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The influences of hydrogen on the performance and emission characteristics of a heavy duty natural gas engine

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ABSTRACT

Because blending hydrogen with natural gas can allow the mixture to burn leaner, reducing the emission of nitrogen oxide (NO_x), hydrogen blended with natural gas (HCNG) is a viable alternative to pure fossil fuels because of the effective reduction in total pollutant emissions and the increased engine efficiency.

In this research, the performance and emission characteristics of an 11-L heavy duty lean burn engine using HCNG were examined, and an optimization strategy for the control of excess air ratio and of spark advance timing was assessed, in consideration of combustion stability. The thermal efficiency increased with the hydrogen addition, allowing stable combustion under leaner operating conditions. The efficiency of NO_x reduction is closely related to the excess air ratio of the mixture and to the spark advance timing. With the optimization of excess air ratio and spark advance timing, HCNG can effectively reduce NO_x as much as 80%.

1. Introduction

Natural gas is a clean fuel that features the lowest carbon content of all hydrocarbon fuels. Therefore, it is thought to be a good alternative to diesel and gasoline due to its lower emission levels of non-methane hydrocarbons (NMHC) and carbon monoxide (CO) [1,2]. Generally, turbocharged natural gas engines operate under lean burn conditions to prevent thermal damage and increase thermal efficiency in heavy duty vehicle applications. Although a natural gas engine functioning in the lean burn operation region uses the margin of the lean operating condition for the reduction of NO_x emissions, it is difficult to satisfy restrictive exhaust gas regulations, such as EURO-VI, when only this approach to lean burn combustion is utilized [3,4].

Hydrogen shows combustion characteristics different from those of hydrocarbon fuels, including a wide range of flammability, rapid combustion, a short quenching distance, and a high adiabatic flame temperature. Such characteristics of hydrogen enable ultra-lean burning, high efficiency, and low exhaust emissions in the low load region [5,6]. Hydrogen is a clean fuel that neither generates harmful exhaust gases such as particulate matter (PM), total hydrocarbon (THC), and CO nor emits the green house gas carbon dioxide (CO_2). Adding hydrogen to natural gas will improve engine performance due to its lean operational capacity. Technologies for using hydrogen in the internal combustion engines have been studied since the 1800s, and various techniques for enriching hydrogen to natural gas in order to widen the lean operation range and to reduce the harmful emission of exhaust gases have been introduced.

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Hydrogen mixed with compressed natural gas (HCNG) can extend the lean burn limit while maintaining stable, efficient combustion and achieving lower NO_x, hydrocarbon (HC) and green house gas (GHG) emissions [7–12].

However, because simply adding hydrogen to natural gas does not guarantee the reduction of harmful emissions, several parameters should be evaluated and optimized for the development of lean burn engines that can operate in the field of practical application. It is believed that HCNG can be used to meet the requirements of EURO-VI by controlling the lean limit of combustion and through the application of a proper aftertreatment system. This study analyzes the effect of mixing natural gas with hydrogen on the combustion of an engine in order to understand its applicability to a hydrogen–natural gas engine and to optimize the hydrogen–natural gas engine. The study examines the characteristics of emissions and efficiency with respect to various percentages of hydrogen in the HCNG fuel and various engine control parameters such as excess air ratio and spark ignition timing. The applicability of hydrogen-blended fuel to the field is also assessed.

2. Experimental procedures

2.1. Experimental setup

The test engine selected for this study was an 11 L, 6-cylinder natural gas engine for a city bus. The test engine was coupled with an eddy current (EC) dynamometer (Schenck) and equipped with a fuel supply system and an electronic control system to observe the effects of hydrogen addition. Table 1 shows the main specifications of the test engine and Fig. 1 shows a schematic of the experimental setup for the engine. Regular natural gas supplied from Korea gas corporation and 99.9 percent normal hydrogen was used as fuels. The properties of natural gas were shown in a Table 2. Natural gas, the main fuel, was supplied to the intake manifold through the metering valve and mixer after being decompressed to 0.8 MPa by the regulator from the compressed fuel vessel, which was charged to around 20 MPa. The amount of fuel was controlled by a metering valve module consisting of eight gas injectors. To prevent cooling resulting from the expansion of the compressed fuel by the regulator, a heat exchanger was installed. In this way, the gas temperature was maintained at 40 °C. The hydrogen fuel was supplied through a mass flow controller (MFC) at 0.8 MPa using the pressure controller, mixed with the intake air at the mixer upstream, and then fed into the suction manifold. The original HCNG fuel is based on a premixed fuel of hydrogen and natural gas. However, it is believed that this layout causes the premixed fuel to be sufficiently charged with air upstream of the intake manifold and that there is no significant influence on the results due to the mixing of hydrogen and natural gas within the manifold.

In order to evaluate the applicability of gas fuel in the practical field, the typical fuel regulator and injector for commercial vehicle were used as fuel system for natural gas. The regulator provides a precise fuel pressure control with 0.683–0.863 MPa nominal outlet pressure range at various inlet pressures and the maximum supply effect of the regulator is 0.004 P_{Outlet}/P_{Inlet}. The components of injector are designed to cope with the high gas volume and the high throughflow velocity. The flow rate of the natural gas was measured using a mass flow meter (MFM), while that of the hydrogen was directly controlled using the MFC. The output of mass flow sensor which consists of a stainless steel capillary tube with resistance thermometer elements is directly proportional to the total mass flow rate and a proportional, electromagnetic control valve with extremely fast and sooth control characteristics is actuated with 1–2 s of settling time. Its accuracy is 0.8% Reading plus 0.2% Full scale (Max. flow 1500 L/min). The injection amount, injection time and ignition time of the natural gas fuel were controlled using a computer-operated engine control system (ECS). During the test, the amounts of hydrogen were controlled by the MFC, and the speed and the load of the engine were controlled using an EC dynamometer system. The engine control variables, such as the engine speed and intake throttle position, were monitored, and the excess air ratio was measured using an LA 4 lambda meter (ETAS Co.).

Gas leak detectors and flashback arrestors for each fuel as safety devices were used. The leak detector for hydrogen with ion-source filaments guaranteed the sensitivity from $1 \cdot 10^{-7}$ atm.cc/s with 5% H_2/95% N_2. Natural gas detector allows less than 1 s of response time for 10 ppm methane. Both leak detection systems comprise a gas leak detector and a motor actuated valve. The valve installed at the supplying line upstream automatically shuts off the fuel gas if detector detects leakage of fuel. To prevent flashback and sudden gas reverse flow, flashback arrestors were installed at the supplying pipe upstream for each fuel. It is equipped with sintered metal filter and large capacity non return valve and operates at 0.8 MPa working pressure.

A gas sampling probe was installed in the exhaust pipe and was connected to an exhaust gas analyzer (MEXA 7000, IHORIBA). The gaseous emissions were analyzed using a nondispersive infrared absorption (NDIR) analyzer for the CO and CO_2 measurements, a chemiluminescence detector (CLD) analyzer for the NO_x measurements, and a flame ionization detector (FID) analyzer for the THC measurements.

To analyze in-cylinder combustion phenomena, the combustion pressure was measured and analyzed using a spark plug-type pressure sensor and a combustion analyzer in real time. For the analysis of this experiment, a data acquisition program was written using National Instruments LabView software which allows the acquisition of numerous channels of data and provides real-time measurement displays.

Table 1 – Engine specifications.

Bore (mm)	123
Stroke (mm)	155
Compression ratio	10.5
Displacement (cc)	11,0550
Idle speed (rpm)	600 ± 50
Max. power	213 kW/2200 rpm
Max. torque	1226 Nm/1260 rpm

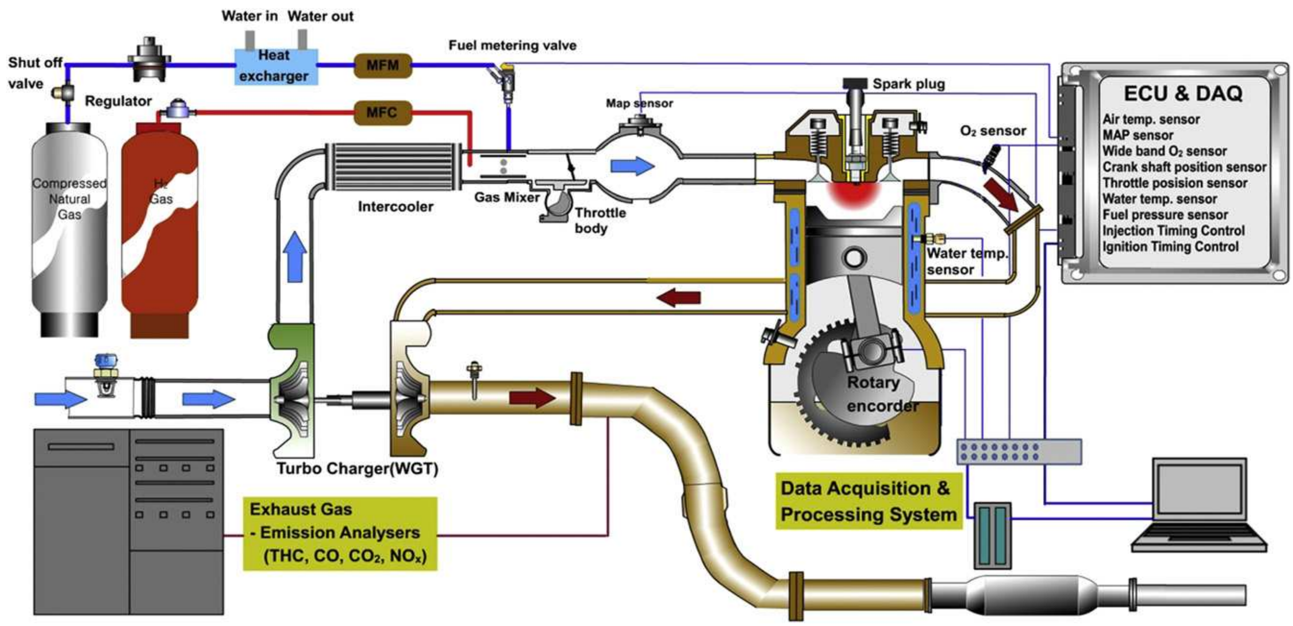

Fig. 1 – Schematic diagram of the experimental engine setup.

2.2. *Experimental methods*

The testing conditions were set at 1260 rpm, which is the engine revolution speed of maximum torque, and 570 Nm, which is half of the maximum torque, reflecting typical and frequently used operating conditions. After the engine had been sufficiently warmed up, the cooling water control system was set to maintain the cooling water temperature at $82.5 \pm 2.5\,^{\circ}C$. The excess air ratio was varied from $\lambda = 1.3$ to the lean burn limit by an increment of 0.1. Under each test condition, ignition time sweeps were carried out in order to determine the minimum advance for best torque (MBT) requirements. The thermal efficiency, exhaust emissions and combustion stability were analyzed for each fuel of different excess air ratio. One important criterion of combustion stability that can be represented by the cyclic variability derived from pressure data is the coefficient of variation (COV) for the indicated mean effective pressure (IMEP) [5]. This value is calculated from the standard deviation of IMEP divided by the mean IMEP and is usually expressed in percent.

The fuels for the test were natural gas and various HCNG (10–40 vol% hydrogen). The test conditions were determined by the manipulation of a throttle valve while maintaining constant torque and excess air ratio. The thermal efficiency and exhaust emissions for HCNG fuel were compared to those of a natural gas baseline.

Table 2 – The properties of natural gas.

Methane (CH_4)	91.332%
Ethane (C_2H_6)	5.363%
Propane (C_3H_8)	2.136%
Iso-butane (i-C_4H_{10})	0.459%
Normal-butane (n-C_4H_{10})	0.476%
Iso-pentane (i-C_5H_{12})	0.015%
Normal-pentane (n-C_5H_{12})	0.002%
Nitrogen (N_2)	0.217%
Higher heating value	43.54 MJ/Nm^3 (10,400 kcal/Nm^3)
Lower heating value	39.33 MJ/Nm^3 (9393 kcal/Nm^3)
Specific gravity	0.6169
Gas density	0.7976 kg/Nm^3

3. Results and discussion

In order to meet the improved efficiency and emissions standards using lean burn combustion, it is important to secure combustion stability in the lean operation condition so that the stable lean burn range is expanded. Unstable lean operation may cause deterioration in efficiency and an increase of unburned hydrocarbon emissions. The NO_x reduction will also be limited by the narrow flammability limit. As reported in previous research and in this study, the purpose of adding hydrogen to a natural gas engine is to take advantage of the wider lean burn characteristics of hydrogen. In this research, the lean burn characteristics were accessed according to the hydrogen addition ratio when the hydrogen addition was varied from 10 vol% to 40 vol%. The COV_{IMEP} values in Fig. 2 show changes in combustion stability with the hydrogen addition ratio in the fuel at the MBT spark timing of each operation condition. If we set the COV_{IMEP} value of 2% as the standard for an operable stable combustion, it is observed that the flammability limit is extended by the hydrogen addition.

Fig. 3 shows the thermal efficiencies versus excess air ratio as hydrogen was added. As the leanness was increased, the thermal efficiency increased; however, when the excess air ratio exceeded a certain level of leanness, the degree of

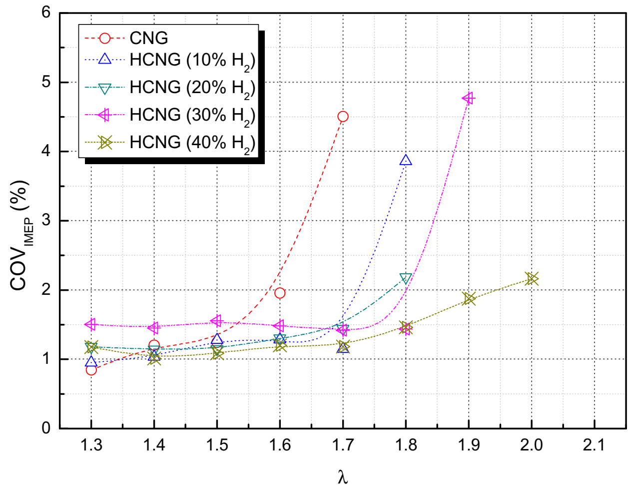

Fig. 2 – Variation in combustion stability versus excess air ratio at different blending ratios of hydrogen.

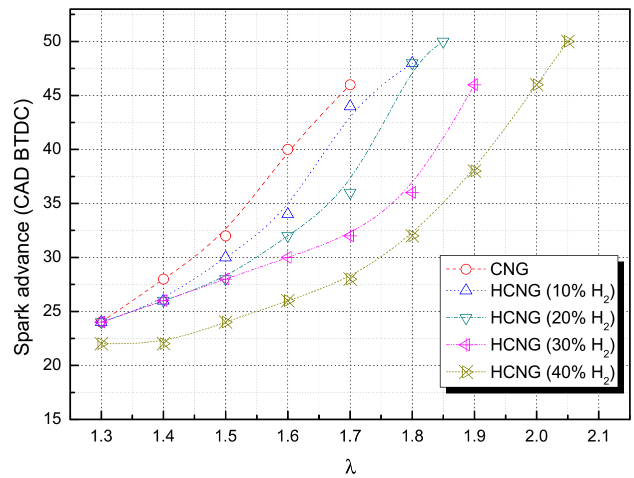

Fig. 4 – Variation in MBT timing versus excess air ratio at different blending ratios of hydrogen.

efficiency decreased due to the degradation in combustion stability. The excess air ratio of maximum efficiency for each hydrogen addition ratio also indicates a leaner condition with the increase of the hydrogen addition ratio. It is noticeable that the thermal efficiency of 40 vol% hydrogen addition was considerably lower, by 1.6%, than that of the natural gas alone. The reason why the hydrogen addition at certain excess air ratios was effective in improving thermal efficiency is the efficient expansion work due to the increased laminar flame speed of the mixture, as presented in Fig. 4, which shows the MBT spark advance timing at each operating condition. An increase of thermal efficiency is expected with an increase of the hydrogen addition ratio because the anti-knocking characteristics of hydrogen are beneficial to making a higher compression ratio possible as more hydrogen is added [13–15].

Figs. 5–7 show the results of the harmful emissions NO_x, THC and CO, respectively, as a function of changes in the excess air ratio under the same operating condition as in Fig. 3. As shown in Fig. 5 (enlarged scale for more detailed comparison), under certain excess air ratio conditions, the level of NO_x increased as hydrogen was added. This is because NO_x generation is promoted as the temperature of the combustion gas increases due to the high adiabatic flame temperature of hydrogen. On the other hand, if the conditions of best efficiency, indicated with a bold symbol in Fig. 5, are considered, NO_x emissions are likely to decrease since stable operation is possible with hydrogen addition under higher excess air ratio conditions. A comparison of NO_x emissions with natural gas only under the conditions of best efficiency indicated an approximately 67% reduction of the NO_x with the addition of 30 vol% hydrogen and an 84% NO_x reduction with 40 vol% hydrogen.

The characteristics of THC emissions are a little different from those of NO_x emissions, as shown in Fig. 6. Although THC emissions decreased with hydrogen addition at certain excess air ratio conditions, THC emissions are similar or higher at the operating condition of best efficiency (bold symbol in Fig. 5).

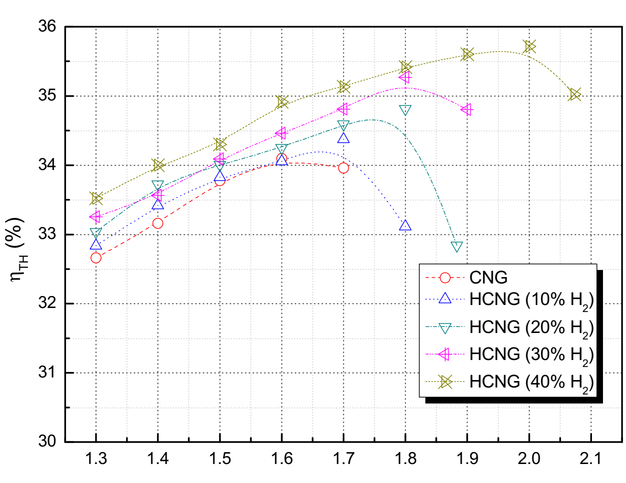

Fig. 3 – Variation in thermal efficiency versus excess air ratio at different blending ratios of hydrogen.

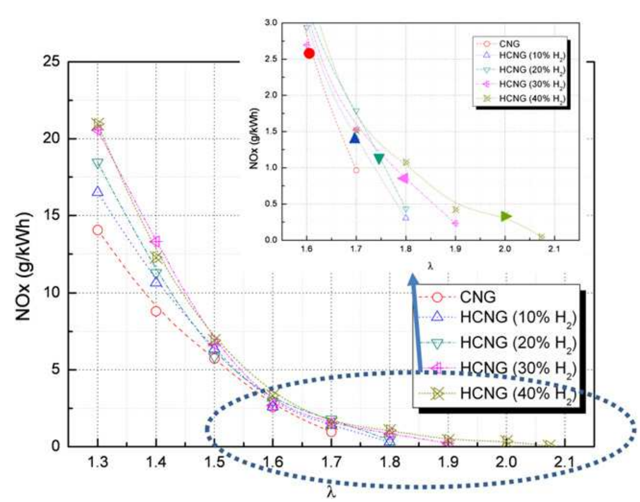

Fig. 5 – Variation in tail-pipe NO_x emissions versus excess air ratio at different blending ratios of hydrogen.

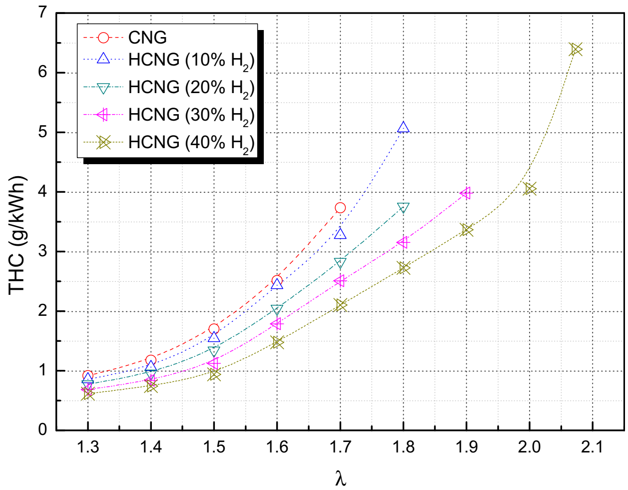

Fig. 6 – Variation in tail-pipe THC emissions versus excess air ratio at different blending ratios of hydrogen.

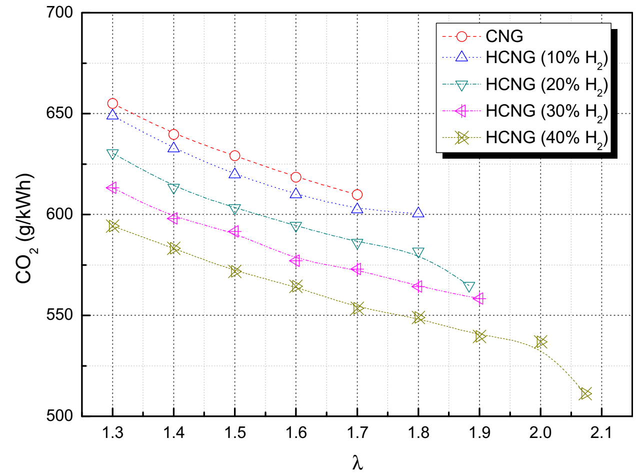

Fig. 8 – Variation in tail-pipe CO_2 emissions versus excess air ratio at different blending ratios of hydrogen.

Considering currently available aftertreatment system technologies, because oxidation catalysts are superior in price and performance to De-NO_x catalysts, it is believed that achieving NO_x reduction with lean combustion and THC reduction with oxidation catalysts is the most pertinent way to meet the emission regulations.

Fig. 7 describes the decrease of CO emissions as the hydrogen addition ratio is increased. With the addition of 30 vol% hydrogen, CO emissions were decreased by 15.7% compared with natural gas only and decreased by 16.2% with the addition of 40 vol% hydrogen. CO_2 emissions are also decreased with the increased hydrogen addition ratio, as shown in Fig. 8. The reason why CO and CO_2 emissions are reduced is the reduced carbon content in the fuel and the promoted combustion of the mixture as the hydrogen addition ratio is increased.

It was observed that combustion stability and thermal efficiency were improved and NO_x, CO, and CO_2 emissions were reduced without the increase of THC emissions as the hydrogen addition ratio was increased. However, the possible mileage per charging must be shortened with the original natural gas fuel tank because the increase in the hydrogen addition ratio leads to an increase in the volume flow rate of the fuel. Fig. 9 shows that, although thermal efficiency increases, the flow rate of the total fuel is increased because the calorific value per volume of hydrogen is low. The increase of the total fuel flow rate is 23.7% for the addition of 30 vol% hydrogen and 35.4% for the addition of 40 vol% hydrogen. When the total fuel flow rate increases, another concern related to the capacity requirement of the fuel supply system arises. The fuel regulator and injector are typical parts of the fuel supply system and their capacity should be assessed under the maximum power and the maximum torque conditions. Verification testing of the regulator and injector revealed that the regulator used for the city bus functions adequately, but that the capacity of the injector should be increased by 10% with the addition of 30 vol% hydrogen. For the addition of 40 vol% hydrogen, the capacity of the regulator should also be increased.

From the results above, a summary of the comparison under the operable maximum excess air ratio condition for

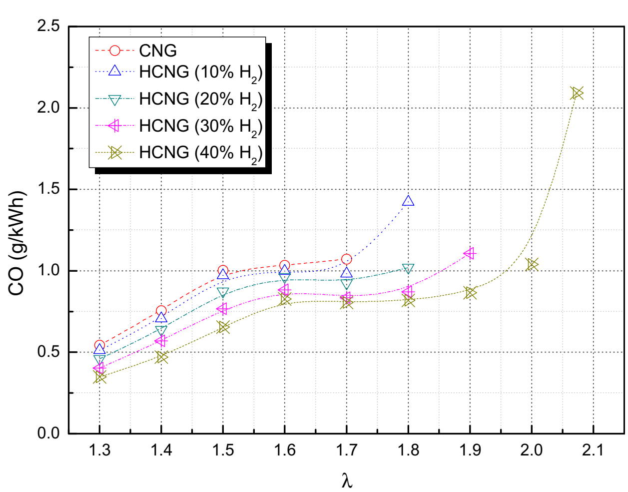

Fig. 7 – Variation in tail-pipe CO emissions versus excess air ratio at different blending ratios of hydrogen.

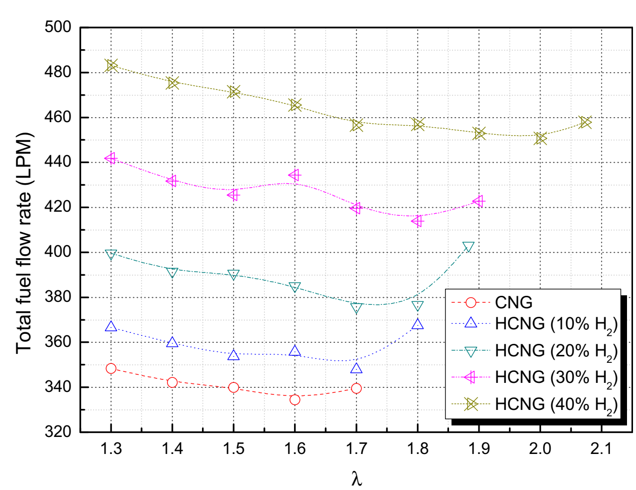

Fig. 9 – Variation in total fuel flow rate versus excess air ratio at different blending ratios of hydrogen.

each hydrogen addition ratio is described in Fig. 10. The thermal efficiency and emissions data are represented as values normalized to the standard data of natural gas alone. As the leanness of the burn increased, the thermal efficiency increased. NO_x, CO, and CO_2 emissions (but not THC emissions) are significantly decreased for leaner burns. When emissions regulations are tightened from EURO-V to EURO-VI, the most important factor to consider is NO_x emissions, for which the reduction level should be more than 80%. As shown in Fig. 10, the result of NO_x reduction for the addition of 20 vol % hydrogen is far short of the stricter emissions standards, and even with the addition of 30 vol% hydrogen NO_x, the emissions cannot satisfy the EURO-VI emissions standard of (0.4 g/kWh) at the maximum efficiency condition. Although the addition of 40 vol% hydrogen can allow the engine to meet the EURO-VI emissions standards, the reduction in the possible mileage per charging that results from adding that volume of hydrogen renders such a solution impractical.

Although NO_x reduction of over 80% could not be accomplished with the addition of 30 vol% hydrogen, it is possible through the sacrifice of the increased thermal efficiency by employing retarded spark timing. Fig. 11 shows the results of retarded spark timing with the addition of 30 vol% hydrogen. Although the effect of increased thermal efficiency disappears and the total fuel flow rate slightly increases, it is apparent that NO_x reduction of over 80% is possible through this approach. Retarding the spark timing offers the advantage of THC and CO reduction due to the promotion of after-burn in the cylinder. Therefore, when applying HCNG in practice, using retarded spark timing is more effective for the reduction of harmful emissions such as NO_x, THC, and CO than using MBT spark timing.

NMHC emissions should also be considered from the emission results. Fig. 10 shows that when the hydrogen addition ratio increases, NMHC emissions are further reduced than in the standard condition despite the relatively slight increase in THC emissions. This implies that the portion of methane in the THC emissions became greater as the hydrogen addition ratio increased. Because of the structural stability of methane, the oxidation treatment control of methane emissions from natural gas vehicles is more difficult than is the control of NMHC emissions. Many natural gas oxidation studies have been carried out at various operating conditions, ranging from investigations of the exact stoichiometric excess air ratio to the lean mixture limit [16,17]. The addition of hydrogen to natural gas results in an increase in the temperature of the combustion chamber as the fraction of hydrogen increases, leading to a reduction in NMHC [18,19]. Because methane is also regulated as a pollutant (GHG), and the relative increase of methane emissions with the addition of hydrogen is a shortcoming of the HCNG engine, supplementary technology should be sought to satisfy the next generation emissions standards. If oxidation catalyst systems, which presently contain a significant amount of precious metals such as platinum, can enable engines to meet emissions standards, the advantage of HCNG might disappear due to deteriorating price competitiveness. An improved oxidation catalyst system for the effective reduction of methane emissions should be developed.

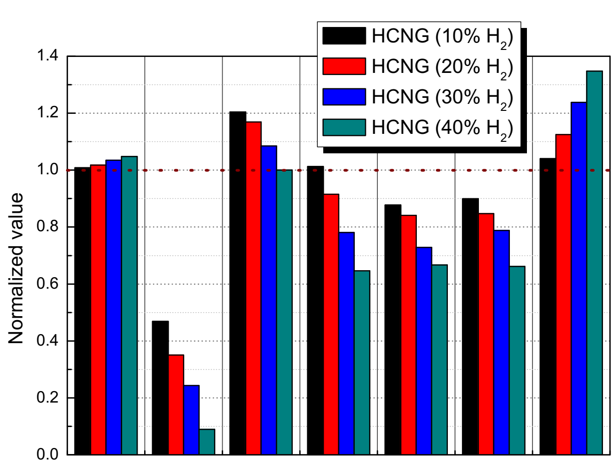

Fig. 10 – Comparison of lean burn characteristics as a function of the hydrogen fraction.

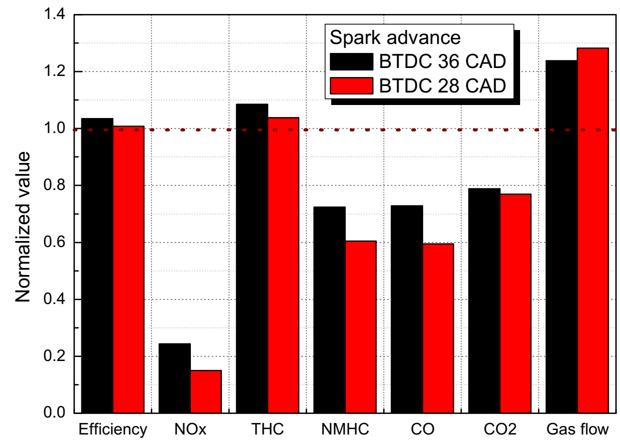

Fig. 11 – Improvement in exhaust emissions with retarded spark timing at 30 vol% blending ratio of hydrogen.

4. Conclusions

An experiment to obtain NO_x reduction was performed in an 11 L heavy duty natural gas engine through the addition of hydrogen to natural gas fuel. Several parameters were evaluated to develop lean burn engines that can operate in the field. To validate the applicability of hydrogen-blended fuel, the thermal efficiency and emissions characteristics were assessed.

Results from the experiments can be summarized as follows:

1) When hydrogen was added to natural gas, the lean burn limit was expanded to $\lambda = 2.0$. As the operation range approached the lean burn limit, efficiency tended to increase and then decrease after the maximum point was reached.
2) It was observed that combustion stability and thermal efficiency were improved and that NO_x, CO and CO_2 emissions were reduced without the increase of THC emissions

as the hydrogen addition ratio was increased. However, the possible mileage per charging must be shortened when using the original natural gas fuel tank because the increase in the hydrogen addition ratio leads to an increase in the volume flow rate of the fuel.

3) Although NO_x reduction of over 80% could not be accomplished with the addition of 30 vol% hydrogen, such a reduction is possible through the sacrifice of the increased thermal efficiency by retarding the spark timing. When applying HCNG in practice, using retarded spark timing is more effective for the reduction of harmful emissions such as NO_x, THC, and CO than using MBT spark timing.
4) Because methane is also regulated as a pollutant and because a relative increase of methane emissions accompanies the addition of hydrogen in an HCNG engine, a supplemental method of reducing emissions should be sought to satisfy emissions standards. If oxidation catalyst systems, which presently contain a significant amount of precious metals such as platinum, can enable engines to meet emissions standards, the advantage of HCNG might disappear due to deteriorating price competitiveness.

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