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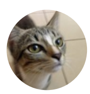

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Experimental Study on Emissions of a Spark-Ignition Engine Fueled with Natural Gas−Hydrogen Blends

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An experimental investigation on the influence of the excess air ratio and hydrogen fraction on the emissions characteristics of a spark-ignition engine fueled with natural gas−hydrogen blends was conducted. The results indicate that the excess air ratio has a significant effect on the hydrocarbon (HC), NO_x, CO, and CO_2 concentration for both pure natural gas and natural gas−hydrogen blends. For a specified excess air ratio, HC emissions decrease with the increase of hydrogen fraction; the behavior is more obvious under the lean burn operation. The NO_x concentration increases with the increase of hydrogen fraction, and NO_x gets its peak value at an excess air ratio of 1.1. CO_2 emissions decrease with increasing hydrogen fraction. Meanwhile, the addition of hydrogen into natural gas can extend the lean burn limit of a mixture. Thus, an engine fueled with natural gas−hydrogen blends operating under lean mixture conditions can get low emissions of HC, CO, CO_2, and NO_x.

1. Introduction

With the increasing concerns over the environmental protection and the shortage of crude oil supplying, much effort has been focused on the utilization of alternative fuels in engines. Natural gas is a potential alternative fuel due to its higher octane number, low emissions, low price, and abundant reserve. Although natural gas is considered as the clean fuel, natural gas vehicles are far from clean and NO_x emission still remains a high value under high load operation. In order to meet more stringent emission regulations,[1,2] many technologies, such as multipoint fuel injection, oxygen sensor feedback control, exhaust gas recirculation (EGR), and special catalyst, have been adopted in the natural gas engines. Besides, natural gas has its disadvantages, like poor lean burn capability and low flame propagation speed. Hydrogen is an excellent additive to natural gas due to its unique characteristics. It has a much better lean burn capability, high flame propagation speed, small quenching distance, and so forth.

Many researchers studied the effect of the addition of hydrogen into natural gas on performances and emissions in the past several years. Shudo et al.[3] found that an increase in the amount of premixed hydrogen could stabilize the combustion process and reduced hydrocarbon (HC) and CO emissions but increase the NO_x emissions in a methane stratified charge spark-ignition engine with hydrogen premixing under the same excess air ratio operation. Allenby et al.[4] reported that the addition of hydrogen into methane can improve combustion stability and the engine can tolerate up to 25% EGR, while maintaining a coefficient of variability of indicated mean effective pressure below 5%. This level of EGR gives a reduction in NO emissions greater than 80% at the stoichiometric air−fuel ratio. Wong et al.[5] analytically examined the effect of hydrogen enrichment and hydrogen addition on cyclic variations in homogeneously charged compression ignition engines. The results indicated that the addition of hydrogen can reduce cyclic variations while extending the operating region of the engine. Karim et al.[6] theoretically studied the addition of hydrogen on methane combustion characteristics at different spark timings. The theoretical results show that the addition of hydrogen into natural gas can decrease the average ignition lag and the average combustion duration at the same equivalence ratio. It indicated that the addition of hydrogen can promote the flame propagation speed, thus stabilizing the combustion process, especially the lean combustion process. Hoekstra et al.[7] and Sierens et al.[8] found in the experiment that the addition of hydrogen into natural gas could extend the lean burn limit of an engine due to the effect of the addition of hydrogen on the improvement of combustion stability. The results also indicated low NO_x

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Table 1. Engine Specifications

type	HH368Q spark-ignition engine
bore, mm	68.5
stroke, mm	72
displacement, cm^3	796
compression ratio	9.4
ignition sequence	1–3–2
rated power, kW	26.5
rated speed, rpm	5500

emissions could be obtained under lean combustion for natural gas−hydrogen blends. In order to evaluate the emissions of the vehicle using the natural gas−hydrogen blends, the U.S. Department of Energy launched the Advanced Vehicle Testing Activity (AVTA) program.[9] The results of the FTP-75 (FTP-75 is the abbreviation for the United States Federal Test Procedure 75) emission test in the Ford 150F vehicle using 30% hydrogen and 70% natural gas as a fuel were below the SULEV (SULEV is the abbreviation for super ultralow emission vehicle) emission standards for CO and NO_x emissions.

In this paper, the emissions of a spark-ignition engine fueled with various fractions of natural gas−hydrogen blends and under various excess air ratios are studied. The study is expected to clarify the behaviors of engines fueled with natural gas−hydrogen blends at various hydrogen fractions and can provide emission data over wide hydrogen addition and excess air ratio ranges and provide practical guidance for the engine operation.

2. Test Engine and Experimental Apparatus

The specifications of the test engine are listed in Table 1. The schematic diagram of the engine and control system is shown in Figure 1. The gas engine was modified from an HH368Q gasoline engine. The CNG−H_2 supplying system of the engine includes a fuel tank, pressure regulator, and gas mixer. A step motor was used to adjust the amount of CNG−H_2 blends entering the cylinder and the excess air ratio (excess air ratio is the ratio of the actual air−fuel ratio to the stoichiometric air−fuel ratio). An electronic control unit was used to control the spark timing and the step motor.

The hydrogen used in this study has a purity of 99.995%, and the compositions of natural gas are listed in Table 2. For the natural gas, we used the commercial product from Shaanxi Province of China. The fuel properties of natural gas and hydrogen are listed in Table 3. The lean burn capability of hydrogen is much better than that of natural gas, and the laminar burning velocity of hydrogen is 7 times that of natural gas. Thus, the addition of hydrogen to natural gas can increase the flame propagation speed and stabilize the combustion process, especially under lean mixture combustion. The quenching distance of hydrogen is one-third that of natural gas, and this is beneficial to reducing the unburned HC near the wall and the top-land crevices. Although the mass heating value of hydrogen is larger than that of natural gas, the volumetric heating value of hydrogen has a lower value than that of natural gas. Moreover, the volumetric fraction of hydrogen at stoichiometric air−fuel ratio is greater than that of natural gas and hydrogen occupies a large proportion of volume at the stoichiometric air−fuel ratio. A stoichiometric hydrogen−air mixture contains less energy than a stoichiometric natural gas−air mixture.

In this study, five natural gas−hydrogen blends were studied. The volumetric fractions of hydrogen in the natural gas−hydrogen blends were 0, 12, 23, 30, and 40%, respectively. The natural gas−hydrogen blends were prepared in the fuel tank in advance, and they were supplied to the engine intake system through a gas mixture. The engine was operated at a speed of 3000 rpm and a brake mean effective pressure (bmep) of 0.16 and 0.32 MPa. A Horiba MEXA-700λ instrument for detecting excess air ratio was used to measure the mixture concentration; this analyzer has a measuring accuracy of 0.1 air−fuel ratio at an air−fuel ratio of 14.7. A Horiba MEXA-554JA instrument was used to measure the exhaust HC (hydrocarbon), CO, and CO_2 concentration; this analyzer has a measuring accuracy of 12 ppm for HC, 0.06% for CO, and 0.5% for CO_2. A Horiba MEXA-720 NO_x instrument with a measuring accuracy of 30 ppm was used to measure the exhaust NO_x concentration.

3. Results and Discussion

Figure 2 shows the unburned hydrocarbon concentration at various hydrogen fractions in natural gas−hydrogen blends. The variation of HC versus excess air ratio (λ) shows a similar trend at various hydrogen fractions. HC gets its lowest value at excess air ratios between 1.1 and 1.2 in the case of both natural gas−air mixture combustion and natural gas−hydrogen-air mixture combustion. In the case of rich mixture combustion, the insufficient oxygen in the cylinder increases the HC concentration. In the case of leaner mixture combustion, the occurrence of bulk quenching increases the HC concentration. Increasing the hydrogen fraction can extend the mixture lean burn limit and causes low HC concentrations at even large excess air ratios. Increasing the engine load (bmep) can extend the lean burn limit, and this is due to the increase of cylinder gas temperature. If the mixture approaches its lean burn limit, a remarkable increase in HC concentration is presented. The advantage of natural gas−hydrogen blends is to extend the lean burn limit and make the engine operate under lean mixture conditions. This is beneficial to increasing engine thermal efficiency and decreasing engine NO_x concentration.

HC concentration versus hydrogen fraction is shown in Figure 3. When the excess air ratio is larger than 1.5, the HC concentration decreases with the increase of the hydrogen fraction in the blends. When the excess air ratio is smaller than 1.5, the HC concentration maintains an almost constant low value regardless of the hydrogen fraction in the blends. This suggests that hydrogen addition can contribute to HC reduction at leaner mixture combustion. In this case, the bulk-quenching phenomenon will occur in the natural gas−air mixture combustion.

Figure 4 illustrates the engine NO_x concentration at various hydrogen fractions. Similar to the HC behavior, the variation of NO_x to excess air ratio shows a similar trend at all hydrogen fractions. The NO_x concentration reaches it peak value at an excess ratio of 1.1. For rich mixture combustion, the insufficient oxygen in the cylinder decreases the NO_x concentration. When the excess air ratio is larger than 1.1, the NO_x concentration decreases remarkably with the increase of excess air ratio due to the decrease in combustion temperature. When the excess air ratio is larger than 1.6, a very low NO_x concentration is presented. Increasing the engine load increases the NO_x concentration but has little influence on the pattern of NO_x concentration to excess air ratio.

Figure 5 shows the NO_x concentration versus the hydrogen fraction in the natural gas−hydrogen blends. The NO_x concentration increases with the increase of hydrogen fraction in the blends. The combustion improvement and high cylinder gas temperature are responsible for this.[10] At excess air ratios between 1.0 and 1.2, the behavior of NO_x increasing with the increase of hydrogen fraction becomes remarkable, and this suggests that the cylinder gas temperature makes a rapid increase with the increase of hydrogen

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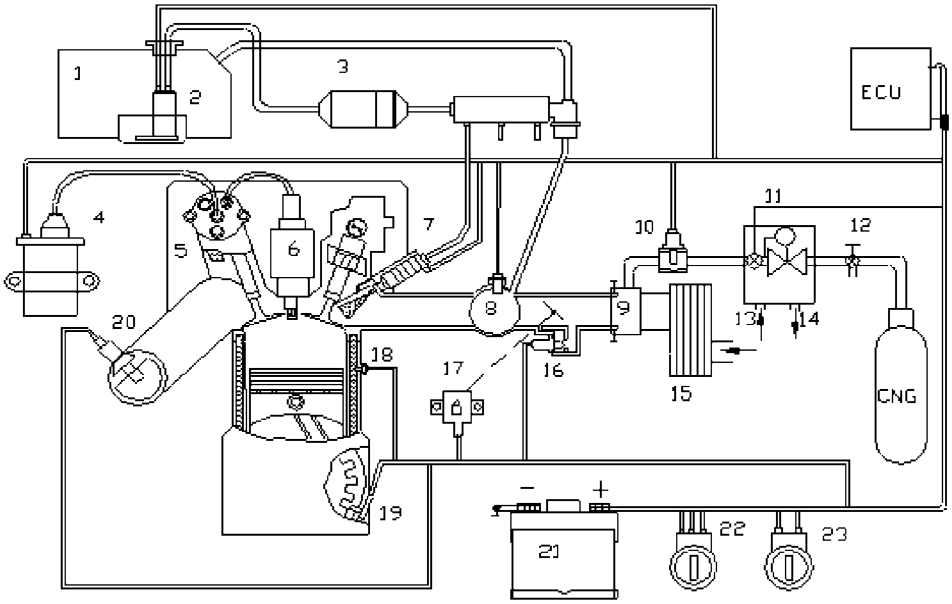

Figure 1. Schematic diagram of the engine and control system: 1, fuel tank; 2, fuel pump; 3, fuel filter; 4, ignition coil; 5, distributor; 6, spark plug; 7, fuel injector; 8, manifold pressure sensor; 9, gas mixer; 10, CNG−H_2 control step motor; 11, CNG−H_2 regulator; 12, CNG−H_2 valve; 13, coolant in; 14, coolant out; 15, air filter; 16, idling control step motor; 17, throttle position sensor; 18, coolant water temperature sensor; 19, crankshaft position sensor; 20, oxygen sensor; 21, battery; 22, ignition switch; 23, gasoline/CNG selection switch.

Table 2. Compositions of Natural Gas[a]

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

[a] Data source is mainly from ref 10.

Table 3. Fuel Properties of Natural Gas and Hydrogen[a]

fuel	natural gas	hydrogen
equivalence ratio of lean burn lower limit at 293 K and 1 atm	0.53	0.1
density at 300 K and 1 atm, kg/m^3	0.754	0.083
mass lower heating value, MJ/kg	43.726	119.930
volumetric lower heating value at 300 K and 1 atm, MJ/ m^3	32.97	9.82
stoichiometric air−fuel ratio, kg/kg	17.19	34.20
volumetric fraction of fuel at stoichiometric air−fuel ratio, %	9.5	29.0
molar carbon-to-hydrogen ratio	0.25	0
quenching distance in NTP air, mm	2.03	0.64
laminar burning velocity, m/s	0.38	2.9
adiabatic flame temperature in air, K	2148	2318
conductivity at 300 K and 1 atm, mW/(m^2 K)	34	182
minimum energy required for ignition in air, mJ	0.29	0.02
octane number	127	

[a] Data source is mainly from ref 11.

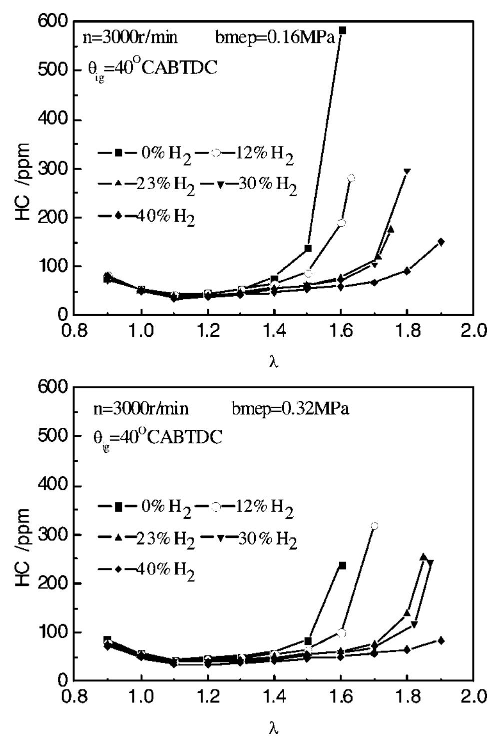

Figure 2. HC concentration at various hydrogen fractions.

fraction in the blends. In the case of leaner mixture combustion, only a slow increase in NO_x concentration with the increase of hydrogen fraction is demonstrated. This indicates that hydrogen addition at lean mixture combustion does not accompany high NO_x concentration. It can be seen that NO_x concentration is less than 20 ppm when the excess air ratio is larger than 1.7 for the blend with 40% hydrogen and 60% natural gas. This indicates that the lean limit can be extended while introducing hydrogen into natural gas and lower NO_x can be obtained for the lean mixtures even if no postcatalyst is used.

Figure 6 gives the CO concentration at various hydrogen fractions. The results clearly show that the mixture concentration strongly influences the CO concentration. At excess air ratios larger than 1, CO remains at a very low value regardless of the mixture concentration and hydrogen fraction. The CO concentration increases remarkably with increasing excess air ratio at excess air ratios smaller than 1. The insufficient oxygen is responsible for this increasing behavior.

The CO_2 concentration at various hydrogen fractions is plotted in Figure 7. The CO_2 concentration gets its peak value at the stoichiometric mixture. An increase in the excess air ratio will decrease the CO_2 concentration due to mixture

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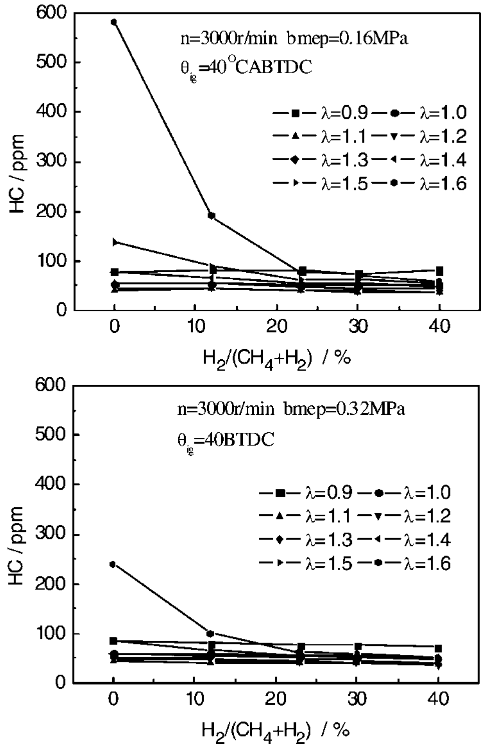

Figure 3. HC concentration versus hydrogen fraction.

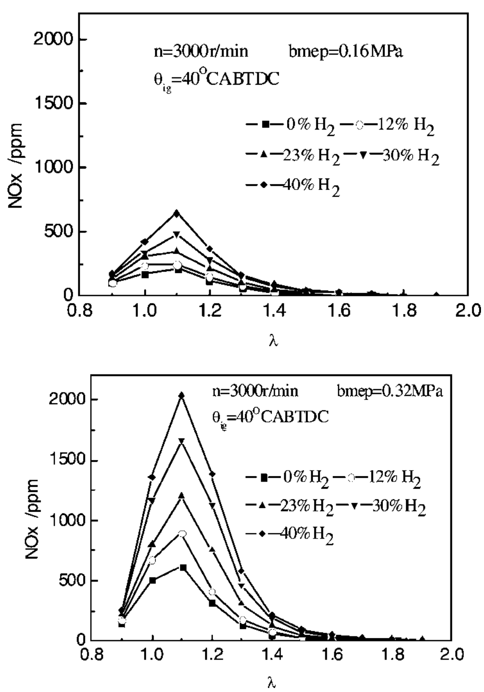

Figure 4. NO_x concentration at various hydrogen fractions.

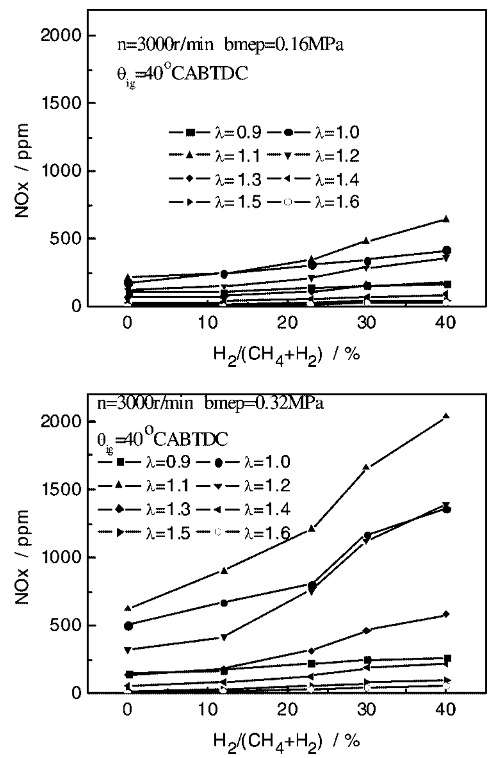

Figure 5. NO_x concentration versus hydrogen fraction.

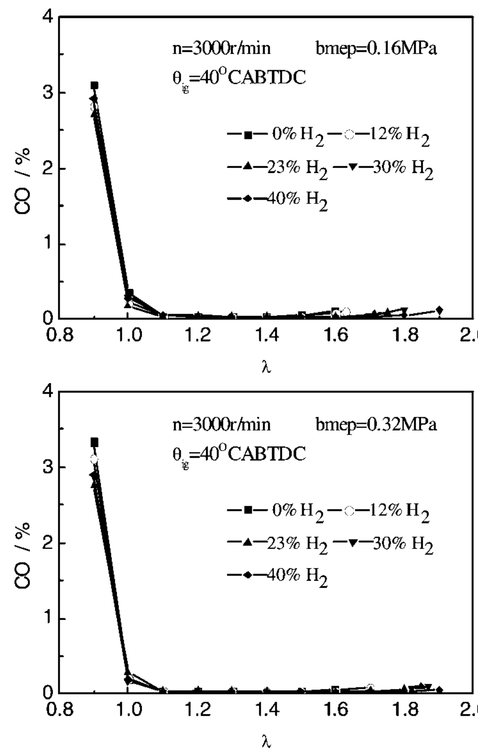

Figure 6. CO concentration at various hydrogen fractions.

dilution, while rich mixture combustion causes a high concentration of CO and a low CO_2 concentration. For lean mixture combustion, the bulk quenching leads to a sharp decrease in CO_2 concentration, and this is due to the increase of unburned hydrocarbons.

Figure 8 shows CO_2 concentration versus hydrogen fraction. The results show that the CO_2 concentration decreases with the increase of hydrogen fraction in most cases; the decrease in C/H ratio in the blends leads to the decrease in CO_2 concentration. In the case of leaner mixture combustion ($\lambda = 1.6$), pure natural gas combustion may accompany the bulk-quenching phenomenon, resulting in the increase in HC concentration and subsequently low CO_2 concentration.

4. Conclusions

An experimental study on the emissions of a spark-ignition engine fueled with natural gas−hydrogen blends was conducted. The main conclusions are summarized as follows:

(1) For a specified excess air ratio, the HC concentration decreases with the increase of the hydrogen fraction. Addi-

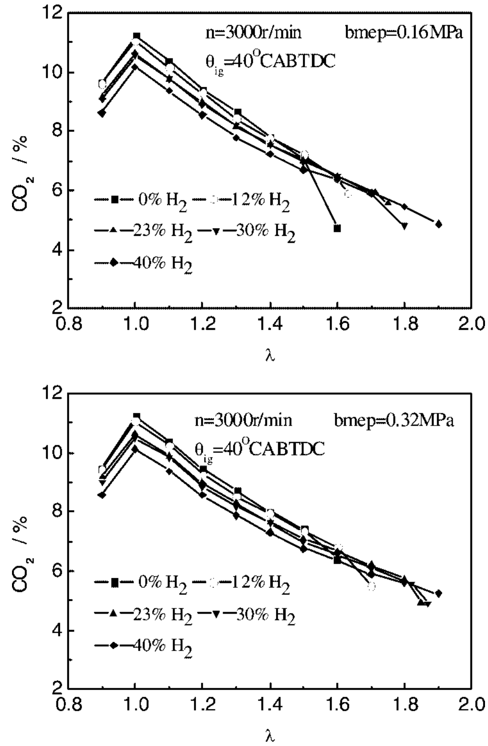

Figure 7. CO_2 concentration at various hydrogen fractions.

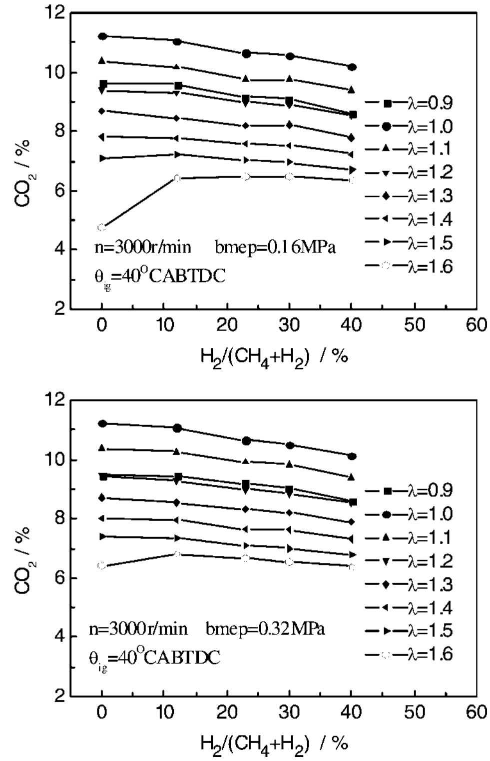

Figure 8. CO_2 concentration versus hydrogen fraction.

tion of hydrogen into natural gas can extend the lean burn limit.

(2) The NO_x concentration gets its peak value at an excess air ratio of 1.1 and increases with the increase of hydrogen fraction in the blends. Lower NO_x emission can be obtained for lean combustion even if no postcatalyst is used.

(3) The CO concentration is mainly influenced by the excess air ratio and has little influence from hydrogen addition. The CO_2 concentration decreases with the increase of the hydrogen fraction in the blends.

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