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Hydrogen effects on the combustion stability, performance and emissions of a turbo gasoline direct injection engine in various air/fuel ratios

Joonsuk Kim[a], Kwang Min Chun[b], Soonho Song[b,*], Hong-Kil Baek[c], Seung Woo Lee[c]

[a] The Graduate School, Department of Mechanical Engineering, Yonsei University, 50 Yonsei-ro, Seodaemun-gu, Seoul 03722, Republic of Korea
[b] Department of Mechanical Engineering, Yonsei University, 50 Yonsei-ro, Seodaemun-gu, Seoul 03722, Republic of Korea
[c] Hyundai Motor Company, 150 Hyundaiyeonguso-ro, Namyang-eup, Hwaseong-si, Gyeonggi-do 18280, Republic of Korea

HIGHLIGHTS

- H_2 effects on performance of T-GDI engine with various air/fuel ratios are studied.
- Spark timing could be retarded on MBT zone and could be advanced on knock zone.
- With H_2, CA0-50 was improved on all, but CA50-90 worsened on BMEP 8 to 16 bar.
- Variations of cylinder-to-cylinder and cycle-to-cycle are reduced by H_2 addition.
- Thermal efficiency is improved by H_2 addition but optimum conditions differed.

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ABSTRACT

Lean combustion has been identified as a way of improving the thermal efficiency of gasoline engines. However, it is difficult to operate gasoline engines in lean conditions, because the stability of the combustion deteriorates. Hydrogen has properties that can be exploited to improve the combustion stability, and hence solve the combustion instability problem. In this paper, we describe an experimental study into the effects of hydrogen on the combustion stability, performance, and emissions at various air/fuel ratios using a four-cylinder 2.0-L turbo gasoline direct-ignition engine. The engine speed was fixed at 2000 RPM with a brake mean effective pressure of 2–18 bar. The results of these experiments indicate that combustion is improved, thermal efficiency enhanced, and emissions reduced except for NO_X. Especially, in terms of combustion stability, the cycle-to-cycle variation and cylinder-to-cylinder variation decreased when hydrogen was added. This was caused by two effects. First, when the brake mean effective pressure was 10 bar or below, the burn duration was reduced by the fast flame speed, high adiabatic temperature, and high diffusion coefficient of hydrogen. This led to an increase in the combustion stability and thermal efficiency. Second, when the brake mean effective pressure was above 10 bar, knock resistance was enhanced by hydrogen: spark timing could be advanced upon hydrogen addition. Therefore, the combustion stability and performance improved. Additionally, the improvements to the combustion, especially in terms of combustion stability, became more pronounced as the in-cylinder air–fuel mixture became leaner: the lean limit was thus extended.

Abbreviations: MBT, maximum brake torque; ITE, indicated thermal efficiency; BMEP, brake mean effective pressure; IMEP, indicated mean effective pressure; T-GDI, Turbo Gasoline Direct Injection; LNT, Lean NO_X Trap; SCR, Selective Catalytic Reduction; TWC, Three Way Catalyst; DECU, Developed Electronic Control Unit; $m'_{gasoline}$, gasoline mass flow; m'_{H_2}, hydrogen mass flow; m'_{air}, intake air mass flow; $AF_{st,gasoline}$, stoichiometric air fuel ratio of gasoline; AF_{st,H_2}, stoichiometric air fuel ratio of hydrogen; $Q_{LHV,gasoline}$, low heating value of gasoline; Q_{LHV,H_2}, low heating value of hydrogen; $\dot{W}_{ind}$, indicated work output of the engine; CA, crank angle; CA0-10, CA duration of 010% heat release; CA0-50, CA duration of 050% heat release; CA10-50, CA duration of 1050% heat release; CA50-90, CA duration of 5090% heat release; CA10-90, CA duration of 1090% heat release; ATDC, after top dead center; COV, coefficient of variation; HC, hydro carbon; NO_x, nitrogen oxides; CO, carbon monoxide; CO_2, carbon dioxide

* Corresponding author.
E-mail address: soonhosong@yonsei.ac.kr (S. Song).

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1. Introduction

Globally, emissions and fuel economy regulations are being strengthened. Lean combustion significantly increases thermal efficiency [1–3] and reduces raw emissions of gasoline engines. However, combustion stability deteriorates under lean conditions. In addition, an exhaust after-treatment device such as an LNT or SCR is required when the engine operates under lean conditions to reduce NO_X emissions because TWC is only operable on stoichiometry. These concerns constitute barriers to the use of lean-combustion gasoline engines [1].

Hydrogen has received much attention for use in fuel cells [4–7], hydrogen engines [8,9], and as an additive [10,11]. Due to its unique properties, hydrogen is more suitable for use in spark-ignition (SI) engines than compression-ignition (CI) engines [12,13]. For example, the adiabatic flame speed of hydrogen is much faster than that of gasoline, which improves the combustion stability and thermal efficiency of the engine [13,14]. Additionally, the diffusion coefficient of hydrogen is much larger than that of gasoline, which leads to a more homogeneous air–fuel mixture; also, the wide flammability of hydrogen facilitates engine operation in lean conditions [15]. Moreover, the high ignition temperature of hydrogen is higher than that of gasoline, which reduces knocking. The properties of hydrogen and gasoline are compared in Table 1 [16].

There are many publications detailing the effect of hydrogen on SI engines. For ethanol engines, Yousufuddin investigated the performance and combustion characteristics of a hydrogen-ethanol fueled engine [17]. For methanol engines, Changmin investigated hydrogen effects on a DISI methanol engine [18], and Zhang studied hydrogen effects on a hydrogen-methanol engine in terms of loads [19], spark timings [20] and cold start characteristics [21]. For natural gas engines, Liu studied hydrogen effects on a natural gas engine in terms of emissions [22,23], performance [23], developing a combustion model [24], thermal efficiency [25] and cycle-by-cycle variations [26], and Navarro investigated carbon dioxide emissions of a natural gas-hydrogen engine [27]. For gasoline engines, Wang investigated the effect of hydrogen in terms of the idle performance of a gasoline engine [16], the fixed manifold absolute pressure (MAP) under lean conditions on 1400 RPM [15], with a wide-open throttle under lean conditions [28], and various operating conditions [29]. Yang studied the combustion processes of a gasoline rotary engine with hydrogen [30]. D'Andrea studied the effect of various engine speeds and equivalence ratios on combustion in a hydrogen-blended gasoline engine [31]. The results of these experiments showed that hydrogen has a positive effect on combustion in gasoline engines. After the addition of hydrogen, the cycle-to-cycle variation decreased and the combustion speed increased.

Previous studies had certain limitations. First, combustion characteristics against engine load of turbo gasoline direct injection (T-GDI) engines are different from that of natural aspirated (N/A) engines. In addition, T-GDI engines operate on high mean effective pressure when conventional N/A engines are not able to be operated. It means that an influence of hydrogen addition on T-GDI engines is different from that on conventional N/A engines. However, most experiments employed N/A or port-fuel injection engines. A few recent works evaluated T-GDI engines, but most did not consider the boosting zone which was unique characteristic of T-GDI engines. Second, the previous studies were conducted at specified engine loads; thus, a study in which the engine load is varied from low to high would be informative. Third, little attention has been paid to the cylinder-to-cylinder variation. The cylinder-to-cylinder variation is as important as the cycle-to-cycle variation because there are wide variations in cylinders under lean conditions. Therefore, the effects of hydrogen as an additive to T-GDI engines under low-to-high loads, including the impact on boosting zone and the cylinder-to-cylinder variation, should be studied.

In this study, we examined the effect of adding hydrogen to a four-cylinder 2.0-L T-GDI engine operating under stoichiometric-to-lean conditions at 2000 RPM with a BMEP of 2 to 18 bar. The maximum BMEP was 18 bar of engine output torque at the lean limit. Hydrogen was added to the front of an intercooler, and the hydrogen flow rate was controlled using a mass flow controller.

Table 1
Properties of hydrogen and gasoline.

Properties	Hydrogen	Gasoline
Molecular weight	2.015	110
Stoichiometric A/F ratio	34.3	14.6
Ignition temperature (K)	858	530
Adiabatic flame temperature (K)	2384	2270
Flame speed at 293 K (cm/s)	237	41.5
Limits of flammability (vol% in air)	4.1–75	1.5–7.6
Quenching gap (cm)	0.06	0.2
Lower heating value (MJ/kg)	120	44
Diffusion coefficient at stoichiometric condition (cm^2/s)	0.61	0.05

2. Equipment and condition

2.1. Equipment

A schematic diagram of the experimental setup is shown in Fig. 1.

The test engine was manufactured by the Hyundai Motor Company. The specifications of the engine are as follows: 2.0 L, twin scroll turbo charged, side-mounted direct gasoline fuel injection using a solenoid injector, dual variable-valve timing, and four cylinders. Additional specifications are listed in Table 2.

The engine speed was controlled by an eddy current dynamometer (accuracy: ± 0.5% of the full scale). Kistler pressure sensors 6041A were installed on all cylinders. A Kistler combustion analyzer Ki-Box (accuracy: ± 1% of the full scale) was used to analyze the combustion characteristics, based on signals from the crank position sensors and pressure sensors. The engine was equipped with an electronic control unit (DECU) for parameter modification. Lambda in this study was calculated using Eq. (1). The oxygen sensor was installed on exhaust line to obtain the lambda value in real time.

$$\text{Lambda} = \frac{m'_{air}}{m'_{gasoline} \times AF_{st,gasoline} + m'_{H_2} \times AF_{st,H_2}} \tag{1}$$

A gas analyzer, Horiba MEXA9100DEGR (accuracy, ± 1% of the full scale), detected NO_X, CO_2, CO, HC, and O_2 exhaust emissions. NO_X was measured by chemiluminescent detection method, CO_2 and CO were monitored by non-dispersive infrared (NDIR) method, HC was monitored by hydrogen flame ionization detection (FID) method, and O_2 was monitored by magneto-pneumatic detection (MPD) method. Hydrogen from a high-pressure pure hydrogen tank was added to the front of the intercooler. The hydrogen flow rate was controlled using a mass flow controller (accuracy: ± 2% of the full scale). A hydrogen volume fraction analyzer was installed at the outlet of the intercooler to verify the hydrogen intake flow rate.

The fraction of hydrogen energy was calculated using Eq. (2).

$$\text{Hydrogen energy fraction (\%)} = \frac{m'_{H_2} \times Q_{LHV,H_2}}{m'_{gasoline} \times Q_{LHV,gasoline} + m'_{H_2} \times Q_{LHV,H_2}} \times 100 \tag{2}$$

2.2. Experimental conditions

The engine was operated at 2000 RPM with a BMEP of 2 to 18 bar at 2-bar intervals, because 2000 RPM is the common driving condition. The maximum torque for measuring the lean limit occurred at BMEP 18 bar: this was the maximum engine output torque at 2000 RPM at the lean limit with a wide-open throttle under maximum boost pressure. We postulated in future, an on-board reformer would supply hydrogen.