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Effect of split injections coupled with swirl on combustion performance in DI diesel engines

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ABSTRACT

Engine-out emissions (NOx and soot) have led to serious air pollution problems, and consequently, increasingly stringent emission norms. In order to decrease the emissions and improve the combustion performance of diesel engines, the effect of split injections with swirl (swirl rate of 0 and 1) was experimentally researched, and the mechanism of the fuel/air mixture of split injections with swirl (swirl rate of 0.5–2.5) was numerically researched in this study. The experimental research was carried out on a modified 1132Z single cylinder diesel engine, equipped with an endoscope system. A two-color method was applied to record flame temperature distribution and KL factor. The experimental results indicate that the flame observed using split injections with swirl rotated obviously. Split injections with swirl had a positive influence on improving the fuel/air mixture, accelerating combustion progress and shortening combustion duration. The combustion duration decreased at swirl rate of 1, with a reduction in the range of 19.5–25.7% at various pilot quantities. In addition, brake-specific fuel consumption (BSFC) and soot emission were reduced. BSFC was lower at swirl rate of 1 than that at swirl rate of 0, with a reduction in the range of 1.1–2.01 g/(kW h)$^{-1}$. For KL factor at 12°CA ATDC, it was also observed that at 12°CA ATDC, the KL factor was lower at swirl rate of 1 than that at swirl rate of 0, with a reduction of 34.9%. Related numerical research on split injections with swirl was performed, and the results show that at a specific swirl, the main injection deviated from the pilot injection and entered the area between two sprays, enhancing the utilization of air in the chamber. In addition, the main combustion process accelerated due to the better thermal-atmosphere provided by the pilot injection, so that a better engine performance and lower soot concentration was achieved.

1. Introduction

Diesel engines are widely used in the field of transportation and engineering machinery because of their excellent thermal efficiency and practical economic viability [1–3]. However, emissions, such as NOx and soot, have led to several serious air pollution problems [4,5]. Stricter regulations and control measures aimed at reducing exhaust emissions have been implemented in many countries all over the world [6,7]. Conventional fuel injection systems have been studied at great length and several strategies have been proposed to reduce the emissions of diesel engines [8,9]. One of the most promising strategies is split or multiple injections, which is created by dividing a conventional injection into two or more sections [10,11]. The results show that split or multiple injections have a positive effect on increasing the fuel/air mixture area and decreasing the air/fuel mixture concentration [12–14]. Advantages, such as brake-specific fuel consumption (BSFC) and soot reduction have also been obtained [15,16].

Prior investigations on split injections were focused on the improvement of fuel economy and emission performance of direct injection (DI) diesel engines. By multidimensional modeling, Reitz et al. [17,18] found that if the dwell time was too short, fuel from the second injection would penetrate into the rich zone formed by the first injection; thus, a split injection would be similar to a single injection. On the contrary, if the dwell time was too long, the second injection could not be ignited rapidly, and the long delay would have a negative effect on soot emissions and thermal efficiency. Ehleskog et al. [19] also studied the influence of split injection dwell time on emissions and engine performance at low load in a heavy-duty direct injection (HDDI) diesel engine and found

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that NOx emissions and BSFC increased with dwell time. Therefore, it was valuable and necessary to propose optimal dwell times when adapting a split injection strategy.

Additionally, investigations of air swirl in diesel engines have attracted the interest of many scholars over the past few decades. Air swirl is beneficial towards decreasing CO and soot emissions, especially when the injection timing is earlier or later than the optimal timing [20–23]. Sjöberg et al. [24] also studied the influence of swirl on the combustion process of a DI diesel engine and found that the ignition delay decreased with increasing swirl. Sharma et al. [25] found that the swirl ratio had a considerable impact on increasing the heat transfer rates in a homogenous charge compression ignition (HCCI) engine. The results show that the heat transfer rates increased by 0.88% to the wall at a swirl ratio of 1, and 45.66% to the dome and 39.99% to the piston at a swirl ratio of 4, which reduced in-cylinder temperature and NOx formation in the HCCI mode.

As mentioned above, it is possible to reduce CO and soot emissions by using split injections or swirl. However, to the authors' knowledge, studies on the effect of split injections coupled with swirl on the combustion progress and the emissions are scarce, and studies about the mechanism of the fuel/air mixture of split injections with swirl are rarely found. In this study, the BSFC, the heat release rate (HRR), the combustion duration were studied; flame temperature distribution and KL factor of split injections with swirl (swirl rate of 0 and 1) were analyzed by two color method and compared. Furthermore, numerical research on the in-cylinder fuel/air equivalence ratio and in-cylinder temperature of split injections with swirl (swirl rate of 0.5–2.5) was conducted to reveal the mechanism of the fuel/air mixture of split injections with swirl, and the match of split injections with swirl was also put forward. This study will be of sound reference for improving the utilization of in-cylinder air as well as soot reduction research of direct injection (DI) diesel engines.

2. Analytical methodology

2.1. Experimental study and data processing

The tests were conducted on a 132 mm diameter single cylinder engine (One cylinder research diesel engine Model 1132Z). The connecting rod length and stroke length were 262 mm and 145 mm respectively. The nozzle diameter and nozzle number of the injector was 0.195 mm × 7.

A VHN-16/8 air compressor was setup to create a supercharged environment. In order to heat the intake air and establish a constant temperature environment, an AEH100 was installed. In addition, a common-rail system was setup to supply fuel to the engine. The fuel injection pressure was 140 MPa and was measured by a Kistler 4067B. The number of injections could be varied from 1 to 5 in a single engine cycle; however, one pilot injection and one main injection were chosen in this study. The injection rate and fuel injection quantities were measured by an EFS8246 single jet measuring instrument, with an accuracy of 0.01 mm^3/cycle. The location of the crank-shaft position was provided by a crankshaft transducer (Kistler 2613B), with a sensitivity of 0.1°CA. A MEXA-720 NOx analyzer provided by HORIBA was used to measure the NOx concentration with an accuracy within ±1 ppm, and the sample volume for each measurement was 1 L. The pressure were measured by a Kistler 6025B. Because the endoscope cannot work in high temperature and pressure, loading low load was chosen for operation. The engine speed and brake power were 1500 rpm and about 25 kW, respectively. All measurements were recorded when the engine was at steady state. The test conditions for this study and the specifications of the test engine are summarized in Table. 1.

The engine was modified with the installation of an endoscope system to observe the real-time in-cylinder combustion flame, which required a minor modification to the engine head. The Visioscope 513D endoscope system (AVL513D) was provided by AVL. The system consists of an endoscope, a high-speed camera, light and an optical fiber. The lighting system was not used since the entire field of view was brightly illuminated by the flame. The schematic diagram of the apparatus is shown in Fig. 1. A line of sight of 0°from the axis with an 80° field of view was created and two whole sprays were visualized. The overhead view of the combustion chamber is shown in Fig. 2. Cooling was very important for the endoscope; most prior reported failures were due to the optical property deterioration at high temperatures. In the present experiment, fresh air was supplied from an air compressor. Heat was convected away through the air flow. The endoscope is easy to get fouled during operation, and in order to take clear and effective flame pictures, the number of cycles was controlled to about 16 times.

The two-color method was utilized to obtain flame temperature and KL factor information. The basis of the two-color method was the relationship between the temperature of glowing particles and the intensity of their thermal radiation. The spectral intensity emitted by a black body, *Io*, was given by Planck's radiation law:

$$Io(\lambda, T) = C_1 \lambda^{-5} \frac{1}{\exp(C_2/\lambda T) - 1} \tag{1}$$

where C_1 and C_2 are Planck's radiation constants, having values of 1.191×10^8 wμ m^4/m^2 and 14388 μ mk, respectively.

Real objects, such as soot particles, are not perfect emitters and radiate less energy at each wave length than the equivalent black body value. The spectral emissivity ε_λ was defined as the ratio of the actual intensity emitted by a measured object at a given wavelength to the black body intensity, so that:

$$I(\lambda, T) = \varepsilon_\lambda Io(\lambda, T) = \varepsilon_\lambda C_1 \lambda^{-5} \frac{1}{\exp(C_2/\lambda T) - 1} \tag{2}$$

Since $\exp(C_2/\lambda T) \gg 1$, Wien's approximation to Planck's law could be used:

$$I(\lambda, T) = \varepsilon_\lambda C_1 \lambda^{-5} \exp(-C_2/\lambda T) \tag{3}$$

At the same time, T_a was defined as the luminous temperature, so that:

$$I(\lambda, T) = Io(\lambda, T_a) = C_1 \lambda^{-5} \exp(-C_2/\lambda T_a) \tag{4}$$

ε_λ was a function of wavelength λ and temperature T and was given by:

$$\varepsilon_\lambda = 1 - \exp(-KL/\lambda^\alpha) \tag{5}$$

where K is the absorption coefficient, which was proportional to the intensity of the object radiation. L is the thickness of the object radiation. α was constant in specific wavelength range, having a value of 1.38 in the a diesel engine.

Based on the above assumption and by letting $KL_{\lambda 1} = KL_{\lambda 2}$, the projection temperature T at wavelengths λ_1 and λ_2 was given by:

$$\begin{aligned} &\left[1 - \exp\left\{-\frac{C_2}{\lambda_1}\left(\frac{1}{T_{a_1}} - \frac{1}{T}\right)\right\}\right]^{\lambda_1 a} \\ &= \left[1 - \exp\left\{-\frac{C_2}{\lambda_2}\left(\frac{1}{T_{a2}} - \frac{1}{T}\right)\right\}\right]^{\lambda_2 a} \end{aligned} \tag{6}$$

Then KL was given by:

$$KL = -\lambda^a \ln\left[1 - \exp\left\{-C_2/\lambda\left(\frac{1}{T_a} - \frac{1}{T}\right)\right\}\right] \tag{7}$$

where T was the temperature calculated by Eq. (6).