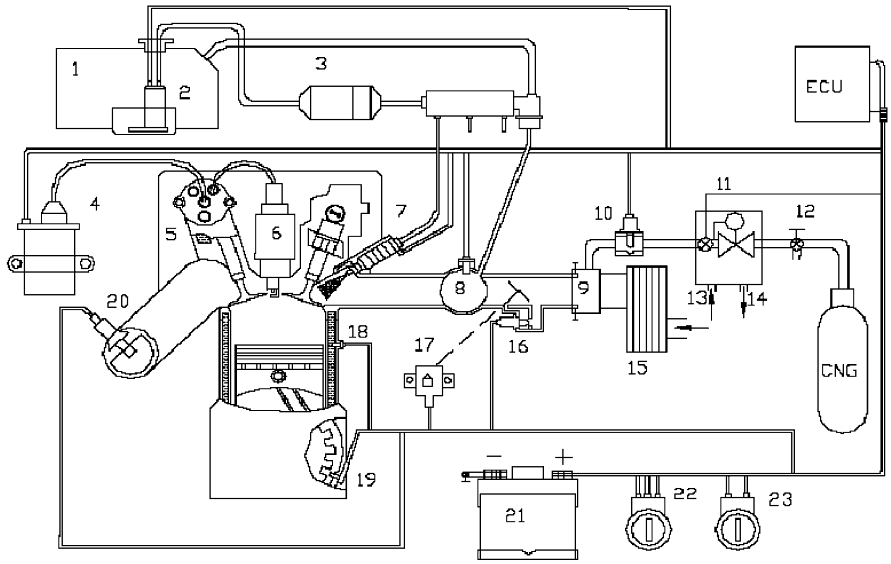

Figure 1. Schematic diagram of the experimental setup. 1, fuel tank; 2, fuel pump; 3, filter; 4, ignition coil; 5, destributor; 6, spark plug; 7, injector; 8, pressure sensor; 9, mixer; 10, CNG step motor; 11, pressure releasing valve; 12, CNG feeding valve; 13, coolant in; 14, coolant out; 15, air filter; 16, idling step motor; 17, throttle valve position sensor; 18, coolant temperature sensor; 19, speed sensor; 20, oxygen sensor; 21, battery; 22, ignition switch; 23, fuel switch.

CO_2 concentrations could be decreased when exhaust concentration from an engine operated on natural gas−hydrogen blends were compared to those of the natural gas engine. However, NO_x may increase for the natural gas−hydrogen combustion at a rich mixture condition as the improvement of lean burning ability and increased flame propagation speed, and NO_x concentration can be greatly decreased through lean combustion and retarding of the ignition advance angle. The previous work mainly concentrated on engine performances and emissions fueled by homogeneous natural gas−hydrogen mixtures at the stoichiometric equivalence ratio, and few literatures were found for various equivalence ratios using natural gas−hydrogen blends. 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 makes the system complicated and difficult to control. This paper will investigate the performances and emissions of a spark-ignited engine fueled with various fractions of natural gas−hydrogen mixtures under various equivalence ratios; various fractions of natural gas−hydrogen are prepared in advance in a fuel tank and are supplied to the intake port of the engine. The study is expected to clarify the behaviors of engine fueled with various fractions of natural gas−hydrogen mixtures at various equivalence ratios. The study will enrich the understanding of the natural gas−hydrogen fueled engine and provide a practical guide to engine application.

Experimental Setup and Procedures

A three-cylinder automotive compressed natural gas (CNG) spark-ignited engine was used in the experiment, and the specifications of the engine are as follows: cylinder bore of 68.5 mm, stroke of 72 mm, displacement of 796 mL, compression ratio of 9.4, the speed of 5500 rpm, and rate of power of 26.5 kW. The natural gas−hydrogen blends with different fractions of hydrogen were prepared in advance in the CNG tank. Four kinds of fuels were prepared in this study; they are, the pure natural gas, the fuel blend with 90% natural gas and 10% hydrogen in volume, the fuel blend with 80% natural gas and 20% hydrogen, and the fuel blend with 74% natural gas and 26% hydrogen. Figure 1 shows the schematic diagram of the experimental setup; fuel is supplied to the engine intake port through a gas mixer to the intake port, and the amount of fuel and excessive air ratios are controlled by a designed electronically controlled unit (ECU) unit and regulated by a step motor in the mixer. The experiments were conducted at wide opening throttle (WOT), maximum brake torque ignition timing (MBT), and an engine speed of 2000 rpm. A Horiba 7100 exhaust analyzer was used to measure exhaust HC, CO, CO_2, and NO_x concentration, and the analyzer has the measuring accuracy of 1 ppm for HC, 0.01% for CO, 0.01% for CO_2, and 1 ppm for NO_x. The Horiba sensor for the excessive air ratio was used to detect the mixture concentration with the measuring accuracy of 5%. In the experiments, the exhaust gases were measured when the engine operating parameters were adjusted at the specified conditions; that is, exhaust gases were measured at steady operating conditions.

Table 1. Fuel Properties of Natural Gas and Hydrogen

types of fuel	natural gas	hydrogen
density, kg/m^3 at 1 atm and 300 K	0.754	0.082
stoichiometric A/F ratio, vol %	9.396	2.387
laminar burning velocity, m/s	0.38	2.9
quenching distance, mm	2.03	0.64
conductivity at 300 K, mW/(m^2 K)	34	182
lower heating value, MJ/kg	43.726	119.7
lower heating value, MJ/m^3	32.97	9.82
cetane number	127	
C/H ratio	0.25	0

Table 1 gives the fuel properties of natural gas and hydrogen. Hydrogen with purity of 99.995% is used, while natural gas constitution is listed in Table 2. It can be seen that the laminar burning velocity of hydrogen is five times that of natural gas, and the quenching distance of hydrogen is one-third that of natural gas; the latter will be beneficial to reducing unburned HCs near the wall and top-land crevice. Meanwhile, hydrogen has higher conductivity than that of natural gas, and this may increase the heat transfer to the coolant in the case of natural gas−hydrogen combustion

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Table 2. Compositions of Natural Gas

item	CH_4	C_2H_6	C_3H_8	N_2	CO_2	others
volumetric fraction (%)	96.160	1.096	0.136	0.001	2.540	0.067

compared to that of natural gas combustion. The mass lower heating value of hydrogen is larger than that of natural gas, but the volumetric lower heating value of hydrogen is smaller than that of natural gas.

Because the volumetric heating value of hydrogen is less than that of natural gas, the volumetric heat value of natural gas−hydrogen mixtures will decrease with the increase of hydrogen fraction in the fuel blends. For a given fuel injection duration, the amount of heat release will be decrease with the increase of hydrogen fraction in fuel blends, and to maintain the same equivalence ratio, more fuel must be injected for natural gas−hydrogen mixture combustion.

Results and Discussion

Figure 2 gives the experimental results of the optimum ignition timings versus the excessive air ratio for various hydrogen fractions in the fuel blends. The optimum ignition timing is the ignition timing for getting the maximum brake torque (MBT). For all mixtures, the optimum ignition timing increases or advances with the increase of the excessive air ratio. The optimum ignition timing will be postponed with the increase of hydrogen addition in natural gas−hydrogen blend. A lean mixture with large excessive air ratio reduces the burning velocity, and this needs the advancement of the ignition timing to avoid power loss due to the extended combustion duration. Addition of hydrogen in natural gas will shorten the ignition delay and increase the burning velocity of the mixture, and the reduction in ignition delay requires the postponing of ignition timing and avoidance of the large increase of cylinder pressure at a late stage of the compression stroke, which may increase the power loss for overcoming the high cylinder pressure. In the case of natural gas−hydrogen combustion, properly delaying the ignition timing does not postpone the combustion duration because the burning velocity of the mixtures will increase by hydrogen addition. For both natural gas combustion and natural gas−hydrogen combustion, the shortest MBT is presented at a stoichiometric equivalence ratio ($\lambda = 1.0$) or equivalence ratio slightly less than the stoichiometric equivalence ratio ($\lambda = 0.9$), and it increases with the increase of the equivalence ratio. The experimental results show that the lean-burn limit can be extended with the increase of the hydrogen fraction in the fuel blend, compared to the lean-burn limit of the excessive air ratio

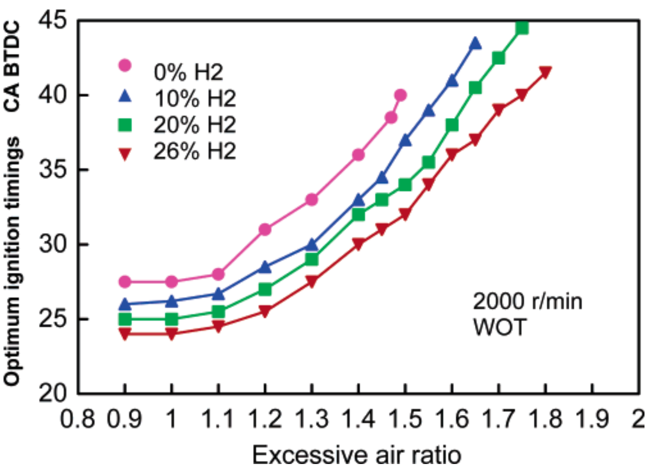

Figure 2. Optimum ignition timing versus excessive air ratio.

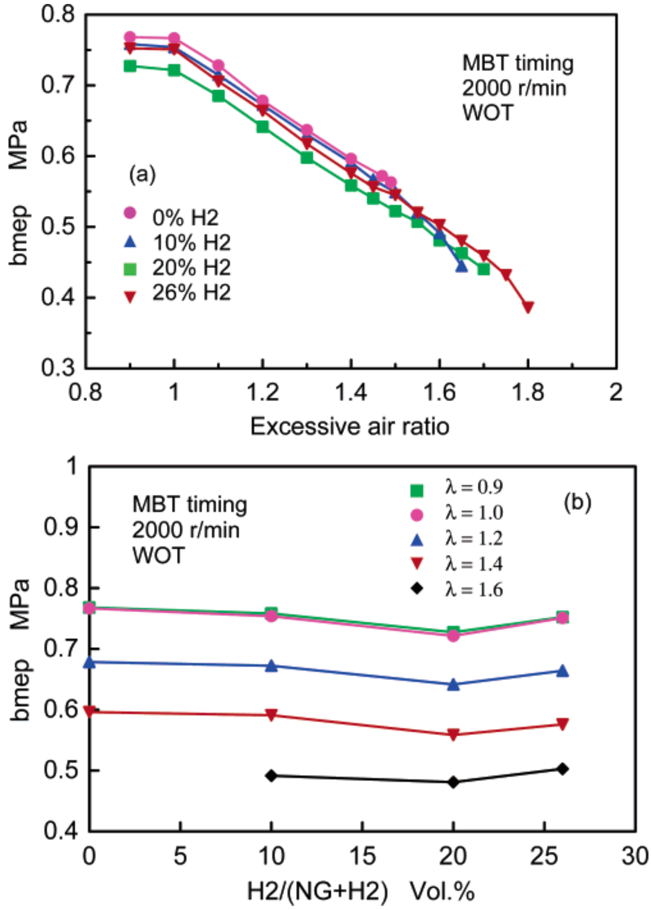

Figure 3. Brake mean effective pressure for natural gas−hydrogen combustion.

of 1.5, and the addition of 10, 20, and 26% hydrogen in natural gas will extend the lean-burn limits to the excessive air ratios of 1.65, 1.75, and 1.8, respectively. The findings by Gauducheau et al. in their numerical analysis also revealed the extension of the lean-burn limit by hydrogen addition.[15] This is due to the improvement of ignition and combustion when hydrogen in the mixture is available.

Figure 3a illustrates the experimental results of brake mean effective pressure (bmep) versus excessive air ratio (λ) for natural gas−hydrogen combustion. The results show that bmep decreases with the increase of excessive air ratio in the case of both natural gas combustion and natural gas−hydrogen combustion. No decreasing in bmep is observed from $\lambda = 0.9$ to $\lambda = 1.0$, while a linear reduction in bmep is presented from the stoichiometric mixture ($\lambda = 1.0$) to the lean mixtures. The latter behavior is coordinated to the reduction of the heat value which decreases linearly with the increase of the excessive air ratio. The results also reveal that bmep decreases with the increase of the hydrogen fraction when the hydrogen fraction is less than 20%, and bmep will increase with further increase of the hydrogen fraction when the hydrogen fraction is larger than 20%; this behavior can clearly be observed in Figure 3b. For a given excessive air ratio, the increase of the hydrogen fraction will reduce the volumetric heating value of the natural gas−hydrogen blend and result in the decrease in power. Hydrogen

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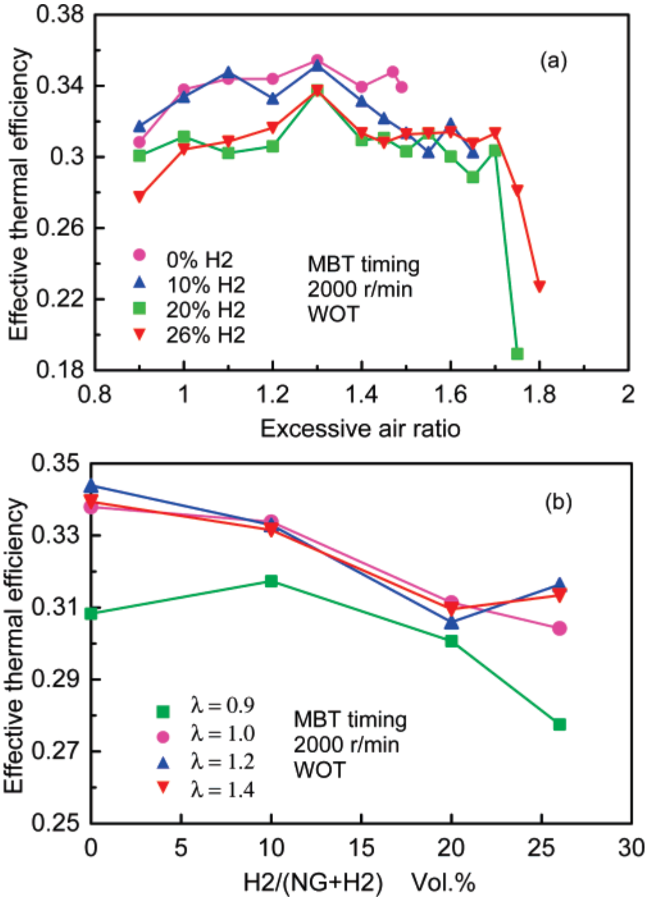

Figure 4. Effective thermal efficiency for natural gas−hydrogen combustion.

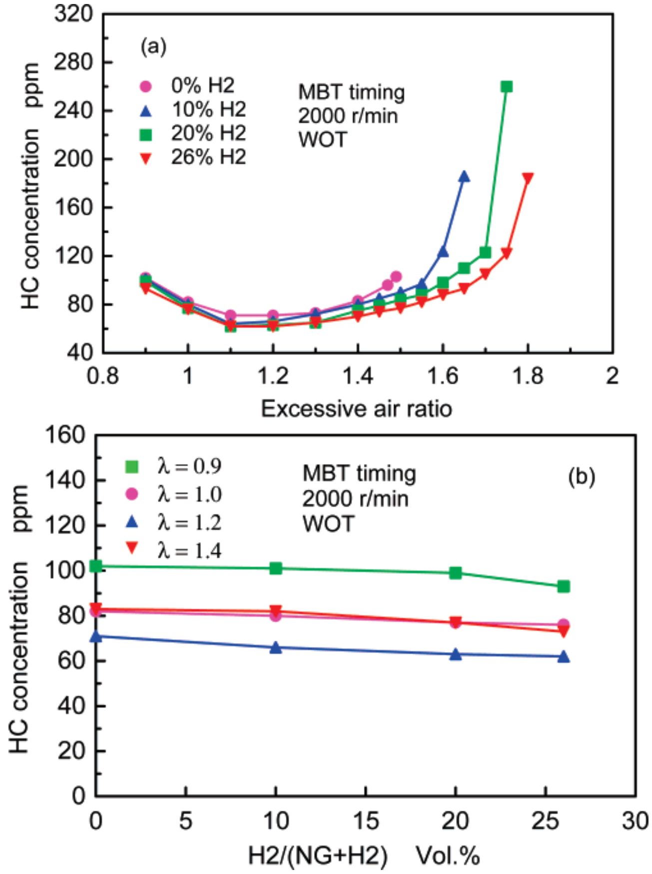

Figure 5. Exhaust HC concentration for natural gas−hydrogen combustion.

addition will increase the burning velocity of mixture, but the effect is not remarkable at small hydrogen addition.[11] Meanwhile, quenching distance decreases and heat transfer increases with the addition of hydrogen.[14] The effects of all these factors along with the decrease of heating value make bmep decrease with increase of hydrogen fraction when the hydrogen fraction is less than 20%. The improvement in combustion is remarkable when hydrogen fraction is larger than 20%, and this has been verified by the previous other study.[5,9,16] Thus, the gain in combustion improvement will be greater than the power loss from heat transfer and heating value reduction, causing the increase of bmep with the increase of the hydrogen fraction when the hydrogen fraction is larger than 20%. Experimental results conclude that, for a specific equivalence ratio, the addition of hydrogen in natural gas can only cause the increase in the power output when the hydrogen fraction is over a certain value.

Figure 4a shows the experimental results of effective thermal efficiency versus hydrogen fractions. When the excessive air ratio (λ) is less than 1.3, the effective thermal efficiency shows an increase with the increase of the excessive air ratio, and this is due to the increase of the specific heat ratio of the mixture. When λ is in the range between 1.3 and 1.7, the effective thermal efficiency decreases with the increase of the excessive air ratio, and this is due to the decrease in the mixture burning velocity and the increase in the combustion duration. When λ is larger than 1.7, a sharp decrease of the effective thermal efficiency is presented with the increase of the excessive air ratio, and this is due to the possible occurrence of bulk mixture quenching and partial burn phenomenon. For the specific excessive air ratio, the addition of hydrogen in natural gas causes the decrease in effective thermal efficiency, and as was explained above, the increase in combustion temperature and conductivity will enhance heat transfer from cylinder gases to coolant, increasing the fraction of heat transferred to coolant and reducing the effective thermal efficiency.[14]

Figure 4b gives the experimental results of the effective thermal efficiency versus hydrogen fractions. In the case of a rich mixture ($\lambda = 0.9$), maximum thermal efficiency is presented at 10% hydrogen fraction, and further increase in the hydrogen fraction will decrease the effective thermal efficiency. The reason may be due to the decrease in quenching distance which in turn decreases unburned fuel in the quenched layer and top-land crevice. The effective thermal efficiency decreases with the increase of the hydrogen fraction when the hydrogen fraction is larger than 10%, and this is due to the increase of combustion temperature and heat transfer with the increase of the hydrogen fraction. Moreover, postponing the ignition timing will to some extent reduce the effective thermal efficiency. Reduction in effective thermal efficiency due to an increase in heat transfer is larger than the improvement in thermal effective efficiency by burning velocity enhancement, and this consequently makes the decrease of the effective thermal efficiency with the increase of the hydrogen fraction. In the case of stoichiometric mixture ($\lambda = 1.0$), the effective thermal efficiency decreases monotonically with the increase of the hydrogen fraction. In the case of lean mixtures ($\lambda = 1.2$ and $\lambda = 1.4$), the effective thermal efficiency decreases with the increase of the hydrogen fraction when the hydrogen fraction is less than 20%, while it increases with the increase of the hydrogen fraction when hydrogen

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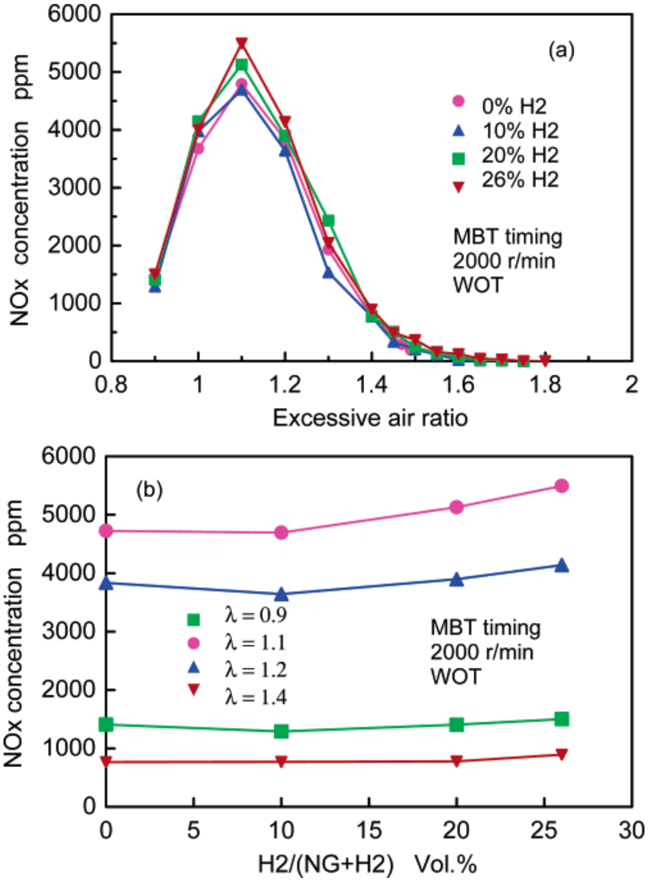

Figure 6. Exhaust NO_x concentration for natural gas−hydrogen combustion.

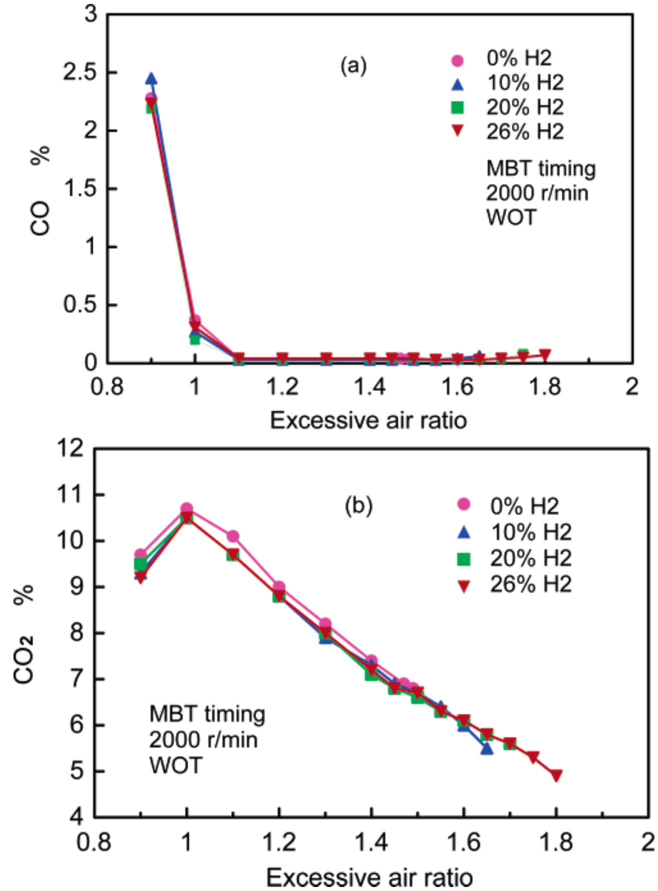

Figure 7. Exhaust CO and CO_2 concentrations for natural gas−hydrogen combustion.

fraction is larger than 20%. For lean mixture combustion, the influentual degree of hydrogen addition on heat transfer decreases with the increase of hydrogen fraction, but the influentual degree of hydrogen addition on combustion improvement increases with the increase of hydrogen fraction. Thus, in the case of a hydrogen fraction larger than 20%, an increase in the effective thermal efficiency is presented.

Figure 5a shows the experimental results of exhaust HC concentration versus excessive air ratios, and the exhaust HC concentration versus hydrogen fractions is plotted in Figure 5b. The figure shows that a similar trend of HC concentration versus excessive air ratio is demonstrated, and the minimum value of HC is presented in the range of $\lambda = 1.1$ to $\lambda = 1.3$ in the case of both natural gas combustion and natural gas−hydrogen combustion. HC concentration increases with the decrease of excessive air ratio when λ is less than 1.0, and this is due to the increase of unburned fuel for a rich mixture combustion. The HC concentration is increased with the increase of λ when λ is larger than 1.3, while the HC concentration turns to a remarkable increase when λ reaches a certain excessive air ratio, and this excessive air ratio will shift to the leaner mixture in the case of higher hydrogen fraction. The turning points for remarkable HC increase for the mixtures with 10% hydrogen fraction, 20% hydrogen fraction, and 26% hydrogen fraction in the fuel blends are $\lambda = 1.55$, $\lambda = 1.7$, and $\lambda = 1.75$, respectively. The results indicate that the lean-burn limit can be extended with the increase of hydrogen addition in natural gas. Figure 5b shows that the exhaust HC concentration decreases with the increase of hydrogen fraction in natural gas−hydrogen blends, and this can be explained by the following interpretations. In one aspect, the fraction of natural gas is decreased with the increase of the hydrogen fraction, and this will reduce the unburned fuel from natural gas. While in another aspect, the quenching distance of the mixture is decreased with the increase of the hydrogen fraction in the fuel blend, and this will decrease the unburned fuel in the quenching layer on the combustion chamber surface, enabling the flame to propagate into the top-land crevice and burn the fuel in the crevice. Moreover, HC post-flame oxidation is promoted due to the increase in combustion temperature by hydrogen addition. All these factors decrease the exhaust HC concentration with the increase of hydrogen fraction.

Figure 6a gives the experimental results of the exhaust NO_x concentration versus excessive air ratios. For both natural gas combustion and natural gas−hydrogen combustion, NO_x concentration reaches its peak value at $\lambda = 1.1$, and this is consistent with the spark ignition engine fueled with gasoline and alcohols. In the case of an excessive air ratio less than 1.1, the rich mixture combustion with insufficient oxygen in the cylinder decreases the NO_x concentration. In the case of excessive air ratio greater than 1.1, the dilution of the mixture reduces the combustion temperature and decreases the NO_x concentration. For a specific excessive air ratio, NO_x concentration shows a slight increase with the increase of the hydrogen fraction, and this is due to the increase of combustion temperature by hydrogen addition. Figure 6b gives the exhaust NO_x concentration versus hydrogen fractions. The figure clearly shows that NO_x has little variation from natural gas combustion to the natural gas−hydrogen combustion with 10% hydrogen fraction. NO_x concentration increases with the increase of hydrogen fraction in the fuel blend when hydrogen fraction is larger than 10%, and this phenomenon is more obviously for mixtures at $\lambda = 1.1$ and $\lambda = 1.2$. For rich mixture ($\lambda = 0.9$) and leaner mixture ($\lambda = 1.4$) combustion, a small variation of the NO_x concentration with hydrogen fraction is demonstrated, and this is suggested by the suppression in NO_x formation in the case of either a rich mixture (suppressed by low oxygen concentration) or a leaner mixture (suppressed by low combustion temperature). The study also reveals that the influence of mixture concentration on NO_x formation is greater than hydrogen addition, especially at rich mixture combustion and lean mixture combustion.

Figure 7 illustrates the experimental results of the exhaust CO and CO_2 concentrations versus excessive air ratios. It can be seen that CO concentrations depend strongly on excessive air ratio. When λ is less than 1.1, the CO concentration increases remarkably with the decrease of λ for both the natural gas mixture combustion and the natural gas−hydrogen mixture combustion. The CO concentration remains a very low value at the excessive air ratio larger than 1.1, and this is due to the sufficient oxygen in the cylinder to make the combustion complete. For a specific excessive air ratio, little difference is observed between the natural gas mixture combustion and the natural gas−hydrogen mixture combustion. CO_2 concentration gives a peak value at a stoichiometric mixture; when the mixture is rich, there is not enough oxygen to permit CO to CO_2. When the mixture is lean, there is a large amount air available which causes the decrease of CO_2 with the increase of the excessive air ratio, and this is consistent to the findings by Gauducheau et al. in the study in ref 15.

Conclusions

An experimental study on performance and emissions of a spark ignition engine operating on the natural gas−hydrogen mixtures was conducted. The main results are summarized as follows:

(1) The engine lean-burn limit is extended by addition of hydrogen into natural gas. For a specific excessive air ratio, engine power output and thermal efficiency decrease with the increase of hydrogen fraction in natural gas when the hydrogen fraction is less than a certain value (20%), whereas engine power output and thermal efficiency increase with further increase of hydrogen fraction when the hydrogen fraction is larger than a certain value (20%).

(2) Addition of hydrogen into natural gas decreases the exhaust HC concentration. However, addition of hydrogen into natural gas will increase NO_x concentration. Thus, an engine operating on lean-burn natural gas−hydrogen combustion is favorable for getting higher thermal efficiency and lower emissions.

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