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Research article

Effect of direct injection dimethyl ether on the micro-flame ignited (MFI) hybrid combustion and emission characteristics of a 4-stroke gasoline engine

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ABSTRACT

The low temperature combustion is required for an internal combustion gasoline engine to achieve higher thermal efficiency and lower NOx emissions. The gasoline controlled auto-ignition (CAI) has been extensively researched to achieve low temperature combustion but its lack of control in combustion process has limited its applications. To overcome this problem, dimethyl ether (DME) is directly injected into the cylinder to generate multiple micro-flame ignition sites and regulate the entire heat release of premixed gasoline mixture. This micro-flame ignited (MFI) hybrid combustion was achieved in a 4-stroke gasoline engine in this work. The results show that the heat release process consists of three stages: stage I of low temperature oxidation reactions, followed with a short delay of a few crank angles by stage II of high temperature reactions of DME and the main combustion of premixed gasoline mixture in stage III. With the increased amount of DME in the late injection, the start of main combustion advances and the combustion duration reduces for both single and split injections, which results in lower HC and CO emissions. The introduction of split DME injections improves the combustion process and reduces HC and CO emissions as well as the maximum thermal efficiency.

1. Introduction

Over the last 30 years, vehicle emissions legislation has been becoming more and more stringent in the main industrial countries. Although a conventional spark-ignition (SI) gasoline engine equipped with a three-way catalytic converter can easily meet stringent emissions legislation at stoichiometry, their efficiencies need to be improved significantly to meet the future carbon dioxide (CO_2) emissions or fuel economy requirements [1,2].

Fuel economy of conventional SI gasoline engines is much lower than that of compression ignition (CI) diesel engines, due to high pumping losses at part loads and operation at stoichiometry [3,4]. Significant research and development works have been carried out to overcome the pumping loss and improve the engine efficiency through lean or diluted combustion [5,6]. Amongst those potential combustion technologies studied, controlled auto-ignition (CAI), also known as homogeneous charge compression ignition (HCCI), has been shown as a promising alternative combustion concept to achieve high fuel efficiency with near zero nitrogen oxides (NOx) emissions [7,8].

Although CAI combustion have many advantages over conventional SI combustion, its commercial applications still face various obstacles including its narrow operating range in engine speed-load map, lack of direct control over the combustion phasing, high unburned hydrocarbon and carbon monoxide emissions. Ignition timing of CAI combustion could not be directly regulated like the conventional gasoline engine controlled by spark timing or diesel engine by direct injection timing since the auto-ignition process is governed by complicated chemical kinetics [9,10].

Spark assisted auto-ignition hybrid combustion, in which a flame kernel formed by the ignition of stratified fuel near the spark plug is used to increase the in-cylinder temperature and trigger the auto-ignition of the remaining fuel, is a promising way to achieve diluted or lean burn combustion and control the process of heat release in the cylinder. The spark assisted CAI hybrid combustion was demonstrated at low load and speed conditions in a single-cylinder 4-stroke gasoline engine, in which the premixed homogeneous mixture formed by port fuel injection was burned by multiple auto-ignition promoted by the formation of an expanding flame front near the spark plug [11]. However, Law et al. [12] found that spark timing showed no effect on ignition and combustion processes at 2000 rpm and indicated mean effective pressure (IMEP) in a range from 2 bar to 5 bar. In addition, the spark assisted auto-ignition combustion suffered from high cycle-to-cycle variation because of the cyclic variation in the flame kernel development [13–15], which could be reduced by the employment of stratified

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charge near the spark plug to enhance the initial flame speed [16]. Both modelling [17] and experimental studies [18,19] showed that the early flame propagation and auto-ignition process in stratified charge flame ignition combustion could be controlled by fuel stratification in the cylinder and the advance of spark timing. However, gasoline stratification may result in high particulate matter (PM) and NOx emissions in the stratified regions [18,19].

Although dimethyl ether (DME) and diesel fuel are two of the ignitable fuels with low auto-ignition temperature [20,21], DME is easy to evaporate at low temperature and low injection pressure when liquid-phase DME is directly injected into the cylinder. Besides, DME has high oxygen content and no C—C bonds [22], which is benefit to achieve smokeless combustion. Thus, DME is used to trigger the auto-ignition of premixed gasoline mixture rather than diesel fuel. When injected directly into a premixed gasoline/air mixture, the auto-ignition of DME could start the heat release process and trigger the auto-ignition of premixed gasoline mixture in gasoline engines with trapped hot burned gases [23,24], either through exhaust throttling in the exhaust pipe or intake re-breathing by the secondary opening of the intake valve during the exhaust stroke. The direct injection timing and the amount of DME affected the timing of auto-ignition and NOx emissions can be decreased by over 90% compared to the spark ignition mode. However, indicated thermal efficiency was lower than that in SI mode due to high pumping losses in the case of high back pressure operations [23]. In order to optimize the engine fuel economy and reduce the emissions, Seo et al. [24] investigated the effect of the timings and quantity of split DME injections and found that the second injection timing of DME could control the phasing of auto-ignition whilst the early injection of a small amount of DME helped improve the combustion efficiency of the premixed gasoline mixture.

Compared to the intake re-breathing method, the negative valve overlap (NVO) strategy is capable of getting more and hotter internally recycled burned gases, in which exhaust valves are closed before top dead center (BTDC) to trap burned gases and intake valves are opened after top dead center (ATDC) to avoid backflows burned gases into intake ports [25]. Thus, the current study was planned and carried out to investigate the characteristics of DME enabled micro-flame ignition (MFI) in a gasoline engine with negative valve overlap. The effect of DME injection strategy on ignition, heat release rate, efficiency and emissions was analyzed and discussed.

2. Experimental setup

The experiments were carried out in a single cylinder 4-stroke gasoline engine equipped with variable valve timing and duration systems on both the intake and exhaust valves based on the BMW's Valvetronic and Vanos devices. The specifications of the engine are shown in Table 1.

The engine could be operated with either or both port fuel injection (PFI) and direct injection (DI). Gasoline was injected into the intake ports through the PFI injector and DME directly injected into the cylinder via a side mounted multi-hole DI gasoline injector. The properties of DME and gasoline are shown in Table 2.

Table 1
Engine specifications.

Parameters	Value
Bore (mm)	86
Stroke (mm)	86
Number of valves	4
Sweep volume (cc)	500
Geometric compression ratio	12.4
Coolant temperature (°C)	80 ± 1
Oil temperature (°C)	55 ± 1
Intake temperature (°C)	26–28
Intake pressure (MPa)	0.1
Gasoline injection pressure (MPa)	0.35
DME injection pressure (MPa)	4

Table 2
Properties of dimethyl ether and gasoline.

Fuel type	Dimethyl ether	Gasoline
Chemical formula	CH_3OCH_3 [21]	C_4–C_{12}
Stoichiometric air/fuel ratio	9.0 [21]	14.7
Cetane number	55–60 [22]	10–15
Low heating value (MJ/kg)	27.6 [21]	43.96

The DME supply system as shown in Fig. 1 has been carefully designed to keep DME in the liquid state by pressurizing the DME in the fuel line to 4 MPa by compressed gaseous nitrogen (N_2) through adjusting the pressure relief valve 2 between a high pressure nitrogen bottle and DME bottles. If the bubbles are observed in the observation chamber, they were discharged through a manually operated bleed valve on the observation chamber in order to avoid the pressure fluctuation and changes to the DME injector. DME was directly injected into the cylinder through a commercial gasoline direct injection (GDI) injector during the experiment, and a sensitive MAX 213 positive displacement piston type flow meter was connected in the supply line in series for DME measurement. There is no leakage in the nozzle tip of GDI injector for DME during the experiments.

As shown in Fig. 2(a), the first direct injection of DME was at 220°CA BTDC to form uniform air-DME mixture in the cylinder and the second direct injection timing of DME was 60°CA BTDC. The bowl in the piston was designed for generating micro-flame by assisted spark discharge in the case of late direct injection timing based on the 3D-CFD simulations [16]. The total lower heating value (LHV) of gasoline and DME in a cycle was kept constant at 475 J, equal to the LHV of 10.8 mg gasoline.

Fig. 2(b) shows schematically the in-cylinder distribution of gasoline and DME, which can be altered by the injection timing, number and the mass of DME in the cylinder. The 1st_DI DME has long time to mix with the air-gasoline and forms relatively homogenous air-gasoline-DME mixture. The second injection of DME tends to locate near the center of the chamber to form a stratified charge near the spark plug.

To find the effect of DME on the combustion characteristics of the engine, the ratio of DME in the first direct injection to the total energy released in a cycle (γ_{1st_DI}), and the ratio of DME in the second direct injection to the total energy released in a cycle (γ_{2nd_DI}) are defined as follows:

$$\gamma_{1st\,DI} = \frac{m_{1st\,DI} \times LHV_{DME}}{m_G \times LHV_G + (m_{1st\,DI} + m_{2nd\,DI}) \times LHV_{DME}} \times 100\% \quad (1)$$

$$\gamma_{2nd\,DI} = \frac{m_{2nd\,DI} \times LHV_{DME}}{m_G \times LHV_G + (m_{1st\,DI} + m_{2nd\,DI}) \times LHV_{DME}} \times 100\% \quad (2)$$

where, m_{1st_DI} and m_{2nd_DI} are the mass of DME in the first direct injection and the second direct injection, respectively. m_G is the mass of gasoline burned in a cycle. LHV_{DME} and LHV_G are low heating value of DME and gasoline, respectively.

In-cylinder pressure data were measured by a flush-mounted piezoelectric pressure transducer (Kistler 6125B) coupled with a Kistler 5011B charge amplifier and recorded by an in-house data acquisition system at a resolution of 0.36°CA using a shaft encoder.

To analyze the combustion characteristics of the engine, heat release rate (HRR) is obtained based on the first law of thermodynamics as follows:

$$\frac{dQ}{d\theta} = \frac{1}{\gamma - 1} \times V \times \frac{dP}{d\theta} + \frac{\gamma}{\gamma - 1} \times P \times \frac{dV}{d\theta} \quad (3)$$

where, γ is the ratio of specific heat, P and V are the in-cylinder pressure and volume at a crank angle (θ), respectively.

CA10 and CA90 are used to define the start and end of the main heat