

Experimental Investigation of Homogeneous Charge Induced Ignition (HCII) with Low-Pressure Injection to Reduce PM Emissions in a Heavy-Duty Engine

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ABSTRACT

Homogeneous Charge Induced Ignition (HCII) combustion utilizes a port injection of high-volatile fuel to form a homogeneous charge and a direct injection of high ignitable fuel near the Top Dead Center (TDC) to trigger combustion. Compared to Conventional Diesel Combustion (CDC) with high injection pressures, HCII has the potential to achieve diesel-like thermal efficiency with significant reductions in NO_x and PM emissions with relatively low-pressure injections, which would benefit the engine cost saving remarkably. In the first part of current investigation, experiments were conducted at medium load with single diesel injection strategy. HCII exhibited great potential of using low injection pressures to achieve low soot emissions. But the engine load for HCII was limited by high heat release rate. Thus, in the second and third part, experiments were performed at high and low load with double diesel injection strategy. At high load, HCII with relatively low injection pressures (600-1000 bar) could obtain the same level of soot emissions as CDC with relatively high injection pressures (1200-1400 bar). The soot reduction could be over 50%. The particle number-size distribution analyses of PM emissions illustrated that the accumulation mode particulates were significantly reduced in HCII. Compared to CDC, indicated thermal efficiency for HCII was improved about 1-2% at high engine load. In addition, HCII could achieve Low Temperature Combustion (LTC) when combined with EGR, especially at low load. NO_x emissions for HCII increased slightly at medium and high load, but decreased considerably at low load.

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INTRODUCTION

Driven by the global concern over energy saving and environmental protection, advanced combustion technologies have been the focus of Internal Combustion Engine (ICE) development. However, it is always a big challenge for Compression Ignition (CI) engines to defeat the trade-off between NO_x and soot emissions caused by diffusion combustion. As for Spark Ignition (SI) engines, due to low compression ratio, high pumping loss, difficulty to achieve lean-burn and large cycle variation, the thermal efficiency can hardly catch up with that of CI engines. To solve these conventional problems in ICEs, some novel combustion modes have been proposed, such as Low Temperature Combustion (LTC), Homogeneous Charge Compression Ignition (HCCI) and Premixed Charge Compression Ignition (PCCI), etc.

Among these concepts, dual-fuel combustion using high-ignitable fuel to ignite low-ignitable fuel is considered as a possible way to merge the advantages of gasoline and diesel, achieving high efficiency and low emissions. Dual-fuel combustion has been studied by many research institutions and received increasing interest in recent years. Homogeneous Charge Induced Ignition (HCII) was proposed by Wang et al. [1, 2] and this mode utilizes a port injection

of high-volatile fuel to form a homogeneous charge and a direct injection of high ignitable fuel near the Top Dead Center (TDC) to trigger combustion. Wang et al. [1, 2] found that HCII was capable of achieving lean combustion at higher compression ratios and heat release rate close to constant volume combustion. In this way, HCII avoids several drawbacks that lower the thermal efficiency of SI engines, and may achieve the high thermal efficiency comparable to diesel engines. Yu et al. [3] studied the three different sequential heat release modes with HCII and found that two-stage combustion mode in HCII featured higher thermal efficiency, reduced NO_x and soot emissions and smoother pressure rise rate. Yu et al. [4] also compared HCII combustion with gasoline/diesel blend fuels combustion and found that both modes could achieve LTC with extremely low soot and NO_x emissions when combined with large amount of Exhaust Gas Recirculation (EGR). Gao et al. [5] compared the single and double injection strategies in HCII over diesel injection timing sweep and EGR sweep. It was found that compared to single injection strategy, the HC, CO and NO_x emissions of double injection HCII were significantly lower and the combustion efficiency was improved remaining a low level of soot emissions.

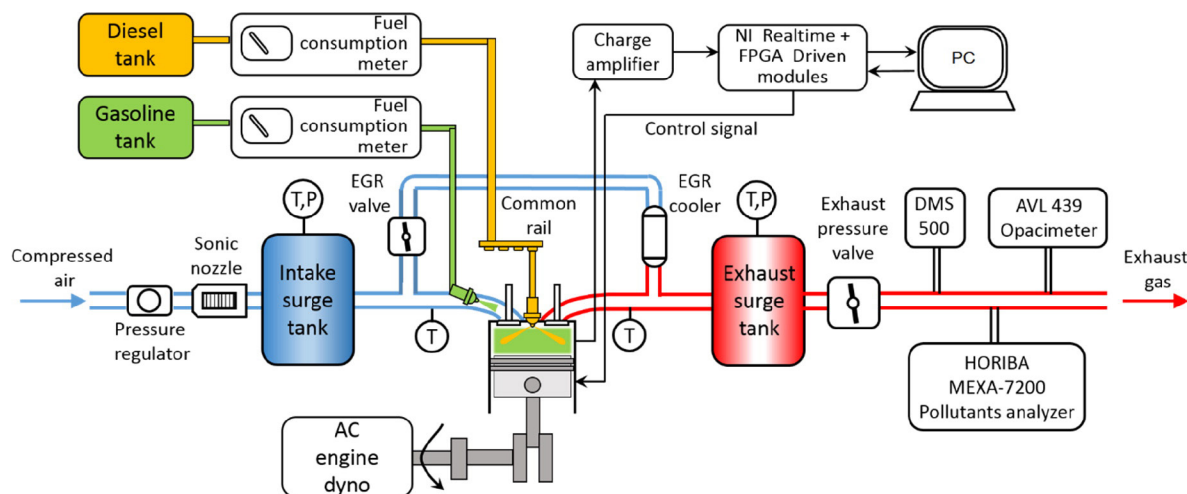


Figure 1. Schematic of the engine test bench.

HCII mode utilized the diesel direct-injection near TDC as an effective control of the combustion phasing, duration and magnitude. Therefore, certain amount of diffusion combustion may still exist in the combustion process, which makes HCII combustion distinguished from Reactivity Controlled Compression Ignition (RCCI), another dual-fuel combustion strategy proposed by Kokjohn et al. [6]. RCCI uses the injection of low reactivity fuel in the intake port and early cycle direct injection of high reactivity fuel. With multiple injections, RCCI can control the incylinder fuel reactivity to optimize the combustion process [7]. Kokjohn et al. [8] made direct comparisons between Conventional Diesel Combustion (CDC) and RCCI. The results showed that RCCI reduced NO_x and soot emissions significantly and achieve higher gross indicated efficiency. Splitter et al. [9] found that the gross thermal efficiency for RCCI could be near 60%. Both HCII and RCCI exhibit the potentials of dual-fuel combustion to realize clean and highly efficient combustion which was widely studied, even with other fuel combinations [10, 11, 12, 13, 14, 15, 16].

As the governmental regulations on exhaust emissions are getting increasingly stringent, the common rail pressure used on CI engines keeps getting higher (normally around 200 MPa) to increase the premixed combustion ratio in order to reduce PM emissions. However, such an expensive fuel injection system would considerably increase the manufacturing costs for ICEs. As for HCII combustion, well-premixed gasoline charge is already formed when diesel is injected. Thus, the need for high pressure diesel injection can be relieved and this may decrease the engine costing. Ryu et al. [17] studied the effects of injection pressure in a diesel engine using biodiesel-CNG dual fuel and found that the combustion stability increased and ignition delay decreased with the increase of the injection pressure, while smoke and NO_x emissions were decreased and increased, respectively. In Liu et al.'s study [12] about the effects of diesel injection pressure on diesel/methanol dual fuel, it was found that combustion duration became shorter and the maximum cylinder pressure and the peak pressure rise rate (PPRR) increased as the injection pressure increased. Walker et al. [18] researched RCCI strategy with a low-pressure GDI injector and the experimental results showed comparable engine performances and emission outputs for both low-pressure and high-pressure direct-injection RCCI with an optimized piston geometry.

In Walker et al.'s study [18], with GDI technology, the diesel injection pressures were much lower than that commonly used in CI engines. In the current study, the possibility of using relatively low common-rail injection pressure to achieve low PM emissions with HCII combustion strategy was explored on a single-cylinder Heavy Duty (HD) engine. The combustion and emission performances were analyzed in detail to understand the underlying process.

EXPERIMENTAL SETUP AND METHOD

The experiments were conducted on a single-cylinder engine and the major engine specifications are listed in Table 1. An external air compressor was used to supply intake air. The intake air temperature was controlled at 30 °C and the intake mass flow rate was fixed at 66 kg/h. The lubricating oil temperature was controlled at 80 °C. The test fuels were Beijing market diesel (0#) and gasoline (95#). The detailed fuel properties are shown in Table 2. The diesel and gasoline fuel supply systems were specially designed for the targeted single cylinder. Diesel was provided by an external high pressure common rail, whose fuel pump was driven by a 7.5 kW inverter motor. PID control was used to get a precise and steady rail pressure. Gasoline was port-injected and controlled electronically. The injection timing of gasoline was controlled at 300 °CA BTDC to make sure a homogeneous charge was formed.

Table 1. Engine Specifications.

Cylinder bore	123 mm
Stroke	156 mm
Displacement	1.85 L
Compression ratio	17.3
Swirl ratio	1.85
Diesel injection pressure	40-180 MPa
Gasoline injection pressure	0.3 MPa
Maximum speed	2300 r/min
Maximum torque	360 N·m

Figure 1 shows the schematic of the engine test bench, which was refitted from a HD engine. The dynamometer used for the test bench was Horiba LI250, the maximum torque of which was up to 440 N·m and maximum speed was 10000 r/min, covering all the operation conditions of the engine. Two 100L surge tanks were used to make the intake and exhaust pressure steady. HORIBA MEXA-7200 was used to measure NO_x, CO, CO₂, HC and O₂ emissions. AVL 439 opacimeter was used to measure smoke emissions, the sensitivity of which was 0.1% and response time was 100ms.

The EGR rate was determined by the intake CO₂ concentration divided by the exhaust CO₂ concentration. An exhaust pressure valve and an EGR flow regulating valve were installed to adjust EGR rate. A water-cooled exhaust gas cooler was installed in the EGR loop. Since the clean air mass flow was fixed, the intake pressure was 1.1 bar and the exhaust pressure was 1 bar for cases without EGR. And for cases with 30% EGR the intake and exhaust pressure was 1.4 bar and 1.5 bar, respectively.

Combustion DMS500 sampled the raw exhaust and measured particle size distributions. The nucleation and accumulation mode particles were discriminated automatically by the DMS500 software through fitting each log-normally distributed mode from the number-size spectrum. The first dilution ratio was fixed at 5 and the second dilution ratio was adjusted to keep the dynamic range within 0.5%-5%. The temperature of the sampling system was controlled at 80 °C.

Table 2. Fuel properties.

Property	Gasoline	Diesel
CN	-	56.5
RON	95.1	-
g _c /%	86.56	86.45
g _H /%	13.01	13.49
g _c /g _H	6.65	6.41
Density (20 °C) /(kg/L)	0.7525	0.8304
Heating value /(kJ/kg)	42,840	42,680
Kinematic viscosity (20 °C) /(mm ² /s)	0.75	4.127
Sulfur content /ppm	21.0	4.3

The engine controller was managed using the National Instruments Labview based on the NI real-time and FPGA Driven modules. The configuration of engine controller is shown in Table 3. The NI system could both perform flexible control and combustion analysis to replace the combustion analyzer. The cylinder pressure was measured with a Kistler 6125C pressure transducer and combustion pressure data were recorded with the resolution of 0.2 °Crank Angle (CA).

Table 3. Configuration of the engine controller.

Driven module	Function
ESTTL	Engine Synchronization and spark timing control
AD Combo	Engine crank and cam sensors and other analog signal inputs
PFI	Gasoline injector control
DI	Diesel injector control

The engine operation was evaluated by the indicated performance to exclude the friction and accessory discrepancies. Combustion analysis was performed based on the average cylinder pressure of 50 consecutive cycles. Heat Release Rate (HRR) was calculated using the following equation:

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta}$$

where, γ is the specific heat ratio, V is the instantaneous cylinder volume, and p is the in-cylinder pressure. CA₁₀, CA₅₀, and CA₉₀ are defined as the crank angles at which 10%, 50%, and 90% fuel mass fraction was consumed, respectively. Ignition delay is defined as the crank angle interval between Start of Injection (SOI) and CA₁₀. SOI is the CA position of the electric signal that marks when the injection starts. Combustion duration is defined as the crank angle interval between CA₉₀ and CA₁₀. Energy ratio γ_g of the gasoline to the total fuel injected during each engine cycle, may be obtained using the following equation.

$$\gamma_g = (m_g \times Hu_g) / (m_g \times Hu_g + m_d \times Hu_d)$$

where m_d is the diesel mass, m_g is the gasoline mass, Hu_d is the low heating value of diesel, Hu_g is the low heating value of gasoline.

The Indicated Thermal Efficiency (ITE) was calculated according to the indicated work and the measured fuel energy:

$$\eta_i = W_i / (m_g \times Hu_g + m_d \times Hu_d)$$

where, W_i is the indicated work.

RESULTS AND DISCUSSION

Comparison of Low-Pressure Injection HCII with High-Pressure Injection CDC at Medium Engine Load

An exploratory experiment was conducted at medium engine load with 30% EGR to study the potential of using low injection pressure in HCII to reduce PM emissions. The Indicated Mean Effective Pressure (IMEP) was fixed at 9 bar and the engine speed was fixed at 1000 r/min. Then CA₅₀ was swept from 6 °CA ATDC to 14 °CA ATDC with a single diesel injection by adjusting the injection timing. The injection pressure for CDC operation was 1000 bar, and that was 600 bar for HCII mode. The γ_g in HCII mode was controlled at 58%.

Figure 2(a) and 2(b) show the cylinder pressure and HRR for CDC mode and HCII mode, respectively, together with the diesel injection for the diesel injector (CA₅₀=6, 10, 14 °CA ATDC). For both modes,

it can be seen that the maximum pressure and maximum HRR decreased as the combustion phasing delayed from TDC. HCII was more likely to show the trend of transition from fast single-stage combustion to two-stage combustion as the injection timing was delayed, which was described in detail by Yu et al. [3]. Thus, HCII demonstrated a faster heat release process than CDC, especially in the fast single-stage combustion. This was mainly due to the high equivalence ratio of gasoline-air mixture, which was 0.45 in this experiment. After the diesel was injected, a part of gasoline was ignited in diesel/gasoline mixture region. However, then multipoint auto-ignition started in the rest gasoline mixture and the HRR of rapid gasoline auto-ignition was coupled into the diesel HRR peak, resulting a high peak of total HRR. The fast HRR may be relieved by using double injection strategy and lower gasoline equivalence ratio.

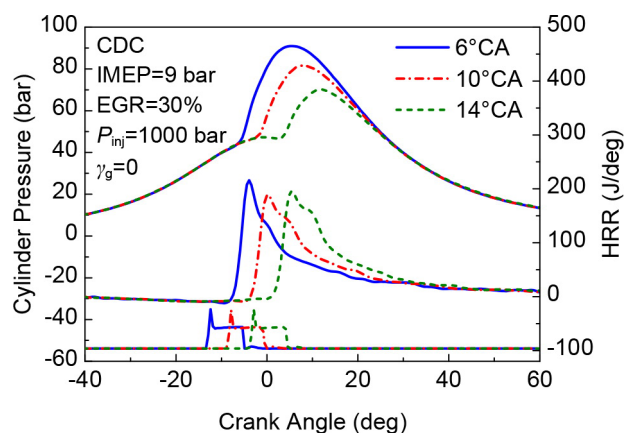


Figure 2(a). Cylinder pressure and HRR for CDC with single diesel injection. The driving currents for diesel injector are also shown.

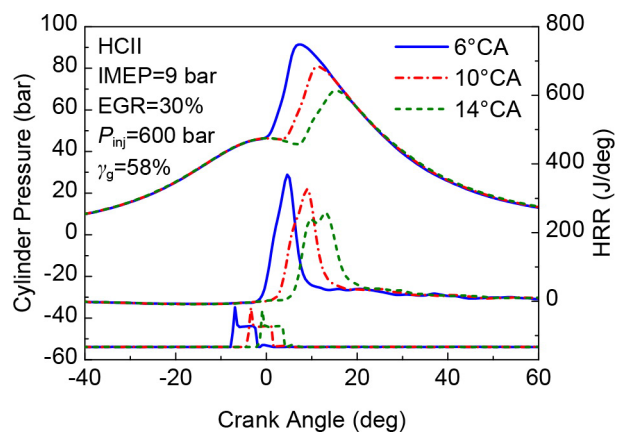


Figure 2(b). Cylinder pressure and HRR for HCII with single diesel injection. The driving currents for diesel injector are also shown.

Figure 3(a) shows the PPRR for HCII and CDC operation. It is illustrated that the PPRR for both modes decreased noticeably as the CA50 increased because the combustion phasing was delayed. It was noteworthy that the PPRR for HCII mode was higher than CDC mode and the discrepancy narrowed as the CA50 increased. When CA50 was relatively early (6 to 10 °CA ATDC), HCII shows fast single stage combustion, resulting in a high PPRR near 10 bar/°CA. As discussed above, HCII demonstrated the trend of two-stage combustion with the increase of CA50 and the PPRR declined to the same level as CDC.

Figure 3(b) shows the ITE for HCII and CDC with a single injection. As can be seen in the figure, HCII had lower ITE than CDC for all CA50 concerned (about 1-2 % lower). One reason is that the rapid HRR release in HCII mode resulted in higher temperature in the cylinder, thus, the heat transfer loss was increased in HCII mode. Furthermore, another reason for the lower ITE in HCII might be the fact that the HC and CO for HCII would be noticeably high, which might lower the combustion efficiency.

Figure 3(c) presents the Coefficient of Variation (COV) in HCII and CDC. Because gasoline auto-ignition process might vary from cycle to cycle, the COV for HCII was higher than that for CDC. However, all the considered COVs were less than 3.0, which was quite acceptable.

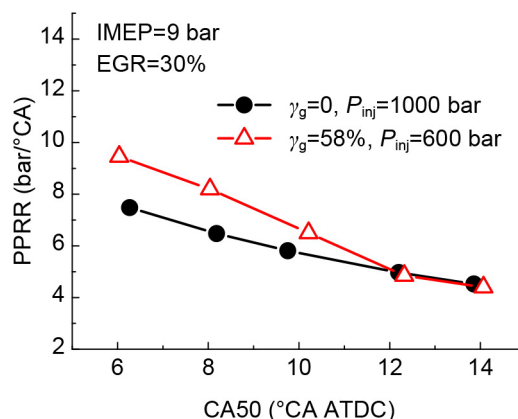


Figure 3(a). PPRR vs. CA50 for HCII and CDC with single diesel injection.

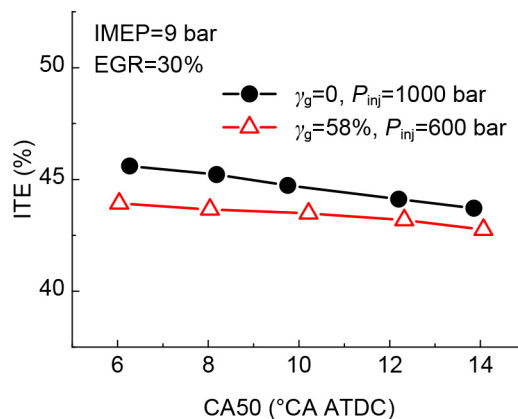


Figure 3(b). ITE vs. CA50 for HCII and CDC with single diesel injection.

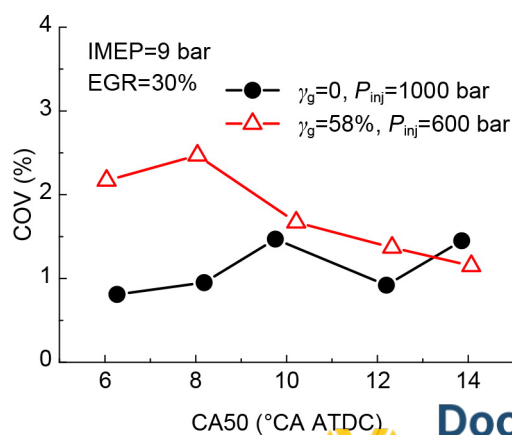


Figure 3(c). COV vs. CA50 for HCII and CDC with single diesel injection.