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An experimental investigation of the effects of fuel injection strategy on the efficiency and emissions of a heavy-duty engine at high load with gasoline compression ignition

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ABSTRACT

Based on a single-cylinder engine modified from a heavy-duty compression ignition engine with gasoline compression ignition (GCI), the influences of (i) delaying the intake valve closing timing (IVCT), (ii) a two-pulse fuel injection strategy and (iii) the injection pressure on combustion and emissions under high-load conditions are studied. By delaying the IVCT, the effective compression ratio and cylinder temperature during the compression stroke can be reduced, which can increase the fuel ignition delay and reduce soot emission and the heat-transfer loss rate, contributing to expand the engine's high-load limit for high efficiency and clean combustion. With IVCT delay at an inlet pressure of 3.0 bar and a gross indicated mean effective pressure (IMEPg) of 16.5 bar, the gross indicated thermal efficiency (ITEg) is 52.3%, which increases by 2.4% compared to the case without IVCT delay. Combining IVCT delay, two-pulse injection, and an appropriate injection pressure can reduce the fuel mixing time, which can decrease the ringing intensity (RI) and achieve clean combustion at high engine load. Using IVCT delay, an intake pressure of 3.4 bar, an exhaust-gas recirculation (EGR) rate of 47%, a pilot injection of 10% at (−70, −7), an injection pressure of 80 MPa, and an IMEPg of 15 bar, the ITEg is 53.1% and the emissions of soot and NOx are 0.0068 g/kWh and 0.21 g/kWh, respectively.

1. Introduction

Homogeneous charge compression ignition (HCCI) is an effective way to achieve clean and efficient combustion. To circumvent the difficulty of controlling the combustion phase of the HCCI combustion, low-temperature combustion (LTC) involving premixed combustion ignition and partially premixed combustion (PPC) has been proposed [1,2]. It is difficult to prepare a premixed diesel mixture for LTC. However, substantial progress has been made toward clean diesel combustion using the methods of early injection, multi-pulse injection, high exhaust-gas recirculation (EGR) rate, and delaying the intake valve closing timing (IVCT) [3–10]. Because diesel fuel is highly reactive, extensive EGR is required for high-load PPC combustion. Noehre et al. [11] used an EGR rate of 78% with a high swirl ratio and low compression ratio, at 15 bar IMEPg, they realized soot emission of 0.27 FSN. The mixture was close to the theoretical equivalence ratio and the combustion efficiency was only 85.27%, which restricted further expansion of the engine load.

In an attempt to expand the load range of high-efficiency clean combustion, Kalghatgi [12,13] used gasoline PPC ignition in a heavy-duty single-cylinder engine with a compression ratio of 14:1. Using an EGR rate of 25% at 15.95 bar IMEPg, they achieved soot and NOx emissions of 0.07 FSN and 0.58 g/kWh, respectively, and an indicated fuel consumption rate of 179 g/kWh. Since then, the gasoline PPC [14–16] combustion model has become the focus for high-efficiency clean combustion. Using a heavy-duty single-cylinder engine with a compression ratio of 9.1:1 at an engine load of 50%, Hanson et al. [17] showed that gasoline PPC can lead to lower emissions, higher thermal efficiency and lower maximum pressure rise rate (MPRR) compared with HCCI/PCCI (premixed charge compression ignition). Using a heavy-duty compression engine at 14.8 bar IMEPg with gasoline compression ignition (GCI), Johansson et al. [18] realized NOx emission in line with Euro VI emission standards. The efficiency was equivalent to that achieved by Kalghatgi [13] but the soot emission increased to 1.5 FSN. By reducing the compression ratio to 14.3:1, and using an injection pressure of 180 MPa, Johansson et al. [19] achieved highly efficient combustion at 18 bar IMEPg. The indicated thermal efficiency increased to 56% but with up to 2 FSN soot emission and a 2 MPa/deg MPRR limit to further improve engine load. Using a 240 MPa ultra-high injection pressure with an advanced engine platform, Johansson et al.

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[20] achieved highly efficient combustion at 26 bar IMEPg with an indicated thermal efficiency of 52% and soot emission below 0.5 FSN. However, when the IMEPg is up to 15 bar with GCI, the MPRR exceeds 2 MPa/deg, which most engines cannot tolerate.

In the past few years, researchers in Saudi Arabian have studied the low octane number fuel compression ignition combustion technology [21–26], especially naphtha fuel GCI combustion, based on gasoline and diesel prototypes. The results show that low octane number fuel GCI combustion can also achieve the same high efficiency combustion as diesel engine, and effectively reduce soot and NOx emissions. Meanwhile, low octane number fuel is lesser processed than commercial gasoline and diesel fuel, and helps saving the total energy consumption from well to wheel. Hence, GCI combustion has a bright prospect in high efficiency and clean combustion.

Delaying IVCT [5,27] developed by our research group can reduce an engine's effective compression ratio while maintaining a constant expansion ratio. This is the well-known quasi Miller cycle, which can effectively reduce the temperature and pressure during an engine's compression stroke as well as help promote fuel mixing, which is helpful for diesel engine. This study focuses on how to realize clean combustion with high efficiency and maintain acceptable MPRR under high engine load with GCI combustion. Based on this, the present study considers the influence of (i) delaying IVCT, (ii) a two-pulse injection strategy, and (iii) injection pressure on GCI in a heavy-duty engine under high engine load.

2. Experimental apparatus

The test engine is a single-cylinder engine modified from a six-cylinder heavy-duty compression ignition engine; the parameters are given in detail in Table 1. The intake system is comprise of an air compressor, a pressure-regulating valve, a TOCEIL-LFE125 intake airflow meter (measurement uncertainty [28]: < ± 0.315 m^3/h) and a surge tank, which is used for simulating the supercharger. The exhaust system is comprise of an exhaust surge tank, an EGR valve, and an exhaust pressure-regulating valve. To change the effective compression ratio of the test engine, the test cylinder is equipped with an IVCT delaying system [27] developed by our research group. The test engine system is shown in Fig. 1. The combined intake and exhaust system of the engine can flexibly regulate the EGR rate and the pressure and flow rate of the inlet and exhaust system. During the experiments, the exhaust back pressure is higher than the intake pressure by 0.1 bar, thereby ensuring that the EGR system works well. The EGR rate is defined as

$$\mathrm{EGR} = \frac{\mathrm{m_r}}{m_r + m_{in}} \times 100\% \tag{1}$$

where m_r is the mass of recirculated exhaust gas in the cylinder and $\mathrm{m_{in}}$ is the corresponding mass of fresh air.

The engine is equipped with an independent high-pressure common rail fuel injection system (maximum injection pressure is 160 MPa with diesel fuel) that can flexibly adjust the fuel injection pressure (Pcr), the injection timing (SOI: start of injection), and the number of injection pulses to achieve multi-pulse injection. The injector is a standard injector for heavy-duty diesel engines with eight orifices. The orifice diameter is 0.217 mm and the spray angle is 143°.

The fuel is 92# commercial gasoline (92 RON). Table 2 gives details. To avoid wear of the fuel supply system, 200 ppm of lubricating additive is added to the fuel. The maximum injection pressure for gasoline fuel is 140 MPa.

The cylinder pressure is measured with a Kistler 6125B pressure transducer (measurement uncertainty: < ± 0.3 bar) and a single-channel charge amplifier. The IMEPg is calculated by P–V data with more than 200 cycles for each case. The exhaust system uses a Horiba MEXA-7100DEGR exhaust-gas analyzer (the measurement sensitivities for all emissions are 1 ppm) and an AVL 415S smoke meter (measurement sensitivity: 0.001 FSN) to measure the exhaust components and soot emission.

In this study, the gross indicated thermal efficiency (ITEg) [29] of the engine, the incomplete-combustion loss rate η_C, the heat-transfer loss rate η_{HT} and the exhaust loss rate η_{EL} are defined as follows:

$$\mathrm{ITEg} = \frac{W_g}{Q_f} \times 100\% \tag{2}$$

$$\eta_C = \frac{\mathrm{Q_{ub}}}{\mathrm{Q_f}} \times 100\% \tag{3}$$

$$\eta_{HT} = \frac{\mathrm{Q_{HT}}}{\mathrm{Q_f}} \times 100\% \tag{4}$$

$$\eta_{EL} = (1 - \mathrm{ITE_g} - \eta_C - \eta_{HT}) \times 100\% \tag{5}$$

where W_g is the work done over the compression and expansion strokes per cycle, Q_f is the low heat value of injected fuel per cycle, $\mathrm{Q_{ub}}$ is the low heat value of unburned fuel that is converted by unburned hydrocarbons (HC) and CO, and $\mathrm{Q_{HT}}$ is the heat lost to cooling water, which can be calculated using the Sitkei formulas [30].

3. Results

3.1. Effect of IVCT system on combustion and emissions

Our previous research [31] demonstrated that the effective compression ratio of the engine was reduced with IVCT delay, and the charge in-cylinder was reduced under the same intake pressure and the mass of fuel injected per cycle, which increased the equivalence ratio and soot and unburned exhaust pollutants. It is necessary to increase the intake pressure for clean combustion under high engine loads. In this subsection, we investigate the influence of IVCT on GCI combustion and emissions.

During the experiments, the engine speed is fixed at 1300 rpm, the intake temperature is around 40 °C, the EGR rate is 40% and the IVCT is −70° after top dead center (ATDC) (at this time, the effective compression ratio is 7.4). With a single injection, we change the fuel mass to change the engine load. The test conditions are quantified in Table 3.

As the IMEPg is increased, the fuel/oxygen equivalence ratio φ [27] of the mixture increases and the intake-oxygen concentration decreases. Comparing the cases with/without IVCT delay, the oxygen concentration and the φ value are almost coincident under the same load, as shown in Fig. 2.

The fuel/oxygen equivalence ratio φ is defined as

$$\varphi = \frac{\frac{m_f}{m_{oxy}}}{\left(\frac{F}{O_2}\right)_{stoic}} \tag{6}$$

where m_f is the fuel mass injected per cycle, m_{oxy} is the total oxygen mass in the cylinder per cycle, and $(F/O_2)_{stoic}$ is the stoichiometric fuel/oxygen ratio (0.293 for gasoline).

Table 1
Engine parameters.

Bore × stroke	126 × 155
Swirl ratio	1.2
Maximum cylinder pressure	16.5 MPa
Compression ratio	17:1
Combustion chamber	Open ω shape with BUMP ring [27]
Valve timing	IVO: 340° ATDC
(four valves)	IVC: 146° BTDC
ATDC: after top dead center	EVO: 131° ATDC
BTDC: before top dead center	EVC: 339° BTDC
Fuel injection system	Common rail
Maximum injection pressure	160 MPa
Injector	8 × 0.217 × 143°