Improving cold start, combustion and emission characteristics of a lean burn spark ignition natural gas engine with multi-point hydrogen injection

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Abstract

The cold start, combustion and emission characteristics of a four-cylinder spark ignition (SI) natural gas (CNG) engine were optimized using multi-point hydrogen gas (H_2) injection, exhaust gas recirculation (EGR), and an oxidation catalyst (OC). The transient HC emission was also collected with Fast FID measurement within the period of cold start. The results indicate that multi-point H_2 injection could improve the cold start performance of CNG SI engine. The peak cylinder pressure and HC emissions maintain stable quickly with hydrogen addition in the stage of cold start. The maximum in-cylinder pressure and the heat release rate increased linearly with the H_2 injection duration. Multi-point H_2 injection shortens the flame development duration and the combustion duration. The coefficient of variation (COV) of the indicated mean effective pressure (IMEP) increased with the EGR rate and decreased with H_2 multi-point injection. Multi-point H_2 injection can improve EGR tolerance for CNG engine. The OC could effectively reduce CO emissions for all of the different conditions, but the conversion efficiency for the HC emissions was insufficient. The combination of lean burn combustion, EGR, multi-point H_2 injection, and a more effective OC is a potential pathway for CNG engine optimization.

Keywords: Natural gas; Hydrogen; Multi-point injection; Cold start; Cycle-to-cycle variation; Emission;

1. Introduction

Due to the energy security, the search for alternative fuels for use in the internal combustion engine, such as biodiesel, natural gas, ethanol, dimethyl ether, hydrogen gas (H_2) etc., is a hot topic for researchers. Natural gas has potential for transportation applications [1], because of its reduced environmental burden relative to conventional fuels and its applicability to conventional diesel (dual fuel) [2] and gasoline engines (port or direct injection) [3, 4]. Moreover, the use of natural gas could result in lower nitrogen oxides (NO_x), carbon monoxide (CO), and non-methane hydrocarbon (HC) emission levels, by using a simple after-treatment device, and it is almost free of particulate emissions [5]. Because of the high octane number (between 120 and 130) and high anti-knocking potential of natural gas, CNG engine has high tolerance for high compression ratios [6]. However, due to the lower flame velocity and the narrower flammability limits of natural gas [7], a natural gas spark ignition engine has the weaknesses, such as large cycle-to-cycle variations, poor lean-burn capability, and poor exhaust gas recirculation (EGR) tolerance, which could decrease the engine power output and the thermal efficiency and increase fuel

consumption. Lean burn natural gas engines have become more attractive as they are highly efficient due to the lower heat and pumping losses and the excess air. In addition, lean burn combustion reduces the flame speed, prolongs the burn duration, and lowers the peak combustion temperatures in cylinder. The lower combustion temperature in cylinder could suppress the heat transfer, leading to lower NO_x [8]. However, because of the poor lean burn capability of CNG, many obstacles, such as the lean burn limit and methane slip, still exist during the application of this mode in a natural gas engine.

H₂ gas is considered to be an energy carrier that can be produced from many sources. As it is carbon-free, and thus burns without emitting any harmful emissions other than water vapor, it has potential as a fuel in several respects. Its broader flammability limits, higher flame velocity, and smaller quenching distance allow it to burn in more dilute conditions [9-10]. Hydrogen also improves flame propagation, leading to higher engine efficiency and lower fuel consumption. Unfortunately, its low ignition energy makes stoichiometric air-fuel mixtures prone to pre-ignition and even backfire [11-13].

A combination of natural gas and H₂ could be applied to avoid these problems. The features of CNG and H₂ complement each other, potentially improving the lean burn combustion and the emission characteristics. Many studies have been carried out on

spark ignition engine fueled with H₂-CNG blended fuel [13-18]. The results showed that the maximum cylinder pressure rose, while CNG consumption rate decreased with hydrogen-CNG blend. Because of the properties of hydrogen, leaner fuel-air mixtures combined with proper ignition timing can be used to improve engine emissions with no loss of performance. Moreover, except port injection, direct injection of methane and hydrogen and premixed hydrogen-methane blend with port injection was also studied in a spark ignition engine [19]. The results showed that lean limits depend on direct injection direction and timing. Direct injection of comparably small amounts of methane and hydrogen has the similar effect to the lean limit extension.

However, in these studies, the H₂ and natural gas were premixed in a bottle or mixer before the experiment and supplied using the same fuel system. There is almost no literature on multi-point H₂ injection used with a natural gas engine. In consideration

of lack of research results on the cold-start performance for natural gas engine with multi-point H_2 injection, the present work tried to be complementary in this issue. Moreover, natural gas engines suffer from methane slip, i.e. the release of a small amount of unburned methane into the exhaust. Although methane oxidation is difficult and limited in the after stream of the exhaust, the combination of multi-point H_2 injection, EGR, and an oxidation catalyst (OC) could have potential for the optimization of combustion and emissions in a SI CNG engine.

2. Experimental

A four-cylinder SI CNG engine coupled with an eddy-current dynamometer was used in this study. The experimental setup is shown in Figure 1. The engine parameters are given in Table 1. The CNG was supplied to the six natural gas injectors installed in the intake manifold via a pressure reducer from the CNG bottles. The injection parameters, such as the injection duration and timing, were set by an electronic unit. A H_2 common rail and four injectors were used to inject the H_2 from multiple points near the intake valve, with another electronic unit controlling the injection timing and duration. All experiments were conducted at 1500 r/min and 323 Nm. To

maintain a stable H_2 /fuel ratio, the H_2 injection system was calibrated before the experiment. Four H_2 injection durations were set for the multiple point injection, 1.2 ms, 2 ms, 2.5 ms, and 3 ms, corresponding to H_2 /fuel ratios of 5%, 12%, 15.5%, and 19%, respectively. The fuel properties are given in Table 2. The composition of natural gas used in this study is given in Table 3.

A shaft encoder, a Kistler piezoelectric transducer (6118B), and a Kistler charge amplifier (5115) were used to transmit the crank angle and in-cylinder pressure to the D2T combustion analyzer (Evolution 3). The piezoelectric transducer was mounted in position on the spark plug. The cylinder pressure was recorded every 0.5 CA. CO, carbon dioxide, oxygen, unburned HC, and NO_x were analyzed by California Analyze Instruments CAI600 (NIDR, HFID, HCLD). The hydrocarbon emissions of cold-start were measured with Fast FID HC analyzer (CAMBUSTION HFR500). A self-developed procedure with Labview Software was used for emission data acquisition with a data acquisition card. Also, a commercial OC, combined with a noise silencer supplied by SDEC, was used. The OC is a flow-through canister device, which is filled with honeycomb substrate and active catalyst layer on the surface of

substrate. The physical parameters of the OC are as follows, the monolith length was 100 mm, the cell density was 400 CPSI, the substrate material was cordierite, and the active metals were platinum (Pt) and palladium (Pd) (Pt:Pd = 1:5). The standard volume of the OC was 5.76 L. The cold start test was based on standard test procedure of China (HJ 689-2014). The test was conducted in cold start engine room equipped with air cooling system, water cooling system and lube oil cooling system. Each cold start test was conducted after 6 hours stationary cooling period to ensure the temperature of circulation water and lube oil maintain 20°C -30°C. And the ambient temperature was set at 20°C to ensure experimental condition stable and repeatable. The accuracy and experimental uncertainty have been given in Table 4 by the classic method [21-22]. Student's t-test was employed to analyze whether the difference between the results obtained from different conditions are statistically significant at 95% confidence level.

3. Results and discussion

3.1 Cold start performance

Fig. 2 shows the engine speed within 150 s after cold start with different H₂ injection

durations. The engine speeds are increased within first two seconds and then decreased continuously until maintaining at idle speed. As seen in the figure, the peak speed rises with increasing of H_2 injection during cold start period, because of the higher laminar flame speed of H_2 . While, after maintaining idle mode, the 15.5% H_2 injection results in lower and more stable idle speed than those of pure CNG and 5% H_2 injection. The variations of peak cylinder pressure within first 100 cycles are plotted in Figure 3. According to the figure, the peak cylinder pressure increases and maintains stable quickly with hydrogen addition, which corresponds to the variation of HC emissions detected with fast-FID in Fig. 3. It can be shown that HC emissions within the first 10 cycles of pure CNG are higher than that of H_2 injection. Moreover, HC emission of pure CNG seems to be stabilized after 60 cycles, while the HC emission of 15.5% H_2 injection appears to be stabilized after only 40 cycles. These results confirm that multi-point H_2 injection could improve the cold start performance of CNG SI engine.

3.2 Combustion analysis

The effect of multi-point H_2 injection on combustion characteristics measured at

1500 rpm and 323 Nm with 12% EGR is depicted in Figure 4. As seen in the figure, the maximum heat release rate and in-cylinder combustion pressure increased linearly with the H₂ injection duration. The flame development duration, from the spark plug timing to CA10, increased continuously with the EGR rate at various H₂ injection durations, as seen in Figure 5. For a specific EGR rate, multi-injection H₂ shortened the flame development duration, due to the higher laminar flame speed of H₂ than that of methane (CH₄) [24]. The combustion duration is also shown in Figure 5. For both operating conditions, the combustion duration, calculated from CA10 to CA90, decreased with H₂ injection duration. The slow reaction $\text{CH}_4 + \text{O}^*$ to $\text{CH}_3^* + \text{OH}^*$ of CH₄ results in lower laminar flame speed [9]. The higher concentration of OH* radicals due to the addition of H₂ enhanced the removal of the first hydrogen atom from the methane molecule, leading to a shorter and faster combustion than that of natural gas [25-26]. Meanwhile, the combustion process of H₂ injection was phased closer to the top dead center, suggesting a faster combustion process, leading to a higher brake thermal efficiency, as shown in Figure. 5. The brake thermal efficiency decreased significantly with EGR rate. The multi-point H₂ injection could

increase the thermal efficiency at certain EGR rates and increase the EGR tolerance of the natural gas engine. For instance, when the EGR rate was higher than 5%, the brake thermal efficiency fell off for the 1.2 ms H₂ injection duration. While for the 3 ms duration, the brake thermal efficiency began to decay, with an EGR rate higher than 12.5%.

3.3 Cycle-to-cycle variations

Combustion stability is an important parameter used to evaluate combustion performance. The effects of the multi-point H₂ injection on cycle-to-cycle variations were studied. Figure 6 shows the IMEP of 200 cycles with different EGR and multi-point H₂ injection conditions. As seen in the figure, compared with the 0% EGR rate, an IMEP of 200 cycles of 12% EGR rate distributed in a wider range. After using multi-point H₂ injection, the points of 200 cycles IMEP became more concentrated.

The coefficient of variation of the IMEP, COV_{IMEP} , calculated using Equation 1, which is an important index of cyclic variability, is shown in Figure 7.

$$COV = \frac{\sigma_{IMEP}}{\overline{IMEP}} \times 100 \quad (1)$$

where $\overline{IMEP}$ is the average IMEP for cycles, n , while σ_{IMEP} is the standard

deviation.

$$IMEP = \frac{\sum_{i=1}^{i=n} IMEP(i)}{n} \quad (2)$$

$$\sigma_{IMEP} = \sqrt{\frac{1}{n-1} \sum_{i=1}^{i=n} (IMEP(i) - \overline{IMEP})^2} \quad (3)$$

The COV_{IMEP} increased with the EGR rate increase due to misfire occurrence. The COV_{IMEP} can be reduced with H_2 multi-point injection. The unstable combustion process is defined as a COV_{IMEP} higher than 5%. For the 1.2 ms H_2 injection duration, the stable EGR operation range ($COV_{IMEP} < 5\%$) was from 0% to 12.5%, while for the 3 ms H_2 injection duration it was 0% to 18%, implying that there is higher EGR tolerance for multi-point H_2 injection. The extension of the multi-point H_2 injection duration increased the engine tolerance to the EGR dilution from 12.5% to 18%.

The ignition energy of H_2 is only 1/15 of that of CH_4 in normal temperature and pressure air [11]. The lower ignition energy and the higher laminar flame speed of H_2 could accelerate combustion process, shorten flame development duration and combustion duration, and stabilize the kernel of flame, leading to the lower COV_{IMEP} . Meanwhile, the high flame diffusion velocity of H_2 can improve the fuel-air mixture homogeneity, and then reduce the occurrences of misfire or partial burning cycles

[27], which results in lower cycle-to-cycle variations compared with pure natural gas.

On the other hand, H^* radicals are key factors for the flame stability during in-cylinder combustion, because of the chain branching reaction $H^* + O_2 \leftrightarrow OH^* + O^*$ [9]. H_2 addition could increase the concentration of radical pool, more H^* , O^* , and OH^* [28], confirming the lower COV for multi-point H_2 injection compared to pure natural gas. Schefer [26] studied lean flame stability in premixed, swirl-stabilized flames and reported that after adding 20% H_2 to methane, OH^* concentration could increase nearly 20% and lean stability limit could decrease nearly 15%, resulting in stable burner operation.

3.4 Emission analysis

Figure 8 shows the NO_x emissions versus the EGR rate and H_2 injection duration. In all cases tested, the NO_x emissions increased with H_2 injection and decreased with the EGR rate. Due to the higher adiabatic temperature of the H_2 [29], multi-point H_2 injection could improve combustion and increase the combustion temperature, leading to the increase in NO_x emissions.

The CO emissions versus the EGR rate and the H_2 injection duration are presented in

Figure 9. Although the CO emissions decreased slightly at EGR rates of 0% to 10%, overall, they increased with EGR. Multi-point H_2 injection could reduce the CO emissions, especially at a high EGR rate, because of the lower carbon atom content.

The HC emission characteristics are also presented in Fig.10. HC emissions remain stable at the 0% to 12% EGR rates and then increase rapidly with an EGR rate higher than 12%. With increased H_2 injection duration, the HC emissions decreased at certain EGR rates. Moreover, the catastrophe point of EGR for the HC emissions moved towards the higher EGR rate region, because of the higher homogeneity of the fuel-air mixture, the increased stability of combustion, and the shorter quenching distance of H_2 compared with that of methane, leading to more complete combustion, especially close to the combustion chamber wall [30]. Moreover, H_2 injection could replace some of the natural gas, resulting in the lower C/H ratio of the blended fuel, leading to the lower HC emissions. On the other hand, due to the lower H_2 ratio used in this study, fuel mixture in cylinder contains a higher methane mole fraction relative to hydrogen and thus methyl radical chemistry will tend to dominate. During the oxidation of methane, hydroperoxyl radical chemistry (Equation 4) is

significant because of HO_2^* formation (Equation 5) [31]. H_2 in combustion chamber could improve reactivity due to the formation of H^* and OH^* (Equation 6), and accelerate methane oxidation and reduce HC emissions.

The effects of the OC are also shown in Fig.9 and 10. The CO emissions could be almost totally removed by the OC, while the reduction of the HC emissions only ranged from 0% to 25%. Due to the lean burn combustion, unburned methane is a predominating constituent of the hydrocarbon emissions. Among alkanes, methane is the least kinetically reactive molecule [32], resulting in the lower HC reduction by the OC. On the other hand, the inlet temperature of OC drops significantly because of the space limitation, leading to the lower reduction efficiency for HC emissions. Thus, the performance of the OC could be improved by increasing the OC entry temperature and the formula design of the catalyst.

3.5 Optimization of emissions, thermal efficiency and coefficient of variation

Figure 11 shows the optimization for NO_x and HC emission. The original equivalent ratio was 0.85 for 1500 rpm and 323 Nm. When using lean burn combustion, NO_x emissions could be greatly reduced, with a 0.73 equivalent ratio. EGR could then suppress the NO_x emissions further to a low level. Due to the considerable HC emission increase for lean burn combustion, EGR, multi-point H₂ injection, and an OC was used to reduce them. Although the HC emissions could be reduced by only 26.7% using multi-point H₂ injection, cyclic variability and thermal efficiency were improved (Figure 12). The OC could reduce the HC emissions further, although not sufficiently. Multi-point H₂ injection has the potential to change the H₂/CNG ratio on-line flexibly at different engine conditions, further improving engine combustion and emission performance. Meanwhile, the catalyst for methane oxidation needs further study if a lean burn combustion mode is to be used in the CNG engine. The combination of lean burn combustion, EGR, multi-point H₂ injection, and a more effective OC is a potential pathway for CNG engine optimization. The flexible use of a multi-point H₂ injection system offers a method to take the advantages of lean-burn CNG SI engine, while avoiding the drawbacks of using pure CNG or H₂.

4. Conclusion

The effects of multi-point H₂ injection system on cold start, combustion and emission characteristics of a four-cylinder CNG SI engine were studied. The following conclusions can be drawn.

1). Multi-point H₂ injection could improve the cold start performance and stability of CNG SI engine. The peak speed rises with increasing of H₂ injection during the cold start period. While, after reaching idle mode, the 15.5% H₂ injection results in lower and more stable idle speed. The peak cylinder pressure and HC emissions become stable quickly with hydrogen addition within cold start.

2). The multi-point H₂ injection shortens the flame development duration and the combustion duration. The heat release of the CNG engine becomes a faster and more centralized combustion process, leading to a higher brake thermal efficiency after multi-point H₂ injection. The COV_{IMEP} increase with the EGR rate, but decrease with H₂ multi-point injection. Multi-point H₂ injection can improve EGR tolerance for CNG engine.

3). Multi-point H₂ injection could reduce CO and HC emissions. The OC could

effectively reduce CO emissions, while the conversion efficiency for HC emissions was far from enough, due to the low reactivity of methane and the lower entry temperature of the OC.

4). Multi-point H₂ injection has the potential to change the H₂/CNG ratio on-line flexibly for different engine conditions, further improving the engine combustion and emission performance. The combination of lean burn combustion, EGR, multi-point H₂ injection, and a more effective OC is a potential pathway for CNG engine optimization.

Acknowledgements

The authors would like to thank the National Science Foundation of China (Project No. 51306117; 51436005), the Ministry of Science and Technology (MOST) of China (Project No. 2012BAD47B04), and the Foundation of Shanghai Jiao Tong University for supporting this research.

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Figure captions

Fig. 1. Experimental setup

Fig. 2. Engine speed versus time of cold start

Fig. 3. Maximum cylinder pressure and Fast FID HC emission for the first 100 cycles of cold start

Fig. 4. In-cylinder pressure and heat release rate

Fig. 5. Flame development duration, combustion duration and brake thermal efficiency

Fig. 6. IMEP of 200 cycles

Fig. 7. Coefficient of variation in IMEP

Fig. 8. NO_x emissions

Fig. 9. CO emissions

Fig. 10. HC emissions

Fig. 11. Optimization of NO_x and HC emissions

Fig. 12. Optimization of cyclic variability and thermal efficiency

Table 1 Engine specifications

Model	SC5DT180Q4
Number of cylinders	4
Suction type	Turbo intercooler
Maximum power	135 kW/2300 rev/min
Maximum torque	645 Nm /1500 rev/min
Bore x stroke	114mm x 130 mm
Displacement	5300cc
Compression ratio	11: 1
CNG Injection type	Single point injection
Ignition timing	20°CA BTDC

Table 2. Test fuel properties

Properties	CNG	Hydrogen
Relative air–fuel ratio (Stoichiometric)	17.2	34.3
Density (kg/m ³) @ stp	0.748	0.085
Octane number	130	<130
Lower calorific value (MJ/kg)	48.6	120
Auto-ignition temperature (°C)	650 ^[20]	536
Laminar burning velocity (cm/s) @stp	37–45	265–325

Table 3. Composition of natural gas used in this study

CH ₄	C ₂ H ₆	C ₃ H ₈	n-C ₄ H ₁₀	CO ₂	Others
96.04%	2.58%	0.56%	0.23%	0.41%	0.18%

Table 4. The features of instruments

	Type	Accuracy	Uncertainty (95% confidence level)
Natural gas mass flow	Emerson CMF025	0.35%	5.7%
CO	CAI NDIR 602P	1%	3.12%
NOx	CAI HCLD 600	1%	2.74%
THC	CAI HFID 600	1%	2.74%
Fast FID (HC)	CAMBUSTION HFR500	1%	6.32%

Figure(s)

Figure(2)

Figure(3)

Figure(4)

Figure(5)

Figure(6)

Figure(7)

Figure(8)

Figure(9)

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